

8. Internal Forced Convection

8.1. General Procedure

- Find fluid properties from Appendix 1 at bulk mean temperature $T_b = (T_i + T_e)/2$
 - $\rho, \mu, k, c_p, Pr, \nu$
- Determine mean velocity V_{avg}
- Determine the type of flow (laminar or turbulent)
 - Laminar: $Re < 2300$
 - Turbulent: $Re > 4000$
- Determine the Nusselt number, Nu , using the appropriate correlation
 - Check if $l_{h,laminar}$ and $l_{t,laminar}$ is less than L . If so, use Table 1
 - Else, use empirical correlations
- Determine the heat transfer coefficient h using Nu , k , and A_s

8.2. Variable Definitions

- Nu : Nusselt number
- Re : Reynolds number
- Pr : Prandtl number
- μ : Dynamic viscosity
- ν : Kinematic viscosity
- k : Thermal conductivity
- h : Convection heat transfer coefficient
- D_h : Hydraulic diameter
- A_s : Surface area
- A_c : Cross-sectional area
- V_{avg} : Average velocity
- T_b : Bulk mean temperature
- T_i : Inlet temperature
- T_e : Exit temperature
- \dot{m} : Mass flow rate
- \dot{q} : Heat flux
- ΔT_{lm} : Log mean temperature difference

8.3. Formulas

8.3.1. General Formulas

$$\begin{aligned}\dot{m} &= \rho V_{avg} A_c \\ Re &= \frac{\rho V_{avg} D_h}{\mu} = \frac{V_{avg} D_h}{\nu} \\ D_h &= \frac{4A_c}{\text{Perimeter}} = D|_{\text{circular}} = a|_{\text{square}} \\ &= \frac{2ab}{a+b} \Big|_{\text{rectangular}} = \frac{4ab}{a+b} \Big|_{\text{channel}} \\ Nu &= \frac{hD_h}{k} \\ A_s &= \pi DL|_{\text{circular}} = 4ab|_{\text{rectangular}} \\ A_c &= \pi \frac{D^2}{4} |_{\text{circular}} = ab|_{\text{rectangular}} \\ l_{h,laminar} &= 0.05 Re D_h \\ l_{t,laminar} &= 0.05 Re Pr D_h = Pr l_{h,laminar} \\ l_{h,turbulent} &\approx l_{t,turbulent} = 10 D_h\end{aligned}$$

8.3.2. Constant \dot{q}

$$\begin{aligned}T_e &= T_i + \frac{\dot{q}}{\dot{m} c_p} \\ \dot{q} &= h(T_s - T_b)\end{aligned}$$

8.3.3. Constant T_s

$$\begin{aligned}T_e &= T_s - (T_s - T_i) \exp\left(-\frac{hA_s}{\dot{m} c_p}\right) \\ T_s &= \frac{T_e - T_i \exp\left(-\frac{\dot{m} c_p}{hA_s}\right)}{1 - \exp\left(-\frac{\dot{m} c_p}{hA_s}\right)} \\ \dot{Q} &= hA_s \Delta T_{lm} \\ T_{lm} &= \frac{T_i - T_e}{\ln[(T_s - T_e)/(T_s - T_i)]}\end{aligned}$$

8.3.4. Correlations for Nu

For fully developed laminar flow, use Table 1.

For entry region in a circular tube where $T_s = \text{constant}$, use:

$$(\text{Edwards et al., 1979}) \quad Nu = 3.66 + \frac{0.0658(D/L)RePr}{1 + 0.04[(D/L)RePr]^{2/3}}$$

For entry region in a circular tube where the difference between T_s and T_b is large, use:

$$\begin{aligned}(\text{Sieder and Tate, 1936}) \quad Nu &= 1.86 \left(\frac{RePrD}{L} \right)^{1/3} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} \\ 0.6 &< Pr < 5, \quad 0.0044 < \frac{\mu_b}{\mu_s} < 9.75\end{aligned}$$

All properties for Sieder and Tate should be evaluated at T_b except μ_s which should be evaluated at T_s .

For entry region between two isothermal parallel plates,
use:

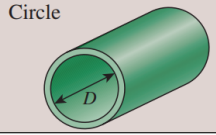
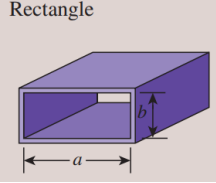
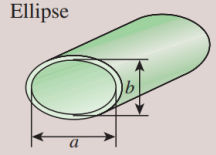
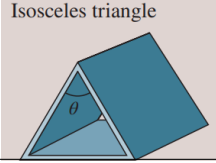
$$\text{(Edwards et al., 1979) } \text{Nu} = 7.54 + \frac{0.03(D_h/L)\text{RePr}}{1 + 0.016[(D_h/L)\text{RePr}]^{2/3}}$$
$$\text{Re} \leq 2800$$

For turbulent flow in a circular tube, use:

$$\text{(Dittus-Boelter, 1930) } \text{Nu} = 0.023\text{Re}^{0.8}\text{Pr}^n$$
$$n = 0.4 \text{ (Heating), } \quad n = 0.3 \text{ (Cooling)}$$

Tables

Table 1: Nusselt number and friction factor for fully developed laminar flow in tubes of various cross sections ($D_h = 4A_c/P$, $Re = V_{avg}D_h/\nu$, and $Nu = hD_h/k$) (**Table 8-1 in textbook**)

Tube Geometry	a/b or θ°	Nu		f
		$T_s = \text{constant}$	$\dot{q}_s = \text{constant}$	
Circle 	—	4.36	3.66	64/Re
Rectangle 	a/b			
	1	2.98	3.61	56.92/Re
	2	3.39	4.12	62.20/Re
	3	3.96	4.79	68.36/Re
	4	4.44	5.33	72.92/Re
	6	5.14	6.05	78.80/Re
	8	5.60	6.49	82.32/Re
	∞	7.54	8.24	96.00/Re
Ellipse 	a/b			
	1	3.66	4.36	64.00/Re
	2	3.74	4.56	67.28/Re
	4	3.79	4.88	72.96/Re
	8	3.72	5.09	76.60/Re
	16	3.65	5.18	78.16/Re
Isosceles triangle 	θ°			
	10	1.61	2.45	50.80/Re
	30	2.26	2.91	52.28/Re
	60	2.47	3.11	53.32/Re
	90	2.34	2.98	52.60/Re
	120	2.00	2.68	50.96/Re

9. Natural Convection

9.1. General Procedure

9.1.1. Over Surfaces

1. Find Rayleigh number, Ra_L , using fluid properties at film temperature $T_f = (T_s + T_\infty)/2$
2. Use the appropriate correlation in Table 2 to find the Nusselt number, Nu
3. Determine the heat transfer coefficient h using Nu , k , and L_c

9.1.2. In Enclosures

1. Find Rayleigh number, Ra_L , using fluid properties at average temperature $T_{avg} = (T_1 + T_2)/2$ where T_1 and T_2 are the temperatures of the hot and cold surfaces respectively.
2. Use the appropriate correlation to find the Nusselt number, Nu
3. Determine the heat transfer coefficient h using Nu , k , and L_c

9.2. Variable Definitions

- Ra_L : Rayleigh number
- Gr_L : Grashof number
- T_s : Surface temperature
- T_∞ : Ambient temperature
- T_f : Film temperature
- L_c : Characteristic length
- β : Coefficient of volume expansion
- ν : Kinematic viscosity
- α : Thermal diffusivity
- g : Gravitational acceleration

9.3. Formulas

$$Ra_L = Gr_L Pr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} Pr$$

$$\beta = \frac{1}{T}, \quad \text{for ideal gases}$$

$$h = \frac{kNu}{L_c}$$

See Table 2 for L_c and Nu .

9.3.1. Over Surfaces

For convection

$$\dot{Q} = hA_s(T_s - T_\infty)$$

Use Table 2 to find Nu .

9.3.2. In Rectangular Enclosures

For convection in rectilinear enclosures,

$$\dot{Q} = hA_s(T_1 - T_2)$$

where T_1 and T_2 are the temperatures of the hot and cold surfaces respectively.

In **horizontal rectangular enclosures** ($L_c = L$, where L is the gap between plates),

$$Nu = 1 + 1.44 \left[1 - \left(\frac{1708}{Ra_L} \right) \right]^+ + \left[\frac{Ra_L}{18} - 1 \right]^+ \\ Ra_L < 10^8 \text{ (gases)}, \quad Ra_L < 10^5 \text{ (liquids)}$$

For large aspect ratios ($H/L \geq 12$), this equation (Hollands et al., 1976) correlates experimental data extremely well for tilt angles up to 70° . $[\]^+$ indicates that if the quantity in the bracket is negative, it should be set equal to zero.

$$Nu_L = 1 + 1.44 \left[1 - \left(\frac{1708}{Ra_L \cos \theta} \right) \right]^+ + \left[\frac{Ra_L \cos \theta}{18} - 1 \right]^+ \\ Ra_L < 10^8 \text{ (gases)}, \quad Ra_L < 10^5 \text{ (liquids)}, \\ 0 < \theta < 70^\circ, \quad \frac{H}{L} \geq 12$$

In **vertical rectangular enclosures** ($L_c = L$, where L is the gap between plates),

$$Nu_L = 0.18 \left(\frac{Pr}{0.2 + Pr} Ra_L \right)^{0.29} \\ \frac{Pr}{0.2 + Pr} > 10^3, \quad 1 < \frac{H}{L} < 2$$

or

$$Nu_L = 0.22 \left(\frac{Pr}{0.2 + Pr} Ra_L \right)^{0.28} \left(\frac{H}{L} \right)^{-0.25} \\ Ra_L < 10^{10}, \quad 2 < \frac{H}{L} < 10$$

or

$$Nu_L = 0.42 \left(\frac{Pr}{0.2 + Pr} Ra_L \right)^{0.25} Pr^{0.012} \left(\frac{H}{L} \right)^{-0.3} \\ 1 < Pr < 2 \times 10^4, \quad 10^4 < Ra_L < 10^7, \quad 10 < \frac{H}{L} < 40$$

9.3.3. In Concentric Horizontal Cylinders

In **concentric horizontal cylinders** ($L_c = (D_o - D_i)/2$, where D_o and D_i are the outer and inner diameters respectively),

$$\dot{Q}_{cylinder} = \frac{2\pi kNu}{\ln(D_o/D_i)} (T_i - T_o)$$

where T_i and T_o are the temperatures of the inner and outer surfaces respectively.

$$Nu = \max \left\{ 1, 0.386 \left(\frac{Pr}{0.861 + Pr} \right)^{0.25} (F_{cyl} Ra_L)^{0.25} \right\}$$

9.3.4. Combined Natural and Forced Convection

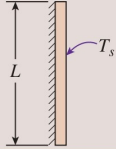
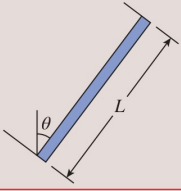


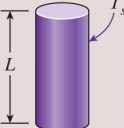
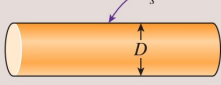
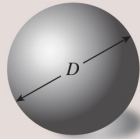
For combined natural and forced convection,

$$\text{Nu}_{\text{Overall}} = (\text{Nu}_{\text{Forced}}^n \pm \text{Nu}_{\text{Natural}}^n)^{1/n}$$

where the plus sign is for assisting flows and the minus sign is for opposing flows. $n = 3.5$ for horizontal plates and $n = 4$ for cylinders and spheres. Else use $n = 3$.

Tables

Table 2: Empirical correlations for the average Nusselt number for natural convection over surfaces (**Table 9-1 in textbook**)

TABLE 9-1			
Empirical correlations for the average Nusselt number for natural convection over surfaces			
Geometry	Characteristic Length L_c	Range of Ra	Nu
Vertical plate 	L	10^4 – 10^9 10^9 – 10^{13} Entire range	$Nu = 0.59 Ra_L^{1/4}$ (9-19) $Nu = 0.1 Ra_L^{1/3}$ (9-20) $Nu = \left\{ 0.825 + \frac{0.387 Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2$ (9-21) (complex but more accurate)
Inclined plate 	L		Use vertical plate equations for the upper surface of a cold plate and the lower surface of a hot plate Replace g with $g \cos \theta$ for $0 < \theta < 60^\circ$
Horizontal plate (surface area A and perimeter p) (a) Upper surface of a hot plate (or lower surface of a cold plate) 	A_s/p	10^4 – 10^7 10^7 – 10^{11}	$Nu = 0.54 Ra_L^{1/4}$ (9-22) $Nu = 0.15 Ra_L^{1/3}$ (9-23)
(b) Lower surface of a hot plate (or upper surface of a cold plate) 		10^5 – 10^{11}	$Nu = 0.27 Ra_L^{1/4}$ (9-24)
Vertical cylinder 	L		A vertical cylinder can be treated as a vertical plate when $D \geq \frac{35L}{Gr_L^{1/4}}$
Horizontal cylinder 	D	$Ra_D \leq 10^{12}$	$Nu = \left\{ 0.6 + \frac{0.387 Ra_D^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right\}^2$ (9-25)
Sphere 	D	$Ra_D \leq 10^{11}$ $(Pr \geq 0.7)$	$Nu = 2 + \frac{0.589 Ra_D^{1/4}}{[1 + (0.469/Pr)^{9/16}]^{4/9}}$ (9-26)

11. Heat Exchangers

11.1. General Procedure

11.1.1. Log Mean Temperature Difference Method

1. See if \dot{Q} is given or can be obtained by energy balance

$$\dot{Q} = \dot{m}c_p(T_{\text{out},h} - T_{\text{in},h}) = \dot{m}c_p(T_{\text{in},c} - T_{\text{out},c})$$

2. Find the log mean temperature difference, ΔT_{lm}
3. Find the heat transfer coefficient, U . For multi-pass shell-and-tube heat exchangers, use $A_s = NA_{s,i}$ where N is the number of passes and $A_{s,i}$ is the surface area of one pass.

11.1.2. ε -NTU Method

1. Find NTU using the given heat exchanger type and c
2. Find ε using the given heat exchanger type and NTU
3. Find \dot{Q} using ε and \dot{Q}_{max}
- 4.

11.2. Variable Definitions

- \dot{Q} : Heat transfer rate
- \dot{Q}_{max} : Maximum heat transfer rate
- $T_{\text{in},h}$: Hot inlet temperature
- $T_{\text{out},h}$: Hot outlet temperature
- $T_{\text{in},c}$: Cold inlet temperature
- $T_{\text{out},c}$: Cold outlet temperature
- U : Overall heat transfer coefficient
- ΔT_{lm} : Log mean temperature difference
- F : Correction factor
- ε : Effectiveness
- \dot{Q}_{max} : Maximum heat transfer rate
- C_c : Cold heat capacity rate
- C_h : Hot heat capacity rate
- C : Heat capacity rate ratio
- NTU: Number of transfer units

11.3. Formulas

11.3.1. Log Mean Temperature Difference Method

For a heat exchanger,

$$\dot{Q} = \dot{Q}_c = -\dot{Q}_h$$

Overall heat transfer coefficient,

$$\frac{1}{UA_s} = \frac{1}{U_i A_{s,i}} = \frac{1}{U_o A_{s,o}} = R = \frac{1}{h_i A_{s,i}} + R_{\text{wall}} + \frac{1}{h_o A_{s,o}}$$

Log mean temperature difference,

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

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

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

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

Correction factor,

$$\Delta T_{\text{lm}} = F \Delta T_{\text{lm, CF}}$$

where $\Delta T_{\text{lm, CF}}$ is the log mean temperature difference for counterflow and F is the correction factor which can be found in Figure 1.

11.3.2. ε -NTU Method

$$C_{\text{min}} = \min(\dot{m}_c c_{p,c}, \dot{m}_h c_{p,h})$$

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

$$\dot{Q} = C_c(T_{\text{out},c} - T_{\text{in},c}) = C_h(T_{\text{in},h} - T_{\text{out},h})$$

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}}$$

if $C_c = C_{\text{min}}$,

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{T_{\text{out},c} - T_{\text{in},c}}{T_{\text{in},h} - T_{\text{in},c}}$$

if $C_h = C_{\text{min}}$,

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{T_{\text{in},h} - T_{\text{out},h}}{T_{\text{in},h} - T_{\text{in},c}}$$

NTU is defined as,

$$\text{NTU} = \frac{UA_s}{C_{\text{min}}} = \frac{UA_s}{\dot{m}c_{p,\text{min}}}$$

$$c = \frac{C_{\text{min}}}{C_{\text{max}}}$$

The effectiveness, ε , can be found using Table 3.

Tables

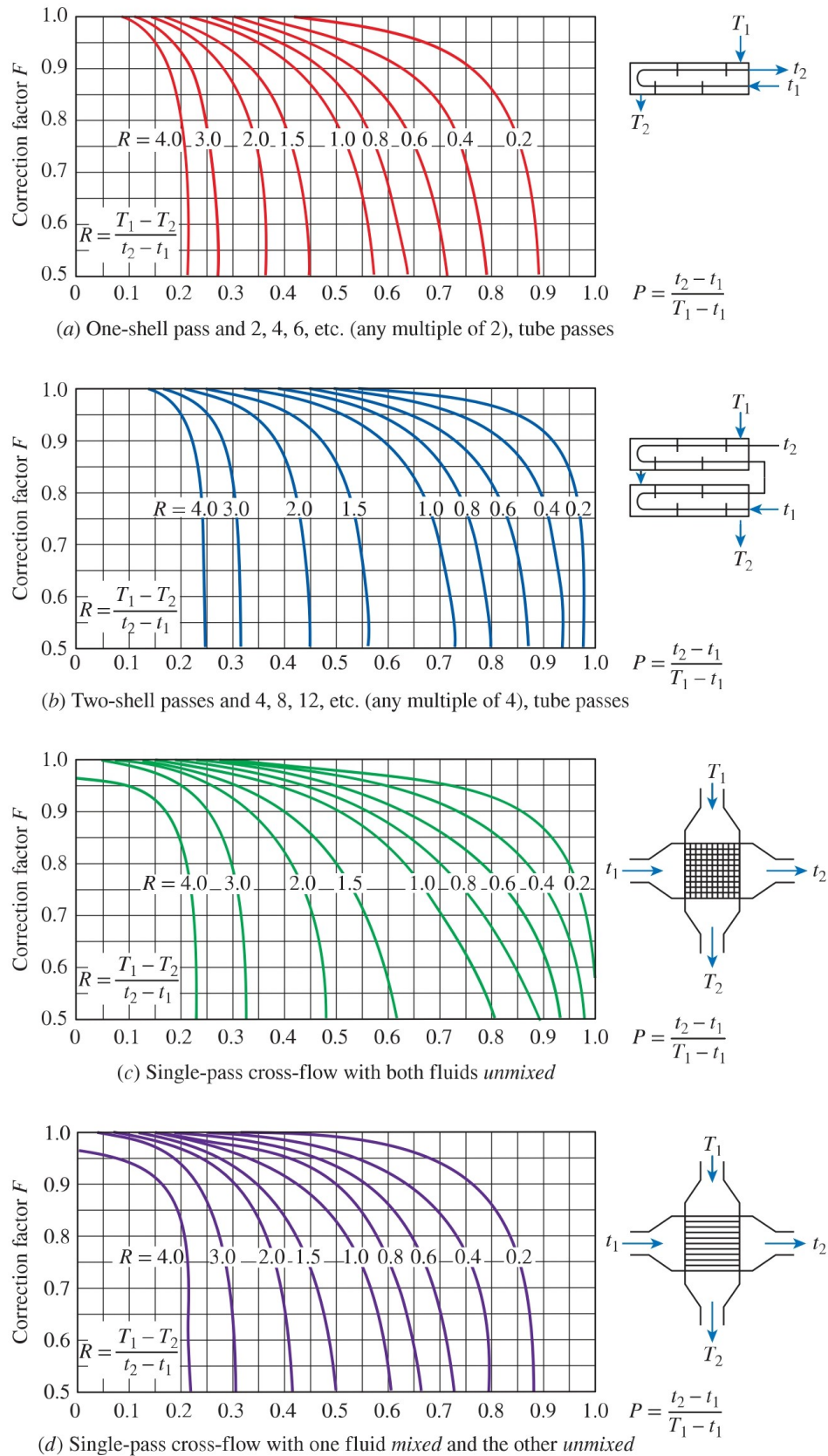
Figure 1: Correction factor F charts for common shell-and-tube and crossflow heat exchangers.

Table 3: Effectiveness relations for heat exchangers: $NTU = UA_s/C_{\min}$ and $c = C_{\min}/C_{\max} = (\dot{m}c_p)_{\min}/(\dot{m}c_p)_{\max}$

TABLE 11-4	
Effectiveness relations for heat exchangers: $NTU = UA_s/C_{\min}$ and $c = C_{\min}/C_{\max} = (\dot{m}c_p)_{\min}/(\dot{m}c_p)_{\max}$	
Heat Exchanger Type	Effectiveness Relation
1 <i>Double pipe:</i>	
Parallel-flow	$\varepsilon = \frac{1 - \exp[-NTU(1 + c)]}{1 + c}$
Counterflow	$\varepsilon = \frac{1 - \exp[-NTU(1 - c)]}{1 - c \exp[-NTU(1 - c)]} \quad (\text{for } c < 1)$
	$\varepsilon = \frac{NTU}{1 + NTU} \quad (\text{for } c = 1)$
2 <i>Shell-and-tube:</i>	
One-shell pass 2, 4, . . . tube passes	$\varepsilon_1 = 2 \left\{ 1 + c + \sqrt{1 + c^2} \frac{1 + \exp[-NTU_1 \sqrt{1 + c^2}]}{1 - \exp[-NTU_1 \sqrt{1 + c^2}]} \right\}^{-1}$
n -shell passes $2n$, $4n$, . . . tube passes	$\varepsilon = \left[\left(\frac{1 - \varepsilon_1 c}{1 - \varepsilon_1} \right)^n - 1 \right] \left[\left(\frac{1 - \varepsilon_1 c}{1 - \varepsilon_1} \right)^n - c \right]^{-1}$
3 Crossflow (<i>single-pass</i>) Both fluids unmixed	$\varepsilon = 1 - \exp \left\{ \frac{NTU^{0.22}}{c} [\exp(-c NTU^{0.78}) - 1] \right\}$
C_{\max} mixed, C_{\min} unmixed	$\varepsilon = \frac{1}{c} (1 - \exp \{-c[1 - \exp(-NTU)]\})$
C_{\min} mixed, C_{\max} unmixed	$\varepsilon = 1 - \exp \left\{ -\frac{1}{c} [1 - \exp(-c NTU)] \right\}$
4 All heat exchangers with $c = 0$	$\varepsilon = 1 - \exp(-NTU)$

12. Thermal Radiation Fundamentals

12.1. General Procedure

Most questions will ask for irradiation given some geometry and temperatures.

1. Determine absorptivity α , reflectivity ρ , and transmissivity τ using the geometry and material properties. Note that if two of these are known, the third can be found using $\alpha + \rho + \tau = 1$.
2. Use solid angle ω , irradiation G , and intensity I to find whatever is asked for

12.2. Variable Definitions

- ω : Solid angle
- G : Irradiation
- I : Intensity
- J : Radiosity
- \dot{Q} : Heat transfer rate
- α : Absorptivity
- ρ : Reflectivity
- τ : Transmissivity
- ϵ : Emissivity
- σ : Stefan-Boltzmann constant

12.3. Formulas

For $A \ll r^2$ (small area, far away from surface),

$$\omega_{2-1} := \frac{A_2 \cos \theta_2}{r^2}$$

$$J_1 := \pi I_{1,e+r} = E_b + G_{\text{ref}} = \epsilon \sigma T_1^4 + \rho G_1$$

$$\dot{Q}_{1-2} := I_1 (A_1 \cos \theta_1) \omega_{2-1}$$

$$G_2 := \frac{\dot{Q}_{1-2}}{A_2}$$

Combining the above equations into G_2 ,

$$G_2 = \frac{(\epsilon \sigma T_1^4 + \rho G_1) A_1 \cos \theta_1 \cos \theta_2}{\pi r^2}$$

For $\theta_1 = \theta_2 = \rho = 0$ and $\epsilon = 1$ for a blackbody,

$$G_2 = \frac{\sigma T_1^4 A_1}{\pi r^2}$$

$$\implies T_1 = \left(\frac{G_2 \pi r^2}{\sigma A_1} \right)^{1/4}$$

13. Thermal Radiation Heat Transfer

13.1. General Procedure

Most questions will ask for irradiation given some geometry and temperatures.

1. Determine view factors F_{ij} from inspection using the geometry. Check for special cases such as:
 - $F_{ii} = 0$ when the surface i is planar or convex
 - $F_{ij} = 1$ when the surface i is a concave and fully encloses surface j
2. Check Figures section for view factors that cannot be obtained from inspection
3. Use the reciprocity relation and summation rule to find the remaining view factors

Solvability condition: Given N surfaces, there are N^2 view factors. $N(N-1)/2$ view factors must be given by inspection, problem statement, or tables. The rest can be determined using reciprocity and summation.

13.2. Variable Definitions

- F_{ij} : View factor, the fraction of radiation leaving surface i that strikes surface j
- ω : Solid angle
- G : Irradiation, the rate of radiant energy incident on a surface per unit area
- I : Intensity
- J : Radiosity, the total rate of radiant energy leaving a surface per unit area
- E_b : Black body emissive power
- \dot{Q} : Heat transfer rate
- α : Absorptivity
- ρ : Reflectivity
- τ : Transmissivity
- ϵ : Emissivity
- σ : Stefan-Boltzmann constant

13.3. Formulas

Reciprocity relation:

$$A_{1,s}F_{12} = A_{2,s}F_{21}$$

Summation rule:

$$\sum_{j=1}^n F_{ij} = 1$$

Crossed-strings method:

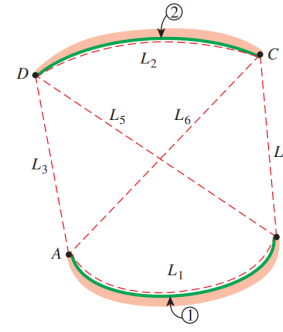


Figure 2: Determination of the view factor F_{12} by the application of the crossed-strings method.

$$F_{12} = \frac{(L_5 + L_6) - (L_3 + L_4)}{2L_1}$$

On a black body, net radiation heat transfer is

$$E_{bi} = \sigma T_i^4$$

$$\dot{Q}_{12} = A_{1,s}F_{12}\sigma(T_1^4 - T_2^4) = A_{2,s}F_{21}\sigma(T_1^4 - T_2^4)$$

On a grey body ($\epsilon_i = \alpha_i$, $\tau = 0$, $\alpha_i + \rho_i = 1$), net radiation heat transfer is

$$J_i = \pi I_i = \epsilon_i \sigma T_i^4 + \rho_i G_i$$

$$\dot{Q}_{12} = \frac{A_{i,s}\epsilon_i}{1 - \epsilon_i}(E_{bi} - J_i)$$

For radiation in two-surface enclosures, net radiation heat transfer is

$$\begin{aligned} \dot{Q}_{12} &= \dot{Q}_1 = -\dot{Q}_2 \\ \dot{Q}_{12} &= \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{\epsilon_1 A_{1,s}} + \frac{1}{A_{2,s}F_{12}} + \frac{1-\epsilon_2}{\epsilon_2 A_{2,s}}} \\ &= \frac{E_{b1} - E_{b2}}{R_1 + R_{12} + R_2} \end{aligned}$$

For some familiar geometries, the above equation simplifies, which is given in Table 4.

Figures

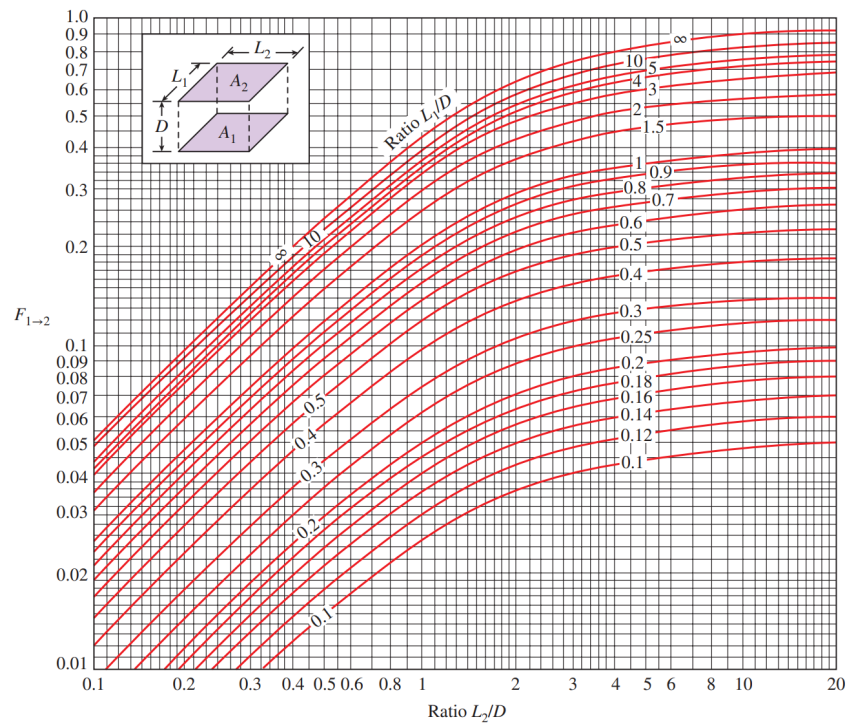


Figure 3: View factor between two aligned parallel rectangles of equal size

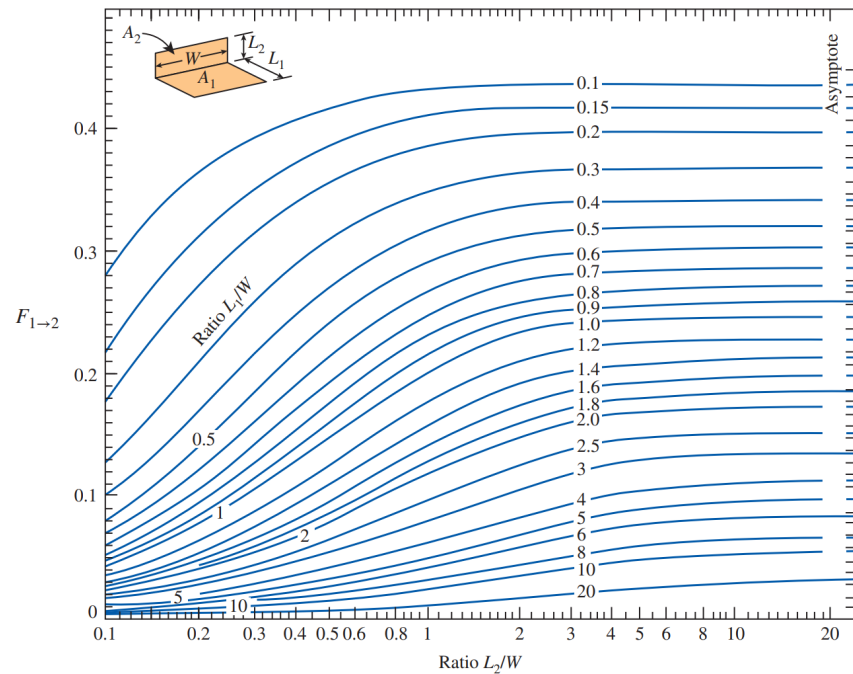


Figure 4: View factor between two perpendicular rectangles with a common edge

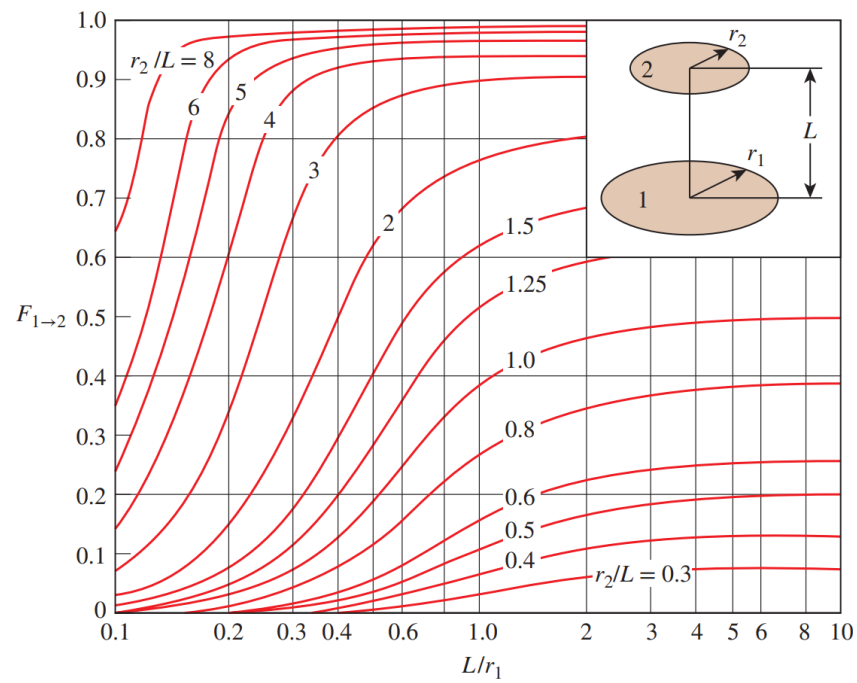


Figure 5: View factor between two coaxial parallel disks

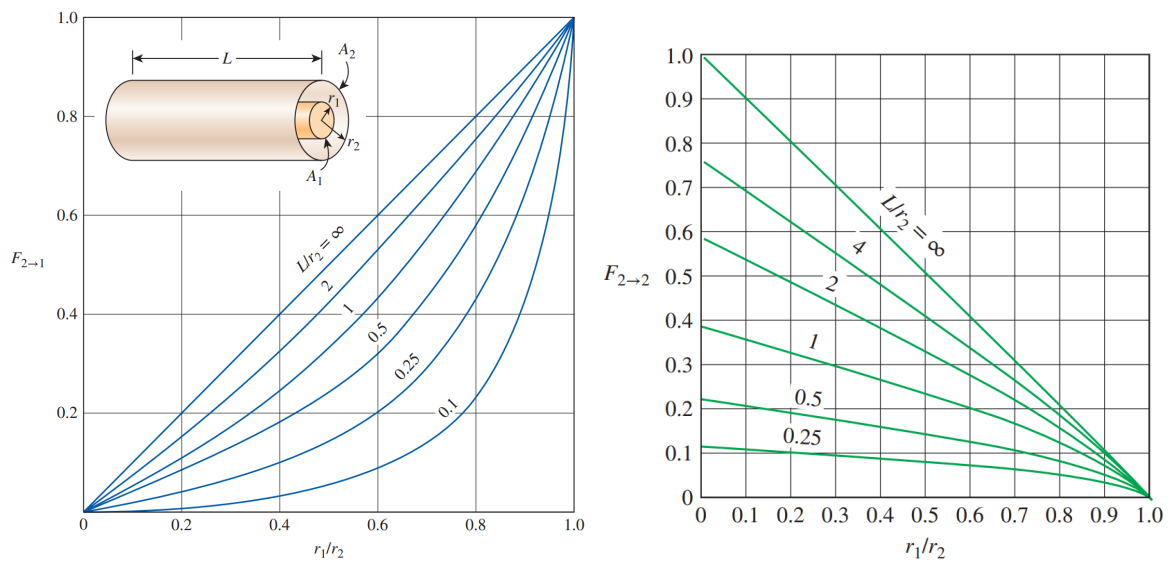
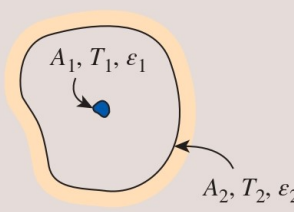
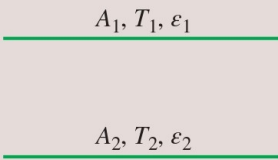
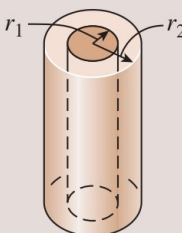
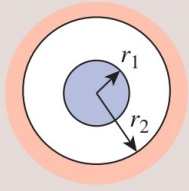


Figure 6: View factors for two concentric cylinders of finite length: (a) outer cylinder to inner cylinder; (b) outer cylinder to itself.

Tables

Table 4: Radiation heat transfer relations for some familiar two-surface arrangements

TABLE 13-3				
Radiation heat transfer relations for some familiar two-surface arrangements				
Small object in a large cavity				
	$\frac{A_1}{A_2} \approx 0$ $F_{12} = 1$	$\dot{Q}_{12} = A_1 \sigma \varepsilon_1 (T_1^4 - T_2^4)$	(13-37)	
Infinitely large parallel plates				
	$A_1 = A_2 = A$ $F_{12} = 1$	$\dot{Q}_{12} = \frac{A \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$	(13-38)	
Infinitely long concentric cylinders				
	$\frac{A_1}{A_2} = \frac{r_1}{r_2}$ $F_{12} = 1$	$\dot{Q}_{12} = \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1}{r_2} \right)}$	(13-39)	
Concentric spheres				
	$\frac{A_1}{A_2} = \left(\frac{r_1}{r_2} \right)^2$ $F_{12} = 1$	$\dot{Q}_{12} = \frac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1}{r_2} \right)^2}$	(13-40)	