Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

5th Edition

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Chapter 10

BOILING AND CONDENSATION

Boiling Heat Transfer

10-1C Boiling is the liquid-to-vapor phase change process that occurs at a solid-liquid interface when the surface is heated above the saturation temperature of the liquid. The formation and rise of the bubbles and the liquid entrainment coupled with the large amount of heat absorbed during liquid-vapor phase change at essentially constant temperature are responsible for the very high heat transfer coefficients associated with nucleate boiling.

10-2C The different boiling regimes that occur in a vertical tube during flow boiling are forced convection of liquid, bubbly flow, slug flow, annular flow, transition flow, mist flow, and forced convection of vapor.

10-3C Both boiling and evaporation are liquid-to-vapor phase change processes, but evaporation occurs at the *liquid-vapor interface* when the vapor pressure is less than the saturation pressure of the liquid at a given temperature, and it involves no bubble formation or bubble motion. Boiling, on the other hand, occurs at the *solid-liquid interface* when a liquid is brought into contact with a surface maintained at a temperature T_s sufficiently above the saturation temperature T_{sat} of the liquid.

10-4C Boiling is called *pool boiling* in the absence of bulk fluid flow, and *flow boiling* (or *forced convection boiling*) in the presence of it. In pool boiling, the fluid is stationary, and any motion of the fluid is due to natural convection currents and the motion of the bubbles due to the influence of buoyancy.

10-5C The boiling curve is given in Figure 10-6 in the text. In the *natural convection boiling* regime, the fluid motion is governed by natural convection currents, and heat transfer from the heating surface to the fluid is by natural convection. In the *nucleate boiling* regime, bubbles form at various preferential sites on the heating surface, and rise to the top. In the *transition boiling* regime, part of the surface is covered by a vapor film. In the *film boiling* regime, the heater surface is completely covered by a continuous stable vapor film, and heat transfer is by combined convection and radiation.

10-6C In the *film boiling* regime, the heater surface is completely covered by a continuous stable vapor film, and heat transfer is by combined convection and radiation. In the *nucleate boiling* regime, the heater surface is covered by the liquid. The boiling heat flux in the stable film boiling regime can be higher or lower than that in the nucleate boiling regime, as can be seen from the boiling curve.

10-7C The boiling curve is given in Figure 10-6 in the text. The burnout point in the curve is point C. The *burnout* during boiling is caused by the heater surface being blanketed by a continuous layer of vapor film at increased heat fluxes, and the resulting rise in heater surface temperature in order to maintain the same heat transfer rate across a low-conducting vapor film. Any attempt to increase the heat flux beyond $\dot{q}_{\max}$ will cause the operation point on the boiling curve to jump suddenly from point C to point E. However, the surface temperature that corresponds to point E is beyond the melting point of most heater materials, and burnout occurs. The burnout point is avoided in the design of boilers in order to avoid the disastrous explosions of the boilers.

10-8C Pool boiling heat transfer can be increased *permanently* by increasing the number of nucleation sites on the heater surface by *coating* the surface with a thin layer (much less than 1 mm) of very porous material, or by *forming cavities* on the surface mechanically to facilitate continuous vapor formation. Such surfaces are reported to enhance heat transfer in the nucleate boiling regime by a factor of up to 10, and the critical heat flux by a factor of 3. The use of finned surfaces is also known to enhance nucleate boiling heat transfer and the critical heat flux.

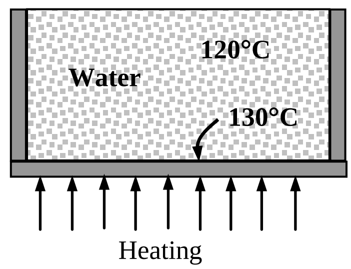
10-9C Yes. Otherwise we can create energy by alternately vaporizing and condensing a substance.

10-10 Water is boiled at T_{sat} = 120°C in a mechanically polished stainless steel pressure cooker whose inner surface temperature is maintained at T_s = 130°C. The heat flux on the surface is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 120°C are (Tables 10-1 and A-9)

$$\rho_l = 943.4\ \text{kg/m}^3 \quad h_{fg} = 2203\times 10^3\ \text{J/kg}$$
$$\rho_v = 1.121\ \text{kg/m}^3 \quad \mu_l = 0.232\times 10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\sigma = 0.0550\ \text{N/m} \quad c_{pl} = 4244\ \text{J/kg}\cdot{}^\circ\text{C}$$
$$\text{Pr}_l = 1.44$$



Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}} = 130 - 120 = 10^\circ\text{C}$ which is relatively low (less than 30°C). Therefore, nucleate boiling will occur. The heat flux in this case can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n}\right)^3$$
$$= (0.232\times 10^{-3})(2203\times 10^3)\left[\frac{9.81(943.4\text{-}1.121)}{0.0550}\right]^{1/2}\left(\frac{4244(130-120)}{0.0130(2203\times 10^3)1.44}\right)^3$$
$$= 228{,}400\ \text{W/m}^2 = \mathbf{228.4\ kW/m^2}$$

10-11 The nucleate pool boiling heat transfer rate per unit length and the rate of evaporation per unit length of water being boiled by a rod that is maintained at 10°C above the saturation temperature are to be determined.

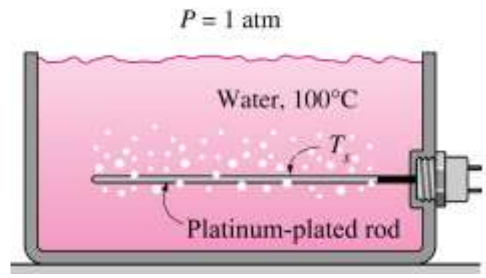
Assumptions **1** Steady operating condition exists. **2** Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are σ = 0.0589 N/m (Table 10-1) and, from Table A-9,

ρ_l = 957.9 kg/m^3 $\qquad$ h_{fg} = 2257 × 10^3 J/kg

ρ_v = 0.5978 kg/m^3 $\qquad$ μ_l = 0.282 × 10^{-3} kg/m·s

Pr$_l$ = 1.75 $\qquad$ c_{pl} = 4217 J/kg·K



Also, C_{sf} = 0.0130 and n = 1.0 for the boiling of water on platinum surface (Table 10-3).

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}}$ = 10°C, which is less than 30°C for water from Fig. 10-6. Therefore, nucleate boiling will occur. The heat flux in this case can be determined from the Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left[\frac{c_{pl}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \Pr_l^n}\right]^3$$

$$= (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2}\left[\frac{4217(10)}{(0.013)(2257\times10^3)(1.75)}\right]^3$$

$$= 1.408\times10^5 \text{ W/m}^2$$

Finally, the nucleate pool boiling heat transfer rate per unit length is

$$\dot{Q}_{\text{boiling}}/L = \pi D\dot{q}_{\text{nucleate}} = \pi(0.010\text{ m})(1.408\times10^5 \text{ W/m}^2) = \mathbf{4420\ W/m}$$

The rate of evaporation per unit length is

$$\frac{\dot{m}_{\text{evaporation}}}{L} = \frac{(\dot{Q}_{\text{boiling}}/L)}{h_{fg}} = \frac{4420 \text{ J/s}\cdot\text{m}}{2257\times10^3 \text{ J/kg}} = \mathbf{1.96\times10^{-3}\ kg/s\cdot m}$$

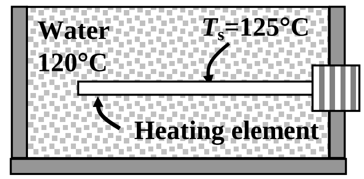
Discussion The value for the rate of evaporation per unit length indicates that 1 m of the platinum-plated rod would boil water at a rate of about 2 grams per second.

10-12 Water is boiled at a saturation (or boiling) temperature of T_{sat} = 120°C by a brass heating element whose temperature is not to exceed T_s = 125°C. The highest rate of steam production is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the boiler are negligible. **3** The boiling regime is nucleate boiling since $\Delta T = T_s - T_{sat} = 125 - 120 = 5°\text{C}$ which is in the nucleate boiling range of 5 to 30°C for water.

Properties The properties of water at the saturation temperature of 120°C are (Tables 10-1 and A-9)

$$\rho_l = 943.4\ \text{kg/m}^3$$
$$\rho_v = 1.12\ \text{kg/m}^3$$
$$\sigma = 0.0550\ \text{N/m}$$
$$\text{Pr}_l = 1.44$$
$$h_{fg} = 2203 \times 10^3\ \text{J/kg}$$
$$\mu_l = 0.232 \times 10^{-3}\ \text{kg}\cdot\text{m/s}$$
$$c_{pl} = 4244\ \text{J/kg}\cdot°\text{C}$$



Also, $C_{sf} = 0.0060$ and n = 1.0 for the boiling of water on a brass surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis Assuming nucleate boiling, the heat flux in this case can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg}\ \text{Pr}_l^n}\right)^3$$

$$= (0.232\times10^{-3})(2203\times10^3)\left[\frac{9.81(943.4-1.12)}{0.0550}\right]^{1/2}\left(\frac{4244(125-120)}{0.0060(2203\times10^3)1.44}\right)^3$$

$$= 290{,}300\ \text{W/m}^2$$

The surface area of the heater is

$$A_s = \pi DL = \pi(0.02\ \text{m})(0.65\ \text{m}) = 0.04084\ \text{m}^2$$

Then the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A_s \dot{q}_{\text{nucleate}} = (0.04084\ \text{m}^2)(290{,}300\ \text{W/m}^2) = 11{,}856\ \text{W}$$

(b) The rate of evaporation of water is determined from

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{11{,}856\ \text{J/s}}{2203\times10^3\ \text{J/kg}}\left(\frac{3600\ \text{s}}{1\ \text{h}}\right) = \mathbf{19.4\ kg/h}$$

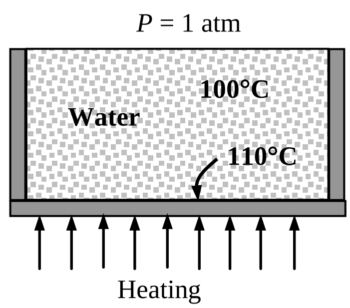
Therefore, steam can be produced at a rate of about 20 kg/h by this heater.

10-13 Water is boiled at 1 atm pressure and thus at a saturation (or boiling) temperature of T_{sat} = 100°C in a mechanically polished stainless steel pan whose inner surface temperature is maintained at T_s = 110°C. The rate of heat transfer to the water and the rate of evaporation of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)

$$\rho_l = 957.9 \text{ kg/m}^3$$
$$\rho_v = 0.60 \text{ kg/m}^3$$
$$\sigma = 0.0589 \text{ N/m}$$
$$\text{Pr}_l = 1.75$$
$$h_{fg} = 2257 \times 10^3 \text{ J/kg}$$
$$\mu_l = 0.282 \times 10^{-3} \text{ kg} \cdot \text{m/s}$$
$$c_{pl} = 4217 \text{ J/kg} \cdot °\text{C}$$



Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}} = 110 - 100 = 10°\text{C}$ which is relatively low (less than 30°C). Therefore, nucleate boiling will occur. The heat flux in this case can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right)^3$$
$$= (0.282 \times 10^{-3})(2257 \times 10^3) \left[\frac{9.8(957.9 - 0.60)}{0.0589} \right]^{1/2} \left(\frac{4217(110 - 100)}{0.0130(2257 \times 10^3)1.75} \right)^3$$
$$= 140{,}700 \text{ W/m}^2$$

The surface area of the bottom of the pan is

$$A_s = \pi D^2 / 4 = \pi (0.30 \text{ m})^2 / 4 = 0.07069 \text{ m}^2$$

Then the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A_s \dot{q}_{\text{nucleate}} = (0.07069 \text{ m}^2)(140{,}700 \text{ W/m}^2) = \mathbf{9945 \text{ W}}$$

(b) The rate of evaporation of water is determined from

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{9945 \text{ J/s}}{2257 \times 10^3 \text{ J/kg}} = \mathbf{0.00441 \text{ kg/s}}$$

That is, water in the pan will boil at a rate of 4.4 grams per second.

10-14 The nucleate pool boiling heat transfer coefficient of water being boiled by a horizontal platinum-plated rod is to be determined.

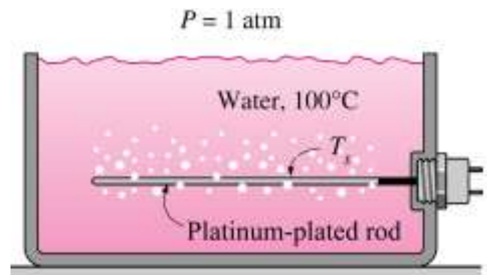
Assumptions **1** Steady operating condition exists. **2** Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are σ = 0.0589 N/m (Table 10-1) and, from Table A-9,

ρ_l = 957.9 kg/m^3 $\qquad$ h_{fg} = 2257 × 10^3 J/kg

ρ_v = 0.5978 kg/m^3 $\qquad$ μ_l = 0.282 × 10^{-3} kg/m·s

Pr$_l$ = 1.75 $\qquad$ c_{pl} = 4217 J/kg·K



Also, C_{sf} = 0.0130 and n = 1.0 for the boiling of water on platinum surface (Table 10-3).

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\rm sat}$ = 10°C, which is less than 30°C for water from Fig. 10-6. Therefore, nucleate boiling will occur. The heat flux in this case can be determined from the Rohsenow relation to be

$$\dot{q}_{\rm nucleate} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left[\frac{c_{pl}(T_s-T_{\rm sat})}{C_{sf}h_{fg}\,{\rm Pr}_l^n}\right]^3$$

$$= (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2}\left[\frac{4217(10)}{(0.013)(2257\times10^3)(1.75)}\right]^3$$

$$= 1.408\times10^5\ {\rm W/m^2}$$

Using the Newton's law of cooling, the boiling heat transfer coefficient is

$$\dot{q}_{\rm nucleate} = h(T_s - T_{\rm sat}) \qquad \rightarrow \qquad h = \frac{\dot{q}_{\rm nucleate}}{T_s - T_{\rm sat}}$$

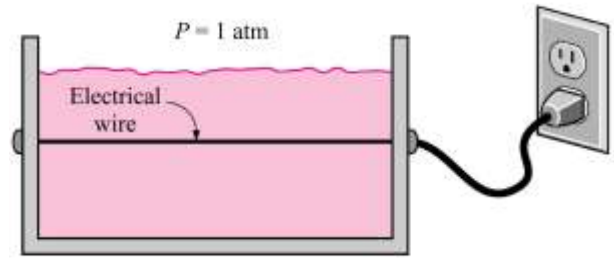
$$h = \frac{1.408\times10^5\ {\rm W/m^2}}{(110-100)\ {\rm K}} = \mathbf{14{,}100\ W/m^2\cdot K}$$

Discussion Heat transfer coefficient on the order of 10^4 W/m^2·K can be obtained in nucleate boiling with a temperature difference of just 10°C.

10-15 The nucleate boiling heat transfer coefficient and the value of C_{sf} for water being boiled by a long electrical wire are to be determined.

Assumptions **1** Steady operating condition exists. **2** Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are σ = 0.0589 N/m (Table 10-1) and, from Table A-9,



ρ_l = 957.9 kg/m^3 $\qquad$ h_{fg} = 2257 × 10^3 J/kg

ρ_v = 0.5978 kg/m^3 $\qquad$ μ_l = 0.282 × 10^{-3} kg/m·s

Pr$_l$ = 1.75 $\qquad$ c_{pl} = 4217 J/kg·K

Also, n = 1.0 is given.

Analysis The excess temperature in this case is $\Delta T = T_s - T_{sat}$ = 28°C, which is less than 30°C for water from Fig. 10-6. Therefore, nucleate boiling will occur. The nucleate boiling heat transfer coefficient can be determined using

$$\dot{q}_{\text{boiling}} = h(T_s - T_{\text{sat}}) \qquad \rightarrow \qquad h = \frac{\dot{q}_{\text{boiling}}}{T_s - T_{\text{sat}}}$$

Also, we know

$$\dot{Q}_{\text{boiling}} / L = \pi D \dot{q}_{\text{boiling}} = 4100 \text{ W/m}$$

$$\dot{q}_{\text{boiling}} = \frac{4100 \text{ W/m}}{\pi D} = \frac{4100 \text{ W/m}}{\pi(0.001 \text{ m})} = 1.305 \times 10^6 \text{ W/m}^2$$

Hence, the nucleate boiling heat transfer coefficient is

$$h = \frac{1.305 \times 10^6 \text{ W/m}}{(128 - 100) \text{ K}} = \mathbf{46{,}600\ W/m^2 \cdot K}$$

The value of the experimental constant C_{sf} can be determined from the Rohsenow relation to be

$$\dot{q}_{\text{boiling}} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{c_{pl}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right]^3$$

or

$$C_{sf} = \left\{ \frac{\mu_l h_{fg}}{\dot{q}_{\text{boiling}}} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{c_{pl}(T_s - T_{\text{sat}})}{h_{fg} \text{Pr}_l^n} \right]^3 \right\}^{1/3}$$

$$C_{sf} = \left\{ \frac{(0.282 \times 10^{-3})(2257 \times 10^3)}{1.305 \times 10^6} \left[\frac{9.81(957.9 - 0.5978)}{0.0589} \right]^{1/2} \left[\frac{4217(128 - 100)}{(2257 \times 10^3)(1.75)} \right]^3 \right\}^{1/3}$$

$$= \mathbf{0.0173}$$

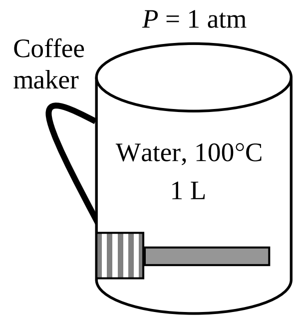
Discussion The boiling heat transfer coefficient of 46,600 W/m^2·K is within the range suggested by Table 1-5 for boiling and condensation (2500 to 100,000 W/m^2·K).

10-16 Water is boiled at sea level (1 atm pressure) and thus at a saturation (or boiling) temperature of T_{sat} = 100°C by a stainless steel heating element. The surface temperature of the heating element and its power rating are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the coffee maker are negligible. **3** The boiling regime is nucleate boiling (this assumption will be checked later).

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)

$$\rho_l = 957.9\ \text{kg/m}^3$$
$$\rho_v = 0.60\ \text{kg/m}^3$$
$$\sigma = 0.0589\ \text{N/m}$$
$$\Pr_l = 1.75$$
$$h_{fg} = 2257\times 10^3\ \text{J/kg}$$
$$\mu_l = 0.282\times 10^{-3}\ \text{kg}\cdot\text{m/s}$$
$$c_{pl} = 4217\ \text{J/kg}\cdot°\text{C}$$



Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 connection with their definitions in order to avoid unit manipulations.

Analysis The density of water at room temperature is very nearly 1 kg/L, and thus the mass of 1 L water at 18°C is nearly 1 kg. The rate of energy transfer needed to evaporate half of this water in 25 min and the heat flux are

$$Q = \dot{Q}\Delta t = mh_{fg} \;\rightarrow\; \dot{Q} = \frac{mh_{fg}}{\Delta t} = \frac{(0.5\,\text{kg})(2257\ \text{kJ/kg})}{(25\times 60\,\text{s})} = 0.7523\,\text{kW}$$

$$A_s = \pi DL = \pi(0.004\,\text{m})(0.20\,\text{m}) = 0.002513\,\text{m}^2$$

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{0.7523\,\text{kW}}{0.002513\,\text{m}^2} = 299.36\,\text{kW/m}^2 = 299{,}360\ \text{W/m}^2$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given.

Assuming nucleate boiling, the temperature of the inner surface of the pan is determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf}h_{fg}\Pr_l^n}\right)^3$$

$$299{,}360 = (0.282\times 10^{-3})(2257\times 10^3)\left[\frac{9.81(957.9-0.60)}{0.0589}\right]^{1/2}\left(\frac{4217(T_s-100)}{0.0130(2257\times 10^3)1.75}\right)^3$$

It gives

$$T_s = \mathbf{112.9°C}$$

which is in the nucleate boiling range (5 to 30°C above surface temperature). Therefore, the nucleate boiling assumption is valid.

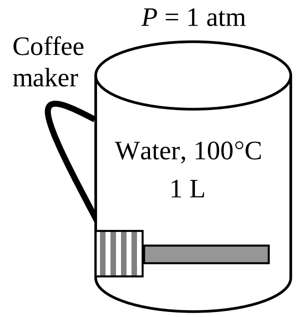
The specific heat of water at the average temperature of (14+100)/2 = 57°C is c_p = 4.184 kJ/kg·°C. Then the time it takes for the entire water to be heated from 14°C to 100°C is determined to be

$$Q = \dot{Q}\Delta t = mc_p\Delta T \;\rightarrow\; \Delta t = \frac{mc_p\Delta T}{\dot{Q}} = \frac{(1\,\text{kg})(4.184\,\text{kJ/kg}\cdot°\text{C})(100-14)°\text{C}}{0.7523\,\text{kJ/s}} = 478\,\text{s} = \mathbf{7.97\,min}$$

10-17 Water is boiled at sea level (1 atm pressure) and thus at a saturation (or boiling) temperature of T_{sat} = 100°C by a copper heating element. The surface temperature of the heating element and its power rating are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the coffee maker are negligible. **3** The boiling regime is nucleate boiling (this assumption will be checked later).

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)



$$\rho_l = 957.9 \text{ kg/m}^3$$
$$\rho_v = 0.60 \text{ kg/m}^3$$
$$\sigma = 0.0589 \text{ N/m}$$
$$\text{Pr}_l = 1.75$$
$$h_{fg} = 2257\times 10^3 \text{ J/kg}$$
$$\mu_l = 0.282\times 10^{-3} \text{ kg}\cdot\text{m/s}$$
$$c_{pl} = 4217 \text{ J/kg}\cdot°\text{C}$$

Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a copper surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 connection with their definitions in order to avoid unit manipulations.

Analysis The density of water at room temperature is very nearly 1 kg/L, and thus the mass of 1 L water at 18°C is nearly 1 kg. The rate of energy transfer needed to evaporate half of this water in 25 min and the heat flux are

$$Q = \dot{Q}\Delta t = mh_{fg} \rightarrow \dot{Q} = \frac{mh_{fg}}{\Delta t} = \frac{(0.5\text{ kg})(2257\text{ kJ/kg})}{(25\times 60\text{ s})} = 0.7523\text{ kW}$$

$$A_s = \pi DL = \pi(0.004\text{ m})(0.20\text{ m}) = 0.002513\text{ m}^2$$

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{0.7523\text{ kW}}{0.002513\text{ m}^2} = 299.36\text{ kW/m}^2 = 299{,}360\text{ W/m}^2$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given.

Assuming nucleate boiling, the temperature of the inner surface of the pan is determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$

$$299{,}360 = (0.282\times 10^{-3})(2257\times 10^3)\left[\frac{9.81(957.9-0.60)}{0.0589}\right]^{1/2}\left(\frac{4217(T_s-100)}{0.0130(2257\times 10^3)1.75}\right)^3$$

It gives

$$T_s = \mathbf{112.9°C}$$

which is in the nucleate boiling range (5 to 30°C above surface temperature). Therefore, the nucleate boiling assumption is valid.

The specific heat of water at the average temperature of (14+100)/2 = 57°C is c_p = 4.184 kJ/kg·°C. Then the time it takes for the entire water to be heated from 14°C to 100°C is determined to be

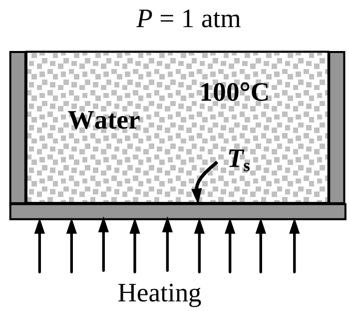
$$Q = \dot{Q}\Delta t = mc_p\Delta T \rightarrow \Delta t = \frac{mc_p\Delta T}{\dot{Q}} = \frac{(1\text{ kg})(4.184\text{ kJ/kg}\cdot°\text{C})(100-14)°\text{C}}{0.7523\text{ kJ/s}} = 478\text{ s} = \mathbf{7.97\text{ min}}$$

10-18 Water is boiled at sea level (1 atm pressure) and thus at a saturation (or boiling) temperature of T_{sat} = 100°C in a teflon-pitted stainless steel pan placed on an electric burner. The water level drops by 10 cm in 30 min during boiling. The inner surface temperature of the pan is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the pan are negligible. **3** The boiling regime is nucleate boiling (this assumption will be checked later).

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)

$$\rho_l = 957.9\ \text{kg/m}^3$$
$$\rho_v = 0.60\ \text{kg/m}^3$$
$$\sigma = 0.0589\ \text{N/m}$$
$$\text{Pr}_l = 1.75$$
$$h_{fg} = 2257\times10^3\ \text{J/kg}$$
$$\mu_l = 0.282\times10^{-3}\ \text{kg}\cdot\text{m/s}$$
$$c_{pl} = 4217\ \text{J/kg}\cdot°\text{C}$$



Also, $C_{sf} = 0.0058$ and n = 1.0 for the boiling of water on a teflon-pitted stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 connection with their definitions in order to avoid unit manipulations.

Analysis The rate of heat transfer to the water and the heat flux are

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{\rho \Delta V}{\Delta t} = \frac{(957.9\ \text{kg/m}^3)(\pi \times (0.2\ \text{m})^2\ /4 \times 0.10\ \text{m})}{15 \times 60\ \text{s}} = 0.003344\ \text{kg/s}$$

$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} = (0.03344\ \text{kg/s})(2257\ \text{kJ/kg}) = 7.547\ \text{kW}$$

$$A_s = \pi D^2/4 = \pi(0.20\ \text{m})^2/4 = 0.03142\ \text{m}^2$$

$$\dot{q} = \dot{Q}/A_s = (7547\ \text{W})/(0.03142\ \text{m}^2) = 240{,}200\ \text{W/m}^2$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given. Assuming nucleate boiling, the temperature of the inner surface of the pan is determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} \left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n}\right)^3$$

$$240{,}200 = (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.8(957.9 - 0.60)}{0.0589}\right]^{1/2} \left(\frac{4217(T_s - 100)}{0.0058(2257\times10^3)1.75}\right)^3$$

It gives

T_s = **105.3°C**

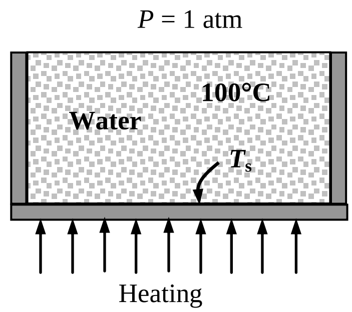
which is in the nucleate boiling range (5 to 30°C above surface temperature). Therefore, the nucleate boiling assumption is valid.

10-19 Water is boiled at sea level (1 atm pressure) and thus at a saturation (or boiling) temperature of T_{sat} = 100°C in a polished copper pan placed on an electric burner. The water level drops by 10 cm in 30 min during boiling. The inner surface temperature of the pan is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the pan are negligible. **3** The boiling regime is nucleate boiling (this assumption will be checked later).

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)

$$\begin{aligned}\rho_l &= 957.9\ \text{kg/m}^3\\ \rho_v &= 0.60\ \text{kg/m}^3\\ \sigma &= 0.0589\ \text{N/m}\\ \text{Pr}_l &= 1.75\\ h_{fg} &= 2257\times10^3\ \text{J/kg}\\ \mu_l &= 0.282\times10^{-3}\ \text{kg}\cdot\text{m/s}\\ c_{pl} &= 4217\ \text{J/kg}\cdot°\text{C}\end{aligned}$$



Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a copper surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 connection with their definitions in order to avoid unit manipulations.

Analysis The rate of heat transfer to the water and the heat flux are

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{\rho\Delta V}{\Delta t} = \frac{(957.9\ \text{kg/m}^3)(\pi\times(0.2\ \text{m})^2\ /4\times 0.10\ \text{m})}{15\times 60\ \text{s}} = 0.003344\ \text{kg/s}$$

$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} = (0.03344\ \text{kg/s})(2257\ \text{kJ/kg}) = 7.547\ \text{kW}$$

$$A_s = \pi D^2\ /\ 4 = \pi(0.20\ \text{m})^2\ /\ 4 = 0.03142\ \text{m}^2$$

$$\dot{q} = \dot{Q}\ /\ A_s = (7547\ \text{W})/(0.03142\ \text{m}^2) = 240{,}200\ \text{W/m}^2$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given. Assuming nucleate boiling, the temperature of the inner surface of the pan is determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\,\text{Pr}_l^n}\right)^3$$

$$240{,}200 = (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.8(957.9-0.60)}{0.0589}\right]^{1/2}\left(\frac{4217(T_s-100)}{0.0130(2257\times10^3)1.75}\right)^3$$

It gives

T_s = **111.9°C**

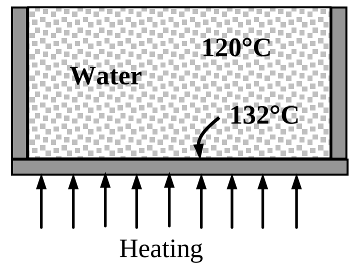
which is in the nucleate boiling range (5 to 30°C above surface temperature). Therefore, the nucleate boiling assumption is valid.

10-20 Water is boiled at T_{sat} = 120°C in a mechanically polished stainless steel pressure cooker whose inner surface temperature is maintained at T_s = 132°C. The boiling heat transfer coefficient is to be determined.

Assumptions 1 Steady operating conditions exist. **2** Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 120°C are (Tables 10-1 and A-9)

$$\rho_l = 943.4 \text{ kg/m}^3 \quad h_{fg} = 2203\times10^3 \text{ J/kg}$$
$$\rho_v = 1.121 \text{ kg/m}^3 \quad \mu_l = 0.232\times10^{-3} \text{ kg/m}\cdot\text{s}$$
$$\sigma = 0.0550 \text{ N/m} \quad c_{pl} = 4244 \text{ J/kg}\cdot°\text{C}$$
$$\text{Pr}_l = 1.44$$



Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis The excess temperature in this case is $\Delta T = T_s - T_{sat} = 132 - 120 = 12°\text{C}$ which is relatively low (less than 30°C). Therefore, nucleate boiling will occur. The heat flux in this case can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{sat})}{C_{sf} h_{fg} \text{Pr}_l^n}\right)^3$$

$$= (0.232\times10^{-3})(2203\times10^3)\left[\frac{9.81(943.4 - 1.121)}{0.0550}\right]^{1/2}\left(\frac{4244(132-120)}{0.0130(2203\times10^3)1.44}\right)^3$$

$$= 394{,}600 \text{ W/m}^2$$

The boiling heat transfer coefficient is

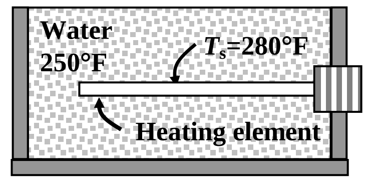
$$\dot{q}_{\text{nucleate}} = h(T_s - T_{sat}) \longrightarrow h = \frac{\dot{q}_{\text{nucleate}}}{T_s - T_{sat}} = \frac{394{,}600 \text{ W/m}^2}{(132-120)°\text{C}} = 32{,}880 \text{ W/m}^2\cdot°\text{C} = \mathbf{32.9 \text{ kW/m}^2\cdot\text{C}}$$

10-21E Water is boiled at a temperature of T_{sat} = 250°F by a nickel-plated heating element whose surface temperature is maintained at T_s = 280°F. The boiling heat transfer coefficient, the electric power consumed, and the rate of evaporation of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the boiler are negligible. **3** The boiling regime is nucleate boiling since $\Delta T = T_s - T_{sat} = 280 - 250 = 30°\text{ F}$ which is in the nucleate boiling range of 9 to 55°F for water.

Properties The properties of water at the saturation temperature of 250°F are (Tables 10-1 and A-9E)

$$\rho_l = 58.82\text{ lbm/ft}^3$$
$$\rho_v = 0.0723\text{ lbm/ft}^3$$
$$\sigma = 0.003755\text{ lbf/ft} = 0.1208\text{ lbm/s}^2$$
$$\text{Pr}_l = 1.43$$
$$h_{fg} = 946\text{ Btu/lbm}$$
$$\mu_l = 1.544\times10^{-4}\text{ lbm/ft}\cdot\text{s} = 0.556\text{ lbm/ft}\cdot\text{h}$$
$$c_{pl} = 1.015\text{ Btu/lbm}\cdot°\text{F}$$



Also, g = 32.2 ft/s^2 and C_{sf} = 0.0060 and n = 1.0 for the boiling of water on a nickel plated surface (Table 10-3). Note that we expressed the properties in units that will cancel each other in boiling heat transfer relations.

Analysis (*a*) Assuming nucleate boiling, the heat flux can be determined from Rohsenow relation to be

$$\begin{aligned}\dot{q}_{\text{nucleate}} &= \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\,\text{Pr}_l^n}\right)^3\\ &= (0.556)(946)\left[\frac{32.2(58.82-0.0723)}{0.1208}\right]^{1/2}\left(\frac{1.015(280-250)}{0.0060(946)1.43}\right)^3\\ &= 3{,}475{,}221\text{ Btu/h}\cdot\text{ft}^2\end{aligned}$$

Then the convection heat transfer coefficient becomes

$$\dot{q} = h(T_s - T_{\text{sat}}) \rightarrow h = \frac{\dot{q}}{T_s - T_{\text{sat}}} = \frac{3{,}475{,}221\text{Btu/h}\cdot\text{ft}^2}{(280-250)°\text{F}} = \mathbf{115{,}840\ Btu/h\cdot ft^2\cdot °F}$$

(*b*) The electric power consumed is equal to the rate of heat transfer to the water, and is determined from

$$\begin{aligned}\dot{W}_e &= \dot{Q} = \dot{q}A_s = (\pi DL)\dot{q} = (\pi\times0.5/12\text{ ft}\times2\text{ ft})(3{,}475{,}221\text{ Btu/h}\cdot\text{ft}^2) = 909{,}811\text{ Btu/h}\\ &= \mathbf{266.7\ kW} \quad (\text{since } 1\text{ kW} = 3412\text{ Btu/h})\end{aligned}$$

(*c*) Finally, the rate of evaporation of water is determined from

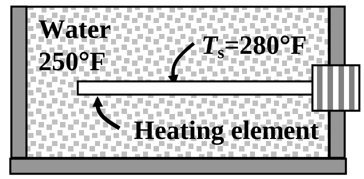
$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{909{,}811\text{ Btu/h}}{946\text{ Btu/lbm}} = \mathbf{961.7\ lbm/h}$$

10-22E Water is boiled at a temperature of T_{sat} = 250°F by a platinum-plated heating element whose surface temperature is maintained at T_s = 280°F. The boiling heat transfer coefficient, the electric power consumed, and the rate of evaporation of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the boiler are negligible. **3** The boiling regime is nucleate boiling since $\Delta T = T_s - T_{\text{sat}} = 280 - 250 = 30°\text{F}$ which is in the nucleate boiling range of 9 to 55°F for water.

Properties The properties of water at the saturation temperature of 250°F are (Tables 10-1 and A-9E)

$$\rho_l = 58.82 \text{ lbm/ft}^3$$
$$\rho_v = 0.0723 \text{ lbm/ft}^3$$
$$\sigma = 0.003755 \text{ lbf/ft} = 0.1208 \text{ lbm/s}^2$$
$$\text{Pr}_l = 1.43$$
$$h_{fg} = 946 \text{ Btu/lbm}$$
$$\mu_l = 1.544 \times 10^{-4} \text{ lbm/ft} \cdot \text{s} = 0.556 \text{ lbm/ft} \cdot \text{h}$$
$$c_{pl} = 1.015 \text{ Btu/lbm} \cdot °\text{F}$$



Also, g = 32.2 ft/s^2 and $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a platinum plated surface (Table 10-3). Note that we expressed the properties in units that will cancel each other in boiling heat transfer relations.

Analysis (*a*) Assuming nucleate boiling, the heat flux can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right)^3$$
$$= (0.556)(946) \left[\frac{32.2(58.82 - 0.0723)}{0.1208} \right]^{1/2} \left(\frac{1.015(280 - 250)}{0.0130(0.1208 \times 10^3)1.43} \right)^3$$
$$= 341{,}670 \text{ Btu/h} \cdot \text{ft}^2$$

Then the convection heat transfer coefficient becomes

$$\dot{q} = h(T_s - T_{\text{sat}}) \rightarrow h = \frac{\dot{q}}{T_s - T_{\text{sat}}} = \frac{341{,}670 \text{ Btu/h} \cdot \text{ft}^2}{(280 - 250)°\text{F}} = \mathbf{11{,}390 \text{ Btu/h} \cdot ft^2 \cdot °F}$$

(*b*) The electric power consumed is equal to the rate of heat transfer to the water, and is determined from

$$\dot{W}_e = \dot{Q} = \dot{q}A_s = (\pi DL)\dot{q} = (\pi \times 0.5/12 \text{ ft} \times 2 \text{ ft})(341{,}670 \text{ Btu/h} \cdot \text{ft}^2) = 89{,}449 \text{ Btu/h}$$
$$= \mathbf{26.2 \text{ kW}} \quad (\text{since } 1 \text{ kW} = 3412 \text{ Btu/h})$$

(*c*) Finally, the rate of evaporation of water is determined from

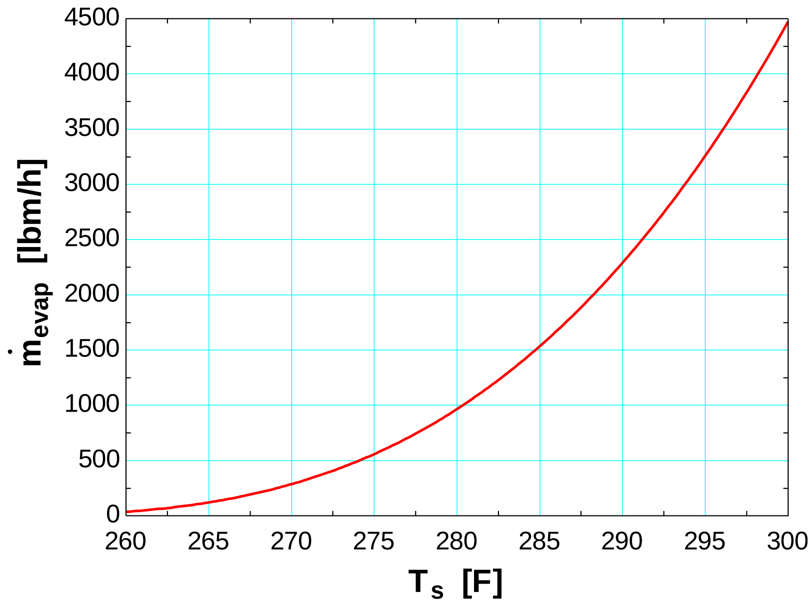
$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{89{,}449 \text{ Btu/h}}{946 \text{ Btu/lbm}} = \mathbf{94.6 \text{ lbm/h}}$$

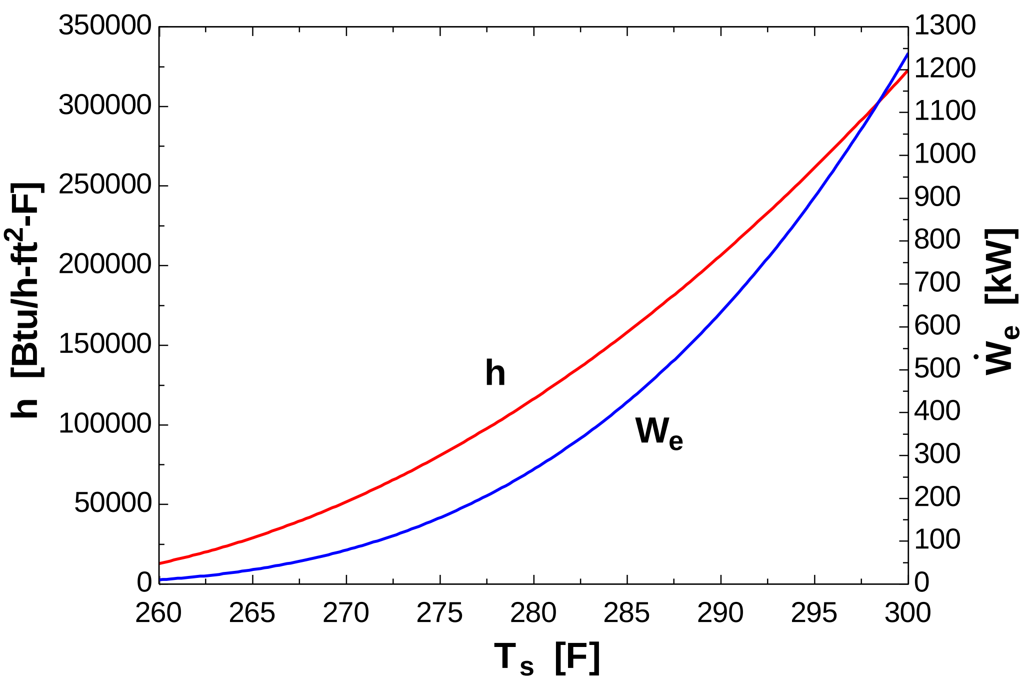
10-23E Prob. 10-22E is reconsidered. The effect of surface temperature of the heating element on the boiling heat transfer coefficient, the electric power, and the rate of evaporation of water is to be investigated.

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

```
"GIVEN"
T_sat=250 [F]
L=2 [ft]
D=0.5/12 [ft]
T_s=280 [F]

"PROPERTIES"
Fluid$='steam_IAPWS'
P_sat=pressure(Fluid$, T=T_sat, x=1)
rho_l=density(Fluid$, T=T_sat, x=0)
rho_v=density(Fluid$, T=T_sat, x=1)
sigma=SurfaceTension(Fluid$, T=T_sat)*Convert(lbf/ft, lbm/s^2)
mu_l=Viscosity(Fluid$,T=T_sat, x=0)
Pr_l=Prandtl(Fluid$, T=T_sat, P=P_sat+1) "P=P_sat+1 is used to get liquid state"
C_l=CP(Fluid$, T=T_sat, x=0)
h_f=enthalpy(Fluid$, T=T_sat, x=0)
h_g=enthalpy(Fluid$, T=T_sat, x=1)
h_fg=h_g-h_f
C_sf=0.0060 "from Table 10-3 of the text"
n=1 "from Table 10-3 of the text"
g=32.2 [ft/s^2]

"ANALYSIS"
"(a)"
q_dot_nucleate=mu_l*h_fg*(((g*(rho_l-rho_v))/sigma)^0.5)*((C_l*(T_s-T_sat))/(C_sf*h_fg*Pr_l^n))^3
q_dot_nucleate=h*(T_s-T_sat)
"(b)"
W_dot_e=q_dot_nucleate*A*Convert(Btu/h, kW)
A=pi*D*L
"(c)"
m_dot_evap=Q_dot_boiling/h_fg
Q_dot_boiling=W_dot_e*Convert(kW, Btu/h)
```

T_s [F]	h [Btu/h.ft^2.F]	$\dot{W}_e$ [kW]	$\dot{m}_{evap}$ [lbm/h]
260	12919	9.912	35.77
262	18603	17.13	61.82
264	25321	27.2	98.17
266	33073	40.6	146.5
268	41857	57.81	208.6
270	51676	79.3	286.2
272	62528	105.5	380.9
274	74413	137	494.5
276	87332	174.2	628.8
278	101285	217.6	785.3
280	116271	267.6	965.9
282	132290	324.8	1172
284	149343	389.6	1406
286	167430	462.5	1669
288	186550	543.9	1963
290	206704	634.4	2290
292	227891	734.4	2650
294	250111	844.4	3047
296	273366	964.8	3482
298	297653	1096	3956
300	322974	1239	4472





10-24 Hot mechanically polished stainless steel ball bearings are cooled by submerging them in water at 1 atm. The rate of heat removed from a ball bearing at the instant it is submerged in the water is to be determined.

Assumptions **1** Steady operating conditions exist at the instant of submersion. **2** Surface temperature is uniform. **3** The boiling regime is nucleate boiling since $\Delta T_{\text{excess}} = T_s - T_{\text{sat}} = 25°\text{C}$, which is in the nucleate boiling range of 5 to 30°C for water.

Properties At 1 atm, the saturation temperature of water is $T_{\text{sat}} = 100°\text{C}$. The properties of water at $T_{\text{sat}} = 100°\text{C}$ are $\sigma = 0.0589$ N/m (Tables 10-1) and, from Table A-9,

$$\rho_l = 957.9\ \text{kg/m}^3 \qquad h_{fg} = 2257\times 10^3\ \text{J/kg}$$
$$\rho_v = 0.5978\ \text{kg/m}^3 \qquad \mu_l = 0.282\times 10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\text{Pr}_l = 1.75 \qquad c_{pl} = 4217\ \text{J/kg}\cdot\text{K}$$

Also, $C_{sf} = 0.0130$ and $n = 1$ for the boiling of water on a mechanically polished stainless steel surface (Table 10-3).

Analysis The instant a ball bearing is submerged in the water, with $\Delta T_{\text{excess}} = 25°\text{C}$, nucleate boiling would occur. The heat flux can be determined from Rohsenow relation to be

$$\begin{aligned}\dot{q}_{\text{nucleate}} &= \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\,\text{Pr}_l^n}\right)^3\\ &= (0.282\times 10^{-3})(2257\times 10^3)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2}\left[\frac{4217(125-100)}{0.0130(2257\times 10^3)1.75}\right]^3\\ &= 2.1998\times 10^6\ \text{W/m}^2\end{aligned}$$

The heat transfer surface area is

$$A_s = \pi D^2 = \pi(0.05\ \text{m})^2 = 0.007854\ \text{m}^2$$

The rate of heat removed from a ball bearing at the instant it is submerged in the water is

$$\dot{Q}_{\text{boiling}} = A_s\dot{q}_{\text{nucleate}} = (0.007854\ \text{m}^2)(2.1998\times 10^6\ \text{W/m}^2) = \mathbf{17.3\ kW}$$

Discussion Note that a 5-cm-diameter stainless steel ball can release 17.3 kW of heat at the instant it is submerged in water. This high value of heat rate removal, even with a temperature difference of only 25°C, is a result from nucleate boiling.

10-25 A polished copper tube is used to generate 1.5 kg/s of steam at 270 kPa. The surface temperature of the tube, with the interest to minimize the excess temperature, is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The boiling regime is nucleate boiling since ΔT_{excess} is to be minimized (this assumption will be verified).

Properties At 270 kPa, the saturation temperature of water is T_{sat} = 130°C. The properties of water at T_{sat} = 130°C are σ = 0.05295 N/m (Tables 10-1) and, from Table A-9,

$$\rho_l = 934.6\ \text{kg/m}^3 \qquad h_{fg} = 2174\times 10^3\ \text{J/kg}$$
$$\rho_v = 1.496\ \text{kg/m}^3 \qquad \mu_l = 0.213\times 10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\text{Pr}_l = 1.33 \qquad c_{pl} = 4263\ \text{J/kg}\cdot\text{K}$$

Also, C_{sf} = 0.013 and n = 1 for the boiling of water on a mechanically polished copper surface (Table 10-3).

Analysis The heat flux can be determined from the rate of vaporization to be

$$\dot{q}_{\text{nucleate}} = \frac{\dot{m}_{\text{vapor}} h_{fg}}{A_s} = \frac{\dot{m}_{\text{vapor}} h_{fg}}{\pi DL} = \frac{(1.5\ \text{kg/s})(2174\times 10^3\ \text{J/kg})}{\pi(0.05\ \text{m})(15\ \text{m})} = 1.384\times 10^6\ \text{W/m}^2$$

In the interest of minimizing the excess temperature, the boiling regime would be nucleate boiling. The heat flux can be expressed using the Rohsenow relation,

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg}\ \text{Pr}_l^n}\right)^3$$

$$1.384\times 10^6\ \text{W/m}^2 = (0.213\times 10^{-3})(2174\times 10^3)\left[\frac{9.81(934.6-1.496)}{0.05295}\right]^{1/2}\left[\frac{4263(T_s - 130)}{0.0130(2174\times 10^3)1.33}\right]^3$$

Solving for T_s yields

$$T_s = \mathbf{147°C}$$

Discussion With T_s = 147°C, the excess temperature would be since $\Delta T_{\text{excess}} = T_s - T_{\text{sat}}$ = 17°C, which is in the nucleate boiling range of 5 to 30°C for water. Thus the assumption of nucleate boiling regime is verified.

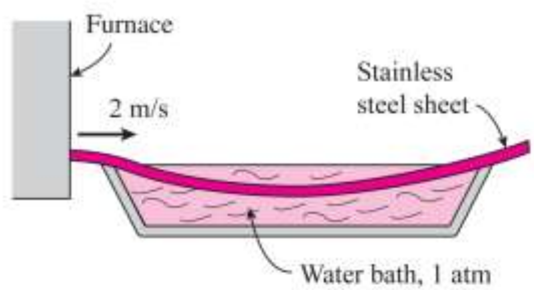
10-26 A long hot mechanically polished stainless steel sheet is being cooled in a water bath. The temperature of the stainless steel sheet leaving the water bath is to be determined whether or not it has the risk of thermal burn hazard.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform. **3** The boiling regime is nucleate boiling since $\Delta T_{\text{excess}} = T_s - T_{\text{sat}} = 25°\text{C}$, which is in the nucleate boiling range of 5 to 30°C for water.

Properties The specific heat and the density of stainless steel are given as $c_{p,ss}$ = 450 J/kg·K and ρ_{ss} = 7900 kg/m^3, respectively.

At 1 atm, the saturation temperature of water is T_{sat} = 100°C. The properties of water at T_{sat} = 100°C are σ = 0.0589 N/m (Tables 10-1) and, from Table A-9,

$$\rho_l = 957.9\ \text{kg/m}^3 \qquad h_{fg} = 2257\times10^3\ \text{J/kg}$$
$$\rho_v = 0.5978\ \text{kg/m}^3 \qquad \mu_l = 0.282\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\text{Pr}_l = 1.75 \qquad c_{pl} = 4217\ \text{J/kg}\cdot\text{K}$$



Also, C_{sf} = 0.013 and n = 1 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3).

Analysis The mass of the stainless steel sheet being conveyed enters and exits the water bath at a rate of

$$\dot{m} = \rho_{ss} V w t$$

The rate of heat that needs to be removed from the sheet so that it leaves the water bath below 45°C is

$$\dot{Q}_{\text{removed}} = \dot{m} c_{p,ss}(T_{\text{in}} - T_{\text{out}})$$

Then,

$$\dot{Q}_{\text{removed}} = \rho_{ss} V w t c_{p,ss}(T_{\text{in}} - T_{\text{out}})$$
$$= (7900\ \text{kg/m}^3)(2\ \text{m/s})(0.5\ \text{m})(0.005\ \text{m})(450\ \text{J/kg}\cdot\text{K})(125-45)\ \text{K}$$
$$= 1.422\times10^6\ \text{W}$$

With ΔT_{excess} = 25°C, nucleate boiling would occur in the water bath. The heat flux can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg}\,\text{Pr}_l^n}\right)^3$$
$$= (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2}\left[\frac{4217(125-100)}{0.0130(2257\times10^3)1.75}\right]^3$$
$$= 2.1998\times10^6\ \text{W/m}^2$$

The heat transfer surface area of the sheet submerged in the water bath is

$$A_s = 2(1\ \text{m})(0.5\ \text{m}) + 2(1\ \text{m})(0.005\ \text{m}) = 1.01\ \text{m}^2$$

The rate of heat that could be removed from the sheet in the water bath is

$$\dot{Q}_{\text{boiling}} = A_s \dot{q}_{\text{nucleate}} = (1.01\ \text{m}^2)(2.1998\times10^6\ \text{W/m}^2) = \mathbf{2.222\times10^6\ W} > \dot{Q}_{\text{removed}} = 1.422\times10^6\ \text{W}$$

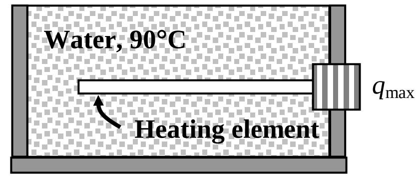
Discussion The rate of heat that could be removed from the stainless steel sheet in the water bath via nucleate boiling is greater than the heat that needs to be removed from the sheet so that it leaves the water bath below 45°C. Thus, there is no risk of thermal burn on the stainless steel sheet as it leaves the water bath.

Note that this analysis is simplified to steady state conditions, but the actual cooling process is transient.

10-27 Water is boiled at the saturation (or boiling) temperature of T_{sat} = 90°C by a horizontal brass heating element. The maximum heat flux in the nucleate boiling regime is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 90°C are (Tables 10-1 and A-9)



$$\rho_l = 965.3\ \text{kg/m}^3 \qquad h_{fg} = 2283\times10^3\ \text{J/kg}$$
$$\rho_v = 0.4235\ \text{kg/m}^3 \qquad \mu_l = 0.315\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\sigma = 0.0608\ \text{N/m} \qquad c_{pl} = 4206\ \text{J/kg}\cdot°\text{C}$$
$$\text{Pr}_l = 1.96$$

Also, $C_{sf} = 0.0060$ and n = 1.0 for the boiling of water on a brass heating (Table 10-3). Note that we expressed the properties in units specified under Eqs. 10-2 and 10-3 in connection with their definitions in order to avoid unit manipulations. For a large horizontal heating element, C_{cr} = 0.12 (Table 10-4). (It can be shown that L^* = 1.58 > 1.2 and thus the restriction in Table 10-4 is satisfied).

Analysis The maximum or critical heat flux is determined from

$$\begin{aligned}\dot{q}_{max} &= C_{cr} h_{fg}[\sigma g \rho_v^2(\rho_l - \rho_v)]^{1/4}\\ &= 0.12(2283\times10^3)[0.0608\times9.81\times(0.4235)^2(965.3-0.4235)]^{1/4}\\ &= 873{,}200\ \text{W/m}^2 = \mathbf{873.2\ kW/m^2}\end{aligned}$$

10-28 The electrical current at which a nickel wire would be in danger of burnout in nucleate boiling is to be determined.

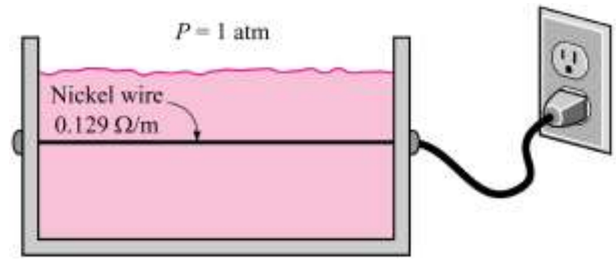
Assumptions **1** Steady operating condition exists. **2** Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are σ = 0.0589 N/m (Table 10-1) and, from Table A-9,

ρ_l = 957.9 kg/m^3

h_{fg} = 2257 × 10^3 J/kg

ρ_v = 0.5978 kg/m^3



Analysis The danger of burnout occurs when the heat flux is at maximum in nucleate boiling, which can be determined using

$$\dot{q}_{\max} = C_{cr} h_{fg}[\sigma g \rho_v^2 (\rho_l - \rho_v)]^{1/4}$$

Using Table 10-4, the parameter L^* and the constant C_{cr} are determined to be

$$L^* = L\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} = (0.0005)\left[\frac{9.81(957.9 - 0.5978)}{0.0589}\right]^{1/2} = 0.1997$$

which correspond to

$$C_{cr} = 0.12 L^{*-0.25} = 0.12(0.1997)^{-0.25} = 0.1795$$

Hence, the maximum heat flux is

$$\begin{aligned}\dot{q}_{\max} &= (0.1795)(2257 \times 10^3)[(0.0589)(9.81)(0.5978)^2(957.9 - 0.5978)]^{1/4} \\ &= 1.519 \times 10^6 \text{ W/m}^2\end{aligned}$$

We know that

$$\dot{q} = \frac{I^2 R}{A_s} = \frac{I^2 R}{\pi D L}$$

Thus, the electrical current at which the wire would be in danger of burnout is

$$I = \left[\frac{\dot{q}_{\max} \pi D}{(R/L)}\right]^{1/2} = \left[\frac{(1.519 \times 10^6 \text{ W/m}^2)\pi(0.001\,\text{m})}{0.129\,\Omega/\text{m}}\right]^{1/2} = \mathbf{192\ A}$$

Discussion The electrical current at which burnout could occur will decrease if the resistance of the wire increases.

10-29 Water is boiled at a temperature of T_{sat} = 100°C by hot gases flowing through a tube submerged in water. The maximum rate of vaporization is to be determined.

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

Properties The properties of water at T_{sat} = 100°C are σ = 0.0589 N/m (Tables 10-1) and, from Table A-9, ρ_l = 957.9 kg/m^3, ρ_v = 0.5978 kg/m^3, h_{fg} = 2257 × 10^3 J/kg.

Analysis The maximum rate of vaporization occurs at the maximum heat flux.

For a horizontal cylindrical heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = \frac{D}{2}\left(\frac{g(\rho_l - \rho_v)}{\sigma}\right)^{1/2} = (0.050/2)\left[\frac{9.81(957.9 - 0.5978)}{0.0589}\right]^{1/2} = 9.98 > 1.2$$

$$C_{cr} = 0.12 \quad (\text{since } L^* > 1.2 \text{ and thus large cylinder})$$

Then the maximum heat flux is determined from

$$\begin{aligned}\dot{q}_{\max} &= C_{cr} h_{fg} [\sigma g \rho_v^2 (\rho_l - \rho_v)]^{1/4} \\ &= 0.12(2257 \times 10^3)[0.0589 \times 9.81 \times (0.5978)^2 (957.9 - 0.5978)]^{1/4} = 1.0155 \times 10^6 \text{ W/m}^2\end{aligned}$$

The heat transfer surface area is

$$A_s = \pi DL = \pi(0.05 \text{ m})(10 \text{ m}) = 1.5708 \text{ m}^2$$

Then, the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A_s \dot{q}_{\max} = (1.5708 \text{ m}^2)(1.0155 \times 10^6 \text{ W/m}^2) = 1.5952 \times 10^6 \text{ W}$$

The maximum rate of vaporization of water is determined from

$$\dot{m}_{\text{vapor}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{1.5952 \times 10^6 \text{ J/s}}{2257 \times 10^3 \text{ J/kg}} = \mathbf{0.707 \text{ kg/s}}$$

Discussion The rate of vaporization can be increased by increasing the tube diameter, thereby increasing the heat transfer surface area.

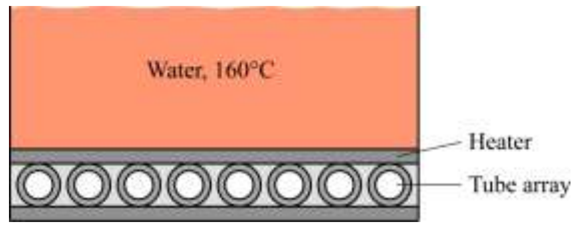
10-30 Water is boiled at a temperature of T_{sat} = 160°C by a 3 m × 3 m horizontal flat heater heated by hot gases flowing through an array of tubes embedded in it. The maximum rate of vaporization is to be determined.

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

Properties The properties of water at T_{sat} = 160°C are σ = 0.0466 N/m (Tables 10-1) and, from Table A-9, ρ_l = 907.4 kg/m^3, ρ_v = 3.256 kg/m^3, h_{fg} = 2083 × 10^3 J/kg.

Analysis The maximum rate of vaporization occurs at the maximum heat flux.

For a horizontal flat heating element, the coefficient C_{cr} is determined from Table 10-4 to be



$$L^* = L\left(\frac{g(\rho_l - \rho_v)}{\sigma}\right)^{1/2} = (3)\left[\frac{9.81(907.4 - 3.256)}{0.0466}\right]^{1/2} = 1309 > 27$$

$$C_{cr} = 0.149 \quad \text{(since } L^* > 27 \text{ and thus large flat heater)}$$

Then the maximum heat flux is determined from

$$\begin{aligned}\dot{q}_{\max} &= C_{cr} h_{fg} [\sigma g \rho_v^2 (\rho_l - \rho_v)]^{1/4} \\ &= 0.149(2083\times10^3)[0.0466\times9.81\times(3.256)^2(907.4-3.256)]^{1/4} \\ &= 2.5252\times10^6 \text{ W/m}^2\end{aligned}$$

The heat transfer surface area is

$$A_s = L\times L = 3\text{ m}\times3\text{ m} = 9\text{ m}^2$$

Then, the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A_s \dot{q}_{\max} = (9\text{ m}^2)(2.5252\times10^6 \text{ W/m}^2) = 2.2727\times10^7 \text{ W}$$

The maximum rate of vaporization of water is determined from

$$\dot{m}_{\text{vapor}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{2.2727\times10^7 \text{ J/s}}{2083\times10^3 \text{ J/kg}} = \mathbf{10.9\ kg/s}$$

Discussion The rate of vaporization can be increased by increasing the surface area of the plate.

10-31 Water is to be boiled at 1 atm by a spherical heater and a square horizontal flat heater of the same surface area, and which heater geometry would produce higher maximum rate of vaporization is to be determined.

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

Properties At 1 atm, the saturation temperature of water is T_{sat} = 100°C. The properties of water at T_{sat} = 100°C are σ = 0.0589 N/m (Tables 10-1) and, from Table A-9, ρ_l = 957.9 kg/m^3, ρ_v = 0.5978 kg/m^3, h_{fg} = 2257 × 10^3 J/kg.

Analysis The maximum rate of vaporization occurs at the maximum heat flux.

For a spherical heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = \frac{D}{2}\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (1/2)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2} = 199.7 > 4.26$$

$$C_{cr} = 0.11 \quad (\text{since } L^* > 4.26 \text{ and thus large sphere})$$

Then the maximum heat flux is determined from

$$\dot{q}_{\max} = C_{cr} h_{fg}[\sigma g \rho_v^2(\rho_l-\rho_v)]^{1/4}$$
$$= 0.11(2257\times10^3)[0.0589\times9.81\times(0.5978)^2(957.9-0.5978)]^{1/4} = 9.3092\times10^5 \text{ W/m}^2$$

The heat transfer surface area for a sphere is

$$A_s = \pi D^2 = \pi(1\,\text{m})^2 = 3.142\,\text{m}^2$$

The maximum rate of vaporization of water is determined from

$$\dot{m}_{\text{vapor}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{A_s\dot{q}_{\max}}{h_{fg}} = \frac{(3.142\,\text{m}^2)(9.3092\times10^5 \text{ W/m}^2)}{2257\times10^3 \text{ J/kg}} = \mathbf{1.296\ kg/s} \quad \text{(spherical heater)}$$

The width of a square with the same surface area as the spherical heater is

$$A_s = L^2 = 3.142\,\text{m}^2 \quad \rightarrow \quad L = 1.7726\,\text{m}$$

For a horizontal flat heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (1.7726)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2} = 707.8 > 27$$

$$C_{cr} = 0.149 \quad (\text{since } L^* > 27 \text{ and thus large flat heater})$$

Then the maximum heat flux is determined from

$$\dot{q}_{\max} = C_{cr} h_{fg}[\sigma g \rho_v^2(\rho_l-\rho_v)]^{1/4}$$
$$= 0.149(2257\times10^3)[0.0589\times9.81\times(0.5978)^2(957.9-0.5978)]^{1/4} = 1.261\times10^6 \text{ W/m}^2$$

The maximum rate of vaporization of water is determined from

$$\dot{m}_{\text{vapor}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{A_s\dot{q}_{\max}}{h_{fg}} = \frac{(3.142\,\text{m}^2)(1.261\times10^6 \text{ W/m}^2)}{2257\times10^3 \text{ J/kg}} = \mathbf{1.755\ kg/s} \quad \text{(square heater)}$$

Discussion For the same surface area, the square heater would produce about 35% higher in the maximum rate of vaporization than the spherical heater. This is because, for the same surface area, the square heater has a C_{cr} coefficient that is about 35% higher than that of the spherical heater.

10-32 Water is boiled at a temperature of T_{sat} = 180°C by a 3 m × 3 m nickel plated flat heater that is heated by hot gases flowing through an array of tubes embedded in it. The surface temperature that produced the maximum rate of steam generation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the heater are negligible. **3** The boiling regime is nucleate boiling.

Properties The properties of water at T_{sat} = 180°C are σ = 0.0422 N/m (Tables 10-1) and, from Table A-9,

$$\rho_l = 887.3\ \text{kg/m}^3 \qquad h_{fg} = 2015\times10^3\ \text{J/kg}$$
$$\rho_v = 5.153\ \text{kg/m}^3 \qquad \mu_l = 0.150\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\text{Pr}_l = 0.983 \qquad c_{pl} = 4410\ \text{J/kg}\cdot\text{K}$$

Also, C_{sf} = 0.0060 and n = 1 for the boiling of water on a nickel surface (Table 10-3).

Analysis The maximum rate of steam generation occurs at the maximum heat flux.

For a horizontal flat heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (3)\left[\frac{9.81(887.3-5.153)}{0.0422}\right]^{1/2} = 1359 > 27$$
$$C_{cr} = 0.149 \quad (\text{since } L^* > 27 \text{ and thus large flat heater})$$

Then the maximum heat flux is determined from

$$\begin{aligned}\dot{q}_{\max} &= C_{cr} h_{fg}[\sigma g \rho_v^2(\rho_l-\rho_v)]^{1/4}\\ &= 0.149(2015\times10^3)[0.0422\times9.81\times(5.153)^2(887.3-5.153)]^{1/4} = 2.9794\times10^6\ \text{W/m}^2\end{aligned}$$

The heat flux for nucleate boiling can be expressed using the Rohsenow relation to be

$$\dot{q}_{\max} = \dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$

$$2.9794\times10^6\ \text{W/m}^2 = (0.150\times10^{-3})(2015\times10^3)\left[\frac{9.81(887.3-5.153)}{0.0422}\right]^{1/2}\left[\frac{4410(T_s-180)}{0.0060(2015\times10^3)0.983}\right]^3$$

Solving for T_s yields

$$T_s = \mathbf{187.5°C}$$

The convection heat transfer coefficient can be determined as

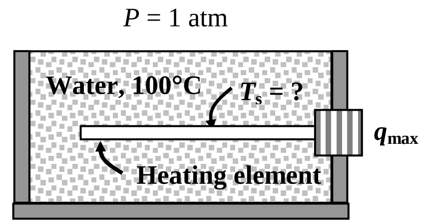
$$\dot{q} = h(T_s - T_{\text{sat}}) \quad \rightarrow \quad h = \frac{2.9794\times10^6\ \text{W/m}^2}{(187.5-180)\ \text{K}} = \mathbf{3.97\times10^5\ W/m^2\cdot K}$$

Discussion Note that a heat transfer coefficient of about 400 kW/m^2·K can be achieved in nucleate boiling with a temperature difference of less than 10°C.

10-33 Water is boiled at 1 atm pressure and thus at a saturation (or boiling) temperature of T_{sat} = 100°C by a mechanically polished stainless steel heating element. The maximum heat flux in the nucleate boiling regime and the surface temperature of the heater for that case are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)



$$\rho_l = 957.9\ \text{kg/m}^3 \qquad h_{fg} = 2257\times10^3\ \text{J/kg}$$
$$\rho_v = 0.60\ \text{kg/m}^3 \qquad \mu_l = 0.282\times10^{-3}\ \text{kg}\cdot\text{m/s}$$
$$\sigma = 0.0589\ \text{N/m} \qquad c_{pl} = 4217\ \text{J/kg}\cdot°\text{C}$$
$$\text{Pr}_l = 1.75$$

Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eqs. 10-2 and 10-3 in connection with their definitions in order to avoid unit manipulations. For a large horizontal heating element, C_{cr} = 0.12 (Table 10-4). (It can be shown that L^* = 3.99 > 1.2 and thus the restriction in Table 10-4 is satisfied).

Analysis The maximum or critical heat flux is determined from

$$\begin{aligned}\dot{q}_{\max} &= C_{cr} h_{fg}[\sigma g \rho_v^2(\rho_l - \rho_v)]^{1/4} \\ &= 0.12(2257\times10^3)[0.0589\times9.8\times(0.6)^2(957.9-0.60)]^{1/4} \\ &= \mathbf{1{,}017{,}000\ W/m^2}\end{aligned}$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given. Substituting the maximum heat flux into the Rohsenow relation together with other properties gives

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg}\,\text{Pr}_l^n}\right)^3$$

$$1{,}017{,}000 = (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.8(957.9-0.60)}{0.0589}\right]^{1/2}\left(\frac{4217(T_s - 100)}{0.0130(2257\times10^3)1.75}\right)^3$$

It gives

$$T_s = \mathbf{119.3°C}$$

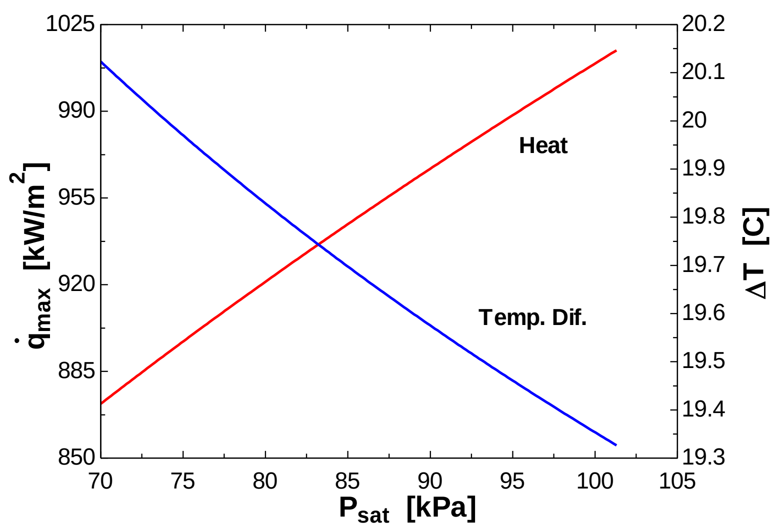
Therefore, the temperature of the heater surface will be only 19.3°C above the boiling temperature of water when burnout occurs.

10-34 Prob. 10-33 is reconsidered. The effect of local atmospheric pressure on the maximum heat flux and the temperature difference $T_s - T_{sat}$ is to be investigated.

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

```
"GIVEN"
D=0.03 [m]
P_sat=101.3 [kPa]

"PROPERTIES"
Fluid$='steam_IAPWS'
T_sat=temperature(Fluid$, P=P_sat, x=1)
rho_l=density(Fluid$, T=T_sat, x=0)
rho_v=density(Fluid$, T=T_sat, x=1)
sigma=SurfaceTension(Fluid$, T=T_sat)
mu_l=Viscosity(Fluid$,T=T_sat, x=0)
Pr_l=Prandtl(Fluid$, T=T_sat, P=P_sat+1[kPa])
c_l=CP(Fluid$, T=T_sat, x=0)
h_f=enthalpy(Fluid$, T=T_sat, x=0)
h_g=enthalpy(Fluid$, T=T_sat, x=1)
h_fg=h_g-h_f
C_sf=0.0130 "from Table 10-3 of the text"
n=1 "from Table 10-3 of the text"
C_cr=0.12 "from Table 10-4 of the text"
g=9.8 [m/s^2] "gravitational acceleraton"
"ANALYSIS"
q_dot_max=C_cr*h_fg*(sigma*g*rho_v^2*(rho_l-rho_v))^0.25
q_dot_nucleate=q_dot_max
q_dot_nucleate=mu_l*h_fg*(((g*(rho_l-rho_v))/sigma)^0.5)*((c_l*(T_s-T_sat))/(C_sf*h_fg*Pr_l^n))^3
DELTAT=T_s-T_sat
```

P_{sat} [kPa]	$\dot{q}_{max}$ [kW/m^2]	ΔT [C]
70	871.9	20.12
71.65	880.3	20.07
73.29	888.6	20.02
74.94	896.8	19.97
76.59	904.9	19.92
78.24	912.8	19.88
79.88	920.7	19.83
81.53	928.4	19.79
83.18	936.1	19.74
84.83	943.6	19.7
86.47	951.1	19.66
88.12	958.5	19.62
89.77	965.8	19.58
91.42	973	19.54
93.06	980.1	19.5
94.71	987.2	19.47
96.36	994.1	19.43
98.01	1001	19.4
99.65	1008	19.36
101.3	1015	19.33



10-35 A 10 cm × 10 cm flat heater is used for vaporizing refrigerant-134a at 350 kPa. The surface temperature of the heater is given as 25°C and the heater is subjected to a heat flux of 0.35 MW/m^2. The coefficient C_{sf} is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the heater are negligible. **3** The boiling regime is nucleate boiling since $\Delta T_{\text{excess}} = T_s - T_{\text{sat}} = 20°\text{C}$.

Properties At 350 kPa, the saturation temperature of R134a is 5°C (Table A-10). The properties of R134a at T_{sat} = 5°C are from Table A-10,

$$\rho_l = 1278\ \text{kg/m}^3 \qquad h_{fg} = 194.8\times10^3\ \text{J/kg}$$
$$\rho_v = 17.12\ \text{kg/m}^3 \qquad \mu_l = 2.589\times10^{-4}\ \text{kg/m}\cdot\text{s}$$
$$\text{Pr}_l = 3.802 \qquad c_{pl} = 1358\ \text{J/kg}\cdot\text{K}$$
$$\sigma = 0.01084\ \text{N/m}$$

Also, n = 1.7 for the boiling of R134a is given.

Analysis The heat flux for nucleate boiling can be expressed using the Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\,\text{Pr}_l^n}\right)^3$$

Using the Rohsenow relation to solve for C_{sf} yields

$$\begin{aligned}C_{sf} &= \frac{c_{p,l}(T_s-T_{\text{sat}})}{h_{fg}\,\text{Pr}_l^n}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/6}\left(\frac{\mu_l h_{fg}}{\dot{q}_{\text{nucleate}}}\right)^{1/3}\\ &= \frac{1358(25-5)}{(194.8\times10^3)(3.802^{1.7})}\left[\frac{9.81(1278-17.12)}{0.01084}\right]^{1/6}\left(\frac{(2.589\times10^{-4})(194.8\times10^3)}{0.35\times10^6}\right)^{1/3} = \mathbf{0.00772}\end{aligned}$$

For a horizontal flat heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (0.1)\left[\frac{9.81(1278-17.12)}{0.01084}\right]^{1/2} = 106.8 > 27$$
$$C_{cr} = 0.149 \quad (\text{since } L^* > 27 \text{ and thus large flat heater})$$

The maximum heat flux in the nucleate boiling regime can be determined from

$$\begin{aligned}\dot{q}_{\max} &= C_{cr}h_{fg}[\sigma g\rho_v^2(\rho_l-\rho_v)]^{1/4}\\ &= 0.149(194.8\times10^3)[0.01084\times9.81\times(17.12)^2(1278-17.12)]^{1/4}\\ &= 4.087\times10^5\ \text{W/m}^2 > 0.35\ \text{MW/m}^2\end{aligned}$$

Discussion Since $\dot{q}_{\max}$ > 0.35 MW/m^2, the Rohsenow relation for nucleate boiling is appropriate for this analysis.

10-36 Water is boiled at a temperature of T_{sat} = 150°C by hot gases flowing through a mechanically polished stainless steel pipe submerged in water whose outer surface temperature is maintained at T_s = 165°C. The rate of heat transfer to the water, the rate of evaporation, the ratio of critical heat flux to current heat flux, and the pipe surface temperature at critical heat flux conditions are to be determined.

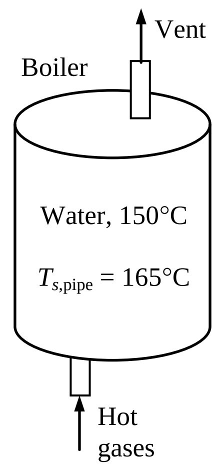
Assumptions **1** Steady operating conditions exist. **2** Heat losses from the boiler are negligible. **3** The boiling regime is nucleate boiling since $\Delta T = T_s - T_{sat} = 165 - 150 = 15°C$ which is in the nucleate boiling range of 5 to 30°C for water.

Properties The properties of water at the saturation temperature of 150°C are (Tables 10-1 and A-9)

$$\rho_l = 916.6 \text{ kg/m}^3 \qquad h_{fg} = 2114\times10^3 \text{ J/kg}$$
$$\rho_v = 2.55 \text{ kg/m}^3 \qquad \mu_l = 0.183\times10^{-3} \text{ kg}\cdot\text{m/s}$$
$$\sigma = 0.0488 \text{ N/m} \qquad c_{pl} = 4311 \text{ J/kg}\cdot°\text{C}$$
$$\text{Pr}_l = 1.16$$

Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis (*a*) Assuming nucleate boiling, the heat flux can be determined from Rohsenow relation to be



$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$
$$= (0.183\times10^{-3})(2114\times10^3)\left[\frac{9.81(916.6-2.55)}{0.0488}\right]^{1/2}\left(\frac{4311(165-150)}{0.0130(2114\times10^3)1.16}\right)^3$$
$$= 1{,}384{,}060 \text{ W/m}^2$$

The heat transfer surface area is

$$A_s = \pi DL = \pi(0.05\text{ m})(50\text{ m}) = 7.854\text{ m}^2$$

Then the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A_s\dot{q}_{\text{nucleate}} = (7.854\text{ m}^2)(1{,}384{,}060\text{ W/m}^2) = 10{,}870{,}400\text{ W} = \mathbf{10{,}870\,kW}$$

(*b*) The rate of evaporation of water is determined from

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{10{,}870\text{ kJ/s}}{2114\text{ kJ/kg}} = \mathbf{5.142\,kg/s}$$

(*c*) For a horizontal cylindrical heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (0.025)\left(\frac{9.8(916.6-2.55)}{0.0488}\right)^{1/2} = 10.7 > 1.2$$

$$C_{cr} = 0.12 \quad (\text{since } L^* > 1.2 \text{ and thus large cylinder})$$

Then the maximum or critical heat flux is determined from

$$\dot{q}_{\max} = C_{cr}h_{fg}[\sigma g\rho_v^2(\rho_l-\rho_v)]^{1/4}$$
$$= 0.12(2114\times10^3)[0.0488\times9.8\times(2.55)^2(916.6-2.55)]^{1/4} = 1{,}852{,}000\text{ W/m}^2$$

Therefore,

$$\frac{\dot{q}_{\max}}{\dot{q}_{\text{current}}} = \frac{1{,}852{,}000}{1{,}384{,}060} = \mathbf{1.338}$$

(*d*) The surface temperature of the pipe at the burnout point is determined from Rohsenow relation at the critical heat flux value to be

$$\dot{q}_{\text{nucleate,cr}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_{s,cr}-T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$

$$1{,}852{,}000 = (0.183\times10^{-3})(2114\times10^3)\left[\frac{9.8(916.6-2.55)}{0.0488}\right]^{1/2}\left(\frac{4311(T_{s,cr}-150)}{0.0130(2114\times10^3)1.16}\right)^3$$

$$T_{s,cr} = \mathbf{166.5°C}$$

10-37 Water is boiled at a temperature of T_{sat} = 160°C by hot gases flowing through a mechanically polished stainless steel pipe submerged in water whose outer surface temperature is maintained at T_s = 165°C. The rate of heat transfer to the water, the rate of evaporation, the ratio of critical heat flux to current heat flux, and the pipe surface temperature at critical heat flux conditions are to be determined.

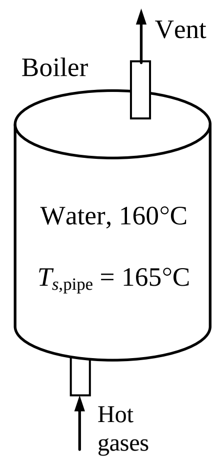
Assumptions **1** Steady operating conditions exist. **2** Heat losses from the boiler are negligible. **3** The boiling regime is nucleate boiling since $\Delta T = T_s - T_{\text{sat}} = 165 - 160 = 5°\text{C}$ which is in the nucleate boiling range of 5 to 30°C for water.

Properties The properties of water at the saturation temperature of 160°C are (Tables 10-1 and A-9)

$$\rho_l = 907.4 \text{ kg/m}^3 \qquad h_{fg} = 2083 \times 10^3 \text{ J/kg}$$
$$\rho_v = 3.256 \text{ kg/m}^3 \qquad \mu_l = 0.170 \times 10^{-3} \text{ kg} \cdot \text{m/s}$$
$$\sigma = 0.0466 \text{ N/m} \qquad c_{pl} = 4340 \text{ J/kg} \cdot °\text{C}$$
$$\text{Pr}_l = 1.09$$

Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis (*a*) Assuming nucleate boiling, the heat flux can be determined from Rohsenow relation to be



$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right)^3$$
$$= (0.170 \times 10^{-3})(2083 \times 10^3) \left[\frac{9.81(907.4 - 3.256)}{0.0466} \right]^{1/2} \left(\frac{4340(165 - 160)}{0.0130(2083 \times 10^3)1.09} \right)^3$$
$$= 61{,}390 \text{ W/m}^2$$

The heat transfer surface area is

$$A_s = \pi DL = \pi(0.05 \text{ m})(50 \text{ m}) = 7.854 \text{ m}^2$$

Then the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A_s \dot{q}_{\text{nucleate}} = (7.854 \text{ m}^2)(61{,}390 \text{ W/m}^2) = \mathbf{482{,}200 W}$$

(*b*) The rate of evaporation of water is determined from

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{482.2 \text{ kJ/s}}{2083 \text{ kJ/kg}} = \mathbf{0.2315 kg/s}$$

(*c*) For a horizontal cylindrical heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l - \rho_v)}{\sigma} \right)^{1/2} = (0.025)\left(\frac{9.81(907.4 - 3.256)}{0.0466} \right)^{1/2} = 10.9 > 0.12$$
$$C_{cr} = 0.12 \quad (\text{since } L^* > 1.2 \text{ and thus large cylinder})$$

Then the maximum or critical heat flux is determined from

$$\dot{q}_{\max} = C_{cr} h_{fg} [\sigma g \rho_v^2 (\rho_l - \rho_v)]^{1/4}$$
$$= 0.12(2083 \times 10^3)[0.0466 \times 9.81 \times (3.256)^2 (907.4 - 3.256)]^{1/4} = 2.034 \times 10^6 \text{ W/m}^2$$

Therefore,

$$\frac{\dot{q}_{\max}}{\dot{q}_{\text{current}}} = \frac{2.034 \times 10^6 \text{ W/m}^2}{61{,}390 \text{ W/m}^2} = \mathbf{33.13}$$

(*d*) The surface temperature of the pipe at the burnout point is determined from Rohsenow relation at the critical heat flux value to be

$$2.034 \times 10^6 = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{p,l}(T_{s,cr} - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right)^3$$
$$= (0.170 \times 10^{-3})(2083 \times 10^3) \left[\frac{9.81(907.4 - 3.256)}{0.0466} \right]^{1/2} \left(\frac{4340(T_{s,cr} - 160)}{0.0130(2083 \times 10^3)1.09} \right)^3$$
$$T_{s,cr} = \mathbf{176.1°C}$$

10-38 Steam is generated by a 1 m × 1 m flat heater boiling water at 1 atm with an excess temperature above 300°C. The range of the steam generation rate that can be achieved by the heater is to be determined.

Assumptions **1** Steady operating conditions exists. **2** Heat losses from the heater are negligible. **3** The boiling regime is film boiling since $\Delta T_{\text{excess}} > 300°\text{C}$, which is much larger than 30°C.

Properties At 1 atm, the saturation temperature of water is T_{sat} = 100°C. The properties of water at T_{sat} = 100°C are σ = 0.0589 N/m (Tables 10-1) and, from Table A-9, ρ_l = 957.9 kg/m^3, ρ_v = 0.5978 kg/m^3, h_{fg} = 2257 × 10^3 J/kg.

Analysis For a horizontal flat heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l - \rho_v)}{\sigma}\right)^{1/2} = (1)\left[\frac{9.81(957.9 - 0.5978)}{0.0589}\right]^{1/2} = 399.4 > 27$$

$$C_{cr} = 0.149 \quad \text{(since } L^* > 27 \text{ and thus large flat heater)}$$

The range of steam generation rate not exceeding the burnout point can be determined from the minimum and maximum boiling heat fluxes.

The minimum rate of vaporization occurs at the minimum heat flux, which can be determined from

$$\dot{q}_{\min} = 0.09\rho_v h_{fg}\left[\frac{\sigma g(\rho_l - \rho_v)}{(\rho_l + \rho_v)^2}\right]^{1/4}$$

$$= 0.09(0.5978)(2257\times 10^3)\left[\frac{(0.0589)(9.81)(957.9 - 0.5978)}{(957.9 + 0.5978)^2}\right]^{1/4} = 19021 \text{ W/m}^2$$

The maximum rate of vaporization occurs at the maximum heat flux, which can be determined from

$$\dot{q}_{\max} = C_{cr} h_{fg}[\sigma g \rho_v^2(\rho_l - \rho_v)]^{1/4}$$

$$= 0.149(2257\times 10^3)[0.0589\times 9.81\times (0.5978)^2(957.9 - 0.5978)]^{1/4} = 1.261\times 10^6 \text{ W/m}^2$$

The heat transfer surface area is

$$A_s = L\times L = 1\text{ m}\times 1\text{ m} = 1\text{ m}^2$$

Then, the rate of heat transfer during boiling is

$$\dot{Q}_{\text{boiling}} = A_s\dot{q}$$

Thus, the range of the steam generation rate in the film boiling regime is

$$\frac{\dot{Q}_{\text{boiling,min}}}{h_{fg}} \le \dot{m}_{\text{vapor}} \le \frac{\dot{Q}_{\text{boiling,max}}}{h_{fg}}$$

$$\frac{A_s\dot{q}_{\min}}{h_{fg}} \le \dot{m}_{\text{vapor}} \le \frac{A_s\dot{q}_{\max}}{h_{fg}}$$

$$\frac{(1\text{ m}^2)(19021\text{ W/m}^2)}{2257\times 10^3\text{ J/kg}} \le \dot{m}_{\text{vapor}} \le \frac{(1\text{ m}^2)(1.261\times 10^6\text{ W/m}^2)}{2257\times 10^3\text{ J/kg}}$$

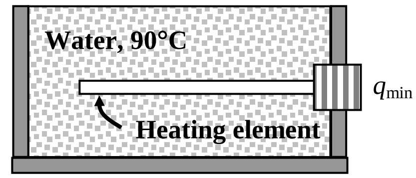
or $\mathbf{0.00843\ kg/s \le \dot{m}_{vapor} \le 0.559\ kg/s}$

Discussion The maximum rate of steam generation is more than 66 times larger than the minimum rate of steam generation.

10-39 Water is boiled at T_{sat} = 90°C in a brass heating element. The surface temperature of the heater is to be determined.

Assumptions 1 Steady operating conditions exist. **2** Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 90°C are (Tables 10-1 and A-9)



$$\rho_l = 965.3\ \text{kg/m}^3 \qquad h_{fg} = 2283\times10^3\ \text{J/kg}$$
$$\rho_v = 0.4235\ \text{kg/m}^3 \qquad \mu_l = 0.315\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\sigma = 0.0608\ \text{N/m} \qquad c_{pl} = 4206\ \text{J/kg}\cdot°\text{C}$$
$$\text{Pr}_l = 1.96$$

Also, $C_{sf} = 0.0060$ and n = 1.0 for the boiling of water on a brass heating (Table 10-3).

Analysis The minimum heat flux is determined from

$$\dot{q}_{\min} = 0.09\rho_v h_{fg}\left[\frac{\sigma g(\rho_l - \rho_v)}{(\rho_l + \rho_v)^2}\right]^{1/4}$$
$$= 0.09(0.4235)(2283\times10^3)\left[\frac{(0.0608)(9.81)(965.3 - 0.4235)}{(965.3 + 0.4235)^2}\right]^{1/4} = 13{,}715\ \text{W/m}^2$$

The surface temperature can be determined from Rohsenow equation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$
$$13{,}715\ \text{W/m}^2 = (0.315\times10^{-3})(2283\times10^3)\left[\frac{9.81(965.3\text{-}0.4235)}{0.0608}\right]^{1/2}\left(\frac{4206(T_s - 90)}{0.0060(2283\times10^3)1.96}\right)^3$$
$$T_s = \mathbf{92.3°C}$$

10-40 The power dissipation per unit length of a metal rod submerged horizontally in water, when electric current is passed through it, is to be determined.

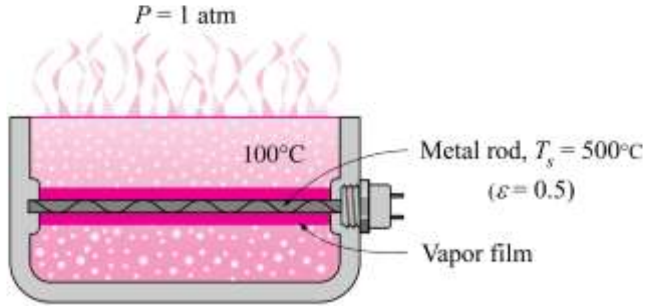
Assumptions **1** Steady operating condition exists. **2** Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_l = 957.9 kg/m^3 (Table A-9).

The properties of vapor at the film temperature of $T_f = (T_{sat} + T_s)/2$ = 300°C are, from Table A-16,

ρ_v = 0.3831 kg/m^3 $\quad$ c_{pv} = 1997 J/kg·K

$\mu_v = 2.045 \times 10^{-5}$ kg/m·s $\quad$ k_v = 0.04345 W/m·K



Analysis The excess temperature in this case is $\Delta T = T_s - T_{sat}$ = 400°C, which is much larger than 30°C for water from Fig. 10-6. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from

$$\dot{q}_{film} = C_{film}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{sat})]}{\mu_v D(T_s-T_{sat})}\right]^{1/4}(T_s-T_{sat})$$

$$= 0.62\left[\frac{9.81(0.04345)^3(0.3831)(957.9-0.3831)[2257\times10^3+0.4(1997)(400)]}{(2.045\times10^{-5})(0.002)(400)}\right]^{1/4}(400)$$

$$= 1.152\times10^5 \text{ W/m}^2$$

The radiation heat flux is determined from

$$\dot{q}_{rad} = \varepsilon\sigma(T_s^4 - T_{sat}^4) = (0.5)(5.67\times10^{-8} \text{ W/m}^2\cdot\text{K}^4)(773^4 - 373^4)\text{ K}^4 = 9573 \text{ W/m}^2$$

Then the total heat flux becomes

$$\dot{q}_{total} = \dot{q}_{film} + \frac{3}{4}\dot{q}_{rad} = 1.152\times10^5 \text{ W/m}^2 + \frac{3}{4}(9573 \text{ W/m}^2) = 1.224\times10^5 \text{ W/m}^2$$

Finally, the power dissipation per unit length of the metal rod is

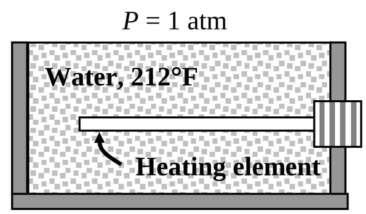
$$\dot{Q}_{total}/L = \pi D\dot{q}_{total} = \pi(0.002\text{ m})(1.224\times10^5 \text{ W/m}^2) = \mathbf{769\ W/m}$$

Discussion The contribution of radiation to the total heat flux is about 8%.

10-41E Water is boiled at 1 atm pressure and thus at a saturation (or boiling) temperature of T_{sat} = 212°F by a horizontal polished copper heating element whose surface temperature is maintained at T_s = 788°F. The rate of heat transfer to the water per unit length of the heater is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 212°F are $\rho_l = 59.82\ \text{lbm/ft}^3$ and $h_{fg} = 970\ \text{Btu/lbm}$ (Table A-9E). The properties of the vapor at the film temperature of $T_f = (T_{sat} + T_s)/2 = (212+788)/2 = 500°\text{F}$ are (Table A-16E)



$$\rho_v = 0.02571\ \text{lbm/ft}^3$$
$$\mu_v = 1.267\times10^{-5}\ \text{lbm/ft}\cdot\text{s} = 0.04561\ \text{lbm/ft}\cdot\text{h}$$
$$c_{pv} = 0.4707\ \text{Btu/lbm}\cdot°\text{F}$$
$$k_v = 0.02267\ \text{Btu/h}\cdot\text{ft}\cdot°\text{F}$$

Also, g = 32.2 ft/s^2 = 32.2×(3600)^2 ft/h^2. Note that we expressed the properties in units that will cancel each other in boiling heat transfer relations. Also note that we used vapor properties at 1 atm pressure from Table A-16E instead of the properties of saturated vapor from Table A-9E since the latter are at the saturation pressure of 680 psia (46 atm).

Analysis The excess temperature in this case is $\Delta T = T_s - T_{sat} = 788 - 212 = 576°\text{F}$, which is much larger than 30°C or 54°F. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined to be

$$\dot{q}_{\text{film}} = 0.62\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{sat})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})$$
$$= 0.62\left[\frac{32.2(3600)^2(0.02267)^3(0.02571)(59.82-0.02571)[970+0.4\times0.4707(788-212)]}{(0.04561)(0.5/12)(788-212)}\right]^{1/4}(788-212)$$
$$= 18{,}600\ \text{Btu/h}\cdot\text{ft}^2$$

The radiation heat flux is determined from

$$\dot{q}_{\text{rad}} = \varepsilon\sigma(T_s^4 - T_{\text{sat}}^4)$$
$$= (0.05)(0.1714\times10^{-8}\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}^4)\left[(788+460\ \text{R})^4-(212+460\ \text{R})^4\right]$$
$$= 190.4\ \text{Btu/h}\cdot\text{ft}^2$$

Note that heat transfer by radiation is very small in this case because of the low emissivity of the surface and the relatively low surface temperature of the heating element. Then the total heat flux becomes

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 18{,}600 + \frac{3}{4}\times190.4 = 18{,}743\ \text{Btu/h}\cdot\text{ft}^2$$

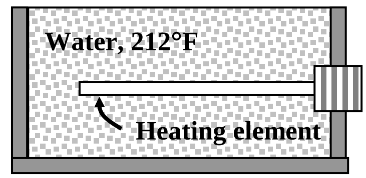
Finally, the rate of heat transfer from the heating element to the water is determined by multiplying the heat flux by the heat transfer surface area,

$$\dot{Q}_{\text{total}} = A_s\dot{q}_{\text{total}} = (\pi DL)\dot{q}_{\text{total}}$$
$$= (\pi\times0.5/12\ \text{ft}\times1\ \text{ft})(18{,}743\ \text{Btu/h}\cdot\text{ft}^2)$$
$$= \mathbf{2453\ Btu/h}$$

10-42E Water is boiled at 1 atm pressure and thus at a saturation (or boiling) temperature of T_{sat} = 212°F by a horizontal polished copper heating element whose surface temperature is maintained at T_s = 988°F. The rate of heat transfer to the water per unit length of the heater is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 212°F are $\rho_l = 59.82\ \text{lbm/ft}^3$ and $h_{fg} = 970\ \text{Btu/lbm}$ (Table A-9E). The properties of the vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (212+988)/2 = 600°\text{F}$ are, by interpolation, (Table A-16E)



$$\rho_v = 0.02395\ \text{lbm/ft}^3$$
$$\mu_v = 1.416\times10^{-5}\ \text{lbm/ft}\cdot\text{s} = 0.05099\ \text{lbm/ft}\cdot\text{h}$$
$$c_{pv} = 0.4799\ \text{Btu/lbm}\cdot°\text{F}$$
$$k_v = 0.02640\ \text{Btu/h}\cdot\text{ft}\cdot°\text{F}$$

Also, g = 32.2 ft/s^2 = 32.2×(3600)^2 ft/h^2. Note that we expressed the properties in units that will cancel each other in boiling heat transfer relations. Also note that we used vapor properties at 1 atm pressure from Table A-16E instead of the properties of saturated vapor from Table A-9E since the latter are at the saturation pressure of 1541 psia (105 atm).

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}} = 988 - 212 = 776°\text{F}$, which is much larger than 30°C or 54°F. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from

$$\dot{q}_{\text{film}} = 0.62\left[\frac{g k_v^3 \rho_v(\rho_l - \rho_v)[h_{fg} + 0.4C_{pv}(T_s - T_{sat})]}{\mu_v D(T_s - T_{\text{sat}})}\right]^{1/4}(T_s - T_{\text{sat}})$$
$$= 0.62\left[\frac{32.2(3600)^2(0.0264)^3(0.02395)(59.82-0.02395)[970+0.4\times0.4799(988-212)]}{(0.05099)(0.5/12)(988-212)}\right]^{1/4}(988-212)$$
$$= 25{,}147\ \text{Btu/h}\cdot\text{ft}^2$$

The radiation heat flux is determined from

$$\dot{q}_{\text{rad}} = \varepsilon\sigma(T_s^4 - T_{\text{sat}}^4)$$
$$= (0.05)(0.1714\times10^{-8}\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}^4)\left[(988+460\ \text{R})^4 - (212+460\ \text{R})^4\right]$$
$$= 359.3\ \text{Btu/h}\cdot\text{ft}^2$$

Note that heat transfer by radiation is very small in this case because of the low emissivity of the surface and the relatively low surface temperature of the heating element. Then the total heat flux becomes

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 25{,}147 + \frac{3}{4}\times359.3 = 25{,}416\ \text{Btu/h}\cdot\text{ft}^2$$

Finally, the rate of heat transfer from the heating element to the water is determined by multiplying the heat flux by the heat transfer surface area,

$$\dot{Q}_{\text{total}} = A_s\dot{q}_{\text{total}} = (\pi DL)\dot{q}_{\text{total}}$$
$$= (\pi\times0.5/12\ \text{ft}\times1\ \text{ft})(25{,}416\ \text{Btu/h}\cdot\text{ft}^2)$$
$$= \mathbf{3327\ Btu/h}$$

10-43 The initial heat transfer rate from a hot metal sphere that is suddenly submerged in a water bath is to be determined.

Assumptions **1** Steady operating condition exists. **2** The metal sphere has uniform initial surface temperature.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_l = 957.9 kg/m^3 (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 400°C are, from Table A-16,

ρ_v = 0.3262 kg/m^3 $\quad$ c_{pv} = 2066 J/kg·K

$\mu_v = 2.446 \times 10^{-5}$ kg/m·s $\quad$ k_v = 0.05467 W/m·K

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}}$ = 600°C, which is much larger than 30°C for water from Fig. 10-6. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from

$$
\begin{aligned}
\dot{q}_{\text{film}} &= C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}}) \\
&= 0.67\left[\frac{9.81(0.05467)^3(0.3262)(957.9-0.3262)[2257\times10^3+0.4(2066)(600)]}{(2.446\times10^{-5})(0.02)(600)}\right]^{1/4}(600) \\
&= 1.052\times10^5\ \text{W/m}^2
\end{aligned}
$$

The radiation heat flux is determined from

$$
\begin{aligned}
\dot{q}_{\text{rad}} &= \varepsilon\sigma(T_s^4-T_{\text{sat}}^4) \\
&= (0.75)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(973^4-373^4)\ \text{K}^4 \\
&= 3.729\times10^4\ \text{W/m}^2
\end{aligned}
$$

Then the total heat flux becomes

$$
\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 1.052\times10^5\ \text{W/m}^2 + \frac{3}{4}(3.729\times10^4\ \text{W/m}^2) = 1.332\times10^5\ \text{W/m}^2
$$

Finally, the initial heat transfer rate from the submerged metal sphere is

$$
\dot{Q}_{\text{total}} = \dot{q}_{\text{total}}\pi D^2 = (1.332\times10^5\ \text{W/m}^2)\pi(0.02\ \text{m})^2 = \mathbf{167\ W}
$$

Discussion The contribution of radiation to the total heat flux is about 21%, which is significant and cannot be neglected.

10-44 The initial heat transfer rate from a hot steel rod that is suddenly submerged in a water bath is to be determined.

Assumptions **1** Steady operating condition exists. **2** The steel rod has uniform initial surface temperature.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_l = 957.9 kg/m^3 (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{sat} + T_s)/2$ = 300°C are, from Table A-16,

ρ_v = 0.3831 kg/m^3 $\quad$ c_{pv} = 1997 J/kg·K

$\mu_v = 2.045 \times 10^{-5}$ kg/m·s $\quad$ k_v = 0.04345 W/m·K

Analysis The excess temperature in this case is $\Delta T = T_s - T_{sat}$ = 400°C, which is much larger than 30°C for water from Fig. 10-6. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from

$$\begin{aligned}
\dot{q}_{\text{film}} &= C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}}) \\
&= 0.62\left[\frac{9.81(0.04345)^3(0.3831)(957.9-0.3831)[2257\times10^3+0.4(1997)(400)]}{(2.045\times10^{-5})(0.02)(400)}\right]^{1/4}(400) \\
&= 6.476\times10^4 \text{ W/m}^2
\end{aligned}$$

The radiation heat flux is determined from

$$\begin{aligned}
\dot{q}_{\text{rad}} &= \varepsilon\sigma(T_s^4-T_{\text{sat}}^4) \\
&= (0.9)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)(773^4-373^4)\text{ K}^4 \\
&= 1.723\times10^4 \text{ W/m}^2
\end{aligned}$$

Then the total heat flux becomes

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 6.476\times10^4 \text{ W/m}^2 + \frac{3}{4}(1.723\times10^4 \text{ W/m}^2) = 7.768\times10^4 \text{ W/m}^2$$

Finally, the initial heat transfer rate from the submerged steel rod is

$$\dot{Q}_{\text{total}} = \dot{q}_{\text{total}}\pi DL = (7.768\times10^4 \text{ W/m}^2)\pi(0.02\text{ m})(0.2\text{ m}) = \mathbf{976\ W}$$

Discussion The contribution of radiation to the total heat flux is about 17%, which is significant and cannot be neglected.

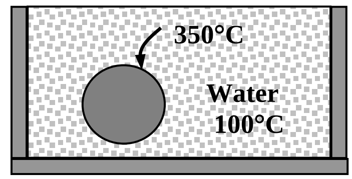
10-45 Water is boiled at T_{sat} = 100°C by a spherical platinum heating element immersed in water. The surface temperature is T_s = 350°C. The rate of heat transfer is to be determined.

Assumptions 1 Steady operating conditions exist. **2** Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are (Table A-9)

$$h_{fg} = 2257\times10^3 \text{ J/kg}$$
$$\rho_l = 957.9 \text{ kg/m}^3$$

The properties of water vapor at (350+100)/2 = 225°C are (Table A-16)



$$\rho_v = 0.444 \text{ kg/m}^3$$
$$\mu_v = 1.749\times10^{-5} \text{ kg/m}\cdot\text{s}$$
$$c_{pv} = 1951 \text{ J/kg}\cdot°\text{C}$$
$$k_v = 0.03581 \text{ W/m}\cdot°\text{C}$$

Analysis The film boiling occurs since the temperature difference between the surface and the fluid. The heat flux in this case can be determined from

$$\dot{q}_{\text{film}} = 0.67\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)\left[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})\right]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})$$
$$= 0.67\left[\frac{(9.81)(0.03581)^3(0.444)(957.9-0.444)\left[2257\times10^3+0.4(1951)(350-100)\right]}{(1.749\times10^{-5})(0.15)(350-100)}\right]^{1/4}(350-100)$$
$$= 25{,}207 \text{ W/m}^2$$

The radiation heat transfer is

$$\dot{q}_{\text{rad}} = \varepsilon\sigma(T_s^4-T_{\text{sat}}^4) = (0.10)(5.67\times10^{-8})\left[(350+273)^4-(100+273)^4\right] = 745 \text{ W/m}^2$$

The total heat flux is

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 25{,}207 + \frac{3}{4}(745) = 25{,}766 \text{ W/m}^2$$

Then the total rate of heat transfer becomes

$$\dot{Q}_{\text{total}} = A\dot{q}_{\text{total}} = \pi(0.15)^2(25{,}766 \text{ W/m}^2) = \mathbf{1821 W}$$

10-46 Cylindrical stainless steel rods are heated to 700°C and then suddenly quenched in water at 1 atm. The convection heat transfer coefficient and the total rate of heat removed from a rod at the instant it is submerged in the water are to be determined.

Assumptions **1** Steady operating conditions exist at the instant of submersion. **2** Surface temperature is uniform. **3** The boiling regime is film boiling since $\Delta T_{\text{excess}} = T_s - T_{\text{sat}}$ = 600°C, which is much larger than 30°C.

Properties At 1 atm, the saturation temperature of water is T_{sat} = 100°C. The properties of water at T_{sat} = 100°C are $h_{fg} = 2257 \times 10^3$ J/kg and ρ_l = 957.9 kg/m^3 (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (100 + 700)/2$ = 400°C are, from Table A-16,

$$\rho_v = 0.3262\ \text{kg/m}^3 \qquad c_{pv} = 2066\ \text{J/kg}\cdot\text{K}$$
$$\mu_v = 2.446\times10^{-5}\ \text{kg/m}\cdot\text{s} \qquad k_v = 0.05467\ \text{W/m}\cdot\text{K}$$

Analysis The film boiling heat flux can be determined from

$$\dot{q}_{\text{film}} = C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})$$

$$where\ C_{film} = 0.62\ (horizontal\ cylinders)$$

$$= 0.62\left[\frac{9.81(0.05467)^3(0.3262)(957.9-0.3262)[2257\times10^3+0.4(2066)(700-100)]}{(2.446\times10^{-5})(0.025)(700-100)}\right]^{1/4}(700-100)$$
$$= 92097\ \text{W/m}^2$$

Thus, the convection heat transfer coefficient is

$$\dot{q}_{\text{film}} = h(T_s - T_{\text{sat}}) \quad \rightarrow \quad h = \frac{92097\ \text{W/m}^2}{(700-100)\ \text{K}} = \mathbf{153.5\ W/m^2\cdot K}$$

The radiation heat flux is determined from

$$\begin{aligned}\dot{q}_{\text{rad}} &= \varepsilon\sigma(T_s^4 - T_{\text{sat}}^4)\\ &= (0.3)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(973^4 - 373^4)\ \text{K}^4\\ &= 14917\ \text{W/m}^2\end{aligned}$$

Then the total heat flux becomes

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 92097\ \text{W/m}^2 + \frac{3}{4}(14917\ \text{W/m}^2) = 1.0328\times10^5\ \text{W/m}^2$$

The total rate of heat removed from a rod at the instant it is submerged in the water is

$$\dot{Q}_{\text{total}} = (\pi DL)\dot{q}_{\text{total}} = \pi(0.025\ \text{m})(0.25\ \text{m})(1.0328\times10^5\ \text{W/m}^2) = \mathbf{2028\ W}$$

Discussion Convection heat transfer coefficient in film boiling is generally lower than that of nucleate boiling, because the excess temperature of film boiling is much larger than that of nucleate boiling.

10-47 A long cylindrical stainless steel rod with mechanically polished surface is being quenched in a water bath. The temperature of the rod leaving the water bath is to be determined whether or not it has the risk of thermal burn hazard.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform. **3** The boiling regime is film boiling since $\Delta T_{\text{excess}} = T_s - T_{\text{sat}}$ = 700°C − 100°C = 600°C, which is much larger than 30°C.

Properties The specific heat and the density of stainless steel are given as $c_{p,ss}$ = 450 J/kg·K and ρ_{ss} = 7900 kg/m^3, respectively.

At 1 atm, the saturation temperature of water is T_{sat} = 100°C. The properties of water at T_{sat} = 100°C are h_{fg} = 2257 × 10^3 J/kg and ρ_l = 957.9 kg/m^3 (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = (100 + 700) = 400°C are, from Table A-16,

$$\rho_v = 0.3262\ \text{kg/m}^3 \qquad c_{pv} = 2066\ \text{J/kg}\cdot\text{K}$$
$$\mu_v = 2.446\times10^{-5}\ \text{kg/m}\cdot\text{s} \qquad k_v = 0.05467\ \text{W/m}\cdot\text{K}$$

Analysis With ΔT_{excess} = 600°C, film boiling would occur in the water bath. The heat flux can be determined from

$$\dot{q}_{\text{film}} = C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})$$

$$where\ C_{film} = 0.62\ (horizontal\ cylinders)$$

$$= 0.62\left[\frac{9.81(0.05467)^3(0.3262)(957.9-0.3262)[2257\times10^3+0.4(2066)(700-100)]}{(2.446\times10^{-5})(0.025)(700-100)}\right]^{1/4}(700-100)$$
$$= 92097\ \text{W/m}^2$$

The radiation heat flux is determined from

$$\dot{q}_{\text{rad}} = \varepsilon\sigma(T_s^4 - T_{\text{sat}}^4) = (0.3)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(973^4-373^4)\ \text{K}^4 = 14917\ \text{W/m}^2$$

Then the total heat flux becomes

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 92097\ \text{W/m}^2 + \frac{3}{4}(14917\ \text{W/m}^2) = 1.0328\times10^5\ \text{W/m}^2$$

The rate of heat that could be removed from the rod in the water bath is

$$\dot{Q}_{\text{total}} = A_s\dot{q}_{\text{total}} = (\pi DL)\dot{q}_{\text{total}} = \pi(0.025\ \text{m})(3\ \text{m})(1.0328\times10^5\ \text{W/m}^2) = 24335\ \text{W}$$

The mass of the stainless steel rod being conveyed enters and exits the water bath at a rate of

$$\dot{m} = \rho_{ss}V(\pi D^2/4)$$

The rate of heat that needs to be removed from the rod so that it leaves the water bath below 45°C can be determined using

$$\dot{Q}_{\text{total}} = \dot{m}c_{p,ss}(T_{\text{in}} - T_{\text{out}}) = \rho_{ss}V(\pi D^2/4)c_{p,ss}(T_{\text{in}} - T_{\text{out}})$$

Thus, the speed of the rod conveying through the water bath is

$$V = \frac{\dot{Q}_{\text{total}}}{\rho_{ss}(\pi D^2/4)c_{p,ss}(T_{\text{in}} - T_{\text{out}})}$$
$$= \frac{24335\ \text{W}}{(7900\ \text{kg/m}^3)[\pi(0.025\ \text{m})^2/4](450\ \text{J/kg}\cdot\text{K})(700-45)\text{K}} = 0.0213\ \text{m/s} = \mathbf{76.7\ m/hr}$$

Discussion To ensure that the stainless steel rod leaves the water bath below 45°C, in order to prevent thermal burn hazard, the speed of the rod conveying through the water bath should be about 77 m/hr or slower.

Note that this analysis is simplified to steady state conditions, but the actual quenching process is transient.

10-48 A cylindrical heater is used for boiling water at 1 atm. The film boiling convection heat transfer coefficient at the burnout point is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the heater are negligible. **3** The boiling regime is film boiling.

Properties At 1 atm, the saturation temperature of water is T_{sat} = 100°C. The properties of water at T_{sat} = 100°C are σ = 0.0589 N/m (Tables 10-1) and, from Table A-9, ρ_l = 957.9 kg/m^3, $\rho_{v,sat}$ = 0.5978 kg/m^3, h_{fg} = 2257 × 10^3 J/kg.

The properties of vapor at the film temperature of T_f = 1150°C are, from Table A-16,

$$\rho_v = 0.1543 \text{ kg/m}^3 \qquad c_{pv} = 2571 \text{ J/kg}\cdot\text{K}$$
$$\mu_v = 5.283\times10^{-5} \text{ kg/m}\cdot\text{s} \qquad k_v = 0.1588 \text{ W/m}\cdot\text{K}$$

Analysis For a cylindrical heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = \frac{D}{2}\left(\frac{g(\rho_l-\rho_{v,\text{sat}})}{\sigma}\right)^{1/2} = (0.01/2)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2} = 1.997 > 1.2$$
$$C_{cr} = 0.12 \quad (\text{since } L^* > 1.2 \text{ and thus large cylinder})$$

The burnout point occurs at the maximum heat flux, which is

$$\dot{q}_{\max} = C_{cr} h_{fg}[\sigma g \rho_{v,\text{sat}}^2(\rho_l-\rho_{v,\text{sat}})]^{1/4}$$
$$= 0.12(2257\times10^3)[0.0589\times9.81\times(0.5978)^2(957.9-0.5978)]^{1/4} = 1.0155\times10^6 \text{ W/m}^2$$

To determine the film boiling convection heat transfer coefficient, the knowledge of T_s is needed, which can be determined from the heat transfer in the film boiling region:

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{sat})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}}) + \frac{3}{4}\varepsilon\sigma(T_s^4-T_{\text{sat}}^4)$$
$$where\, C_{film} = 0.62\,(horizontal\, cylinders)$$

Substituting the values,

$$1.0155\times10^6 = 0.62\left[\frac{9.81(0.1588)^3(0.1543)(957.9-0.1543)[2257\times10^3+0.4(2571)(\text{T}_s-100)]}{(5.283\times10^{-5})(0.01)(\text{T}_s-100)}\right]^{1/4}(\text{T}_s-100)$$
$$+\frac{3}{4}(0.3)(5.67\times10^{-8})[(\text{T}_s+273)^4-(373)^4]$$

Solving for the surface temperature yield T_s = 2231°C

The film boiling heat flux is

$$\dot{q}_{\text{film}} = C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{sat})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})$$
$$= 0.62\left[\frac{9.81(0.1588)^3(0.1543)(957.9-0.1543)[2257\times10^3+0.4(2571)(2231-100)]}{(5.283\times10^{-5})(0.01)(2231-100)}\right]^{1/4}(2231-100)$$
$$= 5.1419\times10^5 \text{ W/m}^2$$

Thus, the film boiling convection heat transfer coefficient is

$$h = \frac{\dot{q}_{\text{film}}}{T_s-T_{\text{sat}}} = \frac{5.1419\times10^5 \text{ W/m}^2}{(2231-100)\text{ K}} = \mathbf{241.3\ W/m^2\cdot K}$$

Discussion Note that the film temperature T_f = (2231 + 100)/2 = 1166°C, is close to the assumed value of 1150°C for the evaluation of vapor properties. Therefore, 1150°C is a reasonable film temperature for the vapor properties.

10-49 A cylindrical heater is used for boiling water at 100°C. The boiling convection heat transfer coefficients at the maximum heat flux for nucleate boiling and film boiling are to be determined.

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

Properties The properties of water at T_{sat} = 100°C are σ = 0.0589 N/m (Tables 10-1) and, from Table A-9,

$$\rho_l = 957.9\ \text{kg/m}^3 \qquad h_{fg} = 2257\times10^3\ \text{J/kg}$$
$$\rho_{v,\text{sat}} = 0.5978\ \text{kg/m}^3 \qquad \mu_l = 0.282\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\text{Pr}_l = 1.75 \qquad c_{pl} = 4217\ \text{J/kg}\cdot\text{K}$$

Also, C_{sf} = 0.0130 and n = 1 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). The properties of vapor at the film temperature of T_f = 1150°C are, from Table A-16,

$$\rho_v = 0.1543\ \text{kg/m}^3 \qquad c_{pv} = 2571\ \text{J/kg}\cdot\text{K}$$
$$\mu_v = 5.283\times10^{-5}\ \text{kg/m}\cdot\text{s} \qquad k_v = 0.1588\ \text{W/m}\cdot\text{K}$$

Analysis For a cylindrical heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = \frac{D}{2}\left(\frac{g(\rho_l-\rho_{v,\text{sat}})}{\sigma}\right)^{1/2} = (0.003/2)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2} = 0.599 < 1.2$$

$$C_{cr} = 0.12L^{*-0.25} = 0.1364 \quad (\text{since } L^* < 1.2 \text{ and thus small cylinder})$$

The maximum heat flux can be determined as

$$\dot{q}_{\max} = C_{cr}h_{fg}[\sigma g\rho_{v,\text{sat}}^2(\rho_l-\rho_{v,\text{sat}})]^{1/4}$$
$$= 0.1364(2257\times10^3)[0.0589\times9.81\times(0.5978)^2(957.9-0.5978)]^{1/4} = 1.1543\times10^6\ \text{W/m}^2$$

(a) The surface temperature T_s for nucleate boiling at $\dot{q}_{\max}$ can be solved as

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_{v,\text{sat}})}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$

Substituting the values,

$$1.1543\times10^6 = (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2}\left[\frac{4217(T_s-100)}{0.0130(2257\times10^3)1.75}\right]^3$$

$$\therefore \quad T_s = 120.2°\text{C}$$

Thus, the nucleate boiling convection heat transfer coefficient is

$$h_{\text{nucleate}} = \frac{\dot{q}_{\text{nucleate}}}{T_s-T_{\text{sat}}} = \frac{1.1543\times10^6\ \text{W/m}^2}{(120.2-100)\ \text{K}} = \mathbf{57{,}144\ W/m^2\cdot K}$$

(b) The surface temperature T_s for film boiling at $\dot{q}_{\max}$ can be solved as

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{sat})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}}) + \frac{3}{4}\varepsilon\sigma(T_s^4-T_{\text{sat}}^4)$$

Substituting the values,

$$1.1543\times10^6 = 0.62\left[\frac{9.81(0.1588)^3(0.1543)(957.9-0.1543)[2257\times10^3+0.4(2571)(T_s-100)]}{(5.283\times10^{-5})(0.003)(T_s-100)}\right]^{1/4}$$
$$\times(T_s-100) + \frac{3}{4}(0.3)(5.67\times10^{-8})[(T_s+273)^4-(373)^4]$$

$\therefore \quad T_s = 2192°C$

The film boiling heat flux is

$$\dot{q}_{film} = C_{film}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{sat})]}{\mu_v D(T_s-T_{sat})}\right]^{1/4}(T_s-T_{sat})$$

where $C_{film} = 0.62\,(\text{horizontal cylinders})$

$$= 0.62\left[\frac{9.81(0.1588)^3(0.1543)(957.9-0.1543)[2257\times10^3+0.4(2571)(2192-100)]}{(5.283\times10^{-5})(0.003)(2192-100)}\right]^{1/4}(2192-100)$$

$$= 6.8366\times10^5\ \text{W/m}^2$$

Thus, the film boiling convection heat transfer coefficient is

$$h = \frac{\dot{q}_{film}}{T_s - T_{sat}} = \frac{6.8366\times10^5\ \text{W/m}^2}{(2192-100)\ \text{K}} = \mathbf{327\ W/m^2\cdot K}$$

Discussion The nucleate boiling convection heat transfer coefficient is about 175 times higher than that of film boiling. This is because the vapor film surrounding the heater surface during film boiling impedes convection heat transfer.

Note that the film temperature T_f = (2192 + 100)/2 = 1146°C, is close to the assumed value of 1150°C used in film boiling for the evaluation of vapor properties.

10-50 Water is boiled at 1 atm pressure and thus at a saturation (or boiling) temperature of T_{sat} = 100°C by a horizontal nickel plated copper heating element. The maximum (critical) heat flux and the temperature jump of the wire when the operating point jumps from nucleate boiling to film boiling regime are to be determined.

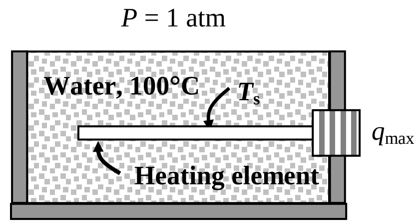
Assumptions 1 Steady operating conditions exist. 2 Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)

$$\rho_l = 957.9\ \text{kg/m}^3 \qquad h_{fg} = 2257\times10^3\ \text{J/kg}$$
$$\rho_v = 0.5978\ \text{kg/m}^3 \qquad \mu_l = 0.282\times10^{-3}\ \text{kg}\cdot\text{m/s}$$
$$\sigma = 0.0589\ \text{N/m} \qquad c_{pl} = 4217\ \text{J/kg}\cdot°\text{C}$$
$$\text{Pr}_l = 1.75$$

Also, $C_{sf} = 0.0060$ and n = 1.0 for the boiling of water on a nickel plated surface (Table 10-3). Note that we expressed the properties in units specified under Eqs. 10-2 and 10-3 in connection with their definitions in order to avoid unit manipulations. The vapor properties at the anticipated film temperature of $T_f = (T_s+T_{sat})/2$ of 1000°C (will be checked) (Table A-16)

$$\rho_v = 0.1725\ \text{kg/m}^3 \qquad c_{pv} = 2471\ \text{J/kg}\cdot°\text{C}$$
$$k_v = 0.1362\ \text{W/m}\cdot°\text{C} \qquad \mu_v = 4.762\times10^{-5}\ \text{kg/m}\cdot\text{s}$$



Analysis (*a*) For a horizontal heating element, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (0.0015)\left(\frac{9.81(957.9-0.5978)}{0.0589}\right)^{1/2} = 0.5990 < 1.2$$

$$C_{cr} = 0.12L^{*-0.25} = 0.12(0.5990)^{-0.25} = 0.1364$$

Then the maximum or critical heat flux is determined from

$$\begin{aligned}\dot{q}_{\max} &= C_{cr}h_{fg}[\sigma g\rho_v^2(\rho_l-\rho_v)]^{1/4}\\ &= 0.1364(2257\times10^3)[0.0589\times9.81\times(0.5978)^2(957.9-0.5978)]^{1/4}\\ &= \mathbf{1{,}154{,}000\ W/m^2}\end{aligned}$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given. Substituting the maximum heat flux into the Rohsenow relation together with other properties gives

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$

$$1{,}154{,}000 = (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.81(957.9-0.5978)}{0.0589}\right]^{1/2}\left(\frac{4217(T_s-100)}{0.0060(2257\times10^3)1.75}\right)^3$$

It gives $T_s = 109.3°\text{C}$

(*b*) Heat transfer in the film boiling region can be expressed as

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 0.62\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{sat})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}}) + \frac{3}{4}\varepsilon\sigma(T_s^4-T_{\text{sat}}^4)$$

Substituting,

$$1{,}154{,}000 = 0.62\left[\frac{9.81(0.1362)^3(0.1725)(957.9-0.1725)[2257\times10^3+0.4\times2471(T_s-100)]}{(4.762\times10^{-5})(0.003)(T_s-100)}\right]^{1/4}$$
$$\times(T_s-100) + \frac{3}{4}(0.5)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)\left[(T_s+273)^4-(100+273)^4\right]$$

Solving for the surface temperature gives T_s = 1999°C. Therefore, the temperature jump of the wire when the operating point jumps from nucleate boiling to film boiling is

Temperature jump: $\Delta T = T_{s,\text{film}} - T_{s,\text{crit}} = 1999 - 109 = \mathbf{1890°C}$

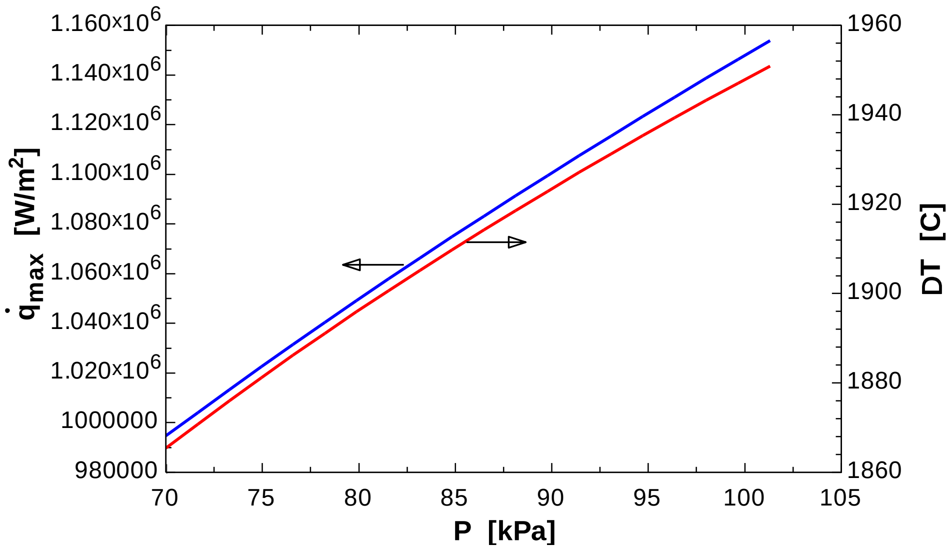
Note that the film temperature is (1999+100)/2=1050°C, which is close enough to the assumed value of 1000°C for the evaluation of vapor paroperties.

10-51 Prob. 10-50 is reconsidered. The effects of the local atmospheric pressure and the emissivity of the wire on the critical heat flux and the temperature rise of wire are to be investigated.

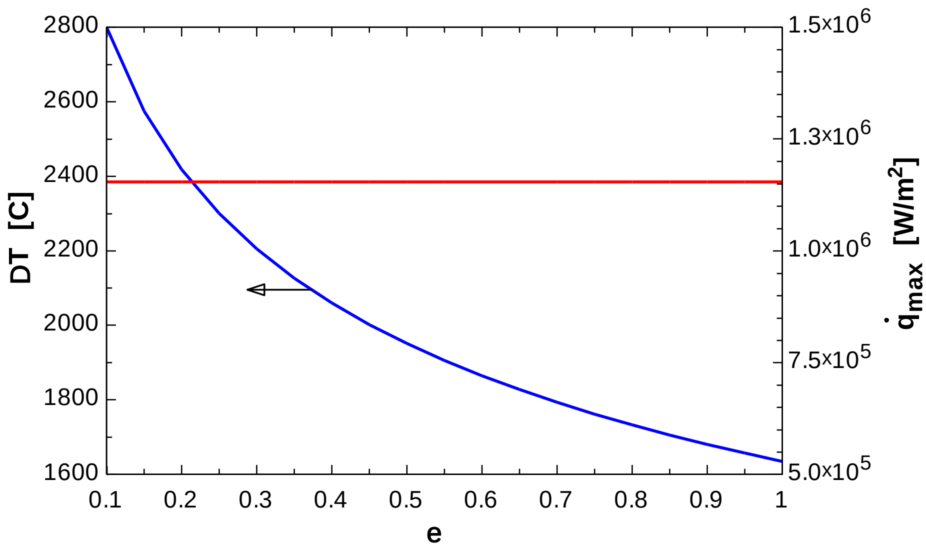
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=0.3 [m]
D=0.003 [m]
epsilon=0.5
P=101.3 [kPa]

"PROPERTIES"
Fluid$='steam_IAPWS'
T_sat=temperature(Fluid$, P=P, x=1)
rho_l=density(Fluid$, T=T_sat, x=0)
rho_v=density(Fluid$, T=T_sat, x=1)
sigma=SurfaceTension(Fluid$, T=T_sat)
mu_l=Viscosity(Fluid$,T=T_sat, x=0)
Pr_l=Prandtl(Fluid$, T=T_sat, P=P+1)
c_l=CP(Fluid$, T=T_sat, x=0)*Convert(kJ/kg-C, J/kg-C)
h_f=enthalpy(Fluid$, T=T_sat, x=0)
h_g=enthalpy(Fluid$, T=T_sat, x=1)
h_fg=(h_g-h_f)*Convert(kJ/kg, J/kg)
C_sf=0.0060 "from Table 10-3 of the text"
n=1 "from Table 10-3 of the text"

T_vapor=1000-273 "[C], assumed vapor temperature in the film boiling region"
rho_v_f=density(Fluid$, T=T_vapor, P=P) "f stands for film"
c_v_f=CP(Fluid$, T=T_vapor, P=P)*Convert(kJ/kg-C, J/kg-C)
k_v_f=Conductivity(Fluid$, T=T_vapor, P=P)
mu_v_f=Viscosity(Fluid$,T=T_vapor, P=P)

g=9.81 [m/s^2] “gravitational acceleraton"
sigma_rad=5.67E-8 [W/m^2-K^4] “Stefan-Boltzmann constant"

"ANALYSIS"
"(a)"
"C_cr is to be determined from Table 10-4 of the text"
C_cr=0.12*L_star^(-0.25)
L_star=D/2*((g*(rho_l-rho_v))/sigma)^0.5
q_dot_max=C_cr*h_fg*(sigma*g*rho_v^2*(rho_l-rho_v))^0.25
q_dot_nucleate=q_dot_max
q_dot_nucleate=mu_l*h_fg*(((g*(rho_l-rho_v))/sigma)^0.5)*((c_l*(T_s_crit-T_sat))/(C_sf*h_fg*Pr_l^n))^3
"(b)"
q_dot_total=q_dot_film+3/4*q_dot_rad "Heat transfer in the film boiling region"
q_dot_total=q_dot_nucleate
q_dot_film=0.62*((g*k_v_f^3*rho_v_f*(rho_l-rho_v_f)*(h_fg+0.4*c_v_f*(T_s_film-
T_sat)))/(mu_v_f*D*(T_s_film-T_sat)))^0.25*(T_s_film-T_sat)
q_dot_rad=epsilon*sigma_rad*((T_s_film+273)^4-(T_sat+273)^4)
DELTAT=T_s_film-T_s_crit
```

P [kPa]	$\dot{q}_{max}$ [kW/m^2]	ΔT [C]
70	994682	1865
71.65	1004096	1871
73.29	1013373	1876
74.94	1022516	1881
76.59	1031531	1886
78.24	1040422	1891
79.88	1049194	1896
81.53	1057848	1901
83.18	1066391	1905
84.83	1074825	1910
86.47	1083153	1914
88.12	1091379	1919
89.77	1099505	1923
91.42	1107535	1927
93.06	1115471	1931
94.71	1123316	1935
96.36	1131072	1939
98.01	1138742	1943
99.65	1146328	1947
101.3	1153832	1951



ε	$\dot{q}_{max}$ [kW/m^2]	ΔT [C]
0.1	1153832	2800
0.15	1153832	2574
0.2	1153832	2418
0.25	1153832	2300
0.3	1153832	2205
0.35	1153832	2127
0.4	1153832	2060
0.45	1153832	2002
0.5	1153832	1951
0.55	1153832	1905
0.6	1153832	1864
0.65	1153832	1827
0.7	1153832	1793
0.75	1153832	1762
0.8	1153832	1733
0.85	1153832	1706
0.9	1153832	1681
0.95	1153832	1657
1	1153832	1635



Condensation Heat Transfer

10-52C Condensation is a vapor-to-liquid phase change process. It occurs when the temperature of a vapor is reduced *below* its saturation temperature T_{sat}. This is usually done by bringing the vapor into contact with a solid surface whose temperature T_s is *below* the saturation temperature T_{sat} of the vapor.

10-53C In *film condensation*, the condensate wets the surface and forms a liquid film on the surface which slides down under the influence of gravity. The thickness of the liquid film increases in the flow direction as more vapor condenses on the film. This is how condensation normally occurs in practice. In *dropwise condensation*, the condensed vapor forms droplets on the surface instead of a continuous film, and the surface is covered by countless droplets of varying diameters. Dropwise condensation is a much more effective mechanism of heat transfer.

10-54C The presence of noncondensable gases in the vapor has a detrimental effect on condensation heat transfer. Even small amounts of a noncondensable gas in the vapor cause significant drops in heat transfer coefficient during condensation.

10-55C The modified latent heat of vaporization h^*_{fg} is the amount of heat released as a unit mass of vapor condenses at a specified temperature, plus the amount of heat released as the condensate is cooled further to some average temperature between T_{sat} and T_s. It is defined as $h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$ where c_{pl} is the specific heat of the liquid at the average film temperature.

10-56C During film condensation on a vertical plate, heat flux at the top will be higher since the thickness of the film at the top, and thus its thermal resistance, is lower.

10-57C The condensation heat transfer coefficient for the tubes will be the highest for the case of horizontal side by side (case b) since (1) for long tubes, the horizontal position gives the highest heat transfer coefficients, and (2) for tubes in a vertical tier, the average thickness of the liquid film at the lower tubes is much larger as a result of condensate falling on top of them from the tubes directly above, and thus the average heat transfer coefficient at the lower tubes in such arrangements is smaller.

10-58C In condensate flow, the wetted perimeter is defined as the length of the surface-condensate interface at a cross-section of condensate flow. It differs from the ordinary perimeter in that the latter refers to the entire circumference of the condensate at some cross-section.

10-59 The hydraulic diameter D_h for all 4 cases are expressed in terms of the boundary layer thickness δ as follows:

(*a*) Vertical plate: $$D_h = \frac{4A_c}{p} = \frac{4w\delta}{w} = 4\delta$$

(*b*) Tilted plate: $$D_h = \frac{4A_c}{p} = \frac{4w\delta}{w} = 4\delta$$

(*c*)Vertical cylinder: $$D_h = \frac{4A_c}{p} = \frac{4\pi D\delta}{\pi D} = 4\delta$$

(*d*) Horizontal cylinder: $$D_h = \frac{4A_c}{p} = \frac{4(2L\delta)}{2L} = 4\delta$$

(*e*) Sphere: $$D_h = \frac{4A_c}{p} = \frac{4\pi D\delta}{\pi D} = 4\delta$$

Therefore, the Reynolds number for all 5 cases can be expressed as

$$\mathrm{Re} = \frac{4\dot{m}}{p\mu_l} = \frac{4A_c\rho_l V_l}{p\mu_l} = \frac{D_h\rho_l V_l}{\mu_l} = \frac{4\delta\rho_l V_l}{\mu_l}$$

10-60 The local heat transfer coefficients at the middle and at the bottom of a vertical plate undergoing film condensation are to be determined.

Assumptions **1** Steady operating condition exists. **2** The plate surface has uniform temperature. **3** The flow is laminar.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_v = 0.5978 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 90°C are, from Table A-9,

ρ_l = 965.3 kg/m^3 $\qquad$ c_{pl} = 4206 J/kg·K

$\mu_l = 0.315 \times 10^{-3}$ kg/m·s $\qquad$ k_l = 0.675 W/m·K

$\nu_l = \mu_l / \rho_l = 0.326 \times 10^{-6}$ m^2/s

Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68 c_{pl}(T_{\text{sat}} - T_s) \\ &= 2257 \times 10^3 + 0.68(4206)(100 - 80) \\ &= 2314 \times 10^3 \text{ J/kg} \end{aligned}$$

The local heat transfer coefficient can be calculated using

$$\begin{aligned} h_x &= \left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg} k_l^3}{4\mu_l(T_{\text{sat}} - T_s)x} \right]^{1/4} \\ &= \left[\frac{(9.81)(965.3)(965.3 - 0.5978)(2314 \times 10^3)(0.675)^3}{4(0.315 \times 10^{-3})(100 - 80)x} \right]^{1/4} \\ &= 4008\left(\frac{1}{x}\right)^{1/4} \text{ W/m}^2 \cdot \text{K} \end{aligned}$$

The local heat transfer coefficient at the middle of the plate (x = 0.1 m) is

$$h_x = 4008\left(\frac{1}{x}\right)^{1/4} \text{ W/m}^2 \cdot \text{K} = 4008\left(\frac{1}{0.1}\right)^{1/4} \text{ W/m}^2 \cdot \text{K} = \mathbf{7130 \ W/m^2 \cdot K}$$

The local heat transfer coefficient at the bottom of the plate (x = 0.2 m) is

$$h_x = 4008\left(\frac{1}{x}\right)^{1/4} \text{ W/m}^2 \cdot \text{K} = 4008\left(\frac{1}{0.2}\right)^{1/4} \text{ W/m}^2 \cdot \text{K} = \mathbf{5990 \ W/m^2 \cdot K}$$

Discussion The assumption that the flow is laminar is verified to be appropriate:

$$\text{Re} \cong \frac{4g\rho_l^2}{3\mu_l^2}\left(\frac{k_l}{h_{x=L}}\right)^3 = \frac{4(9.81)(965.3)^2}{3(0.315 \times 10^{-3})^2}\left(\frac{0.675}{5990}\right)^3 = 176 < 1800$$

10-61 The necessary surface temperature of the plate used to condensate saturated water vapor at a desired condensation rate is to be determined.

Assumptions **1** Steady operating condition exists. **2** The plate surface has uniform temperature. **3** The film temperature is 90°C.

Properties Based on the problem statement, we take film temperature to be $T_f = (T_{sat} + T_s)/2 = 90$°C and the surface temperature to be $T_s = 80$°C. The properties of liquid water at the film temperature of $T_f = 90$°C are, from Table A-9,

$\rho_l = 965.3$ kg/m^3 $\quad$ $c_{pl} = 4206$ J/kg·K

$\mu_l = 0.315 \times 10^{-3}$ kg/m·s $\quad$ $k_l = 0.675$ W/m·K

$\nu_l = \mu_l / \rho_l = 0.326 \times 10^{-6}$ m^2/s

The properties of water at the saturation temperature of $T_{sat} = 100$°C are $h_{fg} = 2257$ kJ/kg (Table A-2) and $\rho_v = 0.5978$ kg/m^3 (Table A-9).

Analysis The calculation of the modified latent heat of vaporization requires the knowledge of the T_s. Hence, we assume T_s = 80°C, and iterate the solution, if necessary, until good agreement with the calculated value of T_s is achieved:

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68 c_{pl}(T_{\text{sat}} - T_s) \\ &= 2257 \times 10^3 + 0.68(4206)(100 - 80) \\ &= 2314 \times 10^3 \text{ J/kg} \end{aligned}$$

The Reynolds number is

$$\text{Re} = \frac{4\dot{m}}{p\mu_l} = \frac{4(0.016 \text{ kg/s})}{(0.5 \text{ m})(0.315 \times 10^{-3} \text{ kg/m}\cdot\text{s})} = 406.3$$

which is between 30 and 1800, and thus the flow is wavy-laminar. The heat transfer coefficient is

$$\begin{aligned} h = h_{\text{vert, wavy}} &= \frac{\text{Re}\, k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3} \\ &= \frac{(406.3)(0.675 \text{ W/m}\cdot\text{K})}{1.08(406.3)^{1.22} - 5.2}\left[\frac{9.81 \text{ m/s}^2}{(0.326 \times 10^{-6} \text{ m}^2\text{/s})^2}\right]^{1/3} \\ &= 7558 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

Hence, the surface temperature can be calculated using

$$hA_s(T_{\text{sat}} - T_s) = \dot{m}h^*_{fg} \quad \rightarrow \quad T_s = T_{\text{sat}} - \frac{\dot{m}h^*_{fg}}{hA_s}$$

$$T_s = 100°\text{C} - \frac{(0.016 \text{ kg/s})(2314 \times 10^3 \text{ J/kg})}{(7558 \text{ W/m}^2\cdot\text{K})(0.5 \text{ m})^2} = \mathbf{80.4°C}$$

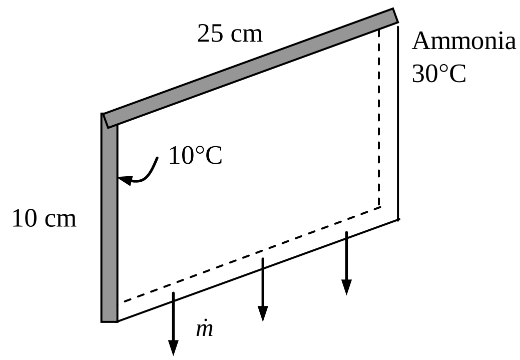
Discussion The assumed T_s = 80°C and T_f = 90°C are good, thus the solution does not require iteration.

10-62 Saturated ammonia at a saturation temperature of T_{sat} = 30°C condenses on vertical plates which are maintained at 10°C. The average heat transfer coefficient and the rate of condensation of ammonia are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The plate is isothermal. **3** The condensate flow is wavy-laminar over the entire plate (this assumption will be verified). **4** The density of vapor is much smaller than the density of liquid, $\rho_v \ll \rho_l$.

Properties The properties of ammonia at the saturation temperature of 30°C are $h_{fg} = 1144\times10^3$ J/kg and ρ_v = 9.055 kg/m^3. The properties of liquid ammonia at the film temperature of $T_f = (T_{sat} + T_s)/2 = (30 + 10)/2$ = 20°C are (Table A-11),

$$\rho_l = 610.2 \text{ kg/m}^3$$
$$\mu_l = 1.519\times10^{-4} \text{ kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l / \rho_l = 2.489\times10^{-7} \text{ m}^2\text{/s}$$
$$c_{pl} = 4745 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 0.4927 \text{ W/m}\cdot°\text{C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68 c_{pl}(T_{sat} - T_s)$$
$$= 1144\times10^3 \text{ J/kg} + 0.68\times 4745 \text{ J/kg}\cdot°\text{C}(30-10)°\text{C}$$
$$= 1209\times10^3 \text{ J/kg}$$

Assuming wavy-laminar flow, the Reynolds number is determined from

$$\text{Re} = \text{Re}_{\text{vertical,wavy}} = \left[4.81 + \frac{3.70 L k_l (T_{sat} - T_s)}{\mu_l h^*_{fg}}\left(\frac{g}{\nu_l^2}\right)^{1/3}\right]^{0.820}$$

$$= \left[4.81 + \frac{3.70\times(0.1\text{ m})\times(0.4927 \text{ W/m}\cdot°\text{C})\times(30-10)°\text{C}}{(1.519\times10^{-4} \text{ kg/m}\cdot\text{s})(1209\times10^3 \text{ J/kg})}\left(\frac{9.8 \text{ m/s}^2}{(2.489\times10^{-7} \text{ m}^2\text{/s})^2}\right)^{1/3}\right]^{0.82} = 307.0$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined to be

$$h = h_{\text{vertical,wavy}} = \frac{\text{Re}\, k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3}$$

$$= \frac{307\times(0.4927 \text{ W/m}\cdot°\text{C})}{1.08(307)^{1.22} - 5.2}\left(\frac{9.8 \text{ m/s}^2}{(2.489\times10^{-7} \text{ m}^2\text{/s})^2}\right)^{1/3} = \mathbf{7032 \text{ W/m}^2\cdot°\text{C}}$$

The total heat transfer surface area of the plates is

$$A_s = W\times L = 30(0.25\text{ m})(0.10\text{ m}) = 0.75\text{ m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (7032 \text{ W/m}^2\cdot°\text{C})(0.75\text{ m}^2)(30-10)°\text{C} = 105{,}480 \text{ W}$$

(*b*) The rate of condensation of steam is determined from

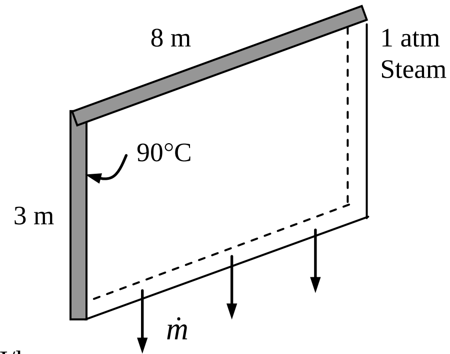
$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{105{,}480 \text{ J/s}}{1209\times10^3 \text{ J/kg}} = \mathbf{0.0872 \text{ kg/s}}$$

10-63 Saturated steam at atmospheric pressure thus at a saturation temperature of T_{sat} = 100°C condenses on a vertical plate which is maintained at 90°C by circulating cooling water through the other side. The rate of heat transfer to the plate and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The plate is isothermal. **3** The condensate flow is wavy-laminar over the entire plate (this assumption will be verified). **4** The density of vapor is much smaller than the density of liquid, $\rho_v \ll \rho_l$.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257×10³ J/kg and ρ_v = 0.60 kg/m³. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (100 + 90)/2 = 95°C$ are (Table A-9),

$$\rho_l = 961.5\ \text{kg/m}^3$$
$$\mu_l = 0.297\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 0.309\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4212\ \text{J/kg}\cdot°\text{C}$$
$$k_l = 0.677\ \text{W/m}\cdot°\text{C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2257\times10^3\ \text{J/kg} + 0.68\times4212\ \text{J/kg}\cdot°\text{C}(100-90)°\text{C} = 2{,}286\times10^3\ \text{J/kg}$$

Assuming wavy-laminar flow, the Reynolds number is determined from

$$\text{Re} = \text{Re}_{\text{vertical,wavy}} = \left[4.81 + \frac{3.70Lk_l(T_{sat}-T_s)}{\mu_l h^*_{fg}}\left(\frac{g}{\nu_l^2}\right)^{1/3}\right]^{0.820}$$

$$= \left[4.81 + \frac{3.70\times(3\ \text{m})\times(0.677\ \text{W/m}\cdot°\text{C})\times(100-90)°\text{C}}{(0.297\times10^{-3}\ \text{kg/m}\cdot\text{s})(2286\times10^3\ \text{J/kg})}\left(\frac{9.81\ \text{m/s}^2}{(0.309\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3}\right]^{0.820} = 1113$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined to be

$$h = h_{\text{vertical,wavy}} = \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3}$$

$$= \frac{1113\times(0.677\ \text{W/m}\cdot°\text{C})}{1.08(1113)^{1.22} - 5.2}\left(\frac{9.81\ \text{m/s}^2}{(0.309\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3} = 6279\ \text{W/m}^2\cdot°\text{C}$$

The heat transfer surface area of the plate is

$$A_s = W\times L = (3\ \text{m})(8\ \text{m}) = 24\ \text{m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (6279\ \text{W/m}^2\cdot°\text{C})(24\ \text{m}^2)(100-90)°\text{C} = 1{,}506{,}960\ \text{W} = \mathbf{1507\ kW}$$

(*b*) The rate of condensation of steam is determined from

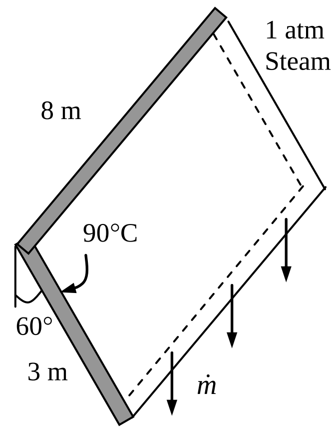
$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{1{,}506{,}960\ \text{J/s}}{2286\times10^3\ \text{J/kg}} = \mathbf{0.659\ kg/s}$$

10-64 Saturated steam at a saturation temperature of T_{sat} = 100°C condenses on a plate which is tilted 60° from the vertical and maintained at 90°C by circulating cooling water through the other side. The rate of heat transfer to the plate and the rate of condensation of the steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The plate is isothermal. **3** The condensate flow is wavy-laminar over the entire plate (this assumption will be verified). **4** The density of vapor is much smaller than the density of liquid, $\rho_v \ll \rho_l$.

Properties The properties of water at the saturation temperature of 100°C are $h_{fg} = 2257\times10^3$ J/kg and ρ_v = 0.60 kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (100 + 90)/2 = 95$°C are (Table A-9),

$$\begin{aligned}
\rho_l &= 961.5\ \text{kg/m}^3\\
\mu_l &= 0.297\times10^{-3}\ \text{kg/m}\cdot\text{s}\\
\nu_l &= \mu_l/\rho_l = 0.309\times10^{-6}\ \text{m}^2/\text{s}\\
c_{pl} &= 4212\ \text{J/kg}\cdot°\text{C}\\
k_l &= 0.677\ \text{W/m}\cdot°\text{C}
\end{aligned}$$



Analysis The modified latent heat of vaporization is

$$\begin{aligned}
h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{sat} - T_s)\\
&= 2257\times10^3\ \text{J/kg} + 0.68\times4212\ \text{J/kg}\cdot°\text{C}(100-90)°\text{C}\\
&= 2{,}286\times10^3\ \text{J/kg}
\end{aligned}$$

Assuming wavy-laminar flow, the Reynolds number is determined from the vertical plate relation by replacing g by $g\cos\theta$ where θ = 60° to be

$$\text{Re} = \text{Re}_{\text{tilted,wavy}} = \left[4.81 + \frac{3.70Lk_l(T_{sat}-T_s)}{\mu_l h^*_{fg}}\left(\frac{g\cos 60}{\nu_l^2}\right)^{1/3}\right]^{0.820}$$

$$= \left[4.81 + \frac{3.70\times(3\ \text{m})\times(0.677\ \text{W/m}\cdot°\text{C})\times(100-90)°\text{C}}{(0.297\times10^{-3}\ \text{kg/m}\cdot\text{s})(2286\times10^3\ \text{J/kg})}\left(\frac{(9.81\ \text{m/s}^2)\cos 60}{(0.309\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3}\right]^{0.82} = 920.8$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined from

$$h = h_{\text{tilted,wavy}} = \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g\cos\theta}{\nu_l^2}\right)^{1/3}$$

$$= \frac{920.8\times(0.677\ \text{W/m}\cdot°\text{C})}{1.08(920.8)^{1.22} - 5.2}\left(\frac{(9.81\ \text{m/s}^2)\cos 60}{(0.309\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3} = 5198\ \text{W/m}^2\cdot°\text{C}$$

The heat transfer surface area of the plate is

$$A_s = W\times L = (3\ \text{m})(8\ \text{m}) = 24\ \text{m}^2.$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (5198\ \text{W/m}^2\cdot°\text{C})(24\ \text{m}^2)(100-90)°\text{C} = 1{,}247{,}520\ \text{W} = \mathbf{1248\ kW}$$

(*b*) The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{1{,}247{,}520\ \text{J/s}}{2286\times10^3\ \text{J/kg}} = \mathbf{0.546\ kg/s}$$

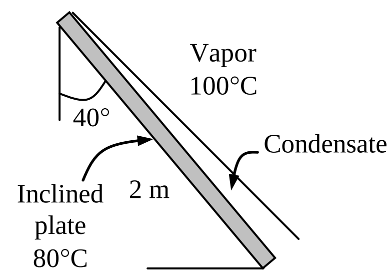
Discussion Using the heat transfer coefficient determined in the previous problem for the vertical plate, we could also determine the heat transfer coefficient from $h_{\text{inclined}} = h_{\text{vert}}(\cos\theta)^{1/4}$. It would give 5280 W/m^2·°C, which is 1.6% different than the value determined above.

10-65 Saturated steam at a saturation temperature of T_{sat} = 100°C condenses on a plate which is tilted 40° from the vertical and maintained at 80°C by circulating cooling water through the other side. The rate of heat transfer to the plate and the rate of condensation of the steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The plate is isothermal. **3** The condensate flow is wavy-laminar over the entire plate (this assumption will be verified). **4** The density of vapor is much smaller than the density of liquid, $\rho_v \ll \rho_l$.

Properties The properties of water at the saturation temperature of 100°C are $h_{fg} = 2257\times10^3$ J/kg and $\rho_v = 0.60$ kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (100 + 80)/2 = 90$°C are (Table A-9),

$$\rho_l = 965.3\ \text{kg/m}^3$$
$$\mu_l = 0.315\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 0.326\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4206\ \text{J/kg}\cdot°\text{C}$$
$$k_l = 0.675\ \text{W/m}\cdot°\text{C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2257\times10^3\ \text{J/kg} + 0.68\times4206\ \text{J/kg}\cdot°\text{C}(100-80)°\text{C} = 2{,}314\times10^3\ \text{J/kg}$$

Assuming wavy-laminar flow, the Reynolds number is determined from the vertical plate relation by replacing g by $g\cos\theta$ where θ= 40° to be

$$\text{Re} = \text{Re}_{\text{tilted,wavy}} = \left[4.81 + \frac{3.70Lk_l(T_{sat} - T_s)}{\mu_l h^*_{fg}}\left(\frac{g\cos\theta}{\nu_l^2}\right)^{1/3}\right]^{0.820}$$
$$= \left[4.81 + \frac{3.70\times(2\ \text{m})\times(0.675\ \text{W/m}\cdot°\text{C})\times(100-80)°\text{C}}{(0.315\times10^{-3}\ \text{kg/m}\cdot\text{s})(2314\times10^3\ \text{J/kg})}\left(\frac{(9.81\ \text{m/s}^2)\cos 40}{(0.326\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3}\right]^{0.82} = 1197$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined from

$$h = h_{\text{tilted,wavy}} = \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g\cos\theta}{\nu_l^2}\right)^{1/3}$$
$$= \frac{1197\times(0.675\ \text{W/m}\cdot°\text{C})}{1.08(1197)^{1.22} - 5.2}\left(\frac{(9.81\ \text{m/s}^2)\cos 40}{(0.326\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3} = \mathbf{5440\ W/m^2\cdot°C}$$

The heat transfer surface area of the plate is:

$$A = w\times L = (2\ \text{m})(2\ \text{m}) = 4\ \text{m}^2.$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA(T_{sat} - T_s) = (5440\ \text{W/m}^2\cdot°\text{C})(4\ \text{m}^2)(100-80)°\text{C} = 435{,}200\ \text{W}$$

(*b*) The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{435{,}200\ \text{J/s}}{2314\times10^3\ \text{J/kg}} = \mathbf{0.188\ kg/s}$$

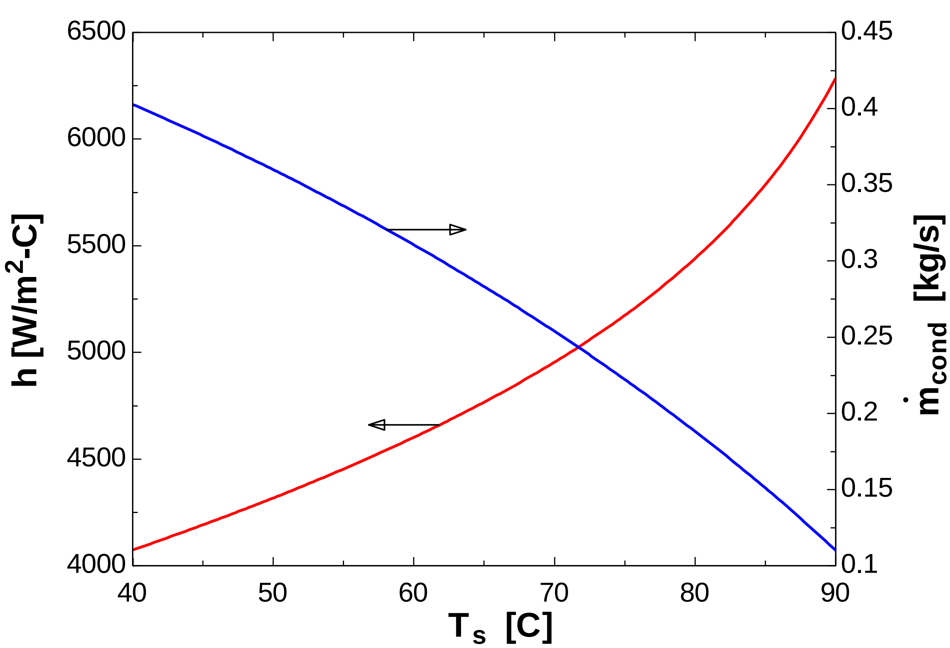
Discussion We could also determine the heat transfer coefficient from $h_{\text{inclined}} = h_{\text{vert}}(\cos\theta)^{1/4}$.

10-66 Prob. 10-65 is reconsidered. The effects of plate temperature and the angle of the plate from the vertical on the average heat transfer coefficient and the rate at which the condensate drips off are to be investigated.

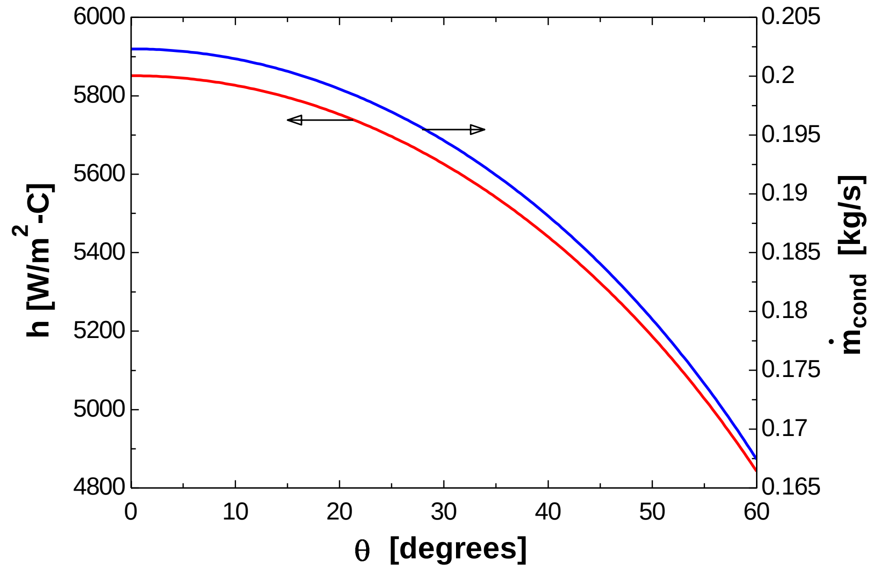
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_sat=100 [C]
L=2 [m]
theta=40 [degrees]
T_s=80 [C]

"PROPERTIES"
Fluid$='steam_IAPWS'
T_f=1/2*(T_sat+T_s)
P_sat=pressure(Fluid$, T=T_sat, x=1)
rho_l=density(Fluid$, T=T_f, x=0)
mu_l=Viscosity(Fluid$,T=T_f, x=0)
nu_l=mu_l/rho_l
c_l=CP(Fluid$, T=T_f, x=0)*Convert(kJ/kg-C, J/kg-C)
k_l=Conductivity(Fluid$, T=T_f, P=P_sat+1)
h_f=enthalpy(Fluid$, T=T_sat, x=0)
h_g=enthalpy(Fluid$, T=T_sat, x=1)
h_fg=(h_g-h_f)*Convert(kJ/kg, J/kg)
g=9.8 [m/s^2]

"ANALYSIS"
"(a)"
h_fg_star=h_fg+0.68*c_l*(T_sat-T_s)
Re=(4.81+(3.7*L*k_l*(T_sat-T_s))/(mu_l*h_fg_star)*((g*Cos(theta))/nu_l^2)^(1/3))^0.820
h=(Re*k_l)/(1.08*Re^1.22-5.2)*((g*Cos(theta))/nu_l^2)^(1/3)
Q_dot=h*A*(T_sat-T_s)
A=L^2
"(b)"
m_dot_cond=Q_dot/h_fg_star
```

T_s [C]	h [W/m^2.C]	$\dot{m}_{cond}$ [kg/s]
40	4073	0.4027
42.5	4132	0.3926
45	4191	0.3821
47.5	4253	0.3712
50	4317	0.3599
52.5	4384	0.3482
55	4453	0.3361
57.5	4526	0.3236
60	4602	0.3106
62.5	4682	0.2971
65	4767	0.2832
67.5	4857	0.2688
70	4954	0.2538
72.5	5059	0.2383
75	5174	0.2222
77.5	5300	0.2055
80	5441	0.1881
82.5	5601	0.17
85	5787	0.151
87.5	6009	0.1311
90	6286	0.11



θ [degrees]	h [W/m^2.C]	$\dot{m}_{cond}$ [kg/s]
0	5851	0.2023
3	5849	0.2022
6	5842	0.202
9	5831	0.2016
12	5816	0.2011
15	5796	0.2004
18	5771	0.1996
21	5742	0.1986
24	5708	0.1974
27	5670	0.196
30	5626	0.1945
33	5577	0.1928
36	5522	0.1909
39	5462	0.1889
42	5396	0.1866
45	5323	0.1841
48	5243	0.1813
51	5156	0.1783
54	5061	0.175
57	4957	0.1714
60	4842	0.1674

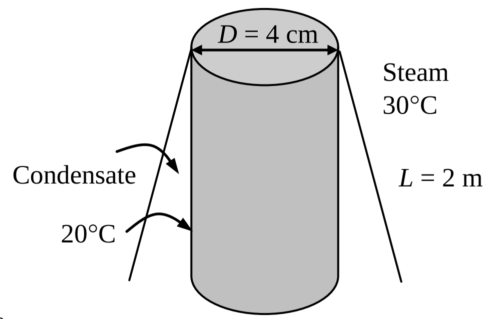


10-67 Saturated steam condenses outside of vertical tube. The rate of heat transfer to the coolant, the rate of condensation and the thickness of the condensate layer at the bottom are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal. **3** The tube can be treated as a vertical plate. **4** The condensate flow is wavy-laminar over the entire tube (this assumption will be verified). **5** Nusselt's analysis can be used to determine the thickness of the condensate film layer. **6** The density of vapor is much smaller than the density of liquid, $\rho_v << \rho_l$.

Properties The properties of water at the saturation temperature of 30°C are $h_{fg} = 2431\times10^3$ J/kg and $\rho_v = 0.03$ kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (30 + 20)/2 = 25$°C are (Table A-9),

$$\rho_l = 997.0\ \text{kg/m}^3$$
$$\mu_l = 0.891\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 0.894\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4180\ \text{J/kg}\cdot°\text{C}$$
$$k_l = 0.607\ \text{W/m}\cdot°\text{C}$$



Analysis (*a*)The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2431\times10^3\ \text{J/kg} + 0.68\times4180\ \text{J/kg}\cdot°\text{C}(30-20)°\text{C} = 2459\times10^3\ \text{J/kg}$$

Assuming wavy-laminar flow, the Reynolds number is determined from

$$\text{Re} = \text{Re}_{\text{vertical,wavy}} = \left[4.81 + \frac{3.70Lk_l(T_{sat}-T_s)}{\mu_l h^*_{fg}}\left(\frac{g}{\nu_l^2}\right)^{1/3}\right]^{0.820}$$

$$= \left[4.81 + \frac{3.70\times(2\ \text{m})\times(0.607\ \text{W/m}\cdot°\text{C})\times(30-20)°\text{C}}{(0.891\times10^{-3}\ \text{kg/m}\cdot\text{s})(2459\times10^3\ \text{J/kg})}\left(\frac{9.8\ \text{m/s}^2}{(0.894\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3}\right]^{0.82} = 157.3$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined to be

$$h = h_{\text{vertical,wavy}} = \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3}$$

$$= \frac{157.3\times(0.607\ \text{W/m}\cdot°\text{C})}{1.08(157.3)^{1.22} - 5.2}\left(\frac{9.8\ \text{m/s}^2}{(0.894\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3} = 4302\ \text{W/m}^2\cdot°\text{C}$$

The heat transfer surface area of the tube is $A_s = \pi DL = \pi(0.04\ \text{m})(2\ \text{m}) = 0.2513\ \text{m}^2$. Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (4302\ \text{W/m}^2\cdot°\text{C})(0.2513\ \text{m}^2)(30-20)°\text{C} = \mathbf{10{,}811\ W}$$

(*b*) The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{10{,}811\ \text{J/s}}{2459\times10^3\ \text{J/kg}} = \mathbf{4.40\times10^{-3}\ kg/s}$$

(*c*) Combining equations $\delta_L = k_l/h_l$ and $h = (4/3)h_L$, the thickness of the liquid film at the bottom of the tube is determined to be

$$\delta_L = \frac{4k_l}{3h} = \frac{4(0.607\ \text{W/m}\cdot°\text{C})}{3(4302\ \text{W/m}^2\cdot°\text{C})} = 0.188\times10^{-3}\ \text{m} = \mathbf{0.2\ mm}$$

Discussion The assumption of wavy laminar flow is verified since Reynolds number is between 30 and 1800. The assumption that the tube diameter is large relative to the thickness of the liquid film at the bottom of the tube is verified since the thickness of the liquid film is 0.2 mm, which is much smaller than the diameter of the tube (4 cm).

10-68 The rate of condensation and the heat transfer rate for a vertical pipe, with specified surface temperature, are to be determined.

Assumptions **1** Steady operating condition exists. **2** The surface has uniform temperature. **3** The pipe can be treated as a vertical plate. **4** The condensate flow is wavy-laminar over the entire tube (this assumption will be verified). **5** Nusselt's analysis can be used to determine the thickness of the condensate film layer. **6** The density of vapor is much smaller than the density of liquid, $\rho_v << \rho_l$.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_v = 0.5978 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 90°C are, from Table A-9,

$\rho_l = 965.3\ \text{kg/m}^3$ $\qquad c_{pl} = 4206\ \text{J/kg·K}$

$\mu_l = 0.315 \times 10^{-3}\ \text{kg/m·s}$ $\qquad k_l = 0.675\ \text{W/m·K}$

$\nu_l = \mu_l / \rho_l = 0.326 \times 10^{-6}\ \text{m}^2\text{/s}$

Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) \\ &= 2257\times 10^3 + 0.68(4206)(100-80) \\ &= 2314\times 10^3\ \text{J/kg} \end{aligned}$$

Assuming wavy-laminar flow, the Reynolds number is determined from

$$\begin{aligned} \text{Re}_{\text{vert, wavy}} &= \left[4.81 + \frac{3.70Lk_l(T_{\text{sat}}-T_s)}{\mu_l h^*_{fg}}\left(\frac{g}{\nu_l^2}\right)^{1/3}\right]^{0.820} \\ &= \left[4.81 + \frac{3.70(1)(0.675)(100-80)}{(0.315\times 10^{-3})(2314\times 10^3)}\left(\frac{9.81}{(0.326\times 10^{-6})^2}\right)^{1/3}\right]^{0.820} = 729.7 \end{aligned}$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined to be

$$\begin{aligned} h = h_{\text{vert, wavy}} &= \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3} = \frac{(729.7)(0.675\ \text{W/m}\cdot\text{K})}{1.08(729.7)^{1.22} - 5.2}\left[\frac{9.81\ \text{m/s}^2}{(0.326\times 10^{-6}\ \text{m}^2\text{/s})^2}\right]^{1/3} \\ &= 6633\ \text{W/m}^2\cdot\text{K} \end{aligned}$$

Then the rate of heat transfer during this condensation process becomes

$$\begin{aligned} \dot{Q} &= \pi DLh(T_{\text{sat}} - T_s) \\ &= \pi(0.1\ \text{m})(1\ \text{m})(6633\ \text{W/m}^2\cdot\text{K})(100-80)\ \text{K} \\ &= \mathbf{4.168\times 10^4\ W} \end{aligned}$$

The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{4.168\times 10^4\ \text{W}}{2314\times 10^3\ \text{J/kg}} = \mathbf{0.018\ kg/s}$$

Discussion Combining equations $\delta_L = k_l / h_L$ and $h = (4/3)h_L$, the thickness of the liquid film at the bottom of the tube is determined to be

$$\delta_L = \frac{4k_l}{3h} = \frac{4(0.675\ \text{W/m}\cdot\text{K})}{3(6633\ \text{W/m}^2\cdot\text{K})} = 0.136\ \text{mm} << 100\ \text{mm}$$

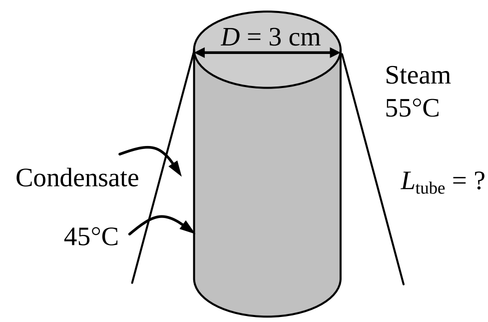
Since $\delta_L << D$, the pipe can be treated as a vertical plate.

10-69 Saturated steam at a saturation temperature of T_{sat} = 55°C condenses on the outer surface of a vertical tube which is maintained at 45°C. The required tube length to condense steam at a rate of 10 kg/h is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal. **3** The vertical tube can be treated as a vertical plate. **4** The density of vapor is much smaller than the density of liquid, $\rho_v \ll \rho_l$.

Properties The properties of water at the saturation temperature of 55°C are $h_{fg} = 2371\times10^3$ J/kg and $\rho_v = 0.1045$ kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (55 + 45)/2 = 50$°C are (Table A-9),

$$\rho_l = 988.1\ \text{kg/m}^3$$
$$\mu_l = 0.547\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 0.554\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4181\ \text{J/kg}\cdot°\text{C}$$
$$k_l = 0.644\ \text{W/m}\cdot°\text{C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2371\times10^3\ \text{J/kg} + 0.68\times4181\ \text{J/kg}\cdot°\text{C}(55-45)°\text{C} = 2399\times10^3\ \text{J/kg}$$

The Reynolds number is determined from its definition to be

$$\text{Re} = \frac{4\dot{m}}{p\mu_l} = \frac{4(10/3600\ \text{kg/s})}{\pi(0.03\ \text{m})(0.547\times10^{-3}\ \text{kg/m}\cdot\text{s})} = 215.5$$

which is between 30 and 1800. Therefore the condensate flow is wavy laminar, and the condensation heat transfer coefficient is determined from

$$h = h_{\text{vertical,wavy}} = \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3}$$
$$= \frac{215.5\times(0.644\ \text{W/m}\cdot°\text{C})}{1.08(215.5)^{1.22} - 5.2}\left(\frac{9.8\ \text{m/s}^2}{(0.554\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3} = 5644\ \text{W/m}^2\cdot°\text{C}$$

The rate of heat transfer during this condensation process is

$$\dot{Q} = \dot{m}h^*_{fg} = (10/3600\ \text{kg/s})(2399\times10^3\ \text{J/kg}) = 6{,}664\ \text{W}$$

Heat transfer can also be expressed as

$$\dot{Q} = hA_s(T_{sat} - T_s) = h(\pi DL)(T_{sat} - T_s)$$

Then the required length of the tube becomes

$$L = \frac{\dot{Q}}{h(\pi D)(T_{sat} - T_s)} = \frac{6664\ \text{W}}{(5844\ \text{W/m}^2\cdot°\text{C})\pi(0.03\ \text{m})(55-45)°\text{C}} = \mathbf{1.21\,m}$$

Discussion Combining equations $\delta_L = k_l/h_l$ and $h = (4/3)h_L$, the thickness of the liquid film at the bottom of the tube is determined to be

$$\delta_L = \frac{4k_l}{3h} = \frac{4(0.644\ \text{W/m}\cdot°\text{C})}{3(5844\ \text{W/m}^2\cdot°\text{C})} = 0.147\times10^{-3}\ \text{m} = 0.15\ \text{mm}$$

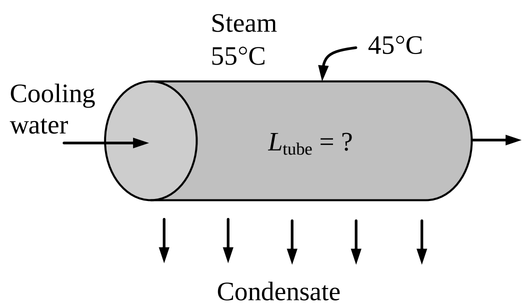
The assumption that the tube diameter is large relative to the thickness of the liquid film at the bottom of the tube is verified since the thickness of the liquid film is 0.15 mm, which is much smaller than the diameter of the tube (3 cm). Also, the assumption of wavy laminar flow is verified since Reynolds number is between 30 and 1800.

10-70 Saturated steam at a saturation temperature of T_{sat} = 55°C condenses on the outer surface of a horizontal tube which is maintained at 45°C. The required tube length to condense steam at a rate of 10 kg/h is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal.

Properties The properties of water at the saturation temperature of 55°C are $h_{fg} = 2371\times10^3$ J/kg and ρ_v = 0.1045 kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (55 + 45)/2 = 50°C$ are (Table A-9),

$$\rho_l = 988.1\,\text{kg/m}^3$$
$$\mu_l = 0.547\times10^{-3}\,\text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 0.554\times10^{-6}\,\text{m}^2/\text{s}$$
$$c_{pl} = 4181\,\text{J/kg}\cdot°\text{C}$$
$$k_l = 0.644\ \text{W/m}\cdot°\text{C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2371\times10^3\ \text{J/kg} + 0.68\times4181\,\text{J/kg}\cdot°\text{C}(55-45)°\text{C} = 2399\times10^3\ \text{J/kg}$$

Noting that the tube is horizontal, the condensation heat transfer coefficient is determined from

$$h = h_{\text{horizontal}} = 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat}-T_s)D}\right]^{1/4}$$
$$= 0.729\left[\frac{(9.8\,\text{m/s}^2)(988.1\,\text{kg/m}^3)(988.1-0.10\ \text{kg/m}^3)(2399\times10^3\ \text{J/kg})(0.644\ \text{W/m}\cdot°\text{C})^3}{(0.547\times10^{-3}\ \text{kg/m}\cdot\text{s})(55-45)°\text{C}(0.03\,\text{m})}\right]^{1/4}$$
$$= 10{,}135\ \text{W/m}^2.°\text{C}$$

The rate of heat transfer during this condensation process is

$$\dot{Q} = \dot{m}h^*_{fg} = (10/3600\ \text{kg/s})(2399\times10^3\ \text{J/kg}) = 6{,}664\ \text{W}$$

Heat transfer can also be expressed as

$$\dot{Q} = hA_s(T_{sat}-T_s) = h(\pi DL)(T_{sat}-T_s)$$

Then the required length of the tube becomes

$$L = \frac{\dot{Q}}{h(\pi D)(T_{sat}-T_s)} = \frac{6664\ \text{W}}{(10{,}135\ \text{W/m}^2\cdot°\text{C})\pi(0.03\,\text{m})(55-45)°\text{C}} = \mathbf{0.70\,m}$$

10-71 Saturated vapor condenses on the outer surface of a 1.5-m-long vertical tube at 60°C that is maintained with a surface temperature of 40°C. The rate of heat transfer to the tube and the required tube diameter to condense 12 kg/h of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Isothermal tube surface. **3** The vertical tube can be treated as a vertical plate (this assumption will be verified). **4** The condensate flow is wavy-laminar over the entire plate (this assumption will be verified). **5** The density of vapor is much smaller than the density of liquid, $\rho_v << \rho_l$.

Properties The properties of water at the saturation temperature of 60°C are h_{fg} = 2359 × 10^3 J/kg and ρ_v = 0.1304 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (60 + 40)/2 = 50$°C are (Table A-9)

$$\rho_l = 988.1\ \text{kg/m}^3$$
$$\mu_l = 0.547\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 0.554\times10^{-6}\ \text{m}^2/\text{s} \quad 0$$
$$c_{pl} = 4181\ \text{J/kg}\cdot\text{K}$$
$$k_l = 0.644\ \text{W/m}\cdot\text{K}$$

Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s) = 2359\times10^3\ \text{J/kg} + 0.68(4181\ \text{J/kg}\cdot\text{K})(60-40)\ \text{K} = 2.4159\times10^6\ \text{J/kg}$$

Assuming wavy-laminar flow, the Reynolds number is determined from

$$\text{Re} = \text{Re}_{\text{vertical,wavy}} = \left[4.81 + \frac{3.70Lk_l(T_{sat}-T_s)}{\mu_l h^*_{fg}}\left(\frac{g}{\nu_l^2}\right)^{1/3}\right]^{0.820}$$

$$= \left[4.81 + \frac{(3.70)(1.5\ \text{m})(0.644\ \text{W/m}\cdot\text{K})(60-40)\ \text{K}}{(0.547\times10^{-3}\ \text{kg/m}\cdot\text{s})(2.4159\times10^6\ \text{J/kg})}\left(\frac{9.81\ \text{m/s}^2}{(0.554\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3}\right]^{0.820} = 454.73$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined to be

$$h = h_{\text{vertical,wavy}} = \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3}$$

$$= \frac{(454.73)(0.644\ \text{W/m}\cdot\text{K})}{1.08(454.73)^{1.22} - 5.2}\left[\frac{9.81\ \text{m/s}^2}{(0.554\times10^{-6}\ \text{m}^2/\text{s})^2}\right]^{1/3} = 4937.5\ \text{W/m}^2\cdot\text{K}$$

The rate of heat transfer to the tube during this condensation process is

$$\dot{Q} = \dot{m}h^*_{fg} = (12/3600\ \text{kg/s})(2.4159\times10^6\ \text{J/kg}) = \mathbf{8053\ W}$$

Heat transfer can also be expressed as

$$\dot{Q} = hA_s(T_{sat} - T_s) = h(\pi DL)(T_{sat} - T_s)$$

Then the required diameter of the tube becomes

$$D = \frac{\dot{Q}}{h(\pi L)(T_{sat} - T_s)} = \frac{8053\ \text{W}}{(4937.5\ \text{W/m}^2\cdot\text{K})\pi(1.5\ \text{m})(60-40)\ \text{K}} = \mathbf{0.0173\ m}$$

To verify the assumption that vertical tube can be treated as a vertical plate (D >> δ), calculate δ from

$$\delta = (4k_1)/3h = 4(0.664\ \text{W/m}\cdot\text{K})/(3)(4937.5\ \text{W/m}^2\cdot\text{K}) = 1.79\times10^{-4}\ \text{m} << D = 0.0173\ \text{m}$$

Thus our assumption of D >> δ is verified.

Discussion With diameter known, the Reynolds number can also be verified to be wavy- laminar flow

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu_l} = \frac{4(12/3600\ \text{kg/s})}{\pi(0.0173\ \text{m})(0.547\times10^{-3}\ \text{kg/m}\cdot\text{s})} = 448$$

Note that the Re value obtained based on equation 10-27 will not exactly match the above calculated value of Re since equation 10-27 is based on experimental data and several approximations. However, the two values are very close (within 2%).

10-72 Repeat Prob.10-71 for a horizontal tube. Saturated vapor condenses on the outer surface of a 1.5-m-long horizontal tube at 60°C that is maintained with a surface temperature of 40°C. The rate of heat transfer to the tube and the required tube diameter to condense 12 kg/h of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Isothermal tube surface.

Properties The properties of water at the saturation temperature of 60°C are h_{fg} = 2359 × 10^3 J/kg and ρ_v = 0.1304 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (60 + 40)/2 = 50$°C are (Table A-9)

$$\begin{aligned}
\rho_l &= 988.1\,\text{kg/m}^3\\
\mu_l &= 0.547\times10^{-3}\,\text{kg/m}\cdot\text{s}\\
\nu_l &= \mu_l/\rho_l = 0.554\times10^{-6}\,\text{m}^2/\text{s}\\
c_{pl} &= 4181\,\text{J/kg}\cdot\text{K}\\
k_l &= 0.644\ \text{W/m}\cdot\text{K}
\end{aligned}$$

Analysis The modified latent heat of vaporization is

$$\begin{aligned}
h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s)\\
&= 2359\times10^3\ \text{J/kg} + 0.68(4181\,\text{J/kg}\cdot\text{K})(60-40)\,\text{K} = 2.4159\times10^6\ \text{J/kg}
\end{aligned}$$

The rate of heat transfer to the tube during this condensation process is

$$\dot{Q} = \dot{m}h^*_{fg} = (12/3600\,\text{kg/s})(2.4159\times10^6\ \text{J/kg}) = \mathbf{8053\ W}$$

Heat transfer can also be expressed as

$$\dot{Q} = hA_s(T_{\text{sat}} - T_s) = h(\pi DL)(T_{\text{sat}} - T_s) \quad\to\quad h = \frac{\dot{Q}}{(\pi DL)(T_{\text{sat}} - T_s)}$$

Noting that the tube is horizontal, the condensation heat transfer coefficient is determined from

$$h = h_{\text{horiz}} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}} - T_s)D}\right]^{1/4} = \frac{\dot{Q}}{(\pi DL)(T_{\text{sat}} - T_s)}$$

Solving for the required tube diameter,

$$\begin{aligned}
D &= \left\{0.729\frac{(\pi L)(T_{\text{sat}} - T_s)}{\dot{Q}}\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}} - T_s)}\right]^{1/4}\right\}^{-4/3}\\
&= \left\{0.729\frac{\pi(1.5\,\text{m})(60-40)\text{K}}{8053\,\text{J/s}}\right.\\
&\quad\left.\times\left[\frac{(9.81\,\text{m/s}^2)(988.1\,\text{kg/m}^3)(988.1-0.1304)\text{kg/m}^3(2.4159\times10^6\ \text{J/kg})(0.644\ \text{W/m}\cdot\text{K})^3}{(0.547\times10^{-3}\,\text{kg/m}\cdot\text{s})(60-40)\text{K}}\right]^{1/4}\right\}^{-4/3}\\
&= \mathbf{0.00694\ m}
\end{aligned}$$

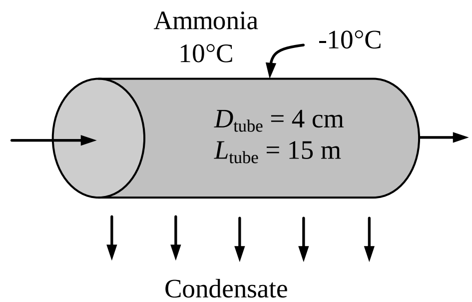
Discussion When placed vertically, the required tube diameter is about 2.5 times larger than that of a horizontal tube. Due to the higher heat transfer coefficient for a horizontal tube, in comparison with a vertical tube, the horizontal tube requires a smaller diameter for the same length to achieve the same rate of condensation.

10-73 Saturated ammonia vapor at a saturation temperature of T_{sat} = 10°C condenses on the outer surface of a horizontal tube which is maintained at -10°C. The rate of heat transfer from the ammonia and the rate of condensation of ammonia are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal.

Properties The properties of ammonia at the saturation temperature of 10°C are h_{fg} = 1226×10^3 J/kg and ρ_v = 4.870 kg/m^3 (Table A-11). The properties of liquid ammonia at the film temperature of $T_f = (T_{sat} + T_s)/2 = (10 + (-10))/2 = 0$°C are (Table A-11),

$$\rho_l = 638.6\ \text{kg/m}^3$$
$$\mu_l = 1.896\times10^{-4}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = 0.2969\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4617\ \text{J/kg}\cdot°\text{C}$$
$$k_l = 0.5390\ \text{W/m}\cdot°\text{C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 1226\times10^3\ \text{J/kg} + 0.68\times4617\ \text{J/kg}\cdot°\text{C}[10-(-10)]°\text{C} = 1288\times10^3\ \text{J/kg}$$

Noting that the tube is horizontal, the condensation heat transfer coefficient is determined from

$$h = h_{\text{horizontal}} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4}$$
$$= 0.729\left[\frac{(9.81\ \text{m/s}^2)(638.6\ \text{kg/m}^3)(638.6 - 4.870\ \text{kg/m}^3)(1288\times10^3\ \text{J/kg})(0.5390\ \text{W/m}\cdot°\text{C})^3}{(1.896\times10^{-4}\ \text{kg/m}\cdot\text{s})[10-(-10)]°\text{C}(0.02\ \text{m})}\right]^{1/4}$$
$$= 7390\ \text{W/m}^2.°\text{C}$$

The heat transfer surface area of the tube is

$$A_s = \pi DL = \pi(0.04\ \text{m})(15\ \text{m}) = 1.885\ \text{m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (7390\ \text{W/m}^2.°\text{C})(1.885\ \text{m}^2)[10-(-10)]°\text{C} = \mathbf{278{,}600\ W}$$

(*b*) The rate of condensation of ammonia is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{278{,}600\ \text{J/s}}{1288\times10^3\ \text{J/kg}} = \mathbf{0.216\ kg/s}$$

10-74 A spherical tank containing cold fluid is causing condensation of moist air on the outer surface. The rate of moisture condensation is to be determined whether or not risk of electrical hazard exists.

Assumptions **1** Steady operating conditions exist. **2** Isothermal tank surface. **3** Film condensation occurs on the tank surface.

Properties The properties of water at the saturation temperature of 25°C are h_{fg} = 2442 × 10^3 J/kg and ρ_v = 0.0231 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (25 + 5)/2 = 15$°C are (Table A-9)

$$\rho_l = 999.1\ \text{kg/m}^3$$
$$\mu_l = 1.138\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 1.139\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4185\ \text{J/kg}\cdot\text{K}$$
$$k_l = 0.589\ \text{W/m}\cdot\text{K}$$

Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s)$$
$$= 2442\times10^3\ \text{J/kg} + 0.68(4185\ \text{J/kg}\cdot\text{K})(25-5)\ \text{K} = 2.4989\times10^6\ \text{J/kg}$$

The film condensation heat transfer coefficient for a sphere is determined from

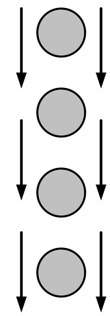
$$h = h_{\text{horiz}} = 0.815\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}}-T_s)D}\right]^{1/4}$$
$$= 0.815\left[\frac{(9.81\ \text{m/s}^2)(999.1\ \text{kg/m}^3)(999.1-0.0231)\text{kg/m}^3(2.4989\times10^6\ \text{J/kg})(0.589\ \text{W/m}\cdot\text{K})^3}{(1.138\times10^{-3}\ \text{kg/m}\cdot\text{s})(25-5)\text{K}(3\ \text{m})}\right]^{1/4}$$
$$= 2384.1\ \text{W/m}^2\cdot\text{K}$$

Thus, the rate of film condensation is

$$\dot{m} = \frac{\dot{Q}}{h^*_{fg}} = \frac{h(\pi D^2)(T_{\text{sat}}-T_s)}{h^*_{fg}}$$
$$= \frac{(2384.1\ \text{W/m}^2\cdot\text{K})\pi(3\ \text{m})^2(25-5)\ \text{K}}{2.4989\times10^6\ \text{J/kg}} = \mathbf{0.5395\ kg/s} > 0.5\ \text{kg/s}$$

Discussion The rate of condensation from the tank surface is greater than the capability of the system in removing the condensate. Thus, there is a risk of excess condensate coming in contact with the high voltage device and cause electrical hazard. To prevent electrical hazard, the preventive system should be capable of removing more than 0.54 kg/s of condensate.

10-75 There is film condensation on the outer surfaces of *N* horizontal tubes arranged in a vertical tier. The value of *N* for which the average heat transfer coefficient for the entire tier be equal to half of the value for a single horizontal tube is to be determined.



Assumptions Steady operating conditions exist.

Analysis The relation between the heat transfer coefficients for the two cases is given to be

$$h_{\text{horizontal, N tubes}} = \frac{h_{\text{horizontal, 1 tube}}}{N^{1/4}}$$

Therefore,

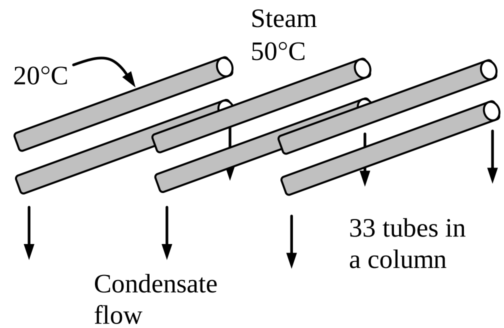
$$\frac{h_{\text{horizontal, N tubes}}}{h_{\text{horizontal, 1 tube}}} = \frac{1}{2} = \frac{1}{N^{1/4}} \longrightarrow N = \mathbf{16}$$

10-76 Saturated steam at a saturation temperature of T_{sat} = 50°C condenses on the outer surfaces of a tube bank with 33 tubes in each column maintained at 20°C. The average heat transfer coefficient and the rate of condensation of steam on the tubes are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal.

Properties The properties of water at the saturation temperature of 50°C are h_{fg} = 2383×10^3 J/kg and ρ_v = 0.0831 kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (50 + 20)/2 = 35$°C are (Table A-9),

$$\begin{aligned} \rho_l &= 994.0 \text{ kg/m}^3 \\ \mu_l &= 0.720\times10^{-3} \text{ kg/m}\cdot\text{s} \\ \nu_l &= \mu_l / \rho_l = 0.724\times10^{-6} \text{ m}^2\text{/s} \\ c_{pl} &= 4178 \text{ J/kg}\cdot°\text{C} \\ k_l &= 0.623 \text{ W/m}\cdot°\text{C} \end{aligned}$$



Analysis (*a*) The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68 c_{pl}(T_{sat} - T_s) \\ &= 2383\times10^3 \text{ J/kg} + 0.68\times4178 \text{ J/kg}\cdot°\text{C}(50-20)°\text{C} \\ &= 2468\times10^3 \text{ J/kg} \end{aligned}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$\begin{aligned} h = h_{\text{horizontal}} &= 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4} \\ &= 0.729\left[\frac{(9.8 \text{ m/s}^2)(994 \text{ kg/m}^3)(994 - 0.08 \text{ kg/m}^3)(2468\times10^3 \text{ J/kg})(0.623 \text{ W/m}\cdot°\text{C})^3}{(0.720\times10^{-3} \text{ kg/m}\cdot\text{s})(50-20)°\text{C}(0.015 \text{ m})}\right]^{1/4} \\ &= 8425 \text{ W/m}^2\cdot°\text{C} \end{aligned}$$

Then the average heat transfer coefficient for a 33-tube high vertical tier becomes

$$h_{\text{horiz, N tubes}} = \frac{1}{N^{1/4}} h_{\text{horiz,1 tube}} = \frac{1}{33^{1/4}}(8425 \text{ W/m}^2\cdot°\text{C}) = \mathbf{3515\ W/m^2\cdot°C}$$

The surface area for all 33 tubes per unit length is

$$A_s = N_{\text{total}}\pi DL = 33\pi(0.015 \text{ m})(1 \text{ m}) = 1.555 \text{ m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (3515 \text{ W/m}^2\cdot°\text{C})(1.555 \text{ m}^2)(50-20)°\text{C} = 164{,}000 \text{ W}$$

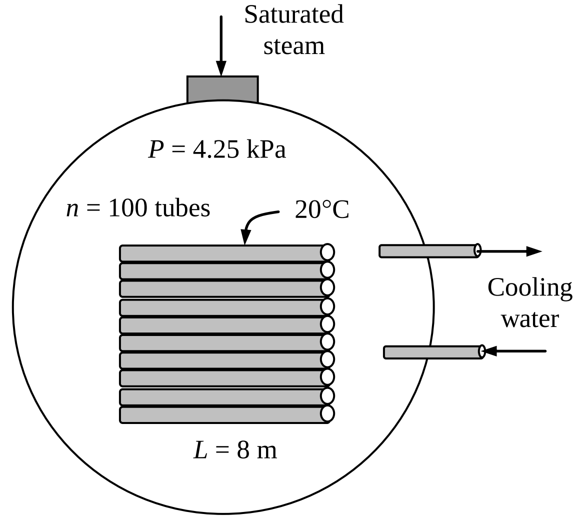
(*b*) The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{164{,}000 \text{ J/s}}{2468\times10^3 \text{ J/kg}} = \mathbf{0.0664\ kg/s}$$

10-77 Saturated steam at a pressure of 4.25 kPa and thus at a saturation temperature of T_{sat} = 30°C (Table A-9) condenses on the outer surfaces of 100 horizontal tubes arranged in a 10×10 square array maintained at 20°C by circulating cooling water. The rate of heat transfer to the cooling water and the rate of condensation of steam on the tubes are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal.

Properties The properties of water at the saturation temperature of 30°C are $h_{fg} = 2431\times10^3$ J/kg and $\rho_v = 0.03$ kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (30 + 20)/2 = 25°C$ are (Table A-9),



$$\rho_l = 997.0 \text{ kg/m}^3$$
$$\mu_l = 0.891\times10^{-3} \text{ kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l / \rho_l = 0.894\times10^{-6} \text{ m}^2/\text{s}$$
$$c_{pl} = 4180 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 0.607 \text{ W/m}\cdot°\text{C}$$

Analysis (*a*) The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2431\times10^3 \text{ J/kg} + 0.68\times4180 \text{ J/kg}\cdot°\text{C}(30-20)°\text{C} = 2{,}459\times10^3 \text{ J/kg}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$h = h_{\text{horizontal}} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4}$$
$$= 0.729\left[\frac{(9.8 \text{ m/s}^2)(997 \text{ kg/m}^3)(997 - 0.03 \text{ kg/m}^3)(2459\times10^3 \text{ J/kg})(0.607 \text{ W/m}\cdot°\text{C})^3}{(0.891\times10^{-3} \text{ kg/m}\cdot\text{s})(30-20)°\text{C}(0.03 \text{ m})}\right]^{1/4}$$
$$= 8674 \text{ W/m}^2.°\text{C}$$

Then the average heat transfer coefficient for a 10-pipe high vertical tier becomes

$$h_{\text{horiz, N tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{10^{1/4}}(8674 \text{ W/m}^2\cdot°\text{C}) = 4878 \text{ W/m}^2\cdot°\text{C}$$

The surface area for all 100 tubes is

$$A_s = N_{\text{total}}\pi DL = 100\pi(0.03 \text{ m})(8 \text{ m}) = 75.40 \text{ m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (4878 \text{ W/m}^2.°\text{C})(75.40 \text{ m}^2)(30-20)°\text{C} = 3{,}678{,}000 \text{ W} = \mathbf{3678\ kW}$$

(*b*) The rate of condensation of steam is determined from

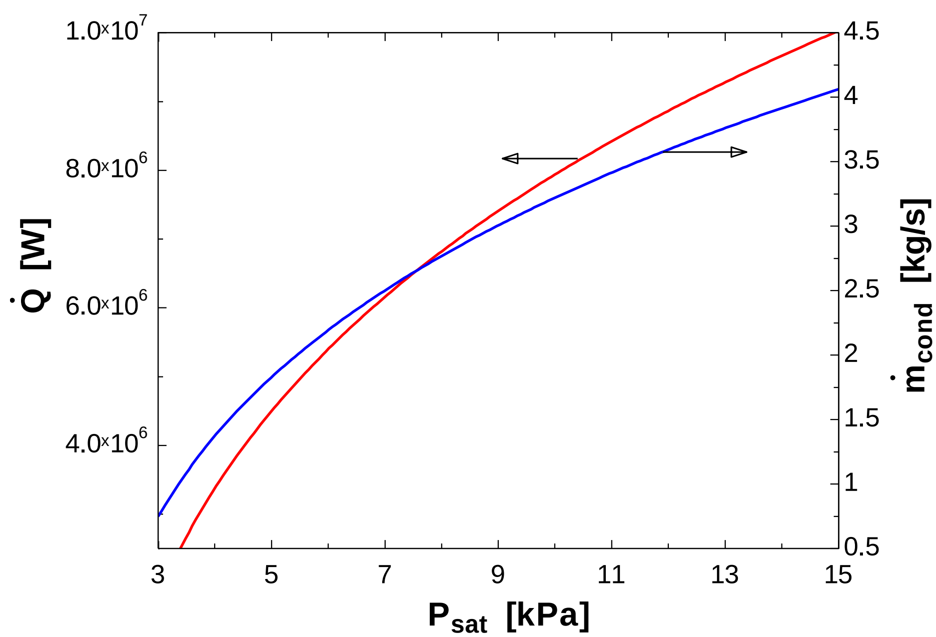
$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{3{,}678{,}000 \text{ J/s}}{2459\times10^3 \text{ J/kg}} = \mathbf{1.496\ kg/s}$$

10-78 Prob. 10-77 is reconsidered. The effect of the condenser pressure on the rate of heat transfer and the rate of condensation of the steam is to be investigated.

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

```
"GIVEN"
P_sat=4.25 [kPa]
n_tube=100
N=10
L=8 [m]
D=0.03 [m]
T_s=20 [C]

"PROPERTIES"
Fluid$='steam_IAPWS'
T_sat=temperature(Fluid$, P=P_sat, x=1)
T_f=1/2*(T_sat+T_s)
h_f=enthalpy(Fluid$, T=T_sat, x=0)
h_g=enthalpy(Fluid$, T=T_sat, x=1)
h_fg=(h_g-h_f)*Convert(kJ/kg, J/kg)
rho_v=density(Fluid$, T=T_sat, x=1)
rho_l=density(Fluid$, T=T_f, x=0)
mu_l=Viscosity(Fluid$,T=T_f, x=0)
nu_l=mu_l/rho_l
c_l=CP(Fluid$, T=T_f, x=0)*Convert(kJ/kg-C, J/kg-C)
k_l=Conductivity(Fluid$, T=T_f, P=P_sat+1)
g=9.8 [m/s^2]

"ANALYSIS"
h_fg_star=h_fg+0.68*c_l*(T_sat-T_s)
h_1tube=0.729*((g*rho_l*(rho_l-rho_v)*h_fg_star*k_l^3)/(mu_l*(T_sat-T_s)*D))^0.25
h=1/N^0.25*h_1tube
Q_dot=h*A*(T_sat-T_s)
A=n_tube*pi*D*L
m_dot_cond=Q_dot/h_fg_star
```

P_{sat} [kPa]	$\dot{Q}$ [W]	$\dot{m}_{cond}$ [kg/s]
3	1834387	0.7471
4	3374608	1.373
5	4495906	1.828
6	5397473	2.193
7	6158471	2.501
8	6820321	2.769
9	7408006	3.006
10	7937849	3.22
11	8421171	3.415
12	8866242	3.594
13	9279144	3.76
14	9664649	3.916
15	10026498	4.061

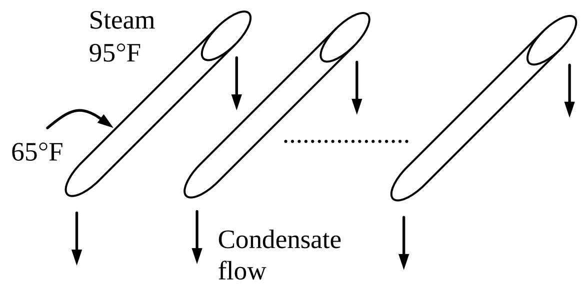


10-79E Saturated steam at a saturation temperature of T_{sat} = 95°F condenses on the outer surfaces of horizontal pipes which are maintained at 65°F by circulating cooling water. The rate of heat transfer to the cooling water and the rate of condensation per unit length of a single horizontal pipe are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The pipe is isothermal. **3** There is no interference between the pipes (no drip of the condensate from one tube to another).

Properties The properties of water at the saturation temperature of 95°F are h_{fg} = 1040 Btu/lbm and ρ_v = 0.0025 lbm/ft^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (95 + 65)/2 = 80$°F are (Table A-9E),

$$\rho_l = 62.22 \text{ lbm/ft}^3$$
$$\mu_l = 5.764\times10^{-4} \text{ lbm/ft}\cdot\text{s} = 2.075 \text{ lbm/ft}\cdot\text{h}$$
$$\nu_l = \mu_l / \rho_l = 0.03335 \text{ ft}^2\text{/h}$$
$$c_{pl} = 0.999 \text{ Btu/lbm}\cdot°\text{F}$$
$$k_l = 0.352 \text{ Btu/h}\cdot\text{ft}\cdot°\text{F}$$



Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{sat} - T_s) \\ &= 1040 \text{ Btu/lbm} + 0.68\times(0.999 \text{ Btu/lbm}\cdot°\text{F})(95-65)°\text{F} \\ &= 1060 \text{ Btu/lbm} \end{aligned}$$

Noting that we have condensation on a horizontal tube, the heat transfer coefficient is determined from

$$\begin{aligned} h = h_{\text{horiz}} &= 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4} \\ &= 0.729\left[\frac{(32.2 \text{ ft/s}^2)(62.22 \text{ lbm/ft}^3)(62.22 - 0.0025 \text{ lbm/ft}^3)(1060 \text{ Btu/lbm})(0.352 \text{ Btu/h}\cdot\text{ft}\cdot°\text{F})^3}{[(1 \text{ h/3600 s})^2](2.075 \text{ lbm/ft}\cdot\text{h})(95-65)°\text{F}(1/12 \text{ ft})}\right]^{1/4} \\ &= 1420 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F} \end{aligned}$$

The heat transfer surface area of the tube per unit length is

$$A_s = \pi DL = \pi(1/12 \text{ ft})(1 \text{ ft}) = 0.2618 \text{ ft}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (1420 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F})(0.2618 \text{ ft}^2)(95-65)°\text{F} = \mathbf{11{,}150 \text{ Btu/h}}$$

(*b*) The rate of condensation of steam is determined from

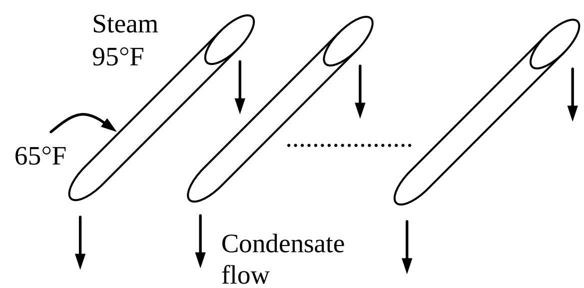
$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{11{,}150 \text{ Btu/h}}{1060 \text{ Btu/lbm}} = \mathbf{10.5 \text{ lbm/h}}$$

10-80E Saturated steam at a saturation temperature of T_{sat} = 95°F condenses on the outer surfaces of 20 horizontal pipes which are maintained at 65°F by circulating cooling water and arranged in a rectangular array of 4 pipes high and 5 pipes wide. The rate of heat transfer to the cooling water and the rate of condensation per unit length of the pipes are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The pipes are isothermal.

Properties The properties of water at the saturation temperature of 95°F are h_{fg} = 1040 Btu/lbm and ρ_v = 0.0025 lbm/ft^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (95 + 65)/2 = 80$°F are (Table A-9E),

$$\rho_l = 62.22 \text{ lbm/ft}^3$$
$$\mu_l = 5.764\times10^{-4} \text{ lbm/ft}\cdot\text{s} = 2.075 \text{ lbm/ft}\cdot\text{h}$$
$$\nu_l = \mu_l/\rho_l = 0.03335 \text{ ft}^2\text{/h}$$
$$c_{pl} = 0.999 \text{ Btu/lbm}\cdot°\text{F}$$
$$k_l = 0.352 \text{ Btu/h}\cdot\text{ft}\cdot°\text{F}$$



Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{sat} - T_s) \\ &= 1040 \text{ Btu/lbm} + 0.68\times(0.999 \text{ Btu/lbm}\cdot°\text{F})(95-65)°\text{F} \\ &= 1060 \text{ Btu/lbm} \end{aligned}$$

Noting that we have condensation on a horizontal tube, the heat transfer coefficient is determined from

$$h = h_{horiz} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4}$$

$$= 0.729\left[\frac{(32.2 \text{ ft/s}^2)(62.22 \text{ lbm/ft}^3)(62.22 - 0.0025 \text{ lbm/ft}^3)(1060 \text{ Btu/lbm})(0.352 \text{ Btu/h}\cdot\text{ft}\cdot°\text{F})^3}{[(1 \text{ h}/3600 \text{ s})^2](2.075 \text{ lbm/ft}\cdot\text{h})(95-65)°\text{F}(1/12 \text{ ft})}\right]^{1/4}$$

$$= 1420 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

Then the average heat transfer coefficient for a 4-pipe high vertical tier becomes

$$h_{\text{horiz, N tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{4^{1/4}}(1420 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F}) = 1004 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

The surface area for all 32 pipes per unit length of the pipes is

$$A_s = N_{total}\pi DL = 32\pi(1/12 \text{ ft})(1 \text{ ft}) = 8.378 \text{ ft}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (1004 \text{ Btu/h.ft}^2\cdot°\text{F})(8.378 \text{ ft}^2)(95-65)°\text{F} = \mathbf{252{,}300 \ Btu/h}$$

(*b*) The rate of condensation of steam is determined from

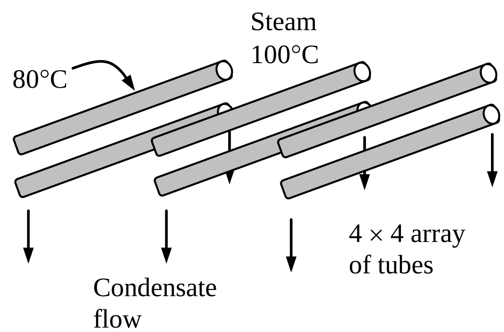
$$\dot{m}_{condensation} = \frac{\dot{Q}}{h^*_{fg}} = \frac{252{,}300 \text{ Btu/h}}{1060 \text{ Btu/lbm}} = \mathbf{238 \ lbm/h}$$

10-81 Saturated steam at a saturation temperature of T_{sat} = 100°C condenses on the outer surfaces of a tube bank maintained at 80°C. The rate of condensation of steam on the tubes are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257×10^3 J/kg and ρ_v = 0.5978 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 =$ (100 + 80)/2 = 90°C are (Table A-9),

$$\rho_l = 965.3\ \text{kg/m}^3$$
$$\mu_l = 0.315\times10^{-3}\ \text{kg/m}\cdot\text{s}$$
$$c_{pl} = 4206\ \text{J/kg}\cdot°\text{C}$$
$$k_l = 0.675\ \text{W/m}\cdot°\text{C}$$



Analysis (*a*) The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{sat} - T_s) \\ &= 2357\times10^3\ \text{J/kg} + 0.68\times4206\ \text{J/kg}\cdot°\text{C}(100-80)°\text{C} \\ &= 2314\times10^3\ \text{J/kg} \end{aligned}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$\begin{aligned} h = h_{\text{horizontal}} &= 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat}-T_s)D}\right]^{1/4} \\ &= 0.729\left[\frac{(9.8\ \text{m/s}^2)(965.3\ \text{kg/m}^3)(965.3-0.5978\ \text{kg/m}^3)(2314\times10^3\ \text{J/kg})(0.675\ \text{W/m}\cdot°\text{C})^3}{(0.315\times10^{-3}\ \text{kg/m}\cdot\text{s})(100-80)°\text{C}(0.05\ \text{m})}\right]^{1/4} \\ &= 8736\ \text{W/m}^2\cdot°\text{C} \end{aligned}$$

Then the average heat transfer coefficient for a 4-pipe high vertical tier becomes

$$h_{\text{horiz, N tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{4^{1/4}}(8736\ \text{W/m}^2\cdot°\text{C}) = 6177\ \text{W/m}^2\cdot°\text{C}$$

The surface area for all 16 tubes is

$$A_s = N_{\text{total}}\pi DL = 16\pi(0.05\ \text{m})(2\ \text{m}) = 5.027\ \text{m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (6177\ \text{W/m}^2\cdot°\text{C})(5.027\ \text{m}^2)(100-80)°\text{C} = 621{,}000\ \text{W}$$

The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{621{,}000\ \text{J/s}}{2314\times10^3\ \text{J/kg}} = 0.2684\ \text{kg/s} = \mathbf{966\ kg/h}$$

10-82 Saturated water vapor at a pressure of 12.4 kPa condenses on a rectangular array of 100 horizontal tubes at 30°C. The condensation rate per unit length of is to be determined.

Assumptions **1** Steady operating condition exists. **2** The tube surfaces are isothermal.

Properties The properties of water at the saturation temperature of 50°C corresponding to 12.4 kPa are h_{fg} = 2383 kJ/kg and ρ_v = 0.0831 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2$ = 40°C are, from Table A-9,

ρ_l = 992.1 kg/m^3 $\qquad$ c_{pl} = 4179 J/kg·K

μ_l = 0.653 × 10^{-3} kg/m·s $\qquad$ k_l = 0.631 W/m·K

Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) \\ &= 2383\times10^3 + 0.68(4179)(50-30) \\ &= 2440\times10^3 \text{ J/kg} \end{aligned}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$\begin{aligned} h_{\text{horiz,1 tube}} &= 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}}-T_s)D}\right]^{1/4} \\ &= 0.729\left[\frac{(9.81)(992.1)(992.1-0.0831)(2440\times10^3)(0.631)^3}{(0.653\times10^{-3})(50-30)(0.008)}\right]^{1/4} \\ &= 11{,}250 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

Then the average heat transfer coefficient for a 5-tube high vertical tier becomes

$$h = h_{\text{horiz},N\text{ tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{5^{1/4}}(11{,}250 \text{ W/m}^2\cdot\text{K}) = 7523 \text{ W/m}^2\cdot\text{K}$$

The rate of heat transfer per unit length during this condensation process becomes

$$\begin{aligned} \dot{Q}/L &= N_{\text{total}}\pi Dh(T_{\text{sat}} - T_s) \\ &= (100)\pi(0.008\text{ m})(7523\text{ W/m}^2\cdot\text{K})(50-30)\text{ K} \\ &= 3.781\times10^5 \text{ W/m} \end{aligned}$$

The rate of condensation per unit length is

$$\frac{\dot{m}_{\text{condensation}}}{L} = \frac{\dot{Q}/L}{h^*_{fg}} = \frac{3.781\times10^5 \text{ W/m}}{2440\times10^3 \text{ J/kg}} = \mathbf{0.155\,kg/s\cdot m}$$

Discussion Therefore, water vapor condenses at a rate of 155 g/s per meter length of the tubes.

10-83 Saturated water vapor at a pressure of 12.4 kPa condenses on an array of 100 horizontal tubes at 30°C. The condensation rates for (*a*) a rectangular array of 5 tubes high and 20 tubes wide and (*b*) a square array of 10 tubes high and 10 tubes wide are to be determined.

Assumptions **1** Steady operating condition exists. **2** The tube surfaces are isothermal.

Properties The properties of water at the saturation temperature of 50°C corresponding to 12.4 kPa are h_{fg} = 2383 kJ/kg and ρ_v = 0.0831 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 40°C are, from Table A-9,

ρ_l = 992.1 kg/m^3 $\quad$ c_{pl} = 4179 J/kg·K

$\mu_l = 0.653 \times 10^{-3}$ kg/m·s $\quad$ k_l = 0.631 W/m·K

Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) \\ &= 2383\times10^3 + 0.68(4179)(50-30) \\ &= 2440\times10^3 \text{ J/kg} \end{aligned}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$\begin{aligned} h_{\text{horiz,1 tube}} &= 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}}-T_s)D}\right]^{1/4} \\ &= 0.729\left[\frac{(9.81)(992.1)(992.1-0.0831)(2440\times10^3)(0.631)^3}{(0.653\times10^{-3})(50-30)(0.008)}\right]^{1/4} \\ &= 11{,}250 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

(*a*) For a rectangular array, the average heat transfer coefficient for a 5-tube high vertical tier becomes

$$h = h_{\text{horiz},N\text{ tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{5^{1/4}}(11{,}250 \text{ W/m}^2\cdot\text{K}) = 7523 \text{ W/m}^2\cdot\text{K}$$

The rate of heat transfer during this condensation process becomes

$$\dot{Q} = N_{\text{total}}\pi DLh(T_{\text{sat}} - T_s) = (100)\pi(0.008\text{ m})(1\text{ m})(7523\text{ W/m}^2\cdot\text{K})(50-30)\text{ K} = 3.781\times10^5 \text{ W}$$

The rate of condensation is

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{3.781\times10^5 \text{ W}}{2440\times10^3 \text{ J/kg}} = \mathbf{0.155\ kg/s} \qquad \text{(rectangular array)}$$

(*b*) For a square array, the average heat transfer coefficient for a 10-tube high vertical tier becomes

$$h = h_{\text{horiz},N\text{ tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{10^{1/4}}(11{,}250 \text{ W/m}^2\cdot\text{K}) = 6326 \text{ W/m}^2\cdot\text{K}$$

The rate of heat transfer during this condensation process becomes

$$\dot{Q} = N_{\text{total}}\pi DLh(T_{\text{sat}} - T_s) = (100)\pi(0.008\text{ m})(1\text{ m})(6326\text{ W/m}^2\cdot\text{K})(50-30)\text{ K} = 3.180\times10^5 \text{ W}$$

The rate of condensation is

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{3.180\times10^5 \text{ W}}{2440\times10^3 \text{ J/kg}} = \mathbf{0.130\ kg/s} \qquad \text{(square array)}$$

Discussion The condensation rate of the rectangular array tube bank is about 20% higher than that of the square array tube bank:

$$\frac{\dot{m}_{\text{condensation}}(\text{rectangular})}{\dot{m}_{\text{condensation}}(\text{square})} = \frac{0.155\text{ kg/s}}{0.130\text{ kg/s}} = 1.19$$

10-84 Saturated refrigerant-134a vapor is condensed as it is flowing through a tube. With a given vapor flow rate at the entrance, the flow rate of the vapor at the exit is to be determined.

Assumptions **1** Steady operating condition exists. **2** The tube surfaces are isothermal.

Properties The properties of refrigerant-134a at the saturation temperature of 35°C corresponding to 888 kPa are h_{fg} = 168.2 kJ/kg, ρ_v = 43.41 kg/m^3, and $\mu_v = 1.327 \times 10^{-5}$ kg/m·s (Table A-10). The properties of liquid refrigerant-134a at the film temperature of $T_f = (T_{sat} + T_s)/2$ = 25°C are, from Table A-10,

ρ_l = 1207 kg/m^3 $\qquad$ c_{pl} = 1427 J/kg·K

$\mu_l = 2.012 \times 10^{-4}$ kg/m·s $\qquad$ k_l = 0.0833 W/m·K

Analysis The modified latent heat of vaporization for film condensation inside horizontal tube is

$$\begin{aligned} h^*_{fg} &= h_{fg} + \frac{3}{8}c_{pl}(T_{\text{sat}} - T_s) \\ &= 168.2\times10^3 + \frac{3}{8}(1427)(35-15) \\ &= 178.9\times10^3 \text{ J/kg} \end{aligned}$$

The Reynolds number associated with film condensation inside a horizontal tube is

$$\text{Re}_{\text{vapor}} = \left(\frac{\rho_v V_v D}{\mu_v}\right)_{\text{inlet}} = \frac{4\dot{m}_{v,\text{ inlet}}}{\pi D\mu_v} = \frac{4(0.003\text{ kg/s})}{\pi(0.012\text{ m})(1.327\times10^{-5}\text{ kg/m}\cdot\text{s})} = 24{,}000 < 35{,}000$$

Hence, the heat transfer coefficient for film condensation inside a horizontal tube can be determined using

$$\begin{aligned} h = h_{\text{internal}} &= 0.555\left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h^*_{fg}}{\mu_l(T_{\text{sat}} - T_s)D}\right]^{1/4} \\ &= 0.555\left[\frac{(9.81)(1207)(1207-43.41)(0.0833)^3(178.9\times10^3)}{(2.012\times10^{-4})(35-15)(0.012)}\right]^{1/4} \\ &= 1293\text{ W/m}^2\cdot\text{K} \end{aligned}$$

The rate of heat transfer during this condensation process becomes

$$\begin{aligned} \dot{Q} &= \pi DLh(T_{\text{sat}} - T_s) \\ &= \pi(0.012\text{ m})(0.25\text{ m})(1293\text{ W/m}^2\cdot\text{K})(35-15)\text{ K} \\ &= 243.7\text{ W} \end{aligned}$$

Then, the rate of condensation can be calculated as

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{243.7\text{ W}}{178.9\times10^3\text{ J/kg}} = 0.00136\text{ kg/s}$$

Applying the conservation of mass, the flow rate of vapor leaving the tube can be determined as

$$\dot{m}_{v,\text{ inlet}} = \dot{m}_{v,\text{ outlet}} + \dot{m}_{\text{condensation}} \qquad \rightarrow \qquad \dot{m}_{v,\text{ outlet}} = \dot{m}_{v,\text{ inlet}} - \dot{m}_{\text{condensation}}$$

$$\dot{m}_{v,\text{ outlet}} = 0.003\text{ kg/s} - 0.00136\text{ kg/s} = \mathbf{0.00164\text{ kg/s}}$$

Discussion About 45% of the refrigerant-134a vapor that entered the tube is condensed inside it.

10-85 Saturated ammonia vapor is condensed as it flows through a tube. With a given vapor flow rate at the exit, the flow rate of the vapor at the inlet is to be determined.

Assumptions **1** Steady operating condition exists. **2** The tube surfaces are isothermal. **3** The Reynolds number of the vapor at the inlet is less than 35,000 (this assumption will be verified).

Properties The properties of ammonia at the saturation temperature of 25°C corresponding to 1003 kPa are h_{fg} = 1166 kJ/kg, ρ_v = 7.809 kg/m^3, and $\mu_v = 1.037 \times 10^{-5}$ kg/m·s (Table A-11). The properties of liquid ammonia at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 15°C are, from Table A-11,

ρ_l = 617.5 kg/m^3

c_{pl} = 4709 J/kg·K

$\mu_l = 1.606 \times 10^{-4}$ kg/m·s

k_l = 0.5042 W/m·K

Analysis The modified latent heat of vaporization for film condensation inside horizontal tube is

$$\begin{aligned} h^*_{fg} &= h_{fg} + \frac{3}{8}c_{pl}(T_{\text{sat}} - T_s) \\ &= 1166 \times 10^3 + \frac{3}{8}(4709)(25 - 5) \\ &= 1201 \times 10^3 \text{ J/kg} \end{aligned}$$

Assuming Re_{vapor} < 35,000 and the heat transfer coefficient for film condensation inside a horizontal tube can be determined using

$$\begin{aligned} h = h_{\text{internal}} &= 0.555\left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h^*_{fg}}{\mu_l(T_{\text{sat}} - T_s)D}\right]^{1/4} \\ &= 0.555\left[\frac{(9.81)(617.5)(617.5-7.809)(0.5042)^3(1201\times10^3)}{(1.606\times10^{-4})(25-5)(0.025)}\right]^{1/4} \\ &= 5091 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

The rate of heat transfer during this condensation process becomes

$$\begin{aligned} \dot{Q} &= \pi D L h(T_{\text{sat}} - T_s) \\ &= \pi(0.025\text{ m})(0.5\text{ m})(5091\text{ W/m}^2\cdot\text{K})(25-5)\text{ K} \\ &= 3998\text{ W} \end{aligned}$$

Then, the rate of condensation can be calculated as

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{3998\text{ W}}{1201\times10^3\text{ J/kg}} = 0.00333\text{ kg/s}$$

Applying the conservation of mass, the flow rate of vapor leaving the tube can be determined as

$$\dot{m}_{v,\text{inlet}} = \dot{m}_{v,\text{outlet}} + \dot{m}_{\text{condensation}}$$

$$\dot{m}_{v,\text{inlet}} = 0.002\text{ kg/s} + 0.00333\text{ kg/s} = \mathbf{0.00533\text{ kg/s}}$$

Discussion The Reynolds number associated with film condensation inside a horizontal tube is

$$\text{Re}_{\text{vapor}} = \left(\frac{\rho_v V_v D}{\mu_v}\right)_{\text{inlet}} = \frac{4\dot{m}_{v,\text{inlet}}}{\pi D\mu_v} = \frac{4(0.00533\text{ kg/s})}{\pi(0.025\text{ m})(1.037\times10^{-5}\text{ kg/m}\cdot\text{s})} = 26{,}200 < 35{,}000$$

Thus, the Re_{vapor} < 35,000 assumption is appropriate for this problem.

10-86 A copper tube transporting cold coolant has a surface temperature of 5°C and condenses moist air at 25°C. The rate of condensation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Isothermal tube surface.

Properties The required property of water at the saturation temperature T_{sat} = 25°C is h_{fg} = 2442 × 10^3 J/kg (Table A-9). The required property of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (25 + 5)/2$ = 15°C is c_{pl} = 4185 J/kg·K (Table A-9).

Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2442\times10^3 \text{ J/kg} + 0.68(4185 \text{ J/kg}\cdot\text{K})(25-5)\text{ K} = 2.4989\times10^6 \text{ J/kg}$$

The dropwise condensation heat transfer coefficient, for 22°C < T_{sat} < 100°C, is determined to be

$$h = h_{\text{dropwise}} = 51{,}104 + 2044T_{sat}$$
$$= 51{,}104 + 2044(25°\text{C}) = 102{,}204 \text{ W/m}^2\cdot\text{K}$$

The rate of heat transfer to the tube during this condensation process is

$$\dot{Q} = hA_s(T_{sat} - T_s) = h(\pi DL)(T_{sat} - T_s)$$
$$= (102{,}204 \text{ W/m}^2\cdot\text{K})\pi(0.025\text{ m})(10\text{ m})(25-5)\text{ K}$$
$$= 1.6054\times10^6 \text{ W}$$

Thus, the rate of condensation is

$$\dot{m} = \frac{\dot{Q}}{h^*_{fg}} = \frac{1.6054\times10^6 \text{ J/s}}{2.4989\times10^6 \text{ J/kg}} = \mathbf{0.6424\ kg/s}$$

Discussion The heat transfer rate during the dropwise condensation can be increased by increasing the surface area of the tube, and thereby increasing the rate of condensation as well.

10-87 A copper tube that is used for transporting cold fluid is causing condensation of moist air on its outer surface. The rate of moisture condensation is to be determined so that a system for removing the condensate can be sized to alleviate the risk of electrical hazard.

Assumptions **1** Steady operating conditions exist. **2** Isothermal tube surface.

Properties The required property of water at the saturation temperature T_{sat} = 27°C is h_{fg} = 2438 × 10^3 J/kg (Table A-9). The required property of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (27 + 3)/2$ = 15°C is c_{pl} = 4185 J/kg·K (Table A-9).

Analysis Since dropwise condensation can occur, and it will have higher rate of heat transfer and therefore higher condensation rate than film condensation. The system must be able to handle the dropwise condensation rate.

The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68 c_{pl}(T_{\text{sat}} - T_s) \\ &= 2438\times 10^3 \text{ J/kg} + 0.68(4185 \text{ J/kg}\cdot\text{K})(27-3)\text{ K} = 2.5063\times 10^6 \text{ J/kg} \end{aligned}$$

The dropwise condensation heat transfer coefficient, for 22°C < T_{sat} < 100°C, is determined to be

$$\begin{aligned} h = h_{\text{dropwise}} &= 51{,}104 + 2044 T_{\text{sat}} \\ &= 51{,}104 + 2044(27°\text{C}) = 106{,}292 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

The rate of heat transfer during the condensation process is

$$\begin{aligned} \dot{Q} &= hA_s(T_{\text{sat}} - T_s) \\ &= h(\pi DL)(T_{\text{sat}} - T_s) \\ &= (106{,}292 \text{ W/m}^2\cdot\text{K})\pi(0.025\text{ m})(10\text{ m})(27-3)\text{ K} \\ &= 2.0036\times 10^6 \text{ W} \end{aligned}$$

Thus, the rate of dropwise condensation is

$$\dot{m} = \frac{\dot{Q}}{h^*_{fg}} = \frac{2.0036\times 10^6 \text{ J/s}}{2.5063\times 10^6 \text{ J/kg}} = \mathbf{0.799\ kg/s}$$

Discussion In order to alleviate the risk of electrical hazard, the system must be able to remove the condensate at a rate of 0.8 kg/s. Note that dropwise condensation rate are higher than film condensation rate (by 10 times or more), thus a system capable of removing the condensate at a rate of 0.8 kg/s can also handle the condensate from film condensation.

10-88 Steam condenses at 60°C on a copper tube that is maintained with a surface temperature of 40°C. The condensation rates during film condensation and dropwise condensation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Isothermal tube surface.

Properties The properties of water at the saturation temperature of 60°C are h_{fg} = 2359 × 10^3 J/kg and ρ_v = 0.1304 kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (60 + 40)/2 = 50$°C are (Table A-9)

$$\begin{aligned}
\rho_l &= 988.1\,\text{kg/m}^3 \\
\mu_l &= 0.547\times10^{-3}\,\text{kg/m}\cdot\text{s} \\
\nu_l &= \mu_l/\rho_l = 0.554\times10^{-6}\,\text{m}^2/\text{s} \\
c_{pl} &= 4181\,\text{J/kg}\cdot\text{K} \\
k_l &= 0.644\,\text{W/m}\cdot\text{K}
\end{aligned}$$

Analysis The modified latent heat of vaporization is

$$\begin{aligned}
h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) \\
&= 2359\times10^3\ \text{J/kg} + 0.68(4181\,\text{J/kg}\cdot\text{K})(60-40)\,\text{K} = 2.4159\times10^6\ \text{J/kg}
\end{aligned}$$

(*a*) The film condensation heat transfer coefficient is determined from

$$\begin{aligned}
h_{\text{film}} &= h_{\text{horiz}} = 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}}-T_s)D}\right]^{1/4} \\
&= 0.729\left[\frac{(9.81\,\text{m/s}^2)(988.1\,\text{kg/m}^3)(988.1-0.1304)\text{kg/m}^3(2.4159\times10^6\ \text{J/kg})(0.644\ \text{W/m}\cdot\text{K})^3}{(0.547\times10^{-3}\,\text{kg/m}\cdot\text{s})(60-40)\text{K}(0.025\,\text{m})}\right]^{1/4} \\
&= 8937.7\ \text{W/m}^2\cdot\text{K}
\end{aligned}$$

Thus, the rate of film condensation is

$$\begin{aligned}
\dot{m}_{\text{film}} &= \frac{\dot{Q}}{h^*_{fg}} = \frac{h_{\text{film}}(\pi DL)(T_{\text{sat}}-T_s)}{h^*_{fg}} \\
&= \frac{(8937.7\ \text{W/m}^2\cdot\text{K})\pi(0.025\,\text{m})(15\,\text{m})(60-40)\ \text{K}}{2.4159\times10^6\ \text{J/kg}} = \mathbf{0.08717\ kg/s}
\end{aligned}$$

(*b*) The dropwise condensation heat transfer coefficient, for 22°C < T_{sat} < 100°C, is determined from

$$\begin{aligned}
h_{\text{dropwise}} &= 51{,}104 + 2044T_{\text{sat}} \\
&= 51{,}104 + 2044(60°\text{C}) = 173{,}744\ \text{W/m}^2\cdot\text{K}
\end{aligned}$$

Thus, the rate of dropwise condensation is

$$\begin{aligned}
\dot{m}_{\text{dropwise}} &= \frac{\dot{Q}}{h^*_{fg}} = \frac{h_{\text{dropwise}}(\pi DL)(T_{\text{sat}}-T_s)}{h^*_{fg}} \\
&= \frac{(173{,}744\ \text{W/m}^2\cdot\text{K})\pi(0.025\,\text{m})(15\,\text{m})(60-40)\ \text{K}}{2.4159\times10^6\ \text{J/kg}} = \mathbf{1.695\ kg/s}
\end{aligned}$$

Discussion The dropwise condensation rate is 19.4 times greater than that of film condensation. This is because the convection heat transfer coefficient for dropwise condensation is greater than that of film condensation at the same factor, $h_{\text{dropwise}}/h_{\text{film}}$ = 19.4.

With dropwise condensation, there is no liquid film to impede heat transfer. Therefore, a much higher heat transfer coefficient can be achieved in dropwise condensation than in film condensation.

Special Topic: Non-Boiling Two-Phase Flow Heat Transfer

10-89 The flow quality of a non-boiling two-phase flow in a tube with $\dot{m}_l / \dot{m}_g = 300$ is to be determined.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Analysis The flow quality is given as

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g}$$

Hence, the equation can be rearranged as

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} = \frac{\dot{m}_g / \dot{m}_g}{(\dot{m}_l + \dot{m}_g)/\dot{m}_g} = \frac{1}{\dot{m}_l / \dot{m}_g + 1}$$

Thus, the flow quality is

$$x = \frac{1}{300 + 1} = \mathbf{0.00332}$$

Discussion The flow quality is a dimensionless parameter.

10-90 The flow quality and the mass flow rates of the gas and the liquid for a non-boiling two-phase flow, where $V_{sl} = 3V_{sg}$, are to be determined.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Properties The densities of the gas and liquid are given to be ρ_g = 8.5 kg/m^3 and ρ_l = 855 kg/m^3, respectively.

Analysis The mass flow rate of gas can be calculated using

$$\dot{m}_g = \rho_g V_{sg} A_c = \rho_g V_{sg} \pi \frac{D^2}{4}$$

$$= (8.5\ \text{kg/m}^3)(0.8\ \text{m/s})\pi \frac{(0.102\ \text{m})^2}{4} = \mathbf{0.0556\ kg/s}$$

Then, the mass flow rate of liquid is (with $V_{sl} = 3V_{sg}$)

$$\dot{m}_l = \rho_l V_{sl} A_c = 3\rho_l V_{sg} \pi \frac{D^2}{4}$$

$$= 3(855\ \text{kg/m}^3)(0.8\ \text{m/s})\pi \frac{(0.102\ \text{m})^2}{4} = \mathbf{16.8\ kg/s}$$

Thus, the flow quality is

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} = \frac{0.0556}{16.8 + 0.0556} = \mathbf{0.00330}$$

Discussion The total mass flow rate of gas and liquid for this two-phase flow is simply

$$\dot{m}_{\text{tot}} = \dot{m}_l + \dot{m}_g = 16.8\ \text{kg/s} + 0.0556\ \text{kg/s} = 16.86\ \text{kg/s}$$

10-91 The mass flow rate of air and the superficial velocities of air and engine oil for a non-boiling two-phase flow in a tube are to be determined.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Properties The densities of air and engine oil at the bulk mean temperature T_b = 140°C are ρ_g = 0.8542 kg/m^3 (Table A-15) and ρ_l = 816.8 kg/m^3 (Table A-13), respectively.

Analysis The flow quality is given as

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} = \frac{\dot{m}_g / \dot{m}_g}{(\dot{m}_l + \dot{m}_g)/\dot{m}_g} = \frac{1}{\dot{m}_l / \dot{m}_g + 1}$$

or

$$\frac{\dot{m}_l}{\dot{m}_g} + 1 = \frac{1}{x} \qquad \rightarrow \qquad \frac{\dot{m}_l}{\dot{m}_g} = \frac{1-x}{x}$$

With known liquid (engine oil) mass flow rate and flow quality, the gas (air) mass flow rate is determined using

$$\dot{m}_g = \frac{x}{1-x}\dot{m}_l = \frac{2.1\times10^{-3}}{1-2.1\times10^{-3}}(0.9\text{ kg/s}) = \mathbf{0.00189\ kg/s}$$

From the gas and liquid mass flow rates, the superficial gas and liquid velocities can be calculated:

$$V_{sg} = \frac{\dot{m}_g}{\rho_g A} = \frac{4\dot{m}_g}{\rho_g \pi D^2} = \frac{4(0.00189\text{ kg/s})}{(0.8542\text{ kg/m}^3)\pi(0.025\text{ m})^2} = \mathbf{4.51\ m/s}$$

$$V_{sl} = \frac{\dot{m}_l}{\rho_l A} = \frac{4\dot{m}_l}{\rho_l \pi D^2} = \frac{4(0.9\text{ kg/s})}{(816.8\text{ kg/m}^3)\pi(0.025\text{ m})^2} = \mathbf{2.25\ m/s}$$

Discussion The superficial velocity of air is twice that of the engine oil.

10-92 Starting with the two-phase non-boiling heat transfer correlation, the expression that is appropriate for the case when only water is flowing in the tube is to be determined.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Analysis The two-phase non-boiling heat transfer correlation is given as

$$h_{tp} = h_l F_p \left[1 + 0.55 \left(\frac{x}{1-x} \right)^{0.1} \left(\frac{1-F_p}{F_p} \right)^{0.4} \left(\frac{Pr_g}{Pr_l} \right)^{0.25} \left(\frac{\mu_l}{\mu_g} \right)^{0.25} \left(I^* \right)^{0.25} \right]$$

where

$$F_p = (1-\alpha) + \alpha \left[\frac{2}{\pi} \tan^{-1} \left(\sqrt{\frac{\rho_g (V_g - V_l)^2}{gD(\rho_l - \rho_g)}} \right) \right]^2 \qquad \text{and} \qquad x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g}$$

For the situation when the air flow is shut off and only water is flowing in the pipe, we have $\dot{m}_g = 0$ and $\alpha = 0$. Hence, we get

$$F_p = (1-0) + 0 = 1 \qquad \text{and} \qquad x = 0$$

Thus, the two-phase non-boiling heat transfer correlation becomes

$$h_{tp,\ \alpha=0} = h_l (1) \left[1 + 0.55 \left(\frac{0}{1-0} \right)^{0.1} \left(\frac{1-1}{1} \right)^{0.4} \left(\frac{Pr_g}{Pr_l} \right)^{0.25} \left(\frac{\mu_l}{\mu_g} \right)^{0.25} \left(I^* \right)^{0.25} \right] = h_l (1) \left[1 + 0 \right] = h_l$$

The liquid phase heat transfer coefficient is calculated using:

$$h_l = 0.027 \, \mathrm{Re}_l^{4/5} \, \mathrm{Pr}_l^{1/3} \left(\frac{k_l}{D} \right) \left(\frac{\mu_l}{\mu_s} \right)^{0.14}$$

where the *in situ* liquid Reynolds number is

$$\mathrm{Re}_l = \frac{4\dot{m}_l}{\pi \sqrt{1-\alpha}\, \mu_l D} = \frac{4\dot{m}_l}{\pi \sqrt{1-0}\, \mu_l D} = \frac{4\dot{m}_l}{\pi \mu_l D}$$

Therefore, we have

$$h_{tp,\ \alpha=0} = h_l = 0.027 \, \mathrm{Re}_l^{4/5} \, \mathrm{Pr}_l^{1/3} \left(\frac{k_l}{D} \right) \left(\frac{\mu_l}{\mu_s} \right)^{0.14}$$

Discussion When only water is flowing in the tube, the two-phase non-boiling heat transfer correlation is reduced to a familiar equation for internal forced convection.

10-93 Air-water slug flows through a 25.4-mm diameter horizontal tube in microgravity condition. Using the non-boiling two-phase heat transfer correlation, the two-phase heat transfer coefficient (h_{tp}) is to be determined

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Properties The properties of water (liquid) are given to be μ_l = 85.5×10^{-5} kg/m·s, μ_s = 73.9×10^{-5} kg/m·s, ρ_l = 997 kg/m^3, k_l = 0.613 W/m·K, and Pr$_l$ = 5.0. The properties of air (gas) are given to be μ_g = 18.5×10^{-6} kg/m·s, ρ_g = 1.16 kg/m^3, and Pr$_g$ = 0.71.

Analysis From the superficial gas and liquid velocities, and void fraction, the gas and liquid velocities can be calculated as

$$V_g = \frac{V_{sg}}{\alpha} = \frac{0.3\ \text{m/s}}{0.27} = 1.11\ \text{m/s}$$

$$V_l = \frac{V_{sl}}{1-\alpha} = \frac{0.544\ \text{m/s}}{1-0.27} = 0.745\ \text{m/s}$$

The gas and liquid mass flow rates are calculated as

$$\dot{m}_g = \rho_g V_{sg} A_c = \rho_g V_{sg} \pi \frac{D^2}{4} = (1.16\ \text{kg/m}^3)(0.3\ \text{m/s})\pi \frac{(0.0254\ \text{m})^2}{4} = 1.76\times10^{-4}\ \text{kg/s}$$

$$\dot{m}_l = \rho_l V_{sl} A_c = \rho_l V_{sl} \pi \frac{D^2}{4} = (997\ \text{kg/m}^3)(0.544\ \text{m/s})\pi \frac{(0.0254\ \text{m})^2}{4} = 0.275\ \text{kg/s}$$

Using the gas and liquid mass flow rates, the quality is determined to be

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} = \frac{1.76\times10^{-4}}{0.275 + 1.76\times10^{-4}} = 6.40\times10^{-4}$$

The flow pattern factor (F_p) can be calculated using

$$F_p = (1-\alpha) + \alpha\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{\rho_g (V_g - V_l)^2}{gD(\rho_l - \rho_g)}}\right)\right]^2$$

$$= (1-0.27) + (0.27)\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{(1.16\ \text{kg/m}^3)(1.11\ \text{m/s} - 0.745\ \text{m/s})^2}{(9.81\ \text{m/s}^2)(0.0254\ \text{m})(997\ \text{kg/m}^3 - 1.16\ \text{kg/m}^3)}}\right)\right]^2 = 0.730$$

The liquid phase heat transfer coefficient is calculated using:

$$h_l = 0.027\,\text{Re}_l^{4/5}\,\text{Pr}_l^{1/3}\left(\frac{k_l}{D}\right)\left(\frac{\mu_l}{\mu_s}\right)^{0.14}$$

$$= 0.027(18870)^{4/5}(5.0)^{1/3}\left(\frac{0.613\ \text{W/m}\cdot\text{K}}{0.0254\ \text{m}}\right)\left(\frac{85.5\times10^{-5}\ \text{kg/m}\cdot\text{s}}{73.9\times10^{-5}\ \text{kg/m}\cdot\text{s}}\right)^{0.14} = 2995\ \text{W/m}^2\cdot\text{K}$$

where the *in situ* liquid Reynolds number is

$$\text{Re}_l = \frac{4\dot{m}_l}{\pi\sqrt{1-\alpha}\,\mu_l D} = \frac{4(0.275\ \text{kg/s})}{\pi\sqrt{1-0.27}(85.5\times10^{-5}\ \text{kg/m}\cdot\text{s})(0.0254\ \text{m})} = 18870$$

The inclination factor (I^*) has a value of one for horizontal tube (θ = 0). Thus, using the general two-phase heat transfer correlation, the value for h_{tp} is estimated to be

$$\frac{h_{tp}}{h_l} = F_p\left[1 + 0.55\left(\frac{x}{1-x}\right)^{0.1}\left(\frac{1-F_p}{F_p}\right)^{0.4}\left(\frac{Pr_g}{Pr_l}\right)^{0.25}\left(\frac{\mu_l}{\mu_g}\right)^{0.25}\left(I^*\right)^{0.25}\right]$$

$$= (0.730)\left[1 + 0.55\left(\frac{6.40\times10^{-4}}{1-6.40\times10^{-4}}\right)^{0.1}\left(\frac{1-0.730}{0.730}\right)^{0.4}\left(\frac{0.71}{5.0}\right)^{0.25}\left(\frac{85.5\times10^{-5}\ \text{kg/m}\cdot\text{s}}{18.5\times10^{-6}\ \text{kg/m}\cdot\text{s}}\right)^{0.25}\right] = 0.9369$$

or $\quad h_{tp} = 0.9369 h_l = 0.9369(2995\ \text{W/m}^2\cdot\text{K}) = \mathbf{2810\ W/m^2\cdot K}$

Discussion The non-boiling two-phase heat transfer coefficient (h_{tp}) is about 7% lower than the liquid phase heat transfer coefficient (h_l).

10-94 A two-phase flow of air and silicone (Dow Corning 200® Fluid, 5 cs) is being transported in an 11.7-mm diameter horizontal tube, where condensation occurs on the tube outer surface. The overall convection heat transfer coefficient is to be determined.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow inside tube is non-boiling and does not involve phase change. **3** Fluid properties are constant. **4** The thermal resistance of the tube wall is negligible. **5** Isothermal tube surface. **6** Film condensation occurs on the tube outer surface.

Properties Inside the tube: The properties of liquid silicone (liquid) are given to be $\mu_l = 45.7\times10^{-4}$ kg/m·s, $\rho_l = 913$ kg/m^3, $k_l = 0.117$ W/m·K, $\sigma = 19.7\times10^{-3}$ N/m and $\Pr_l = 64$. The properties of air (gas) are given to be $\mu_g = 18.4\times10^{-6}$ kg/m·s, $\rho_g = 1.19$ kg/m^3, $\Pr_g = 0.71$.

Outside the tube: The properties of water at the saturation temperature of 40°C are $h_{fg} = 2407\times10^3$ J/kg and $\rho_v = 0.0512$ kg/m^3 (Table A-9). The properties of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (40 + 20)/2 = 30$°C are (Table A-9),

$$\begin{aligned}
\rho_{l,f} &= 996.0\ \text{kg/m}^3\\
\mu_{l,f} &= 0.798\times10^{-3}\ \text{kg/m}\cdot\text{s}\\
\nu_{l,f} &= \mu_l/\rho_l = 0.801\times10^{-6}\ \text{m}^2/\text{s}\\
c_{pl,f} &= 4178\ \text{J/kg}\cdot\text{K}\\
k_{l,f} &= 0.615\ \text{W/m}\cdot\text{K}\\
\Pr_f &= 5.42
\end{aligned}$$

Analysis Inside the tube: The flow pattern factor (F_P) can be calculated using

$$F_P = (1-\alpha) + \alpha\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{\rho_g(V_g - V_l)^2}{gD(\rho_l - \rho_g)}}\right)\right]^2$$

$$= (1-0.011) + (0.011)\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{(1.19\ \text{kg/m}^3)(13.5\ \text{m/s} - 9.34\ \text{m/s})^2}{(9.81\ \text{m/s}^2)(0.0117\ \text{m})(913\ \text{kg/m}^3 - 1.19\ \text{kg/m}^3)}}\right)\right]^2$$

$$= 0.9898$$

The inclination factor (I^*) for horizontal tube ($\theta = 0°$) is calculated to be

$$I^* = 1 + \frac{(\rho_i - \rho_g)gD^2}{\sigma}|\sin\theta| = 1$$

The liquid phase heat transfer coefficient is calculated using the Seder and Tate (1936) equation:

$$h_l = 0.027\,\text{Re}_l^{0.8}\,\Pr^{1/3}\left(\frac{\mu_l}{\mu_s}\right)^{0.14}\left(\frac{k_l}{D}\right)$$

$$= 0.027(21718)^{0.8}(64)^{1/3}\left(\frac{45.7}{39.8}\right)^{0.14}\left(\frac{0.117\ \text{W/m}\cdot\text{K}}{0.0117\ \text{m}}\right)$$

$$= 3246\ \text{W/m}^2\cdot\text{K}$$

where the *in situ* liquid Reynolds number is

$$\text{Re}_l = \frac{4\dot{m}_l}{\pi\sqrt{1-\alpha}\,\mu_l D} = \frac{4(0.907\ \text{kg/s})}{\pi\sqrt{1-0.011}(45.7\times10^{-4}\ \text{kg/m}\cdot\text{s})(0.0117\ \text{m})} = 21718$$

Using the general two-phase heat transfer correlation, Eq. (10-38), the two-phase heat transfer coefficient (h_{tp}) is estimated to be

$$h_{tp} = h_l F_P \left[1 + 0.55 \left(\frac{x}{1-x} \right)^{0.1} \left(\frac{1-F_P}{F_P} \right)^{0.4} \left(\frac{Pr_g}{Pr_l} \right)^{0.25} \left(\frac{\mu_l}{\mu_g} \right)^{0.25} \left(I^* \right)^{0.25} \right]$$

$$= (3246 \text{ W/m}^2 \cdot \text{K})(0.9898) \left[1 + 0.55 \left(\frac{2.08 \times 10^{-5}}{1 - 2.08 \times 10^{-5}} \right)^{0.1} \left(\frac{1 - 0.9898}{0.9898} \right)^{0.4} \left(\frac{0.71}{64} \right)^{0.25} \left(\frac{45.7 \times 10^{-4}}{18.4 \times 10^{-6}} \right)^{0.25} \right]$$

$$h_i = h_{tp} = 3337 \text{ W/m}^2 \cdot \text{K}$$

Outside the tube: The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68 c_{pl,f} (T_{\text{sat}} - T_s) = 2407 \times 10^3 \text{ J/kg} + 0.68 \times (4182 \text{ J/kg} \cdot \text{K})(40 - 20)\text{K} = 2464 \times 10^3 \text{ J/kg}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$h_o = h_{\text{horizontal}} = 0.729 \left[\frac{g \rho_{l,f} (\rho_{l,f} - \rho_v) h^*_{fg} k^3_{l,f}}{\mu_{l,f} (T_{\text{sat}} - T_s) D} \right]^{1/4}$$

$$= 0.729 \left[\frac{(9.81 \text{ m/s}^2)(996 \text{ kg/m}^3)(996 - 0.05 \text{ kg/m}^3)(2464 \times 10^3 \text{ J/kg})(0.615 \text{ W/m} \cdot \text{K})^3}{(0.798 \times 10^{-3} \text{ kg/m} \cdot \text{s})(40 - 20)\text{K}(0.0117 \text{ m})} \right]^{1/4}$$

$$= 9584 \text{ W/m}^2 \cdot \text{K}$$

Noting that the thermal resistance of the tube is negligible, the overall heat transfer coefficient becomes

$$U = \frac{1}{1/h_i + 1/h_o} = \frac{1}{1/3337 + 1/9584} = \mathbf{2475 \ W/m^2 \cdot K}$$

Discussion The condensation heat transfer coefficient is almost 3 times higher than the non-boiling heat transfer coefficient. This is expected, as heat transfer coefficient involving phase change is much higher than that without phase change.

10-95 Air-water mixture is flowing in a 5° inclined tube with diameter of 25.4 mm, and the mixture superficial gas and liquid velocities are 1 m/s and 2 m/s, respectively. The two-phase heat transfer coefficient (h_{tp}) is to be determined.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Properties The properties of water (liquid) at bulk mean temperature $T_b = (T_i + T_e)/2 = 45°C$ are, from Table A-9, $\mu_l = 0.596\times10^{-3}$ kg/m·s, $\rho_l = 990.1$ kg/m^3, $k_l = 0.637$ W/m·K, and $\Pr_l = 3.91$. The properties of air (gas) at bulk mean temperature $T_b = 45°C$ are, from Table A-15, $\mu_g = 1.941\times10^{-5}$ kg/m·s, $\rho_g = 1.109$ kg/m^3, and $\Pr_g = 0.7241$. Also, at $T_s = 80°C$ we get $\mu_s = 0.355\times10^{-3}$ kg/m·s from Table A-9.

Analysis From the superficial gas and liquid velocities, and void fraction, the gas and liquid velocities can be calculated as

$$V_g = \frac{V_{sg}}{\alpha} = \frac{1\text{ m/s}}{0.33} = 3.030\text{ m/s} \qquad V_l = \frac{V_{sl}}{1-\alpha} = \frac{2\text{ m/s}}{1-0.33} = 2.985\text{ m/s}$$

The gas and liquid mass flow rates are calculated as

$$\dot{m}_g = \rho_g V_{sg} A_c = \rho_g V_{sg} \pi \frac{D^2}{4} = (1.109\text{ kg/m}^3)(1\text{ m/s})\pi \frac{(0.0254\text{ m})^2}{4} = 5.619\times10^{-4}\text{ kg/s}$$

$$\dot{m}_l = \rho_l V_{sl} A_c = \rho_l V_{sl} \pi \frac{D^2}{4} = (990.1\text{ kg/m}^3)(2\text{ m/s})\pi \frac{(0.0254\text{ m})^2}{4} = 1.003\text{ kg/s}$$

Using the gas and liquid mass flow rates, the quality is determined to be

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} = \frac{5.619\times10^{-4}}{1.003 + 5.619\times10^{-4}} = 5.599\times10^{-4}$$

The flow pattern factor (F_p) can be calculated using

$$F_p = (1-\alpha) + \alpha\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{\rho_g (V_g - V_l)^2}{gD(\rho_l - \rho_g)}}\right)\right]^2$$

$$= (1-0.33) + (0.33)\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{(1.109\text{ kg/m}^3)(3.03\text{ m/s} - 2.985\text{ m/s})^2}{(9.81\text{ m/s}^2)(0.0254\text{ m})(990.1\text{ kg/m}^3 - 1.109\text{ kg/m}^3)}}\right)\right]^2 = 0.670$$

The inclination factor (I^*) for $\theta = 5°$ is calculated to be

$$I^* = 1 + \frac{(\rho_l - \rho_g)gD^2}{\sigma}|\sin\theta| = 1 + \frac{(990.1\text{ kg/m}^3 - 1.109\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.0254\text{ m})^2}{0.068\text{ N/m}}|\sin 5°| = 9.023$$

The liquid phase heat transfer coefficient is calculated using:

$$h_l = 0.027\,\mathrm{Re}_l^{4/5}\,\Pr_l^{1/3}\left(\frac{k_l}{D}\right)\left(\frac{\mu_l}{\mu_s}\right)^{0.14}$$

$$= 0.027(103100)^{4/5}(3.91)^{1/3}\left(\frac{0.637\text{ W/m}\cdot\text{K}}{0.0254\text{ m}}\right)\left(\frac{0.596\times10^{-3}\text{ kg/m}\cdot\text{s}}{0.355\times10^{-3}\text{ kg/m}\cdot\text{s}}\right)^{0.14} = 11754\text{ W/m}^2\cdot\text{K}$$

where the *in situ* liquid Reynolds number is

$$\mathrm{Re}_l = \frac{4\dot{m}_l}{\pi\sqrt{1-\alpha}\,\mu_l D} = \frac{4(1.003\text{ kg/s})}{\pi\sqrt{1-0.33}(0.596\times10^{-3}\text{ kg/m}\cdot\text{s})(0.0254\text{ m})} = 103100$$

Thus, using the general two-phase heat transfer correlation, the value for h_{tp} is estimated to be

$$\frac{h_{tp}}{h_l} = F_p\left[1 + 0.55\left(\frac{x}{1-x}\right)^{0.1}\left(\frac{1-F_p}{F_p}\right)^{0.4}\left(\frac{Pr_g}{Pr_l}\right)^{0.25}\left(\frac{\mu_l}{\mu_g}\right)^{0.25}\left(I^*\right)^{0.25}\right]$$

$$= (0.670)\left[1 + 0.55\left(\frac{5.599\times10^{-4}}{1-5.599\times10^{-4}}\right)^{0.1}\left(\frac{1-0.670}{0.670}\right)^{0.4}\left(\frac{0.7241}{3.91}\right)^{0.25}\left(\frac{59.6}{1.941}\right)^{0.25}(9.023)^{0.25}\right] = 1.021$$

or $\quad h_{tp} = 1.021 h_l = 1.021(11754\text{ W/m}^2\cdot\text{K}) = \mathbf{12{,}000\ W/m^2\cdot K}$

Discussion The inclination factor is $I^* = 1$ when the tube is at horizontal position, since sin(0°) = 0. The two-phase heat transfer coefficient for horizontal tube would be $h_{tp} = 10300$ W/m^2·K, which is about 14% lower than that of 5° inclined tube.

10-96 Mixture of petroleum and natural gas is being transported in a pipeline that is located in a terrain that caused it to have an average inclination angle of 10°. The two-phase heat transfer coefficient is to be determined.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Properties The properties of petroleum (liquid) are given to be $\mu_l = 297.5\times10^{-4}$ kg/m·s, $\mu_s = 238\times10^{-4}$ kg/m·s, $\rho_l = 853$ kg/m^3, $k_l = 0.163$ W/m·K, $\sigma = 0.020$ N/m, and $\mathrm{Pr}_l = 405$. The properties of natural gas are given to be $\mu_g = 9.225\times10^{-6}$ kg/m·s, $\rho_g = 9.0$ kg/m^3, and $\mathrm{Pr}_g = 0.80$.

Analysis From the gas and liquid mass flow rates, the superficial gas and liquid velocities can be calculated:

$$V_{sg} = \frac{\dot{m}_g}{\rho_g A} = \frac{4\dot{m}_g}{\rho_g \pi D^2} = \frac{4(0.055\ \text{kg/s})}{(9.0\ \text{kg/m}^3)\pi(0.102\ \text{m})^2} = 0.748\ \text{m/s}$$

$$V_{sl} = \frac{\dot{m}_l}{\rho_l A} = \frac{4\dot{m}_l}{\rho_l \pi D^2} = \frac{4(16\ \text{kg/s})}{(853\ \text{kg/m}^3)\pi(0.102\ \text{m})^2} = 2.296\ \text{m/s}$$

Using the superficial velocities and void fraction, the gas and liquid velocities are found to be

$$V_g = \frac{V_{sg}}{\alpha} = \frac{0.748\ \text{m/s}}{0.22} = 3.400\ \text{m/s} \qquad V_l = \frac{V_{sl}}{1-\alpha} = \frac{2.296\ \text{m/s}}{1-0.22} = 2.944\ \text{m/s}$$

Using the gas and liquid mass flow rates, the quality is determined to be

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} = \frac{0.055}{16+0.055} = 3.426\times10^{-3}$$

The flow pattern factor (F_p) can be calculated using

$$F_p = (1-\alpha) + \alpha\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{\rho_g\,(V_g - V_l)^2}{gD\,(\rho_l - \rho_g)}}\right)\right]^2$$

$$= (1-0.22) + (0.22)\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{(9.0\ \text{kg/m}^3)(3.40\ \text{m/s} - 2.944\ \text{m/s})^2}{(9.81\ \text{m/s}^2)(0.102\ \text{m})(853\ \text{kg/m}^3 - 9.0\ \text{kg/m}^3)}}\right)\right]^2 = 0.7802$$

The liquid phase heat transfer coefficient is calculated using:

$$h_l = 0.027\,\mathrm{Re}_l^{4/5}\,\mathrm{Pr}_l^{1/3}\left(\frac{k_l}{D}\right)\left(\frac{\mu_l}{\mu_s}\right)^{0.14}$$

$$= 0.027(7601)^{4/5}(405)^{1/3}\left(\frac{0.163\ \text{W/m·K}}{0.102\ \text{m}}\right)\left(\frac{297.5\times10^{-4}\ \text{kg/m}\cdot\text{s}}{238\times10^{-4}\ \text{kg/m}\cdot\text{s}}\right)^{0.14} = 419.1\ \text{W/m}^2\cdot\text{K}$$

where the *in situ* liquid Reynolds number is

$$\mathrm{Re}_l = \frac{4\dot{m}_l}{\pi\sqrt{1-\alpha}\,\mu_l D} = \frac{4(16\ \text{kg/s})}{\pi\sqrt{1-0.22}(297.5\times10^{-4}\ \text{kg/m}\cdot\text{s})(0.102\ \text{m})} = 7601$$

The inclination factor (I^*) for $\theta = 10°$ is calculated to be

$$I^* = 1 + \frac{(\rho_l - \rho_g)gD^2}{\sigma}|\sin\theta| = 1 + \frac{(853\ \text{kg/m}^3 - 9.0\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.102\ \text{m})^2}{0.020\ \text{N/m}}|\sin 10°| = 748.9$$

Thus, using the general two-phase heat transfer correlation, the value for h_{tp} is estimated to be

$$\frac{h_{tp}}{h_l} = F_p\left[1 + 0.55\left(\frac{x}{1-x}\right)^{0.1}\left(\frac{1-F_p}{F_p}\right)^{0.4}\left(\frac{Pr_g}{Pr_l}\right)^{0.25}\left(\frac{\mu_l}{\mu_g}\right)^{0.25}\left(I^*\right)^{0.25}\right]$$

$$= (0.7802)\left[1 + 0.55\left(\frac{3.426\times10^{-3}}{1-3.426\times10^{-3}}\right)^{0.1}\left(\frac{1-0.7802}{0.7802}\right)^{0.4}\left(\frac{0.80}{405}\right)^{0.25}\left(\frac{29750}{9.225}\right)^{0.25}(748.9)^{0.25}\right] = 1.999$$

or $\qquad h_{tp} = 1.999 h_l = 1.999(419.1\ \text{W/m}^2\cdot\text{K}) = \mathbf{838\ W/m^2\cdot K}$

Discussion Since $V_g > V_l$, there will be slippage between the gas and liquid phases. When $V_g \neq V_l$, slippage between the gas and liquid phases exists. When $V_g = V_l$, slippage between the gas and liquid phases is negligible, and the flow is called homogeneous two-phase flow.

10-97 Air-water mixture is flowing in a tube with diameter of 25.4 mm, and the mixture superficial gas and liquid velocities are 1 m/s and 2 m/s, respectively. The two-phase heat transfer coefficient (h_{tp}) for (*a*) horizontal tubee (θ = 0°) and (*b*) vertical tube (θ = 90°), are to be determined and compared.

Assumptions **1** Steady operating condition exists. **2** Two-phase flow is non-boiling and it does not involve phase change. **3** Fluid properties are constant.

Properties The properties of water (liquid) at bulk mean temperature $T_b = (T_i + T_e)/2$ = 45°C are, from Table A-9, μ_l = 0.596×10^{-3} kg/m·s, ρ_l = 990.1 kg/m^3, k_l = 0.637 W/m·K, and Pr$_l$ = 3.91. The properties of air (gas) at bulk mean temperature T_b = 45°C are, from Table A-15, μ_g = 1.941×10^{-5} kg/m·s, ρ_g = 1.109 kg/m^3, and Pr$_g$ = 0.7241. Also, at T_s = 80°C we get μ_s = 0.355×10^{-3} kg/m·s from Table A-9.

Analysis From the superficial gas and liquid velocities, and void fraction, the gas and liquid velocities can be calculated as

$$V_g = \frac{V_{sg}}{\alpha} = \frac{1\text{ m/s}}{0.33} = 3.030\text{ m/s}$$

$$V_l = \frac{V_{sl}}{1-\alpha} = \frac{2\text{ m/s}}{1-0.33} = 2.985\text{ m/s}$$

The gas and liquid mass flow rates are calculated as

$$\dot{m}_g = \rho_g V_{sg} A_c = \rho_g V_{sg} \pi \frac{D^2}{4} = (1.109\text{ kg/m}^3)(1\text{ m/s})\pi \frac{(0.0254\text{ m})^2}{4} = 5.619\times10^{-4}\text{ kg/s}$$

$$\dot{m}_l = \rho_l V_{sl} A_c = \rho_l V_{sl} \pi \frac{D^2}{4} = (990.1\text{ kg/m}^3)(2\text{ m/s})\pi \frac{(0.0254\text{ m})^2}{4} = 1.003\text{ kg/s}$$

Using the gas and liquid mass flow rates, the quality is determined to be

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} = \frac{5.619\times10^{-4}}{1.003 + 5.619\times10^{-4}} = 5.599\times10^{-4}$$

The flow pattern factor (F_p) can be calculated using

$$F_p = (1-\alpha) + \alpha\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{\rho_g (V_g - V_l)^2}{gD(\rho_l - \rho_g)}}\right)\right]^2$$

$$= (1-0.33) + (0.33)\left[\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{(1.109\text{ kg/m}^3)(3.03\text{ m/s} - 2.985\text{ m/s})^2}{(9.81\text{ m/s}^2)(0.0254\text{ m})(990.1\text{ kg/m}^3 - 1.109\text{ kg/m}^3)}}\right)\right]^2$$

$$= 0.670$$

The liquid phase heat transfer coefficient is calculated using:

$$h_l = 0.027\,\text{Re}_l^{4/5}\,\text{Pr}_l^{1/3}\left(\frac{k_l}{D}\right)\left(\frac{\mu_l}{\mu_s}\right)^{0.14}$$

$$= 0.027(103100)^{4/5}(3.91)^{1/3}\left(\frac{0.637\text{ W/m}\cdot\text{K}}{0.0254\text{ m}}\right)\left(\frac{0.596\times10^{-3}\text{ kg/m}\cdot\text{s}}{0.355\times10^{-3}\text{ kg/m}\cdot\text{s}}\right)^{0.14}$$

$$= 11754\text{ W/m}^2\cdot\text{K}$$

where the *in situ* liquid Reynolds number is

$$\text{Re}_l = \frac{4\dot{m}_l}{\pi\sqrt{1-\alpha}\,\mu_l D} = \frac{4(1.003\text{ kg/s})}{\pi\sqrt{1-0.33}(0.596\times10^{-3}\text{ kg/m}\cdot\text{s})(0.0254\text{ m})} = 103100$$

(*a*) The inclination factor (I^*) for horizontal tube (θ = 0°) is $I^* = 1$. Thus, using the general two-phase heat transfer correlation, the value for h_{tp} is estimated to be

$$\frac{h_{tp,\,\mathrm{horiz}}}{h_l} = F_p\left[1+0.55\left(\frac{x}{1-x}\right)^{0.1}\left(\frac{1-F_p}{F_p}\right)^{0.4}\left(\frac{Pr_g}{Pr_l}\right)^{0.25}\left(\frac{\mu_l}{\mu_g}\right)^{0.25}\left(I^*\right)^{0.25}\right]$$
$$= (0.670)\left[1+0.55\left(\frac{5.599\times10^{-4}}{1-5.599\times10^{-4}}\right)^{0.1}\left(\frac{1-0.670}{0.670}\right)^{0.4}\left(\frac{0.7241}{3.91}\right)^{0.25}\left(\frac{59.6}{1.941}\right)^{0.25}\right]$$
$$= 0.8727$$

or

$$h_{tp,\,\mathrm{horiz}} = 0.87271 h_l = 0.8727(11754\ \mathrm{W/m^2\cdot K}) = \mathbf{10{,}300\ W/m^2\cdot K}$$

(*b*) The inclination factor (I^*) for vertical tube (θ = 90°) is calculated to be

$$I^* = 1+\frac{(\rho_l-\rho_g)gD^2}{\sigma}\left|\sin\theta\right|$$
$$= 1+\frac{(990.1\ \mathrm{kg/m^3}-1.109\ \mathrm{kg/m^3})(9.81\ \mathrm{m/s^2})(0.0254\ \mathrm{m})^2}{0.068\ \mathrm{N/m}}\left|\sin 90°\right|$$
$$= 93.05$$

Thus, using the general two-phase heat transfer correlation, the value for h_{tp} is estimated to be

$$\frac{h_{tp,\,\mathrm{vert}}}{h_l} = F_p\left[1+0.55\left(\frac{x}{1-x}\right)^{0.1}\left(\frac{1-F_p}{F_p}\right)^{0.4}\left(\frac{Pr_g}{Pr_l}\right)^{0.25}\left(\frac{\mu_l}{\mu_g}\right)^{0.25}\left(I^*\right)^{0.25}\right]$$
$$= (0.670)\left[1+0.55\left(\frac{5.599\times10^{-4}}{1-5.599\times10^{-4}}\right)^{0.1}\left(\frac{1-0.670}{0.670}\right)^{0.4}\left(\frac{0.7241}{3.91}\right)^{0.25}\left(\frac{59.6}{1.941}\right)^{0.25}(93.05)^{0.25}\right]$$
$$= 1.30$$

or

$$h_{tp,\,\mathrm{vert}} = 1.30 h_l = 1.30(11754\ \mathrm{W/m^2\cdot K}) = \mathbf{15{,}300\ W/m^2\cdot K}$$

Discussion The two-phase heat transfer coefficient of vertical pipe is about 49% higher than that of horizontal pipe:

$$\frac{h_{tp,\,\mathrm{vert}}}{h_{tp,\,\mathrm{horiz}}} = \frac{15{,}300}{10{,}300} = 1.49$$

10-98 Air-water mixture flows through a 0.0254 m stainless steel pipe at specified flow condition. Using the concept of Reynolds analogy, the two phase convective heat transfer coefficient is to be determined.

Assumptions **1** Steady state operating conditions exist. **2** Two phase flow is non-boiling in nature and does not undergo any phase change. **3** Fluid properties are constant.

Properties The properties of water and air are calculated at a system temperature and pressure of 25°C and 201 kPa using EES. The thermo physical properties of water and air are,

$$\rho_l = 997.1\,\text{kg/m}^3,\ \mu_l = 8.9\times10^{-4}\ \text{kg/m.s},Pr_l = 6.26,\ \text{k}_l = 0.595\,\text{W/m}\cdot\text{K}\ \ \text{and}\ \sigma = 0.0719\ \text{N/m}$$
$$\rho_g = 2.35\,\text{kg/m}^3,\ \mu_g = 1.84\times10^{-5}\ \text{kg/m.s}$$

Analysis We first need to determine superficial Reynolds number and the actual velocities for each phase.

$$Re_{sl} = \frac{\rho_l V_{sl} D}{\mu_l} = \frac{997.1(\text{kg/m}^3)\times 0.3(\text{m/s})\times 0.0254(\text{m})}{8.9\times10^{-4}(\text{kg/m}\cdot\text{s})} = 8537$$

$$Re_{sg} = \frac{\rho_g V_{sg} D}{\mu_g} = \frac{2.35(\text{kg/m}^3)\times 23(\text{m/s})\times 0.0254(\text{m})}{1.84\times10^{-5}(\text{kg/m}\cdot\text{s})} = 74612$$

The actual phase velocities are calculated from the known values of superficial velocities and void fraction.

$$V_l = \frac{V_{sl}}{1-\alpha} = \frac{0.3(\text{m/s})}{1-0.86} = 2.14\,\text{m/s}\ \text{ and }\ V_g = \frac{V_{sg}}{\alpha} = \frac{23(\text{m/s})}{0.86} = 26.74\,\text{m/s}$$

The mass flow rate of each phase is,

$$\dot{m}_l = \rho_l V_{sl}\frac{\pi}{4}D^2 = 997.1(\text{kg/m}^3)\times 0.3(\text{m/s})\times\frac{\pi}{4}\times 0.0254^2(\text{m}^2) = 0.15\,\text{kg/s}$$

$$\dot{m}_g = \rho_g V_{sg}\frac{\pi}{4}D^2 = 2.35(\text{kg/m}^3)\times 23(\text{m/s})\times\frac{\pi}{4}\times 0.0254^2(\text{m}^2) = 0.027\,\text{kg/s}$$

Thus the total mass flow rate is, $\dot{m} = \dot{m}_l + \dot{m}_g = 0.15 + 0.027 = 0.177\ \text{kg/s}$

Since this is a vertical upward flow of air water we can use the Reynolds analogy given by Equation (10-40).

$$\frac{h_{tp}}{h_l} = F_p{}^m\left(\frac{\dot{m}_l}{\dot{m}}\right)\left(\frac{\rho_{tp}}{\rho_l}\right)^n \phi_l{}^p$$

The single phase heat transfer coefficient is calculated as,

$$h_l = 0.027\,\text{Re}_l^{4/5}\,\text{Pr}_l^{1/3}\,(k_l/D)\,(\mu_l/\mu_s)^{0.14}$$

The in-situ Reynolds number required in single phase heat transfer equation is calcutaed as,

$$\text{Re}_l = \frac{4\dot{m}_l}{\mu_l D\pi\sqrt{1-\alpha}} = \frac{4\times 0.15(\text{kg/s})}{8.9\times10^{-4}(\text{kg/m.s})\times 0.0254(\text{m})\times\pi\sqrt{1-0.86}} = 22579$$

$$h_l = 0.027\times 22579^{4/5}\times 6.26^{1/3}\times\frac{0.595(\text{W/m}\cdot\text{K})}{0.0254(\text{m})}\times\left(\frac{8.9\times10^{-4}}{4.66\times10^{-4}}\right)^{0.14} = 3878\ \text{W/m}^2\cdot\text{K}$$

$$\text{F}_\text{p} = (1-\alpha)+\alpha\left(\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{\rho_\text{g}(\text{V}_\text{g}-\text{V}_\text{l})^2}{\text{gD}(\rho_\text{l}-\rho_\text{g})}}\right)\right)^2$$

$$\text{F}_\text{p} = (1-0.86)+0.86\left(\frac{2}{\pi}\tan^{-1}\left(\sqrt{\frac{2.35(\text{kg/m}^3)(26.74-2.14)^2}{9.81\times 0.0254(\text{m})\times(997.1-2.35)(\text{kg/m}^3)}}\right)\right)^2$$

$$\therefore\ \text{F}_\text{p} = 0.621$$

The two phase density required in Reynolds analogy equation is calculated as,

$$\rho_{tp} = (1-\alpha)\rho_l + \alpha\rho_g = (1-0.86)\times 997.1(\text{kg/m}^3) + 0.86\times 2.35(\text{kg/m}^3) = 141.6\,\text{kg/m}^3$$

Single phase pressure drop for turbulent pipe flow (with superficial Reynolds number of Re_{sl} = 8537) is calculated by first calculating the single phase friction factor from either the Mood chart (Fig. A-20) or the Colebrook equation (Eq.8-76) as follows,

$$\frac{1}{\sqrt{f_l}} = -2\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}_{sl}\sqrt{f_l}}\right) = -2\log\left(\frac{0.002/25.4}{3.7} + \frac{2.51}{8537\sqrt{f_l}}\right) = 0.0323$$

Where from Table 8-3 for stainless steel pipe, the roughness is ε = 0.002.

Single phase pressure drop is then calculated as,

$$(dP/dL)_{f,l} = \frac{f_l\rho_l V_{sl}^2}{2D} = \frac{0.0323\times 997.1(\text{kg/m}^3)\times 0.3^2(\text{m}^2)}{2\times 0.0254(\text{m})} = 57.06\,\text{Pa/m}$$

Thus the two-phase friction multiplier is,

$$\phi_l = \sqrt{\frac{(dP/dL)_{f,tp}}{(dP/dL)_{f,l}}} = \sqrt{\frac{2700(\text{Pa/m})}{57.06(\text{Pa/m})}} = 6.87$$

Thus the two-phase heat transfer coefficient calculated using Reynolds analogy (Eq. 10-40) is,

$$\frac{\mathrm{h_{tp}}}{\mathrm{h_l}} = \mathrm{F_p}^{\mathrm{m}}\left(\frac{\dot{\mathrm{m}}_\mathrm{l}}{\dot{\mathrm{m}}}\right)\left(\frac{\rho_{\mathrm{tp}}}{\rho_\mathrm{l}}\right)^{\mathrm{n}}\phi_\mathrm{l}^{\mathrm{p}} = 0.621^{0.5}\left(\frac{0.15}{0.177}\right)\left(\frac{141.6(\text{kg/m}^3)}{997.1(\text{kg/m}^3)}\right)^{-0.5} 6.87^{0.2}$$

$$\frac{\mathrm{h_{tp}}}{\mathrm{h_l}} = 2.605$$

Thus the two phase heat transfer coefficient is: h_{tp} = **10102 W/m²·K**

Discussion The use of Reynolds analogy strongly depends on the correct estimation of the two-phase pressure drop. Most of the correlations available for two-phase pressure drop fail to correctly estimate the two-phase pressure drop and hence based on the calculated value of two-phase pressure drop, the Reynolds analogy may not predict the two- phase heat transfer coefficient correctly.

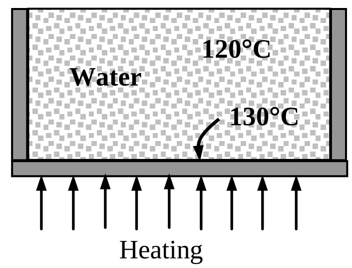
Review Problems

10-99 Water is boiled at T_{sat} = 120°C in a mechanically polished stainless steel pressure cooker whose inner surface temperature is maintained at T_s = 130°C. The time it will take for the tank to empty is to be determined.

Assumptions 1 Steady operating conditions exist. **2** Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 120°C are (Tables 10-1 and A-9)

$$\rho_l = 943.4 \text{ kg/m}^3 \quad h_{fg} = 2203\times10^3 \text{ J/kg}$$
$$\rho_v = 1.121 \text{ kg/m}^3 \quad \mu_l = 0.232\times10^{-3} \text{ kg/m}\cdot\text{s}$$
$$\sigma = 0.0550 \text{ N/m} \quad c_{pl} = 4244 \text{ J/kg}\cdot°\text{C}$$
$$\text{Pr}_l = 1.44$$



Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis The excess temperature in this case is $\Delta T = T_s - T_{sat} = 130 - 120 = 10°\text{C}$ which is relatively low (less than 30°C). Therefore, nucleate boiling will occur. The heat flux in this case can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\,\text{Pr}_l^n}\right)^3$$
$$= (0.232\times10^{-3})(2203\times10^3)\left[\frac{9.8(943.4\text{-}1.121)}{0.0550}\right]^{1/2}\left(\frac{4244(130-120)}{0.0130(2203\times10^3)1.44}\right)^3$$
$$= 228{,}400 \text{ W/m}^2$$

The rate of heat transfer is

$$\dot{Q} = A\dot{q}_{\text{nucleate}} = \frac{1}{4}\pi(0.20\text{ m})^2(228{,}400 \text{ W/m}^2) = 7174 \text{ W}$$

The rate of evaporation is

$$\dot{m}_{\text{evap}} = \frac{\dot{Q}}{h_{fg}} = \frac{7174 \text{ W}}{2203\times10^3 \text{ kJ/kg}} = 0.003256 \text{ kg/s}$$

Noting that the tank is half-filled, the mass of the water and the time it will take for the tank to empty are

$$m = \frac{1}{2}\rho_l \boldsymbol{V} = \frac{1}{2}(943.4\text{ kg/m}^3)\left[\pi(0.20\text{ m})^2 / 4\times(0.30\text{ m})\right] = 4.446 \text{ kg}$$

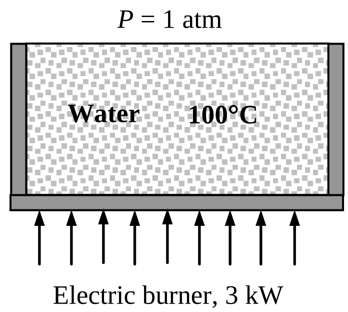
$$t = \dot{m}_{\text{evap}} = \frac{m}{\dot{m}_{\text{evap}}} = \frac{4.446\text{ kg}}{0.003256\text{ kg/s}} = 1365\text{ s} = \mathbf{22.8\,min}$$

10-100 Water is boiled at sea level (1 atm pressure) and thus at a saturation (or boiling) temperature of T_{sat} = 100°C in a mechanically polished AISI 304 stainless steel pan placed on top of a 3-kW electric burner. Only 60% of the heat (1.8 kW) generated is transferred to the water. The inner surface temperature of the pan and the temperature difference across the bottom of the pan are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the boiler are negligible. **3** The boiling regime is nucleate boiling (this assumption will be checked later). **4** Heat transfer through the bottom of the pan is one-dimensional.

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)

$$\rho_l = 957.9 \text{ kg/m}^3$$
$$\rho_v \approx 0.60 \text{ kg/m}^3$$
$$\sigma = 0.0589 \text{ N/m}$$
$$\text{Pr}_l = 1.75$$
$$h_{fg} = 2257 \times 10^3 \text{ J/kg}$$
$$\mu_l = 0.282 \times 10^{-3} \text{ kg}\cdot\text{m/s}$$
$$c_{pl} = 4217 \text{ J/kg}\cdot K$$



Also, k_{steel} = 14.9 W/m·K (Table A-3), $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis The rate of heat transfer to the water and the heat flux are

$$\dot{Q} = 0.60 \times 3 \text{ kW} = 1.8 \text{ kW} = 1800 \text{ W}$$
$$A_s = \pi D^2/4 = \pi(0.30 \text{ m})^2/4 = 0.07069 \text{ m}^2$$
$$\dot{q} = \dot{Q}/A_s = (1800 \text{ W})/(0.07069 \text{ m}^2) = 25.46 \text{ W/m}^2$$

Then temperature difference across the bottom of the pan is determined directly from the steady one-dimensional heat conduction relation to be

$$\dot{q} = k_{steel}\frac{\Delta T}{L} \rightarrow \Delta T = \frac{\dot{q}L}{k_{steel}} = \frac{(25{,}460 \text{ W/m}^2)(0.006 \text{ m})}{14.9 \text{ W/m}\cdot K} = \mathbf{10.3°C}$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given.

Assuming nucleate boiling, the temperature of the inner surface of the pan is determined from Rohsenow relation to be

$$\dot{q}_{nucleate} = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s - T_{sat})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$

$$25{,}460 = (0.282 \times 10^{-3})(2257 \times 10^3)\left[\frac{9.81(957.9 - 0.60)}{0.0589}\right]^{1/2}\left(\frac{4217(T_s - 100)}{0.0130(2257 \times 10^3)1.75}\right)^3$$

It gives

$$T_s = \mathbf{105.7°C}$$

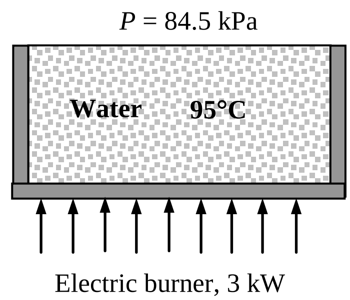
which is in the nucleate boiling range (5 to 30°C above surface temperature). Therefore, the nucleate boiling assumption is valid.

10-101 Water is boiled at 84.5 kPa pressure and thus at a saturation (or boiling) temperature of T_{sat} = 95°C in a mechanically polished AISI 304 stainless steel pan placed on top of a 3-kW electric burner. Only 60% of the heat (1.8 kW) generated is transferred to the water. The inner surface temperature of the pan and the temperature difference across the bottom of the pan are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the boiler are negligible. **3** The boiling regime is nucleate boiling (this assumption will be checked later). **4** Heat transfer through the bottom of the pan is one-dimensional.

Properties The properties of water at the saturation temperature of 95°C are (Tables 10-1 and A-9)

$$\rho_l = 961.5 \text{ kg/m}^3$$
$$\rho_v \approx 0.50 \text{ kg/m}^3$$
$$\sigma = 0.0599 \text{ N/m}$$
$$\Pr_l = 1.85$$
$$h_{fg} = 2270\times10^3 \text{ J/kg}$$
$$\mu_l = 0.297\times10^{-3} \text{ kg}\cdot\text{m/s}$$
$$c_{pl} = 4212 \text{ J/kg}\cdot K$$



Also, k_{steel} = 14.9 W/m·K (Table A-3), $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis The rate of heat transfer to the water and the heat flux are

$$\dot{Q} = 0.60\times 3\text{ kW} = 1.8\text{ kW} = 1800\text{ W}$$
$$A_s = \pi D^2/4 = \pi(0.30\text{ m})^2/4 = 0.07069\text{ m}^2$$
$$\dot{q} = \dot{Q}/A_s = (1800\text{ W})/(0.07069\text{ m}^2) = 25{,}460\text{ W/m}^2 = 25.46\text{ kW/m}^2$$

Then temperature difference across the bottom of the pan is determined directly from the steady one-dimensional heat conduction relation to be

$$\dot{q} = k_{steel}\frac{\Delta T}{L} \rightarrow \Delta T = \frac{\dot{q}L}{k_{steel}} = \frac{(25{,}460\text{ W/m}^2)(0.006\text{ m})}{14.9\text{ W/m}\cdot K} = \mathbf{10.3°C}$$

The Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given.

Assuming nucleate boiling, the temperature of the inner surface of the pan is determined from Rohsenow relation to be

$$\dot{q}_{nucleate} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{sat})}{C_{sf}h_{fg}\Pr_l^n}\right)^3$$

$$25{,}460 = (0.297\times10^{-3})(2270\times10^3)\left[\frac{9.81(961.5-0.50)}{0.0599}\right]^{1/2}\left(\frac{4212(T_s-95)}{0.0130(2270\times10^3)1.85}\right)^3$$

It gives

$$T_s = \mathbf{100.9°C}$$

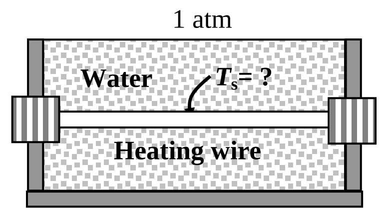
which is in the nucleate boiling range (5 to 30°C above surface temperature). Therefore, the nucleate boiling assumption is valid.

10-102 Water is boiled at 1 atm pressure and thus at a saturation temperature of T_{sat} = 100°C by a nickel electric heater whose diameter is 2 mm. The highest temperature at which this heater can operate without burnout is to be determined.

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

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)

$$\rho_l = 957.9 \text{ kg/m}^3 \qquad h_{fg} = 2257\times10^3 \text{ J/kg}$$
$$\rho_v \approx 0.60 \text{ kg/m}^3 \qquad \mu_l = 0.282\times10^{-3} \text{ kg}\cdot\text{m/s}$$
$$\sigma = 0.0589 \text{ N/m} \qquad c_{pl} = 4217 \text{ J/kg}\cdot K$$
$$\text{Pr}_l = 1.75$$



Also, $C_{sf} = 0.0060$ and n = 1.0 for the boiling of water on a nickel surface (Table 10-3).

Analysis The maximum rate of heat transfer without the burnout is simply the critical heat flux. For a horizontal heating wire, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (0.001)\left(\frac{9.8(957.9-0.60}{0.0589}\right)^{1/2} = 0.399 < 1.2$$
$$C_{cr} = 0.12L^{*-0.25} = 0.12(0.399)^{-0.25} = 0.151$$

Then the maximum or critical heat flux is determined from

$$\begin{aligned}\dot{q}_{\max} &= C_{cr} h_{fg}[\sigma g \rho_v^2(\rho_l-\rho_v)]^{1/4} \\ &= 0.151(2257\times10^3)[0.0589\times9.8\times(0.6)^2(957.9-0.60)]^{1/4} \\ &= 1{,}280{,}000 \text{ W/m}^2\end{aligned}$$

Rohsenow relation which gives the nucleate boiling heat flux for a specified surface temperature can also be used to determine the surface temperature when the heat flux is given. Substituting the maximum heat flux into Rohsenow relation together with other properties gives

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n}\right)^3$$

$$1{,}280{,}000 = (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.8(957.9\text{-}0.60)}{0.0589}\right]^{1/2}\left(\frac{4217(T_s-100)}{0.0060(2257\times10^3)1.75}\right)^3$$

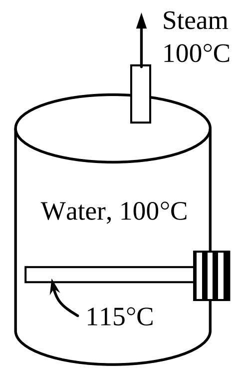
It gives the maximum temperature to be:

$$T_s = \mathbf{109.6^\circ C}$$

10-103 Water is boiled at T_{sat} = 100°C by a chemically etched stainless steel electric heater whose surface temperature is maintained at T_s = 115°C. The rate of heat transfer to the water, the rate of evaporation of water, and the maximum rate of evaporation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are (Tables 10-1 and A-9)



$$\rho_l = 957.9\ \text{kg/m}^3$$
$$\rho_v = 0.60\ \text{kg/m}^3$$
$$\sigma = 0.0589\ \text{N/m}$$
$$\text{Pr}_l = 1.75$$
$$h_{fg} = 2257\times10^3\ \text{J/kg}$$
$$\mu_l = 0.282\times10^{-3}\ \text{kg}\cdot\text{m/s}$$
$$c_{pl} = 4217\ \text{J/kg}\cdot°\text{C}$$

Also, $C_{sf} = 0.0130$ and n = 1.0 for the boiling of water on a chemically etched stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis (*a*) The excess temperature in this case is $\Delta T = T_s - T_{sat} = 115 - 100 = 15°\text{C}$ which is relatively low (less than 30°C). Therefore, nucleate boiling will occur. The heat flux in this case can be determined from Rohsenow relation to be

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}(T_s-T_{\text{sat}})}{C_{sf}h_{fg}\text{Pr}_l^n}\right)^3$$
$$= (0.282\times10^{-3})(2257\times10^3)\left[\frac{9.8(957.9-0.60)}{0.0589}\right]^{1/2}\left(\frac{4217(115-100)}{0.0130(2257\times10^3)1.75}\right)^3$$
$$= 474{,}900\ \text{W/m}^2$$

The surface area of the bottom of the heater is $A_s = \pi DL = \pi(0.002\ \text{m})(0.8\ \text{m}) = 0.005027\ \text{m}^2$.

Then the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A_s\dot{q}_{\text{nucleate}} = (0.005027\ \text{m}^2)(474{,}900\ \text{W/m}^2) = \mathbf{2387\ W}$$

The rate of evaporation of water is determined from

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{2387\ \text{J/s}}{2257\times10^3\ \text{J/kg}} = \mathbf{1.058\times10^{-3}\ kg/s = 3.81\,kg/h}$$

(*b*) For a horizontal heating wire, the coefficient C_{cr} is determined from Table 10-4 to be

$$L^* = L\left(\frac{g(\rho_l-\rho_v)}{\sigma}\right)^{1/2} = (0.001)\left(\frac{9.8(957.9-0.60)}{0.0589}\right)^{1/2} = 0.399 < 1.2$$
$$C_{cr} = 0.12L^{*-0.25} = 0.12(0.399)^{-0.25} = 0.151$$

Then the maximum or critical heat flux is determined from

$$\dot{q}_{\max} = C_{cr}h_{fg}[\sigma g\rho_v^2(\rho_l-\rho_v)]^{1/4} = 0.151(2257\times10^3)[0.0589\times9.8\times(0.6)^2(957.9-0.60)]^{1/4}$$
$$= 1{,}280{,}000\ \text{W/m}^2 = \mathbf{1280\ kW/m^2}$$

10-104 The initial boiling heat transfer coefficient and the total heat transfer coefficient, when a heated steel rod was submerged in a water bath, are to be determined.

Assumptions **1** Steady operating condition exists. **2** The steel rod has uniform initial surface temperature.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_l = 957.9 kg/m^3 (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 300°C are, from Table A-16,

ρ_v = 0.3831 kg/m^3 $\quad$ c_{pv} = 1997 J/kg·K

$\mu_v = 2.045 \times 10^{-5}$ kg/m·s $\quad$ k_v = 0.04345 W/m·K

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}}$ = 400°C, which is much larger than 30°C for water from Fig. 10-6. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from

$$\begin{aligned}\dot{q}_{\text{film}} &= C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})\\
&= 0.62\left[\frac{9.81(0.04345)^3(0.3831)(957.9-0.3831)[2257\times10^3+0.4(1997)(400)]}{(2.045\times10^{-5})(0.02)(400)}\right]^{1/4}(400)\\
&= 6.476\times10^4\ \text{W/m}^2\end{aligned}$$

Using the Newton's law of cooling, the boiling heat transfer coefficient is

$$\dot{q}_{\text{film}} = h_{\text{film}}(T_s-T_{\text{sat}}) \quad\rightarrow\quad h_{\text{film}} = \frac{\dot{q}_{\text{film}}}{T_s-T_{\text{sat}}}$$

$$h_{\text{film}} = \frac{6.476\times10^4\ \text{W/m}^2}{(500-100)\ \text{K}} = \mathbf{162\ W/m^2\cdot K}$$

The radiation heat transfer coefficient can be determined using

$$\dot{q}_{\text{rad}} = \varepsilon\sigma(T_s^4-T_{\text{sat}}^4) = h_{\text{rad}}(T_s-T_{\text{sat}}) \quad\rightarrow\quad h_{\text{rad}} = \frac{\varepsilon\sigma(T_s^4-T_{\text{sat}}^4)}{T_s-T_{\text{sat}}}$$

$$h_{\text{rad}} = \frac{\varepsilon\sigma(T_s^4-T_{\text{sat}}^4)}{T_s-T_{\text{sat}}} = \frac{(0.9)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(773^4-373^4)\ \text{K}^4}{(500-100)\ \text{K}} = 43.08\ \text{W/m}^2\cdot\text{K}$$

Then, the total heat transfer coefficient can be determined using

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} \quad\rightarrow\quad h_{\text{total}}(T_s-T_{\text{sat}}) = h_{\text{film}}(T_s-T_{\text{sat}}) + \frac{3}{4}h_{\text{rad}}(T_s-T_{\text{sat}})$$

or

$$\begin{aligned}h_{\text{total}} &= h_{\text{film}} + \frac{3}{4}h_{\text{rad}}\\
&= 162\ \text{W/m}^2\cdot\text{K} + \frac{3}{4}(43.08\ \text{W/m}^2\cdot\text{K})\\
&= \mathbf{194\ W/m^2\cdot K}\end{aligned}$$

Discussion The boiling heat transfer coefficient (h_{film}) is 3.76 times the radiation heat transfer coefficient (h_{rad}).

10-105 The boiling heat transfer coefficient and the total heat transfer coefficient for water being boiled by a cylindrical metal rod are to be determined.

Assumptions **1** Steady operating condition exists. **2** Heat losses from the boiler are negligible.

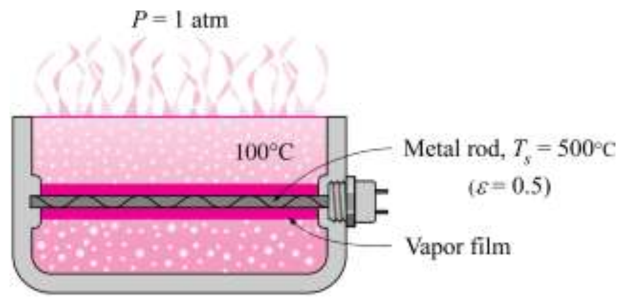
Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_l = 957.9 kg/m^3 (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 300°C are, from Table A-16,

ρ_v = 0.3831 kg/m^3

c_{pv} = 1997 J/kg·K

$\mu_v = 2.045 \times 10^{-5}$ kg/m·s

k_v = 0.04345 W/m·K



Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}}$ = 400°C, which is much larger than 30°C for water from Fig 10-6. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from

$$\dot{q}_{\text{film}} = C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})$$

$$= 0.62\left[\frac{9.81(0.04345)^3(0.3831)(957.9-0.3831)[2257\times10^3+0.4(1997)(400)]}{(2.045\times10^{-5})(0.002)(400)}\right]^{1/4}(400)$$

$$= 1.152\times10^5 \text{ W/m}^2$$

Using the Newton's law of cooling, the boiling heat transfer coefficient is

$$\dot{q}_{\text{film}} = h_{\text{film}}(T_s - T_{\text{sat}}) \quad\rightarrow\quad h_{\text{film}} = \frac{\dot{q}_{\text{film}}}{T_s - T_{\text{sat}}}$$

$$h_{\text{film}} = \frac{1.152\times10^5 \text{ W/m}^2}{(500-100)\text{ K}} = \mathbf{288\ W/m^2\cdot K}$$

The radiation heat transfer coefficient can be determined using

$$\dot{q}_{\text{rad}} = \varepsilon\sigma(T_s^4 - T_{\text{sat}}^4) = h_{\text{rad}}(T_s - T_{\text{sat}}) \quad\rightarrow\quad h_{\text{rad}} = \frac{\varepsilon\sigma(T_s^4 - T_{\text{sat}}^4)}{T_s - T_{\text{sat}}}$$

$$h_{\text{rad}} = \frac{\varepsilon\sigma(T_s^4 - T_{\text{sat}}^4)}{T_s - T_{\text{sat}}} = \frac{(0.5)(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)(773^4 - 373^4)\text{ K}^4}{(500-100)\text{ K}} = 23.93 \text{ W/m}^2\cdot\text{K}$$

Then, the total heat transfer coefficient can be determined using

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} \quad\rightarrow\quad h_{\text{total}}(T_s - T_{\text{sat}}) = h_{\text{film}}(T_s - T_{\text{sat}}) + \frac{3}{4}h_{\text{rad}}(T_s - T_{\text{sat}})$$

or

$$h_{\text{total}} = h_{\text{film}} + \frac{3}{4}h_{\text{rad}}$$

$$= 288 \text{ W/m}^2\cdot\text{K} + \frac{3}{4}(23.93 \text{ W/m}^2\cdot\text{K})$$

$$= \mathbf{306\ W/m^2\cdot K}$$

Discussion The boiling heat transfer coefficient (h_{film}) is about 12 times the radiation heat transfer coefficient (h_{rad}).

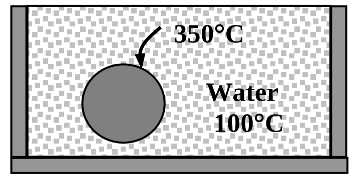
10-106 Water is boiled at T_{sat} = 100°C by a spherical platinum heating element immersed in water. The surface temperature is T_s = 350°C. The boiling heat transfer coefficient is to be determined.

Assumptions 1 Steady operating conditions exist. **2** Heat losses from the heater and the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are (Table A-9)

$$h_{fg} = 2257 \times 10^3 \text{ J/kg}$$
$$\rho_l = 957.9 \text{ kg/m}^3$$

The properties of water vapor at (350+100)/2 = 225°C are (Table A-16)



$$\rho_v = 0.444 \text{ kg/m}^3$$
$$\mu_v = 1.749 \times 10^{-5} \text{ kg/m}\cdot\text{s}$$
$$c_{pv} = 1951 \text{ J/kg}\cdot°\text{C}$$
$$k_v = 0.03581 \text{ W/m}\cdot°\text{C}$$

Analysis The film boiling occurs since the temperature difference between the surface and the fluid. The heat flux in this case can be determined from

$$\dot{q}_{\text{film}} = 0.67\left[\frac{gk_v^3\rho_v(\rho_l - \rho_v)\left[h_{fg} + 0.4c_{pv}(T_s - T_{\text{sat}})\right]}{\mu_v D(T_s - T_{\text{sat}})}\right]^{1/4}(T_s - T_{\text{sat}})$$

$$= 0.67\left[\frac{(9.81)(0.03581)^3(0.444)(957.9 - 0.444)\left[2257 \times 10^3 + 0.4(1951)(350 - 100)\right]}{(1.749 \times 10^{-5})(0.15)(350 - 100)}\right]^{1/4}(350 - 100)$$

$$= 27{,}399 \text{ W/m}^2$$

The boiling heat transfer coefficient is

$$\dot{q}_{\text{film}} = h(T_s - T_{\text{sat}}) \longrightarrow h = \frac{\dot{q}_{\text{film}}}{T_s - T_{\text{sat}}} = \frac{27{,}399 \text{ W/m}^2}{(350 - 100)°\text{C}} = \mathbf{110\ W/m^2 \cdot C}$$

10-107 The initial boiling heat transfer coefficient and the total heat transfer coefficient, when heated steel ball bearings are submerged in a water bath, are to be determined.

Assumptions **1** Steady operating condition exists. **2** The steel ball bearings have uniform initial surface temperature.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257 kJ/kg (Table A-2) and ρ_l = 957.9 kg/m^3 (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2$ = 400°C are, from Table A-16,

ρ_v = 0.3262 kg/m^3 $\quad$ c_{pv} = 2066 J/kg·K

μ_v = 2.446 × 10^{-5} kg/m·s $\quad$ k_v = 0.05467 W/m·K

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}}$ = 600°C, which is much larger than 30°C for water in Fig. 10-6. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from

$$\dot{q}_{\text{film}} = C_{\text{film}}\left[\frac{gk_v^3\rho_v(\rho_l-\rho_v)[h_{fg}+0.4c_{pv}(T_s-T_{\text{sat}})]}{\mu_v D(T_s-T_{\text{sat}})}\right]^{1/4}(T_s-T_{\text{sat}})$$

$$= 0.67\left[\frac{9.81(0.05467)^3(0.3262)(957.9-0.3262)[2257\times10^3+0.4(2066)(600)]}{(2.446\times10^{-5})(0.02)(600)}\right]^{1/4}(600)$$

$$= 1.052\times10^5\ \text{W/m}^2$$

Using the Newton's law of cooling, the boiling heat transfer coefficient is

$$\dot{q}_{\text{film}} = h_{\text{film}}(T_s-T_{\text{sat}}) \quad\rightarrow\quad h_{\text{film}} = \frac{\dot{q}_{\text{film}}}{T_s-T_{\text{sat}}}$$

$$h_{\text{film}} = \frac{1.052\times10^5\ \text{W/m}^2}{(700-100)\ \text{K}} = \mathbf{175\ W/m^2\cdot K}$$

The radiation heat transfer coefficient can be determined using

$$\dot{q}_{\text{rad}} = \varepsilon\sigma(T_s^4-T_{\text{sat}}^4) = h_{\text{rad}}(T_s-T_{\text{sat}}) \quad\rightarrow\quad h_{\text{rad}} = \frac{\varepsilon\sigma(T_s^4-T_{\text{sat}}^4)}{T_s-T_{\text{sat}}}$$

$$h_{\text{rad}} = \frac{\varepsilon\sigma(T_s^4-T_{\text{sat}}^4)}{T_s-T_{\text{sat}}} = \frac{(0.75)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(973^4-373^4)\ \text{K}^4}{(700-100)\ \text{K}} = 62.15\ \text{W/m}^2\cdot\text{K}$$

Then, the total heat transfer coefficient can be determined using

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} \quad\rightarrow\quad h_{\text{total}}(T_s-T_{\text{sat}}) = h_{\text{film}}(T_s-T_{\text{sat}}) + \frac{3}{4}h_{\text{rad}}(T_s-T_{\text{sat}})$$

or

$$h_{\text{total}} = h_{\text{film}} + \frac{3}{4}h_{\text{rad}}$$

$$= 175\ \text{W/m}^2\cdot\text{K} + \frac{3}{4}(62.15\ \text{W/m}^2\cdot\text{K})$$

$$= \mathbf{222\ W/m^2\cdot K}$$

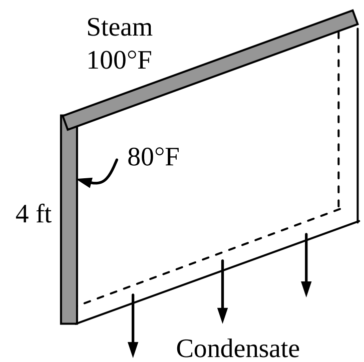
Discussion The boiling heat transfer coefficient (h_{film}) is 2.82 times the radiation heat transfer coefficient (h_{rad}).

10-108E Steam at a saturation temperature of T_{sat} = 100°F condenses on a vertical plate which is maintained at 80°F. The rate of heat transfer to the plate and the rate of condensation of steam per ft width of the plate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The plate is isothermal. **3** The condensate flow is wavy-laminar over the entire plate (this assumption will be verified). **4** The density of vapor is much smaller than the density of liquid, $\rho_v \ll \rho_l$.

Properties The properties of water at the saturation temperature of 100°F are h_{fg} = 1037 Btu/lbm and ρ_v = 0.00286 lbm/ft^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (100 + 80)/2 = 90$°F are (Table A-9E),

$$\rho_l = 62.12 \text{ lbm/ft}^3$$
$$\mu_l = 5.117\times10^{-4} \text{ lbm/ft}\cdot\text{s} = 1.842 \text{ lbm/ft}\cdot\text{h}$$
$$\nu_l = \mu_l/\rho_l = 0.02965 \text{ ft}^2\text{/h}$$
$$c_{pl} = 0.999 \text{ Btu/lbm}\cdot°\text{F}$$
$$k_l = 0.358 \text{ Btu/h}\cdot\text{ft}\cdot°\text{F}$$



Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{sat} - T_s) \\ &= 1037 \text{ Btu/lbm} + 0.68\times(0.999 \text{ Btu/lbm}\cdot°\text{F})(100-80)°\text{F} \\ &= 1051 \text{ Btu/lbm} \end{aligned}$$

Assuming wavy-laminar flow, the Reynolds number is determined from

$$\text{Re} = \text{Re}_{\text{vertical,wavy}} = \left[4.81 + \frac{3.70Lk_l(T_{sat}-T_s)}{\mu_l h^*_{fg}}\left(\frac{g}{\nu_l^2}\right)^{1/3}\right]^{0.820}$$

$$= \left[4.81 + \frac{3.70\times(4\text{ ft})\times(0.358 \text{ Btu/h}\cdot\text{ft}\cdot°\text{F})\times(100-80)°\text{F}}{(1.842 \text{ lbm/ft}\cdot\text{h})(1051 \text{ Btu/lbm})}\left(\frac{32.2 \text{ ft/s}^2}{(0.02965 \text{ ft}^2/\text{h})^2}\frac{(3600\text{ s})^2}{(1\text{ h})^2}\right)^{1/3}\right]^{0.82} = 145$$

which is between 30 and 1800, and thus our assumption of wavy laminar flow is verified. Then the condensation heat transfer coefficient is determined from

$$h = h_{\text{vertical,wavy}} = \frac{\text{Re}\,k_l}{1.08\,\text{Re}^{1.22} - 5.2}\left(\frac{g}{\nu_l^2}\right)^{1/3}$$

$$= \frac{145\times(0.358 \text{ Btu/h}\cdot\text{ft}\cdot°\text{F})}{1.08(145)^{1.22} - 5.2}\left(\frac{32.2 \text{ ft/s}^2}{(0.02965 \text{ ft}^2/\text{h})^2}\frac{(3600\text{ s})^2}{(1\text{ h})^2}\right)^{1/3} = 875 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

The heat transfer surface area of the plate is

$$A_s = W\times L = (4\text{ ft})(1\text{ ft}) = 4\text{ ft}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (875 \text{ Btu/h}\cdot\text{ft}^2\cdot°\text{F})(4\text{ ft}^2)(100-80)°\text{F} = \mathbf{70{,}000\ Btu/h}$$

The rate of condensation of steam is determined from

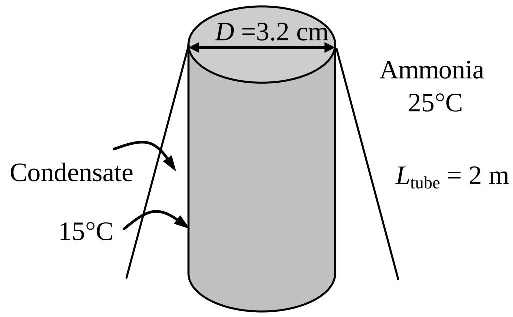
$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{70{,}000 \text{ Btu/h}}{1051 \text{ Btu/lbm}} = \mathbf{66.6\ lbm/h}$$

10-109 Saturated ammonia at a saturation temperature of T_{sat} = 25°C condenses on the outer surface of vertical tube which is maintained at 15°C by circulating cooling water. The rate of heat transfer to the coolant and the rate of condensation of ammonia are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal. **3** The tube can be treated as a vertical plate. **4** The condensate flow is turbulent over the entire tube (this assumption will be verified). **5** The density of vapor is much smaller than the density of liquid, $\rho_v \ll \rho_l$.

Properties The properties of ammonia at the saturation temperature of 25°C are $h_{fg} = 1166\times10^3$ J/kg and ρ_v = 7.809 kg/m^3. The properties of liquid ammonia at the film temperature of $T_f = (T_{sat} + T_s)/2 = (25 + 15)/2 = 20$°C are (Table A-11),

$$\rho_l = 610.2\ \text{kg/m}^3$$
$$\mu_l = 1.519\times10^{-4}\ \text{kg/m}\cdot\text{s}$$
$$\nu_l = \mu_l/\rho_l = 0.2489\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4745\ \text{J/kg}\cdot°\text{C}$$
$$k_l = 0.4927\ \text{W/m}\cdot°\text{C}$$
$$\text{Pr}_l = 1.463$$



Analysis (*a*) The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 1166\times10^3\ \text{J/kg} + 0.68\times4745\ \text{J/kg}\cdot°\text{C}(25-15)°\text{C} = 1198\times10^3\ \text{J/kg}$$

Assuming turbulent flow, the Reynolds number is determined from

$$\text{Re} = \text{Re}_{\text{vertical,turb}} = \left[\frac{0.0690Lk_l\ \text{Pr}^{0.5}(T_{sat}-T_s)}{\mu_l h^*_{fg}}\left(\frac{g}{\nu_l^2}\right)^{1/3} - 151\,\text{Pr}^{0.5} + 253\right]^{4/3}$$

$$= \left[\frac{0.0690\times2\times0.4927(1.463)^{0.5}(25-15)}{(1.519\times10^{-4}\ \text{kg/m.s})(1198\times10^3\ \text{J/kg})}\left(\frac{9.81}{(0.2489\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3} - 151(1.463)^{0.5} + 253\right]^{4/3}$$

$$= 2142$$

which is greater than 1800, and thus our assumption of turbulent flow is verified. Then the condensation heat transfer coefficient is determined from

$$h = h_{\text{vertical,turbulent}} = \frac{\text{Re}\,k_l}{8750 + 58\,\text{Pr}^{-0.5}(\text{Re}^{0.75} - 253)}\left(\frac{g}{\nu_l^2}\right)^{1/3}$$

$$= \frac{2142\times(0.4927\ \text{W/m}\cdot°\text{C})}{8750 + 58\times1.463^{-0.5}(2142^{0.75} - 253)}\left(\frac{9.81\ \text{m/s}^2}{(0.2489\times10^{-6}\ \text{m}^2/\text{s})^2}\right)^{1/3} = 4873\ \ \text{W/m}^2\cdot°\text{C}$$

The heat transfer surface area of the tube is $A_s = \pi DL = \pi(0.032\ \text{m})(2\ \text{m}) = 0.2011\ \text{m}^2$. Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (4873\ \text{W/m}^2\cdot°\text{C})(0.2011\ \text{m}^2)(25-15)°\text{C} = \mathbf{9800\ W}$$

(*b*) The rate of condensation of ammonia is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{9800\ \text{J/s}}{1198\times10^3\ \text{J/kg}} = \mathbf{8.180\times10^{-3}\ kg/s}$$

Discussion Combining equations $\delta_L = k_l/h_l$ and $h = (4/3)h_L$, the thickness of the liquid film at the bottom of the tube is determined to be

$$\delta_L = \frac{4k_l}{3h} = \frac{4(0.4927\ \text{W/m}\cdot°\text{C})}{3(4873\ \text{W/m}^2\cdot°\text{C})} = 0.135\times10^{-3}\ \text{m} = 0.135\ \text{mm}$$

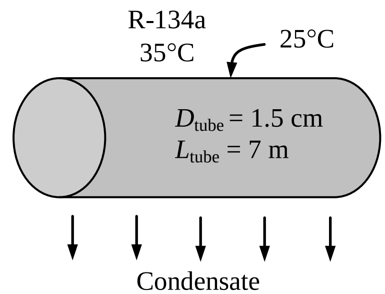
The assumption that the tube diameter is large relative to the thickness of the liquid film at the bottom of the tube is verified since the thickness of the liquid film is 0.135 mm, which is much smaller than the diameter of the tube (3.2 cm). Also, the assumption of turbulent flow is verified since Reynolds number is greater than 1800.

10-110 Saturated refrigerant-134a vapor condenses on the outside of a horizontal tube maintained at a specified temperature. The rate of condensation of the refrigerant is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal.

Properties The properties of refrigerant-134a at the saturation temperature of 35°C are $h_{fg} = 168.2\times10^3$ J/kg and $\rho_v = 43.41$ kg/m^3. The properties of liquid R-134a at the film temperature of $T_f = (T_{sat} + T_s)/2 = (35 + 25)/2 = 30$°C are (Table A-10),

$$\rho_l = 1188 \text{ kg/m}^3$$
$$\mu_l = 1.888\times10^{-4} \text{ kg/m.s}$$
$$c_{pl} = 1448 \text{ J/kg.°C}$$
$$k_l = 0.0808 \text{ W/m.°C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 168.2\times10^3 \text{ J/kg} + 0.68\times1448 \text{ J/kg}\cdot\text{°C}(35-25)\text{°C} = 178.0\times10^3 \text{ J/kg}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$h = h_{\text{horizontal}} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4}$$
$$= 0.729\left[\frac{(9.81 \text{ m/s}^2)(1188 \text{ kg/m}^3)(1188 - 43.41 \text{ kg/m}^3)(178.0\times10^3 \text{ J/kg})(0.0808 \text{ W/m}\cdot\text{°C})^3}{(1.888\times10^{-4} \text{ kg/m}\cdot\text{s})(35-25)\text{°C}(0.015 \text{ m})}\right]^{1/4}$$
$$= 1880 \text{ W/m}^2\cdot\text{°C}$$

The heat transfer surface area of the pipe is

$$A_s = \pi DL = \pi(0.015 \text{ m})(7 \text{ m}) = 0.3299 \text{ m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (1880 \text{ W/m}^2\cdot\text{°C})(0.3299 \text{ m}^2)(35-25)\text{°C} = 6202 \text{ W}$$

The rate of condensation of steam is determined from

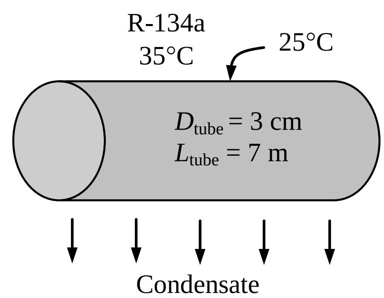
$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{6202 \text{ J/s}}{178.0\times10^3 \text{ J/kg}} = 0.03484 \text{ kg/s} = \mathbf{2.09 \ kg/min}$$

10-111 Saturated refrigerant-134a vapor condenses on the outside of a horizontal tube maintained at a specified temperature. The rate of condensation of the refrigerant is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal.

Properties The properties of refrigerant-134a at the saturation temperature of 35°C are h_{fg} = 168.2×10^3 J/kg and ρ_v = 43.41 kg/m^3. The properties of liquid R-134a at the film temperature of $T_f = (T_{sat} + T_s)/2 = (35 + 25)/2 = 30°C$ are (Table A-10),

$$\rho_l = 1188\ \text{kg/m}^3$$
$$\mu_l = 1.888\times10^{-4}\ \text{kg/m.s}$$
$$\nu_l = \mu_l/\rho_l = 0.1590\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 1448\ \text{J/kg.°C}$$
$$k_l = 0.0808\ \text{W/m.°C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 168.2\times10^3\ \text{J/kg} + 0.68\times1448\ \text{J/kg}\cdot°\text{C}(35-25)°\text{C} = 178.0\times10^3\ \text{J/kg}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$h = h_{\text{horizontal}} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4}$$
$$= 0.729\left[\frac{(9.81\ \text{m/s}^2)(1188\ \text{kg/m}^3)(1188 - 43.41\ \text{kg/m}^3)(178.0\times10^3\ \text{J/kg})(0.0808\ \text{W/m}\cdot°\text{C})^3}{(1.888\times10^{-4}\ \text{kg/m}\cdot\text{s})(35-25)°\text{C}(0.03\ \text{m})}\right]^{1/4}$$
$$= 1581\ \text{W/m}^2\cdot°\text{C}$$

The heat transfer surface area of the pipe is

$$A_s = \pi DL = \pi(0.03\ \text{m})(7\ \text{m}) = 0.6597\ \text{m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (1581\ \text{W/m}^2\cdot°\text{C})(0.6597\ \text{m}^2)(35-25)°\text{C} = 10{,}430\ \text{W}$$

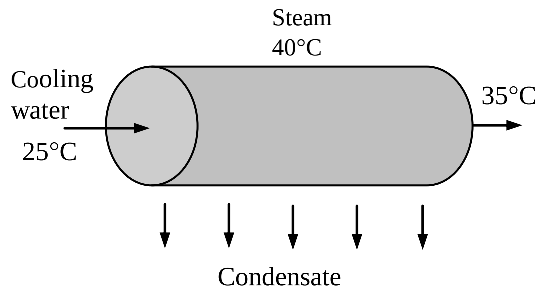
The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{10{,}430\ \text{J/s}}{178.0\times10^3\ \text{J/kg}} = 0.05859\ \text{kg/s} = \mathbf{3.52\ kg/min}$$

10-112 Steam at a saturation temperature of T_{sat} = 40°C condenses on the outside of a thin horizontal tube. Heat is transferred to the cooling water that enters the tube at 25°C and exits at 35°C. The rate of condensation of steam, the average overall heat transfer coefficient, and the tube length are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube can be taken to be isothermal at the bulk mean fluid temperature in the evaluation of the condensation heat transfer coefficient. **3** Liquid flow through the tube is fully developed. **4** The thickness and the thermal resistance of the tube is negligible.

Properties The properties of water at the saturation temperature of 40°C are h_{fg} = 2407×10^3 J/kg and ρ_v = 0.05 kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (40+30)/2$ = 35°C and at the bulk fluid temperature of $T_b = (T_{in} + T_{out})/2 = (25$ $+ 35)/2$ = 30°C are (Table A-9),



At 35°C :	**At 30°C :**
$\rho_l = 994.0\ \text{kg/m}^3$	$\rho_l = 996.0\ \text{kg/m}^3$
$\mu_l = 0.720\times10^{-3}\ \text{kg/m}\cdot\text{s}$	$\mu_l = 0.798\times10^{-3}\ \text{kg/m}\cdot\text{s}$
$\nu_l = \mu_l/\rho_l = 0.724\times10^{-6}\ \text{m}^2/\text{s}$	$\nu_l = \mu_l/\rho_l = 0.801\times10^{-6}\ \text{m}^2/\text{s}$
$c_{pl} = 4178\ \text{J/kg}\cdot°\text{C}$	$c_{pl} = 4178\ \text{J/kg}\cdot°\text{C}$
$k_l = 0.623\ \text{W/m}\cdot°\text{C}$	$k_l = 0.615\ \text{W/m}\cdot°\text{C}$
Pr = 4.83	Pr = 5.42

Analysis The mass flow rate of water and the rate of heat transfer to the water are

$$\dot{m}_{\text{water}} = \rho V A_c = (996\ \text{kg/m}^3)(2\ \text{m/s})[\pi(0.03\ \text{m})^2/4] = 1.408\ \text{kg/s}$$

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (1.408\ \text{kg/s})(4178\ \text{J/kg}\cdot°\text{C})(35-25)°\text{C} = \mathbf{58{,}830\ \ W}$$

The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s) = 2407\times10^3\ \text{J/kg} + 0.68\times4178\ \text{J/kg}\cdot°\text{C}(40-30)°\text{C} = 2435\times10^3\ \text{J/kg}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$h_o = h_{\text{horizontal}} = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4}$$

$$= 0.729\left[\frac{(9.81\ \text{m/s}^2)(994\ \text{kg/m}^3)(994-0.05\ \text{kg/m}^3)(2435\times10^3\ \text{J/kg})(0.623\ \text{W/m}\cdot°\text{C})^3}{(0.720\times10^{-3}\ \text{kg/m}\cdot\text{s})(40-30)°\text{C}(0.03\ \text{m})}\right]^{1/4} = 11{,}775\ \text{W/m}^2\cdot°\text{C}$$

The average heat transfer coefficient for flow inside the tube is determined as follows:

$$\text{Re} = \frac{V_{avg}D}{\nu} = \frac{(2\ \text{m/s})(0.03\ \text{m})}{0.801\times10^{-6}} = 74{,}906$$

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(74{,}906)^{0.8}(5.42)^{0.4} = 358.9$$

$$h_i = \frac{k\text{Nu}}{D} = \frac{(0.615\ \text{W/m}\cdot°\text{C})(358.9)}{0.03\ \text{m}} = 7357\ \text{W/m}^2\cdot°\text{C}$$

Noting that the thermal resistance of the tube is negligible, the overall heat transfer coefficient becomes

$$U = \frac{1}{1/h_i + 1/h_o} = \frac{1}{1/7357 + 1/11{,}775} = \mathbf{4528\ \ W/m^2.°C}$$

The logarithmic mean temperature difference is:

$$\Delta T_{lm} = \frac{\Delta T_i - \Delta T_e}{\ln(\Delta T_i/\Delta T_o)} = \frac{15-5}{\ln(15/5)} = 9.102°\text{C}$$

The tube length is determined from

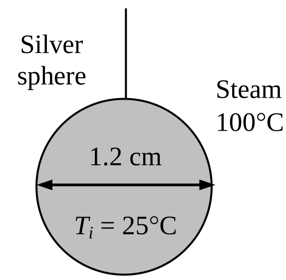
$$\dot{Q} = UA_s\Delta T_{lm} \rightarrow L = \frac{\dot{Q}}{U(\pi D)\Delta T_{lm}} = \frac{58{,}830\ \text{W}}{(4528\ \text{W/m}^2\cdot°\text{C})\pi(0.03\ \text{m})(9.102°\text{C})} = \mathbf{15.1\ m}$$

Note that the flow is turbulent, and thus the entry length in this case is 10D = 0.3 m is much shorter than the total tube length. This verifies our assumption of fully developed flow.

10-113 Saturated steam condenses on a suspended silver sphere which is initially at 25°C. The time needed for the temperature of the sphere to rise to 50°C and the amount of steam condenses are to be determined.

Assumptions **1** The temperature of the sphere changes uniformly and thus the lumped system analysis is applicable. **2** The average condensation heat transfer coefficient evaluated for the average temperature can be used for the entire process. **3** Constant properties at room temperature can be used for the silver ball.

Properties The properties of water at the saturation temperature of 100°C are h_{fg} = 2257×10^3 J/kg and ρ_v = 0.60 kg/m^3. The properties of the silver ball at room temperature and the properties of liquid water at the average film temperature of $T_f = (T_{\text{sat}} + T_{s,avg})/2 = (100 + 37.5)/2 = 69°\text{C} \approx 70°\text{C}$ are (Tables A-3 and A-9),



Silver Ball:

$$\rho = 10{,}500 \text{ kg/m}^3$$
$$\alpha = 174\times10^{-6} \text{ m}^2/\text{s}$$
$$c_p = 235 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 429 \text{ W/m}\cdot°\text{C}$$

Liquid Water:

$$\rho_l = 977.5 \text{ kg/m}^3$$
$$\mu_l = 0.404\times10^{-3} \text{ kg/m}\cdot\text{s}$$
$$c_{pl} = 4190 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 0.663 \text{ W/m}\cdot°\text{C}$$

Analysis The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) \\ &= 2257\times10^3 \text{ J/kg} + 0.68\times4190 \text{ J/kg}\cdot°\text{C}(100-37.5)°\text{C} = 2435\times10^3 \text{ J/kg} \end{aligned}$$

Noting that the tube is horizontal, the condensation heat transfer coefficient is determined from

$$\begin{aligned} h = h_{\text{sph}} &= 0.815\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}}-T_s)D}\right]^{1/4} \\ &= 0.815\left[\frac{(9.8 \text{ m/s}^2)(977.5 \text{ kg/m}^3)(977.5-0.60 \text{ kg/m}^3)(2435\times10^3 \text{ J/kg})(0.663 \text{ W/m}\cdot°\text{C})^3}{(0.404\times10^{-3} \text{ kg/m}\cdot\text{s})(100-37.5)°\text{C}(0.012 \text{ m})}\right]^{1/4} \\ &= 9916 \text{ W/m}^2\cdot°\text{C} \end{aligned}$$

The characteristic length and the Biot number for the lumped system analysis is (see Chap. 4)

$$L_c = \frac{\boldsymbol{V}}{A} = \frac{\pi D^3/6}{\pi D^2} = \frac{D}{6} = \frac{0.012 \text{ m}}{6} = 0.002 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(9916 \text{ W/m}^2\cdot°\text{C})(0.002 \text{ m})}{(429 \text{ W/m}\cdot°\text{C})} = 0.0462 < 0.1$$

The lumped system analysis is applicable since Bi < 0.1. Then the time needed for the temperature of the sphere to rise from 25 to 50°C is determined to be

$$b = \frac{hA_s}{\rho c_p \boldsymbol{V}} = \frac{h}{\rho c_p L_c} = \frac{9916 \text{ W/m}^2\cdot°\text{C}}{(10{,}500 \text{ kg/m}^3)(235 \text{ J/kg}\cdot°\text{C})(0.002 \text{ m})} = 2.009 \text{ s}^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt} \longrightarrow \frac{50-100}{25-100} = e^{-2.009t} \longrightarrow t = \mathbf{0.202 \ s}$$

The total heat transfer to the ball and the amount of steam that condenses become

$$m_{\text{sphere}} = \rho\boldsymbol{V} = \rho\frac{\pi D^3}{6} = (10{,}500 \text{ kg/m}^3)\frac{\pi(0.012 \text{ m})^3}{6} = 0.009500 \text{ kg}$$

$$Q = mc_p[T(t)-T_i]_{\text{sphere}} = (0.009500 \text{ kg})(235 \text{ J/kg}\cdot°\text{C})(50-25)°\text{C} = 55.81 \text{ J}$$

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{55.81 \text{ J/s}}{2435\times10^3 \text{ J/kg}} = \mathbf{2.29\times10^{-5} \ kg/s}$$

10-114 Steam at a saturation temperature of T_{sat} = 100°C condenses on a suspended silver sphere which is initially at 25°C. The time needed for the temperature of the sphere to rise to 50°C and the amount of steam condenses during this process are to be determined.

Assumptions **1** The temperature of the sphere changes uniformly and thus the lumped system analysis is applicable. **2** The average condensation heat transfer coefficient evaluated for the average temperature can be used for the entire process. **3** Constant properties at room temperature can be used for the silver ball.

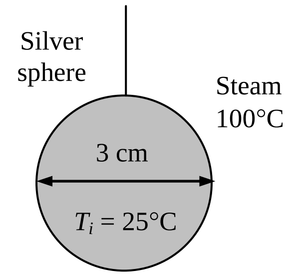
Properties The properties of water at the saturation temperature of 100°C are $h_{fg} = 2257\times10^3$ J/kg and $\rho_v = 0.60$ kg/m^3. The properties of the silver ball at room temperature and the properties of liquid water at the average film temperature of $T_f = (T_{sat} + T_{s,avg})/2 = (100 + 37.5)/2 = 69°C \approx 70°C$ 70°C are (Tables A-3 and A-9),

Copper Ball:

$$\rho = 8933 \text{ kg/m}^3$$
$$\alpha = 117\times10^{-6} \text{ m}^2\text{/s}$$
$$c_p = 385 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 401 \text{ W/m}\cdot°\text{C}$$

Liquid Water:

$$\rho_l = 977.5 \text{ kg/m}^3$$
$$\mu_l = 0.404\times10^{-3} \text{ kg/m}\cdot\text{s}$$
$$c_{pl} = 4190 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 0.663 \text{ W/m}\cdot°\text{C}$$



Analysis The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{sat} - T_s)$$
$$= 2257\times10^3 \text{ J/kg} + 0.68\times4190 \text{ J/kg}\cdot°\text{C}(100-37.5)°\text{C} = 2435\times10^3 \text{ J/kg}$$

Noting that the tube is horizontal, the condensation heat transfer coefficient is determined from

$$h = h_{sph} = 0.815\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat}-T_s)D}\right]^{1/4}$$

$$= 0.815\left[\frac{(9.8 \text{ m/s}^2)(977.5 \text{ kg/m}^3)(977.5-0.60 \text{ kg/m}^3)(2435\times10^3 \text{ J/kg})(0.663 \text{ W/m}\cdot°\text{C})^3}{(0.404\times10^{-3} \text{ kg/m}\cdot\text{s})(100-37.5)°\text{C}(0.03 \text{ m})}\right]^{1/4}$$

$$= 7886 \text{ W/m}^2\cdot°\text{C}$$

The characteristic length and the Biot number for the lumped system analysis is (see Chap. 4)

$$L_c = \frac{V}{A_s} = \frac{\pi D^3/6}{\pi D^2} = \frac{D}{6} = \frac{0.03 \text{ m}}{6} = 0.005 \text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(7886 \text{ W/m}^2\cdot°\text{C})(0.005 \text{ m})}{(401 \text{ W/m}\cdot°\text{C})} = 0.098 < 0.1$$

The lumped system analysis is applicable since Bi < 0.1. Then the time needed for the temperature of the sphere to rise from 25 to 50°C is determined to be

$$b = \frac{hA_s}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{7886 \text{ W/m}^2\cdot°\text{C}}{(8933 \text{ kg/m}^3)(385 \text{ J/kg}\cdot°\text{C})(0.005 \text{ m})} = 0.4586 \text{ s}^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt} \longrightarrow \frac{50-100}{25-100} = e^{-0.4586t} \longrightarrow t = \mathbf{0.884\ s}$$

The total heat transfer to the ball and the amount of steam that condenses become

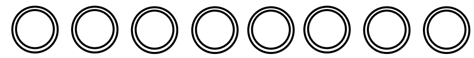
$$m_{sphere} = \rho V = \rho\frac{\pi D^3}{6} = (8933 \text{ kg/m}^3)\frac{\pi(0.03 \text{ m})^3}{6} = 0.1263 \text{ kg}$$

$$Q = mc_p[T(t)-T_i]_{sphere} = (0.1263 \text{ kg})(385 \text{ J/kg}\cdot°\text{C})(50-25)°\text{C} = 1216 \text{ J}$$

$$\dot{m}_{condensation} = \frac{\dot{Q}}{h^*_{fg}} = \frac{1216 \text{ J/s}}{2435\times10^3 \text{ J/kg}} = \mathbf{4.99\times10^{-4}\ kg/s}$$

10-115 There is film condensation on the outer surfaces of 8 horizontal tubes arranged in a horizontal or vertical tier. The ratio of the condensation rate for the cases of the tubes being arranged in a horizontal tier versus in a vertical tier is to be determined.

Assumptions Steady operating conditions exist.



Horizontal tier

Analysis The heat transfer coefficients for the two cases are related to the heat transfer coefficient on a single horizontal tube by

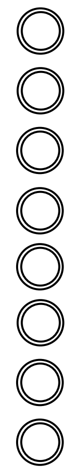
Horizontal tier:

$$h_{\text{horizontal tier of N tubes}} = h_{\text{horizontal, 1 tube}}$$

Vertical tier:

$$h_{\text{vertical tier of N tubes}} = \frac{h_{\text{horizontal, 1 tube}}}{N^{1/4}}$$

Vertical tier

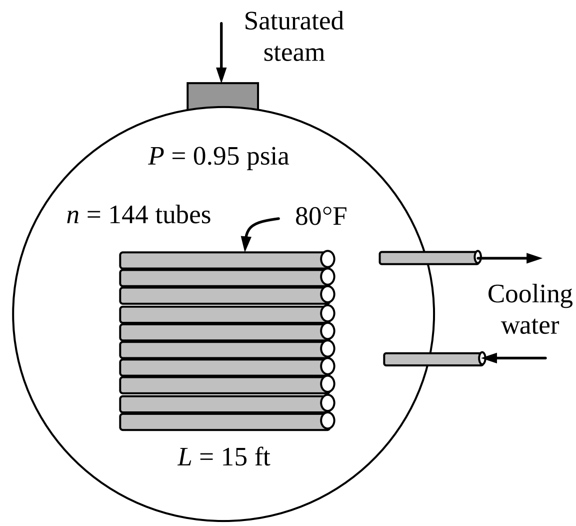


Therefore,

$$\begin{aligned}\text{Ratio} &= \frac{\dot{m}_{\text{horizontal tier of N tubes}}}{\dot{m}_{\text{vertical tier of N tubes}}}\\ &= \frac{h_{\text{horizontal tier of N tubes}}}{h_{\text{vertical tier of N tubes}}}\\ &= \frac{h_{\text{horizontal, 1 tube}}}{h_{\text{horizontal, 1 tube}}/N^{1/4}}\\ &= N^{1/4}\\ &= 8^{1/4} = \mathbf{1.68}\end{aligned}$$

10-116E Saturated steam at a saturation pressure of 0.95 psia and thus at a saturation temperature of T_{sat} = 100°F (Table A-9E) condenses on the outer surfaces of 144 horizontal tubes which are maintained at 80°F by circulating cooling water and arranged in a 12 × 12 square array. The rate of heat transfer to the cooling water and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal.



Properties The properties of water at the saturation temperature of 100°F are h_{fg} = 1037 Btu/lbm and ρ_v = 0.00286 lbm/ft^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (100 + 80)/2 = 90$°F are (Table A-9E),

$$\begin{aligned}
\rho_l &= 62.12\ \text{lbm/ft}^3 \\
\mu_l &= 5.117\times10^{-4}\ \text{lbm/ft}\cdot\text{s} = 1.842\ \text{lbm/ft}\cdot\text{h} \\
\nu_l &= \mu_l/\rho_l = 0.02965\ \text{ft}^2\text{/h} \\
c_{pl} &= 0.999\ \text{Btu/lbm}\cdot°\text{F} \\
k_l &= 0.358\ \text{Btu/h}\cdot\text{ft}\cdot°\text{F}
\end{aligned}$$

Analysis (*a*) The modified latent heat of vaporization is

$$\begin{aligned}
h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{sat} - T_s) \\
&= 1037\ \text{Btu/lbm} + 0.68\times(0.999\ \text{Btu/lbm}\cdot°\text{F})(100-80)°\text{F} \\
&= 1051\ \text{Btu/lbm}
\end{aligned}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$\begin{aligned}
h = h_{\text{horiz}} &= 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat}-T_s)D}\right]^{1/4} \\
&= 0.729\left[\frac{(32.2\ \text{ft/s}^2)(62.12\ \text{lbm/ft}^3)(62.12-0.00286\ \text{lbm/ft}^3)(1051\ \text{Btu/lbm})(0.358\ \text{Btu/h}\cdot\text{ft}\cdot°\text{F})^3}{[(1\ \text{h}/3600\ \text{s})^2](1.842\ \text{lbm/ft}\cdot\text{h})(100-80)°\text{F}(1.2/12\ \text{ft})}\right]^{1/4} \\
&= 1562\ \text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}
\end{aligned}$$

Then the average heat transfer coefficient for a 4-tube high vertical tier becomes

$$h_{\text{horiz, N tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{12^{1/4}}(1562\ \text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}) = 839.2\ \text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$$

The surface area for all 144 tubes is

$$A_s = N_{\text{total}}\pi DL = 144\pi(1.2/12\ \text{ft})(15\ \text{ft}) = 678.6\ \text{ft}^2$$

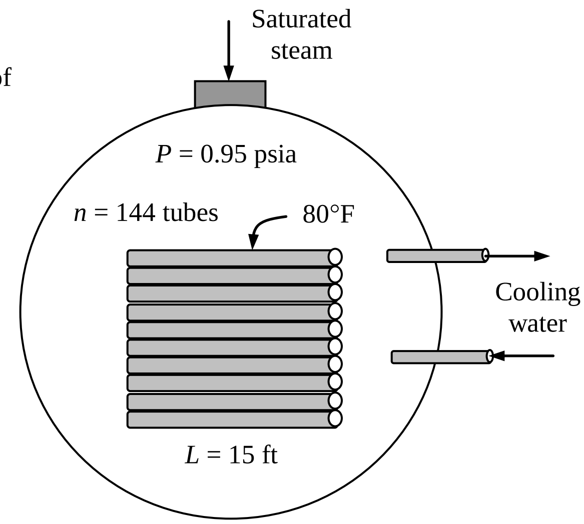
Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat}-T_s) = (839.2\ \text{Btu/h.ft}^2\cdot°\text{F})(678.6\ \text{ft}^2)(100-80)°\text{F} = \mathbf{11{,}390{,}000\ Btu/h}$$

(*b*) The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{11{,}390{,}000\ \text{Btu/h}}{1051\ \text{Btu/lbm}} = \mathbf{10{,}837\ lbm/h}$$

10-117E Saturated steam at a saturation pressure of 0.95 psia and thus at a saturation temperature of T_{sat} = 100°F (Table A-9E) condenses on the outer surfaces of 144 horizontal tubes which are maintained at 80°F by circulating cooling water and arranged in a 12 × 12 square array. The rate of heat transfer to the cooling water and the rate of condensation of steam are to be determined.



Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal.

Properties The properties of water at the saturation temperature of 100°F are h_{fg} = 1037 Btu/lbm and ρ_v = 0.00286 lbm/ft^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 =$ (100 + 80)/2 = 90°F are (Table A-9E),

$$\begin{aligned}
\rho_l &= 62.12\ \text{lbm/ft}^3 \\
\mu_l &= 5.117\times10^{-4}\ \text{lbm/ft}\cdot\text{s} = 1.842\ \text{lbm/ft}\cdot\text{h} \\
\nu_l &= \mu_l/\rho_l = 0.02965\ \text{ft}^2/\text{h} \\
c_{pl} &= 0.999\ \text{Btu/lbm}\cdot{}^\circ\text{F} \\
k_l &= 0.358\ \text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}
\end{aligned}$$

Analysis (*a*) The modified latent heat of vaporization is

$$\begin{aligned}
h^*_{fg} &= h_{fg} + 0.68c_{pl}(T_{sat} - T_s) \\
&= 1037\ \text{Btu/lbm} + 0.68\times(0.999\ \text{Btu/lbm}\cdot{}^\circ\text{F})(100-80)^\circ\text{F} \\
&= 1051\ \text{Btu/lbm}
\end{aligned}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$\begin{aligned}
h = h_{\text{horiz}} &= 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat}-T_s)D}\right]^{1/4} \\
&= 0.729\left[\frac{(32.2\ \text{ft/s}^2)(62.12\ \text{lbm/ft}^3)(62.12-0.00286\ \text{lbm/ft}^3)(1051\ \text{Btu/lbm})(0.358\ \text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F})^3}{[(1\ \text{h}/3600\ \text{s})^2](1.842\ \text{lbm/ft}\cdot\text{h})(100-80)^\circ\text{F}(2.0/12\ \text{ft})}\right]^{1/4} \\
&= 1375\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}
\end{aligned}$$

Then the average heat transfer coefficient for a 4-tube high vertical tier becomes

$$h_{\text{horiz, N tubes}} = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{12^{1/4}}(1375\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}) = 738.8\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

The surface area for all 144 tubes is

$$A_s = N_{\text{total}}\pi DL = 144\pi(2/12\ \text{ft})(15\ \text{ft}) = 1131\ \text{ft}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat} - T_s) = (738.8\ \text{Btu/h.ft}^2\cdot{}^\circ\text{F})(1131\ \text{ft}^2)(100-80)^\circ\text{F} = \mathbf{16{,}712{,}000\ Btu/h}$$

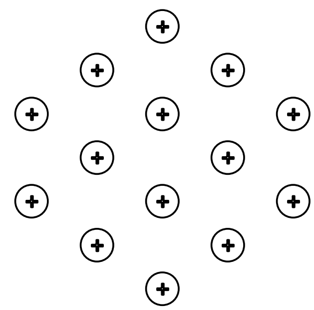
(*b*) The rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h^*_{fg}} = \frac{16{,}712{,}000\ \text{Btu/h}}{1051\ \text{Btu/lbm}} = \mathbf{15{,}900\ lbm/h}$$

10-118 Ammonia is liquefied in a horizontal condenser at 37°C by a coolant at 20°C. The average value of overall heat transfer coefficient and the tube length are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal. **3** The thermal resistance of the tube walls is negligible.

Properties The properties of ammonia at the saturation temperature of 310 K (37°C) are h_{fg} = 1113×10^3 J/kg and ρ_v = 11.09 kg/m^3 (Table A-11). We assume a tube outer surface temperature of 31°C. The properties of liquid ammonia at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (37 + 31)/2 = 34$°C are (Table A-11)



$$\rho_l = 589.0 \text{ kg/m}^3$$
$$\mu_l = 1.303\times10^{-4} \text{ kg/m}\cdot\text{s}$$
$$c_{pl} = 4867 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 0.4602 \text{ W/m}\cdot°\text{C}$$

The thermal conductivity of copper is 401 W/m·°C (Table A-3).

Analysis (*a*) The modified latent heat of vaporization is

$$h^*_{fg} = h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s)$$
$$= 1113\times10^3 \text{ J/kg} + 0.68\times4867 \text{ J/kg}\cdot°\text{C}(37-31)°\text{C} = 1133\times10^3 \text{ J/kg}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$h = h_{\text{horizontal}} = 0.729\left[\frac{g\rho_l(\rho_l-\rho_v)h^*_{fg}k_l^3}{\mu_l(T_{\text{sat}}-T_s)D}\right]^{1/4}$$
$$= 0.729\left[\frac{(9.8 \text{ m/s}^2)(589.0 \text{ kg/m}^3)(589.0-11.09 \text{ kg/m}^3)(1133\times10^3 \text{ J/kg})(0.4602 \text{ W/m}\cdot°\text{C})^3}{(1.303\times10^{-4} \text{ kg/m}\cdot\text{s})(37-31)°\text{C}(0.038 \text{ m})}\right]^{1/4}$$
$$= 7693 \text{ W/m}^2\cdot°\text{C}$$

Noting that there are two 2-pipe high, two 3-pipe high, and one 4-pipe high vertical tiers in the tube-layout, the average heat transfer coefficient is to be determined as follows

$$h_1 = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{2^{1/4}}(7693 \text{ W/m}^2\cdot°\text{C}) = 6469 \text{ W/m}^2\cdot°\text{C}$$
$$h_2 = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{3^{1/4}}(7693 \text{ W/m}^2\cdot°\text{C}) = 5845 \text{ W/m}^2\cdot°\text{C}$$
$$h_1 = \frac{1}{N^{1/4}}h_{\text{horiz,1 tube}} = \frac{1}{4^{1/4}}(7693 \text{ W/m}^2\cdot°\text{C}) = 5440 \text{ W/m}^2\cdot°\text{C}$$
$$h_o = \frac{2\times2h_1 + 2\times3h_2 + 1\times4h_3}{2\times2+2\times3+1\times4} = \frac{4\times6469+6\times5845+4\times5440}{14} = 5908 \text{ W/m}^2\cdot°\text{C}$$

Let us check if the assumed value for the rube temperature was reasonable

$$h_iA_i\Delta T_i = h_oA_o\Delta T_o$$
$$(4000)\pi(0.030)L(T_{\text{tube}}-20) = (5908)\pi(0.038)L(37-T_{\text{tube}}) \longrightarrow T_{\text{tube}} = 31.1°\text{C}$$

which is very close to the assumed value of 31°C. Therefore, the assumption was good. The overall heat transfer coefficient based on the outer surface is determined from

$$U_o = \left(\frac{D_o}{D_ih_i} + \frac{D_o\ln(D_o/D_i)}{2k} + \frac{1}{h_o}\right)^{-1} = \left(\frac{0.038}{0.030\times4000} + \frac{0.038\ln(3.8/3.0)}{2(401)} + \frac{1}{5908}\right)^{-1} = \mathbf{2012\ W/m^2\cdot°C}$$

(*b*) The rate of heat transfer is

$$\dot{Q} = \dot{m}_{\text{condensation}}h^*_{fg} = (900/3600 \text{ kg/s})(1133\times10^3 \text{ J/kg}) = 2.833\times10^5 \text{ W}$$

Then the tube length may be determined from

$$\dot{Q} = U_oA_o\Delta T$$
$$2.833\times10^5 \text{ W} = (2012 \text{ W/m}^2\cdot\text{C})(14)\pi(0.038 \text{ m})L(37-20) \longrightarrow L = \mathbf{4.96\ m}$$

10-119 Saturated ammonia vapor at a saturation temperature of T_{sat} = 25°C condenses on the outer surfaces of a tube bank in which cooling water flows. The rate of condensation of ammonia, the overall heat transfer coefficient, and the tube length are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tubes are isothermal. **3** The thermal resistance of the tube walls is negligible.

Properties The properties of ammonia at the saturation temperature of 25°C are h_{fg} = 1166×10^3 J/kg and ρ_v = 7.809 kg/m^3 (Table A-11). We assume that the tube temperature is 20°C. Then, the properties of liquid ammonia at the film temperature of $T_f = (T_{sat} + T_s)/2 = (25 + 20)/2 = 22.5$°C are (Table A-11)

$$\rho_l = 606.5 \text{ kg/m}^3$$
$$\mu_l = 1.479 \times 10^{-4} \text{ kg/m}\cdot\text{s}$$
$$c_{pl} = 4765 \text{ J/kg}\cdot°\text{C}$$
$$k_l = 0.4869 \text{ W/m}\cdot°\text{C}$$

The water properties at the average temperature of (14+17)/2 = 15.5°C are (Table A-9)

$$\rho = 999.0 \text{ kg/m}^3$$
$$c_p = 4185 \text{ J/kg}\cdot°\text{C}$$
$$\mu = 1.124 \times 10^{-3} \text{ kg/m}\cdot\text{s}$$
$$k = 0.590 \text{ W/m}\cdot°\text{C}$$
$$\text{Pr} = 7.98$$

Analysis (*a*) The modified latent heat of vaporization is

$$\begin{aligned} h^*_{fg} &= h_{fg} + 0.68 c_{pl}(T_{sat} - T_s) \\ &= 1166 \times 10^3 \text{ J/kg} + 0.68 \times 4765 \text{ J/kg}\cdot°\text{C}(25-20)°\text{C} \\ &= 1182 \times 10^3 \text{ J/kg} \end{aligned}$$

The heat transfer coefficient for condensation on a single horizontal tube is

$$\begin{aligned} h = h_{\text{horizontal}} &= 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)h^*_{fg}k_l^3}{\mu_l(T_{sat} - T_s)D}\right]^{1/4} \\ &= 0.729\left[\frac{(9.8 \text{ m/s}^2)(606.5 \text{ kg/m}^3)(606.5 - 7.809 \text{ kg/m}^3)(1182\times10^3 \text{ J/kg})(0.4869 \text{ W/m}\cdot°\text{C})^3}{(1.479\times10^{-4} \text{ kg/m}\cdot\text{s})(25-20)°\text{C}(0.025\text{ m})}\right]^{1/4} \\ &= 9280 \text{ W/m}^2\cdot°\text{C} \end{aligned}$$

Then the average heat transfer coefficient for a 4-pipe high vertical tier becomes

$$h_o = h_{\text{horiz, N tubes}} = \frac{1}{N^{1/4}} h_{\text{horiz,1 tube}} = \frac{1}{4^{1/4}}(9280 \text{ W/m}^2\cdot°\text{C}) = 6562 \text{ W/m}^2\cdot°\text{C}$$

The rate of heat transfer in the condenser is

$$\dot{m} = 16\rho A_c \boldsymbol{V} = 16(999 \text{ kg/m}^3)\pi(0.25)(0.025\text{ m})^2(2 \text{ m/s}) = 15.69 \text{ kg/s}$$
$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (15.69 \text{ kg/s})(4185 \text{ J/kg}\cdot°\text{C})(17-14) = 1.970\times10^5 \text{ W}$$

Then the rate of condensation becomes

$$\dot{m}_{cond} = \frac{\dot{Q}}{h^*_{fg}} = \frac{1.970\times10^5 \text{ W}}{1182\times10^3 \text{ J/kg}} = \mathbf{0.167 kg/s}$$

(*b*) For the calculation of the heat transfer coefficient on the inner surfaces of the tubes, we first determine the Reynolds number

$$\text{Re} = \frac{VD\rho}{\mu} = \frac{(2\text{ m/s})(0.025\text{ m})(999.0\text{ kg/m}^3)}{1.124\times10^{-3}\text{ kg/m}\cdot\text{s}} = 44{,}440$$

which is greater than 10,000. Therefore, the flow is turbulent. Assuming fully developed flow, the Nusselt number and the heat transfer coefficient are determined to be

$$Nu = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(44{,}440)^{0.8}(7.98)^{0.4} = 275.9$$

$$h_i = \frac{k}{D}Nu = \frac{(0.590\text{ W/m}\cdot{}^\circ\text{C})}{0.025\text{ m}}(275.9) = 6511\text{ W/m}^2\cdot{}^\circ\text{C}$$

Let us check if the assumed value for the rube temperature was reasonable

$$h_i\Delta T_i = h_o\Delta T_o$$
$$(6511)(T_{\text{tube}} - 15.5) = (6562)(25 - T_{\text{tube}})$$
$$T_{\text{tube}} = 20.3^\circ\text{C}$$

which is sufficiently close to the assumed value of 20°C. Disregarding thermal resistance of the tube walls, the overall heat transfer coefficient is determined from

$$U = \left(\frac{1}{h_i} + \frac{1}{h_o}\right)^{-1} = \left(\frac{1}{6511} + \frac{1}{6562}\right)^{-1} = \mathbf{3268\text{ W/m}^2\cdot{}^\circ C}$$

(*c*) The tube length may be determined from

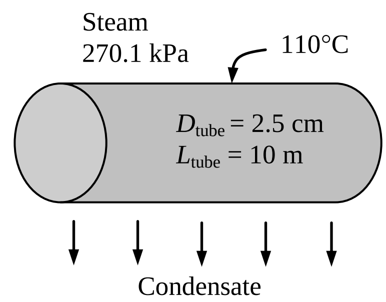
$$\dot{Q} = UA\Delta T$$
$$1.970\times10^5\text{ W} = (3268\text{ W/m}^2\cdot\text{C})(16)\pi(0.025\text{ m})L\left[25 - \frac{1}{2}(14+17)\right]$$
$$L = \mathbf{5.05\text{ m}}$$

10-120 Saturated steam at 270.1 kPa pressure and thus at a saturation temperature of T_{sat} = 130°C (Table A-9) condenses inside a horizontal tube which is maintained at 110°C. The average heat transfer coefficient and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The tube is isothermal. **3** The vapor velocity is low so that Re_{vapor} < 35,000.

Properties The properties of water at the saturation temperature of 130°C are $h_{fg} = 2174\times10^3$ J/kg and $\rho_v = 1.50$ kg/m^3. The properties of liquid water at the film temperature of $T_f = (T_{sat} + T_s)/2 = (130 + 110)/2 = 120°C$ are (Table A-9),

$$\rho_l = 943.4\ \text{kg/m}^3$$
$$\mu_l = 0.232\times10^{-3}\ \text{kg/m.s}$$
$$\nu_l = \mu_l/\rho_l = 0.246\times10^{-6}\ \text{m}^2/\text{s}$$
$$c_{pl} = 4244\ \text{J/kg.°C}$$
$$k_l = 0.683\ \text{W/m.°C}$$



Analysis The condensation heat transfer coefficient is determined from

$$h = h_{internal} = 0.555\left[\frac{g\rho_l(\rho_l-\rho_v)k_l^3}{\mu_l(T_{sat}-T_s)D}\left(h_{fg}+\frac{3}{8}c_{pl}(T_{sat}-T_s)\right)\right]^{1/4}$$

$$= 0.555\left[\frac{(9.8\ \text{m/s}^2)(943.4\ \text{kg/m}^3)(943.4-1.50)\ \text{kg/m}^3)(0.683\ \text{W/m}\cdot°\text{C})^3}{(0.232\times10^{-3}\ \text{kg/m}\cdot\text{s})(130-110)°\text{C}(0.025\ \text{m})}\right.$$
$$\left.\times\left(2174\times10^3\ \text{J/kg} + \frac{3}{8}(4244\ \text{J/kg}\cdot°\text{C})(130-110)°\text{C}\right)\right]^{1/4}$$
$$= \mathbf{8413\ W/m^2\cdot°C}$$

The heat transfer surface area of the pipe is

$$A_s = \pi DL = \pi(0.025\ \text{m})(10\ \text{m}) = 0.7854\ \text{m}^2$$

Then the rate of heat transfer during this condensation process becomes

$$\dot{Q} = hA_s(T_{sat}-T_s) = (8413\ \text{W/m}^2\cdot°\text{C})(0.7854\ \text{m}^2)(130-110)°\text{C} = 132{,}151\ \text{W}$$

The rate of condensation of steam is determined from

$$\dot{m}_{condensation} = \frac{\dot{Q}}{h_{fg}} = \frac{132{,}151\ \text{J/s}}{2174\times10^3\ \text{J/kg}} = \mathbf{0.0608\ kg/s}$$

Fundamentals of Engineering (FE) Exam Problems

10-121 When boiling a saturated liquid, one must be careful while increasing the heat flux to avoid "burnout." Burnout occurs when the boiling transitions from _____ boiling.

(a) convection to nucleate (b) convection to film (c) film to nucleate

(d) nucleate to film (e) none of them

Answer (d) nucleate to film

10-122 Heat transfer coefficients for a vapor condensing on a surface can be increased by promoting

(a) film condensation (b) dropwise condensation (c) rolling action (d) none of them

Answer (b) dropwise condensation

10-123 At a distance x down a vertical, isothermal flat plate on which a saturated vapor is condensing in a continuous film, the thickness of the liquid condensate layer is δ. The heat transfer coefficient at this location on the plate is given by

(a) k_l / δ (b) δh_f (c) δh_{fg} (d) δh_g (e) none of them

Answer (a) k_l / δ

10-124 When a saturated vapor condenses on a vertical, isothermal flat plate in a continuous film, the rate of heat transfer is proportional to

(a) $(T_s - T_{\text{sat}})^{1/4}$ (b) $(T_s - T_{\text{sat}})^{1/2}$ (c) $(T_s - T_{\text{sat}})^{3/4}$ (d) $(T_s - T_{\text{sat}})$ (e) $(T_s - T_{\text{sat}})^{2/3}$

Answer (c) $(T_s - T_{\text{sat}})^{3/4}$

10-125 Saturated water vapor is condensing on a 0.5 m^2 vertical flat plate in a continuous film with an average heat transfer coefficient of 7 kW/m^2·K. The temperature of the water is 80^{o}C (h_{fg} = 2309 kJ/kg) and the temperature of the plate is 60^{o}C. The rate at which condensate is being formed is

(a) 0.0303 kg/s (b) 0.07 kg/s (c) 0.15 kg/s (d) 0.24 kg/s (e) 0.28 kg/s

Answer (a) 0.0303 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
hfg=2309 [kJ/kg]
dT=20 [C]
A=0.5 [m^2]
h=7 [kJ/m^2-K-s]
mdot=h*A*dT/hfg
```

10-126 Steam condenses at 50ºC on a 0.8-m-high and 2.4-m-wide vertical plate that is maintained at 30ºC. The condensation heat transfer coefficient is

(a) 3975 W/m^2·ºC (b) 5150 W/m^2·ºC (c) 8060 W/m^2·ºC (d) 11,300 W/m^2·ºC (e) 14,810 W/m^2·ºC

(For water, use ρ_l = 992.1 kg/m^3, μ_l = 0.653×10^{-3} kg/m·s, k_l = 0.631 W/m·°C, c_{pl} = 4179 J/kg·°C, $h_{fg\ @\ Tsat}$ = 2383 kJ/kg)

Answer (b) 5150 W/m^2·ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_sat=50 [C]
T_s=30 [C]
L=0.8 [m]
w=2.4 [m]
h_fg=2383E3 [J/kg] "at 50 C from Table A-9"
"The properties of water at (50+30)/2=40 C are (Table A-9)"
rho_l=992.1 [kg/m^3]
mu_l=0.653E-3 [kg/m-s]
nu_l=mu_l/rho_l
c_p_l=4179 [J/kg-C]
k_l=0.631 [W/m-C]
g=9.81 [m/s^2]
h_fg_star=h_fg+0.68*c_p_l*(T_sat-T_s)
Re=(4.81+(3.70*L*k_l*(T_sat-T_s))/(mu_l*h_fg_star)*(g/nu_l^2)^(1/3))^0.820
"Re is between 30 and 1800, and therefore there is wavy laminar flow"
h=(Re*k_l)/(1.08*Re^1.22-5.2)*(g/nu_l^2)^(1/3)
```

10-127 An air conditioner condenser in an automobile consists of 2 m^2 of tubular heat exchange area whose surface temperature is 30°C. Saturated refrigerant 134a vapor at 50°C (h_{fg} = 152 kJ/kg) condenses on these tubes. What heat transfer coefficient must exist between the tube surface and condensing vapor to produce 1.5 kg/min of condensate?

(a) 95 W/m^2·K (b) 640 W/m^2·K (c) 727 W/m^2·K (d) 799 W/m^2·K (e) 960 W/m^2·K

Answer (a) 95 W/m^2·K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
hfg=152000 [J/kg]
dT=20 [C]
A=2 [m^2]
mdot=(1.5/60) [kg/s]
Q=mdot*hfg
Q=h*A*dT
```

10-128 Saturated water vapor at 40°C is to be condensed as it flows through a tube at a rate of 0.2 kg/s. The condensate leaves the tube as a saturated liquid at 40°C. The rate of heat transfer from the tube is

(a) 34 kJ/s (b) 268 kJ/s (c) 453 kJ/s (d) 481 kJ/s (e) 515 kJ/s

Answer (d) 481 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T1=40 [C]
m_dot=0.2 [kg/s]
h_f=ENTHALPY(Steam_IAPWS,T=T1,x=0)
h_g=ENTHALPY(Steam_IAPWS,T=T1,x=1)
h_fg=h_g-h_f
Q_dot=m_dot*h_fg

"Wrong Solutions:"
W1_Q=m_dot*h_f  "Using hf"
W2_Q=m_dot*h_g "Using hg"
W3_Q=h_fg "not using mass flow rate"
W4_Q=m_dot*(h_f+h_g) "Adding hf and hg"
```

10-129 Steam condenses at 50ºC on the outer surface of a horizontal tube with an outer diameter of 6 cm. The outer surface of the tube is maintained at 30ºC. The condensation heat transfer coefficient is

(a) 5493 W/m^2·ºC (b) 5921 W/m^2·ºC (c) 6796 W/m^2·ºC (d) 7040 W/m^2·ºC (e) 7350 W/m^2·ºC

(For water, use ρ_l = 992.1 kg/m^3, μ_l = 0.653×10^{-3} kg/m·s, k_l = 0.631 W/m·°C, c_{pl} = 4179 J/kg·°C, $h_{fg\ @\ Tsat}$ = 2383 kJ/kg)

Answer (c) 6796 W/m^2·ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_sat=50 [C]
T_s=30 [C]
D=0.06 [m]
h_fg=2383E3 [J/kg] "at 50 C from Table A-9"
rho_v=0.0831 [kg/m^3]
"The properties of water at (50+30)/2=40 C are (Table A-9)"
rho_l=992.1 [kg/m^3]
mu_l=0.653E-3 [kg/m-s]
c_p_l=4179 [J/kg-C]
k_l=0.631 [W/m-C]
g=9.81 [m/s^2]
h_fg_star=h_fg+0.68*c_p_l*(T_sat-T_s)
h=0.729*((g*rho_l*(rho_l-rho_v)*h_fg_star*k_l^3)/(mu_l*(T_sat-T_s)*D))^0.25
```

10-130 Steam condenses at 50ºC on a tube bank consisting of 20 tubes arranged in a rectangular array of 4 tubes high and 5 tubes wide. Each tube has a diameter of 6 cm and a length of 3 m and the outer surfaces of the tubes are maintained at 30ºC. The rate of condensation of steam is

(a) 0.054 kg/s (b) 0.076 kg/s (c) 0.315 kg/s (d) 0.284 kg/s (e) 0.446 kg/s

(For water, use ρ_l = 992.1 kg/m^3, μ_l = 0.653×10^{-3} kg/m·s, k_l = 0.631 W/m·°C, c_{pl} = 4179 J/kg·°C, $h_{fg\ @\ Tsat}$ = 2383 kJ/kg)

Answer (c) 0.315 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_sat=50 [C]
T_s=30 [C]
D=0.06 [m]
L=3 [m]
N=4
N_total=5*N

h_fg=2383E3 [J/kg] "at 50 C from Table A-9"
rho_v=0.0831 [kg/m^3] "at 50 C from Table A-9"
"The properties of water at (50+30)/2=40 C are (Table A-9)"
rho_l=992.1 [kg/m^3]
mu_l=0.653E-3 [kg/m-s]
c_p_l=4179 [J/kg-C]
k_l=0.631 [W/m-C]
g=9.81 [m/s^2]
h_fg_star=h_fg+0.68*c_p_l*(T_sat-T_s)
h_1tube=0.729*((g*rho_l*(rho_l-rho_v)*h_fg_star*k_l^3)/(mu_l*(T_sat-T_s)*D*N))^0.25
h_Ntubes=1/N^0.25*h_1tube
A_s=N_total*pi*D*L
Q_dot=h_Ntubes*A_s*(T_sat-T_s)
m_dot_cond=Q_dot/h_fg_star
```

10-131 ... 10-136 Design and Essay Problems