Heat Exchangers

Notes based on practice, tutorial and past year examination papers

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Typical Assumptions

1. Steady heat transfer
2. Constant properties (fluid specific heats, overall heat transfer coefficients...)
3. No heat generation
4. Negligible conduction and radiation (please check!)
5. Flow is incompressible.
6. Flow is laminar or turbulent or mixed (please check!)
7. Velocity does not vary in direction perpendicular to the flow
8. Fully (thermal or hydrodynamic) developed flow or (thermal or hydrodynamic) developing (please check!)
9. No heat generation (from frictional heating)
10. No work done by viscous forces. (please check!)
11. No body forces on fluid (please check!)
12. Negligible heat conduction in direction perpendicular to the flow (please check!)
13. Smooth surface (please check!)
14. Gas (if there is any) is standard and calorically perfect.
15. Heat exchanger wall has no contact resistance or thermal resistance (please check!) because it is thin?
16. Negligible kinetic energy and potential energy change in fluid (example fluid)
17. Fluid in question has similar properties to air or water (example: blood)
18. Properties are obtained by linear interpolation.
19. No fouling.
20. No heat loss to surroundings.
21. Flow is inviscid (please check!)
22. Average velocity is constant
23. Axial conduction along tube is negligible

Fouling Factors

TABLE 11.1 Representative fouling factors [1]

Fluid	R''_f (m^2 · K/W)
Seawater and treated boiler feedwater (below 50°C)	0.0001
Seawater and treated boiler feedwater (above 50°C)	0.0002
River water (below 50°C)	0.0002–0.001
Fuel oil	0.0009
Refrigerating liquids	0.0002
Steam (nonoil bearing)	0.0001

TABLE 11–2

Representative fouling factors (thermal resistance due to fouling for a unit surface area)

Fluid	R_f, m^2·K/W
Distilled water, seawater, river water, boiler feedwater:	
Below 50°C	0.0001
Above 50°C	0.0002
Fuel oil	0.0009
Transformer, lubricating or hydraulic oil	0.0002
Quenching oil	0.0007
Vegetable oil	0.0005
Steam (oil-free)	0.0001
Steam (with oil traces)	0.0002
Organic solvent vapors, natural gas	0.0002
Engine exhaust and fuel gases	0.0018
Refrigerants (liquid)	0.0002
Refrigerants (vapor)	0.0004
Ethylene and methylene glycol (antifreeze) and amine solutions	0.00035
Alcohol vapors	0.0001
Air	0.0004

Heat Transfer Coefficient

TABLE 11.2 Representative values of the overall heat transfer coefficient

Fluid Combination	U (W/m$^2 \cdot$ K)
Water to water	850–1700
Water to oil	110–350
Steam condenser (water in tubes)	1000–6000
Ammonia condenser (water in tubes)	800–1400
Alcohol condenser (water in tubes)	250–700
Finned-tube heat exchanger (water in tubes, air in cross flow)	25–50

TABLE 11–1

Representative values of the overall heat transfer coefficients in heat exchangers

Type of Heat Exchanger	U, W/m^2·K*
Water-to-water	850–1700
Water-to-oil	100–350
Water-to-gasoline or kerosene	300–1000
Water-to-brine	600–1200
Feedwater heaters	1000–8500
Steam-to-light fuel oil	200–400
Steam-to-heavy fuel oil	50–200
Steam condenser	1000–6000
Freon condenser (water cooled)	300–1000
Ammonia condenser (water cooled)	800–1400
Alcohol condensers (water cooled)	250–700
Gas-to-gas	10–40
Gas-to-brine	10–250
Oil-to-oil	50–400
Organic vapors-to-water	700–1000
Organic solvents-to-organic solvents	100–300
Water-to-air in finned tubes (water in tubes)	30–60†
	400–850‡
Steam-to-air in finned tubes (steam in tubes)	30–300†
	400–4000‡

*Multiply the listed values by 0.176 to convert them to Btu/h·ft^2·°F.
†Based on air-side surface area.
‡Based on water- or steam-side surface area.

Log-Mean Temp Difference

To design or to predict the performance of a heat exchanger, it is essential to relate the total heat transfer rate to quantities such as the inlet and outlet fluid temperatures, the overall heat transfer coefficient, and the total surface area for heat transfer. Two such relations may readily be obtained by applying overall energy balances to the hot and cold fluids, as shown in Figure 11.6. In particular, if q is the total rate of heat transfer between the hot and cold fluids and there is negligible heat transfer between the exchanger and its surroundings, as well as negligible potential and kinetic energy changes, application of the steady flow energy equation, Equation 1.12d, gives

$$q = \dot{m}_h(i_{h,i} - i_{h,o}) \tag{11.6a}$$

and

$$q = \dot{m}_c(i_{c,o} - i_{c,i}) \tag{11.7a}$$

where i is the fluid enthalpy. The subscripts h and c refer to the hot and cold fluids, whereas the subscripts i and o designate the fluid inlet and outlet conditions. If the fluids are not undergoing a phase change and constant specific heats are assumed, these expressions reduce to

$$q = \dot{m}_h c_{p,h}(T_{h,i} - T_{h,o}) \tag{11.6b}$$

and

$$q = \dot{m}_c c_{p,c}(T_{c,o} - T_{c,i}) \tag{11.7b}$$

Another useful expression may be obtained by relating the total heat transfer rate q to the temperature difference ΔT between the hot and cold fluids, where

$$\Delta T \equiv T_h - T_c \tag{11.8}$$

Such an expression would be an extension of Newton's law of cooling, with the overall heat transfer coefficient U used in place of the single convection coefficient h. However,

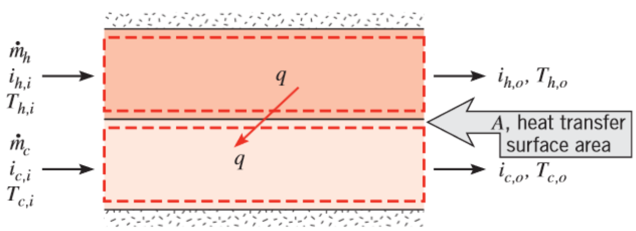


FIGURE 11.6 Overall energy balances for the hot and cold fluids of a two-fluid heat exchanger.

since ΔT varies with position in the heat exchanger, it is necessary to work with a rate equation of the form

$$q = UA\Delta T_m \tag{11.9}$$

where ΔT_m is an appropriate *mean* temperature difference. Equation 11.9 may be used with Equations 11.6 and 11.7 to perform a heat exchanger analysis. Before this can be done, however, the specific form of ΔT_m must be established.

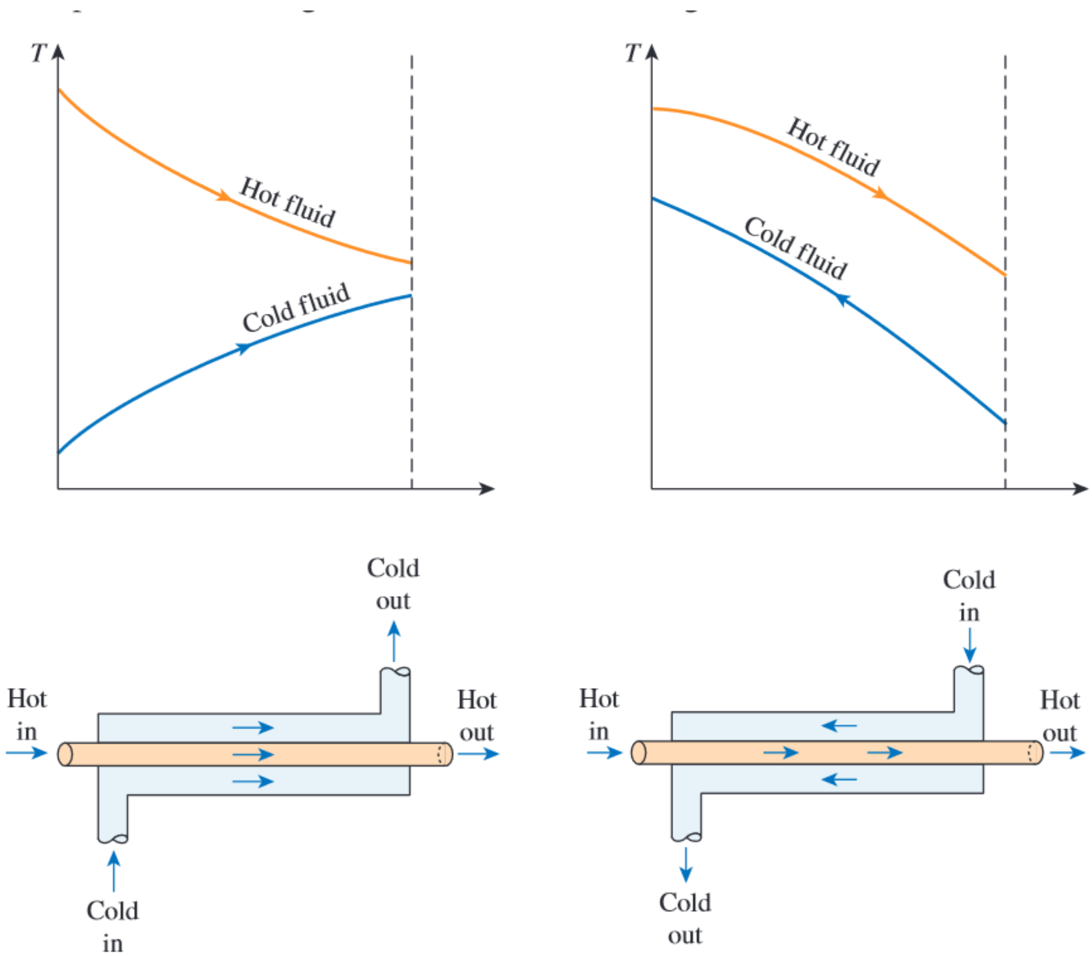
Comparing the above expression with Equation 11.9, we conclude that the appropriate average temperature difference is a *log mean temperature difference,* ΔT_{lm}. Accordingly, we may write

$$q = UA\,\Delta T_{lm} \tag{11.14}$$

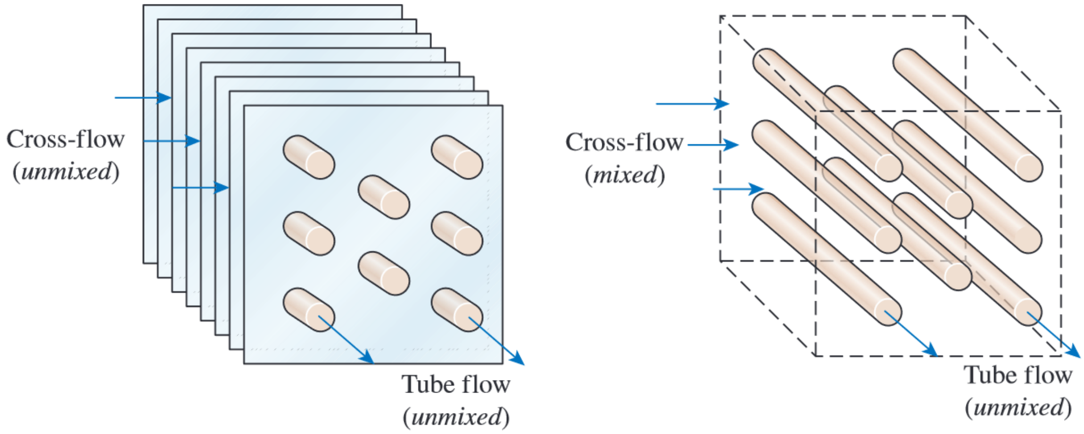
where

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \tag{11.15}$$

Pictures of Heat Exchangers



(*a*) Parallel flow (*b*) Counter flow



(*a*) Both fluids unmixed (*b*) One fluid mixed, one fluid unmixed

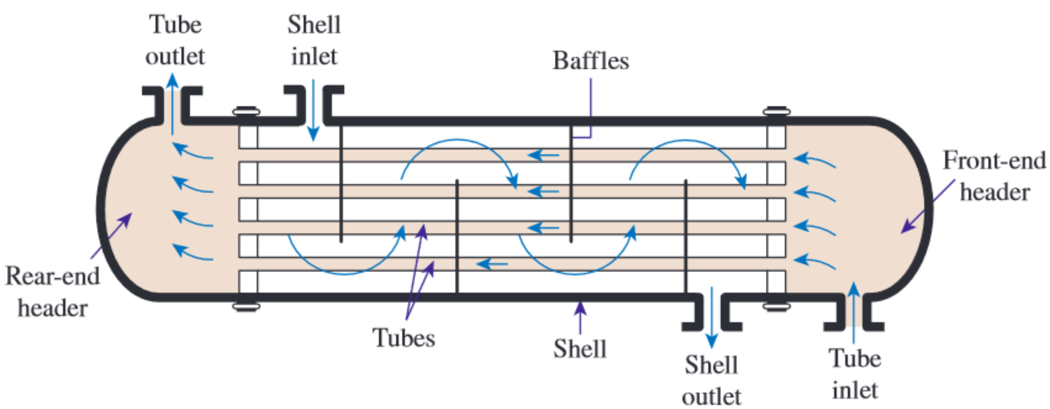
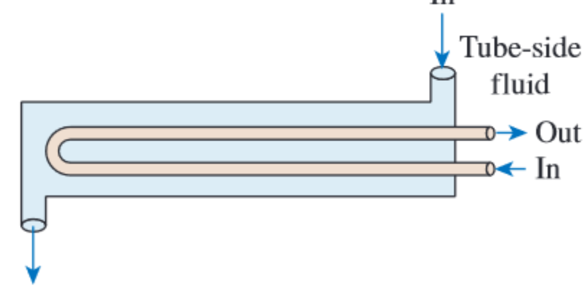
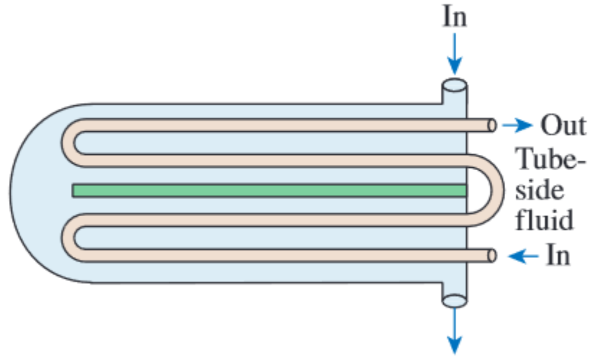


FIGURE 11–5
The schematic of a shell-and-tube heat exchanger (one-shell pass and one-tube pass).



(*a*) One-shell pass and two-tube passes



(*b*) Two-shell passes and four-tube passes

FIGURE 11–6

Multipass flow arrangements in shell-and-tube heat exchangers.

Circuits and Overall Thermal Heat Coefficient

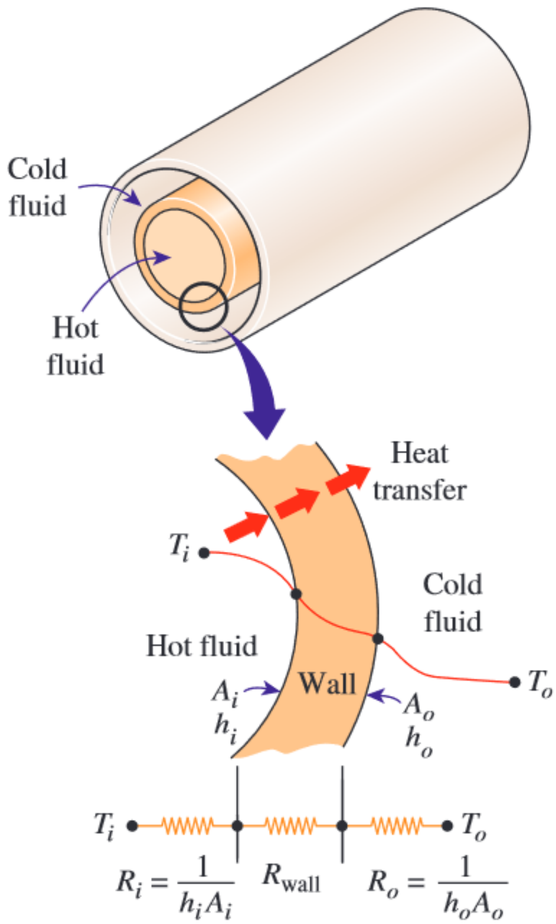


FIGURE 11–8
Thermal resistance network associated with heat transfer in a double-pipe heat exchanger.

$$R = R_{total} = R_i + R_{wall} + R_o = \frac{1}{h_i A_i} + \frac{\ln (D_o/D_i)}{2\pi k L} + \frac{1}{h_o A_o} \qquad (11\text{–}2)$$

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + R_{wall} + \frac{1}{h_o A_o} \qquad (11\text{–}4)$$

Log-Mean Temperature Difference Relations

Hint: LMTD is good for sizing

The log mean temperature difference (LMTD) method discussed in Sec. 11–4 is easy to use in heat exchanger analysis when the inlet and the outlet temperatures of the hot and cold fluids are known or can be determined from an energy balance. Once ΔT_{lm}, the mass flow rates, and the overall heat transfer coefficient are available, the heat transfer surface area of the heat exchanger can be determined from

$$\dot{Q} = UA_s \Delta T_{lm}$$

Therefore, the LMTD method is very suitable for determining the *size* of a heat exchanger to realize prescribed outlet temperatures when the mass flow rates and the inlet and outlet temperatures of the hot and cold fluids are specified.

With the LMTD method, the task is to *select* a heat exchanger that will meet the prescribed heat transfer requirements. The procedure to be followed by the selection process is:

1. Select the type of heat exchanger suitable for the application.
2. Determine any unknown inlet or outlet temperature and the heat transfer rate using an energy balance.
3. Calculate the log mean temperature difference ΔT_{lm} and the correction factor F, if necessary.
4. Obtain (select or calculate) the value of the overall heat transfer coefficient U.
5. Calculate the heat transfer surface area A_s.

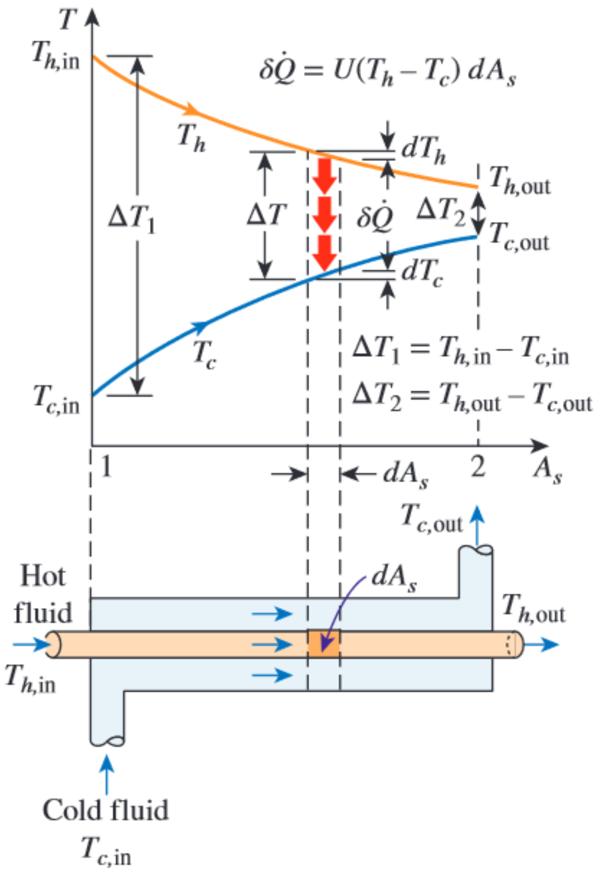
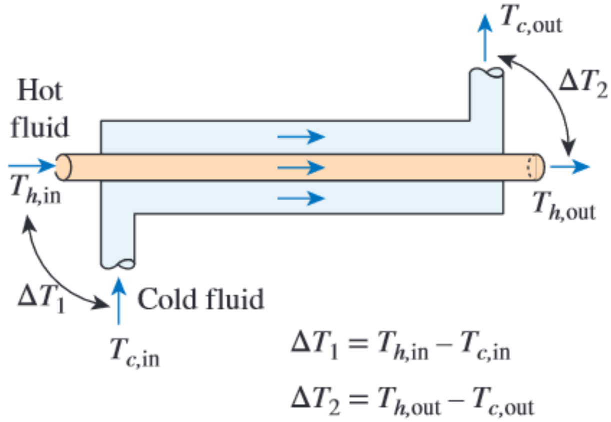
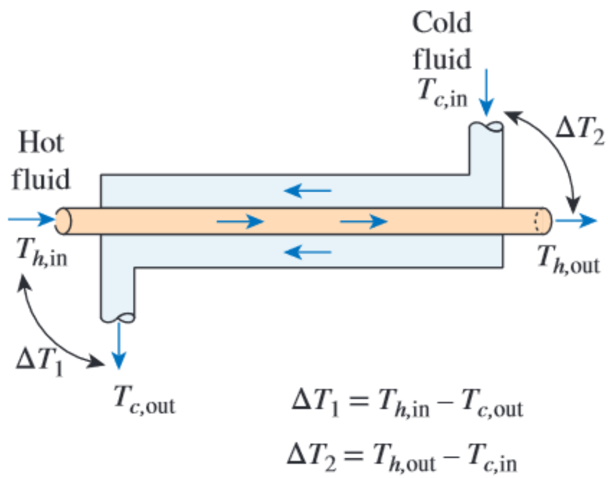


FIGURE 11–15

Variation of the fluid temperatures in a parallel-flow double-pipe heat exchanger.



(a) Parallel-flow heat exchangers



(b) Counter-flow heat exchangers

Multipass and Crossflow Heat Exchangers: Use of a Correction Factor

The log mean temperature difference ΔT_{lm} relation developed earlier is limited to parallel-flow and counterflow heat exchangers only. Similar relations are also developed for *crossflow* and *multipass shell-and-tube* heat exchangers, but the resulting expressions are too complicated because of the complex flow conditions.

In such cases, it is convenient to relate the equivalent temperature difference to the log mean temperature difference relation for the counterflow case as

$$\Delta T_{lm} = F \Delta T_{lm,\,CF} \tag{11–26}$$

where F is the **correction factor**, which depends on the *geometry* of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. The $\Delta T_{lm,\,CF}$ is the log mean temperature difference for the case of a *counterflow* heat exchanger with the same inlet and outlet temperatures and is determined from Eq. 11–25 by taking $\Delta T_1 = T_{h,\,in} - T_{c,\,out}$ and $\Delta T_2 = T_{h,\,out} - T_{c,\,in}$ (Fig. 11–18).

The correction factor is less than unity for a crossflow and multipass shell-and-tube heat exchanger. That is, $F \le 1$. The limiting value of $F = 1$ corresponds to the counterflow heat exchanger. Thus, the correction factor F for a heat exchanger is *a measure of deviation of the ΔT_{lm} from the corresponding values for the counterflow case.*

The correction factor F for common crossflow and shell-and-tube heat exchanger configurations is given in Fig. 11–19 versus two temperature ratios P and R defined as

$$P = \frac{t_2 - t_1}{T_1 - t_1} \tag{11–27}$$

and

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}c_p)_{\text{tube side}}}{(\dot{m}c_p)_{\text{shell side}}} \tag{11–28}$$

where the subscripts 1 and 2 represent the *inlet* and *outlet,* respectively. Note that for a shell-and-tube heat exchanger, T and t represent the *shell-* and *tube-side* temperatures, respectively, as shown in the correction factor charts.

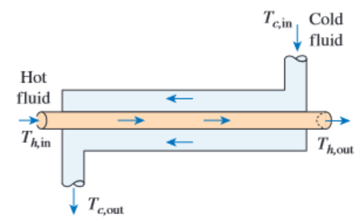
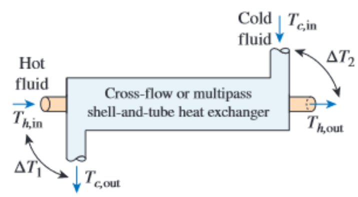


FIGURE 11–17
The variation of the fluid temperatures in a counterflow double-pipe heat exchanger.



Heat transfer rate:

$$\dot{Q} = UA_s F \Delta T_{lm,CF}$$

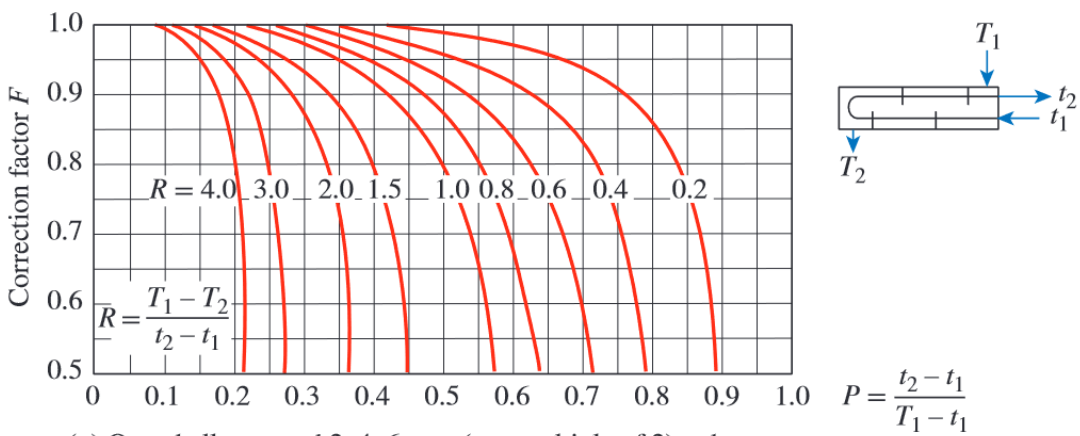
where $\Delta T_{lm,CF} = \dfrac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$

$\Delta T_1 = T_{h,in} - T_{c,out}$

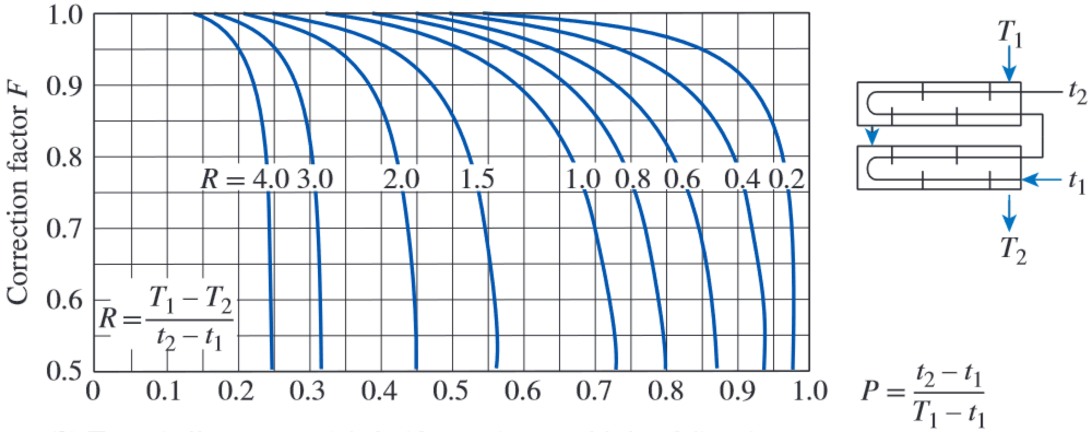
$\Delta T_2 = T_{h,out} - T_{c,in}$

and $F = \ldots$ (Fig. 11–19)

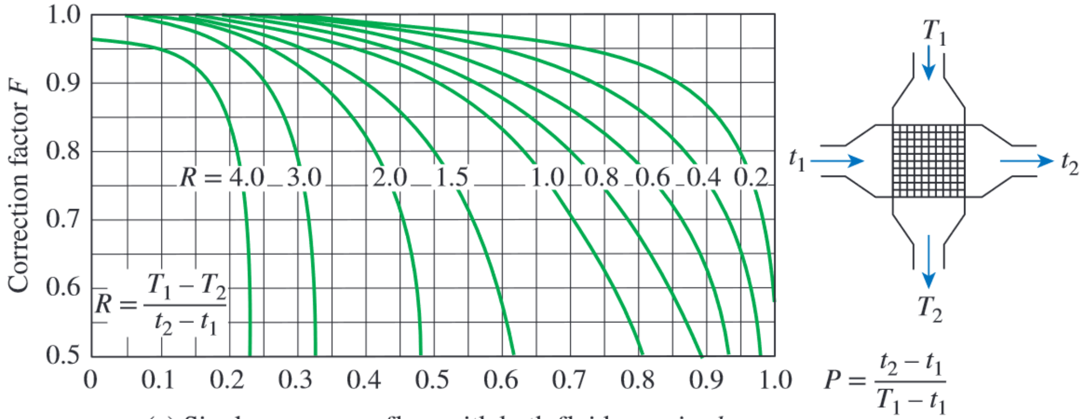
FIGURE 11–18
The determination of the heat transfer rate for crossflow and multipass shell-and-tube heat exchangers using the correction factor.



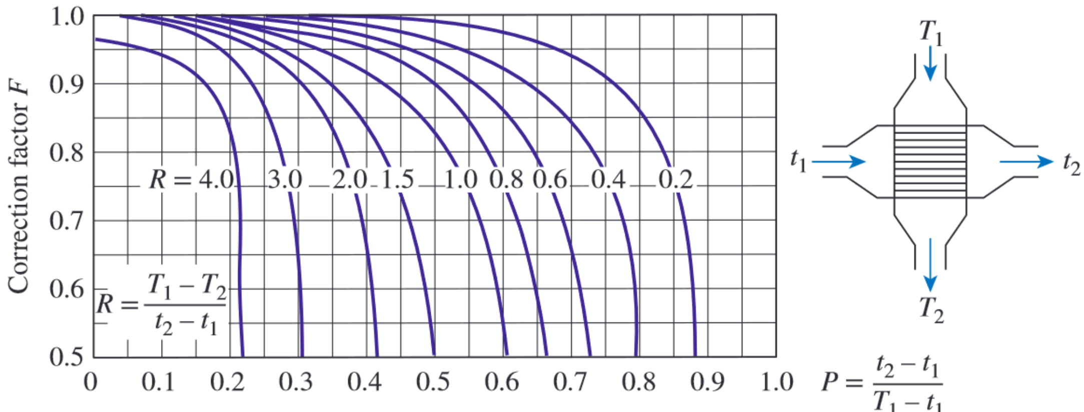
(*a*) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes



(b) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes



(c) Single-pass cross-flow with both fluids *unmixed*



(d) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

Effectiveness-NTU Relations

Heat transfer in a heat exchanger usually involves convection in each fluid and conduction through the wall separating the two fluids. In the analysis of heat exchangers, it is convenient to work with an *overall heat transfer coefficient* U or a *total thermal resistance* R, expressed as

$$\frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o} = R = \frac{1}{h_iA_i} + R_{\text{wall}} + \frac{1}{h_oA_o}$$

where the subscripts i and o stand for the inner and outer surfaces of the wall that separates the two fluids, respectively. When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, the relation simplifies to

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o}$$

where $U \approx U_i \approx U_o$. The effects of fouling on both the inner and the outer surfaces of the tubes of a heat exchanger can be accounted for by

$$\frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o} = R$$
$$= \frac{1}{h_iA_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{f,o}}{A_o} + \frac{1}{h_oA_o}$$

where $A_i = \pi D_i L$ and $A_o = \pi D_o L$ are the areas of the inner and outer surfaces and $R_{f,i}$ and $R_{f,o}$ are the fouling factors at those surfaces.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}}$$

$$\Delta T_{\max} = T_{h,\text{in}} - T_{c,\text{in}}$$

$$\dot{Q}_{\max} = C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})$$

Typical calculations are:

$$C_c = \dot{m}_c c_{pc} = 104.5 \text{ kW/K}$$

$$C_h = \dot{m}_h c_{ph} = 92 \text{ kW/K}$$

$$C_{\min} = 92 \text{ kW/K}$$

$$\Delta T_{\max} = T_{h,\text{in}} - T_{c,\text{in}} = 110°\text{C}$$

$$\dot{Q}_{\max} = C_{\min}\, \Delta T_{\max} = 10{,}120 \text{ kW}$$

TABLE 11.3 Heat exchanger effectiveness relations [5]

Flow Arrangement	Relation		
Parallel flow	$\varepsilon = \dfrac{1 - \exp[-\text{NTU}(1 + C_r)]}{1 + C_r}$		(11.28a)
Counterflow	$\varepsilon = \dfrac{1 - \exp[-\text{NTU}(1 - C_r)]}{1 - C_r \exp[-\text{NTU}(1 - C_r)]}$	$(C_r < 1)$	
	$\varepsilon = \dfrac{\text{NTU}}{1 + \text{NTU}}$	$(C_r = 1)$	(11.29a)
Shell-and-tube			
One shell pass (2, 4, . . . tube passes)	$\varepsilon_1 = 2\left\{1 + C_r + (1 + C_r^2)^{1/2} \times \dfrac{1 + \exp[-(\text{NTU})_1(1 + C_r^2)^{1/2}]}{1 - \exp[-(\text{NTU})_1(1 + C_r^2)^{1/2}]}\right\}^{-1}$		(11.30a)
n shell passes ($2n$, $4n$, . . . tube passes)	$\varepsilon = \left[\left(\dfrac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1}\right)^n - 1\right]\left[\left(\dfrac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1}\right)^n - C_r\right]^{-1}$		(11.31a)
Cross-flow (single pass)			
Both fluids unmixed	$\varepsilon = 1 - \exp\left[\left(\dfrac{1}{C_r}\right)(\text{NTU})^{0.22}\{\exp[-C_r(\text{NTU})^{0.78}] - 1\}\right]$		(11.32)
$C_{\max}$ (mixed), $C_{\min}$ (unmixed)	$\varepsilon = \left(\dfrac{1}{C_r}\right)(1 - \exp\{-C_r[1 - \exp(-\text{NTU})]\})$		(11.33a)
$C_{\min}$ (mixed), $C_{\max}$ (unmixed)	$\varepsilon = 1 - \exp(-C_r^{-1}\{1 - \exp[-C_r(\text{NTU})]\})$		(11.34a)
All exchangers ($C_r = 0$)	$\varepsilon = 1 - \exp(-\text{NTU})$		(11.35a)

Effectiveness relations for heat exchangers: $\text{NTU} = UA_s/C_{\min}$ and $c = C_{\min}/C_{\max} = (\dot{m}c_p)_{\min}/(\dot{m}c_p)_{\max}$

Heat Exchanger Type	Effectiveness Relation
1 *Double pipe:*	
Parallel-flow	$\varepsilon = \dfrac{1 - \exp[-\text{NTU}(1+c)]}{1+c}$
Counterflow	$\varepsilon = \dfrac{1 - \exp[-\text{NTU}(1-c)]}{1 - c\exp[-\text{NTU}(1-c)]}$ (for $c < 1$) $\varepsilon = \dfrac{\text{NTU}}{1+\text{NTU}}$ (for $c = 1$)
2 *Shell-and-tube:* One-shell pass 2, 4, . . . tube passes	$\varepsilon_1 = 2\left\{1 + c + \sqrt{1+c^2}\,\dfrac{1+\exp\left[-\text{NTU}_1\sqrt{1+c^2}\right]}{1-\exp\left[-\text{NTU}_1\sqrt{1+c^2}\right]}\right\}^{-1}$
n-shell passes 2*n*, 4*n*, . . . tube passes	$\varepsilon = \left[\left(\dfrac{1-\varepsilon_1 c}{1-\varepsilon_1}\right)^n - 1\right]\left[\left(\dfrac{1-\varepsilon_1 c}{1-\varepsilon_1}\right)^n - c\right]^{-1}$
3 Crossflow (*single-pass*) Both fluids unmixed	$\varepsilon = 1 - \exp\left\{\dfrac{\text{NTU}^{0.22}}{c}\left[\exp(-c\text{NTU}^{0.78}) - 1\right]\right\}$
$C_{\max}$ mixed, $C_{\min}$ unmixed	$\varepsilon = \dfrac{1}{c}(1 - \exp\{-c[1 - \exp(-\text{NTU})]\})$
$C_{\min}$ mixed, $C_{\max}$ unmixed	$\varepsilon = 1 - \exp\left\{-\dfrac{1}{c}[1 - \exp(-c\,\text{NTU})]\right\}$
4 All heat exchangers with $c = 0$	$\varepsilon = 1 - \exp(-\text{NTU})$

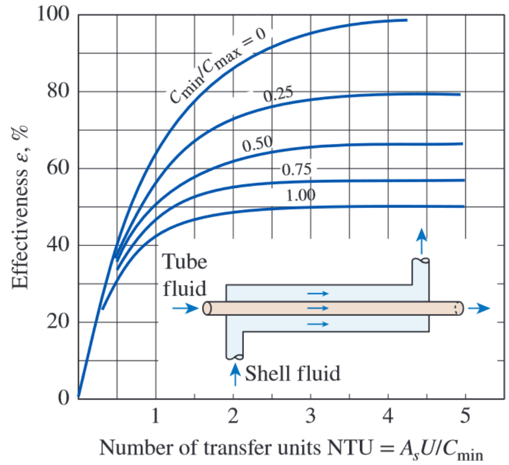
Table 11.4 Heat exchanger NTU relations

Flow Arrangement	Relation		
Parallel flow	$\text{NTU} = -\dfrac{\ln[1-\varepsilon(1+C_r)]}{1+C_r}$		(11.28b)
Counterflow	$\text{NTU} = \dfrac{1}{C_r - 1}\ln\left(\dfrac{\varepsilon - 1}{\varepsilon C_r - 1}\right)$	$(C_r < 1)$	
	$\text{NTU} = \dfrac{\varepsilon}{1-\varepsilon}$	$(C_r = 1)$	(11.29b)
Shell-and-tube			
One shell pass (2, 4, . . . tube passes)	$(\text{NTU})_1 = -(1+C_r^2)^{-1/2}\ln\left(\dfrac{E-1}{E+1}\right)$		(11.30b)
	$E = \dfrac{2/\varepsilon_1 - (1+C_r)}{(1+C_r^2)^{1/2}}$		(11.30c)
n shell passes ($2n$, $4n$, . . . tube passes)	Use Equations 11.30b and 11.30c with		
	$\varepsilon_1 = \dfrac{F-1}{F-C_r} \quad F = \left(\dfrac{\varepsilon C_r - 1}{\varepsilon - 1}\right)^{1/n} \quad \text{NTU} = n(\text{NTU})_1$		(11.31b, c, d)
Cross-flow (single pass)			
$C_{\max}$ (mixed), $C_{\min}$ (unmixed)	$\text{NTU} = -\ln\left[1 + \left(\dfrac{1}{C_r}\right)\ln(1-\varepsilon C_r)\right]$		(11.33b)
$C_{\min}$ (mixed), $C_{\max}$ (unmixed)	$\text{NTU} = -\left(\dfrac{1}{C_r}\right)\ln[C_r\ln(1-\varepsilon)+1]$		(11.34b)
All exchangers ($C_r = 0$)	$\text{NTU} = -\ln(1-\varepsilon)$		(11.35b)

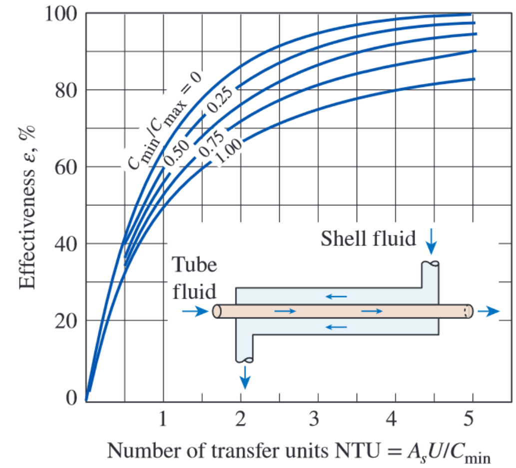
TABLE 11–5

NTU relations for heat exchangers: $\text{NTU} = UA_s/C_{\min}$ and $c = C_{\min}/C_{\max} = (\dot{m}c_p)_{\min}/(\dot{m}c_p)_{\max}$

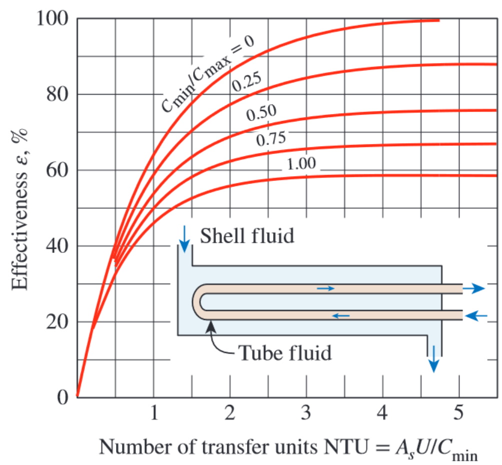
Heat Exchanger Type		NTU Relation
1	*Double-pipe:*	
	Parallel-flow	$\text{NTU} = -\dfrac{\ln[1 - \varepsilon(1 + c)]}{1 + c}$
	Counterflow	$\text{NTU} = \dfrac{1}{c - 1}\ln\left(\dfrac{\varepsilon - 1}{\varepsilon c - 1}\right)$ (for $c < 1$)
		$\text{NTU} = \dfrac{\varepsilon}{1 - \varepsilon}$ (for $c = 1$)
2	*Shell and tube:* One-shell pass 2, 4, . . . tube passes	$\text{NTU}_1 = -\dfrac{1}{\sqrt{1 + c^2}}\ln\left(\dfrac{2/\varepsilon_1 - 1 - c - \sqrt{1 + c^2}}{2/\varepsilon_1 - 1 - c - \sqrt{1 + c^2}}\right)$
	n-shell passes $2n$, $4n$, . . . tube passes	$\text{NTU} = n(\text{NTU})_1$ To find effectiveness of the heat exchanger with one-shell pass use, $\varepsilon_1 = \dfrac{F - 1}{F - c}$ where $F = \left(\dfrac{\varepsilon c - 1}{\varepsilon - 1}\right)^{1/n}$
3	*Crossflow (single-pass):*	
	$C_{\max}$ mixed, $C_{\min}$ unmixed	$\text{NTU} = -\ln\left[1 + \dfrac{\ln(1 + \varepsilon c)}{c}\right]$
	$C_{\min}$ mixed, $C_{\max}$ unmixed	$\text{NTU} = -\dfrac{\ln[c\ln(1 - \varepsilon) + 1]}{c}$
4	All heat exchangers with $c = 0$	$\text{NTU} = -\ln(1 - \varepsilon)$



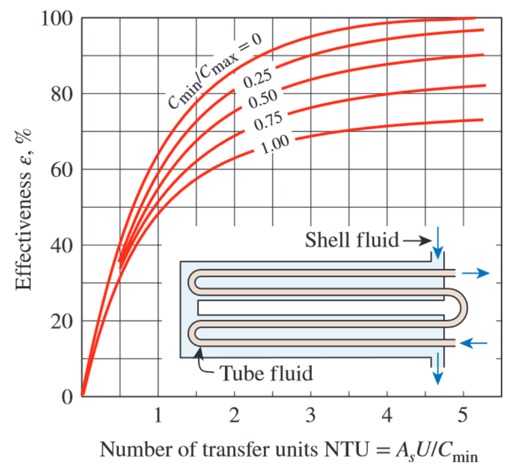
(*a*) Parallel-flow



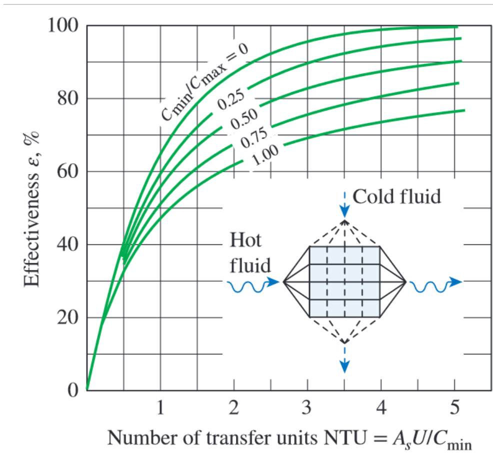
(*b*) Counter-flow



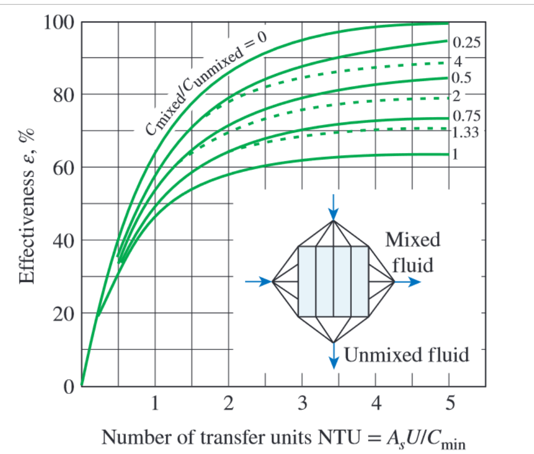
(*c*) One-shell pass and 2, 4, 6, … tube passes



(*d*) Two-shell passes and 4, 8, 12, … tube passes



(*e*) Cross-flow with both fluids unmixed



(*f*) Cross-flow with one fluid mixed and the other unmixed

We make these observations from the effectiveness relations and charts already given:

1. The value of the effectiveness ranges from 0 to 1. It increases rapidly with NTU for small values (up to about NTU = 1.5) but rather slowly for

larger values. Therefore, the use of a heat exchanger with a large NTU (usually larger than 3) and thus a large size cannot be justified economically, since a large increase in NTU in this case corresponds to a small increase in effectiveness. Thus, a heat exchanger with a very high effectiveness may be desirable from a heat transfer point of view but undesirable from an economical point of view.

2. For a given NTU and capacity ratio $c = C_{min}/C_{max}$, the *counterflow* heat exchanger has the *highest* effectiveness, followed closely by the cross-flow heat exchangers with both fluids unmixed. As you might expect, the lowest effectiveness values are encountered in parallel-flow heat exchangers (Fig. 11–28).
3. The effectiveness of a heat exchanger is independent of the capacity ratio c for NTU values of less than about 0.3.
4. The value of the capacity ratio c ranges between 0 and 1. For a given NTU, the effectiveness becomes a *maximum* for $c = 0$ and a *minimum* for $c = 1$. The case $c = C_{min}/C_{max} \to 0$ corresponds to $C_{max} \to \infty$, which is realized during a phase-change process in a *condenser* or *boiler.* All effectiveness relations in this case reduce to

$$\varepsilon = \varepsilon_{max} = 1 - \exp(-\text{NTU}) \tag{11–41}$$

regardless of the type of heat exchanger (Fig. 11–29). Note that the temperature of the condensing or boiling fluid remains constant in this case. The effectiveness is the *lowest* in the other limiting case of $c = C_{min}/C_{max} = 1$, which is realized when the heat capacity rates of the two fluids are equal. In industrial applications such as hot water cooling or chilled water heating pertaining to the heating, ventilating, and air conditioning (HVAC) industry, the hot or cold fluid is channeled from one location to another with the pipe immersed in ambient environment. Typical examples of this flow situation are the fluid-carrying heat exchanger pipe exposed to ambient air or submerged in large liquid media such as a lake. In such cases, since the effective change in reference temperature of the ambient environment is virtually zero and the relative mass of the ambient environment is infinitely large, the maximum heat capacity rate C_{max} is a very large number. Thus the capacity ratio, $c = C_{min}/C_{max} \to 0$ and hence Eq. 11–41 is applicable in cases when the heat exchanger is in contact with the ambient environment.