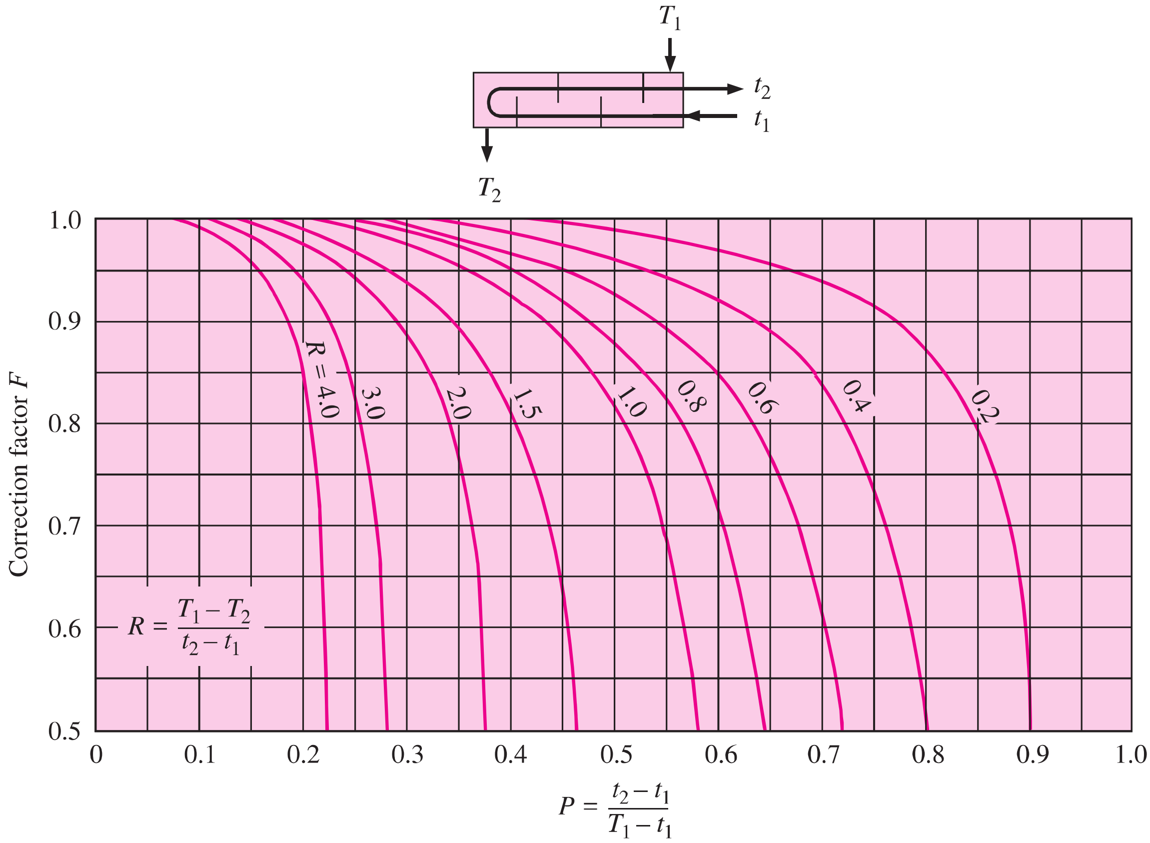
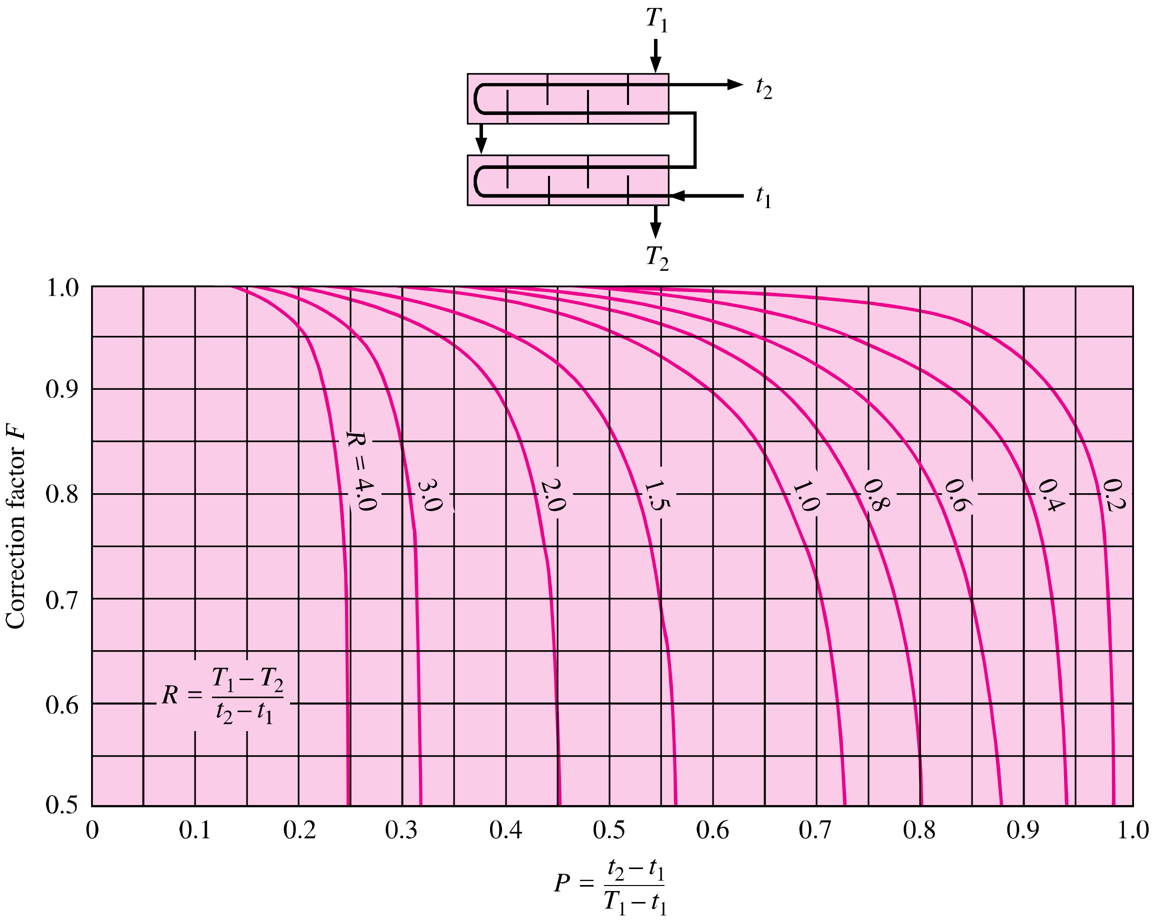


T1
t2
t1
T2
1.0
0.9
0.8
0.7
0.6
0.5
Correction factor F
R = 4.0
3.0
2.0
1.5
1.0
0.8
0.6
0.4
0.2
R = (T1 − T2)/(t2 − t1)
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
P = (t2 − t1)/(T1 − t1)



T1
t2
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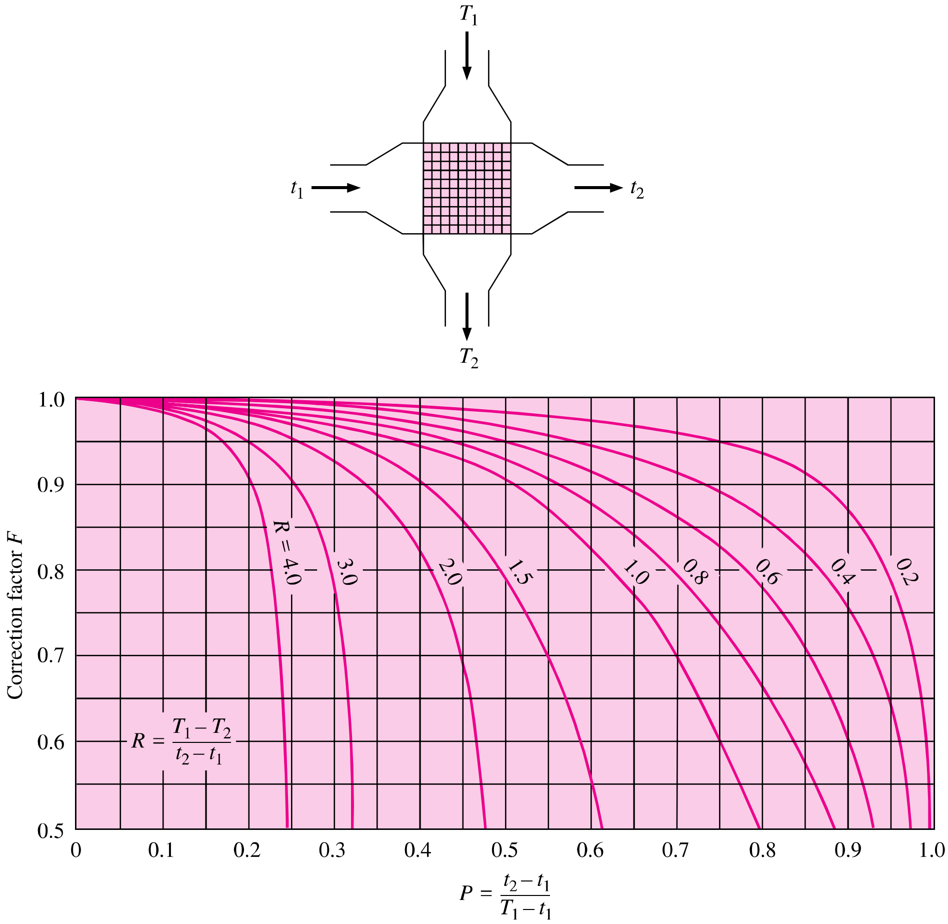
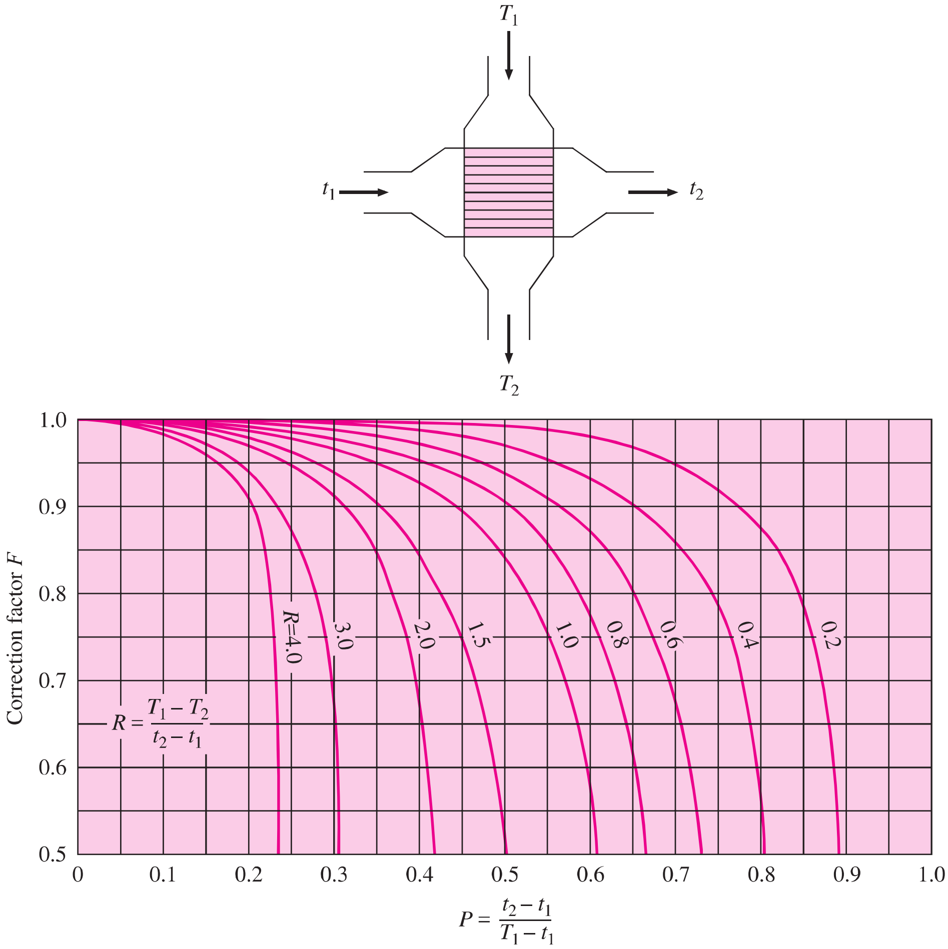
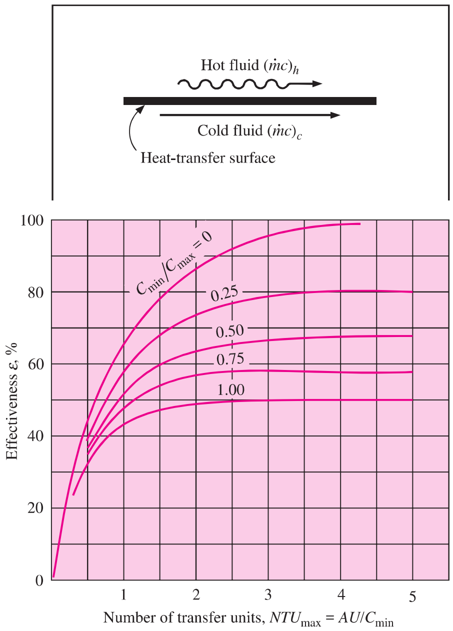
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
P = (t2 − t1)/(T1 − t1)



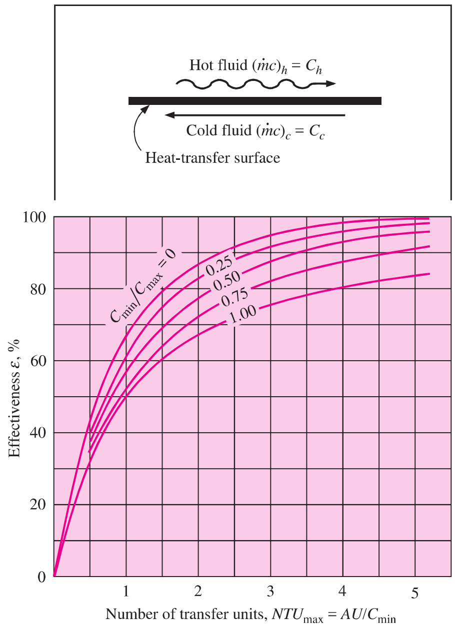
T_1
t_1
t_2
T_2
Correction factor F
$R = \frac{T_1 - T_2}{t_2 - t_1}$
R = 4.0
3.0
2.0
1.5
1.0
0.8
0.6
0.4
0.2
$P = \frac{t_2 - t_1}{T_1 - t_1}$



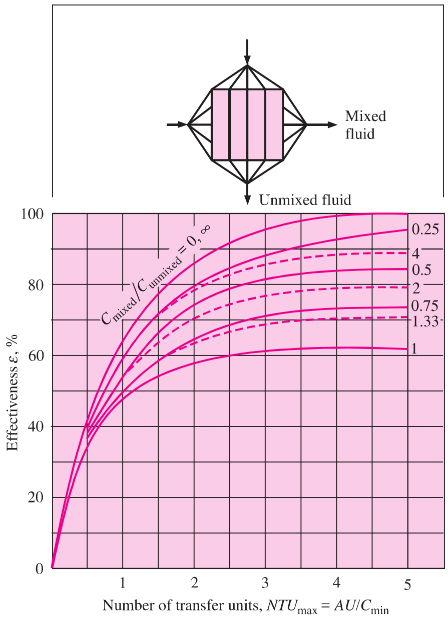
T_1
t_1
t_2
T_2
Correction factor F
$R = \frac{T_1 - T_2}{t_2 - t_1}$
R=4.0
3.0
2.0
1.5
1.0
0.8
0.6
0.4
0.2
$P = \frac{t_2 - t_1}{T_1 - t_1}$



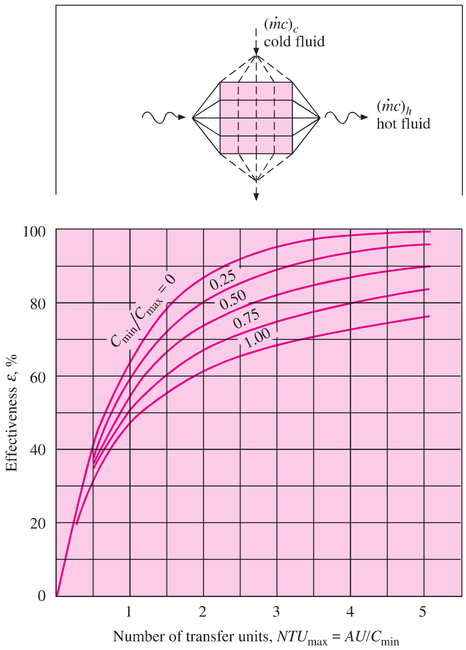
Hot fluid $(\dot{m}c)_h$
Cold fluid $(\dot{m}c)_c$
Heat-transfer surface
$C_{min}/C_{max} = 0$
0.25
0.50
0.75
1.00
Effectiveness ε, %
Number of transfer units, $NTU_{max} = AU/C_{min}$



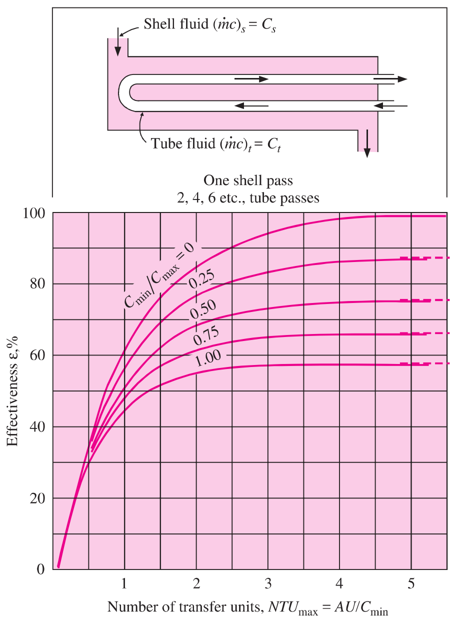
Hot fluid $(\dot{m}c)_h = C_h$
Cold fluid $(\dot{m}c)_c = C_c$
Heat-transfer surface
$C_{min}/C_{max} = 0$
0.25
0.50
0.75
1.00
Effectiveness ε, %
Number of transfer units, $NTU_{max} = AU/C_{min}$



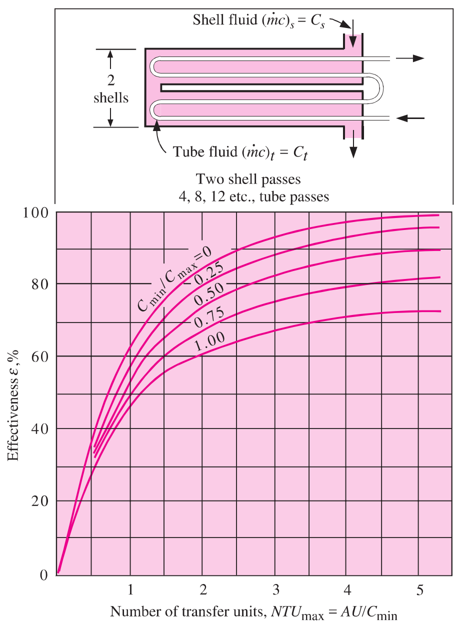
Mixed fluid
Unmixed fluid
$C_{mixed}/C_{unmixed} = 0, \infty$
0.25
4
0.5
2
0.75
1.33
1
Effectiveness ε, %
Number of transfer units, $NTU_{max} = AU/C_{min}$



$(\dot{m}c)_c$ cold fluid
$(\dot{m}c)_h$ hot fluid
$C_{min}/C_{max} = 0$
0.25
0.50
0.75
1.00
Effectiveness ε, %
Number of transfer units, $NTU_{max} = AU/C_{min}$

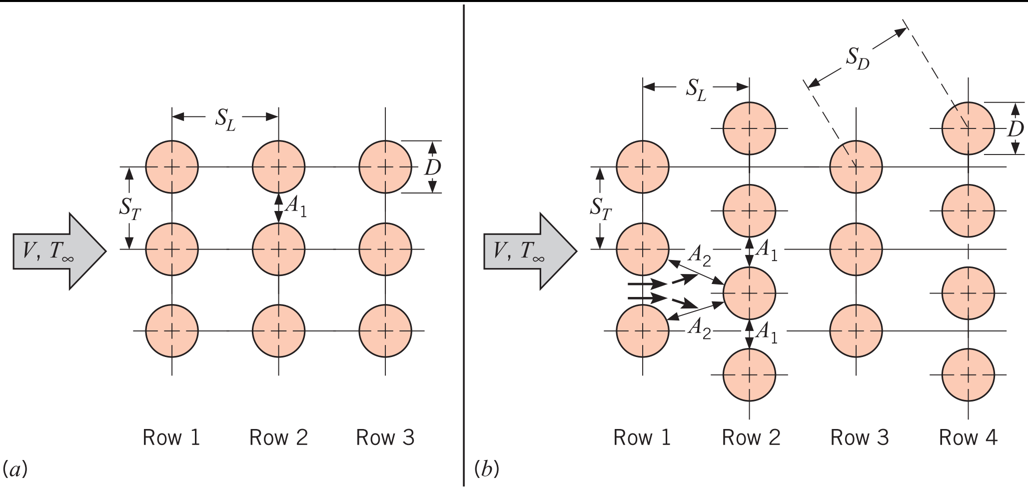


Shell fluid $(\dot{m}c)_s = C_s$
Tube fluid $(\dot{m}c)_t = C_t$
One shell pass
2, 4, 6 etc., tube passes
$C_{min}/C_{max} = 0$
0.25
0.50
0.75
1.00
Effectiveness ε,%
Number of transfer units, $NTU_{max} = AU/C_{min}$



Shell fluid $(\dot{m}c)_s = C_s$
2 shells
Tube fluid $(\dot{m}c)_t = C_t$
Two shell passes
4, 8, 12 etc., tube passes
$C_{min}/C_{max} = 0$
0.25
0.50
0.75
1.00
Effectiveness ε,%
Number of transfer units, $NTU_{max} = AU/C_{min}$

Cross-Flow Heat Exchanger



- $Re_{D,max} = \frac{V_{max} D}{\nu}$

(a) In-line rows, $V_{max} = u_\infty \left[\frac{S_T}{S_T - D}\right]$

(b) Staggered row, $V_{max} = max\left(u_\infty \left[\frac{S_T}{S_T - D}\right],\ u_\infty \left[\frac{S_T}{2(S_D - D)}\right]\right)$, $S_D = \sqrt{S_L^2 + (S_T/2)^2}$

- $Nu_m \equiv \frac{h_m D}{k} = C_1 Re_{D,max}^m Pr^{0.36} \left(\frac{Pr}{Pr_w}\right)^{1/4}$

Configuratio	$Re_{D,\max}$	C_1	m
Aligned	10–10^2	0.80	0.40
Staggered	10–10^2	0.90	0.40
Aligned	10^2–10^3	Approximate as a single	
Staggered	10^2–10^3	(isolated) cylinder	
Aligned	10^3–2×10^5	0.27	0.63
$(S_T/S_L > 0.7)$[a]			
Staggered	10^3–2×10^5	$0.35(S_T/S_L)^{1/5}$	0.60
$(S_T/S_L < 2)$			
Staggered	10^3–2×10^5	0.40	0.60
$(S_T/S_L > 2)$			
Aligned	2×10^5–2×10^6	0.021	0.84
Staggered	2×10^5–2×10^6	0.022	0.84

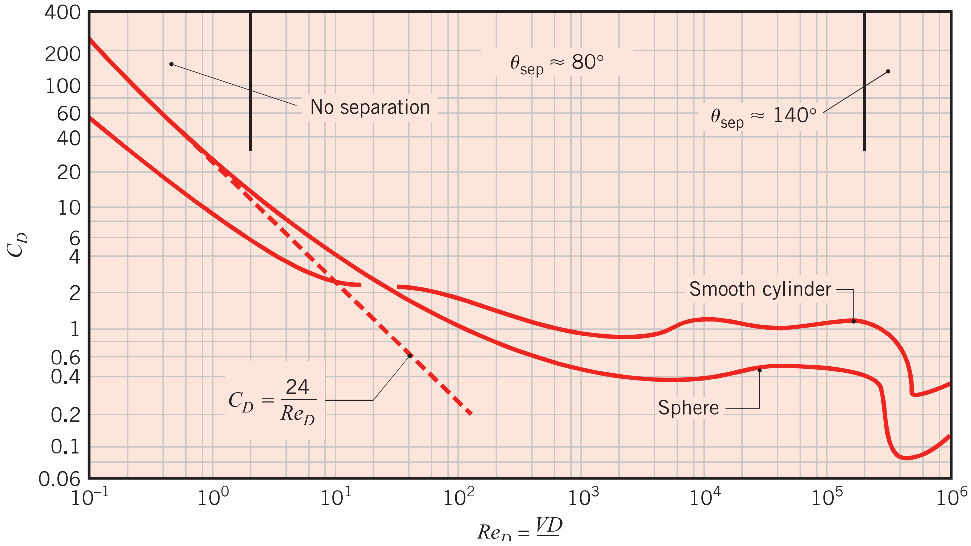
[a]For $S_T/S_L < 0.7$, heat transfer is inefficient and aligned tubes should not be used.

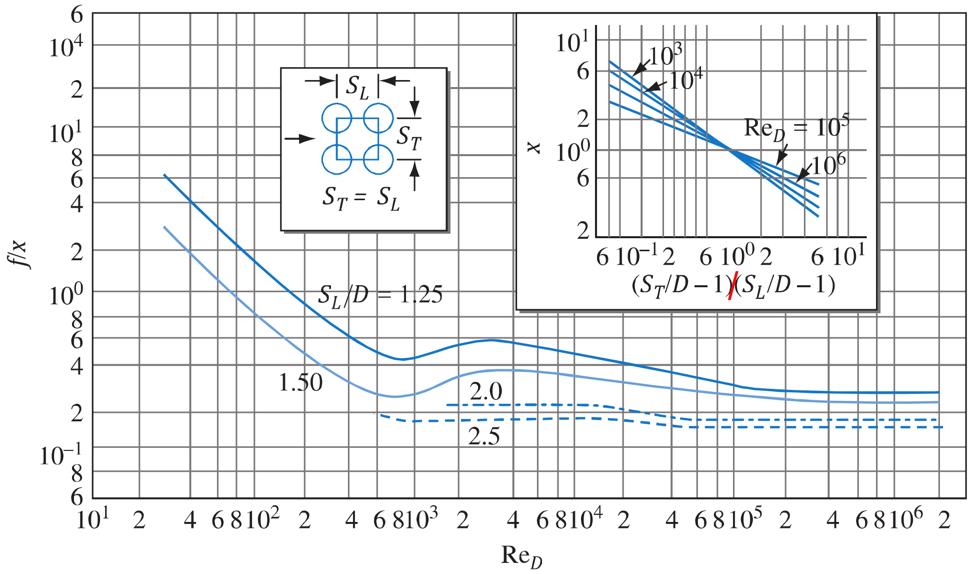
- $Nu_m|_{(N<20)} = C_2 Nu_m|_{(N \geq 20)}$

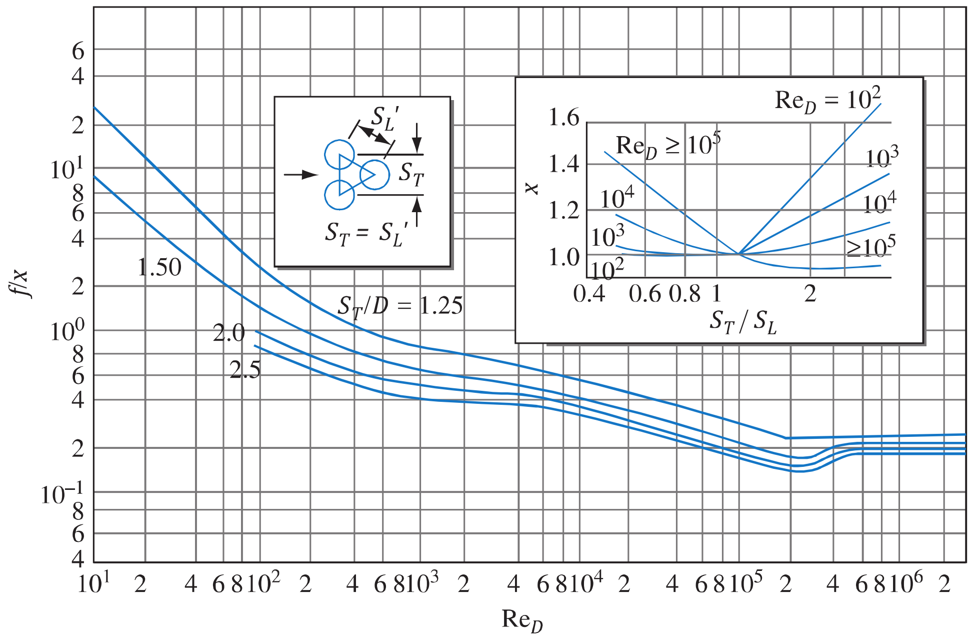
N	2	3	4	5	6	8	10	16	20
Staggered	0.77	0.84	0.89	0.92	0.94	0.97	0.98	0.99	1.0
In-line	0.70	0.80	0.90	0.92	0.94	0.97	0.98	0.99	1.0

- Pressure drop, $\Delta P = f\left(\frac{1}{2}\rho V_{max}^2\right) N_L$
- Pumping work, $\dot{W}_p = \dot{V}\Delta P = V_{av} A \Delta P$
- Motor power, $\dot{W}_{motor} = \frac{\dot{W}_p}{\eta_{pump}}$

$$F_D = C_D A \left(\frac{1}{2}\rho u_\infty^2\right)$$







Double Pipe Heat Exchanger

- Laminar: $Nu = 1.86(Gz)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}; \quad Gz = \frac{RePr}{L/D}$
- Turbulent: $Nu = 0.023Re^{0.8}Pr^n; \quad n = \begin{cases} 0.4 & : heating \\ 0.3 & : cooling \end{cases}$
- Friction factor, $f = \begin{cases} \xi_{corr}\ [64/Re] & : \ laminar\ flow \\ (1.82\log_{10} Re - 1.64)^{-2} & : \ turbulent\ flow \end{cases}$

Pipe area

- Equivalent diameter, $De_p = Dh_p = ID_p$
- Pressure drop, $\Delta P_p = f_p \frac{L}{ID_p}(\frac{1}{2}\rho_p V_p^2)$, $\xi_{corr} = 1$

Annular area

- $Dh_a = ID_a - OD_p, \quad De_a = \frac{ID_a^2 - OD_p^2}{OD_p}$
- $\frac{1}{\xi_{corr}} = \frac{1+\kappa^2}{(1-\kappa)^2} + \frac{1+\kappa}{(1-\kappa)\ln(\kappa)}$, $\kappa = OD_p/ID_a$
- $\Delta P_a = (f_a \frac{L}{Dh_a} + 1)(\frac{1}{2}\rho_a V_a^2)$

Shell & Tube Heat Exchanger

- Tube pitch ratio, $PR \equiv P_T/OD_t$, $1.2 \leq PR \leq 1.5$
- Preferred tube length, L:
  6 ft (1.83 m), 8 ft (2.44 m), 12 ft (3.66 m), 16 ft (4.88 m), 20 ft (6.1 m), 24 ft (7.32 m).
- Total outside tube area, $A_o = N_t(\pi OD_t L)$
- $CL = \begin{cases} 1.0 & for\ 90^o,\ 45^o \\ 0.87 & for\ 30^o,\ 60^o \end{cases}$
- $CTP = \begin{cases} 0.93 & : \ 1\ tube-pass \\ 0.90 & : \ 2\ tube-passes \\ 0.85 & : \ 3\ tube-passes \end{cases}$
- Shell inside diameter, $D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_o(PR)^2 OD_t}{L}\right]^{1/2}$

Tube-side

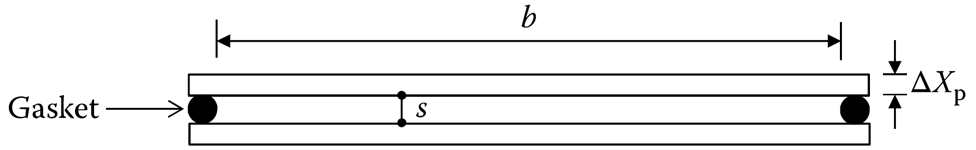
- Total tube x-section area, $A_t = \left(\frac{N_t}{N_p}\right)\frac{\pi}{4}ID_t^2 \quad N_p =$ no. of pass
- Laminar: $Nu = 1.86(Gz)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}; \quad Gz = \frac{RePr}{L/OD_t}$
- Turbulent: $Nu = 0.023Re^{0.8}Pr^n; \quad n = \begin{cases} 0.4 & : heating \\ 0.3 & : cooling \end{cases}$
- For smooth pipes, $f_t = \begin{cases} 0.316Re^{-0.25} & : \ Re \leq 2\times10^4 \\ 0.184Re^{-0.20} & : \ 2\times10^4 \leq Re \leq 3\times10^5 \end{cases}$
- Pressure drop, $\Delta P_t = N_p\left(f_t \frac{L}{ID_t} + 4\right)(\frac{1}{2}\rho_t V_t^2)$

Shell-side

$$Dh_s = \frac{4A}{P} = \begin{cases} \frac{4P_T^2}{\pi OD_t} - OD_t & : \ square\ pitch \\ \frac{2\sqrt{3}P_T^2}{\pi OD_t} - OD_t & : \ triangular\ pitch \end{cases}$$

- $0.4D_s \leq B \leq 0.6D_s, \quad B =$ baffle spacing, Number of baffles, $N_b = \frac{L}{B} - 1$
- Bundle cross-flow area, $A_s = \frac{D_s CB}{P_T}$
- $Nu = 0.36Re_s^{0.55}Pr^{1/3}$, $\quad Re_s = V_s Dh_s/\nu_s$
- $f_s = \exp\left[0.576 - 0.19\ln(Re_s)\right]$
- $\Delta P_s = f_s(N_b + 1)\frac{D_s}{D_e}(\frac{1}{2}\rho_s V_s^2)$

Plate & Frame Heat Exchanger



- $NTU = \frac{U_o A_o N_s}{(\dot{m} C_p)_{min}}, \quad A_o = bl$
- Effective $NTU = F_{corr} NTU, \quad F_{corr} = 1 - 0.0166 NTU$
- $\frac{1}{U_o} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{\Delta X_p}{k} + R_{di} + R_{do}$
- Flow velocity, $V = \begin{cases} \frac{2\dot{m}}{\rho s b (N_s+1)} & : \text{ odd number of plates} \\ \frac{2\dot{m}}{\rho s b N_s}, \ \frac{2\dot{m}}{\rho s b (N_s+2)} & : \text{ even number of plates} \end{cases}$
- $D_h = \frac{4(cross-sectional\ area)}{wetted\ perimeter} = \frac{4sb}{2s+2b} \simeq \frac{4sb}{2b} = 2s, \quad Re = \frac{V D_h}{\nu}$
- Laminar flow: $Nu = 1.86 (Gz)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}; \ Gz = \frac{RePr}{L/D}$ $Re < 100$
- Turbulent flow: $Nu = 0.374 Re^{0.668} Pr^{1/3};$ $Re > 100$
- Friction factor, $f = \begin{cases} \frac{280}{Re} & : 1 < Re < 10 \\ \frac{100}{Re^{0.589}} & : 10 < Re < 100 \\ \frac{12}{Re^{0.183}} & : Re > 100 \end{cases}$
- Pressure drop, $\Delta P = \frac{fL}{D_h}(\frac{1}{2}\rho V^2) + 1.3(\frac{1}{2}\rho V_p^2) \simeq \frac{fL}{D_h}(\frac{1}{2}\rho V^2)$

Condensation Correlations

- *Condensation on vertical surfaces*

$$h_m = 1.2 \times 0.943 \left[\frac{g\rho_l(\rho_l - \rho_v) h_{fg} k_l^3}{\mu_l (T_v - T_w) L}\right]^{1/4}$$

- *Condensation on inclined surfaces*

$$h_m = 0.943 \left[\frac{g\rho_l(\rho_l - \rho_v) h_{fg} k_l^3}{\mu_l (T_v - T_w) L} \sin\varphi\right]^{1/4}$$

- *Condensation on horizontal tube*

$$h_m = 0.725 \left[\frac{g\rho_l(\rho_l - \rho_v) h_{fg} k_l^3}{\mu_l (T_v - T_w) D}\right]^{1/4}$$

- *Condensation on horizontal tube banks*

$$h_m|_{N\ tubes} = \frac{1}{N^{1/4}} h_m|_{1\ tube}$$

- *Condensation inside horizontal tube*

$$h_m = 0.555 \left[\frac{g\rho_l(\rho_l - \rho_v) h'_{fg} k_l^3}{\mu_l (T_v - T_w) D}\right]^{1/4}$$