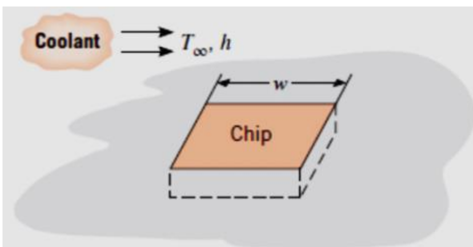
Suggested Problem
Heat Transfer I

1.14 A wall has inner and outer surface temperatures of 16 and 6°C, respectively. The interior and exterior air temperatures are 20 and 5°C, respectively. The inner and outer convection heat transfer coefficients are 5 and 20 W/m2 · K, respectively. Calculate the heat flux from the interior air to the wall, from the wall to the exterior air, and from the wall to the interior air. Is the wall under steady-state conditions?

1.17 A cartridge electrical heater is shaped as a cylinder of length L = 300 mm and outer diameter D = 30 mm. Under normal operating conditions, the heater dissipates 2 kW while submerged in a water flow that is at 20°C and provides a convection heat transfer coefficient of h = 5000 W/m2 · K. Neglecting heat transfer from the ends of the heater, determine its surface temperature Ts and the thermal resistance due to convection. If the water flow is inadvertently terminated while the heater continues to operate, the heater surface is exposed to air that is also at 20°C but for which h = 50 W/m2 · K. What are the corresponding thermal resistance due to convection and surface temperature? What are the consequences of such an event?

1.19 A square isothermal chip is of width w = 5 mm on a side and is mounted in a substrate such that its side and back surfaces are well insulated; the front surface is exposed to the flow of a coolant at T∞ = 15°C. From reliability considerations, the chip temperature must not exceed T = 85°C.



If the coolant is air and the corresponding convection coefficient is h = 200 W/m2 · K, what is the maximum allowable chip power? If the coolant is a dielectric liquid for which h = 3000 W/m2 · K, what is the maximum allowable power?

1.22 An overhead 25-m-long, uninsulated industrial steam pipe of 100-mm diameter is routed through a building whose walls and air are at 25°C. Pressurized steam maintains a pipe surface temperature of 150°C, and the coefficient associated with natural convection is h = 10 W/m2 · K. The surface emissivity is ε = 0.8.

(a) What is the rate of heat loss from the steam line?

(b) If the steam is generated in a gas-fired boiler operating at an efficiency of ηf = 0.90 and natural gas is priced at Cg = $0.02 per MJ, what is the annual cost of heat loss from the line?

1.15 The free convection heat transfer coefficient on a thin hot vertical plate suspended in still air can be determined from observations of the change in plate temperature with time as it cools. Assuming the plate is isothermal and radiation exchange with its surroundings is negligible, evaluate the convection coefficient at the instant of time when the plate temperature is 245°C and the change in plate temperature with time (dT/dt) is −0.028 K/s. The ambient air temperature is 25°C and the plate measures 0.4 × 0.4 m with a mass of 4.25 kg and a specific heat of 2770 J/kg · K.

Now the plate is in a vacuum with a surrounding temperature of 25°C. What is the emissivity of the plate? What is the rate at which radiation is emitted by the surface?

1.30 Pressurized water (pin = 10 bar, Tin = 110°C) enters the bottom of an L = 12-m-long vertical tube of diameter D = 110 mm at a mass flow rate of . The tube is located inside a combustion chamber, resulting in heat transfer to the tube. Superheated steam exits the top of the tube at pout = 7 bar, Tout = 600°C. Determine the change in the rate at which the following quantities enter and exit the tube: (a) the combined thermal and flow work, (b) the mechanical energy, and (c) the total energy of the water. Also, (d) determine the heat transfer rate, q. Hint: Relevant properties may be obtained from a thermodynamics text.

1.31 Consider the tube and inlet conditions of Problem 1.30. Heat transfer at a rate of q = 3.89 MW is delivered to the tube. For an exit pressure of p = 8 bar, determine (a) the temperature of the water at the outlet as well as the change in (b) combined thermal and flow work, (c) mechanical energy, and (d) total energy of the water from the inlet to the outlet of the tube. Hint: As a first estimate, neglect the change in mechanical energy in solving part (a). Relevant properties may be obtained from a thermodynamics text.

1.33 Approximately 40 percent of the water that is pumped in the United States is used to cool power plants. Sufficient quantities of water may not be available in arid regions, necessitating use of air-cooled condensers. Consider the power plant and operating conditions of **Example 1.7,** but with the plant's water-cooled condenser replaced with an air-cooled condenser. The cold-side thermal resistance can be approximated as Rt,c = 1/(hcA), where hc is the convection heat transfer coefficient for the flow in the condenser tubes and A is the heat transfer surface area. The heat transfer surface area of the air condenser is 10 times larger than that of the water condenser, and the convection heat transfer coefficient for the water condenser is 25 times larger than that of the air condenser. Determine the modified efficiency and power output of the plant with the air-cooled condenser. Knowing that the condenser cost is proportional to its heat transfer area, is the higher cost of the air-cooled condenser offset by better power plant efficiency and/or higher power output with air cooling? Assume clean conditions.

1.35 Chips of width L = 15 mm on a side are mounted to a substrate that is installed in an enclosure whose walls and air are maintained at a temperature of Tsur = 25°C. The chips have an emissivity of ε = 0.60 and a maximum allowable temperature of Ts = 85°C.

(a) If heat is rejected from the chips by radiation and natural convection, what is the maximum operating power of each chip? The convection coefficient depends on the chip-to-air temperature difference and may be approximated as h = C(Ts − T∞)1/4, where C = 4.2 W/m2 · K5/4.

(b) If a fan is used to maintain airflow through the enclosure and heat transfer is by forced convection, with h = 250 W/m2 · K, what is the maximum operating power?

1.40 An aluminum plate 4 mm thick is mounted in a horizontal position, and its bottom surface is well insulated. A special, thin coating is applied to the top surface such that it absorbs 80% of any incident solar radiation, while having an emissivity of 0.25. The density ρ and specific heat c of aluminum are known to be 2700 kg/m3 and 900 J/kg · K, respectively.

(a) Consider conditions for which the plate is at a temperature of 25°C and its top surface is suddenly exposed to ambient air at T∞ = 20°C and to solar radiation that provides an incident flux of 900 W/m2. The convection heat transfer coefficient between the surface and the air is h = 20 W/m2 · K. What is the initial rate of change of the plate temperature?

(b) What will be the equilibrium temperature of the plate when steady-state conditions are reached?

1.42 Liquid oxygen, which has a boiling point of 90 K and a latent heat of vaporization of 214 kJ/kg, is stored in a spherical container whose outer surface is of 500-mm diameter and at a temperature of −10°C. The container is housed in a laboratory whose air and walls are at 25°C.
(a) If the surface emissivity is 0.20 and the heat transfer coefficient associated with free convection at the outer surface of the container is 10 W/m2 · K, what is the rate, in kg/s, at which oxygen vapor must be vented from the system?

(b) Moisture in the ambient air will result in frost formation on the container, causing the surface emissivity to increase. Assuming the surface temperature and convection coefficient to remain at −10°C and 10 W/m2 · K, respectively, compute the oxygen evaporation rate (kg/s) as a function of surface emissivity over the range $0.2 \leq \varepsilon \leq 0.94$.

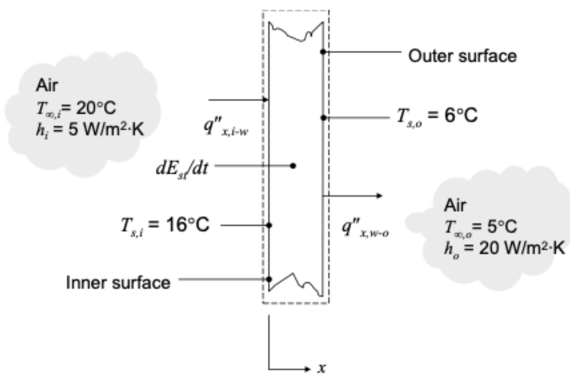
Suggested Problem_Solution
Heat Transfer I

PROBLEM 1.14

KNOWN: Inner and outer surface temperatures of a wall. Inner and outer air temperatures and convection heat transfer coefficients.

FIND: Heat flux from inner air to wall. Heat flux from wall to outer air. Heat flux from wall to inner air. Whether wall is under steady-state conditions.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible radiation, (2) No internal energy generation.

ANALYSIS: The heat fluxes can be calculated using Newton's law of cooling. Convection from the inner air to the wall occurs in the positive x-direction:

$$q''_{x,i-w} = h_i(T_{\infty,i} - T_{s,i}) = 5\ \text{W/m}^2\cdot\text{K} \times (20°\text{C} - 16°\text{C}) = 20\ \text{W/m}^2 \quad <$$

Convection from the wall to the outer air also occurs in the positive x-direction:

$$q''_{x,w-o} = h_o(T_{s,o} - T_{\infty,o}) = 20\ \text{W/m}^2\cdot\text{K} \times (6°\text{C} - 5°\text{C}) = 20\ \text{W/m}^2 \quad <$$

From the wall to the inner air:

$$q''_{w-i} = h_i(T_{s,i} - T_{\infty,i}) = 5\ \text{W/m}^2\cdot\text{K} \times (16°\text{C} - 20°\text{C}) = -20\ \text{W/m}^2 \quad <$$

An energy balance on the wall gives

$$\frac{dE_{st}}{dt} = \dot{E}_{in} - \dot{E}_{out} = A(q''_{x,i-w} - q''_{x,w-o}) = 0$$

Since $dE_{st}/dt = 0$, the wall *could be* at steady-state and the *spatially-averaged* wall temperature is not changing. However, it is possible that stored energy is increasing in one part of the wall and decreasing in another, therefore we cannot tell if the wall is at steady-state or not. If we found $dE_{st}/dt \neq 0$, we would know the wall was not at steady-state. $<$

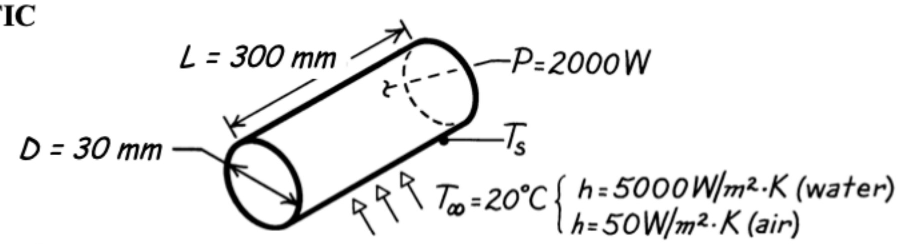
COMMENTS: The heat flux from the wall to the inner air is equal and opposite to the heat flux from the inner air to the wall.

PROBLEM 1.17

KNOWN: Dimensions of a cartridge heater. Heater power. Convection coefficients in air and water at a prescribed temperature.

FIND: Thermal convection resistance and heater surface temperatures in water and air.

SCHEMATIC



ASSUMPTIONS: (1) Steady-state conditions, (2) All of the electrical power is transferred to the fluid by convection, (3) Negligible heat transfer from ends.

ANALYSIS: With P = q_{conv}, Newton's law of cooling yields

$$P = hA(T_S - T_\infty) = h\pi DL(T_S - T_\infty)$$
$$T_S = T_\infty + \frac{P}{h\pi DL}.$$

From Eq. 1.11, the thermal resistance due to convection is given by

$$R_{t,conv} = \Delta T / q_x = (T_s - T_\infty)/P = 1/h\pi DL$$

In water,

$$T_s = 20°C + \frac{2000\ W}{5000\ W/m^2 \cdot K \times \pi \times 0.03\ m \times 0.3\ m} = 34.2°C \qquad <$$

$$R_{t,conv} = 1/h\pi DL = 1/(5000\ W/m^2 \cdot K \times \pi \times 0.03\ m \times 0.3\ m) = 0.00707\ K/W \qquad <$$

In air,

$$T_s = 20°C + \frac{2000\ W}{50\ W/m^2 \cdot K \times \pi \times 0.03\ m \times 0.3\ m} = 1435°C \qquad <$$

$$R_{t,conv} = 1/h\pi DL = 1/(50\ W/m^2 \cdot K \times \pi \times 0.03\ m \times 0.3\ m) = 0.707\ K/W \qquad <$$

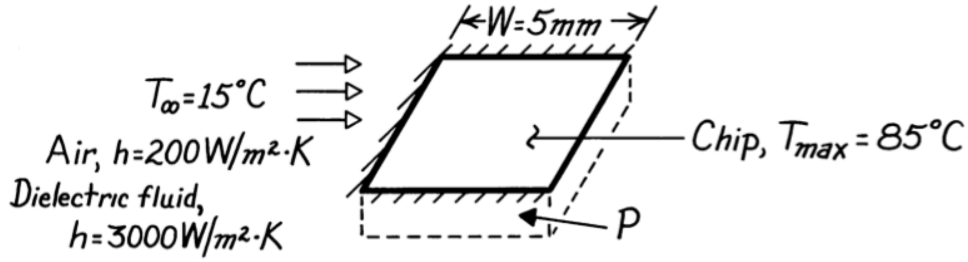
COMMENTS: (1) Air is much less effective than water as a heat transfer fluid. Hence, the cartridge temperature is much higher in air, so high, in fact, that the cartridge would melt. (2) In air, the high cartridge temperature would render radiation significant. (3) Larger thermal resistance corresponds to less effective heat transfer.

PROBLEM 1.19

KNOWN: Chip width and maximum allowable temperature. Coolant conditions.

FIND: Maximum allowable chip power for air and liquid coolants.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Negligible heat transfer from sides and bottom, (3) Chip is at a uniform temperature (isothermal), (4) Negligible heat transfer by radiation in air.

ANALYSIS: All of the electrical power dissipated in the chip is transferred by convection to the coolant. Hence,

$$P = q$$

and from Newton's law of cooling,

$$P = hA(T - T_\infty) = h\, W^2(T - T_\infty).$$

In *air*,

$$P_{max} = 200\ W/m^2\cdot K(0.005\ m)^2(85 - 15)\ °C = 0.35\ W. \qquad <$$

In the *dielectric liquid*

$$P_{max} = 3000\ W/m^2\cdot K(0.005\ m)^2(85-15)\ °C = 5.25\ W. \qquad <$$

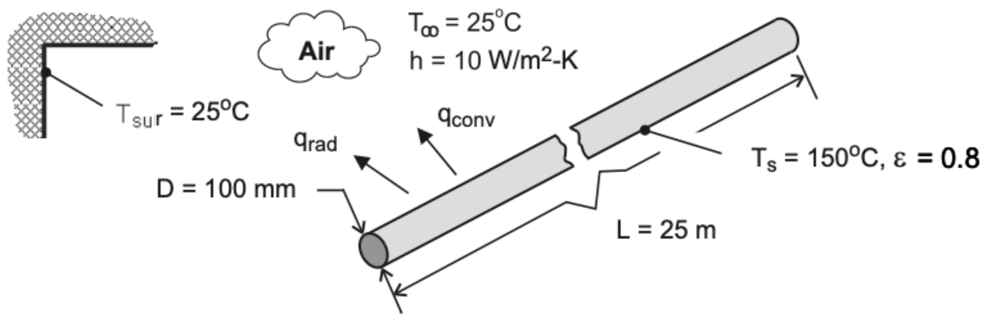
COMMENTS: Relative to liquids, air is a poor heat transfer fluid. Hence, in air the chip can dissipate far less energy than in the dielectric liquid.

PROBLEM 1.22

KNOWN: Length, diameter, surface temperature and emissivity of steam line. Temperature and convection coefficient associated with ambient air. Efficiency and fuel cost for gas fired furnace.

FIND: (a) Rate of heat loss, (b) Annual cost of heat loss.

SCHEMATIC:



ASSUMPTIONS: (1) Steam line operates continuously throughout year, (2) Net radiation transfer is between small surface (steam line) and large enclosure (plant walls).

ANALYSIS: (a) From Eqs. (1.3a) and (1.7), the heat loss is

$$q = q_{conv} + q_{rad} = A\left[h\left(T_s - T_\infty\right) + \varepsilon\sigma\left(T_s^4 - T_{sur}^4\right)\right]$$

where $A = \pi DL = \pi\left(0.1\text{m} \times 25\text{m}\right) = 7.85\text{m}^2$.

Hence,

$$q = 7.85\text{m}^2\left[10\ \text{W/m}^2 \cdot \text{K}\left(150 - 25\right)\text{K} + 0.8 \times 5.67 \times 10^{-8}\ \text{W/m}^2 \cdot \text{K}^4\left(423^4 - 298^4\right)\text{K}^4\right]$$

$$q = 7.85\text{m}^2\left(1,250 + 1,095\right)\text{W/m}^2 = \left(9813 + 8592\right)\text{W} = 18,405\ \text{W}$$ <

(b) The annual energy loss is

$$E = qt = 18,405\ \text{W} \times 3600\ \text{s/h} \times 24\text{h/d} \times 365\ \text{d/y} = 5.80 \times 10^{11}\ \text{J}$$

With a furnace energy consumption of $E_f = E/\eta_f = 6.45 \times 10^{11}$ J, the annual cost of the loss is

$$C = C_g E_f = 0.02\ \text{\$/MJ} \times 6.45 \times 10^5 \text{MJ} = \$12,900$$ <

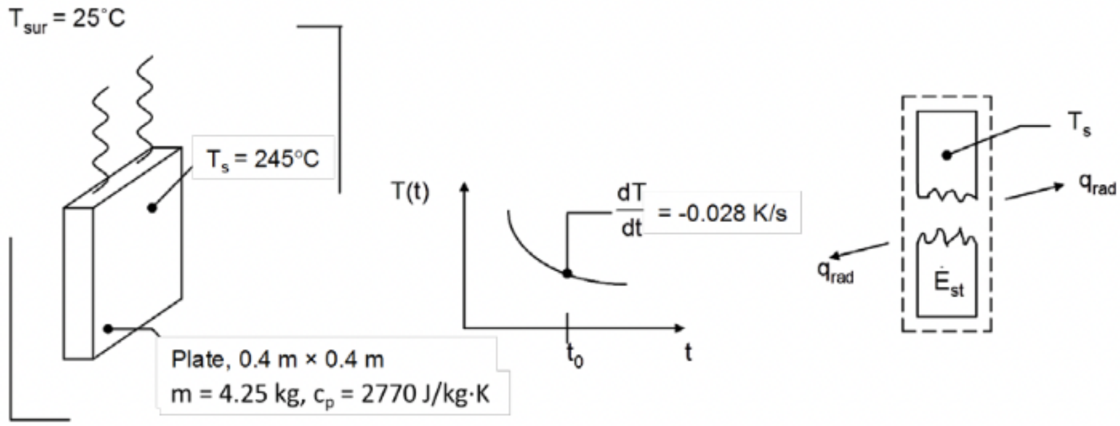
COMMENTS: The heat loss and related costs are unacceptable and should be reduced by insulating the steam line.

PROBLEM 1.26

KNOWN: Hot plate suspended in vacuum and surroundings temperature. Mass, specific heat, area and time rate of change of plate temperature.

FIND: (a) The emissivity of the plate, and (b) The rate at which radiation is emitted from the plate.

SCHEMATIC:



ASSUMPTIONS: (1) Plate is isothermal and at uniform temperature, (2) Large surroundings, (3) Negligible heat loss through suspension wires.

ANALYSIS: For a control volume about the plate, the conservation of energy requirement is

$$\dot{E}_{in} - \dot{E}_{out} = \dot{E}_{st} \qquad (1)$$

where $\dot{E}_{st} = mc_p \dfrac{dT}{dt}$ (2)

and for large surroundings $\dot{E}_{in} - \dot{E}_{out} = A\varepsilon\sigma(T_{sur}^4 - T_s^4)$ (3)

Combining Eqns. (1) through (3) yields

$$\varepsilon = \frac{mc_p}{A\sigma}\frac{\dfrac{dT}{dt}}{(T_{sur}^4 - T_s^4)}$$

Noting that $T_{sur} = 25^\circ C + 273\ K = 298\ K$ and $T_s = 245^\circ C + 273\ K = 518\ K$, we find

$$\varepsilon = \frac{4.25\ kg \times 2770 \dfrac{J}{kg \cdot K} \times (-0.028 \dfrac{K}{s})}{2 \times 0.4\ m \times 0.4\ m \times 5.67 \times 10^{-8} \dfrac{W}{m^2 \cdot K^4} (298^4 - 518^4)\ K^4} = 0.28 \qquad <$$

The rate at which radiation is emitted from the plate is

$$q_{rad,e} = \varepsilon A\sigma T_s^4 = 0.28 \times 2 \times 0.4\ m \times 0.4\ m \times 5.67 \times 10^{-8} \frac{W}{m^2 \cdot K^4} \times (518\ K)^4 = 370\ W \qquad <$$

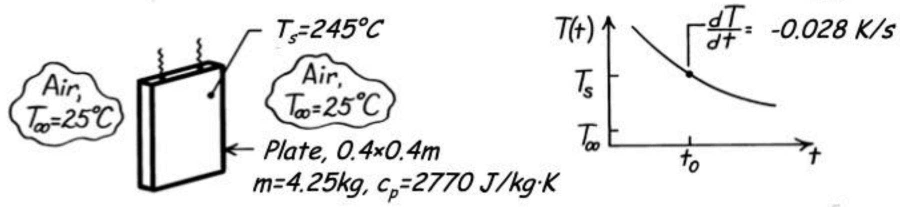
COMMENTS: Note the importance of using kelvins when working with radiation heat transfer.

PROBLEM 1.15

KNOWN: Hot vertical plate suspended in cool, still air. Change in plate temperature with time at the instant when the plate temperature is 245°C.

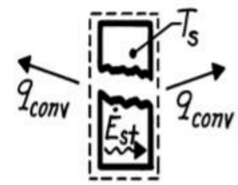
FIND: Convection heat transfer coefficient for this condition.

SCHEMATIC:



ASSUMPTIONS: (1) Plate is isothermal, (2) Negligible radiation exchange with surroundings, (3) Negligible heat lost through suspension wires.

ANALYSIS: As shown in the cooling curve above, the plate temperature decreases with time. The condition of interest is for time t_o. For a control surface about the plate, the conservation of energy requirement is



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

$$-2hA_s\left(T_s - T_\infty\right) = mc_p\frac{dT}{dt}$$

where A_s is the surface area of one side of the plate. Solving for h, find

$$h = \frac{mc_p}{2A_s\left(T_s - T_\infty\right)}\left(\frac{-dT}{dt}\right)$$

$$h = \frac{4.25\text{ kg} \times 2770\text{ J/kg}\cdot\text{K}}{2 \times \left(0.4 \times 0.4\right)\text{m}^2\left(245 - 25\right)\text{K}} \times 0.028\text{ K/s} = 4.7\text{ W/m}^2\cdot\text{K}$$ <

COMMENTS: (1) Assuming the plate is very highly polished with emissivity of 0.08, determine whether radiation exchange with the surroundings at 25°C is negligible compared to convection.

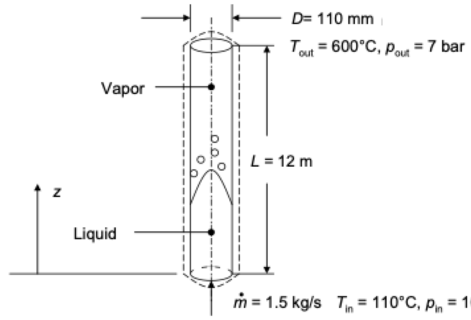
(2) We will later consider the criterion for determining whether the isothermal plate assumption is reasonable. If the thermal conductivity of the present plate were high (such as aluminum or copper), the criterion would be satisfied.

PROBLEM 1.30

KNOWN: Inlet and outlet conditions for flow of water in a vertical tube.

FIND: (a) Change in combined thermal and flow work, (b) change in mechanical energy, and (c) change in total energy of the water from the inlet to the outlet of the tube, (d) heat transfer rate, q.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Uniform velocity distributions at the tube inlet and outlet.

PROPERTIES: Table A.6 water (T = 110°C): ρ = 950 kg/m^3, (T = (179.9°C + 110 °C)/2 = 145°C): c_p = 4300 J/kg·K, ρ = 919 kg/m^3. Other properties are taken from Moran, M.J. and Shapiro, H.N., *Fundamentals of Engineering Thermodynamics*, 6th Edition, John Wiley & Sons, Hoboken, 2008 including (p_{sat} = 10 bar): T_{sat} = 179.9°C, i_f = 762.81 kJ/kg; (p = 7 bar, T = 600°C): i = 3700.2 kJ/kg, υ = 0.5738 m^3/kg.

ANALYSIS: The steady-flow energy equation, in the absence of work (other than flow work), is

$$\dot{m}\left(u + pv + \tfrac{1}{2}V^2 + gz\right)_{in} - \dot{m}\left(u + pv + \tfrac{1}{2}V^2 + gz\right)_{out} + q = 0$$
$$\dot{m}\left(i + \tfrac{1}{2}V^2 + gz\right)_{in} - \dot{m}\left(i + \tfrac{1}{2}V^2 + gz\right)_{out} + q = 0 \qquad (1)$$

while the conservation of mass principle yields

$$V_{in} = \frac{4\dot{m}}{\rho\pi D^2} = \frac{4\times 1.5 \text{ kg/s}}{950 \text{ kg/m}^3 \times \pi \times (0.110 \text{ m})^2} = 0.166 \text{ m/s} \,;\; V_{out} = \frac{\upsilon 4\dot{m}}{\pi D^2} = \frac{0.5738 \text{ m}^3/\text{kg}\times 4\times 1.5 \text{ kg/s}}{\pi\times(0.110 \text{ m})^2} = 90.6 \text{ m/s}$$

(a) The change in the combined thermal and flow work energy from inlet to outlet:

$$E_{i,out} - E_{i,in} = \dot{m}(i)_{out} - \dot{m}(i)_{in} = \dot{m}(i)_{out} - \dot{m}[i_{f,sat} + c_p(T_{in} - T_{sat})]$$
$$= 1.5 \text{ kg/s}\times\left(3700.2 \text{ kJ/kg} - [762.81 \text{ kJ/kg} + 4.3 \text{ kJ/kg}\cdot\text{K}\times(110-179.9)^\circ\text{C}]\right) \quad <$$
$$= 4.86 \text{ MW}$$

where $i_{f,sat}$ is the enthalpy of saturated liquid at the phase change temperature and pressure.

(b) The change in mechanical energy from inlet to outlet is:

Continued…

PROBLEM 1.30 (cont.)

$$E_{m,out} - E_{m,in} = \dot{m}(\tfrac{1}{2}V^2 + gz)_{out} - \dot{m}(\tfrac{1}{2}V^2 + gz)_{in}$$
$$= 1.5 \text{ kg/s}\times\left(\tfrac{1}{2}\left[(90.6 \text{ m/s})^2 - (0.166 \text{ m/s})^2\right] + 9.8 \text{ m/s}^2\times 12 \text{ m}\right) = 6.33 \text{ kW} \quad <$$

(c) The change in the total energy is the summation of the thermal, flow work, and mechanical energy change or

$$E_{in} - E_{out} = 4.86 \text{ MW} + 6.33 \text{ kW} = 4.87 \text{ MW} \quad <$$

(d) The total heat transfer rate is the same as the total energy change, $q = E_{in} - E_{out} = 4.87$ MW $<$

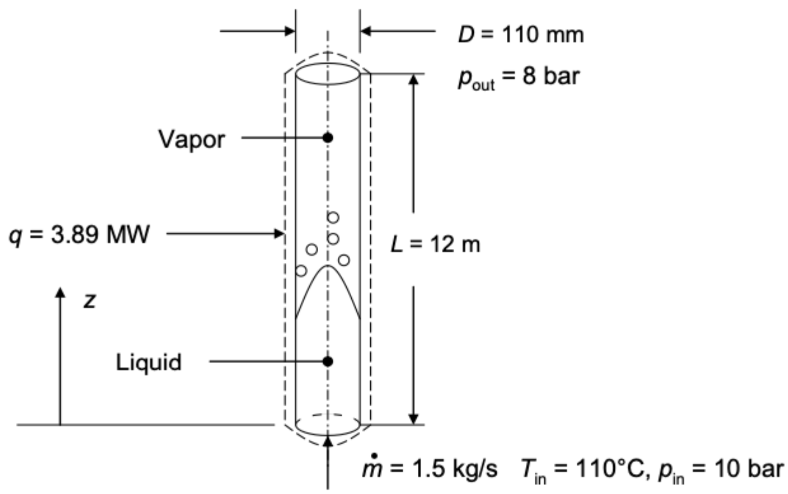
COMMENTS: (1) The change in mechanical energy, consisting of kinetic and potential energy components, is negligible compared to the change in thermal and flow work energy. (2) The average heat flux at the tube surface is $q'' = q/(\pi DL) = 4.87\text{MW}/(\pi\times 0.110 \text{ m}\times 12 \text{ m}) = 1.17 \text{ MW/m}^2$, which is very large. (3) The change in the velocity of the water is inversely proportional to the change in the density. As such, the outlet velocity is very large, and large pressure drops will occur in the vapor region of the tube relative to the liquid region of the tube.

PROBLEM 1.31

KNOWN: Flow of water in a vertical tube. Tube dimensions. Mass flow rate. Inlet pressure and temperature. Heat rate. Outlet pressure.

FIND: (a) Outlet temperature, (b) change in combined thermal and flow work, (c) change in mechanical energy, and (d) change in total energy of the water from the inlet to the outlet of the tube.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Negligible change in mechanical energy. (3) Uniform velocity distributions at the tube inlet and outlet.

PROPERTIES: Table A.6 water (T = 110°C): ρ = 950 kg/m^3, (T = (179.9°C + 110 °C)/2 = 145°C): c_p = 4300 J/kg·K, ρ = 919 kg/m^3. Other properties are taken from Moran, M.J. and Shapiro, H.N., *Fundamentals of Engineering Thermodynamics*, 6th Edition, John Wiley & Sons, Hoboken, 2008 including (p_{sat} = 10 bar): T_{sat} = 179.9°C, i_f = 762.81 kJ/kg; (p = 8 bar, i = 3056 kJ/kg): T = 300°C, υ = 0.3335 m^3/kg.

ANALYSIS: (a) The steady-flow energy equation, in the absence of work (other than flow work), is

$$\dot{m}\left(u + pv + \tfrac{1}{2}V^2 + gz\right)_{\text{in}} - \dot{m}\left(u + pv + \tfrac{1}{2}V^2 + gz\right)_{\text{out}} + q = 0$$
$$\dot{m}\left(i + \tfrac{1}{2}V^2 + gz\right)_{\text{in}} - \dot{m}\left(i + \tfrac{1}{2}V^2 + gz\right)_{\text{out}} + q = 0 \qquad (1)$$

Neglecting the change in mechanical energy yields

$$\dot{m}(i_{\text{in}} - i_{\text{out}}) + q = 0$$

The inlet enthalpy is

$$i_{\text{in}} = i_{f,\text{in}} + c_p(T_{\text{in}} - T_{\text{sat}}) = 762.81 \text{ kJ/kg} + 4.3 \text{ kJ/kg}\cdot\text{K} \times (110 - 179.9)°\text{C} = 462.2 \text{ kJ/kg}$$

Thus the outlet enthalpy is

$$i_{\text{out}} = i_{\text{in}} + q / \dot{m} = 462.2 \text{ kJ/kg} + 3890 \text{ kW} / 1.5 \text{ kg/s} = 3056 \text{ kJ/kg}$$

Continued…

and the outlet temperature can be found from thermodynamic tables at $p = 8$ bars, $i = 3056$ kJ/kg, for which

$$T_{\text{out}} = 300°\text{C} \qquad <$$

(b) The change in the combined thermal and flow work energy from inlet to outlet:

$$E_{i,\text{out}} - E_{i,\text{in}} = \dot{m}(i_{\text{out}} - i_{\text{in}}) = q = 3.89 \text{ MW} \qquad <$$

(c) The change in mechanical energy can now be calculated. First, the outlet specific volume can be found from thermodynamic tables at $T_{\text{out}} = 300°\text{C}$, $p_{\text{out}} = 8$ bars, $\upsilon = 0.3335 \text{ m}^3/\text{kg}$. Next, the conservation of mass principle yields

$$V_{\text{in}} = \frac{4\dot{m}}{\rho\pi D^2} = \frac{4\times 1.5 \text{ kg/s}}{950 \text{ kg/m}^3 \times \pi \times (0.110 \text{ m})^2} = 0.166 \text{ m/s} \; ; \; V_{\text{out}} = \frac{\upsilon 4\dot{m}}{\pi D^2} = \frac{0.3335 \text{ m}^3/\text{kg} \times 4 \times 1.5 \text{ kg/s}}{\pi \times (0.110 \text{ m})^2} = 52.6 \text{ m/s}$$

The change in mechanical energy from inlet to outlet is:

$$\begin{aligned} E_{m,\text{out}} - E_{m,\text{in}} &= \dot{m}(\tfrac{1}{2}V^2 + gz)_{\text{out}} - \dot{m}(\tfrac{1}{2}V^2 + gz)_{\text{in}} \\ &= 1.5 \text{ kg/s} \times \left(\tfrac{1}{2}\left[(52.6 \text{ m/s})^2 - (0.166 \text{ m/s})^2\right] + 9.8 \text{ m/s}^2 \times 12 \text{ m}\right) = 2.25 \text{ kW} \qquad < \end{aligned}$$

(d) The change in the total energy is the summation of the thermal, flow work, and mechanical energy change or

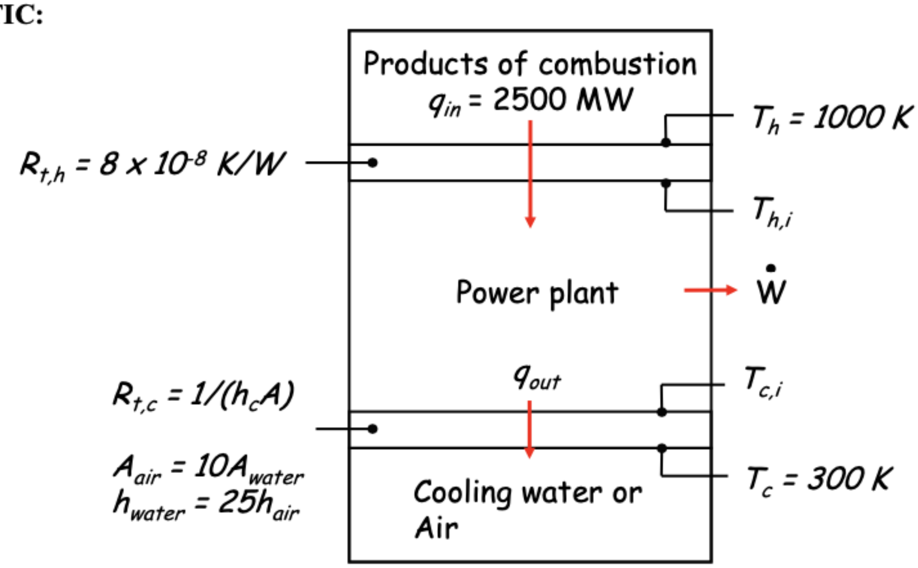
$$E_{\text{in}} - E_{\text{out}} = 3.89 \text{ MW} + 2.25 \text{ kW} = 3.89 \text{ MW} \qquad <$$

COMMENTS: (1) The change in mechanical energy, consisting of kinetic and potential energy components, is negligible compared to the change in thermal and flow work energy. (2) The average heat flux at the tube surface is $q'' = q/(\pi DL) = 3.89 \text{ MW}/(\pi \times 0.110 \text{ m} \times 12 \text{ m}) = 0.94 \text{ MW/m}^2$, which is very large. (3) The change in the velocity of the water is inversely proportional to the change in the density. As such, the outlet velocity is very large, and large pressure drops will occur in the vapor region of the tube relative to the liquid region of the tube.

KNOWN: Power plant and operating conditions of Example 1.7. Change in cold-side heat transfer surface area and convection heat transfer coefficient.

FIND: Modified efficiency and power output.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) power plant operates as an internally reversible heat engine, (3) clean operating conditions.

ANALYSIS: The cold-side thermal resistance for water cooling (for design conditions) is provided in Example 1.7 and is $R_{t,c} = 2 \times 10^{-8}$ K/W. The cold side thermal resistance is given by $R_{t,c} = 1/(h_c A)$, therefore

$$\frac{R_{t,c,\text{air}}}{R_{t,c,\text{water}}} = \frac{(hA)_{\text{water}}}{(hA)_{\text{air}}} = \left(\frac{h_{\text{water}}}{h_{\text{air}}}\right) \times \left(\frac{A_{\text{water}}}{A_{\text{air}}}\right) = \frac{25}{10} = 2.5$$

Hence, $R_{t,c,\text{air}} = 2.5 \times 2 \times 10^{-8}$ K/W $= 5 \times 10^{-8}$ K/W and $R_{\text{tot,air}} = 8 \times 10^{-8} + 5 \times 10^{-8} = 13 \times 10^{-8}$ K/W.

The modified efficiency for the air-cooled condenser is

$$\eta_m = 1 - \frac{T_c}{T_h - q_{in} R_{\text{tot,air}}} = 1 - \frac{300\text{K}}{1000\text{K} - 2500 \times 10^6\,\text{W} \times 1.3 \times 10^{-7}\,\text{K/W}} = 0.556 \qquad <$$

The power output is

$$\dot{W} = q_{\text{in}} \eta_m = 2500\text{ MW} \times 0.556 = 1390\text{ MW} \qquad <$$

Continued …

The air-cooled condenser is both (1) more expensive and (2) leads to a lower plant efficiency and power output relative to the water-cooled condenser of Example 1.7.

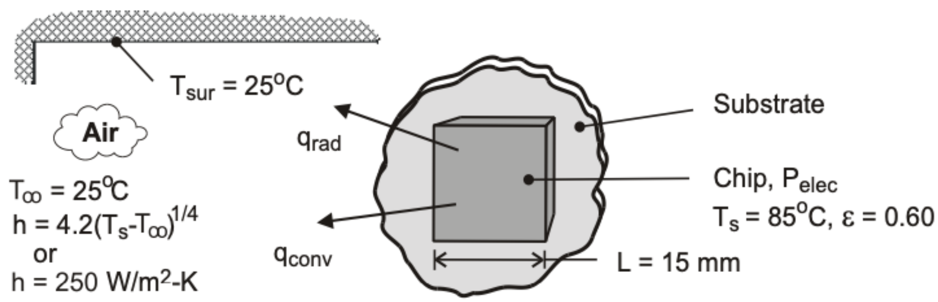
COMMENT: The diminished performance and higher cost of the air-cooled condenser, relative to the water-cooled condenser, is typical. This problem illustrates the profound linkage between power generation and water usage, and is referred to as "the water-energy nexus."

PROBLEM 1.35

KNOWN: Width, surface emissivity and maximum allowable temperature of an electronic chip. Temperature of air and surroundings. Convection coefficient.

FIND: (a) Maximum power dissipation for free convection with $h(W/m^2 \cdot K) = 4.2(T - T_\infty)^{1/4}$, (b) Maximum power dissipation for forced convection with $h = 250\ W/m^2 \cdot K$.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Radiation exchange between a small surface and a large enclosure, (3) Negligible heat transfer from sides of chip or from back of chip by conduction through the substrate.

ANALYSIS: Subject to the foregoing assumptions, electric power dissipation by the chip must be balanced by convection and radiation heat transfer from the chip. Hence, from Eq. (1.10),

$$P_{elec} = q_{conv} + q_{rad} = hA(T_s - T_\infty) + \varepsilon A\sigma\left(T_s^4 - T_{sur}^4\right)$$

where $A = L^2 = (0.015m)^2 = 2.25\times10^{-4}m^2$.

(a) If heat transfer is by natural convection,

$$q_{conv} = C\,A(T_s - T_\infty)^{5/4} = 4.2\ W/m^2 \cdot K^{5/4}\left(2.25\times10^{-4}m^2\right)(60K)^{5/4} = 0.158\ W$$

$$q_{rad} = 0.60\left(2.25\times10^{-4}m^2\right)5.67\times10^{-8}\ W/m^2 \cdot K^4\left(358^4 - 298^4\right)K^4 = 0.065\ W$$

$$P_{elec} = 0.158\ W + 0.065\ W = 0.223\ W$$ <

(b) If heat transfer is by forced convection,

$$q_{conv} = hA(T_s - T_\infty) = 250\ W/m^2 \cdot K\left(2.25\times10^{-4}m^2\right)(60K) = 3.375\ W$$

$$P_{elec} = 3.375\ W + 0.065\ W = 3.44\ W$$ <

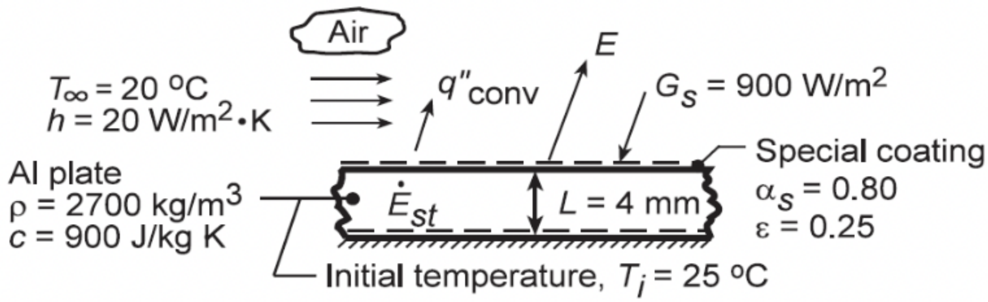
COMMENTS: Clearly, radiation and natural convection are inefficient mechanisms for transferring heat from the chip. For $T_s = 85°C$ and $T_\infty = 25°C$, the natural convection coefficient is 11.7 $W/m^2 \cdot K$. Even for forced convection with $h = 250\ W/m^2 \cdot K$, the power dissipation is well below that associated with many of today's processors. To provide acceptable cooling, it is often necessary to attach the chip to a highly conducting substrate and to thereby provide an additional heat transfer mechanism due to conduction from the back surface.

PROBLEM 1.40

KNOWN: Thickness and initial temperature of an aluminum plate whose thermal environment is changed.

FIND: (a) Initial rate of temperature change, (b) Steady-state temperature of plate.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible end effects, (2) Uniform plate temperature at any instant, (3) Constant properties, (4) Adiabatic bottom surface, (5) Negligible radiation from surroundings, (6) No internal heat generation.

ANALYSIS: (a) Applying an energy balance, Eq. 1.12c, at an instant of time to a control volume about the plate, $\dot{E}_{in} - \dot{E}_{out} = \dot{E}_{st}$, it follows for a unit surface area.

$$\alpha_S G_S \left(1\,m^2\right) - E\left(1\,m^2\right) - q''_{conv}\left(1\,m^2\right) = (d/dt)(McT) = \rho\left(1\,m^2 \times L\right)c\,(dT/dt).$$

Rearranging and substituting from Eqs. 1.3 and 1.7, we obtain

$$dT/dt = (1/\rho Lc)\left[\alpha_S G_S - \varepsilon\sigma T_i^4 - h\left(T_i - T_\infty\right)\right].$$

$$dT/dt = \left(2700\,kg/m^3 \times 0.004\,m \times 900\,J/kg\cdot K\right)^{-1} \times$$

$$\left[0.8 \times 900\,W/m^2 - 0.25 \times 5.67 \times 10^{-8}\,W/m^2\cdot K^4\,(298\,K)^4 - 20\,W/m^2\cdot K\,(25-20)^\circ C\right]$$

$$dT/dt = 0.052^\circ\,C/s.$$ <

(b) Under steady-state conditions, $\dot{E}_{st} = 0$, and the energy balance reduces to

$$\alpha_S G_S = \varepsilon\sigma T^4 + h\left(T - T_\infty\right) \quad (2)$$

$$0.8 \times 900\,W/m^2 = 0.25 \times 5.67 \times 10^{-8}\,W/m^2\cdot K^4 \times T^4 + 20\,W/m^2\cdot K\left(T - 293\,K\right)$$

The solution yields T = 321.4 K = 48.4°C. <

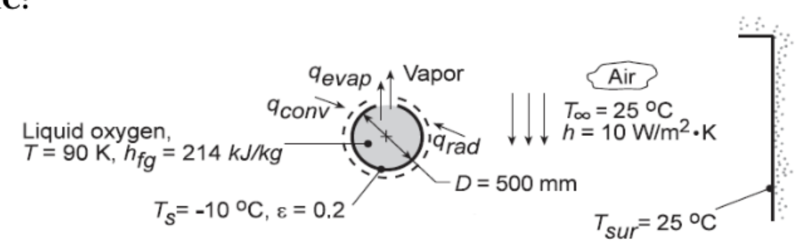
COMMENTS: The surface radiative properties have a significant effect on the plate temperature, which would decrease with increasing ε and decreasing α_S. If a low temperature is desired, the plate coating should be characterized by a large value of ε/α_S. The temperature would also decrease with increasing h.

PROBLEM 1.42

KNOWN: Boiling point and latent heat of liquid oxygen. Diameter and emissivity of container. Free convection coefficient and temperature of surrounding air and walls.

FIND: Mass evaporation rate.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Temperature of container outer surface equals boiling point of oxygen.

ANALYSIS: (a) Applying mass and energy balances to a control surface about the container, it follows that, at any instant,

$$\frac{dm_{st}}{dt} = -\dot{m}_{out} = -\dot{m}_{evap} \qquad \frac{dE_{st}}{dt} = \dot{E}_{in} - \dot{E}_{out} = q_{conv} + q_{rad} - q_{evap}. \qquad (1a,b)$$

With h_f as the enthalpy of liquid oxygen and h_g as the enthalpy of oxygen vapor, we have

$$E_{st} = m_{st} h_f \qquad q_{evap} = \dot{m}_{out} h_g \qquad (2a,b)$$

Combining Equations (1a) and (2a,b), Equation (1b) becomes (with $h_{fg} = h_g - h_f$)

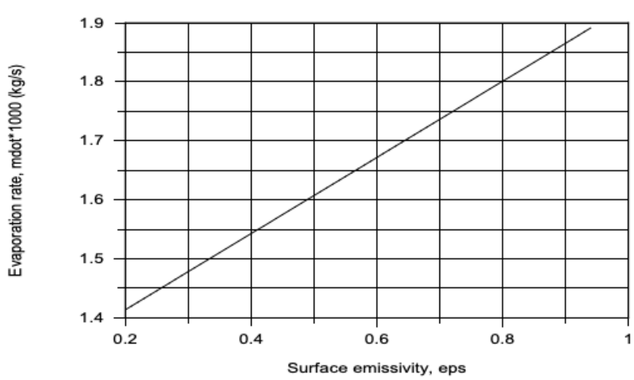
$$\dot{m}_{out} h_{fg} = q_{conv} + q_{rad}$$

$$\dot{m}_{evap} = (q_{conv} + q_{rad})/h_{fg} = \left[h(T_\infty - T_s) + \varepsilon\sigma\left(T_{sur}^4 - T_s^4\right)\right]\pi D^2/h_{fg} \qquad (3)$$

$$\dot{m}_{evap} = \frac{\left[10\,W/m^2\cdot K(298-263)K + 0.2\times5.67\times10^{-8}\,W/m^2\cdot K^4\left(298^4-263^4\right)K^4\right]\pi(0.5\,m)^2}{214\,kJ/kg}$$

$$\dot{m}_{evap} = (350+35.2)\,W/m^2\left(0.785\,m^2\right)/214\,kJ/kg = 1.41\times10^{-3}\,kg/s.$$ <

(b) Using Equation (3), the mass rate of vapor production can be determined for the range of emissivity 0.2 to 0.94. The effect of increasing emissivity is to increase the heat rate into the container and, hence, increase the vapor production rate.



COMMENTS: To reduce the loss of oxygen due to vapor production, insulation should be applied to the outer surface of the container, in order to reduce q_{conv} and q_{rad}. Note from the calculations in part (a), that heat transfer by convection is greater than by radiation exchange.