Quiz HT107

BOILING AND CONDENSATION

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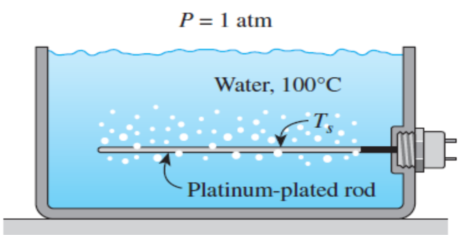
Problems

Problem 1 (Çengel & Ghajar, 2015, w/ permission)

Water is boiled at atmospheric pressure by a horizontal platinum-plated rod with diameter of 10 mm. If the surface temperature of the rod is maintained at 110°C, determine the nucleate pool boiling heat transfer coefficient. Use as properties enthalpy of vaporization h_{fg} = 2257 kJ/kg, density of water ρ_l = 957.9 kg/m^3, viscosity of water μ_l = 0.282×10^{-3} kg·m/s, specific heat of water c_{pl} = 4217 J/kg°C, Prandtl number of water Pr_l = 1.75, and density of water vapor ρ_v = 0.60 kg/m^3.



A) h = 5400 W/m^{2}K

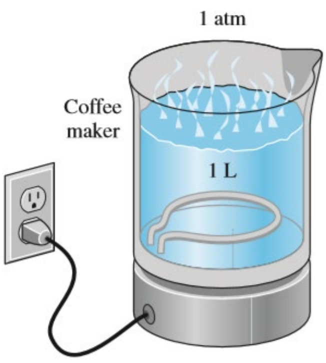
B) h = 8090 W/m^{2}K

C) h = 11,200 W/m^{2}K

D) h = 14,080 W/m^{2}K

Problem 2A (Çengel & Ghajar, 2015, w/ permission)

Water is boiled at sea level in a coffee maker equipped with a 20-cm-long, 0.4-cm-diameter immersion-type electric heating element made of mechanically polished stainless steel. The coffee maker initially contains 1L of water at 14°C. Once boiling starts, it is observed that half of the water in the coffee maker evaporates in 25 min. Determine the power rating of the electric heating element immersed in the water and the surface temperature of the heating element. Use as properties enthalpy of vaporization h_{fg} = 2257 kJ/kg, density of water ρ_l = 957.9 kg/m^3, density of water vapor ρ_v = 0.598 kg/m^3, specific heat of water c_{pl} = 4217 J/kg°C, and Prandtl number of water Pr = 1.75.



A) $\dot{Q}$ = 150 kW/m^2 and T_s = 113°C
B) $\dot{Q}$ = 150 kW/m^2 and T_s = 132°C
C) $\dot{Q}$ = 300 kW/m^2 and T_s = 113°C
D) $\dot{Q}$ = 300 kW/m^2 and T_s = 132°C

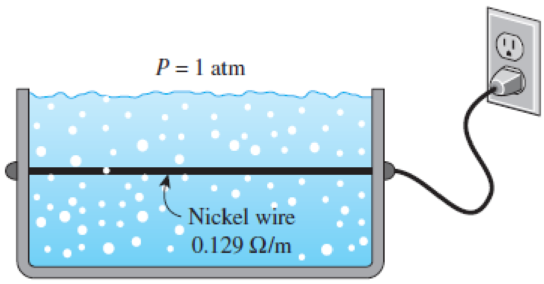
Problem 2B

Determine how long it will take for the heater to raise the temperature of 1 L of cold water from 14°C to the boiling temperature. Use c_{pl} = 4.184 kJ/kg as the mean specific heat of liquid water.

A) Δt = 1.82 min
B) Δt = 3.51 min
C) Δt = 5.65 min
D) Δt = 7.97 min

Problem 3 (Çengel & Ghajar, 2015, w/ permission)

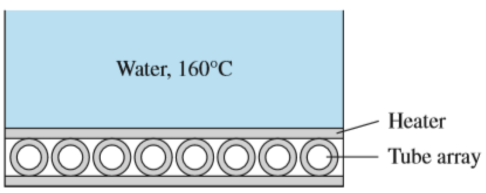
A 1-mm diameter nickel wire with electrical resistance of 0.129 Ω/m is submerged horizontally in water at atmospheric pressure. Determine the electrical current at which the wire would be in danger of burnout in nucleate boiling. Use as properties density of water ρ_l = 957.9 kg/m^3, density of water vapor ρ_v = 0.60 kg/m^3, and enthalpy of vaporization h_{fg} = 2257 kJ/kg.



A) I = 90.6 A
B) I = 142 A
C) I = 193 A
D) I = 244 A

Problem 4 (Çengel & Ghajar, 2015, w/ permission)

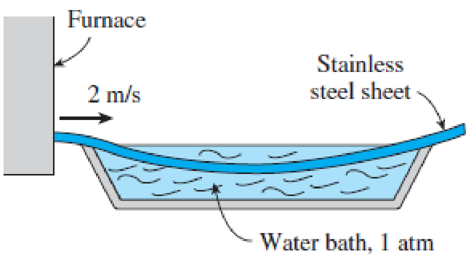
Hot gases flow inside an array of tubes that are embedded in a 3 m × 3 m horizontal flat heater. The heater is used for boiling water in a tank at 160°C. With the interest of avoiding burnout, determine the maximum rate of water vaporization that can be achieved by the heater. Use as properties density of water ρ_l = 907.4 kg/m^3, density of water vapor ρ_v = 3.26 kg/m^3, and enthalpy of vaporization h_{fg} = 2083 kJ/kg.



A) $\dot{m}_{\text{vapor}}$ = 5.45 kg/s
B) $\dot{m}_{\text{vapor}}$ = 10.9 kg/s
C) $\dot{m}_{\text{vapor}}$ = 15.4 kg/s
D) $\dot{m}_{\text{vapor}}$ = 20.6 kg/s

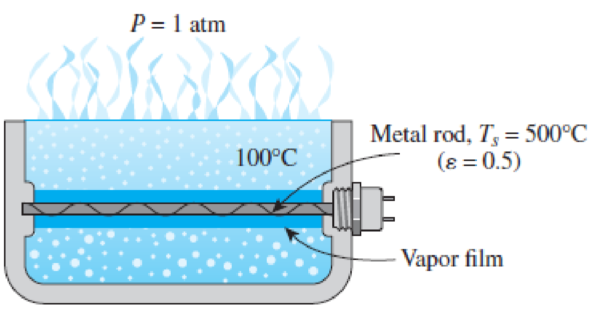
Problem 5 (Çengel & Ghajar, 2015, w/ permission)

A long mechanically polished stainless steel sheet is being conveyed at 2 m/s through a water bath to be cooled. The 0.5-m-wide and 5-mm-thick stainless steel sheet has a temperature of 125°C as it enters the water bath. The length of the sheet submerged in water is 1 m as it is being conveyed through the water bath. As the hot sheet enters the water bath, boiling would occur at 1 atm. In order to prevent thermal burn on people handling the sheet, it must exit the water bath at a temperature below 45°C. Is the rate of heat that could be removed from the stainless steel sheet in the water bath sufficient to cool it to below 45°C as it leaves the water bath? Use as properties density of liquid water ρ_l = 957.9 kg/m^3, density of water vapor ρ_v = 0.60 kg/m^3, enthalpy of vaporization h_{fg} = 2257 kJ/kg, viscosity of water μ_l = 0.282 × 10^{-3} kg/m·s, specific heat of water c_{pl} = 4217 J/kg, and Prandtl number of water Pr_l = 1.75. Use ρ_{ss} = 7900 kg/m^3 , $c_{p,ss}$ = 450 J/kg·K, and ε = 0.5 as the density, specific heat and emissivity of steel, respectively.



Problem 6 (Çengel & Ghajar, 2015, w/ permission)

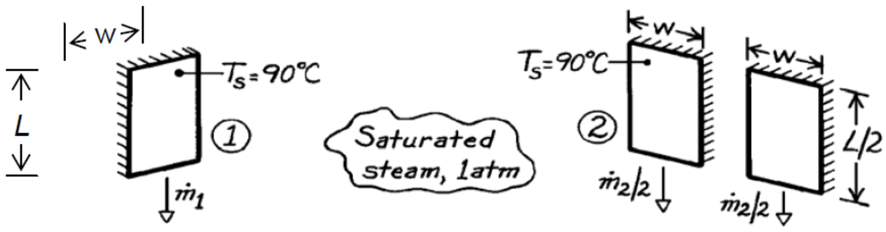
A 2-mm-diameter cylindrical metal rod with emissivity of 0.5 is submerged horizontally in water under atmospheric pressure. When electric current is passed through the metal rod, the surface temperature reaches 500°C. Determine the power dissipation per unit length of the metal rod. Use properties density of water ρ_l = 957.9 kg/m^3, density of water vapor ρ_v = 0.383 kg/m^3, specific heat of water vapor c_{pv} = 1997 kJ/kg, thermal conductivity of water vapor k_v = 0.0435 W/mK, and viscosity of water vapor μ_v = 2.05×10^{-5} kg/m·s.



A) $\dot{Q}'$ = 454 W/m
B) $\dot{Q}'$ = 769 W/m
C) $\dot{Q}'$ = 1080 W/m
D) $\dot{Q}'$ = 1400 W/m

Problem 7 (Bergman et al., 2011, w/ permission)

Two configurations are being considered in the design of a condensing system for steam at 1 atm employing a vertical plate maintained at 90°C. The first configuration is a single vertical plate L×w and the second consists of two vertical plates (L/2)×w, where L and w are the vertical and horizontal dimensions, respectively. Which configuration would you choose? If need be for a numerical comparison, use as properties enthalpy h_{fg} = 2257 kJ/kg for saturated water vapor, density of water ρ_l = 962 kg/m^3, dynamic viscosity of water μ_l = 296×10^{-6} N·s/m^2, thermal conductivity of water k_l = 0.678 W/mK, specific heat of water $c_{p,l}$ = 4212 J/kg·K, and kinematic viscosity of water ν_l = 3.08×10^{-7} m^2/s.



Problem 8A (Bergman et al., 2011, w/ permission)

A 2 m × 2 m vertical plate is exposed on one side to saturated steam at atmospheric pressure and on the other side to cooling water that maintains a plate temperature of 50°C. What is the rate of heat transfer to the water? What is the rate at which steam condenses on the plate? Use as properties enthalpy of vaporization h_{fg} = 2257 kJ/kg, density of water ρ_l = 975 kg/m^{3}dynamic viscosity of water μ_l = 375×10^{-6} N·s/m^2, thermal conductivity of water k_l = 0.668 W/mK, specific heat of water $c_{p,l}$ = 4193 J/kg·K, kinematic viscosity of water ν_l = 3.85×10^{-7} m^2/s, and Prandtl number of water Pr_l = 2.35.

A) $\dot{q}$ = 0.52 MW and $\dot{m}$ = 0.433 kg/s
B) $\dot{q}$ = 0.52 MW and $\dot{m}$ = 0.866 kg/s
C) $\dot{q}$ = 1.04 MW and $\dot{m}$ = 0.433 kg/s
D) $\dot{q}$ = 1.04 MW and $\dot{m}$ = 0.866 kg/s

Problem 8B

For plates inclined at an angle θ from the vertical, the average convection coefficient for condensation on the upper surface, $\bar{h}_{L,\text{inc}}$, may be approximated by an expression of the form $\bar{h}_{L,\text{inc}} \approx \bar{h}_{L,\text{vert}}(\cos\theta)^{1/4}$, where $h_{L,\text{vert}}$ is the average coefficient for the vertical orientation. If the 2 m × 2 m plate is inclined 45° from the normal, what are the rates of heat transfer and condensation?

A) $\dot{q}$ = 0.46 MW and $\dot{m}$ = 0.397 kg/s
B) $\dot{q}$ = 0.46 MW and $\dot{m}$ = 0.791 kg/s
C) $\dot{q}$ = 0.95 MW and $\dot{m}$ = 0.397 kg/s
D) $\dot{q}$ = 0.95 MW and $\dot{m}$ = 0.791 kg/s

Problem 9A (Bergman et al., 2011, w/ permission)

Saturated steam at 1 atm condenses on the outer surface of a vertical, 100-mm-diameter pipe 1 m long, having a uniform surface temperature of 94°C. Estimate the heat transfer rate to the pipe and the condensation rate. Use as properties enthalpy of vaporization h_{fg} = 2257 kJ/kg, specific heat of water c_{pl} = 4214 J/kg·K, thermal conductivity of water k_l = 0.679 W/mK, dynamic viscosity of water μ_l = 289 ×10^{-6} N·s/m, and kinematic viscosity of water v_l = 3.01×10^{-7} m^2/s.

A) $\dot{q}$ = 16.0 kW and $\dot{m}$ = 7.04 g/s
B) $\dot{q}$ = 16.0 kW and $\dot{m}$ = 12.4 g/s
C) $\dot{q}$ = 25.1 kW and $\dot{m}$ = 7.04 kg/s
D) $\dot{q}$ = 25.1 kW and $\dot{m}$ = 12.4 g/s

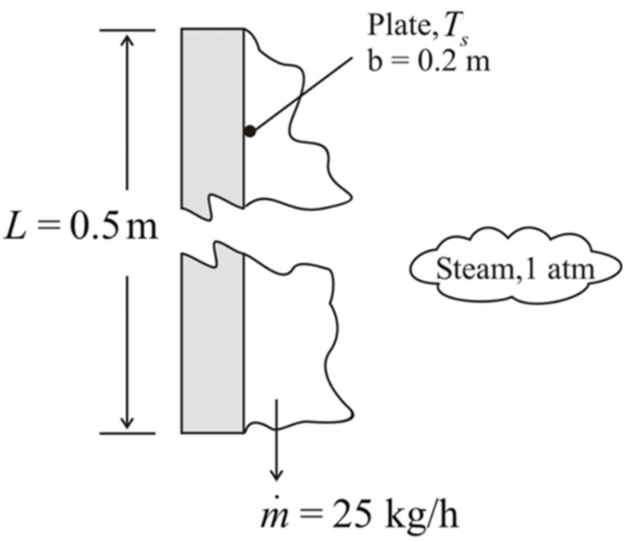
Problem 9B

Repeat the previous problem if the steam in contact with the pipe is saturated at 1.5 bars. The corresponding saturation temperature is 385 K, the enthalpy of vaporization is h_{fg} = 2225 kJ/kg, and the properties of liquid water at the film temperature are now c_{pl} = 4220 J/kg·K, k_l = 0.681 W/mK, μ_l = 271 × 10^{-6} N·s/m, and v_l = 2.83 × 10^{-7} m^2/s.

A) $\dot{q}$ = 32.8 kW and $\dot{m}$ = 17.5 g/s
B) $\dot{q}$ = 32.8 kW and $\dot{m}$ = 25.8 g/s
C) $\dot{q}$ = 39.9 kW and $\dot{m}$ = 17.5 kg/s
D) $\dot{q}$ = 39.9 MW and $\dot{m}$ = 25.8 kg/s

Problem 10 (Bergman et al., 2011, w/ permission)

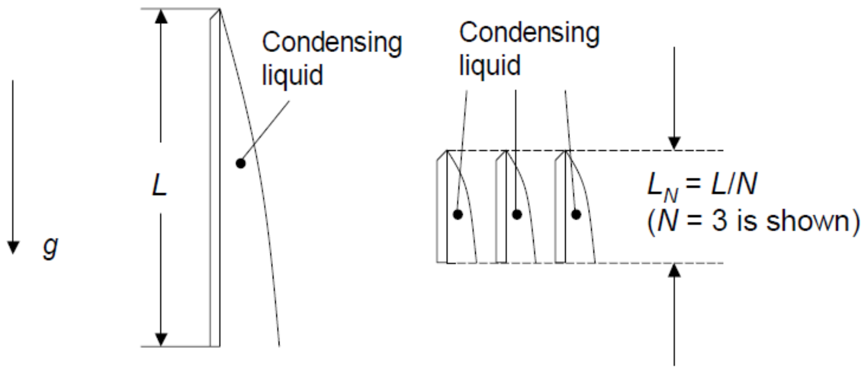
A vertical plate 500 mm high and 200 mm wide is to be used to condense saturated steam at 1 atm. At what surface temperature must the plate be maintained to achieve a condensation rate of $\dot{m}$ = 25 kg/h? Use as properties enthalpy of vaporization h_{fg} = 2257 kJ/kg, density of water ρ_l = 967.1 kg/m^3, specific heat of water c_{pl} = 4203 J/kg·K, dynamic viscosity of water μ_l = 324×10^{-6} N·s/m^2, thermal conductivity of water k_l = 0.674 W/mK, and kinematic viscosity of water ν_l = 3.35×10^{-7} m^2/s.



A) T_s = 77.3°C
B) T_s = 98.4°C
C) T_s = 120°C
D) T_s = 141°C

Problem 11 (Bergman et al., 2011, w/ permission)

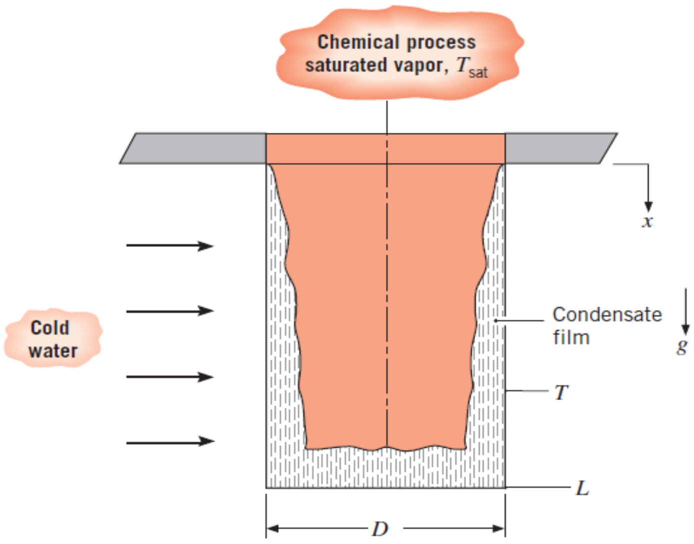
Consider wave-free laminar condensation on a vertical isothermal plate of length L, providing an average heat transfer coefficient of $\bar{h}_L$. If the plate is divided into N smaller plates, each of length $L_N = L/N$, determine an expression for the ratio of the heat transfer coefficient averaged over N plates to the heat transfer coefficient averaged over the single plate, $\bar{h}_{L,N}/\bar{h}_{L,1}$.



A) $\bar{h}_{L,N}/\bar{h}_{L,1} = N^{1/4}$
B) $\bar{h}_{L,N}/\bar{h}_{L,1} = N^{1/2}$
C) $\bar{h}_{L,N}/\bar{h}_{L,1} = N^{3/5}$
D) $\bar{h}_{L,N}/\bar{h}_{L,1} = N^{3/4}$

Problem 12 (Bergman et al., 2011, w/ permission)

Saturated vapor from a chemical process condenses at a slow rate on the inner surface of a vertical, thin-walled cylindrical container of length L and diameter D. The container wall is maintained at a uniform temperature T_s by flowing cold water across its outer surface. Derive an expression for the time t_f required to fill the container with condensate, assuming that the condensate film is laminar. Express your result in terms of D, L, $(T_{sat} - T_s)$, g, and appropriate fluid properties.



A) $t_f = \dfrac{2.12 h_{fg} \rho_l A_c}{\pi D \left[\dfrac{g \rho_l (\rho_l - \rho_v) k_l^3 h_{fg}}{\mu_l (T_{sat} - T_s)^3 L} \right]^{1/4}}$

B) $t_f = \dfrac{4.24 h_{fg} \rho_l A_c}{\pi D \left[\dfrac{g \rho_l (\rho_l - \rho_v) k_l^3 h_{fg}}{\mu_l (T_{sat} - T_s)^3 L} \right]^{1/4}}$

C) $t_f = \dfrac{2.12 h_{fg} \rho_l A_c}{\pi D \left[\dfrac{g \rho_l (\rho_l - \rho_v) k_l^3 h_{fg}}{\mu_l (T_{sat} - T_s)^3} \right]^{1/4} \dfrac{1}{L^{3/4}}}$

D) $t_f = \dfrac{4.24 h_{fg} \rho_l A_c}{\pi D \left[\dfrac{g \rho_l (\rho_l - \rho_v) k_l^3 h_{fg}}{\mu_l (T_{sat} - T_s)^3} \right]^{1/4} \dfrac{1}{L^{3/4}}}$

Problem 13 (Çengel & Ghajar, 2015, w/ permission)

A horizontal tube 1 m long with a surface temperature of 70°C is used to condense saturated steam at 1 atm. What diameter is required to achieve a condensation rate of 125 kg/h? Use as properties enthalpy of vaporization h_{fg} = 2257 kJ/kg, density of water vapor ρ_v = 0.596 kg/m^3, vapor temperature T_{sat} = 100°C, density of water ρ_l = 968.6 kg/m^3, specific heat of water c_{pl} = 4201 J/kg·K, dynamic viscosity of water μ_l = 332×10^{-6} N·s/m^2, and thermal conductivity of water k_l = 0.673 W/mK.

A) D = 83 mm

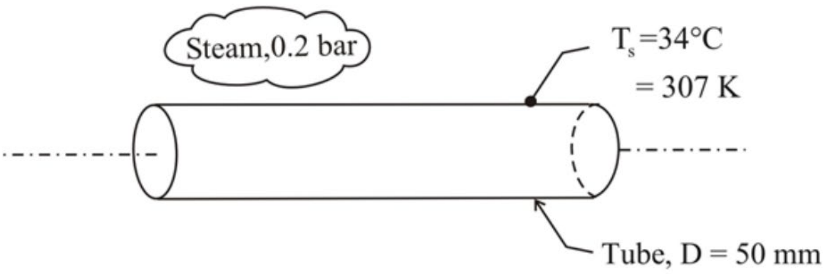
B) D = 144 mm

C) D = 206 mm

D) D = 267 mm

Problem 14A (Bergman et al., 2011, w/ permission)

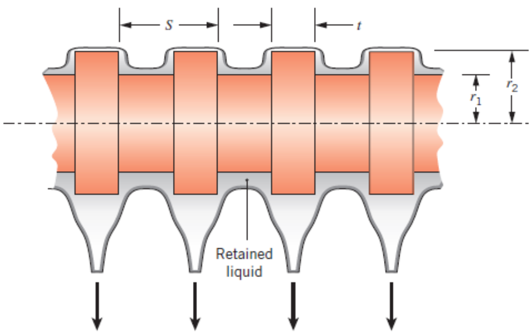
A horizontal tube of 50-mm outer diameter, with a surface temperature of 34°C, is exposed to steam at 0.2 bar. Estimate the heat transfer rate per unit length of the tube and the condensation rate. For saturated steam at 0.2 bar, the saturation temperature is T_{sat} = 333 K. In addition, use as properties density of vapor ρ_v = 0.129 kg/m^3, enthalpy of vaporization h_{fg} = 2358 kJ/kg, density of water ρ_l = 989.1 kg/m^3, specific heat of water c_{pl} = 4180 J/kg·K, viscosity of water μ_l = 577×10^{-6} N·s/m^2, and thermal conductivity of water k_l = 0.640 W/mK.



A) $\dot{q}'$ = 14.1 kW/m and $\dot{m}'$ = 5.1 g/s·m
B) $\dot{q}'$ = 14.1 kW/m and $\dot{m}'$ = 11.6 g/s·m
C) $\dot{q}'$ = 28.3 kW/m and $\dot{m}'$ = 5.1 g/s·m
D) $\dot{q}'$ = 28.3 kW/m and $\dot{m}'$ = 11.6 g/s·m

Problem 14B

The tube of the previous problem is modified by milling sharp-cornered grooves around its periphery, as in the figure below. The 2-mm-deep grooves are each 2 mm wide with a pitch of S = 4 mm. Estimate the minimum condensation and heat transfer rates per unit length that would be expected for the modified tube. How much is the performance enhanced relative to the original tube of the previous problem? Compute the heat enhancement ratio E. In addition to the data in the previous part, use as surface tension σ = 0.0661 N/m.



A) E = 1.09
B) E = 1.28
C) E = 1.47
D) E = 1.66

Additional Information

Table 1 Surface tension of liquid-vapor interface for water

T, °C	σ, N/m*
0	0.0757
20	0.0727
40	0.0696
60	0.0662
80	0.0627
100	0.0589
120	0.0550
140	0.0509
160	0.0466
180	0.0422
200	0.0377
220	0.0331
240	0.0284
260	0.0237
280	0.0190
300	0.0144
320	0.0099
340	0.0056
360	0.0019
374	0.0

Table 2 Surface tension of some fluids

Substance and Temp. Range	Surface Tension, σ, N/m* (T in °C)
Ammonia, –75 to –40°C:	$0.0264 + 0.000223T$
Benzene, 10 to 80°C:	$0.0315 - 0.000129T$
Butane, –70 to –20°C:	$0.0149 - 0.000121T$
Carbon dioxide, –30 to –20°C:	$0.0043 - 0.000160T$
Ethyl alcohol, 10 to 70°C:	$0.0241 - 0.000083T$
Mercury, 5 to 200°C:	$0.4906 - 0.000205T$
Methyl alcohol, 10 to 60°C:	$0.0240 - 0.000077T$
Pentane, 10 to 30°C:	$0.0183 - 0.000110T$
Propane, –90 to –10°C:	$0.0092 - 0.000087T$

*Multiply by 0.06852 to convert to lbf/ft or by 2.2046 to convert to lbm/s^2.

Table 3 Values of C_{sf} and n for various fluid-surface combinations

Fluid–Heating Surface Combination	C_{sf}	n
Water–copper (polished)	0.0130	1.0
Water–copper (scored)	0.0068	1.0
Water–stainless steel (mechanically polished)	0.0130	1.0
Water–stainless steel (ground and polished)	0.0060	1.0
Water–stainless steel (teflon pitted)	0.0058	1.0
Water–stainless steel (chemically etched)	0.0130	1.0
Water–brass	0.0060	1.0
Water–nickel	0.0060	1.0
Water–platinum	0.0130	1.0
n-Pentane–copper (polished)	0.0154	1.7
n-Pentane–chromium	0.0150	1.7
Benzene–chromium	0.1010	1.7
Ethyl alcohol–chromium	0.0027	1.7
Carbon tetrachloride–copper	0.0130	1.7
Isopropanol–copper	0.0025	1.7

Table 4 Values of coefficient C_{cr} for use with the formula for heat transfer in nucleate boiling

Heater Geometry	C_{cr}	Charac. Dimension of Heater, L	Range of L^*
Large horizontal flat heater	0.149	Width or diameter	$L^* > 27$
Small horizontal flat heater[1]	$18.9K_1$	Width or diameter	$9 < L^* < 20$
Large horizontal cylinder	0.12	Radius	$L^* > 1.2$
Small horizontal cylinder	$0.12L^{*-0.25}$	Radius	$0.15 < L^* < 1.2$
Large sphere	0.11	Radius	$L^* > 4.26$
Small sphere	$0.227L^{*-0.5}$	Radius	$0.15 < L^* < 4.26$

[1] $K_1 = \sigma/[g(\rho_l - \rho_v)A_{heater}]$

Solutions

P.1 Solution

The excess temperature is ΔT_{sat} = 110 – 100 = 10°C, which is relatively low (less than 30°C), so that we can surmise that nucleate boiling will occur. The properties are liquid density ρ_l = 957.9 kg/m^3, vapor density ρ_v = 0.60 kg/m^3, surface tension σ = 0.0589 N/m (Table 1), Prandtl number Pr_l = 1.75, latent heat h_{fg} = 2257 kJ/kg, liquid water viscosity μ_l = 0.282×10^{-3} kg·m/s, and liquid water specific heat c_{pl} = 4217 J/kg·°C. We have C_{sf} = 0.0130 and n = 1.0 for the boiling of water. Substituting in the Rohsenow relationship gives

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

$$\dot{q}_{\text{nucleate}} = \left(0.282\times10^{-3}\right)\times\left(2257\times10^3\right)\times\left[\frac{9.81\times(957.9-0.60)}{0.0589}\right]^{0.5}\times\left[\frac{4217\times(110-100)}{0.0130\times\left(2257\times10^3\right)\times1.75^{1.0}}\right]^3$$

$$\therefore\ \dot{q}_{\text{nucleate}} = 140.8\ \text{kW/m}^2$$

The heat transfer coefficient is obtained by solving Newton's law of cooling for h,

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

$$\therefore\ h = \frac{140{,}800}{(110-100)} = \boxed{14{,}080\ \text{W/m}^2\text{K}}$$

♦ The correct answer is **D**.

P.2 Solution

Part A: The rate of condensation heat transfer is

$$\dot{q} = \frac{m\times h_{fg}}{\Delta t} = \frac{(0.5\times1.0)\times2257}{(25\times60)} = 0.752\ \text{kW}$$

Given the surface area of the coffee maker $A_s = \pi$ × 0.004 × 0.20 = 2.51 × 10^{-3} m^2, the power rating of the heating element is determined as

$$\dot{Q} = \frac{0.752}{2.51\times10^{-3}} = \boxed{300\ \text{kW/m}^2}$$

The surface temperature is determined next. The properties to use are water viscosity μ_l = 0.282×10^{-3} kg·m/s, enthalpy of vaporization h_{fg} = 2257 kJ/kg, density of water ρ_l = 957.9 kg/m^3, density of water vapor ρ_v = 0.598 kg/m^3, specific heat of water c_{pl} = 4217 kJ/kg°C, Prandtl number of water Pr = 1.75, surface tension σ = 0.0589 N/m (Table 1). For boiling of water, we have C_{sf} = 0.0130 and n = 1.0. Applying the Rohsenow relationship gives

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

$$\therefore 300{,}000 = \left(0.282\times10^{-3}\right)\times\left(2257\times10^3\right)\times\left[\frac{9.81\times(957.9-0.598)}{0.0589}\right]^{1/2}\times\left[\frac{4217\times(T_s-100)}{0.013\times\left(2257\times10^3\right)\times1.75^{1.0}}\right]^3$$

Solving for T_s with a CAS such as Mathematica yields T_s = 113°C. The temperature of the heating element is just above 110 degrees Celsius.

♦ The correct answer is **C**.

Part B: The heat transferred to the water is given by the elementary relation

$$Q = mc_p \Delta T \rightarrow \dot{Q}\Delta t = mc_p \Delta T$$

$$\therefore \Delta t = \frac{mc_p \Delta T}{\dot{Q}} = \frac{1.0 \times 4.184 \times (100-14)}{0.752} = 478 \text{ s} \approx \boxed{7.97 \text{ min}}$$

♦ The correct answer is **D**.

P.3 Solution

Properties for water at the conditions in question include ρ_l = 957.9 kg/m^3, ρ_v = 0.60 kg/m^3, h_{fg} = 2257 kJ/kg, and σ = 0.0589 N/m (Table 1). Parameter $L*$ is obtained by the following expression, noting that the radius r = 0.0005 m is the characteristic dimension of the wire,

$$L* = L\left[\frac{g(\rho_l - \rho_v)}{\sigma_s}\right]^{1/2} = 0.0005 \times \left[\frac{9.81 \times (957.9 - 0.60)}{0.0589}\right]^{1/2} = 0.20 \text{ m}$$

Since 0.15 < $L*$ < 1.2, Coefficient C_{cr} can be obtained from the relation (see Table 4)

$$C_{cr} = 0.12 L*^{-0.25} = 0.12 \times 0.20^{-0.25} = 0.18$$

The maximum heat flux is such that

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

$$\therefore \dot{q}_{\max} = 0.18 \times (2257 \times 10^3) \times \left[0.0589 \times 9.81 \times 0.60^2 \times (957.9 - 0.60)\right]^{1/4} = 1526 \text{ kW/m}^2$$

Noting that electric power is given by $P = R \times I^2$, the electrical current at which the wire would be in danger of burnout in nucleate boiling is

$$\dot{q}_{\max} = \frac{R \times I^2}{A_s} \rightarrow I = \sqrt{\frac{\dot{q}_{\max} \times A_s}{R}} = \sqrt{\frac{\dot{q}_{\max} \times \pi DL}{R}} = \sqrt{\frac{\dot{q}_{\max} \times \pi D}{\underbrace{\left(\frac{R}{L}\right)}_{=0.129}}}$$

$$\therefore I = \sqrt{\frac{(1526 \times 10^3) \times \pi \times 0.001}{0.129}} = \boxed{193 \text{ A}}$$

The current needed for the wire to be in danger of burnout in nuclear boiling is 193 ampères.

♦ The correct answer is **C**.

P.4 Solution

The characteristic length of the heater is

$$L* = L\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}$$

Here, length L = 3 m, density of water ρ_l = 907.4 kg, density of water vapor ρ_v = 3.26 kg/m^3, and surface tension σ = 0.0466 N/m (Table 1). Thus,

$$L* = L\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} = 3 \times \left[\frac{9.81 \times (907.4 - 3.26)}{0.0466}\right]^{1/2} = 1309 \text{ m}$$

Since this is greater than 27, the cylinders can be treated as horizontal flat heaters and coefficient C_{cr} = 0.149 (Table 4). The maximum heat flux is given by

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

Here, enthalpy of vaporization h_{fg} = 2083 kJ/kg, surface tension σ = 0.0466 N/m, density of water vapor ρ_v = 3.26 kg/m^3, and density of water ρ_l = 907.4 kg/m^3, with the result that

$$\dot{q}_{\max} = 0.149\times\left(2083\times10^3\right)\times\left[0.046\times9.81\times3.26^2\times\left(907.4-3.26\right)\right]^{1/4} = 2.52\times10^6 \text{ W/m}^2$$

The surface area of the heater is A_s = 3 × 3 = 9 m^2, and the rate of heat transfer follows as

$$\dot{q}_{\text{heater}} = \dot{q}_{\max}A_s = \left(2.52\times10^6\right)\times9 = 2.27\times10^7 \text{ W}$$

The rate of water vaporization can be obtained by dividing $\dot{q}_{\text{heater}}$ by the heat of vaporization h_{fg},

$$\dot{m}_{\text{vapor}} = \frac{\dot{q}_{\text{heater}}}{h_{fg}} = \frac{2.27\times10^7}{2083\times10^3} = \boxed{10.9 \text{ kg/s}}$$

♦ The correct answer is **B**.

P.5 Solution

Properties of water at 100°C include ρ_l = 957.9 kg/m^3, ρ_v = 0.60 kg/m^3, Pr_l = 1.75, h_{fg} = 2257 kJ/kg, μ_l = 0.282×10^{-3} kg·s/m, c_{pl} = 4217 J/kg, and σ = 0.0589 N/m (Table 1). The heat that needs to me removed from the sheet can be established with the heat formula

$$\dot{q}_{\text{removed}} = \rho_{ss}c_{p,ss}Vwt\left(T_{\text{in}}-T_{\text{out}}\right)$$

in which ρ_{ss} = 7900 kg/m^3 is the density of the stainless steel, $c_{p,ss}$ = 450 J/kg·K is the specific heat of the stainless steel, V = 2 m/s is the velocity with which the sheet is introduced in the bath, w = 0.5 m is its width, t = 5 mm is its thickness, T_{in} = 125°C is the entrance temperature, and T_{out} = 45°C is the exit temperature. Substituting, we obtain

$$\dot{q}_{\text{removed}} = 7900\times450\times2\times0.5\times0.005\times\left(125-45\right) = 1422 \text{ kW}$$

Thence, we can compute the nucleate boiling heat flux with the usual formula

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

where we substitute μ_l = 0.282×10^{-3} kg/m·s, h_{fg} = 2257 kJ/kg, ρ_l = 957.9 kg/m^3, ρ_v = 0.60 kg/m^3, c_{pl} = 4217 J/kg, C_{sf} = 0.0130, Pr = 1.75, n = 1.0, σ = 0.0589 N/m (Table 1), T_s = 125°C as the temperature of the water bath. Thus,

$$\dot{q}_{\text{nucleate}} = \left(0.282\times10^{-3}\right)\times\left(2257\times10^3\right)\times\left[\frac{9.81\times\left(957.9-0.60\right)}{0.0589}\right]^{1/2}\times\left[\frac{4217\times\left(125-100\right)}{0.0130\times\left(2257\times10^3\right)\times1.75^{1.0}}\right]^3$$

$$\therefore \dot{q}_{\text{nucleate}} = 2199 \text{ kW/m}^2$$

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

$$A_s = 2lw+2lt = 2\times1\times0.5+2\times1\times0.005 = 1.01\,\text{m}^2$$

The rate of heat removed from the sheet in the water bath, $\dot{Q}$, follows as

$$\dot{Q}_{\text{boiling}} = A_s\dot{q}_{\text{nucleate}} = 1.01\times2199 = 2221 \text{ kW}$$

Notice that heat is removed from the sheet at a rate of 2221 kW, which is considerably greater than the rate of 1422 kW with which heat needs to be removed for the sheet to exit the water bath at the minimum acceptable temperature, 45°C.

P.6 Solution

Under these conditions, the excess temperature is ΔT_{sat} = 500 – 100 = 400°C, which is considerably greater than 30°C and hence allows us to surmise that there will be film boiling. The film boiling heat flux is given by

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

Here, we have k_v = 0.0435 W/mK, ρ_l = 957.9 kg/m³, ρ_v = 0.383 kg/m³, h_{fg} = 2257 kJ/kg, c_{pv} = 1997 J/kg, μ_v = 2.05×10⁻⁵ kg/m·s, D = 2 mm as the diameter of the cylindrical rod, T_s = 500°C as the temperature on the surface of the rod, and T_{sat} = 100°C; finally, coefficient C_{film} = 0.62 for a horizontal cylinder. Substituting each variable, we obtain

$$\dot{q}_{\text{film}} = 0.62 \times \left\{ \frac{9.81 \times 0.0435^3 \times 0.383 \times (957.9 - 0.383) \times \left[\left(2257 \times 10^3\right) + 0.4 \times 1997 \times (500-100)\right]}{\left(2.05 \times 10^{-5}\right) \times 0.002 \times (500-100)} \right\}^{0.25} (500-100)$$

$$\therefore \dot{q}_{\text{film}} = 115,200 \text{ W/m}^2$$

Thence, we compute the radiation heat flux with the Stefan-Boltzmann law,

$$\dot{q}_{\text{rad}} = \varepsilon\sigma\left(T_s^4 - T_{\text{sat}}^4\right) = 0.5 \times \left(5.87 \times 10^{-8}\right) \times \left[(500+273)^4 - (100+273)^4\right] = 9,570 \text{ W/m}^2$$

The total heat flux is given by the simple sum

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4}\dot{q}_{\text{rad}} = 115,200 + 0.75 \times 9,570 = 122,400 \text{ W/m}^2$$

Power dissipation per unit length in the metal rod easily follows,

$$\dot{Q} = \dot{q}_{\text{total}} A \rightarrow \dot{Q} = \dot{q}_{\text{total}} \times \pi D L \rightarrow \frac{\dot{Q}}{L} = \dot{Q}' = \dot{q}_{\text{total}} \times \pi D$$

$$\therefore \dot{Q}' = 122,400 \times \pi \times 0.002 = \boxed{769 \text{ W/m}}$$

♦ The correct answer is **B**.

P.7 Solution

The heat transfer and condensation rates are respectively given by

$$\dot{q} = \bar{h}_L A_s \left(T_{\text{sat}} - T_s\right)$$

and

$$\dot{m} = \frac{\dot{q}}{h_{fg}}$$

where the total area A_s is the same for each case. Hence, the following ratio applies,

$$\frac{\dot{q}_1}{\dot{q}_2} = \frac{\dot{m}_1}{\dot{m}_2} = \frac{\bar{h}_{L,1}}{\bar{h}_{L,2}}$$

In which the average convection coefficients $\bar{h}_{L,1}$ and $\bar{h}_{L,2}$ are evaluated for plate lengths of L and $L/2$, respectively. For laminar film condensation on both plates, using the correlation below,

$$\bar{h}_L = 0.943\left[\frac{g\rho_l\left(\rho_l - \rho_v\right)k_l^3 h'_{fg}}{\mu_l\left(T_{\text{sat}} - T_s\right)L}\right]^{1/4}$$

we see that $\bar{h}_L \propto L^{-1/4}$ and the ratio of heat transfers is, consequently,

$$\frac{\dot{q}_1}{\dot{q}_2}=\left(\frac{L}{L/2}\right)^{-1/4}=0.84$$

Hence, case 2 is preferred and provides 19% more heat transfer (since $\dot{q}_2/\dot{q}_1$ = 1/0.84 = 1.19). It should be noted, however, that the laminar solution is valid provided that parameter $P < 15.8$, therefore we require that L be

$$L<15.8\frac{\mu_l h'_{fg}\left(\nu_l^2/g\right)^{1/3}}{k_l\left(T_{\text{sat}}-T_s\right)}=15.8\times\frac{\left(296\times10^{-6}\right)\times\left(2286\times10^3\right)\times\left[\left(3.08\times10^{-7}\right)^2/9.81\right]^{1/3}}{0.678\times(100-90)}=0.034=34\,\text{mm}$$

Note that the corrected latent heat $h'_{fg} = h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s)$ = 2257 + 0.68×4212× 10^{-3} ×(100-90) = 2286 kJ/kg, hence the value used above. We can anticipate that for other, larger values of L the comparison of $\bar{h}_L$ values cannot be so easily made. Nevertheless, an examination of graphical trends in condensation regimes (Fig. 10-26 in Çengel & Ghajar, or Fig. 10-13 in Bergman et al.) shows that we can expect the same behavior of $\bar{h}_L$ in the *wavy* region since $\bar{h}_L$ decreases with increasing Re_δ (corresponding to increasing L), and anticipate that indeed case 2 will provide the greater condensation rate. Note, however, that in the turbulent region, with the increase in $\bar{h}_L$ with Re_δ, we cannot establish with certainty which case is preferred.

In dealing with single-phase, forced or free convection, we associate thin thermal boundary layers with higher heat transfer rates. For vertical plates, we would expect the shorter plate to have the higher convection heat transfer coefficient. The results of this problem suggest the same is true for condensation on the vertical plate.

P.8 Solution

Part A: The modified latent heat is $h'_{fg} = h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s)$ = 2257 + 0.68×4193×(100 – 50) = 2400 kJ/kg. Parameter P is then

$$P=\frac{k_l L\left(T_{\text{sat}}-T_s\right)}{\mu_l h'_{fg}\left(\nu_l^2/g\right)^{1/3}}=\frac{0.668\times2\times(100-50)}{\left(375\times10^{-6}\right)\times\left(2400\times10^3\right)\times\left[\left(3.85\times10^{-7}\right)^2/9.81\right]^{1/3}}=3002$$

which is within the range $P \geq 2530$ (along with $Pr \geq 1$), so the Nusselt number can be obtained with the correlation

$$Nu_L=\frac{1}{P}\left[(0.024P-53)\Pr^{1/2}+89\right]^{4/3}=\frac{1}{3002}\times\left[(0.024\times3002-53)\times2.35^{1/2}+89\right]^{4/3}=0.193$$

and the heat transfer coefficient is

$$\bar{h}_L=\frac{k_l}{\left(\nu_l^2/g\right)^{1/3}}\times Nu_L=\frac{0.668}{\left[\left(3.85\times10^{-7}\right)^2/9.81\right]^{1/3}}\times0.193=5215\ \text{W/m}^2\text{K}$$

The heat and condensation rates are then

$$\dot{q}=\bar{h}_L A\left(T_{\text{sat}}-T_s\right)=5215\times(2\times2)\times(100-50)=\boxed{1.04\ \text{MW}}$$

$$\dot{m}=\dot{q}/h_{fg}=1{,}040{,}000/\left(2400\times10^3\right)=\boxed{0.433\ \text{kg/s}}$$

♦ The correct answer is **C.**

Part B: Using the relationship proposed, the convection coefficient for the inclined configuration is

$$\bar{h}_{L,\text{inc}}=5215\times\left(\cos45^\circ\right)^{1/4}=4782\ \text{W/m}^2\text{K}$$

The heat transfer coefficient therefore reduces by 1 – 4782/5215 = 8.3% and the heat transfer and condensation rates should reduce accordingly, yielding

$$\dot{q}_{\text{inc}} = 0.917 \times \dot{q} = \boxed{0.95 \text{ MW}}$$

$$\dot{m}_{\text{inc}} = \boxed{0.397 \text{ kg/s}}$$

♦ The correct answer is **C**.

P.9 Solution

Part A: The Jakob number is given by

$$\text{Ja} = \frac{c_{pl}\left(T_{\text{sat}} - T_s\right)}{h_{fg}} = \frac{4214 \times (100 - 94)}{2257 \times 10^3} = 0.0112$$

and the corrected enthalpy of vaporization follows as

$$h'_{fg} = h_{fg}\left(1 + 0.68\text{Ja}\right) = 2257 \times (1 + 0.68 \times 0.0112) = 2274 \text{ kJ/kg}$$

The next step is to compute parameter P,

$$P = \frac{k_l L\left(T_{\text{sat}} - T_s\right)}{\mu_l h'_{fg}\left(v_l^2/g\right)^{1/3}}$$

$$P = \frac{0.679 \times 1.0 \times (100 - 94)}{\left(289 \times 10^{-6}\right) \times \left(2274 \times 10^3\right) \times \left[\left(3.01 \times 10^{-7}\right)^2 / 9.81\right]^{1/3}} = 295$$

Since $15.8 < P < 2530$, the correlation to use for the heat transfer coefficient is

$$\bar{h}_L = \frac{k_l}{\left(v_l^2/g\right)^{1/3}} \frac{1}{P}\left(0.68P + 0.89\right)^{0.82}$$

$$\therefore \bar{h}_L = \frac{0.679}{\left[\left(3.01 \times 10^{-7}\right)^2 / 9.81\right]^{1/3}} \times \frac{1}{295} \times \left(0.68 \times 295 + 0.89\right)^{0.82} = 8500 \text{ W/m}^2\text{K}$$

The heat transfer rate follows from Newton's law of cooling,

$$\dot{q} = \bar{h}_L A\left(T_{\text{sat}} - T_s\right) = 8500 \times (\pi \times 0.1 \times 1.0) \times (100 - 94) = \boxed{16.0 \text{ kW}}$$

The condensation rate, in turn, is obtained if we divide $\dot{q}$ by the enthalpy of vaporization,

$$\dot{m} = \frac{\dot{q}}{h'_{fg}} = \frac{16.0 \times 10^3}{2.274 \times 10^6} = \boxed{7.04 \text{ g/s}}$$

♦ The correct answer is **A**.

Part B: The Jakob number is now

$$\text{Ja} = \frac{c_{pl}\left(T_{\text{sat}} - T_s\right)}{h_{fg}} = \frac{4220 \times (385 - 367)}{2225 \times 10^3} = 0.0341$$

and the enthalpy of vaporization becomes

$$h'_{fg} = 2225 \times (1 + 0.68 \times 0.0341) = 2277 \text{ kJ/kg}$$

Parameter P is calculated as

$$P = \frac{0.681 \times 1.0 \times (385 - 367)}{\left(271 \times 10^{-6}\right) \times \left(2277 \times 10^3\right) \times \left[\left(2.83 \times 10^{-7}\right)^2 / 9.81\right]^{1/3}} = 987$$

Since 15.8 < P < 2530, the correlation to use for the heat transfer coefficient is the same as before, namely,

$$\bar{h}_L = \frac{0.681}{\left[\left(2.83\times10^{-7}\right)^2/9.81\right]^{1/3}}\times\frac{1}{987}\times\left(0.86\times987+0.89\right)^{0.82} = 7060\ \mathrm{W/m^2K}$$

The heat transfer rate is determined as

$$\dot{q} = 7060\times\left(\pi\times0.1\times1.0\right)\times\left(385-367\right) = \boxed{39.9\ \mathrm{kW}}$$

and the condensation rate follows as

$$\dot{m} = \frac{39.9\times10^3}{2.277\times10^6} = \boxed{17.5\ \mathrm{g/s}}$$

The calculations are summarized below.

	Part A	Part B
Pressure	1.01	1.5
T_{sat} (K)	373	385
T_{sat} - T_s (K)	6	18
h_L (W/m²K)	8500	7060
q (kW)	16	39.9
m (g/s)	7.04	17.5

Clearly, increasing the steam pressure by about 50% causes the excess temperature to increase three-fold, the heat transfer coefficient to drop 17%, and the heat transfer and condensation rates to increase by a factor of 2.5.

♦ The correct answer is **C**.

P.10 Solution

With knowledge of $\dot{m}$ = 25 kg/h = 6.94×10^{-3} kg/s, the Reynolds number Re_δ can be computed with the relation

$$\mathrm{Re}_\delta = \frac{4\dot{m}}{\mu_l b} = \frac{4\times\left(6.94\times10^{-3}\right)}{\left(324\times10^{-6}\right)\times0.2} = 428$$

Hence, flow is wavy-laminar and the correlation to apply for the heat transfer coefficient is

$$h = \frac{k_l}{\left(\nu_l^2/g\right)^{1/3}}\frac{\mathrm{Re}_\delta}{1.08\,\mathrm{Re}_\delta^{1.22}-5.2}$$

in which k_l = 0.674 W/mK and ν_l = 3.35×10^{-7} m²/s as given, with the result that

$$h = \frac{0.674}{\left[\left(3.35\times10^{-7}\right)^{1/3}/9.81\right]}\frac{428}{1.08\times428^{1.22}-5.2} = 7325\ \mathrm{W/m^2K}$$

The heat transfer rate is obtained with Newton's law of cooling,

$$\dot{q} = hA_s\left(T_{\mathrm{sat}}-T_s\right) = 7325\times\left(0.5\times0.2\right)\times\left(100-T_s\right)\ \ (\mathrm{I})$$

where we have substituted the convection coefficient along with the plate dimensions in A_s = 0.5×0.2. The surface temperature T_s is the variable we seek. The modified latent heat h'_{fg} is

$$h'_{fg} = h_{fg}+0.68c_{pl}\left(T_{\mathrm{sat}}-T_s\right) = 2257+0.68\times4203\times\left(100-50\right) = 2400\ \mathrm{kJ/kg}$$

and the equation for condensation rate allows us to write

$$\dot{m} = \frac{\dot{q}}{h'_{fg}} \rightarrow \dot{q} = \dot{m}h'_{fg}$$

$$\dot{q} = \left(6.94\times10^{-3}\right)\times\left(2400\times10^{3}\right) = 16,656 \text{ W (II)}$$

Equations (I) and (II) must yield the same result. Equating one to the other, we obtain

$$16,656 = 7325\times(0.5\times0.2)\times(100 - T_s)$$

$$\therefore \boxed{T_s = 77.3^{\circ}\text{C}}$$

♦ The correct answer is **A**.

P.11 Solution

The dimensionless parameter *P*, which is used in Bergman & Lavine's discussion of film condensation, is given by

$$P = \frac{k_l L\left(T_{\text{sat}} - T_s\right)}{\mu_l h'_{fg}\left(\nu^2/g\right)^{1/3}}$$

Equation 10.43 in the aforementioned textbook states that the Nusselt number is

$$Nu = 0.943P^{-1/4}$$

hence the proportion

$$\overline{h}_{L,1} \propto P^{-1/4} \propto L^{-1/4}$$

For multiple plates, each of length $L_N = L/N$, the following proportion holds,

$$\overline{h}_{L,N} \propto L_N^{-1/4} \propto \left(\frac{L}{N}\right)^{-1/4}$$

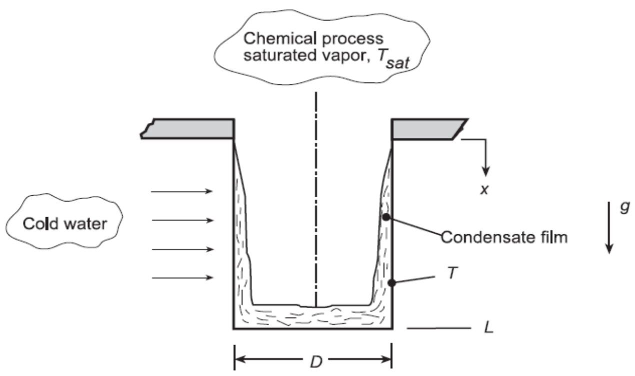
Dividing the two previous proportions, we can state that

$$\boxed{\frac{\overline{h}_{L,N}}{\overline{h}_{L,1}} = N^{1/4}}$$

Notice that by breaking the single plate into shorter segments, the average liquid film thickness is reduced, resulting in a modest increase in the average heat transfer coefficient, i.e., resulting in a heat transfer *enhancement*. For instance, dividing the plate into 3 segments would produce a system whose convection coefficient is $3^{1/4} \approx 1.32$ times the heat transfer coefficient for condensation of a single plate.

♦ The correct answer is **A**.

P.12 Solution



The length and diameter of the cylindrical container are *L* and *D*, respectively, and the surface temperature of the wall is maintained at T_s by cold water flowing across its outer surface. The liquid mass in the container is expressed as

$$M = \rho_l A_c \left(L - x\right)$$

where *x* is the vertical coordinate. The rate of change of mass, *dM/dt*, with respect to time is obtained as

$$\dot{m}\left(t\right) = \frac{dM}{dt}$$

which, using the expression we just posed for *M*, becomes

$$\frac{dM}{dt} = -\rho_l A_c \frac{dx}{dt} = \dot{m}\left(t\right) \;\; \text{(I)}$$

Using Newton's law of cooling and the latent heat h_{fg}, we can express the condensation rate as

$$\dot{m}\left(t\right) = \frac{\overline{h} A_s \left(T_{\text{sat}} - T_s\right)}{h_{fg}} \;\; \text{(II)}$$

Here, the average convection coefficient is expressed by the correlation

$$\overline{h} = 0.943 \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right) x}\right]^{1/4}$$

The surface area subject to condensation is $A_s = \pi D x$. Substituting the relations we have for $\overline{h}$ and A_s in equation (II), it follows that

$$\dot{m}\left(t\right) = \frac{0.943 \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right) x}\right]^{1/4} \pi D x \left(T_{\text{sat}} - T_s\right)}{h_{fg}}$$

$$\therefore \dot{m}\left(t\right) = \frac{0.943}{h_{fg}} \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right)^3 L}\right]^{1/4} \times \left(\frac{L}{x}\right)^{1/4} \times \pi D x$$

$$\therefore \dot{m}\left(t\right) = \frac{0.943\pi D}{h_{fg}} \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right)^3 L}\right]^{1/4} \left(\frac{L}{x^{-3}}\right)^{1/4}$$

Next, we equate the expression above and (I) to obtain

$$\therefore -\rho_l A_c \frac{dx}{dt} = \frac{0.943\pi D}{h_{fg}} \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right)^3 L}\right]^{1/4} \left(\frac{L}{x^{-3}}\right)^{1/4}$$

$$\therefore -x^{3/4} dx = \frac{0.943\pi D}{h_{fg}} \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right)^3 L}\right]^{1/4} L^{1/4} dt$$

By integrating the above relation gives

$$-\int_L^0 x^{3/4} dx = \frac{0.943\pi D}{h_{fg} \rho_l A_c} \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right)^3 L}\right]^{1/4} L^{1/4} \int_0^{t_f} dt$$

$$\therefore -\left(\frac{x^{1/4}}{1/4}\right)\Bigg|_L^0 = \frac{0.943\pi D}{h_{fg} \rho_l A_c} \left[\frac{g\rho_l \left(\rho_l - \rho_v\right) k_l^3 h_{fg}}{\mu_l \left(T_{\text{sat}} - T_s\right)^3 L}\right]^{1/4} L^{1/4} \times t_f$$

$$\therefore 4L^{1/4} = \frac{0.943\pi D}{h_{fg}\rho_l A_c}\left[\frac{g\rho_l(\rho_l-\rho_v)k_l^3 h_{fg}}{\mu_l(T_{\text{sat}}-T_s)^3 L}\right]^{1/4} L^{1/4}\times t_f$$

$$\therefore \boxed{t_f = \frac{4.24 h_{fg}\rho_l A_c}{\pi D\left[\frac{g\rho_l(\rho_l-\rho_v)k_l^3 h_{fg}}{\mu_l(T_{\text{sat}}-T_s)^3 L}\right]^{1/4}}}$$

The expression above provides the time required to fill the container with condensate.

♦ The correct answer is **B**.

P.13 Solution

We first compute the Jakob number, Ja,

$$\text{Ja} = \frac{c_{pl}(T_{\text{sat}}-T_s)}{h_{fg}} = \frac{4201\times(100-70)}{(2257\times10^3)} = 0.0558$$

The modified latent heat of vaporization easily follows,

$$h'_{fg} = h_{fg}(1+0.68\text{Ja}) = 2257\times(1+0.68\times0.0558) = 2343 \text{ kJ/kg}$$

The heat transfer rate can be given as the product of the mass flow and the modified latent heat,

$$\dot{q} = \dot{m}h'_{fg}$$

Alternatively, the heat transfer rate can be determined by means of Newton's law of cooling,

$$\dot{q} = h\times\pi DL\times(T_{\text{sat}}-T_s)$$

Equating the two previous expressions, we can establish the diameter required to achieve the prescribed condensation rate,

$$\dot{m}\times h_{fg} = h\times\pi DL\times(T_{\text{sat}}-T_s)$$

$$\therefore D = \frac{\dot{m}h_{fg}}{\pi Lh(T_{\text{sat}}-T_s)}$$

Substituting $\dot{m}$ = 125 kg/h = 0.0347 kg/s, h_{fg} = 2343 kJ/kg, L = 1 m, T_{sat} = 100°C, and T_s = 70°C gives

$$\therefore D = \frac{\dot{m}h_{fg}}{\pi Lh(T_{\text{sat}}-T_s)} = \frac{0.0347\times(2343\times10^3)}{\pi\times1\times h\times(100-70)} = \frac{862.64}{h}$$

$$\therefore D = 862.64h^{-1} \quad \text{(I)}$$

Hence, we cannot determine the pipe diameter without the heat transfer coefficient. This quantity can be established with the correlation for condensation in radial systems,

$$h = C\left[\frac{g\rho_l(\rho_l-\rho_v)h'_{fg}k_l^2}{\mu_l(T_{\text{sat}}-T_s)D}\right]^{1/4}$$

For a tube, coefficient C = 0.729. Furthermore, we have ρ_l = 968.6 kg/m^3, ρ_v = 0.596 kg/m^3, c_{pl} = 4201 J/kg·K, μ_l = 332×10^{-6} N·s/m^2, and k_l = 0.673 W/mK. Substituting, we get

$$h = 0.729\times\left[\frac{9.81\times968.6\times(968.6-0.596)\times(2343\times10^3)\times0.673^3}{(332\times10^{-6})\times(100-70)\times D}\right]^{1/4} = 3694.36D^{-1/4}$$

We then substitute *h* in equation (I), giving

$$D = 862.64h^{-1} = 862.64 \times \left(3694.36D^{-1/4}\right)^{-1} = 0.234D^{1/4}$$

$$\therefore \frac{D}{D^{1/4}} = 0.234$$

$$\therefore D^{3/4} = 0.234$$

$$\therefore D = (0.234)^{\frac{4}{3}} = 0.144\,\text{m}$$

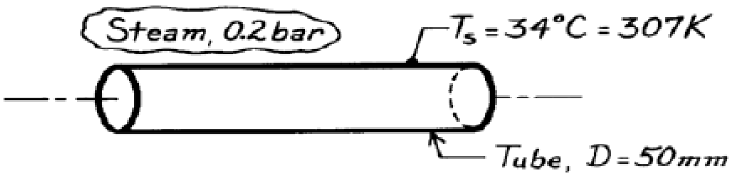
$$\therefore \boxed{D = 144\,\text{mm}}$$

The pipe should have a diameter of 144 millimeters.

♦ The correct answer is **B**.

P.14 Solution

Part A:



The heat transfer and condensate forming rates per unit length of the tube are, respectively,

$$\dot{q}' = h \times \pi D \times \left(T_{\text{sat}} - T_s\right)$$

$$\dot{m}' = \frac{\dot{q}'}{h'_{fg}}$$

The Jakob number, Ja, is

$$\text{Ja} = \frac{c_{pl}\left(T_{\text{sat}} - T_s\right)}{h_{fg}} = \frac{4180 \times (333 - 307)}{\left(2358 \times 10^3\right)} = 0.0461$$

and the modified latent heat of vaporization follows as

$$h'_{fg} = h_{fg}\left(1 + 0.68\text{Ja}\right) = 2358 \times (1 + 0.68 \times 0.0461) = 2432\ \text{kJ/kg}$$

For laminar film condensation, we use the following equation for the heat transfer coefficient,

$$h = C\left[\frac{g\rho_l\left(\rho_l - \rho_v\right)k_l^3 h'_{fg}}{\mu_l\left(T_{\text{sat}} - T_s\right)D}\right]^{1/4}$$

where coefficient C = 0.729 for a tube. Substituting this and other data gives

$$h = 0.729 \times \left[\frac{9.81 \times 989.1 \times (989.1 - 0.129) \times 0.640^3 \times \left(2432 \times 10^3\right)}{\left(577 \times 10^{-6}\right) \times (333 - 307) \times 0.050}\right]^{1/4} = 6927.82\ \text{W/m}^2\text{K}$$

Lastly, substituting this and other data in the expression for heat transfer per unit length yields

$$\dot{q}' = 6927.82 \times (\pi \times 0.05) \times (333 - 307) = \boxed{28.3\ \text{kW/m}}$$

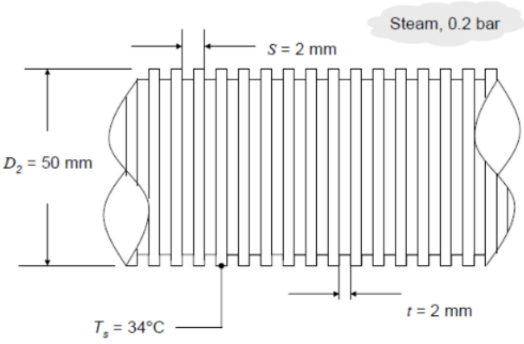
Similarly, substituting the quantities we have in the expression for $\dot{m}'$, the result is

$$\dot{m}' = \frac{\dot{q}'}{h'_{fg}} = \frac{28.3}{2432} = 0.0116 \text{ kg/s} = \boxed{11.6 \text{ g/s}\cdot\text{m}}$$

In each second, the system will lose about 11.6 grams of condensate per unit length of tube.

♦ The correct answer is **D**.

Part B: The system in question is illustrated below.



In the previous problem, the heat transfer rate and the condensate formation rates per unit length were determined as $\dot{q}'$ = 28.3 kW/m and $\dot{m}'$ = 11.6 g/s·m. The portions of the larger tube that are not milled away serve as fins. Therefore, the heat transfer rate from the grooved large tube is related to the heat transfer rate from a corresponding smooth tube of smaller diameter D_1 = 46 mm, modified by the enhancement ratio. We calculate the heat transfer rate for a smooth tube of diameter D_1 as follows. The heat transfer coefficient is obtained with the usual formula

$$h_{D_1} = 0.729\frac{k_l}{D_1}\left[\frac{g\rho_l(\rho_l-\rho_v)D_l^3 h'_{fg}}{k_l(T_{\text{sat}}-T_s)\mu_l}\right]$$

$$h_{D_1} = 0.729\times\frac{0.640}{0.046}\times\left[\frac{9.81\times 989.1\times(989.1-0.129)\times 0.046^3\times\left(2432\times 10^3\right)}{0.640\times(333-307)\times\left(577\times 10^{-6}\right)}\right]^{1/4}$$

$$\therefore h_{D_1} = 7073.75 \text{ W/m}^2\text{K}$$

The heat transfer rate for the smooth tube per unit length is computed as

$$\dot{q}' = h_{D_1}\times\pi D_1\times(T_{\text{sat}}-T_s) = 7073.75\times\pi\times 0.046\times(333-307) = 26.6 \text{ kW/m}$$

Thence, the minimum enhancement factor for the finned tube is calculated with the relation

$$\varepsilon_{\text{ft,min}} = \frac{tr_2}{Sr_1}\left[\frac{r_1}{r_2}+1.02\frac{\sigma r_1}{(\rho_l-\rho_v)gt^3}\right]$$

Here, t is the width of the fin, r_2 is the radius of the tube, r_1 is the radius of the un-finned tube, σ is the surface tension, ρ_l is the density of liquid in the tube, ρ_v is the density of vapor, and S is the pitch. Substituting t = 0.002 m, r_2 = 0.025 m, r_1 = 0.023 m, ρ_l = 989.1 kg/m^3, ρ_v = 0.129 kg/m^3, σ = 0.0661 N/m, and S = 0.004 m yields

$$\varepsilon_{\text{ft,min}} = \frac{0.002\times 0.025}{0.004\times 0.023}\times\left[\frac{0.023}{0.025}+1.02\times\frac{0.0661\times 0.023}{(989.1-0.129)\times 9.81\times 0.002^3}\right]^{1/4} = 1.16$$

Thus, the minimum heat transfer rate for the grooved tube is

$$\dot{q}'_{\text{ft,min}} = \varepsilon_{\text{ft,min}}\dot{q}' = 1.16\times 26.6 = 30.9 \text{ kW/m}$$

The corresponding condensation rate is

$$\dot{m}' = \frac{\left(30.9\times10^3\right)}{\left(2432\times10^3\right)} = 0.0127\ \text{kg/s}\cdot\text{m} = 12.7\ \text{g/s}\cdot\text{m}$$

The enhancement of heat transfer due to milling of the tube, for either heat transfer or condensation rate, is therefore

$$E = \frac{\dot{q}'_{\text{ft,min}}}{\dot{q}'} = \frac{30.9}{28.3} = \boxed{1.09}$$

♦ The correct answer is **A**.

Answer Summary

Problem 1		**D**
Problem 2	2A	**C**
	2B	**D**
Problem 3		**C**
Problem 4		**B**
Problem 5		Open-ended pb.
Problem 6		**B**
Problem 7		Open-ended pb.
Problem 8	8A	**C**
	8B	**C**
Problem 9	9A	**A**
	9B	**C**
Problem 10		**A**
Problem 11		**A**
Problem 12		**B**
Problem 13		**B**
Problem 14	14A	**D**
	14B	**A**

References

- BERGMAN, T., LAVINE, A., INCROPERA, F., and DEWITT, D. (2011). *Fundamentals of Heat and Mass Transfer*. 7th edition. Hoboken: John Wiley and Sons.
- ÇENGEL, Y. and GHAJAR, A. (2015) Heat and *Mass Transfer: Fundamentals and Applications.* 5th edition. New York: McGraw-Hill.

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