Solutions Manual

for

Introduction to Thermodynamics and Heat Transfer

Yunus A. Cengel

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Chapter 9

MECHANISMS OF HEAT TRANSFER

Heat Transfer Mechanisms

9-1C 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 9-1) 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.

9-2C 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.

9-3C 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.

9-4C 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.

9-5C 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.

9-6C Conduction is expressed by Fourier's law of conduction as $\dot{Q}_{\text{cond}} = -kA\frac{dT}{dx}$ 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.

9-7C Convection involves fluid motion, conduction does not. In a solid we can have only conduction.

9-8C No. It is purely by radiation.

9-9C 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.

9-10C 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.

9-11C 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.

9-12C 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.

9-13C 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.

9-14C Diamond is a better heat conductor.

9-15C The rate of heat transfer through both walls can be expressed as

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

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

Therefore, heat transfer through the brick wall will be larger despite its higher thickness.

9-16C 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.

9-17C 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.

9-18C 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.

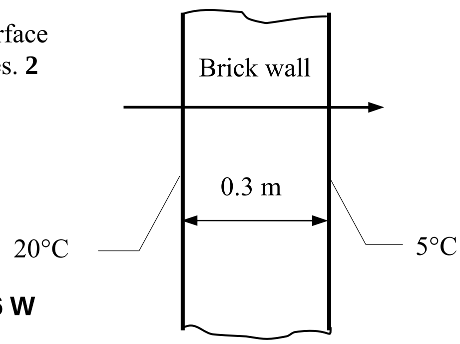
9-19C The thermal conductivity of an alloy of two metals will most likely be less than the thermal conductivities of both metals.

9-20 The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of heat transfer through the wall is to be determined.

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

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

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



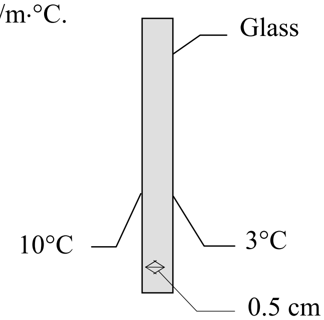
$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.69\text{ W/m}\cdot\text{°C})(4\times 7\text{ m}^2)\frac{(20-5)\text{°C}}{0.3\text{ m}} = \mathbf{966\ W}$$

9-21 The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat transfer through the glass in 5 h is to be determined.

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

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

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



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

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

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

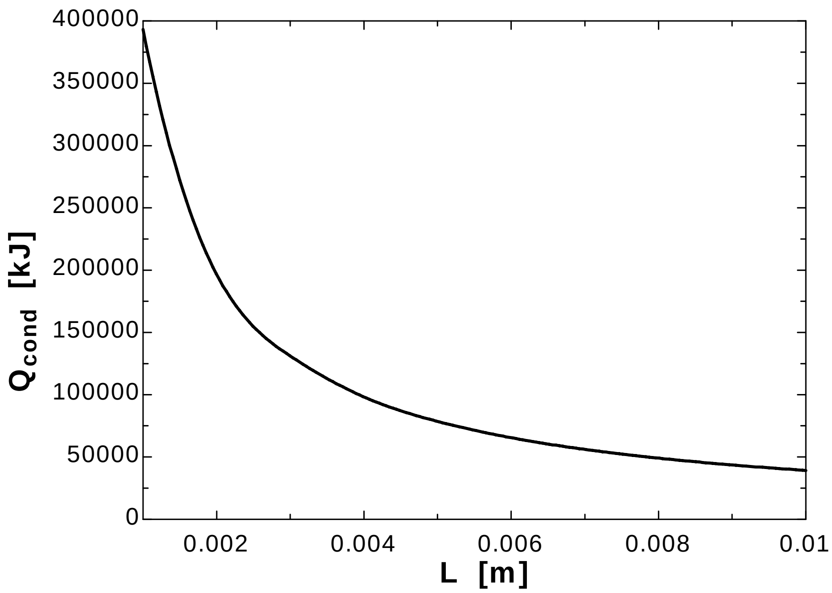
If the thickness of the glass doubled to 1 cm, then the amount of heat transfer will go down by half to **39,310 kJ**.

9-22 EES Prob. 9-21 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_9-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



9-23 Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface temperature of the bottom of the pan is given. The temperature of the outer surface is to be determined.

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

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

Analysis The heat transfer area is

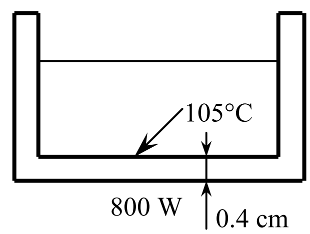
$$A = \pi r^2 = \pi (0.075\text{ m})^2 = 0.0177\text{ m}^2$$

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

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

Substituting,

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



which gives

$$T_2 = \mathbf{105.76°C}$$

9-24E 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

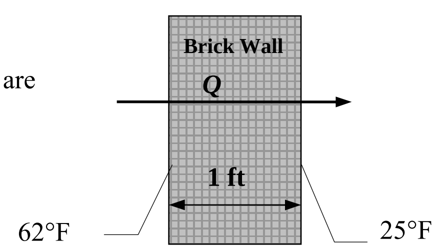
$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.42\text{ Btu/h.ft.°F})(200\text{ ft}^2)\frac{(62 - 25)°\text{F}}{1\text{ ft}} = \mathbf{3108\text{ Btu/h}}$$

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

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

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

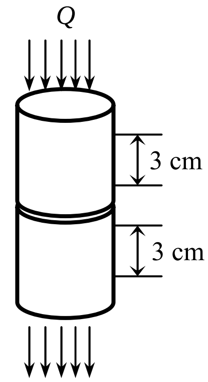


Therefore, the cost of the heat loss through the wall to the home owner that night is $0.51.

9-25 The thermal conductivity of a material is to be determined by ensuring one-dimensional heat conduction, and by measuring temperatures when steady operating conditions are reached.

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

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



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

The rate of heat flow through each sample is

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

Then the thermal conductivity of the sample becomes

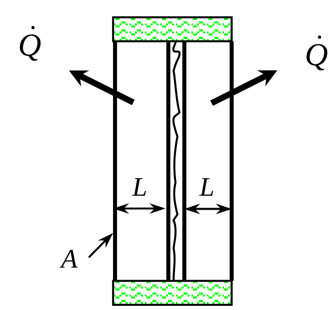
$$A = \frac{\pi D^2}{4} = \frac{\pi(0.04\text{ m})^2}{4} = 0.001257\text{ m}^2$$

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

9-26 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

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

9-27 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.

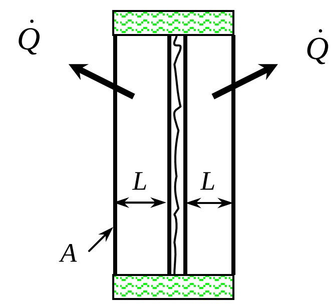
Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. **2** Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples.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

Analysis For each sample we have

$$\dot{Q} = 20/2 = 10 \text{ W}$$

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

$$\Delta T = 82 - 74 = 8°\text{C}$$



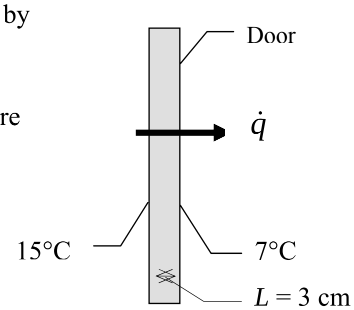
Then the thermal conductivity of the material becomes

$$\dot{Q} = kA\frac{\Delta T}{L} \longrightarrow k = \frac{\dot{Q}L}{A\Delta T} = \frac{(10\text{ W})(0.005\text{ m})}{(0.01\text{ m}^2)(8°\text{C})} = \mathbf{0.625\ W/m\cdot°C}$$

9-28 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 directly from Fourier's relation to be



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

9-29 The rate of radiation heat transfer between a person and the surrounding surfaces at specified temperatures is to be determined in summer and in winter.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer by convection is not considered. **3** The person is completely surrounded by the interior surfaces of the room. **4** The surrounding surfaces are at a uniform temperature.

Properties The emissivity of a person is given to be ε = 0.95

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

(*a*) Summer: T_{surr} = 23+273=296

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A_s (T_s^4 - T_{\text{surr}}^4)$$
$$= (0.95)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.6\ \text{m}^2)[(32+273)^4 - (296\ \text{K})^4]\text{K}^4$$
$$= \mathbf{84.2\ W}$$

(*b*) Winter: T_{surr} = 12+273= 285 K

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

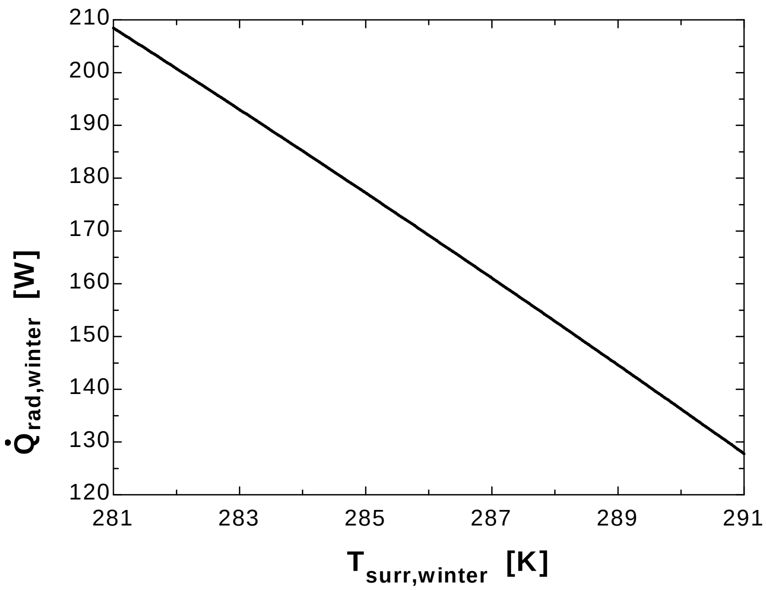
Discussion Note that the radiation heat transfer from the person more than doubles in winter.

9-30 EES Prob. 9-29 is reconsidered. The rate of radiation heat transfer in winter as a function of the temperature of the inner surface of the room is to be plotted.

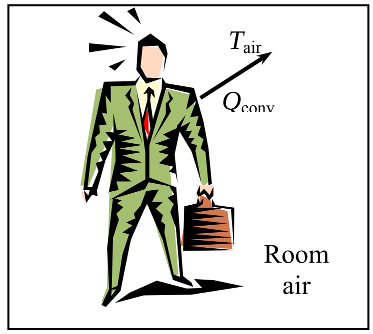
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_infinity=(20+273) [K]
T_surr_winter=(12+273) [K]
T_surr_summer=(23+273) [K]
A=1.6 [m^2]
epsilon=0.95
T_s=(32+273) [K]

"ANALYSIS"
sigma=5.67E-8 [W/m^2-K^4] "Stefan-Boltzman constant"
Q_dot_rad_summer=epsilon*sigma*A*(T_s^4-T_surr_summer^4)
Q_dot_rad_winter=epsilon*sigma*A*(T_s^4-T_surr_winter^4)
```

$T_{surr, winter}$ [K]	$Q_{rad, winter}$ [W]
281	208.5
282	200.8
283	193
284	185.1
285	177.2
286	169.2
287	161.1
288	152.9
289	144.6
290	136.2
291	127.8



9-31 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.

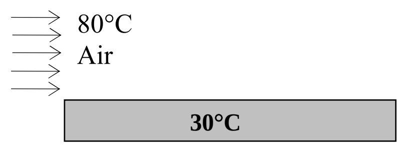
Analysis The heat transfer surface area of the person is

$$A_s = \pi DL = \pi(0.3\text{ m})(1.70\text{ m}) = 1.602\text{ m}^2$$

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

$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (20\text{ W/m}^2\cdot{}^\circ\text{C})(1.602\text{ m}^2)(34-18)^\circ\text{C} = \mathbf{513\text{ W}}$$

9-32 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

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

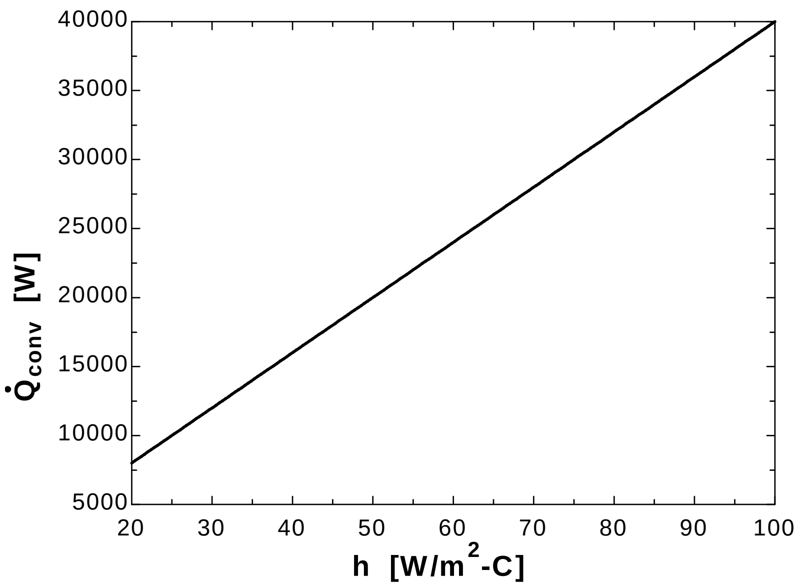
9-33 EES Prob. 9-32 is reconsidered. The rate of heat transfer as a function of the heat transfer coefficient is to be plotted.

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)

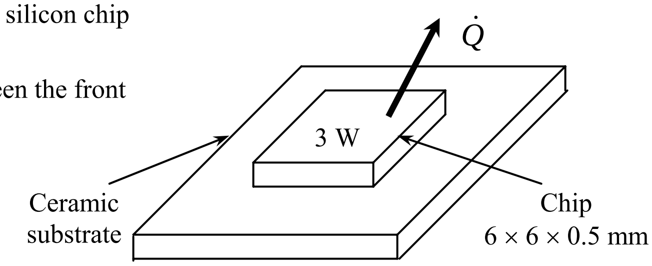
h [W/m^2.C]	Q_{conv} [W]
20	8000
30	12000
40	16000
50	20000
60	24000
70	28000
80	32000
90	36000
100	40000



9-34 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

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

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

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

9-35 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.

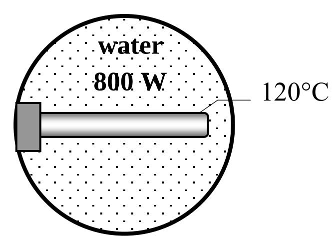
Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-15).

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

$$Q_{\text{in}} = mc(T_2 - T_1)$$

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

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



The surface area of the wire is

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

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

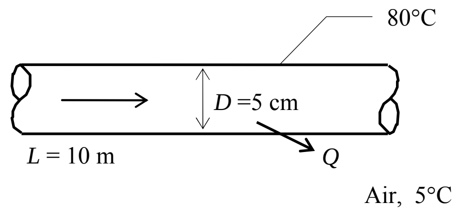
$$h_1 = \frac{\dot{Q}}{A_s(T_s - T_{\infty 1})} = \frac{800\text{ W}}{(0.00628\text{ m}^2)(120-20)°\text{C}} = \mathbf{1274\text{ W/m}^2\cdot°C}$$

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

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

9-36 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

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

9-37 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.

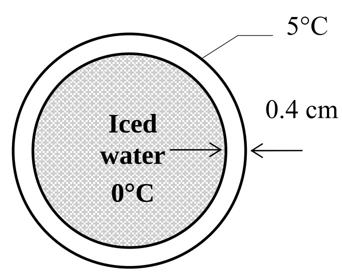
Properties The thermal conductivity of iron is k = 80.2 W/m·°C (Table A-24). The heat of fusion of water is given to be 333.7 kJ/kg.

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

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



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

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

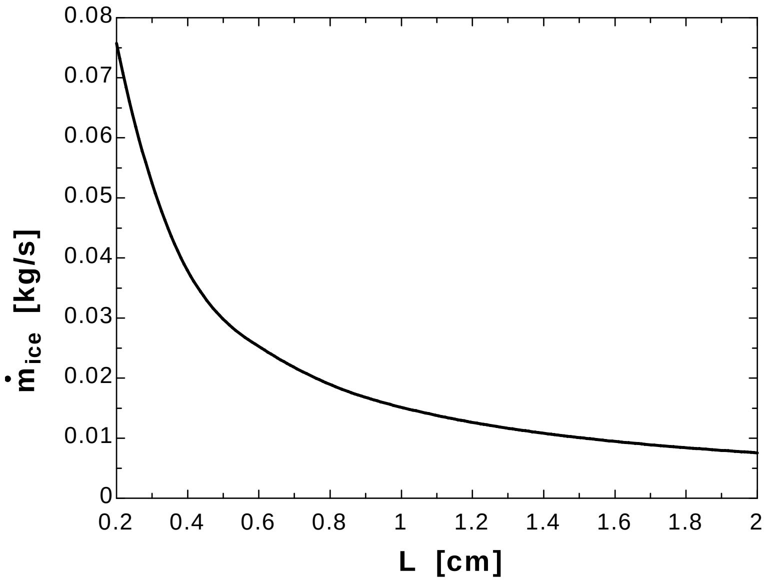
Discussion We should point out that this result is slightly in error for approximating a curved wall as a plain wall. The error in this case is very small because of the large diameter to thickness ratio. For better accuracy, we could use the inner surface area (D = 19.2 cm) or the mean surface area (D = 19.6 cm) in the calculations.

9-38 EES Prob. 9-37 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=0.4 [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]
0.2	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

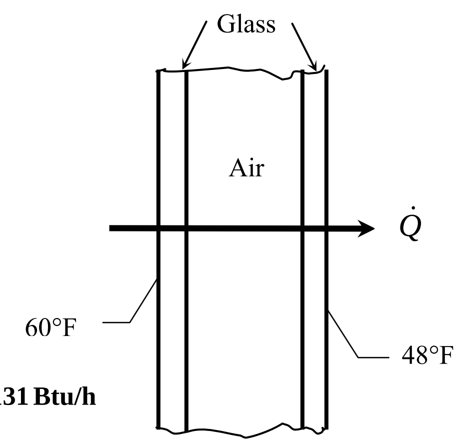


9-39E The inner and outer glasses of a double pane window with a 0.5-in air space are at specified temperatures. The rate of heat transfer through the window is to be determined

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-22E).

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



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

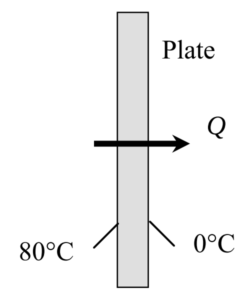
$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.01419\text{ Btu/h.ft.°F})(16\text{ ft}^2)\frac{(60-48)\text{°F}}{0.25/12\text{ ft}} = \mathbf{131\text{ Btu/h}}$$

9-40 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

$$\dot{Q} = kA\frac{T_1 - T_2}{L} \rightarrow k = \frac{(\dot{Q}/A)L}{(T_1 - T_2)} = \frac{(500\text{ W/m}^2)(0.02\text{ m})}{(80-0)\text{°C}} = \mathbf{0.125\text{ W/m}\cdot\text{°C}}$$



9-41 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.

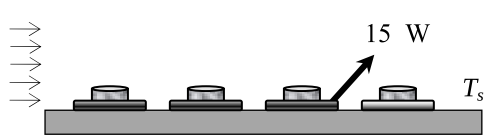
Assumptions **1** Steady operating conditions exist. **2** The entire plate is nearly isothermal. **3** Thermal properties of the wall are constant. **4** The exposed surface area of the transistor can be taken to be equal to its base area. **5** Heat transfer by radiation is disregarded. **6** The convection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



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

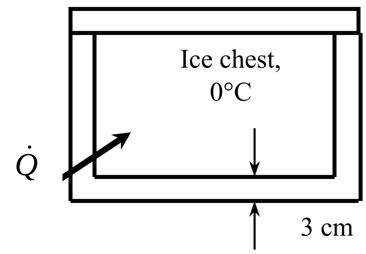
9-42 A styrofoam ice chest is initially filled with 40 kg of ice at 0°C. The time it takes for the ice in the chest to melt completely is to be determined.

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

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

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

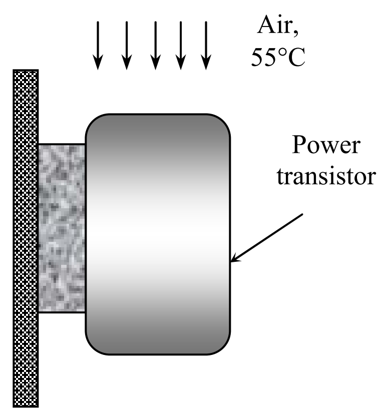
$$Q = mh_{if} = (28\ \text{kg})(333.7\ \text{kJ/kg}) = 9344\ \text{kJ}$$

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

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{9344{,}000\ \text{J}}{4.72\ \text{J/s}} = 1.98\times10^6\ \text{s} = \mathbf{22.9\ days}$$

9-43 A transistor mounted on a circuit board is cooled by air flowing over it. The transistor case temperature is not to exceed 70°C when the air temperature is 55°C. The amount of power this transistor can dissipate safely is to be determined.

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



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

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

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

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

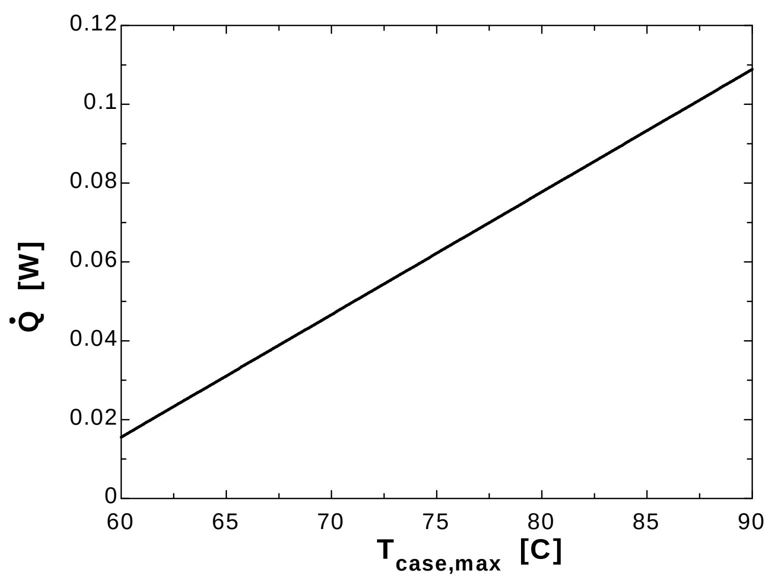
Therefore, the amount of power this transistor can dissipate safely is 0.047 W.

9-44 EES Prob. 9-43 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.

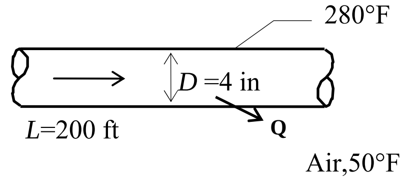
```
"GIVEN"
L=0.004 [m]
D=0.006 [m]
h=30 [W/m^2-C]
T_infinity=55 [C]
T_case_max=70 [C]

"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



9-45E 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

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

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

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

$$Q = \dot{Q}\Delta t = (289{,}000\text{ Btu/h})(365\times 24\text{ h/yr}) = 2.531\times 10^9\text{ Btu/yr}$$

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

$$\text{Annual Energy Loss} = \frac{2.531\times 10^9\text{ Btu/yr}}{0.86}\left(\frac{1\text{ therm}}{100{,}000\text{ Btu}}\right) = 29{,}435\text{ therms/yr}$$

Then the annual cost of the energy lost becomes

$$\text{Energy cost} = (\text{Annual energy loss})(\text{Unit cost of energy})$$
$$= (29{,}435\text{ therms/yr})(\$1.10/\text{therm}) = \mathbf{\$32{,}380/yr}$$

9-46 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.

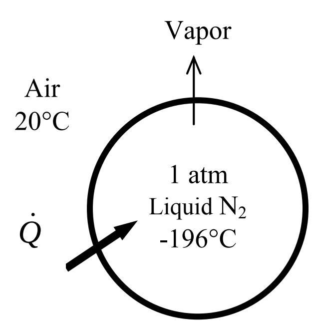
Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is disregarded. **3** The convection heat transfer coefficient is constant and uniform over the surface. **4** The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the nitrogen inside.

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

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

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



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

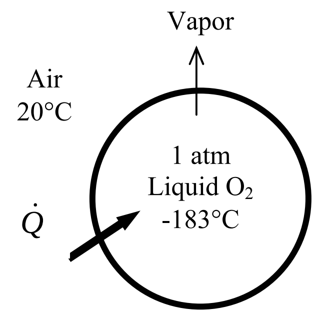
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{271.430\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{1.37\text{ kg/s}}$$

9-47 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 determined to be

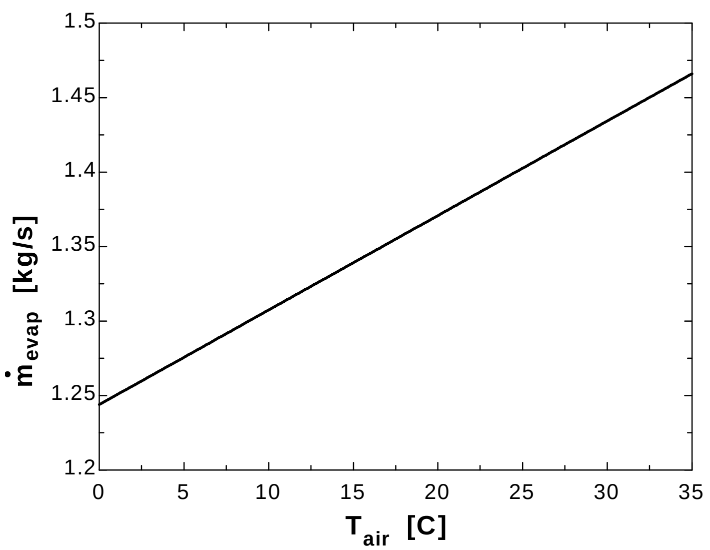
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{255.120\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{1.20\ kg/s}$$

9-48 EES Prob. 9-46 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



9-49 A person with a specified surface temperature is subjected to radiation heat transfer in a room at specified wall temperatures. The rate of radiation heat loss from the person is to be determined.

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

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

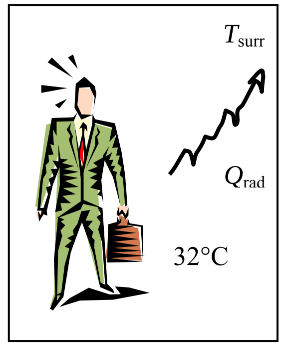
Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are

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

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

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

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



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

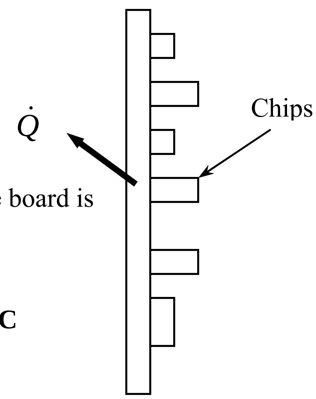
9-50 A circuit board houses 80 closely spaced logic chips on one side, each dissipating 0.06 W. All the heat generated in the chips is conducted across the circuit board. The temperature difference between the two sides of the circuit board is to be determined.

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

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

Analysis The total rate of heat dissipated by the chips is

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



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

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

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

Discussion Note that the circuit board is nearly isothermal.

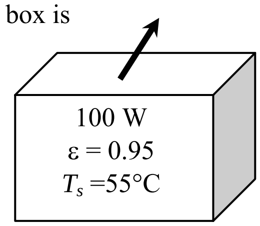
9-51 A sealed electronic box dissipating a total of 100 W of power is placed in a vacuum chamber. If this box is to be cooled by radiation alone and the outer surface temperature of the box is not to exceed 55°C, the temperature the surrounding surfaces must be kept is to be determined.

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

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

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

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



The radiation heat transfer from the box can be expressed as

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

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

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

9-52E Using the conversion factors between W and Btu/h, m and ft, and K and R, the Stefan-Boltzmann constant $\sigma = 5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4$ is to be expressed in the English unit, $\text{Btu/h}\cdot\text{ft}^2\cdot\text{R}^4$.

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

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

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

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

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

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

9-53E Using the conversion factors between W and Btu/h, m and ft, and °C and °F, the convection coefficient in SI units is to be expressed in Btu/h·ft^2·°F.

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

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

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

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

$$1°\text{C} = 1.8°\text{F}$$

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

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

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

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

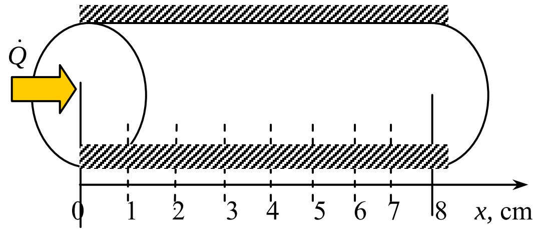
9-54 A cylindrical sample of a material is used to determine its thermal conductivity. The temperatures measured along the sample are tabulated. The variation of temperature along the sample is to be plotted and the thermal conductivity of the sample material is to be calculated.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is one-dimensional (axial direction).

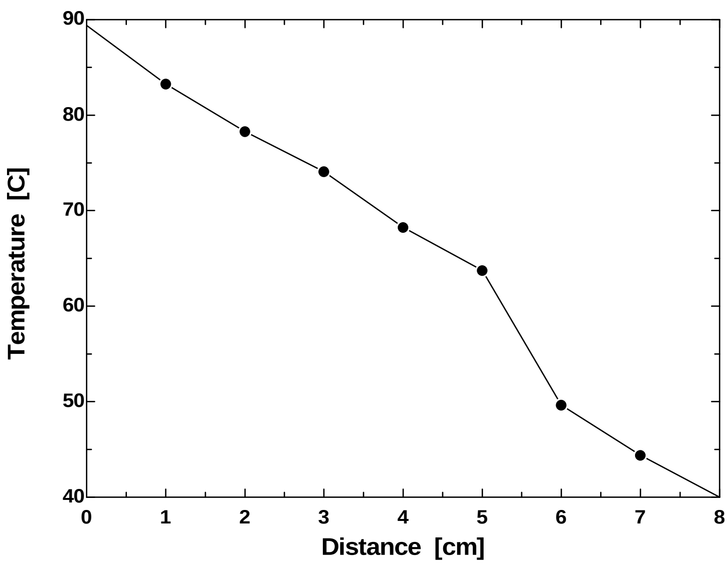
Analysis The following table gives the results of the calculations. The plot of temperatures is also given below. A sample calculation for the thermal conductivity is as follows:

$$A = \frac{\pi D^2}{4} = \frac{\pi (0.025\text{ m})^2}{4} = 0.00049\text{ m}^2$$

$$k_{12} = \frac{\dot{Q}L}{A(T_1 - T_2)} = \frac{(83.45\text{ W})(0.010\text{ m})}{(0.00049\text{ m}^2)(6.13°\text{C})} = 277.8\text{ W/m}\cdot°\text{C})$$



Distance from left face, cm	Temperature, °C	Temperature difference (ºC)	Thermal conductivity (W/m·ºC)
0	T_1= 89.38	$T_{9\text{-}}T_2$= 6.13	277.8
1	T_2= 83.25	T_2-T_3= 4.97	342.7
2	T_3= 78.28	T_3-T_4= 4.18	407.4
3	T_4= 74.10	T_4-T_5= 5.85	291.1
4	T_5= 68.25	T_5-T_6= 4.52	376.8
5	T_6=63.73	T_6-T_7= 14.08	120.9
6	T_7= 49.65	T_7-T_8= 5.25	324.4
7	T_8= 44.40	T_8-T_9= 4.40	387.1
8	T_9= 40.00	$T_{9\text{-}}T_2$= 6.13	277.8



Discussion It is observed from the calculations in the table and the plot of temperatures that the temperature reading corresponding to the calculated thermal conductivity of 120.9 is probably not right, and it should be discarded.

9-55 An aircraft flying under icing conditions is considered. The temperature of the wings to prevent ice from forming on them is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer coefficient is constant.

Properties The heat of fusion and the density of ice are given to be 333.7 kJ/kg and 920 kg/m^3, respectively.

Analysis The temperature of the wings to prevent ice from forming on them is determined to be

$$T_{\text{wing}} = T_{\text{ice}} + \frac{\rho V h_{if}}{h} = 0°\text{C} + \frac{(920\ \text{kg/m}^3)(0.001/60\ \text{m/s})(333{,}700\ \text{J/kg})}{150\ \text{W/m}^2\cdot°\text{C}} = \mathbf{34.1°C}$$

Simultaneous Heat Transfer Mechanisms

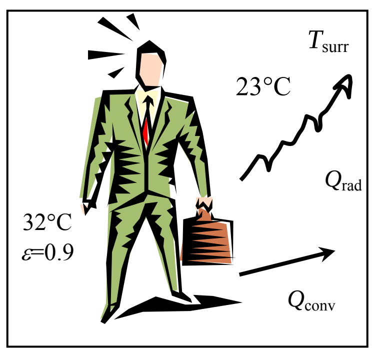
9-56C All three modes of heat transfer can not occur simultaneously in a medium. A medium may involve two of them simultaneously.

9-57C (*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.

9-58C 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.

9-59C 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.

9-60 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

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

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

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

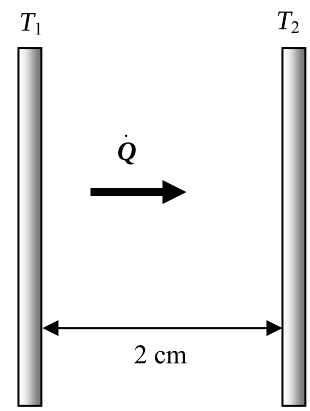
Discussion Note that heat transfer from the person by evaporation, which is of comparable magnitude, is not considered in this problem.

9-61 Two large plates at specified temperatures are held parallel to each other. The rate of heat transfer between the plates is to be determined for the cases of still air, evacuation, regular insulation, and super insulation between the plates.

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

Properties The thermal conductivities are k = 0.00015 W/m·°C for super insulation, k = 0.01979 W/m·°C at -50°C (Table A-22) for air, and k = 0.036 W/m·°C for fiberglass insulation.

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



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

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

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

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

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

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

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

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

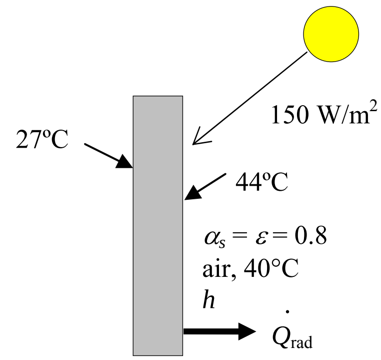
$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{cond}} = kA\frac{T_1 - T_2}{L} = (0.00015\text{ W/m}\cdot{}^\circ\text{C})(1\text{ m}^2)\frac{(290-150)\text{ K}}{0.02\text{ m}} = \mathbf{1.05\ W}$$

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

9-62 The outer surface of a wall is exposed to solar radiation. The effective thermal conductivity of the wall is to be determined.

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

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



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

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

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

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

$$+(0.8)(150\text{ W/m}^2)$$

Solving for k gives

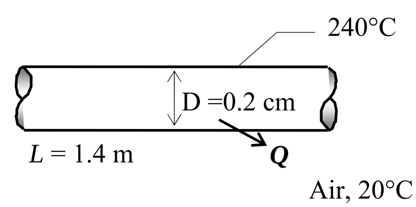
$$k = \mathbf{0.961\ W/m\cdot°C}$$

9-63 The convection heat transfer coefficient for heat transfer from an electrically heated wire to air is to be determined by measuring temperatures when steady operating conditions are reached and the electric power consumed.

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

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

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



The surface area of the wire is

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

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

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

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

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

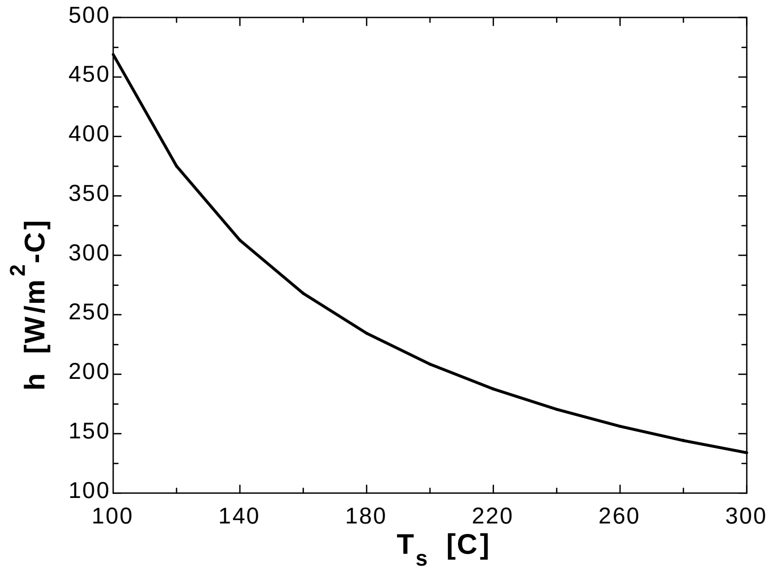
Discussion If the temperature of the surrounding surfaces is equal to the air temperature in the room, the value obtained above actually represents the combined convection and radiation heat transfer coefficient.

9-64 EES Prob. 9-63 is reconsidered. The convection heat transfer coefficient as a function of the wire surface temperature is to be plotted.

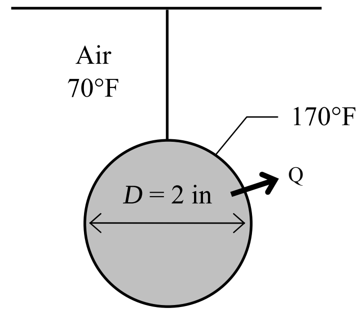
Analysis The problem is solved using EES, and the solution is given below.

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

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

T_s [C]	h [W/m^2.C]
100	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



9-65E A spherical ball whose surface is maintained at a temperature of 170°F is suspended in the middle of a room at 70°F. The total rate of heat transfer from the ball is to be determined.



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

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

Analysis The heat transfer surface area is

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

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

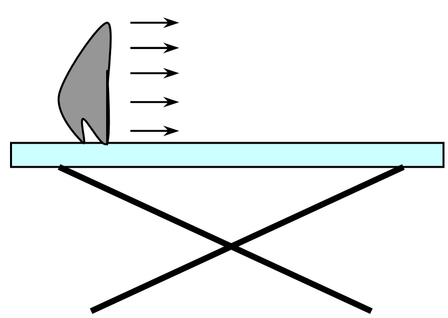
$$\dot{Q}_{\text{conv}} = hA_s\Delta T = (15 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F})(0.08727 \text{ ft}^2)(170-70)°\text{F} = 130.9 \text{ Btu/h}$$

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

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

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.

9-66 CD EES 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 $\dot{Q}_{\text{conv}} = hA_s\Delta T = (35 \text{ W/m}^2\cdot\text{K})(0.02 \text{ m}^2)(T_s - 293 \text{ K}) = 0.7(T_s - 293 \text{ K})$

and

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

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

Solving by trial and error gives

$$T_s = \mathbf{947\ K} = \mathbf{674°C}$$

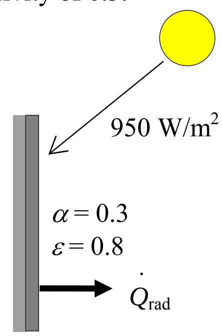
Discussion We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 947 K.

9-67 A spacecraft in space absorbs solar radiation while losing heat to deep space by thermal radiation. The surface temperature of the spacecraft is to be determined when steady conditions are reached.

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

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

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



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

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

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

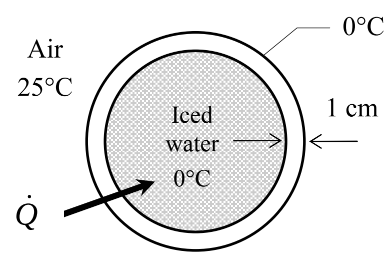
Canceling the surface area A and solving for T_s gives

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

9-68 A spherical tank located outdoors is used to store iced water at 0°C. The rate of heat transfer to the iced water in the tank and the amount of ice at 0°C that melts during a 24-h period are to be determined.

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

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



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

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

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

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

$$\dot{Q}_{\text{rad}} = \varepsilon A_s \sigma (T_{\text{surr}}^4 - T_s^4) = (0.75)(28.65 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(288 \text{ K})^4 - (273 \text{ K})^4] = 1614 \text{ W}$$

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

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

$$Q = \dot{Q}\Delta t = (23.102 \text{ kJ/s})(24 \times 3600 \text{ s}) = 1{,}996{,}000 \text{ kJ}$$

Then the amount of ice that melts during this period becomes

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

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

9-69 CD EES 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.

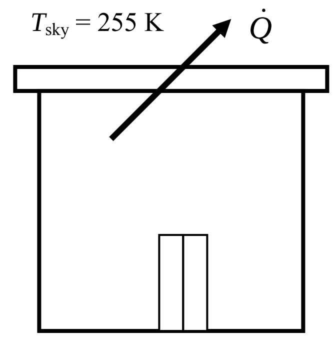
Assumptions **1** Steady operating conditions exist. **2** The emissivity and thermal conductivity of the roof are constant.

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



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

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

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

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

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

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

Then the amount of natural gas consumption during a 9-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

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

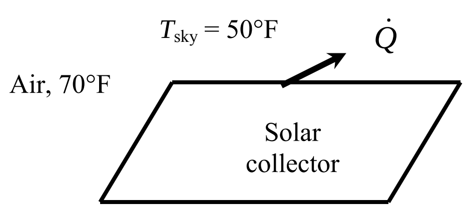
9-70E A flat plate solar collector is placed horizontally on the roof of a house. The rate of heat loss from the collector by convection and radiation during a calm day are to be determined.

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

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

Analysis The exposed surface area of the collector is

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



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

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

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

$$= 3551\text{ Btu/h}$$

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

Review Problems

9-71 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

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

Windy weather

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

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

In windy air it would be

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

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

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

which gives

$$T_{\text{effective}} = -12.7^\circ\text{C}$$

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

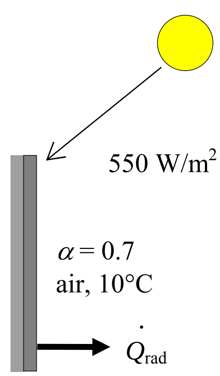
$$F_{\text{wind-chill}} = 20 - (-12.7) = \mathbf{32.7^\circ C}$$

9-72 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 α = 0.7.

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



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

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

$$0.7\times A\times 550\text{ W/m}^2 = (25\text{ W/m}^2\cdot{}^\circ\text{C})A_s(T_s - 10)$$

Canceling the surface area A_s and solving for T_s gives

$$T_s = \mathbf{25.4^\circ C}$$

9-73 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.

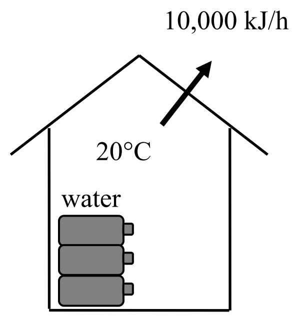
Properties The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-15).

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

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

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

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



or

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

Substituting,

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

It gives

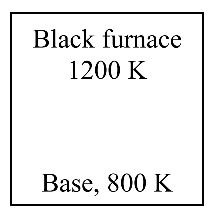
$$T_1 = \mathbf{77.4°C}$$

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

9-74 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

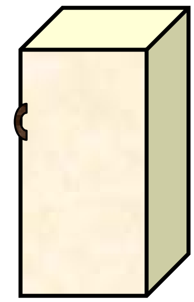
$$\begin{aligned}\dot{Q}_{rad,base} &= \varepsilon A\sigma(T_{base}^4 - T_{surr}^4)\\ &= (0.7)(3\times3 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2.\text{K}^4)[(1200 \text{ K})^4 - (800 \text{ K})^4]\\ &= 594{,}400 \text{ W} = \mathbf{594\,kW}\end{aligned}$$

9-75 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

$$A_{\text{total}} = 2\left[(1.8\times1.2)+(1.8\times0.8)+(1.2\times0.8)\right] = 9.12\ \text{m}^2$$



Since the refrigerator has a COP of 1.5, the rate of heat removal from the refrigerated space, which is equal to the rate of heat gain in steady operation, is

$$\dot{Q} = \dot{W}_e \times \text{COP} = (600\ \text{W})\times1.5 = 900\ \text{W}$$

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

$$\dot{Q}_{\text{ave}} = \dot{Q}/4 = (900\ \text{W})/4 = 225\ \text{W}$$

Then the thermal conductivity of refrigerator walls is determined to be

$$\dot{Q}_{\text{ave}} = kA\frac{\Delta T_{\text{ave}}}{L} \longrightarrow k = \frac{\dot{Q}_{\text{avg}}L}{A\Delta T_{\text{avg}}} = \frac{(225\ \text{W})(0.03\ \text{m})}{(9.12\ \text{m}^2)(17-6)^\circ\text{C}} = \mathbf{0.0673\ W/m\cdot{}^\circ C}$$

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

$$\Delta t = 365\times24/4 = 2190\ \text{h}$$

Then the total amount of electricity consumed during a one-year period and the annular cost of operating this refrigerator are

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

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

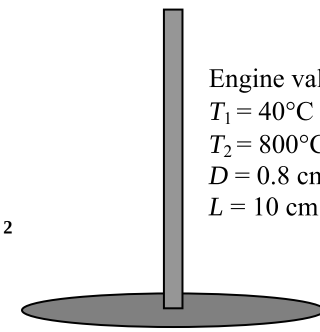
9-76 Engine valves are to be heated in a heat treatment section. The amount of heat transfer, the average rate of heat transfer, the average heat flux, and the number of valves that can be heat treated daily are to be determined.

Assumptions Constant properties given in the problem can be used.

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

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

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



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

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

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

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

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

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

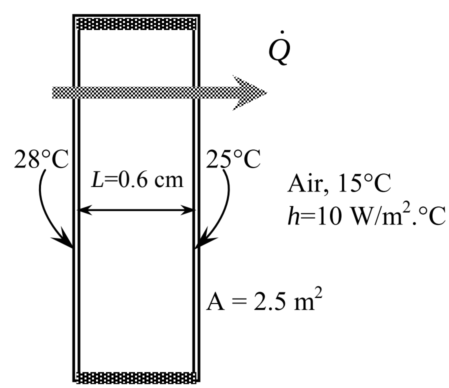
9-77 The glass cover of a flat plate solar collector with specified inner and outer surface temperatures is considered. The fraction of heat lost from the glass cover by radiation is to be determined.

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

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.7\text{ W/m}\cdot°\text{C})(2.5\text{ m}^2)\frac{(28-25)°\text{C}}{0.006\text{ m}} = 875\text{ W}$$



The rate of heat transfer from the glass by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T = (10\text{ W/m}^2\cdot°\text{C})(2.5\text{ m}^2)(25-15)°\text{C} = 250\text{ W}$$

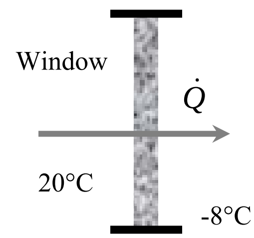
Under steady conditions, the heat transferred through the cover by conduction should be transferred from the outer surface by convection and radiation. That is,

$$\dot{Q}_{\text{rad}} = \dot{Q}_{\text{cond}} - \dot{Q}_{\text{conv}} = 875 - 250 = 625\text{ W}$$

Then the fraction of heat transferred by radiation becomes

$$f = \frac{\dot{Q}_{\text{rad}}}{\dot{Q}_{\text{cond}}} = \frac{625}{875} = \mathbf{0.714} \quad \text{(or 71.4\%)}$$

9-78 The range of U-factors for windows are given. The range for the rate of heat loss through the window of a house is to be determined.



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

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

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

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

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

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

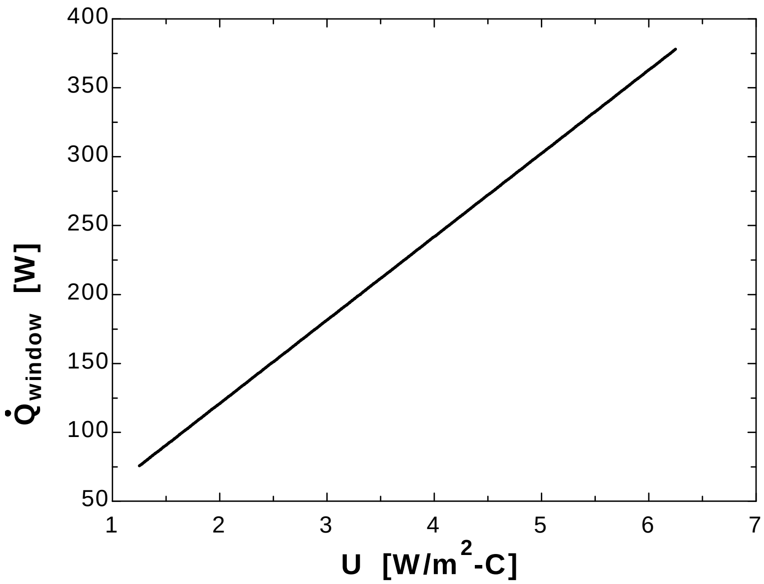
Discussion Note that the rate of heat loss through windows of identical size may differ by a factor of 5, depending on how the windows are constructed.

9-79 EES Prob. 9-78 is reconsidered. The rate of heat loss through the window as a function of the *U*-factor is to be plotted.

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

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

"ANALYSIS"
Q_dot_window=U*A*(T_9-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

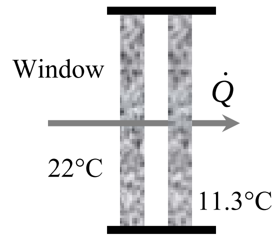


9-80 The windows of a house in Atlanta are of double door type with wood frames and metal spacers. The average rate of heat loss through the windows in winter is to be determined.

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

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

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



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

$$\dot{Q}_{\text{window, avg}} = (2.50 \text{ W/m}^2 \cdot °\text{C})(20 \text{ m}^2)(22 - 11.3)°\text{C} = \mathbf{535\ W}$$

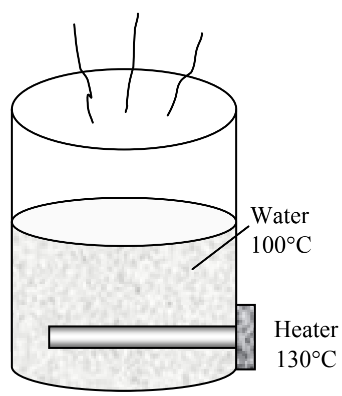
Discussion This is the "average" rate of heat transfer through the window in winter in the absence of any infiltration.

9-81 Boiling experiments are conducted by heating water at 1 atm pressure with an electric resistance wire, and measuring the power consumed by the wire as well as temperatures. The boiling heat transfer coefficient is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the water container are negligible.

Analysis The heat transfer area of the heater wire is

$$A = \pi DL = \pi(0.002 \text{ m})(0.50 \text{ m}) = 0.003142 \text{ m}^2$$

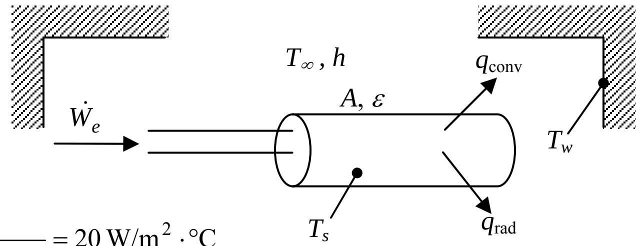


Noting that 4100 W of electric power is consumed when the heater surface temperature is 130°C, the boiling heat transfer coefficient is determined from Newton's law of cooling $\dot{Q} = hA(T_s - T_{\text{sat}})$ to be

$$h = \frac{\dot{Q}}{A(T_s - T_{\text{sat}})} = \frac{4100 \text{ W}}{(0.003142 \text{ m}^2)(130 - 100)°\text{C}} = \mathbf{43{,}500\ W/m^2 \cdot °C}$$

9-82 An electric heater placed in a room consumes 500 W power when its surfaces are at 120°C. The surface temperature when the heater consumes 700 W is to be determined without and with the consideration of radiation.

Assumptions **1** Steady operating conditions exist. **2** The temperature is uniform over the surface.



Analysis (*a*) Neglecting radiation, the convection heat transfer coefficient is determined from

$$h = \frac{\dot{Q}}{A(T_s - T_\infty)} = \frac{500\text{ W}}{(0.25\text{ m}^2)(120-20)^\circ\text{C}} = 20\text{ W/m}^2\cdot{}^\circ\text{C}$$

The surface temperature when the heater consumes 700 W is

$$T_s = T_\infty + \frac{\dot{Q}}{hA} = 20^\circ\text{C} + \frac{700\text{ W}}{(20\text{ W/m}^2\cdot{}^\circ\text{C})(0.25\text{ m}^2)} = \mathbf{160^\circ C}$$

(*b*) Considering radiation, the convection heat transfer coefficient is determined from

$$h = \frac{\dot{Q} - \varepsilon A\sigma(T_s^4 - T_{surr}^4)}{A(T_s - T_\infty)}$$

$$= \frac{500\text{ W - }(0.75)(0.25\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)\left[(393\text{ K})^4 - (283\text{ K})^4\right]}{(0.25\text{ m}^2)(120-20)^\circ\text{C}} = 12.58\text{ W/m}^2\cdot{}^\circ\text{C}$$

Then the surface temperature becomes

$$\dot{Q} = hA(T_s - T_\infty) + \varepsilon A\sigma(T_s^4 - T_{surr}^4)$$

$$700 = (12.58)(0.25)(T_s - 293) + (0.75)(0.25)(5.67\times10^{-8})\left[T_s^4 - (283\text{ K})^4\right]$$

$$T_s = 425.9\text{ K} = \mathbf{152.9^\circ C}$$

Discussion Neglecting radiation changed T_s by more than 7°C, so assumption is not correct in this case.

9-83 . . . 9-85 Design and Essay Problems