Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

5th Edition

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Chapter 1

INTRODUCTION AND BASIC CONCEPTS

Thermodynamics and Heat Transfer

1-1C Thermodynamics deals with the amount of heat transfer as a system undergoes a process from one equilibrium state to another. Heat transfer, on the other hand, deals with the rate of heat transfer as well as the temperature distribution within the system at a specified time.

1-2C (a) The driving force for heat transfer is the temperature difference. (b) The driving force for electric current flow is the electric potential difference (voltage). (a) The driving force for fluid flow is the pressure difference.

1-3C The caloric theory is based on the assumption that heat is a fluid-like substance called the "caloric" which is a massless, colorless, odorless substance. It was abandoned in the middle of the nineteenth century after it was shown that there is no such thing as the caloric.

1-4C The *rating* problems deal with the determination of the *heat transfer rate* for an existing system at a specified temperature difference. The *sizing* problems deal with the determination of the *size* of a system in order to transfer heat at a *specified rate* for a *specified temperature difference*.

1-5C The experimental approach (testing and taking measurements) has the advantage of dealing with the actual physical system, and getting a physical value within the limits of experimental error. However, this approach is expensive, time consuming, and often impractical. The analytical approach (analysis or calculations) has the advantage that it is fast and inexpensive, but the results obtained are subject to the accuracy of the assumptions and idealizations made in the analysis.

1-6C The description of most scientific problems involves equations that relate the *changes* in some key variables to each other, and the smaller the increment chosen in the changing variables, the more accurate the description. In **the limiting case of infinitesimal changes in variables**, we obtain *differential equations*, which provide precise mathematical formulations for the physical principles and laws by representing the rates of changes as *derivatives*.

As we shall see in later chapters, the differential equations of fluid mechanics are known, but very difficult to solve except for very simple geometries. Computers are extremely helpful in this area.

1-7C Modeling makes it possible to predict the course of an event before it actually occurs, or to study various aspects of an event mathematically without actually running expensive and time-consuming experiments. When preparing a mathematical model, all the variables that affect the phenomena are identified, reasonable assumptions and approximations are made, and the interdependence of these variables are studied. The relevant physical laws and principles are invoked, and the problem is formulated mathematically. Finally, the problem is solved using an appropriate approach, and the results are interpreted.

1-8C The right choice between a crude and complex model is usually the *simplest* model which yields *adequate* results.

Preparing very accurate but complex models is not necessarily a better choice since such models are not much use to an analyst if they are very difficult and time consuming to solve. At the minimum, the model should reflect the essential features of the physical problem it represents.

1-9C Warmer. Because energy is added to the room air in the form of electrical work.

1-10C Warmer. If we take the room that contains the refrigerator as our system, we will see that electrical work is supplied to this room to run the refrigerator, which is eventually dissipated to the room as waste heat.

1-11C For the constant pressure case. This is because the heat transfer to an ideal gas is $mc_p\Delta T$ at constant pressure and $mc_v\Delta T$ at constant volume, and c_p is always greater than c_v.

1-12C Thermal energy is the sensible and latent forms of internal energy, and it is referred to as heat in daily life.

1-13C The rate of heat transfer per unit surface area is called heat flux $\dot{q}$. It is related to the rate of heat transfer by $\dot{Q} = \int_A \dot{q} dA$.

1-14C Energy can be transferred by heat, work, and mass. An energy transfer is heat transfer when its driving force is temperature difference.

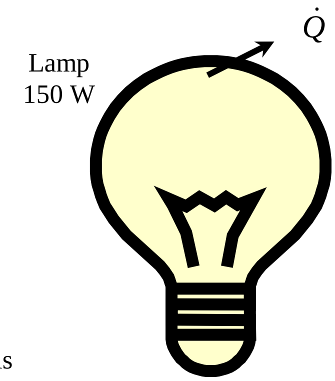
1-15 The filament of a 150 W incandescent lamp is 5 cm long and has a diameter of 0.5 mm. The heat flux on the surface of the filament, the heat flux on the surface of the glass bulb, and the annual electricity cost of the bulb are to be determined.

Assumptions Heat transfer from the surface of the filament and the bulb of the lamp is uniform.

Analysis (*a*) The heat transfer surface area and the heat flux on the surface of the filament are

$$A_s = \pi DL = \pi(0.05\ \text{cm})(5\ \text{cm}) = 0.785\ \text{cm}^2$$

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{150\ \text{W}}{0.785\ \text{cm}^2} = 191\ \text{W/cm}^2 = \mathbf{1.91 \times 10^6\ W/m^2}$$



(*b*) The heat flux on the surface of glass bulb is

$$A_s = \pi D^2 = \pi(8\ \text{cm})^2 = 201.1\ \text{cm}^2$$

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{150\ \text{W}}{201.1\ \text{cm}^2} = 0.75\ \text{W/cm}^2 = \mathbf{7500\ W/m^2}$$

(*c*) The amount and cost of electrical energy consumed during a one-year period is

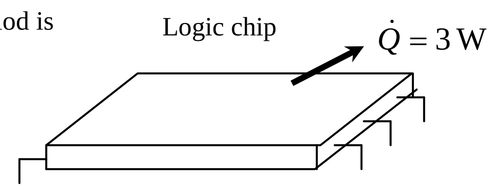
$$\text{Electricity Consumption} = \dot{Q}\Delta t = (0.15\ \text{kW})(365 \times 8\ \text{h/yr}) = 438\ \text{kWh/yr}$$
$$\text{Annual Cost} = (438\ \text{kWh/yr})(\$0.08/\text{kWh}) = \mathbf{\$35.04/yr}$$

1-16E A logic chip in a computer dissipates 3 W of power. The amount heat dissipated in 8 h and the heat flux on the surface of the chip are to be determined.

Assumptions Heat transfer from the surface is uniform.

Analysis (*a*) The amount of heat the chip dissipates during an 8-hour period is

$$Q = \dot{Q}\Delta t = (3\ \text{W})(8\ \text{h}) = 24\ \text{Wh} = \mathbf{0.024\ kWh}$$



(*b*) The heat flux on the surface of the chip is

$$\dot{q} = \frac{\dot{Q}}{A} = \frac{3\ \text{W}}{0.08\ \text{in}^2} = \mathbf{37.5\ W/in^2}$$

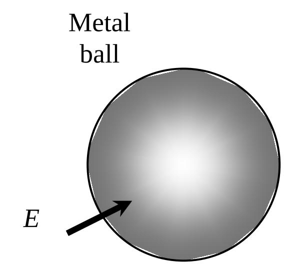
1-17 An aluminum ball is to be heated from 80°C to 200°C. The amount of heat that needs to be transferred to the aluminum ball is to be determined.

Assumptions The properties of the aluminum ball are constant.

Properties The average density and specific heat of aluminum are given to be ρ = 2700 kg/m^3 and c_p = 0.90 kJ/kg·°C.

Analysis The amount of energy added to the ball is simply the change in its internal energy, and is determined from

$$E_{\text{transfer}} = \Delta U = mc_p(T_2 - T_1)$$



where

$$m = \rho \boldsymbol{V} = \frac{\pi}{6}\rho D^3 = \frac{\pi}{6}(2700\ \text{kg/m}^3)(0.15\ \text{m})^3 = 4.77\ \text{kg}$$

Substituting,

$$E_{\text{transfer}} = (4.77\ \text{kg})(0.90\ \text{kJ/kg}\cdot°\text{C})(200 - 80)°\text{C} = \mathbf{515\ kJ}$$

Therefore, 515 kJ of energy (heat or work such as electrical energy) needs to be transferred to the aluminum ball to heat it to 200°C.

1-18 One metric ton of liquid ammonia in a rigid tank is exposed to the sun. The initial temperature is 4°C and the exposure to sun increased the temperature by 2°C. Heat energy added to the liquid ammonia is to be determined.

Assumptions The specific heat of the liquid ammonia is constant.

Properties The average specific heat of liquid ammonia at (4 + 6)°C / 2 = 5°C is c_p = 4645 J/kg·K (Table A-11).

Analysis The amount of energy added to the ball is simply the change in its internal energy, and is determined from

$$Q = mc_p(T_2 - T_1)$$

where

$$\text{m} = 1\ \text{metric ton} = 1000\ \text{kg}$$

Substituting,

$$Q = (1000\ \text{kg})(4645\ \text{J/kg}\cdot°\text{C})(2°\text{C}) = \mathbf{9290\ kJ}$$

Discussion Therefore, 9290 kJ of heat energy is required to transfer to 1 metric ton of liquid ammonia to heat it by 2°C. Also, the specific heat units J/kg·ºC and J/kg·K are equivalent, and can be interchanged.

1-19 A 2 mm thick by 3 cm wide AISI 1010 carbon steel strip is cooled in a chamber from 527 to 127°C. The heat rate removed from the steel strip is 100 kW and the speed it is being conveyed in the chamber is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The stainless steel sheet has constant properties. **3** Changes in potential and kinetic energy are negligible.

Properties For AISI 1010 steel, the specific heat of AISI 1010 steel at (527 + 127)°C / 2 = 327°C = 600 K is 685 J/kg·K (Table A-3), and the density is given as 7832 kg/m^3.

Analysis The mass of the steel strip being conveyed enters and exits the chamber at a rate of

$$\dot{m} = \rho V w t$$

The rate of heat loss from the steel strip in the chamber is given as

$$\dot{Q}_{\text{loss}} = \dot{m} c_p (T_{\text{in}} - T_{\text{out}}) = \rho V w t c_p (T_{\text{in}} - T_{\text{out}})$$

Thus, the velocity of the steel strip being conveyed is

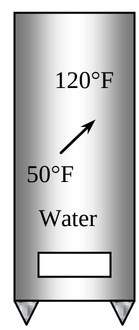
$$V = \frac{\dot{Q}_{\text{loss}}}{\rho w t c_p (T_{\text{in}} - T_{\text{out}})} = \frac{100 \times 10^3 \text{ W}}{(7832 \text{ kg/m}^3)(0.030 \text{ m})(0.002 \text{ m})(685 \text{ J/kg}\cdot\text{K})(527 - 127)\text{K}} = \mathbf{0.777 \text{ m/s}}$$

Discussion A control volume is applied on the steel strip being conveyed in and out of the chamber.

1-20E A water heater is initially filled with water at 50°F. The amount of energy that needs to be transferred to the water to raise its temperature to 120°F is to be determined.

Assumptions **1** Water is an incompressible substance with constant specific. **2** No water flows in or out of the tank during heating.

Properties The density and specific heat of water at 85ºF from Table A-9E are: ρ = 62.17 lbm/ft^3 and c_p = 0.999 Btu/lbm·R.



Analysis The mass of water in the tank is

$$m = \rho \boldsymbol{V} = (62.17 \text{ lbm/ft}^3)(60 \text{ gal})\left(\frac{1 \text{ ft}^3}{7.48 \text{ gal}}\right) = 498.7 \text{ lbm}$$

Then, the amount of heat that must be transferred to the water in the tank as it is heated from 50 to 120°F is determined to be

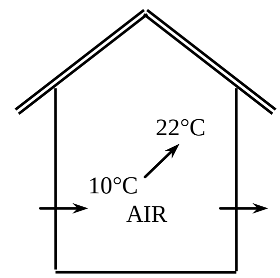
$$Q = m c_p (T_2 - T_1) = (498.7 \text{ lbm})(0.999 \text{ Btu/lbm}\cdot{}^\circ\text{F})(120 - 50)^\circ\text{F} = \mathbf{34{,}874 \text{ Btu}}$$

Discussion Referring to Table A-9E the density and specific heat of water at 50ºF are: ρ = 62.41 lbm/ft^3 and c_p = 1.000 Btu/lbm·R and at 120ºF are: ρ = 61.71 lbm/ft^3 and c_p = 0.999 Btu/lbm·R. We evaluated the water properties at an average temperature of 85ºF. However, we could have assumed constant properties and evaluated properties at the initial temperature of 50ºF or final temperature of 120ºF without loss of accuracy.

1-21 A house is heated from 10°C to 22°C by an electric heater, and some air escapes through the cracks as the heated air in the house expands at constant pressure. The amount of heat transfer to the air and its cost are to be determined.

Assumptions **1** Air as an ideal gas with a constant specific heats at room temperature. **2** The volume occupied by the furniture and other belongings is negligible. **3** The pressure in the house remains constant at all times. **4** Heat loss from the house to the outdoors is negligible during heating. **5** The air leaks out at 22°C.

Properties The specific heat of air at room temperature is c_p = 1.007 kJ/kg·°C.



Analysis The volume and mass of the air in the house are

$$V = \text{(floor space)(height)} = (200\,\text{m}^2)(3\,\text{m}) = 600\,\text{m}^3$$

$$m = \frac{PV}{RT} = \frac{(101.3\,\text{kPa})(600\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(10+273.15\,\text{K})} = 747.9\,\text{kg}$$

Noting that the pressure in the house remains constant during heating, the amount of heat that must be transferred to the air in the house as it is heated from 10 to 22°C is determined to be

$$Q = mc_p(T_2 - T_1) = (747.9\,\text{kg})(1.007\,\text{kJ/kg}\cdot{}^\circ\text{C})(22-10)^\circ\text{C} = \mathbf{9038\,kJ}$$

Noting that 1 kWh = 3600 kJ, the cost of this electrical energy at a unit cost of \$0.075/kWh is

$$\text{Enegy Cost} = \text{(Energy used)(Unit cost of energy)} = (9038\,/\,3600\,\text{kWh})(\$0.075/\text{kWh}) = \mathbf{\$0.19}$$

Therefore, it will cost the homeowner about 19 cents to raise the temperature in his house from 10 to 22°C.

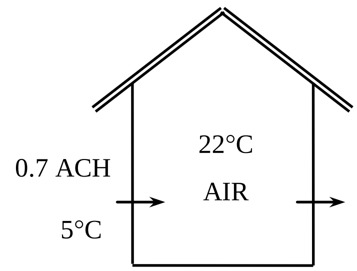
1-22 An electrically heated house maintained at 22°C experiences infiltration losses at a rate of 0.7 ACH. The amount of energy loss from the house due to infiltration per day and its cost are to be determined.

Assumptions **1** Air as an ideal gas with a constant specific heats at room temperature. **2** The volume occupied by the furniture and other belongings is negligible. **3** The house is maintained at a constant temperature and pressure at all times. **4** The infiltrating air exfiltrates at the indoors temperature of 22°C.

Properties The specific heat of air at room temperature is c_p = 1.007 kJ/kg·°C.

Analysis The volume of the air in the house is

$$V = (\text{floor space})(\text{height}) = (150\ \text{m}^2)(3\ \text{m}) = 450\ \text{m}^3$$



Noting that the infiltration rate is 0.7 ACH (air changes per hour) and thus the air in the house is completely replaced by the outdoor air 0.7×24 = 16.8 times per day, the mass flow rate of air through the house due to infiltration is

$$\begin{aligned}\dot{m}_{\text{air}} &= \frac{P_o \dot{V}_{\text{air}}}{RT_o} = \frac{P_o(\text{ACH} \times V_{\text{house}})}{RT_o} \\ &= \frac{(89.6\ \text{kPa})(16.8 \times 450\ \text{m}^3/\text{day})}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(5+273.15\ \text{K})} = 8485\ \text{kg/day}\end{aligned}$$

Noting that outdoor air enters at 5°C and leaves at 22°C, the energy loss of this house per day is

$$\begin{aligned}\dot{Q}_{\text{infilt}} &= \dot{m}_{\text{air}} c_p (T_{\text{indoors}} - T_{\text{outdoors}}) \\ &= (8485\ \text{kg/day})(1.007\ \text{kJ/kg.°C})(22-5)\text{°C} = 145{,}260\ \text{kJ/day} = \mathbf{40.4\ kWh/day}\end{aligned}$$

At a unit cost of \$0.082/kWh, the cost of this electrical energy lost by infiltration is

$$\text{Enegy Cost} = (\text{Energy used})(\text{Unit cost of energy}) = (40.4\ \text{kWh/day})(\$0.082/\text{kWh}) = \mathbf{\$3.31/day}$$

1-23 Water is heated in an insulated tube by an electric resistance heater. The mass flow rate of water through the heater is to be determined.

Assumptions **1** Water is an incompressible substance with a constant specific heat. **2** The kinetic and potential energy changes are negligible, Δke ≅ Δpe ≅ 0. **3** Heat loss from the insulated tube is negligible.

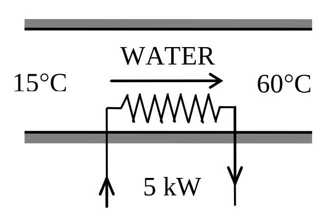
Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg·°C.

Analysis We take the tube as the system. This is a *control volume* since mass crosses the system boundary during the process. We *observe* that this is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\text{CV}} = 0$ and $\Delta E_{\text{CV}} = 0$, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$, and the tube is insulated. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{e,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} = \dot{m}c_p(T_2 - T_1)$$



Thus,

$$\dot{m} = \frac{\dot{W}_{\text{e,in}}}{c_p(T_2 - T_1)} = \frac{5\ \text{kJ/s}}{(4.18\ \text{kJ/kg}\cdot\text{°C})(60-15)\text{°C}} = \mathbf{0.0266\ \ kg/s}$$

1-24 Liquid ethanol is being transported in a pipe where heat is added to the liquid. The volume flow rate that is necessary to keep the ethanol temperature below its flashpoint is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The specific heat and density of ethanol are constant.

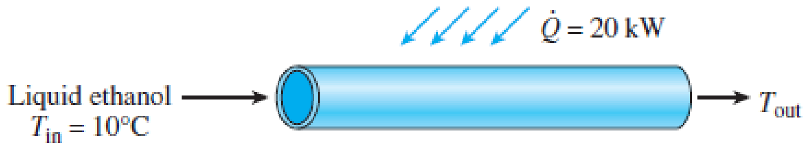
Properties The specific heat and density of ethanol are given as 2.44 kJ/kg·K and 789 kg/m^3, respectively.

Analysis The rate of heat added to the ethanol being transported in the pipe is

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

or

$$\dot{Q} = \dot{V} \rho c_p (T_{\text{out}} - T_{\text{in}})$$



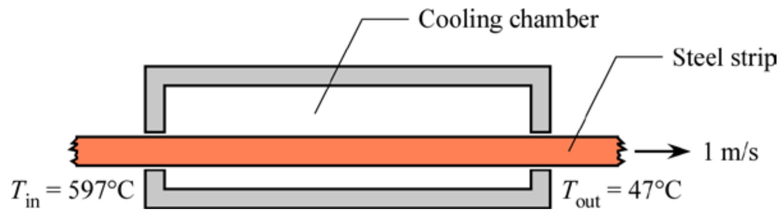
For the ethanol in the pipe to be below its flashpoint, it is necessary to keep T_{out} below 16.6°C. Thus, the volume flow rate should be

$$\dot{V} > \frac{\dot{Q}}{\rho c_p (T_{\text{out}} - T_{\text{in}})} = \frac{20 \text{ kJ/s}}{(789 \text{ kg/m}^3)(2.44 \text{ kJ/kg}\cdot\text{K})(16.6 - 10) \text{ K}}$$

$$\dot{V} > \mathbf{0.00157\ m^3/s}$$

Discussion To maintain the ethanol in the pipe well below its flashpoint, it is more desirable to have a much higher flow rate than 0.00157 m^3/s.

1-25 A 2 mm thick by 3 cm wide AISI 1010 carbon steel strip is cooled in a chamber from 597 to 47°C to avoid instantaneous thermal burn upon contact with skin tissue. The amount of heat rate to be removed from the steel strip is to be determined.



Assumptions **1** Steady operating conditions exist. **2** The stainless steel sheet has constant specific heat and density. **3** Changes in potential and kinetic energy are negligible.

Properties For AISI 1010 carbon steel, the specific heat of AISI 1010 steel at (597 + 47)°C / 2 = 322°C = 595 K is 682 J/kg·K (by interpolation from Table A-3), and the density is given as 7832 kg/m^3.

Analysis The mass of the steel strip being conveyed enters and exits the chamber at a rate of

$$\dot{m} = \rho V w t$$

The rate of heat being removed from the steel strip in the chamber is given as

$$\begin{aligned}\dot{Q}_{\text{removed}} &= \dot{m} c_p (T_{\text{in}} - T_{\text{out}}) \\ &= \rho V w t c_p (T_{\text{in}} - T_{\text{out}}) \\ &= (7832\ \text{kg/m}^3)(1\ \text{m/s})(0.030\ \text{m})(0.002\ \text{m})(682\ \text{J/kg}\cdot\text{K})(597 - 47)\ \text{K} \\ &= \mathbf{176\ kW}\end{aligned}$$

Discussion By slowing down the conveyance speed of the steel strip would reduce the amount of heat rate needed to be removed from the steel strip in the cooling chamber. Since slowing the conveyance speed allows more time for the steel strip to cool.

1-26 Liquid water is to be heated in an electric teapot. The heating time is to be determined.

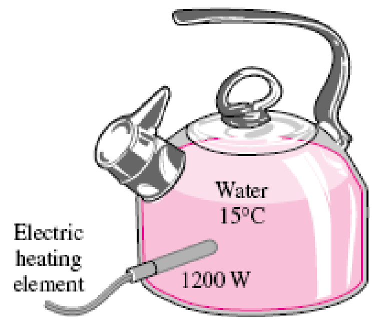
Assumptions **1** Heat loss from the teapot is negligible. **2** Constant properties can be used for both the teapot and the water.

Properties The average specific heats are given to be 0.7 kJ/kg·K for the teapot and 4.18 kJ/kg·K for water.

Analysis We take the teapot and the water in it as the system, which is a closed system (fixed mass). The energy balance in this case can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$E_{\text{in}} = \Delta U_{\text{system}} = \Delta U_{\text{water}} + \Delta U_{\text{teapot}}$$



Then the amount of energy needed to raise the temperature of water and the teapot from 15°C to 95°C is

$$\begin{aligned} E_{\text{in}} &= (mc\Delta T)_{\text{water}} + (mc\Delta T)_{\text{teapot}} \\ &= (1.2\text{ kg})(4.18\text{ kJ/kg}\cdot°\text{C})(95-15)°\text{C} + (0.5\text{ kg})(0.7\text{ kJ/kg}\cdot°\text{C})(95-15)°\text{C} \\ &= 429.3\text{ kJ} \end{aligned}$$

The 1200-W electric heating unit will supply energy at a rate of 1.2 kW or 1.2 kJ per second. Therefore, the time needed for this heater to supply 429.3 kJ of heat is determined from

$$\Delta t = \frac{\text{Total energy transferred}}{\text{Rate of energy transfer}} = \frac{E_{\text{in}}}{\dot{E}_{\text{transfer}}} = \frac{429.3\text{ kJ}}{1.2\text{ kJ/s}} = 358\text{ s} = \mathbf{6.0\ min}$$

Discussion In reality, it will take more than 6 minutes to accomplish this heating process since some heat loss is inevitable during heating. Also, the specific heat units kJ/kg · °C and kJ/kg · K are equivalent, and can be interchanged.

1-27 It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 9000 kJ/h. The power rating of the heater is to be determined.

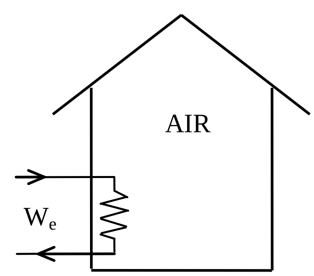
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The temperature of the room remains constant during this process.

Analysis We take the room as the system. The energy balance in this case reduces to

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,in} - Q_{out} = \Delta U = 0$$

$$W_{e,in} = Q_{out}$$



since $\Delta U = mc_v\Delta T = 0$ for isothermal processes of ideal gases. Thus,

$$\dot{W}_{e,in} = \dot{Q}_{out} = 9000\text{ kJ/h}\left(\frac{1\text{ kW}}{3600\text{ kJ/h}}\right) = \mathbf{2.5\ kW}$$

1-28 The resistance heating element of an electrically heated house is placed in a duct. The air is moved by a fan, and heat is lost through the walls of the duct. The power rating of the electric resistance heater is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

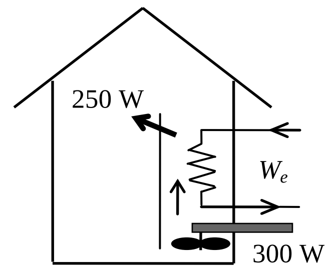
Properties The specific heat of air at room temperature is c_p = 1.007 kJ/kg·°C (Table A-15).

Analysis We take the heating duct as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that this is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. Also, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} - \dot{W}_{\text{fan,in}} + \dot{m}c_p(T_2 - T_1)$$



Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{\text{e,in}} = (0.25\,\text{kW}) - (0.3\,\text{kW}) + (0.6\text{ kg/s})(1.007\text{ kJ/kg}\cdot°\text{C})(5°\text{C})$$
$$= \mathbf{2.97\ kW}$$

1-29 A room is heated by an electrical resistance heater placed in a short duct in the room in 15 min while the room is losing heat to the outside, and a 300-W fan circulates the air steadily through the heater duct. The power rating of the electric heater and the temperature rise of air in the duct are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, Δke ≅ Δpe ≅ 0. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **3** Heat loss from the duct is negligible. **4** The house is air-tight and thus no air is leaking in or out of the room.

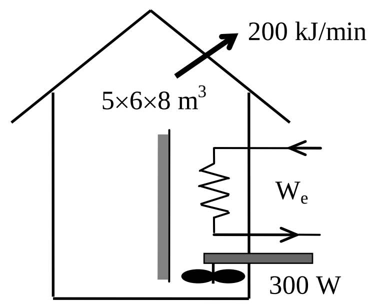
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.007 kJ/kg·K for air at room temperature (Table A-15) and $c_v = c_p - R$ = 0.720 kJ/kg·K.

Analysis (a) We first take the air in the room as the system. This is a constant volume *closed system* since no mass crosses the system boundary. The energy balance for the room can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} + W_{\text{fan,in}} - Q_{\text{out}} = \Delta U$$

$$(\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}})\Delta t = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$



The total mass of air in the room is

$$V = 5 \times 6 \times 8 \text{ m}^3 = 240 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(98 \text{ kPa})(240 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 284.6 \text{ kg}$$

Then the power rating of the electric heater is determined to be

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} - \dot{W}_{\text{fan,in}} + mc_v(T_2 - T_1)/\Delta t$$
$$= (200/60 \text{ kJ/s}) - (0.3 \text{ kJ/s}) + (284.6 \text{ kg})(0.720 \text{ kJ/kg} \cdot {}^\circ\text{C})(25 - 15^\circ\text{C})/(18 \times 60 \text{ s}) = \mathbf{4.93 \text{ kW}}$$

(b) The temperature rise that the air experiences each time it passes through the heater is determined by applying the energy balance to the duct,

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} + \dot{m}h_1 = \dot{Q}_{\text{out}}^{\nearrow 0} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} = \dot{m}\Delta h = \dot{m}c_p \Delta T$$

Thus,

$$\Delta T = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{\dot{m}c_p} = \frac{(4.93 + 0.3)\text{kJ/s}}{(50/60 \text{ kg/s})(1.007 \text{ kJ/kg} \cdot \text{K})} = \mathbf{6.2^\circ C}$$

1-30 The ducts of an air heating system pass through an unheated area, resulting in a temperature drop of the air in the duct. The rate of heat loss from the air to the cold environment is to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

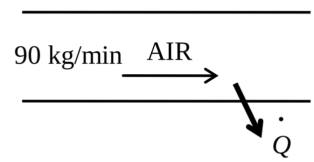
Properties The specific heat of air at room temperature is c_p = 1.007 kJ/kg·°C (Table A-15).

Analysis We take the heating duct as the system. This is a *control volume* since mass crosses the system boundary during the process. We *observe* that this is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\text{CV}} = 0$ and $\Delta E_{\text{CV}} = 0$. Also, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Substituting,

$$\dot{Q}_{\text{out}} = \dot{m}c_p \Delta T = (90 \text{ kg/min})(1.007 \text{ kJ/kg} \cdot °\text{C})(3°\text{C}) = \mathbf{272 \text{ kJ/min}}$$

1-31 Air is moved through the resistance heaters in a 900-W hair dryer by a fan. The volume flow rate of air at the inlet and the velocity of the air at the exit are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, Δke ≅ Δpe ≅ 0. **3** Constant specific heats at room temperature can be used for air. **4** The power consumed by the fan and the heat losses through the walls of the hair dryer are negligible.

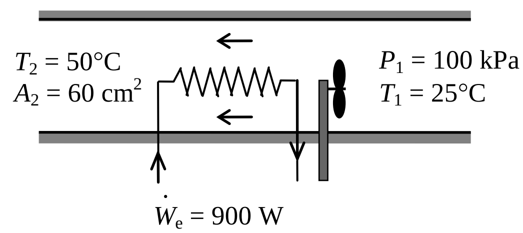
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.007 kJ/kg·K for air at room temperature (Table A-15).

Analysis (a) We take the hair dryer as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that this is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$, and there is only one inlet and one exit and thus $\dot m_1 = \dot m_2 = \dot m$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{in} - \dot E_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\rm system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot E_{in} = \dot E_{out}$$

$$\dot W_{\rm e,in} + \dot W_{\rm fan,in}^{\nearrow 0} + \dot m h_1 = \dot Q_{out}^{\nearrow 0} + \dot m h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot W_{\rm e,in} = \dot m c_p (T_2 - T_1)$$



Thus,

$$\dot m = \frac{\dot W_{\rm e,in}}{c_p(T_2 - T_1)} = \frac{0.9 \text{ kJ/s}}{(1.007 \text{ kJ/kg}\cdot{}^\circ\text{C})(50-25)^\circ\text{C}} = 0.03575 \text{ kg/s}$$

Then,

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})}{100 \text{ kPa}} = 0.8553 \text{ m}^3/\text{kg}$$

$$\dot{\boldsymbol{V}}_1 = \dot m \boldsymbol{v}_1 = (0.03575 \text{ kg/s})(0.8553 \text{ m}^3/\text{kg}) = \mathbf{0.0306\ \ m^3/s}$$

(b) The exit velocity of air is determined from the conservation of mass equation,

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323 \text{ K})}{100 \text{ kPa}} = 0.9270 \text{ m}^3/\text{kg}$$

$$\dot m = \frac{1}{\boldsymbol{v}_2} A_2 V_2 \longrightarrow V_2 = \frac{\dot m \boldsymbol{v}_2}{A_2} = \frac{(0.03575 \text{ kg/s})(0.9270 \text{ m}^3/\text{kg})}{60\times 10^{-4} \text{ m}^2} = \mathbf{5.52\ m/s}$$

1-32E Air gains heat as it flows through the duct of an air-conditioning system. The velocity of the air at the duct inlet and the temperature of the air at the exit are to be determined.

Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -222°F and 548 psia. **2** The kinetic and potential energy changes are negligible, Δke ≅ Δpe ≅ 0. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

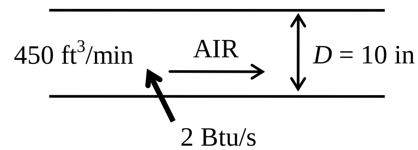
Properties The gas constant of air is R = 0.3704 psia·ft^3/lbm·R (Table A-1E). Also, c_p = 0.240 Btu/lbm·R for air at room temperature (Table A-15E).

Analysis We take the air-conditioning duct as the system. This is a *control volume* since mass crosses the system boundary during the process. We *observe* that this is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$, there is only one inlet and one exit and thus $\dot m_1 = \dot m_2 = \dot m$, and heat is lost from the system. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{in} - \dot E_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\rm system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot E_{in} = \dot E_{out}$$

$$\dot Q_{\rm in} + \dot m h_1 = \dot m h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot Q_{\rm in} = \dot m c_p (T_2 - T_1)$$



(*a*) The inlet velocity of air through the duct is determined from

$$V_1 = \frac{\dot{\boldsymbol{V}}_1}{A_1} = \frac{\dot{\boldsymbol{V}}_1}{\pi r^2} = \frac{450 \text{ ft}^3/\text{min}}{\pi (5/12 \text{ ft})^2} = \mathbf{825\ ft/min}$$

(*b*) The mass flow rate of air becomes

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510 \text{ R})}{15 \text{ psia}} = 12.6 \text{ ft}^3/\text{lbm}$$

$$\dot m = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{450 \text{ ft}^3/\text{min}}{12.6 \text{ ft}^3/\text{lbm}} = 35.7 \text{ lbm/min} = 0.595 \text{ lbm/s}$$

Then the exit temperature of air is determined to be

$$T_2 = T_1 + \frac{\dot Q_{\rm in}}{\dot m c_p} = 50°\text{F} + \frac{2 \text{ Btu/s}}{(0.595 \text{ lbm/s})(0.240 \text{ Btu/lbm}\cdot°\text{F})} = \mathbf{64.0°F}$$

Heat Transfer Mechanisms

1-33C The thermal conductivity of a material is the rate of heat transfer through a unit thickness of the material per unit area and per unit temperature difference. The thermal conductivity of a material is a measure of how fast heat will be conducted in that material.

1-34C No. Such a definition will imply that doubling the thickness will double the heat transfer rate. The equivalent but "more correct" unit of thermal conductivity is $W \cdot m/m^2 \cdot °C$ that indicates product of heat transfer rate and thickness per unit surface area per unit temperature difference.

1-35C Diamond is a better heat conductor.

1-36C The thermal conductivity of gases is proportional to the square root of absolute temperature. The thermal conductivity of most liquids, however, decreases with increasing temperature, with water being a notable exception.

1-37C Superinsulations are obtained by using layers of highly reflective sheets separated by glass fibers in an evacuated space. Radiation heat transfer between two surfaces is inversely proportional to the number of sheets used and thus heat loss by radiation will be very low by using this highly reflective sheets. At the same time, evacuating the space between the layers forms a vacuum under 0.000001 atm pressure which minimize conduction or convection through the air space between the layers.

1-38C Most ordinary insulations are obtained by mixing fibers, powders, or flakes of insulating materials with air. Heat transfer through such insulations is by conduction through the solid material, and conduction or convection through the air space as well as radiation. Such systems are characterized by apparent thermal conductivity instead of the ordinary thermal conductivity in order to incorporate these convection and radiation effects.

1-39C The thermal conductivity of an alloy of two metals will most likely be less than the thermal conductivities of both metals.

1-40C The mechanisms of heat transfer are conduction, convection and radiation. Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas which is in motion, and it involves combined effects of conduction and fluid motion. Radiation is energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules.

1-41C Conduction is expressed by Fourier's law of conduction as $\dot{Q}_{\text{cond}} = -kA\dfrac{dT}{dx}$ where *dT/dx* is the temperature gradient, *k* is the thermal conductivity, and *A* is the area which is normal to the direction of heat transfer.

Convection is expressed by Newton's law of cooling as $\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty)$ where h is the convection heat transfer coefficient, A_s is the surface area through which convection heat transfer takes place, T_s is the surface temperature and T_∞ is the temperature of the fluid sufficiently far from the surface.

Radiation is expressed by Stefan-Boltzman law as $\dot{Q}_{\text{rad}} = \varepsilon\sigma A_s(T_s^4 - T_{\text{surr}}^4)$ where ε is the emissivity of surface, A_s is the surface area, T_s is the surface temperature, T_{surr} is the average surrounding surface temperature and $\sigma = 5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4$ is the Stefan-Boltzman constant.

1-42C Convection involves fluid motion, conduction does not. In a solid we can have only conduction.

1-43C No. It is purely by radiation.

1-44C In forced convection the fluid is forced to move by external means such as a fan, pump, or the wind. The fluid motion in natural convection is due to buoyancy effects only.

1-45C In solids, conduction is due to the combination of the vibrations of the molecules in a lattice and the energy transport by free electrons. In gases and liquids, it is due to the collisions of the molecules during their random motion.

1-46C The parameters that effect the rate of heat conduction through a windowless wall are the geometry and surface area of wall, its thickness, the material of the wall, and the temperature difference across the wall.

1-47C In a typical house, heat loss through the wall with glass window will be larger since the glass is much thinner than a wall, and its thermal conductivity is higher than the average conductivity of a wall.

1-48C The house with the lower rate of heat transfer through the walls will be more energy efficient. Heat conduction is proportional to thermal conductivity (which is 0.72 W/m.°C for brick and 0.17 W/m.°C for wood, Table 1-1) and inversely proportional to thickness. The wood house is more energy efficient since the wood wall is twice as thick but it has about one-fourth the conductivity of brick wall.

1-49C The rate of heat transfer through both walls can be expressed as

$$\dot{Q}_{\text{wood}} = k_{\text{wood}} A \frac{T_1 - T_2}{L_{\text{wood}}} = (0.16\ \text{W/m}\cdot{}^\circ\text{C}) A \frac{T_1 - T_2}{0.1\ \text{m}} = 1.6 A (T_1 - T_2)$$

$$\dot{Q}_{\text{brick}} = k_{\text{brick}} A \frac{T_1 - T_2}{L_{\text{brick}}} = (0.72\ \text{W/m}\cdot{}^\circ\text{C}) A \frac{T_1 - T_2}{0.25\ \text{m}} = 2.88 A (T_1 - T_2)$$

where thermal conductivities are obtained from Table A-5. Therefore, heat transfer through the brick wall will be larger despite its higher thickness.

1-50C Emissivity is the ratio of the radiation emitted by a surface to the radiation emitted by a blackbody at the same temperature. Absorptivity is the fraction of radiation incident on a surface that is absorbed by the surface. The Kirchhoff's law of radiation states that the emissivity and the absorptivity of a surface are equal at the same temperature and wavelength.

1-51C A blackbody is an idealized body which emits the maximum amount of radiation at a given temperature and which absorbs all the radiation incident on it. Real bodies emit and absorb less radiation than a blackbody at the same temperature.

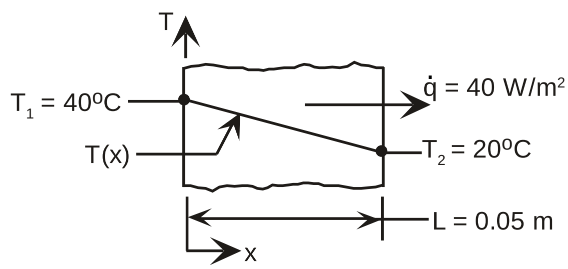
1-52 The thermal conductivity of a wood slab subjected to a given heat flux of 40 W/m^2 with constant left and right surface temperatures of 40ºC and 20ºC is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the wood slab remain constant at the specified values. **2** Heat transfer through the wood slab is one dimensional since the thickness of the slab is small relative to other dimensions. **3** Thermal conductivity of the wood slab is constant.

Analysis The thermal conductivity of the wood slab is determined directly from Fourier's relation to be

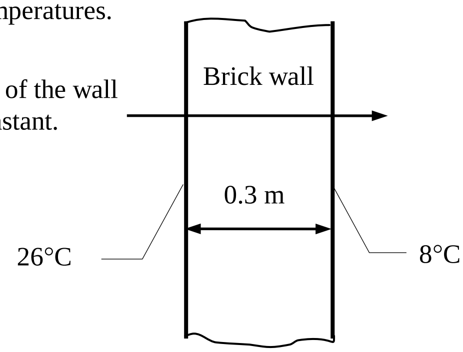
$$k = \dot{q}\frac{L}{T_1 - T_2} = \left(40\ \frac{\text{W}}{\text{m}^2}\right)\frac{0.05\ \text{m}}{(40 - 20)^\circ\text{C}} =$$

0.10 W/m·K



Discussion Note that the ºC or K temperature units may be used interchangeably when evaluating a temperature difference.

1-53 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of heat transfer through the wall is to be determined.



Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Thermal properties of the wall are constant.

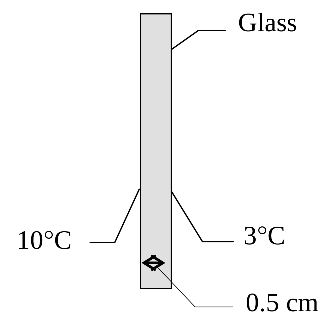
Properties The thermal conductivity of the wall is given to be k = 0.69 W/m·°C.

Analysis Under steady conditions, the rate of heat transfer through the wall is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.69 \text{ W/m}\cdot\text{°C})(4\times 7 \text{ m}^2)\frac{(26-8)\text{°C}}{0.3 \text{ m}} = \mathbf{1159 \text{ W}}$$

1-54 The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat transfer through the glass in 5 h is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Thermal properties of the glass are constant.



Properties The thermal conductivity of the glass is given to be k = 0.78 W/m·°C.

Analysis Under steady conditions, the rate of heat transfer through the glass by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.78 \text{ W/m}\cdot\text{°C})(2\times 2 \text{ m}^2)\frac{(10-3)\text{°C}}{0.005\text{m}} = 4368 \text{ W}$$

Then the amount of heat transfer over a period of 5 h becomes

$$Q = \dot{Q}_{\text{cond}}\Delta t = (4.368 \text{ kJ/s})(5\times 3600 \text{ s}) = \mathbf{78{,}620 \text{ kJ}}$$

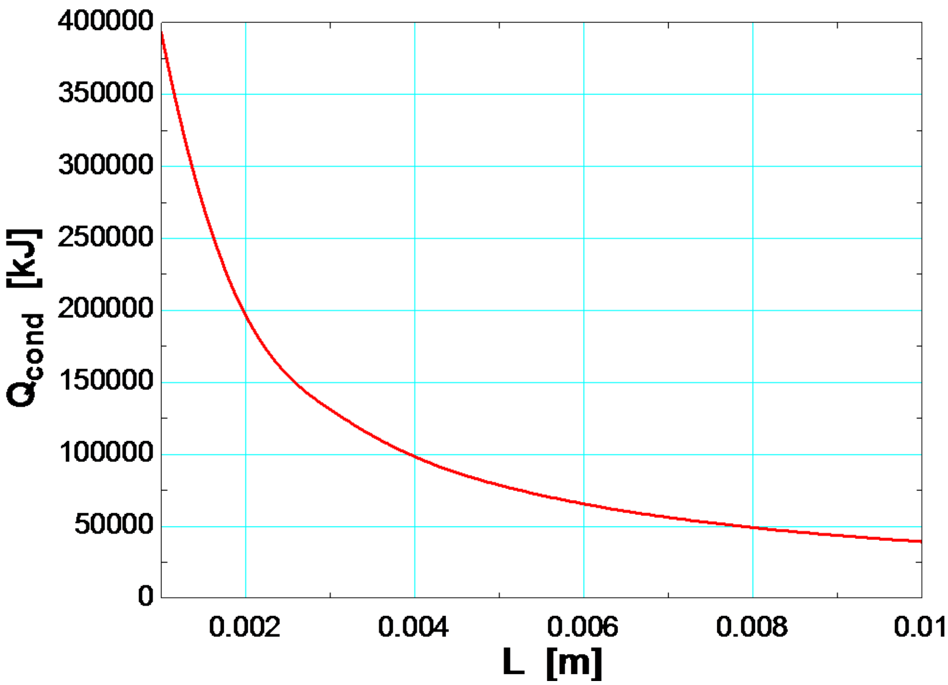
If the thickness of the glass doubled to 1 cm, then the amount of heat transfer will go down by half to **39,310 kJ**.

1-55 Prob. 1-54 is reconsidered. The amount of heat loss through the glass as a function of the window glass thickness is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=0.005 [m]
A=2*2 [m^2]
T_1=10 [C]
T_2=3 [C]
k=0.78 [W/m-C]
time=5*3600 [s]

"ANALYSIS"
Q_dot_cond=k*A*(T_1-T_2)/L
Q_cond=Q_dot_cond*time*Convert(J, kJ)
```

L [m]	Q_{cond} [kJ]
0.001	393120
0.002	196560
0.003	131040
0.004	98280
0.005	78624
0.006	65520
0.007	56160
0.008	49140
0.009	43680
0.01	39312



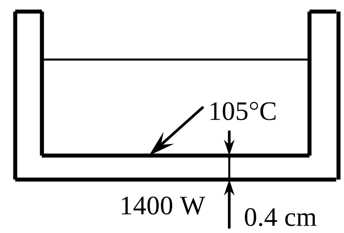
1-56 Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface temperature of the bottom of the pan is given. The temperature of the outer surface is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values. **2** Thermal properties of the aluminum pan are constant.

Properties The thermal conductivity of the aluminum is given to be k = 237 W/m·°C.

Analysis The heat transfer area is

$$A = \pi r^2 = \pi (0.075 \text{ m})^2 = 0.0177 \text{ m}^2$$



Under steady conditions, the rate of heat transfer through the bottom of the pan by conduction is

$$\dot{Q} = kA\frac{\Delta T}{L} = kA\frac{T_2 - T_1}{L}$$

Substituting,

$$1400 \text{ W} = (237 \text{ W/m}\cdot°\text{C})(0.0177 \text{ m}^2)\frac{T_2 - 105°\text{C}}{0.004 \text{ m}}$$

which gives

$$T_2 = \mathbf{106.33°C}$$

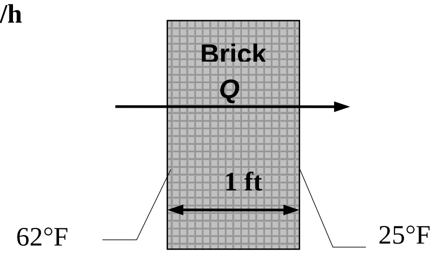
1-57E The inner and outer surface temperatures of the wall of an electrically heated home during a winter night are measured. The rate of heat loss through the wall that night and its cost are to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values during the entire night. **2** Thermal properties of the wall are constant.

Properties The thermal conductivity of the brick wall is given to be k = 0.42 Btu/h·ft·°F.

Analysis (*a*) Noting that the heat transfer through the wall is by conduction and the surface area of the wall is $A = 20 \text{ ft} \times 10 \text{ ft} = 200 \text{ ft}^2$, the steady rate of heat transfer through the wall can be determined from

$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.42 \text{ Btu/h.ft.°F})(200 \text{ ft}^2)\frac{(62 - 25)°\text{F}}{1 \text{ ft}} = \mathbf{3108 \text{ Btu/h}}$$



or 0.911 kW since 1 kW = 3412 Btu/h.

(*b*) The amount of heat lost during an 8 hour period and its cost are

$$Q = \dot{Q}\Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$$

$$\begin{aligned} \text{Cost} &= (\text{Amount of energy})(\text{Unit cost of energy}) \\ &= (7.288 \text{ kWh})(\$0.07/\text{kWh}) \\ &= \mathbf{\$0.51} \end{aligned}$$

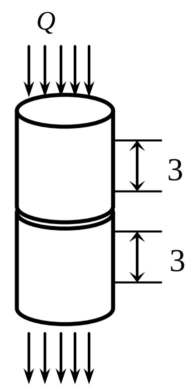
Therefore, the cost of the heat loss through the wall to the home owner that night is $0.51.

1-58 The thermal conductivity of a material is to be determined by ensuring one-dimensional heat conduction, and by measuring temperatures when steady operating conditions are reached.

Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. **2** Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples. **3** The apparatus possesses thermal symmetry.

Analysis The electrical power consumed by the heater and converted to heat is

$$\dot{W}_e = \mathbf{V}I = (110\text{ V})(0.6\text{ A}) = 66\text{ W}$$



The rate of heat flow through each sample is

$$\dot{Q} = \frac{\dot{W}_e}{2} = \frac{66\text{ W}}{2} = 33\text{ W}$$

Then the thermal conductivity of the sample becomes

$$A = \frac{\pi D^2}{4} = \frac{\pi(0.04\text{ m})^2}{4} = 0.001257\text{ m}^2$$

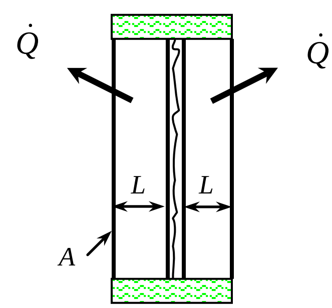
$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(33\text{ W})(0.03\text{ m})}{(0.001257\text{ m}^2)(8°\text{C})} = \mathbf{98.5\ W/m.°C}$$

1-59 The thermal conductivity of a material is to be determined by ensuring one-dimensional heat conduction, and by measuring temperatures when steady operating conditions are reached.

Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. **2** Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples. **3** The apparatus possesses thermal symmetry.

Analysis For each sample we have

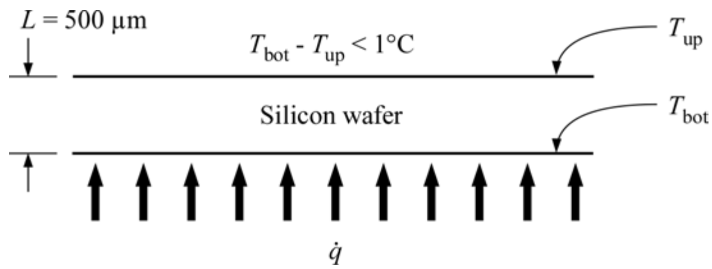
$$\dot{Q} = 25/2 = 12.5\text{ W}$$
$$A = (0.1\text{ m})(0.1\text{ m}) = 0.01\text{ m}^2$$
$$\Delta T = 82 - 74 = 8°\text{C}$$



Then the thermal conductivity of the material becomes

$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(12.5\text{ W})(0.005\text{ m})}{(0.01\text{ m}^2)(8°\text{C})} = \mathbf{0.781\ W/m.°C}$$

1-60 To prevent a silicon wafer from warping, the temperature difference across its thickness cannot exceed 1°C. The maximum allowable heat flux on the bottom surface of the wafer is to be determined.



Assumptions **1** Heat conduction is steady and one-dimensional. **2** There is no heat generation. **3** Thermal conductivity is constant.

Properties The thermal conductivity of silicon at 27°C (300 K) is 148 W/m·K (Table A-3).

Analysis For steady heat transfer, the Fourier's law of heat conduction can be expressed as

$$\dot{q} = -k\frac{dT}{dx} = -k\frac{T_{up} - T_{bot}}{L}$$

Thus, the maximum allowable heat flux so that $T_{bot} - T_{up} < 1°C$ is

$$\dot{q} \le k\frac{T_{bot} - T_{up}}{L} = (148\ \text{W/m}\cdot\text{K})\frac{1\,\text{K}}{500\times10^{-6}\ \text{m}}$$

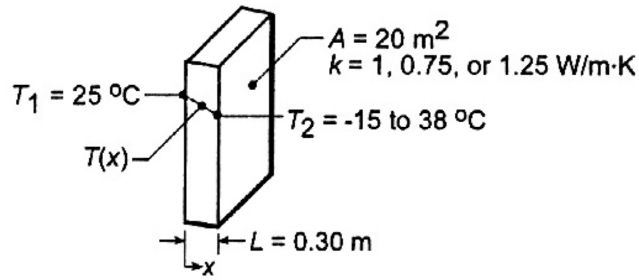
$$\dot{q} \le \mathbf{2.96\times10^5\ W/m^2}$$

Discussion With the upper surface of the wafer maintained at 27°C, if the bottom surface of the wafer is exposed to a flux greater than 2.96×10^5 W/m^2, the temperature gradient across the wafer thickness could be significant enough to cause warping.

1-61 Heat loss by conduction through a concrete wall as a function of ambient air temperatures ranging from -15 to 38°C is to be determined.

Assumptions **1** One-dimensional conduction. **2** Steady-state conditions exist. **3** Constant thermal conductivity. **4** Outside wall temperature is that of the ambient air.

Properties The thermal conductivity is given to be k = 0.75, 1 or 1.25 W/m·K.



Analysis From Fourier's law, it is evident that the gradient, $dT/dx = -\dot{q}/k$, is a constant, and hence the temperature distribution is linear, if $\dot{q}$ and k are each constant. The heat flux must be constant under one-dimensional, steady-state conditions; and k are each approximately constant if it depends only weakly on temperature. The heat flux and heat rate for the case when the outside wall temperature is $T_2 = -15°C$ and k = 1 W/m·K are:

$$\dot{q} = -k\frac{dT}{dx} = k\frac{T_1 - T_2}{L} = (1\,\mathrm{W/m\cdot K})\frac{25°C - (-15°C)}{0.30\,\mathrm{m}} = 133.3\,\mathrm{W/m^2} \quad (1)$$

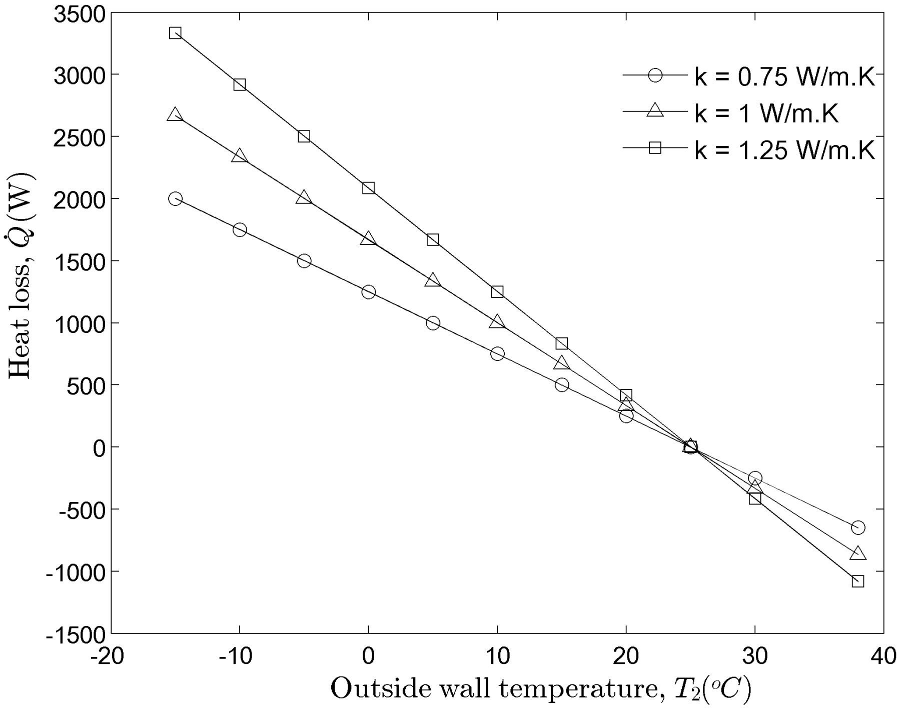
$$\dot{Q} = \dot{q}\cdot A = (133.3\,\mathrm{W/m^2})\cdot(20\,\mathrm{m^2}) = \mathbf{2667\ W} \quad (2)$$

Combining Eqs. (1) and (2), the heat rate $\dot{Q}$ can be determined for the range of ambient temperature, $-15 \le T_2 \le 38°C$, with different wall thermal conductivities, k.

Discussion (1) Notice that from the graph, the heat loss curves are linear for all three thermal conductivities. This is true because under steady-state and constant k conditions, the temperature distribution in the wall is linear. (2) As the value of k increases, the slope of the heat loss curve becomes steeper. This shows that for insulating materials (very low k), the heat loss curve would be relatively flat. The magnitude of the heat loss also increases with increasing thermal conductivity. (3) At $T_2 = 25°C$, all the three heat loss curves intersect at zero; because $T_1 = T_2$ (when the inside and outside temperatures are the same), thus there is no heat conduction through the wall. This shows that heat conduction can only occur when there is temperature difference.

The results for the heat loss $\dot{Q}$ with different thermal conductivities k are tabulated and plotted as follows:

	$\dot{Q}$ [W]		
T_2 [°C]	k = 0.75 W/m·K	k = 1 W/m·K	k = 1.25 W/m·K
-15	2000	2667	3333
-10	1750	2333	2917
-5	1500	2000	2500
0	1250	1667	2083
5	1000	1333	1667
10	750	1000	1250
15	500	666.7	833.3
20	250	333.3	416.7
25	0	0	0
30	-250	-333.3	-416.7
38	-650	-866.7	-1083



k = 0.75 W/m.K
k = 1 W/m.K
k = 1.25 W/m.K
Heat loss, $\dot{Q}$(W)
Outside wall temperature, T_2(^oC)
3500
3000
2500
2000
1500
1000
500
0
-500
-1000
-1500
-20
-10
0
10
20
30
40

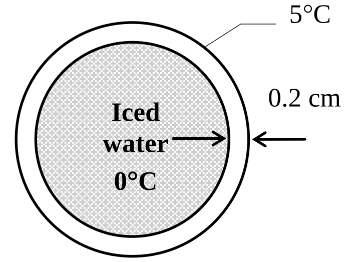
1-62 A hollow spherical iron container is filled with iced water at 0°C. The rate of heat loss from the sphere and the rate at which ice melts in the container are to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Heat transfer through the shell is one-dimensional. **3** Thermal properties of the iron shell are constant. **4** The inner surface of the shell is at the same temperature as the iced water, 0°C.

Properties The thermal conductivity of iron is k = 80.2 W/m·°C (Table A-3). The heat of fusion of water is given to be 333.7 kJ/kg.

Analysis This spherical shell can be approximated as a plate of thickness 0.4 cm and area

$$A = \pi D^2 = \pi (0.2\text{ m})^2 = 0.126\text{ m}^2$$



Then the rate of heat transfer through the shell by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (80.2\text{ W/m}\cdot{}^\circ\text{C})(0.126\text{ m}^2)\frac{(5-0)^\circ\text{C}}{0.002\text{ m}} = 25{,}263\text{ W} = \mathbf{25.3\,kW}$$

Considering that it takes 333.7 kJ of energy to melt 1 kg of ice at 0°C, the rate at which ice melts in the container can be determined from

$$\dot{m}_{\text{ice}} = \frac{\dot{Q}}{h_{if}} = \frac{25.263\text{ kJ/s}}{333.7\text{ kJ/kg}} = \mathbf{0.0757\ \ kg/s}$$

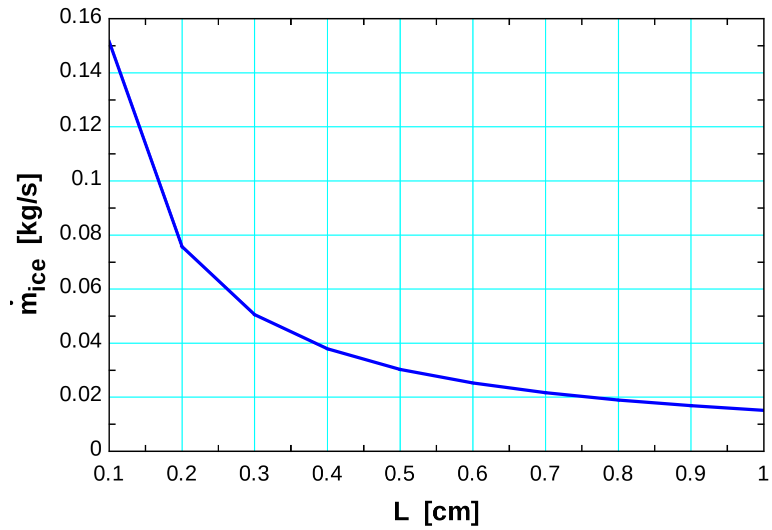
Discussion We should point out that this result is slightly in error for approximating a curved wall as a plain wall. The error in this case is very small because of the large diameter to thickness ratio. For better accuracy, we could use the inner surface area (D = 19.6 cm) or the mean surface area (D = 19.8 cm) in the calculations.

1-63 Prob. 1-62 is reconsidered. The rate at which ice melts as a function of the container thickness is to be plotted.

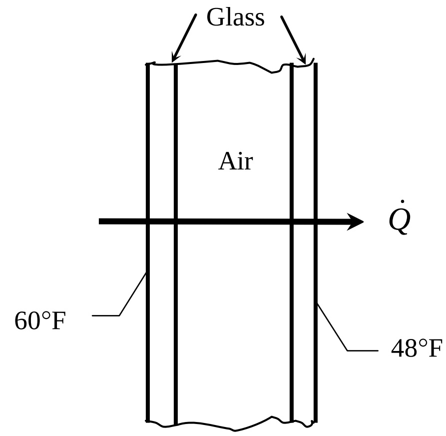
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
D=0.2 [m]
L=0.2 [cm]
T_1=0 [C]
T_2=5 [C]

"PROPERTIES"
h_if=333.7 [kJ/kg]
k=k_(Iron, 25)

"ANALYSIS"
A=pi*D^2
Q_dot_cond=k*A*(T_2-T_1)/(L*Convert(cm, m))
m_dot_ice=(Q_dot_cond*Convert(W, kW))/h_if
```

L [cm]	m_{ice} [kg/s]
0.1	0.1515
0.2	0.07574
0.3	0.0505
0.4	0.03787
0.5	0.0303
0.6	0.02525
0.7	0.02164
0.8	0.01894
0.9	0.01683
1	0.01515



1-64E The inner and outer glasses of a double pane window with a 0.5-in air space are at specified temperatures. The rate of heat transfer through the window is to be determined



Assumptions **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Heat transfer through the window is one-dimensional. **3** Thermal properties of the air are constant.

Properties The thermal conductivity of air at the average temperature of (60+48)/2 = 54°F is k = 0.01419 Btu/h·ft·°F (Table A-15E).

Analysis The area of the window and the rate of heat loss through it are

$$A = (4\ \text{ft}) \times (4\ \text{ft}) = 16\ \text{ft}^2$$

$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.01419\ \text{Btu/h.ft.°F})(16\ \text{ft}^2)\frac{(60-48)\text{°F}}{0.25/12\ \text{ft}} = \mathbf{131\ Btu/h}$$

1-65E Using the conversion factors between W and Btu/h, m and ft, and °C and °F, the convection coefficient in SI units is to be expressed in Btu/h·ft^2·°F.

Analysis The conversion factors for W and m are straightforward, and are given in conversion tables to be

$$1\ \text{W} = 3.41214\ \text{Btu/h}$$
$$1\ \text{m} = 3.2808\ \text{ft}$$

The proper conversion factor between °C into °F in this case is

$$1\text{°C} = 1.8\text{°F}$$

since the °C in the unit W/m^2·°C represents *per °C change in temperature*, and 1°C change in temperature corresponds to a change of 1.8°F. Substituting, we get

$$1\ \text{W/m}^2\cdot\text{°C} = \frac{3.41214\ \text{Btu/h}}{(3.2808\ \text{ft})^2(1.8\ \text{°F})} = 0.1761\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{°F}$$

which is the desired conversion factor. Therefore, the given convection heat transfer coefficient in English units is

$$h = 22\ \text{W/m}^2\cdot\text{°C} = 22 \times 0.1761\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{°F} = \mathbf{3.87\ Btu/h\cdot ft^2\cdot °F}$$

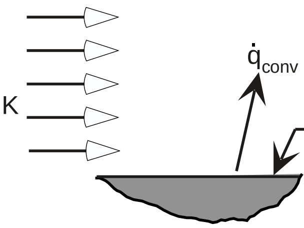
1-66 The heat flux between air with a constant temperature and convection heat transfer coefficient blowing over a pond at a constant temperature is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Convection heat transfer coefficient is uniform. **3** Heat transfer by radiation is negligible. **4** Air temperature and the surface temperature of the pond remain constant.

Analysis From Newton's law of cooling, the heat flux is given as

$$\dot{q}_{conv} = h\,(T_s - T_\infty)$$

$$\dot{q}_{conv} = 20\ \frac{\text{W}}{\text{m}^2\cdot\text{K}}\ (40 - 20)°\text{C} = \mathbf{400\ W/m^2}$$



Discussion (1) Note the direction of heat flow is out of the surface since $T_s > T_\infty$; (2) Recognize why units of K in h and units of ºC in (T_s - T_∞) cancel.

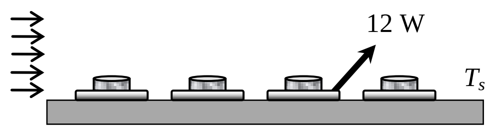
1-67 Four power transistors are mounted on a thin vertical aluminum plate that is cooled by a fan. The temperature of the aluminum plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The entire plate is nearly isothermal. **3** Thermal properties of the wall are constant. **4** The exposed surface area of the transistor can be taken to be equal to its base area. **5** Heat transfer by radiation is disregarded. **6** The convection heat transfer coefficient is constant and uniform over the surface.

Analysis The total rate of heat dissipation from the aluminum plate and the total heat transfer area are

$$\dot{Q} = 4\times 12\ \text{W} = 48\ \text{W}$$

$$A_s = 2(0.22\ \text{m})(0.22\ \text{m}) = 0.0968\ \text{m}^2$$



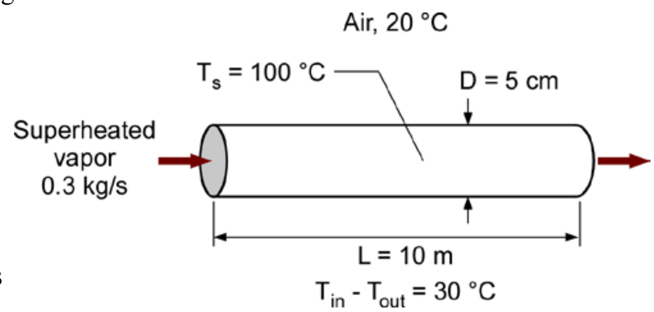
Disregarding any radiation effects, the temperature of the aluminum plate is determined to be

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 25°\text{C} + \frac{48\ \text{W}}{(25\ \text{W/m}^2\cdot°\text{C})(0.0968\ \text{m}^2)} = 44.8°\mathbf{C}$$

1-68 The convection heat transfer coefficient heat transfer between the surface of a pipe carrying superheated vapor and the surrounding is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** Rate of heat loss from the vapor in the pipe is equal to the heat transfer rate by convection between pipe surface and the surrounding.

Properties The specific heat of vapor is given to be 2190 J/kg · °C.



Analysis The surface area of the pipe is

$$A_s = \pi DL = \pi(0.05\,\text{m})(10\,\text{m}) = 1.571\,\text{m}^2$$

The rate of heat loss from the vapor in the pipe can be determined from

$$\begin{aligned}\dot{Q}_{\text{loss}} &= \dot{m}c_p(T_{\text{in}} - T_{\text{out}})\\ &= (0.3\,\text{kg/s})(2190\,\text{J/kg}\cdot{}^\circ\text{C})(30)\,{}^\circ\text{C} = 19710\,\text{J/s}\\ &= 19710\,\text{W}\end{aligned}$$

With the rate of heat loss from the vapor in the pipe assumed equal to the heat transfer rate by convection, the heat transfer coefficient can be determined using the Newton's law of cooling:

$$\dot{Q}_{\text{loss}} = \dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty)$$

Rearranging, the heat transfer coefficient is determined to be

$$h = \frac{\dot{Q}_{\text{loss}}}{A_s(T_s - T_\infty)} = \frac{19710\,\text{W}}{(1.571\,\text{m}^2)(100-20)\,{}^\circ\text{C}} = \mathbf{157\ W/m^2\cdot{}^\circ C}$$

Discussion By insulating the pipe surface, heat loss from the vapor in the pipe can be reduced.

1-69 An electrical resistor with a uniform temperature of 90 °C is in a room at 20 °C. The heat transfer coefficient by convection is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation heat transfer is negligible. **3** No hot spot exists on the resistor.

Analysis The total heat transfer area of the resistor is

$$A_s = 2(\pi D^2/4) + \pi DL = 2\pi(0.025\,\text{m})^2/4 + \pi(0.025\,\text{m})(0.15\,\text{m}) = 0.01276\,\text{m}^2$$

The electrical energy converted to thermal energy is transferred by convection:

$$\dot{Q}_{\text{conv}} = IV = (5\,\text{A})(6\,\text{V}) = 30\,\text{W}$$

From Newton's law of cooling, the heat transfer by convection is given as

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

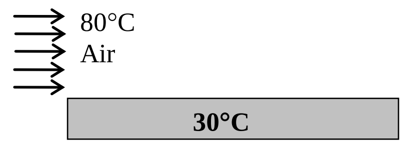
Rearranging, the heat transfer coefficient is determined to be

$$h = \frac{\dot{Q}_{\text{conv}}}{A_s(T_s - T_\infty)} = \frac{30\,\text{W}}{(0.01276\,\text{m}^2)(90-20)\,{}^\circ\text{C}} = \mathbf{33.6\ W/m^2\cdot{}^\circ C}$$

Discussion By comparing the magnitude of the heat transfer coefficient determined here with the values presented in Table 1-5, one can conclude that it is likely that forced convection is taking place rather than free convection.

1-70 Hot air is blown over a flat surface at a specified temperature. The rate of heat transfer from the air to the plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The convection heat transfer coefficient is constant and uniform over the surface.



Analysis Under steady conditions, the rate of heat transfer by convection is

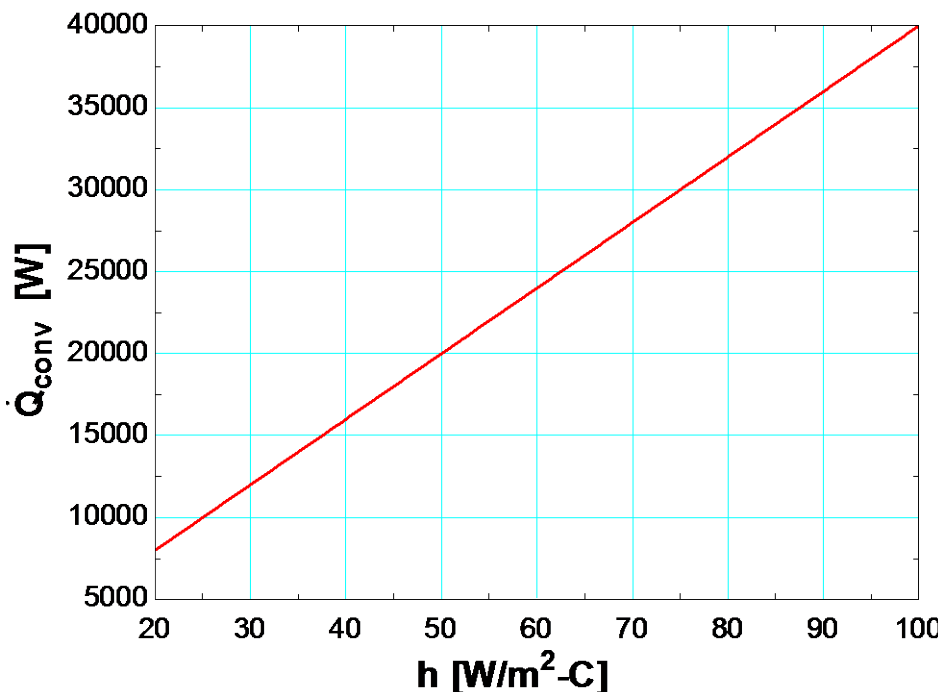
$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (55\ \text{W/m}^2\cdot°\text{C})(2\times 4\ \text{m}^2)(80-30)°\text{C} = \mathbf{22{,}000\ W}$$

1-71 Prob. 1-70 is reconsidered. The rate of heat transfer as a function of the heat transfer coefficient is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

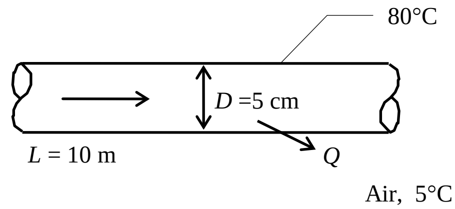
```
"GIVEN"
T_infinity=80 [C]
A=2*4 [m^2]
T_s=30 [C]
h=55 [W/m^2-C]

"ANALYSIS"
Q_dot_conv=h*A*(T_infinity-T_s)
```

h [W/m^2.C]	Q_{conv} [W]
20	8000
30	12000
40	16000
50	20000
60	24000
70	28000
80	32000
90	36000
100	40000



1-72 A hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 W/m^2·°C. The rate of heat loss from the pipe by convection is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The convection heat transfer coefficient is constant and uniform over the surface.



Analysis The heat transfer surface area is

$$A_s = \pi DL = \pi(0.05 \text{ m})(10 \text{ m}) = 1.571 \text{ m}^2$$

Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (25 \text{W/m}^2 \cdot °\text{C})(1.571 \text{ m}^2)(80-5)°\text{C} = \mathbf{2945\ W}$$

1-73 An AISI 316 spherical container is used for storing chemical undergoing exothermic reaction that provide a uniform heat flux to its inner surface. The necessary convection heat transfer coefficient to keep the container's outer surface below 50°C is to be determined.

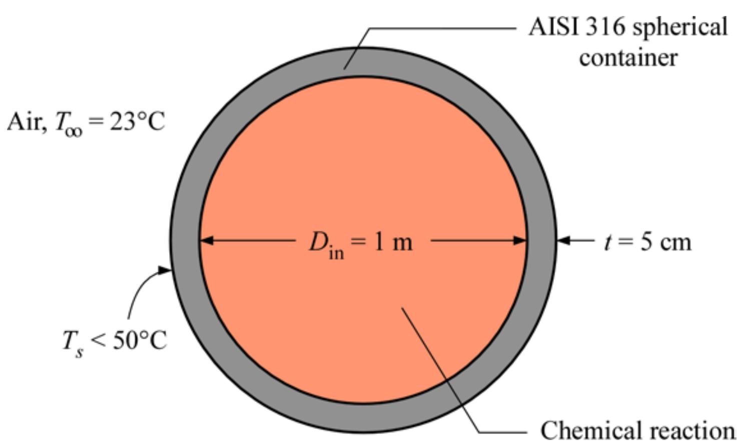
Assumptions **1** Steady operating conditions exist. **2** Negligible thermal storage for the container. **3** Temperature at the surface remained uniform.

Analysis The heat rate from the chemical reaction provided to the inner surface equal to heat rate removed from the outer surface by convection

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{out}}$$

$$\dot{q}_{\text{reaction}} A_{s,\text{in}} = hA_{s,\text{out}}(T_s - T_\infty)$$

$$\dot{q}_{\text{reaction}}(\pi D_{\text{in}}^2) = h(\pi D_{\text{out}}^2)(T_s - T_\infty)$$



The convection heat transfer coefficient can be determined as

$$h = \frac{\dot{q}_{\text{reaction}}}{T_s - T_\infty}\left(\frac{D_{\text{in}}}{D_{\text{out}}}\right)^2$$

$$= \frac{60000 \text{ W/m}^2}{(50-23) \text{ K}}\left(\frac{1 \text{ m}}{1 \text{ m} + 2 \times 0.05 \text{ m}}\right)^2$$

$$= 1840 \text{ W/m}^2 \cdot \text{K}$$

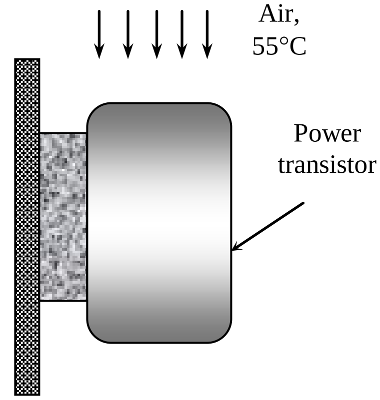
To keep the container's outer surface temperature below 50°C, the convection heat transfer coefficient should be

$$h > \mathbf{1840\ W/m^2 \cdot K}$$

Discussion From Table 1-5, the typical values for free convection heat transfer coefficient of gases are between 2–25 W/m^2·K. Thus, the required h > 1840 W/m^2·K is not feasible with free convection of air. To prevent thermal burn, the container's outer surface temperature should be covered with insulation.

1-74 A transistor mounted on a circuit board is cooled by air flowing over it. The transistor case temperature is not to exceed 70°C when the air temperature is 55°C. The amount of power this transistor can dissipate safely is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is disregarded. **3** The convection heat transfer coefficient is constant and uniform over the surface. **4** Heat transfer from the base of the transistor is negligible.



Analysis Disregarding the base area, the total heat transfer area of the transistor is

$$
\begin{aligned}
A_s &= \pi DL + \pi D^2/4 \\
&= \pi(0.6\,\text{cm})(0.4\,\text{cm}) + \pi(0.6\,\text{cm})^2/4 = 1.037\,\text{cm}^2 \\
&= 1.037\times10^{-4}\ \text{m}^2
\end{aligned}
$$

Then the rate of heat transfer from the power transistor at specified conditions is

$$\dot{Q} = hA_s(T_s - T_\infty) = (30\ \text{W/m}^2\cdot°\text{C})(1.037\times10^{-4}\ \text{m}^2)(70-55)°\text{C} = \mathbf{0.047\ W}$$

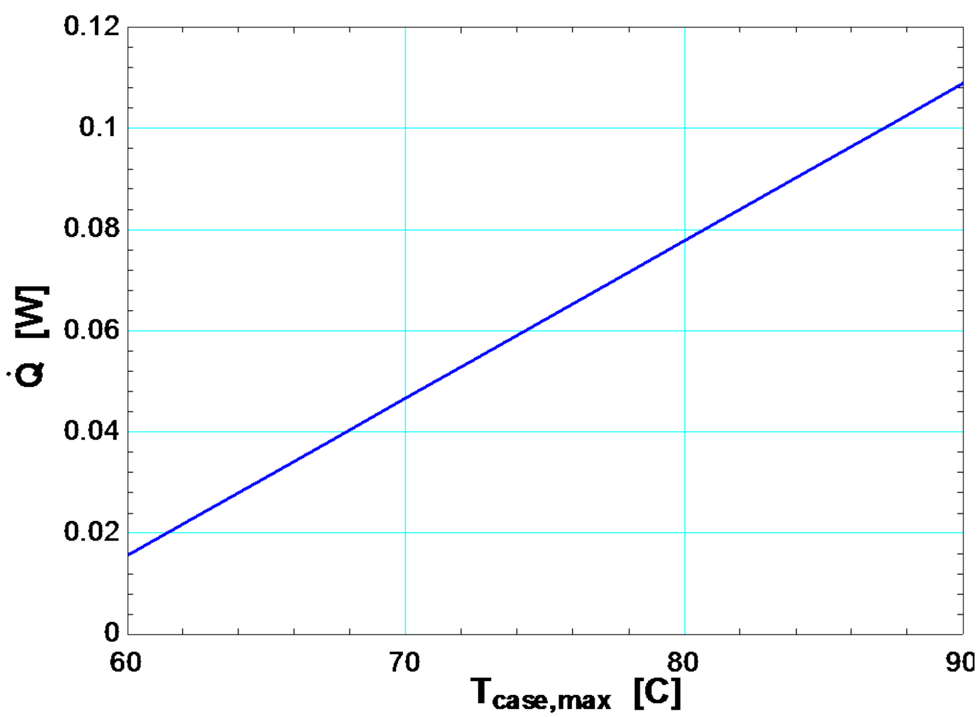
Therefore, the amount of power this transistor can dissipate safely is 0.047 W.

1-75 Prob. 1-74 is reconsidered. The amount of power the transistor can dissipate safely as a function of the maximum case temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=0.004 [m]
D=0.006 [m]
h=30 [W/m^2-C]
T_infinity=55 [C]
T_case_max=70 [C]

"ANALYSIS"
A=pi*D*L+pi*D^2/4
Q_dot=h*A*(T_case_max-T_infinity)
```

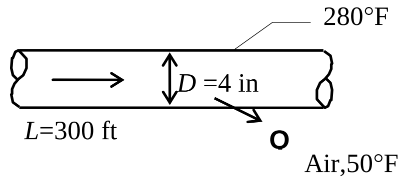
$T_{case, max}$ [C]	Q [W]
60	0.01555
62.5	0.02333
65	0.0311
67.5	0.03888
70	0.04665
72.5	0.05443
75	0.0622
77.5	0.06998
80	0.07775
82.5	0.08553
85	0.09331
87.5	0.1011
90	0.1089



1-76E A 300-ft long section of a steam pipe passes through an open space at a specified temperature. The rate of heat loss from the steam pipe and the annual cost of this energy lost are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is disregarded. **3** The convection heat transfer coefficient is constant and uniform over the surface.

Analysis (*a*) The rate of heat loss from the steam pipe is



$$A_s = \pi DL = \pi(4/12\text{ ft})(300\text{ ft}) = 314.2\text{ ft}^2$$

$$\dot{Q}_{\text{pipe}} = hA_s(T_s - T_{\text{air}}) = (6\text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})(314.2\text{ ft}^2)(280-50)^\circ\text{F}$$
$$= 433{,}540\text{ Btu/h} \cong \mathbf{433{,}500\ Btu/h}$$

(*b*) The amount of heat loss per year is

$$Q = \dot{Q}\Delta t = (433{,}540\text{ Btu/h})(365\times 24\text{ h/yr}) = 3.798\times 10^9\text{ Btu/yr}$$

The amount of gas consumption per year in the furnace that has an efficiency of 86% is

$$\text{Annual Energy Loss} = \frac{3.798\times 10^9\text{ Btu/yr}}{0.86}\left(\frac{1\text{ therm}}{100{,}000\text{ Btu}}\right) = 44{,}161\text{ therms/yr}$$

Then the annual cost of the energy lost becomes

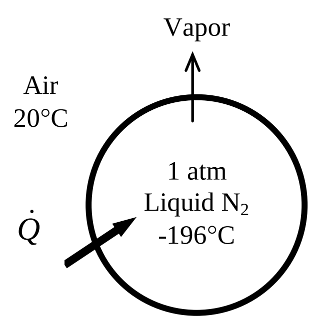
$$\text{Energy cost} = (\text{Annual energy loss})(\text{Unit cost of energy})$$
$$= (44{,}161\text{ therms/yr})(\$1.10/\text{therm}) = \$48{,}576/\text{yr} \cong \mathbf{\$48{,}600}$$

1-77 A 4-m diameter spherical tank filled with liquid nitrogen at 1 atm and -196°C is exposed to convection with ambient air. The rate of evaporation of liquid nitrogen in the tank as a result of the heat transfer from the ambient air is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is disregarded. **3** The convection heat transfer coefficient is constant and uniform over the surface. **4** The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the nitrogen inside.

Properties The heat of vaporization and density of liquid nitrogen at 1 atm are given to be 198 kJ/kg and 810 kg/m^3, respectively.

Analysis The rate of heat transfer to the nitrogen tank is



$$A_s = \pi D^2 = \pi(4\text{ m})^2 = 50.27\text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_{\text{air}}) = (25\text{ W/m}^2\cdot{}^\circ\text{C})(50.27\text{ m}^2)[20-(-196)]^\circ\text{C}$$
$$= 271{,}430\text{ W}$$

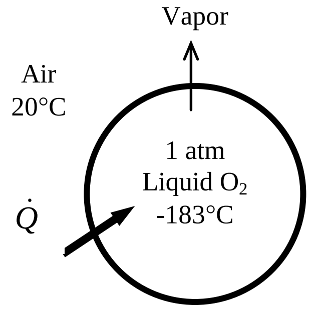
Then the rate of evaporation of liquid nitrogen in the tank is determined to be

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{271.430\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{1.37\ kg/s}$$

1-78 A 4-m diameter spherical tank filled with liquid oxygen at 1 atm and -183°C is exposed to convection with ambient air. The rate of evaporation of liquid oxygen in the tank as a result of the heat transfer from the ambient air is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is disregarded. **3** The convection heat transfer coefficient is constant and uniform over the surface. **4** The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the oxygen inside.

Properties The heat of vaporization and density of liquid oxygen at 1 atm are given to be 213 kJ/kg and 1140 kg/m^3, respectively.



Analysis The rate of heat transfer to the oxygen tank is

$$A_s = \pi D^2 = \pi(4\text{ m})^2 = 50.27\text{ m}^2$$

$$\begin{aligned}\dot{Q} &= hA_s(T_s - T_{\text{air}}) = (25\text{ W/m}^2.°\text{C})(50.27\text{ m}^2)[20-(-183)]°\text{C} \\ &= 255{,}120\text{ W}\end{aligned}$$

Then the rate of evaporation of liquid oxygen in the tank is determined to be

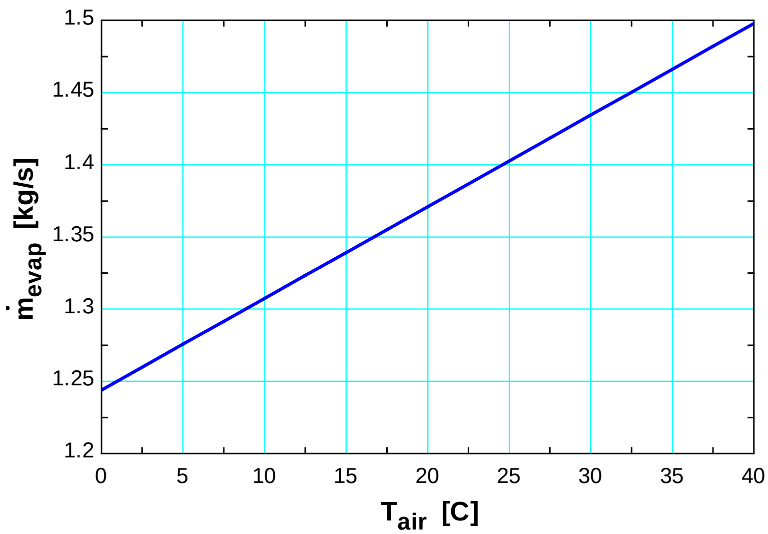
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{255.120\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{1.20\text{ kg/s}}$$

1-79 Prob. 1-77 is reconsidered. The rate of evaporation of liquid nitrogen as a function of the ambient air temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
D=4 [m]
T_s=-196 [C]
T_air=20 [C]
h=25 [W/m^2-C]

"PROPERTIES"
h_fg=198 [kJ/kg]

"ANALYSIS"
A=pi*D^2
Q_dot=h*A*(T_air-T_s)
m_dot_evap=(Q_dot*Convert(J/s, kJ/s))/h_fg
```

T_{air} [C]	m_{evap} [kg/s]
0	1.244
2.5	1.26
5	1.276
7.5	1.292
10	1.307
12.5	1.323
15	1.339
17.5	1.355
20	1.371
22.5	1.387
25	1.403
27.5	1.418
30	1.434
32.5	1.45
35	1.466
37.5	1.482
40	1.498

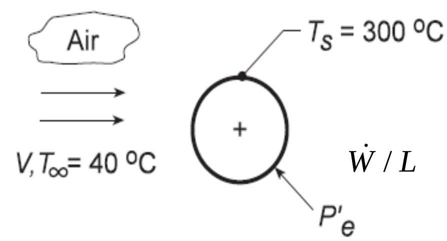


1-80 Power required to maintain the surface temperature of a long, 25 mm diameter cylinder with an imbedded electrical heater for different air velocities.

Assumptions 1 Temperature is uniform over the cylinder surface. **2** Negligible radiation exchange between the cylinder surface and the surroundings. **3** Steady state conditions.

Analysis (*a*) From an overall energy balance on the cylinder, the power dissipated by the electrical heater is transferred by convection to the air stream. Using Newton's law of cooling on a per unit length basis,

$$\dot{W}/L = h\,A_s\,(T_s - T_\infty) = h\,(\pi D)\,(T_s - T_\infty)$$



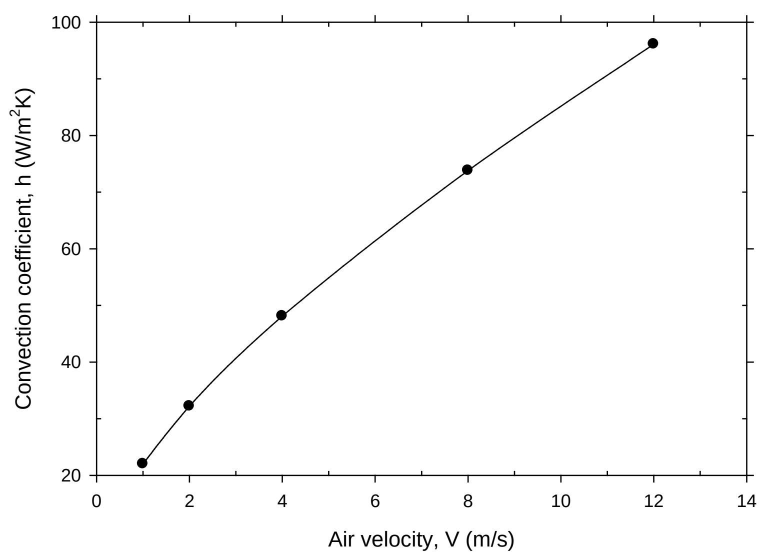
where $\dot{W}/L$ is the electrical power dissipated per unit length of the cylinder. For the V = 1 m/s condition, using the data from the table given in the problem statement, find

$$h = (\dot{W}/L\,)\,/\,(\pi D)\,(T_s - T_\infty)$$

$$h = 450 \text{ W/m} / (\pi \times 0.25 \text{ m})\,(300 - 40)\text{ °C} = 22.0 \text{ W/m}^2\cdot\text{K}$$

Repeating the calculations for the rest of the *V* values given, find the convection coefficients for the remaining conditions in the table. The results are tabulated and plotted below. Note that h is not linear with respect to the air velocity.

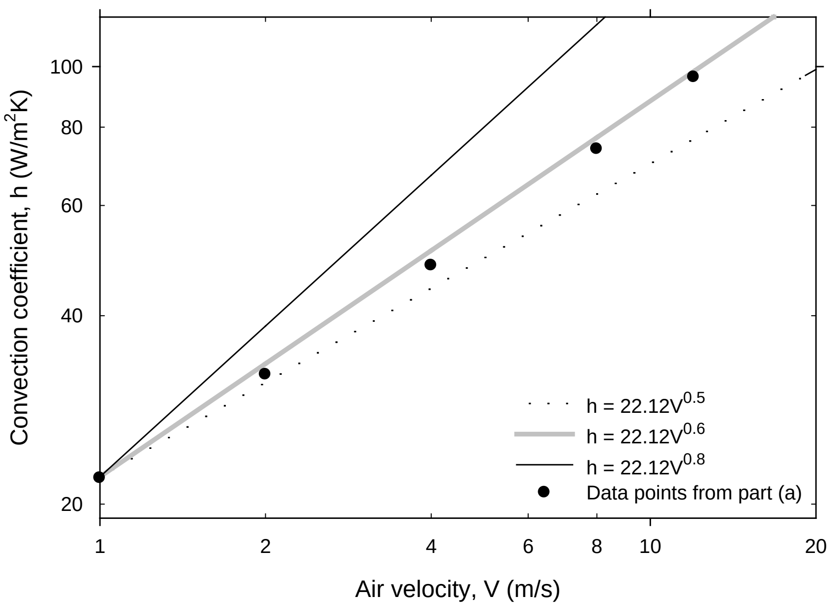
V (m/s)	$\dot{W}/L$ (W/m)	h (W/m^2·K)
1	450	22.0
2	658	32.2
4	983	48.1
8	1507	73.8
12	1963	96.1



Plot of convection coefficient (h) versus air velocity (V)

(*b*) To determine the constants C and n, plot *h* vs. *V* on log-log coordinates. Choosing C = 22.12 W/m^2.K(s/m)^n, assuring a match at V = 1, we can readily find the exponent n from the slope of the *h* vs. *V* curve. From the trials with n = 0.8, 0.6 and 0.5, we recognize that n = 0.6 is a reasonable choice. Hence, the best values of the constants are: **C = 22.12** and **n = 0.6**. The details of these trials are given in the following table and plot.

V (m/s)	$\dot{W}/L$ (W/m)	h (W/m^2·K)	$h = 22.12V^n$ (W/m^2·K)		
			n = 0.5	n = 0.6	n = 0.8
1	450	22.0	22.12	22.12	22.12
2	658	32.2	31.28	33.53	38.51
4	983	48.1	44.24	50.82	67.06
8	1507	73.8	62.56	77.03	116.75
12	1963	96.1	76.63	98.24	161.48



Plots for $h = CV^n$ with C = 22.12 and n = 0.5, 0.6, and 0.8

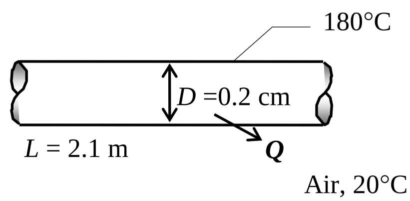
Discussion Radiation may not be negligible, depending on the surface emissivity.

1-81 The convection heat transfer coefficient for heat transfer from an electrically heated wire to air is to be determined by measuring temperatures when steady operating conditions are reached and the electric power consumed.

Assumptions **1** Steady operating conditions exist since the temperature readings do not change with time. **2** Radiation heat transfer is negligible.

Analysis In steady operation, the rate of heat loss from the wire equals the rate of heat generation in the wire as a result of resistance heating. That is,

$$\dot{Q} = \dot{E}_{\text{generated}} = \mathbf{V}I = (110\text{ V})(3\text{ A}) = 330\text{ W}$$



The surface area of the wire is

$$A_s = \pi DL = \pi(0.002\text{ m})(2.1\text{ m}) = 0.01319\text{ m}^2$$

The Newton's law of cooling for convection heat transfer is expressed as

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

Disregarding any heat transfer by radiation, the convection heat transfer coefficient is determined to be

$$h = \frac{\dot{Q}}{A_s(T_1 - T_\infty)} = \frac{330\text{ W}}{(0.01319\text{ m}^2)(180-20)^\circ\text{C}} = \mathbf{156\ W/m^2 \cdot {}^\circ C}$$

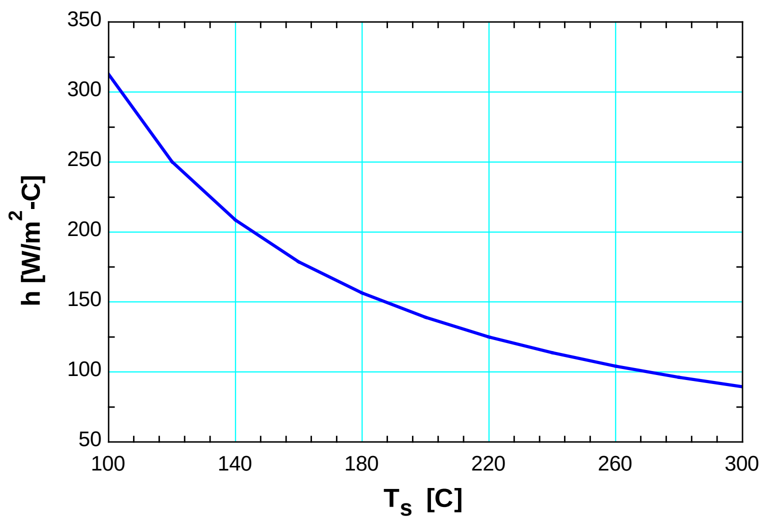
Discussion If the temperature of the surrounding surfaces is equal to the air temperature in the room, the value obtained above actually represents the combined convection and radiation heat transfer coefficient.

1-82 Prob. 1-81 is reconsidered. The convection heat transfer coefficient as a function of the wire surface temperature is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=2.1 [m]
D=0.002 [m]
T_infinity=20 [C]
T_s=180 [C]
V=110 [Volt]
I=3 [Ampere]

"ANALYSIS"
Q_dot=V*I
A=pi*D*L
Q_dot=h*A*(T_s-T_infinity)
```

T_s [C]	h [W/m^2.C]
100	312.6
120	250.1
140	208.4
160	178.6
180	156.3
200	138.9
220	125.1
240	113.7
260	104.2
280	96.19
300	89.32



1-83E Using the conversion factors between W and Btu/h, m and ft, and K and R, the Stefan-Boltzmann constant $\sigma = 5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4$ is to be expressed in the English unit, $\text{Btu/h} \cdot \text{ft}^2 \cdot \text{R}^4$.

Analysis The conversion factors for W, m, and K are given in conversion tables to be

$$1\,\text{W} = 3.41214\,\text{Btu/h}$$

$$1\,\text{m} = 3.2808\,\text{ft}$$

$$1\,\text{K} = 1.8\,\text{R}$$

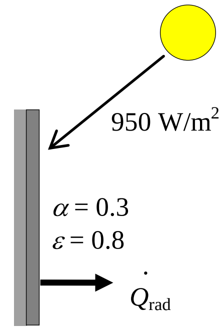
Substituting gives the Stefan-Boltzmann constant in the desired units,

$$\sigma = 5.67\ \text{W/m}^2 \cdot \text{K}^4 = 5.67 \times \frac{3.41214\ \text{Btu/h}}{(3.2808\ \text{ft})^2 (1.8\ \text{R})^4} = \mathbf{0.171\ Btu/h \cdot ft^2 \cdot R^4}$$

1-84 A spacecraft in space absorbs solar radiation while losing heat to deep space by thermal radiation. The surface temperature of the spacecraft is to be determined when steady conditions are reached.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Thermal properties of the wall are constant.

Properties The outer surface of a spacecraft has an emissivity of 0.8 and an absorptivity of 0.3.



Analysis When the heat loss from the outer surface of the spacecraft by radiation equals the solar radiation absorbed, the surface temperature can be determined from

$$\dot{Q}_{\text{solarabsorbed}} = \dot{Q}_{\text{rad}}$$

$$\alpha\dot{Q}_{\text{solar}} = \varepsilon\sigma A_s (T_s^4 - T_{\text{space}}^4)$$

$$0.3\times A_s \times (950\ \text{W/m}^2) = 0.8\times A_s \times (5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[T_s^4 - (0\ \text{K})^4]$$

Canceling the surface area A and solving for T_s gives

$$T_s = \mathbf{281.5\ K}$$

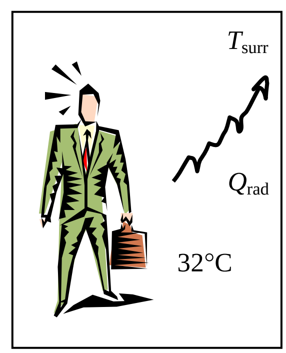
1-85 A person with a specified surface temperature is subjected to radiation heat transfer in a room at specified wall temperatures. The rate of radiation heat loss from the person is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by convection is disregarded. **3** The emissivity of the person is constant and uniform over the exposed surface.

Properties The average emissivity of the person is given to be 0.5.

Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are

(*a*) T_{surr} = 300 K



$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4)\\ &= (0.5)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.7\ \text{m}^2)[(32+273)^4 - (300\ \text{K})^4]\text{K}^4\\ &= \mathbf{26.7\ W}\end{aligned}$$

(*b*) T_{surr} = 280 K

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4)\\ &= (0.5)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.7\ \text{m}^2)[(32+273)^4 - (280\ \text{K})^4]\text{K}^4\\ &= \mathbf{121\ W}\end{aligned}$$

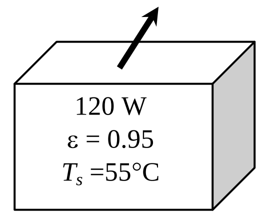
Discussion Note that the radiation heat transfer goes up by more than 4 times as the temperature of the surrounding surfaces drops from 300 K to 280 K.

1-86 A sealed electronic box dissipating a total of 120 W of power is placed in a vacuum chamber. If this box is to be cooled by radiation alone and the outer surface temperature of the box is not to exceed 55°C, the temperature the surrounding surfaces must be kept is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by convection is disregarded. **3** The emissivity of the box is constant and uniform over the exposed surface. **4** Heat transfer from the bottom surface of the box to the stand is negligible.

Properties The emissivity of the outer surface of the box is given to be 0.95.

Analysis Disregarding the base area, the total heat transfer area of the electronic box is



$$A_s = (0.5\,\text{m})(0.5\,\text{m}) + 4\times(0.2\,\text{m})(0.5\,\text{m}) = 0.65\,\text{m}^2$$

The radiation heat transfer from the box can be expressed as

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A_s(T_s^4 - T_{\text{surr}}^4)$$
$$120\text{ W} = (0.95)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)(0.65\,\text{m}^2)\left[(55+273\,\text{K})^4 - T_{\text{surr}}^4\right]$$

which gives T_{surr} = **300.4 K = 27.4°C**. Therefore, the temperature of the surrounding surfaces must be less than 27.4°C.

1-87 One highly polished surface at 1070°C and one heavily oxidized surface are emitting the same amount of energy per unit area. The temperature of the heavily oxidized surface is to be determined.

Assumptions The emissivity of each surface is constant and uniform.

Properties The emissivity of the highly polished surface is ε_1 = 0.1, and the emissivity of heavily oxidized surface is ε_2 = 0.78.

Analysis The rate of energy emitted by radiation is

$$\dot{Q}_{\text{emit}} = \varepsilon\sigma A_s T_s^4$$

For both surfaces to emit the same amount energy per unit area

$$(\dot{Q}_{\text{emit}}/A_s)_1 = (\dot{Q}_{\text{emit}}/A_s)_2$$

or

$$\varepsilon_1 T_{s,1}^4 = \varepsilon_2 T_{s,2}^4$$

The temperature of the heavily oxidized surface is

$$T_{s,2} = \left(\frac{\varepsilon_1}{\varepsilon_2}T_{s,1}^4\right)^{1/4} = \left[\frac{0.1}{0.78}(1070+273)^4\right]^{1/4}\text{ K} = \mathbf{803.6\ K}$$

Discussion If both surfaces are maintained at the same temperature, then the highly polished surface will emit less energy than the heavily oxidized surface.

1-88 A spherical probe in space absorbs solar radiation while losing heat to deep space by thermal radiation. The incident radiation rate on the probe surface is to be determined.

Assumptions **1** Steady operating conditions exist and surface temperature remains constant. **2** Heat generation is uniform.

Properties The outer surface the probe has an emissivity of 0.9 and an absorptivity of 0.1.

Analysis The rate of heat transfer at the surface of the probe can be expressed as

$$\dot{Q}_{\text{gen}} = \dot{Q}_{\text{rad}} - \dot{Q}_{\text{absorbed}}$$

$$\dot{e}_{\text{gen}} \boldsymbol{V} = \varepsilon\sigma A_s (T_s^4 - T_{\text{space}}^4) - \alpha A_s \dot{q}_{\text{solar}}$$

$$\dot{e}_{\text{gen}} \left(\frac{4}{3}\pi r^3 \right) = \varepsilon\sigma(4\pi r^2)(T_s^4 - T_{\text{space}}^4) - \alpha(4\pi r^2)\dot{q}_{\text{solar}}$$

Thus, incident radiation rate on the probe surface is

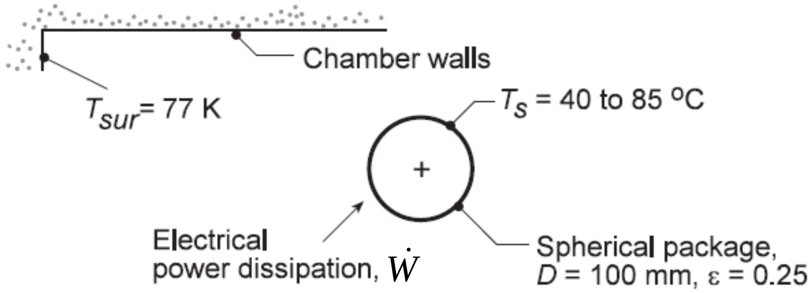
$$\dot{q}_{\text{solar}} = \frac{1}{\alpha}\left[\varepsilon\sigma(T_s^4 - T_{\text{space}}^4) - \frac{r}{3}\dot{e}_{\text{gen}} \right]$$

$$\dot{q}_{\text{solar}} = \frac{1}{0.1}\left[(0.9)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[(-40+273)^4 - 0)]\,\text{K}^4 - \frac{(1\,\text{m})(100\ \text{W/m}^3)}{3} \right] = \mathbf{1171\ W/m^2}$$

Discussion By adjusting the emissivity or absorptivity of the probe surface, the amount of incident radiation rate on the surface can be changed.

1-89 Spherical shaped instrumentation package with prescribed surface emissivity within a large laboratory room having walls at 77 K.

Assumptions 1 Uniform surface temperature. **2** Laboratory room walls are large compared to the spherical package. **3** Steady state conditions.



Analysis From an overall energy balance on the package, the internal power dissipation $\dot{W}$ will be transferred by radiation exchange between the package and the laboratory walls. The net rate of radiation between these two surfaces is given by

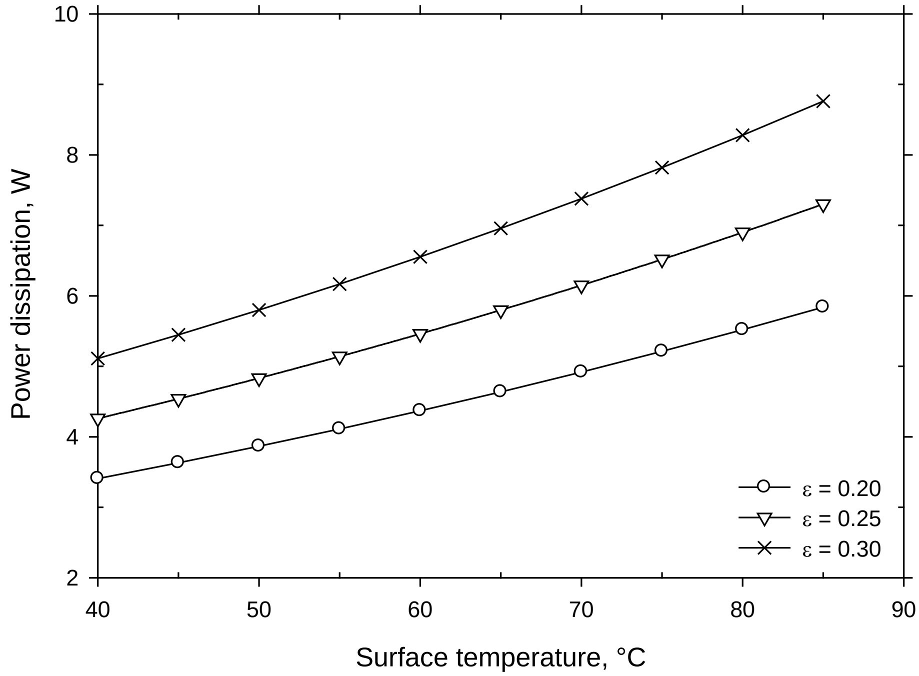
$$\dot{Q}_{rad} = \dot{W} = \varepsilon \sigma A_s (T_s^4 - T_{sur}^4)$$

For the condition when T_s = 40°C, ε = 0.25, with $A_s = \pi D^2$, the power dissipation will be

$$\dot{W} = 0.25\ (\pi \times 0.10^2\ \text{m}^2)\ (5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K})\ [(40 + 273)^4 - 77^4]\ \text{K}^4 = 4.3\ \text{W}$$

Repeating this calculation for the range 40ºC ≤ T_s ≤ 85°C, we can obtain the power dissipation as a function of surface temperature for the ε = 0.25 condition. Similarly, the calculation can be repeated for with 0.2 or 0.3. The details of these trials are given in the following table and plot.

	$\dot{W}$ (W)		
T_s (°C)	ε = 0.20	ε = 0.25	ε = 0.30
40	3.41	4.26	5.11
45	3.63	4.54	5.45
50	3.87	4.83	5.80
55	4.11	5.14	6.17
60	4.37	5.46	6.55
65	4.64	5.80	6.96
70	4.92	6.15	7.38
75	5.21	6.52	7.82
80	5.52	6.90	8.28
85	5.84	7.30	8.76



Family of curves for power dissipation ($\dot{W}$) versus surface temperature (T_s) for different values of emissivity (ε)

Discussion:

(1) As expected, the internal power dissipation increases with increasing emissivity and surface temperature. Because the radiation rate equation is non-linear with respect to temperature, the power dissipation will likewise not be linear with surface temperature.

(2) At a constant surface temperature, the power dissipation is linear with respect to the emissivity. The trends of the $\dot{W}$ versus T_s curves remained similar at different values of emissivity. By increasing the emissivity, the $\dot{W}$ versus T_s curve is shifted higher.

(3) The emissivity of various materials is listed in Tables A-18 and A-19 of the text.

(4) The maximum power dissipation possible if the surface temperature is not to exceed 85 °C is $\dot{W}$ = 8.76, where $\varepsilon = 0.30$. Similarly, the minimum power dissipation possible for the surface temperature range 40ºC $\leq T_s \leq$ 85°C is $\dot{W}$ = 3.41 W, where T_s = 40 °C and $\varepsilon = 0.20$. Therefore, the possible range of power dissipation for 40ºC $\leq T_s \leq$ 85 °C and $0.2 \leq \varepsilon \leq 0.3$ is $3.41 \leq \dot{W} \leq 8.76$ W.

Simultaneous Heat Transfer Mechanisms

1-90C All three modes of heat transfer cannot occur simultaneously in a medium. A medium may involve two of them simultaneously.

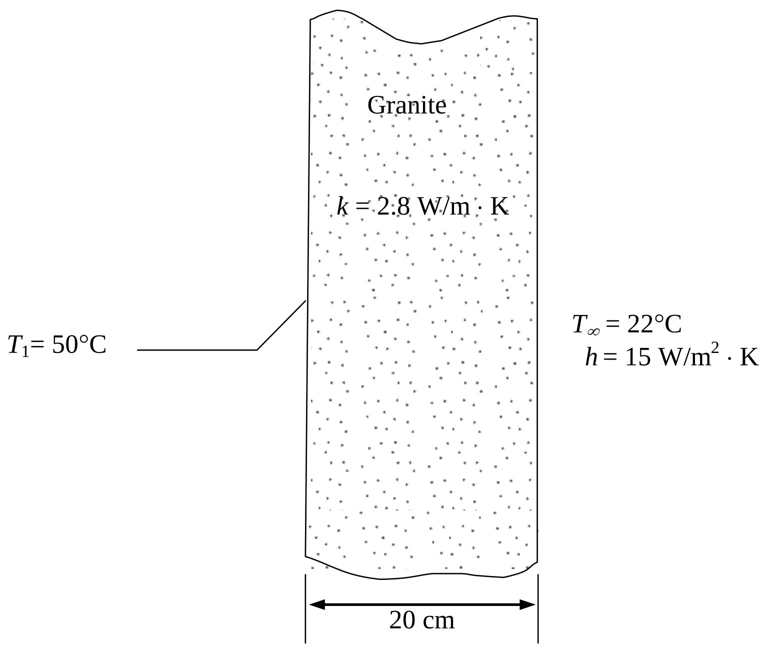
1-91C (*a*) Conduction and convection: No. (*b*) Conduction and radiation: Yes. Example: A hot surface on the ceiling. (*c*) Convection and radiation: Yes. Example: Heat transfer from the human body.

1-92C The human body loses heat by convection, radiation, and evaporation in both summer and winter. In summer, we can keep cool by dressing lightly, staying in cooler environments, turning a fan on, avoiding humid places and direct exposure to the sun. In winter, we can keep warm by dressing heavily, staying in a warmer environment, and avoiding drafts.

1-93C The fan increases the air motion around the body and thus the convection heat transfer coefficient, which increases the rate of heat transfer from the body by convection and evaporation. In rooms with high ceilings, ceiling fans are used in winter to force the warm air at the top downward to increase the air temperature at the body level. This is usually done by forcing the air up which hits the ceiling and moves downward in a gently manner to avoid drafts.

1-94 The right surface of a granite wall is subjected to convection heat transfer while the left surface is maintained as a constant temperature. The right wall surface temperature and the heat flux through the wall are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the granite wall is one dimensional. **3** Thermal conductivity of the granite wall is constant. **4** Radiation hest transfer is negligible.



Analysis The heat transfer through the wall by conduction is equal to heat transfer to the outer wall surface by convection:

$$\dot{q}_{\text{cond}} = \dot{q}_{\text{conv}}$$

$$k\frac{T_1 - T_2}{L} = h(T_2 - T_\infty)$$

$$T_2 = \frac{(kT_1 / L) + hT_\infty}{(k / L) + h}$$

$$T_2 = \frac{(2.8\,\text{W/m}\cdot\text{K})(50°\text{C})/(0.20\,\text{m}) + (15\,\text{W/m}^2\cdot\text{K})(22°\text{C})}{(2.8\,\text{W/m}\cdot\text{K})/(0.20\,\text{m}) + 15\,\text{W/m}^2\cdot\text{K}}$$

$$T_2 = \mathbf{35.5\,°C}$$

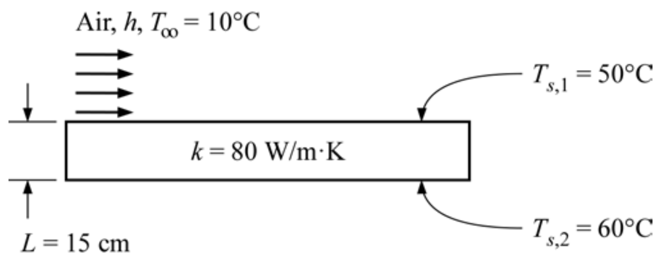
Now that T_2 is known, we can calculate the heat flux. Since the heat transfer through the wall by conduction is equal to heat transfer to the outer wall surface by convection, we may use either the Fourier's law of heat conduction or the Newton's law of cooling to find the heat flux. Using Fourier's law of heat conduction:

$$\dot{q}_{\text{cond}} = k\frac{T_1 - T_2}{L} = (2.8\,\text{W/m}\cdot\text{K})\frac{(50 - 35.5)\,°\text{C}}{0.20\,\text{m}} = \mathbf{203\ W/m^2}$$

Discussion Alternatively using Newton's law of cooling to find the heat flux, we obtain the same result:

$$\dot{q}_{conv} = h(T_2 - T_\infty) = (15\,\text{W/m}^2\cdot\text{K})\,(35.5 - 22) = 203\,\text{W/m}^2$$

1-95 The upper surface of a solid plate is being cooled by air. The air convection heat transfer coefficient at the upper plate surface is to be determined.



Assumptions **1** Steady operating conditions exist. **2** Thermal conductivity of the plate is constant. **3** Heat conduction in solid is one-dimensional. **4** Temperatures at the surfaces remained constant.

Properties The thermal conductivity of the solid plate is given as k = 80 W/m·K.

Analysis Applying energy balance on the upper surface of the solid plate

$$\dot{q}_{\text{cond}} = \dot{q}_{\text{conv}}$$

$$k\frac{T_{s,2} - T_{s,1}}{L} = h(T_{s,1} - T_\infty)$$

The convection heat transfer of the air is

$$h = \frac{k}{L}\frac{T_{s,2} - T_{s,1}}{T_{s,1} - T_\infty} = \frac{80\ \text{W/m}\cdot\text{K}}{0.15\ \text{m}}\left(\frac{60-50}{50-10}\right) = \mathbf{133\ W/m^2\cdot K}$$

Discussion A convection heat transfer coefficient of 133 W/m^2·K for forced convection of gas is reasonable when compared with the values listed in Table 1-5.

1-96 Air is blown over a hot horizontal plate which is maintained at a constant temperature. The surface also loses heat by radiation. The inside plate temperature is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the steel plate is one dimensional. **3** Thermal conductivity of the steel plate is constant.

Analysis The heat transfer by conduction through the plate is equal to the sum of convection and radiation heat loses:

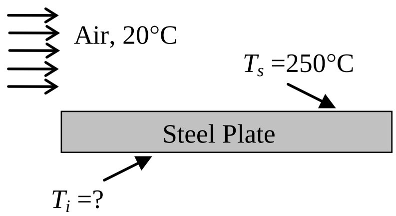
$$\dot{Q}_{\text{cond}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}}$$

where

$$\dot{Q}_{conv} = hA(T_s - T_\infty) = (25\,\text{W/m}^2\cdot\text{K})(0.38\,\text{m}^2)(250-20)\text{K} = 2185\,\text{W}$$

$$\dot{Q}_{rad} = 300\,\text{W}$$

Then,



$$\dot{Q}_{cond} = kA\frac{T_i - T_s}{L} = 2185\,\text{W} + 300\,\text{W} = 2485\,\text{W}$$

Solving for the inside plate temperature

$$T_i = T_s + \frac{\dot{Q}_{cond}\ L}{kA} = 250°\text{C} + \frac{(2485\,\text{W})(0.02\,\text{m})}{(43\,\text{W/m}\cdot\text{K})(0.38\,\text{m}^2)} = \mathbf{253°C}$$

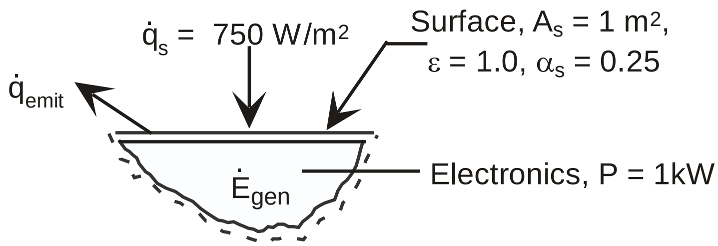
Discussion Heat loss by convection is much more dominant than heat loss by radiation. If we had not accounted for the heat loss by radiation in our calculation, the inside plate temperature would be 252.7ºC, which is only 0.3ºC less than the actual value. In this case we could have neglected the heat loss by radiation.

1-97 For an electronic package with given surface area, power dissipation, surface emissivity and absorptivity to solar radiation and the solar flux, the surface temperature with and without incident solar radiation is to be determined.

Assumptions **1** Steady operating conditions exist.

Analysis Apply conservation of energy (heat balance) to a control volume about the electronic package in rate form

$$\dot{Q}_{in} - \dot{Q}_{out} + \dot{E}_{gen} = \dot{E}_{stored} = 0$$



With the solar input, we have

$$\alpha_s A_s \dot{q}_s - A_s \dot{q}_{emit} + P = 0$$

where

$$\dot{q}_{emit} = \varepsilon\sigma T_s^4$$

Solving for the surface temperature T_s, we have

$$T_s = \left(\frac{\alpha_s A_s \dot{q}_s + P}{A_s \varepsilon\sigma}\right)^{1/4}$$

(a) Surface Temperature in the sun ($\dot{q}_s$ = 750 W/m^2)

$$T_s = \left[\frac{(0.25)(1\text{m}^2)(750\text{W/m}^2) + 1000\text{W}}{(1\text{m}^2)(1.0)(5.67\text{x}10^{-8}\text{W/m}^2\cdot\text{K}^4)}\right]^{1/4} = \mathbf{380.4\ K}$$

(b) Surface Temperature in the shade ($\dot{q}_s$ =0)

$$T_s = \left[\frac{1000\text{W}}{(1\text{m}^2)(1.0)(5.67\text{x}10^{-8}\text{W/m}^2\cdot\text{K}^4)}\right]^{1/4} = \mathbf{364.4\ K}$$

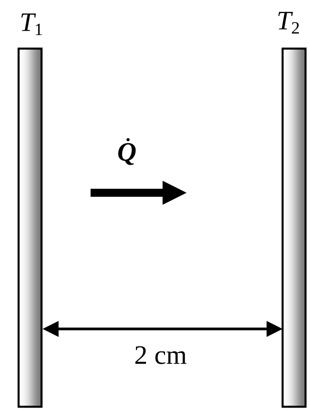
Discussion In orbit, the space station would be continuously cycling between shade and sunshine, and a steady-state condition would not exist.

1-98 Two large plates at specified temperatures are held parallel to each other. The rate of heat transfer between the plates is to be determined for the cases of still air, evacuation, regular insulation, and super insulation between the plates.

Assumptions **1** Steady operating conditions exist since the plate temperatures remain constant. **2** Heat transfer is one-dimensional since the plates are large. **3** The surfaces are black and thus ε = 1. **4** There are no convection currents in the air space between the plates.

Properties The thermal conductivities are k = 0.00015 W/m·°C for super insulation, k = 0.01979 W/m·°C at -50°C (Table A-15) for air, and k = 0.036 W/m·°C for fiberglass insulation (Table A-6).

Analysis (a) Disregarding any natural convection currents, the rates of conduction and radiation heat transfer



$$\dot{Q}_{\text{cond}} = kA\frac{T_1 - T_2}{L} = (0.01979\ \text{W/m}^2\cdot{}^\circ\text{C})(1\ \text{m}^2)\frac{(290-150)\ \text{K}}{0.02\ \text{m}} = 139\ \text{W}$$

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A_s(T_1^{\ 4} - T_2^{\ 4})$$
$$= 1(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(1\text{m}^2)\left[(290\ \text{K})^4 - (150\ \text{K})^4\right] = 372\ \text{W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{cond}} + \dot{Q}_{\text{rad}} = 139 + 372 = \mathbf{511\ W}$$

(b) When the air space between the plates is evacuated, there will be radiation heat transfer only. Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{rad}} = \mathbf{372\ W}$$

(c) In this case there will be conduction heat transfer through the fiberglass insulation only,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{cond}} = kA\frac{T_1 - T_2}{L} = (0.036\ \text{W/m}\cdot{}^\text{o}\text{C})(1\ \text{m}^2)\frac{(290-150)\ \text{K}}{0.02\ \text{m}} = \mathbf{252\ W}$$

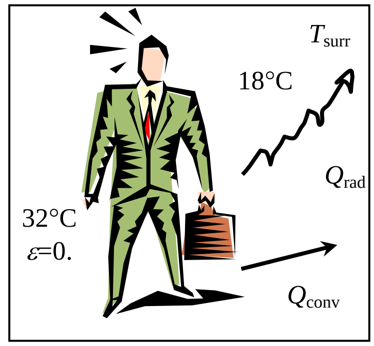
(d) In the case of superinsulation, the rate of heat transfer will be

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{cond}} = kA\frac{T_1 - T_2}{L} = (0.00015\ \text{W/m}\cdot{}^\circ\text{C})(1\ \text{m}^2)\frac{(290-150)\ \text{K}}{0.02\ \text{m}} = \mathbf{1.05\ W}$$

Discussion Note that superinsulators are very effective in reducing heat transfer between to surfaces.

1-99 The total rate of heat transfer from a person by both convection and radiation to the surrounding air and surfaces at specified temperatures is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The person is completely surrounded by the interior surfaces of the room. **3** The surrounding surfaces are at the same temperature as the air in the room. **4** Heat conduction to the floor through the feet is negligible. **5** The convection coefficient is constant and uniform over the entire surface of the person.



Properties The emissivity of a person is given to be ε = 0.9.

Analysis The person is completely enclosed by the surrounding surfaces, and he or she will lose heat to the surrounding air by convection and to the surrounding surfaces by radiation. The total rate of heat loss from the person is determined from

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4) = (0.90)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.7\ \text{m}^2)[(32+273)^4 - (18+273)^4]\text{K}^4 = 128.6\ \text{W}$$

$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (5\ \text{W/m}^2\cdot\text{K})(1.7\ \text{m}^2)(32-18)°\text{C} = 119\ \text{W}$$

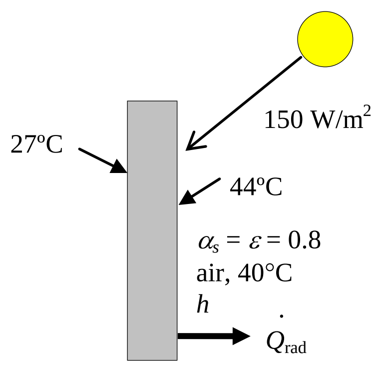
and

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 128.6 + 119 = \mathbf{247.6\ W}$$

Discussion Note that heat transfer from the person by evaporation, which is of comparable magnitude, is not considered in this problem.

1-100 The outer surface of a wall is exposed to solar radiation. The effective thermal conductivity of the wall is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat transfer coefficient is constant and uniform over the surface.



Properties Both the solar absorptivity and emissivity of the wall surface are given to be 0.8.

Analysis The heat transfer through the wall by conduction is equal to net heat transfer to the outer wall surface:

$$\dot{q}_{\text{cond}} = \dot{q}_{\text{conv}} + \dot{q}_{\text{rad}} + \dot{q}_{\text{solar}}$$

$$k\frac{T_2 - T_1}{L} = h(T_o - T_2) + \varepsilon\sigma(T_{\text{surr}}^4 - T_2^4) + \alpha_s q_{\text{solar}}$$

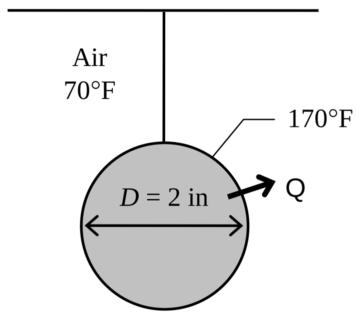
$$k\frac{(44\text{-}27)°\text{C}}{0.25\ \text{m}} = (8\ \text{W/m}^2\cdot°\text{C})(40-44)°\text{C} + (0.8)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)\left[(40+273\ \text{K})^4 - (44+273\ \text{K})^4\right]$$
$$+ (0.8)(150\ \text{W/m}^2)$$

Solving for k gives

$$k = \mathbf{0.961\ W/m\cdot°C}$$

1-101E A spherical ball whose surface is maintained at a temperature of 170°F is suspended in the middle of a room at 70°F. The total rate of heat transfer from the ball is to be determined.

Assumptions **1** Steady operating conditions exist since the ball surface and the surrounding air and surfaces remain at constant temperatures. **2** The thermal properties of the ball and the convection heat transfer coefficient are constant and uniform.



Properties The emissivity of the ball surface is given to be ε = 0.8.

Analysis The heat transfer surface area is

$$A_s = \pi D^2 = \pi(2/12\text{ ft})^2 = 0.08727\text{ ft}^2$$

Under steady conditions, the rates of convection and radiation heat transfer are

$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (15\text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})(0.08727\text{ ft}^2)(170-70)^\circ\text{F} = 130.9\text{ Btu/h}$$

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s(T_s^4 - T_o^4)\\ &= 0.8(0.08727\text{ ft}^2)(0.1714\times10^{-8}\text{ Btu/h}\cdot\text{ft}^2\cdot\text{R}^4)[(170+460\text{ R})^4-(70+460\text{ R})^4]\\ &= 9.4\text{ Btu/h}\end{aligned}$$

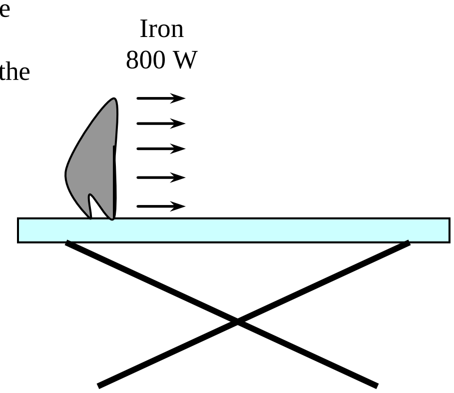
Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 130.9 + 9.4 = \mathbf{140.3\text{ Btu/h}}$$

Discussion Note that heat loss by convection is several times that of heat loss by radiation. The radiation heat loss can further be reduced by coating the ball with a low-emissivity material.

1-102 An 800-W iron is left on the iron board with its base exposed to the air at 20°C. The temperature of the base of the iron is to be determined in steady operation.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the iron base and the convection heat transfer coefficient are constant and uniform. **3** The temperature of the surrounding surfaces is the same as the temperature of the surrounding air.



Properties The emissivity of the base surface is given to be ε = 0.6.

Analysis At steady conditions, the 800 W energy supplied to the iron will be dissipated to the surroundings by convection and radiation heat transfer. Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 800 \text{ W}$$

where

$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (35 \text{ W/m}^2 \cdot \text{K})(0.02 \text{ m}^2)(T_s - 293 \text{ K}) = 0.7(T_s - 293 \text{ K})$$

and

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s(T_s^4 - T_o^4) = 0.6(0.02 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[T_s^4 - (293 \text{ K})^4] \\ &= 0.06804\times10^{-8}[T_s^4 - (293 \text{ K})^4]\end{aligned}$$

Substituting,

$$800 \text{ W} = 0.7(T_s - 293 \text{ K}) + 0.06804\times10^{-8}[T_s^4 - (293 \text{ K})^4]$$

Solving by trial and error gives

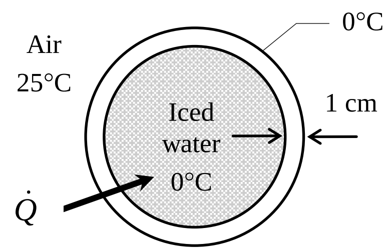
$$T_s = 874 \text{ K} = \mathbf{601°C}$$

Discussion We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 874 K.

1-103 A spherical tank located outdoors is used to store iced water at 0°C. The rate of heat transfer to the iced water in the tank and the amount of ice at 0°C that melts during a 24-h period are to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Thermal properties of the tank and the convection heat transfer coefficient is constant and uniform. **3** The average surrounding surface temperature for radiation exchange is 15°C. **4** The thermal resistance of the tank is negligible, and the entire steel tank is at 0°C.

Properties The heat of fusion of water at atmospheric pressure is $h_{if} = 333.7$ kJ/kg . The emissivity of the outer surface of the tank is 0.75.



Analysis (a) The outer surface area of the spherical tank is

$$A_s = \pi D^2 = \pi(3.02\text{ m})^2 = 28.65\text{ m}^2$$

Then the rates of heat transfer to the tank by convection and radiation become

$$\dot{Q}_{\text{conv}} = hA_s(T_\infty - T_s) = (30\text{ W/m}^2\cdot°\text{C})(28.65\text{ m}^2)(25-0)°\text{C} = 21{,}488\text{ W}$$

$$\dot{Q}_{\text{rad}} = \varepsilon A_s\sigma(T_{\text{surr}}^4 - T_s^4) = (0.75)(28.65\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)[(288\text{ K})^4 - (273\text{ K})^4] = 1614\text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 21{,}488 + 1614 = 23{,}102\text{ W} = \mathbf{23.1\,kW}$$

(b) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q}\Delta t = (23.102\text{ kJ/s})(24\times3600\text{ s}) = 1{,}996{,}000\text{ kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{1{,}996{,}000\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{5980\,kg}$$

Discussion The amount of ice that melts can be reduced to a small fraction by insulating the tank.

1-104 A draw batch furnace front is subjected to uniform heat flux on the inside surface, while the outside surface is subjected to convection and radiation heat transfer. The outside surface temperature is to be determined.

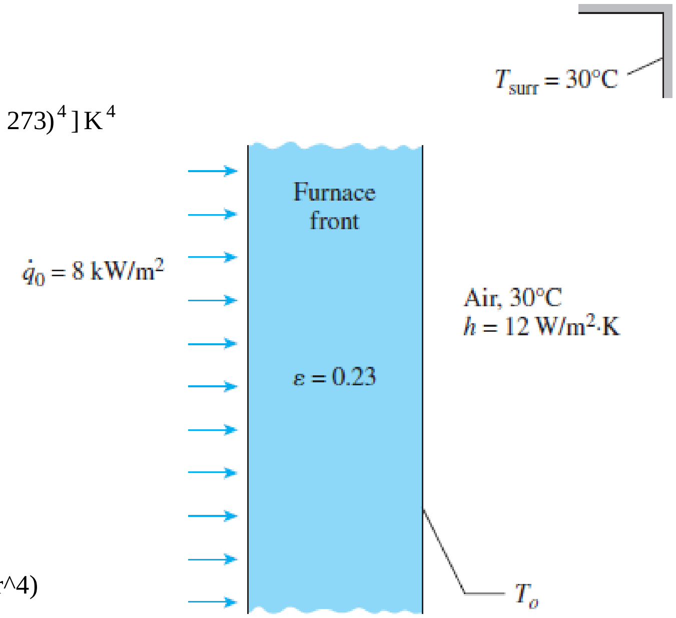
Assumptions **1** Heat conduction is steady. **2** One dimensional heat conduction across the furnace front thickness. **3** Uniform heat flux on inside surface.

Properties Emissivity and convective heat transfer coefficient are given to be 0.23 and 12 $W/m^2 \cdot K$, respectively.

Analysis The uniform heat flux subjected on the inside surface is equal to the sum of heat fluxes transferred by convection and radiation on the outside surface

$$\dot{q}_0 = h(T_o - T_\infty) + \varepsilon\sigma(T_o^4 - T_{surr}^4)$$

$$8000 \text{ W/m}^2 = (12 \text{ W/m}^2 \cdot \text{K})[T_o - (30+273)]\text{ K} \\ + (0.23)(5.67\times10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[T_o^4 - (30+273)^4]\text{ K}^4$$



Solving for the outer surface temperature yields

$$T_o = \mathbf{707\ K = 434°C}$$

Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
h=12 [W/m^2-K]
q_dot_0=8000 [W/m^2]
T_surr=303 [K]
epsilon=0.23
sigma=5.67e-8 [W/m^2-K^4]
q_dot_0=h*(T_o-T_surr)+epsilon*sigma#*(T_o^4-T_surr^4)
```

Discussion By insulating the furnace front, heat loss from the outer surface can be reduced.

1-105 A flat-plate solar absorber is exposed to an incident solar radiation. The efficiency of the solar absorber (the ratio of the usable heat collected by the absorber to the incident solar radiation on the absorber) is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Temperature at the surface remained constant.

Properties The absorber surface has an absorptivity of 0.93 and an emissivity of 0.9.

Analysis The rate of usable heat at the absorber plate can be expressed as

$$\dot{Q}_{\text{usable}} = \dot{Q}_{\text{absorbed}} - \dot{Q}_{\text{rad}} - \dot{Q}_{\text{conv}}$$

$$\dot{Q}_{\text{usable}} = \alpha A_s \dot{q}_{\text{solar}} - \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4) - hA_s (T_s - T_\infty)$$

Expressed in terms of heat flux, we have

$$\dot{q}_{\text{usable}} = \alpha\dot{q}_{\text{solar}} - \varepsilon\sigma(T_s^4 - T_{\text{surr}}^4) - h(T_s - T_\infty)$$

$$\dot{q}_{\text{usable}} = (0.93)(800\ \text{W/m}^2) - (0.9)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[(40+273)^4 - (-5+273)^4]\,\text{K}^4 - (7\ \text{W/m}^2\cdot\text{K})(40-20)\,\text{K}$$

$$\dot{q}_{\text{usable}} = 377.5\ \text{W/m}^2$$

Thus, the efficiency of the solar absorber is

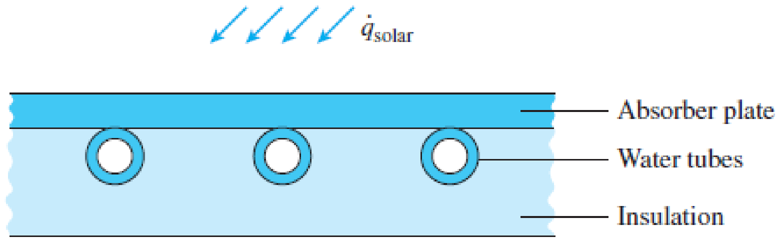
$$\eta = \frac{\dot{q}_{\text{usable}}}{\dot{q}_{\text{solar}}} = \frac{377.5\ \text{W/m}^2}{800\ \text{W/m}^2} = \mathbf{0.472}$$

Discussion The efficiency of the solar absorber is influenced by the surrounding and ambient temperatures, as well as the convective heat transfer coefficient. If the weather is particularly windy, thus causing higher level of heat loss via convection, then the efficiency of the solar absorber could be adversely affected.

1-106 A flat-plate solar collector is used to heat water. The temperature rise of the water heated by the net heat rate from the solar collector is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Specific heat of water is constant. **3** Temperature at the surface remained constant. **4** Conduction through the solar absorber is negligible. **5** Heat loss through the sides and back of the absorber is negligible.

Properties The absorber surface has an absorptivity of 0.9 and an emissivity of 0.9. The specific heat of water is given as 4.2 kJ/kg·K.



Analysis The net heat rate absorbed by the solar collector is

$$\dot{Q}_{net} = \dot{Q}_{absorbed} - \dot{Q}_{rad} - \dot{Q}_{conv}$$

$$\dot{Q}_{net} = A_s[\alpha\dot{q}_{solar} - \varepsilon\sigma(T_s^4 - T_{surr}^4) - h(T_s - T_\infty)]$$

$$\dot{Q}_{net} = (2\text{ m}^2)[(0.9)(500\text{ W/m}^2) - (0.9)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)[(35+273)^4 - (0+273)^4)]\text{K}^4$$
$$- (5\text{ W/m}^2\cdot\text{K})(35-25)\text{ K}]$$

$$\dot{Q}_{net} = 448.4\text{ W}$$

The temperature rise can be determined using

$$\dot{Q}_{net} = \dot{m}c_p(T_{out} - T_{in})$$

Thus,

$$T_{out} - T_{in} = \frac{\dot{Q}_{net}}{\dot{m}c_p} = \frac{448.4\text{ W}}{(0.005\text{ kg/s})(4200\text{ J/kg}\cdot\text{K})} = \mathbf{21.4°C}$$

Discussion The temperature rise of the water is influenced by the usable net heat rate absorbed by the solar collector and the water flow rate.

1-107 A draw batch furnace front is subjected to uniform heat flux on the inside surface, while the outside surface is subjected to convection and radiation heat transfer. The outside surface temperature is to be determined whether it is below 50°C or not.

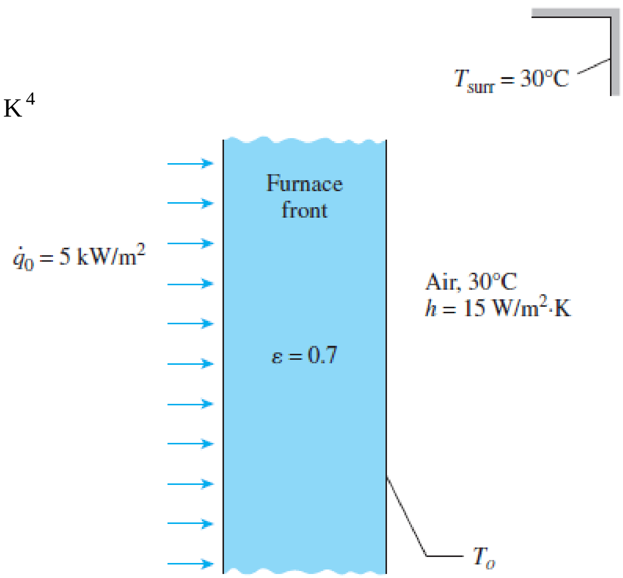
Assumptions **1** Heat conduction is steady. **2** One dimensional heat conduction across the furnace front thickness. **3** Uniform heat flux on inside surface.

Properties Emissivity and convective heat transfer coefficient are given to be 0.7 and 15 W/m^2·K, respectively.

Analysis The uniform heat flux subjected on the inside surface is equal to the sum of heat fluxes transferred by convection and radiation on the outside surface

$$\dot{q}_0 = h(T_o - T_\infty) + \varepsilon\sigma(T_o^4 - T_{\rm surr}^4)$$

$$5000\ \mathrm{W/m^2} = (15\ \mathrm{W/m^2\cdot K})[T_o - (30+273)]\ \mathrm{K}$$
$$+ (0.7)(5.67\times10^{-8}\ \mathrm{W/m^2\cdot K^4})[T_o^4 - (30+273)^4]\ \mathrm{K}^4$$



Solving for the outer surface temperature yields

$$T_o = \mathbf{497\ K} = \mathbf{224°C} > 50°\mathrm{C}$$

Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
h=15 [W/m^2-K]
q_dot_0=5000 [W/m^2]
T_surr=303 [K]
epsilon=0.7
sigma=5.67e-8 [W/m^2-K^4]
q_dot_0=h*(T_o-T_surr)+epsilon*sigma#*(T_o^4-T_surr^4)
```

Discussion Yes, insulation is required on the furnace front surface to avoid thermal burn upon contact with skin.

1-108 The roof of a house with a gas furnace consists of a 22-cm thick concrete that is losing heat to the outdoors by radiation and convection. The rate of heat transfer through the roof and the money lost through the roof that night during a 14 hour period are to be determined.

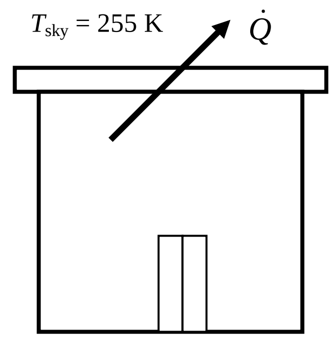
Assumptions **1** Steady operating conditions exist. **2** The emissivity and thermal conductivity of the roof are constant.

Properties The thermal conductivity of the concrete is given to be k = 2 W/m·°C. The emissivity of the outer surface of the roof is given to be 0.9.

Analysis In steady operation, heat transfer from the outer surface of the roof to the surroundings by convection and radiation must be equal to the heat transfer through the roof by conduction. That is,

$$\dot{Q} = \dot{Q}_{\text{roof, cond}} = \dot{Q}_{\text{roof to surroundings, conv+rad}}$$

The inner surface temperature of the roof is given to be $T_{s,in}$ = 15°C. Letting $T_{s,out}$ denote the outer surface temperatures of the roof, the energy balance above can be expressed as



$$\dot{Q} = kA\frac{T_{s,in} - T_{s,out}}{L} = h_o A(T_{s,out} - T_{surr}) + \varepsilon A\sigma(T_{s,out}{}^4 - T_{surr}{}^4)$$

$$\dot{Q} = (2\text{ W/m}\cdot°\text{C})(300\text{ m}^2)\frac{15°\text{C} - T_{s,out}}{0.22\text{ m}}$$
$$= (15\text{ W/m}^2.°\text{C})(300\text{ m}^2)(T_{s,out} - 10)°\text{C}$$
$$+ (0.9)(300\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)\left[(T_{s,out} + 273\text{ K})^4 - (255\text{ K})^4\right]$$

Solving the equations above using an equation solver (or by trial and error) gives

$$\dot{Q} = \mathbf{19{,}830\ W} \text{ and } T_{s,\,out} = \mathbf{7.8°C}$$

Then the amount of natural gas consumption during a 16-hour period is

$$E_{gas} = \frac{Q_{total}}{0.85} = \frac{\dot{Q}\Delta t}{0.85} = \frac{(19.83\text{ kJ/s})(14\times3600\text{ s})}{0.85}\left(\frac{1\text{ therm}}{105{,}500\text{ kJ}}\right) = 11.15\text{ therms}$$

Finally, the money lost through the roof during that period is

$$\text{Money lost} = (11.15\text{ therms})(\$1.20/\text{therm}) = \mathbf{\$13.4}$$

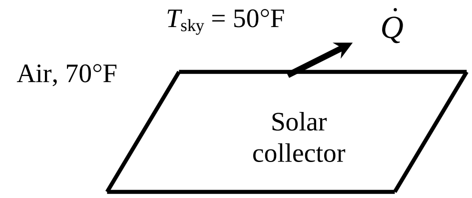
1-109E A flat plate solar collector is placed horizontally on the roof of a house. The rate of heat loss from the collector by convection and radiation during a calm day are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The emissivity and convection heat transfer coefficient are constant and uniform. **3** The exposed surface, ambient, and sky temperatures remain constant.

Properties The emissivity of the outer surface of the collector is given to be 0.9.

Analysis The exposed surface area of the collector is

$$A_s = (5\,\text{ft})(15\,\text{ft}) = 75\,\text{ft}^2$$



Noting that the exposed surface temperature of the collector is 100°F, the total rate of heat loss from the collector to the environment by convection and radiation becomes

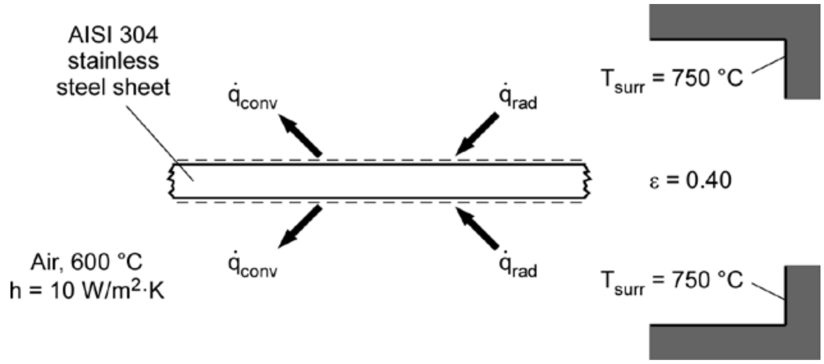
$$\dot{Q}_{\text{conv}} = hA_s(T_\infty - T_s) = (2.5\,\text{Btu/h.ft}^2\cdot{}^\circ\text{F})(75\,\text{ft}^2)(100-70)^\circ\text{F} = 5625\,\text{Btu/h}$$

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon A_s\sigma(T_{\text{surr}}^4 - T_s^4) = (0.9)(75\,\text{ft}^2)(0.1714\times10^{-8}\,\text{Btu/h}\cdot\text{ft}^2\cdot\text{R}^4)[(100+460\,\text{R})^4 - (50+460\,\text{R})^4] \\ &= 3551\,\text{Btu/h}\end{aligned}$$

and

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 5625 + 3551 = \mathbf{9176\,Btu/h}$$

1-110 Temperature of the stainless steel sheet going through an annealing process inside an electrically heated oven is to be determined.



Assumptions **1** Steady operating conditions exist. **2** Temperature of the stainless steel sheet is uniform. **3** Radiation heat transfer between stainless steel sheet and surrounding oven surfaces is between a small object and a large enclosure.

Properties The emissivity of the stainless steel sheet is given to be 0.40.

Analysis The amount of heat transfer by radiation between the sheet and the surrounding oven surfaces is balanced by the convection heat transfer between the sheet and the ambient air:

$$\dot{q}_{\text{rad}} - \dot{q}_{\text{conv}} = 0$$

$$\varepsilon\sigma(T_{\text{surr}}^4 - T_s^4) - h(T_s - T_\infty) = 0$$

$$(0.40)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[(750+273)^4 - T_s^4]\,\text{K}^4 - (10\ \text{W/m}^2\cdot\text{K})[T_s - (600+273)]\,\text{K} = 0$$

Solving the above equation by EES software (Copy the following line and paste on a blank EES screen to verify solution):

```
0.40*5.67e-8*((750+273)^4-T_s^4)-10*(T_s-(600+273))=0
```

The temperature of the stainless steel sheet is

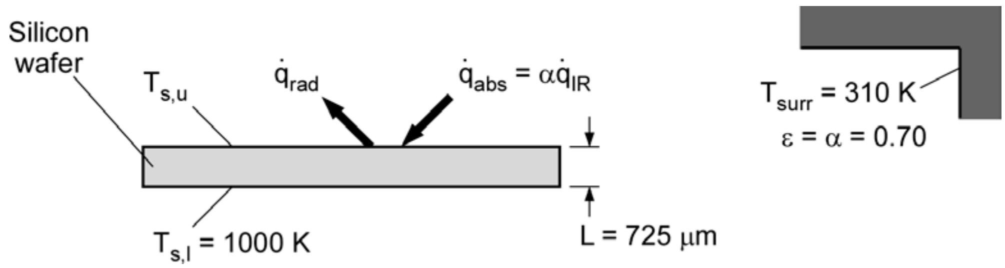
$$T_s = 1009\ \text{K} = \mathbf{736\ °C}$$

Discussion Note that the energy balance equation involving radiation heat transfer used for solving the stainless steel sheet temperature must be used with absolute temperature.

1-111 The upper surface temperature of a silicon wafer undergoing heat treatment in a vacuum chamber by infrared heater is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation heat transfer between upper wafer surface and surroundings is between a small object and a large enclosure. **3** One-dimensional conduction in wafer. **4** The silicon wafer has constant properties. **5** No hot spot exists on the wafer.

Properties The thermal conductivity of silicon at 1000 K is 31.2 W/m · K (Table A-3).



Analysis The heat transfer through the thickness of the wafer by conduction is equal to net heat transfer at the upper wafer surface:

$$\dot{q}_{\text{cond}} = \dot{q}_{\text{abs}} - \dot{q}_{\text{rad}}$$

$$k\frac{T_{s,u} - T_{s,l}}{L} = \alpha\dot{q}_{\text{IR}} - \varepsilon\sigma(T_{s,u}^4 - T_{\text{surr}}^4)$$

$$(31.2\text{ W/m}\cdot\text{K})\frac{(T_{s,u} - 1000)\text{ K}}{(725\times 10^{-6}\text{ m})} = (0.70)(200000\text{ W/m}^2) - (0.70)(5.67\times 10^{-8}\text{ W/m}^2\cdot\text{K}^4)(T_{s,u}^4 - 310^4)\text{ K}^4$$

Copy the following line and paste on a blank EES screen to solve the above equation:

```
31.2*(T_su-1000)/725e-6=0.70*200000-0.70*5.67e-8*(T_su^4-310^4)
```

Solving by EES software, the upper surface temperature of silicon wafer is

$$T_{s,u} = \mathbf{1002\ K}$$

Discussion Excessive temperature difference across the wafer thickness will cause warping in the silicon wafer.

Problem Solving Techniques and EES

1-112C (*a*) Despite the convenience and capability the engineering software packages offer, they are still just tools, and they will not replace the traditional engineering courses. They will simply cause a shift in emphasis in the course material from mathematics to physics. (*b*) They are of great value in engineering practice, however, as engineers today rely on software packages for solving large and complex problems in a short time, and perform optimization studies efficiently.

1-113 We are to determine a positive real root of the following equation using EES: $3.5x^3 - 10x^{0.5} - 3x = -4$.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
3.5*x^3-10*x^0.5-3*x = -4
```

Answer: ***x* = 1.554**

Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

1-114 We are to solve a system of 2 equations and 2 unknowns using EES.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
x^3-y^2=10.5
3*x*y+y=4.6
```

Answers: ***x* = 2.215**, ***y* = 0.6018**

Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

1-115 We are to solve a system of 3 equations with 3 unknowns using EES.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
2*x-y+z=5
3*x^2+2*y=z+2
x*y+2*z=8
```

Answers: ***x* = 1.141, *y* = 0.8159, *z* = 3.535.**

Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

1-116 We are to solve a system of 3 equations with 3 unknowns using EES.

Analysis Using EES software, copy the following lines and paste on a blank EES screen to verify the solution:

```
x^2*y-z=1.5
x-3*y^0.5+x*z=-2
x+y-z=4.2
```

Answers: ***x* = 0.9149, *y* = 10.95, *z* = 7.665**

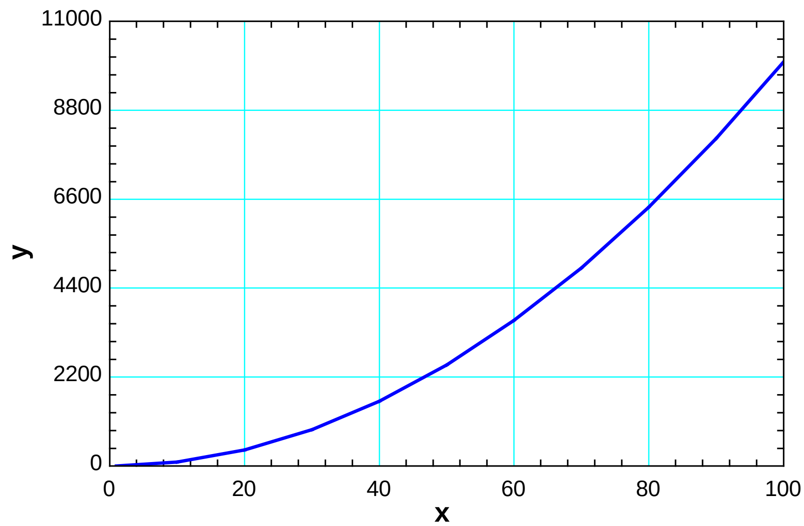
Discussion To obtain the solution in EES, click on the icon that looks like a calculator, or Calculate-Solve.

1-117 The squares of the number from 1 to 100 in increments of 10 are to be evaluated using the parametric table and plot features of EES.

Analysis The problem is solved using EES, and the solution is given below.

```
x=1
y=x^2
```

x	y
1	1
10	100
20	400
30	900
40	1600
50	2500
60	3600
70	4900
80	6400
90	8100
100	10000



Special Topic: Thermal Comfort

1-118C The metabolism refers to the burning of foods such as carbohydrates, fat, and protein in order to perform the necessary bodily functions. The metabolic rate for an average man ranges from 108 W while reading, writing, typing, or listening to a lecture in a classroom in a seated position to 1250 W at age 20 (730 at age 70) during strenuous exercise. The corresponding rates for women are about 30 percent lower. Maximum metabolic rates of trained athletes can exceed 2000 W. We are interested in metabolic rate of the occupants of a building when we deal with heating and air conditioning because the metabolic rate represents the rate at which a body generates heat and dissipates it to the room. This body heat contributes to the heating in winter, but it adds to the cooling load of the building in summer.

1-119C The metabolic rate is proportional to the size of the body, and the metabolic rate of women, in general, is lower than that of men because of their smaller size. Clothing serves as insulation, and the thicker the clothing, the lower the environmental temperature that feels comfortable.

1-120C Asymmetric thermal radiation is caused by the *cold surfaces* of large windows, uninsulated walls, or cold products on one side, and the *warm surfaces* of gas or electric radiant heating panels on the walls or ceiling, solar heated masonry walls or ceilings on the other. Asymmetric radiation causes discomfort by exposing different sides of the body to surfaces at different temperatures and thus to different rates of heat loss or gain by radiation. A person whose left side is exposed to a cold window, for example, will feel like heat is being drained from that side of his or her body.

1-121C (*a*) Draft causes undesired local cooling of the human body by exposing parts of the body to high heat transfer coefficients. (*b*) Direct contact with *cold floor surfaces* causes localized discomfort in the feet by excessive heat loss by conduction, dropping the temperature of the bottom of the feet to uncomfortable levels.

1-122C Stratification is the formation of vertical still air layers in a room at difference temperatures, with highest temperatures occurring near the ceiling. It is likely to occur at places with high ceilings. It causes discomfort by exposing the head and the feet to different temperatures. This effect can be prevented or minimized by using destratification fans (ceiling fans running in reverse).

1-123C It is necessary to ventilate buildings to provide adequate fresh air and to get rid of excess carbon dioxide, contaminants, odors, and humidity. Ventilation increases the energy consumption for heating in winter by replacing the warm indoors air by the colder outdoors air. Ventilation also increases the energy consumption for cooling in summer by replacing the cold indoors air by the warm outdoors air. It is not a good idea to keep the bathroom fans on all the time since they will waste energy by expelling conditioned air (warm in winter and cool in summer) by the unconditioned outdoor air.

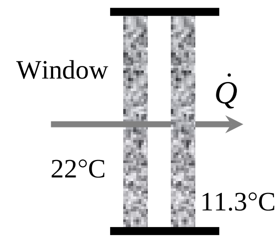
Review Problems

1-124 The windows of a house in Atlanta are of double door type with wood frames and metal spacers. The average rate of heat loss through the windows in winter is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses associated with the infiltration of air through the cracks/openings are not considered.

Analysis The rate of heat transfer through the window can be determined from

$$\dot{Q}_{\text{window, avg}} = U_{\text{overall}} A_{\text{window}} (T_{\text{i}} - T_{\text{o}})$$



where T_i and T_o are the indoor and outdoor air temperatures, respectively, U_{overall} is the *U*-factor (the overall heat transfer coefficient) of the window, and A_{window} is the window area. Substituting,

$$\dot{Q}_{\text{window, avg}} = (2.5\ \text{W/m}^2 \cdot {}^\circ\text{C})(20\ \text{m}^2)(22 - 11.3)^\circ\text{C} = \mathbf{535\ W}$$

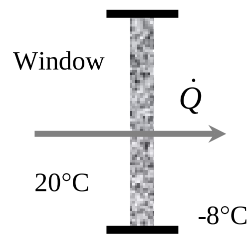
Discussion This is the "average" rate of heat transfer through the window in winter in the absence of any infiltration.

1-125 The range of U-factors for windows are given. The range for the rate of heat loss through the window of a house is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses associated with the infiltration of air through the cracks/openings are not considered.

Analysis The rate of heat transfer through the window can be determined from

$$\dot{Q}_{\text{window}} = U_{\text{overall}} A_{\text{window}} (T_i - T_o)$$



where T_i and T_o are the indoor and outdoor air temperatures, respectively, U_{overall} is the U-factor (the overall heat transfer coefficient) of the window, and A_{window} is the window area. Substituting,

Maximum heat loss: $\dot{Q}_{\text{window, max}} = (6.25\ \text{W/m}^2 \cdot {}^\circ\text{C})(1.2 \times 1.8\ \text{m}^2)[20 - (-8)]^\circ\text{C} = \mathbf{378\ W}$

Minimum heat loss: $\dot{Q}_{\text{window, min}} = (1.25\ \text{W/m}^2 \cdot {}^\circ\text{C})(1.2 \times 1.8\ \text{m}^2)[20 - (-8)]^\circ\text{C} = \mathbf{76\ W}$

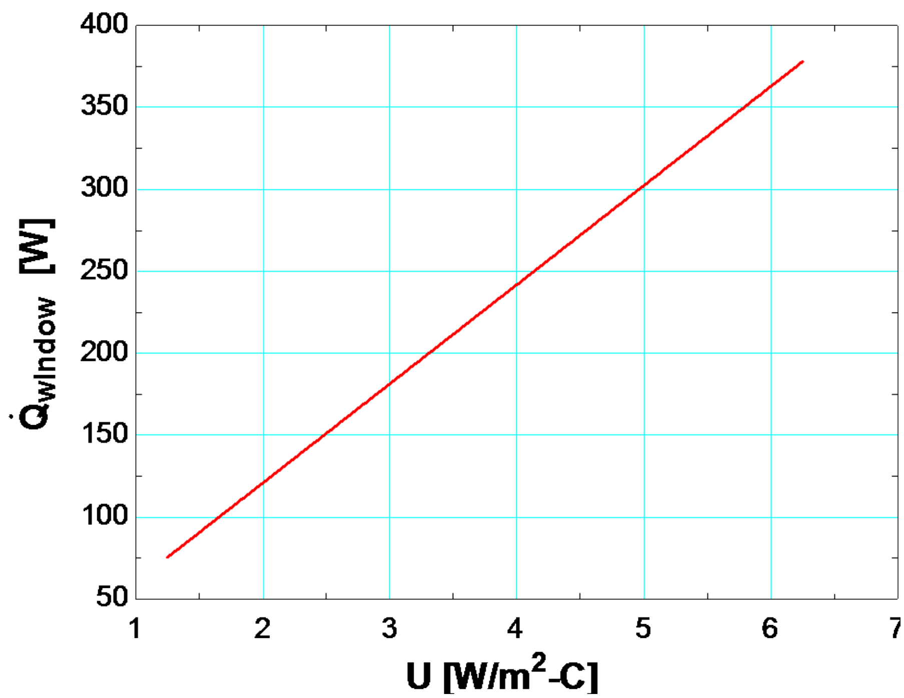
Discussion Note that the rate of heat loss through windows of identical size may differ by a factor of 5, depending on how the windows are constructed.

1-126 Prob. 1-125 is reconsidered. The rate of heat loss through the window as a function of the *U*-factor is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
A=1.2*1.8 [m^2]
T_1=20 [C]
T_2=-8 [C]
U=1.25 [W/m^2-C]

"ANALYSIS"
Q_dot_window=U*A*(T_1-T_2)
```

U [W/m^2.C]	Q_{window} [W]
1.25	75.6
1.75	105.8
2.25	136.1
2.75	166.3
3.25	196.6
3.75	226.8
4.25	257
4.75	287.3
5.25	317.5
5.75	347.8
6.25	378



Review Problems

1-127 A room is to be heated by 1 ton of hot water contained in a tank placed in the room. The minimum initial temperature of the water is to be determined if it to meet the heating requirements of this room for a 24-h period.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 24-h period.

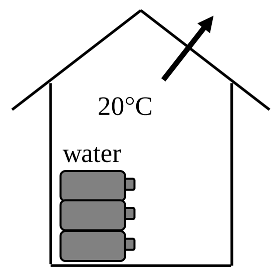
Properties The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-9).

Analysis Heat loss from the room during a 24-h period is

$$Q_{\text{loss}} = (10{,}000 \text{ kJ/h})(24 \text{ h}) = 240{,}000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \rightarrow -Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}^{\nearrow 0}$$



or

$$-Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$-240{,}000 \text{ kJ} = (1000 \text{ kg})(4.18 \text{ kJ/kg·°C})(20 - T_1)$$

It gives

$$T_1 = \mathbf{77.4°C}$$

where T_1 is the temperature of the water when it is first brought into the room.

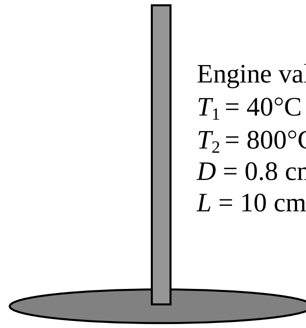
1-128 Engine valves are to be heated in a heat treatment section. The amount of heat transfer, the average rate of heat transfer, the average heat flux, and the number of valves that can be heat treated daily are to be determined.

Assumptions Constant properties given in the problem can be used.

Properties The average specific heat and density of valves are given to be c_p = 440 J/kg.°C and ρ = 7840 kg/m^3.

Analysis (*a*) The amount of heat transferred to the valve is simply the change in its internal energy, and is determined from

$$Q = \Delta U = mc_p(T_2 - T_1)$$
$$= (0.0788\text{ kg})(0.440\text{ kJ/kg}\cdot{}^\circ\text{C})(800-40)^\circ\text{C} = \mathbf{26.35\ kJ}$$



(*b*) The average rate of heat transfer can be determined from

$$\dot{Q}_{\text{avg}} = \frac{Q}{\Delta t} = \frac{26.35\text{ kJ}}{5\times 60\text{ s}} = 0.0878\text{ kW} = \mathbf{87.8\ W}$$

(*c*) The average heat flux is determined from

$$\dot{q}_{\text{ave}} = \frac{\dot{Q}_{\text{avg}}}{A_s} = \frac{\dot{Q}_{\text{avg}}}{2\pi DL} = \frac{87.8\text{ W}}{2\pi(0.008\text{ m})(0.1\text{ m})} = \mathbf{1.75\times 10^4\ W/m^2}$$

(*d*) The number of valves that can be heat treated daily is

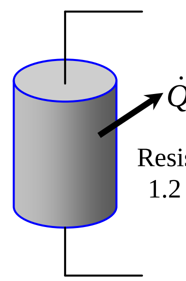
$$\text{Number of valves} = \frac{(10\times 60\text{ min})(25\text{ valves})}{5\text{ min}} = \mathbf{3000\ \ valves}$$

1-129 A cylindrical resistor on a circuit board dissipates 1.2 W of power. The amount of heat dissipated in 24 h, the heat flux, and the fraction of heat dissipated from the top and bottom surfaces are to be determined.

Assumptions Heat is transferred uniformly from all surfaces.

Analysis (*a*) The amount of heat this resistor dissipates during a 24-hour period is

$$Q = \dot{Q}\Delta t = (1.2\text{ W})(24\text{ h}) = \mathbf{28.8\ Wh} = \mathbf{104\ kJ} \quad \text{(since 1 Wh = 3600 Ws = 3.6 kJ)}$$



(*b*) The heat flux on the surface of the resistor is

$$A_s = 2\frac{\pi D^2}{4} + \pi DL = 2\frac{\pi(0.4\text{ cm})^2}{4} + \pi(0.4\text{ cm})(2\text{ cm}) = 0.251 + 2.513 = 2.764\text{ cm}^2$$

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{1.2\text{ W}}{2.764\text{ cm}^2} = \mathbf{0.434\ W/cm^2}$$

(*c*) Assuming the heat transfer coefficient to be uniform, heat transfer is proportional to the surface area. Then the fraction of heat dissipated from the top and bottom surfaces of the resistor becomes

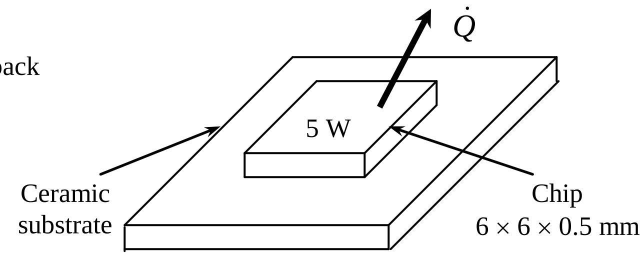
$$\frac{Q_{\text{top-base}}}{Q_{\text{total}}} = \frac{A_{\text{top-base}}}{A_{\text{total}}} = \frac{0.251}{2.764} = \mathbf{0.091} \ \text{ or } (9.1\%)$$

Discussion Heat transfer from the top and bottom surfaces is small relative to that transferred from the side surface.

1-130 The heat generated in the circuitry on the surface of a 5-W silicon chip is conducted to the ceramic substrate. The temperature difference across the chip in steady operation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Thermal properties of the chip are constant.

Properties The thermal conductivity of the silicon chip is given to be k = 130 W/m·°C.



Analysis The temperature difference between the front and back surfaces of the chip is

$$A = (0.006\text{ m})(0.006\text{ m}) = 0.000036\text{ m}^2$$

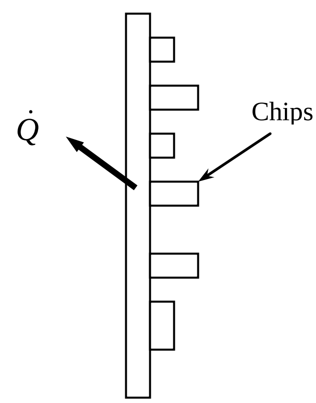
$$\dot{Q} = kA\frac{\Delta T}{L}$$

$$\Delta T = \frac{\dot{Q}L}{kA} = \frac{(5\text{ W})(0.0005\text{ m})}{(130\text{ W/m}\cdot{}^\circ\text{C})(0.000036\text{ m}^2)} = \mathbf{0.53^\circ C}$$

1-131 A circuit board houses 80 closely spaced logic chips on one side, each dissipating 0.06 W. All the heat generated in the chips is conducted across the circuit board. The temperature difference between the two sides of the circuit board is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Thermal properties of the board are constant. **3** All the heat generated in the chips is conducted across the circuit board.

Properties The effective thermal conductivity of the board is given to be k = 16 W/m·°C.



Analysis The total rate of heat dissipated by the chips is

$$\dot{Q} = 80 \times (0.06\text{ W}) = 4.8\text{ W}$$

Then the temperature difference between the front and back surfaces of the board is

$$A = (0.12\text{ m})(0.18\text{ m}) = 0.0216\text{ m}^2$$

$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow \Delta T = \frac{\dot{Q}L}{kA} = \frac{(4.8\text{ W})(0.003\text{ m})}{(16\text{ W/m}\cdot{}^\circ\text{C})(0.0216\text{ m}^2)} = \mathbf{0.042^\circ C}$$

Discussion Note that the circuit board is nearly isothermal.

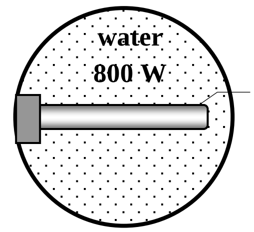
1-132 An electric resistance heating element is immersed in water initially at 20°C. The time it will take for this heater to raise the water temperature to 80°C as well as the convection heat transfer coefficients at the beginning and at the end of the heating process are to be determined.

Assumptions **1** Steady operating conditions exist and thus the rate of heat loss from the wire equals the rate of heat generation in the wire as a result of resistance heating. **2** Thermal properties of water are constant. **3** Heat losses from the water in the tank are negligible.

Properties The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-9).

Analysis When steady operating conditions are reached, we have $\dot{Q} = \dot{E}_{\text{generated}} = 800\text{ W}$. This is also equal to the rate of heat gain by water. Noting that this is the only mechanism of energy transfer, the time it takes to raise the water temperature from 20°C to 80°C is determined to be

$$Q_{\text{in}} = mc(T_2 - T_1)$$
$$\dot{Q}_{\text{in}}\Delta t = mc(T_2 - T_1)$$
$$\Delta t = \frac{mc(T_2 - T_1)}{\dot{Q}_{\text{in}}} = \frac{(75\text{ kg})(4180\text{ J/kg}\cdot\text{°C})(80-20)\text{°C}}{800\text{ J/s}} = 23{,}510\text{ s} = \mathbf{6.53\ h}$$



The surface area of the wire is

$$A_s = \pi DL = \pi(0.005\text{ m})(0.4\text{ m}) = 0.00628\text{ m}^2$$

The Newton's law of cooling for convection heat transfer is expressed as $\dot{Q} = hA_s(T_s - T_\infty)$. Disregarding any heat transfer by radiation and thus assuming all the heat loss from the wire to occur by convection, the convection heat transfer coefficients at the beginning and at the end of the process are determined to be

$$h_1 = \frac{\dot{Q}}{A_s(T_s - T_{\infty 1})} = \frac{800\text{ W}}{(0.00628\text{ m}^2)(120-20)\text{°C}} = \mathbf{1274\ W/m^2\cdot °C}$$
$$h_2 = \frac{\dot{Q}}{A_s(T_s - T_{\infty 2})} = \frac{800\text{ W}}{(0.00628\text{ m}^2)(120-80)\text{°C}} = \mathbf{3185\ W/m^2\cdot °C}$$

Discussion Note that a larger heat transfer coefficient is needed to dissipate heat through a smaller temperature difference for a specified heat transfer rate.

1-133 A standing man is subjected to high winds and thus high convection coefficients. The rate of heat loss from this man by convection in still air at 20°C, in windy air, and the wind chill temperature are to be determined.

Assumptions **1** A standing man can be modeled as a 30-cm diameter, 170-cm long vertical cylinder with both the top and bottom surfaces insulated. **2** The exposed surface temperature of the person and the convection heat transfer coefficient is constant and uniform. **3** Heat loss by radiation is negligible.

Analysis The heat transfer surface area of the person is

$$A_s = \pi DL = \pi(0.3\text{ m})(1.70\text{ m}) = 1.60\text{ m}^2$$

Windy weather

The rate of heat loss from this man by convection in still air is

$$Q_{\text{still air}} = hA_s\Delta T = (15\text{ W/m}^2\cdot°\text{C})(1.60\text{ m}^2)(34 - 20)°\text{C} = \mathbf{336\ W}$$

In windy air it would be

$$Q_{\text{windy air}} = hA_s\Delta T = (30\text{ W/m}^2\cdot°\text{C})(1.60\text{ m}^2)(34 - 20)°\text{C} = \mathbf{672\ W}$$

To lose heat at this rate in still air, the air temperature must be

$$672\text{ W} = (hA_s\Delta T)_{\text{still air}} = (15\text{ W/m}^2\cdot°\text{C})(1.60\text{ m}^2)(34 - T_{\text{effective}})°\text{C}$$

which gives

$$T_{\text{effective}} = \mathbf{6°C}$$

That is, the windy air at 20°C feels as cold as still air at 6°C as a result of the wind-chill effect.

1-134 The surface temperature of an engine block that generates 50 kW of power output is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Temperature inside the engine compartment is uniform. **3** Heat transfer by radiation is not considered.

Analysis With a net engine efficiency of 35%, which means 65% of the generated power output are heat loss by convection:

$$\dot{Q}_{\text{conv}} = \dot{W}_{\text{out}}(1-\eta) = (50\text{ kW})(1-0.35) = 32.5\text{ kW}$$

From Newton's law of cooling, the heat transfer by convection is given as

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

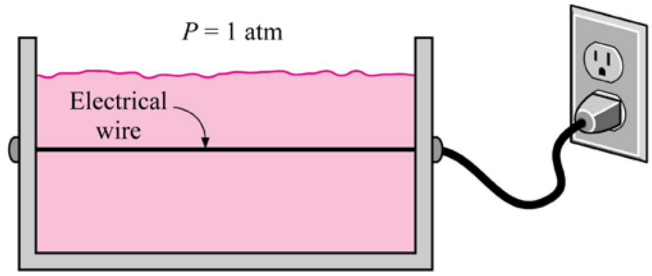
Rearranging, the engine block surface temperature is

$$T_s = \frac{\dot{Q}_{\text{conv}}}{hA_s} + T_\infty = \frac{32.5\times 10^3\text{ W}}{(50\text{ W/m}^2\cdot°\text{C})(0.95\text{ m}^2)} + 157\ °\text{C} = \mathbf{841\ °C}$$

Discussion Due to the complex geometry of the engine block, hot spots are likely to occur with temperatures much higher than 841 °C.

1-135 Electric power required to maintain the surface temperature of an electrical wire submerged in boiling water at 115ºC.

Assumptions **1** Steady operating conditions exist. **2** Convection heat transfer coefficient is uniform. **3** Heat transfer by radiation is negligible. **4** Heat losses from the boiler are negligible.



Analysis From an overall energy balance on the electrical wire, the power dissipated by the wire is transferred by convection to the water. Using Newton's law of cooling,

$$\dot{Q}_{conv} = h\,A_s\,(T_s - T_\infty)$$

where the surface area of the wire is

$$A_s = \pi DL = \pi(1 \times 10^{-3})\,(15 \times 10^{-2}) = 4.712 \times 10^{-4}\ \mathrm{m^2}$$

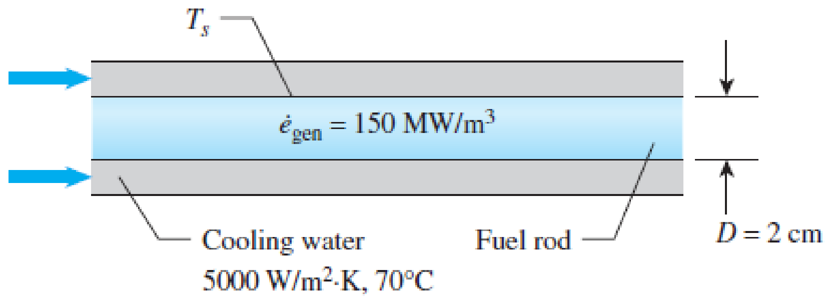
The heat transfer by convection with h = 51,250 W/m^2·K (average of the upper and lower values of the convection heat transfer coefficients given in Table 1-5) is

$$\dot{Q}_{conv} = h\,A_s\,(T_s - T_\infty) = (51{,}250\ \mathrm{W/m^2{\cdot}K})\,(4.712 \times 10^{-4}\ \mathrm{m^2})\,(115 - 100)\ \mathrm{K} = \mathbf{362.2\ W}$$

This is equal to the electric power that must be supplied to maintain the surface temperature of the wire at 115ºC.

Discussion If we had used the two extreme values of the convection heat transfer coefficient for boiling and condensation given in Table 1-5, the required electric power would range from 17.7 W (for h = 2500 W/m^2·K) to 706.8 W (for h = 100,000 W/m^2·K).

1-136 A cylindrical fuel rod is cooled by water flowing through its encased concentric tube. The surface temperature of the fuel rod is to be determined.



Assumptions **1** Steady operating conditions exist. **2** Heat generation in the fuel rod is uniform.

Analysis The total heat transfer area of the fuel rod is

$$A_s = \pi DL$$

The heat transfer rate from the fuel rod is equal to the rate of heat generation multiplied by the fuel rod volume

$$\dot{Q} = \dot{e}_{\text{gen}}(\pi D^2 L/4)$$

The rate of heat removal from the fuel rod by the cooling water is

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

Thus, $\quad \dot{e}_{\text{gen}}(\pi D^2 L/4) = h\pi DL(T_s - T_\infty)$

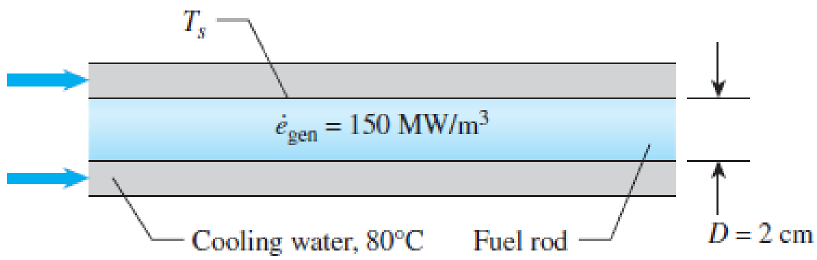
The surface temperature of the fuel rod is

$$T_s = T_\infty + \frac{D\dot{e}_{\text{gen}}}{4h}$$

$$T_s = 70°\text{C} + \frac{(0.02\text{ m})(150\times 10^6\text{ W/m}^3)}{4(5000\text{ W/m}^2\cdot\text{K})} = \mathbf{220°C}$$

Discussion A convection heat transfer coefficient of 5000 W/m^2·K for forced convection of liquid is reasonable when compared with the values listed in Table 1-5.

1-137 A cylindrical fuel rod is cooled by water flowing through its encased concentric tube. The surface temperature of the fuel rod must be maintained below 300°C and the convection heat transfer coefficient is to be determined.



Assumptions **1** Steady operating conditions exist. **2** Heat generation in the fuel rod is uniform.

Analysis The total heat transfer area of the fuel rod is

$$A_s = \pi DL$$

The heat transfer rate from the fuel rod is equal to the rate of heat generation multiply the fuel rod volume

$$\dot{Q} = \dot{e}_{gen}(\pi D^2 L/4)$$

The heat transfer rate removed from the fuel rod by the cooling water is

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

Thus,

$$\dot{e}_{gen}(\pi D^2 L/4) = h\pi DL(T_s - T_\infty)$$

The convection heat transfer coefficient can be determined as

$$\begin{aligned} h &= \frac{D\dot{e}_{gen}}{4(T_s - T_\infty)} \\ &= \frac{(0.02\text{ m})(150\times 10^6\text{ W/m}^3)}{4(300-80)\text{ K}} \\ &= 3410\text{ W/m}^2\cdot\text{K} \end{aligned}$$

To maintained the fuel rod surface temperature below 300°C, the convection heat transfer coefficient should be

$$h > \mathbf{3410\ W/m^2\cdot K}$$

Discussion For the fuel rod surface temperature to be kept below 300°C, the cooling water convection heat transfer coefficient should be higher than 3410 W/m^2·K. Higher value of convection heat transfer coefficient can be achieved by increasing the flow rate of cooling water.

1-138 The rate of radiation heat transfer between a person and the surrounding surfaces at specified temperatures is to be determined in summer and in winter.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by convection is not considered. **3** The person is completely surrounded by the interior surfaces of the room. **4** The surrounding surfaces are at a uniform temperature.

Properties The emissivity of a person is given to be ε = 0.95

Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are:

(*a*) Summer: T_{surr} = 23+273=296

$$
\begin{aligned}
\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4) \\
&= (0.95)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.6\ \text{m}^2)[(32+273)^4 - (296\ \text{K})^4]\text{K}^4 \\
&= \mathbf{84.2\ W}
\end{aligned}
$$

(*b*) Winter: T_{surr} = 12+273= 285 K

$$
\begin{aligned}
\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4) \\
&= (0.95)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.6\ \text{m}^2)[(32+273)^4 - (285\ \text{K})^4]\text{K}^4 \\
&= \mathbf{177.2\ W}
\end{aligned}
$$

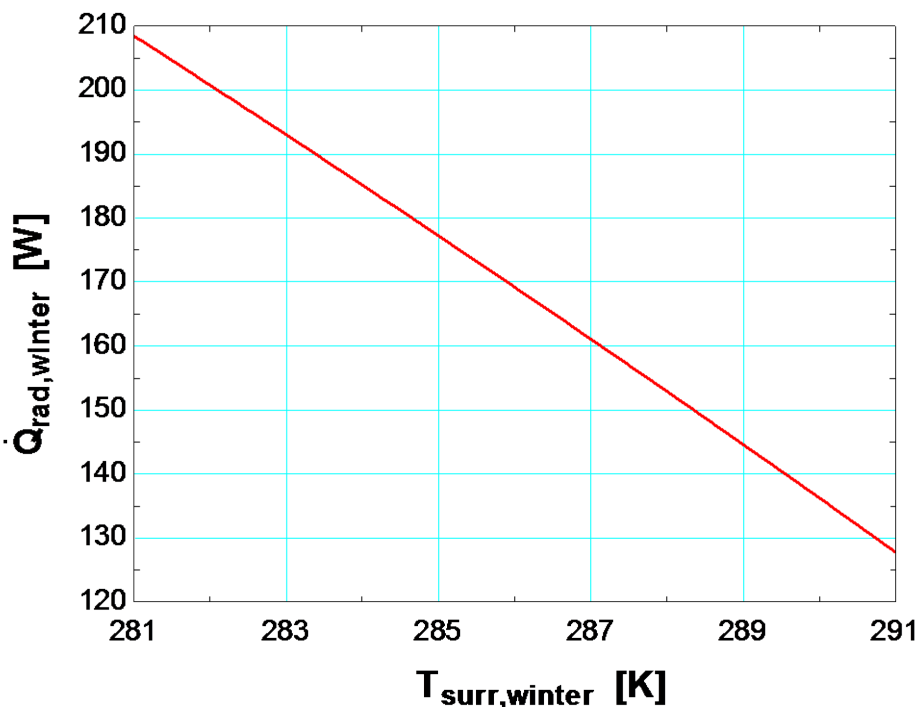
Discussion Note that the radiation heat transfer from the person more than doubles in winter.

1-139  Prob. 1-138is reconsidered. The rate of radiation heat transfer in winter as a function of the temperature of the inner surface of the room is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_infinity=(20+273) [K]
T_surr_winter=(12+273) [K]
T_surr_summer=(23+273) [K]
A=1.6 [m^2]
epsilon=0.95
T_s=(32+273) [K]

"ANALYSIS"
sigma=5.67E-8 [W/m^2-K^4] "Stefan-Boltzman constant"
Q_dot_rad_summer=epsilon*sigma*A*(T_s^4-T_surr_summer^4)
Q_dot_rad_winter=epsilon*sigma*A*(T_s^4-T_surr_winter^4)
```

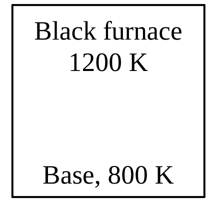
$T_{surr,\ winter}$ [K]	$Q_{rad,\ winter}$ [W]
281	208.5
282	200.8
283	193
284	185.1
285	177.2
286	169.2
287	161.1
288	152.9
289	144.6
290	136.2
291	127.8



1-140 The base surface of a cubical furnace is surrounded by black surfaces at a specified temperature. The net rate of radiation heat transfer to the base surface from the top and side surfaces is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The top and side surfaces of the furnace closely approximate black surfaces. **3** The properties of the surfaces are constant.

Properties The emissivity of the base surface is $\varepsilon = 0.4$.



Analysis The base surface is completely surrounded by the top and side surfaces. Then using the radiation relation for a surface completely surrounded by another large (or black) surface, the net rate of radiation heat transfer from the top and side surfaces to the base is determined to be

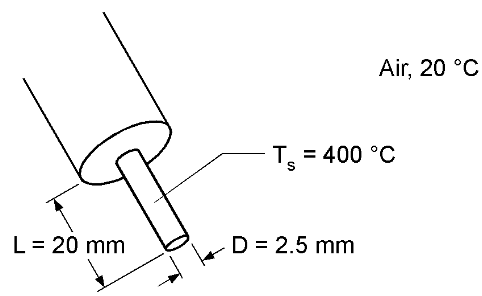
$$\begin{aligned}\dot{Q}_{\text{rad,base}} &= \varepsilon A\sigma(T_{\text{base}}^4 - T_{\text{surr}}^4) \\ &= (0.4)(3\times 3\ \text{m}^2)(5.67\times 10^{-8}\ \text{W/m}^2.\text{K}^4)[(1200\ \text{K})^4 - (800\ \text{K})^4] \\ &= 339{,}660\ \text{W} = \mathbf{340\ kW}\end{aligned}$$

1-141 The power required to maintain the soldering iron tip at 400 °C is to be determined.

Assumptions **1** Steady operating conditions exist since the tip surface and the surrounding air temperatures remain constant. **2** The thermal properties of the tip and the convection heat transfer coefficient are constant and uniform. **3** The surrounding surfaces are at the same temperature as the air.

Properties The emissivity of the tip is given to be 0.80.

Analysis The total heat transfer area of the soldering iron tip is



$$\begin{aligned}A_s &= \pi D^2/4 + \pi DL \\ &= \pi(0.0025\ \text{m})^2/4 + \pi(0.0025\ \text{m})(0.02\ \text{m}) \\ &= 1.62\times 10^{-4}\ \text{m}^2\end{aligned}$$

The rate of heat transfer by convection is

$$\begin{aligned}\dot{Q}_{\text{conv}} &= hA_s(T_{\text{tip}} - T_\infty) \\ &= (25\ \text{W/m}^2\cdot{}^\circ\text{C})(1.62\times 10^{-4}\ \text{m}^2)(400-20)\ {}^\circ\text{C} \\ &= 1.54\ \text{W}\end{aligned}$$

The rate of heat transfer by radiation is

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s(T_{\text{tip}}^4 - T_{\text{surr}}^4) \\ &= (0.80)(5.67\times 10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(1.62\times 10^{-4}\ \text{m}^2)[(400+273)^4 - (20+273)^4]\ \text{K}^4 \\ &= 1.45\ \text{W}\end{aligned}$$

Thus, the power required is equal to the total rate of heat transfer from the tip by both convection and radiation:

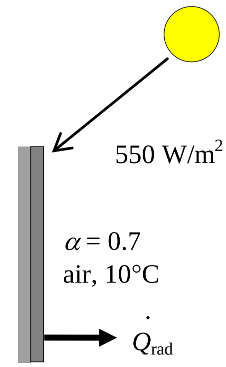
$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 1.54\ \text{W} + 1.45\ \text{W} = \mathbf{2.99\ W}$$

Discussion If the soldering iron tip is highly polished with an emissivity of 0.05, the power required to maintain the tip at 400 °C will reduce to 1.63 W, or by 45.5%.

1-142 The backside of the thin metal plate is insulated and the front side is exposed to solar radiation. The surface temperature of the plate is to be determined when it stabilizes.

Assumptions 1 Steady operating conditions exist. **2** Heat transfer through the insulated side of the plate is negligible. **3** The heat transfer coefficient is constant and uniform over the plate. **4** Radiation heat transfer is negligible.

Properties The solar absorptivity of the plate is given to be $\alpha = 0.7$.



Analysis When the heat loss from the plate by convection equals the solar radiation absorbed, the surface temperature of the plate can be determined from

$$\dot{Q}_{\text{solarabsorbed}} = \dot{Q}_{\text{conv}}$$
$$\alpha\dot{Q}_{\text{solar}} = hA_s(T_s - T_o)$$
$$0.7 \times A \times 550\ \text{W/m}^2 = (25\ \text{W/m}^2\cdot{}^\circ\text{C})A_s(T_s - 10)$$

Canceling the surface area A_s and solving for T_s gives

$$T_s = \mathbf{25.4^\circ C}$$

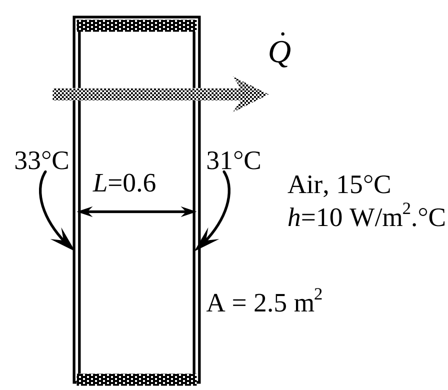
1-143 The glass cover of a flat plate solar collector with specified inner and outer surface temperatures is considered. The fraction of heat lost from the glass cover by radiation is to be determined.

Assumptions **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Thermal properties of the glass are constant.

Properties The thermal conductivity of the glass is given to be $k = 0.7$ W/m·°C.

Analysis Under steady conditions, the rate of heat transfer through the glass by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.7\ \text{W/m}\cdot{}^\circ\text{C})(2.5\ \text{m}^2)\frac{(33-31)^\circ\text{C}}{0.006\ \text{m}} = 583\ \text{W}$$



The rate of heat transfer from the glass by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T = (10\ \text{W/m}^2\cdot{}^\circ\text{C})(2.5\ \text{m}^2)(31-15)^\circ\text{C} = 400\ \text{W}$$

Under steady conditions, the heat transferred through the cover by conduction should be transferred from the outer surface by convection and radiation. That is,

$$\dot{Q}_{\text{rad}} = \dot{Q}_{\text{cond}} - \dot{Q}_{\text{conv}} = 583 - 400 = 183\ \text{W}$$

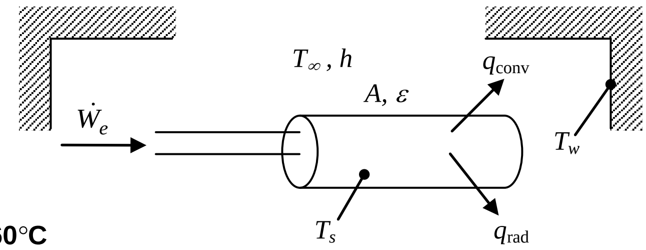
Then the fraction of heat transferred by radiation becomes

$$f = \frac{\dot{Q}_{\text{rad}}}{\dot{Q}_{\text{cond}}} = \frac{183}{583} = \mathbf{0.314} \quad \text{(or 31.4\%)}$$

1-144 An electric heater placed in a room consumes 500 W power when its surfaces are at 120°C. The surface temperature when the heater consumes 700 W is to be determined without and with the consideration of radiation.

Assumptions **1** Steady operating conditions exist. **2** The temperature is uniform over the surface.

Analysis (*a*) Neglecting radiation, the convection heat transfer coefficient is determined from



$$h = \frac{\dot{Q}}{A(T_s - T_\infty)} = \frac{500\text{ W}}{(0.25\text{ m}^2)(120-20)^\circ\text{C}} = 20\text{ W/m}^2\cdot{}^\circ\text{C}$$

The surface temperature when the heater consumes 700 W is

$$T_s = T_\infty + \frac{\dot{Q}}{hA} = 20^\circ\text{C} + \frac{700\text{ W}}{(20\text{ W/m}^2\cdot{}^\circ\text{C})(0.25\text{ m}^2)} = \mathbf{160^\circ C}$$

(*b*) Considering radiation, the convection heat transfer coefficient is determined from

$$\begin{aligned} h &= \frac{\dot{Q} - \varepsilon A\sigma(T_s^4 - T_{surr}^4)}{A(T_s - T_\infty)} \\ &= \frac{500\text{ W} - (0.75)(0.25\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)\left[(393\text{ K})^4 - (283\text{ K})^4\right]}{(0.25\text{ m}^2)(120-20)^\circ\text{C}} \\ &= 12.58\text{ W/m}^2\cdot{}^\circ\text{C} \end{aligned}$$

Then the surface temperature becomes

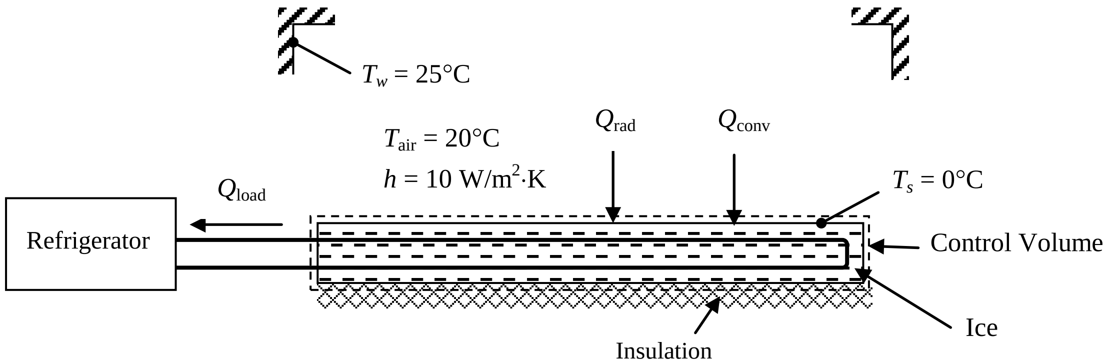
$$\begin{aligned} \dot{Q} &= hA(T_s - T_\infty) + \varepsilon A\sigma(T_s^4 - T_{surr}^4) \\ 700 &= (12.58)(0.25)(T_s - 293) + (0.75)(0.25)(5.67\times10^{-8})\left[T_s^4 - (283\text{ K})^4\right] \\ T_s &= 425.9\text{ K} = \mathbf{152.9^\circ C} \end{aligned}$$

Discussion Neglecting radiation changed T_s by more than 7°C, so assumption is not correct in this case.

1-145 An ice skating rink is located in a room is considered. The refrigeration load of the system and the time it takes to melt 3 mm of ice are to be determined.

Assumptions **1** Steady operating conditions exist in part (a). **2** The surface is insulated on the back side in part (b).

Properties The heat of fusion and the density of ice are given to be 333.7 kJ/kg and 920 kg/m^3, respectively.



Analysis (*a*) The refrigeration load is determined from

$$\dot{Q}_{load} = hA(T_{air} - T_s) + \varepsilon A\sigma(T_w^4 - T_s^4)$$
$$= (10)(40\times12)(20-0) + (0.95)(40\times12)(5.67\times10^{-8})\left[298^4 - 273^4\right] = \mathbf{156{,}300\ W}$$

(*b*) The time it takes to melt 3 mm of ice is determined from

$$t = \frac{LW\delta\rho h_{if}}{\dot{Q}_{load}} = \frac{(40\times12\ \text{m}^2)(0.003\ \text{m})(920\ \text{kg/m}^3)(333.7\times10^3\ \text{J/kg})}{156{,}300\ \text{J/s}} = 2831\ \text{s} = \mathbf{47.2\ min}$$

Fundamentals of Engineering (FE) Exam Problems

1-146 A 2-kW electric resistance heater in a room is turned on and kept on for 50 minutes. The amount of energy transferred to the room by the heater is

(a) 2 kJ (b) 100 kJ (c) 6000 kJ (d) 7200 kJ (e) 12,000 kJ

Answer (c) 6000 kJ

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
We= 2 [kJ/s]
time=50*60 [s]
We_total=We*time [kJ]
"Wrong Solutions:"
W1_Etotal=We*time/60 "using minutes instead of s"
W2_Etotal=We "ignoring time"
```

1-147 A 2-kW electric resistance heater submerged in 30-kg water is turned on and kept on for 10 min. During the process, 500 kJ of heat is lost from the water. The temperature rise of water is

(a) 5.6°C (b) 9.6°C (c) 13.6°C (d) 23.3°C (e) 42.5°C

Answer (a) 5.6°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
C=4.18 [kJ/kg-K]
m=30 [kg]
Q_loss=500 [kJ]
time=10*60 [s]
W_e=2 [kJ/s]
"Applying energy balance E_in-E_out=dE_system gives"
time*W_e-Q_loss = dU_system
dU_system=m*C*DELTAT
“Some Wrong Solutions with Common Mistakes:”
time*W_e = m*C*W1_T "Ignoring heat loss"
time*W_e+Q_loss = m*C*W2_T  "Adding heat loss instead of subtracting"
time*W_e-Q_loss = m*1.0*W3_T "Using specific heat of air or not using specific heat"
```

1-148 Eggs with a mass of 0.15 kg per egg and a specific heat of 3.32 kJ/kg·°C are cooled from 32°C to 10°C at a rate of 200 eggs per minute. The rate of heat removal from the eggs is

(a) 7.3 kW (b) 53 kW (c) 17 kW (d) 438 kW (e) 37 kW

Answer (e) 37 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
C=3.32 [kJ/kg-K]
m_egg=0.15 [kg]
T1=32 [C]
T2=10 [C]
n=200 "eggs/min"
m=n*m_egg/60 "kg/s"
"Applying energy balance E_in-E_out=dE_system gives"
"-E_out = dU_system"
Qout=m*C*(T1-T2) "kJ/s"
“Some Wrong Solutions with Common Mistakes:”
W1_Qout = m*C*T1  "Using T1 only"
W2_Qout = m_egg*C*(T1-T2)  "Using one egg only"
W3_Qout = m*C*T2  "Using T2 only"
W4_Qout=m_egg*C*(T1-T2)*60   "Finding kJ/min"
```

1-149 A cold bottled drink (m = 2.5 kg, c_p = 4200 J/kg·°C) at 5°C is left on a table in a room. The average temperature of the drink is observed to rise to 15°C in 30 minutes. The average rate of heat transfer to the drink is

(a) 23 W (b) 29 W (c) 58 W (d) 88 W (e) 122 W

Answer: (c) 58 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
c=4200 [J/kg-K]
m=2.5 [kg]
T1=5 [C]
T2=15 [C]
time = 30*60 [s]
"Applying energy balance E_in-E_out=dE_system gives"
Q=m*c*(T2-T1)
Qave=Q/time
“Some Wrong Solutions with Common Mistakes:”
W1_Qave = m*c*T1/time  "Using T1 only"
W2_Qave = c*(T2-T1)/time  "Not using mass"
W3_Qave = m*c*T2/time  "Using T2 only"
```

1-150 Water enters a pipe at 20ºC at a rate of 0.50 kg/s and is heated to 60ºC. The rate of heat transfer to the water is

(a) 20 kW (b) 42 kW (c) 84 kW (d) 126 kW (e) 334 kW

Answer (c) 84 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_in=20 [C]
T_out=60 [C]
m_dot=0.50 [kg/s]
c_p=4.18 [kJ/kg-C]
Q_dot=m_dot*c_p*(T_out-T_in)
"Some Wrong Solutions with Common Mistakes"
W1_Q_dot=m_dot*(T_out-T_in) "Not using specific heat"
W2_Q_dot=c_p*(T_out-T_in) "Not using mass flow rate"
W3_Q_dot=m_dot*c_p*T_out "Using exit temperature instead of temperature change"
```

1-151 Air enters a 12-m-long, 7-cm-diameter pipe at 50ºC at a rate of 0.06 kg/s. The air is cooled at an average rate of 400 W per m^2 surface area of the pipe. The air temperature at the exit of the pipe is

(a) 4.3ºC (b) 17.5ºC (c) 32.5ºC (d) 43.4ºC (e) 45.8ºC

Answer (c) 32.5ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
L=12 [m]
D=0.07 [m]
T1=50 [C]
m_dot=0.06 [kg/s]
q=400 [W/m^2]
A=pi*D*L
Q_dot=q*A
c_p=1007 [J/kg-C] "Table A-15"
Q_dot=m_dot*c_p*(T1-T2)
"Some Wrong Solutions with Common Mistakes"
q=m_dot*c_p*(T1-W1_T2) "Using heat flux, q instead of rate of heat transfer, Q_dot"
Q_dot=m_dot*4180*(T1-W2_T2) "Using specific heat of water"
Q_dot=m_dot*c_p*W3_T2 "Using exit temperature instead of temperature change"
```

1-152 Heat is lost steadily through a 0.5-cm thick 2 m × 3 m window glass whose thermal conductivity is 0.7 W/m·°C. The inner and outer surface temperatures of the glass are measured to be 12°C to 9°C. The rate of heat loss by conduction through the glass is

(a) 420 W (b) 5040 W (c) 17,600 W (d) 1256 W (e) 2520 W

Answer (e) 2520 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
A=3*2 [m^2]
L=0.005 [m]
T1=12 [C]
T2=9 [C]
k=0.7 [W/m-C]
Q=k*A*(T1-T2)/L
"Some Wrong Solutions with Common Mistakes:"
W1_Q=k*(T1-T2)/L "Not using area"
W2_Q=k*2*A*(T1-T2)/L  "Using areas of both surfaces"
W3_Q=k*A*(T1+T2)/L  "Adding temperatures instead of subtracting"
W4_Q=k*A*L*(T1-T2) "Multiplying by thickness instead of dividing by it"
```

1-153 Steady heat conduction occurs through a 0.3-m thick 9 m by 3 m composite wall at a rate of 1.2 kW. If the inner and outer surface temperatures of the wall are 15°C and 7°C, the effective thermal conductivity of the wall is

(a) 0.61 W/m·°C (b) 0.83 W/m·°C (c) 1.7 W/m·°C (d) 2.2 W/m·°C (e) 5.1 W/m·°C

Answer (c) 1.7 W/m·°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
A=9*3 [m^2]
L=0.3 [m]
T1=15 [C]
T2=7 [C]
Q=1200 [W]
Q=k*A*(T1-T2)/L
"Wrong Solutions:"
Q=W1_k*(T1-T2)/L "Not using area"
Q=W2_k*2*A*(T1-T2)/L  "Using areas of both surfaces"
Q=W3_k*A*(T1+T2)/L  "Adding temperatures instead of subtracting"
Q=W4_k*A*L*(T1-T2) "Multiplying by thickness instead of dividing by it"
```

1-154 Heat is lost through a brick wall (k = 0.72 W/m·ºC), which is 4 m long, 3 m wide, and 25 cm thick at a rate of 500 W. If the inner surface of the wall is at 22ºC, the temperature at the midplane of the wall is

(a) 0ºC (b) 7.5ºC (c) 11.0ºC (d) 14.8ºC (e) 22ºC

Answer (d) 14.8ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
k=0.72 [W/m-C]
Length=4 [m]
Width=3 [m]
L=0.25 [m]
Q_dot=500 [W]
T1=22 [C]
A=Length*Width
Q_dot=k*A*(T1-T_middle)/(0.5*L)
"Some Wrong Solutions with Common Mistakes"
Q_dot=k*A*(T1-W1_T_middle)/L "Using L instead of 0.5L"
W2_T_middle=T1/2 "Just taking the half of the given temperature"
```

1-155 A 10-cm high and 20-cm wide circuit board houses on its surface 100 closely spaced chips, each generating heat at a rate of 0.12 W and transferring it by convection and radiation to the surrounding medium at 40°C. Heat transfer from the back surface of the board is negligible. If the combined convection and radiation heat transfer coefficient on the surface of the board is 22 W/m^2·°C, the average surface temperature of the chips is

(a) 41°C (b) 54°C (c) 67°C (d) 76°C (e) 82°C

Answer (c) 67°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
A=0.1*0.2 [m^2]
Q= 100*0.12 [W]
Tair=40 [C]
h=22 [W/m^2-C]
Q= h*A*(Ts-Tair)
 "Wrong Solutions:"
Q= h*(W1_Ts-Tair) "Not using area"
Q= h*2*A*(W2_Ts-Tair) "Using both sides of surfaces"
Q= h*A*(W3_Ts+Tair) "Adding temperatures instead of subtracting"
Q/100= h*A*(W4_Ts-Tair) "Considering 1 chip only"
```

1-156 A 40-cm-long, 0.4-cm-diameter electric resistance wire submerged in water is used to determine the convection heat transfer coefficient in water during boiling at 1 atm pressure. The surface temperature of the wire is measured to be 114°C when a wattmeter indicates the electric power consumption to be 7.6 kW. The heat transfer coefficient is

(a) 108 kW/m^2·°C (b) 13.3 kW/m^2·°C (c) 68.1 kW/m^2·°C (d) 0.76 kW/m^2·°C (e) 256 kW/m^2·°C

Answer (a) 108 kW/m^2·°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
L=0.4 [m]
D=0.004 [m]
A=pi*D*L [m^2]
We=7.6 [kW]
Ts=114 [C]
Tf=100 [C]  "Boiling temperature of water at 1 atm"
We= h*A*(Ts-Tf)
"Wrong Solutions:"
We= W1_h*(Ts-Tf) "Not using area"
We= W2_h*(L*pi*D^2/4)*(Ts-Tf) "Using volume instead of area"
We= W3_h*A*Ts "Using Ts instead of temp difference"
```

1-157 While driving down a highway early in the evening, the air flow over an automobile establishes an overall heat transfer coefficient of 18 W/m^2·K. The passenger cabin of this automobile exposes 9 m^2 of surface to the moving ambient air. On a day when the ambient temperature is 33°C, how much cooling must the air conditioning system supply to maintain a temperature of 20°C in the passenger cabin?

(a) 670 W (b) 1284 W (c) 2106 W (d) 2565 W (e) 3210 W

Answer (c) 2106 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
h=18 [W/m^2-C]
A=9 [m^2]
T_1=33 [C]
T_2=20 [C]
Q=h*A*(T_2-T_1)
```

1-158 Over 90 percent of the energy dissipated by an incandescent light bulb is in the form of heat, not light. What is the temperature of a vacuum-enclosed tungsten filament with an exposed surface area of 2.03 cm^2 in a 100 W incandescent light bulb? The emissivity of tungsten at the anticipated high temperatures is about 0.35. Note that the light bulb consumes 100 W of electrical energy, and dissipates all of it by radiation.

(a) 1870 K (b) 2230 K (c) 2640 K (d) 3120 K (e) 2980 K

Answer (b) 2230 K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
e =0.35
Q=100 [W]
A=2.03E-4 [m^2]
Q=e*A*sigma#*T^4
```

1-159 On a still clear night, the sky appears to be a blackbody with an equivalent temperature of 250 K. What is the air temperature when a strawberry field cools to 0°C and freezes if the heat transfer coefficient between the plants and the air is 6 $W/m^2\cdot{}^oC$ because of a light breeze and the plants have an emissivity of 0.9?

(a) 14°C (b) 7°C (c) 3°C (d) 0°C (e) –3°C

Answer (a) 14°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
e=0.9
h=6 [W/m^2-K]
T_1=273 [K]
T_2=250 [K]
h*(T-T_1)=e*sigma#*(T_1^4-T_2^4)
```

1-160 A 25-cm diameter black ball at 130°C is suspended in air, and is losing heat to the surrounding air at 25°C by convection with a heat transfer coefficient of 12 W/m^2·°C, and by radiation to the surrounding surfaces at 15°C. The total rate of heat transfer from the black ball is

(a) 217 W (b) 247 W (c) 251 W (d) 465 W (e) 2365 W

Answer: (d) 465 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
sigma=5.67E-8 [W/m^2-K^4]
eps=1
D=0.25 [m]
A=pi*D^2
h_conv=12 [W/m^2-C]
Ts=130 [C]
Tf=25 [C]
Tsurr=15 [C]
Q_conv=h_conv*A*(Ts-Tf)
Q_rad=eps*sigma*A*((Ts+273)^4-(Tsurr+273)^4)
Q_total=Q_conv+Q_rad
"Wrong Solutions:"
W1_Ql=Q_conv "Ignoring radiation"
W2_Q=Q_rad "ignoring convection"
W3_Q=Q_conv+eps*sigma*A*(Ts^4-Tsurr^4) "Using C in radiation calculations"
W4_Q=Q_total/A "not using area"
```

1-161 A 3-m^2 black surface at 140°C is losing heat to the surrounding air at 35°C by convection with a heat transfer coefficient of 16 W/m^2·°C, and by radiation to the surrounding surfaces at 15°C. The total rate of heat loss from the surface is

(a) 5105 W (b) 2940 W (c) 3779 W (d) 8819 W (e) 5040 W

Answer (d) 8819 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
sigma=5.67E-8 [W/m^2-K^4]
eps=1
A=3 [m^2]
h_conv=16 [W/m^2-C]
Ts=140 [C]
Tf=35 [C]
Tsurr=15 [C]
Q_conv=h_conv*A*(Ts-Tf)
Q_rad=eps*sigma*A*((Ts+273)^4-(Tsurr+273)^4)
Q_total=Q_conv+Q_rad
“Some Wrong Solutions with Common Mistakes:”
W1_Ql=Q_conv "Ignoring radiation"
W2_Q=Q_rad "ignoring convection"
W3_Q=Q_conv+eps*sigma*A*(Ts^4-Tsurr^4) "Using C in radiation calculations"
W4_Q=Q_total/A "not using area"
```

1-162 A person's head can be approximated as a 25-cm diameter sphere at 35°C with an emissivity of 0.95. Heat is lost from the head to the surrounding air at 25°C by convection with a heat transfer coefficient of 11 W/m^2·°C, and by radiation to the surrounding surfaces at 10°C. Disregarding the neck, determine the total rate of heat loss from the head.

(a) 22 W (b) 27 W (c) 49 W (d) 172 W (e) 249 W

Answer: (c) 49 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
sigma=5.67E-8 [W/m^2-K^4]
eps=0.95
D=0.25 [m]
A=pi*D^2
h_conv=11 [W/m^2-C]
Ts=35 [C]
Tf=25 [C]
Tsurr=10 [C]
Q_conv=h_conv*A*(Ts-Tf)
Q_rad=eps*sigma*A*((Ts+273)^4-(Tsurr+273)^4)
Q_total=Q_conv+Q_rad
"Wrong Solutions:"
W1_Ql=Q_conv "Ignoring radiation"
W2_Q=Q_rad "ignoring convection"
W3_Q=Q_conv+eps*sigma*A*(Ts^4-Tsurr^4) "Using C in radiation calculations"
W4_Q=Q_total/A "not using area"
```

1-163 A room is heated by a 1.2 kW electric resistance heater whose wires have a diameter of 4 mm and a total length of 3.4 m. The air in the room is at 23ºC and the interior surfaces of the room are at 17ºC. The convection heat transfer coefficient on the surface of the wires is 8 W/m^2·ºC. If the rates of heat transfer from the wires to the room by convection and by radiation are equal, the surface temperature of the wires is

(a) 3534ºC (b) 1778ºC (c) 1772ºC (d) 98ºC (e) 25ºC

Answer (b) 1778ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
D=0.004 [m]
L=3.4 [m]
W_dot_e=1200 [W]
T_infinity=23 [C]
T_surr=17 [C]
h=8 [W/m^2-C]
A=pi*D*L
Q_dot_conv=W_dot_e/2
Q_dot_conv=h*A*(T_s-T_infinity)
"Some Wrong Solutions with Common Mistakes"
Q_dot_conv=h*A*(W1_T_s-T_surr) "Using T_surr instead of T_infinity"
Q_dot_conv/1000=h*A*(W2_T_s-T_infinity) "Using kW unit for the rate of heat transfer"
Q_dot_conv=h*(W3_T_s-T_infinity) "Not using surface area of the wires"
W_dot_e=h*A*(W4_T_s-T_infinity) "Using total heat transfer"
```

1-164 A person standing in a room loses heat to the air in the room by convection and to the surrounding surfaces by radiation. Both the air in the room and the surrounding surfaces are at 20ºC. The exposed surfaces of the person is 1.5 m^2 and has an average temperature of 32ºC, and an emissivity of 0.90. If the rates of heat transfer from the person by convection and by radiation are equal, the combined heat transfer coefficient is

(a) 0.008 $W/m^2·ºC$ (b) 3.0 $W/m^2·ºC$ (c) 5.5 $W/m^2·ºC$ (d) 8.3 $W/m^2·ºC$ (e) 10.9 $W/m^2·ºC$

Answer (e) 10.9 $W/m^2·ºC$

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=20 [C]
T_surr=20 [C]
T_s=32 [C]
A=1.5 [m^2]
epsilon=0.90
sigma=5.67E-8 [W/m^2-K^4]
Q_dot_rad=epsilon*A*sigma*((T_s+273)^4-(T_surr+273)^4)
Q_dot_total=2*Q_dot_rad
Q_dot_total=h_combined*A*(T_s-T_infinity)
"Some Wrong Solutions with Common Mistakes"
Q_dot_rad=W1_h_combined*A*(T_s-T_infinity) "Using radiation heat transfer instead of total heat transfer"
Q_dot_rad_1=epsilon*A*sigma*(T_s^4-T_surr^4) "Using C unit for temperature in radiation calculation"
2*Q_dot_rad_1=W2_h_combined*A*(T_s-T_infinity)
```

1-165 . . . 1-168 Design and Essay Problems