بسم الله الرحمن الرحيم

King Abdulaziz University
Faculty of Engineering
Mechanical Engineering Department

Basics of Rating and Thermal Design of Heat Exchangers

General notes prepared for the course

MEP 460: Design of Heat Exchangers

by

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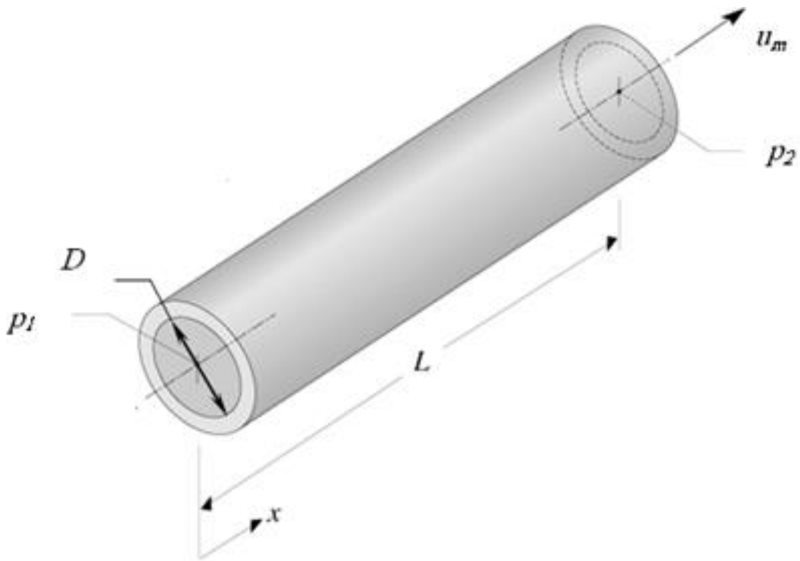
General Information



- Mass flow rate, $\dot{m}$:

$$\dot{m} = \rho u_m A_C ,$$

A_C : Flow area; u_m : Mean flow velocity normal to cross-section

- Mean bulk Temperature, T_m:

$$T_m = \frac{\int_{A_C} \rho u c_p T dA_C}{\dot{m} c_p}$$

- Newton's law of cooling for the Local Heat Flux, q''

$$q'' = h(T_s - T_m)$$

- Reynolds number, Re_D:

$$Re_D \equiv \frac{\rho u_m D_h}{\mu} = \frac{4\dot{m}}{P_w \mu}$$

- Hydraulic diameter, D_h:

$$D_h \equiv \frac{4A_C}{P_w} , \qquad P_w : \text{Wetted perimeter}$$

- For circular pipe,

$$A_C = \frac{\pi}{4} d_i^2; \quad P_w = P = \pi d_i; \quad D_h = d_i$$

- Onset of turbulence occurs at a critical Reynolds number of

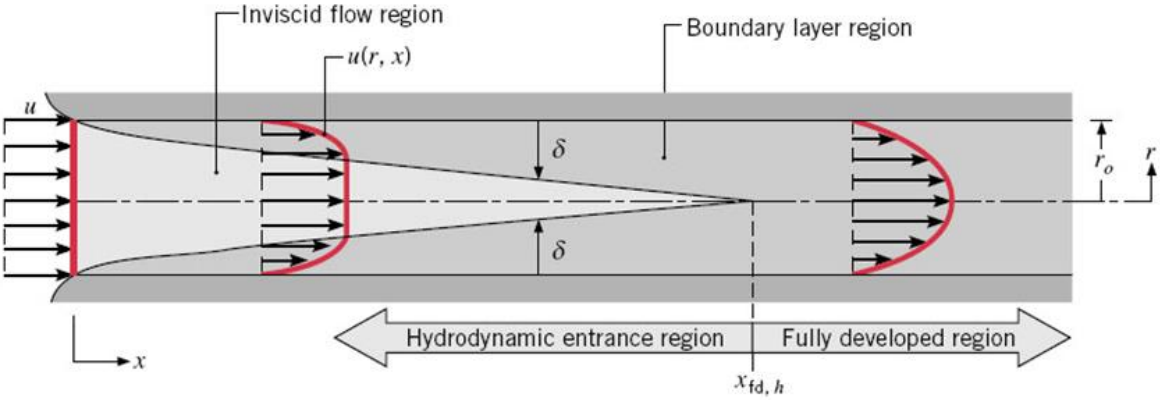
$$Re_{D,cr} \approx 2300$$

- Fully turbulent conditions occurs when

$$Re_D \geq 10{,}000$$

Fully developed flow

A) Hydrodynamically fully developed flow



Laminar, hydrodynamic boundary layer development in a circular tube

- Hydrodynamic Entry Length

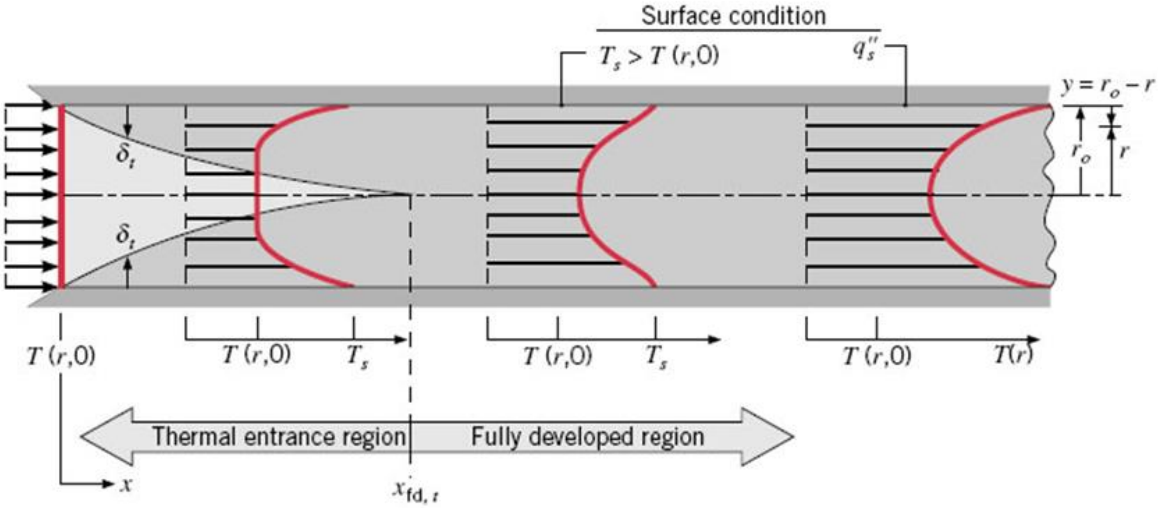
  Laminar: $x_{fd,h} / D \approx 0.05\, Re_D$

  Turbulent: $10 < x_{fd,h} / D < 60$

- Requirement for fully developed hydrodynamically flow condition:

$$\left.\frac{\partial u}{\partial x}\right|_{fd,h} = \left.\frac{d\tau_w}{\partial x}\right|_{fd,h} = 0$$

B) Thermally fully developed flow



Thermal boundary layer development in a heated circular tube

- Thermal Entry Length

  Laminar: $x_{fd,t}/D \approx 0.05\, Re_D\, Pr$

  Turbulent: $10 < x_{fd,t}/D < 60$

- For laminar flow, how do hydrodynamic and thermal entry lengths compare for a gas? And oil? A liquid metal?

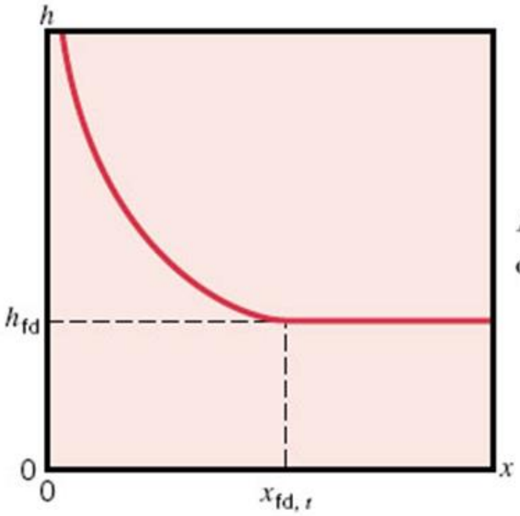
- Can a flow be developing hydrodynamically *and* be thermally fully-developed?

- Requirement for thermally fully developed flow condition:

$$\frac{\partial}{\partial x}\left(\frac{T_s(x)-T(r,x)}{T_s(x)-T_m(x)}\right)_{fd,t} = 0$$

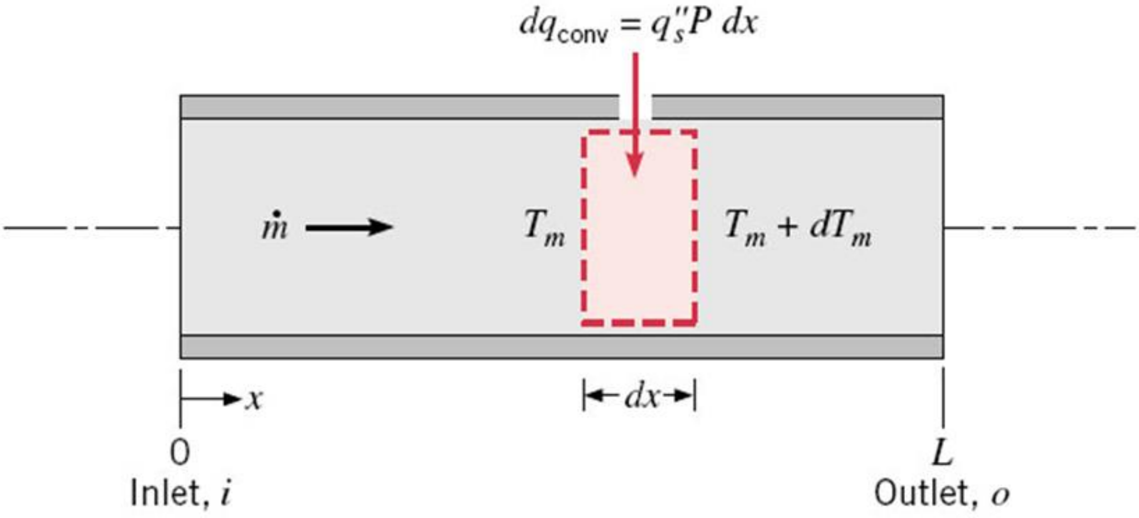
$$\frac{\partial}{\partial r}\left(\frac{T_s - T}{T_s - T_m}\right)\bigg|_{r=r_i} = -\frac{\partial T/\partial r|_{r=r_i}}{T_s - T_m} = \frac{q''_s/k}{T_s - T_m} = \frac{h}{k} \neq f(x)$$

$$h \neq f(x)$$



Axial variation of the convection heat transfer coefficient for flow in a tube.

Calculation of mean bulk temperature



Control volume for internal flow in a tube

$$dq_{conv} = \dot{m}c_p\left[\left(T_m + dT_m\right) - T_m\right] = \dot{m}c_p dT_m$$

Integrating from the tube inlet to outlet $(x = 0 \rightarrow x = L)$: $q_{conv} = \dot{m}c_p\left(T_{m,o} - T_{m,i}\right)$

Uniform Heat Flux $(q_s'' = constant)$

$$dq_{conv} = \dot{m}c_p dT_m = q_s'' P dx$$

$$\frac{dT_m}{dx} = \frac{q_s'' P}{\dot{m}c_p}$$

Integrating from the tube inlet to outlet $(x = 0 \rightarrow x = L)$

$$T_{m,o} = T_{m,i} + \frac{q_s'' PL}{\dot{m}c_p}; \quad q_s'' = h(T_s - T_m)$$

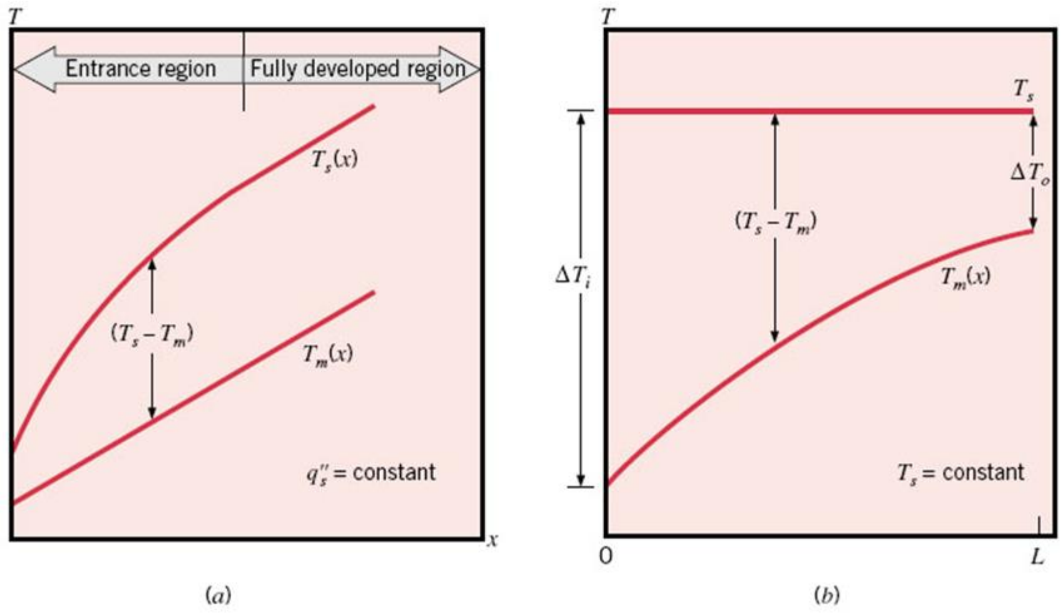
Uniform Surface Temperature $(T_s = constant)$

$$dq_{conv} = \dot{m}c_p dT_m = hP(T_s - T_m)dx$$

$$\frac{dT_m}{T_s - T_m} = \frac{hP}{\dot{m}c_p} dx$$

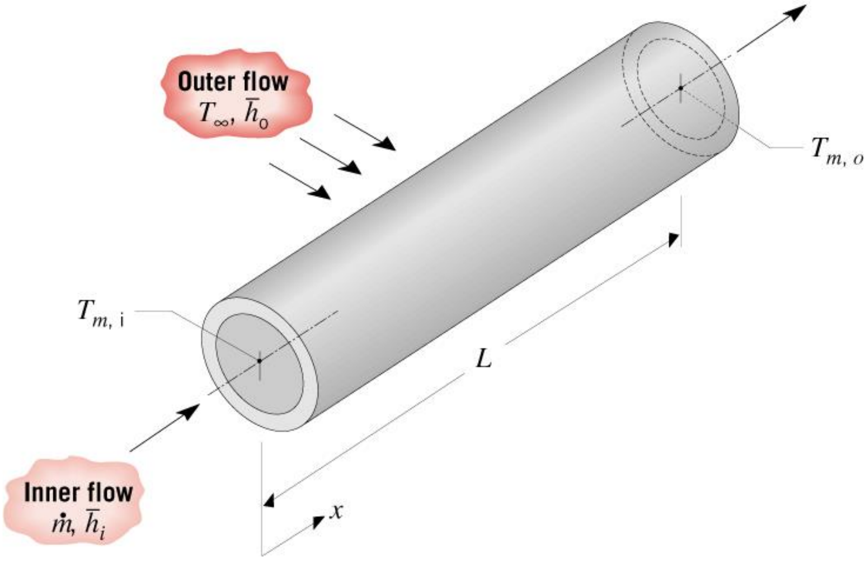
Integrating from the tube inlet to outlet $(x = 0 \rightarrow x = L)$

$$\frac{T_s - T_{m,o}}{T_s - T_{m,i}} = exp\left(-\frac{\bar{h}PL}{\dot{m}c_p}\right); \quad \bar{h} = \frac{1}{L}\int_0^L h dx$$



Axial temperature variations for heat transfer in a tube. (*a*) Constant surface heat flux. (*b*) Constant surface temperature

Uniform External Fluid Temperature $(T_\infty = constant)$



$$\frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}} = exp\left(-\frac{1}{\dot{m}c_p R_{tot}}\right)$$

$$R_{tot} = \frac{1}{\bar{h}_i A_i} + R_W + \frac{1}{\bar{h}_o A_o}$$

$$A_i = \pi d_i L\,; \qquad A_o = \pi d_o L\,; \qquad R_W = \frac{ln(d_o/d_i)}{2\pi k_{tube} L}$$

$$q_{conv} = \dot{m}c_p(T_{m,o} - T_{m,i}) = \frac{1}{R_{tot}} \frac{(T_\infty - T_{m,o}) - (T_\infty - T_{m,i})}{ln\left(\frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}}\right)}$$

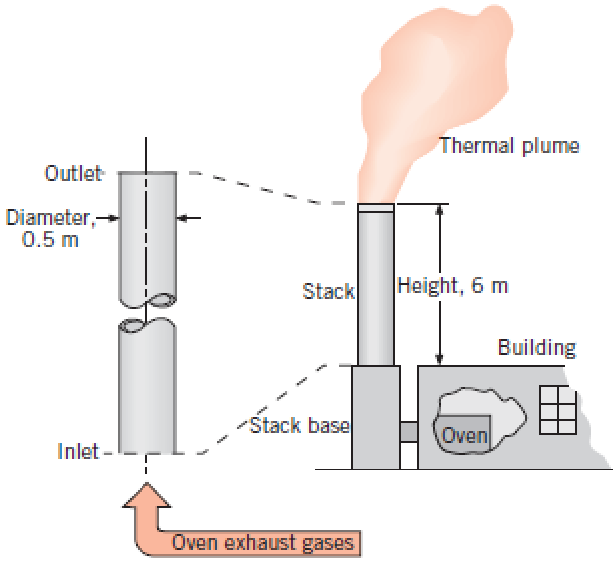
Overall heat transfer coefficients (U_i, U_o, U)

$$R_{tot} = \frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o}$$

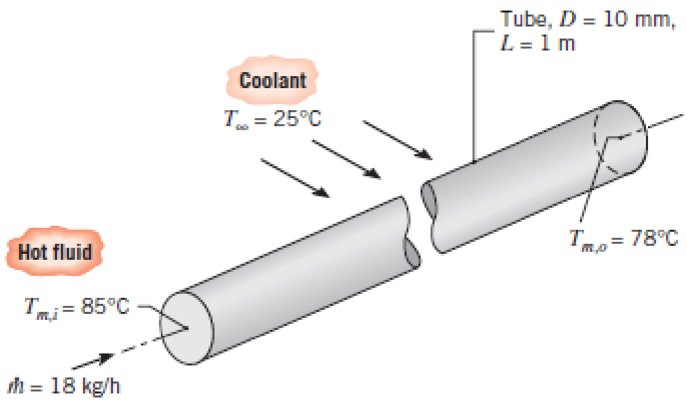
$$\frac{1}{U_o} = \frac{1}{\bar{h}_o} + \frac{d_o\,ln(d_o/d_i)}{2k_{tube}} + \left(\frac{d_o}{d_i}\right)\frac{1}{\bar{h}_i}; \quad \frac{1}{U_i} = \left(\frac{d_i}{d_o}\right)\frac{1}{\bar{h}_o} + \frac{d_i\,ln(d_o/d_i)}{2k_{tube}} + \frac{1}{\bar{h}_i}$$

Problems

P1. Exhaust gases from a wire processing oven are discharged into a tall stack, and the gas and stack surface temperatures at the outlet of the stack must be estimated. Knowledge of the outlet gas temperature $T_{m,o}$ is useful for predicting the dispersion of effluents in the thermal plume, while knowledge of the outlet stack surface temperature $T_{s,o}$ indicates whether condensation of the gas products will occur. The thin-walled, cylindrical stack is 0.5 m in diameter and 6.0 m high. The exhaust gas flow rate is 0.5 kg/s, and the inlet temperature is 600 °C. Estimate the outlet gas and stack surface temperatures if the convection heat transfer coefficients are $h_i = 10.2\, W/m^2K$ and $h_o = 13.9\, W/m^2K$. Take $T_\infty = 4°C$ and $c_{p,air} = 1104\, J/kgK$.

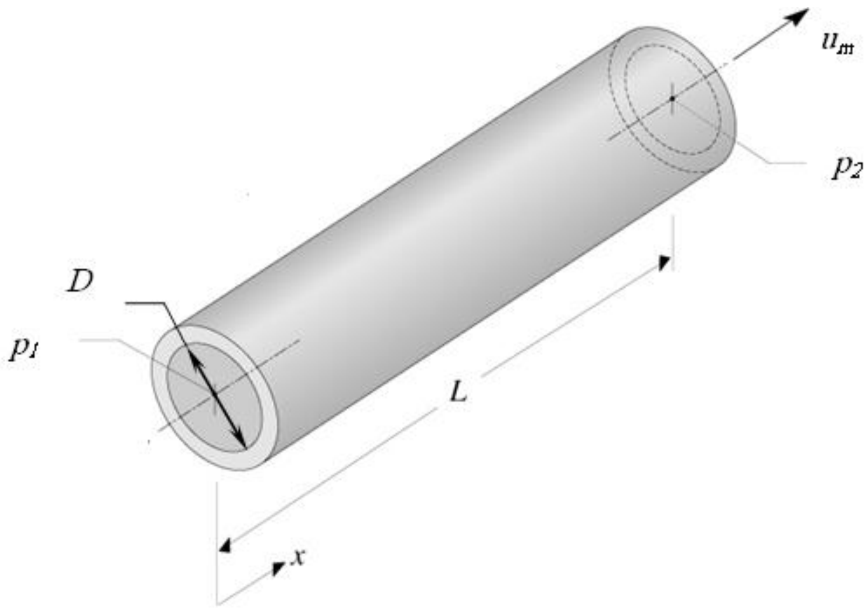


P2: A hot fluid passes through a thin-walled tube of 10-mm diameter and 1-m length, and a coolant at free stream temperature of 25 °C is in cross flow over the tube. When the flow rate is 18 kg/h and the inlet temperature is 85 °C, the outlet temperature is 78 °C. Assuming fully developed flow, determine the outlet temperature if the flow rate is increased by a factor of 2. That is, the flow rate is 36 kg/h, with all other conditions the same.



General Correlations for Internal Flow

Friction factor for hydrodynamically fully developed flow



- The pressure drop may be determined from knowledge of the friction factor *f*, where,

$$f \equiv -\frac{(dp/dx)D}{\rho u_m^2/2}$$

- Fanning friction factor f_a:

$$f_a = f/4$$

- Laminar flow in a circular tube:

$$f = \frac{64}{Re_D}; \quad f_a = \frac{16}{Re_D}$$

- Turbulent flow in a smooth circular tube:

$$f = [0.79\, ln(Re_D) - 1.64]^{-2}, \quad 3000 \le Re_D \le 5\times10^6$$

$$f_a = [1.58\, ln(Re_D) - 3.28]^{-2}, \quad 3000 \le Re_D \le 5\times10^6$$

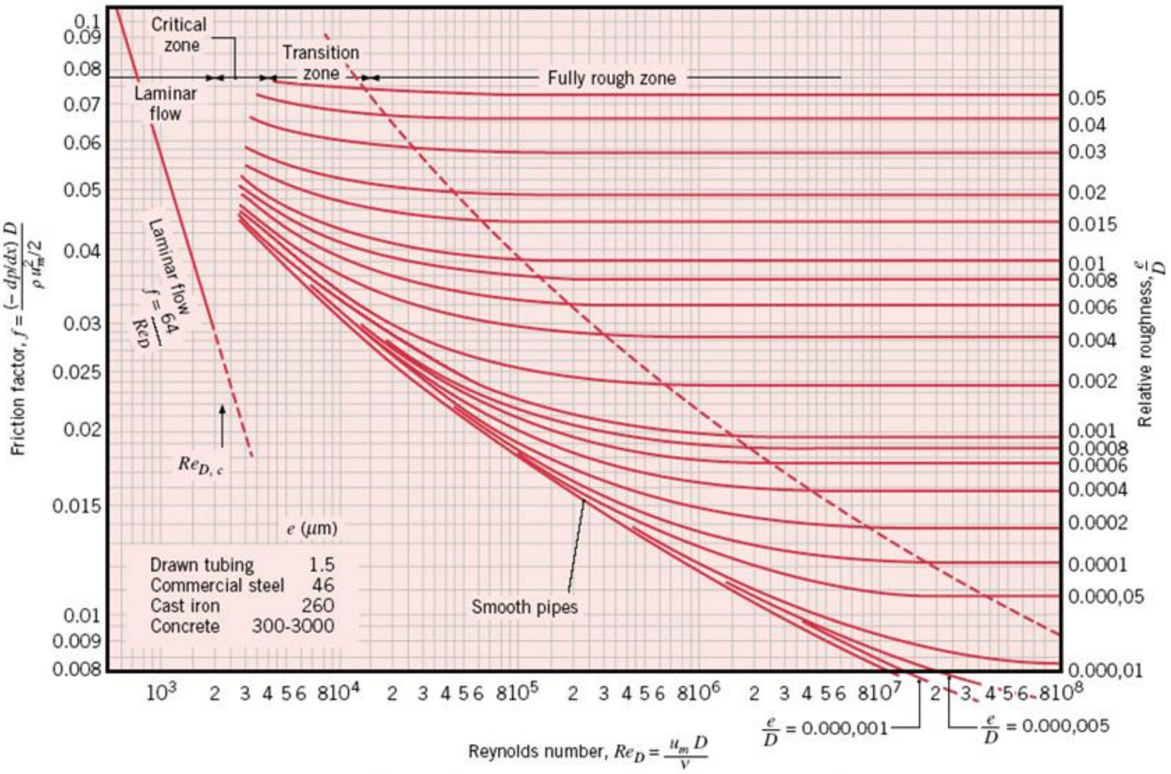
- Turbulent flow in a roughened circular tube (e: surface roughness):

$$\frac{1}{\sqrt{f}} = -2.0\log_{10}\left[\frac{(e/D)}{3.7} + \frac{2.51}{Re_D\sqrt{f}}\right], \quad \textit{Colebrook equation}$$

$$\frac{1}{\sqrt{f_a}} = -4.0\log_{10}\left[\frac{(e/D)}{3.7} + \frac{1.255}{Re_D\sqrt{f_a}}\right]$$

$$\frac{1}{\sqrt{f}} \cong -1.8\log_{10}\left[\frac{6.9}{Re_D} + \left(\frac{e/D}{3.7}\right)^{1.11}\right], \quad \textit{Haaland equation}$$

$$\frac{1}{\sqrt{f_a}} \cong -3.6\log_{10}\left[\frac{6.9}{Re_D} + \left(\frac{e/D}{3.7}\right)^{1.11}\right]$$



Friction factor for fully developed flow in a circular tube [3]. Used with permission.

Pressure drop for fully developed flow from x=0 to x=L

$$\Delta p = p(x=0) - p(x=L) = p_1 - p_2$$

$$\Delta p = f\,\frac{L}{D}\,\rho\,\frac{u_m^2}{2}$$

$$\Delta p = 4 f_a\,\frac{L}{D}\,\rho\,\frac{u_m^2}{2}$$

Pumping power requirement $(\dot{P})$

$$\dot{P} = \frac{\dot{m}\Delta p}{\rho\eta_p}$$

η_p : Pump total efficiency

Nusselt number for fully developed flow condition

- **Laminar flow in a circular tube**

  - **Uniform surface heat flux** (q_s'')**:** $Nu_D \equiv \dfrac{hD}{k} = 4.36$

  - **Uniform surface temperature** (T_s)**:** $Nu_D \equiv \dfrac{hD}{k} = 3.66$

- **Turbulent flow in a circular tube**

$$Nu_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}, \quad Gnielinski \quad correlation$$

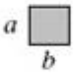
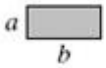
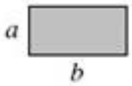
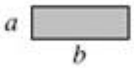
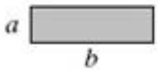
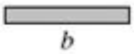
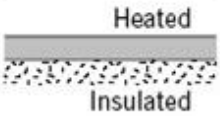
$$Nu_D = \frac{(f_a/2)(Re_D - 1000)Pr}{1 + 12.7(f_a/2)^{1/2}(Pr^{2/3} - 1)}$$

$$3\times10^3 \le Re_D \le 5\times10^6, \quad 0.5 \le Pr \le 2000$$

- **<u>Laminar flow</u> in a <u>noncircular tube</u>**

$$D_h \equiv \frac{4A_C}{P_w}$$

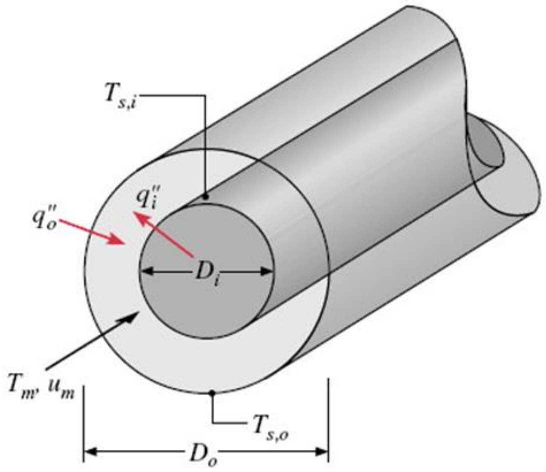
Nusselt numbers and friction factors for fully developed laminar flow in tubes of differing cross section

Cross Section	$\frac{b}{a}$	$Nu_D \equiv \frac{hD_h}{k}$ (Uniform q_s'')	$Nu_D \equiv \frac{hD_h}{k}$ (Uniform T_s)	$f\,Re_{D_h}$
	—	4.36	3.66	64
	1.0	3.61	2.98	57
	1.43	3.73	3.08	59
	2.0	4.12	3.39	62
	3.0	4.79	3.96	69
	4.0	5.33	4.44	73
	8.0	6.49	5.60	82
	∞	8.23	7.54	96
	∞	5.39	4.86	96
	—	3.11	2.49	53

Used with permission from W. M. Kays and M. E. Crawford, *Convection Heat and Mass Transfer*, 3rd ed. McGraw-Hill, New York, 1993.

- **Laminar flow in annulus region with insulated outer surface** $(q_o''=0)$

$$q_i''=h_o\left[T_{s,i}-T_m\right];\quad Nu_o=\frac{h_o D_{h,o}}{k};\quad D_{h,o}=D_o-D_i$$



- **Uniform surface heat flux** (q_i'')

$$Nu_o=\frac{72.2081+382.499(D_i/D_o)}{1+82.5237(D_i/D_o)+1.08936(D_i/D_o)^2}$$

- **Uniform surface temperature** $(T_{s,i})$

$$Nu_o=\frac{71.5905+355.517(D_i/D_o)}{1+81.133(D_i/D_o)+4.85985(D_i/D_o)^2}$$

- **Turbulent flow in a noncircular tube and in an annulus region**

$$D_h\equiv\frac{4A_C}{P_w},\qquad D_e\equiv\frac{4A_C}{P_h},\qquad Re_{D_h}\equiv\frac{\rho u_m D_h}{\mu},\qquad Nu_{D_e}\equiv\frac{hD_e}{k}$$

P_w : Wetted perimeter; P_h : Heat transfer perimeter

$$Nu_{De}=\frac{(f/8)(Re_{D_h}-1000)Pr}{1+12.7(f/8)^{1/2}(Pr^{2/3}-1)},\quad \textit{Gnielinski correlation}$$

$$Nu_{De}=\frac{(f_a/2)(Re_{D_h}-1000)Pr}{1+12.7(f_a/2)^{1/2}(Pr^{2/3}-1)}$$

$$3\times10^3\le Re_{D_h}\le5\times10^6,\quad 0.5\le Pr\le2000$$

Nusselt number for developing flows (Entry region effects)

- **Laminar flow in a circular tube**
    - **Thermal Entry region**

$$Nu_D = 3.66 + \frac{0.0668(D/L)Re_D\,Pr}{1.0 + 0.04[(D/L)Re_D\,Pr]^{2/3}}$$

    - **Combined Entry region**

$$Nu_D = 1.86\left(\frac{Re_D\,Pr\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14}$$

Problems

3.1. A fluid flows steadily with a velocity of 6 m/s through a commercial iron rectangular duct whose sides are 1 in. by 2 in., and the length of the duct is 6 m. The average temperature of the fluid is 60°C. The fluid completely fills the duct. Calculate the surface heat transfer coefficient if the fluid is

a. Water

b. Air at atmospheric pressure

c. Engine oil (ρ = 864 kg/m³, c_p = 2047 J/(kg · K), ν = 0.0839 × 10^{-3}m²/s, Pr = 1050, k = 0.140 W/m · K)

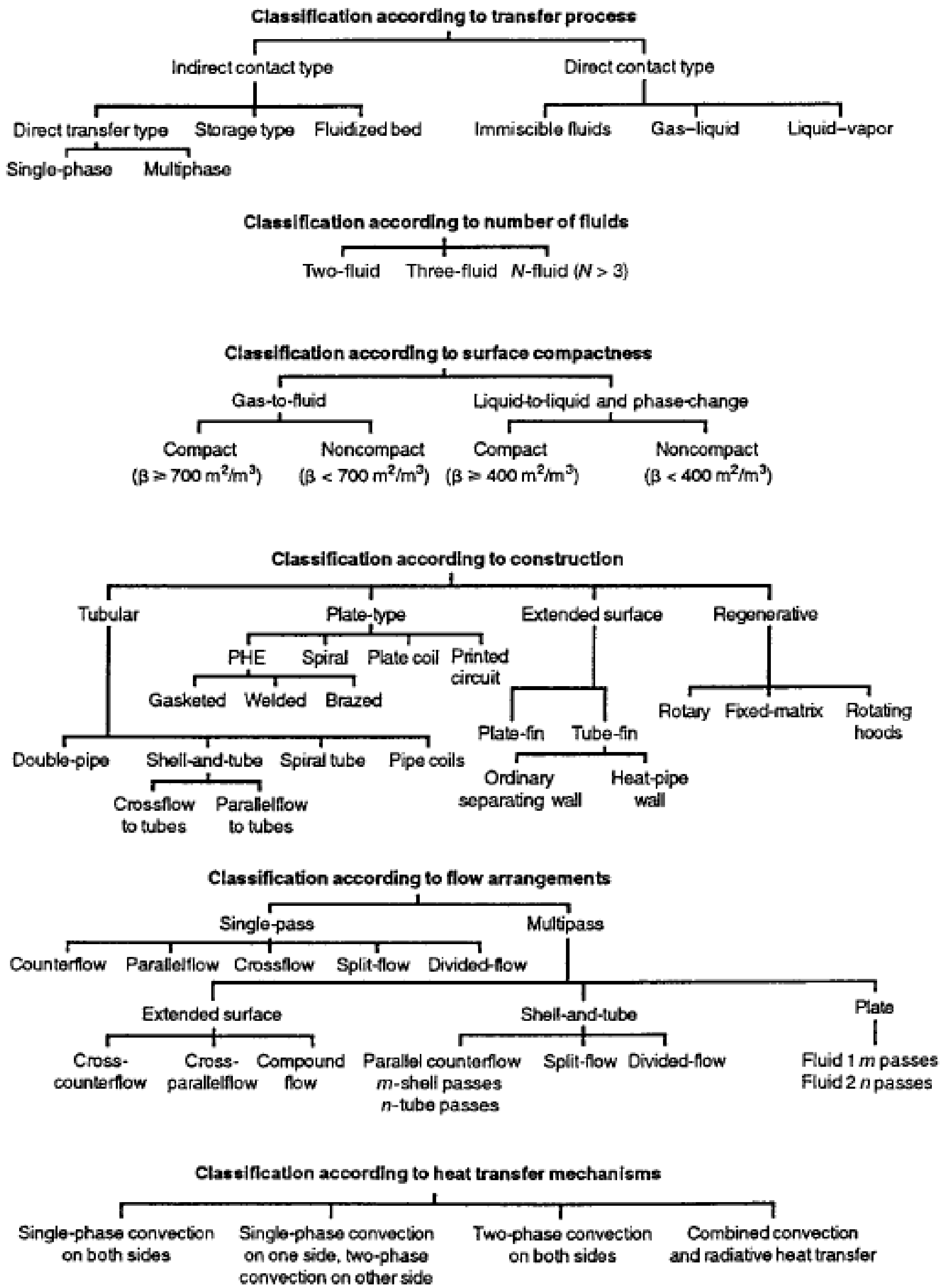
3.2. Air at 1.5 atm and 40°C flows through a 10 m rectangular duct of 40 cm by 25 cm with a velocity of 5 m/s. The surface temperature of the duct is maintained at 120°C and the average air temperature at exit is 80°C. Calculate the total heat transfer using Gnielinski's correlation and check your result by energy balance.

3.10. Consider the laminar flow of an oil inside a duct with a Reynolds number of 1000. The length of the duct is 2.5 m and the diameter is 2 cm. The duct is heated electrically by the use of its walls as an electrical resistance. Properties of the oil at the average oil temperature are ρ = 870 kg/m³, μ = 0.004 N · s/m², and c_p= 1959 kJ/kg · K, and k = 0.128 W/m · K. Obtain the local Nusselt number at the end of the duct.

4.1. Consider the flow of 20°C water through a circular duct with an inner diameter of 2.54 cm. The average velocity of the water is 4 m/s. Calculate the pressure drop per unit length ($\Delta p/L$).

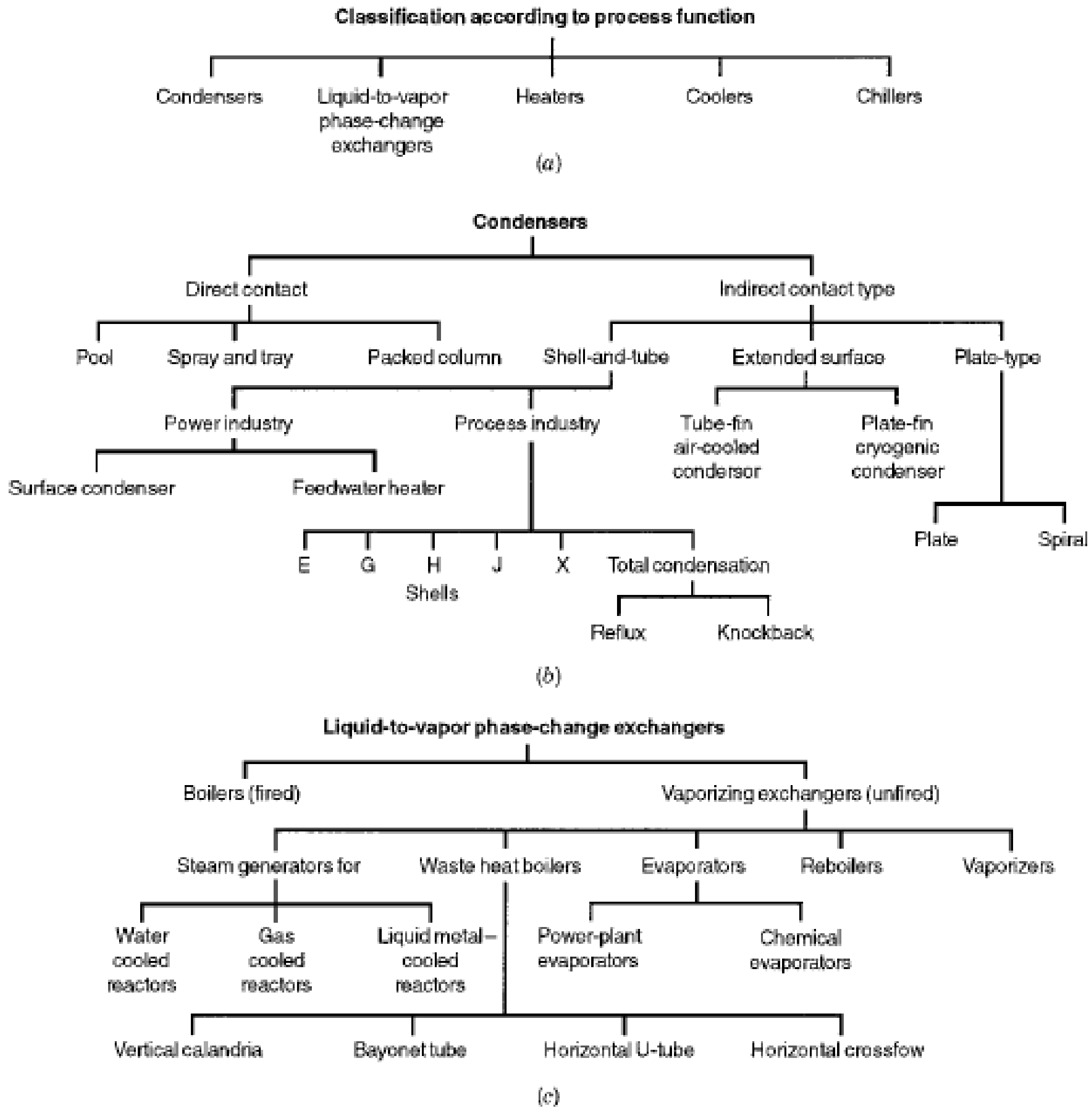
4.2. Water at 5°C flows through a parallel-plate channel in a flat-plate heat exchanger. The spacing between the plates is 2 cm, and the mean velocity of the water is 3.5 m/s. Calculate the pressure drop per unit length in the hydrodynamically fully developed region.

Classifications of Heat Exchangers



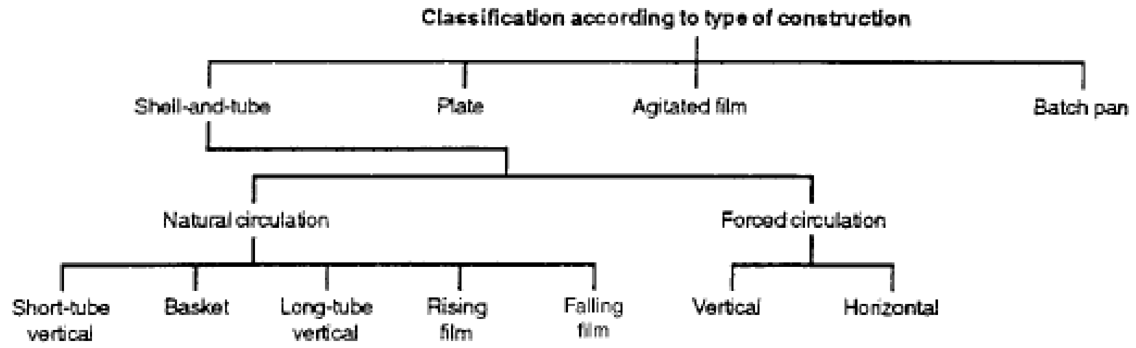
Classification of heat exchangers (Shah, 1981).

Classifications according to transfer processes

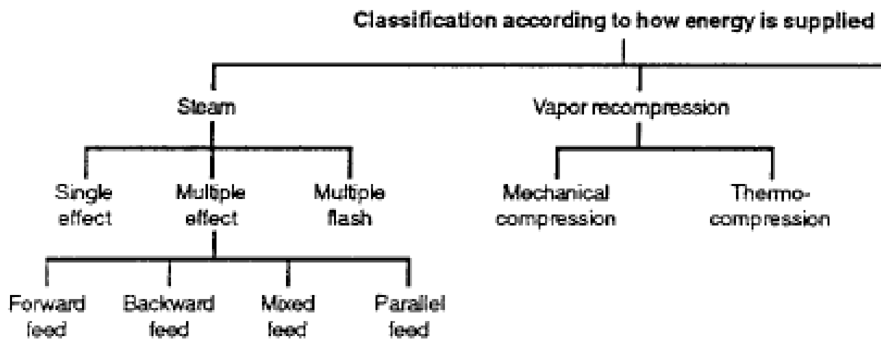


(a) Classification according to process function; (b) classification of condensers; (c) classification of liquid-to-vapor phase-change exchangers.

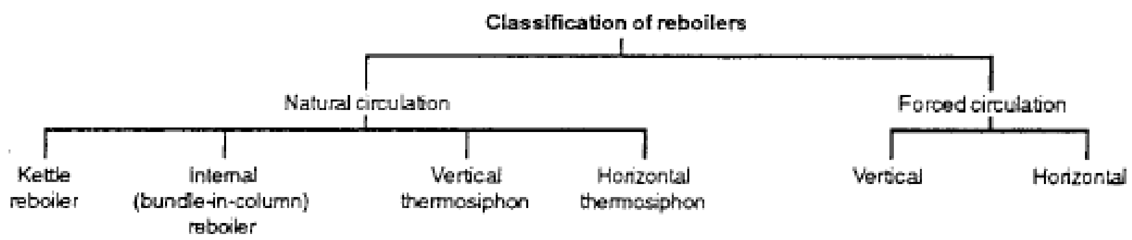
Classifications according to construction features



(d) (i)

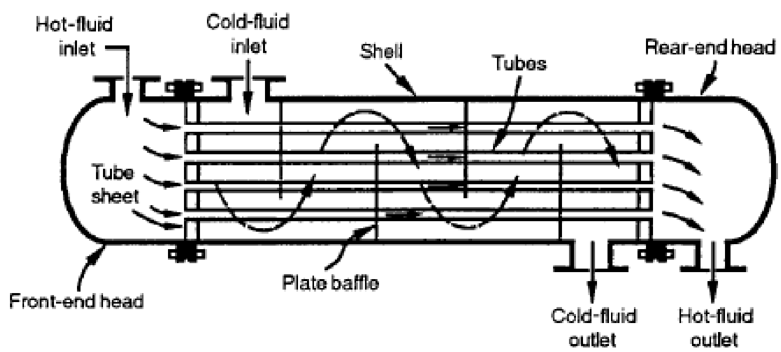


(d) (ii)

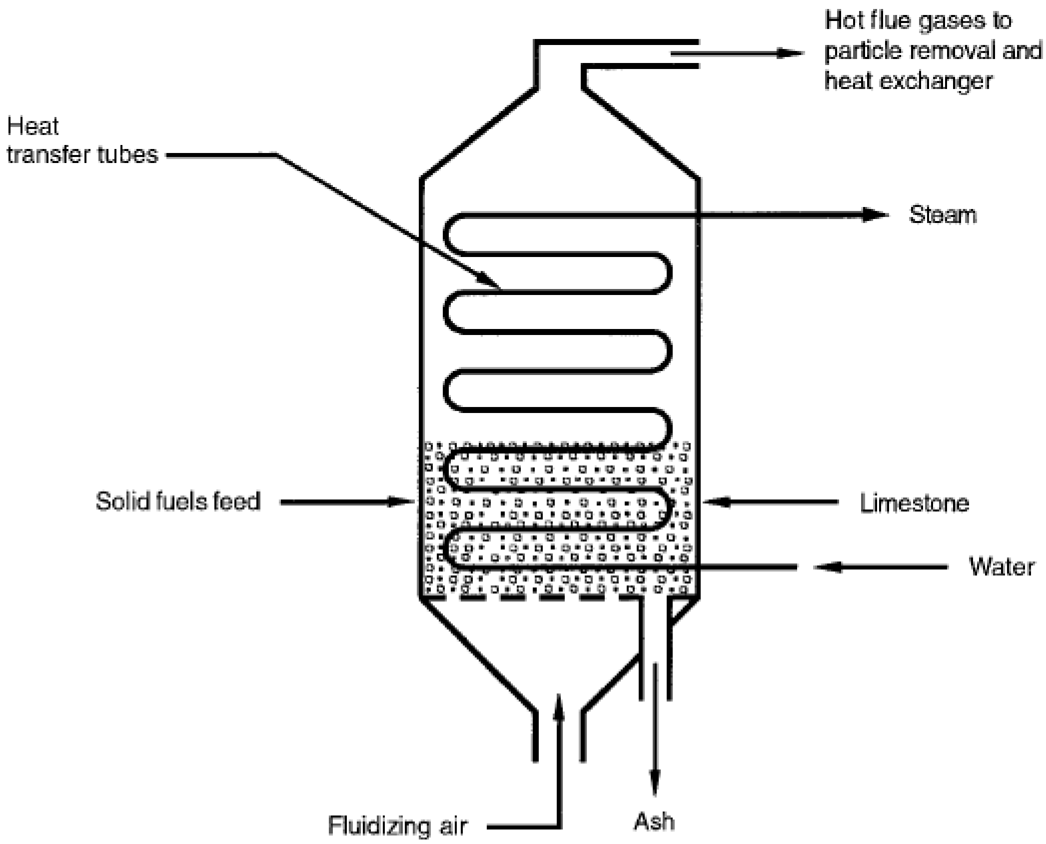


(e)

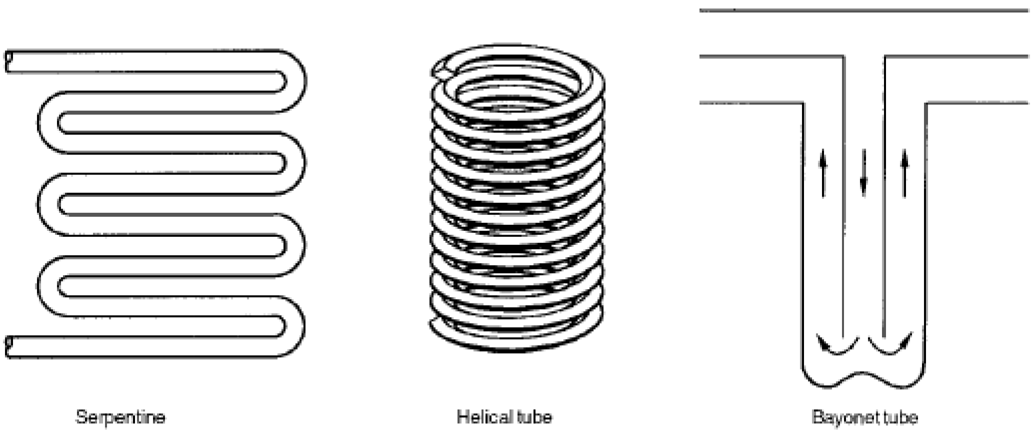
(*d*) classification of chemical evaporators according to (i) the type of construction, and (ii) how energy is supplied (Shah and Mueller, 1988); (*e*) classification of reboilers.



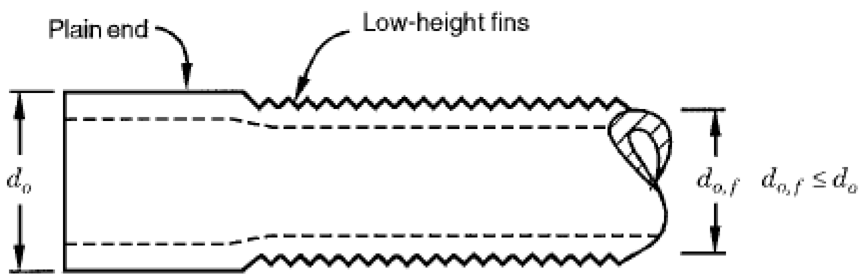
Shell-and-tube exchanger (BEM) with one shell pass and one tube pass;



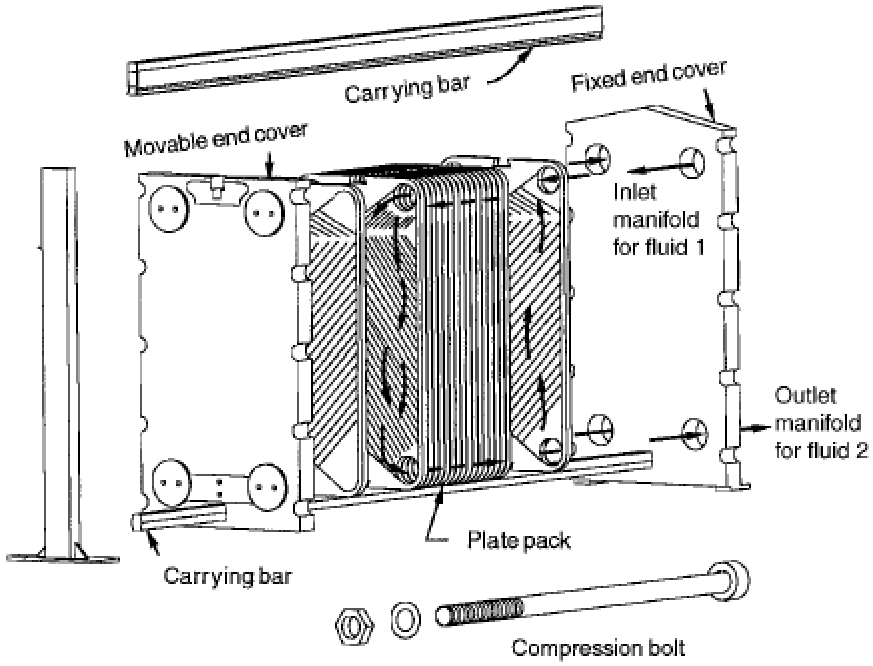
Fluidized-bed heat exchanger.



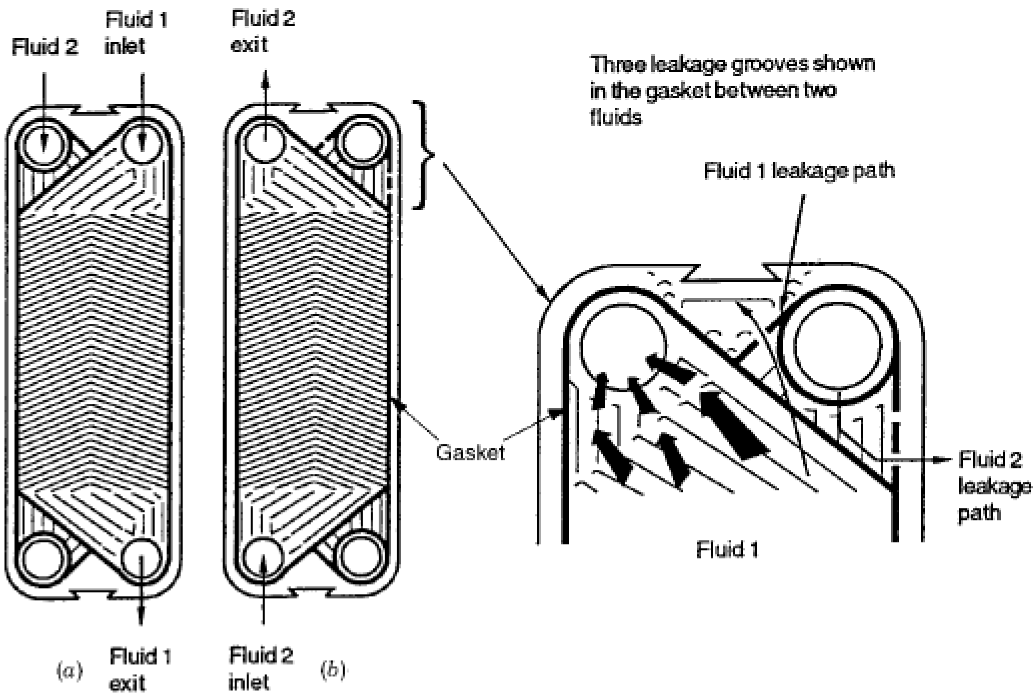
Additional tube configurations used in shell-and-tube exchangers.



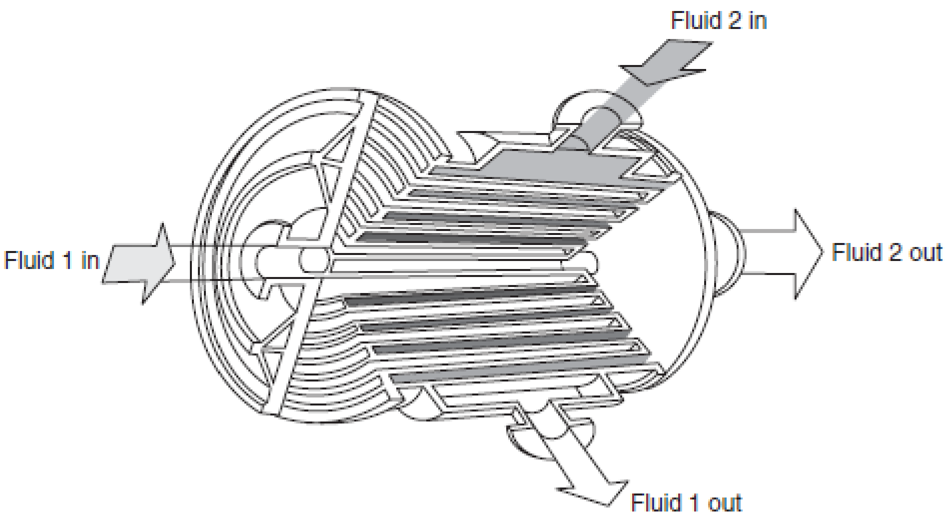
Low-finned tubing. The plain end goes into the tubesheet.



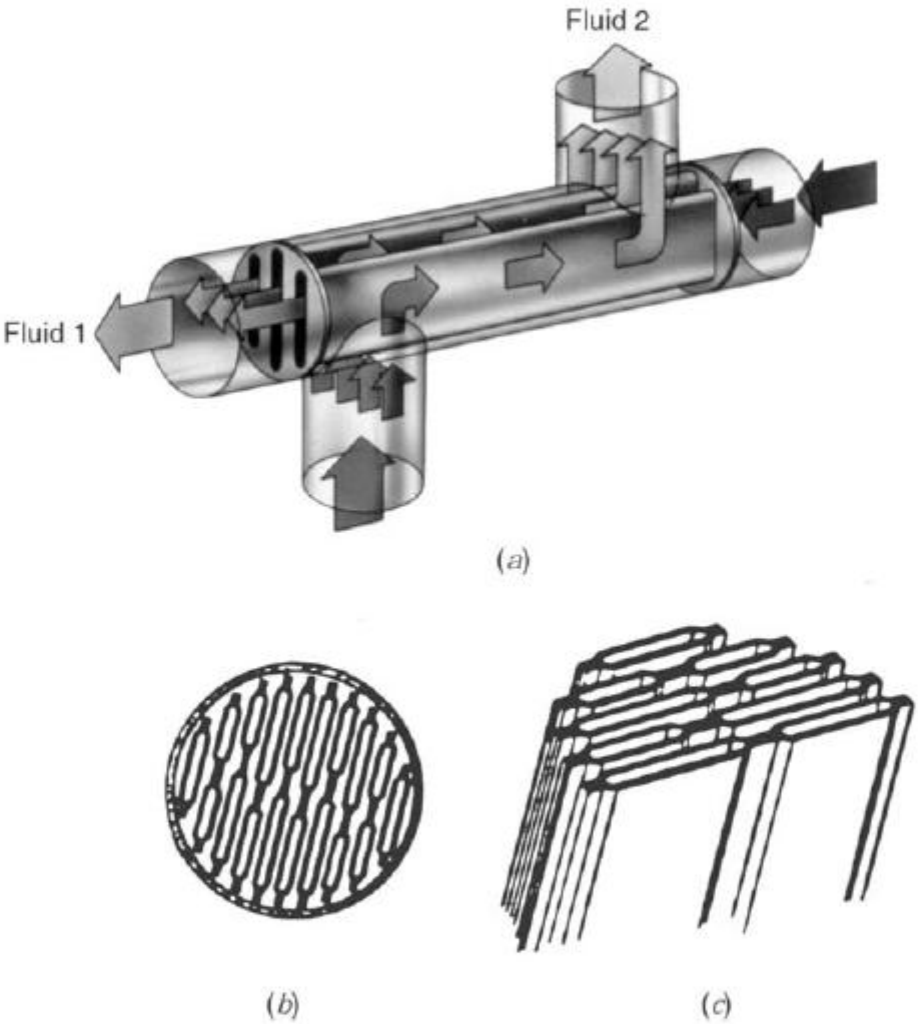
Gasketed plate- and-frame heat exchanger.



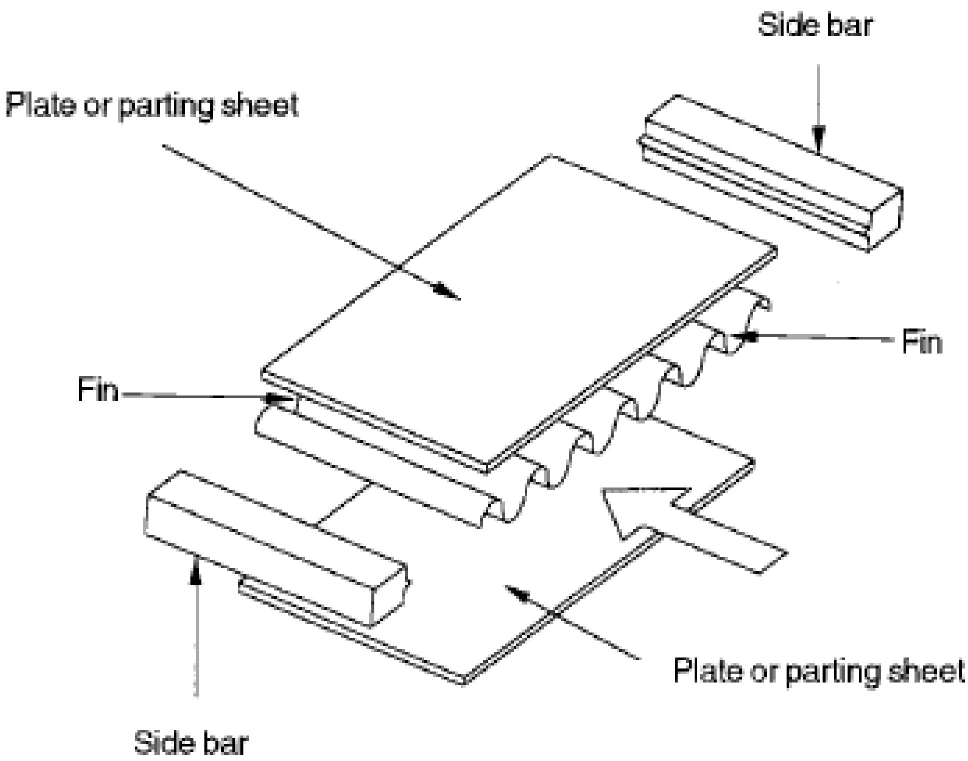
Plates showing gaskets around the ports (Shah and Focke, 1988).



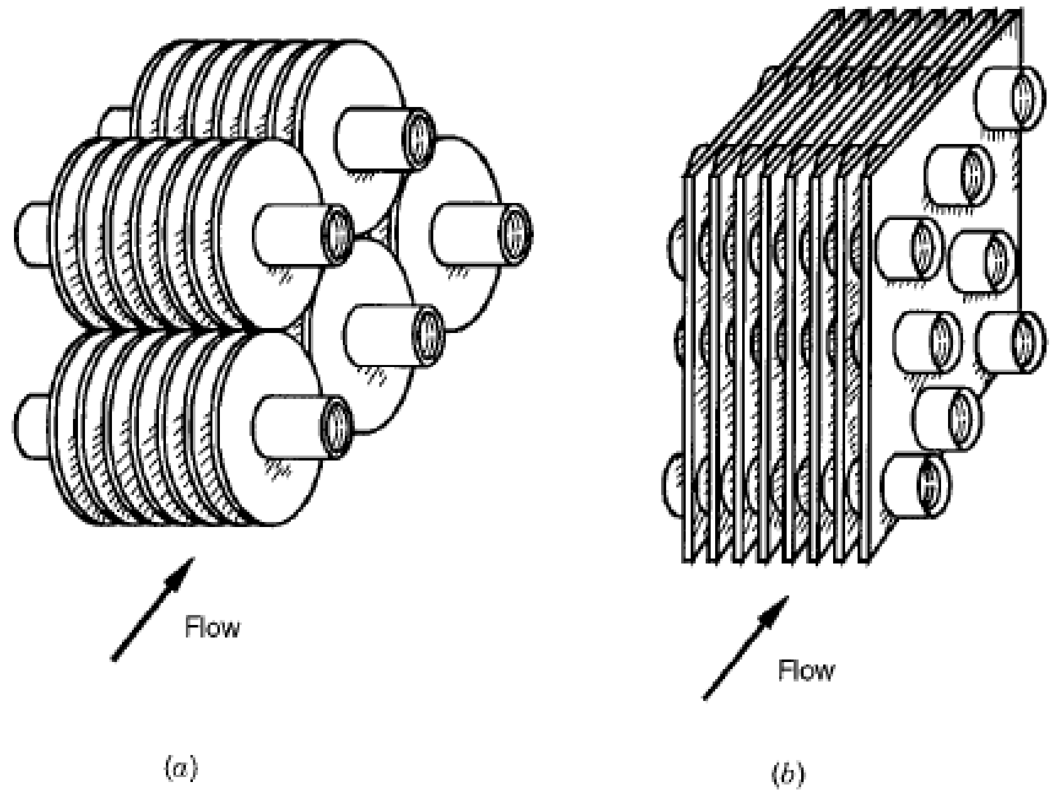
Spiral plate heat exchanger with both fluids in spiral counterflow.



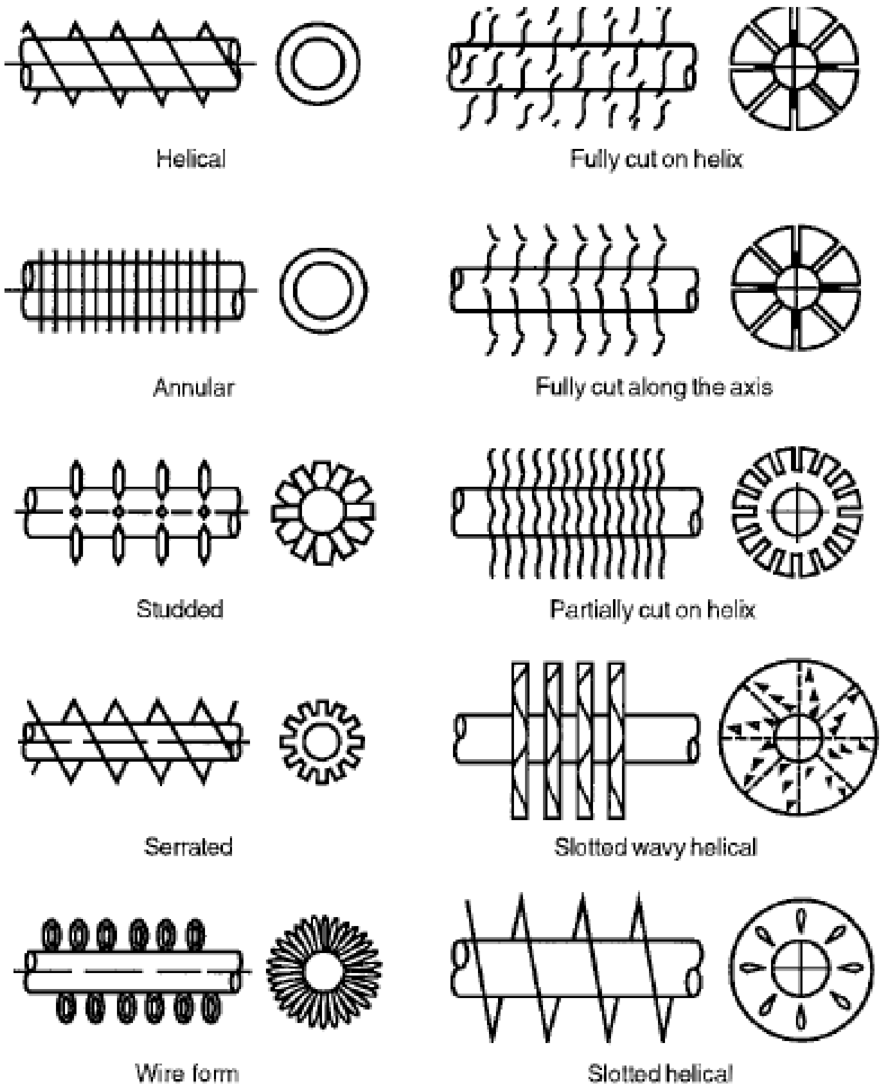
(*a*) Lamella heat exchanger; (*b*) cross section of a lamella heat exchanger; (*c*) lamellas. (Courtesy of Alfa Laval Thermal, Inc., Lund, Sweden.)



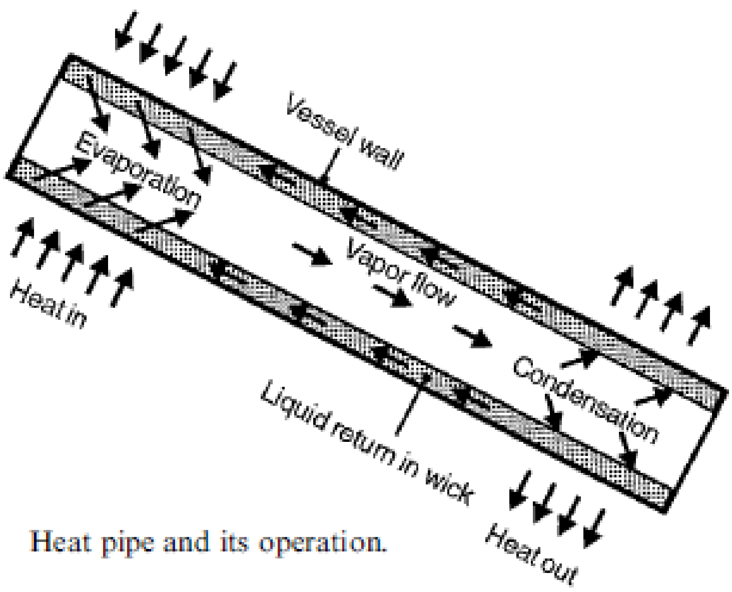
Basic components of a plate-fin heat exchanger (Shah and Webb, 1983).



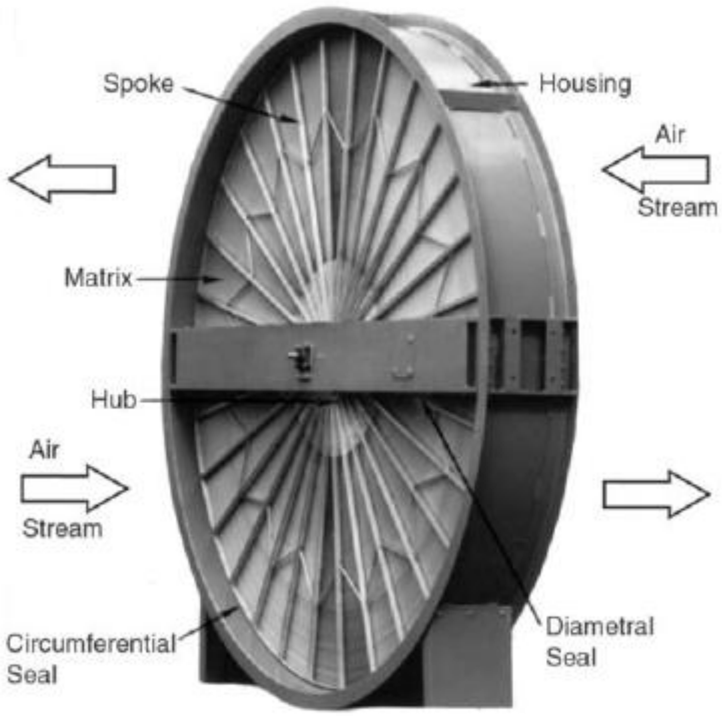
(*a*) Individually finned tubes; (*b*) flat (continuous) fins on an array of tubes. The flat fins are shown as plain fins, but they can be wavy, louvered, or interrupted.



Individually finned tubes (Shah, 1981).

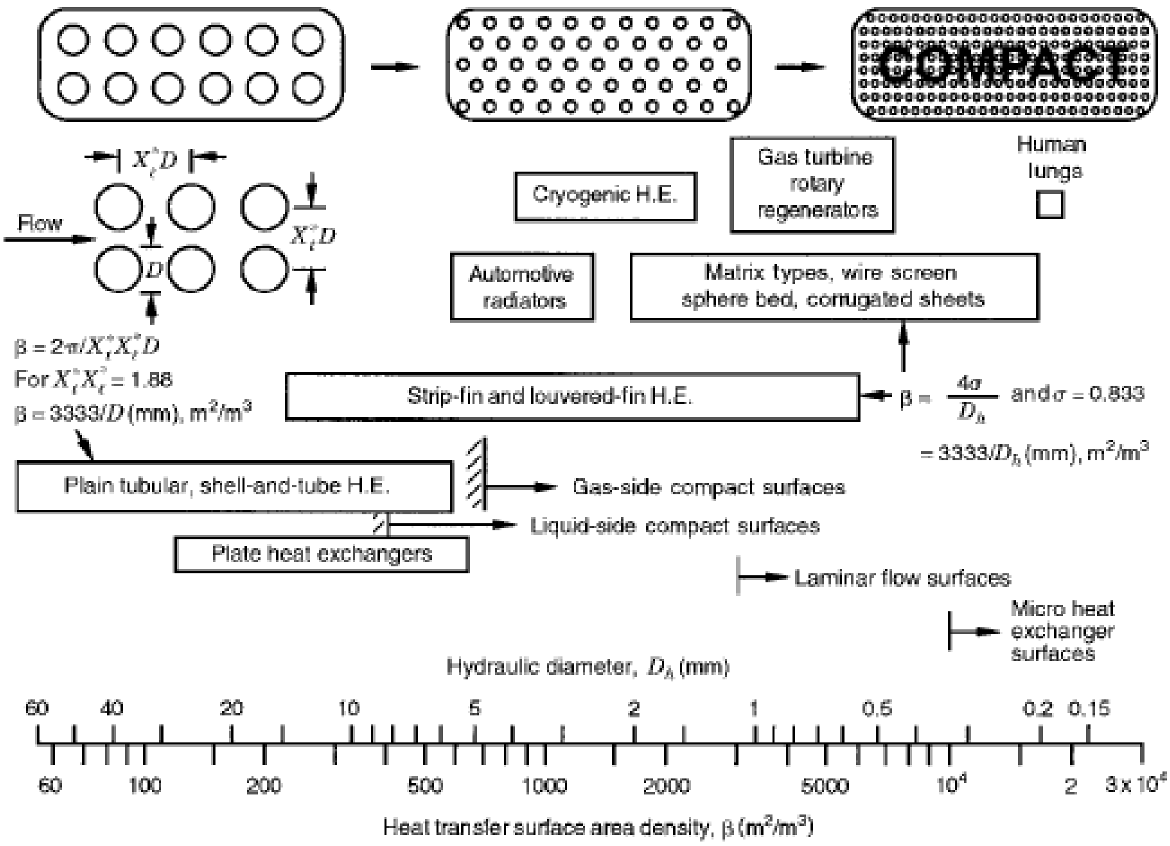


Heat pipe and its operation.



Heat wheel or a rotary regenerator made from a polyester film.

Classifications according to surface compactness

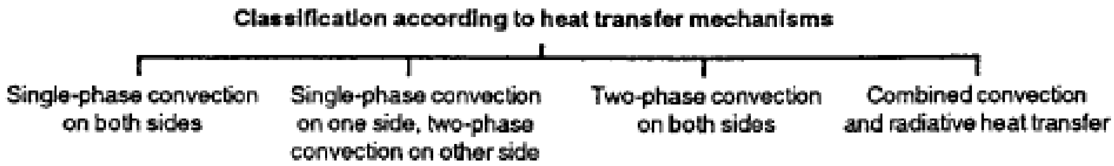


Heat transfer surface area density spectrum of exchanger surfaces (Shah, 1981).

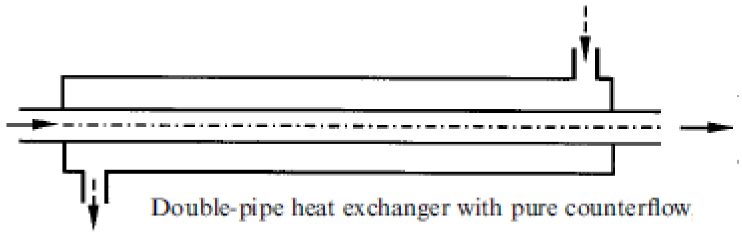
Classifications according to number of fluids

- Heat exchangers with as many as 12 fluids streams have been used in some chemical processes applications.
- The design theory of three- and multi fluid heat exchangers is algebraically very complex.
- The present notes covers only design theory of two-fluid heat exchangers.

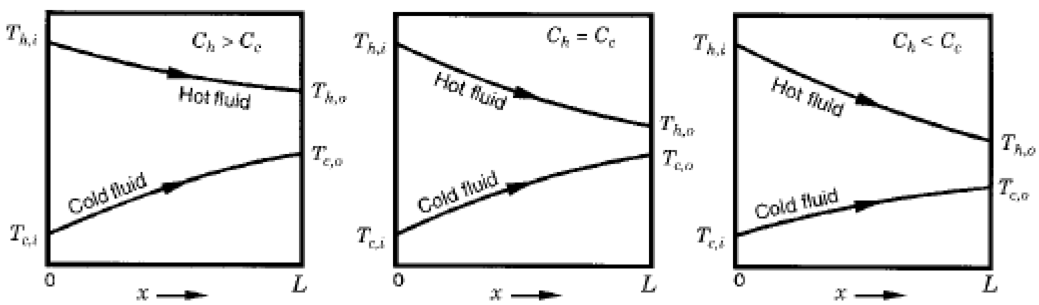
Classifications according to heat transfer mechanism



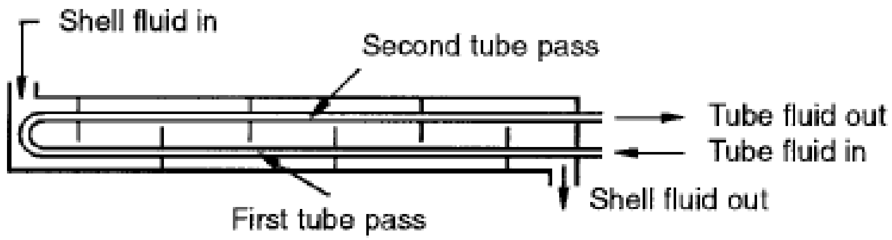
Classifications according to flow arrangements



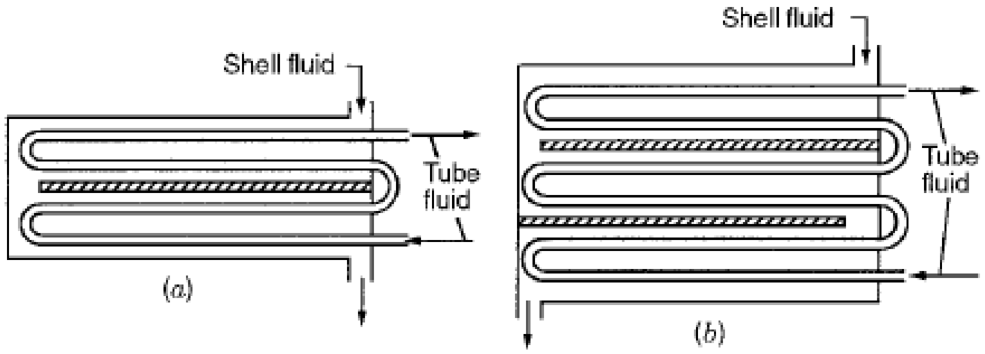
Double-pipe heat exchanger with pure counterflow



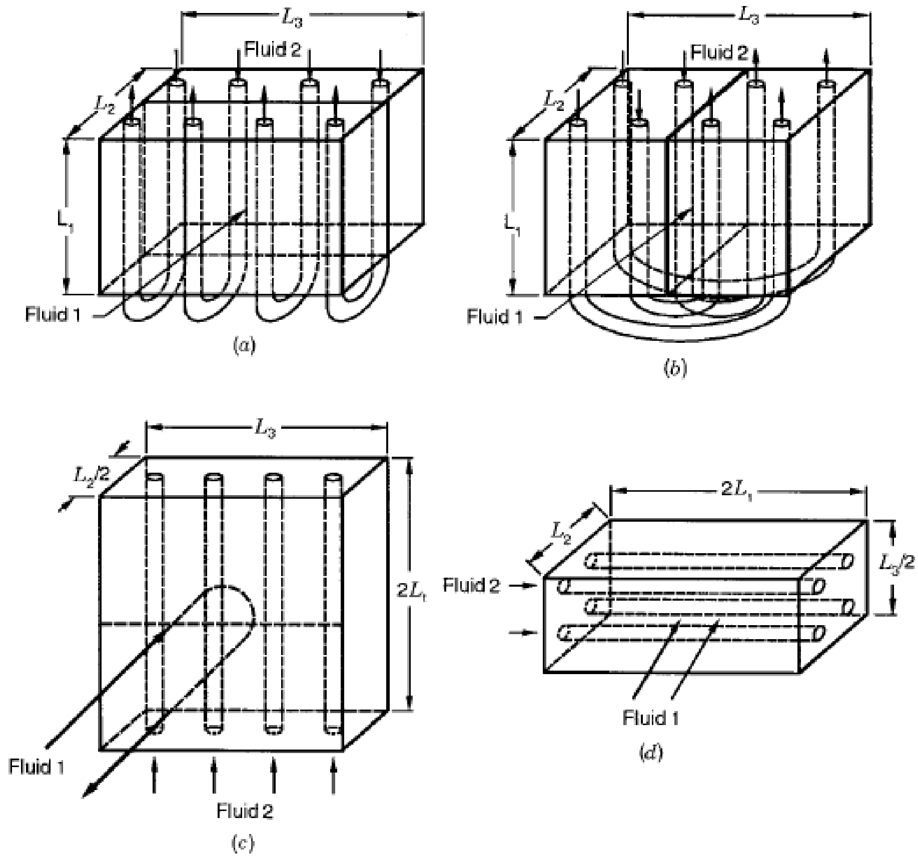
Temperature distributions in a parallelflow heat exchanger (Shah, 1981).



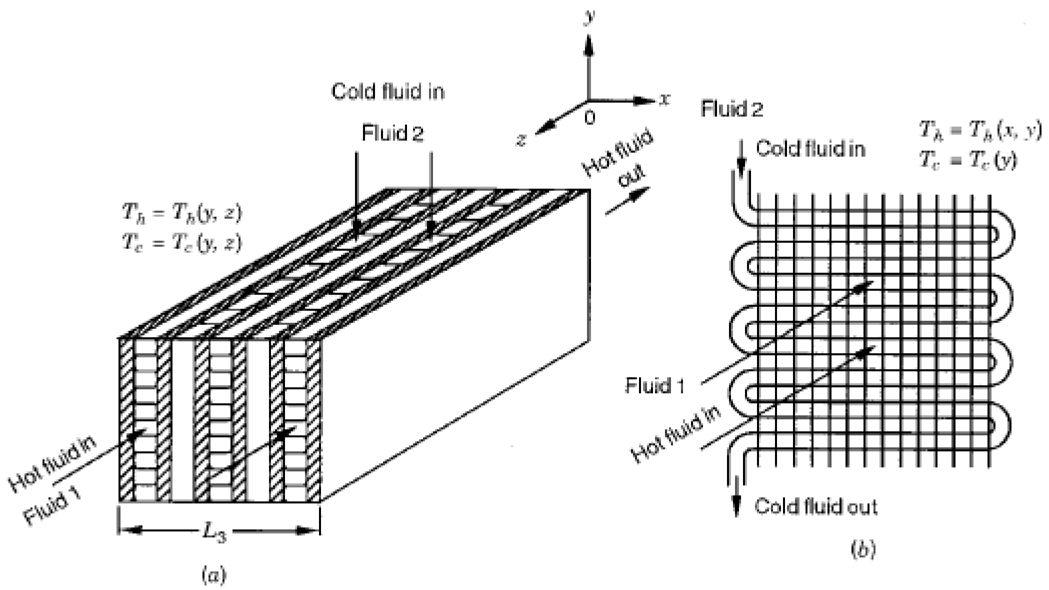
A 1–2 TEMA E heat exchanger (one shell pass and two tube passes)



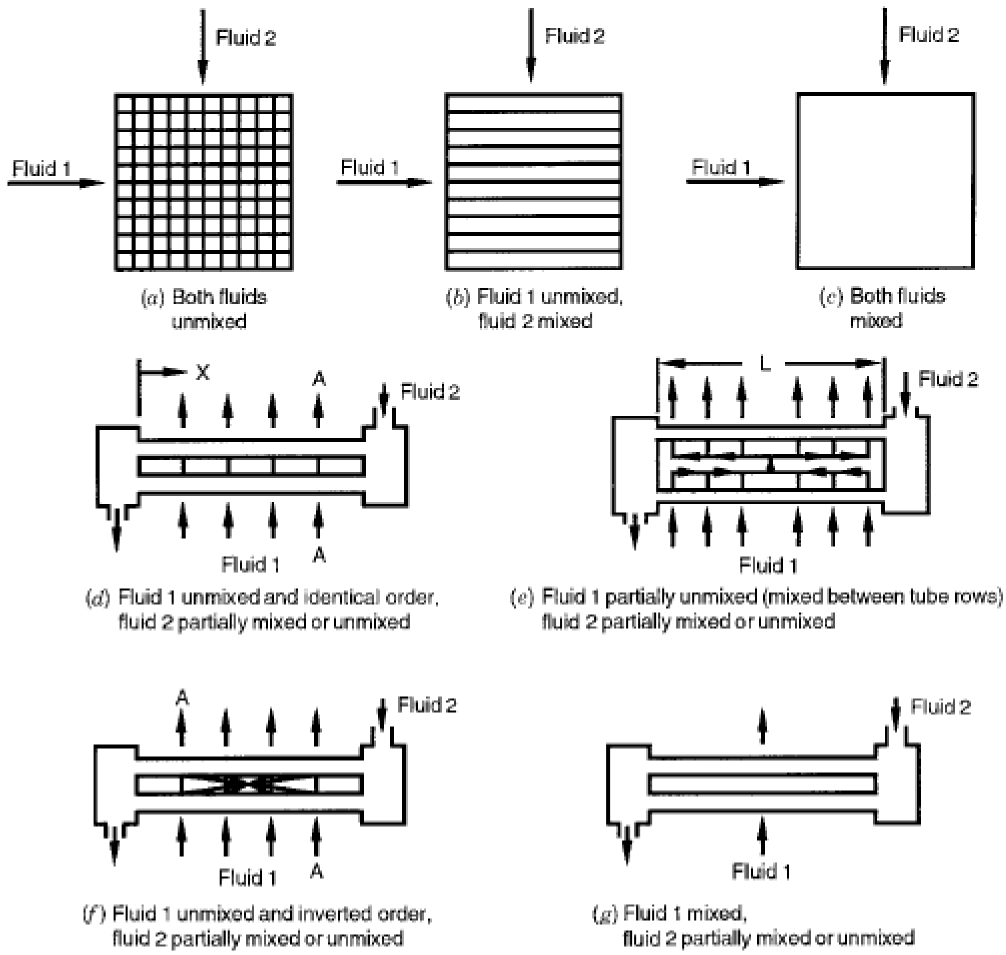
(a) Two shell pass–four tube pass exchanger; (b) three shell pass–six tube pass excnanger.



(a) Two-pass cross-counterflow exchanger; (b) single-pass crossflow exchanger; (c, d) unfolded exchangers of (a) and (b), respectively.



(a) Plate-fin unmixed–unmixed crossflow heat exchanger; (b) serpentine (one tube row) tube-fin unmixed–mixed crossflow heat exchanger (Shah, 1981).

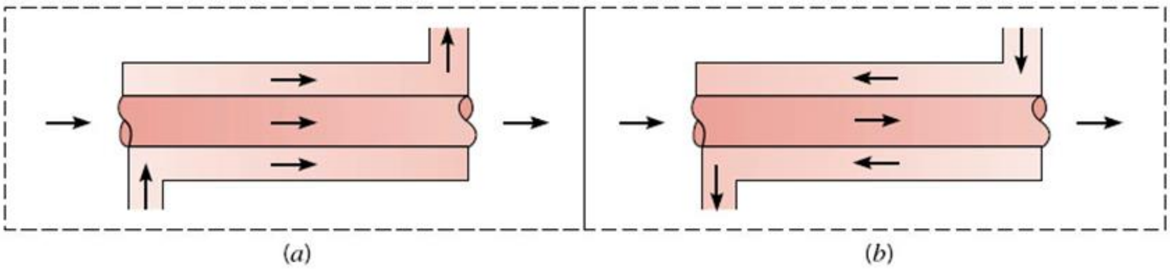


Symbolic presentation of various degrees of mixing in a single-phase crossflow exchanger.

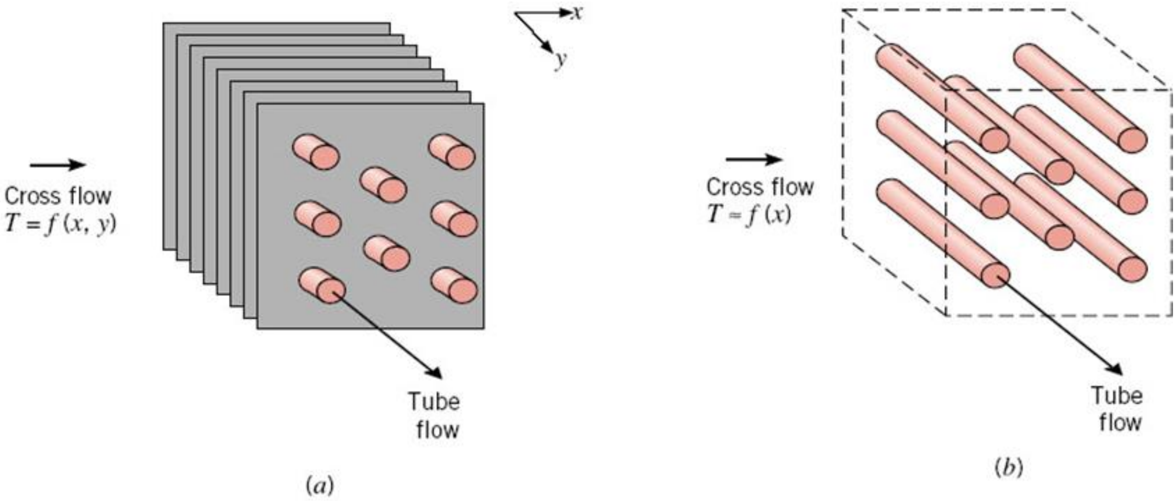
Basic Equations for Heat Exchangers

Heat Exchangers: Devices used to exchange thermal energy between at least two fluids. They encompass a wide range of flow configurations.

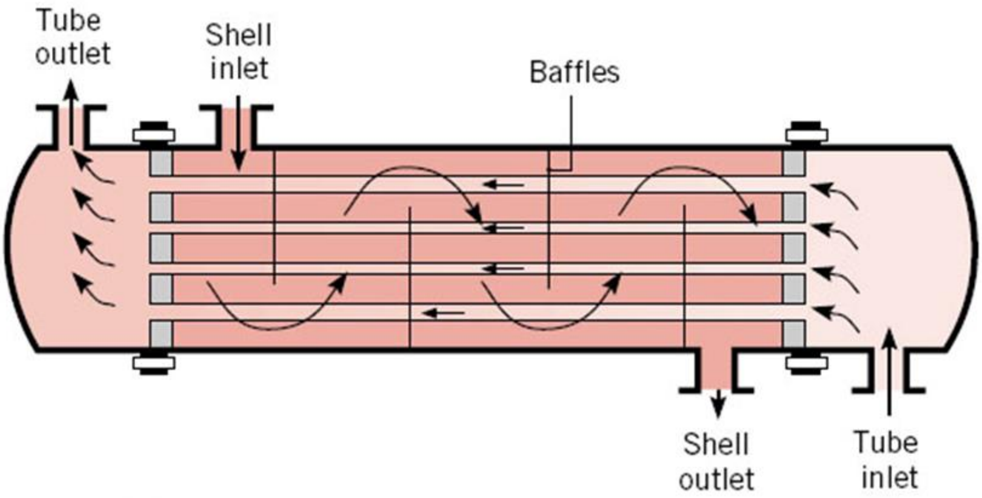
Classifications of Heat exchangers based on flow arrangement



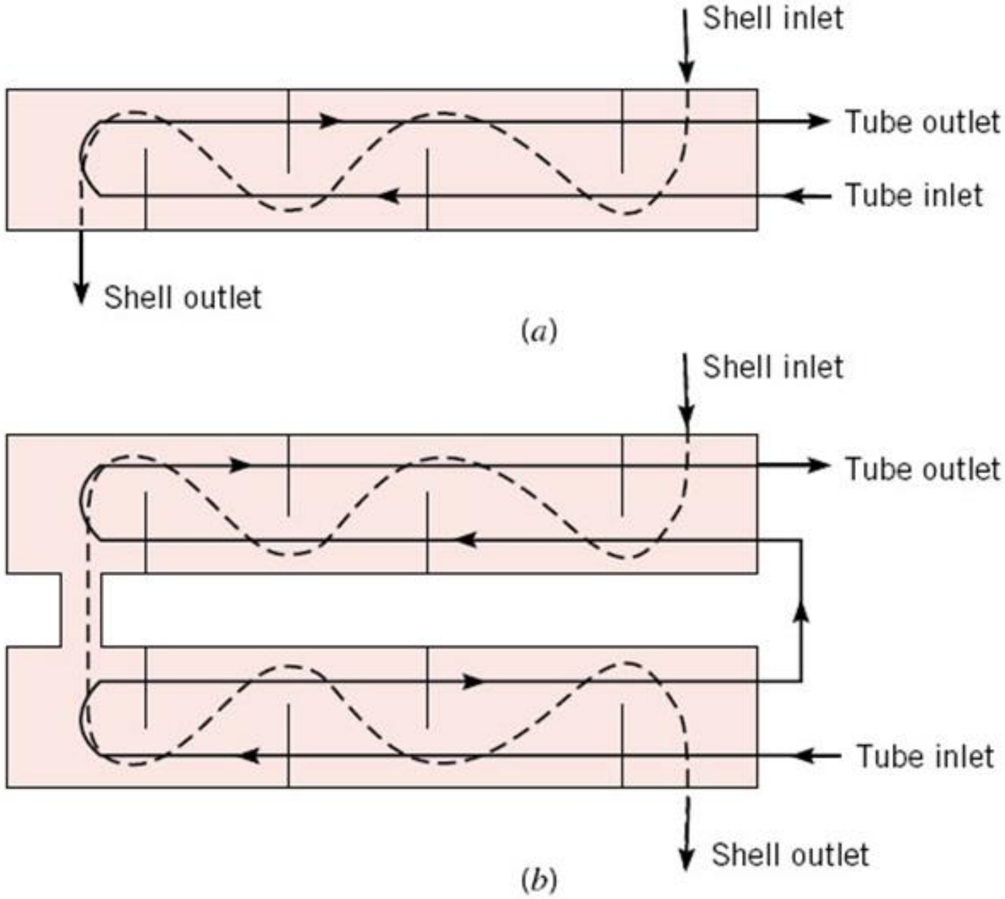
Concentric tube heat exchangers. (*a*) Parallel flow. (*b*) Counterflow.



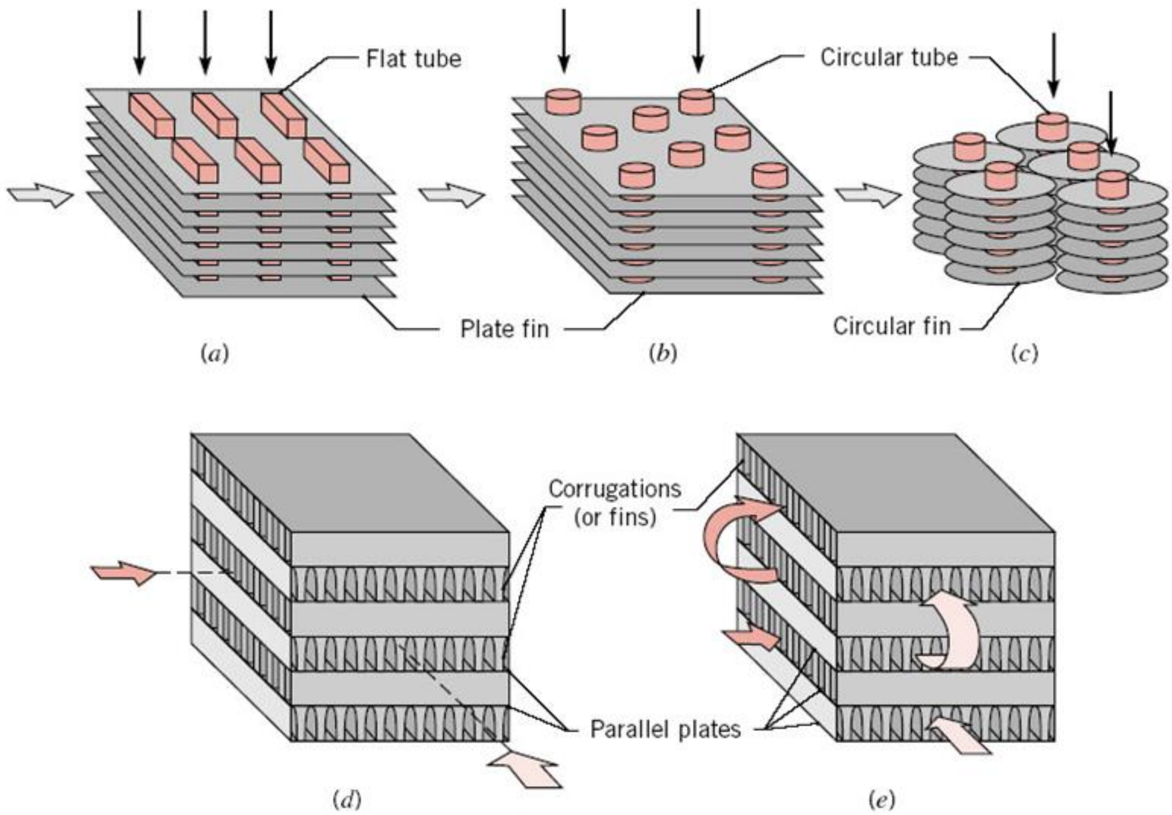
Cross-flow heat exchangers. (*a*) Finned with both fluids unmixed. (*b*) Unfinned with one fluid mixed and the other unmixed.



Shell-and-tube heat exchanger with one shell pass and one tube pass (cross-counterflow mode of operation).



Shell-and-tube heat exchangers. (*a*) One shell pass and two tube passes. (*b*) Two shell passes and four tube passes.



Compact heat exchanger cores. (*a*) Fin–tube (flat tubes, continuous plate fins). (*b*) Fin–tube (circular tubes, continuous plate fins). (*c*) Fin–tube (circular tubes, circular fins). (*d*) Plate–fin (single pass). (*e*) Plate–fin (multipass).

The overall heat transfer coefficient

- **Bare circular clean tubes of N_t number of tubes**

$$\frac{1}{UA}=\frac{1}{U_oA_o}=\frac{1}{U_iA_i}=\frac{1}{h_oA_o}+R_W+\frac{1}{h_iA_i}$$

$$A_o=\pi d_oN_tL; \qquad A_i=\pi d_iN_tL$$

$$R_W=\frac{ln(d_o/d_i)}{2\pi k_{tube}N_tL}$$

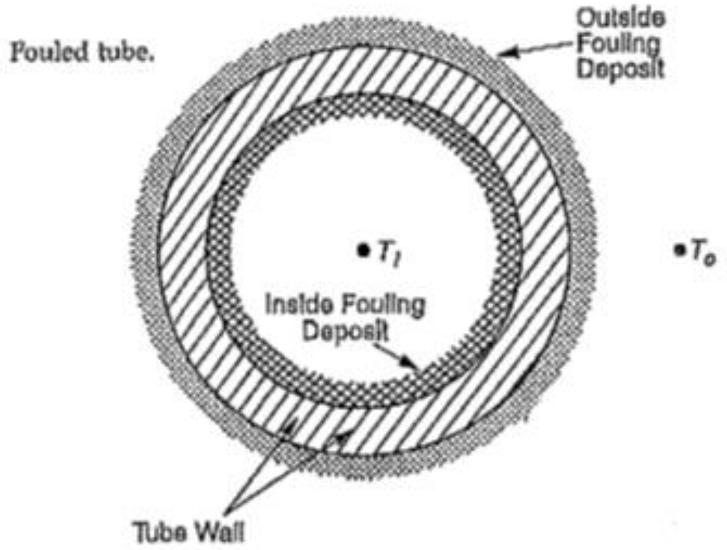
$$\frac{1}{U_o}=\frac{1}{h_o}+\frac{d_o\,ln(d_o/d_i)}{2k_{tube}}+\left(\frac{d_o}{d_i}\right)\frac{1}{h_i}$$

$$\frac{1}{U_i}=\left(\frac{d_i}{d_o}\right)\frac{1}{h_o}+\frac{d_i\,ln(d_o/d_i)}{2k_{tube}}+\frac{1}{h_i}$$

- **<u>Bare</u> <u>circular</u> <u>fouled</u> tubes of $\underline{N_t}$ number of tubes**

$$\frac{1}{UA}=\frac{1}{U_oA_o}=\frac{1}{U_iA_i}=\frac{1}{h_oA_o}+\frac{R_{fo}}{A_o}+R_W+\frac{R_{fi}}{A_i}+\frac{1}{h_iA_i}$$

$$R_{fo}: outside\ \ fouling\ unit\ resistance$$
$$R_{fi}: inside\ \ fouling\ unit\ resistance$$



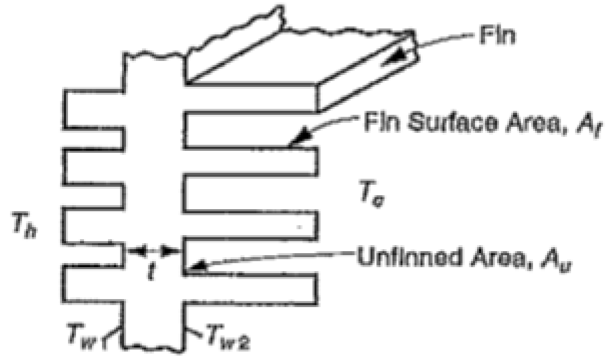
$$\frac{1}{U_o}=\frac{1}{h_o}+R_{fo}+\frac{d_o\,ln(d_o/d_i)}{2k_{tube}}+\left(\frac{d_o}{d_i}\right)R_{fi}+\left(\frac{d_o}{d_i}\right)\frac{1}{h_i}$$

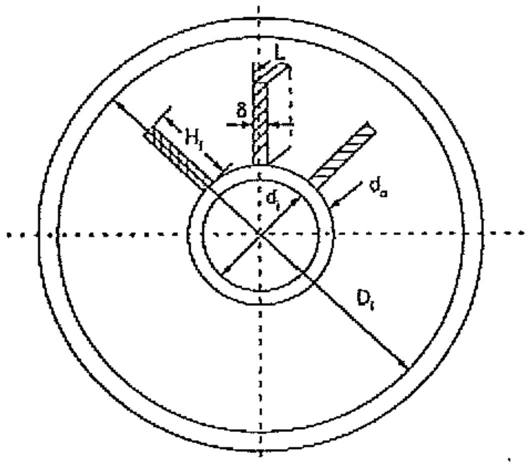
$$\frac{1}{U_i}=\left(\frac{d_i}{d_o}\right)\frac{1}{h_o}+\left(\frac{d_i}{d_o}\right)R_{fo}+\frac{d_i\,ln(d_o/d_i)}{2k_{tube}}+R_{fi}+\frac{1}{h_i}$$

- **<u>Finned</u> <u>circular</u> <u>fouled</u> tubes of $\underline{N_t}$ number of tubes (<u>rectangular fins</u>)**

$$\frac{1}{UA}=\frac{1}{U_oA_o}=\frac{1}{U_iA_i}=\frac{1}{\eta_oh_oA_o}+\frac{R_{fo}}{\eta_oA_o}+R_W+\frac{R_{fi}}{\eta_iA_i}+\frac{1}{\eta_ih_iA_i}$$

$$\eta_o=1-(1-\eta_{fo})\frac{A_{fo}}{A_o};\quad \eta_i=1-(1-\eta_{fi})\frac{A_{fi}}{A_i}$$





$$A_o = (\pi d_o + 2H_{fo}N_{fo})N_t L; \quad A_i = (\pi d_i + 2H_{fi}N_{fi})N_t L$$

$$A_{fo} = (2H_{fo} + \delta_o)N_{fo}N_t L; \quad A_{fi} = (2H_{fi} + \delta_i)N_{fi}N_t L$$

$$A_{uo} = (\pi d_o - N_{fo}\delta_o)N_t L; \quad A_{ui} = (\pi d_i - N_{fi}\delta_i)N_t L$$

$$A_o = A_{uo} + A_{fo}; \quad A_i = A_{ui} + A_{fi}$$

$$(\eta_o, \eta_i): \ (outer, inner)\ surface\ efficiency$$

$$(N_{fo}, N_{fi}): no. of \ \ fins\ on\ (outer, inner)\ surface$$

$$(H_{fo}, H_{fi}): \ (outer, inner)\ fin\ length$$

$$(\delta_{fo}, \delta_{fi}): \ (outer, inner)\ fin\ thickness$$

$$(k_{fin,o}, k_{fin,i}): \ (outer, inner)\ fin\ thermal\ conductivity$$

$$(\eta_{fo}, \eta_{fi}): \ (outer, inner)\ fin\ thermal\ efficiency$$

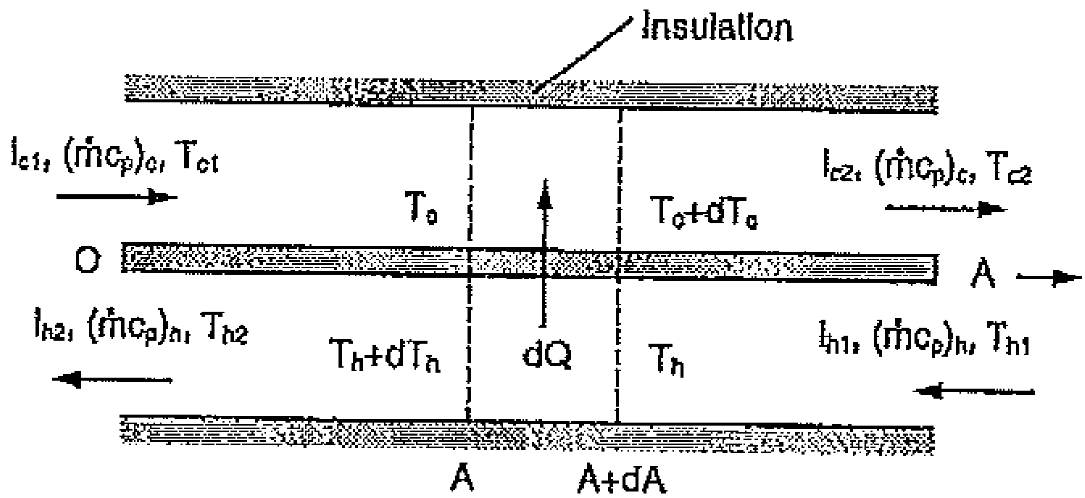
$$\eta_{fo} = \frac{tanh\left([H_{fo} + \delta_o/2]\sqrt{\frac{2h_o}{k_{fin,o}\delta_o}}\right)}{[H_{fo} + \delta_o/2]\sqrt{\frac{2h_o}{k_{fin,o}\delta_o}}}; \quad \eta_{fi} = \frac{tanh\left([H_{fi} + \delta_i/2]\sqrt{\frac{2h_i}{k_{fin,i}\delta_i}}\right)}{[H_{fi} + \delta_i/2]\sqrt{\frac{2h_i}{k_{fin,i}\delta_i}}}$$

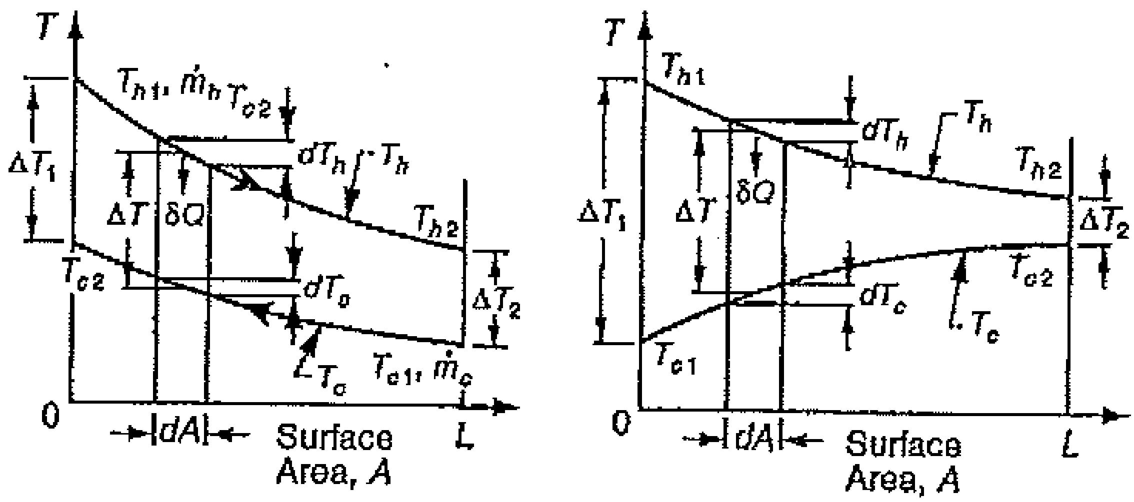
$$\frac{1}{U_o}=\frac{1}{\eta_o h_o}+\frac{R_{fo}}{\eta_o}+\frac{A_o\,ln\left(d_o/d_i\right)}{2\pi k_{tube}N_t L}+\left(\frac{A_o}{A_i}\right)\frac{R_{fi}}{\eta_i}+\left(\frac{A_o}{A_i}\right)\frac{1}{\eta_i h_i}$$

$$\frac{1}{U_i}=\left(\frac{A_i}{A_o}\right)\frac{1}{\eta_o h_o}+\left(\frac{A_i}{A_o}\right)\frac{R_{fo}}{\eta_o}+\frac{A_i\,ln\left(d_o/d_i\right)}{2\pi k_{tube}N_t L}+\frac{R_{fi}}{\eta_i}+\frac{1}{\eta_i h_i}$$

Log Mean Temperature Difference (LMTD) method of analysis

Assumptions: a) Fully developed conditions, b) Constant cross-sectional areas, c) Constant properties





(a) counterflow

(b) parallel flow

1) Counter Flow Heat Exchangers

$$Q = \dot{m}_c c_{pc}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{ph}\left(T_{h1} - T_{h2}\right) = U_o A_o \frac{\left(T_{h2} - T_{c1}\right) - \left(T_{h1} - T_{c2}\right)}{ln\left(\dfrac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}}\right)}$$

$$\frac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}} = exp\left(U_o A_o\left[\frac{1}{\dot{m}_c c_{pc}} - \frac{1}{\dot{m}_h c_{ph}}\right]\right)$$

2) Parallel Flow Heat Exchangers

$$Q = \dot{m}_c c_{pc}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{ph}\left(T_{h1} - T_{h2}\right) = U_o A_o \frac{\left(T_{h2} - T_{c2}\right) - \left(T_{h1} - T_{c1}\right)}{ln\left(\dfrac{T_{h2} - T_{c2}}{T_{h1} - T_{c1}}\right)}$$

$$\frac{T_{h2} - T_{c2}}{T_{h1} - T_{c1}} = exp\left(-U_o A_o\left[\frac{1}{\dot{m}_c c_{pc}} + \frac{1}{\dot{m}_h c_{ph}}\right]\right)$$

3) Other Types Heat Exchangers

$$Q = \dot{m}_c c_{pc}\left(T_{c2} - T_{c1}\right) = \dot{m}_h c_{ph}\left(T_{h1} - T_{h2}\right) = U_o A_o \Delta T_{lm}$$

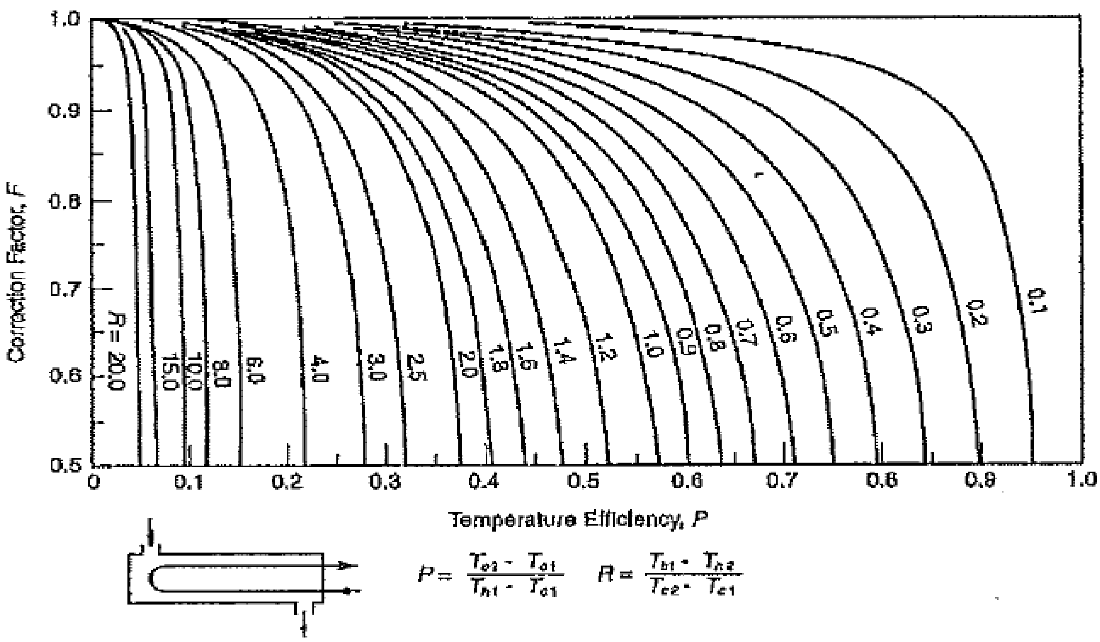
$$\Delta T_{lm} = F \times \Delta T_{lm,cf} = F \times \left[\frac{\left(T_{h2} - T_{c1}\right) - \left(T_{h1} - T_{c2}\right)}{ln\left(\dfrac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}}\right)}\right]$$

$$F : correction\ \ factor\ \left(F \le 1\right)$$

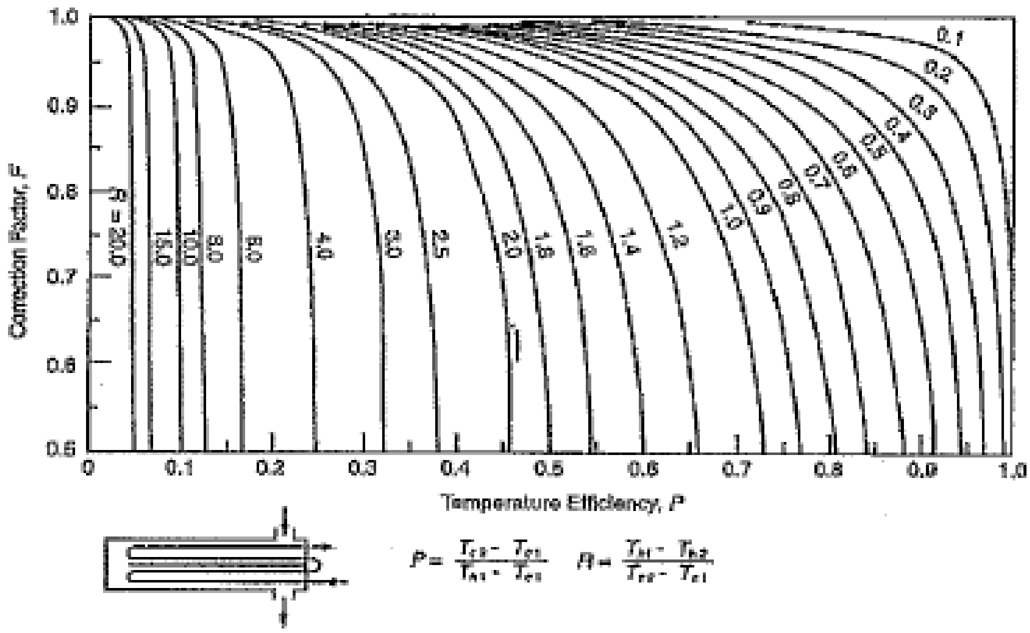
$$F = f\left(R \equiv \frac{\dot{m}_c c_{p,c}}{\dot{m}_h c_{p,h}} = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}}, P = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}}, flow\ arrangement\right)$$

$$P : Temperature\ efficiency$$

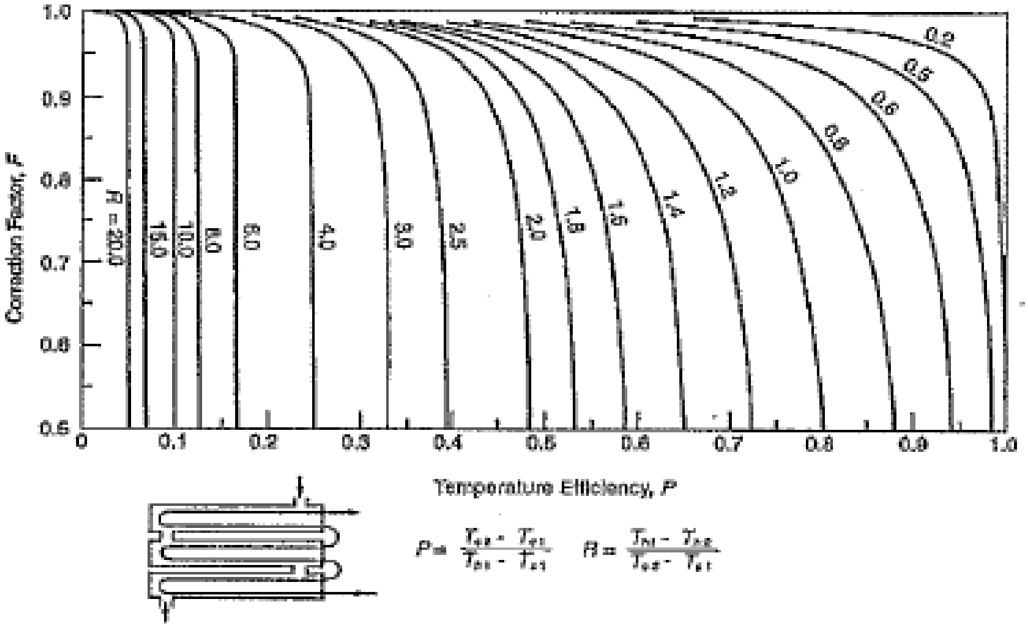
$$R : Heat\ capacity\ \ ratio$$



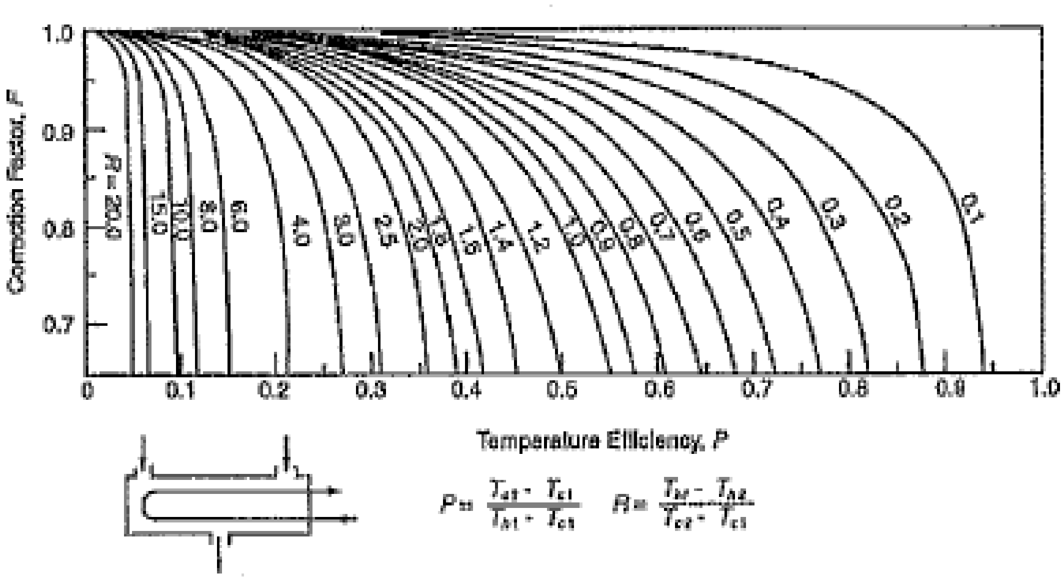
LMTD correction factor F for a shell-and-tube heat exchanger with one shell pass and two or a multiple of two tube passes. (From *Standards of the Tubular Exchanger Manufacturers Association* [1988]. ©1988 by Tubular Exchanger Manufacturers Association. With permission.)



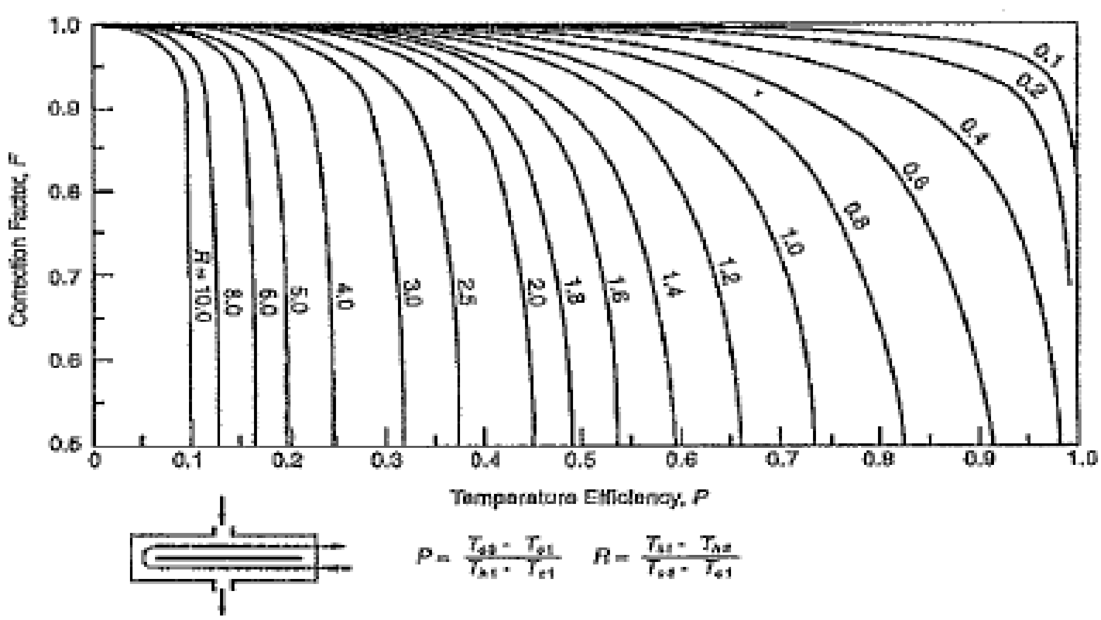
LMTD correction factor F for a shell-and-tube heat exchanger with two shell passes and four or a multiple of four tube passes. (From *Standards of the Tubular Exchanger Manufacturers Association* [1988]. ©1988 by Tubular Exchanger Manufacturers Association. With permission.)



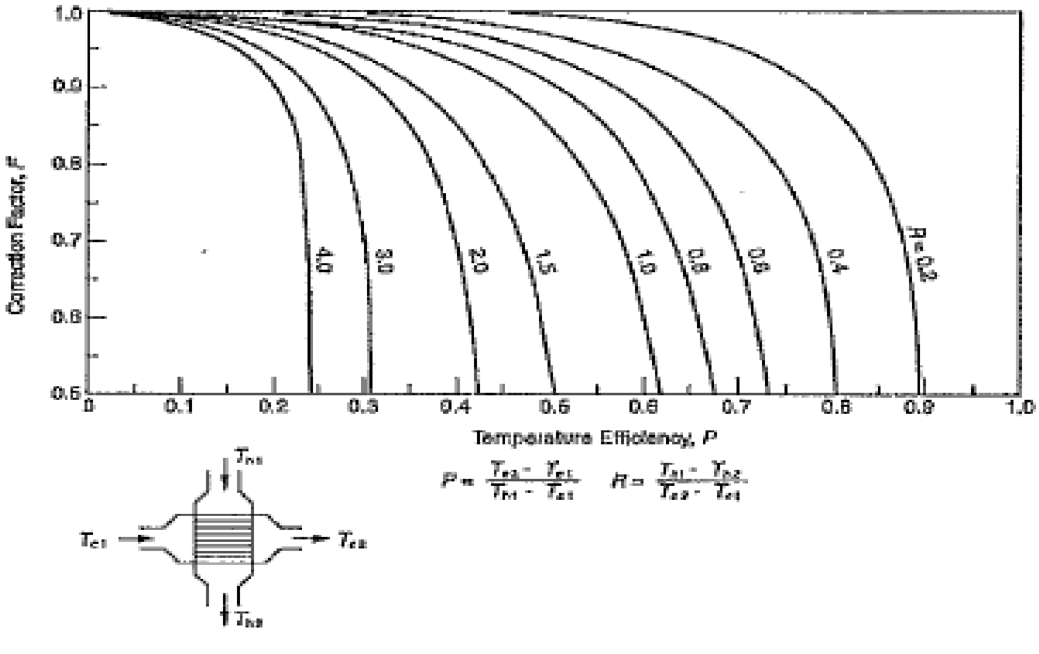
LMTD correction factor F for a shell-and-tube heat exchanger with three two-shell passes and six or more even number tube passes. (From *Standards of the Tubular Exchanger Manufacturers Association* [1988]. ©1988 by Tubular Exchanger Manufacturers Association. With permission.)



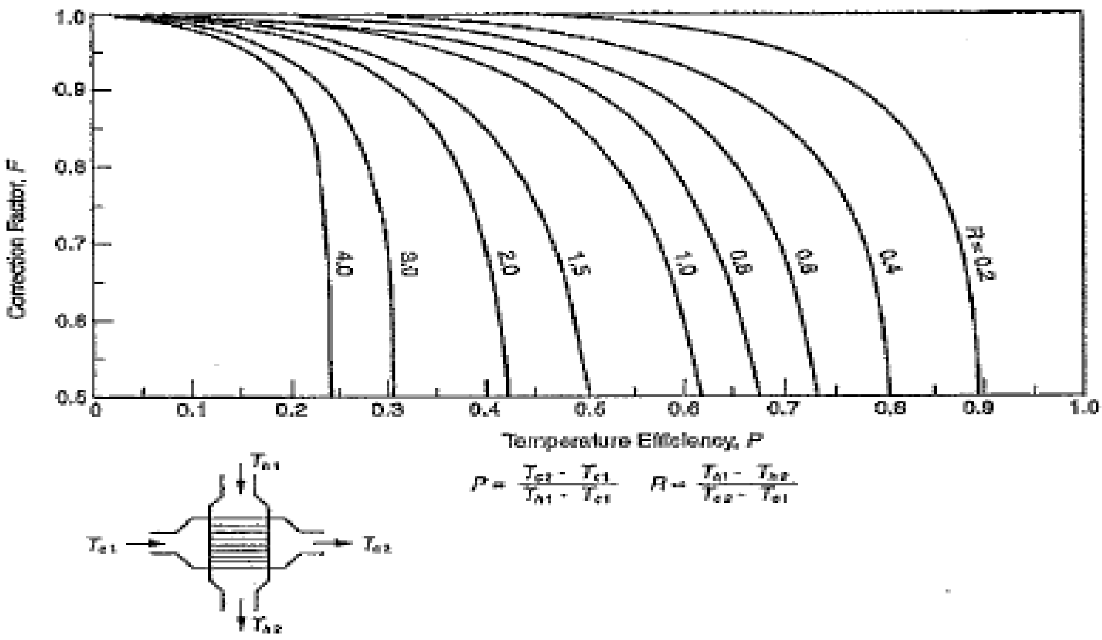
LMTD correction factor F for a divided-flow shell-type heat exchanger with one divided-flow shell pass and an even number of tube passes. (From *Standards of the Tubular Exchanger Manufacturers Association* [1988]. ©1988 by Tubular Exchanger Manufacturers Association. With permission.)



LMTD correction factor F for a split-flow shell-type heat exchanger with one split-flow shell pass and two-tube passes. (From *Standards of the Tubular Exchanger Manufacturers Association* [1988]. ©1988 by Tubular Exchanger Manufacturers Association. With permission.)



LMTD correction factor F for a crossflow heat exchanger with both fluids unmixed. (From Bowman, R. A., Mueller, A. C., and Nagle, W. M. [1940] *Trans. ASME*, Vol. 62, 283–294.)



LMTD correction factor F for a single-pass crossflow heat exchanger with one fluid mixed and the other unmixed. (From Bowman, R. A., Mueller, A. C., and Nagle, W. M. [1940] *Trans. ASME*, Vol. 62, 283–294.)

The ε-NTU method of heat exchanger analysis

Assumptions: a) Fully developed conditions, b) Constant cross-sectional areas, c) Constant properties

Definitions and relationships

$$NTU \equiv \frac{U_o A_o}{C_{min}}; \quad C_r \equiv \frac{C_{min}}{C_{max}}; \quad \varepsilon \equiv \frac{Q}{Q_{max}}$$

$$NTU : Number\ of\ Transfer\ Units$$

$$C_{min} = Minimum\ of\ \left(C_c, C_h\right)$$

$$C_{max} = Maximum\ of\ \left(C_c, C_h\right)$$

$$C_c = \dot{m}_c c_{pc}; \quad C_h = \dot{m}_h c_{ph}$$

$$C_c, C_h : thermal\ capacity\ of\ \left(cold, hot\right)\ fluid\ flow$$

$$\varepsilon : heat\ exchanger\ effectiveness$$

$$Q_{max} = C_{min}\left(T_{h1} - T_{c1}\right)$$

$$Q = C_c\left(T_{c2} - T_{c1}\right) = C_h\left(T_{h1} - T_{h2}\right)$$

$$\varepsilon = \frac{Q}{C_{min}\left(T_{h1} - T_{c1}\right)} = \frac{C_c\left(T_{c2} - T_{c1}\right)}{C_{min}\left(T_{h1} - T_{c1}\right)} = \frac{C_h\left(T_{h1} - T_{h2}\right)}{C_{min}\left(T_{h1} - T_{c1}\right)}$$

$$\varepsilon = f\left(NTU, C_r,\ flow\ arragement\right)$$

$$NTU = g\left(\varepsilon, C_r,\ flow\ arragement\right)$$

Heat Exchanger Effectiveness Relations

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

$\varepsilon_1 \equiv \varepsilon(n=1)$;
$NTU_1 \equiv NTU(n=1)$
$NTU = n \times NTU_1$

(1) C_r
(2) NTU
(3) NTU_1
(4) ε_1
(5) ε
(6) Q

Heat Exchanger NTU Relations

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

$\varepsilon_1 \equiv \varepsilon(n=1);$
$NTU_1 \equiv NTU(n=1)$
$NTU = n \times NTU_1$

(1) C_r
(2) ε
(3) F
(4) ε_1
(5) E
(6) NTU_1
(7) NTU
(8) U_oA_o

Heat exchanger with condensing fluid

$$Q = \dot{m}_c c_{pc}(T_{c2}-T_{c1}) = \dot{m}_h h_{fg} = U_oA_o\frac{(T_h-T_{c1})-(T_h-T_{c2})}{ln\left(\dfrac{T_h-T_{c1}}{T_h-T_{c2}}\right)}$$

$$F = 1.0; \quad C_h \to \infty; \quad C_r \to 0$$

Heat exchanger with evaporating fluid

$$Q = \dot{m}_c h_{fg} = \dot{m}_h c_{ph}(T_{h1}-T_{h2}) = U_oA_o\frac{(T_{h2}-T_c)-(T_{h1}-T_c)}{ln\left(\dfrac{T_{h2}-T_c}{T_{h1}-T_c}\right)}$$

$$F = 1.0; \quad C_c \to \infty; \quad C_r \to 0$$

Basic heat exchanger equation under variable U_o-coefficient

$$Q = U_{o,m} A_o F \frac{\left(T_{h2} - T_{c1}\right) - \left(T_{h1} - T_{c2}\right)}{ln\left(\dfrac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}}\right)}$$

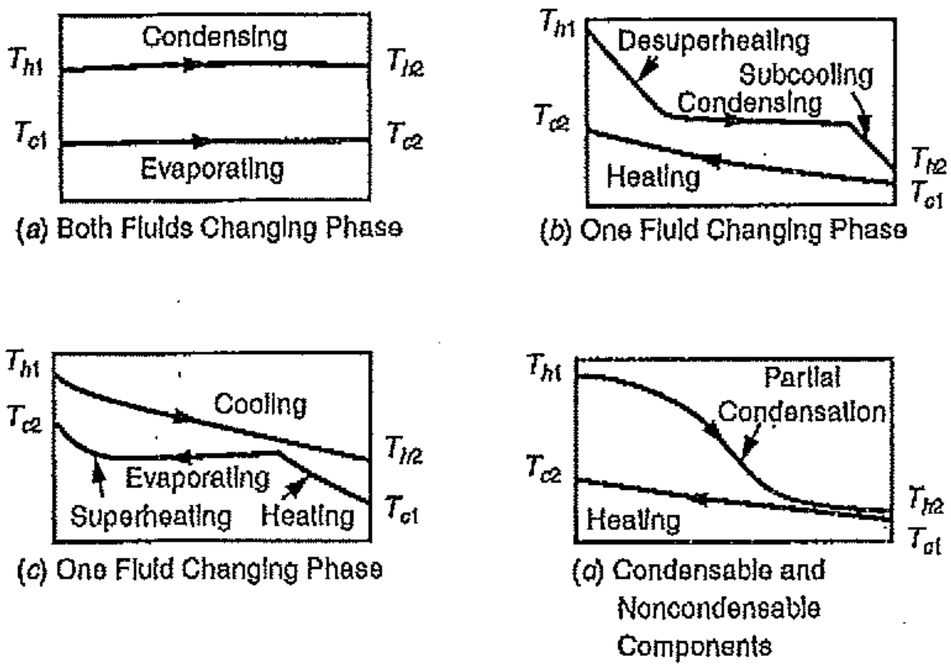
$$U_{o,m} = \frac{1}{A_o} \int_{A_o} U_o dA$$

- **Linear variation of U_o-coefficient with A**

$$U_{o,m} = \frac{U_{o,1} + U_{o,2}}{2}$$

- **Small variation of U_o-coefficient with A**

$$U_{o,m} = U_o\left(A = A_o/2\right)$$



Typical cases of a heat exchanger with variable U. (From Walker, G. [1990] *Industrial Heat Exchanger, A Basic Guide*, Hemisphere, Washington, D.C.)

Heat exchanger rating calculations

- **Known quantities:** Inlet temperatures, mass flow rates, outer surface area
  1) Calculate the overall heat transfer coefficient U_o.
  2) Calculate thermal capacities C_c and C_h.
  3) Calculate C_{min}, C_{max} and C_r.
  4) Calculate NTU.
  5) Calculate the heat exchanger effectiveness ε using ε-NTU relationships.
  6) Determine the heat transfer rate Q.

Heat exchanger design calculations using LMTD method

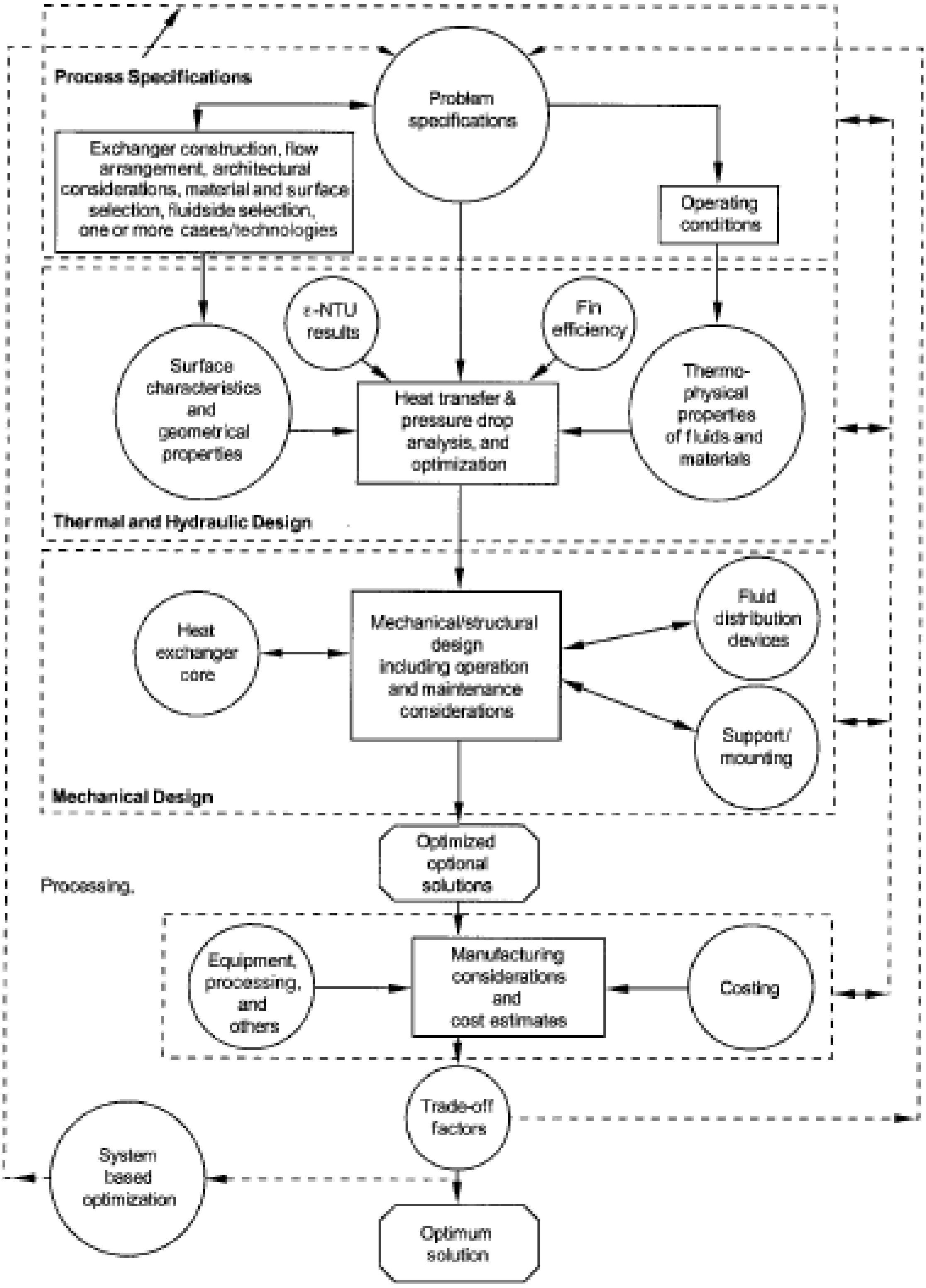
- **Known quantities:** Inlet temperatures, mass flow rates, one outlet temperature
  1) Calculate the heat transfer rate Q.
  2) Calculate the unknown outlet temperature using the conservation of energy principle.
  3) Calculate $\Delta T_{lm,cf}$.
  4) Calculate the correction factor F.
  5) Calculate the overall heat transfer coefficient U_o.
  6) Determine A_o.

Heat exchanger design calculations using ε-NTU method

- **Known quantities:** Inlet temperatures, mass flow rates, one outlet temperature

1) Calculate the heat transfer rate Q.
2) Calculate the unknown outlet temperature using the conservation of energy principle.
3) Calculate the overall heat transfer coefficient U_o.
4) Calculate thermal capacities C_c and C_h.
5) Calculate C_{min}, C_{max} and C_r.
6) Calculate the heat exchanger effectiveness ε.
7) Calculate the NTU using NTU-ε relationships.
8) Determine A_o.

Heat Exchanger Design Methodology



Heat exchanger design methodology. (Modified from Shah, 1982; Taborek, 1988; and Kays and London, 1998.)

Problems

2.10. 5000 kg/hr of water will be heated from 20°C to 35°C by hot water at 140°C. A 15°C hot water temperature drop is allowed. A number of double-pipe heat exchangers with annuli and pipes, each connected in series, will be used. Hot water flows through the inner tube. The thermal conductivity of the material is 50 W/m · K.

Fouling factors:	$R_i = 0.000176\ m^2 \cdot K/W$
	$R_o = 0.000352\ m^2 \cdot K/W$
Inner tube diameters:	ID = 0.0525 m, OD = 0.0603 m
Annulus diameters:	ID = 0.0779 m, OD = 0.0889 m

The heat transfer coefficients in the inner tube and in the annulus are 4620 $W/m^2 \cdot K$ and 1600 $W/m^2 \cdot K$, respectively. Calculate the overall heat transfer coefficient and the surface area of the heat exchanger.

2.13. Water (c_p = 4182 J/kg · K) at a flow rate of 5000 kg/hr is heated from 10°C to 35°C in an oil cooler by engine oil (c_p = 2072 J/kg · K) with an inlet temperature of 65°C and a flow rate of 6000 kg/hr. Take the overall heat transfer coefficient to be 3500 $W/m^2 \cdot K$. What are the areas required for:

a. Parallel flow

b. Counterflow

2.14. In order to cool a mass flow rate of 9.4 kg/h of air from 616°C to 232°C, it is passed through the inner tube of double-pipe heat exchanger with counterflow, which is 1.5 m long with an outer diameter of the inner tube of 2.5 cm.

a. Calculate the heat transfer rate. For air, c_{ph} = 1060 J/kg · K.

b. The cooling water enters the annular side at 16°C with a volume flow rate of 0.3 liter/min. Calculate the exit temperature of the water. For water, c_{pc} = 4180 J/kg · K.

c. Determine the effectiveness of this heat exchanger, then determine NTU. The overall heat transfer coefficient based on the outside heat transfer surface area is 38.5 $W/m^2 \cdot K$. Calculate the surface area of the heat exchanger and the number of hairpins.

2.15. A shell-and-tube heat exchanger is designed to heat water from 40°C to 60°C with a mass flow rate of 20,000 kg/h. Water at 180°C flows through tubes with a mass flow rate of 10,000 kg/hr. The tubes have an inner diameter of $d_i = 20$ mm; the Reynolds number is $Re = 10{,}000$. The overall heat transfer coefficient based outside heat transfer surface area is estimated to be $U = 450\ \mathrm{W/m^2 \cdot K}$.

a. Calculate the heat transfer rate Q of the heat exchanger and the exit temperature of the hot fluid.

b. If the heat exchanger is counterflow with one tube and one shell pass, determine (by use of the LMTD and ε-NTU methods):

i. The outer heat transfer area

ii. The velocity of the fluid through the tubes

iii. The cross-sectional area of the tubes

iv. The number of the tubes and the length of the heat exchanger

2.16. An oil cooler is used to cool lubricating oil from 70°C to 40°C. The cooling water enters the exchanger at 15°C and leaves at 25°C. The specific heat capacities of the oil and water are 2 and 4.2 kJ/kg · K, respectively, and the oil flow rate is 4 kg/s.

a. Calculate the water flow rate required.

b. Calculate the true mean temperature difference for two-shell-pass-and-four-tube passes and one-shell-pass-and-two-tube passes shell-and-tube heat exchangers and an unmixed-unmixed crossflow configuration, respectively.

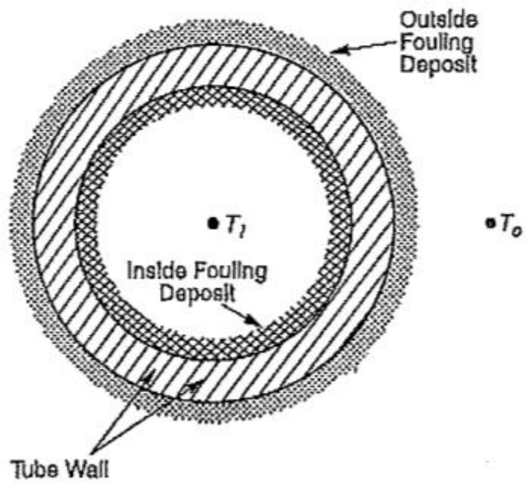
2.18. In an oil cooler, oil flows through the heat exchanger with a mass flow rate of 8 kg/s and inlet temperature of 70°C. The specific heat of oil is 2 kJ/kg · K. The cooling stream is treated cooling water that has a specific heat capacity of 4.2 kJ/kg · K, a flow rate of 20 kg/s, and an inlet temperature of 15°C. Assuming a total heat exchanger surface area of 150 m² and an overall heat transfer coefficient of 150 W/m² · K, calculate the outlet temperature for two-pass shell-and-tube and unmixed-unmixed crossflow units, respectively. Estimate the respective F-correction factors.

Fouling of Heat Exchangers

Fouling: It is accumulation of undesirable substances on a surface.

Examples on fouling

- Deposit of cholesterol on the inner surface of the artery wall. This deposit results in narrowing the blood flow cross-sectional area. Thus more pumping power is required by the heart to circulate the blood.
- Deposit of ashes on the tube surface which forms of a less thermally conducting layer on the surface. Thus, heat transfer rate is reduced.



Fouled tube

Disadvantages of fouling on heat exchanger operation

- Reducing heat transfer rate due to the increased thermal resistance of the deposit layer.
- Increasing pumping power requirement due to narrowing of the flow passages.

Definitions

U_{of} : *Overall heat transfer coefficient based on fouled condition*

U_{oc} : *Overall heat transfer coefficient based on clean condition*

$$\frac{1}{U_{of}} = \frac{1}{U_{oc}} + R_{ft}$$

$$R_{ft} : Total \;\; fouling \;\; factor$$

$$R_{ft} = R_{fo} + R_{fi}\left(\frac{d_o}{d_i}\right), \quad bare\,circular\;tube$$

$$R_{ft} = \frac{R_{fo}}{\eta_o} + \frac{R_{fi}}{\eta_i}\left(\frac{A_o}{A_i}\right), \quad finned\;tube$$

Design of heat exchangers based clean condition

$$Q = U_{oc} A_{oc} \Delta T_{lm}$$

Design of heat exchangers based fouled condition

$$Q = U_{of} A_{of} \Delta T_{lm}$$

Relationship between clean and fouled conditions based designs

$$U_{of} A_{of} \Delta T_{lm} = U_{oc} A_{oc} \Delta T_{lm}$$

$$U_{of} A_{of} = U_{oc} A_{oc}$$

$$\frac{A_{of}}{A_{oc}} = \frac{U_{oc}}{U_{of}} = 1 + U_{oc} R_{ft} \geq 1.0$$

Heat exchanger operation under fouled condition

A. Effect of fouling on pressure drop and pumping power

$$\Delta p_c = f_c \frac{L}{d_{ic}} \rho \frac{u_{m,c}^2}{2}; \quad \Delta p_f = f_f \frac{L}{d_{if}} \rho \frac{u_{m,f}^2}{2}$$

$$f_c \cong f_f$$

$$\dot{m}_c = \dot{m}_f \rightarrow \rho u_{m,c} \frac{\pi}{4} d_{ic}^2 = \rho u_{m,f} \frac{\pi}{4} d_{if}^2$$

$$\frac{u_{m,f}}{u_{m,c}} = \left(\frac{d_{ic}}{d_{if}}\right)^2$$

$$\frac{\dot{P}_f}{\dot{P}_c} = \frac{\Delta p_f}{\Delta p_c} \cong \left(\frac{d_{ic}}{d_{if}}\right)^5 \geq 1.0$$

$$\boxed{\frac{d_{ic}}{d_{if}} = 1.25 \qquad \frac{\dot{P}_f}{\dot{P}_c} \cong (1.25)^5 = 3.05}$$

B. Effect of fouling on flow rate

$$\dot{m}_c = \rho \frac{\pi}{128} d_{ic}^4 \left(\frac{\Delta p_c}{\mu L}\right); \quad \dot{m}_f = \rho \frac{\pi}{128} d_{if}^4 \left(\frac{\Delta p_f}{\mu L}\right)$$

$$\Delta p_c = \Delta p_f$$

$$\frac{\dot{m}_f}{\dot{m}_c} = \left(\frac{d_{if}}{d_{ic}}\right)^4$$

$$\boxed{\frac{d_{ic}}{d_{if}} = 1.25 \qquad \frac{\dot{m}_f}{\dot{m}_c} = (0.8)^4 = 0.4096}$$

C. Effect of fouling on heat transfer

$$Q_f = U_{of} A_o \Delta T_{lm,f}$$

$$Q_c = U_{oc} A_o \Delta T_{lm,c}$$

$$\frac{Q_f}{Q_c} = \left(\frac{U_{of}}{U_{oc}}\right)\left(\frac{\Delta T_{lm,f}}{\Delta T_{lm,c}}\right) = \left[\frac{1}{1 + U_{oc} R_{ft}}\right]\left(\frac{\Delta T_{lm,f}}{\Delta T_{lm,c}}\right)$$

Cost of fouling on industrial sector

- Increased capital cost (large pumps are required, large heat transfer area is required, stand by heat exchangers are required).
- Increased maintenance cost (to clean the deposits).
- Loss of production cost (shutting down the plant for cleaning the heat exchangers).
- Energy losses (Large electrical energy is required to operate the large pumps).

Categories of fouling

A. **Particulate fouling:** e.g. dust, ash.
B. **Crystallization fouling:** This arises due to presence of salts in the fluid.
C. **Corrosion fouling:** corrosive fluids may react with tube material producing corrosion products.
D. **Biofouling:** e.g. bacteria, algae and molds.
E. **Chemical reaction fouling:** The tube surface can act as catalyst expediting chemical reaction on the surface. e.g. polymerization.

Fundamental process of fouling

1. **Initiation of the surface:** e.g. removing of any protective layers.
2. **Transport of undesirable substances:** Mechanisms of this process are: a) mass diffusion, b) Inertial impaction, c) Sedimentation, d) Thermophoresis (motion of particles due to temperature gradients), d) Electrophoresis (motion of particles due to electrical potential gradients).
3. **Attachment:** depends on adhering forces between the deposits and the surface.
4. **Removal:** removal mechanisms are: a) Dissolution (removal by ions), b) Erosion (removal by small fouling masses), c) Spalling (removal by large fouling masses).
5. **Aging:** growing effect.

(Fouling is an extremely complex phenomenon)

Cleanliness Factor (CF) of the heat exchanger:

$$CF = \frac{U_{of}}{U_{oc}} = \frac{1}{1 + U_{oc} R_{ft}}$$

$$CF \approx 0.85, \quad Typical \quad designs$$

Percent Over Surface (OS) of the heat exchanger:

$$OS = 100\left(\frac{A_{of}}{A_{oc}} - 1\right)\% = 100 U_{oc} R_{ft} \%$$

$$OS \approx 17.6\%, \quad Typical \quad designs$$

TEMA values of fouling factors to be used in heat exchangers design

- **TEMA:** Tubular Exchangers Manufacturers Association
- Cleaning of heat exchanger can be started once the fouling factors reach TEMA values shown in the next tables.

Fouling Resistances for Water

Temperature of Heating Medium / Temperature of Water / Water Velocity (m/s)	Up to 115°C, 50°C: 0.9 and Less	Up to 115°C, 50°C: Over 0.9	R_f (m² · K/W) 115 to 205°C, Over 50°C: 0.9 and Less	R_f (m² · K/W) 115 to 205°C, Over 50°C: Over 0.9
Seawater	0.000088	0.000088	0.000176	0.000176
Brackish water	0.000352	0.000176	0.000528	0.000352
Cooling tower and artificial spray pond				
Treated make up	0.000176	0.000176	0.000352	0.000352
Untreated	0.000528	0.000528	0.000881	0.000705
City or well water	0.000176	0.000176	0.000352	0.000352
River water				
Minimum	0.000352	0.000176	0.000528	0.000352
Average	0.000528	0.000352	0.000705	0.000528
Muddy or silty	0.000528	0.000352	0.000705	0.000528
Hard (over 15 grains/gal)	0.000528	0.000528	0.000881	0.000881
Engine jacket	0.000176	0.000176	0.000176	0.000176
Distilled or closed cycle				
Condensate	0.000088	0.000088	0.000088	0.000088
Treated boiler feedwater	0.000176	0.000088	0.000176	0.000176
Boiler blowdown	0.000352	0.000352	0.000352	0.000352

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TEMA Design Fouling Resistances for Industrial Fluids

Industrial Fluids	R_f ($m^2 \cdot K/W$)
Oils	
Fuel oil no. 2	0.000352
Fuel oil no. 6	0.000881
Transformer oil	0.000176
Engine lube oil	0.000176
Quench oil	0.000705
Gases and Vapors	
Manufactured gas	0.001761
Engine exhaust gas	0.001761
Steam (nonoil bearing)	0.000088
Exhaust steam (oil bearing)	0.000264–0.000352
Refrigerant vapors (oil bearing)	0.000352
Compressed air	0.000176
Ammonia vapor	0.000176
CO_2 vapor	0.000176
Chlorine vapor	0.000352
Coal flue gas	0.001761
Natural gas flue gas	0.000881
Liquids	
Molten heat transfer salts	0.000088
Refrigerant liquids	0.000176
Hydraulic fluid	0.000176
Industrial organic heat transfer media	0.000352
Ammonia liquid	0.000176
Ammonia liquid (oil bearing)	0.000528
Calcium chloride solutions	0.000528
Sodium chloride solutions	0.000528
CO_2 liquid	0.000176
Chlorine liquid	0.000352
Methanol solutions	0.000352
Ethanol solutions	0.000352
Ethylene glycol solutions	0.000352

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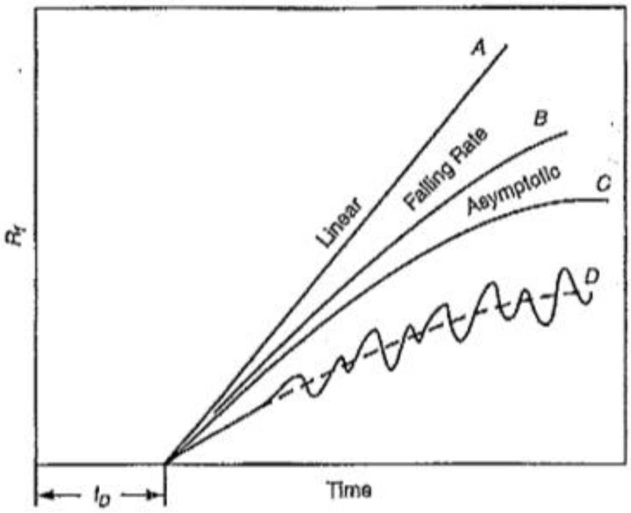
Prediction of fouling



Typical fouling factor-time curve. (From Chenoweth, J. M. [1987] In *Heat Transfer in High Technology and Power Engineering*, pp. 406-419. Hemisphere, New York. With permission.)

Effect of Parameters on Fouling

Parameter Increased	Deposition Rate	Removal Rate	Asymptotic Fouling
Stickness	Increases	Decreases	Increases
Surface temperature	Increases	Questionable	Increases
Toughness	Questionable	Decreases	Increases
Roughness	Increases (?)	Increases	Questionable
In situ corrosion	Increases	Questionable	Increases
Ex situ corrosion	Increases	Questionable	Increases
Velocity	Decreases	Increases	Decreases

Techniques to control fouling

Various Techniques to Control Fouling

On-Line Techniques	Off-Line Techniques
Use and control of appropriate additives	Disassembly and manual cleaning
Inhibitors	
Antiscalants	Lances
Dispursants	Liquid jet
Acids	Steam
	Air jet
On-Line Cleaning	Mechanical cleaning
Sponge balls	Drills
Brushes	Scrapers
Sonic horns	
Soot blowers	
Chains and scrapers	Chemical cleaning
Thermal shock	
Air bumping	

From Chenoweth, J. M. (1988) In *Fouling Science and Technology*, pp. 477-494. Kluwer, Dordrecht, The Netherlands. With permission.

Problems

5.1. The heat transfer coefficient of a steel ($k = 43$ W/m · K) tube (1.9 cm ID and 2.3 cm OD) in a shell-and-tube heat exchanger is 500 W/m^2 · K on the inside and 120 W/m^2 · K on the shell side, and it has a deposit with a total fouling factor of 0.000176 m^2 · K/W. Calculate

a. The overall heat transfer coefficient

b. The cleanliness factor, and percent over surface

5.3. Assume the water for a boiler is preheated using flue gases from the boiler stack. The flue gases are available at a rate of 0.25 kg/s at 150°C, with a specific heat of 1000 J/kg · K. The water entering the exchanger at 15°C at the rate of 0.05 kg/s is to be heated to 90°C. The heat exchanger is to be of the type with one shell pass and four tube passes. The water flows inside the tubes, which are made of copper (2.5 cm ID, 3.0 cm OD). The heat transfer coefficient on the gas side is 115 W/m^2 · K, while the heat transfer coefficient on the water side is 1150 W/m^2 · K. A scale on the water side and gas side offer an additional total thermal resistance of 0.000176 m^2 · K/W.

a. Determine the overall heat transfer coefficient based on the outer tube diameter.

b. Determine the appropriate mean temperature difference for the heat exchanger.

c. Estimate the required tube length.

d. Calculate the percent over surface design and the cleanliness factor.

5.5. In a double-pipe heat exchanger, deposits of calcium carbonate with a thickness of 1.12 mm and magnesium phosphate with a thickness of 0.88 mm on the inside and outside of the inner tube, respectively, have formed over time. Tubes (ID = 1.9 cm, OD = 2.3 cm) are made of carbon steel (k = 43 W/m · K). Calculate the total fouling resistance based on the outside surface area of the heat exchanger.

5.8. A counterflow double-pipe heat exchanger is designed to cool lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube is 0.2 kg/s, and the water enters the tubes at 20°C, while the flow rate of oil through the outlet annulus is 0.4 kg/s. The oil and water enter the heat exchanger at temperatures of 60°C and 30°C, respectively. The heat transfer coefficients in the annulus and in the inner tube have been estimated as 8 $W/m^2 \cdot K$ and 2445 $W/m^2 \cdot K$, respectively. The outer diameter of the inner tube is 25 mm, and the inner diameter of the outer tube is 45 mm. The total length of the double-pipe heat exchanger (total length per hairpin) is 15 m. In the analysis, the tube wall resistance and the curvature of the wall are neglected. What is the total value of the fouling resistance used in this design?

5.10. In a single-phase double-pipe heat exchanger, water is to be heated from 35°C to 95°C. The water flow rate is 4 kg/s. Condensing steam at 200°C in the annulus is used to heat the water. The overall heat transfer coefficient used in the design of this heat exchanger is 3500 $W/m^2 \cdot K$. After an operation of six months, the outlet temperature of the hot water drops to 90°C. Maintaining the outlet temperature of the water is essential for the purpose of this heat exchanger, therefore fouling is not acceptable. Calculate the total fouling factor under these operations and comment on whether the cleaning cycle must be extended.

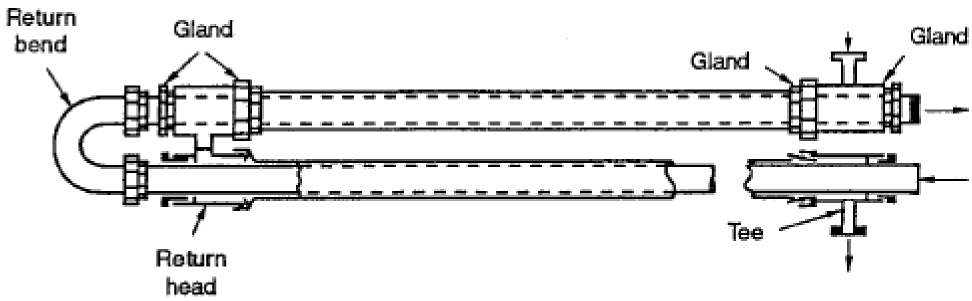
Double Pipe Heat Exchangers (DPHXs)

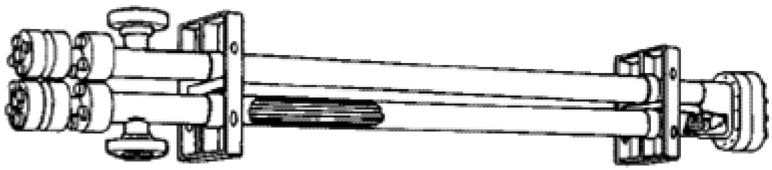
DPHX consists of one set of pipes of smaller diameter (tubes) placed concentrically inside a pipe of larger diameter (pipe or shell) with appropriate fittings to direct the flow from one section to the next.

Annulus is the volume between the outer surface of the tubes and the inner surface of the pipe.

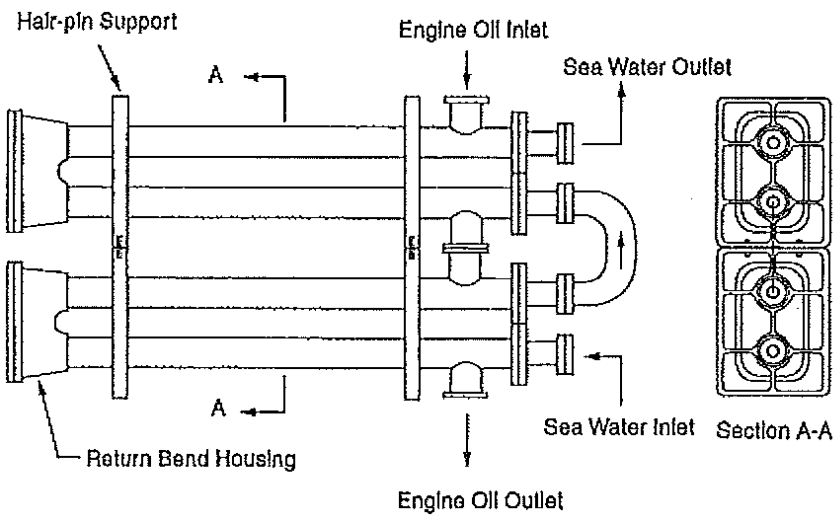
DPHXs are used majorly for sensible cooling/heating processes.

DPHXs exist in industry in form of hairpin heat exchangers as shown in the figures below.

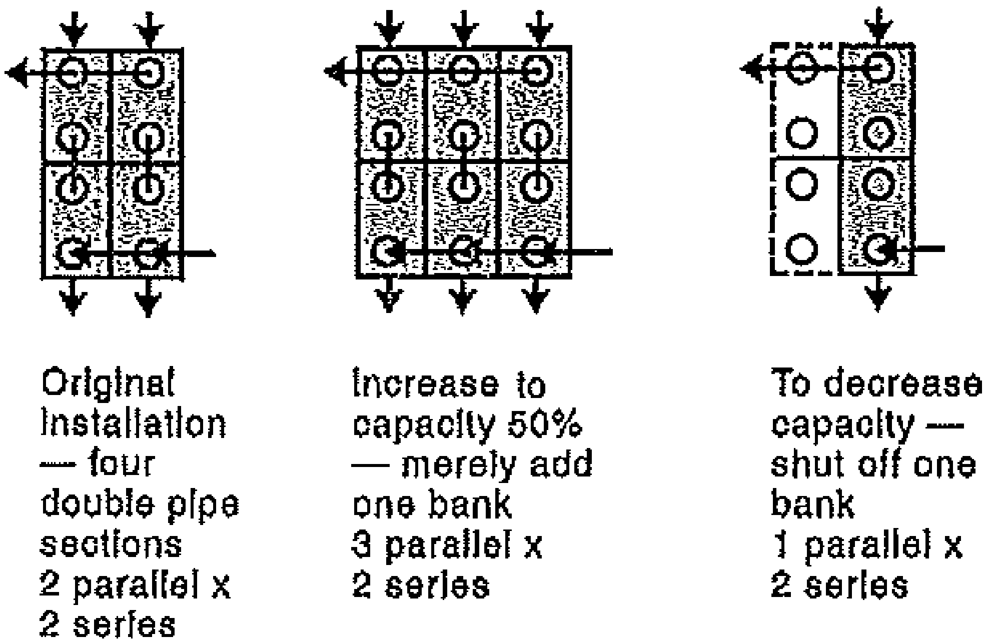




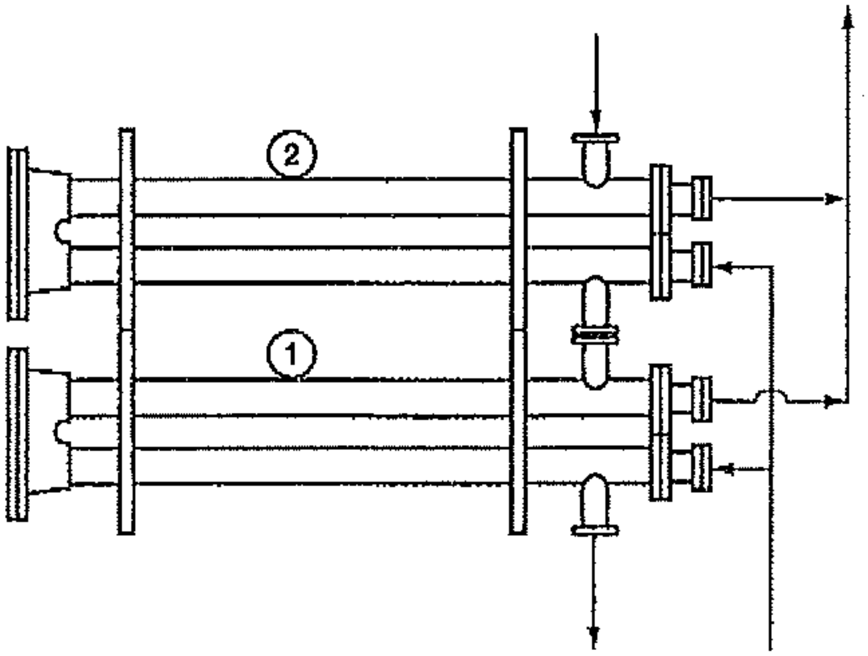
DPHXs can be arranged in series and parallel arrangements or combined arrangements.



Two hairpin sections arranged in series.



How hairpin sections are changed to meet new requirements

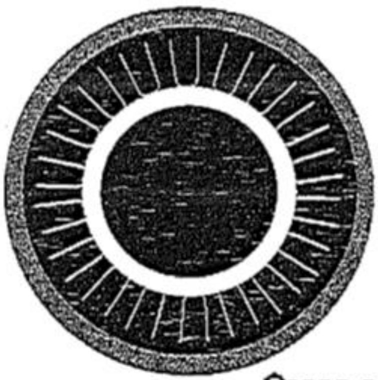


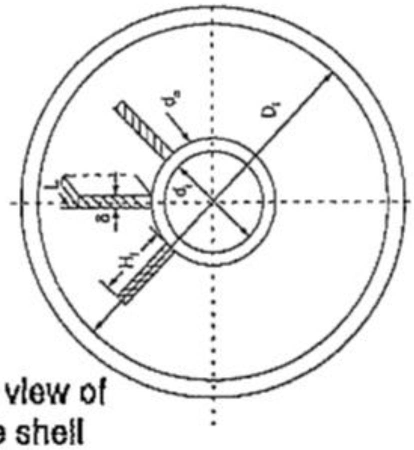
Two double pipe units in series on the annulus (shell) side and parallel on the tube side

Advantages of DPHXs: a) easy to be cleaned, b) easy to be maintained, c) fluids can be gases or liquids, d) tubes may have fins on their inner surface, outer surface or on both inner and outer surfaces (fins are usually placed on the surface that has minimum convection heat transfer coefficients).

Disadvantages of DPHXs: a) bulky, b) have small heat transfer area per unit volume, c) expensive per heat transfer area.

DPHX cross sections; See next figures for various cross-sections





Cross section view of fintube inside shell

Cross section view of bare tubes inside shell

Cross section view of fintubes inside shell

Pressure drop in single hairpin heat exchanger

$$u_{m,i} = \frac{\dot{m}_i}{\rho_i A_{C,i}}; \quad u_{m,o} = \frac{\dot{m}_o}{\rho_o A_{C,o}}$$

$$\Delta p_i = \left(4 f_{a,i} \frac{2L}{d_i} + K_{bend,S} + K_{inlet} + K_{outlet} \right) \rho_i \frac{u_{m,i}^2}{2}$$

$$\Delta p_o = \left(4 f_{a,o} \frac{2L}{D_{h,o}} + K_{bend,S} + K_{inlet} + K_{outlet} \right) \rho_o \frac{u_{m,o}^2}{2}$$

$$K_{inlet} \cong 1.0; \quad K_{outlet} \cong 0.5; \quad K_{bend,S} \cong 2.0$$

Pressure drop in N_{HP} hairpin heat exchangers arranged in series

$$u_{m,i}=\frac{\dot{m}_i}{\rho_i A_{C,i}};\quad u_{m,o}=\frac{\dot{m}_o}{\rho_o A_{C,o}}$$

$$\Delta p_i=\left(\left\{4f_{a,i}\frac{2L}{d_i}+K_{bend(i),S}+\frac{K_{bend(i),L}}{2}\right\}N_{HP}+K_{inlet(i)}+K_{outlet(i)}\right)\rho_i\frac{u_{m,i}^2}{2}$$

$$\Delta p_o=\left(\left\{4f_{a,o}\frac{2L}{D_{h,o}}+K_{bend(o),S}+\frac{K_{bend(o),L}}{2}\right\}N_{HP}+K_{inlet(o)}+K_{outlet(o)}\right)\rho_o\frac{u_{m,o}^2}{2}$$

$$K_{inlet(i,o)}\cong 1.0;\quad K_{outlet(i,o)}\cong 0.5;\quad K_{bend(i,o),S}\cong 2.0;\quad K_{bend(i,o),L}\cong 1.5$$

Thermal/hydraulic analysis of tubes with $N_{HP}=1$

$$D_{h,i}=\frac{4A_{C,i}}{P_{w,i}};\quad D_{e,i}=\frac{4A_{C,i}}{P_{h,i}}$$

$$Re_{D_{h,i}}=\frac{\rho_i u_{m,i} D_{h,i}}{\mu_i};\quad Nu_{D_{e,i}}=\frac{h_i D_{e,i}}{k_i}$$

- **Bare circular tubes of number N_t**

$$A_{C,i}=\frac{\pi}{4}d_i^2 N_t$$

$$P_{w,i}=P_{h,i}=\pi d_i N_t$$

$$A_i=\pi d_i N_t(2L)$$

- **Finned circular tubes of number N_t (rectangular fins)**

$$A_{C,i}=\frac{\pi}{4}d_i^2 N_t-H_{f,i}\delta_i N_{f,i}N_t$$

$$\boxed{(A_i)_1=A_i(N_{HP}=1)}$$

$$P_{w,i}=P_{h,i}=\pi d_i N_t+2H_{f,i}N_{f,i}N_t$$

$$\left(A_{u,i}\right)_1 = 2N_t L\left(\pi d_i - N_{f,i}\delta_i\right); \quad \left(A_{f,i}\right)_1 = 2N_t N_{f,i} L\left(2H_{f,i} + \delta_i\right)$$

$$\left(A_i\right)_1 = 2N_t L\left(\pi d_i + 2N_{f,i} H_{f,i}\right)$$

Convection heat transfer coefficient inside tubes (h_i)

- **Developing Laminar flow in a circular tube**
  - **Thermal Entry region**

$$Re_{d_i} = \frac{\rho_i u_{m,i} d_i}{\mu_i}$$

$$\overline{Nu}_{d_i} = \frac{\bar{h}_i d_i}{k_i} = 3.66 + \frac{0.0668\left(d_i/L\right) Re_{d_i}\; Pr_i}{1 + 0.04\left[\left(d_i/L\right) Re_{d_i}\; Pr_i\right]^{2/3}}$$

  - **Combined Entry region**

$$\overline{Nu}_{d_i} = \frac{\bar{h}_i d_i}{k_i} = 1.86\left(\frac{Re_{d_i}\; Pr_i\; d_i}{L}\right)^{1/3}\left(\frac{\mu_{b,i}}{\mu_{s,i}}\right)^{0.14}$$

- **Turbulent fully developed flow**

$$Nu_{D_{e,i}} = \frac{\left(f_{a,i}/2\right)\left(Re_{D_{h,i}} - 1000\right) Pr_i}{1 + 12.7\left(f_{a,i}/2\right)^{1/2}\left(Pr_i^{2/3} - 1\right)}$$

$$3\times10^3 \le Re_{D_{h,i}} \le 5\times10^6, \quad 0.5 \le Pr_i \le 2000$$

$$f_{a,i} = \left[1.58\,ln\left(Re_{D_{h,i}}\right) - 3.28\right]^{-2}, \quad 3000 \le Re_{D_{h,i}} \le 5\times10^6; \textit{ smooth tube}$$

$$\frac{1}{\sqrt{f_{a,i}}} \cong -3.6\,log_{10}\left[\frac{6.9}{Re_{D_{h,i}}} + \left(\frac{e/D}{3.7}\right)^{1.11}\right]; \textit{ roughened tube}$$

Thermal/hydraulic analysis of annulus with $N_{HP}=1$

$$D_{h,o} = \frac{4A_{C,o}}{P_{w,o}}; \quad D_{e,o} = \frac{4A_{C,o}}{P_{h,o}}$$

$$Re_{D_{h,o}} = \frac{\rho_o u_{m,o} D_{h,o}}{\mu_o}; \quad Nu_{D_{e,o}} = \frac{h_o D_{e,o}}{k_o}$$

- **Bare circular tubes of number N_t**

$$A_{C,o} = \frac{\pi}{4}\left(D_i^2 - d_o^2 N_t\right)$$

$$P_{w,o} = \pi\left(D_i + d_o N_t\right); \quad P_{h,o} = \pi d_o N_t$$

$$\left(A_o\right)_1 = \pi d_o N_t (2L)$$

- **Finned circular tubes of number N_t (rectangular fins)**

$$A_{C,o} = \frac{\pi}{4}\left(D_i^2 - d_o^2 N_t\right) - H_{f,o}\delta_o N_{f,o} N_t$$

$$\boxed{\left(A_o\right)_1 = A_o\left(N_{HP} = 1\right)}$$

$$P_{w,o} = \pi\left(D_i + d_o N_t\right) + 2H_{f,o} N_{f,o} N_t; \quad P_{h,o} = \pi d_o N_t + 2H_{f,o} N_{f,o} N_t$$

$$\left(A_{u,o}\right)_1 = 2N_t L\left(\pi d_o - N_{f,o}\delta_o\right); \quad \left(A_{f,o}\right)_1 = 2N_t N_{f,o} L\left(2H_{f,o} + \delta_o\right)$$

$$\left(A_o\right)_1 = 2N_t L\left(\pi d_o + 2N_{f,o} H_{f,o}\right)$$

Convection heat transfer coefficient outside tubes (h_o)

- **Developing Laminar flow**
  - **Thermal Entry region**

$$\overline{Nu}_{D_{e,o}} = \frac{\bar{h}_o D_{e,o}}{k_o} = 3.66 + \frac{0.0668\left(D_{h,o}/L\right) Re_{D_{h,o}}\, Pr_o}{1 + 0.04\left[\left(D_{h,o}/L\right) Re_{D_{h,o}}\, Pr_o\right]^{2/3}}$$

  - **Combined Entry region**

$$\overline{Nu}_{D_{e,o}} = \frac{\bar{h}_o D_{e,o}}{k_o} = 1.86\left(\frac{Re_{D_{h,o}}\, Pr_o\, D_{h,o}}{L}\right)^{1/3}\left(\frac{\mu_{b,o}}{\mu_{s,o}}\right)^{0.14}$$

- **<u>Turbulent</u> <u>fully developed flow</u>**

$$Nu_{D_{e,o}}=\frac{(f_{a,o}/2)(Re_{D_{h,o}}-1000)Pr_o}{1+12.7(f_{a,o}/2)^{1/2}(Pr_o^{2/3}-1)}$$

$$3\times10^3\le Re_{D_{h,o}}\le 5\times10^6,\quad 0.5\le Pr_o\le 2000$$

$$f_{a,o}=\left[1.58\,ln\left(Re_{D_{h,o}}\right)-3.28\right]^{-2},\quad 3000\le Re_{D_{h,o}}\le 5\times10^6;\ smooth\ tube$$

$$\frac{1}{\sqrt{f_{a,o}}}\cong -3.6\,log_{10}\left[\frac{6.9}{Re_{D_{h,o}}}+\left(\frac{e/D}{3.7}\right)^{1.11}\right]$$

<u>Thermal analysis of N_{HP} hairpin series heat exchangers</u>

$$Q=\dot{m}_c c_{pc}(T_{c2}-T_{c1})=\dot{m}_h c_{ph}(T_{h1}-T_{h2})=U_oA_o\frac{(T_{h2}-T_{c1})-(T_{h1}-T_{c2})}{ln\left(\frac{T_{h2}-T_{c1}}{T_{h1}-T_{c2}}\right)}$$

$$A_o=(A_o)_1N_{HP};\quad (A_o)_1\equiv A_o(N_{HP}=1)$$

- **<u>Bare</u> <u>circular</u> tubes**

$$\frac{1}{U_{of}}=\frac{1}{h_o}+R_{fo}+\frac{d_o\,ln(d_o/d_i)}{2k_{tube}}+\left(\frac{d_o}{d_i}\right)R_{fi}+\left(\frac{d_o}{d_i}\right)\frac{1}{h_i}$$

- **<u>Circular</u> tubes with <u>fins on their outer surfaces</u>**

$$\frac{1}{U_{of}}=\frac{1}{\eta_oh_o}+\frac{R_{fo}}{\eta_o}+\frac{(A_o)_1\,ln(d_o/d_i)}{2\pi k_{tube}N_t(2L)}+\left[\frac{(A_o)_1}{(A_i)_1}\right]R_{fi}+\left[\frac{(A_o)_1}{(A_i)_1}\right]\frac{1}{h_i}$$

- **<u>Circular</u> tubes with <u>fins on their inner surfaces</u>**

$$\frac{1}{U_{of}}=\frac{1}{h_o}+R_{fo}+\frac{(A_o)_1\,ln(d_o/d_i)}{2\pi k_{tube}N_t(2L)}+\left[\frac{(A_o)_1}{(A_i)_1}\right]\frac{R_{fi}}{\eta_i}+\left[\frac{(A_o)_1}{(A_i)_1}\right]\frac{1}{\eta_ih_i}$$

- **<u>Finned</u> <u>circular</u> tubes with <u>fins on inner and outer surfaces</u>**

$$\frac{1}{U_{of}}=\frac{1}{\eta_o h_o}+\frac{R_{fo}}{\eta_o}+\frac{(A_o)_1\, ln(d_o/d_i)}{2\pi k_{tube}N_t(2L)}+\left[\frac{(A_o)_1}{(A_i)_1}\right]\frac{R_{fi}}{\eta_i}+\left[\frac{(A_o)_1}{(A_i)_1}\right]\frac{1}{\eta_i h_i}$$

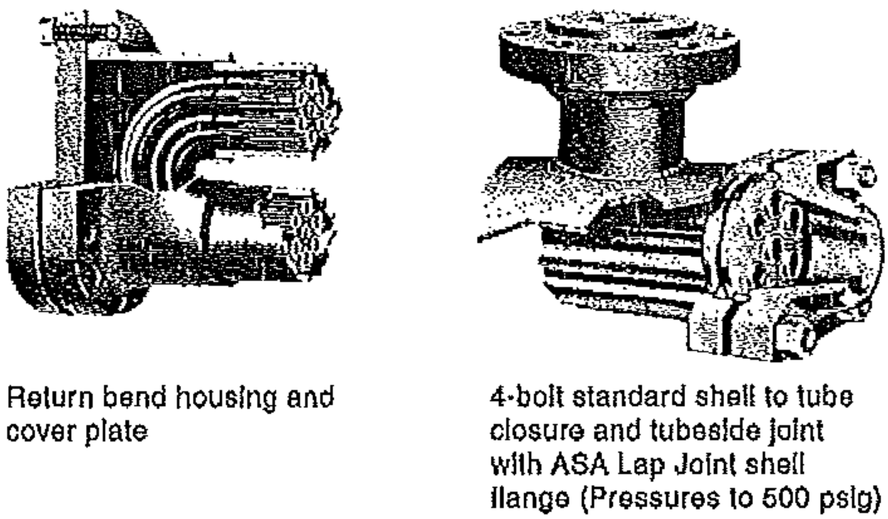
$$\eta_o=1-(1-\eta_{fo})\frac{(A_{fo})_1}{(A_o)_1};\quad \eta_i=1-(1-\eta_{fi})\frac{(A_{fi})_1}{(A_i)_1}$$

$$\eta_{fo}=\frac{tanh\left([H_{fo}+\delta_o/2]\sqrt{\frac{2h_o}{k_{fin,o}\delta_o}}\right)}{[H_{fo}+\delta_o/2]\sqrt{\frac{2h_o}{k_{fin,o}\delta_o}}};\quad \eta_{fi}=\frac{tanh\left([H_{fi}+\delta_i/2]\sqrt{\frac{2h_i}{k_{fin,i}\delta_i}}\right)}{[H_{fi}+\delta_i/2]\sqrt{\frac{2h_i}{k_{fin,i}\delta_i}}}$$

<u>Design and operational features</u>

DPHXs have four key design components:

1. Shell nozzles
2. Tube nozzles
3. Shell-to-tube closure
4. Return-bend housing



(a) Return-bend housing and the cover plate; (b) shell-to-tube closure and tube-side joint. (Courtesy of Brown Fintube, Inc.)

Problems

6.1. A counterflow double-pipe heat exchanger is used to cool the lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube is $m_c = 0.2$ kg/s, while the flow rate of oil through the outer annulus is $m_h = 0.4$ kg/s. The oil and water enter at temperatures of 60°C and 30°C, respectively. The heat transfer coefficient in the annulus is calculated to be 15 $W/m^2 \cdot K$. The inner tube diameter is 25 mm and the inside diameter of the outer annulus is 45 mm. The outlet temperature of the oil is 40°C. Assume $c_p = 4178$ $J/kg \cdot K$ for water and $c_p = 2006$ $J/kg \cdot K$ for oil. The tube wall resistance and the curvature of the wall are neglected. Calculate the length of the heat exchanger if fouling is neglected.

6.8. Sea water at 30°C flows on the inside of a 25 mm ID steel tube with a 0.8 mm wall thickness at a flow rate of 0.4 kg/s. The tube forms the inside of a double-pipe heat exchanger. Engine oil (refined lube oil) at 100°C flows in the annular space with a flow rate of 0.1 kg/s. The outlet temperature of the oil is 60°C. The material is carbon steel. The inner diameter of the outer tube is 45 ~~m~~ mm. Calculate:

a. The heat transfer coefficient in the annulus
b. The heat transfer coefficient inside the tube
c. The overall heat transfer coefficient with fouling
d. The area of the heat exchanger; assuming the length of a hairpin to be 4 m, calculate the number of hairpins
e. The pressure drops and pumping powers for both streams

Design Project 6.1

Oil Cooler: Finned-Tube Double-Pipe Heat Exchanger

The objective of this example is to design an oil cooler with sea water. The decision was made to use a hairpin heat exchanger.

Fluid	Annulus Fluid, Engine Oil	Tube-Side Fluid, Sea Water
Flow rate, kg/s	5	—
Inlet temperature, °C	65	20
Outlet temperature, °C	55	30
Density, kg/m³	885.27	1013.4
Specific heat, kJ/kg · °C	1.902	4.004
Viscosity, kg/m · s	0.075	9.64×10^{-4}
Thermal conductivity, W/m · K	0.1442	0.639
Prandtl number (Pr)	1050	6.29

Length of the hairpin = 3 m

Annulus nominal diameter = 2 inches

Nominal diameter of the inner tube = 3/4 inches

Fin height H_f = 0.0127 m

Fin thickness δ = 0.9 mm

Number of fins per tube = 30

Material throughout = carbon steel

Thermal conductivity, k = 52 W/m · K

Number of tubes inside the annulus (varied, starting with N_t = 1)

Select the proper fouling factors. The geometrical information is provided to initiate the analysis. These will be taken as variable parameters to come up with a suitable design; one can start with one inner tube and complete the hand calculations by then checking geometrical parameters to determine the effects of these changes on the design. Cost analysis of the selected design will be made. The final report will include material selection, mechanical design, and technical drawings of the components and the assembly. Calculate:

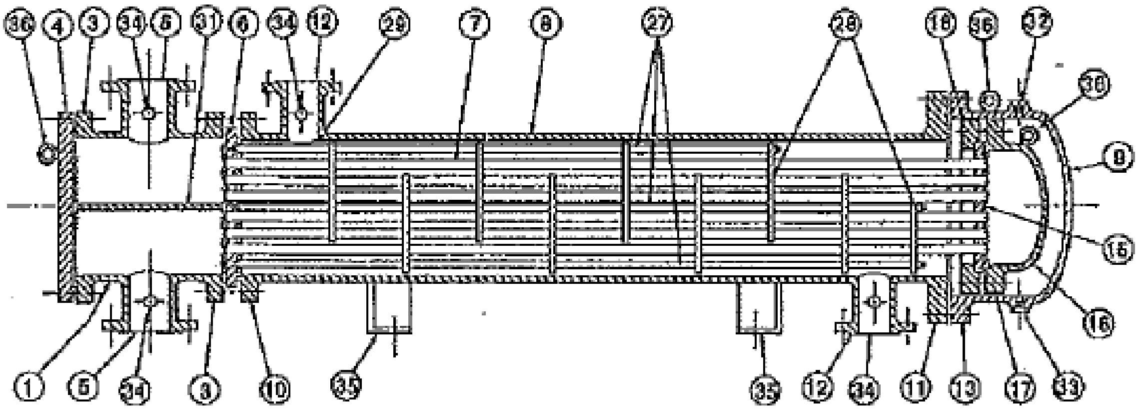
a. The velocities in the tube and in the annulus
b. The overall heat transfer coefficient for a clean and fouled heat exchanger
c. The total heat transfer area of the heat exchanger with and without fouling (OS design)

d. The surface area of a hairpin and the number of hairpins;
e. Pressure drop inside the tube and in the annulus
f. Pumping powers for both streams
g. Technical drawings

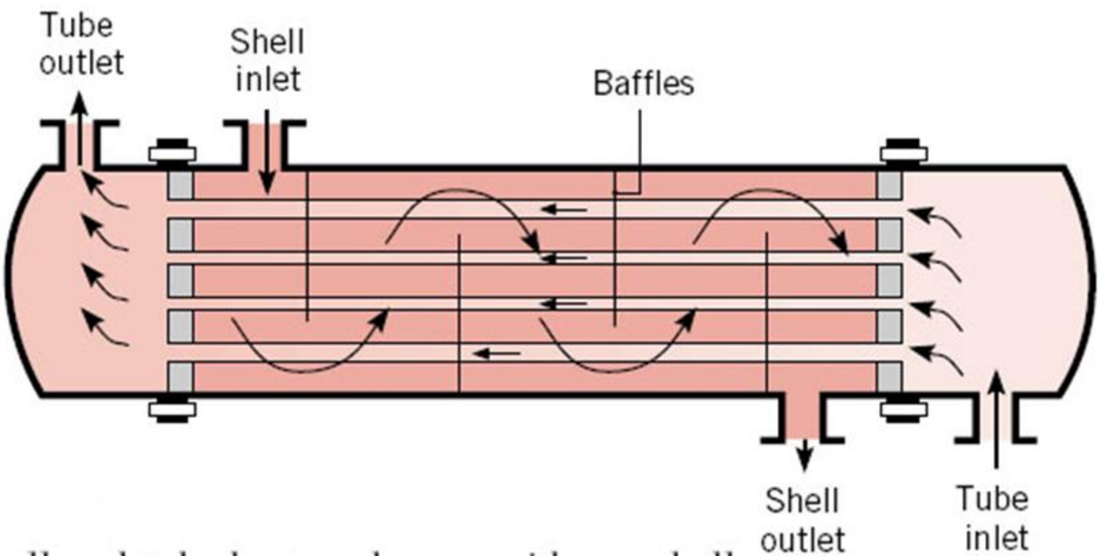
Shell-and-Tube Heat Exchangers (STHEs)

STHE basic components:

- Tube bundle
- Front end head
- Baffles
- Shell
- Rear end head
- Tube sheets



Constructional parts and connections: (1) stationary head — channel; (2) stationary head — bonnet; (3) stationary head flange — channel or bonnet; (4) channel cover; (5) stationary head nozzle; (6) stationary tube sheet; (7) tubes; (8) shell; (9) shell cover; (10) shell flange — rear head end; (11) shell flange — rear head end; (12) shell nozzle; (13) shell cover flange; (14) expansion joint; (15) floating tube sheet; (16) floating head cover; (17) floating head backing device; (18) floating head backing device; (19) split shear ring; (20) slip-on backing flange; (21) floating head cover — external; (22) floating tubesheet skirt; (23) packing box; (24) packing; (25) packing gland; (26) lantern ring; (27) tierods and spacers; (28) transverse baffle or support plates; (29) impingement plate; (30) longitudinal baffle; (31) pass partition; (32) vent connection; (33) drain connection; (34) instrument connection; (35) support saddle; (36) lifting lug; (37) support bracket; (38) weir; (39) liquid level connection. (Courtesy of the Tubular Exchanger Manufacturers Association.)



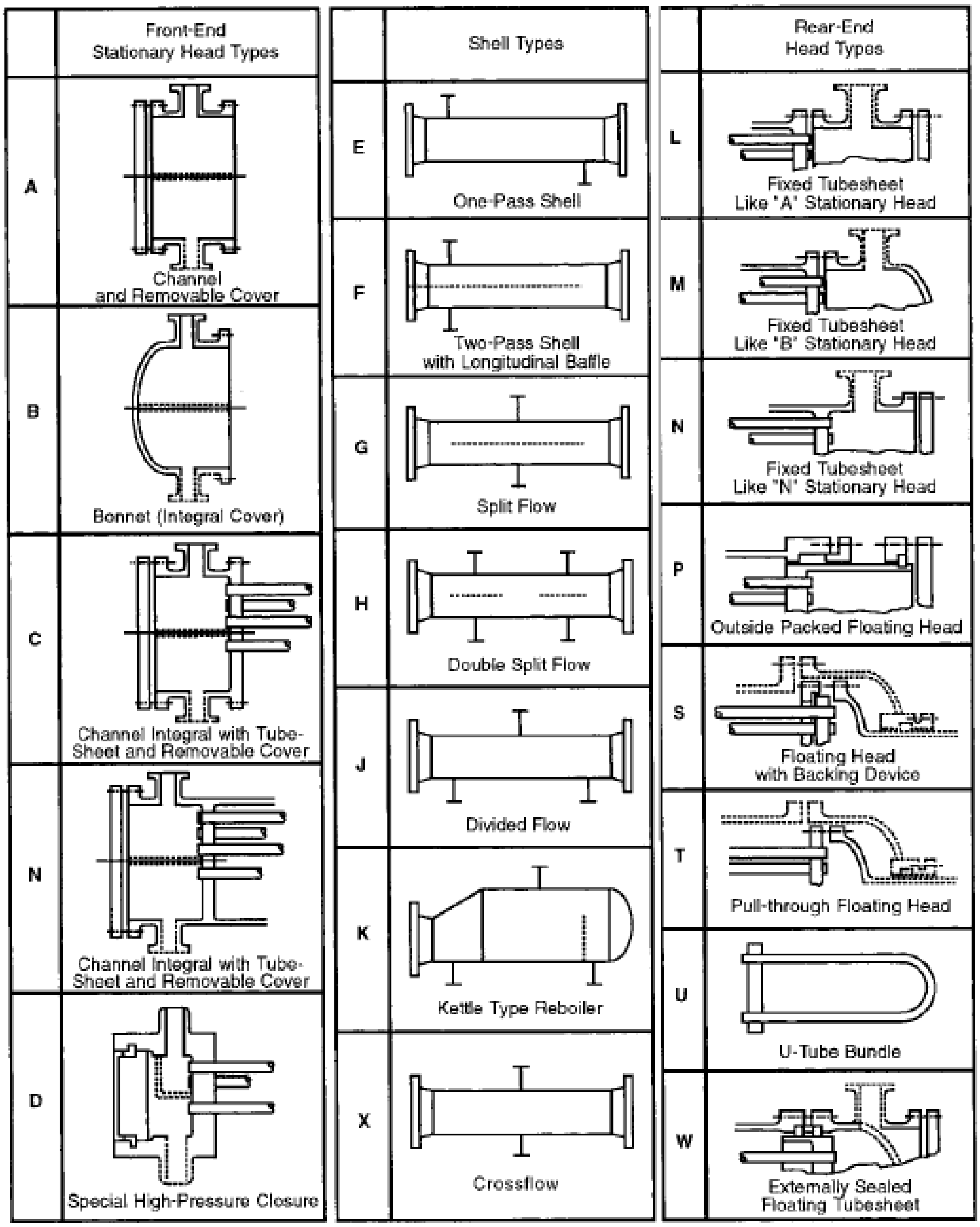
Shell-and-tube heat exchanger with one shell pass and one tube pass (cross-counterflow mode of operation).

STHEs have larger heat transfer area per unit volume than double pipe heat exchangers.

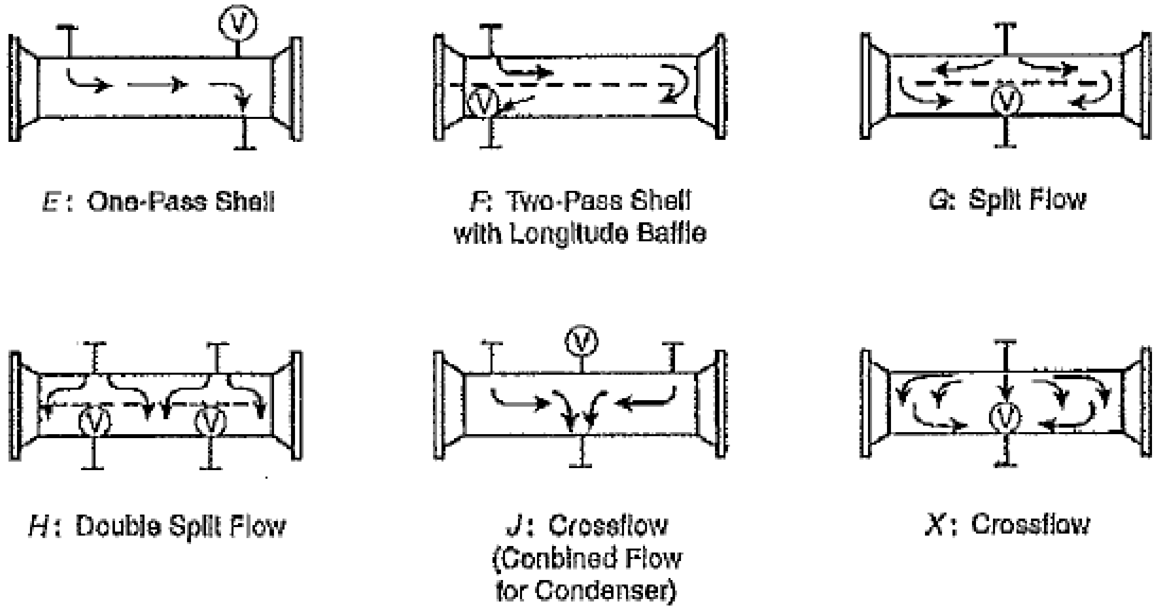
STHEs can be used for large pressure applications.

STHEs are easier to be cleaned than many types of heat exchangers such as compact heat exchangers.

TEMA Shell and end heads types



Standard shell types and front- and rear-end head types (From TEMA, 1999).



Schematic sketches of most common TEMA shell types. (From Butterworth, D. [1988] in *Two-Phase Flow Heat Exchangers: Thermal-Hydraulic Fundamentals and Design*, Kluwer Publishers, The Netherlands. With permission.)

$$\Delta p_{F-Shell} \approx 8\Delta p_{E-Shell}$$

$$\Delta p_{G-Shell} \approx \Delta p_{E-Shell}$$

$$\Delta p_{J-Shell} \approx \frac{1}{8}\Delta p_{E-Shell}$$

E-Shell: least expensive shell

E-Shell and 1P tubes: counter flow heat exchanger

F-Shell and 2P tubes: counter flow heat exchanger

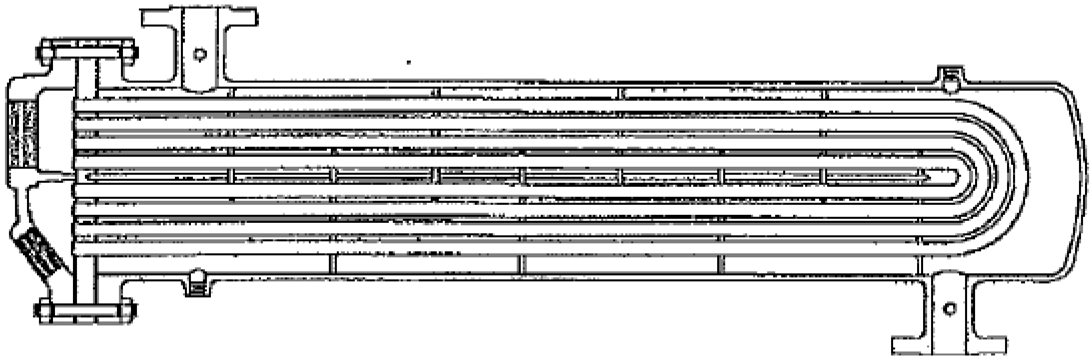
J-Shell: can be used as shell side horizontal condenser

G-Shell, J-Shell and X-Shell: cross flow heat exchanger

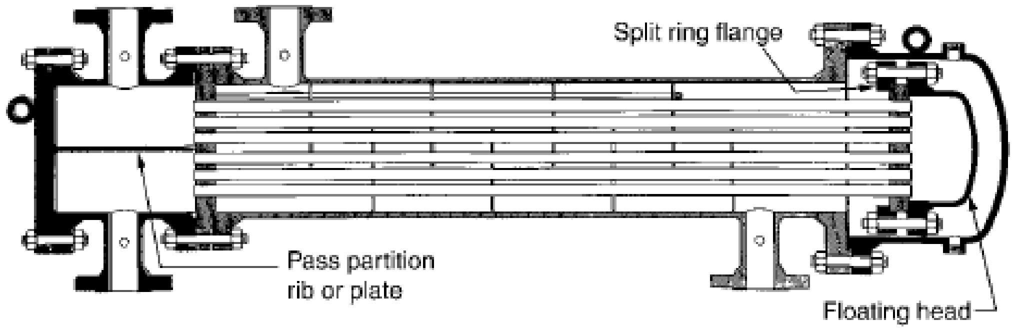
Tubes covers 60% of the shell diameter of K-Shell

<u>**Tube bundle**</u>

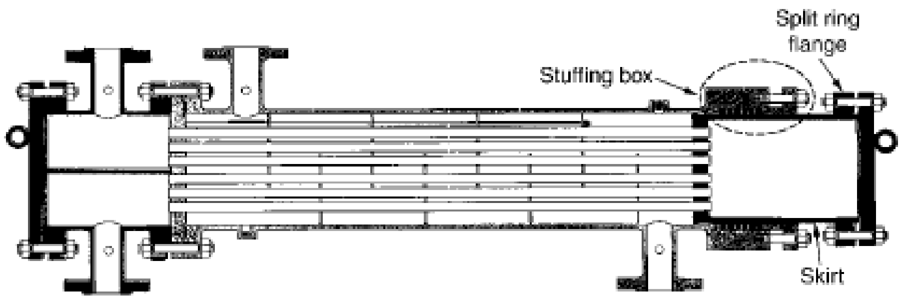
- They should accommodate thermal expansion.
- They should furnish ease of cleaning.
- They should provide least expensive construction.



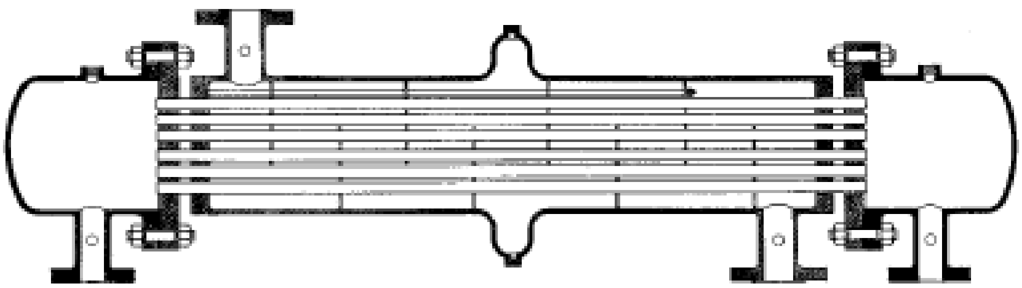
A bare U-tube, baffled single pass shell, shell-and-tube heat exchanger (courtesy of the Patterson-Kelley Co.)



Two-pass exchanger (AES) with a split-ring (S) floating head. (Courtesy of Patternson-Kelley CO., Division of HARSCO Corporation, East Stroudsburg, Pennsylvania.)



Two-pass exchanger (AEP) with an outside packed (P) floating head. (Courtesy of Patternson-Kelley Co., Division of HARSCO Corporation, East Stroudsburg, Pennsylvania.)



Two-pass exchanger (BET) with a pull-through (T) rear-end head. (Courtesy of Patternson-Kelley Co., Division of HARSCO Corporation, East Stroudsburg, Pennsylvania.)

Tubes and tube passes

- **Tube material:** low carbon steel, low alloy steel, stainless steel, copper, admiralty, cupronickel, inconel, alumnium (in form of alloys) or titanum.
- **Tube standards**

Heat Exchanger and Condenser Tube Data

Nominal Pipe Size (in.)	Outside Diameter (in.)	Schedule Number or Weight	Wall Thickness (in.)	Inside Diameter (in.)	Surface Area		Cross-Sectional Area	
					Outside (ft^2/ft)	Inside (ft^2/ft)	Metal Area ($in.^2$)	Flow Area ($in.^2$)
		40	0.113	0.824	0.275	0.216	0.333	0.533
3/4	1.05	80	0.154	0.742	0.275	0.194	0.434	0.432
		40	0.133	1.049	0.344	0.275	0.494	0.864
1	1.315	80	0.179	0.957	0.344	0.250	0.639	0.719
		40	0.140	1.38	0.434	0.361	0.668	1.496
1-1/4	1.660	80	0.191	1.278	0.434	0.334	0.881	1.283
		40	0.145	1.61	0.497	0.421	0.799	2.036
1-1/2	1.900	80	0.200	1.50	0.497	0.393	1.068	1.767
		40	0.154	2.067	0.622	0.541	1.074	3.356
2	2.375	80	0.218	1.939	0.622	0.508	1.477	2.953
		40	0.203	2.469	0.753	0.646	1.704	4.79
2-1/2	2.875	80	0.276	2.323	0.753	0.608	2.254	4.24
		40	0.216	3.068	0.916	0.803	2.228	7.39
3	3.5	80	0.300	2.900	0.916	0.759	3.106	6.60
		40	0.226	3.548	1.047	0.929	2.680	9.89
3-1/2	4.0	80	0.318	3.364	1.047	0.881	3.678	8.89
		40	0.237	4.026	1.178	1.054	3.17	12.73
4	4.5	80	0.337	3.826	1.178	1.002	4.41	11.50
		10 S	0.134	5.295	1.456	1.386	2.29	22.02
5	5.563	40	0.258	5.047	1.456	1.321	4.30	20.01
		80	0.375	4.813	1.456	1.260	6.11	18.19
		10 S	0.134	6.357	1.734	1.664	2.73	31.7
6	6.625	40	0.280	6.065	1.734	1.588	5.58	28.9
		80	0.432	5.761	1.734	1.508	8.40	26.1
		10 S	0.148	8.329	2.258	2.180	3.94	54.5
8	8.625	30	0.277	8.071	2.258	2.113	7.26	51.2
		80	0.500	7.625	2.258	1.996	12.76	45.7
		10 S	0.165	10.420	2.81	2.73	5.49	85.3
10	10.75	30	0.279	10.192	2.81	2.67	9.18	81.6
		Extra heavy	0.500	9.750	2.81	2.55	16.10	74.7
		10 S	0.180	12.390	3.34	3.24	7.11	120.6
	12.75	30	0.330	12.09	3.34	3.17	12.88	114.8
		Extra heavy	0.500	11.75	3.34	3.08	19.24	108.4
		10	0.250	13.5	3.67	3.53	10.80	143.1
14	14.0	Standard	0.375	13.25	3.67	3.47	16.05	137.9
		Extra heavy	0.500	13.00	3.67	3.40	21.21	132.7
		10	0.250	15.50	4.19	4.06	12.37	188.7
16	16.0	Standard	0.375	15.25	4.19	3.99	18.41	182.7
		Extra heavy	0.500	15.00	4.19	3.93	24.35	176.7
		10 S	0.188	17.624	4.71	4.61	10.52	243.9
18	18.0	Standard	0.375	17.25	4.71	4.52	20.76	233.7
		Extra heavy	0.500	17.00	4.71	4.45	27.49	227.0

Source: Courtesy of the Tubular Exchanger Manufacturers Association.

Dimensional Data for Commercial Tubing

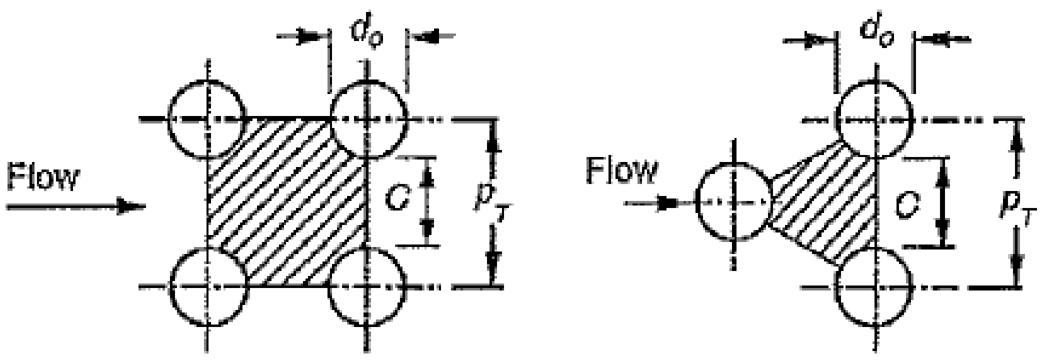
OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.^{2})	Sq. Ft. External Surface per Ft. Length	Sq. Ft. Internal Surface per Ft. Length	Weight per Ft. Length, Steel (lb.)	ID Tubing (in.)	OD/ID
1/4	22	0.028	0.0295	0.0655	0.0508	0.066	0.194	1.289
1/4	24	0.022	0.0333	0.0655	0.0539	0.054	0.206	1.214
1/4	26	0.018	0.0360	0.0655	0.0560	0.045	0.214	1.168
3/8	18	0.049	0.0603	0.0982	0.0725	0.171	0.277	1.354
3/8	20	0.035	0.0731	0.0982	0.0798	0.127	0.305	1.233
3/8	22	0.028	0.0799	0.0982	0.0835	0.104	0.319	1.176
3/8	24	0.022	0.0860	0.0982	0.0867	0.083	0.331	1.133
1/2	16	0.065	0.1075	0.1309	0.0969	0.302	0.370	1.351
1/2	18	0.049	0.1269	0.1309	0.1052	0.236	0.402	1.244
1/2	20	0.035	0.1452	0.1309	0.1126	0.174	0.430	1.163
1/2	22	0.028	0.1548	0.1309	0.1162	0.141	0.444	1.126
5/8	12	0.109	0.1301	0.1636	0.1066	0.602	0.407	1.536
5/8	13	0.095	0.1486	0.1636	0.1139	0.537	0.435	1.437
5/8	14	0.083	0.1655	0.1636	0.1202	0.479	0.459	1.362
5/8	15	0.072	0.1817	0.1636	0.1259	0.425	0.481	1.299
5/8	16	0.065	0.1924	0.1636	0.1296	0.388	0.49s	1.263
5/8	17	0.058	0.2035	0.1636	0.1333	0.350	0.509	1.228
5/8	18	0.049	0.2181	0.1636	0.1380	0.303	0.527	1.186
5/8	19	0.042	0.2298	0.1636	0.1416	0.262	0.541	1.155
5/8	20	0.035	0.2419	0.1636	0.1453	0.221	0.555	1.136
3/4	10	0.134	0.1825	0.1963	0.1262	0.884	0.482	1.556
3/4	11	0.120	0.2043	0.1963	0.1335	0.809	0.510	1.471
3/4	12	0.109	0.2223	0.1963	0.1393	0.748	0.532	1.410
3/4	13	0.095	0.2463	0.1963	0.1466	0.666	0.560	1.339
3/4	14	0.083	0.2679	0.1963	0.1529	0.592	0.584	1.284
3/4	15	0.072	0.2884	0.1963	0.1587	0.520	0.606	1.238
3/4	16	0.065	0.3019	0.1963	0.1623	0.476	0.620	1.210
3/4	17	0.058	0.3157	0.1963	0.1660	0.428	0.634	1.183
3/4	18	0.049	0.3339	0.1963	0.1707	0.367	0.652	1.150
3/4	20	0.035	0.3632	0.1963	0.1780	0.269	0.680	1.103
7/8	10	0.134	0.2892	0.2291	0.1589	1.061	0.607	1.441
7/8	11	0.120	0.3166	0.2291	0.1662	0.969	0.635	1.378
7/8	12	0.109	0.3390	0.2291	0.1720	0.891	0.657	1.332
7/8	13	0.095	0.3685	0.2291	0.1793	0.792	0.685	1.277
7/8	14	0.083	0.3948	0.2291	0.1856	0.704	0.709	1.234
7/8	16	0.065	0.4359	0.2291	0.1950	0.561	0.745	1.174
7/8	18	0.049	0.4742	0.2291	0.2034	0.432	0.777	1.126
7/8	20	0.035	0.5090	0.2291	0.2107	0.313	0.805	1.087
1	8	0.165	0.3526	0.2618	0.1754	1.462	0.670	1.493
1	10	0.134	0.4208	0.2618	0.1916	1.237	0.732	1.366
1	11	0.120	0.4536	0.2618	0.1990	1.129	0.760	1.316
1	12	0.109	0.4803	0.2618	0.2047	1.037	0.782	1.279
1	13	0.095	0.5153	0.2618	0.2121	0.918	0.810	1.235
1	14	0.083	0.5463	0.2618	0.2183	0.813	0.834	1.199
1	15	0.072	0.5755	0.2618	0.2241	0.714	0.856	1.167
1	16	0.065	0.5945	0.2618	0.2278	0.649	0.870	1.119

Dimensional Data for Commercial Tubing

OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.²)	Sq. Ft. External Surface per Ft. Length	Sq. Ft. Internal Surface per Ft. Length	Weight per Ft. Length, Steel (lb.)	ID Tubing (in.)	OD/ID
1	18	0.049	0.6390	0.2618	0.2361	0.496	0.902	1.109
1	20	0.035	0.6793	0.2618	0.2435	0.360	0.930	1.075
1-1/4	7	0.180	0.6221	0.3272	0.2330	2.057	0.890	1.404
1-1/4	8	0.165	0.6648	0.3272	0.2409	1.921	0.920	1.359
1-1/4	10	0.134	0.7574	0.3272	0.2571	1.598	0.982	1.273
1-1/4	11	0.120	0.8012	0.3272	0.2644	1.448	1.010	1.238
1-1/4	12	0.109	0.8365	0.3272	0.2702	1.329	1.032	1.211
1-1/4	12	0.095	0.8825	0.3272	0.2773	1.173	1.060	1.179
1-1/4	14	0.083	0.9229	0.3272	0.2838	1.033	1.084	1.153
1-1/4	16	0.065	0.9852	0.3272	0.2932	0.823	1.120	1.116
1-1/4	18	0.049	1.042	0.3272	0.3016	0.629	1.152	1.085
1-1/4	20	0.035	1.094	0.3272	0.3089	0.456	1.180	1.059
1-1/2	10	0.134	1.192	0.3927	0.3225	1.955	1.232	1.218
1-1/2	12	0.109	1.291	0.3927	0.3356	1.618	1.282	1.170
1-1/2	14	0.083	1.398	0.3927	0.3492	1.258	1.334	1.124
1-1/2	16	0.065	1.474	0.3927	0.3587	0.996	1.370	1.095
2	11	0.120	2.433	0.5236	0.4608	2.410	1.760	1.136
2	13	0.095	2.573	0.5236	0.4739	1.934	1.810	1.105
2-1/2	9	0.148	3.815	0.6540	0.5770	3.719	2.204	1.134

Source: Courtesy of the Tubular Exchanger Manufacturers Association.

Tube layouts



Square and triangular pitch-tube layouts

P_T: tube pitch; C: clearence; $C = P_T - d_o$

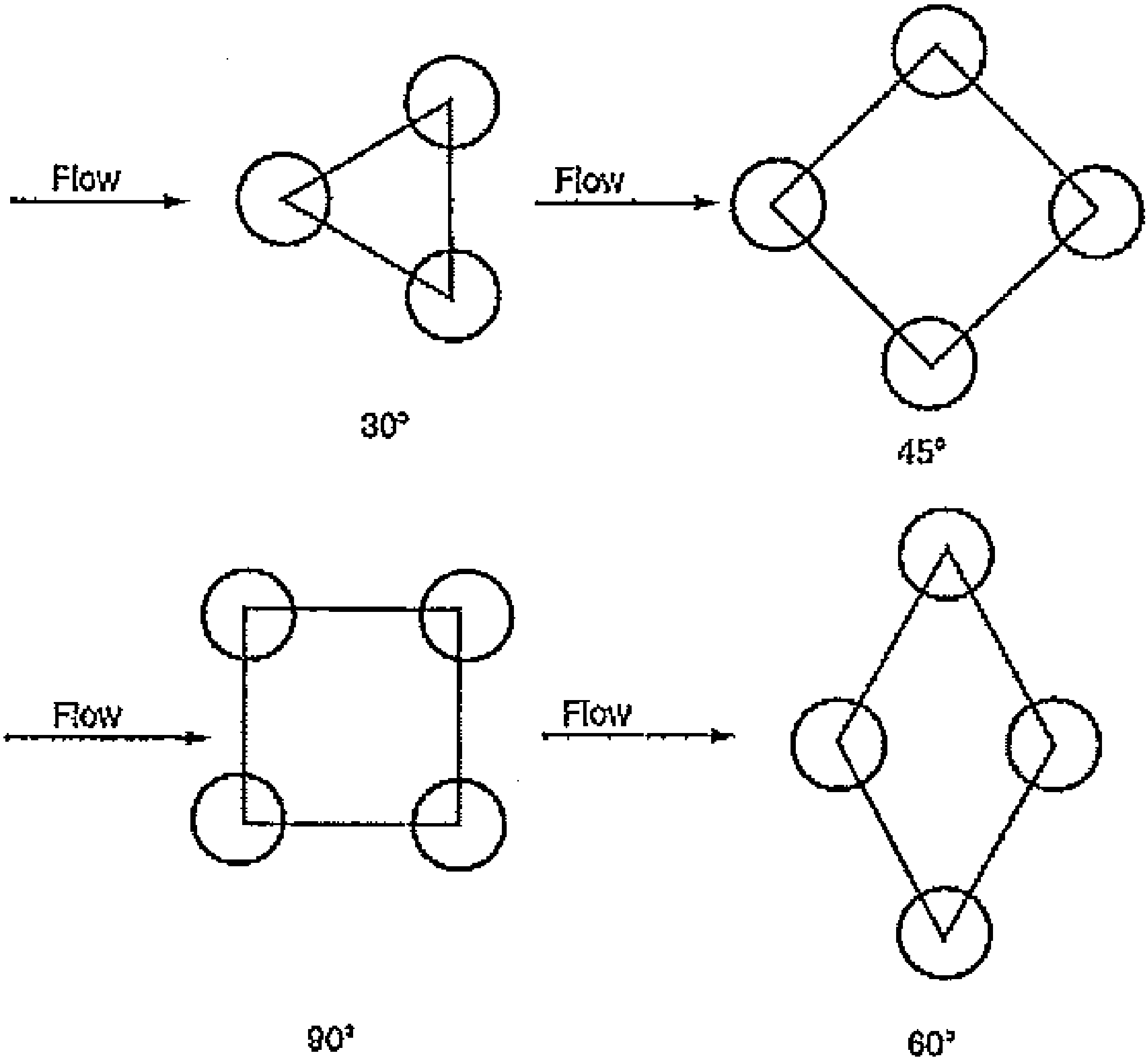
$C \geq 7\ mm$: cleaning requirment for square pitch

Traingular pitch results in largest tube density inside the shell

$1.25 \le \frac{P_T}{d_o} \le 1.5$: to avoid weak structurally tube sheets while enhancing heat transfer

$$(N_t)_{max} = f\left(D_i, d_o, N_P, P_T, tube\ layout\right)$$

$(N_t)_{max}$: Maximum tubes to be fit inside the shell – (Tube count)



Tube layout angles

Tube-Shell Layouts (Tube Counts)

3/4-in. O.D. Tubes on 1-in. Triangular Pitch

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
8	37	30	24	24	
10	61	52	40	36	
12	92	82	76	74	70
13 1/4	109	106	86	82	74
15 1/4	151	138	122	118	110
17 1/4	203	196	178	172	166
19 1/4	262	250	226	216	210
21 1/4	316	302	278	272	260
23 1/4	384	376	352	342	328
25	470	452	422	394	382
27	559	534	488	474	464
29	630	604	556	538	508
31	745	728	678	666	640
33	856	830	774	760	732
35	970	938	882	864	848
37	1074	1044	1012	986	870
39	1206	1176	1128	1100	1078

1-in. O.D. Tubes on 1 1/4-in. Triangular Pitch

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
8	21	16	16	14	
10	32	32	26	24	
12	55	52	48	46	44
13 1/4	68	66	58	54	50
15 1/4	91	86	80	74	72
17 1/4	131	118	106	104	94
19 1/4	163	152	140	136	128
21 1/4	199	188	170	164	160
23 1/4	241	232	212	212	202
25	294	282	256	252	242
27	349	334	302	296	286
29	397	376	338	334	316
31	472	454	430	424	400
33	538	522	486	470	454
35	608	592	562	546	532
37	674	664	632	614	598
39	766	736	700	688	672

3/4-in. O.D. Tubes on 1-in. Square Pitch

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
8	32	26	20	20	
10	52	52	40	36	
12	81	76	68	68	60
13 1/4	97	90	82	76	70
15 1/4	137	124	116	108	108
17 1/4	177	166	158	150	142
19 1/4	224	220	204	192	188

Tube-Shell Layouts (Tube Counts)

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
3/4-in. O.D. Tubes on 1-in. Square Pitch ? ($P_T \stackrel{?}{=}$ 1in. tube pitch)					
21 1/4	277	270	246	240	234
23 1/4	341	324	308	302	292
25	413	394	370	356	346
27	481	460	432	420	408
29	553	526	480	468	456
31	657	640	600	580	560
33	749	718	688	676	648
35	845	824	780	766	748
37	934	914	886	866	838
39	1049	1024	982	968	948
1-in. O.D. Tubes on 1 1/4-in. Square Pitch					
8	21	16	14		
10	32	32	26	24	
12	48	45	40	38	36
13 1/4	61	56	52	48	44
15 1/4	81	76	68	68	64
17 1/4	112	112	96	90	82
19 1/4	138	132	128	122	116
21 1/4	177	166	158	152	148
23 1/4	213	208	192	184	184
25	260	252	238	226	222
27	300	288	278	268	260
29	341	326	300	294	286
31	406	398	380	368	358
33	465	460	432	420	414
35	522	518	488	484	472
37	596	574	562	544	532
39	665	644	624	612	600
3/4-in. O.D. Tubes on 15/16-in. Triangular Pitch					
8	36	32	26	24	18
10	62	56	47	42	36
12	109	98	86	82	78
13 1/4	127	114	96	90	86
15 1/4	170	160	140	136	128
17 1/4	239	224	194	188	178
19 1/4	301	282	252	244	234
21 1/4	361	342	314	306	290
23 1/4	442	420	386	378	364
25	532	506	468	446	434
27	637	602	550	536	524
29	721	692	640	620	594
31	847	822	766	722	720
33	974	938	878	852	826
35	1102	1068	1004	988	958
37	1240	1200	1144	1104	1072
39	1377	1330	1258	1248	1212

Tube-Shell Layouts (Tube Counts)

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
1 1/4-in. O.D. Tubes on 1 9/16-in. Square Pitch					
10	16	12	10		
12	30	24	22	16	16
13 1/4	32	30	30	22	22
15 1/4	44	40	37	35	31
17 1/4	56	53	51	48	44
19 1/4	78	73	71	64	56
21 1/4	96	90	86	82	78
23 1/4	127	112	106	102	96
25	140	135	127	123	115
27	166	160	151	146	140
29	193	188	178	174	166
31	226	220	209	202	193
33	258	252	244	238	226
35	293	287	275	268	258
37	334	322	311	304	293
39	370	362	348	342	336
1 1/2-in. O.D. Tubes on 1 7/8-in. Square Pitch					
12	16	16	12	12	
13 1/4	22	22	16	16	
15 1/4	29	29	24	24	22
17 1/4	29	39	34	32	29
19 1/4	50	48	45	43	39
21 1/4	62	60	57	54	50
23 1/4	78	74	70	66	62
25	94	90	86	84	78
27	112	108	102	98	94
29	131	127	120	116	112
31	151	146	141	138	131
33	176	170	164	160	151
35	202	196	188	182	176
37	224	220	217	210	202
39	252	246	237	230	224
1 1/2-in. O.D. Tubes on 1 7/8-in. Triangular Pitch					
12	18	14	14	12	12
13 1/4	27	22	18	16	14
15 1/4	26	34	32	30	27
17 1/4	48	44	42	38	36
19 1/4	61	58	55	51	48
21 1/4	76	78	70	66	61
23 1/4	95	91	86	80	76
25	115	110	105	98	95
27	136	131	125	118	115
29	160	154	147	141	136
31	184	177	172	165	160
33	215	206	200	190	184
35	246	238	230	220	215
37	275	268	260	252	246
39	307	299	290	284	275

Tube-Shell Layouts (Tube Counts)

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
1 1/4-in. O.D. Tubes on 9/16-in. Triangular Pitch					
10					
10	20	18	14		
12 1/4	32	30	26	22	20
13 1/4	38	36	32	28	26
15 1/4	54	51	45	42	38
17 1/4	69	66	62	58	54
19 1/4	95	91	86	78	69
21 1/4	117	112	105	101	95
23 1/4	140	136	130	123	117
25	170	164	155	150	140
27	202	196	185	179	170
29	235	228	217	212	202
31	275	270	255	245	235
33	315	305	297	288	275
35	357	348	335	327	315
37	407	390	380	374	357
39	449	436	425	419	407

From Kern, D. Q. (1950) *Process Heat Transfer*, McGraw-Hill, New York. With permission.

Allocations of streams

- The more seriously fouling fluid flows through the tubes.
- The high pressure fluid flows through the tubes.
- The corrosive fluid must flow through the tubes.
- Fluids producing lower heat transfer coefficients flows on the shell side.

Baffle types and geometry

Baffles function

- Supporting the tubes to prevent vibrations.
- Diverting shell side fluid flow across the bundle to obtain higher convection heat transfer coefficients.

Baffle types

- Transverse baffles (Plate baffles, Rod baffles)
- Longitudinal baffles.

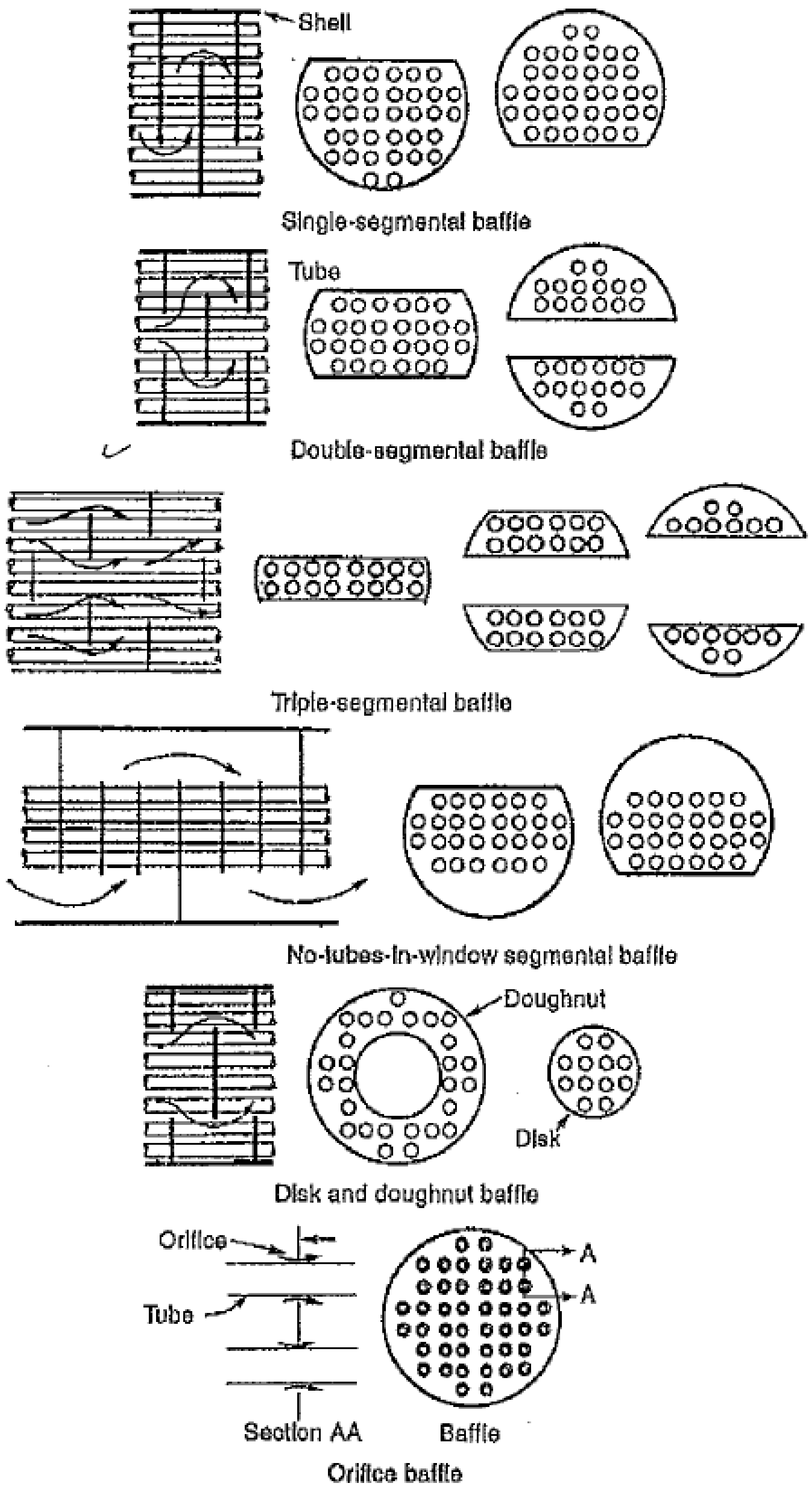
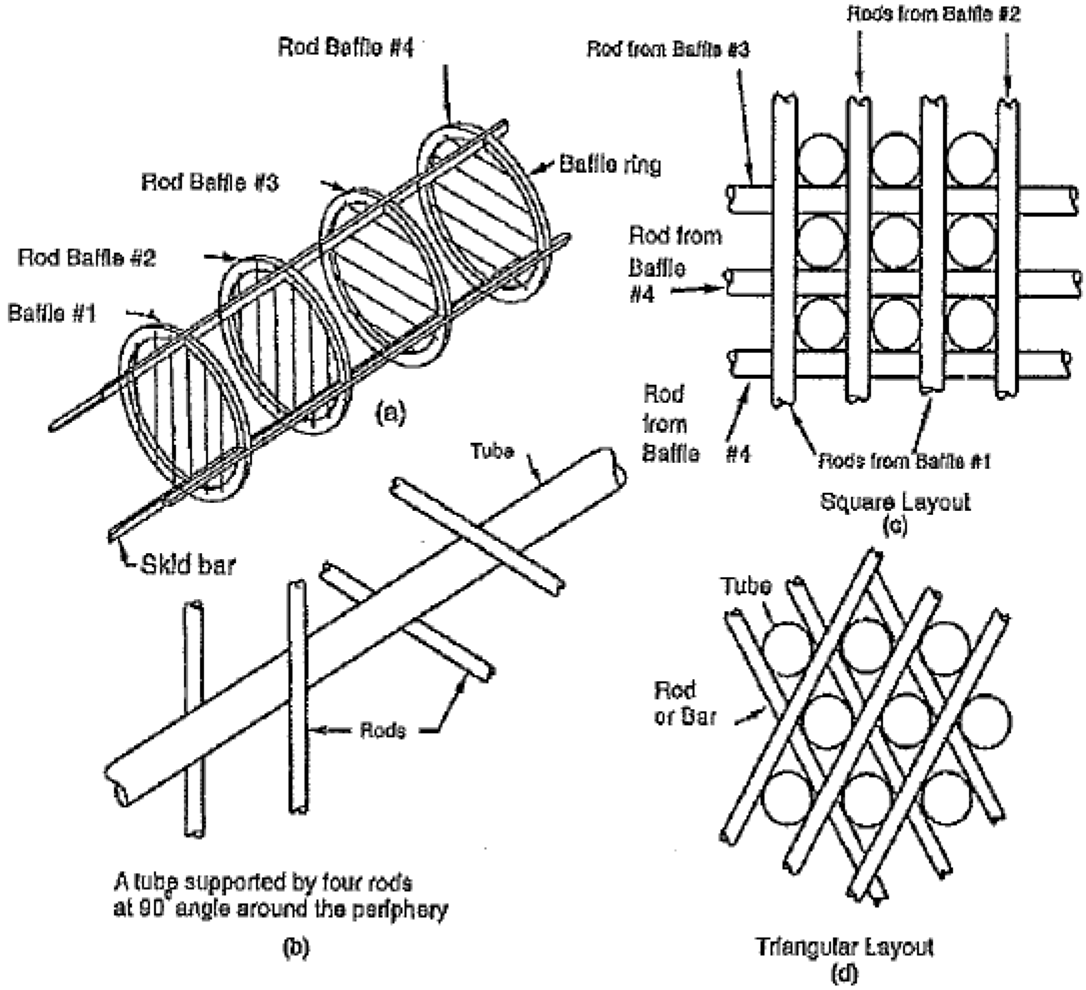
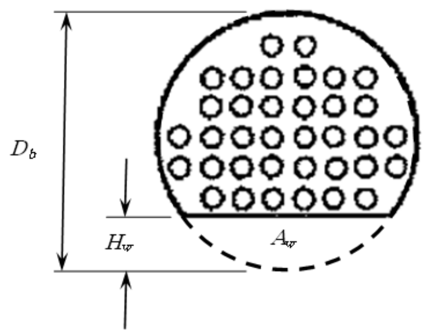


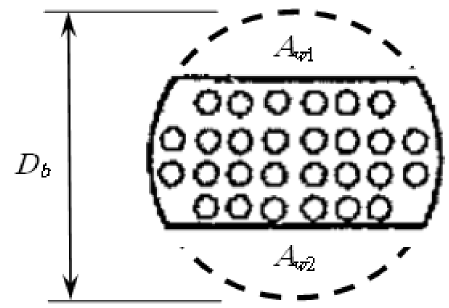
Plate baffle types. (Adapted from Kakaç, S., Bergles, A. E., and Mayinger, F., Eds. [1981] *Heat Exchangers: Thermal-Hydraulic Fundamentals and Design*, Taylor and Francis, Washington, D.C.)



(a) Four rod baffles held by skid bars (no tube shown), (b) a tube supported by four rods, (c) a square layout of tubes with rods, and (d) a triangle layout of tubes with rods. (Adapted from Kakaç, S., Bergles, A. E., and Mayinger, F., Eds. [1981] *Heat Exchangers: Thermal-Hydraulic Fundamentals and Design*, Taylor and Francis, Washington, D.C.)

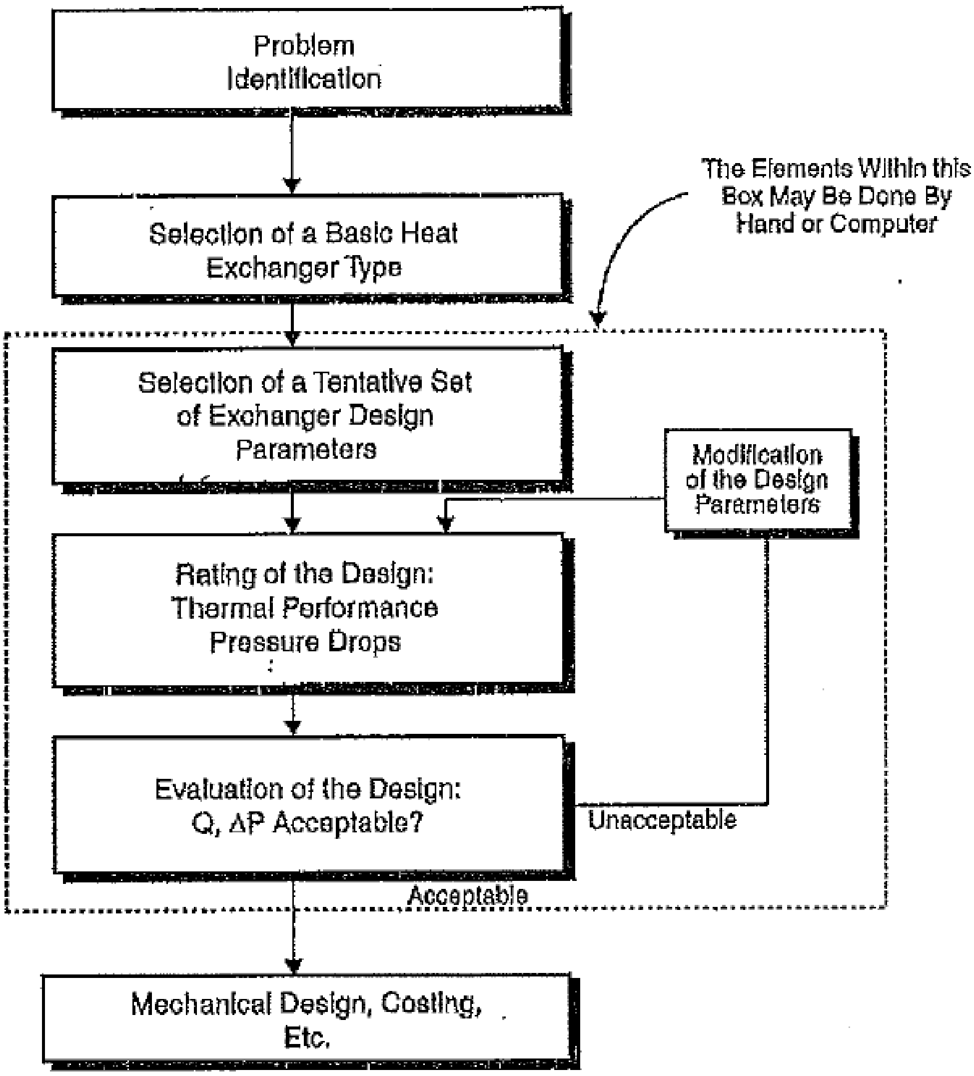
Baffle window (single/others segmental):





B : Baffle spacing	A_W : area of baffle window
Single segmental baffle	$25\% \le Baffle\ cut = \dfrac{H_W}{D_b} \times 100\% \le 30\%$
Other segmental baffle	$Baffle\ cut = \dfrac{A_{W1} + A_{W2}}{\pi D_b^2 / 4} \times 100\%$
For strong structurally tube sheets, less vibration, and augmented heat transfer	$0.4 \le \dfrac{B}{D_i} \le 0.6$

Basic design procedure of a heat exchanger



Basic logic structure for process heat exchanger design. (Based on Bell, K. J. [1981] *Heat Exchangers: Thermal-Hydraulic Fundamentals and Design*, Taylor and Francis, Washington, D.C.)

Typical Film Heat Transfer Coefficients for Shell-and-Tube Heat Exchangers

	Fluid Condition	W/(m² · K)
Sensible Heat Transfer		
Water	Liquid	5,000–7,500
Ammonia	Liquid	6,000–8,000
Light organics	Liquid	1,500–2,000
Medium organics	Liquid	750–1,500
Heavy organics	Liquid	
	Heating	250–750
	Cooling	150–400
Very heavy organics	Liquid	
	Heating	100–300
	Cooling	60–150
Gas	1–2 bar abs	80–125
Gas	10 bar abs	250–400
Gas	100 bar abs	500–800
Condensing Heat Transfer		
Steam, ammonia	No noncondensable	8,000–12,000
Light organics	Pure component, 0.1 bar abs, no noncondensable	2,000–5,000
Light organics	0.1 bar, 4% noncondensable	750–1,000
Medium organics	Pure or narrow condensing range, 1 bar abs	1,500–4,000
Heavy organics	Narrow condensing range, 1 bar abs	600–2,000
Light multicomponent mixture, all condensable	Medium condensing range, 1 bar abs	1,000–2,500
Medium multicomponent mixture, all condensable	Medium condensing range, 1 bar abs	600–1,500
Heavy multicomponent mixture, all condensable	Medium condensing range, 1 bar abs	300–600
Vaporizing Heat Transfer		
Water	Pressure < 5 bar abs, $\Delta T = 25$ K	5,000–10,000
Water	Pressure 5–100 bar abs, $\Delta T = 20$ K	4,000–15,000
Ammonia	Pressure < 30 bar abs, $\Delta T = 20$ K	3,000–5,000
Light organics	Pure component, pressure < 30 bar abs, $\Delta T = 20$ K	2,000–4,000
Light organics	Narrow boiling range, pressure 20–150 bar abs, $\Delta T = 15$–20 K	750–3,000
Medium organics	Narrow boiling range, pressure < 20 bar abs, $\Delta T_{max} = 15$ K	600–2,500
Heavy organics	Narrow boiling range, pressure < 20 bar abs, $\Delta T_{max} = 15$ K	400–1,500

Shell side pressure drop

$$\Delta p_o = \frac{f_o G_o^2 (N_b + 1) D_i}{2 \rho_o D_{e,o} \left(\mu_o / \mu_{o,w}\right)^{0.14}}$$

Square pitch:	$D_{e,o} = \dfrac{4\left(P_T^2 - \pi d_o^2/4\right)}{\pi d_o}$
Triangular pitch:	$D_{e,o} = \dfrac{4\left(P_T^2 \sqrt{3}/4 - \pi d_o^2/8\right)}{\pi d_o/2}$
$A_{C,o} = B D_i \left(1 - \left[\dfrac{d_o}{P_T}\right]\right)$	$G_o = \dfrac{\dot{m}_o}{A_{C,o}}$
$N_b = \dfrac{L}{B} - 1$; N_b : no. of baffles	$f_o = exp\left(0.576 - 0.19\, ln\left[Re_o\right]\right)$ $Re_o = \dfrac{D_{e,o} G_o}{\mu_o}$ $400 < Re_o < 1 \times 10^6$

Tube side pressure drop (bare circular tubes)

$$u_{m,i} = \frac{\dot{m}_i}{\rho_i A_{C,i}}; \quad A_{C,i} = \frac{\pi}{4} d_i^2 \frac{N_t}{N_P}; \quad N_P \text{ : no. of tube passes}$$

$$\Delta p_i = \left(4 f_{a,i} \frac{L}{d_i} + 4\right) \rho_i \frac{u_{m,i}^2}{2} N_P$$

Convection heat transfer coefficient inside tubes (h_i)

- **Developing Laminar flow in a circular tube**
  - **Thermal Entry region**

$$Re_{d_i} = \frac{\rho_i u_{m,i} d_i}{\mu_i}$$

$$Nu_{d_i} = \frac{\bar{h}_i d_i}{k_i} = 3.66 + \frac{0.0668 (d_i/L) Re_{d_i} Pr_i}{1 + 0.04\left[(d_i/L) Re_{d_i} Pr_i\right]^{2/3}}$$

- **Combined Entry region**

$$Nu_{d_i} = \frac{\bar{h}_i d_i}{k_i} = 1.86\left(\frac{Re_{d_i}\, Pr_i\, d_i}{L}\right)^{1/3}\left(\frac{\mu_{i,b}}{\mu_{i,w}}\right)^{0.14}$$

- **<u>Turbulent fully developed flow</u>**

$$Nu_{D_{e,i}} = \frac{\left(f_{a,i}/2\right)\left(Re_{D_{h,i}} - 1000\right)Pr_i}{1 + 12.7\left(f_{a,i}/2\right)^{1/2}\left(Pr_i^{2/3} - 1\right)}$$

$$3\times10^3 \le Re_{D_{h,i}} \le 5\times10^6,\quad 0.5 \le Pr_i \le 2000$$

$$f_{a,i} = \left[1.58\,ln\left(Re_{D_{h,i}}\right) - 3.28\right]^{-2},\quad 3000 \le Re_{D_{h,i}} \le 5\times10^6\,;\ smooth\ tube$$

$$\frac{1}{\sqrt{f_{a,i}}} \cong -3.6\,log_{10}\left[\frac{6.9}{Re_{D_{h,i}}} + \left(\frac{e/D}{3.7}\right)^{1.11}\right],\quad roughened\ tube$$

<u>Shell side heat transfer coefficient using Kern method</u>

$$\frac{h_o D_{e,o}}{k_o} = 0.36\left(\frac{D_{e,o}G_o}{\mu_o}\right)^{0.55} Pr_o^{1/3}\left(\frac{\mu_o}{\mu_{o,w}}\right)^{0.14}$$

$$2\times10^3 < Re_o = \frac{D_{e,o}G_o}{\mu_o} < 1\times10^6$$

<table>
<tr><td>Square pitch:</td><td>$D_{e,o} = \frac{4\left(P_T^2 - \pi d_o^2/4\right)}{\pi d_o}$</td></tr>
<tr><td>Triangular pitch:</td><td>$D_{e,o} = \frac{4\left(P_T^2\sqrt{3}/4 - \pi d_o^2/8\right)}{\pi d_o/2}$</td></tr>
<tr><td>$A_{C,o} = BD_i\left(1 - \left[\frac{d_o}{P_T}\right]\right)$</td><td>$G_o = \frac{\dot{m}_o}{A_{C,o}}$</td></tr>
<tr><td colspan="2">$Baffle\ cut = 25\%$</td></tr>
</table>

Thermal analysis of shell-and-tube heat exchangers

$$Q=\dot{m}_c c_{pc}\left(T_{c2}-T_{c1}\right)=\dot{m}_h c_{ph}\left(T_{h1}-T_{h2}\right)=U_o A_o F\,\frac{\left(T_{h2}-T_{c1}\right)-\left(T_{h1}-T_{c2}\right)}{ln\left(\dfrac{T_{h2}-T_{c1}}{T_{h1}-T_{c2}}\right)}$$

- **Bare circular tubes**

$$\frac{1}{U_o}=\frac{1}{h_o}+R_{fo}+\frac{d_o\,ln\left(d_o/d_i\right)}{2k_{tube}}+\left(\frac{d_o}{d_i}\right)R_{fi}+\left(\frac{d_o}{d_i}\right)\frac{1}{h_i}$$

$$A_o=\pi d_o N_t L$$

- **Finned circular tubes with fins on inner and outer surfaces (rectangular fins)**

$$\frac{1}{U_o}=\frac{1}{\eta_o h_o}+\frac{R_{fo}}{\eta_o}+\frac{\left(A_o\right)_{L=1\,m}\,ln\left(d_o/d_i\right)}{2\pi k_{tube}N_t}+\left[\frac{\left(A_o\right)_{L=1\,m}}{\left(A_i\right)_{L=1\,m}}\right]\frac{R_{fi}}{\eta_i}+\left[\frac{\left(A_o\right)_{L=1\,m}}{\left(A_i\right)_{L=1\,m}}\right]\frac{1}{\eta_i h_i}$$

$$\eta_o=1-\left(1-\eta_{fo}\right)\frac{\left(A_{fo}\right)_{L=1\,m}}{\left(A_o\right)_{L=1\,m}};\quad \eta_i=1-\left(1-\eta_{fi}\right)\frac{\left(A_{fi}\right)_{L=1\,m}}{\left(A_i\right)_{L=1\,m}}$$

$$\eta_{fo}=\frac{tanh\left(\left[H_{fo}+\delta_o/2\right]\sqrt{\dfrac{2h_o}{k_{fin,o}\delta_o}}\right)}{\left[H_{fo}+\delta_o/2\right]\sqrt{\dfrac{2h_o}{k_{fin,o}\delta_o}}};\quad \eta_{fi}=\frac{tanh\left(\left[H_{fi}+\delta_i/2\right]\sqrt{\dfrac{2h_i}{k_{fin,i}\delta_i}}\right)}{\left[H_{fi}+\delta_i/2\right]\sqrt{\dfrac{2h_i}{k_{fin,i}\delta_i}}}$$

$$A_{u,i}=N_t L\left(\pi d_i-N_{f,i}\delta_i\right);\quad A_{f,i}=N_t N_{f,i}L\left(2H_{f,i}+\delta_i\right)$$

$$A_i=N_t L\left(\pi d_i+2N_{f,i}H_{f,i}\right)$$

$$A_{u,o}=N_t L\left(\pi d_o-N_{f,o}\delta_o\right);\quad A_{f,o}=N_t N_{f,o}L\left(2H_{f,o}+\delta_o\right)$$

$$A_o=N_t L\left(\pi d_o+2N_{f,o}H_{f,o}\right)$$

Problems

8.1. Crude oil at a flow rate of 63.77 kg/s enters the exchanger at 102°C and leaves at 65°C. The heat will be transferred to 45 kg/s of tube water (city water) coming from a supply at 21°C. The exchanger data is given below: 3/4" OD, 18 BWG tubes on a 1 inch square pitch; 2 tube passes and 4 tube passes will be considered. Tube material is carbon steel. The heat exchanger has one shell. Two different shell diameters of ID 35 and 37 inches should be studied. Baffle spacing is 275 mm. Calculate the length of the heat exchanger for clean and fouled surfaces. Also calculate:

a. Tube side velocity for 1–2 and 1–4 arrangements

b. Overall heat transfer coefficients for clean and fouled surfaces

c. Pressure drops

d. Pumping powers

The allowable shell-side and tube-side pressure drops are 60 kPa and 45 kPa, respectively. The following properties are given:

	Shell Side	Tube Side
Specific heat, J/kg · K	2177	4186.8
Dynamic viscosity, N · s/m²	0.00189	0.00072
Thermal conductivity, W/m · K	0.122	0.605
Density, kg/m³	786.4	995
Prandtl number	33.73	6.29
Maximum pressure loss, Pa	60,000	45,000

8.4. A heat exchanger is available to heat raw water by the use of condensed water at 67°C which flows in the shell side with a mass flow rate of 50,000 kg/h. Shell-side dimensions are: $ID_s = D_s =$ 19.25 in., $P_T = 1.25$ in. (square), and baffle spacing = 0.3 m. The raw water enters the tubes at 17°C with a mass flow rate of 30,000 kg/h. Tube dimensions are: d_o = 1 in. = 0.0254 m (18 BWG tubes), d_i = 0.902 in. The length of the heat exchanger is 6 m with two passes. The permissible maximum pressure drop on the shell side is 1.5 psi. Water outlet temperature should not be less than 40°C. Calculate:

a. Outlet temperatures

b. Heat load of the heat exchanger

Is the heat exchanger appropriate for this purpose?

Design Project 8.1

Design of an Oil Cooler for Marine Applications

The following specifications are given:

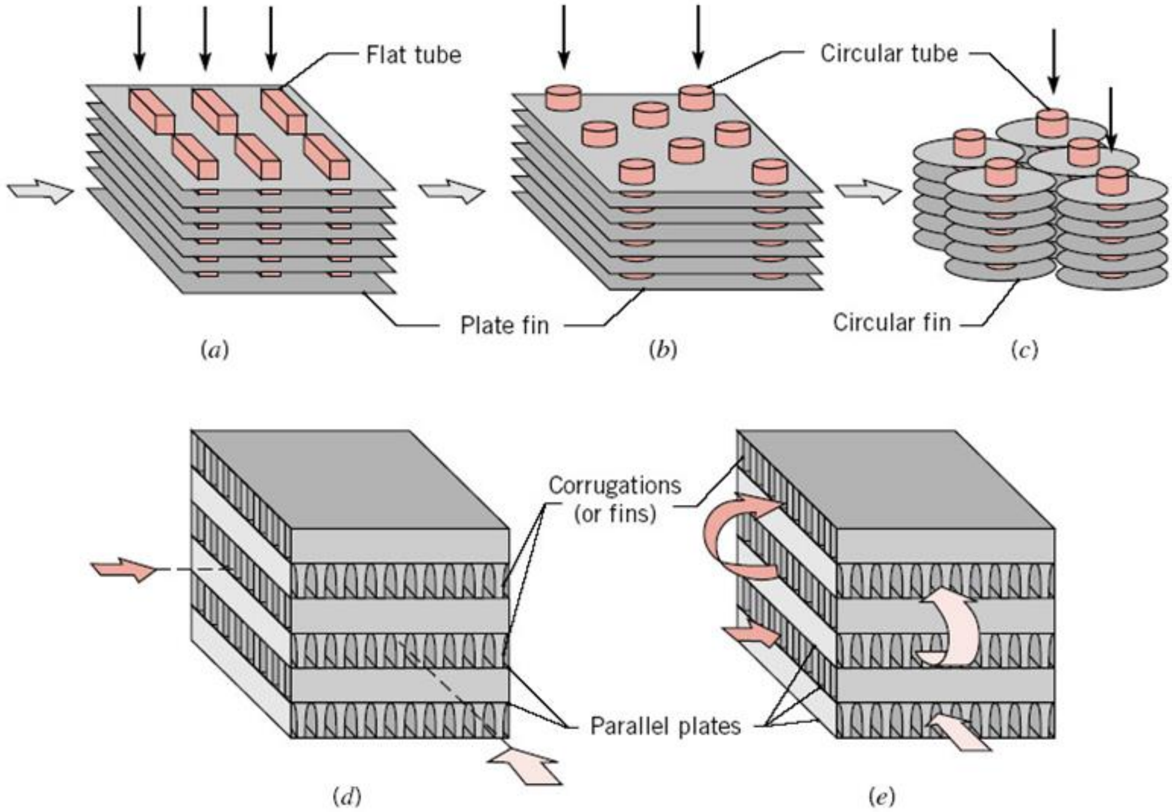
Fluid	SAE-30 Oil	Sea Water
Inlet temperature, °C	65	20
Outlet temperature, °C	56	32
Pressure drop limit (kPa)	140	40
Total mass flow rate (kg/s)	20	—

A shell-and-tube heat exchanger type with geometrical parameters can be selected. The heat exchanger must be designed and rated. Different configurations of shell-and-tube types can be tested. A parametrical study is expected to develop with a suitable final design; mechanical design will be performed and the cost will be estimated;

Compact Heat Exchangers (Compact HXs)

Compact HXs is used for gas flow applications.

Compact HX types are: a) Plate-fin type, and b) Tube-fin type.



Compact heat exchanger cores. (*a*) Fin–tube (flat tubes, continuous plate fins). (*b*) Fin–tube (circular tubes, continuous plate fins). (*c*) Fin–tube (circular tubes, circular fins). (*d*) Plate–fin (single pass). (*e*) Plate–fin (multipass).

Compact HX has heat transfer surface area per unit volume larger than 700 m^2/m^3.

$$A_t/V \geq 700\ m^2/m^3$$

Microheat exchanger has heat transfer surface area per unit volume larger than 10000 m^2/m^3.

$$A_t/V \geq 10{,}000\ m^2/m^3$$

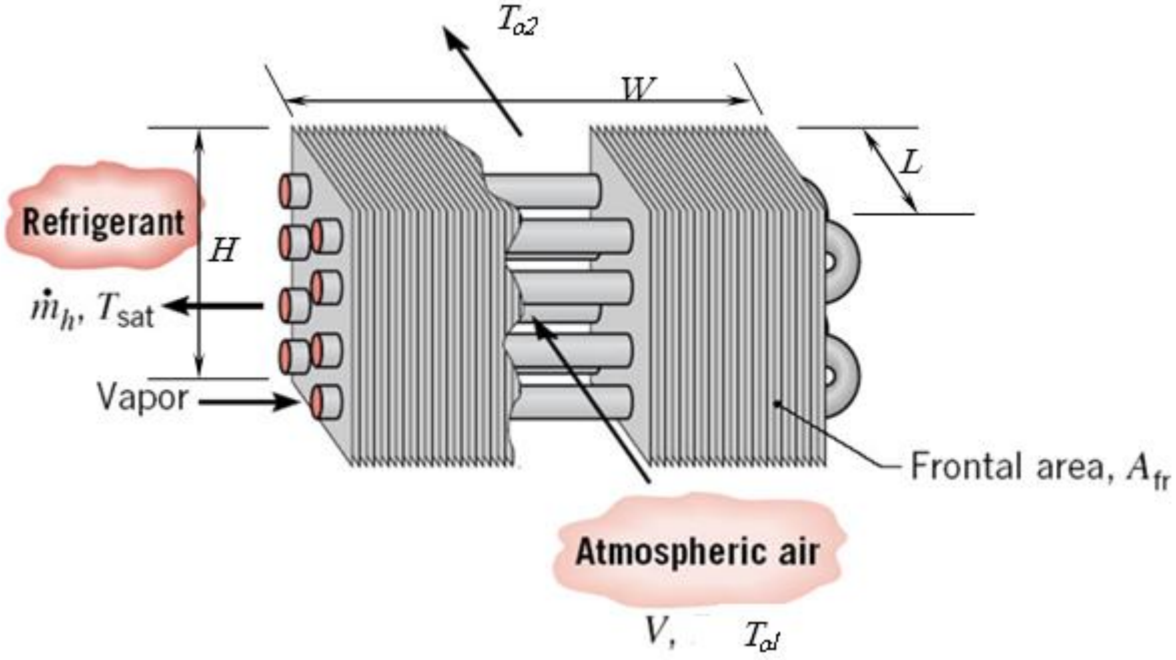
Compact HXs are used for gas to gas or liquid to gas heat exchangers.

Compact HXs applications: a) air conditioning condensers and evaporators, b) automotive radiators, c) intercoolers of compressors.

Heat transfer enhancement techniques

- **Active techniques** (requiring external power to enhance heat transfer), e.g. surface vibration, acoustic techniques, using electrohydrodynamic or hydromagetic effects.
- **Passive techniques** (do not require external power to enhance heat transfer), e.g. using fins, twist tapes to gererate turbulence, surface roughness.

Heat transfer in compact heat exchangers



$$Stanton\ number\ (St): \quad St_o = \frac{h_o}{G_o c_{p,o}}$$

$$Pr_o = \frac{\mu_o c_{p,o}}{k_o}; \quad Re_o = \frac{G_o D_{h,o}}{\mu_o}$$

$$G_o = \rho_o U_{max,o} = \frac{\dot{m}_o}{A_{min,o}}$$

$$A_{min,o} : minimum\ external\ flow\ area$$

$$A_t \equiv A_o : external\ heat\ transfer\ area$$

$$A_{fr,o} : external\ flow\ frontal\ area$$

$$A_{fr,o} = HW$$

$$D_{ho} = 4L\frac{A_{min}}{A_t}$$

$$j_{H,o} = St_o\,Pr_o^{2/3} = \frac{h_o}{G_o c_{p,o}}Pr_o^{2/3}$$

Important relationships

$$\beta = \frac{A_t}{V};\quad \sigma = \frac{A_{min,o}}{A_{fr,o}};\quad V = HWL$$

$$\frac{A_t}{A_{min,o}} = \frac{\beta}{\sigma}L;\quad D_{h,o} = \frac{4\sigma}{\beta}$$

$$G_o = \frac{\dot{m}_o}{A_{min,o}} = \frac{\dot{m}_o}{\sigma A_{fr,o}} = \frac{\rho_{o,1}U_{\infty,o}}{\sigma}$$

$$Re_{h,o} = \frac{G_o D_{h,o}}{\mu_o} = \frac{4\dot{m}_o}{\mu_o \beta A_{fr,o}} = \frac{4\rho_{o,1}U_{\infty,o}}{\mu_o \beta}$$

Overall heat transfer coefficient

$$\delta_o :\ fin\ thickness\ [cm]$$

$$b_o :\ fin\ pitch\ [cm^{-1}]$$

$$\alpha_o = \frac{A_{f,o}}{A_t}$$

$$\eta_o = 1 - (1 - \eta_{fo})\alpha_o$$

$$\delta_{tube} : tube\ thickness$$

$$P_i : tube\ inner\ perimeter$$

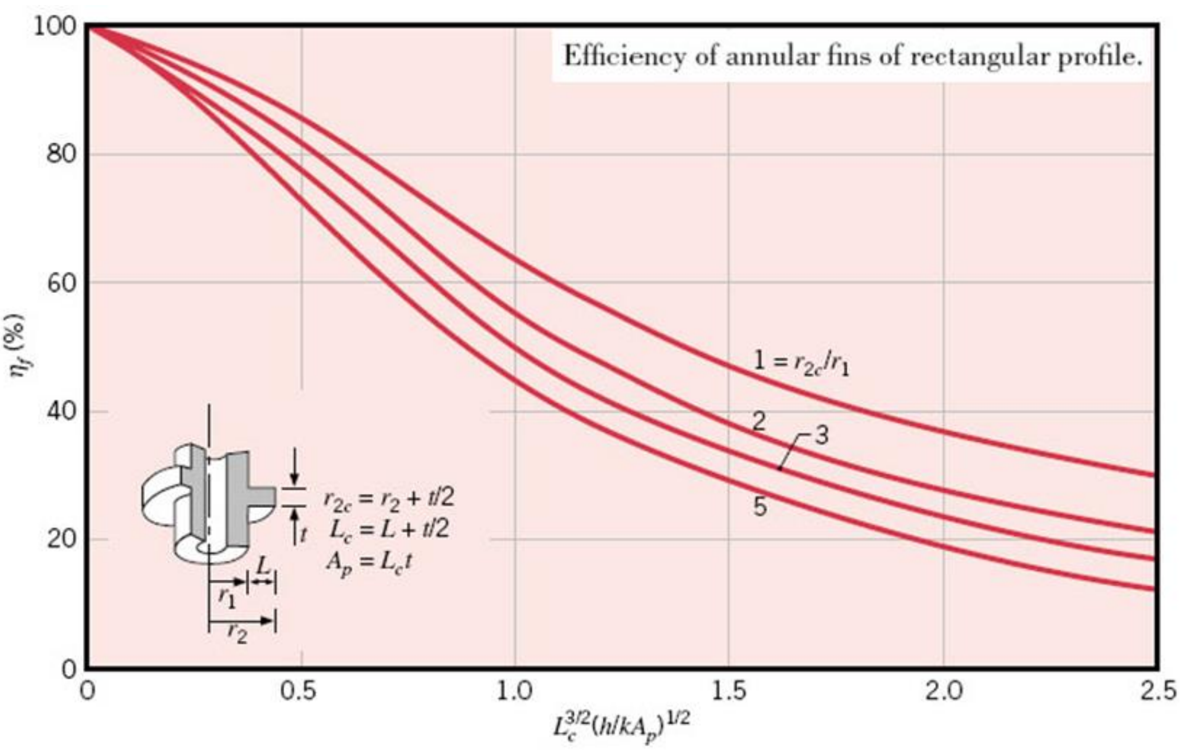
$$P_o : tube\ outer\ perimeter$$

- **Circular** tube with **fins on outer surface and no-fins on inner surface**

$$\frac{1}{U_o} = \left(\frac{d_o}{d_i}\right)\left(\frac{1-\delta_o b_o}{1-\alpha_o}\right)\frac{1}{h_i} + d_o\left(\frac{1-\delta_o b_o}{1-\alpha_o}\right)\frac{\ln(d_o/d_i)}{2k_{tube}} + \frac{1}{\eta_o h_o}$$

- **Rectangular** tube with **fins only on outer surface**

$$\frac{1}{U_o} = \left(\frac{P_o}{P_i}\right)\left(\frac{1-\delta_o b_o}{1-\alpha_o}\right)\frac{1}{h_i} + \left(\frac{P_o}{P_i}\right)\left(\frac{1-\delta_o b_o}{1-\alpha_o}\right)\frac{\delta_{tube}}{k_{tube}} + \frac{1}{\eta_o h_o}$$



Thermal analysis of compact heat exchangers

$$Q = \dot{m}_c c_{pc}(T_{c2} - T_{c1}) = \dot{m}_h c_{ph}(T_{h1} - T_{h2}) = U_o A_t F \frac{(T_{h2} - T_{c1}) - (T_{h1} - T_{c2})}{ln\left(\dfrac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}}\right)}$$

Gas-side pressure drop for compact heat exchangers

$$\frac{1}{\rho_o} = \frac{1}{2}\left[\frac{1}{\rho_{o,1}} + \frac{1}{\rho_{o,2}}\right]$$

- **Tube-fin compact heat exchangers**

$$\Delta p_o = \frac{G_o^2}{2\rho_{o,1}}\left[f_o \frac{A_t}{A_{min,o}} \frac{\rho_{o,1}}{\rho_o} + (1 + \sigma^2)\left(\frac{\rho_{o,1}}{\rho_{o,2}} - 1\right)\right]$$

- **Plate-fin compact heat exchangers**

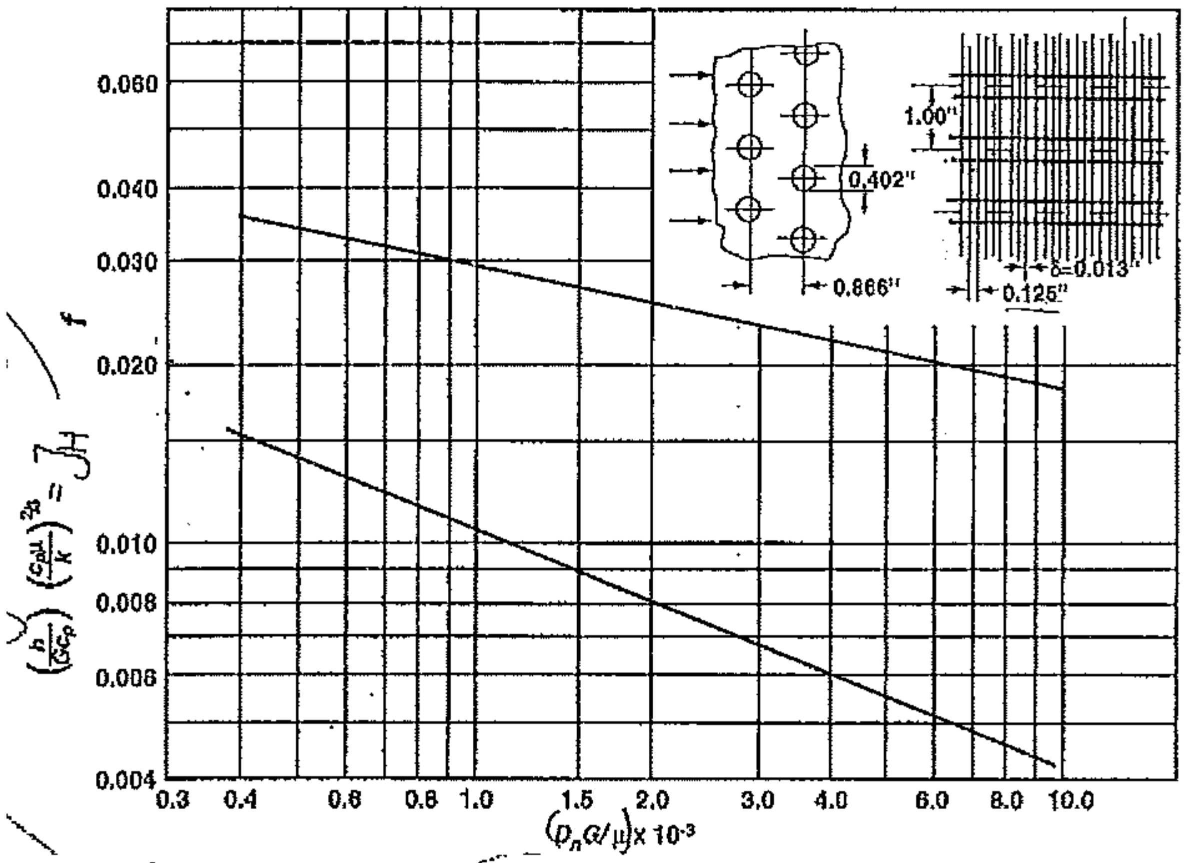
$$\Delta p_o = \frac{G_o^2}{2\rho_{o,1}}\left[(k_c + 1 - \sigma^2) + 2\left(\frac{\rho_{o1}}{\rho_{o2}} - 1\right) + f_o \frac{A_t}{A_{min,o}} \frac{\rho_{o,1}}{\rho_o} - (1 - k_e - \sigma^2)\left(\frac{\rho_{o,1}}{\rho_{o,2}}\right)\right]$$

$$k_c : inlet\ contraction\ loss\ coefficient$$

$$k_e : oulet\ expansion\ loss\ coefficient$$

Gas-side convection coefficient, h_o, using different charts

- The h-coefficient shown in the subsequent figures is the h_o-coefficient.
- The f-factor shown in the subsequent figures is the f_o-factor.



Heat transfer and friction factor for a circular tube continuous fin heat exchanger. Surface 8.0-3/8 T: tube O.D. = 1.02 cm; fin pitch = 3.15/cm; fin thickness = 0.033 cm; fin area/total area = 0.839; air-passage hydraulic diameter = 0.3633 cm; free-flow area/frontal area, σ = 0.534; heat transfer area/total volume = 587 m²/m³. (From Kays, W. M. and London, A. L. [1984], *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York. With permission.)

$$N_L : no.\ of\ rows$$

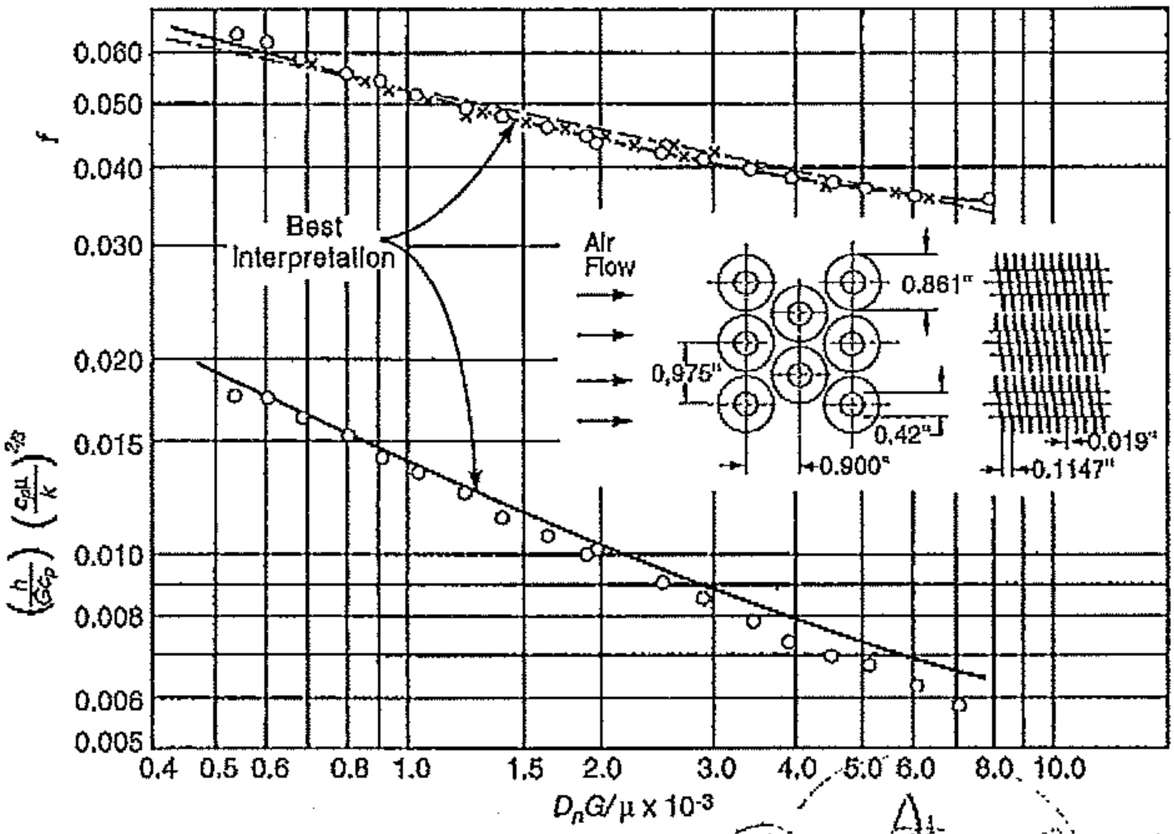
$$N_T : maximum\ no.\ of\ tubes\ per\ row$$

$$P_L = 0.866\ inch$$

$$P_T = 1.00\ inch$$

$$L = (N_L - 1)P_L + P_L$$

$$H = (N_T - 1)P_T + P_T$$



Heat transfer and friction factor for flow across circular finned-tube matrix. Surface CF-8.72(c): tube OD = 1.07 cm; fin pitch = 3.43/cm; fin thickness = 0.048 cm; fin area/total area = 0.876; air-passage hydraulic diameter, d_h = 0.443 cm; free-flow area/frontal area, σ = 0.494; heat transfer area/total volume = 446 m²/m³. (From Kays, W. M. and London, A. L. [1984], *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York. With permission.)

$$N_L : no.\ of\ rows$$

$$N_T : maximum\ no.\ of\ tubes\ per\ row$$

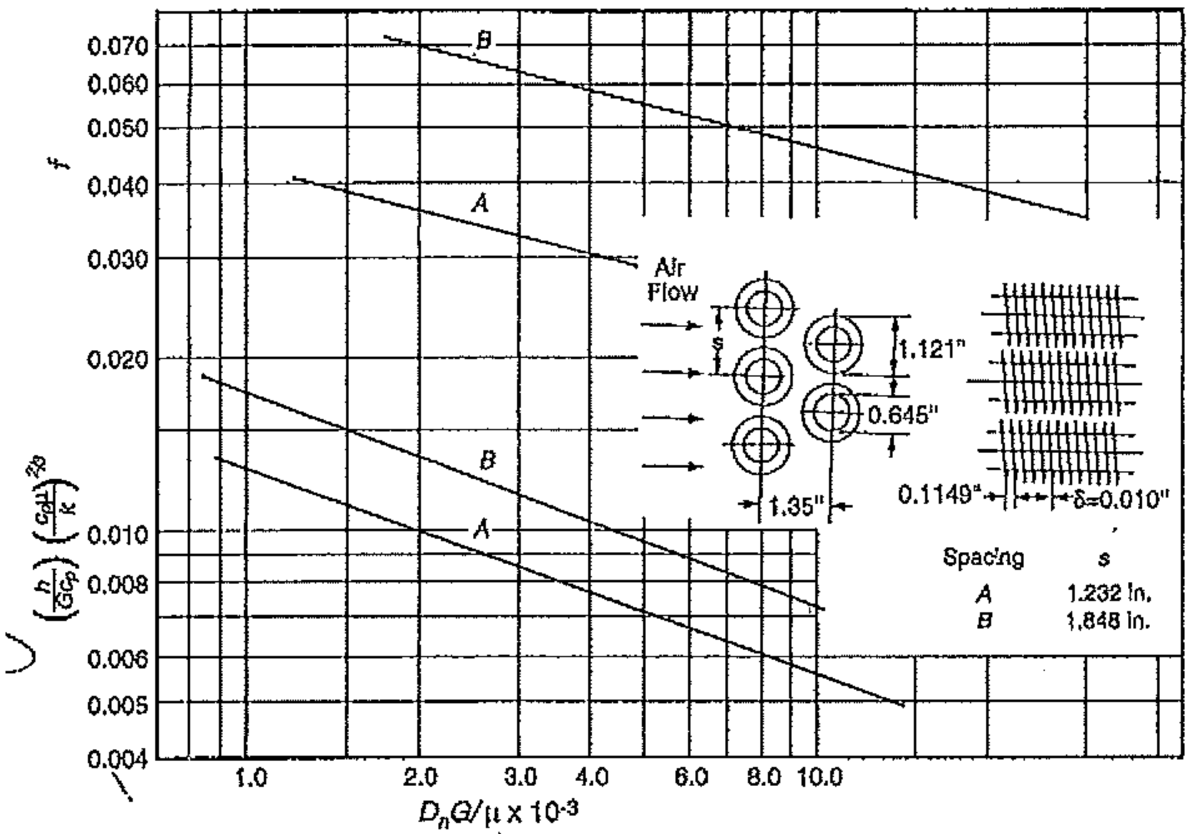
$$D_f = 0.861\, inch$$

$$P_L = 0.900\, inch$$

$$P_T = 0.975\, inch$$

$$L = (N_L - 1)P_L + D_f$$

$$H = (N_T - 1)P_T + D_f$$



Heat transfer and friction factor for flow across finned-tube matrix. Surface CF-8.7-5/8 J: tube OD = 1.638 cm; fin pitch = 3.43/cm; fin thickness = 0.0254 cm; fin area/total area = 0.862; air-passage hydraulic diameter, D_h = 0.5477 cm (A), 1.1673 cm (B); free-flow area/frontal area, σ = 0.443 (A), 0.628 (B); heat transfer area/total volume = 323.8 m²/m³ (A), 215.6 m²/m³ (B). (From Kays, W. M. and London, A. L. [1984], *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York. With permission.)

$$N_L : no.\ of\ rows$$

$$N_T : maximum\ no.\ of\ tubes\ per\ row$$

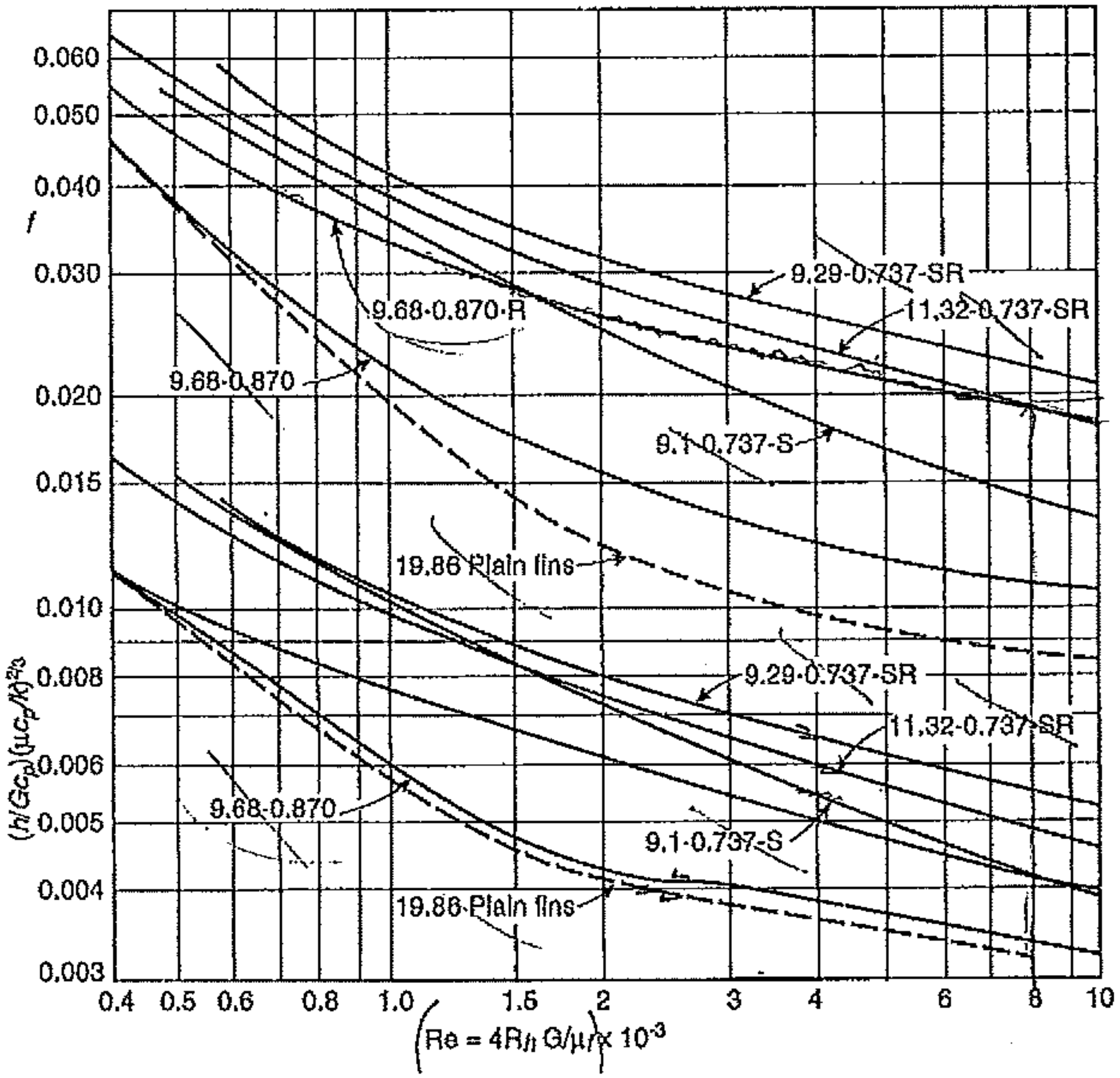
$$D_f = 1.121\ inch$$

$$P_L = 1.35\ inch$$

$$P_{T,A} = 1.232\ inch; \quad P_{T,B} = 1.848\ inch$$

$$L = (N_L - 1)P_L + D_f$$

$$H = (N_T - 1)P_T + D_f$$

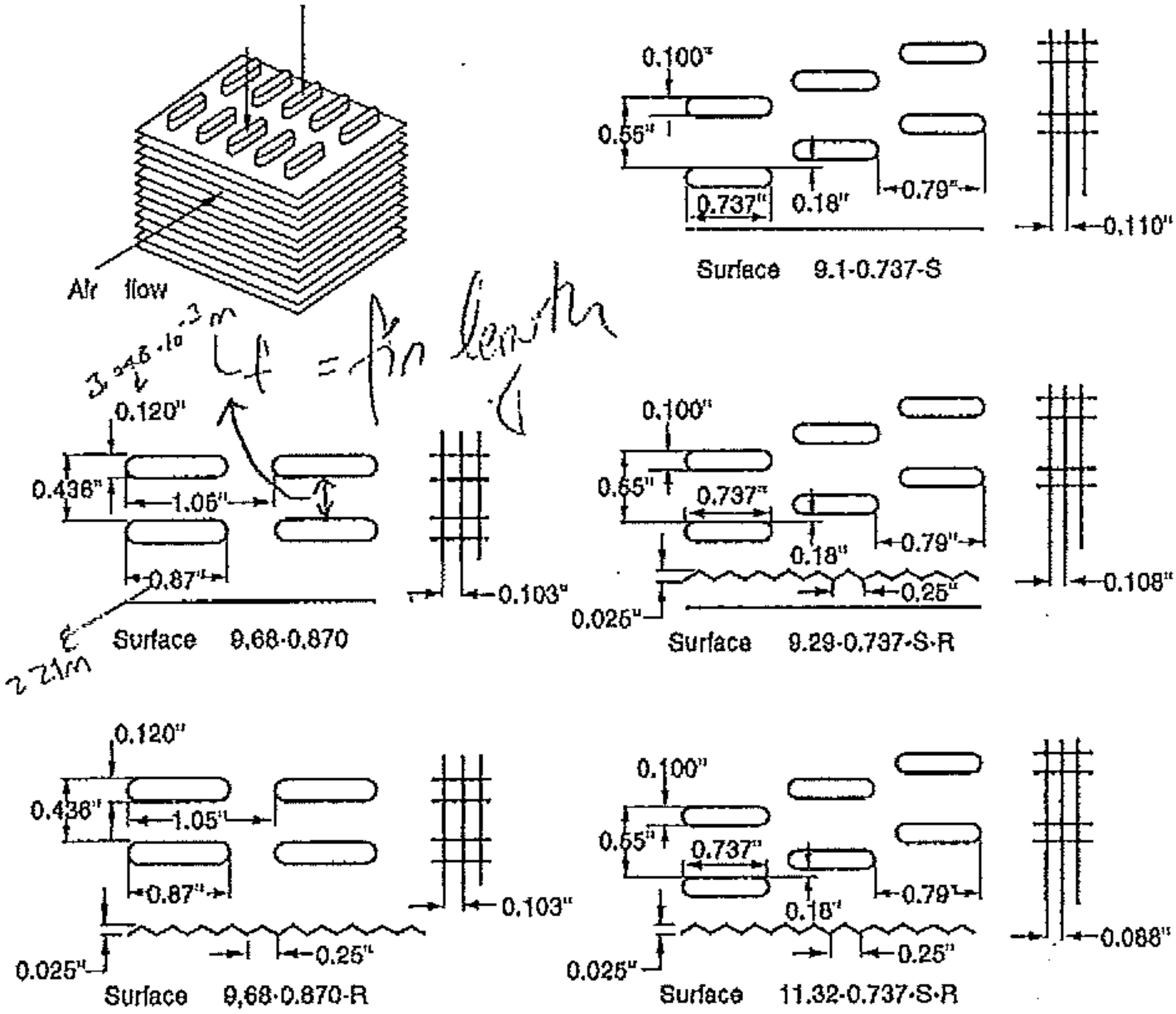


Heat transfer and friction factor for flow across finned flat-tube matrix for the surfaces shown in Figure 9.8 and Table 9.1. (From Kays, W. M. and London, A. L. [1984], *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York. With permission.)

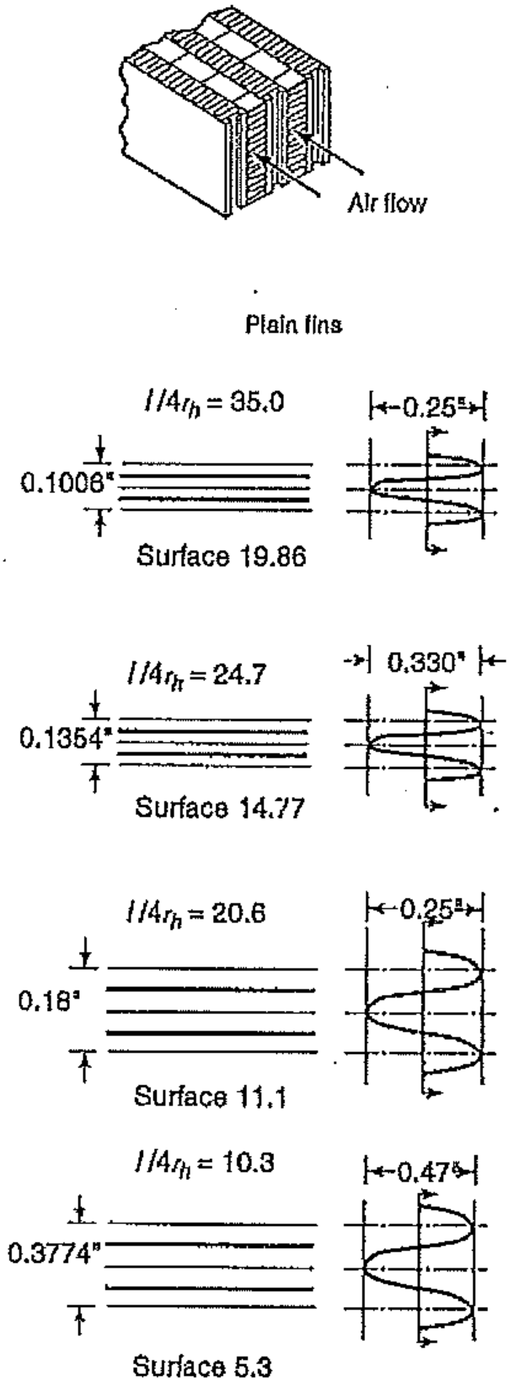
Heat Transfer Matrix Geometries for Plate Plain-Fin and Fin Flat-Tube Types for Which Test Data are Presented in Figures 9.7 and 9.9

Surface Designation	Fins (per cm)	Hydraulic Diameter (D_h, cm)	Plate Spacing (b, cm)	Tube or Fin Thickness (cm)	Extended Total Area	Area Volume Between Plates (β, m²/m³)	Area Core Volume (β, m²/m³)	Free-Flow Frontal Area (σ)
Plate Plain-Fin Type								
5.30	13.46	0.051	1.194	0.0152	0.719	511.8		
11.10	28.19	0.257	0.635	0.0152	0.730	1095.8		
14.77	37.52	0.215	0.838	0.0152	0.831	1210.6		
19.86	50.44	0.152	0.635	0.0152	0.833	1493.0		
Fin Flat-Tube Type								
9.68-0.870	24.587	0.2997		0.0102	0.795		751.3	0.697
9.68-0.870-R	24.587	0.2997		0.0102	0.795		751.3	0.697
9.1-0.737-S	23.114	0.3565		0.0102	0.813		734.9	0.788
9.29-0.737-S-R	28.753	0.3510		0.0102	0.845		885.8	0.788
11.32-0.737-S-R	23.596	0.3434		0.0102	0.814		748.0	0.780

From Kays, W. M. and London, A. L. [1984], *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York. With permission.



Various flattened tube-plate-fin compact surfaces for which test data are presented in Figure 9.7. (From Kays, W. M. and London, A. L. [1984], *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York. With permission.)



Heat transfer and friction factor for four plain plate-fin heat transfer matrixes of Table 9.1. (From Kays, W. M. and London, A. L. [1984], *Compact Heat Exchangers*, 3rd ed., McGraw-Hill, New York. With permission.)

Problems

9.1. Air at 1 atm and 400 K and with a velocity of 10 m/s flows across the compact heat exchanger shown in Figure 9.6a and exits with a mean temperature of 300 K. The core is 0.6 m long. Calculate the total frictional pressure drop between the air inlet and outlet and the average heat transfer coefficient on the air side.

9.3. Hot air at 2 atm and 500 K at a rate of 8 kg/s flows across a circular finned-tube matrix configuration shown in Figure 9.6. The frontal area of the heat exchanger is 0.8 m × 0.5 m and the core is 0.5 m long. Geometrical configurations are shown in Figure 9.6. Calculate:

a. The heat transfer coefficient

b. The total frictional pressure drop between the air inlet and outlet

9.13. Design a heat exchanger asking the surface given in Figure 9.8 (surface 9.29-0.737-S-R). Fins are continuous aluminum. The geometrical data for the air side are given in Table 9.1. On the water side, the flatted tube is 0.2 cm × 1.6 cm. The inside diameter of the tube before it was flattened was 1.23 cm, with a wall thickness of 0.025 cm. Water velócity inside is 1.5 m/s. The design should specify the core size and the core pressure drop.

Design Project 9.1

Design of the Cooling System and Radiator of a Truck

Some of the design specifications as applicable to a typical truck are given as:

Heat load:	100 kW
Water inlet temperature:	80°C
Water outlet temperature:	70°C
Air inlet temperature:	35°C
Air outlet temperature:	46°C

Air pressure:	100 kPa
Air pressure drop:	0.3 kPa
Core matrix of the radiator:	to be selected

Different compact surfaces must be studied and compared for the thermal and hydraulic analysis of the radiator. A parametric study is expected to develop an acceptable final design which includes the selection of a pump, materials selection, mechanical design, drawings, and cost estimation.

Boiling

- Boiling is associated with transformation of liquid to vapor at a solid/liquid interface due to convection heat transfer from the solid.
- Agitation of the liquid by vapor bubbles is one mechanism that provides for large convection coefficients and hence large heat fluxes at low-to-moderate surface-to-fluid temperature differences.
- Special form of Newton's law of cooling:

$$q_s'' = h\left(T_s - T_{sat}\right) = h\Delta T_e$$

$$T_{sat} : \textit{saturation temperature of liquid}$$

$$\Delta T_e \equiv \left(T_s - T_{sat}\right) : \textit{excess temperature}$$

Special cases

- **Pool boiling**
  Liquid motion is due to natural convection and bubble-induced mixing.

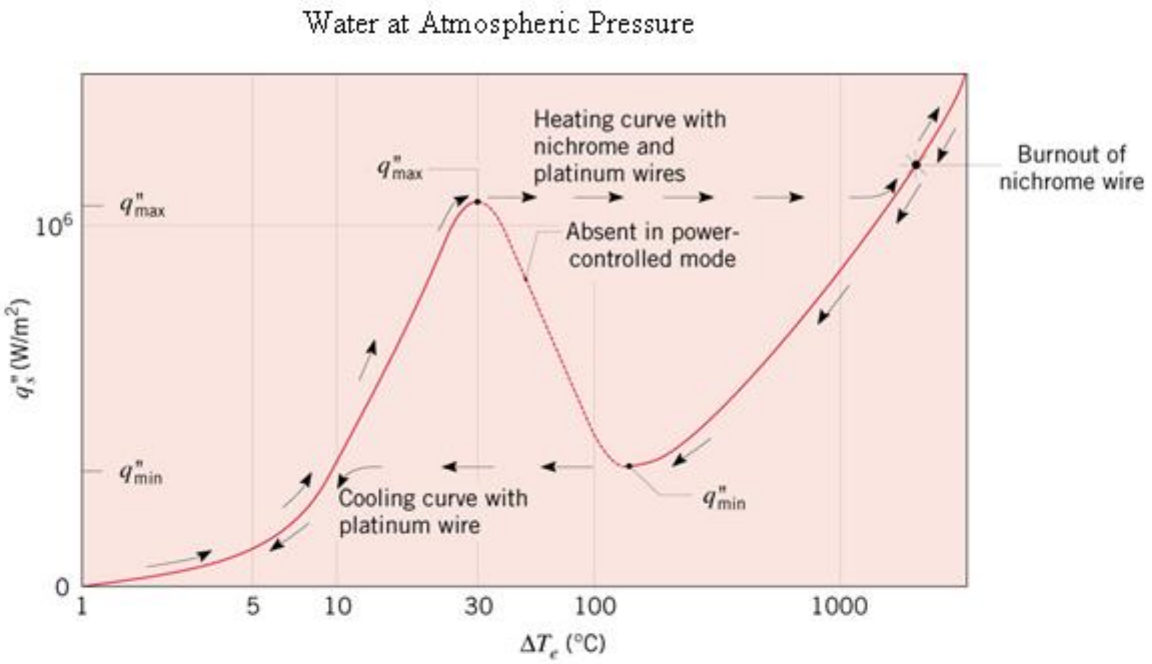
- **Forced convection boiling**
  Fluid motion is induced by external means, as well as by bubble-induced mixing.

- **Saturated boiling**
  Liquid temperature is slightly higher than saturation temperature.

- **Subcooled boiling**
  Liquid temperature is less than saturation temperature.

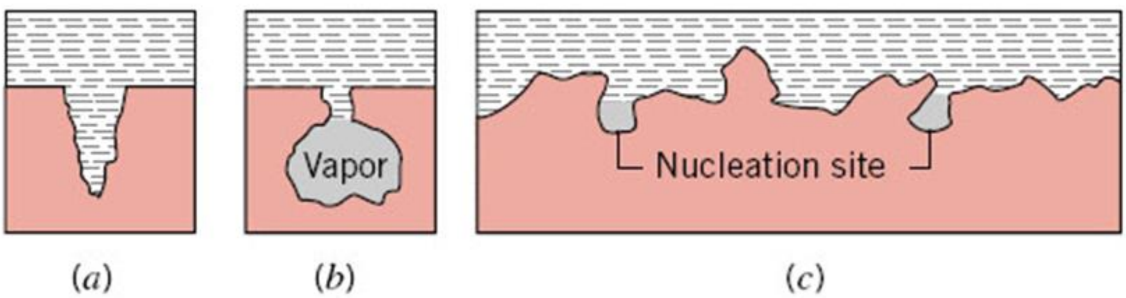
The boiling curve

Reveals range of conditions associated with saturated pool boiling on $q_s'' - \Delta T_e$ plot.

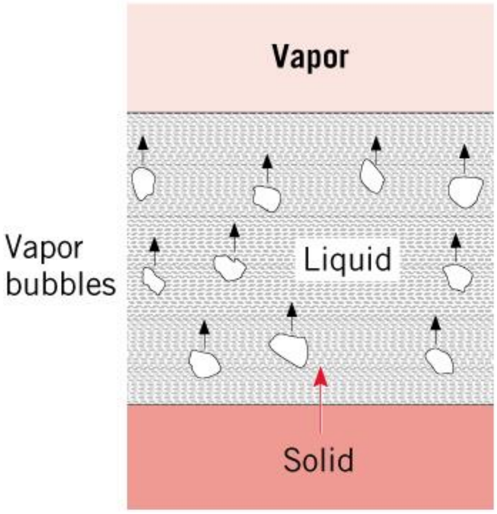


- **Free convection boiling** $(\Delta T_e < 5\ ^\circ C, water\ @1atm)$
  - Little vapor formation
  - Liquid motion is due principally to single-phase natural convection.
- **Onset of Nucleate Boiling - ONB** $(\Delta T_e \approx 5\ ^\circ C, water\ @1atm)$
- **Nucleate boiling** $(5\ ^\circ C < \Delta T_e < 30\ ^\circ C, water\ @1atm)$

  Isolated vapor bubbles $(5\ ^\circ C < \Delta T_e < 10\ ^\circ C, water\ @1atm)$
  - Liquid motion is strongly influenced by nucleation of bubbles at the surface.
  - h and q_s'' increase sharply with increasing ΔT_e.
  - Heat transfer is principally due to contact of liquid with the surface (single-phase convection) and not to vaporization.

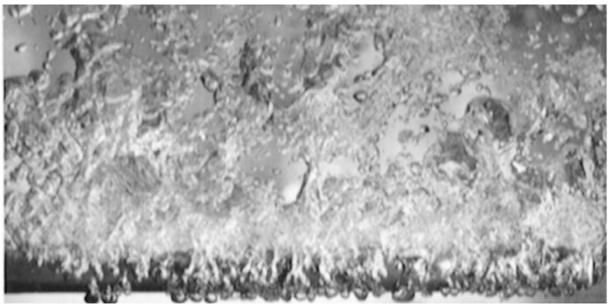


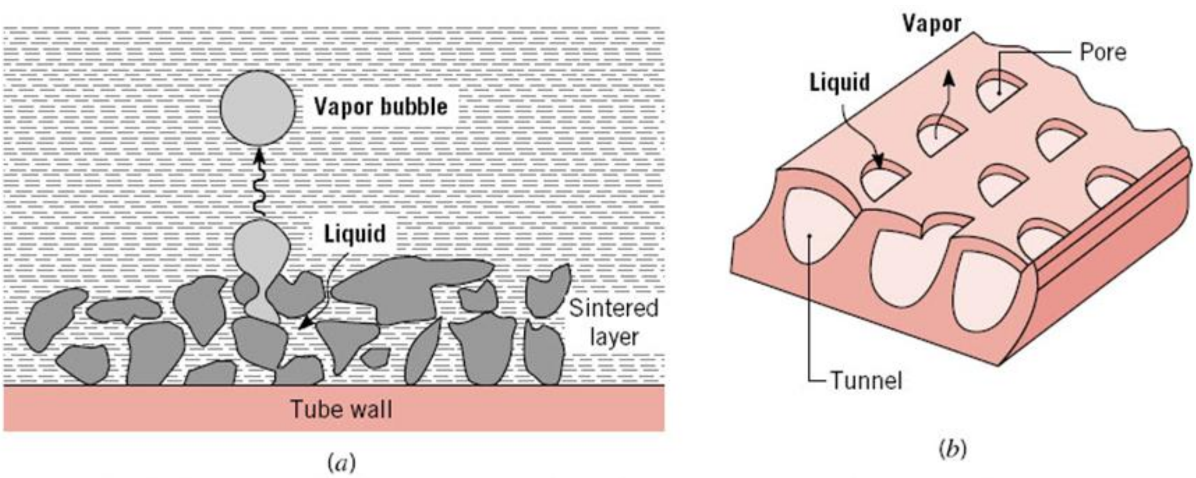
Formation of nucleation sites. (*a*) Wetted cavity with no trapped vapor. (*b*) Reentrant cavity with trapped vapor. (*c*) Enlarged profile of a roughened surface.



Jets and columns $\left(10\ ^{\circ}C < \Delta T_e < 30\ ^{\circ}C, water\ @1atm\right)$

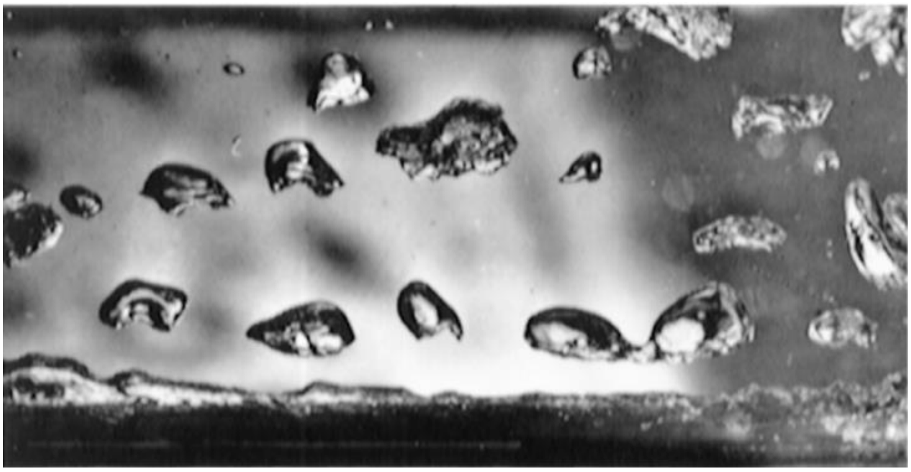
- Increasing number of nucleation sites causes bubble interactions and coalescence into jets and slugs.
- Liquid/surface contact is impaired.
- q''_s continues to increase with ΔT_e but h begins to decrease.





Typical structured enhancement surfaces for augmentation of nucleate boiling. (*a*) Sintered metallic coating. (*b*) Mechanically formed double-reentrant cavity.

- **Critical Heat Flux - CHF,** $q''_{max}\ \left(\Delta T_e \cong 30\ ^\circ C, water\ @1atm\right)$
  - Maximum attainable heat flux in nucleate boiling.
  - $q''_{max} \cong 1\ MW/m^2$ for water at atmospheric pressure.
- **Potential burnout for power - Controlled heating**
  - An increase in beyond q''_{max} causes the surface to be blanketed by vapor, and the surface temperature can spontaneously achieve a value that potentially exceeds its melting point for water at atmospheric pressure $\left(\Delta T_e > 1000\ ^\circ C\right)$.
  - If the surface survives the temperature shock, conditions are characterized by *film boiling.*
- **Film boiling** $\left(\Delta T_e \geq 120\ ^\circ C, water\ @1atm\right)$
  - Heat transfer is by conduction and radiation across the vapor blanket.
  - A reduction in q''_s follows the cooling curve continuously to the Leidenfrost point corresponding to the minimum heat flux q''_{min} for film boiling.
  - A reduction in q''_s below q''_{min} causes an abrupt reduction in surface temperature to the nucleate boiling regime.



- **Transition boiling for temperature - Controlled heating**
  - Characterized by a continuous decay of q''_s (from q''_{max} to q''_{min}) with increasing ΔT_e.
  - Surface conditions oscillate between nucleate and film boiling, but portion of surface experiencing film boiling increases with ΔT_e.
  - Also termed unstable or partial film boiling.

Pool boiling correlations

- **Nucleate boiling**
  - **Rohsenow Correlation**

$$q''_s = \mu_L h_{fg}\left[\frac{g(\rho_L-\rho_v)}{\sigma_f}\right]^{1/2}\left(\frac{c_{p,L}\Delta T_e}{C_{s,f}\,h_{fg}\,Pr_l^n}\right)^3 = (\dot{m}_v/A_s)h_{fg}$$

μ_L	Dynamic visocity of liquid phase
h_{fg}	Enyhalpy of vaporization
g	Gravitional acceleration
ρ_L	Density of liquid phase
ρ_v	Density of vapor phase
σ_f	Surface tension
$c_{p,L}$	Specific heat of liquid phase
Pr_L	Prandtl number of liquid phase
$C_{s,f}, n$	Surface/fluid combination

Values of $C_{s,f}$ for various surface–fluid combinations [5–7]

Surface–Fluid Combination	$C_{s,f}$	n
Water–copper		
Scored	0.0068	1.0
Polished	0.0128	1.0
Water–stainless steel		
Chemically etched	0.0133	1.0
Mechanically polished	0.0132	1.0
Ground and polished	0.0080	1.0
Water–brass	0.0060	1.0
Water–nickel	0.006	1.0
Water–platinum	0.0130	1.0
n-Pentane–copper		
Polished	0.0154	1.7
Lapped	0.0049	1.7
Benzene–chromium	0.0101	1.7
Ethyl alcohol–chromium	0.0027	1.7

  - **Critical heat flux of Zuber**

$$(q''_s)_{max} = C h_{fg}\rho_v\left[\frac{\sigma_f g(\rho_L-\rho_v)}{\rho_v^2}\right]^{1/4} = \left[(\dot{m}_v)_{max}/A_s\right]h_{fg}$$

$C=0.131$	Horizontal cylinder or sphere
$C=0.149$	Large horizontal plate

- **Film boiling**

$$q_s=\bar{h}A_s\left(T_s-T_{sat}\right)=\dot{m}_v h'_{fg}$$

$$\bar{h}^{4/3}=\bar{h}_{conv}^{4/3}+\bar{h}_{rad}\bar{h}^{1/3}$$

$$\bar{h}\cong\bar{h}_{conv}+\frac{3}{4}\bar{h}_{rad}\quad if\quad \bar{h}_{conv}>\bar{h}_{rad}$$

$$Nu=\frac{\bar{h}_{conv}D}{k_v}=C\left[\frac{g\left(\rho_L-\rho_v\right)h'_{fg}D^3}{\nu_v k_v\left(T_s-T_{sat}\right)}\right]^{1/4}$$

$C=0.62$	Horizontal cylinder	$A_s=\pi DL$
$C=0.67$	Sphere	$A_s=\pi D^2$

$$h'_{fg}=h_{fg}+0.8c_{p,v}\left(T_s-T_{sat}\right)$$

$$\bar{h}_{rad}=\frac{\varepsilon\sigma\left(T_s^4-T_{sat}^4\right)}{T_s-T_{sat}}$$

$$\sigma=5.67\times10^{-8}\ W/m^2K^4:\quad Stefan-Boltzmann\quad constant$$

Correlation for boiling inside tubes

$$\bar{X}=\frac{\int_{A_C}\rho uX dA_C}{\dot{m}}=\frac{q''_s\pi Dx}{\dot{m}h_{fg}}$$

$\bar{X}$: Mean vapor mass fraction at a given section

$$\frac{h_1}{h_{sp}}=0.6683\left(\frac{\rho_l}{\rho_v}\right)^{0.1}\bar{X}^{0.16}\left(1-\bar{X}\right)^{0.64}f\left(Fr\right)+1058\left(\frac{q''_s}{\left[\dot{m}/A_C\right]h_{fg}}\right)^{0.7}\left(1-\bar{X}\right)^{0.8}G_{s,f}$$

$$\frac{h_2}{h_{sp}} = 1.136\left(\frac{\rho_l}{\rho_v}\right)^{0.45} \bar{X}^{0.72}\left(1-\bar{X}\right)^{0.08} f\left(Fr\right) + 667.2\left(\frac{q_s''}{\left[\dot{m}/A_C\right]h_{fg}}\right)^{0.7}\left(1-\bar{X}\right)^{0.8} G_{s,f}$$

$$Fr = \left(\left[\dot{m}/A_C\right]/\rho_l\right)^2/\left(gD\right)$$

$f\left(Fr\right) = 1.0$	Vertical tubes
$f\left(Fr\right) = 1.0$	Horizontal tubes with $Fr > 0.04$
$f\left(Fr\right) = 2.63Fr^{0.3}$	Horizontal tubes with $Fr < 0.04$

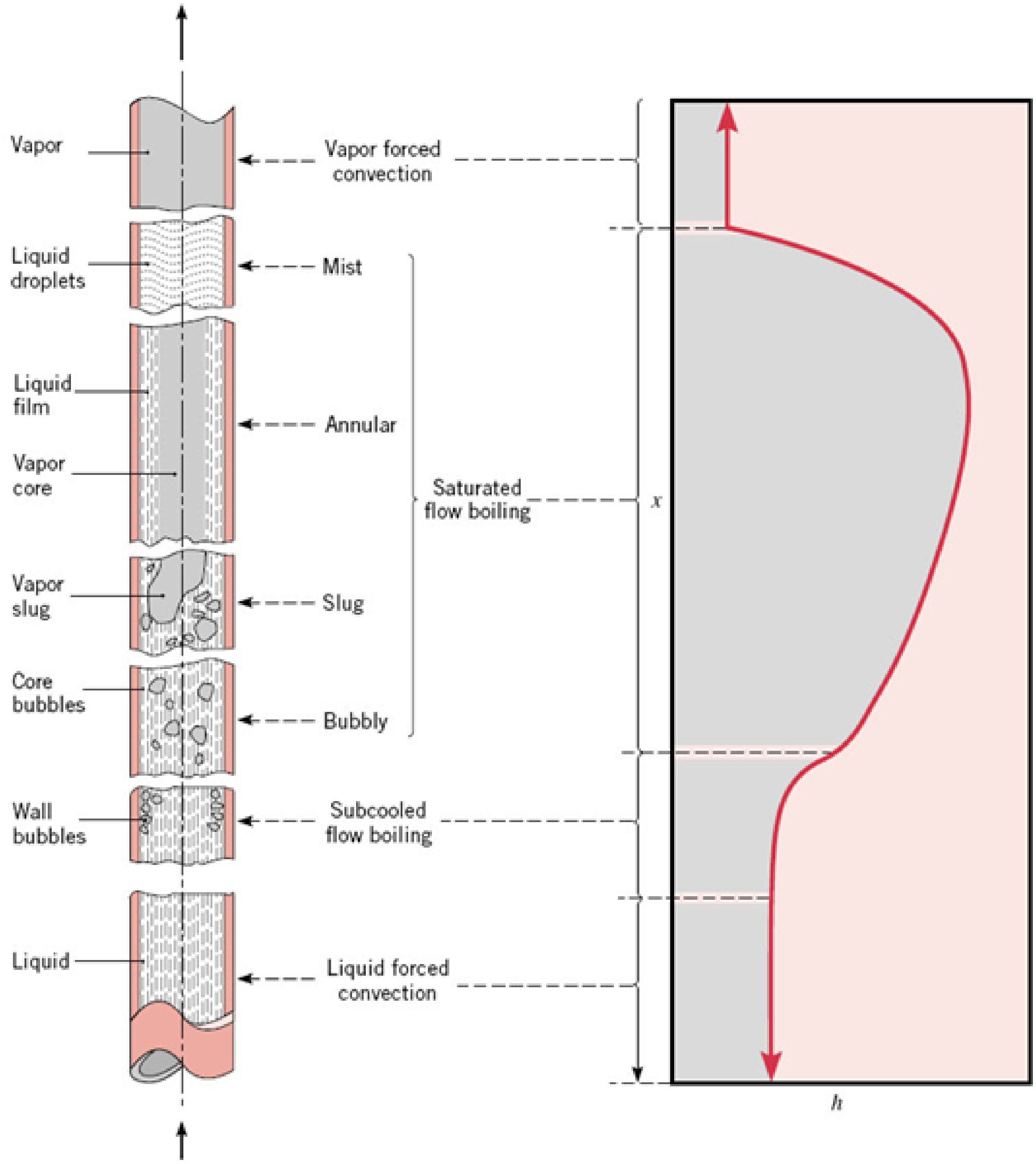
$$\frac{h_{sp}D}{k_i} = 0.023\left(\frac{\rho_l u_{m,l} D}{\mu_l}\right)^{0.8} Pr_i^{0.4}$$

$$h = MAX\left[h_1, h_2\right]$$

$$0 < \bar{X} \le 0.8$$

Values of $G_{s,f}$ for various surface–liquid combinations

Fluid in Commercial Copper Tubing	$G_{s,f}$
Kerosene	0.488
Refrigerant R-134a	1.63
Refrigerant R-152a	1.10
Water	1.00
For stainless steel tubing, use $G_{s,f} = 1$.	



Flow regimes for forced convection boiling in a tube.

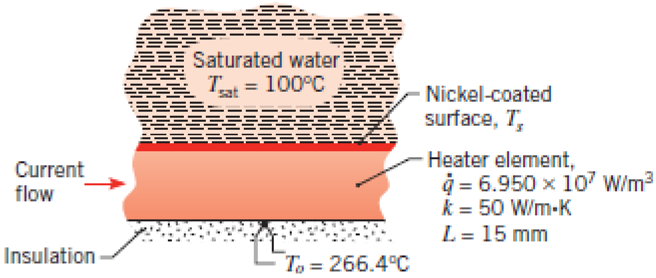
Problems

P1 A long, 1-mm-diameter wire passes an electrical current dissipating 3150 W/m and reaches a surface temperature of 126 °C when submerged in water at 1 atm. What is the boiling heat transfer coefficient? Estimate the value of the correlation coefficient $C_{s,f}$.

P2 A nickel-coated heater element with a thickness of 15 mm and a thermal conductivity of 50 W/mK is exposed to saturated water at atmospheric pressure. A thermocouple is attached to the back surface, which is well insulated. Measurements at a particular operating condition yield an electrical power dissipation in the heater element of 6.950x10^7 W/m^3 and a temperature of T_o= 266.4 °C.

(a) From the foregoing data, calculate the surface temperature, T_s, and the heat flux at the exposed surface.

(b) Using the surface heat flux determined in part (a), estimate the surface temperature by applying an appropriate boiling correlation.



P3 A 1-mm-diameter horizontal platinum wire of emissivity 0.25 is operated in saturated water at 1-atm pressure. What is the surface heat flux if the surface temperature is 800 K?

P4 Copper tubes 25 mm in diameter and 0.75 m long are used to boil saturated water at 1 atm. (a) If the tubes are operated at 75% of the critical heat flux, how many tubes are needed to provide a vapor production rate of 750 kg/h? What is the corresponding tube surface temperature?

P5 A vertical steel tube carries water at a pressure of 10 bars. Saturated liquid water is pumped into the =0.1-m diameter tube at its bottom end (x =0) with a mean velocity of u_m=0.05 m/s. The tube is exposed to combusting pulverized coal, providing a uniform heat flux of q_s=100,000 W/m^2.

(a) Determine the tube wall temperature and the quality of the flowing water at x =15 m. Assume $G_{s,f}$=1.

(b) Determine the tube wall temperature at a location beyond x=15 m where single-phase flow of the vapor exists at a mean temperature of T_{sat}. Assume the vapor at this location is also at a pressure of 10 bars.

Condensation

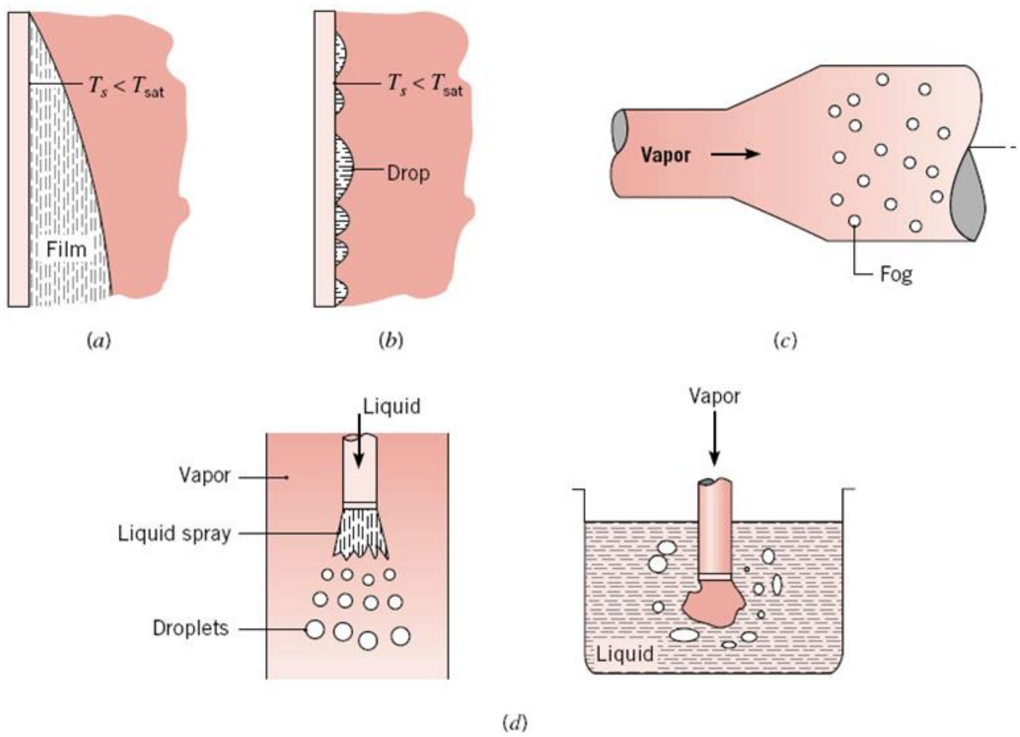
Heat transfer to a surface occurs by condensation when the surface temperature is less than the saturation temperature of an adjoining vapor.

Film condensation

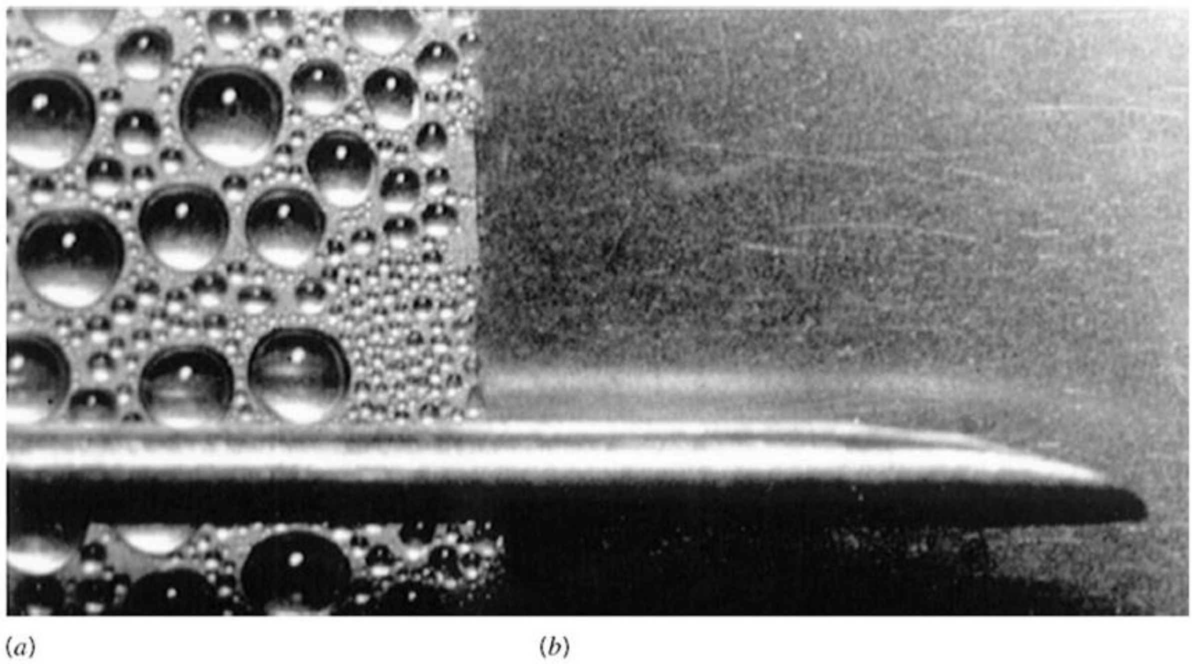
- Entire surface is covered by the condensate, which flows continuously from the surface and provides a resistance to heat transfer between the vapor and the surface.
- Thermal resistance is reduced through use of short vertical surfaces and horizontal cylinders.
- Characteristic of clean, uncontaminated surfaces.

Dropwise condensation

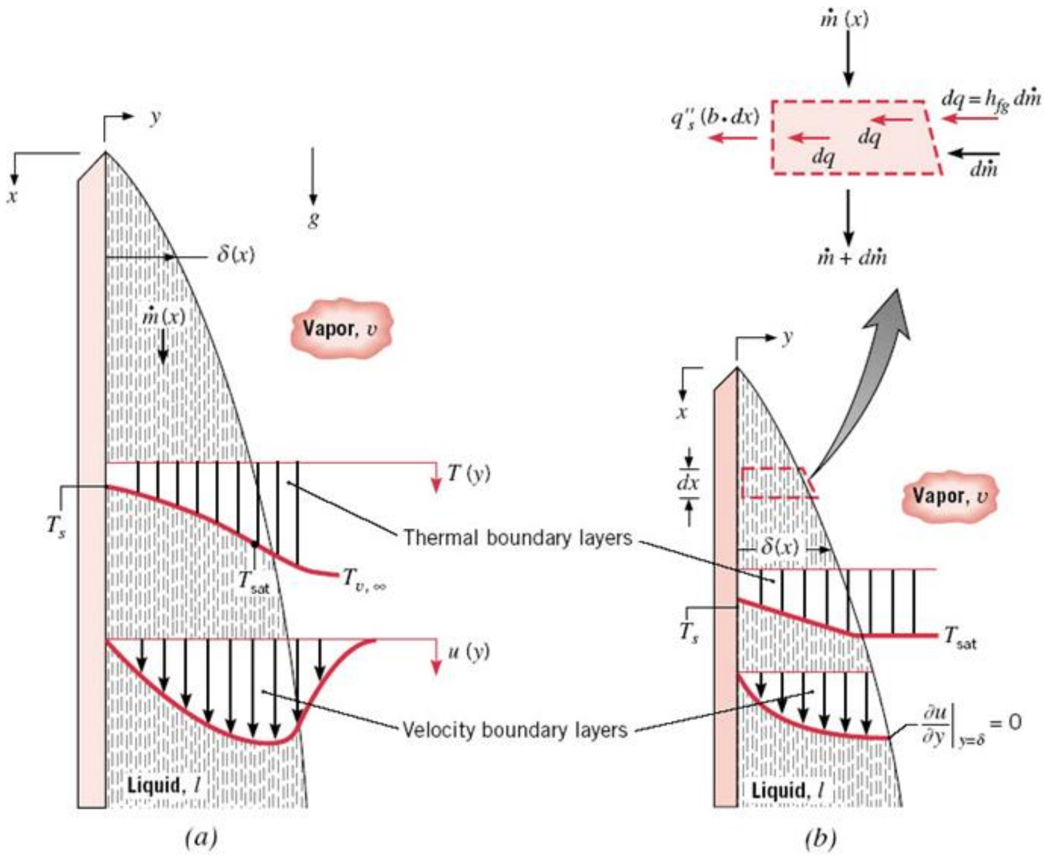
- Surface is covered by drops ranging from a few micrometers to agglomerations visible to the ordinary eyes.
- Thermal resistance is greatly reduced due to absence of a continuous film.
- Surface coatings may be applied to inhibit wetting and stimulate dropwise condensation.



Modes of condensation. (*a*) Film. (*b*) Dropwise condensation on a surface. (*c*) Homogeneous condensation or fog formation resulting from increased pressure due to expansion. (*d*) Direct contact condensation.

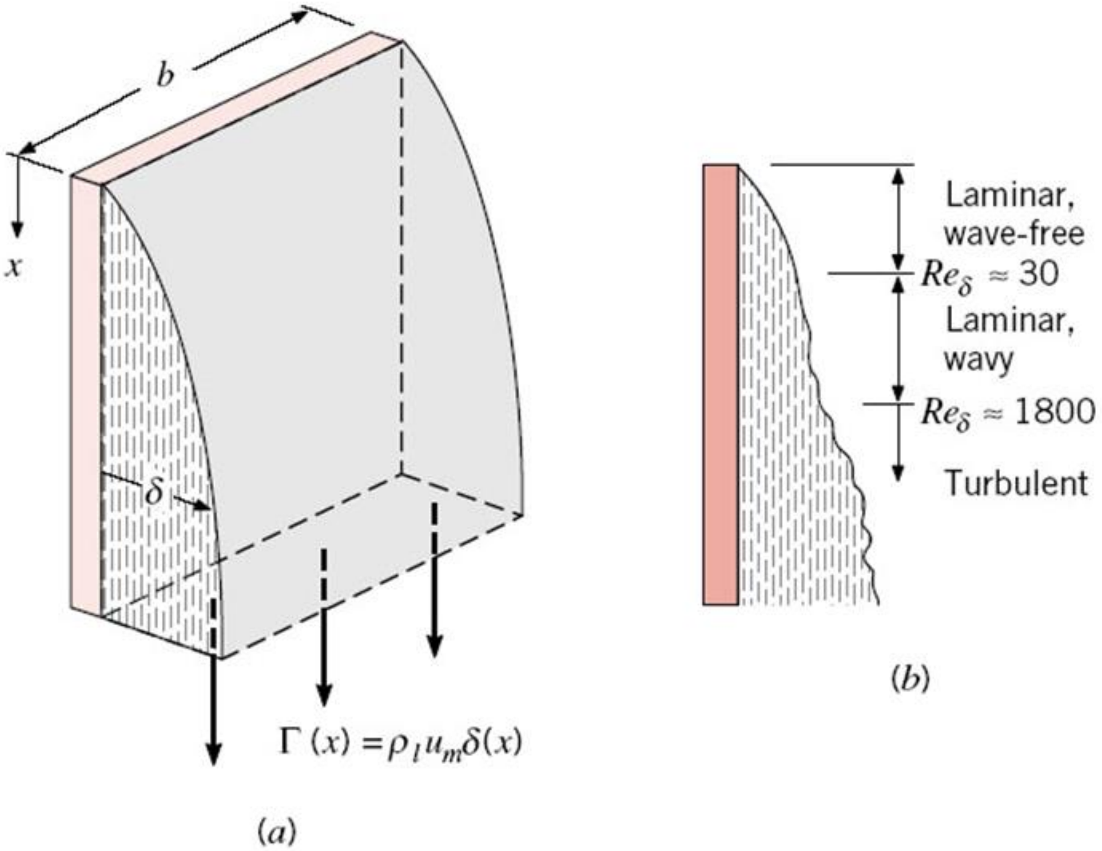


Condensation on a vertical surface. (*a*) Dropwise. (*b*) Film.
Photograph courtesy of Professor J. W. Westwater, University of Illinois at Champaign-Urbana.



Boundary layer effects related to film condensation on a vertical surface.
(*a*) Without approximation. (*b*) With assumptions associated with Nusselt's analysis, for a vertical plate of width *b*.

General correlations for film condensation on vertical surface



Film condensation on a vertical plate.
(*a*) Condensate rate for plate of width *b*. (*b*) Flow regimes.

$$q = \bar{h}_L A_s (T_{sat} - T_s) = \dot{m}_L h'_{fg}$$

$$h'_{fg} = h_{fg} + 0.68 c_{p,l} (T_{sat} - T_s)$$

$$P = \frac{k_l L (T_{sat} - T_s)}{\mu_l h'_{fg} (\nu_l^2 / g)^{1/3}}$$

- **Wavy free laminar region** $(P < 15.8)$

$$\overline{Nu}_L = \frac{\bar{h}_L (\nu_l^2 / g)^{1/3}}{k_l} = 0.943 P^{-1/4}$$

- **Wavy laminar region** $(15.8 < P < 2530)$

$$\overline{Nu}_L = \frac{\bar{h}_L \left(\nu_l^2/g\right)^{1/3}}{k_l} = \frac{1}{P}\left(0.68P + 0.89\right)^{0.82}$$

- **Turbulent region** $(P > 2530;\ Pr_l \geq 1.0)$

$$\overline{Nu}_L = \frac{\bar{h}_L \left(\nu_l^2/g\right)^{1/3}}{k_l} = \frac{1}{P}\left[\left(0.024P - 53\right)Pr_l^{1/2} + 89\right]^{4/3}$$

μ_l	Dynamic visocity of liquid phase
ν_l	kinematic visocity of liquid phase
h_{fg}	Enyhalpy of vaporization
$c_{p,l}$	Specific heat of liquid phase
g	Gravitional acceleration
L	Surface length along g-direction
k_l	Thermal conductivity of liquid phase
$\dot{m}_L$	Condensate mass flow rate at bottom section
A	Surface area
$\bar{h}_L$	Average convection heat transfer coefficent

Correlations for film condensation on radial systems

$$q = \bar{h}_{D,N} A_s \left(T_{sat} - T_s\right) = \dot{m}_l h'_{fg}$$

$$h'_{fg} = h_{fg} + 0.68 c_{p,l}\left(T_{sat} - T_s\right)$$

- **Single smooth sphere and single horizontal tube (N=1)**

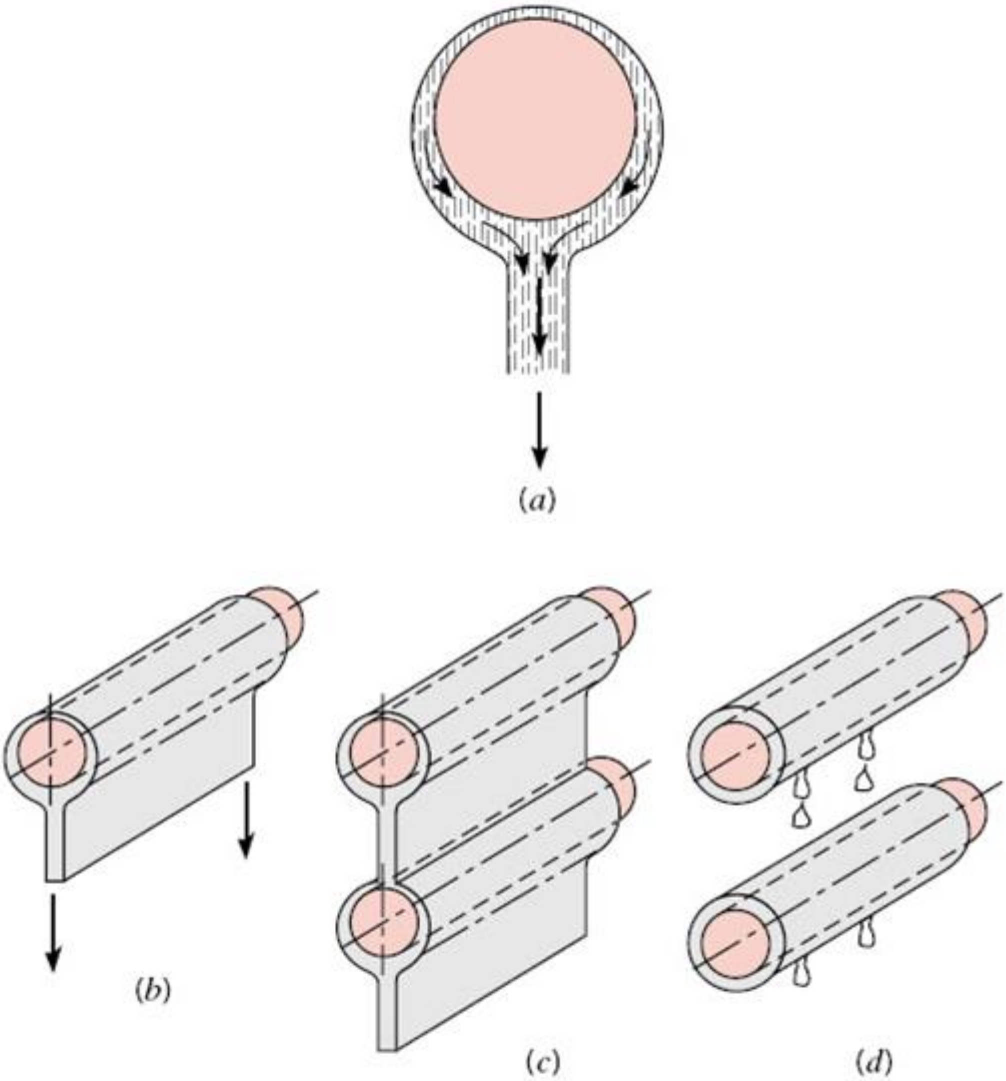
$$\overline{Nu}_D = \frac{\bar{h}_D D}{k_l} = C\left[\frac{\rho_l g\left(\rho_l - \rho_v\right) h'_{fg} D^3}{\mu_l k_l \left(T_{sat} - T_s\right)}\right]^{1/4}$$

$C = 0.729$	Horizontal tube $\left(A_s = \pi D L\right)$
$C = 0.826$	sphere $\left(A_s = \pi D^2\right)$

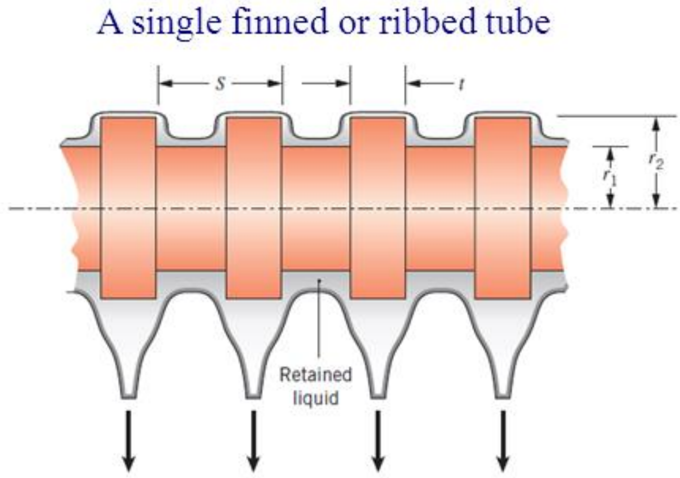
- **<u>Vertical tier</u> of <u>*N*</u> <u>horizontal tubes</u>**

$$\bar{h}_{D,N} = \bar{h}_D N^n$$

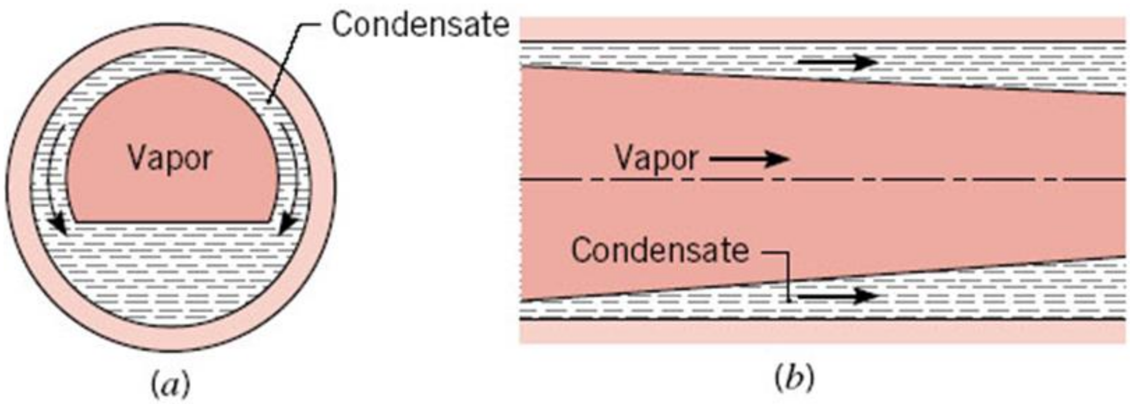
$n = -1/4$	Analytical value, assuming continuous film between smooth tubes
$n = -1/6$	Experimental value, due to dripping between smooth tubes
$-1/6 < n < 0$	Experimental value, finned or ribbed tubes



Film condensation on (*a*) a sphere, (*b*) a single horizontal tube, (*c*) a vertical tier of horizontal tubes with a continuous condensate sheet, and (*d*) with dripping condensate.



Correlation for film condensation in horizontal tube



Film condensation in a horizontal tube.
(*a*) Cross section of condensate flow for low vapor velocities.
(*b*) Longitudinal section of condensate flow for large vapor velocities.

(a) Small vapor mass flow rate

$$Re_{v,i} = \left(\frac{\rho_v u_{m,v} D}{\mu_v}\right) < 35{,}000$$

$$h'_{fg} = h_{fg} + 0.375 c_{p,l}\left(T_{sat} - T_s\right)$$

$$\overline{Nu_D} = \frac{\bar{h}_D D}{k_l} = 0.55\left[\frac{\rho_l g(\rho_l - \rho_v)h'_{fg}D^3}{\mu_l k_l (T_{sat} - T_s)}\right]^{1/4}$$

(b) Large vapor mass flow rate

$$Nu_D = \frac{h_D D}{k_l} = 0.023\, Re_{D,l}^{0.8}\, Pr_i^{0.4}\left[1 + \frac{2.22}{X_{tt}^{0.89}}\right]$$

$$Re_{D,l} = \frac{4\dot{m}(1 - X)}{\pi D \mu_l}; \quad X \equiv \frac{\dot{m}_v}{\dot{m}}$$

$$Martinelli \quad parameter : X_{tt} = \left(\frac{1 - X}{X}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}$$

<u>Correlation for dropwise steam condensation on copper surfaces</u>

$$q = \bar{h}_{dc} A_s (T_{sat} - T_s)$$

$$\bar{h}_{dc} = 51{,}104 + 2044 \times T_{sat}, \quad 22^\circ C < T_{sat} < 100^\circ C$$

$$\bar{h}_{dc} = 255{,}510 \quad W/m^2K \quad , \quad T_{sat} > 100^\circ C$$

Problems

P1 Saturated steam at 1 atm condenses on the outer surface of a vertical, 100-mm-diameter pipe 1 m long, having a uniform surface temperature of 94 °C. Estimate the total condensation rate and the heat transfer rate to the pipe.

P2 The condenser of a steam power plant consists of a square (in-line) array of 625 tubes, each of 25-mm diameter. Consider conditions for which saturated steam at 0.105 bars condenses on the outer surface of each tube, while a tube wall temperature of 17 °C is maintained by the flow of cooling water through the tubes. What is the rate of heat transfer to the water per unit length of the tube array? What is the corresponding condensation rate?

P3 The condenser of a steam power plant consists of AISI 302 stainless steel tubes (k_s=15 W/m.K), each of outer and inner diameters d_o=30 mm and d_i=26 mm, respectively. Saturated steam at 0.135 bar condenses on the outer surface of a tube, while water at a mean temperature of T_m=290 K is in fully developed flow through the tube. For a water flow rate iside the tube of 0.25 kg/s, what is the outer surface temperature $T_{s,o}$ of the tube and the rates of heat transfer and steam condensation per unit tube length? As a first estimate, you may evaluate the properties of the liquid film at the saturation temperature. If one wishes to increase the transfer rates, what is the limiting factor that should be addressed?

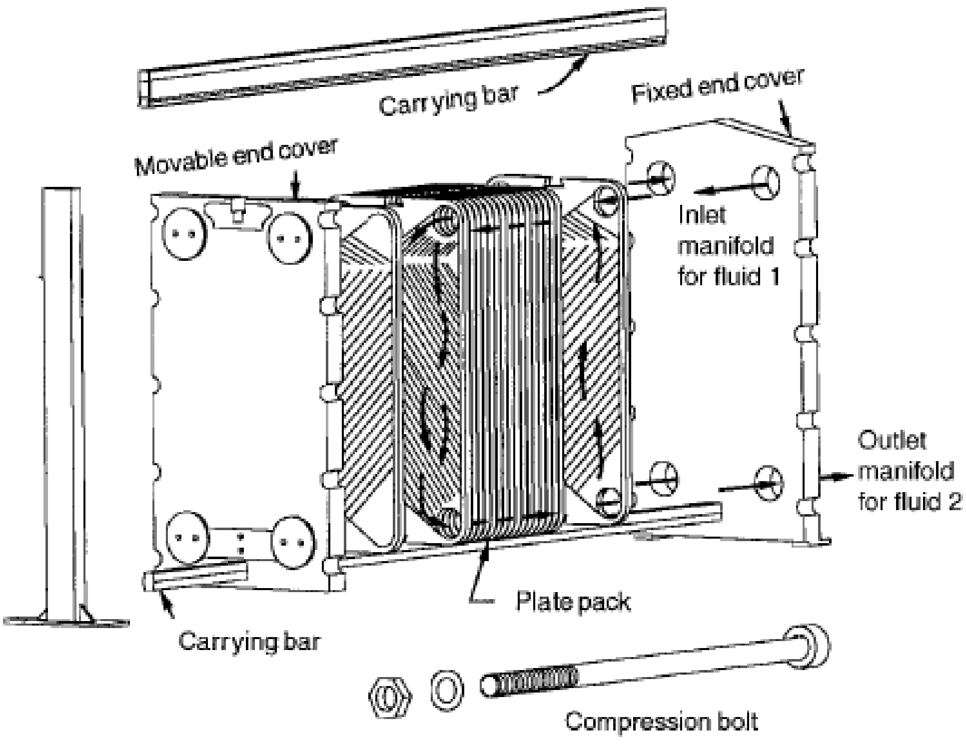
P4 Refrigerant R-22 with a mass flow rate of $8.75\text{x}10^{-3}$ kg/s is condensed inside a 7-mm-diameter tube. Annular flow is observed. The saturation temperature of the pressurized refrigerant is T_{sat}=45 °C, and the wall temperature is T_s=40 °C. Vapor properties are ρ_v=77 kg/m^3 and μ_v=15x10^{-6} N.s/m^2. Determine the heat transfer coefficient and the heat transfer and condensation rates per unit length at a quality of X=0.5.

The Gasketed-Plate Heat Exchangers

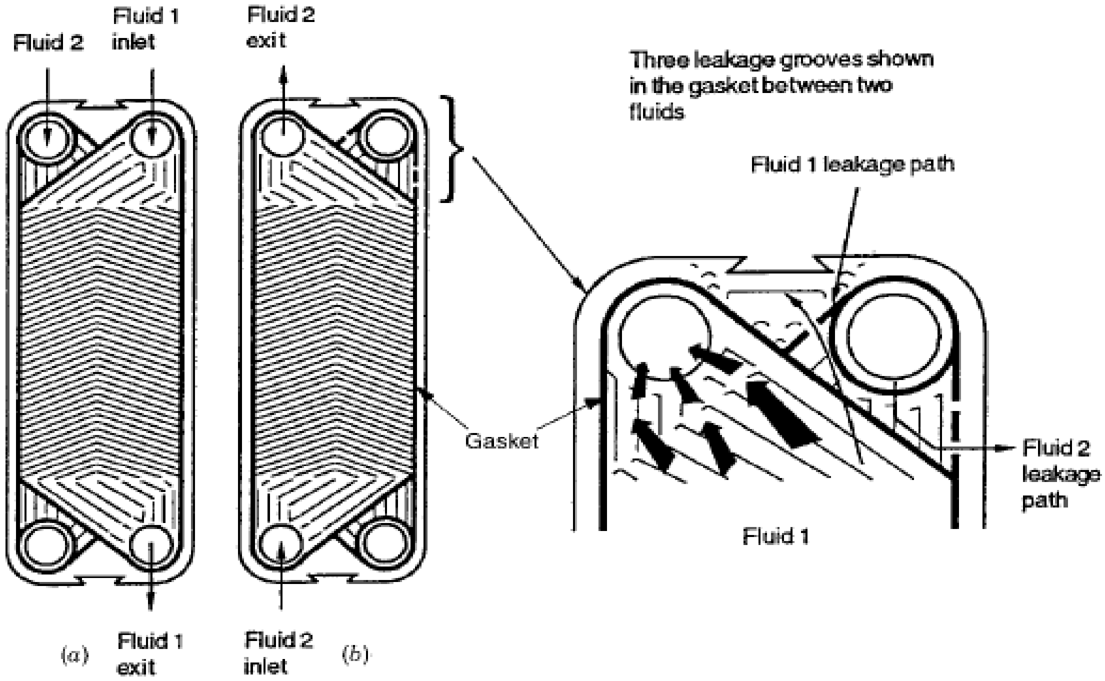
- Gasketed Plate Heat Exchangers (**G-P HX**s) were introduced mainly for the food industry because of their ease of cleaning.
- **G-P HX**s design becomes well identified in 1960s with the development of more effective plate geometries, assemblies, and improved gasket materials.
- **G-P HX**s can be used as an alternative to tube-and-shell type heat exchangers for low-and medium-pressure liquid-to-liquid heat trasfer applications.
- Unlike to common design of heat exchangers, manufactrures of **G-P HX**s have developed their own computerized design procedures applicable to the exchangers they market.

Main Components

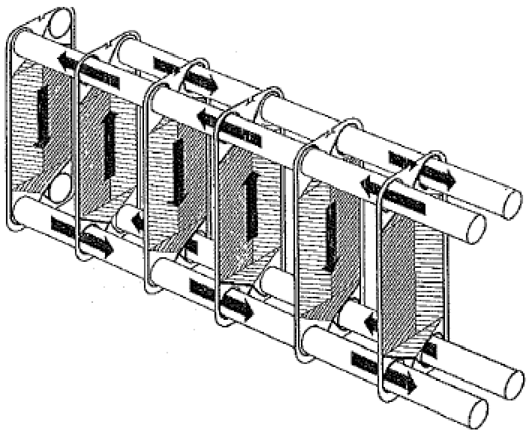
- **G-P HX** components are the plate (the heat transfer solid surface) and the frame.
- Elements of the frame: fixed plate, compression plate, pressing equipment, and connecting ports.
- Elements of the heat transfer soild surface: series of plates, parts for fluid entry and exit in the four corners.



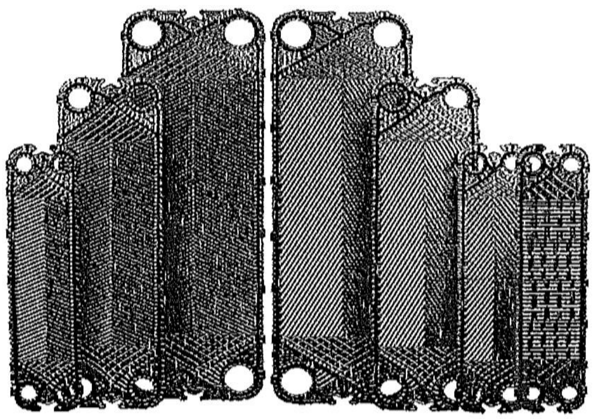
Gasketed plate- and-frame heat exchanger



Plates showing gaskets around the ports (Shah and Focke, 1988).



Flow diagram in a single-pass counterflow arrangement. (Courtesy of Alfa-Laval Thermal AB.)

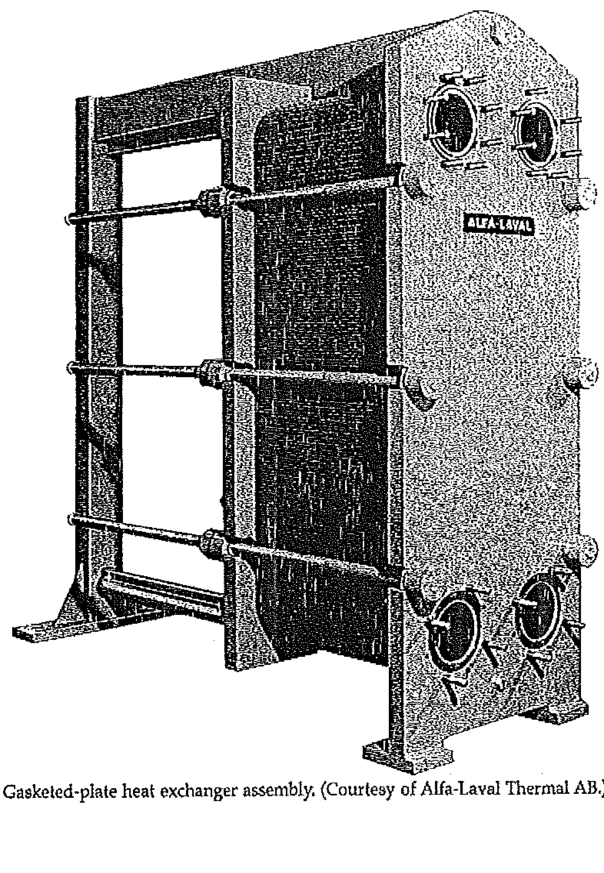


A chevron type heat exchanger plate. (Courtesy of Alfa-Laval Thermal AB.)

<u>The Plate Pack</u>

- Packing the plates make the holes at the corners to form continuous tunnels or mainfolds.
- The plate pack is tightnend by means of either a mechanical or hydraulic tightening device.
- The passages formed between the plates and corner ports are arranged so that the two heat transfer media can flow through alternate channels, always in counter-current flow.
- The warmer medium will give some of its thermal energy through the thin plate wall to the colder medium on the other side.

- The medium are led into similar hole-tunels as in the inlets at the other end of the plate package and are then discharged from the heat exchanger.
- Several hundereds of plates can be stacked in a single frame which are held together by the bolts that hold the stack in compression.
- The two sides of the plate heat exchanger are normaly of identical hydrodynamic characteristics.
- The plate is a sheet of metal precison-pressed into a corrugated pattern.
- The largest single plate is the order of 4.3 m heigh x 1.1 m wide.
- The heat transfer area for a single plate lies in the range 0.01-3.6 m^2.
- The plate thickness ranges between 0.5 and 1.2 mm.
- The plates are spaced with nominal gaps of 2.5-5.0 mm.
- The hydraulic diameters for the flow channels ranges between 4-10 mm.
- The fluid should be equally distributed over the full width of the plate. This requires the minmum length/width ratio of the order of 1.8.
- Leakage from the channels between the plates to the surrounding atmosphere is prevented by the gasketing around the exterior of the plate.
- The number and size of the plates are determined by the flow rate, physical properties of the fluids, pressure drop, and the temperature requirements.

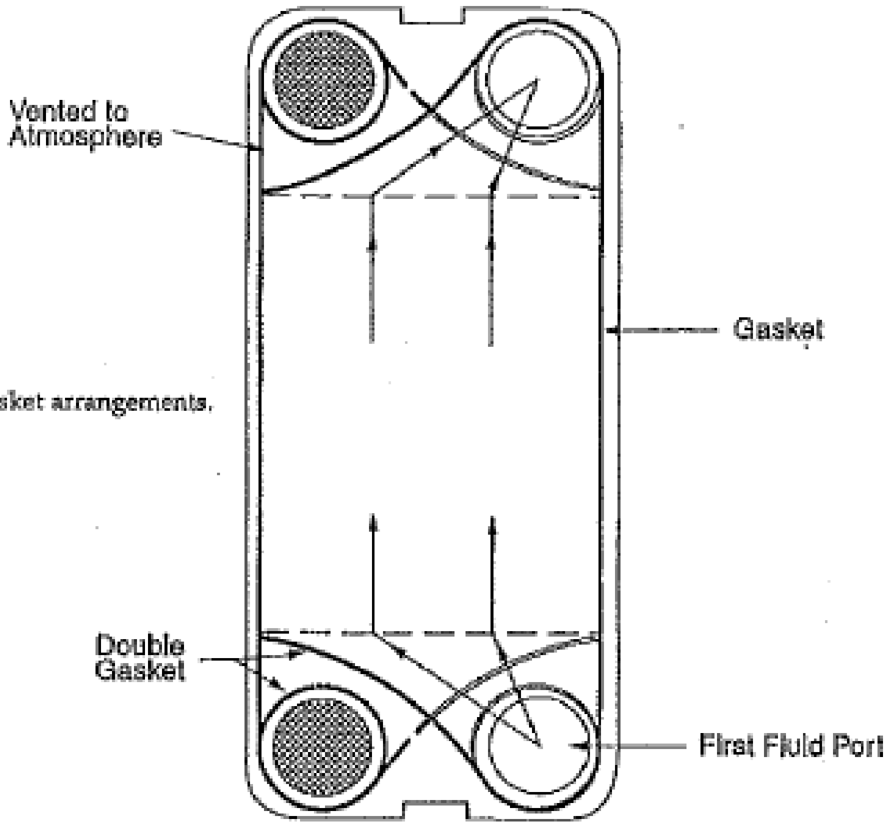


Gasketed-plate heat exchanger assembly. (Courtesy of Alfa-Laval Thermal AB.)

Plate Materials

Material	Thermal Conductivity (W/m^2 · K)
Stainless steel (316)	16.5
Titanium	20
Inconel 600	16
Incolay 825	12
Hastelloy C-276	10.6
Monel 400	66
Nickel 200	66
9/10 Cupro-nickel	52
70/30 Cupro-nickel	35

From Raju, K. S. N. and Jagdish, C. B. [1983] In *Low Reynolds Number Flow Heat Exchangers*, Hemisphere, Washington, D.C. With permission.



Gasket arrangements.

The Plate Types

- Wide types of corrugated plates are available including chevron and washboard types.
- The most used type is the chevron type.
- In washboard type, turbulence is promoted by continuously altering flow direction and fluid's velocity.

- In chevron type, adjacent plates are assembled such that the flow in the channels provides swirling motion to the fluids.
- The Chevron angle (β) varies between the extremes of about 65° and 25°.
- The Chevron angle (β) determines the pressure drop and heat transfer characteristics of the plate.
- The chevron angle (β) is reversed on adjacent plates so that when plates are clamped together, the corrugations provide numerous contact points.
- Because of the many supporting contact points, the plates can be made from very thin material, usually 0.6 mm.

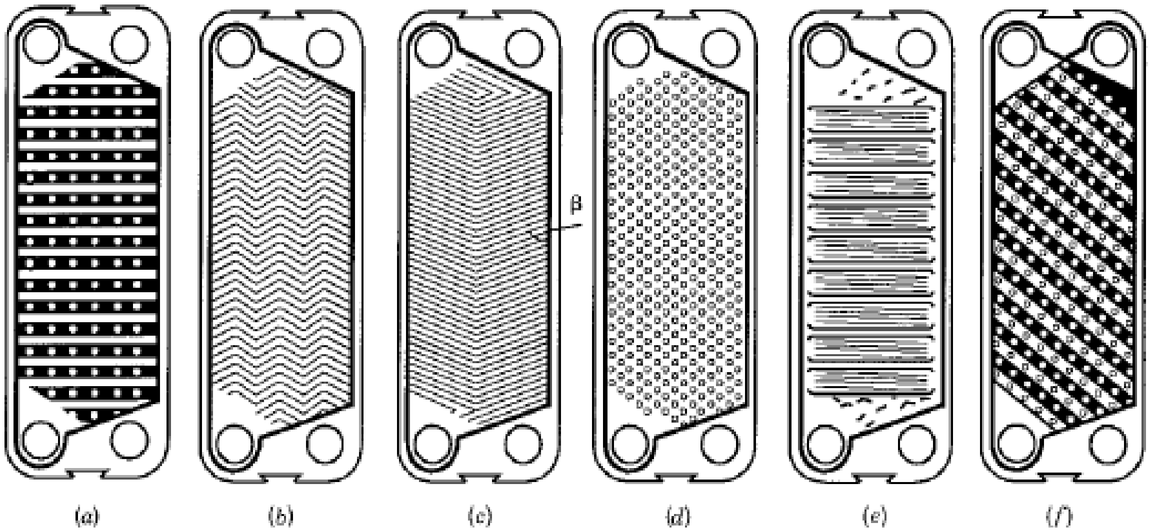
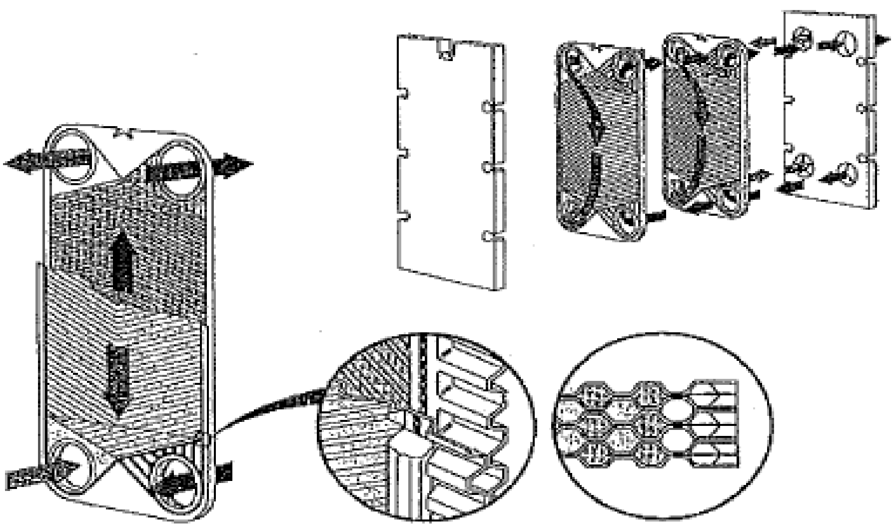
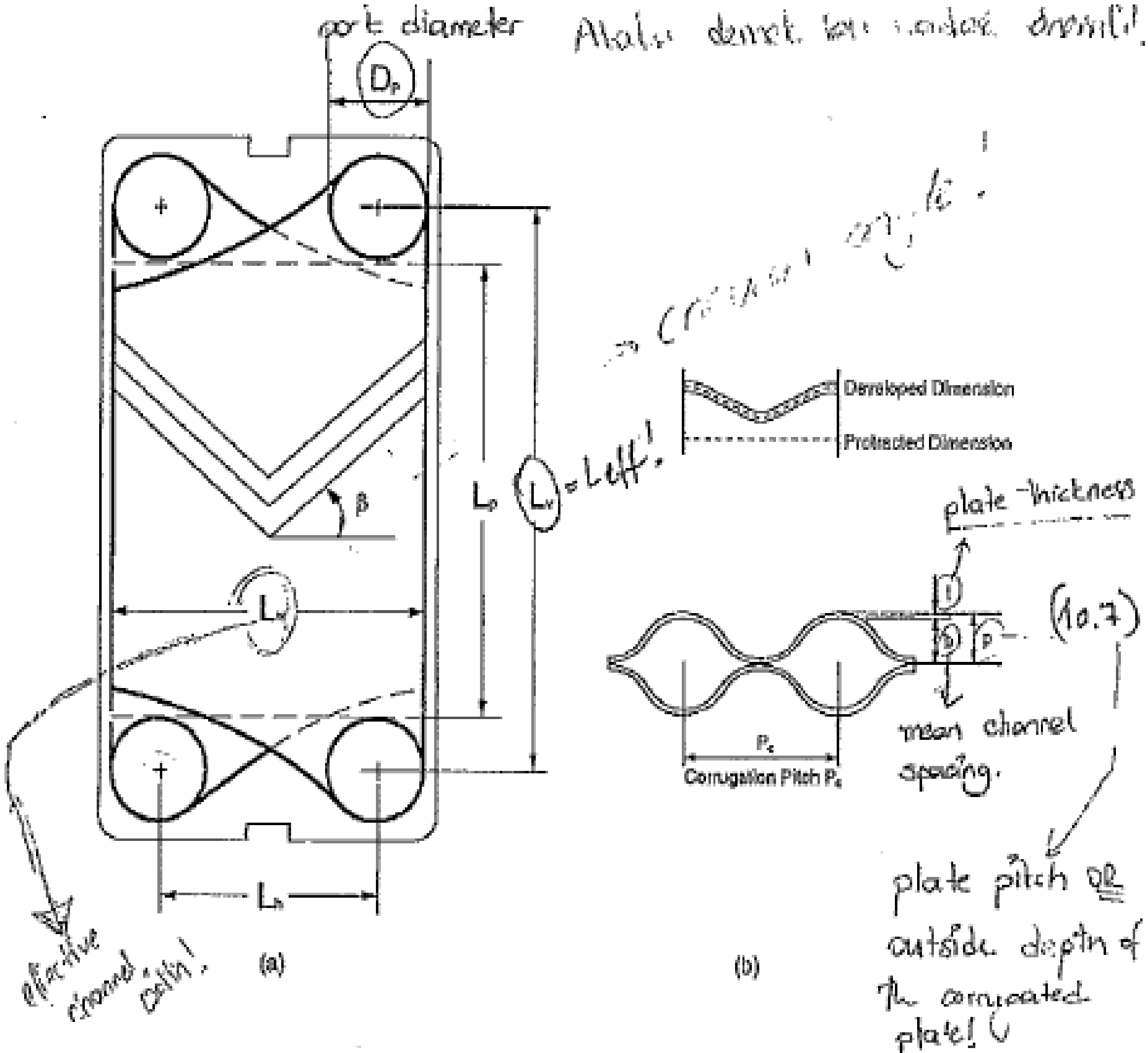


Plate patterns:
(a) washboard; (b) zigzag; (c) chevron or herringbone;
(d) protrusions and depressions;
(e) washboard with secondary corrugations; (f) oblique washboard (Shah and Focke, 1988).



The chevron angle is reversed on adjacent plates. (Courtesy of Alfa-Laval Thermal AB.)



(a) Main dimensions of a chevron plate; (b) developed and projected dimensions of a chevron plate cross-section normal to the direction of troughs.

Main Advantages

- Flexibility of design through a variety of plate sizes and passes arrangements.
- Easily accessible heat transfer area, allowing changes in configuration to suit changes in processes requirements through changes in the number of plates.
- Efficient heat transfer; high heat transfer coefficients for both fluids because of turbulence and a small hydraulic diameter.
- Very compact (large heat transfer area/volume ratio yet 2500 m^2 of surface area is available in a single unit), and low in weight.
- Only the plate edges are exposed to the atmosphere, no insulation is required as heat losses are negligible.
- Inter-mixing of the two fluids cannot occur under gasket failure.

- Plate units exhibit low fouling characteristics due to high turbulence and low residence time. The transition to turbulence occurs at low Reynolds number of 10 to 400.
- More than two fluids may be processed in a single unit.
- Less expensive than tubular heat exchangers if the tubes are made from stainless steel.

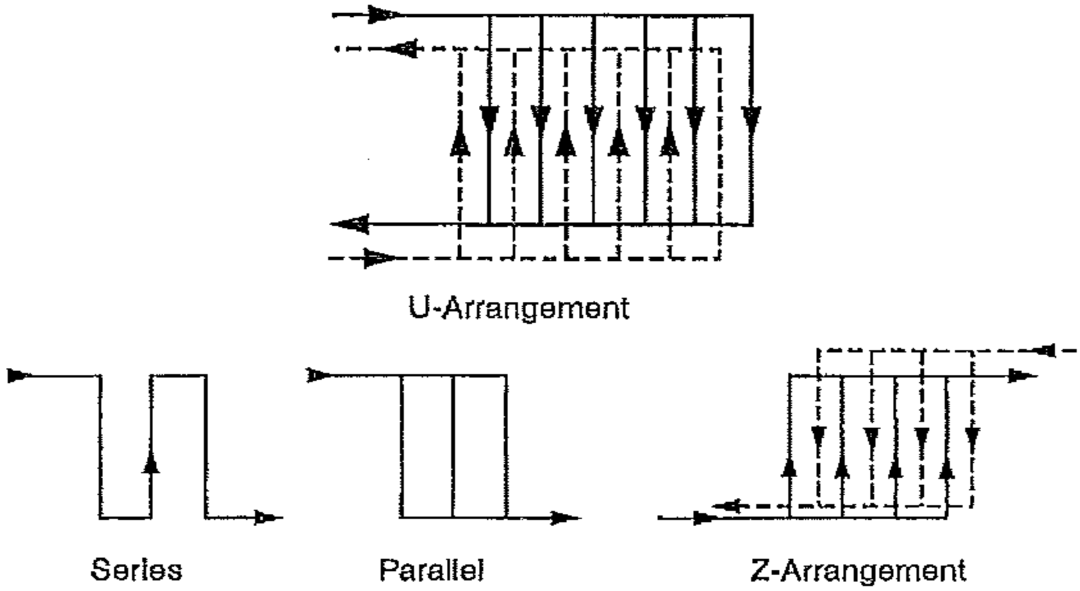
Performance limits

- The gaskets impose restriction on operating temperatures (160°C-250°C).
- The gaskets impose restriction on operating pressures (25-30 bar).
- The gaskets impose restriction on nature of fluids that can be handled.
- The upper size of the G-P HX is limited by the presses available to stamp out the plates from the sheet metal.
- G-P HXs with sizes larger than 1500 m^2 are not normally available.
- It is possible to have a maximum design pressure of up to 2.5 MPa; normally, the design pressure is around 1.0 MPa.
- Operating temperatures are limited by the availability of suitable gasket materials.
- G-P HXs are not suitable for air coolers or gas-to-gas applications.
- Velocities lower than 0.1 m/s are not used in plate heat exchangers.
- High viscous fluids are not preferred to be used in G-P HXs.
- Specially designed G-P HXs are now available for duties involving evaporation and condensation systems.

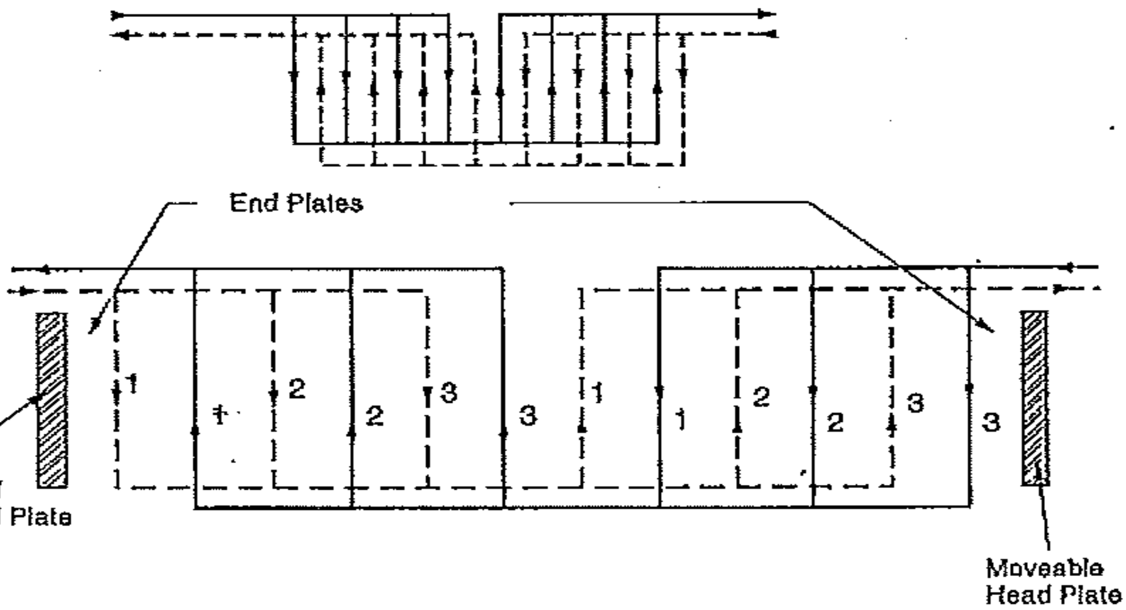
Passes and Flow Arrangements

- The term "pass" refers to a group of channels in which the flow is in the same direction.
- Single pass arrangement can be of **U-** and **Z-arrangement** types.
- **The U-arrangement:** all four ports in this arrangement will be on the fixed-head plate. This permits disassembly of the heat exchanger for cleaning or repair without disturbing any external piping. The flow distribution for this arrangement is less uniform than the **Z-arrangement**.
- **The Multipass arrangement:** consists of passes connected in series.
- Arrangements abbreviation is written as:

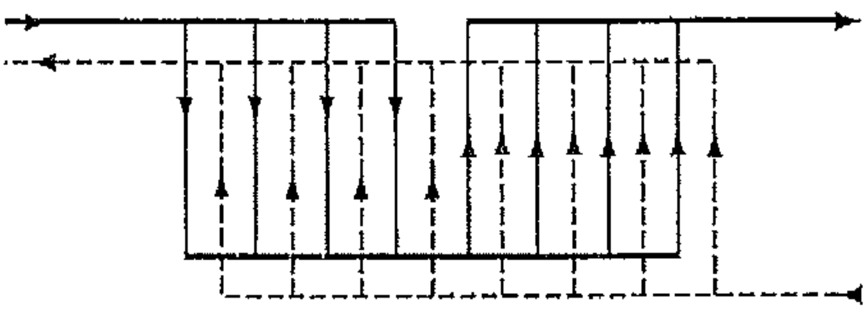
$$N_p \text{ of } 1^{st} \text{ fluid} \times \text{ no. of channels for } 1^{st} \text{ fluid/ } N_p \text{ of } 2^{nd} \text{ fluid} \times \text{ no. of channels for } 2^{nd} \text{ fluid}$$



Flow pattern: (a) schematic of a U-type arrangement — counterflow, single-pass flow ($1\times6/1\times6$) (b) Z-arrangement ($1\times4/1\times4$ configuration).



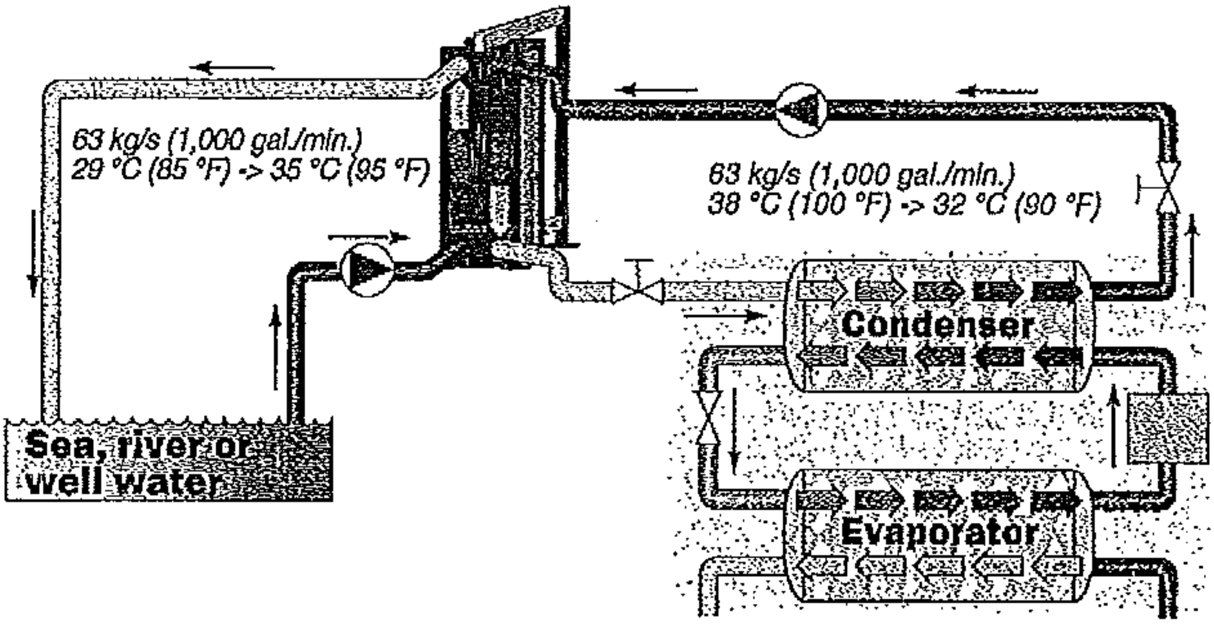
Schematic arrangement of a 2 pass/2 pass flow system ($2\times3/2\times3$ configuration).



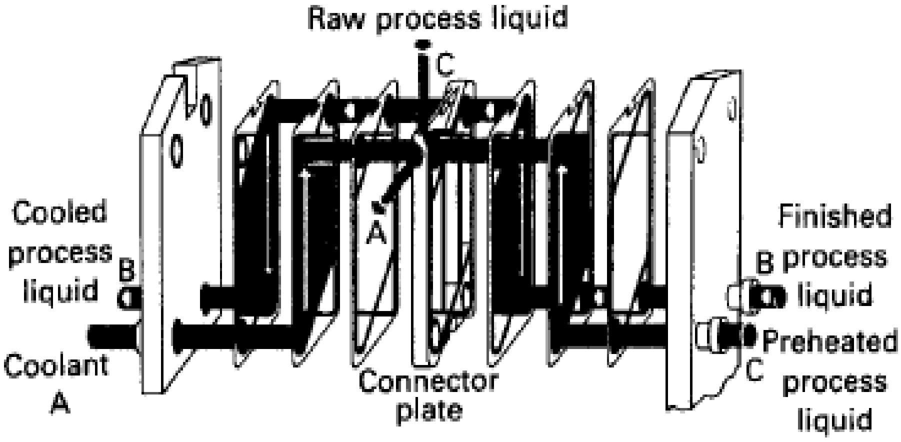
Schematic arrangement of a two-pass/one-pass flow system ($2\times4/1\times8$ configuration).

Applications

- **G-P HX**s are used in chemical, pharmaceutical, hygiene products, biochemical processing, food, and dairy industries as they can meet health and sanitation requirements.
- **G-P HX**s are mainly used as liquid-to-liquid turbulent flow **HX**s.



Closed-circuit cooling system. (Courtesy of Alfa-Laval Thermal AB.)

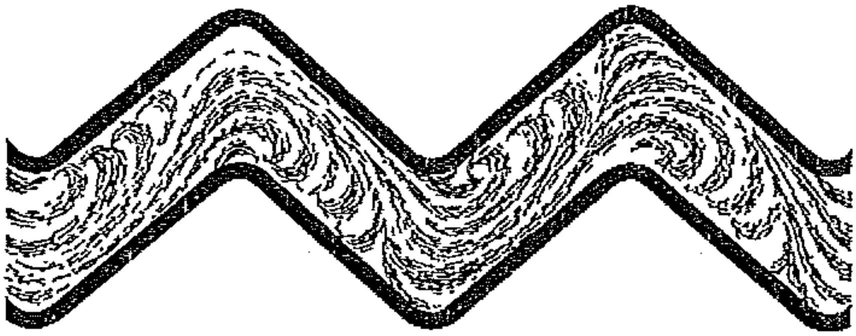


A three-fluid plate heat exchanger. (Courtesy of Alfa Laval Thermal, Inc., Lund, Sweden.)

Some Geometrical and Operating Condition Characteristics of Plate-and-Frame Heat Exchangers

Unit		Operation	
Maximum surface area	2500 m^2	Pressure	0.1 to 3.0 MPa
Number of plates	3 to 700	Temperature	−40 to 260°C
Port size	Up to 400 mm (for liquids)	Maximum port velocity	6 m/s (for liquids)
		Channel flow rates	0.05 to 12.5 m^3/h
		Maximum unit flow rate	2500 m^3/h
Plates		**Performance**	
Thickness	0.5 to 1.2 mm	Temperature approach	As low as 1°C
Size	0.03 to 3.6 m^2	Heat exchanger efficiency	Up to 93%
Spacing	1.5 to 7 mm	Heat transfer coefficients for water–water duties	3000 to 8000 $W/m^2 \cdot K$
Width	70 to 1200 mm		
Length	0.4 to 5 m		
Hydraulic diameter	2 to 10 mm		
Surface area per plate	0.02 to 5 m^2		

Source: Data from Shah (1994).



Flow regime between plates. (Courtesy of Alfa-Laval Thermal AB.)

Materials Selection

Application	Material
Natural cooling water, cooling tower water, or demineralized water	Stainless steel 316
Sea or brackish water	Titanium
Dilute sulphuric and nitric acids up to 10% concentration, and for temperatures up to 70°C	Titanium, titanium-palladium alloy, Incoloy 825, Hastelloy
Chloride solution	
Chloride content < 200 ppm	Stainless steel
Chloride content > 200 ppm	Titanium
Caustic solutions (50 to 70%)	Nickel
Wet chloride, chlorinated brines, hypochlorite solutions	Titanium
Copper sulphate solution in electrolyte refining	Stainless steel
Cooling hydrogen gas saturated with water vapor and mercury carryover in electrolysis plants	Incoloy

From Raju, K. S. N. and Jagdish, C. B. [1983] In *Low Reynolds Number Flow Heat Exchangers*, Hemisphere, Washington, D.C. With permission.

Fouling of G-P HXs

G-P HXs have less fouling than tubular heat exchangers because of the following reasons:

- High turbulence maintains solids in suspension.
- Velocity profiles across a plate are uniform with almost absent zones of low velocities.
- The plate surfaces are generally smooth and can be further electro-polished.
- Deposit of corrosion products are absent because of low corrosion rates.
- The high film coefficients maintain a moderately low metal wall temperature. This helps preventing crystallization growth.
- The plates can be easily cleaned.

Recommended Fouling Factors for Plate Heat Exchangers

Service	Fouling Factor, $m^2 \cdot K/W$
Water	
Demineralized or distilled	0 0000017
Soft	0.0000034
Hard	0.0000086
Cooling tower (treated)	0.0000069
Sea (coastal) or estuary	0.000086
Sea (ocean)	0.000052
River, canal, tube well, etc.	0.000086
Engine jacket	0.0000103
Steam	0.0000017
Lubricating oils	0.0000034–0.0000086
Vegetable oils	0.0000017–0.0000052
Organic solvents	0.0000017–0.0000103
General process fluids	0.0000017–0.00000103

From Raju, K. S. N. and Jagdish, C. B. [1983] In *Low Reynolds Number Flow Heat Exchangers*, Hemisphere, Washington, D.C.; Cooper, A. et al. [1980] *Heat Transfer Eng.*, Vol. 1, No. 3, 50–55. With permission.

Heat Transfer and Pressure Drop Calculations

- Surface enlargement factor $(\phi), \left\{ \phi = \dfrac{Developed \quad length}{Projected \quad length} \right\}, \; 1.15 \le \phi \le 1.25.$

$$\phi = \frac{A_1}{A_{1p}};$$

A_1: Actual effective area provided by manufacturer;

A_{1p}: The projected plate area;

$$A_{1p} = L_p \cdot L_w; \qquad L_p \approx L_v - D_p; \qquad L_w \approx L_h + D_p;$$

L_v: The vertical ports distance;

L_w: The horizontal ports distance.

- The mean channel spacing (b):

$$b = p - t;$$

p: The plate pitch; t: the plate thickness

- The compressed plate pact length (L_c), $p = \dfrac{L_c}{N_t}$, N_t: total number of plates.
- The hydraulic diameter of the channel (D_h):

$$D_h = \frac{4bL_w}{2(b + L_w\phi)} \approx \frac{2b}{\phi};$$

$$Re = \frac{G_c D_h}{\mu};$$

$$G_c = \frac{\dot{m}}{N_{cp} b L_w};$$

N_{cp}: Number of channels per pass;

$$N_{cp} = \frac{N_t - 1}{2N_p},$$ N_p: number of passes

- Convection HT Coefficients Correlation:

$$\frac{hD_h}{k} = C_h\, Re^n\, Pr^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.17}$$

- The frictional pressure drop (Δp_c):

$$\Delta p_c = 4f\,\frac{L_v N_p}{D_h}\frac{G_c^2}{2\rho}\left(\frac{\mu_b}{\mu_w}\right)^{-0.17}, \qquad f = \frac{K_p}{Re^m}$$

- The port pressure drop (Δp_p):

$$\Delta p_p = 1.4 N_p \frac{G_p^2}{2\rho}, \qquad G_p = \frac{\dot{m}}{\pi D_p^2 / 4}.$$

- The total pressure drop (Δp_t):

$$\Delta p_t = \Delta p_c + \Delta p_p.$$

- The overall heat transfer coefficient:

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{t}{k_w};$$

$$\frac{1}{U_f} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{t}{k_w} + R_{fh} + R_{fc}$$

- Required Heat Duty, A_e is the total developed area for all thermally effective plates $(N_t - 2)$:

$$Q_r = (\dot{m}c_p)_c (T_{c2} - T_{c1}) = (\dot{m}c_p)_h (T_{h1} - T_{h2});$$

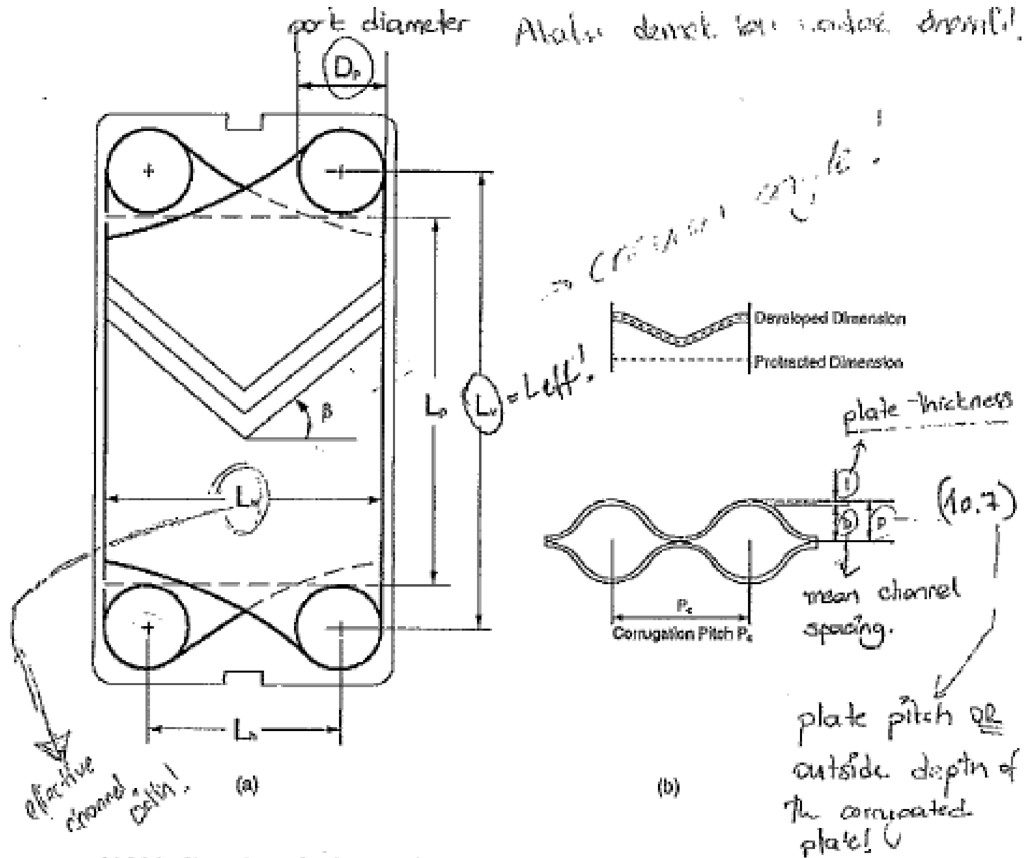
$$Q_f = U_f A_e F \Delta T_{lm,cf};$$

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{ln\,\frac{\Delta T_1}{\Delta T_2}};$$

$$\Delta T_1 = T_{h2} - T_{c1};\, \Delta T_2 = T_{h1} - T_{c2}$$

- The safety factor of the design (C_s),

$$C_s = \frac{Q_f}{Q_r}$$



(a) Main dimensions of a chevron plate; (b) developed and projected dimensions of a chevron plate cross-section normal to the direction of troughs.

Constants for Single-Phase Heat Transfer and Pressure Loss Calculation in Gasketed-Plate Heat Exchangers[2,10]

Chevron Angle (degree)	Heat Transfer: Reynolds Number	C_h	n	Pressure Loss: Reynolds Number	K_p	m
≤ 30	≤ 10	0.718	0.349	< 10	50.000	1.000
	> 10	0.348	0.663	10–100	19.400	0.589
				> 100	2.990	0.183
45	< 10	0.718	0.349	< 15	47.000	1.000
	10–100	0.400	0.598	15–300	18.290	0.652
	> 100	0.300	0.663	> 300	1.441	0.206
50	< 20	0.630	0.333	< 20	34.000	1.000
	20–300	0.291	0.591	20–300	11.250	0.631
	> 300	0.130	0.732	> 300	0.772	0.161
60	< 20	0.562	0.326	< 40	24.000	1.000
	20–400	0.306	0.529	40–400	3.240	0.457
	> 400	0.108	0.703	> 40	0.760	0.215
≥ 65	< 20	0.562	0.326	50	24.000	1.000
	20–500	0.331	0.503	50–500	2.800	0.451
	> 500	0.087	0.718	> 500	0.639	0.213

Problems

10.1. The following constructional information is available for a gasketed-plate heat exchanger:

Chevron angle	50°
Enlargement factor	1.17
All port diameters	15 cm
Plate thickness	0.0006 m
Vertical port distance	1.50 m
Horizontal port distance	0.50 m
Plate pitch	0.0035 m

Calculate:

a. Mean channel flow gap
b. One channel flow area
c. Channel equivalent diameter
d. Projected plate area
e. Effective surface area per plate

10.2. A gasketed-plate heat exchanger will be used for heating city water (R_{fc} = 0.00006 $m^2 \cdot K/W$) using the waste water available at 90°C. The vertical distance between the ports of the plate is 1.60 m and the width of the plate is 0.50 m with a gap between the plates of 6 mm. The enlargement factor is given by the manufacturer as 1.17 and the chevron angle is 50°. The plates are made of titanium (k = 20 W/m · K) with a thickness of 0.0006 m. The port diameter is 0.15 m. The cold water enters the plate heat exchanger at 15°C and leaves at 45°C at a rate of 6 kg/s and it will be heated by the hot water available at 90°C, flowing at a rate of 12 kg/s. Considering single-pass arrangements for both streams, calculate:

a. The effective surface area and the number of plates of this heat exchanger
b. The pressure drop for both streams

10.8. A one-pass counter-current flow heat exchanger has 201 plates. The exchanger has a vertical port distance of 2 m and is 0.6 m wide, with a gap between the plates of 6 mm. This heat exchanger will be used for the following process: cold water from the city supply with an inlet temperature of 10°C is fed to the heat exchanger at a rate of 15 kg/s and will be heated to 75°C with wastewater entering at an inlet temperature of 90°C. The flow rate of hot water is 30 kg/s, which is distilled water. The other construction parameters are as given in Problem 10.2. There is no limitation on the pressure drop. Is this heat exchanger suitable for this purpose (larger or smaller)?

10.9. In Example 10.1, a new mass flow rate is provided as 90 kg/s. Assume that the inlet temperatures of the hot and cold fluid are changed to 120°C and 20°C respectively. If the flow arrangement is single pass for both streams, use the ε-NTU method to calculate the outlet temperatures for the specific heat exchanger. There may be a constraint on the outlet temperature of the cold stream, therefore this type of analysis is important to see if the required outlet temperature of the cold stream is satisfied.

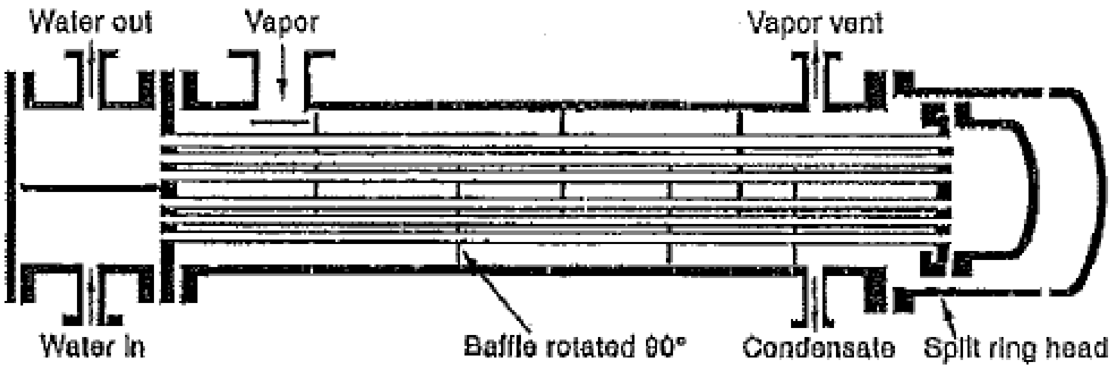
Design Project 10.3

Design a Heat Exchanger for An Open Heart Operation — Blood Heat Exchanger

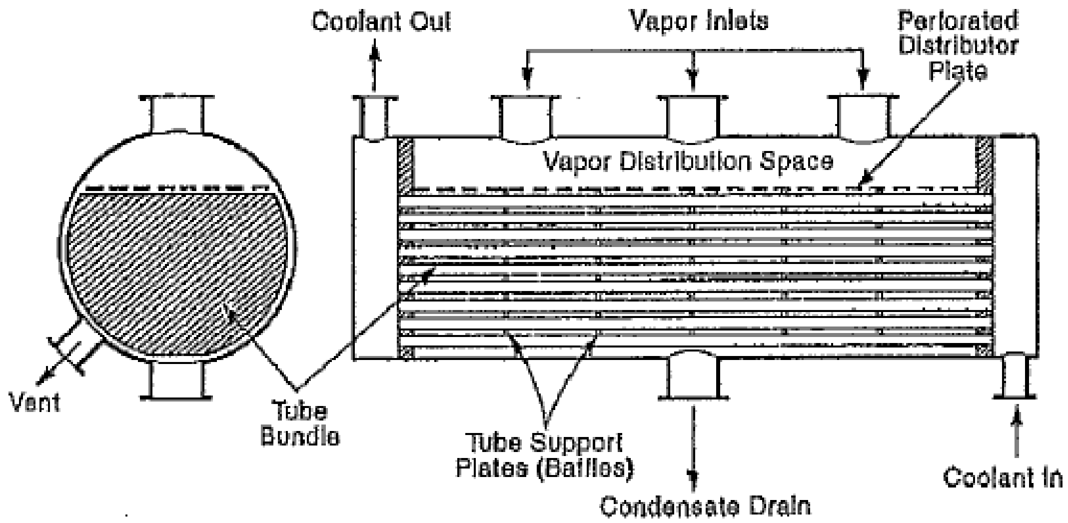
The objective of a blood heat exchanger is to shorten the time normally required to cool a patient's blood prior to open-heart surgery. Since blood behaves in a non-Newtonian manner, necessary assumptions are made in order to complete the calculations. The material in contact needs to have the smoothest possible surface, therefore a stainless steel with a specific non-wetting silicon resin shell-and-tube heat exchanger is chosen for the design. For a sufficient design, the pressure drop must be minimal due to the fragile nature of blood, and the heat transfer rate must be high due to the required rapid temperature change. This design project will take into account thermal, dynamic heat transfer and a parametric analysis for a blood heat exchanger. Assume that the patient's blood will be cooled from 37°C to 27°C by cooling water available at 15°C. The mass flow rate of blood is 0.03 kg/s and the properties at 32°C are: ρ = 1055 kg/m^3, μ = 0.00045 kg/m · s, k = 0.426 W/m · K, Pr = 3.52, and c_p = 3330 J/kg · K. Cooling water may be available at 0.20 kg/s. One can directly select a gasketed-plate heat exchanger, or a comparative study as in design Projects 10.1 and 10.2 can be carried out.

Condensers and Evaporators

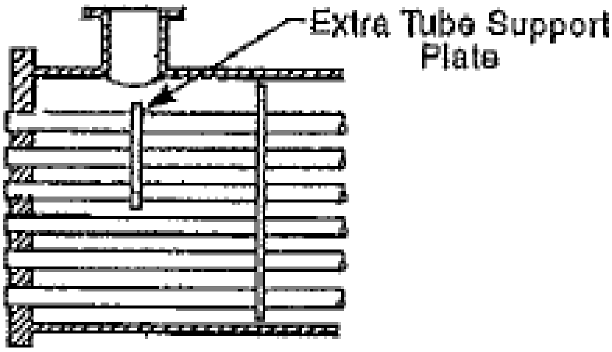
Horizontal shell side condensers



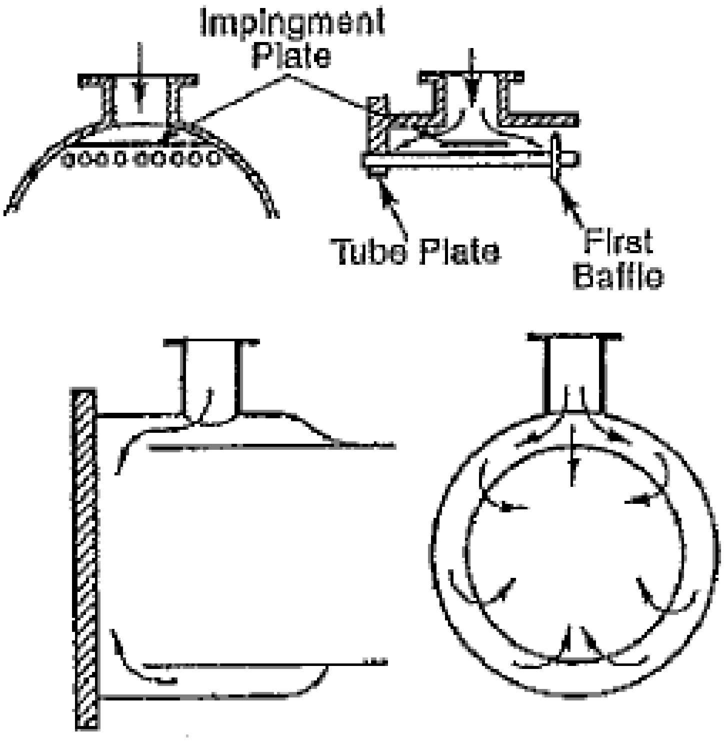
Horizontal shell-side condenser. (From Mueller, A. C. [1983], *Heat Exchanger Design Handbook*, Hemisphere, Washington, D.C.)



Main features of a crossflow condenser (TEMA X-type). (From Butterworth, D. [1991], *Boilers, Evaporators and Condensers*, John Wiley & Sons, New York, 571. Butterworth, D. [1988], *Two-Phase Flow Heat Exchangers*, Kluwer Publishers, The Netherlands.)

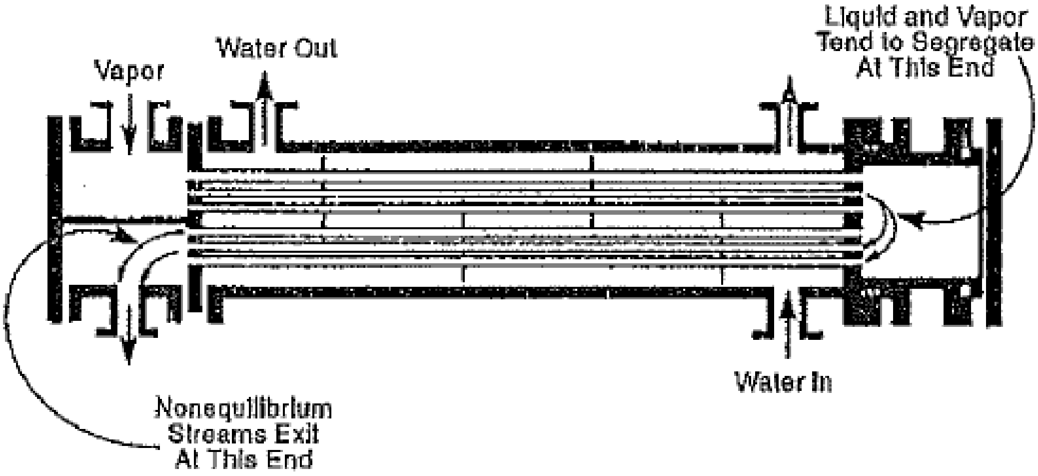


Extra tube support plate to help prevent vibration of tubes near the inlet nozzle.



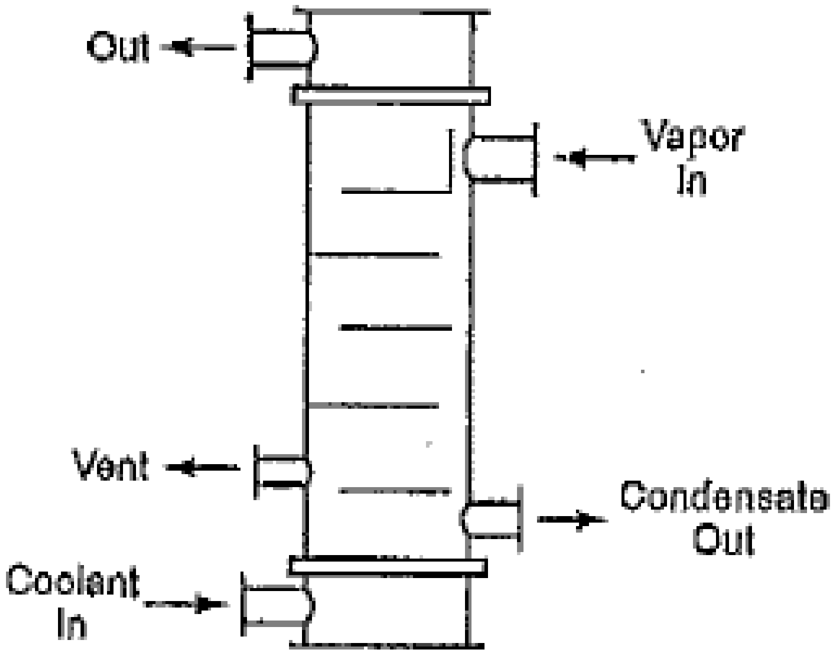
Impingement plate and vapor belt. (From Butterworth, D. [1991], *Boilers, Evaporators and Condensers*, John Wiley & Sons, New York, 571. Butterworth, D. [1988], *Two-Phase Flow Heat Exchangers*, Kluwer Publishers, The Netherlands.)

Horizontal tube side condensers



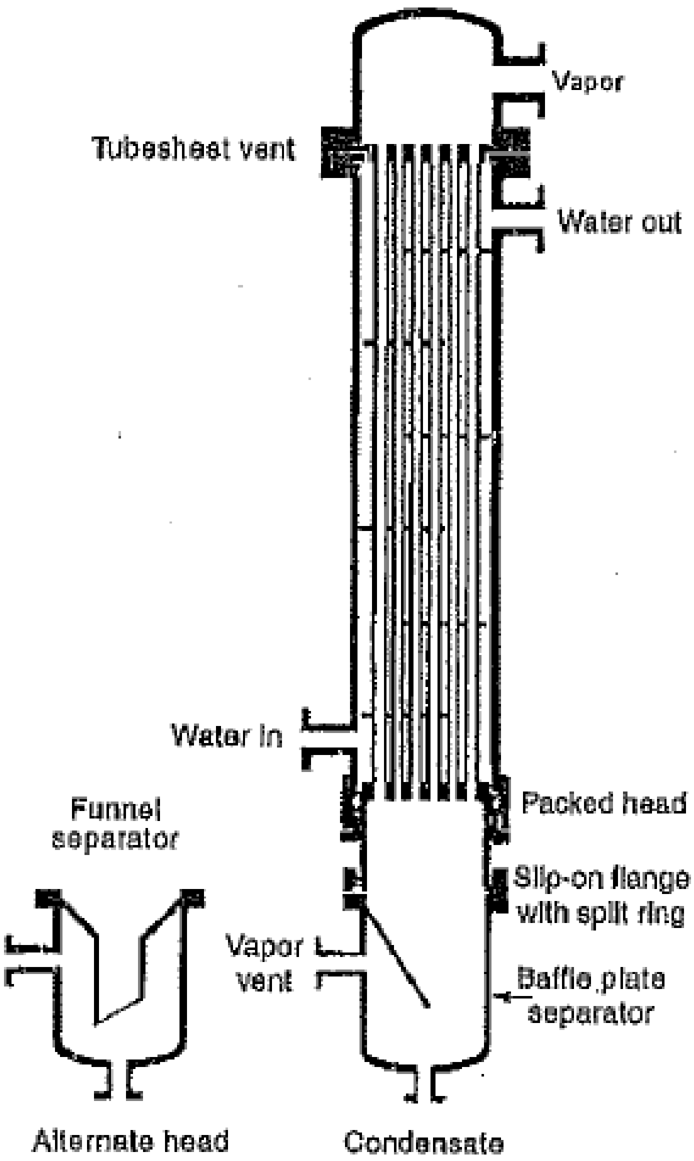
Horizontal in-tube condenser. (From Mueller, A. C. [1983], *Heat Exchanger Design Handbook*, Hemisphere, Washington, D.C. Breber, G. [1988], *Heat Transfer Equipment Design*, Hemisphere, Washington, D.C., 477.)

Vertical shell side condensers

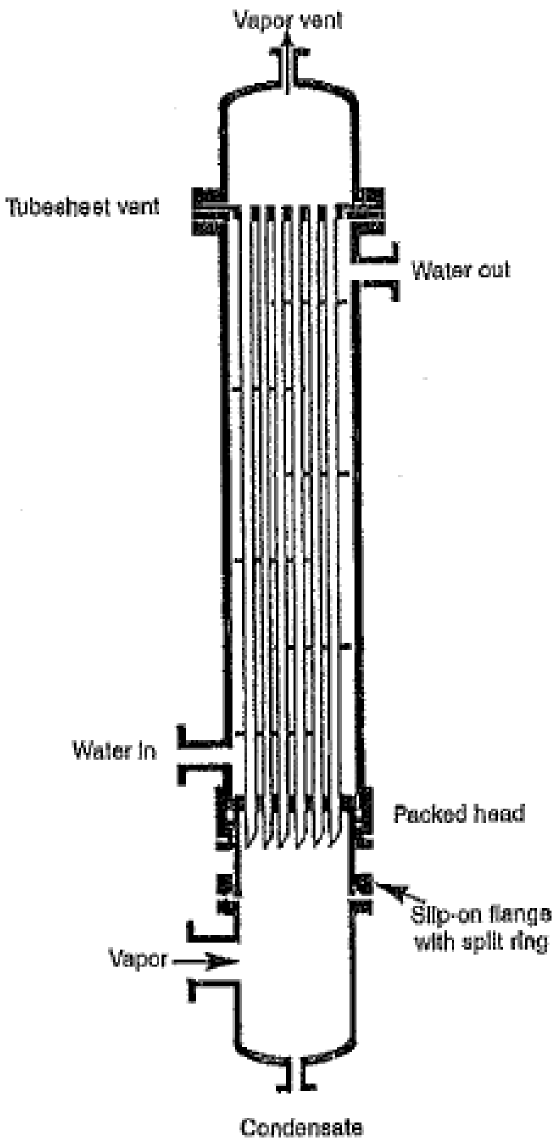


Vertical tube side condensers

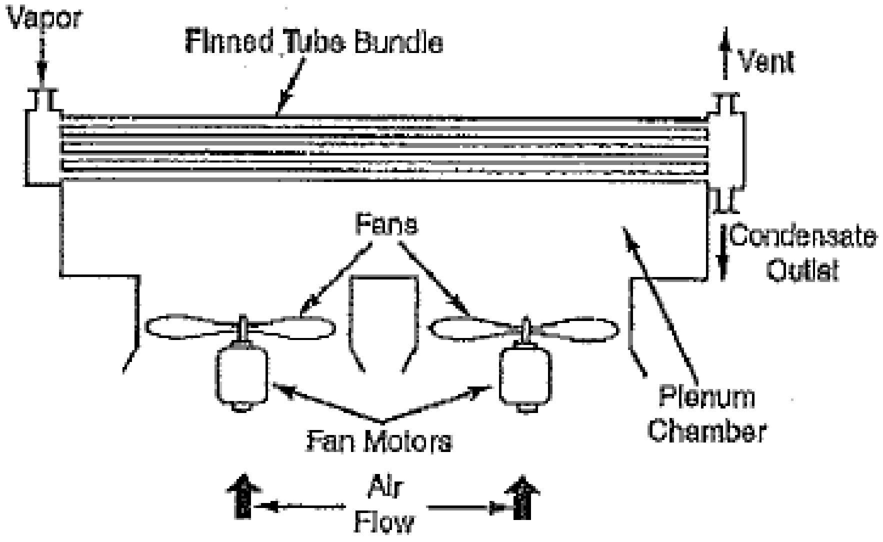
1. Vertical in-tube down flow condenser



2. Reflux condenser

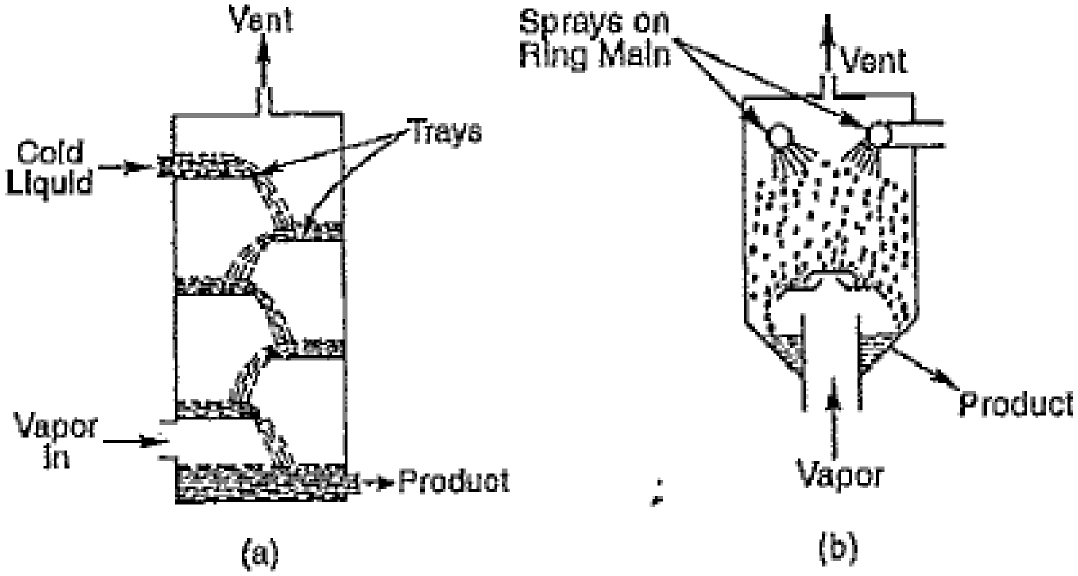


Air cooled condensers



Forced-draft, air-cooled exchanger used as a condenser.

Direct contact condensers



Direct contact type heat exchangers: (a) tray condenser; (b) spray condenser. (Adapted from Butterworth, D. [1988] In *Two-Phase Flow Heat Exchangers: Thermal-Hydraulic Fundamentals and Design*, Kluwer, Dordrecht, The Netherlands.)

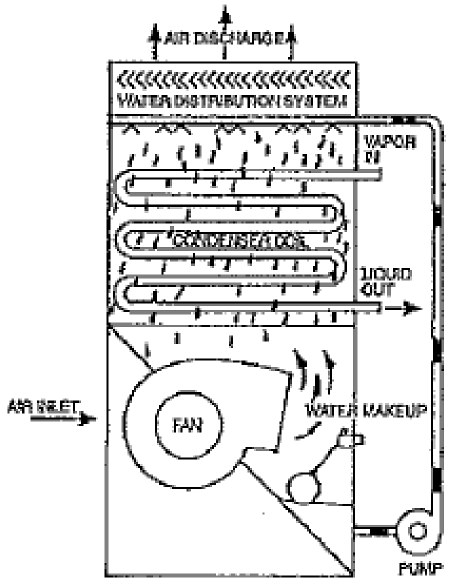
Failure of condenser operation

1. The tubes may be more fouled than expected – a problem not unique to condensers.
2. The condensate may not be drained properly, causing tubes to be flooded. This could mean that the condensate outlet is too small or too high.
3. Venting of non-condensable gases may be inadequate.
4. The condenser was designed on the basis of end temperatures without noticing that design duty would involve a temperature cross in the middle of the range.
5. Flooding limits have been exceeded for condensers with backflow of liquid against upward vapor flow.
6. Excessive fogging may be occurring. This can be problem when condensing high molecular weight vapors in the presence of non-condensable gases.
7. Severe maldistribution in parallel condensing paths is possible, particularly with vacuum operation. This occurs because there can be two flow rates which satisfy the imposed pressure drop.

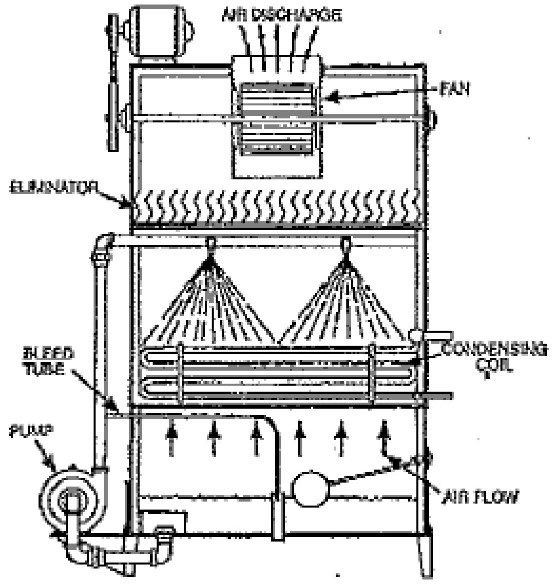
Condensers for refrigeration and air conditioning

1. Water-cooled condensers
    - Horizontal shell-and-tube
    - Vertical shell-and-tube
    - Shell-and-coil
    - Double pipe
2. Air-cooled condensers
3. Evaporative condensers

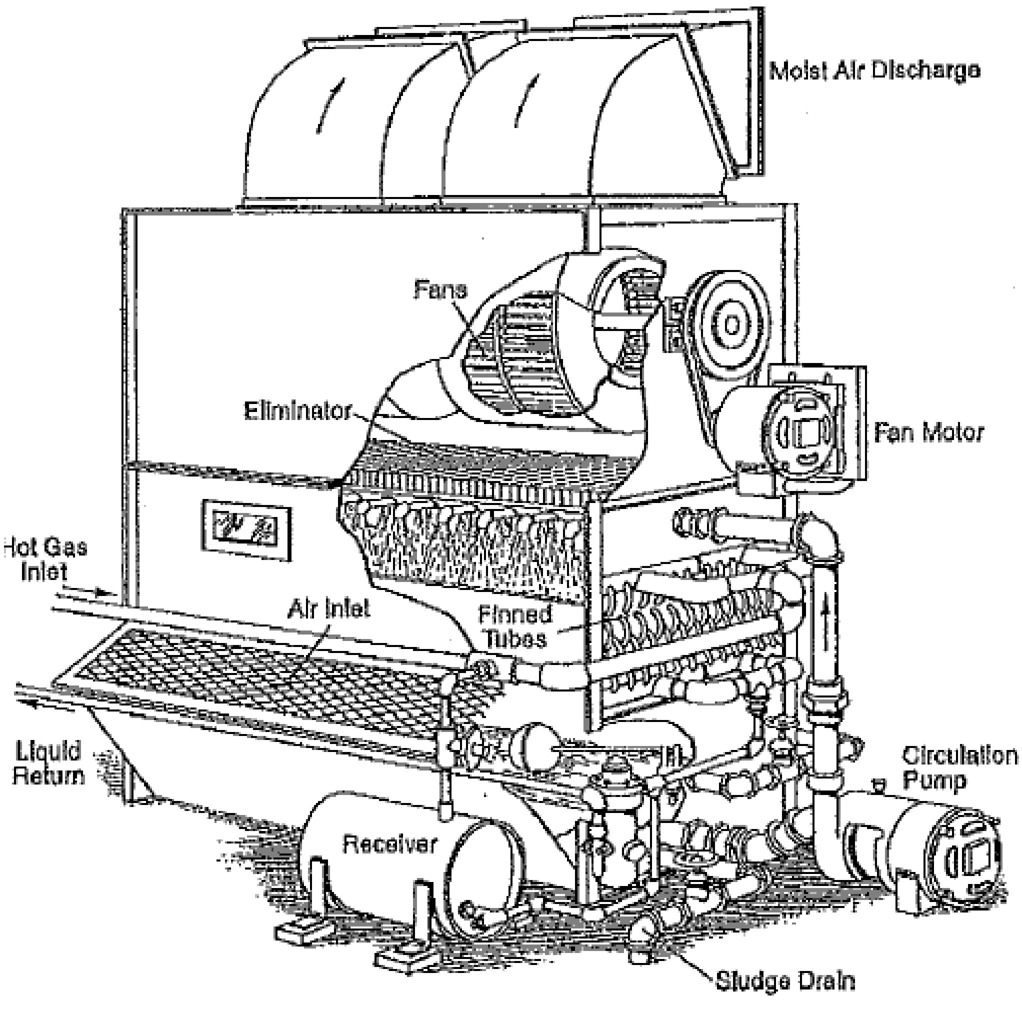
Evaporative condensers



Blow-Through-Type

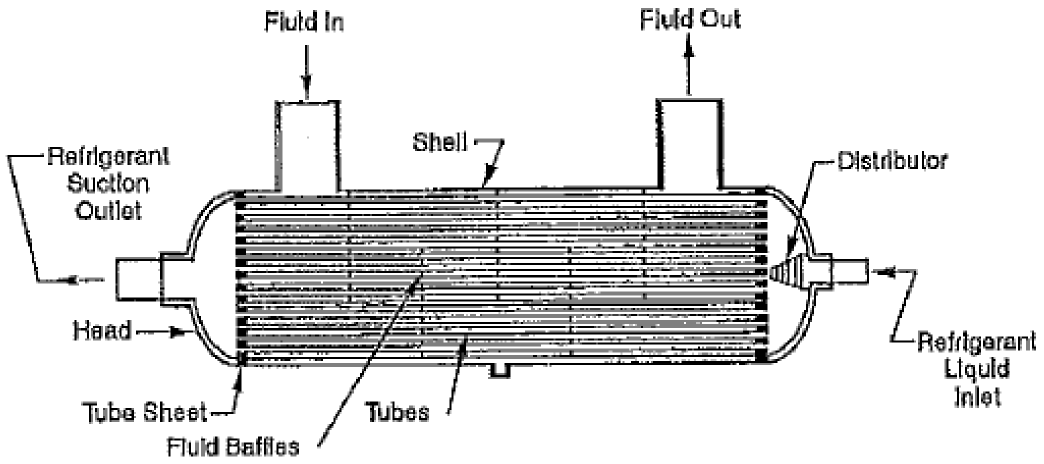


Draw-Through-Type

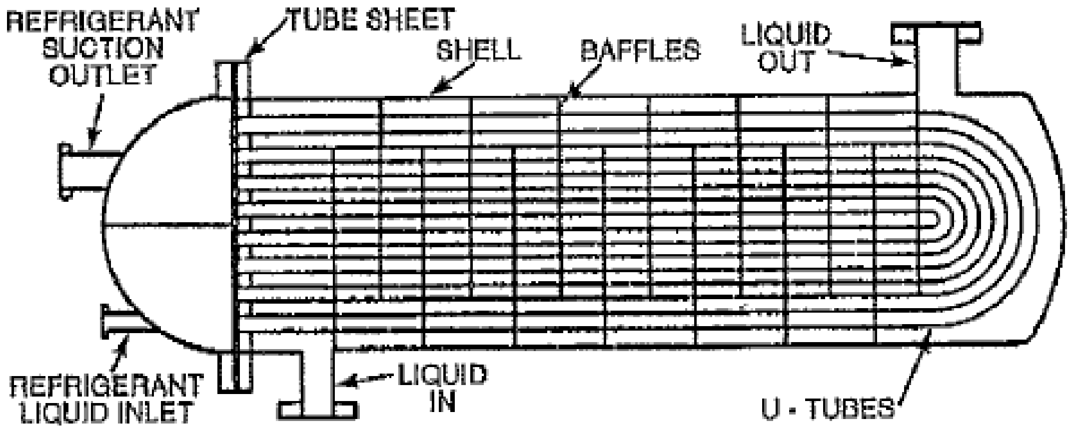


Evaporators for refrigeration and air conditioning

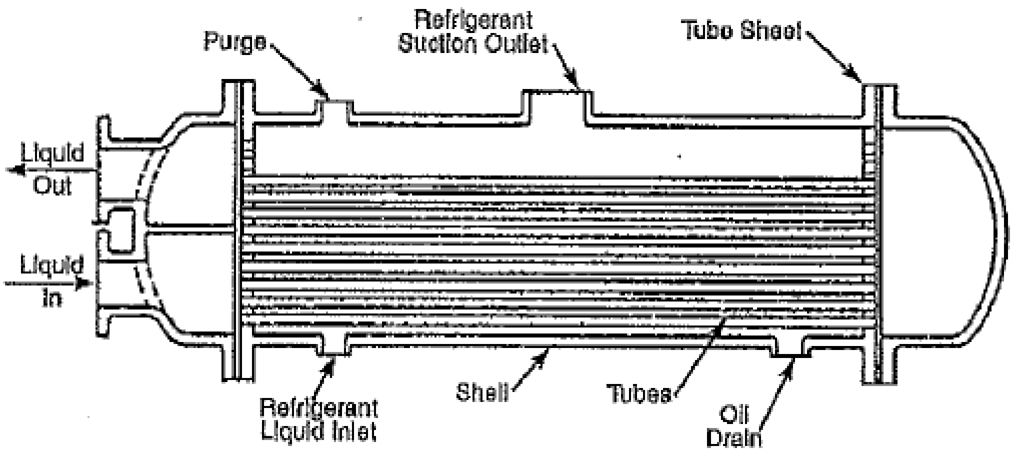
- **Water cooling evaporators (Chillers)**



Direct (straight-tube type) expansion evaporator.

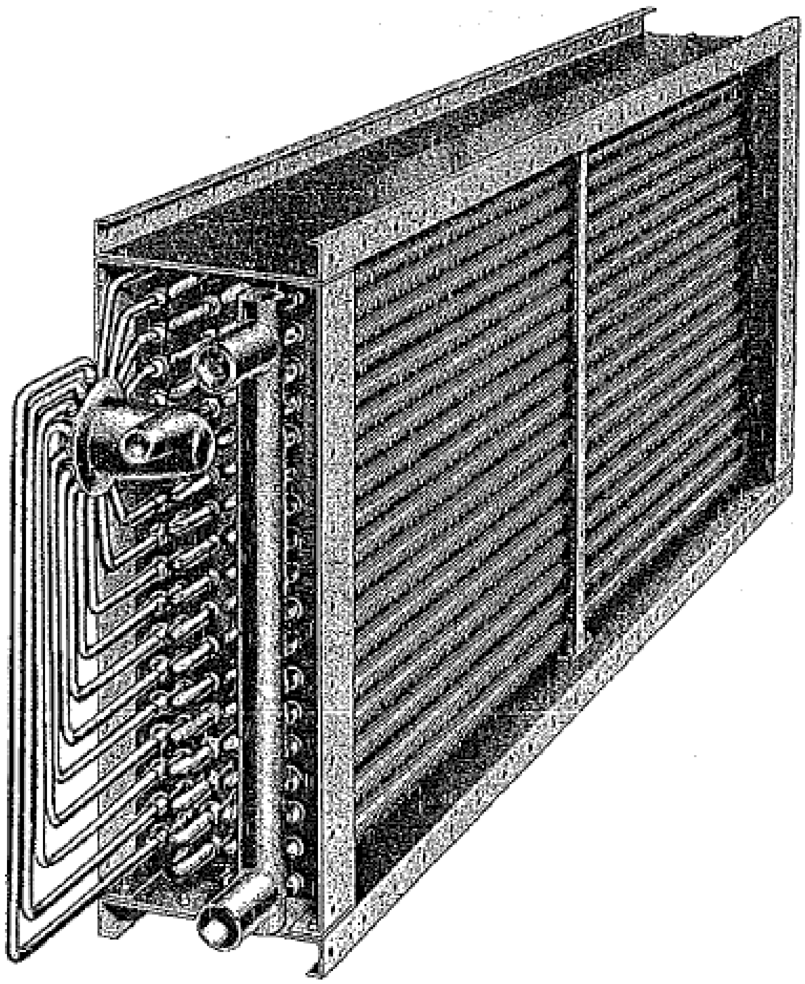


Direct (U-type) expansion evaporator.

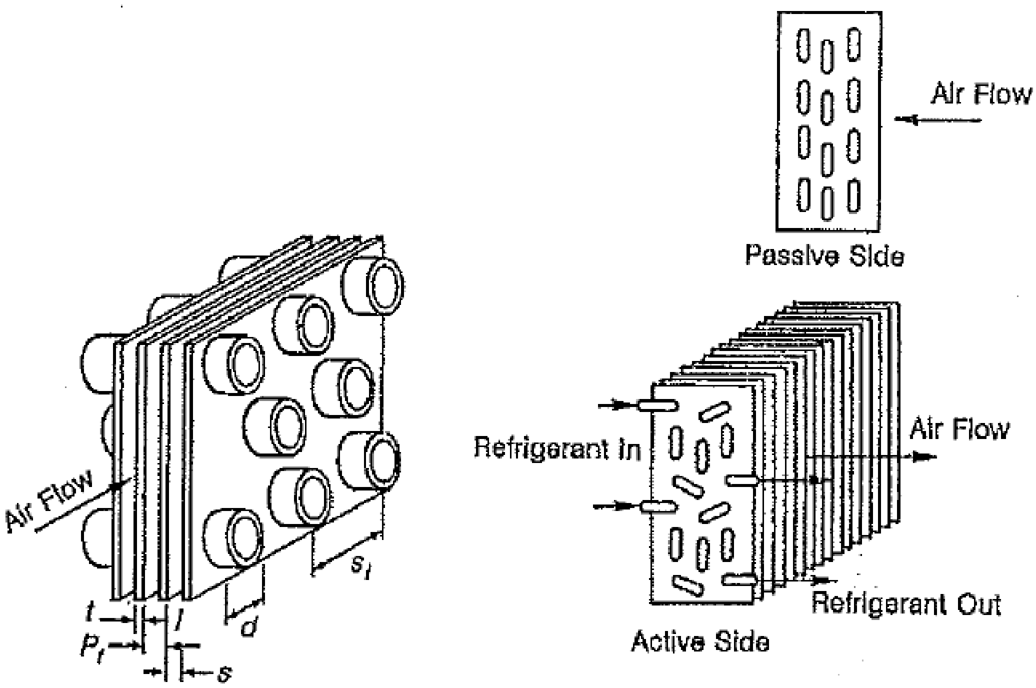


Flooded shell-and-tube evaporator.

- **Air cooling evaporators (Air coolers)**



Dry expansion coil-finned tube bank for an air-conditioning system.



Dry expansion coil (air evaporator)

Standards for evaporators and condensers

- Air-conditioning and Refrigerating Institute (ARI) Standards
- American Society for Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standards

Problems

11.2. In a power plant, a shell-and-tube type heat exchanger is used as a condenser. This heat exchanger consists of 20,000 tubes and the fluid velocity through the tubes is 2 m/s. The tubes are made of admiralty metal and have 18 BWG, 7/8" OD. The cooling water enters at 20°C and exits at 30°C. The average temperature of the tube walls is 55°C and the shell-side heat transfer coefficient is 4000 $W/m^2 \cdot K$. Fouling on both sides is neglected. Making acceptable engineering assumptions, perform the thermal design of this condenser.

11.3. A surface condenser is designed as a two-tube-pass shell-and-tube type heat exchanger at 10 kPa (h_{fg} = 2007.5 kJ/kg, T_s = 45°C). Coolant water enters the tubes at 15°C and leaves at 25°C. The coefficient is 300 $W/m^2 \cdot K$. The tubing is thin walled, 5 cm inside diameter, made of carbon-steel, and the length of the heat exchanger is 2 m. Calculate the:

a. LMTD

b. Steam mass flow rate

c. Surface area of the condenser

d. Number of tubes

e. Effectiveness, ε

Design Project 11.1

Design of a Compact Air-Cooled Refrigerant Condenser

Specifications:

Cooling load (heat duty):	Q = 125 kW
Refrigerant:	R-134A condensing inside tubes at T_s = 37°C (310 K)
Coolant:	Air
	Inlet temperature, T_{c1} = 18°C
	Inlet temperature, T_{c1} = 26°C
	Mean pressure P = 2 atm (0.2027 MPa)
Heat transfer matrix:	to be selected from Chapter 9

At least two different surfaces must be studied for thermal and hydraulic analysis and then compared; the primary attention must be given to obtaining the smallest possible heat exchanger. A parametrical study is expected to develop a suitable final design. The final design will include materials selection, mechanical design, technical drawings, and the cost estimation.

Design Project 11.3

Water-Cooled Shell-and-Tube Type Freon Condenser

Specifications:

Cooling load:	200 kW
Refrigerant:	Freon-134A
Temperature:	27°C
Mean pressure:	0.702 MPa
Coolant:	Water
Inlet temperature:	18°C
Outlet temperature:	26°C
Mean pressure:	0.4 MPa
Heat transfer matrix:	to be selected from Chapter 9

Condensation will be on the shell side of the heat exchanger. A shell type must be selected. The size will be estimated first and then it will be rated, as described in Chapter 8. The final report should include thermal-hydraulic analysis, optimization, materials selection, mechanical design, drawings, sizing, and cost estimation.

References

[1] F. P. Incorpera, D. P. DeWitt, T. L. Bergman, A. S. Lavine, “Fundamentals of Heat and Mass Transfer-6th Edition”, John Wiley, 2006, New York.

[2] F. P. Incorpera, D. P. DeWitt, T. L. Bergman, A. S. Lavine, “Fundamentals of Heat and Mass Transfer-7th Edition”, John Wiley, New, 2011, New York.

[3] R. K. Shah, D. P. Sekulic, Fundamental of Heat Exchanger Design, John Wiley, 2003, New York.

[4] S. Kakaς, H. Liu, Heat Exchangers: Selection, Rating, and Thermal Design, CRC Press, 2002, Florida.