Exercise 1(Points: 5.5)

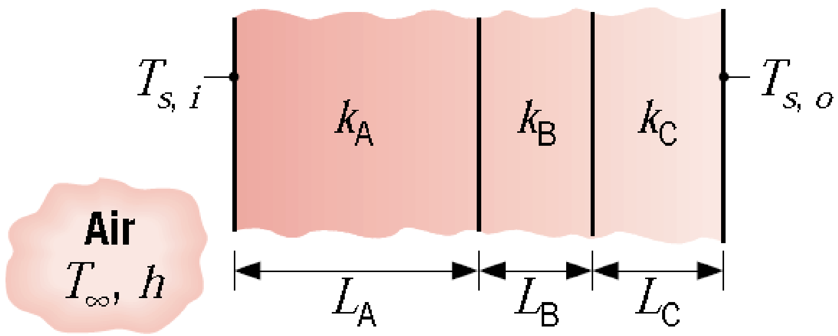
The composite wall of an oven consists of three materials (A,B and C) two of which are of known thermal conductivity, $k_A = 20W/mK$ and $k_C = 50W/mK$, and known thickness, $L_A = 0.3m$ and $L_C = 0.15m$. The third material, B, which is sandwiched between material A and C, is of known thickness $L_B = 0.15m$ but unknown thermal conductivity k_B. Under steady-state operating conditions, measurements reveal an outer surface temperature of $T_{s,o} = 20C$, an inner surface temperature $T_{s,i} = 600C$ and an oven air temperature $T_\infty = 800C$. The inside convection coefficient is known and equal to $h_i = 25W/m^2K$.

a) Draw the equivalent electrical circuit for the entire system (oven and wall). Indicate clearly the relevant temperatures in the circuit.

b) Write the expression for every thermal resistance in the system

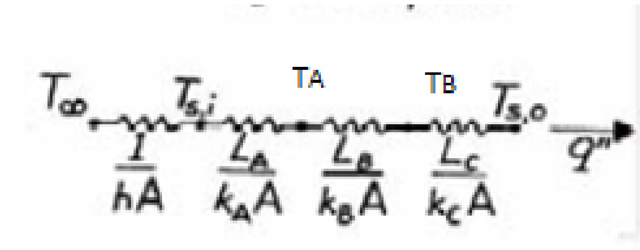
c) Calculate the heat flux flowing through the wall as well as the temperatures at the interfaces within the wall

d) Determine the thermal conductivity of the layer B



Solution

a) **1pt** : 0.5 for the three conduction resistors, 0.5 for the convection resistor.



b) **1pt**: 0.25pts for every resistance expression

$$R_{conv} = \frac{1}{h_i}$$
$$R_{cond,A} = \frac{L_A}{k_A}$$
$$R_{cond,B} = \frac{L_B}{k_B}$$
$$R_{cond,C} = \frac{L_C}{k_C}$$

c) **2.5pt** : 0.5 for realizing that the heat transfer through the wall is the same as by convection; 0.5 for the expression and 0.5 for the value; 0.5 for T_A and 0.5 for T_B (0.25 for expression and 0.25 for value).

$$q^" = h_i(T_\infty - T_{s,i})$$
$$q^" = 25W/mK(800 - 600) = 5000W/m^2$$

$$q^" = \frac{k_A}{L_A}(T_{s,i} - T_A)$$
$$T_A = 525C = 798K$$

$$q^" = \frac{k_C}{L_C}(T_B - T_{s,o})$$
$$T_B = 35C = 308K$$

d) **1pt**: 0.5 for the expression and 0.25 for k_B expression and 0.25 for the value

$$q^" = \frac{T_{s,i} - T_{s,o}}{R_{cond,A} + R_{cond,B} + R_{cond,C}} = \frac{T_{s,i} - T_{s,o}}{\frac{L_A}{k_A} + \frac{L_B}{k_B} + \frac{L_C}{k_C}}$$

$$k_B = L_B \left(\frac{T_{s,i} - T_{s,o}}{q^"} - \frac{L_A}{k_A} - \frac{L_C}{k_C} \right)^{-1} = 1.53W/mK$$

or

$$k_B = \frac{q^" L_B}{T_A - T_B} = 1.53W/mK$$

Exercise 2(Points 5.5)

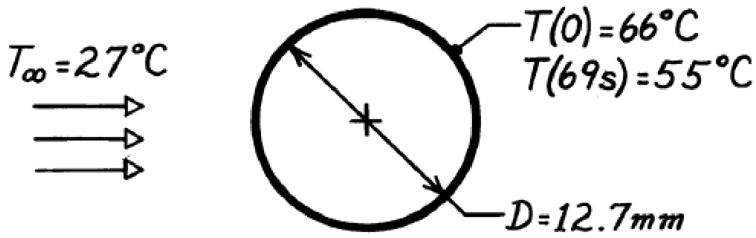
The heat transfer coefficient for air flowing over a sphere is to be determined by observing the temperature-time history of a sphere fabricated from pure copper. The sphere, which has a diameter of $D = 12.7mm$ is at an initial temperature of $T_i = 66C$ before it is inserted into an airstream having a temperature of $T_\infty = 27C$. After $69s$ in the airstream, a thermocouple on the outer surface of the sphere indicates $55C$.

a) After choosing a suitable reference temperature, and justifying your choice, list the following thermophysical properties of copper: density, specific heat and thermal conductivity.

b) Assume that the sphere behaves as an isothermal object and calculate the heat transfer coefficient

c) Verify that the assumption of part b) was correct

Solution



a) **1.5pt** : 0.25 for choosing $T_{mean,Cu}$, 0.25 for right Ts, 0.25 for $T_{mean,Cu}$ value. 0.25 for each correct interpolation.

$$T_{ref,Cu} = T_{mean,Cu} = (66 + 55)/2 + 273.15 = 333.65K$$

. From Table $A1$ (interpolation):

$$\rho = 8933kg/Km$$

$$c_p = 389J/kgK$$

$$k = 398W/mK$$

b) **2.5pt**: 0.5 for lumped capacitance model; 0.5 for correct expression; 0.25 for volume and 0.25 for area; 0.5 for recognizing $T(69s) = 55C$; 0.5 for calculating h (0.25 for τ and 0.25 for h).

Isothermal sphere implies we can use the lumped capacitance model.

$$\frac{\theta(t)}{\theta_0} = \frac{T(t) - T_\infty}{T_i - T_\infty} = exp(-\frac{hA_S}{\rho V c_p}t) = exp(-\frac{t}{\tau})$$

where

$$V = \frac{\pi D^3}{6}$$

$$A_s = \pi D^2$$

At $t = 69s$:

$$\frac{\theta(69s)}{\theta_0} = \frac{T(69s) - T_\infty}{T(0) - T_\infty} = \frac{55 - 27}{66 - 27} = 0.718$$

Solving for τ we have:

$$\tau = -\frac{69s}{ln(0.718)} = 208s$$

.

Therefore from the expression of tau we obtain:

$$h = \frac{\rho V c_p}{A_s \tau} = \frac{8933(\pi 0.0127^3/6)389}{\pi 0.0127^2 208} = 35.3W/m^2K$$

c) **1.5pt** ; 0.5 for choosing to calculate Bi; 0.5 for the right value; 0.5 for the conclusion

$$Bi = \frac{hV}{kA} = \frac{hD}{6k} = 0.000188 << 0.1$$

Therefore the initial assumption is valid.

Exercise 3(Points: 13)

Saturated steam at $0.15 bar$ is condensed in a shell-and-tube heat exchanger with one shell pass and two tube passes consisting of 130 brass (i.e Copper - Zinc alloy with 70/30 composition) tubes, each with a length per pass of $L = 2m$. The tubes have inner and outer diameters of $D_i = 13.4mm$ and $D_o = 15.9mm$, respectively. Cooling water enters the tubes at $20C$ with a mean velocity of $v = 1.25m/s$. The heat transfer coefficient for condensation on the outer surfaces of the tubes is $h_o = 13500W/m^2K$.

a) On the same graph, draw a qualitative sketch of the evolution of the steam and water temperatures as the two fluids go through the heat exchanger

b) Using a reference average temperature for the water side equal to $T_{ref,c} = 305K$, list the values of density, specific heat c_p, viscosity, thermal conductivity, Prandtl.

c) For the steam side indicate the reference temperature $T_{ref,h}$ and list the values of latent heat and specific heat c_p.

d) List the value of the thermal conductivity of the brass tubes estimated at the average temperature between the hot and cold side reference temperatures ($T_{ref,h}$ and $T_{ref,c}$).

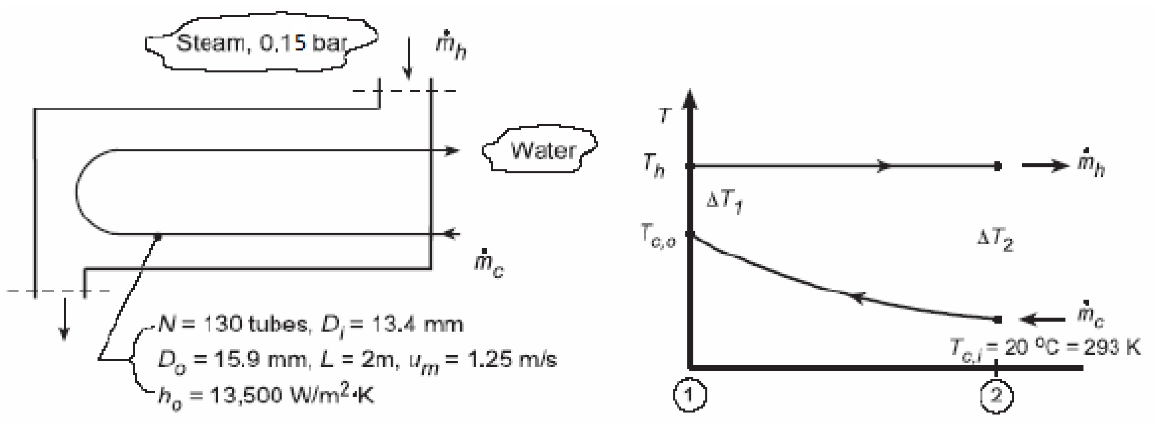
e) Write the expression for the overall heat transfer coefficient and calculate its value with respect to the outside area of the tube. Clearly explain your choice of the convection correlations.

f) Determine the cooling water outlet temperature

g) Determine the steam condensation rate

Solution

a) **1.5pt**: 1 for constant temperature of the steam side; 0.5 for the water side.



b) **1.5pt**: 0.5 for ρ; 0.25 for each of the other 4 values

From table $A6$ with $T_{ref,c} = 305K$:

$$\rho = 1/v_f = 995kg/m^3$$

$$c_p = 4178J/kgK$$
$$\mu = 769 \cdot 10^{-6}Ns/m^2$$
$$k = 0.620W/mK$$
$$Pr = 5.2$$

c) **1pt**: 0.5 for the ref temperature; 0.25 for each of the 2 values

From table $A6$ we estimate the steam properties at (interpolation):

$$T_{ref,h} = T_{sat}(0.15bar) = 327K$$

$$h_{fg} = 2373kJ/kg = 2.373 \cdot 10^6 J/kg$$
$$c_p = 1906J/kgK$$

d) **0.5pt**: 0.25 for the reference temperature value; 0.25 for the value

From table $A1$:

$$T_{ref,tube} = (T_{ref,c} + T_{ref,h})/2 = 316K$$
$$k = 119.4W/mK$$

e) **4.5pt**: 2.25 U expression (0.5 for each term (3x0.5) - 0.25 for type of resistance, 0.25 for expression - and 0.75 for the correct areas: 0.25 for N in A_o, 0.25 for $2L$ and 0.25 for N in R_{cond}); 0.25 for convection type; 0.25 for Re value; 0.25 for flow condition; 0.25 for convection correlation; 0.25 for correct exponent; 0.25 for value of Nu; 0.25 for value of h; 0.5 for value of U.

$$\frac{1}{U_oA_o} = \frac{1}{A_oh_o} + \frac{1}{2\pi k(2L)N} ln\frac{r_o}{r_i} + \frac{1}{A_ih_i} = \frac{1}{\pi D_o(2L)Nh_o} + \frac{1}{2\pi k(2L)N} ln\frac{r_o}{r_i} + \frac{1}{\pi D_i(2L)Nh_i}$$

$$U_o = \left[\frac{1}{h_o} + \frac{r_o}{k} ln\frac{r_o}{r_i} + \frac{r_o}{r_i}\frac{1}{h_i}\right]^{-1}$$

Forced internal convection (circular tubes)

$$Re_{D,i} = \frac{\rho v D_i}{\mu} = 21673 >> 2300$$

The flow is turbulent.

$$Nu_D = \frac{h_iD_i}{k_f} = 0.023Re_D^{0.8}Pr_f^n$$

The water is heating:

$$n = 0.4$$

$$Nu_D = 0.023 \cdot (21673)^{0.8} \cdot (5.2)^{0.4} = 130.9$$

$$h_i = \frac{k_f}{D_i} Nu_D = 6057 W/m^2K$$

and so:

$$U_o = \left[\frac{1}{13500} + \frac{15.9 \cdot 10^{-3}}{119.4} ln\frac{15.9}{13.4} + \frac{15.9}{13.4}\frac{1}{6057}\right]^{-1} = 3557 W/m^2K$$

f) **3.5pt**: 0.5 expression for the energy balance on cold side Q; 0.5 choice of $\epsilon - NTU$ method (expression of $Q = \epsilon Q_{max}$); 0.25 for C_h and 0.5 for C_c values - 0.25 for mass flow rate expression and 0.25 for value; 0.25 for correct C_{min}; 0.25 for NTU value; 0.5 for correct ϵ expression; 0.25 for ϵ value; 0.5 for correct $T_{c,o}$ value

Energy balance on the cold (water) side:

$$Q = C_c(T_{c,o} - T_{c,i})$$

We use $\epsilon - NTU$ method:

$$Q = \epsilon Q_{max} = \epsilon C_{min}(T_{h,i} - T_{c,i})$$

$$C_h \to \infty$$

$$C_c = (\dot{m}c_p)_c = (\rho A v N c_p)_c = (\rho \pi D_i^2/4 v N c_p)_c = 22.8 \cdot 4178 = 95270 W/K$$

$$C_{min} = C_c$$

$$NTU = \frac{U_o A_o}{C_{min}} = \frac{U_o(\pi D_o 2LN)}{C_{min}} = \frac{3557(\pi 0.0159 \cdot 4 \cdot 130)}{95270} = 0.968$$

From Table 11.3

$$\epsilon = 1 - epx(-NTU) = 0.62$$

$$T_{c,o} = T_{c,i} + \epsilon(T_{h,i} - T_{c,i}) = 20 + 0.62(327 - 293) = 41.1C$$

g) **0.5pt**: 0.25 expression, 0.25 value

The condensation rate is

$$\dot{m}_h = Q/h_{fg} = C_c(T_{c,o} - T_{c,i})/h_{fg} = 95270(41.1 - 20)/(2.373 \cdot 10^6) = 0.85kg/s$$

Exercise 4(Points: 10)

To augment the heat transfer between two flowing fluids, it is proposed to insert a 2024 aluminum pin ($L = 100mm$, $D = 5mm$) through the wall separating the two fluids. The pin is inserted to a depth of $d = 50mm$ into fluid 1. Fluid 1 is carbon dioxide (CO_2) with a mean temperature of $280K$ and uniform velocity of $10m/s$. Fluid 2 is air with a mean temperature of $350K$ and mean velocity of $3m/s$.

a) List the values of the physical properties of the fluids and solid. For Fluid 1 and 2 use their temperatures as reference. For the aluminum pin use the average temperature between the hot and cold air streams as reference temperature.

b) Motivating your choice of the convection correlation, determine the convection coefficients between CO_2 and the pin on one side and the air and the pin on the other side

c) Treating the pin as a fin, write the expression for the heat transfer rate between the pin and the fluid on each side

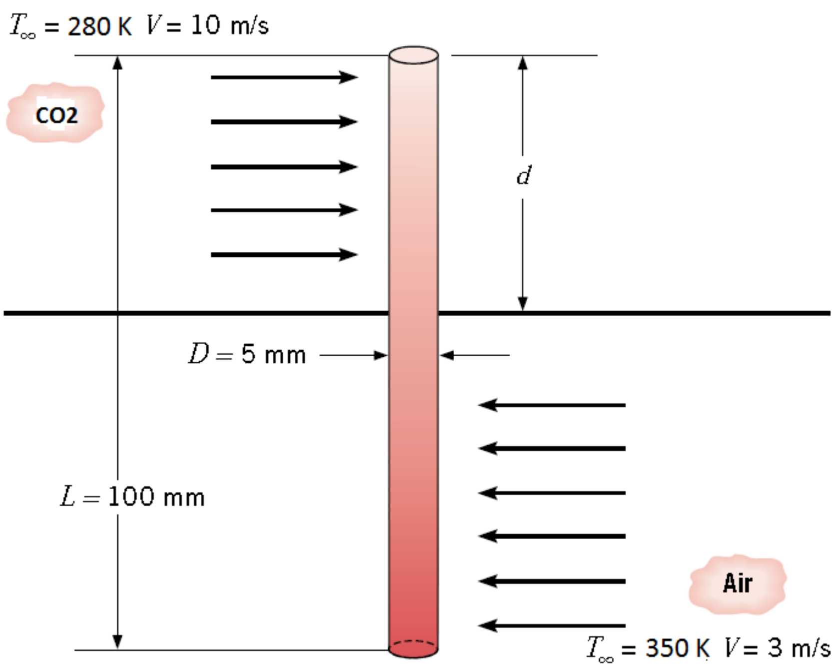
d) Determine the temperature at the base of the fin (i.e. at the wall)

e) Determine the heat transfer rate from the warm air to the cool air through the pin-fin

Note: remember that the hyperbolic functions have the following expressions:

$$sinh(x) = \frac{e^x - e^{-x}}{2}$$

$$cosh(x) = \frac{e^x + e^{-x}}{2}$$



Solution

a) **1.5pt**: 0.5 for all values of CO2, 0.5 for all values Air and 0.25 for ref temperature for Al and 0.25 for the k value of aluminum

Table $A4$ CO_2 ($T_{f1} = 280K$):

$$\rho_1 = 1.9022 kg/m^3$$
$$\mu_1 = 140 \cdot 10^{-7} Ns/m^2$$
$$k_1 = 0.0152 W/mK$$
$$Pr_1 = 0.765$$
$$c_p = 830 J/Kkg$$
$$\nu = 7.36 \cdot 10^{-6} m^2/s$$
$$\alpha = 9.63 \cdot 10^{-6} m^2/s$$

Table $A4$ Air ($T_{f2} = 350K$):

$$\rho_2 = 0.995 kg/m^3$$
$$\mu_2 = 208.2 \cdot 10^{-7} Ns/m^2$$
$$k_2 = 0.03 W/mK$$
$$Pr_2 = 0.7$$

$$c_p = 1009 J/Kkg$$
$$\nu = 20.92 \cdot 10^{-6} m^2/s$$
$$\alpha = 29.9 \cdot 10^{-6} m^2/s$$

Table $A1$, Aluminum 2024 ($T_{Al} = (280 + 350)/2 = 315K$):

$$k_{Al} = 176.23 W/mK$$

b) **2.5pt**: 0.25 for forced external convection; 0.25 for cylinder geometry; 0.5 for right correlation; 0.5 for Re (0.25 each); 0.5 for Nu (0.25 each); 0.5 for convection coeff (0.25 each).

External forced convection around a cylinder (Churchill-Bernstein correlation):

$$Nu_D = 0.3 + \frac{0.62 Re_D^{1/2} Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \left[1 + \left(\frac{Re_D}{282000}\right)^{5/8}\right]^{4/5}$$

We get:

$$Re_1 = \frac{\rho_1 v_1 D}{\mu_1} = 6793$$
$$Re_2 = \frac{\rho_2 v_2 D}{\mu_2} = 716.9$$

$$Nu_{D,1} = 44.7$$
$$h_1 = \frac{Nu_{D,1} k_1}{D} = 135.88 W/m^2K$$

$$Nu_{D,2} = 13.48$$
$$h_2 = \frac{Nu_{D,2} k_2}{D} = 80.9 W/m^2K$$

c) **1pt**: 0.25 for the correct boundary condition; 0.25 for $L = d$ and 0.5 for $k = k_{Al}$

Each half of the pin behaves as a fin. Convection occurs onto the entire surface so the correct boundary condition is convection at the tip.

$$Q_1 = M_1 \frac{sinh(m_1 d) + (h_1/m_1 k_{Al}) cosh(m_1 d)}{cosh(m_1 d) + (h_1/m_1 k_{Al}) sinh(m_1 d)}$$
$$Q_2 = M_2 \frac{sinh(m_2 d) + (h_2/m_2 k_{Al}) cosh(m_2 d)}{cosh(m_2 d) + (h_2/m_2 k_{Al}) sinh(m_2 d)}$$

as $L_1 = L_2 = L/2 = d = 0.05m$

d) **4pt**: 1 for the heat transfer equality condition (0.5 for equality, 0.5 for sign), 0.25 for P and 0.25 for Ac, 0.5 for re-writing the equality as a function for Tb, 0.5 for the 2 values of m, 0.5 for the 2 values of B, 1 for the value of Tb

The base of the fin is coincident for the two sides. Also:

$$Q_1 = -Q_2$$

The fin is cylindrical with $P = \pi D$ and $A_c = \pi D^2/4$ so:

$$m_i = \sqrt{\frac{h_i P}{k A_c}} = 2\sqrt{\frac{h_i}{k_{Al} D}}$$

$$M_i = \sqrt{h_i P k A_c}\theta_{b,i} = \sqrt{h_i D k}\frac{\pi D}{2}(T_b - T_{\infty,i})$$

So:

$$\sqrt{h_1 D k}\frac{\pi D}{2}(T_b - T_{\infty,1})\frac{sinh(m_1 L_1) + (h_1/m_1 k) cosh(m_1 L_1)}{cosh(m_1 L_1) + (h_1/m_1 k) sinh(m_1 L_1)} =$$
$$-\sqrt{h_2 D k}\frac{\pi D}{2}(T_b - T_{\infty,2})\frac{sinh(m_2 L_2) + (h_2/m_2 k) cosh(m_2 L_2)}{cosh(m_2 L_2) + (h_2/m_2 k) sinh(m_2 L_2)}$$

This can be re-written as:

$$\sqrt{h_1}(T_b - T_{\infty,1})B_1 = -\sqrt{h_2}(T_b - T_{\infty,2})B_2$$

Solving for T_b we get:

$$T_b = \frac{\sqrt{h_1}B_1 T_{\infty,1} + \sqrt{h_2}B_2 T_{\infty,2}}{\sqrt{h_1}B_1 + \sqrt{h_2}B_2}$$

Substituting the values of the physical parameters:

$$m_1 = 22.74$$

$$m_2 = 19.16$$

So: $m_1 d = 1.137$ and $m_2 d = 0.958$. Also: $sinh(m_1 d) = 1.398$, $cosh(m_1 d) = 1.719$, $sinh(m_2 d) = 1.111$ and $cosh(m_2 d) = 1.495$. Finally $\frac{h_1}{m_1 k_{Al}} = 0.0284$ and $\frac{h_2}{m_2 k_{Al}} = 0.024$.

Hence:

$$B1 = 0.823$$

$$B_2 = 0.753$$

And

$$T_b = 310.5K$$

e) **1pt**: 0.5 for the values of M, 0.5 for the value of Q

We just need to calculate:

$$Q = Q_1 = -Q_2 = \sqrt{h_1 D k}\frac{\pi D}{2}(T_b - T_{\infty,1})\frac{sinh(m_1 L_1) + (h_1/m_1 k)cosh(m_1 L_1)}{cosh(m_1 L_1) + (h_1/m_1 k)sinh(m_1 L_1)}$$

Substituting the values of the physical parameters:

$$M_1 = 0.0789(T_b - 280K)$$

$$M_2 = 0.0663(T_b - 350K)$$

Therefore:

$$Q = Q_1 = M_1 B_1 = 1.98W$$

or

$$Q = -Q_2 = -M_2 B_2 = 1.972W$$

Exercise 5(Points: 9)

Exhaust gases from a wire processing oven are discharged into a tall stack. 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 has a diameter $D = 0.5m$ and a height $H = 6m$. The exhaust gas flow rate is $\dot{m} = 0.5kg/s$ and the inlet gas temperature is $T_{m,i} = 600C$. The environmental conditions outside the stack are characterized by an air temperature of $T_\infty = 4C$ and a wind velocity of $v = 5m/s$.

Assuming that the exhaust gas has the same thermophysical properties of air and assuming that the wall of the stack has a negligible thermal resistance:

a) Using an average temperature of $T_{m,g} = 550C$, list the values of specific heat, viscosity, thermal conductivity and Prandtl number for the exhaust gases.

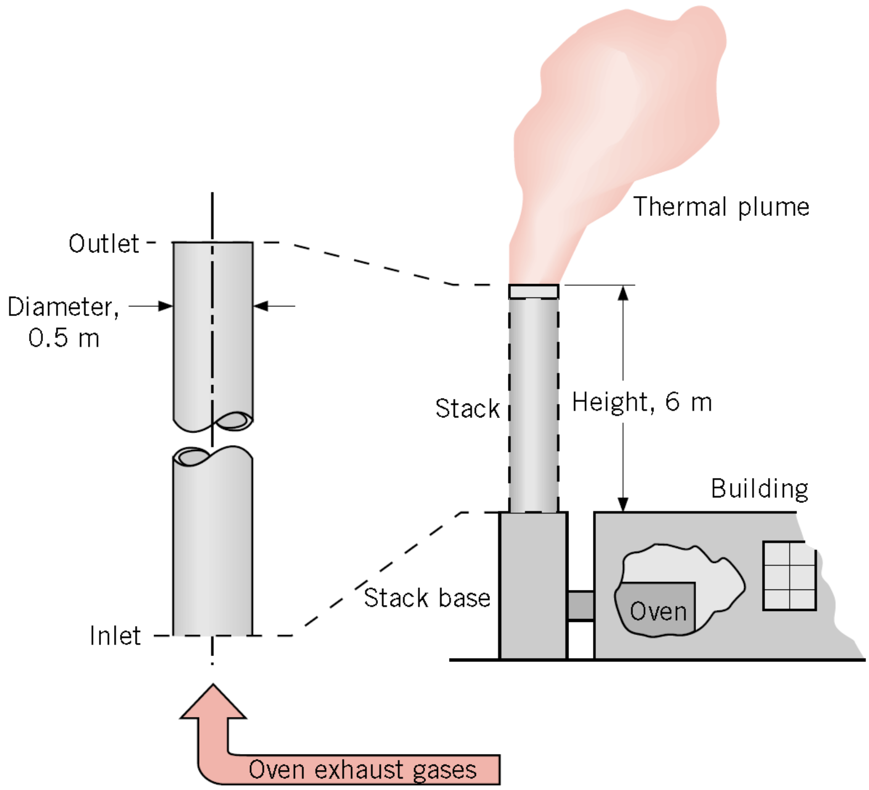
b) Assuming a reference surface temperature of the stack of $T_{ref,s} = 250C$, calculate the reference temperature for the air boundary layer on the outside of the stack and list the values of density, specific heat, viscosity, thermal conductivity and Prandtl number.

c) Calculate the inner and outer convection coefficients, h_i and h_o. Write clearly the motivation for your choice of a specific correlation.

d) Calculate the average gas outlet temperature, $T_{m,o}$

e) Calculate the outlet stack surface temperature, $T_{s,o}$

f) Comment on how adequate are the reference temperatures we used to estimate the thermophysical properties.



Solution

a) **1pt**: 0.25 for every value

From table $A1$, at $T_{m,g} = 823K$ (interpolation):

$$c_{p,g} = 1.104kJ/kgK = 1104J/kgK$$

$$\mu_g = 376.4 \cdot 10^{-7} Ns/m^2$$

$$k_g = 0.0584W/mK$$

$$Pr_g = 0.712$$

b) **0.5pt**: 0.25 for the correct reference temperature; 0.25 for the values

The boundary layer reference temperature is:

$$T_f = (T_\infty + T_{s,o})/2 = 400K$$

From table $A4$ we obtain:

$$\rho_f = 0.8711kg/m^3$$

$$c_{p,f} = 1.014kJ/kgK$$

$$\mu_f = 2.3 \cdot 10^{-5} Ns/m^2$$

$$k_f = 0.0338W/mK$$

$$Pr_f = 0.690$$

c) **3pt**: 0.25 for type of convection; 0.25 for converting Re as a function of mass flow rate; 0.25 for value of Re; 0.25 for recognizing the turbulent flow choosing the right correlation; 0.25 for the choice of the right exponent; 0.25 for the value of Nu_i; 0.25 for the value of h_i. 0.25 for type of convection; 0.25 for the choice of the correlation; 0.25 for the value of Re; 0.25 for the value of Nu_o; 0.25 for the value of h_o.

Internal Flow. Forced internal convection → calculate Re

$$Re_{D,i} = \frac{\rho_g v D}{\mu_g} = \frac{4\dot{m}}{\pi D \mu_g}$$

$$Re_{D,i} = \frac{4 \cdot 0.5}{\pi \cdot 0.5 \cdot 376.4 \cdot 10^{-7}} = 33827 >> 2300$$

The flow is turbulent and $Re_{D,i} > 10000$:

$$Nu_i = 0.023 Re_{D,i}^{4/5} Pr_g^n$$

The exhaust gas is cooling:

$$n = 0.3$$

$$Nu_i = \frac{h_i D}{k_g} = 87.26$$

$$h_i = \frac{k}{D} 0.023 Re_{D,i}^{4/5} Pr_g^{0.3} = \frac{58.4 \cdot 10^{-3}}{0.5} 0.023 (33827)^{4/5} (0.712)^{0.3} = 10.2W/m^2K$$

External Flow. Forced external convection. Flow around a cylinder.

$$Nu_{D,o} = 0.3 + \frac{0.62 Re_{D,o}^{1/2} Pr^{1/3}}{\left[1 + (0.4/Pr_f)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re_o}{282000}\right)^{5/8}\right]^{4/5}$$

where

$$Re_{D,o} = \frac{\rho v D}{\mu} = \frac{5 \cdot 0.5}{26.41 \cdot 10^{-6}} = 94660$$

$$Nu_{D,o} = 205$$

$$h_o = \frac{Nu_{D,o} k_f}{D} = \frac{0.0338205}{0.5} = 13.9 W/m^2 K$$

d) **2pt**: 0.5 for choosing the right equation; 0.25 for the right expression of the external area; 1 for the right expression of U (0.25 for each convection term, 0.25 for neglecting the conduction, 0.25 for the U value or UA); 0.25 for the value for $T_{m,o}$.

Internal convection in a cylindrical pipe with heat removed by external convection:

$$\frac{T_\infty - T_{m,o}}{T_\infty - T_{m,i}} = exp\left(-\frac{UA}{\dot{m} c_{p,g}}\right)$$

where

$$A = PH = \pi D H$$

.

Negligible conduction at the wall. The overall heat transfer coefficient:

$$R_{tot} = \frac{1}{UA} = \frac{1}{Ah_i} + \frac{1}{Ah_o}$$

$$UA = 55.33 W/K$$

or

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = 5.88$$

$$T_{m,o} = T_\infty - (T_\infty - T_{m,i}) exp\left(-\frac{UA}{\dot{m} c_{p,g}}\right) = 4 - (4 - 600) exp\left(-\frac{5.88 \cdot \pi 0.5 \cdot 6}{0.5 \cdot 1104}\right) = 543C = 816K$$

e) **1.5**: 1.25 for the expression of heat transfer from the gas to the air at the outlet section (0.5 for each side of the equation and 0.25 for the equality); 0.25 for the value of the outlet temperature

At the outlet section:

$$q_o^{"} = h_i (T_{m,o} - T_{s,o}) = h_o (T_{s,o} - T_\infty)$$

$$T_{s,o} = \frac{h_i T_{m,o} + h_o T_\infty}{h_i + h_o} = \frac{10.2543 + 13.94}{10.2 + 13.9} = 232C = 505K$$

f) **1pt**

The temperatures used to estimate the fluid properties were adequate ($T_{m,g} = (600 + 543)/2 = 571.5C \approx 550C$ and $T_{s,o} = 232C \approx T_{ref,s} = 250C$