Complete Solution Manual to Accompany

HEAT TRANSFER

A Practical Approach

SECOND EDITION

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Preface

This manual is prepared as an aide to the instructors in correcting homework assignments, but it can also be used as a source of additional example problems for use in the classroom. With this in mind, all solutions are prepared in full detail in a systematic manner, using a word processor with an equation editor. The solutions are structured into the following sections to make it easy to locate information and to follow the solution procedure, as appropriate:

Solution -	The problem is posed, and the quantities to be found are stated.
Assumptions -	The significant assumptions in solving the problem are stated.
Properties -	The material properties needed to solve the problem are listed.
Analysis -	The problem is solved in a systematic manner, showing all steps.
Discussion -	Comments are made on the results, as appropriate.

A sketch is included with most solutions to help the students visualize the physical problem, and also to enable the instructor to glance through several types of problems quickly, and to make selections easily.

Problems designated with the CD icon in the text are also solved with the EES software, and electronic solutions complete with parametric studies are available on the CD that accompanies the text. Comprehensive problems designated with the computer-EES icon [**pick one of the four given**] are solved using the EES software, and their solutions are placed at the *Instructor Manual* section of the *Online Learning Center* (OLC) at www.mhhe.com/cengel. Access to solutions is limited to instructors only who adopted the text, and instructors may obtain their passwords for the OLC by contacting their McGraw-Hill Sales Representative at http://www.mhhe.com/catalogs/rep/.

Every effort is made to produce an error-free Solutions Manual. However, in a text of this magnitude, it is inevitable to have some, and we will appreciate hearing about them. We hope the text and this Manual serve their purpose in aiding with the instruction of Heat Transfer, and making the Heat Transfer experience of both the instructors and students a pleasant and fruitful one.

We acknowledge, with appreciation, the contributions of numerous users of the first edition of the book who took the time to report the errors that they discovered. All of their suggestions have been incorporated. Special thanks are due to Dr. Mehmet Kanoglu who checked the accuracy of most solutions in this Manual.

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Chapter 1
BASICS OF HEAT TRANSFER

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 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.

1-7C 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.

Heat and Other Forms of Energy

1-8C 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 $\dot{Q} = \int_A \dot{q} dA$.

1-9C Energy can be transferred by heat, work, and mass. An energy transfer is heat transfer when its driving force is temperature difference.

1-10C Thermal energy is the sensible and latent forms of internal energy, and it is referred to as heat in daily life.

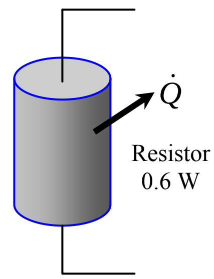
1-11C For the constant pressure case. This is because the heat transfer to an ideal gas is $mC_p\Delta T$ at constant pressure and $mC_p\Delta T$ at constant volume, and C_p is always greater than C_v.

1-12 A cylindrical resistor on a circuit board dissipates 0.6 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

$$Q = \dot{Q}\Delta t = (0.6\ \text{W})(24\ \text{h}) = \mathbf{14.4\ Wh = 51.84\ kJ} \quad (\text{since } 1\ \text{Wh} = 3600\ \text{Ws} = 3.6\ \text{kJ})$$



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

$$A_s = 2\frac{\pi D^2}{4} + \pi DL = 2\frac{\pi(0.4\ \text{cm})^2}{4} + \pi(0.4\ \text{cm})(1.5\ \text{cm}) = 0.251 + 1.885 = 2.136\ \text{cm}^2$$

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{0.60\ \text{W}}{2.136\ \text{cm}^2} = \mathbf{0.2809\ W/cm^2}$$

(*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

$$\frac{Q_{\text{top-base}}}{Q_{\text{total}}} = \frac{A_{\text{top-base}}}{A_{\text{total}}} = \frac{0.251}{2.136} = \mathbf{0.118} \quad \text{or} \quad (11.8\%)$$

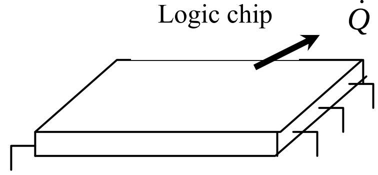
Discussion Heat transfer from the top and bottom surfaces is small relative to that transferred from the side surface.

1-13E 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

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

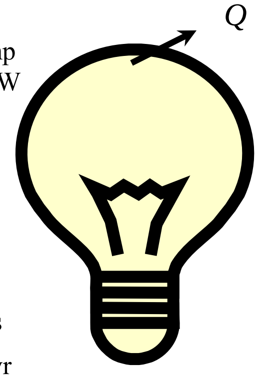
1-14 The filament of a 150 W incandescent lamp is 5 cm long and has a diameter of 0.5 mm. The heat flux on the surface of the filament, the heat flux on the surface of the glass bulb, and the annual electricity cost of the bulb are to be determined.

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

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

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



(*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

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

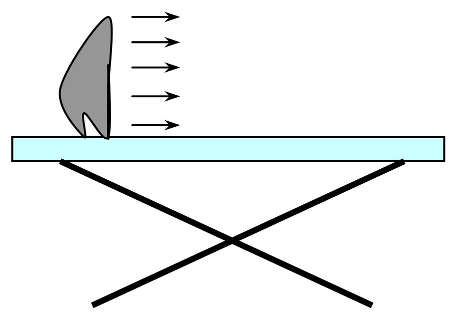
$$\text{Annual Cost} = (438\text{ kWh / yr})(\$0.08\text{ / kWh}) = \mathbf{\$35.04\ /\ yr}$$

1-15 A 1200 W iron is left on the ironing board with its base exposed to the air. The amount of heat the iron dissipates in 2 h, the heat flux on the surface of the iron base, and the cost of the electricity are to be determined.

Assumptions Heat transfer from the surface is uniform.

Analysis (*a*) The amount of heat the iron dissipates during a 2-h period is

$$Q = \dot{Q}\Delta t = (1.2\text{ kW})(2\text{ h}) = \mathbf{2.4\ kWh}$$



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

$$\dot{Q}_{\text{base}} = (0.9)(1200\text{ W}) = 1080\text{ W}$$

$$\dot{q} = \frac{\dot{Q}_{\text{base}}}{A_{\text{base}}} = \frac{1080\text{ W}}{0.015\text{ m}^2} = \mathbf{72{,}000\ W\ /\ m^2}$$

(*c*) The cost of electricity consumed during this period is

$$\text{Cost of electricity} = (2.4\text{ kWh}) \times (\$0.07\text{ / kWh}) = \mathbf{\$0.17}$$

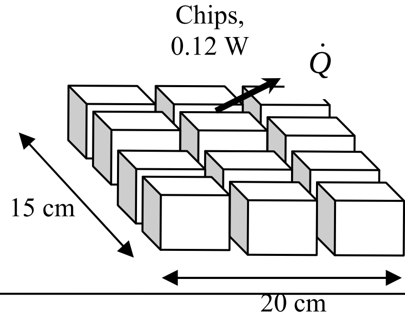
1-16 A 15 cm × 20 cm circuit board houses 120 closely spaced 0.12 W logic chips. The amount of heat dissipated in 10 h and the heat flux on the surface of the circuit board are to be determined.

Assumptions **1** Heat transfer from the back surface of the board is negligible. **2** Heat transfer from the front surface is uniform.

Analysis (*a*) The amount of heat this circuit board dissipates during a 10-h period is

$$\dot{Q} = (120)(0.12\text{ W}) = 14.4\text{ W}$$

$$Q = \dot{Q}\Delta t = (0.0144\text{ kW})(10\text{ h}) = \mathbf{0.144\,kWh}$$



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

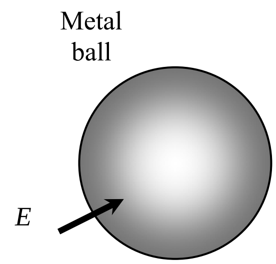
$$A_s = (0.15\text{ m})(0.2\text{ m}) = 0.03\text{ m}^2$$

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{14.4\text{ W}}{0.03\text{ m}^2} = \mathbf{480\ W/m^2}$$

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 = 2,700 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

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

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,

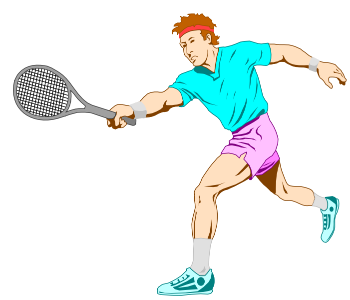
$$E_{\text{transfer}} = (4.77\text{ kg})(0.90\text{ kJ/kg.°C})(200\text{ - }80)\text{°C} = \mathbf{515\ kJ}$$

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 The body temperature of a man rises from 37°C to 39°C during strenuous exercise. The resulting increase in the thermal energy content of the body is to be determined.

Assumptions The body temperature changes uniformly.

Properties The average specific heat of the human body is given to be 3.6 kJ/kg.°C.



Analysis The change in the sensible internal energy content of the body as a result of the body temperature rising 2°C during strenuous exercise is

$$\Delta U = mC\Delta T = (70\text{ kg})(3.6\text{ kJ/kg.°C})(2\text{°C}) = \mathbf{504\ kJ}$$

1-19 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.

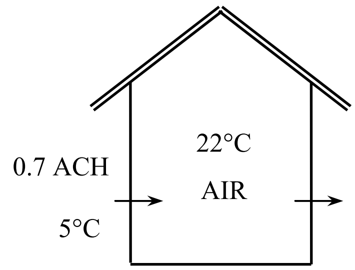
Assumptions **1** Air as an ideal gas with a constant specific heats at room temperature. **2** The volume occupied by the furniture and other belongings is negligible. **3** The house is maintained at a constant temperature and pressure at all times. **4** The infiltrating air exfiltrates at the indoors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

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

Analysis The volume of the air in the house is

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

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



$$\dot{m}_{\text{air}} = \frac{P_o \dot{V}_{\text{air}}}{RT_o} = \frac{P_o(\text{ACH} \times V_{\text{house}})}{RT_o}$$
$$= \frac{(89.6\ \text{kPa})(16.8 \times 600\ \text{m}^3/\text{day})}{(0.287\ \text{kPa.m}^3/\text{kg.K})(5 + 273.15\ \text{K})} = 11{,}314\ \text{kg/day}$$

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

$$\dot{Q}_{\text{infilt}} = \dot{m}_{\text{air}} C_{\text{p}} (T_{\text{indoors}} - T_{\text{outdoors}})$$
$$= (11{,}314\ \text{kg/day})(1.007\ \text{kJ/kg.°C})(22 - 5)\text{°C} = 193{,}681\ \text{kJ/day} = \mathbf{53.8\ kWh/day}$$

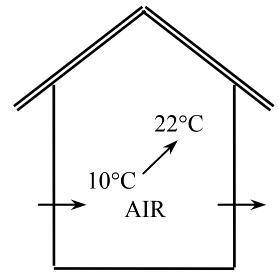
At a unit cost of $0.082/kWh, the cost of this electrical energy lost by infiltration is

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

1-20 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 (Table A-15).



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

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

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

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

$$Q = mC_p(T_2 - T_1) = (747.9\text{ kg})(1.007\text{ kJ/kg.°C})(22-10)\text{°C} = \mathbf{9038\ kJ}$$

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

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

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

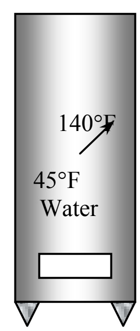
1-21E A water heater is initially filled with water at 45°F. The amount of energy that needs to be transferred to the water to raise its temperature to 140°F is to be determined.

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

Properties The density and specific heat of water are given to be 62 lbm/ft^3 and 1.0 Btu/lbm.°F.

Analysis The mass of water in the tank is

$$m = \rho V = (62\text{ lbm/ft}^3)(60\text{ gal})\left(\frac{1\text{ ft}^3}{7.48\text{ gal}}\right) = 497.3\text{ lbm}$$



Then, the amount of heat that must be transferred to the water in the tank as it is heated from 45 to140°F is determined to be

$$Q = mC(T_2 - T_1) = (497.3\text{ lbm})(1.0\text{ Btu/lbm.°F})(140-45)\text{°F} = \mathbf{47{,}250\ Btu}$$

The First Law of Thermodynamics

1-22C Warmer. Because energy is added to the room air in the form of electrical work.

1-23C 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.

1-24C Mass flow rate $\dot{m}$ is the amount of mass flowing through a cross-section per unit time whereas the volume flow rate $\dot{V}$ is the amount of volume flowing through a cross-section per unit time. They are related to each other by $\dot{m} = \rho\dot{V}$ where ρ is density.

1-25 Two identical cars have a head-on collusion on a road, and come to a complete rest after the crash. The average temperature rise of the remains of the cars immediately after the crash is to be determined.

Assumptions **1** No heat is transferred from the cars. **2** All the kinetic energy of cars is converted to thermal energy.

Properties The average specific heat of the cars is given to be 0.45 kJ/kg.°C.

Analysis We take both cars as the system. This is a *closed system* since it involves a fixed amount of mass (no mass transfer). Under the stated assumptions, the energy balance on the system can be expressed as

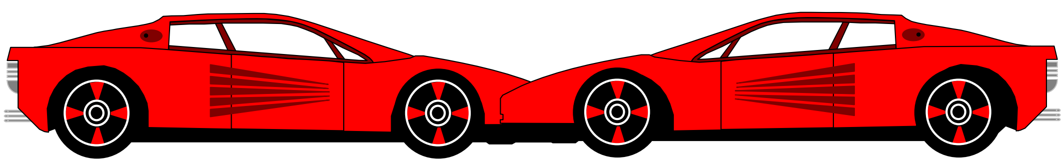
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U_{\text{cars}} + \Delta KE_{\text{cars}}$$

$$0 = (mC\Delta T)_{\text{cars}} + [m(0 - V^2)/2]_{\text{cars}}$$

That is, the decrease in the kinetic energy of the cars must be equal to the increase in their internal energy. Solving for the velocity and substituting the given quantities, the temperature rise of the cars becomes

$$\Delta T = \frac{mV^2/2}{mC} = \frac{V^2/2}{C} = \frac{(90{,}000/3600\ \text{m/s})^2/2}{0.45\ \text{kJ/kg.°C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{0.69°C}$$



1-26 A classroom is to be air-conditioned using window air-conditioning units. The cooling load is due to people, lights, and heat transfer through the walls and the windows. The number of 5-kW window air conditioning units required is to be determined.

Assumptions There are no heat dissipating equipment (such as computers, TVs, or ranges) in the room.

Analysis The total cooling load of the room is determined from

$$\dot{Q}_{\text{cooling}} = \dot{Q}_{\text{lights}} + \dot{Q}_{\text{people}} + \dot{Q}_{\text{heat gain}}$$

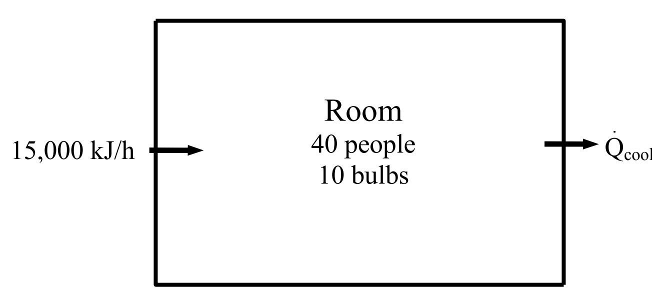
where

$$\dot{Q}_{\text{lights}} = 10 \times 100\ \text{W} = 1\ \text{kW}$$

$$\dot{Q}_{\text{people}} = 40 \times 360\text{kJ/h} = 14{,}400\ \text{kJ/h} = 4\text{kW}$$

$$\dot{Q}_{\text{heatgain}} = 15{,}000\ \text{kJ/h} = 4.17\ \text{kW}$$

Substituting, $\dot{Q}_{\text{cooling}} = 1 + 4 + 4.17 = 9.17\ \text{kW}$



Thus the number of air-conditioning units required is

$$\frac{9.17\ \text{kW}}{5\ \text{kW/unit}} = 1.83 \longrightarrow \mathbf{2\ units}$$

1-27E The air in a rigid tank is heated until its pressure doubles. The volume of the tank and the amount of heat transfer 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, $\Delta pe \cong \Delta ke \cong 0$. **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 gas constant of air is R = 0.3704 psia.ft^3/lbm.R = 0.06855 Btu/lbm.R (Table A-1).

Analysis (*a*) We take the air in the tank as our system. This is a *closed system* since no mass enters or leaves. The volume of the tank can be determined from the ideal gas relation,

$$V = \frac{mRT_1}{P_1} = \frac{(20\text{lbm})(0.3704\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(80+460\text{R})}{50\text{psia}} = \mathbf{80.0ft^3}$$

(b) Under the stated assumptions and observations, the energy balance becomes

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

$$Q_{in} = \Delta U \longrightarrow Q_{in} = m(u_2 - u_1) \cong mC_v(T_2 - T_1)$$

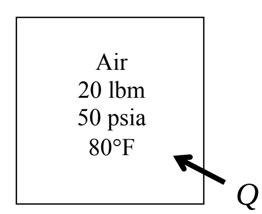
The final temperature of air is

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \quad\longrightarrow\quad T_2 = \frac{P_2}{P_1}T_1 = 2 \times (540\text{ R}) = 1080\ R$$

The specific heat of air at the average temperature of T_{ave} = (540+1080)/2= 810 R = 350°F is

$C_{\text{v,ave}} = C_{\text{p,ave}} - R$ = 0.2433 - 0.06855 = 0.175 Btu/lbm.R. Substituting,

Q = (20 lbm)(0.175 Btu/lbm.R)(1080 - 540) R = **1890 Btu**



1-28 The hydrogen gas in a rigid tank is cooled until its temperature drops to 300 K. The final pressure in the tank and the amount of heat transfer are to be determined.

Assumptions **1** Hydrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -240°C and 1.30 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$.

Properties The gas constant of hydrogen is R = 4.124 kPa.m^3/kg.K (Table A-1).

Analysis (*a*) We take the hydrogen in the tank as our system. This is a *closed system* since no mass enters or leaves. The final pressure of hydrogen can be determined from the ideal gas relation,

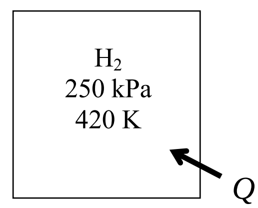
$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{300 \text{ K}}{420 \text{ K}} (250 \text{ kPa}) = \mathbf{178.6 \text{ kPa}}$$

(*b*) The energy balance for this system can be expressed as

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

$$-Q_{out} = \Delta U$$

$$Q_{out} = -\Delta U = -m(u_2 - u_1) \cong mC_v (T_1 - T_2)$$



where

$$m = \frac{P_1 V}{R T_1} = \frac{(250 \text{ kPa})(1.0 \text{ m}^3)}{(4.124 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(420 \text{ K})} = 0.1443 \text{ kg}$$

Using the C_v (=C_p – R) = 14.516 – 4.124 = 10.392 kJ/kg.K value at the average temperature of 360 K and substituting, the heat transfer is determined to be

Q_{out} = (0.1443 kg)(10.392 kJ/kg·K)(420 - 300)K = **180.0 kJ**

1-29 A resistance heater is to raise the air temperature in the room from 7 to 25°C within 20 min. The required power rating of the 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. **4** Heat losses from the room 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 We observe that the pressure in the room remains constant during this process. Therefore, some air will leak out as the air expands. However, we can take the air to be a closed system by considering the air in the room to have undergone a constant pressure expansion process. The energy balance for this steady-flow system can be expressed as

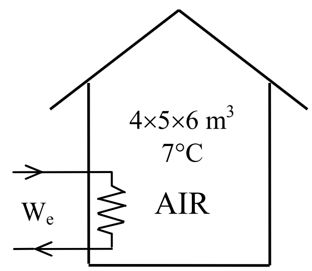
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,in} - W_b = \Delta U$$

$$W_{e,in} = \Delta H = m(h_2 - h_1) \cong mC_p(T_2 - T_1)$$

or,

$$\dot{W}_{e,in}\Delta t = mC_{p,ave}(T_2 - T_1)$$



The mass of air is

$$V = 4 \times 5 \times 6 = 120 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(280 \text{ K})} = 149.3 \text{ kg}$$

Using C_p value at room temperature, the power rating of the heater becomes

$$\dot{W}_{e,in} = (149.3 \text{ kg})(1.007 \text{ kJ/kg}\cdot^\circ\text{C})(25-7)^\circ\text{C}/(15\times 60 \text{ s}) = \mathbf{3.01 \text{ kW}}$$

1-30 A room is heated by the radiator, and the warm air is distributed by a fan. Heat is lost from the room. The time it takes for the air temperature to rise to 20°C 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, C_p = 1.007 and C_v = 0.720 kJ/kg·K. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa.

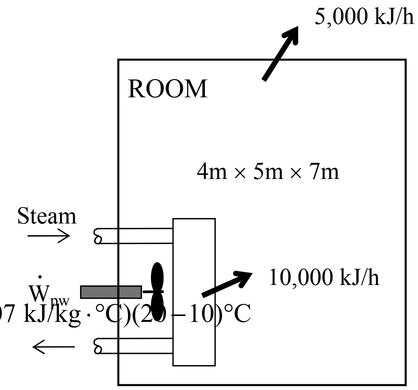
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_p = 1.007 kJ/kg·K for air at room temperature (Table A-15).

Analysis We take the air in the room as the system. This is a *closed system* since no mass crosses the system boundary during the process. We observe that the pressure in the room remains constant during this process. Therefore, some air will leak out as the air expands. However we can take the air to be a closed system by considering the air in the room to have undergone a constant pressure process. The energy balance for this system can be expressed as

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

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

$$(\dot{Q}_{in} + \dot{W}_{e,in} - \dot{Q}_{out})\Delta t = \Delta H = m(h_2 - h_1) \cong mC_p(T_2 - T_1)$$



The mass of air is

$$V = 4 \times 5 \times 7 = 140 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(140 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3 / \text{kg} \cdot \text{K})(283 \text{ K})} = 172.4 \text{ kg}$$

Using the C_p value at room temperature,

$$\left[(10{,}000 - 5000)/3600 \text{ kJ/s} + 0.1 \text{ kJ/s}\right]\Delta t = (172.4 \text{ kg})(1.007 \text{ kJ/kg} \cdot °\text{C})(20 - 10)°\text{C}$$

It yields

Δt = **1163 s**

1-31 A student living in a room turns his 150-W fan on in the morning. The temperature in the room when she comes back 10 h later 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. **4** All the doors and windows are tightly closed, and heat transfer through the walls and the windows is disregarded.

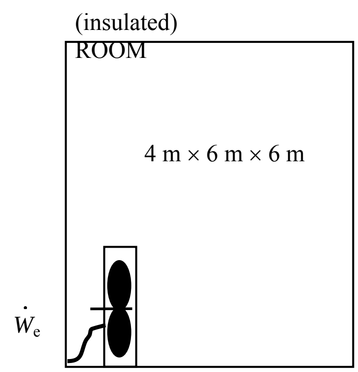
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_p = 1.007 kJ/kg·K for air at room temperature (Table A-15) and $C_v = C_p - R$ = 0.720 kJ/kg·K.

Analysis We take the room as the system. This is a *closed system* since the doors and the windows are said to be tightly closed, and thus no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

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

$$W_{e,in} = \Delta U$$

$$W_{e,in} = m(u_2 - u_1) \cong mC_v(T_2 - T_1)$$



The mass of air is

$$V = 4 \times 6 \times 6 = 144 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(144 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 174.2 \text{ kg}$$

The electrical work done by the fan is

$$W_e = \dot{W}_e \Delta t = (0.15 \text{ kJ/s})(10 \times 3600 \text{ s}) = 5400 \text{ kJ}$$

Substituting and using C_v value at room temperature,

$$5400 \text{ kJ} = (174.2 \text{ kg})(0.720 \text{ kJ/kg.°C})(T_2 - 15)\text{°C}$$

$$T_2 = \mathbf{58.1°C}$$

1-32E A paddle wheel in an oxygen tank is rotated until the pressure inside rises to 20 psia while some heat is lost to the surroundings. The paddle wheel work done is to be determined.

Assumptions **1** Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -181°F and 736 psia. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The energy stored in the paddle wheel is negligible. **4** This is a rigid tank and thus its volume remains constant.

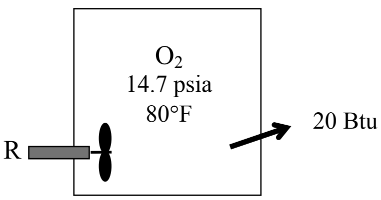
Properties The gas constant of oxygen is R = 0.3353 psia.ft^3/lbm.R = 0.06206 Btu/lbm.R (Table A-1E).

Analysis We take the oxygen in the tank as our system. This is a *closed system* since no mass enters or leaves. The energy balance for this system can be expressed as

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

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

$$W_{pw,in} = Q_{out} + m(u_2 - u_1) \cong Q_{out} + mC_v(T_2 - T_1)$$



The final temperature and the number of moles of oxygen are

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{20 \text{ psia}}{14.7 \text{ psia}} (540 \text{ R}) = 735 \text{ R}$$

$$m = \frac{P_1 V}{RT_1} = \frac{(14.7 \text{ psia})(10 \text{ ft}^3)}{(0.3353 \text{ psia}\cdot\text{ft}^3 / \text{lbmol}\cdot\text{R})(540 \text{ R})} = 0.812 \text{ lbm}$$

The specific heat ofoxygen at the average temperature of T_{ave} = (735+540)/2= 638 R = 178°F is

$C_{v,\text{ave}} = C_p - R$ = 0.2216-0.06206 = 0.160 Btu/lbm.R. Substituting,

$W_{\text{pw,in}}$ = (20 Btu) + (0.812 lbm)(0160 Btu/lbm.R)(735 - 540) Btu/lbmol = **45.3 Btu**

Discussion Note that a "cooling" fan actually causes the internal temperature of a confined space to rise. In fact, a 100-W fan supplies a room as much energy as a 100-W resistance heater.

1-33 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** We 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

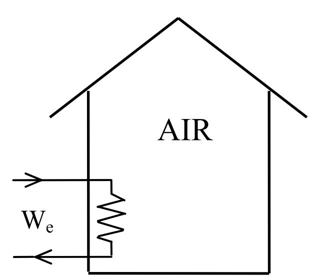
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

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

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

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

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

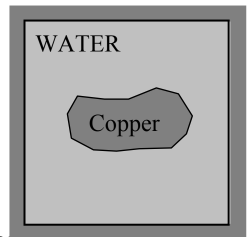


1-34 A hot copper block is dropped into water in an insulated tank. The final equilibrium temperature of the tank is to be determined.

Assumptions **1** Both the water and the copper block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energy changes are zero, $\Delta KE = \Delta PE = 0$ and $\Delta E = \Delta U$. **3** The system is well-insulated and thus there is no heat transfer.

Properties The specific heats of water and the copper block at room temperature are $C_{\text{p, water}}$ = 4.18 kJ/kg·°C and $C_{\text{p, Cu}}$ = 0.386 kJ/kg·°C (Tables A-3 and A-9).

Analysis We observe that the volume of a rigid tank is constant We take the entire contents of the tank, water + copper block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance on the system can be expressed as



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

$$0 = \Delta U$$

or,
$$\Delta U_{Cu} + \Delta U_{water} = 0$$

$$[mC(T_2 - T_1)]_{Cu} + [mC(T_2 - T_1)]_{water} = 0$$

Using specific heat values for copper and liquid water at room temperature and substituting,

$$(50 \text{ kg})(0.386 \text{ kJ/kg}\cdot{}^\circ\text{C})(T_2 - 70)^\circ\text{C} + (80 \text{ kg})(4.18 \text{ kJ/kg}\cdot{}^\circ\text{C})(T_2 - 25)^\circ\text{C} = 0$$

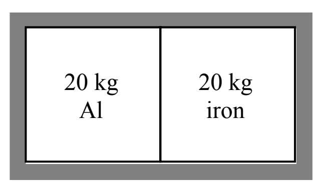
$$T_2 = \mathbf{27.5^\circ C}$$

1-35 An iron block at 100°C is brought into contact with an aluminum block at 200°C in an insulated enclosure. The final equilibrium temperature of the combined system is to be determined.

Assumptions **1** Both the iron and aluminum block are incompressible substances with constant specific heats. **2** The system is stationary and thus the kinetic and potential energy changes are zero, $\Delta KE = \Delta PE = 0$ and $\Delta E = \Delta U$. **3** The system is well-insulated and thus there is no heat transfer.

Properties The specific heat of iron is given in Table A-3 to be 0.45 kJ/kg.°C, which is the value at room temperature. The specific heat of aluminum at 450 K (which is somewhat below 200°C = 473 K) is 0.973 kJ/kg.°C.

Analysis We take the entire contents of the enclosure iron + aluminum blocks, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance on the system can be expressed as



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

$$0 = \Delta U$$

$$\Delta U_{\text{iron}} + \Delta U_{\text{Al}} = 0$$

or,
$$[mC(T_2 - T_1)]_{\text{iron}} + [mC(T_2 - T_1)]_{\text{Al}} = 0$$

Substituting,

$$(20 \text{ kg})(0.450 \text{ kJ / kg}\cdot{}^\circ\text{C})(T_2 - 100)^\circ\text{C} + (20 \text{ kg})(0.973 \text{ kJ / kg}\cdot{}^\circ\text{C})(T_2 - 200)^\circ\text{C} = 0$$

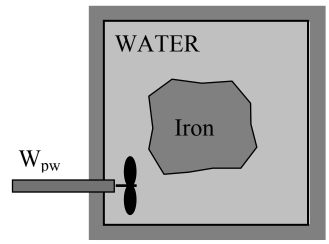
$$T_2 = \mathbf{168\ ^\circ C}$$

1-36 An unknown mass of iron is dropped into water in an insulated tank while being stirred by a 200-W paddle wheel. Thermal equilibrium is established after 25 min. The mass of the iron is to be determined.

Assumptions **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energy changes are zero, $\Delta KE = \Delta PE = 0$ and $\Delta E = \Delta U$. **3** The system is well-insulated and thus there is no heat transfer.

Properties The specific heats of water and the iron block at room temperature are $C_{p,\text{ water}}$ = 4.18 kJ/kg·°C and $C_{p,\text{ iron}}$ = 0.45 kJ/kg·°C (Tables A-3 and A-9). The density of water is given to be 1000 kg/m³.

Analysis We take the entire contents of the tank, water + iron block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance on the system can be expressed as



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

$$W_{\text{pw,in}} = \Delta U$$

or,

$$W_{\text{pw,in}} = \Delta U_{\text{iron}} + \Delta U_{\text{water}}$$

$$W_{\text{pw,in}} = [mC(T_2 - T_1)]_{\text{iron}} + [mC(T_2 - T_1)]_{\text{water}}$$

where

$$m_{\text{water}} = \rho V = (1000 \text{ kg/m}^3)(0.08 \text{ m}^3) = 80 \text{ kg}$$

$$W_{\text{pw}} = \dot{W}_{\text{pw}} \Delta t = (0.2 \text{ kJ/s})(25 \times 60 \text{ s}) = 300 \text{ kJ}$$

Using specific heat values for iron and liquid water and substituting,

$$(300\text{kJ}) = m_{\text{iron}}(0.45 \text{ kJ/kg}\cdot{}^\circ\text{C})(27-90)^\circ\text{C} + (80 \text{ kg})(4.18 \text{ kJ/kg}\cdot{}^\circ\text{C})(27-20)^\circ\text{C} = 0$$

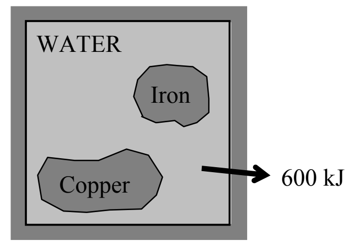
$$m_{\text{iron}} = \mathbf{72.1 \text{ kg}}$$

1-37E A copper block and an iron block are dropped into a tank of water. Some heat is lost from the tank to the surroundings during the process. The final equilibrium temperature in the tank is to be determined.

Assumptions **1** The water, iron, and copper blocks are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energy changes are zero, $\Delta KE = \Delta PE = 0$ and $\Delta E = \Delta U$.

Properties The specific heats of water, copper, and the iron at room temperature are $C_{p,\ water}$ = 1.0 Btu/lbm·°F, $C_{p,\ Copper}$ = 0.092 Btu/lbm·°F, and $C_{p,\ iron}$ = 0.107 Btu/lbm·°F (Tables A-3E and A-9E).

Analysis We take the entire contents of the tank, water + iron + copper blocks, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance on the system can be expressed as



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

$$-Q_{\text{out}} = \Delta U = \Delta U_{\text{copper}} + \Delta U_{\text{iron}} + \Delta U_{\text{water}}$$

or $$-Q_{\text{out}} = [mC(T_2 - T_1)]_{\text{copper}} + [mC(T_2 - T_1)]_{\text{iron}} + [mC(T_2 - T_1)]_{\text{water}}$$

Using specific heat values at room temperature for simplicity and substituting,

$$-600\text{Btu} = (90\text{lbm})(0.092\text{Btu/lbm}\cdot°\text{F})(T_2 - 160)°\text{F} + (50\text{lbm})(0.107\text{Btu/lbm}\cdot°\text{F})(T_2 - 200)°\text{F}$$
$$+ (180\text{lbm})(1.0\text{Btu/lbm}\cdot°\text{F})(T_2 - 70)°\text{F}$$

T_2 = **74.3 °F**

1-38 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 200-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, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **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.

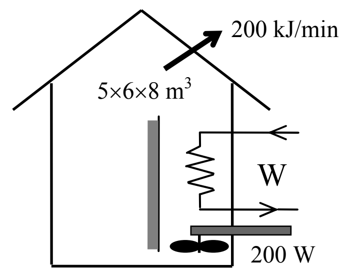
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_p = 1.007 kJ/kg·K for air at room temperature (Table A-15) and $C_v = C_p - R$ = 0.720 kJ/kg·K.

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

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} - \dot{W}_{\text{fan,in}} + mC_v(T_2 - T_1)/\Delta t$$

$$= (200/60\text{kJ/s}) - (0.2\text{kJ/s}) + (284.6\text{kg})(0.720\text{ kJ/kg}\cdot°\text{C})(25-15)°\text{C}/(15\times 60\text{s}) = \mathbf{5.41\ kW}$$

(*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,

$$\Delta T = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{\dot{m}C_p} = \frac{(5.41 + 0.2)\text{kJ/s}}{(50/60\text{kg/s})(1.007\text{ kJ/kg}\cdot\text{K})} = \mathbf{6.7°C}$$

1-39 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.

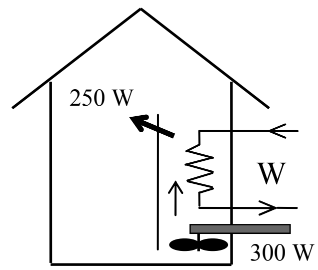
Properties The specific heat of air at room temperature is C_p = 1.007 kJ/kg·°C (Table A-15).

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 $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$. 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

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

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

$$\dot W_{\rm e,in} = \dot Q_{\rm out} - \dot W_{\rm fan,in} + \dot m C_p (T_2 - T_1)$$



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

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

1-40 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, $\Delta ke \cong \Delta pe \cong 0$. **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.

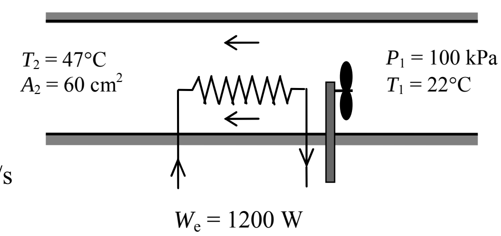
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, C_p = 1.007 kJ/kg·K for air at room temperature (Table A-15).

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

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

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

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



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,

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

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

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

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

$$\dot{m} = \frac{1}{v_2}A_2\mathbf{V}_2 \longrightarrow \mathbf{V}_2 = \frac{\dot{m}v_2}{A_2} = \frac{(0.04767 \text{ kg/s})(0.9187 \text{ m}^3/\text{kg})}{60\times10^{-4}\text{ m}^2} = \mathbf{7.30\ m/s}$$

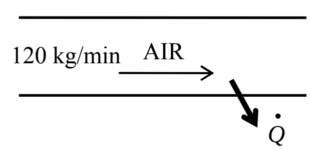
1-41 The ducts of an air heating system pass through an unheated area, resulting in a temperature drop of the air in the duct. The rate of heat loss from the air to the cold environment is to be determined.

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 $\Delta m_{CV} = 0$ 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

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



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

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

Substituting,

$$\dot{Q}_{\text{out}} = \dot{m}C_p\Delta T = (120\text{kg/min})(1.007\text{kJ/kg}\cdot°\text{C})(3°\text{C}) = \textbf{363 kJ/min}$$

1-42E 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, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air, C_p = 0.2404 and C_v = 0.1719 Btu/lbm·R. This assumption results in negligible error in heating and air-conditioning applications.

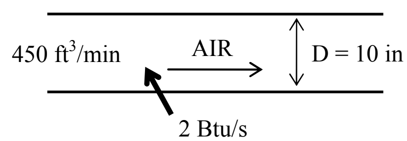
Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1). Also, C_p = 0.2404 Btu/lbm·R for air at room temperature (Table A-15E).

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

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

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

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



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

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

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510\text{R})}{(15\text{psia})} = 12.6\, ft^3\ /\ lbm$$

$$\dot m = \frac{\dot V_1}{v_1} = \frac{450\text{ft}^3/\text{min}}{12.6\text{ft}^3/\text{lbm}} = 35.7\text{lbm/min} = 0.595\text{lbm/s}$$

Then the exit temperature of air is determined to be

$$T_2 = T_1 + \frac{\dot Q_{in}}{\dot m C_p} = 50°\text{F} + \frac{2 \text{ Btu/s}}{(0.595\text{lbm/s})(0.2404\text{Btu/lbm}\cdot°\text{F})} = \textbf{64.0°F}$$

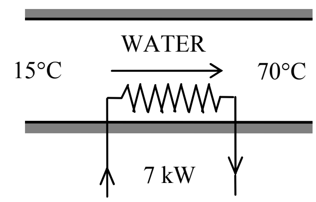
1-43 Water is heated in an insulated tube by an electric resistance heater. The mass flow rate of water through the heater is to be determined.

Assumptions **1** Water is an incompressible substance with a constant specific heat. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **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 (Table A-9).

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

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



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

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

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

Heat Transfer Mechanisms

1-44C 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.

1-45C 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.

1-46C 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.

1-47C 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.

1-48C Conduction is expressed by Fourier's law of conduction as $\dot{Q}_{cond} = -kA\dfrac{dT}{dx}$ 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 $\dot{Q}_{conv} = hA_s(T_s - T_\infty)$ 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 $\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^{\,4} - T_{surr}^{\,4})$ where ε is the emissivity of surface, A_s is the surface area, T_s is the surface temperature, T_{surr} is average surrounding surface temperature and $\sigma = 5.67 \times 10^{-8}\ \mathrm{W/m^2.K^4}$ is the Stefan-Boltzman constant.

1-49C Convection involves fluid motion, conduction does not. In a solid we can have only conduction.

1-50C No. It is purely by radiation.

1-51C 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.

1-52C 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.

1-53C 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.

1-54C 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.m/m^2.°C that indicates product of heat transfer rate and thickness per unit surface area per unit temperature difference.

1-55C 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.

1-56C Diamond is a better heat conductor.

1-57C The rate of heat transfer through both walls can be expressed as

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

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

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

1-58C 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.

1-59C 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.

1-60C 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.

1-61C The thermal conductivity of an alloy of two metals will most likely be less than the thermal conductivities of both metals.

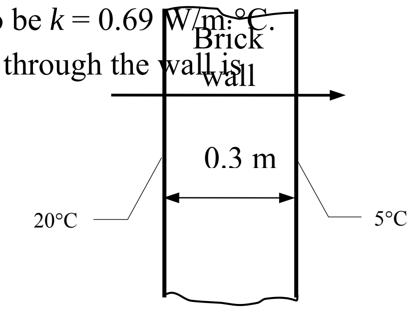
1-62 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of heat transfer through the wall is to be determined.

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

$$\dot{Q}_{cond} = kA\frac{\Delta T}{L} = (0.69\text{W/m}\cdot°\text{C})(5\times 6\text{m}^2)\frac{(20-5)°\text{C}}{0.3\text{m}} = \mathbf{1035W}$$



1-63 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.

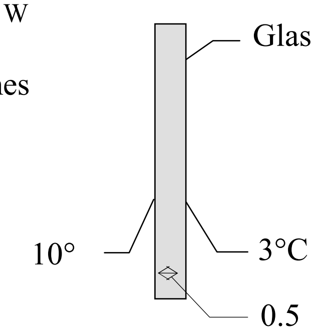
Analysis Under steady conditions, the rate of heat transfer through the glass by conduction is

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

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

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

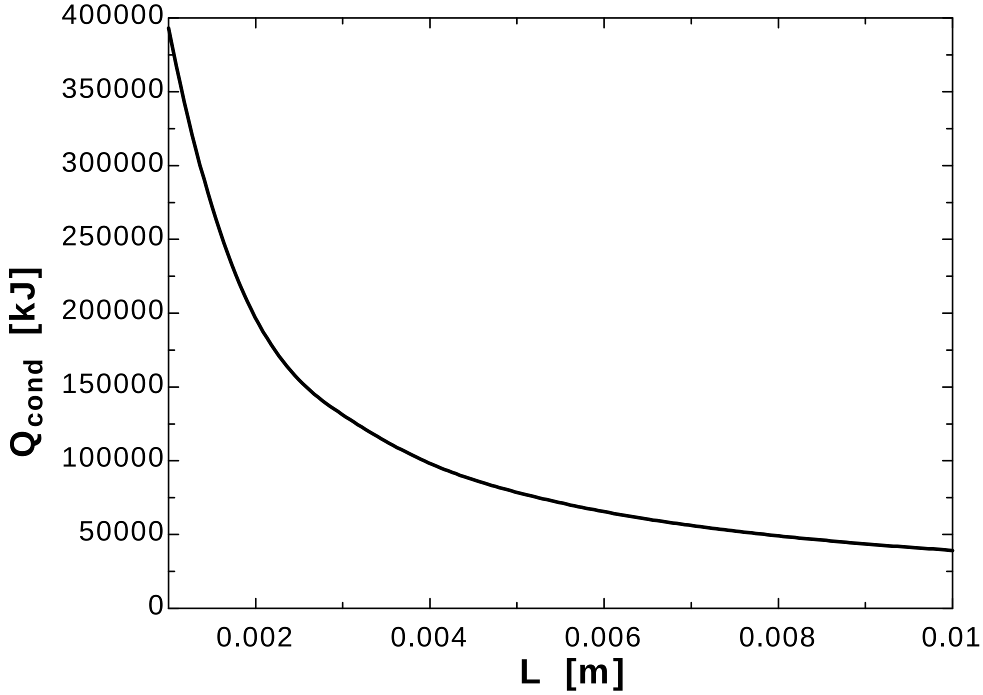
If the thickness of the glass doubled to 1 cm, then the amount of heat transfer will go down by half to **39,310 kJ**.



1-64

```
"GIVEN"
"L=0.005 [m], parameter to be varied"
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


400000
350000
300000
250000
200000
150000
100000
50000
0
0.002
0.004
0.006
0.008
0.01
L [m]
Q_{cond} [kJ]

1-65 Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface of the bottom of the pan is given. The temperature of the outer surface is to be determined.

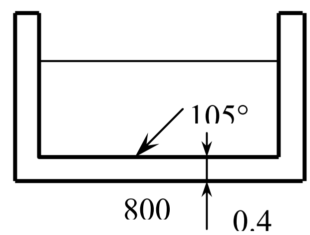
Assumptions **1** Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values. **2** Thermal properties of the aluminum pan are constant.

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

Analysis The heat transfer area is

$$A = \pi r^2 = \pi(0.1 \text{ m})^2 = 0.0314 \text{ m}^2$$

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,

$$800\text{W} = (237\text{W/m}\cdot{}^\circ\text{C})(0.0314\text{m}^2)\frac{T_2 - 105^\circ\text{C}}{0.004\text{m}}$$

which gives

$$T_2 = \mathbf{105.43\ ^\circ C}$$

1-66E 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.

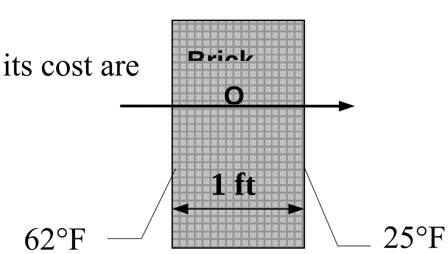
Properties The thermal conductivity of the brick wall is given to be $k = 0.42$ Btu/h.ft.°F.

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

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

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



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

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

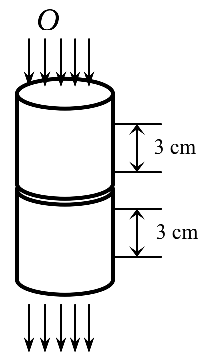
Therefore, the cost of the heat loss through the wall to the home owner that night is \$0.51.

1-67 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

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



The rate of heat flow through each sample is

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

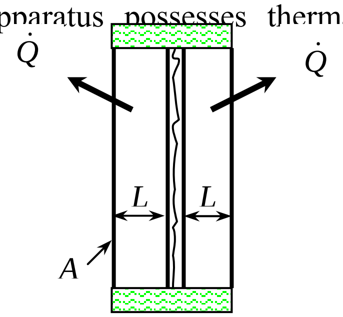
Then the thermal conductivity of the sample becomes

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

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

1-68 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

$$\dot{Q} = 35/2 = 17.5\text{ W}$$

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

$$\Delta T = 82 - 74 = 8°\text{C}$$

Then the thermal conductivity of the material becomes

$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(17.5\text{ W})(0.005\text{ m})}{(0.01\text{ m}^2)(8°\text{C})} = \mathbf{1.09\ W/m.°C}$$

1-69 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.

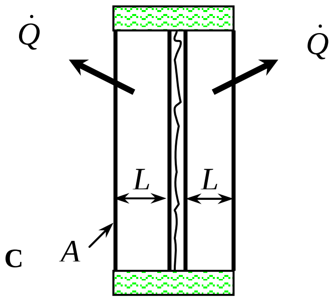
Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. **2** Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples.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

Analysis For each sample we have

$$\dot{Q} = 28/2 = 14 \text{ W}$$

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

$$\Delta T = 82 - 74 = 8°\text{C}$$



Then the thermal conductivity of the material becomes

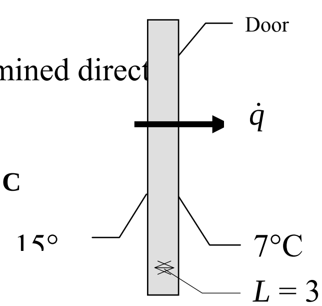
$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(14 \text{ W})(0.005 \text{ m})}{(0.01 \text{ m}^2)(8°\text{C})} = \mathbf{0.875 \text{ W/m.°C}}$$

1-70 The thermal conductivity of a refrigerator door is to be determined by measuring the surface temperatures and heat flux when steady operating conditions are reached.

Assumptions 1 Steady operating conditions exist when measurements are taken. **2** Heat transfer through the door is one dimensional since the thickness of the door is small relative to other dimensions.

Analysis The thermal conductivity of the door material is determined direct from Fourier's relation to be

$$\dot{q} = k\frac{\Delta T}{L} \longrightarrow k = \frac{\dot{q}L}{\Delta T} = \frac{(25 \text{ W/m}^2)(0.03 \text{ m})}{(15-7)°\text{C}} = \mathbf{0.09375 \text{ W/m.°C}}$$



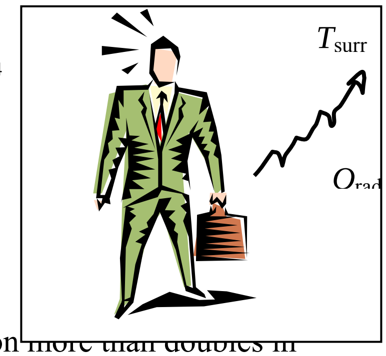
1-71 The rate of radiation heat transfer between a person and the surrounding surfaces at specified temperatur es is to be determined in summer and in winter.

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

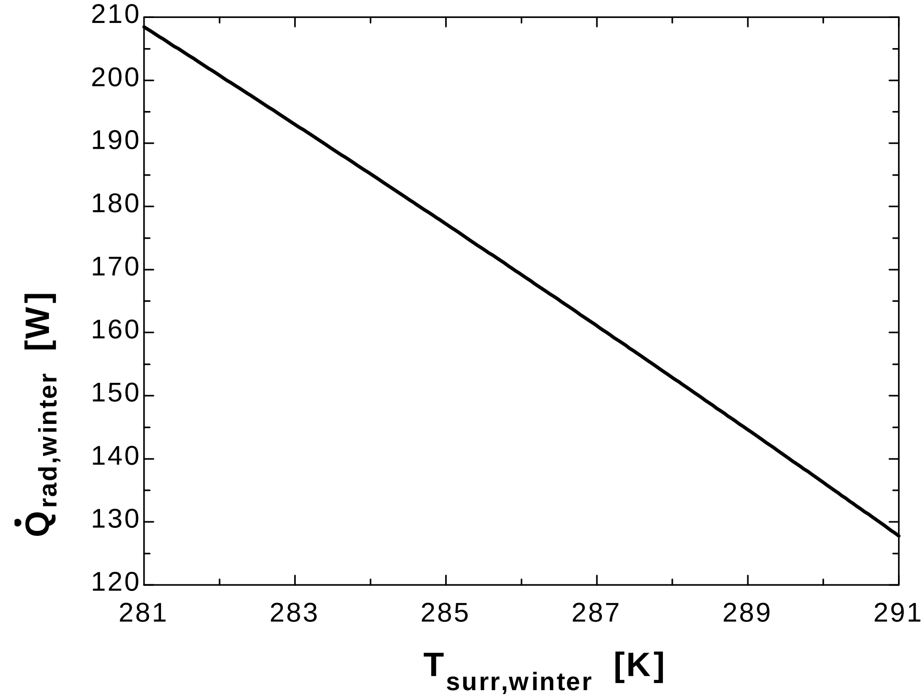
$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4) \\ &= (0.95)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.6\ \text{m}^2)[(32+273)^4 - (285\ \text{K})^4]\text{K}^4 \\ &= \mathbf{177.2\ W}\end{aligned}$$

Discussion Note that the radiation heat transfer from the person more than doubles in winter.

1-72

```
"GIVEN"
T_infinity=20+273 "[K]"
"T_surr_winter=12+273 [K], parameter to be varied"
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"
"(a)"
Q_dot_rad_summer=epsilon*sigma*A*(T_s^4-T_surr_summer^4)
"(b)"
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

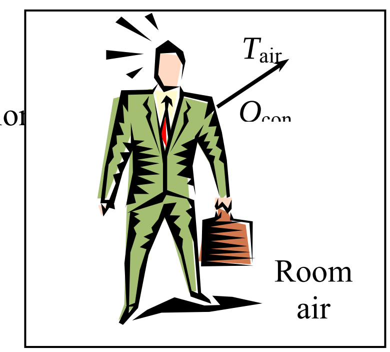


1-73 A person is standing in a room at a specified temperature. The rate of heat transfer between a person and the surrounding air by convection is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The environment is at a uniform temperature.

Analysis The heat transfer surface area of the person is

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



Under steady conditions, the rate of heat transfer by convection

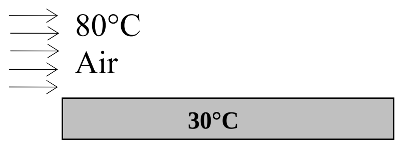
$$\dot{Q}_{conv} = hA_s\Delta T = (15 \text{W/m}^2\cdot^\circ\text{C})(1.60\text{m}^2)(34-20)^\circ\text{C} = \mathbf{336 W}$$

1-74 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

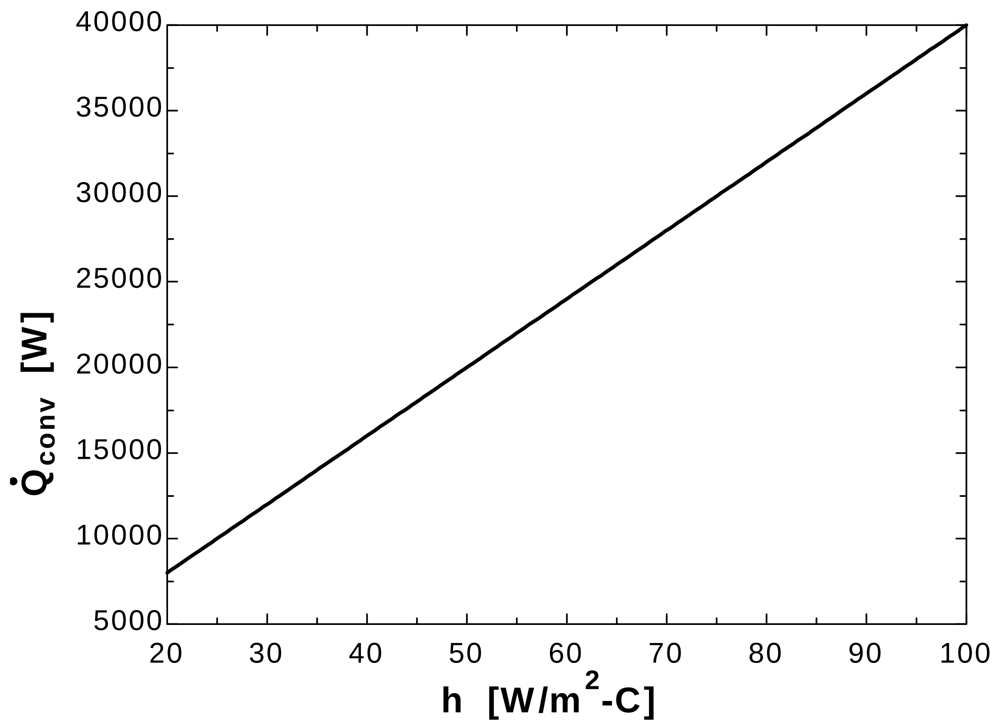
$$\dot{Q}_{conv} = hA_s\Delta T = (55 \text{W/m}^2\cdot^\circ\text{C})(2\times 4\text{m}^2)(80-30)^\circ\text{C} = \mathbf{22{,}000W}$$



1-75

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

"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



1-76 The heat generated in the circuitry on the surface of a 3-W silicon chip is conducted to the ceramic substrate. The temperature difference across the chip in steady operation is to be determined.

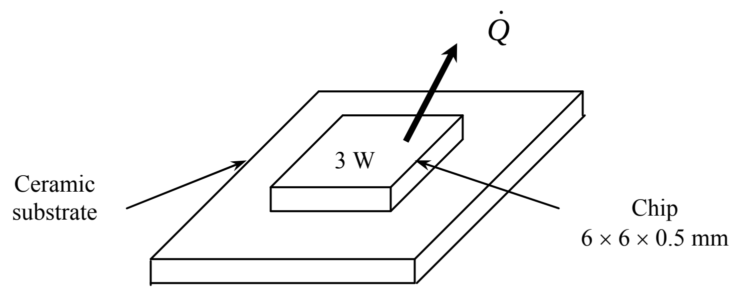
Assumptions **1** Steady operating conditions exist. **2** Thermal properties of the chip are constant.

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

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

$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow \Delta T = \frac{\dot{Q}L}{kA} = \frac{(3\text{ W})(0.0005\text{ m})}{(130\text{ W/m.°C})(0.000036\text{ m}^2)} = \mathbf{0.32\ °C}$$



1-77 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-2).

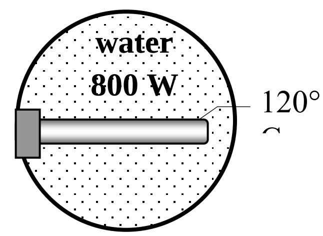
Analysis When steady operating conditions are reached, we have $\dot{Q} = \dot{E}_{\text{generated}} = 800\ \text{W}$.

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

$$Q_{in} = mC(T_2 - T_1)$$

$$\dot{Q}_{in}\Delta t = mC(T_2 - T_1)$$

$$\Delta t = \frac{mC(T_2 - T_1)}{\dot{Q}_{in}} = \frac{(60\ \text{kg})(4180\ \text{J/kg.°C})(80-20)\text{°C}}{800\ \text{J/s}} = 18{,}810\ \text{s} = \mathbf{5.225\ h}$$



The surface area of the wire is

$$A_s = (\pi D)L = \pi(0.005\ \text{m})(0.5\ \text{m}) = 0.00785\ \text{m}^2$$

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

$$h_1 = \frac{\dot{Q}}{A_s(T_s - T_{\infty 1})} = \frac{800\ \text{W}}{(0.00785\ \text{m}^2)(120-20)\text{°C}} = \mathbf{1020\ W/m^2.°C}$$

$$h_2 = \frac{\dot{Q}}{A_s(T_s - T_{\infty 2})} = \frac{800\ \text{W}}{(0.00785\ \text{m}^2)(120-80)\text{°C}} = \mathbf{2550\ W/m^2.°C}$$

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

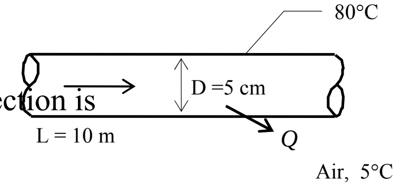
1-78 A hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 W/ m^2.°C. The rate of heat loss from the pipe by convection is to be determined.

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



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

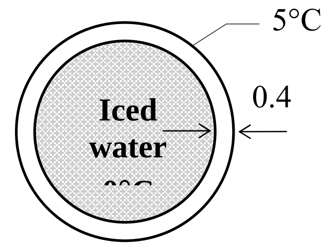
1-79 A hollow spherical iron container is filled with iced water at 0°C. The rate of heat loss from the sphere and the rate at which ice melts in the container are to be determined.

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

$$\dot{Q}_{cond} = kA\frac{\Delta T}{L} = (80.2\text{W/m}\cdot°\text{C})(0.126\text{m}^2)\frac{(5-0)°\text{C}}{0.004\text{m}} = 12{,}632\text{W}$$

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

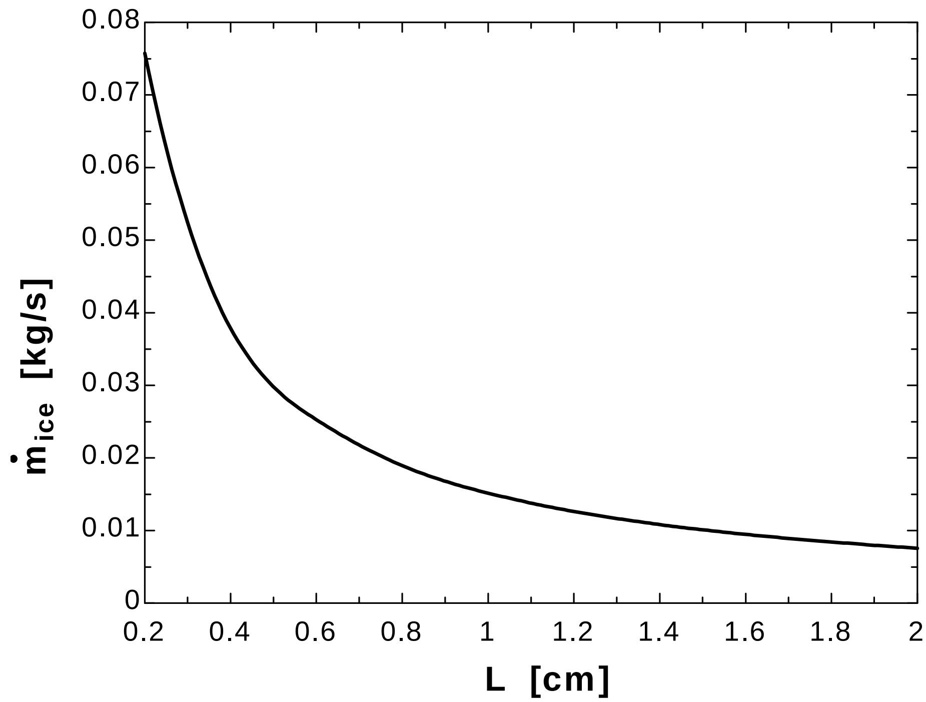
$$\dot{m}_{ice} = \frac{\dot{Q}}{h_{if}} = \frac{12.632 \text{ kJ/s}}{333.7 \text{ kJ/kg}} = \mathbf{0.038 \text{ kg/s}}$$

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 = 19.2 cm) or the mean surface area (D = 19.6 cm) in the calculations.

1-80

```
"GIVEN"
D=0.2 "[m]"
"L=0.4 [cm], parameter to be varied"
T_1=0 "[C]"
T_2=5 "[C]"

"PROPERTIES"
h_if=333.7 "[kJ/kg]"
k=k_('Iron', 25) "[W/m-C]"

"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]
0.2	0.07574
0.4	0.03787
0.6	0.02525
0.8	0.01894
1	0.01515
1.2	0.01262
1.4	0.01082
1.6	0.009468
1.8	0.008416
2	0.007574

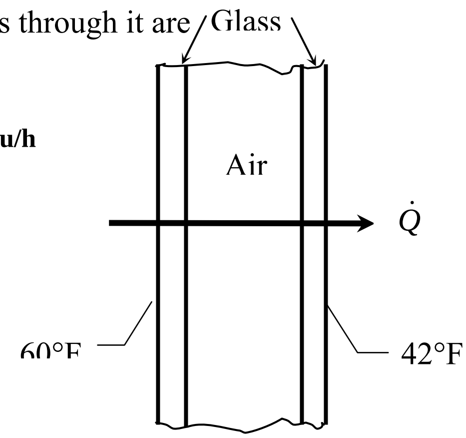


1-81E The inner and outer glasses of a double pane window with a 0.5-in air space are at specified temperatures. The rate of heat transfer through the window is to be determined

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+42)/2 = 51°F is $k = 0.01411$ Btu/h.ft.°F (Table A-15).

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



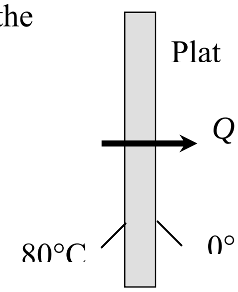
$$A = (6\text{ ft}) \times (6\text{ ft}) = 36\text{ m}^2$$

$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.01411\text{ Btu/h.ft.°F})(36\text{ ft}^2)\frac{(60-42)\text{°F}}{0.25/12\text{ ft}} = \mathbf{439\text{ Btu/h}}$$

1-82 Two surfaces of a flat plate are maintained at specified temperatures, and the rate of heat transfer through the plate is measured. The thermal conductivity of the plate material is to be determined.

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

Analysis The thermal conductivity is determined directly from the steady one-dimensional heat conduction relation to be



$$\dot{Q} = kA\frac{T_1 - T_2}{L} \rightarrow k = \frac{\dot{Q}/A}{L(T_1 - T_2)} = \frac{500 \text{ W/m}^2}{(0.02 \text{ m})(80 - 0)°\text{C}} = \mathbf{313 \text{ W/m.°C}}$$

1-83 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.

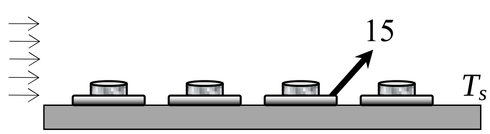
Analysis The total rate of heat dissipation from the aluminum plate and the total heat transfer area are

$$\dot{Q} = 4 \times 15 \text{ W} = 60 \text{ W}$$

$$A_s = (0.22 \text{ m})(0.22 \text{ m}) = 0.0484 \text{ m}^2$$

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

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 25°\text{C} + \frac{60 \text{ W}}{(25 \text{ W/m}^2.°\text{C})(0.0484 \text{ m}^2)} = \mathbf{74.6°C}$$



1-84 A styrofoam ice chest is initially filled with 40 kg of ice at 0°C. The time it takes for the ice in the chest to melt completely is to be determined.

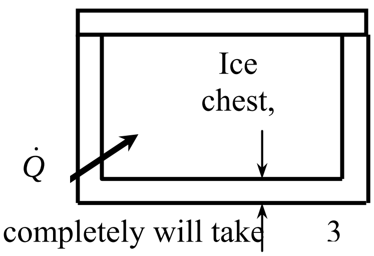
Assumptions **1** Steady operating conditions exist. **2** The inner and outer surface temperatures of the ice chest remain constant at 0°C and 8°C, respectively, at all times. **3** Thermal properties of the chest are constant. **4** Heat transfer from the base of the ice chest is negligible.

Properties The thermal conductivity of the styrofoam is given to be k = 0.033 W/m·°C. The heat of fusion of ice at 0°C is 333.7 kJ/kg.

Analysis Disregarding any heat loss through the bottom of the ice chest and using the average thicknesses, the total heat transfer area becomes

$$A = (40-3)(40-3) + 4\times(40-3)(30-3) = 5365 \text{ cm}^2 = 0.5365 \text{ m}^2$$

The rate of heat transfer to the ice chest becomes



$$\dot{Q} = kA\frac{\Delta T}{L} = (0.033 \text{ W/m.°C})(0.5365 \text{ m}^2)\frac{(8-0)\text{°C}}{0.03 \text{ m}} = 4.72 \text{ W}$$

The total amount of heat needed to melt the ice completely is

$$Q = mh_{if} = (40 \text{ kg})(333.7 \text{ kJ/kg}) = 13{,}348 \text{ kJ}$$

Then transferring this much heat to the cooler to melt the ice completely will take

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{13{,}348{,}000 \text{ J}}{4.72 \text{ J/s}} = 2{,}828{,}000 \text{ s} = 785.6 \text{ h} = \mathbf{32.7 \text{ days}}$$

1-85 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.

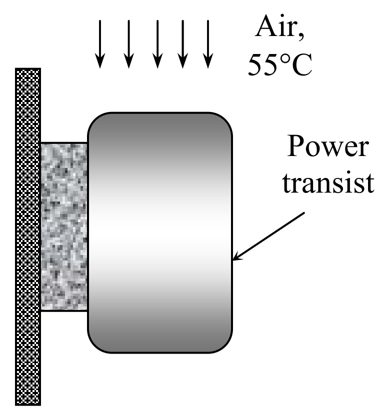
Analysis Disregarding the base area, the total heat transfer area of the transistor is

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

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

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

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

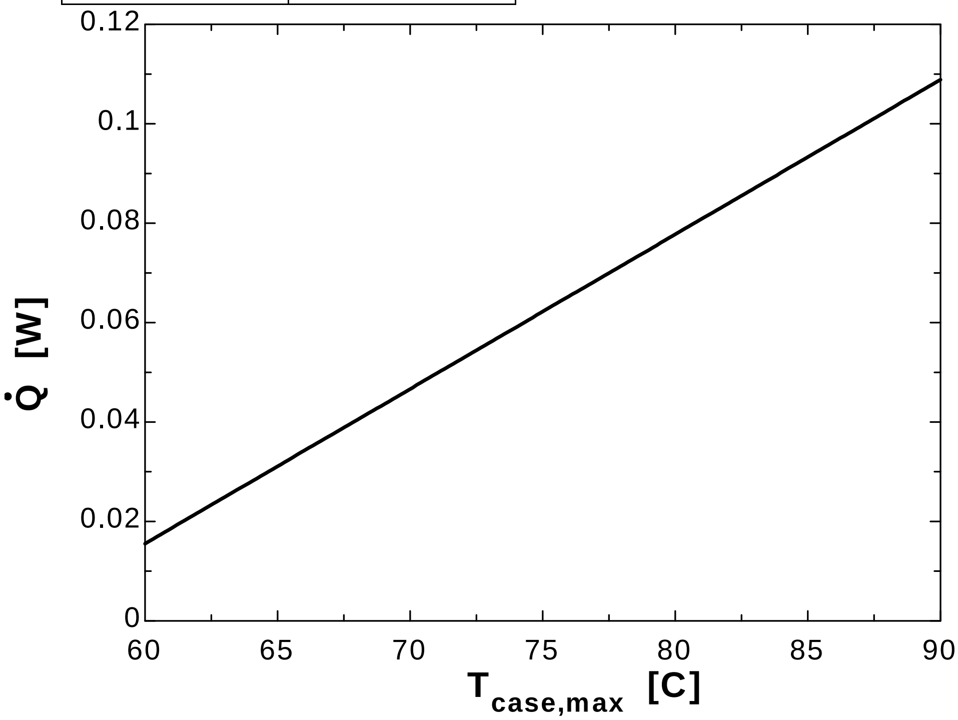


1-86

"GIVEN"
L=0.004 "[m]"
D=0.006 "[m]"
h=30 "[W/m^2-C]"
T_infinity=55 "[C]"
"T_case_max=70 [C], parameter to be varied"

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

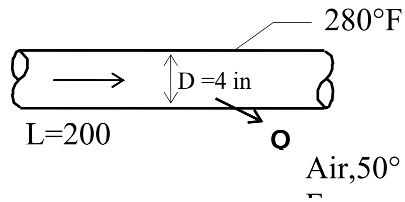
$T_{case,\ max}$ [C]	Q [W]
60	0.01555
62.5	0.02333
65	0.0311
67.5	0.03888
70	0.04665
72.5	0.05443
75	0.0622
77.5	0.06998
80	0.07775
82.5	0.08553
85	0.09331
87.5	0.1011
90	0.1089



1-87E A 200-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



$$A_s = \pi DL = \pi(4/12\text{ ft})(200\text{ ft}) = 209.4\text{ ft}^2$$

$$\dot{Q}_{\text{pipe}} = hA_s(T_s - T_{air}) = (6\text{ Btu/h.ft}^2.°\text{F})(209.4\text{ ft}^2)(280-50)°\text{F}$$
$$= \mathbf{289{,}000\ Btu/h}$$

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

$$Q = \dot{Q}\Delta t = (289{,}000\text{ Btu/h})(365\times 24\text{ h/yr}) = 2.532\times 10^9\text{ Btu/yr}$$

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

$$\text{Annual Energy Loss} = \frac{2.532\times 10^9\text{ Btu/yr}}{0.86}\left(\frac{1\text{ therm}}{100{,}000\text{ Btu}}\right) = 29{,}438\text{ therms/yr}$$

Then the annual cost of the energy lost becomes

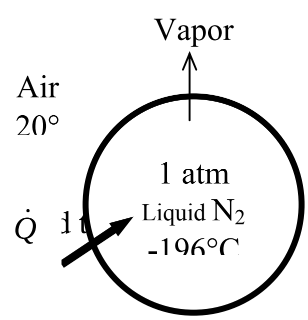
$$\text{Energy cost} = (\text{Annual energy loss})(\text{Unit cost of energy})$$
$$= (29{,}438\text{ therms/yr})(\$0.58/\text{therm}) = \mathbf{\$17{,}074/yr}$$

1-88 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



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

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

Then the rate of evaporation of liquid nitrogen in the tank is deter

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

1-89 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.

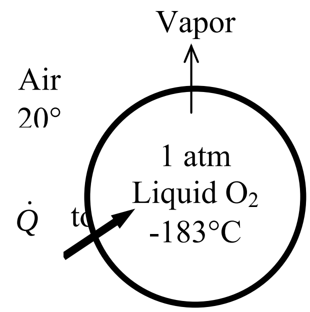
Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is disregarded. **3** The convection heat transfer coefficient is constant and uniform over the surface. **4** The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the oxygen inside.

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

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

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



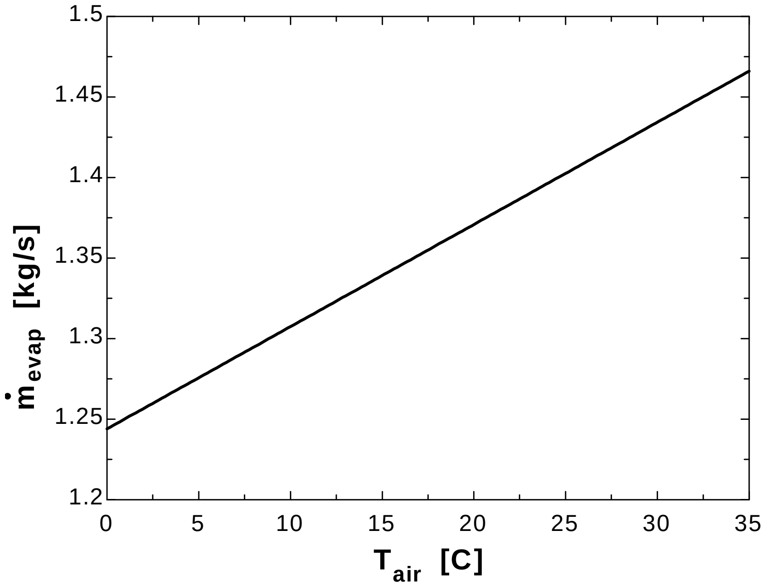
Then the rate of evaporation of liquid oxygen in the tank is deterr to

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

1-90

```
"GIVEN"
D=4 "[m]"
T_s=-196 "[C]"
"T_air=20 [C], parameter to be varied"
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


1.5
1.45
1.4
1.35
1.3
1.25
1.2
ṁevap [kg/s]
0
5
10
15
20
25
30
35
Tair [C]

1-91 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 0.7.

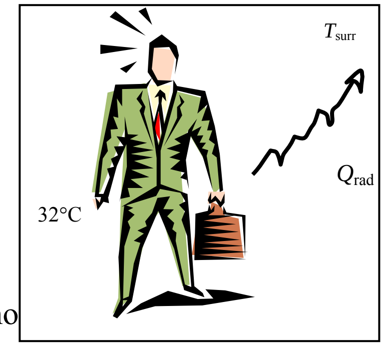
Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are

(*a*) $T_{\text{surr}} = 300$ K

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s(T_s^4 - T_{\text{surr}}^4)\\ &= (0.7)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.7\ \text{m}^2)[(32+273)^4 - (300\ \text{K})^4]\text{K}^4\\ &= \mathbf{37.4\ W}\end{aligned}$$

(*b*) $T_{\text{surr}} = 280$ K

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon\sigma A_s(T_s^4 - T_{\text{surr}}^4)\\ &= (0.7)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.7\ \text{m}^2)[(32+273)^4 - (280\ \text{K})^4]\text{K}^4\\ &= \mathbf{169\ W}\end{aligned}$$



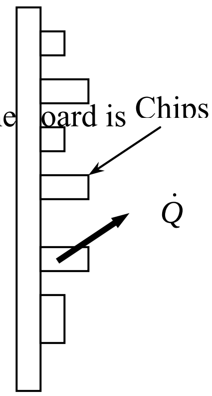
Discussion Note that the radiation heat transfer goes up by mo as the temperature of the surrounding surfaces drops from 300 K to 280 K.

1-92 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

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

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

Discussion Note that the circuit board is nearly isothermal.

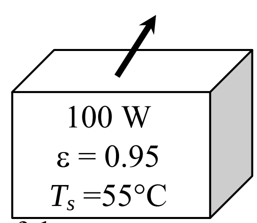
1-93 A sealed electronic box dissipating a total of 100 W of power is placed in a vacuum chamber. If this box is to be cooled by radiation alone and the outer surface temperature of the box is not to exceed 55°C, the temperature the surrounding surfaces must be kept is to be determined.

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

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

$$100\text{ W} = (0.95)(5.67 \times 10^{-8}\text{ W/m}^2.\text{K}^4)(0.48\text{ m}^2)\left[(55 + 273\text{ K})^4 - T_{surr}^{\,4}\right]$$

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

1-94 Using the conversion factors between W and Btu/h, m and ft, and K and R, the Stefan-Boltzmann constant $\sigma = 5.67 \times 10^{-8}\ \mathrm{W/m^2.K^4}$ is to be expressed in the English unit, $\mathrm{Btu/h.ft^2.R^4}$.

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

$$1\ \mathrm{W} = 3.41214\ \mathrm{Btu/h}$$
$$1\ \mathrm{m} = 3.2808\ \mathrm{ft}$$
$$1\ \mathrm{K} = 1.8\ \mathrm{R}$$

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

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

1-95 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

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

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

$$1°\mathrm{C} = 1.8°\mathrm{F}$$

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

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

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

$$h = 20\ \mathrm{W/m^2.°C} = 20 \times 0.1761\ \mathrm{Btu/h.ft^2.°F} = \mathbf{3.52\ Btu/h.ft^2.°F}$$

Simultaneous Heat Transfer Mechanisms

1-96C All three modes of heat transfer can not occur simultaneously in a medium. A medium may involve two of them simultaneously.

1-97C (*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.

1-98C 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.

1-99C 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.

1-100 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.

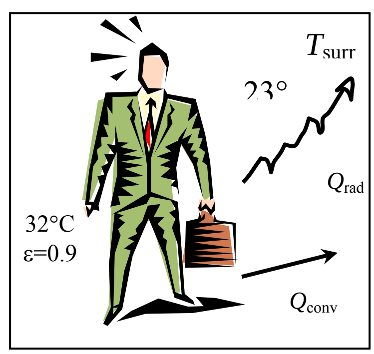
Analysis The person is completely enclosed by the surrounding surfaces, and he or she will lose heat to the surrounding air by convection, and to the surrounding surfaces by radiation. The total rate of heat loss from the person is determined from

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A_s(T_s^4 - T_{\text{surr}}^4) = (0.90)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.7\ \text{m}^2)[(32+273)^4-(23+273)^4]\text{K}^4$$
$$= 84.8\ \text{W}$$

$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (5\text{W/m}^2\cdot\text{K})(1.7\text{m}^2)(32-23)°\text{C} = 76.5\text{W}$$

and

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 84.8 + 76.5 = \mathbf{161.3\ W}$$

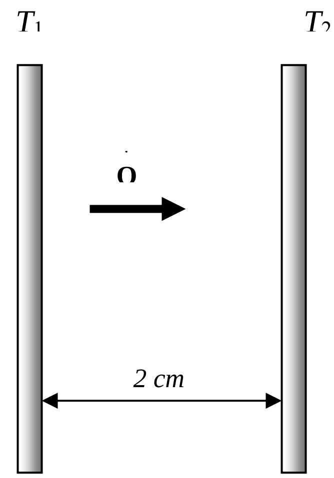


Discussion Note that heat transfer from the person by evaporation, which is of comparable magnitude, is not considered in this problem.

1-101 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, 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-16).



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

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

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

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

(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,

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

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

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

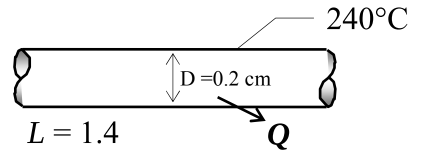
Discussion Note that superinsulators are very effective in reducing heat transfer between to surfaces.

1-102 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,

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



The surface area of the wire is

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

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

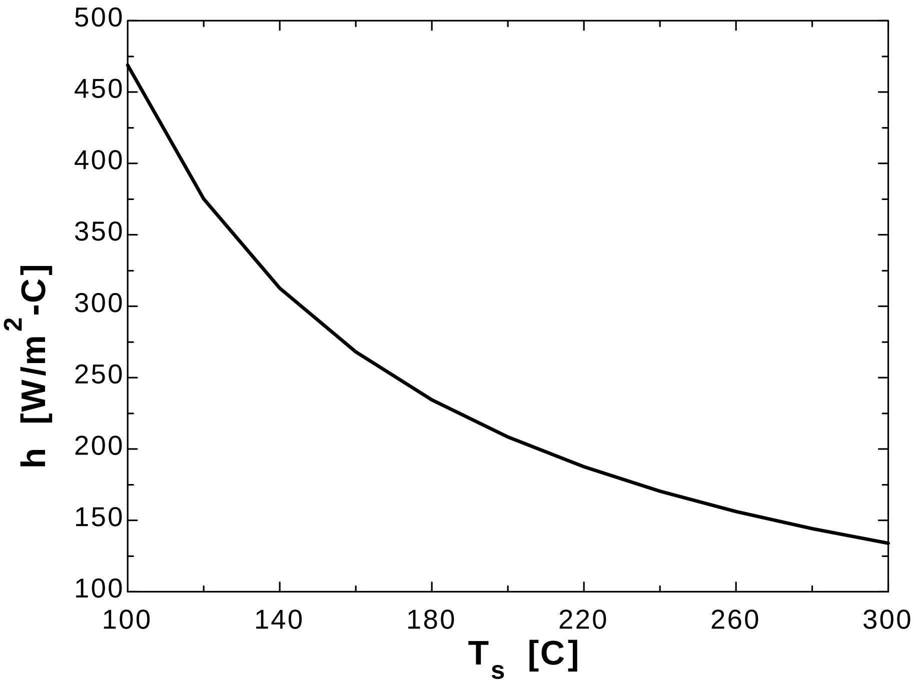
$$h = \frac{\dot{Q}}{A_s(T_1 - T_\infty)} = \frac{330 \text{ W}}{(0.00880 \text{ m}^2)(240-20)°\text{C}} = \mathbf{170.5\ W/m^2.°C}$$

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.

1-103

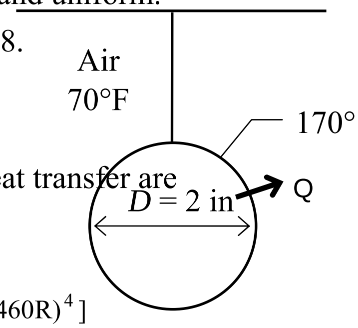
```
"GIVEN"
L=1.4 "[m]"
D=0.002 "[m]"
T_infinity=20 "[C]"
"T_s=240 [C], parameter to be varied"
V=110 "[Volt]"
I=3 "[Ampere]"

"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	468.9
120	375.2
140	312.6
160	268
180	234.5
200	208.4
220	187.6
240	170.5
260	156.3
280	144.3
300	134



1-104E 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 $\varepsilon = 0.8$.

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

$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (12 \text{Btu/h.ft}^2\cdot{}^\circ\text{F})(0.08727\text{ft}^2)(170-70)^\circ\text{F} = 104.7 \text{ Btu/h}$$

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

Therefore, $\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 104.7 + 9.4 = \mathbf{114.1 \ Btu/h}$

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.

1-105 A 1000-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 1000 W energy supplied to the iron will be dissipated to the surroundings by convection and radiation heat transfer. Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 1000 \text{ W}$$

where

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

and

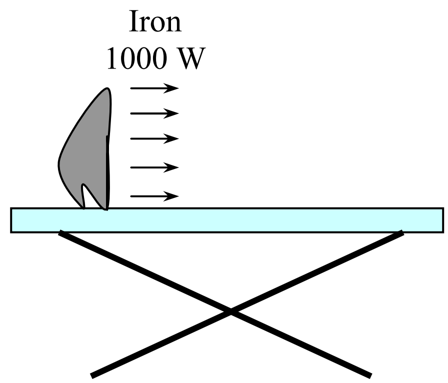
$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A_s(T_s^4 - T_o^4) = 0.6(0.02\text{m}^2)(5.67\times10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[\text{T}_\text{s}^4 - (293\text{K})^4]$$
$$= 0.06804\times10^{-8}[T_s^4 - (293\text{K})^4] \text{ W}$$

Substituting,

$$1000 \text{ W} = 0.7(T_s - 293 \text{ K}) + 0.06804\times10^{-8}[T_s^4 - (293 \text{ K})^4]$$

Solving by trial and error gives

$$T_s = \mathbf{947\ K = 674^\circ\ C}$$



Discussion We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 947 K.

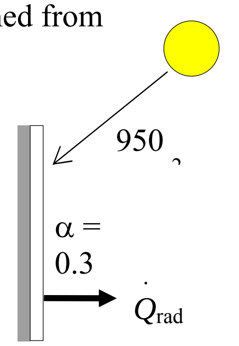
1-106 A spacecraft in space absorbs solar radiation while losing heat to deep space by thermal radiation. The surface temperature of the spacecraft is to be determined when steady conditions are reached..

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

$$\dot{Q}_{\text{solar absorbed}} = \dot{Q}_{\text{rad}}$$
$$\alpha\dot{Q}_{\text{solar}} = \varepsilon\sigma A_s(T_s^4 - T_{\text{space}}^4)$$
$$0.3\times A_s\times(950\text{W/m}^2) = 0.8\times A_s\times(5.67\times10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[\text{T}_\text{s}^4 - (0\text{K})^4]$$



Canceling the surface area A and solving for T_s gives

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

1-107 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 $h_{if} = 333.7 \text{ kJ/kg}$. The emissivity of the outer surface of the tank is 0.6.

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

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

$$\dot{Q}_{\text{rad}} = \varepsilon A_s \sigma(T_{surr}^4 - T_s^4) = (0.6)(28.65 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2.\text{K}^4)[(288 \text{ K})^4 - (273 \text{ K})^4] = 1292 \text{ W}$$

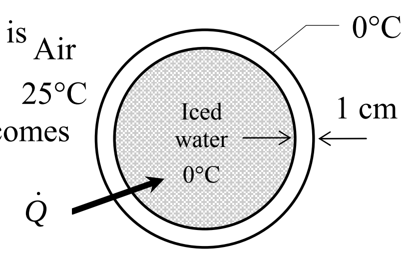
$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 21{,}488 + 1292 = 22{,}780 \text{ W}$$

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

$$Q = \dot{Q}\Delta t = (22.78 \text{ kJ/s})(24\times3600 \text{ s}) = 1{,}968{,}000 \text{ kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{1{,}968{,}000 \text{ kJ}}{333.7 \text{ kJ/kg}} = \mathbf{5898 \text{ kg}}$$



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

1-108 The roof of a house with a gas furnace consists of a 15-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.

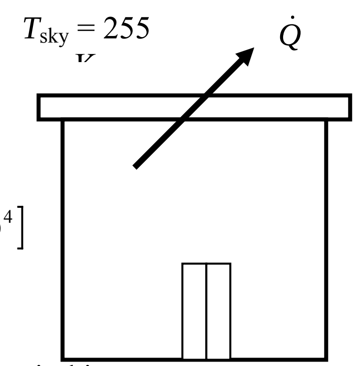
Assumptions **1** Steady operating conditions exist. **2** The emissivity and thermal conductivity of the roof are constant.

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



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

$$\dot{Q} = (2\ \text{W/m.°C})(300\ \text{m}^2)\frac{15\text{°C} - T_{s,out}}{0.15\ \text{m}}$$

$$= (15\ \text{W/m}^2\text{.°C})(300\ \text{m}^2)(T_{s,out} - 10)\text{°C}$$

$$+ (0.9)(300\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\text{.K}^4)\left[(T_{s,out} + 273\ \text{K})^4 - (255\ \text{K})^4\right]$$

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

$$\dot{Q} = \mathbf{25{,}450\ W} \ \text{ and } \ T_{s,\ out} = \mathbf{8.64°C}$$

Then the amount of natural gas consumption during a 1-hour period is

$$E_{gas} = \frac{Q_{total}}{0.85} = \frac{\dot{Q}\Delta t}{0.85} = \frac{(25.450\ \text{kJ/s})(14\times3600\ \text{s})}{0.85}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = 14.3\ \text{therms}$$

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

$$\text{Money lost} = (14.3\ \text{therms})(\$0.60/\text{therm}) = \mathbf{\$8.58}$$

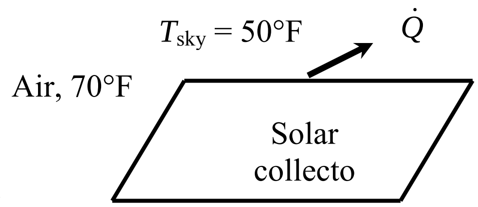
1-109E A flat plate solar collector is placed horizontally on the roof of a house. The rate of heat loss from the collector by convection and radiation during a calm day are to be determined.

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 the environment by convection and radiation becomes

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

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

and

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

Problem Solving Techniques and EES

1-110C 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. 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.

1-111 Determine a positive real root of the following equation using EES:

$$2x^3 - 10x^{0.5} - 3x = -3$$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
2*x^3-10*x^0.5-3*x = -3
```

Answer: x = 2.063 (using an initial guess of x=2)

1-112 Solve the following system of 2 equations with 2 unknowns using EES:

$$x^3 - y^2 = 7.75$$

$$3xy + y = 3.5$$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
x^3-y^2=7.75
3*x*y+y=3.5
```

Answer x=2 y=0.5

1-113 Solve the following system of 3 equations with 3 unknowns using EES:

$2x - y + z = 5$

$3x^2 + 2y = z + 2$

$xy + 2z = 8$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

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

Answer x=1.141, y=0.8159, z=3.535

1-114 Solve the following system of 3 equations with 3 unknowns using EES:

$x^2y - z = 1$

$x - 3y^{0.5} + xz = -2$

$x + y - z = 2$

Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
x^2*y-z=1
x-3*y^0.5+x*z=-2
x+y-z=2
```

Answer x=1, y=1, z=0

Special Topic: Thermal Comfort

1-115C 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.

1-116C 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.

1-117C 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.

1-118C (*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.

1-119C 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).

1-120C 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.

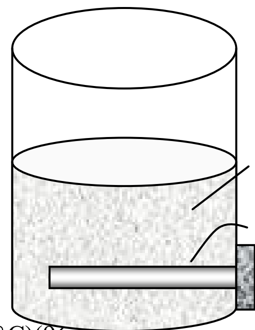
Review Problems

1-121 Cold water is to be heated in a 1200-W teapot. The time needed to heat the water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Thermal properties of the teapot and the water are constant. **3** Heat loss from the teapot is negligible.

Properties The average specific heats are given to be 0.6 kJ/kg.°C for the teapot and 4.18 kJ/kg.°C for water.

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



$$E_{in} - E_{out} = \Delta E_{system}$$

$$E_{in} = \Delta U_{system} = \Delta U_{water} + \Delta U_{tea\ pot}$$

Then the amount of energy needed to raise the temperature of water and the teapot from 18°C to 96°C is

$$\begin{aligned} E_{in} &= (mC\Delta T)_{water} + (mC\Delta T)_{teapot} \\ &= (2.5\text{ kg})(4.18\text{ kJ / kg.°C})(96-18)\text{°C} + (0.8\text{ kg})(0.6\text{ kJ / kg.°C})(96-18)\text{°C} \\ &= 853\text{ kJ} \end{aligned}$$

The 1500 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 853 kJ of heat is determined from

$$\Delta t = \frac{\text{Total energy transferred}}{\text{Rate of energy transfer}} = \frac{E_{in}}{\dot{E}_{transfer}} = \frac{853\text{ kJ}}{1.2\text{ kJ/s}} = 710\text{ s} = \mathbf{11.8\ min}$$

Discussion In reality, it will take longer to accomplish this heating process since some heat loss is inevitable during the heating process.

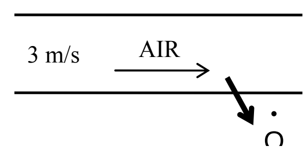
1-122 The duct of an air heating system of a house passes through an unheated space in the attic. The rate of heat loss from the air in the duct to the attic and its cost under steady conditions are to be determined.

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** Steady operating conditions exist since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** 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 kJ/kg.K (Table A-1). 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. There is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the energy balance for this steady-flow system can be expressed in the rate form as

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



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

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

The density of air at the inlet conditions is determined from the ideal gas relation to be

$$\rho = \frac{P}{RT} = \frac{100 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(65+273)\text{K}} = 1.031 \text{ kg/m}^3$$

The cross-sectional area of the duct is

$$A_c = \pi D^2/4 = \pi(0.20 \text{ m})^2/4 = 0.0314 \text{ m}^2$$

Then the mass flow rate of air through the duct and the rate of heat loss become

$$\dot{m} = \rho A_c \mathbf{V} = (1.031 \text{ kg/m}^3)(0.0314 \text{ m}^2)(3 \text{ m/s}) = 0.0971 \text{ kg/s}$$

and

$$\begin{aligned}\dot{Q}_{\text{loss}} &= \dot{m}C_p(T_{\text{in}} - T_{\text{out}})\\ &= (0.0971 \text{ kg/s})(1.007 \text{ kJ/kg.°C})(65-60)\text{°C}\\ &= \mathbf{0.489 \text{ kJ/s}}\end{aligned}$$

or 1760 kJ/h. The cost of this heat loss to the home owner is

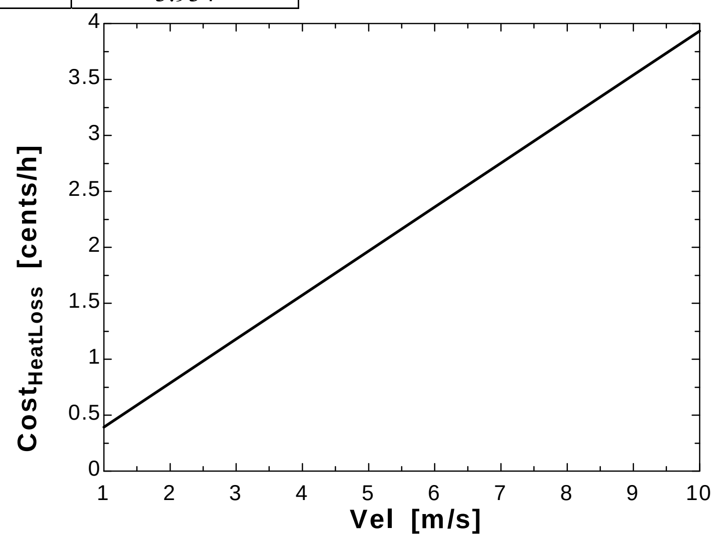
$$\begin{aligned}\text{Cost of Heat Loss} &= \frac{(\text{Rate of heat loss})(\text{Unit cost of energy input})}{\text{Furnace efficiency}}\\ &= \frac{(1760 \text{ kJ/h})(\$0.58/\text{therm})}{0.82}\left(\frac{1 \text{ therm}}{105{,}500 \text{ kJ}}\right)\\ &= \mathbf{\$0.012/h}\end{aligned}$$

Discussion The heat loss from the heating ducts in the attic is costing the homeowner 1.2 cents per hour. Assuming the heater operates 2,000 hours during a heating season, the annual cost of this heat loss adds up to \$24. Most of this money can be saved by insulating the heating ducts in the unheated areas.

1-123

```
"GIVEN"
L=4 "[m]"
D=0.2 "[m]"
P_air_in=100 "[kPa]"
T_air_in=65 "[C]"
"Vel=3 [m/s], parameter to be varied"
T_air_out=60 "[C]"
eta_furnace=0.82
Cost_gas=0.58 "[$/therm]"

"PROPERTIES"
R=0.287 "[kJ/kg-K], gas constant of air"
C_p=CP(air, T=25) "at room temperature"

"ANALYSIS"
rho=P_air_in/(R*(T_air_in+273))
A_c=pi*D^2/4
m_dot=rho*A_c*Vel
Q_dot_loss=m_dot*C_p*(T_air_in-T_air_out)*Convert(kJ/s, kJ/h)
Cost_HeatLoss=Q_dot_loss/eta_furnace*Cost_gas*Convert(kJ, therm)*Convert($, cents)
```

Vel [m/s]	$Cost_{HeatLoss}$ [Cents/h]
1	0.3934
2	0.7868
3	1.18
4	1.574
5	1.967
6	2.361
7	2.754
8	3.147
9	3.541
10	3.934



1-124 Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. The electric power input to the heater, and the money that will be saved during a 10-min shower by installing a heat exchanger with an effectiveness of 0.50 are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$,. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** Heat losses from the pipe are negligible.

Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and C = 4.18 kJ/kg·°C (Table A-9).

Analysis We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the energy balance for this steady-flow system can be expressed in the rate form as

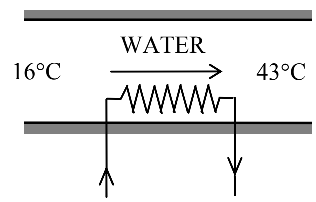
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \quad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

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

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1) = \dot{m}[C(T_2 - T_1) + v(P_2 - P_1)^{\nearrow 0}] = \dot{m}C(T_2 - T_1)$$

where

$$\dot{m} = \rho \dot{V} = (1 \text{ kg/L})(10 \text{ L/min}) = 10 \text{ kg/min}$$



Substituting,

$$\dot{W}_{\text{e,in}} = (10/60\text{kg/s})(4.18 \text{ kJ/kg}\cdot°\text{C})(43 - 16)°\text{C} = \mathbf{18.8 \text{ kW}}$$

The energy recovered by the heat exchanger is

$$\begin{aligned}\dot{Q}_{\text{saved}} &= \varepsilon \dot{Q}_{\text{max}} = \varepsilon \dot{m}C(T_{\text{max}} - T_{\text{min}}) \\ &= 0.5(10/60 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(39 - 16)°\text{C} \\ &= 8.0 \text{ kJ/s} = 8.0 \text{ kW}\end{aligned}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot{W}_{\text{in,new}} = \dot{W}_{\text{in,old}} - \dot{Q}_{\text{saved}} = 18.8 - 8.0 = \mathbf{10.8 \text{ kW}}$$

The money saved during a 10-min shower as a result of installing this heat exchanger is

$$(8.0 \text{ kW})(10/60 \text{ h})(8.5 \text{ cents/kWh}) = \mathbf{11.3 \text{ cents}}$$

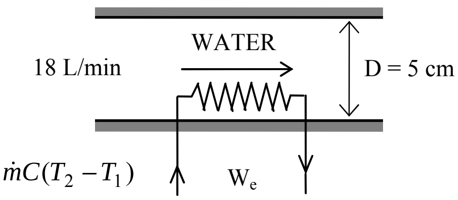
1-125 Water is to be heated steadily from 15°C to 50°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$,. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** The pipe is insulated and thus the heat losses are negligible.

Properties The density and specific heat of water at room temperature are ρ = 1000 kg/m^3 and C = 4.18 kJ/kg·°C (Table A-9).

Analysis (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. 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

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



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

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[C(T_2 - T_1) + v(P_2 - P_1)^{\nearrow 0}] = \dot{m}C(T_2 - T_1)$$

The mass flow rate of water through the pipe is

$$\dot{m} = \rho\dot{V}_1 = (1000 \text{ kg/m}^3)(0.018 \text{ m}^3\text{/min}) = 18 \text{ kg/min}$$

Therefore,

$$\dot{W}_{e,in} = \dot{m}C(T_2 - T_1) = (18/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot°\text{C})(50 - 15)°\text{C} = \mathbf{43.9 \text{ kW}}$$

(*b*) The average velocity of water through the pipe is determined from

$$\text{V}_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}}{\pi r^2} = \frac{0.018 \text{ m}^3\text{/min}}{\pi(0.025 \text{ m})^2} = \mathbf{9.17 \text{ m/min}}$$

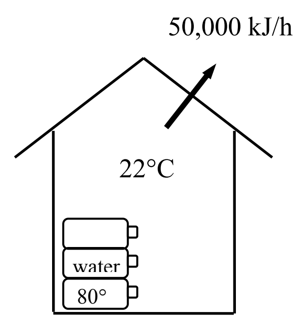
1-126 The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night with or without solar heating are to be determined.

Assumptions **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times.

Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and C = 4.18 kJ/kg·°C (Table A-9).

Analysis (*a*) The total mass of water is

$$m_w = \rho V = (1\text{ kg/L})(50\times 20\text{ L}) = 1000\text{ kg}$$



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

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

$$W_{e,in} - Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}^{\nearrow 0}$$

$$\dot{W}_{e,in}\Delta t - Q_{out} = [mC(T_2 - T_1)]_{\text{water}}$$

Substituting,

(15 kJ/s)Δt - (50,000 kJ/h)(10 h) = (1000 kg)(4.18 kJ/kg·°C)(22 - 80)°C

It gives

Δt = 17,170 s = **4.77 h**

(*b*) If the house incorporated no solar heating, the 1st law relation above would simplify further to

$$\dot{W}_{e,in}\Delta t - Q_{out} = 0$$

Substituting,

(15 kJ/s)Δt - (50,000 kJ/h)(10 h) = 0

It gives

Δt = 33,330 s = **9.26 h**

1-127 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 factor 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

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

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

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

In windy air it would be

$$Q_{\text{windy air}} = hA_s\Delta T = (50\text{ W/m}^2\cdot°\text{C})(1.60\text{ m}^2)(34 - 20)°\text{C} = \mathbf{1120\text{ W}}$$

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

$$1120\text{ W} = (hA_s\Delta T)_{\text{still air}} = (15\text{ W/m}^2\cdot°\text{C})(1.60\text{ m}^2)(34 - T_{\text{effective}})°\text{C}$$

Windy weather

which gives

$$T_{\text{effective}} = -12.7°\text{C}$$

That is, the windy air at 20°C feels as cold as still air at -12.7°C as a result of the wind-chill effect. Therefore, the wind-chill factor in this case is

$$F_{\text{wind-chill}} = 20 - (-12.7) = \mathbf{32.7°C}$$

1-128 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.

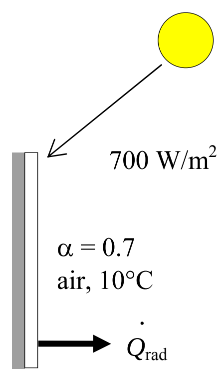
Properties The solar absorptivity of the plate is given to be $\alpha = 0.7$.

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

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

$$\alpha\dot{Q}_{\text{solar}} = hA_s(T_s - T_o)$$

$$0.7 \times A \times 700\text{W/m}^2 = (30\text{W/m}^2\cdot°\text{C})A_s(T_s - 10)$$



Canceling the surface area A_s and solving for T_s gives

$$T_s = \mathbf{26.3°C}$$

1-129 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.

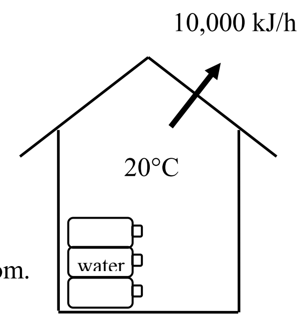
Properties The specific heat of water at room temperature is $C = 4.18$ kJ/kg·°C (Table A-9).

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

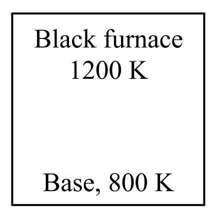
$$T_1 = \mathbf{77.4°C}$$

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

1-130 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 $\varepsilon = 0.7$.



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

$$\begin{aligned}\dot{Q}_{\text{rad,base}} &= \varepsilon A\sigma(T_{\text{base}}^4 - T_{\text{surr}}^4)\\ &= (0.7)(3\times 3\text{ m}^2)(5.67\times 10^{-8}\text{ W/m}^2.\text{K}^4)[(1200\text{ K})^4 - (800\text{ K})^4]\\ &= \mathbf{594{,}400\text{ W}}\end{aligned}$$

1-131 A refrigerator consumes 600 W of power when operating, and its motor remains on for 5 min and then off for 15 min periodically. The average thermal conductivity of the refrigerator walls and the annual cost of operating this refrigerator are to be determined.

Assumptions **1** Quasi-steady operating conditions exist. **2** The inner and outer surface temperatures of the refrigerator remain constant.

Analysis The total surface area of the refrigerator where heat transfer takes place is

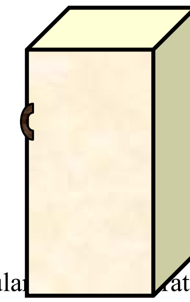
$$A_{\text{total}} = 2\left[(1.8\times 1.2)+(1.8\times 0.8)+(1.2\times 0.8)\right] = 9.12\ \text{m}^2$$

Since the refrigerator has a COP of 2.5, the rate of heat removal from the refrigerated space, which is equal to the rate of heat gain in steady operation, is

$$\dot{Q} = \dot{W}_e \times \text{COP} = (600\ \text{W})\times 2.5 = 1500\ \text{W}$$

But the refrigerator operates a quarter of the time (5 min on, 15 min off). Therefore, the average rate of heat gain is

$$\dot{Q}_{\text{ave}} = \dot{Q}/4 = (1500\ \text{W})/4 = 375\ \text{W}$$



Then the thermal conductivity of refrigerator walls is determined to be

$$\dot{Q}_{\text{ave}} = kA\frac{\Delta T_{\text{ave}}}{L} \longrightarrow k = \frac{\dot{Q}_{\text{ave}}L}{A\Delta T_{\text{ave}}} = \frac{(375\ \text{W})(0.03\ \text{m})}{(9.12\ \text{m}^2)(17-6)^\circ\text{C}} = \mathbf{0.112\ W/m.^\circ C}$$

The total number of hours this refrigerator remains on per year is

$$\Delta t = 365\times 24/4 = 2190\ \text{h}$$

Then the total amount of electricity consumed during a one-year period and the annual [illegible]rating this refrigerator are

$$\text{Annual Electricity Usage} = \dot{W}_e\Delta t = (0.6\ \text{kW})(2190\ \text{h/yr}) = 1314\ \text{kWh/yr}$$

$$\text{Annual cost} = (1314\ \text{kWh/yr})(\$0.08/\text{kWh}) = \mathbf{\$105.1/yr}$$

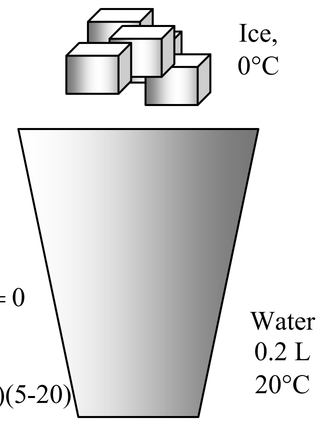
1-132 A 0.2-L glass of water at 20°C is to be cooled with ice to 5°C. The amounts of ice or cold water that needs to be added to the water are to be determined.

Assumptions **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water is negligible.

Properties The density of water is 1 kg/L, and the specific heat of water at room temperature is C = 4.18 kJ/kg·°C (Table A-9). The heat of fusion of ice at atmospheric pressure is 333.7 kJ/kg,.

Analysis The mass of the water is

$$m_w = \rho V = (1\text{kg/L})(0.2\text{ L}) = 0.2\text{kg}$$



We take the ice and the water as our system, and disregard any heat and mass transfer. This is a reasonable assumption since the time period of the process is very short. Then the energy balance can be written as

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

$$\left[mC(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mC(T_2 - 0^\circ\text{C})_{\text{liquid}}\right]_{\text{ice}} + [mC(T_2 - T_1)]_{\text{water}} = 0$$

Noting that $T_{1,\text{ ice}}$ = 0°C and T_2 = 5°C and substituting

$$m[0 + 333.7 \text{ kJ/kg} + (4.18 \text{ kJ/kg·°C})(5\text{-}0)\text{°C}] + (0.2 \text{ kg})(4.18 \text{ kJ/kg·°C})(5\text{-}20)$$

It gives m = 0.0354 kg = **35.4 g**

Cooling with cold water can be handled the same way. All we need to do is replace the terms for ice by the ones for cold water at 0°C:

$$(\Delta U)_{\text{coldwater}} + (\Delta U)_{\text{water}} = 0$$

$$[mC(T_2 - T_1)]_{\text{coldwater}} + [mC(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$[m_{\text{cold water}} (4.18 \text{ kJ/kg·°C})(5 - 0)\text{°C}] + (0.2 \text{ kg})(4.18 \text{ kJ/kg·°C})(5\text{-}20)\text{°C} = 0$$

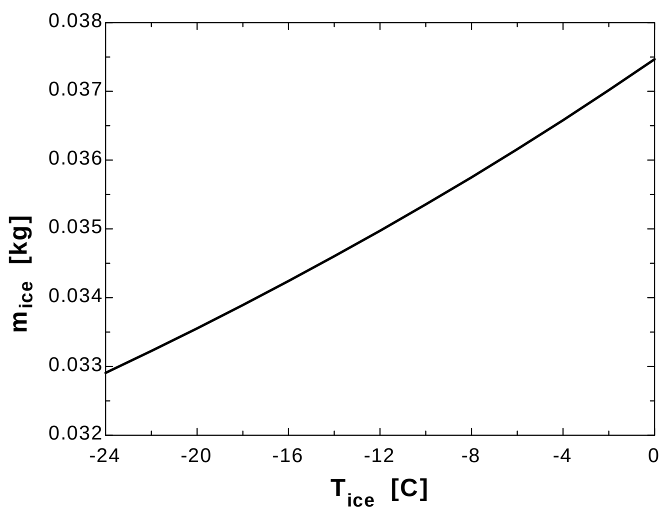
It gives m = 0.6 kg = **600 g**

Discussion Note that this is 17 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks.

1-133

```
"GIVEN"
V=0.0002 "[m^3]"
T_w1=20 "[C]"
T_w2=5 "[C]"
"T_ice=0 [C], parameter to be varied"
T_melting=0 "[C]"

"PROPERTIES"
rho=density(water, T=25, P=101.3) "at room temperature"
C_w=CP(water, T=25, P=101.3) "at room temperature"
C_ice=c_('Ice', T_ice)
h_if=333.7 "[kJ/kg]"

"ANALYSIS"
m_w=rho*V
DELTAU_ice+DELTAU_w=0 "energy balance"
DELTAU_ice=m_ice*C_ice*(T_melting-T_ice)+m_ice*h_if
DELTAU_w=m_w*C_w*(T_w2-T_w1)
```

T_{ice} [C]	m_{ice} [kg]
-24	0.03291
-22	0.03323
-20	0.03355
-18	0.03389
-16	0.03424
-14	0.0346
-12	0.03497
-10	0.03536
-8	0.03575
-6	0.03616
-4	0.03658
-2	0.03702
0	0.03747



1-134E A 1-short ton (2000 lbm) of water at 70°F is to be cooled in a tank by pouring 160 lbm of ice at 25°F into it. The final equilibrium temperature in the tank is to be determined. The melting temperature and the heat of fusion of ice at atmospheric pressure are 32°F and 143.5 Btu/lbm, respectively

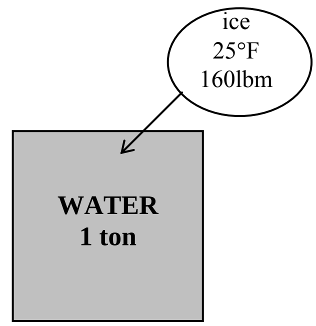
Assumptions **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water is negligible.

Properties The density of water is 62.4 lbm/ft^3, and the specific heat of water at room temperature is C = 1.0 Btu/lbm·°F (Table A-9). The heat of fusion of ice at atmospheric pressure is 143.5 Btu/lbm and the specific heat of ice is 0.5 Btu/lbm.°F.

Analysis We take the ice and the water as our system, and disregard any heat transfer between the system and the surroundings. Then the energy balance for this process can be written as

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

$$\left[mC(32°\text{F}-T_1)_{\text{solid}}+mh_{if}+mC(T_2-32°\text{F})_{\text{liquid}}\right]_{\text{ice}}+\left[mC(T_2-T_1)\right]_{\text{water}}=0$$



Substituting,

$$(160\text{lbm})\left[(0.50\text{Btu/lbm}\cdot°\text{F})(32-25)°\text{F}+143.5\text{Btu/lbm}+(1.0\text{Btu/lbm}\cdot°\text{F})(T_2-32)°\text{F}\right]$$
$$+(2000\text{lbm})(1.0\text{Btu/lbm}\cdot°\text{F})(T_2-70)°\text{F}=0$$

It gives $\qquad T_2 = \mathbf{56.3°F}$

which is the final equilibrium temperature in the tank.

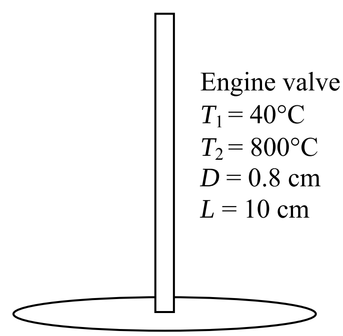
1-135 Engine valves are to be heated in a heat treatment section. The amount of heat transfer, the average rate of heat transfer, the average heat flux, and the number of valves that can be heat treated daily are to be determined.

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

$$Q=\Delta U=mC_p(T_2-T_1)$$
$$=(0.0788\text{ kg})(0.440\text{ kJ/kg.°C})(800-40)°\text{C}=\mathbf{26.35\ kJ}$$



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

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

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

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

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

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

1-136 Somebody takes a shower using a mixture of hot and cold water. The mass flow rate of hot water and the average temperature of mixed water are to be determined.

Assumptions The hot water temperature changes from 80°C at the beginning of shower to 60°C at the end of shower. We use an average value of 70°C for the temperature of hot water exiting the tank.

Properties The properties of liquid water are C_p = 4.18 kJ/kg.°C and ρ = 977.6 kg/m^3 (Table A-2).

Analysis We take the water tank as the system. The energy balance for this system can be expressed as

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{sys}}$$

$$\left[\dot{W}_{\text{e,in}} + \dot{m}_{\text{hot}} C(T_{\text{in}} - T_{\text{out}})\right]\Delta t = m_{\text{tank}} C(T_2 - T_1)$$

where T_{out} is the average temperature of hot water leaving the tank: (80+70)/2=70°C and

$$m_{\text{tank}} = \rho V = (977.6\ \text{kg/m}^3)(0.06\ \text{m}^3) = 58.656\ \text{kg}$$

Substituting,

$$\left[1.6\ \text{kJ/s} + \dot{m}_{\text{hot}}(4.18\ \text{kJ/kg.°C})(20-70)\text{°C}\right](8\times 60\ \text{s}) = (58.656\ \text{kg})(4.18\ \text{kJ/kg.°C})(60-80)\text{°C}$$

$$\dot{m}_{\text{hot}} = \mathbf{0.0565\ kg/s}$$

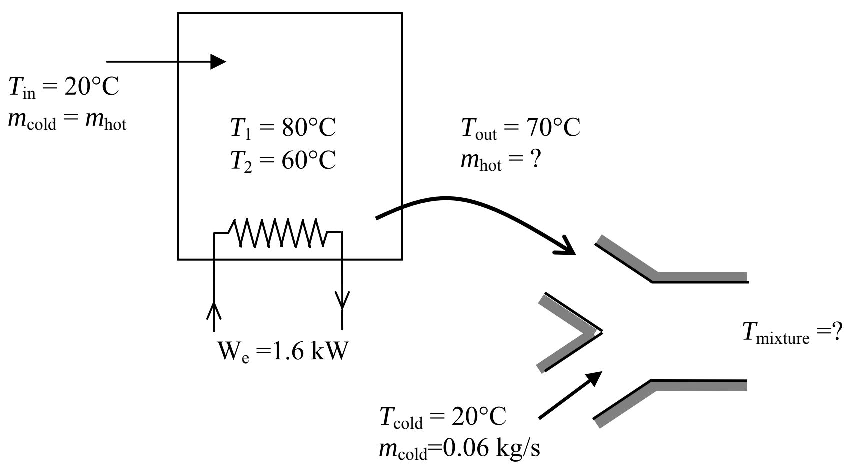
To determine the average temperature of the mixture, an energy balance on the mixing section can be expressed as

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

$$\dot{m}_{\text{hot}} C T_{\text{hot}} + \dot{m}_{\text{cold}} C T_{\text{cold}} = (\dot{m}_{\text{hot}} + \dot{m}_{\text{cold}}) C T_{\text{mixture}}$$

$$(0.0565\ \text{kg/s})(4.18\ \text{kJ/kg.°C})(70\text{°C}) + (0.06\ \text{kg/s})(4.18\ \text{kJ/kg.°C})(20\text{°C}) = (0.0565 + 0.06\ \text{kg/s})(4.18\ \text{kJ/kg.°C})T_{\text{mixture}}$$

$$T_{\text{mixture}} = \mathbf{44.2°C}$$



1-137 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.

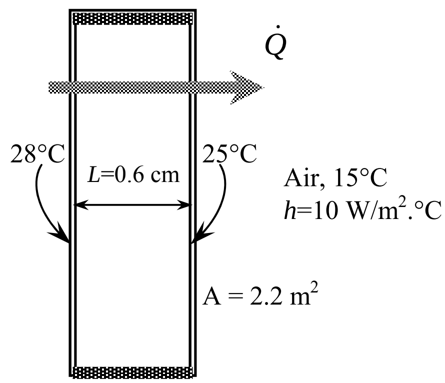
Properties The thermal conductivity of the glass is given to be k = 0.7 W/m·°C.

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

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.7 \text{ W/m}\cdot{}^\circ\text{C})(2.2 \text{ m}^2)\frac{(28-25)^\circ\text{C}}{0.006 \text{ m}} = 770 \text{ W}$$

The rate of heat transfer from the glass by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T = (10 \text{ W/m}^2\cdot{}^\circ\text{C})(2.2 \text{ m}^2)(25-15)^\circ\text{C} = 220 \text{ W}$$



Under steady conditions, the heat transferred through the cover by conduction should be transferred from the outer surface by convection and radiation. That is,

$$\dot{Q}_{\text{rad}} = \dot{Q}_{\text{cond}} - \dot{Q}_{\text{conv}} = 770 - 220 = 550 \text{ W}$$

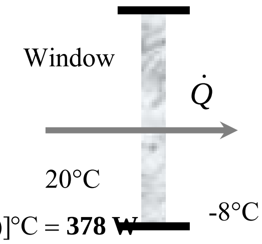
Then the fraction of heat transferred by radiation becomes

$$f = \frac{\dot{Q}_{\text{rad}}}{\dot{Q}_{\text{cond}}} = \frac{550}{770} = \mathbf{0.714} \quad \text{(or 71.4\%)}$$

1-138 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



$$\dot{Q}_{\text{window}} = U_{\text{overall}} A_{\text{window}} (T_{in} - T_{out})$$

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: $\dot{Q}_{\text{window, max}} = (6.25\ \text{W/m}^2\cdot°\text{C})(1.2\times1.8\ \text{m}^2)[20-(-8)]°\text{C} = \mathbf{378\ W}$

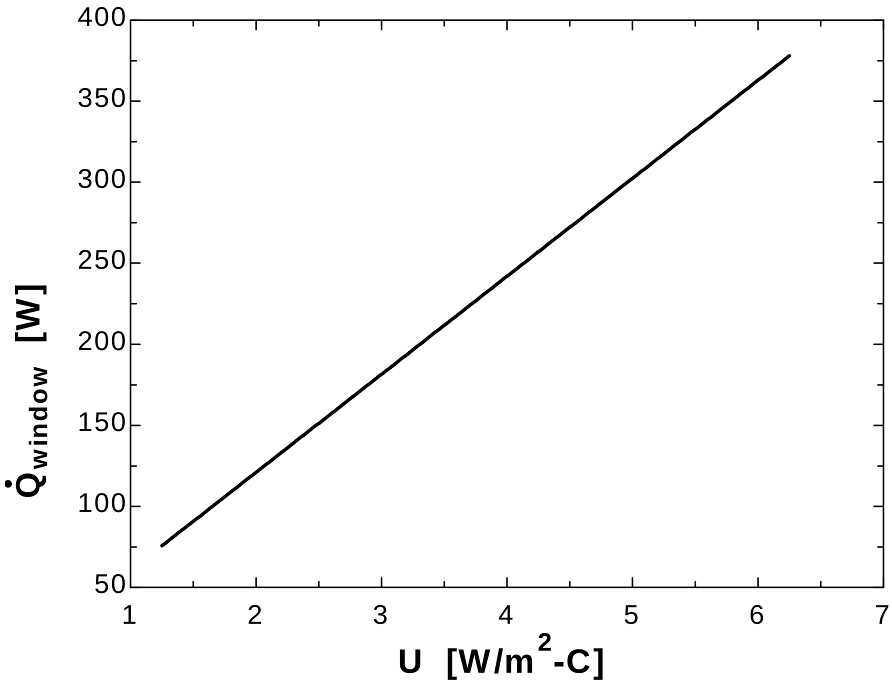
Minimum heat loss: $\dot{Q}_{\text{window, min}} = (1.25\ \text{W/m}^2\cdot°\text{C})(1.2\times1.8\ \text{m}^2)[20-(-8)]°\text{C} = \mathbf{76\ W}$

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.

1-139

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

"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



1-140 . . . 1-144 Design and Essay Problems

Chapter 2
HEAT CONDUCTION EQUATION

Introduction

2-1C Heat transfer is a *vector* quantity since it has direction as well as magnitude. Therefore, we must specify both direction and magnitude in order to describe heat transfer completely at a point. Temperature, on the other hand, is a scalar quantity.

2-2C The term *steady* implies *no change with time* at any point within the medium while *transient* implies *variation with time* or *time dependence*. Therefore, the temperature or heat flux remains unchanged with time during steady heat transfer through a medium at any location although both quantities may vary from one location to another. During transient heat transfer, the temperature and heat flux may vary with time as well as location. Heat transfer is one-dimensional if it occurs primarily in one direction. It is two-dimensional if heat tranfer in the third dimension is negligible.

2-3C Heat transfer to a canned drink can be modeled as two-dimensional since temperature differences (and thus heat transfer) will exist in the radial and axial directions (but there will be symmetry about the center line and no heat transfer in the azimuthal direction. This would be a transient heat transfer process since the temperature at any point within the drink will change with time during heating. Also, we would use the cylindrical coordinate system to solve this problem since a cylinder is best described in cylindrical coordinates. Also, we would place the origin somewhere on the center line, possibly at the center of the bottom surface.

2-4C Heat transfer to a potato in an oven can be modeled as one-dimensional since temperature differences (and thus heat transfer) will exist in the radial direction only because of symmetry about the center point. This would be a transient heat transfer process since the temperature at any point within the potato will change with time during cooking. Also, we would use the spherical coordinate system to solve this problem since the entire outer surface of a spherical body can be described by a constant value of the radius in spherical coordinates. We would place the origin at the center of the potato.

2-5C Assuming the egg to be round, heat transfer to an egg in boiling water can be modeled as one-dimensional since temperature differences (and thus heat transfer) will primarily exist in the radial direction only because of symmetry about the center point. This would be a transient heat transfer process since the temperature at any point within the egg will change with time during cooking. Also, we would use the spherical coordinate system to solve this problem since the entire outer surface of a spherical body can be described by a constant value of the radius in spherical coordinates. We would place the origin at the center of the egg.