Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

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Chapter 16

HEATING AND COOLING OF BUILDINGS

A Brief History

16-1C Ice can be made by evacuating the air in a water tank. During evacuation, vapor is also thrown out, and thus the vapor pressure in the tank drops, causing a difference between the vapor pressures at the water surface and in the tank. This pressure difference is the driving force of vaporization, and forces the liquid to evaporate. But the liquid must absorb the heat of vaporization before it can vaporize, and it absorbs it from the liquid and the air in the neighborhood, causing the temperature in the tank to drop. The process continues until water starts freezing. The process can be made more efficient by insulating the tank well so that the entire heat of vaporization comes essentially from the water.

16-2C The first ammonia absorption refrigeration system was developed in 1851 by Ferdinand Carre. The formulas related to dry-bulb, wet-bulb, and dew-point temperatures were developed by Willis Carrier in 1911.

16-3C The concept of *heat pump* was conceived by Sadi Carnot in 1824. The first heat pump was built by T. G. N. Haldane in 1930, and the heat pumps were mass produced in 1952.

Human Body and Thermal Comfort

16-4C The metabolism refers to the burning of foods such as carbohydrates, fat, and protein in order to perform the necessary bodily functions. The metabolic rate for an average man ranges from 108 W while reading, writing, typing, or listening to a lecture in a classroom in a seated position to 1250 W at age 20 (730 at age 70) during strenuous exercise. The corresponding rates for women are about 30 percent lower. Maximum metabolic rates of trained athletes can exceed 2000 W. We are interested in metabolic rate of the occupants of a building when we deal with heating and air conditioning because the metabolic rate represents the rate at which a body generates heat and dissipates it to the room. This body heat contributes to the heating in winter, but it adds to the cooling load of the building in summer.

16-5C The metabolic rate is proportional to the size of the body, and the metabolic rate of women, in general, is lower than that of men because of their smaller size. Clothing serves as insulation, and the thicker the clothing, the lower the environmental temperature that feels comfortable.

16-6C Asymmetric thermal radiation is caused by the *cold surfaces* of large windows, uninsulated walls, or cold products on one side, and the *warm surfaces* of gas or electric radiant heating panels on the walls or ceiling, solar heated masonry walls or ceilings on the other. Asymmetric radiation causes discomfort by exposing different sides of the body to surfaces at different temperatures and thus to different rates of heat loss or gain by radiation. A person whose left side is exposed to a cold window, for example, will feel like heat is being drained from that side of his or her body.

16-7C (*a*) Draft causes undesired local cooling of the human body by exposing parts of the body to high heat transfer coefficients. (*b*) Direct contact with *cold floor surfaces* causes localized discomfort in the feet by excessive heat loss by conduction, dropping the temperature of the bottom of the feet to uncomfortable levels.

16-8C Stratification is the formation of vertical still air layers in a room at difference temperatures, with highest temperatures occurring near the ceiling. It is likely to occur at places with high ceilings. It causes discomfort by exposing the head and the feet to different temperatures. This effect can be prevented or minimized by using destratification fans (ceiling fans running in reverse).

16-9C It is necessary to ventilate buildings to provide adequate fresh air and to get rid of excess carbon dioxide, contaminants, odors, and humidity. Ventilation increases the energy consumption for heating in winter by replacing the warm indoors air by the colder outdoors air. Ventilation also increases the energy consumption for cooling in summer by replacing the cold indoors air by the warm outdoors air. It is not a good idea to keep the bathroom fans on all the time since they will waste energy by expelling conditioned air (warm in winter and cool in summer) by the unconditioned outdoor air.

Heat Transfer from the Human Body

16-10C Yes, roughly one-third of the metabolic heat generated by a person who is resting or doing light work is dissipated to the environment by convection, one-third by evaporation, and the remaining one-third by radiation.

16-11C Sensible heat is the energy associated with a temperature change. The sensible heat loss from a human body increases as (*a*) the skin temperature increases, (*b*) the environment temperature decreases, and (*c*) the air motion (and thus the convection heat transfer coefficient) increases.

16-12C Latent heat is the energy released as water vapor condenses on cold surfaces, or the energy absorbed from a warm surface as liquid water evaporates. The latent heat loss from a human body increases as (*a*) the skin wettedness increases and (*b*) the relative humidity of the environment decreases. The rate of evaporation from the body is related to the rate of latent heat loss by $\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg}$ where h_{fg} is the latent heat of vaporization of water at the skin temperature.

16-13C The insulating effect of clothing is expressed in the unit **clo** with 1 clo = 0.155 m^2.°C/W = 0.880 ft^2.°F.h/Btu. Clothing serves as insulation, and thus reduces heat loss from the body by convection, radiation, and evaporation by serving as a resistance against heat flow and vapor flow. Clothing decreases heat gain from the sun by serving as a radiation shield.

16-14C (*a*) Heat is lost through the skin by convection, radiation, and evaporation. (*b*) The body loses both sensible heat by convection and latent heat by evaporation from the lungs, but there is no heat transfer in the lungs by radiation.

16-15C The *operative temperature* $T_{\text{operative}}$ is the average of the mean radiant and ambient temperatures weighed by their respective convection and radiation heat transfer coefficients, and is expressed as

$$T_{\text{operative}} = \frac{h_{\text{conv}} T_{\text{ambient}} + h_{\text{rad}} T_{\text{surr}}}{h_{\text{conv}} + h_{\text{rad}}} \cong \frac{T_{\text{ambient}} + T_{\text{surr}}}{2}$$

When the convection and radiation heat transfer coefficients are equal to each other, the operative temperature becomes the arithmetic average of the ambient and surrounding surface temperatures. Another environmental index used in thermal comfort analysis is the effective temperature, which combines the effects of temperature and humidity.

16-16 The convection heat transfer coefficient for a clothed person while walking in still air at a velocity of 0.5 to 2 m/s is given by $h = 8.6V^{0.53}$ where V is in m/s and h is in W/m^2.°C. The convection coefficients in that range vary from 5.96 W/m^2.°C at 0.5 m/s to 12.42 W/m^2.°C at 2 m/s. Therefore, at low velocities, the radiation and convection heat transfer coefficients are comparable in magnitude. But at high velocities, the convection coefficient is much larger than the radiation heat transfer coefficient.

Velocity, m/s	$h = 8.6V^{0.53}$ W/m^2.°C
0.50	5.96
0.75	7.38
1.00	8.60
1.25	9.68
1.50	10.66
1.75	11.57
2.00	12.42

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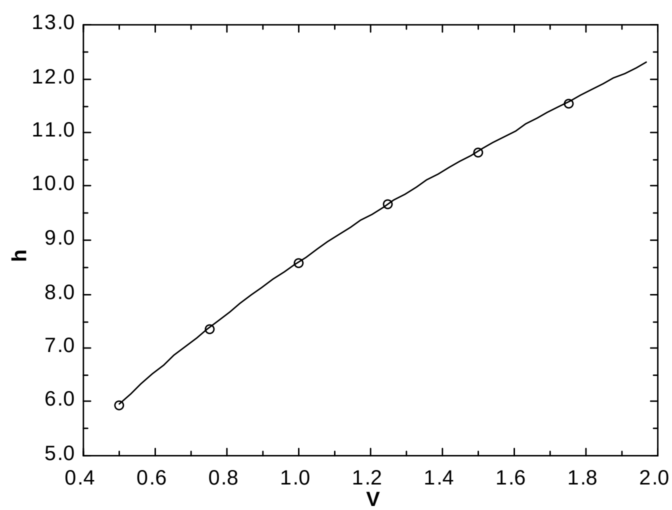
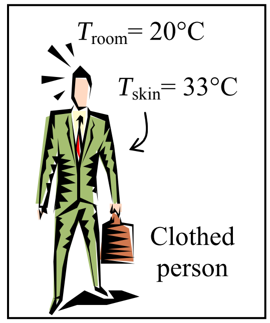


16-17 A man wearing summer clothes feels comfortable in a room at 20°C. The room temperature at which this man would feel thermally comfortable when unclothed is to be determined.

Assumptions **1** Steady conditions exist. **2** The latent heat loss from the person remains the same. **3** The heat transfer coefficients remain the same. **4** The air in the room is still (there are no winds or running fans). **5** The surface areas of the clothed and unclothed person are the same.



Analysis At low air velocities, the convection heat transfer coefficient for a standing man is given in Table 13-3 to be 4.0 W/m^2.°C. The radiation heat transfer coefficient at typical indoor conditions is 4.7 W/m^2.°C. Therefore, the heat transfer coefficient for a standing person for combined convection and radiation is

$$h_{\text{combined}} = h_{\text{conv}} + h_{\text{rad}} = 4.0 + 4.7 = 8.7 \text{ W/m}^2.°\text{C}$$

The thermal resistance of the clothing is given to be

$$R_{\text{cloth}} = 1.1 \text{ clo} = 1.1 \times 0.155 \text{ m}^2.°\text{C/W} = 0.171 \text{ m}^2.°\text{C/W}$$

Noting that the surface area of an average man is 1.8 m^2, the sensible heat loss from this person when clothed is determined to be

$$\dot{Q}_{\text{sensible,clothed}} = \frac{A_s(T_{\text{skin}} - T_{\text{ambient}})}{R_{\text{cloth}} + \dfrac{1}{h_{\text{combined}}}} = \frac{(1.8 \text{ m}^2)(33-20)°\text{C}}{0.171 \text{ m}^2.°\text{C/W} + \dfrac{1}{8.7 \text{ W/m}^2.°\text{C}}} = 82 \text{ W}$$

From heat transfer point of view, taking the clothes off is equivalent to removing the clothing insulation or setting $R_{\text{cloth}} = 0$. The heat transfer in this case can be expressed as

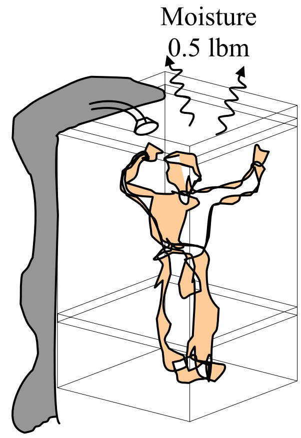
$$\dot{Q}_{\text{sensible,unclothed}} = \frac{A_s(T_{\text{skin}} - T_{\text{ambient}})}{\dfrac{1}{h_{\text{combined}}}} = \frac{(1.8 \text{ m}^2)(33 - T_{\text{ambient}})°\text{C}}{\dfrac{1}{8.7 \text{ W/m}^2.°\text{C}}}$$

To maintain thermal comfort after taking the clothes off, the skin temperature of the person and the rate of heat transfer from him must remain the same. Then setting the equation above equal to 82 W gives

$$T_{\text{ambient}} = \mathbf{27.8°C}$$

Therefore, the air temperature needs to be raised from 22 to 27.8°C to ensure that the person will feel comfortable in the room after he takes his clothes off. Note that the effect of clothing on latent heat is assumed to be negligible in the solution above. We also assumed the surface area of the clothed and unclothed person to be the same for simplicity, and these two effects should counteract each other.

16-18E An average person produces 0.50 lbm of moisture while taking a shower. The contribution of showers of a family of four to the latent heat load of the air-conditioner per day is to be determined.



Assumptions All the water vapor from the shower is condensed by the air-conditioning system.

Properties The latent heat of vaporization of water is given to be 1050 Btu/lbm.

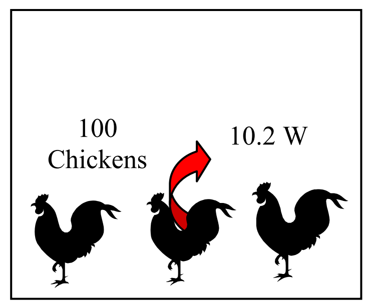
Analysis The amount of moisture produced per day is

$$\dot{m}_{\text{vapor}} = (\text{Moisture produced per person})(\text{No. of persons})$$
$$= (0.5\text{ lbm/person})(4\text{ persons/day}) = 2\text{ lbm/day}$$

Then the latent heat load due to showers becomes

$$\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg} = (2\text{ lbm/day})(1050\text{ Btu/lbm}) = \mathbf{2100\ Btu/day}$$

16-19 There are 100 chickens in a breeding room. The rate of total heat generation and the rate of moisture production in the room are to be determined.



Assumptions All the moisture from the chickens is condensed by the air-conditioning system.

Properties The latent heat of vaporization of water is given to be 2430 kJ/kg. The average metabolic rate of chicken during normal activity is 10.2 W (3.78 W sensible and 6.42 W latent).

Analysis The total rate of heat generation of the chickens in the breeding room is

$$\dot{Q}_{\text{gen, total}} = \dot{q}_{\text{gen, total}}(\text{No. of chickens})$$
$$= (10.2\text{ W/chicken})(100\text{ chickens}) = \mathbf{1020\ W}$$

The latent heat generated by the chicken and the rate of moisture production are

$$\dot{Q}_{\text{gen, latent}} = \dot{q}_{\text{gen, latent}}(\text{No. of chickens})$$
$$= (6.42\text{ W/chicken})(100\text{ chickens}) = 642\text{ W}$$
$$= 0.642\text{ kW}$$

$$\dot{m}_{\text{moisture}} = \frac{\dot{Q}_{\text{gen, latent}}}{h_{fg}} = \frac{0.642\text{ kJ/s}}{2430\text{ kJ/kg}} = 0.000264\text{ kg/s} = \mathbf{0.264\ g/s}$$

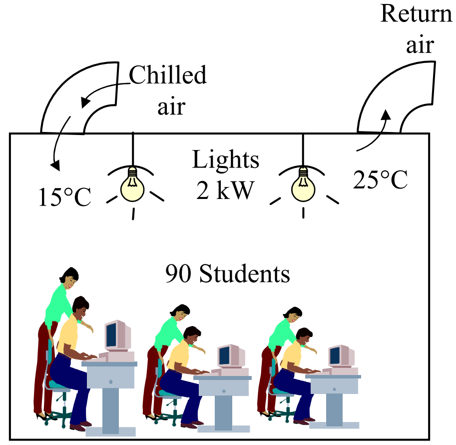
16-20 Chilled air is to cool a room by removing the heat generated in a large insulated classroom by lights and students. The required flow rate of air that needs to be supplied to the room is to be determined.

Assumptions **1** The moisture produced by the bodies leave the room as vapor without any condensing, and thus the classroom has no latent heat load. **2** Heat gain through the walls and the roof is negligible.

Properties The specific heat of air at room temperature is 1.00 kJ/kg·°C (Table A-15). The average rate of metabolic heat generation by a person sitting or doing light work is 115 W (70 W sensible, and 45 W latent).

Analysis The rate of sensible heat generation by the people in the room and the total rate of sensible internal heat generation are

$$\dot{Q}_{\text{gen, sensible}} = \dot{q}_{\text{gen, sensible}}(\text{No. of people})$$
$$= (70 \text{ W/person})(90 \text{ persons}) = 6300 \text{ W}$$
$$\dot{Q}_{\text{total, sensible}} = \dot{Q}_{\text{gen, sensible}} + \dot{Q}_{\text{lighting}}$$
$$= 6300 + 2000 = 8300 \text{ W}$$



Then the required mass flow rate of chilled air becomes

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{total, sensible}}}{c_p \Delta T}$$
$$= \frac{8.3 \text{ kJ/s}}{(1.0 \text{ kJ/kg}\cdot{}^\circ\text{C})(25-15)^\circ\text{C}} = \mathbf{0.83 \text{ kg/s}}$$

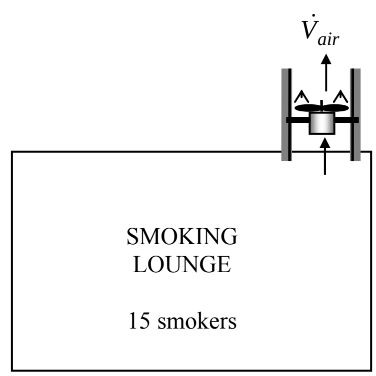
Discussion The latent heat will be removed by the air-conditioning system as the moisture condenses outside the cooling coils.

16-21 A smoking lounge that can accommodate 15 smokers is considered. The required minimum flow rate of air that needs to be supplied to the lounge is to be determined.

Assumptions Infiltration of air into the smoking lounge is negligible.

Properties The minimum fresh air requirements for a smoking lounge is 30 L/s per person (Table 16-2).

Analysis The required minimum flow rate of air that needs to be supplied to the lounge is determined directly from

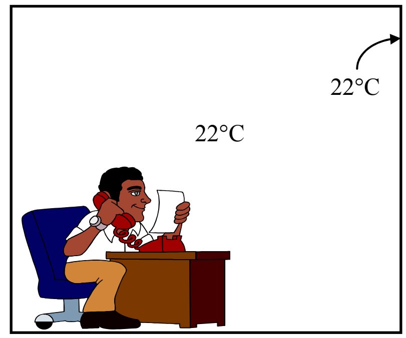


$$\dot{V}_{\text{air}} = \dot{V}_{\text{air per person}}(\text{No. of persons})$$
$$= (30 \text{ L/s}\cdot\text{person})(15 \text{ persons})$$
$$= 450 \text{ L/s} = \mathbf{0.45 \text{ m}^3\text{/s}}$$

16-22 The average mean radiation temperature during a cold day drops to 18°C. The required rise in the indoor air temperature to maintain the same level of comfort in the same clothing is to be determined.

Assumptions **1** Air motion in the room is negligible. **2** The average clothing and exposed skin temperature remains the same. **3** The latent heat loss from the body remains constant. **4** Heat transfer through the lungs remain constant.

Properties The emissivity of the person is 0.95 (Table A-15). The convection heat transfer coefficient from the body in still air or air moving with a velocity under 0.2 m/s is h_{conv} = 3.1 W/m^2·°C (Table 16-3).



Analysis The total rate of heat transfer from the body is the sum of the rates of heat loss by convection, radiation, and evaporation,

$$\dot{Q}_{\text{body, total}} = \dot{Q}_{\text{sensible}} + \dot{Q}_{\text{latent}} + \dot{Q}_{\text{lungs}}$$
$$= (\dot{Q}_{\text{conv}} + Q_{\text{rad}}) + \dot{Q}_{\text{latent}} + \dot{Q}_{\text{lungs}}$$

Noting that heat transfer from the skin by evaporation and from the lungs remains constant, the sum of the convection and radiation heat transfer from the person must remain constant.

$$\dot{Q}_{\text{sensible,old}} = hA(T_s - T_{\text{air, old}}) + \varepsilon A\sigma(T_s^4 - T_{\text{surr,old}}^4) = hA(T_s - 22) + 0.95A\sigma[(T_s + 273)^4 - (22 + 273)^4]$$
$$\dot{Q}_{\text{sensible,new}} = hA(T_s - T_{\text{air, new}}) + \varepsilon A\sigma(T_s^4 - T_{\text{surr,new}}^4) = hA(T_s - T_{\text{air, new}}) + 0.95A\sigma[(T_s + 273)^4 - (18 + 273)^4]$$

Setting the two relations above equal to each other, canceling the surface area *A*, and simplifying gives

$$-22h - 0.95\sigma(22 + 273)^4 = -hT_{\text{air, new}} - 0.95\sigma(18 + 273)^4$$
$$3.1(T_{\text{air, new}} - 22) + 0.95 \times 5.67 \times 10^{-8}(291^4 - 295^4) = 0$$

Solving for the new air temperature gives

$$T_{\text{air, new}} = \mathbf{29.0°C}$$

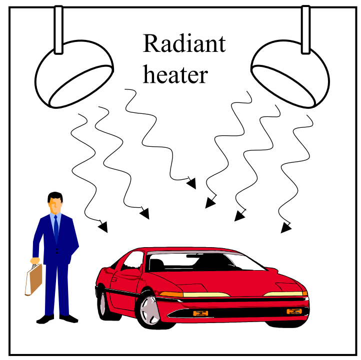
Therefore, the air temperature must be raised to 29°C to counteract the increase in heat transfer by radiation.

16-23 A car mechanic is working in a shop heated by radiant heaters in winter. The lowest ambient temperature the worker can work in comfortably is to be determined.

Assumptions **1** The air motion in the room is negligible, and the mechanic is standing. **2** The average clothing and exposed skin temperature of the mechanic is 33°C.

Properties The emissivity and absorptivity of the person is given to be 0.95. The convection heat transfer coefficient from a standing body in still air or air moving with a velocity under 0.2 m/s is h_{conv} = 4.0 W/m^2·°C (Table 13-3).

Analysis The equivalent thermal resistance of clothing is



$$R_{\text{cloth}} = 0.7\text{ clo} = 0.7 \times 0.155\text{ m}^2.°\text{C/W} = 0.1085\text{ m}^2.°\text{C/W}$$

Radiation from the heaters incident on the person and the rate of sensible heat generation by the person are

$$\dot{Q}_{\text{rad, incident}} = 0.05 \times \dot{Q}_{\text{rad, total}} = 0.05(4\text{ kW}) = 0.2\text{ kW} = 200\text{ W}$$

$$\dot{Q}_{\text{gen, sensible}} = 0.5 \times \dot{Q}_{\text{gen, total}} = 0.5(350\text{ W}) = 175\text{ W}$$

Under steady conditions, and energy balance on the body can be expressed as

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

$$\dot{Q}_{\text{rad from heater}} - \dot{Q}_{\text{conv+rad from body}} + \dot{Q}_{\text{gen, sensible}} = 0$$

or

$$\alpha\dot{Q}_{\text{rad, incident}} - h_{\text{conv}}A_s(T_s - T_{\text{surr}}) - \varepsilon A_s\sigma(T_s^4 - T_{\text{surr}}^4) + \dot{Q}_{\text{gen, sensible}} = 0$$

$$0.95(200\text{ W}) - (4.0\text{ W/m}^2\cdot\text{K})(1.8\text{ m}^2)(306 - T_{\text{surr}})$$
$$- 0.95(1.8\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)[(306\text{ K})^4 - T_{\text{surr}}^4) + 175\text{ W} = 0$$

Solving the equation above gives

$$T_{\text{surr}} = 284.8\text{ K} = \mathbf{11.8°C}$$

Therefore, the mechanic can work comfortably at temperatures as low as 12°C.

Design conditions for Heating and Cooling

16-24C The extreme outdoor temperature under which a heating or cooling system must be able to maintain a building at the indoor design conditions is called the *outdoor design temperature.* It differs from the average winter temperature in that the average temperature represents the arithmetic average of the hourly outdoors temperatures. The 97.5% winter design temperature ensures that the heating system will provide thermal comfort 97.5 percent of the time, but may fail to do so during 2.5 percent of the time. The 99% winter design temperature, on the other hand, ensures that the heating system will provide thermal comfort 99 percent of the time, but may fail to do so during 1 percent of the time in an average year.

16-25C Yes, it is possible for a city A to have a lower winter design temperature but a higher average winter temperature than another city B. In that case, a house in city A will require a larger heating system, but it will use less energy during a heating season.

16-26C The solar radiation has no effect on the design heating load in winter since the coldest outdoor temperatures occur before sunrise, but it may reduce the annual energy consumption for heating considerably. Similarly, the heat generated by people, lights, and appliances has no effect on the design heating load in winter since the heating system should be able to meet the heating load of a house even when there is no internal heat generation, but it will reduce the annual energy consumption for heating.

16-27C The solar radiation constitutes a major part of the cooling load, and thus it increases both the design cooling load in summer and the annual energy consumption for cooling. Similarly, the heat generated by people, lights, and appliances constitute a significant part of the cooling load, and thus it increases both the design cooling load in summer and the annual energy consumption for cooling.

16-28C The moisture level of the outdoor air contributes to the latent heat load, and it affects the cooling load in summer. This is because the humidity ratio of the outdoor air is higher than that of the indoor air in summer, and the outdoor air that infiltrates into the building increases the amount of moisture inside. This excess moisture must be removed by the air-conditioning system. The moisture level of the outdoor air, in general, does affect the heating load in winter since the humidity ratio of the outdoor air is much lower than that of the indoor air in winter, and the moisture production in the building is sufficient to keep the air moist. However, in some cases, it may be necessary to add moisture to the indoor air. The heating load in this case will increase because of the energy needed to vaporize the water.

16-29C The reason for different values of recommended design heat transfer coefficients for combined convection and radiation on the outer surface of a building in summer and in winter is the wind velocity. In winter, the wind velocity and thus the heat transfer coefficient is higher.

16-30C The *sol-air temperature* is defined as the equivalent outdoor air temperature that gives the same rate of heat flow to a surface as would the combination of incident solar radiation, convection with the ambient air, and radiation exchange with the sky and the surrounding surfaces. It is used to account for the effect of solar radiation by considering the outside temperature to be higher by an amount equivalent to the effect of solar radiation. The higher the solar absorptivity of the outer surface of a wall, the higher is the amount of solar radiation absorption and thus the sol-air temperature.

16-31C Most of the solar energy absorbed by the walls of a brick house will be transferred to the outdoors since the thermal resistance between the outer surface and the indoor air (the wall resistance + the convection resistance on the inner surface) is much larger than the thermal resistance between the outer surface and the outdoor air (just the convection resistance).

16-32 The climatic conditions for major cities in the U.S. are listed in Table 16-4, and for the indicated design levels we read

Winter: $T_{outdoor}$ = -19°C (97.5 percent level)

Summer: $T_{outdoor}$ = 35°C

$T_{wet\text{-}bulb}$ = 23°C (2.5 percent level)

Therefore, the heating and cooling systems in Lincoln, Nebraska for common applications should be sized for these outdoor conditions. Note that when the wet-bulb and ambient temperatures are available, the relative humidity and the humidity ratio of air can be determined from the psychrometric chart.

16-33 The climatic conditions for major cities in the U.S. are listed in Table 16-4, and for the indicated design levels we read

Winter: $T_{outdoor}$ = -16°C (99 percent level)

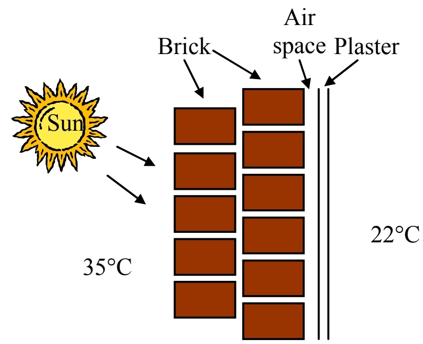
Summer: $T_{outdoor}$ = 37°C

$T_{wet\text{-}bulb}$ = 23°C (2.5 percent level)

Therefore, the heating and cooling systems in Wichita, Kansas for common applications should be sized for these outdoor conditions. Note that when the wet-bulb and ambient temperatures are available, the relative humidity and the humidity ratio of air can be determined from the psychrometric chart.

16-34 The south wall of a house is subjected to solar radiation at summer design conditions. The design heat gain, the fraction of heat gain due to solar heating, and the fraction of solar radiation that is transferred to the house are to be determined.

Assumptions **1** Steady conditions exist. **2** Thermal properties of the wall and the heat transfer coefficients are constant.



Properties The overall heat transfer coefficient of the wall is given to be 1.6 W/m^2·°C.

Analysis (*a*) The house is located at 40°N latitude, and thus we can use the sol-air temperature data directly from Table 16-7. At 15:00 the tabulated air temperature is 35°C, which is identical to the air temperature given in the problem. Therefore, the sol-air temperature on the south wall in this case is 43.6°C, and the heat gain through the wall is determined to be

$$\dot{Q}_{\text{wall}} = UA(T_{\text{sol-air}} - T_{\text{inside}}) = (1.6\ \text{W/m}^2\cdot{}^\circ\text{C})(20\ \text{m}^2)(43.6-22)^\circ\text{C} = \mathbf{691.2\ W}$$

(*b*) Heat transfer is proportional to the temperature difference, and the overall temperature difference in this case is 43.6 - 22 = 21.6°C. Also, the difference between the sol-air temperature and the ambient air temperature is

$$\Delta T_{\text{solar}} = T_{\text{sol-air}} - T_{\text{ambient}} = 43.6 - 35 = 8.6^\circ\text{C}$$

which is the equivalent temperature rise of the ambient air due to solar heating. The fraction of heat gain due to solar heating is equal to the fraction of the solar temperature difference to the overall temperature difference, and is determined to be

$$\text{Solar fraction} = \frac{\dot{Q}_{\text{wall,solar}}}{\dot{Q}_{\text{wall,total}}} = \frac{UA\Delta T_{\text{solar}}}{UA\Delta T_{\text{total}}} = \frac{\Delta T_{\text{solar}}}{\Delta T_{\text{total}}} = \frac{8.6^\circ\text{C}}{21.6^\circ\text{C}} = \mathbf{0.40}\ \text{(or 40\%)}$$

Therefore, almost half of the heat gain through the west wall in this case is due to solar heating of the wall.

(*c*) The outer layer of the wall is made of red brick which is dark colored. Therefore, the value of α_s / h_o is 0.052 m^2.°C/W. Then the fraction of incident solar energy transferred to the interior of the house is determined directly from Eq. 16-20 to be

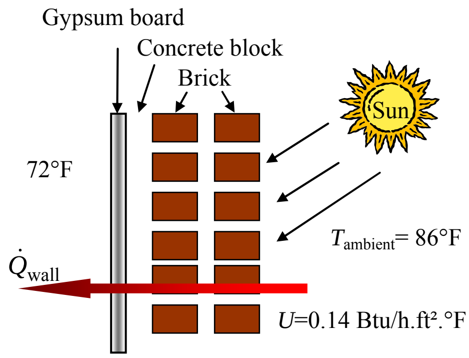
$$\text{Solar fraction tranferred} = U\frac{\alpha_s}{h_o} = (1.6\ \text{W/m}^2\cdot{}^\circ\text{C})(0.052\ \text{m}^2\cdot{}^\circ\text{C/W}) = \mathbf{0.0832}$$

Discussion Less than 10 percent of the solar energy incident on the surface will be transferred to the house in this case. Note that a glass wall would transmit about 10 times more energy into the house.

16-35E The west wall of a house is subjected to solar radiation at summer design conditions. The design heat gain and the fraction of heat gain due to solar heating are to be determined.

Assumptions **1** Steady conditions exist. **2** Thermal properties of the wall and the heat transfer coefficients are constant.

Properties The overall heat transfer coefficient of the wall is given to be 0.14 Btu/h·ft²·°F.



Analysis (*a*) The house is located at 40°N latitude, and thus we can use the sol-air temperature data directly from Table 16-7. At 16:00 the tabulated air temperature is 94°F, which is 8°F higher than the air temperature given in the problem. But we can still use the data in that table provided that we subtract 8 °F from all temperatures. Therefore, the sol-air temperature on the west wall in this case is 159 - 8 = 151°F, and the heat gain through the wall is determined to be

$$\dot{Q}_{\text{wall}} = UA(T_{\text{sol-air}} - T_{\text{inside}}) = (0.14\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})(150\times 12\ \text{ft}^2)(151-72)^\circ\text{F} = \mathbf{19{,}908\ Btu/h}$$

(*b*) Heat transfer is proportional to the temperature difference, and the overall temperature difference in this case is 151 - 72 = 79°F. Also, the difference between the sol-air temperature and the ambient air temperature is

$$\Delta T_{\text{solar}} = T_{\text{sol-air}} - T_{\text{ambient}} = 151 - 86 = 65^\circ\text{F}$$

which is the equivalent temperature rise of the ambient air due to solar heating. The fraction of heat gain due to solar heating is equal to the ratio of the solar temperature difference to the overall temperature difference, and is determined to be

$$\text{Solar fraction} = \frac{\dot{Q}_{\text{wall,solar}}}{\dot{Q}_{\text{wall,total}}} = \frac{UA\Delta T_{\text{solar}}}{UA\Delta T_{\text{total}}} = \frac{\Delta T_{\text{solar}}}{\Delta T_{\text{total}}} = \frac{65^\circ\text{F}}{79^\circ\text{F}} = \mathbf{0.823}\ (\text{or } 82.3\%)$$

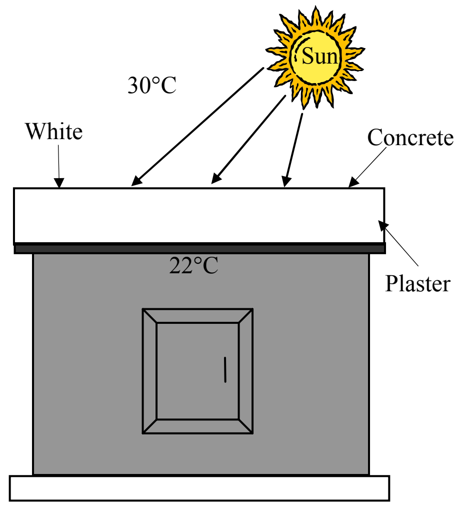
Therefore, almost the entire heat gain through the west wall in this case is due to solar heating of the wall.

16-36 The roof of a house is subjected to solar radiation at summer design conditions. The design heat gain and the fraction of heat gain due to solar heating are to be determined.

Assumptions **1** Steady conditions exist. **2** Thermal properties of the wall and the heat transfer coefficients are constant.

Properties The overall heat transfer coefficient of the roof is given to be 1.8 W/m^2·°C.

Analysis (*a*) The house is located at 40°N latitude, and thus we can use the sol-air temperature data directly from Table 16-7. At 16:00 the tabulated air temperature is 34.7°C, which is 4.7°C higher than the air temperature given in the problem. But we can still use the data in that table provided that we subtract 4.7°C from all temperatures. Therefore, the sol-air temperature on the roof in this case is 42.7 - 4.7 = 38.0°C, and the heat gain through the roof is determined to be



$$\dot{Q}_{\text{roof}} = UA(T_{\text{sol-air}} - T_{\text{inside}})$$
$$= (1.8\ \text{W/m}^2 \cdot {}^\circ\text{C})(150\ \text{m}^2)(38-22)^\circ\text{C} = \mathbf{4320\ W}$$

(*b*) Heat transfer is proportional to the temperature difference, and the overall temperature difference in this case is 38 - 22 = 16°C. Also, the difference between the sol-air temperature and the ambient air temperature is

$$\Delta T_{\text{solar}} = T_{\text{sol-air}} - T_{\text{ambient}} = 38 - 30 = 8^\circ\text{C}$$

which is the equivalent temperature rise of the ambient air due to solar heating. The fraction of heat gain due to solar heating is equal to the ratio of the solar temperature difference to the overall temperature difference, and is determined to be

$$\text{Solar fraction} = \frac{\dot{Q}_{\text{wall,solar}}}{\dot{Q}_{\text{wall,total}}} = \frac{UA\Delta T_{\text{solar}}}{UA\Delta T_{\text{total}}} = \frac{\Delta T_{\text{solar}}}{\Delta T_{\text{total}}} = \frac{8^\circ\text{C}}{16^\circ\text{C}} = \mathbf{0.50}\ \text{(or 50\%)}$$

Therefore, half of the heat gain through the roof in this case is due to solar heating of the roof.

Heat Gain from People, Lights, and Appliances

16-37C The heat given off by people in a concert hall is an important consideration in the sizing of the air-conditioning system for that building because of the high density of people in the hall, and the heat generation from the people contributes a significant amount to the cooling load.

16-38C By replacing the incandescent lamps of a building by high-efficiency fluorescent lamps, (a) the design cooling load will decrease, (*b*) annual energy consumption for cooling will also decrease, and (*c*) annual energy consumption for heating for the building will increase since fluorescent lamps generate much less heat than incandescent lamps for the same light output.

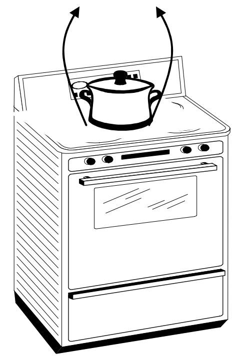
16-39C It is usually a good idea to replace incandescent light bulbs by compact fluorescent bulbs that may cost 40 times as much to purchase since incandescent lights waste energy by (1) consuming more electricity for the same amount of lighting, and (2) making the cooling system work harder and longer to remove the heat given off.

16-40C The motors and appliances in a building generate heat, and thus (*a*) they increase the design cooling load, (*b*) they increase the annual energy consumption for cooling, and (*c*) they reduce the annual energy consumption for heating of the building.

16-41C The motor efficiency η_{motor} is defined as the ratio of the shaft power delivered to the electrical power consumed by the motor. The higher the motor efficiency, the lower is the amount of heat generated by the motor. Therefore, high efficiency motors decrease the design cooling load of a building and the annual energy consumption for cooling.

16-42C The heat generated by a hooded range in a kitchen with a powerful fan that exhausts all the air heated and humidified by the range still needs to be considered in the determination of the cooling load of the kitchen although all the heated air is exhausted since part of the energy (32% of it) is radiated to the surroundings from the hot surfaces.

16-43 A hooded electric open burner and a gas burner are considered. The amount of the electrical energy used directly for cooking, the cost of energy per "utilized" kWh, and the contribution of this burner to the design cooling load are to be determined.



Analysis The efficiency of the electric heater is given to be 78 percent. Therefore, a burner that consumes 3-kW of electrical energy will supply

$$\dot{Q}_{\text{utilized}} = (\text{Energy input}) \times (\text{Efficiency}) = (3\text{ kW})(0.73) = \mathbf{2.19\ kW}$$

of useful energy. The unit cost of utilized energy is inversely proportional to the efficiency, and is determined from

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$0.09\ /\ \text{kWh}}{0.73} = \mathbf{\$0.123/kWh}$$

The design heat gain from a hooded appliance is taken to be 32% of the half of its rated energy consumption, and is determined to be

$$\dot{Q}_{\text{hooded appliance}} = 0.5 \times 0.32 \dot{Q}_{\text{appliance, input}} = 0.5 \times 0.32 \times (3\text{ kW}) = \mathbf{0.48\ kW} \quad \text{(electric burner)}$$

Noting that the efficiency of a gas burner is 38 percent, the energy input to a gas burner that supplies utilized energy at the same rate (2.19 kW) is

$$\dot{Q}_{\text{input, gas}} = \frac{\dot{Q}_{\text{utilized}}}{\text{Efficiency}} = \frac{2.19\text{ kW}}{0.38} = \mathbf{5.76\ kW} \ \ (= 19{,}660\text{ Btu/h})$$

since 1 kW = 3412 Btu/h. Therefore, a gas burner should have a rating of at least 19,660 Btu/h to perform as well as the electric unit.

Noting that 1 therm = 29.3 kWh, the unit cost of utilized energy in the case of gas burner is determined the same way to be

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$1.10\ /(29.3\text{ kWh})}{0.38} = \mathbf{\$0.099/kWh}$$

which is about 20 percent less than the unit cost of utilized electricity. The design heat gain from this hooded gas burner is determined similarly to be

$$\dot{Q}_{\text{unhooded appliance}} = 0.5 \times 0.32 \dot{Q}_{\text{appliance, input}} = 0.5 \times 0.32 \times (5.76\text{ kW}) = \mathbf{0.922\ kW} \quad \text{(gas burner)}$$

which is almost twice as much as that of the electric burner. Therefore, a hooded gas appliance will contribute more to the heat gain than a comparable electric appliance.

16-44 Several people are working out in an exercise room. The rate of heat gain from people and the equipment, and the fraction of that heat in the latent form are to be determined.

Analysis The 8 weight lifting machines do not have any motors, and thus they do not contribute to the internal heat gain directly. The usage factors of the motors of the treadmills are taken to be unity since they are used constantly during peak periods. Noting that 1 hp = 746 W, the total heat generated by the motors is

$$\dot{Q}_{\text{motors}} = (\text{No. of motors}) \times \dot{W}_{\text{motor}} \times f_{\text{load}} \times f_{\text{usage}} / \eta_{\text{motor}} = 4 \times (2.5 \times 746 \text{ W}) \times 0.70 \times 1.0 / 0.77 = 6782 \text{ W}$$

The average rate of heat dissipated by people in an exercise room is given in Table 16-8 to be 525 W, of which 315 W is in latent form. Therefore, the heat gain from 14 people is

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 14 \times (525 \text{ W}) = 7350 \text{ W}$$

Then the total rate of heat gain (or the internal heat load) of the exercise room during peak period becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{motors}} + \dot{Q}_{\text{people}} = 6782 + 7350 = \mathbf{14{,}132\ W}$$

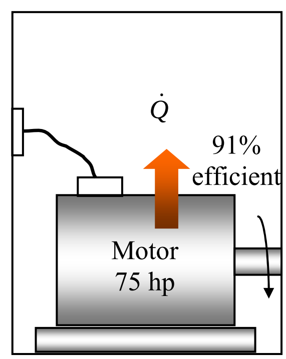
The entire heat given off by the motors is in sensible form. Therefore, the latent heat gain is due to people only, which is determined to be

$$\dot{Q}_{\text{latent}} = (\text{No. of people}) \times \dot{Q}_{\text{latent, per person}} = 14 \times (315 \text{ W}) = \mathbf{4410\ W}$$

The remaining 14,132 - 4410 = 9722 W of heat gain is in the sensible form.

16-45 A worn out standard motor is replaced by a high efficiency one. The reduction in the internal heat gain due to higher efficiency under full load conditions is to be determined.

Assumptions **1** The motor and the equipment driven by the motor are in the same room. **2** The motor operates at full load so that $f_{\text{load}} = 1$.



Analysis The heat generated by a motor is due to its inefficiency, and the difference between the heat generated by two motors that deliver the same shaft power is simply the difference between the electric power drawn by the motors,

$$\dot{W}_{\text{in, electric, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746 \text{ W})/0.91 = 61{,}484 \text{ W}$$

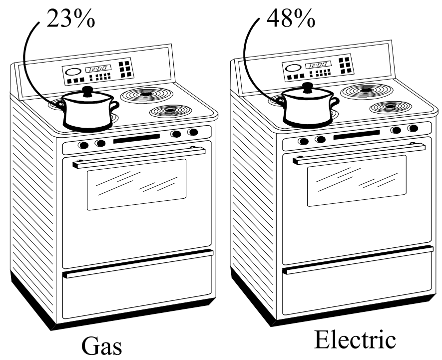
$$\dot{W}_{\text{in, electric, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746 \text{ W})/0.954 = 58{,}648 \text{ W}$$

Then the reduction in heat generation becomes

$$\dot{Q}_{\text{reduction}} = \dot{W}_{\text{in, electric, standard}} - \dot{W}_{\text{in, electric, efficient}} = 61{,}484 - 58{,}648 = \mathbf{2836\ W}$$

16-46 An electric hot plate and a gas hot plate are considered. For the same amount of "utilized" energy, the ratio of internal heat generated by gas hot plates to that by electric ones is to be determined.

Assumptions Hot plates are not hooded and thus the entire energy they consume is dissipated to the room they are in.



Analysis The utilized energy can be expressed in terms of the energy input and the efficiency as

$$\dot{E}_{\text{utilized}} = (\text{Energy input}) \times (\text{Efficiency})$$
$$= \dot{E}_{\text{in}} \times \eta \quad \rightarrow \quad \dot{E}_{\text{in}} = \dot{E}_{\text{utilized}} / \eta$$

Noting that the utilized energy is the same for both the electric and gas hot plates, the ratio of internal heat generated by gas hot plates to that by electric ones is determined to be

$$\text{Ratio of heat generation} = \frac{\dot{E}_{\text{in, gas}}}{\dot{E}_{\text{in, electric}}} = \frac{\dot{E}_{\text{utilized}} / \eta_{\text{gas}}}{\dot{E}_{\text{utilized}} / \eta_{\text{electric}}} = \frac{\eta_{\text{electric}}}{\eta_{\text{gas}}} = \frac{0.48}{0.23} = \mathbf{2.07}$$

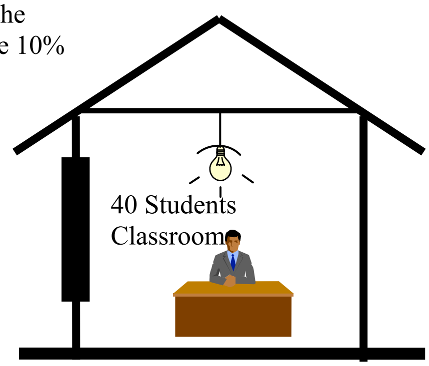
Therefore, the gas hot plate will contribute twice as much to the internal heat gain of the room.

16-47 A classroom has 40 students, one instructor, and 18 fluorescent light bulbs. The rate of internal heat generation in this classroom is to be determined.

Assumptions **1** There is a mix of men, women, and children in the classroom. **2** The amount of light (and thus energy) leaving the room through the windows is negligible.

Properties The average rate of heat generation from people seated in a room/office is 115 W (Table 16-8).

Analysis The amount of heat dissipated by the lamps is equal to the amount of electrical energy consumed by the lamps, including the 10% additional electricity consumed by the ballasts. Therefore,



$$\dot{Q}_{\text{lighting}} = (\text{Energy consumed per lamp}) \times (\text{No. of lamps})$$
$$= (40\text{ W})(1.1)(18) = 792\text{ W}$$
$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 41 \times (115\text{ W}) = 4715\text{ W}$$

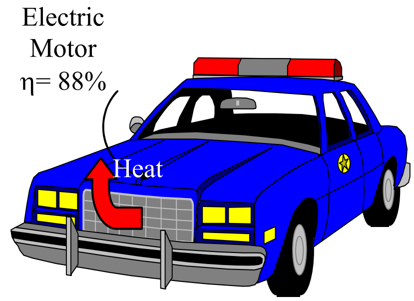
Then the total rate of heat gain (or the internal heat load) of the classroom from the lights and people become

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{lighting}} + \dot{Q}_{\text{people}} = 792 + 4715 = \mathbf{5507\ W}$$

16-48 An electric car is powered by an electric motor mounted in the engine compartment. The rate of heat supply by the motor to the engine compartment at full load conditions is to be determined.

Assumptions The motor operates at full load so that $f_{load} = 1$.

Analysis The heat generated by a motor is due to its inefficiency, and the heat generated by a motor is equal to the difference between the electrical energy it consumes and the shaft power it delivers,



$$\dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (60\text{ hp})/0.88 = 68.18\text{ hp}$$

$$\dot{Q}_{\text{generation}} = \dot{W}_{\text{in, electric}} - \dot{W}_{\text{shaft out}}$$

$$= 68.18 - 60 = 8.18\text{ hp} = \mathbf{6.10\ kW}$$

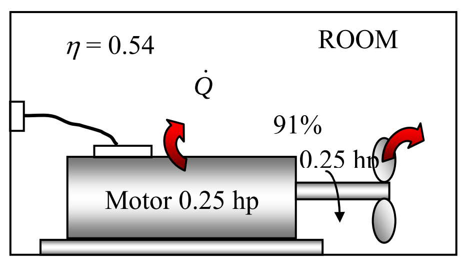
since 1 hp = 0.746 kW.

Discussion The motor will supply as much heat to the compartment as a 6.1 kW resistance heater.

16-49 A room is cooled by circulating chilled water, and the air is circulated through the heat exchanger by a fan. The contribution of the fan-motor assembly to the cooling load of the room is to be determined.

Assumptions The fan motor operates at full load so that $f_{load} = 1$.

Analysis The entire electrical energy consumed by the motor, including the shaft power delivered to the fan, is eventually dissipated as heat. Therefore, the contribution of the fan-motor assembly to the cooling load of the room is equal to the electrical energy it consumes,



$$\dot{Q}_{\text{internal generation}} = \dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}}$$

$$= (0.25\text{ hp})/0.54 = 0.463\text{ hp} = \mathbf{345\ W}$$

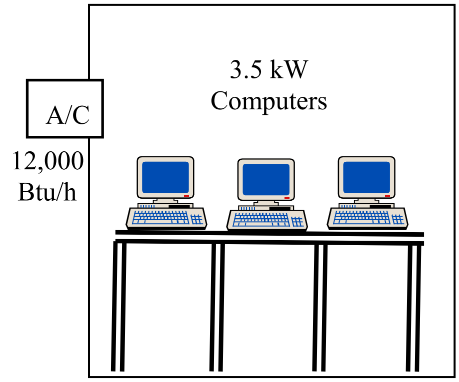
since 1 hp = 746 W.

16-50 An office that is being cooled adequately by a 12,000 Btu/h window air-conditioner is converted to a computer room. The number of additional air-conditioners that need to be installed is to be determined.

Assumptions **1** The computer are operated by 4 adult men. **2** The computers consume 40 percent of their rated power at any given time.

Properties The average rate of heat generation from a man seated in a room/office is 130 W (Table 16-8).

Analysis The amount of heat dissipated by the computers is equal to the amount of electrical energy they consume. Therefore,



$$\begin{aligned}\dot{Q}_{\text{computers}} &= (\text{Rated power})\times(\text{Usage factor})\\ &= (3.5\text{ kW})(0.4) = 1.4\text{ kW}\\ \dot{Q}_{\text{people}} &= (\text{No. of people})\times\dot{Q}_{\text{person}}\\ &= 4\times(130\text{ W}) = 520\text{ W}\\ \dot{Q}_{\text{total}} &= \dot{Q}_{\text{computers}} + \dot{Q}_{\text{people}}\\ &= 1400 + 520 = 1920\text{ W} = 6551\text{ Btu/h}\end{aligned}$$

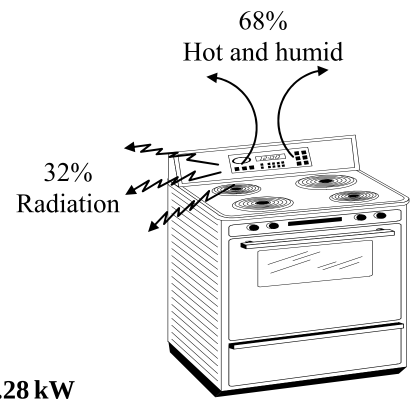
since 1 W = 3.412 Btu/h. Then noting that each available air conditioner provides 4,000 Btu/h cooling, the number of air-conditioners needed becomes

$$\text{No. of air conditioners} = \frac{\text{Cooling load}}{\text{Cooling capacity of A/C}} = \frac{6551\text{ Btu/h}}{4000\text{ Btu/h}} = 1.6 \approx \mathbf{2\ Air\ conditioners}$$

16-51 A restaurant purchases a new 8-kW electric range for its kitchen. The increase in the design cooling load is to be determined for the cases of hooded and unhooded range.

Assumptions **1** The contribution of an appliance to the design cooling is half of its rated power. **2** The hood of an appliance removes all the heated air and moisture generated, except the radiation heat that constitutes 38 percent of the heat generated.

Analysis The design cooling load due to an unhooded range is half of its rated power, and the design cooling of a hooded range is half of the radiation component of heat dissipation, which is taken to be 38 percent. Then the increase in the design cooling load for both cases becomes



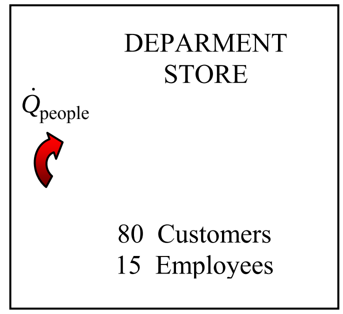
$$\dot{Q}_{\text{unhooded appliance}} = 0.5\times\dot{Q}_{\text{appliance, input}} = 0.5\times(8\text{ kW}) = \mathbf{4\ kW}$$

$$\dot{Q}_{\text{hooded appliance}} = 0.5\times 0.32\dot{Q}_{\text{appliance, input}} = 0.5\times 0.32\times(8\text{ kW}) = \mathbf{1.28\ kW}$$

16-52 A department store expects to have 95 people at peak times in summer. The contribution of people to the sensible, latent, and total cooling load of the store is to be determined.

Assumptions There is a mix of men, women, and children in the classroom.

Properties The average rate of heat generation from people in a shopping center is 130 W, and 75 W of it is in sensible form and 55 W in latent form (Table 16-8).



Analysis The contribution of people to the sensible, latent, and total cooling load of the store are

$$\dot{Q}_{\text{people, total}} = (\text{No. of people}) \times \dot{Q}_{\text{person, total}} = 95 \times (130\text{ W}) = \mathbf{12{,}350\ W}$$

$$\dot{Q}_{\text{people, sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person, sensible}} = 95 \times (70\text{ W}) = \mathbf{7125\ W}$$

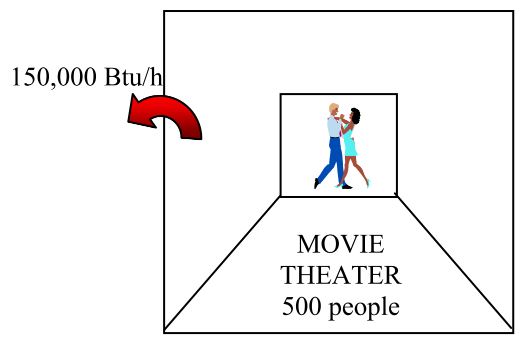
$$\dot{Q}_{\text{people, latent}} = (\text{No. of people}) \times \dot{Q}_{\text{person, latent}} = 95 \times (35\text{ W}) = \mathbf{5225\ W}$$

16-53E There are 500 people in a movie theater in winter. It is to be determined if the theater needs to be heated or cooled.

Assumptions There is a mix of men, women, and children in the classroom.

Properties The average rate of heat generation from people in a movie theater is 105 W, and 70 W of it is in sensible form and 35 W in latent form (Table 16-8).

Analysis Noting that only the sensible heat from a person contributes to the heating load of a building, the contribution of people to the heating of the building is



$$\dot{Q}_{\text{people, sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person, sensible}}$$
$$= 500 \times (70\text{ W}) = 35{,}000\text{ W}$$
$$= \mathbf{119{,}420\ Btu/h}$$

since 1 W = 3.412 Btu/h. The building needs to be heated since the heat gain from people is less than the rate of heat loss of 150,000 Btu/h from the building.

Heat Transfer through the Walls and Roofs

16-54C The *R*-value of a wall is the thermal resistance of the wall per unit surface area. It is the same as the unit thermal resistance of the wall. It is the inverse of the *U*-factor of the wall, $R = 1/U$.

16-55C The effective emissivity for a plane-parallel air space is the "equivalent" emissivity of one surface for use in the relation $\dot{Q}_{\text{rad}} = \varepsilon_{\text{effective}}\sigma A(T_2^4 - T_1^4)$ that results in the same rate of radiation heat transfer between the two surfaces across the air space. It is determined from

$$\frac{1}{\varepsilon_{\text{effective}}} = \frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1$$

where ε_1 and ε_2 are the emissivities of the surfaces of the air space. When the effective emissivity is known, the radiation heat transfer through the air space is determined from the $\dot{Q}_{\text{rad}}$ relation above.

16-56C The unit thermal resistances (*R*-value) of both 40-mm and 90-mm vertical air spaces are given to be the same, which implies that more than doubling the thickness of air space in a wall has no effect on heat transfer through the wall. This is not surprising since the convection currents that set in in the thicker air space offset any additional resistance due to a thicker air space.

16-57C Radiant barriers are highly reflective materials that minimize the radiation heat transfer between surfaces. Highly reflective materials such as aluminum foil or aluminum coated paper are suitable for use as radiant barriers. Yes, it is worthwhile to use radiant barriers in the attics of homes by covering at least one side of the attic (the roof or the ceiling side) since they reduce radiation heat transfer between the ceiling and the roof considerably.

16-58C The roof of a house whose attic space is ventilated effectively so that the air temperature in the attic is the same as the ambient air temperature at all times will still have an effect on heat transfer through the ceiling since the roof in this case will act as a radiation shield, and reduce heat transfer by radiation.

16-59 The *R*-value and the *U*-factor of a wood frame wall are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

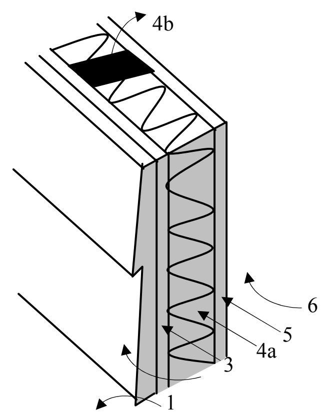
Properties The R-values of different materials are given in Table 16-10.

Analysis The schematic of the wall as well as the different elements used in its construction are shown below. Heat transfer through the insulation and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Once the unit thermal resistances and the *U*-factors for the insulation and stud sections are available, the overall average thermal resistance for the entire wall can be determined from

$$R_{\text{overall}} = 1/U_{\text{overall}} \quad \text{where} \quad U_{\text{overall}} = (Uf_{\text{area}})_{\text{insulation}} + (Uf_{\text{area}})_{\text{stud}}$$

and the value of the area fraction f_{area} is 0.80 for insulation section and 0.20 for stud section since the headers that constitute a small part of the wall are to be treated as studs. Using the available *R*-values from Table 16-10 and calculating others, the total *R*-values for each section is determined in the table below.

Construction	*R*-value, m^2.°C/W Between studs	*R*-value, m^2.°C/W At studs
1. Outside surface, 12 km/h wind	0.044	0.044
2. Wood bevel lapped siding	0.14	0.14
3. Fiberboard sheathing, 13 mm	0.23	0.23
4a. Mineral fiber insulation, 140 mm	3.696	--
4b. Wood stud, 38 mm by 140 mm	--	0.98
5. Gypsum wallboard, 13 mm	0.079	0.079
6. Inside surface, still air	0.12	0.12



Total unit thermal resistance of each section, *R* (in m^2.°C/W)	4.309	1.593
The *U*-factor of each section, $U = 1/R$, in W/m^2.°C	0.232	0.628
Area fraction of each section, f_{area}	0.80	0.20
Overall *U*-factor, $U = \Sigma f_{\text{area},i} U_i = 0.80 \times 0.232 + 0.20 \times 0.628$	**0.311 W/m^2.°C**	
Overall unit thermal resistance, $R = 1/U$	**3.213 m^2.°C/W**	

Therefore, the *R*-value and *U*-factor of the wall are $R = 3.213$ m^2.°C/W and $U = 0.311$ W/m^2.°C.

16-60 The change in the *R*-value of a wood frame wall due to replacing fiberwood sheathing in the wall by rigid foam sheathing is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

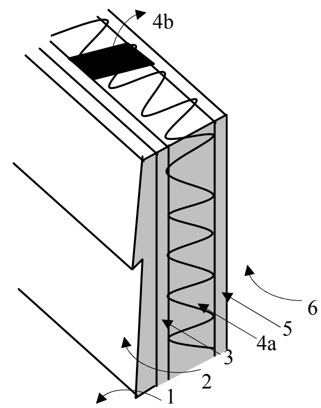
Properties The R-values of different materials are given in Table 16-10.

Analysis The schematic of the wall as well as the different elements used in its construction are shown below. Heat transfer through the insulation and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Once the unit thermal resistances and the *U*-factors for the insulation and stud sections are available, the overall average thermal resistance for the entire wall can be determined from

$$R_{\text{overall}} = 1/U_{\text{overall}} \quad \text{where} \quad U_{\text{overall}} = (Uf_{\text{area}})_{\text{insulation}} + (Uf_{\text{area}})_{\text{stud}}$$

and the value of the area fraction f_{area} is 0.80 for insulation section and 0.20 for stud section since the headers that constitute a small part of the wall are to be treated as studs. Using the available *R*-values from Table 16-10 and calculating others, the total *R*-values for each section of the existing wall is determined in the table below.

Construction	*R* -value, $m^2.°C/W$	
	Between studs	At studs
1. Outside surface, 12 km/h wind	0.044	0.044
2. Wood bevel lapped siding	0.14	0.14
3. Rigid foam, 25 mm	0.98	0.98
4a. Mineral fiber insulation, 140 mm	3.696	--
4b. Wood stud, 38 mm by 140 mm	--	0.98
5. Gypsum wallboard, 13 mm	0.079	0.079
6. Inside surface, still air	0.12	0.12



Total unit thermal resistance of each section, *R* (in $m^2.°C/W$)	5.059	2.343
The *U*-factor of each section, $U = 1/R$, in $W/m^2.°C$	0.198	0.426
Area fraction of each section, f_{area}	0.80	0.20
Overall *U*-factor, $U = \Sigma f_{\text{area,i}} U_{\text{i}} = 0.80 \times 0.232 + 0.20 \times 0.628$	$0.2436\ W/m^2.°C$	
Overall unit thermal resistance, $R = 1/U$	$4.105\ m^2.°C/W$	

The *R*-value of the existing wall is $R = 3.213\ m^2.°C/W$. Then the change in the *R*-value becomes

$$\%\ \text{Change} = \frac{\Delta R - \text{value}}{R - \text{value, old}} = \frac{4.105 - 3.213}{4.105} = 0.217 \quad (\text{or } \mathbf{21.7\%})$$

16-61E The *R*-value and the *U*-factor of a masonry cavity wall are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

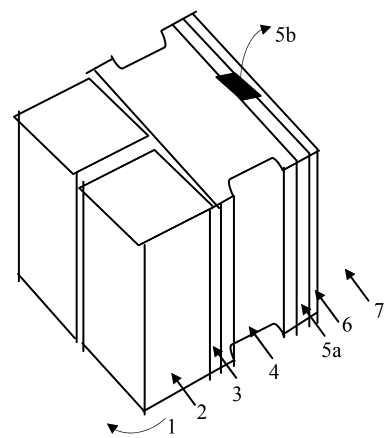
Properties The R-values of different materials are given in Table 16-10.

Analysis The schematic of the wall as well as the different elements used in its construction are shown below. Heat transfer through the air space and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Once the unit thermal resistances and the *U*-factors for the air space and stud sections are available, the overall average thermal resistance for the entire wall can be determined from

$$R_{\text{overall}} = 1/U_{\text{overall}} \quad \text{where} \quad U_{\text{overall}} = (Uf_{\text{area}})_{\text{air space}} + (Uf_{\text{area}})_{\text{stud}}$$

and the value of the area fraction f_{area} is 0.80 for air space and 0.20 for the ferrings and similar structures. Using the available *R*-values from Table 16-10 and calculating others, the total *R*-values for each section of the existing wall is determined in the table below.

Construction	*R*-value, h.ft^2.°F/Btu	
	Between furring	At furring
1. Outside surface, 15 mph wind	0.17	0.17
2. Face brick, 4 in	0.43	0.43
3. Cement mortar, 0.5 in	0.10	0.10
4. Concrete block, 4-in	1.51	1.51
5a. Air space, 3/4-in, nonreflective	2.91	--
5b. Nominal 1 × 3 vertical furring	--	0.94
6. Gypsum wallboard, 0.5 in	0.45	0.45
7. Inside surface, still air	0.68	0.68



Total unit thermal resistance of each section, *R*	6.25	4.28
The *U*-factor of each section, $U = 1/R$, in Btu/h.ft^2.°F	0.160	0.234
Area fraction of each section, f_{area}	0.80	0.20
Overall *U*-factor, $U = \Sigma f_{\text{area,i}} U_{\text{i}} = 0.80\times0.160+0.20\times0.234$	**0.175 Btu/h.ft^2.°F**	
Overall unit thermal resistance, $R = 1/U$	**5.72 h.ft^2.°F/Btu**	

Therefore, the overall unit thermal resistance of the wall is R = 5.72 h.ft^2.°F/Btu and the overall U-factor is U = 0.175 Btu/h.ft^2.°F. These values account for the effects of the vertical ferring.

16-62 The winter *R*-value and the *U*-factor of a flat ceiling with an air space are to be determined for the cases of air space with reflective and nonreflective surfaces.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the ceiling is one-dimensional. **3** Thermal properties of the ceiling and the heat transfer coefficients are constant.

Properties The R-values of different materials are given in Table 16-10. The R-values of different air layers are given in Table 16-13.

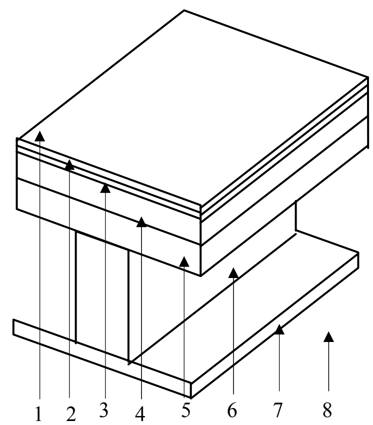
Analysis The schematic of the ceiling as well as the different elements used in its construction are shown below. Heat transfer through the air space and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Once the unit thermal resistances and the *U*-factors for the air space and stud sections are available, the overall average thermal resistance for the entire wall can be determined from

$$R_{overall} = 1/U_{overall} \quad \text{where} \quad U_{overall} = (Uf_{area})_{air\ space} + (Uf_{area})_{stud}$$

and the value of the area fraction f_{area} is 0.82 for air space and 0.18 for stud section since the headers which constitute a small part of the wall are to be treated as studs.

(a) Nonreflective surfaces, $\varepsilon_1 = \varepsilon_2 = 0.9$ and thus $\varepsilon_{effective} = \dfrac{1}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} = \dfrac{1}{1/0.9 + 1/0.9 - 1} = 0.82$

Construction	*R* -value, m^2.°C/W Between studs	At studs
1. Still air above ceiling	0.12	0.044
2. Linoleum (R = 0.009 m^2.°C/W)	0.009	0.14
3. Felt (R = 0.011 m^2.°C/W)	0.011	0.23
4. Plywood, 13 mm	0.11	
5. Wood subfloor (R = 0.166 m^2.°C/W)	0.166	
6a. Air space, 90 mm, nonreflective	0.16	---
6b. Wood stud, 38 mm by 90 mm	---	0.63
7. Gypsum wallboard, 13 mm	0.079	0.079
8. Still air below ceiling	0.12	0.12



Total unit thermal resistance of each section, R (in m^2.°C/W)	0.775	1.243
The U-factor of each section, $U = 1/R$, in W/m^2.°C	1.290	0.805
Area fraction of each section, f_{area}	0.82	0.18
Overall U-factor, $U = \Sigma f_{area,i} U_i$ = 0.82×1.290+0.18×0.805	**1.203 W/m^2.°C**	
Overall unit thermal resistance, $R = 1/U$	**0.831 m^2.°C/W**	

(*b*) One-reflective surface, $\varepsilon_1 = 0.05$ and $\varepsilon_2 = 0.9 \rightarrow \varepsilon_{effective} = \dfrac{1}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} = \dfrac{1}{1/0.05 + 1/0.9 - 1} = 0.05$

In this case we replace item 6a from 0.16 to 0.47 m^2.°C/W. It gives R = 1.085 m^2.°C/W and U = 0.922 W/ m^2.°C for the air space. Then,

Overall U-factor, $U = \Sigma f_{area,i} U_i$ = 0.82×1.085+0.18×0.805	**1.035 W/m^2.°C**
Overall unit thermal resistance, $R = 1/U$	**0.967 m^2.°C/W**

(*c*) Two-reflective surface, $\varepsilon_1 = \varepsilon_2 = 0.05 \rightarrow \varepsilon_{effective} = \dfrac{1}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} = \dfrac{1}{1/0.05 + 1/0.05 - 1} = 0.03$

In this case we replace item 6a from 0.16 to 0.49 m^2.°C/W. It gives R = 1.105 m^2.°C/W and U = 0.905 W/ m^2.°C for the air space. Then,

Overall U-factor, $U = \Sigma f_{area,i} U_i$ = 0.82×1.105+0.18×0.805	**1.051 W/m^2.°C**
Overall unit thermal resistance, $R = 1/U$	**0.951 m^2.°C/W**

16-63 The winter *R*-value and the *U*-factor of a masonry cavity wall are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

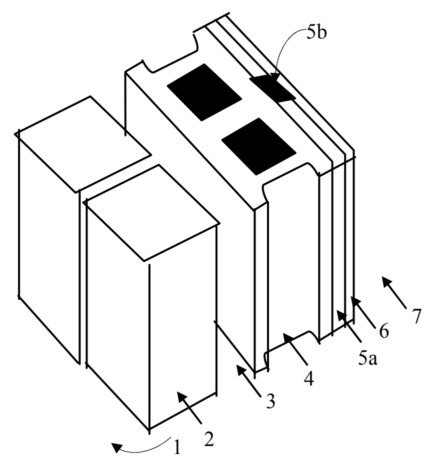
Properties The R-values of different materials are given in Table 16-10.

Analysis The schematic of the wall as well as the different elements used in its construction are shown below. Heat transfer through the air space and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Once the unit thermal resistances and the *U*-factors for the air space and stud sections are available, the overall average thermal resistance for the entire wall can be determined from

$$R_{overall} = 1/U_{overall} \quad \text{where} \quad U_{overall} = (Uf_{area})_{air\ space} + (Uf_{area})_{stud}$$

and the value of the area fraction f_{area} is 0.84 for air space and 0.16 for the ferrings and similar structures. Using the available *R*-values from Tables 16-10 and 16-13 and calculating others, the total *R*-values for each section of the existing wall is determined in the table below.

Construction	*R*-value, $m^2.°C/W$	
	Between furring	At furring
1. Outside surface, 24 km/h	0.030	0.030
2. Face brick, 100 mm	0.12	0.12
3. Air space, 90-mm, nonreflective	0.16	0.16
4. Concrete block, lightweight, 100-mm	0.27	0.27
5a. Air space, 20 mm, nonreflective	0.17	---
5b. Vertical ferring, 20 mm thick	---	0.94
6. Gypsum wallboard, 13	0.079	0.079
7. Inside surface, still air	0.12	0.12



Total unit thermal resistance of each section, *R*	0.949	1.719
The *U*-factor of each section, $U = 1/R$, in $W/m^2.°C$	1.054	0.582
Area fraction of each section, f_{area}	0.84	0.16
Overall *U*-factor, $U = \Sigma f_{area,i} U_i = 0.84 \times 1.054 + 0.16 \times 0.582$	**0.978 W/m².°C**	
Overall unit thermal resistance, $R = 1/U$	**1.02 m².°C/W**	

Therefore, the overall unit thermal resistance of the wall is $R = 1.02\ m^2.°C/W$ and the overall U-factor is $U = 0.978\ W/m^2.°C$. These values account for the effects of the vertical ferring.

16-64 The winter *R*-value and the *U*-factor of a masonry cavity wall with a reflective surface are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

Properties The R-values of different materials are given in Table 16-10. The *R*-values of air spaces are given in Table 16-13.

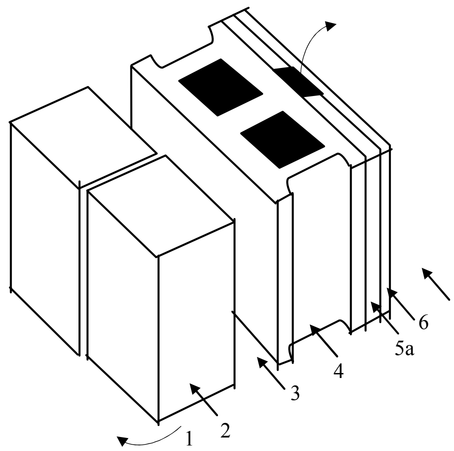
Analysis The schematic of the wall as well as the different elements used in its construction are shown below. Heat transfer through the air space and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Once the unit thermal resistances and the *U*-factors for the air space and stud sections are available, the overall average thermal resistance for the entire wall can be determined from

$R_{overall} = 1/U_{overall}$

where $U_{overall} = (Uf_{area})_{air\ space} + (Uf_{area})_{stud}$

and the value of the area fraction f_{area} is 0.84 for air space and 0.16 for the ferrings and similar structures. For an air space with one-reflective surface, we have $\varepsilon_1 = 0.05$ and $\varepsilon_2 = 0.9$, and thus

$$\varepsilon_{effective} = \frac{1}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} = \frac{1}{1/0.05 + 1/0.9 - 1} = 0.05$$



Using the available *R*-values from Tables 16-10 and 16-13 and calculating others, the total *R*-values for each section of the existing wall is determined in the table below.

Construction	*R*-value, m^2.°C/W Between furring	At furring
1. Outside surface, 24 km/h	0.030	0.030
2. Face brick, 100 mm	0.12	0.12
3. Air space, 90-mm, reflective with ε = 0.05	0.45	0.45
4. Concrete block, lightweight, 100-mm	0.27	0.27
5a. Air space, 20 mm, reflective with ε =0.05	0.49	---
5b. Vertical ferring, 20 mm thick	---	0.94
6. Gypsum wallboard, 13	0.079	0.079
7. Inside surface, still air	0.12	0.12

Total unit thermal resistance of each section, *R*	1.559	2.009
The *U*-factor of each section, *U* = 1/*R*, in W/m^2.°C	0.641	0.498
Area fraction of each section, f_{area}	0.84	0.16
Overall *U*-factor, $U = \Sigma f_{area,i} U_i = 0.84\times1.05+0.16\times0.582$	**0.618 W/m^2.°C**	
Overall unit thermal resistance, *R* = 1/*U*	**1.62 m^2.°C/W**	

Therefore, the overall unit thermal resistance of the wall is *R* = 1.62 m^2.°C/W and the overall U-factor is *U* = 0.618 W/m^2.°C. These values account for the effects of the vertical ferring.

Discussion The change in the *U*-value as a result of adding reflective surfaces is

$$\text{Change} = \frac{\Delta U - \text{value}}{U - \text{value, nonreflective}} = \frac{0.978 - 0.618}{0.978} = 0.368$$

Therefore, the rate of heat transfer through the wall will decrease by 36.8% as a result of adding a reflective surface.

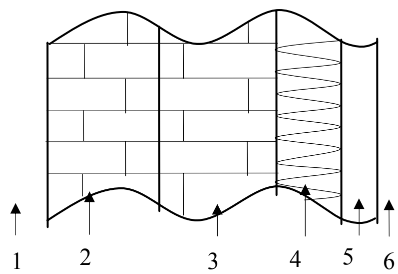
16-65 The winter *R*-value and the *U*-factor of a masonry wall are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

Properties The R-values of different materials are given in Table 16-10.

Analysis Using the available *R*-values from Tables 16-10, the total *R*-value of the wall is determined in the table below.

Construction	*R*-value, $m^2.°C/W$
1. Outside surface, 24 km/h	0.030
2. Face brick, 100 mm	0.075
3. Common brick, 100 mm	0.12
4. Urethane foam insulation, 25-mm	0.98
5. Gypsum wallboard, 13 mm	0.079
6. Inside surface, still air	0.12



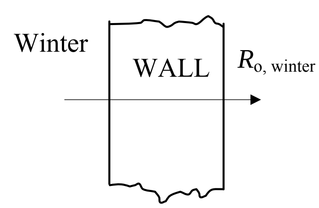
Total unit thermal resistance of each section, *R*	**1.404 m²·°C/W**
The *U*-factor of each section, *U* = 1/*R*	**0.712 W/m²·°C**

Therefore, the overall unit thermal resistance of the wall is $R = 1.404\ m^2.°C/W$ and the overall U-factor is $U = 0.712\ W/m^2.°C$.

16-66 The *U*-value of a wall under winter design conditions is given. The *U*-value of the wall under summer design conditions is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant except the one at the outer surface.

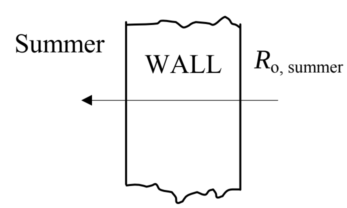
Properties The R-values at the outer surface of a wall for summer (12 km/h winds) and winter (24 km/h winds) conditions are given in Table 3-6 to be $R_{o,\ summer} = 0.044\ m^2.°C/W$ and $R_{o,\ winter} = 0.030\ m^2.°C/W$.



Analysis The *R*-value of the existing wall is

$$R_{winter} = 1/U_{winter} = 1/1.40 = 0.714\ m^2 \cdot °C/W$$

Noting that the added and removed thermal resistances are in series, the overall *R*-value of the wall under summer conditions becomes



$$\begin{aligned} R_{summer} &= R_{winter} - R_{o,winter} + R_{o,summer} \\ &= 0.714 - 0.030 + 0.044 \\ &= 0.728\ m^2 \cdot °C/W \end{aligned}$$

Then the summer *U*-value of the wall becomes

$$R_{summer} = 1/U_{summer} = 1/0.728 = \mathbf{1.37\ m^2 \cdot °C/W}$$

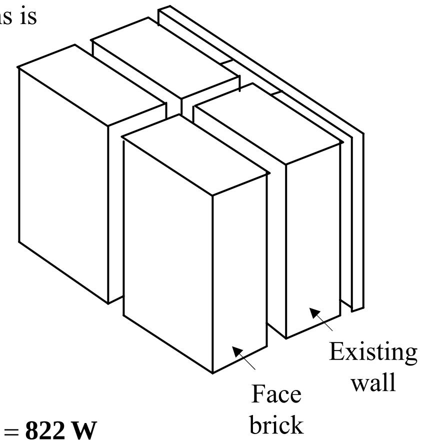
16-67 The *U*-value of a wall is given. A layer of face brick is added to the outside of a wall, leaving a 20-mm air space between the wall and the bricks. The new *U*-value of the wall and the rate of heat transfer through the wall is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

Properties The *U*-value of a wall is given to be $U = 2.25$ W/m^2.°C. The *R* - values of 100-mm face brick and a 20-mm air space between the wall and the bricks various layers are 0.075 and 0.170 m^2.°C/W, respectively.

Analysis The *R*-value of the existing wall for the winter conditions is

$$R_{\text{existing wall}} = 1/U_{\text{existing wall}} = 1/2.25 = 0.444 \text{ m}^2 \cdot \text{°C/W}$$



Noting that the added thermal resistances are in series, the overall *R*-value of the wall becomes

$$R_{\text{modified wall}} = R_{\text{existing wall}} + R_{\text{brick}} + R_{\text{air layer}}$$

$$= 0.44 + 0.075 + 0.170 = 0.689 \text{ m}^2 \cdot \text{°C/W}$$

Then the *U*-value of the wall after modification becomes

$$R_{\text{modified wall}} = 1/U_{\text{modified wall}} = 1/0.689 = \mathbf{1.45\ m^2 \cdot °C/W}$$

The rate of heat transfer through the modified wall is

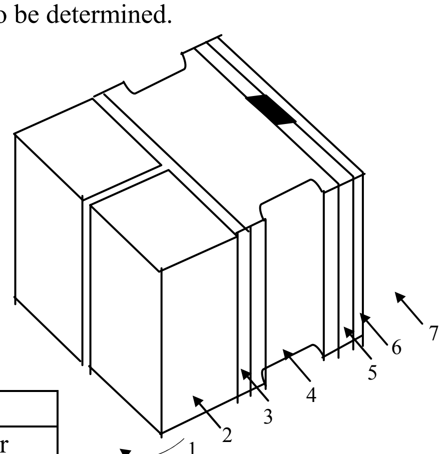
$$\dot{Q}_{\text{wall}} = (UA)_{\text{wall}}(T_i - T_o) = (1.45 \text{ W/m}^2 \cdot \text{°C})(21 \text{ m}^2)[22-(-5)\text{°C}] = \mathbf{822\ W}$$

16-68 The summer and winter *R*-values of a masonry wall are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant. **4** The air cavity does not have any reflecting surfaces.

Properties The R-values of different materials are given in Table 16-10.

Analysis Using the available *R*-values from Tables 16-10, the total *R*-value of the wall is determined in the table below.



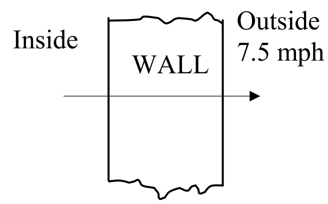
Construction	*R*-value, m^2.°C/W Summer	Winter
1a. Outside surface, 24 km/h (winter)	---	0.030
1b. Outside surface, 12 km/h (summer)	0.044	---
2. Face brick, 100 mm	0.075	0.075
3. Cement mortar, 13 mm	0.12	0.12
4. Concrete block, lightweight, 100 mm	0.27	0.27
5. Air space, nonreflecting, 40-mm	0.16	0.16
5. Plaster board, 20 mm	0.122	0.122
6. Inside surface, still air	0.12	0.12
Total unit thermal resistance of each section (the *R*-value) , m^2.°C/W	**0.809**	**0.795**

Therefore, the overall unit thermal resistance of the wall is $R = 0.809$ m^2.°C/W in summer and $R = 0.795$ m^2.°C/W in winter.

16-69E The *U*-value of a wall for 7.5 mph winds outside are given. The *U*-value of the wall for the case of 15 mph winds outside is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant except the one at the outer surface.

Properties The R-values at the outer surface of a wall for summer (7.5 mph winds) and winter (15 mph winds) conditions are given in Table 16-10 to be

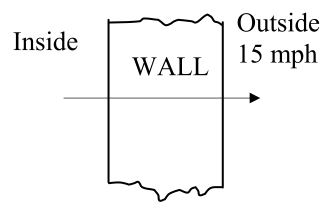


$$R_{\text{o, 7.5 mph}} = R_{\text{o, summer}} = 0.25\ \text{h.ft}^2.°\text{F/Btu}$$

and $$R_{\text{o, 15 mph}} = R_{\text{o, winter}} = 0.17\ \text{h.ft}^2.°\text{F/Btu}$$

Analysis The *R*-value of the wall at 7.5 mph winds (summer) is

$$R_{\text{wall, 7.5 mph}} = 1/U_{\text{wall, 7.5 mph}} = 1/0.075 = 13.33\ \text{h.ft}^2\cdot°\text{F/Btu}$$



Noting that the added and removed thermal resistances are in series, the overall ***R***-value of the wall at 15 mph (winter) conditions is obtained by replacing the summer value of outer convection resistance by the winter value,

$$\begin{aligned} R_{\text{wall,15 mph}} &= R_{\text{wall, 7.5 mph}} - R_{\text{o, 7.5 mph}} + R_{\text{o,15 mph}} \\ &= 13.33 - 0.25 + 0.17 = 13.25\ \text{h.ft}^2\cdot°\text{F/Btu} \end{aligned}$$

Then the U-value of the wall at 15 mph winds becomes

$$R_{\text{wall,15 mph}} = 1/U_{\text{wal,15 mph}} = 1/13.25 = \mathbf{0.0755\ h.ft^2\cdot°F/Btu}$$

Discussion Note that the effect of doubling the wind velocity on the *U*-value of the wall is less than 1 percent since

$$\text{Change} = \frac{\Delta U - \text{value}}{U - \text{value}} = \frac{0.0755 - 0.075}{0.075} = 0.0067 \quad (\text{or } 0.67\%)$$

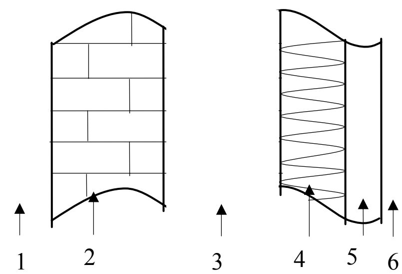
16-70 Two homes are identical, except that their walls are constructed differently. The house that is more energy efficient is to be determined.

Assumptions **1** The homes are identical, except that their walls are constructed differently. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

Properties The R-values of different materials are given in Table 16-10.

Analysis Using the available *R*-values from Tables 16-10, the total *R*-value of the masonry wall is determined in the table below.

Construction	*R*-value, $m^2.°C/W$
1. Outside surface, 24 km/h (winter)	0.030
2. Concrete block, light weight, 200 mm	2×0.27=0.54
3. Air space, nonreflecting, 20 mm	0.17
4. Urethane foam insulation, 25-mm	0.98
5. Plasterboard, 20 mm	0.12
6. Inside surface, still air	0.12



Total unit thermal resistance (the *R*-value)	**0.98 $m^2.°C/W$**

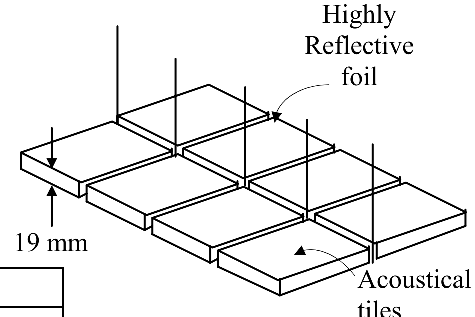
which is less than 2.4 $m^2.°C/W$. Therefore, the standard *R*-2.4 $m^2.°C/W$ wall is better insulated and thus it is more energy efficient.

16-71 A ceiling consists of a layer of reflective acoustical tiles. The *R*-value of the ceiling is to be determined for winter conditions.

Assumptions **1** Heat transfer through the ceiling is one-dimensional. **3** Thermal properties of the ceiling and the heat transfer coefficients are constant.

Properties The R-values of different materials are given in Tables 16-10 and 16-11.

Analysis Using the available *R*-values, the total *R*-value of the ceiling is determined in the table below.



Construction	*R*-value, $m^2.°C/W$
1. Still air, reflective horizontal surface facing up	R = 1/h = 1/4.32 = 0.23
2. Acoustic tile, 19 mm	0.32
3. Still air, horizontal surface, facing down	R = 1/h = 1/9.26 = 0.11

Total unit thermal resistance (the *R*-value)	**0.66 $m^2.°C/W$**

Therefore, the R-value of the hanging ceiling is 0.66 $m^2.°C/W$.

Heat Loss from Basement Walls and Floors

16-72C The mechanism of heat transfer from the basement walls and floors to the ground is *conduction heat transfer* because of the direct contact between the walls and the floor. The rate of heat loss through the ground depends on the thermal conductivity of the soil, which depends on the composition and moisture content of the soil. The higher the moisture content, the higher the thermal conductivity, and the higher the rate of heat transfer.

16-73C For a basement wall that is completely below grade, the heat loss through the upper half of the wall will be greater than the heat loss through the lower half since the heat at a lower section must pass thorough a longer path to reach the ground surface, and thus overcome a larger thermal resistance.

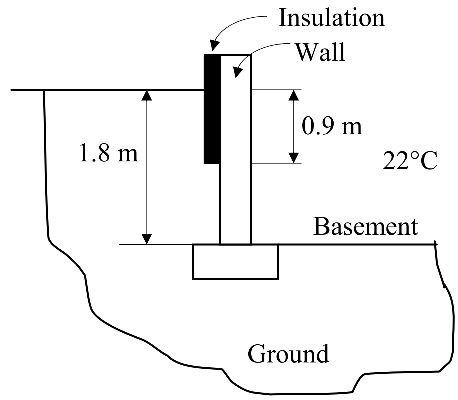
16-74C A building loses more heat to the ground through the below grade section of the basement wall than it does through the floor of a basement per unit surface area. This is because the floor has a very long path for heat transfer to the ground surface compared to the wall.

16-75C Heat transfer from a floor on grade at ground level is proportional to the perimeter of the floor, not the surface area.

16-76C Venting a crawl space increases heat loss through the floor since it will expose the bottom of the floor to a lower temperature in winter. Venting a crawl space in summer will increase heat gain through the floor since it will expose the bottom of the floor to a higher temperature.

16-77C The cold water pipes in an unheated crawl space in winter does not need to be insulated to avoid the danger of freezing in winter if the vents of the crawl space are tightly closed since, in this case, the temperature in the crawl space will be somewhere between the house temperature and the ambient temperature that will normally be above freezing.

16-78 The peak heat loss from a below grade basement in Anchorage, Alaska to the ground through its walls and the floor is to be determined.



Assumptions **1** Steady operating conditions exist. **2** The basement is maintained at 20°C.

Properties The heat transfer coefficients are given in Table 16-14a, and the amplitudes in Fig. 16-37.

Solution The floor and wall areas of the basement are

$$A_{\text{wall}} = \text{Height} \times \text{Perimeter} = 2 \times (1.8\ \text{m})(7 + 10\ \text{m}) = 61.2\ \text{m}^2$$

$$A_{\text{floor}} = \text{Length} \times \text{Width} = (7\ \text{m})(10\ \text{m}) = 70\ \text{m}^2$$

The amplitude of the annual soil temperature is determined from Fig. 16-37 to be 10°C. Then the ground surface temperature for the design heat loss becomes

$$T_{\text{ground surface}} = T_{\text{winter, mean}} - A = -5 - 10 = -15°\text{C}$$

The top 0.9-m section of the wall below the grade is insulated with *R*-2.2, and the heat transfer coefficients through that section are given in Table 16-14a to be 1.27, 1.20, and 1.00 W/m^2.°C through the 1st, 2nd, and 3rd 0.3-m wide depth increments, respectively. The heat transfer coefficients through the uninsulated section of the wall which extends from 0.9 m to 1.8 m level is determined from the same table to be 2.23, 1.80, and 1.50 W/m^2.°C for each of the remaining 0.3-m wide depth increments. The average overall heat transfer coefficient is

$$U_{\text{wall, ave}} = \frac{\sum U_{\text{wall}}}{\text{No. of increments}} = \frac{1.27 + 1.2 + 1.0 + 2.23 + 1.8 + 1.5}{6} = 1.50\ \text{W/m}^2.°\text{C}$$

Then the heat loss through the basement wall becomes

$$\dot{Q}_{\text{basement walls}} = U_{\text{wall, ave}} A_{\text{wall}} (T_{\text{basement}} - T_{\text{ground surface}})$$
$$= (1.50\ \text{W/m}^2.°\text{C})(61.2\ \text{m}^2)[20 - (-15)°\text{C}] = 3213\ \text{W}$$

The shortest width of the house is 7 m, and the depth of the foundation below grade is 1.8 m. The floor heat transfer coefficient is given in Table 16-14b to be 0.15 W/m^2.°C. Then the heat loss through the floor of the basement becomes

$$\dot{Q}_{\text{basement floor}} = U_{\text{floor}} A_{\text{floor}} (T_{\text{basement}} - T_{\text{ground surface}})$$
$$= (0.15\ \text{W/m}^2.°\text{C})(70\ \text{m}^2)[20 - (-15)]°\text{C} = 368\ \text{W}$$

which is considerably less than the heat loss through the wall. The total heat loss from the basement is then determined to be

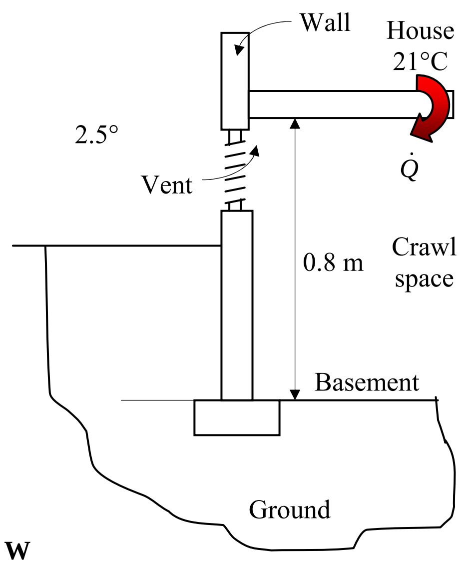
$$\dot{Q}_{\text{basement}} = \dot{Q}_{\text{basement wall}} + \dot{Q}_{\text{basement floor}} = 3213 + 368 = \mathbf{3581\ W}$$

Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-79 The vent of the crawl space is kept open. The rate of heat loss to the crawl space through insulated and uninsulated floors is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The vented crawl space temperature is the same as the ambient temperature.

Properties The overall heat transfer coefficient for the insulated floor is given in Table 16-15 to be 0.432 W/m^2.°C. It is 1.42 W/m^2.°C for the uninsulated floor.



Analysis (*a*) The floor area of the house (or the ceiling area of the crawl space) is

$$A_{\text{floor}} = \text{Length} \times \text{Width} = (8\text{ m})(12\text{ m}) = 96\text{ m}^2$$

Then the heat loss from the house to the crawl space becomes

$$\begin{aligned}\dot{Q}_{\text{insulated floor}} &= U_{\text{insulated floor}} A_{\text{floor}} (T_{\text{indoor}} - T_{\text{crawl}})\\ &= (0.432\text{ W/m}^2\cdot°\text{C})(180\text{ m}^2)(21-2.5)°\text{C}\\ &= \mathbf{1439\ W}\end{aligned}$$

(*b*) The heat loss for the uninsulated case is determined similarly to be

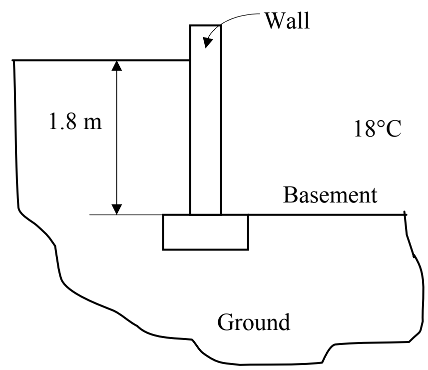
$$\begin{aligned}\dot{Q}_{\text{uninsulated floor}} &= U_{\text{uninsulated floor}} A_{\text{floor}} (T_{\text{indoor}} - T_{\text{crawl}})\\ &= (1.42\text{ W/m}^2.°\text{C})(180\text{ m}^2)(21-2.5)°\text{C} = \mathbf{4729\ W}\end{aligned}$$

Discussion Note that heat loss through the uninsulated floor is more than 3 times the heat loss through the insulated floor. Therefore, it is a good practice to insulate floors when the crawl space is ventilated to conserve energy and enhance comfort.

16-80 The peak heat loss from a below grade basement in Boise, Idaho to the ground through its walls and the floor is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The basement is maintained at 18°C.

Properties The heat transfer coefficients are given in Table 16-14a, and the amplitudes in Fig. 16-37. The mean winter temperature in Boise is 4.6°C (Table 16-5).



Analysis The floor and wall areas of the basement are

$$A_{\text{wall}} = \text{Height} \times \text{Perimeter} = 2 \times (1.8\text{ m})(8+12\text{ m}) = 108\text{ m}^2$$

$$A_{\text{floor}} = \text{Length} \times \text{Width} = (8\text{ m})(12\text{ m}) = 96\text{ m}^2$$

The amplitude of the annual soil temperature is determined from Fig. 16-37 to be 11°C. Then the ground surface temperature for the design heat loss becomes

$$T_{\text{ground surface}} = T_{\text{winter, mean}} - A = 4.6 - 11 = -6.4°\text{C}$$

The entire 1.8-m section of the wall below the grade is uninsulated, and the heat transfer coefficients through that section are given in Table 16-14a to be 7.77, 4.20, 2.93, 2.23, 1.80, and 1.50 W/m^2.°C for each 0.3-m wide depth increments. The average overall heat transfer coefficient is

$$U_{\text{wall, ave}} = \frac{\sum U_{\text{wall}}}{\text{No. of increments}} = \frac{7.77+4.20+2.93+2.23+1.8+1.5}{6} = 3.405\text{ W/m}^2\cdot°\text{C}$$

Then the heat loss through the basement wall becomes

$$\begin{aligned}\dot{Q}_{\text{basement walls}} &= U_{\text{wall, ave}} A_{\text{wall}} (T_{\text{basement}} - T_{\text{ground surface}}) \\ &= (3.405\text{ W/m}^2\cdot°\text{C})(108\text{ m}^2)[18-(-6.4)°\text{C}] = 8973\text{ W}\end{aligned}$$

The shortest width of the house is 12 m, and the depth of the foundation below grade is 1.8 m. The floor heat transfer coefficient is given in Table 16-14b to be 0.10 W/m^2.°C. Then the heat loss through the floor of the basement becomes

$$\begin{aligned}\dot{Q}_{\text{basement floor}} &= U_{\text{floor}} A_{\text{floor}} (T_{\text{basement}} - T_{\text{ground surface}}) \\ &= (0.10\text{ W/m}^2.°\text{C})(96\text{ m}^2)[18-(-6.4)]°\text{C} = 234\text{ W}\end{aligned}$$

which is considerably less than the heat loss through the wall. The total heat loss from the basement is then determined to be

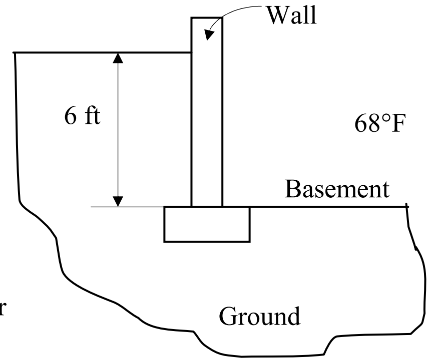
$$\dot{Q}_{\text{basement}} = \dot{Q}_{\text{basement wall}} + \dot{Q}_{\text{basement floor}} = 8973 + 234 = \mathbf{9207\ W}$$

Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-81E The peak heat loss from a below grade basement in Boise, Idaho to the ground through its walls and the floor is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The basement is maintained at 68°F.

Properties The heat transfer coefficients are given in Table 16-14a, and the amplitudes in Fig. 16-37. The mean winter temperature in Boise is 39.7°F (Table 16-5).



Solution The floor and wall areas of the basement are

$$A_{\text{wall}} = \text{Height} \times \text{Perimeter} = 2 \times (6\text{ ft})(32 + 60\text{ ft}) = 1104\text{ ft}^2$$

$$A_{\text{floor}} = \text{Length} \times \text{Width} = (32\text{ ft})(60\text{ ft}) = 1920\text{ ft}^2$$

The amplitude of the annual soil temperature is determined from Fig. 16-37 to be 19.8°F. Then the ground surface temperature for the design heat loss becomes

$$T_{\text{ground surface}} = T_{\text{winter, mean}} - A = 39.7 - 19.8 = 19.9°\text{F}$$

The entire 6-ft section of the wall below the grade is uninsulated, and the heat transfer coefficients through that section are given in Table 16-14a to be 0.410, 0.222, 0.155, 0.119, 0.096, 0.079 Btu/h.ft^2.°F for each 1-ft wide depth increments. The average overall heat transfer coefficient is

$$U_{\text{wall, ave}} = \frac{\sum U_{\text{wall}}}{\text{No. of increments}} = \frac{0.410 + 0.222 + 0.155 + 0.119 + 0.096 + 0.079}{6} = 0.180\text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

Then the heat loss through the basement wall becomes

$$\begin{aligned}\dot{Q}_{\text{basement walls}} &= U_{\text{wall, ave}} A_{\text{wall}} (T_{\text{basement}} - T_{\text{ground surface}}) \\ &= (0.180\text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F})(1104\text{ ft}^2)(68 - 19.9)°\text{F} = 9558\text{ Btu/h}\end{aligned}$$

The shortest width of the house is 32 ft, and the depth of the foundation below grade is 6 ft. The floor heat transfer coefficient is given in Table 16-14b to be 0.022 Btu/h.ft^2.°F. Then the heat loss through the floor of the basement becomes

$$\begin{aligned}\dot{Q}_{\text{basement floor}} &= U_{\text{floor}} A_{\text{floor}} (T_{\text{basement}} - T_{\text{ground surface}}) \\ &= (0.022\text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F})(1920\text{ ft}^2)(68 - 19.9)°\text{F} = 2032\text{ Btu/h}\end{aligned}$$

which is considerably less than the heat loss through the wall. The total heat loss from the basement is then determined to be

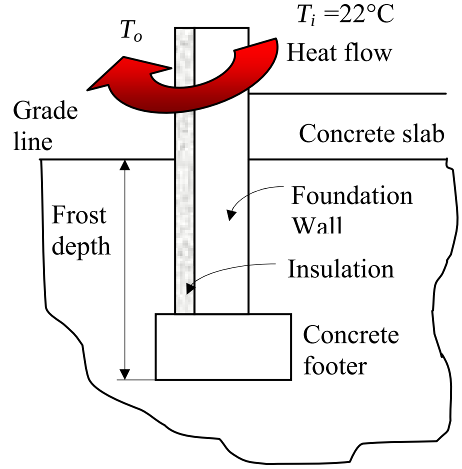
$$\dot{Q}_{\text{basement}} = \dot{Q}_{\text{basement wall}} + \dot{Q}_{\text{basement floor}} = 9558 + 2032 = \mathbf{11{,}590\ Btu/h}$$

Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-82 A house with a concrete slab floor sits directly on the ground at grade level, and the wall below grade is insulated. The heat loss from the floor at winter design conditions is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The house is maintained at 22°C. **3** The weather in Baltimore is moderate.

Properties The 97.5% winter design conditions in Baltimore is -11°C (Table 16-4). The heat transfer coefficient for the insulated wall below grade is $U = 0.86$ W/m.°C (Table 16-14c).



Solution Heat transfer from a floor on the ground at the grade level is proportional to the perimeter of the floor, and the perimeter in this case is

$$p_{\text{floor}} = 2 \times (\text{Length} + \text{Width}) = 2(15+18)\text{ m} = 66\text{ m}$$

Then the heat loss from the floor becomes

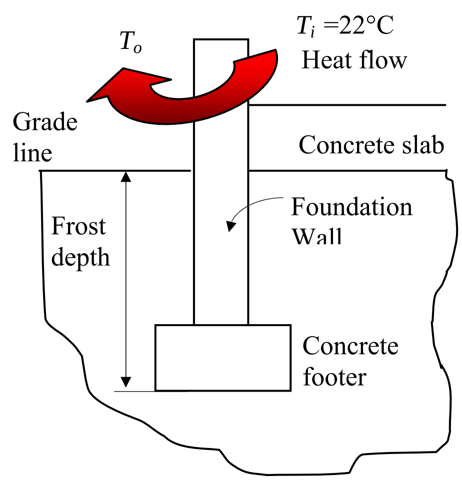
$$\begin{aligned}\dot{Q}_{\text{floor}} &= U_{\text{floor}}\, p_{\text{floor}} (T_{\text{indoor}} - T_{\text{outdoor}})\\ &= (0.86\text{ W/m}\cdot°\text{C})(66\text{ m})[22-(-11)]°\text{C} = \mathbf{1873\ W}\end{aligned}$$

Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-83 A house with a concrete slab floor sits directly on the ground at grade level, and the wall below grade is uninsulated. The heat loss from the floor at winter design conditions is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The house is maintained at 22°C. **3** The weather in Baltimore is moderate.

Properties The 97.5% winter design conditions in Baltimore is -11°C (Table 16-4). The heat transfer coefficient for the uninsulated wall below grade is $U = 1.17$ W/m.°C (Table 16-14c).



Solution Heat transfer from a floor on the ground at the grade level is proportional to the perimeter of the floor, and the perimeter in this case is

$$p_{\text{floor}} = 2 \times (\text{Length} + \text{Width}) = 2(15+18)\text{ m} = 66\text{ m}$$

Then the heat loss from the floor becomes

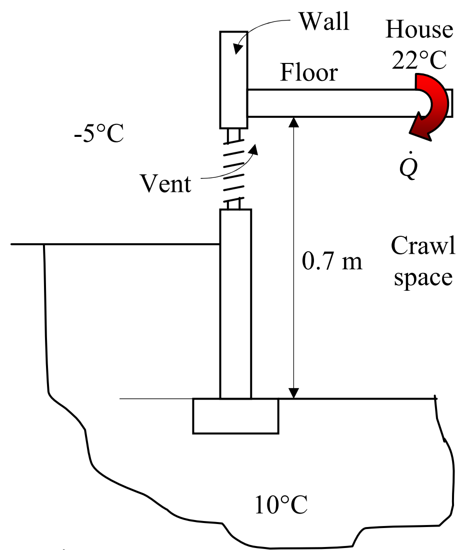
$$\begin{aligned}\dot{Q}_{\text{floor}} &= U_{\text{floor}}\, p_{\text{floor}} (T_{\text{indoor}} - T_{\text{outdoor}})\\ &= (1.17\text{ W/m}\cdot°\text{C})(66\text{ m})[22-(-11)]°\text{C} = \mathbf{2548\ W}\end{aligned}$$

Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-84 The vents of the crawl space of a house are kept closed, but air still infiltrates The heat loss from the house to the crawl space and the crawl space temperature are to be determined for the cases of insulated and uninsulated walls, floor, and ceiling of the crawl space.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties and heat transfer coefficients remain constant. **3** The atmospheric pressure is 1 atm.

Properties The indoor and outdoor design temperatures are given to be 22°C and -5°C, respectively, and the deep-down ground temperature is 10°C. The properties of air at -5°C and 1 atm are $\rho = 1.328\ \text{kg/m}^3$ and c_p = 1.004 kJ/kg.°C (Table A-11). The overall heat transfer coefficients (the U-values) for insulated and uninsulated crawl spaces are given in Table 16-15 to be



	U, W/m^2.°C	
Application	uninsulated	insulated*
Floor above crawl space	1.42	0.432
Ground of crawl space	0.437	0.437
Wall of crawl space	2.77	1.07

*An insulation R-value of 1.94 m^2.°C/W is used on the floor, and 0.95 m^2.°C/W on the walls.

Analysis (a) The floor (ceiling), ground, and wall areas of the crawl space are

$$A_{\text{floor}} = A_{\text{ground}} = \text{Length} \times \text{Width} = (12\ \text{m})(20\ \text{m}) = 240\ \text{m}^2$$

$$A_{\text{wall}} = \text{Height} \times \text{Perimeter} = (0.7\ \text{m})[2 \times (12 + 20\ \text{m})] = 44.8\ \text{m}^2$$

The volume of the crawl space and the infiltration heat loss is

$$V_{\text{crawl}} = (\text{Height})(\text{Width})(\text{Length}) = (0.7\ \text{m})(12\ \text{m})(20\ \text{m}) = 168\ \text{m}^3$$

$$\begin{aligned}\dot{Q}_{\text{infiltration}} &= \rho \dot{V} c_p (T_{\text{ambient}} - T_{\text{crawl}}) = \rho (V_{\text{crawl}} \times ACH) c_p (T_{\text{ambient}} - T_{\text{crawl}}) \\ &= (1.328\ \text{kg/m}^3)(168\ \text{m}^3)(1.2/\text{h})(1004\ \text{J/kg}\cdot°\text{C})(-5 - T_{\text{crawl}})°\text{C} \\ &= 268{,}800(-5 - T_{\text{crawl}})\ \text{J/h} = 74.7(-5 - T_{\text{crawl}})\ \text{W} \quad (\text{since } 1\ \text{h} = 3600\ \text{s})\end{aligned}$$

Noting that under steady conditions the net heat transfer to the crawl space is zero, the energy balance for the crawl space can be written as

$$\dot{Q}_{\text{floor}} + \dot{Q}_{\text{wall}} + \dot{Q}_{\text{ground}} + \dot{Q}_{\text{infiltration}} = 0$$

or

$$[UA(T_{\text{indoor}} - T_{\text{crawl}})]_{\text{floor}} + [UA(T_{\text{ambient}} - T_{\text{crawl}})]_{\text{wall}} + [UA(T_{\text{ground}} - T_{\text{crawl}})]_{\text{ground}} + \rho \dot{V} c_p (T_{\text{ambient}} - T_{\text{crawl}}) = 0$$

Using the U-values from the table above for the **insulated case** and substituting,

$$\begin{aligned}&(0.432\ \text{W/m}^2\cdot°\text{C})(240\ \text{m}^2)(22 - T_{\text{crawl}})°\text{C} + (1.07\ \text{W/m}^2\cdot°\text{C})(44.8\ \text{m}^2)(-5 - T_{\text{crawl}})°\text{C} \\ &\quad + (0.437\ \text{W/m}^2\cdot°\text{C})(240\ \text{m}^2)(10 - T_{\text{crawl}})°\text{C} + 74.7(-5 - T_{\text{crawl}})\ \text{W} = 0\end{aligned}$$

Solving for the equation above for the crawl space temperature gives T_{crawl} = **1.3°C.**

(b) Similarly, using the U-values from the table above for the **uninsulated case** and substituting,

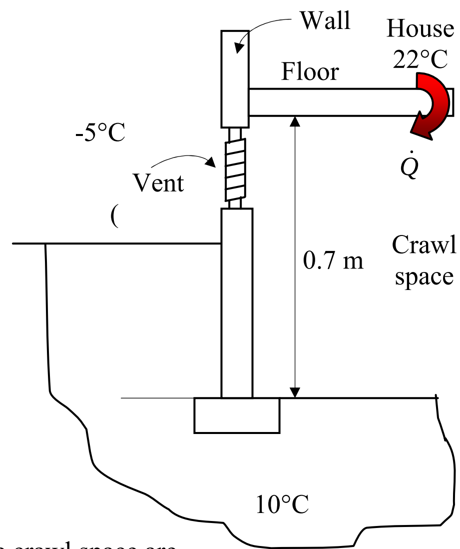
$$\begin{aligned}&(1.42\ \text{W/m}^2\cdot°\text{C})(240\ \text{m}^2)(22 - T_{\text{crawl}})°\text{C} + (2.77\ \text{W/m}^2\cdot°\text{C})(44.8\ \text{m}^2)(-5 - T_{\text{crawl}})°\text{C} \\ &\quad + (0.437\ \text{W/m}^2\cdot°\text{C})(240\ \text{m}^2)(10 - T_{\text{crawl}})°\text{C} + 74.7(-5 - T_{\text{crawl}})\ \text{W} = 0\end{aligned}$$

Solving for the equation above for the crawl space temperature gives T_{crawl} = **11.7°C.**

16-85 The vents of the crawl space of a house are tightly sealed, and no air infiltrates. The heat loss from the house to the crawl space and the crawl space temperature are to be determined for the cases of insulated and uninsulated walls, floor, and ceiling of the crawl space.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties and heat transfer coefficients remain constant. **3** The atmospheric pressure is 1 atm. **4** Air infiltration is negligible.

Properties The indoor and outdoor design temperatures are given to be 22°C and -5°C, respectively, and the deep-down ground temperature is 10°C. The properties of air at -5°C and 1 atm are $\rho = 1.328 \text{ kg/m}^3$ and C_p = 1.004 kJ/kg.°C (Table A-11). The overall heat transfer coefficients (the *U*-values) for insulated and uninsulated crawl spaces are given in Table 16-15 to be



Application	*U*, W/m^2.°C	
	uninsulated	insulated*
Floor above crawl space	1.42	0.432
Ground of crawl space	0.437	0.437
Wall of crawl space	2.77	1.07

*An insulation *R*-value of 1.94 m^2.°C/W is used on the floor, and 0.95 m^2.°C/W on the walls.

Analysis (*a*) The floor (ceiling), ground, and wall areas of the crawl space are

$$A_{\text{floor}} = A_{\text{ground}} = \text{Length} \times \text{Width} = (12\text{ m})(20\text{ m}) = 240\text{ m}^2$$

$$A_{\text{wall}} = \text{Height} \times \text{Perimeter} = (0.7\text{ m})[2 \times (12 + 20\text{ m})] = 44.8\text{ m}^2$$

Noting that under steady conditions the net heat transfer to the crawl space is zero, the energy balance for the crawl space can be written as

$$\dot{Q}_{\text{floor}} + \dot{Q}_{\text{wall}} + \dot{Q}_{\text{ground}} = 0$$

or

$$[UA(T_{\text{indoor}} - T_{\text{crawl}})]_{\text{floor}} + [UA(T_{\text{ambient}} - T_{\text{crawl}})]_{\text{wall}} + [UA(T_{\text{ground}} - T_{\text{crawl}})]_{\text{ground}} = 0$$

Using the *U*-values from the table above for the **insulated case** and substituting,

$$(0.432\text{ W/m}^2\cdot°\text{C})(240\text{ m}^2)(22 - T_{\text{crawl}})°\text{C} + (1.07\text{ W/m}^2\cdot°\text{C})(44.8\text{ m}^2)(-5 - T_{\text{crawl}})°\text{C}$$
$$+ (0.437\text{ W/m}^2\cdot°\text{C})(240\text{ m}^2)(10 - T_{\text{crawl}})°\text{C} = 0$$

Solving for the equation above for the crawl space temperature gives T_{crawl} = **6.5°C.**

(*b*) Similarly, using the *U*-values from the table above for the **uninsulated case** and substituting,

$$(1.42\text{ W/m}^2\cdot°\text{C})(240\text{ m}^2)(22 - T_{\text{crawl}})°\text{C} + (2.77\text{ W/m}^2\cdot°\text{C})(44.8\text{ m}^2)(-5 - T_{\text{crawl}})°\text{C}$$
$$+ (0.437\text{ W/m}^2\cdot°\text{C})(240\text{ m}^2)(10 - T_{\text{crawl}})°\text{C} = 0$$

Solving for the equation above for the crawl space temperature gives T_{crawl} = **13.9°C.**

Heat Transfer through Windows

16-86C Windows are considered in three regions when analyzing heat transfer through them because the structure and properties of the frame are quite different than those of the glazing. As a result, heat transfer through the frame and the edge section of the glazing adjacent to the frame is two-dimensional. Even in the absence of solar radiation and air infiltration, heat transfer through the windows is more complicated than it appears to be. Therefore, it is customary to consider the windows in three regions when analyzing heat transfer through them: (1) the *center-of-glass*, (2) the *edge-of-glass*, and (3) the *frame* regions. When the heat transfer coefficient for all three regions are known, the overall U-value of the window is determined from

$$U_{\text{window}} = (U_{\text{center}}A_{\text{center}} + U_{\text{edge}}A_{\text{edge}} + U_{\text{frame}}A_{\text{frame}}) / A_{\text{window}}$$

where A_{window} is the window area, and A_{center}, A_{edge}, and A_{frame} are the areas of the center, edge, and frame sections of the window, respectively, and U_{center}, U_{edge}, and U_{frame} are the heat transfer coefficients for the center, edge, and frame sections of the window.

16-87C Of the three similar double pane windows with air gab widths of 5, 10, and 20 mm, the U-factor and thus the rate of heat transfer through the window will be a minimum for the window with 10-mm air gab, as can be seen from Fig. 16-44.

16-88C In an ordinary double pane window, about half of the heat transfer is by radiation. A practical way of reducing the radiation component of heat transfer is to reduce the emissivity of glass surfaces by coating them with low-emissivity (or "low-e") material.

16-89C When a thin polyester film is used to divide the 20-mm wide air of a double pane window space into two 10-mm wide layers, both (*a*) convection and (*b*) radiation heat transfer through the window will be reduced.

16-90C When a double pane window whose air space is flashed and filled with argon gas, (*a*) convection heat transfer will be reduced but (*b*) radiation heat transfer through the window will remain the same.

16-91C The heat transfer rate through the glazing of a double pane window is higher at the edge section than it is at the center section because of the two-dimensional effects due to heat transfer through the frame.

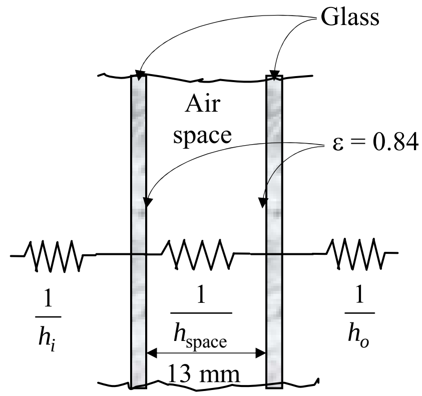
16-92C The *U*-factors of windows with aluminum frames will be highest because of the higher conductivity of aluminum. The U-factors of wood and vinyl frames are comparable in magnitude.

16-93 The U-factor for the center-of-glass section of a double pane window is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the window is one-dimensional. **3** The thermal resistance of glass sheets is negligible.

Properties The emissivity of clear glass is given to be 0.84. The values of h_i and h_o for winter design conditions are h_i = 8.29 W/m^2.°C and h_o = 34.0 W/m^2.°C (Table 16-11).

Analysis Disregarding the thermal resistance of glass sheets, which are small, the U-factor for the center region of a double pane window is determined from



$$\frac{1}{U_{center}} \cong \frac{1}{h_i} + \frac{1}{h_{space}} + \frac{1}{h_o}$$

where h_i, h_{space}, and h_o are the heat transfer coefficients at the inner surface of window, the air space between the glass layers, and the outer surface of the window, respectively. The effective emissivity of the air space of the double pane window is

$$\varepsilon_{effective} = \frac{1}{1/\varepsilon_1 + 1/\varepsilon_2 - 1} = \frac{1}{1/0.84 + 1/0.84 - 1} = 0.72$$

For this value of emissivity and an average air space temperature of 10°C with a temperature difference across the air space to be 15°C, we read h_{space} = 5.7 W/m^2.°C from Table 9-3 for 13-mm thick air space. Therefore,

$$\frac{1}{U_{center}} = \frac{1}{8.29} + \frac{1}{5.7} + \frac{1}{34.0} \longrightarrow U_{center} = \mathbf{3.07\ W/m^2 \cdot °C}$$

Discussion The overall U-factor of the window will be higher because of the edge effects of the frame.

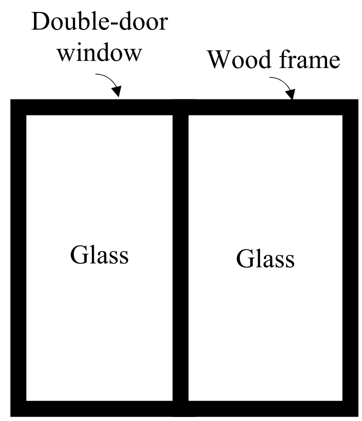
16-94 The rate of heat loss through an double-door wood framed window and the inner surface temperature are to be determined for the cases of single pane, double pane, and low-e triple pane windows.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the window is one-dimensional. **3** Thermal properties of the windows and the heat transfer coefficients are constant. **4** Infiltration heat losses are not considered.

Properties The U-factors of the windows are given in Table 16-19.

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 which is determined to be

$$A_{window} = \text{Height} \times \text{Width} = (1.2\text{ m})(1.8\text{ m}) = 2.16\text{ m}^2$$

The U-factors for the three cases can be determined directly from Table 9-6 to be 5.57, 2.86, and 1.46 W/m^2.°C, respectively. Also, the inner surface temperature of the window glass can be determined from Newton's law,

$$\dot{Q}_{\text{window}} = h_i A_{\text{window}} (T_i - T_{\text{glass}}) \longrightarrow T_{\text{glass}} = T_i - \frac{\dot{Q}_{\text{window}}}{h_i A_{\text{window}}}$$

where h_i is the heat transfer coefficient on the inner surface of the window which is determined from Table 9-5 to be h_i = 8.3 W/m^2.°C. Then the rate of heat loss and the interior glass temperature for each case are determined as follows:

(*a*) Single glazing:

$$\dot{Q}_{\text{window}} = (5.57\text{ W/m}^2\cdot°\text{C})(2.16\text{ m}^2)[20-(-8)°\text{C}] = \mathbf{337\ W}$$

$$T_{\text{glass}} = T_i - \frac{\dot{Q}_{\text{window}}}{h_i A_{\text{window}}} = 20°\text{C} - \frac{337\text{ W}}{(8.29\text{ W/m}^2\cdot°\text{C})(2.16\text{ m}^2)} = \mathbf{1.2°C}$$

(*b*) Double glazing (13 mm air space):

$$\dot{Q}_{\text{window}} = (2.86\text{ W/m}^2\cdot°\text{C})(2.16\text{ m}^2)[20-(-8)°\text{C}] = \mathbf{173\ W}$$

$$T_{\text{glass}} = T_i - \frac{\dot{Q}_{\text{window}}}{h_i A_{\text{window}}} = 20°\text{C} - \frac{173\text{ W}}{(8.29\text{ W/m}^2\cdot°\text{C})(2.16\text{ m}^2)} = \mathbf{10.3°C}$$

(*c*) Triple glazing (13 mm air space, low-e coated):

$$\dot{Q}_{\text{window}} = (1.46\text{ W/m}^2\cdot°\text{C})(2.16\text{ m}^2)[20-(-8)°\text{C}] = \mathbf{88.3\ W}$$

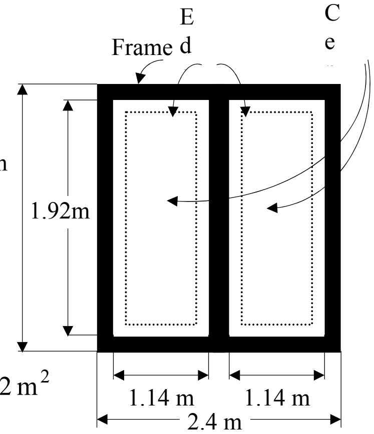
$$T_{\text{glass}} = T_i - \frac{\dot{Q}_{window}}{h_i A_{window}} = 20 - \frac{88.3\text{ W}}{(8.3\text{ W/m}^2.°\text{C})(2.16\text{ m}^2)} = \mathbf{15.1°C}$$

Discussion Note that heat loss through the window will be reduced by 49 percent in the case of double glazing and by 74 percent in the case of triple glazing relative to the single glazing case. Also, in the case of single glazing, the low inner glass surface temperature will cause considerable discomfort in the occupants because of the excessive heat loss from the body by radiation. It is raised from 1.2°C to 10.3°C in the case of double glazing and to 15.1°C in the case of triple glazing.

16-95 The overall U-factor for a double-door type window is to be determined, and the result is to be compared to the value listed in Table 16-19.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the window is one-dimensional.

Properties The U-factors for the various sections of windows are given in Tables 16-11 and 16-19.



Analysis The areas of the window, the glazing, and the frame are

$$A_{\text{window}} = \text{Height} \times \text{Width} = (2 \text{ m})(2.4 \text{ m}) = 4.80 \text{ m}^2$$

$$A_{\text{glazing}} = 2 \times \text{Height} \times \text{Width} = 2(1.92 \text{ m})(1.14 \text{ m}) = 4.38 \text{ m}^2$$

$$A_{\text{frame}} = A_{\text{window}} - A_{\text{glazing}} = 4.80 - 4.38 = 0.42 \text{ m}^2$$

The edge-of-glass region consists of a 6.5-cm wide band around the perimeter of the glazings, and the areas of the center and edge sections of the glazing are determined to be

$$A_{\text{center}} = 2(\text{Height} \times \text{Width}) = 2(1.92 - 0.13 \text{ m})(1.14 - 0.13 \text{ m}) = 3.62 \text{ m}^2$$

$$A_{\text{edge}} = A_{\text{glazing}} - A_{\text{center}} = 4.38 - 3.62 = 0.76 \text{ m}^2$$

The U-factor for the frame section is determined from Table 9-4 to be $U_{\text{frame}} = 2.8$ W/m^2.°C. The U-factor for the center and edge sections are determined from Table 9-6 to be $U_{\text{center}} = 2.78$ W/m^2.°C and $U_{\text{edge}} = 3.40$ W/m^2.°C. Then the overall U-factor of the entire window becomes

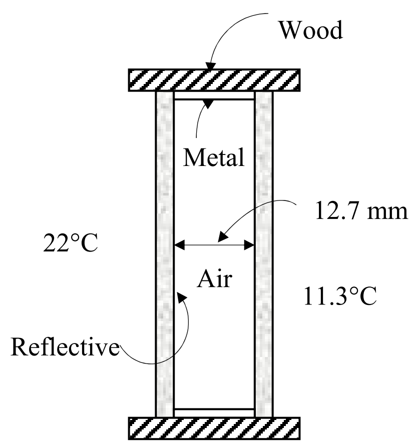
$$\begin{aligned} U_{\text{window}} &= (U_{\text{center}} A_{\text{center}} + U_{\text{edge}} A_{\text{edge}} + U_{\text{frame}} A_{\text{frame}}) / A_{\text{window}} \\ &= (2.78 \times 3.62 + 3.40 \times 0.76 + 2.8 \times 0.42) / 4.80 \\ &= \mathbf{2.88 \ W/m^2 \cdot {}^\circ C} \end{aligned}$$

Discussion The overall U-factor listed in Table 9-6 for the specified type of window is 2.86 W/m^2.°C, which is sufficiently close to the value obtained above.

16-96 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 transfer through the window is one-dimensional. **3** Thermal properties of the windows and the heat transfer coefficients are constant. **4** Infiltration heat losses are not considered.

Properties The U-factor of the window is given in Table 9-6 to be 2.13 W/m^2.°C.



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

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

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,

$$\dot{Q}_{\text{window, avg}} = (2.13 \text{ W/m}^2.{}^\circ\text{C})(14 \text{ m}^2)(22 - 11.3){}^\circ\text{C} = \mathbf{319 \ W}$$

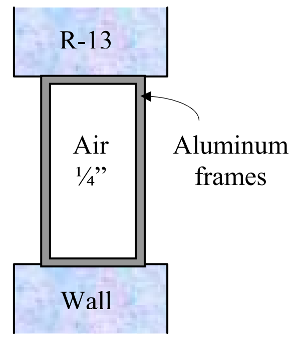
Discussion This is the "average" rate of heat transfer through the window in winter in the absence of any infiltration.

16-97E The *R*-value of the common double door windows that are double pane with 1/4-in of air space and have aluminum frames is to be compared to the *R*-value of *R*-13 wall. It is also to be determined if more heat is transferred through the windows or the walls.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the window is one-dimensional. **3** Thermal properties of the windows and the heat transfer coefficients are constant. **4** Infiltration heat losses are not considered.

Properties The U-factor of the window is given in Table 16-19 to be 4.55×0.176 = 0.801 Btu/h.ft^2.°F.

Analysis The *R*-value of the windows is simply the inverse of its *U*-factor, and is determined to be



$$R_{\text{window}} = \frac{1}{U} = \frac{1}{0.801\ \text{Btu/h}\cdot\text{ft}^2.°\text{F}} = 1.25\ \text{h}\cdot\text{ft}^2.°\text{F/Btu}$$

which is less than 13. Therefore, the *R*-value of a double pane window is **much less** than the *R*-value of an *R*-13 wall.

Now consider a 1-ft^2 section of a wall. The solid wall and the window areas of this section are A_{wall} = 0.8 ft^2 and A_{window} = 0.2 ft^2. Then the rates of heat transfer through the two sections are determined to be

$$\dot{Q}_{\text{wall}} = U_{\text{wall}} A_{\text{wall}} (T_i - T_o) = A_{\text{wall}} \frac{T_i - T_o}{R - \text{value, wall}} = (0.8\ \text{ft}^2)\frac{\Delta T(°\text{F})}{13\ \text{h.ft}^2.°\text{F/Btu}} = 0.0615\Delta T\ \text{Btu/h}$$

$$\dot{Q}_{\text{window}} = U_{\text{window}} A_{\text{window}} (T_i - T_o) = A_{\text{window}} \frac{T_i - T_o}{R - \text{value}} = (0.2\ \text{ft}^2)\frac{\Delta T(°\text{F})}{1.25\ \text{h.ft}^2.°\text{F/Btu}} = 0.160\Delta T\ \text{Btu/h}$$

Therefore, the rate of heat transfer through a double pane window is **much more** than the rate of heat transfer through an *R*-13 wall.

Discussion The ratio of heat transfer through the wall and through the window is

$$\frac{\dot{Q}_{\text{window}}}{\dot{Q}_{\text{wall}}} = \frac{0.160\ \text{Btu/h}}{0.0615\ \text{Btu/h}} = 2.60$$

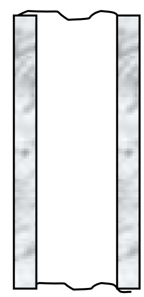
Therefore, 2.6 times more heat is lost through the windows than through the walls although the windows occupy only 20% of the wall area.

16-98 The overall U-factor of a window is given to be U = 2.76 W/m^2.°C for 12 km/h winds outside. The new U-factor when the wind velocity outside is doubled is to be determined.

Assumptions Thermal properties of the windows and the heat transfer coefficients are constant.

Properties The heat transfer coefficients at the outer surface of the window are h_o = 22.7 W/m^2.°C for 12 km/h winds, and h_o = 22.7 W/m^2.°C for 24 km/h winds (Table 16-11).

Analysis The corresponding convection resistances for the outer surfaces of the window are



$$R_{\text{o, 12 km/h}} = \frac{1}{h_{\text{o, 12 km/h}}} = \frac{1}{22.7\ \text{W/m}^2.°\text{C}} = 0.044\ \text{m}^2.°\text{C/W}$$

$$R_{\text{o, 24 km/h}} = \frac{1}{h_{\text{o, 24 km/h}}} = \frac{1}{34.0\ \text{W/m}^2.°\text{C}} = 0.029\ \text{m}^2.°\text{C/W}$$

Also, the R-value of the window at 12 km/h winds is

$$R_{\text{window, 12 km/h}} = \frac{1}{U_{\text{window, 12 km/h}}} = \frac{1}{2.76\ \text{W/m}^2.°\text{C}} = 0.362\ \text{m}^2.°\text{C/W}$$

Noting that all thermal resistances are in series, the thermal resistance of the window for 24 km/h winds is determined by replacing the convection resistance for 12 km/h winds by the one for 24 km/h:

$$R_{\text{window, 24 km/h}} = R_{\text{window, 12 km/h}} - R_{\text{o, 12 km/h}} + R_{\text{o, 24 km/h}} = 0.362 - 0.044 + 0.029 = 0.347\ \text{m}^2.°\text{C/W}$$

Then the U-factor for the case of 24 km/h winds becomes

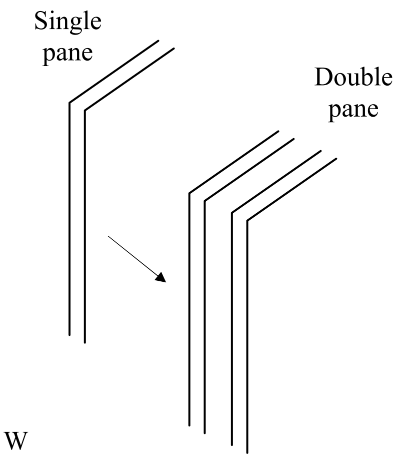
$$U_{\text{window, 24 km/h}} = \frac{1}{R_{\text{window, 24 km/h}}} = \frac{1}{0.347\ \ \text{m}^2.°\text{C/W}} = \mathbf{2.88\ W/m^2.°C}$$

Discussion Note that doubling of the wind velocity increases the U-factor only slightly (about 4%) from 2.76 to 2.88 W/m^2.°C.

16-99 The existing wood framed single pane windows of an older house in Wichita are to be replaced by double-door type vinyl framed double pane windows with an air space of 6.4 mm. The amount of money the new windows will save the home owner per month is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the window is one-dimensional. **3** Thermal properties of the windows and the heat transfer coefficients are constant. **4** Infiltration heat losses are not considered.

Properties The U-factors of the windows are 5.57 W/m^2.°C for the old single pane windows, and 3.20 W/m^2.°C. for the new double pane windows (Table 9-6).



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. Noting that the heaters will turn on only when the outdoor temperature drops below 18°C, the rates of heat transfer due to electric heating for the old and new windows are determined to be

$$\dot{Q}_{\text{window, old}} = (5.57 \text{ W/m}^2.°\text{C})(17 \text{ m}^2)(18 - 7.1)°\text{C} = 1032 \text{ W}$$

$$\dot{Q}_{\text{window, new}} = (3.20 \text{ W/m}^2.°\text{C})(17 \text{ m}^2)(18 - 7.1)°\text{C} = 593 \text{ W}$$

$$\dot{Q}_{\text{saved}} = \dot{Q}_{\text{window, old}} - \dot{Q}_{\text{window, new}} = 1032 - 593 = 439 \text{ W}$$

Then the electrical energy and cost savings per month becomes

$$\text{Energy savings} = \dot{Q}_{\text{saved}} \Delta t = (0.439 \text{ kW})(30 \times 24 \text{ h/month}) = 316 \text{ kWh/month}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of energy}) = (316 \text{ kWh/month})(\$0.085/\text{kWh}) = \mathbf{\$26.9/month}$$

Discussion We would obtain the same result if we used the actual indoor temperature (probably 22°C) for T_{i} instead of the balance point temperature of 18°C.

Solar Heat Gain through Windows

16-100C (*a*) The spectral distribution of solar radiation beyond the earth's atmosphere resembles the energy emitted by a black body at 5982°C, with about 39 percent in the visible region (0.4 to 0.7 μm), and the 52 percent in the near infrared region (0.7 to 3.5 μm). (*b*) At a solar altitude of 41.8°, the total energy of direct solar radiation incident at sea level on a clear day consists of about 3 percent ultraviolet, 38 percent visible, and 59 percent infrared radiation.

16-101C A window that transmits visible part of the spectrum while absorbing the infrared portion is ideally suited for minimizing the air-conditioning load since such windows provide maximum daylighting and minimum solar heat gain. The ordinary window glass approximates this behavior remarkably well.

16-102C The **solar heat gain coefficient (**SHGC) is defined as the fraction of incident solar radiation that enters through the glazing. The solar heat gain of a glazing relative to the solar heat gain of a reference glazing, typically that of a standard 3 mm (1/8 in) thick double-strength clear window glass sheet whose SHGC is 0.87, is called the **shading coefficient**. They are related to each other by

$$\mathrm{SC} = \frac{\text{Solar heat gain of product}}{\text{Solar heat gain of reference glazing}} = \frac{\mathrm{SHGC}}{\mathrm{SHGC}_{\mathrm{ref}}} = \frac{\mathrm{SHGC}}{0.87} = 1.15 \times \mathrm{SHGC}$$

For single pane clear glass window, SHGC = 0.87 and SC = 1.0.

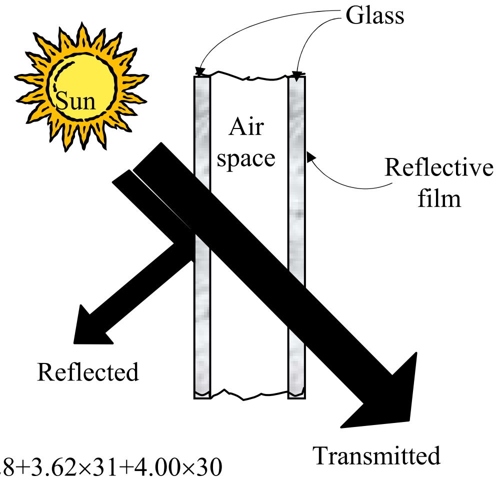
16-103C The SC (shading coefficient) of a device represents the solar heat gain relative to the solar heat gain of a reference glazing, typically that of a standard 3 mm (1/8 in) thick double-strength clear window glass sheet whose SHGC is 0.87. The shading coefficient of a 3-mm thick *clear glass* is SC = 1.0 whereas SC = 0.88 for 3-mm thick *heat absorbing glass*.

16-104C A device that blocks solar radiation and thus reduces the solar heat gain is called a shading device. External shading devices are more effective in reducing the solar heat gain since they intercept sun's rays before they reach the glazing. The solar heat gain through a window can be reduced by as much as 80 percent by exterior shading. *Light colored* shading devices maximize the back reflection and thus minimize the solar gain. *Dark colored* shades, on the other hand, minimize the back refection and thus maximize the solar heat gain.

16-105C A low-e coating on the inner surface of a window glass reduces both the (*a*) heat loss in winter and (*b*) heat gain in summer. This is because the radiation heat transfer to or from the window is proportional to the emissivity of the inner surface of the window. In winter, the window is colder and thus radiation heat loss from the room to the window is low. In summer, the window is hotter and the radiation transfer from the window to the room is low.

16-106C Glasses coated with reflective films on the outer surface of a window glass reduces solar heat both in summer and in winter.

16-107 The net annual cost savings due to installing reflective coating on the West windows of a building and the simple payback period are to be determined.



Assumptions **1** The calculations given below are for an average year. **2** The unit costs of electricity and natural gas remain constant.

Analysis Using the daily averages for each month and noting the number of days of each month, the total solar heat flux incident on the glazing during summer and winter months are determined to be

$$Q_{\text{solar, summer}} = 4.24\times30 + 4.16\times31 + 3.93\times31 + 3.48\times30 = 482 \text{ kWh/year}$$

$$Q_{\text{solar, winter}} = 2.94\times31 + 2.33\times30 + 2.07\times31 + 2.35\times31 + 3.03\times28 + 3.62\times31 + 4.00\times30 = 615 \text{ kWh/year}$$

Then the decrease in the annual cooling load and the increase in the annual heating load due to reflective film become

$$\begin{aligned}\text{Cooling load decrease} &= Q_{\text{solar, summer}} A_{\text{glazing}} (\text{SHGC}_{\text{without film}} - \text{SHGC}_{\text{with film}})\\ &= (482 \text{ kWh/year})(60 \text{ m}^2)(0.766-0.35)\\ &= 12{,}031 \text{ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Heating load increase} &= Q_{\text{solar, winter}} A_{\text{glazing}} (\text{SHGC}_{\text{without film}} - \text{SHGC}_{\text{with film}})\\ &= (615 \text{ kWh/year})(60 \text{ m}^2)(0.766-0.35)\\ &= 15{,}350 \text{ kWh/year} = 523.7 \text{ therms/year}\end{aligned}$$

since 1 therm = 29.31 kWh. The corresponding decrease in cooling costs and increase in heating costs are

$$\begin{aligned}\text{Decrease in cooling costs} &= (\text{Cooling load decrease})(\text{Unit cost of electricity})/\text{COP}\\ &= (12{,}031 \text{ kWh/year})(\$0.09/\text{kWh})/3.2 = \$338/\text{year}\end{aligned}$$

$$\begin{aligned}\text{Increase in heating costs} &= (\text{Heating load increase})(\text{Unit cost of fuel})/\text{Efficiency}\\ &= (523.7 \text{ therms/year})(\$0.45/\text{therm})/0.90 = \$262/\text{year}\end{aligned}$$

Then the net annual cost savings due to reflective films become

$$\text{Cost Savings} = \text{Decrease in cooling costs} - \text{Increase in heating costs} = \$338 - 262 = \mathbf{\$76/year}$$

The implementation cost of installing films is

$$\text{Implementation Cost} = (\$20/\text{m}^2)(60 \text{ m}^2) = \$1200$$

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$1200}{\$76/\text{year}} = \mathbf{16 \text{ years}}$$

Discussion The reflective films will pay for themselves in this case in about 16 years, which is unacceptable to most manufacturers since they are not usually interested in any energy conservation measure which does not pay for itself within 3 years.

16-108 A house located at 40º N latitude has ordinary double pane windows. The total solar heat gain of the house at 9:00, 12:00, and 15:00 solar time in July and the total amount of solar heat gain per day for an average day in January are to be determined.

Assumptions The calculations are performed for an average day in a given month.

Properties The shading coefficient of a double pane window with 6-mm thick glasses is SC = 0.82 (Table 16-21). The incident radiation at different windows at different times are given as (Table 16-20)

Month	Time	Solar radiation incident on the surface, W/m^2			
		North	East	South	West
July	9:00	117	701	190	114
July	12:00	138	149	395	149
July	15:00	117	114	190	701
January	Daily total	446	1863	5897	1863

Analysis The solar heat gain coefficient (SHGC) of the windows is determined from Eq.16-40 to be

$$\text{SHGC} = 0.87\times\text{SC} = 0.87\times 0.82 = 0.7134$$

The rate of solar heat gain is determined from

$$\dot{Q}_{\text{solar gain}} = SHGC \times A_{\text{glazing}} \times \dot{q}_{\text{solar, incident}} = 0.7134 \times A_{\text{glazing}} \times \dot{q}_{\text{solar, incident}}$$

Then the rates of heat gain at the 4 walls at 3 different times in July become

North wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.7134\times(4\text{ m}^2)\times(117\text{ W/m}^2) = \mathbf{334\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.7134\times(4\text{ m}^2)\times(138\text{ W/m}^2) = \mathbf{394\ W}$$

$$\dot{Q}_{\text{solar gain, 15:00}} = 0.7134\times(4\text{ m}^2)\times(117\text{ W/m}^2) = \mathbf{334\ W}$$

East wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.7134\times(6\text{ m}^2)\times(701\text{ W/m}^2) = \mathbf{3001\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.7134\times(6\text{ m}^2)\times(149\text{ W/m}^2) = \mathbf{638\ W}$$

$$\dot{Q}_{\text{solar gain, 15:00}} = 0.7134\times(6\text{ m}^2)\times(114\text{ W/m}^2) = \mathbf{488\ W}$$

South wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.7134\times(8\text{ m}^2)\times(190\text{ W/m}^2) = \mathbf{1084\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.7134\times(8\text{ m}^2)\times(395\text{ W/m}^2) = \mathbf{2254\ W}$$

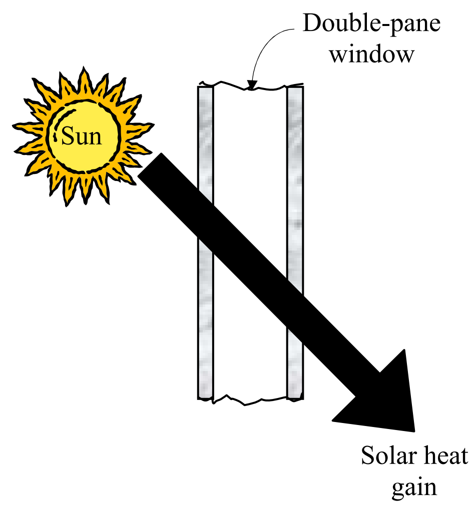
$$\dot{Q}_{\text{solar gain, 15:00}} = 0.7134\times(8\text{ m}^2)\times(190\text{ W/m}^2) = \mathbf{1084\ W}$$

West wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.7134\times(6\text{ m}^2)\times(114\text{ W/m}^2) = \mathbf{488\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.7134\times(6\text{ m}^2)\times(149\text{ W/m}^2) = \mathbf{638\ W}$$

$$\dot{Q}_{\text{solar gain, 15:00}} = 0.7134\times(6\text{ m}^2)\times(701\text{ W/m}^2) = \mathbf{3001\ W}$$



Similarly, the solar heat gain of the house through all of the windows in January is determined to be

January:

$$\dot{Q}_{\text{solar gain, North}} = 0.7134\times(4\text{ m}^2)\times(446\text{ Wh/m}^2\cdot\text{day}) = 1273\text{ Wh/day}$$

$$\dot{Q}_{\text{solar gain,East}} = 0.7134\times(6\text{ m}^2)\times(1863\text{ Wh/m}^2\cdot\text{day}) = 7974\text{ Wh/day}$$

$$\dot{Q}_{\text{solar gain, South}} = 0.7134\times(8\text{ m}^2)\times(5897\text{ Wh/m}^2\cdot\text{day}) = 33{,}655\text{ Wh/day}$$

$$\dot{Q}_{\text{solar gain,West}} = 0.7134\times(6\text{ m}^2)\times(1863\text{ Wh/m}^2\cdot\text{day}) = 7974\text{ Wh/day}$$

Therefore, for an average day in January,

$$\dot{Q}_{\text{solar gain per day}} = 1273 + 7974 + 33{,}655 + 7974 = 58{,}876\text{ Wh/day} \cong \mathbf{58.9\ kWh/day}$$

16-109 A house located at 40º N latitude has gray-tinted double pane windows. The total solar heat gain of the house at 9:00, 12:00, and 15:00 solar time in July and the total amount of solar heat gain per day for an average day in January are to be determined.

Assumptions The calculations are performed for an average day in a given month.

Properties The shading coefficient of a gray-tinted double pane window with 6-mm thick glasses is SC = 0.58 (Table 16-21). The incident radiation at different windows at different times are given as (Table 16-20)

Month	Time	Solar radiation incident on the surface, W/m^2			
		North	East	South	West
July	9:00	117	701	190	114
July	12:00	138	149	395	149
July	15:00	117	114	190	701
January	Daily total	446	1863	5897	1863

Analysis The solar heat gain coefficient (SHGC) of the windows is determined from Eq.16-40 to be

$$SHGC = 0.87\times SC = 0.87\times 0.58 = 0.5046$$

The rate of solar heat gain is determined from

$$\dot{Q}_{\text{solar gain}} = SHGC\times A_{\text{glazing}}\times \dot{q}_{\text{solar, incident}} = 0.5046\times A_{\text{glazing}}\times \dot{q}_{\text{solar, incident}}$$

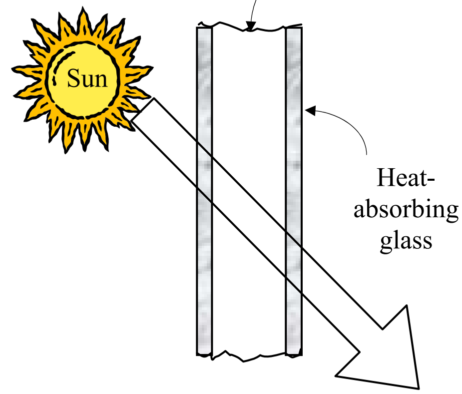
Then the rates of heat gain at the 4 walls at 3 different times in July become

North wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.5046\times(4\ \text{m}^2)\times(117\ \text{W/m}^2) = \mathbf{236\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.5046\times(4\ \text{m}^2)\times(138\ \text{W/m}^2) = \mathbf{279\ W}$$

$$\dot{Q}_{\text{solar gain,15:00}} = 0.5046\times(4\ \text{m}^2)\times(117\ \text{W/m}^2) = \mathbf{236\ W}$$



East wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.5046\times(6\ \text{m}^2)\times(701\ \text{W/m}^2) = \mathbf{2122\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.5046\times(6\ \text{m}^2)\times(149\ \text{W/m}^2) = \mathbf{451\ W}$$

$$\dot{Q}_{\text{solar gain,15:00}} = 0.5046\times(6\ \text{m}^2)\times(114\ \text{W/m}^2) = \mathbf{345\ W}$$

South wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.5046\times(8\ \text{m}^2)\times(190\ \text{W/m}^2) = \mathbf{767\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.5046\times(8\ \text{m}^2)\times(395\ \text{W/m}^2) = \mathbf{1595\ W}$$

$$\dot{Q}_{\text{solar gain,15:00}} = 0.5046\times(8\ \text{m}^2)\times(190\ \text{W/m}^2) = \mathbf{767\ W}$$

West wall:

$$\dot{Q}_{\text{solar gain, 9:00}} = 0.5046\times(6\ \text{m}^2)\times(114\ \text{W/m}^2) = \mathbf{345\ W}$$

$$\dot{Q}_{\text{solar gain,12:00}} = 0.5046\times(6\ \text{m}^2)\times(149\ \text{W/m}^2) = \mathbf{451\ W}$$

$$\dot{Q}_{\text{solar gain,15:00}} = 0.5046\times(6\ \text{m}^2)\times(701\ \text{W/m}^2) = \mathbf{2122\ W}$$

Similarly, the solar heat gain of the house through all of the windows in January is determined to be

January:

$$\dot{Q}_{\text{solar gain, North}} = 0.5046\times(4\ \text{m}^2)\times(446\ \text{Wh/m}^2\cdot\text{day}) = 900\ \text{Wh/day}$$

$$\dot{Q}_{\text{solar gain,East}} = 0.5046\times(6\ \text{m}^2)\times(1863\ \text{Wh/m}^2\cdot\text{day}) = 5640\ \text{Wh/day}$$

$$\dot{Q}_{\text{solar gain, South}} = 0.5046\times(8\ \text{m}^2)\times(5897\ \text{Wh/m}^2\cdot\text{day}) = 23{,}805\ \text{Wh/day}$$

$$\dot{Q}_{\text{solar gain,West}} = 0.5046\times(6\ \text{m}^2)\times(1863\ \text{Wh/m}^2\cdot\text{day}) = 5640\ \text{Wh/day}$$

Therefore, for an average day in January,

$$\dot{Q}_{\text{solar gain per day}} = 900+5640+23{,}805+5640 = 35{,}985\ \text{Wh/day} = \mathbf{35.895\ kWh/day}$$

16-110 A building at 40º N latitude has double pane heat absorbing type windows that are equipped with light colored venetian blinds. The total solar heat gains of the building through the south windows at solar noon in April for the cases of with and without the blinds are to be determined.

Assumptions The calculations are performed for an "average" day in April, and may vary from location to location.

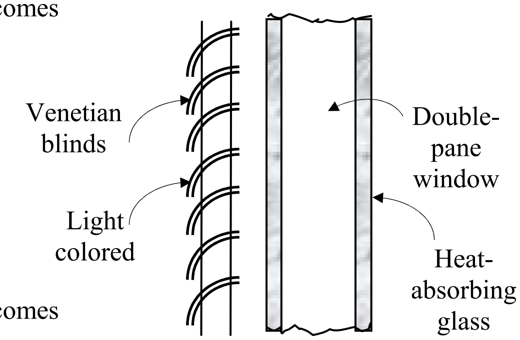
Properties The shading coefficient of a double pane heat absorbing type windows is SC = 0.58 (Table 11-5). It is given to be SC = 0.30 in the case of blinds. The solar radiation incident at a South-facing surface at 12:00 noon in April is 559 W/m^2 (Table 11-4).

Analysis The solar heat gain coefficient (SHGC) of the windows without the blinds is determined from Eq.16-40 to be

$$\text{SHGC} = 0.87\times\text{SC} = 0.87\times0.58 = 0.5046$$

Then the rate of solar heat gain through the window becomes

$$\begin{aligned}\dot{Q}_{\text{solar gain, no blinds}} &= SHGC\times A_{\text{glazing}}\times\dot{q}_{\text{solar, incident}}\\ &= 0.5046(130\text{ m}^2)(559\text{ W/m}^2)\\ &= \mathbf{36{,}670\ W}\end{aligned}$$



In the case of windows equipped with venetian blinds, the SHGC and the rate of solar heat gain become

$$\text{SHGC} = 0.87\times\text{SC} = 0.87\times0.30 = 0.261$$

Then the rate of solar heat gain through the window becomes

$$\begin{aligned}\dot{Q}_{\text{solar gain, no blinds}} &= SHGC\times A_{\text{glazing}}\times\dot{q}_{\text{solar, incident}}\\ &= 0.261(130\text{ m}^2)(559\text{ W/m}^2)\\ &= \mathbf{18{,}970\ W}\end{aligned}$$

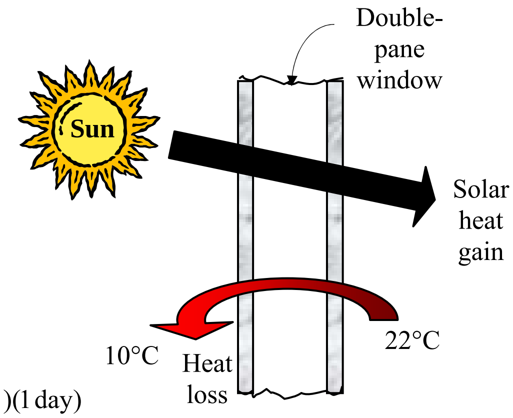
Discussion Note that light colored venetian blinds significantly reduce the solar heat, and thus air-conditioning load in summers.

16-111 A house has double door type windows that are double pane with 6.4 mm of air space and aluminum frames and spacers. It is to be determined if the house is losing more or less heat than it is gaining from the sun through an east window in a typical day in January.

Assumptions **1** The calculations are performed for an "average" day in January. **2** Solar data at 40° latitude can also be used for a location at 39° latitude.

Properties The shading coefficient of a double pane window with 3-mm thick clear glass is SC = 0.88 (Table 16-21). The overall heat transfer coefficient for double door type windows that are double pane with 6.4 mm of air space and aluminum frames and spacers is 4.55 W/m^2.°C (Table 16-19). The total solar radiation incident at an East-facing surface in January during a typical day is 1863 Wh/m^2 (Table 16-20).

Analysis The solar heat gain coefficient (SHGC) of the windows is determined from Eq.16-40 to be



$$\text{SHGC} = 0.87\times\text{SC} = 0.87\times0.88 = 0.7656$$

Then the solar heat gain through the window per unit area becomes

$$\begin{aligned} Q_{\text{solar gain}} &= SHGC \times A_{\text{glazing}} \times q_{\text{solar, daily total}} \\ &= 0.7656(1\ \text{m}^2)(1863\ \text{Wh/m}^2) \\ &= \mathbf{1426\ Wh} = \mathbf{1.426\ kWh} \end{aligned}$$

The heat loss through a unit area of the window during a 24-h period is

$$\begin{aligned} Q_{\text{loss, window}} &= \dot{Q}_{\text{loss, window}}\Delta t = U_{\text{window}} A_{\text{window}} (T_i - T_{0,\text{ave}})(1\ \text{day}) \\ &= (4.55\ \text{W/m}^2\cdot°\text{C})(1\ \text{m}^2)(22-10)°\text{C}(24\ \text{h}) \\ &= \mathbf{1310\ Wh} = \mathbf{1.31\ kWh} \end{aligned}$$

Therefore, the house is loosing **less** heat than it is gaining through the East windows during a typical day in January.

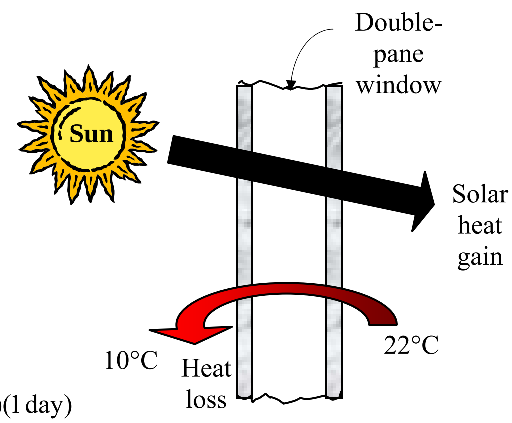
16-112 A house has double door type windows that are double pane with 6.4 mm of air space and aluminum frames and spacers. It is to be determined if the house is losing more or less heat than it is gaining from the sun through a South window in a typical day in January.

Assumptions **1** The calculations are performed for an "average" day in January. **2** Solar data at 40° latitude can also be used for a location at 39° latitude.

Properties The shading coefficient of a double pane window with 3-mm thick clear glass is SC = 0.88 (Table 16-21). The overall heat transfer coefficient for double door type windows that are double pane with 6.4 mm of air space and aluminum frames and spacers is 4.55 $W/m^2.°C$ (Table 16-19). The total solar radiation incident at a South-facing surface in January during a typical day is 5897 Wh/m^2 (Table 16-20).

Analysis The solar heat gain coefficient (SHGC) of the windows is determined from Eq. 16-40 to be

$$\text{SHGC} = 0.87\times\text{SC} = 0.87\times0.88 = 0.7656$$



Then the solar heat gain through the window per unit area becomes

$$\begin{aligned} Q_{\text{solar gain}} &= SHGC\times A_{\text{glazing}}\times q_{\text{solar, daily total}} \\ &= 0.7656(1\text{ m}^2)(5897\text{ Wh/m}^2) \\ &= \mathbf{4515\ Wh = 4.515\ kWh} \end{aligned}$$

The heat loss through a unit area of the window during a 24-h period is

$$\begin{aligned} Q_{\text{loss, window}} &= \dot{Q}_{\text{loss, window}}\Delta t = U_{\text{window}}A_{\text{window}}(T_i - T_{0,\text{ave}})(1\text{ day}) \\ &= (4.55\text{ W/m}^2\cdot°\text{C})(1\text{ m}^2)(22-10)°\text{C}(24\text{ h}) \\ &= \mathbf{1310\ Wh = 1.31\,kWh} \end{aligned}$$

Therefore, the house is **loosing** much less heat than it is gaining through the South windows during a typical day in January.

16-113E A house has 1/8-in thick single pane windows with aluminum frames on a West wall. The rate of net heat gain (or loss) through the window at 3 PM during a typical day in January is to be determined.

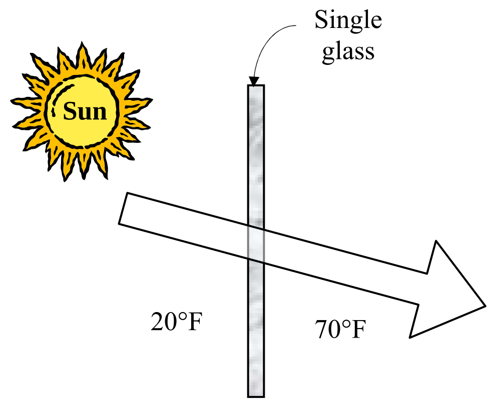
Assumptions **1** The calculations are performed for an "average" day in January. **2** The frame area relative to glazing area is small so that the glazing area can be taken to be the same as the window area.

Properties The shading coefficient of a 1/8-in thick single pane window is SC = 1.0 (Table 16-21). The overall heat transfer coefficient for 1/8-in thick single pane windows with aluminum frames is 6.63 W/m^2.°C = 1.17 Btu/h.ft^2.°F (Table 9-6). The total solar radiation incident at a West-facing surface at 3 PM in January during a typical day is 557 W/m^2 = 177 Btu/h.ft^2 (Table 16-20).

Analysis The solar heat gain coefficient (SHGC) of the windows is determined from Eq. 16-40 to be

$$\text{SHGC} = 0.87\times\text{SC} = 0.87\times1.0 = 0.87$$

The window area is: $A_{\text{window}} = (9\text{ ft})(15\text{ ft}) = 135\text{ ft}^2$



Then the rate of solar heat gain through the window at 3 PM becomes

$$\begin{aligned}\dot{Q}_{\text{solar gain, 3 PM}} &= SHGC\times A_{\text{glazing}}\times\dot{q}_{\text{solar, 3 PM}}\\ &= 0.87(135\text{ ft}^2)(177\text{ Btu/h.ft}^2)\\ &= 20{,}789\text{ Btu/h}\end{aligned}$$

The rate of heat loss through the window at 3 PM is

$$\begin{aligned}\dot{Q}_{\text{loss, window}} &= U_{\text{window}}A_{\text{window}}(T_i - T_0)\\ &= (1.17\text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F})(135\text{ ft}^2)(70-20)°\text{F}\\ &= 7898\text{ Btu/h}\end{aligned}$$

The house will be gaining heat at 3 PM since the solar heat gain is larger than the heat loss. The rate of net heat gain through the window is

$$\dot{Q}_{\text{net}} = \dot{Q}_{\text{solar gain, 3 PM}} - \dot{Q}_{\text{loss, window}} = 20{,}789 - 7898 = \mathbf{12{,}890\text{ Btu/h}}$$

Discussion The actual heat gain will be less because of the area occupied by the window frame.

16-114 A building located near 40º N latitude has equal window areas on all four sides. The side of the building with the highest solar heat gain in summer is to be determined.

Assumptions The shading coefficients of windows on all sides of the building are identical.

Analysis The reflective films should be installed on the side that receives the most incident solar radiation in summer since the window areas and the shading coefficients on all four sides are identical. The incident solar radiation at different windows in July are given to be (Table 16-20)

Month	Time	The daily total solar radiation incident on the surface, Wh/m^2			
		North	East	South	West
July	Daily total	1621	4313	2552	4313

Therefore, the reflective film should be installed on the **East** or **West** windows (instead of the South windows) in order to minimize the solar heat gain and thus the cooling load of the building.

Infiltration Heat Load and Weatherizing

16-115C The uncontrolled entry of outside air into a building through unintentional openings is called *infiltration*. It differs from ventilation in that ventilation is intentional and controlled whereas infiltration is unintentional and uncontrolled. Infiltration increases the heating loss in winter and the cooling load in summer since the air entering must be heated in winter and cooled in summer. Therefore, the warm air leaving the house represents *energy loss*. This is also the case for cool air leaving in summer since some electricity is used to cool that air.

16-116C Air infiltration rate of a building can be determined by direct measurements by (1) *injecting a tracer gas* into a building and observing the decline of its concentration with time, or (2) *pressurizing the building* to 10 to 75 Pa gage pressure by a large fan mounted on a door or window, and measuring the air flow required to maintain a specified indoor-outdoor pressure difference. The larger the airflow to maintain a pressure difference, the more the building may leak. Sulfur hexafluoride (SF_6) is commonly used as a tracer gas because it is inert, nontoxic, and easily detectable at concentrations as low as 1 part per billion. Pressurization testing is easier to conduct, and thus preferable to tracer gas testing.

The *design infiltration rate* determined at design conditions is used to size a heating or cooling equipment where as the *seasonal average infiltration rate* is used to properly estimate the seasonal energy consumption for heating or cooling.

16-117C The infiltration unit ACH (air changes per hour) is defined as

$$\text{ACH} = \frac{\text{Flow rate of outdoor air into the building (per hour)}}{\text{Internal volume of the building}} = \frac{\dot{V}\ (\text{m}^3/\text{h})}{V\ (\text{m}^3)}$$

Therefore, the quantity ACH represents the number of building volumes of outdoor air that infiltrates (and eventually exfiltrates) per hour.

Too low and too high values of ACH should be avoided since too little fresh air will cause health and comfort problems such as the sick-building syndrome, which is experienced in super airtight buildings, and too much of it will waste energy. Therefore, the rate of fresh air supply should be just enough to maintain the indoor air quality at an acceptable level.

16-118C The energy of the air vented out from the kitchens and bathrooms can be saved by installing an *air-to-air heat exchanger* (also called "economizer" or "heat recuperator") that transfers the heat from the exhausted stale air to the incoming fresh air without any mixing. Such heat exchangers are commonly used in superinsulated houses, but the benefits of such heat exchangers must be weighed against the cost and complexity of their installation.

16-119C The latent heat load of infiltration **is not** necessarily zero when the relative humidity of the hot outside air in summer is the same as that of inside air since $\dot{Q}_{\text{infiltration, latent}} = \rho_o h_{fg}(ACH)(V_{\text{building}})(w_i - w_o)$ where $w_i - w_o$ is the humidity ratio difference between the indoor and outdoor air, and w is higher at higher temperatures for the same relative humidity.

16-120C The latent heat load of infiltration **is** necessarily zero when the humidity ratio w of the hot outside air in summer is the same as that of inside air since $\dot{Q}_{\text{infiltration, latent}} = \rho_o h_{fg}(ACH)(V_{\text{building}})(w_i - w_o) = 0$ when $w_i = w_o$.

16-121C Some practical ways of preventing infiltration in homes are (1) *caulking* that can be applied with a caulking gun inside and outside where two stationary surfaces such as a wall and a window frame meet and (2) *weather-stripping* with a narrow piece of metal, vinyl, rubber, felt or foam that seals the contact area between the fixed and movable sections of a joint.

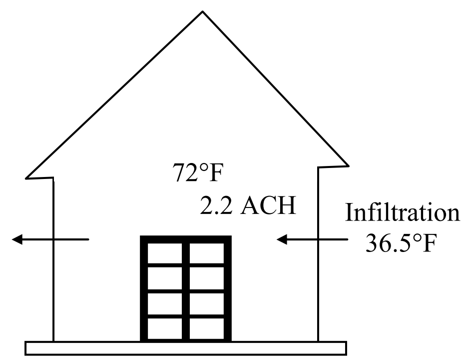
16-122C Yes, it is true that the infiltration rate and infiltration losses can be reduced by using radiant panel heaters since the air temperature can be lowered without sacrificing comfort, and the lower the temperature difference between the indoors and the outdoors the lower the infiltration loss. It is also true that radiant panels will increase the heat losses through the wall and the roof by conduction as a result of increased surface temperature if this turns out to be the case since heat conduction through the wall is proportional to the temperature difference across the wall.

16-123E A winterizing project is to reduce the infiltration rate of a house from 2.2 ACH to 1.1 ACH. The resulting cost savings are to be determined.

Assumptions **1** The house is maintained at 72°F at all times. **2** The latent heat load during the heating season is negligible. **3** The infiltrating air is heated to 72°F before it exfiltrates.

Properties The gas constant of air is 0.3704 psia.ft^3/lbm·R (Table A-1E). The specific heat of air at room temperature is 0.24 Btu/lbm·°F (Table A-11E).

Analysis The density of air at the outdoor conditions is



$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5 \text{ psia}}{(0.3704 \text{ psia.ft}^3\text{/lbm.R})(496.5 \text{ R})} = 0.0734 \text{ lbm/ft}^3$$

The volume of the building is

$$\mathbf{V}_{\text{building}} = (\text{Floor area})(\text{Height}) = (3000 \text{ ft}^2)(9 \text{ ft}) = 27{,}000 \text{ ft}^3$$

The reduction in the infiltration rate is 2.2 – 1.1 = 1.1 ACH.

The sensible infiltration heat load corresponding to it is

$$\begin{aligned}\dot{Q}_{\text{infiltration, saved}} &= \rho_o c_p (ACH_{\text{saved}})(\mathbf{V}_{\text{building}})(T_i - T_o) \\ &= (0.0734 \text{ lbm/ft}^3)(0.24 \text{ Btu/lbm.°F})(1.1\text{/h})(27{,}000 \text{ ft}^3)(72 - 36.5)\text{°F} \\ &= 18{,}573 \text{ Btu/h} = 0.18573 \text{ therm/h}\end{aligned}$$

since 1 therm = 100,000 Btu. The number of hours during a six month period is 6×30×24 = 4320 h. Noting that the furnace efficiency is 0.65 and the unit cost of natural gas is $1.20/therm, the energy and money saved during the 6-month period are

$$\begin{aligned}\text{Energy savings} &= (\dot{Q}_{\text{infiltration, saved}})(\text{No. of hours per year})/\text{Efficiency} \\ &= (0.18573 \text{ therm/h})(4320 \text{ h/year})/0.65 \\ &= 1234 \text{ therms/year}\end{aligned}$$

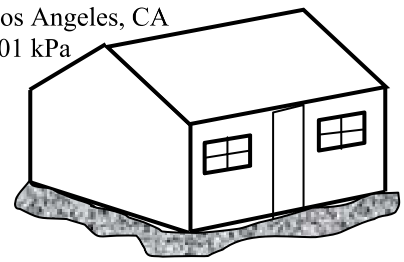
$$\begin{aligned}\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy}) \\ &= (1234 \text{ therms/year})(\$1.20\text{/therm}) \\ &= \mathbf{\$1480/year}\end{aligned}$$

Therefore, reducing the infiltration rate by one-half will reduce the heating costs of this homeowner by $1480 per year.

16-124 Two identical buildings in Los Angeles and Denver have the same infiltration rate. The ratio of the heat losses by infiltration at the two cities under identical conditions is to be determined.

Assumptions Both homes are identical and both are subjected to the same conditions except the atmospheric conditions.

Analysis The sensible infiltration heat loss is given as



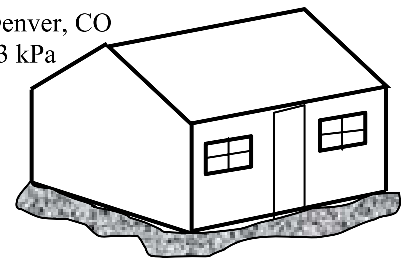
$$\dot{Q}_{\text{infiltration}} = \dot{m}_{\text{air}} c_p (T_i - T_o) = \rho_{o,\text{air}} (ACH)(\boldsymbol{V}_{\text{building}}) c_p (T_i - T_o)$$

Therefore, the infiltration heat loss is proportional to the density of air, and thus the ratio of infiltration heat losses at the two cities is simply the densities of outdoor air at those cities,

$$\text{Infiltration heat loss ratio} = \frac{\dot{Q}_{\text{infiltration, Los Angeles}}}{\dot{Q}_{\text{infiltration, Denver}}} = \frac{\rho_{o,\text{ air, Los Angeles}}}{\rho_{o,\text{ air, Denver}}}$$

$$= \frac{(P_0 / RT_0)_{\text{Los Angeles}}}{(P_0 / RT_0)_{\text{Denver}}} = \frac{P_{o,\text{ Los Angeles}}}{P_{0,\text{ Denver}}}$$

$$= \frac{101\,\text{kPa}}{83\,\text{kPa}} = \mathbf{1.22}$$



Therefore, the infiltration heat loss in Los Angeles will be 22% higher than that in Denver under identical conditions.

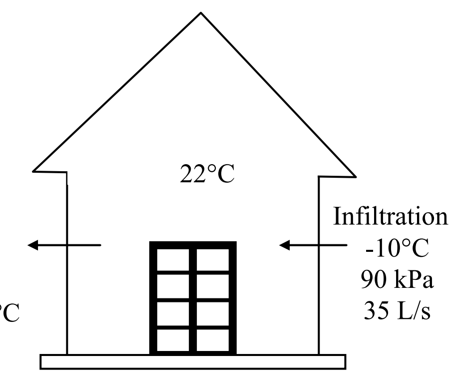
16-125 Outdoor air at -10°C and 90 kPa enters the building at a rate of 35 L/s when the indoors is maintained at 22°C. The rate of sensible heat loss from the building due to infiltration is to be determined.

Assumptions **1** The house is maintained at 22°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 22°C before it exfiltrates.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg·K (Table A-1). The specific heat of air at room temperature is c_p = 1.0 kJ/kg·°C (Table A-11).

Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{90\,\text{kPa}}{(0.287\,\text{kPa.m}^3/\text{kg.K})(-10 + 273\,\text{K})} = 1.19\,\text{kg/m}^3$$



Then the sensible infiltration heat load corresponding to an infiltration rate of 35 L/s becomes

$$\dot{Q}_{\text{infiltration}} = \rho_o \dot{\boldsymbol{V}}_{air} c_p (T_i - T_o)$$

$$= (1.19\,\text{kg/m}^3)(0.035\,\text{m}^3/\text{s})(1.0\,\text{kJ/kg.°C})[22 - (-10)]\text{°C}$$

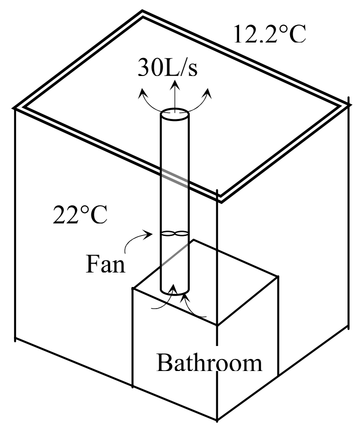
$$= \mathbf{1.335\,kW}$$

Therefore, sensible heat will be lost at a rate of 1.335 kJ/s due to infiltration.

16-126 The ventilating fan of the bathroom of an electrically heated building in San Francisco runs continuously. The amount and cost of the heat "vented out" per month in winter are to be determined.

Assumptions **1** We take the atmospheric pressure to be 1 atm = 101.3 kPa since San Francisco is at sea level. **2** The house is maintained at 22°C at all times. **3** The infiltrating air is heated to 22°C before it exfiltrates.

Properties The gas constant of air is R = 0.287 kPa.m^3/kg·K (Table A-1). The specific heat of air at room temperature is c_p = 1.0 kJ/kg·°C (Table A-11).



Analysis The density of air at the indoor conditions of 1 atm and 22°C is

$$\rho_o = \frac{P_o}{RT_o} = \frac{(101.3\text{ kPa})}{(0.287\text{ kPa.m}^3/\text{kg.K})(22+273\text{ K})} = 1.20\text{ kg/m}^3$$

Then the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (1.20\text{ kg/m}^3)(0.030\text{ m}^3/\text{s}) = 0.036\text{ kg/s}$$

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 12.2°C, this corresponds to energy loss at a rate of

$$\begin{aligned}\dot{Q}_{\text{loss by fan}} &= \dot{m}_{\text{air}} c_p (T_{\text{indoors}} - T_{\text{outdoors}}) \\ &= (0.036\text{ kg/s})(1.005\text{ kJ/kg.°C})(22-12.2)\text{°C} = 0.355\text{ kJ/s} = 0.355\text{ kW}\end{aligned}$$

Then the amount and cost of the heat "vented out" per month (1 month = 30×24 = 720 h) becomes

$$\text{Energy loss} = \dot{Q}_{\text{loss by fan}}\Delta t = (0.355\text{ kW})(720\text{ h/month}) = \mathbf{256\text{ kWh/month}}$$

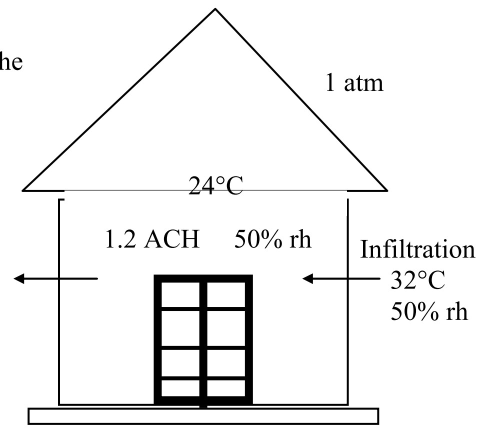
$$\text{Money loss} = (\text{Energy loss})(\text{Unit cost of energy}) = (256\text{ kWh/month})(\$0.09/\text{kWh}) = \mathbf{\$23.0/month}$$

Discussion Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used with care.

16-127 The infiltration rate of a building is estimated to be 1.2 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at 5°C. **4** The effect of water vapor on air density is negligible.

Properties The gas constant and the specific heat of air are R = 0.287 kPa.m^3/kg.K and c_p = 1.0 kJ/kg·°C (Tables A-1 and A-15). The heat of vaporization of water at 5°C is $h_{fg} = h_{fg@5°C} = 2490\,\text{kJ/kg}$ (Table A-9). The properties of the ambient and room air are determined from the psychrometric chart to be



$$\left.\begin{array}{l} T_{\text{ambient}} = 32^\circ\text{C} \\ \phi_{\text{ambient}} = 50\% \end{array}\right\} \omega_{\text{ambient}} = 0.0150\,\text{kg/kg dryair}$$

$$\left.\begin{array}{l} T_{\text{room}} = 24^\circ\text{C} \\ \phi_{\text{room}} = 50\% \end{array}\right\} \omega_{\text{room}} = 0.0093\,\text{kg/kg dryair}$$

Analysis Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 0.8 times every hour, the air will enter the room at a mass flow rate of

$$\rho_{\text{ambient}} = \frac{P_o}{RT_o} = \frac{101.325\,\text{kPa}}{(0.287\,\text{kPa.m}^3/\text{kg.K})(32+273\,\text{K})} = 1.16\,\text{kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_{\text{ambient}} V_{\text{room}} ACH = (1.16\,\text{kg/m}^3)(20\times 13\times 3\,\text{m}^3)(1.2\,\text{h}^{-1}) = 1085\,\text{kg/s} = 0.3016\,\text{kg/s}$$

Then the sensible, latent, and total infiltration heat loads of the room are determined to be

$$\dot{Q}_{\text{infiltration,sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (0.3016\,\text{kg/s})(1.0\,\text{kJ/kg.°C})(32-24)\text{°C} = \mathbf{2.41\,kW}$$

$$\dot{Q}_{\text{infiltration,latent}} = \dot{m}_{\text{air}} (\omega_{\text{ambient}} - \omega_{\text{room}}) h_{fg} = (0.3016\,\text{kg/s})(0.0150-0.0093)\,(2490\,\text{kJ/kg}) = \mathbf{4.28\,kW}$$

$$\dot{Q}_{\text{infiltration, total}} = \dot{Q}_{\text{infiltration,sensible}} + \dot{Q}_{\text{infiltration,latent}} = 2.41 + 4.28 = \mathbf{6.69\,kW}$$

Discussion The specific volume of the dry air at the ambient conditions could also be determined from the psychrometric chart at ambient conditions.

Annual Energy Consumption

16-128C Yes, it is possible for a building in a city to have a higher peak heating load but a lower energy consumption for heating in winter than an identical building in another city. This will be the case for a city with severe but relatively short heating seasons.

16-129C No, we cannot determine the annual energy consumption of a building for heating by simply multiplying the design heating load of the building by the number of hours in the heating season. This is because the design heating load represents the heat loss under extreme conditions, not average conditions.

16-130C No, as the manager of a large commercial building, I would not lower the thermostat setting in winter and raise it in summer by a few degrees. Although this practice will save energy and thus money, it may cost much more in reduced productivity.

16-131C Taking the balance point temperature to be 18°C, as is commonly done, the number of heating degree-days for a winter day during which the average outdoor temperature was 10°C, and never went above 18°C, is determined to be

$$DD_{\text{1 day}} = (T_{\text{balance point}} - T_{\text{daily average}})(\text{1 day}) = (18 - 10)°\text{C}(\text{1 day}) = \mathbf{8°C\text{-}day}$$

16-132C Taking the balance point temperature to be 18°C, as is commonly done, the number of heating degree-days for a winter month during which the average outdoor temperature was 12°C, and never rose above 18°C, is determined to be

$$DD_{\text{1 mont}} = (T_{\text{balance point}} - T_{\text{monthly average}})(\text{1 month}) = (18 - 12)°\text{C}(\text{30 days}) = \mathbf{180°C\text{-}day}$$

16-133C The °C-days are based on temperature differences, and $\Delta T(°\text{F}) = 1.8\Delta T(°\text{C})$ for temperature differences. Therefore, we should not add 32 to the result.

16-134C The outdoor temperature above which no heating is required is called the *balance point temperature* T_{balance}. The balance-point temperature is used in the determination of degree-days instead of the actual thermostat setting of a building since the internal heat generated by people, lights, and appliances in occupied buildings as well as the heat gain from the sun during the day, $\dot{Q}_{\text{gain}}$, will be sufficient to compensate for the heat losses from the building until the outdoor temperature drops below T_{balance}.

16-135C It is proper to use the degree-day method to determine the annual energy consumption of a building under relatively *steady* conditions. The method is based on constant indoor conditions during the heating or cooling season, and it assumes the efficiency of the heating or cooling equipment is not affected by the variation of outdoor temperature. These conditions will be closely approximated if all the thermostats in a building are set at the same temperature at the beginning of a heating or cooling season, and are never changed, and a seasonal average efficiency is used (rather than the full-load or design efficiency) for the furnaces or coolers.

16-136 A person offers to his roommate in Syracuse, New York, to pay the heating bills during the upcoming year (starting January 1st) if he pays the heating bills for the current calendar year until Dec. 31. It is to be determined if this is a good offer.

Assumptions The calculations are performed for an "average" year. **2** The time value of money is not considered.

Properties The annual heating degree-days of Syracuse, NY, is 6756°F-days (Table 16-5). The monthly distribution of degree-days are 6, 28, 132, 415, 744, 1153, 1271, 1140, 1004, 570, 248, and 45°F-days for July through June, respectively.

Analysis It makes sense to accept this offer if the cost of heating before December 31st is less than the cost of heating after December 31st. The amount and cost of energy consumption of a building for heating is proportional to the heating degree days. For Syracuse, we have

$$DD_{\text{heating, before Dec. 31}} = 6 + 28 + 132 + 415 + 744 + 1153 = 2478°\text{F-days} \quad (2478/6756 = 0.367)$$

$$DD_{\text{heating, after Dec. 31}} = 1271 + 1140 + 1004 + 570 + 248 + 45 = 4278°\text{F-days} \quad (4278/6756 = 0.633)$$

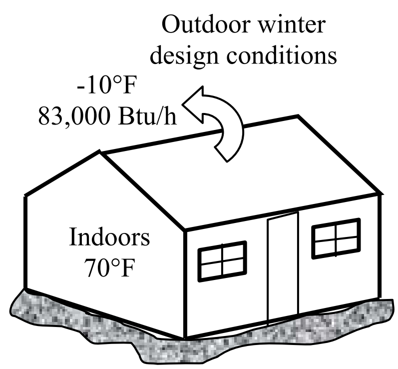
This is clearly a **good offer** for the roommate since 63.3% of the heating load occurs after December 31st, and the proposer is offering to pay for it. Therefore, the offer should be accepted.

16-137E A house whose design heat load is 83,000 Btu/h is heated by a high-efficiency natural gas furnace. The annual gas consumption of this house and its cost are to be determined.

Assumptions **1**The house is maintained at 70°F at all times during the heating season. **2** The calculations are performed for an "average" year.

Properties The annual heating degree-days of Billing, Montana, is 7049°F-days (Table 16-5). The winter design temperature of Billing is given to be -10°F.

Analysis The fuel (natural gas) consumption rate of the house for heating at design conditions is



$$\dot{Q}_{\text{design}} = \frac{\dot{Q}_{\text{design, load}}}{\eta_{\text{heating}}} = \frac{83{,}000\ \text{Btu/h}}{0.95} = 87{,}368\ \text{Btu/h} = 0.874\ \text{therm/h}$$

Then the annual natural gas usage of the house and its cost can be determined from Eq. 16-48 to be

$$\begin{aligned}\text{Annual fuel consumption} = Q_{\text{heating,year}} &= \frac{DD_{\text{heating}}}{(T_i - T_o)_{\text{design}}}\dot{Q}_{\text{design}} \\ &= \frac{7049°\text{F-day}}{[70-(-10)]°\text{F}}\left(\frac{24\ \text{h}}{1\ \text{day}}\right)(0.874\ \text{therm/h}) \\ &= \mathbf{1848\ therms/year}\end{aligned}$$

and

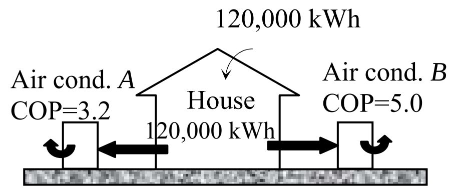
$$\begin{aligned}\text{Annual fuel cost} &= (\text{Annual fuel consumption})(\text{Unit cost of fuel}) \\ &= (1848\ \text{therms/year})(\$1.10/\text{therm}) = \mathbf{\$2033/year}\end{aligned}$$

Therefore, it will cost $2033 per year to heat this house during an average year.

16-138 A decision is to be made between a cheaper but inefficient and an expensive but efficient air-conditioner for a building.

Assumptions The two air conditioners are comparable in all aspects other than the initial cost and the efficiency.

Analysis The unit that will cost less during its lifetime is a better buy. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period. The energy and cost savings of the more efficient air conditioner in this case is



$$\begin{aligned}\text{Energy savings} &= (\text{Annual energy usage of A}) - (\text{Annual energy usage of B})\\ &= (\text{Annual cooling load})(1/\text{COP}_\text{A} - 1/\text{COP}_\text{B})\\ &= (120{,}000\text{ kWh/year})(1/3.2 - 1/5.0)\\ &= 13{,}500\text{ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy})\\ &= (13{,}500\text{ kWh/year})(\$0.10/\text{kWh}) = \mathbf{\$1350/year}\end{aligned}$$

The installation cost difference between the two air-conditioners is

Cost difference = Cost of B – cost of A = 7000 – 5500 = $1500

Therefore, the more efficient air-conditioner B will pay for the $1500 cost differential in this case in about 1 year.

Discussion A cost conscious consumer will have no difficulty in deciding that the more expensive but more efficient air-conditioner B is clearly a better buy in this case since air conditioners last at least 15 years. But the decision would not be so easy if the unit cost of electricity at that location was much less than $0.10/kWh, or if the annual air-conditioning load of the house was much less than 120,000 kWh.

16-139 An industrial facility is to replace its 40-W standard fluorescent lamps by their 34-W high efficiency counterparts. The amount of energy and money that will be saved a year as a result of switching to the high efficiency fluorescent lamps as well as the simple payback period are to be determined.

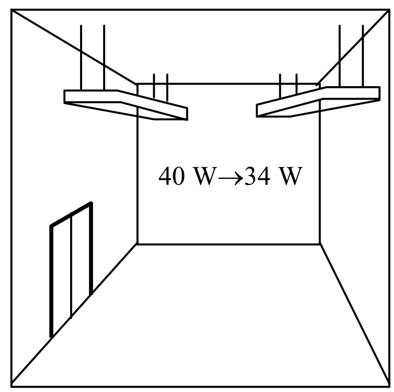
Analysis The reduction in the total electric power consumed by the lighting as a result of switching to the high efficiency fluorescent is

Wattage reduction = (Wattage reduction per lamp)(Number of lamps)
= (40 - 34 W/lamp)(700 lamps)
= 4200 W

Then using the relations given earlier, the energy and cost savings associated with the replacement of the high efficiency fluorescent lamps are determined to be

Energy Savings = (Total wattage reduction)(Ballast factor)(Operating hours)
= (4.2 kW)(1.1)(2800 h/year)
= **12,936 kWh/year**

Cost Savings = (Energy savings)(Unit electricity cost)
= (12,936 kWh/year)($0.08/kWh)
= **$1035**



The implementation cost of this measure is simply the extra cost of the energy efficient fluorescent bulbs relative to standard ones, and is determined to be

Implementation Cost = (Cost difference of lamps)(Number of lamps)
= [($2.26-$1.77)/lamp](700 lamps)
= $343

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$343}{\$1035\text{ / year}} = \mathbf{0.33\ year}\ \ (4.0\text{ months})$$

Discussion Note that if all the lamps were burnt out today and are replaced by high-efficiency lamps instead of the conventional ones, the savings from electricity cost would pay for the cost differential in about 4 months. The electricity saved will also help the environment by reducing the amount of CO_2, CO, NO_x, etc. associated with the generation of electricity in a power plant.

16-140 The lighting energy consumption of a storage room is to be reduced by installing motion sensors. The amount of energy and money that will be saved as a result of installing motion sensor as well as the simple payback period are to be determined.

Assumptions The electrical energy consumed by the ballasts is negligible.

Analysis The plant operates 12 hours a day, and thus currently the lights are on for the entire 12 hour period. The motion sensors installed will keep the lights on for 3 hours, and off for the remaining 9 hours every day. This corresponds to a total of 9×365 = 3285 off hours per year. Disregarding the ballast factor, the annual energy and cost savings become

Energy Savings = (Number of lamps)(Lamp wattage)(Reduction of annual operating hours)

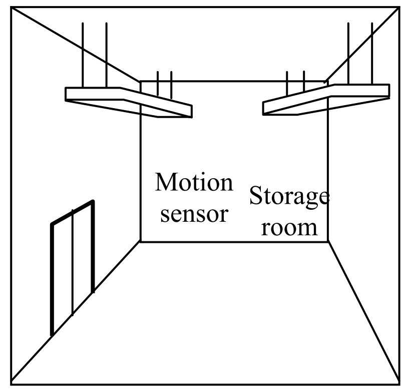
= (24 lamps)(60 W/lamp)(3285 hours/year)

= **4730 kWh/year**

Cost Savings = (Energy Savings)(Unit cost of energy)

= (5,203 kWh/year)($0.08/kWh)

= **$378/year**



The implementation cost of this measure is the sum of the purchase price of the sensor plus the labor,

Implementation Cost = Material + Labor = $32 + $40 = $72

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$72}{\$378\ /\ \text{year}}$$

$$= \mathbf{0.19\ year}\ \ (2.3\ \text{months})$$

Therefore, the motion sensor will pay for itself in about 2 months.

16-141 The existing manual thermostats of an office building are to be replaced by programmable ones to reduce the heating costs by lowering the temperature setting shortly before closing at 6 PM and raising it shortly before opening at 8 AM. The annual energy and cost savings as result of installing programmable thermostats as well as the simple payback period are to be determined.

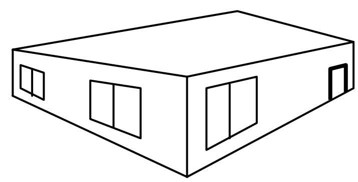
Assumptions **1** The ambient temperature remains below 7.2ºC during the entire heating season (Nov.-Apr). This assumption will most likely be violated some days, and thus the result is optimistic. **2** The balance point temperature is 18ºC so that no heating is required at temperatures above 18ºC.

Properties The annual heating degree-days for Reno, Nevada is given to be 3346°C-days.

Analysis The energy usage for heating is proportional to degree days, and the reduction in the degree days due to lowering the thermostat setting to 7.2ºC from about 6 PM to 8 AM for 10 h for everyday during the heating season for 180 days is

$$DD_{\text{reduction}} = (22 - 7.2)\text{ºC}(10\text{ h/day})(180\text{ days}) = 32{,}614\text{ºC-h} = 1359\text{ºC-day}$$

which is



$$\text{Reduction fraction} = \frac{DD_{\text{reduction}}}{DD_{\text{annual}}} = \frac{1359\text{°C - days}}{3346\text{°C - days}} = 0.406 \text{ (or 40.6\%)}$$

Therefore, the energy usage for heating will be reduced by 40.6%. Then the reduction in the amount and cost of heating energy as a result of installing programmable thermostats become

Energy Savings = (Reduction fraction)(Annual heating energy usage)

= 0.406(3530 therms/year) **= 1433 therms/year**

and

Cost Savings = (Reduction fraction)(Annual heating bill) = 0.406($4060/year) = **$1648/year**

The total implementation cost of installation of 5 programmable thermostats is

Implementation Cost = 5×$325 = $1625

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$1625}{\$1648\text{ / year}} = \mathbf{0.986\ year}$$

Therefore, the programmable thermostats will pay for themselves during the first heating season.

16-142 A worn out standard motor is to be replaced by a high efficiency one. The amount of electrical energy and money savings as a result of installing the high efficiency motor instead of the standard one as well as the simple payback period are to be determined.

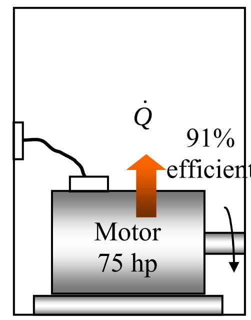
Assumptions The load factor of the motor remains constant at 0.75.

Analysis The electric power drawn by each motor and their difference can be expressed as

$$\dot{W}_{\text{electric in, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{standar}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{standard}}$$

$$\dot{W}_{\text{electric in, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{efficien}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{efficient}}$$

$$\begin{aligned}\text{Power savings} &= \dot{W}_{\text{electric in, standard}} - \dot{W}_{\text{electric in, efficient}} \\ &= (\text{Power rating})(\text{Load factor})[1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}}]\end{aligned}$$



where η_{standard} is the efficiency of the standard motor, and $\eta_{\text{efficient}}$ is the efficiency of the comparable high efficiency motor. Then the annual energy and cost savings associated with the installation of the high efficiency motor are determined to be

Energy Savings = (Power savings)(Operating Hours)

= (Power Rating)(Operating Hours)(Load Factor)($1/\eta_{\text{standard}}$- $1/\eta_{\text{efficient}}$)

= (75 hp)(0.746 kW/hp)(4,368 hours/year)(0.75)(1/0.91 - 1/0.954)

= **9,290 kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

= (9,290 kWh/year)($0.08/kWh)

= **$743/year**

The implementation cost of this measure consists of the excess cost the high efficiency motor over the standard one. That is,

Implementation Cost = Cost differential = $5,520 - $5,449 = $71

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$71}{\$743/\text{year}} = \mathbf{0.096\ year}\ (\text{or } 1.1\ \text{months})$$

Therefore, the high-efficiency motor will pay for its cost differential in about one month.

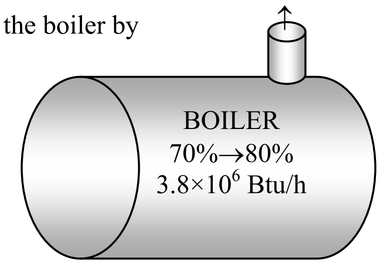
16-143E The combustion efficiency of a furnace is raised from 0.7 to 0.8 by tuning it up. The annual energy and cost savings as a result of tuning up the boiler are to be determined.

Assumptions The boiler operates at full load while operating.

Analysis The heat output of boiler is related to the fuel energy input to the boiler by

Boiler output = (Boiler input)(Combustion efficiency)

or $\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}}\eta_{\text{furnace}}$



The current rate of heat input to the boiler is given to be $\dot{Q}_{\text{in, current}} = 3.8\times10^6$ Btu/h . Then the rate of useful heat output of the boiler becomes

$$\dot{Q}_{\text{out}} = (\dot{Q}_{\text{in}}\eta_{\text{furnace}})_{\text{current}} = (3.8\times10^6 \text{ Btu/h})(0.7) = 2.66\times10^6 \text{ Btu/h}$$

The boiler must supply useful heat at the same rate after the tune up. Therefore, the rate of heat input to the boiler after the tune up and the rate of energy savings become

$$\dot{Q}_{\text{in, new}} = \dot{Q}_{\text{out}} / \eta_{\text{furnace, new}} = (2.66\times10^6 \text{ Btu/h})/0.8 = 3.325\times10^6 \text{ Btu/h}$$

$$\dot{Q}_{\text{in, saved}} = \dot{Q}_{\text{in, current}} - \dot{Q}_{\text{in, new}} = 3.8\times10^6 - 3.325\times10^6 = 0.475\times10^6 \text{ Btu/h}$$

Then the annual energy and cost savings associated with tuning up the boiler become

Energy Savings $= \dot{Q}_{\text{in, saved}}$ (Operation hours)

$= (0.475\times10^6$ Btu/h)(1500 h/year) = $\mathbf{712.5\times10^6}$ **Btu/yr**

Cost Savings = (Energy Savings)(Unit cost of energy)

$= (712.5\times10^6$ Btu/yr)($4.35 per 10^6 Btu) = **$3099/year**

Discussion Notice that tuning up the boiler will save $3099 a year, which is a significant amount. The implementation cost of this measure is negligible if the adjustment can be made by in-house personnel. Otherwise it is worthwhile to have an authorized representative of the boiler manufacturer to service the boiler twice a year.

16-144 The gas space heating of a facility is to be supplemented by air heated in a liquid-to-air heat exchanger of a compressor. The amount of money that will be saved by diverting the compressor waste heat into the facility during the heating season is to be determined.

Assumptions The atmospheric pressure at that location is 1 atm.

Analysis The mass flow rate of air through the liquid-to-air heat exchanger is

Mass flow rate of air = (Density of air)(Average velocity)(Flow area)
= (1.21 kg/m^3)(3 m/s)(1.0 m^2)
= 3.63 kg/s = 13,068 kg/h

Noting that the exit temperature of air is 52°C, the rate at which heat can be recovered (or the rate at which heat is transferred to air) is

Rate of Heat Recovery = (Mass flow rate of air)(Specific heat of air)(Temperature rise)
= (13,068 kg/h)(1.0 kJ/kg.°C)(52 – 20)°C
= 418,176 kJ/h

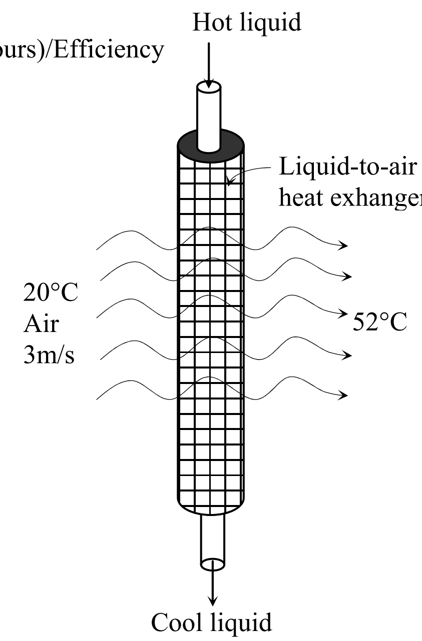
The number of operating hours of this compressor during the heating season is

Operating hours = (20 hours/day)(5 days/week)(26 weeks/year) = 2600 hours/year

Then the annual energy and cost savings become

Energy Savings = (Rate of Heat Recovery)(Annual Operating Hours)/Efficiency
= (418,176 kJ/h)(2600 h/year)/0.8
= 1,359,100,000 kJ/year
= 12,882 therms/year

Cost Savings = (Energy savings)(Unit cost of energy saved)
= (12,882 therms/year)($1.00/therm)
= $12,882/year



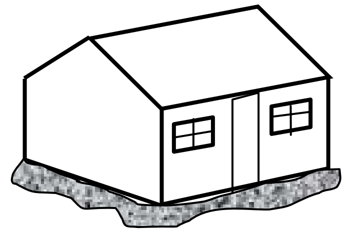
Discussion Notice that utilizing the waste heat from the compressor will save $12,882 per year from the heating costs. The implementation of this measure requires the installation of an ordinary sheet metal duct from the outlet of the heat exchanger into the building. The installation cost associated with this measure is relatively low. Several manufacturing facilities already have this conservation system in place. A damper is used to direct the air into the building in winter and to the ambient in summer. Combined compressor/heat-recovery systems are available in the market for both air-cooled (greater than 50 hp) water cooled (greater than 125 hp) systems.

16-145 An Atlanta family has moved to an identical house in Denver, CO where the fuel and electricity prices are the same. The annual heating cost of this family in their new house is to be determined.

Assumptions Calculations are performed for an average year.

Properties The annual heating degree-days are 2961°F-days for Atlanta, and 6283°F-days for Denver (Table 16-5).

Solution The heating cost is proportional to the energy consumption, which is proportional to the degree-days. The ratio of the degree-days in the two cities is



$$DD_{\text{ratio}} = \frac{\text{°C-days for Atlanta}}{\text{°C-day for Denver}} = \frac{6283\text{°F-days}}{2961\text{°F-days}} = 2.12$$

Therefore, the heating load will increase by factor of 2.12 in Denver. Then the annual heating cost of this new house in Denver becomes

Annual heating cost in Denver = 2.12×(Annual heating cost in Atlanta) = 2.12($600/yr) **= $1272**

16-146E The annual gas consumption and its cost for a house in Cleveland, Ohio with a design heat load of 65,000 Btu/h and a furnace efficiency of 90% are to be determined.

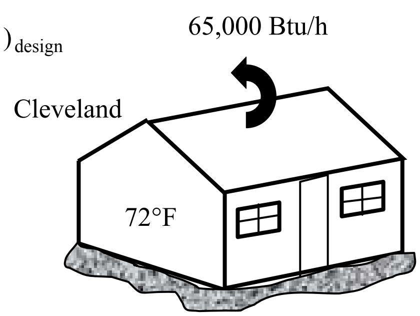
Assumptions The house is maintained at 72°F at all times during the heating season.

Properties The annual heating degree-days of Cleveland, Ohio is 6351°F-days (Table 16-5). The 97.5% winter design temperature of Cleveland is 5°F.

Analysis The overall heat loss coefficient K_{overall} of the building is determined from

$$\dot{Q}_{\text{design}} = UA\Delta T_{\text{design}} = UA(T_i - T_o)_{\text{design}} = K_{\text{overall}}(T_i - T_o)_{\text{design}}$$

Substituting,



$$K_{\text{overall}} = \frac{\dot{Q}_{\text{design}}}{(T_i - T_o)_{\text{design}}} = \frac{65{,}000\text{ Btu/h}}{(72-5)\text{°F}} = 970\text{ Btu/h.°F}$$

Then the annual gas consumption of the house for heating is determined to be

$$\begin{aligned}\text{Annual Gas Consumption} &= \frac{K_{\text{overall}}}{\eta_{\text{heating}}} DD_{\text{heating}} \\ &= \frac{970\text{ Btu/h.°F}}{0.90}(6351\text{°F-days})\left(\frac{24\text{ h}}{1\text{day}}\right)\left(\frac{1\text{ therm}}{100{,}000\text{ Btu}}\right) \\ &= 1643\text{ therms}\end{aligned}$$

Therefore, the house will consume 1643 therms of natural gas for heating.

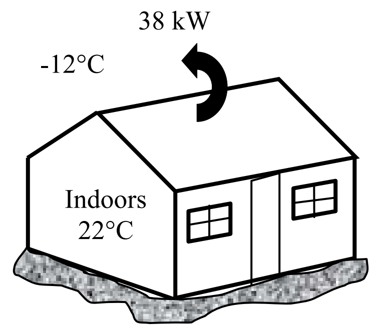
16-147 A house in Boise, Idaho is heated by electric resistance heaters. The amount of money the home owner will save if she lowers the thermostat from 22°C to 14°C every night in December is to be determined.

Assumptions **1** The house is maintained at 22°C during the day, and 14°C for 8 hours at night. **2** The efficiency of electric resistance heating system is 100%.

Properties The annual heating degree-days of Cleveland, Ohio is 6351°F-days (Table 16-5). The 97.5% winter design temperature of Boise is given to be -12°C.

Analysis The overall heat loss coefficient K_{overall} of the building is determined from

$$\dot{Q}_{\text{design}} = UA\Delta T_{\text{design}} = UA(T_{\text{i}} - T_o)_{\text{design}} = K_{\text{overall}}(T_{\text{i}} - T_o)_{\text{design}}$$



Substituting,

$$K_{\text{overall}} = \frac{\dot{Q}_{\text{design}}}{(T_{\text{i}} - T_o)_{\text{design}}} = \frac{38\text{ kW}}{[72-(-12)]°\text{C}} = 1.12\text{ kW/°C}$$

The rate at which energy is saved at night is

$$\dot{Q}_{\text{saved}} = UA\Delta T_{\text{reduction}} = K_{\text{overall}}\Delta T_{\text{reduction}}$$
$$= (1.12\text{ kW/°C})(22 - 14)°\text{C} = 8.96\text{ kW}$$

Then the energy and cost savings in December due to lowering the thermostat becomes

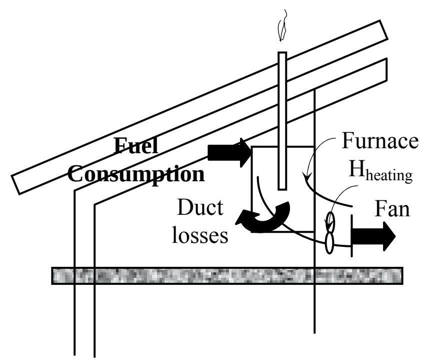
$$\text{Energy savings} = \dot{Q}_{\text{saved}}\Delta t = (8.96\text{ kW})(31\times 8\text{ h}) = \mathbf{2222\ kWh}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of eergy}) = (2222\text{ kWh})(\$0.06/\text{kWh}) = \mathbf{\$133}$$

Discussion Note that thermostat setback results in considerable savings in winter, and is commonly used in practice.

Review Problems

16-148 A decision is to be made between a cheaper but inefficient and an expensive but efficient natural gas heater for a house.



Assumptions The two heaters are comparable in all aspects other than the initial cost and efficiency.

Analysis Other things being equal, the logical choice is the heater that will cost less during its lifetime. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period.

The annual heating cost is given to be \$1100. Noting that the existing heater is 60% efficient, only 60% of that energy (and thus money) is delivered to the house, and the rest is wasted due to the inefficiency of the heater. Therefore, the monetary value of the heating load of the house is

Cost of useful heat = (60%)(Current annual heating cost) = 0.60×(\$1100/yr)=\$660/yr

This is how much it would cost to heat this house with a heater that is 100% efficient. For heaters that are less efficient, the annual heating cost is determined by dividing \$660 by the efficiency:

82% heater: Annual cost of heating = (Cost of useful heat)/Efficiency = (\$660/yr)/0.82 = \$805/yr

95% heater: Annual cost of heating = (Cost of useful heat)/Efficiency = (\$660/yr)/0.95 = \$695/yr

Annual cost savings with the efficient heater = 805 - 695 = \$110

Excess initial cost of the efficient heater = 2700 - 1600 = \$1100

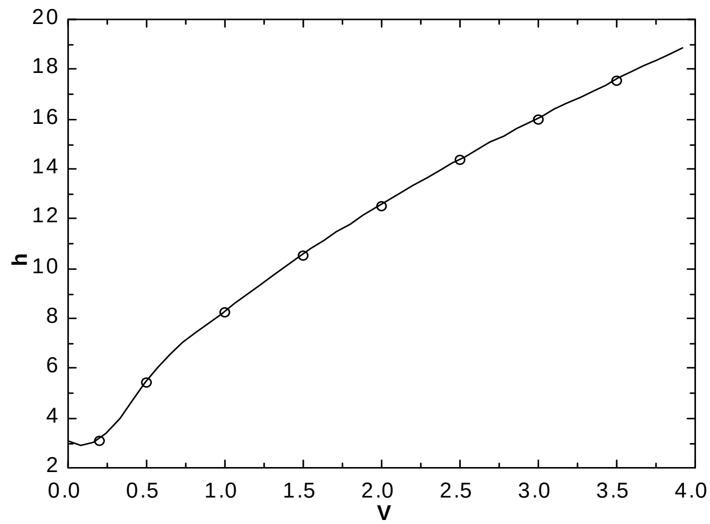
The simple payback period becomes

$$\text{Simple payback period} = \frac{\text{Excess initial cost}}{\text{Annaul cost savings}} = \frac{\$1100}{\$110\,/\,\text{yr}} = \mathbf{10\ years}$$

Therefore, the more efficient heater will pay for the \$1100 cost differential in this case in 10 years, which is more than the 8-year limit. Therefore, the purchase of the cheater and less efficient heater is indicated in this case.

16-149 The convection heat transfer coefficient for a clothed person seated in moving air at a velocity of 0.2 to 4 m/s is given by $h = 8.3V^{0.6}$ where V is in m/s and h is in W/m^2.°C. The convection coefficients in that range vary from 3.16 W/m^2.°C at 0.2 m/s to 19.07 W/m^2.°C at 4.0 m/s. Therefore, at low velocities, the radiation and convection heat transfer coefficients are comparable in magnitude. But at high velocities, the convection coefficient is much larger than the radiation heat transfer coefficient.

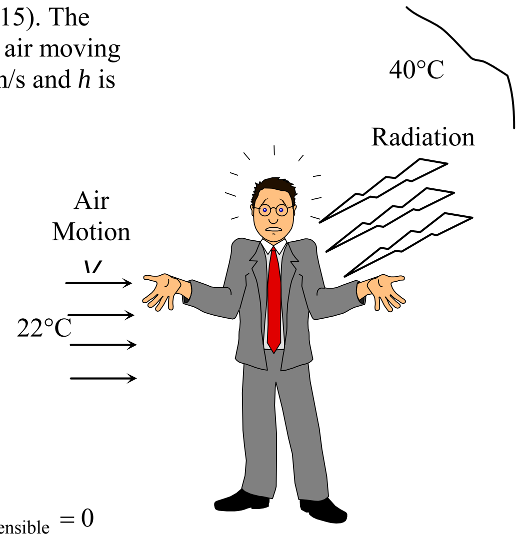
Velocity, m/s	$h = 8.3V^{0.6}$ W/m^2.°C
0.0	3.10
0.20	3.16
0.5	5.48
1.00	8.30
1.5	10.59
2.0	12.58
2.5	14.40
3.0	16.05
3.5	17.60
4.0	19.07



16-150 Workers in a casting facility are surrounded with hot surfaces. The velocity of air needed to provide comfort for the workers is to be determined.

Assumptions **1** The average clothing and exposed skin temperature of the workers is 30°C. **2** The workers are standing in moving air.

Properties The emissivity of the person is 0.95 (Table A-15). The convection heat transfer coefficient for a standing man in air moving with a velocity V is given by $h = 14.8V^{0.69}$ where V is in m/s and h is in W/m^2.°C (Table 16-3).



Analysis The rate of sensible heat transfer from the person is

$$\dot{Q}_{\text{gen, sensible}} = 0.5 \times \dot{Q}_{\text{gen, total}} = 0.5(300\text{ W}) = 150\text{ W}$$

Under steady conditions, and energy balance on the body can be expressed as

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

$$\dot{Q}_{\text{in, radiation}} - \dot{Q}_{\text{out, convection}} + \dot{Q}_{\text{gen, sensible}} = 0$$

or

$$\varepsilon A\sigma(T_{\text{surr}}^4 - T_{\text{person}}^4) - h_{\text{conv}} A(T_{\text{person}} - T_{\text{ambient}}) + \dot{Q}_{\text{gen, sensible}} = 0$$

$$0.95(1.8\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)[(40+273\text{ K})^4 - (30+273\text{ K})^4]$$
$$- h(1.8\text{ m}^2)(30-22)°\text{C} + 150\text{ W} = 0$$

Solving the equation above for h gives

$$h = 18.3\text{ W/m}^2\cdot°\text{C}$$

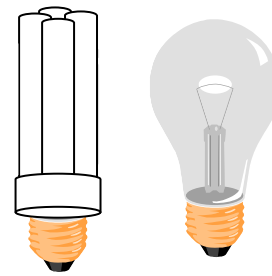
Then the velocity of air needed to provide comfort for the workers is determined from $h = 14.8V^{0.69} = 18.3$ to be

V = **1.36 m/s**

Therefore, the velocity of air must be at least 1.36 m/s.

16-151 Switching to energy efficient lighting reduces the electricity consumed for lighting as well as the cooling load in summer, but increases the heating load in winter. It is to be determined if switching to efficient lighting will increase or decrease the total heating and cooling cost of the building whose annual heating load is roughly equal to the annual cooling load.

Assumptions **1** The annual heating load of the building is roughly equal to the annual cooling load. **2** The light escaping through the windows is negligible so that the entire lighting energy becomes part of the internal heat generation.



Analysis Consider 1 h of operation of lighting in summer and 1 h of operation winter.

Current lighting:

Lighting cost: (Energy used)(Unit cost)= (2 kWh)($0.08/kWh) = $0.16

Increase in the air conditioning load: = 1 kWh

Increase in the air conditioning cost: (Increase in load/COP)(unit cost)=(1 kWh/3.5)($0.08/kWh)=$0.0229

Decrease in the heating cost = (1/29.3 therm)($1.25/therm) = 0.0427

Net cost of 1 h of operation of lighting in summer and 1 h of operation in winter is

Current net cost = Cost of lighting - Cost of heating + Cost of Air-conditioning.

= 0.16-0.0427+0.0239

= **$0.14**

(*a*) *Energy-efficient lighting* (consumes 1/4th of the electricity for the same lighting)

Lighting cost: (Energy used)(Unit cost)= (0.5 kWh)($0.08/kWh) = $0.04

Increase in the air conditioning load: = 0.25 kWh

Increase in air conditioning cost: (Increase in load/COP)(unit cost)=(0.25 kWh/3.5)($0.08/kWh) = $0.0057

Decrease in the heating cost = (0.25/29.3 therm)($1.25/therm) = $0.0107

Net cost of 1 h of operation of lighting in summer and 1 h of operation in winter is

Current net cost = Cost of lighting - Cost of heating + Cost of Air-conditioning.

= 0.04-0.0057+0.0107

= **$0.045**

Therefore, the energy efficient lighting will reduce the total energy usage and cost of this family considerably.

16-152 The outer surfaces of the walls of a brick farmhouse are exposed to 24 km/h winds. The rate of heat transfer through a 20-m^2 section of the wall is to be determined.

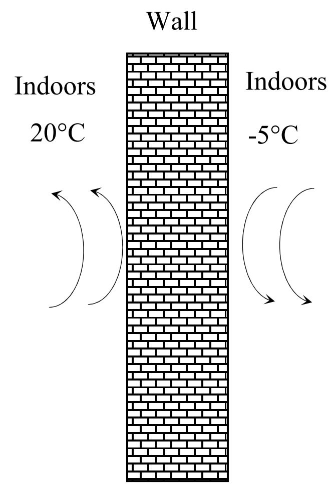
Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the wall is one-dimensional. **3** Thermal properties of the wall and the heat transfer coefficients are constant.

Properties The R-values of different materials are given in Table 16-10.

Analysis The schematic of the wall as well as the different elements used in its construction are shown below. Heat transfer through the insulation and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Once the unit thermal resistances and the *U*-factors for the insulation and stud sections are available, the overall average thermal resistance for the entire wall can be determined from

$$R_{\text{overall}} = 1/U_{\text{overall}} \quad \text{where} \quad U_{\text{overall}} = (Uf_{\text{area}})_{\text{insulation}} + (Uf_{\text{area}})_{\text{stud}}$$

and the value of the area fraction f_{area} is 0.80 for insulation section and 0.20 for stud section since the headers that constitute a small part of the wall are to be treated as studs. Using the available *R*-values from Table 16-10 and calculating others, the total *R*-values for each section is determined in the table below.



Construction	*R*-value, m^2.°C/W
1. Outside surface, 24 km/h winds	0.030
2. Common brick, 200 mm	0.12×2 =0.24
3. Inside surface, still air	0.12
TOTAL	0.39

Then the *U*-factor of the wall after the rate of heat transfer through the wall become

$$U_{\text{wall}} = 1/R_{\text{wall}} = 1/0.39 = 2.56\,\text{m}^2\cdot\text{°C/W}$$

$$\dot{Q}_{\text{wall}} = UA(T_i - T_o) = (2.56\ \text{W/m}^2\cdot\text{°C})(20\,\text{m}^2)[20-(-5)\text{°C}] = \mathbf{1280\ W}$$

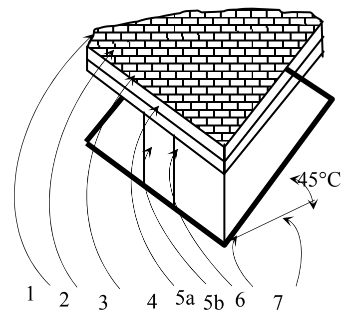
16-153E The *R*-value and the *U*-factor of a 45° pitched roof are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the roof is one-dimensional. **3** Thermal properties of the roof and the heat transfer coefficients are constant.

Properties The R-values of different materials are given in Table 16-10.

Analysis The schematic of the pitched roof as well as the different elements used in its construction are shown below. Using the available *R*-values from Table 16-10, the overall *R*-value of the roof can be determined as shown in the table below.

Construction	*R*-value, h.ft^2.°F/Btu	
	Between studs	At studs
1. Outside surface, 15 mph wind	0.17	0.17
2. Asphalt shingle roofing	0.44	0.44
3. Building paper	0.06	0.06
4. Plywood deck, 5/8 in	0.78	0.78
5a. Reflective air space, 3.5-in	2.17	----
5b. Wood stud, 2 in by 4 in	----	3.58
6. Gypsum wallboard, 0.5 in	0.45	0.45
7. Inside surface, still air	0.68	0.68



Total unit thermal resistance of each section, R	4.75	6.16
The U-factor of each section, $U = 1/R$, in Btu/h.ft^2.°F	0.211	0.162
Area fraction of each section, f_{area}	0.80	0.20
Overall U-factor, $U = \Sigma f_{\text{area,i}} U_i = 0.80\times0.211+0.20\times0.162$	**0.201 Btu/h.ft^2.°F**	
Overall unit thermal resistance, $R = 1/U$	**4.97 h.ft^2.°F/Btu**	

Therefore, the *R*-value and *U*-factor of the roof are R = 4.97 h.ft^2.°F/Btu and U = 0.201 Btu/h.ft^2.°F

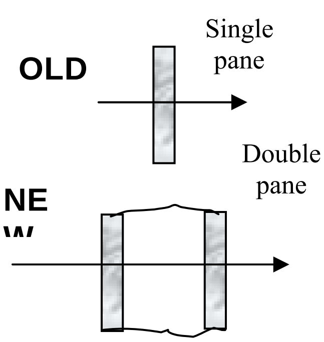
16-154 Heat losses through the windows of a house with aluminum frame single pane windows account for 26 percent of the total. The reduction in the heat load of the house as a result of switching to vinyl frame double pane windows is to be determined.

Properties The *U*-factors of the aluminum frame single pane and vinyl frame double pane windows are 7.16 and 2.74 W/m^2.°C, respectively.

Analysis The rate of heat transfer through the windows for the existing house is

$$\dot{Q}_{\text{window, old}} = (\text{Fraction of heat loss through windows})\dot{Q}_{\text{house, total}} = 0.26\times(32\text{ kW}) = 8.32\text{ kW}$$

Then the rate of heat transfer through the new windows becomes



$$\frac{\dot{Q}_{\text{window, new}}}{\dot{Q}_{\text{window, old}}} = \frac{U_{\text{new}} A\Delta T}{U_{\text{old}} A\Delta T}$$

which gives

$$\dot{Q}_{\text{window, new}} = \frac{U_{\text{new}}}{U_{\text{old}}}\dot{Q}_{\text{window, old}} = \frac{2.74}{7.14}(8.32\text{ kW}) = 3.19\text{ kW}$$

Therefore, the reduction in the heat load of the house is

$$\dot{Q}_{\text{reducdtion}} = \dot{Q}_{\text{window, old}} - \dot{Q}_{\text{window, new}} = 8.32 - 3.19 = \mathbf{5.13\,kW}$$

Discussion Note that the heat load from the house will go down by 16% since 5.13/32 = 0.16.

16-155 The attic of a house in Thessaloniki, Greece is not vented in summer. The rate of heat gain through the roof in late July is to be determined assuming the roof is (*a*) light colored and (*b*) dark colored.

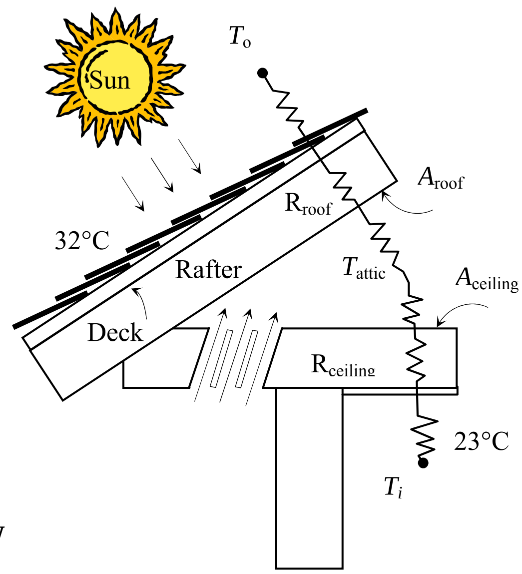
Assumptions **1** Steady operating conditions exist. **2** Thermal properties of the roof and the heat transfer coefficients are constant. **3** The effect of air infiltration on the attic temperature is negligible (this will result in a higher rate of heat transfer than actual). **4** The sol-air temperature for a horizontal surface can be used for the tilted roof.

Properties The *R*-values of the roof and the ceiling are given to be 1.4 m^2.°C/W and 0.50 m^2.°C/W, respectively.

Analysis Noting that the roof-to-ceiling area ratio is 1.4, the thermal resistance of the roof-attic-ceiling combination per unit area of the ceiling is

$$R_{\text{total}} = R_{\text{ceiling}} + \left(\frac{A_{\text{ceiling}}}{A_{\text{roof}}}\right) R_{\text{roof}} = 0.50 + \tfrac{1}{4} 1.40 = 1.50\ \text{m}^2 \cdot °\text{C/W}$$

(*a*) The house is located at 40°N latitude, and thus we can use the sol-air temperature data directly from Table 16-7. At 16:00, the tabulated sol-air temperatures for a light-colored horizontal surface is 42.7°C. Also, the tabulated air temperature is 34.7° C, which is 2.7°C higher than the air temperature given in the problem. But we can still use the data in that table provided that we subtract 2.7°C from all temperatures. Therefore, the sol-air temperature on the roof in this case is 42.7 - 2.7 = 40.0°C, and the heat gain through the roof is determined to be



$$\begin{aligned}\dot{Q}_{\text{roof, light color}} &= UA(T_{\text{sol-air, light color}} - T_{\text{inside}}) \\ &= A\frac{T_{\text{sol-air, light color}} - T_{\text{inside}}}{R} \\ &= (150\ \text{m}^2)\frac{(40-23)°\text{C}}{1.50\ \text{m}^2 \cdot °\text{C/W}} = \mathbf{1700\ W}\end{aligned}$$

(*b*) The tabulated sol-air temperatures for a dark-colored horizontal surface is 54.7°C. Therefore, the sol-air temperature on the roof in this case is 54.7 - 2.7 = 52.0°C, and the heat gain through the roof is determined to be

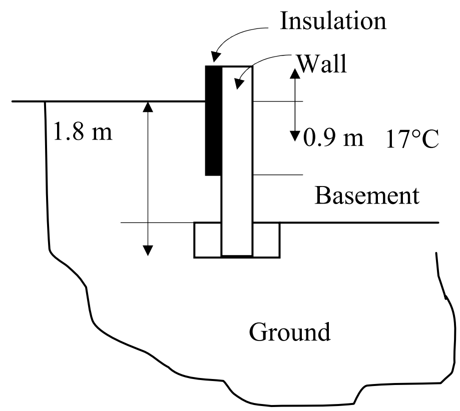
$$\begin{aligned}\dot{Q}_{\text{roof, dark color}} &= UA(T_{\text{sol-air, dark color}} - T_{\text{inside}}) \\ &= A\frac{T_{\text{sol-air, darh color}} - T_{\text{inside}}}{R} = (150\ \text{m}^2)\frac{(52-23)°\text{C}}{1.50\ \text{m}^2 \cdot °\text{C/W}} = \mathbf{2900\ W}\end{aligned}$$

Discussion Note that the color of the exposed surface of the roof has a major effect on the rate of heat gain through the roof.

16-156 The peak heat loss from a below grade basement in Norfolk, Virginia to the ground through its walls and floor is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The basement is maintained at 17°C.

Properties The winter average temperature of Norfolk is 9.9°C (Table 16-5). The heat transfer coefficients are given in Table 16-14a, and the amplitudes in Fig. 16-37.



Solution The floor and wall areas of the basement are

$$A_{\text{wall}} = \text{Height} \times \text{Perimeter} = 2 \times (1.8\text{ m})(10+19\text{ m}) = 104.4\text{ m}^2$$

$$A_{\text{floor}} = \text{Length} \times \text{Width} = (10\text{ m})(19\text{ m}) = 190\text{ m}^2$$

The amplitude of the annual soil temperature is determined from Fig. 16-37 to be 10°C. Then the ground surface temperature for the design heat loss becomes

$$T_{\text{ground surface}} = T_{\text{winter, mean}} - A = 9.9 - 10 = -0.1°\text{C}$$

The top 0.9-m section of the wall below the grade is insulated with *R*-0.73, and the heat transfer coefficients through that section are given in Table 16-14a to be 2.87and 2.20 W/m^2.°C through the 1st and 2nd 0.3-m wide depth increments, respectively. The heat transfer coefficients through the uninsulated section of the wall which extends from 0.6 m to 1.8 m level is determined from the same table to be 2.93, 2.23, 1.80, and 1.50 W/m^2.°C for each of the remaining 0.3-m wide depth increments. The average overall heat transfer coefficient is

$$U_{\text{wall, ave}} = \frac{\sum U_{\text{wall}}}{\text{No. of increments}} = \frac{2.87+2.20+2.93+2.23+1.8+1.5}{6} = 2.255\text{ W/m}^2.°\text{C}$$

Then the heat loss through the basement wall becomes

$$\dot{Q}_{\text{basement walls}} = U_{\text{wall, ave}} A_{\text{wall}} (T_{\text{basement}} - T_{\text{ground surface}})$$
$$= (2.255\text{ W/m}^2 \cdot °\text{C})(104.4\text{ m}^2)[17-(-0.1)°\text{C}] = 4026\text{ W}$$

The shortest width of the house is 10 m, and the depth of the foundation below grade is 1.8 m. The floor heat transfer coefficient is determined from Table 16-14b to be 0.12 W/m^2.°C. Then the heat loss through the floor of the basement becomes

$$\dot{Q}_{\text{basement floor}} = U_{\text{floor}} A_{\text{floor}} (T_{\text{basement}} - T_{\text{ground surface}})$$
$$= (0.12\text{ W/m}^2.°\text{C})(104.4\text{ m}^2)[17-(-0.1)]°\text{C} = 390\text{ W}$$

which is considerably less than the heat loss through the wall. The total heat loss from the basement is then determined to be

$$\dot{Q}_{\text{basement}} = \dot{Q}_{\text{basement wall}} + \dot{Q}_{\text{basement floor}} = 4026 + 390 = \mathbf{4416\ W}$$

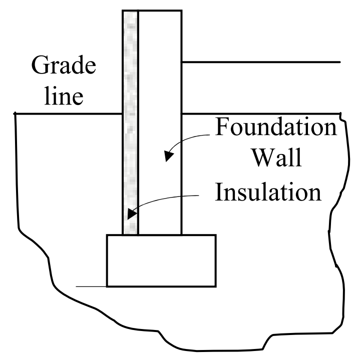
Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-157 A house with a concrete slab floor sits directly on the ground at grade level, and the wall below grade is insulated. The heat loss from the floor at winter design conditions is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The house is maintained at 22°C. **3** The weather in Anchorage is severe.

Properties The 97.5% winter design conditions in Anchorage is -28°C (Table 16-4). The heat transfer coefficient for the insulated wall below grade is U = 0.86 W/m.°C (Table 16-14c).

Solution Heat transfer from a floor on the ground at the grade level is proportional to the perimeter of the floor, and the perimeter in this case is



$$p_{\text{floor}} = 2 \times (\text{Length} + \text{Width}) = 2(15 + 20) \text{ m} = 70 \text{ m}$$

Then the heat loss from the floor becomes

$$\dot{Q}_{\text{floor}} = U_{\text{floor}} p_{\text{floor}} (T_{\text{indoor}} - T_{\text{outdoor}})$$
$$= (0.86 \text{ W/m} \cdot °\text{C})(70 \text{ m})[22 - (-28)]°\text{C} = \mathbf{2890 \text{ W}}$$

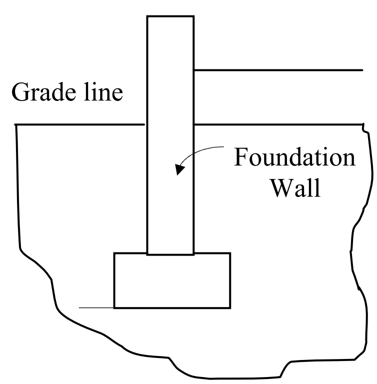
Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-158 A house with a concrete slab floor sits directly on the ground at grade level, and the wall below grade is uninsulated. The heat loss from the floor at winter design conditions is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The house is maintained at 22°C. **3** The weather in Anchorage is severe.

Properties The 97.5% winter design conditions in Baltimore is -28°C (Table 16-4). The heat transfer coefficient for the uninsulated wall below grade is U = 1.17 W/m.°C (Table 16-14c).

Analysis Heat transfer from a floor on the ground at the grade level is proportional to the perimeter of the floor, and the perimeter in this case is



$$p_{\text{floor}} = 2 \times (\text{Length} + \text{Width}) = 2(15 + 20) \text{ m} = 70 \text{ m}$$

Then the heat loss from the floor becomes

$$\dot{Q}_{\text{floor}} = U_{\text{floor}} p_{\text{floor}} (T_{\text{indoor}} - T_{\text{outdoor}})$$
$$= (1.17 \text{ W/m} \cdot °\text{C})(70 \text{ m})[22 - (-28)]°\text{C} = \mathbf{3931 \text{ W}}$$

Discussion This is the *design* or *peak* rate of heat transfer from below-grade section of the basement, and this is the value to be used when sizing the heating system. The actual heat loss from the basement will be much less than that most of the time.

16-159 The classrooms and faculty offices of a university campus are not occupied an average of 4 hours a day, but the lights are kept on. The amounts of electricity and money the campus will save a year if the lights are turned off during unoccupied periods are to be determined.

Analysis The total electric power consumed by the lights in the classrooms and faculty offices is

$$\dot{E}_{\text{lighting, classroom}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (200 \times 12 \times 110 \text{ W}) = 264{,}000 = 264 \text{ kW}$$

$$\dot{E}_{\text{lighting, offices}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (400 \times 6 \times 110 \text{ W}) = 264{,}000 = 264 \text{ kW}$$

$$\dot{E}_{\text{lighting, total}} = \dot{E}_{\text{lighting, classroom}} + \dot{E}_{\text{lighting, offices}} = 264 + 264 = 528 \text{ kW}$$

Noting that the campus is open 240 days a year, the total number of unoccupied work hours per year is

Unoccupied hours = (4 hours/day)(240 days/year) = 960 h/yr

Then the amount of electrical energy consumed per year during unoccupied work period and its cost are

$$\text{Energy savings} = (\dot{E}_{\text{lighting, classroom}})(\text{Unoccupied hours}) = (528 \text{ kW})(960 \text{ h/yr}) = \mathbf{506{,}880 \text{ kWh}}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of energy}) = (506{,}880 \text{ kWh})(\$0.075/\text{kWh}) = \mathbf{\$38{,}016/yr}$$

Discussion Note that simple conservation measures can result in significant energy and cost savings.

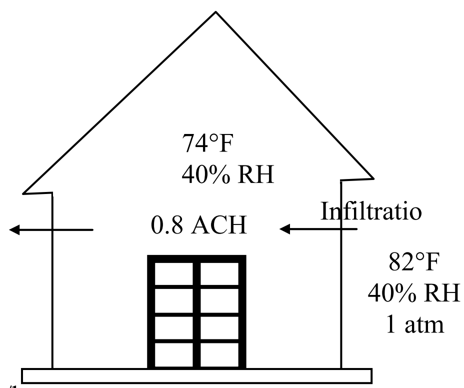
16-160E The infiltration rate of a building is estimated to be 0.8 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at 40°F. **4** The effect of water vapor on air density is negligible.

Properties The gas constant and the specific heat of air are R = 0.3704 psia.ft^3/lbm.R and C_p = 0.24 Btu/lbm·°F (Tables A-1E and A-11E). The heat of vaporization of water at 40°F is $h_{fg} = h_{fg@40°F} = 1071$ Btu/lbm (Table A-9E). The properties of the ambient and room air are determined from the psychrometric chart (Fig. A-13E) to be

$$\left.\begin{aligned} T_{\text{ambient}} &= 82^\circ \text{F} \\ \phi_{\text{ambient}} &= 40\% \end{aligned}\right\} \begin{aligned} \omega_{\text{ambient}} &= 0.094 \text{ lbm/lbm dryair} \\ v &= 13.7 \text{ ft}^3\text{/lbm dry air} \end{aligned}$$

$$\left.\begin{aligned} T_{\text{room}} &= 74^\circ \text{F} \\ \phi_{\text{room}} &= 40\% \end{aligned}\right\} \omega_{\text{room}} = 0.073 \text{ lbm/lbm dryair}$$



Analysis Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 0.8 times every hour, the air will enter the room at a mass flow rate of

$$\dot{m}_{\text{air}} = \frac{V_{\text{room}}}{v_{\text{ambient}}} ACH = \frac{9 \times 50 \times 60 \text{ ft}^3}{13.7 \text{ ft}^3\text{/lbm dry air}} (0.8 \text{ h}^{-1}) = 176 \text{ lbm/h}$$

Then the sensible, latent, and total infiltration heat loads of the room are

$$\dot{Q}_{\text{infiltration,sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (176 \text{ lbm/h})(0.24 \text{ Btu/lbm.°F})(82 - 74)\text{°F} = \mathbf{337 \text{ Btu/h}}$$

$$\dot{Q}_{\text{infiltration,latent}} = \dot{m}_{\text{air}} (\omega_{\text{ambient}} - \omega_{\text{room}}) h_{fg} = (176 \text{ lbm/h})(0.094 - 0.073)(1071 \text{ Btu/lbm}) = \mathbf{3952 \text{ Btu/h}}$$

$$\dot{Q}_{\text{infiltration}} = \dot{Q}_{\text{infiltration,sensible}} + \dot{Q}_{\text{infiltration,latent}} = 3952 + 337 = \mathbf{4289 \text{ Btu/h}}$$

Discussion The specific volume of the dry air at the ambient conditions could also be determined from the ideal gas relation, $v = RT/P = (0.3704 \text{ psia} \cdot \text{ft}^3\text{/lbm} \cdot \text{R})(82 + 460 \text{ R})/14.7 \text{ psia} = 13.7 \text{ ft}^3/\text{lbm}$.

16-161 It is believed that January is the coldest month in the Northern hemisphere. On the basis of Table 16-5, it is to be determined if this is true for all locations.

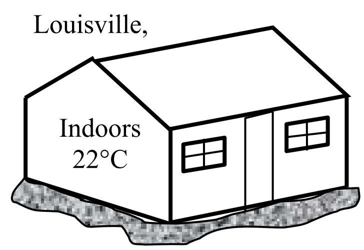
Analysis Degree-days is a measure of coldness of a location. We notice from Table 16-5 that in Atlanta, Georgia the heating degree-days are 648°F-days in December and 636°F-days in January. Therefore, January is **not** necessarily the coldest month in Northern Hemisphere.

16-162 The December space heating bill of a fully occupied house is given. The heating bill of this house is to be determined if it were not occupied.

Assumptions The outdoors temperature never rises above 18°C in January.

Properties The heating degree-days for Louisville, Kentucky for the month of January is 890°F-days (Table 16-5). It is equivalent to 890/1.8 = 494°C-days.

Analysis The degree-days method is based on the assumption that the internal heat gain can meet the heating needs of a house when the outdoors temperature is above 18°C. Therefore, for an indoor temperature of 22°C, the number of degree-days that correspond to the temperature range of 18 to 22°C is



$$DD_{internal} = (22-18)\ °C(31 \text{ days}) = 124°C\text{-days}$$

Then the total °C-days for an unoccupied building becomes

$$DD_{total} = DD_{standard} + DD_{internal} = 494+124 = 618°C\text{-days}$$

Noting that 618/494 = 1.251, we conclude that the number of heating degree-days and thus the heating bill in January will increase by 25.1%. Therefore, the January heating bill for the unoccupied house will be

$$\text{Heating bill of unoccupied house} = 1.251\times(\text{Heating bill of occupied house})$$

$$= 1.251\times\$110 = \mathbf{\$138}$$

Discussion Note that the heating bill of this house will increase by 25.1% in the absence of any internal heat gain.

16-163 The annual gas consumption and its cost for a house in Charlotte, NC with a design heat load of 28 kW and a furnace efficiency of 80% are to be determined.

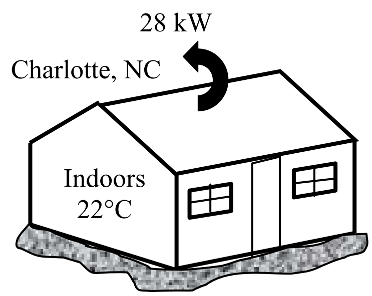
Assumptions The house is maintained at 22°C at all times during the heating season.

Properties The annual heating degree-days of Charlotte, NC is 3191°F-days (Table 16-5). It is equivalent to 3191/1.8 = 1773°C-days. The 97.5% winter design temperature of Charlotte is -6°C.

Analysis (*a*) The overall heat loss coefficient K_{overall} of the building is determined from

$$\dot{Q}_{\text{design}} = UA\Delta T_{\text{design}} = UA(T_{\text{i}} - T_o)_{\text{design}} = K_{\text{overall}}(T_{\text{i}} - T_o)_{\text{design}}$$

Substituting,



$$K_{\text{overall}} = \frac{\dot{Q}_{\text{design}}}{(T_{\text{i}} - T_o)_{\text{design}}} = \frac{28{,}000\text{ W}}{[22-(-6)]°\text{C}} = 1000\text{W} = \mathbf{1\,kW/°C}$$

(*b*) The annual gas consumption of the house for heating and its cost are determined to be

$$\text{Annual Gas Consumption} = \frac{K_{\text{overall}}}{\eta_{\text{heating}}} DD_{\text{heating}}$$

$$= \frac{1000\text{ W/°C}}{0.80}(1773°\text{ C - days})\left(\frac{24 \times 3600\text{ s}}{1\text{day}}\right)$$

$$= 1.915 \times 10^8 = \mathbf{1815\ therms} \qquad (\text{since 1 therm} = 105{,}500\text{ kJ})$$

Then the annual heating cost becomes

$$\text{Annual heating cost} = (\text{Annual energy consumption})(\text{Unit cost of energy})$$

$$= (1815\text{ therms/year})(\$1.20/\text{therm}) = \mathbf{\$2178/year}$$

Therefore, it will cost $2178 per year to heat this house.

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