Solutions Manual for

Heat and Mass Transfer: Fundamentals & Applications

6th Edition in SI Units

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Chapter 1
INTRODUCTION AND BASIC CONCEPTS

Thermodynamics and Heat Transfer

1-1C (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.

1-2C 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*.

1-3C 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.

1-4C 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.

1-5C 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.

1-6C Warmer. Because energy is added to the room air in the form of electrical work.

1-7C 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.

1-8C The claim is false. The heater of a house supplies the energy that the house is losing, which is proportional to the temperature difference between the indoors and the outdoors. A turned off heater consumes no energy. The heat lost from a house to the outdoors during the warming up period is less than the heat lost from a house that is already at the temperature that the thermostat is set because of the larger cumulative temperature difference in the latter case. For best practice, the heater should be turned off when no one is at home during day (at subfreezing temperatures, the heater should be kept on at a low temperature to avoid freezing of water in pipes). Also, the thermostat should be lowered during bedtime to minimize the temperature difference between the indoors and the outdoors at night and thus the amount of heat that the heater needs to supply to the house.

1-9C No. The thermostat tells an airconditioner (or heater) at what interior temperature to stop. The air conditioner will cool the house at the same rate no matter what the thermostat setting is. So, it is best to set the thermostat at a comfortable temperature and then leave it alone. Setting the thermostat too low a home owner risks wasting energy and money (and comfort) by forgetting it at the set low temperature.

1-10C No. Since there is no temperature drop of water, the heater will never kick into make up for the heat loss. Therefore, it will not waste any energy during times of no use, and there is no need to use a timer. But if the family were on a time-of-use tariff, it would be possible to save money (but not energy) by turning on the heater when the rate was lowest at night and off during peak periods when rate is the highest.

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 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 $\dot{Q} = \int_A \dot{q} dA$.

1-13C Energy can be transferred by heat and work to a close system. An energy transfer is heat transfer when its driving force is temperature difference.

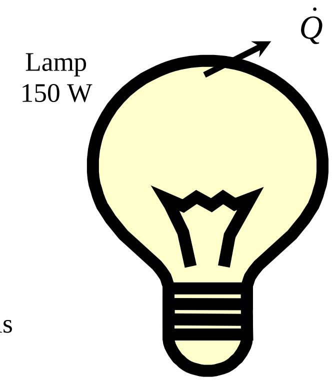
1-14 The filament of a 150 W incandescent lamp is 5 cm long and has a diameter of 0.5 mm. The heat flux on the surface of the filament, the heat flux on the surface of the glass bulb, and the annual electricity cost of the bulb are to be determined.

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

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

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{150\,\text{W}}{0.785\,\text{cm}^2} = 191\,\text{W/cm}^2 = \mathbf{1.91\times10^6\ W/m^2}$$



(*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

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

$$\text{Annual Cost} = (438\,\text{kWh/yr})(\$0.08/\text{kWh}) = \mathbf{\$35.04/yr}$$

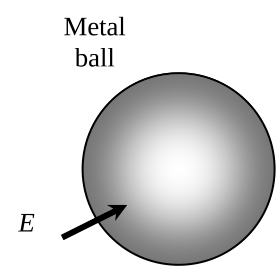
1-15 An aluminum ball is to be heated from 80°C to 200°C. The amount of heat that needs to be transferred to the aluminum ball is to be determined.

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

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

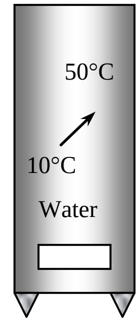
Substituting,

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

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.

1-16 A water heater is initially filled with water at 10°C. The amount of energy that needs to be transferred to the water to raise its temperature to 50°C 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 30ºC from Table A-9 are: ρ = 996 kg/m^3 and c_p = 4178 J/kg·K.

Analysis The mass of water in the tank is

$$m = \rho \boldsymbol{V} = (996\,\text{kg/m}^3)(0.230\,\text{m}^3) = 229.1\,\text{kg}$$

Then, the amount of heat that must be transferred to the water in the tank as it is heated from 10 to 50°C is determined to be

$$Q = mc_p(T_2 - T_1) = (229.1\,\text{kg})(4.178\ \text{kJ/kg}\cdot{}^\circ\text{C})(50 - 10)^\circ\text{C} = \mathbf{38{,}284kJ}$$

Discussion Referring to Table A-9 the density and specific heat of water at 10ºC are: ρ = 999.7 kg/m^3 and c_p = 4194 J/kg·K and at 50ºC are: ρ = 988.1 kg/m^3 and c_p = 4181 J/kg·K. We evaluated the water properties at an average temperature of 30ºC. However, we could have assumed constant properties and evaluated properties at the initial temperature of 10ºC or final temperature of 50ºC without loss of accuracy.

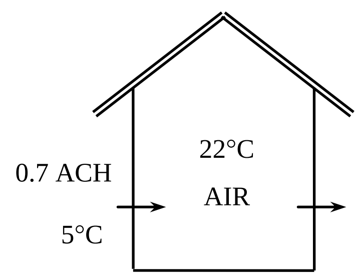
1-17 An electrically heated house maintained at 22°C experiences infiltration losses at a rate of 0.7 ACH. The amount of energy loss from the house due to infiltration per day and its cost are to be determined.

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

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



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

$$\dot{m}_{\text{air}} = \frac{P_o \dot{\boldsymbol{V}}_{\text{air}}}{RT_o} = \frac{P_o(\text{ACH} \times \boldsymbol{V}_{\text{house}})}{RT_o}$$

$$= \frac{(89.6\,\text{kPa})(16.8 \times 600\,\text{m}^3\,/\,\text{day})}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(5 + 273.15\text{K})} = 11{,}314\,\text{kg/day}$$

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

$$\dot{Q}_{\text{infilt}} = \dot{m}_{\text{air}} c_p (T_{\text{indoors}} - T_{\text{outdoors}})$$

$$= (11{,}314\,\text{kg/day})(1.007\,\text{kJ/kg.}^\circ\text{C})(22 - 5)^\circ\text{C} = 193{,}681\,\text{kJ/day} = \mathbf{53.8\,kWh/day}$$

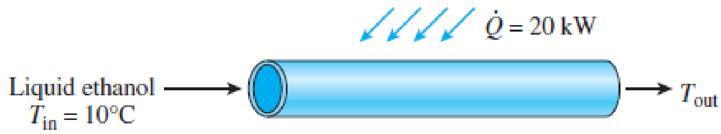
At a unit cost of \$0.082/kWh, the cost of this electrical energy lost by infiltration is

$$\text{Enegy Cost} = (\text{Energy used})(\text{Unit cost of energy}) = (53.8\,\text{kWh/day})(\$0.082/\text{kWh}) = \mathbf{\$4.41/day}$$

1-18 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

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

or

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

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

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

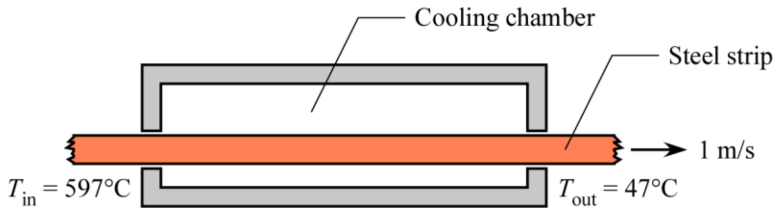
$$\dot{V} > \mathbf{0.00157\ m^3/s}$$

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.

1-19 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

$$\dot{m} = \rho Vwt$$

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

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

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.

1-20 Liquid water is to be heated in an electric teapot. The heating time is to be determined.

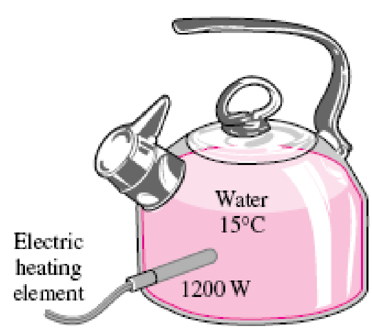
Assumptions **1** Heat loss from the teapot is negligible. **2** Constant properties can be used for both the teapot and the water.

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.

1-21 A water heater uses 100 kW to heat 60 gallon (0.2271 m^3) of liquid water initially at 20°C. Determine the heating duration such that the water exiting the heater would be in compliance with the ASME Boiler and Pressure Vessel Code (ASME BPVC.IV-2015) service restrictions.

Assumptions **1** Heating of the heater material is negligible (i.e. 100 kW is for heating the water only). **2** Constant properties are used for the water. **3** No water flowing out of the heater during the heating.

Properties The average density and specific heat of water are given to be 970 kg/m^3 and 4.18 kJ/kg·K, respectively.

Analysis The mass of the water in the heater is

$$m = \rho \boldsymbol{V} = (970\,\text{kg/m}^3)(0.2271\,\text{m}^3) = 220.29\,\text{kg}$$

The energy balance in this case can be expressed as

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

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

$$E_{\text{in}} = (mc\Delta T)_{\text{water}}$$

In terms of heat transfer rate

$$\dot{Q} = \frac{E_{\text{in}}}{\Delta t} = \frac{(mc\Delta T)_{\text{water}}}{\Delta t}$$

Solving for the heating duration from 20°C to 120°C,

$$\Delta t = \frac{(mc)_{\text{water}}(T_2 - T_1)}{\dot{Q}}$$

$$\Delta t = \frac{(220.29\,\text{kg})(4.18\,\text{kJ/kg}\cdot°\text{C})(100°\text{C})}{100\,\text{kJ/s}} = 920.8\,\text{s} = \mathbf{15.3\,min}$$

Discussion It takes about 15 minutes at 100 kW to heat 60 gallons of liquid water from 20°C to the ASME Boiler and Pressure Vessel Code service restrictions temperature of 120°C. If the heating duration is more than 15.3 minutes, then the final temperature of the water that would exit the heater would be higher than 120°C, which exceeds the code restrictions.

1-22 A boiler (10 kg) is used to heat 20 gallon (0.07571 m^3) of liquid water with 50 kW for 30 minutes. Determine whether this operating condition would be in compliance with the ASME Boiler and Pressure Vessel Code (ASME BPVC.IV-2015) service restrictions.

Assumptions **1** Heat loss from the boiler is negligible. **2** Constant properties are used for both the boiler and the water. **3** The raise in temperature for the boiler and the water is equal. **4** No water flowing out of the boiler during the heating.

Properties The average specific heats are given to be 0.48 kJ/kg·K for the boiler material and 4.18 kJ/kg·K for the water. The average density of the water is 850 kg/m^3.

Analysis We take the boiler and the water in it as a closed system. The energy balance in this case can be expressed as

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

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

$$E_{\text{in}} = (mc\Delta T)_{\text{system}} = (mc\Delta T)_{\text{water}} + (mc\Delta T)_{\text{boiler}}$$

$$E_{\text{in}} = [(mc)_{\text{water}} + (mc)_{\text{boiler}}]\Delta T$$

Solving for the raise in temperature

$$\Delta T = \frac{E_{\text{in}}}{[(\rho \boldsymbol{V} c)_{\text{water}} + (mc)_{\text{boiler}}]}$$

$$\Delta T = \frac{(50\text{ kJ/s})(30 \times 60\text{ s})}{[(850\text{ kg/m}^3)(0.07571\text{m}^3)(4.18\text{ kJ/kg}\cdot°\text{C}) + (10\text{ kg})(0.48\text{ kJ/kg}\cdot°\text{C})]} = 329°\text{C}$$

The final temperature is

$$T_2 = T_1 + \Delta T = 15°\text{C} + 329°\text{C} = \mathbf{344°C} > 120°\text{C}$$

Discussion The final temperature of the water after 30 minutes of heating at 50 kW is 224°C greater than the ASME Boiler and Pressure Vessel Code service restrictions of 120°C. Thus, the operating condition would not be in compliance with the code. A temperature control mechanism should be implemented to ensure that the water exiting the boiler stays below 120°C.

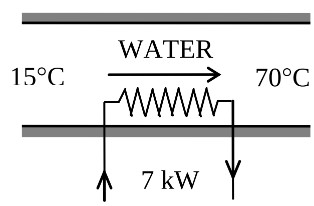
1-23 Water is heated in an insulated tube by an electric resistance heater. The mass flow rate of water through the heater is to be determined.

Assumptions **1** Water is an incompressible substance with a constant specific heat. **2** The kinetic and potential energy changes are negligible, $\Delta\text{ke} \cong \Delta\text{pe} \cong 0$. **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

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



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

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

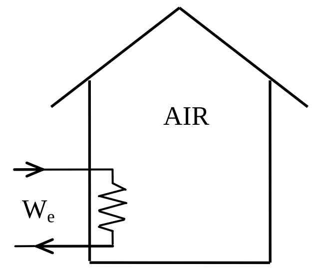
Thus,

$$\dot{m} = \frac{\dot{W}_{\text{e,in}}}{c_p(T_2 - T_1)} = \frac{7\text{ kJ/s}}{(4.18\text{ kJ/kg}\cdot°\text{C})(70-15)°\text{C}} = \mathbf{0.0304\,kg/s}$$

1-24 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 7000 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,

$$\dot{W}_{e,in} = \dot{Q}_{out} = 7000\text{kJ/h}\left(\frac{1\text{kW}}{3600\text{kJ/h}}\right) = \mathbf{1.94\ kW}$$

1-25 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.

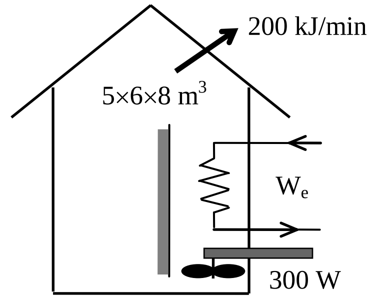
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.007 kJ/kg·K for air at room temperature (Table A-15) and $c_v = c_p - R$ = 0.720 kJ/kg·K.

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

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

(*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,

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

1-26 Air is moved through the resistance heaters in a 1200-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.

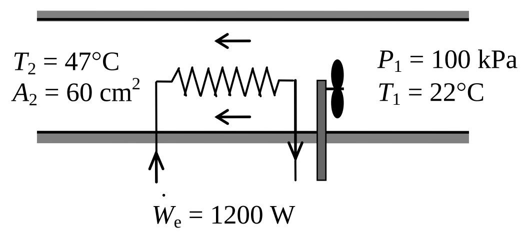
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.007 kJ/kg·K for air at room temperature (Table A-15).

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 $\Delta m_{\text{CV}} = 0$ and $\Delta E_{\text{CV}} = 0$, 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

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

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

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



Thus,

$$\dot{m} = \frac{\dot{W}_{\text{e,in}}}{c_p(T_2 - T_1)} = \frac{1.2 \text{ kJ/s}}{(1.007 \text{ kJ/kg} \cdot °\text{C})(47 - 22)°\text{C}} = 0.04767 \text{ kg/s}$$

Then,

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(295 \text{ K})}{100 \text{ kPa}} = 0.8467 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}v_1 = (0.04767 \text{ kg/s})(0.8467\text{m}^3/\text{kg}) = \mathbf{0.0404 \text{ m}^3/\text{s}}$$

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

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(320 \text{ K})}{100 \text{ kPa}} = 0.9184 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_2} A_2 V_2 \longrightarrow V_2 = \frac{\dot{m}v_2}{A_2} = \frac{(0.04767 \text{ kg/s})(0.9184 \text{ m}^3/\text{kg})}{60 \times 10^{-4} \text{ m}^2} = \mathbf{7.30 \text{ m/s}}$$

1-27 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 -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.

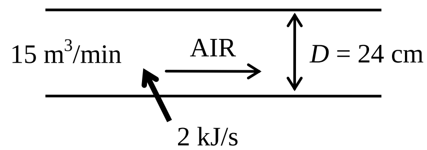
Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1). Also, c_p = 1.007 kJ/kg·K for air at room temperature (Table A-15).

Analysis 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 $\Delta m_{\text{CV}} = 0$ 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 heat is lost from the system. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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



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

$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi r^2} = \frac{15 \text{ m}^3/\text{min}}{\pi (0.12 \text{ m})^2} = \mathbf{332 \text{ m/min}}$$

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(10 + 273 \text{ K})}{100 \text{ kPa}} = 0.812 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{15 \text{ m}^3/\text{min}}{0.812 \text{ m}^3/\text{kg}} = 18.5 \text{ kg/min} = 0.308 \text{ kg/s}$$

Then the exit temperature of air is determined to be

$$T_2 = T_1 + \frac{\dot{Q}_{\text{in}}}{\dot{m}c_p} = 10°\text{C} + \frac{2 \text{ kJ/s}}{(0.308 \text{ kg/s})(1.007 \text{ kJ/kg} \cdot °\text{C})} = \mathbf{16.5°C}$$

1-28 Liquid water entering at 10°C and flowing at 10 g/s(0.01 kg/s) is heated in a circular tube by an electrical heater at 10 kW. Determine whether the water exit temperature would be below 79°C and comply with the ASME Code for Process Piping, and the minimum mass flow rate to keep the exit temperature below 79°C.

Assumptions **1** Water is an incompressible substance with constant properties. **2** Heat loss from the tube is negligible. **3** Steady operating conditions.

Properties The specific heat of water is given as 4.18 kJ/kg·K.

Analysis The tube is taken as a control volume. For steady state flow, the mass flow rate at the inlet is equal to the mass flow rate at the exit:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

The energy balance for the control volume is

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

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = 0$$

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

$$\dot{Q}_{\text{heater}} + \dot{m}h_1 = \dot{m}h_2$$

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

Solving for the exit temperature,

$$T_2 = \frac{\dot{Q}_{\text{heater}}}{\dot{m}c_p} + T_1 = \frac{10\,\text{kJ/s}}{(0.01\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot{}^\circ\text{C})} + 10^\circ\text{C} = \mathbf{249^\circ C} > 79^\circ\text{C}$$

Thus, the exit temperature is not in compliance with the ASME Code for Process Piping for PVDC lining.

The minimum mass flow rate needed to keep the water exit temperature from exceeding 79°C is

$$\dot{m} \ge \frac{\dot{Q}_{\text{heater}}}{(T_2 - T_1)c_p}$$

$$\ge \frac{10\,\text{kJ/s}}{(79-10)^\circ\text{C}\,(4.18\,\text{kJ/kg}\cdot{}^\circ\text{C})}$$

$$\ge \mathbf{0.0347\,kg/s}$$

The higher the value of mass flow rate, the lower the water exit temperature is achieved.

Discussion If the desire is to have higher exit temperature, then a different thermoplastic lining should be used. Polytetrafluoroethylene (PTFE) lining has a recommended maximum temperature of 260°C by the ASME Code for Process Piping (ASME B31.3-2014, A323).

Heat Transfer Mechanisms

1-29C 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.

1-30C Diamond is a better heat conductor.

1-31C 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.

1-32C 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.

1-33C 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.

1-34C 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.

1-35C 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.

1-36C Convection involves fluid motion, conduction does not. In a solid we can have only conduction.

1-37C No. It is purely by radiation.

1-38C 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.

1-39C 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.

1-40C 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.

1-41C 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.

1-42C 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.

1-43C The rate of heat transfer through both walls can be expressed as

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

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

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

1-44C 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.

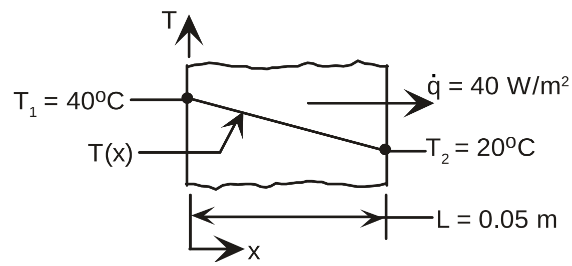
1-45C A blackbody is an idealized body which emits the maximum amount of radiation at a given temperature and which absorbs all the radiation incident on it. Real bodies emit and absorb less radiation than a blackbody at the same temperature.

1-46 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

$$k = \dot{q}\frac{L}{T_1 - T_2} = \left(40\frac{\text{W}}{\text{m}^2}\right)\frac{0.05\,\text{m}}{(40-20)°\text{C}} = \mathbf{0.10\ W/m{\cdot}K}$$



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

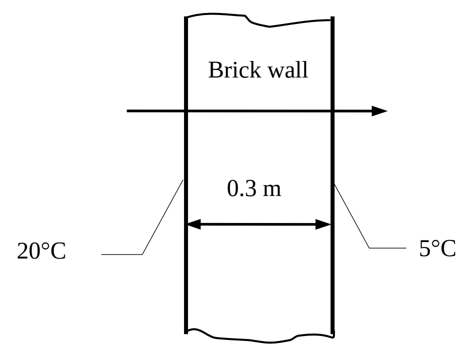
1-47 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of heat transfer through the wall is to be determined.

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

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.69\ \text{W/m}\cdot°\text{C})(4\times 7\ \text{m}^2)\frac{(20-5)°\text{C}}{0.3\ \text{m}} = \mathbf{966\ W}$$



1-48 The inner and outer glasses of a double pane window with a 6-mm air space are at specified temperatures. The rate of heat transfer through the window is to be determined

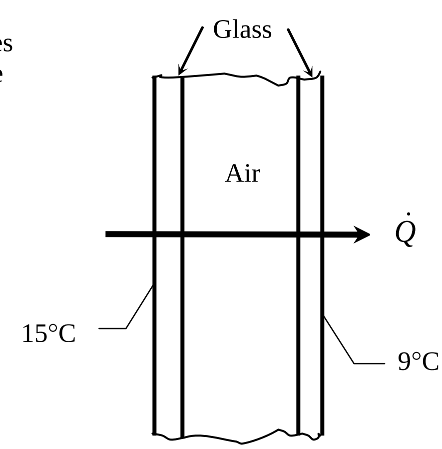
Assumptions **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Heat transfer through the window is one-dimensional. **3** Thermal properties of the air are constant.

Properties The thermal conductivity of air at the average temperature of (15+9)/2 = 12°C is k = 0.02454 W/m·°C (Table A-15).

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

$$A = (1.2\,\text{m})\times(1.2\,\text{m}) = 1.44\,\text{m}^2$$

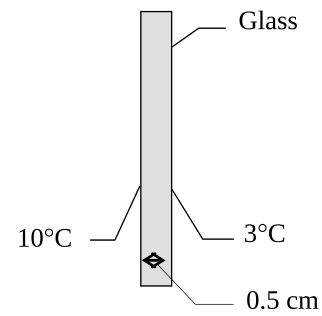
$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.02454\,\text{W/m.°C})(1.44\,\text{m}^2)\frac{(15-9)°\text{C}}{0.006\,\text{m}} = \mathbf{35.3\,W}$$



1-49 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

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

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

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

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**.

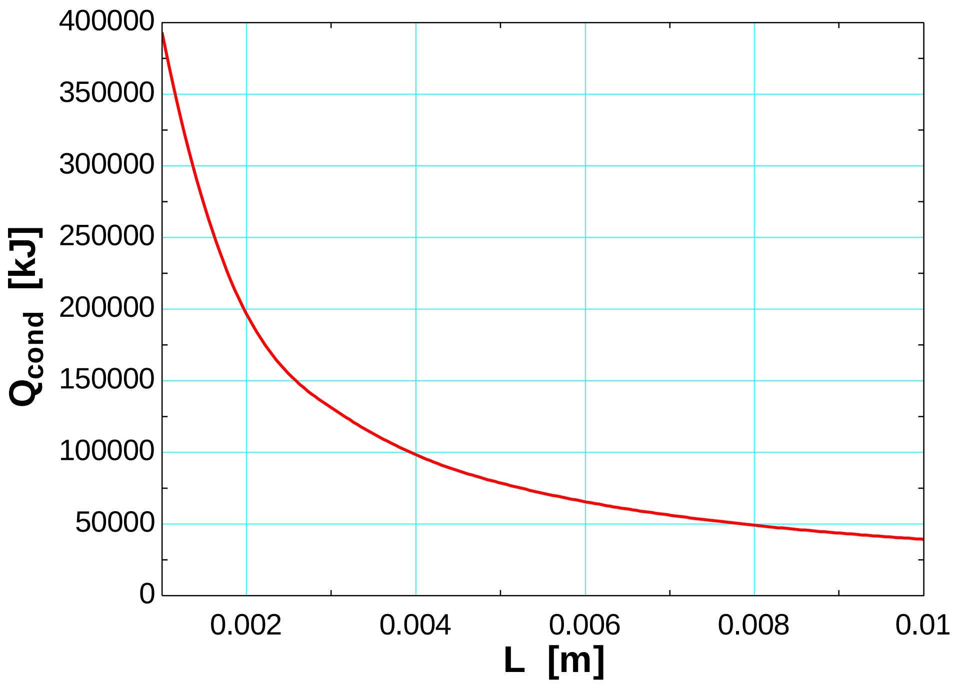
1-50 Prob. 1-49 is reconsidered. The amount of heat loss through the glass as a function of the window glass thickness is to be plotted.

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



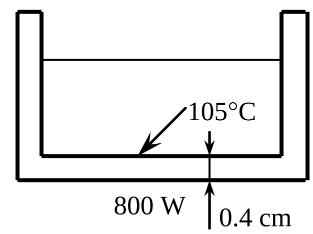
1-51 Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface temperature of the bottom of the pan is given. The temperature of the outer surface is to be determined.

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,

$$800 \text{ W} = (237 \text{ W/m}\cdot{}^\circ\text{C})(0.0177 \text{ m}^2)\frac{T_2 - 105^\circ\text{C}}{0.004 \text{ m}}$$

which gives

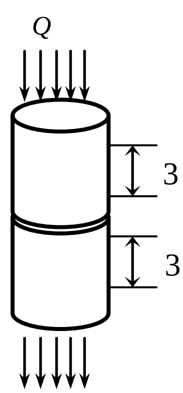
$$T_2 = \mathbf{105.76^\circ C}$$

1-52 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

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

$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(33 \text{ W})(0.03 \text{m})}{(0.001257 \text{m}^2)(10^\circ\text{C})} = \mathbf{78.8 \text{ W/m.}^\circ C}$$

1-53 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.

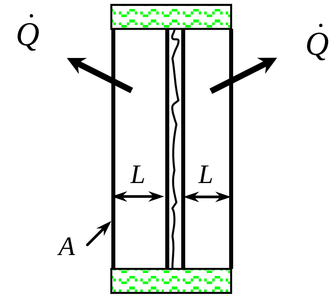
Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. **2** Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples.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

Analysis For each sample we have

$$\dot{Q} = 25/2 = 12.5\,\text{W}$$

$$A = (0.1\,\text{m})(0.1\,\text{m}) = 0.01\,\text{m}^2$$

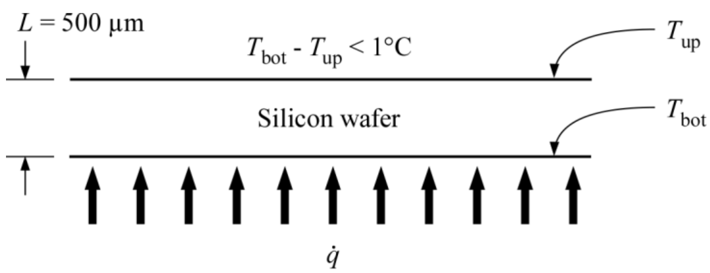
$$\Delta T = 82 - 74 = 8°\text{C}$$



Then the thermal conductivity of the material becomes

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

1-54 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

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

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

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

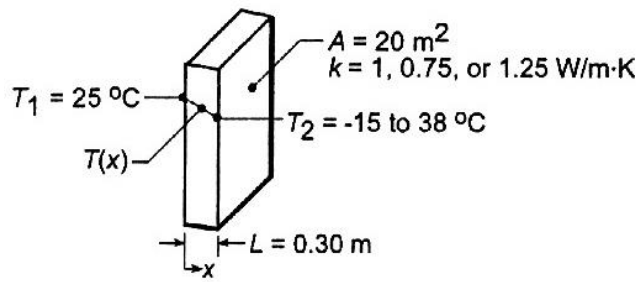
$$\dot{q} \le \mathbf{2.96\times10^5\ W/m^2}$$

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.

1-55 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:

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

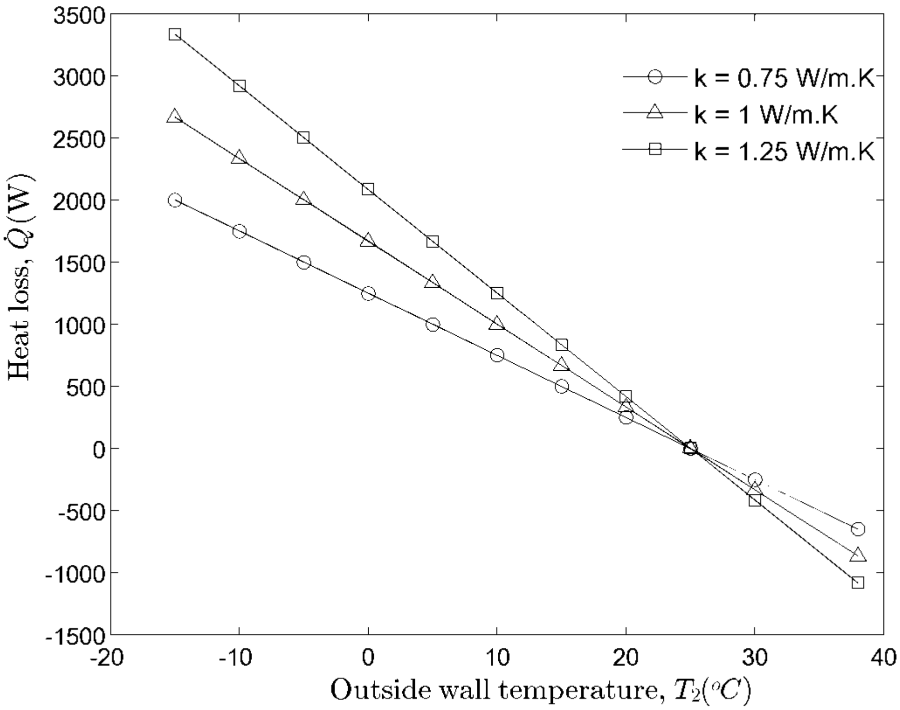
$$\dot{Q} = \dot{q}\cdot A = (133.3\,\text{W/m}^2)\cdot(20\,\text{m}^2) = \mathbf{2667\,W} \quad (2)$$

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


3500
3000
2500
2000
1500
1000
500
0
-500
-1000
-1500
-20
-10
0
10
20
30
40
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(°C)

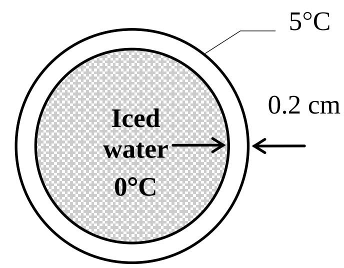
1-56 A hollow spherical iron container is filled with iced water at 0°C. The rate of heat gain by the iced water and the rate at which ice melts in the container are to be determined.

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. **5** Treat the spherical shell as a plain wall and use the outer area.

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

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

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.

1-57 Prob. 1-56 is reconsidered. The rate at which ice melts as a function of the container thickness is to be plotted.

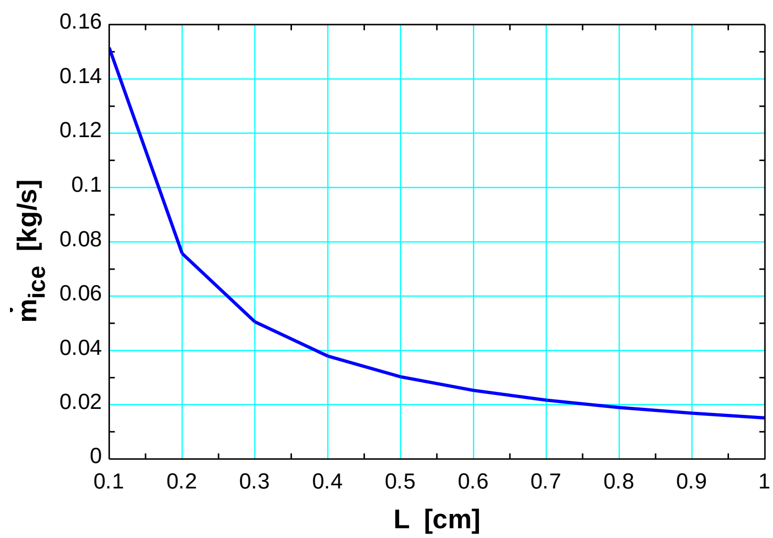
Analysis The problem is solved using EES, and the solution is given below.

```
"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
```

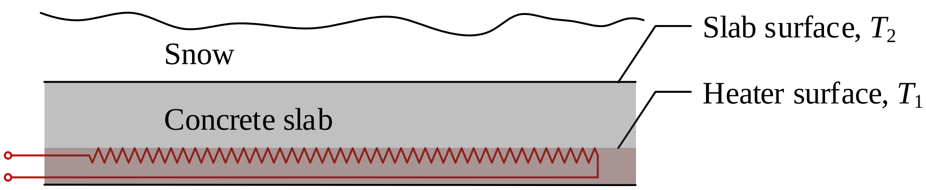
L [cm]	m_{ice} [kg/s]
0.1	0.1515
0.2	0.07574
0.3	0.0505
0.4	0.03787
0.5	0.0303
0.6	0.02525
0.7	0.02164
0.8	0.01894
0.9	0.01683
1	0.01515



1-58 C&S A 5 m × 5 m concrete slab with embedded heating cable melts snow at a rate of 0.1 kg/s. The power density (heat flux) for the embedded heater is to be determined whether it is in compliance with the NFPA 70 code. Also, the temperature difference between the heater surface and the slab surface is to be determined whether it exceeds 21°C, as recommended in the ASHRAE Handbook to minimize thermal stress.

Assumptions **1** Steady operating conditions. **2** Slab surface and heater surface temperatures are uniform. **3** Heat transfer through the concrete layer is one-dimensional. **3** Properties of the concrete are constant. **4** The heater heats the surface uniformly.

Properties The thermal conductivity of concrete is given as 1.4 W/m·K. The latent heat of fusion for water is 333.7 kJ/kg (Table A-2).



Analysis The heat rate required for melting snow at 0.1 kg/s is

$$\dot{Q} = \dot{m}_{\text{ice}} h_{if} = (0.1\,\text{kg/s})(333700\,\text{J/kg}) = 33370\,\text{W}$$

For a surface area of 5 m × 5 m, the power density (heat flux) is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{33370\,\text{W}}{25\,\text{m}^2} = \mathbf{1335\,W/m^2} > 1300\,\text{W/m}^2$$

To determine the temperature difference between the heater surface(T_1) and the slab surface(T_2), we use the Fourier law of conduction:

$$\dot{q} = k\frac{T_1 - T_2}{L}$$

or

$$T_1 - T_2 = \frac{\dot{q}\,L}{k} = \frac{(1335\,\text{W/m}^2)(0.05\,\text{m})}{1.4\,\text{W/m}\cdot\text{K}} = \mathbf{47.7°C} > 21°\text{C}$$

Discussion The power density for the embedded heating cable in the concrete slab slightly exceeds the limit set by the National Electrical Code® (NFPA 70) of 1300 W/m^2. The temperature difference between the heater surface and the slab surface is about 27°C higher than the recommended value by the 2015 ASHRAE Handbook—HVAC Applications, Chapter 51.

1-59 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

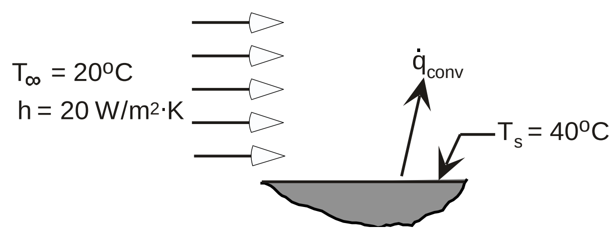
$$1\,\text{W/m}^2\cdot°\text{C} = \frac{3.41214\,\text{Btu/h}}{(3.2808\,\text{ft})^2(1.8\,°\text{F})} = 0.1761\,\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

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

$$h = 14\,\text{W/m}^2\cdot°\text{C} = 14\times 0.1761\,\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F} = \mathbf{2.47\,Btu/h\cdot ft^2\cdot °F}$$

1-60 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

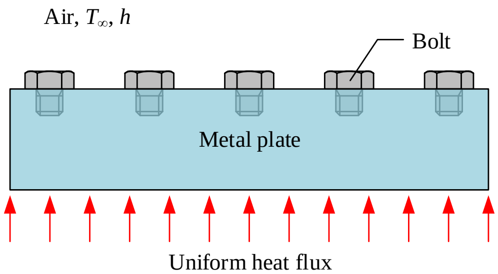
$$\dot{q}_{conv} = h\,(T_s - T_\infty)$$

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

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.

1-61 A series of ASME SA-193 carbon steel bolts are bolted to the upper surface of a metal plate. The upper surface is exposed to convection with the ambient air. The bottom surface is subjected to a uniform heat flux. Determine whether the use of the bolts complies with the ASME Boiler and Pressure Vessel Code, where 260°C is the maximum allowable use temperature.

Assumptions **1** Heat transfer is steady. **2** One dimensional heat conduction through the metal plate. **3** Uniform heat flux on the bottom surface. **4** Uniform surface temperature at the upper plate surface. **5** The temperature of the bolts is equal to the upper surface temperature of the plate.



Analysis The uniform heat flux subjected on the bottom plate surface is equal to the heat flux transferred by convection on the upper surface.

$$\dot{q}_0 = \dot{q}_{\text{cond}} = \dot{q}_{\text{conv}} = 5000\,\text{W/m}^2$$

From the Newton's law of cooling, we have

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

Assuming the temperature of the bolts is equal to the upper surface temperature of the plate,

$$T_{\text{bolt}} = T_s = \frac{\dot{q}_{\text{conv}}}{h} + T_\infty = \frac{5000\,\text{W/m}^2}{10\,\text{W/m}^2\cdot\text{K}} + 30°\text{C} = \mathbf{530°C} > 260°\text{C}$$

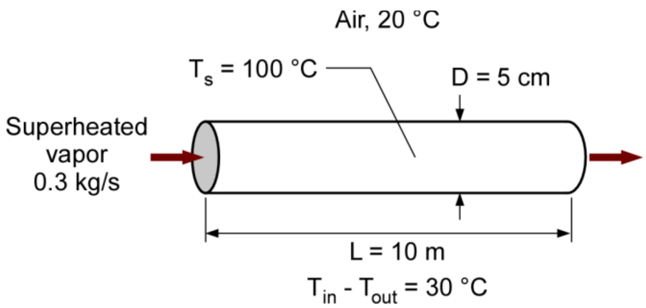
Discussion The temperature of the bolts exceeds the maximum allowable use temperature by 260°C. One way to keep the temperature of the bolts below 260°C is by increasing the convection heat transfer coefficient. Higher convection heat transfer coefficient can be achieved by having forced convection. To keep the upper surface temperature of the plate at 260°C or lower, the convection heat transfer coefficient should be higher than 21.7 W/m^2·K.

$$h > \frac{\dot{q}_{\text{conv}}}{T_s - T_\infty} > \frac{5000\,\text{W/m}^2}{(260-30)\,\text{K}} > 21.7\,\text{W/m}^2\cdot\text{K}$$

1-62 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.

1-63 A boiler supplies hot water to a dishwasher through a pipe at 60 g/s. The pipe dimensions are given. The water exits the boiler at 95°C. The pipe section between the boiler and the dishwasher is exposed to convection. The water temperature entering the dishwasher is to be determined whether it meets the ANSI/NSF 3 standard.

Assumptions **1** Constant properties are used for the water. **2** Steady operating conditions. **3** Surface temperature of the pipe is uniform.

Properties The average specific heat of water is given to be 4.20 kJ/kg·K.

Analysis From energy balance, the rate of heat loss from the pipe is equal to the heat transfer rate by convection on the pipe surface:

$$\dot{Q}_{\text{pipe}} = \dot{Q}_{\text{conv}}$$

$$\dot{m}\,c_p\,(T_1 - T_2) = h\,A_s\,(T_s - T_\infty)$$

Solving for the water temperature entering the dishwasher T_2, we have

$$T_2 = T_1 - \frac{h\,A_s}{\dot{m}\,c_p}(T_s - T_\infty)$$

$$= T_1 - \frac{\pi\,D\,L\,h}{\dot{m}\,c_p}(T_s - T_\infty)$$

$$= 95°\text{C} - \frac{\pi(0.02\text{ m})(20\text{ m})(100\text{ W/m}^2\cdot\text{K})}{(0.06\text{ kg/s})(4200\text{ J/kg}\cdot\text{K})}(50-20)°\text{C}$$

$$= \mathbf{80°C} < 82°\text{C}$$

Discussion The hot water entering the dishwasher is 2°C lower than the temperature required by the ANSI/NSF 3 standard. To increase the water temperature entering the dishwasher, one or combination of the following steps can be taken: (a) add insulation on the pipe wall to reduce the heat loss from the pipe surface; (b) increase the water mass flow rate; (c) reduce the pipe distance between the boiler and the dishwasher; and (d) increase the water temperature coming out from the boiler.

1-64 Hot liquid flows in a pipe with PVDF lining on the inner surface. The pipe outer surface is subjected to uniform heat flux. The liquid mean temperature and convection heat transfer coefficient are given. Determine whether the surface temperature of the lining complies with the ASME Code for Process Piping.

Assumptions **1** Steady operating conditions. **2** Heat transfer is one-dimensional through the pipe wall. **3** Surface temperature is uniform. **4** Thermal properties are constant.

Analysis The surface energy balance on the PVDF lining is

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

$$\dot{Q}_o = \dot{Q}_{\text{conv}}$$

$$\dot{q}_0 A_{s,o} = hA_{s,i}(T_s - T_f)$$

The outer and inner surface areas of the pipe are

$$A_{s,o} = \pi\,D_o\,L \quad \text{and} \quad A_{s,i} = \pi\,D_i\,L$$

Solving for the lining surface temperature T_s,

$$\dot{q}_0(\pi\,D_o\,L) = h(\pi\,D_i\,L)(T_s - T_f)$$

$$T_s = \frac{\dot{q}_0}{h}\frac{D_o}{D_i} + T_f = \frac{(1200\text{ W/m}^2)(27\text{ mm})}{(50\text{ W/m}^2\cdot\text{K})(22\text{ mm})} + 120°\text{C} = \mathbf{149.5°C} > 135°\text{C}$$

Discussion The surface temperature of the lining exceeds the maximum temperature recommended by the ASME Process Piping code for PVDF lining. A different thermoplastic lining should be used. Polytetrafluoroethylene (PTFE) lining has a recommended maximum temperature of 260°C by the ASME Code for Process Piping (ASME B31.3-2014, A323), which would meet these conditions.

1-65 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

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

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.

1-66 An electrical cable is covered with polyethylene insulation and is subjected to convection with the ambient air. Determine whether the insulation surface temperature meets the ASTM D1351 standard for polyethylene insulation.

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

AnalysisThe electrical energy that is converted to thermal energy is determined using the Joule heating relation:

$$\dot{Q} = IV = (1\,\text{A})(30\,\text{V}) = 30\,\text{W}$$

The thermal energy for the joule heating is then transferred through the insulation layer by conduction, and then by convection at the outer surface of the insulation. From the Newton's law of cooling for convection, we have

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

Rearranging the equation and solving for the surface temperature,

$$T_s = \frac{\dot{Q}}{h\,A_s} + T_\infty = \frac{30\,\text{W}}{(5\,\text{W/m}^2\cdot\text{K})(0.1\,\text{m}^2)} + 20°\text{C} = \mathbf{80°C} > 75°\text{C}$$

Discussion With the surface temperature being 5°C higher than the specification of the ASTM D1351 standard for polyethylene insulation that means the temperature at the inner surface of the insulation being in contact with the cable would be higher than 80°C. To solve this problem, we will need to use a larger diameter (or thicker) cable. The electrical resistance decreases with increasing cable thickness, which would reduce joule heating. We can also use a different insulation material with a higher temperature rating. From the ASTM database, the crosslinked polyethylene insulation (ASTM D2655) is rated up to 90°C for normal operation.

1-67 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.

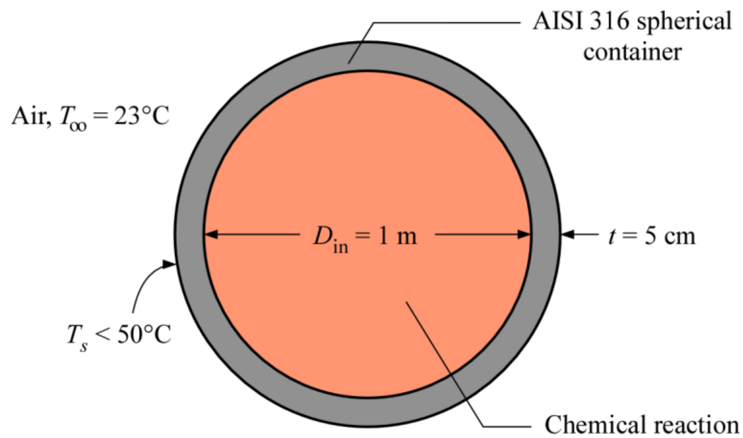
Assumptions **1** Steady operating conditions exist. **2** Negligible thermal storage for the container. **3** Temperature at the surface remained uniform.

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

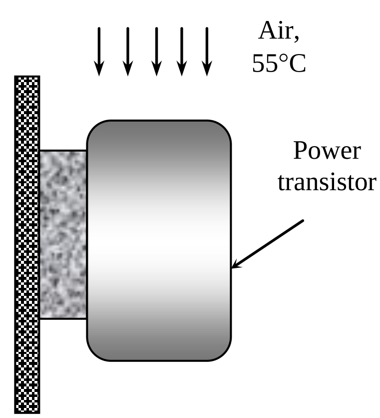
$$h > \mathbf{1840\,W/m^2\cdot K}$$

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.

1-68 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



$$A_s = \pi DL + \pi D^2/4$$

$$= \pi(0.6\,\text{cm})(0.4\,\text{cm}) + \pi(0.6\,\text{cm})^2/4 = 1.037\,\text{cm}^2$$

$$= 1.037\times 10^{-4}\,\text{m}^2$$

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

$$\dot{Q} = hA_s(T_s - T_\infty) = (30\,\text{W/m}^2\cdot°\text{C})(1.037\times 10^{-4}\,\text{m}^2)(70-55)°\text{C} = \mathbf{0.047\,W}$$

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

1-69 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.

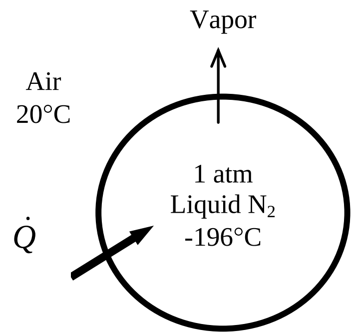
Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is disregarded. **3** The convection heat transfer coefficient is constant and uniform over the surface. **4** The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the nitrogen inside.

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

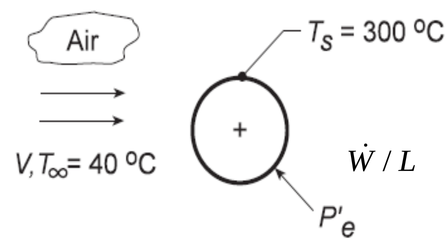
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{271.430\,\text{kJ/s}}{198\,\text{kJ/kg}} = \mathbf{1.37\,kg/s}$$

1-70 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,

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



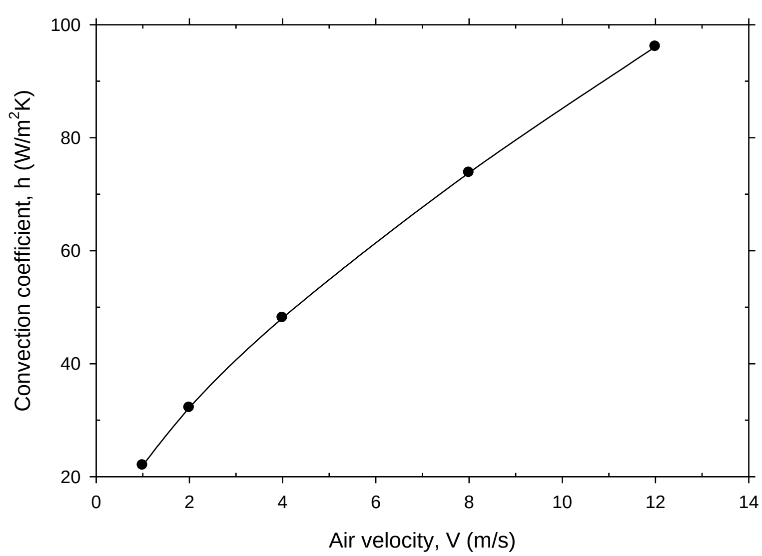
where $\dot{W}/L$ is the electrical power dissipated per unit length of the cylinder. For the V = 1 m/s condition, using the data from the table given in the problem statement, find

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

$$h = 450\text{ W/m} / (\pi \times 0.025\text{ m})\,(300 - 40)\text{ ºC} = 22.0\text{ W/m}^2\cdot\text{K}$$

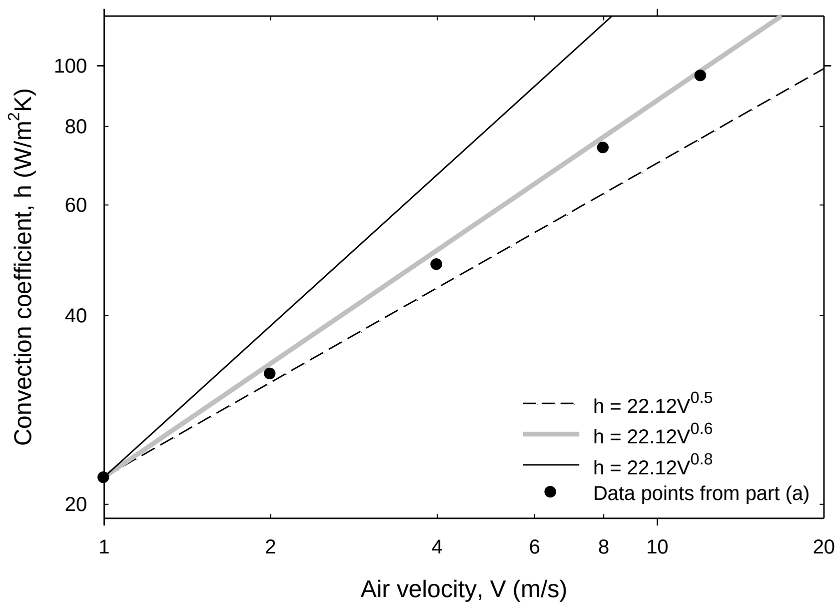
Repeating the calculations for the rest of the *V* values given, find the convection coefficients for the remaining conditions in the table. The results are tabulated and plotted below. Note that h is not linear with respect to the air velocity.

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

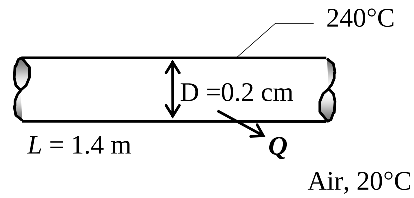
Discussion Radiation may not be negligible, depending on the surface emissivity.

1-71 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

$$A_s = \pi DL = \pi(0.002\,\text{m})(1.4\,\text{m}) = 0.00880\,\text{m}^2$$

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

$$h = \frac{\dot{Q}}{A_s(T_1 - T_\infty)} = \frac{330\,\text{W}}{(0.00880\,\text{m}^2)(240-20)°\text{C}} = \mathbf{170.5\,W/m^2\cdot°C}$$

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.

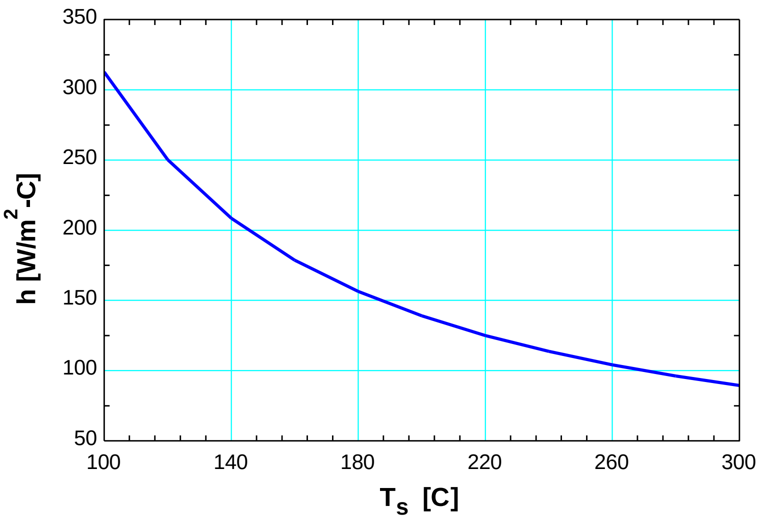
1-72 Prob. 1-71 is reconsidered. The convection heat transfer coefficient as a function of the wire surface temperature is to be plotted.

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)
```

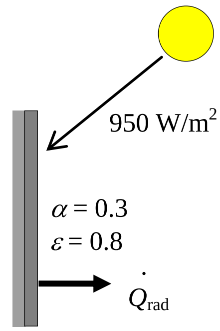
T_s [C]	h [W/m^2.C]
100	312.6
120	250.1
140	208.4
160	178.6
180	156.3
200	138.9
220	125.1
240	113.7
260	104.2
280	96.19
300	89.32



1-73 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

$$\dot{Q}_{\text{solar absorbed}} = \dot{Q}_{\text{rad}}$$

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

$$0.3 \times A_s \times (950 \text{ W/m}^2) = 0.8 \times A_s \times (5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[T_s^4 - (0 \text{ K})^4]$$

Canceling the surface area A and solving for T_s gives

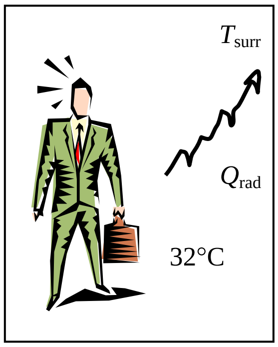
$$T_s = \mathbf{281.5\ K}$$

1-74 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

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4) \\ &= (0.5)(5.67 \times 10^{-8} \text{ W/m}^2.\text{K}^4)(1.7 \text{ m}^2)[(32+273)^4 - (300 \text{ K})^4]\text{K}^4 \\ &= \mathbf{26.7\ W}\end{aligned}$$

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

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon \sigma A_s (T_s^4 - T_{\text{surr}}^4) \\ &= (0.5)(5.67 \times 10^{-8} \text{ W/m}^2.\text{K}^4)(1.7 \text{ m}^2)[(32+273)^4 - (280 \text{ K})^4]\text{K}^4 \\ &= \mathbf{121\ W}\end{aligned}$$

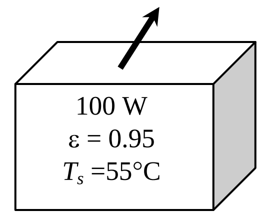
Discussion Note that the radiation heat transfer goes up by more than 4 times as the temperature of the surrounding surfaces drops from 300 K to 280 K.

1-75 A sealed electronic box dissipating a total of 100 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



$$A_s = (0.4\,\text{m})(0.4\,\text{m}) + 4\times(0.2\,\text{m})(0.4\,\text{m}) = 0.48\,\text{m}^2$$

The radiation heat transfer from the box can be expressed as

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

$$100\,\text{W} = (0.95)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(0.48\,\text{m}^2)\left[(55+273\,\text{K})^4 - T_{\text{surr}}^4\right]$$

which gives T_{surr} = **296.3 K = 23.3°C**. Therefore, the temperature of the surrounding surfaces must be less than 23.3°C.

1-76 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 $\varepsilon_1 = 0.1$, and the emissivity of heavily oxidized surface is $\varepsilon_2 = 0.78$.

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.

1-77 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

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

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

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

Thus, incident radiation rate on the probe surface is

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

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

$$\dot{Q}_{\text{solar}} = A_s\dot{q}_{\text{solar}} = (4\pi r^2)\dot{q}_{\text{solar}} = 4\pi(1\,\text{m})^2(1171\,\text{W/m}^2) = \mathbf{14{,}715\,W}$$

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