Review of Heat Transfer & Fluid Flow

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ME 307: Heat Transfer Equipment Design

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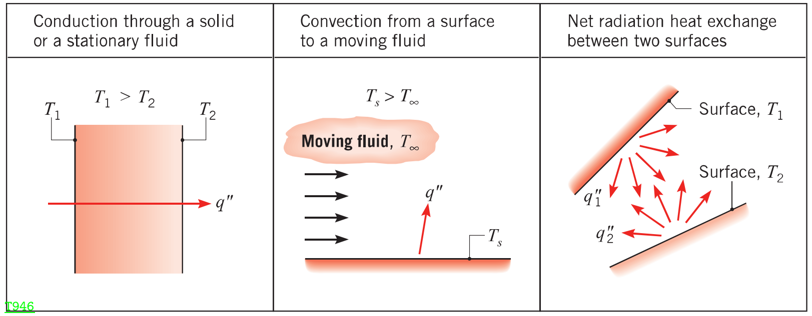
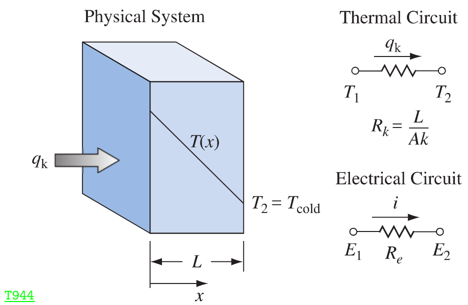
Modes of Heat Transfer



T946

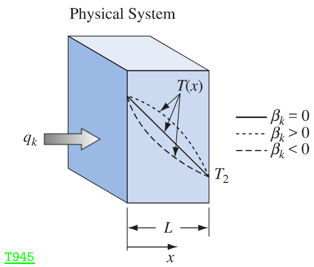
Heat Conduction Through Plain Wall



T944

Temperature distribution for steady-state conduction through a plane wall, and analogy between thermal and electrical circuits.

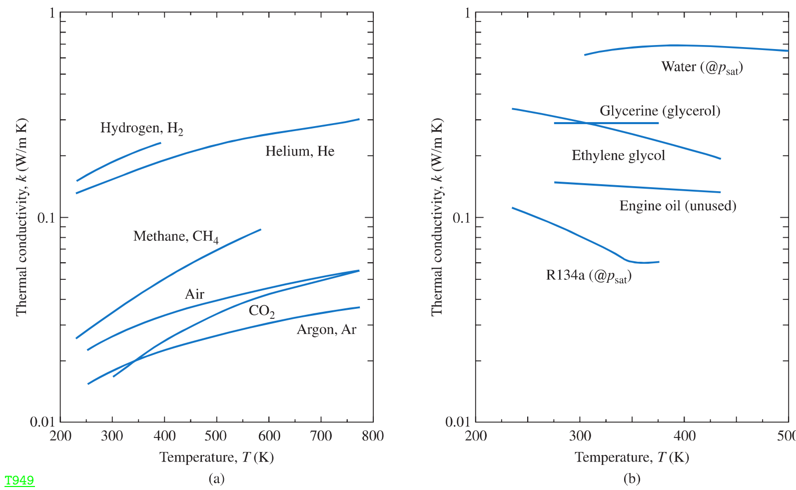
$$k(T) = k_0(1 + \beta_k T)$$



T945

Variable thermal conductivity

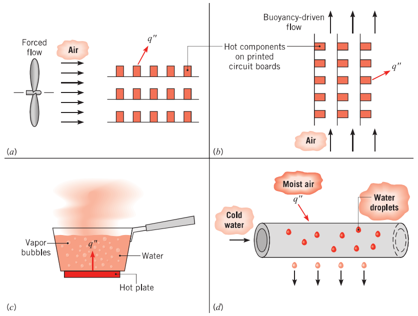
$$\dot{Q}_{cond} = kA\frac{\Delta T}{\Delta x} = UA\Delta T = \frac{\Delta T}{1/UA} = \frac{\Delta T}{R_{cond}} :\to R_{cond} = \frac{\Delta x}{kA} = \frac{1}{UA}$$



T949

Variation of thermal conductivity. (a) gases and (b) liquids.

Heat Convection



T947

(a) Forced convection. (b) Natural convection. (c) Boiling. (d) Condensation.

Process	h ($W/m^2 \cdot K$)
Free convection	
Gases	2–25
Liquids	50–1000
Forced convection	
Gases	25–250
Liquids	100–20,000
Convection with phase change	
Boiling or condensation	2500–100,000

T948

$$\dot{Q}_{conv} = hA(T_s - T_\infty) = \frac{(T_s - T_\infty)}{1/hA} = \frac{(T_s - T_\infty)}{R_{conv}} :\to R_{conv} = \frac{1}{hA}$$

Non-dimensional Numbers in Heat Convection

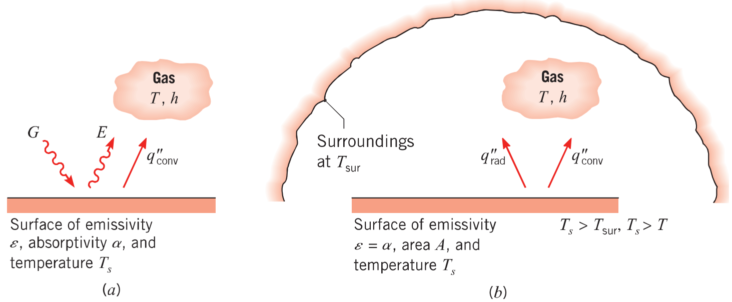
$$\frac{hL}{k} \equiv Nu = f(Re, Pr, \cdots)$$

- Reynolds number, $Re \equiv \frac{\rho VL}{\mu} = \frac{inertia\ force}{viscous\ force}$
- Prandtl number $Pr \equiv \frac{\nu}{\alpha} = \frac{c_p \mu}{k} = \frac{molecular\ diffusivity\ of\ momentum}{molecular\ diffusivity\ of\ heat}$
- Nusselt number $Nu \equiv \frac{hL}{k}$
- Peclet number $Pe \equiv RePr$
- Graetz number, $Gz \equiv Pe\frac{d}{x}$

Heat Radiation

$$\dot{Q}_{rad} = \epsilon\sigma A(T_2^4 - T_1^4)$$

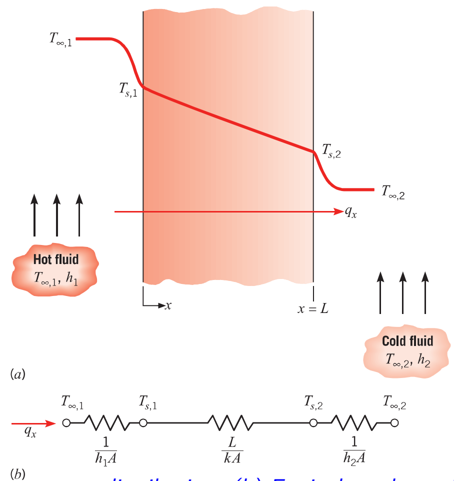
$$\dot{Q}_{rad} = h_r A(T_s - T_{sur}) \quad : \quad h_r \equiv \epsilon\sigma(T_s + T_{sur})(T_s^2 + T_{sur}^2)$$



T954

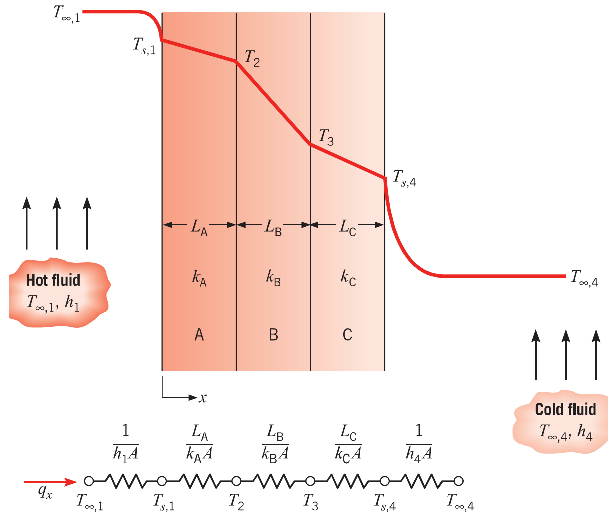
Radiation exchange: (a) at a surface and (b) between a surface and large surroundings.

Heat Conduction + Convection through Plain Wall



T950

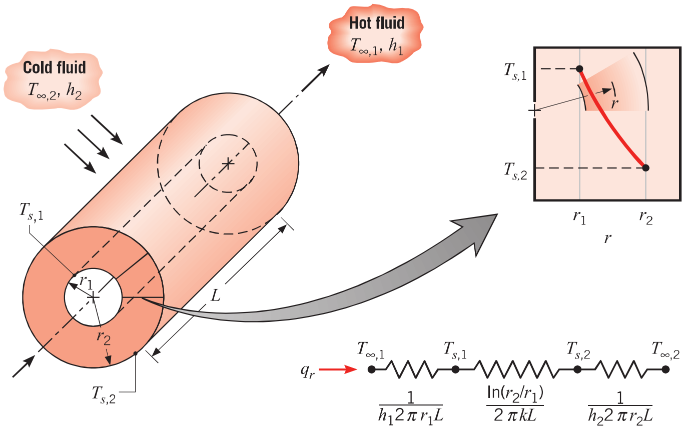
(a) Temperature distribution. (b) Equivalent thermal circuit.



T951

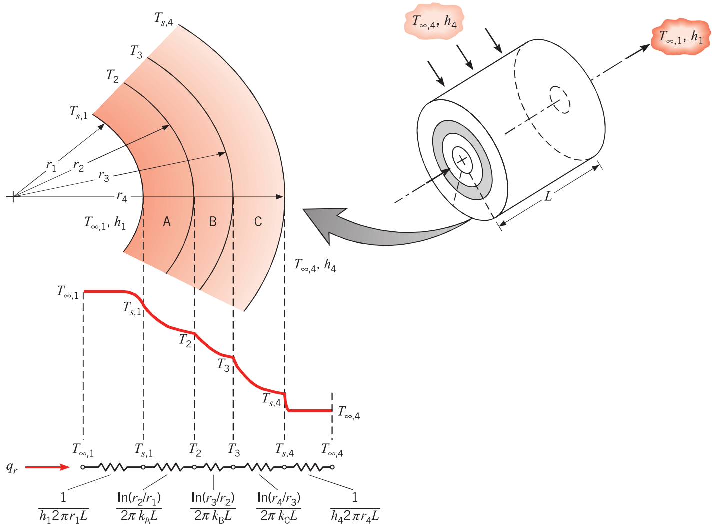
Equivalent thermal circuit for a series composite wall.

Heat Conduction + Convection: Hollow Cylinder



T952

Hollow cylinder with convective surface conditions.



T953

Temperature distribution for a composite cylindrical wall.

Simplified Equations for Air: Free Convection

Surface	**Laminar,** $10^4 < Gr_f Pr_f < 10^9$	**Turbulent,** $Gr_f Pr_f > 10^9$
Vertical plane or cylinder	$h = 1.42\left(\frac{\Delta T}{L}\right)^{1/4}$	$h = 1.31(\Delta T)^{1/3}$
Horizontal cylinder	$h = 1.32\left(\frac{\Delta T}{d}\right)^{1/4}$	$h = 1.24(\Delta T)^{1/3}$
Horizontal plate:		
Heated plate facing upward or cooled plate facing downward	$h = 1.32\left(\frac{\Delta T}{L}\right)^{1/4}$	$h = 1.52(\Delta T)^{1/3}$
Heated plate facing downward or cooled plate facing upward	$h = 0.59\left(\frac{\Delta T}{L}\right)^{1/4}$	

T661

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Flow inside Circular Duct

Sieder & Tate: [Laminar flow]

$$Nu_m = 1.86(Gz)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}; \quad Gz = \frac{RePr}{L/D}$$

$0.48 < Pr < 16700;\ 0.044 < \mu_b/\mu_w < 9.75;\ (Gz)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} > 2$

Dittus-Boelter: [Turbulent flow]

$$Nu_m = 0.023Re^{0.8}Pr^n; \quad n = \begin{cases} 0.4 & : heating \\ 0.3 & : cooling \end{cases}$$

$0.7 < Pr < 16700;\ Re > 10000;\ L/D > 60$

Holman Ex. 10-1 ▷ Hot water at 98°C flows through a 2-in schedule 40 horizontal steel pipe [k = 54 W/m°C] and is exposed to atmospheric air at 20°C. The water velocity is 25 cm/s. Calculate the overall heat transfer coefficient for this situation, based on the outer area of pipe. [U_o = 7.84 W/m^2°C]

Heat Exchanger: Fouling

During operation, heat exchangers become fouled with an accumulation of deposits of one kind or another on the heat transfer surface areas. Major categories of fouling:

1. scaling or precipitation fouling
2. particulate fouling
3. chemical reaction fouling
4. corrosion fouling
5. biological fouling
6. solidification fouling

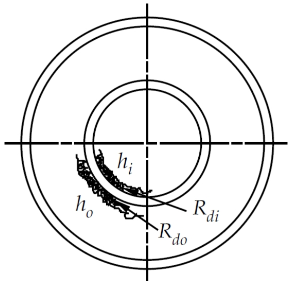
Economic fluid velocity (m/s)

Fluid	Min	Max
Water	1.40	3.00
Engine oil	1.40	2.80
Acetone, alcohols	1.55	3.29
Ammonia	1.68	3.60
Benzene, toluene	1.46	3.17
Octane	1.55	3.38
R-134a	1.37	2.90
NaCl sol. (20%)	1.31	2.86
Ethylene glycol sol.	1.37	2.96

T1013

Fouling factor (m^{2}K/W)

Engine oil	0.00088
Engine exhaust gas	0.00176
Steam	0.00088
Refrigerant (liq)	0.00176
Refrigerant (gas)	0.00035
Air, CO, CO_2, NH_3	0.00176
Ethylene glycol sol.	0.00035
Hydraulic fluid	0.00176
Water (city, well)	0.00035
Water (distilled)	0.00009



T1248

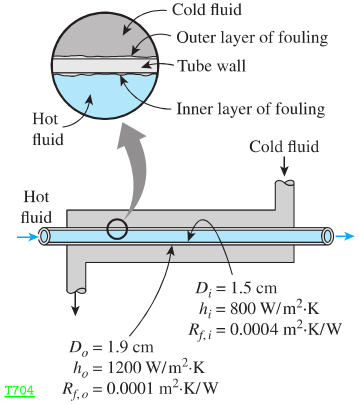
$$\dot{Q} = U_o A_o \Delta T_{LM}$$

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} = R = \frac{1}{h_i A_i} + \frac{R_{d,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{R_{d,o}}{A_o} + \frac{1}{h_o A_o}$$

$$\frac{1}{U_o} \simeq \frac{1}{h_i} + \frac{1}{h_o}; \qquad \frac{1}{U} \simeq \frac{1}{U_o} + R_{d,i} + R_{d,o}$$

Cengel Ex. 11-2 ▷ Effect of fouling on overall heat transfer coefficient. Double-pipe (shell-and-tube) heat exchanger made of steel, k = 15.1 W/m-K, inner-diameter of outside shell is 3.2 cm.

[U_o = 315 W/m^2°C, $U_{o,new}$ = 389.5 W/m^2°C]



T704

Condensation Correlations

$$Rate\ of\ condensation,\ \dot{m} = \frac{A h_m (T_v - T_w)}{h_{fg}}; \qquad Re = \frac{4\dot{m}}{\mu_l P}$$

h_{fg} evaluated at T_v

$$Wetted\ perimeter,\ P = \begin{cases} \pi D & \text{for vertical tube of outside diameter } D \\ 2L & \text{for horizontal tube of length } L \\ w & \text{for vertical or inclined plate of width } w \end{cases}$$

Condensation on vertical surfaces: [Laminar (Re <1800)]

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

- Properties are evaluated at film temperature, $T_f = \frac{1}{2}(T_w + T_v)$
- Flow is laminar if $Re < 1800$.

Condensation on inclined surfaces: [Laminar (Re <1800)]

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

Condensation on horizontal tube: [Laminar (Re <1800)]

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

Comparison of vertical tube of length L and horizontal tube of diameter D:

$$\frac{h_{m,vert}}{h_{m,horz}} = 1.56 \left[\frac{D}{L} \right]^{1/4}$$

If $L = 100D \longrightarrow h_{m,horz} \simeq 2.0 h_{m,vert}$. For condensation, horizontal tube arrangements are generally preferred.

Condensation on horizontal tube banks:

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

Condensation inside horizontal tube:

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

$h'_{fg} = h_{fg} + \frac{3}{8}c_{p,l}(T_v - T_w)$; $Re_v = \frac{\rho_v u_v D}{\mu_v} \leqslant 35000$

Turbulent film-wise condensation: vertical tube

$$h_m \left[\frac{\mu_l^2}{k_l^3\rho_l^2 g}\right]^{1/3} = 0.0077 Re^{0.4}$$

Ozikik Ex. 10.1 ▷ Air free saturated steam at 65°C, P = 25.03 kPa condenses on the outer surface of a 2.5 cm OD, 3-m long vertical tube maintained at constant temperature of 35°C by the flow of the cooling water. Estimate average heat transfer coefficient and the rate of condensate flow at the bottom of the tube.

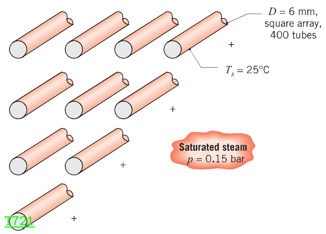
[$\dot{m}_V = 0.01124$ kg/s]

Ozikik Ex. 10.2 ▷ Redo Ozisik Ex. 10.1 assuming horizontal tube.

[$\dot{m}_H = 0.02386$ kg/s]

Dewitt Ex. 10-4 ▷ The tube bank of a steam condenser consists of a square array of 400 tubes, each 6 mm in diameter. If horizontal, unfinned tubes are exposed to saturated steam at a pressure of 0.15 bar and the tube surface is maintained at 25°C, what is the rate at which steam is condensed per unit length of the tube bank?

[$\dot{m} = 0.474$ kg/s]

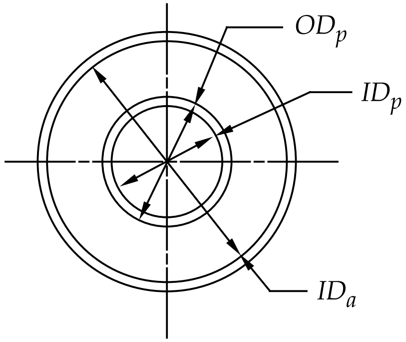


Flow Parameters: Flow inside Tubes/Ducts

- Turbulent flow inside circular duct, if $Re_f \geqslant 2300$.
- Pressure drop, $\Delta P = f\frac{L}{D}(\frac{1}{2}\rho u_m^2)$
- Pumping power, $\dot{W} = \dot{V}\Delta P = V_{av}A\Delta P$
- $f = \begin{cases} \xi_{corr}\ [64/Re_f] & \textit{for laminar flow} \\ (1.82\log_{10} Re_f - 1.64)^{-2} & \textit{for turbulent flow} \end{cases}$

$Re_f \equiv \frac{\rho u_m D_h}{\mu}$ = Reynolds number for friction analysis
$Re_Q \equiv \frac{\rho u_m D_e}{\mu}$ = Reynolds number for heat transfer analysis
f = (Darcy) friction factor
$D_h \equiv 4A/P_{wetted}$ = hydraulic diameter
$D_e \equiv 4A/P_{heat}$ = equivalent diameter
u_m = mean flow velocity

$\Rightarrow Re_Q = Re_f\left[\frac{D_e}{D_h}\right]$



T761

- **Pipe area:**
  - $D_e = D_h = ID_p$
  - $\Delta P = f \frac{L}{ID_p}(\frac{1}{2}\rho_p V_p^2), \quad \xi_{corr} = 1$
- **Annular area:**
  - $D_h = ID_a - OD_p, \quad D_e = \frac{ID_a^2 - OD_p^2}{OD_p}$
  - $\frac{1}{\xi_{corr}} = \frac{1+\kappa^2}{(1-\kappa)^2} + \frac{1+\kappa}{(1-\kappa)\ln(\kappa)}, \; \kappa = OD_p/ID_a$
  - $\Delta P = (f\frac{L}{D_h} + 1)(\frac{1}{2}\rho_a V_a^2)$

Penoncello Ex. 5-10 ▷ The outer pipe of a double pipe heat exchanger is 4-in schedule 40 commercial steel. The inside tube is 3 std type M copper. Water at an average temperature of 30°C is flowing in the annulus at a rate of 10 m^3/h. Determine the Reynolds numbers of the annular flow for use in hydraulic analysis and heat transfer analysis.

[$Re_f = 2.43 \times 10^5, \; Re_e = 5.56 \times 10^5$]