PROBLEM 1.1

Equations

```
$UnitSystem SI Mass J K Pa
```

Problem 1.1

$$th_g = 0.5\ [\text{in}] \cdot \left|0.0254\ \frac{\text{m}}{\text{in}}\right| \quad \text{thickness of glass} \tag{1}$$

$$th_b = 1\ [\text{in}] \cdot \left|0.0254\ \frac{\text{m}}{\text{in}}\right| \quad \text{thickness of brick} \tag{2}$$

$$L = 20\ [\text{in}] \cdot \left|0.0254\ \frac{\text{m}}{\text{in}}\right| \quad \text{size of side} \tag{3}$$

$$T_{in} = \text{ConvertTemp}(F,\ K,\ 400\ [\text{F}]) \quad \text{inner temperature} \tag{4}$$

$$T_{out} = \text{ConvertTemp}(F,\ K,\ 70\ [\text{F}]) \quad \text{outer temperature} \tag{5}$$

$$A_{side} = L^2 \quad \text{area of side} \tag{6}$$

$$k_g = 0.937\ [\text{W/m·K}] \quad \text{conductivity of glass, from Appendix A} \tag{7}$$

$$k_b = 0.72\ [\text{W/m·K}] \quad \text{conductivity of brick, from Appendix A} \tag{8}$$

$$\dot{q}_{cond} = k_g \cdot A_{side} \cdot \frac{T_{in} - T_{out}}{th_g} + k_b \cdot 5 \cdot A_{side} \cdot \frac{T_{in} - T_{out}}{th_b} \tag{9}$$

rate of heat transfer

PROBLEM 1.2

Equations

`$UnitSystem SI Mass J K Pa`

Problem 1.2

$$T_s = \text{ConvertTemp}(F,\ K,\ 200\ [\text{F}]) \quad \text{radiator temperature} \tag{1}$$

$$\dot{w} = 25\ [\text{hp}] \cdot \left| 745.7\ \frac{\text{W}}{\text{hp}} \right| \quad \text{engine power} \tag{2}$$

$$T_\infty = \text{ConvertTemp}(F,\ K,\ 70\ [\text{F}]) \quad \text{air temperature} \tag{3}$$

$$\dot{q}_{conv} = \dot{w} \quad \text{heat transfer} \tag{4}$$

$$A_s = 6\ \left[\text{m}^2\right] \quad \text{area} \tag{5}$$

$$\dot{q}_{conv} = \bar{h} \cdot A_s \cdot (T_s - T_\infty) \quad \text{Newton's Law of Cooling} \tag{6}$$

PROBLEM 1.3

Equations

```
$UnitSystem J K Pa
```

Problem 1.3

$$A_s = 20\ [\text{inch}^2] \cdot \left| 6.4516 \times 10^{-4}\ \frac{\text{m}^2}{\text{inch}^2} \right| \quad \text{surface area} \tag{1}$$

$$T_s = 145\ [\text{C}] \quad \text{surface temperature} \tag{2}$$

$$T_\infty = 20\ [\text{C}] \quad \text{air temperature} \tag{3}$$

$$\bar{h} = 6.2\ [\text{W/m}^2\cdot\text{K}] \quad \text{heat transfer coefficient} \tag{4}$$

$$\dot{q}_{conv} = \bar{h} \cdot A_s \cdot (T_s - T_\infty) \quad \text{Newton's Law of Cooling} \tag{5}$$

$$\dot{w} = \dot{q}_{conv} \quad \text{energy balance} \tag{6}$$

PROBLEM 1.4

Equations

```
$UnitSystem J K Pa
```

Problem 1.4

$\dot{q}''_s = 375 \ [\mathrm{W/m^2}]$ solar flux (1)

$T_\infty = 25 \ [\mathrm{C}]$ ambient temperature (2)

$T_s = 42 \ [\mathrm{C}]$ surface temperature (3)

$\dot{q}''_s = \bar{h} \cdot (T_s - T_\infty)$ Newton‘s Law of Cooling (4)

PROBLEM 1.5

Equations

`$UnitSystem SI Mass J K Pa`

Problem 1.5

$\dot{w} = 60\ [\mathrm{W}]$ electricity provided to bulb (1)

$f_{light} = 0.1\ [\text{-}]$ fraction of power converted to light (2)

$A_s = 24.5\ \left[\mathrm{inch}^2\right] \cdot \left|6.4516 \times 10^{-4}\ \frac{\mathrm{m}^2}{\mathrm{inch}^2}\right|$ surface area (3)

$T_\infty = \mathrm{ConvertTemp}(F,\ K,\ 70\ [\mathrm{F}])$ temperature of surroundings (4)

$e = 0.9\ [\text{-}]$ emissivity of glass (5)

$\bar{h} = 12.5\ \left[\mathrm{W/m^2 \cdot K}\right]$ heat transfer coefficient (6)

$\dot{q}_{conv} = \bar{h} \cdot A_s \cdot (T_s - T_\infty)$ convection (7)

$\dot{q}_{rad} = sigma\# \cdot e \cdot A_s \cdot \left(T_s^4 - T_\infty^4\right)$ radiation (8)

$\dot{w} = \dot{q}_{conv} + \dot{q}_{rad} + f_{light} \cdot \dot{w}$ energy balance (9)

Radiation is very important

PROBLEM 1.6

Equations

```
$UnitSystem J K Pa
```

Problem 1.6

$$D = 1.5\ [\text{inch}] \cdot \left|0.0254\ \frac{\text{m}}{\text{inch}}\right| \quad \text{diameter of sphere} \tag{1}$$

$$T_s = \text{ConvertTemp}(C,\ K,\ 46\ [\text{C}]) \quad \text{surface temperature} \tag{2}$$

$$T_\infty = \text{ConvertTemp}(C,\ K,\ 25\ [\text{C}]) \quad \text{air temperature} \tag{3}$$

$$\dot{w} = 7.2\ [\text{W}] \quad \text{electrical power to heaters} \tag{4}$$

a.) heat transfer coefficient neglecting radiation

$$A_s = 4 \cdot \pi \cdot (D/2)^2 \quad \text{surface area} \tag{5}$$

$$\dot{w} = \dot{q}_{conv,a} \quad \text{energy balance} \tag{6}$$

$$\dot{q}_{conv,a} = \bar{h}_a \cdot A_s \cdot (T_s - T_\infty) \quad \text{Newton's Law of cooling} \tag{7}$$

b.) radiation

$$e = 0.21 \quad \text{emissivity} \tag{8}$$

$$T_{ext} = T_\infty \quad \text{external temperature} \tag{9}$$

$$\dot{q}_{rad} = A_s \cdot sigma\# \cdot e \cdot \left(T_s^4 - T_{ext}^4\right) \quad \text{radiation heat transfer} \tag{10}$$

c.) heat transfer coefficient including radiation

$$\dot{w} = \dot{q}_{rad} + \dot{q}_{conv,c} \quad \text{energy balance} \tag{11}$$

$$\dot{q}_{conv,c} = \bar{h}_c \cdot A_s \cdot (T_s - T_\infty) \quad \text{Newton's Law of cooling} \tag{12}$$

Solution

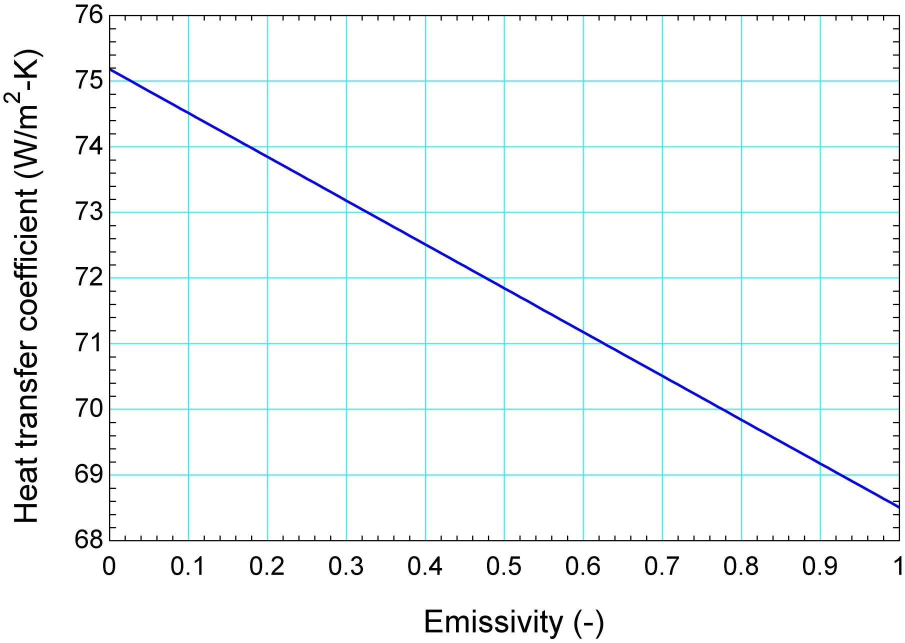
Variables in Main program

$A_s = 0.00456\ [\text{m}^2]$	$D = 0.0381\ [\text{m}]$
$e = 0.21\ [\text{-}]$	$\bar{h}_a = 75.18\ [\text{W/m}^2\cdot\text{K}]$
$\bar{h}_c = 73.78\ [\text{W/m}^2\cdot\text{K}]$	$\dot{q}_{conv,a} = 7.2\ [\text{W}]$
$\dot{q}_{conv,c} = 7.066\ [\text{W}]$	$\dot{q}_{rad} = 0.1343\ [\text{W}]$
$T_{ext} = 298.2\ [\text{K}]$	$T_\infty = 298.2\ [\text{K}]$
$T_s = 319.2\ [\text{K}]$	$\dot{w} = 7.2\ [\text{W}]$

Key Variables

$\dot{q}_{rad} = 0.1343\ [\text{W}]$	*b.) radiation heat transfer*
$\bar{h}_c = 73.78\ [\text{W/m}^2\cdot\text{K}]$	*c.) heat transfer coefficient including radiation*
$q''_s = -9999\ [\text{W/m}^2]$	

Plot Window 1: $Plot\ 1$



PROBLEM 1.7

Equations

```
$UnitSystem Mass J K Pa
```

$$W = 45\ [\text{mm}] \cdot \left|0.001\ \frac{\text{m}}{\text{mm}}\right| \quad \text{width} \tag{1}$$

$$H = 42.5\ [\text{mm}] \cdot \left|0.001\ \frac{\text{m}}{\text{mm}}\right| \quad \text{height} \tag{2}$$

$$\dot{g} = 95\ [\text{W}] \quad \text{rate of generation} \tag{3}$$

$$T_{\infty} = 25\ [\text{C}] \quad \text{ambient temperature} \tag{4}$$

a.) free convection

$$\dot{q}_{conv} = \dot{g} \quad \text{energy balance} \tag{5}$$

$$A_s = W \cdot H \quad \text{surface area} \tag{6}$$

$$\bar{h}_a = 6.0\ [\text{W/m}^2\cdot\text{K}] \quad \text{heat transfer coefficient} \tag{7}$$

$$\dot{q}_{conv} = \bar{h}_a \cdot A_s \cdot (T_{s,a} - T_{\infty}) \quad \text{Newton's Law of Cooling} \tag{8}$$

b.) advanced cooling

$$T_{s,b} = 42\ [\text{C}] \quad \text{surface temperature} \tag{9}$$

$$\dot{q}_{conv} = \bar{h}_b \cdot A_s \cdot (T_{s,b} - T_{\infty}) \quad \text{Newton's Law of Cooling} \tag{10}$$

Solution

Variables in Main program

$A_s = 0.001913\ [\text{m}^2]$	$\dot{g} = 95\ [\text{W}]$
$H = 0.0425\ [\text{m}]$	$\bar{h}_a = 6\ [\text{W/m}^2\cdot\text{K}]$
$\bar{h}_b = 2922\ [\text{W/m}^2\cdot\text{K}]$	$\dot{q}_{conv} = 95\ [\text{W}]$
$T_{\infty} = 25\ [\text{C}]$	$T_{s,a} = 8304\ [\text{C}]$
$T_{s,b} = 42\ [\text{C}]$	$W = 0.045\ [\text{m}]$

Key Variables

$T_{s,a} = 8304\ [\text{C}]$	*a.) Chip temperature*
$\bar{h}_b = 2922\ [\text{W/m}^2\cdot\text{K}]$	*b.) Heat transfer coefficient*

PROBLEM 1.8

Equations

```
$UnitSystem SI Mass J K Pa
```

$T_{in} = \text{ConvertTemp}(C,\ K,\ 25\ [\text{C}])$ air temperature on inside (1)

$T_{out} = \text{ConvertTemp}(C,\ K,\ -4\ [\text{C}])$ air temperature outdoors (2)

$\bar{h}_{in} = 6\ \left[\text{W/m}^2\cdot\text{K}\right]$ average heat transfer coefficient on inside surface (3)

$\bar{h}_{out} = 18\ \left[\text{W/m}^2\cdot\text{K}\right]$ average heat transfer coefficient on outside surface (4)

$e = 0.9\ [\text{-}]$ emissivity on both surfaces (5)

$k_g = 0.95\ [\text{W/m}\cdot\text{K}]$ glass conductivity (6)

$th_g = 6\ [\text{mm}] \cdot \left|0.001\ \frac{\text{m}}{\text{mm}}\right|$ glass thickness (7)

a.) neglect conduction through the glass

$\dot{q''}_{conv,in} = \bar{h}_{in} \cdot (T_{in} - T_g)$ convection heat flux to inside surface (8)

$\dot{q''}_{rad,in} = sigma\# \cdot e \cdot \left(T_{in}^4 - T_g^4\right)$ radiation heat flux to inside surface (9)

$\dot{q''}_{conv,out} = \bar{h}_{out} \cdot (T_g - T_{out})$ convection heat flux from outside surface (10)

$\dot{q''}_{rad,out} = sigma\# \cdot e \cdot \left(T_g^4 - T_{out}^4\right)$ radiation heat flux from outside surface (11)

$\dot{q''}_{conv,in} + \dot{q''}_{rad,in} = \dot{q''}_{conv,out} + \dot{q''}_{rad,out}$ energy balance (12)

$\dot{q''}_{total} = \dot{q''}_{conv,in} + \dot{q''}_{rad,in}$ total rate of heat transfer per area lost through the window (13)

b.) determine temp. difference across the glass

$\dot{q''}_{cond} = \dot{q''}_{total}$ conduction heat flux (14)

$k_g \cdot \frac{\Delta T_g}{th_g} = \dot{q''}_{cond}$ Fourier‘s Law (15)

Solution

Variables in Main program

$\Delta T_g = 1.339$ [K]	$e = 0.9$ [-]
$\bar{h}_{in} = 6$ $[\text{W/m}^2\cdot\text{K}]$	$\bar{h}_{out} = 18$ $[\text{W/m}^2\cdot\text{K}]$
$k_g = 0.95$ $[\text{W/m}\cdot\text{K}]$	$\dot{q''}_{cond} = 212$ $[\text{W/m}^2]$
$\dot{q''}_{conv,in} = 116.7$ $[\text{W/m}^2]$	$\dot{q''}_{conv,out} = 172$ $[\text{W/m}^2]$
$\dot{q''}_{rad,in} = 95.36$ $[\text{W/m}^2]$	$\dot{q''}_{rad,out} = 40.09$ $[\text{W/m}^2]$
$\dot{q''}_{total} = 212$ $[\text{W/m}^2]$	$th_g = 0.006$ [m]
$T_g = 278.7$ [K] {5.553 [C]}	$T_{in} = 298.2$ [K]
$T_{out} = 269.2$ [K]	

Key Variables

$T_g = 278.7$ [K] {5.553 [C]} *a.) glass temperature*
$\dot{q}''_{total} = 212$ [W/m^2] *a.) rate of heat loss per area*
$\Delta T_g = 1.339$ [K] *b.) temperature difference across glass*

PROBLEM 9

Equations

```
$UnitSystem SI Mass J K Pa
```

Problem 1.9

$L = 0.5$ [m] wall thickness (1)

$k = 0.72$ [W/m·K] conductivity (2)

$a_0 = 900$ [C] coefficient (3)

$a_1 = -300$ [C/m] (4)

$a_2 = -50$ [C/m^2] (5)

$dTdx_0 = a_1$ dTdx at x=0 (6)

$\dot{q''}_{cond,0} = -k \cdot dTdx_0$ rate of conduction heat flux at x=0 (7)

$dTdx_L = a_1 + 2 \cdot a_2 \cdot L$ dTdx at x=L (8)

$\dot{q''}_{cond,L} = -k \cdot dTdx_L$ rate of conduction heat flux at x=L (9)

$\dot{q''}_{in} = \dot{q''}_{cond,0} - \dot{q''}_{cond,L}$ net rate of heat transfer into the wall (10)

$T_\infty = 950$ [C] fluid temperature (11)

$T_0 = a_0$ surface temperature (12)

$\bar{h} \cdot (T_\infty - T_0) = \dot{q''}_{cond,0}$ Newton‘s Law of Cooling (13)

Solution

Variables in Main program

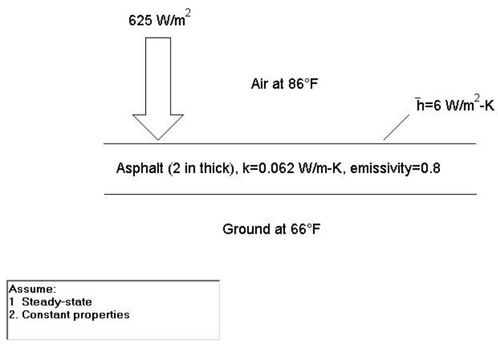
$a_0 = 900$ [C]	$a_1 = -300$ [C/m]
$a_2 = -50$ [C/m^2]	$dTdx_0 = -300$ [C/m]
$dTdx_L = -350$ [C/m]	$\bar{h} = 4.32$ [W/m^2·K]
$k = 0.72$ [W/m·K]	$L = 0.5$ [m]
$\dot{q''}_{cond,0} = 216$ [W/m^2]	$\dot{q''}_{cond,L} = 252$ [W/m^2]
$\dot{q''}_{in} = -36$ [W/m^2]	$T_0 = 900$ [C]
$T_\infty = 950$ [C]	

Key Variables

$\dot{q''}_{cond,0} = 216$ [W/m^2]	*a.) conduction at x=0*
$\dot{q''}_{cond,L} = 252$ [W/m^2]	*b.) conduction at x=L*
$\dot{q''}_{in} = -36$ [W/m^2]	*c.) net heat transfer into wall (is negative so net heat transfer is OUT of wall)*
$\bar{h} = 4.32$ [W/m^2·K]	*d.) heat transfer coefficient*

Problem 1.10

SOLUTION



Equations

```
$UnitSystem SI Mass J K Pa
```

$T_\infty = \text{ConvertTemp}(F,\ K,\ 86\ [\text{F}])$ surrounding temperature (1)

$\dot{q''}_s = 625\ \left[\text{W/m}^2\right]$ solar flux (2)

$\bar{h} = 6\ \left[\text{W/m}^2\cdot\text{K}\right]$ heat transfer coefficient (3)

$e = 0.8$ emissivity (4)

$k = 0.062\ [\text{W/m}\cdot\text{K}]$ thermal conductivity of asphalt (5)

$th = 2\ [\text{inch}] \cdot \left|0.0254\ \frac{\text{m}}{\text{inch}}\right|$ thickness of asphalt (6)

$T_g = \text{ConvertTemp}(F,\ K,\ 66\ [\text{F}])$ ground temperature (7)

$\dot{q''}_{conv} = \bar{h} \cdot (T_s - T_\infty)$ convection (8)

$\dot{q''}_{rad} = sigma\# \cdot e \cdot \left(T_s^4 - T_\infty^4\right)$ radiation (9)

$\dot{q''}_{cond} = k \cdot \frac{T_s - T_g}{th}$ conduction (10)

$\dot{q''}_s = \dot{q''}_{conv} + \dot{q''}_{rad} + \dot{q''}_{cond}$ energy balance (11)

Solution

Variables in Main program

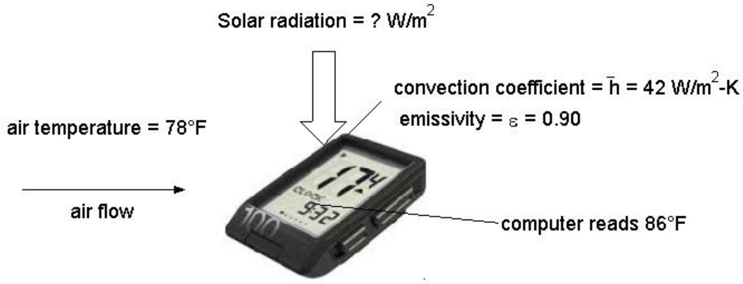
$e = 0.8$ [-]
$\bar{h} = 6$ $[\text{W/m}^{2K}]$
$k = 0.062$ [W/m-K]
$\dot{q''}_{cond} = 68.74$ $[\text{W/m}^2]$
$\dot{q''}_{conv} = 271.3$ $[\text{W/m}^2]$
$\dot{q''}_{rad} = 285$ $[\text{W/m}^2]$
$\dot{q''}_s = 625$ $[\text{W/m}^2]$
$th = 0.0508$ [m]
$T_g = 292$ [K]
$T_\infty = 303.2$ [K]
$T_s = 348.4$ [K] {167.4 [F]}

Key Variables

$T_s = 348.4$ [K] {167.4 [F]} *Asphalt surface temperature*

Problem 1.11

SOLUTION



Equations

`$UnitSystem SI Mass J K Pa`

$T_{\infty} = \text{ConvertTemp}(F,\ K,\ 78\ [\text{F}])$ air temperature (1)

$T_s = \text{ConvertTemp}(F,\ K,\ 86\ [\text{F}])$ temperature indicated on computer (2)

$\bar{h} = 42\ [\text{W/m}^2{\cdot}\text{K}]$ convection coefficient (3)

$e = 0.9$ emissivity (4)

$\dot{q''}_{conv} = \bar{h} \cdot (T_s - T_{\infty})$ convective heat transfer rate (5)

$\dot{q''}_{rad} = e \cdot sigma\# \cdot \left(T_s^4 - T_{\infty}^4\right)$ radiative heat transfer rate (6)

$\dot{q''}_s = \dot{q''}_{conv} + \dot{q''}_{rad}$ total heat transfer rate (7)

Assuming steady-state conditions, the rate of heat loss by convection and radiation must equal the absorbed solar flux.

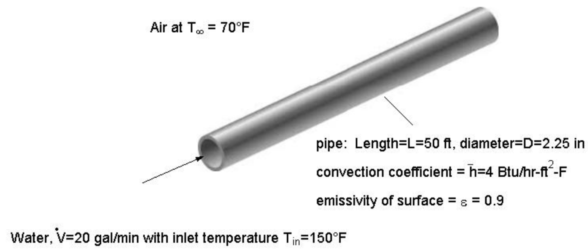
Solution

Variables in Main program

$e = 0.9$ [-]	$\bar{h} = 42$ [W/m^{2K}]
$\dot{q''}_{conv} = 186.7$ [W/m^2]	$\dot{q''}_{rad} = 24.73$ [W/m^2]
$\dot{q''}_s = 211.4$ [W/m^2]	$T_{\infty} = 298.7$ [K]
$T_s = 303.2$ [K] {86 [F]}	

Problem 1.12

SOLUTION



Equations

Assumptions:
1. Steady state
2. Constant properties
3. Small drop in temperature so heat loss can be estimated using the inlet temperature

```
$UnitSystem SI MASS RAD PA  K J
```

Inputs

$$\dot{V} = 20\ [\text{gal/min}] \cdot \left| 6.30902 \times 10^{-5}\ \frac{\text{m}^3/\text{s}}{\text{gal/min}} \right| \quad \text{volumetric flow rate} \tag{1}$$

$$D = 2.25\ [\text{inch}] \cdot \left| 0.0254\ \frac{\text{m}}{\text{inch}} \right| \quad \text{diameter} \tag{2}$$

$$L = 50\ [\text{ft}] \cdot \left| 0.3048\ \frac{\text{m}}{\text{ft}} \right| \quad \text{length} \tag{3}$$

$$\rho = 1000\ \left[\text{kg/m}^3\right] \quad \text{density of water} \tag{4}$$

$$\mu = 0.001\ [\text{kg/m}\cdot\text{s}] \quad \text{viscosity of water} \tag{5}$$

$$c = 4200\ [\text{J/kg}\cdot\text{K}] \quad \text{specific heat capacity} \tag{6}$$

$$e = 0.9\ [\ ] \quad \text{emissivity} \tag{7}$$

$$f = 0.016\ [\ ] \quad \text{friction factor} \tag{8}$$

$$T_{in} = \text{ConvertTemp}(F,\ K,\ 150\ [\text{F}]) \quad \text{inlet temperature} \tag{9}$$

$$T_{\infty} = \text{ConvertTemp}(F,\ K,\ 70\ [\text{F}]) \quad \text{ambient temperature} \tag{10}$$

$$\bar{h} = 4\ \left[\text{Btu/ft}^2\cdot\text{hr}\cdot\text{R}\right] \cdot \left| 5.67826409\ \frac{\text{W/m}^2\cdot\text{K}}{\text{Btu/ft}^2\cdot\text{hr}\cdot\text{R}} \right| \quad \text{heat transfer coefficient} \tag{11}$$

$$A_c = \pi \cdot \frac{D^2}{4} \quad \text{cross-sectional area of pipe} \tag{12}$$

$$u_m = \dot{V}/A_c \quad \text{average velocity} \tag{13}$$

$$\Delta P = \left(\rho \cdot \frac{u_m^2}{2}\right) \cdot (f \cdot L/D) \quad \text{pressure drop} \tag{14}$$

$A_s = \pi \cdot D \cdot L$ surface area (15)

$\dot{q}_{conv} = \bar{h} \cdot A_s \cdot (T_{in} - T_\infty)$ convection (16)

$\dot{q}_{rad} = sigma\# \cdot e \cdot A_s \cdot \left(T_{in}^4 - T_\infty^4\right)$ radiation (17)

$\dot{m} = u_m \cdot A_c \cdot rho$ mass flow rate (18)

$\dot{m} \cdot c \cdot \Delta T = \dot{q}_{conv} + \dot{q}_{rad}$ energy balance (19)

Solution

Variables in Main program

$A_c = 0.002565$ $\left[\text{m}^2\right]$	$A_s = 2.736$ $\left[\text{m}^2\right]$
$c = 4200$ [J/kg-K]	$D = 0.05715$ [m]
$\Delta P = 516.2$ [Pa] {0.07486 [psi]}	$\Delta T = 0.6704$ [K] {1.207 [R]}
$e = 0.9$ [-]	$f = 0.016$ [-]
$\bar{h} = 22.71$ $\left[\text{W/m}^{2\text{K}}\right]$	$L = 15.24$ [m]
$\mu = 0.001$ [kg/m-s]	$\dot{m} = 1.262$ [kg/s]
$\dot{q}_{conv} = 2762$ [W]	$\dot{q}_{rad} = 790.8$ [W]
$\rho = 1000$ $\left[\text{kg/m}^3\right]$	$T_{in} = 338.7$ [K]
$T_\infty = 294.3$ [K]	$u_m = 0.4919$ [m/s]
$\dot{V} = 0.001262$ $\left[\text{m}^3\text{/s}\right]$	

Key Variables

$\Delta P = 516.2$ [Pa] {0.07486 [psi]} *a.) pressure drop*

$\Delta T = 0.6704$ [K] {1.207 [R]} *b.) temperature change. Note that this small temperature changes justifies assumption 3.*

Problem 1.13

SOLUTION

Equations

```
$UnitSystem SI Mass J K Pa
```

$T_\infty = \text{ConvertTemp}(C,\ K,\ 20\ [\text{C}])$ ambient temperature (1)

$\dot{w}_e = 1.2\ [\text{kW}] \cdot \left|1000\ \frac{\text{W}}{\text{kW}}\right|$ burner power (2)

$D = 8\ [\text{inch}] \cdot \left|0.0254\ \frac{\text{m}}{\text{inch}}\right|$ diameter (3)

$H = 6\ [\text{inch}] \cdot \left|0.0254\ \frac{\text{m}}{\text{inch}}\right|$ height (4)

$\bar{h} = 8\ \left[\text{W/m}^2{\cdot}\text{K}\right]$ heat transfer coefficient (5)

$e = 0.7$ emissivity of pan surface (6)

$T_s = \text{T}_{\text{sat}}\left(Water,\ \text{P} = Po\#\right)$ temperature of pan surface (7)

$A_s = \pi \cdot D \cdot H + \pi \cdot (D/2)^2$ surface area (8)

$\dot{q}_{conv} = \bar{h} \cdot A_s \cdot (T_s - T_\infty)$ convection heat transfer rate (9)

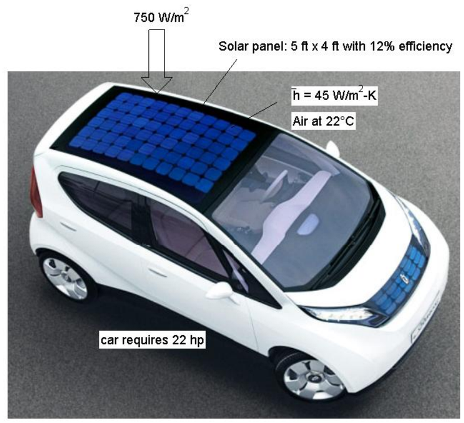
$\dot{q}_{rad} = e \cdot sigma\# \cdot A_s \cdot \left(T_s^4 - T_\infty^4\right)$ radiation heat transfer rate (10)

$\eta = \frac{\dot{W}_e - \dot{q}_{conv} - \dot{q}_{rad}}{\dot{w}_e}$ efficiency of burner (11)

Solution

$A_s = 0.1297$ [m^2]	$D = 0.2032$ [m]	$e = 0.7$ [-]	$\eta = 0.8794$ [-]
$H = 0.1524$ [m]	$\bar{h} = 8$ [W/m^2-·K]	$\dot{q}_{conv} = 82.99$ [W]	$\dot{q}_{rad} = 61.77$ [W]
$T_\infty = 293.2$ [K]	$T_s = 373.1$ [K]	$\dot{w}_e = 1200$ [W]	

PROBLEM 1.14



Equations

Assumptions:
Steady state
Constant conditions as specified in problem statement
No radiation losses

```
$UnitSystem SI MASS RAD PA  C J
```

$$L = 5 \ [\text{ft}] \cdot \left| 0.3048 \ \frac{\text{m}}{\text{ft}} \right| \quad \text{length of panel} \tag{1}$$

$$W = 4 \ [\text{ft}] \cdot \left| 0.3048 \ \frac{\text{m}}{\text{ft}} \right| \quad \text{width of panel} \tag{2}$$

$$\dot{q}''_s = 750 \ \left[\text{W/m}^2\right] \quad \text{solar flux} \tag{3}$$

```
$ifnot ParametricTable
```

$$\eta_p = 0.12 \ [\ ] \quad \text{efficiency of solar panel} \tag{4}$$

```
$endif
```

$$\dot{w}_{car} = 22 \ [\text{hp}] \cdot \left| 745.7 \ \frac{\text{W}}{\text{hp}} \right| \quad \text{cruising power required} \tag{5}$$

$$\bar{h} = 45 \ \left[\text{W/m}^2\text{·K}\right] \quad \text{heat transfer coefficient} \tag{6}$$

$$T_\infty = 22 \ [\text{C}] \quad \text{ambient temperature} \tag{7}$$

$$A_{panel} = L \cdot W \quad \text{panel area} \tag{8}$$

$$\dot{w}_{panel} = A_{panel} \cdot \dot{q}''_s \cdot \eta_p \quad \text{power received from solar panel} \tag{9}$$

$$f_{power} = \dot{w}_{panel} / \dot{w}_{car} \quad \text{fraction of power provided} \tag{10}$$

$$\dot{q}''_s \cdot A_{panel} = \dot{w}_{panel} + \dot{q}_{conv} \quad \text{first law} \tag{11}$$

$\dot{q}_{conv} = A_{panel} \cdot \bar{h} \cdot (T_s - T_\infty)$ Newton's Law of Cooling (12)

$t_{sit} = 6 \ [\text{hr}] \cdot \left| 3600 \ \frac{\text{s}}{\text{hr}} \right|$ sitting time (13)

$w_{panel} = t_{sit} \cdot \dot{w}_{panel}$ energy collected by panel (14)

$t_{drive} = 30 \ [\text{min}] \cdot \left| 60 \ \frac{\text{s}}{\text{min}} \right|$ driving time (15)

$w_{car} = t_{drive} \cdot \dot{w}_{car}$ energy required by car (16)

$f_{energy} = w_{panel}/w_{car}$ fraction of energy provided (17)

Solution

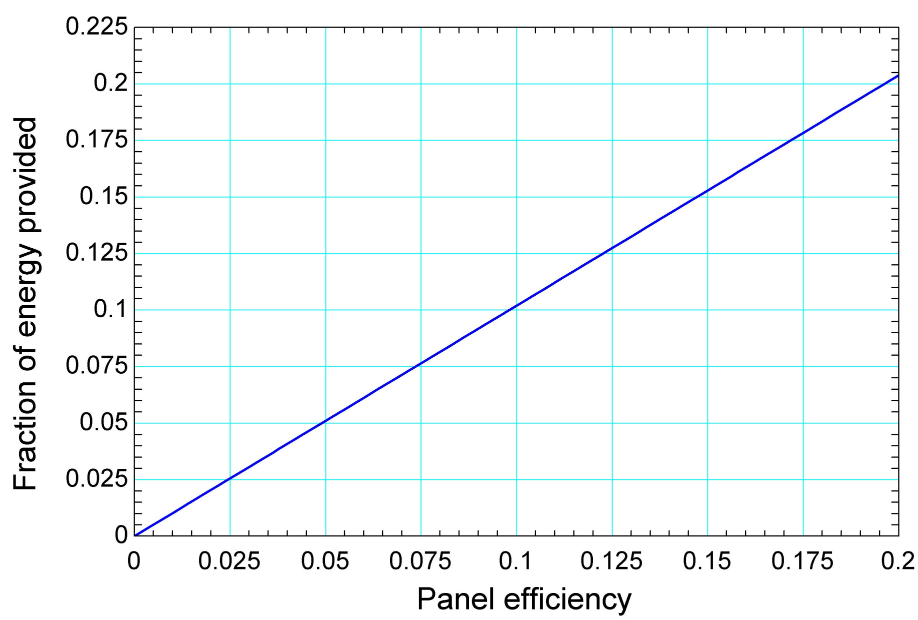
Variables in Main program

$A_{panel} = 1.858$ [m^2]
$f_{power} = 0.01019$ [-] {1.019 [%]}
$\dot{q}_{conv} = 1226$ [W]
$T_\infty = 22$ [C]
$W = 1.219$ [m]
$\dot{w}_{panel} = 167.2$ [W]
$\eta_p = 0.12$ [-]
$\bar{h} = 45$ [W/m^2·K]
$\dot{q}''_s = 750$ [W/m^2]
$T_s = 36.67$ [C] {98 [F]}
$w_{car} = 2.953 \times 10^7$ [J] {8.203 [kW·hr]}
$w_{panel} = 3.612 \times 10^6$ [J] {1.003 [kW·hr]}
$f_{energy} = 0.1223$ [-] {12.23 [%]}
$L = 1.524$ [m]
$t_{drive} = 1800$ [s]
$t_{sit} = 21600$ [s]
$\dot{w}_{car} = 16405$ [W]

Key Variables

$\dot{w}_{panel} = 167.2$ [W] *a.) power produced by panel*
$f_{power} = 0.01019$ [-] {1.019 [%]} *b) fraction of power reqd. by car produced by panel*
$T_s = 36.67$ [C] {98 [F]} *c.) panel temperature*
$w_{panel} = 3.612 \times 10^6$ [J] {1.003 [kW·hr]} *d.) Energy produced by panel per day*
$w_{car} = 2.953 \times 10^7$ [J] {8.203 [kW·hr]} *e.) energy reqd. by car per day*
$f_{energy} = 0.1223$ [-] {12.23 [%]} *f.) fraction of energy provided by panel*

Plot Window 1: *Plot 1*



Problem 1.15

SOLUTION

Equations

$$HDD = 7800 \; [\mathrm{F \cdot days}] \cdot \left| 0.55555556 \, \frac{\Delta \mathrm{K}}{\Delta \mathrm{F}} \right| \quad \text{heating degree days in K-days} \tag{1}$$

$$A_{wall} = 600 \; \left[\mathrm{ft}^2\right] \cdot \left| 0.09290304 \, \frac{\mathrm{m}^2}{\mathrm{ft}^2} \right| \quad \text{area of walls} \tag{2}$$

$$R = 18 \; \left[\mathrm{ft}^2 \cdot \mathrm{F} \cdot \mathrm{hr/Btu}\right] \cdot \left| 0.17611016 \, \frac{\mathrm{m}^2 \cdot \mathrm{K/W}}{\mathrm{ft}^2 \cdot \mathrm{F} \cdot \mathrm{hr/Btu}} \right| \tag{3}$$

$$Q = A_{wall}/R \cdot HDD \cdot \left| 0.0864 \, \frac{\mathrm{MJ}}{\mathrm{W \cdot day}} \right| \quad \text{total heat loss in winter} \tag{4}$$

$$GasUnitCost = 0.75 \; [\$/\mathrm{therm}] \cdot \left| 0.00947817 \, \frac{\$/\mathrm{MJ}}{\$/\mathrm{therm}} \right| \quad \text{unit gas cost} \tag{5}$$

$$CostwithGas = Q \cdot GasUnitCost \quad \text{total cost to heat house with gas} \tag{6}$$

$$ElectricityUnitCost = 0.136 \; [\$/\mathrm{kW \cdot hr}] \cdot \left| 0.277777778 \, \frac{\$/\mathrm{MJ}}{\$/\mathrm{kW \cdot hr}} \right| \quad \text{unit electricity cost} \tag{7}$$

$$CostwithElectricity = Q \cdot ElectricityUnitCost \quad \text{total cost to heat house with electricity} \tag{8}$$

Solution

Variables in Main program

$A_{wall} = 55.74 \; [\mathrm{m}^2]$
$CostwithGas = 46.8 \; [\$]$
$GasUnitCost = 0.007109 \; [\$/\mathrm{MJ}]$
$Q = 6584 \; [\mathrm{MJ}]$
$CostwithElectricity = 248.7 \; [\$]$
$ElectricityUnitCost = 0.03778 \; [\$/\mathrm{MJ}]$
$HDD = 4333 \; [\mathrm{K}^{\mathrm{days}}]$
$R = 3.17 \; [\mathrm{m}^{2\mathrm{K}}/\mathrm{W}]$

Key Variables

$Q = 6584 \; [\mathrm{MJ}]$ *a) amount of heat required*
$CostwithGas = 46.8 \; [\$]$ *b) heating cost with gas*
$CostwithElectricity = 248.7 \; [\$]$ *c) heating cost with electricity*

Problem 1.16

Equations

Assumptions:
1. The metal tank wall offers no thermal resistance
2. The temperature is uniform at all locations in the tank
3. There are no losses from the bottom of the tank
4. Heat loss is calculated assuming steady-state

```
$TabStops 0.2 4 in
$UnitSystem SI C Pa J
```

$V = 50\ [\text{gal}] \cdot \left|0.003785412\ \frac{\text{m}^3}{\text{gal}}\right|$ volume of tank (1)

$D = 18\ [\text{in}] \cdot \left|0.0254\ \frac{\text{m}}{\text{in}}\right|$ diameter of tank (2)

$V = \pi \cdot \frac{D^2}{4} \cdot L$ determine height of tank (3)

$T_w = 52\ [\text{C}]$ temperature of water (4)

$T_b = 20\ [\text{C}]$ temperature of basement (5)

$T_{cold} = 10\ [\text{C}]$ temperature of supply water (6)

$A = \pi \cdot D \cdot L + \pi \cdot \frac{D^2}{4}$ surface area of side and top of tank (7)

$\bar{h} = 4.5\ \left[\text{W/m}^2 \cdot \text{K}\right]$ heat transfer coefficient (8)

a)

$\dot{Q}_{loss} = {}^{-}h \cdot A \cdot (T_w - T_b)$ rate of heat loss from surface to the basement = 262.1 W (9)

b)

$Q_{loss,day} = \dot{Q}_{loss} \cdot 24\ [\text{hr}] \cdot \left|3600\ \frac{\text{s}}{\text{hr}}\right|$ energy loss in one day (10)

$m_{day} = 42\ [\text{gal}] \cdot \left|0.003785412\ \frac{\text{m}^3}{\text{gal}}\right| \cdot \rho\left(water,\ \text{T} = \text{T}_\text{w},\ \text{P} = \text{Po\#}\right)$ mass of water used in one day (11)

$Q_{use,day} = m_{day} \cdot \left(\text{h}\left(Water,\ \text{T} = T_w,\ \text{P} = Po\#\right) - \text{h}\left(Water,\ \text{T} = T_{cold},\ \text{P} = Po\#\right)\right)$ (12)

energy needed to heat water each day excluding losses

$Loss_{fraction} = \frac{Q_{loss,day}}{(Q_{loss,day} + Q_{use,day})}$ fraction of total energy attributable to losses = 0.451 (13)

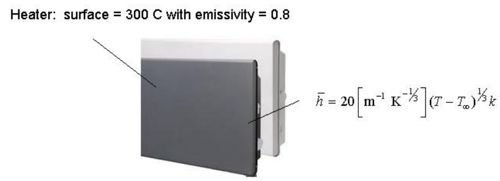
c)

$ElecCost = 0.15\ [\$/\text{kW} \cdot \text{hr}]$ electrical cost (14)

$Cost = Q_{loss,day} \cdot 365 \cdot \left|2.77778 \times 10^{-7}\ \frac{\text{kW} \cdot \text{hr}}{\text{J}}\right| \cdot ElecCost$ annual cost of losses= $344.4 (15)

The rate of heat loss can be substantially reduced by adding insulation to the walls and top of the tank.

Problem 1.17



Equations

Assume steady state and constant properties

$e = 0.8$ [-] emissivity (1)

$T = \text{ConvertTemp}(C,\ K,\ 300\ [\text{C}])$ temperature (2)

$T_\infty = \text{ConvertTemp}(C,\ K,\ 20\ [\text{C}])$ ambient temperature (3)

$T_{film} = \dfrac{T + T_\infty}{2}$ film temperature (4)

$k = \text{k}\left(Air,\ \text{T} = T_{film}\right)$ conductivity (5)

$\bar{h} = 20\ \left[1/\text{m}\cdot\text{K}^{1/3}\right] \cdot (T - T_\infty)^{1/3} \cdot k$ heat transfer coefficient (6)

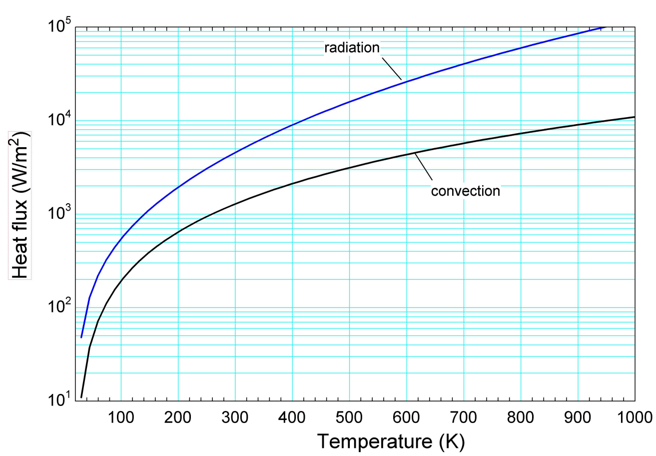
$\dot{q}''_{conv} = \bar{h} \cdot (T - T_\infty)$ heat flux due to convection (7)

$\dot{q}''_{rad} = e \cdot sigma\# \cdot \left(T^4 - T_\infty^4\right)$ heat flux due to radiation (8)

Solution

$e = 0.8$ [-]	$\bar{h} = 4.594$ $[\text{W/m}^{2\text{K}}]$	$k = 0.03511$ [W/m-K]	$\dot{q}''_{conv} = 1286$ $[\text{W/m}^2]$
$\dot{q}''_{rad} = 4560$ $[\text{W/m}^2]$	$T = 573.2$ [K] {300 [C]}	$T_{film} = 433.2$ [K]	$T_\infty = 293.2$ [K]

Plot Window 1: *Plot 1*



Problem **1.18**

$\dot{q}_{rad} = A_s\,\sigma\,\varepsilon\left(T_s^4 - T_{ext}^4\right)$ "rate of radiative heat transfer"

$\dot{q}_{conv} = \bar{h}\,A_s\left(T_s - T_\infty\right)$ "rate of convective heat transfer"

Assume: $T_s = T_\infty$

Set: $\dot{q}_{rad} = \dot{q}_{conv}$

Solve for h: $h = \sigma\varepsilon\left(T_s^2 + T_{ext}^2\right)\left(T_s + T_{ext}\right)$

Equations

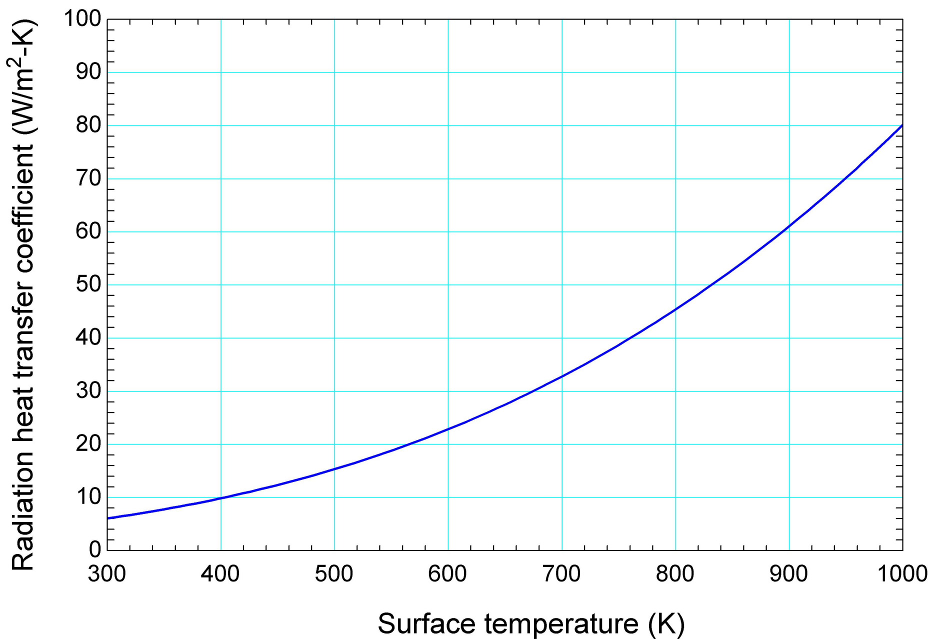
```
$UnitSystem SI Mass J K Pa
```

$T_{ext} = 298.1$ [K] external surroundings temperature (1)

$e = 1$ emissivity (2)

$\bar{h}_{rad} = sigma\# \cdot e \cdot \left(T_s^2 + T_{ext}^2\right) \cdot \left(T_s + T_{ext}\right)$ radiation heat transfer coefficient (3)

Plot Window 1: *Plot* 1



Problem 1.19

Equations

```
$UnitSystem SI C J
$TabStops 0.2 4 in
```

function $\bar{h}(T_s)$ function to determine the heat transfer coefficient (1)

$\bar{h} = 285\ [\text{W/m}^2\text{·K}] + 10\ [\text{W/m}^2\text{·K}^2] \cdot T_s$ (2)

end (3)

$D = 0.015\ [\text{m}]$ diameter of coil (4)

$L = 0.6\ [\text{m}]$ total length of coil (5)

$A = \pi \cdot D \cdot L$ surface area of each coil (6)

$T_{ini} = 10\ [\text{C}]$ initial water temperature (7)

$T_{final} = 55\ [\text{C}]$ final water temperature (8)

$\dot{Q} = 2500\ [\text{W}]$ power of each heating element (9)

$V = 40\ [\text{gal}] \cdot \left| 0.003785412\ \frac{\text{m}^3}{\text{gal}} \right|$ volume of water (10)

$m = V \cdot \rho\left(Water,\ \text{T} = 35\ [\text{C}],\ \text{P} = Po\#\right)$ mass of water (11)

$Q = m \cdot \left(\text{h}\left(Water,\ \text{T} = T_{final},\ \text{P} = Po\#\right) - \text{h}\left(Water,\ \text{T} = T_{ini},\ \text{P} = Po\#\right)\right)$ energy balance on water (12)

a)

$Q = 2 \cdot \dot{Q} \cdot time$ determines time for two heating elements (13)

b)

$\dot{Q} = \bar{h}(T_{sb}) \cdot A \cdot (T_{sb} - T_{ini})$ Newton‘s Law of Cooling at the start when the water is at 10°C (14)

c)

$\dot{Q} = \bar{h}(T_{sc}) \cdot A \cdot (T_{sc} - T_{final})$ Newton‘s Law of Cooling at the start when the water is at 55°C (15)

d) Vary time in Parametric table to produce plot. $time_2$ is varied between 0 and time.

```
$if ParametricTable
```

$\dot{Q} = \bar{h}(T_s) \cdot A \cdot (T_s - T_w)$ convective heat transfer rate when water is at T_w (16)

$\dot{Q} \cdot 2 \cdot time_2 = m \cdot (\mathrm{h}(Water,\ \mathrm{T} = T_w,\ \mathrm{P} = Po\#) - \mathrm{h}(Water,\ \mathrm{T} = T_{ini},\ \mathrm{P} = Po\#))$ energy balance on water(17)

```
$endif
```

See \\EES_{plot}

Solution

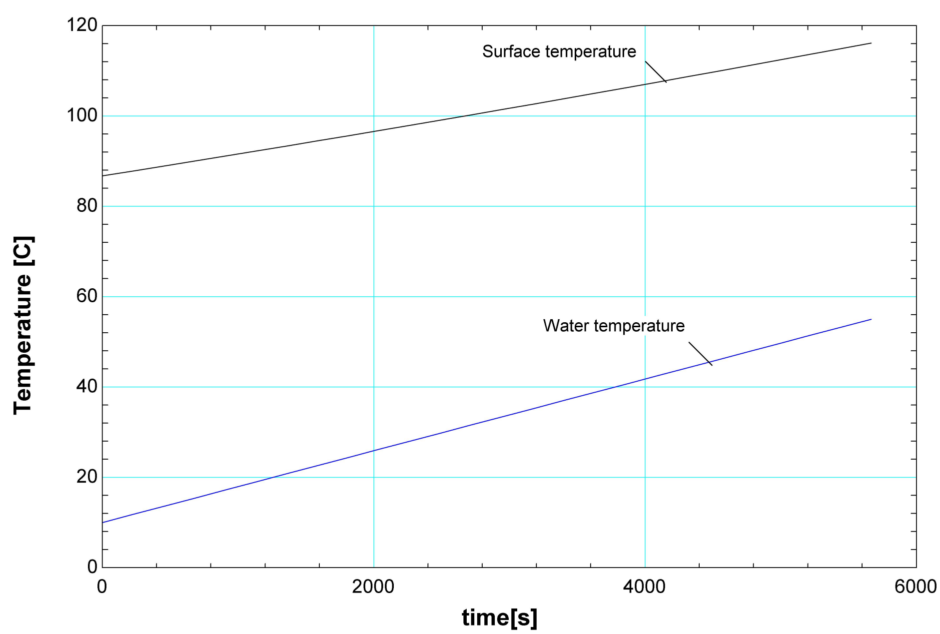
Variables in Main program

$A = 0.02827$ [m^2] $D = 0.015$ [m]
$L = 0.6$ [m] $m = 150.5$ [kg]
$Q = 2.833 \times 10^7$ [J] $\dot{Q} = 2500$ [W]
$time = 5666$ [s] {1.574 [hr]} $time_2 = 5666$ [s]
$T_{final} = 55$ [C] $T_{ini} = 10$ [C]
$T_s = 116.1$ [C] $T_{sb} = 86.73$ [C]
$T_{sc} = 116.1$ [C] $T_w = 55$ [C]
$V = 0.1514$ [m^3]

Key Variables

$time = 5666$ [s] {1.574 [hr]} *a) time required to heat the tank*
$T_{sb} = 86.73$ [C] *b) temperature of the heater surface shortly after heating is initiated*
$T_{sc} = 116.1$ [C] *c) temperature of the heating element when the water reaches 55°C*

Plot Window 1: *Plot 1*



PROBLEM 1.20

Equations

Problem 1.20

```
$UnitSystem SI J K Pa Radian
```

Inputs

$\dot{q} = 0.36\ [W]$ Steady-state power dissipation (1)

$T_w = \text{ConvertTemp}(C,\ K,\ 230\ [C])$ wire temperature (2)

$T_\infty = \text{ConvertTemp}(C,\ K,\ 24\ [C])$ ambient temperature (3)

$D = 0.125\ [mm] \cdot \left| 0.001\ \frac{m}{mm} \right|$ diameter (4)

$L = 6.5\ [mm] \cdot \left| 0.001\ \frac{m}{mm} \right|$ length (5)

$Po = 1\ [atm] \cdot \left| 101325\ \frac{Pa}{atm} \right|$ pressure (6)

a)

$\dot{q}_{conv} = \dot{q}$ energy balance on wire (7)

$\dot{q} = \bar{h} \cdot A_s \cdot (T_w - T_\infty)$ Newton's Law of cooling (8)

$A_s = \pi \cdot D \cdot L$ surface area (9)

b)

$T_{film} = \frac{T_w + T_\infty}{2}$ film temperature (10)

$\rho = \rho\,(Air,\ T = T_{film},\ P = Po)$ density (11)

$k = k\,(Air,\ T = T_{film})$ thermal conductivity (12)

$\mu = \mu\,(Air,\ T = T_{film})$ viscosity (13)

$Re = \rho \cdot u_{inf} \cdot D / mu$ defn of Reynold's number (14)

$\bar{h} \cdot D / k = 0.82 \cdot Re^{0.385}$ known relation (15)

Solution

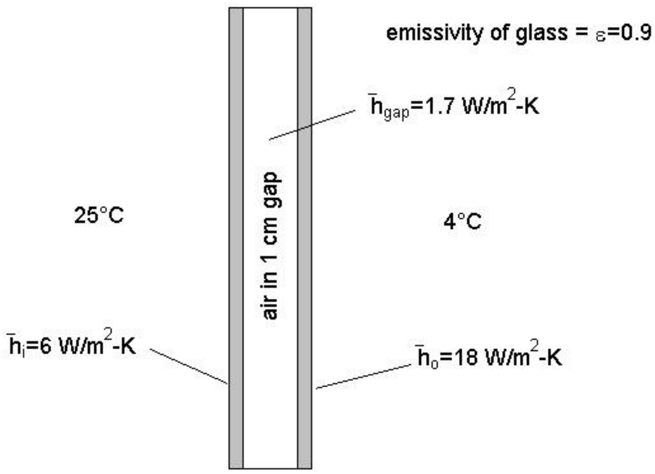
Variables in Main program

$A_s = 0.000002553$ [m^2]	$D = 0.000125$ [m]
$\bar{h} = 684.6$ [W/m^2·K]	$k = 0.03284$ [W/m·K]
$L = 0.0065$ [m]	$\mu = 0.00002293$ [kg/m·s]
$Po = 101325$ [Pa]	$\dot{q} = 0.36$ [W]
$\dot{q}_{conv} = 0.36$ [W]	$Re = 20.15$ [-]
$\rho = 0.8821$ [kg/m^3]	$T_{film} = 400.2$ [K]
$T_\infty = 297.2$ [K]	$T_w = 503.2$ [K]
$u_{inf} = 4.189$ [m/s]	

Key Variables

$\bar{h} = 684.6$ $[\mathrm{W/m^2{\cdot}K}]$ *a.) heat transfer coefficient*
$u_{inf} = 4.189$ $[\mathrm{m/s}]$ *b.) air velocity*

Problem 1.21



Assumptions:

1. Each pane of glass is at a uniform temperature
2. Steady-state
3. Constant properties

Equations

```
$UnitSystem SI Mass J K Pa
```

$T_{in} = \text{ConvertTemp}(C,\ K,\ 25\ [\text{C}])$ air temperature on inside (1)

$T_{out} = \text{ConvertTemp}(C,\ K,\ -4\ [\text{C}])$ air temperature outdoors (2)

$\bar{h}_{in} = 6\ \left[\text{W/m}^2\cdot\text{K}\right]$ average heat transfer coefficient on inside surface (3)

$\bar{h}_{out} = 18\ \left[\text{W/m}^2\cdot\text{K}\right]$ average heat transfer coefficient on outside surface (4)

$\bar{h}_{gap} = 1.7\ \left[\text{W/m}^2\cdot\text{K}\right]$ average heat transfer coefficient in the gap (5)

$e = 0.9\ [-]$ emissivity on both surfaces (6)

$\dot{q}''_{conv,in} = \bar{h}_{in}\cdot(T_{in} - T_{g,in})$ convection heat flux from inside to inner pane (7)

$\dot{q}''_{rad,in} = sigma\# \cdot e \cdot \left(T_{in}^4 - T_{g,in}^4\right)$ radiation heat flux from inside to inner pane (8)

$$\dot{q}''_{rad,gap} = sigma\# \cdot \frac{T_{g,in}^4 - T_{g,out}^4}{(2/e-1)}$$ radiation heat flux from inner pane to outer pane (9)

$$\dot{q''}_{conv,gap} = \bar{h}_{gap} \cdot (T_{g,in} - T_{g,out})$$ convection heat flux from inner pane to outer pane (10)

$$\dot{q''}_{conv,out} = \bar{h}_{out} \cdot (T_{g,out} - T_{out})$$ convection heat flux from outer pane to outside (11)

$$\dot{q''}_{rad,out} = sigma\# \cdot e \cdot \left(T_{g,out}^4 - T_{out}^4\right)$$ radiation heat flux from outer pane to outside (12)

$$\dot{q''}_{conv,in} + \dot{q''}_{rad,in} = \dot{q''}_{rad,gap} + \dot{q''}_{conv,gap}$$ energy balance on inner pane (13)

$$\dot{q''}_{rad,gap} + \dot{q''}_{conv,gap} = \dot{q''}_{conv,out} + \dot{q''}_{rad,out}$$ energy balance on outer pane (14)

$$\dot{q''}_{total} = \dot{q''}_{conv,in} + \dot{q''}_{rad,in}$$ total rate of heat transfer per area lost through the window (15)

Solution

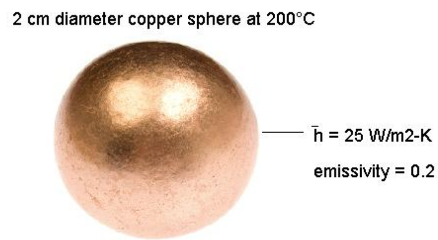
Variables in Main program

$e = 0.9$ [-]
$\bar{h}_{gap} = 1.7$ [W/m2K]
$\bar{h}_{in} = 6$ [W/m2K]
$\bar{h}_{out} = 18$ [W/m2K]
$\dot{q''}_{conv,gap} = 27.58$ $[\mathrm{W/m^2}]$
$\dot{q''}_{conv,in} = 50.88$ $[\mathrm{W/m^2}]$
$\dot{q''}_{conv,out} = 77.33$ $[\mathrm{W/m^2}]$
$\dot{q''}_{rad,gap} = 67.26$ $[\mathrm{W/m^2}]$
$\dot{q''}_{rad,in} = 43.96$ $[\mathrm{W/m^2}]$
$\dot{q''}_{rad,out} = 17.51$ $[\mathrm{W/m^2}]$
$\dot{q''}_{total} = 94.84$ $[\mathrm{W/m^2}]$
$T_{g,in} = 289.7$ [K] {16.52 [C]}
$T_{g,out} = 273.4$ [K] {0.2959 [C]}
$T_{in} = 298.2$ [K]
$T_{out} = 269.2$ [K]

Key Variables

$T_{g,in} = 289.7$ [K] {16.52 [C]} *inner pane temperature*
$T_{g,out} = 273.4$ [K] {0.2959 [C]} *outer pane temperature*
$\dot{q''}_{total} = 94.84$ $[\mathrm{W/m^2}]$ *total heat flux*

Problem 1.22



Equations

Assumption steady-state conditions

$\rho = 8958 \ \left[\text{kg/m}^3\right]$ density of copper (1)

$c = 389.3 \ [\text{J/kg}\cdot\text{K}]$ specific heat capacity of copper (2)

$e = 0.2 \ [-]$ emissivity (3)

$D = 2 \ [\text{cm}] \cdot \left| 0.01 \ \frac{\text{m}}{\text{cm}} \right|$ diameter (4)

$\bar{h} = 25 \ \left[\text{W/m}^2\cdot\text{K}\right]$ heat transfer coefficient (5)

$T = \text{ConvertTemp}(C, \ K, \ 200 \ [\text{C}])$ temperature (6)

$T_\infty = \text{ConvertTemp}(C, \ K, \ 20 \ [\text{C}])$ surrounding temperature (7)

$A_s = 4 \cdot \pi \cdot (D/2)^2$ surface area (8)

$V = 4 \cdot \pi \cdot \frac{(D/2)^3}{3}$ volume (9)

$\dot{q}_{rad} = sigma\# \cdot e \cdot A_s \cdot \left(T^4 - T_\infty^4\right)$ radiation heat transfer rate (10)

$\dot{q}_{conv} = \bar{h} \cdot A_s \cdot (T - T_\infty)$ convection heat transfer rate (11)

$0 = \dot{q}_{rad} + \dot{q}_{conv} + dUdt$ energy balance (12)

$dUdt = \rho \cdot c \cdot V \cdot dTdt$ incompressible (13)

Solution

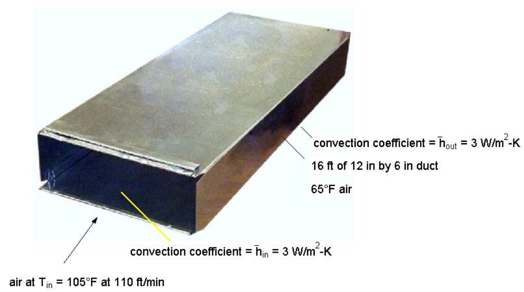
Variables in Main program

$A_s = 0.001257$ $\left[\text{m}^2\right]$	$c = 389.3$ [J/kg-K]	$D = 0.02$ [m]	$dTdt = -0.4288$ [K/s]
$dUdt = -6.264$ [J/s]	$e = 0.2$ [-]	$\bar{h} = 25$ $\left[\text{W/m}^{2\text{K}}\right]$	$\dot{q}_{conv} = 5.655$ [W]
$\dot{q}_{rad} = 0.609$ [W]	$\rho = 8958$ $\left[\text{kg/m}^3\right]$	$T = 473.2$ [K]	$T_\infty = 293.2$ [K]
$V = 0.000004189$ $\left[\text{m}^3\right]$			

Key Variables

$dTdt = -0.4288$ [K/s] *rate of temperature change with time*

Problem 1.23



Equations

Assmptions:
1. Steady-state
2. Small change in temperature of air so heat loss can be accurately calculated using the inlet temperature

```
$UnitSystem SI Mass J K Pa
```

$T_f = \text{ConvertTemp}(F,\ K,\ 105\ [\text{F}])$ temperature of air in duct (1)

$L = 16\ [\text{ft}] \cdot \left| 0.3048\ \frac{\text{m}}{\text{ft}} \right|$ length of duct (2)

$W = 12\ [\text{inch}] \cdot \left| 0.0254\ \frac{\text{m}}{\text{inch}} \right|$ width of duct (3)

$H = 6\ [\text{inch}] \cdot \left| 0.0254\ \frac{\text{m}}{\text{inch}} \right|$ height of duct (4)

$u_m = 110\ [\text{ft/min}] \cdot \left| 0.00508\ \frac{\text{m/s}}{\text{ft/min}} \right|$ mean velocity (5)

$\bar{h}_i = 3\ [\text{W/m}^2\cdot\text{K}]$ internal heat transfer coefficient (6)

$\bar{h}_o = 2.75\ [\text{W/m}^2\cdot\text{K}]$ external heat transfer coefficient (7)

$T_\infty = \text{ConvertTemp}(F,\ K,\ 65\ [\text{F}])$ ambient temperature (8)

$\rho = 1.177\ [\text{kg/m}^3]$ density of air - Appendix C-1 (9)

$c = 1007\ [\text{J/kg}\cdot\text{K}]$ specific heat capacity of air - Appendix C-1 (10)

$A_s = L \cdot (2 \cdot W + 2 \cdot H)$ surface area of duct (11)

$\dot{q}_{conv,i} = \bar{h}_i \cdot A_s \cdot (T_f - T_s)$ convection heat transfer from air in duct to duct surface (12)

$\dot{q}_{conv,o} = \bar{h}_o \cdot A_s \cdot (T_s - T_\infty)$ convection heat transfer from duct surface to surroundings (13)

$\dot{q}_{conv,i} = \dot{q}_{conv,o}$ energy balance on duct (14)

$\dot{m} = u_m \cdot \rho \cdot W \cdot H$ mass flow rate (15)

$\dot{m} \cdot c \cdot \Delta T = \dot{q}_{conv,o}$ energy balance on fluid (16)

$T_{f,out} = T_f - \Delta T$ outlet air temperature (17)

A more accurate result would result by recalculating heat loss using the average of the inlet and outlet air temperatures

Solution

Variables in Main program

$A_s = 4.459$ $[\text{m}^2]$	$c = 1007$ [J/kg-K]
$\Delta T = 4.621$ [K] {8.319 [R]}	$H = 0.1524$ [m]
$\bar{h}_i = 3$ $[\text{W/m}^{2\text{K}}]$	$\bar{h}_o = 2.75$ $[\text{W/m}^{2\text{K}}]$
$L = 4.877$ [m]	$\dot{m} = 0.03055$ [kg/s]
$\dot{q}_{conv,i} = 142.2$ [W]	$\dot{q}_{conv,o} = 142.2$ [W]
$\rho = 1.177$ $[\text{kg/m}^3]$	$T_f = 313.7$ [K]
$T_{f,out} = 309.1$	$T_\infty = 291.5$ [K]
$T_s = 303.1$ [K] {85.87 [F]}	$u_m = 0.5588$ [m/s]
$W = 0.3048$ [m]	

Key Variables

$\dot{q}_{conv,o} = 142.2$ [W]	*a.) rate of heat loss to surroundings*
$\Delta T = 4.621$ [K] {8.319 [R]}	*b.) Temperature change of fluid*

Problem 1.24

Equations

Assume steady state and constant properties

$$\dot{V}_a = 1600\ [\mathrm{ft^3/min}] \cdot \left| 4.71947 \times 10^{-4}\ \frac{\mathrm{m^3/s}}{\mathrm{ft^3/min}} \right| \quad \text{fan flow rate} \tag{1}$$

$$T_{a,in} = \mathrm{ConvertTemp}(F,\ K,\ 75\ [\mathrm{F}]) \quad \text{inlet air temperature} \tag{2}$$

$$\dot{V}_{eg} = 5\ [\mathrm{gal/min}] \cdot \left| 6.30902 \times 10^{-5}\ \frac{\mathrm{m^3/s}}{\mathrm{gal/min}} \right| \quad \text{pump flow rate} \tag{3}$$

$$T_{eg,in} = \mathrm{ConvertTemp}(F,\ K,\ 150\ [\mathrm{F}]) \quad \text{ethylene glycol inlet temperature} \tag{4}$$

$$\rho_{eg} = 1110\ [\mathrm{kg/m^3}] \quad \text{density from Appendix B} \tag{5}$$

$$c_{eg} = 2392\ [\mathrm{J/kg \cdot K}] \quad \text{specific heat capacity from Appendix B} \tag{6}$$

$$\rho_a = 1.177\ [\mathrm{kg/m^3}] \quad \text{density from Appendix C} \tag{7}$$

$$c_a = 1007\ [\mathrm{J/kg \cdot K}] \quad \text{specific heat capacity from Appendix C} \tag{8}$$

$$T_{a,out} = \mathrm{ConvertTemp}(F,\ K,\ 105\ [\mathrm{F}]) \quad \text{exit air temperature} \tag{9}$$

$$\dot{m}_a = \dot{V}_a \cdot \rho_a \quad \text{mass flow rate of air} \tag{10}$$

$$\dot{q} = \dot{m}_a \cdot c_a \cdot (T_{a,out} - T_{a,in}) \quad \text{heat transfer rate to air} \tag{11}$$

$$\dot{m}_{eg} = \dot{V}_{eg} \cdot \rho_{eg} \quad \text{mass flow rate of ethylene glycol} \tag{12}$$

$$\dot{q} = \dot{m}_{eg} \cdot c_{eg} \cdot (T_{eg,in} - T_{eg,out}) \quad \text{outlet ethylene glycol flow rate} \tag{13}$$

Solution

Variables in Main program

$c_a = 1007$ [J/kg-K] $\quad c_{eg} = 2392$ [J/kg-K] $\quad \dot{m}_a = 0.8888$ [kg/s] $\quad \dot{m}_{eg} = 0.3502$ [kg/s]
$\dot{q} = 14917$ [W] $\quad \rho_a = 1.177$ [kg/m^3] $\quad \rho_{eg} = 1110$ [kg/m^3] $\quad T_{a,in} = 297$ [K]
$T_{a,out} = 313.7$ [K] $\quad T_{eg,in} = 338.7$ [K] $\quad T_{eg,out} = 320.9$ [K] {117.9 [F]} $\quad \dot{V}_a = 0.7551$ [m^3/s]
$\dot{V}_{eg} = 0.0003155$ [m^3/s]

Key Variables

$\dot{q} = 14917$ [W] *a) rate of heat transfer from ethylene glycol*
$T_{eg,out} = 320.9$ [K] {117.9 [F]} *b) exit temperature of ethylene glycol*

Problem 1.25

Equations

```
$UnitSystem SI Mass J K Pa
```

Assumptions:
1. steady state
2. constant properties
3. neglect radiation

EES is used because this problem involves iterative calculations. Note that reasonable guess values are required. They can be set with the $VarInfo directives.

```
$VarInfo T_f Guess=310 Lower=298.15

$VarInfo T_s Guess=320 Lower=298.15

$VarInfo Re Lower=0
```

$D = 0.025\ [\text{m}]$ diameter (1)

$L = 0.3\ [\text{m}]$ length (2)

$\dot{w} = 220\ [\text{W}]$ power (3)

$T_\infty = \text{ConvertTemp}(C,\ K,\ 25\ [\text{C}])$ fluid temperature (4)

$u_f = 2\ [\text{m/s}]$ fluid velocity (5)

$P = 1\ [\text{bar}] \cdot \left| 100000\ \frac{\text{Pa}}{\text{bar}} \right|$ water pressure (6)

```
$ifnot ParametricTable
```

$F\$ = \text{'Air'}$ fluid is air (7)

```
$endif
```

$T_f = \frac{T_s + T_\infty}{2}$ film or average temperature (8)

$\rho = \rho\,(F\$,\ \text{T} = T_f,\ \text{P} = P)$ density (9)

```
$if F$='air'
```

$$\mu = \mu\left(F\$,\ \mathrm{T} = T_f\right) \quad \text{viscosity} \tag{10}$$

$$k = \mathrm{k}\left(F\$,\ \mathrm{T} = T_f\right) \quad \text{conductivity} \tag{11}$$

```
$else
```

$$\mu = \mu\left(F\$,\ \mathrm{T} = T_f,\ \mathrm{P} = P\right) \quad \text{viscosity} \tag{12}$$

$$k = \mathrm{k}\left(F\$,\ \mathrm{T} = T_f,\ \mathrm{P} = P\right) \quad \text{conductivity} \tag{13}$$

```
$endif
```

$$Re = \rho \cdot u_f \cdot D/mu \quad \text{Reynolds number} \tag{14}$$

$$Nu\bar{s}selt = 0.3 \cdot Re^{0.6} \quad \text{average Nusselt number} \tag{15}$$

$$Nu\bar{s}selt = \bar{h} \cdot D/k \quad \text{average heat transfer coefficient} \tag{16}$$

$$A_s = \pi \cdot D \cdot L \quad \text{surface area} \tag{17}$$

$$\dot{q}_{conv} = \bar{h} \cdot A_s \cdot (T_s - T_\infty) \quad \text{convection heat transfer} \tag{18}$$

$$\dot{q}_{conv} = \dot{w} \quad \text{energy balance} \tag{19}$$

Note that the heat transfer coefficient between water and the tube surface is much higher than for air, resulting in a much lower surface temperature.

Parametric Table: $Table\ 1$

Run	$F\$$	Re [-]	$Nu\bar{s}selt$ [-]	$\dot{q}_{conv}$ [W]	T_s [K] {[C]}
1	air	*1697*	*25.99*	*220*	*556.8* {*283.6*}
2	water	*57142*	*214.4*	*220*	*299.9* {*26.79*}

Problem 1.26

Figure 1 illustrates a Solar Electrical Generation System (a SEGS plant).

Figure 1: SEGS plant.

A field of independently controlled mirrors focus the solar energy onto the central receiver in order to elevate the temperature of a molten salt which subsequently provides heat to a power cycle that produces electricity. We will model the central receiver as shown in Figure 2.

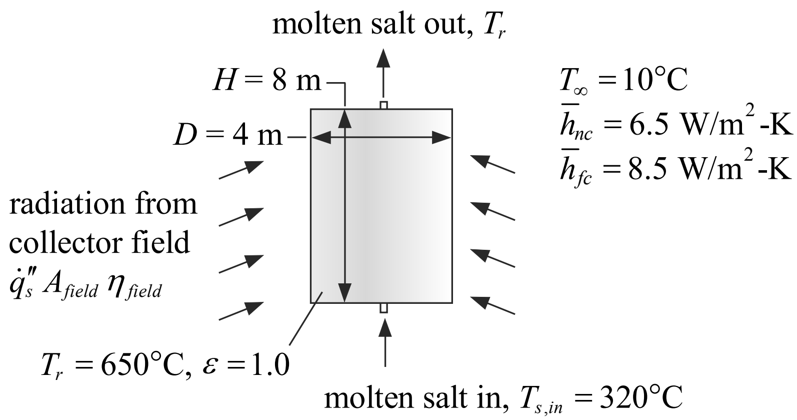


Figure 2: Model of central receiver.

The receiver is a cylinder with diameter D = 4 m and height H = 8 m. The emissivity of the surface is ε = 1. The collector field has an area of A_{field} = 20 acre and an overall efficiency relative to transferring the incident solar flux to the receiver surface of η_{field} = 0.76. The solar flux that is incident on the field is $\dot{q}_s''$ = 750 W/m^2. The receiver absorbs all of the radiation that strikes its surface. The temperature of the receiver surface is T_r = 650ºC. The molten salt is composed of 58%NaCl and 42% $MgCl_2$. The salt enters the receiver at $T_{s,in}$ = 320ºC and leaves at the receiver surface temperature. Note that the properties of this molten salt are available in the solid/liquid library of EES. The flow rate of the molten salt is controlled in order to maintain the required receiver temperature. The molten salt flows into the boiler of a power cycle. The efficiency of the power cycle is given by:

$$\eta = \eta_2 \left(1 - \frac{T_\infty}{T_r}\right) \tag{1}$$

where η_2 = 0.25 is the second law efficiency of the power cycle.

The receiver experiences losses that are related to forced and natural convection to the surrounding fluid at T_∞ = 10ºC. The natural convection heat transfer coefficient is $\bar{h}_{nc}$ = 6.5 W/m^2-K and the forced convection heat transfer coefficient is $\bar{h}_{fc}$ = 8.5 W/m^2-K. The heat transfer coefficient that characterizes the combined effect of natural and forced convection can be calculated approximately according to:

$$\bar{h} = \left(\bar{h}_{nc}^3 + \bar{h}_{fc}^3\right)^{1/3} \tag{2}$$

The receiver also experiences radiation loss to surroundings at T_∞ = 10ºC. Assume that the top and bottom surfaces are well-insulated so that losses only occur from the sides.
a.) Determine the total rate of heat loss from the receiver.

The inputs are entered in EES:

```
$UnitSystem SI MASS RAD PA K J
$Tabstops 0.2 0.4 0.6 3.5 in

"Inputs"
D=4 [m]                                  "outer diameter of receiver"
H=8 [m]                                  "height of receiver"
e=1.0 [-]                                "emissivity"
T_s_in=converttemp(C,K,320 [C])          "molten salt inlet temperature"
T_r_C=650 [C]                            "receiver and molten salt outlet temperature, in C"
T_r=converttemp(C,K,T_r_C)               "receiver and molten salt outlet  temperature"
T_infinity=converttemp(C,K,10 [C])       "surrounding air temperature"
A_field_acre=20 [acre]                   "size of collector field, in acre"
A_field=A_field_acre*convert(acre,m^2)   "area of collector field"
eta_field=0.76 [-]                       "overall efficiency of field relative to delivering flux to receiver"
q``_s=750 [W/m^2]                        "solar flux"
h_bar_nc=6.5 [W/m^2-K]                   "natural convection heat transfer coefficient"
h_bar_fc=8.5 [W/m^2-K]                   "forced convection heat transfer coefficient"
eta_2=0.25 [-]                           "2nd law efficiency of the power cycle"
```

The mixed convection coefficient is computed according to Eq. (2). The resistance to convection is:

$$R_{conv} = \frac{1}{\bar{h}\,\pi\,D\,H} \tag{3}$$

The resistance to radiation is:

$$R_{rad} = \frac{1}{\varepsilon\,\sigma\,\pi\,D\,H\left(T_r + T_\infty\right)\left(T_r^2 + T_\infty^2\right)} \tag{4}$$

The heat transfer due to convection and radiation are given by:

$$\dot{q}_{conv} = \frac{\left(T_r - T_\infty\right)}{R_{conv}} \tag{5}$$

$$\dot{q}_{rad} = \frac{\left(T_r - T_\infty\right)}{R_{rad}} \tag{6}$$

The total loss is:

$$\dot{q}_{loss} = \dot{q}_{rad} + \dot{q}_{conv} \tag{7}$$

```
h_bar=(h_bar_nc^3+h_bar_fc^3)^(1/3)                              "convection heat transfer coefficient"
R_conv=1/(pi*D*H*h_bar)                                          "convection resistance"
R_rad=1/(e*sigma#*pi*D*H*(T_r+T_infinity)*(T_r^2+T_infinity^2))  "radiation resistance"
q_dot_conv=(T_r-T_infinity)/R_conv                               "convective losses"
q_dot_rad=(T_r-T_infinity)/R_rad                                 "radiation losses"
q_dot_loss=q_dot_conv+q_dot_rad                                  "total losses"
```

which leads to $\dot{q}_{loss} = 4.72$ MW.

b.) Determine the mass flow rate of molten salt.

The total rate at which solar energy is hitting the collector field is:

$$\dot{q}_{s,field} = A_{field}\,\dot{q}''_s \tag{8}$$

and the total rate at which solar energy is hitting the receiver is:

$$\dot{q}_{s,r} = \dot{q}_{s,field}\,\eta_{field} \tag{9}$$

The rate at which heat is transferred to the molten salt is:

$$\dot{q}_{salt} = \dot{q}_{s,r} - \dot{q}_{loss} \tag{10}$$

```
q_dot_s_field=A_field*q``_s            "incident solar energy on field"
q_dot_s_r=q_dot_s_field*eta_field      "incident solar energy on receiver"
q_dot_salt=q_dot_s_r-q_dot_loss        "heat transfer to molten salt"
```

The specific heat capacity of the molten salt (c_{salt}) is evaluated at the average salt temperature. The mass flow rate of salt is obtained from an energy balance:

$$\dot{q}_{salt} = \dot{m}_{salt}\, c_{salt} \left(T_r - T_{s,in}\right) \tag{11}$$

```
q_dot_s_field=A_field*q``_s                              "incident solar energy on field"
q_dot_s_r=q_dot_s_field*eta_field                        "incident solar energy on receiver"
q_dot_salt=q_dot_s_r-q_dot_loss                          "heat transfer to molten salt"
T_salt_avg=(T_r+T_s_in)/2                                "average molten salt temperature"
c_salt=c_('Salt (58% NaCl, 42% MgCl2)', T_salt_avg)      "molten salt heat capacity"
m_dot_salt*c_salt*(T_r-T_s_in)=q_dot_salt                "salt flow rate"
```

which leads to $\dot{m}_{salt}$ = 116.2 kg/s.

c.) Determine the rate at which power is produced by the power cycle.

The efficiency of the cycle is computed according to Eq. (1). The rate of power production is given by:

$$\dot{w} = \eta\, \dot{q}_{salt} \tag{12}$$

```
eta=eta_2*(1-T_infinity/T_r)                             "efficiency of power cycle"
w_dot=eta*q_dot_salt                                     "power produced by power cycle"
w_dot_MW=w_dot*convert(W,MW)                             "in MW"
```

which leads to $\dot{w}$ = 7.2 MW.

d.) Determine the overall efficiency of the SEGS plant; this is defined as the ratio of the power produced to the total solar energy incident on the field.

The total efficiency of the SEGS plant is:

$$\eta_{overall} = \frac{\dot{w}}{\dot{q}_{s,field}} \tag{13}$$

```
eta_overall=w_dot/q_dot_s_field                    "overall efficiency of the SEGS plant"
```

which leads to $\eta_{overall}$ = 0.118.

e.) Plot the overall efficiency as a function of the receiver temperature for various values of the heat flux. You should see that there is an optimal receiver temperature for each value of heat flux. Explain why this occurs.

Figure 3 illustrates the requested plot. As the receiver temperature increases, the efficiency of the cycle is improved but the losses also increase. The optimal receiver temperature balances these effects. Notice that the optimal receiver temperature is reduced with decreasing solar flux because the losses that can be tolerated is reduced.

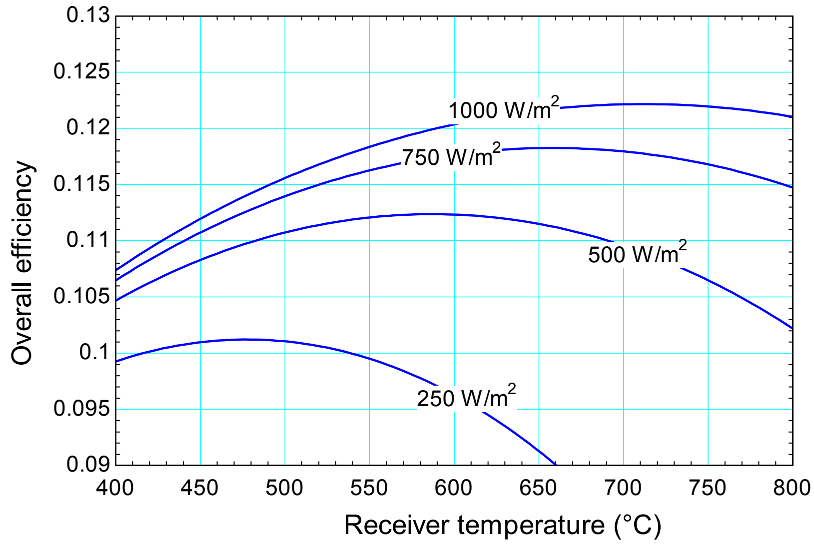


Figure 3: Overall efficiency as a function of the receiver temperature for various values of the solar flux.

f.) Plot the overall efficiency as a function of solar flux if the plant is operated with a constant receiver temperature of T_r = 650ºC.

Figure 4 illustrates the overall efficiency as a function of the solar flux with T_r = 650ºC. Note that this plot corresponds to traversing along a line of constant receiver temperature in Figure 3.

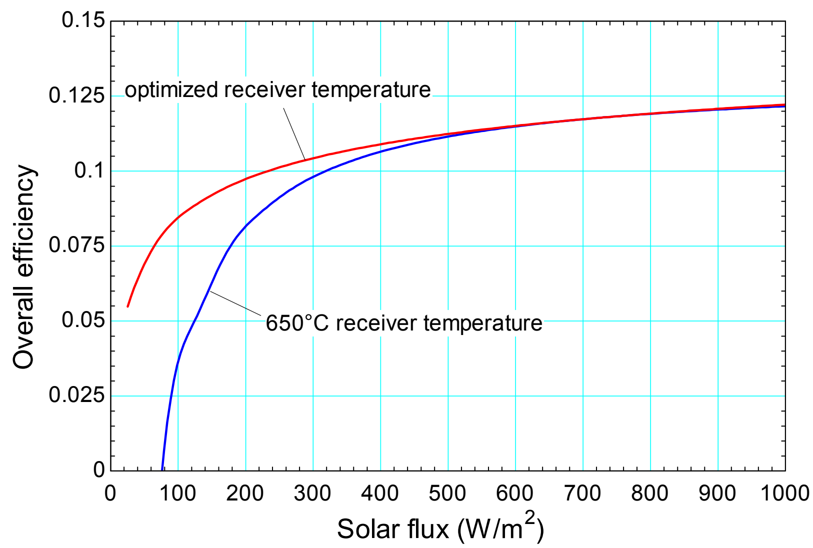


Figure 4: Overall efficiency as a function of the solar flux for the case where T_r = 650ºC and the case where the receiver temperature is optimized.

g.) Overlay on your plot from (f) the overall efficiency as a function of solar flux if the receiver temperature is optimized for the instantaneous value of the solar flux. You may want to use the Min/Max Table feature in EES to accomplish this.

Figure 4 illustrates the case overall efficiency as a function of the solar flux for the case where the receiver temperature is adjusted in order to maximize the overall efficiency. Note that this corresponds to moving along the apex of the curves in Figure 3.

Problem 1.27

Computer chips tend to work better if they are kept cold. You are examining the feasibility of maintaining the processor of a personal computer at the sub-ambient temperature of T_{chip} = 0°F. Assume that the operation of the computer chip itself generates $\dot{q}_{chip}$ =10 W of power. Model the processor unit as a box that is a = 2 inch x b = 6 inch x c = 4 inch. Assume that all six sides of the box is exposed to air at T_{air} = 70°F with a convection heat transfer coefficient of $\overline{h}$ =10 W/m^2-K. The box experiences a radiation heat transfer with surroundings that are at T_{sur} = 70°F. The emissivity of the processor surface is ε=0.7 and all six sides experience the radiation heat transfer. You are asked to size the refrigeration system required to maintain the temperature of the processor.

a.) What is the refrigeration load that your refrigeration system must be able to remove to maintain the processor at a steady-state temperature (W)?

The input parameters are entered in EES; notice that the units of each parameter are immediately converted into SI and the units of the associated variables are set (by you) in the Variable Information Window (Figure 2).

```
$UnitSystem SI MASS RAD PA  K J
$TABSTOPS   0.2 0.4 0.6 0.8 3.5 in

"INPUTS"
T_chip = converttemp(F,K,0)              "chip temperature"
q_dot_chip = 10 [W]                      "chip generation"
a = 2 [inch]*convert(inch,m)             "dimensions of processor"
b = 6 [inch]*convert(inch,m)
c = 4 [inch]*convert(inch,m)
h = 10 [W/m^2-K]                         "heat transfer coefficient"
T_air=converttemp(F,K,70)                "air temperature"
T_sur=converttemp(F,K,70)                "temperature of surroundings"
e = 0.7                                  "emissivity of surface"
```

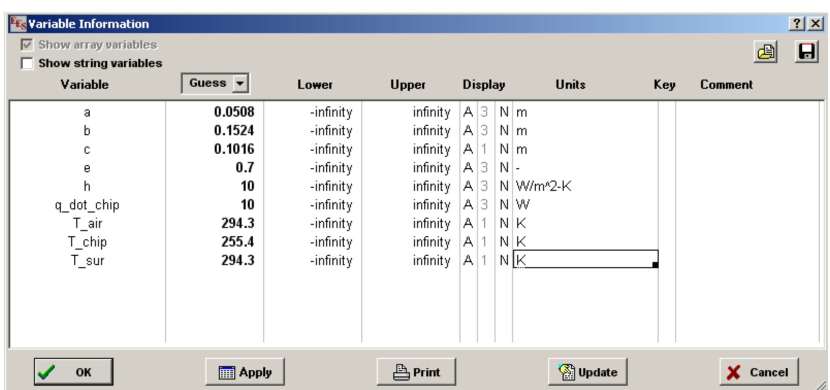


Figure 2:Variable Information window showing the units for each variable set.

A control volume encompasses just the processor and includes the internal generation from operating the chip ($\dot{q}_{chip}$) as well as convection ($\dot{q}_{conv}$) and radiation ($\dot{q}_{rad}$) and the heat transfer removed by the refrigeration system ($\dot{q}_{load}$). The energy balance is:

$$\dot{q}_{chip} + \dot{q}_{conv} + \dot{q}_{rad} = \dot{q}_{load} \tag{1}$$

The convection and radiation heat transfer rates may be evaluated using the associated rate equations:

$$\dot{q}_{conv} = h\,A_s\left(T_{air} - T_{chip}\right) \tag{2}$$

$$\dot{q}_{rad} = \sigma\,\varepsilon\,A_s\left(T_{sur}^4 - T_{chip}^4\right) \tag{3}$$

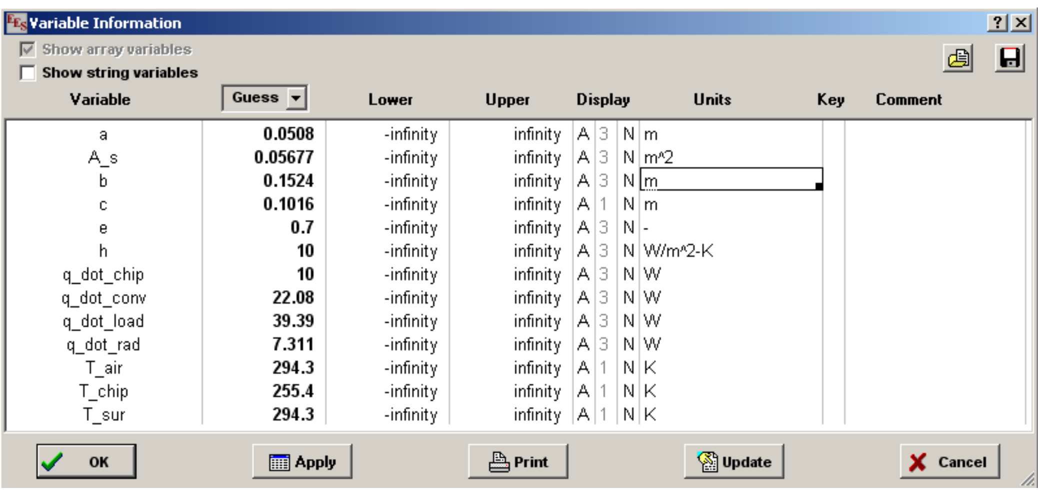
where σ is Stefan-Boltzmann's constant and A_s is the surface area of the processor:

$$A_s = 2\left(a\,b + b\,c + a\,c\right) \tag{4}$$

These equations are programmed in EES:

```
"part (a)"
A_s=2*(a*b+b*c+a*c)                                   "surface area of processor"
q_dot_conv=h*A_s*(T_air-T_chip)                       "convective heat transfer"
q_dot_rad=sigma#*e*A_s*(T_sur^4-T_chip^4)             "radiation heat transfer"
q_dot_chip+q_dot_conv+q_dot_rad=q_dot_load            "energy balance"
```

The units of the variables that have been added are also entered in the Variable Information window (Figure 3).



Variable	Guess	Lower	Upper	Display			Units	Key	Comment
a	0.0508	-infinity	infinity	A	3	N	m		
A_s	0.05677	-infinity	infinity	A	3	N	m^2		
b	0.1524	-infinity	infinity	A	3	N	m		
c	0.1016	-infinity	infinity	A	1	N	m		
e	0.7	-infinity	infinity	A	3	N	-		
h	10	-infinity	infinity	A	3	N	W/m^2-K		
q_dot_chip	10	-infinity	infinity	A	3	N	W		
q_dot_conv	22.08	-infinity	infinity	A	3	N	W		
q_dot_load	39.39	-infinity	infinity	A	3	N	W		
q_dot_rad	7.311	-infinity	infinity	A	3	N	W		
T_air	294.3	-infinity	infinity	A	1	N	K		
T_chip	255.4	-infinity	infinity	A	1	N	K		
T_sur	294.3	-infinity	infinity	A	1	N	K		

Figure 3: Variable Information window with additional units entered.

You can check that your solution is dimensionally consistent by selecting Check Units from the Calculate menu (Figure 4).

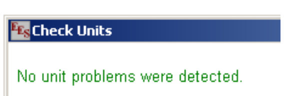


Figure 4: Check Units message

Solving the problem (Solve from the Calculate menu) will bring up the Solution Window (Figure 5) and shows that the refrigeration load is 39.4 W.

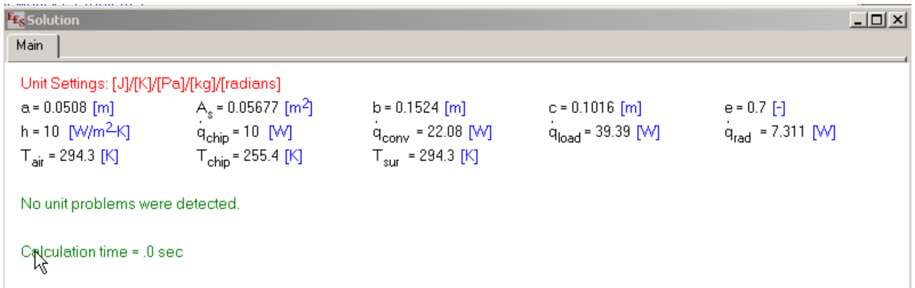


Figure 5: Solution Window

b.) If the coefficient of performance (COP) of the refrigeration system is nominally 3.5, then how much heat must be rejected to the ambient air (W)? Recall that COP is the ratio of the amount of refrigeration provided to the amount of input power required.

The definition of COP is:

$$\text{COP} = \frac{\dot{q}_{load}}{\dot{w}_{ref}} \tag{5}$$

which is programmed in EES:

```
"part (b)"
COP = 3.5                                 "specified COP"
COP = q_dot_load/w_dot_ref                "refrigeration power"
```

and solved to show that the refrigeration power will be 11.3 W.

c.) If electricity costs 12¢/kW-hr, how much does it cost to run the refrigeration system for a year, assuming that the computer is never shut off.

The cost of electricity and time of operation are both converted to SI units and used to evaluate the cost per year.

```
"part (c)"
ecost = 12 [cents/kW-hr]*convert(cents/kW-hr,$/J)    "cost of electricity"
time=1 [year]*convert(year,s)                         "time of operation"
cost=time*ecost*w_dot_ref                             "cost of operating system for 1 year"
```

The cost of operating the system for 1 year is $11.8.

Problem 1.28

You are designing a cubical case that contains electronic components that drive remotely located instruments. You have been asked to estimate the maximum and minimum operating temperature limits that should be used to specify the components within the case. The case is W = 8 inch on a side. The emissivity of the paint used on the case is $\varepsilon = 0.85$. The operation of the electronic components within the case generates between $\dot{q} = 5$ and $\dot{q} = 10$ W due to ohmic heating, depending on the intensity of the operation. The top surface of the case is exposed to a solar flux $\dot{q}''$. All of the surfaces of the case convect (with average heat transfer coefficient $\bar{h}$) and radiate to surroundings at T_∞. The case will be deployed in a variety of climates, ranging from very hot ($T_{\infty,max}$ = 110°F) to very cold ($T_{\infty,max}$ = -40°F), very sunny ($\dot{q}''_{max}$ = 850 W/m^2) to night ($\dot{q}''_{min}$ = 0 W/m^2), and very windy ($\bar{h}_{max}$ = 100 W/m^2-K) to still ($\bar{h}_{min}$ = 5 W/m^2-K). For the following questions, assume that the case is at a single, uniform temperature and at steady state.

a.) Come up with an estimate for the maximum operating temperature limit.

The case temperature will be highest when the case generation is maximum, the ambient temperature is maximum, the solar flux is maximum, and the heat transfer coefficient is minimum. These inputs are entered in EES:

```
$UnitSystem SI MASS RAD PA K J
$Tabstops 0.2 0.4 0.6 3.5 in

"Inputs"
W=8 [inch]*convert(inch,m)                    "side dimension"
e=0.85 [-]                                    "emissivity"
q_dot=10 [W]                                  "dissipation in case"
q``=850 [W/m^2]                               "solar flux"
h_bar=5 [W/m^2-K]                             "heat transfer coefficient"
T_infinity=converttemp(F,K,120 [F])           "ambient temperature"
```

The surface area of the case is:

$$A_s = 6W^2 \tag{1}$$

The resistance to convection from the case is:

$$R_{conv} = \frac{1}{\bar{h}\, A_s} \tag{2}$$

```
A_s=6*W^2                                     "surface area"
R_conv=1/(h_bar*A_s)                          "convection resistance"
```

The radiation resistance cannot be calculated without knowing the surface temperature of the case, T. Therefore, a reasonable value of the surface temperature is assumed. The radiation resistance is:

$$R_{rad} = \frac{1}{\varepsilon\, \sigma\, A_s \left(T^2 + T_\infty^2\right)\left(T + T_\infty\right)} \tag{3}$$

```
T=350 [K]                                                    "guess for the case temperature"
R_rad=1/(e*A_s*sigma#*(T^2+T_infinity^2)*(T+T_infinity))     "radiation resistance"
```

The guess values are updated and the assumed value of the case temperature is commented out. An energy balance on the case leads to:

$$\dot{q}_s + \dot{q} = \frac{\left(T - T_\infty\right)}{R_{conv}} + \frac{\left(T - T_\infty\right)}{R_{rad}} \tag{4}$$

where $\dot{q}_s$ is the absorbed solar irradiation.

$$\dot{q}_s = W^2\, q'' \tag{5}$$

```
{T=350 [K]}                                                  "guess for the case temperature"
q``*W^2+q_dot=(T-T_infinity)/R_conv+(T-T_infinity)/R_rad     "energy balance on case"
T_F=converttemp(K,F,T)                                       "case temperature in F"
```

which leads to a maximum operating temperature limit of T = 147.5°F.

b.) Plot the maximum operating temperature as a function of the case size, W. Explain the shape of your plot (why does the temperature go up or down with W? if there is an asymptotic limit, explain why it exists).

The value of W is commented out and a parametric table is generated that includes W and T. Figure 1 illustrates the maximum operating temperature as a function of the size of the enclosure. As the size of the enclosure is reduced, the maximum operating temperature increases because the 10 W of dissipation must be rejected but the area available for convection and radiation is reduced. As the size is increased, the maximum operating temperature reaches an asymptote. The limiting value of T is higher than T_∞ because the absorbed solar irradiation and the surface area for convection and radiation both increase in proportion to W^2; therefore the limit is consistent with the situation where the solar flux is exactly balanced by the heat flux associated with radiation and convection (the dissipation becomes insignificant relative to the solar flux).

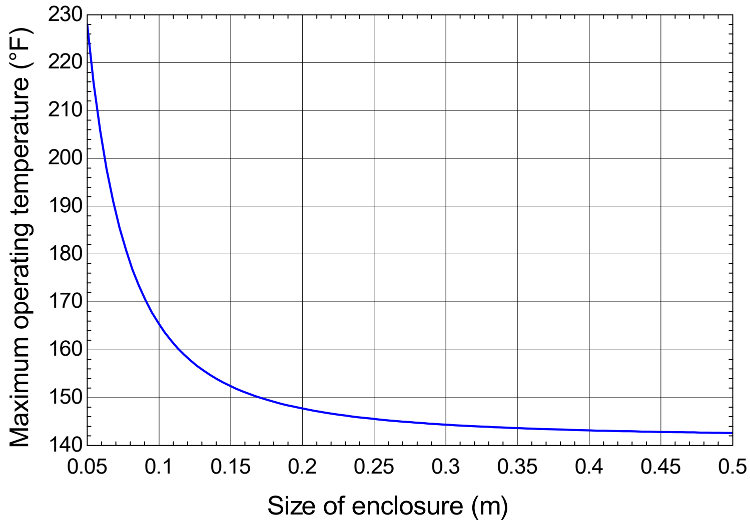


Figure 1: Maximum operating temperature as a function of the size of the enclosure.

c.) Come up with an estimate for the minimum operating temperature limit (with W = 8 inch).

The case temperature will be lowest when the case generation is minimum, the ambient temperature is minimum, the solar flux is minimum, and the heat transfer coefficient is maximum. These inputs are entered in EES:

```
q_dot=5 [W]                                   "dissipation in case"
q``=0 [W/m^2]                                 "solar flux"
h_bar=100 [W/m^2-K]                           "heat transfer coefficient"
T_infinity=converttemp(F,K,-40 [F])           "ambient temperature"
```

The solution is run again at the predicted temperature is T= -39.7°F

d.) Do you feel that the emissivity of the case surface is very important for determining the minimum operating temperature? Justify your answer.

The emissivity is not important because radiation is not important. To see this, look at the resistance to convection, R_{conv}= 0.040 K/W, and the resistance to radiation, R_{rad}= 1.65 K/W. These two heat transfer mechanisms occur in parallel; the largest resistance in a parallel network is not important - therefore, radiation is much less important than convection.