21

CONVECTIVE HEAT-TRANSFER COEFFICIENT

Introduction

In most transport processes
heat transfer in fluids is accompanied by some form of fluid motion
so that the heat transfer does not occur by conduction alone.

Forced convection:	fluid motion arises principally from a pressure gradient caused by a pump or blower
Natural convection:	fluid motion arises only from density differences associated with the temperature field

Heat transfer in fluids

Heat transfer in fluids

- conduction
- convection
  - Forced convection
  - Natural convection
  - Mixed convection

Introduction

Whether the heat-transfer mechanism is natural or forced convection, the fluid motion can be described by the equations of fluid mechanics.

@ low velocities → the flow is laminar throughout the system
@ high velocities → laminar near the heating surface
+ turbulent some distance away

Although fluid velocities in natural convection are usually lower than in forced convection, it is incorrect to think of natural convection as causing only laminar flow; turbulence dose occur when critical Reynolds number for the system is exceeded.

Analytical Solution

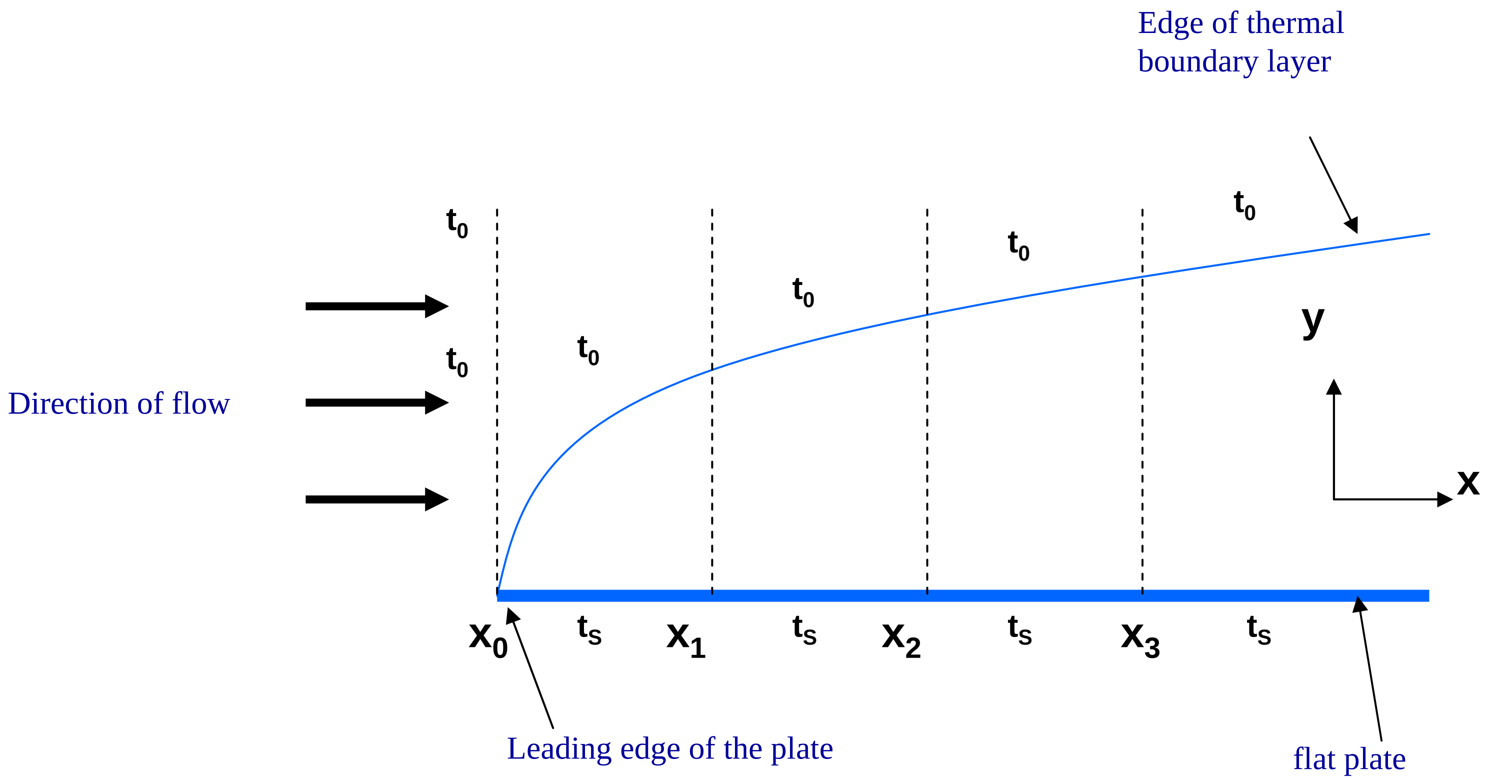
All problems of convective heat transfer can be expressed in terms of the differential mass, energy, and momentum balances. However, the mathematical difficulties connected with integration of these simultaneous nonlinear partial differential equations are such that analytical solutions exist only for simplified cases.

One of the problems simplest to deal with analytically is that of

Heat transfer between a fluid and a flat plate when the fluid is flowing parallel to the plate. (Fig. 21-1)

Heat transfer between a fluid and a flat plate when the fluid is flowing parallel to the plate.

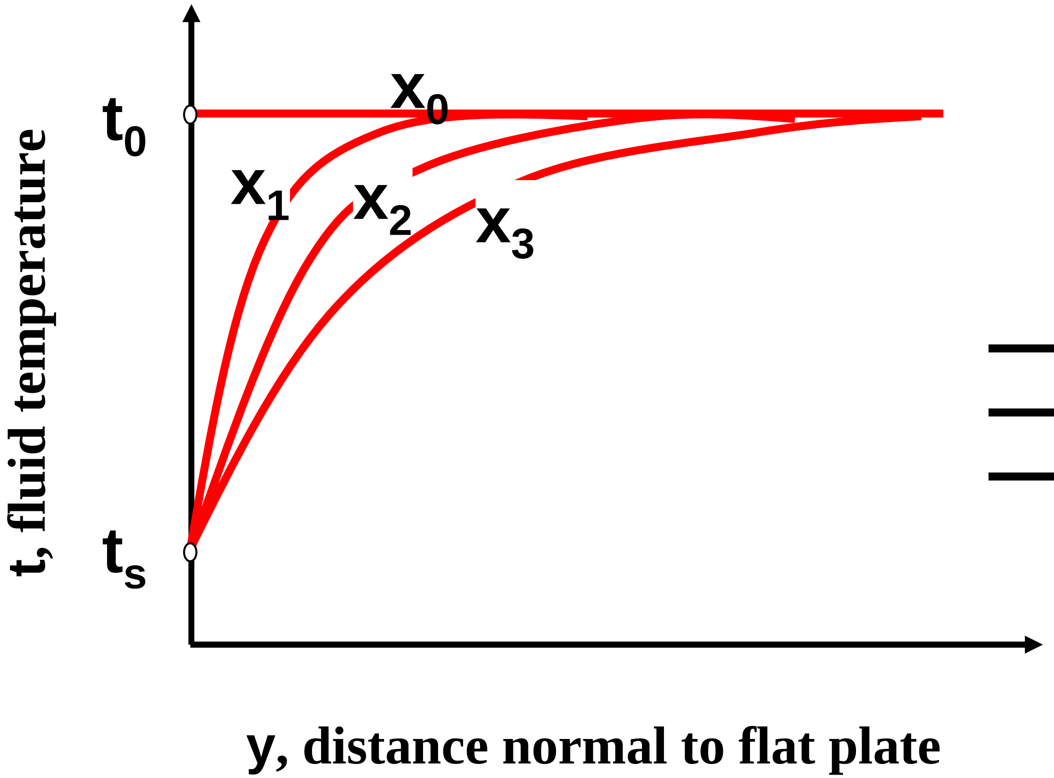
(Fig. 21-1)

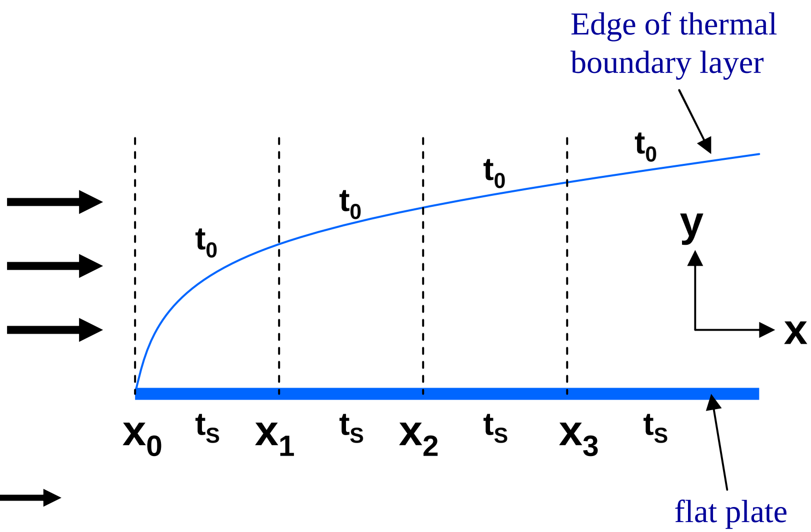


$t_0 > t_s$

Fig. 21-1
Development of a thermal boundary layer for flow over a flat plate

Temperature profiles in a developing thermal boundary layer on a flat plate

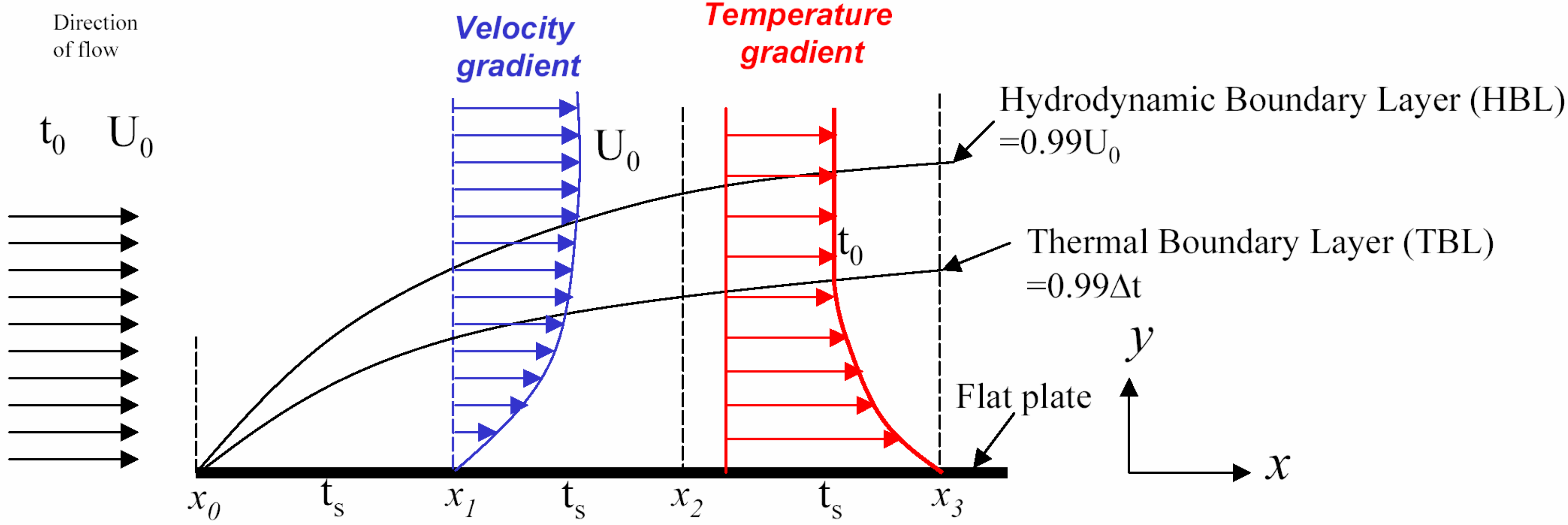




Development of thermal boundary layer for flow over a flat plate

Blasius flow (laminar flow)

$$Re_x = \frac{\rho u_0 x}{\mu} \leq 5\times 10^5 \text{ (Laminar flow)}$$



$$Pr = \frac{\nu}{\alpha} > 1 \cdots HBL > TBL$$

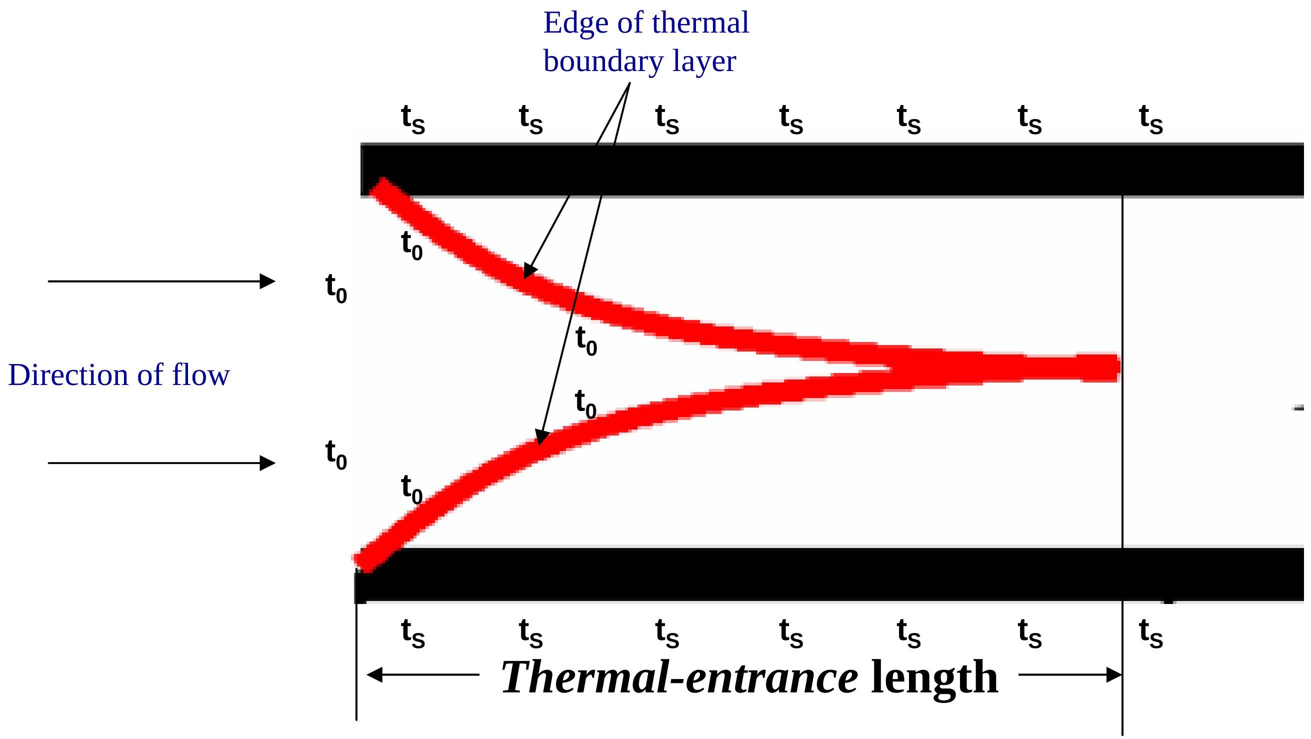
$$Pr = \frac{\nu}{\alpha} < 1 \cdots HBL < TBL$$

$$Pr : \text{water} = 6.5, \text{air} = 0.7, Hg = 0.025$$

$$\text{water } (350^{o} F) = 1$$

Heat transfer system

Heat is transferred between a fluid and the wall of a pipe

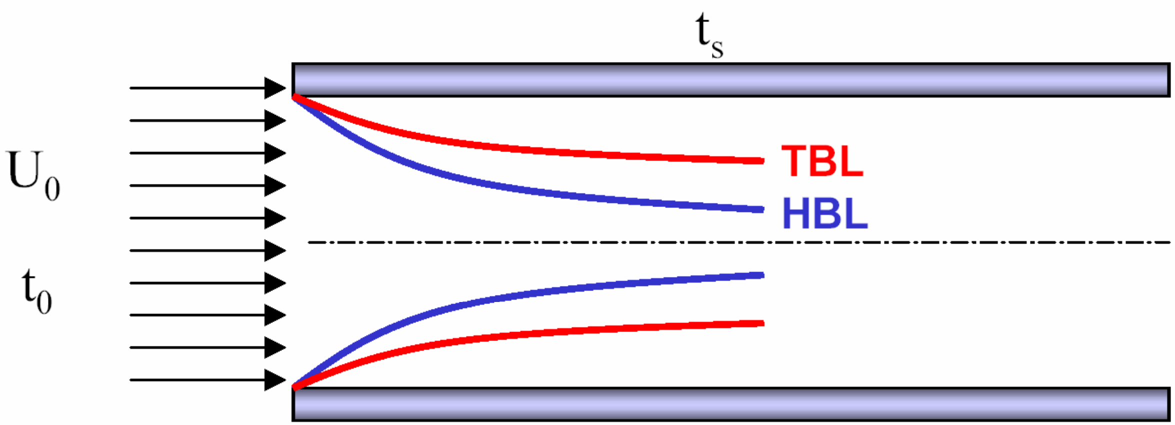


Temperature profile become flatter.

Development of thermal boundary layer for flow in a pipe

Entrance region

$$\mathrm{Re}_D = \frac{\rho u_0 x}{\mu} \le 2100 \text{ (Laminar flow)}$$

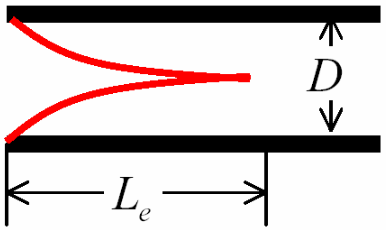


Thermal Entrance Length

$$HBLT: \quad \frac{L_e}{D} \cong 0.05\,\mathrm{Re}_D$$

$$TBLT: \quad \frac{L_e}{D} \cong 0.05\,\mathrm{Re}_D \cdot \mathrm{Pr}$$

$$in\ tubulent\ flow: \frac{L_e}{D} \cong 40 \sim 50$$



Temperature profiles near the entrance of a pipe

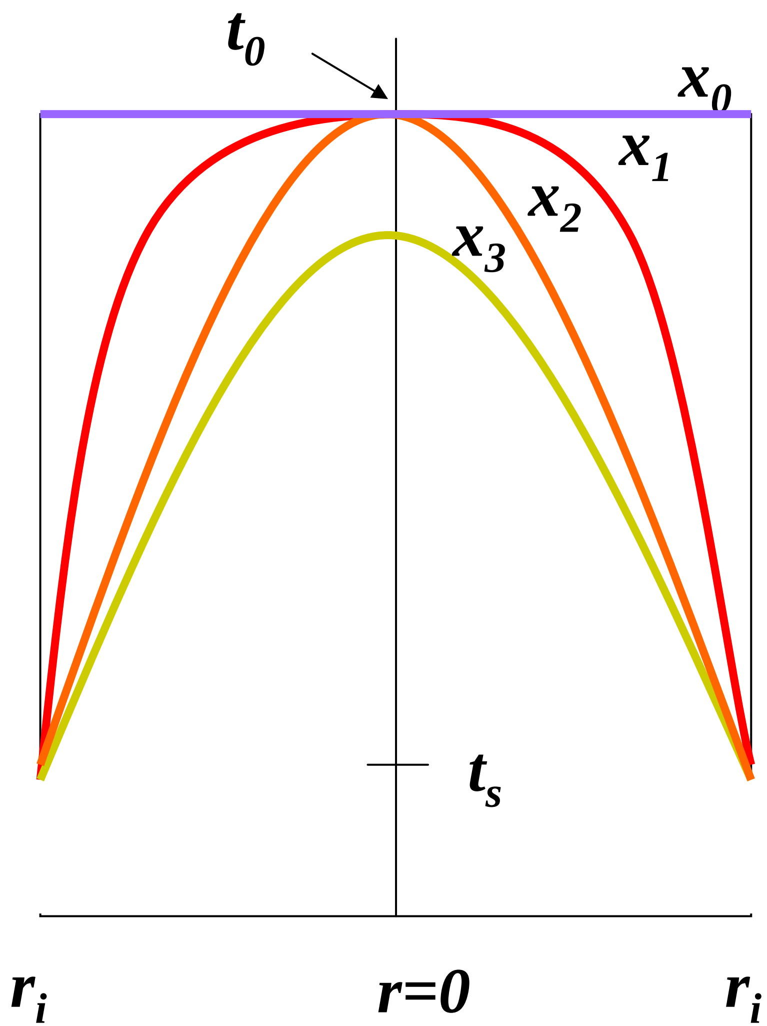


Figure 21-4

Temperature profiles vs Velocity profiles

The temperature profile at a point in a flow system is influenced by the velocity profile.

The velocity profile is influenced by the temperature profile. (The velocity profile of an isothermal system may differ substantially from the velocity of a system in which heat is being transferred.)

Temperature profile in a fluid

Temperature profile in a fluid is influenced by the velocity profile

Differential energy balance

$$\underbrace{u_x \frac{\partial t}{\partial x} + u_y \frac{\partial t}{\partial y} + u_z \frac{\partial t}{\partial z}}_{\text{velocity profile}} + \frac{\partial t}{\partial \theta} = \frac{k}{\rho C_p} \underbrace{\left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2} \right)}_{\text{Temperature profile}} \quad (8\text{-}11)$$

velocity profile **Temperature profile**

Velocity profile is influenced by the temperature profile in a fluid

Navier-Stokes equation

$$u_x \frac{\partial u_x}{\partial x} + u_y \frac{\partial u_y}{\partial y} + u_z \frac{\partial u_z}{\partial z} + \frac{\partial u_x}{\partial \theta} = g_c X - \frac{g_c}{\rho} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 u_x}{\partial x^2} + \frac{\partial^2 u_x}{\partial y^2} + \frac{\partial^2 u_x}{\partial z^2} \right)$$

↑ **Note that viscosity is temp dependent !**

Simplification

The non-isothermal velocity profile can be used in solving the differential momentum and energy balances for a non-isothermal system.

This simplification may introduce a serious error when the viscosity of the fluid is strongly dependent on temperature.

Frequently the temperature gradient is greatest near the wall, and it is in this region that the velocity gradient is also greatest. The effect of temperature on the viscosity of the fluid at the wall may therefore have a pronounced effect on both the velocity and temperature profiles of the system.

Individual Heat-transfer Coefficients

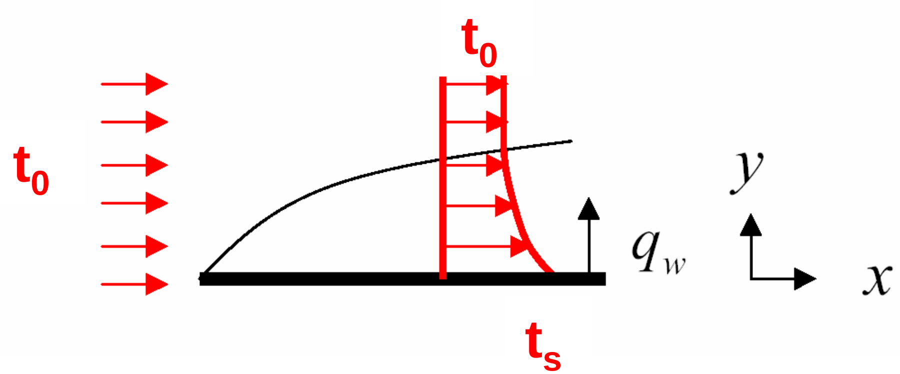
Fourier's law

At the surface, there is no fluid motion and energy transfer can only by conduction

Newton's law of cooling

$$q_w = hA(t_s - t_m) = -k_f A \left.\frac{\partial t}{\partial y}\right|_{y=0}$$

- Rate of Heat transfer: q_w
- Heat transfer coefficient: h
- Surface temperature: t_s
- bulk temperature: t_m



Individual heat transfer coefficient, h

h **is a function of**

the properties of the fluid,
the geometry and roughness of the surface, and
the flow pattern of the fluid.

Evaluation of individual heat transfer coefficient, *h*

Several methods are available for evaluating *h*

Laminar flow systems → Analytical methods

Turbulent systems → Integral methods
→ Mixing-length theory
→ Dimensional analysis

Evaluation of individual heat transfer coefficient, *h*

$$dq = k\,dA\left(\frac{dt}{dy}\right)_{y=0} = h(t_0 - t_s)\,dA$$

$$h = \frac{k}{t_0 - t_s}\left(\frac{dt}{dy}\right)_{y=0} = k\left\{\frac{d\left[(t - t_s)/(t_0 - t_s)\right]}{dy}\right\}_{y=0}$$

Velocity of flow past the heated surface ↑
Temperature gradient at the wall ↑
Heat transfer coefficient, *h* ↑

Individual Heat-transfer Coefficients

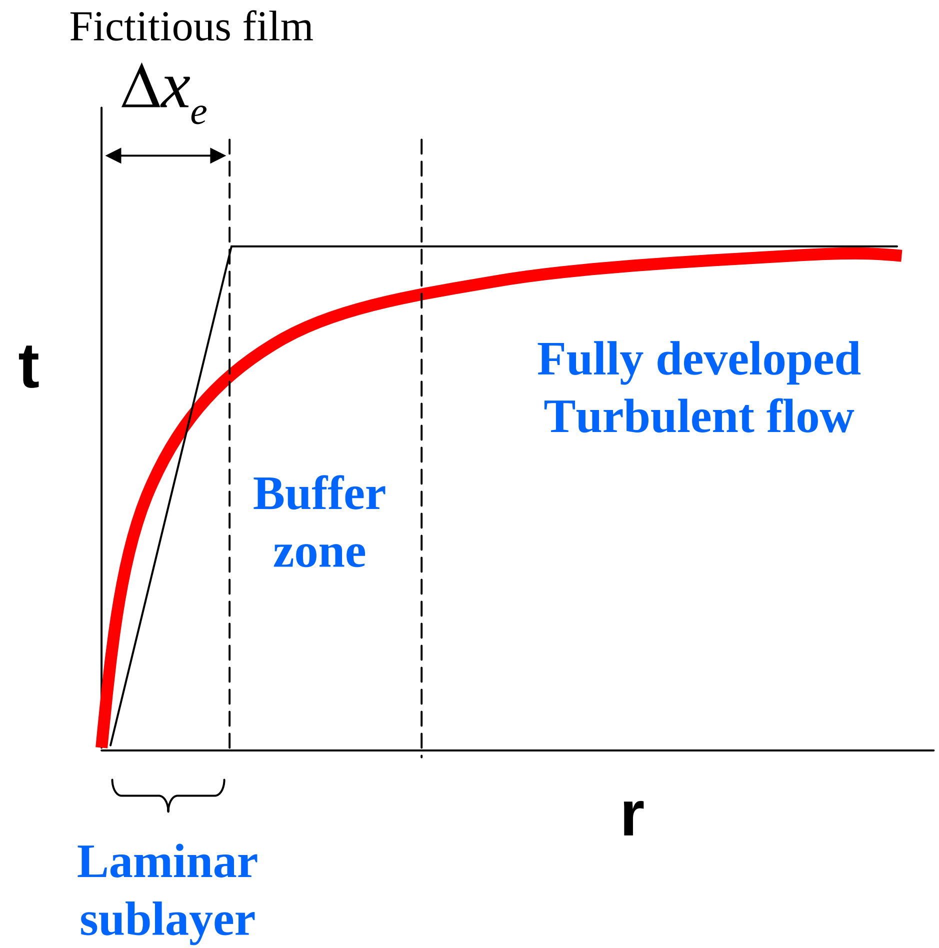
$$h = \frac{-k_f \left.\frac{\partial t}{\partial y}\right|_{y=0}}{t_s - t_0}$$

- The thermal boundary layer strongly influence the wall temperature gradient $\partial t / \partial y|_{y=0}$
- The wall temperature gradient will determine the rate of heat transfer across the boundary layer
- Since (t_s-t_0) is a constant, independent of *x*, while δ_t increases with increasing *x*, temperature gradients in the boundary layer must decrease with increasing *x*.
- Accordingly, the magnitude of $\partial t / \partial y|_{y=0}$ decreases with increasing *x*, and if follows that q and h decreases with increasing *x*.

Individual film coefficient, h

$$h = \frac{k}{\Delta x_e}$$

If the resistance to heat flow is through of as existing only in a laminar film, h is $k/\Delta x_e$ where Δx_e is the equivalent thickness of a stationary film just thick enough to offer the resistance corresponding to observed value of h. Since there is often an appreciable resistance in the turbulent core and since the stationary film has not even an approximate physical counterpart in laminar flow, boiling and radiation, we shall refer *h* as an individual heat transfer coefficient.



Individual Heat-transfer Coefficients

Definition of *h*

Flow through a heated conduit

$$q = hA(t_s - t_b)$$

$$t_b = \frac{1}{Au_{av}} \int_A u_x t dA$$

Mixing cup temp

Natural convection

$$q = hA(t_s - t_\infty)$$

t_∞=Temp far from surface

Condensation

$$q = hA(t_{sv} - t_s)$$

t_{sv}=Temp of sat vapor

Boiling fluid

$$q = hA(t_s - t_{sl})$$

t_{sl}=Temp of sat liquid

Radiation

$$q = h_r A(t_{s1} - t_{s2})$$

Individual Heat-transfer Coefficients

Convection in Circular pipes, Laminar : Eq. (22-40) pp352

Convection in Circular pipes, turbulent : Eq. (24-4) pp384

Convection from spheres : Eq. (24-18) pp395

Convection from a plane surface : Eq. (24-22)- Eq.(24-26)

Boiling fluid : Eq. (25-5) pp412

Condensation on vertical tubes : Eq. (25-29) pp420

Condensation on horizontal tubes : Eq. (25-32) pp421

Individual heat transfer coefficients

	h, Btu/(h)(ft^2)(^oF)
Steam, dropwise condensation	5,000-20,000
Steam, film-type condensation	1,000-3,000
Water, boiling	300-9,000
Organic vapors, condensing	200-400
Water, heating	50-3,000
Oils, heating or cooling	10-300
Steam, superheating	5-20
Air, forced convection	2-15
Air, natural convection	0.5-2

Multiply values of coefficients given in table by 5.678 to get W/(m^2)(K)

Example 21-1

The analysis of unsteady-state heating or cooling of solid objects can be often be simplified by assuming that the resistance to heat conduction inside the solid is negligible compared to the convective heat-transfer resistance in the surrounding fluid.

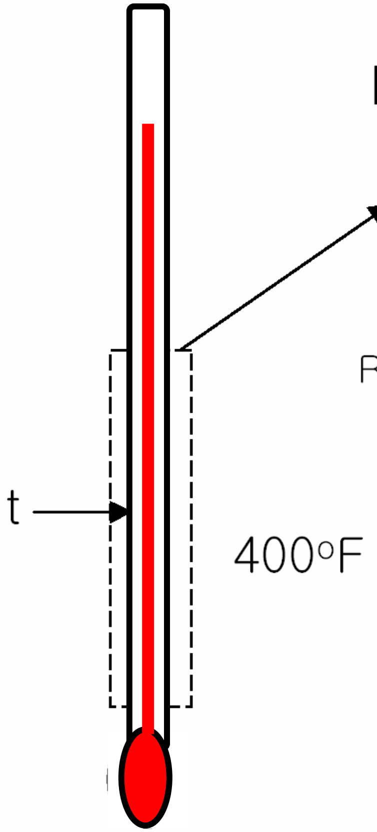
The circumstances necessary to justify this assumption are a high thermal conductivity for the solid and a low convective heat transfer coefficient in the adjacent fluid. A metallic object being heated or cooled in air often constitute such a system.

Example 21-1

To illustrate, we shall calculate the time required for a mercury thermometer initially at 70°F, placed in an oven at 400°F, to reach 279°F. The thermometer bulb has a diameter of 0.24 in and will be assumed to be adequately represented as an infinitely long cylinder with negligible resistance to heat transfer either in the glass which is very thin or in the mercury, which has an adequately high thermal conductivity [$\boldsymbol{k}$=5.6 Btu/(h)(ft)(°F) at 140°F]. The mean convective heat-transfer coefficient between the outside of the thermometer and the air in the oven will be taken as $\boldsymbol{h}$=2 Btu/(h)(ft^2)(°F).

Example 21-1: time required for a mercury thermometer

thermometer bulb ➡ infinite long cylinder with negligible resistance to heat transfer either in the glass or in the mercury



Heat balance on the thermometer bulb

$$hA(400 - t) = \rho C_p V \frac{dt}{d\theta}$$

Rearrange and integrate

$$\frac{dt}{400 - t} = \frac{hA}{\rho C_p V} d\theta \quad \Rightarrow \quad \ln \frac{400 - 279}{400 - 70} = -\frac{hA\theta}{\rho C_p V}$$

For a cylinder,

$$\frac{A}{V} = \frac{\pi DL}{\pi D^2 L / 4} = \frac{4}{D}$$

Example 21-1: time required for a mercury thermometer

$$\theta = \frac{\rho C_p}{h(A/V)} \ln \frac{400-279}{400-70}$$

$$\rho = 849 \ lb/ft^3$$

$$C_p = 0.033 \quad Btu/(lb)(^oF)$$

$$\theta = \frac{(849)(0.033)}{(2)(4/\frac{0.24}{12})} \ln \frac{330}{121} = 0.0707 \ln(2.72) = 0.0707 \ h$$

$$= 4.24 \text{ min}$$

For an infinite cylinder, A/V=(πDL)/(πD²/4)= 4/D

Example 21-1: time required for a mercury thermometer

$$Y = e^{-Bi \cdot Fo}$$

$$\ln \frac{400-279}{400-70} = -\frac{hA\theta}{\rho C_p V}$$

Biot number   Fourier number

$$\frac{400-279}{400-70} = \exp\left(-\frac{hA\theta}{\rho C_p V}\right) = \exp\left[-\left(\frac{hL}{k}\right)\left(\frac{k\theta}{\rho C_p L^2}\right)\right]$$

L : characteristic length scale

For an infinite cylinder, L=V/A=D/4

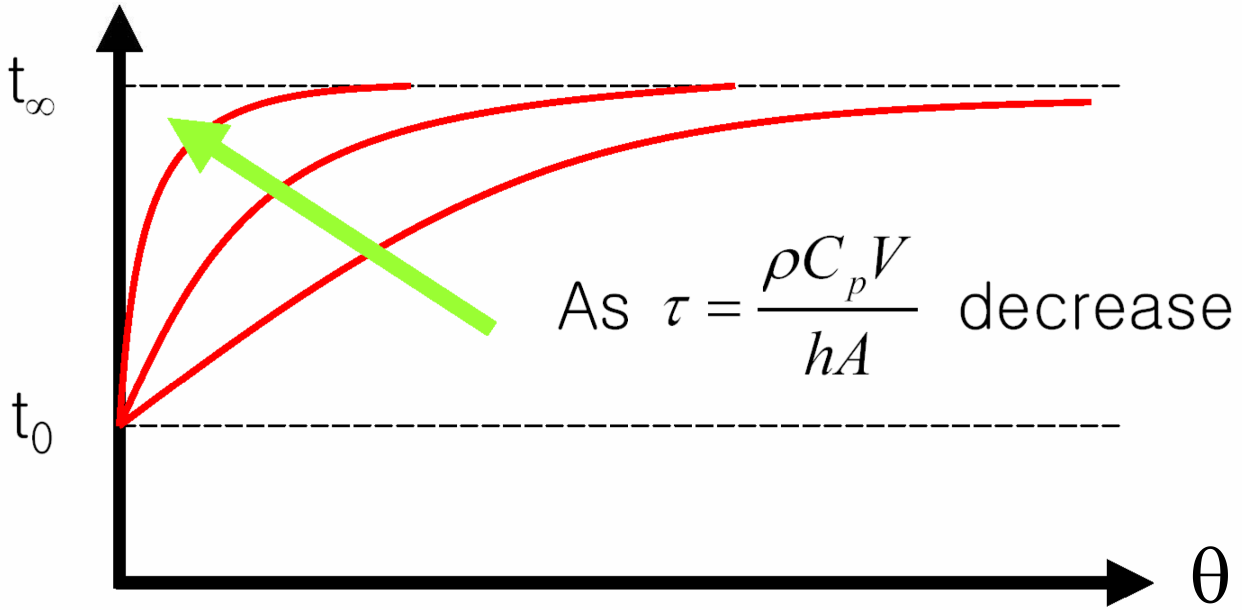
$$Y = e^{-\frac{\theta}{\tau}}$$

$$\tau = 4.24\,\text{min}$$

$$\tau = \frac{\rho C_p V}{hA}$$

: time constant (time required for the temperature difference driving force to fall to e^{-1} or 0.368 of its initial value.

Time constant



$$\frac{t_\infty - t}{t_\infty - t_0} = \exp\left(-\frac{\theta}{\tau}\right)$$

For fast response on change in temperature $\tau = \frac{\rho C_p V}{hA} \downarrow$

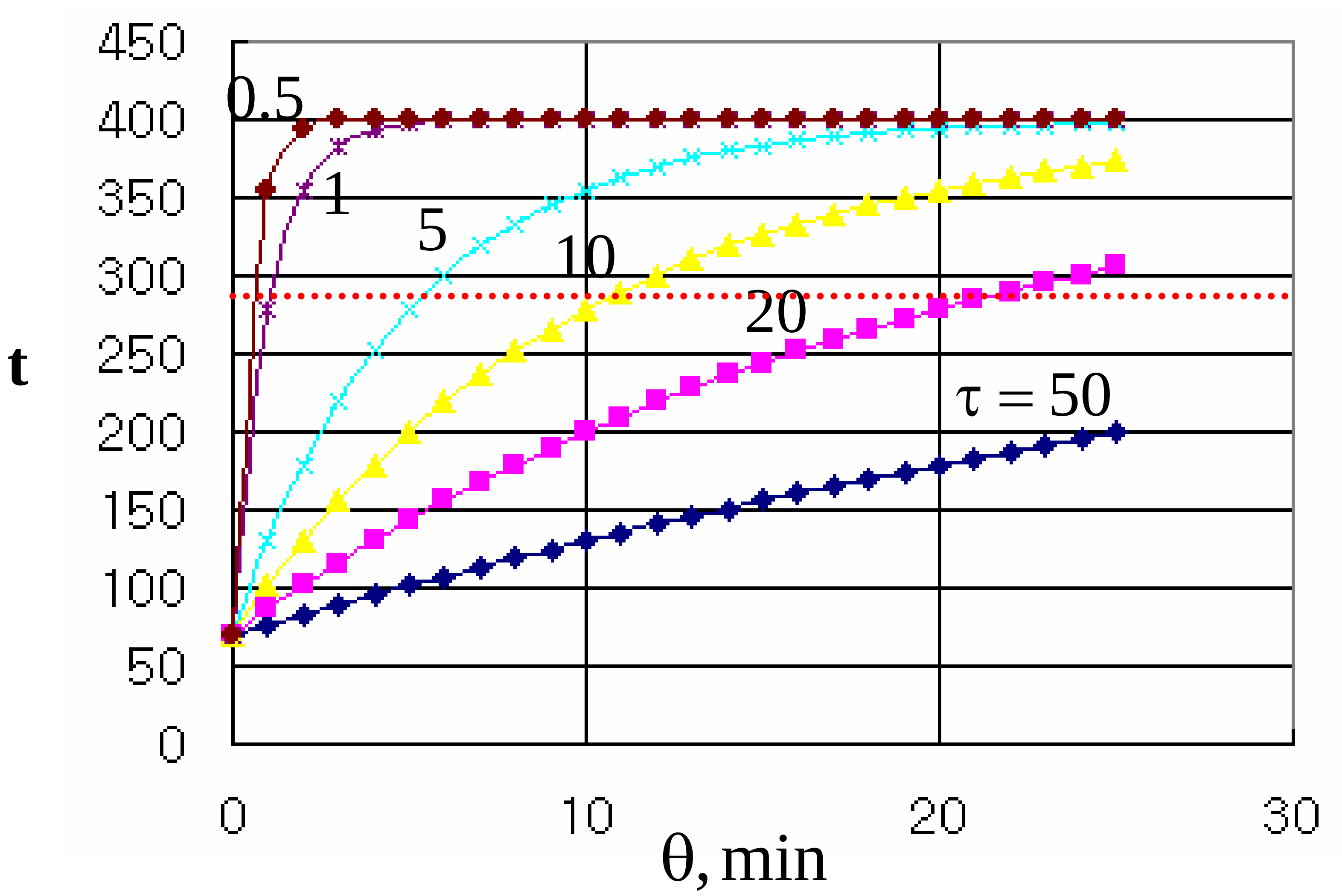
Temperature response, t

τ =time constant

θ

	50	20	10	5	1	0.5
0	70	70	70	70	70	70
1	76.53444	86.09429	101.4037	129.8189	278.5998	355.3394
2	82.93949	101.4037	129.8189	178.7944	355.3394	393.9558
3	89.2177	115.9664	155.53	218.8922	383.5703	399.182
4	95.37161	129.8189	178.7944	251.7214	393.9558	399.8893
5	101.4037	142.9957	199.8449	278.5998	397.7765	399.985
6	107.3163	155.53	218.8922	300.6059	399.182	399.998
7	113.1118	167.4529	236.1268	318.623	399.6991	399.9997
8	118.7925	178.7944	251.7214	333.3741	399.8893	400
9	124.3608	189.5827	265.832	345.4514	399.9593	400
10	129.8189	199.8449	278.5998	355.3394	399.985	400
11	135.1688	209.6066	290.1525	363.435	399.9945	400
12	140.4128	218.8922	300.6059	370.0631	399.998	400
13	145.553	227.7249	310.0645	375.4897	399.9993	400
14	150.5914	236.1268	318.623	379.9327	399.9997	400
15	155.53	244.119	326.367	383.5703	399.9999	400
16	160.3708	251.7214	333.3741	386.5485	400	400
17	165.1158	258.9531	339.7144	388.9868	400	400
18	169.7668	265.832	345.4514	390.9832	400	400
19	174.3257	272.3755	350.6424	392.6176	400	400
20	178.7944	278.5998	355.3394	393.9558	400	400
21	183.1745	284.5205	359.5894	395.0515	400	400
22	187.468	290.1525	363.435	395.9485	400	400
23	191.6764	295.5099	366.9146	396.6829	400	400
24	195.8015	300.6059	370.0631	397.2842	400	400
25	199.8449	305.4534	372.912	397.7765	400	400

Temperature response, t, with time



Biot Number (Bi)

$$Bi = \frac{hL}{k_s}$$

$$Bi = \frac{\text{the internal thermal resistance of a solid}}{\text{the boundary layer thermal resistance}}$$

Fourier Number (Fo)

$$Fo = \frac{\alpha\theta}{L^2}$$

$$F_O = \frac{\text{the rate of heat conduction}}{\text{the rate thermal energy storage in a solid}}$$

Reynolds Number (Re)

$$\mathrm{Re} = \frac{u_\infty L}{\nu}$$

$$\mathrm{Re} = \frac{\text{the inertia forces}}{\text{the viscous forces}}$$

Prandtl Number (Pr)

$$\mathrm{Pr} = \frac{C_p \mu}{k} = \frac{\nu}{\alpha}$$

$$\mathrm{Pr} = \frac{\text{the molecular momentum}}{\text{the thermal diffusivity}}$$

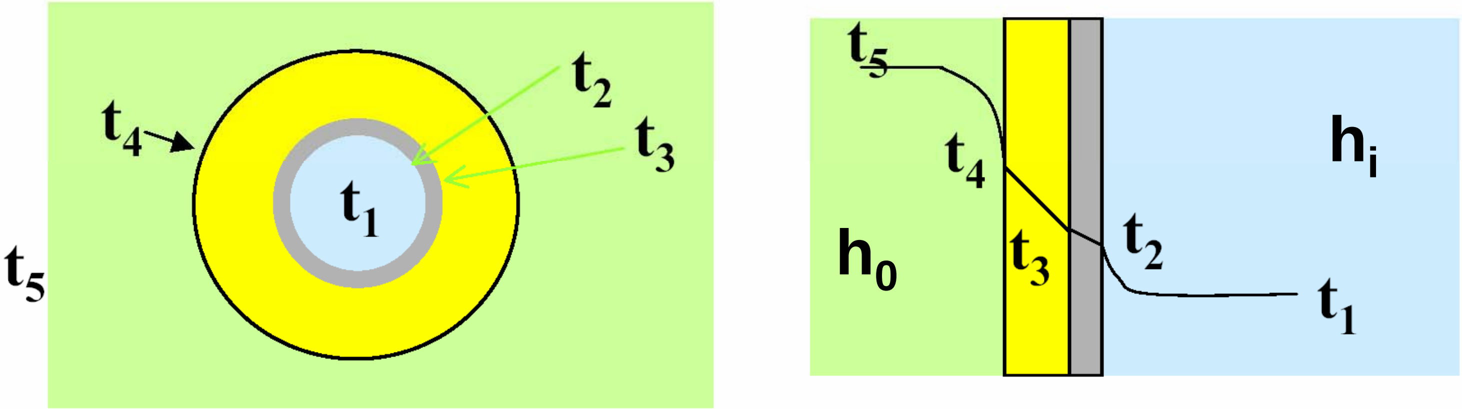
Individual heat transfer coefficient, *h*
Overall heat transfer coefficient, *U*

$$q = hA\Delta t_{individual}$$

$$q = UA\Delta t_{overall}$$

Overall Heat-Transfer Coefficient

Heat is transfer by a series of conduction and convection mechanisms



$$q = h_i A_i (t_1 - t_2) = k_b A_{b,\,lm} \frac{t_2 - t_3}{\Delta r_b} = k_c A_{c,\,lm} \frac{t_3 - t_4}{\Delta r_c} = h_0 A_0 (t_4 - t_5)$$

(21-12)

Thermal resistance $\dfrac{1}{h_i A_i}$ $\dfrac{\Delta r_b}{k_b A_{b,\,lm}}$ $\dfrac{\Delta r_c}{k_c A_{c,\,lm}}$ $\dfrac{1}{h_0 A_0}$

Individual temperature drops

$$t_1 - t_2 = q\frac{1}{h_i A_i}$$

$$t_2 - t_3 = q\frac{\Delta r_b}{k_b A_{b,lm}}$$

$$t_3 - t_4 = q\frac{\Delta r_c}{k_c A_{c,lm}}$$

$$t_4 - t_5 = q\frac{1}{h_0 A_0}$$

Overall temperature drops

$$\Delta t_{overall} = t_1 - t_5$$

$$= q\frac{1}{h_i A_i} + q\frac{\Delta r_b}{k_b A_{b,lm}} + q\frac{\Delta r_c}{k_c A_{c,lm}} + q\frac{1}{h_0 A_0}$$

$$= q\left(\frac{1}{h_i A_i} + \frac{\Delta r_b}{k_b A_{b,lm}} + \frac{\Delta r_c}{k_c A_{c,lm}} + \frac{1}{h_0 A_0}\right)$$

$$q = \frac{t_1 - t_5}{\left(\frac{1}{h_i A_i} + \frac{\Delta r_b}{k_b A_{b,lm}} + \frac{\Delta r_c}{k_c A_{c,lm}} + \frac{1}{h_0 A_0}\right)}$$

$$q = \frac{\Delta t_{overall}}{\Sigma R}$$

Overall coefficients of heat transfer based on the outside area U_0

$$q = \frac{A_0 \Delta t_{overall}}{\left(\frac{A_0}{h_i A_i} + \frac{A_0 \Delta r_b}{k_b A_{b,lm}} + \frac{A_0 \Delta r_c}{k_c A_{c,lm}} + \frac{1}{h_0} \right)}$$

$$\frac{1}{U_0} = \frac{A_0}{h_i A_i} + \frac{A_0 \Delta r_b}{k_b A_{b,lm}} + \frac{A_0 \Delta r_c}{k_c A_{c,lm}} + \frac{1}{h_0}$$

$$q = U_0 A_0 \Delta t_{overall}$$

Overall coefficients of heat transfer based on the inside area U_0

$$q = \frac{A_i \Delta t_{overall}}{\left(\frac{1}{h_i} + \frac{A_i \Delta r_b}{k_b A_{b,lm}} + \frac{A_i \Delta r_c}{k_c A_{c,lm}} + \frac{A_i}{h_0 A_0} \right)}$$

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{A_i \Delta r_b}{k_b A_{b,lm}} + \frac{A_i \Delta r_c}{k_c A_{c,lm}} + \frac{A_i}{h_0 A_i}$$

$$q = U_i A_i \Delta t_{overall}$$

Overall coefficients of heat transfer

$$q = U_0 A_0 \Delta t_{overall}$$

$$= U_i A_i \Delta t_{overall}$$

Overall coefficients of heat transfer

$$q = \frac{\Delta t_{overall}}{\Sigma R}$$

$$\Sigma R = \frac{1}{U_0 A_0} = \frac{1}{U_i A_i}$$

$$U_0 A_0 = U_i A_i$$

Approximation of overall coefficients of heat transfer based on the inside area U_i

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{A_i \Delta r_b}{k_b A_{b,lm}} + \frac{A_i \Delta r_c}{k_c A_{c,lm}} + \frac{A_i}{h_0 A_i}$$

$$\frac{1}{h_i} >> \frac{\Delta r_b}{k_b}, \frac{\Delta r_c}{k_c}, \frac{1}{h_0}$$

Approximation

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{\Delta r_b}{k_b} + \frac{\Delta r_c}{k_c} + \frac{1}{h_0}$$

Overall coefficients of heat transfer of the flat parallel walls

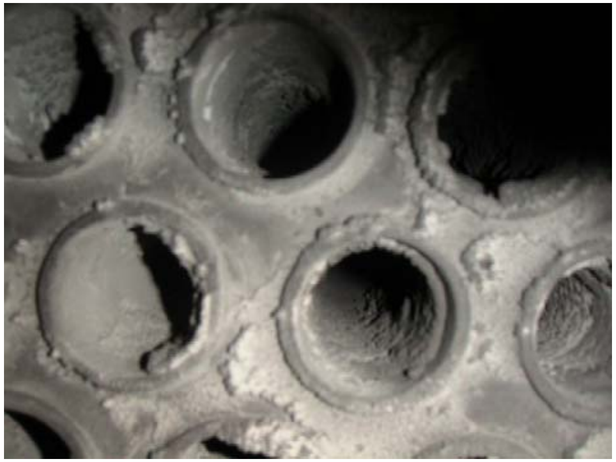
$$\frac{1}{U_i} = \frac{1}{U_0} = \frac{1}{h_i} + \frac{\Delta x_b}{k_b} + \frac{\Delta x_c}{k_c} + \frac{1}{h_0}$$

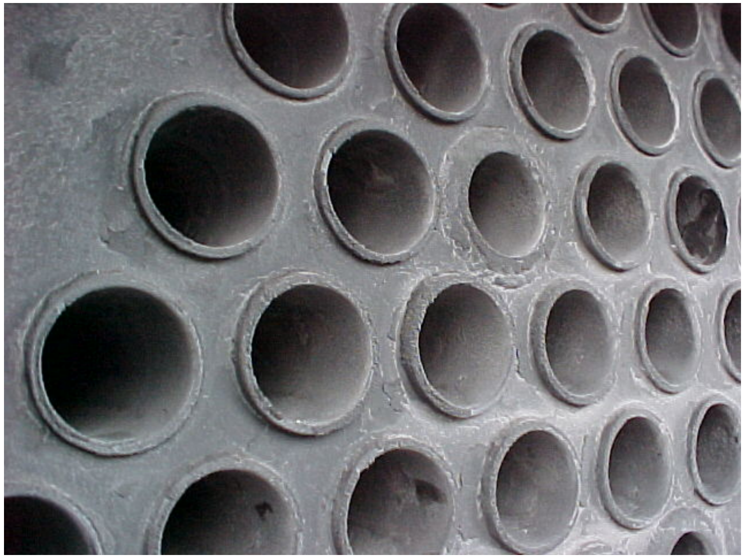
Overall coefficients of heat transfer

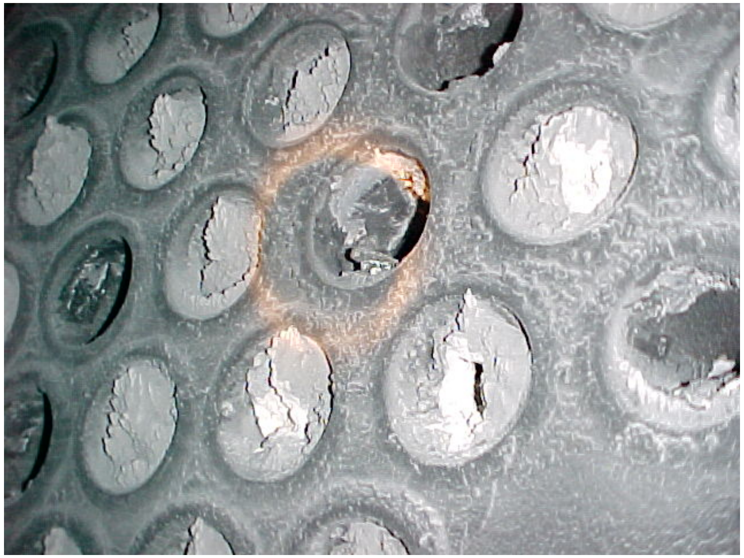
Table 21-2

	U, Btu/(h)(ft^2)(^{o}F)
Stabilizer reflux condenser	94
Oil pre-heater	108
Reboiler (condensing steam to re-boiling water)	300-800
Air heater (molten salt to air)	6
Steam-jacketed vessel evaporating milk	500

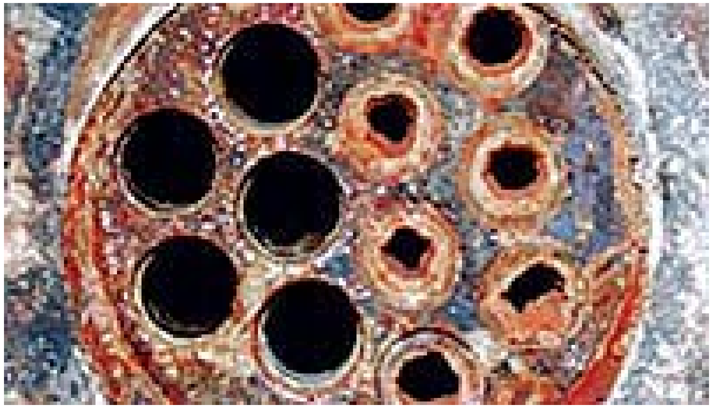
Fouling in heat exchanger

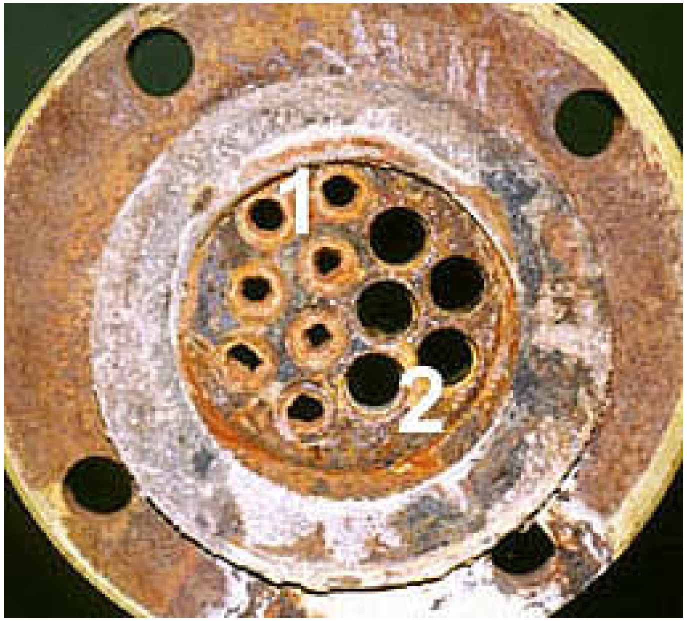


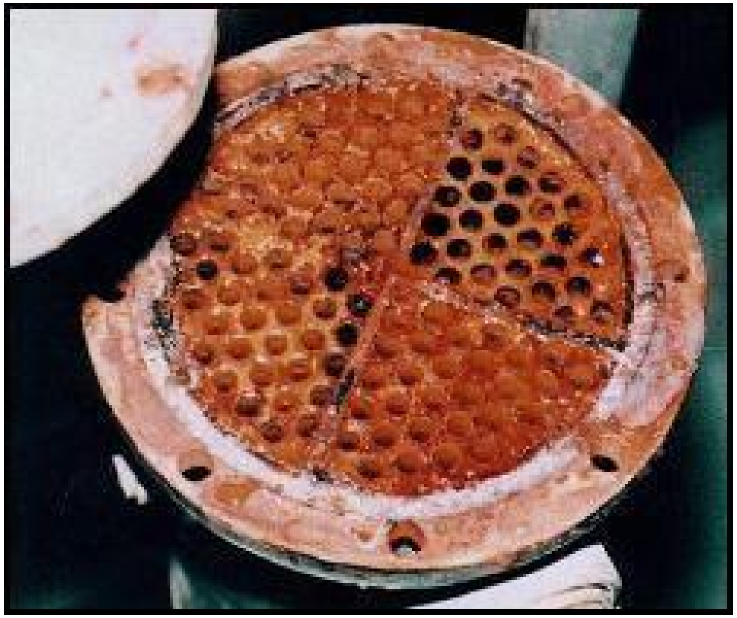




Fouling in heat exchanger







Fouling Coefficients

Fouling: deposits on heat transfer surface

- hard scale
  (boiler or evaporator)
- coke
  (oil heater in a refinery)

$$\text{Heat Transfer Coefficient} = \frac{\text{thermal conductivity}}{\text{thickness of scale}}$$

Sandblasting, pneumatic cleaning tools, chemical cleaning

- porous deposits
  (mud, soot, vegetable)

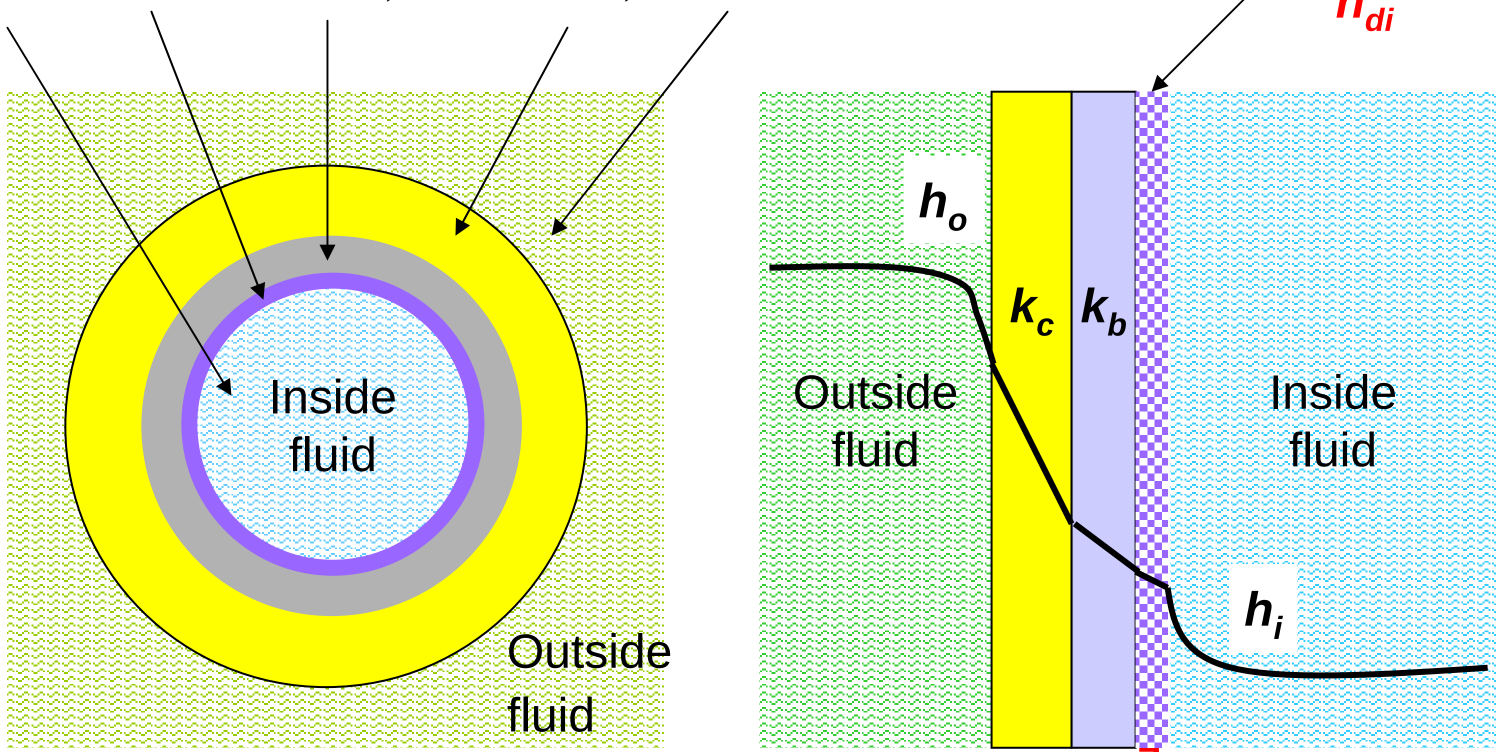
Thermal conductivity itself may be high
But effective thermal conductivity may be almost as low as that of the fluid.

Steam (air, hot water) blowing

Fouling Coefficients

$$q = h_d A \Delta t_{scale}$$

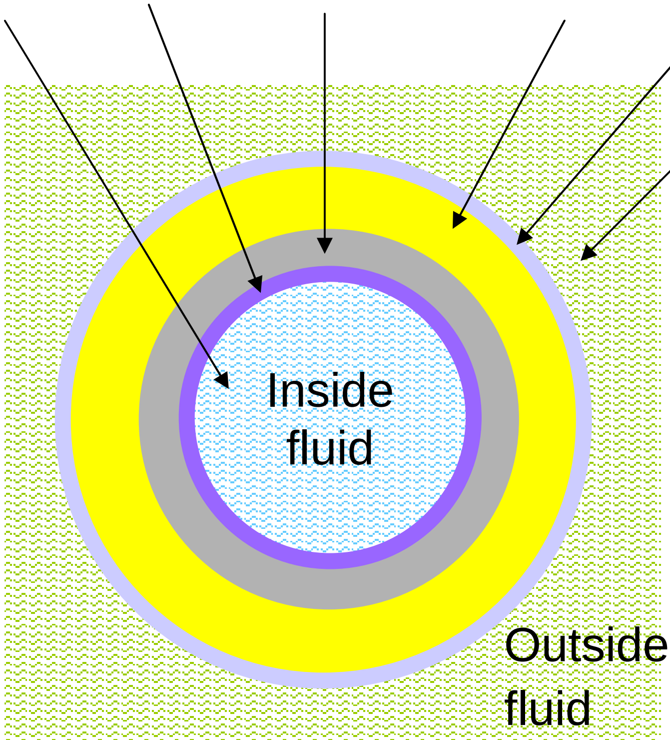
$$U_o = \frac{1}{\frac{A_o}{h_i A_i} + \frac{A_o}{h_{di} A_i} + \frac{\Delta r_b}{k_b}\frac{A_o}{A_{b,lm}} + \frac{\Delta r_c}{k_c}\frac{A_o}{A_{c,lm}} + \frac{1}{h_o}}$$

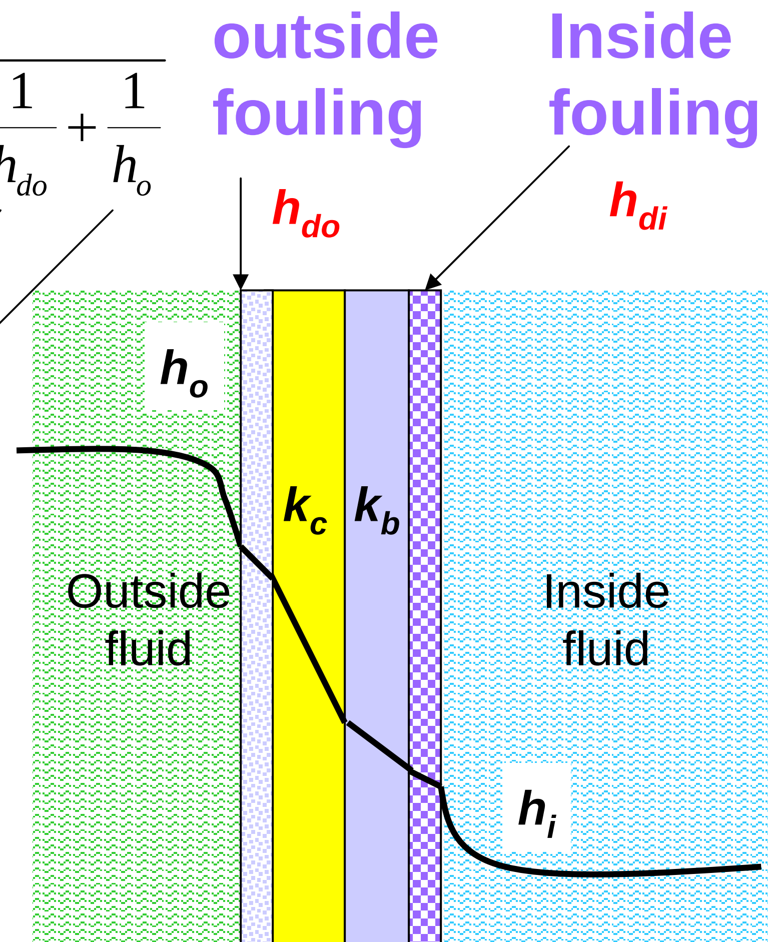


Fouling Coefficients

$$q = h_d A \Delta t_{scale}$$

$$U_o = \frac{1}{\frac{A_o}{h_i A_i} + \frac{A_o}{h_{di} A_i} + \frac{\Delta r_b}{k_b}\frac{A_o}{A_{b,lm}} + \frac{\Delta r_c}{k_c}\frac{A_o}{A_{c,lm}} + \frac{1}{h_{do}} + \frac{1}{h_o}}$$





Fouling coefficients

Table 21-3

	h_d, Btu/(h)(ft^2)(^{o}F)
Overhead vapors from crude-oil distillation	1000
Dry crude oil (300-100^{o}F):	
Velocity under 2 ft/sec	250
Velocity 2-4 ft/sec	330
Velocity over 4 ft/sec	500
Air	500
Steam (non-oil-bearing)	200
Water, Great Lakes, over 125^{o}F	500

Tubular Exchanger Manufacturers Association, New York, 1949

Example 21-2

A reflux condenser contains ¾ in. 16-gauge copper tubes in which cooling water circulates. Hydrocarbon vapors condense on the exterior surfaces of the tubes. Find the overall heat-transfer coefficient U_o. The inside convective coefficient can be taken as 4500 W/m^2·K, and the outside coefficient as 1500 W/m^2·K.

Approximately fouling coefficients from Table 21-3 are

$$h_{d0} = 5700 \text{ W/m}^2\cdot\text{K}$$
$$h_{di} = 2840 \text{ W/m}^2\cdot\text{K}$$

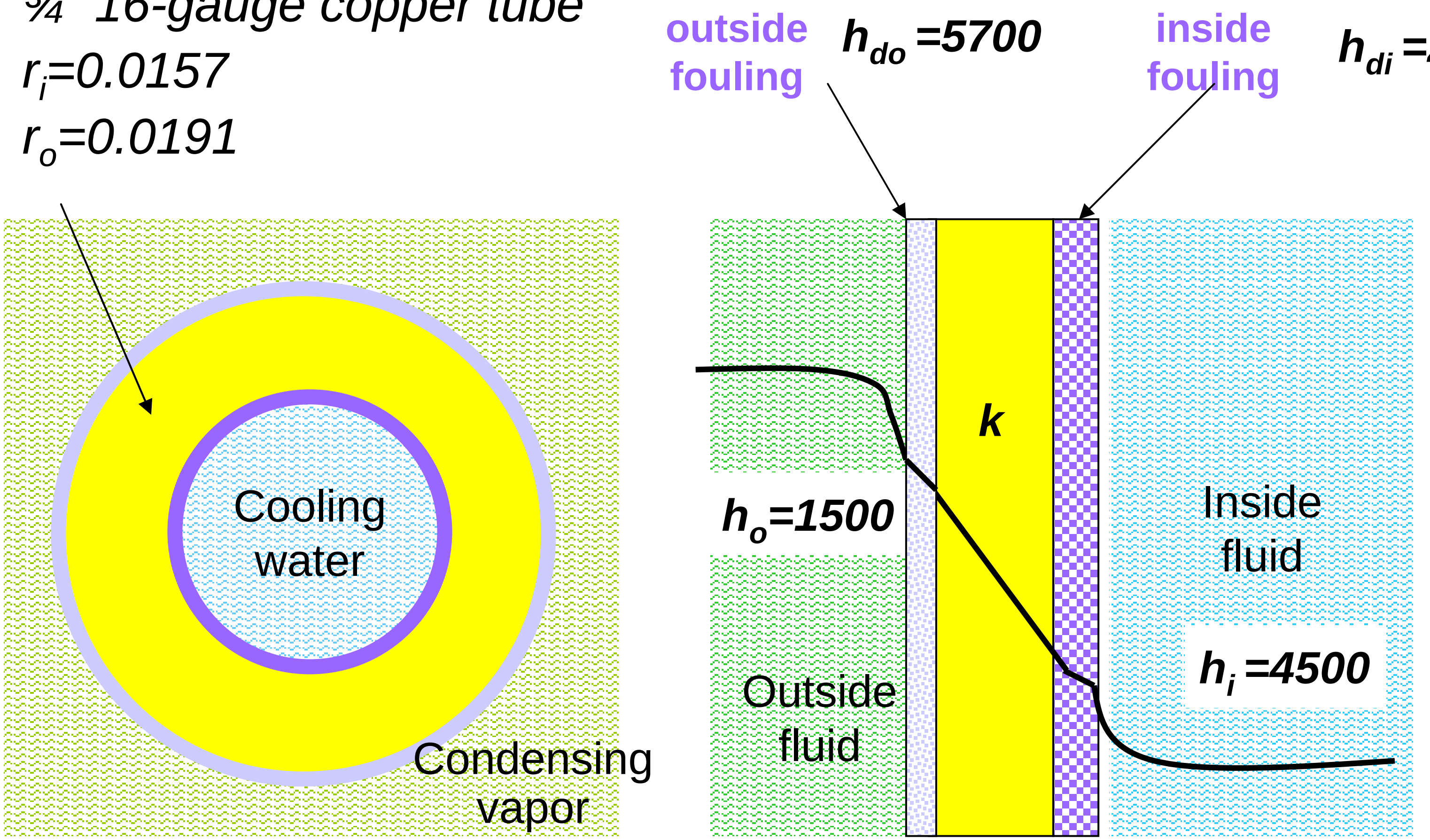
	Btu/(h)(ft^2)(^{o}F)
Overhead vapors from crude-oil distillation	1000
Water, Great Lakes, over 125^{o}F	500

Example 21-2 Find U_o

¾" 16-gauge copper tube

r_i=0.0157

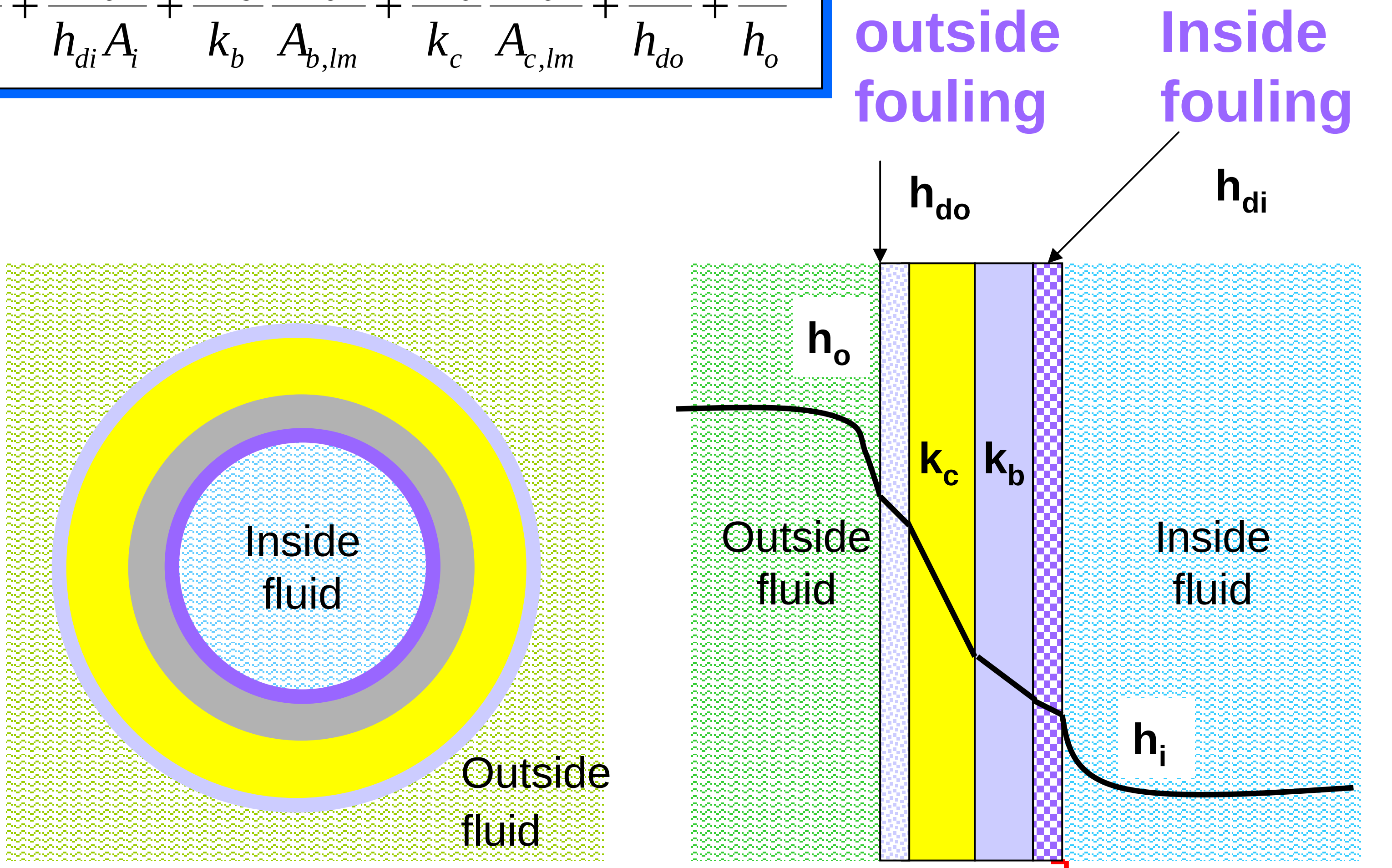
r_o=0.0191



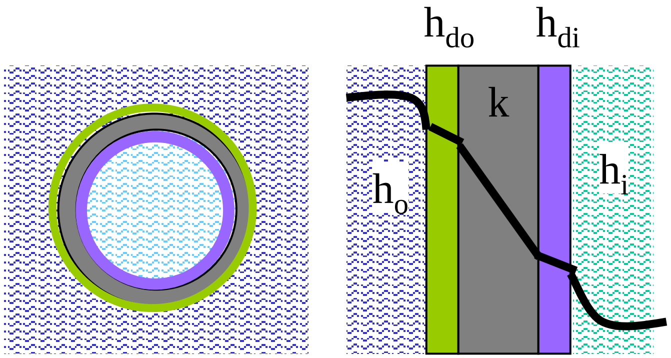
unit: $W/m^2 \cdot K$

Fouling Coefficients

$$U_o = \cfrac{1}{\cfrac{A_o}{h_i A_i} + \cfrac{A_o}{h_{di} A_i} + \cfrac{\Delta r_b}{k_b}\cfrac{A_o}{A_{b,lm}} + \cfrac{\Delta r_c}{k_c}\cfrac{A_o}{A_{c,lm}} + \cfrac{1}{h_{do}} + \cfrac{1}{h_o}}$$



Example 21-2



$$U_0 = \frac{1}{\dfrac{r_0}{h_i r_i} + \dfrac{r_0}{h_{d_i} r_i} + \dfrac{\Delta r \cdot r_0}{k_b r_{lm}} + \dfrac{1}{h_{d_0}} + \dfrac{1}{h_0}}$$

$$h_i = 4500 W/m^2 \cdot K, \quad h_0 = 1500 W/m^2 \cdot K$$

$$h_{d_0} = 1000 \times 5.678 = 5700 W/m^2 \cdot K$$

$$h_{d_i} = 500 \times 5.678 = 2840 W/m^2 \cdot K$$

$$k = 380\ W/m \cdot K \quad , \quad r_i = 0.0157m \quad , \quad r_0 = 0.0191m$$

$$r_{l_m} = \frac{r_0 - r_i}{\ln \dfrac{r_0}{r_i}} = \frac{0.0191 - 0.0157}{\ln \dfrac{0.0191}{0.0157}} = 0.0175m$$

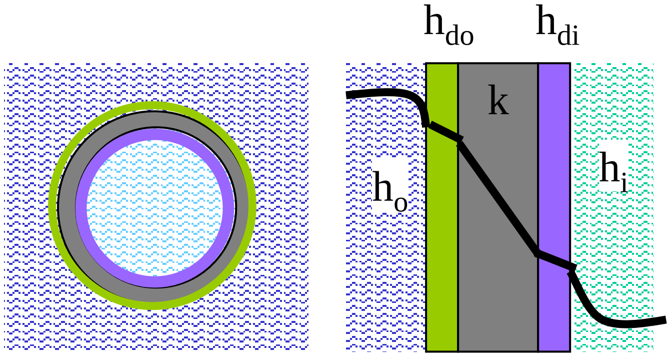
$$\frac{r_0}{h_i r_i} = \frac{0.0191}{(4500)(0.0157)} = 0.00027$$

$$\frac{r_0}{h_{di} r_i} = \frac{0.0191}{(2840)(0.0157)} = 0.00043$$

$$\frac{\Delta r \cdot r_0}{k_b r_{lm}} = \frac{(0.00165)(0.0191)}{(380)(0.0175)} = 0.0000047$$

$$\frac{1}{h_{d_0}} = \frac{1}{5700} = 0.00018, \quad \frac{1}{h_0} = \frac{1}{1500} = 0.00067$$

Example 21-2



$$U_0 = \frac{1}{\frac{r_0}{h_i r_i} + \frac{r_0}{h_{d_i} r_i} + \frac{\Delta r \cdot r_0}{k_b r_{lm}} + \frac{1}{h_{d_0}} + \frac{1}{h_0}}$$

$$= \frac{1}{0.00027 + 0.00043 + 0.0000047 + 0.00018 + 0.00067}$$

Resistance of metal wall is negligible

Fouling Resistance is significant~39%

$$= \frac{1}{0.00155} = 645\, W/m^2 \cdot K$$

$$U_i A_i = U_o A_o$$

$$U_i = (645)(0.0191)/((0.0157) = 785\, W/m^2 \cdot K$$

Heat Exchangers

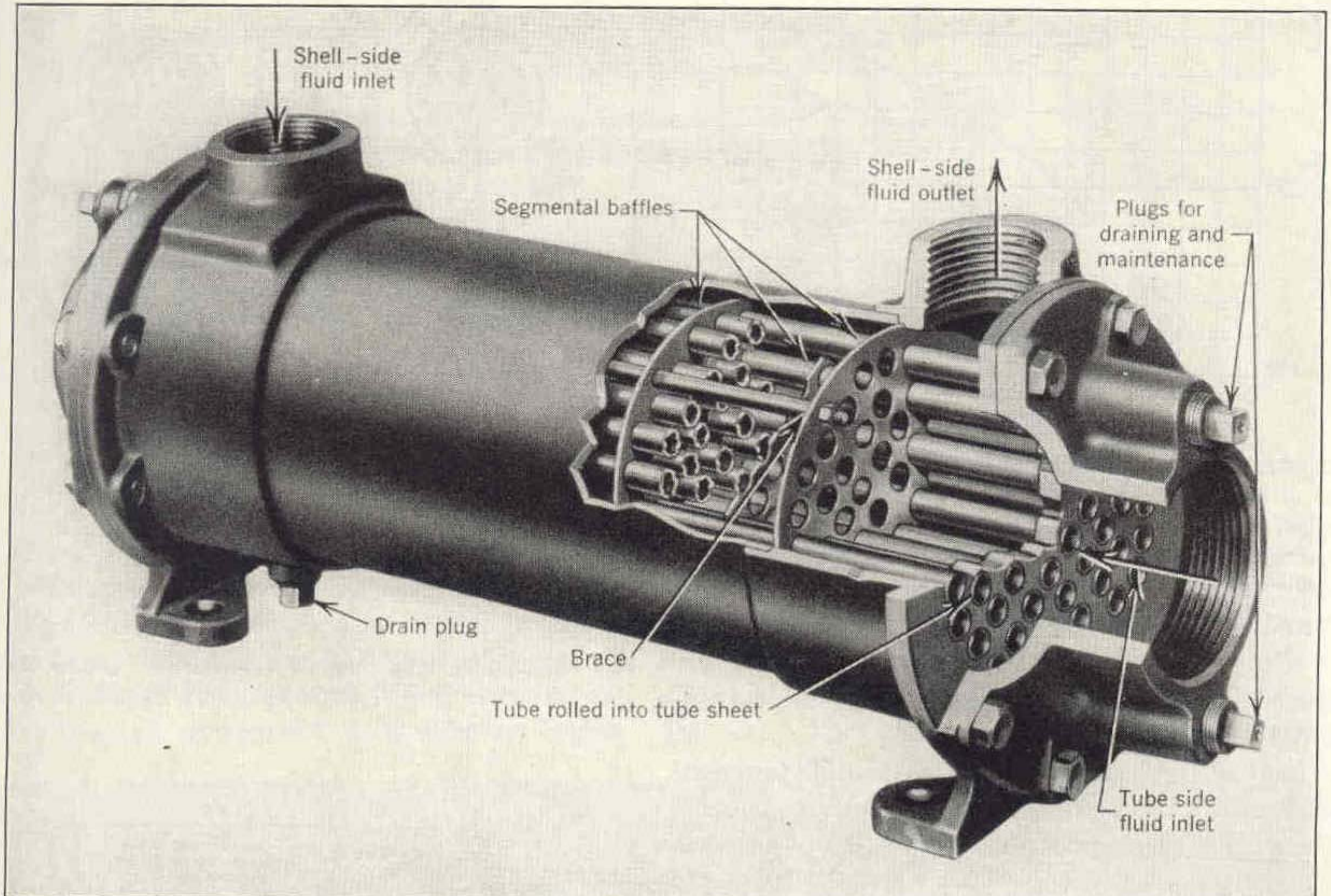


FIG. 403. Cutaway view of a single-pass shell and tube exchanger. (*Ross Heater and Mfg. Co.*)

Heat Exchangers

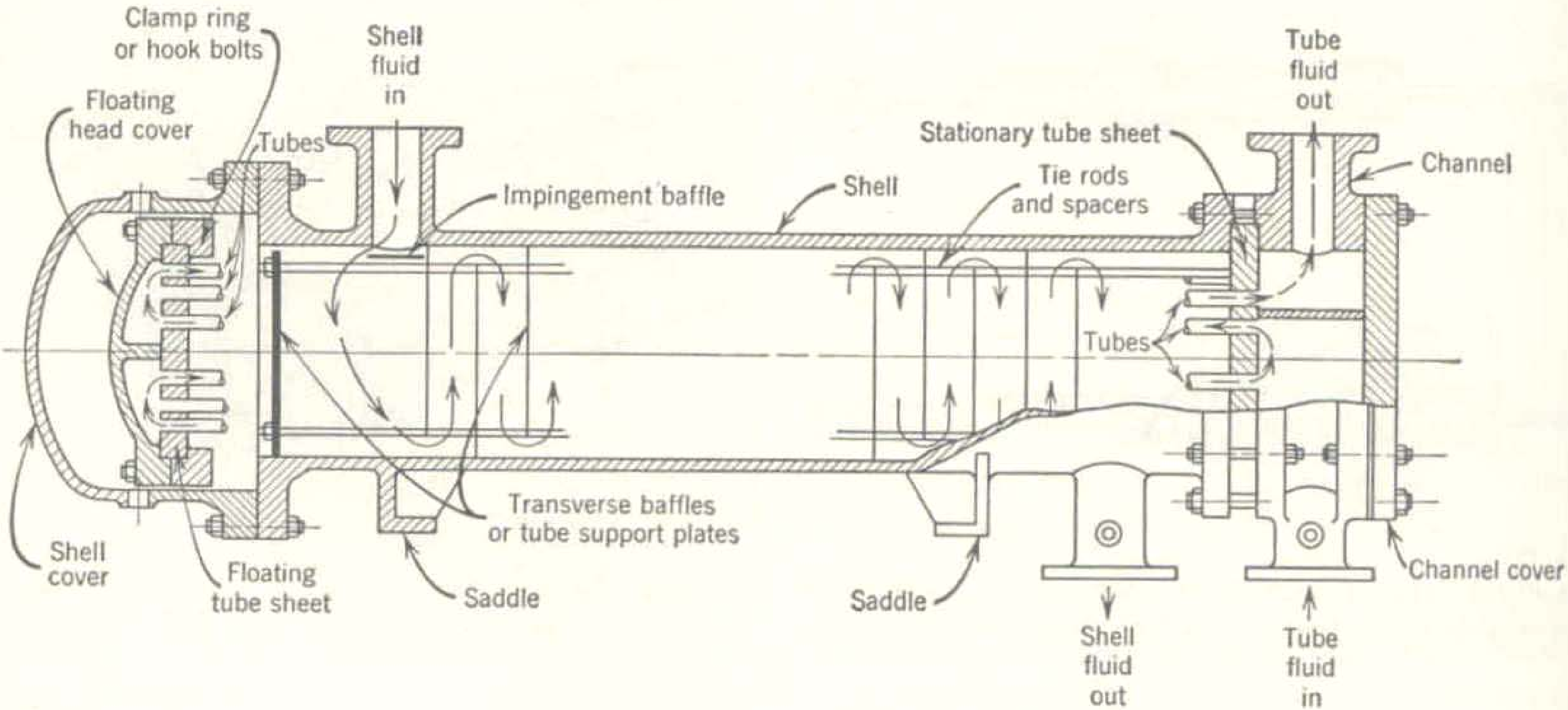


FIG. 410. Cross-sectional drawing of a typical four-pass tube side, single-pass shell side, floating head heat exchanger. (*Tubular Exchanger Manufacturers Association.*)

Heat Exchangers

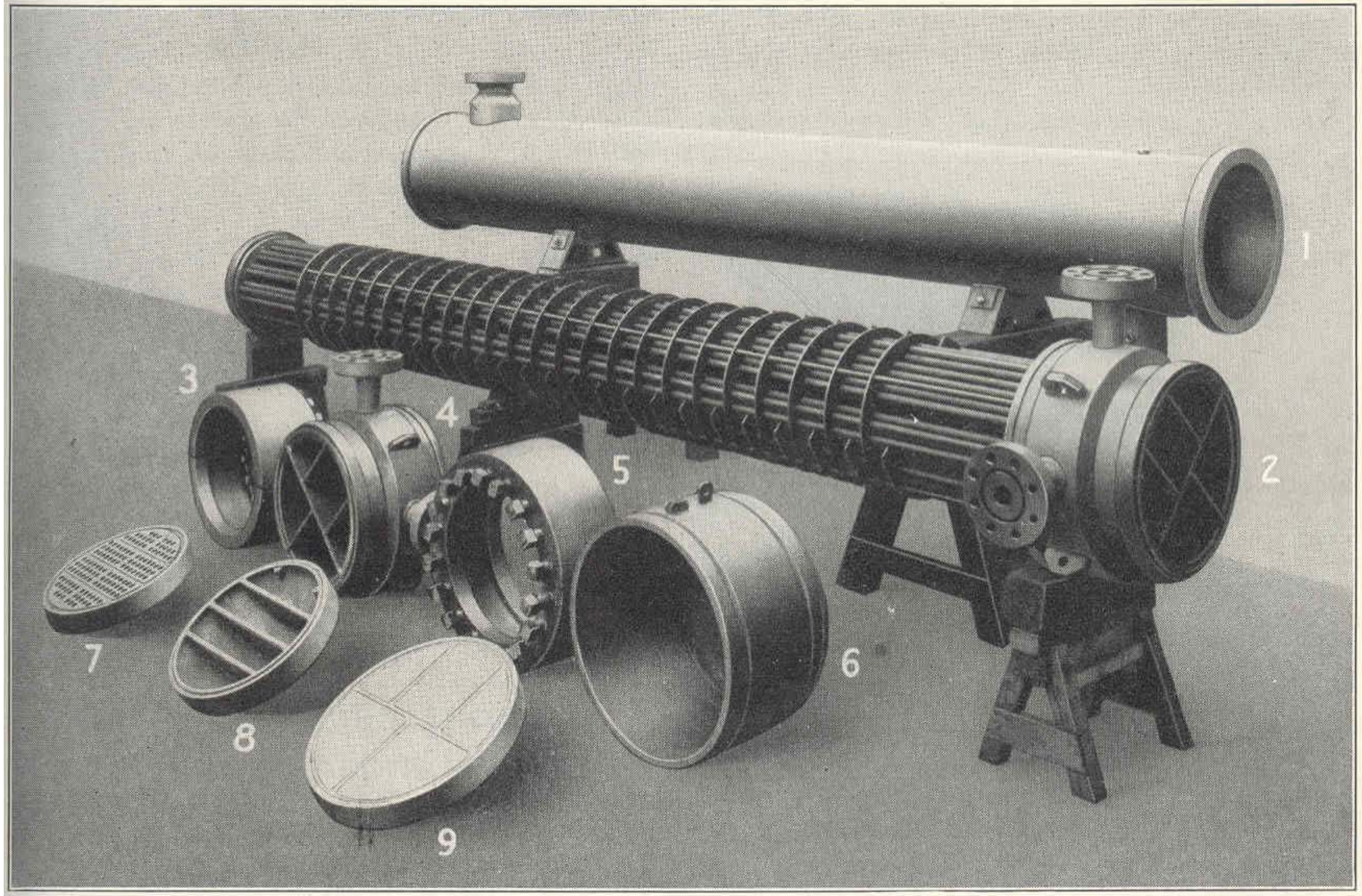
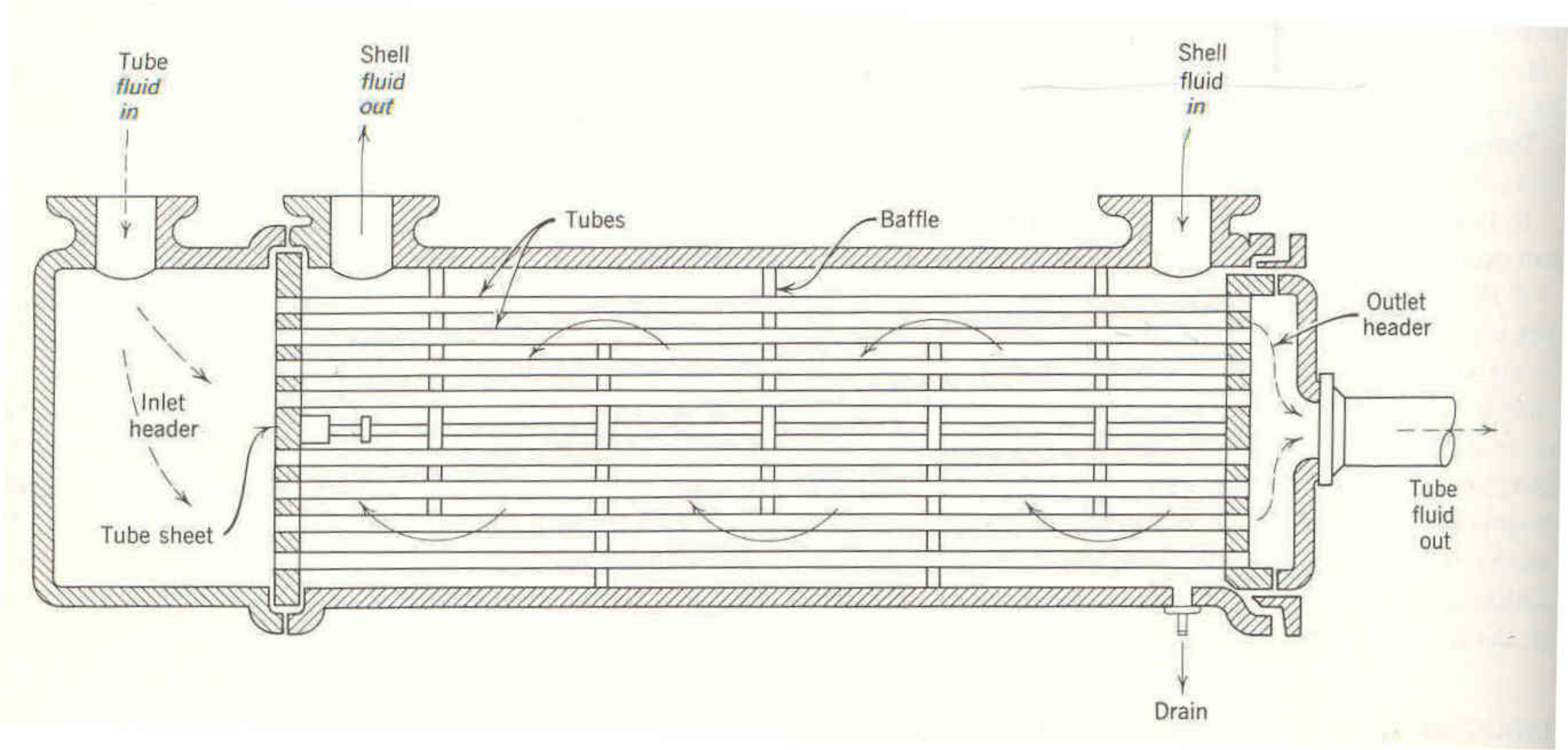


FIG. 412. Component parts of a feed water heater. (*American Locomotive Co.*) 1, shell; 2, assembled tube bundle; 3, clamp ring for floating head; 4, channel with integral tube sheet; 5, clamp ring for channel cover; 6, shell cover; 7, floating head tube sheet; 8, floating head cover; 9, channel cover.

Heat Exchangers



Heat Exchangers

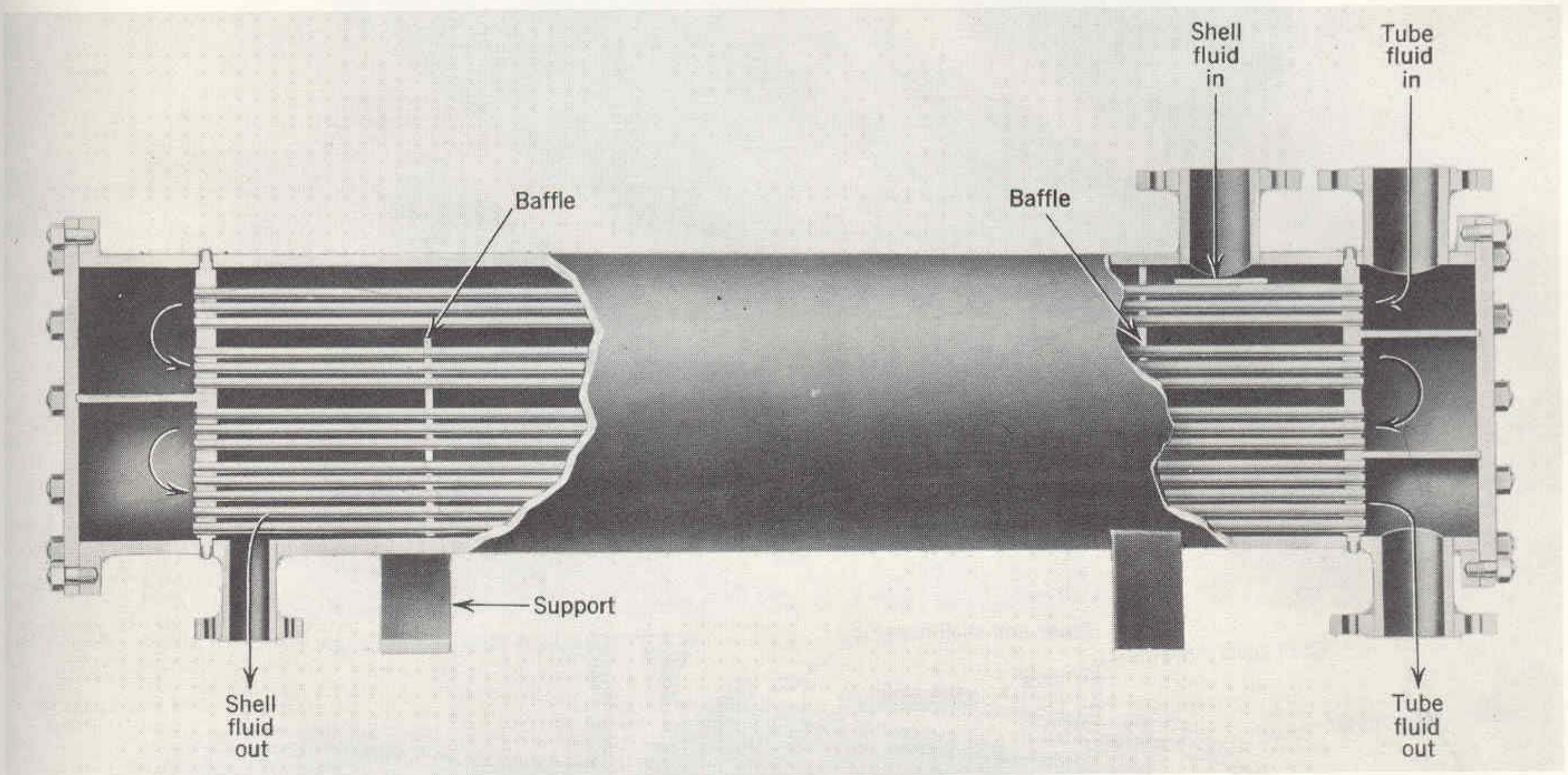
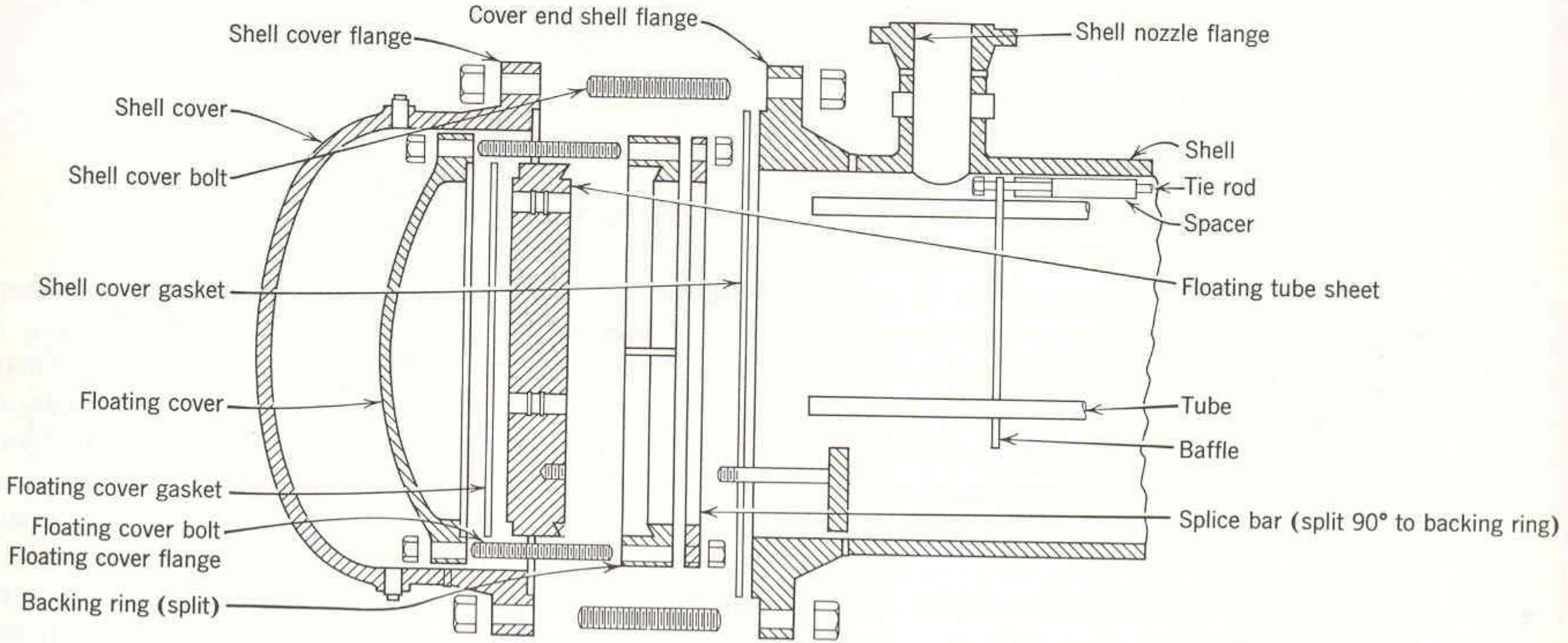


Figure 15.3. Heat exchanger with four tube passes and one shell pass. (Courtesy The Whitlock Manufacturing Co.)

Heat Exchangers



(*c*) Detail of floating head.

Figure 15.4. Two-tube-pass, one-shell-pass, floating-head heat exchangers. (Courtesy National U.S. Radiator Corp., Heat Transfer Division.)

Shell–and–Tube Heat Exchangers

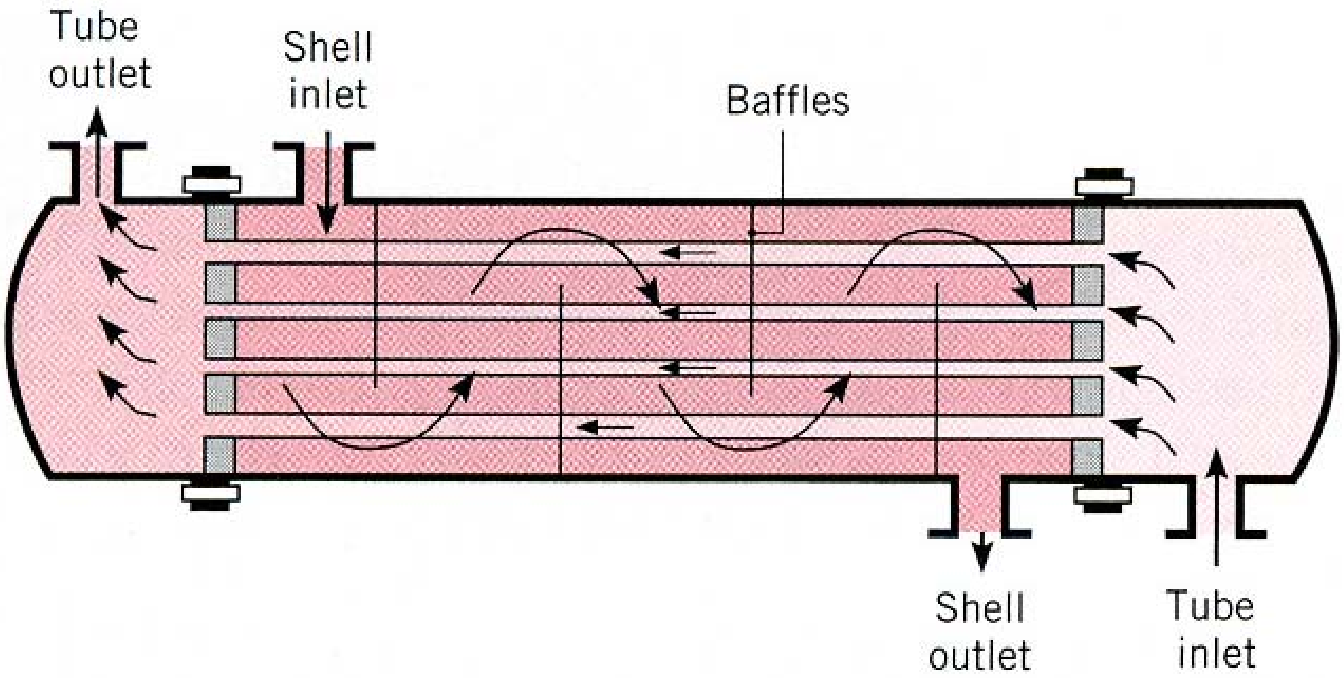


FIGURE 11.3 Shell-and-tube heat exchanger with one shell pass and one tube pass (cross-counterflow mode of operation).

Shell–and–Tube Heat Exchangers

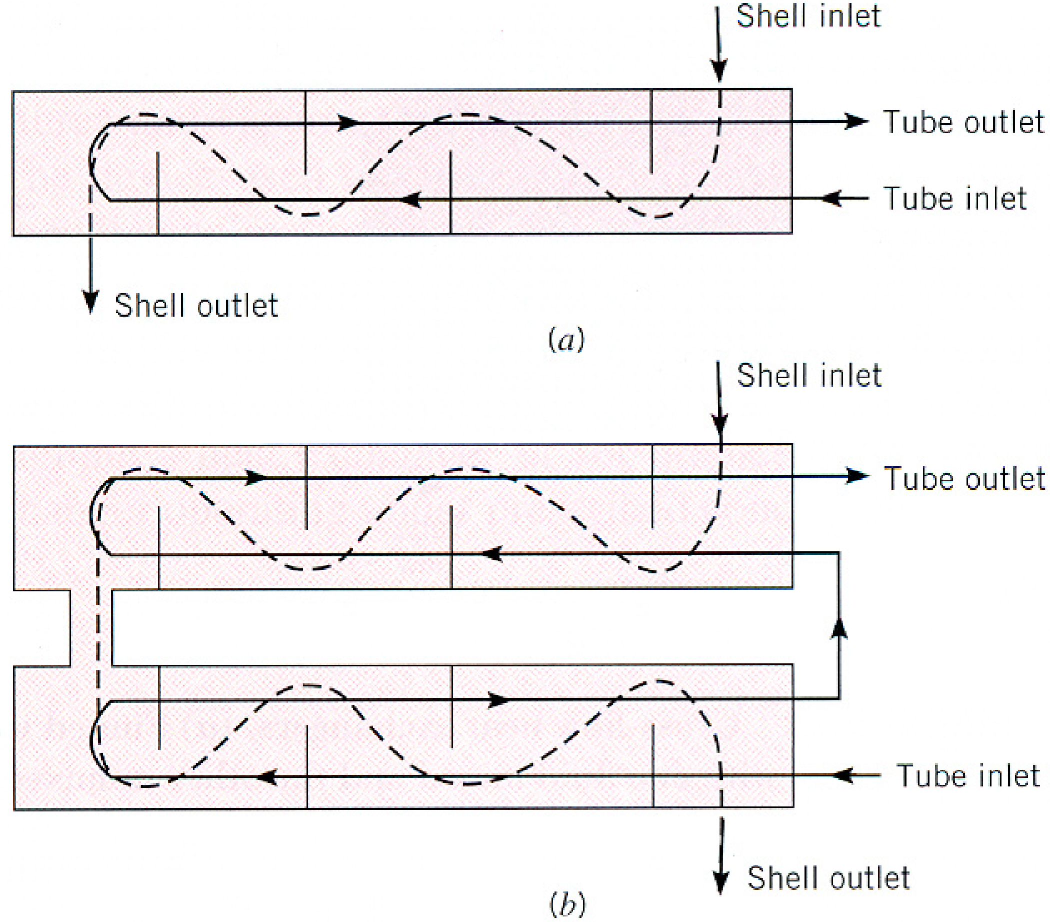
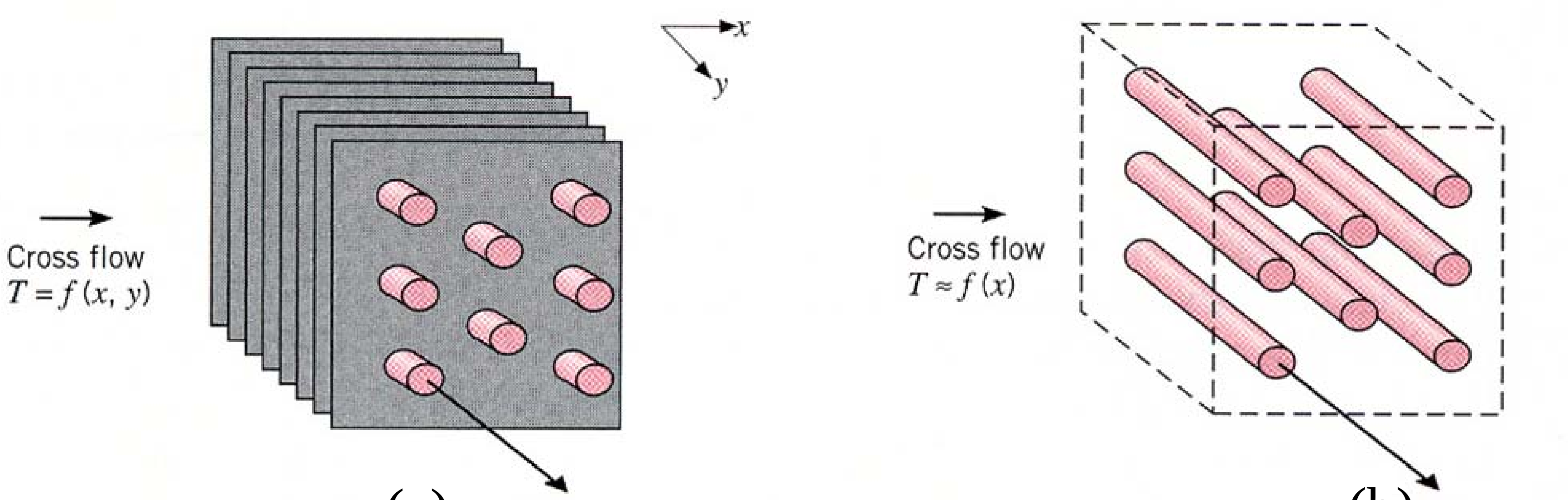


FIGURE 11.4 Shell-and-tube heat exchangers. (*a*) One shell pass and two tube passes. (*b*) Two shell passes and four tube passes.

Heat Exchangers

Figure 15.5 Extended surface tubing. (Courtesy The Griscom-Russell Co.)

Crossflow heat exchangers



(a) (b)

(a) Finned with both fluid unmixed

(b) Unfinned with one fluid mixed and the other unmixed

Compact heat exchangers

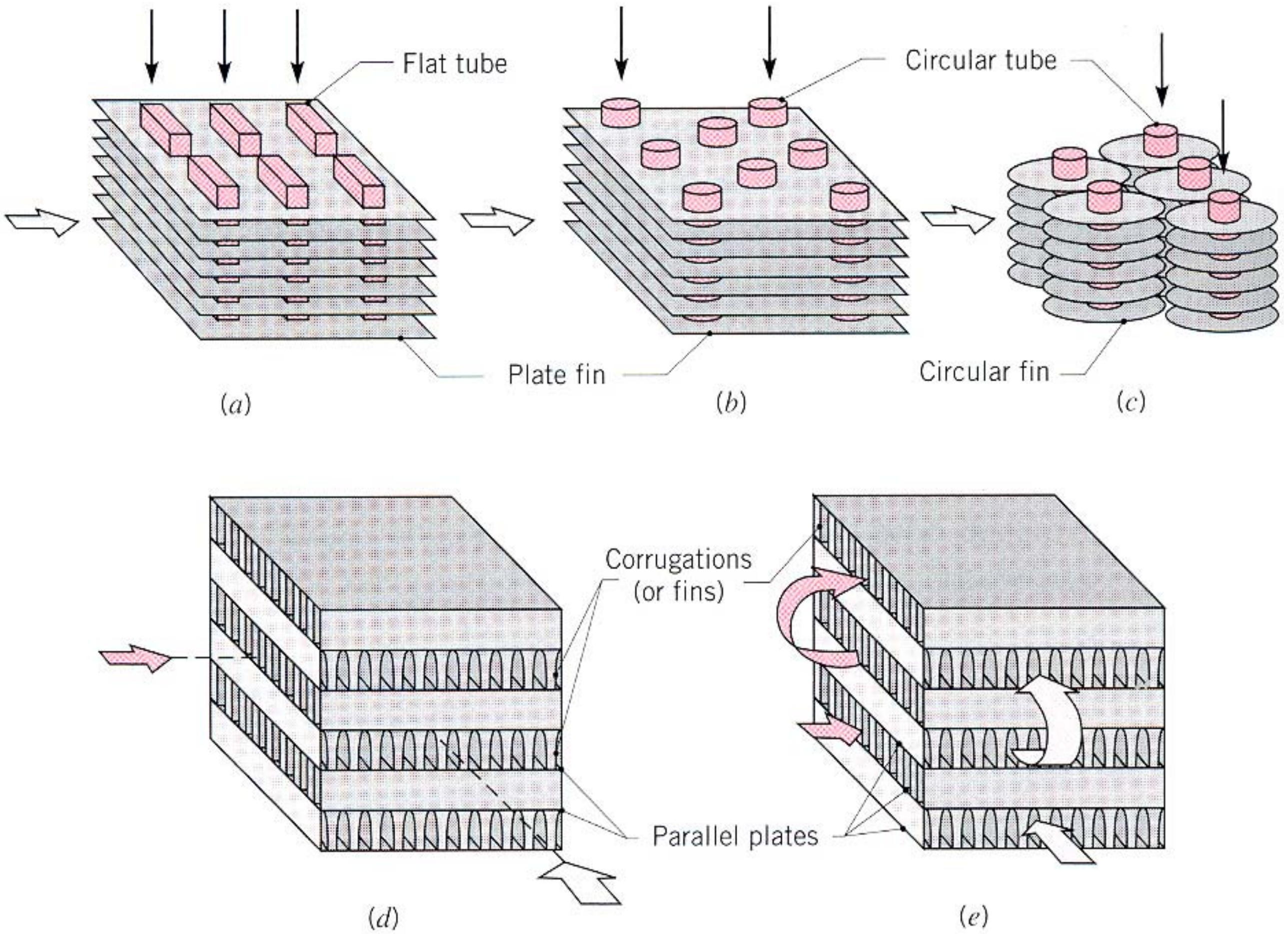
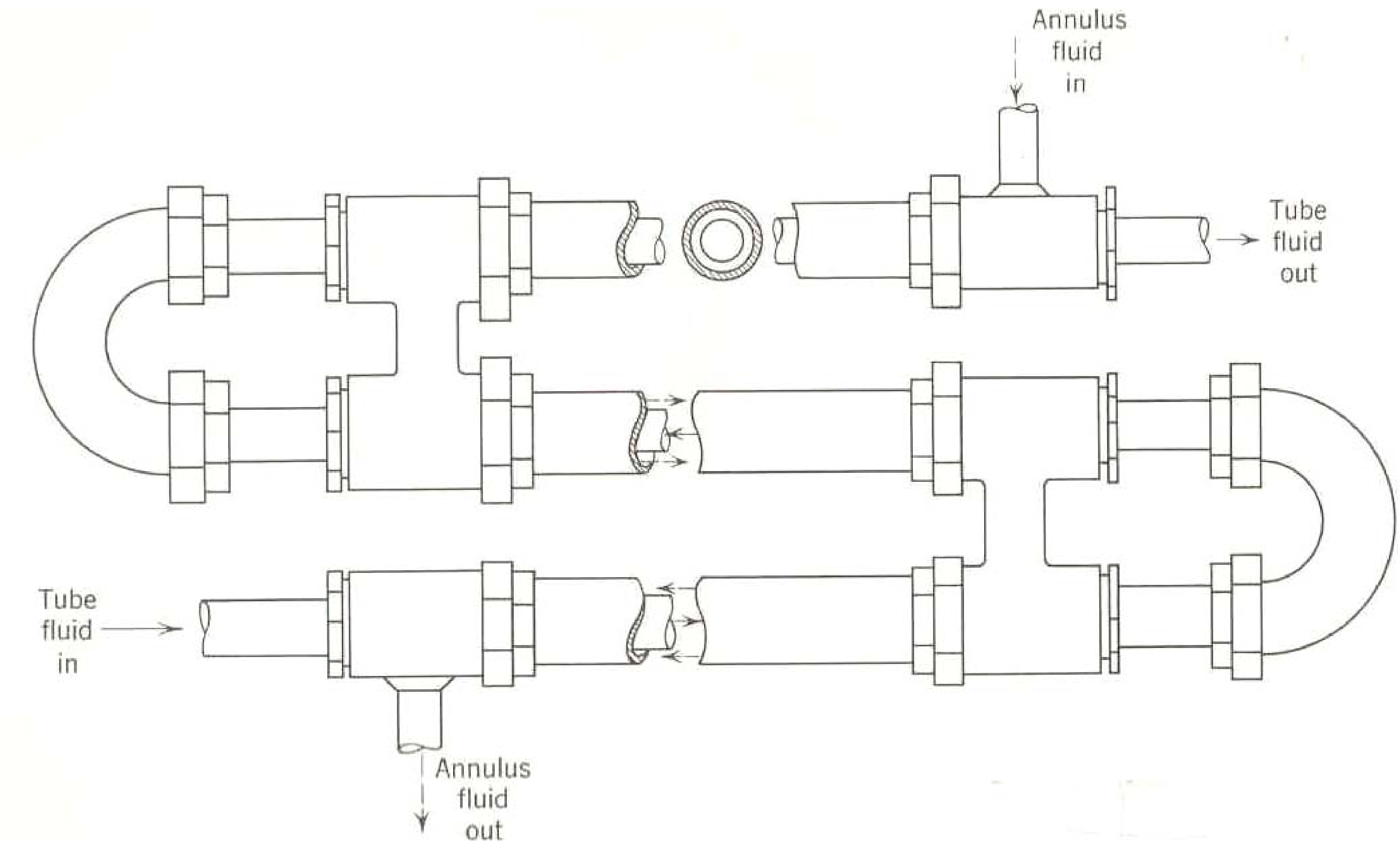
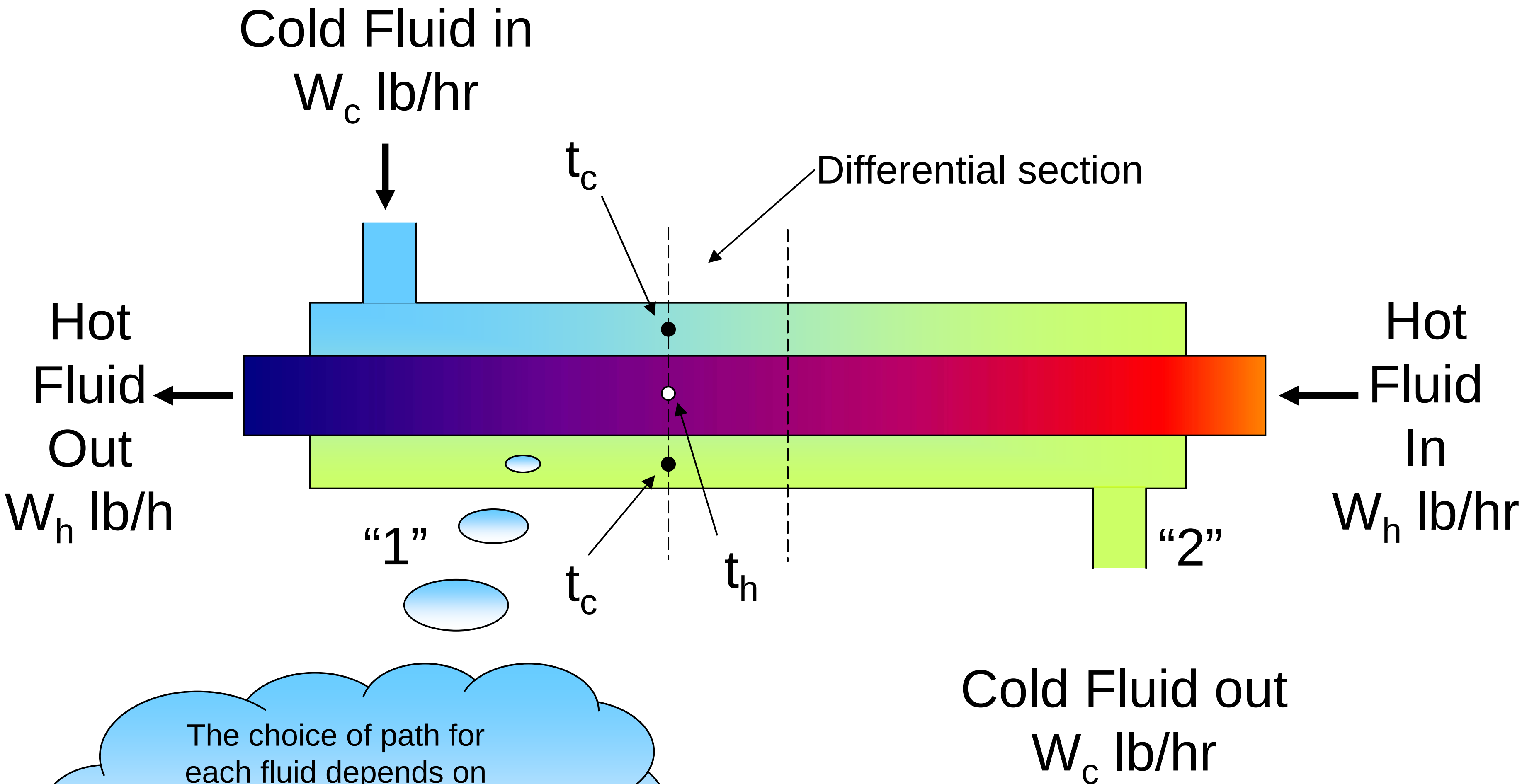


FIGURE 11.5 Compact heat exchanger cores. (*a*) Fin–tube (flat tubes, continuous plate fins). (*b*) Fin–tube (circular tubes, continuous plate fins). (*c*) Fin–tube (circular tubes, circular fins). (*d*) Plate–fin (single pass). (*e*) Plate–fin (multipass).

Double-pipe heat exchanger

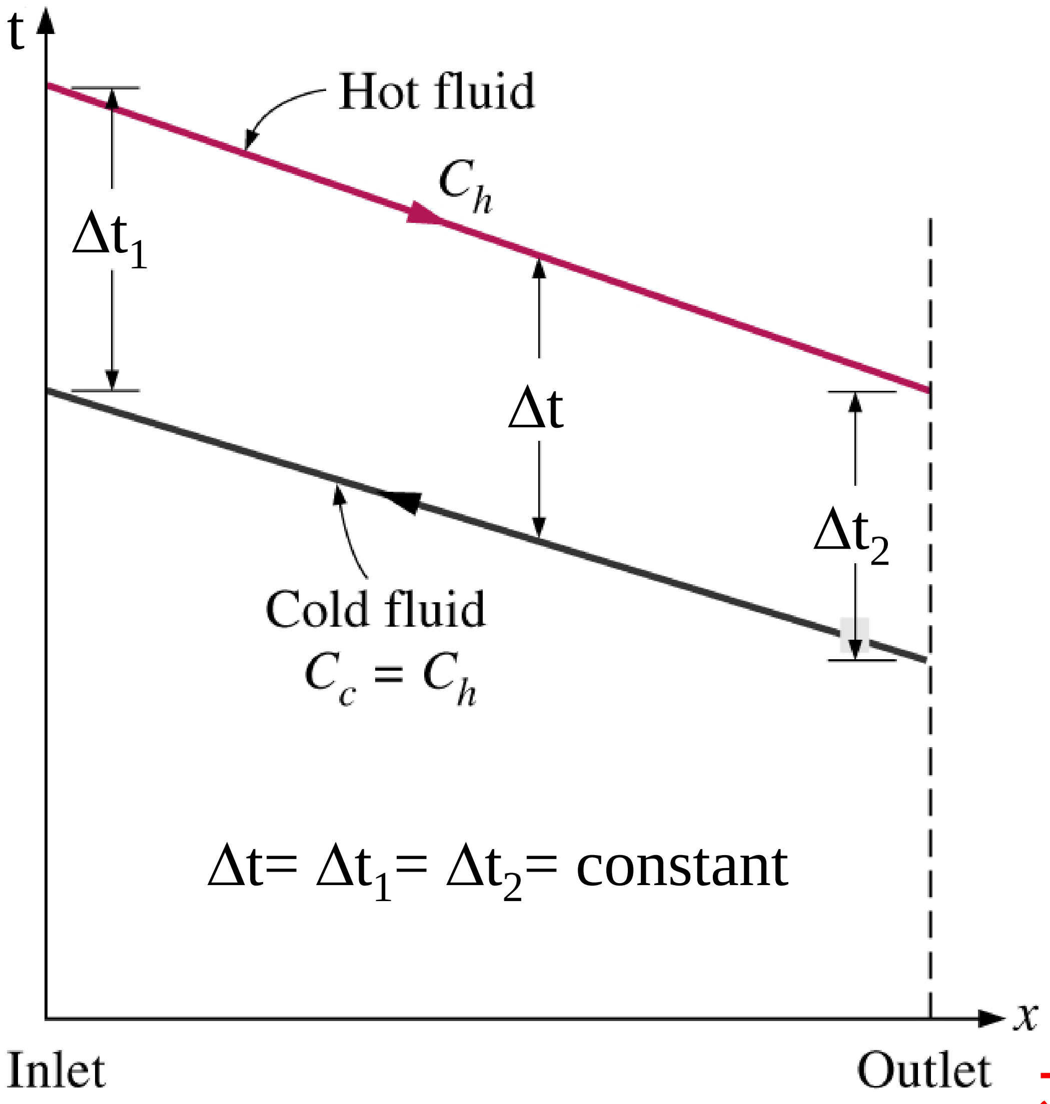


Double-pipe heat exchanger

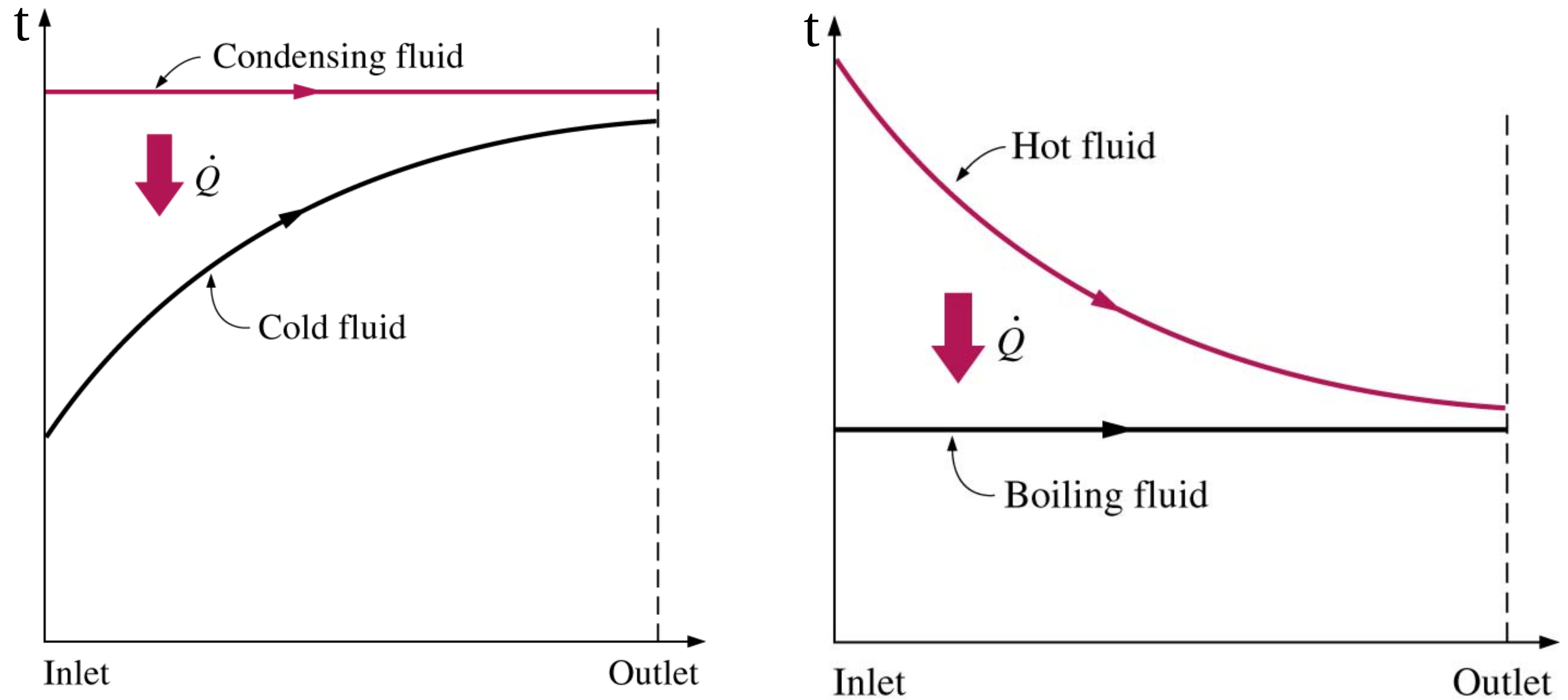


The choice of path for each fluid depends on considerations such as corrosion, plugging, fluid pressure, and permissible pressure drop.

Double-pipe heat exchanger



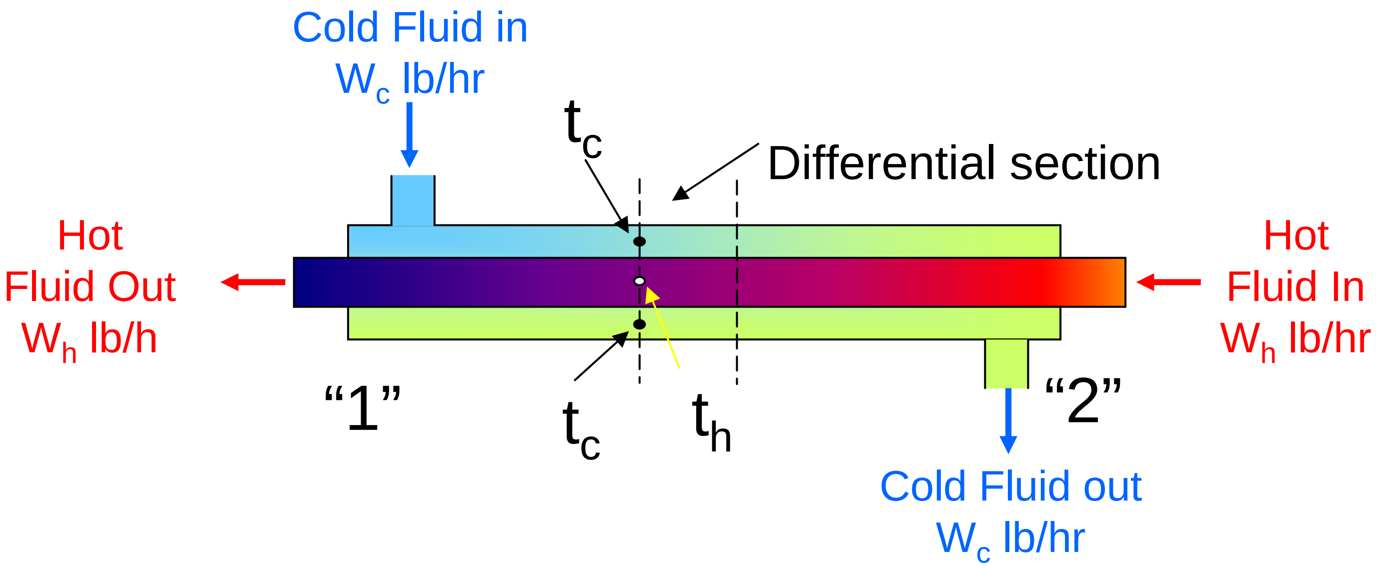
Double-pipe heat exchanger



(*a*) Condenser ($C_h \rightarrow \infty$)

(*b*) Boiler ($C_c \rightarrow \infty$)

Double-pipe heat exchanger



The determination of the required heat transfer area is one of the principal objective in the design of heat exchangers. For a differential segment of the exchanger for which the outside tube area is dA_0,

$$q = U_0 A_0 \Delta t_{overall}$$

Generally,

$$dq = U_0 \Delta t_{overall} dA_0 \quad \leftarrow \quad U_0 \ \& \ \Delta t \text{ are varing}$$

$$= U_i \Delta t dA_i$$

$$\int_0^q \frac{dq}{U_0 \Delta t_{overall}} = \int_0^{A_0} dA_0$$

(21-23)

Double-pipe heat exchanger

$$\int_0^q \frac{dq}{U_0 \Delta t_{overall}} = \int_0^{A_0} dA_0$$

Outside heat transfer area, A_0

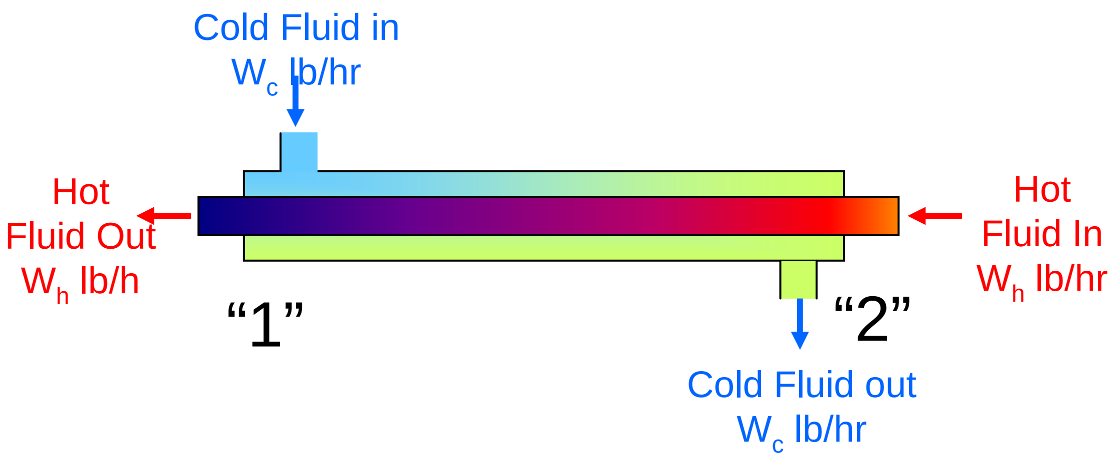
The overall coefficient of heat transfer U_0

The difference between the mixing-cup temperatures of the hot and cold fluids, Δt

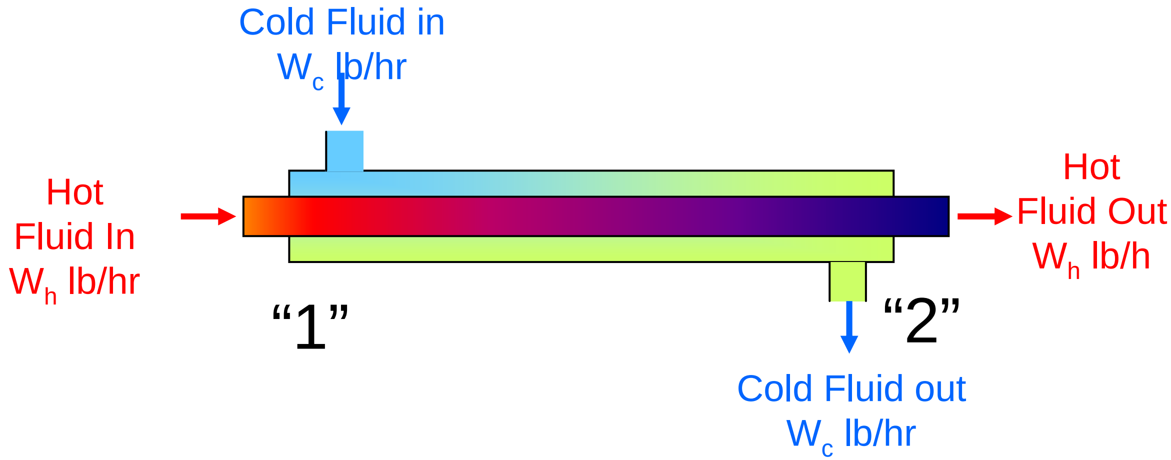
Under steady conditions, the mixing-cup temperature of the hot and cold fluids in a heat exchanger are assumed to be fixed at any cross section normal to the flow. The overall temperature difference is $\Delta t = t_h - t_c$.

Countercurrent vs. Concurrent

Countercurrent

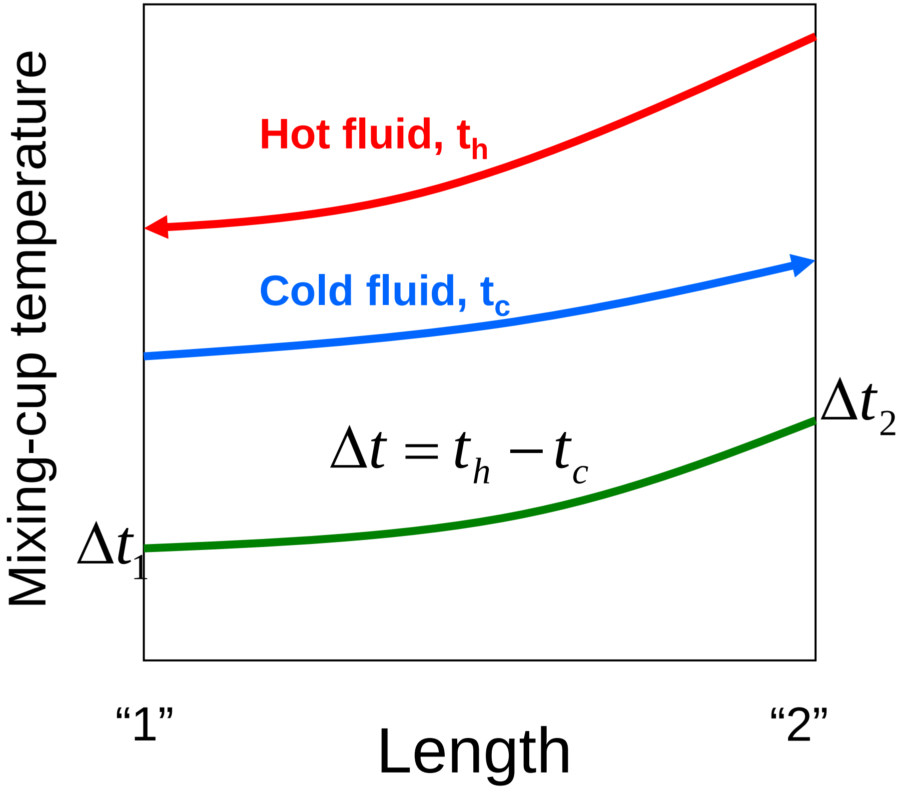


Concurrent
(Parallel-flow)

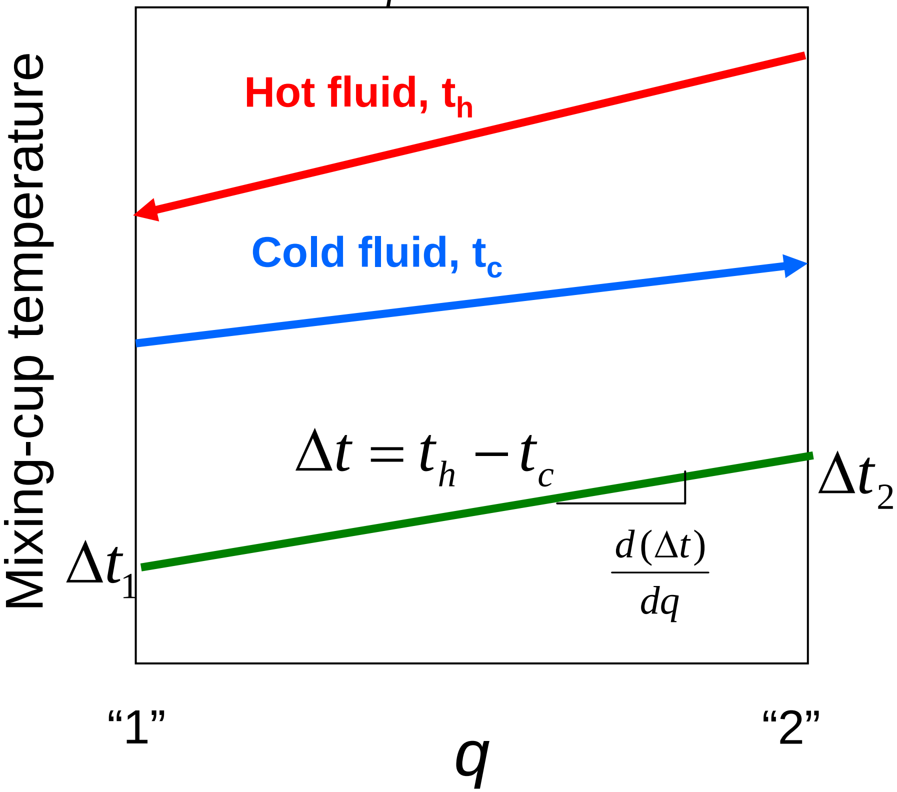


Double-pipe heat exchanger

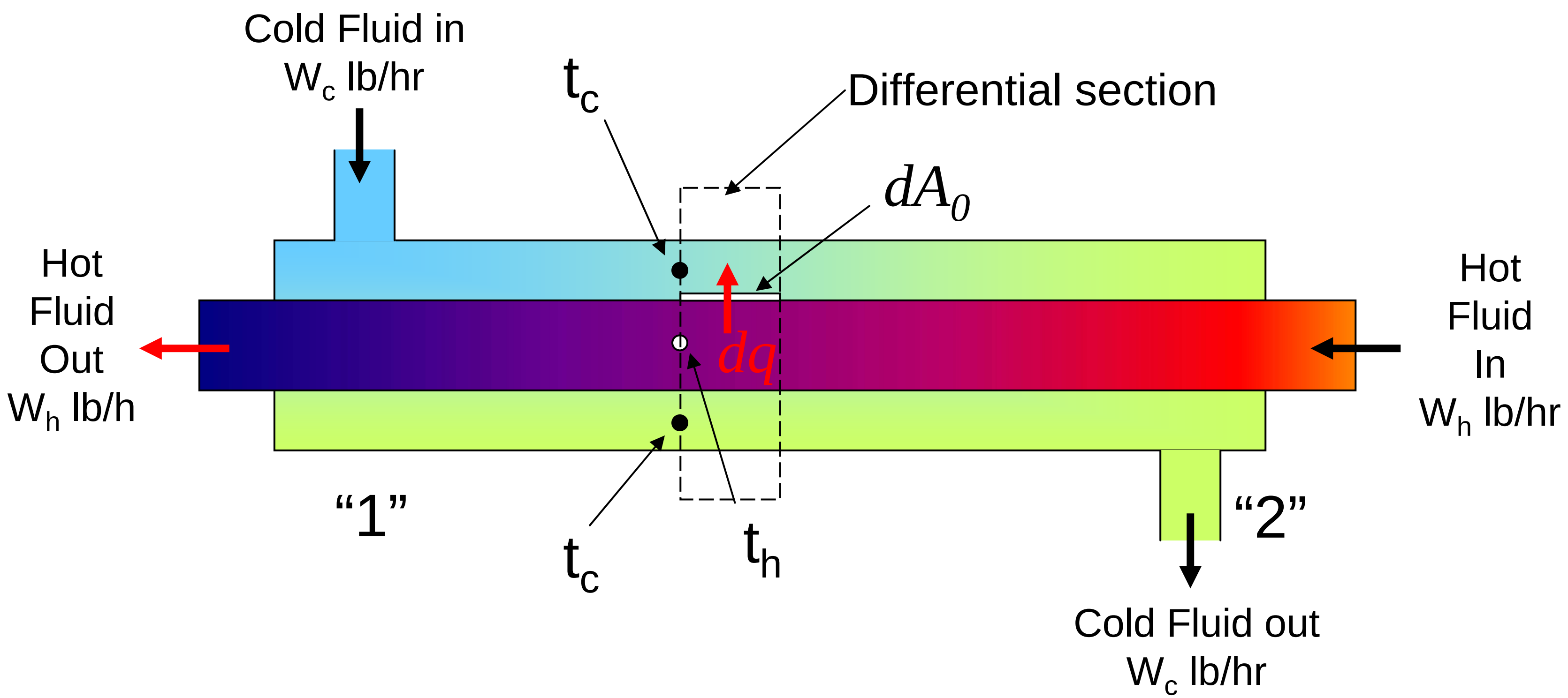
Countercurrent



If C_p*=constant*

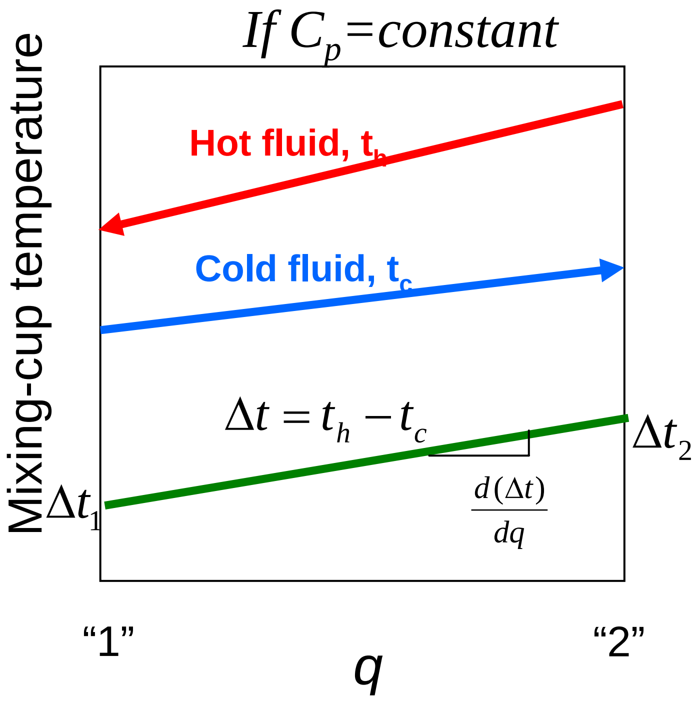


Heat balance for the differential section



$$dq = w_c C_{pc} dt_c = w_h C_{ph} dt_h = U_0 (t_h - t_c) dA_0$$

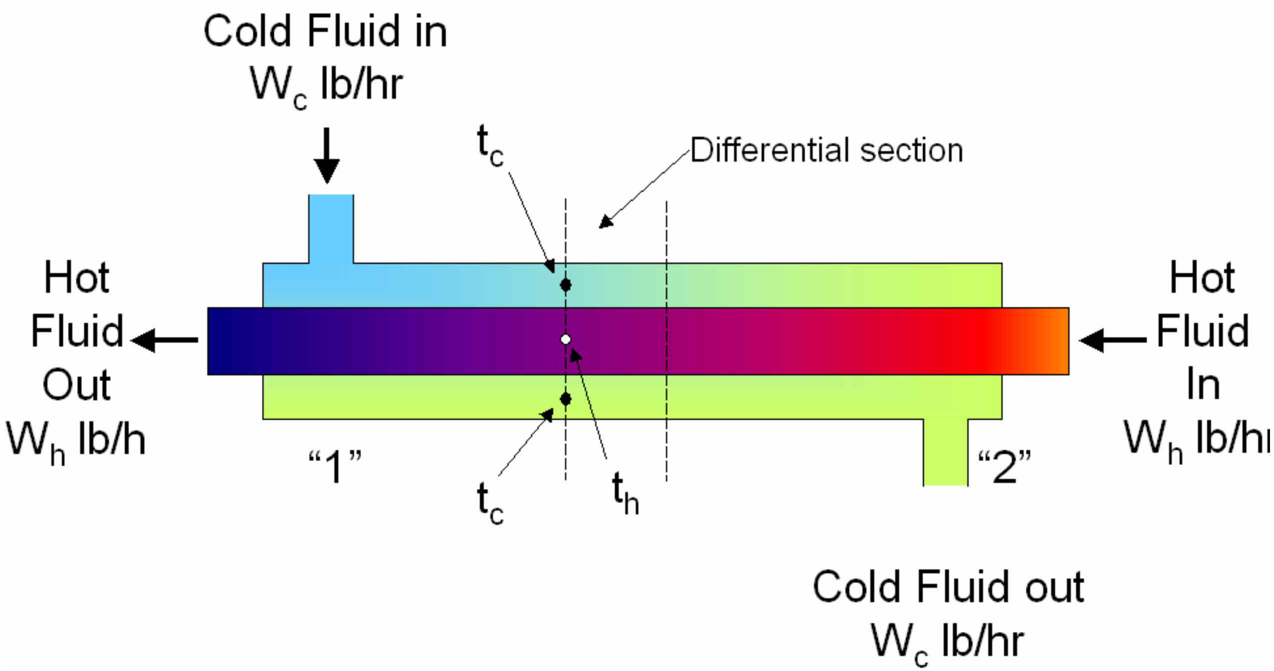
Double-pipe heat exchanger



The difference between the mixing cup temperatures, ***Δt***, is also linear w.r.t. ***q.***

$$\frac{d(\Delta t)}{dq} = \frac{\Delta t_2 - \Delta t_1}{q}$$

Heat Balance in the differential section of the exchanger



$$dq = w_c C_{pc} dt_c = w_h C_{ph} dt_h = U_0 (t_h - t_c) dA_0 = U_0 \Delta t dA_0 \quad (21\text{-}25)$$

$$\frac{d(\Delta t)}{dq} = \frac{\Delta t_2 - \Delta t_1}{q} \quad (21\text{-}26)$$

Heat Balance in the differential section of the exchanger

$dq = U_0 \Delta t dA_0$

$$\frac{d(\Delta t)}{dq} = \frac{d(\Delta t)}{U_o \Delta t dA_o} = \frac{\Delta t_2 - \Delta t_1}{q} \qquad (21\text{-}27)$$

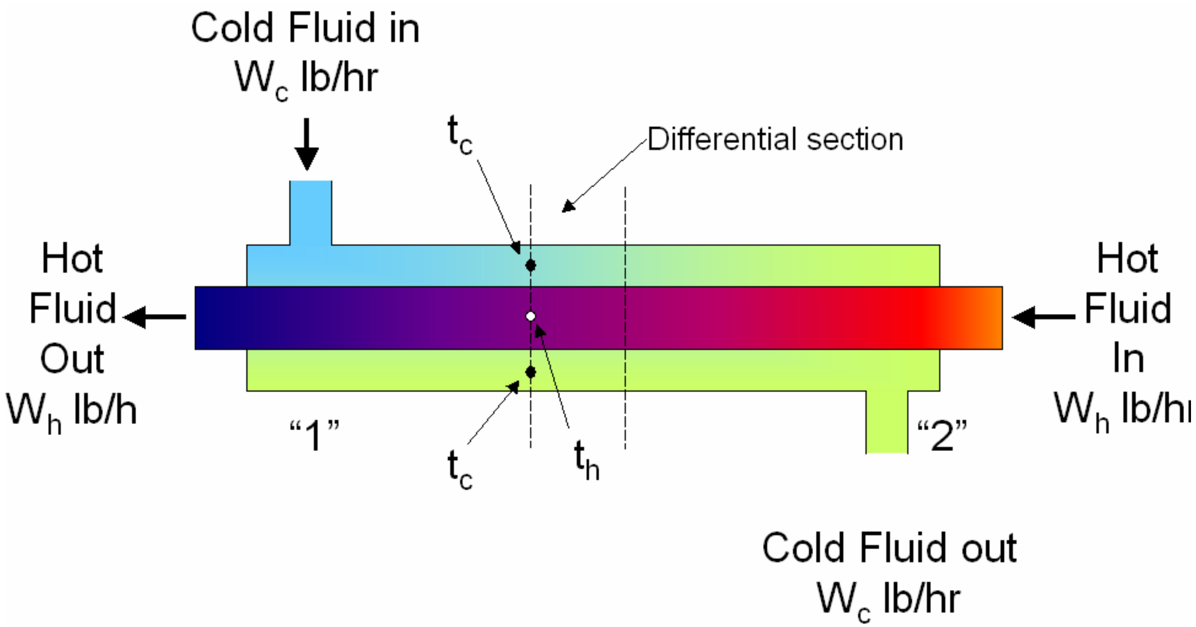
$$\int_{\Delta t_1}^{\Delta t_2} \frac{d(\Delta t)}{U_o \Delta t} = \frac{\Delta t_2 - \Delta t_1}{q} \int_0^{A_o} dA_o \qquad (21\text{-}28)$$

If U_o=constant

$$q = U_o A_o \left[\frac{\Delta t_2 - \Delta t_1}{\ln(\Delta t_2 / \Delta t_1)} \right] \qquad (21\text{-}29)$$

LMTD
Logarithmic
Mean
Temperature
Difference

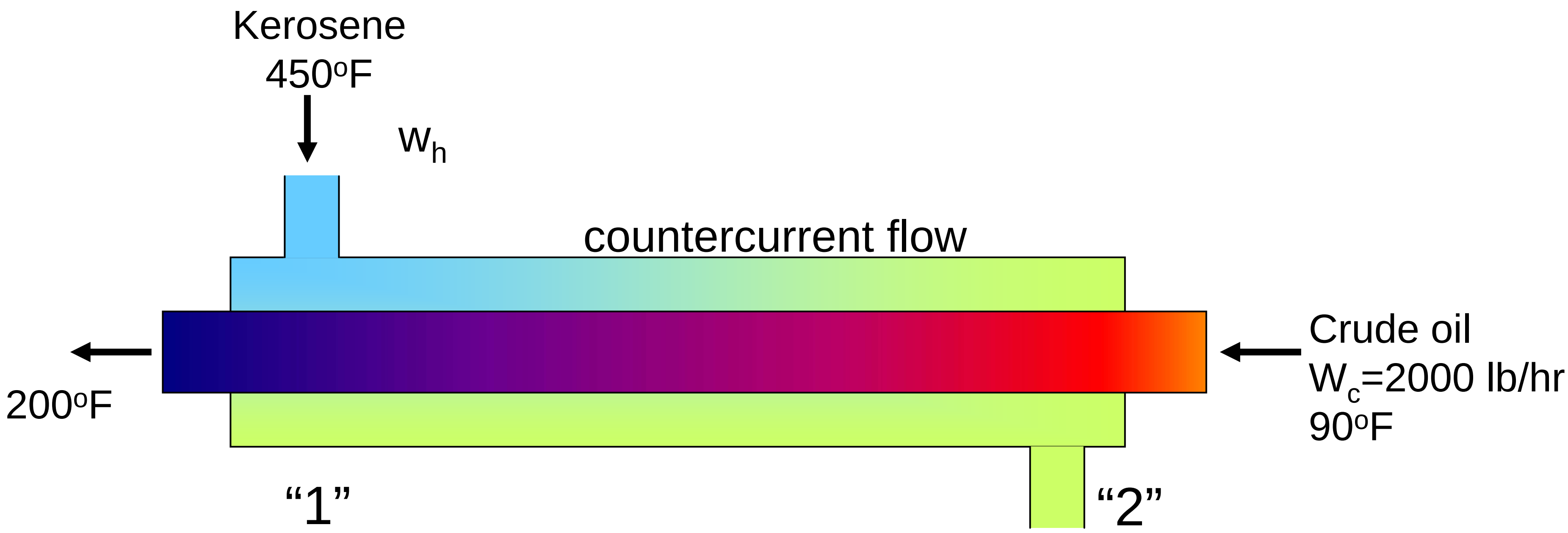
Heat Exchange in double pipe heat exchanger



$$q = U_o A_o \left[\frac{\Delta t_2 - \Delta t_1}{\ln\left(\Delta t_2 / \Delta t_1\right)} \right] = w_c C_{pc} \Delta t_c = w_h C_{ph} \Delta t_h$$

Example 21-3

If the temperature of approach (minimum temperature difference between fluids) is 20°F, determine the A_o, w_h for (a) concurrent flow and (b) countercurrent flow.



Overall coefficient U_o=80 Btu/(h)(ft^2)(°F)
Specific heat of crude oil = 0.56 Btu/(ib)(°F)
Specific heat of kerosene = 0.60 Btu/(ib)(°F)

Example 21-3 (a) Concurrent flow

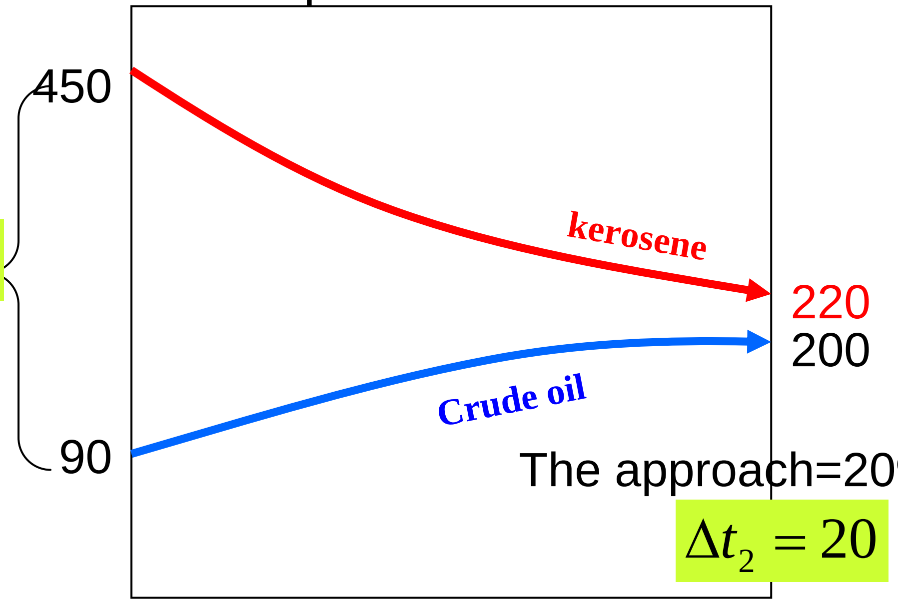
Total heat load:

$$q = w_c C_{pc} \Delta t_c = (2000)(0.56)(200-90) = 123{,}000\ Btu/h$$

$$w_h = \frac{q}{C_{ph}\Delta t_h} = \frac{123{,}000}{(0.60)(450-220)} = 891\ lb/h$$

$$A_o = \frac{q}{U_o\left[\dfrac{\Delta t_2 - \Delta t_1}{\ln(\Delta t_2/\Delta t_1)}\right]} = \frac{123{,}000}{80\left[\dfrac{360-20}{\ln(360/20)}\right]} = 13.1\ ft^2$$

The temperature distribution



$\Delta t_1 = 360$

The approach=20°F

$\Delta t_2 = 20$

Example 21-3 (b) Countercurrent flow

Total heat load:

$$q = w_c C_{pc} \Delta t_c = (2000)(0.56)(200-90) = 123{,}000 \ Btu/h$$

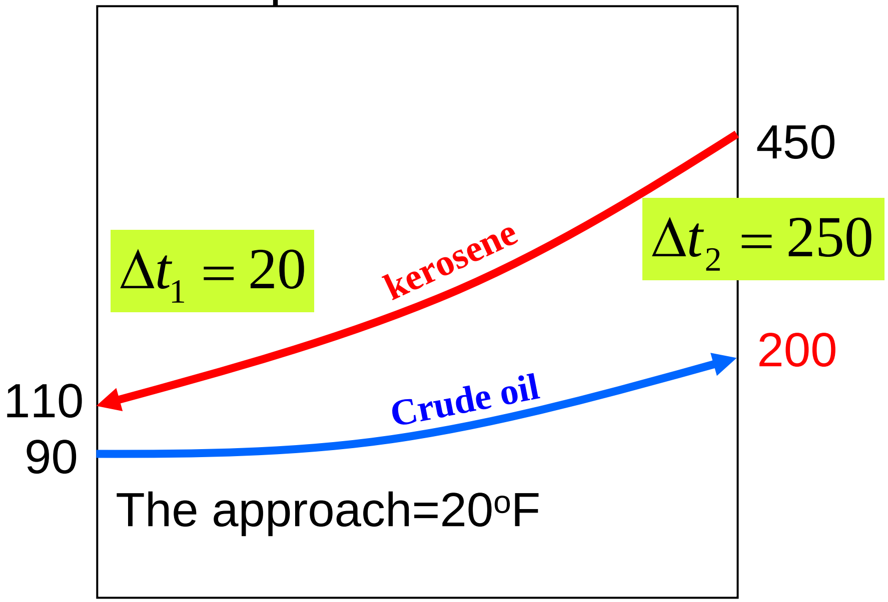
$$w_h = \frac{q}{C_{ph}\Delta t_h} = \frac{123{,}000}{(0.60)(450-110)} = 603 \ lb/h$$

$$A_o = \frac{q}{U_o\left[\dfrac{\Delta t_2 - \Delta t_1}{\ln(\Delta t_2/\Delta t_1)}\right]}$$

$$= \frac{123{,}000}{80\left[\dfrac{250-20}{\ln(250/20)}\right]}$$

$$= 16.9 \ ft^2$$

The temperature distribution



If U_o is a function of temperature

$$U_o = a + b\Delta t$$

(21-30) $$\int_{\Delta t_1}^{\Delta t_2} \frac{d(\Delta t)}{U_o \Delta t} = \frac{\Delta t_2 - \Delta t_1}{q} \int_0^{A_o} dA_o \rightarrow \int_{\Delta t_1}^{\Delta t_2} \frac{d(\Delta t)}{(a + b\Delta t)\Delta t} = \frac{\Delta t_2 - \Delta t_1}{q} A_o$$

$$\int_{\Delta t_1}^{\Delta t_2} \frac{d(\Delta t)}{(a + b\Delta t)\Delta t} = \int_{\Delta t_1}^{\Delta t_2} \left(\frac{1/a}{\Delta t} + \frac{-b/a}{a + b\Delta t} \right) d(\Delta t)$$

$$= \left[\frac{1}{a} \ln \Delta t - \frac{1}{a} \ln(a + b\Delta t) \right]_{\Delta t_1}^{\Delta t_2}$$

$$= \left[\frac{1}{a} \ln \frac{\Delta t}{a + b\Delta t} \right]_{\Delta t_1}^{\Delta t_2}$$

$$= \frac{1}{a} \ln \left(\frac{\Delta t_2}{a + b\Delta t_2} \right) \left(\frac{a + b\Delta t_1}{\Delta t_1} \right)$$

$$= \frac{1}{a} \ln \frac{U_{o1} \Delta t_2}{U_{o2} \Delta t_1} \qquad (21\text{-}31)$$

If U_o is a function of temperature

$$U_o = a + b\Delta t$$

$$U_{o2} = a + b\Delta t_2$$

$$U_{o1} = a + b\Delta t_1$$

$$U_{o2} - U_{o1} = b(\Delta t_2 - \Delta t_1)$$

$$b = \frac{U_{o2} - U_{o1}}{\Delta t_2 - \Delta t_1}$$

$$a = \frac{U_{o1}\Delta t_2 - U_{o2}\Delta t_1}{\Delta t_2 - \Delta t_1} \qquad (21\text{-}32)$$

If U_o is a function of temperature

$$U_o = a + b\Delta t$$

$$\frac{\Delta t_2 - \Delta t_1}{q} A_o = \frac{1}{a} \ln \frac{U_{o1}\Delta t_2}{U_{o2}\Delta t_1}$$

$$a = \frac{U_{o1}\Delta t_2 - U_{o2}\Delta t_1}{\Delta t_2 - \Delta t_1}$$

$$q = \frac{U_{o1}\Delta t_2 - U_{o2}\Delta t_1}{\ln \dfrac{U_{o1}\Delta t_2}{U_{o2}\Delta t_1}} A_o \qquad (21\text{-}34)$$

Heat transfer rate in double pipe heat exchanger

$U_o = const$

$$q = U_o A_o \frac{\Delta t_2 - \Delta t_1}{\ln\left(\frac{\Delta t_2}{\Delta t_1}\right)} \quad (21\text{-}29)$$

$U_o = a + b\Delta t$

$$q = \frac{U_{o1}\Delta t_2 - U_{o2}\Delta t_1}{\ln \frac{U_{o1}\Delta t_2}{U_{o2}\Delta t_1}} A_o \quad (21\text{-}34)$$

Homework

PROBLEMS

21-1

21-6

21-10

21-15

21-17

Due on November 2