16–34 A transistor with a height of 0.4 cm and a diameter of 0.6 cm is mounted on a circuit board. The transistor is cooled by air flowing over it with an average heat transfer coefficient of 30 W/m^2·K. If the air temperature is 55°C and the transistor case temperature is not to exceed 70°C, determine the amount of power this transistor can dissipate safely. Disregard any heat transfer from the transistor base.

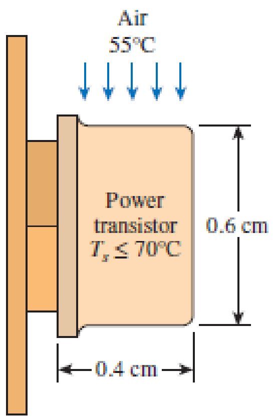


FIGURE P16–34

16–36 The boiling temperature of nitrogen at atmospheric pressure at sea level (1 atm) is −196°C. Therefore, nitrogen is commonly used in low-temperature scientific studies since the temperature of liquid nitrogen in a tank open to the atmosphere remains constant at −196°C until the liquid nitrogen in the tank is depleted. Any heat transfer to the tank results in the evaporation of some liquid nitrogen, which has a heat of vaporization of 198 kJ/kg and a density of 810 kg/m^3 at 1 atm.

Consider a 4-m-diameter spherical tank initially filled with liquid nitrogen at 1 atm and −196°C. The tank is exposed to 20°C ambient air with a heat transfer coefficient of 25 W/m^2·K. The temperature of the thin-shelled spherical tank is observed to be almost the same as the temperature of the nitrogen inside. Disregarding any radiation heat exchange, determine the rate of evaporation of the liquid nitrogen in the tank as a result of the heat transfer from the ambient air.

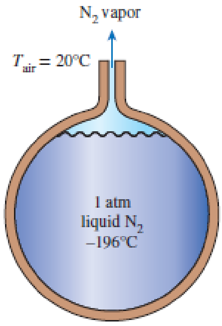


FIGURE P16–36

16–44 A spherical interplanetary probe with a diameter of 2 m is sent out into the solar system. The probe surface is made of material having an emissivity of 0.9 and an absorptivity of 0.1. Signals from the sensors monitoring the probe surface temperatures are indicating an average value of $-40°C$ for a space temperature of 0 K. If the electronics inside the probe are generating heat at a rate of 100 W/m^3, determine the incident radiation rate on the probe surface.

16–53 An electronic package with a surface area of 1 m^2 placed in an orbiting space station is exposed to space. The electronics in this package dissipate all 1 kW of its power to the space through its exposed surface. The exposed surface has an emissivity of 1.0 and an absorptivity of 0.25. Determine the steady-state exposed surface temperature of the electronic package (*a*) if the surface is exposed to a solar flux of 750 W/m^2, and (*b*) if the surface is not exposed to the sun.

16–60 Solar radiation is incident on a 5-m^2 solar absorber plate surface at a rate of 800 W/m^2. Ninety-three percent of the solar radiation is absorbed by the absorber plate, while the remaining 7 percent is reflected away. The solar absorber plate has a surface temperature of 40°C with an emissivity of 0.9 that experiences radiation exchange with the surrounding temperature of −5°C. In addition, convective heat transfer occurs between the absorber plate surface and the ambient air of 20°C with a convection heat transfer coefficient of 7 $W/m^2 \cdot K$. Determine the efficiency of the solar absorber, which is defined as the ratio of the usable heat collected by the absorber to the incident solar radiation on the absorber.

16–70 It is well known that wind makes the cold air feel much colder as a result of the *wind-chill effect* that is due to an increase in the convection heat transfer coefficient with increasing air velocity. The wind-chill effect is usually expressed in terms of the *wind-chill temperature* (WCT), which is the apparent temperature felt by exposed skin. For an outdoor air temperature of 0°C, for example, the wind-chill temperature is −5°C with 20 km/h winds and −9°C with 60 km/h winds. That is, a person exposed to 0°C windy air at 20 km/h will feel as cold as a person exposed to −5°C calm air (air motion under 5 km/h).

For heat transfer purposes, a standing man can be modeled as a 30-cm-diameter, 170-cm-long vertical cylinder with both the top and bottom surfaces insulated and with the side surface at an average temperature of 34°C. For a convection heat transfer coefficient of 15 $W/m^2 \cdot K$, determine the rate of heat loss from this man by convection in still air at 20°C. What will your answer be if the convection heat transfer coefficient is increased to 30 $W/m^2 \cdot K$ as a result of winds? What is the wind-chill temperature in this case? *Answers:* 336 W, 672 W, 6°C

16–73 A cylindrical fuel rod 2 cm in diameter is encased in a concentric tube and cooled by water. The fuel generates heat uniformly at a rate of 150 MW/m³. The convection heat transfer coefficient on the fuel rod is 5000 W/m²·K, and the average temperature of the cooling water, sufficiently far from the fuel rod, is 70°C. Determine the surface temperature of the fuel rod, and discuss whether the value of the given convection heat transfer coefficient on the fuel rod is reasonable.

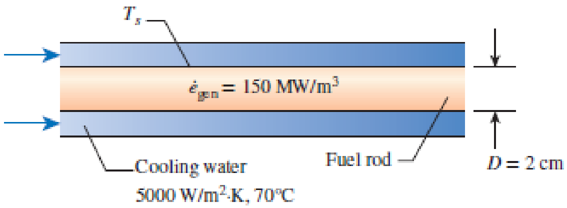
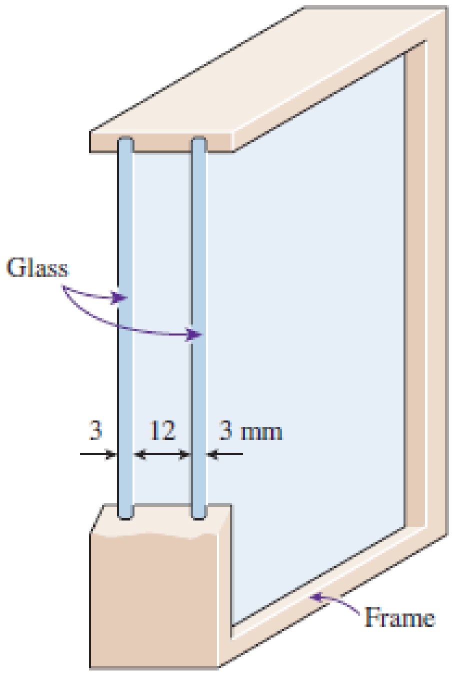
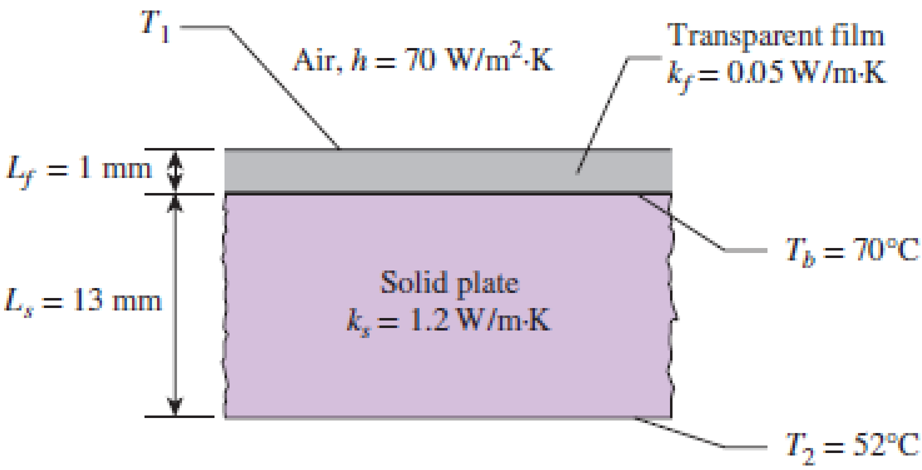


FIGURE P16–73

17–22 Consider a 1.5-m-high and 2.4-m-wide double-pane window consisting of two 3-mm-thick layers of glass ($k = 0.78$ W/m·K) separated by a 12-mm-wide stagnant airspace ($k = 0.026$ W/m·K). Determine the steady rate of heat transfer through this double-pane window and the temperature of its inner surface for a day during which the room is maintained at 21°C while the temperature of the outdoors is −5°C. Take the convection heat transfer coefficients on the inner and outer surfaces of the window to be $h_1 = 10$ W/m²·K and $h_2 = 25$ W/m²·K, and disregard any heat transfer by radiation. *Answers:* 154 W, 16.7°C



17–27 A transparent film is to be bonded onto the top surface of a solid plate inside a heated chamber. For the bond to cure properly, a temperature of 70°C is to be maintained at the bond, between the film and the solid plate. The transparent film has a thickness of 1 mm and thermal conductivity of 0.05 W/m·K, while the solid plate is 13 mm thick and has a thermal conductivity of 1.2 W/m·K. Inside the heated chamber, the convection heat transfer coefficient is 70 W/m²·K. If the bottom surface of the solid plate is maintained at 52°C, determine the temperature inside the heated chamber and the surface temperature of the transparent film. Assume thermal contact resistance is negligible.

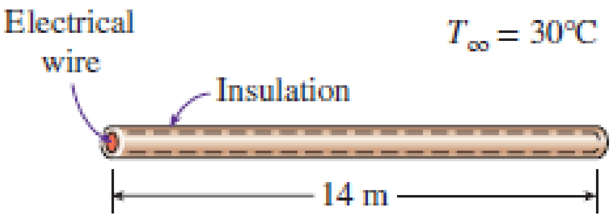


T=127ºC

17–49 An aluminum plate and a stainless steel plate are pressed against each other at an average pressure of 20 MPa. Both plates have a surface roughness of 2 μm. Determine the impact on the temperature drop at the interface if the surface roughness of the plates is increased tenfold.

$\Delta T_{20} = 8 \times \Delta T_2$

17–76 A 2.2-mm-diameter and 14-m-long electric wire is tightly wrapped with a 1-mm-thick plastic cover whose thermal conductivity is $k = 0.15$ W/m·K. Electrical measurements indicate that a current of 13 A passes through the wire, and there is a voltage drop of 8 V along the wire. If the insulated wire is exposed to a medium at $T_\infty = 30°C$ with a heat transfer coefficient of $h = 24$ W/m²·K, determine the temperature at the interface of the wire and the plastic cover in steady operation. Also determine if doubling the thickness of the plastic cover will increase or decrease this interface temperature.



T_i= 58,6ºC

Aumentando a espessura do plástico para o dobro, virá:

T_i= 54,1ºC

17–119 Steam in a heating system flows through tubes whose outer diameter is 5 cm and whose walls are maintained at a temperature of 130°C. Circular aluminum alloy 2024-T6 fins ($k = 186$ W/m·K) of outer diameter 6 cm and constant thickness 1 mm are attached to the tube. The space between the fins is 3 mm, and thus there are 250 fins per meter length of the tube. Heat is transferred to the surrounding air at $T_\infty = 25°C$, with a heat transfer coefficient of 40 W/m^2·K. Determine the increase in heat transfer from the tube per meter of its length as a result of adding fins. *Answer:* 1788 W

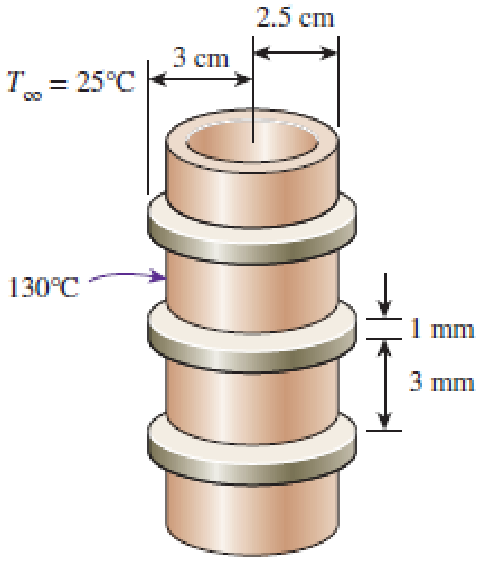
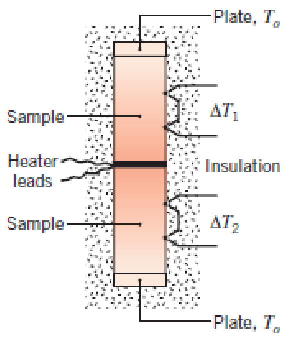


FIGURE P17–119

2.17 An apparatus for measuring thermal conductivity employs an electrical heater sandwiched between two identical samples of diameter 30 mm and length 60 mm, which are pressed between plates maintained at a uniform temperature $T_o = 77°C$ by a circulating fluid. A conducting grease is placed between all the surfaces to ensure good thermal contact. Differential thermocouples are imbedded in the samples with a spacing of 15 mm. The lateral sides of the samples are insulated to ensure one-dimensional heat transfer through the samples.



(a) With two samples of SS316 in the apparatus, the heater draws 0.353 A at 100 V, and the differential thermocouples indicate $\Delta T_1 = \Delta T_2 = 25.0°C$. What is the thermal conductivity of the stainless steel sample material? What is the average temperature of the samples? Compare your result with the thermal conductivity value reported for this material in Table A.1.

(b) By mistake, an Armco iron sample is placed in the lower position of the apparatus with one of the SS316 samples from part (a) in the upper portion. For this situation, the heater draws 0.601 A at 100 V, and the differential thermocouples indicate $\Delta T_1 = \Delta T_2 = 15.0°C$. What are the thermal conductivity and average temperature of the Armco iron sample?

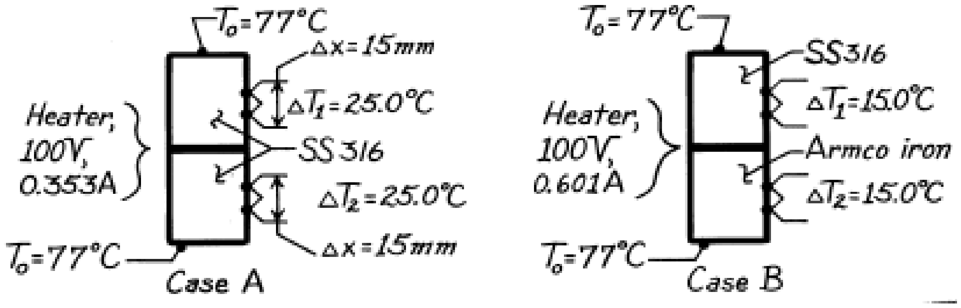
(c) What is the advantage in constructing the apparatus with two identical samples sandwiching the heater rather than with a single heater–sample combination? When would heat leakage out of the lateral surfaces of the samples become significant? Under what conditions would you expect $\Delta T_1 \neq \Delta T_2$?

PROBLEM 2.17

KNOWN: Electrical heater sandwiched between two identical cylindrical (30 mm dia. × 60 mm length) samples whose opposite ends contact plates maintained at T_o.

FIND: (a) Thermal conductivity of SS316 samples for the prescribed conditions (A) and their average temperature, (b) Thermal conductivity of Armco iron sample for the prescribed conditions (B), (c) Comment on advantages of experimental arrangement, lateral heat losses, and conditions for which $\Delta T_1 \neq \Delta T_2$.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional heat transfer in samples, (2) Steady-state conditions, (3) Negligible contact resistance between materials.

PROPERTIES: *Table A.2*, Stainless steel 316 $(T=400\text{ K})$: $k_{ss} = 15.2\text{ W/m}\cdot\text{K}$; Armco iron $(T=380\text{ K})$: $k_{iron} = 67.2\text{ W/m}\cdot\text{K}$.

ANALYSIS: (a) For Case A recognize that half the heater power will pass through each of the samples which are presumed identical. Apply Fourier's law to a sample

$$q = kA_c\frac{\Delta T}{\Delta x}$$

$$k = \frac{q\Delta x}{A_c\Delta T} = \frac{0.5(100\text{V}\times 0.353\text{A})\times 0.015\text{ m}}{\pi(0.030\text{ m})^2/4\times 25.0^\circ\text{C}} = 15.0\text{ W/m}\cdot\text{K}. \qquad <$$

The total temperature drop across the length of the sample is $\Delta T_1(L/\Delta x)$ = 25°C (60 mm/15 mm) = 100°C. Hence, the heater temperature is T_h = 177°C. Thus the average temperature of the sample is

$$T = (T_o + T_h)/2 = 127^\circ\text{C} = 400\text{ K}. \qquad <$$

We compare the calculated value of k with the tabulated value (see above) at 400 K and note the good agreement.

(b) For Case B, we assume that the thermal conductivity of the SS316 sample is the same as that found in Part (a). The heat rate through the Armco iron sample is

Continued .....

PROBLEM 2.17 (Cont.)

$$q_{iron} = q_{heater} - q_{ss} = 100\,V \times 0.601\,A - 15.0\ W/m \cdot K \times \frac{\pi(0.030\ m)^2}{4} \times \frac{15.0^\circ C}{0.015\ m}$$

$$q_{iron} = (60.1 - 10.6)\,W = 49.5\ W$$

where

$$q_{ss} = k_{ss} A_c \Delta T_2 / \Delta x_2.$$

Applying Fourier's law to the iron sample,

$$k_{iron} = \frac{q_{iron} \Delta x_2}{A_c \Delta T_2} = \frac{49.5\ W \times 0.015\ m}{\pi(0.030\ m)^2 / 4 \times 15.0^\circ C} = 70.0\ W/m \cdot K \qquad <$$

The total drop across the iron sample is 15°C(60/15) = 60°C; the heater temperature is (77 + 60)°C = 137°C. Hence the average temperature of the iron sample is

$$\bar{T} = (137 + 77)^\circ C/2 = 107^\circ C = 380\ K. \qquad <$$

We compare the computed value of k with the tabulated value (see above) at 380 K and note the good agreement.

(c) The principal advantage of having two identical samples is the assurance that all the electrical power dissipated in the heater will appear as equivalent heat flows through the samples. With only one sample, heat can flow from the backside of the heater even though insulated.

Heat leakage out the lateral surfaces of the cylindrically shaped samples will become significant when the sample thermal conductivity is comparable to that of the insulating material. Hence, the method is suitable for metallics, but must be used with caution on nonmetallic materials.

For any combination of materials in the upper and lower position, we expect $\Delta T_1 = \Delta T_2$. However, if the insulation were improperly applied along the lateral surfaces, it is possible that heat leakage will occur, causing $\Delta T_1 \neq \Delta T_2$.

2.28 Uniform internal heat generation at $\dot{q} = 5 \times 10^7\ \mathrm{W/m^3}$ is occurring in a cylindrical nuclear reactor fuel rod of 50-mm diameter, and under steady-state conditions the temperature distribution is of the form $T(r) = a + br^2$, where T is in degrees Celsius and r is in meters, while $a = 800°\mathrm{C}$ and $b = -4.167 \times 10^5\ °\mathrm{C/m^2}$. The fuel rod properties are $k = 30\ \mathrm{W/m \cdot K}$, $\rho = 1100\ \mathrm{kg/m^3}$, and $c_p = 800\ \mathrm{J/kg \cdot K}$.

(a) What is the rate of heat transfer per unit length of the rod at $r = 0$ (the centerline) and at $r = 25$ mm (the surface)?

(b) If the reactor power level is suddenly increased to $\dot{q}_2 = 10^8\ \mathrm{W/m^3}$, what is the initial time rate of temperature change at $r = 0$ and $r = 25$ mm?

PROBLEM 2.28

KNOWN: Steady-state temperature distribution in a cylindrical rod having uniform heat generation of $\dot{q}_1 = 5\times10^7\ \mathrm{W/m^3}$.

FIND: (a) Steady-state centerline and surface heat transfer rates per unit length, q_r'. (b) Initial time rate of change of the centerline and surface temperatures in response to a change in the generation rate from $\dot{q}_1$ to $\dot{q}_2 = 10^8\ \mathrm{W/m^3}$.

SCHEMATIC:

ASSUMPTIONS: (1) One-dimensional conduction in the r direction, (2) Uniform generation, and (3) Steady-state for $\dot{q}_1 = 5\times10^7\ \mathrm{W/m^3}$.

ANALYSIS: (a) From the rate equations for cylindrical coordinates,

$$q_r'' = -k\frac{\partial T}{\partial r} \qquad q = -kA_r\frac{\partial T}{\partial r}.$$

Hence,

$$q_r = -k(2\pi r L)\frac{\partial T}{\partial r}$$

or

$$q_r' = -2\pi k r\frac{\partial T}{\partial r} \tag{1}$$

where $\partial T/\partial r$ may be evaluated from the prescribed temperature distribution, T(r).

At r = 0, the gradient is $(\partial T/\partial r) = 0$. Hence, from Equation (1) the heat rate is

$$q_r'(0) = 0. \qquad <$$

At $r = r_o$, the temperature gradient is

$$\left.\frac{\partial T}{\partial r}\right]_{r=r_o} = -2\left[4.167\times10^5\frac{\mathrm{K}}{\mathrm{m^2}}\right](r_o) = -2\left(4.167\times10^5\right)(0.025\mathrm{m})$$

$$\left.\frac{\partial T}{\partial r}\right]_{r=r_o} = -0.208\times10^5\ \mathrm{K/m.}$$

PROBLEM 2.28 (Cont.)

Hence, the heat rate at the outer surface ($r = r_o$) per unit length is

$$q_r'(r_o) = -2\pi[30 \text{ W/m}\cdot\text{K}](0.025\text{m})\left[-0.208\times10^5 \text{ K/m}\right]$$

$$q_r'(r_o) = 0.980\times10^5 \text{ W/m.}$$ <

(b) Transient (time-dependent) conditions will exist when the generation is changed, and for the prescribed assumptions, the temperature is determined by the following form of the heat equation, Equation 2.26

$$\frac{1}{r}\frac{\partial}{\partial r}\left[kr\frac{\partial T}{\partial r}\right]+\dot{q}_2 = \rho c_p\frac{\partial T}{\partial t}$$

Hence

$$\frac{\partial T}{\partial t} = \frac{1}{\rho c_p}\left[\frac{1}{r}\frac{\partial}{\partial r}\left[kr\frac{\partial T}{\partial r}\right]+\dot{q}_2\right].$$

However, initially (at t = 0), the temperature distribution is given by the prescribed form, $T(r) = 800 - 4.167\times10^5 r^2$, and

$$\frac{1}{r}\frac{\partial}{\partial r}\left[kr\frac{\partial T}{\partial r}\right] = \frac{k}{r}\frac{\partial}{\partial r}\left[r\left(-8.334\times10^5\cdot r\right)\right]$$

$$= \frac{k}{r}\left(-16.668\times10^5\cdot r\right)$$

$$= 30 \text{ W/m}\cdot\text{K}\left[-16.668\times10^5 \text{ K/m}^2\right]$$

$$= -5\times10^7 \text{ W/m}^3 \quad (\text{the original } \dot{q}=\dot{q}_1).$$

Hence, everywhere in the wall,

$$\frac{\partial T}{\partial t} = \frac{1}{1100 \text{ kg/m}^3\times800 \text{ J/kg}\cdot\text{K}}\left[-5\times10^7+10^8\right] \text{ W/m}^3$$

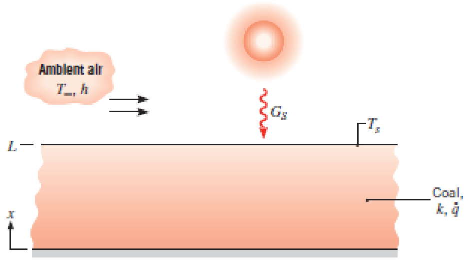
or

$$\frac{\partial T}{\partial t} = 56.82 \text{ K/s.}$$ <

COMMENTS: (1) The value of $(\partial T/\partial t)$ will decrease with increasing time, until a new steady-state condition is reached and once again $(\partial T/\partial t) = 0$. (2) By applying the energy conservation requirement, Equation 1.12c, to a unit length of the rod for the steady-state condition, $\dot{E}_{in}' - \dot{E}_{out}' + \dot{E}_{gen}' = 0$.

Hence $q_r'(0) - q_r'(r_o) = -\dot{q}_1\left(\pi r_o^2\right)$.

2.42 A plane layer of coal of thickness $L = 1$ m experiences uniform volumetric generation at a rate of $\dot{q} = 20$ W/m^3 due to slow oxidation of the coal particles. Averaged over a daily period, the top surface of the layer transfers heat by convection to ambient air for which $h = 5$ W/m$^2 \cdot$K and $T_\infty = 25$°C, while receiving solar irradiation in the amount $G_S = 400$ W/m^2. Irradiation from the atmosphere may be neglected. The solar absorptivity and emissivity of the surface are each $\alpha_S = \varepsilon = 0.95$.



(a) Write the steady-state form of the heat diffusion equation for the layer of coal. Verify that this equation is satisfied by a temperature distribution of the form

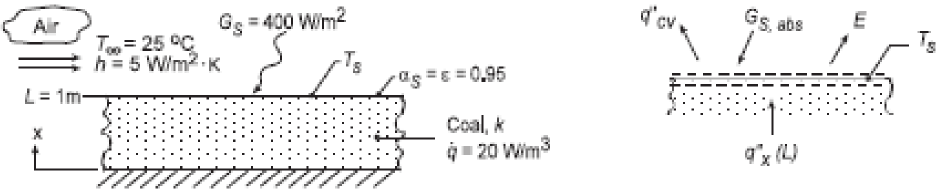
$$T(x) = T_s + \frac{\dot{q}L^2}{2k}\left(1 - \frac{x^2}{L^2}\right)$$

From this distribution, what can you say about conditions at the bottom surface ($x = 0$)? Sketch the temperature distribution and label key features.

(b) Obtain an expression for the rate of heat transfer by conduction per unit area at $x = L$. Applying an energy balance to a control surface about the top surface of the layer, obtain an expression for T_s. Evaluate T_s and $T(0)$ for the prescribed conditions.

(c) Daily average values of G_S and h depend on a number of factors, such as time of year, cloud cover, and wind conditions. For $h = 5$ W/m$^2 \cdot$K, compute and plot T_S and $T(0)$ as a function of G_S for $50 \le G_S \le 500$ W/m^2. For $G_S = 400$ W/m^2, compute and plot T_S and $T(0)$ as a function of h for $5 \le h \le 50$ W/m$^2 \cdot$K.

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional conduction, (2) Uniform volumetric heat generation, (3) Constant properties, (4) Negligible irradiation from the surroundings, and (5) Steady-state conditions.

PROPERTIES: *Table A.3*, Coal (300K): k = 0.26 W/m.K

ANALYSIS: (a) For one-dimensional, steady-state conduction with uniform volumetric heat generation and constant properties the heat diffusion equation (HDE) follows from Eq. 2.22,

$$\frac{d}{dx}\left(\frac{dT}{dx}\right)+\frac{\dot{q}}{k}=0 \qquad (1)$$ <

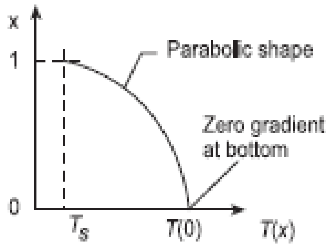
Substituting the temperature distribution into the HDE, Eq. (1),

$$T(x)=T_s+\frac{\dot{q}L^2}{2k}\left(1-\frac{x^2}{L^2}\right) \qquad \frac{d}{dx}\left[0+\frac{\dot{q}L^2}{2k}\left(0-\frac{2x}{L^2}\right)\right]+\frac{\dot{q}}{k}?=?0 \qquad (2,3)$$

we find that it does indeed satisfy the HDE for all values of x. <

From Eq. (2), note that the temperature distribution must be quadratic, with maximum value at x = 0. At x = 0, the heat flux is

$$q''_x(0)=-k\left.\frac{dT}{dx}\right)_{x=0}=-k\left[0+\frac{\dot{q}L^2}{2k}\left(0-\frac{2x}{L^2}\right)\right]_{x=0}=0$$



so that the gradient at x = 0 is zero. Hence, the bottom is insulated.

(b) From an overall energy balance on the pile, the conduction heat flux at the surface must be

$$q''_x(L)=\dot{E}''_g=\dot{q}L$$ <

From a surface energy balance per unit area shown in the schematic above,

$$\dot{E}_{in} - \dot{E}_{out} + \dot{E}_g = 0 \qquad q''_x(L) - q''_{conv} + G_{S,abs} - E = 0$$

$$\dot{q}L - h(T_s - T_\infty) + 0.95G_S - \varepsilon\sigma T_s^4 = 0 \tag{4}$$

$$20\,W/m^3 \times 1\,m - 5\,W/m^2K(T_s - 298\,K) + 0.95 \times 400\,W/m^2 - 0.95 \times 5.67 \times 10^{-8}\,W/m^2K^4 T_s^4 = 0$$

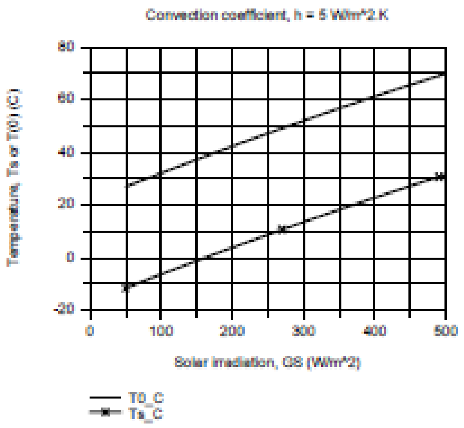
$$T_s = 295.7\,K = 22.7°C$$ <

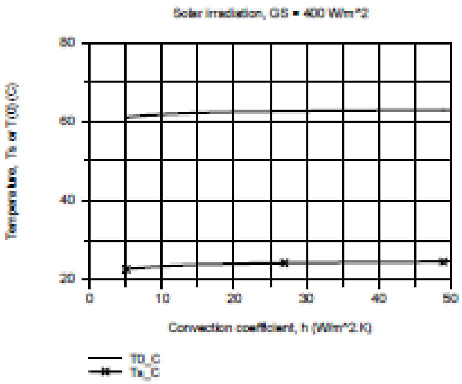
From Eq. (2) with x = 0, find

$$T(0) = T_s + \frac{\dot{q}L^2}{2k} = 22.7°C + \frac{20\,W/m^3 \times (1m)^2}{2 \times 0.26\,W/m\cdot K} = 61.1°C \tag{5}$$ <

where the thermal conductivity for coal was obtained from Table A.3.

(c) Two plots are generated using Eq. (4) and (5) for T_s and T(0), respectively; (1) with h = 5 W/m^2·K for $50 \le G_S \le 500$ W/m^2 and (2) with G_S = 400 W/m^2 for $5 \le h \le 50$ W/m^2·K.

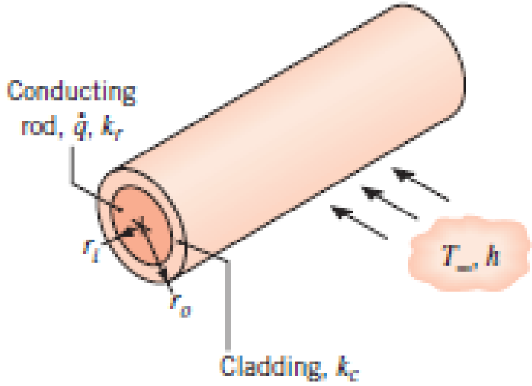




From the T vs. h plot with G_S = 400 W/m^2, note that the convection coefficient does not have a major influence on the surface or bottom coal pile temperatures. From the T vs. G_S plot with h = 5 W/m^2·K, note that the solar irradiation has a very significant effect on the temperatures. The fact that T_s is less than the ambient air temperature, T_∞, and, in the case of very low values of G_S, below freezing, is a consequence of the large magnitude of the emissive power E.

COMMENTS: In our analysis we ignored irradiation from the sky, an environmental radiation effect you'll consider in Chapter 12. Treated as large isothermal surroundings, $G_{sky} = \sigma T_{sky}^4$ where T_{sky} = -30°C for very clear conditions and nearly air temperature for cloudy conditions. For low G_S conditions we should consider G_{sky}, the effect of which will be to predict higher values for T_s and T(0).

2.48 Passage of an electric current through a long conducting rod of radius r_i and thermal conductivity k_r results in uniform volumetric heating at a rate of $\dot{q}$. The conducting rod is wrapped in an electrically nonconducting cladding material of outer radius r_o and thermal conductivity k_c, and convection cooling is provided by an adjoining fluid.



For steady-state conditions, write appropriate forms of the heat equations for the rod and cladding. Express appropriate boundary conditions for the solution of these equations.

ANALYSIS: From Equation 2.26, the appropriate forms of the heat equation are

Conducting Rod:

$$\frac{k_r}{r}\frac{d}{dr}\left(r\frac{dT_r}{dr}\right)+\dot{q}=0 \qquad <$$

Cladding:

$$\frac{d}{dr}\left(r\frac{dT_c}{dr}\right)=0. \qquad <$$

Appropriate boundary conditions are:

(a) $dT_r/dr|_{r=0}=0$ $\qquad <$

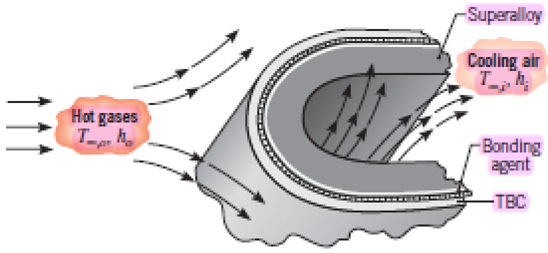
(b) $T_r(r_i)=T_c(r_i)$ $\qquad <$

(c) $k_r\frac{dT_r}{dr}|_{r_i}=k_c\frac{dT_c}{dr}|_{r_i}$ $\qquad <$

(d) $-k_c\frac{dT_c}{dr}|_{r_o}=h\left[T_c(r_o)-T_\infty\right]$ $\qquad <$

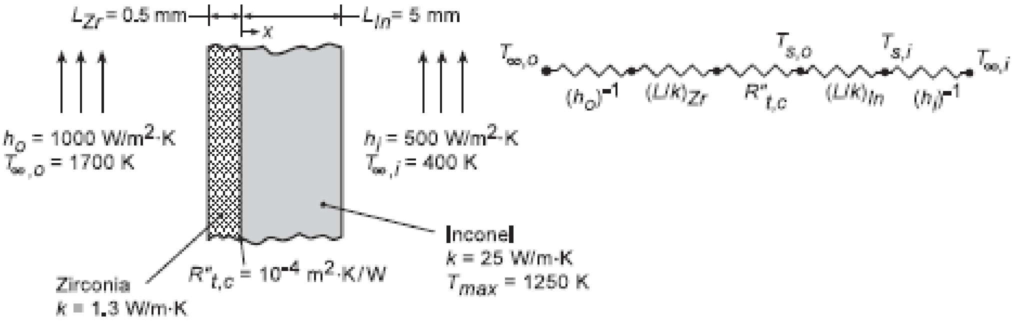
COMMENTS: Condition (a) corresponds to symmetry at the centerline, while the interface conditions at $r = r_i$ (b,c) correspond to requirements of thermal equilibrium and conservation of energy. Condition (d) results from conservation of energy at the outer surface. Note that contact resistance at the interface between the rod and cladding has been neglected.

3.30 The performance of gas turbine engines may be improved by increasing the tolerance of the turbine blades to hot gases emerging from the combustor. One approach to achieving high operating temperatures involves application of a *thermal barrier coating* (TBC) to the exterior surface of a blade, while passing cooling air through the blade. Typically, the blade is made from a high-temperature superalloy, such as Inconel ($k \approx 25$ W/m·K), while a ceramic, such as zirconia ($k \approx 1.3$ W/m·K), is used as a TBC.



Consider conditions for which hot gases at $T_{\infty,o}$ = 1700 K and cooling air at $T_{\infty,i}$ = 400 K provide oute and inner surface convection coefficients of h_o = 1000 W/m²·K and h_i = 500 W/m²·K, respectively. If a 0.5-mm-thick zirconia TBC is attached to a 5-mm-thick Inconel blade wall by means of a metallic bond-ing agent, which provides an interfacial thermal resis-tance of $R''_{t,c} = 10^{-4}$ m²·K/W, can the Inconel be main-tained at a temperature that is below its maximum allowable value of 1250 K? Radiation effects may be neglected, and the turbine blade may be approximated as a plane wall. Plot the temperature distribution with and without the TBC. Are there any limits to the thick-ness of the TBC?

SCHEMATIC:



ASSUMPTIONS: (1) One-dimensional, steady-state conduction in a composite plane wall, (2) Constant properties, (3) Negligible radiation.

ANALYSIS: For a unit area, the total thermal resistance with the TBC is

$$R''_{tot,w} = h_o^{-1} + (L/k)_{Zr} + R''_{t,c} + (L/k)_{In} + h_i^{-1}$$

$$R''_{tot,w} = \left(10^{-3} + 3.85\times10^{-4} + 10^{-4} + 2\times10^{-4} + 2\times10^{-3}\right)\text{m}^2\cdot\text{K/W} = 3.69\times10^{-3}\,\text{m}^2\cdot\text{K/W}$$

With a heat flux of

$$q''_w = \frac{T_{\infty,o} - T_{\infty,i}}{R''_{tot,w}} = \frac{1300\,\text{K}}{3.69\times10^{-3}\,\text{m}^2\cdot\text{K/W}} = 3.52\times10^5\,\text{W/m}^2$$

the inner and outer surface temperatures of the Inconel are

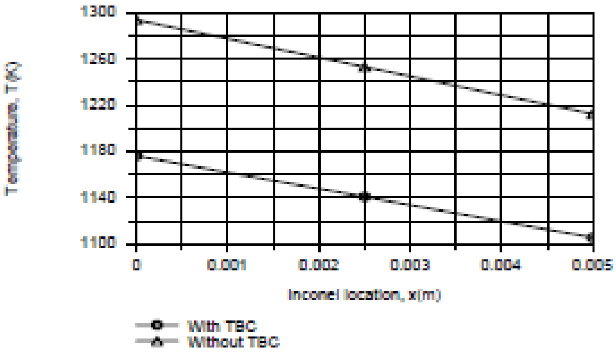
$$T_{s,i(w)} = T_{\infty,i} + \left(q''_w/h_i\right) = 400\,\text{K} + \left(3.52\times10^5\,\text{W/m}^2/500\,\text{W/m}^2\cdot\text{K}\right) = 1104\,\text{K}$$

$$T_{s,o(w)} = T_{\infty,i} + \left[(1/h_i) + (L/k)_{In}\right]q''_w = 400\,\text{K} + \left(2\times10^{-3} + 2\times10^{-4}\right)\text{m}^2\cdot\text{K/W}\left(3.52\times10^5\,\text{W/m}^2\right) = 1174\,\text{K}$$

Without the TBC, $R''_{tot,wo} = h_o^{-1} + (L/k)_{In} + h_i^{-1} = 3.20\times10^{-3}\,\text{m}^2\cdot\text{K/W}$, and $q''_{wo} = \left(T_{\infty,o} - T_{\infty,i}\right)/R''_{tot,wo}$ = (1300 K)/3.20×10⁻³ m²·K/W = 4.06×10⁵ W/m². The inner and outer surface temperatures of the Inconel are then

$$T_{s,i(wo)} = T_{\infty,i} + \left(q''_{wo}/h_i\right) = 400\,\text{K} + \left(4.06\times10^5\,\text{W/m}^2/500\,\text{W/m}^2\cdot\text{K}\right) = 1212\,\text{K}$$

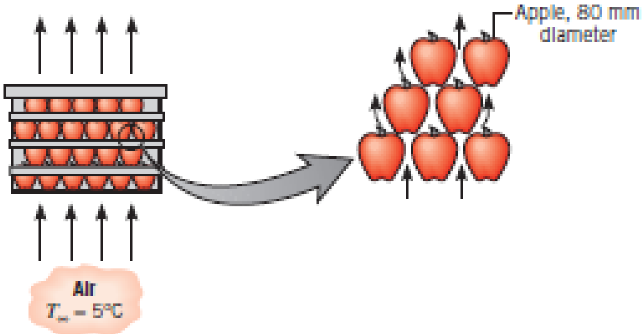
$$T_{s,o(wo)} = T_{\infty,i} + \left[(1/h_i) + (L/k)_{In}\right]q''_{wo} = 400\,\text{K} + \left(2\times10^{-3} + 2\times10^{-4}\right)\text{m}^2\cdot\text{K/W}\left(4.06\times10^5\,\text{W/m}^2\right) = 1293\,\text{K}$$



Use of the TBC facilitates operation of the Inconel below T_{max} = 1250 K.

COMMENTS: Since the durability of the TBC decreases with increasing temperature, which increases with increasing thickness, limits to the thickness are associated with reliability considerations.

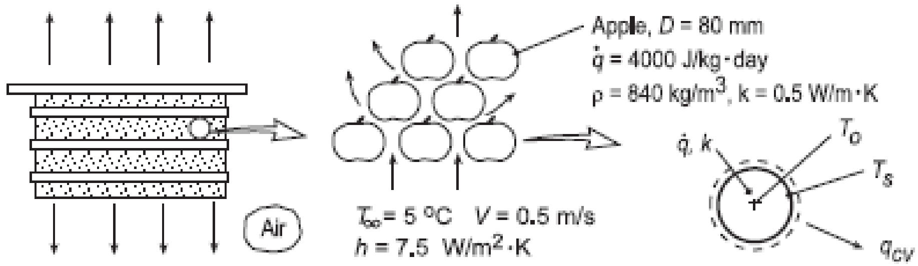
3.105 Unique characteristics of biologically active materials such as fruits, vegetables, and other products require special care in handling. Following harvest and separation from producing plants, glucose is catabolized to produce carbon dioxide, water vapor, and heat, with attendant internal energy generation. Consider a carton of apples, each of 80-mm diameter, which is ventilated with air at 5°C and a velocity of 0.5 m/s. The corresponding value of the heat transfer coefficient is $7.5\ \mathrm{W/m^2 \cdot K}$. Within each apple thermal energy is uniformly generated at a total rate of 4000 J/kg·day. The density and thermal conductivity of the apple are $840\ \mathrm{kg/m^3}$ and $0.5\ \mathrm{W/m \cdot K}$, respectively.



(a) Determine the apple center and surface temperatures.

(b) For the stacked arrangement of apples within the crate, the convection coefficient depends on the velocity as $h = C_1 V^{0.425}$, where $C_1 = 10.1\ \mathrm{W/m^2 \cdot K \cdot (m/s)^{0.425}}$. Compute and plot the center and surface temperatures as a function of the air velocity for $0.1 \le V \le 1$ m/s.

SCHEMATIC:



ASSUMPTIONS: (1) Apples can be modeled as spheres, (2) Each apple experiences flow of ventilation air at T_∞ = 5°C, (3) One-dimensional radial conduction, (4) Constant properties and (5) Uniform heat generation.

ANALYSIS: (a) From Eq. C.24, the temperature distribution in a solid sphere (apple) with uniform generation is

$$T(r) = \frac{\dot{q}r_o^2}{6k}\left(1 - \frac{r^2}{r_o^2}\right) + T_s \tag{1}$$

To determine T_s, perform an energy balance on the apple as shown in the sketch above, with volume V = $4/3\,\pi r_o^3$,

$$\dot{E}_{in} - \dot{E}_{out} + \dot{E}_g = 0 \qquad\qquad -q_{cv} + \dot{q}\forall = 0$$

$$-h\left(4\pi r_o^2\right)\left(T_s - T_\infty\right) + \dot{q}\left(4\pi r_o^3/3\right) = 0 \tag{2}$$

$$-7.5\,\mathrm{W/m^2\cdot K}\left(4\pi\times 0.040^2\,\mathrm{m^2}\right)\left(T_s - 5^\circ\mathrm{C}\right) + 38.9\,\mathrm{W/m^3}\left(4\pi\times 0.040^3\,\mathrm{m^3}/3\right) = 0$$

where the volumetric generation rate is

$$\dot{q} = 4000\,\mathrm{J/kg\cdot day}$$

$$\dot{q} = 4000\,\mathrm{J/kg\cdot day}\times 840\,\mathrm{kg/m^3}\times(1\,\mathrm{day}/24\,\mathrm{hr})\times(1\,\mathrm{hr}/3600\,\mathrm{s})$$

$$\dot{q} = 38.9\,\mathrm{W/m^3}$$

and solving for T_s, find

$$T_s = 5.14^\circ\mathrm{C}$$ <

From Eq. (1), at r = 0, with T_s, find

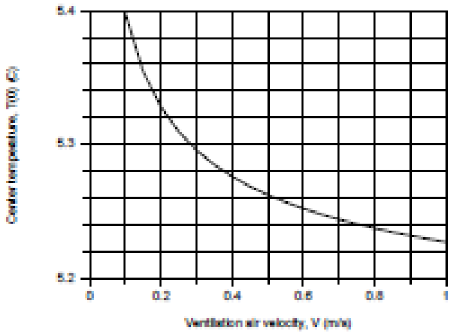
$$T(0) = \frac{38.9\,\mathrm{W/m^3}\times 0.040^2\,\mathrm{m^2}}{6\times 0.5\,\mathrm{W/m\cdot K}} + 5.14^\circ\mathrm{C} = 0.12^\circ\mathrm{C} + 5.14^\circ\mathrm{C} = 5.26^\circ\mathrm{C}$$ <

Continued...

(b) With the convection coefficient depending upon velocity,

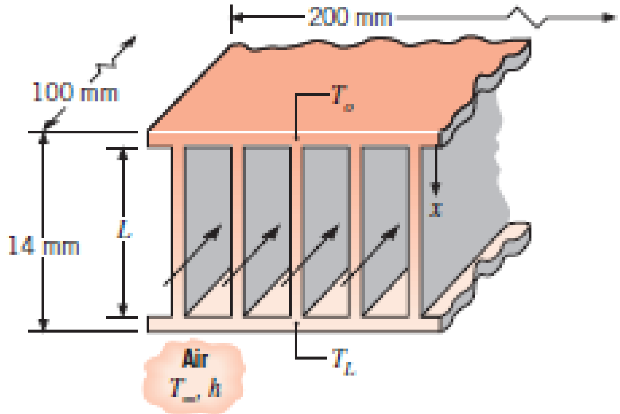
$$h = C_1 V^{0.425}$$

with $C_1 = 10.1$ W/m^2·K·(m/s)$^{0.425}$, and using the energy balance of Eq. (2), calculate and plot T_s as a function of ventilation air velocity V. With very low velocities, the center temperature is nearly 0.5°C higher than the air. From our earlier calculation we know that $T(0) - T_s = 0.12$°C and is independent of V.



COMMENTS: (1) While the temperature within the apple is nearly isothermal, the center temperature will track the ventilation air temperature which will increase as it passes through stacks of cartons.

3.142 Finned passages are frequently formed between parallel plates to enhance convection heat transfer in compact heat exchanger cores. An important application is in electronic equipment cooling, where one or more air-cooled stacks are placed between heat-dissipating electrical components. Consider a single stack of rectangular fins of length L and thickness t, with convection conditions corresponding to h and T_∞.



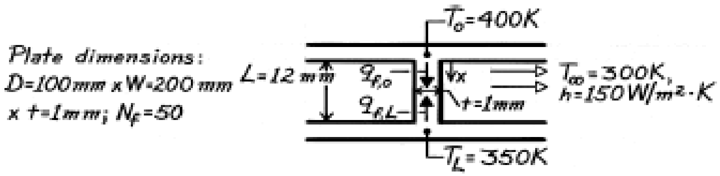
(a) Obtain expressions for the fin heat transfer rates, $q_{f,o}$ and $q_{f,L}$, in terms of the base temperatures, T_o and T_L.

(b) In a specific application, a stack that is 200 mm wide and 100 mm deep contains 50 fins, each of length $L = 12$ mm. The entire stack is made from aluminum, which is everywhere 1.0 mm thick. If temperature limitations associated with electrical components joined to opposite plates dictate maximum allowable plate temperatures of $T_o = 400$ K and $T_L = 350$ K, what are the corresponding maximum power dissipations if $h = 150$ W/m^2-K and $T_\infty = 300$ K?

KNOWN: Arrangement of fins between parallel plates. Temperature and convection coefficient of air flow in finned passages. Maximum allowable plate temperatures.

FIND: (a) Expressions relating fin heat transfer rates to end temperatures, (b) Maximum power dissipation for each plate.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional conduction in fins, (3) Constant properties, (4) Negligible radiation, (5) Uniform h, (6) Negligible variation in T_∞, (7) Negligible contact resistance.

PROPERTIES: *Table A.1*, Aluminum (pure), 375 K: k = 240 W/m·K.

ANALYSIS: (a) The general solution for the temperature distribution in a fin is

$$\theta(x) \equiv T(x) - T_\infty = C_1 e^{mx} + C_2 e^{-mx}$$

Boundary conditions: $\theta(0) = \theta_o = T_o - T_\infty$, $\quad \theta(L) = \theta_L = T_L - T_\infty$.

Hence $\quad \theta_o = C_1 + C_2 \qquad \theta_L = C_1 e^{mL} + C_2 e^{-mL}$

$$\theta_L = C_1 e^{mL} + (\theta_o - C_1) e^{-mL}$$

$$C_1 = \frac{\theta_L - \theta_o e^{-mL}}{e^{mL} - e^{-mL}} \qquad C_2 = \theta_o - \frac{\theta_L - \theta_o e^{-mL}}{e^{mL} - e^{-mL}} = \frac{\theta_o e^{mL} - \theta_L}{e^{mL} - e^{-mL}}.$$

Hence
$$\theta(x) = \frac{\theta_L e^{mx} - \theta_o e^{m(x-L)} + \theta_o e^{m(L-x)} - \theta_L e^{-mx}}{e^{mL} - e^{-mL}}$$

$$\theta(x) = \frac{\theta_o\left[e^{m(L-x)} - e^{-m(L-x)}\right] + \theta_L\left(e^{mx} - e^{-mx}\right)}{e^{mL} - e^{-mL}}$$

$$\theta(x) = \frac{\theta_o \sinh m(L-x) + \theta_L \sinh mx}{\sinh mL}.$$

The fin heat transfer rate is then

$$q_f = -kA_c \frac{dT}{dx} = -kDt\left[-\frac{\theta_o m}{\sinh mL}\cosh m(L-x) + \frac{\theta_L m}{\sinh mL}\cosh mx\right].$$

Hence
$$q_{f,o} = kDt\left(\frac{\theta_o m}{\tanh mL} - \frac{\theta_L m}{\sinh mL}\right) \qquad <$$

$$q_{f,L} = kDt\left(\frac{\theta_o m}{\sinh mL} - \frac{\theta_L m}{\tanh mL}\right). \qquad <$$

(b) $$m = \left(\frac{hP}{kA_c}\right)^{1/2} = \left(\frac{150\ \text{W/m}^2\cdot\text{K}(2\times0.1\ \text{m}+2\times0.001\ \text{m})}{240\ \text{W/m}\cdot\text{K}\times0.1\ \text{m}\times0.001\ \text{m}}\right)^{1/2} = 35.5\ \text{m}^{-1}$$

$$mL = 35.5\ \text{m}^{-1}\times0.012\ \text{m} = 0.43$$

$$\sinh mL = 0.439 \qquad \tanh mL = 0.401 \qquad \theta_o = 100\ \text{K} \qquad \theta_L = 50\ \text{K}$$

$$q_{f,o} = 240\ \text{W/m}\cdot\text{K}\times0.1\ \text{m}\times0.001\ \text{m}\left(\frac{100\ \text{K}\times35.5\ \text{m}^{-1}}{0.401} - \frac{50\ \text{K}\times35.5\ \text{m}^{-1}}{0.439}\right)$$

$q_{f,o} = 115.4\ \text{W}$ (*from* the top plate)

$$q_{f,L} = 240\ \text{W/m}\cdot\text{K}\times0.1\ \text{m}\times0.001\ \text{m}\left(\frac{100\ \text{K}\times35.5\ \text{m}^{-1}}{0.439} - \frac{50\ \text{K}\times35.5\ \text{m}^{-1}}{0.401}\right)$$

$q_{f,L} = 87.8\ \text{W}.$ (*into* the bottom plate)

Maximum power dissipations are therefore

$$q_{o,max} = N_f q_{f,o} + (W - N_f t)Dh\theta_o$$

$$q_{o,max} = 50\times115.4\ \text{W} + (0.200 - 50\times0.001)\text{m}\times0.1\ \text{m}\times150\ \text{W/m}^2\cdot\text{K}\times100\ \text{K}$$

$$q_{o,max} = 5770\ \text{W} + 225\ \text{W} = 5995\ \text{W}$$ <

$$q_{L,max} = -N_f q_{f,L} + (W - N_f t)Dh\theta_o$$

$$q_{L,max} = -50\times87.8\text{W} + (0.200 - 50\times0.001)\text{m}\times0.1\ \text{m}\times150\ \text{W/m}^2\cdot\text{K}\times50\ \text{K}$$

$$q_{L,max} = -4390\ \text{W} + 112\text{W} = -4278\ \text{W}.$$ <

COMMENTS: (1) It is of interest to determine the air velocity needed to prevent excessive heating of the air as it passes between the plates. If the air temperature change is restricted to $\Delta T_\infty = 5$ K, its flowrate must be

$$\dot{m}_{air} = \frac{q_{tot}}{c_p \Delta T_\infty} = \frac{1717\ \text{W}}{1007\ \text{J/kg}\cdot\text{K}\times5\ \text{K}} = 0.34\ \text{kg/s}.$$

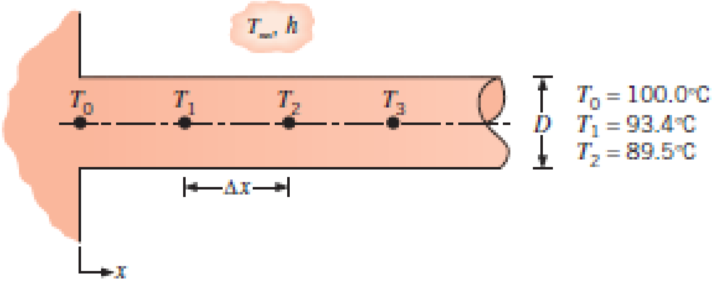
Its mean velocity is then

$$V_{air} = \frac{\dot{m}_{air}}{\rho_{air} A_c} = \frac{0.34\ \text{kg/s}}{1.16\ \text{kg/m}^3\times0.012\ \text{m}(0.2 - 50\times0.001)\text{m}} = 163\ \text{m/s}.$$

Such a velocity would be impossible to maintain. To reduce it to a reasonable value, e.g. 10 m/s, A_c would have to be increased substantially by increasing W (and hence the space between fins) and by increasing L. The present configuration is impractical from the standpoint that 1717 W could not be transferred to air in such a small volume.

(2) A negative value of $q_{L,max}$ implies that the bottom plate must be cooled externally to maintain the plate at 350 K.

4.62 A steady-state, finite-difference analysis has been performed on a cylindrical fin with a diameter of 12 mm and a thermal conductivity of 15 W/m·K. The convection process is characterized by a fluid temperature of 25°C and a heat transfer coefficient of 25 W/m^2·K.



(a) The temperatures for the first three nodes, separated by a spatial increment of $x = 10$ mm, are given in the sketch. Determine the fin heat rate.

(b) Determine the temperature at node 3, T_3.

18–22 To warm up some milk for a baby, a mother pours milk into a thin-walled cylindrical container whose diameter is 6 cm. The height of the milk in the container is 7 cm. She then places the container into a large pan filled with hot water at 70°C. The milk is stirred constantly so that its temperature is uniform at all times. If the heat transfer coefficient between the water and the container is 120 $W/m^2 \cdot K$, determine how long it will take for the milk to warm up from 3°C to 38°C. Assume the entire surface area of the cylindrical container (including the top and bottom) is in thermal contact with the hot water. Take the properties of the milk to be the same as those of water. Can the milk in this case be treated as a lumped system? Why? *Answer:* 4.50 min

18–43 A heated 6-mm-thick Pyroceram plate ($\rho = 2600$ kg/m^3, $c_p = 808$ J/kg·K, $k = 3.98$ W/m·K, and $\alpha = 1.89 \times 10^{-6}$ m^2/s) is being cooled in a room with air temperature of 25°C and convection heat transfer coefficient of 13.3 W/m^2·K. The heated Pyroceram plate had an initial temperature of 500°C, and it is allowed to cool for 286 s. If the mass of the Pyroceram plate is 10 kg, determine the heat transfer from the Pyroceram plate during the cooling process using the analytical one-term approximation method.

18–54 For heat transfer purposes, an egg can be considered to be a 5.5-cm-diameter sphere having the properties of water. An egg that is initially at 4.3°C is dropped into boiling water at 100°C. The heat transfer coefficient at the surface of the egg is estimated to be 800 $W/m^2 \cdot K$. If the egg is considered cooked when its center temperature reaches 71°C, determine how long the egg should be kept in the boiling water. Solve this problem using the analytical one-term approximation method.

18–75 Refractory bricks are used as linings for furnaces, and they generally have low thermal conductivity to minimize heat loss through the furnace walls. Consider a thick furnace wall lined with refractory bricks ($k = 1.0$ W/m·K and $\alpha = 5.08 \times 10^{-7}$ m²/s), where initially the wall has a uniform temperature of 15°C. If the wall surface is subjected to uniform heat flux of 20 kW/m², determine the temperature at the depth of 10 cm from the surface after an hour of heating time.

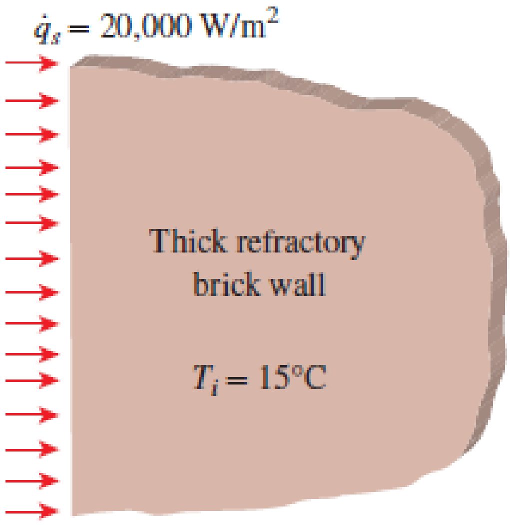


FIGURE P18–75

18–103 Water mains must be placed at sufficient depth below the earth's surface to avoid freezing during extended periods of subfreezing temperatures. Determine the minimum depth at which the water main must be placed at a location where the soil is initially at 15°C and the earth's surface temperature under the worst conditions is expected to remain at −10°C for 75 days. Take the properties of soil at that location to be $k = 0.7$ W/m·K and $\alpha = 1.4 \times 10^{-5}$ m²/s. *Answer:* 7.05 m

5.89 An insurance company has hired you as a consultant to improve their understanding of burn injuries. They are especially interested in injuries induced when a portion of a worker's body comes into contact with machinery that is at elevated temperatures in the range of 50 to 100°C. Their medical consultant informs them that irreversible thermal injury (cell death) will occur in any living tissue that is maintained at $T \geq 48°C$ for a duration $\Delta t \geq 10$ s. They want information concerning the extent of irreversible tissue damage (as measured by distance from the skin surface) as a function of the machinery temperature and the time during which contact is made between the skin and the machinery. Assume that living tissue has a normal temperature of 37°C, is isotropic, and has constant properties equivalent to those of liquid water.

(a) To assess the seriousness of the problem, compute locations in the tissue at which the temperature will reach 48°C after 10 s of exposure to machinery at 50°C and 100°C.

(b) For a machinery temperature of 100°C and $0 \leq t \leq 30$ s, compute and plot temperature histories at tissue locations of 0.5, 1, and 2 mm from the skin.

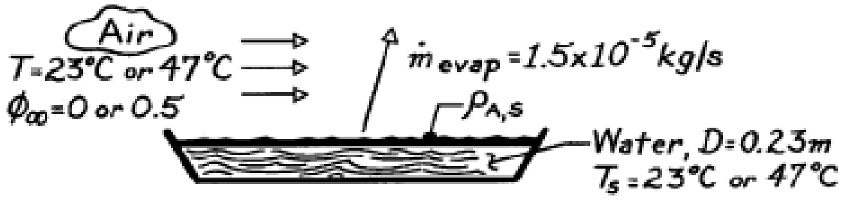
6.53 It is observed that a 230-mm-diameter pan of water at 23°C has a mass loss rate of 1.5×10^{-5} kg/s when the ambient air is dry and at 23°C.

(a) Determine the convection mass transfer coefficient for this situation.

(b) Estimate the evaporation mass loss rate when the ambient air has a relative humidity of 50%.

(c) Estimate the evaporation mass loss rate when the water and ambient air temperatures are 47°C, assuming that the convection mass transfer coefficient remains unchanged and the ambient air is dry.

SCHEMATIC:



ASSUMPTIONS: (1) Water vapor is saturated at liquid interface and may be approximated as a perfect gas.

PROPERTIES: *Table A-6*, Saturated water vapor (T_s = 296K): $\rho_{A,sat} = v_g^{-1} = (49.4\ m^3/kg)^{-1}$ = 0.0202 kg/m^3; (T_s = 320 K): $\rho_{A,sat} = v_g^{-1} = \left(13.98\ m^3/kg\right)^{-1} = 0.0715\ kg/m^3$.

ANALYSIS: (a) Since evaporation is a convection mass transfer process, the rate equation has the form $\dot{m}_{evap} = \bar{h}_m A\left(\rho_{A,s} - \rho_{A,\infty}\right)$ and the mass transfer coefficient is

$$\bar{h}_m = \frac{\dot{m}_{evap}}{\left(\pi D^2/4\right)\left(\rho_{A,s} - \rho_{A,\infty}\right)} = \frac{1.5\times10^{-5}\ kg/s}{(\pi/4)(0.23\ m)^2\ 0.0202\ kg/m^3} = 0.0179\ m/s \quad <$$

with $T_s = T_\infty$ = 23°C and ϕ_∞ = 0.

(b) If the relative humidity of the ambient air is increased to 50%, the ratio of the evaporation rates is

$$\frac{\dot{m}_{evap}(\phi_\infty = 0.5)}{\dot{m}_{evap}(\phi_\infty = 0)} = \frac{\bar{h}_m A\left[\rho_{A,s}(T_s) - \phi_\infty \rho_{A,s}(T_\infty)\right]}{\bar{h}_m A\ \rho_{A,s}(T_s)} = 1 - \phi_\infty \frac{\rho_{A,s}(T_\infty)}{\rho_{A,s}(T_s)}.$$

Hence, $\dot{m}_{evap}(\phi_\infty = 0.5) = 1.5\times10^{-5}\ kg/s\left[1 - 0.5\dfrac{0.0202\ kg/m^3}{0.0202\ kg/m^3}\right] = 0.75\times10^{-5}\ kg/s.$

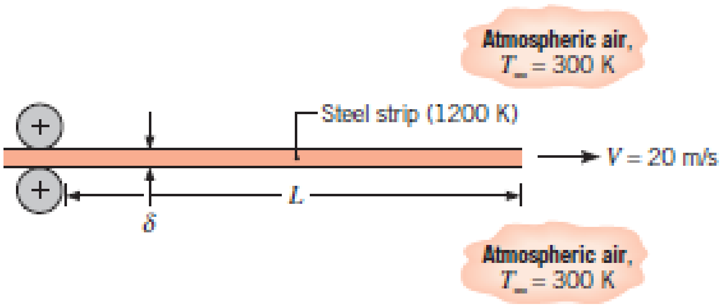
(c) If the temperature of the ambient air is increased from 23°C to 47°C, with ϕ_∞ = 0 for both cases, the ratio of the evaporation rates is

$$\frac{\dot{m}_{evap}\left(T_s = T_\infty = 47^\circ C\right)}{\dot{m}_{evap}\left(T_s = T_\infty = 23^\circ C\right)} = \frac{\bar{h}_m A \rho_{A,s}\left(47^\circ C\right)}{\bar{h}_m A \rho_{A,s}\left(23^\circ C\right)} = \frac{\rho_{A,s}\left(47^\circ C\right)}{\rho_{A,s}\left(23^\circ C\right)}.$$

Hence, $\dot{m}_{evap}\left(T_s = T_\infty = 47^\circ C\right) = 1.5\times10^{-5} kg/s \dfrac{0.0715\ kg/m^3}{0.0202\ kg/m^3} = 5.31\times10^{-5}\ kg/s. \quad <$

COMMENTS: Note the highly nonlinear dependence of the evaporation rate on the water temperature. For a 24°C rise in T_s, $\dot{m}_{evap}$ increases by 350%.

7.32 A steel strip emerges from the hot roll section of a steel mill at a speed of 20 m/s and a temperature of 1200 K. Its length and thickness are $L = 100$ m and $\delta = 0.003$ m, respectively, and its density and specific heat are 7900 kg/m^3 and 640 J/kg·K, respectively.



Accounting for heat transfer from the top and bottom surfaces and neglecting radiation and strip conduction effects, determine the time rate of change of the strip temperature at a distance of 1 m from the leading edge and at the trailing edge. Determine the distance from the leading edge at which the minimum cooling rate is achieved.

ASSUMPTIONS: (1) Constant properties, (2) Negligible radiation, (3) Negligible longitudinal conduction in strip, (4) Critical Reynolds number is 5×10^5.

PROPERTIES: Steel (given): $\rho = 7900$ kg/m^3, $c_p = 640$ J/kg·K. *Table A-4*, Air $(\bar{T} = 750\text{K, 1 atm})$: $\nu = 76.4 \times 10^{-6}$ m^2/s, k = 0.0549 W/m·K, Pr = 0.702.

ANALYSIS: Performing an energy balance for a control mass of unit surface area A_s riding with the strip,

$$-\dot{E}_{out} = dE_{st} / dt$$

$$-2h_x A_s (T - T_\infty) = \rho\delta A_s c_p (dT/dt)$$

$$dT/dt = \frac{-2h_x (T - T_\infty)}{\rho\delta c_p} = -\frac{2(900\text{K})h_x}{7900\ \text{kg/m}^3 (0.003\ \text{m}) 640\ \text{J/kg}\cdot\text{K}} = -0.119 h_x\ (\text{K/s}).$$

At x = 1 m, $\quad Re_x = \dfrac{Vx}{\nu} = \dfrac{20\ \text{m/s}(1\text{m})}{76.4\times10^{-6}\ \text{m}^2/\text{s}} = 2.62\times10^5 < Re_{x,c}$. Hence,

$$h_x = (k/x) 0.332 Re_x^{1/2} Pr^{1/3} = \frac{0.0549\ \text{W/m}\cdot\text{K}}{1\ \text{m}} (0.332)\left(2.62\times10^5\right)^{1/2} (0.702)^{1/3} = 8.29\ \text{W/m}^2\cdot\text{K}$$

and at x = 1m, dT/dt = -0.987 K/s. $\quad <$

At the trailing edge, $Re_x = 2.62\times10^7 > Re_{x,c}$. Hence

$$h_x = (k/x) 0.0296 Re_x^{4/5} Pr^{1/3} = \frac{0.0549\ \text{W/m}\cdot\text{K}}{100\ \text{m}} (0.0296)\left(2.62\times10^7\right)^{4/5} (0.702)^{1/3} = 12.4\ \text{W/m}^2\cdot\text{K}$$

and at x = 100 m, dT/dt = -1.47 K/s. $\quad <$

The minimum cooling rate occurs just before transition; hence, for $Re_{x,c} = 5\times10^5$

$$x_c = 5\times10^5 (\nu/V) = \frac{5\times10^5 \times 76.4\times10^{-6}\ \text{m}^2/\text{s}}{20\ \text{m/s}} = 1.91\ \text{m} \quad <$$

COMMENTS: The cooling rates are very low and would remain low even if radiation were considered. For this reason, hot strip metals are quenched by water and not by air.

7.58 A fine wire of diameter *D* is positioned across a passage to determine flow velocity from heat transfer characteristics. Current is passed through the wire to heat it, and the heat is dissipated to the flowing fluid by convection. The resistance of the wire is determined from electrical measurements, and the temperature is known from the resistance.

(a) For a fluid of arbitrary Prandtl number, develop an expression for its velocity in terms of the difference between the temperature of the wire and the free stream temperature of the fluid.

(b) What is the velocity of an airstream at 1 atm and 25°C, if a wire of 0.5-mm diameter achieves a temperature of 40°C while dissipating 35 W/m?

ASSUMPTIONS: (1) Steady-state conditions, (2) Uniform wire temperature, (3) Negligible radiation.

PROPERTIES: *Table A-4*, Air (T_∞ = 298 K, 1 atm): $\nu = 15.8 \times 10^{-6}$ m^2/s, k = 0.0262 W/m·K, Pr = 0.71; (T_s = 313 K, 1 atm): Pr = 0.705.

ANALYSIS: (a) The rate of heat transfer per unit cylinder length is

$$q' = (q/L) = \bar{h}(\pi D)\ (T_s - T_\infty)$$

where, from the Zhukauskas relation, with Pr ≈ Pr_s,

$$\bar{h} = \frac{k}{D} C\, Re_D^m\, Pr^n = \frac{k}{D} C \left(\frac{VD}{\nu}\right)^m Pr^n$$

Hence,

$$V = \left[\frac{q'}{(k/D) C\, Pr^n (\pi D)\ (T_s - T_\infty)}\right]^{1/m} \left(\frac{\nu}{D}\right).$$ <

(b) Assuming ($10^3 < Re_D < 2 \times 10^5$), C = 0.26, m = 0.6 from Table 7.4. Hence,

$$V = \left[\frac{35\ \text{W/m}}{0.0262\ \text{W/m}\cdot\text{K} \times 0.26 (0.71)^{0.37}\ \pi (40-25)^\circ\text{C}}\right]^{1/0.6} \left(\frac{15.8 \times 10^{-6}\text{m}^2/\text{s}}{5 \times 10^{-4}\text{m}}\right)$$

$$V = 97\ \text{m/s}.$$ <

To verify the assumption of the Reynolds number range, calculate

$$Re_D = \frac{VD}{\nu} = \frac{97\ \text{m/s}\ (5 \times 10^{-4}\text{m})}{15.8 \times 10^{-6}\text{m}^2/\text{s}} = 3074.$$

Hence the assumption was correct.

COMMENTS: The major uncertainty associated with using this method to determine V is that associated with use of the correlation for $\overline{Nu}_D$.

7.94 A tube bank uses an aligned arrangement of 30-mm-diameter tubes with $S_T = S_L = 60$ mm and a tube length of 1 m. There are 10 tube rows in the flow direction ($N_L = 10$) and 7 tubes per row ($N_T = 7$). Air with upstream conditions of $T_\infty = 27°C$ and $V = 15$ m/s is in cross flow over the tubes, while a tube wall temperature of 100°C is maintained by steam condensation inside the tubes. Determine the temperature of air leaving the tube bank, the pressure drop across the bank, and the fan power requirement.

ASSUMPTIONS: (1) Steady-state conditions, (2) Negligible radiation, (3) Air pressure is approximately one atmosphere, (4) Uniform surface temperature.

PROPERTIES: *Table A-4*, Air (300 K, 1 atm): $\rho = 1.1614$ kg/m^3, $c_p = 1007$ J/kg·K, $\nu = 15.89 \times 10^{-6}$ m^2/s, k = 0.0263 W/m·K, Pr = 0.707; (373K): Pr = 0.695.

ANALYSIS: (a) The air temperature increases exponentially, with

$$T_o = T_s - (T_s - T_i)\exp\left(-\frac{\pi D N \bar{h}}{\rho V N_T S_T c_p}\right).$$

With $V_{max} = \frac{S_T}{S_T - D} V = \frac{60}{30} 15 \frac{m}{s} = 30 \frac{m}{s}$; $Re_{D,max} = \frac{30\text{ m/s} \times 0.03\text{ m}}{15.89 \times 10^{-6}\text{ m}^2/\text{s}} = 56,639.$

Tables 7.5 and 7.6 give $C_1 = 0.27$, m = 0.63 and $C_2 = 0.97$. Hence from the Zukauskas correlation,

$$\overline{Nu}_D = 0.27(0.97)(56,639)^{0.63}(0.707)^{0.36}(0.707/0.695)^{1/4} = 229$$

$$\bar{h} = \overline{Nu}_D k/D = 229 \times 0.0263\text{ W/m}\cdot\text{K}/0.03\text{ m} = 201\text{ W/m}^2\cdot\text{K}.$$

Hence,

$$T_o = 373\text{K} - (373 - 300)\text{K}\exp\left(-\frac{\pi \times 0.03\text{ m} \times 70 \times 201\text{ W/m}^2\cdot\text{K}}{1.1614\text{ kg/m}^3 \times 15\text{ m/s} \times 7 \times 0.06\text{ m} \times 1007\text{ J/kg}\cdot\text{K}}\right)$$

$$T_o = 373\text{K} - 73\text{K} \times 0.835 = 312\text{K} = 39°\text{C}.$$ <

(b) With $Re_{D,max} = 5.66 \times 10^4$, $P_L = 2$, $(P_T - 1)/(P_L - 1) = 1$, Fig. 7.14 yields $f \approx 0.19$ and $\chi = 1$. Hence,

$$\Delta p = N_L \chi \left(\frac{\rho V_{max}^2}{2}\right) f = 10\left(\frac{1.1614\text{ kg/m}^3 \times (30\text{ m/s})^2}{2}\right) 0.19 = 993\text{ N/m}^2 = 0.00993\text{ bar}.$$ <

The fan power requirement is

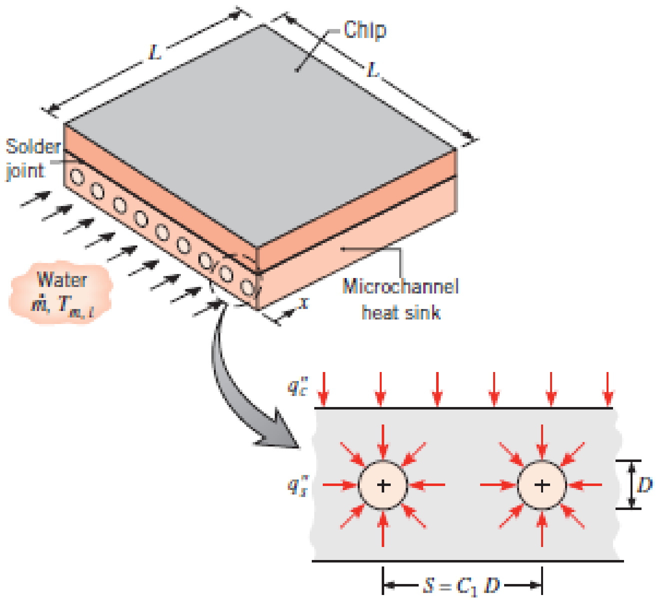
$$P = \dot{m}_a \Delta p/\rho = \rho V N_T S_T L\, \Delta p/\rho = 15\text{ m/s} \times 7 \times 0.06\text{ m} \times 1\text{m} \times 993\text{ N/m}^2 = 6.26\text{ kW}.$$ <

COMMENTS: The heat rate is

$$q = \dot{m}_a c_p (T_o - T_i) = \rho V N_T S_T L\, c_p (T_o - T_i)$$

$$q = 1.1614\text{ kg/m}^3 \times 15\text{ m/s} \times 7 \times 0.06\text{m} \times 1\text{m} \times 1007\text{ J/kg}\cdot\text{K}\ (312 - 300)\text{K} = 88.4\text{ kW}.$$

8.55 A common procedure for cooling a high-performance computer chip involves joining the chip to a heat sink within which circular microchannels are machined. During operation, the chip produces a uniform heat flux q''_c at its interface with the heat sink, while a liquid coolant (water) is routed through the channels. Consider a square chip and heat sink, each L on a side, with microchannels of diameter D and pitch $S = C_1 D$, where the constant C_1 is greater than unity. Water is supplied at an inlet temperature $T_{m,i}$ and a total mass flow rate $\dot{m}$ (for the entire heat sink).

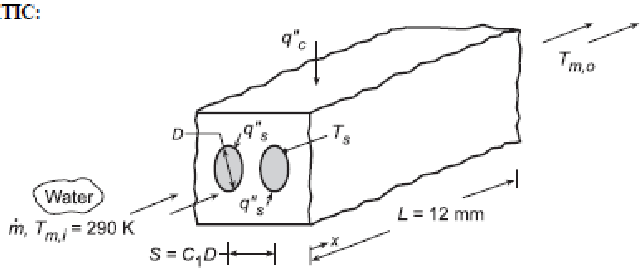


(a) Assuming that q''_c is dispersed in the heat sink such that a uniform heat flux q''_s is maintained at the surface of each channel, obtain expressions for the longitudinal distributions of the mean fluid, $T_m(x)$, and surface, $T_s(x)$, temperatures in each channel. Assume laminar, fully developed flow throughout each channel, and express your results in terms of $\dot{m}$, q''_c, C_1, D, and/or L, as well as appropriate thermophysical properties.

(b) For $L = 12$ mm, $D = 1$ mm, $C_1 = 2$, $q''_c = 20$ W/cm^2, $\dot{m} = 0.010$ kg/s, and $T_{m,i} = 290$ K, compute and plot the temperature distributions $T_m(x)$ and $T_s(x)$.

(c) A common objective in designing such heat sinks is to maximize q''_c while maintaining the heat sink at an acceptable temperature. Subject to prescribed values of $L = 12$ mm and $T_{m,i} = 290$ K and the constraint that $T_{s,max} \leq 50°$C, explore the effect on q''_c of variations in heat sink design and operating conditions.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state, (2)Incompressible liquid with negligible viscous dissipation, (3) All of the chip power dissipation is transferred to the coolant, with a uniform surface heat flux, q_s'', (4) Laminar, fully developed flow, (5) Constant properties.

PROPERTIES: *Table A.6*, Water (assume $\overline{T}_m = T_{m,i} = 290$ K): $c_p = 4184$ J/kg·K, $\mu = 1080 \times 10^{-6}$ N·s/m², k = 0.598 W/m·K, Pr = 7.56.

ANALYSIS: (a) The number of channels passing through the heat sink is $N = L/S = L/C_1D$, and conservation of energy dictates that

$$q_c'' L^2 = N(\pi DL) q_s'' = \pi L^2 q_s''/C_1$$

which yields

$$q_s'' = \frac{C_1 q_c''}{\pi} \qquad (1)$$

With the mass flowrate per channel designated as $\dot{m}_1 = \dot{m}/N$, Eqs. 8.40 and 8.27 yield

$$T_m(x) = T_{m,i} + \frac{q_s'' \pi D}{\dot{m}_1 c_p} x = T_{m,i} + \frac{L q_c''}{\dot{m} c_p} x \qquad (2) <$$

$$T_s(x) = T_m(x) + \frac{q_s''}{h} = T_m(x) + \frac{C_1 q_c''}{\pi h} \qquad (3) <$$

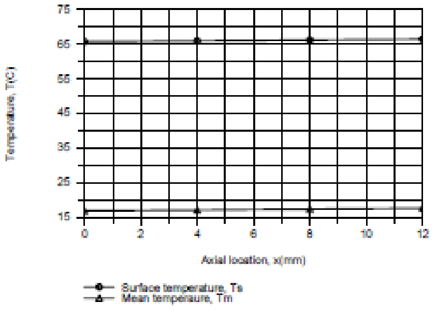
where, for laminar, fully developed flow with uniform q_s'', Eq. 8.53 yields h = 4.36 k/D.

(b) With L = 12 mm, D = 1 mm, $C_1 = 2$ and $\dot{m} = 0.01$ kg/s, it follows that S = 2 mm, N = 6 and $Re_D = 4\dot{m}_1/\pi D\mu = 4(0.01\,\text{kg/s})/6\pi(0.001\,\text{m})1.08\times10^{-3}\,\text{N}\cdot\text{s/m}^2 = 1965$. Hence, the flow is laminar, as assumed, and h = 4.36(0.598 W/m·K/0.001 m) = 2607 W/m²·K. From Eqs. (2) and (3) the outlet mean and surface temperatures are

$$T_{m,o} = 290\,\text{K} + \frac{(0.012\,\text{m})^2\, 20\times10^4\,\text{W/m}^2}{0.01\,\text{kg/s}(4184\,\text{J/kg}\cdot\text{K})} = 290.7\,\text{K} = 17.7^\circ\text{C}$$

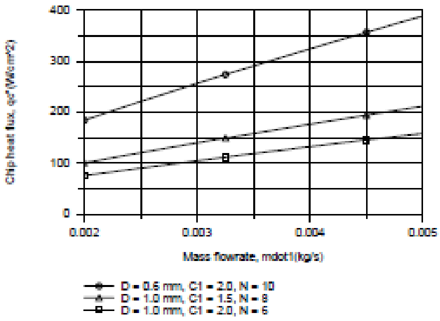
$$T_{s,o} = T_{m,o} + \frac{2}{\pi}\times\frac{20\times10^4\,\text{W/m}^2}{2607\,\text{W/m}^2\cdot\text{K}} = 339.5\,\text{K} = 66.5^\circ\text{C}$$

The axial temperature distributions are as follows



The flowrate is sufficiently large (and the convection coefficient sufficiently low) to render the increase in T_m and T_s with increasing x extremely small.

(c) The desired constraint of $T_{s,max} \le 50°C$ is not met by the foregoing conditions. An obvious and logical approach to achieving improved performance would involve increasing $\dot{m}_1$ such that turbulent flow is maintained in each channel. A value of $\dot{m}_1 > 0.002$ kg/s would provide $Re_D > 2300$ for D = 0.001. Using Eq. 8.60 with n = 0.4 to evaluate Nu_D and accessing the Correlations Toolpad of IHT to explore the effect of variations in $\dot{m}_1$ for different combinations of D and C_1, the following results were obtained.

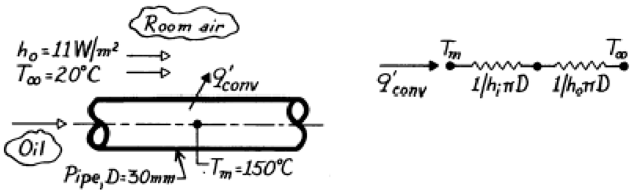


We first note that a significant increase in q_c'' may be obtained by operating the channels in turbulent flow. In addition, there is an obvious advantage to reducing C_1, thereby increasing the number of channels for a fixed channel diameter. The biggest enhancement is associated with reducing the channel diameter, which significantly increases the convection coefficient, as well as the number of channels for fixed C_1. For $\dot{m}_1 = 0.005$ kg/s, h increases from 32,400 to 81,600 $W/m^2 \cdot K$ with decreasing D from 1.0 to 0.6 mm. However, for fixed $\dot{m}_1$, the mean velocity in a channel increases with decreasing D and care must be taken to maintain the flow pressure drop within acceptable limits.

COMMENTS: Although the distribution computed for $T_m(x)$ in part (b) is correct, the distribution for $T_s(x)$ represents an upper limit to actual conditions due to the assumption of fully developed flow throughout the channel.

8.58 Oil at 150°C flows *slowly* through a long, thin-walled pipe of 30-mm inner diameter. The pipe is suspended in a room for which the air temperature is 20°C and the convection coefficient at the outer tube surface is 11 W/m$^2\cdot$K. Estimate the heat loss per unit length of tube.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Tube wall thermal resistance negligible, (3) Fully developed flow, (4) Radiation exchange between pipe and room negligible.

PROPERTIES: *Table A-5*, Unused engine oil (T_m = 150°C = 423K): k = 0.133 W/m·K.

ANALYSIS: The rate equation, for a unit length of the pipe, can be written as

$$q'_{conv} = \frac{(T_m - T_\infty)}{R'_t}$$

where the thermal resistance is comprised of two elements,

$$R'_t = \frac{1}{h_i \pi D} + \frac{1}{h_o \pi D} = \frac{1}{\pi D}\left(\frac{1}{h_i} + \frac{1}{h_o}\right).$$

The convection coefficient for internal flow, h_i, must be estimated from an appropriate correlation. From practical considerations, we recognize that the oil flow rate cannot be large enough to achieve turbulent flow conditions. Hence, the flow is *laminar*, and if the pipe is very long, the flow will be *fully developed*. The appropriate correlation is

$$Nu_D = \frac{h_i D}{k} = 3.66$$

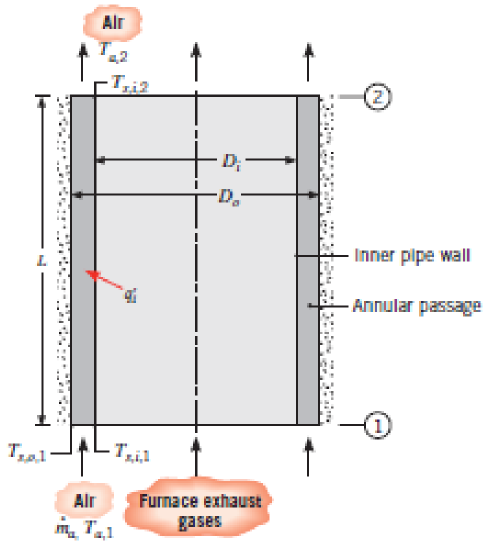
$$h_i = Nu_D\ k/D = 3.66 \times 0.133 \frac{W}{m\cdot K}/0.030\ m = 16.2\ W/m^2\cdot K$$

The heat rate per unit length of the pipe is

$$q'_{conv} = \frac{(150-20)^\circ C}{\frac{1}{\pi(0.030m)}\left(\frac{1}{16.2}+\frac{1}{11}\right)\frac{m^2\cdot K}{W}} = 80.3\ W/m \qquad <$$

COMMENTS: This problem requires making a judgment that the oil flow will be laminar rather than turbulent. Why is this a reasonable assumption? Recognize that the correlation applies to a constant surface temperature condition.

8.98 It is common practice to recover waste heat from an oil- or gas-fired furnace by using the exhaust gases to preheat the combustion air. A device commonly used for this purpose consists of a concentric pipe arrangement for which the exhaust gases are passed through the inner pipe, while the cooler combustion air flows through an annular passage around the pipe.

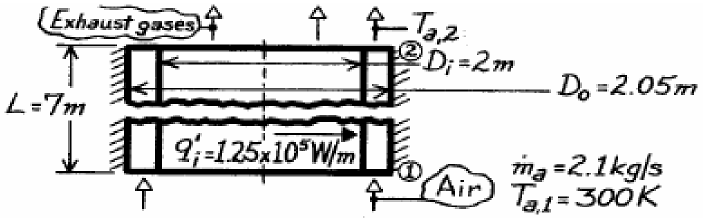


Consider conditions for which there is a uniform heat transfer rate per unit length, $q_i' = 1.25 \times 10^5$ W/m, from the exhaust gases to the pipe inner surface, while air flows through the annular passage at a rate of $\dot{m}_a = 2.1$ kg/s. The thin-walled inner pipe is of diameter $D_i = 2$ m, while the outer pipe, which is well insulated from the surroundings, is of diameter $D_o = 2.05$ m. The air properties may be taken to be $c_p = 1030$ J/kg·K, $\mu = 270 \times 10^{-7}$ N·s/m², $k = 0.041$ W/m·K, and $Pr = 0.68$.

(a) If air enters at $T_{a,1} = 300$ K and $L = 7$ m, what is the air outlet temperature $T_{a,2}$?

(b) If the airflow is fully developed throughout the annular region, what is the temperature of the inner pipe at the inlet ($T_{s,i,1}$) and outlet ($T_{s,i,2}$) sections of the device? What is the outer surface temperature $T_{s,o,1}$ at the inlet?

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Constant properties, (3) Uniform heating of recuperator inner surface, (4) Adiabatic outer surface, (5)Air is ideal gas with negligible viscous dissipation and pressure variation, (6) Fully developed air flow throughout.

PROPERTIES: *Table A-4*, Air (given): $c_p = 1030$ J/kg·K, $\mu = 270 \times 10^{-7}$ N·s/m^2, k = 0.041 W/m·K, Pr = 0.68.

ANALYSIS: (a) From an energy balance on the air

$$q_i' L = \dot{m}_a c_{p,a}\left(T_{a,2} - T_{a,1}\right)$$

$$T_{a,2} = T_{a,1} + \frac{q_i' L}{\dot{m}_a c_{p,a}} = 300\text{ K} + \frac{1.25\times10^5 \text{W/m}\times 7\text{m}}{2.1\text{ kg/s}\times 1030\text{ J/kg}\cdot\text{K}} = 704.5\text{ K} \qquad <$$

(b) The surface temperatures may be evaluated from Eqs. 8.67 and 8.68 with

$$Re_D = \frac{\rho u_m D_h}{\mu} = \frac{\dot{m}_a \left(D_o - D_i\right)}{(\pi/4)\left(D_o^2 - D_i^2\right)\mu} = \frac{4\,\dot{m}_a}{\pi\left(D_o + D_i\right)\mu} = \frac{4(2.1\text{ kg/s})}{\pi(4.05\text{m})\ 270\times10^{-7}\text{N}\cdot\text{s/m}^2}$$

$$Re_D = 24,452$$

the flow is turbulent and from Eq. 8.60

$$h_i \approx h_o \approx \frac{k}{D_h} 0.023\, Re_D^{4/5} Pr^{0.4} = \frac{0.041\text{ W/m}\cdot\text{K}}{0.05\text{ m}} 0.023(24,452)^{4/5}(0.68)^{0.4} = 52\text{ W/m}^2\cdot\text{K}.$$

With $q_i'' = q_i' / \pi D_i = 1.25\times10^5 \text{W/m}/\pi\times 2\text{m} = 19{,}900\text{ W/m}^2$

Eq. 8.67 gives

$$\left(T_{s,i} - T_m\right) = q_i'' / h_i = 19,900\text{ W/m}^2 / 52\text{ W/m}^2\cdot\text{K} = 383\text{K}$$

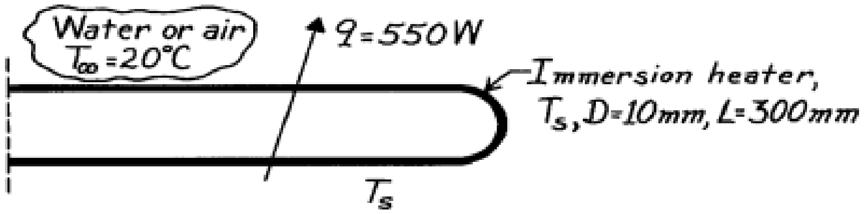
$$T_{s,i,1} = 683\text{K} \qquad T_{s,i,2} = 1087\text{K}. \qquad <$$

From Eq. 8.68, with $q_o'' = 0$, $\left(T_{s,o} - T_m\right) = 0$. Hence

$$T_{s,o,1} = T_{a,1} = 300\text{K}. \qquad <$$

9.61 An electric immersion heater, 10 mm in diameter and 300 mm long, is rated at 550 W. If the heater is horizontally positioned in a large tank of water at 20°C, estimate its surface temperature. Estimate the surface temperature if the heater is accidentally operated in air at 20°C.

SCHEMATIC:



ASSUMPTIONS: (1) Quiescent ambient fluid, (2) Negligible radiative exchange.

PROPERTIES: *Table A-6*, Water and *Table A-4*, Air:

	T(K)	$k \cdot 10^3$(W/m·K)	$\nu \cdot 10^7$(μ/ρ,m^2/s)	$\alpha \cdot 10^7$($k/\rho c_p$,m^2/s)	Pr	$\beta \cdot 10^6$(K^{-1})
Water	315	634	6.25	1.531	4.16	400.4
Air	1500	100	2400	3500	0685	666.7

ANALYSIS: From the rate equation, the surface temperature, T_s, is

$$T_s = T_\infty + q / \left(\pi D L \bar{h}\right) \tag{1}$$

where $\bar{h}$ is estimated by an appropriate correlation. Since such a calculation requires knowledge of T_s, an iteration procedure is required. Begin by assuming for *water* that T_s = 64°C such that T_f = 315K. Calculate the Rayleigh number,

$$Ra_D = \frac{g\beta\Delta T D^3}{\nu\alpha} = \frac{9.8\,m/s^2 \times 400.4 \times 10^{-6}\,K^{-1}(64-20)K(0.010m)^3}{6.25\times10^{-7}\,m^2/s \times 1.531\times10^{-7}\,m^2/s} = 1.804\times10^6. \tag{2}$$

Using the Churchill-Chu relation, find

$$\overline{Nu}_D = \frac{\bar{h}D}{k} = \left\{0.60 + \frac{0.387\,Ra_D^{1/6}}{\left[1+(0.559/Pr)^{9/16}\right]^{8/27}}\right\}^2 \tag{3}$$

$$\bar{h} = \frac{0.634\,W/m\cdot K}{0.01m}\left\{0.60 + \frac{0.387\left(1.804\times10^6\right)^{1/6}}{\left[1+(0.559/4.16)^{9/16}\right]^{8/27}}\right\}^2 = 1301\,W/m^2\cdot K.$$

Substituting numerical values into Eq. (1), the calculated value for T_s in *water* is

$$T_s = 20°C + 550\,W/\pi \times 0.010m \times 0.30m \times 1301\,W/m^2\cdot K = 64.8°C.$$ <

Continued …

Our initial assumption of $T_s = 64°C$ is in excellent agreement with the calculated value.

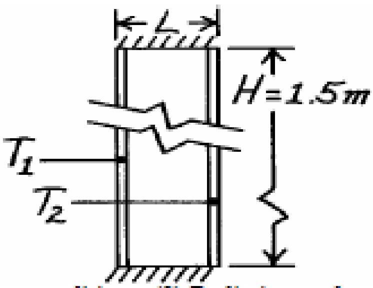
With accidental operation in *air*, the heat transfer coefficient will be nearly a factor of 100 less. Suppose $\overline{h} \approx 25\ W/m^2 \cdot K$, then from Eq. (1), $T_s \approx 2360°C$. Very likely the heater will burn out. Using air properties at $T_f \approx 1500K$ and Eq. (2), find $Ra_D = 1.815 \times 10^2$. Using Eq. 9.33, $Nu_D = C\,Ra_D^n$ with C= 0.85 and n = 0.188 from Table 9.1, find $\overline{h} = 22.6\ W/m^2 \cdot K$. Hence, our first estimate for the surface temperature in *air* was reasonable,

$$T_s \approx 2300°C.$$ <

However, radiation exchange will be the dominant mode, and would reduce the estimate for T_s. Generally such heaters could not withstand operating temperatures above 1000°C and safe operation in air is not possible.

9.94 The space between the panes of a double-glazed window can be filled with either air or carbon dioxide at atmospheric pressure. The window is 1.5 m high and the spacing between the panes can be varied. Develop an analysis to predict the convection heat transfer rate across the window as a function of pane spacing and determine, under otherwise identical conditions, whether air or carbon dioxide will yield the smaller rate. Illustrate the results of your analysis for two surface-temperature conditions: winter (−10°C, 20°C) and summer (35°C, 25°C).

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Radiation exchange is negligible, (3) Gases are at atmospheric pressure, (4) Perfect gas behavior.

PROPERTIES: *Table A-4*: Winter, $\overline{T} = (-10+20)°C/2 = 288\,K$, Summer, $\overline{T} = (35+25)°C/2 = 303\,K$:

Gas (1 atm)	T (K)	α ($m^2/s \times 10^6$)	ν ($m^2/s \times 10^6$)	$k \times 10^3$ (W/m·K)
Air	288	20.5	14.82	24.9
Air	303	22.9	16.19	26.5
CO_2	288	10.2	7.78	15.74
CO_2	303	11.2	8.55	16.78

ANALYSIS: The heat flux by convection across the window is

$$q'' = h(T_1 - T_2)$$

where the convection coefficient is estimated from the correlation of Eq. 9.53 for large aspect ratios $10 < H/L < 40$, for which $\overline{h}$ is independent of L,

$$\overline{Nu}_L = \overline{h}L/k = 0.046 Ra_L^{1/3}.$$

Substituting numerical values for winter (w) and summer (s) conditions,

$$Ra_{L,w,air} = \frac{9.8\,m/s^2\,(1/288\,K)(20-(-10))\,KL^3}{20.5\times10^{-6}\,m^2/s \times 14.82\times10^{-6}\,m^2/s} = 3.360\times10^9 L^3$$

$$Ra_{L,s,air} = 8.724\times10^8 L^3 \qquad Ra_{L,w,CO_2} = 1.286\times10^{10} L^3 \qquad Ra_{L,s,CO_2} = 3.378\times10^9 L^3$$

the heat transfer coefficients are

$$\overline{h}_{w,air} = (0.0249\,W/m\cdot K/L)\times 0.046\left(3.360\times10^9 L^3\right)^{1/3} = 1.72\,W/m^2\cdot K$$

$$h_{s,air} = 1.16\,W/m^2\cdot K \quad h_{w,CO_2} = 1.70\,W/m^2\cdot K \qquad h_{s,CO_2} = 1.16\,W/m^2\cdot K.$$

Thus,

$$q''_{w,air} = 51.5\,W/m^2 \quad q''_{s,air} = 11.6\,W/m^2 \quad q''_{w,CO_2} = 50.9\,W/m^2 \quad q''_{s,CO_2} = 11.6\,W/m^2.$$

COMMENTS: (1) The correlation is valid for $10^6 < Ra_L < 10^9$. As an example, for a spacing L = 10 mm, the Rayleigh number would be less than 10^6 in all four cases, and Eq. 9.52 should be used instead. However, note that H/L = 150, which is out of the range of validity of both correlations. (2) For this particular case, the smaller k for CO_2 is almost exactly offset by the smaller α and ν which lead to larger Ra_L, and there is very little difference between the results for air and CO_2.

19–68 Inside a condenser, there is a bank of seven copper tubes with cooling water flowing in them. Steam condenses at a rate of 0.6 kg/s on the outer surfaces of the tubes, which are at a constant temperature of 68°C. Each copper tube is 5 m long and has an inner diameter of 25 mm. Cooling water enters each tube at 5°C and exits at 60°C. Determine the average heat transfer coefficient of the cooling water flowing inside each tube and the cooling water mean velocity needed to achieve the indicated heat transfer rate in the condenser.

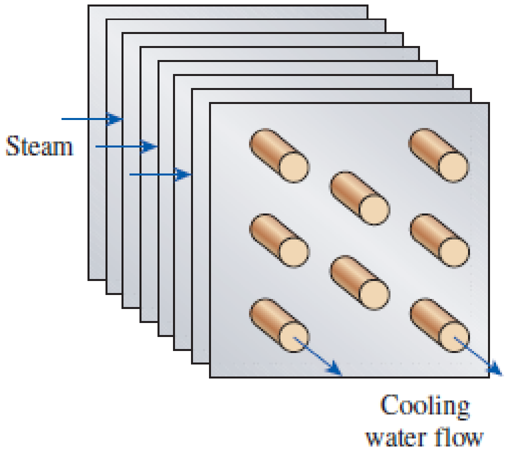


FIGURE P19–68

19–121 Hot water at 90°C enters a 15-m section of a cast iron pipe ($k = 52$ W/m·K) whose inner and outer diameters are 4 and 4.6 cm, respectively, at an average velocity of 1.2 m/s. The outer surface of the pipe, whose emissivity is 0.7, is exposed to the cold air at 10°C in a basement, with a convection heat transfer coefficient of 12 W/m^2·K. Taking the walls of the basement to be at 10°C also, determine (*a*) the rate of heat loss from the water and (*b*) the temperature at which the water leaves the basement.

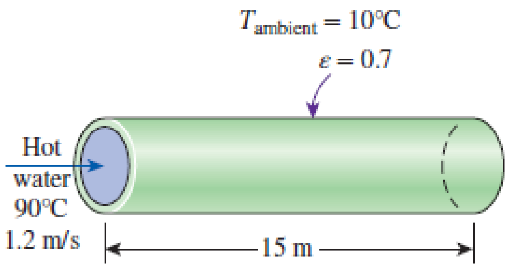


FIGURE P19–121

20–29 Consider a 15-cm × 20-cm printed circuit board (PCB) that has electronic components on one side. The board is placed in a room at 20°C. The heat loss from the back surface of the board is negligible. If the circuit board is dissipating 8 W of power in steady operation, determine the average temperature of the hot surface of the board, assuming the board is (*a*) vertical, (*b*) horizontal with hot surface facing up, and (*c*) horizontal with hot surface facing down. Take the emissivity of the surface of the board to be 0.8 and assume the surrounding surfaces to be at the same temperature as the air in the room. Evaluate air properties at a film temperature of 32.5°C and 1 atm pressure. Is this a good assumption? *Answers:* (*a*) 46.6°C, (*b*) 42.6°C, (*c*) 50.7°C

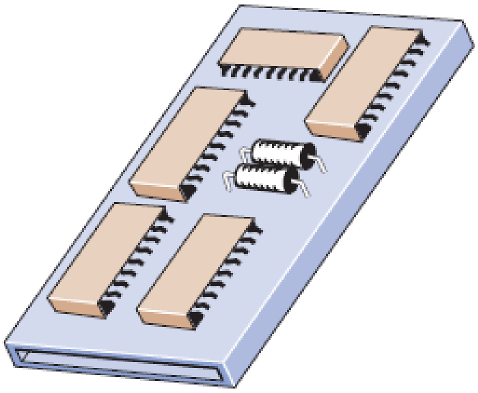
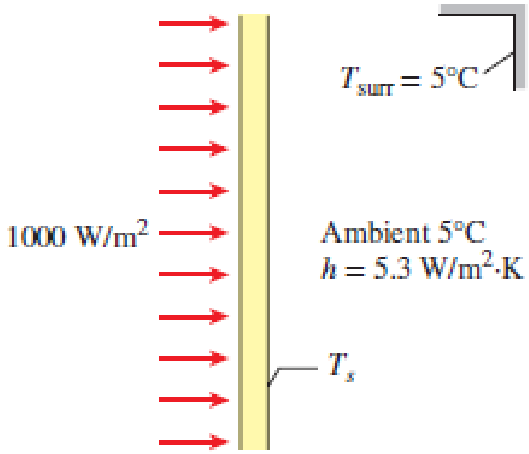


FIGURE P20–29

20–69 A vertical 1.5-m-high, 2.8-m-wide double-pane window consists of two layers of glass separated by a 2.0-cm air gap at atmospheric pressure. The room temperature is 26°C while the inner glass temperature is 18°C. Disregarding radiation heat transfer, determine the temperature of the outer glass layer and the rate of heat loss through the window by natural convection.

21–23 A thin vertical copper plate is subjected to a uniform heat flux of 1000 W/m^2 on one side, while the other side is exposed to ambient surroundings at 5°C. The surface of the plate is oxidized black and can be treated as a blackbody. The heat transfer coefficient due to natural convection on the plate surface is 5.3 W/m^2·K. Determine the surface temperature of the plate. Discuss the contribution of the net radiation heat transfer on the total heat loss from the plate.



21–26 The temperature of the filament of an incandescent lightbulb is 2500 K. Assuming the filament to be a blackbody, determine the fraction of the radiant energy emitted by the filament that falls in the visible range. Also, determine the wavelength at which the emission of radiation from the filament peaks.

21–31 A 3-mm-thick glass window transmits 90 percent of the radiation between $\lambda = 0.3$ and $3.0\ \mu m$ and is essentially opaque for radiation at other wavelengths. Determine the rate of radiation transmitted through a 3-m × 3-m glass window from blackbody sources at (*a*) 5800 K and (*b*) 1000 K.

21–38 The spectral emissivity function of an opaque surface at 1000 K is approximated as

$$\varepsilon_\lambda = \begin{cases} \varepsilon_1 = 0.4, & 0 \le \lambda < 3\ \mu\text{m} \\ \varepsilon_2 = 0.7, & 3\ \mu\text{m} \le \lambda < 6\ \mu\text{m} \\ \varepsilon_3 = 0.3, & 6\ \mu\text{m} \le \lambda < \infty \end{cases}$$

Determine the average emissivity of the surface and the rate of radiation emission from the surface, in W/m^2.

21–44 An opaque horizontal plate is well insulated on the edges and the lower surface. The irradiation on the plate is 3000 W/m^2, of which 500 W/m^2 is reflected. The plate has a uniform temperature of 700 K and has an emissive power of 5000 W/m^2. Determine the total emissivity and absorptivity of the plate.

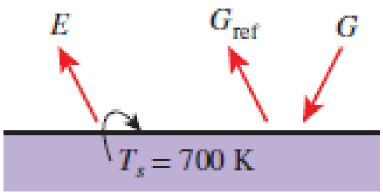


FIGURE P21–44

21–45 Irradiation on a semitransparent medium is at a rate of 520 W/m^2. If 160 W/m^2 of the irradiation is reflected from the medium and 130 W/m^2 is transmitted through the medium, determine the medium's absorptivity, reflectivity, transmissivity, and emissivity.