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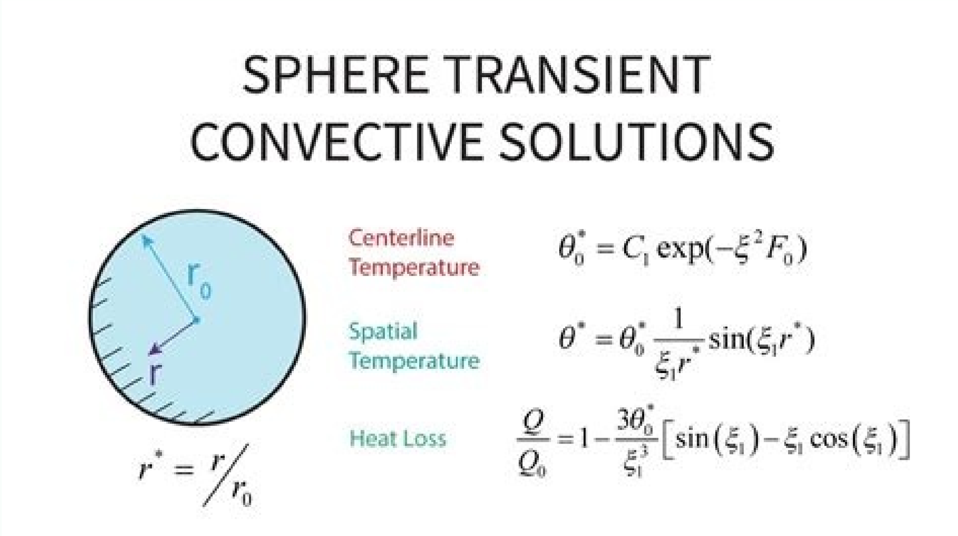

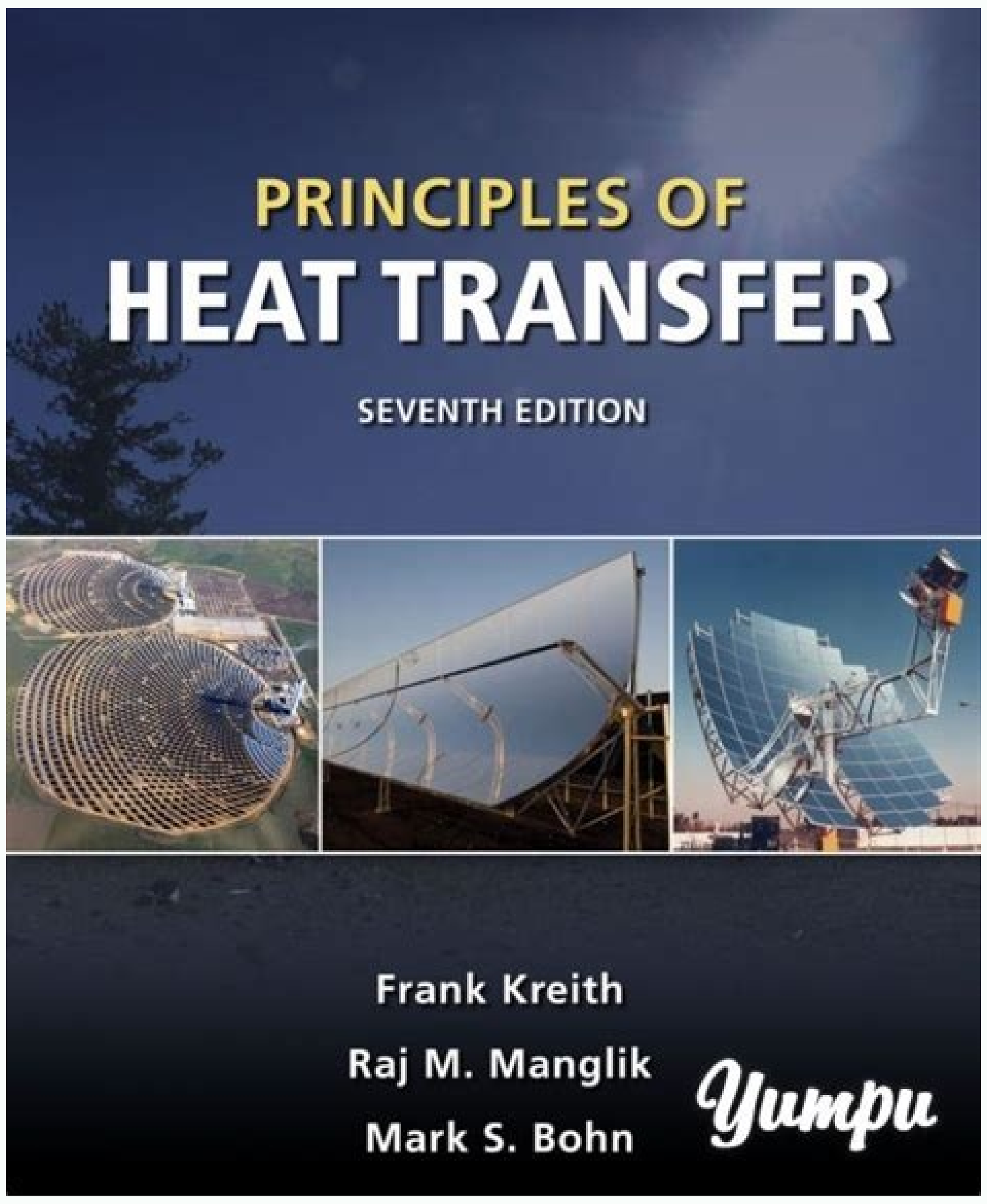

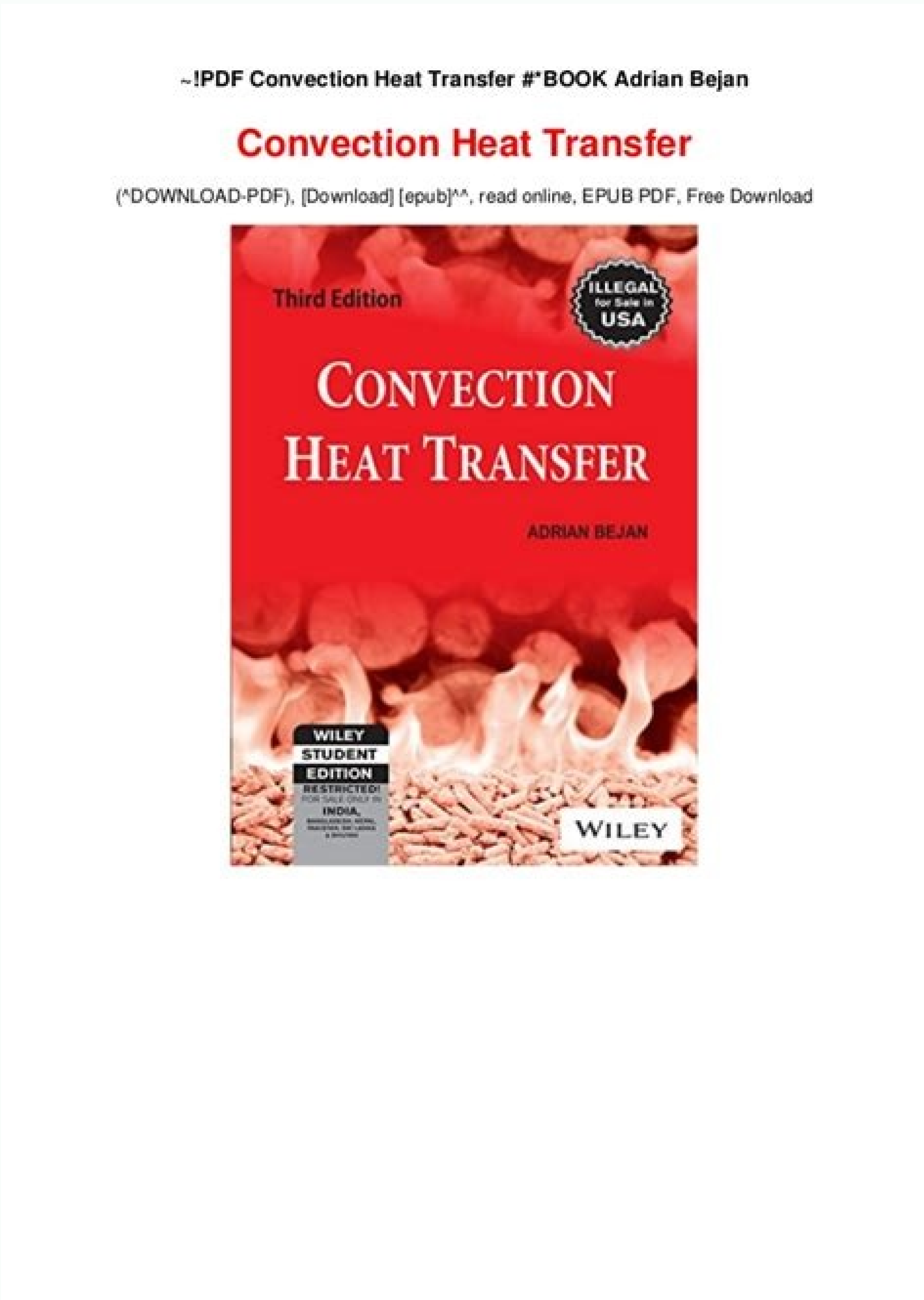

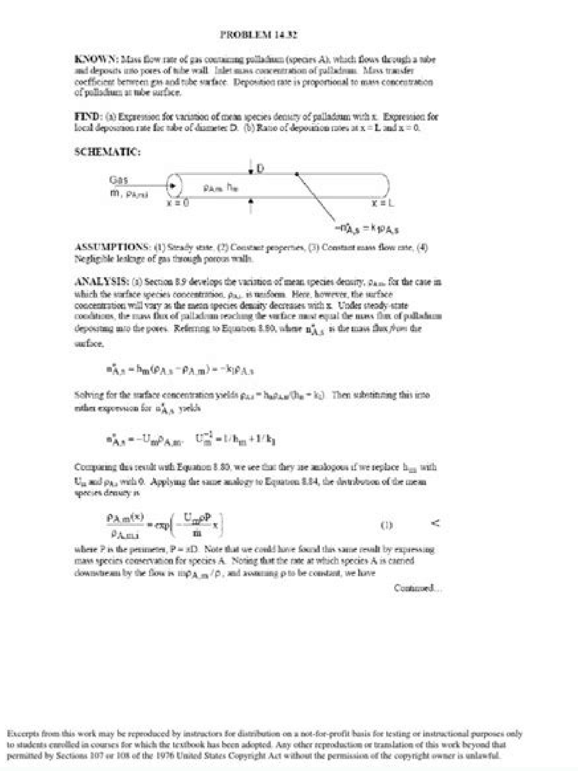

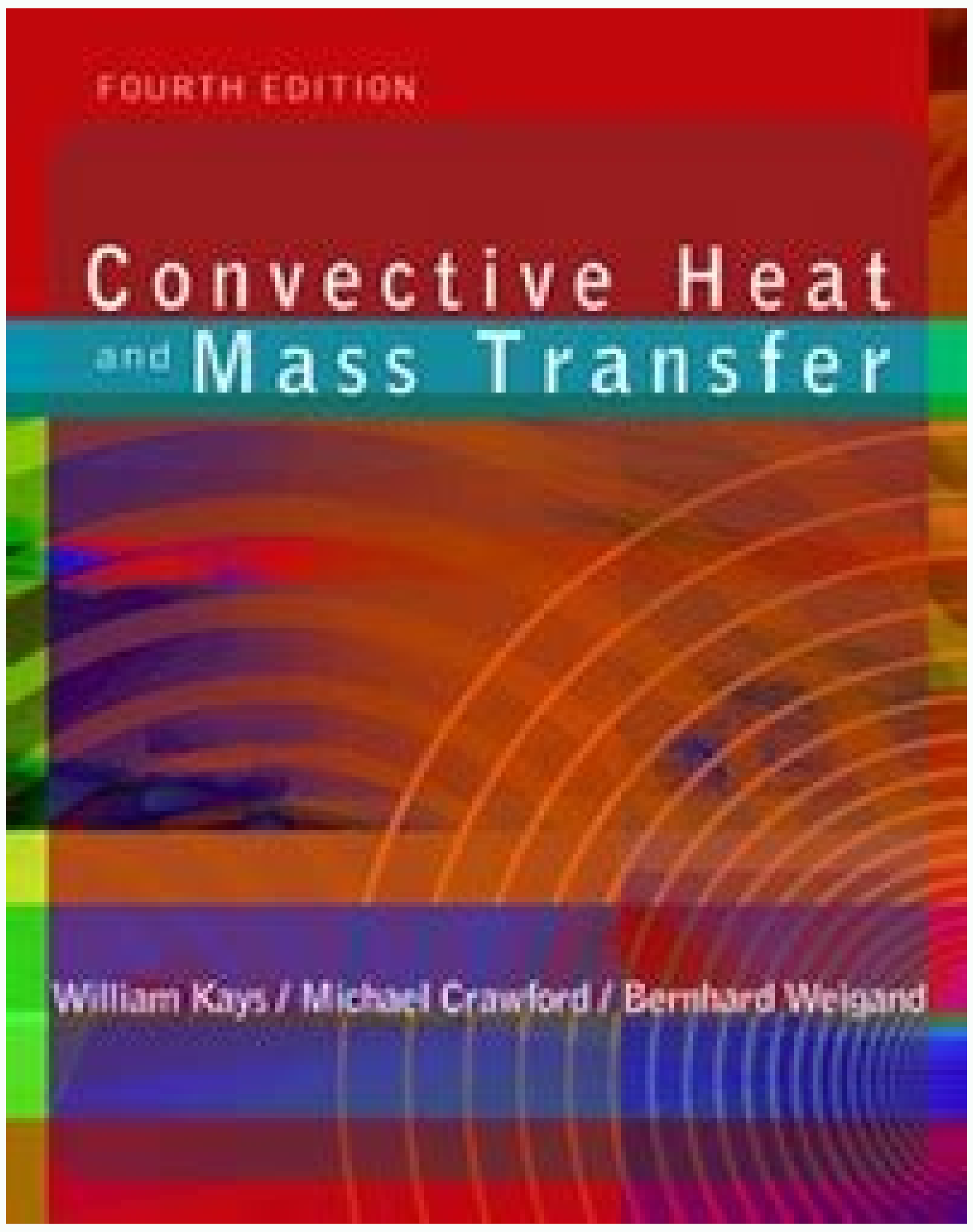

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3 Radiation effects are negligible 4 Heat transfer from the backside of the plate is negligible. Then the continuity equation reduces to ∂u ∂v ∂u + = 0 → Continuity: = 0 → u = u(y) 3000 rpm ∂x ∂x ∂y Therefore, the x-component of velocity does not change 12 m/s 6 cm in the flow direction (i.e., the velocity profile remains unchanged). ° C 2°C 4 / From Table 4-1 we read, for a sphere, λ1 = 1.635 and A1 = 1.302. 5-14C A node on an insulated boundary can be treated as an interior node in the finite difference formulation of a plane wall by replacing the insulation on the boundary by a mirror, and considering the reflection of the medium as its extension. 0 Properties The thermal conductivity is given to be k = 4.5 Btu/h·ft·°F. Properties The properties of air at 1 atm and the film temperature of (Ts + T∞)/2 = (12+5)/2 = 8.5°C are (Table A-15) Air V∞ = 55 km/h T∞ = 5°C k = 0.02428 W/m.°C υ = 1.413 × 10 -5 m 2 /s Pr = 0.7340 Ts = 12°C Analysis Air flows parallel to the 10 m side: The Reynolds number in this case is V L [(55 × 1000 / 3600)m/s](10 m) Re L = ∞ = = 1.081 × 10 7 υ 1.413 × 10 −5 m 2 /s L which is greater than the critical Reynolds number. Finite difference formulation is to be obtained, and the top and bottom surface temperatures under steady conditions are to be determined. Assumptions 1 Heat conduction in the plates is one-dimensional since the plate is large relative to its thickness and there is thermal symmetry about the center plane. (c) The temperature at the center of the sphere (r = 0) is determined by substituting the known quantities to be g&r 2 g& 2 (4 × 10 7 W/m 3)(0.04 m) 2 T (0) = T s + (r0 − 0 2) = Ts + 0 = 80°C + = 791°C 6k 6k 6 × (15 W/ m.°C) Thus the temperature at center will be about 711°C above the temperature of the outer surface of the sphere. o qs, W/m2 Analysis The nodal spacing is given to be Δx = 0.25 m. Substituting the given quantities, the maximum allowable time step becomes Δt ≤ (0.125 / 12 ft) 2 2(4.2 ×10 −6 ft 2 /s)[1 + (2.6 Btu/h.ft 2 .°F)(0.125 / 12 m) /(0.48 Btu/h.ft.°F)] = 12.2 s Therefore, any time step less than 12.2 s can be used to solve this problem. ° F)(2.5 / 12 ft) = = 56.8 k (0.44 Btu / h. ⎬ Ti − T∞ 0.45 W/m.º C 1 k ⎪ = = = 0.0136 ⎪ Bi hro (440W/m².º C)(0.0753m) ⎭ τ= αt = 4-55 (Fig.4 − 15) Chapter 4 Transient Heat Conduction 4-56 Chapter 4 Transient Heat Conduction Transient Heat Conduction in Semi-Infinite Solids 4-60C A semi-infinite medium is an idealized body which has a single exposed plane surface and extends to infinity in all directions. Assumptions The body temperature changes uniformly. 3 For exhaust gases, air properties are used. The finite k difference formulation for the steady case is obtained by simply setting Tmi +1 = Tmi and disregarding the time index i. Substituting, we get 1 W / m2 . Analysis The Biot number is Bi = hro (467 W/m 2 .°C)(0.011 m) = = 6.66 k (0.771 W/m.°C) Water 94°C The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, Hot dog λ1 = 2.0785 and A1 = 1.5357 The Fourier number is τ= αt L2 = (2.017 × 10 −7 m 2 /s)(4 min × 60 s/min) (0.011 m) 2 = 0.4001 > 0.2 Then the temperature at the center of the hot dog is determined to be θ o,cyl = 2 2 T0 − T∞ = A1e −λ1 τ = (1.5357)e −(2.0785) (0.4001) = 0.2727 Ti − T∞ T0 − 94 = 0.2727 → T0 = 73.8 °C 20 − 94 From Table 4-2 we read J 0 =0.2194 corresponding to the constant λ 1 =2.0785. Assumptions Constant properties given in the problem can be used. 7-80C The optimum thickness of insulation is the thickness that corresponds to a minimum combined cost of insulation and heat lost. ° C k From Table 4-1 we read, for a plane wall, λ1 = 1.308 and A1=1.239. Using energy balances, the finite difference equations for each of the 9 nodes are obtained as follows: Node 1: hi Thermal symmetry 1 • • Thermal symmetry i +1 i +1 i +1 i Δy Δy T2i +1 − T1i +1 Δx T4 − T1 Δx Δy T1 − T1 +k (Ti − T1i +1) + k = ρC Δy 2 2 2 Δx 2 2 Δt T i +1 − T2i +1 Δy T1i +1 − T2i +1 Δy T3i +1 − T2i +1 Δy T2i +1 − T2i +k + kΔx 5 = ρCΔx Δx Δx 2 2 Δy 2 Δt Node 3: ho i +1 i +1 i +1 i Δy Δy T2i +1 − T3i +1 Δx T6 − T3 Δx Δy T3 − T3 +k (To − T3i +1) + k = ρC Δy 2 2 2 Δx 2 2 Δt N4: hi Δy (Ti − T4i +1) + k Node 5: kΔy T i +1 − T4i +1 Δx T4i +1 − T4i Δx T1i +1 − T4i +1 Δx T7i +1 − T4i +1 = ρCΔy +k + kΔy 5 2 Δy 2 Δy Δx 2 Δt T4i +1 − T5i +1 T i +1 − T5i +1 T i +1 − T5i +1 T i +1 − T5i +1 T i +1 − T5i + kΔy 6 + kΔx 8 + kΔx 2 = ρCΔxΔy 5 Δx Δx Δy Δy Δt N6: ho Δy (Ti − T6i +1) + k Node 7: 5 W + hi T i +1 − T6i +1 Δx T3i +1 − T6i +1 Δx T9i +1 − T6i +1 Δx T6i +1 − T6i +k + kΔy 5 = ρCΔy 2 Δy 2 Δy Δx 2 Δt i +1 i +1 i +1 i Δy Δy T8i +1 − T7i +1 Δx T4 − T7 Δx Δy T7 − T7 +k (Ti − T7i +1) + k = ρC Δy 2 2 2 Δx 2 2 Δt 5-94 Chapter 5 Numerical Methods in Heat Conduction Node 8: k T i +1 − T8i +1 Δy T7i +1 − T8i +1 Δy T9i +1 − T8i +1 Δy T8i +1 − T8i +k + kΔx 5 = ρCΔx Δx Δx Δy 2 2 2 Δt Node 9: ho i +1 i +1 i +1 i Δy Δy T8i +1 − T9i +1 Δx T6 − T9 Δx Δy T9 − T9 (To − T9i +1) + k +k = ρC 2 2 Δy 2 Δx 2 2 Δt where k = 0.84 W/m.°C, α = k / ρC = 0.39 × 10 −6 m 2 / s , Ti = To = -3°C hi = 6 W/m2.°C, ho = 20 W/m2.°C, Δx = 0.002 m, and Δy = 0.01 m. 1-23C Warmer. T2 = 15°C Analysis The shape factor for this configuration is given in Table 3-5 to be 2πL S= 2 ⎛ 4 z − D1 2 − D 2 2 ⎞ cosh −1 ⎜ ⎟ ⎜ 2 D1 D 2 ⎟ ⎝ ⎠ D = 5 cm z = 40 cm 2π(8 m) L=8m = = 9.078 m 2 2 2 ⎞ ⎛ −1 ⎜ 4(0.4 m) − (0.05 m) − (0.05 m) ⎟ cosh ⎜ ⎟ 2(0.05 m)(0.05 m) ⎠ ⎝ ⎠ Then the steady rate of heat transfer between the pipes becomes Q& = Sk (T − T) = (9.078 m)(0.75 W/m.°C)(60 − 15)°C = 306 W 1 2 3-93 Chapter 3 Steady Heat Conduction 3-125 "!PROBLEM 3-125" "GIVEN" L=8 "[m]" D_1=0.05 "[m]" D_2=D_1 "z=0.40 [m], parameter to be varied" T_1=60 "[C]" T_2=15 "[C]" k=0.75 "[W/m-C]" "ANALYSIS" S=(2*pi*L)/(arccosh((4*z^2-D_1^2-D_2^2)/(2*D_1*D_2))) Q_dot=S*k*(T_1-T_2) z [m] 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1 Q [W] 644.1 411.1 342.3 306.4 283.4 267 254.7 244.8 236.8 230 650 600 550 Q [W] 500 450 400 350 300 250 200 0.1 0.2 0.3 0.4 0.5 0.6 z [m] 3-94 0.7 0.8 0.9 1 Chapter 3 Steady Heat Conduction 3-126E A row of used uranium fuel rods are buried in the ground parallel to each other. Q& = 1 ∞ 2 → T1 = T∞ 2 + QR ° C / W) = 40.5° C total = 40° C + (3.2 W)(01484 Rtotal T −T & Q& = 1 2 → T2 = T1 − QR board = 40.5° C − (3.2 W)(0.00694 ° C / W) = 40.5 − 0.02 ≅ 40.5° C Rboard 3-115 A circuit board houses 80 logic chips on one side, dissipating 0.04 W each through the back side of the board to the surrounding medium. 3 Heat loss from the fin tip is given to benegligible. 4-57 Chapter 4 Transient Heat Conduction 4-64 An area is subjected to cold air for a 10-h period. 1-47C The parameters that effect the rate of heat conduction through a windowless wall are the geometry and surface area of wall, its thickness, the material of the wall, and the temperature difference across the wall. Therefore, we will use a different but intuitive approach. ° F / Btu 2 ho A3 ho (2πr3) (5 Btu / h.ft . 4-77 Chapter 4 Transient Heat Conduction 4-79 "!PROBLEM 4-79" "GIVEN" 2*L_1=0.04 "[m]" L_2=L_1 2*L_3=0.10 "[m]" "T_i=-20 [C], parameter to be varied" T_infinity=18 "[C]" h=12 "[W/m^2-C]" T_L1_L2_L3=0 "[C]" "PROPERTIES" k=2.22 "[W/m-C]" alpha=0.124E-7 "[m^2/s]" "ANALYSIS" "This block can physically be formed by the intersection of two infinite plane wall of thickness 2L=4 cm and an infinite plane wall of thickness 2L=10 cm" "For the two plane walls" Bi_w1=(h*L_1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda_1_w1=0.3208 "w stands for wall" A_1_w1=1.0173 time*Convert(min, s)=tau_w1*L_1^2/alpha "For the third plane wall" Bi_w3=(h*L_3)/k "From Table 4-1 corresponding to this Bi number, we read" lambda_1_w3=0.4951 A_1_w3=1.0408 time*Convert(min, s)=tau_w3*L_3^2/alpha theta_L_w1=A_1_w1*exp(lambda_1_w1^2*tau_w1)*Cos(lambda_1_w1*L_1/L_1) "theta_L_w1=(T_L_w1T_infinity)/(T_i-T_infinity)" theta_L_w3=A_1_w3*exp(lambda_1_w3^2*tau_w3)*Cos(lambda_1_w3*L_3/L_3) "theta_L_w3=(T_L_w3T_infinity)/(T_i-T_infinity)" (T_L1_L2_L3-T_infinity)/(T_i-T_infinity)=theta_L_w1^2*theta_L_w3 "corner temperature" Ti [C] -26 -24 -22 -20 -18 time [min] 1614 1512 1405 1292 1173 4-78 Chapter 4 Transient Heat Conduction -16 -14 -12 -10 -8 -6 -4 1048 914.9 773.3 621.9 459.4 283.7 92.84 1800 1600 tim e [m in] 1400 1200 1000 800 600 400 200 0 -30 -25 -20 -15 T i [C] 4-79 -10 -5 0 Chapter 4 Transient Heat Conduction 4-80 A cylindrical ice block is placed on a table. Therefore, T1 = T9, T2 = T10, T3 = T11, T4 = T7, and T5 = T8. 3 The heat transfer coefficient also includes the radiation effects. The rate of heat loss from the collector by convection and radiation during a calm day are to be determined. Assumptions 1 Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values. 1-135 Engine valves are to be heated in a heat treatment section. 7-74C In cold surfaces such as chilled water lines, refrigerated trucks, and air conditioning ducts, insulation saves energy since the source of "coldness" is refrigeration that requires energy input. 4-12 Chapter 4 Transient Heat Conduction 4-23 A number of carbon steel balls are to be annealed by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The density and latent heat of fusion of water at 0°C are ρ = 1000 kg/m3 and hif = 333.7 kJ/kg (Table A-9). 4 The heat transfer coefficient is constant and uniform over the entire spoon surface. Properties The thermal conductivity is given to be k = 16 Btu/h.ft·°C. Multiplying the number of time steps N by the time step Δt = 5 s will give the defrosting time. Properties The thermal conductivities are given to be k = 0.16 W/m·°C for plastic pipe and k = 0.035 W/m·°C for fiberglass insulation. The density of the cylinder is ρ, the specific heat is C, and the area of the cylinder normal to the direction of heat transfer is A = πD 2 / 4 , which is constant.)e − (0.8784) (0.243) = 0.912 Ti − T∞ T0 − 1700 = 0.912 → T0 = 228°F 85 − 1700 4-29 Chapter 4 Transient Heat Conduction 4-41 Steaks are cooled by passing them through a refrigeration room. Assumptions 1 Heat transfer in both the steaks and the defrosting plate is one-dimensional since heat transfer from the lateral surfaces is negligible. The temperatures at the center of the top surface and at the corner after a specified period of cooling are to be determined. The properties of air at 1 atm and at this temperature are (Table A-15) k = 0.02735 W/m.°C υ = 1.798 × 10 -5 m 2 /s Air V∞ = 150 m/min T∞ = 40°C Pr = 0.7228 Analysis The Reynolds number is V D (150/60 m/s)(0.003 m) Re = ∞ = = 417.1 υ 1.798 × 10 −5 m 2 /s The proper relation for Nusselt number corresponding to this Reynolds number is hD 0.62 Re 0.5 Pr 1 / 3 Nu = = 0.3 + 1/ 4 k 1 + (0.4 / Pr) 2 / 3 [ ] ⎡ ⎛ Re ⎞ 5 / 8 ⎤ ⎥ 1 + ⎢ ⎜ ⎟ ⎥ ⎢ ⎝ 282,000 ⎠ ⎦ ⎣] Resistor 0.4 W D = 0.3 cm 4/5 5/8 0.62(417.1) 0.5 (0.7228)1 / 3 ⎡ ⎛ 417.1 ⎞ ⎤ ⎥ = 0.3 + 1 + ⎢ ⎜ ⎟ 1 / 4 ⎢ ⎝ 282,000 ⎠ ⎥ ⎦ 1 + (0.4 / 0.7228) 2 / 3 The heat transfer coefficient is 0.02735 W/m.°C k h = Nu = (10.43) = 95.09 W/m 2 .°C D 0.003 m Then the surface temperature of the component becomes [Q& 4/5 = 10.43 As = πDL = π (0.003 m)(0.018 m) = 0.0001696 m 2 Q& 0.4 W → Ts = T∞ + = 40 °C + = 64.8°C Q& = hAs (T s − T∞) 2 hA (95.09 W/m .°C)(0.0001696 m 2) 7-49 Chapter 7 External Forced Convection 7-58 A cylindrical hot water tank is exposed to windy air. 2-3C Heat transfer to a canned drink can be modeled as two-dimensional since temperature differences (and thus heat transfer) will exist in the radial and axial directions (but there will be symmetry about the center line and no heat transfer in the azimuthal direction. Note that we will use the base area and we assume the temperature of the surrounding surfaces are at the same temperature with the air (Tsurr = 25 ° C) L = 10 cm Q& rad = εAs σ (Ts 4 − Tsurr 4) = (0.90)[(0.1 m)(0.062 m)](5.67 × 10 −8 W/m 2 .°C)[(60 + 273 K) 4 − (25 + 273 K) 4] = 1.4 W The heat transfer rate by convection will be 1.4 W less than total rate of heat transfer from the transformer. Also, the mesh Fourier number is τ= αΔt Δx 2 = (015 . Then the energy equation with viscous dissipation reduce to 2 2 ⎛ ∂u ⎞ d 2T ⎛V⎞ ⎟ ⎜ ⎟ + μ → k = − μ ⎜ ∂y 2 dy 2 ⎝L⎠ ⎝ ∂y ⎠ since ∂u / ∂y = V / L . 4 The pipe is insulated and thus the heat losses are negligible. Solar energy is incident on the pond surface at for 4 h The temperature distribution in the pond under the most favorable conditions is to be determined. Dividing both sides by k and integrating twice give Energy: 0=k ∂ 2T 2 T (y) = − μ ⎛ y ⎞ ⎛ V⎞ + C3 y + C 4 2k ⎝ L ⎠ Applying the boundary conditions T(0) = T0 and T(L) = T0 gives the temperature distribution to be T (y) = T0 + μV 2 2k ⎛ y y2 ⎞ − ⎟ ⎜ L L2 ⎜ ⎝ ⎠ The temperature gradient is determined by differentiating T(y) with respect to y, y⎞ dT μV 2 ⎛ = ⎜1 − 2 ⎟ dy 2kL ⎝ L⎠ The location of maximum temperature is determined by setting dT/dy = 0 and solving for y, y⎞ dT μV 2 ⎛ L = → y= ⎜1 − 2 ⎟ = 0 2 dy 2kL ⎝ L⎠ 6-11 Chapter 6 Fundamentals of Convection Therefore, maximum temperature will occur at mid plane in the oil. at r = r1 : 4k 4k 2-72 Chapter 2 Heat Conduction Equation Substituting this C2 relation into Eq. (b) and rearranging give g& Twire (r) = TI + (r12 − r 2) (c) 4 k wire Plastic layer The mathematical formulation of heat transfer problem in the plastic can be expressed as d ⎛ dT ⎞ ⎟ r ⎜ =0 dr ⎝ dr ⎠ dT (r2) = h[T (r2) − T∞] dr The solution of the differential equation is determined by integration to be dT C1 r = C1 → = → T (r) = C1 ln r + C2 dr r dr where C1 and C2 are arbitrary constants. 3-45 The thickness of copper plate whose thermal resistance is equal to the thermal contact resistance is to be determined. 3-26 Chapter 3 Steady Heat Conduction 3-52 A wall consists of horizontal bricks separated by plaster layers. • • . ° C 0.035 W / m . 6 The pressure of air is 1 atm. Then T0 the number of nodes becomes M = L / Δx + 1 = 0.3/0.1+1 = 4. 1-98C The human body loses heat by convection, radiation, and evaporation in both summer and winter. ° C)(70 − 67)° C = 13,296 W The thermal resistances for convection in the pipe and the pipe itself are Rpipe Rconv,i Rconv ,i Rpipe Rcombined ,o T∞1 ln(r2 / r1) ln(175 . 7-110 Design and Essay Problems KJ 7-101 88 110 Chapter 8 Internal Forced Convection Chapter 8 INTERNAL FORCED CONVECTION General Flow Analysis 8-1C Liquids are usually transported in circular pipes because pipes with a circular cross-section can withstand large pressure differences between the inside and the outside without undergoing any distortion. 1-64 Chapter 1 Basics of Heat Transfer 1-65 Chapter 1 Basics of Heat Transfer Review Problems 1-121 Cold water is to be heated in a 1200-W teapot. 2-55C Yes, this claim is reasonable since no heat is entering the cylinder and thus there can be no heat transfer from the cylinder in steady operation. Then heat will continue to be transferred from the outer parts of the turkey to the inner as a result of temperature difference. 5 The Biot number in this case is large (much larger than 0.1). × 10 −6 m2 / s)(20 min × 60 s / min) (0.025 m) 2 4-83 = 2.208 > 0.2 Hot gases 500°C 5 cm × 5 }= 0.284 Ti = 20°C Chapter 4 Transient Heat Conduction { 2 T (0,0,0, t) − 500 = (1.0580)e −(0.5932) (2.208) 20 − 500 } {(1.1016)e 2 − (0.7910) 2 (2.208) }= 0.0654 → T (0,0,0, t) = 469°C After 60 minutes τ= αt 2 L = (115 . Then the pressure drop across the tube bank becomes ΔP = N L fχ 2 ρVmax (0.6746 kg/m 3)(6.102 m/s) 2 = 16(0.18)(1) 2 2 ⎛ 1N ⎞ ⎜ 1 kg · m/s 2 ⎟ ⎝ ⎠ = 36.2 Pa ⎟ ⎠ (c) The temperature rise of water is Q& = m& water C p , water ΔTwater → ΔTwater = Q& 115.425 kW = = 4.6°C m& water C p , water (6 kg/s)(4.18 kJ/kg · °C) Discussion The arithmetic mean fluid temperature is (Ti + Te)/2 = (300 + 237)/2 = 269°C, which is sufficiently close to the assumed value of 250°C. The finite difference formulation of the problem is to be obtained, and the tip temperature of the spoon as well as the rate of heat transfer from the exposed surfaces are to be determined. Analysis (a) Approximate solution This problem can be solved approximately by using an average temperature Electronic for the device when evaluating the heat loss. 2 The energy stored in the glass containers themselves is negligible relative to the energy stored in water. 2 Heat transfer is one-dimensional since there is thermal symmetry about the mid point. Assumptions 1 Heat transfer is steady since there is no change r with time. 4 The given heat transfer coefficient accounts for the radiation effects. The time it will take for the temperature of the inner surfaces of the house to start changing is to be determined. 1-56C Diamond is a better heat conductor. These properties will be used for both fresh and frozen meat. 7-10C At sufficiently high velocities, the fluid stream detaches itself from the surface of the body. Analysis Using the energy balance approach and taking the direction of all heat transfers to be towards the node under consideration, the g(x, t) implicit finite difference formulations become q0 Left boundary node: h, T∞ Δx i +1 i +1 i +1 i T1 − T0 Δx T0 − T0 i +1 + q& 0 A + g& 0 (AΔx / 2) = ρA C kA • • • • • Δx 2 Δt 0 1 2 3 4 Right boundary node: kA T3i +1 − T4i +1 Δx T4i +1 − T4i + hA(T∞i +1 − T4i +1) + g& 4i +1 (AΔx / 2) = ρA C Δx 2 Δt 5-76 A plane wall with variable heat generation and constant thermal conductivity is subjected to insulation at the left (node 0) and radiation at the right boundary (node 5). The product solution for this problem can be written as θ (L, ro , t) block = θ (L, t) wall θ (ro , t) cyl 2 2 0 − 20 ⎡ = A1e −λ1 τ cos(λ1 L / L)⎤ ⎡ A1 e −λ1 τ J 0 (λ1 ro / ro)⎤ ⎥ ⎢ ⎥ ⎦ ⎣ Ti − 20 ⎣ [ ]⎡ 2 2 0 − 20 = (1.0187)e −(0.3318) (0.2232) cos(0.3318) (1.0146)e −(0.3407) (0.8928) (0.9707) Ti − 20 which gives ⎦ Ti = −4° C Therefore, the ice will not start melting for at least 2 hours if its initial temperature is -4°C or below. Analysis In this problem there is a question of which surface area to use. Such docks save valuable storage space from being used for shipping purpose, and provide a more acceptable working environment for the employees. The bearing is cooled externally by a liquid. Column 1 contains the time, column 2 the value of T[1], column 3, the value of T[2], etc., and column 10 the Row." Time=TableValue('Table 1',Row-1,#Time)+DELTAt Duplicate i=1,8 T_old[i]=TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight equations for the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_infinity-T_old[1])+k*l/2*(T_old[2]T_old[1])/l+k*l/2*(T_old[4]-T_old[1])/l+g_dot*l^2/4=RhoC*l^2/4*(T[1]T_old[1])/DELTAt "Node 1" h*l*(T_infinity-T_old[2])+k*l/2*(T_old[1]-T_old[2])/l+k*l/2*(T_old[3]T_old[2])/l+k*l*(T_old[5]-T_old[2])/l+g_dot*l^2/2=RhoC*l^2/2*(T[2]T_old[2])/DELTAt "Node 2" h*l*(T_infinity-T_old[3])+k*l/2*(T_old[2]-T_old[3])/l+k*l/2*(T_old[6]T_old[3])/l+g_dot*l^2/4=RhoC*l^2/4*(T[3]-T_old[3])/DELTAt "Node 3" q_dot_L*l+k*l/2*(T_old[1]-T_old[4])/l+k*l/2*(T_bottomT_old[4])/l+k*l*(T_old[5]-T_old[4])/l+g_dot*l^2/2=RhoC*l^2/2*(T[4]T_old[4])/DELTAt "Node 4" T[5]=(14*tau)*T_old[5]+tau*(T_old[2]+T_old[4]+T_old[6]+T_bottom+g_dot*l^2/k) "Node 5" 5-85 Chapter 5 Numerical Methods in Heat Conduction h*l*(T_infinity-T_old[6])+k*l/2*(T_old[3]-T_old[6])/l+k*l/2*(T_old[5]T_old[6])/l+k*l/2*(T_bottom-T_old[6])/l+k*l/2*(T_old[7]T_old[6])/l+g_dot*3/4*l^2=RhoC*3/4*l^2*(T[6]-T_old[6])/DELTAt "Node 6" h*l*(T_infinity-T_old[7])+k*l/2*(T_old[6]-T_old[7])/l+k*l/2*(T_old[8]T_old[7])/l+k*l*(T_bottom-T_old[7])/l+g_dot*l^2/2=RhoC*l^2/2*(T[7]T_old[7])/DELTAt "Node 7" h*l/2*(T_infinity-T_old[8])+k*l/2*(T_old[7]-T_old[8])/l+k*l/2*(T_bottomT_old[8])/l+g_dot*l^2/4=RhoC*l^2/4*(T[8]-T_old[8])/DELTAt "Node 8" 5-86 Chapter 5 Numerical Methods in Heat Conduction Time [s] 0 15 30 45 60 75 90 105 120 135 … … 1650 1665 1680 1695 1710 1725 1740 1755 1770 1785 T1 [C] T2 [C] T3 [C] T4 [C] T5 [C] T6 [C] T7 [C] T8 [C] Row 140 203.5 265 319 365.5 404.6 437.4 464.7 487.4 506.2 … … 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 140 200.1 259.7 312.7 357.4 394.9 426.1 451.9 473.3 491 … … 575.7 575.7 575.7 575.7 575.7 575.7 575.7 575.7 575.7 575.7 140 196.1 252.4 300.3 340.3 373.2 400.3 422.5 440.9 456.1 … … 528.5 528.5 528.5 528.5 528.5 528.5 528.5 528.5 528.5 528.5 140 207.4 258.2 299.9 334.6 363.6 387.8 407.9 424.5 438.4 … … 504.6 504.6 504.6 504.6 504.6 504.6 504.6 504.6 504.6 504.6 140 204 253.7 293.5 326.4 353.5 375.9 394.5 409.8 422.5 … … 483.1 483.1 483.1 483.1 483.1 483.1 483.1 483.1 483.1 483.1 140 201.4 243.7 275.7 300.7 320.6 336.7 349.9 360.7 369.6 … … 411.9 411.9 411.9 411.9 411.9 411.9 411.9 411.9 411.9 411.9 140 200.1 232.7 252.4 265.2 274.1 280.8 286 290.1 293.4 … … 308.8 308.8 308.8 308.8 308.8 308.8 308.8 308.8 308.8 308.8 140 200.1 232.5 250.1 260.4 267 271.6 275 277.5 279.6 … … 288.9 288.9 288.9 288.9 288.9 288.9 288.9 288.9 288.9 288.9 550 500 450 T 3 [C] 400 350 300 250 200 150 100 0 200 400 600 800 1000 1200 1400 1600 1800 Time [s] 5-87 1 2 3 4 5 6 7 8 9 10 … … 111 112 113 114 115 116 117 118 119 120 Chapter 5 Numerical Methods in Heat Conduction 5-89 A long solid bar is subjected to transient two-dimensional heat transfer. The enthalpy of vaporization of water at 100°C is hfg = 2257 kJ/kg-K (Table A9). Assumptions 1 Heat conduction in the wires is one-dimensional in the radial direction. However, it is not proper to use this model when finding temperatures near the bottom and the top of the cylinder. energies & 1 = mh & 2 (since Δke ≅ Δpe ≅ 0) Q& in + mh & p (T2 − T1) Q& in = mC 450 ft3/min (a) The inlet velocity of air through the duct is determined from V& V& 450 ft 3/min V1 = 1 = 12 = = 825 ft/min A1 πr π(5/12 ft)2 (b) The mass flow rate of air becomes ( ) RT1 0.3704psia · ft 3 /lbm · R (510R) = = 12.6 ft 3 / lbm P1 (15psia) V& 450ft 3 /min m& = 1 = = 35.7lbm/min = 0.595lbm/s v1 12.6ft 3 /lbm v1 = Then the exit temperature of air is determined to be Q& 2 Btu/s = 64.0°F T2 = T1 + in = 50°F + & (0.595lbm/s)(0.2404Btu/lbm · °F) mC p 1-21 AIR 2 Btu/s D = 10 in Chapter 1 Basics of Heat Transfer 1-43 Water is heated in an insulated tube by an electric resistance heater. The thermal resistance of the wall and its R-value of insulation are to be determined. 5-2C The analytical solutions are based on (1) driving the governing differential equation by performing an energy balance on a differential volume element, (2) expressing the boundary conditions in the proper mathematical form, and (3) solving the differential equation and applying the boundary conditions to determine the integration constants. and A1 = 1.2732 Orange The Fourier number is τ= αt L2 = (0.136 × 10 −6 m 2 /s)(4 h × 3600 s/h) (0.04 m) 2 = 1.224 > 0.2 Ti = 15°C Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. 5 The heat transfer coefficient on the top and bottom surfaces is the same as that on the side surfaces. The time for the plate temperature to reach 140°C and whether it is realistic to assume the plate temperature to be uniform at all times are to be determined. 1-75 Water 0.2 L 20°C Chapter 1 Basics of Heat Transfer 1-133 "GIVEN" V=0.0002 "[m^3]" T_w1=20 "[C]" T_w2=5 "[C]" "T_ice=0 [C], parameter to be varied" T_melting=0 "[C]" "PROPERTIES" rho=density(water, T=25, P=101.3) "at room temperature" C_w=CP(water, T=25, P=101.3) "at room temperature" C_ice=c_('Ice', T_ice) h_if=333.7 "[kJ/kg]" "ANALYSIS" m_w=rho*V DELTAU_ice+DELTAU_w=0 "energy balance" DELTAU_ice=m_ice*C_ice*(T_melting-T_ice)+m_ice*h_if DELTAU_w=m_w*C_w*(T_w2-T_w1) Tice [C] -24 -22 -20 -18 -16 -14 -12 -10 -8 -6 -4 -2 0 mice [kg] 0.03291 0.03323 0.03355 0.03389 0.03424 0.0346 0.03497 0.03536 0.03575 0.03616 0.03658 0.03702 0.03747 0.038 0.037 m ice [kg] 0.036 0.035 0.034 0.033 0.032 -24 -20 -16 -12 T ice [C] 1-76 -8 -4 0 Chapter 1 Basics of Heat Transfer 1-134E A 1-short ton (2000 lbm) of water at 70°F is to be cooled in a tank by pouring 160 lbm of ice at 25°F into it. k (2.5 W/m.°C) Bi = The Biot number and the corresponding constants for h = 80 W / m2 . The thermal properties of the defrosting plate are k = 401 W/m.°C, α = 117 × 10 −6 m 2 / s , and ε = 0.90 (Table A-3). The finite difference formulation of this problem is to be obtained, and the nodal temperatures under steady conditions are to be determined. Ti = 20°C The Fourier number is τ= αt L2 = (115 . Assumptions 1 The temperature in the wall is affected by the thermal conditions at outer surfaces only. Considering a unit surface area, Q& = h(A(T − T) = (9 W / m 2 . Refrigeration prevents or delays the spoilage of foods by reducing the rate of growth of microorganisms. = A1e − λ1 τ = (15618 Ti − T∞ Then the center temperature of the short cylinder becomes ⎡ T (0,0, t) − T∞ ⎤ = θ o, wall × θ o,cyl = 1× 0.106 = 0.106 ⎥ ⎢ Ti − T∞ ⎦ short cylinder T (0,0, t) − 212 = 0.106 → T (0,0, t) = 194°F 40 − 212 4-73 Chapter 4 Transient Heat Conduction After 10 minutes τ= αt L2 = 0.0,wall = τ= αt ro 2 θ o ,cyl = (0.0077 ft 2 / h)(10 / 60 h) (2.5 / 12 ft) 2 = 0.03 < 0.2 (Be cautious!) 2 2 T0 − T∞ = A1e −λ1 τ = (1.2728)e −(1.5421) (0.03) ≅ 1 Ti − T∞ = (0.0077 ft 2 / h)(10 / 60 h) (0.4 / 12 ft) 2 = 1156 > 0.2 . ° C)(0.01 m2) Q& The total thermal resistance is then Rtotal = Rcontact + Rplate + Rconvection = 0.0227 + 0.0031 + 3.333 = 3.359 ° C / W Note that the thermal resistance of copper plate is very small and can be ignored all together. Because energy is added to the room air in the form of electrical work. 4-89C Cooking kills the microorganisms in foods, and thus prevents spoilage of foods. Then the minimum thickness of fiberglass insulation required is t = r3 - r2 = 0.0362 - 0.0230 = 0.0132 m = 1.32 cm Therefore, insulating the pipe with at least 1.32 cm thick fiberglass insulation will ensure that the outer surface temperature of the pipe will be at 30°C or below. The length of time the block should be kept in the furnace and the amount of heat transferred to the block are to be determined. Therefore, taking T2 the outer surface temperature of the plate to be T2 (absolute, in R), 520 R T −T k As 1 2 = εσAs T24 L Canceling the area A and substituting the known quantities, (520 R) − T2 (12 . 1 2 3 4 Analysis The nodal spacing is given to be Δx = 0.01 m. Therefore, the oranges will freeze. Heat is transferred from the inner surface of the pipe to the water by convection. Analysis The differential equation and the boundary conditions for this heat conduction problem can be expressed as ∂ 2 T ∂ 2 T 1 ∂T + = ∂x 2 ∂y 2 α ∂t h, T∞ b ∂T (x ,0, t) =0 ∂x ∂T (0, y , t) =0 ∂x ∂T (a , y , t) −k = h[T (a , y , t) − T∞] ∂x ∂T (x , b, t) −k = h[T (x , b, t) − T∞] ∂x h, T∞ a Insulated T (x , y ,0) = Ti 2-122 Heat is generated at a constant rate in a short cylinder. 4-84 Chapter 4 Transient Heat Conduction 4-83 A cylindrical aluminum block is heated in a furnace. The number of transistors that can be placed on this plate is to be determined. Node 1 (interior) : Node 3 (interior) : g&l 2 =0 k g&l 2 150 + 200 + T1 + T4 − 4T3 + =0 k 100 + 120 + T2 + T3 − 4T1 + Noting that T1 = T2 and T3 = T4 and substituting, (10 7 W/m 3)(0.1 m) 2 =0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 350 + T1 − 3T3 + =0 180 W/m · °C The solution of the above system is 100 • 120 • 220 + T3 − 3T1 + 100 • • 100 • 1 100 • • 2 • 120 • 4 • 150 g 150 • • 3 0.1 m • • 200 200 T1 = T2 = 411.5°C T3 = T4 = 439.0°C • 200 • 200 (b) The total rate of heat transfer from the top surface Q& top can be determined from an energy balance on a volume element at the top surface whose height is l/2, length 0.3 m, and depth 1 m: T − 100 ⎞ ⎛ 1 × 1 120 − 100 Q& top + g& 0 (0.3 × 1× l / 2) + ⎜ ⎜ 2k + 2kl × 1 1 ⎟ ⎟ = 0 l 2 l ⎠ ⎝ ⎛1m ⎞ Q& top = −(10 7 W/m 3)(0.3 × 0.1 / 2)m 3 − 2(180 W/m · °C)⎢ (120 − 100)°C + (1 m)(411.5 - 100)°C ⎥ 2 ⎝ ⎠ = 265,750 W (per m depth) 5-42 Chapter 5 Numerical Methods in Heat Conduction 5-51 "!PROBLEM 5-51" "GIVEN" k=180 "[W/m-C], parameter to be varied" g_dot=1E7 "[W/m^3], parameter to be varied" DELTAx=0.10 "[m]" DELTAy=0.10 "[m]" d=1 "[m], depth" "Temperatures at the selected nodes are also given in the figure" "ANALYSIS" "(a)" l=DELTAx T_1=T_2 "due to symmetry" T_3=T_4 "due to symmetry" "Using the finite difference method, the two equations for the two unknown temperatures are determined to be" 120+120+T_2+T_3-4*T_1+(g_dot*l^2)/k=0 150+200+T_1+T_4-4*T_3+(g_dot*l^2)/k=0 "(b)" "The rate of heat loss from the top surface can be determined from an energy balance on a volume element whose height is l/2, length 3*l, and depth d=1 m" -Q_dot_top+g_dot*(3*l*d*l/2)+2*(k*(l*d)/2*(120-100)/l+k*l*d*(T_1-100)/l)=0 k [W/m.C] 10 30.53 51.05 71.58 92.11 112.6 133.2 153.7 174.2 194.7 215.3 235.8 256.3 276.8 297.4 317.9 338.4 358.9 379.5 400 T1 [C] 5134 1772 1113 832.3 676.6 577.7 509.2 459.1 420.8 390.5 366 345.8 328.8 314.4 301.9 291 281.5 273 265.5 258.8 T3 [C] 5161 1799 1141 859.8 704.1 605.2 536.7 486.6 448.3 418 393.5 373.3 356.3 341.9 329.4 318.5 309 300.5 293 286.3 5-43 Qtop [W] 250875 252671 254467 256263 258059 259855 261651 263447 265243 267039 268836 270632 272428 274224 276020 277816 279612 281408 283204 285000 Chapter 5 Numerical Methods in Heat Conduction g [W/m3] 100000 5.358E+06 1.061E+07 1.587E+07 2.113E+07 2.639E+07 3.165E+07 3.691E+07 4.216E+07 4.742E+07 5.268E+07 5.794E+07 6.319E+07 6.845E+07 7.371E+07 7.897E+07 8.423E+07 8.948E+07 9.474E+07 1.000E+08 T1 [C] 136.5 282.6 428.6 574.7 720.7 866.8 1013 1159 1305 1451 1597 1743 1889 2035 2181 2327 2473 2619 2765 2912 T3 [C] 164 310.1 456.1 602.2 748.2 894.3 1040 1186 1332 1479 1625 1771 1917 2063 2209 2355 2501 2647 2793 2939 Qtop [W] 18250 149697 281145 412592 544039 675487 806934 938382 1.070E+06 1.201E+06 1.333E+06 1.464E+06 1.596E+06 1.727E+06 1.859E+06 1.990E+06 2.121E+06 2.253E+06 2.384E+06 2.516E+06 6000 285000 280000 5000 heat 275000 temperature 270000 3000 265000 2000 260000 1000 0 0 255000 50 100 150 200 250 k [W /m -C] 5-44 300 350 250000 400 Q top [W] T 1 [C] 4000 Chapter 5 Numerical Methods in Heat Conduction 6000 5000 T 3 [C] 4000 3000 2000 1000 0 0 50 100 150 200 250 300 350 400 3000 3.0 x 10 6 2500 2.5 x 10 6 2000 2.0 x 10 6 1500 1.5 x 10 6 1000 1.0 x 10 6 500 5.0 x 10 5 0.0 x 10 0 tem perature 1500 heat 0 2.2 x 10 7 4.4 x 10 7 6.6 x 10 7 g [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m -C] Chapter 5 Numerical Methods in Heat Conduction 3000 2500 T 3 [C] 2000 1500 1000 500 0 0 2.200 x 10 7 6.600 x 10 7 g [W /m ^3] 5-46 1.100 x 10 8 Chapter 5 Numerical Methods in Heat Conduction 5-52 A long solid body is subjected to steady two-dimensional heat transfer. 3 All the heat generated in the chips is conducted across the circuit board, and is dissipated from the backside of the board. Plasterboard, 20 mm 6. 2 Heat conduction in the orange is one-dimensional because of symmetry about the midpoint. Analysis (a) We take the pipe as the system. Substituting these values into the one-term solution gives T −T θ 0 = o ∞ = A1e − λ21τ → Ti − T∞ -6°C 18 / 4 − (−6) = 1517 . 7-72 Air is heated by hot tubes in a tube bank. The problem is solved in a systematic manner, showing all steps. °C)(22 − 10)°C = 9038 kJ Noting that 1 kWh = 3600 kJ, the cost of this electrical energy at a unit cost of $0.075/kWh is Enegy Cost = (Energy used)(Unit cost of energy) = (9038 / 3600 kWh)($0.075/kWh) = $0.19 Therefore, it will cost the homeowner about 19 cents to raise the temperature in his house from 10 to 22°C. 6-7C A fluid flow during which the density of the fluid remains nearly constant is called incompressible flow. Properties The thermal conductivity of the brick wall is given to be k = 0.40 Btu/h·ft·°F. Noting that D = D0 = 4.04 m in this case, the Nusselt number becomes Re = V∞ D [(40 × 1000/3600) m/s](4.04 m) = = 2.961× 10 6 υ 1.516 × 10 −5 m 2 /s [ ] ⎛ μ hD Nu = = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 ⎜ ∞ k ⎝ μs [= 2 + 0.4(2.961 × 10) 6 0.5 1/ 4 ⎞ ⎟ ⎟ ⎠ + 0.06(2.961 × 10) 6 2/3](0.713) 0.4 ⎛ ⎜ 1.825 × 10 −5 ⎜ 1.729 × 10 −5 ⎝ 1/ 4 ⎞ ⎟ ⎟ ⎠ 0.02514 W/m.°C k Nu = (1724) = 10.73 W/m 2 .°C D 4.04 m The rate of heat transfer to the liquid nitrogen is As = πD 2 = π (4.04 m) 2 = 51.28 m 2 T∞ − Ts , tan k T∞ − Ts , tan k Q& = = r 1 Rinsulation + Rconv 2 − r1 + 4πkr1 r2 hAs [20 − (−183)]°C = = 25.8 W (2.02 − 2) m 1 + 4π (0.00005 W/m.°C)(2.02 m)(2 m) (10.73 W/m 2 .°C)(51.28 m 2) and h= The rate of evaporation of liquid oxygen then becomes 0.0258 kJ/s Q& Q& = m& hif → m& = = = 1.21 × 10 -4 kg/s 213 kJ/kg hif 7-97 = 1724 Chapter 7 External Forced Convection 7-103 A circuit board houses 80 closely spaced logic chips on one side. Analysis First we find the Biot number: Bi = . ° C)(012 = = 5.62 k 0.47 W / m. 4 The tank surface is assumed to be at the same temperature as the iced water because of negligible resistance through the steel. Analysis The inner and outer surface areas of the insulated pipe are Ai = πDi L = π (35 . It can also be expressed in terms of the temperatures at the neighboring nodes in the following easy-to-remember form: i +1 i +1 i +1 i +1 i +1 + Ttop + Tright + Tbottom − 4Tnode = Tleft Discussion We note that setting i +1 Tnode i = Tnode i +1 i − Tnode Tnode τ gives the steady finite difference formulation. 3-43C Heat transfer through the voids at an interface is by conduction and radiation. Therefore, the Reynolds number is defined on the basis of maximum velocity. Noting that u = u(y), v = 0, and ∂P / ∂x = 0 (flow is maintained by the motion of the upper plate rather than the pressure gradient), the xmomentum equation reduces to ⎛ ∂u ∂ 2u ∂u ⎞ ∂ 2 u ∂P x-momentum: ρ⎜ u → =0 + v ⎟ = μ 2 − ∂y ⎠ ∂x dy 2 ∂y ⎝ ∂x 12 m/s 5 cm 15 cm This is a second-order ordinary differential equation, and integrating it twice gives u (y) = C1 y + C 2 The fluid velocities at the plate surfaces must be equal to the velocities of the plates because of the no-slip condition. The equilibrium temperature of the top surface is to be determined. Assuming steady one-dimensional conduction in the radial direction, the convection boundary condition on the outer surface of the pipe can be expressed as −k dT (r2) = h[T (r2) − T∞] dr 2-13 h T∞ r1 r2 Chapter 2 Heat Conduction Equation 2-43 A spherical shell of inner radius r1 , outer radius r2 , and thermal conductivity k is considered. Analysis (a) We take the hydrogen in the tank as our system. = 01342 °C / W Rtotal = Rboard + Repoxy + Rcopper + Rconv = 0.00694 + 0.0051 + 0.00024 + 01342 . 2-33 Chapter 2 Heat Conduction Equation 2-69 "GIVEN" L=6 "[m]" r_1=0.037 "[m]" r_2=0.04 "[m]" k=14 "[W/m-C]" Q_dot=300 "[W]" T_infinity=-10 "[C]" h=30 "[W/m^2-C]" f_loss=0.15 "ANALYSIS" q_dot_s=((1-f_loss)*Q_dot)/A A=2*pi*r_2*L T=T_infinity+(ln(r/r_1)+k/(h*r_1))*(q_dot_s*r_2)/k "Variation of temperature" "r is the parameter to be varied" r [m] 0.037 0.03733 0.03767 0.038 0.03833 0.03867 0.039 0.03933 0.03967 0.04 T [C] 3.906 3.902 3.898 3.893 3.889 3.885 3.881 3.877 3.873 3.869 -3.87 T [C] -3.879 -3.888 -3.897 -3.906 0.037 0.0375 0.038 0.0385 0.039 0.0395 0.04 r [m] 2-70 A spherical container is subjected to uniform heat flux on the outer surface and specified temperature on the inner surface. Properties The U-value of a wall is given to be U = 2.25 W/m2.°C. The complete transient explicit finite difference formulation of this problem is to be obtained. The temperature of the body is specified at a point that is at the center of the plane wall but at the surface of the cylinder. Δx ε Assumptions 1 Heat transfer through the wall is given • • 0• to be steady and one-dimensional, and the thermal 2 1 conductivity and heat generation to be variable. 6-36C For steady, laminar, two-dimensional, incompressible flow with constant properties and a Prandtl number of unity and a given geometry, yes, it is correct to say that both the average friction and heat transfer coefficients depend on the Reynolds number only since C f = f 4 (Re L) and Nu = g 3 (Re L , Pr) from non-dimensionalized momentum and energy equations. Properties The properties of the wall are given to be k=1.2 W/m·°C, ε = 0.80, and αs = 0.45. Then, Overall U-factor, U = Σfarea,iUi = 0.82×1.105+0.18×0.805 1.051 W/m2.°C Overall unit thermal resistance, R = 1/U 0.951 m2.°C/W 3-143 The winter R-value and the U-factor of a masonry cavity wall are to be determined. 4 The inner surface temperature of the ice chest can be taken to be 0°C at all times. 5 The heat transfer coefficient is constant and uniform over the entire fin surface. 7-78C Yes, hair or any other cover reduces heat loss from the head, and thus serves as insulation for the head. Dividing both sides by k and integrating twice give Energy: 0=k ∂ 2T 2 2 dT μ⎛V⎞ = − ⎜ ⎟ y + C3 dy k⎝L⎠ 2 T (y) = − μ ⎛ y ⎞ ⎛ V⎞ + C3 y + C 4 2k ⎝ L ⎠ Applying the two boundary conditions give dT −k =0 → C 3 = 0 B.C. 1: y=0 dy y =0 B.C. 2: y=L → C 4 = T0 + T (L) = T0 μV 2 2k Substituting the constants give the temperature distribution to be T (y) = T0 + μV 2 2k ⎛ y2 ⎞ ⎜1 − ⎟ ⎜ L2 ⎟ ⎠ ⎝ The temperature gradient is determined by differentiating T(y) with respect to y, dT − μV 2 = y dy kL2 6-18 Chapter 6 Fundamentals of Convection The location of maximum temperature is determined by setting dT/dy = 0 and solving for y, dT − μV 2 = y=0 → y = 0 dy kL2 Therefore, maximum temperature will occur at the lower plate surface, and it s value is Tmax = T (0) = T0 + μV 2 2k .The heat flux at the upper plate is q& L = −k dT dy =k y=L μV 2 kL2 L= μV 2 L 6-19 Chapter 6 Fundamentals of Convection 6-44 The flow of fluid between two large parallel plates is considered. The density of the cylinder is ρ and the specific heat is C. Properties The thermal conductivity of the silicon chip is given to be k = 130 W/m·°C. The cooking time of the egg is to be determined for a location at 1610-m elevation. 2-7 Chapter 2 Heat Conduction

Equation 2-23 We consider a thin spherical shell element of thickness Δr in a sphere (see Fig. We can improve the result obtained by reevaluating the radiation resistance and repeating the calculations. Analysis The one-dimensional transient temperature distribution in the wall for that time period can be determined from ⎛ x T (x, t) − Ti = erfc⎜⎜ Ts − Ti ⎝ 2 αt ⎞ ⎟ ⎟ ⎠ Ice chest Hot water 60°C But, T (x, t) − Ti 0.1 − 0 = = 0.00167 → 0.00167 = erfc(2.225) 60 − 0 Ts − Ti (Table 4-3) Ice, 0°C Therefore, x 2 αt = 2.225 → t = x2 4 × (2.225) 2 α = (0.05 m) 2 4(2.225) 2 (1.7 × 10 −5 m 2 /s) = 7.4 s The rate of heat transfer to the ice when steady operation conditions are reached can be determined by applying the thermal resistance network concept as 1 1 = = 0.00167°C/W 2 hi A (250 W/m .°C)(1.2 × 2 m 2) 0.05 m L = = = 0.00040°C/W kA (52 W/m.°C)(1.2 × 2 m 2) Rconv ,i = R wall Rconv ,0 Rtotal Rconv 1 1 T1 = = ≅ 0°C/W ho A (∞)(1.2 × 2 m 2) = Rconv ,1 + R wall + Rconv , 2 = 0.00167 + 0.00040 + 0 = 0.00207°C/W T −T (60 − 0)°C Q& = 2 1 = = 28,990 W Rtotal 0.00207 o C/W 4-63 Rwall Rconv T2 Chapter 4 Transient Heat Conduction Transient Heat Conduction in Multidimensional Systems 4-69C The product solution enables us to determine the dimensionless temperature of two- or three-dimensional heat transfer problems as the product of dimensionless temperatures of one-dimensional heat transfer problems. Assumptions 1 Heat conduction in the block is three-dimensional, and thus the temperature varies in all three directions. ° C)(0.0015 m)] = 0.00117 W/ ° C k eff = (kt) copper + (kt) epoxy t copper + t epoxy = Copper (0.3088 + 0.00117)W/ ° C = 58.48 W / m. 4 Heat transfer by radiation is disregarded. 2 Heat conduction in the potato is one-dimensional in the radial direction because of the symmetry about the midpoint. 2 The thermal properties of chickens are constant. Then the energy balance for this steady-flow system can be expressed in the rate form as ΔE& system Ê0 (steady) = = 0 → E& in = E& out E& − E& 1in424out 3 144 42444 3 Rate of net energy transfer by heat, work, and mass Rate of change in internal, kinetic, potential, etc. The time it will take to reduce the temperature of turkey breast at a depth of 3.8 cm to -18°C and the amount of heat transfer per turkey are to be determined. 4 The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified). 5-31E A large plate lying on the ground is subjected to convection from its exposed surface. energies & 1 = Q& out + mh & 2 (since Δke ≅ Δpe ≅ 0) mh & p (T1 − T2) Q& out = mC Substituting, Q& out = m& C p ΔT = (120kg/min)(1.007 kJ/kg · °C)(3°C) = 363 kJ/min 1-20 120 kg/min AIR · Q Chapter 1 Basics of Heat Transfer 1-42E Air gains heat as it flows through the duct of an air-conditioning system. This is a closed system since it involves a fixed amount of mass (no mass transfer). Analysis Using the energy balance approach and taking the direction of all heat transfers to be towards the node under consideration, the explicit transient finite difference formulations become Left boundary node: 4 εσA[Tsurr − (T0i) 4] + hA(T∞i − T0i) + kA T3i − T4i Δx Δx T4i +1 − T4i i i & + = ρ + g A A C Q& rightt kA surface 4 Δx 2 2 Δt Heat transfer at right surface: Noting that Q = Q& Δt = i +1 i T1i − T0i Δx Δx T 0 − T 0 + g& 0i A = ρA C Δx 2 2 Δt ∑ Q& Δt , the total amount of heat i Tsurr i 20 Q right surface = ∑ TL g(x, t) transfer becomes q0 i Q& right surface Δt Δx i =1 i +1 i ⎛ T4i − T3i Δx Δx T4 − T4 ⎞ kA − g& 4i A + ρA = C ⎟ 2 2 Δx Δt i =1 ⎝ 20 ∑ 5-69 ⎠ ⎟ Δt ⎟ ⎠ h, T∞ • 0 • 1 • 2 • 3 4 • Chapter 5 Numerical Methods in Heat Conduction 5-78 Starting with an energy balance on a volume element, the two-dimensional transient explicit finite difference equation for a general interior node in rectangular coordinates for T (x , y , t) for the case of constant thermal conductivity and no heat generation is to be obtained. Therefore, we need to Insulated consider only one-fourth of the geometry in the • • • • solution whose nodal network consists of 10 equally 5 6 7 8 spaced nodes. 5-33 The handle of a stainless steel spoon partially immersed in boiling water loses heat by convection and radiation. 4b Construction 6 3 4a 5 1. Nodes 1, 2, 3, 4, and 5 are interior nodes, and thus for them we can use the general explicit finite difference relation expressed as g& mi Δx 2 Tmi+1 − Tmi = → Tmi +1 = τ (Tmi −1 + Tmi +1) + (1 − 2τ)Tmi k τ The finite difference equation for boundary nodes 0 and 6 are obtained by applying an energy balance on the half volume elements and taking the direction of all heat transfers to be towards the node under consideration: Tmi−1 − 2Tmi + Tmi+1 + Node 0: hin A(Tini − T0i) + kA or i +1 i T1i − T0i Δx T0 − T0 = ρA C Δx 2 Δt h Δx ⎞ h Δx ⎛ T0i +1 = ⎜⎜1 − 2τ − 2τ in ⎟⎟T0i + 2τT1i + 2τ in Tin k ⎠ k ⎝ 5-78 Chapter 5 Numerical Methods in Heat Conduction T1i +1 = τ (T0i + T2i) + (1 − 2τ)T1i Node 1 (m = 1) : Node 2 (m = 2) : T2i +1 = τ (T1i + T3i) + (1 − 2τ)T2i Node 3 (m = 3) : T3i +1 = τ (T2i + T4i) + (1 − 2τ)T3i Node 4 (m = 4) : T4i +1 = τ (T3i + T5i) + (1 − 2τ)T4i Node 5 (m = 5) : T5i +1 = τ (T4i + T6i) + (1 − 2τ)T5i i i hout A(Tout − T6i) + κAq& solar + kA Node 6 or i +1 i T5i − T6i Δx T6 − T6 = ρA C Δx 2 Δt κq& i Δx h Δx ⎞ h Δx i ⎛ + 2τ solar T6i +1 = ⎜⎜1 − 2τ − 2τ out ⎟⎟T6i + 2τT5i + 2τ out Tout k ⎠ k k ⎝ where L = 0.30 m, k = 0.70 W/m.°C, α = 0.44 × 10 −6 m2 / s , Tout and q& solar are as given in the table, κ = 0.76 hout = 3.4 W/m2.°C, Tin = 20°C, hin = 9.1 W/m2.°C, and Δx = 0.05 m. Analysis (a) The total mass of water is m w = ρV = (1 kg/L)(50 × 20 L) = 1000 kg 50,000 kJ/h Taking the contents of the house, including the water as our system, the energy balance relation can be written as E −E 1in424out 3 Net energy transfer by heat, work, and mass = ΔE system 1 424 3 22°C Change in internal, kinetic, potential, etc. Properties The thermal conductivity of chimney is given to be k = 1.4 W/m·°C. 5 When calculating the conduction thermal resistance of aluminum, the average of inner and outer surface areas will be used. Then the temperature at the center of the box if the box contains margarine becomes 2 2 T (0, t) − T∞ = A1 e − λ1 τ = (1.2431)e − (1.3269) (0.9504) θ(0, t) wall = Ti − T∞ The Fourier number is = Margarine, Ti = 30°C T (0, t) − 0 = 0.233 → T (0, t) = 7.0 °C 30 − 0 (b) Repeating the calculations for white cake, hL (25 W/m 2 .°C)(0.05 m) Bi = = = 15.24 → λ1 = 1.4641 and A1 = 1.2661 k (0.082 W/m.°C) τ= αt L2 = (0.10 × 10 −6 m 2 /s)(6 h × 3600 s/h) (0.05 m) 2 θ(0, t) wall = = 0.864 > 0.2 2 2 T (0, t) − T∞ = A1 e − λ1 τ = (1.2661)e − (1.4641) (0.864) Ti − T∞ T (0, t) − 0 = 0.199 → T (0, t) = 6.0 °C 30 − 0 (c) Repeating the calculations for chocolate cake, hL (25 W/m 2 .°C)(0.05 m) Bi = = = 11.79 → λ1 = 1.4356 and A1 = 1.2634 k (0.106 W/m.°C) τ= αt L2 = (0.12 × 10 −6 m 2 /s)(6 h × 3600 s/h) θ(0, t) wall = (0.05 m) 2 = 1.0368 > 0.2 2 2 T (0, t) − T∞ = A1 e − λ1 τ = (1.2634)e − (1.4356) (1.0368) Ti − T∞ T (0, t) − 0 = 0.149 → T (0, t) = 4.5 °C 30 − 0 4-114 Chapter 4 Transient Heat Conduction 4-121 A cold cylindrical concrete column is exposed to warm ambient air during the day. Assumptions 1 The temperature in the wall is affected by the thermal conditions at outer surfaces only, and thus the wall can be considered to be a semi-infinite medium with a specified outer surface temperature of 18°C. Analysis (a) It is given that D = 0.021 m, SL = ST = 0.08 m, and V = 4.5 m/s. Properties The density and specific heat of water are given to be 62 lbm/ft3 and 1.0 Btu/lbm.°F. This assumption cannot be justified in this case since the walls are very thick and the corner sections constitute a considerable portion of the chimney structure. Properties The thermal conductivity is given to be k = 386 W/m·°C. If the changes are not significant, we conclude that the round-off error is not a problem. 3 The thermal properties of the watermelon are motion is caused by natural means such as the buoyancy effect that manifests itself as the rise of the warmer fluid and the fall of the cooler fluid. The maximum length of the duct for a specified temperature increase in the duct is to be determined. 2 Convection heat transfer coefficient is constant over the entire surface. Qrad 1-58 Chapter 1 Basics of Heat Transfer 1-107 A spherical tank located outdoors is used to store iced water at 0°C. Analysis This short cylinder can physically be formed by the intersection of a long cylinder of radius D/2 = 4 cm and a plane wall of thickness 2L = 15 cm. Again assuming the temperatures between the adjacent nodes to vary linearly, the energy balance relation above becomes k (Δy × Δz) Tm −1, n, r − Tm, n, r + k (Δy × Δz) + k (Δx × Δy) Δx + k (Δx × Δz) Tm +1, n, r − Tm, n, r Δx Tm, n, r −1 − Tm, n, r Tm, n +1, r − Tm, n, r + k (Δx × Δz) Δy Tm, n −1, r − Tm, n, r Δy Tm, n, r +1 − Tm, n, r + k (Δx × Δy) + g& 0 (Δx × Δy × Δz) = 0 Δz Δz Dividing each term by k Δx × Δy × Δz and simplifying gives Tm −1, n , r − 2Tm, n, r + Tm +1, n, r Δx 2 + Tm, n −1, r − 2Tm, n, r + Tm, n +1, r Δy 2 + Tm, n, r −1 − 2Tm, n, r + Tm, n , r +1 Δz 2 For a cubic mesh with Δx = Δy = Δz = l, and the relation above simplifies to Tm −1, n, r + Tm +1, n, r + Tm, n −1, r + Tm,n −1, r + Tm, n, r −1 + Tm,n ,r +1 − 6Tm,n ,r + It can also be expressed in the following easy-to-remember form: Tleft + Ttop + Tright + Tbottom + Tfront + Tback − 6Tnode + 5-105 g& 0 l 2 =0 k g& 0 l 2 =0 k + g& 0 =0 k Chapter 5 Numerical Methods in Heat Conduction 5-107 Starting with an energy balance on a volume element, the three-dimensional transient explicit finite difference equation for a general interior node in rectangular coordinates for T(x, y, z, t) for the case of constant thermal conductivity k and no heat generation is to be obtained. Properties The R-values at the outer surface of a wall for summer (7.5 mph winds) and winter (15 mph winds) conditions are given in Table 3-6 to be Ro, 7.5 mph = Ro, summer = 0.25 h.ft2.°F/Btu Inside WALL 3-109 Outside 7.5 mph Chapter 3 Steady Heat Conduction and Ro, 15 mph = Ro, winter = 0.17 h.ft2.°F/Btu Analysis The R-value of the wall at 7.5 mph winds (summer) is Rwall, 7.5 mph = 1 / U wall, 7.5 mph = 1 / 0.09 = 1111 . 2 Thermal conductivity and emissivity are constant. Then the energy balance for this process can be written as E −E 1in424out 3 = Net energy transfer by heat, work, and mass [mC (32°F − T) ΔE system 1 424 3 → 0 = ΔU → (ΔU)ice + (ΔU)water = 0 Change in internal, kinetic, potential, etc. Then the rate of heat transfer is determined to be Rcontact ΔT (85 − 15)° C Q& = = = 20.8 W Rtotal 3.359 ° C / W Tcase Rplate Rconv T∞ Therefore, the power transistor should not be operated at power levels greater than 20.8 W if the case temperature is not to exceed 85°C. The distance from the leading edge of the plate where the flow becomes turbulent is the distance xcr where the Reynolds number becomes equal to the critical Reynolds number, Re cr = ρV∞ x cr μ → x cr = μ Re cr (0.891 × 10 −3 kg/m ⋅ s)(5 × 10 5) = = 0.056 m = 5.6 cm ρV∞ (997 kg/m 3)(8 m/s) The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation, δ cr = 5x Re 1x/ 2 → δ cr = 5 x cr Re 1cr/ 2 = 5(0.056 m) (5 × 10 5) 1 / 2 = 0.00040 m = 0.4 mm Therefore, the flow becomes turbulent after about 5 cm from the leading edge of the plate, and the thickness of the boundary layer at that location is 0.4 mm. J / s)(14 × 3600 s) = 8357 . ° C L ΔT (15 − 7)° C 15° 1-30 7°C L=3 Chapter 1 Basics of Heat Transfer 1-71 The rate of radiation heat transfer between a person and the surrounding surfaces at specified temperatur es is to be determined in summer and in winter. The heat of fusion of ice at atmospheric pressure is 143.5 Btu/lbm and the specific heat of ice is 0.5 Btu/lbm.°F. 2-119C The general solution of a 3rd order linear and homogeneous differential equation will involve 3 arbitrary constants. 4 The local atmospheric pressure is 1 atm. The time it will take for the average temperature of the drink to rise to 10°C with and without rubber insulation is to be determined. This would be a transient heat transfer process since the temperature at any point within the drink will change with time during heating. 3-12C Yes, it is. Properties The conductivity and diffusivity are given to be k = . o C / W . The cooling time and if any part of the oranges will freeze during this cooling process are to be determined. 3-23 Chapter 3 Steady Heat Conduction 3-46 Six identical power transistors are attached on a copper plate. At the minimum, the model should reflect the essential features of the physical problem it represents. Assumptions 1 Heat transfer along the fin is given to be steady, and the temperature along the fin to vary in the x direction only so that T = T(x). Assumptions 1 Heat conduction in the cylindrical block is two-dimensional, and thus the temperature varies in both axial x- and radial r- directions. Discussion We could also solve this problem using transient temperature charts, but the data in this case falls at a point on the chart which is very difficult to read: ⎤ (0.13 × 10 −6 m²/s)(2.5 × 3600 s) = 0.206⎥ 2 (0.0753 m)² r0 ⎥ To − T∞ = 0.15....0.30 ?? 4 ⎞ 2 / 3 ⎤⎥1 + ⎢ ⎢ ⎥⎣⎢ ⎝ Pr ⎠ ⎦⎥⎡ ⎛ Re ⎞⎤1 + ⎢⎢ ⎥⎥⎣ ⎝ 282,000 ⎠ 5/8 ⎤ Ts = 86°F 4/5 ⎥ ⎥⎦ 5/8 ⎤⎡ 0.62(4.463 × 10 4) 0.5 (0.7306)1 / 3⎥ ⎛⎜ 4.463 × 10 4 ⎞⎟ ⎥ = 0. The heat flux on the surface of the plate is to be determined. 4-14b we have T − T∞ 93 − 100 ⎤ = = 0.28⎥ To − T∞ 75 − 100 k ⎥ 1 = = 0.25 ⎬ Bi hr x ro o ⎥ = =1 ⎥⎦ ro ro From Fig. Then the continuity equation reduces to ∂u ∂v ∂u Continuity: + = 0 → = 0 → u = u(y) 4500 rpm ∂x ∂x ∂y Therefore, the x-component of velocity does not change in the flow direction (i.e., the velocity profile remains unchanged). 4 − 14b) 4-50 (Fig.4 − 14a) Chapter 4 Transient Heat Conduction . Also, Cp = 0.2404 Btu/lbm·R for air at room temperature (Table A-15E). Then the annual fuel cost of this furnace before insulation becomes Annual Cost = Q in × Unit cost = (18,526 therm / yr)($0.50 / therm) = $9,263 / yr We expect the surface temperature of the furnace to increase, and the heat transfer coefficient to decrease somewhat when insulation is installed. k (2.5 W/m.°C) To determine the center temperature, the product solution method can be written as [θ (0,0, t) block = [θ (0, t) wall]θ (0, t) cyl] 2 T (0,0, t) − T∞ ⎛ = ⎜ A1e −λ1 τ ⎞⎟ ⎛⎜ A1 e − λ1 τ ⎞⎟ ⎝ ⎠ wall ⎝ ⎠ cyl Ti − T∞ 2 { }{ } { }{ } { }{ } 2 2 T (0,0, t) − 500 = (1.1016)e −(0.7910) (1.104) (1.0931)e −(0.8516) (1.104) = 0.271 20 − 500 T (0,0, t) = 370°C After 20 minutes 2 2 T (0,0, t) − 500 = (1.1016)e − (0.7910) (2.208) (1.0931)e − (0.8516) (2.208) = 0.06094 → T (0,0, t) = 471°C 20 − 500 After 60 minutes 2 2 T (0,0, t) − 500 = (1.1016)e − (0.7910) (6.624) (1.0931)e − (0.8516) (6.624) = 0.0001568 → T (0,0, t) = 500°C 20 − 500 Note that τ > 0.2 in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable. 3 Thermal conductivity of the concrete is constant. 4-32C When the Biot number is less than 0.1, the temperature of the sphere will be nearly uniform at all times. It is to be determined if the plastic insulation on the ball will increase or decrease heat transfer from it. The finite difference formulation of the problem is to be obtained, and the unknown nodal temperatures are to be determined. Assumptions 1 The thermal properties of the balls are constant. By simplifying and solving the continuity, momentum, and energy equations it is found in Example 6-1 that dT W& mech = Q& 0 = −Q& L = −kA dy = −kA y =0 5 cm 10 cm 2 2 μV 2 (1 − 0) = − A μV = − A μV 2kL 2L 2L First, the velocity and the surface area are ⎛ 1 min ⎞ V = πDN& = π(0.05 m)(2500 rev/min)⎜ ⎟ = 6.545 m/s ⎝ 60 s ⎠ A = πDL bearing = π(0.05 m)(0.10 m) = 0.01571 m 2 (a) Air: (1.918 ×10 −5 N ⋅ s/m 2)(6.545 m/s) 2 μV 2 = −(0.01571 m 2) W&

T_infinity)=theta_o_w*theta_o_c "center temperature of short cylinder" "(b)" theta_L_w=A_1_w*exp(-lambda_1_w^2*tau_w)*Cos(lambda_1_w*L/L) "theta_L_w=(T_L_w-T_infinity)/(T_i-T_infinity)" (T_L_o-T_infinity)/(T_i-T_infinity)=theta_L_w*theta_o_c "center temperature of the top surface" "(c)" V=pi*r_o^2*(2*L) 4-67 Chapter 4 Transient Heat Conduction m=rho*V Q_max=m*C_p*(T_i-T_infinity) Q_w=1-theta_o_w*Sin(lambda_1_w)/lambda_1_w "Q_w=(Q/Q_max)_w" Q_c=1-2*theta_o_c*J_1/lambda_1_c "Q_c=(Q/Q_max)_c" J_1=0.0846 "From Table 4-2, at lambda_1_c" Q/Q_max=Q_w+Q_c*(1-Q_w) "total heat transfer" 4-68 Chapter 4 Transient Heat Conduction To,o [C] 119.3 95.18 76.89 63.05 52.58 44.66 38.66 34.12 30.69 28.09 26.12 24.63 time [min] 5 10 15 20 25 30 35 40 45 50 55 60 TL,o [C] 116.8 93.23 75.42 61.94 51.74 44.02 38.18 33.75 30.41 27.88 25.96 24.51 Q [kJ] 80.58 140.1 185.1 219.2 245 264.5 279.3 290.5 298.9 305.3 310.2 313.8 120 350 300 100 heat 250 200 60 150 tem perature 40 20 0 100 10 20 30 40 tim e [m in] 4-69 50 50 60 Q [kJ] T o,o [C] 80 Chapter 4 Transient Heat Conduction 120 100 T L,o [C] 80 60 40 20 0 10 20 30 40 tim e [m in] 4-70 50 60 Chapter 4 Transient Heat Conduction 4-75 A semi-infinite aluminum cylinder is cooled by water. L =1.5 Analysis The one-dimensional transient temperature distribution in the wall for that time period can be Q& determined from ⎛ x ⎞ T (x, t) − Ti ⎟ = erfc⎜⎜ ⎟ Ts − Ti ⎝ 2 αt ⎠ 1800° But, T (x, t) − Ti 70.1 − 70 = = 0.00006 → 0.00006 = erfc(2.85) 1800 − 70 Ts − Ti (Table 4-3) Therefore, x 2 αt = 2.85 → t = x2 4 × (2.85) 2 α = (1.5 ft) 2 4 × (2.85) 2 (0.023 ft 2 /h) 4-61 = 3.01 h = 181 min 70°F Chapter 4 Transient Heat Conduction 4-67 A thick wood slab is exposed to hot gases for a period of 5 minutes. The initial value of the temperature gradient at the potato surface is Potato Ti = 20°C ⎛ ∂T ⎞ q& conv = q& cond = − k ⎟ = h(Ts − T∞) ⎟ ⎝ ∂r ⎠ r = R ∂T ∂r =− r =R h(Ts − T∞) (19.1 W/m 2 .°C)(20 − 5)°C =− = −585 °C/m k (0.49 W/m.°C) 6-9 The rate of heat loss from an average man walking in still air is to be determined at different walking velocities. It truly reflects the exponential decay of the local temperature difference. Vapor Analysis The rate of heat transfer to the nitrogen tank is As = πD 2 = π (4 m) 2 = 50.27 m 2 Q& = hAs (T s − Tair) = (25 W/m 2 .°C)(50.27 m 2)[20 − (−196)]°C = 271,430 W Air 20° 1 atm Liquid N2 Then the rate of evaporation of liquid nitrogen in the tank is determined Q& to be Q& 271.430 kJ/s → m& = = = 1.37 kg/s Q& = m& h fg 198 kJ/kg h fg 1-46 -196°C Chapter 1 Basics of Heat Transfer 1-89 A 4-m diameter spherical tank filled with liquid oxygen at 1 atm and -183°C is exposed to convection with ambient air. Then the maximum velocity and the Reynolds number based on the maximum velocity become Vmax ST 0.015 = V= (4 m/s) = 8.571 m/s 0.015 − 0.008 ST − D Re D = V=4 m/s Ti=0°C ρVmax D (1.316 kg/m 3)(8.571 m/s)(0.008 m) = = 5294 μ 1.705 × 10 −5 kg/m ⋅ s Ts=-20°C SL ST The average Nusselt number is determined using the proper relation from Table 7-2 to be Nu D = 0.27 Re 0D.63 Pr 0.36 (Pr/ Prs) 0.25 = 0.27(5294) 0.63 (0.7375) 0.36 (0.7375 / 0.7408) 0.25 = 53.61 D Since NL > 16. For convenience, let us choose the time step to be Δt = 900 s = 15 min. Analysis Noting that heat transfer is steady and one-dimensional in the radial r direction, the mathematical formulation of this problem can be expressed as 1 d ⎛ dT ⎞ g& ⎟ r ⎟+ = 0 r dr ⎝ dr ⎠ k dT (r0) = h[T (r0) − T∞] (convection at the outer surface) dr dT (0) = 0 (thermal symmetry about the centerline) dr Multiplying both sides of the differential equation by r and rearranging gives and − k g& d ⎛ dT ⎞ ⎟ r ⎟=− r dr ⎝ dr ⎠ k r Water ro T∞ h Integrating with respect to r gives 0 dT g& r 2 r (a) =− + C1 dr k 2 Heater It is convenient at this point to apply the second boundary condition since it is related to the first derivative of the temperature by replacing all occurrences of r and dT/dr in the equation above by zero. Properties The properties of aluminum are given to be k = 236 W/m.°C, ρ = 2702 kg/m3, Cp = 0.896 kJ/kg.°C, and α = 9.75×10-5 m2/s. 5-83 A pin fin with negligible heat transfer from its tip is considered. 4-28C The solution for determination of the one-dimensional transient temperature distribution involves many variables that make the graphical representation of the results impractical. Properties The thermal conductivity of the glass is given to be kglass = 0.45 Btu/h.ft·°F. Air space, nonreflecting, 20 mm 5. We assume the Nusselt number is proportional to the nth power of the Reynolds number with 0.33 < n < 0.805. Nominal 1 × 3 vertical ferring 6. Analysis The area of the window and the individual resistances are Glass A = (1.2 m) × (2 m) = 2.4 m 2 1 1 = = 0.04167 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.006 m L = = 0.00321 °C/W Rglass = k1 A (0.78 W/m.°C)(2.4 m 2) 1 1 = = 0.01667 °C/W Ro = R conv,2 = 2 h2 A (25 W/m 2 .°C)(2.4 m 2) Rtotal = R conv,1 + R glass + Rconv,2 L Ri = R conv,1 = = 0.04167 + 0.00321 + 0.01667 = 0.06155 °C/W The steady rate of heat transfer through window glass is then T −T [24 − (−5)]° C = 471 W Q& = ∞1 ∞ 2 = Rtotal 0.06155 ° C / W Q& T1 Ri Rglass T∞1 The inner surface temperature of the window glass can be determined from T −T & Q& = ∞1 1 → T1 = T∞1 − QR conv ,1 = 24° C − (471 W)(0.04167 ° C / W) = 4.4° C Rconv ,1 3-3 Ro T∞2 Chapter 3 Steady Heat Conduction 3-19 A double-pane window consists of two 3-mm thick layers of glass separated by a 12-mm wide stagnant air space. 3 The thermal properties of the chickens are constant. The rate of heat transfer from the top surface is Δt = Q& top ,ave = ho Atop (Tair − Tcan ,ave) = (10 W / m 2 . 2 There is no heat generation in the plate and the soil. 3-16C The blanket will introduce additional resistance to heat transfer and slow down the heat gain of the drink wrapped in a blanket. 5-4 Chapter 5 Numerical Methods in Heat Conduction 5-13C In the finite difference formulation of a problem, an insulated boundary is best handled by replacing the insulation by a mirror, and treating the node on the boundary as an interior node. 4-107 The water pipes are buried in the ground to prevent freezing. Column 1 contains the time, column 2 the value of T[1], column 3, the value of T[2], etc., and column 7 the Row." Time=TableValue(Row-1,#Time)+DELTAt Duplicate i=1,5 T_old[i]=TableValue(Row-1,#T[i]) end "Using the explicit finite difference approach, the six equations for the six unknown temperatures are determined to be" T[1]=tau*(T_old[2]+T_old[2])+(1-2*tau)*T_old[1]+tau*(g_dot*DELTAx^2)/k "Node 1, insulated" T[2]=tau*(T_old[1]+T_old[3])+(1-2*tau)*T_old[2]+tau*(g_dot*DELTAx^2)/k "Node 2" T[3]=tau*(T_old[2]+T_old[4])+(1-2*tau)*T_old[3]+tau*(g_dot*DELTAx^2)/k "Node 3" T[4]=tau*(T_old[3]+T_old[5])+(1-2*tau)*T_old[4]+tau*(g_dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau2*tau*(h*DELTAx)/k)*T_old[5]+2*tau*T_old[4]+2*tau*(h*DELTAx)/k*T_infinity +tau*(g_dot*DELTAx^2)/k "Node 4, convection" 5-76 Chapter 5 Numerical Methods in Heat Conduction T1 [C] 100 106.7 113.4 120.1 126.8 133.3 139.9 146.4 152.9 159.3 ... ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.2 139.6 146.2 152.6 159.1 ... ... 1213 1216 1220 1223 1227 1230 1233 1237 1240 1243 T3 [C] 100 106.7 113.4 119.7 126.3 132.6 139.1 145.4 151.8 158.1 ... ... 1203 1206 1209 1213 1216 1219 1223 1226 1229 1233 T4 [C] 100 106.7 112.5 119 125.1 131.5 137.6 144 150.2 156.5 ... ... 1185 1188 1192 1195 1198 1201 1205 1208 1211 1214 T5 [C] 100 104.8 111.3 117 123.3 129.2 135.5 141.5 147.7 153.7 ... ... 1160 1163 1167 1170 1173 1176 1179 1183 1186 1189 1200 1200 T right 1000 1000 T left 800 T left [C] Row 1 2 3 4 5 6 7 8 9 10 ... ... 232 233 234 235 236 237 238 239 240 241 800 600 600 400 400 200 200 0 0 500 1000 1500 2000 2500 Tim e [s] 5-77 3000 3500 0 4000 T right [C] Time [s] 0 15 30 45 60 75 90 105 120 135 ... ... 3465 3480 3495 3510 3525 3540 3555 3570 3585 3600 Chapter 5 Numerical Methods in Heat Conduction 5-86 The passive solar heating of a house through a Trombe wall is studied. 4 Heat generation in the heater is uniform. 1-67 Chapter 1 Basics of Heat Transfer 1-123 "GIVEN" L=4 "[m]" D=0.2 "[m]" P_air_in=100 "[kPa]" T_air_in=65 "[C]" "Vel=3 [m/s], parameter to be varied" T_air_out=60 "[C]" eta_furnace=0.82 Cost_gas=0.58 "[$/therm]" "PROPERTIES" R=0.287 "[kJ/kg-K], gas constant of air" C_p=(Fluid$, T=25) "at room temperature" "ANALYSIS" rho=P_air_in/(R*(T_air_in+273)) A_c=pi*D^2/4 m_dot=rho*A_c*Vel Q_dot_loss=m_dot*C_p*(T_air_in-T_air_out)*Convert(kJ/s, kJ/h) Cost_HeatLoss=Q_dot_loss/eta_furnace*Cost_gas*Convert(kJ, therm)*Convert($, cents) Vel [m/s] CostHeatLoss [Cents/h] 0.3934 0.7868 1.18 1.574 1.967 2.361 2.754 3.147 3.541 3.934 1 2 3 4 5 6 7 8 9 10 4 3.5 Cost HeatLoss [cents/h] 3 2.5 2 1.5 1 0.5 0 1 2 3 4 5 6 Vel [m /s] 1-68 7 8 9 10 Chapter 1 Basics of Heat Transfer 1-124 Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. Properties The thermal conductivity is given to be k = 1.5 W/m·°C. Therefore, the rate of heat D = 0.08 in generation in a resistance wire is simply equal to the power rating of a resistance heater. Then the R-value of insulation of the wall becomes equivalent to its thermal resistance, which is determined from. 2 The thermal properties of the geometries are constant. 2 Heat transfer by convection is not considered. Analysis (a) The characteristic length of the wire and Air the Biot number are 30°C πr 2 L ro 0.0015 m V 350°C Lc = = o = = = 0.00075 m 10 m/min As 2πro L 2 2 Bi = hLc (35 W/m 2 .°C)(0.00075 m) = = 0.00011 < 0.1 k 236 W/m.°C Aluminum wire Since Bi < 0.1, the lumped system analysis is applicable. 2 Water is an incompressible substance with constant specific heats. Analysis The nodal spacing is given to be Δx=Δx=l=0.02 m, and the general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat generation is expressed as Tleft + Ttop + Tright + Tbottom − 4Tnode + g& node l 2 = 0 → Tnode = (Tleft + Ttop + Tright + Tbottom) / 4 k (a) There is symmetry about the insulated surfaces as well as about the diagonal line. 4-82 Chapter 4 Transient Heat Conduction 4-82 A cubic block and a cylindrical block are exposed to hot gases on all of their surfaces. However, 3°C 75°F the lumped system analysis is still applicable since the milk is stirred constantly, so that its temperature remains uniform at all times. ° F) λ 1 = 2.1589 and A1 = 15618 . Assumptions 1 Heat transfer through the ceiling is one-dimensional. Properties The thermal conductivity of copper tube is given to be k = 223 Btu/h·ft·°F. 4-27C Yes. It gives R = 1.085 m2.°C/W and U = 0.922 W/ m2.°C for the air space. kg / s)(4180 J / kg. Analysis After 5 minutes Plate: First the Biot number is calculated to be Bi = hL (7 Btu/h.ft 2 .°F)(0.5 / 12 ft) = = 0.01006 k (29 Btu/h.ft.°F) 2 ro The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, 2 ro λ1 = 0.0998 and A1 = 1.0017 The Fourier number is τ= αt 2 = (0.61 ft 2 /h)(5 min/60 min/h) (0.5 / 12 ft) 2 L = 29.28 > 0.2 Then the center temperature of the plate becomes θ 0, wall = 2L T − 75 T0 − T ∞ = A1 e −λ1 τ → 0 = (1.0017)e −(0.0998) Ti − T∞ 400 − 75 2 2 (29.28) = 0.748 → T0 = 318°F Cylinder: −1 Bi = 0.01 Table 4 → λ1 = 0.1412 and A1 = 1.0025 θ 0,cyl = 2 2 T − 75 T0 − T∞ = A1e − λ1 τ → 0 = (1.0025)e − (0.1412) (29.28) = 0.559 → T0 = 257°F Ti − T∞ 400 − 75 Sphere: −1 Bi = 0.01 Table 4 → λ1 = 0.1730 and A1 = 1.0030 θ 0, sph = 2 2 T − 75 T0 − T∞ = A1 e −λ1 τ → 0 = (1.0030)e −(0.1730) (29.28) = 0.418 → T0 = 211°F Ti − T∞ 400 − 75 After 10 minutes τ= αt L2 = (0.61 ft 2 /h)(10 min/60 min/h) (0.5 / 12 ft) 2 = 58.56 > 0.2 Plate: θ 0, wall = 2 2 T − 75 T0 − T ∞ = A1e −λ1 τ → 0 = (1.0017)e −(0.0998) (58.56) = 0.559 → T0 = 257°F Ti − T∞ 400 − 75 4-108 Chapter 4 Transient Heat Conduction as the solution at time t = 0 (for i = 0), sweeping through the 3 equations above will give the solution at intervals of 1 min. = = 1843 k 0.26 Btu / h.ft. 2 Thermal resistance of the tank is negligible. 4 The outdoor temperature remains constant in the 4-h blocks. All the heat generated in the chips is conducted across the circuit board. 2 The transistor case is isothermal at 80 °C . Analysis (a) From Fig. But the large thermal conductivity of copper will minimize this effect. > 52.4) . ° F)[2π (1.75 / 12 ft)(1 ft)] ln(r2 / r1) ln(2 / 175 . Yunus A. Noting that D = D0 = 4.1 m, the Nusselt number becomes Re = V∞ D [(40 × 1000/3600) m/s](4.1 m) = = 3.005 × 10 6 υ 1.516 × 10 −5 m 2 /s Nu = ⎛μ hD = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 ⎜⎜ ∞ k ⎝ μs [ ] ⎞ 1/ 4 ⎟ ⎟⎟ ⎠ ⎛ 1.825 × 10 −5 = 2 + 0.4(3.005 × 10 6) 0.5 + 0.06(3.005 × 10 6) 2 / 3 (0.7309) 0.4 ⎜⎜ −5 ⎝ 1.189 × 10 0.02514 W/m.°C k (1910) = 11.71 W/m 2 .°C h = Nu = and D 4.1 m The rate of heat transfer to the liquid nitrogen is 7-94 1/ 4 ⎞ ⎟ ⎟ ⎠ = 1910 Chapter 7 External Forced Convection As = πD 2 = π (4.1 m) 2 = 52.81 m 2 T∞ − Ts ,tan k T∞ − Ts ,tan k = Q& = r 1 Rinsulation + Rconv 2 − r1 + 4πkr1 r2 hAs [20 − (−196)]°C = = 1311 W (2.05 − 2) m 1 + 4π (0.035 W/m.°C)(2.05 m)(2 m) (11.71 W/m 2 .°C)(52.81 m 2) The rate of evaporation of liquid nitrogen then becomes Q& 7.361 kJ/s Q& = m& hif → m& = = = 0.0372 kg/s hif 198 kJ/kg (c) We use the dynamic viscosity value at the new estimated surface temperature of 0°C to be μ = 1.729 × 10 −5 kg/m.s . Analysis (a) Noting that the 85% of the 300 W generated by the strip heater is transferred to the pipe, the heat flux through the outer surface is determined to be q& s = Q& s Q& s 0.85 × 300 W = = = 169.1 W/m 2 A2 2πr2 L 2π (0.04 m)(6 m) Noting that heat transfer is one-dimensional in the radial r direction and heat flux is in the negative r direction, the mathematical formulation of this problem can be expressed as r d ⎛ dT ⎞ ⎟=0 ⎜r dr ⎝ dr ⎠ −k and k dT (r1) = h[T∞ − T (r1)] dr Air, -10°C r1 dT (r2) = q& s dr (b) Integrating the differential equation once with respect to r gives r Heater r2 L=6 m dT = C1 dr Dividing both sides of the equation above by r to bring it to a readily integrable form and then integrating, dT C1 = dr r T (r) = C1 ln r + C2 where C1 and C2 are arbitrary constants. Also, we would place the origin somewhere on the center line, possibly at the center of the bottom surface. ° C)(80 − 0)° C = 2748 W S= and the total rate of heat loss from the hot water becomes Q& = 498 + 413 + 2748 = 3659 W total (b) Using the water properties at the room temperature, the temperature drop of the hot water as it flows through this 25-m section of the wall becomes Q& = m& C ΔT p Q& Q& Q& ΔT = = = = m& C p (ρV&)C p (ρVAc)C p 3659 J/s = 0.30°C ⎡ π(0.05 m) 2 ⎤ 3 (1000 kg/m)(1.5 m/s) ⎢ ⎥ (4180 J/kg.°C) 4 ⎦ ⎣ 3-96 Chapter 3 Steady Heat Conduction 3-129 The walls and the roof of the house are made of 20-cm thick concrete, and the inner and outer surfaces of the house are maintained at specified temperatures. 4-9C The lumped system analysis is more likely to be applicable in air than in water since the convection heat transfer coefficient and thus the Biot number is much smaller in air. 3 Thermal properties are constant Properties The thermal conductivity of plastic cover is given to be k = 0.075 Btu/h·ft·°F. 2 The thermal contact resistance at the interface is negligible. 5-16 Chapter 5 Numerical Methods in Heat Conduction 5-28 A plate is subjected to specified heat flux on one side and specified temperature on the other. (b) The average temperature of the inner surface of the roof can be taken to be T1,ave = T1 @ 6 PM + T1 @ 6 AM 2 = 18 + 10.3 = 14.15°C 2 Then the average rate of heat loss through the roof that night becomes 5-96 Chapter 5 Numerical Methods in Heat Conduction [4 − (T1i + 273) 4 Q& ave = hi As (Ti − T1,ave) + εσAs Twall] = (5 W/m 2 · °C)(20 × 20 m 2)(20 - 14.15)°C + 0.9(20 × 20 m 2)(5.67 × 10 -8 W/m 2 · K 4)[(293 K) 4 − (14.15 + 273 K) 4] = 23,360 W 5-97 Chapter 5 Numerical Methods in Heat Conduction 5-94 A refrigerator whose walls are constructed of 3-cm thick urethane insulation malfunctions, and stops running for 6 h. Analysis (a) The heat transfer rate and the rate of evaporation of the liquid without insulation are A = πD 2 = π (3 m) 2 = 28.27 m 2 1 1 Ro = = = 0.00101 °C/W 2 ho A (35 W/m .°C)(28.27 m 2) Ts1 T −T [15 − (−183)]°C Q& = s1 ∞ 2 = = 196,040 W Ro 0.00101 °C/W Q& 196.040 kJ/s Q& = m& h fg → m& = = = 0.920 kg/s h fg 213 kJ/kg Ro T∞2 (b) The heat transfer rate and the rate of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are A = πD 2 = π (31 . Assumptions 1 Heat transfer is transient, but can be treated as steady at average conditions. 4 The human body is modeled as a cylinder. kg / m3)(0.0314 m2)(3 m / s) = 0.0971 kg / s and Q& loss = m& C p (Tin − Tout) = (0.0971 kg/s)(1.007 kJ/kg.°C)(65 − 60)°C = 0.489 kJ/s or 1760 kJ/h. m) Raluminum = Repoxy Q& = T1 − Tbase (T1 − 39.5)°C = R aluminum + R epoxy + R board (0.00028 + 0.00555 + 0.011) °C/W → T1 = 39.5°C + (15 W)(0.0168 °C/W) = 39.8°C T − T2 Q& = 1 → T2 = T1 − Q& R board = 39.8°C − (15 W)(0.011 °C/W) = 39.6°C R board 3-165 A circuit board houses electronic components on one side, dissipating a total of 15 W through the backside of the board to the surrounding medium. 3 Heat loss from the duct is negligible. Analysis The exact analytical solution to this problem is ⎛ x ⎞ T (x, t) − Ti = erfc⎜ ⎟ Ts − Ti ⎝ 2 αt ⎠ Wall 30 cm Substituting, ⎛ 0.3 m 5.1 − 5 ⎜ = 0.01 = erfc⎜ − 6 2 15 − 5 ⎝ 2 (0.45 ×10 m /s)t ⎞ ⎟ ⎟ Noting from Table 4-3 that 0.01 = erfc(1.8215), the time is determined to be ⎛ 0.3 m ⎜ ⎜ −6 2 ⎝ 2 (0.45 ×10 m /s)t ⎞ ⎟ → t = 15,070 s = 251 min ⎟ = 1.8215 ⎠ 4-118 Ti = 5°C 15°C 0 x Chapter 4 Transient Heat Conduction 4-125 A thick wall is exposed to cold outside air. 5-3C The energy balance method is based on subdividing the medium into a sufficient number of volume elements, and then applying an energy balance on each element. 5-48 Chapter 5 Numerical Methods in Heat Conduction 5-54E A long solid bar is subjected to steady two-dimensional heat transfer. We measure x from the bottom surface of the block since this surface represents the adiabatic center surface of the plane wall of thickness 2L = 4 cm. Finite Difference Formulation of Differential Equations 5-6C A point at which the finite difference formulation of a problem is obtained is called a node, and all the nodes for a problem constitute the nodal network. 3 The heat transfer coefficients are constant. Properties The average surface temperature is (350+250)/2 = 300°C, and the properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-15) k = 0.02588 W/m.°C υ = 1.608 × 10 -5 m 2 /s μ ∞ = 1.872 × 10 −5 Air V∞ = 6 m/s T∞ = 30°C kg/m.s μ s , @ 300 °C = 2.934 × 10 −5 kg/m.s D = 15 cm Ts = 350°C Pr = 0.7282 D Analysis The Reynolds number is V D (6 m/s)(0.15 m) Re = ∞ = = 5.597 × 10 4 υ 1.57 × 10 −5 m 2 /s The Nusselt number corresponding this Reynolds number is determined to be Nu = [ ] ⎛μ hD = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 ⎜⎜ ∞ k ⎝ μs ⎞ 1/ 4 ⎛ 1.872 × 10 −5 ⎟⎟ ⎠ ⎞ ⎟ = 145.6 = 2 + 0.4(5.597 × 10 4) 0.5 + 0.06(5.597 × 10 4) 2 / 3 (0.7282) 0.4 ⎜⎜ −5 ⎝ 2.934 × 10 ⎠ Heat transfer coefficient is 0.02588 W/m.°C k h = Nu = (145.6) = 25.12 W/m 2 . 39 × 10 −5 m 2 / s . An energy balance on the device device can be expressed as 30 W E −E +E = ΔE → − Q& Δt + E& Δt = mC ΔT in or, out generation device out generation p device ⎛ T + T∞ ⎞ E& generation Δt − hAs ⎜ − T∞ ⎟Δt = mC p (T − T∞) 2 ⎝ ⎠ Substituting the given values, ⎛ T − 25 ⎞ o (30 J/s)(5 × 60 s) − (12 W/m 2 .°C)(0.0005 m 2)⎜ ⎟ C(5 × 60 s) = (0.02 kg)(850 J/kg.°C)(T − 25)°C ⎝ 2 ⎠ which gives T = 527.8°C If the device were attached to an aluminum heat sink, the temperature of the device would be ⎛ T − 25 ⎞ (30 J/s)(5 × 60 s) − (12 W/m 2 .°C)(0.0085m 2)⎜ ⎟°C(5 × 60 s) = (0.20 + 0.02) kg × (850 J/kg.°C)(T − 25)°C ⎝ 2 ⎠ which gives T = 69.5°C Note that the temperature of the electronic device drops considerably as a result of attaching it to a heat sink. m2) L 0.0015 m Rair = R2 = R4 = R6 = R8 = = = 0.0524 ° C / W kA (0.026 W / m.o C)(11 . Properties The thermal conductivity of the insulating material is given to be k = 0.02 Btu/h·ft·°F. 2 The thermal properties of the fin are constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. ° C)(1 m) R1 = Rpipe = Rinsulation = ln(r3 / r2) ln(r3 / 0.033) = = 4.55 ln(r3 / 0.033) ° C / W 2πk 2 L 2π (0.035 W / m. m2 T∞1 T∞2 Ao = πDo L = π (0.046 m)(15 m) = 2.168 m2 1 1 = = 0.0044 °C/W 2 hi A (120 W/m .°C)(1.885 m 2) ln(r2 / r1) ln(2.3 / 2) = = = 0.00003 °C/W 2πk1 L 2π (52 W/m.°C)(15 m) Ri = R pipe The outer surface temperature of the pipe will be somewhat below the water temperature. Solar energy is incident on the pond surface at 45° at an average rate of 500 W/m2 for a period of 4 h The temperature distribution in the pond under the most favorable conditions is to be determined. ° C)(0.0025 m) 4h = kD tanh aL tanh(14.40 m-1 × 0.02 m) = 0.973 = aL 14.40 m -1 × 0.02 m The fins can be assumed to be at base temperature provided that the fin area is modified by multiplying it by 0.973. Then the answer of this problem is zero since the problem states to disregard radiation. ° C) T∞ h T∞ h k g 0 ro r (b) The mathematical formulation of this problem can be expressed as 1 d ⎛ dT ⎞ g& ⎜r ⎟+ = 0 r dr ⎝ dr ⎠ k dT (r0) = h[T (r0) − T∞] (convection at the outer surface) dr dT (0) = 0 (thermal symmetry about the centerline) dr Multiplying both sides of the differential equation by r and integrating gives g& dT g& r 2 d ⎛ dT ⎞ (a) =− + C1 ⎜r ⎟=− r → r dr k 2 dr ⎝ dr ⎠ k Applying the boundary condition at the center line, dT (0) g& 0× =− × 0 + C1 → C1 = 0 B.C. at r = 0: dr 2k Dividing both sides of Eq. (a) by r to bring it to a readily integrable form and integrating, g& 2 dT g& =− r T (r) = − r + C2 → (b) 4k dr 2k Applying the boundary condition at r = r0 , and − k g&r0 g& r g& 2 ⎛ g& 2 ⎞ = h⎜ − r0 + C 2 − T∞ ⎟ → C 2 = T∞ + 0 + r0 2k 2 h 4k ⎝ 4k ⎠ Substituting this C2 relation into Eq. (b) and rearranging give & gr g& 2 T (r) = T∞ + (r0 − r 2) + 0 4k 2h which is the temperature distribution in the wire as a function of r. C. 4 Heat generation in the rod is uniform. Analysis (a) The inner and the outer surface areas of sphere are Ai = πDi 2 = π (5 m) 2 = 78.54 m 2 Ao = πDo 2 = π (5.03 m) 2 = 79.49 m 2 We assume the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surfaces of the tank. Properties The thermal conductivity is given to be k (T) = k0 (1 + βT) . 2 Thermal properties, heat transfer coefficients, and the indoor and outdoor temperatures are constant. The temperature of the aluminum plate is to be determined. The surface temperature of the plate is to be determined when it stabilizes. h, T∞ Convectio T 0 Analysis The nodal network consists of 3 nodes, and the base Δx temperature T0 at node 0 is specified. The thermal resistance Analysis (a) The representative surface area is A = 012 network and the individual thermal resistances Rare 2 R5 R1 R3 T1 R4 R7 R6 T2 0.01 m ⎛ L ⎞ = 0.04 °C/W R1 = R A = ⎜ ⎟ = 2 ⎝ kA ⎠ A (2 W/m.°C)(0.12 m) 0.05 m ⎛ L⎞ = 0.06 °C/W R 2 = R 4 = RC = ⎜ ⎟ = 2 ⎝ kA ⎠ C (20 W/m.°C)(0.04 m) 0.05 m ⎛ L ⎞ = 0.16 °C/W R3 = R B = ⎜ ⎟ = 2 ⎝ kA ⎠ B (8 W/m.°C)(0.04 m) 0.1 m ⎛ L ⎞ R5 = R D = ⎜ ⎟ = = 0.11 °C/W ⎝ kA ⎠ D (15 W/m.o C)(0.06 m 2) 0.1 m ⎛ L ⎞ R6 = R E = ⎜ ⎟ = = 0.05 o C/W 2 ⎝ kA ⎠ E (35 W/m.°C)(0.06 m) 0.06 m ⎛ L ⎞ R7 = R F = ⎜ ⎟ = = 0.25 °C/W 2 ⎝ kA ⎠ F (2 W/m.°C)(0.12 m) 1 1 1 1 1 1 1 = + + = + + → Rmid ,1 = 0.025 °C/W Rmid ,1 R2 R3 R4 0.06 0.16 0.06 1 Rmid , 2 = 1 1 1 1 = + → Rmid , 2 = 0.034 °C/W R5 R6 0.11 0.05 Rtotal = R1 + Rmid ,1 + Rmid , 2 + R7 = 0.04 + 0.025 + 0.034 + 0.25 = 0.349 °C/W T − T∞ 2 (300 − 100)°C Q& = ∞1 = = 572 W (for a 0.12 m × 1 m section) 0.349 °C/W Rtotal Then steady rate of heat transfer through entire wall becomes (5 m)(8 m) Q& total = (572 W) = 1.91 × 10 5 W 0.12 m 2 (b) The total thermal resistance between left surface and the point where the sections B, D, and E meet is Rtotal = R1 + Rmid ,1 = 0.04 + 0.025 = 0.065 °C/W Then the temperature at the point where the sections B, D, and E meet becomes T −T Q& = 1 → T = T1 − Q& Rtotal = 300°C − (572 W)(0.065 °C/W) = 263°C Rtotal (c) The temperature drop across the section F can be determined from 3-34 Chapter 3 Steady Heat Conduction ΔT Q& = → ΔT = Q& R F = (572 W)(0.25 °C/W) = 143°C RF 3-35 Chapter 3 Steady Heat Conduction 3-58 A composite wall consists of several horizontal and vertical layers. Therefore, there are only 6 unknown nodal temperatures, and thus we need only 6 equations to determine them uniquely. 2-2C The term steady implies no change with time at any point within the medium while transient implies variation with time or time dependence. Therefore, even if we eliminate the thermal contact resistance at the interface completely, we will lower the operating temperature of the transistor in this case by less than 1°C. Properties The emissivity of the ball surface is given to be ε = 0.8. Air Analysis The heat transfer surface area is 70°F 170° As = πD² = π(2/12 ft)² = 0.08727 ft² Under steady conditions, the rates of convection and radiation heat transfer are D = 2 in o 2 o 2 Q Q& conv = hAs ΔT = (12Btu/h.ft · F)(0.08727ft)(170 − 70) F = 104.7 Btu/h Q& rad = εσAs (Ts4 − To4) = 0.8(0.08727ft 2)(0.1714 × 10 −8 Btu/h.ft 2 · R 4)[(170 + 460R) 4 − (70 + 460R) 4] = 9.4 Btu/h Therefore, Q& = Q& + Q& = 104.7 + 9.4 = 114.1 Btu / h total conv rad Discussion Note that heat loss by convection is several times that of heat loss by radiation. • • • The finite difference equations for boundary nodes are obtained by applying an balance on h, energy T∞ the volume elements and taking the direction of all heat transfers to be towards the node under consideration: Tleft + Ttop + Tright + Tbottom − 4Tnode + Node 1 (convection) : 2k g& l 2 l T2 − T1 l + 2h (T∞ − T1) + 0 = 0 2 2 4 l T − T2 g& l 2 l T1 − T2 + kl 5 + hl (T∞ − T2) + 0 = 0 2 2 l 1 2 g& l 4T2 − 4T5 + 0 = 0 k Node 2 (convection) : 2k Node 5 (interior) : where g& 0 = 0.19 ×10 5 Btu/h · ft 3 , l = 0.2 ft, k = 16 Btu/h.ft·°F, h =7.9 Btu/h.ft2·°F, and T∞ =70°F. The temperature is assumed to vary linearly between the nodes, especially when expressing heat conduction between the elements using Fourier's law. The time it takes to register 99 percent of the initial ΔT is to be determined. [4(0.0002) + 3(0.0015)] = 0.000954 m 2 k A Q& = eff (T1 − T2) L & QL (27 / 2 W)(018 . ° C)(0.000954 m 2) 3-117 Epoxy Chapter 3 Steady Heat Conduction 3-158 The plumbing system of a house involves some section of a plastic pipe exposed to the ambient air. 727b, f = 0.44. 2 Heat transfer is one-dimensional. Two half pieces of the roast have a much larger surface area than the single piece and thus a higher rate of heat transfer. The final equilibrium temperature in the tank is to be determined. Analysis The one-dimensional transient temperature distribution in the ground can be determined from ⎛ x T (x, t) − Ti = erfc⎜ T ∞ − Ti ⎝ 2 αt ⎛ ⎞⎟ ⎛ hx h 2αt ⎞ ⎡ ⎜ ⎜erfc⎜ x + h αt ⎜ − exp⎜ + ⎜ k ⎜ 2 αt k ⎠⎟⎢ k 2 ⎝⎟⎠⎣ ⎝ ⎞⎤ ⎟⎥ ⎟ ⎠⎦ Winds T∞ =-10°C where 2 -5 2 h αt (40 W/m .°C) (1.6 × 10 m / s)(10 × 3600 s) = = 33.7 0.9 W/m.°C k Soil Ti 2 ⎛ h αt ⎞ = 33.7 2 = 1138 ⎜ ⎟ = ⎜ k ⎟ ⎝ ⎠ h 2αt k2 Then we conclude that the last term in the temperature distribution relation above must be zero regardless of x despite the exponential term tending to infinity since (1) erfc (ξ) → 0 for ξ > 4 (see Table 4-3) and (2) the term has to remain less than 1 to have physically meaningful solutions. Analysis We consider a volume element of size Δx × Δy × 1 centered about a general interior node (m, n) in a region in which heat is generated at a constant rate of g& and the thermal conductivity k is variable (see Fig. 3-56 Chapter 3 Steady Heat Conduction 3-75E A steam pipe covered with 2-in thick fiberglass insulation is subjected to convection on its surfaces. The local and global discretization errors are identical for the first time step. Noting that Pr It is valid for a Prandtl number range of 0.6 < Pr < 60. When the diameter and length of the cylinder are comparable, it is not proper to treat the cylinder as being infinitely long. 3 Heat transfer through the base is negligible. 4 The water in the pipe is stationary, and its initial temperature is 0°C. 3-84C It will decrease. 2-115C A differential equation is said to be linear if the dependent variable and all of its derivatives are of the first degree, and their coefficients depend on the independent variable only. 3 Thermal properties of the wall and the heat transfer coefficients are constant. = = 0.00037 < 01 k (35 W / m. Properties The thermal conductivity and solar absorptivity are given to be k = 0.6 W/m·°C and αs = 0.7. Analysis The nodal spacing is given to be Δx=Δx=l=1 m, and all nodes are boundary nodes. 2 Convection heat transfer coefficient is constant and Radiation h, T∞ uniform. Assumptions 1 The orange is spherical in shape with a diameter of 8 cm. 5-65 Chapter 5 Numerical Methods in Heat Conduction 5-69C For transient one-dimensional heat conduction in a plane wall with both sides of the wall at specified temperatures, the stability criteria for the explicit method can be expressed in its simplest form as αΔt 1 τ= ≤ 2 2 (Δx) 5-70C For transient one-dimensional heat conduction in a plane wall with specified heat flux on both sides, the stability criteria for the explicit method can be expressed in its simplest form as αΔt 1 τ= ≤ 2 2 (Δx) which is identical to the one for the interior nodes. The area of the sphere normal to the direction of heat transfer at any location is A = 4πr 2 where r is the value of the radius at that location. Asymmetric radiation causes discomfort by exposing different sides of the body to surfaces at different temperatures and thus to different rates of heat loss or gain by radiation. (b) Direct contact with cold floor surfaces causes localized discomfort in the feet by excessive heat loss by conduction, dropping the temperature of the bottom of the feet to uncomfortable levels. / 12) ft = 13473 = . 4 This is a rigid tank and thus its volume remains constant. Their values are determined directly from k g T∞ =30°C g&L (5 × 10 5 W/m 3)(0.015 m) = 155°C = 30°C + h=60 W/m2.°C 2 h 60 W/m .°C 2L=3 cm g&L2 (5 × 10 5 W/m 3)(0.015 m) 2 To = T s + = 155°C + = 158.7 °C 2k 2(15.1 W/m.°C) Ts = T∞ + T∞ =30°C h=60 W/m2.°C 2-84 Heat is generated uniformly in a large brass plate. The temperatures at the top, middle, and bottom of the exposed surface of the damn are to be determined. There is only one inlet and one exit and thus m& 1 = m& 2 = m& . The thickness of insulation and the outer surface temperature of the wall are to be determined for two different insulating materials. There is uniform heat generation in the medium, and the thermal conductivity k of the medium is constant. Properties The thermal conductivity is given to be k (T) = k 0 (1 + βT 2) . 90 W + εAs σ (Tsurr 4 − Ts 4) = hAs (Ts − T∞) 90 + (0.9)(1.7)(5.67 × 10 −8)[(40 + 273) 4 − Ts 4] = (18.02)(1.7)[Ts − (32 + 273)] Ts = 309.2 K = 36.2°C 7-87 Chapter 7 External Forced Convection 7-95 The heat generated by four transistors mounted on a thin vertical plate is dissipated by air blown over the plate on both surfaces. 0063) exp − (0 . 5-53 Chapter 5 Numerical Methods in Heat Conduction 5-57 "!PROBLEM 5-57" "GIVEN" k=1.4 "[W/m-C]" A_flow=0.20*0.40 "[m^2]" t=0.10 "[m]" T_i=280 "[C], parameter to ve varied" h_i=75 "[W/m^2-C]" T_o=15 "[C]" h_o=18 "[W/m^2-C]" epsilon=0.9 "parameter to ve varied" T_sky=250 "[K]" DELTAx=0.10 "[m]" DELTAy=0.10 "[m]" d=1 "[m], unit depth is considered" sigma=5.67E-8 "[W/m^2-K^4], Stefan-Boltzmann constant" "ANALYSIS" "(b)" l=DELTAx "We consider only one-fourth of the geometry whose nodal network consists of 10 nodes. 2-141 Design and Essay Problems 2-77 Chapter 3 Steady Heat Conduction Chapter 3 STEADY HEAT CONDUCTION Steady Heat Conduction In Plane Walls 3-1C (a) If the lateral surfaces of the rod are insulated, the heat transfer surface area of the cylindrical rod is the bottom or the top surface area of the rod, As = πD 2 / 4 . The surface temperature of the shaft, the rate of heat transfer to the coolant, and the mechanical power wasted are to be determined. The rate of heat transfer per foot length of the tube when a 0.01 in thick layer of limestone is formed on the inner surface of the tube is to be determined. Assumptions 1 The temperature in the wall is affected by the thermal conditions at inner surfaces only and the convection heat transfer coefficient inside is very large. The fractions of heat lost from each person's body by respiration are to be determined. 2 Heat transfer through the insulation is onedimensional. (b) The Reynolds number is V x (5 m/s)(0.15 m) Re x = ∞ = = 4.532 × 10 4 5 2 − υ 1.655 × 10 m /s which is less than the critical Reynolds number but we assume the flow to be turbulent since the electronic components are expected to act as turbulators. T (t) = T∞ + Q& in hA 4-9 Chapter 4 Transient Heat Conduction 4-21 "!PROBLEM 4-21" "GIVEN" E_dot=1000 "[W]" L=0.005 "[m]" A=0.03 "[m^2]" T_infinity=22 "[C]" T_i=T_infinity h=12 "[W/m^2-C], parameter to be varied" f_heat=0.85 T_f=140 "[C], parameter to be varied" "PROPERTIES" rho=2770 "[kg/m^3]" C_p=875 "[J/kg-C]" alpha=7.3E-5 "[m^2/s]" "ANALYSIS" V=L*A m=rho*V Q_dot_in=f_heat*E_dot Q_dot_out=h*A*(T_ave-T_infinity) T_ave=1/2*(T_i+T_f) (Q_dot_in-Q_dot_out)*time=m*C_p*(T_f-T_i) "energy balance on the plate" h [W/m2.C] 5 7 9 11 13 15 17 19 21 23 25 time [s] 51 51.22 51.43 51.65 51.88 52.1 52.32 52.55 52.78 53.01 53.24 Tf [C] 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 time [s] 3.428 7.728 12.05 16.39 20.74 25.12 29.51 33.92 38.35 42.8 47.28 51.76 56.27 60.8 65.35 69.92 74.51 79.12 4-10 Chapter 4 Transient Heat Conduction 53.25 52.8 tim e [s] 52.35 51.9 51.45 51 5 9 13 2 17 21 25 h [W /m -C] 80 70 60 tim e [s] 50 40 30 20 10 0 20 40 60 80 100 120 T f [C] 4-11 140 160 180 200 Chapter 4 Transient Heat Conduction 4-22 Ball bearings leaving the oven at a uniform temperature of 900°C are exposed to air for a while before they are dropped into the water for quenching. 3 The surrounding surfaces are at the same temperature as the air in the room. The rate of heat transfer from the plastic sheet is to be determined. Analysis The individual resistances are Rpipe Ri Ro Ai = πDi L = π (0.4 / 12 ft)(1 ft) = 0105 . Properties The thermal conductivity and diffusivity of the eggs are given to be k = 0.6 W/m.°C and α = 0.14×10-6 m2/s. (b) Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be a= hp = kAc hπD kπD / 4 2 = 4h = kD 4(50 W / m2 . Also, the energy balance expressions can be simplified using the definitions of thermal diffusivity α = k / (ρC) and the dimensionless mesh Fourier number τ = αΔt / l 2 where Δx = Δy = l . 3 Thermal properties of the chest are constant. The house that is more energy efficient is to be determined. °C / W Then the temperatures on the two sides of the circuit board becomes T −T & . Analysis Using the energy balance approach and taking the direction of all heat transfers to be towards the node under consideration, the finite difference formulations become Left boundary node (all temperatures are in K): 4 εσA(Tsurr − T04) + hA(T∞ − T0) + kA Heat transfer at right surface: Q& right surface T1 − T0 + q& 0 A + g& 0 (AΔx / 2) = 0 Δx + kA T 4 − T5 + g& 5 (AΔx / 2) = 0 Δx 5-20 A composite plane wall consists of two layers A and B in perfect contact at the interface where node 1 is. 3 The thermal properties of the meat slab are constant. Analysis The initial rate of heat transfer from a potato is As = πD 2 = π (0.10 m) 2 = 0.03142 m 2 Air V∞ = 1 m/s T∞ = 5°C Q& = hAs (Ts − T∞) = (19.1 W/m 2 .°C)(0.03142 m 2)(20 − 5)°C = 9.0 W where the heat transfer coefficient is obtained from the table at 1 m/s velocity. The outer surface of the shell is subjected to radiation to surrounding surfaces at Tsurr . 4-95C This claim is reasonable since the lower the storage temperature, the longer the storage life of beef. Assumptions 1 Heat conduction in the short cylinder is two-dimensional, and thus the temperature varies in both the axial x- and the radial r- directions. Properties Assuming a film temperature of Tf = 10°C for the outdoors, the properties of air are evaluated to be (Table A-15) k = 0.02439 W/m.°C υ = 1.426 × 10 -5 m 2 /s Air V∞ = 50 km/h T∞2 = 4°C T∞1 = 22°C WALL Pr = 0.7336 Analysis Air flows along 8-m side. Analysis (a) The amount of heat transferred to the valve is simply the change in its internal energy, and is determined from Q = ΔU = mC p (T2 − T1) = (0.0788 kg)(0.440 kJ/kg.°C)(800 - 40)°C = 26.35 kJ (b) The average rate of heat transfer can be determined from Q 26.35 kJ Q& ave = = = 0.0878 kW = 87.8 W Δt 5 × 60 s (c) The average heat flux is determined from Q& Q& 87.8 W q& ave = ave = ave = = 1.75× 10 4 W/m 2 2π DL 2π (0.008 m)(0.1 m) As (d) The number of valves that can be heat treated daily is Number of valves = (10 × 60 min)(25 valves) = 3000 valves (5 min) 1-77 Engine valve T1 = 40°C T2 = 800°C D = 0.8 cm L = 10 cm Chapter 1 Basics of Heat Transfer 1-136 Somebody takes a shower using a mixture of hot and cold water. The finite difference formulation of the boundary nodes and the finite difference formulation for the rate of heat transfer at the left boundary are to be determined. The atmospheric pressure in atm is 1 atm P = (83.4 kPa) = 0.823 atm 101.325 kPa The kinematic viscosity at this atmospheric pressure will be Air V∞ = 4 m/s T∞ = 35°C υ = (1.798 × 10 −5 m 2 /s) / 0.823 = 2.184 × 10 −5 m 2 /s Transistors Ts = 65°C L = 25 cm Analysis The Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = = 4.579 × 10 4 υ 2.184 × 10 −5 m 2 /s which is less than the critical Reynolds number (5 × 105). The temperature distribution throughout the glass 15 min after the strip heaters are turned on and also when steady conditions are reached are to be determined using the explicit method. Noting that the R-values of the wood fiberboard and the rigid foam insulation are 0.23 m2.°C/W and 0.98 m2.°C/W, respectively, and the added and removed thermal resistances are in series, the overall Rvalue of the wall after modification becomes Rnew = Rold − Rremoved + Radded = 3.213 − 0.23 + 0.98 = 3.963 m2 . 7-4C The force a flowing fluid exerts on a body in the normal direction to flow that tend to move the body in that direction is called lift. Note that the second derivative of temperature at a node n is expressed in terms of the temperatures at node n and its two neighboring nodes 5-8 The finite difference formulation of steady two-dimensional heat conduction in a medium with heat generation and constant thermal conductivity is given by Tm−1,n − 2Tm,n + Tm+1,n Tm,n −1 − 2Tm,n + Tm,n +1 g& m,n + + =0 k Δx 2 Δy 2 in rectangular coordinates. Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow. 4 Thermal properties and heat transfer coefficients are constant. (c) Convection and radiation: Yes. Analysis (a) Noting that heat transfer is steady and one-dimensional in the radial r direction, the mathematical formulation of this problem can be expressed as d dr ⎛ dT ⎞ ⎜r ⎟=0 ⎝ dr ⎠ dT (r1) = hi [Ti − T (r1)] dr dT (r2) −k = ho [T (r2) − To] dr −k and (b) Integrating the differential equation once with respect to r gives r Ti hi r1 r2 r To ho dT = C1 dr Dividing both sides of the equation above by r to bring it to a readily integrable form and then integrating, dT C1 = dr r T (r) = C1 ln r + C2 where C1 and C2 are arbitrary constants. It is determined from 1 1 1 = + −1 ε effective ε1 ε2 where ε1 and ε2 are the emissivities of the surfaces of the air space. Using the energy balance approach and taking the direction of all heat transfers to be towards the node under Δx • • D 2• 0 1 consideration, the finite difference formulations become ε Node 1 (at midpoint): Radiation T −T T −T 4 kA 0 1 + kA 2 1 + h(pΔx / 2)(T∞ − T1) + εσA[Tsurr − (T1 + 273) 4] = 0 Tsurr Δx Δx Node 2 (at fin tip): T − T2 4 kA 1 + h(pΔx / 2 + A)(T∞ − T2) + εσ (pΔx / 2 + A)[Tsurr − (T2 + 273) 4] = 0 Δx where A = πD 2 / 4 is the cross-sectional area and p = πD is the perimeter of the fin. 4-109 Chapter 4 Transient Heat Conduction 4-117E "!PROBLEM 4-117E" "GIVEN" 2*L=1/12 "[ft]" 2*r_o_c=1/12 "[ft], c stands for cylinder" 2*r_o_s=1/12 "[ft], s stands for sphere" T_i=400 "[F]" T_infinity=75 "[F]" h=7 "[Btu/h-ft^2-F]" "time=5 [min], parameter to be varied" "PROPERTIES" k=15 "[Btu/h-ft-F]" alpha=0.333*Convert(ft^2/h, ft^2/min) "[ft^2/min]" "ANALYSIS" "For plane wall" Bi_w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda_1_w=0.1410 A_1_w=1.0033 tau_w=(alpha*time)/L^2 (T_o_w-T_infinity)/(T_i-T_infinity)=A_1_w*exp(-lambda_1_w^2*tau_w) "For long cylinder" Bi_c=(h*r_o_c)/k "From Table 4-1 corresponding to this Bi number, we read" lambda_1_c=0.1995 A_1_c=1.0050 tau_c=(alpha*time)/r_o_c^2 (T_o_c-T_infinity)/(T_i-T_infinity)=A_1_c*exp(-lambda_1_c^2*tau_c) "For sphere" Bi_s=(h*r_o_s)/k "From Table 4-1 corresponding to this Bi number, we read" lambda_1_s=0.2445 A_1_s=1.0060 tau_s=(alpha*time)/r_o_s^2 (T_o_s-T_infinity)/(T_i-T_infinity)=A_1_s*exp(-lambda_1_s^2*tau_s) time [min] 5 10 15 20 25 30 35 40 45 50 55 60 To,w [F] 312.3 247.7 200.7 166.5 141.6 123.4 110.3 100.7 93.67 88.59 84.89 82.2 To,c [F] 247.9 166.5 123.4 100.6 88.57 82.18 78.8 77.01 76.07 75.56 75.3 75.16 4-110 To,s [F] 200.7 123.4 93.6 82.15 77.75 76.06 75.41 75.16 75.06 75.02 75.01 75 T o [F] Chapter 4 Transient Heat Conduction 350 350 300 300 250 250 200 200 w all 150 150 cylinder 100 50 0 100 sphere 10 20 30 40 tim e [m in] 4-111 50 50 60 Chapter 4 Transient Heat Conduction 4-118 Internal combustion engine valves are quenched in a large oil bath. Assumptions 1 Heat conduction in the rod is one-dimensional since the rod is sufficiently long, and thus temperature varies in the radial direction only. Also, the mesh Fourier number for the steaks is τ steak = αΔt Δx 2 = (0.93 × 10 −6 m 2 / s)(5 s) (0.005 m) 2 = 0186 . 3 Air is an ideal gas. Properties The thermal conductivity of the aluminum is given to be k = 237 W/m·°C. 7-75C The R-value of insulation is the thermal resistance of the insulating material per unit surface area. = 16.5 kW Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least m& water = Q& water 16.5kW = = 1.97 kg/s (C p ΔT) water (4.18 kJ/kg.º C)(2º C) If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C. This is because the heat flux boundary conditions have no effect on the stability criteria. Analysis Disregarding the base area, the total heat transfer area of the electronic box is As = (0.4 m)(0.4 m) + 4 × (0.2 m)(0.4 m) = 0.48 m 2 The radiation heat transfer from the box can be expressed as 100 W ε = 0.95 100 W = (0.95)(5.67 × 10 −8 W/m 2 .K 4)(0.48 m 2)[(55 + 273 K) 4 − Tsurr 4] Ts =55°C which gives Tsurr = 296.3 K = 23.3°C. The energy balance in this case can be expressed as E in − E out = ΔE system E in = ΔU system = ΔU water + ΔU tea pot Then the amount of energy needed to raise the temperature of water and the teapot from 18°C to 96°C is Water 18°C Heater 1200 W E in = (mCΔT) water + (mCΔT) teapot = (2.5 kg)(4.18 kJ / kg. Analysis Disregarding the base area, the total heat transfer area of the transistor is As = πDL + πD 2 / 4 = π (0.6 cm)(0.4 cm) + π (0.6 cm) 2 / 4 = 1.037 cm 2 = 1.037 × 10 −4 m 2 Then the rate of heat transfer from the power transistor at specified conditions is Q& = hAs (Ts − T∞) = (30 W/m 2 .°C)(1.037 × 10 -4 m 2)(70 − 55)°C = 0.047 W Therefore, the amount of power this transistor can dissipate safely is 0.047 W. Properties The thermal conductivity and emissivity are given to be k = 6 • 15.1 W/m·°C and ε = 0.8. Tsurr 5 • Analysis The nodal spacing is given to be Δx=3 cm. Properties The heat of fusion of water at atmospheric pressure is hif = 333. The heat of vaporization of water at 100°F is given to be 1037 Btu/lbm. ° F / Btu 2 hi A1 hi (2πr1 L) (30 Btu / h.ft . For the case of steady heat conduction with no heat generation it reduces to 1 ∂ ⎛ ∂T ⎞ ∂ 2 T =0 ⎜+ ⎜r r ∂r ⎝ ∂r ⎠ ∂z 2 2-10 Chapter 2 Heat Conduction Equation 2-30 Consider a thin disk element of thickness Δz and diameter D in a long cylinder (Fig. 2 Thermal properties of the wall are constant. Tsky Radiation ε Plate Convection h, T∞ 0 1 2 3 4 5 • • • • 1 in • • 6 • Soil 7 • 0.6 ft 8 • This system of 10 equations with 10 unknowns constitute the 9 • finite difference formulation of the problem. 3 The thermal properties of the granite are constant. Properties The conductivity and diffusivity are given to be k = 28 W/m·°C and α = 12 × 10 −6 m2 / s .) = = 0.00244 h. The flow of compressible fluid (such as air) is not necessarily compressible since the density of a compressible fluid may still remain constant during flow. kJ / s)(4160 × 3600 s / yr) = 1.524 × 109 kJ / yr Rinsulation Ro Ts The efficiency of the furnace is given to be 78 percent. 3-2C In steady heat conduction, the rate of heat transfer into the wall is equal to the rate of heat transfer out of it. Dividing the equation above by AΔr gives − T − Tt 1 Q& r + Δr − Q& r + g& = ρC t + Δt A Δr Δt Taking the limit as Δr → 0 and Δt → 0 yields ∂T 1 ∂ ⎛ ∂T ⎞ ⎜ kA ⎜ + g& = ρC A ∂r ⎝ ∂t ∂r ⎠ since, from the definition of the derivative and Fourier's law of heat conduction, Q& r + Δr − Q& r ∂Q ∂ ⎛ ∂T ⎞ = = ⎜ − kA ⎜ Δr → 0 ∂ r ∂r ⎝ ∂r ⎠ Δr lim Noting that the heat transfer area in this case is A = 2πrL and the thermal conductivity is constant, the onedimensional transient heat conduction equation in a cylinder becomes 1 ∂ ⎛ ∂T ⎞ 1 ∂T ⎜r ⎜ + g& = r ∂r ⎝ ∂r ⎠ α ∂t where α = k / ρC is the thermal diffusivity of the material. 8-12C The logarithmic mean temperature difference ΔTln is an exact representation of the average temperature difference between the fluid and the surface for the entire tube. When preparing a mathematical model, all the variables that affect the phenomena are identified, reasonable assumptions and approximations are made, and the interdependence of these variables are studied. Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. m2 Rcond Ro T∞1 1 1 = = 0.0032 °C/W 2 hi Ai (180 W/m .°C)(1.73 m 2) ln(r2 / r1) ln(5 / 4.6) = = 0.00004 °C/W Rcond = 2πkL 2π (52 W/m.°C)(6 m) 1 1 = = 0.0213 °C/W Ro = 2 ho Ao (25 W/m .°C)(1.88 m 2) Ri = T∞2 T1 T2 R total = Ri + Rcond + Ro = 0.0032 + 0.00004 + 0.0213 = 0.0245 °C/W The rate of heat transfer and average outer surface temperature of the pipe are T −T (200 − 12)°C Q& = ∞1 ∞ 2 = = 7673 W R total 0.0245 °C/W T − T∞ 2 Q& = 2 → T2 = T∞ 2 + Q& Ro = 12 °C + (7673 W)(0.0213 °C/W) = 174.8°C Ro (b) The fin efficiency can be determined from Fig. Substituting the given quantities, the maximum allowable value of the time step is determined to be Δt ≤ (0.015 m) 2 4(3.2 × 10 −6 m 2 /s)[1 + (80 W/m 2 .°C)(0.015 m) /(15 W/m.°C)] = 16.3 s Therefore, any time step less than 16.3 s can be used to solve this problem. 2 Heat transfer is one-dimensional since heat transfer from the side surfaces are disregarded 3 Thermal conductivities are constant. 7-88 Chapter 7 External Forced Convection 7-96 A spherical tank used to store iced water is subjected to winds. 3-77 Chapter 3 Steady Heat Conduction 3-110 Circular aluminum fins are to be attached to the tubes of a heating system. Therefore, the wood will not ignite. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are As = NπDL = 200π(0.016 m)(1 m) = 10.05 m 2 m& = m& i = ρ i V(N T S T L) = (1.204 kg/m 3)(5.2 m/s)(10)(0.05 m)(1 m) = 3.130 kg/s Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become ⎛ Ah Te = Ts − (Ts − Ti) exp⎜ − s ⎜ m& C p ⎝ 2 2 ⎞ ⎛ ⎞ ⎟ = 100 − (100 − 20) exp⎜ − (10.05 m)(108.7 W/m · °C) ⎟ = 43.44°C ⎜ (3.130 kg/s)(1007 J/kg · °C) ⎟ ⎟ ⎝ ⎠ ⎠ 7-61 Chapter 7 External Forced Convection ΔTln = (Ts − Ti) − (Ts − Te) (100 − 20) − (100 − 43.44) = = 67.6°C ln[(Ts − Ti) /(Ts − Te)] ln[(100 − 20) /(100 − 43.44)] Q& = hAs ΔTln = (108.7 W/m 2 · °C) (10.05 m 2)(67.6°C) = 73,882 W (b) For this in-line arrangement tube bank, the friction coefficient corresponding to ReD = 6806 and SL/D = 5/1.6 = 3.125 is, from Fig. The phrase "energy generation," however, is vague since the form of energy generated is not clear. The amount of heat dissipated in 10 h and the heat flux on the surface of the circuit board are to be determined. The average friction coefficient is to be determined. The rate of evaporation of liquid nitrogen in the tank as a result of the heat transfer from the ambient air is to be determined. 2-24 Chapter 2 Heat Conduction Equation Assumptions 1 Heat conduction is steady and one-dimensional since the pipe is long relative to its thickness, and there is thermal symmetry about the center line. 2 2 T0 − T∞ . 2 The thermal properties of the block are constant. Properties The properties of steaks are given to be k = 0.45 W/m.°C and α = 0.91×10-7 m2/s Analysis The Biot number is Bi = hL (9 W/m 2 .°C)(0.01 m) = = 0.200 k (0.45 W/m.°C) Steaks 25°C The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, λ 1 = 0.4328 and A1 = 10311 . We measure x from the midplane. T (r) = 257.2 − 473.68(0.175) 2 + 98.34 ln(0.175) = 71.2°C 2-137 A spherical ball in which heat is generated uniformly is exposed to iced-water. This would be a transient heat transfer process since the temperature at any point within the egg will change with time during cooking. 5-99 Chapter 5 Numerical Methods in Heat Conduction 5-95 "!PROBLEM 5-95" "GIVEN" t_ins=0.03 "[m]" k=0.026 "[W/m-C]" alpha=0.36E-6 "[m^2/s]" T_i=3 "[C]" h_i=6 "[W/m^2-C]" h_o=9 "[W/m^2-C]" T_infinity=25 "[C]" m_food=15 "[kg]" C_food=3600 "[J/kg-C]" DELTAx=0.01 "[m]" DELTAt=60 "[s]" "time=6*3600 [s], parameter to be varied" "PROPERTIES" rho_air=density(air, T=T_i, P=101.3) C_air=CP(air, T=T_i)*Convert(kJ/kg-C, J/kg-C) "ANALYSIS" M=t_ins/DELTAx+1 "Number of nodes" tau=(alpha*DELTAt)/DELTAx^2 RhoC=k/alpha "RhoC=rho*C" "The technique is to store the temperatures in the parametric table and recover them (as old temperatures) using the variable ROW. Therefore, T∞ =30°C the flow is laminar. Then the energy equation with dissipation (Eqs. 4-54 Chapter 4 Transient Heat Conduction 4-59 Chickens are to be chilled by holding them in agitated brine for 2.5 h. 5 The Fourier number is τ > 0.2 so that the one-term approximate solutions are applicable (this assumption will be verified). In the absence of any heat generation, an energy balance on this thin element of thickness Δx during a small time interval Δt can be expressed as ΔE element Q& x − Q& x +Δx = Δt where ΔEelement = Et + Δt − Et = mC (Tt + Δt − Tt) = ρCAΔx (Tt + Δt − Tt) Substituting, T − Tt Q& x − Q& x + Δx = ρCAΔx t + Δt Δt

Dividing by AΔx gives − T − Tt 1 Q& x + Δx − Q& x = ρC t + Δt A Δx Δt Taking the limit as Δx → 0 and Δt → 0 yields ∂T 1 ∂ (∂T) kA = ρC A ∂x ∂x ∂t since, from the definition of the derivative and Fourier's law of heat conduction, Q& x + Δx − Q& x ∂Q ∂ (∂T) = = − kA Δx →0 Δx ∂x ∂x ∂x lim Noting that the area A of a plane wall is constant, the one-dimensional transient heat conduction equation in a plane wall with constant thermal conductivity k becomes ∂ 2 T 1 ∂T = ∂x 2 α ∂t where the property α = k / ρC is the thermal diffusivity of the material. 5 The thermal properties of the fins are constant. ° C) The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, λ1 = 3.0877 and A1 = 1.9969 Water 97°C Egg Ti = Then the Fourier number becomes θ 0, sph = 2 2 T 0 − T∞ 70 − 97 = A1e − λ1 τ → = (1.9969)e −(3.0877) τ → τ = 0.198 ≈ 0.2 Ti − T∞ 8 − 97 Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. 2 Heat transfer is one-dimensional since there is thermal symmetry about the center line and no change in g the axial direction. Analysis The inner radius of the pipe is r1 = 3.0 cm and the outer radius of the pipe and thus the inner radius of insulation is r2 = 3.3 cm. 4951) (0.05) 2 → t = 108,135 s = 30.04 hours Therefore, the ice will start melting in about 30 hours. 5 Heat loss through the floor is negligible. ° C) = 12.04 m-1 (386 W / m. Outside surface, 24 km/h (winter) 2. 4 The top surface at x = L is subjected to specified temperature and the bottom surface at x = 0 is subjected to uniform heat flux. Properties The properties of air at 1 atm pressure and the free stream temperature of 20°C are (Table A-15) k = 0.02514 W/m.°C υ = 1.516 × 10 -5 m 2 /s Insulation μ ∞ = 1.825 × 10 −5 kg/m.s μ s , @ −183°C = 6.127 × 10 −5 kg/m.s Pr = 0.7309 Analysis (a) When there is no insulation, D = Di = 4 m, and the Reynolds number is V D [(40 × 1000/3600) m/s](4 m) Re = ∞ = = 2.932 × 10 6 υ 1.516 × 10 −5 m 2 /s The Nusselt number is determined from Nu = [ ] (μ hD = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 ∞ k μs [= 2 + 0.4(2.932 × 10) 6 0.5 Do Wind 20°C 40 km/h Di Oxygen tank -183°C 1/4 + 0.06(2.932 × 10) 6 2/3 (1.825 × 10 −5 (0.7309) 0.4 −5 1.05 × 10 1/4 = 2220 0.02514 W/m.°C k Nu = (2220) = 13.95 W/m 2 .°C D 4m The rate of heat transfer to the liquid oxygen is Q& = hA (T − T) = h(πD 2)(T − T) = (13.95 W/m 2 .°C)[π (4 m) 2][(20 − (−183)] °C = 142,372 W and h= s s ∞ s ∞ The rate of evaporation of liquid oxygen then becomes Q& 142.4 kJ/s Q& = m& hif → m& = = = 0.668 kg/s hif 213 kJ/kg (b) Note that after insulation the outer surface temperature and diameter will change. The energy balance for this steady-flow system can be expressed in the rate form as E& − E& = ΔE& systemÊ0 (steady) = 0 → E& in = E& out 1in424out 3 144 42444 3 Rate of net energy transfer by heat, work, and mass Rate of change in internal, kinetic, potential, etc. °C) 4-45 Chapter 4 Transient Heat Conduction 4-54 The center temperature of potatoes is to be lowered to 6°C during cooling. The thermal resistances involved and the rate of heat transfer are ln(r2 / r1) ln(12 . The heat flux on the surface of the filament, the heat flux on the surface of the glass bulb, and the annual electricity cost of the bulb are to be determined. Analysis (a) The nodal spacing is given to be Δx=0.5 cm. Analysis In steady operation, heat conduction through the wall must be equal to net heat transfer from the outer surface. Properties We assume the film temperature to be 200 ° F . 3-15C The new design introduces the thermal resistance of the copper layer in addition to the thermal resistance of the aluminum which has the same value for both designs. The total discretization error at any step is called the global or accumulated discretization error. Assuming the heater operates 2,000 hours during a heating season, the annual cost of this heat loss adds up to $24. The dimensionless temperature for a two-dimensional problem is determined by determining the dimensionless temperatures in both directions, and taking their product. 3 Thermal properties of 32°F the milk are constant at room temperature. A variable whose value can be changed arbitrarily is called an independent variable (or argument). × 105) 0.5 (1505)1/ 3 = 2908 k k 0141 . and Heat loss through the end surface of the kiln with styrofoam: Ri Rstyrofoa Ro Tin Tout 1 1 = = 0.201× 10 − 4 °C/W 2 hi Ai (3000 W/m .°C)[(4 − 0.4)(5 − 0.4) m 2] L 0.02 m = = = 0.0332 °C/W kAave (0.033 W/m.°C)[(4 − 0.2)(5 − 0.2) m 2] Ri = R styrofoam Ro = 1 1 = = 0.0020 °C/W ho Ao (25 W/m 2 .°C)[4 × 5 m 2] Rtotal = Ri + R styrpfoam + Ro = 0.201× 10 − 4 + 0.0332 + 0.0020 = 0.0352 °C/W and T − Tout [40 − (−4)]° C = = 1250 W Q& end surface = in 0.0352 ° C / W Rtotal Then the total rate of heat transfer from the kiln becomes Q& total = Q& top + sides + 2Q& side = 85,500 + 2 × 1250 = 88,000 W 3-62 "GIVEN" width=5 "[m]" 3-39 Chapter 3 Steady Heat Conduction height=4 "[m]" length=40 "[m]" L_wall=0.2 "[m], parameter to be varied" k_concrete=0.9 "[W/m-C]" T_in=40 "[C]" T_out=-4 "[C]" L_sheet=0.003 "[m]" L_styrofoam=0.02 "[m]" k_styrofoam=0.033 "[W/m-C]" h_i=3000 "[W/m^2-C]" "h_o=25 [W/m^2-C], parameter to be varied" "ANALYSIS" R_conv_i=1/(h_i*A_1) A_1=(2*height+width-3*L_wall)*length R_concrete=L_wall/(k_concrete*A_2) A_2=(2*height+width-1/2*3*L_wall)*length R_conv_o=1/(h_o*A_3) A_3=(2*height+width)*length R_total_top_sides=R_conv_i+R_concrete+R_conv_o Q_dot_top_sides=(T_in-T_out)/R_total_top_sides "Heat loss from top and the two side surfaces" R_conv_i_end=1/(h_i*A_4) A_4=(height-2*L_wall)*(width-2*L_wall) R_styrofoam=L_styrofoam/(k_styrofoam*A_5) A_5=(height-L_wall)*(width-L_wall) R_conv_o_end=1/(h_o*A_6) A_6=height*width R_total_end=R_conv_i_end+R_styrofoam+R_conv_o_end Q_dot_end=(T_in-T_out)/R_total_end "Heat loss from one end surface" Q_dot_total=Q_dot_top_sides+2*Q_dot_end

= Q& conv + Q& rad = hAs (Ts − T∞) + εAs σ (Ts 4 − Tsurr 4) 90 W = (17.36 W/m 2 .°C)(0.0314 m 2)[Ts − (25 + 273)K] + (0.9)(0.0314 m 2)(5.67 × 10 -8 W/m 2 .K 4) Ts 4 − (25 + 273 K) 4 Ts = 406.2 K = 133.2°C 7-52 A steam pipe is exposed to a light winds in the atmosphere. Then 45° 0 the number of nodes becomes M = L / Δx + 1 = 1/0.25+1 = 4. Outside surface, 12 km/h wind 2. The coefficient of T4i is smaller in this case, and thus the stability criteria for this problem can be expressed as 1 − 2τ − 2τ hΔx k ≥0 → τ≤ 1 2(1 + hΔx / k) since τ = αΔt / Δx 2 from the stability criteria that requires all primary coefficients to be greater than or equal to zero. ° C are 5 cm × 5 cm × 5 cm in size. Assumptions 1 The thermal resistance of the tank and the internal convection resistance are negligible. Their centerline temperature when they leave the oven is to be determined. 245.65 > 52.4) . This corresponds to Q& 249 W %change = difference × 100 = × 100 = 3.8% (increase) & 6559 W Q total,0° m 2)(5.67 × 10 -8 W/m 2 .K 4) (75 + 273 K) 4 − (25 + 273 K) 4 = 1159 W = Q& + Q& = 5001 + 1159 = 6160 W conv] rad 0.191 kg/min 0 Then the rate of heat transfer from the wire to the air becomes Q& = m& C p [T (t) − T∞] = (0.191 kg/min) Assumptions 1 The thermal properties of the valves are constant. ° F)(1 ft) 2πkL 2π (17 = Rtotal,new + Rlimestone,i = 0.005 the pipe and thus the internal convection resistance slightly, but this effect should be negligible. 3-88 An electric wire is tightly 0.664(4.579 × 10 4) 0.5 (0.7228)1 / 3 = 127.5 Nu = k k 0.02735 W/m.°C h = Nu = (127.5) = 13.95 W/m 2 .°C L 0.25 m As = this plate becomes 26.2 W n= = 4.4 → 4 6W This result is conservative since the transistors will cause the flow to be turbulent calculations. Properties The thermal conductivity of the plate and the soil are given to be kplate = 7.2 Btu/h·ft·°F and ksoil = conditions exist. This system of 8 equations with 8 unknowns constitutes the finite difference formulation of the problem. 3 B 0.00039 + 01361 . Analysis (a) The radius of the roast is determined to be m = ρV → V = V= m ρ = 3.2 kg 1200 kg/m 3 = 0.0 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding occur on the surface (r/r0 = 1), and is determined to be 2 sin(λ1 r / r0) T (r0) − T∞ sin(λ1 r0 / r0) T0 − T∞ sin(λ1 r0 / r0) T to 4 °C for chilling injury for potatoes. The increase in heat transfer from the tubes per unit length as a result of adding fins i dependent variable or its derivatives involve the independent variable as a coefficient, that equation is said to have variable c equivalent thickness is determined from the relation above to be L = kR = kRc = (386 W / m. 3 The system is well-insulated a wall is subjected to specified temperature on the left surface and convection on the right surface. This is known as the no-slip reflective sheets. 4 Thermal contact resistances at the interfaces are disregarded. ° F)(1 ft) 2πkL 2π (17 ln(ro / r2) ln(0.66 / Tmi+1 − Tmi = → Tmi +1 = τ (Tmi −1 + Tmi +1) + (1 − 2τ)Tmi k τ The finite difference equations for other nodes are obtained 2 (interior) : T2i +1 = τ steak (T1i + T3i) + (1 − 2τ steak)T2i Node 3 (interior) : T3i +1 = τ steak (T2i + T4i) + (1 − 2τ steak)

98.1 W Q& = As q& L = (nDW) 0.001 m L (b) This is equivalent to the rate of heat transfer through the cylindrical sleeve by conduction, which is expressed as 2πW (T0 − Ts) Q& = k ln(D0 / D) − (70 W/m · °C) 2π (0.15 m)(T0 − 40°C) = 98.1 W ln(8 / 5) which gives the surface temperature of the shaft to be T0 = 40.7°C (c) The mechanical power wasted by the viscous dissipation in oil is equivalent to the rate of heat generation, W& lost = g& = 98.1 W 6-24 Chapter 6 Fundamentals of Convection Momentum and Heat Transfer Analogies 6-47C Reynolds analogy is expressed as C f , x Re L = Nu x . The amount of heat transfer to the air and its cost are to be determined. Assumptions 1 The water supplied to the house is at room temperature. Noting that D = D0 = 4.1 m, the Nusselt number becomes Re = V∞ D (40 × 1000/3600 m/s)(4.1 m) ... The rate of heat transfer is to be determined when steady operating conditions are reached. Then the Nusselt number and the heat transfer coefficient are determined to be h L N u = ... The mechanical power ... The convection resistance ... Chapter 7 External Forced Convection ... The fin efficiency ... Properties The thermal conductivity of the aluminum alloy is given to be k = ... Analysis The nodal network of this problem consists of 5 nodes, and 3 4 the base temperature of the fin ... The thermal properties of the turkey are ... the specific heat ... Assumptions 1 Heat transfer through the wall is steady since the surface temperatures remain constant at the specified values. ... The properties of air at 1 atm and the film temperature ... Properties The thermal conductivity of fiberglass insulation ... Analysis The electrical resistance of the wire ... The total rate of heat generation in the wire ... Chapter 2 Heat Conduction Equation ... Chapter 3 Steady Heat Conduction ... Chapter 4 Transient Heat Conduction ... Chapter 5 Numerical Methods in Heat Conduction ... We observe that this is a steady-flow process since there is no change with time at any point and thus ΔmCV = 0 and ΔE CV = 0 , there is only one inlet and one exit and thus m& 1 = m& 2 = m& , and the tube is insulated. 7-6C The frontal area of a body is the area seen by a person when looking from upstream. The entry length is much longer in laminar flow than it is in turbulent flow. 2 Thermal properties of the aluminum pan are constant. Then surface temperatures on the two sides of the circuit board becomes T −T & Q& = 1 ∞ → T1 = T∞ + QR total = 37° C + (15 W)(1.492 ° C / W) = 59.4° C Rtotal T −T & Q& = 1 2 → T2 = T1 − QR board = 59.4° C − (15 W)(0.011 ° C / W) = 59.2° C Rboard 2 cm (b) Noting that the cross-sectional areas of the fins are constant, the efficiency of these rectangular fins is determined to be hp ≅ kAc a= h(2w) = k (tw) 2h = kt 2(45 W/m 2 .°C) = 13.78 m -1 (237 W/m.°C)(0.002 m) tanh aL tanh(13.78 m -1 × 0.02 m) = = 0.975 aL 13.78 m -1 × 0.02 m The finned and unfinned surface areas are η fin = 0.002 ⎞ t⎞ ⎛ ⎛ 2 Afinned = (20)2 w⎜ L + ⎟ = (20)2(0.15)⎜ 0.02 + ⎟ = 0.126 m 2⎠ 2 ⎠ ⎝ ⎝ Aunfinned = (0.1)(0.15) − 20(0.002)(0.15) = 0.0090 m 2 Then, Raluminum Repoxy Q& finned = η fin Q& fin,max = η fin hAfin (Tbase − T∞) T1 Rboard T∞ Q& unfinned = hAunfinned (Tbase − T∞) Q& total = Q& unfinned + Q& finned = h(Tbase − T∞)(η fin Afin + Aunfinned) Substituting, the base temperature of the finned surfaces is determined to be Q& total Tbase = T∞ + h(η fin Afin + Aunfinned) = 37°C + 15 W (45 W/m 2 .°C)[(0.975)(0.126 m 2) + (0.0090 m 2)] = 39.5°C Then the temperatures on both sides of the board are determined using the thermal resistance network to be 3-124 Chapter 3 Steady Heat Conduction L 0.001 m = = 0.00028 ° C / W kA (237 W / m. The primary freezing methods of poultry are the air blast tunnel freezing, cold plates, immersion freezing, and cryogenic cooling. ° C)(1 m2) 0.02 m L Discussion Note that superinsulators are very effective in reducing heat transfer between to surfaces. Properties The thermal conductivity and emissivity are given to be k =7.2 Btu/h·ft·°F and ε = 0.6. Analysis (a) Taking the direction normal to the surface of the plate to be the x direction with x = 0 at the bottom surface, and the mathematical formulation of this problem can be expressed as and d 2T =0 dx 2 dT (L) 4 4 −k = h[T (L) − T∞] + εσ [T (L) 4 − Tsky] = h[T2 − T∞] + εσ [(T2 + 460) 4 − Tsky] dx T (L) = T2 = 75° F Tsky (b) Integrating the differential equation twice with respect to x yields x dT = C1 dx 75°F ε T∞ h L T (x) = C1x + C2 where C1 and C2 are arbitrary constants. 4 Heat transfer from the base of the transistor is negligible. ° C) The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, Plates 25°C λ 1 = 01039 . Analysis The thermal resistances of different layers for unit surface area of 1 m2 are Copp er 1 1 = = 0.00017 °C/W Rcontact = Epox Epox 2 2 h A (6000 W/m .°C)(1 m) c c Rplate = 0.001 m L = = 2.6 × 10 −6 ° C / W kA (386 W / m. Analysis Heat cannot be conducted through an evacuated space since the thermal conductivity of vacuum is zero (no medium to conduct heat) and thus its thermal resistance is zero. 6 5 7 Construction 1a. It yields dT (0) g& 0× =− × 0 + C1 → C1 = 0 B.C. at r = 0: dr 2k Dividing both sides of Eq. (a) by r to bring it to a readily integrable form and integrating, dT g& =− r dr 2k g& 2 T (r) = − r + C2 (b) and 4k Applying the other boundary condition at r = r0 , B. 5 The air leaks out at 22°C. energies 0 = ΔU cars + ΔKE cars 0 = (mCΔT) cars + [m(0 − V 2) / 2]cars That is, the decrease in the kinetic energy of the cars must be equal to the increase in their internal energy. 2 The thermal properties of the junction are constant. The smallest and thus the most restrictive primary coefficient in this case is the coefficient of T0i in the formulation of node 0 since hin > hout, and thus 1 − 2τ − 2τ h Δx hin Δx < 1 − 2τ − 2τ out k k Therefore, the stability criteria for this problem can be expressed as 1 − 2τ − 2τ hin Δx ≥0 → k τ≤ 1 2(1 + hin Δx / k) → Δt ≤ Δx 2 2α (1 + hin Δx / k) since τ = αΔt / Δx 2 . m2 = 0.0174 m2 1 1 . 13-37C Friction drag is due to the shear stress at the surface whereas the pressure drag is due to the pressure differential between the front and back sides of the body when a wake is formed in the rear. The rate of heat loss from the wall is to be determined. In gases and liquids, it is due to the collisions of the molecules during their random motion. 2 The thermal properties of the iron base and the convection heat transfer coefficient are constant and uniform. 5-10 Chapter 5 Numerical Methods in Heat Conduction 5-25 A long triangular fin attached to a surface is considered. Plaster board, 20 mm 6. T2=40°C Noting that u = u(y), v = 0, and ∂P / ∂x = 0 (flow is maintained by the motion of the upper plate rather than the pressure L=0.7 mm Oil gradient), the x-momentum equation (Eq. 6-28) reduces to ⎛ ∂u d 2u ∂u ⎞ ∂ 2 u ∂P → 2 = 0 + v ⎟⎟ = μ 2 − ρ⎜⎜ u ∂y ⎠ ∂x dy ∂y ⎝ ∂x This is a second-order ordinary differential equation, and integrating it twice gives x-momentum: T1=25°C u (y) = C1 y + C 2 The fluid velocities at the plate surfaces must be equal to the velocities of the plates because of the no-slip condition. Properties The conductivity and diffusivity are given to be k = 1.4 W/m.°C and α = 0.69 × 10 −6 m2 / s . Analysis The time required to freeze the turkeys Brine from 1°C to -18ºC with brine at -29ºC can be -29°C determined directly from Fig. Then the heat transfer surface area becomes As = N tubenDL = (1× N L)π (0.01 m)(4 m) = 0.1257N L Then the log mean temperature difference, and the expression for the rate of heat transfer become ΔTln = (Ts − Ti) − (Ts − Te) (90 − 15) − (90 − 65) = = 45.51°C ln[(Ts − Ti) /(Ts − Te)] ln[(90 − 15) /(90 − 65)] Q& = hAs ΔTln = (16,994 W/m 2 ⋅ °C)(0.1257N L)(45.51°C) = 97,220 N L The mass flow rate of water through a cross-section corresponding to NT =1 and the rate of heat transfer are m& = ρAcV = (999.1 kg/m 3)(4 × 0.03 m 2)(0.8 m/s) = 95.91 kg/s Q& = m& C p (Te − Ti) = (95.91 kg/s)(417 9 J/kg.C) (65 − 15)°C = 2.004 × 10 7 W Substituting this result into the heat transfer expression above we find th e number of tube rows 7-65 D Chapter 7 External Forced Convection Q& = hAs ΔTln → 2.004 × 10 7 W = 97,220 N L → N L = 206 7-66 Chapter 7 External Forced Convection 7-70 Air is cooled by an evaporating refrigerator. Assumptions 1 The temperature in the container walls is affected by the thermal conditions at outer surfaces only and the convection heat transfer coefficient outside inside is given to be very large. Therefore, we first need to determine the dimensionless temperature at the surface of the wall. 4-22 -1)(600 s) = 448 °C Chapter 4 Transient Heat Conduction 4-37 "!PROBLEM 4-37" "GIVEN" L=0.03/2 "[m]" T_i=25 "[C]" T_infinity=700 "[C], parameter to be varied" time=10 "[min], parameter to be varied" h=80 "[W/m^2-C]" "PROPERTIES" k=110 "[W/m-C]" alpha=33.9E-6 "[m^2/s]" "ANALYSIS" Bi=(h*L)/k "From Table 4-1, corresponding to this Bi number, we read" lambda_1=0.1039 A_1=1.0018 tau=(alpha*time*Convert(min, s))/L^2 (T_L-T_infinity)/(T_i-T_infinity)=A_1*exp(-lambda_1^2*tau)*Cos(lambda_1*L/L) T∞ [C] 500 525 550 575 600 625 650 675 700 725 750 775 800 825 850 875 900 TL [C] 321.6 337.2 352.9 368.5 384.1 399.7 415.3 430.9 446.5 462.1 477.8 493.4 509 524.6 540.2 555.8 571.4 time [min] 2 4 6 8 10 12 TL [C] 146.7 244.8 325.5 391.9 446.5 491.5 4-23 Chapter 4 Transient Heat Conduction 14 16 18 20 22 24 26 28 30 528.5 558.9 583.9 604.5 621.4 635.4 646.8 656.2 664 600 550 T L [C] 500 450 400 350 300 550 600 650 700 T ∞ 4-24 [C] 750 800 850 900 Chapter 4 Transient Heat Conduction 700 600 T L [C] 500 400 300 200 100 0 5 10 15 20 tim e [m in] 4-25 25 30 Chapter 4 Transient Heat Conduction 4-38 A long cylindrical shaft at 400°C is allowed to cool slowly. The mass flow rate of hot water and the average temperature of mixed water are to be determined. Assumptions 1 Steady operating conditions exist since the ball surface and the surrounding air and surfaces remain at constant temperatures. The emissivity of the outer surface of the roof is given to be 0.9. Analysis In steady operation, heat transfer from the outer surface of the roof to the surroundings by convection and radiation must be equal to the heat transfer through the roof by conduction. ° C)(140 − 22)° C Δt = = = 51.8 s (850 − 21.2) J / s Q& − Q& in out which is the time required for the plate temperature to reach 140 °C . Assumptions 1 Heat conduction in the trunks is one-dimensional since it is long and it has thermal symmetry about the center line. Properties The gas constant of hydrogen is R = 4.124 kPa.m3/kg.K (Table A-1). On windy days, the external convection heat transfer coefficient is greater compared to calm days. Then the maximum velocity and the Reynolds number based on the maximum velocity become V=4.5 m/s Vmax = Re D Ts=80°C SL Ti=300°C ST 0.08 V= (4.5 m/s) = 6.102 m/s 0.08 − 0.021 ST − D ρV D (0.6746 kg/m 3)(6.102 m/s)(0.021 m) = max = 3132 μ 2.76 × 10 −5 kg/m ⋅ s ST The average Nusselt number is determined using the proper relation from Table 7-2 to be Nu D = 0.27 Re 0D.63 Pr 0.36 (Pr/ Prs) 0.25 = 0.27(3132) 0.63 (0.6946) 0.36 D (0.6946 / 0.7154) 0.25 = 37.46 Since NL =16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become Nu D, N L = Nu D = 37.46 h= Nu D , N L k D = 37.46(0.04104 W/m ⋅ °C) = 73.2 W/m 2 ⋅ °C 0.021 m The total number of tubes is N = NL ×NT = 16×8 = 128. Special thanks are due to Dr. Mehmet Kanoglu who checked the accuracy of most solutions in this Manual. 2 Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. It is to be determined whether the wood will ignite. Assumptions 1 Heat conduction in the hot dog is two-dimensional, and thus the temperature varies in both the axial x- and the radial r- directions. Assumptions 1 Steady operating conditions exist since the plate temperatures remain constant. This requires the identification of the smallest primary coefficient in the system. 1-3C The caloric theory is based on the assumption that heat is a fluid-like substance called the "caloric" which is a massless, colorless, odorless substance. It is to be determined whether the temperature at the outer surfaces of the kiln changes during the curing period. Analysis From Fig. The finite difference formulation of the left Δx q0 boundary node (node 0) and the finite difference formulation for the rate of heat transfer at the right • • • • • • 0 1 2 boundary (node 5) are to be determined. In summer, we can keep cool by dressing lightly, staying in cooler environments, turning a fan on, avoiding humid places and direct exposure to the sun. g = 7×107 W/m3 Analysis The total rate of heat generation in the rod is D = 5 cm determined by multiplying the rate of heat generation per unit volume by the volume of the rod L=1m & G& = gV = g& (πD 2 / 4) L = (7 × 107 W / m3)[π (0.05 m) 2 / 4](1 m) = 1.374 × 105 W = 137.4 kW rod 2-17 The variation of the absorption of solar energy in a solar pond with depth is given. m2 Aunfinned = 0.0216 − 864 πD 2 4 Atotal,with fins = Afinned + Aunfinned Rconv = 1 hAtotal,with fins = 0.0216 − 864 × π (0.0025) 2 4 = 0132 . Inside surface, still air R-value, m2.°C/W Summer Winter --0.030 0.044 --0.075 0.075 0.018 0.018 0.27 0.27 0.16 0.16 0.122 0.122 0.12 0.12 4 3 2 Total unit thermal resistance of each section (the R-value) , m2.°C/W 1 0.809 0.795 2 Therefore, the overall unit thermal resistance of the wall is R = 0.809 m .°C/W in summer and R = 0.795 m2.°C/W in winter. 7-64 Combustion air is preheated by hot water in a tube bank. Properties The thermal conductivity of the concrete is given to be k = 0.55 W/m·°C. We try to minimize drag in order to reduce fuel consumption in vehicles, improve safety and durability of structures subjected to high winds, and to reduce noise and vibration. Assumptions 1 Heat transfer is transient and two-dimensional. The average heat transfer coefficient and the cooling time of the potato if it is loosely wrapped completely in a towel are to be determined. The thermal resistance network and individual resistances are Ri R3 R2 R1 R4 T∞ 1 R6 R5 1 1 = = 0.303 ° C / W 2 h1 A (10 W / m . 4-7 Chapter 4 Transient Heat Conduction 4-19E A person shakes a can of drink in a iced water to cool it. Analysis Expressing all the temperatures in Kelvin, the differential equation and the boundary conditions for this heat conduction problem can be expressed as d 2T =0 dx 2 dT (0) −k = h1[T∞1 − T (0)] dx dT (L) 4 −k = h1[T (L) − T∞ 2] + ε 2σ T (L) 4 − Tsky dx Tsky T∞1 h1 T∞2 h2 L 2-17 x Chapter 2 Heat Conduction Equation Solution of Steady One-Dimensional Heat Conduction Problems 2-52C Yes, this claim is reasonable since in the absence of any heat generation the rate of heat transfer through a plain wall in steady operation must be constant. 2 Heat transfer from the fin tips is negligible. h° F / Btu 4πkr1r2 4π (0.035 Btu / h.ft. 7 Heat transfer through the bottom surface of the refrigerator is negligible. The midplane in the latter case will behave like an insulated surface because of thermal symmetry. 3 Thermal properties and heat transfer coefficients are constant. 4-6C Biot number represents the ratio of conduction resistance within the body to convection resistance at the surface of the body. During a differential time interval dt, the temperature of the body rises by a differential amount dT. 2 Heat transfer is one-dimensional in the axial Interface direction since the lateral surfaces of both cylinders are well-insulated. In turbulent flow, on the other hand, it will be highest when θ is between 90° and 120° . Properties The thermal properties of the brick are given to be k = 0.72 W/m.°C and α = 1.6×10-7 m2/s. The mass of water in the tank is D2 L = (971.8 kg/m 3)π(0.50 m) 2 (0.95 m)/4 = 181.27 kg 4 The amount of heat transfer from the water is determined from Q = mC p (T2 − T1) = (181.27 kg)(4197 J/kg.°C)(80 - T2)°C m = ρV = ρπ Then average rate of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = = Δt 45 × 60 s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of water 7-50 Chapter 7 External Forced Convection (181.27 kg)(4197 J/kg.°C)(80 - T2)°C ⎛ 80 + T2 ⎞ − 18 ⎟°C = Q& = (26.53 W/m 2 .°C)(1.885 m 2)⎜ 2 45 × 60 s ⎝ ⎠ → T2 = 69.9°C 7-51 Chapter 7 External Forced Convection 7-59 "!PROBLEM 7-59" "GIVEN" D=0.50 "[m]" L=0.95 "[m]" T_w1=80 "[C]" T_infinity=18 "[C]" Vel=40 "[km/h]" "time=45 [min], parameter to be varied" "PROPERTIES" Fluid$='air' k=Conductivity(Fluid$, T=T_film) Pr=Prandtl(Fluid$, T=T_film) rho=Density(Fluid$, T=T_film, P=101.3) mu=Viscosity(Fluid$, T=T_film) nu=mu/rho T_film=1/2*(T_w_ave+T_infinity) rho_w=Density(water, T=T_w_ave, P=101.3) C_p_w=CP(Water, T=T_w_ave, P=101.3)*Convert(kJ/kg-C, J/kg-C) T_w_ave=1/2*(T_w1+T_w2) "ANALYSIS" Re=(Vel*Convert(km/h, m/s)*D)/nu Nusselt=0.3+(0.62*Re^0.5*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^0.25*(1+(Re/282000)^(5/8))^(4/5) h=k/D*Nusselt A=pi*D*L+2*pi*D^2/4 Q_dot=h*A*(T_w_ave-T_infinity) m_w=rho_w*V_w V_w=pi*D^2/4*L Q=m_w*C_p_w*(T_w1-T_w2) Q_dot=Q/(time*Convert(min, s)) time [min] 30 45 60 75 90 105 120 135 150 165 180 195 210 225 240 255 270 285 300 Tw2 [C] 73.06 69.86 66.83 63.96 61.23 58.63 56.16 53.8 51.54 49.39 47.33 45.36 43.47 41.65 39.91 38.24 36.63 35.09 33.6 7-52 Chapter 7 External Forced Convection 75 70 65 T w 2 [C] 60 55 50 45 40 35 30 0 50 100 150 200 tim e [m in] 7-53 250 300 Chapter 7 External Forced Convection 7-60 Air flows over a spherical tank containing iced water. The total rate of heat loss from the person is determined from 4) = (0.90)(5.67 × 10 −8 W/m 2 .K 4)(1.7 m 2)[(32 + 273) 4 − (23 + 273) 4]K 4 Q& rad = εσAs (Ts4 − Tsurr = 84.8 W & Qconv = hAs ΔT = (5W/m 2 ⋅ K)(1.7m 2)(32 − 23)°C = 76.5W Tsurr and 23° Q& total = Q& conv + Q& rad = 84.8 + 76.5 = 161.3 W Qrad Discussion Note that heat transfer from the person by evaporation, which is of comparable magnitude, is not considered in this problem. Analysis (See Figure 5-49 in the text). 2 The thermal properties of the water are constant. 4-18 Chapter 4 Transient Heat Conduction 4-34 An egg is dropped into boiling water. Analysis The shape factor for this configuration is given in Table 3-5 to be a 16 2πL 2π (10 m) = = 0.8 < 1.41 → S = = = 358.7 m b 20 ⎛ a ⎞ 0.785 ln 0.8 0.785 ln⎜ ⎟ 16 cm ⎝b⎠ Then the steady rate of heat transfer through the walls of the duct becomes Q& = Sk (T1 − T2) = (358.7 m)(0.75 W/m.°C)(100 − 15)°C = 2.29 ×10 4 W = 22.9 kW 3-97 20 cm Chapter 3 Steady Heat Conduction 3-131 A spherical tank containing some radioactive material is buried in the ground. The rate of heat transfer and the amount of ice that melts per day are to be determined. 4-43, the center and the surface temperatures of the hot dog 4 min after the start of the cooking and the amount of heat transferred to the hot dog are to be determined. Using the explicit finite difference method, the time it takes to defrost the steaks is to be determined. The temperatures at the center and at the surface of the ball are to be determined. Use the DUPLICATE statement to reduce the number of equations that need to be typed. Assumptions 1 Heat transfer through the wall is steady since the surface temperatures remain constant at the specified values. Analysis (a) The heat transfer area of the stud is A = (0.1 m)(2.5 m) = 0.25 m2. But we will assume turbulent flow because of the constant agitation of the engine block. Properties The thermal conductivity is given to be k = 30 W/m·°C. Then the number 4 • of nodes M becomes 3 cm h, T∞ 3 • L 18 cm 2 • M = +1 = +1 = 7 Δx 3 cm 1 • 0 • The base temperature at node 0 is given to be T0 = 95°C. = = 01361 °C / W hAtotal,with fins (50 W / m 2 . That is, Q& = Q& = Q& = Q& room to roof, conv+rad roof, cond roof to surroundings, conv+rad Taking the inner and outer surface temperatures of the roof to be Ts,in and Ts,out , respectively, the quantities above can be expressed as 4 Q& = h A (T − T) + εA σ (T − T 4) = (5 W/m 2 .°C)(300 m 2)(20 − T)°C room to roof, conv + rad i s room Q& s ,in s s ,in room + (0.9)(300 m)(5.67 × 10 2 −8 [s ,in W/m .K) (20 + 273 K) − (Ts ,in + 273 K) 4 2 4 4] Ts ,in − Ts ,out Ts ,in − Ts ,out = (2 W/m.°C)(300 m 2) Q& roof, cond = kAs L 0.15 m Q& roof to surr, conv + rad = ho As (Ts ,out − Tsurr) + εAs σ (Ts ,out 4 − Tsurr 4) = (31.0 W/m 2 .°C)(300 m 2)(Ts ,out − 10)°C [+ (0.9)(300 m 2)(5.67 × 10 −8 W/m 2 .K 4) (Ts ,out + 273 K) 4 − (100 K) 4 Solving the equations above simultaneously gives Q& = 28,025 W = 28.03 kW, T = 10.6°C, and T s ,in s ,out = 3.5°C The total amount of natural gas consumption during a 14-hour period is Q Q& Δt (28.03 kJ/s)(14 × 3600 s) ⎛ 1 therm ⎞ Q gas = total = = ⎜⎜ = 15.75 therms ⎟⎟ 0.85 0.85 0.85 ⎝ 105,500 kJ ⎠ Finally, the money lost through the roof during that period is Money lost = (15.75 therms)($0.60 / therm) = $9.45 7-92] Chapter 7 External Forced Convection 7-100 Steam is flowing in a stainless steel pipe while air is flowing across the pipe. Therefore, to generate this much heat, the furnace must consume energy (in the form of natural gas) at a rate of Qin = Q / η oven = (2.001 × 109 kJ / yr) / 0.78 = 2.565 × 109 kJ / yr = 24,314 therms / yr since 1 therm = 105,500 kJ. The optimum thickness of insulation and the amount of money saved per year are to be determined. It yields dT (0) g& 0× =− × 0 + C1 → C1 = 0 B.C. at r = 0: dr 2k Dividing both sides of Eq. (a) by r to bring it to a readily integrable form and integrating, dT g& =− r dr 2k g& 2 T (r) = − r + C2 and (b) 4k Applying the second boundary condition at r = r0 , B. The energy balance in this case reduces to = ΔE system E −E 1in424out 3 1 424 3 Net energy transfer by heat, work, and mass Change in internal, kinetic, potential, etc. air" A_i=2*(1.8-0.03)*(0.8-0.03)+2*(1.8-0.03)*(0.7-0.03)+(0.8-0.03)*(0.7-0.03) m_air=rho_air*V_air V_air=(1.8-0.03)*(0.8-0.03)*(0.7-0.03) 5-101 Chapter 5 Numerical Methods in Heat Conduction Time [s] T1 [C] 3 35.9 5.389 36.75 6.563 37 7.374 37.04 8.021 36.97 ... ... 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.82 0 60 120 180 240 300 360 420 480 540 ... ... 35460 35520 35580 35640 35700 35760 35820 35880 35940 36000 T2 [C] 3 3 10.11 7.552 13.21 9.968 15.04 11.55 16.27 12.67 ... ... 24.23 24.24 24.23 24.24 24.24 24.25 24.25 24.26 24.25 24.26 T3 [C] 3 3 3 4.535 4.855 6.402 6.549 7.891 7.847 8.998 ... ... 23.65 23.65 23.66 23.67 23.67 23.68 23.68 23.69 23.69 23.7 T4 [C] 3 3 3 3 3.663 3.517 4.272 4.03 4.758 4.461 ... ... 23.09 23.1 23.11 23.12 23.12 23.13 23.14 23.15 23.15 23.16 T5 [C] 3 3 3 3 3 3.024 3.042 3.087 3.122 3.182 ... ... 22.86 22.87 22.88 22.88 22.89 22.9 22.91 22.92 22.93 22.94 25 20.5 T 5 [C] 16 11.5 7 2.5 0 5000 10000 15000 20000 25000 30000 35000 40000 Tim e [s] 5-102 Row 1 2 3 4 5 6 7 8 9 10 ... ... 592 593 594 595 596 597 598 599 600 601 Chapter 5 Numerical Methods in Heat Conduction Special Topic: Controlling the Numerical Error 5-96C The results obtained using a numerical method differ from the exact results obtained analytically because the results obtained by a numerical method are approximate. Therefore, (20.26-3.81)/20.26 = 81.2% of the heat transfer occurs through the steel bars across the wall despite the negligible space that they occupy, and obviously their effect cannot be neglected. Analysis The convection heat transfer coefficients and the rate of heat losses at different wind velocities are (a) h = 14.8V 0.53 = 14.8(0.5 m/s) 0.69 = 9.174 W/m 2 .°C Q& = hAs (Ts − T∞) = (9.174 W/m 2 .°C)(1.7 m 2)(29 − 10)°C = 296.3 W (b) h = 14.8V 0.53 = 14.8(1.0 m/s) 0.69 = 14.8 W/m 2 .°C Ts = 29°C Air V∞ T∞ = 10°C Q& = hAs (Ts − T∞) = (14.8 W/m 2 .°C)(1.7 m 2)(29 − 10)°C = 478.0 W (c) h = 14.8V 0.53 = 14.8(1.5 m/s) 0.69 = 19.58 W/m 2 .°C Q& = hAs (Ts − T∞) = (19.58 W/m 2 .°C)(1.7 m 2)(29 − 10)°C = 632.4 W 6-11 The expression for the heat transfer coefficient for air cooling of some fruits is given. This way the node next to the boundary node appears on both sides of the boundary node because of symmetry, converting it into an interior node. m)(6 m) + 2π (1.2 m) 2 / 4 = 24.88 m2 Q& = hA (T − T) = (25 W/m 2 .°C)(24.88 m 2)[30 − (−42)]°C = 44,787 W tank ∞1 ∞2 Propane The volume of the tank and the mass of the propane are V = πr 2 L = π (0.6 m) 2 (6 m) = 6.786 m 3 m = ρV = (581 kg/m 3)(6.786 m 3) = 3942.6 kg The rate of vaporization of propane is Q& 44.787 kJ/s Q& = m& h fg → m& = = = 0.1054 kg/s 425 kJ/kg h fg tank, -42°C Rins, ends Rconv, o Ts T∞ Then the time period for the propane tank to empty becomes 3942.6 kg m Rsins, sides Δt = = = 37,413 s = 10.4 hours m& 0.1054 kg/s (b) We now repeat calculations for the case of insulated tank with 7.5-cm thick insulation. 2 Heat transfer is one-dimensional since the wall is large relative to its thickness, and the thermal conditions on both sides of the wall are T2 uniform. A 1-9C Energy can be transferred by heat, work, and mass. 3-101C If the fin is too long, the temperature of the fin tip will approach the surrounding temperature and we can neglect heat transfer from the fin tip. 4-13b we obtain x 5 cm = =1 L 5 cm T (L, t) − T∞ − 1 − (−12) = = 0.65 5 − (−12) To − T∞ ⎤ ⎥ ⎥ ⎬ ⎥ ⎦ ⎥ ⎦ 1 k = = 0.95 Bi hL Air -12°C Meat which gives k 0.47 W/m.°C ⎛ 1 ⎞ 2 h = Bi = ⎜ = 9.9 W/m .°C ⎟ L 0.05 m 0.95 ⎝ ⎠ 15°C Therefore, the convection heat transfer coefficient should be kept below this value to satisfy the constraints on the temperature of the steak during refrigeration. 2 Heat transfer is one-dimensional since the plate is large. Analysis The nodal spacing is given to be Δx = 0.02 m. For specified top and bottom surface temperatures, the rate of heat transfer along the cylinders and the temperature drop at the interface are to be determined. Assumptions 1 Heat transfer through the wall is given to be transient and one-dimensional, and the thermal conductivity to be constant. The complete finite difference formulation for the determination of nodal temperatures is to be obtained. 3 Any direct radiation gain or loss through the windows is negligible. Assumptions 1 Heat conduction in the plate is one-dimensional since the plate is large relative to its thickness and there is thermal symmetry about the center plane. It offers the convenience of incorporating the effects of radiation in the convection heat transfer coefficient, and to ignore radiation in heat transfer calculations. 4 The pipe is at the same temperature as the hot water. Analysis The lowest temperature in the steak will occur at the surfaces and the highest temperature at the center at a given time since the inner part of the steak will be last place to be cooled. °F) 2-48 Chapter 2 Heat Conduction Equation 2-89 Heat is generated uniformly in a spherical radioactive material with specified surface temperature. (c) Using Eq. (1), the heat flux on the surface of the cylinder at r = r0 is determined from its definition to be q& s = −k ⎡ g&r 2 dT (r0) = −k ⎢ 0 dr ⎢⎣ k ⎡ g&r 2 ⎛ 2r ⎞ ⎤ ⎥ ⎥ − ⎢ 0 k = − ⎢ r 2 ⎥ ⎥ ⎢⎣ k ⎝ 0 ⎠ ⎦ r = r0 ⎛ 2r0 ⎞ − ⎢ r2 ⎝ 0 2-39 k g r D ⎞⎟ ⎥ = 2 g&r0 = 2(35 W/cm 3)(4 cm) = 280 W/cm 2 ⎥ ⎠ ⎦ Chapter 2 Heat Conduction Equation 2-79 "GIVEN" r_0=0.04 "[m]" k=25 "[W/m-C]" g_dot_0=35E+6 "[W/m^3]" T_s=80 "[C]" "ANALYSIS" T=(g_dot_0*r_0^2)/k*(1-(r/r_0)^2)+T_s "Variation of temperature" "r is the parameter to be varied" r [m] 0 0.004444 0.008889 0.01333 0.01778 0.02222 0.02667 0.03111 0.03556 0.04 T [C] 2320 2292 2209 2071 1878 1629 1324 964.9 550.1 80 2500 T [C] 2000 1500 1000 500 0 0 0.005 0.01 0.015 0.02 r [m] 2-40 0.025 0.03 0.035 0.04 Chapter 2 Heat Conduction Equation 2-80E A long homogeneous resistance heater wire with specified convection conditions at the surface is used to boil water. 5-9 surr Chapter 5 Numerical Methods in Heat Conduction 5-24 A uranium plate is subjected to insulation on one side and convection on the other. Assumptions 1 Heat transfer is steady since there is no indication of any change with time. Therefore, taking the outer surface temperature of the plate to be T2 (absolute, in R), T −T x L kAs 1 2 = εσAsT24 − α s As q&solar L Canceling the area A and substituting the known quantities, (520 R) − T2 (1.2 Btu/h ⋅ ft ⋅ °F) = 0.8(0.1714 × 10 −8 Btu/h ⋅ ft 2 ⋅ R 4)T24 − 0.45(300 Btu/h ⋅ ft 2) 0.5 ft T2 = 530.9 R Solving for T2 gives the outer surface temperature to be Then the rate of heat transfer through the wall becomes T −T (520 − 530.9) R q& = k 1 2 = (12 . Properties The thermal conductivity, density, and specific heat of the balls are given to be k = 54 W/m.°C, ρ = 7833 kg/m3, and Cp = 0.465 kJ/kg.°C. Properties The thermal conductivities of the leather and synthetic fabric are given to be k = 0.159 W/m·°C and k = 0.13 W/m·°C, respectively. 2-117C A differential equation is said to have constant coefficients if the coefficients of all the terms which involve the dependent variable or its derivatives are constants. × 108 kJ / yr) / 0.78 = 2.197 × 108 kJ / yr = 2082 therms Annual Cost = Qin,ins × Unit cost = (2082 therm / yr)($0.50 / therm) = $1041 / yr Cost savings = Energy cost w/o insulation - Energy cost w/insulation = 9263 - 1041 = $8222/yr The unit cost of insulation is given to be $10/m2 per cm thickness, plus $30/m2 for labor. Wood subfloor (R = 0.166 m2.°C/W) 6a. energies 10,000 kJ/h or -Qout = [mC(T2 - T1)]water Substituting, -240,000 kJ = (1000 kg)(4.18 kJ/kg·°C)(20 - T1) It gives T1 = 77.4°C where T1 is the temperature of the water when it is first brought into the room. Therefore, we must specify both direction and magnitude in order to describe heat transfer completely at a point. 3 The combined heat transfer coefficient is constant and uniform over the entire surface. × 109 kJ / yr) / 0.78 = 1.954 × 109 kJ / yr = 18,526 therms / yr since 1 therm = 105,500 kJ. Under the stated assumptions, the energy balance on the system can be expressed as E −E 1in424out 3 ΔE system 1 424 3 = Net energy transfer by heat, work, and mass Change in internal, kinetic, potential, etc. Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be hp = kAc a= hπD = kπD 2 / 4 4h = kD 4(35 W / m2 . Liquids have higher dynamic viscosities than gases. Comprehensive problems designated with the computer-EES icon [pick one of the four given] are solved using the EES software, and their solutions are placed at the Instructor Manual section of the Online Learning Center (OLC) at www.mhhe.com/cengel. The fraction of the hot water cost that is due to the heat loss from the tank and the payback period of the do-it-yourself insulation kit are to be determined. 7-71 Chapter 7 External Forced Convection 7-81 The thickness of flat R-8 insulation in SI units is to be determined when the thermal conductivity of the material is known. The mathematical formulation, the variation of temperature, and the rate of heat transfer are to be determined for steady one-dimensional heat transfer. 6-14C A fluid in direct contact with a solid surface sticks to the surface and there is no slip. Column 1 contains the time, column 2 the value of T[1], column 3, the value of T[2], etc., and column 7 the Row." Time=TableValue('Table 1',Row-1,#Time)+DELTAt Duplicate i=1,5 T_old[i]=TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the six equations for the six unknown temperatures are determined to be" h_o*(T_infinity-T_old[1])+k*(T_old[2]T_old[1])/DELTAx=RhoC*DELTAx/2*(T[1]-T_old[1])/DELTAt "Node 1, convection" T[2]=tau*(T_old[1]+T_old[3])+(1-2*tau)*T_old[2] "Node 2" T[3]=tau*(T_old[2]+T_old[4])+(1-2*tau)*T_old[3] "Node 3" h_i*(T_old[5]-T_old[4])+k*(T_old[3]T_old[4])/DELTAx=RhoC*DELTAx/2*(T[4]-T_old[4])/DELTAt "Node 4, convection" 5-100 Chapter 5 Numerical Methods in Heat Conduction h_i*A_i*(T_old[4]-T_old[5])=m_air*C_air*(T[5]T_old[5])/DELTAt+m_food*C_food*(T[5]-T_old[5])/DELTAt "Node 5, refrig. Applying the boundary conditions give − g&r1 2 + C1 ln r1 + C 2 4k r = r1: T (r1) = r = r2: T (r2) = − g&r2 2 + C1 ln r2 + C 2 4k Substituting the given values, these equations can be written as 60 = − (37,894)(0.15) 2 + C1 ln(0.15) + C 2 4(20) 80 = − (37,894)(0.20) 2 + C1 ln(0.20) + C 2 4(20) Solving for C1 and C2 simultaneously gives C1 = 98.34 C 2 = 257.2 Substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = − 37,894r 2 + 98.34 ln r + 257.2 = 257.2 − 473.68r 2 + 98.34 ln r 4(20) 2-76 Chapter 2 Heat Conduction Equation The temperature at the center surface of the pipe is determined by setting radius r to be 17.5 cm, which is the average of the inner radius and outer radius. Then from Fig. The R-value of the ceiling is to be determined for winter conditions. The hydraulic diameter is defined such that it reduces to ordinary diameter D for circular tubes since Dh = 4 Ac 4πD 2 / 4 = =D. Assumptions 1 Heat transfer through the wall is given to be steady and one-dimensional, and the thermal conductivity and heat generation to be variable. 3 Heat generation is uniform. ° C / W) = 0.0214 m = 2.14 cm Therefore, the interface between the two plates offers as much resistance to heat transfer as a 2.14 cm thick copper. m) 2 = 30.19 m 2 1 1 Ro = = = 0.000946 °C/W 2 ho A (35 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 − r1 (1.55 − 1.5) m = = 0.0489 °C/W 4πkr1 r2 4π (0.035 W/m.°C)(1.55 m)(1.5 m) Rtotal = Ro + Rinsulation = 0.000946 + 0.0489 = 0.0498 °C/W T −T [15 − (−183)]° C Q& = s1 ∞ 2 = = 3976 W Rtotal 0.0498 ° C / W 3.976 kJ / s Q& & fg → m& = = = 0.0187 kg / s Q& = mh h fg 213 kJ / kg (c) The heat transfer rate and the rate of evaporation of the liquid with a 2-cm superinsulation is A = πD 2 = π (3.04 m) 2 = 29.03 m 2 1 1 Ro = = = 0.000984 °C/W 2 ho A (35 W/m .°C)(29.03 m 2) Rinsulation = Ts1 Rinsulation r2 − r1 (1.52 − 1.5) m = = 13.96 °C/W 4πkr1 r2 4π (0.00005 W/m.°C)(1.52 m)(1.5 m) Rtotal = R o + Rinsulation = 0.000984 + 13.96 = 13.96 °C/W 3-68 Ro T∞2 Chapter 3 Steady Heat Conduction T −T [15 − (−183)]° C Q& = s1 ∞ 2 = = 14.18 W 13.96 ° C / W Rtotal 0.01418 kJ / s Q& & fg Q& = mh → m& = = = 0.000067 kg / s 213 kJ / kg h fg 3-69 Chapter 3 Steady Heat Conduction Critical Radius Of Insulation 3-83C In a cylindrical pipe or a spherical shell, the additional insulation increases the conduction resistance of insulation, but decreases the convection resistance of the surface because of the increase in the outer surface area. The energy balance on the system can be expressed as E −E 1in424out 3 Net energy transfer by heat, work, and mass = ΔE system 1 424 3 Change in internal, kinetic, potential, etc. 3-7C The convection and the radiation resistances at a surface are parallel since both the convection and radiation heat transfers occur simultaneously. Properties The thermal properties of the water is closely approximated by those of water at room temperature, k = 0.607 W/m.°C and α = k / ρC p = 0.146×10-6 m2/s (Table A-9). 3 Thermal conductivity of the soil is constant. The finite difference formulation of the problem for all nodes is to be obtained, and the temperature of the tip of the flange as well as the rate of heat transfer from the exposed surfaces of the flange are to be determined. Assuming no convection and steady one-dimensional conduction in the radial direction, the radiation boundary condition on the outer surface of the shell can be expressed as −k ε dT (r2) 4 = εσ T (r2) 4 − Tsurr dr k r1 r2 Tsurr 2-44 A spherical container consists of two spherical layers A and B that are at perfect contact. Then, Overall U-factor, U = Σfarea,iUi = 0.82×1.085+0.18×1.035 1.035 W/m2.°C Overall unit thermal resistance, R = 1/U 0.967 m2.°C/W (c) Two-reflective surface, ε 1 = ε 2 = 0.05 → ε effective = 1 1 = = 0.03 1 / ε 1 + 1 / ε 2 − 1 1 / 0.05 + 1 / 0.05 − 1 In this case we replace item 6a from 0.16 to 0.49 m2.°C/W. energies We ,in − Wb = ΔU We,in = ΔH = m(h2 − h1) ≅ mC p (T2 − T1) or, W&e,in Δt = mC p , ave (T2 − T1) 4×5×6 m3 7°C The mass of air is V = 4 × 5 × 6 = 120 m3 PV (100 kPa)(120 m3) m= 1 = = 149.3 kg RT1 (0.287 kPa ⋅ m3 / kg ⋅ K)(280 K) We Using Cp value at room temperature, the power rating of the heater becomes W& = (149.3 kg)(1.007 kJ/kg⋅ o C)(25 − 7) o C/(15 × 60 s) = 3.01 kW e,in 1-10 AIR Chapter 1 Basics of Heat Transfer 1-30 A room is heated by the radiator, and the warm air is distributed by a fan. 2-73C Heat generation in a solid is simply conversion of some form of energy into sensible heat energy. 1-28 25°F Chapter 1 Basics of Heat Transfer 1-67 The thermal conductivity of a material is to be determined by ensuring onedimensional heat conduction, and by measuring temperatures when steady operating conditions are reached. 4 − 15b) Tsurface = T∞ + 0.45(To − T∞) = 25 + 0.45(40 − 25) = 31.8º F The slight difference between the two results is due to the reading error of the charts. Analysis (a) It is given that D = 0.016 m, SL = ST = 0.05 m, and V = 5.2 m/s. 7-79 Chapter 7 External Forced Convection 7-87E Steam is flowing through an insulated steel pipe, and it is proposed to add another 1-in thick layer of fiberglass insulation on top of the existing one to reduce the heat losses further and to save energy and money. 4-23 we have Wall 2 -6 2 ⎤ h αt (20 W/m .°C) (1.6 × 10 m / s)(2 × 3600 s) = 2.98 ⎥ = ⎥ T − T∞ 0.72 W/m.°C k = 0.038 ⎬1 − 0.3 m x ⎥ Ti − T∞ ξ= = 1.40 ⎥ 2 αt 2 (1.6 × 10 -6 m 2 / s)(2 × 3600 s) ⎦ 1− 0.038 → T = 17.4°C 18 − 2 For a 40 cm distance from the outer surface, that is for the inner surface, from Fig. Properties Assuming a film temperature of Tf = 80°F, the properties of air are evaluated to be (Table A-15E) k = 0.01481 Btu/h.ft.°F Air V∞ = 60 mph T∞ = 50°F Minivan υ = 0.1697 × 10 ft /s -3 2 Pr = 0.7290 Analysis Air flows along 11 ft long side. The amount of heat the iron dissipates in 2 h, the heat flux on the surface of the iron base, and the cost of the electricity are to be determined. Properties The thermal conductivities are given to be k = 223 Btu/h·ft·°F for copper tube and be k = 0.5 Btu/h·ft·°F for the mineral deposit. ° C)(2πr3 m 2) 188.5r3 Then the rate of average heat transfer from the water can be expressed as Ti ,ave − To Q& = Rtotal → 1.694 W = [7.5 − (−10)]° C [0.0948 + 4.55 ln(r3 / 0.033) + 1 / (188.5r3)]° C / W → r3 = 0.312 m Therefore, the minimum thickness of fiberglass needed to protect the pipe from freezing is t = r3 - r2 = 0.312 - 0.033 = 0.279 m which is too large. 1-52C Emissivity is the ratio of the radiation emitted by a surface to the radiation emitted by a blackbody at the same temperature. Note that the heat transfer area A depends on r in this case, and thus it varies with location. In rooms with high ceilings, ceiling fans are used in winter to force the warm air at the top downward to increase the air temperature at the body level. 4 - 15a) Chapter 4 Transient Heat Conduction 1 k ⎤ = = 0.877⎥ Bi hr0 ⎥ T (r) − T∞ = 0.6 (Fig. Nodes 1, 2, 3, and 4 are interior nodes, and thus for them we can use the general finite difference relation expressed as Tm−1 − 2Tm + Tm+1 + gm + = 0, for m = 0, 1, 2, 3, and 4 Δx 2 Finally, the finite difference equation for node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 and taking the direction of all heat transfers to be towards the node under consideration: Node 0 (Left surface - insulated) : T1 − 2T0 + T1 Δx T0 − 2T1 + T2 + g k0 = 0 Node 1 (interior) : T0 − 2T1 + T2 + g ⋅ Δx 2 = 0 Node 2 (interior) : T1 − 2T2 + T3 + g ⋅ Δx 2 = 0 Node 3 (interior) : T2 − 2T3 + T4 + g ⋅ Δx 2 = 0 Node 4 (interior) : T3 − 2T4 + T5 + g ⋅ Δx 2 = 0 Node 5 (right surface - convection) : hA(T∞ − T5) + kA T4 − T5 + g A(Δx / 2) = 0 Δx Node 1 (interior) : 2 ⋅ ⋅ ⋅ 4 5 h, T∞ where Δx = 0.01 m, g = 4 × 103 kW/m³, k = 28 W/m⋅°C, h = 60 W/m²⋅°C, and T∞ = 30°C. Properties The R-values of different materials are given in Table 3-6. Radiation heat transfer is significant, energies Qin = AU → Qin = m(u2 − u1) = mCv (T2 − T1) The final temperature of air is PV PV 1 = 2 → T1 T2 T2 = T1 P2 = 2 × (540 R) = 1080 R P1 The specific heat of air at the average temperature of Tave = (540+1080)/2= 810 R = 350°F is Cv,ave = Cp,ave − R = 0.2433 − 0.06855 = 0.175 Btu/lbm.R. Substituting, Q = (20 lbm) (0.175 Btu/lbm.R)(1080 − 540) R = 1890 Btu ...

2257 kJ/kg · °C Discussion The arithmetic mean fluid temperature is (Ti + Te)/2 = (20 + 43.4)/2 = 31.7°C, which is fairly close to the assumed value of 35°C. e− (2.027) 2 τ → τ = 0.456 37 − (−6) Beef 37°C which is greater than 0.2 and thus the one-term solution is applicable. Separating the variables in the above equation and integrating from r = r1 where T (r1) = T1 to any r where T (r) = T , we get Q& ∫ r r1 dr = −4π r2 ∫ T k (T)dT T1 Substituting k (T) = k0 (1 + βT) and performing the integrations gives ⎛ 1 1⎞ Q& ⎜ − ⎟ = −4πk 0 [(T − T1) + β (T 2 − T12) / 2] ⎝ r1 r ⎠ Substituting the Q& expression from part (a) and rearranging give T2 + 2 β T+ 2 k ave r2 (r − r1) 2 (T1 − T2) − T12 − T1 = 0 βk 0 r (r2 − r1) β which is a quadratic equation in the unknown temperature T. The heat of fusion of ice at 0°C is 333.7 kJ/kg. 3 The pressure of air is 1 atm. 2 Heat transfer is onedimensional since the plate is large relative to its thickness. 2 q0 Convection heat transfer at the right surface is negligible. The surface temperature of the component is to be determined. In the limiting case, the surface temperature at x = L = 5 cm from the center will be -1°C while the mid plane temperature is 5°C in an environment at -12°C. CENGEL SECOND EDITION Preface This manual is prepared as an aide to the instructors in correcting homework assignments, but it can also be used as a source of additional example problems for use in the classroom. Evacuating the interface eliminates heat transfer by conduction, and thus increases the thermal contact resistance. C)(2.4 m 2) Rtotal = Rconv,1 + 2 R1 + R2 + Rconv,2 = 0.0417 + 2(0.0016) + 0.1923 + 0.0167 Ri = Rconv,1 = Air = 0.2539 °C/W The steady rate of heat transfer through window glass then becomes T −T [24 − (−5)]°C Q& = ∞1 ∞ 2 = = 114 W R1 R2 Ri Rtotal 0.2539°C/W T∞1 The inner surface temperature of the window glass can be determined from T −T Q& = ∞1 1 → T1 = T∞1 − Q& Rconv,1 = 24 o C − (114 W)(0.0417°C/W) = 19.2°C Rconv,1 3-4 R3 Ro T∞2 Chapter 3 Steady Heat Conduction 3-20 A double-pane window consists of two 3-mm thick layers of glass separated by an evacuated space. 3 The thermal properties of the ice block are (ro, L) constant. × 10 −6 m2 / s)(60 min × 60 s / min) (0.025 m) 2 ⎨ 2 T (0,0,0, t) − 500 = (1.0580)e −(0.5932) (6.624) 20 − 500 = 6.624 > 0.2 ⎬ = 0.00109 → T (0,0,0, t) = 500°C 3 Note that τ > 0.2 in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable. m2) 2 = 0.0174 m2 . (c) The temperature at the insulate surface (x = 0) is determined by substituting the known quantities to be T (x) = T2 + g& 0 L2 [4(e − 0.5 − e 0) + (2 − 0 / L)] k (8 × 10 6 W/m 3)(0.05 m) 2 = 30°C + [4(e −0.5 − 1) + (2 − 0)] = 314.1°C (30 W/m · °C) T (0) = T 2 + Therefore, there is a temperature difference of almost 300°C between the two sides of the plate. Properties The conductivity and diffusivity are given to be k = 0.61 W/m.°C and α = 015 . Properties The thermal conductivities are given to be k = 15 W/m·°C for steel and k = 0.038 W/m·°C for glass wool insulation Analysis The inner and the outer surface areas of the insulated pipe per unit length are Ai = πDi L = π (0.05 m)(1 m) = 0157 . The mathematical formulation, and an expression for the outer surface temperature and its value are to be determined for steady one-dimensional heat transfer. Analysis (a)The shape factor for this configuration is given in Table 3-5 to be D =2.5 cm 2π (5 m) 2πL = 16 m S= = ⎛ 8z ⎞ ⎡ 8(0.07 m) ⎤ 60°C ln⎜ ⎟ ln ⎢ ⎥ ⎜ (0 . Properties The thermal conductivity is given to be k = 45 W/m·°C. 3 Thermal conductivities of the glass and air are constant. Then the pressure drop across the tube bank becomes ΔP = N L f χ 2 ρVmax (1.204 kg/m 3)(6.552 m/s) 2 = 8(0.34)(1) 2 2 ⎛ 1N ⎞ ⎜ 1 kg · m/s 2 ⎝ ⎞ ⎟ = 70.3 Pa ⎟ ⎠ Discussion The arithmetic mean fluid temperature is (Ti + Te)/2 = (15 +28.3)/2 = 21.7°C, which is fairly close to the assumed value of 20°C. Using the quadratic formula, the temperature distribution T(r) in the cylindrical shell is determined to be T (r) = − 1 β ± 1 β 2 − 2 k ave ln(r / r1) 2 (T1 − T2) + T12 + T1 βk 0 ln(r2 / r1) β Discussion The proper sign of the square root term (+ or -) is determined from the requirement that the temperature at any point within the medium must remain between T1 and T2 . 6 The heat transfer coefficient accounts for the effect of radiation from the spoon.. Properties The thermal conductivity of the spoon is given to be k = 247 Btu/h·ft·°F. Properties The thermal conductivity and thermal diffusivity of potatoes are given to be k = 0.50 W/m·°C and α = 0.13×10-6 m2/s. 6-49 A flat plate is subjected to air flow, and the drag force acting on it is measured.) + (18 . ° C)(0.0216 m2) 1 1 = = = 0.9259 ° C / W hA (50 W / m2 . Analysis (a) We treat the flanges as fins. Using the definition of the dimensionless αΔt the last equation reduces to mesh Fourier number τ = (Δx) 2 Tmi −1 − (2 + hΔx / k)Tmi + Tmi +1 + g& Δx 2 Tmi +1 − Tmi hΔx T∞ + 0 = k k τ Discussion We note that setting Tmi +1 = Tmi gives the steady finite difference formulation. 5 The given time step Δt = 15 min is less than the critical time step so that the stability criteria is satisfied. The velocity and the surface area are ⎛ 1 min ⎞ V = πDN& = π(0.06 m)(3000 rev/min)⎜ ⎟ = 9.425 m/s ⎝ 60 s ⎠ A = πDLbearing = π(0.06 m)(0.20 m) = 0.0377 m 2 The maximum temperature is Tmax = T (L) = T1 + μV 2 kL 2 2 2 ⎛ ⎜ L − L ⎟ = T1 + μV ⎛⎜1 − 1 ⎞⎟ = T1 + μV ⎜ 2 L ⎟ ⎜2k k ⎝ 2⎠ (0.05 N · s/m 2)(9.425 m/s) 2 ⎛ 1 W ⎞ ⎟ = 63.1°C 2(0.17 W/m · °C) ⎝ 1 N · m/s ⎠ (b) The rate of heat transfer to the bearing is dT μV 2 μV 2 (1 − 0) = − = −kA Q& 0 = −kA dy y =0 kL L = 50°C + (0.05 N · s/m 2)(9.425 m/s) 2 ⎛ 1 W ⎞ ⎟ = −837 W 0.0002 m ⎝ 1 N · m/s ⎠ The rate of heat transfer to the shaft is zero. h° F / Btu Rtotal = Rtotal + Rconv = 13473 T T − Q (250 − 70) ° F 50.8 Btu ∞ Q& = s = = 112.4 Btu/h Δt = = 0.452 h = 27.1 min R total 1.6012 h°F/Btu Q& 112.4 Btu/h This result is conservative since the heat transfer coefficient will be lower in this case because of the smaller exposed surface temperature. Properties The properties of air at the film temperature of (35 + 25) / 2 = 30 °C are (Table A-15) V∞ = 30 km/h T∞ = 25°C k = 0.02588 W/m.°C Tsky = -40°C 700 W/m2 υ = 1.608 × 10 -5 m 2 /s Solar collector Pr = 0.7282 Ts = 35°C Analysis (a) Assuming wind flows across 2 m surface, the Reynolds number is determined from Re L = V∞ L υ = (30 × 1000 / 3600)m/s(2 m) 1.608 × 10 −5 2 m /s L=2m = 1.036 × 10 6 which is greater than the critical Reynolds number (5 × 10 5) . Analysis (See Figure 5-24 in the text). 4 The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the oxygen inside, and thus thermal contact resistance between the tank and the internal convection resistance are negligible. 6-22C The friction coefficient represents the resistance to fluid flow over a flat plate. Felt (R = 0.011 m2.°C/W) 4. Analysis (a) The total rate of heat transfer dissipated by the chips is Q& = 80 × (0.04 W) = 3.2 W 2 cm The individual resistances are Repoxy Rboard RAluminum Rconv T1 T∞2 T2 A = (0.12 m) (0.18 m) = 0.0216 m 2 L 0.003 m = = 0.00694 ° C / W kA (20 W / m. Assumptions 1 Heat transfer is one-dimensional since the walls are large relative to their thickness. Properties The thermal conductivity of the glass is given to be k = 0.78 W/m·°C. The Biot numbers, the corresponding constants, and the Fourier numbers are hL (13 W/m 2 .°C)(0.02 m) = = 0.1171 → λ 1 = 0.3318 and A1 = 10187 . For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are As = NπDL = 128π(0.021 m)(1 m) = 8.445 m 2 m& = m& i = ρi V(N T ST L) = (0.6158 kg/m 3)(4.5 m/s)(8)(0.08 m)(1 m) = 1.773 kg/s Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become ⎛ Ah Te = Ts − (Ts − Ti) exp⎜ − s ⎜ m& C p ⎝ ΔTln = 2 2 ⎞ ⎟ = 80 − (80 − 300) exp⎜ − (8.445 m)(73.2 W/m · °C) ⎟ = 237.0°C ⎜ (1.773 kg/s)(1033 J/kg · °C) ⎟ ⎟ ⎠ (Ts − Ti) − (Ts − Te) (80 − 300) − (80 − 237) = = 186.7°C ln[(Ts − Ti) /(Ts − Te)] ln[(80 − 300) /(80 − 237)] 7-63 Chapter 7 External Forced Convection Q& = hAs ΔTln = (73.2 W/m 2 · °C)(8.445 m 2)(186.7°C) = 115,425 W (b) For this in-line arrangement tube bank, the friction coefficient corresponding to ReD = 3132 and SL/D = 8/2.1 = 3.81 is, from Fig. Assuming the direction of heat conduction to be towards the node under consideration at all surfaces, the energy balance on the volume element can be expressed as ΔE element =0 Q& cond, left + Q& cond, top + Q& cond, right + Q& cond, bottom + G& element = Δt for the steady case. Properties The properties of the junction are given to be k = 35 W / m. Assumptions 1 Heat transfer is one-dimensional since the pond is large relative to its depth. We observe that this is a steady-flow process since there is no change with time at any point and thus ΔmCV = 0 and ΔE CV = 0 . 2-54C Yes, in the case of constant thermal conductivity and no heat generation, the temperature in a solid cylindrical rod whose ends are maintained at constant but different temperatures while the side surface is perfectly insulated will vary linearly during steady one-dimensional heat conduction. 5 The pressure of air is 1 atm. Analysis (a) The nodal spacing is given to be Δx=Δx=l=0.1 m, and all nodes are boundary nodes. 2 Heat conduction in the carcass is one-dimensional in the radial direction because of the symmetry about the centerline. Assumptions 1 Heat conduction in the hot dog is one-dimensional since it is long and it has thermal symmetry about the center line. The numerical solution methods are based on replacing the differential equations by algebraic equations. m2 Ao = πDo L = π (0.046 m)(15 m) = 2.168 m Ri 2 Rpipe Ro T∞1 T∞2 1 1 = = 0.0044 °C/W hi Ai (120 W/m 2 .°C)(1.885 m 2) ln(r2 / r1) ln(2.3 / 2) = = = 0.0000038 °C/W 2πkL 2π(386 W/m.°C)(15 m) Ri = R pipe The outer surface temperature of the pipe will be somewhat below the water temperature. Then the number of nodes becomes M = L / Δx + 1 = 0.03/0.01+1 = 4. 210 kJ/min Immersion chilling, 0.5°C 15°C 3°C Analysis (a) Chickens are dropped into the chiller at a rate of 500 per hour. We would place the origin at the center of the egg. Concrete block, lightweight, 100 mm 5. It is to be determined whether his/her result is reasonable. 3 There is no heat generation in the wall. Analysis (a) The amount of beef mass that needs to be cooled per unit time is Lights, 2 m& beef = (Total beef mass cooled)/(cooling time) k = (350× 280 kg/carcass)/(12h × 3600 s) = 2.27 kg/s The product refrigeration load can be viewed as the energy that needs to be removed from the beef carcass as it is cooled from 35 to 16ºC at a rate of 2.27 kg/s, and is determined to be Q& beef =(m& C p ΔT) beef = (2.27 kg/s)(3.14 kJ/kg.º C)(35 − 16)º C = 135kW 11kW Beef 35°C 280 kg Fans, 22 k Then the total refrigeration load of the chilling room becomes Q& total, chilling room = Q& beef + Q& fan + Q& lights + Q& heat gain = 135 + 22 + 2 + 11 = 170 kW (b) Heat is transferred to air at the rate determined above, and the temperature of air rises from -2.2ºC to 0.5ºC as a result. 3-96C Fins enhance heat transfer from a surface by increasing heat transfer surface area for convection heat transfer. The average friction coefficient of the airfoil is determined from the modified Reynolds analogy to be Cf = 2(1.335 W/m 2 · °C)(0.7296) 2 / 3 2hPr 2/3 = = 0.0001512 ρV C p (1.184 kg/m 3)(12 m/s)(1007 J/kg · °C) 6-26 Chapter 6 Fundamentals of Convection 6-52 The windshield of a car is subjected to parallel winds. Analysis The total surface area of the refrigerator where heat transfer takes place is Atotal = 2 (18 . Properties The thermal conductivity of the ground is given to be k = 1.5 W/m·°C. Therefore critical radius of insulation will be greater on calm days. The steady finite difference formulation of the problem is to be obtained, and the unknown nodal temperatures as well as the rate of heat transfer to the iced water are to be determined. A tennis ball is a blunt body (unless the velocity is very low and we have “creeping flow”). 3 Heat losses from the water in the tank are negligible. 2 The thermal properties of the hot dog are constant. The total amount of heat transfer needed to defrost the steaks is msteak = ρV = (970 kg / m 3)[π (0.075 m) 2 (0.015 m)] = 0.257 kg Qtotal, steak = Qsensible + Qlatent = (mCΔT) steak + (mhif) steak = (0.257 kg)(1.55 kJ / kg. Analysis A short cylinder can be formed by the intersection of a long cylinder of radius D/2 = 14 cm and a plane wall of thickness 2L = 180 cm. The Biot numbers and the corresponding constants are first determined to be Bi = hL (80 W/m 2 .°C)(0.1 m) = = 0.0339 k (236 W/m.°C) Bi = hr0 (80 W/m 2 .°C)(0.075 m) = = 0.0254 → λ1 = 0.2217 and A1 = 1.0063 k 236 W/m.°C → λ1 = 0.1811 and A1 = 1.0056 Noting that τ = αt / L2 and assuming τ > 0.2 in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable, the product solution for this problem can be written as θ(0,0, t) block = θ(0, t) wall θ(0, t) cyl = ⎛⎜ A1 e −λ1 τ ⎞⎟⎛⎜ A1e −λ1 τ ⎞⎟ ⎝ ⎠ wall ⎝ ⎠ cyl 2 2 −5 ⎤ ⎤ ⎡ ⎡ (9.75 ×10 −5)t ⎤ ⎤ ⎡ ⎡ 300 − 1200 ⎡ ⎡ 2 (9.75 × 10)t ⎥ = ⎨(1.0056) exp ⎢− (0.1811) 2 (1 . Btu / h · ft·° F)(25 × 35 ft 2) = 28,875 Btu / h 0.8 ft L Discussion The positive sign indicates that the direction of heat transfer is from the inside to the outside. Properties The conductivity and diffusivity are given to be k = 0.48 Btu/h.ft·°F and α = 4.2 × 10 −6 ft 2 / s . Then the number of nodes M becomes M = L 0. Here T is the temperature, r is the space ⎜r ⎟+ = r ∂r ⎝ ∂r ⎠ k α ∂t variable, g is the heat generation per unit volume, k is the thermal conductivity, α is the thermal diffusivity, and t is the time. The center temperature and the heat transfer per unit length of the cylinder are to be determined. This relation can be modified for the three-dimensional case by simply adding another index j to the temperature in the z direction, and another difference term for the z direction as Tm −1, n, j − 2Tm , n, j + Tm +1, n, j Tm, n −1, j − 2Tm, n, j + Tm, n +1, j Tm, n, j −1 − 2Tm, n, j + Tm, n , j +1 g& m, n , j + + + =0 k Δx 2 Δy 2 Δz 2 5-2 Chapter 5 Numerical Methods in Heat Conduction 5-9 A plane wall with variable heat generation and constant thermal conductivity is subjected to uniform heat flux q& 0 at the left (node 0) and convection at the right boundary (node 4). 3 The thermal properties of the steaks are constant. 3-109 A commercially available heat sink is to be selected to keep the case temperature of a transistor below 80°C in an environment at 35 °C . Analysis The nodal spacing is given to be Δx=Δx=l=0.05 m, and the general finite difference form of an interior node for steady two-dimensional heat conduction is expressed as • • • • g& l 2 Tleft + Ttop + Tright + Tbottom − 4Tnode + node = 0 k 350 260 305 where • • • 200°C • g& node l 2 g& 0 l 2 (8 × 10 6 W/m 3)(0.05 m) 2 g = = = 93.5°C Insulated k k 214 W/m · °C 3 290 2 • • • • The finite difference equations for boundary nodes are obtained by 5 cm applying an energy balance on the volume elements and taking the 325 1 direction of all heat transfers to be towards the node under • • 240 • • consideration: Convection 2 290 − T1 l 240 − T1 l 325 − T1 h,g& Tl + kl +k + hl (T∞ − T1) + 0 ∞ = 0 Node 1 (convection) : k l l l 2k 2 2 &g 0 l 2 Node 2 (interior) : 350 + 290 + 325 + 290 - 4 T2 + =0 k g& l 2 Node 3 (interior) : 260 + 290 + 240 + 200 - 4T3 + 0 = 0 k k = 45 W/m.°C, h = 50 W/m 2 .°C, g& = 8 ×10 6 W/m 3 , T∞ = 20°C where Substituting, T1 = 280.9°C, T2 = 397.1°C, T3 = 330.8°C, (b) The rate of heat loss from the bottom surface through a 1-m long section is Q& = ∑ Q& m element, m = ∑ hA surface,m (Tm − T∞) m = h(l / 2)(200 − T∞) + hl (240 − T∞) + hl (T1 − T∞) + h(l / 2)(325 − T∞) = (50 W/m 2 · °C)(0.05 m × 1 m)[(200 - 20)/2 + (240 - 20) + (280.9 - 20) + (325 - 20)/2]°C = 1808 W 5-38 Chapter 5 Numerical Methods in Heat Conduction 5-47 A long solid body is subjected to steady two-dimensional heat transfer. ° F) Water 202°F The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, Hot dog λ1 = 1.5421 and A1 = 1.2728 r x The Fourier number is τ= αt L2 = (0.0077 ft 2 / h)(5 / 60 h) (2.5 / 12 ft) 2 = 0.015 < 0.2 (Be cautious!) Then the dimensionless temperature at the center of the plane wall is determined from θ o,wall = 2 2 T0 − T∞ = A1e −λ1 τ = (1.2728)e −(1.5421) (0.015) ≅ 1 Ti − T∞ We repeat the same calculations for the long cylinder, Bi = hro (120 Btu / h.ft 2 . Analysis (a) Oil flow in a journal bearing can be approximated as parallel flow between two large plates with one plate moving and the other stationary. 8-9C The friction factor f remains constant along the flow direction in the fully developed region in both laminar and turbulent flow. Properties The thermal properties of body are given to be k = 0.62 W/m.°C and α = 0.15×10-6 m2/s. The density and specific heat of water are to be ρ = 1000 kg/m3 and Cp = 4.18 kJ/kg.°C (Table A-15). Discussion Heat transfer from the top and bottom surfaces is small relative to that transferred from the side surface. Assumptions 1 The temperature of the submerged portion of the spoon is equal to the water temperature. Properties The thermal conductivity of the aluminum pan is given to be k = 237 W/m·°C. The finite difference equation for node 4 subjected to heat flux is obtained from an energy balance by taking the direction of all heat transfers to be towards the node: Node 0 (insulation) : T0i +1 = τ (T1i + T1i) + (1 − 2τ)T0i + τg& 0 (Δx) 2 / k Node 0 (insulation) : T1i +1 = τ (T0i + T2i) + (1 − 2τ)T1i + τg& 1 (Δx) 2 / k Node 2 (interior) : T2i +1 = τ (T1i + T3i) + (1 − 2τ)T2i + τg& 2 (Δx) 2 / k Node 3 (interior) : T3i +1 = τ (T2i + T4i) + (1 − 2τ)T3i + τg& 3 (Δx) 2 / k i +1 i T3i − T4i Δx T4 − T4 + τg& 4 (Δx) 2 / k = ρ C Δx 2 Δt −6 2 . 2 There is no heat generation within the damn. The individual thermal resistances are Ao = πDo L = π (0.46 m)(2 m) = 2.89 m2 1 1 = = 0.029 °C/W ho Ao (12 W/m 2 .°C)(2.89 m 2) ln(r2 / r1) ln(23 / 20) = = 0.37 °C/W R foam = 2πkL 2π (0.03 W/m 2 .°C)(2 m) Ro = Rfoam Ro Tw T∞2 Rtotal = Ro + R foam = 0.029 + 0.37 = 0.40 °C/W The rate of heat loss from the hot water tank is T − T∞ 2 (55 − 27)° C Q& = w = = 70 W Rtotal 0.40 ° C / W The amount and cost of heat loss per year are Q = Q& Δt = (0.07 kW)(365 × 24 h / yr) = 613.2 kWh / yr Cost of Energy = (Amount of energy)(Unit cost) = (613.2 kWh)($0.08 / kWh) = $49.056 f = $49.056 = 0.1752 = 17.5% $280 If 3 cm thick fiber glass insulation is used to wrap the entire tank, the individual resistances becomes Ao = πDo L = π (0.52 m)(2 m) = 3.267 m 2 1 1 = = 0.026 o C/W 2 o ho Ao (12 W/m . 3 Constant specific heats at room temperature can be used for air, Cp = 1.007 and Cv = 0.720 kJ/kg·K. Analysis In steady operation, the rate of heat loss from the wire equals the rate of heat generation in the wire as a result of resistance heating. Yes, it is worthwhile to use radiant barriers in the attics of homes by covering at least one side of the attic (the roof or the ceiling side) since they reduce radiation heat transfer between the ceiling and the roof considerably. Q& 20 W → h = = = 48.43 W/m 2 .°C Q& = ηhAs (T∞ − T s) ηAs (T∞ − Ts) (1)(0.0118 m 2)(60 - 25)°C Starting from heat transfer coefficient, Nusselt number, Reynolds number and finally free-stream velocity will be determined. 3-99C Fins should be attached to the outside since the heat transfer coefficient inside the tube will be higher due to forced convection. Properties The thermal conductivity and emissivity Tsurr ho, To are given to be k = 1.4 W/m·°C and ε = 0.9. Analysis (a) The most striking aspect of this 1 2 3 4 problem is the apparent symmetry about the • • • • horizontal and vertical lines passing through the midpoint of the chimney. ° F) The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, Water 212°F Hot dog λ1 = 1.5421 and A1 = 1.2728 r x The Fourier number is τ= αt L2 = (0.0077 ft 2 / h)(5 / 60 h) (2.5 / 12 ft) 2 = 0.015 < 0.2 (Be cautious!) Then the dimensionless temperature at the center of the plane wall is determined from θ o,wall = 2 2 T0 − T∞ = A1e −λ1 τ = (1.2728)e −(1.5421) (0.015) ≅ 1 Ti − T∞ We repeat the same calculations for the long cylinder, Bi = hro (120 Btu / h.ft 2 . 7-27a, f = 0.27. 6 The thermal contact resistances at the copper-epoxy interfaces are negligible. 4-102 Chapter 4 Transient Heat Conduction 4-111 The trunks of some dry oak trees are exposed to hot gases. Properties The thermal conductivity of the pipe is given to be k = 0.16 W/m·°C. m)3 = 4.77 kg E Substituting, Etransfer = (4.77 kg)(0.90 kJ / kg. Bi = hro kBi (0.45 W/m.°C)(4.3) → h = = = 22.5 W/m 2 .°C k ro (0.08603 m) (b) The temperature at the surface of the rib is 2 2 T (ro , t) − T∞ sin(λ1 ro / ro) sin(2.49) = A1e − λ1 τ = (1.7402)e − (2.49) (0.1881) λ1 ro / ro Ti − T∞ 2.49 T (ro , t) − 163 = 0.132 → T (ro , t) = 142.1 °C 4.5 − 163 θ (ro , t) sph = (c) The maximum possible heat transfer is Q max = mC p (T∞ − Ti) = (3.2 kg)(4.1 kJ/kg. Properties The properties of AISI stainless steel rods are given to be k = 7.74 Btu/h.ft.°F, α = 0.135 ft2/h. A sketch is included with most solutions to help the students visualize the physical problem, and also to enable the instructor to glance through several types of problems quickly, and to make selections easily. energies WATER Wpw,in = ΔU or, Wpw,in = ΔU iron + ΔU water Iron Wpw,in = [mC (T2 − T1)]iron + [mC (T2 − T1)]water Wpw where mwater = ρV = (1000 kg / m3)(0.08 m3) = 80 kg Wpw = W&pw Δt = (0.2 kJ / s)(25 × 60 s) = 300 kJ Using specific heat values for iron and liquid water and substituting, (300 kJ) = m iron (0.45 kJ/kg · °C)(27 − 90)°C + (80 kg)(4.18 kJ/kg · °C)(27 − 20)°C = 0 miron = 72.1 kg 1-15 Chapter 1 Basics of Heat Transfer 1-37E A copper block and an iron block are dropped into a tank of water. 2217) ⎥ ⎥ ⎬ = 0.7627 ⎬ ⎨ 20 − 1200 ⎢ ⎣ (0.1) 2 (0.075) 2 ⎦ ⎦ ⎦ ⎣ ⎦ ⎦ ⎣ ⎦ ⎣ ⎣ Solving for the time t gives t = 241 s = 4.0 min. 7-79C The primary reasons for insulating are energy conservation, personnel protection and comfort, maintaining process temperature, reducing temperature variation and fluctuations, condensation and corrosion prevention, fire protection, freezing protection, and reducing noise and vibration. If the heating element of the oven is large enough to keep the oven at the desired temperature setting under the presumed worst conditions, then it is large enough to do so under all conditions by cycling on and off. 4-92C Cooling the carcass with refrigerated air is at -10ºC would certainly reduce the cooling time, but this proposal should be rejected since it will cause the outer parts of the carcasses to freeze, which is undesirable. Analysis Disregarding any heat loss through the bottom of the ice chest, the total thermal resistance and the heat transfer rate are determined to be Ai = 2(0.3 − 0.03)(0.4 − 0.06) + 2(0.3 − 0.03)(0.5 − 0.06) + (0.4 − 0.06)(0.5 − 0.06) = 0.5708 m 2 Ao = 2(0.3)(0.4) + 2(0.3)(0.5) + (0.4)(0.5) = 0.74 m 2 0.03 m L = = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = = = 0.07508 °C/W 2 hAo (18 W/m .°C)(0.74 m 2) Rchest = Rconv R total = Rchest + Rconv = 1.5927 + 0.07508 = 1.6678 °C/W T − T∞ (30 − 0)°C Q& = s = 20.99 W R total 1.6678 °C/W Ts Rchest Ice The total amount of heat necessary to melt the ice completely is Q = mhif = (45 kg)(333.7 kJ / kg) = 15,016.5 kJ Then the time period to transfer this much heat to the cooler to melt the ice completely becomes Q 15,016,500 J Δt = = = 715,549 s = 198.8 h = 8.28 days 20.99 kJ/s Q& 3-122 Rconv T∞ Chapter 3 Steady Heat Conduction 3-163 A wall is constructed of two large steel plates separated by 1-cm thick steel bars placed 99 cm apart. energies 18 L/min WATER W& e,in + m& h1 = m& h2 (since Δke ≅ Δpe ≅ 0) W& e,in = m& (h2 − h1) = m& [C (T2 − T1) + v(P2 − P1) ©0] = m& C (T2 − T1) The mass flow rate of water through the pipe is m& = ρV&1 = 1000 kg/m 3 0.018 m 3 /min = 18 kg/min (Therefore, W& e,in)( ) = m& C (T2 − T1) = (18/60 kg/s)(4.18 kJ/kg · °C)(50 − 15)°C = 43.9 kW (b) The average velocity of water through the pipe is determined from V& 0.018 m 3 /min V& V1 = 1 = 2 = = 9.17 m/min A1 πr π(0.025 m)2 1-70 We D = 5 cm Chapter 1 Basics of Heat Transfer 1-126 The heating of a passive solar house at night is to be assisted by solar heated water. °F Person, Ts V∞ = 6 ft/s T∞ = 85°F υ = 0.1809 × 10 ft /s -3 2 Pr = 0.7260 300 Btu/h Analysis The Reynolds number is V D (6 ft/s)(1 ft) Re = ∞ = = 3.317 × 10 4 υ 0.1809 × 10 −3 ft 2 /s D = 1 ft The proper relation for Nusselt number corresponding this Reynolds number is hD 0.62 Re 0.5 Pr 1 / 3 Nu = = 0.3 + 1/ 4 k 1 + (0.4 / Pr) 2 / 3 [ ] ⎡ ⎛ Re ⎞ 5 / 8 ⎤ ⎢1 + ⎜ ⎟ ⎥ ⎢ ⎝ 282,000 ⎠ ⎥ ⎣ ⎦ 4/5 ⎡ 0.62(3.317 × 10 4) 0.5 (0.7260)1 / 3 ⎢ ⎛ 3.317 × 10 4 = 0.3 + 1+ 1/ 4 ⎢ ⎜ 282,000 1 + (0.4 / 0.7260) 2 / 3 ⎣ ⎝ [ ] ⎞ ⎟ ⎠ 5/8 ⎤ 4/5 ⎥ ⎥ ⎦ = 107.84 The heat transfer coefficient is k 0.01529 Btu/h.ft.°F h = Nu = (107.84) = 1.649 Btu/h.ft 2 .°F D (1 ft) Then the average temperature of the outer surface of the person becomes Q& 300 Btu/h = 85°F + = 95.1°F Q& = hAs (Ts − T∞) → Ts = T∞ + hAs (1.649 Btu/h.ft 2 .°F)(18 ft 2) If the air velocity were doubled, the Reynolds number would be V D (12 ft/s)(1 ft) Re = ∞ = = 6.633 × 10 4 − 3 2 υ 0.1809 × 10 ft /s The proper relation for Nusselt number corresponding this Reynolds number is hD 0.62 Re 0.5 Pr 1 / 3 Nu = = 0.3 + 1/ 4 k 1 + (0.4 / Pr) 2 / 3 [ ] ⎡ ⎛ Re ⎞ 5 / 8 ⎤ ⎢1 + ⎜ ⎟ ⎥ ⎢ ⎝ 282,000 ⎠ ⎥ 4/5 ⎡ 0.62(6.633 × 10 4) 0.5 (0.7260)1 / 3 ⎢ ⎛ 6.633 × 10 4 = 0.3 + 1+ 1/ 4 ⎢ ⎜ 282,000 1 + (0.4 / 0.7260) 2 / 3 ⎣ ⎝ [ ] ⎞ ⎟ ⎠ 5/8 ⎤ ⎥ ⎥ ⎦ 4/5 = 165.95 Heat transfer coefficient is k 0.01529 Btu/h.ft.°F h = Nu = (165.95) = 2.537 Btu/h.ft 2 .°F D (1 ft) Then the average temperature of the outer surface of the person becomes Q& 300 Btu/h = 85°F + = 91.6°F Q& = hAs (Ts − T∞) → Ts = T∞ + hAs (2.537 Btu/h.ft 2 .°F)(18 ft 2) 7-41 Chapter 7 External Forced Convection 7-51 A light bulb is cooled by a fan. hA (45 W / m. Laminar and Turbulent Flows 6-20C A fluid motion is laminar when it involves smooth streamlines and highly ordered motion of molecules, and turbulent when it involves velocity fluctuations and highly disordered motion. Properties The thermal properties of the soil are given to be k = 0.35 W/m.°C and α = 0.15×10-6 m2/s. The radius of the interface is ro. Properties The thermal conductivity of copper is given to be k = 386 W/m·°C. 5-64 Chapter 5 Numerical Methods in Heat Conduction Transient Heat Conduction 5-63C The formulation of a transient heat conduction problem differs from that of a steady heat conduction problem in that the transient problem involves an additional term that represents the change in the energy content of the medium with time. Thus we have laminar flow. 3 Heat loss from the teapot is negligible. 4 Steady one-dimensional heat conduction relations can be used due to small thickness of the duct wall. The system of 7 equations with 7 unknowns constitutes the finite difference formulation of the problem. Then the time for the annealing process is determined to be b= hAs 75 W/m 2 .°C h = = = 0.01584 s -1 3 ρC pV ρC p Lc (7833 kg/m)(465 J/kg.°C)(0.0013 m) -1 T (t) − T ∞ 100 − 35 = e −bt → = e −(0.01584 s)t → t = 163 s = 2.7 min Ti − T∞ 900 − 35 The amount of heat transfer from a single ball is m = ρV = ρ πD 3 = (7833 kg / m3) π (0.008 m) 3 = 0.0021 kg 6 6 Q = mC p [T f − Ti] = (0.0021 kg)(465 J / kg. Using the proper relations, the Nusselt number, the heat transfer coefficient, and the heat transfer rate are determined to be hL Nu = = 0.037 Re L 0.8 Pr1 / 3 = 0.037(9.888 × 105)0.8 (0.7228)1 / 3 = 2076 k k 0.02735 W/m.°C h = Nu = (2076) = 70.98 W/m 2 .°C L 0.8 m As = wL = (0.8 m)(0.4 m) = 0.32 m 2 Q& conv = hAs (T∞ − Ts) = (70.98 W/m 2 .°C)(0.32 m 2)(80 - 20)°C = 1363 W The radiation heat transfer from the same surface is Q& rad = εAs σ (Ts 4 − Tsurr 4) = (0.95)(0.32 m 2)(5.67 × 10 -8 W/m 2 .K 4)[(80 + 273 K) 4 - (25 + 273 K) 4] = 132 W Then the total rate of heat transfer from that surface becomes Q& = Q& + Q& = (1363 + 132)W = 1495 W total conv rad 7-10 Chapter 7 External Forced Convection 7-21 Air flows on both sides of a continuous sheet of plastic. The warming time of the milk is to be determined. Now we divide • m+1 the x-y-z region into a mesh of nodal points which are spaced Δx, Δy, and Δz apart in the x, y, and z directions, Δz respectively, and consider a general interior node (m, n, r) r • • r+1 whose coordinates are x = mΔx, y = nΔy, are z = rΔz. Also, u = u(y) and v = 0. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be hL Nu = = (0.037 Re L 0.8 − 871) Pr 1 / 3 = [0.037(9.674 × 10 6) 0.8 − 871](0.7282)1 / 3 = 1.212 × 10 4 k k 0.02588 W/m.°C h = Nu = (1.212 × 10 4) = 39.21 W/m 2 .°C L 8m The equilibrium temperature of the top surface is then determined by taking convection and radiation heat fluxes to be equal to each other q& 200 W/m 2 → Ts = T∞ + conv = 30°C + = 35.1°C q& rad = q& conv = h(Ts − T∞) h 39.21 W/m 2 .°C 7-12 Chapter 7 External Forced Convection 7-23 "!PROBLEM 7-23" "GIVEN" Vel=70 "[km/h], parameter to be varied" w=2.8 "[m]" L=8 "[m]" "q_dot_rad=200 [W/m^2], parameter to be varied" T_infinity=30 "[C]" "PROPERTIES" Fluid$='air' k=Conductivity(Fluid$, T=T_film) Pr=Prandtl(Fluid$, T=T_film) rho=Density(Fluid$, T=T_film, P=101.3) mu=Viscosity(Fluid$, T=T_film) nu=mu/rho T_film=1/2*(T_s+T_infinity) "ANALYSIS" Re=(Vel*Convert(km/h, m/s)*L)/nu "Reynolds number is greater than the critical Reynolds number. / 15 .) = = = 0.000031 ° C / W 2πkL 2π (52 W / m. Concrete block, 4-in 5a. Then the number of nodes M becomes L 3 cm M = +1 = +1 = 7 0.5 cm Δx The base temperature at node 0 is given to be T0 = 100°C. 4-100 Chapter 4 Transient Heat Conduction 4-109 A long roll of large 1-Mn manganese steel plate is to be quenched in an oil bath at a specified rate. The thickness of insulation that will protect the water from freezing under worst conditions is to be determined. The explicit transient finite difference formulation of the problem using the energy balance approach method is to be determined. Properties We assume the film temperature to be 35°C. Nodes 1, 2, 3, 4, and 5 are interior nodes, and thus for them we can use the general finite difference relation expressed as T − Tm T − Tm 4 kA m −1 + kA m +1 + h(pΔx)(T∞ − Tm) + εσ (pΔx)[Tsurr − (Tm + 273) 4] = 0 Δx Δx 4 − (Tm + 273) 4] = 0 , m = 1,2,3,4,5 or Tm −1 − 2Tm + Tm +1 + h(pΔx 2 / kA)(T∞ − Tm) + εσ (pΔx 2 / kA)[Tsurr The finite difference equation for node 6 at the fin tip is obtained by applying an energy balance on the half volume element about node 6. 4-1 Chapter 4 Transient Heat Conduction 4-12 Relations are to be obtained for the characteristic lengths of a large plane wall of thickness 2L, a very long cylinder of radius ro and a sphere of radius ro Analysis Relations for the characteristic lengths of a large plane wall of thickness 2L, a very long cylinder of radius ro and a sphere of radius ro are Lc ,wall = Lc ,cylinder = Lc ,sphere = V Asurface V Asurface V Asurface = 2 LA =L 2A = πro 2 h ro = 2πro h 2 = 4πro 3 / 3 4πro 2 = 2ro 2ro ro 3 2L 4-13 A relation for the time period for a lumped system to reach the average temperature (Ti + T∞) / 2 is to be obtained. Assumptions 1 Heat transfer through the wall is given to be transient, and the thermal conductivity and heat Convectio generation to be variables. 3 The thermal properties of the egg are constant. It is defined as Nu = where L is the characteristic length of the surface and k is the k thermal conductivity of the fluid. Fiberboard sheathing, 25 mm 4a. Assuming constant thermal conductivity and one-dimensional heat transfer, the mathematical formulation (the differential equation and the boundary conditions) of this heat conduction problem is to be obtained for steady operation. Assumptions 1 The balls are spherical in shape with a radius of r0 = 1 in. Assumptions 1 Heat transfer through the damn is given to be steady and two-dimensional. Assumptions 1 The egg is spherical in shape with a radius of r0 = 2.75 cm. The amount heat dissipated in 8 h and the heat flux on the surface of the chip are to be determined. Also, there is only one inlet and one exit and thus m& 1 = m& 2 = m& . The specific heat of the aluminum sink is 903 J/kg.°C (Table A-19), but can be taken to be 850 J/kg.°C for simplicity in analysis. Analysis We take the teapot and the water in it as our system that is a closed system (fixed mass). 18°C We measure x from the bottom surface of the block since Ice block this surface represents the adiabatic center surface of the -20°C plane wall of thickness 2L = 10 cm. Analysis (a) The characteristic length of the wire and Air the Biot number are 30°C πr 2 L ro 0.0015 m V 350°C Lc = = o = = = 0.00075 m 10 m/min As 2πro L 2 2 Bi = hLc (35 W/m 2 .°C)(0.00075 m) = = 0.000068 < 0.1 k 386 W/m.°C Copper wire Since Bi < 0.1 the lumped system analysis is applicable. 2-3 Chapter 2 Heat Conduction Equation 2-15E "GIVEN" E_dot=1000 "[W]" L=15 "[in]" "D=0.08 [in], parameter to be varied" "ANALYSIS" g_dot=E_dot/V_wire*Convert(W, Btu/h) V_wire=pi*D^2/4*L*Convert(in^3, ft^3) q_dot=E_dot/A_wire*Convert(W, Btu/h) A_wire=pi*D*L*Convert(in^2, ft^2) q [Btu/h.ft2] 521370 260685 173790 130342 104274 86895 74481 65171 57930 52137 D [in] 0.02 0.04 0.06 0.08 0.1 0.12 0.14 0.16 0.18 0.2 550000 500000 450000 400000 2 q [Btu/h-ft] 350000 300000 250000 200000 150000 100000 50000 0 0.02 0.04 0.06 0.08 0.1 0.12 D [in] 2-4 0.14 0.16 0.18 0.2 Chapter 2 Heat Conduction Equation 2-16 The rate of heat generation per unit volume in the uranium rods is given. The implicit finite difference equations are determined on the basis of the energy balance for the transient case expressed as ∑ Q& i +1 All sides Node 2: k • 1 Inner 4• surfac 7 • Heater 10 • • 2 • 5 Outer surfac 8 Glas • 0.2 • T i +1 − Tmi i+1 + G& element = ρVelement C m Δt We consider only 9 nodes because of symmetry. Analysis (a) Taking the direction normal to the surface of the plate to be the x direction with x = 0 at the bottom surface, the mathematical formulation of this problem can be expressed as and d 2T =0 dx 2 dT (L) −k = h[T (L) − T∞] = h(T2 − T∞) dx x 75°F T∞ h L T (L) = T2 = 75° F (b) Integrating the differential equation twice with respect to x yields dT = C1 dx T (x) = C1x + C2 where C1 and C2 are arbitrary constants. The centerline temperature of the bar after 10 min and after steady conditions are established are to be determined. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined. To describe a heat transfer problem completely, two boundary conditions must be given for each direction of the coordinate system along which heat transfer is significant. 6 All heat losses from the pond are negligible. Every effort is made to produce an error-free Solutions Manual. k (2.22 W/m.°C) The ice will start melting at the corners because of the maximum exposed surface area there. Then the maximum velocity and the Reynolds number based on the maximum velocity become Vmax = ST 0.04 V= (5.2 m/s) = 8.667 m/s 0.04 − 0.016 ST − D since S D > (ST + D) / 2 Re D SL Ts=100°C V=5.2 m/s Ti=20°C ST ρV D (1.145 kg/m 3)(8.667 m/s)(0.016 m) = max = = 8380 μ 1.895 × 10 −5 kg/m · s The average Nusselt number is determined using the proper relation from Table 7-2 to be 7-59 D Chapter 7 External Forced Convection Nu D = 0.35(S T / S L) 0.2 Re 0D.6 Pr 0.36 (Pr/ Prs) 0.25 = 0.35(0.04 / 0.04) 0.2 (8380) 0.6 (0.7268) 0.36 (0.7268 / 0.7111) 0.25 = 70.88 Since NL =20, which is greater than 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become Nu D, N L = Nu D = 70.88 h= Nu D , N L k D = 70.88(0.02625 W/m · °C) = 116.3 W/m 2 · °C 0.016 m The total number of tubes is N = NL ×NT = 20×10 = 200. Properties The thermal conductivity of the concrete is given to be k = 2 W/m·°C. The distance between two consecutive nodes is called the nodal spacing, and a differential equation whose derivatives are replaced by differences is called a difference equation. Properties The thermal conductivity of fiberglass insulation is given to be k = 0.035 W/m·°C. The rate of heat transfer through the roof and the money lost through the roof that night during a 14 hour period are to be determined. 2-134 Heat is generated uniformly in a cylindrical uranium fuel rod. 20°C water 1-130 The base surface of a cubical furnace is surrounded by black surfaces at a specified temperature. 3 The heat transfer from the tip of the spoon is negligible. 4-70C The dimensionless temperature for a three-dimensional heat transfer is determined by determining the dimensionless temperatures of one-dimensional geometries whose intersection is the three dimensional geometry, and taking their product. Assumptions 1 Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -181°F and 736 psia. energies Qin + We,in − Wb − Qout = ΔU (Q& in + W&e,in − Q& out) Δt = ΔH = m(h2 − h1) ≅ mC p (T2 − T1) 5,000 kJ/h ROOM The mass of air is V = 4 × 5 × 7 = 140 m3 m= PV (100 kPa)(140 m3) 1 = = 172.4 kg RT1 (0.287 kPa · m3 / kg · K)(283 K) Using the Cp value at room temperature, 4m × 5m × 7m Steam · Wpw [(10,000 − 5000)/3600 kJ/s + 0.1 kJ/s]Δt = (172.4 kg)(1.007 kJ/kg · °C)(20 − 10)°C It yields Δt = 1163 s 1-11 10,000 kJ/h Chapter 1 Basics of Heat Transfer 1-

31 A student living in a room turns his 150-W fan on in the morning. Therefore, Q& = 6 kW . Highly Reflective foil 19 mm Construction 1. Properties The thermal conductivity of the bar is given to be k = 0.0006 W/m·°C. The heat transfer coefficient at the surface of the rib, the temperature of the outer surface of the rib and the amount of heat transfer when it is wellsolone are to be determined. 2-112C A differential equation involves derivatives, an algebraic equation does not. Therefore, the wall can be considered to be a semi-infinite medium 2 The thermal properties of the wall are constant. Properties We assume the film temperature to be 10 °F. [illegible]

Properties The thermal conductivity and emissivity of copper are given to be k = 386 W/m·°C and ε = 0.7. [illegible] the stated assumptions and observations, the energy balance becomes E −E 1in424out 3 = Net energy transfer by heat, work, and mass ∆E system 1 424 3 Change in internal, kinetic, potential, etc. Analysis The conversion factors for W, m, and K are given in conversion tables to be 1 W = 3.41214 Btu / h 1 m = 3.2808 ft 1 K = 1.8 R Substituting gives the Stefan-Boltzmann constant in the desired units, σ = 5.67 W/m 2 .K 4 = 5.67 × 3.41214 Btu / h (3.2808 ft) 2 (1.8 R) 4 = 0.171 Btu / h.ft 2 .R 4 1-95 Using the conversion factors between W and Btu/h, m and ft, and °C and °F, the convection coefficient in SI units is to be expressed in Btu/h.ft2.°F. Vapor Analysis The rate of heat transfer to the oxygen tank is As = πD 2 = π (4 m) 2 = 50.27 m 2 Q& = hAs (Ts − Tair) = (25 W/m .°C)(50.27 m)[20 − (−183)]°C = 255,120 W 2 2 Air 20° 1 atm Liquid O2 Then the rate of evaporation of liquid oxygen in the tank is determined Q& to be -183°C Q& . 4 Thermal contact resistances at the interfaces are to be considered. q = 1000 W Analysis A 1000 W iron will convert electrical energy into heat in the wire at a rate of 1000 W. Assumptions 1 Heat conduction is steady and one-dimensional. Some examples of heat generations are resistance heating in wires, exothermic chemical reactions in a solid, and nuclear reactions in nuclear fuel rods. 3-14C Convection heat transfer through the wall is expressed as Q& = hAs (Ts − T∞) . 2-9 ⎞ ∂ ⎛ ∂T ⎞ ∂ 2T ⎟⎟ = − ⎜⎜ k ⎟⎟ = − k ∂y ⎝ ∂y ⎠ ∂y 2 ⎠ Chapter 2 Heat Conduction Equation 2-29 We consider a thin ring shaped volume element of width ∆z and thickness ∆r in a cylinder. It can be shown that the Trombe wall will deliver even more heat to the house during the 3rd day since it will start the day at a higher average temperature. However, an average constant temperature as specified in the problem will be used. 5-58 Chapter 5 Numerical Methods in Heat Conduction 5-59E The top and bottom surfaces of a V-grooved long solid bar are maintained at specified temperatures while the left and right surfaces are insulated. Substituting these values into the one-term solution gives θ0 = 2 To − T∞ = A1e − λ 1τ → Ti − T∞ -30°C 14 / Meat 7°C 2 −18 − (−30) = 1239 . 5-114 Chapter 5 Numerical Methods in Heat Conduction 5-118E A plane wall in space is subjected to specified temperature on one side and radiation and heat flux on the other. °F.h/Btu (b) Therefore, this is approximately a R-22 wall in English units. ° C)(0.0216 m2) L 0.002 m RAl = = 0.00039 ° C / W kA (237 W / m. Even in steady flow and thus constant mass flow rate, a fluid may accelerate. Then the total heat transfer during a specified time period is determined by adding the heat transfer amounts for each time step as I QTrumbe wall = ∑ i =1 I i QTrumbe wall = ∑h i in A[(T0 + T0i −1) / 2 − Tin]∆t i =1 where I is the total number of time intervals in the specified time period. Assumptions 1 Heat transfer through the plate is given to be steady and one-dimensional. 2-22 Chapter 2 Heat Conduction Equation 2-60 The base plate of a household iron is subjected to specified heat flux on the left surface and to specified temperature on the right surface. °C (c) From part (a) we have k 1 = = 0.15 . The contact conductance at the interface of copper-aluminum plates for the case of 1.3-1.4 µm roughness and 10 MPa pressure is hc = 49,000 W/m2·°C (Table 3-2). 3 The surface of the plate is smooth. (b) To determine the temperature distribution in the shell, we begin with the Fourier's law of heat conduction expressed as dT Q& = − k (T) A dr where the rate of conduction heat transfer Q& is constant and the heat conduction area A = 2πrL is variable. The cooling time and if any part of the carcass will suffer freezing injury during this cooling process are to be determined. 4 The flow is turbulent over the entire surface because of the constant agitation of the engine block. = 4.201 ° C / W The steady rate of heat transfer through the wall per 0.33 m2 is T −T [(22 − (−4)]° C = 619 Q& = ∞1 ∞ 2 = . 1-24 Chapter 1 Basics of Heat Transfer 1-60C Most ordinary insulations are obtained by mixing fibers, powders, or flakes of insulating materials with air. In other words, a differential equation is linear if it can be written in a form which does not involve (1) any powers of the dependent variable or its derivatives such as y 3 or (y ′) 2 , (2) any products of the dependent variable or its derivatives such as yy′ or y ′y ′′′ , and (3) any other nonlinear functions of the dependent variable such as sin y or e y . 5-102C As the step size is decreased, the discretization error decreases but the round-off error increases. Applying the boundary conditions give C1 = − q& 0 k x = 0: − kC1 = q& 0 → x = L: 4] − kC1 = h[T2 − T∞] + εσ [(T2 + 273) 4 − Tsurr Eliminating the constant C1 from the two relations above gives the following expression for the outer surface temperature T2, 4] = q& h(T2 − T∞) + εσ [(T2 + 273) 4 − Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2 · °C)(T2 − 22) + 0.7(5.67 × 10 −8 W/m 2 · K 4)[(T2 + 273) 4 − 290 4] = 100,000 W/m 2 Using an equation solver (or a trial and error approach), the outer surface temperature is determined from the relation above to be T2 = 895.8°C 2-71 Chapter 2 Heat Conduction Equation 2-131E The concrete slab roof of a house is subjected to specified temperature at the bottom surface and convection and radiation at the top surface. Continuing in this manner, it is observed that steady conditions are reached in the medium after about 6 hours for which the temperature at the center node is 1023°C. Analysis (a) The heat transfer rate and the rate of evaporation of the liquid without insulation are A = πD 2 = π (3 m) 2 = 28.27 m 2 1 1 Ro = = = 0.00101 °C/W 2 ho A (35 W/m .°C)(28.27 m 2) T − T∞ 2 [15 − (−196)]°C Q& = s1 = = 208,910 W Ro 0.00101 °C/W Q& 208.910 kJ/s Q& = m& h fg → m& = = = 1.055 kg/s h fg 198 kJ/kg Ro Ts1 T∞2 (b) The heat transfer rate and the rate of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are A = πD 2 = π (31 . Also, the temperature in each layer varies linearly and thus we could solve this problem by considering 3 nodes only (one at the interface and two at the boundaries). The rate of heat transfer to the tank and the rate at which ice melts are to be determined. The temperature distribution throughout the glass 15 min after the strip heaters are turned on and also when steady conditions are reached are to be determined using the implicit method with a time step of ∆t = 1 min. However, in a text of this magnitude, it is inevitable to have some, and we will appreciate hearing about them. Evacuating the space between the layers forms a vacuum which minimize conduction or convection through the air space. In order to determine radiation heat resistance we assume them to be 5°C and 15°C, respectively, and take the Vacuum emissivity to be 1. 7-27ba, f = 0.34. Properties The thermal conductivities are given to be 0.7 W/m·°C for glass and 0.12 W/m·°C for pine wood. 7-76C The R-value of an insulation represents the thermal resistance of insulation per unit surface area (or per unit length in the case of pipe insulation). The drag coefficient for a flat surface is equivalent to the mean friction coefficient. Analysis (a) We take the hair dryer as the system. The plates are isothermal and there is no change in the flow direction, and thus the temperature depends on y only, T = T(y). Analysis We take the length in the direction of heat transfer to be L and the width of the board to be w. 3 The plant operates every day of the year for 10 h. Properties The thermal conductivity, density, and specific heat of the milk at 20°C are k = 0.607 W/m.°C, ρ = 998 kg/m3, and Cp = 4.182 kJ/kg.°C (Table A-9). 1-120C It is necessary to ventilate buildings to provide adequate fresh air and to get rid of excess carbon dioxide, contaminants, odors, and humidity. = = ρC pV ρC p Lc (8500 kg / m3)(320 J / kg. That is, Q& rad = Q& cond − Q& conv = 770 − 220 = 550 W Then the fraction of heat transferred by radiation becomes f = Q& rad 550 = = 0.714 (or 71.4%) Q& cond 770 28°C L=0.6 cm 25°C Air, 15°C h=10 W/m2.°C A = 2.2 m2 1-79 Chapter 1 Basics of Heat Transfer 1-138 The range of U-factors for windows are given. The air is moved by a fan, and heat is lost through the walls of the duct. Properties The thermal conductivities are given to be k = 386 W/m·°C for copper plates and k = 0.26 W/m·°C for epoxy boards. The pipe is initially filled with stationary water at 0°C. Properties The thermal conductivities are given to be k = 15.1 W/m·°C for sheet metal and 0.035 W/m·°C for fiberglass insulation. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be hL Nu = = (0.037 Re L 0.8 − 871) Pr 1 / 3 = [0.037(2.163 × 10 7) 0.8 − 871](0.7340)1 / 3 = 2.384 × 10 4 k k 0.02428 W/m.°C h = Nu = (2.384 × 10 4) = 57.88 W/m 2 .°C L 10 m As = wL = (10 m)(4 m) = 40 m 2 Q& = hA (T − T) = (57.88 W/m 2 .°C)(40 m 2)(12 - 5)°C = 16,206 W = 16.21 kW s ∞ s 7-4 Chapter 7 External Forced Convection 7-17 "!PROBLEM 7-17" "GIVEN" Vel=55 "[km/h], parameter to be varied" height=4 "[m]" L=10 "[m]" "T_infinity=5 [C], parameter to be varied" T_s=12 "[C]" "PROPERTIES" Fluid$='air' k=Conductivity(Fluid$, T=T_film) Pr=Prandtl(Fluid$, T=T_film) rho=Density(Fluid$, T=T_film, P=101.3) mu=Viscosity(Fluid$, T=T_film) nu=mu/rho T_film=1/2*(T_s+T_infinity) "ANALYSIS" Re=(Vel*Convert(km/h, m/s)*L)/nu "We use combined laminar and turbulent flow relation for Nusselt number" Nusselt=(0.037*Re^0.8-871)*Pr^(1/3) h=k/L*Nusselt A=height*L Q_dot_conv=h*A*(T_s-T_infinity) Vel [km/h] 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 Qconv [W] 1924 2866 3746 4583 5386 6163 6918 7655 8375 9081 9774 10455 11126 11788 12441 T∞ [C] 0 0.5 1 1.5 2 2.5 3 3.5 4 4.5 5 5.5 6 Qconv [W] 15658 14997 14336 13677 13018 12360 11702 11046 10390 9735 9081 8427 7774 7-5 Chapter 7 External Forced Convection 6.5 7 7.5 8 8.5 9 9.5 10 7122 6471 5821 5171 4522 3874 3226 2579 14000 12000 10000 Q conv [W] 8000 6000 4000 2000 0 10 20 30 40 50 60 70 80 Vel [km /h] 16000 14000 Q conv [W] 12000 10000 8000 6000 4000 2000 0 2 4 6 T ∞ 8 [C] 7-6 10 Chapter 7 External Forced Convection 7-18E Air flows over a flat plate. Cylinder: This cylindrical block can physically be formed by the intersection of a long cylinder of radius ro = D/2 = 2.5 cm exposed to the hot gases with a heat transfer coefficient of h = 40 W / m2 .° C and a plane wall of thickness 2L = 5 cm exposed to the hot gases with h = 80 W / m2 . 3 There is no heat generation. 8 s Therefore, any time step less than 4.8 s can be used to solve this problem. 2-28 Chapter 2 Heat Conduction Equation 2-65 A large plane wall is subjected to specified heat flux and temperature on the left surface and no conditions on the right surface. 5-66 Chapter 5 Numerical Methods in Heat Conduction 5-73 A plane wall with no heat generation is subjected to specified temperature at the left (node 0) and heat flux at the right boundary (node 6). Properties The emissivity of the base surface is ε = 0.7. Analysis The base surface is completely surrounded by the top and side surfaces. 6-33C During steady, laminar, two-dimensional flow over an isothermal plate, the thickness of the velocity boundary layer (a) increase with distance from the leading edge, (b) decrease with free-stream velocity, and (c) and increase with kinematic viscosity 6-34C During steady, laminar, two-dimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensionalizing the convection equations is the significant reduction in the number of parameters [the original problem involves 6 parameters (L, V , T∞, Ts, ν, α), but the nondimensionalized problem involves just 2 parameters (ReL and Pr)]. In this case heat is transferred from the surroundings to the cold surfaces, and the refrigeration unit must now work harder and longer to make up for this heat gain and thus it must consume more electrical energy. Analysis The change in the sensible internal energy content of the body as a result of the body temperature rising 2°C during strenuous exercise is ∆U = mC∆T = (70 kg)(3.6 kJ/kg.°C)(2°C) = 504 kJ 1-4 Chapter 1 Basics of Heat Transfer 1-19 An electrically heated house maintained at 22°C experiences infiltration losses at a rate of 0.7 ACH. Analysis The resistance heater converts electric energy into heat at a rate of 3 kW. The center temperature of each geometry is to be determined. Using the average plate temperature, the average rate of heat loss from the plate is determined from ⎛ 140 + 22 ⎞ − 22 ⎟°C = 21.2 W Q& loss = hA(Tplate, ave − T∞) = (12 W/m 2 .°C)(0.03 m 2)⎜ 2 ⎝ ⎠ Energy balance on the plate can be expressed as E in − E out = ∆E plate → Q& in ∆t − Q& out ∆t = ∆E plate = mC p ∆Tplate Solving for ∆t and substituting, mC p ∆Tplate (0.4155 kg)(875 J / kg. Properties The thermal conductivity of the brick wall is given to be k = 0.42 Btu/h.ft.°F. When the effective emissivity is known, the radiation heat transfer through the air space is determined from the Q& rad relation above. Therefore, the wall can be considered to be a semi-infinite medium with a specified surface temperature of 1800°F. ° C)(0.04 m) = = 1.051 ≈ 1.0 k (0.571 W / m. 2 The transistor case is isothermal at 90 °C . Inside surface, still air R-value, m2.°C/W Between At furring furring 0.030 0.030 0.12 0.12 0.45 0.45 0.27 0.27 0.49 ----0.94 0.079 0.079 0.12 0.12 3 Total unit thermal resistance of each section, R 1 2 The U-factor of each section, U = 1/R, in W/m2.°C Area fraction of each section, farea Overall U-factor, U = Σfarea,iUi = 0.84×0.641+0.16×0.498 Overall unit thermal resistance, R = 1/U 1.559 2.009 0.641 0.498 0.84 0.16 0.618 W/m2.°C 1.62 m2.°C/W Therefore, the overall unit thermal resistance of the wall is R = 1.62 m2.°C/W and the overall U-factor is U = 0.618 W/m2.°C. 4 Any heat transfer from the back surface of the board is disregarded. 3 The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be checked). Then the one-term solution can be written in the form θ 0, sph = 2 2 T0 − T∞ 60 − 163 = A1 e − λ1 τ → = 0.65 = A1e −λ1 (0.1217) 4.5 − 163 Ti − T∞ It is determined from Table 4-1 by trial and error that this equation is satisfied when Bi = . Heat transfer is one-dimensional if it occurs primarily in one direction. In the case of a water nozzle, for example, the velocity of water remains constant at a specified point, but it changes from inlet to the exit (water accelerates along the nozzle). Analysis (a) The amount of heat this resistor dissipates during a 24-hour period is Q = Q& ∆t = (0.15 W)(24 h) = 3.6 Wh (b) The heat flux on the surface of the resistor is As = 2 πD 2 + πDL = 2 π (0.003 m) 2 + π (0.003 m) (0.012 m) = 0.000127 m 2 4 4 & Q 0.15 W = = 1179 W/m 2 q& = As 0.000127 m 2 Q& Resistor 0.15 W (c) The surface temperature of the resistor can be determined from Q& 0.15 W → T s = T∞ + = = 171°C Q& = hAs (T s − T∞) 2 hAs (1179 W/m .°C)(0.00012 7 m 2) 3-7 Chapter 3 Steady Heat Conduction 3-24 A power transistor dissipates 0.2 W of power steadily in a specified environment. We observe that this is a steady-flow process since there is no change with time at any point and thus ∆mCV = 0 and ∆E CV = 0 , there is only one inlet and one exit and thus m& 1 = m& 2 = m& , and heat is lost from the system. However, this approach is expensive, time consuming, and often impractical. Analysis (a) The rate of heat loss from the steam pipe is 280°F As = πDL = π (4 / 12 ft)(200 ft) = 209.4 ft 2 Q& pipe = hAs (Ts − Tair) = (6 Btu/h.ft 2 .°F)(209.4 ft 2)(280 − 50)°F = 289,000 Btu/h D =4 in L=200 Q (b) The amount of heat loss per year is Air,50° F Q = Q& ∆t = (289,000 Btu / h)(365 × 24 h / yr) = 2.532 × 109 Btu / yr The amount of gas consumption per year in the furnace that has an efficiency of 86% is Annual Energy Loss = 2.532 × 10 9 Btu/yr ⎛ 1 therm ⎞⎟ ⎜⎜ ⎟ = 29,438 therms/yr 0.86 ⎝ 100,000 Btu ⎠ Then the annual cost of the energy lost becomes Energy cost = (Annual energy loss)(Unit cost of energy) = (29,438 therms / yr)($0.58 / therm) = $17,074 / yr 1-88 A 4-m diameter spherical tank filled with liquid nitrogen at 1 atm and -196°C is exposed to convection with ambient air. Then the mesh Fourier number becomes τ= α∆t l2 = (3.2 × 10 −6 m 2 /s)(15 s) (0.015 m) 2 = 0.2133 (for ∆t = 15 s) Using the specified initial condition as the solution at time t = 0 (for i = 0), sweeping through the 9 equations above will give the solution at intervals of 15 s. 7-57 Chapter 7 External Forced Convection 7-65 Combustion air is preheated by hot water in a tube bank. Thus we have laminar flow over the entire plate. 2 The fluid has constant properties. Properties The thermal conductivity of the tank is given to be k = 18 W/m·°C. Replacing the symmetry lines by insulation and utilizing the mirror-image concept, the finite difference equations for the interior nodes 1 and 2 are determined to be Node 1 (interior) : 100 + 200 + 2T2 − 4T1 = 0 Node 2 (interior) : 100 + 100 + 200 + T1 − 4T4 = 0 100 • 100 • 100 • 100 • Solving the 2 equations above simultaneously gives T1 = T4 = 143°C T2 = T3 = 136°C Insulated 1 • 200 200 • 2 • 3 • 4 • 200 200 200 Discussion Note that taking advantage of symmetry simplified the problem greatly. 2 Heat transfer is one-dimensional since there is thermal symmetry about the center line and no change in the axial direction. Thus we conclude that steady conditions are reached after 3.8 min. Discussion The temperature of the outer parts of the rib is greater than that of the inner parts of the rib after it is taken out of the oven. We observe that this is a steady-flow process since there is no change with time at any point and thus ∆mCV = 0 and ∆E CV = 0 , and there is only one inlet and one exit and thus m& 1 = m& 2 = m& . The power rating of the electric resistance heater is to be determined. The finite difference formulation of this problem is to be obtained, and the temperature of the other side under steady conditions is to be determined. • This problem involves 5 unknown nodal temperatures, and thus 1 Top layer Solar pond • we need to have 5 equations. But the flow is assumed to be turbulent over the entire surface because of the constant agitation of the engine block. 4 Heat loss from the house to the outdoors is negligible during heating. The time it takes for the ice in the chest to melt completely is to be determined. 6-4C The potato will normally cool faster by blowing warm air to it despite the smaller temperature difference in this case since the fluid motion caused by blowing enhances the heat transfer coefficient considerably. Noting that the total thickness of the flange is t = 0.02 m, the heat conduction area at any location along the flange is Acond = 2πrt where the values of radii at the nodes and between the nodes (the mid points) are r0=0.046 m, r1=0.05 m, r2=0.06 m, r3=0.07 m, r4=0.08 m, r5=0.09 m, r6=0.10 m r01=0.048 m, r12=0.055 m, r23=0.065 m, r34=0.075 m, r45=0.085 m, r56=0.095 m Then the finite difference equations for each node are obtained from the energy balance to be as follows: Node 0: hi (2πtr0)(Ti − T0) + k (2πtr01) T1 − T0 =0 ∆x1 Node 1: k (2πtr01) T0 − T1 T −T 4 + k (2πtr12) 2 1 + 2[2π (r1 + r12) / 2)](∆x 2 / 2)}h(T∞ − T1) + εσ [Tsurr − (T1 + 273) 4]} = 0 ∆x1 ∆x 2 Node 2: k (2πtr12) T − T2 T1 − T2 4 + k (2πtr23) 3 + 2(2πtr2 ∆x 2){h(T∞ − T2) + εσ [Tsurr − (T2 + 273) 4]} = 0 ∆x 2 ∆x 2 Node 3: k (2πtr23) T2 − T3 T − T3 4 + k (2πtr34) 4 + 2(2πtr3 ∆x 2){h(T∞ − T3) + εσ [Tsurr − (T3 + 273) 4]} = 0 ∆x 2 ∆x 2 Node 4: k (2πtr34) T3 − T4 T − T4 4 + k (2πtr45) 5 + 2(2πtr4 ∆x 2){h(T∞ − T4) + εσ [Tsurr − (T4 + 273) 4]} = 0 ∆x 2 ∆x 2 Node 5: k (2πtr45) T4 − T5 T − T5 4 + k (2πtr56) 6 + 2(2πtr5 ∆x 2){h(T∞ − T5) + εσ [Tsurr − (T5 + 273) 4]} = 0 ∆x 2 ∆x 2 Node 6: k (2πtr56) T5 − T6 4 + 2[2π (∆x 2 / 2)(r56 + r6) / 2 + 2πr6 t]{h(T∞ − T6) + εσ [Tsurr − (T6 + 273) 4]} = 0 ∆x 2 where ∆x1 = 0.004 m, ∆x 2 = 0.01 m, k = 52 W/m · °C, ε = 0.8, T∞ = 8°C, Tin = 200°C, Tsurr = 290 K and h = 25 W/m2 · °C, hi = 180 W/m2 · °C, σ = 5.67 × 10-8 W/m2 · K 4 . The earth and thick walls can be considered to be semi-infinite media. m) = 0.01 m 2 ∆T = 82 − 74 = 8° C L Then the thermal conductivity of the material becomes & ∆T QL (17.5 W)(0.005 m) Q& = kA → k = = = 1.09 W / m. 1-59 Chapter 1 Basics of Heat Transfer 1-108 The roof of a house with a gas furnace consists of a 15-cm thick concrete that is losing heat to the outdoors by radiation and convection. F)[(7 / 12) × (0.5 / 12)]ft 2 Ri = 9 / 12 ft L = = 5.51 h°F/Btu kA (0.40 Btu/h.ft.°F)[(7 / 12) × (7 / 12)]ft 2 1 1 = = 0.64 h°F/Btu Ro = ho A (4 Btu/h.ft 2 .°F)(0.3906 ft 2) 1 1 1 1 1 1 1 = + + + + → R mid = 5.3135 h°F/Btu R mid R 2 R3 R 4 288 308.57 5.51 Rtotal = Ri + R1 + R mid + R5 + Ro = 1.7068 + 1.0667 + 5.3135 + 1.0667 + 0.64 = 9.7937 h°F/Btu T −T (80 − 30)°F = 5.1053 Btu/h Q& = ∞1 ∞ 2 = 9.7937 h°F/Btu Rtotal R 4 = Rbrick = Then steady rate of heat transfer through entire wall becomes (30 ft)(10 ft) Q& total = (5.1053 Btu/h) = 3921 Btu/h 0.3906 m 2 3-32 T∞2 Chapter 3 Steady Heat Conduction (b) The thermal resistance network and the individual thermal resistances if the wall is constructed of bricks with air holes are R2 Ri R1 R3 T∞1 R6 R4 R7 T∞2 R5 Aairholes = 9(1.25 / 12) × (1.25 / 12) = 0.0977 ft 2 Abricks = (7 / 12 ft) 2 − 0.0977 = 0.2426 ft 2 9 / 12 ft L = = 511.77 h°F/Btu kA (0.015 Btu/h.ft.°F)(0.0977 ft 2) L 9 / 12 ft = = = 7.729 h°F/Btu kA (0.40 Btu/h.ft.°F)(0.2426 ft 2) R 4 = R airholes = R5 = Rbrick 1 1 1 1 1 1 1 1 1 = + + + + + + → R mid = 7.244 h°F/Btu R mid R 2 R3 R 4 R5 288 308.57 511.77 7.729 Rtotal = Ri + R1 + R mid + R6 + Ro = 1.7068 + 1.0667 + 7.244 + 1.0677 + 0.64 = 11.7252 h°F/Btu T −T (80 − 30)°F = 4.2643 Btu/h Q& = ∞1 ∞ 2 = 11.7252 h°F/Btu Rtotal Then steady rate of heat transfer through entire wall becomes (30 ft)(10 ft) Q& total = (4.2643 Btu/h) = 3275 Btu/h 0.3906 ft 2 3-33 Chapter 3 Steady Heat Conduction 3-57 A composite wall consists of several horizontal and vertical layers. ft 2 1 1 = = 0.27211 h °F/Btu 2 hi Ai (35 Btu/h.ft .°F)(0.105 ft 2) ln(r2 / r1) ln(3 / 2) = = 0.00029 h °F/Btu R pipe = 2πkL 2π(223 Btu/h.ft.°F)(1 ft) 1 1 = = 0.00425 h°F/Btu Ro = 2 ho Ao (1500 Btu/h.ft .°F)(0.157 ft 2) Ri = Rtotal = Ri + R pipe + Ro = 0.27211 + 0.00029 + 0.00425 = 0.27665 h °F/Btu The heat transfer rate per ft length of the tube is T − T∞ 2 (100 − 70)°F = = 108.44 Btu/h Q& = ∞1 Rtotal 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length of the tube required is determined to be Q& total = m& h fg = (120 lbm/h)(1037 Btu/lbm) = 124,440 Btu/h Tube length = Q& total 124,440 = = 1148 ft 108.44 Q& 3-60 Chapter 3 Steady Heat Conduction 3-79E Steam exiting the turbine of a steam power plant at 100°F is to be condensed in a large condenser by cooling water flowing through copper tubes. 5-103C The round-off error can be reduced by avoiding extremely small mess sizes (smaller than necessary to keep the discretization error in check) and sequencing the terms in the program such that the addition of small and large numbers is avoided. Applying the boundary conditions give r = r1: T (r1) = − r = r2: −k C1 + C2 = T1 r1 ⎛ C ⎞ C1 = h⎜⎜ − 1 + C2 − T∞ ⎟⎟ 2 r2 ⎝ r2 ⎠ Solving for C1 and C2 simultaneously gives C1 = r2 (T1 − T∞) r k 1− 2 − r1 hr2 and C2 = T1 + C1 = T1 + r1 T1 − T∞ r2 r k r1 1− 2 − r1 hr2 Substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = − ⎛ 1 1 ⎞ C1 C T1 − T∞ + T1 + 1 = C1 ⎜⎜ − ⎟⎟ + T1 = r k r r1 ⎝ r1 r ⎠ 1− 2 − r1 hr2 = 1− (0 − 25)°C 30 W/m · °C 2.1 − 2 (18 W/m 2 · °C)(2.1 m) ⎛ r2 r2 ⎞ ⎜⎜ − ⎟⎟ + T1 ⎝ r1 r ⎠ ⎛ 2.1 2.1 ⎞ − ⎜ ⎟ + 0°C = 29.63(1.05 − 2.1 / r) r ⎠ ⎝ 2 (c) The rate of heat conduction through the wall is 2-26 Chapter 2 Heat Conduction Equation C r (T − T) dT = −k (4πr 2) 21 = −4πkC1 = −4πk 2 1 ∞ Q& = − kA r k dx r 1− 2 − r1 hr2 (2.1 m)(0 − 25)°C = −4π (30 W/m · °C) = 23,460 W 2.1 30 W/m · °C 1− − 2 (18 W/m 2 · °C)(2.1 m) 2-27 Chapter 2 Heat Conduction Equation 2-64 A large plane wall is subjected to specified heat flux and temperature on the left surface and no conditions on the right surface. 4-53 Chapter 4 Transient Heat Conduction 4-58E The center temperature of meat slabs is to be lowered to 36°F during 12-h of cooling. This problem involves 12 unknown nodal temperatures, and thus we need to have 6 equations to determine them uniquely. 2 Convection heat transfer coefficient is constant and uniform. 2 Heat two15°C dimensional (no change in the axial direction). 2 Heat conduction in the meat slabs is one-dimensional because of symmetry about the centerplane. 2 Heat transfer at the edges and corners is two-or threedimensional. Analysis The total cooling load of the room is determined from Q& = Q& + Q& + Q& cooling where Q& lights lights people heat gain = 10 × 100 W = 1 kW Q& people = 40 × 360kJ/h = 14,400 kJ/h = 4kW Room 15,000 kJ/h Q& heat gain = 15,000 kJ/h = 4.17 kW Substituting, Q& cooling = 1 + 4 + 4.17 = 9.17 kW Thus the number of air-conditioning units required is 9.17 kW = 1.83 → 2 units 5 kW/unit 1-7 40 people 10 bulbs · Qcool Chapter 1 Basics of Heat Transfer 1-27E The air in a rigid tank is heated until its pressure doubles. • • • 0 •• 1.5 cm T − Tm Tm −1 − Tm 4 + kA m +1 + h(p∆x)(T∞ − Tm) + εσ (p∆x)[Tsurr − (Tm + 273) 4] = 0 ∆x ∆x 4 Tm −1 − 2Tm + Tm +1 + h(p∆x 2 / kA)(T∞ − Tm) + εσ (p∆x 2 / kA)[Tsurr − (Tm + 273) 4] = 0 , m = 1-12 or The finite difference equation for node 6 at the fin tip is obtained by applying an energy balance on the half volume element about node 13. Hot air is blown over the exposed surface of the plate on the top to melt the ice. and A1 = 10018 . The variation of temperature in the pipe and the center surface temperature of the pipe are to be determined for steady one-dimensional heat transfer. ° C)(1 m 2)(25 − 20)° C = 45 W 1 mm L 1 mm o room s ,out Using the thermal resistance network, heat transfer between the room and the refrigerated space can be expressed as Troom − Trefrig Q& = Rtotal Q& / A = Troom − Trefrig Ri R1 Rins Troom 1 1 ⎛L⎞ ⎛L⎞ + 2⎜ ⎟ +⎜ ⎟ + ho k k h ⎝ ⎠ metal ⎝ ⎠ insulation i Substituting, (25 − 3)° C L 1 2 × 0.001 m 1 + + + 2 2 2 9 W/m . 2 The thermal properties of the shaft are constant. 3-119 Chapter 3 Steady Heat Conduction 3-160E The surface temperature of a baked potato drops from 300°F to 200°F in 5 minutes in an environment at 70°F. The cooling time of the chicken is to be determined for the cases of cooling air being at −40°F and -80°F. Using the proper relation for Nusselt number, heat transfer coefficient and then heat transfer rate are determined to be hL Nu = = (0.037 Re L 0.8 − 871) Pr 1 / 3 = [0.037(1.081 × 10 7) 0.8 − 871](0.7340)1 / 3 = 1.336 × 10 4 k k 0.02428 W/m.°C h = Nu = (1.336 × 10 4) = 32.43 W/m 2 .°C L 10 m As = wL = (4 m)(10 m) = 40 m 2 Q& = hA (T − T) = (32.43 W/m 2 .°C)(40 m 2)(12 - 5)°C = 9081 W = 9.08 kW s ∞ s If the wind velocity is doubled: V L [(110 × 1000 / 3600)m/s](10 m) Re L = ∞ = = 2.163 × 10 7 υ 1.413 × 10 −5 m 2 /s which is greater than the critical Reynolds number. Therefore, the initial nodal temperatures are T00 = 20°C, T10 = 16.66°C, T20 = 13.33°C, T30 = 10°C, T40 = 6.66°C, T50 = 3.33°C, T60 = 0°C Substituting the given and calculated quantities, the nodal temperatures after 6, 12, 18, 24, 30, 36, 42, and 48 h are calculated and presented in the following table and chart. Assumptions 1 Quasi-steady operating conditions exist. Nodes 2, 3, and 4 are interior nodes, and thus for them we can use the general explicit finite difference relation expressed as g& i ∆x 2 g& mi ∆x 2 Tmi+1 − Tmi = → Tmi +1 = τ (Tmi −1 + Tmi +1) + (1 − 2τ)Tmi + τ m k τ k Node 0 can also be treated as an interior node by using the mirror image concept. ° C)(0149 . T1 = 60°C Properties The thermal conductivity of concrete is given to be k = 0.75 W/m·°C. Using the proper relation for Nusselt 400 m/min number, heat transfer coefficient is determined to be hL Nu = = 0.664 Re L 0.5 Pr 1 / 3 = 0.664(7.051 × 10 4) 0.5 (0.7255)1 / 3 = 158.4 k k 0.02662 W/m.°C h = Nu = (158.4) = 23.43 W/m 2 .°C L 0.18 m The temperatures on the two sides of the circuit board are Q& Q& = hAs (T2 − T∞) → T2 = T∞ + hAs (80 × 0.06) W = 30°C + = 39.48°C (23.43 W/m 2 .°C)(0.12 m)(0.18 m) kA Q& L Q& = s (T1 − T2) → T1 = T2 + L kAs (80 × 0.06 W)(0.003 m) = 39.48°C + = 39.52°C (16 W/m.°C)(0.12 m)(0.18 m) 7-98 T1 T2 Chapter 7 External Forced Convection 7-104E The equivalent wind chill temperature of an environment at 10°F at various winds speeds are V = 10 mph: Tequiv = 914 . 4 The thermal contact resistance between the steaks and the plate is negligible. × 10 −6 m 2 / s)(1 s) (0.02 m) 2 = 0.24275 Substituting these values, the nodal temperatures along the fin after 5×60 = 300 time steps (4 h) are determined to be T0 = 120°C, T1 = 110.6°C, T2 = 103.9°C, T3 = 100.0°C, and T4 = 98.5°C. 40 m 1 . The insulating ability of hair or feathers is most visible in birds and hairy animals. Assumptions 1 Heat conduction in the ice block is two-dimensional, and thus the temperature varies in both x- and y- directions. We evaluate the air properties at the assumed mean temperature of -5°C (will be checked later) and 1 atm (Table A-15): k = 0.02326 W/m-K ρ = 1.316 kg/m3 Cp =1.006 kJ/kg-K Pr = 0.7375 µ = 1.705×10 kg/m-s Prs = Ts = 0.7408 -5 Also, the density of air at the inlet temperature of 0°C (for use in the mass flow rate calculation at the inlet) is ρi = 1.292 kg/m3. The remaining space between the steel plates is filled with fiberglass insulation. 4 One-quarter of the heat the person generates is lost from the head. In this case steady operation is established in ---- min, and the nodal temperatures under steady conditions are determined to be T0 =2420°C, T1 =2413°C, T2 =2391°C, T3 =2356°C, and T4 =2306 °C Discussion The steady solution can be checked independently by obtaining the steady finite difference formulation, and solving the resulting equations simultaneously. P2-30).)e − (2.1589) (1.734) = 0.0005 = A1e − λ 1 τ = (15618 Ti − T∞ ⎡ T (0,0, t) − T∞ ⎤ = θ o, wall × θ o,cyl = 1× 0.0005 = 0.0005 ⎢ ⎥⎣ Ti − T∞ ⎦ short cylinder T (0,0, t) − 202 = 0.0005 → T (0,0, t) = 202 °F 40 − 202 (b) Treating the hot dog as an infinitely long cylinder will not change the results obtained in the part (a) since dimensionless temperatures for the plane wall is 1 for all cases. Using the energy balance approach and taking the direction of all heat transfers to be towards the node under consideration, the explicit transient finite difference formulations become Node 1 (interior): hp∆x (T∞ − T1i) + kA T − T1i T2i − T1i T i +1 − T1i + kA 0 = ρA∆xC 1 ∆x ∆x ∆t Node 2 (interior): hp∆x (T∞ − T2i) + kA T3i − T2i T i − T2i T i +1 − T2i + kA 1 = ρA∆xC 2 ∆x ∆x ∆t Node 3 (interior): hp∆x (T∞ − T3i) + kA T4i − T3i T i − T3i T i +1 − T3i + kA 2 = ρA∆xC 3 ∆x ∆x ∆t Node 4 (fin tip): where A = πD 2 / 4 h(p∆x / 2 + A)(T∞ − T4i) + kA T3i − T4i T i +1 − T4i = ρA(∆x / 2)C 4 ∆x ∆t is the cross-sectional area and p = πD is the perimeter of the fin. 8-4C The generally accepted value of the Reynolds number above which the flow in a smooth pipe is turbulent is 4000. Analysis We take the entire contents of the tank, water + iron block, as the system. 0025)(0 . Nodes 2 and 3 are interior nodes in a plain wall, and thus for them we can use the general explicit finite difference relation expressed as g& mi ∆x 2 Tmi+1 − Tmi = → Tmi +1 = τ (Tmi −1 + Tmi +1) + (1 − 2τ)Tmi k τ The finite difference equations for other nodes are obtained from an energy balance by taking the direction of all heat transfers to be towards the node under consideration: Tmi−1 − 2Tmi + Tmi+1 + Node 1: h(T∞ − T1i) + ε steak σ [(T∞ + 273) 4 − (T1i + 273) 4] + k steak Node 2 (interior) : T2i +1 = τ steak (T1i + T3i) + (1 − 2τ steak)T2i Node 3 (interior) : T3i +1 = τ steak (T2i + T4i) + (1 − 2τ steak)T3i T2i − T1i ∆x T1i +1 − T1i = (ρC) steak ∆x 2 ∆t Node 4: π (r452 − r42){h(T∞ − T4i) + ε plateσ [(T∞ + 273) 4 − (T4i + 273) 4]} + k steak (πr42) + k plate (2πr45δ) T3i − T4i ∆x T5i − T4i T i +1 − T4i 2 δ)] 4 = [(ρC) steak (πr42 ∆x / 2) + (ρC) plate (πr45 ∆r ∆t Node 5: 2πr5 ∆r{h(T∞ − T5i) + ε plateσ [(T∞ + 273) 4 − (T5i + 273) 4]} + k plate (2πr56 δ) T6i − T5i T i +1 − T5i = (ρC) plate (πr52 δ) 5 ∆r ∆t Node 6: 2π [(r56 + r6) / 2](∆r / 2){h(T∞ − T6i) + ε plateσ [(T∞ + 273) 4 − (T6i + 273) 4]} + k plate (2πr56 δ) T5i − T6i T i +1 − T6i = (ρC) plate [2π (r56 + r6) / 2](∆r / 2)δ 6 ∆r ∆t 5-118 Chapter 5 Numerical Methods in Heat Conduction where (ρC p) plate = 3439 kW/m 3 · °C, (ρC p) steak = 1504 kW/m 3 · °C, ksteak = 1.40 W/m.°C, εsteak = 0.95, α steak = 0.93 × 10 −6 m 2 / s , hif = 187 kJ/kg, kplate = 401 W/m.°C, α plate = 117 × 10 −6 m 2 / s , and εplate = 0.90, T∞ = 20°C, h =12 W/m2.°C, δ = 0.01 m, ∆x = 0.005 m, ∆r = 0.0375 m, and ∆t = 5 s. dT 1-48C Conduction is expressed by Fourier's law of conduction as Q& cond = − kA where dx dT/dx is the temperature gradient, k is the thermal conductivity, and A is the area which is normal to the direction of heat transfer. Analysis First we find the Biot number: Air Bi = . The mechanical power wasted is equal to the rate of heat transfer. 1-97C (a) Conduction and convection: No. (b) Conduction and radiation: Yes. The density of the cylinder is ρ, the specific heat is C, and the length is L. 2-61 Chapter 2 Heat Conduction Equation 2-116C A linear homogeneous differential equation of order n is expressed in the most general form as y (n) + f 1 (x) y (n −1) + L + f n −1 (x) y ′ + f n (x) y = 0 Each term in a linear homogeneous equation contains the dependent variable or one of its derivatives after the equation is cleared of any common factors. It yields Tm −1 − 2Tm + Tm +1 + g& m ∆x 2 =0 k 5-66C The explicit finite difference formulation of a general interior node for transient two-dimensional g& i l 2 i +1 i i i i i heat conduction is given by Tnode = τ (Tleft + Ttop + Tright + Tbottom) + (1 − 4τ)Tnode + τ node . The cooling time and if any part of the potatoes will suffer chilling injury during this cooling process are to be determined. 4-80 Chapter 4 Transient Heat Conduction 4-81 A cubic block and a cylindrical block are exposed to hot gases on all of their surfaces. We assume the flow is laminar over the entire finned surface of the transformer. The formal finite difference method is based on replacing derivatives by their finite difference approximations. Then, T1 = T2 and T3 = T4 Therefore, there are there are only 2 unknown nodal temperatures, T1 and T3, and thus we need only 2 equations to determine them uniquely. Analysis When a 0.01-in thick layer of deposit forms on the inner surface of the pipe, the inner diameter of the pipe will reduce from 0.4 in to 0.38 in. Acoustic tile, 19 mm 3. 1-57 Chapter 1 Basics of Heat Transfer 1-105 A 1000-W iron is left on the iron board with its base exposed to the air at 20°C. In steady operation, the rate of heat transfer through the insulation refrigerator wall is constant, and thus heat transfer between the room and the refrigerated space is equal to the heat transfer between the room and the outer surface of the refrigerator. For example, y ′′′ is the third order derivative of y, whereas (y ′) 3 is the third degree of the first derivative of y. At -100°C, µ = 1.189 × 10 −5 kg/m.s . 8-17C The hydrodynamic and thermal entry lengths are given as Lh = 0.05 Re D and Lt = 0.05 Re Pr D for laminar flow, and Lh ≈ Lt ≈ 10D in turbulent flow. 2 The potato is completely surrounded by the interior surfaces of the room. 2 Heat transfer by radiation is not considered. ° F Ao (Ts − T∞) π (3 / 12 ft) 2 (250 − 70)° F When the potato is wrapped in a towel, the thermal resistance and heat transfer rate are determined to be r −r [(1.50 + 0.02) / 12]ft − (1.50 / 12)ft = 0.5584 h.°F/Btu Rair = 2 1 = 4πkr1 r2 4π (0.015 Btu/h.ft.°F)[(1.50 + 0.02) / 12]ft (1.50 / 12)ft r −r [(1.52 + 0.12) / 12]ft − (1.52 / 12)ft = 1.3134 h°F/Btu R towel = 3 2 = 4πkr2 r3 4π (0.035 Btu/h.ft.°F)[(1.52 + 0.12) / 12]ft (1.52 / 12)ft 1 1 = = 0.2477 h.°F/Btu Rconv = hA (17.2 Btu/h.ft 2 .°F)π (3.28 / 12) 2 ft 2 R total = Rair + R towel + Rconv = 0.5584 + 1.3134 + 0.2477 = 2.1195 h°F/Btu T − T∞ (250 − 70)°F Q& = s = = 84.9 Btu/h Rtotal 2.1195 h.°F/Btu ∆t = Q 50.8 Btu = = 0.598 h = 35.9 min Q& 84.9 Btu/h This result is conservative since the heat transfer coefficient will be lower because of the smaller exposed surface temperature. ° C)(0.30 × 1 m 2) L . Properties The properties of the plate are given to be k = 1.2 Btu/h·ft·°F and ε = 0.80, and α s = 0.45 . the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become Nu D, N L = FNu D = 53.73 h= Nu D , N L k D = 53.73(0.02326 W/m · °C) = 156.2 W/m 2 · °C 0.008 m The total number of tubes is N = NL ×NT = 30×15 = 450. The number of copper fillings in the board and the area they comprise are Atotal = (6 / 12 ft)(8 / 12 ft) = 0.333 m 2 n copper = 0.33 ft 2 = 13,333 (number of copper fillings) (0.06 / 12 ft)(0.06 / 12 ft) Acopper = n πD 2 4 = 13,333 Aepoxy = Atotal − Acopper π (0.02 / 12 ft) 2 = 0.0291 ft 2 4 = 0.3333 − 0.0291 = 0.3042 ft 2 Rcopper The thermal resistances are evaluated to be L 0.05 / 12 ft = = 0.00064 h.°F/Btu kA (223 Btu/h.ft.°F)(0.0291 ft 2) L 0.05 / 12 ft = = = 0.137 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy Repoxy Then the thermal resistance of the entire epoxy board becomes 1 1 1 1 = + + = → Rboard = 0.00064 h.°F/Btu Rboard Rcopper Repoxy 0.00064 0.137 3-42 Chapter 3 Steady Heat Conduction in Cylinders and Spheres 3-64C When the diameter of cylinder is very small compared to its length, it can be treated as an indefinitely long cylinder.

using the conduction problem, and the result is to be verified. 3 The temperature of the surrounding surfaces is the same as the temperature of the surrounding air. h, Analysis The nodal spacing is given to be 1 2 • • Δx=Δx=l=0.1 m. The temperature at the center of the cylinder 10 cm from the end surface is to be determined. Bi hro Then, k = 0.15r0 = (0.15)(0.011 m) = 0.00165 m h Therefore, the heat transfer coefficient is k 0.771 W/m.°C = 0.00165 → h = = 467 W/m 2 . The energy balance for this steady-flow system can be expressed as E −E = ΔE system 1in424out 3 1 424 3 Net energy transfer by heat, work, and mass Change in internal, kinetic, potential, etc. Analysis We take the heating duct as the system. 3 The furnace operates continuously. Properties The properties of air at 1 atm pressure and the free stream temperature of 20°C are (Table A-15) k = 0.02514 W/m.°C υ = 1.516 × 10 -5 m 2 /s Insulation μ ∞ = 1.825 × 10 −5 kg/m.s μ s , @ −196 °C = 5.023 × 10 −6 kg/m.s Pr = 0.7309 Analysis (a) When there is no insulation, D = Di = 4 m, and the Reynolds number is V D [(40 × 1000/3600) m/s](4 m) Re = ∞ = 2.932 × 10 6 υ 1.516 × 10 −5 m 2 /s The Nusselt number is determined from Nu = [ ] (μ hD = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 ⎛⎜ ∞ k ⎝ μs [= 2 + 0.4(2.932 × 10) 6 0.5 Do Wind 20°C 40 km/h Di Nitrogen tank -196°C 1/ 4 ⎞ ⎟⎠ + 0.06(2.932 × 10) 6 2/3](0.7309) 0.4 ⎛ ⎜ 1.825 × 10 −5 ⎞ ⎜ 5.023 × 10 −6 ⎝ 1/ 4 ⎞ ⎟ ⎠ = 2333 0.02514 W/m.°C k Nu = (2333) = 14.66 W/m 2 .°C D 4m The rate of heat transfer to the liquid nitrogen is Q& = hAs (Ts − T∞) = h(πD 2)(Ts − T∞) and h= = (14.66 W/m 2 .°C)[π (4 m) 2][(20 − (−196)] °C = 159,200 W The rate of evaporation of liquid nitrogen then becomes Q& 159.2 kJ/s Q& = m& hif → m& = = = 0.804 kg/s hif 198 kJ/kg (b) Note that after insulation the outer surface temperature and diameter will change. ° F)(1 ft) Ro = Rconv,2 = 1 1 1 h. Temperature, on the other hand, is a scalar quantity. × 10−8 Btu / h · ft 2 · R 4)T24 x L 0.5 ft Solving for T2 gives the outer surface temperature to be T2 = 487.7 R Then the rate of heat transfer through the wall becomes T −T (520 − 487.7) R q& = k 1 2 = (12 . Assumptions 1 Heat conduction in each geometry is one-dimensional. Then for heat transfer purposes the flange section is equivalent to 214 W Equivalent legth = = 0.167 m = 16.7 cm 1279 W/m Therefore, the flange acts like a fin and increases the heat transfer by 16.7/2 = 8.35 times. 2 W → T s = T∞ + = = 193°C Q& = hAs (T s − T∞) 2 hAs (18 W/m .°C)(0.00010 21 m 2) 3-25 A circuit board houses 100 chips, each dissipating 0.07 W. Analysis The critical radius of plastic insulation for the spherical ball is 2 k 2(013 . Assumptions 1 Heat transfer through the wall is steady since there is no indication of change with time. 3 Radiation effects are negligible 4 Heat transfer from the back side of the plate is negligible. × 12 . The rate of heat transfer to the iced water in the tank and the amount of ice at 0° C that melts during a 24-h period are to be determined. Properties The thermal conductivity and emissivity are given to be k = 8.4 W/m·°C and ε = 0.7. Analysis (a) Taking the direction normal to the surface of the wall to be the x direction with x = 0 at the left surface, and the mathematical formulation of this problem can be expressed as and d 2T =0 dx 2 dT (L) 4] = h[T − T] + εσ [(T + 273) 4 − T 4] −k = h[T(L) − T∞] + εσ [T (L) 4 − Tsurr 2 2 ∞ surr dx T (L) = T2 = 45° C (b) Integrating the differential equation twice with respect to x yields Tsurr dT = C1 dx T (x) = C1x + C2 45°C ε where C1 and C2 are arbitrary constants. Then the heat transfer coefficient 30, which corresponds to λ 1 = 3.0372 and A1 = 19898 can be determined from Bi = hro kBi (0.45 W/m.°C)(30) → h = = = 156.9 W/m 2 .°C k ro (0.08603 m) This value seems to be larger than expected for problems of this kind. 2 Heat transfer from the base of the ice block to the table is negligible. Properties The thermal properties of the steaks are ρ = 970 kg/m3, Cp = 1.55 kJ/kg.°C, k = 1.40 •1 W/m.°C, α = 0.93 × 10 −6 m 2 / s , ε = 0.95, and hif •2 •3 = 187 kJ/kg. 5-75 Chapter 5 Numerical Methods in Heat Conduction 5-85 "!PROBLEM 5-85" "GIVEN" L=0.08 "[m]" k=28 "[W/m-C]" alpha=12.5E-6 "[m^2/s]" T_i=100 "[C]" g_dot=1E6 "[W/m^3]" T_infinity=20 "[C]" h=35 "[W/m^2-C]" DELTAx=0.02 "[m]" "time=300 [s], parameter to be varied" "ANALYSIS" M=L/DELTAx+1 "Number of nodes" DELTAt=15 "[s]" tau=(alpha*DELTAt)/DELTAx^2 "The technique is to store the temperatures in the parametric table and recover them (as old temperatures) using the variable ROW. 3-149E The U-value of a wall for 7.5 mph winds outside are given. The numerical methods are usually more involved and the solutions are obtained at a number of points, but they are applicable to any geometry subjected to any kind of thermal conditions. Analysis The Biot number is Bi = hro (800 W/m 2 .°C)(0.0275 m) = 36.2 = (0.607 W/m.°C) k The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, Water 94 4°C Egg Ti = λ1 = 3.0533 and A1 = 1.9925 Then the Fourier number and the time period become θ 0,sph = 2 2 T0 − T∞ 60 − 94.4 = A1e −λ1 τ → = (1.9925)e −(3.0533) τ → τ = 0.1727 Ti − T∞ 8 − 94.4 which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. 3 The person is completely surrounded by the interior surfaces of the room. The rate of heat loss from the pipe is to be determined. 2 Radiation heat transfer is negligible. 6-10 ⎞ ⎟ ⎠ Chapter 6 Fundamentals of Convection 6-39 The oil in a journal bearing is considered. The left and right surfaces of the wall are maintained at uniform temperatures. Thus, ⎛ 1kW ⎞ ⎟ = 1.94 kW W& e,in = Q& out = 7000kJ/h ⎜ ⎝ 3600kJ/h ⎠ AIR We 1-13 Chapter 1 Basics of Heat Transfer 1-34 A hot copper block is dropped into water in an insulated tank. Assumptions 1Thermal properties of potato are constant, and can be taken to be the properties of water. ° C)(0.5 m) 1 1 = = = 0.6631 ° C / W 2 ho A (40 W / m . The temperature of the wire 2 mm from the center is to be determined in steady operation. The rate of evaporation of liquid nitrogen in the tank as a result of the heat gain from the surroundings for the cases of no insulation, 5-cm thick fiberglass insulation, and 2-cm thick superinsulation are to be determined. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are 7-56 D Chapter 7 External Forced Convection As = NπDL = 64π(0.021 m)(1 m) = 4.222 m 2 m& = m& i = ρ i V(N T S T L) = (1.225 kg/m 3)(3.8 m/s)(8)(0.05 m)(1 m) = 1.862 kg/s Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become ⎛ Ah Te = Ts − (Ts − Ti) exp⎜ − s ⎜ m& C p ⎝ ΔTln = 2 2 ⎞ ⎛ ⎞ = 90 − (90 − 15) exp⎜ − (4.222 m)(87.5 W/m · °C) ⎟ = 28.42°C ⎜ (1.862 kg/s)(1007 J/kg · °C) ⎟ ⎟ ⎝ ⎠ ⎠ (Ts − Ti) − (Ts − Te) (90 − 15) − (90 − 28.42) = = 68.07°C ln[(Ts − Ti) /(Ts − Te)] ln[(90 − 15) /(90 − 28.42)] Q& = hAs ΔTln = (87.5 W/m 2 · °C)(4.222 m 2)(68.07°C) = 25,148 W For this square in-line tube bank, the friction coefficient corresponding to ReD = 9075 and SL/D = 5/2.1 = 2.38 is, from Fig. 2 The thermal properties of turkeys are constant. Analysis This is a transient heat conduction, and the rate of heat transfer will decrease as the drink warms up and the temperature difference between the drink and the surroundings decreases. Concrete block, light weight, 200 mm 3. Inside surface, still air R-value, m2.°C/W Between At studs studs 0.044 0.044 0.14 0.14 0.23 0.23 3.696 --0.98 0.079 0.079 0.12 0.12 2 Total unit thermal resistance of each section, R (in m2.°C/W) The U-factor of each section, U = 1/R, in W/m2.°C Area fraction of each section, farea Overall U-factor, U = Σfarea,iUi = 0.80×0.232+0.20×0.628 Overall unit thermal resistance, R = 1/U 4.309 1.593 0.232 0.628 0.80 0.20 2 0.311 W/m .°C 3.213 m2.°C/W Therefore, the R-value of the existing wall is R = 3.213 m2.°C/W. The rate of heat transfer, the pressure drop of exhaust gases, and the temperature rise of water are to be determined. 4-107 Chapter 4 Transient Heat Conduction 4-116E A plate, a long cylinder, and a sphere are exposed to cool air. Flow over Flat Plates 7-1 Chapter 7 External Forced Convection 7-11C The friction coefficient represents the resistance to fluid flow over a flat plate. The density of the wall is ρ, the specific heat is C, and the area of the wall normal to the direction of heat transfer is A. Assumptions 1 Heat conduction is steady and one-dimensional since there is no change with time and there is thermal symmetry about the midpoint. Then the time required for the temperature of the center of the egg to reach 70°C is determined to be t= τro 2 (0.198)(0.0275 m) 2 = = 1068 s = 17.8 min α (0.14 × 10 −6 m 2 /s) 4-19 Chapter 4 Transient Heat Conduction 4-35 "!PROBLEM 4-35" "GIVEN" D=0.055 "[m]" T_i=8 "[C]" "T_o=70 [C], parameter to be varied" T_infinity=97 "[C]" h=1400 "[W/m^2-C]" "PROPERTIES" k=0.6 "[W/m-C]" alpha=0.14E-6 "[m^2/s]" "ANALYSIS" Bi=(h*r_o)/k r_o=D/2 "From Table 4-1 corresponding to this Bi number, we read" lambda_1=1.9969 A_1=3.0863 (T_o-T_infinity)/(T_i-T_infinity)=A_1*exp(-lambda_1^2*tau) time=(tau*r_o^2)/alpha*Convert(s, min) To [C] 50 55 60 65 70 75 80 85 90 95 time [min] 39.86 42.4 45.26 48.54 52.38 57 62.82 70.68 82.85 111.1 4-20 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 55 60 65 70 75 T o [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 4-36 Large brass plates are heated in an oven. Analysis We take the air in the room as the system. Assumptions 1 Heat conduction is steady and one-dimensional since the wall is large relative to its thickness, and the thermal conditions on both sides of the wall are uniform. The drag force the wind exerts on the windshield is to be determined. Analysis The mass of the water is Ice, m w = ρV = (1kg/L)(0.2 L) = 0.2kg 0°C We take the ice and the water as our system, and disregard any heat and mass transfer. Dividing the equation above by AΔr gives − T − Tt 1 Q& r + Δr − Q& r = ρ C t + Δt A Δr Δt Taking the limit as Δr → 0 and Δt → 0 yields ∂T 1 ∂ ⎛ ∂T ⎞ ⎟ kA ⎟ = ρC A ∂r ⎜ ∂r ⎠ ∂t since, from the definition of the derivative and Fourier's law of heat conduction, Q& r + Δr − Q& r ∂Q ∂ ⎛ ∂T ⎞ = ⎜ − kA ⎟ Δr → 0 ∂ r ∂r ⎝ ∂r ⎠ Δr lim Noting that the heat transfer area in this case is A = 4πr 2 and the thermal conductivity k is constant, the one-dimensional transient heat conduction equation in a sphere becomes 1 ∂ ⎛ 2 ∂T ⎞ 1 ∂T ⎜r ⎟= r 2 ∂r ⎝ ∂r ⎠ α ∂ t where α = k / ρC is the thermal diffusivity of the material. Assumptions Absorption of solar radiation by water is modeled as heat generation. The center temperature of the hot dog is do be determined by treating hot dog as a finite cylinder and an infinitely long cylinder. Using the finite difference method, 10 equations for 10 unknown temperatures are determined to be" h_o*l*1/2*(T_o-T_1)+k*l/2*(T_2-T_1)/l+k*l/2*(T_5-T_1)/l+epsilon*sigma*l/2*(T_sky^4(T_1+273)^4)=0 "Node 1" h_o*l*(T_o-T_2)+k*l/2*(T_1-T_2)/l+k*l/2*(T_3-T_2)/l+k*l*(T_6-T_2)/l+epsilon*sigma*l*(T_sky^4(T_2+273)^4)=0 "Node 2" h_o*l*(T_o-T_3)+k*l/2*(T_2-T_3)/l+k*l/2*(T_4-T_3)/l+k*l*(T_7-T_3)/l+epsilon*sigma*l*(T_sky^4(T_3+273)^4)=0 "Node 3" h_o*l*(T_o-T_4)+k*l/2*(T_3-T_4)/l+k*l/2*(T_8-T_4)/l+epsilon*sigma*l*(T_sky^4-(T_4+273)^4)=0 "Node 4" h_i*l/2*(T_i-T_5)+k*l/2*(T_6-T_5)/l+k*l/2*(T_1-T_5)/l=0 "Node 5" h_i*l*(T_i-T_6)+k*l/2*(T_7-T_6)/l+k*l/2*(T_2-T_6)/l=0 "Node 6" h_i*l*(T_i-T_7)+k*l/2*(T_6-T_7)/l+k*l/2*(T_9-T_7)/l+k*l*(T_3-T_7)/l+k*l*(T_8-T_7)/l=0 "Node 7" h_o*l*(T_o-T_8)+k*l/2*(T_4-T_8)/l+k*l/2*(T_10-T_8)/l+k*l*(T_7-T_8)/l+epsilon*sigma*l*(T_sky^4(T_8+273)^4)=0 "Node 8" h_i*l*(T_i-T_9)+k*l/2*(T_7-T_9)/l+k*l/2*(T_10-T_9)/l=0 "Node 9" h_o*l/2*(T_o-T_10)+k*l/2*(T_8-T_10)/l+k*l/2*(T_9-T_10)/l+epsilon*sigma*l/2*(T_sky^4(T_10+273)^4)=0 "Node 10" "Right top corner is considered. Properties The thermal conductivity of aluminum is given to be 237 W/m·°C. Noting that D = D0 = 4.04 m in this case, the Nusselt number becomes Re = V∞ D [(40 × 1000/3600) m/s](4.04 m) = = 2.961× 10 6 υ 1.516 × 10 −5 m 2 /s [ ] (μ hD Nu = = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 ⎛⎜ ∞ k ⎝ μs [= 2 +

0.4(2.961 × 10) 6 0.5 1/ 4 ⎞ ⎟⎟ ⎠ + 0.06(2.961 × 10) 6 2/3](0.7309) 0.4 ⎛ ⎜ 1.825 × 10 −5 ⎟ 1.729 × 10 −5 ⎝ 0.02514 W/m.°C h = (1724) = 10.73 W/m 2 .°C 0.404 m The rate of heat transfer to the liquid nitrogen is As = πD 2 = π (4.04 m) 2 = 51.28 m 2 T∞ − Ts , tan k T∞ − Ts ,tan k Q& = = 1 Rinsulation + Rconv 2 − r1 + 4πkr1 r2 hAs [20 − (−196)]°C = = 27.4 W (2.02 − 2) m 1 + 4π (0.00005 W/m.°C)(2.02 m)(2 m) (10.73 W/m 2 .°C)(51.28 m 2) and h= The rate of evaporation of liquid nitrogen then becomes Q& = Q& 0.0274 kJ/s m& hif → m& = = = 1.38 × 10 -4 kg/s 198 kJ/kg hf T-95 1 ⎛ 1 ⎞ ⎛ ⎞ = 1724 Chapter 7 External Forced Convection 7-102 A spherical tank filled with liquid oxygen is exposed to ambient winds. 5 The minivan is modeled as a rectangular box. Therefore, there are two unknowns T1 and T2, and we need two equations to determine it. Tw Convection then. Assumptions 1 Heat transfer is steady. 2-31 ... The inner and outer surface temperatures of the ice chest remain constant at 0°C and 8°C, respectively, at all times. 3 Radiation is accounted for in the combined heat transfer coefficient. The rate of heat transfer between the pipes is to be determined. Analysis ... The junction is spherical in shape with a diameter of D = 0.0012 m. 3-24 Chapter 3 Steady Heat Conduction 3-47 Two cylindrical aluminum bars with ground surfaces are pressed against each other in an insulation sleeve. It is important to raise the internal temperature of a roast in an oven above 70°C since most microorganisms, including some that cause diseases, may survive temperatures below 70°C. Analysis The shape factor for this configuration is given in Table 3-5 to be S= 2π(4 m) 2πL = = 13.58 m ⎛ 8z ⎞ ⎛ 8(0.075 m) ⎞ ln⎜ ⎟ ln⎜ ⎟ ⎜ ⎟ ⎝ πD ⎠ ⎝ π(0.03 m) ⎠ Then rate of heat loss from the hot water in 8 parallel pipes becomes Q& = 8Sk (T − T) = 8(13.58 m)(0.75 W/m.°C)(85 − 32)°C = 4318 W 1 2 32°C 85°C z D L=4m The surface temperature of the wall can be determined from As = 4(4 m)(8 m) + 4(8 m) 2 (two sides) Q& 4318 W → T s = T∞ + = 32°C + = 37.6°C Q& = hAs (Ts − T∞) hAs (12 W/m 2 .°C)(64 m 2) 3-100 z Chapter 3 Steady Heat Conduction Special Topic: Heat Transfer Through the Walls and Roofs 3-134C The R-value of a wall is the thermal resistance of the wall per unit surface area. Once the unit thermal resistances and the U-factors for the air space and stud sections are available, the overall average thermal resistance for the entire wall can be determined from Roverall = 1/Uoverall where Uoverall = (Ufarea)air space + (Ufarea)stud and the value of the area fraction farea is 0.80 for air space and 0.20 for the ferming since the headers that constitute a small area The total rate of heat loss from the hot water and the temperature drop of the hot water in the pipe are to be determined. Properties The thermal properties of air are (Table A-15) k = 0.02439 W/m.°C, υ = 1.426 × 10 -5 m 2 /s, Pr = 0.7336 and Analysis The outer diameter of insulated pipe is Do + 4.6+2×3=11.6 cm = 0.116 m. 3 Heat loss by radiation is negligible. 3-50C The thermal resistance network approach will give adequate results for multidimensional heat transfer problems if heat transfer occurs predominantly in one direction. The small and the large cylinders are to be determined. Analysis The temperature at a specified time. 3-137C Radiant barriers are highly reflective materials that minimize the radiation heat transfer between surfaces. Analysis Noting that the surface of the spoon is to be determined. Also, the rate of ... losses. 4-14b we have T − T∞ 88 − 94 ⎫ = = 0.17⎪ To − T∞ 59 − 94 k ⎪ 1 = = 0.15 ⎬ ro Bi hro r ⎪ = =1 ⎪⎭ ro ro Water 94°C ... 94C The rate of freezing can affect color, tenderness, and drip. ° F)(0.000278 ft) 2 = 10.95 ft -1 Noting that x = L = 7/12=0.583 ft ... Therefore, the temperature difference across the exposed section of the spoon handle is ΔT = Tb − Ttip = (200 − 75.4)°F = 124.6 °F ... Analysis The total rate of heat dissipation from the aluminum plate and the total heat transfer area are Q& = 4 × 15 W = 60 W ... m 2) 15 Ts 1-43 Chapter 1 Basics of Heat Transfer 1-84 A styrofoam ice chest is initially filled with 40 kg of ice at 0°C. The ... 1 1 = = 0.0229 °C/W ho Ao (20.167 W/m 2 .°C)(2.168 m 2) = Ri + R pipe + Ro = 0.004 + 0.0000038 + 0.0229 = 0.0273 °C/W ... water must be Q& = m& C p ΔT → m& = m& = ρVAc → V = Q& 2930 J/s = = 0.234 kg/s C p ΔT (4180 J/kg. 2 Heat conduction ... kinetic, potential, etc. / 12) ft 1 1 = = = 0.2539 h. 2-40 A spherical container of inner radius r1 , outer radius r2 , and thermal ... 6 h (1 pm) 24 17.5 16.1 15.9 18.1 24.8 38.8 61.5 12 h (7 48 21.4 22.9 25.8 30.2 34.6 37.2 35.8 pm) 18 h (1 72 22.9 24.6 26.0 ... am) 48 h (7 192 23.0 24.6 25.5 25.2 23.7 20.7 16.3 am) The rate of heat transfer from the Trombe wall to the interior of the house ... A[(T0 + T0) / 2 − Tin]Δt Therefore, the amount of heat transfer during the first time step (i = 1) or during the first 15 min period is ... house which represents a heat loss. Analysis The temperature difference between the center and the surface of the fuel rods ...

)(82 − 8)°C = 6974 W Steam pipe The amount of heat loss during a 10-hour period is Q = Q& Δt = (6.974 kJ/s)(10 × 3600 s) = 2.511× 10 5 kJ (per day) The steam generator has an efficiency of 80%, and steam heating is used for 110 days a year. Analysis The effective conductivity of the multilayer circuit board is first determined to be (kt) copper = 4[(386 W / m. The wall shear also contributes to lift (unless the body is very slim), but its contribution is usually small. 2 Heat transfer is one-dimensional since the wall is large relative to its thickness, and there is thermal symmetry about the center plane. 7-82 Chapter 7 External Forced Convection Review Problems 7-90 Wind is blowing parallel to the walls of a house. 4 Heat loss through the upper part of the iron is negligible. 2 Heat transfer through the wall can be approximated to be one-dimensional. Also, D = 0.008 . We evaluate the air properties at the assumed mean temperature of 250°C (will be checked later) and 1 atm (Table A-15): k = 0.04104 W/m-K ρ = 0.6746 kg/m3 Cp =1.033 kJ/kg-K Pr = 0.6946 μ = 2.76×10 kg/m-s Prs = Ts = 0.7154 -5 Also, the density of air at the inlet temperature of 300°C (for use in the mass flow rate calculation at the inlet) is ρi = 0.6158 kg/m3. 4 The wing is approximated as a cylinder of elliptical cross section whose minor axis is 30 cm. 7-27b, f = 0.33. This is not surprising since the convection currents that set in in the thicker air space offset any additional resistance due to a thicker air space. 2 Radiation effects are negligible. Properties The properties of the orange are approximated by those of water at the average temperature of about 5°C, k = 0.571 W/m.°C and −6 2 α = k / ρC p = 0.571 / (1000 × 4205) = 0.136 × 10 m / s (Table A-9). The exposed surface area of the furnace is Ao = 2 Abase + Aside = 2πr 2 + 2πrL = 2π (1.5 m) 2 + 2π (1.5 m)(6 m) = 70.69 m 2 The rate of heat loss from the furnace before the insulation is installed is Q& = ho Ao (Ts − T∞) = (30 W/m 2 .°C)(70.69 m 2)(75 − 27)°C = 101,794 W Noting that the plant operates 52×80 = 4160 h/yr, the annual heat lost from the furnace is Q = Q& Δt = (101794 . Properties The average specific heats are given to be 0.6 kJ/kg.°C for the teapot and 4.18 kJ/kg.°C for water. ° C)(24 m2) L 0.3 m Q& 14°C 3-2 6° C Chapter 3 Steady Heat Conduction 3-18 The two surfaces of a window are maintained at specified temperatures. 4-3C The lumped system analysis is more likely to be applicable for the body allowed to cool in the air since the Biot number is proportional to the convection heat transfer coefficient, which is larger in water than it is in air because of the larger thermal conductivity of water. Air space, 20 mm, reflective with ε =0.05 5b. Air space, 90 mm, nonreflective 6b. The atmospheric pressure in atm unit is 1 atm P = (18.8 kPa) = 01855 . 7-12C The friction and the heat transfer coefficients change with position in laminar flow over a flat plate. − 914 . The power rating of the resistance heater and the average velocity of the water are to be determined. × 1 = 012 . 1-116C The metabolic rate is proportional to the size of the body, and the metabolic rate of women, in general, is lower than that of men because of their smaller size. Assumptions 1 The rib is a homogeneous spherical object. Analysis We first find the volume and equivalent radius of the chickens: V = m / ρ = 1700g/(0.95g/cm³) = 1789cm³ ⎛ 3 ⎞ ro = ⎜ V ⎟ ⎝ 4π ⎠ 1/ 3 ⎛ 3 ⎞ = ⎜ 1789 cm³ ⎟ ⎝ 4π ⎠ 1/ 3 = 7.53 cm = 0.0753 m Chicken Then the Biot and Fourier numbers become hr0 (440 W/m .°C)(0.0753 m) = = 73.6 k 0.45 W/m.°C α t (0.13 × 10 −6 m²/s)(2.5 × 3600 s) = 0.2063 τ= 2 = (0.0753 m)² r0 Ti = 15°C 2 Bi = Brine -10°C Note that τ = 0.207 > 0.2 , and thus the one-term solution is applicable. Nodes 1 through 12 are interior nodes, and thus for them we can use the general finite difference relation expressed as kA 13 • . The system of 6 equations with 6 unknowns constitutes the finite difference formulation of the problem. Air 500 ft/min 120°F Properties The properties of air at 1 atm and the film temperature of T f = (180 + 120) / 2 = 150°F are (Table A-15) k = 0.01646 Btu/h.ft.°F Power transistor D = 0.22 in υ = 0.210 × 10 -3 ft 2 /s Pr = 0.7188 Analysis The Reynolds number is V D (500/60 ft/s)(0.22/12 ft) Re = ∞ = = 727.5 υ 0.210 × 10 −3 ft 2 /s The Nusselt number corresponding to this Reynolds number is hD 0.62 Re 0.5 Pr 1 / 3 Nu = = 0.3 + 1/ 4 k 1 + (0.4 / Pr)2 / 3 [ ] ⎡ ⎛ Re ⎞ 5 / 8 ⎤ ⎢1 + ⎜ ⎟ ⎥ ⎢ ⎝ 282,000 ⎠ ⎥ ⎦ 4/5 5/8 0.62(727.5) 0.5 (0.7188)1 / 3 ⎡ ⎛ 727.5 ⎞ ⎤ ⎥ = 0.3 + 1 + ⎢ ⎥ ⎥ 1/ 4 ⎢ ⎝ 282,000 ⎠ ⎥ ⎦ 1 + (0.4 / 0.7188)2 / 3 k 0.01646 Btu/h.ft.°F and h = Nu = (13.72) = 12.32 Btu/h.ft 2 .°F D (0.22 / 12 ft) Then the amount of power this transistor can dissipate safely becomes Q& = hAs (T s − T∞) = h(πDL)(Ts − T∞) [ ] 4/5 = 13.72 = (12.32 Btu/h.ft 2 .°F)[π (0.22/12 ft)(0.25/12 ft)](180 − 120)°C = 0.887 Btu/h = 0.26 W (1 W = 3.412 Btu/h) 7-91 Chapter 7 External Forced Convection 7-99 Wind is blowing over the roof of a house. Air V∞ = 6 m/s T∞ = 30°C Ts = 120°C Properties The atmospheric pressure in atm is L 1 atm P = (83.4 kPa) = 0.823 atm 101.325 kPa For an ideal gas, the thermal conductivity and the Prandtl number are independent of pressure, but the kinematic viscosity is inversely proportional to the pressure. Properties The properties of the rib are given to be k = 0.45 W/m.°C, ρ = 1200 kg/m3, Cp = 4.1 kJ/kg.°C, and α = 0.91×10-7 m2/s Analysis (a) The radius of the rib is determined to be m = ρV → V = V= m ρ 3.2 kg = 1200 kg/m 3 = 0.00267 m 3 4 3V 3 3(0.00267 m 3) = = 0.08603 m πro 3 → ro = 3 3 4π 4π R i b The Fourier number is τ= αt ro 2 = (0.91 × 10 −7 m 2 /s)(4 × 3600 + 15 × 60)s (0.08603 m) 2 4-33 = 0.1881 O v Chapter 4 Transient Heat Conduction which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. 3 Thermal conductivity of the walls is constant. Properties The heat of vaporization and density of liquid oxygen at 1 atm are given to be 213 kJ/kg and 1140 kg/m3, respectively. The melting starts at the outer surfaces of the top surface when the temperature drops below 0° C at this location. Assumptions 1 Steady operating conditions exist. The region about a node whose properties are represented by the property values at the nodal point is called the volume element. In order to reduce the number of parameters, some variables are grouped into dimensionless quantities. ° C . 2 Heat transfer is one-dimensional since heat transfer from the side surfaces is disregarded 3 Thermal conductivities are constant. 3 The thermal properties of the carcass are constant. 1 m The left surface temperature is given to be T0 =80°C. Replacing the 3 7 7 symmetry lines by insulation and utilizing the • • • mirror-image concept, the finite difference equations are obtained to be as follows: 4 8 8 • • T2 •− T1 l T0 − T1 l T5 − T1 Node 1 (heat flux): q& 0 l + k +k + kl =0 2 l 2 l l Insulated Symmetry Node 2 (interior): T0 + T1 + T3 + T6 − 4T2 = 0 Node 3 (interior): Node 4 (insulation): T0 + T2 + T4 + T7 − 4T3 = 0 T0 + 2T3 + T8 − 4T4 = 0 Node 5 (heat flux): q& 0 l + k T − T5 l T1 − T5 + kl 6 +0 = 0 2 l l Node 6 (interior): T2 + T5 + T6 + T7 − 4T6 = 0 Node 7 (interior): T3 + T6 + T7 + T8 − 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 − 4T8 = 0 where l = 0.1 m, k = 23 W/m·°C, T0 =0°C, and q& 0 = Q& 0 / A = (6000 W)/(5 × 0.5 m 2) = 2400 W/m 2 . 3 Thermal conductivity of the ground is constant. 5-95 Chapter 5 Numerical Methods in Heat Conduction 5-93 The roof of a house initially at a uniform temperature is subjected to convection and radiation on both sides. Example: A hot surface on the ceiling. We observe that the pressure in the room remains constant during this process. The mathematical formulation, the variation of temperature in the wall, and the temperature of the insulated surface are to be determined for steady one-dimensional heat transfer. ° F Copper (kt) . k (2.22 W/m.°C) αt L2 αt ro 2 = = (0124 . 4 Heat transfer by radiation is negligible. Properties The thermal conductivities are given to be kA = kF = 2, kB = 8, kC = 20, kD = 15, kE = 35 W/m·°C. (b) The driving force for electric current flow is the electric potential difference (voltage). 5 The exposed surface area of the transistor is taken to be equal to its base area. Then the maximum velocity and the Reynolds number V=4 m/s based on the maximum velocity become Vmax ST 0.015 = V= (4 m/s) = 8.571 m/s 0.015 − 0.008 ST − D Re D ρV D (1.316 kg/m 3)(8.571 m/s)(0.008 m) = max = = 5294 μ 1.705 × 10 −5 kg/m · s SL Ts=-20°C Ti=0°C ST The average Nusselt number is determined using the proper relation from Table 7-2 to be D Nu D = 0.35(S T / S L) 0.2 Re 0D.6 Pr 0.36 (Pr/ Prs) 0.25 = 0.35(0.015 / 0.015) 0.2 (5294) 0.6 (0.7375) 0.36 (0.7375 / 0.7408) 0.25 = 53.73 Since NL > 16. Therefore, the watermelon can be considered to be a semi-infinite medium 2 The thermal properties of the watermelon are constant. The pipe is to be insulated with adequate fiber glass insulation to prevent freezing of water in the pipe. 3 The thermal properties of the turkey are constant. Assumptions 1 Heat transfer through the refrigerator walls is steady since the temperatures of the food compartment and the kitchen air remain constant at the specified values. Assumptions 1 Heat transfer through the pin fin is given to be Tsurr steady and one-dimensional, and the thermal conductivity to be Convection constant. ° F / Btu R Q& 2 × 10 4 Btu / h total,new new After 0.01 in thick layer of limestone forms, the new value of thermal resistance and heat transfer rate are determined to be Rlimestone, i Rlimestone,i Rlimestone,i Rtotal,w /lime Rtotal, new HX Rlimestone, o T∞1 T∞2 ln(r1 / ri) ln(0.5 / 0.49) = = = 0.00189 h. 2 Thermal properties of the chip are constant. 2 Heat conduction in the orange is one-dimensional in the radial direction because of the symmetry about the midpoint. Assumptions The properties of the aluminum ball are constant. The finite difference formulation of this problem is to be obtained, and the nodal temperatures under steady conditions as well as the rate of heat transfer through the wall are to be determined. 7-60 Chapter 7 External Forced Convection 7-67 Combustion air is heated by condensing steam in a tube bank. Analysis We treat the surfaces of this cylindrical furnace as plain surfaces since its diameter is greater than 1 m, and disregard the curvature effects. Then the pressure drop across the tube bank becomes ΔP = N L fχ 2 ρVmax (1.204 kg/m 3)(6.552 m/s) 2 = 8(0.22)(1) 2 2 ⎛ 1N ⎞ ⎜ 1 kg · m/s 2 ⎟ ⎝ ⎠ = 45.5 Pa ⎠ Discussion The arithmetic mean fluid temperature is (Ti + Te)/2 = (15 + 29.1)/2 = 22.1°C, which is fairly close to the assumed value of 20°C. Analysis Using the energy balance approach and taking the direction of all heat transfers to be towards the node under consideration, the finite difference formulations become Radiation A B T −T Node 0 (at left boundary): k A A 1 0 = 0 → T1 = T0 Insulated Δx Tsurr Δx ε T0 − T1 T2 − T1 Node 1 (at the interface): k A A + kB A =0 • • Δx Δx 0• 2 1 4 Node 2 (at right boundary): εσA(Tsurr − T24) + k B A T1 − T2 =0 Δx 5-7 Chapter 5 Numerical Methods in Heat Conduction 5-21 A plane wall with variable heat generation and variable thermal conductivity is subjected to specified heat flux q& 0 and convection at the left boundary (node Convectio g(x) Radiation 0) and radiation at the right boundary (node 5). The energy balance for this steady-flow system can be expressed in the rate form as E& − E& out 1in424 3 = Rate of net energy transfer by heat, work, and mass Ē0 (steady) ΔE& system 144 42444 3 =0 → E& in = E& out Rate of change in internal, kinetic, potential, etc. All we need to do is replace the terms for ice by the ones for cold water at 0°C: (ΔU)cold water + (ΔU)water = 0 [mC (T2 − T1)]cold water + [mC (T2 − T1)]water =0 Substituting, [mcold water (4.18 kJ/kg·°C)(5 - 0)°C] + (0.2 kg)(4.18 kJ/kg·°C)(5-20)°C = 0 It gives m = 0.6 kg = 600 g Discussion Note that this is 17 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks. Then the temperature at the center line (r = 0) is determined by substituting the known quantities to be g& 2 g&r0 T (0) = T ∞ + r0 + 4k 2h T (r) = T∞ + 2 = 212°F + (1800 Btu/h.in 3)(0.25 in) 2 ⎛ 12 in ⎞ (1800 Btu/h.in 3)(0.25 in) ⎛ 12 in ⎞ ⎟ + ⎜ ⎟ = 290.8°F 4 × (8.6 Btu/h.ft.°F) 2 × (820 Btu/h · ft 2 · °F) ⎝ 1 ft ⎠ ⎝ 1 ft ⎠ Thus the centerline temperature will be about 80°F above the temperature of the surface of the wire. θ (L, t) wall = 2 2 T (x , t) − T∞ = A1 e −λ1 τ cos(λ1 L / L) = (1.0050)e − (0.164) (5.424) cos(0.164) = 0.857 Ti − T∞ Then the center temperature of the top surface of the cylinder becomes ⎡ T (L,0, t) − T∞ ⎤ = θ (L, t) wall × θ o,cyl = 0.857 × 0.577 = 0.494 ⎥ ⎢ short ⎣ Ti − T∞ ⎦ cylinder T (L,0, t) − 20 = 0.494 → T (L,0, t) = 84.2°C 150 − 20 (c) We first need to determine the maximum heat can be transferred from the cylinder [ ] m = ρV = ρπro 2 L = (8530 kg/m 3) π(0.04 m) 2. This assumption results in negligible error in heating and air-conditioning applications. For convenience, we choose the time step to be Δt = 4 s. That is, ⎛ x ⎛ hx h 2αt ⎞ ⎡ ⎛ x ⎞ h αt ⎞ ⎤ ⎟ ⎛ hx + + 33.3 ⎥ ⎥ ⎥ ≅ 0 exp⎜ + 2 ⎥ ⎥ erfc⎜ = exp⎜ + 1138 ⎥ ⎥ erfc⎜ ⎥ ⎢ k⎟ 2 αt k ⎠ ⎝ ⎠ k ⎠ ⎥ ⎝ ⎠ ⎝ k ⎝ 2 αt ⎠ ⎦ ⎝ ⎝ Therefore, the temperature distribution relation simplifies to ⎛ x T (x, t) − Ti = erfc⎜ T∞ − Ti ⎝ 2 αt ⎞ ⎛ x ⎟ → T (x, t) = Ti + (T∞ − Ti)erfc⎜ ⎟ ⎜ ⎠ ⎝ 2 αt ⎞ ⎟ ⎟ ⎠ Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0: ⎛ 0 T (0,10 h) = Ti + (T∞ − Ti)erfc⎜ ⎜ ⎝ 2 αt x = 0.1m: ⎞ ⎟ = Ti + (T∞ − Ti)erfc (0) = Ti + (T∞ − Ti) × 1 = T∞ = −10°C ⎟ ⎠ ⎛ ⎞ 0.1 m ⎟ T (0.1 m,10 h) = 10 + (−10 − 10)erfc⎜ − 5 2 ⎜ 2 (1.6 × 10 m /s)(10 h × 3600 s/h) ⎟ ⎟ ⎝ ⎠ = 10 − 20erfc(0.066) = 10 − 20 × 0.9257 = −8.5°C x = 0.2 m: ⎛ ⎞ 0 .2 m ⎟ T (0.2 m,10 h) = 10 + (−10 − 10)erfc⎜ − 5 2 ⎜ 2 (1.6 × 10 m /s)(10 h × 3600 s/h) ⎟ ⎟ ⎝ ⎠ = 10 − 20erfc(0.132) = 10 − 20 × 0.8519 = −7.0°C x = 0.5 m: ⎛ ⎞ 0.5 m ⎟ T (0.5 m,10 h) = 10 + (−10 − 10)erfc⎜ − 5 2 ⎜ 2 (1.6 × 10 m /s)(10 h × 3600 s/h) ⎟ ⎟ ⎝ ⎠ = 10 − 20erfc(0.329) = 10 − 20 × 0.6418 = −2.8°C 4-58 Chapter 4 Transient Heat Conduction 4-65 "!PROBLEM 4-65" "GIVEN" T_i=10 "[C]" T_infinity=-10 "[C]" h=40 "[W/m^2-C]" time=10*3600 "[s]" "x=0.1 [m], parameter to be varied" "PROPERTIES" k=0.9 "[W/m-C]" alpha=1.6E-5 "[m^2/s]" "ANALYSIS" (T_x-T_i)/(T_infinity-T_i)=erfc(x/(2*sqrt(alpha*time)))exp((h*x)/k+(h^2*alpha*time)/k^2)*erfc(x/(2*sqrt(alpha*time))+(h*sqrt(alpha*time)/ k)) x [m] 0 0.05 0.1 0.15 0.2 0.25 0.3 0.35 0.4 0.45 0.5 0.55 0.6 0.65 0.7 0.75 0.8 0.85 0.9 0.95 1 Tx [C] -9.666 -8.923 -8.183 -7.447 -6.716 -5.993 -5.277 -4.572 -3.878 -3.197 -2.529 -1.877 -1.24 -0.6207 -0.01894 0.5643 1.128 1.672 2.196 2.7 3.183 4-59 Chapter 4 Transient Heat Conduction 4 2 T x [C] 0 -2 -4 -6 -8 -10 0 0.2 0.4 0.6 x [m] 4-60 0.8 1 Chapter 4 Transient Heat Conduction 4-66 The walls of a furnace made of concrete are exposed to hot gases at the inner surfaces. Nodes 1, 2, and 3 are interior nodes, and thus for them we can use the general finite difference relation expressed as Tm −1 − 2Tm + Tm +1 g& m + = 0 → Tm −1 − 2Tm + Tm +1 = 0 (since g& = 0) , for m = 0, 1, 2, and 3 k Δx 2 The finite difference equation for node 4 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 4 and taking the direction of all heat transfers to be towards the node under consideration: Node 1 (interior) : T0 − 2T1 + T2 = 0 Node 2 (interior) : T1 − 2T2 + T3 = 0 Node 3 (interior) : T2 − 2T3 + T4 = 0 Node 4 (right surface - convection) : h(T∞ − T4) + k g T0 T3 − T4 =0 Δx where Δx = 0.1 m, k = 2.3 W/m · °C, h = 24 W/m 2 · °C, and T∞ = 15°C. °C)(350 − 50)°C = 72.7 kJ/min = 1212 W 4-117 Chapter 4 Transient Heat Conduction 4-124 A brick house made of brick that was initially cold is exposed to warm atmospheric air at the outer surfaces. ° F / Btu 2πk ins L 2π (0.020 Btu / h.ft. Using energy balances, the finite difference equations for each of the 8 nodes are obtained as follows: Node 1: q& L l l l T2i − T1i l T4i − T1i l2 l 2 T i +1 − T1i + h (T∞ − T1i) + k +k + g& 0 =ρ C 1 2 2 2 l 2 l 4 4 Δt Node 2: hl (T∞ − T2i) + k Node 3: hl (T∞ − T3i) + k i i T i − T2i l T1i − T2i l T3 − T2 l2 l 2 T2i +1 − T2i +k + kl 5 + g& 0 =ρ C 2 l 2 l l 2 2 Δt i i i i i +1 i l T2 − T3 l T6 − T3 l2 l 2 T3 − T3 +k + g& 0 =ρ C 2 l 2 l 4 4 Δt (It can be rearranged as T3i +1 = ⎛⎜1 − 4τ − 4τ ⎞ ⎛ i g& 3 l 2 hl ⎞ i hl i ⎤ T T T T τ 2 2 + + + + ⎟ 3 ∞ 6 ⎥ 4 k ⎠ k 2k ⎠ ⎝ ⎟ ⎠ ⎥ ⎠ T i − T4i l T i −T i l 140 − T4i l2 l 2 T i +1 − T4i Node 4: q& L l + k 1 4 + k + kl 5 + g& 0 =ρ C 4 2 l 2 l l ⎛ 2 Node 5 (interior): T5i +1 = (1 − 4τ)T5i + τ ⎥ ⎥ T2i + T4i + T6i + 140 + ⎝ 2 g& 0 l k 2 Δt ⎞ ⎥ ⎥ ⎠ Node 6: hl (T∞ − T6i) + k i i i i i +1 i 140 − T6i T i − T6i l T3 − T6 l T7 − T6 3l 2 3l 2 T6 − T6 + kl 5 + kl +k + g& 0 =ρ C 2 l l l 2 l 4 4 Δt Node 7: hl (T∞ − T7i) + k l 2 i i i i i +1 i 140 − T7i l T6 − T7 l T8 − T7 l2 l 2 T − T7 +k + kl + g& 0 =ρ C 7 2 l 2 l l 2 2 Δt Node 8: h (T∞ − T8i) + k i i i i +1 i l T7 − T8 l 140 − T8 l2 l 2 T − T8 +k + g& 0 =ρ C 8 2 l 2 l 4 4 Δt 5-83 Chapter 5 Numerical Methods in Heat Conduction where g& 0 = 2 ×10 7 W/m 3 , q& L = 8000 W/m 2 , l = 0.015 m, k =15 W/m·°C, h = 80 W/m2·°C, and T∞ =25°C. The convection caused by winds is natural convection for the earth, but it is forced convection for bodies subjected to the winds since for the body it makes no difference whether the air motion is caused by a fan or by the winds. This system of 8 equations with 8 unknowns is the finite difference formulation of the problem. Analysis (a) The amount of heat this resistor dissipates during a 24-hour period is Q = Q& Δt = (0.6 W)(24 h) = 14.4 Wh = 51.84 kJ (since 1 Wh = 3600 Ws = 3.6 kJ) Q& (b) The heat flux on the surface of the resistor is As = 2 q& s = πD 2 4 + πDL = 2 π (0.4 cm) 2 4 + π (0.4 cm)(1.5 cm) = 0.251 + 1.885 = 2.136 cm 2 Resistor 0.6 W Q& 0.60 W = = 0.2809 W/cm 2 As 2.136 cm 2 (c) Assuming the heat transfer coefficient to be uniform, heat transfer is proportional to the surface area. The temperature of the balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature constant are to be determined. Properties The thermal properties of brass are given to be ρ = 8530 kg / m 3 , C p = 0.389 kJ/kg · °C , k = 110 W/m · °C , and α = 3. ° C)[π (0.03)(15)]m2 T∞2 Using arithmetic mean temperature (70+67)/2 = 68.5°C for water, the heat transfer can be expressed as T∞ ,1,ave − T∞ 2 T∞ ,1,ave − T∞ 2 T∞ ,1,ave − T∞ 2 Q& = = = 1 Rtotal Rconv,i + Rpipe + Rcombined,o R conv,i + Rpipe + hcombined Ao Substituting, (68.5 − 15)° C 13,296 W = (0.000031 ° C / W) + (0.001768 ° C / W) + Solving for the combined heat transfer coefficient gives hcombined = 272.5 W/m 2 .°C 3-115 1 hcombined [π (0.035)(15)]m 2 Chapter 3 Steady Heat Conduction 3-156 An 10-m long section of a steam pipe exposed to the ambient is to be insulated to reduce the heat loss through that section of the pipe by 90 percent. An energy balance on this thin cylindrical shell element of thickness Δr during a small time interval Δt can be expressed as ΔE element Q& r − Q& r +Δr + G& element = Δt where ΔEelement = Et + Δt − Et = mC (Tt + Δt − Tt) = ρCAΔr (Tt + Δt − Tt) G& element = g&Velement = g&AΔr Substituting, T − Tt Q& r − Q& r + Δr + g& AΔr = ρCAΔr t + Δt Δt where A = 2πrL . 3 Thermal conductivity of the fiberglass insulation is constant. 4 The exposed surface area of the transistor can be taken to be equal to its base area. Properties The wall properties are given to be k = 0.70 W/m·°C, α = 0.44 × 10 −6 m 2 /s , and κ = 0.76. τ= αt ro 2 θ o ,cyl = = (3.39 × 10 −5 m 2 / s)(15 × 60 s) (0.04 m) 2 = 19.069 > 0.2 2 2 T0 − T∞ = A1e − λ 1 τ = (10038 . The maximum amount of heat transfer is Furnace m = ρV = ρπro L = (2702 kg/m 3) π (0.075 m) 2. Also, Cp = 1.007 kJ/kg·K for air at room temperature (Table A-15). Properties The thermal conductivity of iron is k = 80.2 W/m·°C (Table A-3). 4 Heat generation is uniform. The rate of heat transfer through the roof and the cost of this heat loss for 14-h period are to be determined. Then the number of nodes M becomes L 2 cm M = +1 = +1 = 5 Δx 0.5 cm The base temperature at node 0 is given to be T0 = 130°C.