Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

5th Edition

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McGraw-Hill, 2015

Chapter 1

INTRODUCTION AND BASIC CONCEPTS

Thermodynamics and Heat Transfer

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

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

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

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

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

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

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

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

1-8C The right choice between a crude and complex model is usually the *simplest* model which yields *adequate* results. Preparing very accurate but complex models is not necessarily a better choice since such models are not much use to an analyst if they are very difficult and time consuming to solve. At the minimum, the model should reflect the essential features of the physical problem it represents.

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

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

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

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

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

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

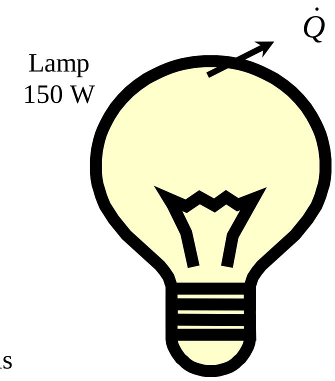
1-15 The filament of a 150 W incandescent lamp is 5 cm long and has a diameter of 0.5 mm. The heat flux on the surface of the filament, the heat flux on the surface of the glass bulb, and the annual electricity cost of the bulb are to be determined.

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

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

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

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



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

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

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

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

$$\text{Electricity Consumption} = \dot{Q}\Delta t = (0.15\text{ kW})(365\times8\text{ h/yr}) = 438\text{ kWh/yr}$$

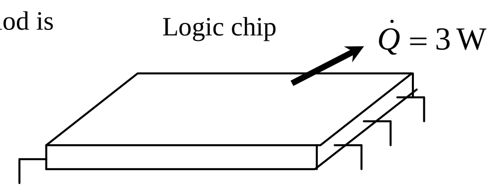
$$\text{Annual Cost} = (438\text{ kWh/yr})(\$0.08/\text{kWh}) = \mathbf{\$35.04/yr}$$

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

Assumptions Heat transfer from the surface is uniform.

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

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



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

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

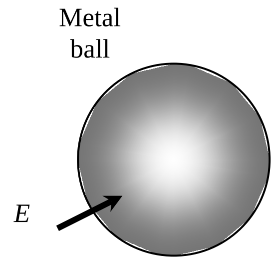
1-17 An aluminum ball is to be heated from 80°C to 200°C. The amount of heat that needs to be transferred to the aluminum ball is to be determined.

Assumptions The properties of the aluminum ball are constant.

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

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

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



where

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

Substituting,

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

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

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

Assumptions The specific heat of the liquid ammonia is constant.

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

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

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

where

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

Substituting,

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

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

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

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

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

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

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

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

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

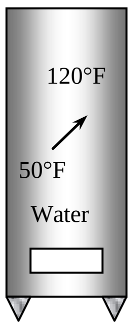
Thus, the velocity of the steel strip being conveyed is

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

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

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

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



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

Analysis The mass of water in the tank is

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

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

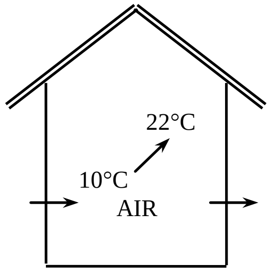
$$Q = mc_p(T_2 - T_1) = (498.7 \text{ lbm})(0.999 \text{ Btu/lbm}\cdot°\text{F})(120 - 50)°\text{F} = \mathbf{34{,}874 \text{ Btu}}$$

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

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

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

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



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

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

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

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

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

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

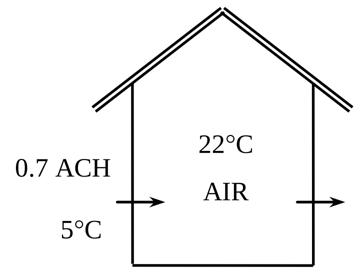
$$\text{Enegy Cost} = \text{(Energy used)(Unit cost of energy)} = (9038/3600\ \text{kWh})(\$0.075/\text{kWh}) = \mathbf{\$0.19}$$

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

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

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

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



Analysis The volume of the air in the house is

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

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

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

$$= \frac{(89.6\text{ kPa})(16.8 \times 450\text{ m}^3/\text{day})}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(5+273.15\text{ K})} = 8485\text{ kg/day}$$

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

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

$$= (8485\text{ kg/day})(1.007\text{ kJ/kg.°C})(22-5)\text{°C} = 145{,}260\text{ kJ/day} = \mathbf{40.4\ kWh/day}$$

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

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

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

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

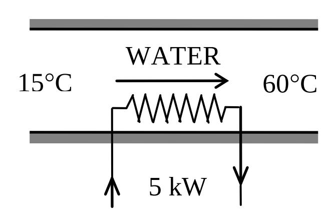
Properties The specific heat of water at room temperature is c_p = 4.18 kJ/kg·°C.

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

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

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

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



Thus,

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

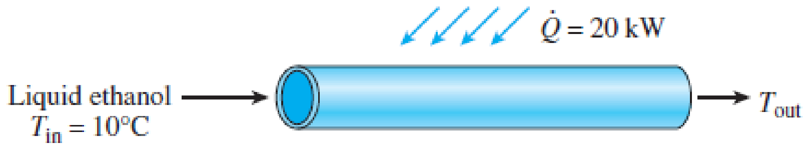
1-24 Liquid ethanol is being transported in a pipe where heat is added to the liquid. The volume flow rate that is necessary to keep the ethanol temperature below its flashpoint is to be determined.

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

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

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

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



or

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

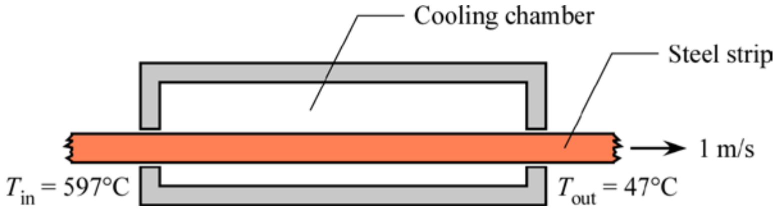
For the ethanol in the pipe to be below its flashpoint, it is necessary to keep T_{out} below 16.6°C. Thus, the volume flow rate should be

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

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

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

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



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

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

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

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

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

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

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

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

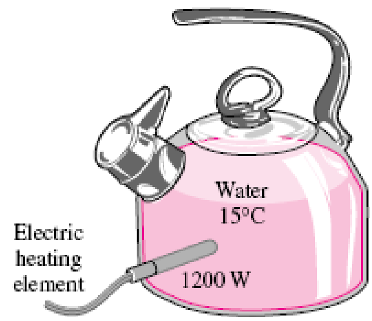
Assumptions **1** Heat loss from the teapot is negligible. **2** Constant properties can be used for both the teapot and the water.

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

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

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

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



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

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

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

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

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

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

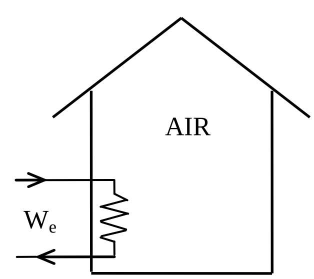
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The temperature of the room remains constant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

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

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

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

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



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

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

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

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

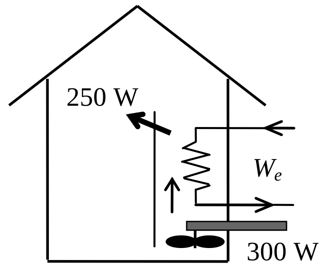
Properties The specific heat of air at room temperature is c_p = 1.007 kJ/kg·°C (Table A-15).

Analysis We take the heating duct as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that this is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\rm CV} = 0$ 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

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

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

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

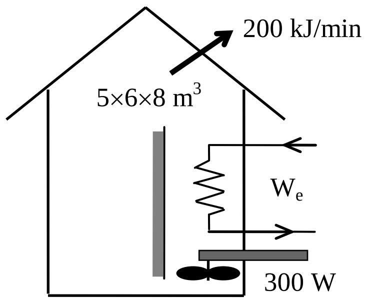
Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). Also, $c_p = 1.007$ kJ/kg·K for air at room temperature (Table A-15) and $c_v = c_p - R = 0.720$ kJ/kg·K.

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

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

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

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



The total mass of air in the room is

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

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

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

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

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

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

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

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

Thus,

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

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

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

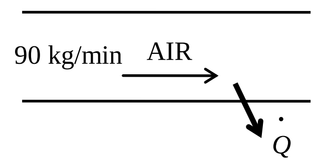
Properties The specific heat of air at room temperature is c_p = 1.007 kJ/kg·°C (Table A-15).

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

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

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

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



Substituting,

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

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

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

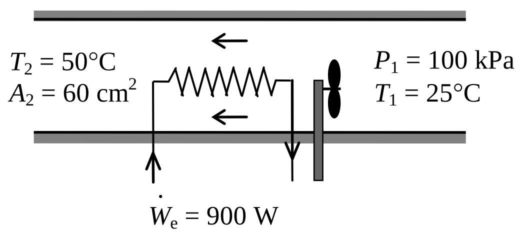
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.007 kJ/kg·K for air at room temperature (Table A-15).

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

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

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

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



Thus,

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

Then,

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})}{100 \text{ kPa}} = 0.8553 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}v_1 = (0.03575 \text{ kg/s})(0.8553 \text{ m}^3/\text{kg}) = \mathbf{0.0306 \ m^3/s}$$

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

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323 \text{ K})}{100 \text{ kPa}} = 0.9270 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_2}A_2V_2 \longrightarrow V_2 = \frac{\dot{m}v_2}{A_2} = \frac{(0.03575 \text{ kg/s})(0.9270 \text{ m}^3/\text{kg})}{60\times 10^{-4} \text{ m}^2} = \mathbf{5.52 \ m/s}$$

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

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

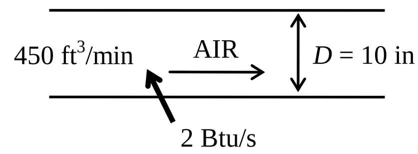
Properties The gas constant of air is R = 0.3704 psia·ft³/lbm·R (Table A-1E). Also, c_p = 0.240 Btu/lbm·R for air at room temperature (Table A-15E).

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

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

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

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



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

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

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

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

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

Then the exit temperature of air is determined to be

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

Heat Transfer Mechanisms

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

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

1-35C Diamond is a better heat conductor.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

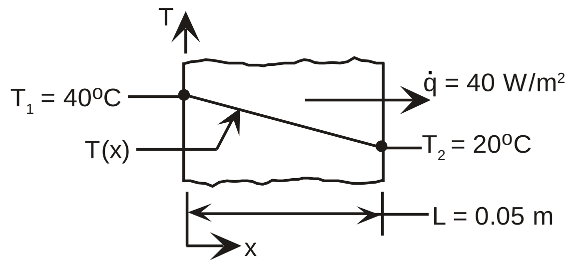
1-52 The thermal conductivity of a wood slab subjected to a given heat flux of 40 W/m^2 with constant left and right surface temperatures of 40ºC and 20ºC is to be determined.

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

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

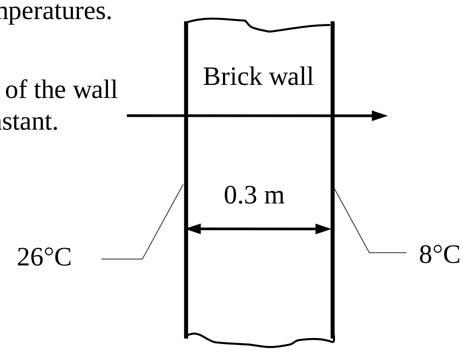
$$k = \dot{q}\frac{L}{T_1 - T_2} = \left(40\ \frac{\text{W}}{\text{m}^2}\right)\frac{0.05\ \text{m}}{(40-20)°\text{C}} =$$

0.10 W/m·K



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

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



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

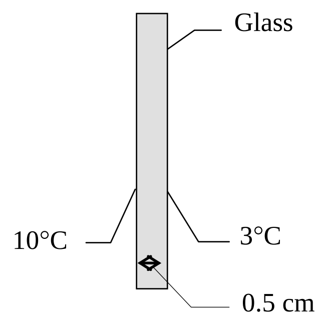
Properties The thermal conductivity of the wall is given to be k = 0.69 W/m·°C.

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

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

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

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



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

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

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

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

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

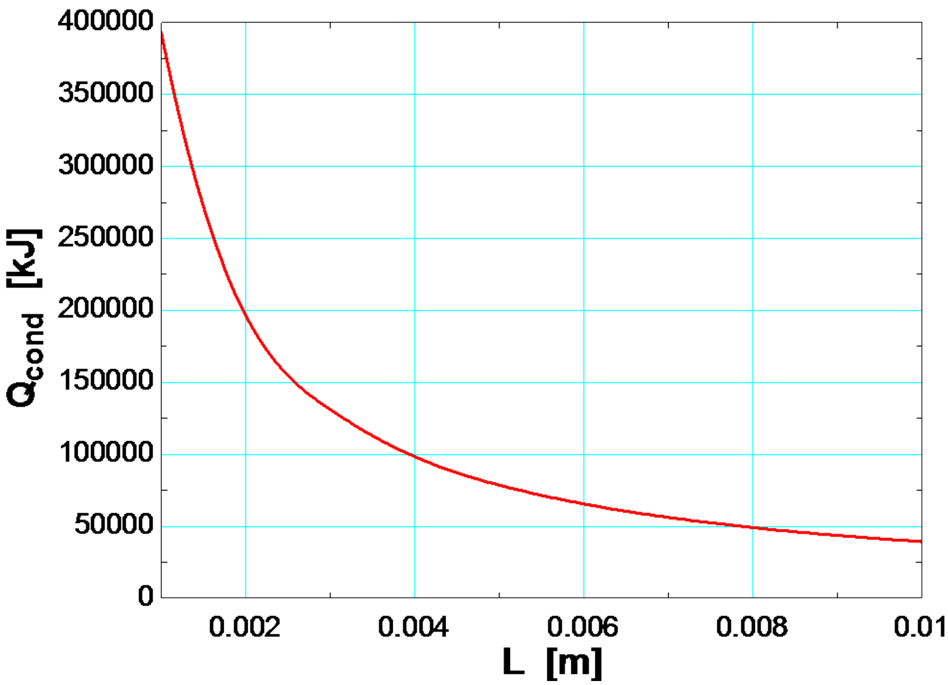
If the thickness of the glass doubled to 1 cm, then the amount of heat transfer will go down by half to **39,310 kJ**.

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

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

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

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

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



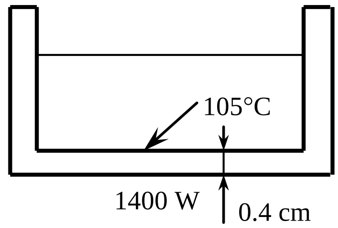
1-56 Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface temperature of the bottom of the pan is given. The temperature of the outer surface is to be determined.

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

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

Analysis The heat transfer area is

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



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

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

Substituting,

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

which gives

$$T_2 = \mathbf{106.33°C}$$

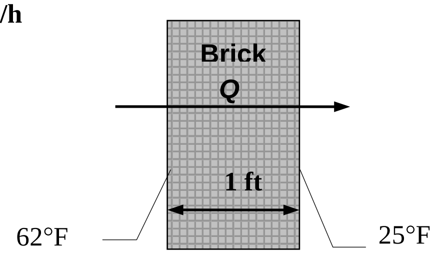
1-57E The inner and outer surface temperatures of the wall of an electrically heated home during a winter night are measured. The rate of heat loss through the wall that night and its cost are to be determined.

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

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

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

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



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

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

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

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

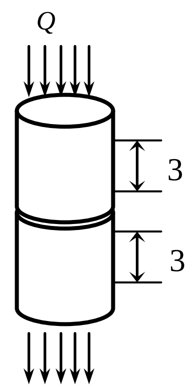
Therefore, the cost of the heat loss through the wall to the home owner that night is \$0.51.

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

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

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

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



The rate of heat flow through each sample is

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

Then the thermal conductivity of the sample becomes

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

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

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

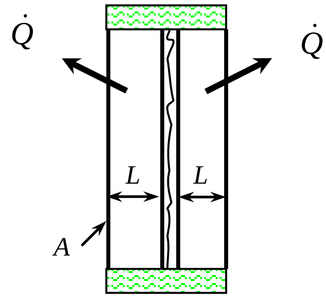
Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. **2** Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples.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

Analysis For each sample we have

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

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

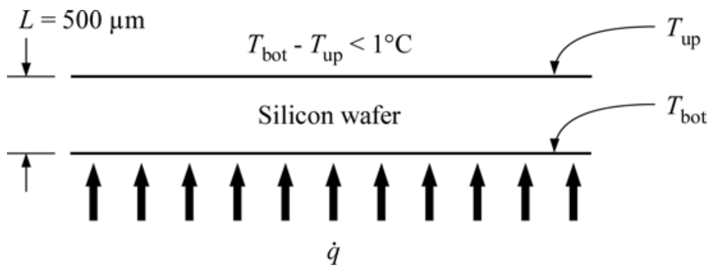
$$\Delta T = 82 - 74 = 8°\text{C}$$



Then the thermal conductivity of the material becomes

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

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



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

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

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

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

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

$$\dot{q} \le k\frac{T_{\text{bot}} - T_{\text{up}}}{L} = (148\ \text{W/m}\cdot\text{K})\frac{1\,\text{K}}{500\times 10^{-6}\ \text{m}}$$

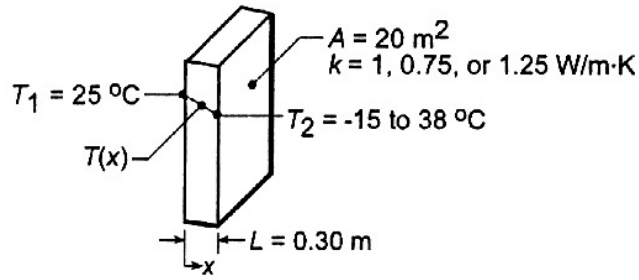
$$\dot{q} \le \mathbf{2.96\times 10^5\ W/m^2}$$

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

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

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

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



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

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

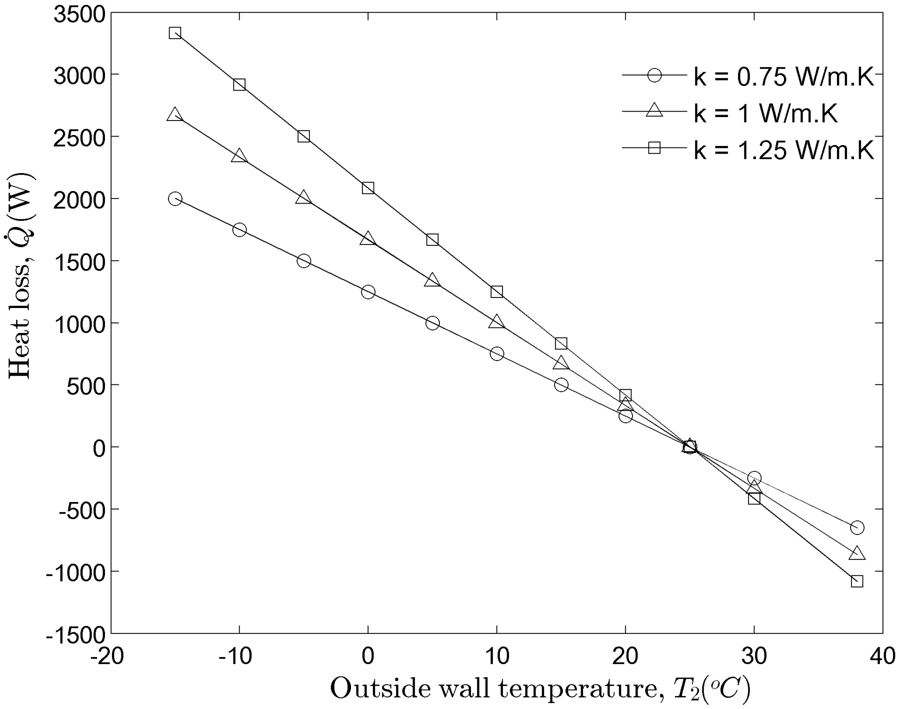
$$\dot{Q} = \dot{q}\cdot A = (133.3\,\text{W/m}^2)\cdot(20\,\text{m}^2) = \mathbf{2667\ W} \qquad (2)$$

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

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

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

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


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

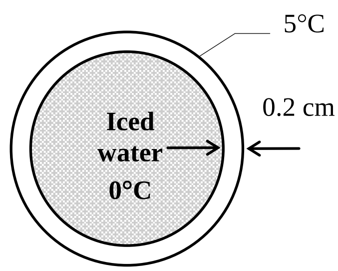
1-62 A hollow spherical iron container is filled with iced water at 0°C. The rate of heat loss from the sphere and the rate at which ice melts in the container are to be determined.

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

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

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

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



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

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

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

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

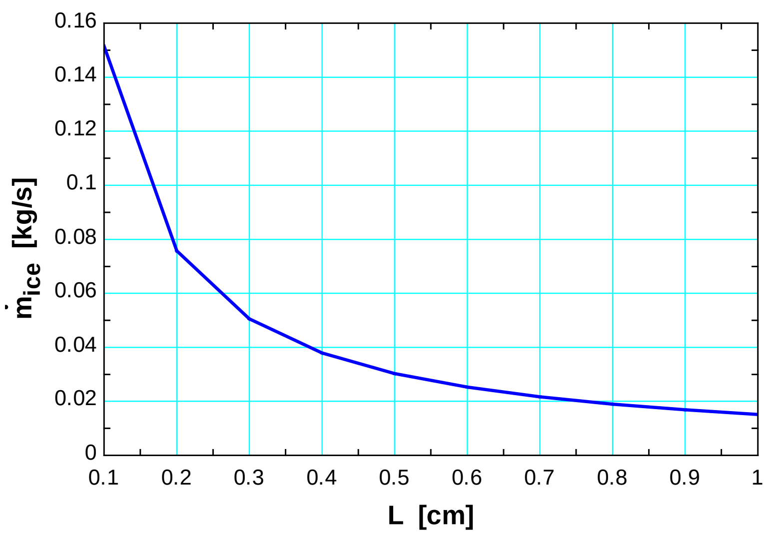
Discussion We should point out that this result is slightly in error for approximating a curved wall as a plain wall. The error in this case is very small because of the large diameter to thickness ratio. For better accuracy, we could use the inner surface area (D = 19.6 cm) or the mean surface area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

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

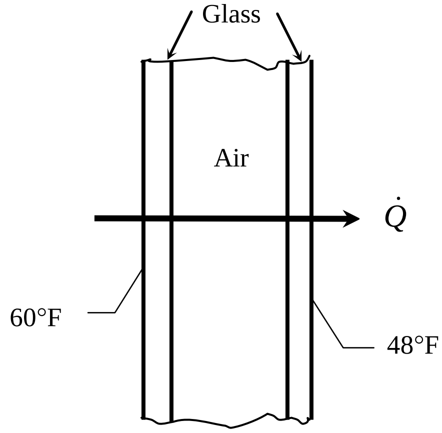
Analysis The problem is solved using EES, and the solution is given below.

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

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

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

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



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



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

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

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

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

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

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

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

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

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

$$1\text{°C} = 1.8\text{°F}$$

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

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

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

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

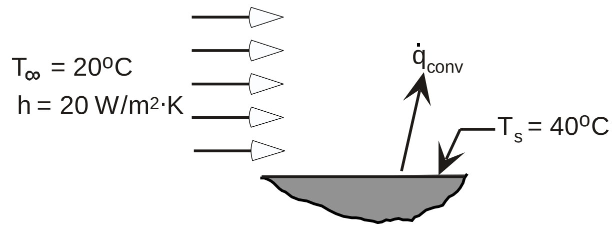
1-66 The heat flux between air with a constant temperature and convection heat transfer coefficient blowing over a pond at a constant temperature is to be determined.

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

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

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

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



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

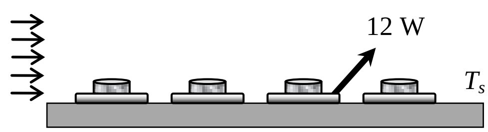
1-67 Four power transistors are mounted on a thin vertical aluminum plate that is cooled by a fan. The temperature of the aluminum plate is to be determined.

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

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

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

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



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

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

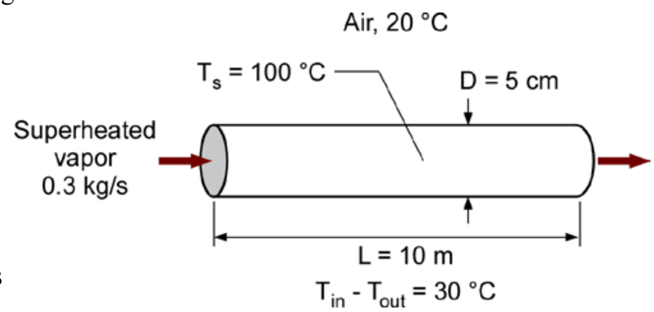
1-68 The convection heat transfer coefficient heat transfer between the surface of a pipe carrying superheated vapor and the surrounding is to be determined.

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

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

Analysis The surface area of the pipe is

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



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

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

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

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

Rearranging, the heat transfer coefficient is determined to be

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

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

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

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

Analysis The total heat transfer area of the resistor is

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

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

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

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

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

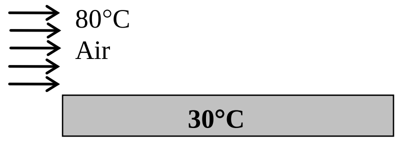
Rearranging, the heat transfer coefficient is determined to be

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

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

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

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



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

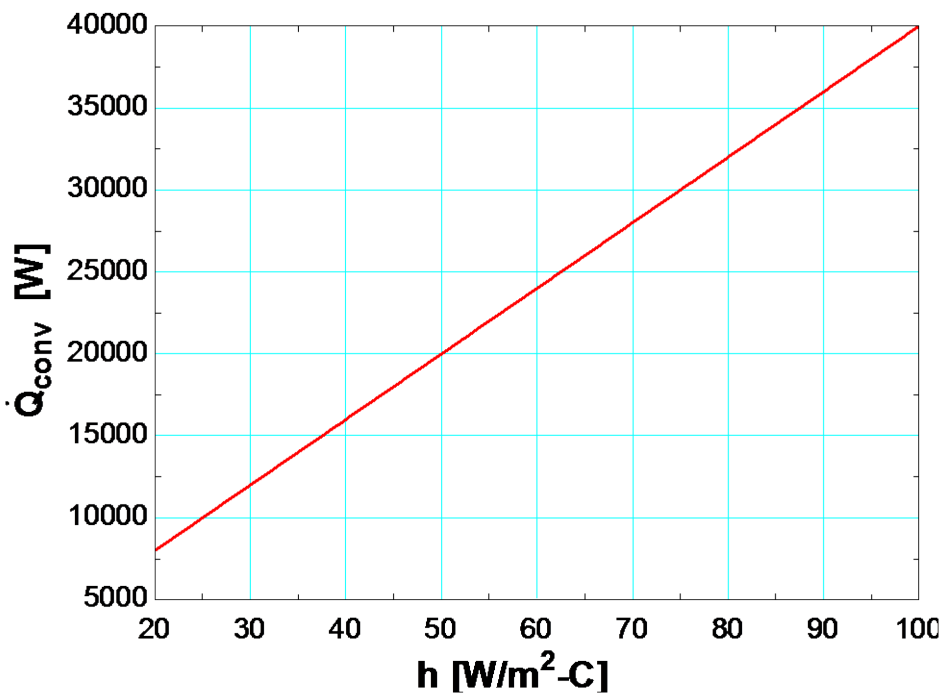
$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (55\ \text{W/m}^2\cdot°\text{C})(2\times 4\ \text{m}^2)(80-30)°\text{C} = \mathbf{22{,}000\ W}$$

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

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

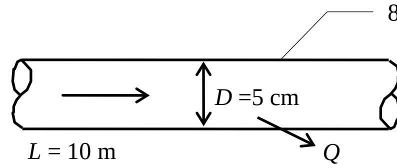
```
"GIVEN"
T_infinity=80 [C]
A=2*4 [m^2]
T_s=30 [C]
h=55 [W/m^2-C]

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

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



1-72 A hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 W/m²·°C. The rate of heat loss from the pipe by convection is to be determined.

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



Analysis The heat transfer surface area is

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

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

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

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

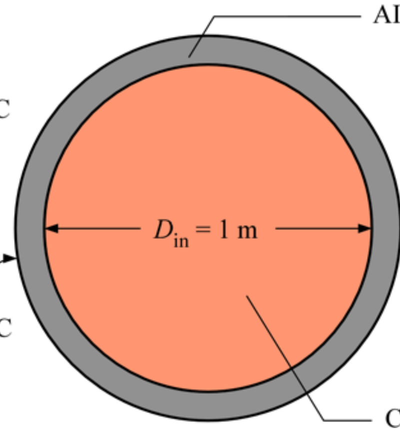
Assumptions **1** Steady operating conditions exist. **2** Negligible thermal storage for the container. **3** Temperature at the surface remained uniform.

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

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

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

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



The convection heat transfer coefficient can be determined as

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

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

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

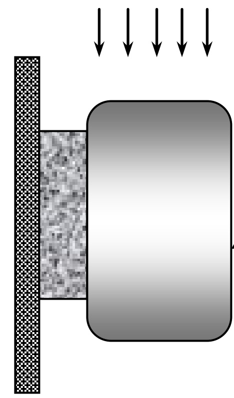
To keep the container's outer surface temperature below 50°C, the convection heat transfer coefficient should be

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

Discussion From Table 1-5, the typical values for free convection heat transfer coefficient of gases are between 2–25 W/m²·K. Thus, the required h > 1840 W/m²·K is not feasible with free convection of air. To prevent thermal burn, the container's outer surface temperature should be covered with insulation.

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

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



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

$$A_s = \pi DL + \pi D^2 / 4$$
$$= \pi(0.6\text{ cm})(0.4\text{ cm}) + \pi(0.6\text{ cm})^2 / 4 = 1.037\text{ cm}^2$$
$$= 1.037 \times 10^{-4}\text{ m}^2$$

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

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

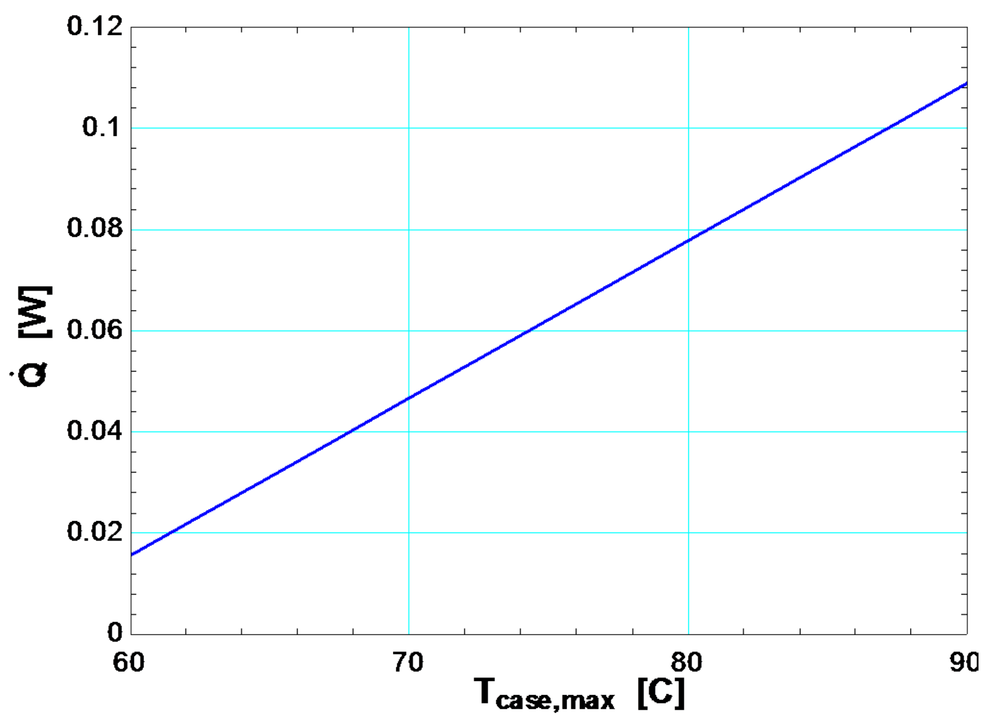
Therefore, the amount of power this transistor can dissipate safely is 0.047 W.

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

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

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

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

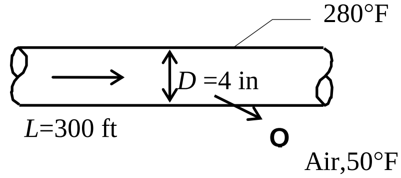
$T_{case, max}$ [C]	Q [W]
60	0.01555
62.5	0.02333
65	0.0311
67.5	0.03888
70	0.04665
72.5	0.05443
75	0.0622
77.5	0.06998
80	0.07775
82.5	0.08553
85	0.09331
87.5	0.1011
90	0.1089



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

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

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



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

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

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

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

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

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

Then the annual cost of the energy lost becomes

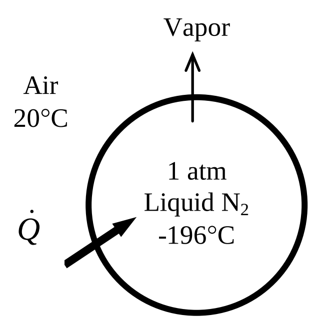
$$\text{Energy cost} = (\text{Annual energy loss})(\text{Unit cost of energy})$$
$$= (44{,}161\text{ therms/yr})(\$1.10/\text{therm}) = \$48{,}576/\text{yr} \cong \mathbf{\$48{,}600}$$

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

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

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

Analysis The rate of heat transfer to the nitrogen tank is



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

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

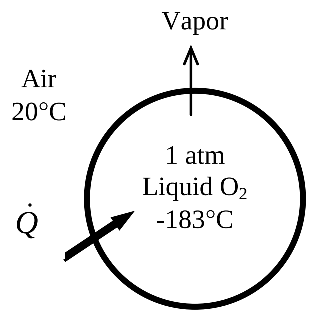
Then the rate of evaporation of liquid nitrogen in the tank is determined to be

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

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

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

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



Analysis The rate of heat transfer to the oxygen tank is

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

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

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

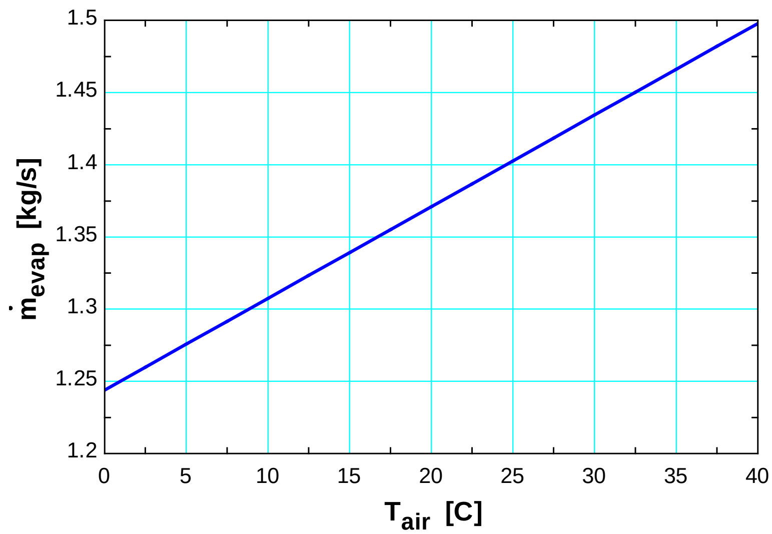
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{255.120\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{1.20\ kg/s}$$

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

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

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

"PROPERTIES"
h_fg=198 [kJ/kg]

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

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

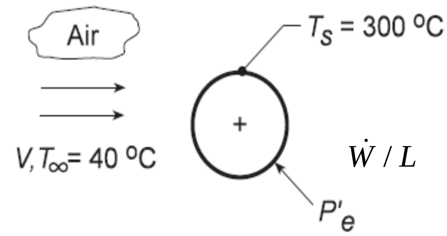


1-80 Power required to maintain the surface temperature of a long, 25 mm diameter cylinder with an imbedded electrical heater for different air velocities.

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

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

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



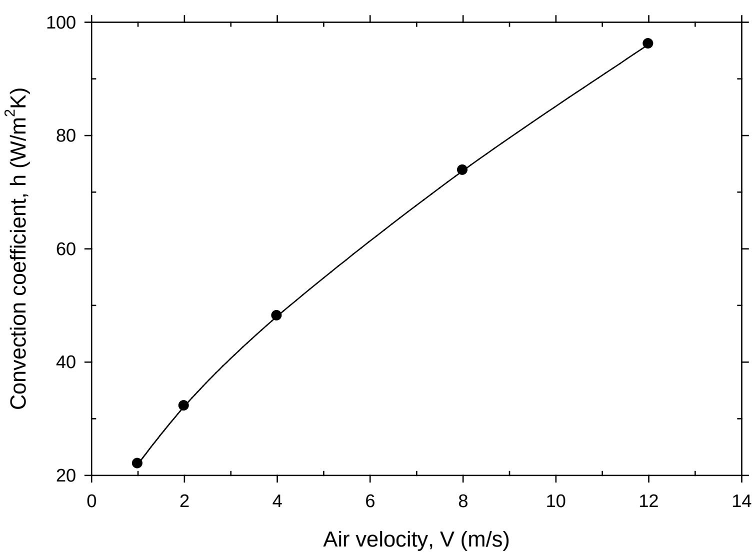
where $\dot{W}/L$ is the electrical power dissipated per unit length of the cylinder. For the V = 1 m/s condition, using the data from the table given in the problem statement, find

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

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

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

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



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