Combined Natural and Forced Convection

9-72C In combined natural and forced convection, the natural convection is negligible when $Gr/\mathrm{Re}^2 < 0.1$. Otherwise it is not.

9-73C In assisting or transverse flows, natural convection enhances forced convection heat transfer while in opposing flow it hurts forced convection.

9-74C When neither natural nor forced convection is negligible, it is not correct to calculate each separately and to add them to determine the total convection heat transfer. Instead, the correlation

$$\mathrm{Nu}_{\mathrm{combined}} = \left(\mathrm{Nu}_{\mathrm{forced}}^n + \mathrm{Nu}_{\mathrm{natural}}^n\right)^{1/n}$$

based on the experimental studies should be used.

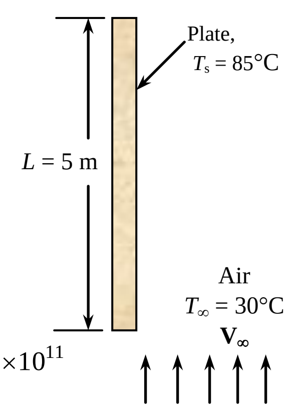
9-75 A vertical plate in air is considered. The forced motion velocity above which natural convection heat transfer from the plate is negligible is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The atmospheric pressure at that location is 1 atm.

Properties The properties of air at 1 atm and 1 atm and the film temperature of $(T_s+T_\infty)/2$ = (85+30)/2 = 57.5°C are (Table A-15)

$$\upsilon = 1.871\times10^{-5}\ \mathrm{m^2/s}$$

$$\beta = \frac{1}{T_f} = \frac{1}{(57.5+273)\mathrm{K}} = 0.003026\mathrm{K^{-1}}$$



Analysis The characteristic length is the height of the plate, $L_c = L = 5$ m. The Grashof and Reynolds numbers are

$$Gr = \frac{g\beta(T_s - T_\infty)L^3}{\upsilon^2} = \frac{(9.81\,\mathrm{m/s^2})(0.003026\mathrm{K^{-1}})(85-30\,\mathrm{K})(5\,\mathrm{m})^3}{(1.871\times10^{-5}\ \mathrm{m^2/s})^2} = 5.829\times10^{11}$$

$$\mathrm{Re} = \frac{V_\infty L}{\upsilon} = \frac{V_\infty(5\,\mathrm{m})}{1.871\times10^{-5}\ \mathrm{m^2/s}} = 2.67\times10^5 V_\infty$$

and the forced motion velocity above which natural convection heat transfer from this plate is negligible is

$$\frac{Gr}{\mathrm{Re}^2} = 0.1 \longrightarrow \frac{5.829\times10^{11}}{(2.67\times10^5 V_\infty)^2} = 0.1 \longrightarrow V_\infty = \mathbf{9.04\ m/s}$$

9-76 "!PROBLEM 9-76"

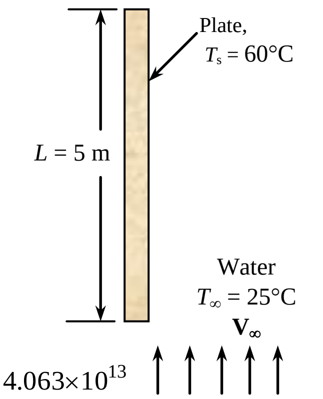
```
"GIVEN"
L=5 "[m]"
"T_s=85 [C], parameter to be varied"
T_infinity=30 "[C]"

"PROPERTIES"
Fluid$='air'
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
beta=1/(T_film+273)
T_film=1/2*(T_s+T_infinity)
g=9.807 "[m/s^2], gravitational acceleration"

"ANALYSIS"
Gr=(g*beta*(T_s-T_infinity)*L^3)/nu^2
Re=(Vel*L)/nu
Gr/Re^2=0.1
```

T_s [C]	Vel [m/s]
50	5.598
55	6.233
60	6.801
65	7.318
70	7.793
75	8.233
80	8.646
85	9.033
90	9.4
95	9.747
100	10.08
105	10.39
110	10.69
115	10.98
120	11.26
125	11.53
130	11.79
135	12.03
140	12.27
145	12.51
150	12.73

9-77 A vertical plate in water is considered. The forced motion velocity above which natural convection heat transfer from the plate is negligible is to be determined.



Assumptions **1** Steady operating conditions exist.

Properties The properties of water at the film temperature of $(T_s+T_\infty)/2$ = (60+25)/2 = 42.5°C are (Table A-15)

$$\upsilon = 0.65\times 10^{-6}\ \text{m}^2/\text{s}$$

$$\beta = 0.00040\ \text{K}^{-1}$$

Analysis The characteristic length is the height of the plate $L_c = L = 5$ m. The Grashof and Reynolds numbers are

$$Gr = \frac{g\beta(T_s - T_\infty)L^3}{\upsilon^2} = \frac{(9.81\,\text{m/s}^2)(0.0004\,\text{K}^{-1})(60-25\,\text{K})(5\,\text{m})^3}{(0.65\times 10^{-6}\ \text{m}^2/\text{s})^2} = 4.063\times 10^{13}$$

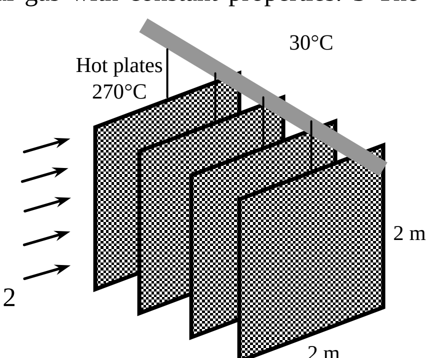
$$\text{Re} = \frac{V_\infty L}{\upsilon} = \frac{V_\infty (5\,\text{m})}{0.65\times 10^{-6}\ \text{m}^2/\text{s}} = 4.6\times 10^6 V_\infty$$

and the forced motion velocity above which natural convection heat transfer from this plate is negligible is

$$\frac{Gr}{\text{Re}^2} = 0.1 \longrightarrow \frac{4.063\times 10^{13}}{(4.6\times 10^6 V_\infty)^2} = 0.1 \longrightarrow V_\infty = \mathbf{2.62\,m/s}$$

9-78 Thin square plates coming out of the oven in a production facility are cooled by blowing ambient air horizontally parallel to their surfaces. The air velocity above which the natural convection effects on heat transfer are negligible is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The atmospheric pressure at that location is 1 atm.



Properties The properties of air at 1 atm and 1 atm and the film temperature of $(T_s+T_\infty)/2$ = (270+30)/2 = 150°C are (Table A-15)

$$\upsilon = 2.859\times 10^{-5}\ \text{m}^2/\text{s}$$

$$\beta = \frac{1}{T_f} = \frac{1}{(150+273)\text{K}} = 0.002364\text{K}^{-1}$$

Analysis The characteristic length is the height of the plate $L_c = L = 2$ m. The Grashof and Reynolds numbers are

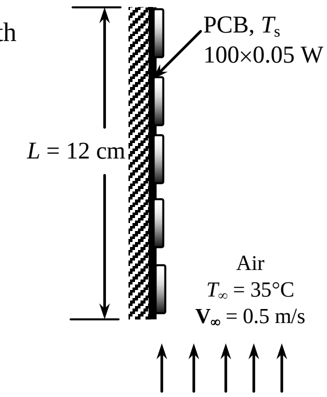
$$Gr = \frac{g\beta(T_s - T_\infty)L^3}{\upsilon^2} = \frac{(9.81\,\text{m/s}^2)(0.002364\text{K}^{-1})(270-30\,\text{K})(2\,\text{m})^3}{(2.859\times 10^{-5}\ \text{m}^2/\text{s})^2} = 5.447\times 10^{10}$$

$$\text{Re} = \frac{V_\infty L}{\upsilon} = \frac{V_\infty (2\,\text{m})}{2.859\times 10^{-5}\ \text{m}^2/\text{s}} = 6.995\times 10^4 V_\infty$$

and the forced motion velocity above which natural convection heat transfer from this plate is negligible is

$$\frac{Gr}{\text{Re}^2} = 0.1 \longrightarrow \frac{5.447\times 10^{10}}{(6.995\times 10^4 V_\infty)^2} = 0.1 \longrightarrow V_\infty = \mathbf{10.6\ m/s}$$

9-79 A circuit board is cooled by a fan that blows air upwards. The average temperature on the surface of the circuit board is to be determined for two cases.



Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The atmospheric pressure at that location is 1 atm.

Properties The properties of air at 1 atm and 1 atm and the anticipated film temperature of $(T_s+T_\infty)/2$ = (60+35)/2 = 47.5°C are (Table A-15)

$$k = 0.02717 \text{ W/m.°C}$$
$$\upsilon = 1.774\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7235$$
$$\beta = \frac{1}{T_f} = \frac{1}{(47.5+273)\text{K}} = 0.00312 \text{ K}^{-1}$$

Analysis We assume the surface temperature to be 60°C. We will check this assumption later on and repeat calculations with a better assumption, if necessary. The characteristic length in this case is the length of the board in the flow (vertical) direction, $L_c = L$ = 0.12 m. Then the Reynolds number becomes

$$\text{Re} = \frac{V_\infty L}{\upsilon} = \frac{(0.5 \text{ m/s})(0.12 \text{ m})}{1.774\times10^{-5} \text{ m}^2\text{/s}} = 3383$$

which is less than critical Reynolds number (5×10^5). Therefore the flow is laminar and the forced convection Nusselt number and h are determined from

$$Nu = \frac{hL}{k} = 0.664\,\text{Re}^{0.5}\Pr^{1/3} = 0.664(3383)^{0.5}(0.7235)^{1/3} = 34.67$$

$$h = \frac{k}{L}Nu = \frac{0.02717 \text{ W/m.°C}}{0.12 \text{ m}}(34.67) = 7.85 \text{ W/m}^2\text{.°C}$$

$$A_s = L\times W = (0.12 \text{ m})(0.2 \text{ m}) = 0.024 \text{ m}^2$$

Then,

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 35°\text{C} + \frac{(100)(0.05 \text{ W})}{(7.85 \text{ W/m}^2\text{.°C})(0.024 \text{ m}^2)} = \mathbf{61.5°C}$$

which is sufficiently close to the assumed value in the evaluation of properties. Therefore, there is no need to repeat calculations.

(b) The Rayleigh number is

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\upsilon^2}\Pr = \frac{(9.81 \text{ m/s}^2)(0.00312 \text{ K}^{-1})(60-35 \text{ K})(0.12 \text{ m})^3}{(1.774\times10^{-5} \text{ m}^2\text{/s})^2}(0.7235) = 3.041\times10^6$$

$$Nu = \left\{0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+\left(\frac{0.492}{\Pr}\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.825 + \frac{0.387(3.041\times10^6)^{1/6}}{\left[1+\left(\frac{0.492}{0.7235}\right)^{9/16}\right]^{8/27}}\right\}^2 = 22.42$$

This is an assisting flow and the combined Nusselt number is determined from

$$Nu_{combined} = (Nu_{forced}{}^n + Nu_{natural}{}^n)^{1/n} = (34.67^3 + 22.42^3)^{1/3} = 37.55$$

Then, $$h = \frac{k}{L}Nu_{combined} = \frac{0.02717 \text{ W/m.°C}}{0.12 \text{ m}}(37.55) = 8.502 \text{ W/m}^2\text{.°C}$$

and $$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 35°\text{C} + \frac{(100)(0.05\,\text{W})}{(8.502\,\text{W/m}^2.°\text{C})(0.024\,\text{m}^2)} = \mathbf{59.5°C}$$

Therefore, natural convection lowers the surface temperature in this case by about 2°C.

Special Topic: Heat Transfer Through Windows

9-80C 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.

9-81C 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. 9-44.

9-82C 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.

9-83C 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.

9-84C 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.

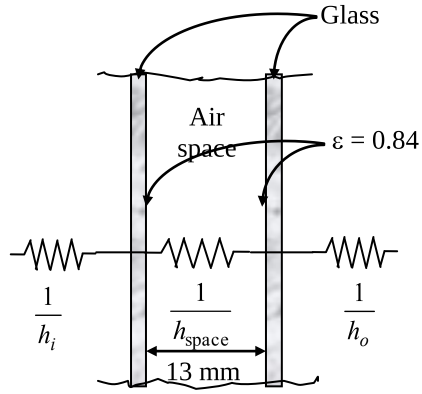
9-85C The heat transfer rate through the glazing of a double pane window is higher at the edge section than it is at the center section because of the two-dimensional effects due to heat transfer through the frame.

9-86C 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.

9-87 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 (from the text).



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

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

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

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

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,

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

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

9-88 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 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 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,

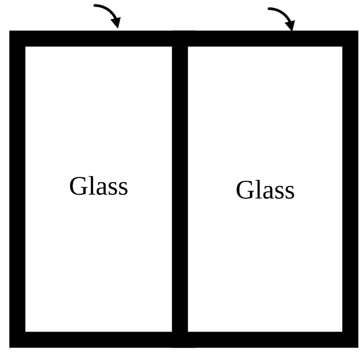
$$\dot{Q}_{\text{window}} = h_i A_{\text{window}} (T_i - T_{\text{glass}}) \quad \rightarrow \quad T_{\text{glass}} = T_i - \frac{\dot{Q}_{\text{window}}}{h_i A_{\text{window}}}$$

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:

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

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



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

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

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

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

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

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

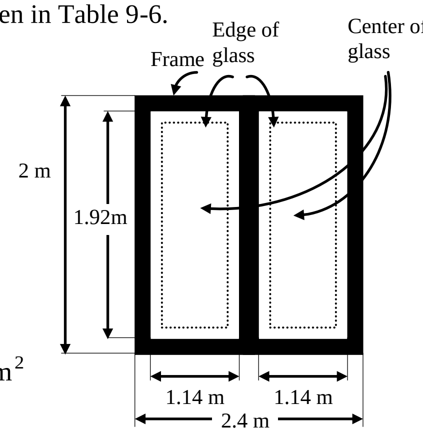
Discussion Note that heat loss through the window will be reduced by 49 percent in the case of double glazing and by 74 percent in the case of triple glazing relative to the single glazing case. Also, in the case of single glazing, the low inner glass surface temperature will cause considerable discomfort 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.

9-89 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 9-6.

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 Table 9-6.

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



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

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

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

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

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

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

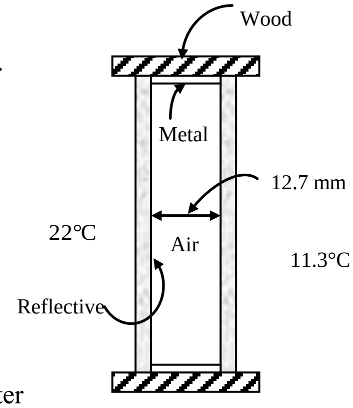
The U-factor for the frame section is determined from Table 9-4 to be U_{frame} = 2.8 W/m^2.°C. The U-factor for the center and edge sections are determined from Table 9-6 to be U_{center} = 2.78 W/m^2.°C and U_{edge} =3.40 W/m^2.°C. Then the overall U-factor of the entire window becomes

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

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.

9-90 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

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

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,

$$\dot{Q}_{\text{window,ave}} = (2.13\,\text{W/m}^2\cdot°\text{C})(20\,\text{m}^2)(22\text{-}11.3)°\text{C} = \mathbf{456\,W}$$

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

9-91E 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 9-6 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

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

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

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

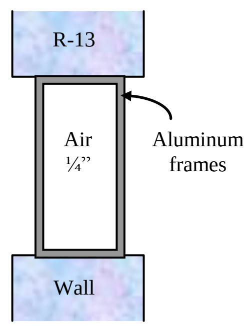
$$\dot{Q}_{\text{window}} = U_{\text{window}} A_{\text{window}}(T_i - T_o) = A_{\text{window}} \frac{T_i - T_o}{\text{R} - \text{value}} = (0.2 \text{ft}^2)\frac{\Delta T(°\text{F})}{(1.25 \text{h.ft}^2.°\text{F/Btu}} = 0.160\Delta T \text{ Btu/h}$$

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.



9-92 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.

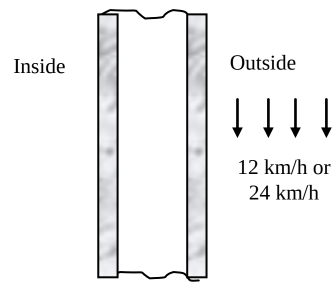
Assumptions Thermal properties of the windows and the heat transfer coefficients are constant.

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 = 34.0 W/m^2.°C for 24 km/h winds (from the text).

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

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

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



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

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

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:

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

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

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

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.

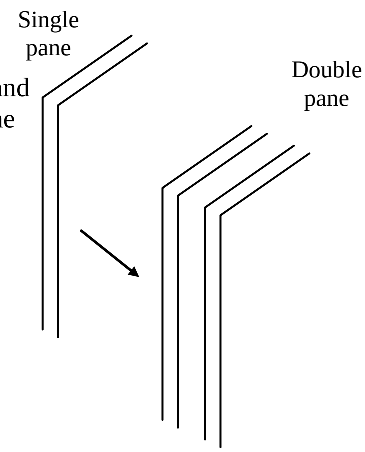
9-93 The existing wood framed single pane windows of an older house in Wichita are to be replaced by double-door type vinyl framed double pane windows with an air space of 6.4 mm. The amount of money the new windows will save the home owner per month is to be determined.

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

$$Q_{\text{window}} = U_{\text{overall}} A_{\text{window}} (T_i - T_o)$$



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

$$\dot{Q}_{\text{window,old}} = (5.57\ \text{W/m}^2.°\text{C})(12\ \text{m}^2)(18-7.1)°\text{C} = 729\ \text{W}$$

$$\dot{Q}_{\text{window,new}} = (3.20\ \text{W/m}^2.°\text{C})(12\ \text{m}^2)(18-7.1)°\text{C} = 419\ \text{W}$$

$$\dot{Q}_{\text{saved}} = \dot{Q}_{\text{window,old}} - \dot{Q}_{\text{window,new}} = 729 - 419 = 310\ \text{W}$$

Then the electrical energy and cost savings per month becomes

$$\text{Energy savings} = \dot{Q}_{\text{saved}} \Delta t = (0.310\ \text{kW})(30 \times 24\ \text{h/month}) = 223\ \text{kWh/month}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of energy}) = (223\ \text{kWh/month})(\$0.07/\text{kWh}) = \mathbf{\$15.62/month}$$

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.