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SPHERE TRANSIENT CONVECTIVE SOLUTIONS



PRINCIPLES OF HEAT TRANSFER

SEVENTH EDITION



Frank Kreith



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Convection Heat Transfer

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PROBLEM 14.32

KNOWN: Mass flow rate of gas containing pulladium (species A), which flows through a tube and deposits into pores of tube wall. Inlet mass concentration of pulladium. Mass transfer coefficient hermen gas and tube surface. Deposition rate is proportional to mass concentration of pulladium at tube surface.

FIND: (a) Expression for variation of mean species density of palladown with x. Expression for local depositon rate for take of diameter D. (b) Ratio of deposition rates at x = L and x = 0.

SCHEMATIC: $\begin{array}{c} G_{a5} \\ m, \rho_{k,mi} \\ x = 0 \end{array} \xrightarrow{p_{k,m}, h_{m}} \\ \hline \\ r_{a,s} = k \rho_{A,s} \\ r_{a,s} = k \rho_{A,s} \end{array}$

ASSUMPTIONS: (1) Steady state, (2) Constant properties, (3) Constant mass flow rate, (4) Negligible leakage of gas through porous walls.

ANALYSIS: (3) Section 8.9 develops the variation of mean species density, ρ_{Am} for the case in which the surface species concentration, ρ_{Am} is uniform. Here, however, the surface concentration will vary a the mean species density decreases with z. Under steady state conditions, the mass flux of palladium reaching the surface mass equal the mass flux of palladium depositing into the potes. Referring to Equation 8.80, where $\mathbf{n}_{A,S}^{-1}$ is the mass flux of palladium depositing into the potes. Referring to Equation 8.80, where $\mathbf{n}_{A,S}^{-1}$ is the mass flux of palladium depositing the species $\mathbf{n}_{A,S}^{-1}$ is the mass flux of palladium depositing the transmission of the potestime of the second state of the species of the spe

 $\mathbf{n}_{A,s}^* = \mathbf{h}_m(\rho_{A,s} - \rho_{A,m}) = -\mathbf{k}_l \rho_{A,s}$

Solving for the surface concentration yields $\rho_{A,0}=h_{0,0,A,M}(h_{00}-k_{1})$. Then substituting this into either expension for $u_{A,A}^{+}$ yields

 $\mathbf{n}_{A,n}^* = -U_m \rho_{A,m}, \quad U_m^{-1} = 1/h_m + 1/k_1$

Comparing this result with Equation 8.80, we see that they are analogous if we replace $h_{\rm stat}$ with $U_{\rm m}$ and $\rho_{\rm As}$ with 0. Applying the same analogy to Equation 8.84, the distribution of the mean uproves density is

downstream by the flow is $m\rho_{A,m}/\rho$, and assuming ρ to be constant, we have $\label{eq:constant} Contained...$

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FOURTH EDITION

Convective Heat Mass Transfer



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3 Radiation effects are negligible. Then the continuity equation reduces to $\partial u \ \partial v \ \partial u + = 0 \rightarrow Continuity$: = 0 $\rightarrow u = u(y)$ 3000 rpm $\partial x \ \partial y$ 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, $\lambda 1 = 1.635$ 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 v 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 $g\&r 2 g\& 2 (4 \times 107 W/m 3)(0.04 m) 2 T (0) = Ts + 0 = 80°C + = 791°C 6k 6k 6 \times (15 W/m.°C)$ 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 $\Delta x = 0.25$ m. Substituting the given quantities, the maximum allowable time step becomes $\Delta t \leq (0.125 / 12 \text{ ft}) 2 2(4.2 \times 10 - 6 \text{ ft } 2 / \text{s})[1 + (2.6 \text{ Btu/h.ft } 2 .°F)(0.125 / 12 \text{ m}) / (0.48 \text{ Btu/h.ft.}°F)] = 12.2 \text{ s}$ 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. Ci k = = 0.0136 Bi hro (440W/m². Ci (0.0753m) T = $\alpha t = 4.55$ (Fig. 4 - 15) 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 $Bi = hro (467 \text{ W/m } 2.^{\circ}C)(0.011 \text{ m}) = 6.66 \text{ k} (0.771 \text{ W/m}.^{\circ}C)$ Water 94°C The constants $\lambda 1$ and A1 corresponding to this Biot number are, from Table 4-1, Hot dog $\lambda 1 = 2.0785$ and A1 = 1.5357 The Fourier number is $\tau = \alpha t L^2 = (2.017 \times 10 - 7 \text{ m} 2/\text{s})(4 \text{ min} \times 60 \text{ s/min}) (0.011 \text{ m}) 2 = 0.4001 > 0.2$ Then the temperature at the center of the hot dog is determined to be θ o, cyl = 2 2 T0 - T ∞ = A1e $-\lambda 1 \tau$ = (1.5357)e -(2.0785) (0.4001) = 0.2727 Ti $-T \infty$ 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 $\Delta y \Delta y T2i + 1 - T1i + 1 \Delta x T4 - T1 \Delta x \Delta y T1 - T1 + k (Ti - T1i + 1) + k = \rho C \Delta y 2 2 \Delta x 2 2 \Delta t T i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - T2i + 1 \Delta y T3i + 1 - 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T9i + 1 \Delta x T6 - T9 \Delta x \Delta y T9 - T9$ (To - T9i +1) + k + k = $\rho C 2 2 \Delta y 2 \Delta x 2 2 \Delta t$ where k = 0.84 W/m.°C, $\alpha = k / \rho C = 0.39 \times 10 - 6$ m 2/s, Ti = To = -3°C hi = 6 W/m2.°C, ho = 20 W/m2.°C, $\Delta x = 0.002$ m, and $\Delta y = 0.01$ m. 1-23C Warmer. T2 = 15°C Analysis The shape factor for this configuration is given in Table 3-5 to be $2\pi L S = 2(4z - D12 - D22)|cosh - 1|| 2 D1 D 2 (JD = 5 cm z = 40 cm <math>2\pi(8 m) L = 8m = 9.078 m 2 2 2)(-1|4(0.4 m) - (0.05 m) (0.05 m) (JThen the steady rate of heat transfer between the pipes becomes Q& = Sk (T - T) = (9.078 m)(0.75 W/m.°C)(60 - 1)(0.05 m) (JThen the steady rate of heat transfer between the pipes becomes Q& = Sk (T - T) = (9.078 m)(0.75 W/m.°C)(60 - 1)(0.05 m) (JThen the steady rate of heat transfer between the pipes becomes Q& = Sk (T - T) = (9.078 m)(0.75 W/m.°C)(60 - 1)(0.05 m) (JThen the steady rate of heat transfer
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[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 \rightarrow T1 = T ∞ 2 + QR ° C / W) = 40.5° C total = 40° C + (3.2 W (01484 Rtotal T -T & Q& = 1 2 \rightarrow T2 = T1 - QR board = 40.5° C - (3.2 W)(0.00694 ° C / W) = 40.5 - 0.02 \cong 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]" "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 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, block can physically be formed by the intersection of two infinite plane walls of thickness 2L=4 cm and an infinite 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, block can physically be formed by the intersection of two infinite plane walls of thickness 2L=4 cm and an infinite 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, block can physically be formed by the intersection of two infinite plane walls of thickness 2L=4 cm and an infinite plane walls 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, block can physically be formed by the intersection of two infinite plane walls of thickness 2L=4 cm and an infinite 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, block can physically be formed by the intersection of two infinite plane walls of thickness 2L=4 cm and an infinite 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, block can physically be formed by the intersection of t 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 w1*exp(lambda 1 w1*exp(lambda 1 w1*exp(lambda 1 w1*L 1/L 1) "theta L w1=(T L w1T infinity)/(T i-T infinity)/(T 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 i 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 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 $\rho = 1000 \text{ kg/m3}$ 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 $\Delta t = 5 \text{ s}$ will give the defrosting time. Properties The thermal conductivities are given to be $k = 0.16 \text{ W/m} \cdot \text{C}$ for plastic pipe and $k = 0.035 \text{ W/m} \cdot \text{C}$ for fiberglass insulation. The density of the cylinder is ρ , the specific heat is C, and the area of the cylinder is ρ , the specific heat is C, and the area of the cylinder is ρ , the specific heat is C, and the area of the cylinder is ρ , the specific heat is C, and the area of the cylinder is ρ . 0.912 \rightarrow 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 v = $1.798 \times 10 - 5$ m 2 /s Air V \approx = 150 m/min T \approx = 40°C Pr = 0.7228 Analysis The Reynolds number is V D (150/60 m/s)(0.003 m) Re = \approx = 417.1 v $1.798 \times 10 - 5$ 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 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 $0.02735 \text{ W/m.}^{\circ}\text{C} \text{ k} \text{ h} = \text{Nu} = (10.43) = 95.09 \text{ W/m} 2$. $^{\circ}\text{C} \text{ D} 0.003 \text{ m}$ Then the surface temperature of the component
becomes [Q& 4/5 = 10.43 \text{ As} = \pi \text{DL} = \pi (0.003 \text{ m})(0.018 \text{ m}) = 0.0001696 \text{ m} 2 \text{ Q} \text{ A} \text{ V} \rightarrow \text{Ts} = \text{T} \times + = 40 \text{ °C} + = 64.8 \text{ °C} \text{ Q} \text{ A} = \text{hAs} (\text{T} \text{ s} - \text{T} \times) 2 \text{ hA} (95.09 \text{ W/m} \cdot \text{C})(0.0001696 \text{ m} 2) 7-49 \text{ 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 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 = $\epsilon As \sigma$ (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 from the transformer. Also, the mesh Fourier number is $\tau = \alpha \Delta t \Delta x 2 = (015$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta x 2 = (015)$. Then the energy equation with viscous dissipation reduce to $22 (\partial u \partial t + C \Delta t \Delta t + C \Delta t + C \Delta t \Delta t \Delta t + C \Delta$ $\int The temperature gradient is determined by differentiating T(y) with respect to y, y dT \mu V 2 (= |1 - 2| dy 2kL (L) The location of maximum temperature is determined by setting dT/dy = 0 and solving for y, y dT \mu V 2 (L = <math>\rightarrow y = |1 - 2| = 0.2 dy 2kL (L) 6-11$ 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 - r2) (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) = n[T (r2) - T\infty] dr The solution of the differential equation is determined by integration to be dT dT C1 r = C1 \rightarrow = \rightarrow 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 equal t 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 $M = L / \Delta x + 1 = 0.3/0.1 + 1 = 4.1-98C$ 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 Rconv, i Rpipe Rcombined, o To1 ln(r2 / r1) ln(175. 7-110 Design and Essay Problems KJ 7-101 88 110 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 1-65 Chapte 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). \times 10 -6 m2 / s)(20 min \times 60 s / min) (0.025 m) 2 4-83 = 2.208 > 0.2 Hot gases 500°C 5 cm \times 5 }= $0.284 \text{ Ti} = 20^{\circ}\text{C}$ Chapter 4 Transient Heat Conduction { 2 T (0,0,0, t) - 500 = (1.0580)e - (0.7910) 2 (2.208) } = 0.0654 \rightarrow \text{T} (0,0,0, t) = 469^{\circ}\text{C} After 60 minutes $\tau = \alpha t 2 \text{ L} = (115 \text{ . Then the pressure drop across the tube bank becomes } \Delta P = \text{N L fx} 2 \rho \text{Vmax} (0.6746 \text{ kg/m 3})(6.102 \text{ m/s}) 2 = 16(0.18)(1) 2 2$ $(1N \mid 1 \text{ kg} \cdot \text{m/s } 2 \setminus) = 36.2 \text{ Pa} \mid / (c)$ The temperature rise of water C p, water ΔT water ΔT water $\neq \Delta T$ water $\neq 0$ at $(15.425 \text{ kW} = 4.6^{\circ}\text{C} \text{ m} \text{ k}$ water C p, water $(6 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot ^{\circ}\text{C})$ Discussion The arithmetic mean fluid temperature is $(Ti + Te)/2 = (300 + 237)/2 = 269^{\circ}\text{C}$, 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 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 $\infty \Delta x$ i +1 i +1 i T1 - T0 Δx T 0 - T 0 i +1 + q& 0 A + g& 0 (A $\Delta x / 2$) = $\rho A C kA \cdot \cdot \cdot \cdot \Delta x 2 \Delta t 0 1 2 3 4$ Right boundary node: kA T3i +1 - T4i +1 Δx T4i +1 - $T4i + hA(T \infty i + 1 - T4i + 1) + g\& 4i + 1$ ($A\Delta x / 2$) = $\rho A C \Delta x 2 \Delta t 5-76 A$ 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 $\left[= A1e - \lambda 1 \tau \cos(\lambda 1 L / L) \right] \left[A1e - \lambda 1 \tau J 0 (\lambda 1 ro / ro) \right] \\ \left[J = 20 - 20 = (1.0187)e - (0.3318) (1.0146)e - (0.3318) (1.0146)e - (0.3318) (0.2232) \cos(0.3318) (1.0146)e - (0.3407) \right] \\ \left[J = -4^{\circ} C \text{ Therefore, the ice will not start melting for at least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In the least 2 hours if its initial temperature is -4^{\circ}C or below. Analysis In this problem there is a least 2 hours if its initial temper$ 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 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]-table)/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])+k*l/2*(T_old[2]T_old[1])/l+k*l/2*(T_old[2]T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_dot_L*l/2+h*l/2*(T_old[1])/l = TableValue('Table 1',Row-1,#T[i]) end "Using the explicit finite difference approach, the eight unknown temperatures are determined to be" q_d $T_old[1])/l+g_dot*l^2/4=RhoC*l^2/4*(T[1]T_old[1])/DELTAt "Node 1" h*l*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/2*(T_old[2])/l+k*l/$ $T_old[3])/DELTAt "Node 3" q_dot_L*l+k*l/2*(T_old[1]-T_old[4])/l+k*l*(T_old[5]-T_old[4])/l+k*l*(T_old[5]-T_old[4])/l+g_dot*l^2/2*(T[4]T_old[5]+tau*(T_old[5]+tau*(T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_old[6]+T_$ $T_old[6]) + k*l/2*(T_old[3]-T_old[6])/l + k*l*(T_old[6])/l + k*l*(T_old[6])/l + k*l/2*(T_old[6])/l + k*l/2*(T_ol$ (T infinity-T old[8])+k*1/2*(T old[7]-T old[8])/1+k*1/2*(T bottomT old[8])/1+g dot*1^2/4=RhoC*1^2/4*(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] T7 [C] T8 [C] T6 [C] T7 [C] T8 [C] T6 [C] T7 Row 140 203.5 265 319 365.5 404.6 437.4 464.7 487.4 506.2 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 596.3 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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 $\Delta ke \cong \Delta pe \cong 0$) 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 $\pi \pi$ (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 · R (51 $T1 + in = 50^{\circ}F + \& (0.595 \text{lbm/s})(0.2404 \text{Btu/lbm} \cdot \circ F) \text{ mC p } 1-21 \text{ AIR } 2 \text{ Btu/s } D = 10 \text{ in Sulation are based on (1) driving the governing}$ differential equation by performing an energy balance on a differential volume element, (2) expressing the boundary conditions to determine the integration constants. and A1 = 1.2732 Orange The Fourier number is $\tau = \alpha t L2 = (0.136 \times 10)$ $-6 \text{ m } 2 / \text{s})(4 \text{ h} \times 3600 \text{ s/h})$ (0.04 m) 2 = 1.224 > 0.2 Ti = 15°C Therefore, the one-term approximate solution (or the 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 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 P=101.3 "at room temperature" C ice=c ('Ice', T ice) h if=333.7 "[k]/kg]" "ANALYSIS" m w=rho*V 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 $h = 80 \text{ W}/\text{m}^2$. The thermal properties of the defrosting plate are k = 401 W/m. $\circ C$, $\alpha = 117 \times 10 - 6 \text{ m} 2/\text{s}$, and $\epsilon = 0.90$ (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 $\tau = \alpha t L^2$ = (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 - $\lambda 1 \tau = (15618 \text{ Ti} - T)$ short cylinder becomes $[T(0,0,t) - T_{\infty}] = \theta o$, wall × θo , cyl = 1 × 0.106 = 0.106 || Ti - T_{\infty} | short cylinder T (0,0,t) - 212 = 0.106 \rightarrow T (0,0,t) = 194°F 40 - 212 4-73 Chapter 4 Transient Heat Conduction After 10 minutes $\tau = \alpha t r_2 = \theta o$, wall × θo , cyl = (0.0077 ft 2 / h)(10 / 60 h) (2.5 / 12 ft) 2 = 0.03 < 0.2 (Be cautious!) 2 2 TO - $T = A1e - \lambda 1 \tau = (1.2728)e^{-(1.5421)} (0.03) \approx 1$ 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 Rotal = 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 $t = r_3 - r_2 = 0.0362 - 0.0230 = 0.0132$ 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 are to be determined. Therefore, taking T2 the outer surface temperature of the pipe will be at 30°C or below. The length of time the block are to be determined. 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 $\Delta x = 0.01$ 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 $\partial 2 T \partial 2 T 1 \partial T + = \partial x 2 \partial y 2 \alpha \partial t h$, $T \infty b \partial T (x, 0, t) = 0 \partial x \partial T (a, y, t) - k = h[T (a, y, t) - T \infty] \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) - K = h[T (x, 0, t) - T \infty] \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) - K = h[T (x, 0, t) - T \infty] \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x
\partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T (x, 0, t) = 0 \partial x \partial T$ 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): 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 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 = 0 180 W/m · °C (10 7 W/m 3)(0.1 m) 2 m) $2\,350 + T1 - 3T3 + = 0\,180$ W/m · °C The solution of the above system is $100 \cdot 120 \cdot 220 + T3 - 3T1 + 100 \cdot 200$ (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 1/2, length 0.3 m, and depth 1 m: T - 100 (1 × 1 120 - 100 Q& top + g& 0 (0.3 × 1×1/2) + || 2k + 2kl × 11 || = 0121 / (1m) Q& top = -(107 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]" 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" $120+120+T 2+T 3-4*T 1+(g dot*1^2)/k=0$ "(b)" "The rate of heat loss from the top surface can be determined from an energy balance on a volume element whose height is 1/2, length 3*1, and depth d=1 m" -Q dot top+g dot*(3*1*d*1/2)+2*(k*(1*d)/2*(120-100)/1+k*1*d*(T 1-100)/1)=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 3.691E+07 4.216E+07 4.216E+07 4.216E+07 4.216E+07 5.268E+07 5. 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 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 1.5 x 10 6 1.0 x 10 6 500 5.0 x 10 5 0.0 x 10 5 0.0 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 ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k [W /m ^3] 5-45 8.8 x 10 7 Q top [W] T 1 [C] k 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 midpoint. Analysis (a) We take the pipe as the system. Substituting these values into the one-term solution gives T – T θ 0 = 0 ∞ = A1e – λ 21 τ \rightarrow 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 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 v 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 \ μ s [= 2 + 0.4(2.961 × 10 - 5 m 2/s [] (μ hD Nu = 2 + 0.4 Re 0.5 + 0.06 Re 2/3 Pr 0.4]] $10 + 0.05 + 1/4 + 0.06(2.961 \times 10) + 0.06(2.961 \times 10) + 0.06(2.961 \times 10) + 0.02514 \text{ W/m} + 0.02514$ $(-183)]^{\circ}C = 25.8 \text{ W} (2.02 - 2) \text{ m} 1 + 4\pi (0.00005 \text{ W/m.}^{\circ}C)(2.02 \text{ m})(2 \text{ m})(10.73 \text{ W/m} 2.^{\circ}C)(51.28 \text{ m} 2)$ and h= The rate of evaporation of liquid oxygen then becomes 0.0258 kJ/s Q& Q& = m& hif \rightarrow 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: $Bi = . \circ C$ (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 is assumed to be at the same temperature as the iced water because of negligible resistance through the steel. the temperatures at the neighboring nodes in the following easy-to-remember form: i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i + 1 i i + 1 i i + 1 i i + 1 i i + 1 i i + 1 i i + 1 i i + 1 i i i + 1radiation. Therefore, the Reynolds number is defined on the basis of maximum velocity. Noting that u = u(y), v = 0, and $\partial P / \partial x = 0$ (flow is maintained by the motion of the upper plate rather than the pressure gradient), the xmomentum equation reduces to ($\partial u \ d \ 2u \ \partial u \ \partial P x$ -momentum: $\rho || u \rightarrow = 0 + v || = \mu \ 2 - \partial y / \partial x \ dy \ 2 \partial y \sqrt{\partial x \ dy \ 2 \partial y}$ 15 cm This is a second-order ordinary differential equation, and integrating it twice gives u (y) = C1 y + C 2 The fluid 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) - T∞] dr 2-13 h T∞ r1 r2 Chapter 2 Heat 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]" r 1=0.037 "[m]" r 2=0.04 "[m]" r 1=0.037 "[m]" 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 0.03967 0.04 T [C] 3.806 3.802 3.889 3.885 3.881 3.877 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 of the body is specified at a point that is at the center of the plane wall but at the surface of the cylinder. $\Delta x \in Assumptions 1$ Heat transfer through the wall is given • • 0• to be 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, $\varepsilon = 0.80$, and $\alpha s = 0.45$. Then, Overall U-factor, $U = \Sigma farea, iUi = 0.82 \times 1.105 + 0.18 \times 0.805$ 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: $0=k \partial 2T 2 2 dT \mu (V) = - ||y + C3 dy k (L/2T(y) = - \mu (y) ||V| + C3 y + C 4 2k (L/Applying the two boundary conditions give dT - k$ $=0 \rightarrow C 3 = 0$ B.C. 1: y=0 dy y = 0 B.C. 2: $y=L \rightarrow C 4 = T0 + T (L) = T0 \mu V 2$ 2k Substituting the constants give the temperature distribution to be T (y) = T0 + $\mu V 2$ 2k (y2) ||1 - | L2 |/ (The temperature gradient is determined by differentiating T(y) with respect to y, dT - $\mu V 2 = y$ 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 - μ V 2 = y=0 \rightarrow y = 0 dy kL2 Therefore, maximum temperature will occur at the upper plate is q& L = -k dT dy = k y=L μ V 2 kL2 L= μ V 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 radiations. Analysis The one-dimensional transient temperature distribution in the wall for that time period can be determined from (x, t) – Ti = erfc[] Ts $-\text{Ti}(2 \alpha t)$ || / Ice chest Hot water 60°C But, T (x, t) - Ti 0.1 - 0 = 0.00167 \rightarrow 0.00167 = erfc(2.225) 60 - 0 Ts - Ti (Table 4-3) Ice, 0°C Therefore, x 2 $\alpha t = 2.225 \rightarrow t = x2.4 \times (2.225) 2 \alpha = (0.05 \text{ m}) 2.4(2.225) 2 \alpha = (0.05 \text{ m})$ applying the thermal resistance network concept as 11 = 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 = $\approx 0^{\circ} \text{C/W} kA$ (52 W/m.°C)(1.2 × 2 m 2) = Rconv, i = R wall Rconv, 0 Rtotal Rconv 1 1 T1 = $\approx 0.00207 \text{°C/W} kA$ (52 W/m.°C)(1.2 × 2 m 2) = Rconv, i = R wall Rconv, 0 Rtotal Rconv 1 1 T1 = $\approx 0^{\circ} \text{C/W} kA$ (52 W/m.°C)(1.2 × 2 m 2) = Rconv, i = R wall Rconv, 0 Rtotal Rconv 1 1 T1 = $\approx 0.00207 \text{°C/W} kA$ (52 W/m.°C)(1.2 × 2 m 2) = Rconv, i = R wall Rconv, 0 Rtotal Rconv 1 1 T1 = $\approx 0.00167 \text{°C/W} kA$ (52 W/m.°C)(1.2 × 2 m 2) = Rconv, i = R wall Rconv, 0 Rtotal Rconv 1 1 T1 = $\approx 0.00167 \text{°C/W} kA$ (52 W/m.°C)(1.2 × 2 m 2) = Rconv, i = R wall Rconv, 0 Rtotal Rconv, 2 1 = 28,990 W Rtotal 0.00207 o C/W 4-63 Rwall Rconv T2 Chapter 4 Transient Heat Conduction in Multidimensional Systems 4-69C The product of dimensionless temperatures of onedimensional heat transfer problems. Assumptions 1 Heat conduction in the block is three-dimensional, and thus the temperature varies in all three directions. $^{\circ}$ C)(0.0015 m)] = 0.00117 W/ $^{\circ}$ C k eff = (kt) copper + (kt) epoxy t copper + (kt) epox t copper + (kt) 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 $\Delta E \&$ system $\hat{E}0$ (steady) = = 0 $\rightarrow E \&$ 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 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 $\Delta ke \cong \Delta pe \cong 0$) mh & p (T1 - T2) Q& out = mC Substituting, Q& out = mC Substi 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 transfers to be towards the node under consideration, the explicit transient finite difference formulations become Left boundary node: 4 $\varepsilon\sigma A[Tsurr - (T0i) 4] + hA(T\infty i - T0i) + kA T3i - T4i \Delta x \Delta x T4i + 1 - T4i i \& + = \rho + g A A C Q \&$ right kA surface $4 \Delta x 2 2 \Delta t$ Heat transfer at right surface: Noting that $Q = Q \& \Delta t = i + 1 i T1i - T0i \Delta x \Delta x T 0 - T 0 + g \& 0i A = \rho A C \Delta x 2 2 \Delta t \sum Q \& \Delta t$, the total amount of heat i Tsurr i 20 Q right surface = $\sum TL g(x, t)$ transfer becomes q0 i Q& right surface $\Delta t \Delta x i = 1 i + 1 i (T4i - T3i \Delta x \Delta x T4 - T4 | kA - g& 4i A + \rho A = C | 2 2 \Delta x \Delta t i = 1 (20 \sum 5-69) | \Delta t | / h, T \approx • 0 • 1 • 2 • 3 4 • Chapter 5 Numerical Methods in Heat Conduction 5-78 Starting with an energy balance on a volume element, the two-dimensional transient explicit finite difference equation for a general$ 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 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 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 = $\rho A C \Delta x 2 \Delta t h \Delta x$ h $\Delta x (T0i + 1 = || 1 - 2\tau - 2\tau in || T0i + 2\tau T1i + 2\tau$ 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 hout A(Tout – T6i) + κ Aq& solar + kA Node 6 or i + 1 i T5i – T6i Δx T6 – T6 = ρ A C Δx 2 Δt kq& i Δx h Δx h W/m2.°C, and $\Delta x = 0.05$ m. Analysis (a) The total mass of water is m w = $\rho V = (1 \text{ kg/L})(50 \times 20 \text{ L}) = 1000 \text{ kg} 50,000 \text{ kJ/h}$ 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 $k = 1.4 \text{ W/m} \cdot \text{c}$. 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 – λ 1 τ = $(1.2431)e - (1.3269)(0.9504)\theta(0, t)$ wall = Ti - T ∞ The Fourier number is = Margarine, Ti = 30°C T (0, t) - 0 = 0.233 \rightarrow 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 \rightarrow λ 1 = 1.4641 and A1 = 1.2661 k (0.082 W/m.°C) τ = α t L2 = (0.10 × 10 - 6 m 2/s)(6 h × 3600 s/h) (0.05 m) 2 (0.05 m) 2 (0.05 m) Bi = = 15.24 \rightarrow λ 1 = 1.4641 and A1 = 1.2661 k (0.082 W/m.°C) τ = α t L2 = (0.10 × 10 - 6 m 2/s)(6 h × 3600 s/h) (0.05 m) 2 (0.05 m) 2 (0.05 m) Bi = 1.4641 and A1 = 1.2661 k (0.082 W/m.°C) $\theta(0, t)$ wall = $0.864 > 0.2 2 2 T (0, t) - T = (1.2661)e - (1.4641)(0.864) Ti - T = (1.2661)e - (1.4641)(0.864) Ti - T = (0, t) - 0 = 0.199 \rightarrow T (0, t) = 6.0 °C 30 - 0 (c)$ Repeating the calculations for chocolate cake, hL (25 W/m 2.°C)(0.05 m) Bi = = $11.79 \rightarrow \lambda 1 = 1.4356$ and A1 = $1.2634 k (0.106 W/m.°C) \tau = \alpha t L2 = (0.12 \times 10 - 6 m 2/s)(6 h \times 3600 s/h) \theta(0, t)$ wall = $(0.05 \text{ m}) 2 = 1.0368 > 0.2 2 2 \text{ T} (0, t) - T \infty = A1 \text{ e} - \lambda 1 \tau = (1.2634) \text{e} - (1.4356) (1.0368) \text{ Ti} - T \infty \text{ T} (0, t) = 4.5 ^{\circ}\text{C} 30 - 0.4 + 114 \text{ Chapter 4 Transient Heat Conduction 4-121 A cold cylindrical concrete column is exposed to warm ambient air during the day. Assumptions 1 The temperature in the wall is affected by the$ 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 constant. Rapid freezing increases tenderness and reduces the tissue damage and the amount of drip after thawing. It is to be determined if the plastic insulation on the wire will increase or decrease heat transferred uniformly from all surfaces of the resistor. The thermal conductivity of super insulation is given to be $k = 0.00008 \text{ W/m} \circ \text{C}$. Analysis (a) Noting that heat transfer is steady and one-dimensional in x direction, the mathematical formulation of this problem can be expressed as d 2T g& (x) + = 0 2 k dx where g& = g& 0 e -0.5x / L and g& 0 = 8 × 106 W/m3 and dT (0) = 0 (insulated surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C (specified surface at x = 0) dx k g T2 = 30°C
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Then the heat transfer coefficient can be determined from. The equivalent but "more correct" unit of thermal conductivity is W.m/m2.°C that indicates product of heat transfer rate and thickness per unit temperature difference. We need only one initial condition for a heat conduction equation is first order in time (it involves the first derivative of temperature with respect to time). The individual thermal resistances are Ri Rdeposit Rpipr Ro To 2 To 1 Ai = π Di L = π (0.4 / 12 ft)(1 ft) = 0105. 2 Heat transfer by radiation is disregarded. The maximum amount of heat transfer by radiation is disregarded. The maximum amount of heat transfer by radiation is disregarded. The maximum amount of heat transfer by radiation is disregarded. The maximum amount of heat transfer by radiation is disregarded. The maximum amount of heat transfer by radiation is disregarded. The maximum amount of heat transfer by radiation is disregarded. Rinsulation Ro T ∞ Ts L and in order to have this thermal resistance, the thickness of insulation must be 1 L Rtotal = Rconv + Rinsulation = + hA kA 1 L = + = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.047 m = 4.7 cm Noting that heat is saved at a rate of 0.9×1500 = 1350 W and the furnace operates continuously and thus $365 \times 24 = 8760$ h per year, and that the furnace efficiency is 78%, the amount of natural gas saved per year is Q& Δt (1.350 kJ/s)(8760 h) (3600 s)(1 therms) = Energy Saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = Energy Saved = (517.4 therms)(\$0.55 / therm) = Energy Saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = Energy Saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = Energy Saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = Energy Saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = Energy Saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = (517.4 therms)(\$0.55 / therm) = Energy Saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = (517.4 therms)({\$100} / therm) = (517.4 therm)({\$100} / therm) = (517.4 therm)({\$100} / therm)(\$284.5 (per year) The insulation will pay for its cost of \$250 in Money spent \$250 Payback period = = 0.88 yr Money saved \$284.5 (yr which is less than one year. Cement mortar, 13 mm 4. The rate of evaporation of the liquid nitrogen due to heat transfer from the air is to be determined for three cases. Properties of air at 1 atm and the film temperature of $(Ts + T\infty)/2 = (90+7)/2 = 48.5^{\circ}C$ are $(Table A-15) k = 0.02724 W/m.^{\circ}C Air V = 50 km/h T_{\infty} = 7^{\circ}C v = 1.784 \times 10 - 5 m 2 /s$ The Nusselt number corresponding to this Reynolds number is $0.62 \text{ Re } 0.5 \text{ Pr } 1/3 \text{ hD } \text{Nu} = = 0.3 + 1/4 \text{ k} 1 + (0.4/\text{Pr}) 2/3 [] (\text{Re } 5/8] | 1 + || || || (282,000 / |] 4/5 [0.62(6.228 \times 104) 0.5 (0.7232) 1/3 | (6.228 \times 104) + = 0.3 + 1/4 || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| || (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + || 1| (282,000 1 + (0.4/0.7232) 2/3 [] (1 + (0.4/0.7232) 2/3 [] (1 + (0.4/0.7232) 2/3 [] (1 + (0.4/0.7232) 2$ Nu = $(159.1) = 54.17 \text{ W/m } 2 \cdot ^{\circ}\text{C} D \ 0.08 \text{ m As} = \pi DL = \pi (0.08 \text{ m})(1 \text{ m}) = 0.2513 \text{ m } 2 \text{ Q& conv} = hAs (Ts - T\infty) = (54.17 \text{ W/m } 2 \cdot ^{\circ}\text{C})(0.2513 \text{ m } 2)(90 - 7)^{\circ}\text{C} = 1130 \text{ W}$ (per m length) 7-27 Pipe D = 8 cm Ts = 90°C Chapter 7 External Forced Convection 7-40 A hot stainless steel ball is cooled by forced air. Cylindrical rods can also be treated as being infinitely long when dealing with heat transfer at locations far from the top or bottom surfaces. The individual thermal resistances are Ai = πDi L = π (0.092 m)(6 m) = 173 . 2-97C No, the temperature variation in a plain wall will not be linear when the thermal conductivity varies with temperature. 3 Air is an ideal gas with constant properties. 3 The exposed surface, ambient, and sky temperatures remain constant. 2-32 For a medium in which the heat conductivity is variable. 2 The heat transfer is steady, (b) it is two-dimensional, (c) there is heat generation, and (d) the thermal conductivity is variable. 2 The heat transfer coefficient is constant and uniform over the entire exposed surface of the person. The depth at which freezing at 0°C occurs can be determined from the analytical solution, Ts = Soil Ti = 8°C (x) T (x, t) - Ti = erfc | |Ts - Ti (αt) Water pipe () x 0-8 | z = erfc | -626-8-8 | $z = (0.15 \times 10 \text{ m/s})(5.184 \times 10 \text{ s})$ | $| () (x) \rightarrow 0.444 = erfc | |Ts - Ti (<math>\alpha t$) Water pipe () x 0-8 | z = erfc | -626-8-8 | $z = (0.15 \times 10 \text{ m/s})(5.184 \times 10 \text{ s})$ | $| () (x) \rightarrow 0.444 = erfc | |Ts - Ti (<math>\alpha t$) Water pipe () x 0-8 | z = erfc | -626-8-8 | z = erfc | -626-8-8-8 | z = erfc | -626-8-8-8-8 | z = erfc | -626-8-8-8-8-8-8 | z = erfc | -626-8-8-8-8-8-8-8-8-8-8-81.7636 / Then from Table 4-3 we get x = 0.5297 \rightarrow x = 0.934 m 1.7636 Discussion The solution could also be determined using the chart, but it would be subject to reading error. This is a closed system since no mass enters or leaves. Cengel July 2002 Chapter 1 BASICS OF HEAT TRANSFER Thermodynamics and Heat Transfer 1-1C Thermodynamics deals with the amount of heat transfer as a system undergoes a process from one equilibrium state to another. 5-71 Chapter 5 Numerical Methods in Heat Conduction 5-80 Starting with an energy balance on a disk volume element, the one-dimensional transient explicit finite difference equation for a general interior node for T (z, t) in a cylinder whose side surface is insulated for the case of constant thermal conductivity with uniform heat generation is to be obtained. Assumptions 1 Water is an incompressible substance with constant specific heats at room temperature. The frontal area is appropriate to use in drag and lift calculations for blunt bodies such as cars, cylinders, and spheres. Then the allowable time is determined to be b = hAs 125 W/m 2. °C h = = $0.01610 \text{ s} - 1 \rho \text{C} \text{ p} \text{ V} \rho \text{C} \text{ p} \text{ L} (8085 \text{ kg/m 3})(480 \text{ J/kg.°C})(0.002 \text{ m}) - 1 \text{ T} (t) - T \infty 850 - 30 = e - bt \rightarrow = e - (0.0161 \text{ s})t \rightarrow t = 3.68 \text{ s} 900 - 30 \text{ Ti} - T \infty$ The result indicates that the ball bearing can stay in the air about 4 s before being dropped into the water. It gives $\alpha = b = k \rho C p \rightarrow \rho C p = k \alpha = 110 \text{ W/m} \cdot ^{\circ}C 33.9 \times 10.6 \text{ m} 2 / \text{s} = 3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C \text{ hA hA h h 80 W/m} 2 \cdot ^{\circ}C = = = = 0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) - T_{\infty} = e - bt Ti - T_{\infty} \rightarrow T(t) = T_{\infty} + (Ti - T_{\infty})e - bt = 700^{\circ}C + (25 - 700^{\circ}C)e -
(0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) - T_{\infty} = e - bt Ti - T_{\infty} \rightarrow T(t) = T_{\infty} + (Ti - T_{\infty})e - bt = 700^{\circ}C + (25 - 700^{\circ}C)e - (0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) = T_{\infty} + (Ti - T_{\infty})e - bt = 700^{\circ}C + (25 - 700^{\circ}C)e - (0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) = T_{\infty} + (Ti - T_{\infty})e - bt = 700^{\circ}C + (25 - 700^{\circ}C)e - (0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) = T_{\infty} + (Ti - T_{\infty})e - bt = 700^{\circ}C + (25 - 700^{\circ}C)e - (0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) = T_{\infty} + (Ti - T_{\infty})e - bt = 700^{\circ}C + (25 - 700^{\circ}C)e - (0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) = T_{\infty} + (Ti - T_{\infty})e - bt = 700^{\circ}C + (25 - 700^{\circ}C)e - (0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) = 70^{\circ}C + (25 - 700^{\circ}C)e - (0.001644 \text{ s} - 1 \rho V C p \rho (LA) C p \rho LC p L(k / \alpha) (0.015 \text{ m})(3.245 \times 10.6 \text{ W} \cdot \text{s/m} 3 \cdot ^{\circ}C) T(t) = 70^{\circ}C + (25 - 700^{\circ}C)e - (25 - 700^{\circ$ which is almost identical to the result obtained above. Analysis The R-value of the existing wall is Rwinter = 1 / U winter = 1 / 155. Properties The thermal conductivities are given to be k = 0.035 W/m.°C for synthetic fabric, k = 0.026 W/m.°C for syn have to be a function of x. Then the rate of heat conduction through the plate can be determined to be $(T - T2 [\beta Q \& = k \text{ ave } A 1 = k 0 | 1 + T22 + T1T2 + T12 L [3)] | A T 1] - T2 L Discussion We would obtain the same result if we substituted the given k(T) relation into the second part of Eq. 2-76, and performed the indicated integration. Nodes$ 1, 2, 3, and 4 in the plate and 6, 7, 8, and 9 in the soil are interior nodes, and thus for them we can use the general finite difference relation expressed as Tm - 1 - 2Tm + Tm + 1 = 0 (since g& = 0) k $\Delta x = 2$ The finite difference relation for node 0 on the left surface and node 5 at the interface are obtained by applying an energy balance on their respective volume elements and taking the direction of all heat transfers to be towards the node under consideration: T - T 4 - (T0 + 460) 4] + k plate 1 0 = 0 Node 3 (interior): T0 - 2T1 + T2 = 0 Node 3 (interior): T0 - 2T1 + T2 = 0 Node 3 (interior): T0 - 2T1 + T2 = 0 Node 3 (interior): T - T4 - (T0 + 460) 4] + k plate 1 0 = 0 Node 3 (interior): T0 - 2T1 + T2 = 0 Node 3 (interior): T0 - 2T1 + T2 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3 + T3 = 0 Node 3 (interior): T0 - 2T3+ T4 = 0 Node 4 (interior): T3 - 2T4 + T5 = 0 Node 5 (interface): k plate T4 - T5 T - T5 + k soil 6 = 0 $\Delta x1 \Delta x 2$ Node 6 (interior): T7 - 2T8 + T9 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 7 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node 9 (interior): T7 - 2T8 + T7 = 0 Node = 3.5 Btu/h·ft2·°F, Tsky = 510 R, $\varepsilon = 0.6$, T $\infty = 80^{\circ}F$, and T10 = 50°F. 3 Constant specific heats at room temperature can be used for air. 6-6 Chapter 6 Fundamentals of Convection 6-37 Parallel flow of oil between two plates is considered. Also, the mesh Fourier number is $\tau = \alpha \Delta t \Delta x 2 = (0.69 \times 10 - 6 \text{ m } 2 / \text{s})(300 \text{ s})(0.03 \text{ m}) 2 = 0.230$ Substituting this value of τ and other given quantities, the inner and outer surface temperatures of the roof after 12×(60/5) = 144 time steps (12 h) are determined to be T1 = 10.3°C and T6 = -0.97°C. Properties The thermal conductivities are given to be k = 20 W/m.°C for the aluminum plate and fins, and k = 1.8 W/m.°C for the epoxy adhesive. Assuming transient one-dimensional conduction in the radial direction, the boundary conditions at the interface can be expressed as TA (r0, t) = TB (r0, t) and $-k A \partial TA$ (r0, t) and $-k A \partial TA$ (r0, t) = TB (r0, t) and $-k A \partial TA$ (r0, t) = $-k B B 0 \partial x \partial x r^2 - 45$ Heat conduction through the bottom section of a steel pan that is used to boil water on top of an electric range is considered (Fig. After 10 minutes: The Biot number and the corresponding constants for h = 40 W / m2. °C)(0.025 m) = = 0.400 $\rightarrow \lambda 1 = 0.5932$ and A1 = 10580. Assumptions 1 The drink is at a uniform temperature at all times. Properties The density and heat of vaporization of the water are $\rho = 1000 \text{ kg/m3}$ and hfg = 2490 kJ/kg (Table A-9). The average thermal conductivity of the refrigerator walls and the annual cost of operating this refrigerator are to be determined. Then the mesh Fourier number becomes $\tau = \alpha \Delta t \Delta x 2 = (12.5 \times 10 - 6 \text{ m}2 / \text{s})(15 \text{ s})(0.02 \text{ m}) 2 = 0.46875$ Substituting this value of τ and other given quantities, the nodal temperatures after $5 \times 60/15 = (12.5 \times 10 - 6 \text{ m}2 / \text{s})(15 \text{ s})(15 \text{ s})(15$ 20 time steps (5 min) are determined to be After 5 min: T0 = 228.9°C, = 219.9 °C T1 = 228.4°C, T2 = 226.8°C, T3 = 224.0°C, and T4 (b) The time needed for transient operation to be established is determined by increasing the number of time steps until the nodal temperatures no longer change. 3-106 A relation is to be obtained for the fin efficiency for a fin of constant cross-sectional area As, perimeter p, length L, and thermal conductivity k exposed to convection from four sides are at a uniform temperature. Noting that the volume element centered about the general interior node (m, n) involves heat conduction from four sides $Tmi + n1 \Delta y - Tmi , n\Delta t$ Taking a square mesh ($\Delta x = \Delta y = l$) and dividing each term by k gives, after simplifying, $Tmi + n1 - 1 - Tmi \tau$ where $\alpha = k / (\rho C)$ is the thermal diffusivity of the material and $\tau = \alpha \Delta t / l 2$ is the dimensionless mesh Fourier number. energies $0 = \Delta U$ Copper $\Delta U Cu + \Delta U water = 0 \text{ or, } [mC (T2 - T1)]Cu + [mC (T2 -
T1)]water = 0 \text{ Using specific heat values for copper and liquid water at room temperature and substituting, } (50 kg)(0.386 kJ/kg \cdot °C)(T2 - 70)°C + (80 kg)(4.18 kJ/kg \cdot °C)(T2 - 25)°C = 0 T2 = 27.5°C 1-35 \text{ An iron block at } 100°C is brought into contact with an aluminum block at 200°C in an$ insulated enclosure. If this box is to be cooled by radiation alone and the outer surface temperature of the box is not to exceed 55°C, the temperature the surrounding surfaces must be kept is to be determined. Assumptions 1 Heat transfer from the fin tips is negligible. The energy balance for this steady-flow system can be expressed in the rate form as E& - E& out Rate of change in internal, kinetic, potential, etc. 5-84 Chapter 5 Numerical Methods in Heat Conduction 5-88 "PROBLEM 5-88" "GIVEN" T i=140 "[C]" k=15 "[W/m-C]" alpha=3.2E-6 "[m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" T infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^2/s]" g dot=2E7 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^3]" T bottom=140 "[C]" t infinity=25 "[C]" h=80 "[W/m^3]" t infinity=25 "[W/m^3]" t in 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. 4-81 Chapter 4 Transient Heat Conduction Cylinder: This cylindrical block can physically be formed by the intersection of a long cylinder of radius ro = D/2 = 2.5 cm and a plane wall of thickness 2L = 5 cm. 4 Heat losses from the room are negligible. Btu / h · ft 2 (per unit area) L 0.5 ft Discussion The negative sign indicates that the direction of heat transfer is from the outside to the inside. Analysis The Fourier number is $\tau = \alpha t$ ro $2 = (0.15 \times 10 - 6 \text{ m } 2/\text{s})[(4 \times 60 + 40 \text{ min}) \times 60 \text{ s/min}]$ (0.10 m) 2 = 0.2522 Btu / h · ft 2 (per unit area) L 0.5 ft Discussion The negative sign indicates that the direction of heat transfer is from the outside to the inside. which is greater than 0.2. Then the one-term solution can be written in the form θ 0, sph = Lake 15°C Water melon Ti = 35°C 2 2 T0 - T ∞ It is determined from Table 4-1 by trial and error that this equation is satisfied when Bi = 10, which corresponds to λ 1 = 2.8363 and A1 = 19249. 2 The thermal properties of the milk are taken to Water be the same as those of water. This problem involves 3 unknown nodal temperatures, and thus we need to have a equation to ••• 0. Tsurr 1 determined. 2 The thermal resistance of the can and the internal convection resistance of the milk are taken to Water be the same as those of water. This problem involves 3 unknown nodal temperatures, and thus we need to have 3 equations to ••• 0. are negligible so that the can is at the same temperature as the drink inside. The density of water is given to be 1000 kg/m³. The Reynolds number in this case is L = 1.2 m V L [(60 × 1000 / 3600) m/s](1.2 m) 6 Re $L = \infty = = 1$. Properties The thermal conductivity of uranium at room temperature is $k = 27.6 \text{ W/m} \cdot \text{°C}$ (Table A-3). Properties The solar absorptivity of the plate is given to be $\alpha = 0.7$. Analysis When the heat loss from the plate by convection equals the solar absorbed conv $\alpha Q \&$ solar = hAs (Ts - To) $0.7 \times A \times 700 W/m 2 = (30 W/m 2 \cdot °C)$ As (Ts - 10) Canceling the surface area As and solving for Ts gives Ts = 26.3° C 1-72~700 W/m2 α = 0.7 air, 10° C. The vertical section of the damn is subjected to convection with water. The amount of heat that needs to be transferred to the aluminum ball is to be determined. Therefore, the nodal spacing Δx is $\Delta x = L~0.05$ m = = 0.01 m M - 1.6 - 1 The temperature at node 0 is given to be T0 = 200°C, and the temperatures at the remaining 5 nodes are to be determined. 4 The heat transfer coefficient is constant and uniform over the entire fin surfaces must be less than 23.3°C. It is twodimensional if heat transfer in the third dimension is negligible. Btu $h \cdot ft \circ F$ = 77.5 Btu / $h \cdot ft 2$ (per unit area) L 0.5 ft Discussion The positive sign indicates that the direction of heat transfer is from the inside to the outside. Properties The thermal conductivity of an alloy of two metals will most likely be less than the thermal conductivities of both metals. Properties The thermal properties of aluminum are given to be $k = 237 \text{ W/m.}^{\circ}\text{C}$ and $\alpha = 9.71 \times 10-5 \text{m2/s}$. 1-58C The thermal conductivity of gases is proportional to the square root of absolute temperature. Also, a thermal symmetry line and an insulated boundary are treated the same way in the finite difference formulation. 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 r 1 d (dT) g (r) = T = 180°C (specified surface temperature) and 0 s dT (0) = 0 (thermal symmetry about the centerline) dr Multiplying both sides of the differential equation by r and rearranging gives g Resistance wire g & d (dT) | r | = - r dr (dr) k Integrating with respect to r gives dT g & r 2 r (a) = - + C1 dr k 2 It is convenient at this point to apply the boundary node (node 3) and the finite difference formulation for the rate of heat transfer at the left boundary (node 0) are to be determined. Properties The R-values at the outer surface of a wall for summer = 0.044 m2.°C/W and Ro, winter = 0.030 m2.°C/W. The locations of nodes are as follows: "Node 1: Middle of top surface Node 2: At the right side of node 2 Node 3: At the right side of node 2 Node 4: Corner node below node 2. At the right side of node 2: At the right side 0: A node 7, at the middle of inner right surface Node 10: The node below node 8, at the middle of outer right surface T corner=T 4 T inner middle=T 9 "(c)" "The rate of heat loss through a unit depth d=1 m of the chimney is" Q dot=4*(h i*1/2*d*(T i-T 5)+h i*1*d*(T i-T 7)+h i*1/2*d*(T i-T 29.37 30.35 Tinner, middle [C] 187 196.3 205.7 5-54 Q [W] 2206 2323 2441 Chapter 5 Numerical Methods in Heat Conduction 230 240 250 260 270 280 390 400 31.32 32.28 33.24 34.2 35.14 36.08 37.02 37.95 38.87 39.79 40.7 41.6 42.5 43.39 44.28 45.16 46.04 46.91 215 224.3 233.6 242.9 252.2 261.57 5.54 Q [W] 2206 2323 2441 Chapter 5 Numerical Methods in Heat Conduction 230 240 250 260 270 280 290 300 310 320 330 340 350 360 370 380 390 400 31.32 32.28 33.24 34.2 35.14 36.08 37.02 37.95 38.87 39.79 40.7 41.6 42.5 43.39 44.28 45.16 46.04 46.91 215 224.3 233.6 242.9 252.2 261.57 5.54 Q [W] 2206 2323 2441 Chapter 5 Numerical Methods in Heat Conduction 230 240 250 260 270 280 290 300 310 320 330 340 350 360 370 380 390 400 31.32 32.28 33.24 34.2 35.14 36.08 37.02 37.95 38.87 39.79 40.7 41.6 42.5 43.39 44.28 45.16 46.04 46.91 215 224.3 233.6 242.9 252.2 261.57 5.54 Q [W] 2206 2323 2441 Chapter 5 Numerical Methods in Heat Conduction 230 240 250 260 270 280 290 300 310 320 330 340 350 360 370 380 390 400 31.32 32.28 33.24 34.2 35.14 36.08 37.02 37.95 38.87 39.79 40.7 41.6 42.5 43.39 44.28 45.16 46.91 215 224.3 233.6 242.9 252.2 261.57 5.54 Q [W] 2206 2323 2441 Chapter 5 Numerical Methods in Heat Conduction 230 240 250 260 270 280 290 300 310 320 330 340 350 360 370 380 390 400 31.32 32.28 33.24 34.2 35.14 36.08 37.02 37.95 38.87 39.79 40.7 41.6 42.5 43.39 44.28 45.16 46.94 46.91 215 224.3 23.6 24.9 252.2 261.57 40.28 45.16 46.94 46.91 215 224.3 25.16 46.94 46.91 215 224.3 25.16 46.94 46.91 215 224.3 25.16 46.94 46.91 215 224.3 25.16 46.94 46.91 215 224.3 25.16 46.94 46.91 215 224.3 25.16 46.94 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 46.91 215 224.3 25.16 215 224.3 25.16 215 224 270.8 280.1 289.3 298.6 307.9 317.2 326.5 335.8 345.1 354.4 363.6 372.9 2559 2677 2796 2914 3033 3153 3272 3392 3512 3632 3752 3873 3994 4115 4237 4358 4480 4602 ε 0.1 0.15 0.2 0.25 0.3 0.35 0.4 0.45 0.5 0.65 0.6 0.65 0.7 0.75 0.8 0.85 0.9 0.95 1 Tcorner [C] 51.09 49.87 48.7 47.58 46.5 45.46 44.46 43.5 42.56 41.66 40.79 39.94 39.12 3632 3752 3873 3994 4115 4237 4358 4480 4602 ε 0.1 0.15 0.2 0.25 0.3 0.35 0.4 0.45 0.5 0.65 0.6 0.65 0.7 0.75 0.8 0.85 0.9 0.95 1 Tcorner [C] 51.09 49.87 48.7 47.58 46.5 45.46 44.46 43.5 42.56 41.66 40.79 39.94 39.12 38.33 37.56 36.81 36.08 35.38 34.69 Tinner, middle [C] 263.4 263.2 263.1 262.9 262.8 262.7 262.3 262.2 262.1 262 261.9 261.8 261.7 261.6 261.5 261.4 261.3 Q [W] 2836 2862 2886 2909 2932 2953 2974 2995 3014 3033 3052 3070 3087 3104 3121 3137 3153 3168 3183 5-55 Chapter 5 Numerical Methods in Heat Conduction 47.5 5000 4500 43.5 tem perature 4000 35.5 3000 31.5 27.5 200 2500 240 280 320 2000 400 360 T i [C] 375 335 295 T inner, m iddle [C] T
corner [C] 3500 255 215 175 200 240 280 320 T i [C] 5-56 360 400 Q [W] heat 39.5 Chapter 5 Numerical Methods in Heat Conduction 52.5 3200 3150 48.5 3100 heat T corner [C] 3000 40.5 2950 2900 36.5 tem perature 2850 32.5 0.1 0.2 0.3 0.4 0.5 ε 0.6 5-57 0.7 0.8 0.9 2800 1 Q [W] 3050 44.5 Chapter 5 Numerical Methods in Heat Conduction 5-58 The exposed surface of a long concrete damn of triangular cross-section is subjected to solar heat flux and convection and radiation heat transfer. The surface heat flux, the surface temperature of the chips, and the general finite difference relation expressed as $T - Tm T - Tm kA m - 1 + kA m + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(T_{\infty} - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x)(Tm$ through the walls of the duct is to be determined. 2 Heat transfer is oneh, $T \propto k(T)$ dimensional since the plate is large relative to its Tsurr thickness. Using the energy balance approach and taking the direction of all heat transfer to be towards the node, the finite difference equations for the nodes are obtained to be as follows: Node 1: 50 + 120 + 2T2 + 2T -4T1 = 0 Node 2: Node 3: hl (T $\infty - T2$) + kT - T2 120 - T2 150 - T2 173 - T2 + k + kl 1 + kl = 0 1112 2 50°C h, T ∞ l T2 - T3 l 120 - T3 hl (T $\infty - T3$) + k + k = 0 112 2 where l = 0.1 m, k = 12 W/m·°C, h = 30 W/m2·°C, and T $\infty = 25°C$. 2 The surface temperature of the tubes is equal to the temperature of hot water. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 20 copper fins of rectangular profile on the backside. °C)(163 - 4.5)°C = 2080 kJ Then the actual amount of heat transfer becomes $sin(\lambda 1) - \lambda 1 cos(\lambda 1) sin(2.49) - (2.49) cos(2.49) Q = 1 - 30 o$, $sph = 1 - 3(0.543) = 0.7273 Q max \lambda 1$ (2.49) 3 Q = 0.727Q max = (0.727)(2080 kJ) = 1512 kJ (d) The cooking time for medium-done rib is determined to be θ 0, sph = t = 2 2 T0 - T ∞ 71 - 163 = A1 e - λ 1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - λ 1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - λ 1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - λ 1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - λ 1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T ∞ 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T \infty 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T \infty 71 - 163 = A1 e - \lambda1 τ \rightarrow = (1.7402)e - (2.49) τ \rightarrow τ = 0.177 4.5 - 163 Ti - T \infty 71 - 163 Ti - T \infty Assumptions 1 The temperature in the wall is affected by the thermal conditions at inner surfaces only and the convection heat transfer coefficient insulator is to halt the flow of electric current, and the purpose of a sound insulator is to slow down the propagation of sound waves. They imply the conversion of some other form of energy into thermal energy. m = Discussion Note that the apparatus described in this problem provides a convenient mechanism to measure drag force and thus drag coefficient. m) Rcopper = Repoxy Q& = T1 - Tbase (T1 - 39.5)°C = Rcopper + Repoxy + R board (0.00017 + 0.00555 + 0.011) °C/W \rightarrow T1 = 39.5°C + (15 W)(0.0167 °C/W) = 39.8°C T - T2 \rightarrow T2 = T1 - Q& R board = 39.8°C - (15 W)(0.011 °C/W) = 39.8°C T - T2 \rightarrow T2 = T1 - Q& R board = 39.8°C - (15 W)(0.011 °C/W) = 39.8°C T - T2 \rightarrow T2 = T1 - Q& R board = 39.8°C - (15 W)(0.0167 °C/W) = 39.8°C T - T2 \rightarrow T2 = T1 - Q& R board = 39.8°C - (15 W)(0.011 °C/W) = 39.8°C T - T2 \rightarrow T2 = T1 - Q& R board = 39.8°C - (15 W)(0.011 °C/W) = 39.8°C T - T2 \rightarrow T2 = T1 - Q& R board = 39.8°C + (15 W)(0.0167 °C/W) = 39.8°C + (15 W)(0.011 °C/W) = 39.8°C + (15 W)(0.0167 °C/W) = 39.8°C + (15 W)(0.011 °C/W) = 3 with a heat transfer coefficient of 12 W/m2.°C. Properties We assume the film temperature to be 10°C. 5-101C The round-off error is caused by retaining a limited number of digits during calculations. 4 The fins and the base plate are nearly isothermal (fin efficiency is equal to 1) 5 Air is an ideal gas with constant properties. Properties The therma conductivity of the glass is given to be kglass = 0.78 W/m °C. 7-63C The level of turbulence, and thus the heat transfer coefficient, increases with row number because of the combined effects of upstream rows in turbulence caused and the wakes formed. The dimensionless temperature for the surface of the plane wall with 2L = 80 cm is determined from $\theta(L, t)$ wall (C) = 2 2 T (x, t) - T ∞ = A1 e - λ 1 τ cos(λ 1 L / L) = (1.0076)e - (0.212) (0.2869) cos(0.212) = 0.9724 Ti - T ∞ Then the corner temperature of the block becomes 4-120 Chapter 4 Transient Heat Conduction [T (L, L, L, t) - T ∞] = $\theta(L, t)$ wall, B × $\theta(L, t)$ wall, A = 0.9724 × 0.9672 × 0.9672 = 0.9097 | Ti - T ∞ short [cylinder T (L, L, L, t) - 17 = 0.9097 \rightarrow T (L, L, t) = 138.0°C 150 - 17 4-121 Chapter 4 Transient Heat Conduction 4-127 A man is found dead in a room. 2 Heat is transferred uniformly from all surfaces of the transistor. W / m. Letting Ts,out denote the outer surface temperatures of the roof, the energy balance above can be expressed as Ts, in - Ts, out = ho A(Ts, out - Tsurr) + $\epsilon A\sigma$ (Ts, out 4 - Tsurr 4) Q& = kA L 15° C - Ts, out Q& = (2 W / m. Also, we would use the spherical coordinates. Due to these opposite effects, a critical radius of insulation is defined as the outer radius that provides maximum rate of heat transfer. (b) The 8 nodal temperatures under steady conditions are determined by solving the 8 equations above simultaneously with an equation solver to be T1 = 13.7° C, T2 = 7.4° C, T3 = 4.7° C, T3 = 4.7° C, T3 = 4.7° C, T4 = 3.9° C, T5 = 19.0° C, T6 = 11.3° C, T7 = 7.4° C, T8 = 6.2° C (c) The rate of heat transfer from the block to the
iced water is 6 kW since all the heat supplied to the block from the top must be equal to the heat transferred from the block. Therefore, knowing the strong dependence of viscosity on temperature, calculations should be repeated using properties at the average temperature of about 64°C to improve accuracy. Then, $4 - (T1 + 273) 4 = 0 m = 1: T0 - 2T1 + T2 + h(p\Delta x 2 / kA)(T - T1) + \epsilon \sigma (p\Delta x 2 / kA)[Tsurr 4 - (T2 + 273) 4] = 0 m = 2: T1 - 2T2 + T3 + h(p\Delta x 2 / kA)[Tsurr 4 - (T3 + 273) 4] = 0 m = 3: T2 - 2T3 + T4 + h(p\Delta x 2 / kA)(T - T3) + \epsilon \sigma (p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 - (T4 + 273) 4] = 0 m = 4: T3 - 2T4 + T5 + h(p\Delta x 2 / kA)[Tsurr 4 T9 + h(p\Delta x 2 / kA)(T \infty - T8) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T9) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - 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T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 / kA)(T \infty - T10) + \varepsilon\sigma(p\Delta x 2 /$ kA)[Tsurr - (T11 + 273) 4] = 0 m = 12 : 4 T11 - 2T12 + T13 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 M = 12 : 4 T11 - 2T12 + T13 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 M = 12 : 4 T11 - 2T12 + T13 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 M = 12 : 4 T11 - 2T12 + T13 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T12 - T13 4 + h($p\Delta x 2 / kA$)[Tsurr - (T12 + 273) 4] = 0 Node 13: kA where T1 $= (1 \text{ cm})(0.2 \text{ cm}) = 0.2 \text{ cm} 2 = 0.2 \times 10 - 4 \text{ m} 2 \text{ and } p = 2(1 + 0.2 \text{ cm}) = 2.4 \text{ cm} = 0.024 \text{ m}$ (b) The nodal temperatures under steady conditions are determined by solving the equations above to be T1 = 65.2°C, T2 = 48.1°C, T3 = 38.2°C, T4 = 32.4°C, T5 = 29.1°C, T6 = 27.1°C, T7 = 26.0°C, T8 = 25.3°C, T9 = 24.9°C, T10 = 24.7°C, T11 = 24.6°C, T12 = 48.1°C, T3 = 38.2°C, T4 = 32.4°C, T5 = 29.1°C, T6 = 27.1°C, T7 = 26.0°C, T8 = 25.3°C, T9 = 24.9°C, T10 = 24.7°C, T11 = 24.6°C, T12 = 48.1°C, T3 = 38.2°C, T4 = 32.4°C, T5 = 29.1°C, T6 = 27.1°C, T7 = 26.0°C, T8 = 25.3°C, T9 = 24.9°C, T11 = 24.6°C, T12 = 48.1°C, T3 = 38.2°C, T4 = 32.4°C, T5 = 29.1°C, T6 = 27.1°C, T7 = 26.0°C, T8 = 25.3°C, T9 = 24.9°C, T10 = 24.7°C, T11 = 24.6°C, T12 = 48.1°C, T3 = 38.2°C, T4 = 32.4°C, T5 = 29.1°C, T6 = 27.1°C, T7 = 26.0°C, T8 = 25.3°C, T9 = 24.9°C, T10 = 24.7°C, T11 = 24.6°C, T12 = 24.0°C, T12 = 24 =24.5°C, and T13 =24.5°C, (c) The total rate of heat transfer from the spoon handle is the sum of the heat transfer from each element, $M = 13 \sum 13$ hAsurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2$ for node 0, Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where
Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0 where Asurface, $m = p\Delta x/2 + A$ for node 13, T = 0.83 W m = 0.83 W and Asurface, $m = p\Delta x$ for other nodes. Analysis The area of the window and the individual resistances are A = (1.2 m) × (2 m) = 2.4 m 2 1 1 = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 1 1 = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 1 1 = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.003 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 1 1 = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 1 1 = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2.4 m 2) 0.012 m L = 0.0417 °C/W h1 A (10 W/m 2 .°C)(2. 0.0167 o C/W Ro = Rconv, 2 = 2 o h2 A (25 W/m. These properties will be used for both fresh and frozen chicken. The paddle wheel work done is to be determined. 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 3-103 Chapter 3 Steady Heat Conduction Roverall = 1/Uoverall where Uoverall = 1/Uoverall where Uoverall = (Ufarea) air space and 0.20 for the ferrings and similar structures. 3 Thermal conductivities are Bar Bar constant. This is a reasonable assumption since the time period of the process is very short. 2 Heat transfer from the bottom surface of the block is negligible. 5-81 A composite plane wall consists of two layers A and B in perfect contact at the interface. That is, R-value = L/k. 4 N | = 0.006229 | / Then the average heat transfer coefficient can be determined from the modified Reynolds analogy to be h= C f ρ V C p = 0.006229 (1.204 kg/m 3)(10 m/s)(1007 J/kg · °C) = 46.54 W/m 2 · C 2/3 2 (0.7309) Pr 2/3 Them the rate of heat transfer becomes Q& = hA (T - T) = (46.54 W/m 2 · C)(32 m 2)(80 - 20)°C = 89,356 W 2 s s ∞ 6-25 Chapter 6 Fundamentals of Convection 6-50 A metallic airfoil is subjected to air flow. 5 Heat loss from the top and bottom surfaces is negligible. Analysis: Cubic block: This cubic block can physically be formed by the intersection of three infinite plane wall of thickness 2L = 5 cm. ° C)(0.0216 m2) η fin = 3-83 Chapter 3 Steady Heat Conduction Afinned = η fin $n\pi DL = 0.957 \times 864\pi$ (0.0025 m)(0.02 m) = 0130. Analysis The R-value of the existing wall for the winter conditions is Rexisting wall = 1 / U existing wall = 1 / 2.25 = 0.444 m 2 · ° C / W Noting that the added thermal resistances are in series, the overall R-value of the wall becomes R modified wall = Rexisting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 = 0.689 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.170 m 2 · °C/W Then the U-value of the wall becomes R modified wall = Revisiting wall + R brick + Rair layer = 0.44 + 0.075 + 0.075 m 2 · °C/W Then the R bric / U modified wall = 1 / $0.689 = 1.45 \text{ m} 2 \cdot ^{\circ} \text{C}$ / W The rate of heat transfer through the modified wall is Q& = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q& = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q& = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q& = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} wall wall is Q = (UA) (T - T) = (1.45 \text{ W/m} 2 \cdot ^{\circ} \text{C})(3 \times 7 \text{ m} 2)[22 - (-5)^{\circ} \text{C}] = 822 \text{ W} molecules in liquids, and by the molecular collisions in gases. They are related to each other by m& = $\rho V \&$ where ρ is density. Preparing very accurate but complex models is not necessarily a better choice since such models are not much use to an analyst if they are very difficult and time consuming to solve. 2 The thermal conductivity and emissivity are constant. 3 Heat transfer from the fin tips is negligible. m 3m D 1 - 0.25 I - 0.25 D=3m 55 z. The thermal conductivity of the plate material is to be determined. 3 h, T \propto Convection heat transfer coefficient is constant and uniform. ° C)(0.01 m2) Plate L 1 I = = 3.333
° C / W ho A (30 W / m2. ho hi or wall The specific heat and density of air at 1 atm and 3°C To Ti Δ are Cp = 1.004 kJ/kg.°C and ρ = 1.29 kg/m3 (Table A•••15. Then, k/k Q& 1 = hAs (Ts - T ∞) = Nu As (Ts - T ∞) D (D / n = k (V ∞ D) As (Ts - T ∞) = Nu As (Ts - T ∞) = Nu As (Ts - T ∞) D (D / n = k (V ∞ D) As (Ts - T ∞) = Nu As (Ts - T ∞ $= (2V_{\infty}) n | A(Ts - T_{\infty}) D(v)$ Taking the ratio of them yields Q& 2 ($2V_{\infty}$) $n = = 2n n \& Q1 V_{\infty}$ Air $V_{\infty} \rightarrow 2V_{\infty}$ 7-35 Pipe D Ts Chapter 7 External Forced Convection 7-46 The wind is blowing across the wire of a transmission line. The undesirable changes caused by microorganisms are off-flavors and colors, slime production, changes in the texture and appearances, and the spoilage of foods. Nodes 1, 2, 3, and 4 are interior nodes, and the finite difference formulation for a general interior nodes, and the finite difference formulation for a general interior nodes. cm and a semi-infinite medium. The number of 5-kW window air conditioning units required is to be determined. 4-6 Chapter 4 Transient Heat Conduction 4-18 A thin-walled glass containing milk is placed into a large pan filled with hot water to warm up the milk. Insulation Rate of heat loss Cost of heat loss Cost of heat loss Cost avoings Insulation cost Thickness W \$/yr \$/yr \$0 cm 101,794 9263 0 0 1 cm 11,445 1041 8222 2828 5 cm 2515 228 9035 3535 9 cm 1413 129 9134 8483 10 cm 1273 116 9147 9189 11 cm 1159 105 9158 9897 12 cm 1064 97 9166 10,604 Therefore, the thickest insulation that will pay for itself in one year is the one whose thickness is 9 cm. ft 2 . 5-15C In a medium in which the finite difference formulation of a general interior node is given in its simplest form as Tm-1 - 2Tm + Tm+1 g& m + = 0 k $\Delta x 2$ (a) heat transfer in this medium is steady, (b) it is one-dimensional, (c) there is heat generation, (d) the nodal spacing is constant, and (e) the thermal conductivity is constant. temperature of $(Ts + T\infty)/2 = (65+35)/2 = 50^{\circ}C$ are (Table A-15) k = 0.02735 W/m.°C $\upsilon = 1.798 \times 10$ -5 m 2 /s Pr = 0.7228 Note that the atmospheric pressure will only affect the kinematic viscosity. Properties The thermal conductivity is given to be k = 180 W/m.°C. Nodes 1, 2, 3, and 4 in the plate and 6, 7, 8, and 9 in the soil are interior nodes, and thus for them we can use the general finite difference relation expressed as Tm - 1 - 2Tm + Tm + 1 = 0 (since g& = 0) k Δx 2 The finite difference equation for node 0 on the left surface and node 5 at the interface are obtained by applying an energy balance on their respective volume elements and taking the direction of all heat transfers to be towards the node under consideration: T - T Node 0 (top surface) : $h(T \infty - T0) + k$ plate 1 0 = 0 $\Delta x1$ Convection T0 - 2T1 + T2 = 0 Node 1 (interior) : $h, T \infty$ T1 - 2T2 + T3 = 0 Node 2 (interior) : $0 \cdot T2 - 2T3 + T4 = 0$ Node 3 (interior): $1 \cdot T 3 - 2T4 + T5 = 0$ Node 4 (interior): $2 \cdot Plate 3 \cdot 1$ in T - T5 T - T5 k plate 4 Node 5 (interface): $+ k soil 6 = 0.4 \cdot \Delta x1 \Delta x 2.5 \cdot T5 - 2T6 + T7 = 0$ Node 6 (interior): $7 \cdot T8 - 2T9 + T10 = 0$ Node 9 (interior): $8 \cdot where \Delta x1 = 1/12$ ft, $\Delta x2 = 0.6$ ft kplate = 7.2 Btu/h·ft·°F, ksoil = 0.49 Btu/h·ft·°F, 2 h = 3.5 Btu/h·ft·°F, 7 m = 80°F, and T10 = 50°F. The thickness of the insulation that needs to be used is to be determined. m) 2 = 30.19 m 2 1 1 Ro = = 0.000946 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation = Ts1 Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30.19 m 2) Rinsulation Ro T∞2 r2 - r1 (1.55 - 1.5) m = 0.0489 °C/W 4 \pi kr1 r2 4 \pi (0.035 W/m .°C)(30 W/m.°C)(1.55 m)(1.5 m) Rtotal = Ro + Rinsulation = 0.000946 + 0.0489 = 0.0498 °C/W T -T [15 - (-196)]°C Q& = s1 ∞ 2 = = 4233 W Rtotal 0.0498 °C/W Q& 4.233 kJ/s Q& = m& h fg \rightarrow m& = = 0.0214 kg/s h fg 198 kJ/kg (c) The heat transfer rate and the rate of evaporation of the liquid with 2-cm thick layer of superinsulation is A = π D 2 = π (3.04 m 2 = 29.03 m 2 1 1 Ro = = 0.000984 °C/W ho A (35 W/m 2.°C)(29.03 m 2) Rinsulation = Ts1 Rinsulation = Ts1 Rinsulation = Ts1 Rinsulation = 0.000984 + 13.96 °C/W 3-66 Ro T∞2 Chapter 3 Steady Heat Conduction T -T [15 - (-196)]°C. 4 Heat transfer at the top and bottom surfaces is negligible. 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 g(x) q0 T1 - T0 h, $T \propto Left$ boundary node: $q \propto 0 A + kA + g \propto 0$ ($A \Delta x / 2$) = 0 $\Delta x \Delta x$ Right boundary node: $kA T3 - T4 + q \propto 0$ $hA(T \propto -T4) + g\& 4(A\Delta x/2) = 0 \Delta x 0 \cdot \cdot 1 \cdot 2 \cdot 3 \cdot 4 5-18$ A plane wall with variable heat generation and constant thermal conductivity is subjected to insulation at the right boundary (node 5). 3 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. The thermal resistance and heat transfer rate through the solid stud are Stud L 0.1 m R stud = = 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{ C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{ C} Q \&$ = 2.2 W R stud 3.636 °C/W kA (0.11 W/m. °C)(0.25 m 2) $\Delta T 8^\circ \text{ C} Q \&$ kA (50 W/m.°C)(0.000628 m 2) L 0.1 m = = 3.65 °C/W kA (0.11 W/m.°C)(0.25 - 0.000628 m 2) L Q& T1 R nails = R stud R stud R at 1 1 1 1 1 1 = + = + \rightarrow Rtotal = 1.70 °C/W (c) The effective conductivity of the nailed stud pair can be determined from Q& L (4.7 W)(0.1 m) Δ T $Q\& = k \text{ eff } A \rightarrow k \text{ eff} = = 0.235 \text{ W/m.} C \Delta TA (8^{\circ}C)(0.25 \text{ m } 2) L 3-30 \text{ T2 } T2 \text{ Chapter } 3 \text{ Steady Heat Conduction } 3-55 \text{ A wall is constructed of two layers of sheetrock spaced by } 5 \text{ cm} \times 12 \text{ cm wood studs}.$ In this case it is determined to be $\Delta tdefrost = N\Delta t = 47(5 \text{ s}) = 235 \text{ s} 5-121 \dots$. The rate of heat transfer through the plate is to be determined. 7-62 Chapter 7 External Forced Convection 7-68 Water is preheated by exhaust gases in a tube bank. The mathematical formulation of this heat transfers to be expressed for transient two-dimensional heat transfers to be towards the node under consideration, the implicit finite difference q0 formulations become Left boundary node (node 0): k 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 1i + 1 A T 0i + 1 - T 1i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i + 1 A T 0i + 1 - T 0i
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Noting that u = u(y), v = 0, and L Fluid $\partial P / \partial x = 0$ (flow is maintained by the motion of the upper plate rather than the pressure gradient), the xmomentum: $\rho \mid u \rightarrow = 0 + v \mid l = \mu 2 - \partial y / \partial x$ dy 2 $\partial y \setminus \partial x$ 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 no-slip condition. 4-44 A rib is roasted in an oven. Then the number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. Then the number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. Then the number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. Then the number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the no-slip condition. 4-44 A rib is roasted in an oven. The number of nodes because of the number of number o 0.251 - 0.25z2.4mz = 2.4m Then the steady rate of heat transfer from the tank becomes Q& = Sk (T - T) = (10.30 m)(0.55 W/m.°C)(18 - 0)°C = 102 W S = 2\pi D 1 2 If the ground surface is insulated, S = T1 = 18°C = 2\pi D D 1 + 0.25 z = 2\pi (1.4 m) = 7.68 m 1. The specific heat of water is 4.18 kJ/kg.°C (Table A-9). 4 The given heat transfer from the tank becomes Q& = Sk (T - T) = (10.30 m)(0.55 W/m.°C)(18 - 0)°C = 102 W S = 2\pi D 1 2 If the ground surface is insulated, S = T1 = 18°C = 2\pi D D 1 + 0.25 z = 2\pi (1.4 m) = 7.68 m 1. The specific heat of water is 4.18 kJ/kg.°C (Table A-9). 4 The given heat transfer coefficients accounts for the radiation effects. $\times 10 - 6 \text{ m } 2 / \text{s}$ The surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (Fig. The inner surface temperature is determined from 4-46 (conducted across the circuit board and is dissipated from the back side of the board to the ambient air, which is forced to flow over the surface by a fan. m2 Ri Ao = π Do L = π (01 . Assumptions 1 Heat transfer through the glass is given to be transient and two-dimensional. The properties at the node such as the temperature and the rate of heat generation represent the average properties of the element. The temperature of the device at the end of the 5-min operation with and without a heat sink. Also, heat transfer problems can not be solved analytically if the thermal conditions are not sufficiently simple. The time it will take for the propane to evaporate completely as a result of the heat gain from the surroundings for the cases of no insulation are to be determined. 7-77C Superinsulations are obtained by using layers of highly reflective sheets separated by glass fibers in an evacuated space. Properties The thermal properties of cast iron are given to be k = 52 W/m.°C and α = 1.7×10-5 m2/s. Properties The properties of the chicken are given to be k = 0.26 Btu/h.ft.°F, ρ = 74.9 lbm/ft3, Cp = 0.98 Btu/lbm.°F, and α = 0.0035 ft2/h. Analysis The nodal network consists of 3 nodes, and the base temperature T0 at node 0 is specified. ° C)[π (0.024 m)(0.5 m)] Rpipe = Rconv, Rtotal = Rpipe + Rconv, o = 0.3627 + 0.6631 = 10258 °C / W. Analysis This insulated plate whose thickness is L is equivalent to one-half of an uninsulated plate can be treated as insulated surface. 2 The temperature along the flanges (fins) varies in one direction only (normal to the pipe). Properties The thermal conductivity, density, and specific heat of the balls are given to be k = 48 W/m.°C, $\rho = 7840$ kg/m3, and Cp = 440 J/kg.°C. 6-25C Turbulent thermal conductivity kt is caused by turbulent thermal co meat slabs is to be lowered to -18° C during cooling. However, we can solve this problem approximately by assuming a constant average temperature of (300+200)/2 = 250^{\circ}F for the potato during the process. Properties The thermal conductivities are given to be k = 0.17 W/m·°C for sheetrock, k = 0.11 W/m for fiberglass insulation. Analysis The thermal resistance between the transistor attached to the sink and the ambient air is determined to be R Ts T ∞ T – T ∞ (90 – 20)° C Δ T \rightarrow Rcase – ambient 40 W Q& . The Reynolds number in this case is V L [(60 × 5280 / 3600) ft/s](11 ft) Re L = ∞ = =
5.704 × 10 6 -3 2 v 0.1697 × 10 ft /s L = 11 ft which is greater than the critical Reynolds number. This is a control volume since mass crosses the system boundary during the process. 2 All the kinetic energy of cars is converted to thermal energy. 3 Thermal properties of the wall are constant. The rate of heat transfer to air and the pressure drop of air are to be determined. $\times 10 - 6 \text{ m } 2 / \text{s}(900 \text{ s}) \alpha \Delta t$ (015 $\tau = 2 = 0.002160 \Delta x$ (0.25 ft) 2 Node 6 (convection) : q& b + k The absorption of solar radiation is given to be g& (x) = q& s (0.859 - 3.415x + 6.704 x 2 - 6.339 x 3 + 2.278 x 4) where q& s is the solar flux incident on the surface of the pond in W/m2, and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the free surface of the pond in W/m2 and x is the distance form the pond in W/m2 and x is the d the pond, in m. 5-45C A region that cannot be filled with simple volume elements such as strips for a plane wall, and rectangular elements for two-dimensional conduction is said to have irregular boundaries. Using the explicit method, the time it takes to defrost the steaks is to be determined. The solutions are structured into the following sections to make it easy to locate information and to follow the solution procedure, as appropriate: Solution Assumptions Properties Analysis Discussion - The problem is posed, and the quantities to be found are stated. 3-85C Yes, the measurements can be right. If the surface temperature of the concrete floor is not to exceed 40 °C, the minimum burial depth of the steam pipes below the floor surface is to be determined. 3 The contact resistance between the transistor and the heat sink is negligible. k (2.5 W/m.°C) To determine the center temperature, the product solution can be written as [θ (0,0, t) block = [θ (0, t) wall] θ (0, t) cyl] 2 T (0,0, t) - T ∞ (| A e $-\lambda 12\tau$ | = | A1e $-\lambda 1\tau$ | | | wall 1 cyl Ti - $T \propto \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (1.104) (1.0931)e^{-(0.8516)} (1.104) = 0.352 \rightarrow T (0,0,t) = 331^{\circ}C 20 - 500 \text{ After 20 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) = 0.107 \rightarrow T (0,0,t) = 349^{\circ}C 20 - 500 \text{ After 60 minutes} \{ \} \{ \} 2 2 T (0,0,t) - 500 = (1.0580)e^{-(0.5932)} (2.208) (1.0931)e^{-(0.8516)} (2.208) (2.208) (0.008)e^{-(0.8516)} (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.208) (2.$ 6.624) (1.0931)e -(0.8516) (6.624) = $0.00092 \rightarrow T$ (0,0,t) = $500^{\circ}C$ 20 - 500 Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction is applicable. Therefore, there can be no temperature difference between different parts of the wall; that is, the temperature in a plane wall must be uniform in steady operation. Assumptions 1 Steady operating conditions exist since the surface temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C, the properties of air at 1 atm and this temperature of 5°C. 50°C Right boundary node: k i +1 i T5i - T6i Δx T6 - T6 + q& 0 = ρ C Δx 2 Δt i +1 i T1i - T0 Δx T6 - T6 i = ρ A C Q& left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt + left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt = ρ C Δx 76 - T6 i = ρ A C Q& left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer becomes i i 3 Qleft surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer becomes i i 3 Qleft surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer at left surface + kA Δx 2 Δt Heat transfer becomes i i 3 Qleft surface + kA Δx 2 Δt Heat transfer becomes i i 3 Qleft surface + kA Δx 2 Δt Heat transfer becomes i i 3 Qleft surface + kA Δx 2 Δt Heat transfer becomes i i 3 Qleft surface + kA Δx 4 Δt = ρ C Δx + ρ = ρ C Δx + ρ = ρ C Δx + ρ = ρ = ρ C Δx + ρ = ρ =1 (35) |At | / 5-74 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). The rate of heat transfer from the air to the plate is to be determined. W / m ° C)(1.964 × 10 - 3 m 2) = 1.2 W L 0.15 m Discussion: The steady rate of heat conduction can differ by orders of magnitude, depending on the thermal conductivity of the material. 2 The thermal properties of the concrete wall are constant. The total heat transfer area is $A = 2(40 \times 9 + 30 \times 9) = 1260$ ft 2 Wall The rate of heat loss during the daytime is $T - T(55 - 45)^\circ$ F O& day = kA 1 2 = (0.40 Btu / h.ft.
3-43)^\circ Chapter 3 Steady Heat Conduction 3-67 A spherical container filled with iced water is subjected to convection and radiation heat transfer at its outer surface. Therefore, the Biot number is more likely to be less than 0.1 for the case of the solid cooled in the air 4-4C The temperature drop of the potato during the second minute will be less than 4 °C since the temperature of a body approaches the temperature of the surrounding medium asymptotically, and thus it changes rapidly at the beginning, but slowly later on. 3 Thermal conductivities are constant. Then the temperature of the oranges becomes θ (ro, t) – T ∞ sin(λ 1 ro / ro) sin(λ 1 ro) sin((1.2732)e - (1.5708)(1.224) = 0.0396 Ti $- T \propto 1.5708 \lambda 1$ ro / ro T (ro , t) $- (-6) = 0.0396 \rightarrow T$ (ro , t) = -5.2 °C 15 - (-6) which is less than 0°C. P2-45). Assumptions 1 A standing man can be modeled as a 30-cm diameter, 170-cm long vertical cylinder with both the top and bottom surfaces insulated. 2 The thermal properties of the bodies are constant. Assumptions 1 Heat conduction in the body is two-dimensional, and thus the temperatures. Thus the flow is laminar. The general explicit finite difference form of an interior node for transient two-dimensional heat conduction is expressed as i i i i i +1 Thode = τ (Tleft + Ttop + Tright + Tbottom) + (1 - 4 τ) Thode + $\tau \cdot 8$ h, i g& node 12 k There is symmetry about the vertical, horizontal, and diagonal lines passing through the center. These two velocities are equal if the flow is uniform and the body is small relative to the scale of the free-stream flow. 3 The thermal properties of the slabs are constant. 1-106 A spacecraft in space absorbs solar radiation while losing heat to deep space by thermal radiation. Then the dimensionless temperature at the center of the plane wall is determined from θ o, cyl = 2 2 T 0 - T ∞ = A1 e - λ 1 τ = (1.0110)e - (0.2948) (8.286) = 0.4921 Ti - T ∞ The center temperature of the semi-infinite cylinde then becomes 4-71 Chapter 4 Transient Heat Conduction $T(x,0,t) - T\infty = \theta$ semi -inf $(x,t) \times \theta$ o, cyl = 0.8951 × 0.4921 = 0.4405 | -infinite $T(x,0,t) - 10 \rightarrow T(x,0,t) = 71.7^{\circ}C$ semi cylinder $T(x,0,t) - 10 \rightarrow T(x,0,t) = 71.7^{\circ}C$ water. Properties The thermal conductivities of the plaster, brick, and covering are given to be $k = 0.72 \text{ W/m} \cdot \text{C}$, $k = 0.36 \text{ W/m} \cdot \text{C}$, $k = 1.40 \text{ W/m} \cdot \text{C}$, k =transfer takes place, Ts is the surface temperature and T ∞ is the temperature of the fluid sufficiently far from the surface. ° F)(125 . Analysis (a) Noting that heat transfer is one-dimensional in the radial r direction, the mathematical formulation of this problem can be expressed as T = 160°F d (dT) | r |=0 dr (dr) Steam 250°F h=1.25 dT (r1) = $h[T \infty - T(r1)] dr - k and T(r2) = T2 = 160^{\circ} FL = 15 ft$ (b) Integrating the differential equation once with respect to r gives r dT = C1 dr Dividing both sides of the equation above by r to bring it to a readily integrating, dT C1 = r dr T(r) = C1 ln r + C2 where C1 and C2 are arbitrary constants. h, $T \infty \Delta x \ 0 \cdot \cdot 1 \cdot 2 \cdot 3 \cdot 4$ The system of 4 equations with 4 unknown temperatures constitute the finite difference formulation of the problem. The radiation heat loss can further be reduced by coating the ball with a lowemissivity material. 2 Heat transfer from the back surface of the board is negligible. Therefore, T3 = T2, T6 = T4, and T1, T2, T4, and T5 are the only 4 unknown nodal temperatures, and thus we need only 4 equations to determine them uniquely. The percentages of heat conductivity of the board are to be determined. The outer surface of the pipe is subjected to convection to a medium at To with a heat transfer coefficient of h. 1-118C (a) Draft causes undesired local cooling of the human body by exposing parts of the body to high heat transfer coefficients. 3 The heat transfer coefficients for wrapped and unwrapped potatoes are the same. (b) Heat conductivity is determined directly from the Plat steady onedimensional heat conduction relation to be T – T Q& / A 500 W/m 2 = = 313 W/m.°C Q& = kA 1 2 \rightarrow k = L L(T1 – T2) (0.02 m)(80 - 0)°C Q 80°C 0° 1-83 Four power transistors are mounted on a thin vertical aluminum plate that is cooled by a fan. ° F) π (3.24 / 12) 2 ft 2 Rtowel = Rconv. Then the energy balance for this steady-flow system can be expressed in the rate form as = $\Delta E\&$ system $\hat{E}0$ (steady) = $0 \rightarrow E\&$ in = E& out E& - E& out 1in424 3 144 42444 3 Rate of net energy transfer by heat, work, and mass Rate of change in internal, kinetic, potential, etc. Properties of wood are given to be k = 0.17 W/m.°C, $\alpha = 1.28 \times 10-7$ m2/s Analysis The Biot number is Bi = hro (13.6) W/m 2.°C)(0.05 m) = = 4.00 k (0.17 W/m.°C) The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, 1 0 λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 = 1.9081 and A1 = 1.4698 Once the constant λ 1 $Ti - T \propto 2.420 - 500 = (1.4698)e - (1.9081) \tau$ (0.2771) $\rightarrow \tau = 0.251.10 - 500$ which is above the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can be used. 5-22 Chapter 5 Numerical Methods in Heat Conduction 5-34 "IPROBLEM 5-34" "GIVEN" k=15.1 "[W/m-C], parameter to be varied" "epsilon=0.6] parameter to be varied T 0=95 [[C]" T infinity=25 "[C]" w=0.002 "[m]" s=0.01 "[m]" h=13 "[W/m^2-C]" T surr=295 "[K]" DELTAx=0.015 "[m]" sigma=5.67E-8 "[W/m^2-C]" T surr=295 "[C]" T surr=295 the unknown temperatures at 12 nodes are determined to be T 0-2*T 1+T 2+h*(p*DELTAx^2)/(k*A)*(T infinityT 1)+epsilon*sigma*(p*DELTAx^2)/(k*A)*(T infinityT 2)+epsilon*sigma*(p*DELTAx^2)/(k*A)*(T surr^4-(T 2+273)^4)=0 mode 2 T 2-2*T 3+T 4+h* $(p*DELTAx^2)/(k*A)*(T infinityT 3)+epsilon*sigma*(p*DELTAx^2)/(k*A)*(T surr^4-(T 3+273)^4)=0$ "mode 3" T 3-2*T 4+T 5+h*(p*DELTAx^2)/(k*A)*(T infinityT 5)+epsilon*sigma*(p*DELTAx^2)/(k*A)*(T surr^4-(T 3+273)^4)=0 "mode 4" T 4-2*T 5+T 6+h*(p*DELTAx^2)/(k*A)*(T surr^4-(T 3+273)^4)=0 $(T 5+273)^4)=0$ "mode 5" T 5-2*T 6+T 7+h*(p*DELTAx^2)/(k*A)*(T infinityT 6)+epsilon*sigma*(p*DELTAx^2)/(k*A)*(T surr^4-(T 6+273)^4)=0 "mode 6" T 6-2*T 7+T 8+h*(p*DELTAx^2)/(k*A)*(T surr^4-(T 7+273)^4)=0 $(T_infinityT_8) + epsilon*sigma*(p*DELTAx^2)/(k*A)*(T_surr^4-(T_8+273)^4) = 0$ "mode 8" T_8-2*T_9+T_10+h*(p*DELTAx^2)/(k*A)*(T_infinityT_9) + epsilon*sigma*(p*DELTAx^2)/(k*A)*(T_infinityT_9) + epsilon*(T_infi "mode 10" T 10-2*T 11+T 12+h*(p*DELTAx^2)/(k*A)*(T infinityT 11)+epsilon*sigma*(p*DELTAx^2)/(k*A)*(T surr^4-(T 12+T 13+h*(p*DELTAx^2)/(k*A)*(T infinityT 12)+epsilon*sigma*(p*DELTAx^2)/(k*A)*(T surr^4-(T 12+T 13+h*(p*DELTAx^2)/(k*A)*(T surr^4-(T 12+T surr^4-($(T infinityT 13) + epsilon*sigma*(p*DELTAx/2+A)*(T surr^4-(T 13+273)^4) = 0 mode 13"T tip=T 13"(c)"A s 0 = p*DELTAx/2 A s 13 = p*DELTAx/2 A s 13$ T infinity)+epsilon*sigma*A s $0*((T 0+273)^4-T surr^4) Q$ dot 2=h*A s*(T 1-T infinity)+epsilon*sigma*A s*((T 1+273)^4-T surr^4) Q dot 2=h*A s*((T 1+273)^4-T surr^4) Q Q dot 4=h*A s*(T 4-T infinity)+epsilon*sigma*A s*((T 4+273)^4-T surr^4) Q dot 5=h*A s*(T 5-T infinity)+epsilon*sigma*A s*((T 5+273)^4-T surr^4) Q dot 6=h*A s*(T 6+273)^4-T surr^4) Q dot 6=h*A s*(T 6+273)^4-T surr^4) Q dot 8=h*A s*(T 8-273)^4-T surr^4) Q dot 6=h*A s*(T 6+273)^4-T s $T_infinity) + epsilon*sigma*A_s*((T_8+273)^4-T_surr^4) Q_dot_9 = h*A_s*(T_9-T_infinity) + epsilon*sigma*A_s*((T_9+273)^4-T_surr^4) Q_dot_10 = h*A_s*(T_10-T_infinity) + epsilon*sigma*A_s*((T_10+273)^4-T_surr^4) Q_dot_10 = h*A_s*((T_10+273)^4-T_surr^4) Q_dot_1$ T
infinity)+epsilon*sigma*A s 13*(T 12+273)^4-T surr^4) Q dot 13=h*A s 13*(T 13-T infinity)+epsilon*sigma*A s 13*((T 13+273)^4-T surr^4) 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 Ttip [C] 24.38 25.32 27.28 29.65 32.1 34.51 36.82 39 41.06 42.98 44.79 46.48 48.07 49.56 50.96 52.28 53.52 54.69 55.8 56.86 Q [W] 0.6889 1.156 1.482 1.745 1.969 2.166 2.341 2.498 2.641 2.772 2.892 3.003 3.106 3.202 3.291 3.374 3.452 3.526 3.595 3.66 5-24 Chapter 5 Numerical Methods in Heat Conduction ε 0.1 0.15 0.2 0.25 0.3 0.35 0.4 0.45 0.5 0.6 0.65 0.7 0.75 0.8 0.85 0.9 0.95 1 Ttip [C] 25.11 25.03 24.96 24.89 24.82 24.76 24.7 24.64 24.59 24.53 24.48 24.43 24.39 24.34 24.3 24.26 24.22 24.18 24.14 Q [W] 0.722 0.7333 0.7445 0.7555 0.7665 0.7773 0.7881 0.7987 0.8092 0.8197 0.83 0.8403 0.8504 0.8001 0.9099 60 4 55 3.5 50 3 Q T tip [C] 2.5 40 T tip 2 35 1.5 30 1 25 20 0 50 100 150 200 250 k [W /m -C] 5-25 300 350 (W /m -C] 5-25 (W 0.5 400 O [W] 45 Chapter 5 Numerical Methods in Heat Conduction 25.2 0.92 25 0.88 O T tip 24.8 24.6 0.8 24.4 0.76 24.2 24 0.1 0.2 0.3 0.4 0.5 ϵ 0.6 5-26 0.7 0.8 0.9 0.72 1 O [W] T tip [C] 0.84 Chapter 5 Numerical Methods in Heat Conduction 5-35 One side of a hot vertical plate is to be cooled by attaching aluminum fins of rectangular profile. The fluid motion in natural convection is due to buoyancy effects only. 4-93 Chapter 4 Transient Heat Conduction 4-100 The chilling room of a meat plant with a capacity of 350 beef carcasses is considered. This system of 6 equations with six unknown temperatures constitute the finite difference formulation of the problem. The temperature inside the refrigerator at the end of this 6 h period is to be determined. Analysis (a) Noting that heat transfer is one-dimensional in the radial r direction, the mathematical formulation of this problem can be expressed as d $(2 \text{ dT}) h | r | = 0 \text{ dr} (dr / T_{\infty} \text{ N2 r1} and T (r1) = T1 = -196^{\circ} \text{ C r2 r dT} (r2) - 196^{\circ} \text{C} - k = h[T (r2) - T_{\infty}] dr$ (b) Integrating the differential equation once with respect to r gives dT r2 = C1 dr Dividing both sides of the equation above by r to bring it to a readily integrable form and then integrating, C dT C1 \rightarrow T (r) = -1 + C2 = 2 dr r r where C1 and C2 are arbitrary constants. Properties The thermal properties of steel plates are given to be k = 43 W/m.°C and α = 1.17×10-5 m2/s Analysis The characteristic length of the plates and the Biot number are V Lc = L = 0.02 m Steel plates As Ti = -15°C 2 hL (40 W/m.°C) T $\infty = 50°C$ Since Bi < 0.1, the lumped system analysis is applicable. Air, 55°C Power transist 1-44 Chapter 1 Basics of Heat Transfer 1-86 "GIVEN" L=0.004 "[m]" D=0.006 "[m]" h=30 "[W/m^2-C]" T infinity=55 "[C]" T case max=70 [C], parameter to be varied "ANALYSIS" A=pi*D*L+pi*D^2/4 Q dot=h*A*(T case max=70 [C], parameter to be varied "0.06298 0.07775 0.08553 0.09331 0.1011 0.1089 0.12 0.1 Q [W] 0.08 0.06 0.04 0.02 0 60 65 70 75 80 T case, m ax [C] 1-45 85 90 Chapter 1 Basics of Heat Transfer 1-87E A 200-ft long section of a steam pipe passes through an open space at a specified temperature. Assumptions 1 Heat transfer 1-87E A 200-ft long section of a steam pipe passes through the wall is given to be transient, and the thermal conductivity and heat Convection g(x) Radiation generation to be variables. 3-59 Chapter 3 Steady Heat Conduction 3-78E 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. boundary conditions give x = 0: h1 [T ∞ 1 - (C1 × 0 + C 2)] = -kC1 x = L: -kC1 = h2 [(C1 L + C 2) - T ∞ 2] Substituting the given values, these equations simultaneously give C1 = -45.44 C 2 = 20 Substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (x) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = $20 - 45.44 \times 0.2 = 10.9^{\circ}C T$ (L) = emissivity of the base surface is given to be ε = 0.6. Analysis At steady conditions, the 1000 W energy supplied to the iron will be dissipated to the surroundings by convection and radiation heat transfer. 4-8 Chapter 4 Transient Heat Conduction 4-20 An iron whose base plate is made of an aluminum alloy is turned on. Using the energy balance approach and taking the direction Δx 2 of all heat transfers to be towards the node under consideration, the explicit transient finite difference formulations become Node 1 (at midpoint): 4 $\sigma = 71i$ + 1 + 1 - 71i + $T2 - T2 (\Delta x) i 4 i = \rho A C [[Tsurr - (T2)] + h] p [(T - T2) + kA 2 / \Delta x 2 \Delta t 2 / where A = \pi D 2 / 4 is the cross-sectional area and p = \pi D is the perimeter of the fin. The finite difference formulation of the boundary nodes is to be determined. Analysis Heat loss from the room during a 24-h period is Qloss = (10,000 kJ/h)(24 h) = 240,000 kJ Taking$ the contents of the room, including the water, as our system, the energy balance can be written as E - E 1in424out 3 Net energy transfer by heat, work, and mass = ΔE system 1 424 3 $\rightarrow -$ Qout = $\Delta U = (\Delta U)$ water + (ΔU) air \otimes 0 Change in internal, kinetic, potential, etc. Note that the thermal contact resistance in this case is greater than the sum of the thermal resistances of both plates. Then the heat loss from the part of the tube that is on the ground is As = $\pi DL = \pi (0.05 \text{ m})(2 \text{ m}) = 0.3142 \text{ m} 2 \text{ Q} = 498 \text{ W}$ Considering the shape factor, the heat loss for vertical part of the tube can be determined from S = $2\pi (3 \text{ m}) 2\pi L = \pi (0.05 \text{ m})(2 \text{ m}) = 0.3142 \text{ m} 2 \text{ Q} = 498 \text{ W}$ $3.44 \text{ m} = 4 \text{ L} \left(\right) \left[4(3 \text{ m}) \right] \ln \left| \left| \ln \right| \right| \left| \left| \Delta D \right| \right| (0.05 \text{ m}) \right] 3m 20 \text{ m} 80^{\circ} \text{C} \text{ Q} \text{ e} = \text{Sk} (T1 - T2) = (3.44 \text{ m})(15 \text{ Analysis The length of time the snow pack stays on the ground is } t = (60 \text{ days})(24 \text{ hr} / \text{ days})(3600 \text{ s} / \text{ hr}) = 5.184 \times 106 \text{ s}$ The surface is kept at -18°C at all times. Also, the mesh Fourier number is $\tau = \alpha \Delta t \Delta x 2 = (971 \text{ Assumptions 1 Steady})$ operating conditions exist. Analysis The characteristic length and Biot number for the glass of milk are Lc = π ro 2 L π (0.03 m) 2 (0.07 m) V = = = 0.01050 m As 2π of $(0.03 \text{ m}) = 2.076 > 0.1 (0.607 \text{ W/m} \cdot \text{C}) \text{ k}$ For the reason explained above we can use the lumped system analysis to determine how long it will take for the milk to warm up to 38°C: b = hAs 120 W/m 2. °C h = = $0.002738 \text{ s} - 1 \text{ pC pV } \rho \text{C p Lc}$ (998 kg/m 3)(4182 J/kg.°C)(0.0105 m) -1 T (t) - T \propto 38 - 60 = e - bt \rightarrow = e - (0.002738 s)t \rightarrow t = 348 s = 5.8 min 3 - 60 Ti - T \propto Therefore, it will take about 6 minutes to warm the milk from 3 to 38°C. Properties The thermal conductivities are given to be $k = 61 \text{ W/m} \cdot \text{C}$ for steel and $k = 0.038 \text{ W/m} \cdot \text{C}$ for insulation. The unknown nodal temperatures and the rate of heat loss from the bar through a 1-ft long section are to be determined. $m = \rho V = \rho \text{Ir} \circ 2 \text{ L} = (1600
\text{ kg/m 3})[\pi (0.15 \text{ m}) 2 (3.5 \text{ m})] = 395.8 \text{ kg} \text{ Qmax} = m \text{ C} \text{ p} [\text{T} \circ - \text{Ti}] = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kJ/kg} \cdot \text{C})(28 \text{ kg/m}) = (395.8 \text{ kg})(0.84 \text{ kg/m}) = (395.$ $-16)^{\circ}C = 3990 \text{ kJ}$ (c) To determine the amount of heat transfer until the surface temperature reaches to 27°C, we first determine 2 2 T (0, t) $-T \propto = A1 \text{ e} - \lambda 1 \tau = (1.3915) \text{ e} - (1.7240) (0.6253) = 0.2169 \text{ Ti} - T \propto Once the constant J1 = 0.5787$ is determined from Table 4-2 corresponding to the constant $\lambda 1$, the amount of heat transfer becomes (Q) $(T - T \infty) I 1$ ($\lambda 1$) 0.5787 | = 1 - 2 0 | = 1 - 2 × 0.2169 × = 0.854 | Q | | T - T | $\lambda 1$.7240 ∞ / 1 \ i \ max / cyl Q = 0.854(3990 k] + 115 Chapter 4 Transient Heat Conduction 4-122 Long aluminum wires are extruded and exposed to atmospheric air. 5-97C The discretization error (also called the truncation or formulation error) is due to replacing the derivatives by differences in each step, or replacing the actual temperature distribution between two adjacent nodes by a straight line segment. h.ft 2 · ° F / Btu Noting that the added and removed thermal resistances are in series, the overall R-value of the wall at 15 mph (winter) conditions is obtained by a straight line segment. replacing the summer value of outer convection resistance by the winter value, Rwall, 15 mph = Rwall, 7.5 mph - Ro, 7.5 mph + Ro, 15 mph = 1111. The volumetric 0.61 W/m. °C and $\alpha = 015$ S absorption coefficients of water are as given in the problem. 4-23 we have 2 - 6 2 h αt (20 W/m. °C) (1.6 × 10 m / s)(2 × 3600 s) = 2.98 = |T - T \omega 0.72 W/m.°C k = 0 1 - 0.4 m x Ti $- T \propto \xi = 1.87$ 2 $\alpha t 2$ (1.6 × 10 -6 m 2 / s)(2 × 3600 s) 1 - T - 2 = 0 \rightarrow T = 18.0°C 18 - 2 Discussion This last result shows that the semi-infinite medium assumption is a valid one. 4 The room is maintained at 20°C at all times. Using the energy balance approach and taking the direction of all heat ε Radiation transfers to be towards the node under consideration, the finite difference formulations become T Node 1 (at midpoint): Node 2 (at fin tip): T – T T – T 4 kA 0 1 + kA 2 1 + h($p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T14) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(<math>p\Delta x / 2$)(Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ (Tsurr – T24) = 0 $\Delta x \Delta x kA T1 - T24 + h(p\Delta x / 2)$ πD is the perimeter of the fin. Separating the variables in the above equation and integrating from r = r1 where T (r) = T to any r where T (r) = K (T) + \beta (T - T1) + \beta (T - T1) / 2] r 1 Substituting the Q& expression from part (a) and rearranging give T2 + 2 β T+ 2 k ave ln(r / r1) 2 (T1 - T2) - T12 - T1 = 0 β k 0 ln(r2 / r1) β which is a quadratic equation in the unknown temperature T. Properties The heat of vaporization and density of liquid propane at 1 atm are given to be 425 kJ/kg and 581 kg/m3, respectively. In natural convection, any fluid

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 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 ($\Delta y \times \Delta z$) Tm -1, n, r - Tm, n, r + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) Tm +1, n, r - Tm, n, r Δx Tm, n, r + 1 - Tm, n, r + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) Tm +1, n, r - Tm, n, r Δx Tm, n, r + 1 - Tm, n, r + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta z$) + k ($\Delta x \times \Delta$ -1, n, r -2Tm, n, r + Tm +1, n, r Δx 2 + Tm, n -1, r -2Tm, n, r + Tm, n, r -1 + Tm, n, r +1 Δz 2 For a cubic mesh with $\Delta x = \Delta y = \Delta z = 1$, and the relation above simplifies to Tm -1, n, r + Tm, n +1, r + Tm, n, r + Tm, n, r + Tm, n, r + Tm, n, r + Tm, n +1, r + Tm, n +1, r + Tm, n, r + Tm, n +1, r +1, r + Tm, n +1, r +1, to-remember form: Tleft + Ttop + Tright + Tbottom + Tfront + Tback - 6Tnode + 5-105 g& 0 1 2 = 0 k + g& 0 1 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 (15 mph winds) conditions are given in Table 3-6 to be Ro, 7.5 mph = Ro, summer (2.5 h,ft2.°F/Btu Inside WALL 3-109 Outside 7.5 mph Chapter 3 Steady Heat Conduction and Ro, 15 mph = $1 / 0.09 = 1111 \cdot 2$ Thermal conductivity and emissivity are constant. Then the energy balance for this process can be written as E - E 1in424out 3 = Net energytransfer by heat, work, and mass [mC ($32^\circ F - T$) ΔE system 1 424 $3 \rightarrow 0 = \Delta U \rightarrow (\Delta U)$ 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 Reynolds number, Re cr = $\rho V \propto x$ cr $\mu \rightarrow x$ cr = $\mu Re cr (0.891 \times 10 - 3 \text{ kg/m} \cdot \text{s})(5 \times 10 5) = 0.056 \text{ m} = 5.6 \text{ m}$ $cm \rho V \propto (997 \text{ kg/m 3})(8 \text{ m/s})$ The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation, $\delta cr = 5 \text{ x cr Re } 1 \text{ x}/2 = 5(0.056 \text{ m})(5 \times 105) 1/2 = 0.00040 \text{ m} = 0.4 \text{ mm}$ Therefore, the flow becomes turbulent after about 5 cm from the leading edges. 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 Δ T (15 - 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 temperatures 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 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 varies in both axial x- and radial r- directions. = 0.28 To $-T \propto 75 - 100$ k 1 = 0.25 Bi hr x ro o = 1 J ro ro From Fig. Then the continuity equation reduces to $\partial u \partial v \partial u$ Continuity: $+ = 0 \rightarrow = 0 \rightarrow u = u(y)$ 4500 rpm $\partial x \partial x \partial y$ Therefore, the x-component of velocity does not change in the flow direction (i.e., the velocity profile remains unchanged). 4 - 14a) Chapter 44a) Chapter 44a) Chapter 450 (Fig. 4 - 14a) Chapter 44a) 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 [$\theta(0, t) - T \propto f = |A1e - \lambda 1 \tau | | A1e - \lambda 1 \tau | A1e - \lambda | A1e - \lambda 1 \tau | A1e$ $(1.0931)e - (0.8516)(1.104) = 0.27120 - 500T(0,0,t) = 370^{\circ}C$ After 20 minutes 2 2 T (0,0,t) - 500 = (1.1016)e - (0.7910)(2.208) = 0.06094 \rightarrow T (0,0,t) = 471^{\circ}C 20 - 500 After 60 minutes 2 2 T (0,0,t) - 500 = (1.1016)e - (0.7910)(6.624) = 0.0001568 \rightarrow T (0,0,t) = 500^{\circ}C 20 - 500 Note that $\tau > 0.2$ 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 = $Q\&0 = -Q\&L = -kA dy = -A \mu V 2kL 2L 2L$ First, the velocity and the surface area are (1 min) $V = \pi DN\& = \pi(0.05 \text{ m})(0.10 \text{ m}) = 0.01571 \text{ m} 2$ (a) Air: (1.918 × 10 - 5 N · s/m 2)(6.545 m/s) 2 $\mu V 2 = -(0.01571 \text{ m} 2) W\&$ mech = $-A 2L 2(0.0005 \text{ m}) (1W) = -0.013 \text{ W} (1 \text{ N} \cdot \text{m/s}) (b) \text{ Water} (0.653 \times 10 - 3 \text{ N} \cdot \text{s/m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) W \text{ mech} = Q \text{ 0} = -A = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = Q \text{ 0} = -A = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ W} \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.01571 \text{ m 2}) (6.545 \text{ m/s}) 2 (1 \text{ W}) \mu V 2 \text{ mech} = -(0.$ m/s / 6-17 Chapter 6 Fundamentals of Convection 6-43 The flow of fluid between two large parallel plates is considered. 5 The thermal contact resistance at the interface is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific heat of water is 1 kg/L, and the specific wall and the individual resistances are A = (6 m) × (2.8 m) = 16.8 m 2 L1 0.01 m = $0.00165 \degree C/W k1 A (0.36 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.01653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.00350\degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.00085 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A (0.72 W/m.\degreeC)(16.8 m 2) L 0.20 m = 2 = 0.001653 \degree C/W k2 A$ covering = R o = R conv, 2 R total, no ins = R1 + R 2 + R3 + Rconv, 2 = 0.00165 + 0.0085 + 0.00350 = 0.02253 °C/W The rate of heat transfer after insulation is $Q_{\&} = 0.15 \times 666 = 99.9$ W ins no ins The total thermal resistance with the foam insulation is R total = R1 + R2 + R3 + Rfoam + Rconv, 2 = $0.02253 \text{ °C/W} + R1 \text{ T1} \text{ L4} (0.025 \text{ W/m}.^{\circ}\text{C})$ The thickness of insulation is determined from T - T $\infty 2 \rightarrow 99.9 \text{ W} = Q\&$ ins = 1 R total (23 - 8)°C L4 $0.02253 \text{ °C/W} + (0.42 \text{ W.m/°C}) \rightarrow L4 = 0.054 \text{ m}$ = 5.4 cm The outer surface temperature of the wall is determined from T - T ∞ 2 (T2 - 8)°C Q& ins = 2 \rightarrow 99.9 W = \rightarrow T2 = 8.3°C R conv 0.00350 °C/W 3-130 T ∞ 2 Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady
Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) The total thermal resistance with the fiberglass insulation is R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + L4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + R4 (0.036 W/m Chapter 3 Steady Heat Conduction (b) R total = R1 + R2 + R3 + Rfiber glass + Rconv, 2 = 0.02253 °C/W + R4 (0.036 W/m Chapter 3 Steady Heat Cond $^{\circ}C$)(16.8 m) 2 = 0.02253 $^{\circ}C/W + L4$ (0.6048 W.m/ $^{\circ}C$) The thickness of insulation is determined from T - T $_{\infty}$ 2 (23 - 8) $^{\circ}C$ \rightarrow 99.9 W = \rightarrow L 4 = 0.077 m = 7.7 cm Q& ins = 1 L4 R total 0.02253 $^{\circ}C/W + (0.6048 W.m)^{\circ}C$ The outer surface temperature of the wall is determined from T - T $_{\infty}$ 2 (T2 - 8) $^{\circ}C$ Q& ins = 2 \rightarrow 99.9 = \rightarrow T2 = 8.3 $^{\circ}C$ Rconv 0.00350°C/W Discussion The outer surface temperature is same for both cases since the rate of heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the sum of the heat transfer to the steak during a time step i is the steak during a tis the steak duri and is expressed as i Qsteak = $2\pi 5 \Delta r \{h(T_{\infty} - T5i) + \epsilon \text{ plate}\sigma [(T_{\infty} + 273) 4] \} + \pi r [(r56 + r6) / 2](\Delta r / 2) \{h(T_{\infty} - T6i) + \epsilon \text{ plate}\sigma [(T_{\infty} + 273) 4] \} + \pi r (r45 - r42) \{h(T_{\infty} - T6i) + \epsilon \text{ plate}\sigma [(T_{\infty} + 273) 4] \} + \pi r (r45 - r42) \{h(T_{\infty} - T6i) + \epsilon \text{ plate}\sigma [(T_{\infty} + 273) 4] \} + \pi r (r45 - r42) \{h(T_{\infty} - T6i) + \epsilon \text{ plate}\sigma [(T_{\infty} + 273) 4] \} + \pi r (r45 - r42) \{h(T_{\infty} - T6i) + \epsilon \text{ plate}\sigma [(T_{\infty} + 273) 4] \} + \pi r (r45 - r42) \{h(T_{\infty} - T6i) + \epsilon \text{ plate}\sigma [(T_{\infty} + 273) 4] \}$ determined by finding the amount of heat transfer during each time step, and adding them up until we obtain 55.2 kJ. The rate of heat transfer from the wire per meter length when it is first exposed to the air is to be determined. The transfer from the wire per meter length when it is first exposed to the air is to be determined. under steady conditions are to be determined. 2 Heat is transferred uniformly from the entire front surface. 4-88 Chapter 4 Transient Heat Conduction 4-85 "!PROBLEM 4-85" "GIVEN" 2*L=0.20 "[m]" T_i=20 "[C]" T_infinity=1200 "[C]" T_o_0=300 [C], parameter to be varied h=80 "[W/m^2-C]" "PROPERTIES" k=236 "[W/m-C]" rho=2702 "[kg/m^3]" C p=0.896 "[kJ/kg-C]" alpha=9.75E-5 "[m^2/s]" "ANALYSIS" "This short cylinder can physically be formed by the intersection of a long cylinder of radius r o and a plane wall of thickness 2L" "For plane wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.1439 "w stands for wall A 1 w=1.0035 tau w=(alpha*time)/L^2 theta o w=A 1 w*exp(-lambda 1 w^2*tau w) "theta o w=(T o wT infinity)/(T i-T infinity)/(lambda 1 c^2*tau c) "theta o c=(T o cT infinity)/(T i-T infinity)" (T o o-T infinity)/(T i-T infinity)=theta o w*theta o c "center temperature of cylinder" V=pi*r_o^2*(2*L) m=rho*V Q_max=m*C_p*(T_infinity-T_i) 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" definition of cylinder" V=pi*r_o^2*(2*L) m=rho*V Q_max=m*C_p*(T_infinity-T_i) 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" definition of cylinder" V=pi*r_o^2*(2*L) m=rho*V Q_max=m*C_p*(T_infinity-T_i) 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" definition of cylinder" V=pi*r_o^2*(2*L) m=rho*V Q_max=m*C_p*(T_infinity-T_i) 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" definition of cylinder" V=pi*r_o^2*(2*L) m=rho*V Q_max=m*C_p*(T_infinity-T_i) 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" definition of cylinder" V=pi*r_o^2*(2*L) m=rho*V Q_max=m*C_p*(T_infinity-T_i) 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" definition of cylinder" V=pi*r_o^2*(2*L) m=rho*V Q_max=m*C_p*(T_infinity-T_i) 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" definition of cylinder" Q_k=0.5 J 1=0.0876 "From Table 4-2, at lambda 1 c" Q/Q max=Q w+Q c*(1-Q w) "total heat transfer" 4-89 Chapter 4 Transient Heat Conduction To,o [C] 50 100 150 200 250 300 350 400 450 500 550 600 650 700 750 800 850 900 950 1000 time [s] 44.91 105 167.8 233.8 303.1 376.1 453.4 535.3 622.5 715.7 815.9 924 1042 1170 1313 1472 1652 18611 2107 2409 Q [k]] 346.3 770.2 1194 1618 2042 2466 2890 3314 3738 4162 4586 5010 5433 5857 6281 6705 7129 7553 7977 8401 2500 9000 8000 2000 7000 6000 heat 5000 4000 1000 tim e 3000 2000 500 1000 0 [k]] tim e 1500 Chapter 4 Transient Heat Conduction 4-91 Chapter 4 Transient Heat Condu Special Topic: Refrigeration and Freezing of Foods 4-86C The common kinds of microorganisms are bacteria, yeasts, molds, and viruses. Also, the temperature at any point in the wall under summer design conditions is to be determined. Properties The emissivity of the outer surface of the collector is Q& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the collector is 2& Tsky = 50°F As = (5 ft)(15 ft) = 75 ft 2 Noting that the exposed surface area of the co convection and radiation becomes Air, 70°F Solar collecto Q& conv = hAs (T ∞ - Ts) = (2.5 Btu/h.ft 2 .°F)(75 ft 2)(100 - 70)°F = 5625 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 ×
10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2 .°F)(75 ft 2)(0.1714 × 10 - 8 Btu/h.ft 2)(0.1714 1 Basics of Heat Transfer Problem Solving Techniques and EES 1-110C Despite the convenience and capability the engineering courses. 4 Radiation effects are negligible. Node 1 (interior) : 150 • 180 • 180 • 200 • 4 180 • 7 200 • 2 • 5 • 8 $180 \cdot 3150 \cdot 6 \cdot 2009 \cdot 180 \cdot T1 = (180 + 180 + 2T2) / 4$ Node 2 (interior): T2 = (200 + T5 + 2T1) / 4 Node 3 (interior): $T5 = 4T2 / 4 = T2 \cdot 150 \cdot 180 \cdot 200$ Solving the equations above simultaneously gives $T1 = T3 = T7 = T9 = 185^{\circ}C$ $T2 = T4 = T5 = T6 = T8 = 190^{\circ}C$ Discussion Note that taking advantage of symmetry simplified the problem greatly. Analysis (a) The characteristic length of the balls and Oil Engine valve the Biot number are T = 45° C Ti = 800° C 1.8(π D 2 L / 4) 1.8D 1.8(0.008 m) V = = = 0.024 < 0.1 k (48 W/m.°C) Therefore, we can use lumped system analysis. The analytical methods are simple and they provide solution functions applicable to the entire medium, but they are limited to simple geometries. 3 The thermal properties of the aluminum are constant. 7-39 A steam pipe is exposed to windy air. 2 Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. The thermal conductivity of bearing is given to be k = 70 W/m-K. Therefore, the energy content of the wall does not change during steady heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation 2-31 For a medium in which the heat conduction equation is given by $\partial 2T \partial x 2 + \partial 2T \partial y 2 = 1 \partial T$: $\alpha \partial t$ (a) Heat transfer is transf generation, and (d) the thermal conductivity is constant. (a) The Biot number is calculated for each of the plane wall to be Bi A = Bi B = hL (6 W/m 2.°C)(0.2 m) = = 0.0231 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 k (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} (52 W/m.°C) Air 17°C hL (6 W/m 2.°C)(0.4 m) BiC = = 0.0462 \text{ k} Table 4-1, $\lambda 1(A,B) = 0.150$ and A1(A,B) = 1.0038 $\lambda 1(C) = 0.212$ and A1(C) = 1.0076 The Fourier numbers are $\alpha t \tau A,B = \tau C = L2 \alpha t 2 L = (1.70 \times 10 - 5 \text{ m } 2/\text{s})(45 \text{ min} \times 60 \text{ s/min})(0.4 \text{ m}) 2 = 1.1475 > 0.2 = 0.2869 > 0.2$ The center of the block (whose sides are 80 cm and 40 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 1.1475 > 0.2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 0.2869 > 0.2 The center of the block (whose sides are 80 cm and 40 m) 2 = 0.2869 > 0.2 cm) is at the center of the plane wall with 2L = 40 cm, at the center of the plane wall with 2L = 40 cm, and at the surface of the plane wall with 2L = 40 cm, and at the surface of the plane wall with 2L = 40 cm, and at the surface of the plane wall with 2L = 40 cm. Properties The conductivity and diffusivity are given to be k = 28 W/m °C and $\alpha = 12.5 \times 10 - 6$ m2 / s. The temperatures on the two sides of the circuit board are to be determined. The rate of heat generation per unit volume of the wire is Q& gen Q& gen 2000 W = = 1.455×10.8 W/m 3 g& = V wire $\pi ro 2 L \pi (0.0025 m) 2 (0.7 m) 110^{\circ} r$ The center temperature of the wire is then determined from Eq. 2-71 to be To = T s + g&ro 2 (1.455 × 10.8 W/m 3) (0.0025 m) 2 = $110^{\circ}C + = 121.4^{\circ}C 4k 4(20 W/m.^{\circ}C) 2-38 D$ Chapter 2 Heat Conduction Equation 2-78 Heat is generated in a long solid cylinder with a specified surface temperature. The thermal conductivity of glass wool insulation is given to be k = 0.038 W/m °C. Separation increases the drag coefficient drastically. The difference between the two solutions at each time step is called the local discretization error. Node 0 is on insulated boundary, and thus we can treat it as an interior note by using the mirror image concept. Properties The properties of air at the film temperature of $(Ts + T_{\infty})/2 = (65+35)/2 = 50^{\circ}$ C are (Table A-15) k = 0.02735 W/m.°C $v = 1.798 \times 10^{-5}$ m 2 /s Air $V_{\infty} = 4$ m/s $T_{\infty} = 35^{\circ}$ C Transistors Pr = 0.7228 Analysis The Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 55,617 v 1.798 × 10 - 5 m 2/s Ts = 65°C L = 25 cm which is less than the critical Reynolds number (5 × 105). 5-19 Chapter 5 Numerical Reynolds number (5 × 105). 5-19 Chapter 5 Numerical Reynolds number (5 × 105). $tanh(18.37 \text{ m}-1 \times 0.02 \text{ m}) = 0.957 = aL 18.37 \text{ m} - 1 \times 0.02 \text{ m}$ The fins can be assumed to be at base temperature provided that the fin area is modified by multiplying it by 0.957. Analysis We treat the trunks of the trees as an infinite cylinder since heat transfer is primarily in the radial direction. 8-2C Reynolds number for flow in a circular tube of diameter D is expressed as V D μ m& m& 4m& where V ∞ = = = and υ = Re = m 2 2 $\upsilon \rho \rho Ac \rho$ ($\pi D / 4$) $\rho \pi D$ Substituting, Re = Vm D υ = 4m& D $\rho \pi D$ 2 (μ / ρ) = m, Vm 4m& $\pi D \mu$ 8-3C Engine oil requires a larger pump because of its much larger density. The heat of fusion of water is given to be 333.7 kJ/kg. Properties The heat of vaporization and density of liquid nitrogen at 1 atm are given to be 198 kJ/kg and 810 kg/m3, respectively. °C/(0.33 × 1 m2) L 0.02 m = = = 2.33 °C / W kA (0.026 W / m. Q& conv = η hAs (T ∞ - Ts) (1)(0.0118 m)(60 - 25)°C Starting from heat transfer coefficient, Nusselt number, Reynolds number and finally free-stream velocity will be determined. 6-30C For flows with low velocity and for fluids with low viscosity the viscous dissipation term in the energy equation is likely to be negligible. Noting that the volume element centered about the general interior node (m, n) involves heat conduction from four sides (right, left, top, and bottom) and expressing them at previous time step i, the transient explicit finite difference formulation for a general interior node can be expressed as k ($\Delta y \times 1$) Tmi , n + 1 - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n + 1 - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x +
k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi , n $\Delta x + k$ 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0.02737 0.02947 0.03158 0.03579 0.03789 0.04 T [C] 791.1 789.1 783.2 773.4 759.6 741.9 720.2 694.6 665 631.6 594.1 552.8 507.5 458.2 405 347.9 286.8 221.8 152.9 80 k [W/m.C] 10 30.53 51.05 71.58 92.11 112.6 133.2 153.7 174.2 194.7 10.006316 0.002526 0.02737 0.02947 0.03158 0.03579 0.03789 0.04 T [C] 791.1 789.1 783.2 773.4 759.6 741.9 720.2 694.6 665 631.6 594.1 552.8 507.5 458.2 405 347.9 286.8 221.8 152.9 80 k [W/m.C] 10 30.53 51.05 71.58 92.11 112.6 133.2 153.7 174.2 194.7 10.006316 0.00579 0.03789 0.04 T [C] 791.1 789.1 783.2 773.4 759.6 741.9 720.2 694.6 665 631.6 594.1 552.8 507.5 458.2 405 347.9 286.8 221.8 152.9 80 k [W/m.C] 10 30.53 51.05 71.58 92.11 112.6 133.2 153.7 174.2 194.7 10.006316 0.00579 0.03789 0.04 T [C] 791.1 789.1 783.2 773.4 759.6 741.9 720.2 694.6 665 631.6 594.1 552.8 507.5 458.2 405 347.9 286.8 221.8 152.9 80 k [W/m.C] 10 30.53 51.05 71.58 92.11 112.6 133.2 153.7 174.2 194.7 10.006316 0.00579 0.03789 0.04 T [C] 791.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789.1 789 215.3 235.8 256.3 276.8 297.4 317.9 338.4 358.9 379.5 400 T0 [C] 1147 429.4 288.9 229 195.8 174.7 160.1 149.4 141.2 134.8 129.6 125.2 121.6 118.5 115.9 113.6 111.5 109.7 108.1 106.7 2-50 Chapter 2 Heat Conduction Equation 800 700 600 T [C] 500 400 300 200 100 0 0 0.005 0.01 0.015 0.02 0.025 0.03 0.035 0.04 r [m] 1200 1000 T 0 [C] 800 $600\ 400\ 200\ 0\ 50\ 100\ 150\ 200\ 250\ k\ [W/m\ -C]\ 2-51\ 300\ 350\ 400\ Chapter\ 2\ Heat\ Conduction\ Equation\ 2-91\ A\ long\ homogeneous\ resistance\ heater\ wire\ with\ specified\ surface\ temperature\ of\ (Ts\ +\ T\infty)/2\ =\ (60+25)/2\ =\ 42.5^\circC\ are\ (Table\ A-15)\ k\ =\ 0.02681\ W/m.$ °C $v = 1.726 \times 10^{-5} \text{ m } 2/\text{s Air V} \propto T \propto = 25^{\circ}\text{C Ts} = 60^{\circ}\text{C Pr} = 0.7248$ Analysis The total heat transfer surface area for this finned surface is As, finned = (0.1 m)(0.005 m) = 0.0048 m L = 10 cm 2 As, total = As, finned + As, unfinned = (0.07 m 2 + 0.0048 m 2 = 0.0118 m2 The convection heat transfer coefficient can be determined from Newton's law of cooling relation for a finned surface. Assuming constant thermal conductivity and transfer, the mathematical formulation (the differential equation and the boundary and initial conditions) of this heat conduction problem is to be obtained. The transistor case temperature is 55°C. The distance x is measured from the midplane. 8-14C The heat flux will be higher near the inlet because the heat transfer coefficient is highest at the tube developed value. 4 The thermal resistances of the water tank and the outer thin sheet metal shell are negligible. - $(10^{\circ} \text{ F}) 0.475 - 0.0203(40 \text{ mph}) + 0.304 40 \text{ mph} = -37.7^{\circ} \text{ F}$ In the last 3 cases, the person needs to be concerned about the possibility of freezing. 025 m) $\pi \left(\pi D \right) \int \left[\text{Then the steady rate of heat transfer from the pipe becomes Q& = -37.7^{\circ} \text{ F}} \right] dt = -37.7^{\circ} \text{ F}$ $(0.006 - x)m = + 85^{\circ}C 20 W/m \cdot ^{\circ}C = 3750(0.006 - x) + 85 T (x) = -(c)$ The temperature is higher than the exposed surface temperature, as expected. Then the average rate of heat transfer into the drink is Ao = π Do L + 2 0.0695 N 2 2 The mass whose weight is 0.06915 kg.m/s 2 = 0.00708 kg = 7.08 g g 9.81 m/s 2 Therefore, the mass of the counterweight must be 7 g to counteract the drag force acting on the plate. 2-24 For a medium in which the heat conduction equation is given in its simplest by 1 d (dT) | rk | + g& = 0 r dr (dr) : (a) Heat transfer is steady, (b) it is one-dimensional, (c) there is heat generations are determined by solving the 10 equations above simultaneously with an equation solver to be 5-52 Chapter 5 Numerical Methods in Heat Conduction T1 = 118.8°C, T2 = 116.7°C, T3 = 103.4°C, T4 = 53.7°C, T5 = 254.4°C, T6 = 253.0°C, T7 = 235.2°C, T8 = 103.5°C, T9 = 263.7°C, T10 = 117.6°C (c) The rate of heat loss through a 1-m long section of the chimney is determined from Q& = 4 Σ Q& one -fourth of chimney = 4 Σ Q& element, inner surface $=4 \sum h A i surface, m (Ti - Tm) m = 4[hi (l/2)(Ti - T5) + hi l (Ti - T6) + hi l (Ti - T6) + hi l (Ti - T7) + hi (l/2)(Ti - T9)] = 4(75 W/m 2 \cdot °C)(0.1 m \times 1 m)[(280 - 253.0) + (280 - 253.0) + (280 - 253.2) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 - 253.0) + (280 -
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n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.20 m) - n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.20 m) - n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.20 m) - n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.20 m) - n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.20 m) - n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.20 m) - n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.20 m) - n fin (0.002 m)(0.002 m) = 0.00032 n fin m 2 Aunfinned = (0.15 m)(0.002 m) = 0.00032 m fin m 2 Aunfinned = (0.15 m)(0.002 m)(0.002 m) = 0.00032 m fin m 2 Aunfinned = (0.15 m)(0.002 m) = 0.00032 m fin m 2 Aunfinned = (0.15 m)(0.002 m)(0.002 m) = 0.00032 m fin m 2 Aunfinned = (0.15 m)(0.002 m) = 0.00032 m fin m 2 Aunfinned = (0.15 m)(0.002 m)(0.002 m) = 0.00032 m fin m 2 Aunfinned = (0.15 m)(0.002 m)(0hAunfinned (Tb – T ∞) = (20 W/m 2.°C)(0.03 – 0.000004n fin m 2)(85 – 25)°C = 36 – 0.0048n fin W Then the total heat transfer from the finned unfinned in fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W Then the total heat transfer from the finned unfinned in fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (0.15 m)(0.20 m) = 0.035328n + 36 – 0.0048n fin W The rate of heat transfer if m 2 Q& no fin = hAno fin (Tb - T ∞) = (20 W/m 2 .°C)(0.03 m 2)(85 - 25)°C = 36 W The number of fins can be determined from the overall fin effectiveness equation Q& 0.35328n fin + 36 - 0.0048n fin ϵ fin = fin \rightarrow 3 = \rightarrow n fin = 207 36 Q& no fin 3-135 2 mm × 2 mm T ∞ = 25°C Chapter 3 Steady Heat Conduction 3-175 "!PROBLEM 3-175" "GIVEN" A surface=0.15*0.20 "[m^2]" T b=85 "[C]" k=237 "[W/m-C]" side=0.002 "[m]" L=0.04 "[m]" T infinity=25 "[C]" h=20 "[W/m^2-C]" "epsilon_fin=3 parameter to be varied" "ANALYSIS" A c=side^2 p=4*side a=sqrt((h*p)/(k*A c)) eta fin=tanh(a*L)/(a*L) A fin=n fin*4*side*L A unfinned=A surface-n fin*side^2 Q dot_finned=eta fin*h*A fin*A fin*A fin*A fin*A fin*A fin*A fin*A fin=n fin*4*side*L A unfinned=A surface-n fin*side^2 Q dot_finned=eta fin*h*A fin*A fin*A fin*A fin*A fin*A fin*A fin*A fin*A fin*A fin=n fin*4*side*L A unfinned=A surface-n fin*side^2 Q dot_finned=eta fin*h*A fin*A (T b-T infinity) Q dot unfinned=h*A unfinned*(T b-T infinity) Q dot total fin=Q dot finned+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot total fin/Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot finned+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot finned+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot finned+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q dot fined+Q dot nofin=h*A surface*(T b-T infinity) epsilon fin=Q do conductivity, α is the thermal diffusivity, and t is the time. m 2 2 = (15 W / m. 3-146 The U-value of a wall under winter design conditions is given. 3-97C Effectiveness of a single fin is the ratio of the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the fin base area. h, T∞ Properties The conductivity and t is the ratio of the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfer rate from the entire exposed surface of the heat transfe diffusivity are 1 2 given to be k = 15 W/m·°C and 3 • • $\alpha = 3.2 \times 10 - 6$ m 2 / s. Considering a unit depth and using the Insulated energy balance approach for the boundary nodes (again assuming all heat transfer to be into the volume element for convenience), the finite difference formulation is obtained to be 1 2 Node 1: h0 (T0 - T1) + k 1 T2 - T1] = 0 2 1 2 1 1 2 Node 5: hi (Ti - T5) + k 1 T6 - T5 | T1 - T5 + k = 0 2 1 2 1 1 Node 6: hi (Ti - T6) + k T - T6 | T7 - T6 + k + k | 2 = 0 2 1 2 1 1 1 Node 7: hi (Ti - T7) + k T - T7 | T9 - TNode 9: hi (Ti - T9) + kl2lT7 - T9lT10 - T9 + k = 0.2l2l Node 10: h0 (T0 - T10) + klT8 - T10lT9 - T10l4 + k + $\varepsilon\sigma$ [T surr - (T10 + 273) 4] = 0.2l2l 2 where l = 0.1 m, k = 1.4 W/m^oC, hi = 75 W/m²·°C, Ti = 280°C, ho = 18 W/m²·°C, ho = 18 W/ exchange on the back surface of the absorber plate is negligible. $|e - (2.1589)(1156 = A1e - \lambda 1 \tau = (15618 = 0.007 \text{ Ti} - T_{\infty}] = 0$, wall $\times \theta$ o, cyl = 1 × 0.007 = 0.007 $|| L \text{ Ti} - T_{\infty}]$ short cylinder T (0,0, t) - 202 = 0.007 $|| L \text{ Ti} - T_{\infty}]$ short cylinder T (0,0, t) - 202 = 0.007 $|| L \text{
Ti} - T_{\infty}]$ / 60 h (2.5 / 12 ft) 2 = 0.045 < 0.2 (Be cautious!) 2 2 T0 - T ∞ = A1e $-\lambda 1 \tau$ = (1.2728)e -(1.5421) (0.045) \cong 1 Ti - T ∞ = (0.0077 ft 2 / h)(15 / 60 h) (0.4 / 12 ft) 2 = 1734 > 0.2. This problem involves 4 unknown nodal temperatures, and thus we need to have 4 equations to determine them uniquely. The boiling heat transfer coefficient and the outer surface temperature of the bottom of the pan are to be determined. $C(1 m) \ln(r3/r2) \ln(r3/0.033) = 4.55 \ln(r3/0.033) + 2.2 dT \mu(V) = - | | y + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 y + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 y + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 y + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T (y) = - \mu 2k 2 (y) | V | + C3 dy k(L/T$ C 4 L Applying the two boundary conditions give dT $-k = 0 \rightarrow C 3 = 0$ B.C. 1: y=0 dy y=0 6-21 Chapter 6 Fundamentals of Convection B.C. 2: $y=L \rightarrow C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C 4 = T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C + T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C + T0 + T(L) = T0 \mu V 2 2k (y2) ||1 - |L2| / L + C + T0 + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| / L + T0 \mu V 2 2k (y2) ||1 - |L2| /$ differentiating T(y) with respect to y, dT - μ V 2 y = dy kL2 . The heat flux at the upper surface is q& L = -k dT dy = k y=L μ V 2 kL2 L= $W Q\& = As q\& L = (\pi DW) 0.0006 m L (b)$ This is equivalent to the rate of heat transfer through the cylindrical sleeve by conduction, which is expressed as $2\pi W (T0 - Ts) Q\& = k \ln(D0 / D) \rightarrow (70 W/m \cdot °C) 2\pi (0.15 m)(T0 - 40°C) = 163.5 W \ln(8 / 5)$ which gives the surface temperature of the shaft to be To = 41.2 °C (c) The mechanical power wasted by the viscous dissipation in oil is equivalent to the rate of heat generation, W& lost = Q& = 163.5 W 6-22 Chapter 6 Fundamentals of Convection 6-46 The oil in a journal bearing is considered. Heat transfer in a very long hot dog could be considered to be one-dimensional in preliminary calculations. 2 The thermal properties of the milk Water are taken to be the same as those of water. 1-99C The fan increases the air motion around the body by convection heat transfer coefficient, which increases 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 and outer surfaces of the tube is to be determined. Series resistances indicate sequential heat transfer (such as two homogeneous layers of a wall). We use an average value of 70°C for the temperature of hot water exiting the tank. The Biot numbers and corresponding constants are first determined to be Bi = hL (80 W/m 2.°C)(0.2 m) = = 0.0678 k (236 m) = 0.0 W/m.°C) Bi = hr0 (80 W/m 2.°C)(0.075 m) = 0.0254 k (236 W/m.°C) $\rightarrow \lambda 1 = 0.2268$ and A1 = $1.0110 \rightarrow \lambda 1 = 0.2217$ and A1 = 1.0063 Noting that $\tau = \alpha t / L2$ and assuming $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for this problem can be written as 2.2 $\theta(0, t)$ block = $\theta(0, t)$ wall $\theta(0, t)$ cyl = $\left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \left(\left| A1e - \lambda 1 \tau \right| \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right| \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau \right) \right) \left(\left| A1e - \lambda 1 \tau
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Analysis (a) The heat flux on the surface of the circuit board is As = (0.12 m)(0.18 m) = 0.0216 m 2 T \propto (100 × 0.07) W Q& = 324 W/m 2 q& = As 0.0216 m 2 Chips Ts (b) The surface temperature of the chips is $Q\& = hA(T - T) s \rightarrow Ts = T\infty + s Q\& \infty$ (100 × 0.07) W $Q\& = 40^{\circ}C + = 72.4^{\circ}C$ hAs (10 W/m 2 .°C)(0.0216 m 2) 3-8 Chapter 3 Steady Heat Conduction 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by natural convection 3-26 A person is dissipating heat at a rate of 150 W by nat and radiation to the surrounding air and surfaces. Assumptions Heat transfer from the surface is uniform. 7-99 Chapter 7 External Forced Convection 7-105E "IPROBLEM 7-105E" "ANALYSIS" T equiv=91.4-(91.4-T ambient)*(0.475 - 0.0203*Vel+0.304*sqrt(Vel)) Vel [mph] 4 14.67 25.33 36 46.67 57.33 68 78.67 89.33 100 4 14.67 25.33 36 46.67 41.02 40.75 41.11 41.96 43.21 44.77 7-100 Chapter 7 External Forced Convection 60 50 60 F 40 T equiv [F] 30 20 40 F 10 0 -10 20 F -20 -30 0 22 44 66 Vel [m ph] 7-106 Also, we can replace the symmetry lines by insulation and utilize the mirror-image concept when writing the finite difference equations for the interior nodes. Assumptions 1 Heat transfer through the body is given to be steady and two-dimensional. Analysis (a) The characteristic length and the Biot number for the brass balls, $250^{\circ}F \ 3 \ \pi D \ / \ 6 \ D \ 2 \ / \ 12 \ ft \ V = = = = 0.02778 \ ft \ Lc = Water bath, 120^{\circ}F \ 6 \ 6 \ As \ \pi D \ 2 \ Bi = hLc \ (42 \ Btu/h.ft \ 2 \ .^{\circ}F)(0.02778 \ ft \) = = 0.01820 < 0.1 \ (64.1 \ Btu/h.ft \ .^{\circ}F) \ k \ The lumped system$ analysis is applicable since Bi < 0.1. Then the temperature of the balls after quenching becomes b = hAs h 42 Btu/h.ft 2.°F = 30.9 h -1 = 0.00858 s -1 = = 3 pC pV pC p Lc (532 lbm/ft)(0.02778 ft) -1 T (t) - T ∞ T (t) - T ∞ from a ball during a 2-minute period is $m = \rho V = \rho \pi D 3 = (532 \text{ lbm/ft } 3) \pi (2/12 \text{ ft}) 3 = 1.290 \text{ lbm } 6.6 \text{ Q} = mC p [Ti - T (t)] = (1.29 \text{ lbm})(0.092 \text{ Btu/lbm}.°F)(250 - 166)°F = 9.97 \text{ Btu}$ Then the vater becomes $Q_{k} = n \& Q = (120 \text{ balls/min}) \times (9.97 \text{ Btu}) = 1196 \text{ Btu/min total ball ball Therefore}$, heat must be removed from the water at a rate of 1196 Btu/min in order to keep its temperature constant at 120 ° F. ° F Ao (Ts - T \propto) 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 r2 - r1 [(15.1-100 The total rate of heat transfer from a person by both convection and radiation to the surrounding air and surfaces at specified temperatures is to be determined. 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 = hnD knD 2 / 4 4h = kD = 4(35 W / m2). Then the length of time for the egg to be kept in boiling water is determined to be hp = kAc a = hnD knD 2 / 4 4h = kD = 4(35 W / m2). to be $t = \tau ro 2$ (0.1633)(0.0275 m) $2 = 846 s = 14.1 min \alpha$ (0.146 × 10 - 6 m 2/s) 4-35 Chapter 4 Transient Heat Conduction 4-46 An egg is cooked in boiling water. Assumptions 1 Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. 2 Steady operating conditions exist since there is no change with time at any point and thus $\Delta mCV = 0$ and $\Delta E CV = 0$. The dimensionless temperature 5 cm from the surface of a semi-infinite medium is first determined from ()] (hx h 2 α t) [(x) T(x, t) - Ti ||erfc| x + h α t || - exp| + = erfc|| || k| 2 α t T ∞ - Ti k |/|] k 2 |/|[α t / () (140)(0.05) (140) 2 (9.71 × 10 - 5)(8 × 60) 0.05 $||| = erfc| exp - + || 237 || 2 (9.71 \times 10 - 5)(8 \times 60) || (237) 2 || || = erfc(0.1158) - exp(0.0458)erfc(0.2433) = 0.8699 - (1.0468)(0.7308) = 0.1049 \theta$ semi -inf = T (x, t) - T ∞ = 1 - 0.1049 = 0.8951 Ti - T ∞ The Biot number is calculated for the long cylinder to be Bi = hro (140 W/m 2.°C)(0.075 m) = 0.0443 k (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m } 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and A1 corresponding to this Biot number is $\tau = \alpha \text{ tro } 2 = (9.71 \times 10 - 5 \text{ m} 2 / \text{s})(8 \times 60 \text{ s})(0.075 \text{ m}) = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1$ and $\lambda 1 = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1 = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1 = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1 = 0.0443 \text{ k}$ (237 W/m.°C) The constants $\lambda 1 = 0$ r D0 = 15 Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Radiation is energy emitted by matter in the form of electronic configurations of the atoms or molecules. The complete finite difference formulation of this problem is to be obtained. Properties The properties of air at 30°C are (Table A-15) k = 0.02588 W/m.°C v = 1.608 × 10 m /s -5 2 Pr = 0.7282 Analysis The rate of convection heat
transfer from the top surface of the car to the air must be equal to the solar radiation absorbed by the same surface in order to reach steady operation conditions. This problem involves 3 unknown nodal temperatures, and thus we need to have 3 equations to determine them uniquely. The thickness of insulation that needs to be installed is to be determined. The final pressure of hydrogen can be determined from the ideal gas relation, P1V P2V T 300 K = \rightarrow P2 = 2 P1 = (250 kPa) = 178.6 kPa T1 T2 T1 420 K (b) The energy balance for this system can be expressed as $E - E = \Delta E$ system 1in424out 3 1 424 3 Net energy transfer by heat, work, and mass Change in internal, kinetic, potential, etc. 1-57C The rate of heat transfer through both walls can be expressed as T - T - T Q brick = k brick A 1 2 = (0.72 W/m.°C) A 1 2 = 2.88 A(T1 - T2) L brick 0.25 m where thermal conductivities are obtained from table A-5. Properties The specific heat of the device is given to be Cp = 850 J/kg.°C. °C A \Delta T (0.001257 m 2)(10° C) L 1-68 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. The energy balance for this steady-flow system can be expressed in the rate form as $\Delta E \&$ system @0 (steady) 144 42444 3 = 0 $\rightarrow E \&$ in = E & out Rate of change in internal, kinetic, potential, etc. The rate of hat transfer through the wall and temperature drops across the plaster, brick, covering, and surface-ambient air are to be determined. Analysis The rate of heat transfer excluding the edges and corners is first determined to be 3°C Atotal = $(12 - 0.4)(12 - 0.4) + 4(12 - 0.4)(6 - 0.2) = 403.7 \text{ m2 kA}(0.75 \text{ W/m. Properties The thermal$ properties of the beef slabs are given to be $\rho = 1090 \text{ kg/m3}$, C p = 3.54 kJ/kg.°C, k = 0.47 W/m.°C, and $\alpha = 0.13 \times 10-6 \text{ m2/s}$. It is expressed as q& t = ρ C p v T' = $-k t 6-4 \partial$ T where T' is the eddy ∂ y Chapter 6 Fundamentals of Convection temperature relative to the mean value, and q& t = ρ C p v T' the rate of thermal energy transport by turbulent eddies. The analytical approach (analysis or calculations) has the advantage that it is fast and inexpensive, but the results obtained are subject to the accuracy of the assumptions and idealizations made in the analysis. Properties The R-values of different materials are given in Tables 3-6 and 3-7. - (10° F) 0.475 - 0.0203(20 mph) + 0.304 20 mph = -24.9° F V = 30 mph: Tequiv = 914. (b) The 10 nodal temperatures under steady conditions are determined by solving the 10 equation solver to be 5-50 Chapter 5 Numerical Methods in Heat Conduction T1 = 94.5°C, T2 = 93.0°C, T3 = 82.1°C, T4 = 36.1°C, T5 = 250.6°C, T6 = 249.2°C, T7 = 229.7°C, T8 = 82.3°C $T9 = 261.5^{\circ}C$, $T10 = 94.6^{\circ}C$ (c) The rate of heat loss through a 1-m long section of the chimney is determined from $Q\& = 4 \sum Q\&$ one -fourth of chimney = 4 $\sum Q\&$ one -fourth of chimney is determined from $Q\& = 4 \sum Q\&$ one -fourth of chimney = 4 $\sum Q\&$ one -fourth of 249.2) + (280 - 229.7) + (280 - 261.5)/2]°C = 3153 W Discussion The rate of heat transfer can also be determined by calculating the heat loss from the outer surface if both the fluid and the surface are at the same temperature since there will be no heat transfer in that case. The wall temperatures at distances 15, 30, and 40 cm from the outer surface at the end of 2-hour cooling period are to be determined. 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. °C)(300 m2)(Ts,out - 10)°C + (0.9)(300 m 2)(5.67 × 10 - 8)(300 m 2)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8)(5.67 × 10 - 8 W/m 2.K 4) (Ts,out + 273 K) 4 - (100 K) 4 Solving the equations above simultaneously gives Q& = 37,440 W, T = 7.3°C, and T = -2.1°C s, in s, out The total amount of natural gas consumption during a 14-hour period is Q Q& Δt (37.440 kJ/s)(14 × 3600 s) (1 therm) Q gas = total = = || || = 22.36 therms 0.80 0.80 (105,500 kJ / Finally, the money lost through the roof during that period is Money lost = (22.36 therms)(\$0.60 / therm) = \$13.4 3-11 Chapter 3 Steady Heat Conduction 3-30 An exposed hot surface of an industrial natural gas furnace is to be insulated to reduce the heat loss through that section of the wall by 90 percent. 9 • This system of 10 equations with 10 unknowns constitute the finite difference formulation of the problem. For convenience, let us choose the time step to be At = 15 s. Considering a unit pipe length of L = 1 ft, the individual thermal resistances are determined to be Ri = Rconv, 1 = 1 1 1 = = 0.0364 h. The emissivity of both surfaces of the concrete roof is 0.9. Analysis The nodal spacing is given to be $\Delta x = 0.03$ m. Analysis The nodal spacing is given to be $\Delta x = 0.125$ in. Assumptions 1 No heat is transferred from the cars. The time it will take for the center of the spacing is given to be $\Delta x = 0.125$ in. Assumptions 1 No heat is transferred from the cars. The time it will take for the center of the space cut surface to cool from 25 to 3°C is to be determined. 3 The heat transfer coefficient is very high so that the temperatures on both sides of the block can be taken to be 0°C. 3-95C Heat transfer rate will decrease since a fin effectiveness smaller than 1 indicates that the fin acts as insulation. The length of time it will take for the ice in the chest to melt completely is to be determined. Analysis The heat flux at the outer surface of the pipe is Q& Q& s 300 W = = 734.6 W / m 2 q& s = s = As $2\pi r^2 L 2\pi (0.065 \text{ cm})(1 \text{ m})$ Noting that there is thermal symmetry about the center line and there is uniform heat flux at the outer surface, the differential equation and the boundary conditions for this heat conduction problem can be expressed as d (dT) | r |=0 dr (dr) dT (r1) = h[T (ri) - T\infty] dr dT (r2) = q& s = 734.6 W/m 2 k dr Q = 300 W k h T\infty r1 r2 2.49 A spherical metal ball that is heated in an oven to a temperature of Ti throughout is dropped into a large body of water at $T \propto$ where it is cooled by convection. Properties The thermal conductivities are given to be k = 223 Btu/h·ft·°F for limestone. 4 The tank surface is at the same temperature as the iced water. Properties The thermal conductivities are given to be k = 0.40 Btu/h·ft·°F for bricks, k = 0.015 Btu/h·ft·°F for air, and k = 0.10 Btu/h·ft·°F for sheetrock. ° C)(01 . Analysis The thickness of flat R-20 insulation (in h·ft2·°F/Btu) is determined from the definition of R-value to be L R value = \rightarrow L = R value k = (20 h.ft 2.°F/Btu)(0.02 Btu/h·ft.°F) = 0.4 ft k R-20 L 7-720 insulation (in h·ft2·°F/Btu) is determined from the definition of R-value to be L R value = \rightarrow L = R value k = (20 h.ft 2.°F/Btu)(0.02 Btu/h·ft.°F) = 0.4 ft k R-20 L 7-720 insulation (in h·ft2·°F/Btu) is determined from the definition of R-value to be L R value = \rightarrow L = R value k = (20 h.ft 2.°F/Btu)(0.02 Btu/h·ft.°F) = 0.4 ft k R-20 L 7-720 insulation (in h·ft2·°F/Btu) is determined from the definition of R-value to be L R value = \rightarrow L = R value k = (20 h.ft 2.°F/Btu)(0.02 Btu/h·ft.°F) = 0.4 ft k R-20 L 7-720 insulation (in h·ft2·°F/Btu) is determined from the definition of R-value to be L R value = \rightarrow L = R value k = (20 h.ft 2.°F/Btu)(0.02 Btu/h·ft.°F) = 0.4 ft k R-20 L 7-720 insulation (in h·ft2·°F/Btu) is determined from the definition of R-value to be L R value = \rightarrow L = R value k = (20 h.ft 2.°F/Btu)(0.02 Btu/h·ft.°F) = 0.4 ft k R-20 L 7-720 insulation (in h·ft2·°F/Btu) is determined from the definition of R-value to be L R value to be R value to be L R value to be R value to Chapter 7 External Forced Convection 7-83 A steam pipe is to be covered with enough insulation to reduce the exposed surface temperature to 30°C. Therefore, the structure is gaining heat. 2-1 Chapter 2 Heat Conduction Equation 2-6C Heat transfer to a hot dog 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 3-38E A thin copper plate is sandwiched between two layers of epoxy boards. That is, Q& = Q& pipe and insulation surface to surroundings Using the thermal resistance network, heat transfer from the steam to the outer surface is expressed as 11 = 0.0995 °C/W Rconv, i = hi Ai (80 W/m 2.°C)[π (0.04 m)(1 m)] ln(r2 / r1) ln(2.3 / 2) = 0.0015 °C/W 2\pi kL 2\pi (15 W/m.°C)(1 m) ln(r3 / r2) ln(5.8 / 2.3) = = 3.874 °C/W 2\pi kL 2\pi (0.038 W/m.°C)(1 m) R pipe = Rinsulation and Q& pipe and ins = Rconv, i T ∞ 1 - Ts (250 - Ts)°C = + R pipe + Rinsulation (0.0995 + 0.0015 + 3.874) °C/W Heat transfer from the outer surface to surr, conv + rad o o s surr o s 4 surr) = (22.50 W/m 2.°C)(0.3644 m 2)(Ts - 3)°C [+ (0.3)(0.3644 m 2)(5.67 × 100 m 2) (0.3644 m 2 -8 W/m 2 .K 4) (Ts + 273 K) 4 -(3 + 273 K) 4 Solving the two equations above simultaneously, the surface temperature and the heat transfer rate per m length is 7-93] Chapter 7 External Forced Convection 7-101 A spherical tank filled with liquid nitrogen is exposed to winds. 2-13 in the text). (c) The chilled water circulated during immersion cooling encourages microbial
growth, and thus immersion cooling encourages microbial growth. The rate of heat transfer through the wall is to be determined, and it is to be determined, and it is to be assessed if the steel bars between the plates can be ignored in heat transfer analysis since they occupy only 1 percent of the heat transfer surface area. Analysis The nodal spacing is given to be $\Delta x = \Delta x = l = 0.01m$, and the general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for the case of no heat general finite difference form of an interior node for steady two-dimensional heat conduction for theat general finite difference form of Tright + Tbottom - 4Tnode = 0 k (a) There is symmetry about a vertical line passing through the nodes 1 and 3. 7 The pressure of air is 1 atm. 4-44 Chapter 4 Transient Heat Conduction 4-53 A hot baked potato is taken out of the oven and wrapped so that no heat is lost from it. Analysis Consider a body of arbitrary shape of mass m, volume V, surface area A, density ρ , and specific heat C p initially at a uniform temperature Ti . 5-109 Chapter 5 Numerical Methods in Heat Conduction 5-113 Starting with an energy balance on a disk volume element, the one-dimensional transient explicit finite difference equation for a general interior node for T (z , t) in a cylinder whose side surface is subjected to convection with a convection coefficient of h and an ambient temperature of To for the case of constant thermal conductivity with uniform heat generation is to be obtained. 7-14 Hot engine oil flows over a flat plate. 2 Heat conductivity with uniform heat generative varies in both axial x- and radial rdirections. ft 2 1 1 = = $0.2711 \text{ h}^{\circ}F/Btu 2 \text{ hi} \text{ Ai} (35 \text{ Btu/h.ft.}^{\circ}F)(0.105 \text{ ft 2}) \ln(r2/r1) \ln(3/2) = = 0.00029 \text{ h}^{\circ}F/Btu 2\pi(2.23 \text{ Btu/h.ft.}^{\circ}F)(1 \text{ ft}) \ln(r1/rdep) \ln(0.2/0.157 \text{ ft 2}) = Ri + R \text{ pipe} + R \text{ deposit} + Rote + Rote$ = 0.27211 + 0.00029 + 0.01633 + 0.00425 = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.27211 + 0.00029 + 0.01633 + 0.00425 = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.27211 + 0.00029 + 0.01633 + 0.00425 = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = 0.29298 °F/Btu Ro = Rtotal The heat transfer required to be Q& total = 0.29298 °F/Btu Ro = 0 m&h fg = (120 lbm/h)(1037 Btu/lbm) = 124,440 Btu/h Tube length = Q& total 124,440 = = 1215 ft 102.40 Q& 3-61 Chapter 3 Steady Heat Conduction 3-80E "GIVEN" T infinity 1=100 "[F]" T infinity 2=70 "[F]" K pipe=223 "[Btu/h-ft-F], parameter to be varied" D i=0.4 "[in]" "D o=0.6 [in], parameter to be varied" T 1=D i/2 r 2=D o/2 h fg=1037 " $[Btu/h-ft^2-F]$ " h i=35 " $[Btu/h-ft^2-F]$ " h i=35 " $[Btu/h-ft^2-F]$ " m dot=120 "[lbm/h]" "ANALYSIS" L=1 "[ft], for 1 ft length of the tube" A i=pi*(D o*Convert(in, ft))*L A o=pi*(D 1180 1175 L tube [ft] 1170 1165 1160 1155 1150 1145 0 50 100 150 200 250 300 350 400 k pipe [Btu/h-ft-F] 1155.0 L tube [ft] 1152.5 1150.0 1147.5 1145.0 0.5 0.6 0.7 0.8 D o [in] 3-65 0.9 1 Chapter 3 Steady Heat Conduction 3-81 A 3-m diameter spherical tank filled with liquid nitrogen at 1 atm and -196°C is exposed to convection and radiation with the surrounding air and surfaces. Otherwise, it is nonlinear. 4 The water in the pipe is stationary, and its initial temperature is 15°C. The contact conductance at the interface of copper-epoxy layers is given to be hc = 6000 W/m2·°C. Assumptions 1 Heat transfer is given to be hc = 6000 W/m2·°C. surface is negligible. Analysis The characteristic length of the steel ball bearings and Biot number are Lc = $\pi D 3 / 6 D 0.012 \text{ mV} = = = 0.002 \text{ m} 6 6 \text{ As } \pi D 2 \text{ Furnace hL} (125 \text{ W/m} 2.°C)(0.002 \text{ m}) = 0.0166 < 0.1 \text{ Bi} = c = (15.1 \text{ W/m}.°C) \text{ k Steel balls 900°C Air, 30°C Therefore, the lumped system analysis is applicable. 5 Heat loss from the fin tip is <math>\Delta x$ considered. Walls without windows : 1 1 Ri = = $0.003571 \circ C/W \ ha (7 W/m .^{C})(10 \times 4 m 2)$ Wall L L R - value 2.31 m 2 °C/W R wall = wall = = $0.001667 \circ C/W \ ha (15 W/m 2 .^{C})(10 \times 4 m 2)$ 1 1 Ro = = $0.003571 + 0.05775 + 0.001667 = 0.062988 \circ C/W \ T - T (22 - 8) \circ C/W \ ha (15 W/m 2 .^{C})(10 \times 4 m 2)$ $= 222.3 \text{ W} \text{ Q}_{\&} = \infty 1 \infty 2 = \text{Rtotal } 0.062988^{\circ}\text{C/W}$ Then Ri Rwall Ro Wall with single pane windows: 1 1 Ri = = 0.001786^{\circ}\text{C/W} + 1.8) m 2 Lglass 0.005 m = = 0.002968^{\circ}\text{C/W} RA (0.78 W/m 2 . 2-16 Chapter 2 Heat Conduction Equation Analysis Noting that there is thermal symmetry about the midpoint and convection and radiation at the outer surface and expressing all temperatures in Rankine, the differential equation and the boundary conditions for this heat conduction problem can be expressed as 1 $\partial (2 \partial T) \partial T | kr | = \rho C 2 \partial r / \partial T | kr | = \rho C 2 \partial r / \partial T (r 0, t) 4] - k = 0$ h[T (r0) - T ∞] + $\varepsilon\sigma$ [T (r0) 4 - Tsurr ∂ r k r2 T ∞ h Ti T (r, 0) = Ti 2-51 The outer surface of the North wall of a house exchanges heat with both convection only. 5-110 Chapter 5 Numerical Methods in Heat Conduction 5-114E The roof of a house initially at a uniform temperature is subjected to convection and radiation on both sides. Solution (a) The rate of heat transfer through the shell is expressed as T - T Q & sphere = $4\pi k$ ave $T + T \downarrow (= k (Tave) = k0 | 1 + \beta 2 1 | 2 / \sqrt{r1 T1 r2}$ r is the average thermal conductivity. Analysis (a) Disregarding any natural convection currents, T1 T2 the rates of conduction and radiation heat transfer T – T (290 – 150) K Q& cond = kA 1 2 = (0.01979 W/m 2 . C)(1 m 2) = 139 W L 0.02 m Q& rad = $\varepsilon \sigma As$ (T1 4 – T2 4) [Q& total] = 1(5.67 × 10 - 8 W/m 2 . K 4)(1m 2) (290 K) 4 - (150 K) 4 = 372 W = Q& + Q& = 139 + 372 = 511 W cond · Q rad (b) When the air space between the plates is evacuated, there will be radiation heat transfer only. 2-14E The power consumed by the resistance wire of an iron is given. Properties The specific heats of water = 4.18 kJ/kg·°C and Cp, Cu = 0.386 kJ/kg·°C (Tables A-3 and A-9). Analysis (a) Noting that heat transfer is one-dimensional in the radial r direction, the mathematical formulation of this problem can be expressed as $d (2 dT) |r| = 0 dr (dr / r1 T (r1) = T1 = 0^{\circ} C$ and $T1 k r2 T \infty h dT (r2) = h[T (r2) - T\infty] dr - k$ (b) Integrating the differential equation once with respect to r gives r2 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 2 T (r) = - C1 + C2 r where C1 and C2 are arbitrary constants. We hope the text and this Manual serve their purpose in aiding with the instructor of Heat Transfer, and making the Heat Transfer, and making the Heat Transfer experience of both the instructors and students a pleasant and fruitful one. Properties It is given that k wire = 15 W / m.° C and k plastic = 12 Analysis Letting TI denote the unknown interface temperature, the mathematical formulation of the heat
transfer problem in the wire can be expressed as 1 d (dT) g& |r| + = 0 r dr (dr) k T \propto dT (0) = 0 with T (r1) = TI and h dr Multiplying both sides of the differential equation by r rearranging, and integrating give g& g& r 2 dT d (dT) = - + C1 (a) |r| = - r \rightarrow r r1 r2 k 2 dr dr (dr k g r Applying the boundary condition at the center (r = 0) gives dT (0) g& 0× = - × 0 + C1 \rightarrow 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 \rightarrow (b) 4k dr 2k Applying the other boundary condition at r = r1, $g\& 2 g\& 2 TI = -r1 + C2 \rightarrow C2 = TI + r1 B$. °C)(0.33 × 1 m 2) Ri = Rconv, 1 = R6 = R plaster = center L 0.02 m = 0.303 °C / W kA (0.22 W / m. At time t = 0, the body is placed into a medium at temperature T ∞ , and heat transfer takes place between the body and its environment with a heat transfer coefficient h. 5 The thermal conductivity of concrete is given to be k = 0.75 W/m.°C. Properties The properties of the apples are given to be k = 0.418 W/m.°C, $\rho = 840$ kg/m3, Cp = 3.81 kJ/kg.°C, and $\alpha = 1.3 \times 10^{-7}$ m2/s. 5-72C The implicit method is unconditionally stable and thus any value of time step Δt can be used in the solution of transient heat conduction problems since there is no danger of unstability. (b) The 8 nodal temperatures under steady conditions are determined by solving the 8 equations above simultaneously with an equation solver to be T1 =163.6°C, T2 =160.5°C, T3 =156.4°C, T4 =154.0°C, T5 =151.0°C, T6 =144.4°C, T7 =134.5°C, T8 =132.6°C Discussion The accuracy of the solution can be improved by using more nodal points. The amount of heat dissipated in 24 h, the heat flux, and the fraction of heat dissipated from the top and bottom surfaces are to be determined. The time it will take for the surface temperature to rise to a specified value, the amounts of heat transfer for specified values of center and surface temperatures are to be determined. Assumptions 1 Heat transfer through the flange is stated to be steady and one-dimensional. ° C)(5.556 × 10 - 5 m2 . + 0.017 = 0149 . We acknowledge, with appreciation, the contributions of numerous users of the first edition of the book who took the time to report the errors that they discovered. The 3 nodal temperatures under steady conditions are determined by solving the 3 equations above simultaneously with an equation solver to be T1 = T3 = T7 = T9 = 304.85°F, T2 = T4 = T6 = T8 = 316.16°F, T5 = 328.04°F (b) The rate of heat loss from the bar through a 1-ft long section is determined from an energy balance on one-eight section of the bar, and multiplying the result by $8:11[1]Q\& = 8 \times Q\&$ one -eight section, conv = $8 \times |h(T1 - T\infty) + h(T2 - T\infty)|(1 ft) = 8 \times h[T1 + T2 - 2T\infty)|(1 ft) = 2 \lfloor 2 \rfloor = 8(7.9 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{°F})(0.2/2 \text{ ft})(1 ft)[304.85 + 316.16 - 2 + 316.16 - 2 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 + 316.16 +$ \times 70]°F = 3040 Btu/h (per ft flength) Discussion Under steady conditions, the rate of heat loss from the bar is equal to the rate of heat loss from the bar is equal to the rate of heat loss from the bar is equal to the rate of heat generation within the bar per unit length, and is determined to be Q& = E& gen = g& 0V = (0.19 \times 10.5 \text{ Btu/h.ft } 3)(0.4 \text{ ft} \times 0.4 \text{ ft} \times 1.4 \text{ ft}) = 3040 \text{ Btu/h} (per ft length) which confirms the results obtained by the finite difference method. (b) The temperatures are determined by solving equations above to be $10 \cdot T0 = 74.63^{\circ}F$, $T3 = 74.63^{\circ$ energy as a 100-W resistance heater. 3-94C The fin efficiency is defined as the ratio of actual heat transfer rate from the fin it transfer rate from the fin it transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin it transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the ratio of actual heat transfer rate from the fin efficiency is defined as the rate of a transfer rate from the fin efficiency is defined as the rate of a transfer rate from the fin efficiency is defined as the rate of a transfer rate from the fin efficiency is defined as the rate of a transfer rate from the fin efficiency is defined as the rate of a transfer EES Software (Copy the following lines and paste on a blank EES screen to verify solution): $2*x^3-10*x^0.5-3*x = -3$ Answer: x = 2.063 (using an initial guess of x=2) 1-112 Solve the following system of 2 equations with 2 unknowns using EES: $x_3 - y_2 = 7.75$ $3x_3 + y = 3.5$ Solution by EES Software (Copy the following lines and paste on a blank EES) screen to verify solution): $x^3-y^2=7.75$ 3* $x^+y+y=3.5$ Answer x=2 y=0.5 1-62 Chapter 1 Basics of Heat Transfer 1-113 Solve the following system of 3 equations with 3 unknowns using EES: 2x - y + z = 5 3 $x^2 + 2y = z + 2$ xy + z = 8 Solution by EES Software (Copy the following lines and paste on a blank EES screen to verify solution): $2^*x-y+z=5$ $3*x^2+2*y=z+2$ x*y+2*z=8 Answer x=1.141, y=0.8159, z=3.535 1-114 Solve the following system of 3 equations with 3 unknowns using EES: $x^2y-z=1$ $x-3y^0.5+xz=-2$ x+y-z=2 Answer x=1, y=1, z=01-63 Chapter 1 Basics of Heat Transfer Special Topic: Thermal Comfort 1-115C The metabolism refers to the burning of foods such as carbohydrates, fat, and protein in order to perform the necessary bodily functions. m Therefore, the minimum thickness of fiberglass needed to protect the pipe from freezing is t = r3 - r2 = 3.50 - 0.033 = 3.467 m 0.06 m, and V = 7 m/s. The rate of heat loss from the steam per unit length of the pipe is to be determined. energies $-Qout = \Delta U = -m(u^2 - u^2)$ where m = (250 kPa)(1.0 m 3) P1V = = 0.1443 kg RT1 (4.124 kPa · m 3/kg · K)(420 K) H2 250 kPa 420 K Q Using the Cv (=Cp - R) = 14.516 - 4.124 = 10.392 kJ/kg.K value at the average temperature of 360 K and substituting, the heat transfer is determined to be Qout = $(0.1443 \text{ kg})(10.392 \text{ kJ/kg} \cdot \text{K})(420 - 300)\text{K} = 180.0 \text{ kJ} + 29 \text{ K}$ initial temperature difference is determined from Gas h, $T \propto T$ (t) $-T \propto = 0.01$ Ti $-T \propto b = hA h 65$ W / m 2. ° C) T = 170°C The constants $\lambda 1$ and A1 corresponding to this Biot number of nodes M becomes M = $\Delta x L 0.6$ cm +1 = +1= 4 Δx 0.2 cm 0 • 1 • 2 The right surface temperature is given to be T3 = 85°C. Noting that the volume element of a general interior node minvolves heat conduction from two sides and the volume of the element is Velement = A Δz , the transient explicit finite difference formulation for an interior node can be expressed as kAT i +1 - Tmi T i - Tmi $Tmi - 1 - Tmi + kAm + 1 + g \leq 0$ $A\Delta x = oA\Delta x C m \Delta t \Delta x \Delta x$ Canceling the surface area A and multiplying by $\Delta x/k$, it simplifies to Disk $g \leq 0 \Delta x 2$ (Δx) 2 i + 1 (Tm - Tmi) = $\alpha \Delta t$ k where $\alpha = k/(\rho C)$ is the thermal diffusivity of the wall material. 2 Thermal properties of the board are constant. Nodes 1 and 2 are interior nodes. 2
3 and thus for them we can use the general finite difference Radiation relation expressed as Tm - 1 - 2Tm + Tm + 1 = 0 (since g& = 0), for m = 1 and $2 \& \Delta x 2$ The finite difference equation for node 3 on the right surface subjected to convection and solar heat flux is obtained by applying an energy balance on the half volume element about node 3 and taking the direction of all heat transfers to be towards the node under consideration: Node 1 (interior): T1 - 2T2 + T3 = 0 Node 3 (right surface): $4 \alpha s q \& s + \varepsilon \sigma$ [Tspace - (T3 + 460) 4] + k T2 - T3 = 0 \Delta x where k = 1.2 Btu/h.ft.°F, $\varepsilon = 0.80$, $\alpha s = 0.45$, q & s = 300 Btu / h.ft 2, Tspace = 0 R, and $\sigma = 0.1714$ Btu/h.ft2.R 4 The system of 3 equations with 3 unknown temperatures constitute the finite difference formulation of the masonry wall is determined in the table below. The temperature difference between the center and the surface of the wire is to be determined. The outer surface of the tank is black and thus its emissivity is $\varepsilon = 1$. Also, the refrigeration unit will consume more power to reduce the temperature to -10° C, and thus it will have a lower efficiency. In general, an energy balance on this ring element during a small time interval Δt can be expressed as $\Delta z \Delta E$ element (Q& r - Q& r + Δr)) + (Q& z - Q& z + Δz) = Δt But the change in the energy content of the element can be expressed as ΔE element = E t + Δt - Tt (Q& r - Q& r + Δr) + (Q& z - Q& z + Δz) = ρC ($2\pi r \Delta r$) Δz (t + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δt - Tt) Δz (Tt + Δ $\Delta r - Q\&r + \Delta z + \Delta z$ $\int \partial t$ since, from the definition of the derivative and Fourier's law of heat conduction, 1 Q& $r + \Delta r - Q$ $r = | -k(2\pi r \Delta z)| = -|k| \Delta z \rightarrow 0$ $2\pi r \Delta z \Delta r (\partial r) + \Delta r (\partial z) = -|k| \Delta z \rightarrow 0$ $2\pi r \Delta z \Delta r (\partial z) + \Delta z (\partial z) = -|k| \Delta z \rightarrow 0$ of constant thermal conductivity the equation above reduces to $1 \partial (\partial T) \partial 2 T 1 \partial T = |r| + r \partial r \sqrt{\partial r} \partial z 2 \alpha \partial t$ where $\alpha = k / \rho C$ is the thermal diffusivity of the material. Air $V \infty = 3 \text{ m/s } T \infty = 30^{\circ} C$ Properties of air at 1 atm and the film temperature of $(Ts + T\infty)/2 = (90+30)/2 = 60^{\circ} C$ are (Table A-15) $\rho = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3 15 \text{ m/min } k = 1.059 \text{ kg/m } 3$ 0.02808 W/m°C Plastic sheet Ts = 90°C $v = 1.896 \times 10 \text{ m/s} + 52 \text{ Pr} = 0.7202 \text{ Analysis The width of the cooling section is first determined from W} = V\Delta t = [(15 / 60) \text{ m/s}](2 \text{ s}) = 0.5 \text{ m}$ The Reynolds number is V L (3 m/s)(1.2 m) Re L = $\infty = 1.899 \times 10.5 \text{ v} + 1.896 \times 10^{-5} \text{ m}^2$ which is less than the critical Reynolds number. Then the pressure drop across the tube bank becomes $\Delta P = N L f\chi 2 \rho V max (1.145 kg/m 3)(8.667 m/s) 2 = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 (1N | | 1 kg \cdot
m/s 2) = 20(0.33)(1) 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 (1N | | 1 kg \cdot m/s 2) = 20(0.33)(1) 2 (1N | | 1 kg \cdot m/s 2) = 20(0.3$ $= (20 + 49.7)/2 = 34.9^{\circ}C$, which is very close to the assumed value of 35^{\circ}C. Analysis (a) It is given that D = 0.04 m, and V = 5.2 m/s. 6 The phase change effects are not considered, and thus the actual cooling time will be much longer than the value determined. Then the annual fuel cost of this furnace before insulation becomes Annual Cost = Q in × Unit cost = (24,314 therm / yr)(\$0.50 / therm) = \$12,157 / yr We expect the surface temperature of the furnace to increase, and the heat transfer are to be determined. The power rating of the heater is to be determined. 3 The thermal conductivity of the pan is constant. Properties The thermal conductivities are given to be k = 0.06 W/m °C for air, and k = 0.026 W/m °C for cotton fabric. Therefore, we should still use the smallest time step practical to minimize the numerical error. °C)(200 - 80)°C = 515 kJ Therefore, 515 kJ of energy (heat or work such as electrical energy) needs to be transferred to the aluminum ball to heat it to 200°C. P2-47). L 0.5 / 12 ft 2 R fiberglass = R 2 = = 20.83 ft .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 + 20.83 = 1 = 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 ft 2 .°F.h/Btu k 2 (0.020 Btu/h.ft.°F) Rtotal = 2 R1 + R 2 = 2 × 0.417 21.66 ft 2. Properties The density and specific heat of water at room temperature are $\rho = 1000 \text{ kg/m3}$ and $C = 4.18 \text{ kJ/kg} \circ C$ (Table A-9). 2 The thermal properties of the rod are constant. The rate of heat transfer per unit length and the temperature drops across the pipe and the insulation are to be determined. 5 Air is an ideal gas with constant. properties. The average heat transfer coefficient and the cooling time of the potato if it is wrapped completely in a towel are to be determined. Heat transfer through the insulation and through the study will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. Analysis The nodal spacing is given to $be \Delta x = \Delta x = l = 0.01 \text{ m}$, 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 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node l 2 = 0 \rightarrow lines passing through the midpoint, and thus we need to consider only 1/8th of the region. The sizing problems deal with the determination of the size of a system in order to transfer heat at a specified rate for a spe 18.9%) R - value, old 3.963 3-141E The R-value and the U-factor of a masonry cavity wall are to be determined. Therefore, there are 4 unknown nodal temperatures, and we need 4 equations to determine them. 6-17C The fluid viscosity is responsible for the development of the velocity boundary layer. The surface temperature of the wire is to be determined using the applicable relations directly and by solving the applicable differential equation. o C)(3 o C) m& = ρ Ac 0.234 kg/s [$\pi(0.04 \text{ m}) 2$] (1000 kg/m) | 4]|[] = 0.186 m/s 3 Discussion The outer surface temperature of the pipe is T – Ts (90 – Ts) °C Q& = $\infty 1 \rightarrow 2930 \text{ W} = \rightarrow \text{Ts} = 77^{\circ}\text{C Ri} + \text{R pipe} (0.0044 + 0.0000) °C/W which is very$ close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Analysis This spherical shell can be approximated as a plate of thickness 0.4 cm and area A = $\pi D^2 = \pi (0.2 \text{ m})^2 = 0.126 \text{ m}^2 5^{\circ} \text{C}$ Then the rate of heat transfer through the shell by conduction is $(5 - 0)^{\circ} \text{C} \Delta T \text{ Q} \&$ cond = $kA = (80.2 \text{ W/m} \cdot °\text{C})(0.126 \text{ m}^2 5 \text{ c}) = 0.126 \text{ m}^2 5^{\circ} \text{C}$ Then the rate of heat transfer through the shell by conduction is $(5 - 0)^{\circ} \text{C} \Delta T \text{ Q} \&$ cond = $kA = (80.2 \text{ W/m} \cdot °\text{C})(0.126 \text{ m}^2 5 \text{ c})$ 12,632W L 0.004m Considering that it takes 333.7 kJ of energy to melt 1 kg of ice at 0°C, the rate at which ice melts in the container can be determined from m& ice = Iced water 0°C 0.4 12.632 kJ / s Q& = = 0.038 kg / s 333.7 kJ / kg hif Discussion We should point out that this result is slightly in error for approximating a curved wall as a plain wall. ° C)(300 m)(Ts,out - 10)° C Tsky = 255 K + (0.9)(300 m 2)(5.67 × 10 - 8 W / m 2).K 4) (Ts,out + 273 K) 4 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation solver (or by trial and error) gives Q& = 25,450 W and Ts, out = 8.64° C Then the amount of natural gas consumption during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation during a 1-hour period is E gas = Q total Q& Δt (25.450 kJ/s)(14 - (255 K) 4 Solving the equation during a 1-hour period is E gas = Q total Q& Δt)(25.450 kJ/s)(14 - \times 3600 s)
(1 therm) = = || || = 14.3 therms 0.85 0.85 (105,500 kJ) Finally, the money lost through the roof during that period is Money lost = (14.3 therms)(\$0.60 / therm) = \$8.58 1-60 Q& Chapter 1 Basics of Heat Transfer 1-109E A flat plate solar collector is placed horizontally on the roof of a house. Properties The properties of air at the film temperature of $(Ts + T_{\infty})/2 = (60+25)/2 = 42.5$ °C are (Table A-15) k = 0.02681 W/m.°C Air V_{\infty} T_{\infty} = 25°C Ts = 60°C $\upsilon = 1.726 \times 10$ -5 m 2/s Pr = 0.7248 20 W Analysis We first need to determine radiation heat transfer rate. Properties The thermal conductivity, thermal diffusivity, and density of chickens are given to be k = 0.45 W/m.°C, $\alpha = 1.726 \times 10^{-5} \text{ m}^2/s$ Pr = 0.7248 20 W Analysis We first need to determine radiation heat transfer rate. 0.13×10^{-6} m2/s, and $\rho = 950$ kg/m3. Properties The thermal conductivity of copper is given to be k = 386 W/m °C (Table A2). Applying the boundary conditions give Convection at x = L Temperature at x = L: h T $\propto 4 - kC1 = h[T2 - T\infty] + \varepsilon \sigma [(T2 + 460) 4 - Tsurr] / k T (L) = C1 \times L + C2 = T2$ \rightarrow L C2 = T2 - C1 L Substituting C1 and C2 into the general solution, the variation of temperature is determined to be 4 h[T2 - T ∞] + $\varepsilon\sigma$ [(T2 + 273) 4 - Tsurr] (L - x) k (14 W/m 2 · °C)(45 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg)(0.389 kJ/kg.°C)(150 - 25)°C + 0.7(5.67 × 10-8 W/m 2 · K 4) [(318 K) 4 - (290 K) 4] = 45°C + (0. (0.15 m) = 6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (6.43 kg Q max = mC p (Ti - T ∞) = (20)°C = 325 kJ Then we determine the dimensionless heat transfer ratios for both geometries as $\left(\begin{array}{c} Q \\ Q \\ Max \end{array} \right) \sin(\lambda 1) \sin(0.164) = 1 - \theta \text{ o}, \text{ wall} = 1 - 2\theta \text{ o}, \text{cyl } 11 = 1 - 2(0.577) = 0.427 \left| \lambda 0.1704 1 \right| \text{ cyl The heat transfer ratio for the short cylinder is } \left(\begin{array}{c} Q \\ Q \\ Max \end{array} \right) \left(\begin{array}{c} Q \\$ || short = |Q| cylinder |max(Q)|| plane + |Q| wall |max[]| (|Q|| long |1 - |Q| cylinder |max[]| (|Q|| long |1 - |Q|| cylinder |M| (|Q|| long |1 - |Q|| cylinder |M| (|Q|| long |1 - |Q|| cylinder |M| (|Q|| long ||Q|| (|Q|| (|Q|| long ||Q|| (|Q|| (Conduction 4-74 "!PROBLEM 4-74" "GIVEN" D=0.08 "[m]" r_o=D/2 height=0.15 "[m]" L=height/2 T_i=150 "[C]" T_infinity=20 "[C]" h=40 "[W/m-C]" rho=8530 "[kg/m^3]" C_p=0.389 "[kJ/kg-C]" alpha=3.39E-5 "[m^2/s]" "ANALYSIS" "(a)" "This short cylinder can physically be formed by the intersection of a long cylinder of radius r o and a plane wall of thickness 2L" "For plane wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" A 1 w=0.2282 "w stands for wall" A 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" A 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" A 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" A 1 w=0.2282 "w stands for wall" A 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=0.2282 "w stands for wall" Bi w=(h*L)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1 w=(h*L)/k "From Table 4-1 corresponding to this Bi numb (T o wT infinity)/(T i-T infinity)/(T i

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) 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 c Q c=(Q/Q max) w Q c=1-2*theta o c*J 1/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 69 50 50 60 Q [k] 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 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 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 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 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 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 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 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 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 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 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 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 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 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 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 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 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 120 100 T L,o [C] 80 60 40 100 T L,o [C] 80 60 40 40 100 T L,o [C] 80 60 40 40 100 T L,o [C] 80 60 40 100 T L,o [C] 80 determined from (x T (x, t) - Ti = erfc) Transient Heat Conduction 4-67 A thick wood slab is $\tau (x, t) - Ti (70.1 - 70 = 0.00006 \rightarrow 0.00006 = erfc(2.85) 2 \alpha = (1.5 \text{ ft}) 2 4 \times (2.85) 2 (0.023 \text{ ft } 2 / h) 4-61 = 3.01 \text{ h} = 181 \text{ min } 70^{\circ}\text{F}$ Chapter 4 Transient Heat Conduction 4-67 A thick wood slab 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 = $\pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2$ 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 heat transfer to the nitrogen tank is As = $\pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2$ 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 heat transfer to the nitrogen tank is As = $\pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2$ 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 heat transfer to the nitrogen tank is As = $\pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2$ 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 heat transfer to the nitrogen tank is As = \pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2 (20 - (-196))[C = 271,430 W Air 20° 1 atm Liquid N2 Then the rate of heat transfer to the nitrogen tank is As = \pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2 (20 - (-196))[C = 271,430 W Air 20° 1 atm Liquid N2 Then the rate of heat transfer to the nitrogen tank is As = \pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2 (20 - (-196))[C = 271,430 W Air 20° 1 atm Liquid N2 Then the rate of heat transfer to the nitrogen tank is As = \pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2 (20 - (-196))[C = 271,430 W Air 20° 1 atm Liquid N2 Then the rate of heat transfer to the nitrogen tank is As = \pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2 (20 - (-196))[C = 271,430 W Air 20° 1 atm Liquid N2 Then the rate of heat transfer to the nitrogen tank is As = \pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2 (20 - (-196))[C = 271,430 W Air 20° 1 atm Liquid N2 Then the rate of heat transfer to the nitrogen tank is As = \pi D 2 (20 - (-196))[C = 271,430 W Air 20° 1 atm Liquid N2 W Air evaporation of liquid nitrogen in the tank is determined Q& to be Q& 271.430 kJ/s \rightarrow 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 $0.015 = V = (4 \text{ m/s}) = 8.571 \text{ m/s} (0.008 \text{ sT} - D \text{ Re } D = V = 4 \text{ m/s} \text{ Ti} = 0^{\circ}\text{C} \text{ sL}$ 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) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0.7375) 0.36 (0convenient 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 $k = 236 \text{ W/m.}^\circ\text{C}$, $\rho = 2702 \text{ kg/m3}$, $\text{Cp} = 0.896 \text{ kJ/kg.}^\circ\text{C}$, and $\alpha = 9.75 \times 10-5 \text{ m2/s}$. 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 = $(1.2 \text{ m}) \times (2 \text{ m}) = 2.4 \text{ m} 2 1 1 = 0.04167 \text{ °C/W} \text{ h} 1 \text{ A} (10 \text{ W/m} 2 . ^{\circ}\text{C})(2.4 \text{ m} 2) 0.006 \text{ m} \text{ L} = 0.00321 \text{ °C/W} \text{ Rglass} = 0.00321 \text{ °C/W} \text{ Rglass}$ 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 .°C)(2.4 m 2) Rtotal = R conv, 1 + R glass + Rconv, 2 L Ri = 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& = $\infty 1 \infty 2$ = 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 \rightarrow T1 = T ∞ 1 – QR conv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C –
(471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C – (471 W)(0.04167 ° C / W) = 4.4° C Rconv, 1 = 24° C / W space. 3 The thermal properties of the chickens are constant. The rate of heat transfer from the top surface is $\Delta t = Q\&$ 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 from the top surface is $\Delta t = Q\&$ 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 from the top surface is $\Delta t = Q\&$ 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 from the top surface is $\Delta t = Q\&$ 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 from the top surface is $\Delta t = Q\&$ 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 from the top surface is $\Delta t = Q\&$ 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 from the top surface is $\Delta t = Q\&$ 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 from the top surface is $\Delta t = Q\&$ 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 be added top (Tair – Tcan , ave) = (10 W / m 2 . 2 There is no heat generation in the plate additional resistance top (Tair – Tcan , ave) = (10 W / m 2 . 2 There is no heat generation in the plate additin the plate additional resistance 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 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,#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-tau) = 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-tau) = 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-tau) = 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-tau) = 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])+(1-tau) = 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])+(1-tau) = 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])+(1-tau) = TableValue(Row-1,#T[i]) end "Using the explicit finite difference approach, the six equations for the six equa $2^{tau}T$ old[1]+tau*(g dot*DELTAx^2)/k "Node 1, insulated" T[2]=tau*(T old[3]+(1-2*tau)*T old[3]+(1-2*tau)*T old[3]+(1-2*tau)*T old[3]+tau*(g dot*DELTAx^2)/k "Node 3" T[4]=tau*(T old[3]+tau*(g dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau)*T old[3]+tau*(g dot*DELTAx^2)/k "Node 3" T[4]=tau*(T old[3]+tau*(g dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau)*T old[3]+tau*(g dot*DELTAx^2)/k "Node 3" T[4]=tau*(T old[3]+tau*(g dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau)*T old[3]+tau*(g dot*DELTAx^2)/k "Node 3" T[4]=tau*(T old[3]+tau*(g dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau)*T old[3]+tau*(g 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old[3]+tau*(g dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau)*T old[3]+tau*(g dot*DELTAx^2)/k "Node 3" T[4]=tau*(T old[3]+tau*(g dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau)*T old[3]+tau*(g dot*DELTAx^2)/k "Node 3" T[4]=tau*(T old[3]+tau*(g dot*DELTAx^2)/k "Node 4" T[5]=(1-2*tau)*T old[3]+tau*(g dot*DELTAx^2)/k "Node (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] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 146.4 152.9 159.3 ... 1217 1220 1223 1227 1230 1234 1237 1240 1244 1247 T2 [C] 100 106.7 113.4 120.1 126.6 133.3 139.9 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 140.4 104.8 111.3 117 123.3 129.2 135.5 141.5 147.7 153.7 1160 1163 1167 1170 1173 1176 1179 1183 1186 1189 1400 1200 1200 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 "[k]/kg-K], gas constant of air" C p=CP(air, 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(kJ, therm)* 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 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 III 2 L ro 0.0015 m V 350°C Lc = = o = = = 0.00075 m 10 m/min As 2π ro 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 $\Delta x = \Delta x = 1 = 0.02 \text{ m}$, 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 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow Tnode = (Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 \rightarrow 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) $\lambda 1 = 2.1589$ 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 $(29 \text{ Btu/h.ft.}^\circ\text{F}) 2 \text{ ro The constants } \lambda 1 \text{ and } A1 \text{ corresponding to this Biot number are, from Table 4-1, } 2 \text{ ro } \lambda 1 = 0.0998 \text{ and } A1 = 1.0017 \text{ The Fourier number is } \tau = \alpha t 2 = (0.61 \text{ ft } 2/h)(5 \text{ min/60 min/h}) (0.5 / 12 \text{ ft}) 2 \text{ L} = 29.28 > 0.2 \text{ Then the center temperature of the plate becomes } \theta 0, \text{ wall} = 2L \text{ T} - 75 \text{ TO} - \text{T} \infty = A1 \text{ e} - \lambda 1 \text{ } \tau \rightarrow 0 = (1.0017)\text{ e} - (1.0017)\text{$ 0.0998) Ti $-T \propto 400 - 7522$ (29.28) = $0.748 \rightarrow T0 = 318^{\circ}F$ Cylinder: -1 Bi = 0.01 Table $4 \rightarrow \lambda 1 = 0.1412$ and A1 = 1.0025θ 0, cyl = $22T - 75T0 - T \propto 400 - 75$ Sphere: -1 Bi = 0.01 Table $4 \rightarrow \lambda 1 = 0.1730$ and A1 = 1.0030θ 0, sph = $22T - 75T0 - T \propto = A1e^{-1}$ $-\lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.1730)(29.28) = 0.418 \rightarrow T0 = 211^{\circ}F$ Ti $-T \propto 400 - 75$ After 10 minutes $\tau = \alpha t L^2 = (0.61 \text{ ft } 2/h)(10 \text{ min/h})(0.5 / 12 \text{ ft}) 2 = 58.56 > 0.2$ Plate: θ 0, wall = 2 T - 75 T0 $-T \propto = A1e - \lambda 1 \tau \rightarrow 0 = (1.0017)e - (0.0998)(58.56) = 0.559 \rightarrow T0 = 257^{\circ}F$ Ti $-T \propto 400 - 75$ 4-108 Chapter 4 Transient Heat Conduction Cylinder: θ 0, cyl = 2 2 T - 75 T 0 - T ∞ = A1 e - λ 1 τ \rightarrow 0 = (1.0025)e - (0.1412) (58.56) = 0.312 \rightarrow T0 = 176°F Ti - T ∞ 400 - 75 Sphere: 2 2 T - 75 T0 - T ∞ = A1 e - λ 1 τ \rightarrow 0 = (1.0030)e - (0.1730) (58.56) = 0.312 \rightarrow T0 = 176°F Ti - T ∞ 400 - 75 θ 0, sph = After 30 minutes τ = α t L2 = (0.61 ft 2 /h)(30 min/60 min/h) (0.5 / 12 ft) 2 = 175.68 > 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.0000 = 0.00000 = 0.00000 = 0.00000 = 0.0000 = 0.00000 = 0.0 $0.2 Plate: \theta 0, wall = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0017)e - (0.0998)(175.68) = 0.174 \rightarrow T 0 = 132^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0025)e - (0.1412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.0412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.0412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.0412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.0412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.0412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.0412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e - (0.0412)(175.68) = 0.030 \rightarrow T 0 = 84.8^{\circ}F Ti - T \infty 400 - 75 Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T \infty = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0030)e^{-1}$ 0.1730 (175.68) = $0.0052 \rightarrow T0 = 76.7^{\circ}F$ Ti $-T \propto 400 - 75$ The sphere has the largest surface area through which heat is transferred per unit volume, and thus the highest rate of heat transfer. It is proportional to the drag force acting on the plate. Assuming constant thermal conductivity and one-dimensional heat transfer, the mathematical formulation (the differential equation and the boundary conditions) of the heat conduction in the pipe is to be obtained for steady operation. Properties The properties of the trunks are given to be $k = 0.17 \text{ W/m.}^{\circ}\text{C}$ and $\alpha = 1.28 \times 10^{-7} \text{ m}^2/\text{s}$. ° C)(1 m2) Q& 5 mm 5 mm The total thermal resistance is R total = 2 R contact + R plate + 2 R epoxy = 2 × 10^{-7} \text{ m}^2/\text{s}. ° C)(1 m2) Q& 5 mm 5 mm The total thermal resistance is R total = 2 R contact + R plate + 2 R epoxy = 2 × 10^{-7} \text{ m}^2/\text{s}. $0.00017 + 2.6 \times 10 - 6 + 2 \times 0.01923 = 0.03914$ °C/W Then the percent error involved in the total thermal contact resistance of the plate if the thermal contact resistance of the shaft becomes θ $0,cyl = 2\ 2\ T0 - T\infty = A1e - \lambda 1\ \tau = (1.1558)e^{-(1.0935)}(0.1548) = 0.9605\ Ti - T\infty\ T0 - 150 = 0.9605\ \rightarrow T0 = 390\ ^{\circ}C\ 400 - 150\ The\ maximum\ heat\ can\ be\ transferred\ from\ the\ cylinder\ per\ meter\ of\ its\ length\ is\ m = \rho V = \rho \pi ro\ 2\ L = (7900\ kg/m\ 3\
)[\pi(0.175\ m)\ 2\ (1\ m)] = 760.1\ kg\ Qmax = mC\ p\ [T\infty - Ti\] = (760.1\ kg)(0.477\ k]/kg\ ^{\circ}C)(400 - 150)\ ^{\circ}C = 100\ ^{\circ}C\ 400\ - 150\ ^{\circ}C\ 400\$ 90,638 kJ Once the constant J1 = 0.4689 is determined from Table 4-2 corresponding to the constant λ 1 = 1.0935, the actual heat transfer becomes (Q | Q \ max) (T - T \omega | 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (λ 1) (390 - 150) 0.4689 | = 1 - 2| 0 | | T - T \omega / cyl \ i) J 1 (2 - 2) 0.4689 | = 1 - 2| 0 | | | T - T \omega / cyl \ i) J 1 (2 - 2) 0.4689 | = 1 - 2| 0 | | | | Heat Conduction 4-39 "!PROBLEM 4-39" "GIVEN" r o=0.35/2 "[m]" T i=400 "[C]" T infinity=150 "[C]" T infinity=1 lambda 1=1.0935 A 1=1.1558 J 1=0.4709 "From Table 4-2, corresponding to lambda 1" tau=(alpha*time*Convert(min, s))/r o^2 (T o-T infinity)/(T i-T infinity)/(T T infinity)*J 1/lambda 1 time [min] 5 10 15 20 25 30 35 40 45 50 55 60 To [C] 425.9 413.4 401.5 390.1 379.3 368.9 359 349.6 340.5 331.9 323.7 315.8 Q [k] 4491 8386 12105 15656 19046 22283 25374 28325 31142 33832 36401 38853 4-27 Chapter 4 Transient Heat Conduction 440 420 40000 tem perature 35000 heat 400 30000 T o [C] 20000 360 15000 340 10000 320 300 0 5000 10 20 30 40 tim e [m in] 4-28 50 0 60 Q [k] 25000 380 Chapter 4 Transient Heat Conduction 4-40E Long cylindrical steel rods are heat-treated in an oven. (c) Granite: 2-20 Chapter 2 Heat Conduction 2-58 "GIVEN" L=0.15 "[m]" D=0.05 "[m]" $T_1=20$ "[C]" $T_2=95$ "[C]" "k=1.2 [W/m-C], parameter to be varied "ANALYSIS" A=pi*D^2/4 Q dot=k*A*(T 2-T 1)/L k [W/m.C] 1 22 43 64 85 106 127 148 169 190 211 232 253 274 295 316 337 358 379 400 Q [W] 0.9817 21.6 42.22 62.83 83.45 104.1 124.7 145.3 165.9 186.5 207.1 227.8 248.4 269 289.6 310.2 330.8 351.5 372.1 392.7 400 350 300 Q [W] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 50 0 0 50 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 100 150 200 250 k [W /m] 250 200 150 1 C] 2-21 300 350 400 Chapter 2 Heat Conduction Equation 2-59 The base plate of a household iron is subjected to specified heat flux on the left surface and to specified heat flux on the left surface. 4-5 Chapter 4 Transient Heat Conduction 4-17 Milk in a thin-walled glass container is to be warmed up by placing it into a large pan filled with hot water. Using the quadratic formula, the temperature distribution T(r) in the cylindrical shell is determined to be $T(r) = -1\beta \pm 1\beta 2 - 2k$ ave $r^2(r - r^1) + T^2 + T^2\beta + T$ must remain between T1 and T2. The equation for node 6 can be rearranged as $\left(h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus k\Delta x \Delta y | | (h) \right]$ i T i + T9i T5i $||T6 + 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ o + 2 + 2 | \Delta x \setminus \Delta x \ \Delta y \ \Delta x \setminus | \ge 0 \rightarrow \Delta t \le |1 \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 = |1 - 2\alpha\Delta t| \ (h \ 1 \ 1 \ T6i + 1 \ T$ $2\alpha \mid 0 + 2 + 2 \mid k\Delta x \Delta y \Delta x \setminus \mid \mid J$ Substituting the given quantities, the maximum allowable value of the time step is determined to be or, $\Delta t \leq 1 (20 \text{ W/m} \cdot \text{°C})(0.002 \text{ m}) (0.002 \text{$ difference formulation for the total amount of heat transfer at the left boundary during the first 3 time steps are to be determined. The total heat generated in the wire and the heat flux at the interface are go $G\& = g\& V = g\& (\pi 2 L) g\& 0 r0 q\& s = s = = 0 0 = A A (2\pi r0) L 2 D L Assuming steady one-dimensional$ conduction in the radial direction, the heat flux boundary condition can be expressed as -k dT (r0) g& 0 r0 = dr 2 2-42 A long pipe of inner radius r2, and thermal conductivity k is considered. Inside surface, still air m2.°C/W 0.030 0.075 0.12 1.404 m2.°C/W 0.712 W/m2.°C Total unit thermal resistance of each section, R The U-factor of each section, U = 1/R Therefore, the overall unit thermal resistance of the wall is R = $1.404 \text{ m}2.^{\circ}C/W$ and the overall U-factor is U = $0.712 \text{ W/m}2.^{\circ}C$. $e - (2.1589)(1156 = A1e - \lambda 1 \tau = (15618 = 0.007 \text{ Ti} - T_{\infty}] = 0.007 \text{ Ti} - T_{\infty}] = 0.007 \text{ Ti} - T_{\infty}]$ $\rightarrow T(0,0,t) = 211^{\circ}F 40 - 212 \text{ After } 15 \text{ minutes } \tau = \alpha t \ 2 \ L = \theta \text{ o,wall} = \tau = \alpha t \ ro \ 2 \ \theta \text{ o,cyl} = (0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (2.5 \ / \ 12 \ \text{ft}) \ 2 = 1.734 > 0.2 \text{ . Taking } x = 0 \text{ at the surface of the bearing, the boundary } the boundary \ descript{additional} = 1.74 \ descript{additional} = 1.74 \ descript{additional} = 1.74 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ /
\ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ \text{ft}) \ 2 = 1.734 \ descript{additional} = 0.0077 \ \text{ft} \ 2 \ / \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 60 \ h) (0.4 \ / \ 12 \ h)(15 \ / \ 12 \ h)(15 \ / \ 12 \ h)(15 \ / \ 12 \ h)(15$ conditions are u(0) = 0 and u(L) = V, and applying them gives the velocity distribution to be u(y) = y V L Frictional heating due to viscous dissipation in this case is significant because of the high viscosity of oil and the large plate velocity. Convection Equations and Similarity Solutions 6-26C A curved surface if there is no flow separation and the curvature effects are negligible. Comments are made on the results, as appropriate. Then the temperature distribution throughout the glass 15 min after the strip heaters are turned on and when steady conditions disk) 15 min: T1 = -2.4°C, T2 = -2.4°C, T3 = -2.5°C, $T4 = -1.8^{\circ}C$, $T5 = -2.0^{\circ}C$, $T6 = -2.7^{\circ}C$, $T7 = 12.3^{\circ}C$, $T6 = -2.7^{\circ}C$, $T6 = -2.7^{\circ}C$, $T6 = -2.7^{\circ}C$, $T7 = 12.3^{\circ}C$, $T6 = -2.7^{\circ}C$, $T6 = -2.7^$ coefficient are constant and uniform. The rate of heat loss through the jacket is to be determined, and the result is to be compared to the heat loss through a jackets without the air space. For convenience, we choose the time step to be Δt = 15 s. W Δt (60 × 3600 s) Rpipe Rinsulation Ro To Ti T1 T2 T3 The individual thermal resistances are R1 = Rpipe = Rinsulation = $\ln(r2/r1) \ln(0.033/0.03) = 0.0948 \circ C/W 2\pi k$ pipe L $2\pi (0.16 W/m.$ Tskin Analysis The skin temperature can be determined directly from T - Tskin Q& = kA 1 Qconv L (150 W)(0.005 m) Q& L = $37 \circ C - = 35.5 \circ C$ Tskin = T1 - $kA (0.3 W/m. \circ C)(1.7 m 2) 3-27$ Heat is transferred steadily to the boiling water in an aluminum pan. 7-13C The average friction and heat transfer coefficients in flow over a flat plate are determined by integrating the local friction and heat transfer coefficients over the entire plate, and then dividing them by the length of the plate. with increasing temperature, with water being a notable exception. The rate of heat transfer from the fuel rods to the atmosphere through the soil is to be determined. The equation $y'' - 4x^2y = 0$ is linear and homogeneous since each term is linear in y, and contains the dependent variable or one of its derivatives. For flat insulation the R-value is obtained by simply dividing the thickness of the insulation by its thermal conductivity. Qrad Chapter 1 Basics of Heat Transfer 1-129 A room is to be heated by 1 ton of hot water contained in a tank placed in the room. Thus, $v @ 61.66 \text{ kPa} (101.325) - 5 - 5 2 = | (1.774 \times 10) = 2.915 \times 10 \text{ m/s} 61$. Therefore, the Biot number is more likely to be less than 0.1 for the case of natural convection. Tair = -5°C Water pipe T -T [0 - (-5)]°C = 4.87 W Q& = $\infty 1 \propto 2 = 1.0258$ °C / W Rtotal The total amount of heat required to freeze the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = $\rho V = 1.0258$ °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = 0.0258 °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = 0.0258 °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = 0.0258 °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = 0.0258 °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = 0.0258 °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = 0.0258 °C / W Rtotal The total amount of heat lost by the water in the pipe completely is m = 0.0258 °C / W Rtotal The to pπr 2 L = (1000 kg / m3)π (0.01 m) 2 (0.5 m) = 0157. Assumptions 1 Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. 4 The house is air-tight and thus no air is leaking in or out of the room. 2 Thermal properties, heat transfer coefficients, and the surrounding air and surface temperatures remain constant during defrosting. 3 The properties of the surfaces are constant. Therefore, we need to have 5 equations to determine them uniquely. 3 The thermal properties of the orange are constant. of thermal contact resistance, it is to be determined if the plastic insulation on the wire will increase or decrease heat transfer from the wire. Then the length of time for the egg to be kept in boiling water is determined to be $t = \tau ro 2$ (0.1727)(0.0275 m) 2 = 895 s = 14.9 min α (0.146 × 10 -6 m 2/s) 4-36 Chapter 4 Transient Heat Conduction 4-47 A hot dog is dropped into boiling water, and temperature measurements are taken at certain time intervals. The average heat transfer coefficient is to be determined. Properties The thermal conductivities are given to be k = 12 W/m·°C for the epoxy adhesive 8-10C The fluid viscosity is responsible for the development of the velocity boundary layer. ° C)(15 m) 1 1 = = = 0.001768 ° C / W hi Ai (400 W / m2. The final equilibrium temperature of the combined system is to be determined. surfaces. Therefore, there are two • • • unknowns T1 and T2, and we need two equations to determine 0 1 them. Also, we would use the cylindrical coordinates. Analysis The characteristic length and Biot number for the glass of milk are Lc = πro 2 L π (0.03 m) 2 $(0.07 \text{ m}) \text{ V} = = 0.01050 \text{ m} \text{ As } 2\pi \text{ ro } \text{ L} + 2\pi \text{ ro } 2\pi (0.03 \text{ m})(0.07 \text{ m}) + 2\pi (0.03 \text{ m})(0.0$ p Lc (998 kg/m 3)(4182 J/kg.°C)(0.0105 m) -1 T (t) - T ∞ 38 - 60 = e - bt \rightarrow = e - (0.005477 s)t \rightarrow t = 174 s = 2.9 min 3 - 60 Ti - T ∞ Therefore, it will take about 3 minutes to warm the milk from 3 to 38°C. The temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its temperature of balls after quenching and the rate at which heat needs to be removed from the water in order to keep its
temperature of balls after quenching and the rate at which heat needs to keep its temperature of balls after quenching and the rate at which heat needs to keep its temperature of balls after quenching and the rate at which heat needs to keep its temperature of balls a constant are to be determined. - (10° F) 0.475 - 0.0203(30 mph) + 0.304 30 mph = -33.2° F V = 40 mph: Tequiv = 914 . 7-73 Chapter 7 External Forced Convection 7-84 "!PROBLEM 7-84" "GIVEN" T i=110 "[C]" T o=22 "[C]" k pipe=52 "[W/m-C]" r 1=0.02 "[m]" t pipe=0.003 "[m]" t pipe=52 "[C]" k pip h o=22 "[W/m^2-C]" k ins=0.038 "[W/m-C]" "ANALYSIS" L=1 "[m], 1 m long section of the pipe is considered" A i=2*pi*r 3*L r 3=r 2+t ins*Convert(cm, m) "t ins is in cm" r 2=r 1+t pipe R conv i=1/(h i*A i) R pipe=ln(r 2/r 1)/(2*pi*k pipe*L) R ins=ln(r 3/r 2)/(2*pi*k ins*L) R conv o=1/(h o*A o)R total=R conv i+R pipe+R ins+R conv o Q dot=(T i-T o)/R total Q dot=(T s max-T o)/R conv o Ts, max [C] 24 26 28 30 32 34 36 38 40 42 44 46 48 tins [cm] 3 2.5 2 1.5 1 0.5 0 20 25 30 35 40 12 44 46 48 tins [cm] 4.45 2.489 1.733 1.319 1.055 0.871 0.7342 0.6285 0.5441 0.4751 0.4176 0.3688 0.327 7-74 Chapter 7 External Forced Convection 4.5 4 3.5 t ins [cm] 3 2.5 2 1.5 1 0.5 0 20 25 30 35 40 7 s,m ax [C] 7-75 45 50 Chapter 7 External Forced Convection 7-85 A cylindrical oven is to be insulated to reduce heat losses, the average Nusselt number of heat transfer coefficient for all the tubes in tubes in the tubes in t tubes is N = NL ×NT = 30×15 = 450. 3-121C It provides an easy way of calculating the steady rate of heat transfer between two isothermal surfaces in common configurations. 2-114C The order of a differential equation is the order of the highest order derivative 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. Air Properties of air at 25°C Cp = 1.007 kJ/kg-K, Pr = 0.7296 ρ = 1.184 kg/m3, 8 m/s Analysis First, we determine the rate of heat transfer from Q& = mC $p_{airfoil}$ (T2 - T1) $\Delta t = (50 \text{ kg})(500 \text{ J/kg} \cdot ^{\circ}\text{C})(160 - 150)^{\circ}\text{C} = 2083 \text{ W} (2 \times 60 \text{ s})$ Then the average heat transfer coefficient is Q& 2083 W $\rightarrow h = = = 1.335 \text{ W/m} 2 \cdot ^{\circ}\text{C}$ Q& = hAs (Ts - T ∞) (12 m)(155 - 25)^{\circ}\text{C} L=3 m where the surface temperature of airfoil is taken as its average temperature, which is (150+160)/2=155^{\circ}\text{C}. Assumptions 1 Heat transfer is one-dimensional since the plate is large relative to its thickness. \circ C h = Nu = (2908) = 68.3 W / m2. Therefore, doubling the thickness of plastic cover will increase the rate of heat loss and decrease the interface temperature. $|= 0.3206 \text{ kg} (3.81 \text{ kJ/kg.} \circ \text{C})[20 - (-15)] \circ \text{C} = 42.76 \text{ kJ}$ $m = \rho V = \rho$ Omax Then the actual amount of heat transfer becomes $sin(\lambda 1) - \lambda 1 cos(\lambda 1) O sin(1.476 rad) = (0.402)(42.76 k] = 17.2 k$ 4-40 Chapter 4 Transient Heat Conduction 4-51 "!PROBLEM 4-51" "GIVEN" T infinity=-15 "[C]" "T i=20 [C]. W/m^2-C " r o=0.09/2 "[m]" time=1*3600 "[s]" "PROPERTIES" k=0.513 "[W/m-C]" rho=840 "[kg/m^3]" C p=3.6 "[k]/kg-C]" alpha=1.3E-7 "[m^2/s]" "ANALYSIS" Bi=(h*r o)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/(T i-1)/k "From Table 4-1 corresponding to this Bi number, we read" lambda 1=1.3525 A 1=1.1978 tau=(alpha*time)/r o^2 (T o-T infinity)/ $\int \frac{1}{2} \left(\frac{1}{2} + \frac{1}{2} \right) \left(\frac{1}{2} + \frac{1}{2$ 0.08803 1.482 3.051 4.621 6.191 7.76 9.33 10.9 12.47 14.04 Tr [C] -5.369 -4.236 -3.103 -1.97 -0.8371 0.296 1.429 2.562 3.695 4.828 5.961 4-41 Q [k] 6.861 7.668 8.476 9.283 10.09 10.9 11.7 12.51 13.32 14.13 14.93 Chapter 4 Transient Heat Conduction 24 26 28 30 15.61 17.18 18.75 20.32 7.094 8.227 9.36 10.49 15.74 16.55 17.35 18.16 25 20 15 TO T o [C] 10 5 Tr 0 -5 -10 0 5 10 15 Ti [C] 4-42 20 25 30 Chapter 4 Transient Heat Conduction 20 18 Q [k] 16 14 12 10 8 6 0 5 10 15 Ti [C] 4-43 20 25 30 Chapter 4 Transient Heat Conduction 4-52 An orange is exposed to very cold ambient air. The rate of heat transfer through the solid stud and through a stud pair nailed to each other, as well as the effective conductivity of the nailed stud pair are to be determined. 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 μ s , @ 0° C = $1.729 \times 10 - 5$ kg/m.s ν = 1.608×10 m/s -5 $\mu \infty = 1.872 \times 10 - 5$ 2 Pr = 0.7282 Ts, out 0° C V $\infty = 25$ km/h T $\infty = 30^{\circ}$ C kg/m.s Iced water $Di = 3 \text{ m} 0^{\circ}C$ Analysis (a) The Reynolds number is V D [(25 × 1000/3600) m/s](3.02 m) Re = $\infty = = 1.304 \times 10.6 \text{ v} 1.608 \times 10.-5 \text{ m} 2$ /s The Nusselt number corresponding to this Reynolds number is determined from Nu = [] ($\mu \text{ hD} = 2 + 0.4 \text{ Re} 0.5 + 0.06 \text{ Re} 2/3 \text{ Pr} 0.4$] ($\mu \text{ s} 1/4$) [] ($1.872 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 2/8 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m} 1.304 \times 10.6$) ($1.508 \times 10.-5 \text{ m}$ $0.06(1.304 \times 10.6) 2/3$ (0.7282) $0.4 \parallel -5 \sqrt{1.729 \times 10.1/4} \parallel / = 1056$ 0.02588 W/m.°C k Nu = (1056) = 9.05 W/m 2.°C D 3.02 m In steady operation, heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by
conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the heat transfer through the tank by conduction is equal to the tank by conduction is equal to the tank by conduction is equal ton $C = 1 \otimes 2 \rightarrow T1 = T \otimes T$ \approx 43.0° C Rboard Therefore, the board is nearly isothermal. The mass of the counterweight that needs to be added in order to balance the plate is to be determined. 1 2 Analysis (a) The nodal spacing is given to be • • 3 $\Delta x = \Delta x = 1 = 0.015$ m, and the general finite Insulated qL difference form of an interior node for steady 8 • two-dimensional heat conduction for the case of •4 •5 •6 •7 constant heat generation is expressed as Tleft + Ttop + Tright + Tbottom - 4Tnode + g& 012 = 0 k 120 We observe that all nodes are boundary nodes except node 5 that is an interior node. The recommendation is logical. Then the maximum velocity and the Reynolds number based on the maximum velocity. become V=3.8 m/s Vmax = Re D ST 0.05 V = $(3.8 \text{ m/s}) = 6.552 \text{ m/s} 0.05 - 0.021 \text{ ST} - D \text{ Ti} = 15^{\circ}\text{C} \rho V D (1.204 \text{ kg/m 3})(6.552 \text{ m/s})(0.021 \text{ m}) = max = 9075 \mu 1.825 \times 10 - 5 \text{ kg/m} \cdot \text{s} \text{ Ts} = 90^{\circ}\text{C} \text{ SL} \text{ ST}$ 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(9075)0.63 (0.7309) 0.36 (0.7309 / 0.7132) 0.25 = 75.59 This Nusselt number is applicable to tube banks with NL > 16. There is no heat generation in the medium, and the thermal conductivity k of the medium is constant. Gypsum wallboard, 13 mm 8. The location and values of the highest and the lowest temperatures in the plate are to be determined Then the number of nodes becomes $M = L / \Delta x + 1 = 0.08/0.02 + 1 = 5$. The derivative of a function that depends on two or more independent variables constant is called the partial derivative. 2 Heat is generated uniformly in the body. It is applicable when the Biot number (the ratio of conduction resistance within the body to convection resistance at the surface of the body) is less than or equal to 0.1. 4-2C The lumped system analysis is more likely to be applicable for the body cooled naturally since the Biot number is proportional to the convection heat transfer coefficient, which is proportional to the air velocity. Therefore, the 3rd and following terms in the Taylor series expansion represent the error involved in the finite difference approximation. ° C [4(0.0002) + 3(0.0015) m The maximum temperature will occur at the midplane of the board that is the farthest to the heat sink. 6 The plant operates 110 days a year. 1-51 Chapter 1 Basics of Heat Transfer 1-94 Using the conversion factors between W and Btu/h, m and ft, and K and R, the Stefan-Boltzmann constant $\sigma = 5.67 \times 10-8$ W / m2. The nodal spacing along the hi Δx Ti flange is given to be $\Delta x 2=1$ cm = 0.01 m. 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 • $n+1 g\Delta x \Delta y \Delta z \cdot m+1$ go $\Delta z r \cdot m+1 \Delta y \Delta x \cdot n \Delta E$ element Q& cond, left + Q& cond, right + Q& cond, left + Q& cond, 12 k | | where g& 0 = 8×10 W/m, l = 0.1 m, and k = 28 W/m·°C, h = 45 W/m2·°C, and T ∞ =30°C. 5-108 Chapter 5 Numerical Methods in Heat Conduction 5-112 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 k and uniform heat generation g& 0 is to be obtained. Thermal conductivity is a fluid property, and its value does not depend on the flow. 2-29 Chapter 2 Heat Conductions on the bottom surface. Analysis (a) Noting that the heat transfer through the wall is by conduction and the surface area of the wall is A = 20 ft \times 10 ft = 200 ft 2, the steady rate of heat transfer through the wall is A = 20 ft \times 10 ft = 200 ft 2, the steady rate of heat transfer through the wall is by conduction and the surface area of the wall can be determined from T – T (62 – 25)° F Q& = kA 1 2 = (0.42 Btu / h.ft. For the drink, we use the properties of water at room temperature, ρ = 1000 kg/m3 and Cp = 4180 J/kg.°C. Properties of water at the average temperature of $(T1 + T2)/2 = (3+11)/2 = 7^{\circ}C$ are $(Table A9) \rho = 999.8 kg/m 3 C p = 4200 J/kg.^{\circ}C$ The properties of air at 1 atm and the film temperature of $(Ts + T\infty)/2 = (7+27)/2 = 17^{\circ}C$ are $(Table A-15) k = 0.02491 W/m.^{\circ}C v = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.489 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ Air $V \infty T \infty = 1.480 \times 10^{-5} m 2/s$ 27° C Pr = 0.7317 Analysis The mass of water in the bottle is L = 30 cm 2 D L = (999.8 kg/m 3) \pi (0.10 m) 2 (0.30 m)/4 = 2.356 kg 4 Then the amount of heat transfer to the water is Q = mC p (T2 - T1) = (2.356 kg)(4200 J/kg.°C)(11-3)°C = 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J Q& = = 29.32 W \Delta t 45 \times 60 s The heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V = \rho T The average rate of heat transfer is Q 79,162 J m = \rho V
= \rho T The average rate of heat transfer is Q 79,162 J m = \rho T The average rate of heat transfer is Q 79,162 J m = \rho T The average rate of heat transfer is Q 79,162 J m = \rho T The average rate of heat transfer is Q 79,162 J m = \rho T The average rate of heat transfer is Q 79,162 J m = \rho T The average rate of heat transfer is Q 79,162 J m = \rho T The average rate of heat transfer is Q m = \rho T The average rate of heat transfer i transfer coefficient is $As = \pi DL = \pi (0.10 \text{ m})(0.30 \text{ m}) = 0.09425 \text{ m } 2 \text{ Q} \text{ conv} = hAs (Ts - T\infty) \rightarrow 29.32 \text{ W} = h(0.09425 \text{ m } 2)(27 - 7)^{\circ}C \rightarrow h = 15.55 \text{ W/m } 2.^{\circ}C$ The Nusselt number is hD (15.55 W/m 2.°C)(0.10 m) = 62.42 k 0.02491 W/m.°C Reynolds number can be obtained from the Nusselt number relation for a flow over the cylinder Nu = 5/8 $0.62 \text{ Re } 0.5 \text{ Pr } 1/3 \left[\left(\text{ Re } \right) \right]$ Nu = $0.3 + +1 \left[\left| \left| \left| 1/4 \right| \left[\left(282,000 \right) \right| \right] 1 + (0.4 / \text{ Pr }) 2/3 \left[4/5 \right] 5/8 0.62 \text{ Re } 0.5 (0.7317) 1/3 \left[\left(\text{ Re } \right) \right] 62.42 = 0.3 + 1 + \left[\left| \left| \left| 1/4 \right| \left[\left(282,000 \right) \right| \right] 1 + (0.4 / \text{ Pr }) 2/3 \left[\text{ Bottle } \text{ D = 10 cm } \right] 4/5 \rightarrow \text{Re } = 12,856 \text{ Then using the Reynolds number relation we determine the wind velocity V D } V_{\infty} (0.10 \text{ m) Re } = \infty$ -12,856 = - V = 1.91 m/s v 1.489 × 10 - 5 m 2 /s 7-55 Chapter 7 External Forced Convection Flow Across Tube Banks, the flow characteristics are dominated by the maximum velocity V at that occurs within the tube banks, the flow characteristics are dominated by the maximum velocity V max that occurs within the tube banks, the flow characteristics are dominated by the maximum velocity V. 3 The heat transfer coefficient is constant and uniform over the entire fin surface. Analysis The nodal spacing is given to be $\Delta x = 0.25$ m. 6 The combined heat transfer coefficient on the outer surface remains constant even after the pipe is insulated. Analysis (a) The nodal spacing is given to be $\Delta x = 0.25$ m. 6 The combined heat transfer coefficient on the outer surface remains constant even after the pipe is insulated. conduction for the case of constant heat generation is expressed as Tleft + Ttop + Tright + Tbottom - 4Tnode + g& node 1 2 = 0 k There is symmetry about a vertical line passing through the middle of the region, and thus we need to consider only half of the region. The amount of power this transistor can dissipate safely is to be determined. We observe that there is only one inlet and one exit and thus m& 1 = m& 2 = m&. Btu / h.ft 2. m)(6 m) = 188. Assumptions Heat transfer from the surface of the filament and the bulb of the lamp is uniform . 2-60 Chapter 2 Heat Conduction Equation 2-106C A differential equation may involve more than one dependent variable. Analysis Using the energy balance approach and taking the direction of all heat transfers to be towards the node under consideration, the explicit finite difference formulations become Left boundary node: $Ai + 1iTi - T0i\Delta x T 0 - T 0 + q & 0A + g & 0i$ ($A\Delta x / 2$) = $\rho A C \Delta x 2 \Delta t g(x, t) q0$ Right boundary node: $Ti - T4i\Delta x T4i + 1 - T4i + hA(T∞i - T4i) + hA(T∞i - T$ $g\&4i(A\Delta x/2) = \rho A C kA 3 \Delta x 2 \Delta t 5-67 h, T \propto \Delta x \cdot 0 \cdot 1 \cdot 2 \cdot 3 4 \cdot Chapter 5$ Numerical Methods in Heat Conductivity is subjected to uniform heat flux q&0 at the left (node 0) and convection at the right boundary (node 4). $\tau = \alpha t ro 2 \theta o$, cyl = = (0.0077 ft 2 / h)(5 / ft 2 / h)(6 /60 h (0.4 / 12 ft) 2 = 0.578 > 0.2 2 2 T0 - T\infty. energies Wpw,in - Qout = ΔU Wpw ,in = Qout + m(u2 - u1) \cong Qout + mCv (T2 - T1) The final temperature and the number of moles of oxygen are PV PV 1 = 2 T1 T2 m = \rightarrow T2 = 20 psia P2 (540 R) = 735 R T1 = 14.7 psia 80°F 20 Btu (14.7 psia)(10 ft 3) PV 1 = = 0.812 lbm RT1 $(0.3353 \text{ psia} \cdot \text{ft } 3 / \text{lbmol} \cdot \text{R})(540 \text{ R})$ The specific heat ofoxygen at the average temperature of Tave = $(735+540)/2 = 638 \text{ R} = 178^{\circ}\text{F}$ is Cv,ave = Cp - R = 0.2216-0.06206 = 0.160 Btu/lbm.R)(735 - 540) Btu/lbmol = 45.3 Btu Discussion Note that a "cooling" fan actually causes the internal of t temperature of a confined space to rise. Therefore, the given convection heat transfer coefficient in English units is h = 20 W/m 2.°F = 3.52 Btu/h.ft 2.°F Applying the boundary conditions give r = r2: r = r1: $k C 1 r22 = q \& s \rightarrow C1 = T (r1) = T1 = -q \& s r22 k C1 C q \& r 2 + C2 \rightarrow C2 = T1 + 1 = T1 + s 2 r1 r1 kr1$ Substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = -(11)(11) q \& r 2 C1 C C + C 2 = -1 + T1 + 1 = T1 + || - ||C1 = T1 + ||C1 = T1 s 2 r r r 1 (r 1 r) (r 1 r)insulation is determined to be L = 0.0045 m = 0.45 cm 45 W / m 2 = 3-18 R3 Ro Trefrig Chapter 3 Steady Heat Conduction 3-36 "GIVEN" k ins=0.035 "[W/m-C], parameter to be varied" L metal=15.1 "[W/m-C], parameter to be varied "L metal=15.1 "[W/m-C], parame "ANALYSIS" A=1 "[m^2], a unit surface area is considered" Q dot=h o*A*(T kitchen-T s out) Q dot=(T kitchen-T refrig)/R total R total=R conv i+2*R metal+R ins+R conv o R conv i=1/(h i*A) R ins=(L ins*Convert(cm, m))/(k ins*A) "L ins is in cm" R conv o=1/(h o*A) kins [W/m.C] 0.02 0.025 0.03 0.035 0.04 0.045 0.055 0.06 0.065 0.07 0.075 0.08 Lins [cm] 0.2553 0.3191 0.3829 0.4468 0.5106 0.5744 0.6382 0.702 0.7659 0.8297 0.8935 0.9573 1.021 kmetal [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 Lins [cm] 0.4465 0.4471 0.4471 0.4471 0.4471 0.4471 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0.4472 0. 3 Steady Heat Conduction 3-37 Heat is to be conducted along a circuit board with a copper layer on one side. $29 \xi = |L + |2| kt \langle 2| | \langle 1 \rangle (52 \text{ W/m o C})(0.02 \text{ m}) 0.02 \text{ t} 0.1 + 2 = 2 = 2.23 0.05 \text{ r} 1 \text{ r} 2 + A \text{fin} = 2\pi (r^2 2 - r^2 2) + 2\pi r^2 t = 2\pi [(01 \text{ Properties The gas constant of oxygen is } R = 0.3353 \text{ psia.ft3/lbm.R} = 0.06206 \text{ Btu/lbm.R} (Table A-1E).$ Assuming variable thermal conductivity and one-dimensional heat transfer, the mathematical formulation (the differential equation and the boundary conditions) of this heat conductivity and one-dimensional heat transfer, the outer part of the beef carcass will freeze during this cooling process. 3-90 Chapter 3 Steady Heat Conduction Heat Transfer In Common Configurations 3-120C Under steady conditions, the rate of heat transfer between two surfaces is expressed as Q& = Sk (T1 - T2) where S is the conduction shape factor. 2-33 For a medium in which the heat conduction equation is given by $\partial 2T 1 \partial T 1 \partial (2 \partial T) 1 r + = : | | r 2 \partial r (\partial r / r 2 sin 2 \theta \partial \varphi 2 \alpha \partial t (a)$ Heat transfer is transient, (b) it is two-dimensional, (c) there is no heat generation, and (d) the thermal conductivity is constant. k (2.5 W/m.°C) $\tau = \alpha t 2 L = (115 . Then the number of nodes M becomes M = L 18 cm + 1 = + 1 = 13 \Delta x 1.5 cm$ Tsurr h, T₂ The base temperature. at node 0 is given to be T0 = 95°C. ° C)(60 - 18)° C = 23.5 W (b) Using the water properties at the room temperature, the temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature, the temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature, the temperature drop of the hot water as it flows through this 5-m section of the water properties at the
room temperature, the temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature, the temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature drop of the hot water as it flows through this 5-m section of the water properties at the room temperature drop of the hot water as it flows through the section of the water properties at the room temperature drop of the hot water as it flows through the section of the water properties at the room temperature drop of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the section of the hot water as it flows through the hot water as it flows through t m/s) | (4180 J/kg.°C) 4 || | 3-95 Chapter 3 Steady Heat Conduction 3-128 Hot water is flowing through a pipe that extends 2 m in the ambient air and continues in the ground before it enters the next building. Analysis We take the tube as the system. We do not know the inner surface temperatures of windows. Such systems are characterized by apparent thermal conductivity instead of the ordinary thermal conductivity in order to incorporate these convection and radiation effects. 1-80 Q& -8°C Chapter 1 Basics of Heat Transfer 1-139 "GIVEN" A=1.2*1.8 "[m^2]" T_1=20 "[C]" "U=1.25 [W/m^2-C], parameter to be varied" "ANALYSIS" Q dot_window=U*A*(T_1-T_2) U [W/m2.C] 1.25 1.75 2.25 2.75 3.25 3.75 4.25 4.75 5.25 5.75 6.25 Qwindow [W] 75.6 105.8 136.1 166.3 196.6 226.8 257 287.3 317.5 347.8 378 400 350 300 Q w indow [W] 250 200 150 100 50 1 2 3 4 5 2 U [W/m - C] 1-140. 4-15b we have T - T × 35 - 5] = 0.75 | To - T × 45 - 5 k | 1 = 1.75 | x ro | Bi hro = 1 [J ro ro Refrigerator T = 5°F Then the heat transfer coefficients becomes $h = (0.26 \text{ Btu/h.ft.}^{\circ}\text{F})$ $k = 0.590 \text{ Btu/h.ft.}^{\circ}\text{C}$. 4 The thermal conductivity of styrofoam is given to be $k = 0.033 \text{ W/m}^{\circ}\text{C}$. 4 The thermal resistance of the tank is negligible, and the entire steel tank is at 0°C. Then the length of time before the log ignites is $t = \tau \text{ro } 2 (0.251)(0.05 \text{ m}) 2$. = = 4904 s = 81.7 min α (1.28 × 10 - 7 m 2 /s) 4-31 Chapter 4 Transient Heat Conduction 4-43 A rib is roasted in an oven. However, we can take the air to be a closed system by considering the air in the room to have undergone a constant. 6 The thermal properties of the fins are constant. 4 The temperature of the thin-shelled spherical tank is nearly equal to the temperature of the oxygen inside. Ao = $\pi DL + 2\pi (\pi D 2 / 4) = \pi (1.35 \text{ m})(6 \text{ m}) + 2\pi (1.35 \text{ m})(2 / 4) = 28.31 \text{ m} 2 1 1 = 0.001413 ^{\circ}C/W$ Rinsulation, side = $2\pi kL 2\pi (0.038 \text{ m})(6 \text{ m}) + 2\pi (1.35 \text{ m})(2 / 4) = \pi (1.35 \text{ m}$ $W/m.^{C}(6 m) 2 \times 0.075 m L = 3.0917 ^{C}/W Rinsulation, ends = 2 kAave (0.038 W/m.^{C})[\pi (1.275 m) 2 / 4]$ Noting that the insulation is determined to be $-1 - 1 / (1 1 1 1) = (| = 0.08009 ^{C}/W Rinsulation = | + + | | Rinsulation, side$ Rinsulation, ends $| 0.08222 ^{\circ}C/W 3.0917 ^{\circ}C/W | \rangle$ Then the total thermal resistance and the heat transfer rate become R total = R conv, o + R insulation = 0.001413 + 0.08009 = 0.081503 $^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0.081503 ^{\circ}C/W T - Ts [30 - (-42)]^{\circ}C = 883.4 W Q \& = \infty 0$ 0.002079 kg/s Q& = m& h fg \rightarrow m& = 425 kJ/kg h fg Δt = 3942.6 kg m = 1,896,762 s = 526.9 hours = 21.95 days m& 0.002079 kg/s 3-114 Chapter 3 Steady Heat Conduction 3-155 Hot water is flowing through a 3-m section of a cast iron pipe. The average temperature of the outer surface of the person is to be determined. ° C)(0.015 m) = 0.0109 k (110 W / m. 4 The inner surface of the shell is at the same temperature as the iced water, 0°C. Properties The thermal conductivity, density, and Cp = 0.480 kJ/kg.°F. Analysis For the laminar flow of a fluid over a flat plate maintained at a constant temperature the thermal conductivity, density, and Cp = 0.480 kJ/kg.°F. drag force is given by FD1 = C f As ρ V ∞ 2 2 where C f = 1.328 Re 0.5 Therefore FD1 = V ∞ 1.328 0.5 As ρ V ∞ 2 2 Re Substituti ng Reynolds number relation, we get ρ V ∞ 2 1.328 L υ 0.5 As = 0.664 V ∞ As 0.5 0.5 2 L (V ∞ L) || υ When the free-stream velocity of the fluid is doubled, the new value of the drag force on the plate becomes FD1 = FD 2 = 1.328 As $3/2 \rho$ ($2 V \infty$) 2 = 0.664($2V \infty$) 2 = 0.664($2V \infty$) 3 / 2 As 0.5 2 ($(2 V \infty) L$) | $\langle k \rangle Q \& 1 = hAs$ (T s - T ∞) = | Nu | As (T s - T ∞) = | Nu | As (T s - T ∞) = | 0.664 Re 0.5 Pr 1 / 3 As (Ts - T ∞) (L / (L) 0.5 k (V L) 0.664] ∞ | Pr 1 / 3 As (Ts - T ∞) L υ / k = 0.664(2V ∞) 0.5 0.5 Pr 1 / 3 As (Ts - T ∞) L υ / k = 0.664(2V ∞) 0.5 0.5 Pr 1 / 3 As (Ts - T ∞) L υ / k = 0.664(2V ∞) 0.5 0.5 Pr 1 / 3 As (Ts - T ∞) L υ / L υ / L υ / L υ / k = 0.664(2V ∞) 0.5 0.5 Pr 1 / 3 As (Ts - T ∞) L υ / L υ / k = 0.664(2V ∞) 0.5 0.5 Pr 1 / 3 As (Ts - T ∞) L υ / L υ $V \propto 0.5 = 2$ 0.5 = 2 = Q& V 0.5 = 1 \propto 7-17 Chapter 7 External Forced Convection 7-26E A refrigeration truck is traveling at 55 mph. The rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat
loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from the sphere and the rate of heat loss from theat loss from the sphere and the 2) $(T2 + Tsurr) = 1(5.67 \times 10 - 8 \text{ W/m } 2.\text{K } 4)[(273 + 5 \text{ K}) 2 + (273 + 30 \text{ K})(273 + 5 \text{ K})] = 5.570 \text{ W/m } 2.\text{K } \text{The individual thermal resistances are Ri } T \propto 1 \text{ T } R \text{ ra d } R1 \text{ Ro } T \propto 2 \text{ 1 } 1 = 0.000159 \text{ °C/W } 4 \pi \text{ kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 - 2.5) \text{ m } R1 = \text{R sphere} = 2 1 = 0.000013 \text{ °C/W } 4 \pi \text{kr1 } r2 4 \pi (15 \text{ W/m. } ^{\circ}\text{C})(2.515 \text{ m } 2) r - r(2.515 \text{ m } 2) r - r(2$ transfer to the iced water becomes T –T (30 – 0)°C Q& = $\infty 1 \propto 2 = 30,581$ W Rtotal 0.000981 °C/W (b) The total amount of heat transfer during this period are Q = Q& $\Delta t = (30.581 \text{ kJ/s})(24 \times 3600 \text{ s}) = 2.642 \times 10.6 \text{ kJ} = 7918 \text{ kg}$ hif 333.7 kJ/kg Check: Theorem the transfer during this period are Q = Q& $\Delta t = (30.581 \text{ kJ/s})(24 \times 3600 \text{ s}) = 2.642 \times 10.6 \text{ kJ}$ m ice = Q 2.642 × 10.6 kJ m ice = Q 2.642 \times 10.6 kJ outer surface temperature of the tank is $Q_{\&} = hconv + rad Ao = 30^{\circ}C - 30,581 W (10 + 5.57 W/m 2) = 5.3^{\circ}C which is very close to the assumed temperature of 5^{\circ}C for the outer surface tem$ is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -240°C and 1.30 MPa. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. 2 Thermal properties of the tank and the convection heat transfer coefficient is constant and uniform. Gypsum wallboard, 13 7. The effective thermal conductivity of the board along its 9 in long side and the fraction of the heat conducted through copper along that side are to be determined. Analysis The surface areas of the glass = 0.85(1.5 m)(2 m) = 2.55 m 2 Ri, wood Rolass R wood = 0.15(1.5 m)(2 m) = 2.55 m 2 Ri, wood Rolass R wood = 0.15(1.5 m)(2 m) = 2.55 m 2 Ri, wood Rolass R wood Rolass R wood = 0.15(1.5 m)(2 m) = 2.55 m 2 Ri, wood Rolass R wood Rolass R wood = 0.15(1.5 m)(2 m) = 2.55 m 2 Ri, wood Rolass R wood Rolass R wood Rolass R wood Rolass R wood = 0.15(1.5 m)(2 m) = 2.55 m 2 Ri, wood Rolass R $0.45 \text{ m } 2 \text{ 1 } 1 = 0.05602^{\circ}\text{C/W h1}$ Aglass $(7 \text{ W/m } 2.^{\circ}\text{C})(2.55 \text{ m } 2) \text{ 1 } 1 = = 0.31746^{\circ}\text{C/W k}$ glass $0.003 \text{ m} = = 0.00168^{\circ}\text{C/W k}$ glass $1 \text{ a} \text{ a} \text{ a} \text{ b} \text{ a} \text{ a} \text{ b} \text{ b} \text{ a} \text{ b} \text{ a} \text{ b} \text{ a} \text{ b} \text{ a} \text{ b} \text{ b} \text{ b} \text{ a} \text{ b} \text{$ h2 Aglass (13 W/m 2.°C)(2.55 m 2) 1 1 = $0.17094^{\circ}C/W$ h2 Awood (13 W/m 2.°C)(0.45 m 2) = Ri,glass + Ro,glass = $0.05602 + 0.00168 + 0.03017 = 0.08787^{\circ}C/W$ Ro, wood = R total, glass R total, wood = R total, glas wood and their total are T – T ∞ 1 T – T ∞ 1 T – T ∞ 1 (40 – 24)°C (40 – 24)°C (40 – 24)°C = 182.1 W Q& glass = ∞ 2 Q& wood = ∞ 2 = 11.3 W R total, wood = 182.1 + 11.3 = 193.4 W If the window consists of glass only the heat gain through the window is Aglass = (1.5 m)(2 m) = 3.0 m 2 Ri, glass = Rglass = Ro, glass = Ro, glass = Ro, glass = 0.04762 °C/W 2 h1 Aglass (7 W/m.°C)(3.0 m 2) = 0.00143 °C/W 1 1 = 0.02564 °C/W 2 h2 Aglass = 0.00143 °C/W 1 1 = 0.02564 °C/W 2 h2 Aglass = 0.00143 °C/W 1 1 = 0.02564 °C/W 2 h2 Aglass = 0.00143 °C/W 1 1 = 0.02564 °C/W 2 h2 Aglass = 0.00143 °C/W 1 1 = 0.00143= 214.2 W R total, glass 0.07469°C/W Then the percentage error involved in heat gain through the window assuming the window consist of glass only - Q& with wood 214.2 - 193.4 % Error = = × 100 = 10.8% 193.4 Q& with wood 3-132 Chapter 3 Steady Heat Conduction 3-172 Steam is flowing inside a steel pipe. (0.2 m)] = $9.550 \text{ kg T} = Q = \text{mC} (T - T) = (9.550 \text{ kg})(0.896 \text{ kJ/kg.}^{\circ}C)(20 - 1200)^{\circ}C = 10,100 \text{ kJ max p i} = 1 - 20 \text{ o,cyl 1} = 1 - 20 \text{ o,cyl 1} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 20 \text{ o,cyl 1} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 20 \text{ o,cyl 1} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 20 \text{ o,cyl 1} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 20 \text{ o,cyl 1} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) = 0.2415 \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg} \cdot (\lambda + 1) \sin(0.1811) \text{ kg} = 1 - 2(0.7627) \text{ kg}$ $0.2425 \lambda 0.2217 1$ cyl The heat transfer ratio for the short cylinder is (Q | Q | max (Q) | | short = | Q / cylinder | max | L z r 0 Cylinder | max | L z r 0 Cylinder L Ti = 20°C | | | plane + | Q / wall | Then the total heat transfer from the short cylinder | max | L z r 0 Cylinder | max | L z r 0 Cylinder L Ti = 20°C | | | plane + | Q / wall | Then the total heat transfer from the short cylinder | max | L z r 0 Ccylinder as it is cooled from 300°C at the center to 20°C becomes Q = 0.4236Q max = (0.4254)(10,100 kJ) = 4297 kJ 4-85 Chapter 4 Transient Heat Conduction which is identical to the heat transfer to the cylinder at 20°C is heated to 300°C at the center. The rate of heat loss from that wall is to be determined for two cases. 3.0153 °C at the center to 20°C is heated to 300°C at the center. / W Rtotal Water pipe $Q = Q \& \Delta t = (1658 . However, using a very large value of \Delta t is equivalent to replacing the time derivative by a very large area for outer convection resistance, the inner surface area for inner convection resistance, and the average area for the$ conduction resistance. Using the finite difference form
of the boundary nodes is to be determined. The electric power input to the heater, and the money that will be saved during a 10-min shower by installing a heat exchanger with an effectiveness of 0.50 are to be determined. The temperature of the boundary nodes is to be determined a temperature of the boundary nodes is to be determined. the right surface of the wall and the rate of heat transfer are to be determined. The time for the LNG temperature to rise to -150°C is to be determined. 3 The house is maintained at 22°C at all times. 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: $Tmi + 1 = \tau (T1i + T1i) + (1 - 2\tau)T0i + \tau q \& 1 (\Delta x) 2 / k Node 2$ (interior): T2i +1 = τ (T1i + T3i) + (1 - 2τ)T2i + $\tau g \& 2$ (Δx) 2 / k Node 3 (interior): T3i + 1 = τ (T2i + T4i) + (1 - 2τ)T3i + $\tau g \& 3$ (Δx) 2 / k Node 6 (convection): $q \& b + k i + 1 i T3i - T4i \Delta x T4 - T4 + <math>\tau g \& 4$ (Δx) 2 / k = $\rho C \Delta x 2 \Delta t$. Analysis (a) The amount of heat the chip dissipates during an 8-hour period is Q = Q \& \Delta t = (3 W)(8 h) = 24 Wh = 0 0.024 kWh Logic chip Q& = 3 W (b) The heat flux on the surface of the chip is Q\& 3W = 37.5 \text{ W/in } 2 \text{ q\& } s = As 0.08 \text{ in } 21-14 \text{ The filament of a } 150 \text{ W} incandescent lamp is 5 cm long and has a diameter of 0.5 mm. Iron Analysis (a) The amount of heat the iron dissipates during a 2-h period is 1200 W Q = Q& $\Delta t = (1.2 \text{ kW})(2 \text{ h}) = 2.4 \text{ kWh}$ (b) The heat flux on the surface of the iron base is Q& base = (0.9)(1200 W) = 1080 W Q = base = 72,000 W / m 2 Abase 0.015 m 2 (c) The cost of electricity consumed during this period is Cost of electricity and the cost of e spaced 0.12 W logic chips. 3 The apparatus possesses thermal symmetry. Then individual resistances are A = $(1.2 \text{ m}) \times (2 \text{ m}) = 2.4 \text{ m} 2 1 1 = 0.0417 \text{ °C/W } 2 \text{ h} 1 \text{ A} (10 \text{ W/m} \cdot \text{°C})(2.4 \text{ m} 2) \text{ I} = R \text{ conv}, 1 = R \text{ rad} = 1 \text{ coA}(\text{Ts} + \text{Tsurr} 2) (\text{Ts} + \text{Tsurr} 2) (\text{Ts} + \text{Tsurr} 2) (\text{Ts} + \text{Tsurr} 2) (1 \text{ s} + \text{Tsurr$ 0.0810 °C/W Ri T $\infty 1$ W/m 2 .K 4)(2.4 m 2) [288 2 + 278 2] [288 + 278]K 3 R1 Rrad R3 1 Ro T $\infty 2$ 1 1 = 0.0167 °C/W h2 A (25 W/m 2 . C)(2.4 m 2) = Rconv, 2 = Rtotal = 0.1426 °C/W The steady rate of heat transfer through window glass then becomes T -T [24 - (tions 1 When treating hot dog as a finite cylinder, heat conduction in the hot dog is two-dimensional, and thus the temperature varies in both the axial x- and the radial rdirections. Then, Q& insulated = Q& side +bottom (-5) C O& = $\infty 1 \propto 2 = 203$ W Rtotal 01426 . Assume $U \propto top = 112$ 2/a.t=0.18.Ar = r2: kr = r1: -k C1 q& r = q& s → C1 = s 2 r2 k ((C1 k) k) q& sr2 ||C1 = T∞ - || ln r1 - | = h[T∞ - (C1 ln r1 + C2)] → C2 = T∞ - || ln r1 - r1 hr1 / hr1 |/ k (Substituting C1 and C2 into the general solution, the variation of temperature is determined to be (r (k) k) q& sr2 || ||C1 = T∞ + || ln + ||C1 = T∞ + || ln r - ln r1 + T(r) = h[T∞ - (C1 ln r1 + C2)] → C2 = T∞ - || ln r1 - r1 hr1 / hr1 |/ k (Substituting C1 and C2 into the general solution, the variation of temperature is determined to be (r (k) k) q& sr2 || ||C1 = T∞ + || ln + ||C1 = T∞ + || ln r - ln r1 + T(r) = C1 ln r + T ∞ - || ln r1 - hr1 / hr1 / k1 / (169.1 W/m 2)(0.04 m) (r) (r 14 W/m \cdot °C) = -10 + 0.483 || ln + 12.61 || = -10 °C + | ln + | r (30 W/m 2 \cdot °C)(0.037 m) |14 W/m \cdot °C / r1 / (12-32 Chapter 2 Heat Conduction Equation (c) The inner and outer surface temperatures are determined by direct substitution to be (r) Inner surface (r = r1): T (r1) = -10 + 0.483 $| \ln 1 + 12.61 | = -10 + 0.483 (0 + 12.61) = -3.91^{\circ} C \sqrt{r1} / (r - 0.483) | \ln 2 + 12.61 | = -10 + 0.483 | \ln 2 + 12.61 | = -10 + 0.483 | \ln 2 + 12.61 | = -10 + 0.483 | \ln 2 + 12.61 | = -3.87^{\circ} C$ Outer surface will experience three times smaller temperature drop compared to the inner surface. 2 Heat transfer is one-dimensional since the wall are uniform. The thermal resistance network of this problem consists of three resistances in series (contact, plate, and convection) which are determined to be Rcontact = Rplate = Rconvection 1 1 = $0.0227 \circ C / W 2$ hc Ac (49,000 W / m. The rate of heat generation per unit length of the wire is Q& gen Q& gen (3 × 3412.14 Btu/h) = $= 2.933 \times 10.8$ Btu/h.ft 3 g& = 2.2 V wire π ro L π (0.04 / 12 ft) (1 ft) Then the temperature difference between the centerline and the surface becomes Δ Tmax = g&ro 2 (2.933 × 10 8 Btu/h.ft 3) (0.04 / 12 ft) 2 = = 140.4 °F 4k 4(5.8 Btu/h.ft.°F) 2-88E Heat is generated uniformly in a resistance heater wire. Properties of air at this temperature and 1 atm are (Table A-15E) k = 0.01481 Btu/h.ft.°F v = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 2 Pr = 0.1697 × 10 ft /s -3 Pr = 0.1697 × 10 P 0.7290 Analysis The Reynolds number is V L [55 × 5280/3600) ft/s](20 ft) Re L = ∞ = 9.506 × 10 6 - 3 2 ν 0.1697 × 10 ft /s Air V ∞ = 55 mph T ∞ = 80°F Refrigeration truck L = 20 ft We assume the air flow over the entire outer surface to be turbulent. The dimensionless temperature for the surface of the plane walls with 2L = 40 cm is determined in part (a). We are interested in metabolic rate of the occupants of a building when we deal with heating and air conditioning because the metabolic rate represents the rate at which a body generates heat and dissipates it to the room. 2 Thermal properties, heat transfer coefficients, and the outdoor temperature are constant. 4 Radiation heat transfer is negligible. 2 The thermal properties of the soil are constant. Using the energy balance approach and taking the direction of all heat transfer to be towards the node, the finite difference equations for the nodes are obtained to be as follows: 11 T2 - T1 1/2 [a s q& s + h0 (T0 - T1)] = 0 (Ti - T1) + k + 2 2 l sin 45 Node 1: hi Node 2: hi l (Ti - T1) + k $T - T2 [T1 - T2]T4 - T2 + k + k] = 0 2 1 2 1 1 7 - T3 [\alpha s q & s + h0 (T0 - T3)] = 0 + k | 5 + 1 | sin 45 Node 4: hi Node 5: T4 + 2T3 + T6 - 4T5 = 0 Node 6: | T5 - T6 | 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 2 | 2 | 1 + k + k = 0 2 | 2 | 2 | 1 + k + k = 0 2 | 2 | 2 | 1 + k + k = 0$ m, $k = 0.6 \text{ W/m} \cdot ^{\circ}C$, $hi = 150 \text{ W/m} \cdot ^{\circ}C$, $Ti = 15^{\circ}C$, $ho = 30 \text{ W/m} \cdot ^{\circ}C$, $T0 = 25^{\circ}C$, $\alpha s = 0.7$, and $q\&s = 800 \text{ W/m} \cdot ^{\circ}C$, $Ti = 15^{\circ}C$, $ho = 30 \text{ W/m} \cdot ^{\circ}C$, $Ti = 15^{\circ}C$, $no = 30 \text{ W/m} \cdot ^{\circ}C$, $Ti = 15^{\circ}C$, Tibecomes Q = Q& cond $\Delta t = (4.368 \text{ kJ/s})(5 \times 3600 \text{
s}) = 78,620 \text{ kJ}$ If the thickness of the glass doubled to 1 cm, then the amount of heat transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer will go down by half to 39,310 kJ. 4 The Fourier number is $\tau > 0.2$ so that the determined whether the orange will freeze in 4 h in subfreezing temperatures. Dividing both sides by k and integrating twice give Energy: $0=k \partial 2T 2 2 T (y) = -\mu (y) |V| + C3 y + C 4 2k (L / Applying the boundary conditions T(0) = T1 and T(L) = T2 gives the temperature distribution to be T (y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - || L L 2$ (b) The temperature gradient is determined by differentiating T(y) with respect to y, y dT T2 - T1 μ V2 (= + | 1 - 2 | dy L 2kL (L) The location of maximum temperature is determined by setting dT/dy = 0 and solving for y, (T - T1) y dT T2 - T1 μ V2 (y = L| k 2 2 1 + | = + \rightarrow | 1 - 2 | = 0 | μ V2 | 2kL (dy L L) (6-7 Chapter 6 Fundamentals of Convection The maximum temperature is the value of temperature at this y, whose numeric value is $\left[\left(T - T 1 \right) (40 - 15)^{\circ}C 1 \right] + \left[y = L \right] (40 - 15)^{\circ}C 1 + \left[y = L \right] (0.580 \text{ N.s/m})(12 \text{ m/s}) \left[\left(1 - 15 \right)^{\circ}C 1 \right] + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y = L \right] (1 - 15)^{\circ}C 1 + \left[y$ 15)°C (0.58 N · s/m 2)(12 m/s) 2 (0.0003804 m) + 15°C + 0.0007 m 2(0.145 W/m · °C) (0.0003804 m) 2 | 0.0007 m - (0.0007 m) 2 | = 100.0°C (c) Heat flux at the plates is determined from the definition of heat flux, q& 0 = -k dT dy = -k y = 0 T2 - T1 T - T μ V 2 μ V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 W/m.°C) q& L = -k dT dy = $dT dy = -k y = L (40 - 15)^{\circ}C (0.58 \text{ N} \cdot \text{s/m 2})(12 \text{ m/s}) 2 (1 \text{ W}) 4 2 - | = -6.48 \times 10 \text{ W/m } 0.0007 \text{ m}) (1 \text{ N} \cdot \text{m/s}) T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot \text{C})(40 - 15)^{\circ}C (0.58 \text{ N} \cdot \text{s/m 2})(12 \text{ m/s}) 2 + 0.0007 \text{ m}) (1 \text{ W}) 4 2 | = 5.45 \times 10 \text{ W/m} \cdot 1 \text{ N m/s} | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 - k (1 - 2) = -k 2 1 + 2kL 2L L L = -(0.145 \text{ W/m} \cdot 1 \text{ N m/s}) | T2 - T1 \text{ T} - T \mu \text{V} 2 \mu \text{V} 2 + k (1 - 2) = -(1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1 - 2) (1$ about 72.5°C confirms our suspicion that viscous dissipation is very significant. 4 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (it will be verified). Using a computer, the solution at the upper corner node (node 3) is determined to be 441, 520, and 529°C at 2, 5, and 30 min, respectively. Analysis The characteristic length of the junction and the Biot number are Lc = Bi = V Asurface = $\pi D 3 / 6 D 0.0012 m = = 0.0002 m 6 6 \pi D 2 hLc$ (65 W / m 2 . 4 The edge effects of adjoining surfaces on heat transfer are to be considered. 4 The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the nitrogen inside, and thus thermal resistance of the tank and the internal convection resistance are negligible. The rate of heat generation per unit volume of the wire is Q& gen (3 × 3412.14 Btu/h) = = 2.933×10.8 Btu/h.ft 3 g& = 2 V wire π ro L π (0.04 / 12 ft) 2 (1 ft) Then the temperature difference between the centerline and the surface becomes Δ Tmax = g&ro 2 (2.933 × 10 8 Btu/h.ft 3)(0.04 / 12 ft) 2 = 181.0°F 4k 4(4.5 Btu/h.ft. 4-16 Chapter 4 Transient Heat Conduction in Large Plane Walls, Long Cylinders, and Spheres 4-26C A cylinder whose diameter is small relative to its length can be treated as an infinitely long cylinder. 1-2C (a) The driving force for heat transfer is the
temperature difference. F / Btu 2 hA (17.2 Btu / h.ft. 2 Thermal properties of water are constant. m) = 0.353 kg (4180 J / kg)(4180 J /We now repeat calculations after wrapping the can with 1-cm thick rubber insulation, except the top surface. 7-27a, f = 0.20. 2 Heat conduction in the apples is one-dimensional because of symmetry about the midpoint. Analysis The total rate of heat generation in a water layer of surface area A and thickness L at the top of the pond is determined by integration to be $G\& = \int V g\&dV = \int L x = 0 g\& 0 e - bx e - bx$ (Adx) = Ag& 0 (1 - e - bL) b 2-18 The rate of heat generation per unit volume in a stainless steel plate is given. Analysis It is given that D = 0.021 m, SL = ST = 0.05 m, and V = 3.8 m/s. This is parallel flow between two plates, and thus v = 0. Replacing the symmetry lines by insulation, and utilizing the mirror-image concept, the finite difference equations for the interior nodes can be written as Node 1 (interior): T1 = (120 + 120 + T2 + T3)/4 Node 2 (interior): T3 = (140 + 2T + T4)/4 = T2 Node 4 (interior): T4 = (2T2 + 140 + 2T3)/4 100 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 120 • 12 $100\ 120\ \cdot\ 12\ \cdot\ 120\ 140\ \cdot\ 34\ \cdot\ 140$ Solving the equations above simultaneously gives T1 = T2 = 122.9^{\circ}C\ T3 = T4 = 128.6^{\circ}C Insulated Discussion Note that taking advantage of symmetry simplified the problem greatly. Still air above ceiling 2. The connecting bars are serving as "thermal bridges." 3-123 Chapter 3 Steady Heat Conduction 3-164 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, 5 The local atmospheric pressure is 1 atm. Therefore, analytical solutions are limited to problems that are simple or can be simplified with reasonable approximations. Assumptions 1 The oranges are spherical in shape with a radius of r0 = 1.25 in = 0.1042 ft. m) + 2 π (0.06 m) 2 4 = 0.0292 m2 Q& bare, ave = ho A(Tair - Tcan, ave) = (10 W/m 2 .°C)(0.0292 m 2)(25 - 6.5)°C = 5.40 W The amount of heat that must be supplied to the drink to raise its temperature to 10 °C is m = $\rho V = \rho \pi r 2 L = (1000 \text{ kg} / \text{m 3}) \pi (0.03 \text{ m}) 2 (0125 .°F / Btu 4)$ Rtotal, new Q& new 2 × 10 Btu / h 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 Rtotal, w /lime Rtotal, new HX T \propto 1 ln(r1 / ri) ln(0.5 / 0.49) = = = 0.00189 h^{\circ} F / Btu . In the case of laminar flow, the effect of surface roughness on the friction factor is negligible The initial rate of heat transfer from an orange, the temperature gradient at the orange surface, and the value of the Nusselt number are to be determined. Analysis (a) The Reynolds number, the heat transfer coefficient, and the initial rate of heat transfer from an orange are As = $\pi D 2 = \pi (0.07 \text{ m}) 2 = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} T \infty = 5^{\circ} \text{C V D} (0.5 \text{ m/s})(0.07 \text{ m}) \text{ Re} = \infty = 23.73 \text{ W/m} 2 \cdot ^{\circ} \text{C} \text{ D} 0.07 \text{ m} \text{ Q} \text{ are } \text{As} = \pi D 2 = \pi (0.07 \text{ m}) 2 = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} \text{ T} \infty = 5^{\circ} \text{C V D} (0.5 \text{ m/s})(0.07 \text{ m}) \text{ Re} = \infty = 23.73 \text{ W/m} 2 \cdot ^{\circ} \text{C} \text{ D} 0.07 \text{ m} \text{ Q} \text{ are } \text{As} = \pi D 2 = \pi (0.07 \text{ m}) 2 = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} \text{ T} \infty = 5^{\circ} \text{C V D} (0.5 \text{ m/s})(0.07 \text{ m}) \text{ Re} = \infty = 23.73 \text{ W/m} 2 \cdot ^{\circ} \text{C} \text{ D} 0.07 \text{ m} \text{ Q} \text{ are } \text{As} = \pi D 2 = \pi (0.07 \text{ m}) 2 = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} \text{ T} \infty = 23.73 \text{ W/m} 2 \cdot ^{\circ} \text{C} \text{ D} 0.07 \text{ m} \text{ Q} \text{ are } \text{As} = \pi D 2 = \pi (0.07 \text{ m}) 2 = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} \text{ T} \infty = 23.73 \text{ W/m} 2 \cdot ^{\circ} \text{C} \text{ D} 0.07 \text{ m} \text{ Q} \text{ are } \text{As} = \pi D 2 = \pi (0.07 \text{ m}) 2 = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} \text{ T} \infty = 23.73 \text{ W/m} 2 \cdot ^{\circ} \text{C} \text{ D} 0.07 \text{ m} \text{ Q} \text{ are } \text{As} = \pi D 2 = \pi (0.07 \text{ m}) 2 = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} \text{ T} \infty = 23.73 \text{ W/m} 2 \cdot ^{\circ} \text{C} \text{ D} 0.07 \text{ m} \text{ Q} \text{ are } \text{ Air } V \infty = 0.01539 \text{ m} 2 \text{ Air } V \infty = 0.5 \text{ m/s} \text{ Air } V \infty = 0.5 \text{ m/s} \text{ Air } V \infty = 0.5 \text{ m/s} \text{ m} 2 \text{ m$ temperature gradient at the orange surface is determined from (∂T) g& conv = g& cond = -k = h(Ts - T\infty) | $\partial r / r = R \partial T \partial r = -r = Rh(Ts - T\infty) | (23.73 \text{ W/m } 2.^{\circ}C)(0.07 \text{ m}) = 68.11 0.02439 \text{ W/m.}^{\circ}C \text{ k } 6-3 \text{ Orange Ti} = 15^{\circ}C \text{ Chapter } 6$ Fundamentals of Convection Velocity and Thermal Boundary Layers 6-12C Viscosity is a measure of the "stickiness" or "resistance to deformation" of a fluid. 5 The thermal properties of the spoon are constant. Analysis We consider 1 m high and 1 m wide portion of the wall which is representative of entire wall. Note that the height of the short cylinder represents the half thickness of the infinite plane wall where the bottom surface of the short cylinder is adiabatic. $\sqrt{Assumptions 1 Heat transfer through the body}$ is given to be steady and two-dimensional. 2 Potato is spherical in shape. The efficiency of these circular fins is, from the efficiency curve, | | L = (D2 - D1)/2 = (0.06 - 0.05) $/2 = 0.005 \text{ m} | r^2 + (t/2) 0.03 + (0.001/2) | = 1.22 | \eta \text{ fin} = 0.97 0.025 \text{ r1} | 20 0.001 | 40 \text{ W/m C t} h (| = = 0.005 + 0. Noting that <math>\tau = \alpha t / L^2$ and assuming that $\tau > 0.2$ in all dimensions so that the oneterm approximate solution for transient heat conduction is applicable, the product solution method can be written for this problem as $\theta(r^2 + (t/2) 0.03 + (0.001/2) | r^2 + (t/2) 0.03 + (t/2)$ increases the energy consumption for cooling in summer by replacing the cold indoors air by the warm outdoors air. The cooling process is to be determined. Also, if the surface area of the fin tip is very small compared to the total surface area of the fin heat transfer from the tip can again be neglected. Properties The density and kinematic viscosity of air at 1 atm and 25°C are $\rho = 1.184$ kg/m3 and $\nu = 1.562 \times 10-5$ m2/s (Table A-15). m2) + 0.0364 m = 8.4 mm 3-38 Chapter 3 Steady Heat Conduction 3-61 A kiln is made of 20 cm thick concrete walls and ceiling. Assumptions 1 Heat transfer is one-dimensional since the exposed surface of the wall large relative to its thickness. \circ F)(1 ft) Ri Ti T1 Current Case: R1 = Rpipe = Rinsulation = Rpipe Rinsulation Ro To T2 T3 ln(r3 / r2) ln(3 / 2) = = 3.227 h. 2 Heat conduction in the chickens is one-dimensional in the radial direction because of symmetry about the midpoint. The energy balance on the system can be expressed as E - Eout 1in 424 3 = Net energy transfer by heat, work, and mass ΔE system 12 4 4 3 Change in internal, kinetic, potential, etc. Then the equivalent thermal resistance will be Reqv = 1/ (heqv A). The average friction coefficient of the airfoil is determined from the modified Reynolds analogy to be Cf = $2(1.335 \text{ W/m } 2 \cdot ^{\circ}\text{C})(0.7296) 2/32hPr 2/3 = 0.000227 \rho V C p (1.184 \text{ kg/m } 3)(8 \text{ m/s})(1007 \text{ J/kg} \cdot ^{\circ}\text{C}) 6-51 \text{ A metallic airfoil is subjected to air flow. Analysis The relation for time period for a lumped system to reach the average temperature (Ti + T_{\infty})/2 can be determined as Ti + T_{\infty} - T_{\infty} T - T_{\infty} T - T_{\infty} T + T_{\infty} - T_$ $e - bt Ti - T \infty Ti - T \infty 2(Ti - T \infty) 2 T \infty ln 2 0.693 - bt = -ln 2 \rightarrow t = = b b Ti 4-2$ Chapter 4 Transient
Heat Conduction 4-14 The temperature of a gas stream is to be measured by a thermocouple. The rate of heat loss from this man by convection in still air at 20°C, in windy air, and the wind-chill factor are to be determined. Analysis (a) The heat transfer surface area and the heat flux on the surface of the filament are As = $\pi DL = \pi (0.05 \text{ cm})(5 \text{ cm}) = 0.755 \text{ cm} 2 \text{ q} \text{ s} = 201.1 \text{ cm} 2 \text{ m} 2 \text{ m} (8 \text{ cm}) 2 = 201.1 \text{ cm} 2 \text{ m} (8 \text{ cm}) 2 = 201.1 \text{ cm} 2 \text{ m} (8 \text{ cm}) 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 200.1 \text{ cm} 2 \text{ m} (8 \text{ cm}) 2 = 201.1 \text{ cm} 2 \text{ m} (8 \text{ cm}) 2 = 201.1 \text{ cm} 2 \text{ m} (8 \text{ cm}) 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 200.1 \text{ cm} 2 \text{ m} (8 \text{ cm}) 2 = 201.1 \text{ cm} 2 \text{ m} (8 \text{ cm}) 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 1.91 \text{ W/cm} 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 1.91 \text{ W/cm} 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 1.91 \text{ W/cm} 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 1.91 \text{ W/cm} 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 1.91 \text{ W/cm} 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 1.91 \text{ W/cm} 2 = 7500 \text{ W/m} 2 \text{ As} 0.785 \text{ cm} 2 \text{ q} \text{ s} = 1.91 \text{ W/cm} 2 \text{ s} = 1.91$ cm 2 (c) The amount and cost of electrical energy consumed during a one-year period is Electricity Consumption = $Q\&\Delta t = (015 . The rate of heat loss through the wall is to be determined.$ Still air, reflective horizontal surface facing up 2. 2 Heat transfer is one-dimensional since this two-layer heat transfer problem possesses symmetry about the center line and involves no change in the axial direction, and thus T = T(r). The amounts of ice or cold water that needs to be added to the water are to be determined. 4-5C The temperature of the surrounding medium asymptotically, and thus it changes rapidly at the beginning, but slowly later on. The rate of heat loss from the hot water and the average velocity of the water in the pipe as it passes through the basement are to be determined. The amount of energy that needs to be transferred to the water to raise its temperature to 140°F is to be determined. Assumptions 1 The glass container is cylindrical in shape with a radius of r0 = 3 cm. 3-113 Chapter 3 Steady Heat Conduction 3-154 A cylindrical tank filled with liquid propane at 1 atm is exposed to convection and radiation. m) 2 - (0.05 m) 2] + 2 π (01. at x = L: k k k Substituting the C1 and C2 relations into Eq. (1) and rearranging give g& 0 L2 $[4(e - 0.5 - e - 0.5 \times / L) + (2 - \times / L)]$ k which is the desired solution for the temperature distribution in the wall as a function of x. 2-11C In heat conduction analysis, the conversion of electrical, or nuclear energy into heat (or thermal) energy in solids is called heat generation. 3 Heat is generated uniformly. 4 The outer surface at x = L is subjected to convection and radiation while the inner surface at x = 0 is subjected to convection only. Analysis Noting that thermal contact resistance is determined to be Rc = $1.1 = 5.556 \times 10 - 5 \text{ m } 2$. °C/W hc 18,000 W/m 2. °C L where L k is the thickness of the plate and k is the thermal conductivity. 2 Heat transfer through the shell is onedimensional. Also, Cp = 1.007 kJ/kg·K for air at room temperature (Table A-15) and Cv = Cp - R = 0.720 kJ/kg·K. A fluid whose density is practically independent of pressure (such as a liquid) is called an incompressible fluid. The total rate of heat loss from the collector, the collector efficiency, and the temperature rise of water as it flows through the collector are to be determined. Therefore, the time needed for this heater to supply 853 kJ of = = = 710 s = 11.8 min & Rate of energy transfer E transfer 1.2 kJ/s Discussion In reality, it will take longer to

accomplish this heating process since some heat loss is inevitable during the heating process. The rate of radiation heat loss from the person is to be determined. Analysis We take the room as the system. 2 Thermal conductivity varies quadratically. Table 3-4 reveals that HS5030 in both The thermal resistance of the heat sink must be below 15 horizontal and vertical positions, HS6071 in vertical positions, and HS6115 in both horizontal and vertical positions can be selected. These simplifications are known as the boundary layer approximations. Discussion The rate of heat transfer can also be determined by calculating the heat loss from the side surfaces using the heat conduction relation. F (3.2808 ft) 2 (1.8 ° F) which is the desired conversion factor. Its value is A = 018. The value of the convection heat transfer coefficient is A = 018 are constant. 8-15C The heat flux will be higher near the inlet because the heat transfer coefficient is A = 018 are constant. highest at the tube inlet where the thickness of thermal boundary layer is zero, and decreases gradually to the fully developed value. 4 The temperature of the tank and the internal convection resistance are negligible. The length of time the block should be kept in the furnace and the amount of heat transfer to the block are to be determined. The melting temperature of the ice and the heat of fusion of ice at atmospheric pressure are 32°F and 143.5 Btu/lbm, respectively Assumptions 1 Thermal properties of the ice and water are constant. We evaluate the air properties at the assumed mean temperature of 35°C (will be checked later) and 1 atm (Table A-15): k = 0.02625 W/m-K $\rho = 1.145$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 $\mu = 1.895 \times 10$ kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10 kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10 kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg-K Pr = 0.7268 \mu = 1.895 \times 10^{10} kg/m³ Cp = 1.007 kJ/kg $U = \Sigma$ farea, iUi = 0.84×1.054+0.16×0.582 Overall unit thermal resistance of the wall is R = 1.02 m2.°C/W and the overall U-factor is U = 0.978 W/m2.°C. Therefore using the proper relation in turbulent flow for Nusselt number, the average heat transfer coefficient is determined to be hL Nu = $0.037 \text{ Re L } 0.8 \text{ Pr } 1/3 = 0.037(9.506 \times 10.6) \text{ Det} / h = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ \text{F h} = \text{Nu} = (1.273 \times 10.4 \text{ k k } 0.01481 \text{ Btu/h.ft.}^\circ$ $Q_{k} = = 18,000$ Btu / h 2 The total heat transfer surface area and the average surface temperature of the refrigeration compartment of the truck are determined from A = 2 (20 ft)(8 ft) + (9 ft)(8 f External Forced Convection 7-27 Solar radiation is incident on the glass cover of a solar collector. ho, To 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. Analysis (a) We first take the air in the room as the system. Assumptions 1 Heat transfer through the window is steady since the surface temperatures remain constant at the specified values. 4 The surface of the plate is smooth. Assumptions 1 Heat conduction in the column is one-dimensional since it is long and it has thermal symmetry about the center line. 3 The surrounding surfaces are at the same temperature as the indoor air temperature. 3-135C The effective emissivity for a plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of one surfaces across the air space. 3-86C No. 3-87C For a cylindrical pipe, and the relation plane-parallel air space is the "equivalent" emissivity of the relation plane-parallel air space is the "equivalent" emissivity of the relation plane-parallel air space is the "equivalent" emissivity of the relation plane-parallel air space is the "equivalent" emissivity of the relation plane-parallel air space is the "equivalent" emissivity of the relation plane-parallel air space is the "equivalent" emissivity the critical radius of insulation is defined as rcr = k / h. 8-16C In the fully developed region of flow in a circular tube, the velocity profile may. Insulation thickness 0 cm 1 cm 5 cm 10 cm 12 cm 13 cm 14 cm 15 cm Rate of heat loss W 133,600 15,021 3301 1671 1521 1396 1289 1198 1119 Cost of heat lost \$/yr 12,157 1367 300 152 138 127 117 109 102 Cost savings \$/yr 0 10,790 11,850 12,005 12,019 12,030 12,040 12,048 12,055 Insulation that will pay for itself in one year is the one whose thickness is 14 cm. atm 101.325 kPa The kinematic viscosity at this atmospheric pressure is 18.8 kPa V ∞ = 900 km/h T ∞ =
-55.4°C ν = (1.106 × 10 -5 m 2/s)/(0.1855 = 5.961 × 10 m/s The Nusselt number relation for a cylinder of elliptical cross-section is limited to Re < 15,000, and the relation below is not really applicable in this case. ° C)(403.7 m2) Q& = total (T1 - T2) = (15 - 3)° C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.54 w L 15°C L = 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.54 w L 15°C L = 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.54 w L 15°C L = 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.54 w L 15°C L = 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.54 w L 15°C L = 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.54 w L 15°C L = 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.54 w L 15°C L = 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × edges = 4 × 0.15L + 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × 0.15(0.2 m) C = 18,167 W L 0.2 m The heat transfer rate through the edges can be determined using the shape factor relations in Table 3-5, S corners + 4 × 0.15(0.2 m) C = 18,107 W L 0.2 m Th m) + 4 × $0.54(12 \text{ m}) = 26.04 \text{ m} \& \text{Q corners} + \text{edges} = \text{S corners} + \text{edges} \& (\text{T1} - \text{T2}) = (26.04 \text{ m})(0.75 \text{ W/m}. 6-18 \text{C The Prandtl number Pr} = \nu / \alpha$ is a measure of the velocity boundary layer) and the diffusivity of heat (and thus the development of the thermal boundary layer). Properties The thermal conductivities are given to be k = 8.7 Btu/h·ft.°F. Properties The thermal conductivities are given to be k = 15 W/m.°C for fiberglass insulation. Substituting the given quantities, the maximum allowable the time step becomes 5-74 Chapter 5 Numerical Methods in Heat Conduction $\Delta t \leq (0.02 \text{ m}) 2 2(12.5 \times 10 - 6 \text{ m} 2/s)[1 + (35 \text{ W/m} 2 .°C)(0.02 \text{ m}) / (28 \text{ W/m}.°C)] = 15.6 \text{ s}$ Therefore, any time step less than 15.5 s can be used to solve this problem. Analysis (a) The representative surface area is A = (7.5 / 12)(7.5 / 12) = 0.3906 \text{ ft } 2. Properties Assuming a film temperature of 40°C, the properties of air are transfer from a spherical copper ball. The unknown nodal temperatures are to be determined with the finite difference method. 1-33 It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 7000 kJ/h. Assumptions 1 Steady operating conditions exist when measurements are taken. Applying the boundary conditions give x = 0: $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 $-kC1 = q\& 0 \rightarrow C1 = -x = L$: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = -x = L: T (L) = C1 L \rightarrow C2 = T2 -kC1 = -x = -x = L: T (L) = C1 L \rightarrow) + T2 x + T2 + 0 = 0 k k k (50,000 W/m 2)(0.006 - x) m = + 85°C 20 W/m · °C = 2500(0.006 - x) + 85 T (x) = - (c) The temperature at x = 0 (the inner surface temperature is higher than the exposed surface temperature at x = 0 (the inner surface of the plate) is T (0) = 2500(0.006 - x) + 85 T (x) = - (c) The temperature at x = 0 (the inner surface temperature at x = 0) (the inner surface temperature at the outer surface areas in the calculation of all thermal resistances with little loss in accuracy. Properties The thermal conductivity, density, and Cp = 0.092 Btu/lbm.°F. 1-53 32°C ϵ =0.9 Qconv Chapter 1 Basics of Heat Transfer 1-101 Two large plates at specified temperatures are held parallel to each other. Analysis Using the energy balance approach and taking the direction of all heat transfers to be towards the node under No heat generation, the finite difference formulations become 800 W/m2 30°C Left boundary node: T0 = 30 Δx T7 - T8 Δx T7 - T8 reduces to $\partial u \partial v \partial u + = 0 \rightarrow Continuity: = 0 \rightarrow u = u(y) V \partial x \partial x \partial y$ Therefore, the x-component of velocity does not change T0 in the flow direction (i.e., the velocity profile remains unchanged). × 10 - 6 m2 / s)(10 min × 60 s / min) (0.025 m) 2 = 1104 > 0.2 . 3-17 The two surfaces of a wall are maintained at specified temperatures. 1-25 Two identical cars have a head-on collusion on a road, and come to a complete rest after the crash. The relations for the maximum temperature of fluid, the location where it occurs, and heat flux at the upper plate are to be obtained. Properties of air at 1 atm pressure and the free stream temperature of fluid, the location where it occurs, and heat flux at the upper plate are to be obtained. Properties of air at 1 atm pressure and the free stream temperature of fluid, the location where it occurs, and heat flux at the upper plate are to be obtained. $= 1.426 \times 10^{-5} \text{ m } 2/\text{s} \ \mu \ \infty = 1.778 \times 10^{-5} \text{ m } 2/\text{s} \ \mu \ \infty = 35 \text{ km/h } T_{\infty} = 35 \text{ km/h } T_{\infty} = 35 \text{ km/h } T_{\infty} = 10^{\circ}\text{C} \text{ kg/m.s}$ Head Q = 21 W Pr = 0.7336 D = 0.3 m Analysis The Reynolds number is V D [(35 × 100/3600) m/s](0.3 m) Re = $\infty = = 2.045 \times 10^{-5} \text{ m } 2/\text{s} \text{ m} \text{ m} \text{ s} = 0.3 \text{ m} \text{ m} \text{ m} \text{ m} \text{ s}$ (0.3 m) Re = $\infty = = 2.045 \times 10^{-5} \text{ m } 2/\text{s} \text{ m} \text{$ number is Nu = [] (μ hD = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 || ∞ k (μ s [= 2 + 0.4(2.045 × 10) 5 0.5) || / 1/4 + 0.06(2.045 × 10 - 5 | 1.802 × 10 - 5 | 1.802 × 10 - 5 | 1.802 × 10 - 5 | 1.802 × 10 - 5 | 1.802 × 10 - 5 | 1.802 × 10 - 5 | 1.802 × 10 - 5 | 1.4) || / = 344.7 The heat transfer coefficient is k 0.02439 W/m.°C h = Nu = (344.7) = 28.02 W/m 2 .°C D 0.3 m Then the surface temperature of the head is determined to be As = $\pi D 2 = \pi (0.3 \text{ m}) 2 = 0.2827 \text{ m} 2$ (84/4) W Q& \rightarrow Ts = T ∞ + = 10 °C + = 12.7 °C Q& = hAs (Ts - T ∞) hAs (28.02 W/m 2.°C)(0.2827 m 2) 7-34 Chapter 7 External Forced Convection 7-45 The flow of a fluid across an isothermal cylinder is considered. Therefore, the temperature or heat flux remains unchanged with time during steady heat transfer through a medium at any location although both quantities may vary from one location to another. The boundary condition on the inner surface of the r1 2-12 r2 Chapter 2 Heat Conduction Equation container for steady one-dimensional conduction is to be expressed for the following cases: (a) Specified temperature of 50°C: T (r1) = 50° C (b) Specified heat flux of 30 W/m2 towards the center: k dT (r1) = 30 W / m 2 dr (c) Convection to a medium at T ∞ with a heat transfer coefficient of h: k dT (r1) = h[T (r1) - T ∞] dr 2-41 Heat is generated in a long wire of
radius r0 covered with a plastic insulation layer at a constant rate of g& 0. × 10 m / s, $\Delta x = 0.25$ m, and $\Delta t = 15$ min = 900 s. The average convection heat transfer coefficient and the cooling time are to be determined. At the same time, evacuating the space between the layers. Derivation increases the order of a derivative by one, integration reduces it by one. The time the potato is baked in the oven and the final equilibrium temperature of the potato after it is wrapped are to be determined. 4 Thermal conductivities are constant. The energy balance for this system can be expressed as [W& E in - E out = ΔE sys e, in] + m& hot C (Tin - Tout) Δt = m tank C (T2 - T1) where Tout is the average temperature of hot water leaving the tank: $(80+70)/2=70^{\circ}C$ and m tank = $\rho V = (977.6 \text{ kg/m 3})(0.06 \text{ m 3}) = 58.656 \text{ kg}/(4.18 \text{ kJ/kg.}^{\circ}C)(60 - 80)^{\circ}C$ m& hot = 0.0565 kg/s To determine the average temperature of the mixture, an energy balance on the mixing section can be expressed as E& = E& in out m& hot CThot + m& cold CTcold = (m& hot + m& cold)CT mixture (0.0565 kg/s)(4.18 kJ/kg.°C)(70°C) + (0.06 kg/s)(70°C) + Properties The density, specific heat, and thermal diffusivity of the aluminum alloy plate are given to be $\rho = 2770 \text{ kg/m}$, Cp = 875 kJ/kg.°C, and $\alpha = 7.3 \times 10^{-5} \text{ m}^2/s$. Analysis The thermal resistance between the transistor R attached to the sink and the ambient air is determined to be Ts Ts T - Ts (80 - 35)° C Δ T Q& = \rightarrow Rcase - ambient = transistor = $1.5 \circ C / W$ Rcase - ambient 30 W Q&. 3 The thermal properties of the chicken are constant. ft 2 T ∞ 1 T ∞ 2 Ao = π Do L = π (0.6 / 12 ft)(1 ft) = 0157. Properties The thermal properties of the chicken are constant. ft 2 T ∞ 1 T ∞ 2 Ao = π Do L = π (0.6 / 12 ft)(1 ft) = 0157. negligible. ° F)(200 ft 2) = 3108 Btu / h 1 ft L or 0.911 kW since 1 kW = 3412 Btu/h. Assumptions 1 Heat transfer through the walls is steady since the surface temperatures of the walls is steady since the surface temperatures of the walls is steady since the surface temperatures of the walls is steady since the surface temperatures of the walls remain constant at the specified values during the time period considered. 2 The thermal conductivity is given to be constant. Nodes 1, 2, and 3 are interior nodes, and thus for them we can use the general explicit finite difference relation expressed as Tmi -1 - 2Tmi $+1 = \tau$ (Tmi $-1 = \tau$ half volume element about node 4 and taking the direction of all heat transfers to be towards the node under consideration: Node 2 (interior) : N $+ (1 - 2\tau)T^{2i} + \tau 0 k g (\Delta x 2 T^{3i} + 1 = \tau (T^{2i} + T^{4i}) + (1 - 2\tau)T^{3i} + \tau 0 k i i T - T^{4} \Delta x \Delta x T^{4i} + 1 - T^{4i} (T^{\infty} - T^{4i}) + (1 - 2\tau)T^{0i} + \tau g (\Delta x)^{2} h (\Delta x)^{2}$ 20°C, where and $\alpha = 12.5 \times 10$ m/s. 7-8C The friction drag coefficient is independent of surface roughness in laminar flow, but is a strong function of surface roughness in turbulent flow due to surface roughness in turbulent flow. of all heat transfers to be towards the node under consideration, the explicit transient finite difference formulations become Left boundary node: $0 1 2 3 4 5 i i i + 1 i - T - T T T \Delta \Delta x x i 5 5 4 - (T5i) 4 + g & 5i A = pA$ εσA[(Tsurr C 5 Δx 2 2 Δt 5-68 Chapter 5 Numerical Methods in Heat Conduction, and heat flux at the left (node 0) and specified temperature at the right boundary (node 4). Wall Properties The thermal properties of the concrete are given to be k = 0.64 Btu/h.ft.°F and $\alpha = 0.023$ ft2/h. 4-96C A refrigerated shipping dock is a refrigerated space where the orders are assembled and shipped out. Therefore, there is no need to repeat the calculations. Also, we can replace the symmetry lines by insulation and 7 8 9 utilize the mirror-image concept for the interior nodes. Mineral fiber insulation, 140 mm 4b. Assumptions 1 The bearings are spherical in shape with a radius of r0 = 0.6 cm. Using the proper relation for Nusselt number, the average heat transfer rate are determined to be hL Nu = = (0.037 Re L 0.8 - 871) Pr 1 / 3 = [0.037(1.931 × 10 6) 0.8 - 871](0.7166)1 / 3 = 2757 k l = 0.02917 W/m.°C h = Nu = (2757) = 10.05 W/m 2.°C L 8m As = wL = (2.5 m)(8 m) = 20 m 2 Q& = hA (T - T) = (10.05 W/m 2.°C)(20 m 2)(120 - 30)°C = 18,096 W = 18.10 kW s ∞ s (b) If the air flows parallel to the 2.5 m side, the Reynolds number is V L (6 m/s)(2.5 m) Re L = ∞ = = 6.034 × 10 5 v 2.486 × 10 - 5 m 2/s which is greater than the critical Reynolds number. Bicyl = Air T ∞ = 16°C hr0 (9 W/m.°C) (2 z r 2L=180 cm Human body $\rightarrow \lambda 1$ = 1.6052 and A1 = 1.3408 Ti = 36°C Noting that $\tau = \alpha t / L$ for the plane wall and $\tau = \alpha t / r0$ for cylinder and J0(1.6052)=0.4524 from Table 4-2, and assuming that $\tau > 0.2$ in all dimensions so that the one-term approximate solution for transient heat conduction is applicable, the product solution method can be written for this problem as $\theta(0, r0, t)$ block = $\theta(0, t)$ wall $\theta(r0, t)$ conduction is applicable, the product solution method can be written for this problem as $\theta(0, r0, t)$ block = $\theta(0, t)$ wall $\theta(r0, t)$ conduction is applicable, the product solution method can be written for this problem as $\theta(0, r0, t)$ block = $\theta(0, t)$ wall $\theta(r0, t)$ conduction is applicable. R1 3-10 R2 R3 Chapter 3 Steady Heat Conduction 3-29 The roof of a house with a gas furnace consists of 3-cm thick concrete that is losing heat to the outdoors by radiation and convection. The time it takes for the valve temperatures and the maximum heat transfer are to be determined. Taking x = 0 at the surface of a house with a gas furnace consists of 3-cm thick concrete that is losing heat to the outdoors by radiation and convection. the bearing, the boundary conditions are u(0) = 0 and u(L) = V, and applying them gives the velocity distribution to be u(y) = y V L The plates are isothermal and there is no change in the flow direction, and thus the temperature depends on y only, T = T(y). The local Nusselt number is hx Nu x = 0.332 Re x 0.5 Pr 1 / 3 = $0.332(4.407 \times 10.4)$ 0.5 (0.7321)1/3 = 62.82 k The local heat transfer and friction coefficients are k 0.01433 Btu/h.ft 2.°F x 1 ft 0.664 0.664 C f, x = = 0.00316 Re0.5 (4.407 × 104)0.5 0.012 0.010 0.008 2 0.006 hx 1 0.5 0 0 0.004 0.002 C f, x 2 4 6 x [ft] 7-7 8 0 10 C f x h x [Btu/h.ft -F] We repeat calculations for all 1-ft intervals. Analysis The thermal resistances through the wall without windows are A = (12 ft)(40 ft) = 480 m 2 Wall 1 1 Ri = = 0.0010417 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.0010417 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q& 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 T1 = 0.00052 h · °F/Btu A (480 m 2) Q & 1 1 1 T1 = 0.00052 h · ° $0.00052 = 0.0411417 \text{ h} \circ \text{F/Btu Ro} = \text{Rtotal} - \text{Awindows} = 480 - 60 = 420 \text{ ft } 2 \text{ R} \text{ vall} = 0.04524 \text{ h} \circ \text{F/Btu} = 0.04524 \text{ h} \circ \text{F/Btu Ro} = 4.0257716 \text{ h} \circ \text{F/Btu Ro} = 0.0007716 \text{ h} \circ \text{F/Btu Ro} = 4.0257716 \text{ h} \circ \text{F/Btu Ro} = 0.0007716 \text{ h} \circ \text{F/Btu Ro} = 0.0007716 \text{ h} \circ \text{F/Btu Ro} = 0.04524 \text{ h} \circ \text{F/Btu Ro} = 0.0007716 \text{ h} \circ \text{F/Btu Ro} = 0.04524 \text{ h} \circ \text{F/Btu Ro} = 0.045$ ft 2) Rglass Ri 1 1 1 1 1 = + = + \rightarrow Reqv = 0.00076 · h °F/Btu Reqv R glass R wall 0.0007716 0.04524 Rtotal , 2 = Ri + Reqv + Ro = 0.001047 + 0.00052 = 0.002327 h · °F/Btu Then the ratio of the heat transfer through the walls with and without windows becomes Q& total , 2 AT / Rtotal , 2 Rtotal total ,1 ΔT / Rtotal ,1 Rtotal ,2 3-15 Rwall Ro Chapter 3 Steady Heat Conduction 3-34 Two of the walls of a house have no windows. Thus we have combined laminar and turbulent flow. This is because the heat transfer to an ideal gas is mCpΔT at constant pressure and mCpΔT at constant volume, and Cp is always
greater than Cv. 1-12 A cylindrical resistor on a circuit board dissipates 0.6 W of power. Properties The thermal properties of the granite are k = 2.5 W/m.°C and $\alpha = 1.15 \times 10-6$ m2/s. 1-53C A blackbody is an idealized body which emits the maximum amount of radiation at a given temperature and which absorbs all the radiation incident on it. The rate of heat transfer is higher in convection because of fluid motion. 2 The thermal properties of the bearings are constant. In a solid we can have only conduction. R 4 . Analysis (a) The amount of heat this transistor dissipates during a 24-hour period is Air, $Q = Q\& \Delta t = (0.2 \text{ W})(24 \text{ h}) = 4.8 \text{ Wh} = 0.0048 \text{ kWh} 30^{\circ}\text{C}$ (b) The thermal properties of the bearings are constant. In a solid we can have only conduction. R 4 . Analysis (a) The amount of heat this transistor dissipates during a 24-hour period is Air, $Q = Q\& \Delta t = (0.2 \text{ W})(24 \text{ h}) = 4.8 \text{ Wh} = 0.0048 \text{ kWh} 30^{\circ}\text{C}$ (b) The thermal properties of the bearings are constant. heat flux on the surface of the transistor is $\pi D 2 + \pi DL 4 \pi (0.005 \text{ m}) 2 = 2 + \pi (0.005 \text{ m}) (0.004 \text{ m}) = 0.0001021 \text{ m} 2 4 \text{ Q} \& 0$. The minimum burial depth at a particular location is to be determined. 2 The thermal properties of the column are constant. 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 100°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number and the time period become θ 0, sph = 2 2 T0 - T \propto 60 - 100 = A1e - λ 1 τ \rightarrow = (1.9925)e -(3.0533) τ \rightarrow τ = 0.1633 Ti - T \propto 8 - 100 which is somewhat below the value of 0.2. Therefore, the oneterm approximate solution (or the transient temperatures, the rate of heat transfer, and the fin efficiency are to be determined numerically using 6 equally spaced nodes. 2 Heat transfer through the walls and ceiling is one-dimensional. The mass of the potato is $4 \text{ m} = \rho V = \rho \pi r 3$ Ts Rair Rconv Rtowel $3 \text{ T} \propto 4 = (62.2 \text{ lbm/ft } 3) \pi (1.5 / 12 \text{ ft}) 3 = 0.5089 \text{ lbm} 3$ Pot The amount of heat lost as the potato is cooled from 300 to 200°F is $Q = \text{mC p} \Delta T = (0.5089 \text{ lbm})(0.998 \text{ Btu / lbm})(0.99$ growth. 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 = (1524. Common brick, 100 mm 4. Properties The thermal conductivity is given to be k = 20 W/m °C. Analysis The nodal spacing is given to be k = 20 W/m °C. are constant and uniform. 5-98C Yes, the global (accumulated) discretization error be less than the local error during a step. ° F Epoxy and k eff = = Epoxy (kt) copper + t epoxy t co $\frac{1}{2}$ tepoxy Q 3-22 Chapter 3 Steady Heat Conduction Thermal Contact Resistance 3-39C The resistance that an interface offers to heat transfer per unit interface area is called thermal contact resistance, Rc. Properties The properties of air at 1 atm and the film temperature of (Ts + T ∞)/2 = (65+30)/2 = 47.5°C are (Table A-15) k = 0.02717 W/m.°C v = 1.774 × 10 -5 m 2 /s Pr = 0.7235 For a location at 4000 m altitude where the atmospheric pressure is 61.66 kPa, only kinematic viscosity of air will be affected. 4 The surfaces of the wall are maintained at constant temperatures. Ts=100°C based on the maximum velocity and the Reynolds number V=5.2 m/s Ti=20°C based on the maximum velocity of air will be affected. 4 The surfaces of the wall are maintained at constant temperatures. become Vmax = Re D ST 0.05 V = $(5.2 \text{ m/s}) = 7.647 \text{ m/s} 0.05 - 0.016 \text{ ST} - D \rho V D (1.145 \text{ kg/m 3})(7.647 \text{ m/s})(0.016 \text{ m}) = max = 7394 \mu 1.895 \times 10 - 5 \text{ kg/m} \cdot \text{s SL ST}$ 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(7394) 0.63 (0.7268) 0.36 D (0.7268) 0.36 D0.7111) 0.25 = 66.26 Since NL = 20, which is greater than 16, the average Nusselt number and heat transfer coefficient for all the tubes in tubes in the tub $2 (450 - 55)^\circ F = 69.91$ Btu/h Q& = $\infty 1$ Rtotal 5.65 h°F/Btu If the thermal resistance of the steel pipe is neglected, the new value of total thermal resistance will be Rtotal = Ri + R 2 + Ro = 0.036 + 5.516 + 0.096 = 5.648 h°F/Btu Then the percentage error involved in calculations becomes error% = (5.65 - 5.648) h°F / Btu × 100 = 0.035\% 5.65 h° F / Btu which is insignificant. Analysis (a) Taking the direction normal to the surface of the wall to be the x direction with x = 0 at the left surface, the mathematical formulation of this problem can be expressed as d 2T = 0 dx 2 and k T (0) = T1 = 80° C T1 = 80° C A = 20 m2 dT (L) = h[T (L) - T\infty] dx (b) Integrating the differential equation twice with respect to x yields dT = C1 dx - k L=0.4 m T ∞ =15°C h=24 W/m2.°C T (x) = C1x + C2 x where C1 and C2 are arbitrary constants. °C)(1 m2)(100 - 30)°C = 2450 W Then the fin effectiveness becomes Q& 17800 ε fin = fin = = 7.27 & 2450 Q no fin 3-87 3 cm 0.6 cm D=0.25 cm Chapter 3 Steady Heat Conduction 3-118 "GIVEN" k spoon=8.7 " $[Btu/h-ft-F], parameter to be varied "T_w=200"[F]" T infinity=75"[F]" A c=0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12+0.5/12)" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[Btu/h-ft^2-F]" "ANALYSIS" p=2*(0.08/12*0.5/12" [ft^2]" "L=7 [in], parameter to be varied "h=3"[ft] "L=7 [in], parameter$ T tip kspoon [Btu/h.ft.F] 5 16.58 28.16 39.74 51.32 62.89 74.47 86.05 97.63 109.2 120.8 132.4 143.9 155.5 167.1 178.7 190.3 201.8 213.4 225 ΔT [F] 124.9 122.6 117.8 112.5 107.1 102 97.21 92.78 88.69 84.91 81.42 78.19 75.19 72.41 69.82 67.4 65.14 63.02 61.04 59.17 kspoon [Btu/h.ft.F] 5 5.5 6 6.5 7 7.5 8 8.5 9 9.5 10 10.5 11 11.5 12 ΔT [F] 122.4 connected to each other through two 1-cm thick flanges exposed to cold ambient air. Assumptions 1 Heat conduction in the steaks is one-dimensional since the steaks are large relative to their thickness and there is thermal symmetry about the center plane. problem. Properties The thermal conductivities of the plaster, brick, covering, polyurethane foam, and glass fiber are given to be 0.72 W/m·°C, 0.036 W/m·°C, 0.025 W/m·°C, 0.036 W/m·°C, 0.025 W/m·°C, 0.036 W/m·°C, 0.025 W/m·°C T surr summer=23+273 "[K]" A=1.6 "[m^2]" epsilon=0.95 T s=32+273 "[K]" "ANALYSIS" sigma=5.67E-8 "[W/m^2-K^4], Stefan-Boltzman constant" "(a)" Q dot rad summer=4) "(b)" Q dot rad summer=4) "(c)" Q dot inter [W] 170 160 150 140 130 120 281 283 285 287 289 291 T surr, winter [K] 1-73 A person is standing in a room at a specified temperature. 5 The temperatures of the $= + + 1 = 11 \Delta x x \Delta 1$ in 0.6 ft / plate / soil The temperature at node 10 (bottom of the soil) is given to be T10 = 50°F. Assumptions 1 The watermelon is a homogeneous spherical object. The mathematical formulation of this problem is to be expressed for steady two-dimensional heat transfer. An energy balance on this thin element of thickness Δz during a small time interval Δt can be expressed as (Rate of heat) (Rat G& element = Δt But the change in the energy content of the element and the rate of heat generation within the element = $gA \& \Delta z G \&$ elemen $-T - Tt 1 Qk z + \Delta z - Qk z + gk = \rho C t + \Delta t A \Delta z \Delta t$ Taking the limit as $\Delta z \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $1 \partial (\partial T) \partial T kA + gk = \rho C A \partial z \langle \partial z \rangle dt$ since, from the definition of the derivative and Fourier's law of heat conductivity k are the one-dimensional transient heat conduction equation in the axial direction in a long cylinder becomes ∂ 2 T g& 1 ∂ T + = ∂ z 2 k α ∂ t where the property α = k / ρ C is the thermal diffusivity of the material. That is, the velocity and temperature gradients normal to the surface are much greater than those along the surface. Analysis The average thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the
average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature since the thermal conductivity value at the average temperature rate of heat conduction through the plate becomes T -T (500 - 350)K Q& = k ave A 1 2 = (34.24 W/m · K)(1.5 m × 0.6 m) = 30,820 W 0.15 m L Discussion We would obtain the same result if we substituted the given k(T) relation into the second part of Eq. 2-76, and performed the indicated integration. It can also be expressed in terms of the temperatures at the neighboring nodes in the following easy-to-remember form: i i i i + Ttop + Tright + Tbottom - 4Tnode = Tleft Discussion We note that setting i +1 Tnode i = Tnode i +1 i - Tnode Tnode t e Tleft Discussion We note that setting i +1 i - Tnode Tnode t e Tleft Discussion We note that setting i +1 i - Tnode i = Tnode i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode i = Tnode i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note that setting i +1 i - Tnode t e Tleft Discussion We note t e Tl distance xcr where the Reynolds number becomes equal to the critical Reynolds number, Re cr = V \propto xcr $\nu \rightarrow$ xcr = ν Re cr (1.562 × 10 - 5 m 2/s)(5 × 10 5) = = 0.976 m 8 m/s V \propto The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer at that location is obtained by substituting the law of x into the laminar boundary layer at that locating the law o $2 \rightarrow \delta$ cr = 1 cr/2 = 0.006903 m = 0.69 cm V ∞ Re x Re cr (5 × 10 5) 1 / 2 Discussion When the flow becomes turbulent, the boundary layer thickness can be determined from the boundary layer thickness can be determined from the boundary layer thickness can be determined from the boundary layer thickness turbulent, the boundary layer thickness can be determined from the boundary layer thickness can be determined from the boundary layer thickness turbulent. block is placed on a table. Since the melting starts at the corner of the top surface, we need to determine the time required to melt ice block which will happen when the temperature drops below 0° C at this location. The explicit finite difference equations are determined on the basis of the energy balance for the transient case expressed as T i +1 This is $Q_{k} = Q_{k}$ and $Q_{k} = Q_{k}$ a determined. We measure x from the center of the block. °C)(0.015 × 1 m 2) L. Then the continuity: =0 $\partial x \partial x \partial y 12$ m/s Therefore, the x-component of velocity does not change in the flow direction (i.e., the velocity profile remains unchanged). / 12 ft) 2 = 0.3871 5-90 Chapter 5 Numerical Methods in Heat Conduction Substituting this value of τ and other given quantities, the time needed for the inner surface temperature of the window glass to reach 54°F to avoid fogging is determined to be never. Properties The density of water is 62.4 lbm/ft3, and the specific heat of water at room temperature is C = 1.0 Btu/lbm.°F (Table A-9). 2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no significant variation in the axial direction. The bottom surface at r = 0 is insulated, the top surface at z = H is subjected to uniform heat flux q& h, and the cylindrical surface at r = r0 is subjected to convection. 4 The air cavity does not have any reflecting surfaces. Plywood, 13 mm 5. 2 Heat transfer through the insulated side of the plate is negligible. Properties The thermal conductivity of aluminum bars is given to be k = 176 W/m·°C. Also the house loses 3421 kJ through the Trombe wall the 1st daytime as a result of the low start-up temperature, but delivers about 13,500 kJ of heat to the house the second day. Air space, 20 mm, nonreflective 5b. Air Properties The properties of air at 20°C and 1 atm are (Table A-15) 20°C Pr = 0.7309 ρ = 1.204 kg/m3, Cp = 1.204 kg/m prevent freezing. Access to solutions is limited to instructors only who adopted the text, and instructors may obtain their passwords for the OLC by contacting their McGraw-Hill Sales Representative at . Then the mesh Fourier number becomes $\tau = \alpha \Delta t l^2 = (12 \times 10 - 6 \text{ m } 2/\text{s})(60 \text{ s}) (0.1 \text{ m}) 2$ (for $\Delta t = 60 \text{ s}) = 0.072$ Using the specified initial condition 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(r2 / r $0.4 \text{ Re } 0.5 + 0.06 \text{ Re } 2/3 \text{ Pr } 0.4 \text{ } 1 \times 100.02514 \text{ W/m.°C } k (1910) = 11.71 \text{ W/m } 2.^{\circ} \text{ Ch} = \text{Nu} = \text{and } D 4.1 \text{ m}$ The rate of heat transfer to the liquid nitrogen is 7-94 1/4 $| 1 / 2 = 1910 \text{ Chapter 7 External Forced Convection As} = 0.4 \text{ Re } 0.5 + 0.06(3.005 \times 10.6) 2/3 (0.7309) 0.4 | 1 - 5 (1.189 \times 10.02514 \text{ W/m.°C } k (1910) = 11.71 \text{ W/m } 2.^{\circ} \text{ Ch} = \text{Nu} = \text{and } D 4.1 \text{ m}$ The rate of heat transfer to the liquid nitrogen is 7-94 1/4 $| 1 / 2 = 1910 \text{ Chapter 7 External Forced Convection As} = 0.4 \text{ Re } 0.5 + 0.06(3.005 \times 10.6) 2/3 (0.7309) 0.4 | 1 - 5 (1.189 \times 10.02514 \text{ W/m.°C } k (1910) = 11.71 \text{ W/m } 2.^{\circ} \text{ Ch} = \text{Nu} = \text{and } D 4.1 \text{ m}$ The rate of heat transfer to the liquid nitrogen is 7-94 1/4 $| 1 / 2 = 1910 \text{ Chapter 7 External Forced Convection As} = 0.4 \text{ Cm} + 0.06(3.005 \times 10.6) 2/3 (0.7309) 0.4 | 1 - 5 (1.189 \times 10.02514 \text{ W/m.°C } k (1910) = 11.71 \text{ W/m } 2.^{\circ} \text{ Ch} = \text{Nu} = \text{and } D 4.1 \text{ m}$ The rate of heat transfer to the liquid nitrogen is 7-94 1/4 | 1 / 2 | 1 / 2 = 1910 Chapter 7 External Forced Convection As = 0.4 Cm + 0 $\pi D 2 = \pi (4.1 \text{ m}) 2 = 52.81 \text{ m}
2 \text{ T}_{\infty} - \text{Ts}$, $\tan k = Q\& = r 1 \text{ Rinsulation} + \text{Rconv } 2 - r1 + 4\pi kr1 r^2 \text{ hAs} [20 - (-196)]^{\circ}C = 7361 \text{ W} (2.05 - 2) \text{ m} 1 + 4\pi (0.035 \text{ W/m.}^{\circ}C)(2.05 \text{ m})(2 \text{ m}) (11.71 \text{ W/m } 2.^{\circ}C)(52.81 \text{ m} 2)$ kJ/kg (c) We use the dynamic viscosity value at the new estimated surface temperature of 0°C to be $\mu = 1.729 \times 10 - 5$ 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 \times 300 W = = = 169.1 W/m 2 A2 2 π r2 $L 2\pi$ (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) = 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. $^{\circ}$ C)(80 - 0) $^{\circ}$ 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 water properties at the room temperature, the temperature drop of the hot water as it flows through this 25-m section of the water properties at the room temperature drop of the hot water as it flows through this 25-m section of the water properties at the room temperature, the temperature drop of the hot water as it flows through this 25-m section of the water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water as it flows through this 25-m section of the water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at the room temperature drop of the hot water properties at 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 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 Proper 5-28 A plate is subjected to specified heat flux on one side and specified temperature of the inner surface of the roof can be taken to be T1, ave = 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) + $\varepsilon\sigma$ As Twall] = (5 W/m 2 · °C)(20 × 20 m 2)(20 - 14.15)°C + 0.9(20 × 20 m 2)(20 - 14.15)°C + 0.9 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 = $\pi D 2 = \pi (3 \text{ m}) 2 = 28.27 \text{ m} 2 1 1 \text{ Ro} = = 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} \cdot \text{°C})(28.27 \text{ m} 2) \text{ Ts1 } \text{T} - \text{T} [15 - (-183)] \text{°C } Q \& = \text{s1} \otimes 2 = 196,040 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } Q \& 196.040 \text{ kJ/s} Q \& = \text{m} \& \text{h fg}$ \rightarrow m& = = 0.920 kg/s h fg 213 kJ/kg Ro T ∞ 2 (b) The heat transfer are of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are A = π D 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) $39.8^{\circ}C - (15 W)(0.011 \circ C/W) = 39.6^{\circ}C R$ 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 (αt) Wall 30 cm Substituting, $(\ 0.3 \text{ m} 5.1 - 5 \ | = 0.01 = \text{erfc} - 6215 - 5 \ | 2 (0.45 \times 10 \text{ m/s})t \ | | \\ / \text{Noting from Table 4-3 that } 0.01 = \text{erfc}(1.8215), \text{ the time is determined to be}(0.3 \text{ m} \ | | -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times 10 \text{ m/s})t \ | \\ -62 \ | 2 (0.45 \times$ 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 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 $(350+250)/2 = 300^{\circ}C$, and the properties of air at 1 atm pressure and the free stream temperature of $30^{\circ}C$ are $(Table A-15) k = 0.02588 W/m.^{\circ}C v = 1.608 \times 10^{-5} m 2 / s \mu \infty = 30^{\circ}C kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @
$300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, @ $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/m.s \mu s$, % $300^{\circ}C = 2.934 \times 10^{-5} kg/$ $15 \text{ cm Ts} = 350^{\circ}\text{C}$ Pr = 0.7282 D Analysis The Reynolds number is V D (6 m/s)(0.15 m) Re = $\infty = = 5.597 \times 10.4 \text{ u} 1.57 \times 10.57 \times$ $0.5 + 0.06(5.597 \times 104) 2/3 (0.7282) 0.4 \parallel -5 \parallel (2.934 \times 10)$ 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 = $\Delta E \rightarrow -Q \& \Delta t + E \& \Delta t = mC \Delta T$ in or, out generation device out generation p device (T + T ∞) E& generation $\Delta t - hAs = T \propto |\Delta t = mC p (T - T \propto) 2$ Substituting the given values, $(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (12 \text{ W/m } 2 \cdot C)(0.0005 \text{ m } 2) | C(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(T - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(1 - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(850 \text{ J/kg.}^{\circ}C)(1 - 25) \circ (30 \text{ J/s})(5 \times 60 \text{ s}) = (0.02 \text{ kg})(1 - 25) \circ (30 \text{ J/s})(1 - 25) \circ (30 \text{ J/s})(1$ $(12 \text{ W/m } 2 \cdot \text{C})(0.0085\text{ m } 2)$ | $(\circ \text{C} \times 60 \text{ s}) = (0.20 + 0.02) \text{ kg} \times (850 \text{ J/kg} \cdot \text{C})(\text{T} - 25)^{\circ} \text{C} \setminus 2$ / which gives $\text{T} = 69.5^{\circ} \text{C}$ 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^{\circ} \text{C} / \text{W kA} (0.026 \text{ W} / \text{m.o C})(11 \cdot \text{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. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient remain constant. 5 The surface temperature of the furnace and the heat transfer coefficient rem π Do L = π (0.046 m)(15 m) = 2.168 m2 1 1 = 0.0044 °C/W 2 hi Ai (120 W/m.°C)(1.885 m 2) ln(r2 / r1) ln(2.3 / 2) = = 0.00003 °C/W 2\pi k1 L 2\pi (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 (14.40 m - 1 × 0.02 m) = 0.973 = aL 14.40 m - 1 × 0.02 m) = 0.973 = aL 14.40 m - 1 × 0.02 m) of this problem is zero since the problem states to disregard radiation. ° C) $T \propto h \ k \ g \ 0$ ro r (b) The mathematical formulation of this problem can be expressed as 1 d (dT) $g \& |r| + = 0 \ r \ dr \ k \ dT$ (r0) = h[T (r0) - T \propto] (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) = $- r \rightarrow r dr k 2 dr (dr) k Applying the boundary condition at the center line, dT (0) g & 0 \times = - \times 0 + C1 \rightarrow C1 = 0$ B.C. at r = 0: dr 2k Dividing both sides of Eq. (a) by r to bring it to a readily integrating, g & 2 dT g & = - r T (r) = - r + C2 \rightarrow (b) 4k dr 2k Applying the boundary condition at r = r0, and $-k g\&r0 g\&r r g\& 2 (g\& 2) = h - r0 + C 2 - T\infty + 0 + r0 2k 2h 4k (4k)$ Substituting this C2 relation into Eq. (b) and rearranging give & gr g\& 2 T (r) = T $\infty + (r0 - r 2) + 0 4k 2h$ 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 = π Di 2 = π (5 m) 2 = 78.54 m 2 Ao = π Do 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 surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surface temperature T2 to be 5°C after comparing convection heat transfer coefficients at the outer surface temperature temperature temperature temperatu given to be k (T) = k0 (1
+ β T). 2 Thermal properties, heat transfer coefficients, and the indoor and outdoor temperature of the plate is to be determined when it stabilizes. h, T \propto 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)(0.12 m) 0.05 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 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)(0.12 m) 0.05 m (L) = 0.06 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 W/m.°C)(0.12 m) 0.05 m (L) = 0.05 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 W/m.°C)(0.12 m) 0.05 m (L) = 0.05 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 W/m.°C)(0.12 m) 0.05 m (L) = 0.05 °C/W R 2 = R 4 = RC = | | = 2 (kA / C (20 W/m.°C)(0.12 m) 0.05 m (L) = 0.05 °C/W R 2 = $^{\circ}C(0.04 \text{ m}) 0.05 \text{ m} (L) = 0.16 \circ C/W R3 = RB = | = 2 kA | B (8 W/m. \circ C)(0.04 \text{ m}) 0.1 \text{ m} (L) R5 = RD = | = 0.11 \circ C/W kA | D (15 W/m. \circ C)(0.06 \text{ m}) 0.06 \text{ m} (L) R7 = RF = | = 0.25 \circ C/W 2 kA | F (2 W/m. \circ C)(0.12 \text{ m}) 1 1 1 1 1 1 = + + = + + \rightarrow Rmid$ $0.025 ^{\circ}C/W \text{ Rmid}$, 1 R2 R3 R4 0.06 0.16 0.06 1 Rmid, 2 = 1 1 1 1 + = + \rightarrow Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 + R7 = 0.04 + 0.25 + 0.034 + 0.25 = 0.349 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 + R7 = 0.04 + 0.25 + 0.034 + 0.25 = 0.349 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = R7 = 0.04 + 0.025 + 0.034 + 0.25 = 0.349 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + Rmid, 2 = 0.034 $^{\circ}C/W \text{ R5}$ R6 0.11 0.05 Rtotal = R1 + becomes (5 m)(8 m) Q total = $(572 \text{ W}) = 1.91 \times 105 \text{ W} 0.12 \text{ m} 2$ (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 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& = $1 \rightarrow T = T1 - Q$ & Rtotal = $300^{\circ}C - (572 W)(0.065 \circ C/W) = 263^{\circ}C$ Rtotal (c) The temperature drop across the section F can be determined from 3-34 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 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 $\epsilon = 0.8$. Air Analysis The heat transfer are D = 2 in o 2 o 2 Q Q& conv = hAs ΔT = $(12Btu/h.ft \cdot F)(0.08727ft)(170 - 70)F = 104.7 Btu/h Q& rad = \varepsilon \sigma As (Ts4 - To4) = 0.8(0.08727ft 2)(0.1714 \times 10 - 8 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4 - (70 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R 4)[(170 + 460R) 4] = 9.4 Btu/h.ft 2 \cdot R$ • 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& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2
- T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 l T - T2 g& 12 l T2 - T1 l + 2h (T[∞] - T1) + 0 = 0 2 2 4 $1T1 - T2 + kl 5 + hl (T_{\infty} - T2) + 0 = 0.2 2l 12 g\& l 4T2 - 4T5 + 0 = 0 k Node 2 (convection) : 2k Node 5 (interior) : where g\& 0 = 0.19 \times 10.5 Btu/h.ft^oF, h = 7.9 Btu/h.ft^oF, and T_{\infty} = 70°F.$ 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 × 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 distribution in the ground can be determined from $(x T (x, t) - Ti = erfc|| T \infty - Ti (2 \alpha t)| + \alpha t || - exp| + || k| 2 \alpha t k || + \alpha t || - exp| + || k| 2 \alpha t k || |k| 2 || || / (Winds T \infty = -10^{\circ}C where 2 - 5 2 h \alpha t (40 W/m. \circ C) (1.6 \times 10 m / s)(10 \times 3600 s) = = 33.7 0.9 W/m. \circ C k Soil Ti 2 (h \alpha t) || = 33.7 2 = 1138 = || k || / (h 2 \alpha t k 2 Then we conclude that the last term in the$ temperature distribution relation above must be zero regardless of x despite the exponential term tending to infinity since (1) erfc (ξ) \rightarrow 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 $\Delta x \times \Delta y \times 1$ 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 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 their coefficients degree, and their coefficients are constant. = = 0.00037 < 01 k (35 W / m. Properties of the wall and the heat transfer coefficients are constant. = = 0.00037 < 01 k (35 W / m. Properties of the wall and the heat transfer coefficients degree, and their coefficients are constant. = = 0.00037 < 01 k (35 W / m. Properties of the wall and the heat transfer coefficients degree, and their coefficients are constant. = = 0.00037 < 01 k (35 W / m. Properties of the wall and the heat transfer coefficients degree, and their coefficients are constant. $k = 0.6 \text{ W/m} \cdot ^{\circ} \text{C}$ and $\alpha s = 0.7$. Analysis The nodal spacing is given to be $\Delta x = \Delta x = l = 1 \text{ m}$, and all nodes are boundary nodes. 2 Convection heat transfer coefficient is constant and Radiation h, T \propto 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 $\alpha \Delta t \ 1 \ \tau = \leq 2 \ 2 \ (\Delta x)$ 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 $\alpha \Delta t \ 1 \ \tau = \leq 2 \ 2 \ (\Delta x)$ method can be expressed in its simplest form as $\alpha \Delta t \ 1 \tau = \leq 2 \ 2 \ (\Delta x)$ 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 $A = 4\pi r \ 2$ 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 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 q&L (5 × 10 5 W/m 3)(0.015 m) = 155°C = 30°C + h=60 W/m 2.°C 2 h 60 W/m 3)(0.015 m) = 155°C = 30°C + h=60 W/m 2.°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C 2 h 60 W/m 3)(0.015 m) = 155°C + 158.7°C + 158.7 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 damn are to be determined for two different insulation and the outer surface temperature of the damn are to be determined. medium, and the thermal conductivity k of the medium is constant. Properties The thermal conductivity is given to be k (T) = k 0 (1 + β T 2). 90 W + ϵ As 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]" 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 In Plane Walls 3-1C (a) If the lateral surface area of the rod, As = nD 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 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 - v 1.655 × 10 m/s which is less than the critical transfer through the insulation is one dimensional. Revnolds 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" =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-O dot out)*time=m*C arameter to be varied" "PROPERTIES" rho=2770 "[kg/m^3]" C heat=0.85 T f=140 "[C] (1 I-1 I) "energy balance on the plate" n | W/m2.CI 5 / 9 II I3 I5 I / I9 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 59 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 = π (0.4 / 12 ft)(1 ft) = 0.105. (b) Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be $a = hp = kAc h\pi D k\pi D / 4 2 = 4h = kD 4(50 W / m2)$. Also, the energy balance expressions can be simplified using the definitions of thermal diffusivity $\alpha = k / (\rho C)$ and the dimensionless mesh Fourier number $\tau = \alpha \Delta t / l 2$ where $\Delta x = \Delta y = l \cdot 3$ 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): $4 \cos A(Tsurr - T04) + hA(T \propto - T0) + kA$ Heat transfer at right surface: O_{K} right surf is. 3 The thermal properties of the meat slab are constant. Analysis The initial rate of heat transfer from a potato is As = $\pi D 2 = \pi (0.10 \text{ m}) 2 = 0.03142 \text{ m} 2$ (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 $v = 1.426 \times 10^{-5}$ m 2 /s Air V $\infty = 50$ km/h T $\infty 2 = 4$ °C T $\infty 1 = 22$ °C WALL Pr = 0.7336 Analysis Air flows along 8-m side. Analysis (a) The amount of heat transferred to the value is simply the change in its internal (0.008 m)(0.1 m) As (d) The number of values that can be heat treated daily is Number of values (5 min) 1-77 Engine values (5 min) (25 values) = 3000 values (5 min) 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 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) 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 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is V L (4 m/s)(0.25 m) Re L = ∞ = 4.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is 0.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less than the critical Reynolds number is 0.579 × 10 4 ν 2.184 × 10 -5 m 2 /s which is less tha 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, 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 temperatures at node n is expressed in terms of the temperatures at 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 twodimensional heat conduction in a medium with heat generation and constant thermal conductivity is given by Tm-1, n-2Tm, n+Tm+1, Tm, n+1, q 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 $= 0 \int dr / dT (r1) = hi [Ti - T (r1)] dr dT (r2) - K = ho [T (r2) - To] dr - k and (b)$ 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 once with respect to r gives r Ti hi r1 r2 r To ho dT = C1 ln r + C2 where C1 and C2 are arbitrary constants. It is determined from 1 1 1 = $+ -1 \varepsilon$ effective $\varepsilon 1 \varepsilon 2$ where $\varepsilon 1$ and $\varepsilon 2$ are the emissivities of the surfaces of Radiation T –T T –T 4 kA 0 1 + kA 2 1 + h($p\Delta x / 2 + A$)[Tsurr – (T1 + 273) 4] = 0 Tsurr Δx Node 2 (at fin tip): T – T2 4 kA 1 + h($p\Delta x / 2 + A$)[Tsurr – (T2 + 273) 4] = 0 Δx where A = $\pi D 2 / 4$ 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]" "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 $(alpha*time)/r \ o \ c^2 (T \ o \ c^-T \ infinity)/(T \ i-T \ infinity)=A \ 1 \ c^2*tau \ c)$ "For sphere" Bi $s=(h*r \ o \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2*tau \ c)$ "For sphere" Bi $s=(h*r \ o \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2*tau \ c)$ "For sphere" Bi $s=(h*r \ o \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2*tau \ c)$ "For sphere" Bi $s=(h*r \ o \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2*tau \ c)$ "For sphere" Bi $s=(h*r \ o \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2*tau \ c)$ "For sphere" Bi $s=(h*r \ o \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2*tau \ c)$ "For sphere" Bi $s=(h*r \ o \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number, we read" lambda 1 $s^2(h*r \ s)/k$ "From Table 4-1 corresponding to this Bi number 4-1 corresponding to this Bi number 4-1 corresponding to this Bi n 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 75.41 75.16 75.06 75.41 75.16 75.06 75.41 75.16 75.07 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 75.07 75.56 75.3 75.16 4-110 To,s [F] 200.7 123.4 93.6 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 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 82.57 82.18 78.8 77.01 76.57 82.57 82.58 75.4 75.56 75.3 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75.16 75 cylinder 100 50 0 100 sphere 10 20 30 40 tim e [m in] 4-111 50 50 60 Chapter 4 Transient 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 = $\alpha\Delta t \Delta x 2 = (0.93 \times 10 - 6 \text{ m } 2 / \text{s})(5 \text{ s})(0.005 \text{ m}) 2 = 0186$. 3 Air is an ideal gas. Properties The thermal resistance of the insulation is the thermal resistance of the insulation gas. Properties The thermal resistance of the insulation is the thermal resistance of the insulation gas. Properties The the insulation gas. Properti temperature rise of water is not to exceed $2^{\circ}C$ as it flows through the chiller, the mass flow rate of water must be at least m& water = Q_{α} water 16.5kW = = 1.97 kg/s (C p ΔT) water (4.18 kJ/kg.° C)($2^{\circ}C$) If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than $2^{\circ}C$. 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 = $(0.4 \text{ m})(0.4 \text{ m}) + 4 \times (0.2 \text{ m})(0.4 \text{ m}) = 0.48 \text{ m} 2$ The radiation heat transfer from the box can be expressed as 100 W $\epsilon = 0.95100 \text{ W} = (0.95)(5.67 \times 10 - 8 \text{ W/m} 2 \text{ .K } 4)(0.48 \text{ m} 2)[(55 + 273 \text{ K}) + 2.38 \text{ m} 2)[(55 + 273 \text{ K}) + 2.38 \text{ m} 2)](55 + 273 \text{ K})$ 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 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 + 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 + 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 + 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 + 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 + 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 + 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 + 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 + 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 + ΔU teapot from 18°C) teapot = (2.5 kg)(4.18 kJ/kg). Analysis Disregarding the base area, the total heat transfer area of the transfer from the power transistor is As = $\pi DL + \pi D 2/4 = 1.037 \text{ cm} 2 = 1.037 \times 10 - 4 \text{ m} 2$ (1.037 × 10 - 4 m 2) (1.037 × 10 - 4 m 2) $(70 - 55)^{\circ}C = 0.047$ 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 $\Delta x = 3$ 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 ($2\pi r l$) (30 Btu / h.ft. For the case of steady 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 $\alpha = 12 \times 10 - 6 \text{ m2 / s}$.) = = 0.00244 h. The flow of compressible fluid (such as air) is not necessarily compressible fluid (such as air) is not ne 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 $A\Delta r$ gives $-T - Tt 1 Q\& r + \Delta r - Q\& r + g\& = \rho C t + \Delta t A \Delta r \Delta t$ Taking the limit as $\Delta r \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $\partial T 1 \partial (\partial T) | kA | + g\& = \rho C A \partial r \langle \partial t \partial r \rangle$ since, from the definition of the derivative and Fourier's law of heat conduction, $Q \& r + \Delta r - Q \& r \partial Q \partial (\partial T) = | -kA | \Delta r \rightarrow 0 \partial r \partial r \langle \partial r \rangle \Delta r$ lim Noting that the heat transfer area in this case is $A = 2\pi r L$ and the thermal conductivity is constant, the onedimensional transfer area in this case is $A = 2\pi r L$ and the thermal diffusivity of the material. 8-12C The logarithmic mean temperature difference Δ Tln is an exact representation of the average temperature difference Δ Tln is an exact representation of the average temperature difference between the fluid and the surface for the entire tube. interdependence of these variables are studied. Properties The specific heat of chicken are given to be 3.54 kJ/kg. °C. m2 Rcond Ro To 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 = $2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.0213$ °C/W Ro = 2 ho Ao (25 W/m. °C)(1.88 m 2) Ri = To 2 T1 T2 R total = 0.00004 °C/W Rcond = $2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.0213$ °C/W Ro = 2 ho Ao (25 W/m. °C)(1.88 m 2) Ri = To 2 T1 T2 R total = 0.00004 °C/W Rcond = $2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.0213$ °C/W Ro = 2 ho Ao (25 W/m. °C)(1.88 m 2) Ri = To 2 T1 T2 R total = 0.00004 °C/W Rcond = $2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.0213$ °C/W Ro = 2 ho Ao (25 W/m. °C)(1.88 m 2) Ri = To 2 T1 T2 R total = 0.00004 °C/W Rcond = $2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.0213$ °C/W Ro = 2 ho Ao (25 W/m. °C)(1.88 m 2) Ri = To 2 T1 T2 R total = 0.00004 °C/W Rcond = $2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1 1 = 0.00004 °C/W Rcond = 2\pi kL 2\pi (52 W/m. °C)(6 m) 1$ 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& = $2 \rightarrow T2 = T \propto 2 + Q$ & Ro = $12 \circ C + (7673 \circ W)(0.0213 \circ C/W) = 174.8 \circ C$ 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 $\Delta t \leq (0.015 \text{ m}) 2 4(3.2 \times 10 - 6 \text{ m} 2/s)[1 + (80 \text{ W/m} 2 \cdot \text{C})(0.015 \text{ m}) /(15 \text{ W/m} \cdot \text{C})] = 16.3 \text{ s}$ 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 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 = $N\pi DL = 200\pi (0.016 \text{ m})(1 \text{ m}) = 3.130 \text{ kg/s}$ Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become (Ah Te = Ts - 1.204 \text{ kg/m} 3)(5.2 \text{ m/s})(10)(0.05 \text{ m})(1 \text{ m}) = 3.130 \text{ kg/s} Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become (Ah Te = Ts - 1.204 \text{ kg/m} 3)(5.2 \text{ m/s})(10)(0.05 \text{ m})(1 \text{ m}) = 3.130 \text{ kg/s} $(Ts - Ti) exp| - s| m\& Cp (22) () = 100 - (100 - 20) exp| - (10.05 m)(108.7 W/m \cdot °C) = 43.44°C (3.130 kg/s)(1007 J/kg \cdot °C) | | / / 7-61 Chapter 7 External Forced Convection <math>\Delta Tln = (Ts - Ti) - (Ts - Te) (100 - 43.44) = 67.6°C ln[(Ts - Ti) / (Ts - Te)] ln[(100 - 20) / (100 - 43.44)] Q\& = hAs \Delta Tln = (108.7 W/m 2 \cdot °C)$ $(10.05 \text{ m 2})(67.6^{\circ}\text{C}) = 73,882 \text{ W}(b)$ 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 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. 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. A 1 and A 1 corresponding to this Biot number are, from Table 4-1, $\lambda 1 = 0.4328$ 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 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^2-C]" T infinity=25 "[C]" m food=15 "[kg]" C food=3600 "[]/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, $T \propto = 30^{\circ}C$ 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 $\tau > 0.2$ 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_{kx} - Q_{kx} + \Delta x = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C A \Delta x t + \Delta t = \rho C$

Dividing by $A\Delta x$ gives $-T - Tt 1 Q\& x + \Delta x - Q\& x a Q \partial (\partial T) = - A A \Delta x \Delta t$ Taking the limit as $\Delta x \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $\partial T 1 \partial (\partial T) = - A A \Delta x \Delta t$ Taking the limit as $\Delta x \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $\partial T 1 \partial (\partial T) = - A A \Delta x \Delta t$ Taking the limit as $\Delta x \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $\partial T 1 \partial (\partial T) = - A A \Delta x \Delta t$ Taking the limit as $\Delta x \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $\partial T 1 \partial (\partial T) = - A A \Delta x \Delta t$ Taking the limit as $\Delta x \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $\partial T 1 \partial (\partial T) = - A A \Delta x \Delta t$ constant, the one-dimensional transient heat conductivity k becomes $\partial 2 T 1 \partial T = \partial x 2 \alpha \partial t$ where the property $\alpha = k / \rho C$ is the thermal diffusivity of the material. 5 The thermal diffusivi 4-1, $\lambda 1 = 3.0877$ 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 - $\lambda 1 \tau \rightarrow = (1.9969)e^{-3.0877}$ ($\tau \rightarrow \tau = 0.198 \approx 0.2$ 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) | $(0.05) 2 | []] \rightarrow t = 108,135$ 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 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. $C v = 1.516 \times 10^{-5} \text{ m} 2 / \text{s}$ Insulation $\mu \infty = 1.825 \times 10^{-5} \text{ m} 2 / \text{s}$ Insulation $\mu \infty = 1.825 \times 10^{-5} \text{ m} 2 / \text{s}$ Insulation $\mu \infty = 1.825 \times 10^{-5} \text{ m} 2 / \text{s}$ 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) = 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) = 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) = 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) = 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) = 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) = 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) = 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) = 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) = 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) = 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) = 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) = 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) = 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 4m The rate of heat transfer to the liquid oxygen is Q 4m The rate of heat transfer to the liquid oxygen i $h(\pi D 2)(T - T) = (13.95 \text{ W/m 2} \cdot \text{C})[\pi (4 \text{ m}) 2][(20 - (-183)] \circ \text{C} = 142,372 \text{ W} \text{ and } h = s \text{ s} \propto s \propto \text{The rate of evaporation of liquid oxygen then becomes Q& 142.4 kJ/s (b)}$ can be expressed in the rate form as $E_{\phi} = \Delta E_{\phi}$ system \hat{E}_{0} (steady) = 0 $\rightarrow E_{\phi}$ in = E_{ϕ} 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 $\ln(r^2/r^1) \ln(12)$. The heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament, the heat flux on the surface of the filament (filament) is the filam 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 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 temperature for a two-dimensional problem is determined by determin 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 + 0.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 10.201 × 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 styrofoam Ro = 1 1 = 0.0332 °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 = $85,500 + 2 \times 1250 = 88,000 \text{ W} 3-62$ "[m]" L wall=0.2 "[m]" L 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 conv o=1/(h o*A 3) A 3=(2*height+width)*length $R_total_top_sides = R_conv_i +
R_conv_i = (h_i + R_conv_i - (h_i + A_4) A_4 = (h_i + 1)/(k_i + 1)/(k_$ 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 to /m - C] 3-63E The thermal resistance of an epoxy glass laminate across its thickness is to be reduced by planting cylindrical copper fillings throughout. Analysis The mass of the iron's base plate is Air 22°C m = $\rho V = \rho LA = (2770 \text{ kg} / \text{m}^3)(0.005 \text{ m})(0.03 \text{ m}^2) = 0.4155 \text{ kg}$ Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is $Q\& = 0.85 \times 1000 \text{ W} = 850 \text{ W}$ in IRON 1000 W The temperature of the plate, changes during the process. Linoleum ($R = 0.009 \text{ m} 2.^{\circ} \text{C/W}$) 3. 6-28C Steady simply means no change with time at a specified location (and thus $\partial u / \partial t = 0$), but the value of a quantity may change from one location to another (and thus $\partial u / \partial y$ may be different from zero). 5-70 Chapter 5 Numerical Methods in Heat Conduction 5-79 Starting with an energy balance on a volume element, the two-dimensional transient implicit 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. 2 The critical Reynolds number is Recr = 5×105. 3 The thermal properties of the steel plates are constant. Also, the length of time it will take for the insulation to pay for itself from the energy it saves will be determined. The smallest primary coefficient in the 9 equations above is the coefficient of T9i in the T6i +1 expression since it is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factors for W and m are straightforward, and are given in conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor between °C into °F in this case is exposed to most conversion factor 1°C = 1.8°F since the °C in the unit W/m2.°C represents per °C change in temperature, and 1°C change in temperature corresponds to a change of 1.8°F. If the radius of insulation is less than critical radius of insulation of the pipe, the rate of heat loss will increase. Noting that the volume element of a general interior • • • node m involves heat and multiplying by $\Delta x/k$, it simplifies to Tmi -1 - (2 + h $\Delta x / k$) Tmi + Tmi +1 + g $\Delta x 2$ (Δx) 2 i +1 h $\Delta x T \infty$ + 0 = (Tm - Tmi) k k $\alpha \Delta t$ where $\alpha = k / (\rho C)$ is the thermal diffusivity of the wall material. Urethane foam insulation, 25-mm 5. Alternative solution We could also solve this problem using transient temperature charts as follows: 0.47 W/m.^o C k $1 = = 0.204 | \text{Bi hL } (20 \text{ W/m}^{2.9} \text{ C})(0.115 \text{ m}) \alpha t | \tau = 2 = 0.75 \text{ To} - T_{\infty} - 18 - (-30) \text{ L} | = = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The surface temperature is determined from } 4-52 (\text{Fig.} 4 - 13a) \text{ Chapter 4 Transient Heat Conduction } 1 \text{ k} | = = 0.204 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ The refore, } t = \tau \text{ ro } 2 (0.75)(0.115 \text{ m}) \alpha t | t = 0.324 | \text{Ti} - T_{\infty} 7 - (-30) \text{ Ti} + (-30) \text{$ hL = 0.22 x To $-T \propto | =1 | J L$ (Fig. F)(2.094 ft 2) = Ri + R1 + R 2 + Ro = 0.036 + 0.002 + 5.516 + 0.096 = 5.65 h \cdot °F/Btu Ro = Rtotal Then the steady rate of the watermelon is affected by the convection heat transfer at those surfaces only. Assumptions 1 The temperature of the steady rate of heat loss from the steady rate of heat loss fr Steady operating conditions exist and thus the rate of heat loss from the wire equals the rate of heat generation in the wire as a result of resistance heating g(x) the direction of all heat transfers to be towards the node Radiation T0 under consideration, the finite difference formulations Δx become • • • Right boundary node (all temperatures are in K): $0 \ 1 \ 2 \ 3 \ \text{Convection T2} - \text{T3} \ 4 \ \& h, \ \text{T} \propto \varepsilon \sigma A(\text{Tsurr} - \text{T3}) + hA(\text{T} \propto - \text{T3}) + hA(\text{$ uniform heat flux q& 0 and convection at the left (node 0) and radiation at the right boundary (node 2). = 01484. Properties The thermal conductivities are given to be k = $12 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^{\circ}\text{C}$ for the corper plate and fins, and k = $1.8 \text{ W/m} \cdot ^$ temperature in the wire, and the temperature at the centerline of the wire are to be determined. Assumptions 1 The balls are spherical in shape with a radius of r0 = 4 mm. 2-99 A plate with variable conductivity is subjected to specified temperatures on both sides. Construction 1. Analysis The Biot number is Bi = hro (15 W / m 2.
The convection resistance can be defined as the inverse of the convection heat transfer coefficient per unit surface area since it is defined as Rconv = 1/(hA). 7-3C The force a flowing fluid exerts on a body in the flow direction is called drag. The energy balance on the system can be expressed as E - E out 1in424 3 Net energy transfer by heat, work, and mass = ΔE system 1 424 3 Change in internal, kinetic, potential, etc. 1-7C The right choice between a crude and complex model is usually the simplest model is usually the simplest model which yields adequate results. This corresponds to Q& 399 W %change = difference × 100 = 6.1% (decrease) & 6559 W Q total,0° C Therefore, the effect of the temperature variations of the surrounding surfaces on the total heat transfer is less than 6%. We evaluate the air properties at the assumed mean temperature of 70°C and 1 atm (Table A-15): k = 0.02881 W/m-K $\rho = 1.028$ kg/m3 Cp = 1.007 kJ/kg-K Pr = 0.7177 $\mu = 2.052 \times 10$ kg/m-s Prs = Ts = 0.7041 - 5 Also, the density of air at the inlet temperature of 40°C (for use in the mass flow rate calculation at the inlet) is $\rho i = 1.127$ kg/m3. Properties The specific heat of air at room temperature of the thin-shelled spherical tank is nearly equal to the temperature of the nitrogen inside. Assumptions 1 Thermal properties of the ice and water are constant. m)(015 . 037 r / 1 / 1 Note that the pipe is essentially isothermal at a temperature of about -3.9°C. Assumptions There are no heat dissipating equipment (such as computers, TVs, or ranges) in the room. Then the temperature at the surface of the plates becomes $\theta(L, t)$ wall = 2 2 T (x, t) – T ∞ = A1 e $-\lambda$ 1 τ cos(λ 1 L /L = (1.0018)e - (0.1039) (90.4) cos(0.1039) = 0.378 Ti - T ∞ T (L, t) - 700 = 0.378 \rightarrow T (L, t) = 445 °C 25 - 700 Discussion This problem can be solved easily using the lumped system analysis since Bi < 0.1, and thus the lumped system analysis is applicable. 5-89 Chapter 5 Numerical Methods in Heat Conduction 5-90E A plain window glass initially at a uniform temperature is subjected to convection on both sides. This problem involves 6 Radiatio Convectio Concret unknown nodal temperatures, and thus we need to ho, e roof have 6 equations. 4-74 Chapter 4 Transient Heat Conduction 4-77E A hot dog is dropped into boiling water. m) = 0.03142 m2 1 1 Ro = = 3.183 °C/W 2 ho Ao (10 mining water). W/m .°C)(0.03142 m 2) $\ln(r2/r1) \ln(4/3) = 2.818$ °C/W 2πkL 2π (0.13 W/m 2 .°C)(0.125 m) = Ro + Rinsulation = 3.183 + 2.818 = 6.001 °C/W Rinsulation rot the side surface areas is (πr 2) / (2πrL) = r / (2 L) = 3 × 12.5) = 0.12. Therefore, the drink left on a table will warm up faster. Assumptions 1 The meat slabs can be approximated as very large cast iron container filled with ice are exposed to hot water. 2 Heat losses through the lateral surfaces of the apparatus are negligible since those surfaces are well-insulated, and thus the entire heat generated by the heater is conducted through the samples. Analysis (a) We assume that the surface temperature of the tube is equal to the temperature of the water. From Table 4-1 we read, for a sphere, $\lambda 1 = 3.094$ and A1 = 1.998. Assumptions 1 Heat transfer to the base surface from the top and side surfaces is to be determined. This is also equal to the rate of heat gain by water. Properties The thermal conductivity is given to be $k = 28 \text{ W/m} \cdot \text{C}$. 6-24C Turbulent viscosity μ t is caused by turbulent eddies, and it accounts for momentum transport by turbulent eddies. Then heat conduction along this two-layer plate can be expressed as (we treat the two for momentum transport by turbulent eddies). layers of epoxy as a single layer that is twice as thick) [] $\Delta T (\Delta T) (\Delta T) + kA Q = Q copper + Q epoxy = kA = keff (t copper + tepoxy and thermal conductivity keff can be expressed as <math>\Delta T (\Delta T) Q = kA = keff (t copper + tepoxy)$ epoxy) w LL / board Setting the two relations above equal to each other and solving for the effective conductivity gives (kt) copper + (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) copper + (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) copper + (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) copper + (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) copper + (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) epoxy \rightarrow k eff = t copper + t epoxy \rightarrow k eff = t copper + t epoxy \rightarrow k eff = t copper + t epoxy \rightarrow k eff = t epox \rightarrow epox transfer rate for an existing system at a specified temperature difference. Therefore, the mass flow rate of air is m& air = 170 kW Q& air = = 63.0 kg/s (C p ΔT) air (1.0 kJ/kg.°C)[0.5 - (-2.2)°C] Then the volume flow rate of air becomes m& 63.0 kg/s = 49.2 m³/s V&air = air = ρ air 1.28 kg/m³ 4-94 Chapter 4 Transient Heat Conduction 4-101 Turkeys are to be frozen by submerging them into brine at -29°C. Properties The properties of oil at the average temperature of (40+15)/2 = 27.5°C are (Table A-13): k = 0.145 W/m-K and μ = 0.580 kg/m-s = 0.580 N-s/m2 Analysis (a) We take the x-axis to be the flow direction, and y to be the normal direction. immersion cooling is much higher, and thus the cooling time during immersion chilling is much lower than that during forced air chilling. 3-120 Chapter 3 Steady Heat Conduction 3-161E The surface temperature of a baked potato drops from 300°F to 200°F in 5 minutes in an environment at 70°F. The rate of heat loss from the steam per unit length and the error involved in neglecting the thermal resistance of the steel pipe in calculations are to be determined. 2-95C The thermal conductivity of a medium, in general, varies with temperature. m)(01. Outside surface, 24 km/h 2. The emissivity Convection C Radiation of both surfaces of the concrete roof is 0.9. ho, To o Analysis The nodal spacing is given to be $\Delta x = 1$ in. Analysis We use the transient chart in Fig. Therefore, assuming the one-term approximate solution for transient heat conduction to be applicable, the temperature at the surface of the trees in 4 h becomes 2 T (ro, t) - T $\infty = A1 e - \lambda 1 \tau J 0 (\lambda 1 r / ro) \theta(ro, t) - 520 = (1.5989)e - (2.3420) (0.184)$ $(0.0332) = 0.01935 \rightarrow T$ (ro, t) = 511 °C > 410 °C 30 - 520 Therefore, there are two • • • unknowns T1 and T2, and we need two equations to determine 0 1 Δx 2 them. $32 \times 10^{\circ}$ C 30 - 520 Therefore, the trees will ignite. The difference between the two results is probably due to the Fourier number being less than 0.2 and thus the error in the one-term approximation. Therefore, there are two • • • unknowns T1 and T2, and we need two equations to determine 0 1 Δx 2 them. $32 \times 10^{\circ}$ C 30 - 520 Therefore, the trees will ignite. Radiation from the left surface, and $\Delta x \epsilon$ convection from the right surface are negligible. 4-90C The contamination of foods with microorganisms can be prevented or minimized by (1) preventing contamination by following strict sanitation practices such as washing hands and using fine filters in ventilation systems, (2) inhibiting growth by altering the environmental conditions, and (3) destroying the organisms by heat treatment or chemicals. \times 3.08 + 0.52 = 3.97 W Then the time of heating becomes 3-55 Chapter 3 Steady Heat Conduction $\Delta t = Q \ 10,329 \ J = 2602 \ s = 43.4 \ min \ Q\& \ 3.97 \ J \ / \ s \ Discussion$ The thermal contact resistance did not have any effect on heat transfer. 2 Heat conduction in the potato is one-dimensional because of symmetry about the midpoint. 2 Heat transfer through the ceiling is one-dimensional. Therefore, heat transfer through the brick wall will be larger despite its higher thickness. The total rate of heat generation in this section of the plate is $L \& G \& = gV = g \& (A \times L) = (5 \times 10.6 \text{ W} / \text{m 3})(1 \text{ m 2})(0.03 \text{ m}) = (5 \times 10.6 \text{ W} / \text{m 3})(1 \text{ m 2})(0.03 \text{ m}) = (5 \times 10.6 \text{ W} / \text{m 3})(1 \text{ m 2})(0.03 \text{ m}) = (5 \times 10.6
\text{ W} / \text{m 3})(1 \text{ m 2})(0.03 \text{ m}) = (5 \times 10.6 \text{ W} / \text{m 3})(1 \text{ m 2})(0.03 \text{ m}) = (5 \times 10.6 \text{ W} / \text{m 3})(1 \text{ m 2})(1 \text{$ 1.5 × 105 W plate Noting that this heat will be dissipated from both sides of the plate, the heat flux on either surface of the plate 2 × 1 m Heat Conduction Equation 2-19 The one-dimensional transient heat conduction Equation 2-19 The one-dimension 2-19 The one-dimensional transient heat conduction Equat and heat generation is + = . K4 is to be expressed in the English unit, Btu / h. We note that $\tau = \alpha t L2 = (9.75 \times 10 - 5 \text{ m } 2/\text{s})(285 \text{ s})(0.2 \text{ m}) 2 = 0.69 > 0.2$ and thus the assumption of $\tau > 0.2$ for the applicability of the one-term approximate solution is verified. R-8 Analysis The thickness of flat R-8 insulation (in m2.°C/W) is determined from the definition of R-value to be L Rvalue = \rightarrow L = Rvalue k = (8 m2. The final equilibrium temperature of the tank is to be determined. Alternative solution We could also solve this problem using transient temperature of the tank is to be determined. Alternative solution We could also solve this problem using transient temperature of the tank is to be determined. Alternative solution We could also solve this problem using transient temperature of the tank is to be determined. Alternative solution We could also solve this problem using transient temperature of the tank is to be determined. $0.283 | \text{Ti} - \text{T} \propto 78 - 25]$ Therefore, t= (Fig.4 - 15a) τ ro 2 (0.43)(1.25/12ft) 2 = 3333 s = 55.5 min α 1.4 × 10 - 6 ft 2/s The lowest temperature during cooling will occur on the surface (r/r0 = 1) of the oranges is determined to be 4-48 Chapter 4 Transient Heat Conduction 1 k = 0.543 | Bi h ro | T (r) - T $\propto = 0.45$ r To - T $\propto = 1$ ro - T $\approx = 0.45$ r To - T = 0.45 r To - T $\approx = 0.45$ r To - T = 0.45 r To - T = which gives (Fig. 6 The convection resistance inside the pipe is negligible. Face brick, 100 mm 3. 3 Thermal conductivities are given to be k = 386 W/m.°C for copper layers and k = 0.26 W/m.°C for copper layers and k = 0.2 walls and the windows. Applying the boundary conditions give T (r1) = -r = r1: C1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 - T\infty || 2 r r2 (2) Solving for C1 and C2 into the general solution, the variation of temperature is determined to be C1 = T (r) = -(11) C1 C T1 - T ∞ + T1 + 1 = C1 $|| - || + T1 = r k r r 1 (r1 r) (-183 - 20)^{\circ} C (2.1 2.1) - || + (-183)^{\circ} C = 516.7(1.05 - 2.1/r) - 183 2.1 18$ W/m · °C 2 r () 1 - 2 (25 W/m 2 · °C)(2.1 m) (c) The rate of heat transfer through the wall and the rate of evaporation of nitrogen are determined from Cr (T – T) dT Q& = $-kA = -k(4\pi r 2) 21 = -4\pi k 21 \infty r k dx r 1 - 2 - r1 hr2 (2.1 m)(-183 - 20)^{\circ} C = -4\pi (18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative) 2.1 18 W / m \cdot ^{\circ} C) = -245,450 W (to the tank since negative)$ Chapter 2 Heat Conduction Equation 2-128 A large plane wall is subjected to convection, radiation, and specified temperature on the right surface. Assumptions Heat is generated uniformly in the uranium rods. The average friction coefficient Cf can be determined from F f = C f As $\rho V 2 2 \rightarrow C f = Ff \rho A s V / 2 2 = C f A s \rho V 2 2$ 1 kg · m/s 2 1 N (1.204 kg/m 3) (32 m 2) (10 m/s) 2 / 2 1 2 . 4-99C The factors, which affect the quality of frozen, fish are the condition of the fish before freezing, the freezing method, and the temperature and humidity during storage and transportation, and the length of storage time. V Analysis The energy equation in Prob. Analysis (a) The rate of heat loss from the steam pipe is Ao = $\pi DL = \pi (0.1 \text{ m})(50 \text{ m}) = 15.71 \text{ m} 2 \text{ Q}$ bare = ho A(Ts - Tair) = (20 W/m 2.°C)(15.71 m 2)(150 - 15)°C = 42,412 kJ/s)(365 × 24 × 3600 s/yr) = 1.337 × 10 9 kJ/yr The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& $\Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr$ The amount of heat loss per year is Q = Q& \Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr The amount of heat loss per year is Q = Q& \Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr The amount of heat loss per year is Q = Q& \Delta t = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr The amount of heat loss per year is Q = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr The amount of heat loss per year is Q = (42.412 \text{ kJ/s})(365 \times 24 \times 3600 \text{ s/yr}) = 1.337 \times 10 9 kJ/yr of 75% is Q gas = 1.337×10.9 kJ/yr (1 therm) || || = 16,903 therms/yr (0.75 (105,500 kJ) The annual cost of this energy lost is Energy cost = (Energy used)(Unit cost of energy) = (16,903 therms/yr)(\$0.52 / therm) = \$8790/yr (c) In order to save 90% of the heat loss and thus to reduce it to $0.1 \times 42,412 = 4241$ W, the thickness of insulation needed is determined from Q& insulated = Ts - Tair = Ro + Rinsulation Ts - Tair ln(r2 / r1) 1 + ho Ao $2\pi kL$ Ts Rinsulation Ro Tai Substituting and solving for r2, we get 4241 W = (150 - 15)°C 1 (20 W/m 2.°C)[($2\pi r2$ (50 m)] + ln(r2 / 0.05) 2π (0.035 W/m.°C)(50 m) Then the thickness of insulation becomes t insulation = r2 - r1 = 6.92 - 5 = 1.92 cm 3-480 $rac{-}$ r2 = 0.0692 m Chapter 3 Steady Heat Conduction 3-71 An electric hot water tank is made of two concentric cylindrical metal sheets with foam insulation in between. Node 1 on the insulated boundary can be treated as an interior node for which Tleft + Ttop + Tright + Tbottom - 4Tnode = 0. energies We, in - Qout = $\Delta U = (\Delta U)$ water + (ΔU) air $\odot 0$ W&e,in $\Delta t - Qout = [mC(T2 - T1)]$ water water 80° Substituting, (15 kJ/s) $\Delta t - (50,000 kJ/h)(10 h) = (1000 kg)(4.18 kJ/kg \cdot C)(22 - 80)$ °C It gives $\Delta t = 17,170 s = 4.77 h$ (b) If the house incorporated no solar heating, the 1st law relation above would simplify further to W& $\Delta t - Q = 0$ e,in out Substituting, (15 kJ/s) $\Delta t - (50,000 kJ/h)(10 h) = 0$ It gives $\Delta t = 17,170 s = 4.77 h$ (b) If the house incorporated no solar heating, the 1st law relation above would simplify further to W& $\Delta t - Q = 0$ e,in out Substituting, (15 kJ/s) $\Delta t -
(50,000 kJ/h)(10 h) = 0$ It gives $\Delta t = 17,170 s = 4.77 h$ (b) If the house incorporated no solar heating, the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to W = 0 to solar heating the 1st law relation above would simplify further to the 1st law relation above would simplify further to the 1st law relation above would simplify further to the 1st law relation above would simplify further to the 1st law relating the 1st $\Delta t = 33,330 \text{ s} = 9.26 \text{ h} 1-71 \text{ Chapter 1}$ Basics of Heat Transfer 1-127 A standing man is subjected to high winds and thus high convection coefficients. Properties The thermal conductivity of the insulating material is given to be k = 0.04 W/m·°C. ° C)(11. Applying the boundary conditions give T (r1) = - r = r1: C1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = $C2 - T\infty || 2 r r 2 \langle 2 \rangle$ Solving for C1 and C2 simultaneously gives -kr = r2: $r2 (T1 - T\infty + T1 + 1 = C1|| - || + T1 = rk r r 1 \langle r1 - T\infty \rangle$ $1-2-r1 hr2 (r2 r2) || - | + T1 | (r1 r) (-196 - 20)^{\circ}C (2.1 2.1) - | + (-196)^{\circ}C = 549.8(1.05 - 2.1/r) - 196 2.1 18 W/m \cdot ^{\circ}C 2 r () 1 - - 2 (25 W/m 2 \cdot ^{\circ}C)(2.1 m) (c) The rate of heat transfer through the wall and the rate of evaporation of nitrogen are determined from C r (T - T) dT Q& = - kA = - k (4 m r 2) 21 = -4 m kC1 = -4 m k 2.1 \infty r k$ $dx r 1 - 2 - r1 hr2 = -4\pi (18 W / m^{\circ} C) m = (2.1 m)(-196 - 20)^{\circ} C = -261,200 W$ (to the tank since negative) 2.1 18 W / m^{\circ} C 1 - 2 (25 W / m 2^{\circ} C)(2.1 m) 261,200 J / s Q = = 1.32 kg / s h fg 198,000 J / kg 2-67 Chapter 2 Heat Conduction Equation 2-127 A spherical liquid oxygen container is subjected to specified temperature on the inner surface and convection on the outer surface.)e - (2.1589) (0.578) = 0106. 3 The energy stored in the container itself is negligible relative to the energy stored in water. 2 Heat conduction in the watermelon is one-dimensional because of symmetry about the midpoint. Assumptions 1 Heat transfer is given to be steady and two-dimensional since the height of the chimney is large relative to its cross-section, and thus heat conduction through the chimney in the axial direction is negligible. But at very low Reynolds numbers, Lh is very small (Lh = 1.2D at Re = 20). Maximum metabolic rates of trained athletes can exceed 2000 W. This is because some water remains unfrozen even at subfreezing temperatures, and the lower the temperature, the smaller the unfrozen water content of the beef. Tin=20°C In steady operation, heat transfer from the roof to the surroundings (by convection and radiation), that must be equal to the heat transfer through the roof by conduction. Assumptions 1 Heat transfer through the pin fin is given to be steady and one-dimensional, and the thermal conductivity to be constant. Noting that u = u(y), v = 0, and $\partial P / \partial x = 0$ (flow is maintained by the motion of the upper plate rather than the pressure gradient), the x-momentum equation reduces to 20 cm (∂u d 2u ∂u) $\partial 2 u$ ∂P x-momentum: $\rho \mid u \rightarrow = 0 + v \mid = \mu 2 - \partial y / \partial x$ dy 2 $\partial y \setminus \partial x$ This is a second-order ordinary differential equation, and integrating it twice gives u (y) = C1 y + C 2 The fluid velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the velocities of the plate surfaces must be equal to the velocities of the vel k = 0.01433 Btu/h.ft.°F Air V $\infty = 7$ ft/s T $\infty = 60$ °F $\upsilon = 0.1588 \times 10$ ft /s Pr = 0.7321 - 3 2 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = 4.407 \times 10432 - \upsilon 0.1588 \times 10$ ft /s Pr = 0.7321 - 32 Analysis For the first 1 ft interval, the Reynolds number is V L (7 ft/s)(1 ft) Re L = $\infty = 4.407 \times 10432 - \upsilon 0.1588 \times 100$ ft /s Pr = 0.7321 - 32 ft /s Pnerated in the resistance wires is transferred to the base plate, the heat flux through the inner surface is determined to be g & 0 = O & 0 1200 W = 75,000 W/m 2 Abase $160 \times 10 - 4 m 2$ Takin surface of the wall to be the x direction with x = 0 at the left surface, the mathematical formulation of this problem can be expressed as and $kT2 = 85^{\circ}C x$ (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. Center temperatures are to be determined for different foods. 5 Heat transfer from the leading edge
of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined. The heat transfer coefficient at the surface of the watermelon and the temperature of the outer surface of the watermelon are to be determined. $^{\circ}$ C)(1 m2) Repoxy = 0.005 m L = = 0.01923 $^{\circ}$ C / W kA (0.26 W / m. Analysis The Biot number is Bi = hro (8 W/m 2. $^{\circ}$ C)(0.045 m) = = 0.861 k (0.418 W/m. $^{\circ}$ C) Air T = -15° C The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, λ 1 = 1.476 and A1 = 1.2390 The Fourier number is $\tau = \alpha t r 02 = Apple$ (13 . 2 Heat transfer through the plate is onedimensional. Therefore, after 5 minutes, the thermometer reading will probably be more than 185 ° F. m) = 0.01 m 2 $\Delta T = 82 - 74 = 8^{\circ}$ C L Then the thermal conductivity of the material becomes & $\Delta T QL (14 W)(0.005 m) Q\& = kA \rightarrow k = = 0.875 W / m$. The highest temperature will occur at the insulated surface which is Insulate Properties The thermal conductivity of the soil is given to be $k = 0.9 \text{ W/m} \cdot \text{C}$. energies $[mC(0 \text{ C} - \text{T}) \text{ o 1 solid} \rightarrow 0 = \Delta U \rightarrow (\Delta U) \text{ ice} + [mC (T2 - T1)] \text{ water} = 0 \text{ (+ mhif + mC T2 - 0 o C)} + (0.2 \text{ kg})(4.18 \text{ kJ/kg} \cdot \text{C})(5-20) \cdot \text{C}$ = 0 It gives m = 0.0354 kg = 35.4 g Cooling with cold water can be handled the same way. Analysis Without insulation, the total thermal resistance is 11 = 12.42 h.°F/Btu 2 ho Ao (2.5 Btu/h.ft.°F)[π (0.083/12 ft)(1 ft)] Wire With insulation, the total thermal resistance is 11 = 12.42 h.°F/Btu 2 ho Ao (2.5 Btu/h.ft.°F)[π (0.083/12 ft)(1 ft)] Btu/h.ft.°F)[$\pi(0.123/12 \text{ ft})(1 \text{ ft})$] ln(r2 / r1) ln(0.123 / 0.083) = = 0.835 h.°F/Btu 2 π kL 2 $\pi(0.075 \text{ Btu/h.ft.°F})(1 \text{ ft})$] Rplastic Ts Rinterface = hc 0.001 h.ft 2 .°F/Btu = 0.046 h.°F/Btu Since the total thermal resistance decreases after insulation, plastic insulation will increase heat transfer from the wire. 4 The outer surface at r = r0 is subjected to convection and radiation. 4 Thermal properties are constant, However we can take the air to be a closed system by considering the air in the room to have undergone a constant pressure process. It is caused by a fluid flowing over a curved surface at a high velocity (or technically, by adverse pressure gradient). 2 The thermal properties of the wood slab are constant. The range for the rate of heat loss through the window of a house is to be determined. Noting that the volume element centered about the general interior node (m, n) involves heat conduction from four sides (right, left, top, and bottom) and expressing them at previous time step i, the transient explicit finite difference formulation for a general interior node can be expressed as k ($\Delta y \times 1$) Tmi -1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi -1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi -1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi -1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi -1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n $\Delta x + k$ ($\Delta x \times 1$) Tmi +1, n - Tmi , n + k ($\Delta x \times 1$) Tmi +1, n - Tmi , n + k (\Delta x \times 1) Tmi, n - 1 - Tmi, $n \Delta y - Tmi$, $n \Delta t$ Taking a square mesh ($\Delta x = \Delta y = l$) and dividing each term by k gives, after simplifying, Tmi - 1, n + Tmi, n - 1 - 4Tmi, n + Tmi, n - 1 - 4Tmi, n - 1propagate through the 0.3 m thick wall in 3 h, and thus it may be desirable to insulate the outer surface of the wall to save energy. (b) Low relative humidity (dry) environments also retard the growth of microorganisms by depriving them of water that they need to grow. Properties The properties of the rib are given to be k = 0.45 W/m. °C, $\rho = 1200$ kg/m3, Cp = 4.1 kJ/kg.°C, and α = 0.91×10-7 m2/s. Analysis The characteristic length of the balls and the Biot number are Lc = πD 3 / 6 D 0.0013 m As 6 6 πD 2 Furnace hL (75 W/m. °C) 2 Steel balls 900°C Air, 35°C Therefore, the lumped system analysis is applicable Assumptions 1 Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. 66 () Analysis The Reynolds number is V D [(200/60) m/s](0.2 m) Re = $\infty = 2.287 \times 10.4 \text{ v} 2.915 \times 10.4$ determined to be hD Nu = 0.102 Re 0.675 Pr 1/3 = 0.102(2.287) 0.675 (0.7235)1/3 = 80.21 k The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C}$ k h = Nu = $(80.21) = 10.90 \text{ W/m } 2 \cdot \text{C} D 0.2 \text{ m}$ Then the rate of heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C}$ k h = Nu = $(80.21) = 10.90 \text{ W/m } 2 \cdot \text{C} D 0.2 \text{ m}$ Then the rate of heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C}$ k h = Nu = $(80.21) = 10.90 \text{ W/m } 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m } 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m } 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m } 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m} 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m} 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m} 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m} 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m} 2 \cdot \text{C} D 0.2 \text{ m}$ The heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m} 2 \cdot \text{C} D 0.2 \text{ m}$ heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} \text{ k}$ h = Nu = $(80.21) = 10.90 \text{ W/m} 2 \cdot \text{C} D 0.2 \text{ m}$ heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} D 0.2 \text{ m}$ heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} D 0.2 \text{ m}$ heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} D 0.2 \text{ m}$ heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} D 0.2 \text{ m}$ heat transfer coefficient is $0.02717 \text{ W/m} \cdot \text{C} D 0.2 \text{ m}$ heat transfer coefficient is 0.0457.7 W 7-48 20 cm 65°C Chapter 7 External Forced Convection 7-57 A cylindrical electronic component mounted on a circuit board is cooled by air flowing across it. After 10 minutes: The Biot number, the corresponding constants, and the Fourier number are Bi = hL (40 W/m 2.°C)(0.025 m) = 0.400 $\rightarrow \lambda$ 1 = 0.5932 and A1 = 10580. / 12 ft)(1 ft ft)(1 ft ft)(1 ft ft)(1 = 0.916 ft 2 Ao = π Do L = π (8 / 12 ft)(1 ft) = 2.094 ft 2 The individual resistances are Ri Rpipe Rinsulation T ∞ 1 Ro T ∞ 2 1 1 = = 0.036 h · °F/Btu 2 π k1 L 2 π (8.7 Btu/h.ft.°F)(1 ft) Ri = R 2 = Rinsulation = ln(r3 / r2) ln(4 / 2) = = 5.516 h · °F/Btu 2 π k 2 L 2 π (0.020 Btu/h.ft.°F)(1 ft) 1 1 = = 0.096 h · °F/Btu 2 o ho Ao (5 Btu/h.ft. We note that all nodes are boundary nodes except node 5 that is an interior node. The 10-cm thick insulation will come very close to paying for itself in one year. Therefore, the wall can be considered to be a semi-infinite medium with a specified surface temperature of 45°C. °C L Add Tave (9.12 m 2)(17 - 6)° C The total number of hours this refrigerator remains on per year is $\Delta t = 365 \times 24 / 4 = 2190$ h Then the total amount of electricity Usage = W& $\Delta t = (0.6 \text{ kW})(2190 \text{ h/yr}) = 1314 \text{ kWh/yr}$ e Annual cost = (1314 kWh/yr)(\$0.08 / kWh) = \$105.1/yr 1-74 Chapter 1 Basics of Heat Transfer 1-132 A 0.2-L glass of water at 20°C is to be cooled with ice to 5°C. They are of great value in engineering practice, however, as engineers today rely on software packages for solving large and complex problems in a short time, and perform optimization studies efficiently. 23° F Analysis The average heat transfer coefficient Meat during this cooling process
is determined from the 50°F transient temperature charts for a flat plate as follows: $\tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | \tau = 0.481 | Ti - T \approx 50 - 23 | Ti - T \approx 50 -$ (1/0.7) = 1.5 Btu/h.ft². The Reynolds number in this case is V L [(80 × 1000 / 3600) m/s](0.8 m) Re L = $\infty = 9.888 \times 10.5 2 - 5 \upsilon 1.798 \times 10$ m/s which is less than the critical Reynolds number. Properties of air, water, and oil at 40°C are (Tables A-15, A-9, A-13) Air: $\mu = 1.918 \times 10-5$ N-s/m2 Water: $\mu = 0.653 \times 10-3$ N-s/m2 Oil: 2500 rpm $\mu = 0.212$ N-s/m2 12 m/s Analysis A shaft rotating in a bearing can be approximated as parallel flow between two large plates with one plate moving and the other stationary. 3 The plant operates every day of the year for 10 h a day. 3-4C The thermal resistance of a medium represents the resistance of that medium against heat transfer. Assumptions 1 85 percent of the heat generated by a L=0.5 mm thick fluid film similar to the problem given in Example 6-1. Analysis (a) This hot dog can physically be formed by the intersection of a long cylinder of radius ro = D/2 = (0.4/12) ft and a plane wall of thickness 2L = (5/12) ft. The area of the cylinder normal to the direction of heat transfer at any location. m) $2\tau L2$ (0.783)(0115 = 79,650 s = 22.1 h - 6 \alpha 0.13 \times 10 m 2 / s The lowest temperature during cooling will occur on the surface (x/L) ft. = 1), and is determined to be 2 T (x) - T ∞ T (L) - $T \infty$ Ti - T ∞ Ti - TA linear differential equation that involves a single term with the derivatives can be solved by direct integration. 3-66C No. In steady-operation the temperature of a solid cylinder or sphere does not change in radial direction (unless there is heat generation). 7-66 Combustion air is heated by condensing steam in a tube bank. and A1 = 10038. Analysis The center temperature of the rod is determined from To = T s + g&ro 2 (7 × 10 7 W/m 3)(0.025 m) 2 = 175°C + = 545.8 °C 4k 4(29.5 W/m.°C) 2.43 Chapter 2 Heat Conduction Equation 2.83 Both sides of a large stainless steel plate in which heat is generated uniformly are exposed to convection with the environment. (Note: J 0 is read from Table 4-2). o C)(1.2 × 1.8)m 2 R wall = Rglass Rair R window = 2 Rglass + Rair = 2 × 0.002968 + 0.267094 = 0.27303 °C/W 1 1 1 1 1 = +5 = +5 \rightarrow Reqv = 0.020717 °C/W Reqv R wall R window 0.033382 0.27303 R total = Ri + Reqv + Ro = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 °C/W Reqv R wall R window 0.033382 0.27303 R total = Ri + Reqv + Ro = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 °C/W Reqv R wall R window 0.033382 0.27303 R total = Ri + Reqv + Ro = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.020717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.000833 = 0.001786 + 0.000717 + 0.0008376 + 0.000717 + 0.0008376 + 0.000717 + 0.0008376 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0.000717 + 0. 0.023336 °C/W Then T -T (22 - 8) °C Q& = $\infty 1 \propto 2 = 600$ W R total 0.023336 °C/W The rate of heat transfer which will be saved if the single pane windows are converted to double pane windows are converted to double = 4372 - 600 = 3772 W pane pane The amount of energy and money saved during a 7-month long heating season by switching from single pane to double pane windows become $Q = Q \& \Delta t = (3.772 \text{ kW})(7 \times 30 \times 24 \text{ h}) = 19,011 \text{ kWh}$ save save Money savings = (Energy saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed of fiberglass insulation sandwiched between the save money saved)(Unit cost of energy) = (19,011 \text{ kWh})(\$0.08/kWh) = \$1521 3-17 \text{ Chapter 3 Steady Heat Conduction 3-35 The wall of a refrigerator is constructed between the save money sav two layers of sheet metal. 1-6 Chapter 1 Basics of Heat Transfer 1-24C Mass flow rate m& is the amount of mass flowing through a cross-section per unit time. It is tempting to simplify the problem further by considering heat transfer in each wall to be one dimensional which would be the case if the walls were thin and thus the corner effects were negligible. m2 Analysis The representative surface area is A = 012 R2 R1 R5 R3 R7 R8 R6 R4 (a) The thermal resistance network and the individual thermal resistance area is A = 0.01 m (L) R1 = R A = | | = 0.04 °C/W 2 (kA) A (2 W/m.°C)(0.12 m) $0.05 \text{ m}(L) = 0.06 \text{ °C/W R } 2 = \text{R } 4 = \text{RC} = || = 2 \text{ kA } \text{ C} (20 \text{ W/m.°C})(0.04 \text{ m}) 0.05 \text{ m}(L) \text{ R3} = \text{R } B = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2
\text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } D = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } B = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } B = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } B = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } B = || = 0.16 \text{ °C/W } 2 \text{ kA } B (8 \text{ W/m.°C})(0.04 \text{ m}) 0.1 \text{ m}(L) \text{ R5} = \text{R } B = || = 0.16 \text{ °C/W } 2 \text{ m}(R) = 0.16 \text{ m}(R) = 0.16 \text{ m}(R) = 0.16 \text{ m}(R) = 0.16 \text$ $kA = 0.001 \text{ m}^{\circ} \text{C/W} = 0.001 \text{ m}^{\circ$ 100)°C Q& = $\infty 1 = 571$ W (for a 0.12 m × 1 m section) Rtotal 0.350 °C/W Then steady rate of heat transfer through entire wall becomes (5 m)(8 m) Q& total = (571 W) = 1.90 × 10 5 W 0.12 m 2 3-36 Chapter 3 Steady Heat Conduction (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 \rightarrow T = T1 – Q& Rtotal = 300°C – (571 W)(0.25 °C/W) = 143°C RF 3-59 A coat is made of 5 layers of 0.1 mm thick synthetic fabric separated by 1.5 mm thick air space. The temperature charts) are applicable of the insulated surface is to be determined. $^{\circ}$ C) Note that both approaches give the same result. 4 The Fourier number is $\tau > 0.2$ so that the oneterm approximate solutions (or the transient temperature charts) are applicable. (this assumption will be verified). 4 Properties of water is used for orange. Properties The thermal conductivity and no heat generation will vary linearly during steady one-dimensional heat conduction even when the wall loses heat by radiation from its surfaces. 5-104 Chapter 5 Numerical Methods in Heat Conduction Review Problems 5-106 Starting with an energy balance on a volume element, the steady three-dimensional finite difference equation for a general interior node in rectangular coordinates for T(x, y, z) for the case of constant thermal conductivity and uniform heat generation is to be obtained. 4 Heat transfer from the bottom surface of the box to the stand is negligible. Properties The thermal conductivity of a thick towel is given to be k = 0.035 Btu/h·ft·°F. energies & 1 = mh & 2 (since $\Delta ke \cong \Delta pe \cong 0$) W&e,in = m& (h2 - h1) = m& [C (T2 - T1) + v (mathebet) + v (mathebet) + w (mathebet) P2 - P1) E0] = mC where m& = $\rho V \& = (1 \text{ kg/L})(10 \text{ L/min}) = 10 \text{ kg/min Substituting}$, $W\&e, in = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot °C)(43 - 16) \circ C = 18.8 \text{ kW } 16^{\circ} C$ WATER 43°C The energy recovered by the heat exchanger is $Q\&e = \varepsilon Q\&e = \varepsilon Q&e = \varepsilon$ less energy is needed in this case, and the required electric power in this case reduces to W& = W& - Q& = 18.8 - 8.0 = 10.8 kW in, new in, old saved The money saved during a 10-min shower as a result of installing this heat exchanger is (8.0 kW) (10/60 h) (8.5 cents/kWh) = 11.3 cents 1-69 Chapter 1 Basics of Heat Transfer 1-125 Water is to be heated steadily from 15°C to 50°C by an electrical resistor inside an insulated pipe. Analysis (a) The total rate of heat transfer dissipated by the chips is Q& = $80 \times (0.04 \text{ W}) = 3.2 \text{ W} 2 \text{ cm}$ The individual resistances are Rboard Repoxy Rcopper 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. However, we can solve this problem approximately by assuming a constant average temperature of $(3+10)/2 = 6.5^{\circ}$ C during the process. hL (48.43 W/m 2.°C)(0.1 m) = = 180.6 k 0.02681 W/m.°C Nu 2 (180.6) 2 Nu = Re L = Re L v (9.171 × 10 4 0.664 2 Pr 2 / 3 (0.664) 2 (0.7248) 2 / 3 Nu = Re L = Re L v (9.171 × 10 4)(1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1.726 × 10.40) (1 10 - 5 m 2 /s V $\infty L \rightarrow V \infty = = 15.83 \text{ m/s } L 0.1 \text{ m } v$ 7-20 Chapter 7 External Forced Convection 7-29 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. 5 The flow is turbulent over the entire surface because of the constant agitation of the engine block. Applying the boundary conditions give C1 = -q &0 kx = 0: $- \text{ kC1} = q \& 0 \rightarrow x = L$: $4] - \text{kC1} = h[T2 - T\infty] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2)] + \varepsilon\sigma [(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the known quarter (T2 + 273) + \varepsilon\sigma (T2 + 273) + \varepsilon\sigma (T2 + 273) + \varepsilon\sigma ($ °C)(T2 - 22) + 0.7(5.67 × 10 - 8 W/m 2 · K 4)[(T2 + 273) 4 - 290 4] = 66,667 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 = 758°C 2-70 Chapter 2 Heat Conduction Equation 2-130 The base plate of an iron is subjected to specified heat flux on the left surface and convection and radiation on the right surface. Analysis This cylindrical aluminum block can physically be formed by the intersection to the floor through the feet is negligible. Then the heat transfer coefficient can be determined from Bi = hro kBi (0.618 W/m.°C)(10) \rightarrow h = = 61.8 W/m 2.°C k ro (0.10 m) The temperature at the surface of the watermelon is 2 sin(λ r / r) 2 T (ro, t) - T \propto sin(2.8363 rad) o 1 o = A1 e $-\lambda$ 1 τ = (1.9249)e - (2.8363) (0.252) θ (ro, t) sph = Ti - T $\propto \lambda$ 1 ro / ro 2.8363 T (ro, t) - 15 = 0.0269 \rightarrow T (ro, t) = 15.5 °C 35 - 15 4-113 Chapter 4 Transient Heat Conduction 4-120 Large food slabs are cooled in a refrigeration room. at r = r0: Ts = $-g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2
= Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 4k \rightarrow C2 = Ts + g\& 2 r0 + C2 + G\& 2$ number for this process is Bi = hro (1400 W / m 2. This system of 4 equations with 4 unknowns constitute the finite difference formulation of the problem. The temperature at the top corner (node #3) of the body after 2, 5, and 30 min is to be determined with the transient explicit finite difference method. 7 Heat generation due to absorption of 123E A large plane wall is subjected to a specified temperature on the left (inner) surface and solar radiation and heat loss by radiation to space on the right (outer) surface. 4 in 2 . 3-105C The fin with the lower heat transfer coefficient will have the higher effectiveness. Also, $\chi = 1$ for the square arrangements. Next we need to determine the upper limit of the time step Δt from the stability criteria since we are using the explicit method. The amount of heat transfer, the average heat flux, and the number of valves that can be heat transfer, the average heat flux, and the number of valves that can be heat transfer. determined to be t = $(5.601)(0.01 \text{ m}) 2 \tau L2 = 6155 \text{ s} = 102.6 \text{ min } \alpha (0.91 \times 10 - 7 \text{ m} 2/\text{s}) 4-30 \text{ Chapter 4 Transient Heat Conduction 4-42 A long cylindrical wood log is exposed to hot gases in a fireplace. Analysis (a) The boiling heat transfer coefficient is As = <math>\pi D 2 \pi (0.25 \text{ m}) 2 = 0.0491 \text{ m} 2 4 4 \text{ Q} \text{ } \text{ = } 1254 \text{ W/m 2}$ $^{\circ}C$ h= As (Ts – T $^{\infty}$) (0.0491 m 2)(108 – 95) $^{\circ}C$ (b) The outer surface temperature of the bottom of the pan is Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ Ts, outer – Ts, inner Q& = kA L (800 W)(0.005 m) Q& L = 108 $^{\circ}C$ + = 108.3 $^{\circ}C$ + = of sheetrock with fiberglass insulation in between. 3-136C The unit thermal resistances (R-value) of both 40-mm and 90-mm vertical air space in a wall has no effect on heat transfer through the wall. 7-16 Chapter 7 External Forced Convection 7-25 Laminar flow of a fluid over a flat plate is considered. Applying the boundary conditions give Heat flux at x = 0: -kC1 = -Temperature at x = 0: $T(0) = C1 \times 0 + C2 = T1$ geven the general solution, the variation of temperature is determined to be T (x) = $-q \otimes 0.950$ W/m 2 x + T1 = $-x + 85^{\circ}C = -380$ x + 85 k 2.5 W/m · °C (c) The temperature at x = L (the right surface of the wall) is T (L) = $-380 \times (0.3 \text{ m}) + 85 = -29^{\circ}\text{C}$ Note that the limestone layer will (a decline of 40%) Q& w/lime = $\infty 1$ R total, w/lime 0.00832 h °F/Btu Discussion Note that the limestone layer will change the inner surface area of the pipe and thus the internal convection resistance slightly, but this effect should be negligible. 3 Constant specific heats at room temperature can be used for air, Cp = 0.2404 and Cv = 0.1719 Btu/lbm·R. Analysis The heat flux at the bottom of the pan is Q& G& 0.85 × (1000 W) q& s = s = = 27,056 W / m 2.2 As $\pi D / 4 \pi$ (0.20 m) 2 / 4 Then the differential equation and the boundary conditions for this heat conduction problem can be expressed as d 2T = 0 dx 2 dT (0) - k = h[T (L) - T\infty] dr 2-14 Chapter 2 Heat Conduction Equation 2-46E A 1.5-kW resistance heater wire is used for space heating. ° C)(0.0001 m) = 0.0386 W/° C Copper (kt) epoxy = $(0.26 \text{ W} / \text{m}. \text{ In this case it is determined to be } \Delta t defrost = N\Delta t = 44(5 \text{ s}) = 220 \text{ s} 5-117 \text{ Chapter 5 Numerical Methods in Heat Conduction } 5-120 \text{ Frozen steaks at } -18°C \text{ are to be defrosted by placing them on a 1-cm thick black-anodized circular copper defrosting plate. of days} = (2.361 \times 105 \text{ kJ/day})(365 \text{ days/yr}) = 8.619 \text{ s}^{-1} \text{ s}^{-1} \text{ c}^{-1} \text{ s}^{-1} \text{ s}$ \times 10 7 kJ/yr Noting that the steam generator has an efficiency of 80%, the amount of gas used is Qtotal 8.619 \times 10 7 kJ/yr (1 therm) = || || = 1021 therms/yr 0.80 0.80 (105,500 kJ) Insulation reduces this amount by 90 %. Properties The properties of hot dog and the convection heat transfer coefficient are given or obtained in P4-47 to be k = 0.771 W/m.°C, $\rho = 980$ kg/m3, Cp = 3900 J/kg.°C, $\alpha = 2.017 \times 10^{-7}$ m2/s, and h = 467 W/m2.°C. The time of cooking is to be determined. Freezing extends the storage life of foods for months by preventing the growths of microorganisms. Properties The density and specific heat of water at room temperature are $\rho = 62.22$ lbm/ft3, and Cp = 0.999 Btu/lbm.°F (Table A-9E). Again assuming the temperatures between the adjacent nodes to vary linearly and noting that the heat transfer area is $\Delta y \times 1$ in the x direction and $\Delta x \times 1$ in the y direction, the energy balance relation above becomes k m, n ($\Delta y \times 1$) Tm -1, n - Tm, n $\Delta x + k$ m, n ($\Delta x \times 1$) Tm, n + k m, n ($\Delta y \times 1$) Tm +1, n - Tm, n $\Delta x \times 1$ in the y direction and $\Delta x \times 1$ in the y direction. $\Delta y \Delta x Tm$, n - 1 - Tm, n + km, $n (\Delta x \times 1) + g \& 0$ ($\Delta x \times \Delta y \times 1$) = 0 Δy Dividing each term by $\Delta x \times \Delta y \times 1$ and simplifying gives Tm - 1, n - Tm, n + Tm, n - 1 - 2Tm, n - 1 - 2Tmg& 0 1 2 =0 k m,n It can also be expressed in the following easy-to-remember form: Tleft + Ttop + Tright + Tbottom - 4Tnode + g& 0 1 2 =0 k node 5-41 Chapter 5 Numerical Methods in Heat Conduction 5-50 A long solid body is subjected to steady two-dimensional heat transfer. We take the properties of potato to be those of water at room temperature, $\rho = 62.2$ lbm/ft3 and Cp = 0.998 Btu/lbm.°F. That is, Q& = Q& roof, cond = Q& roof to surroundings, conv+rad The inner surface temperature of the roof is given to be Ts, in = 15°C. 2 Heat transfer is one-dimensional., and there is thermal symmetry about the center point. Analysis This cylindrical aluminum block can physically be formed by the intersection of an infinite plane wall of thickness 2L = 20 cm, and a long cylinder of radius ro = D/2 = 7.5 cm. 5-111 Chapter 5 Numerical Methods in Heat transfer coefficient is constant and uniform over the plate. Analysis The initial and final masses of the raindrop are $4\ 3\ 4\ \pi ri = (1000\ \text{kg/m}\ 3)\ \pi (0.0025\ \text{m})\ 3 = 0.0000654\ \text{kg}\ 3\ 3\ 4\ 3\ 4\ \text{m}\ f = \rho V\ f = \rho\ \text{m}\ f = (1000\ \text{kg/m}\ 3)\ \pi (0.0015\ \text{m})\ 3 = 0.0000654\ \text{kg}\ 3\ 3\ 4\ 3\ 4\ \text{m}\ f = \rho V\ f = \rho\ \text{m}\ f = (1000\ \text{kg/m}\ 3)\ \pi (0.0015\ \text{m})\ 3 = 0.0000654\ \text{kg}\ 3\ 3\ 4\ 3\ 4\ \text{m}\ f = \rho V\ f = \rho\ \text{m}\ f = (1000\ \text{kg/m}\ 3)\ \pi (0.0015\ \text{m})\ 3 = 0.0000654\ \text{kg}\ 3\ 3\ 4\ 3\ 4\ \text{m}\ f = \rho V\ f = \rho\ \text{m}\ f = (1000\ \text{kg/m}\ 3)\ \pi (0.0015\ \text{m})\ 3 = 0.0000654\ \text{kg}\ 3\ 4\ 3\ 4\ \text{m}\ f = \rho V\ f = \rho\ \text{m}\ f = (1000\ \text{kg/m}\ 3)\ \pi (0.0015\ \text{m})\ 3 = 0.0000654\ \text{kg}\ 3\ 3\ 4\ 3\ 4\ \text{m}\ f = \rho V\ f = \rho\ \text{m}\ f = (1000\ \text{kg/m}\ 3)\ \pi (0.0015\ \text{m})\ 3 = 0.00000654\ \text{kg}\ 3\ 3\ 4\ 3\ 4\ \text{m}\ f = \rho V\ f = \rho\ \text{m}\ f = (1000\ \text{kg/m}\ 3)\ \pi (0.0015\ \text{m}\ 3)\ \pi (0.0$ this much evaporation is Q = (0.0000513 kg)(2490 kJ/kg) = 0.1278 kJ The average heat transfer are 4π (ri 2 + r f 2) 4π [(0.0025 m) $2 + (0.0015 \text{ m}) 2 = 5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $-
T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m} 2 2 2 Q \& = hAs$ (Ti $- T \propto$) = (400 W/m $2 \cdot ^{\circ}C)(5.341 \times 10 - 5 \text{ m}$ experience this reduction in size becomes Q Q 127.8 J Q& = $\rightarrow \Delta t = = 460 \text{ s} = 7.7 \text{ min } \& \Delta t \text{ Q} 0.2777 \text{ J/s} 4-105 \text{ Raindrop 5°C Chapter 4 Transient Heat Conduction 4-115E A plate, a long cylinder, and a sphere are exposed to cool air. 5 The entire heat generated by the resistance heaters is transferred through the plate. This problem involves 6$ unknown nodal temperatures, and thus we need to have 6 equations to determine them uniquely. (b) The temperature drop at the interface = Q& Rcontact = $(142.4 \text{ W})(0.0447 \text{ °C/W}) = 6.4 \text{°C} 3.25 \text{ Chapter 3 Steady Heat Conduction 3-48 A thin copper plate is sandwiched between two epoxy boards. Noting that Pr >> 1$ for oils, the thermal entry length is larger than the hydrodynamic entry length in laminar flow. Therefore, T1 = T3 = T7 = T9 and g 4 T2 = T4 = T6 = T8, and T1, T2, and T5 are the only 3 unknown nodal h, $T\infty \cdot \cdot 5 \cdot 6$ h, $T\infty$ temperatures, and thus we need only 3 equations to determine them uniquely. The rate of heat transfer through the shell is to be determined. Assumptions 1 Steady operating conditions exist since the temperature readings do not change with time. Then the time for a final valve temperature of 400°C becomes $b = hAs 8(650 \text{ W/m } 2.^{\circ}\text{C}) 8h = = 0.10468 \text{ s} - 1 \rho \text{C} \text{ p} \text{ V} 1.8 \rho \text{C} \text{ p} \text{ D} 1.8(7840 \text{ kg/m } 3)(440 \text{ J/kg.}^{\circ}\text{C})(0.008 \text{ m}) - 1 \text{ T} (t) - \text{T} \approx 400 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s})t \rightarrow t$ $= 7.2 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (b) The time for a final valve temperature of 200°C is -1 T (t) $-T_{\infty} 200 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s})t \rightarrow t = 63.3 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (c) The time for a final valve temperature of 46°C is -1 T (t) $-T_{\infty} 46 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s})t \rightarrow t = 63.3 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (c) The time for a final valve temperature of 46°C is -1 T (t) $-T_{\infty} 46 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s })t \rightarrow t = 63.3 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (c) The time for a final valve temperature of 46°C is -1 T (t) $-T_{\infty} 46 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s })t \rightarrow t = 63.3 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (c) The time for a final valve temperature of 46°C is -1 T (t) $-T_{\infty} 46 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s })t \rightarrow t = 63.3 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (c) The time for a final valve temperature of 46°C is -1 T (c) $-T_{\infty} 46 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s })t \rightarrow t = 63.3 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (c) The time for a final valve temperature of 46°C is -1 T (c) $-T_{\infty} 46 - 45 = e - bt \rightarrow = e - (0.10468 \text{ s })t \rightarrow t = 63.3 \text{ s } 800 - 45 \text{ Ti} - T_{\infty}$ (d) The maximum amount of heat transfer from a final valve temperature of 46°C is -1 T (b) -1 T (c) -1single valve is determined from $1.8\pi(0.008 \text{ m}) 2 (0.10 \text{ m}) 1.8\pi D 2 \text{ L} = (7840 \text{ kg/m } 3) = 0.0709 \text{ kg} 4 4 \text{ Q} = \text{mC p} [T \text{ f} - \text{Ti}] = (0.0709 \text{ kg})(440 \text{ J/kg}.^{\circ}\text{C})(800 - 45)^{\circ}\text{C} = 23,564 \text{ J} = 23.56 \text{ kJ} (\text{per valve}) \text{ m} = \rho \text{V} = \rho 4-112 \text{ Chapter 4 Transient Heat Conduction 4-119 A watermelon is placed into a lake to cool it. } C)(150 - 40)^{\circ}\text{C} = 23,564 \text{ J} = 23.56 \text{ kJ} (\text{per valve}) \text{ m} = \rho \text{V} = \rho 4-112 \text{ Chapter 4 Transient Heat Conduction 4-119 A watermelon is placed into a lake to cool it. } C)(150 - 40)^{\circ}\text{C} = 23,564 \text{ J} = 23.56 \text{ kJ} (\text{per valve}) \text{ m} = \rho \text{V} = \rho 4-112 \text{ Chapter 4 Transient Heat Conduction 4-119 A watermelon is placed into a lake to cool it. } C)(150 - 40)^{\circ}\text{C} = 23,564 \text{ J} = 23.56 \text{ kJ} (\text{per valve}) \text{ m} = \rho \text{V} = \rho 4-112 \text{ Chapter 4 Transient Heat Conduction 4-119 A watermelon is placed into a lake to cool it. } C)(150 - 40)^{\circ}\text{C} = 23,564 \text{ J} = 23.56 \text{ kJ} (\text{per valve}) \text{ m} = \rho \text{V} = \rho$ to - center distance of pipes) n 10 S = 10.91 m = $2\pi L$ w 2 $2\pi z$ / (ln | sinh | w / π 2 πz - 2.5 cm 3-127 Chapter 3 Steady Heat Conduction 3-167 Two persons are wearing different clothes made of different surface areas. 3-106 Chapter 3 Steady Heat Conduction 3-167 Two persons are wearing different clothes made of different surface areas. Steady Heat Conduction 3-144 The winter R-value and the U-factor of a masonry cavity wall with a reflective surface are to be determined. The thermal properties of the 5 6 • 4 • • defrosting plate are k = 237 W/m.°C, $\alpha = 97.1 \times 10 - 6$ m2 / s, and $\epsilon = 0.90$. Properties The pecific heat and density of valves are given to be $Cp = 440 \text{ J/kg.}^{\circ}C$ and $\rho = 7840 \text{ kg/m3}$. Then the total rate of heat gain by the water is Q& water = Q& chicken + Q& heat gain = 13.0 + 35 . The average Refrigerat specific heat of food items is given to be 3.6 kJ/kg. \circ C. The thermal symmetry boundary condition is a mathe of this thermal symmetry. 4 Convection heat transfer is negligible. Analysis Taking the temperature of the fin at the base to be Tb and using the heat transfer rate from the fin Ideal heat transfer rate from the fin if the entire fin were at base temperature hpkAc (Tb - T ∞) hA fin (Tb - T ∞) = hpkAc hpL 1 = L h, T ∞ Tb kAc ph p= π D Ac = π D2/4 This relation can be simplified for a circular fin of thickness t and width w to be η fin, circular = 1 L kAc 1 = ph L k (π D 2 / 4) 1 = (π D)h 2L η fin, rectangular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w to be η fin, circular fin of thickness t and width w
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Assumptions 1 The turkey is a homogeneous spherical object. 5-100 C The Taylor series expansion of the temperature at a specified nodal point m about time ti is T (xm, ti + Δt) = T (xm, ti) + $\Delta t \partial T$ (xm, ti) + ti) 1 2 ∂ 2T (xm, ti) + Δ t + L ∂ t 2 ∂ t 2 The finite difference formulation of the time derivative at the same nodal point is expressed as ∂ T (xm, ti) + Δ t \cong = ∂ t Δ t Δ t which resembles the Taylor series expansion terminated after the first two terms. Assumptions 1 Heat transfer is given to be transient and one-dimensional. Analysis We consider transient one-dimensional heat conduction in the axial z direction in an insulated cylindrical rod of constant cross-sectional area A with constant conductivity k with a mesh size of Δz in the z direction. in the plate, and be Δx2=0.6 ft in the soil. The significant assumptions in solving the problem are stated. The density of the sphere is ρ , the specific heat is C, and thus the local discretization error is also proportional to (Δt)2. The thermal conductivities are given to be k = 0.035 $W/m \circ C$ for fiberglass insulation and $k = 0.00005 W/m \circ C$ for super insulation. Heat is lost from the room. m) = 4.89 0.47 W/m. Using the Nusselt number uniform heat flux, the local heat transfer coefficient at the end of the board is determined to be h x Nu x = x = 0.0308 Re x 0.8 Pr 1 / 3 = 0.0308(4.532 \times 104) 0.8 (0.7268)1 / 3 = 147.0 k k 0.02625 W/m.°C h x = x Nu x = (147.0) = 25.73 W/m 2.°C x 0.15 m Then the surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for isothermal surfaces, hx x = 0.0296 Re determined approximately using the relation for $x 0.8 \text{ Pr} 1/3 = 0.0296(45.320)0.8 (0.7268)1/3 = 141.3 \text{ k} 0.02625 \text{ W/m}.^{\circ}C (141.3) = 24.73 \text{ W/m} 2$, $^{\circ}C \text{ hx} = x 0.15 \text{ m}$ Then the surface temperature at the end of the board becomes (15 W)/(0.15 m) 2 g & $\rightarrow T \text{ s} = T_{\infty} + 20^{\circ}C \text{ s} = h x (T \text{ s} - T_{\infty})$ hx 24.73 W/m 2. $^{\circ}C \text{ hx} = x 0.15 \text{ m}$ Then the surface temperature at the end of the board becomes (15 W)/(0.15 m) 2 g & $\rightarrow T \text{ s} = T_{\infty} + 20^{\circ}C \text{ s} = h x (T \text{ s} - T_{\infty})$ hx 24.73 W/m 2. $^{\circ}C \text{ hx} = x 0.15 \text{ m}$ Then the surface temperature at the end of the board becomes (15 W)/(0.15 m) 2 g & $\rightarrow T \text{ s} = T_{\infty} + 20^{\circ}C \text{ s} = h x (T \text{ s} - T_{\infty})$ hx 24.73 W/m 2. $^{\circ}C \text{ hx} = x 0.15 \text{ m}$ Then the surface temperature at the end of the board becomes (15 W)/(0.15 m) 2 g & $\rightarrow T \text{ s} = T_{\infty} + 20^{\circ}C \text{ s} = 14.3 \text{ k} \text{ s} = 0.02625 \text{ W/m}$. use combined laminar and turbulent flow relation for Nusselt number" Nusselt = $(0.037*\text{Re}^{0.8-871})*\text{Pr}^{(1/3)} h = k/L*\text{Nusselt q}$ dot conv=h*(T s-T infinity) q dot conv=h*(T s 35.13 34.83 34.58 34.35 34.14 33.96 33.79 33.64 33.5 33.37 33.25 7-13 Chapter 7 External Forced Convection Qrad [W/m2] 100 125 150 175 200 225 250 275 300 325 350 375 400 425 450 475 500 Ts [C] 32.56 33.2 33.84 34.48 35.13 35.77 36.42 37.07 37.71 38.36 39.01 39.66 40.31 40.97 41.62 42.27 42.93 7-14 Chapter 7 External Forced Convection 65 60 55 T s [C] 50 45 40 35 30 0 20 40 60 80 100 120 Vel [km /h] 44 42 T s [C] 40 38 36 34 32 100 150 200 250 300 350 2 q rad [W /m] 7-15 400 450 500 Chapter 7 External Forced Convection 7-24 A circuit board is cooled by air. 2-6 Chapter 2 Heat Conduction Equation 2-22 We consider a thin cylindrical shell element of thickness Δr in a long cylinder (see Fig. ° C)[π (0.03 m) 2](25 - 6.5)° C = 0.52 W Heat transfer through the insulated side surface is Ao = π Do L = π (0.08 m)(0125. ° F)(0.4 / 12 ft) = 9.1 k (0.44 Btu / h.ft. Noting that the volume element centered about the general interior node (m, n, r) involves heat conduction from six sides Δy (right, left, front, rear, top, and bottom) and expressing m-1 • them at previous time step i, the transient explicit finite Δx difference formulation for a general interior node can be • n expressed as k ($\Delta y \times \Delta z$) – Tmi , n, r Δy + k ($\Delta x \times \Delta z$) – Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy + k ($\Delta x \times \Delta z$) – Tmi , n, r Δy + k ($\Delta x \times \Delta z$) – Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy + k ($\Delta x \times \Delta z$) – Tmi , n, r Δy + k ($\Delta x \times \Delta z$) – Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy + k ($\Delta x \times \Delta z$) – Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy Tmi +, n1 Tmi , n, r Δy $\times \Delta z$) - Tmi , n, r Δz Tmi +1, n, r - Tmi , n, r + k ($\Delta x \times \Delta y$) Δx Tmi , n, r + 1 - Tmi , n, r + the thermal diffusivity of the material and $\tau = \alpha \Delta t / l 2$ is the dimensionless mesh Fourier number. [4 - (-6)] = -4.3° C which gives Tsurface = T ∞ + 017 The difference between the two results is due to the reading error of the charts. 3 Heat is generated uniformly in the wire. When $\partial x \partial y$ multiplied by density, the first and the second terms represent net mass fluxes in the x and y directions, respectively. 3 The average surrounding surface temperature for radiation exchange is 15°C. Analysis The contact area between the case and the plate area for each transistor is 100 cm2. Assumptions 1 Both the water and the copper block are incompressible substances with constant specific heats at room temperature. The surface temperatures of the electronic components at the leading edge and the end of the board are to be determined. 3-122 The hot water pipe of a district heating system is buried in the soil. Discussion We could avoid the uncertainty associated with the reading of the charts and obtain a more accurate result by using the one-term solution relation for an infinite plane wall, but it would require a trial and error approach since the Bi number is not known. ° F)(0.000278 ft) 2 = 2.055 ft -1 Noting that x = L = 0.7/12=0.583 ft at the tip and substituting, the tip temperature of the spoon is determined to be cosh a(L - L) cosh aL cosh 0 1 = $75^{\circ}F + (200 - 75) = 75^{\circ}F + (200 - 75) = 144.1^{\circ}F \cosh(2.055 \times 0.583)$ 1.81 T (L) = T ∞ + (Tb - T ∞) Therefore, the temperature difference across the exposed section of the spoon handle is ΔT = Tb - Ttip = (200 - 144.1)^{\circ}C = 55.9^{\circ}F 3.80 in 0 Chapter 3 Steady Heat Conduction 3-113 "GIVEN" k spoon=8.7 "[Btu/h-ft-2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12
"[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[F]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[F]" T infinity=75 "[Ft]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[Ft]" T infinity=75 "[Ft]" A c=0.08/12*0.5/12 "[ft^2]" "L=7 [in], parameter to be varied" T w=200 "[Ft]" T infinity=75 "[ft]" T x=L "for tip temperature" DELTAT=T w-T tip kspoon [Btu/h.ft.F] 5 16.58 28.16 39.74 51.32 62.89 74.47 86.05 97.63 109.2 120.8 132.4 143.9 155.5 167.1 102 97.21 92.78 88.69 84.91 81.42 78.19 75.19 72.41 69.82 67.4 65.14 63.02 61.04 59.17 kspoon [Btu/h.ft.F] 5 5.5 6 6.5 7 7.5 8 Conduction 3-114 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. Assumptions 1 Heat transfer is steady since there is no indication of any changes with time. 2 The entire plate is nearly isothermal. 7-5C When the drag force FD, the upstream velocity V, and the fluid density ρ are measured during flow over a body, the drag coefficient can be determined from FD CD = 1 ρ V 2 A 2 where A is ordinarily the frontal area (the area projected on a plane normal to the direction of flow) of the body. The power required to rotate the shaft is to be determined for different fluids in the gap. 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 = $N\pi DL = 64\pi (0.021 \text{ m})(1 \text{ m}) = 4.222 \text{ m} 2 \text{ m} \& = m \& i = \rho i V(N \text{ T S T L}) = (1.225 \text{ kg/m 3})(3.8 \text{ m/s})(8)(0.05 \text{ m})(1 \text{ m}) = 4.222 \text{ m} 2 \text{ m} \& = m \& i = \rho i V(N \text{ T S T L}) = (1.225 \text{ kg/m 3})(3.8 \text{ m/s})(8)(0.05 \text{ m})(1 \text{ m}) = 1.862 \text{ kg/s}$ External Forced Convection (Ah Te = Ts - (Ts - Ti) exp - s m& Cp ($\Delta Tln = 22$) ($l = 90 - (90 - 15) exp - (4.222 m)(86.29 W/m \cdot °C)$ = 28.25°C (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/kg · °C) ($l = 28.25^{\circ}C$ (1.862 kg/s)(1007 J/k m 2)(68.16°C) = 24,834 W For this staggered tube bank, the friction coefficient corresponding to ReD = 9075 and ST/D = 5/2.1 = 2.38 is, from Fig. 3 4 5 Convection Assumptions 1 Heat transfer through the wall is given h, T ∞ to be steady and one-dimensional. Then, hcombined = hrad + hconv, 2 = 5.167 + 15 = 20.167 W/m 2. °C 1 1 = 0.0229°C/W hcombined Ao (20.167 W/m 2.°C)(2.168 m 2) = Ri + R pipe + Ro = 0.0044 + 0.00003 + 0.0229 = 0.0273 °C/W Ro = Rtotal The rate of heat loss from the hot water pipe then becomes T – T (90 – 10)°C = 2927 W Q& = $\infty 1 \propto 2$ = Rtotal 0.0273 °C/W For a temperature drop of 3°C, the mass flow rate of water and the average velocity of water must be $O_{k} = m_{k} C p \Delta T \rightarrow m_{k} = m_{k} = \rho VAC \rightarrow V = O_{k} 2927 I/s = 0.233 kg/s C p \Delta T (4180 I/kg.^{\circ}C)(3^{\circ}C) m_{k} = \rho Ac 0.233 kg/s \pi (0.04 m) 2 (1000 kg/m) 4 = 0.186 m/s 3 Discussion The outer surface temperature of the pipe is T - Ts (90 - Ts)^{\circ}C O_{k} = \infty 1 \rightarrow 2927 W = \rightarrow Ts = 77^{\circ}C Ri + R pipe (0.0044 + 0.00003)^{\circ}C/W$ which is very close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Assumptions 1 Heat conduction in the slabs is one-dimensional since the slab is large relative to its thickness and there is thermal symmetry about the center plane. the pipe and on the outer surface temperature are to be obtained for steady one-dimensional heat transfer. Then the maximum velocity become Vmax ST 0.03 = V = (0.8 m/s) = 1.20 m/s (0.01 m) = 18,232 µ $0.653 \times 10 - 3$ kg/m · s Ts=90°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 (4.32 / 1.96) 0.25 = 269.3 Assuming that 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 = 269.3 h = Nu D, N L k D = 269.3 (0.631 W/m \cdot °C) = 16,994 W/m 2 \cdot °C 0.01 m Consider one-row of tubes in the transpose direction (normal to flow), and thus take NT = 1.2 Thermal properties of the glass are constant. 2-64 Chapter 2 Heat Conduction Equation 2-124E A large plane wall is subjected to a specified temperature on the left (inner) surface and heat loss by radiation to space on the right (outer) surface. 2-5C Assuming the egg to be round, heat transfer) will primarily exist in the radial direction only because of symmetry about the center point. The amount of energy loss from the house due to infiltration per day and its cost are to be determined. Analysis The thermal conductivity of the door material is determined directly q& from Fourier's relation to be q& = k & $\Delta T qL$ (25 W / m 2)(0.03 m) $\rightarrow k = = 0.09375 W / m$. The hourly variation of monthly average ambient temperature and solar heat flux incident on a vertical surface is given to be Time of day Ambient Solar insolation Temperature, W/m2 °C 7am-10am 0 375 10am-1pm 4 750 1pm-4pm 6 580 4pm-7pm 1 95 7pm-10pm -2 0 10pm-1am -3 0 1am-4am -4 0 Trom be Heat hin Tin Sun's rays Heat loss hin Tin $\Delta \bullet \bullet \bullet 0$ 1 2 • 3 • 4 hout Tout Glazin hout Tout Analysis The nodal spacing is given to be $\Delta x = 0.05$ m, Then the number of nodes becomes $M = L / \Delta x + 1 = 0.30/0.05 + 1 = 7.3-70$ to be $||| + \eta$ fin = 0.88 20 ||0.02| + 25 W/m C t $h(|| = ||0.05 \text{ m} + \text{m}|| = 0.11 \text{ gives} (||| + ||| + 1.3-70 \text{ for } h + 1.3-70 \text$ of both surfaces of the roof is given Q& to be 0.9. Tair = 10°C Analysis When the surrounding surface temperature is different than the ambient temperature is different temperature is different temperature. conduction problem can be expressed as $d/dT | k | = 0 dx dx / -k dT (0) = q & s = 31,831 W / m2 dr T (L) = TL = 108°C 2-15 Chapter 2 Heat Conduction Equation 2-48 Water flows through a pipe whose outer surface is wrapped with a thin electric heater that consumes 300 W per m length of the pipe. / 1) = = 0.3627 °C / W 2 \pi kL 2 \pi (016.7-23)$ Chapter 7 External Forced Convection 7-32 Air is flowing over a long flat plate with a specified velocity.) h. 8-13C The region of flow over which the thermal entry region, and the length of this region is called the thermal entry length. Using the energy balance approach and taking the direction of all heat transfers to be towards the node under consideration, the implicit transient finite difference formulations become 4 Node 2: $(\varepsilon \sigma \mid p \setminus T \mid +1 - T1 \mid$ +1.4i + 1 = pAC [[Tsurr - (T2)] + h] p [(T ∞ - T2) + kA2 / $\Delta x 2 \Delta t \langle 2 \rangle$ where A = $\pi D 2 / 4$ is the cross-sectional area and p = πD is the perimeter of the fin. The rate of heat transfer between a person and the surrounding air by convection is to be determined. Consequently, the center temperature of the sphere is always the lowest. Also, by approximating the
roast as a spherical object, this heat transfer process can be modeled as one-dimensional since temperature differences (and thus heat transfer) will primarily exist in the radial direction because of symmetry about the center point. The properties of air at this temperature are (Table A-15E) k = 0.01761 Btu/h.ft.°F v = 0.2406 × 10 -3 ft 2 /s Pr = 0.7124 Analysis The Reynolds number is VD (20 ft/s)(0.1/12 ft) Re = ∞ = 692.8 v 0.2406 × 10 - 3 ft 2 /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 / || Resistance wire D = 0.1 in 4/5 5/8 0.62(692.8) 0.5 (0.7124)1/3 [(692.8)] = 0.3 + 1 + || || 1/4 || (282,000 / || 1 + (0.4 / 0.7124) 2 / 3 The heat transfer coefficient is k 0.01761 Btu/h.ft. °F h = Nu = (13.34) = 28.19 Btu/h.ft 2. °F D (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 20 ft/s T ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of the wire becomes As = π DL = π (0.1 / 12 ft) [Air V ∞ = 85°F] 4/5 = 13.34 Then the average temperature of the outer surface of t ft)(12 ft) = 0.3142 ft 2 (1500 × 3.41214) Btu/h Q& \rightarrow Ts = T ∞ + = 85°F + = 662.9°F Q& = hAs (T s - T ∞) hA (28.19 Btu/h.ft 2 .°F)(0.3142 ft 2) Discussion Repeating the calculations at the new film temperature of (85+662.9)/2=374°F gives Ts=668.3°F. The average heat transfer coefficient at the surface of the turkey, the temperature of the skin of the turkey in the oven and the total amount of heat transferred to the turkey in the oven are to be determined. R-value, 3-107 Chapter 3 Steady Heat Conductivity of refrigerator walls is determined to be Q& L ΔT (375) W)(0.03 m) Q& ave = kA ave \rightarrow k = ave = 0.112 W/m. Assumptions 1 The homes are identical, except that their walls are constructed differently. Assumptions 1 Heat transfer through the wall is given to be steady and one-dimensional, and the thermal conductivity to be constant. Therefore, the thermal contact resistance can be ignored for two layers of insulation pressed against each other. + 012. 4-97 Chapter 4 Transient Heat Conduction Review Problems 4-105 Two large steel plates are stuck together because of the glass bulb can be determined by iteration, As = $\pi D 2 = \pi (0.1 \text{ m}) 2 = 0.0314 \text{ m} 2 \text{ Q} \&$ total

= Q& conv + Q& rad = hAs (Ts - T ∞) + ϵ As 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)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K)] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ (0.9)(0.0314 m 2)(Ts 4 - (25 + 273 K))] [+ number of nodes becomes $M = L / \Delta x + 1 = 1/0.25 + 1 = 4$. 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 $1 - 2\tau - 2\tau h\Delta x \ge 0 k \rightarrow \tau \le 12(1 + h\Delta x / k) \Rightarrow \Delta t \le \Delta x + 1 = 1/0.25 + 1 = 4$. Outside surface, 12 km/h wind 2. The upper limit of the time step Δt is determined from the stability criteria that requires all primary coefficients to be greater than or equal to zero. ° C are 5 cm × 5 cm hL (80 W/m 2.°C)(0.025 m) = = 0.800 k (2.5 W/m.°C) Bi = $\rightarrow \lambda 1 = 0.7910$ and A1 = 11016. 4 The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the propane inside, and thus thermal resistance of the tank and the internal convection resistance are negligible. Their centerline temperature when they leave the oven is to be determined. \circ C)(0115. kg)(333.7 kJ / kg) = 52.4 kJ The water completely that night since the amount heat loss is greater than the amount it takes to freeze the water completely (245.65 > 52.4). This corresponds to Q& 249 W %change = difference × 100 = 3.8% (increase) & 6559 W Q total,0° C If the average surrounding temperature is 25°C, the rate of heat loss by radiation and the total rate of heat loss become 7-44 Chapter 7 External Forced Convection Q& rad = ϵ As σ (Ts 4 – Tsurr 4) Q& total [4 = (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.77) + (0.8)(3.7 m 2)(5.67 × 10 -8 W/m 2.K 4)(75 + 273 K) 4 = 1159 W = Q& + Q& = 5001 + 1159 = 6160 W conv] rad which is 6559 - 6160 = 399 W less than the value for a surrounding temperature of 0°C. (c) The mass flow rate of the extruded wire through the air is m& = $\rho V \& = \rho (\pi 2 / 4) V = (2702 \text{ kg/m 3}) \pi (0.0015 \text{ m}) 2 (10 \text{ m/min}) = 6160 \text{ W} conv]$ 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 \infty] = (0.191 \text{ kg/min })(0.896 \text{ kJ/kg}). Assumptions 1 The temperature in the soil is affected by the thermal conditions at one surface temperature. Assumptions 1 The thermal properties of the values are constant. ° F)(1 ft) $2\pi kL 2\pi$ (17 = Rtotal, new + Rlimestone, i = 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = ∞ 1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = \infty1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = \infty1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = \infty1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = \infty1 R total, w/lime 0.00689 h° F / Btu T - T ∞ 2 (350 - 250)°F = = 1.45 × 10 4 Btu/h Q& w/lime = \infty1 R total, w/lime 0.00689 h° F / Btu T - T ∞ the pipe and thus the internal convection resistance slightly, but this effect should be negligible. 3-88 An electric wire is tightly wrapped with a 1-mm thick plastic cover. Using the proper relation in laminar flow for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be hL = 0.664 Re L 0.5 Pr 1 / 3 = $0.664(4.579 \times 104)$ 0.5 (0.7228)1 / 3 = 127.5 Nu = k k 0.02735 W/m 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W
Considering that each transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(65 - 35)°C = 26.2 W Considering that each transistors that can be placed on 2.°C L 0.25 m As = wL = (0.25 m)(0.25 m 2)(0.25 this plate becomes 26.2 W n = = 4.4 \rightarrow 4 6W This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher. 2 The thermal properties of the computer, and even the sequence of calculations. Properties The thermal conductivity of the plate and the soil are given to be kplate = 7.2 Btu/h·ft·°F. Then the cooling time becomes $\tau = \alpha t 2 L \rightarrow t = .$ The heat of fusion of water at 1 atm is hif = 333. °C)(0.0002 m). The rate of heat loss from the steam is to be determined. Assumptions 1 Steady operating conditions exist. This system of 8 equations with 8 unknowns constitutes the finite difference formulation of the problem. 3 Body forces such as gravity are negligible. Therefore, there are two • • D 2 • 0 1 unknowns T1 and T2, and we need two equations to determine them. m2) Rtotal = Rboard + Repoxy + Raluminum + Rconv = 0.00694 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0.0051 + 0 0.00039 + 01361. Analysis (a) The radius of the roast is determined to be m = $\rho V \rightarrow V = V = m \rho = 3.2 \text{ kg } 1200 \text{ kg/m } 3 = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ 4 } 3 \text{ V } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.002667 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.00267 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.00267 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.00267 \text{ m } 3) = 0.002667 \text{ m } 3 \text{ } (0.00267 \text{ m } 3) =$ 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. Then the cooling time becomes $\tau = \alpha t r 02 \rightarrow t = \tau r 02 (0.753)(0.03 \text{ m}) 2 = = 5213 \text{ s} = 1.45 \text{ h} \alpha 0.13 \times 10^{-6} \text{ m} 2 / \text{s}$ The lowest temperature during cooling will occur on the surface (r/r0 = 1), and is determined to be $2 \sin(\lambda 1 r / r0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r0) T (r - T_{\infty} \lambda 1 r 0 / r0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0) T (r - T_{\infty} \sin(\lambda 1 r 0 / r 0$ 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 is to be determined. Properties The thermal conductivity is given to be k = 12 W/m·°C. It gives R = 1.105 m2.°C/W and U = 0.905 W/m2.°C for the air space. If, after cleared of any common factors, any of the terms with the dependent variable or its derivatives involve the independent variable coefficients. Setting R = Rc, the For a unit surface area, the thermal resistance of a flat plate is defined as R = Rc, the For a unit surface area, the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the
thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R = Rc and the thermal resistance of a flat plate is defined as R equivalent thickness is determined from the relation above to be L = kR = kRc = (386 W / m. 3 The system is well-insulated and thus there is no heat transfer. 4 Heat generation within the 0.5-cm thick outer layer of the tissue is negligible. 2 Heat transfer from the front surface is uniform. 2-18 Chapter 2 Heat transfer from the front surface is uniform. wall is subjected to specified temperature on the left surface and convection on the right surface. This is known as the no-slip condition, and it is due to the viscosity of the fluid. Radiation heat transfer between two surfaces is inversely proportional to the number of sheets used and thus heat loss by radiation will be very low by using this highly surface. reflective sheets. 4 Thermal contact resistances at the interfaces are disregarded. ° F)(1 ft) $2\pi kL 2\pi$ (17 ln(ro / r2) ln(0.66 / 0.65) = = = 0.00143 h. 4 The heat transfer coefficients remain constant. 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 Δx 2 $Tmi+1 - Tmi = \rightarrow Tmi + 1 = \tau (Tmi - 1 + Tmi + 1) + (1 - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau 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: <math display="block">Tmi - 1 - 2Tmi + Tmi + 1 + Node 1: h(T - 2\tau)Tmi k \tau The finite difference equation of all heat transfers to be towards the node under consideration of all heat transfers to be towards the node under consideration o$ 2 (interior): T2i +1 = τ steak (T1i + T3i) + (1 - 2 τ steak) T2i Node 3 (interior): T3i +1 = τ steak (T2i + T4i) + (1 - 2 τ steak) T3i T2i - T1i Δx T1i +1 - T1i = (ρC) steak Δx 2 Δt Node 4: π (r452 - r42) {h(T ∞ - T4i) + ϵ plateo [($T\infty$ + 273) 4]} + k steak ($\pi r42$) + k plate ($2\pi r45\delta$) T3i - T4i Δx T5i - T4i T i +1 - T4i 2 δ)] 4 = [(ρC) steak ($\ln 42 \Delta x / 2$) + (ρC) plate ($\ln 45 \Delta r \Delta t$ Node 5: $2 \ln 5 \Delta r \{h(T \infty - T5i) + \epsilon \text{ plateo}[(T \infty + 273) 4]\}$ + k plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($\ln 750 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - T6i T i +1 - T6i = (ρC) plate ($2 \ln 56 \delta$) T6i - T6i T i +1 - $(r56 + r6)/2](\Delta r/2)\delta 6 \Delta r \Delta t 5-116 \text{ Chapter 5 Numerical Methods in Heat Conduction where (<math>\rho C p$) plate = 2441 kW/m 3 · °C, ($\rho C p$) steak = 0.93 × 10 - 6 m 2 / s , hif = 187 kJ/kg, kplate = 237 W/m.°C, α plate = 97.1 × 10 - 6 m 2 / s , and ϵ plate = 0.90, T ∞ = 20°C, h = 12 significant in calculations is to start the calculations with a reasonable mesh size Δx (and time step size Δt for transient problems), based on experience, and then to repeat the calculations using a mesh size of $\Delta x/2$. The dimensionless temperatures at the center of plane wall and at the center of the cylinder are determined first. 6-3C The convection heat transfer coefficient will usually be higher in forced convection since heat transfer coefficient depends on the fluid velocity, and forced convection involves higher fluid velocity, and forced convection since heat transfer by t heat, work, and mass Change in internal, kinetic, potential, etc. Letting r3 represent the outer radius of insulation, the areas of the surfaces exposed to convection for a L = $1 \text{ m} \log \text{ section}$ of the surfaces exposed to convection for a L = $1 \text{ m} \log \text{ section}$ of the surfaces exposed to convection for a L = $1 \text{ m} \log \text{ section}$ of the pipe become A1 = $2 \pi r3 \text{ L} = 2 \pi r3 \text{ m} 2 \text{ m} 2 \pi r3$ m 2 Rpipe Ri Rinsulation Ro To Ti (r3 in m) T1 T2 T3 Then the surfaces exposed to convection for a L = $1 \text{ m} \log \text{ section}$ of the pipe become A1 = $2 \pi r3 \text{ L} = 2 \pi r3 \text{ m} 2 \text{ m} 2 \pi r3$ m 2 Rpipe Ri Rinsulation Ro To Ti (r3 in m) T1 T2 T3 Then the surfaces exposed to convection for a L = $1 \text{ m} \log \text{ section} = 2 \pi r3 \text{ m} 2 \text{ m} 2 \text{ m} 2 \text{ m} 3$ m 2 Rpipe Ri Rinsulation Ro To Ti (r3 in m) T1 T2 T3 Then the surfaces exposed to convection for a L = $1 \text{ m} \log \text{ section} = 2 \pi r3 \text{ m} 2 \text{ m} 3 \text{ m} 2 \text{ m} 3$ m 2 Rpipe Ri Rinsulation Ro To Ti (r3 in m) T1 T2 T3 Then the surfaces exposed to convection for a L = $1 \text{ m} \log \text{ section} = 2 \pi r3 \text{ m} 2 \text{ m} 3 \text{ m} 3$ m 2 Rpipe Ri Rinsulation Ro To Ti (r3 in m) T1 T2 T3 Then the surfaces exposed to convect exposed to co individual thermal resistances are determined to be Ri = Rconv, 1 = R1 = R pipe = R2 = Rinsulation = 1 1 = $0.9944 \ C/W 2 \ ha 1 (80 \ W/m \ C)(1 \ m \ ln(r3 / r2) \ln(r3 / 0.023) = 4.188 \ln(r3 / 0.023) \ C / W 2\pi k 2 \ L 2\pi (0.038 \ W / m. 3 \ Thermal properties of the air are$ constant. The Biot numbers and corresponding constants are first determined to be Bi = hL (600 W/m 2.°C)(0.06 m) = $47.37 \rightarrow \lambda 1 = 1.5381$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow
\lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = $7.895 \rightarrow \lambda 1 = 2.1251$ and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 and A1 = 1.2726 k (0.76 W/m.°C) Bi = hro (600 W/m 2.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 and A1 = 1.2726 k (0.76 W/m.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 and A1 = 1.2726 k (0.76 W/m.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 k (0.76 W/m.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 k (0.76 W/m.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 k (0.76 W/m.°C)(0.01 m) = 7.895 \rightarrow \lambda 1 = 2.1251 exp = (2.1251) 2 exp = 0.2105 (0.01) 2 exp = 0.2and T2 can be determined from k ave $\int = T2 k (T) dT T1 \int = T2 T1 T2 - T1 T2 k 0 (1 + \beta T 2) dT T2 - T1 (= \beta (k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | k 0 | T + T 3 | 3 | T1 T2 - T1 L () \beta | T1 T2 - T1 L ($ average thermal conductivity kave equals the rate of heat transfer through the same medium with variable conductivity k(T). Properties The thermal conductivity k(T). Properties The thermal conductivity k(T). is rare done are to be determined. Node 5 on the insulated boundary can be treated as an interior node for which Tleft + Ttop + Tright + Tbottom - 4Tnode = 0.5 The hot water is to meet the heating requirements of this room for a 24-h period. m)(01. The temperature of the outer surface is to be determined. The first row contains the initial values be Ts, in and Ts, out , respectively, the quantities above can be expressed as $4 Q_{\&} = h A(T - T) + \epsilon A \sigma (T - T 4) = (5 W/m 2.^{\circ}C)(300 m 2)(20 - T)^{\circ}C$ room to roof, conv + rad i room s, in room + (0.9)(300 m 2)(20 - T) + c A \sigma (T - T 4) = (5 W/m 2.^{\circ}C)(300 m 2)(20 - T)^{\circ}C room to roof, conv + rad i room s, in room + (0.9)(300 m 2)(20 - T)^{\circ}C room to roof, conv + rad i room s, in room + (0.9)(300 m 2)(20 - T)^{\circ}C room to roof, conv + rad i room s, in room + (0.9)(300 m 2)(20 - T)^{\circ}C $(300 \text{ m } 2) 0.15 \text{ m } L \text{ Q\& roof to surr, conv+rad} = ho \text{ A}(\text{Ts,out} - \text{Tsurr}) + \varepsilon \text{A}\sigma (\text{Ts,out} 4 - \text{Tsurr} 4) = (12 \text{ W} / \text{m } 2)$. Then the average rate of heat transfer into the drink is Ao = $\pi \text{Do } L + 2 \text{ Q\& bare}$, ave $\pi \text{D} 2 = \pi (0.06 \text{ m})(0.125)$. A relation for the total rate of heat transfer into the drink is Ao = $\pi \text{Do } L + 2 \text{ Q\& bare}$, ave $\pi \text{D} 2 = \pi (0.06 \text{ m})(0.125)$. A relation for the total rate of heat transfer into the drink is Ao = $\pi \text{Do } L + 2 \text{ Q\& bare}$, ave $\pi \text{D} 2 = \pi (0.06 \text{ m})(0.125)$. radiation is disregarded. Assumptions 1 Water is an incompressible substance with constant specific heats. Then the number of • • • • • • nodes M becomes 0 1 2 3 4 5 6 L 5 cm M = +2=+2=71 cm Δx This problem involves 7 unknown nodal temperatures, and thus we need to have 7 equations to determine them uniquely. Analysis (a) Noting that heat transfer is one-dimensional in the radial r direction, the mathematical formulation of this problem can be expressed as $d(2 \text{ dT}) h | r | = 0 \text{ dr} \sqrt{dr} T_{\infty} O2 r r r 2$ and T (r1) = T1 = -183°C dT (r2) - K = h[T (r2) - T_{\infty}] dr (b) Integrating the differential equation once with respect to r gives dT r2 = C1 dr Dividing both sides of the equation above by r to bring it to a readily integrable form and then integrating, C dT C1 \rightarrow T (r) = -1 + C2 = 2 dr r r where C1 and C2 are arbitrary constants. Assumptions 1 Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. 2 The kinetic and potential energy changes are negligible, $\Delta pe \cong \Delta ke \cong 0$. 1-144 Design and Essay Problems KJ 1-81 6 7 Chapter 2 Heat Conduction 2-1C Heat transfer is a vector quantity since it has direction as well as magnitude. The inner radius of the pipe is r1 = 2.0 cm and the outer radius of the pipe and thus the inner radius of insulation is $r_2 = 2.3$ cm. Assumptions 1 Heat transfer is steady since there is no indication of change with time. 2-23 Chapter 2 Heat Conduction Equation 2-61 "GIVEN" Q dot=800 "[W]" L=0.006 "[m]" A base=160E-4 "[m^2]" k=20 "[W/m-C]" T 2=85 "[C]" "ANALYSIS" q dot 0=Q dot/A base T=q dot 0*(L-x)/k+T 2 "Variation of temperature" "x is the parameter to be varied" 0 0.0006667 0.001333 0.002 0.002667 0.003333 0.004 0.005 0.006 x [m] 2-62E A steam pipe is subjected to convection on the inner surface and to specified temperature on the outer surface. This is because the steady heat conduction equation in this case is d 2T / dx 2 = 0 whose solution is T (x) = C1 x + C2 which represents a straight line whose slope is C1. 2 No water flows in or out of the tank during heating. For convenience, we choose the time step to be $\Delta t = 60$ s. 2 The thermal properties of the trunks are constant. ° F / Btu . 5-47 Insulated Chapter 5 Numerical Methods in Heat Conduction 5-53 Heat (Table A-15). Properties The thermal conductivity is given to be $k = 14 \text{ W/m} \cdot \text{C}$. 4 The thermal contact resistance at the interface is negligible. Analysis in this case gives Lc = $b = \pi ro 2 L \pi (1.25 / 12 \text{ ft}) V = = 0.04167 \text{ ft}$ As $2\pi ro L + 2\pi ro 2 2\pi (1.25 / 12 \text{ ft}) + 2\pi$ 2.°F h = = 11.583 h -1 = 0.00322 s -1 ρ C pV ρ C p Lc (62.22 lbm/ft 3) (0.999 Btu/lbm.°F)(0.04167 ft) -1 T (t) - T \propto 45 - 32 = e -bt \rightarrow = e -(0.00322 s) t \rightarrow t = 406 s 80 - 32 Ti - T \propto Therefore, it will take 7 minutes and 46 seconds to cool the canned drink to 45°F. 5°C Analysis Since z>1.5D, the shape factor for this configuration is given in Table 3-5 to be 2π (20 m) $2\pi L$ S = = 34.07 m 80 cm ln(4 z / D) ln[4(0.8 m) /(0.08 m)] 60°C Then the steady rate of heat transfer from the pipe becomes D = 8 cm Q& = Sk (T - T) = (34.07 m)(0.9 W/m.o C)(60 - 5)°C = 1686 W 1 2 L = 20 m 3-91 Chapter 3 Steady Heat Conduction 3-123 "!PROBLEM 3-123" "GIVEN" L=20 "[m]" D=0.08 "[m]" "z=0.80 U = 20 m 3-91 Chapter 3 Steady Heat Conduction 3-123 "!PROBLEM 3-123" "GIVEN" L=20 "[m]" D=0.08 "[m]" "z=0.80 U = 20 m 3-91 Chapter 3 Steady Heat Conduction 3-123 "!PROBLEM 3-123" "GIVEN" L=20 "[m]" D=0.08 "[m]" "z=0.80 U = 20 m 3-91 Chapter 3 Steady Heat Conduction 3-123 "!PROBLEM 3-123" "GIVEN" L=20 "[m]" D=0.08 "[m] 2 Chapter 3 Steady Heat Conduction 3-124 Hot and cold water pipes run parallel to each other in a thick concrete layer. Properties The properties of brass at room temperature are given to be k = 110 W/m. °C, $\alpha = 33.9 \times 10-6 \text{ m}^2/s$ F Analysis The Biot number for this process is u Bi = r hL (80 W / m 2. Noting that this is the only mechanism of energy transfer, the time it takes to raise the water temperature from 20°C to 80°C is determined to be Qin = mC (T2 - T1) Q& in $\Delta t = mC (T2 - T1) (60 \text{ kg})(4180 \text{ J/kg.°C})(80 - 20)^{\circ}C = = 18,810 \text{ s} = 5.225 \text{ h} 800 \text{ J/s} Q& water 800 \text{ W}$ in The surface area of the wire is As = (πD) L = $\pi (0.005 \text{ m})(0.5 \text{ m}) = 0.00785 \text{ m} 2$ The Newton's law of cooling for convection heat transfer is expressed as Q& = hAs (Ts - T∞). Ordinary and partial derivatives are equivalent for functions that depend on a single independent variable. The rate of evaporation of liquid oxygen in the tank as a result of the heat transfer from the ambient air is to be determined. Properties The gas constant of air is R = 0.287 kJ/kg.K (Table A-1). The specific heat of air at the given temperature is Cp = 1006 J/kg.°C (Table A-15). 2-41 Chapter 2 Heat Conduction Equation 2-81E "GIVEN" r 0=0.25/12 "[ft]" k=8.6 "[Btu/h-ft-F]" "g dot=1800 [Btu/h-ft-F]" [Btu/h-ft-F]" "g dot=1800 [Btu/h-ft-F]" [B $(g_dot/Convert(in^3, ft^3))/(4*k)*(r_0^2-r^2)+((g_dot/Convert(in^3,
ft^3))*r_0)/(2*h)$ "Variation of temperature" g [Btu/h.in3] 400 600 800 1000 1200 1400 1600 1800 2000 2200 2400 T0 [F] 229.5 238.3 247 255.8 264.5 273.3 282 290.8 299.5 308.3 317 320 300 T 0 [F] 280 260 240 220 250 700 1150 1600 3 g [Btu/h-in] 2-42 2050 2500 Chapter 2 Heat Conduction Equation 2-82 A nuclear fuel rod with a specified surface temperature is used as the fuel in a nuclear reactor. Applying the boundary conditions give Heat flux at x = 0: -kC1 = - Temperature is used as the fuel in a nuclear reactor. Applying the boundary conditions give Heat flux at x = 0: -kC1 = - Temperature at x = 0: $T(0) = C1 \times 0 + C2 = T1$ gw $0 \times C2 = T1$ gw general solution, the variation of temperature is determined to be T (x) = $-q\&0.700 \text{ W}/m^2x + T1 = -x + 80^\circ \text{ C} = -280 \times (0.3 \text{ m}) + 80 = -4^\circ \text{ C}$ Note that the right surface temperature is lower as expected. The contact conductance at the interface of aluminum-aluminum Rglass Ri Ro plates for the case of ground surfaces and of 20 atm T2 \approx 2 MPa pressure is hc = 11,400 W/m2·°C (Table 3- T1 2). 2 The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties of the case of ground surfaces and of 20 atm T2 \approx 2 MPa pressure is hc = 11,400 W/m2·°C (Table 3- T1 2). 2 The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties of the case of ground surfaces and of 20 atm T2 \approx 2 MPa pressure is hc = 11,400 W/m2·°C (Table 3- T1 2). 2 The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties of the case of ground surfaces and of 20 atm T2 \approx 2 MPa pressure is hc = 11,400 W/m2·°C (Table 3- T1 2). 2 The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties of the case of ground surfaces and of 20 atm T2 \approx 2 MPa pressure is hc = 11,400 W/m2·°C (Table 3- T1 2). 2 The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal properties The thermal properties of the copper are constant. °C)(01571 m2)(180 - 25)°C = 974 W. Ice block Ti Properties The thermal p $0.124 \times 10 \text{ m/s}$. 2 The heat transfer coefficients for wrapped and unwrapped potatoes are the same. The thermal diffusivity and the thermal diffusivity of the rod are to be determined. + $0.017 = 0.147 \times 10 - 6 \text{ m}^2 / \text{s}(60 \text{ min} \times 60 \text{ s} / \text{min}) \{ (0.025 \text{ m}) 2 2 \text{ T} (0,0,0, \text{ t}) - 500 = (1.0580) \text{e} - (0.5932) (6.624) 20 - 500 = 6.624 > 0.2 \} \{ (1.1016) \text{e} 2 - (0.5932) (6.624) 20 - 500 = 6.624 > 0.2 \} \{ (1.1016) \text{e} 2 - (0.5932) (6.624) 20 - 500 = 6.624 > 0.2 \} \}$ 0.7910 2 (6.624) $= 0.000186 \rightarrow T$ (0,0,0, t) = 500°C Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transient heat conduction. The interface temperature TI is determined from the second interface condition that the heat flux in the wire and the plastic layer at r = r1 must be the same: -k wire dTplastic (r1) dTwire (r1) = -k plastic 2 T ∞ - TI 1 r k r1 ln 2 + r1 hr2 Solving for TI and substituting the given values, the interface temperature is determined to be TI = g&r12 2k plastic $| n + | r hr2 1 \rangle | + T_{\infty} | \rangle (0.007 \text{ m} 1.8 \text{ W/m} \cdot ^{\circ}\text{C} | + 25^{\circ}\text{C} = 97.1^{\circ}\text{C} | n + | 0.003 \text{ m} (14 \text{ W/m} 2 \cdot ^{\circ}\text{C})(0.007 \text{ m}) | \rangle \langle \text{Knowing the interface temperature, the temperature at the center line } (r = 0) is obtained by substituting the known quantities into Eq. (c), = (1.5 \times 10.6 \text{ W/m} 3)(0.003 \text{ m}) 2 2(1.8 \text{ W/m} \cdot ^{\circ}\text{C}) \text{ Twire } (0) = \text{TI} + \&$ 12 gr (15. This problem involves 4 unknown nodal temperatures, and thus we need to have 4 equations. 7-7C The part of drag that is due directly to wall shear stress tw is called the skin friction drag FD, friction since it is caused by frictional effects, and the part that is due directly to pressure P and depends strongly on the shape of the body is called the pressure drag FD, pressure. 5-5C The experiments will most likely prove engineer B right since an approximate solution of a more realistic model is more accurate than the exact solution of a crude model of an actual problem. 2 The temperature along the fins varies in one direction only (normal to the plate). 1867 ft) 4 3 V 3 V = πro → ro = 3 = 3 = 0.3545 ft 3 4 π 4 π The Fourier number is $\tau = \alpha t$ ro 2 = (3.5 × 10 - 3 ft 2 /h)(5 h) (0.3545 ft) 2 = 0.1392 Oven T = 325°F which is close to 0.2 but a little below it. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be hL Nu = 0.664 Re L 0.5 Pr 1 / 3 = 0.664(5.386 × 10 4) 0.5 (0.7255)1 / 3 = 138.5 k k 0.02662 W/m.°C h = Nu = (138.5) = 16.75 W/m 2.°C L 0.22 m The temperature of aluminum plate then becomes (4 × 12) W Q& \rightarrow Ts = T ∞ + = 20°C + = 50.0°C Q& = hAs (Ts - T ∞) hAs (16.75 W/m 2.°C)[2(0.22 m) 2] Discussion In reality, the heat transfer coefficient will be higher since the transistors will cause turbulence in the air. The energy balance on the system can be expressed as E - Eout 1in 424 3 = Net energy transfer by heat, work, and mass ΔEsystem 12 4 4 3 WATER Change in internal, kinetic, potential, etc. Assumptions 1 Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values during the entire night. The Reynolds number in this case is V L [$(50 \times 1000 / 3600)$ m/s](8 m) L=8m Re L = ∞ = 7.792 × 10 6 υ 1.426 × 10 -5 m 2 /s which is greater than the critical Reynolds number. Properties The emissivity of a person is completely enclosed by the surrounding surfaces, and he or she will lose heat to the surrounding air by convection, and to the surrounding surfaces by radiation. The thermal conductivity of the styrofoam is given to be k = 0.033 W/m·°C. (b) Exact solution This problem can be solved exactly by obtaining the differential equation from an energy balance on the device for a differential time interval dt. This problem involves 7 unknown nodal temperatures, and thus we need to have 7 equations. 4-98 Chapter 4 Transient Heat Conduction 4-106 A curing kiln is heated by injecting steam into it and raising its inner surface temperature to a specified value. Properties The thermal conductivity of the cast iron is given to be $k = 52 \text{ W/m} \cdot \text{C}$. Inside surface, still air 1 2 3 4 5 R-value, m2.°C/W Total unit thermal resistance (the R-value) which is less than 2.4 m2.°C/W. m) hr0 (22 W / m2. Gypsum wallboard, 0.5 in 7. (c) The mass flow rate of the extruded wire through the air is m& = 52 W/m \cdot \text{C}. $\rho V \&= \rho (\pi 2 / 4) V = (8950 \text{ kg/m 3}) \pi (0.0015 \text{ m}) 2 (10 \text{ m/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) (0.383 \text{ kg/min}) (0.383 \text{ kg/min}) (0.383 \text{ kg/min}) (0.383 \text{ kg/min}) = 0.633 \text{ kg/min} (0.0015 \text{ m}) 2 (10 \text{ m/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) (0.383 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ Then the rate of heat transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ The transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ The transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ The transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = (0.633 \text{ kg/min}) = 0.633 \text{ kg/min} 0$ The transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = 0.633 \text{ kg/min} 0$ The transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = 0.633 \text{ kg/min} 0$ The transfer from the wire to the air becomes $Q \&= m \& C p [T (t) - T \propto] = 0.633 \text{ kg/min} 0$ The transfer from the wire to the air becomes Q &= m & C p [Also, 150 • 180 • 200 • 180 • 12 Insulated • 200 • 1 2 Insulated surface as well as the diagonal line. The length of
time the hot air should be blown is to be determined. 2-55 x Chapter 2 Heat Conduction Equation 2-100 A cylindrical shell with variable conductivity is subjected to specified temperatures on both sides. The differential equation that describes the variation of temperature of the ball with time is to be derived. It is equivalent to insulation or zero heat flux boundary condition, and is expressed at a point x0 as $\partial T(x0, t) / \partial x = 0$. Then the fraction of heat dissipated from the top and bottom surfaces of the resistor becomes Qtop - base Qtotal = Atop - base Atotal = 0.251 = 0.118 or (11.8%) 2136. The desired values are determined directly from T s = T ∞ + g&L (2 × 10 5 W/m 3)(0.05 m) = 25°C + = 252.3°C + = 252.3° 70 75 80 85 90 95 100 Tmin [C] 525 425 358.3 310.7 275 247.2 225 206.8 191.7 178.8 167.9 158.3 150 142.6 136.1 130.3 125 Tmax [C] 527.3 427.3 249.5 227.3 209.1 193.9 181.1 170.1 160.6 152.3 144.9 138.4 132.5 127.3 2-45 Chapter 2 Heat Conduction Equation 550 500 450 T m in [C] 400 350 300 250 200 150 100 20 30 40 50 $60\ 70\ 80\ 90\ 100\ 80\ 90\ 100\ 80\ 90\ 100\ 2\ h\ [W/m\ -C]\ 550\ 500\ 450\ T\ m\ ax\ [C]\ 400\ 350\ 300\ 250\ 200\ 150\ 100\ 20\ 30\ 40\ 50\ 60\ 70\ 2\ h\ [W/m\ -C]\ 2-46\ Chapter\ 2\ Heat\ Conduction\ Equation\ 2-86\ A\ long\ resistance\ heater\ wire\ is\ subjected\ to\ convection\ at\ its\ outer\ surface.\ /\ 2\ m\ The\ pCp\ (volumetric\ specific\ specific$ heat) values of the steaks and of the defrosting plate are $\cdot 1 \cdot 2 \cdot 3 \cdot 4 \cdot 5 \cdot 6 \cdot (\rho C p)$ plate = (8933 kg/m 3)(0.385 kJ/kg $\cdot ^{\circ}$ C) = 1504 kW/m 3 $\cdot ^{\circ}$ C Analysis The nodal spacing is given to be $\Delta x = 0.005$ m in the steaks, and $\Delta r = 0.0375$ m in the plate. Properties of oil at 50° C are given to be k = 0.17 W/m-K and μ = 0.05 N-s/m2 Analysis (a) Oil flow in journal bearing can be approximated as parallel flow between two large plates with one plate moving and the other stationary. 5 The given time step Δt = 5 min is less than the critical time step Δt = 5 min is less than the critical time step Δt = 5 min is less than the critical time step so that the stability criteria is satisfied. Then the cooling time becomes τ = at r02 \rightarrow t= τ r02 (0.456)(0.12 m) 2 = 50,558 s = 14.0 h α 0.13 \times 10 -6 m 2 / s The lowest temperature during cooling will occur on the surface (r/r0 = 1), and is determined to be T (r) - T ∞ T (r0) - T ∞ T -T 2 = θ 0 J0 (λ 1r / r0) = 0 ∞ J0 (λ 1r0 / r0) = A1e - λ 1 τ J0 (λ 1r / r0) \rightarrow Ti - T ∞ Ti - T ∞ Ti - T ∞ Substituting, T (r0) - $0(\lambda 1) = 0.2326 \times 0.2084 = 0.0485 = || \rightarrow T(r0) = -3.9^{\circ}C 37 - (-6) \sqrt{37 - (-6)} \sqrt{37 - (-6)}$ $T k ave \int = T2 k (T) dT T1 \int = T2 T1 T2 - T1 (k 0 (1 + \beta T) dT 2 T2 - T1 = \beta (\lambda 2 k 0 | T + T3 | 3 (J + T12 + T1$ equals the rate of heat transfer through the same medium with variable conductivity k(T). 4 Heat transfer coefficients are constant and uniform over the surfaces. The heat flux boundary condition at the interface (radius r0) in terms of the heat generated is to be expressed. Assumptions 1 The meat slabs can be approximated as very large plane walls of halfthickness L = 3-in. Properties Assuming a film temperature of 40°C, the properties of air are (Table A-15) k = 0.02662 W/m.°C $\nu = 1.702 \times 10 - 5$ m 2/s which is less than the critical Reynolds number. Ti = 400°C The Fourier number is $\tau = \alpha t = L2$ (3.95 × 10 -6 m 2/s)(20 × 60 s) (0.175 m) 2 = 0.1548 which is very close to 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. Then the total cost of insulation becomes Insulation Cost = (Unit cost)(Surface area) = [(\$10 / cm)(1 cm) + \$30 / m 2](70.69 m 2) = \$2828 7-78 T chapter 7 External Forced Convection To determine the thickness of insulation whose cost is equal to annual energy savings, we repeat the calculations above for 2, 3, . Then the temperature at the surface of the hot dog becomes 2 2 T (ro , t) $-T \propto = A1e - \lambda 1 \tau J 0 (\lambda 1 ro / ro) = (1.5357)e - (2.0785) (0.4001) (0.2194) = 0.05982$ Ti $-T \propto T (ro, t) - 94 = 0.05982$ Ti $-T \propto T (ro, t) - 94 = 0.05982$ Ti $-T \propto T (ro, t) = 89.6$ °C 20 - 94 The maximum possible amount of heat transfer is [] m = $\rho V = \rho \pi ro 2 L = (980 \text{ kg/m } 3) \pi (0.011 \text{ m}) 2.3-101$ Chapter 3 Steady Heat Conduction 3-139 The R-value and the U-factor of a wood frame wall are to be determined. Analysis: Cubic block: This cubic block: This cubic block can physically be formed by the intersection of three infinite plane walls of thickness 2L = 5 cm. 1-14 Chapter 1 Basics of Heat Transfer Properties The specific heats of water and the iron block at room temperature are Cp, water = 4.18 kJ/kg·°C and Cp, iron = 0.45 kJ/kg·°C (Tables A-3 and A-9). kg $Q = mh fg = (0157 \cdot 4 The outer surface at r = r0 is subjected to convection. 2-17 in the text)$. Analysis We take the pipe as the system. It is also not proper to use this model when finding the temperatures near the bottom or top surfaces of a cylinder since heat transfer at those locations can be two-dimensional. 2-15 in the text). The nodal spacing is given to be $\Delta x = \Delta y = l = 1$ ft, and the general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference form of an interior node for steady two-dimensional heat conduction with no heat general finite difference for steady twe dit for stead passing through the center. The rate of heat loss through the wall that night and its cost are to be determined. Analysis The rate of heat transfer through each wall can be determined by applying thermal conductivities are given to be k = 20 W/m.°C for the circuit board, k = 386 W/m.°C for the copper plate and fins, and k = 1.8 W/m.°C for the epoxy adhesive. 4 The plates are large so that there is no variation in z direction. Analysis Using water properties at room temperature, the mass flow rate of water and rate of heat transfer from the water are determined to be m& = $\rho V \&$ $= \rho VA = (1000 \text{ kg}/\text{m3})(1.5 \text{ m}/\text{s}) \pi (0.03) 2 / 4 \text{ m2} = 106$. Properties of stainless steel 304 at room temperature are given to be k = 14.9 W/m.°C, $\rho = 7900 \text{ kg/m3}$, Cp = 477 J/kg.°C, $\alpha = 3.95 \times 10-6 \text{ m2/s}$ Analysis First the Biot number is calculated to be Bi = hro (60 W/m 2.°C)(0.175 m) = 0.705 k (14.9 W/m.°C) Air T = 150°C The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, Steel shaft λ 1 = 10935. Taking time step to be Δ t = 1 min, the temperature distribution throughout the glass 15 min after the strip heaters are turned on and when steady conditions are reached are determined to be (from the EES solutions disk) 15 min: T1 = -2.4°C, T2 = -2 -2.4° C, T3 = -2.5° C, T4 = -1.8° C, T5 = -2.0° C, T6 = -2.7° C, T7 = 12.3° C, T8 = 10.7° C, T9 = 9.6° C Steady-state: T1 = -2.4° C, T3 = -2.5° C, T4 = -1.8° C, T5 = -2.0° C, T6 = -2.7° C, T7 = 12.3° C, T8 = 10.7° C, T9 = 9.6° C Steady-state: T1 = -2.4° C, T5 = -2.0° C, T6 = -2.7° C, T7 = 12.3° C, T8 = 10.7° C, T9 = 9.6° C Discussion Steady operating conditions are reached in about 8 min. Then, Δx m = 1: T0 - 2T1 + T2 + h(p\Delta x 2 / kA)(T \infty - 2T) + D(T) T1) = 0 • • • • • 2 m = 2: T1 - 2T2 + T3 + h($p\Delta x / kA$)(T ∞ - T2) = 0 0 1 2 3 4 m = 3: T2 - 2T3 + T4 + h($p\Delta x 2 / kA$)(T ∞ - T3) = 0 Node 4: kA T3 - T4 + h($p\Delta x 2 / kA$)(T ∞ - T4) = 0 Δx where $\Delta x = 0.005$ m, k = 237 W/m · °C, T $\infty =
35^{\circ}$ C, T0 = 130°C, h = 30 W/m 2 · °C and A = (3 m)(0.003 m) = 0.009 m 2 and p = 2(3 + 0.003 m) = 6.006 m . 4-10C m = 0.005 m The lumped system analysis is more likely to be applicable for a golden apple than for an actual apple since the thermal conductivity is much larger and thus the Biot number is much larger and thus the Biot number is much larger and the system analysis is more likely to be applicable for a golden apple since the thermal conductivity is much larger and thus the Biot number is much larger and thus the Biot number is much larger and the system analysis is more likely to be applicable for a golden apple since the thermal conductivity is much larger and thus the Biot number is much larger and thus the Biot number is much larger and the system analysis is more likely to be applicable for a golden apple since the thermal conductivity is much larger and thus the Biot number is much larger and thus the Biot number is much larger and the system analysis is more likely to be applicable for a golden apple since the thermal conductivity is much larger and thus the Biot number is much larger and thus the Biot number is much larger and the system apple since the thermal conductivity is much larger and the system apple since the thermal conductivity is much larger and the system apple since the thermal conductivity is much larger and the system apple since the thermal conductivity is much larger and the system apple since the thermal conductivity is much larger and the system apple since the thermal conductivity is much larger and the system apple since the thermal conductivity is much larger and the system apple since the thermal conductivity is much larger apple since the the thermal conductivity is much larger apple (-5) C = 132.3 W Q& = s1 ∞ 2 = 0.2495 ° C / W Rtotal If the jacket is made of a single layer of 0.5 mm thick synthetic fabric, the rate of heat transfer would be T – T Ts1 – T ∞ 2 [(28 – (-5)] ° C = 827 W Q& = s1 ∞ 2 = 5 × R fabric + Ro (5 × 0.0007 + 0.0364) ° C / W Rtotal The thickness of a wool fabric that has the same thermal resistance is determined from Rtotal = Rwool + Ro = fabric 0.2495 ° C / W = L 1 + kA hA L (0.035 W / m. The Biot numbers and the corresponding constants are first determined to be Bi wall, 1 = hL1 (12 W/m 2.°C)(0.02 m) = = 0.1081 $\rightarrow \lambda 1 = 0.3208$ and A1 = 10173. An energy transfer is heat transfer when its driving force is temperature difference. It is the same as the unit thermal resistance of the wall. Properties The thermal conductivity of the expanded perlite insulation is given to be $k = 0.052 \text{ W/m} \cdot \text{°C}$. (To $-T\infty$) = -6 + 017. Then the rate of heat generation in L = 15 in the wire per unit volume is determined by dividing the total rate of heat generation by the volume of the wire to be 2-2 Chapter 2 Heat Conduction Equation $g_{\&} = G_{\&} V$ wire = $G_{\&} (\pi D/4) L 2 = (3.412 \text{ Btu/h} 7 3) = 7.820 \times 10$ Btu/h \cdot ft 1 W [$\pi(0.08/12 \text{ ft}) / 4$](15 / 12 ft) / 4](15 / 12 $W(3.412 \text{ Btu/h})52 = = q\& = ||= 1.303 \times 10 \text{ Btu/h} \cdot \text{ft}$ whereas heat flux is expressed per unit surface area in Btu/h $\cdot \text{ft}$? Analysis The boundaries can be expressed analytically as -k Atx = 0: -k At x = L: dT(0) = 0 or dx dT(0) = 0 or dx dT(L) = 0 and T(L) = 0 and TInsulated $\Delta x \ 0^{\bullet} T5 - T4 \ \Delta x \ \bullet 1 \ \epsilon \ \bullet 2 \ \bullet 3 \ Tsurr \ \bullet 4 \ 5 \ Substituting, the finite difference formulation of the boundary nodes become At x = 0: At x = L : -k - k \ T1 - T0 = 0 \ \Delta x \ or \ T1 = T0 \ T5 - T \ 4 \ 4 = \varepsilon \sigma \ [T54 - T \ surr \] \ \Delta x \ One-Dimensional \ Steady \ Heat \ Conduction \ 5-11C \ The finite \ difference form of a heat \ conduction \ problem \ by the energy \ balance \ Dimensional \ Steady \ Heat \ Conduction \ 5-11C \ The finite \ difference \ form \ of \ a heat \ conduction \ problem \ by \ the energy \ balance \ Dimensional \ Steady \ Heat \ Conduction \ 5-11C \ The \ finite \ difference \ form \ of \ a heat \ conduction \ balance \ Dimensional \ Steady \ Heat \ Conduction \ Steady \ Heat \ Steady \ Heat \ Conduction \ Steady \ Heat \ Steady \$ method is obtained by subdividing the medium into a sufficient number of volume elements, and then applying an energy balance on each element. Analysis The rate of heat transfer through the window can be determined from Window Q& =U A (T - T) window overall window in out where Ti and To are the indoor air temperatures, respectively, Uoverall is the U-factor (the overall heat transfer coefficient) of the 20°C window, and Awindow is the window area. Still air below ceiling 1 2 3 4 5 6 7 8 Total unit thermal resistance of each section, U = 1/R, in W/m2.°C Area fraction 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, U = 1/R, in W/m2.°C Area fraction 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, U = 2 farea, iU = 2 f $0.82 \times 1.290 + 0.18 \times 0.805$ Overall unit thermal resistance, R = 1/U (b) One-reflective surface, $\epsilon 1 = 0.05$ and $\epsilon 2 = 0.9 \rightarrow \epsilon$ effective R-value, m2.°C/W Between At studes studes 0.12 0.044 0.009 0.14 0.011 0.23 0.11 0.166 0.16 ----0.63 0.079 0.079 0.12 0.12 0.775 1.243 1.290 0.805 0.82 0.18 1.203 W/m2.°C 0.831 m2.°C/W 1 1 = = = 0.05 1 / $\epsilon 1 + 1 / \epsilon 2$ -1 1 / 0.05 + 1 / 0.9 - 1 In this case we replace item 6a from 0.16 to 0.47 m2.°C/W. 3 The fins and the base plate are nearly isothermal (fin efficiency is equal to 1) 4 Air is an ideal gas with constant properties. Properties The thermal conductivities are given to be k = 8.7 Btu/h·ft.°F for steel pipe and k = 0.020 Btu/h·ft.°F for fiberglass insulation. Then the maximum velocity and the Reynolds number based on the maximum velocity become Vmax V=7 m/s Ti=40°C ST $0.06 = V = (7 \text{ m/s}) = 10.50 \text{ m/s}(0.02 \text{ m}) = 10,524 \mu 2.052 \times 10 - 5 \text{ kg/m} \cdot \text{s}$ Ts=140°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(10,524) 0.63 (0.7177) 0.36 (0.7177) 0.7041) 0.25 = 82.33 Since NL > 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tubes are provided by the tubes in the tubes i Chapter 7 External Forced Convection Special Topic: Thermal Insulation 7-73C Thermal insulation is a material that is used primarily to provide resistance to heat flow. / 12 ft) 2 τ r02 (0.426)(125 = 3302 s = 55.0 min 1.4 × 10-6 ft 2 / s α The lowest temperature during cooling will occur on the surface (r/r0 = 1), and is determined to be T (r) – T ∞ $\sin(\lambda \operatorname{1r0}/\operatorname{r0})$ T $- \operatorname{T}_{\infty} \sin(\lambda \operatorname{1r0}/\operatorname{r0})$ T $(\operatorname{r0}) - \operatorname{T}_{\infty} 2\sin(\lambda \operatorname{1r}/\operatorname{r0}) = = \theta = A1e - \lambda \operatorname{1r} + \lambda \operatorname{1r}/\operatorname{r0} \lambda \operatorname{1r0}/\operatorname{r0}$ T $- \operatorname{T}_{\infty} \operatorname{Ti} - \operatorname{T}_{$ pressure drop of air, and the rate of condensation of steam are to be determined. 2-109C For a function f (x, y), the partial derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative $\partial f / \partial x$ will be equal to the ordinary derivative hD Nu = 0.248 Re 0.612 Pr $1/3 =
0.248(1.258 \times 10.6)$ 0.612(0.724)1/3 = 1204 k The average heat transfer coefficient on the wing surface is 0.02152 W/m $2.^{\circ}$ C D 0.3 m Then the average heat transfer per unit surface area becomes $q_{\&} = h(Ts - T_{\infty}) = (86.39 \text{ W/m } 2.^{\circ}\text{C})[0 - (-55.4)]^{\circ}\text{C} = 4786$ W/m $2.^{\circ}\text{C}$ D 0.3 m Then the average rate of heat transfer per unit surface area becomes $q_{\&} = h(Ts - T_{\infty}) = (86.39 \text{ W/m } 2.^{\circ}\text{C})[0 - (-55.4)]^{\circ}\text{C} = 4786$ W/m $2.^{\circ}\text{C}$ D 0.3 m Then the average rate of heat transfer per unit surface area becomes $q_{\&} = h(Ts - T_{\infty}) = (86.39 \text{ W/m } 2.^{\circ}\text{C})[0 - (-55.4)]^{\circ}\text{C} = 4786$ W/m $2.^{\circ}\text{C}$ D 0.3 m Then the average rate of heat transfer per unit surface area becomes $q_{\&} = h(Ts - T_{\infty}) = (86.39 \text{ W/m } 2.^{\circ}\text{C})[0 - (-55.4)]^{\circ}\text{C} = 4786$ W/m $2.^{\circ}\text{C}$ D 0.3 m Then the average rate of heat transfer per unit surface area becomes $q_{\&} = h(Ts - T_{\infty}) = (86.39 \text{ W/m } 2.^{\circ}\text{C})[0 - (-55.4)]^{\circ}$ 39 Chapter 7 External Forced Convection 7-49 A long aluminum wire is cooled by cross air flowing over it. Properties The thermal properties of the rod available are given to be $\rho = 3700 \text{ kg/m3}$ and $\text{Cp} = 920 \text{ J/kg.}^\circ\text{C}$. The length of time that the electric heating system would run that night with or without solar heating are to be determined. The density and the specific heat of LNG are given to be 425 kg/m3 and 3.475 kJ/kg·°C, respectively, Analysis The shape factor for this configuration is given 20°C in Table 3-5 to be S = $2\pi(1.9 \text{ m}) 2\pi L = 12.92 \text{ m} 1.4 \text{ m} \sqrt{1.08 \text{ w}} (\ln | \ln | 1.08 | 0.6 \text{ m} / L = 1.9 \text{ m}$ Then the steady rate of heat transfer to the tank becomes Q& = Sk (T - T) = $(12.92 \text{ m})(0.0006 \text{ W/m}.^{\circ}\text{C})[20 - (-160)]^{\circ}\text{C} = 1.395 \text{ W} 1 2$ The mass of LNG is m = $\rho \text{V} = \rho \pi$ (0.6 m) 3 D3 = (425 kg/m 3) π = 48.07 kg 6 6 The amount heat transfer to the tank for a one-month period is Q = Q& Δt = (1.395 \text{ W})(30 × 24 × 3600 \text{ s}) = 3,615,840 \text{ J} Then the temperature of LNG at the end of the month becomes Q = mC p (T1 - T2) 3,615,840 J = (48.07 kg)(3475 J/kg.°C)[(-160) – T2]°C T2 = -138.4°C 3-178 ··· 3-184 Design and Essay Problems KJ 3-139 1.4 m Chapter 4 Transient Heat Conduction Chapter 4 Tr remains essentially uniform at all times during a heat transfer process. 4-92 Chapter 4 Transient Heat Conduction 4-93C The freezing time could be decreased by (a) lowering the temperature of the meat boxes. ° C (0.5365 m2) Q& = kA 0.03 m L The total amount of heat needed to melt the ice completely is Q = mhif = (40 kg)(333.7 kJ / kg) = 13,348 kJ Ice chest, Q& Then transferring this much heat to the cooler to melt the ice completely will take 3 Q 13,348,000 J Δt = = 2,828,000 s = 785.6 h = 32.7 days 4.72 J/s Q& 1-85 A transistor mounted on a circuit board is cooled by air flowing over it. ° C)(0.04 m) = = 1.67 k (0.6 W / m. Properties The thermal conductivities are given to be k = 380 W/m °C for granite. Therefore, the structure is losing heat as expected. ° C)(300 m 2) 015. 2217) || | } = 0.7627 + 20 - 1200 | (0.2) 2 (0.075) 2 || |] | | | | | | | | | | Solving for the time t gives t = 285 s = 4.7 min. 3-151 A ceiling consists of a layer of reflective acoustical tiles. 5 The pipe temperature remains constant at about 150°C with or without insulation. 5-12C In the energy balance formulation of the finite difference method, it is recommended that all heat transfer at the boundaries of the volume element be assumed to be into the volume element even for steady heat conduction. The rate of heat loss from the hot water, and thus the thermal contact resistance has significance only for highly conducting materials like metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m) (0.038 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ T Δ T (80 - 30)° C Q& = \rightarrow Rtotal = = 0.333 °C/W 2 2 (10 W/m.°C)(3 m 2) L = 0.034 m = 3.4 metals. × 1500 W = 150 W Δ cm Noting that heat is saved at a rate of $0.9 \times 1500 = 1350$ W and the furnace operates continuously and thus $365 \times 24 = 8760$ h per year, and that the furnace efficiency is 78%, the amount of natural gas saved per year is Q& Δt (1.350 kJ/s)(8760 h) (3600 s) (1 therm) = Energy Saved = saved || = 517.4 therms | || Efficiency 0.78 (1 h) (105,500 kJ / The money saved is Money saved = (Energy Saved)(Cost of energy) = (517.4 therms)(\$0.55 / therm) = \$284.5 (per year) The insulation will pay for its cost of \$250 in Money saved \$284.5 / yr which is less than one year. The time for the ignition of the trunks is to be determined. Therefore the flow is laminar. After 5 minutes First the Biot number is calculated for the plane wall to be Bi = hL (120 Btu / h. m2 Ao = π Do L = π (0.055 + 0.06 m)(1 m) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \propto 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(0.157 m 2) ln(r2 / r1) ln(2.75 / 2.5) R1 = R pipe = = = 0.00101 °C/W 2\pi k1 L 2\pi (15 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \propto 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(0.157 m 2) ln(r2 / r1) ln(2.75 / 2.5) R1 = R pipe = = = 0.00101 °C/W 2\pi k1 L 2\pi (15 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(0.157 m 2) ln(r2 / r1) ln(2.75 / 2.5) R1 = R pipe = = = 0.00101 °C/W 2\pi k1 L 2\pi (15 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(0.157 m 2) ln(r2 / r1) ln(2.75 / 2.5) R1 = R pipe = = = 0.00101 °C/W 2\pi k1 L 2\pi (15 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(0.157 m 2) ln(r2 / r1) ln(2.75 / 2.5) R1 = R pipe = = = 0.00101 °C/W 2\pi k1 L 2\pi (15 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T \approx 1 1 1 = = 0.08 °C/W 2 hi Ai (80 W/m. °C)(11 M) = 0.361 m2 Ri The individual thermal resistances are R1 R2 T R2 T R2 Hi R m) Ri = R 2 = Rinsulation = $\ln(r_3/r_2) \ln(5.75/2.75) = 3.089 \text{°C/W} 2\pi k 2 L 2\pi (0.038 \text{ W/m.°C})(1 \text{ m}) 1 1 = 0.1847 \text{°C/W} 2 \text{ ho Ao} (15 \text{ W/m} .°C)(0.361 \text{ m } 2) = \text{Ri} + \text{R1} + \text{R2} + \text{Ro} = 0.08 + 0.00101 + 3.089 + 0.1847 = 3.355 \text{°C/W} Ro = \text{Rtotal Then the steady rate of heat loss from the steam per m. If we take the room that contains the refrigerator$ as our system, we will see that electrical work is supplied to this room to run the refrigerator, which is eventually dissipated to the room as waste heat. Analysis The nodal spacing is given to be $\Delta x = \Delta x = l = 0.1 \text{ m}$, and the general finite difference form of an interior node equation for steady 2-D heat conduction is expressed as Insulated 6 kW heater g& 12 Tleft + Ttop + Tright + Tbottom - 4Tnode + node = 0 k Tleft + Ttop + Tright + Tbottom - 4Tnode = 0 • • • 1 5 5 There is symmetry about a vertical line passing through the middle of the region, and we need to 2 6 10 0°C 6 • • • consider only half of the region. 2 Heat transfer is one-dimensional since there is Ts thermal symmetry about the center line and no change in the ro 2-47 0 Heater Chapter 2 Heat Conductivities are given to be k = 0.035 W/m.°C for fiber glass insulation Analysis We consider only the side surfaces of the water heater for simplicity, and disregard the top and bottom surfaces (it will make difference of about 10 percent). The change in the drag force and the rate of heat transfer when the free-stream velocity of the fluid is doubled is to be determined. 5 The convection coefficient is constant and uniform over the entire surface of the person. 2 Heat transfer is one-dimensional since the thickness of the bottom of the pan is small
relative to its diameter. The results are x hx Cf, x 3 1 0.9005 0.003162 2 0.6367 0.002236 3 0.5199 0.001826 2.5 4 0.4502 0.001195 8 0.3184 0.001118 1.5 9 0.3002 0.001054 10 0.2848 0.001 Chapter 7 External Forced Convection 7-19E "PROBLEM 7-19E" "GIVEN National Convection 7-19E" "GIVE T_air=60 "[F]" "x=10 [ft], parameter to be varied" Vel=7 "[ft/s]" "PROPERTIES" Fluid\$='air' k=Conductivity(Fluid\$, T=T_air) Pr=Prandtl(Fluid\$, critical Re number. Properties The thermal conductivity of cast iron is given to be k = 52 W/m·°C. 3-13 Chapter 3 Steady Heat Conduction 3-32 "GIVEN" A=2*1.5 "[m^2]" T s=80 "[C]" T infinity=30 "[C]" h=10 "[W/m^2-C]" "k ins=0.038 [W/m-C], parameter to be varied" f reduce=0.90 "ANALYSIS" Q dot old=h*A*(T s-T infinity) Q dot new=(1-0.06 k ins [W /m -C] 3-14 0.07 0.08 Chapter 3 Steady Heat Conduction 3-33E Two of the walls of a house have no windows while the other two basic methods of solution of transient problems based on finite differencing are the explicit and the implicit methods. The specific heat of aluminum at 450 K (which is somewhat below 200°C = 473 K) is 0.973 kJ/kg.°C. 4-96 Chapter 4 Transient Heat Conduction 4-104 The center temperature of meat slabs is to be lowered by chilled air to below 5°C while the surface temperature of meat slabs is to be lowered by chilled air to below 5°C while the surface temperature of meat slabs is to be lowered by chilled air to below 5°C while the surface temperature of meat slabs is to be lowered by chilled air to below 5°C while the surface temperature of meat slabs is to be lowered by chilled air to below 5°C while the surface temperature of meat slabs is to be lowered by chilled air to below 5°C while the surface temperature of meat slabs is to be lowered by chilled air to be lowered by chill $A\Delta x$ = $\rho A\Delta x C 1 \Delta x \Delta x \Delta t$ Right boundary node (node 2): k 2i A T1i - T2i $\Delta x T2i + 1 - T2i i + \varepsilon A[(Tsurr + 273) 4 - (T2i + 273) 4] + g & 2i (A\Delta x / 2) = \rho A C \Delta x 2 \Delta t 5-107$ Chapter 5 Numerical Methods in Heat Conduction 5-110 A plane wall with variable heat generation and variable thermal conductivity is subjected to uniform heat flux q 0 and convection at the left (node 0) and radiation at the right boundary (node 2). 2 Heat transfer through the door is small Door relative to other dimensions. 3 The thermal properties of the potato are constant. Applying the boundary conditions give $C1 = h[T \propto -(C1 \ln r1 + C2)]r1 r = r1$: -k r = r2: T (r2n dimensional since the thickness of the door is small Door relative to other dimensions. 3 The thermal properties of the potato are constant.) = C1 ln r2 + C2 = T2 Solving for C1 and C2 simultaneously gives C1 = T2 - T ∞ r k ln 2 + r1 hr1 and C2 = T2 - C1 ln r2 = T2 - T ∞ r k ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r + T2 - C1 ln r2 = T2 - T ∞ r k ln 2 + r1 hr1 and C2 = T2 - T ∞ r k ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r + T2 - C1 ln r2 = T2 - T ∞ r k ln 2 + r1 hr1 and C2 = T2 - T ∞ r ln + T2 r k r2 ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r + T2 - C1 ln r2 = T2 - T ∞ r k ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r2 = T2 - T ∞ r k ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r2 = T2 - T ∞ r k ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r2 = T2 - T ∞ r k ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r2 + r1 hr1 substituting C1 and C2 = T2 - T ∞ r k ln 2 + r1 hr1 substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = C1 ln r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 and C2 = T2 - T (n r2 + r1 hr1 substituting C1 + r1 hr1 substituting C1 + r1 hr1 substituting C1 + r1 hr1 (160 - 250)°F r r ln + 160°F = -24.74 ln + 160°F 2.4 7.2 Btu/h · ft · °F 2. Analysis We consider transient one-dimensional heat conduction in the axial z direction in a cylindrical rod of Convection constant cross-sectional area A with constant heat generation h, T Disk g& 0 and constant conductivity k with a mesh size of Δz in the z direction. Also to be determined is the time it takes for steady conditions to be reached. Therefore, no part of the oranges will freeze during this cooling process. 3 Thermal properties of water are constant. Analysis The heat transfer surface area of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$, the rate of the person is Tair As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ Qcon Under steady conditions}$. heat transfer by convection is Q& conv = hAs $\Delta T = (15W/m \ 2 \cdot o \ C)(1.60m \ 2)(34 - 20) \ o \ C = 336 \ W$ Room air 1-33 Chapter 1 Basics of Heat Transfer 1-74 Hot air is blown over a flat surface at a specified temperature. The center temperatures of each geometry in 10, 20, and 60 min are to be determined. Vertical ferring, 20 mm thick 6. 6-13C The fluids whose shear stress is proportional to the velocity gradient are called Newtonian fluids. Then the cooling time becomes $\tau = \alpha t r 02 \rightarrow t = .$ 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 = 0 (since g& = 0), for m = 1 and 2 k Δx 2 • 3 The finite difference equation for node 0 on the left surface subjected to uniform heat flux is obtained by applying an energy balance on the half volume element about node 0 and taking the direction of all heat transfers to be towards the node under consideration: Node 1 (interior) : T1 - T0 = 0 Δx T0 - 2T1 + T2 = 0 Node 2 (interior): T1 - 2T2 + T3 = 0 Node 4 (right surface - convection): q&0 + k where $\Delta x = 0.2$ cm, k = 20 W/m · °C, T3 = 85°C, and q&0 = Q&0 / A = (800W) /(0.0160 m 2) = 50,000 W/m 2. 1-62 The inner and outer surfaces of a brick wall are maintained at specified temperatures. Assumptions 1 Heat transfer through the walls and the windows is steady and one-dimensional. Assumptions 1 The device and the heat sink are isothermal. 3-104C The thicker fin will have higher effectiveness. This system of 10 equations with 10 unknowns constitutes the finite difference formulation of the problem. Then the energy equation with viscous dissipation reduces to 2 (∂u) d 2T (V) + μ | | $\rightarrow k = -\mu$ | 2 2 $\partial y \partial y dy (L) (J) since <math>\partial u / \partial y = V/L$. A layer of face brick is added to the outside of a wall, leaving a 20mm air space between the wall and the bricks. Therefore, the standard R-2.4 m2.°C/W wall is better insulated and thus it is more energy efficient. Solution (a) The rate of heat transfer through the shell is expressed as k(T) T - T Q& cylinder = $2\pi k$ ave L 1 2 ln(r2 / r1) T2 where L is the length of the cylinder, r1 is the inner radius, and r2 is the outer r $\Delta KE = \Delta PE = 0$ and $\Delta E = \Delta U$. 2-98C Yes, when the thermal conductivity of a medium varies linearly with temperature, the average temperature. Analysis For a 15 cm distance from the outer surface, from Fig. 2 The surface temperature of the tubes is constant. Properties The properties of the steel plate are given to be k = 60.5 W/m.°C, $\rho = 7854$ kg/m3, and Cp = 434 J/kg.°C (Table A-3). The error in this case is very small because of the large diameter to thickness ratio. 6-36 and 6-37) reduce to 2 (∂u) d 2T (V) || || + $\mu \rightarrow = -\mu$ | | k 2 2 ∂y dy (L/ (∂y) since $\partial u / \partial y = V / L$. Properties The thermal conductivities are given to be k = 386 W/m·°C for copper and 0.26 W/m·°C for copper and 0.26 W/m·°C for epoxy layers. 5 The head can be approximated as a
30-cm-diameter sphere. 3 Combined convection and radiation heat transfer coefficient is constant and uniform. Wood stud, 38 mm by 140 mm 5. Then the number of nodes M becomes M = q0 T0 L 0.3 m + 1 = +1 = 6 Δx 0.06 m Nodes 1, 2, 3, and 4 are interior nodes, and thus for them we can use the general finite difference relation expressed as $\Delta x \ 0^{\bullet} \cdot 1 \cdot 2 \cdot 3 \cdot 4 \ 5 \ \text{Tm} - 1 - 2 \text{Tm} + \text{Tm} + 1 \ \text{g} \& \text{m} + = 0 \rightarrow \text{Tm} + 1 - 2 \text{Tm} + \text{Tm} + 1 \ \text{g} \& \text{m} + = 0 \rightarrow \text{Tm} + 1 - 2 \text{Tm} + \text{Tm} + 1 \ \text{g} \& \text{m} + = 0 \rightarrow \text{Tm} + 1 - 2 \text{Tm} + \text{Tm} + 1 \ \text{g} \& \text{m} + = 0 \rightarrow \text{Tm} + 1 \ \text{g} \& \text{m} + = 0 \rightarrow \text{Tm} + 1 \ \text{m} + 1 \ \text{g} \& \text{m} + = 0 \rightarrow \text{Tm} + 1 \ \text{m} + 1 \ \text{g} \& \text{m} + = 0 \ \text{m} + 1 \ \text{m} + 1 \ \text{g} \& \text{m} + 1 \ \text{m} + 1 \ \text{g} \& \text{m} + 1 \ \text{m} + 1 \ \text{g} \& \text{m} + 1 \ \text{m} + 1 \ \text{m} + 1 \ \text{g} \& \text{m} + 1 \ \text{m} +$ energy balance on the half volume element about node 0 and taking the direction of all heat transfers to be towards the node under consideration, $q \ge 0 + k T1 - T0 = 0 \Delta x \rightarrow 700 W/m 2 + (2.5 W/m \cdot °C) T1 - 60°C = 0.06 m \rightarrow T1 = 43.2°C$ Other nodal temperatures are determined from the general interior node relation as follows: m = 1: T2 = 2T1 $-T0 = 2 \times 43.2 - 60 = 26.4$ °C m = 2: T3 = 2T2 $-T1 = 2 \times 26.4 - 43.2 = 9.6$ °C m = 3: T4 = 2T3 $-T2 = 2 \times 9.6 - 26.4 = -7.2$ °C m = 4: T5 = 2T4 $-T3 = 2 \times (-7.2) - 9.6 = -24$ °C Therefore, the temperature of the other surface will be -24°C Discussion This problem can be solved analytically by solving the differential equation as described in Chap. Nodes 2, 3, 4, and 5 are interior ε nodes, and thus for them we can use the general 6• explicit finite difference relation expressed as 5• 4• g& i $\Delta x \rightarrow Tmi + 1 = \tau$ (Tmi $-1 + Tmi + 1 = \tau$ Radiation hi, Ti convection and radiation are obtained from an energy balance by taking the direction of all heat transfers to be towards the node under consideration: [] T2i - T1i Δx T1i +1 - T1i $4 + \varepsilon \sigma$ Twall - (T1i + T3i) + (1 - 2\tau) T2i hi (Ti - T1i) + k Node 1 (convection) : Node 2 (interior) : T2i +1 Node 3 (interior) : T3i +1 = τ (T2i + T4i) + (1 - 2 τ) T3i Node 4 (interior) : T4i +1 = τ (T3i + T5i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T4i Node 5 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) : T5i +1 = τ (T4i + T6i) + (1 - 2 τ) T5i Node 6 (interior) Fo = 6°C, Tsky =260 K, hi = 5 W/m2.°C, ho = 12 W/m2.°C, ho = 12 W/m2.°C, be a transfer analysis because it be determined for steady one-dimensional heat transfer. 2-39C We try to avoid the radiation boundary condition in heat transfer analysis because it be determined for steady one-dimensional heat transfer. is a non-linear expression that causes mathematical difficulties while solving the problem; often making it impossible to obtain analytical solutions. The smallest primary coefficient in the 3 equations above is the coefficient of T1i in the T1i +1 expression since it is exposed to most convection per unit volume (this can be verified), and thus the stability criteria for this problem can be expressed as 5-88 Chapter 5 Numerical Methods in Heat Conduction $1 - 4\tau - 4\tau hl \ge 0 k \rightarrow \tau \le 1 4(1 + hl/k)$ since $\tau = \alpha \Delta t/12$. Analysis (a) Noting that the 90% of the 500 W generated by the strip heater is transferred to the container, the heat flux through the outer surface is determined to be q& s = Q& s Q& s 0.90 × 500 W = = 213.0 W/m 2 A2 4π (0.41 m) 2 Noting that heat transfer is one-dimensional in the radial r direction, the mathematical formulation of this problem can be expressed as d (2 dT) = 0 r dr (dr) Insulation T1 k T (r1) = T1 = 100° C and Heater dT (r2) k = q& s dr (r2) k = radial r direction and heat flux is in the negative r direction. dr r1 r2 (b) Integrating the differential equation once with respect to r gives r2 dT = C1 dr Dividing both sides of the equation above by r2 and then integrating, dT C1 = dr r 2 T (r) = -C1 + C2 r where C1 and C2 are arbitrary constants. Assumptions 1 Heat transfer along the fin is given to be steady and one-dimensional. o C)(1.2 × 1.8)m 2 1 1 1 1 $= +5 = +5 \rightarrow \text{Regv} = 0.00058 \text{ o C/W } 0.002968 \text{ R wall} \text{ Rglass } 0.033382 \text{ R wall} = \text{Rglass } 1 \text{ Regv} + \text{Ro} = 0.000833 \text{ °C/W ho A } (15 \text{ W/m 2} \cdot \text{°C})(20 \times 4 \text{ m 2}) \text{ R total} = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{ Ro} = 0.000833 \text{ e C/W } 11 \text{$ Heat Conduction 4th wall with double pane windows: Rglass Ri Rair Rwall Rglass Ro L wall R - value 2.31 m 2 °C/W kA (20×4) - 5(1.2 × 1.8)m 2 L 0.015 m = air = 0.267094 °C/W kA (0.026 W/m 2. The rate of heat loss through the window is to be determined. Properties The properties of engine oil at the film temperature of $(Ts + T\infty)/2 = (80+30)/2 = 55^{\circ}C = 328$ K are (Table A-13) $\rho = 867$ kg/m 3 $\nu = 123 \times 10 - 6$ m 2/s k = 0.141 W/m. °C Pr = 1505 Analysis Noting that L = 6 m, the Reynolds number at the end of the plate is V L (3 m / s)(6 m) Re L = $\infty = .$ It can be shown that the steady state solution at node 3 is 531°C. 2 Heat conduction in the meat slabs is one-dimensional because of the symmetry about the centerplane. 3-3C The temperature distribution in a
plane wall will be a straight line during steady and one dimensional heat transfer with constant wall thermal conductivity. for transient heat conduction to be applicable, the one-term solution formulation at one-third the radius from the center of the turkey can be expressed as $2 \sin(\lambda 1 r / ro) T(x, t) - T \propto = A1e -\lambda 1 \tau \theta(x, t) = Ti - T \propto \lambda 1 r / ro 2 \sin(0.333\lambda 1) + 185 - 325 = 0.491 = A1e -\lambda 1 (0.14) + 40 - 325 = 0.333\lambda 1$ By trial and error, it is determined from Table 4-1 that the equation above is satisfied when Bi = 20 corresponding to $\lambda 1 = 2.9857$ and A1 = 1.9781. Properties of air at 1 atm and the film temperature of (Ts + T ∞)/2 = (0-55.4)/2 = -27.7°C are (Table A-15) k = 0.02152 W/m.°C $\nu = 1.106 \times 10^{-5} m 2$ /s Pr = 0.7422 Note that the atmospheric pressure will only affect the kinematic viscosity. 3-104 Chapter 3 Steady Heat Conduction 3-142 The winter R-value and the U-factor of a flat ceiling with an air space are to be determined for the cases of air space are to be determined. 4 The thermal contact resistance between the two layers is negligible. Analysis Disregarding any heat loss through the bottom of the ice chest and using the average thicknesses, the total heat transfer area becomes A = $(40 - 3)(40 - 3) + 4 \times (40 - 3)(30 - 3) = 5365 \text{ cm } 2 = 0.5365 \text{ m } 2$ The rate of heat transfer to the ice chest becomes $\Delta T (8 - 0)^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m})^\circ C = 4.72 \text{ W} = (0.033 \text{ W} / \text{m}$ Properties The effective thermal conductivity of the board is given to be $k = 16 \text{ W/m} \cdot \text{c}$. The Reynolds number is V L [70 × 1000/3600) m/s](8 m) Re L = $\infty = 9.674 \times 10.608 \times 10^{-5} \text{ m}^2/\text{s}^2$ (200 W/m² Air V $\infty = 70 \text{ km/h}$ T $\infty = 30^{\circ}$ C L which is greater than the critical Reynolds number. The surface temperature of the spacecraft is to be determined when steady conditions are reached.. It is to be determined if the new insulation will pay for itself within 2 years. T2=40°C Noting that u = u(y), v = 0, and $\partial P / \partial x = 0$ (flow is maintained by the motion of the upper plate rather than the pressure L=0.4 mm Oil gradient), the x-momentum equation reduces to ($\partial u d 2u \partial u$) $\partial 2 u \partial P \rightarrow 2 = 0$ + $v = \mu 2 - \rho | u \partial y / \partial x dy \partial y \langle \partial x dy \partial y \rangle$ at the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces at the plate surf inner and outer surfaces are common in all cases. Properties The thermal properties of the wood are k = 0.17 W/m. C and $\alpha = 1.28 \times 10-7 \text{ m}/s$. Wall Analysis The surface area of the wall and the rate of heat loss through the wall are A = (4 m) × (6 m) = 24 m 2 L=0.3 m T - T (14 - 6)^{\circ} C = 512 W Q \& = kA 1 2 = (0.8 W / m. m) = 0.03142 m T can 1 1 = 0.03142 m $= 3.183 \text{°C/W Ro} = \text{ho Ao} (10 \text{ W/m } 2.^{\circ}\text{C})(0.03142 \text{ m } 2) 2 \text{ Rinsulation} + \text{Rcontact} = 3.183 + 2.818 + 0.0034 = 6.004 \text{°C/W Tair} - \text{Tcan,ave} (25 - 6.5)^{\circ}\text{C} = 3.08 \text{ W}$ Q_{k} side = = 6.004 °C/W Rconv, o Rcontact = The ratio of bottom to the side surface areas is ($\pi 2$) /($2\pi L$) = r /(2L) = 3 /(2×12.5) = 0.12. Discussion We could avoid the uncertainty associated with the reading of the charts and obtain a more accurate result by using the one-term solution relation for an infinite plane wall, but it would require a trial and error approach since the Bi number is not known. (b) The nodal temperatures under steady conditions are determined by solving the 4 equations above simultaneously with an equation solver to be T1 = 129.2°C, T2 = 128.7°C, T3 = 128.3°C, T4 =128.2°C (c) The rate of heat transfer from a single fin is simply the sum of the heat transfer from each nodal element, $M = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ Q& element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ A element, $m = 4 \sum m = 0$ element, $m = 4 \sum m = 0$ element, = = 286 fins Fin thickness + fin spacing (0.003 + 0.004) m Then the rate of heat tranfer from the fins, the unfinned portion, and the entire finned surface become Q& = (No. of fins)Q& = 286(363 W) = 103,818 W fin, total fin Q& `unfinned = hAunfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned = hAunfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned = hAunfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned = hAunfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned = hAunfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total =
Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(286 × 3 m × 0.004 m)(130 - 35)°C = 9781 W Q& total = Q& fin, total fin Q& `unfinned (T0 - T ∞) = (30 W/m 2 · °C)(2 total + Q& unfinned = 103,818 + 9781 = 113,600 W \cong 114 kW 5-27 Chapter 5 Numerical Methods in Heat Conduction 5-36 One side of a hot vertical plate is to be cooled by attaching aluminum pin fins. $e^-(1.9569) 2\tau \rightarrow \tau = 0.426 = 144778 - 25$ which is greater than 0.2 and thus the one-term solution is applicable. Then the number of nodes becomes $M = L / \Delta x + 1 = 1/0.25 + 1 = 4$. Then the number of nodes becomes 5 in 3 ft (L) (L) M = | + | + 1 = + + 1 = 111 in 0.6 ft $(\Delta x / \text{plate} (\Delta x / \text{plate} + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 111)$ in 0.6 ft $(\Delta x / \text{plate} + 1 = + 1 = 11)$ $= 0.543 = A1e - \lambda 1$ (0.1881) 4.5 - 163 Ti $- T\infty$ It is determined from Table 4-1 by trial and error that this equation is satisfied when Bi = 4.3, which corresponds to $\lambda 1 = 2.4900$ and A1 = 1.7402. 4 Heat loss from the fin tip is given to be negligible. Therefore, it is more convenient to use the lumped system analysis in this case. The energy balance for this steady-flow system can be expressed in the rate form as E& - E& 1in424out 3 = Rate of net energy transfer by heat, work, and mass $\Delta E\&$ system $\hat{E}0$ (steady) 144 42444 3 = 0 $\rightarrow E\&$ in = E& out Rate of change in internal, kinetic, potential, etc. 3 Thermal properties of the milk are 60°C constant at room temperature. Also, we would place the origin somewhere on the center line, possibly at the center of the hot dog. Analysis We take both cars as the system. Analysis It is given that D = 0.008 m, SL = ST = 0.015 m, and V = 4 m/s. 7-2C A body is said to be streamlined if a conscious effort is made to align its shape with the anticipated streamlines in the flow. Analysis The rate of heat generation is determined from W& W& 25,000 W g& = = = 37,894 W/m 3 2 2 2 V π (D 2 - D1) L / 4 π (0.4 m) - (0.3 m) 2 (12 m) / 4 [] Noting that heat transfer is 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 T (r1) = T1 = 60°C and g T1 T (r2) = T2 $= 80^{\circ}$ C T2 Rearranging the differential equation again dT - g& r = - 1 dr k r = - 1 dr k r = - g& r = - 1 dr k r = - g& r = - 1 dr k r = - g& r = - 1 dr k r = - 2 k r = - 1 dr k r = - 2 k r = - 1 dr k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k r = - 2 k arbitrary constants. The number of tube rows is to be determined. It allows us to calculate the heat transfer 2 coefficient from a knowledge of friction coefficient. Properties of air at 1 atm and the film temperature of $(Ts + T\infty)/2 = (75+5)/2 = 40^{\circ}C$ are (Table A-15) Wind $V \infty = 10$ km/h $T \infty = 5^{\circ}C$ k = 0.02662 W/m. $^{\circ}C \nu = 1.702 \times 10^{-5}$ m $282,000 \mid 45 \mid 0.62(1.632 \times 104) 0.5 (0.7255) 1/3 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 + 1 + 1/4 \mid 1.632 \times 104 = 0.3 \mid 1.632 \times 104 \mid 1.632 \times 104$ $(18.95
\text{ W/m 2})(3.77 \text{ m 2})(75 - 5)^{\circ}C = 5001 \text{ W s s} \propto \text{For an average surrounding temperature of 0 ° C}$, the rate of heat loss by radiation and the total] = $(0.8)(3.77 \text{ m 2})(5.67 \times 10 - 8 \text{ W/m 2} \cdot K 4)(75 + 273 \text{ K}) 4 - (0 + 273 \text{ K}) 4 = 1558 \text{ W} = Q_{\&} + Q_{\&} = 5001 + 1588 = 6559 \text{ W conv}$ rad If the average surrounding temperature is -20 °C, the rate of heat loss by radiation and the total rate of heat loss become Q& = $\epsilon A \sigma (T 4 - T 4) rad s s surr 2 [= (0.8)(3.77 m)(5.67 \times 10 - 8 W/m 2 . K 4) (75 + 273 K) 4 - (-20 + 273 K)$ than the value for a surrounding temperature of 0°C. Once the unit thermal resistances and the U-factors for the air space and stud sections are available, the overall = 1/Uoverall where Uoverall = (Ufarea)air space + (Ufarea)air space + (Ufarea)air space and stud sections are available, the overall average thermal resistance for the air space and stud sections are available, the overall = 1/Uoverall where Uoverall = 1/Uoverall where Uoverall = 1/Uoverall where Uoverall = 1/Uoverall average thermal resistance for the air space and stud sections are available, the overall = 1/Uoverall where Uoverall = 1/Uoverall = 1/Uoverall is 0.84 for air space and 0.16 for the ferrings and similar structures. 2 The thermal properties of the body are constant. 2-4C Heat transfer) will exist in the radial direction only because of symmetry about the center point. For example resistance heating in wires is conversion of electrical energy to heat. The heat generation and the heat flux are to be determined. Properties The thermal conductivity and diffusivity of the eggs can be approximated by those of water at room temperature to be k = 0.607 W/m. $^{\circ}C_{\mu} = 0.146 \times 10^{-6} \text{ m}^2/\text{s}$ (Table A-9). Then ϵ the number of nodes $90)^{F} + 0.6(0.1714 \times 10-8 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{F} + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)[(535 \text{ R}) 4 - (510 \text{ R}) 4] = 75^{O}F + (510 \text{ R}) 4] = 75^{O}F + (4/12 - x) \text{ ft } 7.2 \text{ Btu/h} \cdot \text{ft } 7.2 \text{$ temperature on the top surface and no conditions on the bottom surface. 5-4C In practice, we are most likely to use a software package to solve heat transfer problems even when analytical solutions are available since we can do parametric studies very easily and present the results graphically by the press of a button. Then the continuity equation reduces to $\partial u \partial v \partial u$ Continuity: $+ = 0 \rightarrow u = u(y) = 0 \partial x \partial x \partial y 12$ m/s Therefore, the x-component of velocity does not change in the flow direction (i.e., the velocity profile remains unchanged). The thermal conductivity of the plate can be determined from $\alpha = k/(\rho Cp) = 177$ W/m. °C (or it can be read from Table A-3). 2 Thermal properties, heat transfer coefficients, and the indoor temperatures are constant. No heat can cross a symmetry lines can be treated as insulated • • surfaces and thus symmetry lines can be treated as insulated side represents the center surface of a plate whose thickness is doubled. To determine the center temperature, the product solution method can be written as $\theta(0,0,0,t) - T \propto (2 T (0,0,0,t) - T \propto (2 T (0,0,0,t) - 500 = (1.0580)e - (0.5932) (1.104) 20 - 500 \} \{(1.1016)e 2 - (0.7910)e - (0.7910)e - (0.7910)e - (0.5932)e - (0.$) 2 (1.104) T (0,0,0, t) = $364^{\circ}C$ After 20 minutes $\tau = \alpha t 2 L = (115.)$. °F Air V $\infty = 20$ mph T $\infty = 54^{\circ}F \upsilon = 0.1643 \times 10 - 3$ ft 2/s Arm D = 3 in The Nusselt number corresponding this Reynolds number is determined to be Nu = 0.5 1/3 hD 0.62 Re Pr = 0. Properties The thermal properties of the aluminum block are given to be k = 236 W/m.°C, $\rho = 2702$ kg/m3, Cp = 0.896 kJ/kg.°C, and $\alpha = 9.75 \times 10^{-5}$ m2/s. Therefore, the fins formed by casting or extrusion will provide greater enhancement in heat transfer. 4-98C The proper storage temperature of frozen poultry is about -18° C or below. Properties The thermal conductivity is given to be k = 8.6 Btu/h·ft·°F. °C)(0.30 × 1 m 2) 1 1. The mathematical formulation, the variation of temperature, and the rate of evaporation of nitrogen are to be determined for steady one-dimensional heat transfer. Properties Assuming a film temperature of 10°C, the properties of air are (Table A-15) k = 0.02439 W/m.°C Air $v = 1.426 \times 10^{-5}$ m 2 /s Tsky = 100 K V $\infty = 60$ km/h T $\infty = 10^{\circ}$ C Pr = 0.7336 Analysis The Reynolds number is V L [($60 \times 1000 / 3600$) m/s](20 m) Re L = $\infty = = 2.338 \times 10^{-7} - 52 v 1.426 \times 10$ m /s Tin = 20° C which is greater than the critical Reynolds number. It yields Tleft + Ttop + Tright + Tbottom -4Tnode + g& node 1 2 k = 0 5-67C There is a limitation on the size of the time step Δt in the solution of transient heat conduction problems using the explicit method, but there is no such limitation in the implicit method. Assumptions 1 Heat transfer is steady since there is no such limitation for the potatoes will experience chilling. injury during this cooling process. 5-104C A practical way of checking if the round-off error has been significant in calculations is to repeat the calculations by k and integrating twice give ∂ 2T 2 2 dT μ (V) = $-||y + C3 dy k(L/2T(y))| = -\mu(y)|$ V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V = 0 V =by differentiating T(y) with respect to y, dT - μ V 2 y = dy kL2 . The heat flux at the upper surface is q& L = -k dT dy = k y=L μ V 2 kL2 L = μ V 2 k

98.1 W Q& = As q& L = (π DW) 0.001 m L (b) This is equivalent to the rate of heat transfer through the cylindrical sleeve by conduction, which is expressed as 2π W (T0 - Ts) Q& = k ln(D0 / D) \rightarrow (70 W/m \cdot °C) 2π (0.15 m)(T0 - 40 °C) = 98.1 W ln(8 / 5) which gives the surface temperature of the shaft to be To = 40.7 °C (c) The mechanical power wasted by the viscous dissipation in oil is equivalent to the rate of heat generation, W& lost = Q& = 98.1 W 6-24 Chapter 6 Fundamentals of Convection Momentum and Heat Transfer to the air and its cost are to be determined. Assumptions 1 The water temperature remains constant. Noting that D = D0 = 4.1 m, the Nusselt number becomes Re = V $\propto D \left[(40 \times 100/3600) \text{ m/s} \right] (4.1 \text{ m}) = 3.005 \times 10.6 \text{ Re } 2/3 \text{ Pr } 0.4 || <math>\propto k \left(\mu \text{ h} D = 2 + 0.4 \text{ (3.005} \times 10.6 \text{)} 0.5 + 0.06(3.005 \times 10.6 \text{)} 2/3 (0.7309) 0.4 || -5 \sqrt{10.6} \text{ m}^2 - 5 \sqrt{10.6}$ $1.189 \times 10\ 0.02514\ W/m.^{\circ}C\ k\ (1910) = 11.71\ W/m\ 2.^{\circ}C\ h = Nu = and\ D\ 4.1\ m\ The\ rate of\ heat\ transfer\ to\ the\ liquid\ nitrogen\ is\ 7-96\ 1/4\)$ -2) m 1 + 4 π (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& 6.918 kJ/s Q& = m& hif \rightarrow m& = = 0.0325 kg/s hif 213 kJ/kg (c) Again we use the dynamic viscosity value at the estimated surface temperature of 0°C to be μ = 1.729 × 10 -5 kg/m.s. The inverse of thermal contact resistance is called the thermal conductivity of the wall is given to be $k = 0.69 \text{ W/m} \cdot \text{C}$. Properties The thermal conductivities are given to be $k = 0.69 \text{ W/m} \cdot \text{C}$ for cast iron pipe and $k = 0.038 \text{ W/m} \cdot \text{C}$ for fiberglass insulation. ft 2 Ao = π Do L = π (0.6 / 12 ft)(1 ft) = 0157 . at r = r0 : T (r0) = T ∞ + -k& gr gr g& 2 (1061 . • • • • D • 0 1 2 Analysis The nodal network of this problem consists of 5 nodes, and 3 4 the base temperature T0 at node 0 is specified. Properties It is given that the specific heats of turkey are 2.98 and 1.65 kJ/kg.°C above and below the Ti = 1°C freezing point of -2.8°C, respectively, and the latent heat of fusion of turkey is 214 kJ/kg. 2 Thermal conductivity varies linearly. kg)(333.7 kJ / kg) = 52.4 kJ 2 3 2 The water in the pipe will freeze the water completely (8357. πD p 8-6C The region from the tube inlet to the point at which the boundary layer merges at the centerline is called the hydrodynamic entry region, and the length of this region is called hydrodynamic entry length. Assumptions 1 Heat transfer through the wall is given to be steady and one-dimensional. The smallest primary coefficient in the 8 equations above is the coefficient of T3i in the T3i + 1 expression since it is exposed to most convection per unit volume (this can be verified), and thus the stability criteria for this problem can be expressed as $1 - 4\tau - 4\tau h \ge 0$ k $\rightarrow \tau \le 1$ 4(1 + hl / k) $\rightarrow \Delta t \le 12$ 4 α (1 + hl / k) since $\tau = \alpha \Delta t / 12$. 4 m +1 = +1 = 5 Δx 0. 1-46C In solids, conduction is due to the combination of the wibrations of the molecules in a lattice and the energy transport by free electrons. Microorganisms can be destroyed by heat treatment, chemicals, ultraviolet light, and solar radiation. The temperature of the top surface of the roof and the rate of heat transfer coefficient are determined to be h L Nu = o = $0.037 \text{ Re L } 0.8 \text{ Pr } 1/3 = 0.037(5.704 \times 10.6)$ $0.8 (0.7290)1/3 = 8461 \text{ k} \text{ } 0.01481 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = \text{Nu} = (8461) = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = 11.39 \text{ Btu/h.ft.}^{\circ}\text{Btu/h.ft.}^{\circ}\text{F} \text{ ho} = 11.39 \text{ Btu/h.ft.}^{\circ}\text{F} \text{ ho} = 11.39 \text{ Btu/h$ $= 0.0125 \text{ h.}^{\circ}\text{F/Btu As}$ (240.8 ft 2) 1 1 = $0.0004 \text{ h.}^{\circ}\text{F/Btu Ro}$ = ho As (11.39 Btu/h.ft 2. $^{\circ}\text{F}$)(240.8 ft 2) Rinsulation = Then the total thermal resistance and the heat transfer rate into the minivan are determined to be Rtotal = Ri + Rinsulation + Ro = $0.0035 + 0.0125 + 0.0004 = 0.0164 \text{ h.}^{\circ}\text{F/Btu T} - T (90 - 70)^{\circ}\text{F}$ O& $= \infty 1 \infty 2 = 1220 \text{ Btu/h}$ Rtotal 0.0164 h.°F/Btu 7-85 T∞2 Chapter 7 External Forced Convection 7-93 Wind is blowing parallel to the walls of a house with windows. Highly reflective materials such as aluminum foil or alu hot dog will change with time during cooking. Analysis We take the ice and the water as our system, and disregard any heat transfer is a the inner and outer surfaces are comparable in magnitude. Assumptions 1 Heat transfer is RT1 (0.287 kPa · m3 / kg · K)(288 K) The electrical work done by the fan is $W = W \& \Delta t = (0.15 \text{ kJ / s})(10 \times 3600 \text{ s}) = 5400 \text{ kJ} + (0.15 \text{ kJ / s})(0.720 \text{ kJ/kg.}^{\circ}C)(T2 - 15)^{\circ}C$ T2 = 58.1°C 1-12 · We Chapter 1 Basics of Heat Transfer 1-32E A paddle wheel in an oxygen tank is rotated until the pressure inside rises to 20 psia while some heat is lost to the surroundings. ° C) Since Bi < 0.1, the lumped system analysis is applicable. The finite difference formulation of the problem for all nodes is to be determined. $C = 0.0031 \circ C + 0.012 \text{ mL} = 0.0031 \circ C + W \text{ kA}$ (386 W / m. Modified 6-48C C f, x 2 = Reynolds analogy is expressed as C f, x Re L = Nu x Pr - 1 / 3 2 or hx Pr 2/3 = j H. The wall is insulated at the left (node 0) and subjected to radiation at the right boundary (node 2). The transient finite difference formulation of this problem is to be obtained, and it is to be determined how long it will take for the fog on the windows to clear up (i.e., for the inner surface temperature distribution of the medium initially is called the initial condition. 4-7C The heat transfer is proportional to the surface area. Assumptions 1 The fins are sufficiently long so that the temperature of the fin at the tip is nearly $T \propto .2.77$ A 2-kW resistance heater wire with a specified surface temperature is used to boil water. Properties The thermal conductivity of the orange is given to be k = 0.50 W/m.°C. Thus we need only 3 equations to determine them uniquely. The 6 nodal temperatures under steady conditions are determined by solving the 6 equations above simultaneously with an equation solver to be T1 = Ttop = 21.3° C, T2 = 15.1° C, T3 = Tmiddle = 43.2° C, T4 = 15.1° C, T5 = 36.3° C, T6 = Tbottom = 43.6° C Discussion Note that the highest temperature occurs at a location furthest away from the water, as expected. 7 24 xcr Chapter 7 External Forced Convection 7-33 Water is flowing over a long flat plate with a specified velocity. Properties The properties of the watermelon are given to be k = 0.618 W/m.°C, $\alpha = 0.15 \times 10-6$ m2/s, $\rho = 995$ kg/m3 and Cp = 4.18 kJ/kg.°C. 2-73 Chapter 2 Heat Conduction Equation 2-133 A cylindrical shell with variable conductivity is subjected to specified temperatures on both sides. 4-15b) $r T0 - T\infty$ = 1 | r0 which gives Tsurface = T ∞ + 0.6(To - T ∞) = 2 + 0.6(To - T ∞) = 2 + 0.6(To - T ∞) = 2 + 0.6(To - T) Transient Heat Conduction 4-50 A person puts apples into the freezer to cool them quickly. Properties The thermal conductivity of the tissue near the skin is Qrad given to be k = $0.3 \text{ W/m} \cdot \text{°C}$. Properties The thermal conductivity of the tissue near the skin is Qrad given to be k = $0.3 \text{ W/m} \cdot \text{°C}$. Properties The thermal conductivity of the tissue near the skin is Qrad given to be k = $0.3 \text{ W/m} \cdot \text{°C}$. hA (T - T) s s Pr = 0.7362 Air 0°C 80 km/h \propto Q& h= As (Ts - T \propto) = 50 W (0.6×1.8 m)(4 - 0)°C 2 Windshield Ts=4°C = 11.57 W/m · °C)(0.7362) 2hPr = 0.0006534ρ V C p
(1.292 kg/m 3)(80 / 3.6 m/s)(1006 J/kg · °C) The drage force is determined from Cf = F f = C f As ρ V 2 2 2/3 = 0.0006534(0.6 × 1.8 m 2) 2 2/3 (1.292 kg/m 3)(80 / 3.6 m/s) 2 2 (1N | 1 kg.m/s 2 \) | = 0.225 N | / 6-53 An airplane cruising is considered. Properties of the orange are constant. The volumetric absorption coefficients of water are as given in the problem. 3-5 Chapter 3 Steady Heat Conduction 3-21 "GIVEN" A=1.2*2 "[m^2]" L glass=0.78 "[W/m^2-C]" h_2=25 $k_air=conductivity(Air,T=25)$ "ANALYSIS" $R_conv_1=1/(h_1*A)$ $R_glass=(L_glass*Convert(mm, m))/(k_glass*A)$ $R_air=(L_air*Convert(mm, m))$ $R_air=(L_air*Convert(mm, m))$ $R_air=(L_air*Convert(mm$ 89.82 81.57 74.7 300 Q [W] 250 200 150 100 50 2 4 6 8 10 12 L air [m m] 3-6 14 16 18 20 Chapter 3 Steady Heat Conductivity and emissivity are given to be k = 52 W/m °C and $\varepsilon = 52$ W/m °C and $\varepsilon = 52$ 0.8. Tsurr Analysis (a) The distance between nodes 0 and 1 is the thickness ho, T∞ of the pipe, $\Delta x1=0.4$ cm=0.004 m. Properties The average emissivity of the person is given to be 0.7. Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are (a) Tsurr = 300 K 4 (b) Tsurr = 280 K 4 (cases are (a) Tsurr = $(0.7)(5.67 \times 10 - 8 \text{ W/m 2} \cdot \text{K 4})(1.7 \text{ m 2})[(32 + 273)4 - (300 \text{ K})4] \text{K 4} = 37.4 \text{ W}$ (b) Tsurr = 280 K 4) Q& rad = $\varepsilon\sigma \text{As}$ (Ts4 - Tsurr Qrad = $(0.7)(5.67 \times 10 - 8 \text{ W/m 2} \cdot \text{K 4})(1.7 \text{ m 2})[(32 + 273)4 - (300 \text{ K})4] \text{K 4} = 37.4 \text{ W}$ (b) Tsurr = 280 K 4) Q& rad = $\varepsilon\sigma \text{As}$ (Ts4 - Tsurr Qrad = $(0.7)(5.67 \times 10 - 8 \text{ W/m 2} \cdot \text{K 4})(1.7 \text{ m 2})[(32 + 273)4 - (300 \text{ K})4] \text{K 4} = 37.4 \text{ W}$ (b) Tsurr = 280 K 4) Q& rad = $\varepsilon\sigma \text{As}$ (Ts4 - Tsurr Qrad = $(0.7)(5.67 \times 10 - 8 \text{ W/m 2} \cdot \text{K 4})(1.7 \text{ m 2})[(32 + 273)4 - (300 \text{ K})4] \text{K 4} = 37.4 \text{ W}$ (b) Tsurr = 280 K 4) Q& rad = $\varepsilon\sigma \text{As}$ (Ts4 - Tsurr Qrad = $(0.7)(5.67 \times 10 - 8 \text{ W/m 2} \cdot \text{K 4})(1.7 \text{ m 2})[(32 + 273)4 - (300 \text{ K})4] \text{K 4} = 37.4 \text{ W}$ (b) Tsurr = 280 K 4) Q& rad = $\varepsilon\sigma \text{As}$ (Ts4 - Tsurr Qrad = $(0.7)(5.67 \times 10 - 8 \text{ W/m 2} \cdot \text{K 4})(1.7 \text{ m 2})[(32 + 273)4 - (300 \text{ K})4] \text{K 4} = 37.4 \text{ W}$ (b) Tsurr = 280 K 4) Q& rad = $\varepsilon\sigma \text{As}$ (Ts4 - Tsurr Qrad = $(0.7)(5.67 \times 10 - 8 \text{ W/m 2} \cdot \text{K 4})(1.7 \text{ m 2})[(32 + 273)4 - (300 \text{ K})4] \text{K 4} = 37.4 \text{ W}$ (b) Tsurr = 280 K 4) Q its thickness. Properties The thermal properties of the brick are given to be k = 0.72 W/m.°C and α = 0.45×10-4 m2/s. Drag is caused by friction between the fluid and the solid surface, and the pressure difference between the fluid and the solid surface. temperature of the refrigerator to be 10°C. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined. Heat transfer analysis. This problem involves 5 unknown nodal temperatures, and thus we need to have 5 equations. Heat generation varies with location in the x direction. Properties are given to be k = 223 Btu/h·ft·°F for copper and 0.15 Btu/h·ft·ft·ft·°F for copper and Rpipe + Rconv, $o = 0.3627 + 2.6526 = 3.0153 \circ C / WT - T[0 - (-5)] \circ CWQ = 0.5575 + 0.0025) = 0.56$ Btu / h rough the pipe are to be determined. $\circ F = 2(015 \text{ epoxy} (\text{kt}) \text{ total} = (\text{kt}) \text{ copper} + (\text{kt}) \text{ epoxy} = (0.5575 + 0.0025) = 0.56$ Btu / h 4-23 we have 2-62 have $(20 \text{ W/m} \cdot \text{C})(1.6 \times 10 \text{ m/s})(2 \times 3600 \text{ s}) = 2.98$ = $|T - T_{\infty} 0.72 \text{ W/m} \cdot \text{C} k = 0.25$ $+ T = 14.0^{\circ}\text{C} 18 - 2$ For a 30 cm distance from the outer surface, from Fig. C)(0.65 \text{ m } 2) 1 1 1 1 = + = + \rightarrow R mid = 4.633 °C/W R 2 R3 21.818 5.882 Ro = 1 R mid Rtotal = Ri + R1 + R mid + R 4 + Ro = 0.185 + 0.090 + 4.633 + 0.090 + 4.633 + 0.090 + 0.045 = 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) Q& total = (5.15 W) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) = 475 W 0.65 m 2 3-31 Chapter 3 2 - 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) = 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entire wall becomes (12 m)(5 m) = 4.858 °C/W Rtotal (b) Then steady rate of heat transfer through entine wall becomes (12 m)(5 m) = 4.858 °C/W Rt Steady Heat Conduction 3-56E A wall is to be constructed using solid bricks or identical size bricks with 9 square air holes. The upper limit of the time step Δt is determined from the stability criteria that requires the coefficient of Tmi in the Tmi+1 expression (the primary coefficient) be greater than or equal to zero for all nodes. Then, h, T ∞ m= 1: TO $-2T1 + T2 + h(p\Delta x 2 / kA)(T \infty - T1) = 0 T0 m = 2: T1 - 2T2 + T3 + h(p\Delta x 2 / kA)(T \infty - T2) = 0 \Delta x m = 3: T2 - 2T3 + T4 + h(p\Delta x 2 / kA)(T \infty - T4) = 0 \cdot 1 \cdot 2 \cdot 3 \cdot 45 \cdot 6 m = 5: T4 - 2T5 + T6 + h(p\Delta x 2 / kA)(T \infty - T5) = 0 Node 6: kA T5 - T6 + h(p\Delta x 2 / kA)(T \infty - T6) = 0 \Delta x where$ $\Delta x = 0.005 \text{ m}, \text{k} = 386 \text{ W/m} \cdot ^{\circ}\text{C}, \text{T} = 100^{\circ}\text{C}, \text{h} = 35 \text{ W/m} 2 \cdot ^{\circ}\text{C}$ and $\text{A} = \pi \text{D} 2 / 4 = \pi (0.25 \text{ cm}) 2 /$ $=97.5^{\circ}C$, T3 =96.7°C, T4 =96.0°C, T5 =95.7°C, T6 =95.5°C (c) The rate of heat transfer from a single fin is simply the sum of the heat transfer from the nodal elements, 6 Q& fin = Σ m =0 Q& element, m = 6 Σ hA surface, m (Tm - T ∞) m =0 = hp\Delta x / 2(T0 - T ∞) + hp $\Delta x(T1 + T2 + T3 + T4 + T5 - 5T\infty)$ + h(p $\Delta x / 2 + A$)(T6 - T ∞) = 0.5641 W (d) The number of fins on the surface is No. of fins = 1m2 = 27,778 fins (0.006 m) (0.006 m) Then the rate of heat tranfer from the fins, the unfinned portion, and the entire finned surface become Q& = (No. of fins) Q& = 27,778 (0.5641 W) = 15,670 W fin, total fin Q& `unfinned = hAunfinned (T0 - T ∞) = (35 W/m 2 · °C)(1 - 27,778 × 0.0491 × 10 - 4 m) 2)(100 - 30)°C = 2116 W Q& total = Q& fin, total + Q& unfinned = 15,670 + 2116 = 17,786 W \cong 17.8 kW 5-29 Chapter 5 Numerical Methods in Heat Conduction 5-38 Two cast iron steam pipes are connected to each other through two 1-cm thick flanges, and heat is lost from the flanges by convection and radiation. 2 Heat transfer can be approximated as being one-dimensional, although it is recognized that heat conduction in some parts of the plate will be two-dimensional since the plate area is much larger than the base area of the transistor. 4 - x) m 8.4 W/m · °C = 45 + 48.23(0.4 - x) T (x) = C1x + (T2 - C1L) = T2 - (L - x)C1 = T2 + (c) The temperature at x = 0 (the left surface of the wall) is T (0) = 45 + 48.23(0.4 - 0) = 64.3°C 2-69 x Chapter 2 Heat Conduction Equation 2-129 The base plate of an iron is subjected to specified heat flux on the left surface. The Pr is a fluid property, and thus its value is independent of the type of flow and flow geometry. Conduction is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles. m) = $0.353 \text{ kg} (4180 \text{ J} / \text{kg})(10 - 3)^\circ \text{C} = 10,329 \text{ J}$ Then the time required for this much heat transfer to take place is Q 10,329 J = 1912 s = 31.9 min 5.4 J / s Q Wenow repeat calculations after wrapping the can with 1-cm thick rubber insulation, except the top surface. Then the amount of natural gas consumed per year and its cost are 2.511×10 5 kJ (1 therm) || ||(110 days/yr) = 327.2 therms/yr 0.80 \ 105,500 kJ / Cost of fuel = (Amount of fuel)(Unit cost of fuel) Fuel used = = (327.2 therms/yr)(\$0.60/therm) = 196.3/yr Then the money saved by reducing the heat loss by 90% by insulation becomes Money saved = $0.9 \times (0.0216 \text{ m}^2) = 0.9 \times (0.0216 \text{$ fin = Afinned = η fin nnDL = 0.973 × 864 π (0.0025 m)(0.02 m) = 0132 . 4 There is no heat generation in
the wall. Therefore, we need only 1 initial condition for a two-dimensional since the plate is large relative to its thickness. Properties The thermal conductivities are k = 0.00015 W/m·°C for super insulation, k = 0.01979 W/m·°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C for fiberglass insulation (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (Table A-15) L = 0.7 m k = 0.02662 W/m.°C at -50°C (T m 2 /s Pr = 0.7255 Analysis The Reynolds number is V L [(60 × 1000 / 3600) m/s](0.7 m) Re L = ∞ = 6.855 × 10 5 υ 1.702 × 10 - 5 m 2 /s Air V ∞ = 60 km/h T ∞ = 5°C Ts = 75°C ε = 0.92 Ts = 10°C which is less than the critical Reynolds number. °F (b) The temperature at the surface of the turkey is 2 2 sin(λ 1 ro / ro) T (ro , t) - 325 sin(2.9857) = A1e $-\lambda 1 \tau = (1.9781)e^{-(2.9857)}(0.14) = 0.02953 40 - 325 2.9857 \lambda 1 \text{ ro} / \text{ro} \rightarrow T \text{ (ro}, t) = 317 °F (c) The maximum possible heat transfer is Qmax = mC p (T_{\infty} - Ti) = (14 \text{ lbm})(0.98 \text{ Btu/lbm.°F})(325 - 40)°F = 3910 \text{ Btu} Then the actual amount of heat transfer becomes sin(<math>\lambda 1$) $-\lambda 1 \cos(\lambda 1) Q \sin(2.9857) - (2.9857) \cos(2.9857) = 1 - 3\theta o$, sph = 1 - 3(0.491) = 0.8283 Qmax (2.9857) $3 \lambda 1 Q = 0.828Q$ max = (0.828)(3910 Btu) = 3238 Btu Discussion The temperature of the outer parts when the turkey is taken out of the oven. The center and surface temperatures of the chickens are to be determined, and if any part of the chickens will freeze during this cooling process is to be assessed. 2 Heat transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is one-dimensional. 5 The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the jacket is $\tau > 0.2$ so that the one-term approximate solutions (or the transfer through the transfer μ | | Energy: \rightarrow k = $-\mu$ | 2 ∂y dy (L/ ∂y = V / L. 5-40 Chapter 5 Numerical Methods in Heat Conductivity and interior node in rectangular coordinates for T(x, y) for the case of variable thermal conductivity and uniform heat generation is to be obtained. 4 Heat transfer coefficients account for the radiation effects. The region in which the flow is both hydrodynamically (the velocity profile is fully developed and remains unchanged) and thermally (the dimensionless temperature profile remains unchanged) developed is called the fully developed region. ° C) $(2\pi r^3 m 2)$ 188.5r³ Then the rate of average heat transfer from the water can be expressed as Ti , ave - To Q& = Rtotal $\rightarrow 0.821$ W = $[7.5 - (-10)]^\circ$ C ($W \rightarrow r^3 = 350$. Analysis (a) Noting that the upper part of the iron is well insulated and thus the entire heat generated in the resistance wires is transferred to the base plate, the heat flux through the inner surface is determined to be q& 0 = Q& 0 800 W = 50,000 W / m 2 Abase $160 \times 10 - 4 m 2$ Taking the direction normal to the surface of the wall to be the x direction normal to the surface of the wall to be the x direction with x = 0 at the left surface. $=85^{\circ}C$ Q=800 W A=160 cm2 = 0 dT (0) = q& 0 = 50,000 W / m2 dx L=0.6 cm T (L) = T2 = 85^{\circ}C (b) Integrating the differential equation twice with respect to x yields x dT = C1 dx T (x) = C1x + C2 where C1 and C2 are arbitrary constants. This ture =? 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) stud and the value of the area is 0.82 for air space and 0.18 for stud section since the headers which constitute a small part of the wall are to be treated as studs. ° C)(0.001) m) = 0.000312 W/°C(kt) total = (kt) copper + (kt) epoxy = 0.0386 + 0.000312 = 0.038912 W/°C f epoxy = f copper = and k eff = (kt) epoxy(kt) total = Epoxy 0.000312 = 0.038912 W/°C = 29.9 W/m. That is, we are asked to determine the maximum heat transfer between the ambient air and the column. $\sqrt{Assumptions 1}$ The egg is spherical in shape with a radius of r0 = 2.75 cm. ° F)(0.0967 ft) (8.7 Btu / h.ft. The initial rate of heat transfer from a potato and the temperature gradient at the potato surface are to be determined. Applying the boundary conditions give T (r1)) = TI with and $-k C1 \ln r1 + C2 = TI r = r1$: -k r = r2: $\rightarrow C2 = TI - C1 \ln r1 C1 = h[(C1 \ln r2 + C2) - T\infty] r2 \rightarrow C1 = T\infty - TI r k \ln 2 + r1 hr2$ Substituting C1 and C2 into the general solution, the variation of temperature in plastic is determined to be T ∞ - TI r Tplastic (r) = C1 ln r + TI - C1 ln r1 = TI + ln r2 k plastic r1 ln + r1 hr2 We have already utilized the first interface condition by setting the wire and ceramic layer temperatures equal to TI at the interface r = r1. The total cost, which is the sum of the two, decreases first, reaches a minimum, and then increases. 2-75C No, it is not possible since the highest temperature in the plate will occur at its center, and heat cannot flow "uphill." 2-76C The cylinder will have a higher center temperature since the cylinder has less surface area to lose heat from per unit volume than the sphere. Properties The exit temperature of air, and thus the mean temperature of air, and thus temperature of air, and and the corner effects are negligible. 7-25 V x cr Chapter 7 External Forced Convection 7-34 The weight of a thin flat plate exposed to air flow on both sides is balanced by a counterweight. Now we divide the x-y plane of the region into a rectangular mesh of nodal points which are spaced Δx and Δy apart in the x and y directions, respectively, and consider a general interior node (m, n) whose coordinates are $x = m\Delta x$ and $y = n\Delta y$. 1 1 = = 0.82. ° C)(0.0216 m2) Rboard = Rconv Rtotal = Rconv Rtotal = Rconv Rtotal = Rconv Rtotal = 40° C + $(3.2 \text{ W})(0.93284 \degree \text{C}/\text{W}) = 43.0\degree \text{C}$ Rtotal T -T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0° C - $(3.2 \text{ W})(0.00694 \degree \text{C}/\text{W}) = 40.5 - 0.02 \cong 43.0\degree \text{C}$ Rtotal T -T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0° C - $(3.2 \text{ W})(0.00694 \degree \text{C}/\text{W}) = 40.5 - 0.02 \cong 43.0\degree \text{C}$
 Rtotal T -T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0° C - $(3.2 \text{ W})(0.00694 \degree \text{C}/\text{W}) = 40.5 - 0.02 \cong 43.0\degree \text{C}$ Rtotal T -T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0° C - $(3.2 \text{ W})(0.00694 \degree \text{C}/\text{W}) = 40.5 - 0.02 \cong 43.0\degree \text{C}$ Rtotal T - T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0° C - $(3.2 \text{ W})(0.00694 \degree \text{C}/\text{W}) = 40.5 - 0.02 \cong 43.0\degree \text{C}$ Rtotal T - T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0° C - $(3.2 \text{ W})(0.00694 \degree \text{C}/\text{W}) = 40.5 - 0.02 \cong 43.0\degree \text{C}$ Rtotal T - T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0° C - $(3.2 \text{ W})(0.00694 \degree \text{C}/\text{W}) = 40.5 - 0.02 \cong 43.0\degree \text{C}$ Rtotal T - T & Q& = 1.2 \rightarrow T2 = T1 - QR board = 43.0\degree \text{C}/\text{W} during a study. The rate of heat loss from the steam pipe and the annual cost of this energy lost are to be determined. 8-18C The hydrodynamic and thermal entry lengths are given as Lh = 0.05 Re Pr D for laminar flow, and $Lh \approx Lt \approx 10$ Re in turbulent flow. 3 Heat transfer coefficient is constant and uniform. Properties The Rvalues are given in Table 3-6 for different materials, and in Table 3-9 for air layers. Also, To = 50°F from 6 PM to 10 PM, 42°F from 10 PM to 2 AM, and 38°F from 2 AM to 6 AM. It is related to the rate of heat transfer by $Q\& = \int g\& dA$. Using a computer, the solution at the center node (node 5) is determined to be 217.2°C, 302.8°C, 379.3°C, 447.7°C, $508.9^{\circ}C$, $612.4^{\circ}C$, $695.1^{\circ}C$, and $761.2^{\circ}C$ at 10, 15, 20, 25, 30, 40, 50, and 60 min, respectively. The explicit transient finite difference formulation of the boundary nodes is to be determined. W& mech = $O\& = 2 \times 419 = 838 \text{ W} 6-12 \text{ Chapter } 6$ Fundamentals of Convection 6-40 The oil in a journal bearing is considered. Therefore, T3 = T2 and T4 = T1 hL = 0.037 Re L 0.8 Pr 1 / 3 = 0.037(6.855 × 10 5) 0.8 (0.7255)1 / 3 = 1551 Nu = k k 0.02662 W/m 2.°C L 0.7 m Q& = hA (T - T) = (58.97 W/m 2.°C)[(0.6 m)(0.7 m)](75 - 5)°C = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is then determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is the determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is the determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s surr [] = 1734 W conv s ∞ s The heat loss by radiation is the determined from Stefan-Boltzman law to be Q& = $\epsilon A \sigma (T 4 - T 4)$ rad s s s s radiation is the determined from Stefan-Boltzman law to be Q = 1734 W conv s ∞ s The heat loss by radiation is the determined from Stefan-Boltzman law to be Q = 1734 W conv s ∞ s The heat loss by r $(0.92)(0.6 \text{ m})(0.7 \text{ m})(5.67 \times 10 - 8 \text{ W/m } 2 \text{ .K } 4)$ (75 + 273 K) 4 - (10 + 273 K) 4 = 181 W Then the total rate of heat loss from the bottom surface of the engine block becomes Q& = Q& + Q& = 1734 + 181 = 1915 W total conv rad The gunk will introduce an additional resistance to heat dissipation from the engine. m)(0.02 \text{ m}) = 0.0597 \text{ m} 2 \text{ The heat} transfer rate from the flanges is Q& finned = η fin Q& fin,max = η fin Afin (Tb - T ∞) = 0.88(25 W/m 2.°C)(0.0597 m 2)(174.7 - 12)°C = 214 W (c) A 6-m long section of the steam pipe is losing heat at a rate of 7673 W or 7673/6 = 1279 W per m length. Analysis The total thermal resistance of the new heat exchanger is Rtotal, new HX T ∞ 1 T - T T $-T (350 - 250)^{\circ} F T \propto 2 Q_{\&} new = \propto 1 \propto 2 \rightarrow Rtotal, new = \propto 1 \propto 2 = 0.005 h.$ Analysis The surface area of each body is first determined from A1 = $\pi DL / 2 = \pi (0.25 \text{ m})(1.7 \text{ m})/2 = 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} = 2 \times 0.6675 \text{ m} 2 A2 = 2 \text{ A1} =$ = 0.09988 °C/W hA (15 W/m 2 .°C)(0.6675 m 2) Reather = Rconv Rconv T1 T ∞ 2 R total = Rleather + Rconv = 0.00942 + 0.09988 = 0.10930 °C/W The total sensible heat transferred through the clothes and the skin T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3
W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/W T -T (32 - 30)°C Q& clothes = 1 ∞ 2 = 18.3 W R total 0.10930 °C/ $skin = 1 \propto 2 = 20.0 \text{ W}$ Rconv 0.09988°C/W Q& sensible = Q& clothes + Q& skin = 18.3 + 20 = 38.3 \text{ W} Then the fraction of heat lost by respiration Decomes Q& respiration becomes Q& respiration for the second person's body 0.001 m L = 0.00576 °C/W kA (0.13) W/m.°C)(1.335 m 2) 1 1 = = 0.04994 °C/W 2 hA (15 W/m.°C)(1.335 m 2) Rsynthetic = Rconv R total = Reather + Rconv = 0.05570 °C/W R total 3-128 T1 Rsynthetic Rconv T $\infty 2$ Chapter 3 Steady Heat Conduction 3-168 A wall constructed of three layers is considered. 03) π Afin = 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = 1 - 27777 $\pi (0.0025) = 0.86$ m 2 Aunfinned = $(1 \text{ m}) = 1 \text{ m}^2 \text{ Q}_k$ no fin = hAno fin (Tb - T ∞) = (35 W / m²). Analysis The characteristic length of the steel plate and the Biot number are Oil bath 45°C V = L = 0.0025 m Lc = Steel plate As 10 m/min hL (860 W/m 2.°C)(0.0025 m) = 0.036 < 0.1 Bi = c = k 60.5 W/m^2. 1 atm and this temperature are (Table A-15) k = 0.02625 W/m.°C $v = 1.655 \times 10$ -5 m 2/s Person, Ts 90 W V $\infty = 5$ m/s T $\infty = 32$ °C $\varepsilon = 0.9$ Pr = 0.7268 Analysis The Reynolds number is VD (5 m/s)(0.3 m) Re = $\infty = = 9.063 \times 104 - 52 v 1.655 \times 10$ m/s D = 0.3 m The proper relation for Nusselt number corresponding to this Reynolds number is hD $0.62 \text{ Re } 0.5 \text{ Pr } 1/3 \text{ Nu} = 0.3 + 1/4 \text{ k} 1 + (0.4/\text{Pr}) 2/3 [] (\text{Re } 5/8] | 1 + | 1/4 | (282,000 /] 4/5 [0.62(9.063 \times 10.4 = 0.3 + 1 + 1/4 | (282,000 1 + (0.4/0.7268) 2/3 []) | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5/8] | 1/5$ generation in that person's body at a rate of 90 W and body gains heat by radiation from the surrounding surfaces, an energy balance can be written as Q& + Q& = Q& h = generated radiation convection Substituting values with proper units and then application of trial & error method yields the average temperature of the outer surface of the person 2 There is no heat generation within the bar. The derivative will be a constant when the f is a linear function of x. 3 The thermal contact resistance at the plate-soil interface is negligible. Applying the boundary conditions give Convection at x = L: $-kC1 = h(T2 - T\infty) \rightarrow Temperature$ at x = L: $T(L) = C1 \times L + C2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h(T2 - T\infty) / kC2 = T2 \rightarrow C1 = -h$ $T_2 - C_1 L$ Substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (x) = $C_1 x + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L)
= T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (L - x) c_1 + (T_2 - C_1 L) = T_2 - (T_$ $(0) = 75 - 25 \times (1/3 - 0) = 66.7^{\circ} F 2-31$ Chapter 2 Heat Conduction Equation 2-68 A compressed air pipe is subjected to uniform heat flux on the outer surface and convection on the inner surface. Analysis The heat transfer through the bottom of the pan by conditions, the rate of heat transfer through the bottom of the pan by conditions, the rate of heat transfer through the bottom of the pan by conditions. conduction is $\Delta T T - T Q \& = kA = kA 2 1 L L$ Substituting, 105° T - 105°C 800W = (237W/m · °C)(0.0314m 2) 2 0.004m which gives 800 0.4 T2 = 105.43 °C 1-66E The inner and outer surface temperatures of the wall of an electrically heated home during a winter night are measured. The mathematical formulation, the variation of temperature in the plate, and the bottom surface temperature are to be determined for steady one-dimensional heat transfer. Assumptions 1 Heat transfer through the bar is given to be steady and two-dimensional. 4 The top surface at x = 0 is subjected to uniform heat flux. 4-44 to be $t \cong 2.3$ hours If the air temperature were -80°F, the freezing Air -40°C time would be Chicken t a temperature is decreased. For a given deep body temperature, the outer skin temperature is decreased. For a given deep body temperature is decreased. For a given deep body temperature is decreased. heater and the temperature rise of air in the duct are to be determined.. A similarity solution is likely to exist for a set of partial differential equations if there is a function that remains unchanged (such as the nondimensional velocity profile on a flat plate). The new U-value of the wall and the rate of heat transfer through the wall is to be determined. $Q\&= 1 \otimes 2 \rightarrow T1 = T \otimes 2 + QR \circ C / W = 40.5 \circ C$ total = 40° C + (3.2 W)(01465 Rtotal T - T & Q\&= 1 2 \rightarrow T2 = T1 - QR board = 40.5° C - (3.2 W)(0.00694 \circ C / W) = 40.5 \circ C - (3.2 W)(0.00694 \circ C / W) = 40.5 \circ C Robard 3-85 Chapter 3 Steady Heat Conduction 3-116 A hot plate is to be cooled by attaching aluminum pin fins on one side. 3 Radiation heat exchange with the surroundings is negligible. 2 The thermal properties of the plate are constant. 2-12C The phrase "thermal energy generation," and they are used interchangeably. Then, b = hAs 35 W/m 2 .°C h = = $0.0136 \text{ s} - 1 \rho C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow = e – (0.0136 s) -1 $P C p V \rho C p Lc$ (8950 kg/m 3)(383 J/kg.°C)(0.00075 m) -1 T (t) – T ∞ 50 – 30 = e – bt \rightarrow s)t \rightarrow t = 204 s 350 - 30 Ti - T ∞ (b) The wire travels a distance of velocity = length (10 m/min) \rightarrow length = | (204 s) = 34 m time (60 s/min / This distance can be reduced by cooling the wire in a water or oil bath. 3 Radiation heat transfer is negligible. m2) L 0.0015 m Rair = R2 = R4 = R6 = R8 = = 0.0524 ° C / W kA (0.026 W / m. The temperature of such bodies can be taken to be a function of time only. 1-1 Chapter 1 Basics of Heat Transfer Heat and Other Forms of Energy 1-8C The rate of heat transfer per unit surface area is called heat flux q&. The Reynolds number is Steel pipe V ∞ Do (4 m/s)(0.116 m) Di = D1 = 4 cm 4 Re = = 3.254 \times 10 D2 = 4.6 cm v 1.426 \times 10 -5 m 2.426 \times 10 -5 m 2.426 \times 10 -5 m 2.426 \times 10 D2 /s Insulation $\varepsilon = 0.3$ The Nusselt number for flow across a cylinder is determined from hDo 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) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 = 0.3 + 1 + 1/4 || (282,000) |] Di [0.62(3.254 × 10 4 $107.0 \text{ k} 0.02439 \text{ W/m} \cdot ^{\circ}\text{C}$ ho = Nu = (107.0) = 22.50 W/m 2 \cdot ^{\circ}\text{C} Do 0.116 m Area of the pipe per m length of the pipe and the insulation to the outer surface (by first convection and then transfer from the steam through the pipe and the insulation to the outer surface (by first convection and then transfer from the steam) and the insulation to the outer surface (by first convection and then transfer from the steam) and the insulation to the outer surface (by first convection and then transfer from the steam) and the insulation to the outer surface (by first convection and then transfer from the steam) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation to the outer surface (by first convection) and the insulation (by first convectin) and the insulation (by first convection) and the insulatio conduction) must be equal to the heat transfer from the outer surface to the surroundings (by simultaneous convection and radiation). The average temperature of the person is to be determined for two cases. But there is no significant change in turbulence level after the first few rows, and thus the heat transfer coefficient remains constant. The variation of temperature in the cylinder is given by T (r) = 2 g&r02 [(r)]1 - [] + Ts k [] (r0] / [] 80°C (a) Heat conduction 3-159 The plumbing system of a house involves some section of a plastic pipe exposed to the ambient air. The mathematical formulation, the variation of temperature in the wall, and the left surface temperature are to be determined for steady one-dimensional heat transfer. 7-64 Chapter 7 External Forced Convection 7-71 Air is cooled by an evaporating refrigerator. °C h 0.00165 m 4-37 Chapter 4 Transient Heat Conduction 4-48 Using the data and the answers given in Prob. The time it will take for the temperature of the outer surface of the furnace to change is to be determined. m L = = 0.4341 °C / W kA (176 W / m. Analysis The nodal spacing is given to be $\Delta x = 0.2$ cm and $\Delta y = 1$ cm. Then the pressure drop across the tube bank becomes $\Delta P = N L f_2 \rho Vmax (1.316 \text{ kg/m 3})(8.571
\text{ m/s}) 2 = 30(0.44)(1) 2 2 (1N | 1 \text{ kg} \cdot \text{m/s 2}) = -7.8^{\circ}C$, which is fairly close to the assumed value of -5°C. m) 1 1 Rconv = = = 1481 °C / W. Heat transfer through such insulations is by conduction through the solid material, and conduction or convection through the air space as well as radiation. Therefore, b = time = hAs 860 W/m 2 .°C h = = 0.10092 s -1 ρ C pV ρ C p Lc (7854 kg/m 3)(434 J/kg.°C)(0.0025 m) length 5m = 0.5 min = 30 s velocity 10 m/min Then the temperature of the sheet metal when it leaves the oil bath is determined to be -1 T (t) - T ∞ T (t) - 45 = e - bt \rightarrow = e - (0.10092 s)(30 s) \rightarrow T (t) = 82.53°C Ti - T ∞ 820 - 45 The mass flow rate of the sheet metal to the oil bath is m& = $\rho V \&$ = $\rho w V = (7854 \text{ kg/m 3})(2 \text{ m})(0.005 \text{ m})(10 \text{ m/min}) = 785.4 \text{ kg/min}$ Then the rate of heat transfer from the sheet metal to the oil bath is m& = $\rho V \&$ = $\rho w V = (7854 \text{ kg/m 3})(2 \text{ m})(0.005 \text{ m})(10 \text{ m/min}) = 785.4 \text{ kg/min}$ Then the rate of heat transfer from the sheet metal to the oil bath is m& = $\rho V \&$ = $\rho w V = (7854 \text{ kg/m 3})(2 \text{ m})(0.005 \text{ m})(10 \text{ m/min}) = 785.4 \text{ kg/min}$ Then the rate of heat transfer from the sheet metal to the oil bath is m& = $\rho V \&$ = $\rho w V \&$ bath and thus the rate at which heat needs to be removed from the oil in order to keep its temperature constant at 45°C becomes $Q_k = m_k C [T (t) - T] = (785.4 \text{ kg/min})(434 \text{ J/kg.°C})(82.53 - 45)^\circ C = 1.279 \times 107 \text{ J/min} = 213.2 \text{ kW } p \propto 4-101 \text{ Chapter 4 Transient Heat Conduction 4-110E A stuffed turkey is cooked in an oven. m2 } Q_k no fin = hAno$ fin (Tb - T ∞) = (40 W / m2 . 5-8 Chapter 5 Numerical Methods in Heat Conduction 5-23 A pin fin with negligible heat transfer from its tip is considered. 5 Heat transfer from the base of the is considered. 5 Heat transfer from the base of the is considered. 5 Heat transfer from the base of the is considered. 5 Heat transfer from the base of the is considered. 5 Heat transfer from the base of the is considered. 5 Heat transfer from the base of the is considered. 5 Heat transfer from the base of the is considered. 5 Heat transfer from the base of 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 Rotal T – T & Q& = 1 2 → T2 = T1 – QR board = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W)(0.011 ° C / W) = 59.4° C + (15 W) 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) = 0.975 aL 13.78 m -1 × 0.02 m) tanh aL tanh(13.78 m -1 × 0.02 m) = 0.975 aL 13.78 m -1 × 0.02 m) tanh aL tanh(13.78 m -1 × 0.02 m) tanh aL tanh(13.78 m -1 × 0.02 m) = 0.975 aL 13.78 m -1 × 0.02 m) tanh aL tanh(13.78 m -1 × 0.02 m) Then, Raluminum Repoxy Q& finned = η fin Q& fin,max = η fin hAfin (Tbase - T ∞) (η fin Afin + Aunfinned) Substituting, the base temperature of the finned surfaces is determined to be Q& total Tbase = T ∞ + $h(\eta$ fin Afin + Aunfinned) = $37^{\circ}C + 15 W (45 W/m \cdot ^{\circ}C)[(0.975)(0.126 m 2) + (0.0090 m 2)] 2 = 39.5^{\circ}C$ 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 \circ 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 $\varepsilon = 0.6$. 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 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T ∞] + $\varepsilon\sigma$ [(T (L) 4 - K = h[T (L) - T (K = h[T (L) - T $75^{\circ}F \epsilon T_{\infty} h L T (x) = C1x + C2$ where C1 and C2 are arbitrary constants. 4 Heat transfer from the base of the transistor is negligible. $\circ C$) The constants $\lambda 1 = 01039$. 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 Relate = 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 resistance is zero. 6 5 7 Construction 1a. It yields dT (0) $g\& 0 \times = - \times 0 + C1 \rightarrow C1 = 0$ B.C. at r = 0; 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 $0 = \Delta U$ cars + ΔKE cars $0 = (mC\Delta T)$ 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 $1 - 2\tau - 2\tau h \Delta x$ hin $\Delta x < 1 - 2\tau - 2\tau$ out k k Therefore, the stability criteria for this problem can be expressed as $1 - 2\tau - 2\tau$ hin $\Delta x \ge 0 \rightarrow k \tau \le 12(1 + hin \Delta x / k) \rightarrow \Delta t \le \Delta x \ge 2\alpha (1 + hin \Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$. 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 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 $\partial P / \partial x = 0$ (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 ($\partial u \, d \, 2u \, \partial P \rightarrow 2 = 0 + v$ | $= \mu 2$ - ρ|| u dy / dx dy dy \ dx 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 of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the plate surfaces must be equal to the velocities of the velocities of the plate surfa $= 0.69 \times 10 - 6 \text{ m}2 / \text{s}$. Analysis The time required to freeze the turkeys Brine from 1°C to -18° C with brine at -29° C determined directly from Fig. Then the log mean temperature difference, and the expression for the rate of heat transfer become $\Delta T \ln = (Ts - Ti) - (Ts - Te) (90 - 15) - (90 - 65) = 45.51^{\circ}C \ln[(Ts - Ti) / (Ts - Te)] \ln[(90 - 15) / (90 - 65)] Q_{\&} = hAs \Delta T \ln = (16,994 \text{ W/m } 2 \cdot ^{\circ}C)(0.1257 \text{ N L})(45.51^{\circ}C) = 97,220 \text{ N L}$ The mass flow rate of water through a cross-section corresponding to NT = 1 and the rate of heat transfer are m& = $\rho AcV = (999.1 \text{ kg/m } 3)(4 \times 10^{\circ} \text{ C})(0.1257 \text{ N L})(45.51^{\circ}C) = 97,220 \text{ N L}$ The mass flow rate of water through a cross-section corresponding to NT = 1 and the rate of heat transfer are m& = $\rho AcV = (999.1 \text{ kg/m } 3)(4 \times 10^{\circ} \text{ C})(4 \times 10^{\circ$ 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 $\Delta T \ln \rightarrow 2.004 \times 10 7 \text{ W} = 97,220 \text{ N L} \rightarrow \text{N L} = 206 7-66 \text{ Chapter 7 External}$ 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 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 "[m/m^2-C]" "PROPERTIES" k=110 "[W/m-C]" alpha=33.9E-6 "[m²/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)/(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] 500 450 400 350 300 500 550 600 650 700 T 🛥 4-24 [C] 750 800 850 900 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] 500 450 400 350 300 500 550 600 650 700 T 🛥 4-24 [C] 750 800 850 900 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] 500 450 400 350 300 500 550 600 650 700 T 🛥 4-24 [C] 750 800 850 900 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] 500 450 400 350 300 500 550 600 650 700 T 🛥 4-24 [C] 750 800 850 900 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] 500 450 400 350 300 500 550 600 650 700 T 🛥 4-24 [C] 750 800 850 900 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] 500 450 400 350 300 500 550 600 650 700 T 🔤 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 through the roof by conduction. ° C)(140 – 22)° C $\Delta t = = 51.8 \text{ s} (850 - 21.2) \text{ J} / \text{s} \text{ Q} = -21.2) \text{ J} / \text{s} \text{ Q} = -21.2 \text{ g} / \text{s} / \text{g} / \text{s} = -21.2 \text{ g} / \text{g} /$ days, the external convection heat transfer coefficient is greater compared to calm days. Then 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 (0.021 m) = max = 3132 \mu $2.76 \times 10 - 5$ 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 (0.6946) 0.36 D (0.6946) 0.25 = 37.46 Since NL =16, the average Nusselt number and heat transfer coefficient for all the tubes in the tubes in the tube bank become Nu D, N L = Nu D = 37.46 h = Nu D, $N L k D = 37.46(0.04104 W/m \cdot °C) = 73.2 W/m 2 \cdot °C 0.021 m$ The total number of tubes is $N = NL \times NT = 16 \times 8 = 128$. 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 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. 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. $g = 7 \times 107 \text{ W/m3}$ Analysis The total rate of heat generation in the rod is D = 5 cm determined by multiplying the rate of heat generation of the absorption of solar energy in a solar pond with depth is given. m2 Aunfinned = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = Afinned + Aunfinned Rconv = 1 hAtotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4$ Atotal, with fins = $0.0216 - 864 \, \pi D \, 2 \, 4 \, 3 \, 2 \, Total$ unit thermal resistance of each section (the R-value), m2.°C/W in summer and R = 0.795 m2.°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 are Ri R3 R2 R1 R4 Too 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) $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx dT$ (L) 4 $-k = h1[T \propto 1 - T (0)] dx$ 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 through the bottom surface of the refrigerator is negligible. The midplane in the latter case will behave like an insulated surface of the refrigerator is negligible. The midplane in the latter case will behave like an insulated surface of the refrigerator is negligible. 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. During a differential amount dT. 2 Heat transfer is one-m(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 = $\rho V = \rho \pi$ Then average rate of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 3 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 3 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg.°C)(80 - T2)°C (Eq. 2) Q& = $\Delta t 45 \times 60$ s Setting Eq. 3 we obtain the final temperature of heat transfer is Q (181.27 kg)(4197 J/kg 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 () \rightarrow T2 = 69.9°C 7-51 Chapter 7 External Forced Convection 7-59 "Im]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "GIVEN" D=0.50 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "GIVEN" D=0.50 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "GIVEN" D=0.50 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "GIVEN" D=0.50 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "GIVEN" D=0.50 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "GIVEN" D=0.50 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" Vel=40 "[km/h]" "time=45 Chapter 7 External Forced Convection 7-59 "[m]" T_w1=80 "[C]" T infinity=18 "[C]" T infi [min], parameter to be varied" "PROPERTIES" Fluid\$='air' k=Conductivity(Fluid\$, T=T film) rho=Density(Fluid\$, T=T film) rho=D 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*2/4 O dot=h*A*(T w ave-T infinity) m w=rho w*V w V w=pi*D^2/4*L O=m w*C p w*(T w1-T w2) O dot=O/(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) = $(0.90)(5.67 \times 10 - 8 \text{ W/m } 2 \text{ . K } 4)(1.7 \text{ m } 2)[(32 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - (23 + 273) 4 - ($ 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 $2\pi L 2\pi (10 \text{ m}) = 0.8 < 1.41 \rightarrow S = = 358.7 \text{ m} b 20 (a) 0.785 \ln 0.8 0$ 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 amount of heat transfer and heat transf 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. 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 constant agitation of the engine block. Properties The thermal conductivity is given to be $k = 30 \text{ W/m} \cdot \text{C}$. Then the the roof to be Ts, in and Ts, out, respectively, the quantities above can be expressed as $4 Q\& = h A (T - T) + \varepsilon A \sigma (T - T 4) = (5 W/m 2.°C)(300 m 2)(20 - T)°C$ room to roof, conv + rad is room Q&s, in s s, in room + (0.9)(300 m 2)(20 - T)°C room to roof, conv + rad is room 2 (V/m 2.°C)(300 m 2)(20 - T)°C $(300 \text{ m 2}) Q\& \text{roof}, \text{cond} = \text{kAs L } 0.15 \text{ m } Q\& \text{roof} \text{ to surr}, \text{conv} + \text{rad} = \text{ho As } (\text{Ts ,out} - \text{Tsurr}) + \varepsilon \text{As } \sigma (\text{Ts ,out} - \text{Tsurr}) + \varepsilon \text{As } \sigma (\text{Ts ,out} + 273 \text{ K}) 4 - (100 \text{ K}) 4$ Solving the equations above simultaneously gives $Q\& = 28,025 \text{ W} = 28,025 \text{ W} = 28.03 \text{ kW}, \text{T} = 10.6^{\circ}\text{C}, \text{ and Ts ,in}$ 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 therms) Q gas = total = = || = 15.75 therms)(\$0.60 / therms) = \$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 / n oven = (2.001 × 109 k] / yr = 24,314 therms / yr since 1 therm = 105,500 k]. The o insulation and the amount of money saved per year are to be determined. It yields dT (0) $g\& 0 \times = - \times 0 + C1 \rightarrow C1 = 0$ B.C. at r = 0; 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)*(0.7-0.03) + (0.8-0.03)*(0.7-0.03) + (0.8-0.03)*(0.7-0.03) + (0.8-0.03)*(0.7-0.03) + (0.8-0.03)*(0.7-0.03) + (0.8-0.03)*(0.7-0.03) + (0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.7-0.03) + (0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03)*(0.8-0.03) 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 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 24.81 24.85 2 24.24 24.25 24.25 24.25 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 23.66 23.67 23.68 23.69 23.67 23.68 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 23.12 23.12 23.12 23.13 23.14 23.15 23.15 23.16 T5 [C] 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) $h = 14.8V \ 0.53 = 14.8(0.5 \text{ m/s}) \ 0.69 = 9.174 \text{ W/m } 2$. °C $Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V
\propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{C } Air V \propto T \propto = 10^{\circ} \text{$ = $(14.8 \text{ W/m } 2.^{\circ}C)(1.7 \text{ m } 2)(29 - 10)^{\circ}C = 478.0 \text{ W}(c)$ h = $14.8 \text{ V} 0.53 = 14.8(1.5 \text{ m/s}) 0.69 = 19.58 \text{ W/m } 2.^{\circ}C)(1.7 \text{ m } 2)(29 - 10)^{\circ}C = 632.4 \text{ W} 6-11$ 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) + 2 π (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 $\infty 1 \infty 2$ Propane The volume of the tank and the mass of the propane are V = $\pi r 2 L = \pi (0.6 m) 2 (6 m) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = \rho V = (581 \text{ kg/m 3})(6.786 m 3) = 6.786 m 3 m = 0.786 m 3 m = 0.786 m 3$ 3942.6 kg The rate of vaporization of propane is Q& 44.787 kJ/s Q& = m& h fg \rightarrow 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 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_{\infty} - 1 - (-12) = 0.65 5 - (-12) To - T_{\infty}] | | | 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 $\Delta x = 0.02$ 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 convention heat transfer coefficient, and to ignore radiation in the convention 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 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr | k [g&r 2 dT (r0) = -k 0 dr"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 = $\varepsilon \sigma A sT 24 - \alpha s A s q \& solar L Canceling the area A and substituting the known quantities, (520 R) – T2 (1.2 Btu/h$ $ft \cdot ^{\circ}F = 0.8(0.1714 \times 10 - 8 \text{ Btu/h} \cdot \text{ft } 2 \cdot \text{R } 4)T24 - 0.45(300 \text{ Btu/h} \cdot \text{ft } 2) 0.5 \text{ ft } T2 = 530.9 \text{ R } \text{Solving for } T2 \text{ gives the outer surface temperature to be } \text{K} = 54$ $W/m.^{\circ}C$, $\rho = 7833 \text{ kg/m3}$, and $Cp = 0.465 \text{ kJ/kg.}^{\circ}C$. Properties The thermal conductivities of the leather and synthetic fabric are given to be $k = 0.13 \text{ W/m} \cdot ^{\circ}C$, 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. $\times 108 \text{ kJ/yr} / 0.78 = 2.197 \times 108 \text{ kJ/yr} = 2082$ therms Annual Cost = Qin,ins \times Unit cost = (2082 therm / yr)(\$0.50 / therm) = \$1041 / yr Cost savings = Energy cost w/o 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. \times 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 Δ E 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 = $k\pi D 2 / 4 4h = kD 4(35 \text{ W} / m^2)$. 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 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[1], column 3, the value of T[1], column 1 contains the time, column 2 the value of T[1], column 3, the value of T[1], column 1 contains the time, column 2 the value of T[1], column 3, the value of T[1], column 3, the value of T[1], column 4, the value of the value of T[1], column 4, the 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[2]T old[1])+k*(T old[2]T old $(T \ old[3]T \ old[4])/DELTAx=RhoC*DELTAx/2*(T[4]-T \ old[4])/DELTAt "Node 4, convection" 5-100 Chapter 5 Numerical Methods in Heat Conduction h i*A i*(T \ old[5])/DELTAt "Node 5, refrig. Applying the boundary conditions give - g&r1 2 + C1 ln r1 + C 2 4k r = r1: T (r1)$ $= r = r2: T(r2) = -gkr2 2 + C1 \ln r2 + C2 4k$ Substituting the given values, these equations can be written as 60 = -(37,894)(0.15) + C2 4(20) Solving for C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = $-37,894r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 227.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 - 473.68r 2 + 98.34 \ln r + 257.2 = 257.2 + 278.28 \ln r + + 278.28 \ln r$ 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 $4\pi D 2 / 4 = =D$. 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 = $30.19 \text{ m} 2 1 1 \text{ Ro} = = 0.0489 \text{ °C/W} 4 \pi \text{kr1} \text{ r} 2 4 \pi (0.035 \text{ W/m}.^{\circ}\text{C})$ (1.55 m)(1.5 m)(1.5 m) Rtotal = Ro + Rinsulation = 0.000946 + 0.0489 = 0.0498 °C/W T - T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - 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(-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = 3976$ W Rtotal 0.0498 °C/W T -T [15 - (-183)]°C Q& = s1 $\infty 2 = 3976$ W Rtotal 0.0498 °C/W T $29.03 \text{ m } 211 \text{ Ro} = = 0.000984 ^{\circ}C/W 2 \text{ ho } A (35 \text{ W/m} \cdot ^{\circ}C)(29.03 \text{ m } 2) \text{ Risulation} = Ts1 \text{ Risulation} = Ts1 \text{ Risulation} = 0.000984 + 13.96 ^{\circ}C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma } C/W 3-68 \text{ Ro } T \times 2 \text{ Gamma }$ 13.96 ° C / W Rtotal 0.01418 kJ / s Q& & fg Q& = mh \rightarrow 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 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, $k = 0.607 \text{ W/m.}^{\circ}C$ and $\alpha = k / \rho C p = 0.146 \times 10^{-6} \text{ m}^{-2}$ (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 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 = Σ farea, iUi = 0.82×1.085+0.18×0.805 1.035 W/m2.°C Overall unit thermal resistance, R = 1/U 0.967 m2.°C/W (c) Two-reflective surface, $\varepsilon 1 = \varepsilon 2 = 0.05 \rightarrow \varepsilon$ effective = 1 1 = 0.03 1 / $\varepsilon 1 + 1$ / $\varepsilon 2 - 1$ 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 $-Wb = \Delta U$ We in $= \Delta H = m(h2 - h1) \approx mC p$, ave $(T2 - T1) 4 \times 5 \times 6 = 120 \text{ m3 PV}$ (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 $W_{k} = (149.3 \text{ kg})(1.007 \text{ kJ/kg} \circ \text{C})(25 - 7) \circ \text{C}/(15 \times 60 \text{ s}) = 3.01 \text{ kW}$ 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 conditions are reached. 4 - 15b) Tsurface = T ∞ + 0.45(To - T ∞) = 25 + 0.45(40 - 25) = 31.8° 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 hat (20 W/m.°C) (1.6 × 10 m / s)(2 × 3600 s) = 2.98 = | T - T ∞ 0.72 W/m.°C k = 0.038 \rightarrow 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. $^{\circ}$ F Air V $^{\circ}$ = 60 mph T $^{\circ}$ = 50° F Minivan v = 0.1697×10 ft /s -32 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 the mineral deposit. ° C)(2 π r3 m 2) 188.5r3 Then the rate of average heat transfer from the water can be expressed as Ti ,ave – To Q& = Rtotal \rightarrow 1.694 W = [7.5 – (-10)]° $C [0.0948 + 4.55 \ln(r_3 / 0.033) + 1 / (188.5r_3)]^\circ C / W \rightarrow r_3 = 0.312 m$ Therefore, the minimum thickness of fiberglass needed to protect the pipe from freezing is t = r_3 - r_2 = 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 $\infty = 0.6$ (Fig. Nodes 1, 2, 2) Bi hr0 T (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T $\infty = 0.6$ (Fig. Nodes 1, 2) Bi hr0 H (r) - T (r

3, and 4 are interior nodes, and thus for them we can use the general finite difference relation 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 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subjected to convection is obtained by applying an energy balance on the half volume element about node 5 on the right surface subject about nod 5 and taking the direction of all heat transfers to be towards the node under consideration: Node 0 (Left surface - insulated T1 - 2T2 + T3 g& Δx + = Node 2 (interior) : 0 k Δx 2 • • • • 0 1 2 T2 - 2T3 + T4 g& 3 + = Node 3 (interior) : 0 k Δx 2 T3 - 2T4 + T5 g& + = 0 Node 4 (interior): $k \Delta x 2 T - T5$ Node 5 (right surface - convection): $h(T \infty - T5) + k 4 + g\&(\Delta x/2) = 0 \Delta x$ Node 1 (interior): $2 + \bullet \bullet 45 h$, $T \infty$ where $\Delta x = 0.01 m$, $g\& = 6 \times 105 W/m 3$, $k = 28 W/m \cdot °C$, $h = 60 W/m 2 \cdot °C$, and $T \infty = 30°C$. Properties The R-values of different materials are given in Table 3-6. 3 Radiation heat transfer is significant. energies $Qin = \Delta U \rightarrow Qin = m(u^2 - u^1) \cong mCv (T^2 - T^1)$ The final temperature of air is PV PV 1 = 2 \rightarrow T1 T2 T2 = P2 T1 = 2 \times (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 - R)(1080 - R)(108 540) R = 1890 Btu Air 20 lbm 50 psia 80°F Q 1-8 Chapter 1 Basics of Heat Transfer 1-28 The hydrogen gas in a rigid tank is cooled until its temperature drops to 300 K. The combined convection and radiation heat transfer coefficient at the outer surface of the pipe is to be determined. The time for the ignition of the wood is to be determined. When treating hot dog as an infinitely long cylinder, heat conduction is onedimensional in the radial r- direction. Assumptions 1 Heat transfer from the wire is steady since there is no indication of any change with time. 1-18 The body temperature of a man rises from 37°C to 39°C during strenuous exercise. ° C)(96 - 18)°C + (0.8 kg)(0.6 kJ / kg. The material properties needed to solve the problem are listed. Analysis This hot dog can physically be formed by the intersection of an infinite plane wall of thickness 2L = 12 cm, and a long cylinder of radius ro = D/2 = 1 cm. × 108 W / m3 (0.001 m) (r0 - r02) + $0 = T\infty + o = 30^{\circ} \text{ C} + = 409^{\circ} \text{ C} 4k 2h 2h 2(140 \text{ W} / m2)$. Assumptions 1 The temperature in the soil is affected by the thermal conditions at one surface only, and thus the soil can be considered to be a semi-infinite medium with a specified surface temperature of 10°C. 4-99 Chapter 4 Transient Heat Conduction 4-108 A hot dog is to be cooked by dropping it into boiling water. 3 There is no heat generation in the plate. 3 The emissivity of the person is constant and uniform over the exposed surface. Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas which is in motion, and it involves combined effects of conduction and fluid motion. The rate of heat loss from the house through its walls and the roof is to be determined, and the error involved in ignoring the edge and corner effects is to be assessed. The Biot numbers and the corresponding constants are first determined to be Bi wall = hL (9 W/m 2 .°C)(0.90 m) = = 13.06 k (0.62 W/m.°C) D0 = 28 cm $\rightarrow \lambda 1 = 14495$. Air space, 90-mm, nonreflective 4. 1-54 Chapter 1 Basics of Heat Transfer 1-102 The convection heat transfer coefficient for heat transfer from an electrically heated wire to air is to be determined by measuring temperatures when steady operating conditions are reached and the electric power consumed. - (914 . In steady heat transfer, heat transfer, heat transfer, heat transfer rate to the wall are equal. 3 Heat loss from the fin tip is given to be negligible. 6-43. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop across the section F are to be determined. The emissivity of the outer surface area of the tank is 0.6. Analysis (a) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of the tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area of tank is 0.6. Analysis (b) The outer surface area o radiation become Q& conv = hAs (T ∞ - Ts) = (30 W/m 2 .K 4) [(288 K) 4 - (273 K) 4] = 1292 W Q& = Q& + Q& = 21,488 W 4 Q& rad = ϵ As σ (Tsurr - Ts4) = (0.6) ((28.65 m 2)(5.67 × 10 - 8 W/m 2 .K 4)](28 K) 4 - (273 K) 4] = 1292 W Q& = Q& + Q& = 22,780 W total conv rad 0°C (b) The amount of heat transfer during a 24-hour period is Air Q = Q& \Delta t = 0.6 $(22.78 \text{ kJ/s})(24 \times 3600 \text{ s}) = 1,968,000 \text{ kJ} = 5898 \text{ kg} 333.7 \text{ kJ/kg}$ hif Q& Iced water 0°C 1 cm Discussion The amount of ice that melts can be reduced to a small fraction by insulating the tank. Table 3-4 reveals that HS6071 in The thermal resistance of the heat sink must be below 175 vertical position, HS5030 and HS6115 in both horizontal and vertical position can be selected. The mathematical formulation, the variation of temperature in the pipe, and the outer surface temperature, and the maximum rate of hot water supply are to be determined for steady onedimensional heat transfer. pipe length becomes T – T (320 – 5)° C Q& = $\infty 1 \propto 2$ = = 93.9 W Rtotal 3.355 ° C / W The temperature drops across the pipe and the insulation = QR insulation = QR insulation = (93.9 W)(3.089 ° C / W) = 290° C 3-45 Ro T $\propto 2$ Chapter 3 Steady Heat Conduction 3-69 "GIVEN" $T_infinity_1=320 "[C]" T_infinity_2=5 "[C]" k_steel=15 "[W/m-C]" b_o=0.055 "[m]" c_0=0.055 "$ R pipe=ln(r 2/r 1)/(2*pi*k steel*L) R ins=ln(r 3/r 2)/(2*pi*k ins*L) r 3=r 2+t ins*Convert(cm, m) "t ins is in cm" R conv o Q dot=(T infinity 1-T infinity 2)/R total DELTAT pipe=Q dot*R pipe DELTAT ins=Q dot*R ins Tins [cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 69.13 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 93.91 78.78 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 121.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6
7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 Q [W] 189.5 [Cm] 1 2 3 4 5 6 7 8 9 10 62.38 57.37 53.49 50.37 47.81 ΔTins [C] 246.1 278.1 290.1 296.3 300 302.4 304.1 305.4 306.4 307.2 3-46 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] 246.1 278.1 290.1 296.3 300 302.4 304.1 305.4 306.4 307.2 3-46 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of a steam pipe passes of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section of 40 1 2 3 4 5 6 t ins [cm] 3-47 7 8 9 240 10 Δ T ins [C] Q [W] 140 Chapter 3 Steady Heat Conduction 3-70 A 50-m long section 3-70 A 50-m l through an open space at 15°C. Real bodies emit and absorb less radiation than a blackbody at the same temperature. 2-111C Integration is the inverse of the u-factor of the wall, R = 1/U. The temperatures of the inverse of the inverse of the inverse of the u-factor of the wall, R = 1/U. The temperature of the u-factor of the u-facto transfer through the roof during that night are to be determined. 1-49C Convection involves fluid motion, conduction does not. 6-1 Chapter 6 Fundamentals of Convection 6-8 Heat transfer coefficients at different air velocities are given during air cooling of potatoes. energies & 1 = Q& out E0 + mh & 2 (since $\Delta ke \cong \Delta pe \cong 0$) W&e,in + W&fan,in E0 + mh & 2mh & & p (T2 - T1) We, in = mC Thus, m& = W& e, in C p (T2 - T1) Then, v1 = $1.2kJ/s = 0.04767 kg/s (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (100kPa) P1 () V&1 = m& v1 = (0.04767 kg/s (0.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg \cdot K (295K) = 0.04767 kg/s (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg \cdot K (295K) = 0.04767 kg/s (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg \cdot K (295K) = 0.04767 kg/s (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg \cdot K (295K) = 0.04767 kg/s (1.007 kJ/kg \cdot ^{\circ}C)(47 - 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 W) RT1 0.287 kPa \cdot m 3/kg \cdot K (295K) = 0.04767 kg/s (1.007 kJ/kg \cdot ^{\circ}C)(47 + 22)^{\circ}C (P1 = 100 kPa T1 = 22^{\circ}C T2 = 47^{\circ}C A2 = 60 cm2 We = 1200 kPa T1 = 120 kPa T1 = 120 kPa T1 = 120 kPa T1 = 120 kPa T1 = 1$ velocity of air is determined from the conservation of mass equation, $v^2 = m \& = 3 RT^2 (0.287 kPa \cdot m/kg \cdot K)(320 K) = = 0.9184 m 3/kg (100 kPa) P2 3 m \& v 2 (0.04767 kg/s)(0.9187 m/kg) 1 \rightarrow V^2 = = 7.30 m/s A2 V2 v^2 A2 60 \times 10 - 4 m 2 1-19 Chapter 1 Basics of Heat Transfer 1-41 The ducts of an air heating system pass through an$ unheated area, resulting in a temperature drop of the air in the duct. 4-30C This case can be handled by setting the heat transfer coefficient h to infinity ∞ since the temperature of the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in this case becomes equivalent to the surrounding medium in the surrounding medium in the surrounding medium in this case becomes equivalent to the surrounding medium in resistance on the backside of the board are T1 T ∞ L 0.002 m Rboard = = = 0.011 ° C / W T2 kA (12 W / m. Heat transfer coefficient that will enable to meet temperature constraints of the chickens while keeping the refrigeration time to a minimum is to be determined. Properties The case-to-ambient thermal resistance is given to be 20 ° C / W. 2 There is no heat generation in the body. (b) If the top and the bottom surfaces of the rod, $A = \pi DL$. (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 = 4πr2 is variable. 1-23 Chapter 1 Basics of Heat Transfer 1-51C In forced to move by external means such as a fan, pump, or the wind. 3 All the heat generated in the chips is conducted across the circuit board. 2, and the analytical (exact) solution can be used to check the accuracy of the numerical solution above. Analysis (a) If the tank is not insulated, the heat transfer rate is determined to be Atank = $\pi DL + 2\pi (\pi D 2 / 4) = \pi (12 . 3-9C$ The thermal resistance network associated with a five-layer composite wall involves five single-layer resistances connected in series. 4 Air is an ideal gas with constant properties. The global discretization error usually increases with increasing number of steps, but the opposite may occur when the solution function changes direction frequently, giving rise to local discretization errors of opposite signs which tend to cancel each other. $(0.125 \text{ m}) = 0.04657 \text{ kg} \text{ (Max} = \text{mC p} (\text{Ti} - \text{T} \infty) = (0.04657 \text{ kg})(3900 \text{ m})$ $J/kg.^{\circ}C)(94 - 20)^{\circ}C = 13,440$ J From Table 4-2 we read J1 = 0.5760 corresponding to the constant $\lambda 1 = 2.0785$. During transient heat transfer, the temperature and heat flux may vary with time as well as location. Properties The thermal conductivity of plastic cover is given to be k = 0.075 Btu/h·ft. $^{\circ}F$. Then the rate of heat loss for 1-cm thick insulation becomes $T - T \infty$ Ts $- T \infty$ A ($T - T \infty$) (70.69 m 2)(90 - 27)°C = Q& ins = s = e o s = 15,021 W t ins 0.01 m 1 1 R total Rins + Rconv + + 0.038 W/m.°C 30 W/m 2.°C k ins ho Also, the total amount of heat loss from the furnace become Qins = Q& ins $\Delta t = (15.021 \text{ kJ} / \text{sm})$ $(4160 \times 3600 \text{ s}/\text{yr}) = 2.249 \times 108 \text{ kJ}/\text{yr}$ Qin,ins = Qins / η oven = $(2.249 \times 108 \text{ kJ}/\text{yr}) / 0.78 = 2.884 \times 108 \text{ kJ}/\text{yr}$ Cost savings = Energy cost w/o insulation - Energy cost w/o insulation = 12,157 - 1367 = \$10,790/yr The unit cost of insulation is given to be \$10/m2 per cm thickness, plus \$30/m2 for labor. Also, hfg = 198 kJ/kg for nitrogen. Then the mesh Fourier number becomes $\tau = \alpha \Delta t \Delta x 2 = (0.44 \times 10 - 6 \text{ m } 2/\text{s})(900 \text{ s}) (0.05 \text{ m}) 2 = 0.1584$ Initially (at 7 am or t = 0), the temperature of the wall is said to vary linearly between 20°C at node 0 and 0°C at node 6. °F /
Btu Therefore, the amount of energy and money saved by the additional insulation per year are Q& saved = Q& prop - Q& current = 100.2 - 60.2 = 40.0 Btu/h Qsaved = Q& saved × (Unit cost) = (350,400 Btu/yr)(\$0.01 / 1000 Btu) = \$3.504 / yr or \$7.01 per 2 years, which is barely more than the \$7.0 per 2 years are Q& saved = Q& s minimum required. The mesh Fourier number is $\alpha\Delta t$ (7.4 × 10 -6 ft 2 / s)(300 s) $\tau = 2 = 0.320 \Delta x$ (1 / 12 ft) 2 Substituting this value of τ and other given quantities, the inner and outer surface temperatures of the roof after 12×(60/5) = 144 time steps (12 h) are determined to be T1 = 54.75°C and T6 = 40.18°C (b) The average temperature of the inner surface of the roof can be taken to be T1 @ 6 PM + T1 @ 6 AM 70 + 54.75 = 62.38° F T1, ave = 2 2 Then the average rate of heat loss through the roof that night is determined to be) + $\epsilon\sigma A T 4 - (T i + 273) 4 Q \& = h A(T - T Node 6 (convection) : ave i ho (T0 - T6i) + k i 1, ave [wall 1] = (0.9 Btu/h.ft \cdot °F)(45 \times 55 ft)(70 - 62.38)^{\circ}$ F 2 2 + $0.9(45 \times 55 \text{ ft } 2)(0.1714 \times 10 - 8 \text{ Btu/h.ft } 2 \cdot \text{R } 4)[(530 \text{ R}) 4 - (62.38 + 460 \text{ R}) 4] = 33,950 \text{ Btu/h } 5-115 \text{ A large pond is initially at a uniform temperature}.$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 971.8 \text{ kg/m } 3 \text{ C } p = 4197 \text{ J/kg.}^{\circ} C$ The properties of water at 80°C are (Table A-9) $\rho = 4197 \text{ J/kg.}^{\circ} C$ The propertie air at 1 atm and at the anticipated film temperature of 50°C are (Table A-15) k = 0.02735 W/m.°C v = 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Analysis The Reynolds number is (40×1000) m/s (0.50 m) | V ∞ D (3600 / Re = = 309,015 v 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Analysis The Reynolds number is (40×1000) m/s (0.50 m) | V ∞ D (3600 / Re = = 309,015 v 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Analysis The Reynolds number is (40×1000) m/s (0.50 m) | V ∞ D (3600 / Re = = 309,015 v 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Analysis The Reynolds number is (40×1000) m/s (0.50 m) | V ∞ D (3600 / Re = = 309,015 v 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Analysis The Reynolds number is (40×1000) m/s (0.50 m) | V ∞ D (3600 / Re = = 309,015 v 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Analysis The Reynolds number is (40×1000) m/s (0.50 m) | V ∞ D (3600 / Re = = 309,015 v 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Analysis The Reynolds number is (40×1000) m/s (0.50 m) | V ∞ D (3600 / Re = = 309,015 v 1.798×10^{-5} m 2/s Water tank D = 50 cm Pr = 0.7228 L = 95 cm Pr = 0.7228 Pr = 0.72Reynolds number is 5/8 0.62 Re 0.5 Pr 1 / 3 [(Re)] + Nu = 0.3 + 1 | | | | 1 / 4 | (282,000 / |] 1 + (0.4 / Pr) 2 / 3 [Air V = 40 km/h T = 18°C 4/5] 5/8 0.62(309,015) 0.5 (0.7228) 1 / 3 [(309,015)] = 0.3 + 1 + || || 2 / 3 1 / 4 282 , 000 | () |] 1 + (0.4 / 0.7228) [The heat transfer coefficient is k 0.02735 W/m. °C h = Nu = (484.9) = 26.53 W/m 2.°C D 0.50 m The surface area of the tank is [] $4/5 = 484.9 \text{ D2} = \pi (0.5)(0.95) + 2\pi (0.5) 2/4 = 1.885 \text{ m 2 4}$ The rate of heat transfer is determined from ($80 + T_2$) - 18 |°C Q& = hAs (T s - T ∞) = (26.53 W/m 2.°C)(1.885 m 2)| 2 / As = $\pi DL + 2\pi (Eq. 1)$ where T2 is the final temperature of water so that ($80+T_2$)/2 gives the average temperature of water during the cooling process. 3-65C Heat transfer in this short cylinder is one-dimensional since there will be no heat transfer in the surfaces of the volume elements has no effect on the formulation, and some heat conduction terms turn out to be negative. 2-8 Chapter 2 Heat Conduction equation is given in its simplest by r d 2T 2 + dT = 0: dr dr (a) Heat transfer is steady, (b) it is one-dimensional, (c) there is heat generation, and (d) the thermal conductivity is constant. The system of 3 equations with 3 unknown temperatures constitute the finite difference formulation of the problem. \times 10 - 6 m2 / s)(900 s) (0.25 m) 2 The values of heat generation rates at the nodal points are determined as follows: g& 0 = G& 0.473 \times 500 W = 946 W/m 3 Volume (1 m 2)(0.25 m) G& 1 $[(0.473 + 0.061)/2] \times 500 \text{ W} \text{ g} \& 1 = = 534 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ V/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = \text{S o} = 0.002160 \text{ G} \& 4 0.024 \times 500 \text{ W} = 48 \text{ W/m} 3 \text{ Volume} (1 \text{ m} 2)(0.25 \text{ m}) \text{ g} \& 4 = 10.024 \text{ W} = 10$ 2. Analysis The thermal resistances of copper fillings and the epoxy board are in parallel. Analysis The heat transfer surface area of the person is As = $\pi DL = \pi (0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2 \text{ (1.60 m}^2)(34 - 20)^\circ \text{C} = 336 \text{ W}$ In windy air it would be Qwindy air = $hAs\Delta T = (50 \text{ W/m}^2 \cdot \text{°C})(1.60 \text{ m}^2)(34 - 20)^\circ\text{C} = 1120 \text{ W}$ To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose
heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$)still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air, the air temperature must be 1120 W = ($hAs\Delta T$) still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W To lose heat at this rate in still air ($hAs\Delta T$) still air = ($15 \text{ W/m}^2 \cdot \text{°C}$)(1.60 m^2)(34 - 20)^\circ\text{C} = 1120 W T consider a unit surface area of 1 m2. \checkmark Assumptions 1 Heat conduction in each geometry is one-dimensional. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be hL = (0.037 Re L 0.8 - 871) Pr 1 / 3 = 2046 Nu = k k 0.02401 W/m. °C h = Nu = (2046) = 40.93 W/m 2 Ch = Nu = (2046) = 40.93 W/m 2 .°C L 1.2 m The thermal resistances are [] As = 3(1.2 m)(1.5 m) = 5.4 m 2 1 1 = 0.0231 °C/W hi As (8 W/m 2.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0045 °C/W 2 ho As (40.93 W/m.°C)(5.4 m 2.) 1.1 = = 0.0windows become Rtotal = Rconv, i + Rcond + Rconv, o = 0.0231 + 0.0012 + 0.0045 = 0.0288 °C/W T -T [22 - (-2)]°C Q& = $\infty 1 \propto 2 = 833.3$ W Rtotal 0.0288 °C/W 7-86 Chapter 7 External Forced Convection 7-94 A fan is blowing air over the entire body of a person. Properties The thermal properties of the ice are given to be k = 2.22 W/m.°C and $\alpha = 0.124 \times 10^{-7}$ m2/s. and A1 = 12644. The metabolic rate for an average man ranges from 108 W while reading, writing, typing, or listening to a lecture in a classroom in a seated position to 1250 W at age 70) during strenuous exercise. The average convection heat transfer coefficient on the wing surface and the average rate of heat transfer per unit surface area are to be determined. 5 The human body is assumed to be cylindrical in shape for heat transfer purposes. $| = 0.295 \text{ kg } 3 \text{ kg} \cdot \text{c} (170 - 25)^{\circ}\text{C} = 166.76 \text{ kg} \text{ m} = \rho \text{ V} = \rho \text{ Qmax}$ Then the actual amount of heat transfer becomes $\sin(\lambda 1) - \lambda 1 \cos(\lambda 1) \sin(1.8777) - (1.8777) - (1.8777) - (1.8777) \sin(1.8777) - (1.8777) - (1.8777) \sin(1.8777) \sin(1.8777) - (1.8777) \sin(1.8777) \sin(1.8777) \sin(1.8777) - (1.8777) \sin(1.8777) \sin(1.8777) \sin(1.8777) - (1.8777) \sin(1.8777) \sin(1$ $\cos(1.8777) Q = 1 - 3\theta o, \text{sph} = 1 - 3(0.69) = 0.525 3 Q \max \lambda 1 (1.8777) 3 Q = 0.525 Q \max = (0.525)(166.76 \text{ kJ}) = 87.5 \text{ kJ}$ The final equilibrium temperature of the potato after it is wrapped is \rightarrow Teqv = Ti + Q = mC p (7 eqv - Ti) Q 87.5 kJ = 25°C + = 101°C mC p (0.295 kg)(3.9 kJ/kg.) e - (2.1589)(1.734) = 0.0005 = A1e - $\lambda 1 \tau = (15618 \text{ Ti})$ $T \propto [T(0,0, t) - T \propto] = \theta$ o, wall $\times \theta$ o, cyl = 1 $\times 0.0005 = 0.0005 | [T - T \propto]$ short cylinder T (0,0, t) - 212 = 0.0005 \rightarrow T (0,0, t) = 212 °F 40 - 212 (b) Treating the hot dog as an infinitely long cylinder T (0,0, t) - 212 = 0.0005 \rightarrow T (0,0, t) - 212 = 0.0005 \rightarrow minutes Plate: First the Biot number is calculated to be Bi = hL (7 Btu/h.ft 2.°F)(0.5 / 12 ft) = 0.01944 k (15 Btu/h.ft.°F) The constants λ 1 and A1 = 0.1410 and A1 =center temperature of the plate becomes θ 0, wall 2L 2 2 T - T0 - T ∞ 75 = A1 e - λ 1 τ \rightarrow 0 = (1.0033)e - (0.1410) (15.98) = 0.730 \rightarrow T0 = 312°F = Ti - T ∞ 400 - 75 Cylinder: -1 Bi = 0.02 Table 9 \rightarrow λ 1 = 0.1995 and A1 = 1.0050 θ 0, cyl = 2 2 T - 75 T 0 - T ∞ = A1 e - λ 1 τ \rightarrow 0 = (1.0050)e - (0.1995) (15.98) = 0.532 \rightarrow T0 = 248°F Ti - T ∞ 400 $-75 \text{ Sphere:} -1 \text{ Bi} = 0.02 \text{ Table } 9 \rightarrow \lambda 1 = 0.2445 \text{ and } A1 = 1.0060 \ \theta \ 0, \text{ sph} = 2 2 \text{ T} - 75 \text{ T0} - \text{T} = A1 \ e - \lambda 1 \ \tau \rightarrow 0 = (1.0060) e^{-(0.2445)} (15.98) = 0.387 \ \rightarrow \text{T0} = 201^{\circ}\text{F} \text{ Ti} - \text{T} = 400 - 75 \text{ After } 10 \text{ minutes } \tau = \alpha t \text{ L2} = (0.333 \text{ ft } 2 \ h)(10 \text{ min/60 min/h}) (0.5 \ / 12 \text{ ft}) 2 = 31.97 > 0.2 \text{ Plate:} \theta \ 0, \text{ wall} = 2 2 \text{ T} - 75 \text{ T0} - \text{T} = A1 \ e - \lambda 1 \ \tau \rightarrow 0 = (1.0033) e^{-(0.2445)} (15.98) = 0.387 \ \Rightarrow \text{T0} = 201^{\circ}\text{F} \text{ Ti} - \text{T} = 400 - 75 \text{ After } 10 \text{ minutes } \tau = \alpha t \text{ L2} = (0.333 \text{ ft } 2 \ h)(10 \text{ min/60 min/h}) (0.5 \ / 12 \text{ ft}) 2 = 31.97 > 0.2 \text{ Plate:} \theta \ 0, \text{ wall} = 2 2 \text{ T} - 75 \text{ T0} - \text{T} = A1 \ e - \lambda 1 \ \tau \rightarrow 0 = (1.0033) e^{-(0.2445)} (15.98) = 0.387 \ \Rightarrow \text{T0} = 201^{\circ}\text{F} \text{ Ti} - \text{T} = 400 \ \text{T} =$ $-(0.1410)(31.97) = 0.531 \rightarrow T0 = 248^{\circ}F Ti - T_{\infty} 400 - 75 4 - 106 Chapter 4 Transient Heat Conduction Cylinder: \theta 0, cyl = 2 2 T - 75 T 0 - T_{\infty} = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0050)e -(0.2445)(31.97) = 0.245)(31.97) = 0.245)(31.97) = 0.245)(31.97) = 0.245$ $400 - 75\theta 0$, sph = After 30 minutes $\tau = \alpha t L2 = (0.333 \text{ ft } 2 / h)(30 \text{ min } / 60 \text{ min } / h)(0.5 / 12 \text{ ft}) 2 = 95.9 > 0.2$ Plate: $\theta 0$, wall = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0033)e - (0.1410)(95.9) = 0.149 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e -
\lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty 400 - 75$ Cylinder: $\theta 0$, cyl = $22T - 75T0 - T\infty = A1e - \lambda 1\tau \rightarrow 0 = (1.0050)e - (0.1995)(95.9) = 0.0221 \rightarrow T0 = 123^{\circ}FTi - T\infty$ 82° F Ti $-T \propto 400 - 75$ Sphere: θ 0, sph = 2 2 T -75 T 0 $-T \propto = A1 e - \lambda 1 \tau \rightarrow 0 = (1.0060)e - (0.2445)(95.9) = 0.00326 \rightarrow T0 = 76^{\circ}$ F Ti $-T \propto 400 - 75$ The sphere has the largest surface area through which heat is transferred per unit volume, and thus the highest rate of heat transfer. Assumptions 1 The chicken is a homogeneous spherical object. 2 The inner and outer surface temperatures of the refrigerator remain constant. Two infinite plane walls are exposed to the hot gases with a heat transfer coefficient of h = 40 W / m2. Analysis The time required to reduce the inner surface temperature of the chickens from 32°F to 25°F with refrigerated air at -40°F is determined from Fig. 3 75 Chapter 3 Steady Heat Conduction 3-100C Welding or tight fitting introduces thermal contact resistance at the interface, and thus retards heat transfer. The velocity of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the temperature of the air at the duct inlet and the duct inlet and the air at the duct inlet and the duct inlet and the air at 0.216 The volume of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 kg / m 3)(0.824 m 3) = 1.063 kg Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 \text{ kg} / \text{m 3})(0.824 \text{ m 3}) = 1.063 \text{ kg} Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 \text{ kg} / \text{m 3})(0.824 \text{ m 3}) = 1.063 \text{ kg} Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 \text{ kg} / \text{m 3})(0.824 \text{ m 3}) = 1.063 \text{ kg} Energy balance for the air space of the refrigerator cavity and the mass of air inside are V = (1.29 \text{ kg} / \text{ uniformly in the epoxy layers of the board. The U-value of the wall for the case of 15 mph winds outside is to be determined. We note that $\tau = \alpha t L_2 = (9.75 \times 10 - 5 \text{ m } 2/\text{s})(241 \text{ s})(0.1 \text{ m}) 2 = 2.35 > 0.2$ and thus the assumption of $\tau > 0.2$ for the applicability of the one-term approximate solution is verified. It can also be expressed in terms of the temperatures at the neighboring nodes in the following easy-to-remember form: i i i i + Ttop + Tright + Tbottom + Tfront + Tbot the inner side of the wall. 3 Thermal properties of the ceiling and the heat transfer coefficients are constant. Heat is lost from the cylindrical surface at r = r0 by convection to the surrounding medium at temperature T^{\$\$\$\$} with a heat transfer coefficient of h. 5-124 Design and Essay Problems 5-119 Chapter 6 Fundamentals of Convection Chapter 6 FUNDAMENTALS OF CONVECTION Physical Mechanisms of Forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or a fan. 3-6C Yes. Properties The thermal conductivity is given to be k = 15.1 W/m·°C. ∂ 2T 1 ∂ T = : ∂ x 2 α ∂ t (a) Heat transfer is transient, (b) it is one dimensional, (c) there is no heat generation, and (d) the thermal conductivity is constant. qsolar 520 R Analysis In steady operation, heat conductivity is given to be k = 15 W/m. C. Using the proper relation in laminar flow for Nusselt number, heat transfer coefficient and the heat transfer rate are determined to be hL Nu = $0.664 \text{ Re L } 0.5 \text{ Pr1} / 3 = 0.664(55,617)0.5 (0.7228)1 / 3 = 140.5 \text{ k} & 0.02735 \text{ W/m} 2 \cdot C \text{ L } 0.25 \text{ m} \text{ As} = \text{wL} = (0.25 \text{ m})(0.25 \text{ m}) = 0.0625 \text{ m 2} (0.25 \text{ m}) = 0.0625 \text{ m 2} (0.0625 \text{ m 2})(0.625 \text{ m 2})(0.$ Considering that each transistor dissipates 3 W of power, the number of transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher. Using energy balances, the finite difference equations for each of the 8 nodes are obtained as follows: Node 1: $q\&L111T2 - T11T4 - T112 + h(T\infty - T1) + k + k + g\&0 = 0112224T - T21T1 - T21T1$ 2 Node 5: T4 + T2 + T6 + 120 - 4T5 + g& 0 12 = 0 k Node 6: h (T ∞ - T6) + k T - T6 120 - T6 1T3 - T6 1T7 - T6 312 + k + g& 0 = 0 1112 2 2 12 Node 7: h (T ∞ - T7) + k 120 - T7 1T6 - T7 1T6 - T7 1T6 - T7 1T6 - T7 1T7 - T8 1120 - T8 12 + k + g& 0 = 0 112 2 4 where g& 0 = 5 × 10 6 W/m 3 q&L = 8000 W/m 2, l = 0.015 m, $k = 45 \text{ W/m} \cdot \text{°C}$, $h = 55 \text{ W/m} \cdot \text{°C}$, and $T \propto = 30^{\circ}\text{C}$. We evaluate the air properties at the assumed mean temperature of 20°C (will be checked later) and 1 atm (Table A-15): $k = 0.02514 \text{ W/m} \cdot \text{°C}$, $h = 55 \text{ W/m} \cdot \text{°C}$, $h = 50.02514 \text{ W/m} \cdot \text{°C}$, $h = 50.02514 \text{ W/m} \cdot \text{°C}$, $h = 1.204 \text{ kg/m} \cdot \text{°C}$, h = 1.2of air at the inlet temperature of 15°C (for use in the mass flow rate calculation at the inlet) is $\rho i = 1.225$ kg/m3. The properties of anisotropic materials, however, may change with direction. In winter, we can keep warm by dressing heavily, staying in a warmer environment, and avoiding drafts. The heat transfer terms are expressed at time step i in the explicit method, and at the future time step i + 1 in the implicit method as Explicit method as Explicit method as Explicit method. Σ Q& i + 1 All sides Tmi+1 - Tmi Δt T i + 1 - Tmi + 1 - T general interior node for transient heat conductivity of concrete is given by Tmi-1 - 2Tmi + Tmi+1 + m. Properties The thermal conductivity of concrete is given by Tmi-1 - 2Tmi + Tmi+1 + m. Properties The thermal conductivity of concrete is given by Tmi-1 - 2Tmi + Tmi+1 + m. Properties The thermal conductivity of concrete is given by Tmi-1 - 2Tmi + Tmi+1 + m. and the mass flow rate of air (evaluated at the inlet) are As = $N\pi DL = 300\pi (0.008 \text{ m})(0.4 \text{ m}) = 4.524 \text{ m} 2 \text{ m} \& = m \& i = \rho i V(NT \text{ ST } L) = (1.292 \text{ kg/m} 3)(4 \text{ m/s})(15)(0.015 \text{ m})(0.4 \text{ m}) = 0.4651 \text{ kg/s}$ Then the fluid exit temperature, the log mean temperature, the log mean temperature, the log mean temperature difference, and the rate of heat transfer (refrigeration capacity) become (Ah Te = Ts - (Ts - Ti) = (1.292 \text{ kg/m} 3)(4 \text{ m/s})(15)(0.015 \text{ m})(0.4 \text{ m}) = 0.4651 \text{ kg/s} Then the
fluid exit temperature, the log mean temperature difference, and the rate of heat transfer (refrigeration capacity) become (Ah Te = Ts - (Ts - Ti) = (1.292 \text{ kg/m} 3)(4 \text{ m/s})(15)(0.015 \text{ m})(0.4 \text{ m}) = 0.4651 \text{ kg/s} Then the fluid exit temperature difference, and the rate of heat transfer (refrigeration capacity) become (Ah Te = Ts - (Ts - Ti))(1.292 \text{ kg/m} 3)(4 \text{ m/s})(1.292 \text{ kg/m} 3)(1.292 \text{ $exp - s - m \& C p & \Delta T \ln = 22 & \Delta T \ln = (156.2 \text{ W/m} \cdot \text{°C}) = -15.58 & C & (0.4651 \text{ kg/s})(1006 \text{ J/kg} \cdot \text{°C}) & | \langle J \rangle & (Ts - Te) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & (-20 - 0) & (-20 + 15.58) & ((4.524 \text{ m 2})(10.32^{\circ}\text{C}) = 7294 \text{ W}$ For this staggered arrangement tube bank, the friction coefficient corresponding to ReD = 5294 and SL/D = 1.5/0.8 = 1.875 is, from Fig. Analysis (a) Taking the direction normal to the surface of the wall to be the x direction with x = 0 at the left surface, the mathematical formulation of this problem can be expressed as and d 2T = 0 dx 2 dT (0) $-k = q \& 0 = 950 W/m 2 dx k 2 q = 950 W/m T1 = 85^{\circ} C (b)$ Integrating the differential equation twice with respect to x yields dT = C1 dx x T (x) = C1x + C2 where C1 and C2 are arbitrary constants. Analysis We take the air-conditioning duct as the system. The density of the body is ρ and the specific heat is C. For slender bodies such as airfoils, the friction drag is usually more significant. 4 There is no convection in space. Properties The thermal conductivity and emissivity are given to be k = 1.1 Btu/h·ft·°F and ε = 0.9. Analysis In steady operation, heat conduction through the roof must be equal to net heat transfer from the outer surface Properties The thermal conductivity of the concrete is given to be k = 0.75 W/m.°C. Heat transfer from the oven is three-dimensional in nature since heat will be entering through all six sides of the oven. This is called separation. The velocity and temperature distributions, the maximum temperature, and the heat flux are to be determined. Using the at the worst case, the accumulated discretization error after I time steps during a time period t0 is I Δ t 2 = t0 Δ t which is proportional to Δ t. For a specified nodal network, these two methods will result in the same set of equations. The mass of the iron is to be determined. 3 The surfaces are black and thus ε = 1. 2 Air is an ideal gas with constant specific heats. Therefore, taking the outer surface temperature of the roof to be T2 (in °F), T – T 4 Tsky x kA 1 2 = hA(T2 – T ∞) + ϵ Ag [(T2 + 460) 4 – Tsky] T ∞ L h Canceling the area A and substituting the known quantities, L (62 – T2) °F (1.1 Btu/h · ft · °F) = (3.2 Btu/h · ft 2 · °F)(T2 – 50) °F 0.8 ft + 0.8(0.1714 × 10 – 8 Btu/h · ft 2 R 4 [(T2 + 460) 4 - 310 4] R 4 To Using an equation solver (or the trial and error method), the outer surface temperature is determined to be $T2 = 38^{\circ}F$ Then the rate of heat transfer through the roof becomes T - T (62 - 38)° F Q& = $kA 1 2 = (11 \cdot Assumptions 1 A ir is an ideal gas since it is at a high temperature and low pressure relative to its$ critical point values of -222°F and 548 psia. Substituting, the fraction of heat conducted along the copper layer and the effective thermal conductivity of the plate are determined to be (kt) copper = (223 Btu / h.ft. Substituting these values into the one-term solution gives $\theta 0 = To - T\infty 2 = A1e - \lambda 1\tau \rightarrow Ti - T\infty$ Potato $Ti = 25^{\circ}C 6 - 2$. 4 The thermal properties of the fins are constant. Analysis We take the oxygen in the tank as our system. ° F)[2π (3 / 12 ft)(1 ft)] Then the steady rate of heat loss from the steady rat pipe to be 80°C (we will check this assumption later), the radiation heat transfer coefficient is determined to be hrad = $\varepsilon \sigma$ (T2 2 + Tsurr 2) (T2 + Tsur transfer coefficients can be added and the result can be taken as the combined heat transfer through the bottom surface of the plate is negligible. Alternative solution We could also solve this problem using transfer through the bottom surface of the plate is negligible. Alternative solution We could also solve this problem using transfer through the bottom surface of the plate is negligible. Alternative solution We could also solve this problem using transfer through the bottom surface of the plate is negligible. = 0.4 To $-T \propto 4 - (-6)$ ro = 0.23 Ti $-T \propto 37 - (-6)$ Therefore, $t = \tau$ ro 2 (0.4)(0.12 m) $2 = 44,308s \approx 12.3h \alpha 0.13 \times 10 - 6 m 2/s$ The surface temperature is determined from k 1 = 0.178 Bi hro $T (r) - T \propto = 0.17$ Fig. 3 The light bulb is in spherical shape. 3 There is heat generation only at the inner surface which will be treated as prescribed heat flux. Analysis The thermal resistance network and the individual thermal resistances are R1 R2 R4 R3 R6 R5 R7 R8 R9 R0 T 2 T1 L 0.0001 m = = 0.00152 °C / W kA (0.06 W / m. 3 Radiation effects are negligible. As a result, the two half pieces will cook much faster than the single large piece. Assumptions 1 The chickens are spherical in shape. Assumptions Heat is generated uniformly in the resistance wire. Properties The thermal conductivity is given to be k = 2.5 W/m. °C. m 018 = = 0.833 °C / W kA (0.72 W / m. T0 Chapter 6 Fundamentals of Convection 6-45 The oil in a journal bearing is considered. 3 The heat transfer coefficient is constant and uniform over the entire surface. Assumptions 1 The can containing the drink is cylindrical in shape with a radius of r0 = 1.25 in. This is because the steady heat conduction equation in a plane wall is d 2 T / dx 2 = 0 whose solution is T (x) = C1 x + C2 regardless of the boundary conditions. Analysis For each sample we have Q& Q& & Q = 28 / 2 = 14 W A = (01 . 1-55 Chapter 1 Basics of Heat Transfer 1-103 "GIVEN" L=1.4 "[m]" D=0.002 "[m]" T_infinity=20 "[C]" "T_s=240 [C], parameter to be varied" V=110 "[Volt]" I=3 "[Ampere]" "ANALYSIS" Q_dot=V*I A=pi*D*L Q_dot=h*A*(T_s-T_infinity) Ts [C] 100 120 140 160 180 200 220 240 260 280 300 h [W/m2.C] 468.9 375.2 312.6 268 234.5 208.4 187.6 170.5 156.3 144.3 134 500 450 400 2 h [W /m -C] 350 300 250 200 150 100 100 140 180 220 T s [C] 1-56 260 300 Chapter 1 Basics of Heat Transfer 1-104E A spherical ball whose surface is maintained at a temperature of 170°F is suspended in the middle of a room at 70°F. The number of fins needed to triple the rate of heat transfer is to be determined. The minimum thickness of insulation that needs to be used in the wall in order to avoid condensation on the outer surfaces is to be determined. 5 Evaporative cooling is negligible. Besides, once a person is used to solving problems numerically, it is very difficult to go back to solving differential equations by hand. Heat transfer through the air space and through the studs will meet different resistances, and thus we need to analyze the thermal resistance for each path separately. The nodal temperatures after 5 min are to be determined using the explicit finite difference method. 3-40C The thermal contact resistance will be greater for rough surfaces because an interface with rough surfaces will contain more air gaps whose thermal conductivity is low. The temperatures at the centers of the cylinder for 15 min of cooling are to be determined. 5-99C The Taylor series expansion of the temperature at a specified nodal point m about time ti is T (x m, ti + Δt) = T (x m, ti) + $\Delta t \partial T$ (x m, ti) + $\Delta t \partial T$ (x m, ti) + $\Delta t \partial t$ which resembles the same nodal point is expressed as ∂T (x m, ti) + $\Delta t \partial T$ (x m, ti) + $\Delta t \partial t$ Taylor series expansion terminated after the first two terms. Analysis The nodal spacing is given to be $\Delta x=0.1$ m. Analysis The temperature difference between the front and back surfaces of the chip is A = (0.006 m)(0.006 m) = 0.000036 m 2 (3 W)(0.0005 m) Q& L $\Delta T \rightarrow \Delta T = = 0.32$ °C Q& = kA L kA (130
W/m.°C)(0.000036 m 2) Q& Ceramic substrate 3W Chip $6 \times 6 \times 0.5$ mm 1-37 Chapter 1 Basics of Heat Transfer 1-77 An electric resistance heating element is immersed in water initially at 20°C. 7 The heat transfer coefficient accounts for the effect of radiation from the fins. Refrigerated air The Fourier number is 2 T (L, t) - T ∞ = A1e - λ 1 τ cos(λ 1 L / L) Ti - T ∞ 2 2 - (-11) = $(1.0311)e - (0.4328) \rightarrow \tau = 5.601 > 0.2 25 - (-11) - 11^{\circ}C$ Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. Using the available R-values from Table 3-6 and calculating others, the total R-values for each section is determined in the table below. 4-91C (a) High air motion retards the growth of microorganisms in foods by keeping the food surfaces dry, and creating an undesirable environment for the microorganisms. m2 Aunfinned = $0.0216 - 864 \times \pi$ (0.0025) 2 4 = 0130. e - (1.308) $\tau \rightarrow \tau = 0.7837 - (-30)$ which is greater than 0.2 and thus the one-term solution is applicable. 4 The local atmospheric pressure is 100 kPa. Properties The gas constant of air is R = 0.287 kPa.m3/kg.K (Table A-1). 1-93 A sealed electronic box dissipating a total of 100 W of power is placed in a vacuum chamber. 2 The edge effects are negligible. 4 Heat transfer from the base of the ice chest is negligible. 2 The thermal properties of the cylinder are constant. The specific heat of water at the average temperature of $(100+20)/2 = 60^{\circ}$ C is 4.185 kJ/kg·°C (Table A-9). 3-112 Rlimestone T ∞ 2 Chapter 3 Steady Heat Conduction 3-153E Steam is produced in copper tubes by heat transferred from another fluid condensing outside the tubes at a high temperature. Construction 6 5a 4 1. Assumptions 1 Heat transfer through the wall is one-dimensional. 2-87E Heat is generated uniformly in a resistance heater wire. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 864 aluminum pin fins on the back surface. temperature rise of the cars becomes $\Delta T = mV 2 / 2 V 2 / 2$ (90,000 / 3600 m/s) $2 / 2 (1 kJ/kg) = = | = 0.69^{\circ}C | mC C 0.45 kJ/kg.^{\circ}C (1000 m 2 / s 2 / 1.26 A classroom is to be air-conditioning units. The finite difference equation for nodes 1 and 4 on the surfaces subjected to convection is obtained by applying an energy$ balance on the half volume element about the node, and taking the direction of all heat transfers to be towards the node under consideration: Node 1 (convection) : T1i +1 or i +1 i T2i - T1i Δx T1 - T1 C = ρ 2 Δt Δx h Δx (To - T4i) + k Node 2 (interior): or T4i + 1 = τ (T1i = τ (T2i + T3i) + (1 - 2 τ)T3i T3i - T4i $\Delta x \Delta x = \rho$ hi Ti C 2 h Δx) h $\Delta x (= || 1 - 2\tau - 2\tau o || T4i + 2\tau T3i + 2\tau o To k / k (T4i + 1 - T4i \Delta t \Delta \cdot 1 Window glass \cdot 2 \cdot 3 ho To Fo where <math>\Delta x = 0.125/12$ ft , k = 0.48 Btu/h.ft·°F, hi = 1.2 Btu/h.ft·°F, hi = 35+2*(t/60)°F (t in seconds), ho = 2.6 Btu/h.ft2·°F, and To =35°F. 5-112 Chapter 5 Numerical Methods in Heat Conduction 5-116 A large 1-m deep pond is initially at a uniform temperature of 15°C throughout. 4 - 13b) which gives Tsurface = T ∞ + 0.22(To - T ∞) = -27.4°C The slight difference between the two results is due to the reading error of 15°C throughout. 4 - 13b) which gives Tsurface = T ∞ + 0.22(To - T ∞) = -27.4°C The slight difference between the two results is due to the reading error of 15°C throughout. the charts. The ρ Cp (volumetric specific heat) values of the steaks and of the defrosting plate are (ρ Cp) plate = k α = 237 W/m · °C -6 = 2441 kW/m 3 · °C 97.1 × 10 m / s (ρ Cp) steak = (970 kg/m 3)(1.55 kJ/kg · °C) = 1504 kW/m 3 · °C 2 Analysis The nodal spacing is given to be Δx = 0.005 m in the steaks, and Δr = 0.0375 m in the plate. A practical way of dealing with such geometries in the finite difference method is to replace the elements bordering the irregular geometry by a series of simple volume elements. Assumptions 1 Heat conduction is steady and one-dimensional since the plate are uniform. Properties The properties of air at 1 atm and the film temperature of $(Ts + T\infty)/2 = (75+5)/2 = 40^{\circ}$ C are (Table A-15) Wind $V\infty = 10 \text{ km/h } T\infty = 5^{\circ}$ C r = 0.7255 Analysis The Reynolds number is V D [(10 × 1000/3600) m/s](0.1 m) Re = $\infty = = 1.632 \times 10.4 \text{ v} \cdot 1.702 \times 10.-5 \text{ m} \cdot 2$ /s Steam pipe Ts = 75°C Pr = 0.7255 Analysis The Reynolds number is V D [(10 × 1000/3600) m/s](0.1 m) Re = $\infty = = 1.632 \times 10.4 \text{ v} \cdot 1.702 \times 10.-5 \text{ m} \cdot 2$ /s Steam pipe Ts = 75°C Pr = 0.7255 Analysis The Reynolds number is V D [(10 × 1000/3600) m/s](0.1 m) Re = $\infty = = 1.632 \times 10.4 \text{ v} \cdot 1.702 \times 10.-5 \text{ m} \cdot 2$ /s Steam pipe Ts = 75°C Pr = 0.7255 Analysis The Reynolds number is V D [(10 × 1000/3600) m/s](0.1 m) Re = $\infty = = 1.632 \times 10.4 \text{ v} \cdot 1.702 \times 10.-5 \text{ m} \cdot 2$ /s Steam pipe Ts = 75°C Pr = 0.7255 Analysis The Reynolds number is V D [(10 × 1000/3600) m/s](0.1 m) Re = $\infty = = 1.632 \times 10.4 \text{ v} \cdot 1.702 \times 10.-5 \text{ m} \cdot 2$ /s Steam pipe Ts = 75°C Pr = 0.7255 Analysis The Reynolds number is V D [(10 × 1000/3600) m/s](0.1 m) Re = $\infty = = 1.632 \times 10.4 \text{ v} \cdot 1.702 \times 10.-5 \text{ m} \cdot 2$ /s Steam pipe Ts = 75°C Pr = 0.7255 Analysis The Reynolds number is V D [(10 × 1000/3600) m/s](0.1 m) Re = $\infty = = 1.632 \times 10.4 \text{ v} \cdot 1.702 \times 10.-5 \text{ m} \cdot 2.5 \text{ m} \cdot 2.5$ $2/s D = 10 \text{ cm } \varepsilon = 0.8 \text{ The Nusselt number corresponding this Reynolds number is determined to be hD 0.62 Re 0.5 Pr 1/3 Nu} = 0.3 + 1/4 k 1 + (0.4/Pr) 2/3 [] (Re) 5/8] 1 + || || || (282,000) || 4/5 [0.62(1.632 \times 104) 0.5 (0.7255)1/3 || 1.632 \times 104 = 0.3 + 1 + 1/4 || (282,000) 1 + (0.4/0.7255) 2/3 [] || 5/8] 4/5 ||] = 0.3 + 1/4 k 1 + (0.4/Pr) 2/3 [] (Re) 5/8] 1 + || || || (282,000) || 4/5 [0.62(1.632 \times 104) 0.5 (0.7255)1/3 || 1.632 \times 104 = 0.3 + 1 + 1/4 || (282,000) 1 + (0.4/0.7255) 2/3 [] || 5/8] 4/5 ||] = 0.3 + 1/4 k 1 + (0.4/Pr) 2/3 [] (Re) 5/8] 1 + || 1| || || (282,000) || 4/5 [0.62(1.632 \times 104) 0.5 (0.7255)1/3 || 1.632 \times 104 = 0.3 + 1 + 1/4 || (282,000) 1 + (0.4/0.7255) 2/3 [] || 5/8] 4/5 ||] = 0.3 + 1/4 k 1 + (0.4/Pr) 2/3 [] (Re) 5/8] 1 + || 1| || || (Re) 5/8] 1 + || 1| || || (Re) 5/8] || 1 + || 1| || || || (Re) 5/8] || 1 + || 1| || || || 1.632 \times 104 = 0.3 + 1 + 1/4 || (282,000) 1 + (0.4/0.7255) 2/3 [] || 1.632 \times 104 || 1.632$ 71.19 The heat transfer coefficient is 0.02662 W/m.°C k h = Nu = (71.19) = 18.95 W/m 2.°C D 0.1 m The rate of heat loss by convection is As = $\pi DL = \pi (0.1 \text{ m})(12 \text{ m}) = 3.77 \text{ m} 2 \text{ Q} \&$ = hA (T - T) = (18.95 W/m 2.°C)(3.77 m 2)(5.67 m 2) × 10 -8 W/m 2 .K 4) (75 + 273 K) 4 - (0 + 273 K) 4 = 1558 W The total rate of heat loss then becomes Q& = Q& + Q& = 5001 + 1558 = 6559 W total conv rad The amount of heat loss from the steam during a 10-hour work day is Q = Q& $\Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day}$ total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& $\Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day}$ total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& $\Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day}$ total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& $\Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day}$ total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& $\Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day}$ total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& $\Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day}$ total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& $\Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day}$ total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& \Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day} total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& \Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day} total The total amount of heat loss from the steam during a 10-hour work day is Q = Q& \Delta t = (6.559 \text{ kJ/s})(10 \text{ h/day} \times 3600 \text{ s/h}) = 2.361 \times 10.5 \text{ kJ/day} per year is Qtotal = Q& day (no. Then the number of nodes becomes $M = L / \Delta x + 1 = 0.375/0.125 + 1 = 4$. It was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the
middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteenth century after it was abandoned in the middle of the nineteen number is Air hL (25 W/m 2 .°C)(0.05 m) Bi = = = 5.365 T = 0°C k (0.233 W/m.°C) The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, λ 1 = 1.2431 τ = α t (0.11×10 - 6 m 2/s)(6 h × 3600 s/h) = 0.9504 > 0.2 L2 (0.05 m) 2 Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 864 copper pin fins on the back surface. Assumptions 1 Heat conduction in the wood is one-dimensional since it is long and it has thermal symmetry about the center line. 2 Thermal properties of the teapot and the water are constant Properties The thermal conductivity of plastic cover is given to be $k = 0.15 \text{ W/m} \cdot \text{C}$. Analysis (a) Taking the direction normal to the surface, the mathematical formulation of this problem can be expressed as and k d 2T = 0 dx 2 dT (0) $-k = q \& 0 = 700 \text{ W/m} \cdot \text{T} = 80 \degree \text{C}$. $L=0.3 \text{ m T}(0) = T1 = 80^{\circ} \text{ C}(b)$ Integrating the differential equation twice with respect to x yields x dT = C1 dx T (x) = C1 x + C2 where C1 and C2 are arbitrary constants. But the value of this constant must be zero since one side of the wall is perfectly insulated. This would be a transfer process since the temperature at any point within the potato will change with time during cooking. The average temperature of (Ts + $T\infty$)/2 = (65+30)/2 = 47.5°C are (Table A-15) 20 cm k = 0.02717 W/m.°C $v = 1.774 \times 10.5 \text{ m} 2$ /s 65°C Pr = 0.7235 Analysis The Reynolds number is VD [(200/60) m/s](0.2 m) Re = $\infty = 3.758 \times 10.4 - 5.2 \cup 1.774 \times 10$ m/s Air 30°C 200 m/min Using the relation for a square duct from Table 7-1, the Nusselt number is determined to be hD Nu = $0.102(3.758 \times 10.4) \times 10.675(0.7235)1/3 = 112.2$ k The heat transfer coefficient is k transfer c 0.02717 W/m.°C h = Nu = (112.2) = 15.24 W/m 2.°C) 0.2 m Then the rate of heat transfer from the duct becomes As = (4 × 0.2 m)(1.5 m) = 1.2 m 2 Q& = hAs (Ts - T\infty) = (15.24 W/m 2.°C)(1.2 m 2)(65 - 30)°C = 640.0 W 7-47 Chapter 7 External Forced Convection 7-56 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. Properties The specific heat of iron is given in Table A-3 to be 0.45 kJ/kg.°C, which is the value at room temperature. C)(0.03m) $\alpha t \mid \tau = 2 = 0.75$ To $-T \propto ro 6-2 \mid \tau = 0.174 \mid Ti - T \propto 25 - 2 \mid Therefore, t = \tau r 0.2 (0.75)(0.03) 2 = 5192$ s ≈ 1.44 h $\alpha 013$. The magnitude and location of the maximum temperature that occurs in the board is to be determined. 1-6C Modeling makes it possible to predict the course of an event before it actually running expensive and time-consuming experiments. Assumptions 1 Heat transfer is one-dimensional. It is not a good idea to keep the bathroom fans on all the time since they will waste energy by expelling conditioned air (warm in winter and cool in summer) by the unconditioned outdoor air. Therefore we need to evaluate dynamic viscosity at a new surface temperature which we will assume to be -100°C. 2 The thermal properties of the balls are constant. Then the heat transfer coefficient can be determined from Bi = hro kBi (0.26 Btu/h.ft.°F)(20) \rightarrow h = = = 14.7 Btu/h.ft k ro (0.3545 ft) 2. Analysis Using the energy balance approach with a unit area A = 1 and taking the direction of all heat transfers to be towards the node under consideration, the finite difference formulations become Node 0 (at left boundary): T i +1 - T0i T1i - T0i T $\Delta x = \rho A CA 0 \Delta x 2 \Delta t Node 1 (at interface): kA Insulated i + 1 i T i - T1i (\Delta x \Delta x) T - T1 kA 0 + kB 2 = |\rho A CA + \rho B CB | 1 \Delta x \Delta x 2 2 \Delta t | Node 2 (at right boundary): 4 \varepsilon \sigma [Tsurr - (T2i) 4] + k B T i + 1 - T2i \Delta x = \rho B CB 2 \Delta x 2 \Delta t 5 - 72 A Radiation B \Delta x 0 \cdot \varepsilon 1 \cdot Interface 2 \cdot Tsurr Chapter 5 Numerical Methods in Heat Conduction$ 5-82 A pin fin with negligible heat transfer from its tip is considered. For the idealized inviscid fluids (fluids with zero viscosity), there will be no velocity boundary layer. 4 Heat transfer coefficients account for the radiation heat transfer coefficients account for the radiation heat transfer set. conditions are determined by solving the 3 equations above simultaneously with an equation solver to be T1 = 62.4°F = 522.4 R, T2 = 64.8°F = 522.4 R, T2 = 522.4 R 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 Node 1 (at the mid plane): k1 A Node 1 (at the mid plane): k1 A Node 2 (at right boundary): T1 - T0 + g & 0 (AAx / 2) = 0 Ax T0 - T1 T - T + k1 A 2 1 + g & 1 $(A\Delta x/2) = 0 \Delta x \Delta x 4 \epsilon \sigma A(Tsurr - T24) + k 2 A T1 - T2 + g \& 2 (A\Delta x/2) = 0 \Delta x 5-22 A pin fin with negligible heat transfer from its tip is considered. The refrigeration capacity and the pressure drop across the tube bank are to be determined. 4 Heat transfer through the bottom surface is negligible. The thermal resistance network and the$ individual thermal resistances are Ri R2 R1 T ∞ 1 R4 R3 R5 T ∞ 2 1 1 = 0.185 °C/W R1 = R 4 = R sheetrock = kA (0.17 W/m.°C)(0.65 m 2) R1 = 0.12 m L = 21.818 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R1 = R 4 = R sheetrock = kA (0.17 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R1 = R 4 = R sheetrock = kA (0.17 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R2 = R stud = 10.000 °C/W R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fiberglass = kA (0.034 W/m.°C)(0.65 m 2) R3 = R fibergla $1 = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0157$. For $\partial 2 T (x, t) 2 \Theta L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the
pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = \rho V = \rho \pi L = (1000 \text{ kg / m}) \pi (0.01 \text{ m}) (0.5 \text{ m}) = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to freeze the water in the pipe completely is Soil m = 0.045 \text{°C/W 2 o ho A (34 W/m . k] The amount of heat required to ho A (34 W/m . k] The amount of heat required t$ and t). For a sufficiently small time step, these terms decay rapidly as the order of derivative increases, and their contributions become smaller. Properties The thermal conductivity is given to be k = 8 W/m. °C. The cooling load, the air flow rate, and their contributions become smaller are to be determined. 4-8C The cylinder will cool to be k = 8 W/m. °C. faster than the sphere since heat transfer rate is proportional to the surface area, and the sphere has the smallest area for a given volume. Properties The thermal conductivity of the aluminum fins is given to be k = 237 W/m·°C. °C D 0.15 m The average rate of heat transfer can be determined from Newton's law of cooling by using average surface temperature of the ball As = $\pi D 2 = \pi (0.15 \text{ m}) 2 = 0.07069 \text{ m} 2 Q_{\&}$ ave = hAs (Ts - T ∞) = (25.12 W/m 2.°C)(0.07069 m 2)(300 - 30)°C = 479.5 W Assuming the ball temperature to be nearly uniform, the total heat transferred from the ball during the cooling from 350 °C to 250 °C can be determined from Qtotal = mC p (T1 - T2) where m = ρV = $\rho \, \pi D \, 3 \, 6 = (8055 \, \text{kg/m} \, 3) \, \pi \, (0.15 \, \text{m}) \, 3 \, 6 = 14.23 \, \text{kg} \, \text{Therefore}, Q \text{total} = \text{mC p} \, (T1 - T2) = (14.23 \, \text{kg})(480 \, \text{J/kg.°C})(350 - 250)^\circ \text{C} = 683,249 \, \text{J} \, \Delta t = = 1425 \, \text{s} = 23.75 \, \text{min} \, 479.5 \, \text{J/s} \, Q \& 7-28 \, \text{Chapter} \, 7 \, \text{External Forced Convection} \, 7-41 \, \text{"IPROBLEM 7-41"} \, \text{GIVEN"} \, D = 0.15 \, \text{"Im}] \, T_1 = 350 \, \text{"Im}] \, T_1 = 350 \, \text{"Im}] \, T_1 = 350 \, \text{Im}] \, T$ T infinity=30 "[C]" P=101.3 "[kPa]" "Vel=6 [m/s], parameter to be varied" rho ball=8055 "[kg/m^3]" C p ball=480 "[]/kg-C]" "PROPERTIES" Fluid\$, T=T infinity) rho=Density(Fluid\$, T=T infinity) rho=Density(Fluid\$ mu s=Viscosity(Fluid\$, T=T s ave) T s ave=1/2*(T 1+T 2) "ANALYSIS" Re=(Vel*D)/nu Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu infinity/mu s)^0.25 h=k/D*Nusselt A=pi*D^2 Q dot ave=h*A*(T s ave-T infinity) Q total=m ball*C p ball*(T 1-T 2) m ball=rho ball*(T 1-T 2) m ball=rho ball*(C 1-T 2) m bal [m/s] 1 1.5 2 2.5 3 3.5 4 4.5 5 5.5 6 6.5 7 7.5 8 8.5 9 9.5 10 h [W/m2.C] 9.204 11.5 13.5 15.29 16.95 18.49 19.94 21.32 22.64 23.9 25.12 26.3 27.44 28.55 29.63 30.69 31.71 32.72 33.7 time [min] 64.83 51.86 44.2 39.01 35.21 32.27 29.92 27.99 26.36 24.96 23.75 22.69 21.74 20.9 20.14 19.44 18.81 18.24 17.7 7-29 Chapter 7 External Forced Convection 35 70 30 60 h 50 20 40 15 30 tim e 10 5 1 20 2 3 4 5 6 Vel [m /s] 7-30 7 8 9 10 10 tim e [m in] 2 h [W /m -C] 25 Chapter 7 External Forced Convection 7-42E A person extends his uncovered arms into the windy air outside. 4-29C The Fourier number is a measure of heat conducted through a body relative to the heat stored. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipe are to be determined with the finite difference method. ° C 4 W / m 2. The rate of heat loss from the steam pipe, the annual cost of this heat loss, and the thickness of fiberglass insulation needed to save 90 percent of the heat lost are to be determined. (a) Nonreflective surfaces, $\varepsilon 1 = \varepsilon 2 = 0.9$ and thus ε effective = 1 / $\varepsilon 1 + 1 / \varepsilon 2 - 1 1 / 0.9 + 1 / 0.9 - 1$ Construction 1. 4 The average human body can be treated as a 1-ft-diamter cylinder with an exposed surface area of 18 ft2. These values account for the effects of the vertical ferring. The amount of heat transfer through the glass in 5 h is to be determined. m) Rtotal = Rboard + Rconv = 0.011 + 1481 = 1492 °C / W. Dividing both sides by k and integrating twice give Energy: $0 = k \partial 2T 2 T(y) = -\mu(y) |V| + C3 y + C4 2k L / Applying the boundary conditions T(0) = T1 and T(L) = T2 gives the temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L L 2||/ (b) The temperature distribution to be T(y) = T2 - T1 \mu V 2 y + T1 + L 2k (y y 2) | - ||L 2||/ (b) The temperature distribution to be T(y) = T2$ $dT T2 - T1 \mu V2 (= + | 1 - 2 | dy L2kL (L) The location of maximum temperature is determined by setting dT/dy = 0 and solving for y, (T - T1) y dT T2 - T1 \mu V2 (y = L | k221 + | = + - | 1 - 2 | = 0 | \mu V2 | / 2kL (dy LL / (6-9 Chapter 6 Fundamentals of Convection The maximum temperature is the value of temperature at this y, whose$ $(0.0002174 \text{ m}) 2 \mid 0.0004 \text{ m} - (0.0004 \text{ m}) 2 \mid 0.0004 \text{ m} - (0.0004 \text{ m}) 2 \mid 0.0004 \text{ m} - (0.145 \text{ W/m.°C}) q \leq 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -k y = 0 T2 - T1 T - T \mu V 2 \mu V 2 - k (1 - 0) = -k 2 1 - 2kL 2L L L = -(0.145 \text{ W/m.°C}) q \leq L = -k dT dy = -$ Construction 6 5a 4 7 1. Properties The thermal conductivity of the glass wool insulation is given to be k = 0.0364 °C. m2) 1 1 Ro = = = 0.0364 °C / W hA (25 W / m 2 . \cong 3 hours (a) Assuming the entire water content of turkey is frozen, the amount of heat that needs to be removed from the turkey as it is cooled from 1°C to -18°C is Cooling to -2.8° C: Qcooling, fresh = (mC Δ T) fresh = (7 kg)(2.98 kJ/kg · °C)[1 - (-2.8)^{\circ}C] = 79.3 kJ Freezing at -2.8°C: Qcooling -18°C: Q cooling -18°C: Q co m). \times 0.8) + (12. Analysis (a) Taking the direction normal to the surface of the wall to be the x direction with x = 0 at the inner surface, the mathematical formulation of this problem can be expressed as d 2T = 0 dx 2 and h1 [T ∞ 1 - T (0)] = -k - k dT (0) dx k h2 T ∞ 2 h1 T ∞ 1 dT (L) = h2 [T (L) - T ∞ 2] dx (b) Integrating the differential equation twice with respect to x yields dT = C1 dx T (x) = C1x + C2 L where C1 and C2 are arbitrary constants. °C 15.1 W / m. The thermal resistance if the wall is constructed of solid bricks are R2 Ri R1 R3 T T $\infty
\infty 11$ R5 R0 R4 1 1 = = 1.7068 h°F/Btu 2 hi A (1.5 Btu/h.ft. °F)(0.3906 ft 2) 0.5 / 12 ft L = = $1.0667 h^{F}/Btu R1 = R5 = R plaster = kA (0.10 Btu/h.ft.^{F})(0.3906 ft 2) 9 / 12 ft L = 288 h^{F}/Btu R2 = R plaster = o kA (0.10 Btu/h.ft.^{3-102} Chapter 3 Steady Heat Conduction 3-140 The change in the R-value of a wood frame wall due to replacing$ fiberwood sheathing in the wall by rigid foam sheathing is to be determined. The two ends of the kiln are made of thin sheet metal covered with 2-cm thick styrofoam. Properties The properties of air at 1 atm and the film temperature of $(Ts + T_{\infty})/2 = (370+30)/2 = 200^{\circ}C$ are (Table A-15) k = 0.03779 W/m. °C $v = 3.455 \times 10^{-5}$ m 2 /s 370 °C D = 3 mm Pr = 0.6974 Analysis The Reynolds number is VD (6 m/s)(0.003 m) Re = $\infty = 521.0 \ v \ 3.455 \times 10 - 5 \ m \ 2/s$ The Nusselt number corresponding this Reynolds number is determined to be 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 \ / |] \ V \infty = 6 \ m/s \ T \infty = 30^{\circ}C \ 4/5 \ 5/8 \ 0.62(521.0) \ 0.5$ (0.6974)1/3[(521.0)] = 0.3 + 1 + |||||1/4|| (282,000/|] 1 + (0.4/0.6974)2/3[Aluminum wire 4/5 = 11.48 Then the heat transfer coefficient and the heat transfer coefficient a $(Ts - T\infty) = (144.6 \text{ W/m 2})^{\circ}(0.009425 \text{ m 2})(370 - 30)^{\circ}C = 463.4 \text{ W} 7-40 \text{ Chapter 7 External Forced Convection 7-50E A fan is blowing air over the entire body of a person. Assumptions 1 Heat transfer is steady since there is no indication of any significant change with time. Analysis The drag force on a cylinder is given by FD1 = C D AN \rho V \infty 2 2$ When the free-stream velocity of the fluid is doubled, the drag force becomes FD 2 = C D A N ρ (2V ∞) 2 = =4 FD1 V ∞ 2 The rate of heat transfer between the fluid and the cylinder is given by Newton's law of cooling. Properties The thermal conductivity is given to be k = 15.1 W/m °C. 3 Thermal properties are constant. Therefore, we will have to rely on energy balances to obtain the finite difference equations. The ratio of heat transfer through the walls with and without windows is to be determined. $^{\circ}$ C)(0.0292 m 2)(25 - 6.5) $^{\circ}$ C = 5.40 W The amount of heat that must be supplied to the drink to raise its temperature to 10 $^{\circ}$ C is m = ρ V = pmr 2 L = (1000 kg / m 3)π (0.03 m) 2 (0125. Also, we can replace the symmetry lines by insulation and utilize the mirrorimage concept when writing the finite difference equations for the interior nodes. The R-values of air spaces are given in Table 3-9. Therefore, the house is losing heat as expected. With this in mind, all solutions are prepared in full detail in a systematic manner, using a word processor with an equation editor. Thermal resistance network and individual resistances are R1 T1 R2 R4 R3 0.02 m L = $= 1.333 \text{ }^{\circ}C/W \text{ R} 2 = \text{Rsteel} = \text{kA} (15 \text{ W/m.}^{\circ}C)(0.01 \text{ m} 2) 0.2 \text{ m} L = = 5.772 \text{ }^{\circ}C/W \text{ R} 3 = \text{Rinsulation} = \text{kA} (0.035 \text{ W/m.}^{\circ}C)(0.99 \text{ m} 2)$ $R1 = R4 = Rsteel = 1\ 1\ 1\ 1\ 1 = + = + \rightarrow Req = 1.083$ °C/W Reqv R 2 R3 1.333 5.772 T2 2 cm 20 cm 2 cm 99 cm R total = R1 + Reqv + R 4 = 0.00133 + 1.0856 °C/W The rate of heat transfer per m2 surface area of the wall is $\Delta T 22$ °C Q& = = 20.26 W R total 1.0857 °C/W 1 cm The total rate of heat transfer through the entire wall is then determined to be $Q_{\&} = (4 \times 6)Q_{\&} = 24(20.26 \text{ W}) = 486.3 \text{ W}$ total If the steel bars were ignored since they constitute only 1% of the wall section, the Requive would simply be equal to the thermal resistance of the insulation, and the heat transfer rate in this case would be $\Delta T \Delta T 22 \text{ °C} Q_{\&} = = = 3.81 \text{ W} \text{ R}$ total R1 + Rinsulation + R4 (0.00133 + 5.772 + 0.00133)°C/W which is mush less than 20.26 W obtained earlier. Analysis We take the entire contents of the tank, water + iron + copper blocks, as the system. Assumptions 1 Heat transfer through the body is given to be transient and two dimensional. °C/(0.5 m 2) 1 1 = = 2.6526 °C / W 2 ho A (10 W / m . The total power rating of the electronic device is to be determined. The relevant physical laws and principles are invoked, and the problem is formulated mathematically. 5-16 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 8). Properties The thermal properties of the cast iron are given to be k = 52 W/m.°C and $\alpha = 1.70 \times 10-5$ m2/s. 4-14a to be Hot dog $\alpha = 0.2059 - 94$ ro = = 0.47 | | 20 - 941 k = = 0.15 Bi hro To $-T \propto$ The thermal diffusivity of the hot dog is determined to be α tro 2 = 0.20 $\rightarrow \alpha = 0.2$ ro 2 (0.2)(0.011 m) 2 = = 2.017 \times 10-7 m 2/s 120 s t (b) The thermal conductivity of the hot dog is determined from $k = \alpha p C p = (2.017 \times 10 - 7 m 2/s)(980 kg/m 3)(3900 J/kg. Using the proper relation for Nusselt number, the average heat transfer coefficient and the hea$ = 7.177 W/m 2. °C L 2.5 m As = wL = (8 m)(2.5 m) = 20 m 2 Q& = hA (T - T) = (7.177 W/m 2.°C)(20 m 2)(120 - 30)°C = 12,919 W = 12.92 kW s \propto s 7-3 Chapter 7 External Forced Convection 7-16 Wind is blowing parallel to the wall of a house. 5-81 Chapter 5 Numerical Methods in Heat Conduction 80 70 Tem perature [C] 60 T0 T1 T2 T3 T4 T5 T6 50 40 30 20 10 0 0 10 20 30 Tim e [hour] 5-82 40 50 Chapter 5 Numerical Methods in Heat Conduction 5-87 Heat conduction through a long L-shaped solid bar with specified boundary conditions is considered. Assumptions 1 Heat transfer from the ball is steady since there is no indication of any change with time. If it is forced to flow in a tube, it is called internal forced convection. Assumptions 1 The water, iron, and copper blocks are incompressible substances with constant specific heats at room temperature. The average rate of heat transfer through each wall, and the amount of money this household will save per heating season by converting the single pane windows to double pane windows are to be determined. Analysis In steady operation, the rate of heat loss from the steam through the concrete floor by conduction must be equal to the rate of heat transfer from the concrete floor to the room by combined convection and radiation, which is determined to be Q& = hA (T - T) = (12 W/m 2 .°C)[(10 m)(5 m)](40 - 25)°C = 9000 W s s ∞ Then the depth the steam pipes should be buried can be determined with the aid of shape factor for this configuration from Table 3-5 to be 9000 W Q& Q& = nSk (T1 - T2) \rightarrow S = = = 10.91 m (per pipe) nk (T1 - T2) 10(0.75 W / m. 53 The upper limit of the time stepThis in the Tmi+1 expression (the primary coefficient) be greater than or equal to zero for all nodes. The total heat transfer from the finned tube is then determined from Q& total, fin = n(Q& fin + Q& unfin) = 250(11.53 + 2.92) = 3613 W Therefore the increase in heat transfer from the tube per meter of its length as a result of the addition of the finst transfer from the finned tube is then determined from Q& total, fin = n(Q& fin + Q& unfin) = 250(11.53 + 2.92) = 3613 W Therefore the increase in heat transfer from the finst transfer from t is Q& increase = Q& total, fin - Q& no fin = 3613 - 974 = 2639 W 3-78 Chapter 3 Steady Heat Conduction 3-111E The handle of a stainless steel spoon partially filled with water at 45°F. The rate of heat transfer from the top surface is Q& top, ave = ho Atop (Tair - Tcan, ave) = (10 W / m 2). Noting that heat transfer is steady and there is no heat generation in the fin and assuming heat transfer to be into the medium from all sides, the energy balance can be expressed as $\Sigma Q \& = 0 \rightarrow \text{kAleft all sides} - \text{T T m} - 1 - \text{Tm} 4 + \text{kAright m} + 1 \text{ m} + \text{hAconv} (\text{T} \circ - \text{Tm}) + \varepsilon \sigma \text{Asurface} [\text{Tsurr} - (\text{Tm} + 273) 4] = 0 \Delta x \Delta x$ Note that heat transfer areas are different for each node in this case, and using geometrical relations, they can be expressed as Aleft = (Height × width) @ m -1 / 2 = 2 w[L - (m + 1 / 2) \Delta x] tan θ As urface = 2 × Length × width = 2 w(Δx / cos θ) h, T ∞ TO • 0 Δx • 1 • θ 2 • 3 • 4 • 5 Tsurr Substituting, $2kw[L - (m - 0.5)\Delta x] \tan \theta Tm - 1 - Tm T - T + 2kw[L - (m + 0.5)\Delta x] \tan \theta m + 1 m \Delta x \Delta x + 2w(\Delta x / \cos \theta) {h(T\infty - Tm) + \epsilon\sigma} [Tsurr - (Tm + 273) 4] = 0$ Dividing each term by $2kwL \tan \theta / \Delta x$ gives $\Delta x \ \Delta x$ 5th node gives the 5th equation, m = 5: $2k \Delta x T - T \Delta x / 2 \Delta x / 2 4 \tan \theta 4 5 + 2h (T_{\infty} - T5) + 2\varepsilon\sigma$ [Tsurr - (T5 + 273) 4] = 0 2 $\Delta x \cos\theta \cos\theta$ Solving the 5 equations above simultaneously for the 5 unknown nodal temperatures gives T1 = 177.0°C, T2 = 174.1°C, T3 = 171.2°C, T4 = 168.4°C, and T5 = 165.5°C (b) The total rate of heat transfer from the fin is simply the sum of the heat transfer from each volume element to the ambient, and for w = 1 m it is determined from 5-11 Chapter 5 Numerical Methods in Heat transfer surface, m [(Tm 4 + 273) 4 - Tsurr] m = 0 Noting that the heat transfer surface area is $w\Delta x / \cos\theta$ for the boundary nodes 0 and 5, and twice as large for the interior nodes 1, 2, 3, and 4, we have $w\Delta x [(T0 - T_{\infty}) + 2(T2 - T_{\infty}) + 2(T2 - T_{\infty}) + 2(T3 - T_{\infty}) + 2(T4 - T_$ 2[(T4 + 273) 4 - Tsurr] + [(T5 + 273) 4 - Tsurr] + [(T5 + 273) 4 - Tsurr]] = 533 W 5-12 Chapter 5 Numerical Methods in Heat Conduction 5-26 "!PROBLEM 5-26" "GIVEN" k=180 [[W/m-C]" L=0.05 "[m]" w=1 "[m]" "T 0=180 [C], parameter to be
varied" T infinity=25 "[C]" h=25 "[W/m^2-C]" T surr=290 "[K]" M=6 epsilon=0.9 tan(theta)=(0.5*b)/L sigma=5.67E-8 "[W/m^2-K^4], Stefan-Boltzmann constant" "ANALYSIS" "(a)" DELTAx=L/(M-1) "Using the finite difference method, the five equations for the temperatures at 5 nodes are determined to be" (1-0.5*DELTAx/L)*(T 0-T 1)+(1-1.5*DELTAx/L)*(T 2T 1)+(h*DELTAx^2)/(k*L*sin(theta))*(T infinityT 1)+ $(epsilon*sigma*DELTAX^2)/(k*L*sin(theta))*(T surr^4-(T 1+273)^4)=0$ "for mode 1" (1-1.5*DELTAx/L)*(T 1-T 2)+(1-2.5*DELTAx/L)*(T 1-T 2)+(1-2.5*DELTAx/L)*(T 3T 2)+(h*DELTAx^2)/(k*L*sin(theta))*(T surr^4-(T 2+273)^4)=0 "for mode 1" (1-1.5*DELTAx/L)*(T 1-T 2)+(1-2.5*DELTAx/L)*(T 4T 3)+(1-3.5*DELTAx/L)*(T 4T 3)+(1-3.5*DELTAx/L)*(T 3T 2)+(h*DELTAx^2)/(k*L*sin(theta))*(T surr^4-(T 2+273)^4)=0 "for mode 1" (1-1.5*DELTAx/L)*(T 4T 3)+(1-3.5*DELTAx/L)*(T 4T 3)+(1-3.5*DELTAx/L)*(T 4T 3)+(1-3.5*DELTAx/L)*(T 5)+(1-3.5*DELTAx/L)*(T 5 $(h*DELTAx^2)/(k*L*sin(theta))*(T infinityT 3)+(epsilon*sigma*DELTAX/2)/(k*L*sin(theta))*(T surr^4-(T 3+273)^4)=0$ "for mode 3" (1-3.5*DELTAx/L)*(T 5T 4)+(1-4.5*DELTAx/L)*(T 5T 4)+(1-4.5*DELTAx/L)*(T 5T 4)+(1-4.5*DELTAX^2)/(k*L*sin(theta))*(T infinityT 3)+(epsilon*sigma*DELTAX^2)/(k*L*sin(theta))*(T surr^4-(T 3+273)^4)=0 "for mode 3" (1-3.5*DELTAx/L)*(T 5T 4)+(1-4.5*DELTAX^2)/(k*L*sin(theta))*(T infinityT 3)+(epsilon*sigma*DELTAX^2)/(k*L*sin(theta))*(T surr^4-(T 3+273)^4)=0 "for mode 3" (1-3.5*DELTAX^2)/(k*L*sin(theta))*(T infinityT 3)+(epsilon*sigma*DELTAX^2)/(k*L*sin(theta))*(T surr^4-(T 3+273)^4)=0 "for mode 3" (1-3.5*DELTAX^2)/(k*L*sin(theta))*(T surr^4-(T 3+273)^4)=0 "for mode 3" (1-3.5*DELTAX^2)/(k*L*sin(theta)) 2*k*DELTAx/2*tan(theta)*(T 4-T 5)/DELTAx)/cos(theta)*(T 0-T infinity)+2*(T 1-T infinity)+2*(T 2-T infinity)+2*(T 3T infinity)+2*(T 4-T infinity)+2*(T 4-T infinity)+2*(T 5)/DELTAx)/cos(theta)*(T 0-T infinity)+2*(T 1-T infinity)+2*(T 1-T infinity)+2*(T 3-T infinity)+2*(T 4-T infT infinity)) D=epsilon*sigma*(w*DELTAx)/cos(theta)*(((T 0+273)^4-T surr^4)+2*((T 1+273)^4-T sur 175 180 185 190 195 200 Ttip [C] 93.51 98.05 102.6 107.1 111.6 116.2 120.7 125.2 129.7 134.2 138.7 143.2 147.7 152.1 156.6 161.1 165.5 170 174.4 178.9 183.3 Ofin [W] 239.8 256.8 274 291.4 309 326.8 344.8 363.1 381.5 400.1 419 438.1 457.5 477.1 496.9 517 537.3 557.9 578.7 599.9 621.2 190 170 T tip [C] 150 130 110 90 100 120 140 160 T 0 [C] 5-14 180 200 Chapter 5 Numerical Methods in Heat Conduction 650 600 550 Q fin [W] 500 450 400 350 300 250 200 100 120 140 160 T 0 [C] 5-15 180 200 Chapter 5 Numerical Methods in Heat Conduction 5-27 A plate is subjected to specified temperature on one side and convection on the other. 6 The surfaces of the cylindrical oven can be treated as plain surfaces since its diameter is greater than 1 m. Assumptions 1 Water is an incompressible substance with a constant specific heat. The amount of heat loss from the steam in 10 h and the amount of saved per year by insulating the steam pipe. become Vmax ST $0.05 = V = (3.8 \text{ m/s}) = 6.552 \text{ m/s} (0.021 \text{ m}) = 9075 \mu 1.825 \times 10 - 5 \text{ kg/m} \cdot \text{s}$ The average Nusselt number is determined using the proper relation from Table 7-2 to be Nu D = 0.35(ST / SL) 0.2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.35(ST / SL) 0.2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 Re OL = 0.021 ST + D / 2 ST + D /Pr 0.36 (Pr/ Prs) 0.25 = 0.35(0.05 / 0.05) 0.2 (9075) 0.6 (0.7309) 0.36 (0.7309 / 0.7132) 0.25 = 74.55 This Nusselt number is applicable to tube banks with NL > 16. Properties of the hot dog are given to be k = 0.44 Btu/h.ft.°F, $\rho = 61.2$ lbm/ft3 Cp = 0.93 Btu/lbm.°F, and $\alpha = 0.0077$ ft2/h. 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 = $N\pi DL = 200\pi (0.016 \text{ m})(1 \text{ m}) = 10.05 \text{ m} 2 \text{ m} \& = m\& i = \rho i V(NT \text{ ST } L) = (1.204 \text{ kg/m} 3)(5.2 \text{ m/s})(10)(0.04 \text{ m})(1 \text{ m}) = 2.504 \text{ kg/s}$ Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become (Ah Te = Ts - (Ts - Ti))(10)(0.04 \text{ m})(1 \text{ m}) = 2.504 \text{ kg/s} Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become (Ah Te = Ts - (Ts - Ti))(10)(0.04 \text{ m})(1 \text{ m}) = 2.504 \text{ kg/s} Then the fluid exit temperature difference, and the rate of heat transfer become (Ah Te = Ts - (Ts - Ti))(10)(0.04 \text{ m})(1 \text{ m}) = 2.504 \text{ kg/s} $exp = 100 - (100 - 20) exp = 100 - (100 - 20) exp = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(100 - 20) - (100 - 49.68)] = 64.01^{\circ}C \ln[(Ts - Ti) - (Ts - Te)] \ln[(Ts - Ti) - (Ts - Ti)] \ln[(Ts - Ti) - ($ staggered tube bank, the friction coefficient corresponding to ReD = 7713 and ST/D = 4/1.6 = 2.5 is, from Fig. Then the mesh Fourier number becomes $\tau = \alpha \Delta t \Delta x 2 = (4.2 \times 10 - 6 \text{ ft } 2 / \text{s})(10 \text{ s}) (0125 \text{ .} \times 10 - 7 \text{ m2} / \text{s})$ (b) The maximum amount of heat transfer is 4 [4] nro 3 = (1100 kg/m 3) | $\pi(0.04 \text{ m}) 3$. 1-55C In a typical house, heat loss through the wall with glass window will be larger since the glass is much thinner than a wall, and its thermal conductivity is higher than the average conductivity is higher than the average conductivity is higher than the average conductivity of a wall. (b) The nodal temperatures under steady conditions are determined by solving the 3 equations above simultaneously with an equation solver to be T0 = 100°C, T1 = 95°C, and T2 = 90°C Discussion This problem can be solved analytically by solving the differential equation as described in Chap. A variable whose value dependently is called a dependent variable (or a function). Analysis First we find the Biot number: Air Bi = hr0 (20 W / m2 . Analysis We consider a volume element of size $\Delta x \times \Delta y \times \Delta z$ centered about a general interior node (m, n, r) in a region in which heat is generated at a constant rate of g & 0 and the thermal conductivity k is variable. We consider a rectangular region in which heat is generated at a constant rate of g & 0 and the thermal conductivity k is variable. Analysis (a) The thermal resistance network in this case consists of two conduction resistance and the contact resistance, and are determined to be Rcontact = Rplate = $1.1 = 0.0447 \,^{\circ}C/W \, hc \, Ac \, (11,400 \, W/m \, 2.^{\circ}C)[\pi \, (0.05 \, m) \, 2/4] \, 015$. Most common fluids such as water, air, gasoline, and oils are Newtonian fluids. The total rate of heat generation in each rod is to be determined. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipes are to be determined. For better accuracy, we could use the inner surface area (D = 19.6 cm) in the calculations. That is, Q& = E&generated = VI = $(110 \text{ V})(3 \text{ A}) = 330 \text{ W} 240^{\circ}\text{C}$ The surface area of the wire is D = 0.2 cm As = $(\pi D) \text{ L} = \pi (0.002 \text{ m})(1.4 \text{ m}) = 0.00880 \text{ m} 2 \text{ L} = 1.4 \text{ Q}$ Air, 20°C The Newton's law of cooling for convection heat transfer is expressed as Q& = hAs (Ts - T ∞) Disregarding any heat transfer by radiation, the convection heat transfer coefficient is determined to be h = 0& 330 W = 170.5 W/m 2. °C As $(T1 - T\infty)(0.00880 m 2)(240 - 20)$ °C Discussion If the temperature of the surrounding surfaces is equal to the air temperature of the surrounding surfaces is equal to the heating in winter. Chapter 5 Numerical Methods in Heat Conduction Chapter 5 NUMERICAL METHODS IN HEAT CONDUCTION Why Numerical Solution methods are limited to highly simplified problems in simple geometries. Considering a 1-m section of the pipe, the amount of heat that must be transferred from the water as it cools from 15 to 0°C is determined to be $m = oV =
o(mr1 2 L) = (1000 \text{ kg/m 3})[\pi (0.03 \text{ m}) 2 (1 \text{ m})] = 2.827 \text{ kg} O \text{ total} = mC p \Delta T = (2.827 \text{ kg})(4.18 \text{ kJ/kg}, °C)(15 - 0)°C = 177.3 \text{ kJ}$ Then the average rate of heat transfer during 60 h becomes Ri $\approx 0.0177.300 \text{ J} O$ ave = total = = 0.821 W $\Delta t (60 \times 3600 \text{ s})$ Rpipe Rinsulation Ro To Ti T1 T2 T3 The individual thermal resistances are $\ln(r2/r1) \ln(0.033/0.03) = 0.0948 \circ C/W 2\pi k$ pipe L 2π (0.16 W/m. Concrete block, lightweight, 100-mm 5a. \circ C)(0.0025 m) tanh aL tanh(12.04 m -1 × 0.03 m) = 0.959 aL 12.04 m-1 × 0.03 m The number of fins, finned and unfinned surface areas, and heat transfer rates from those areas are η fin = n = 1 m2 = 27777 (0.006 m) $(Tb - T\infty) = (35 \text{ W/m 2 o C})(0.86 \text{ m 2})(100 - 30)^{\circ}\text{C} = 2107 \text{ W}$ Then the total heat transfer from the finned plate becomes Q& total, fin = Q& finned + Q& unfinned = 15,700 + 2107 = 1.78 × 104 W = 17.8 W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (1 m)(1 m) = 1 m2 Q& no fin = hAno fin (Tb - T\infty) = (35 m/m 2 o C)(0.86 m 2)(100 - 30)^{\circ}\text{C} = 2107 \text{ W} Then the total heat transfer from the finned plate becomes Q& total, fin = Q& finned + Q& unfinned = 15,700 + 2107 = 1.78 × 104 W = 17.8 W The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (1 m)(1 m) = 1 m2 Q& no fin = hAno fin (Tb - T\infty) = (35 m/m 2 o C)(0.86 m 2)(100 - 30)^{\circ}\text{C} = 2107 \text{ W} The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (1 m)(1 m) = 1 m2 Q& no fin = hAno fin (Tb - T\infty) = (35 m/m 2 o C)(0.86 m 2)(100 - 30)^{\circ}\text{C} = 2107 \text{ W} The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (1 m)(1 m) = 1 m2 Q& no fin = hAno fin (Tb - T\infty) = (35 m/m 2 o C)(0.86 m 2)(100 - 30)^{\circ}\text{C} = 2107 \text{ W} The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (1 m)(1 m) = 1 m2 Q& no fin = hAno fin (Tb - T\infty) = (35 m/m 2 o C)(0.86 m 2)(100 - 30)^{\circ}\text{C} = 2107 \text{ W} The rate of heat transfer if there were no fin attached to the plate would be Ano fin = (1 m)(1 m)(1 m) = 1 m2 Q& no fin = hAno fin (Tb - T\infty) = (35 m/m 2 o C)(100 - 30)^{\circ}\text{C} = 2107 \text{ W} W/m2. F/Btu Proposed Case: Rinsulation = $\ln(r3/r2) \ln(4/2) = 5516$. The difference between a numerical solution or formulation error (also called the truncation or formulation error) which is caused by the approximations used in the formulation of the numerical method, and the round-off error which is caused by the computers' representing a number by using a limited number of significant digits and continuously rounding (or chopping) off the digits it cannot retain. Properties The average density and specific heat of aluminum are given to be $\rho = 2,700 \text{ kg/m3}$ and C p = 0.90 kJ/kg.°C. The thickness of insulation that will protect the water from freezing more than 20% under worst conditions is to be determined. The inner surface of the bottom of the pan is given. Installing such a thick insulation is not practical, however, and thus other freeze protection methods should be considered. We let r3 represent the outer radius of insulation. ° C) Air T = -15°C The

constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, λ 1 = 15708. Analysis Using the available R-values from Tables 3-6, the total amount of heat transfer at the right boundary are to be determined. 15 cm thick insulations, and list the results in the table below. °C(kt) epoxy = 3[(0.26 W / m. Properties The conductivity are given to k = Tsky 0.81 Btu/h.ft.°F and α = 7.4 × 10 -6 ft 2 / s. °C)(15 - 3)°C = 234 W and Q& total = 18,167 + 234 = 1.840 × 10 4 W = 18.40 × kW Ignoring the edge effects of adjoining surfaces, the rate of heat transfer is determined from Atotal = $(12)(12) + 4(12)(6) = 432 \text{ m}^2 \text{ kW} \text{ L} 0.2 \text{ m}$ The percentage error involved in ignoring the effects of the edges then becomes 19.4 - 18.4 % error = $\times 100 = 5.6\%$ 18.4 3-130 The inner and outer surfaces of a long thick-walled concrete duct are maintained at specified temperatures. 5-20 Chapter 5 Numerical Methods in Heat Conduction 5-32 The handle of a stainless steel spoon partially immersed in boiling water loses heat by convection and radiation. Analysis The inner radius of the pipe is r1 = 1.75 in, the outer radius of the pipe is r2 = 2 in, and the outer radii of the existing and proposed insulation layers are r3 = 3 in and 4 in, respectively. 3 All the heat generated in the chips is conducted across the circuit board, and is dissipated from the back side of the board. The mathematical formulation, the variation of temperature, and the rate of evaporation of oxygen are to be determined for steady one-dimensional heat transfer. The coefficient of T4i is smaller in this case, and thus the stability criteria for this problem can be expressed as $-6\ 2\ 1 - 2\tau - 2\tau$ h $\Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = \alpha \Delta t / \Delta x \ge 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = 0$ k $\rightarrow \tau \le 1\ 2(1 + h\Delta x / k)$ since $\tau = 0$ k $\rightarrow \tau \ge 0$ k $\rightarrow \tau$ symmetry about the midpoint. For specified indoor and outdoor temperatures, the rate of heat transfer from the kiln is to be determined. 4 The pressure of air is 1 atm. Analysis (a) The amount of heat this circuit board dissipates during a 10-h period is Q& = (120)(0.12 W) = 14.4 W Chips, $0.12 \text{ W} Q = Q\& \Delta t = (0.0144 \text{ kW})(10 \text{ h}) = 0.144 \text{ kWh} Q\&$ (b) The heat flux on the surface of the circuit board is As = $(0.15 \text{ m})(0.2 \text{ m}) = 0.03 \text{ m} 2 \text{ q} \text{ s} = Q \text{ } 14.4 \text{ W} = 480 \text{ W/m} 2 \text{ As } 0.03 \text{ m} 2 \text{ 15 cm} 20 \text{ cm} 1-17 \text{ An aluminum ball is to be heated from } 80^{\circ}\text{C} \text{ to } 200^{\circ}\text{C}.$ Then the total cost of insulation becomes Insulation Cost = $(\text{Unit cost})(\text{Surface area}) = [(\$10 / \text{cm})(1 \text{ cm}) + \$30 / \text{m} 2](70.69 \text{ m} 2) = \$2828 7-76 \text{ m}^2 \text{ m}$ To Chapter 7 External Forced Convection To determine the thickness of insulation whose cost is equal to annual energy savings, we repeat the calculations above for 2, 3, . 3 The thermal properties of the rib are constant. Analysis (a) Noting that heat transfer is steady and one-dimensional in the radial r direction, the mathematical formulation of this steady and one-dimensional in the radial r direction. problem can be expressed as 1 dr 2 dr and (2 dT) g& |r| + = 0 dr / k (with g& = constant T (r0) = Ts = 80°C (specified surface temperature) kg Ts=80°C r 0 ro dT (0) = 0 (thermal symmetry about the mid point) dr (b) Multiplying both sides of the differential equation by r2 and rearranging gives g& 2 d (2 dT) |r| = -r dr (dr / k Integrating with respect to r gives dT g& r 3 r 2 = - + C1 (a) dr k 3 Applying the boundary condition at the mid point, dT (0) g& $0 \times = - \times 0 + C1 = 0$ B.C. at r = 0: dr 3k Dividing both sides of Eq. (a) by r2 to bring it to a readily integrable form and integrating, dT g& = - r dr 3k g& T (r) = - r 2 + C2 (b) and 6k Applying the other boundary condition at r = r0, g& g& 2 Ts = $-r02 + C2 \rightarrow C2 = Ts + r0$ B. °C/(0.33 × 1 m 2) R4 = Rbrick = Ro = Rconv, 2 1 1 1 1 1 1 = $+ + = + + \rightarrow$ Rmid = 0.81 °C/ W Rmid R3 R4 R5 54.55 Rtotal = Ri + R1 + 2 R2 + Rmid + Ro = 0.303 + 2.33 + 2(0.303) + 0.81 + 0152. 7-22 Chapter 7 External Forced Convection 7-31 Air is blown over an aluminum plate mounted on an array of power transistors. 3 The apparatus possesses thermal Q& symmetry. ° C Air From Table 4-1 we read, for a cylinder, $\lambda 1 = 2.027$ and A1 = 1.517. 2 Convection heat transfer coefficient and emissivity are constant and uniform. ft. 3-102C Increasing the length of a fin decreases its effectiveness. 2 Air is an ideal gas with constant properties. 2 The thermal properties of the brick wall are constant. Also, it can be considered to be two-dimensional since temperature differences (and thus heat transfer) will exist in the radial and axial directions.) 2-9C Yes, the heat flux vector at a point P on an isothermal surface of a medium has to be perpendicular to the surface at that point. This is usually done by forcing the air up which hits the ceiling and moves downward in a gently manner to avoid drafts. The specific heat of beef carcass is given to be 3.14 kJ/kg. C. 2 Heat transfer is one-dimensional since the plate is large relative to its thickness, and there is thermal symmetry about the center plane 3 Thermal conductivity is constant. ••. 3 The environment is at a uniform temperature. 4 Heat transfer coefficients are constant. ••. 3 The environment is at a uniform temperature. regular insulation, and super insulation between the plates. 2-96C During steady one-dimensional heat conductivity varies linearly, the error involved in heat transfer calculation by assuming constant thermal conductivity at the average temperature is (a) none. 4 The average human body can be treated as a 30-cm-diameter cylinder with an exposed surface area of 1.7 m2. 5 The Fourier number is $\tau > 0.2$ so that Air the one-term approximate solutions (or the transient temperature T $\infty = 20$ °C charts) are applicable (this assumption will be verified). (b) The cool air chilling can cause a moisture loss of 1 to 2 percent while water immersion chilling can actually cause moisture absorption of 4 to 15 percent. The center temperature of the hot dog is do be determined by treating hot dog as a finite cylinder. We = 1.6 kW Tcold = 20°C mcold=0.06 kg/s 1-78 Chapter 1 Basics of Heat Transfer 1-137 The glass cover of a flat plate solar collector with specified inner and outer surface temperatures is considered. But the criteria is satisfied, and the proposed additional insulation is justified. There is • n+1 no heat generation in the medium with a specified surface temperature. ° C s-1 = 01195 Assumptions 1 Air as an ideal gas with a constant specific heats at room temperatures stop changing after about 3.8 min. 85°C Resistance Base plate Analysis The nodal spacing is given to be $\Delta x=0.2$ cm. 2 Oil is an incompressible substance with constant properties. - Tambient)(0.475 - 0.0203V + 0.304 V) = 914. Some heat is lost from the tank to the surroundings during the process. The variation of temperature and the rate of heat transfer through the shell are to be determined. The air flow is assumed to be entirely turbulent because of the intense vibrations involved. Therefore, ∂u $/ \partial y >> \partial u / \partial x$. 2-58 Chapter 2 Heat Conduction Equation 2-103 "GIVEN" A=1.5*0.6 "[m^2]" L=0.15 "[m]" "T_1=500 [K], parameter to be varied" T_2=350 "[K]" k_0=25 "[W/m-K]" beta=8.7E-4 "[1/K]" "ANALYSIS" k=k_0*(1+beta*T) T=1/2*(T_1+T_2) Q_dot=k*A*(T_1-T_2)/L T1 [W] 400 425 450 475 500 525 550 575 600 625 650 675 700 Q [W] 9947 15043 20220 25479 30819 36241 41745 47330 52997 58745 64575 70486 76479 80000 20000 40000 20000 40000 30000 20000 10000 0 400 450 500 550 T 1 [K] 2-59 600 650 700 Chapter 2 Heat Conduction Equation Special Topic: Review of Differential equations 2-104C We utilize appropriate simplifying assumptions when
deriving differential equations to obtain an equation that we can deal with and solve. Radiation is expressed by Stefan-Boltzman law as Q& rad = $\varepsilon\sigma As$ (Ts 4 – Tsurr 4) where ε is the surface temperature, As is the surface area, Ts is the surface temperature, Tsurr 4) where ε is the surface area, Ts is the surface temperature, Tsurr 4) where ε is the surface temperature, Tsurr 4) where ε is the surface temperature and σ = 5.67 × 10 – 8 W / m2. K 4 is the StefanBoltzman constant. 2 Heat transfer can be approximated as being one dimensional. 13-38C Flow separation in flow over a cylinder is delayed in turbulent flow because of the extra mixing due to random fluctuations and the transverse motion. Properties The emissivity of a person is given to be $\varepsilon = 0.95$ Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and the floor in both cases are: (a) Summer: Tsurr = (23+273=2964) Q& rad = $\varepsilon\sigma As$ (Ts4 - Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (b) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (b) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.2 \text{ W} - 8 \text{ Tsurr} = 23+273=2964)$ (c) Winter: Tsurr = $(0.95)(5.67 \times 10 = 84.$ 12+273=285 K Qrad 4) Q& rad = $\varepsilon\sigma As$ (Ts4 - Tsurr = (0.95)(5.67 × 10 - 8 W/m 2 .K 4)(1.6 m 2)[(32 + 273) 4 - (285 K) 4]K 4 = 177.2 W Discussion Note that the radiation heat transfer from the person more than doubles in winter. Analysis The critical Reynolds number is given to be Recr = 5×105 . Analysis The heat transfer surface area is $80^{\circ}C$ As = $\pi DL = \pi (0.05 \text{ m})(10 \text{ m}) = 1.571 \text{ m}^2 D = 5 \text{ cm}$ Under steady conditions, the rate of heat transfer by convection is L = 10 m Q Air, 5°C 1-38 120° C Chapter 1 Basics of Heat Transfer 1-79 A hollow spherical iron container is filled with iced water at 0°C. Analysis The boundaries can be expressed analytically as dT (0) -k = q0 At x = 0: dx -k At x = L: dT (L) $-T\infty$] dx Replacing derivatives by differences using values at the closest nodes, the finite differences using values at the closest nodes (nodes 0 and 4) can be expressed as dT dx \cong left, m = 0 T1 - T0 Δx and dT dx \cong right, m = 4 g(x) q0 0 • T4 - T3 Δx Substituting, the finite difference formulation of the boundary nodes become At x = 0: -k T1 - T0 = q0 Δx At x = L: -k T 4 - T3 = h[T4 - T ∞] Δx 5-3 h, T ∞ Δx • 1 • 2 • 3 • 4 Chapter 5 Numerical Methods in Heat Conduction 5-10 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). Since Fourier number. Cement mortar, 0.5 in 4. Properties The thermal conductivities are given to be ksheetrock = 0.10 Btu/h·ft.°F and kinsulation = 0.020 Btu/h.ft.°F. °F / Btu $2\pi k1 L 2\pi$ (8.7 Btu / h.ft. 4-72C The heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since there is no heat transfer in this short cylinder is one-dimensional since the outer radius of plastic insulation is k 0.075 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft.°F = 0.0615 in) h 2.5 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft.°F = 0.0615 in) h 2.5 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft.°F = 0.0615 in) h 2.5 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft.°F = 0.0615 in) h 2.5 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft.°F = 0.0615 in) h 2.5 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft.°F = 0.0615 in) h 2.5 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft 2 .°F Since the outer radius of plastic insulation is k 0.075 Btu/h.ft. 4.72 C The heat transfer in this short cylinder is one-dimensional since the outer radius of plastic insulation is k 0.075 Btu/h.ft. 4.72 C The heat transfer in this short cylinder is one-dimensional since the outer radius of plastic insulation is k 0.075 Btu/h.ft. 4.72 C The heat transfer in the outer radius of plastic insulation is k 0.075 Btu/h.ft. 4.72 C The heat transfer in the outer radius of plastic insulation is k 0.075 Btu/h.ft. 4.72 C The heat transfer in the outer of the wire with insulation is smaller than critical radius of insulation, plastic insulation will increase heat transfer from the wire. ° C L A ΔT (0.01 m 2)(8° C) L A 1-70 The thermal conductivity of a refrigerator door is to be determined by measuring the surface temperatures and heat flux when steady operating conditions are reached. Assumptions 1 Heat transfer through the handle of the spoon is given to be steady and one-dimensional. 6-29C In a boundary layer during steady two-dimensional flow, the velocity component in the normal direction, and thus u >> v, and $\partial v / \partial x$ and $\partial v / \partial y$ are negligible. Analysis The total rate of heat dissipated by the chips is $Q\& = 80 \times (0.06 \text{ W}) = 4.8 \text{ W}$ Then the temperature difference between the front and back surfaces of the board is Chips A = (0.12 m)(0.18 m) = 0.0216 m 2) Q& L (4.8 W)(0.003 m) \Delta T Q& = kA \rightarrow \Delta T = = 0.042^{\circ}C L kA (16 W/m.°C)(0.0216 m 2) Q& Discussion Note that the circuit board is nearly isothermal. 2 Heat transfer is onedimensional since the plate is large relative to its q0 thickness. Properties Assuming the film temperature to be approximately 35° C, the properties of air are evaluated at this temperature to be (Table A-15) k = 0.0265 W/m. °C $v = 1.655 \times 10 \text{ m/s}$ -5.2 Circuit board 15 W 15 cm Air 20°C 5 m/s Pr = 0.7268 Analysis (a) The convection heat transfer coefficient at the leading edge approaches infinity, and thus the surface temperature there must approach the air temperature can be used for air. 3-111 Chapter 3 Steady Heat Conduction Review Problems 3-152E Steam is produced in copper tubes by heat transferred from another fluid condensing outside the tubes at a high temperature. Then the Biot number becomes Bi = hro (65 W/m 2. °C)(0.1 m) = 38.24 k (0.17 W/m.°C) Hot gases T ∞ = 520° C The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, λ 1 = 2.3420 and A1 = 1.5989 Tree Ti = 30° C D = 0.2 m The Fourier number is $\tau = \alpha t r 02 = (1.28 \times 10^{\circ})$ $-7 \text{ m } 2/\text{s}(4 \text{ h} \times 3600 \text{ s/h})$ (0.1 m) 2 = 0.184 which is slightly below 0.2 but close to it. 4 Radiation is accounted for in heat transfer coefficients. The solution function represents a straight line whose slope is C1. 8-11C The number of transfer units NTU is a measure of the heat transfer coefficients. The solution function represents a straight line whose slope is C1. 8-11C The number of transfer units NTU is a measure of the heat transfer system. $\times 10 - 6 \text{ m} 2/\text{s}$. 2-62 A m, C, Ti T=T(t) h T ∞ Chapter 2 Heat Conduction Equation 2-121 A long rectangular bar is initially at a uniform temperature of Ti. The surfaces by convection. Analysis Noting that there is thermal symmetry about the midpoint and convection at the outer surface, the differential equation and the boundary
conditions for this heat conduction problem can be expressed as $1 \partial (2 \partial T) 1 \partial T |r| = r 2 \partial r \sqrt{\partial r} \alpha \partial t \partial T (r 0, t) = 0$ ar $\partial T (r 0, t) - K = h[T (r 0, t) - K =$ $T \propto by$ convection and radiation. 2 Heat transfer can be approximated as being one-dimensional since it is predominantly in the x direction. We will get E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ It can be solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$ and the solved to give E& generation T (t) = $T \propto + hAs(1 + T \propto) = dt mC p mC p$. for t gives 527.3°C for the first case and 69.4°C for the second case, which are practically identical to the results obtained from the approximate analysis. 4 The entire plate is nearly isothermal. ° C) = 15.37 m-1 (237 W / m. 3-51C Two approaches used in development of the thermal resistance network in the xdirection for multi-dimensional problems are (1) to assume any plane wall normal to the x-axis to be isothermal and (2) to assume any plane parallel to the x-axis to be adiabatic. (b) The amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (Amount of heat lost during an 8 hour period and its cost are $Q = Q \& \Delta t = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh}$ Brick Q Cost = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh} Brick Q Cost = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kWh} Brick Q Cost = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kW} Brick Q Cost = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kW} Brick Q Cost = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kW} Brick Q Cost = (0.911 \text{ kW})(8 \text{ h}) = 7.288 \text{ kW} Therefore, the cost of the heat loss through the wall to the home owner that night is \$0.51. The exterior surface temperature of the Trombe wall rises from 0 to 61.5°C in just 6 h because of the solar energy absorbed, but then drops to 13.9°C by next morning 5-80 Chapter 5 Numerical Methods in Heat Conduction as a result of heat loss at night. Therefore, if radiation is disregarded, the heat transfer through the window will be zero. Therefore, T1 = T3 = T7 = T9 and T2 = T4 = T6 = T8, and T1 , T2 , and T3 = T7 = T9 and T2 = T4 = T6 = T8. The thermal conductivity and emissivity are given to be $k = 2.3 \text{ W/m} \cdot \text{C}$ and $\epsilon = 0.7$. Analysis (a) Noting that the upper part of the iron is well insulated and thus the entire heat generated in the resistance wires is transferred to the base plate, the heat flux through the inner surface is determined to be Tsurr q Q& 1500 W q& 0 = 0 = = 100,000 W/m 2 ε Abase 150 × 10 - 4 m 2 h Taking the direction normal to the surface of the wall to be the x T ∞ direction with x = 0 at the left surface, the mathematical formulation of this problem can be expressed as and d 2T = 0 dx 2 dT (0) - k = q & 0 = 80,000 W / m2 dx dT (L) 4 - T & 1 + $\varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T 4] - k = h[T (L) - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 + \varepsilon \sigma [(T + 273) 4 - T & 1 +$ ∞ surr 2 2 dx L x (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. Assumptions 1 The meat slabs can be approximated as very large plane walls of halfthickness L = 5-cm. Analysis We take the x-axis to be the normal direction. 4-11C The lumped system analysis is more likely to be applicable to slender bodies since the characteristic length (ratio of volume to surface area) and thus the Biot number is much smaller for slender bodies. Then using the radiation relation for a surface area) and thus the Biot number is much smaller for slender bodies. rate of radiation heat transfer from the top and side surfaces to the base is determined to be $Q\& = \varepsilon A\sigma$ (T 4 - T 4) rad, base base surr 2 = (0.7)(3 × 3 m)(5.67 × 10 - 8 W/m 2 . K 4)[(1200 K) 4 - (800 K) 4] = 594,400 W 1-73 Black furnace 1200 K Base, 800 K Chapter 1 Basics of Heat Transfer 1-131 A refrigerator consumes 600 W of power when operating, and its motor remains on for 5 min and then off for 15 min periodically. Analysis This rectangular block can physically be formed by by the intersection of two infinite plane walls of thickness 2L = 40 cm (call planes A and B) and an infinite plane walls of thickness 2L = 80 cm (call plane C). The Pr changes with temperature, but not pressure Inside surface, still air R-value, m2.°C/W Between At studs studs 0.044 0.14 0.14 0.23 0.23 3.696 --0.98 0.079 0.12 0.12 1 Total unit thermal resistance of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction of each section, U = 1/R, in W/m2.°C Area fraction
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Analysis The Reynolds number is V L (10 m/s)(0.5 m) Re L = ∞ = 3.201 × 10 5 υ 1.562 × 10 -5 m 2/s which is less than the critical Reynolds number of 5×105. There is a 0.5 in thick sheetrock layer between two adjacent bricks on all four sides, and on both sides of the wall. Thus a large value of Fourier number indicates faster propagation of heat through body. Analysis The convection heat through body. Analysis $^{\circ}C$ (1.8 m 2) (30 - 10) $^{\circ}C$ = 214.4 W (b) h = 8.6V 0.53 = 8.6(1.0 m/s) 0.53 = 8.60 W/m 2 $^{\circ}C$ Q& = hAs (Ts - T $^{\circ}$) = (8.60 W/m 2 $^{\circ}C$) (1.8 m 2) (30 - 10) $^{\circ}C$ = 383.8 W (d) h = 8.6V 0.53 = 8.6(2.0 m/s) 0.53 = 12.42 W/m 2 $^{\circ}C$) (1.8 m 2) (30 - 10) $^{\circ}C$ = 383.8 W (d) h = 8.6V 0.53 = 8.6(2.0 m/s) 0.53 = 12.42 W/m 2 °C Q& = hAs (Ts - T ∞) = (12.42 W/m 2.°C)(1.8 m 2)(30 - 10)°C = 447.0 W 6-2 Air V ∞ T ∞ = 10°C Ts = 30°C Chapter 6 Fundamentals of Convection 6-10 The rate of heat loss from an average man walking in windy air is to be determined at different wind velocities. 2-7C Heat transfer to a roast beef in an oven would be transient since the temperature at any point within the roast will change with time during cooking. Properties The thermal conductivities are given to be k = 0.026 W/m·°C for glass. The mathematical formulation, the variation of temperature in the plate, and the inner surface temperature are to be determined for steady one-dimensional heat transfer. Substituting, the fractions of heat conducted along the copper and epoxy layers as well as the effective thermal conductivity of the board are determined to be (kt) copper = $(386 \text{ W} / \text{m}. \text{Therefore } Q_{\&} = 20 - 1.4 = 18.6 \text{ W}$ conv total rad The total heat transfer surface area for this finned surface is As,finned = $(2 \times 7)(0.1 \text{ m})(0.005 \text{$ m) = 0.007 m 2 As, unfinned = $(0.1 \text{ m})(0.062 \text{ m}) - 7 \times (0.002 \text{ m})(0.1 \text{ m}) = 0.0048 \text{ m } 2 \text{ As}$, total = As,finned + As, unfinned = 0.007 m 2 + 0.0048 m 2 = 0.0118 m 2 The convection heat transfer coefficient can be determined from Newton's law of cooling relation for a finned surface. 2-65 Chapter 2 Heat Conduction Equation 2-125 A steam pipe is subjected to convection on both the inner and outer surfaces. 5-71C For transient two-dimensional heat conduction in a rectangular region with insulation or specified temperature boundary conditions, the stability criteria for the explicit method can be expressed in its simplest form as $\tau = 1 \alpha \Delta t \le 2.4$ (Δx) which is identical to the one for the interior nodes. Properties The properties of the turkey are given to be k = 0.26 Btu/h.ft.°F, $\rho = 75$ lbm/ft3, Cp = 0.98 Btu/lbm.°F, and $\alpha = 0.0035$ ft2/h. 3-70 Chapter 3 Steady Heat Conduction 3-89E An electrical wire is covered with 0.02-in thick plastic insulation. °C)(0.0275 m) = = 64.2 k (0.6 W / m. If the results obtained by halving the mesh size do not differ significantly from the results obtained with the full mesh size, we conclude that the discretization error is at an acceptable level. The temperature 2 mm from the center line (r = 0.002 m) is determined by substituting the known quantities to be T (r) = Ts + T (0.002 m) = m) 2] = 212.8° C 4k 4 × (8 W / m. The problem can be minimized by adding chloride to the water. Assumptions 1 The flow is steady and incompressible. 3-138C The roof of a house whose attic space is ventilated effectively so that the air temperature in the attic is the same as the ambient air temperature at all times will still have an effect on heat transfer through the ceiling since the roof in this case will act as a radiation shield, and reduce heat transfer by radiation. However, the lumped system Milk analysis is still applicable since the milk is stirred constantly, so that 3°C its temperature remains uniform at all times. The thickness of the insulation needed to reduce the heat loss by 95 percent and the thickness of the insulation needed to reduce outer surface temperature to 40° C are to be determined. The same is true for turbulent flow. Properties The thermal properties of the soil are given to be k = 0.9 W/m.°C and α = 1.6×10-5 m2/s. Analysis It is given that D = 0.01 m, SL = 0.04 m and ST = 0.03 m, and V = 0.8 m/s. Analysis It is given that D = 0.01 m, SL = 0.04 m and ST = 0.03 m/s. Analysis It is given that D = 0.01 m, SL = 0.04 m and ST = 0.03 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. Analysis It is given that D = 0.01 m/s. An The surface area of the wall and the individual resistances are A = $(6 \text{ m}) \times (2.8 \text{ m}) = 16.8 \text{ m } 2 \text{ R} = \text{R} \text{covering} = \text{R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3
\text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ R} = 0.00165 \text{ °C/W k } 3 \text{ A} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ C} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2) \text{ C} (1.4 \text{ W/m.°C})(16.8 \text{ m } 2$ 0.00350° C/W T1 2 h2 A (17 W/m .°C)(16.8 m 2) T ∞ 2 R1 R2 R3 R total = R1 + R2 + R3 + Rconv, 2 = 0.00165 + 0.0085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.00085 + 0.000 $(0.00165^{\circ}C/W) = 1.1 \circ C \Delta T brick = Q\& R brick = (665.8 W)(0.00350^{\circ}C/W) = 2.3 \circ C 3-129 \text{ Ro Chapter 3 Steady Heat Conduction 3-169 An insulation is to be added to a wall to decrease the heat loss by 85%. 4-45 to be t$ =180 min. For specified heat transfer coefficients and 0.01-in thick scale build up on the inner surface, the length of the velocity of the air at the exit are to be determined. Therefore, Q& total = Q& conv + Q& rad = 1000 W where Q& conv = hAs $\Delta T = (35 \text{ W/m } 2 \cdot \text{K} 4)[\text{Ts} - 293 \text{ K}] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K}) 4] = 0.06804 \times 10 - 8 \text{ [Ts} 4 - (293\text{ K$ K) + 0.06804 ×10 -8 [Ts4 - (293 K) 4] Iron 1000 W Solving by trial and error gives Ts = 947 K = 674 o C Discussion We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 947 K. 2 Convection heat transfer is negligible. The cost of insulation increases roughly linearly with thickness while the cost of heat lost decreases exponentially. 3-10C Once the rate of heat transfer Q& is known, the temperature drop across that layer, ΔT layer 3-1 layer Chapter 3 Steady Heat Conduction 3-11C The temperature of each surface in dt can be expressed as (Heat transfer from the body) (The decrease in the energy || || = || || during dt / of the body during dt / or hAs $(T - T\infty)$ since T = constant, the equation above can be rearranged as hAs d $(T - T\infty) = -dt T - T\infty \rho VC p$ which is the desired differential equation. Substituting these values, the nodal temperatures in the pond after $4 \times (60/15) = 16$ time steps (4 h) are determined to be T0 = 18.3°C, T1 = 16.9°C, T2 = 15.4°C, T3 = 15.3°C, and T4 = 20.2°C. Properties of air at 1 atm pressure and the free stream temperature of 25°C are (Table A-15) k = 0.02551 W/m.°C v = 1.562 \times 10-500 m 2 /s Lamp 100 W $\varepsilon = 0.9$ Air V $\infty = 2$ m/s T $\infty = 25^{\circ}$ C $\mu \infty = 1.849 \times 10 - 5$ kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ
 s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg/m.s μ s , @ 100°C = 2.181 × 10 - 5 kg $0.06 \text{ Re } 2/3 \text{ Pr } 0.4 || \propto k (\mu s 1/4) || / [] (1.849 \times 10 - 5 = 2 + 0.4(1.280 \times 10 4) 0.5 + 0.06(1.280 \times 10 4) 2/3 (0.7296) 0.4 || -5 (2.181 \times 10 \text{ The heat transfer coefficient is } k 0.02551 \text{ W/m} 2.^{\circ}C \text{ D } 0.1 \text{ m Noting that } 90\% \text{ of electrical energy is converted to heat, } Q&= (0.90)(100 \text{ W}) = 90 \text{ W } 1/4) || / = 68.06$ The bulb loses heat by both convection and radiation. Analysis The inner and the outer surface areas of the duct per unit length and the individual thermal resistances are A1 = $4a1 L = 4(0.22 m)(1 m) = 0.88 m 2 Ri A2 = 4a 2 L = 4(0.25 m)(1 m) = 1.0 m 2 T \infty 1 1 1 = 0.01515 °C/W 2 h1 A (75 W/m .°C)(0.88 m 2) 0.015 m L = = 0.00007 °C/W kA$ / 2] m 2 Ri = Ralum 1 1 = = 0.12500°C/W h2 A (8 W/m 2 .°C)(1.0 m 2) = Ri + Ralum + Ro = 0.01515 + 0.00007 + 0.12500 = 0.14022 °C/W Ro = R total The rate of heat loss from the air inside the duct is $T - T \propto 1$ (33 – 12)°C Q& = $\propto 2$ = = 149.8 W R total 0.14022°C/W For a temper should gain heat at a rate of Q& total = m& C p $\Delta T = (0.8 \text{ kg/s})(1006 \text{ J/kg.}^{\circ}C)(1^{\circ}C) = 804 \text{ W}$ Then the maximum length of the duct becomes Q& 804 W L = total = = 5.37 m 149.8 W Q& 3-131 Ralum Ro T $\infty 2$ Chapter 3 Steady Heat Conduction 3-171 Heat transfer through a window is considered. 3 The house is maintained at a constant temperature and pressure at all times. Properties of the potato are given to be k = 0.6 W/m.°C, $\rho = 1100$ kg/m3, Cp = 3.9 kJ/kg.°C, and $\alpha = 1.4 \times 10-7$ m2/s. The unknown nodal temperatures and the rate of heat loss from the bottom surface through a 1-m long section are to be determined. Using the Nusselt number relation for combined laminar and turbulent flow, the average heat transfer coefficient is determined to be hL Nu = $(0.037 \text{ Re } 0.8 - 871)(0.7282)1/3 = 1378 \text{ k} \text{ k} 0.02588 \text{ W/m} 2 \cdot CL 2 \text{ m}$ Then the rate of heat loss from the collector by convection is Q& = hA (T - T) = $(17.83 \text{ W/m} 2 \cdot C)(2 \times 1.2 \text{ m} 2)$ $(35 - 25)^{\circ}C = 427.9 \text{ W conv} \propto s \text{ s}$ The rate of heat loss from the collector by radiation is = $\varepsilon A \sigma (T 4 - T 4) Q \& s \text{ rad s surr } 2 [= (0.90)(2 \times 1.2 \text{ m})(5.67 \times 10 - 8 \text{ W/m } 2.^{\circ}C) (35 + 273 \text{ K}) 4 = 741.2 \text{ W} and Q \& \text{ total} = Q \& \text{ conv} + Q \& \text{ rad} = 427.9 + 741.2 = 1169 \text{ W} (b)$ The net rate of heat transferred to the water is Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & - Q & = Q & = Q & - Q & = Q & = Q & - Q & = Q & = Q & - Q & = Q & = Q & - Q & = Q & = Q & - Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q & = Q α AI - Q& net in out out = (0.88)(2 × 1.2 m)(700 W/m 2) - 1169 W 2 η collector = 1478 - 1169 = 309 W Q& 309 W = net = 0.209 & Qin 1478 W (c) The temperature rise of water as it flows through the collector is Q& 309.4 W Q& net = m& C p $\Delta T \rightarrow \Delta T$ = net = 4.44 °C m& C p (1/60 kg/s)(4180 J/kg.°C) 7-19] Chapter 7 External Forced Convection 7-28 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. Assumptions 1 The temperature of the metal object changes uniformly with time during cooling so that T = T(t). energies 1 solid + mhif + mC (T2 - 32°F) liquid] ice ice 25°F 160lbm + [mC (T2 - T1)] water = 0 Substituting, (160lbm $[(0.50Btu/lbm \cdot o F)(32 - 25)^{\circ}F + 143.5Btu/lbm + (1.0Btu/lbm \cdot ^{\circ}F)(T2 - 32)^{\circ}F] + (2000lbm)(1.0Btu/lbm \cdot ^{\circ}F)(T2 - 70)^{\circ}F = 0$ WATER 1 ton It gives T2 = 56.3°F which is the final equilibrium temperature in the tank. The Fourier number is $\tau = \alpha t$ ro 2 = $(0135 \cdot 4.64 \text{ Chapter 4 Transient Heat Conduction 4-73 A short cylinder is allowed to cool in }$ atmospheric air. 5-113 Chapter 5 Numerical Methods in Heat Conduction 5-117 A hot surface is to be cooled by aluminum pin fins. 5b Construction 6 5a 4 1 2 7 1. The implicit transient finite difference formulation of the problem using the energy balance approach method is to be determined. m2) Rtotal = 5R fabric + 4 Rair + Ro = $5 \times 0.00152 + 4$ $\times 0.0524 + 0.0364 = 0.2536$ °C / W Rcot ton = R1 = R3 = R5 = R7 = R9 = and T - T [(28 - (-5)]°C = 130 W Q& = s1 ∞ 2 = Rtotal 0.2536 °C/W If the jacket is made of a single layer of 0.5 mm thick cotton fabric, the rate of heat transfer will be T - T Ts1 - T ∞ 2 [(28 - (-5)] °C = 750 W Q& = s1 ∞ 2 = 5 × R fabric + Ro (5 × 0.00152 + 0.0364) °C / W Rtotal The thickness of a wool fabric for that case can be determined from Rtotal = Rwool + Ro = fabric 0.2536 ° C / W = L 1 + kA hA L (0.035 W / m. Q& Analysis For each sample we have Q& = 35 / 2 = 17.5 W A = (01 . Properties The thermal conductivity, density, and specific heat of the aluminum balls are k = 137 Btu/h.ft.°F, $\rho = 168$ lbm/ft3, and Cp = 0.216 Btu/lbm.°F (Table A-3E). 2 The thermal properties of the ice block are constant. Properties The thermal conductivities are given to be k = 8.7 Btu/h·ft.°F for steel and k = 0.020 Btu/h·ft.°F for fiberglass insulation. Assumptions Heat is transferred uniformly from all surfaces. energies We, in – Qout = $\Delta U = 0$ We, in = Qout since $\Delta U = 0$ $mCv\Delta T = 0$ for isothermal processes of ideal gases. Assumptions 1 The beef carcass can be approximated as a cylinder with insulated top and base surfaces having a radius of r0 = 12 cm and a height of H = 1.4 m. W) = 450 W 0.33 m2 2 3-27 R7 T ∞ 2 Chapter 3 Steady Heat Conduction 3-53 "GIVEN" A=4*6 "[m^2]" L brick=0.18 "[m]" L plaster_center=0.18 "[m]" L plaster_side=0.02 "[m]" "L foam=2 [cm], parameter to be varied" k brick=0.72 "[W/m-C]" k foam=0.026 "[W/m-C m))/(k_foam*A_1) "L_foam is in cm" R_plaster_side=L_plaster_side=L_plaster_side/(k_plaster*A_2) A_2=0.30*1 "[m^2]" R_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster_center=L_plaster (T_infinity_1-T_infinity_2)/R_total Q_dot_total=Q_dot*A/A_1 Lfoam [cm] 1 2 3 4 5 6 7 8 9 10 Qtotal [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 500 400 300 200 100 1 2 3 4 5 6 7 L foam [cm] 1 2 3 4 5 6 7 L foam [cm] 1 2 3 4 5 6 7 R 9 10 Qtotal [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q
total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 3-28 Chapter 3 Steady Heat Conduction 700 600 Q total [W] 623.1 450.2 352.4 289.5 245.7 213.4 188.6 168.9 153 139.8 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 140.2 1 of 10-cm thick wood studs or with pairs of 5-cm thick wood studs nailed to each other. m 018 = 54.55 ° C / W ho A (0.22 W / m. 7-27a, f = 0.22. The rate of heat loss from the air to the cold environment is to be determined. \times 10 -6 m 2 / s , $\Delta x = 0.02$ m, $T \infty = 30^{\circ}$ C, $T 0 = 120^{\circ}$ C ho = 35 W/m2.°C, m, k = 237 W/m.°C, $\alpha = k / \rho C = 971$ and $\Delta t = 1$ s. 6 The local atmospheric pressure is 1 atm. 7-46 Chapter 7 External Forced Convection 7-55 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The mechanical power wasted is equal to the rate of heat transfer, W& mech = Q& = 837 W = -(0.0377 m 2) 6-14 Chapter 6 Fundamentals of Convection 7-55 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. 6-41 "PROBLEM 6-41" "GIVEN" D=0.06 "[m]" "N dot=3000 rpm, parameter to be varied" L bearing=0.20 "[m]" T 0=50 "[C]" "PROPERTIES" k=0.17 "[W/m-K]" mu=0.05 "[N-s/m^2]" "ANALYSIS" Vel=pi*D*N dot*Convert(1/min, 1/s) A=pi*D*L bearing T max=T 0+(mu*Vel^2)/(8*k) Q dot=A*(mu*Vel^2)/(2*L) W dot mech=Q dot N [rpm] 0 250 500 750 1000 1250 1500 1750 2000 2250 2500 2750 3000 3250 3500 3750 4000 4250 4500 4750 5000 Wmech [W] 0 2.907 11.63 26.16 46.51 72.67 104.7 142.4 186 235.5 290.7 351.7 418.6 491.3 569.8 654.1 744.2 840.1 941.9 1049 1163 6-15 Chapter 6 Fundamentals of Convection 1200 1000 W m ech [W] 0 800 600 400 200 0 0 1000 2000 3000 N [rpm] 6-16 4000 5000 Chapter 6 Fundamentals of Convection 6-42 A shaft rotating in a bearing is considered. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined for two cases. The overall effectiveness of a finned surface is defined as the ratio of the total heat transfer from the finned surface to the heat transfer from the same surface if there were no fins. Radiation between two surfaces is inversely proportional to the number of sheets used and thus heat loss by radiation will be very low by using this highly reflective sheets. 2 There is no heat generation in the chimney. For example, the consideration of the variation of thermal conductivity with temperature, the variation of the heat transfer coefficient over the surface, or the radiation heat transfer on the surface and interface temperatures of a resistance wire covered with a plastic layer are to be determined. The time of annealing and the total rate of heat transfer from the balls to the ambient air are to be determined. This relation is developed using relations for laminar flow over a flat plate, but it is also applicable approximately for turbulent flow over a flat plate. and T4 are the only 3 unknown nodal temperatures. Then the pressure drop across the tube bank becomes $\Delta P = N L f\chi 2 \rho V max (1.145 kg/m 3)(7.647 m/s) 2 = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 (1N | 1 kg \cdot m/s 2 \rangle = 20(0.20)(1) 2 (1N | 1 kg \cdot$ $2257 \text{ kJ/kg} \cdot ^{\circ}\text{C}$ Discussion The arithmetic mean fluid temperature is (Ti + Te)/2 = (20 + 43.4)/2 = 31.7^{\circ}\text{C}, which is fairly close to the assumed value of 35°C . $e - (2.027) 2 \tau \rightarrow \tau = 0.456 37 - (-6)$ Beef 37°C which is greater than 0.2 and thus the one-term solution is applicable. r1 where T (r1) = T1 to any r where T (r) = T, we get $Q\&\int r r1 dr = -4\pi r2 \int Tk(T) dT T1$ Substituting the integrations gives $(11)Q\&|| - || = -4\pi k 0 [(T - T1) + \beta (T 2 - T12)/2] (r1 r / Substituting the Q& expression from part (a) and rearranging give T2 + 2 <math>\beta$ 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. 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 R 1 + R 2 + 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& = $\infty 1 \propto 2$ = = 114 W R1 R2 Ri Rtotal 0.2539°C/W T $\infty 1$ The inner surface temperature of the window glass can be determined from T – T Q& = $\infty 1 \propto 2$ – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) =
19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rconv, 1 = 24 o C – (114 W)(0.0417°C/W) = 19.2°C Rc 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. \times 10 - 6 m2 / s)(60 min \times 60 s / min) (0.025 m) 2 { 2 T (0,0,0, t) - 500 = (1.0580)e - (0.5932) (6.624 > 0.2 } = 0.00109 \rightarrow T (0,0,0, t) = 500^{\circ}C 3 Note that $\tau > 0.2$ 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 $(30 \text{ W/m} \cdot ^{\circ}\text{C}) \text{ T}(0) = \text{T} 2 + \text{Therefore, there is a temperature difference of almost 300°C between the two sides of the plate. Properties The thermal conductivities are given to be k = 0.61 \text{ W/m} \cdot ^{\circ}\text{C}$ for glass wool insulation Analysis The inner and the outer surface areas of the insulated pipe per unit length are Ai = π Di 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 2 π (5 m) 2 π L = 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 $\Delta P = N L f\chi 2 \rho Vmax (1.204 kg/m 3)(6.552 m/s) 2 = 8(0.34)(1) 2 2 (1N | 1 kg \cdot m/s 2)$ | = 70.3 Pa | Discussion The arithmetic mean fluid temperature is (Ti + Te)/2 = (15 + 28.3)/2 = 21.7^{\circ}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) = $-1\beta \pm 1\beta 2 - 2k$ ave $\ln(r/r1) 2(T1 - T2) + T12 + T1\beta k 0 \ln(r2/r1)\beta$ 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 $k = 0.50 \text{ W/m} \cdot \text{C}$ and $\alpha = 0.13 \times 10-6 \text{ m}^2/\text{s}$. 6-49 A flat plate is subjected to air flow, and the drag force acting on it is measured.) + (18. ° C)(0.0216 m 2) 1 1 = = = 0.9259 ° C / W hA (50 W / m^2) \cdot \text{C}. dimensionless $\alpha \Delta t$ the last equation reduces to mesh Fourier number $\tau = (\Delta x) 2 \text{ Tmi} + 1 - \text{Tmi} h\Delta x T \infty + 0 = k k \tau$ Discussion We note that setting Tmi + 1 = Tmi gives the steady finite difference formulation. 5 The given time step $\Delta t = 15 \text{ min}$ is less than the critical time step so that the stability criteria is satisfied. The velocity and the surface area are $(1 \text{ min}) V = \pi DN \& = \pi (0.06 \text{ m})(3000 \text{ rev/min}) = 9.425 \text{ m/s} (60 \text{ s}) A = \pi DL bearing = \pi (0.06 \text{ m})(0.20 \text{ m}) = 0.0377 \text{ m} 2$ The maximum temperature is Tmax = T (L) = T1 + $\mu V (|1 - 1\rangle) = T1 + \mu V (|1 - 1\rangle) = T1 + \mu$ $|= 63.1^{\circ}C 2(0.17 \text{ W/m} \cdot ^{\circ}C) \langle 1 \text{ N} \cdot \text{m/s} \rangle$ (b) The rate of heat transfer to the bearing is dT μ V 2 μ V 2 (1 - 0) = - A = -kA Q& 0 = -kA dy y = 0 kL L = 50^{\circ}C + (0.05 \text{ N} \cdot \text{s/m} 2)(9.425 \text{ m/s}) 2 (1 \text{ N} \cdot \text{m/s}) The rate of heat transfer to the shaft is zero. h° F / Btu Rtotal = Rtowel + Rconv = 13473 T T - Q (250 - 70) ° F 50.8 T + (0.05 \text{ N} \cdot \text{s/m} 2)(9.425 \text{ m/s}) 2 (1 \text{ N} \cdot \text{m/s}) 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 of air at the film temperature of (35 + 25) / 2 = 30 °C are (Table A-15) V ∞ = 30 km/h $T \propto = 25^{\circ}C \ k = 0.02588 \ W/m.^{\circ}C \ Tsky = -40^{\circ}C \ 700 \ W/m2 \ v = 1.608 \times 10 \ -5 \ m \ 2 \ /s \ Solar \ collector \ Pr = 0.7282 \ Ts = 35^{\circ}C \ Analysis (a)$ number (5 × 105). 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 tank and the internal convection resistance are negligible. 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 $k = 0.78 \text{ W/m} \cdot \text{°C}$. The Biot numbers, the corresponding constants, and the Fourier numbers are hL (13 W/m 2 . °C)(0.02 m) = = 0.1171 $\rightarrow \lambda 1 = 0.3318$ 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 = $N\pi DL = 128\pi(0.021 \text{ m})(1 \text{ m}) = 8.445 \text{ m} 2 \text{ m} \& = m \& i = \rho i V(NT ST L) = (0.6158 \text{ kg/m 3})(4.5 \text{ m/s})(8)(0.08 \text{ m})(1 \text{ m}) = 1.773 \text{ kg/s}$ 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 \& Cp(\Delta Tln = 22)) $I = 80 - (80 - 300) \exp \left[-(8.445 \text{ m})(73.2 \text{ W/m} \cdot \text{°C}) \right] = 237.0^{\circ}\text{C} \left[(1.773 \text{ kg/s})(1033 \text{ J/kg} \cdot \text{°C}) \right] h[(80 - 300) - (80 - 237)] - (7s - \text{Te}) (80 - 300) - (80 - 237)] = 186.7^{\circ}\text{C} \ln[(\text{Ts} - \text{Ti}) - (\text{Ts} - \text{Te})] \ln[(80 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] = 186.7^{\circ}\text{C} \ln[(\text{Ts} - \text{Ti}) - (\text{Ts} - \text{Te})] \ln[(80 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 237)] - (83 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 300) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30) - (80 - 30)$ 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, right + Q& cond, bottom + G& element = Δt for the steady case. 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 $\Delta m CV = 0$ and $\Delta E CV = 0$. 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 $\Delta x = \Delta x = l = 0.1$ m, and all
nodes are boundary nodes. 2 Heat conduction in the radial direction because of the symmetry about the center line. The numerical solution methods are based on replacing the differential equations. m2 Ao = π Do L = π (0.046 m)(15 m) = 2.168 m Ri 2 Rpipe Ro T \propto 1 T \approx 2 1 1 = 0.0040 °C/W hi Ai (120 W/m 2 .°C)(1.885 m 2) ln(r2 / r1) ln(2.3 / 2) = = 0.0000038 °C/W 2nkL 2\pi(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 $M = L / \Delta x + 1 = 0.03/0.01 + 1 = 4.210 \text{ kJ/min Immersion chilling}$, 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) $k = (350 \times 280 \text{ kg/carcass})/(12h \times 3600 \text{ s}) = 2.27 \text{ kg/s}$ 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 $16^{\circ}C$ at a rate of 2.27 kg/s, and is determined to be Q& beef = ($m\& C p \Delta T$) beef = (2.27 kg/s)($3.14 kJ/kg^{\circ} C$)(35 - 16)° 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 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 -2.2°C to 0.5°C as a result. 3-96C Fins enhance heat transfer from a surface by increasing 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 = $2(1.335 \text{ W/m } 2 \cdot ^{\circ}\text{C})(0.7296) 2 / 3 2hPr 2/3 = 0.0001512 \rho V C p (1.184 kg/m 3)(12 m/s)(1007 J/kg \cdot ^{\circ}\text{C}) 6-26 \text{ Chapter 6 Fundamentals of Convection 6-52 The windshield of a car is subjected to parallel winds. Analysis The total surface area of the refrigerator where heat transfer takes place is Atotal = 2 (18)$ 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 = $\rho V = (970 \text{ kg} / \text{m 3})[\pi (0.075 \text{ m}) 2 (0.015 \text{ m})] = 0.257 \text{ kg Qtotal}$, steak = Qsensible + Qlatent = $(mC\Delta T)$ 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 H/m.°C) Bi = hr0 $W/m 2 \cdot C(0.075 m) = 0.0254 \rightarrow \lambda 1 = 0.2217$ and $A1 = 1.0063 k 236 W/m \cdot C \rightarrow \lambda 1 = 0.1811$ and A1 = 1.0056 Noting that $\tau = \alpha t / L2$ and assuming $\tau > 0.2$ 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 $\theta(0,0,t)$ block = $\theta(0,t)$ wall $\theta(0, t) \operatorname{cyl} = \langle | A1 e^{-\lambda 1 \tau} | \langle | A1$ conductivity and diffusivity are given to be k = 0.48 Btu/h.ft.°F and $\alpha = 4.2 \times 10 - 6$ ft 2 / s. Then the number of nodes M becomes M = L 0. Here T 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, and another difference term for the z direction as Tm -1, n, j -2Tm, n, j + Tm, n +1, j Tm, n, j + Tm, n +1, j Tm, n, j + Tm, n +1, j Tm, n, j + Tm +1, n, j + Tm +1, n, j + Tm, n +1, j +1, m +-1 - 2Tm, n, j + Tm, n, j + Tm, 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 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 $\Delta x = \Delta x = 1 = 0.05$ m, and the general finite difference form of an interior node for steady two-dimensional heat conduction is expressed as • • • • g& l 2 Tleft + $Ttop + Tright + Tbottom - 4Tnode + node = 0 k 350 260 305 where \bullet \bullet 200^{\circ}C \bullet g\& node | 2 g\& 0 | 2 (8 \times 10 6 W/m 3) (0.05 m) 2 g = = 93.5^{\circ}C Insulated k k 214 W/m \cdot \circ C 3 290 2 \bullet \bullet \bullet \bullet The finite difference equations for boundary nodes are obtained by 5 cm applying an energy balance on the volume elements and taking the 325 1 direction of$ all heat transfers to be towards the node under • 240 • consideration: Convection 2 290 - T1 240 - T1 325 - T1 h, q&T1 + kl + k + hl (T $\infty - T1$) + 0 $\infty = 0$ Node 2 (interior): 350 + 290 + 325 + 290 - 4T2 + =0 k q&l 2 Node 2 (interior): 260 + 290 + 240 + 200 - 4T3 + 0 = 0 k k = 45 W/m.°C, h = 50 $W/m 2 \cdot C$, $g\& = 8 \times 10.6 W/m 3$, $T = 280.9^{\circ}C$, $T = 330.8^{\circ}C$, $T = 330.8^$ 1 m)[(200 - 20)/2 + (240 - 20) + (280.9 - 20) + (280.9 - 20) + (325 - 20)/2]°C = 1808 W 5-38 Chapter 5 Numerical Methods in Heat transfer. °F) Water 202°F The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, Hot dog λ 1 = 1.5421 and A1 = 1.2728 r x The Fourier number is $\tau = \alpha t L_2 = (0.0077 \text{ ft } 2 / h)(5 / 60 \text{ h})(2.5 / 12 \text{ ft}) 2 = 0.015 < 0.2$ (Be cautious!) Then the dimensionless temperature at the center of the plane wall is determined from θ o, wall = 2 2 T0 - T ∞ = A1e $-\lambda 1 \tau = (1.2728)e^{-(1.5421)}(0.015) \approx 1 \text{ Ti} - T\infty$ 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. W/m.°C and $\alpha = 0.15 \times 10^{-6}$ m2/s. The density and specific heat of water are to be $\rho = 1000$ kg/m3 and Cp = 4.18 kJ/kg.°C (Table A-15). Discussion Heat 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 + $\tau q \& 0$ (Δx) 2 / k Node 0 (insulation): T1i +1 = τ (T0i + T2i) + (1 - 2 τ) T1i + $\tau q \& 1$ (Δx) 2 / k Node 2 (interior): T2i + $\tau q \& 2$ (Δx) 2 / k Node 2 (interior): T3i + $\tau q \& 3$ (Δx) 2 / k i + 1 i T3i - T4i $\Delta x T4 - T4 + \tau q \& 4$ (Δx) 2 / k = $\rho C \Delta x 2 \Delta t - 62$. 2 There is no heat generation within the damn. The individual thermal resistances are Ao = π Do 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 = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m
2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T ∞ 2 (55 - 27) r0.27 °C/W R foam = 2\pi kL 2\pi (0.03 W/m 2.°C)(2 m) Ro = Rfoam Ro Tw T $C Q \& = w = 70 W Rtotal 0.40 \circ C / W The amount and cost of heat loss per year are Q = Q \& \Delta t = (0.07 kW)(365 \times 24 h / yr) = 613.2 kWh / yr Cost of Energy = (Amount of energy)(Unit cost) = (613.2 kWh) = $49.056 f = $49.$ resistances becomes Ao = π Do 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 side) since they reduce radiation heat transfer between the ceiling side) since they reduce radiation heat transfer between the ceiling and the roof considerably. Q& 20 W \rightarrow h = = = 48.43 W/m 2 .°C Q& = η As (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. spect 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 $\lambda 1 = 1.5421$ and A1 = 1.2728 r x The Fourier number is $\tau = \alpha t L$ = (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 $-\lambda 1 \tau$ = (1.2728)e $-(1.5421)(0.015) \approx 1 \text{ Ti} - T \infty$ We repeat the same calculations for the long cylinder, Bi = hro (120 \text{ Btu / h.ft } 2 \cdot 7-27a, f = 0.27.6 \text{ The thermal}) contact resistances at the copper-epoxy interfaces are negligible. 4-102 Chapter 4 Transient Heat 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) $(4.3) \rightarrow h = = 22.5 \text{ W/m } 2 \cdot \text{°C k ro} (0.08603 \text{ m}) (b)$ The temperature at the surface of the rib is 2 2 T (ro, t) $- 163 = 0.132 \rightarrow \text{T}$ () = (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)]ir conservation, personnel protection and comfort, maintaining process temperature variation and fluctuations, condensation and corrosion prevention, fire 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 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)(0.5 - 0.06) + (0.4 - 0.06)(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 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 1.5927 °C/W kAi (0.033 W/m.°C)(0.5708 m 2) 1 1 = 10.07508 °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 \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = s = 20.99 W = R total 1.6678 °C/W T - T \propto (30 - 0)°C Q& = 1.6678 °C/W T - T \propto (30 - 0)°C Q& = 1.6678 °C/W T - T \propto (30 - 0)°C Q& = 1.6678 °C/W T - T \approx (30 - 0)°C Q& = 1.6678 °C/W T - T \approx (30 - 0)°C Q& = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 - 0)°C Q = 1.6678 °C/W T - T \approx (30 much heat to the cooler to melt the ice completely becomes O 15,016,500 J $\Delta t = = 715,549$ s = 198.8 h = 8.28 days 20.99 kJ/s O& 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 $(since \Delta ke \cong \Delta pe \cong 0)$ W& e,in = m& (h2 - h1) = m& (C (T2 - T1) + v(P2 - P1) © 0] = m& C (T2 - T1) + v(P2 - P1) © 0] = m& C (T2 - T1) + v(P2 - P1) © 0] = m& C (T2 - T1) + v(P2 - P1) © 0 through the pipe is determined from V& 0.018 m 3/min V& V1 = 1 = 2 = 9.17 m/min A1 $\pi \pi (0.025 \text{ m}) 21-70 \text{ We D} = 5 \text{ cm}$ 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 ν = 0.1809 × 10 ft/s -3 2 Pr = 0.7260 300 Btu/h Analysis The $1 + \frac{1}{4} = \frac{1}{282,0001 + (0.4 / 0.7260) 2 / 3 []} = 107.84$ The heat transfer coefficient is k 0.01529 Btu/h.ft. °F h = Nu = (107.84) = 1.649 Btu/h.ft. °F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) = 1.649 Btu/h.ft. ?F h = Nu = (107.84) =
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3 2 v 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 - 3 2 v 0.1809 × 10 ft /s The proper relation for Nusselt 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 - 3 2 v 0.1809 × 10 ft /s The proper relation for Nusselt 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 - 3 2 v 0.1809 × 10 ft /s The proper relation for Nusselt 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 - 3 2 v 0.1809 × 10 ft /s The proper relation for Nusselt 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 - 3 2 v 0.1809 × 10 ft /s The proper relation for Nusselt 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 - 3 2 v 0.1809 × 10 ft /s The proper relation for Nusselt 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 - 3 2 v 0.1809 × 10 ft /s The proper relation for Nusselt 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 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 + || 1 $0.5 (0.7260)1/3 | (| 6.633 \times 104 = 0.3 + 1 + 1/4 | | (282,0001 + (0.4/0.7260) 2/3 []) | | / 5/8] | |] 4/5 = 165.95$ Heat transfer coefficient is k 0.01529 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∞) → 1.5 (0.7260) 1/3 | |] 4/5 = 165.95 Heat transfer coefficient is k 0.01529 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∞) → 1.5 (0.7260) 1/3 | |] 4/5 = 165.95 Heat transfer coefficient is k 0.01529 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∞) → 1.5 (0.7260) 1/3 | |] 4/5 = 165.95 Heat transfer coefficient is k 0.01529 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∞) → 1.5 (0.7260) 1/3 | |] 4/5 = 165.95 Heat transfer coefficient is k 0.01529 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∞) → 1.5 (0.7260) 1/3 | |] 4/5 = 165.95 Heat transfer coefficient is k 0.01529 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∞) → 1.5 (0.7260) 1/3 | |] 4/5 = 165.95 Heat transfer coefficient is k 0.01529 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∞) → 1.5 (0.7260) 1/3 | |] 4/5 = 165.95 Heat transfer coefficient is k 0.01529 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& = 1.5 (0.7260) 1/3 | |] 4/5 = 1.5 (0.7260) 1/3 | |] 4/5 = 1.5 (0.7260) 1/3 | |] 4/5 = 1.5 (0.7260) 1/3 | |] 4/5 = 1.5 (0.7260) 1/3 | |] 4/5 | |] 4/5 = 1.5 (0.7260) 1/3 | |] 4/5 | |] 4/5 | $Ts = T\infty + hAs (2.537 Btu/h.ft 2)^{\circ}(18 ft 2) 7-41 Chapter 7 External Forced Convection 7-51 A light bulb is cooled by a fan. hA (45 W / m. Laminar and highly ordered motion of molecules, and turbulent when it involves velocity fluctuations and highly$ disordered motion. Properties The thermal properties of the soil are given to be k = 0.35 W/m.°C and $\alpha = 0.15 \times 10-6$ m2/s. The radius of the interface is ro. Properties The thermal conduction 5-63C The formulation of a transient heat conduction problem in that the transient problem in the transient problem in that the transient problem in that the transient problem in the transient 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 pC pV pC p Lc (7833 kg/m)(465 J/kg.°C)(0.0013 m) - 1 T (t) - T ∞ 100 - 35 = e - bt \rightarrow = e $-(0.01584 \text{ s})t \rightarrow t = 163 \text{ s} = 2.7 \text{ min Ti} - T \propto 900 - 35 \text{ The amount of heat transfer from a single ball is } m = \rho V = \rho \pi D 3 = (7833 \text{ kg}/\text{m}3) \pi (0.008 \text{ m}) 3 = 0.0021 \text{ kg} (465 \text{ J}/\text{kg}) = (0.0021 \text{ kg})(465 \text{ J}/\text{kg}) = (0.0021 \text$ $= 0.037 \text{ Re L } 0.8 \text{ Pr1 } / 3 = 0.037(9.888 \times 105) 0.8 (0.7228)1 / 3 = 2076 \text{ k} \text{ k} 0.02735 \text{ W/m} 2 \cdot \text{C L } 0.8 \text{ m} \text{ As} = \text{wL} = (0.8 \text{ m})(0.4 \text{ m}) = 0.32 \text{ m} 2 \text{ Q\& conv} = \text{hAs} (T \propto - \text{Ts}) = (70.98 \text{ W/m} 2 \cdot \text{C})(0.32 \text{ m} 2)(80 - 20)^{\circ}\text{C} = 1363 \text{ W}$ The radiation heat transfer from the same surface is Q& rad = $\epsilon \text{As} \sigma$ (Ts 4 - Tsurr 4) = (0.95) $(0.32 \text{ m 2})(5.67 \times 10 - 8 \text{ W/m 2} \text{ K 4})[(80 + 273 \text{ K}) 4 - (25 + 273 \text{ K}) 4] = 132 \text{ W}$ Then the 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 = m \Delta x, $y = n \Delta y$, are z = r Δz . 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 = $(0.037 \text{ Re L } 0.8 - 871)(0.7282)1/3 = 1.212 \times 104 \text{ k} \text{ k} 0.02588 \text{ W/m} \cdot 2 \text{ .°C } \text{L 8m The equilibrium temperature of the top surface is then determined by taking convection and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and the heat transfer rate are determined by taking convection and the heat transfer coefficient and theat$ radiation heat fluxes to be equal to each other q& 200 W/m 2 \rightarrow 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]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2],
parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], parameter to be varied" w=2.8 "[m]" "q dot rad=200 [W/m^2], paramete T infinity=30 "[C]" "PROPERTIES" Fluid\$='air' k=Conductivity(Fluid\$, T=T film) Pr=Prandtl(Fluid\$, T=T f 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 guenched 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. The explicit transient finite difference formulation of the problem using the energy balance approach method is to be determined. them we can use the general finite difference relation expressed as $T - Tm T - Tm 4 kA m - 1 + kA m + 1 + h(p\Delta x)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm) + \varepsilon \sigma (p\Delta x 2 / kA)(T \propto - Tm)$ 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 radi 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 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 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 conductivity and one-dimensional 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 $\pi 2$ L ro 0.0015 m V 350° C Lc = = o = = 0.000068 < 0.1 k 386 W/m. $^{\circ}$ 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) A wire=pi*D*L*Convert(in^2, ft^2) g [Btu/h,ft2] 521370 260685 173790 130342 104274 86895 74481 65171 57930 52137 D [in] difference equations are determined on the basis of the energy balance for the transient case expressed as $\Sigma Q\&i + 1$ 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 = ρ Velement 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 $d 2T = 0 dx 2 dT (L) - K = h[T (L) - T\infty] = h(T2 - T\infty) dx x 75^{\circ}F T\infty h L T (L) = T2 = 75^{\circ}F (b)$ 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 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 $0.04 V = (5.2 \text{ m/s}) = 8.667 \text{ m/s} (0.016 \text{ m}) = \text{max} = 8380 \mu 1.895 \times 10 - 5 \text{ kg/m} \cdot \text{s}$ 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(ST/SL) 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) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268) 0.36 (0.7268)
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Properties The thermal conductivity of the concrete is given to be $\text{k} = 2 \text{ W/m} \cdot ^{\circ}\text{C}$. 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 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) $\Delta t = \Delta H = m(h2 - h1) \cong mC p (T2 - T1) 5,000 kJ/h ROOM$ The mass of air is $V = 4 \times 5 \times 7 = 140 \text{ m}3 \text{ m} = PV$ (100 kPa)(140 m3) 1 = 172.4 kg RT1 (0.287 kPa · m3 / kg · K)(283 K) Using the Cp value at room temperature, $4m \times 5m \times 7m$ Steam · Wpw [(10,000 - 5000)/3600 kJ/s + 0.1 kJ/s] $\Delta t = 1163 \text{ s} 1-11 10,000 \text{ kJ/h}$ Chapter 1 Basics of Heat Transfer 131 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 coefficient at the surface of the rib. when it is welldone are to be determined. 2-112C A differential equation involves derivatives, an algebraic equation does not. Therefore, the wall are constant. Properties We assume the film temperature to be a semi-infinite medium 2 The thermal properties of the wall are constant. one-dimensional since the pipe is long relative to its thickness, and there is thermal symmetry about the centerline. $^{\circ}$ C)(900 - 100) $^{\circ}$ C = 781 J = 0.781 kJ (per ball) Then the total rate of heat transfer from the balls to the ambient air becomes Q& = n& Q = (2500 balls/h) × (0.781 kJ/ball) = 1,953 kJ/h = 543 W ball 4-13 Chapter 4 Transient Heat Conduction 4-24 "!PROBLEM 4-24" "GIVEN" D=0.008 "[m]" "T i=900 [C], parameter to be varied" T f=100 "[C]" T infinity=35 "[C]" h=75 "[W/m-2]" C p=465 "[]/kg-C]" alpha=1.474E-6 "[m^2/s]" "ANALYSIS" A=pi*D^2 V=pi*D^3/6 L c=V/A Bi=(h*L c)/k "if Bi < 0.1/s = the lumped sytem analysis is applicable" b=(h*A)/(rho*C p*V) (T f-T infinity)/(T i-T infinity)/(T i-T infinity)/(T i-T f) Q dot=n dot ball*Q*Convert(]/h, W) Ti [C] 500 550 600 650 700 750 800 850 900 950 1000 time [s] 127.4 134 140 145.5 150.6 155.3 159.6 163.7 167.6 171.2 174.7 Q [W] 271.2 305.1 339 372.9 406.9 440.8 474.7 508.6 542.5 576.4 610.3 4-14 Chapter 4 Transient Heat Conduction 180 650 600 170 550 tim e 500 150 450 heat 400 140 350 130 120 500 300 600 700 800 T i [C] 4-15 900 250 1000 Q [W] tim e [s] 160 Chapter 4 Transient Heat Conduction 4-25 An electronic device is on for 5 minutes, and off for several hours. 3 The convection heat transfer coefficient is constant and uniform over the surface. Note that we do not have a square mesh in this case, and thus we will have to rely on energy balances to obtain the finite difference equations. The thermal conductivity are given to be k = 2.3 $W/m \cdot C$ and $\varepsilon = 0.7$. Analysis (a) Noting that the upper part of the iron is well insulated and thus the entire heat generated in the resistance wires is transferred to the base plate, the heat flux through the inner surface is determined Tsurr to be q Q& 1000 W ε q& 0 = 0 = = 66,667 W / m² Abase 150 × 10-4 m² h Taking the direction normal to the surface of the wall to be the x T^{∞} direction with x = 0 at the left surface, the mathematical formulation of this problem can be expressed as and d 2T = 0 dx 2 dT (0) - k = q & 0 = 66,667 W / m² dx dT (L) 4 - Tsurr ∞ surr 2 dx L x (b) Integrating the differential equation twice with x = 0 at the left surface. with respect to x yields dT = C1 dx T(x) = C1x + C2 where C1 and C2 are arbitrary constants. The surface area is $A = 1 \times C2$ where C1 and C2 are arbitrary constants. The surface area is $A = 1 \times C2$ where C1 and C2 area arbitrary constants. 0.65 = 0.65 m2. The flow is laminar." Nusselt x=0.332*Re x^{0.5}*Pr^{(1/3)} h x=k/x*Nusselt x C f x=0.664/Re x^{0.5} x [ft] 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1 ... 0.2985 0.2969 0.2953 0.2937 0.2922 0.2906 0.2891 0.2877 0.2862 0.2848 Cf,x 0.01 0.007071 0.005774 0.005 0.001472 0.004083 0.00378 0.003333 0.003162 0.001048 0.001037 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.001015 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.00105 0.000105 0.000105 0.000105 0.00105 0.000105 0.000 ft -F] 2.5 Chapter 7 External Forced Convection 7-20 A car travels at a velocity of 80 km/h. Assumptions 1 Heat conduction in the shaft is one-dimensional since it is long and it has thermal symmetry about the center line. 3 There is no heat generation in the container. Then the energy equation with dissipation reduces to 2 (∂u) d 2T (V) || || + $\mu \rightarrow k$ = $-\mu$ | 2 2 dy dy (L) (dy) since du / dy = V / L. 2 Heat conduction in the egg is one-dimensional because of symmetry about the midpoint. Analysis The lowest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest
temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature will occur at surfaces of plate while the highest temperature while temperatu at the midplane. Therefore we should always use the logarithmic mean temperature. For top and the two side surfaces: Ri Rconcrete Ro Tin Tout $1 = 0.0067 \times 10 - 4 \text{°C/W}$ kAave $(0.9 \text{ W/m} \cdot 2.\circ)[(40 \text{ m})(13 - 0.3) \text{ m}]$ Ri = Rconcrete Ro = $1 \cdot 1 = 0.769 \times 10 - 4 \text{°C/W}$ ho Ao $(25 \text{ W/m } 2.^{\circ}\text{C})[(40 \text{ m})(13 \text{ m})]$ Rotal = Ri + Rconcrete + Ro = $(0.0067 + 4.37 + 0.769) \times 10 - 4 = 5.146 \times 10 - 4 \circ \text{C/W } \text{T} - \text{Tout} [40 - (-4)]^{\circ} \text{C} = 85,500 \text{ W} \text{ Q} \text{\& top + sides} = \text{in Rtotal } 5146 \times 10 - 4 \circ \text{C/W } \text{T} - \text{Tout} [40 - (-4)]^{\circ} \text{C} = 85,500 \text{ W} \text{ Q} \text{\& top + sides} = \text{in Rtotal } 5146 \times 10 - 4 \circ \text{C/W } \text{T} - \text{Tout} [40 - (-4)]^{\circ} \text{C} = 85,500 \text{ W} \text{ Q} \text{\& top + sides} = 10.05 \text{ m} = 0.01 \text{ m} \text{ M} - 16 - 1 \text{ This problem involves } 6 \text{ unknown nodal temperatures}, and thus$ we need to have 6 equations to determine them uniquely. The interface temperature and the effect of doubling the thickness of the plastic cover on the interface temperature are to be determined. Analysis Under steady conditions, the rate of heat transfer through the glass by conduction is $(28 - 25)^{\circ}C \Delta T Q\&$ cond = kA = $(0.7 \text{ W/m} \cdot \text{°C})(2.2 \text{ m } 2) =$ 770 W L 0.006 m The rate of heat transferred from the glass by convection is $Q\& = hA\Delta T = (10 \text{ W/m } 2 \cdot ^{\circ}\text{C})(2.2 \text{ m } 2)(25 - 15)^{\circ}\text{C} = 220 \text{ W } Q\&$ conv Under steady conduction should be transferred from the outer surface by convection and radiation. The initial temperature of the ice block to avoid melting for 2 h is to be determined. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of m& chicken = (500 chicken + h(2.2 kg/ chicken) = 1100 kg/ h = 0.3056 kg/ s Then the rate of heat removal from the chickens as they are cooled from 15°C to 3°C at this rate becomes Q& chicken = ($m \& C p \Delta T$) chicken = (0.3056 kg/s)(3.54 kJ/kg.º C)(15 - 3)^o C = 13.0 kW (b) The chiller gains heat from the surroundings as a rate of 210 kJ/min = 3.5 kJ/s. 2 Heat transfer is one-dimensional since the plates are large. 6-16C (a) The dynamic viscosity of liquids decreases with temperature. The finite difference equations for boundary nodes are obtained by applying an energy balance on the volume elements and taking the direction of all heat transfers to be towards the node under consideration: Node 2: T1 + 2T4 + T3 - 4T2 = 0 Node 3: T2 + 2T5 - 4T3 = 0 Insulated • 2 • 3 Node 4: 2 × 32 + T2 + T3 - 4T2 = 0 =139.4°F, T6 =130.7°F Therefore, the temperature at the middle of the insulated surface will be T2 =82.8°F. 4-88C The environmental factors that affect of the growth rate of microorganisms are the temperature, the relative humidity, the oxygen level of the environment, and air motion. (b) The dynamic viscosity of gases increases with temperature 5-93 Chapter 5 Numerical Methods in Heat Conduction 5-92 The formation of fog on the glass surfaces of a car is to be prevented by attaching electric resistance heaters to the inner surfaces. Using Eq. 5-6, the first derivative of temperature dT / dx at the midpoints n - 1/2 and n + 1/2 of the sections surrounding the node n can be expressed as Tn+1 $T(x) T - Tn - 1T - Tn dT dT \cong n \cong n + 1$ and $dx n - 1 \Delta x dx n + 1 \Delta x 2 2$ Noting that second derivative is simply the derivative of the first derivative of the first derivative of the first derivative is a d 2T d x 2 \cong n dT d x 1 n + 2 - dT d x Tn Tn - 1 n - 2 Δx Tn + 1 - Tn Tn - Tn - 1 - T - 2Tn + Tn + 1 $\Delta x \Delta x = n - 1 \Delta x \Delta x 2 \Delta x \Delta x x n$ 1 n n+1 which is the finite difference representation of the second derivative at a general internal node n. m) T2 1 1 Rconv = = 1481 °C / W . Properties The thermal properties of the steaks are $\rho = 970 \text{ kg/kg}$. °C, k = 1.40 W/m.°C, $\alpha = 0.93 \times 10 - 6 \text{ m}^2$ / s, $\epsilon = 0.95$, and hif = 187 kJ/kg. °C and one with h = 80 W / m2 . Using the proper relation for Nusselt number, heat transfer coefficient is determined to be h L Nu = 0 = $(0.037 \text{ Re L } 0.8 - 871) \text{ Pr } 1/3 = 0.037(7.792 \times 106) = 30.78 \text{ W/m } 2$. °C L 8m Ri Rinsulation Ro The thermal resistances are T ∞ 1 As = wL = (3 m)(8 m) = 24 m 2 [Ri =] 1 1 = = °C/W T -T (22 - 4)°C = 122.1 W Q& = $\infty 1 \propto 2$ = Rtotal 0.1474 °C/W 7-83 T $\propto 2$ Chapter 7 External Forced Convection 7-91 A car travels at a velocity of 60 km/h. Properties The thermal conductivity is given to be k = 29.5 Uranium rod W/m·°C. Analysis The volume of the air in the house is V = (floor space)(height) = (200 m2)(3 m) = 600 m3 Noting that the infiltration rate is 0.7 ACH (air changes per hour) and thus the air in the house due to infiltration is P V& P (ACH × V house) m& air = 0 air = 0 RTo RTo = 3 (89.6 kPa) $(16.8 \times 600 \text{ m}/\text{day})(0.287 \text{ kPa.m } 3/\text{kg.K})(5 + 273.15 \text{ K}) 0.7 \text{ ACH } 22^{\circ}\text{C} = 11,314 \text{ kg/day} \text{ Noting that outdoor air enters at } 5^{\circ}\text{C} = 11,314 \text{ kg/day}(1.007 \text{ kJ/kg.}^{\circ}\text{C})(22 - 5)^{\circ}\text{C} = 193,681 \text{ kJ/day} = 53.8 \text{ kWh/day} \text{ At a unit cost of } 80.082/\text{kWh/day} \text{ At a unit cost of } 80.082/\text{kWh/day} = 53.8 \text{ kWh/day} = 53.8 \text{ kWh/day} = 53.8 \text{ kWh/day} \text{ At a unit cost of } 80.082/\text{kWh/day} = 53.8 \text{ kWh/day} =$ the cost of this electrical energy lost by infiltration is Energy cost = (Energy used)(Unit cost of energy) = (53.8 kWh/day)(\$0.082/kWh) = \$4.41/day 1-5 Chapter 1 Basics of Heat Transfer 1-20 A house is heated from 10°C to 22°C by an electric heater, and some air escapes through the cracks as the heated air in the house expands at constant pressureThe average friction coefficient and the drag force per unit width are determined from C f = 1.328 Re L $-0.5 \rho V \propto 2.5 - 0.5 = 1.328(1.46 \times 10)$ Oil $V \propto = 3 \text{ m/s}$ T $\propto = 30^{\circ}$ C = 0.00347 (867 kg/m 3)(3 m/s) 2 = 81.3 N 2.2 Similarly, the average Nusselt number and the heat transfer coefficient are determined using the laminar flow relations for a flat plate, hL Nu = = 0.664 Re L 0.5 Pr 1/3 = 0.664(146. We assume these two effects to counteract each other. 4 There are no convection currents in the air space between the plates. It is limited to flow of fluids with a Prandtl number of near unity (such as gases), and negligible pressure gradient in the flow direction (such as flow over a flat plate). We $Q_{k} = s1 \propto 2 = 1511 \ 13.96 \ ^{\circ} C / W$ Rotal 0.01511 kJ / s $Q_{k} \ll fg$ $Q_{k} = mh \rightarrow m_{k} = = 0.000076 \ kg / s \ 198 \ kJ / kg h \ fg \ 3-67$ Chapter 3 Steady Heat Conduction 3-82 A 3-m diameter spherical tank filled with liquid oxygen at 1 atm and -183 $^{\circ}$ C is exposed to convection and radiation with the surrounding air and surfaces. The specific heat of air at room temperature is $Cp = 1.007 \text{ kJ/kg} \circ C$ (Table A-15). 4-62C The total amount of heat transfer from a semi-infinite solid up to a specified time to can be determined by integration from $Q = \int to 0 \text{ Ah}[T(0, t) - T\infty] dt$ where the surface temperature T(0, t) is obtained from Eq. 4-22 by substituting x = 0. Once the unit thermal resistances and the Ufactors for the insulation and stud sections are available, the overall = 1/Uoverall where Uoverall = 1/Uoverall = part of the wall are to be treated as studs. 2 Thermal conductivity is constant. Analysis The time the steel rods stays in the oven can be determined from t = 180 s velocity 10 ft / min Oven, 1700°F The Biot number is Bi = hro (20 Btu/h.ft 2 .°F)(2 / 12 ft) = = 0.4307 k (7.74 Btu/h.ft.°F) Steel rod, 85°F The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, $\lambda 1 = 0.8784$ and A1 = 10995. Properties The specific heat of water at room temperature is C = 4.18 kJ/kg·°C (Table A-2). Then, 4 - (T1 + 273) 4] = 0 m = 1: T0 - 2T1 + T2 + h(p\Delta x 2 / kA)(T - T2) + L(p\Delta x 2 / kA)(T - T2) + L(p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 / kA)(T - T1) + \epsilon\sigma (p\Delta x 2 /
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Therefore, the new design will be a poorer conductor of heat. 4-86 Chapter 4 Transient Heat Conduction is steady and one-dimensional since the surface area of the base plate is large relative to its thickness, and the thermal conditions on both sides of the plate are uniform. Calculations that involve the alternate addition of small and large numbers are most susceptible to round-off error. 5-91 The formation of fog on the glass surfaces of a car is to be prevented by attaching electric resistance heaters to the inner surfaces. 2 The exposed surface temperature of the person and the convection heat transfer coefficient is constant and uniform. Assumptions 1 Heat conduction in the semi-infinite cylinder is two-dimensional, and thus the temperature of the person and the convection heat transfer coefficient is constant and uniform. water at the mean temperature of $(15^{\circ}C + 65^{\circ}C)/2 = 40^{\circ}C$ are (Table A-9): k = 0.631 W/m-K ρ = 999.1 kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the mass flow rate calculation at the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the inlet) is $\rho i = 999.1$ kg/m3. 6 The phase change in the inlet temperature of $15^{\circ}C$ (for use in the inlet) is $\rho i = 999.1$ kg/m3. 7 m set (1000 m effects are not considered, and thus the actual the temperatures will be much higher than the values determined since a considerable part of the cooling process will occur during phase change (freezing of chicken). Then the Nusselt number and the heat transfer coefficient are determined to be hL Nu = = (0.037 Re L 0.8 - 871) Pr 1 / 3 = $[0.037(2.338 \times 107) 0.8 - 871](0.7336)1/3 = 2.542 \times 104$ k k 0.02439 W/m.°C h = Nu = (2.542 × 104) = 31.0 W/m 2.°C L 20 m In steady operation, heat transfer from the roof to the surroundings (by convection and radiation), which must be equal to the heat transfer through the roof by conduction. ° C)(0.0025 m) tanh aL tanh(15.37 m - 1 × 0.03 m) = 0.935 = aL 15.37 m - 1 × 0.03 m The number of fins, finned and unfinned surface areas, and heat transfer rates from those areas are η fin = n = 1 m2 = 27,777 (0.006 m)(0.006 m) [$\int \pi D 2 \int \pi (0.0025) 2 \int 2 = +27777 (0.006 m) (0.0025) 2 \int 2 = +27777 (0.0075) 2 \int 2 = +27777 (0.$ unknown nodal temperatures, and thus we need to have 6 equations. The emissivity of the fin surface is 0.9. Analysis The fin length is given to be L = 5 cm, and the number of nodes is specified to be M = 6. 2 The temperature in the spoon varies in the axial direction only (along the spoon), T(x). 4 The surfaces of the plate are smooth. The space between the studs is filled with fiberglass insulation. kg / s c c & p $\Delta T = (106 \text{ Q} \text{ = mC} \cdot 4.4 \text{ Chapter 4 Transient Heat Conduction 4-16E A number of aluminum balls are to be quenched in a water bath at a specified rate. °C)(3 m 2)(80 - 30)°C = 1500 W In order to reduce heat loss by 90%, the new heat transfer rate and thermal resistance must be$ Q&=010. (b) Noting that the cross-sectional areas of the finit contents of the circular finit contents of the contents of the refrigerator, including the air inside, rises uniformly during this period. 6-2C If the fluid is forced to flow over a surface, it is called external forced convection. 5 The heat transfer coefficient accounts for the effect of radiation from the fins.
3-121 Chapter 3 Steady Heat Conduction 3-162 An ice chest made of 3-cm thick styrofoam is initially filled with 45 kg of ice at 0°C. Therefore, the boundary conditions are u(0) = 0 and u(L) = V, and applying them gives the velocity distribution to be u(y) = y V L Frictional heating due to viscous dissipation in this case is significant because of the high viscosity of oil and the large plate velocity. ° C)(80 - 0)° C = 413 W The shape factor, and the rate of heat loss on the horizontal part that is in the ground are $2\pi (20 \text{ m}) 2\pi L = 22.9 \text{ m} = (42 | [4(3 \text{ m})] \ln | |n| | (D/ [(0.05 \text{ m})] Q \& = Sk (T1 + 1)) Q \& = Sk (T1 + 1)$ -T2) = (22.9 m)(15. Analysis (a) The heat generation per unit volume of the wire is Q& gen Q& gen 2000 W g& = . Analysis The heat flux at the bottom of the pan is Q& G& 0.90 × (900 W) q& s = s = = 31,831 W / m 2.2 As $\pi D / 4 \pi$ (018. Therefore, there is no need to repeat calculations. in the air from the free surface of the water. ° C L 6m The rate of heat transfer is then determined from Newton's law of cooling to be Q& = hA (T - T) = (68.3 W/m 2.°C)(6 × 1 m 2) s 7-2 Ts = 30°C L=6m Chapter 7 External Forced Convection 7-15 The top surface of a hot block is to be cooled by forced air. (b) The nodal temperatures under steady conditions are determined by solving the 6 equations above simultaneously with an equation solver to be T1 = 49.0° C, T2 = 33.0° C, T3 = 27.4° C, T4 = 25.5° C, T5 = 24.8° C, and T6 = 24.6° C, (c) The total rate of heat transfer from the spoon handle is simply the sum of the heat transfer from each nodal element, and is determined from Q& fin = $6 \sum Q\&m = 0$ 6 element, m = $\sum hA$ surface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node 0, Asurface, m = $p\Delta x/2 + A$ for node temperature varies in both x- and rdirections. Air Analysis First we find the Biot number: Bi = hr0 (19 W / m2. However, we would analyze this problem under the worst anticipated lowest temperature in the kitchen (the so called "design" conditions). Properties We assume the film temperature to be 50°C. 4 Air flow is turbulent because of the intense vibrations involved. 5 The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified). The time passed since his death is to be estimated. Then the temperature of the wire at the surface (r = r0) is determined by substituting the known quantities to be B. × 10 - 6 m2 / s)(20 min × 60 s / min) (0.025 m) 2 { 2 T (0,0,0, t) - 500 = (2.208) - (0.5932) (2.208) 20 - 500 = (2.208) - (0.5932) (2.208) 20 - 500 = (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) (2.208) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) - (0.5932) thus the thermal contact resistance is significant for highly conducting materials like metals. Analysis We observe that the volume of a rigid tank is constant We take the entire contents of the tank, water + copper block, as the system. ° C / W)(0.04 W / m. For specified heat transfer coefficients, the length of the tube required to condense steam at a rate of 400 lbm/h is to be determined. The center temperature of the wire is to be determined. = A1e $-\lambda 1 \tau = (15618 \text{ Ti} - T \infty)$ a bort cylinder T (0,0, t) $- 202 = 0.106 \rightarrow T (0,0, t) = 185^{\circ} \text{F} 40 - 202 \text{ After 10 minutes}$ 4-75 Chapter 4 Transient Heat Conduction $\tau = \alpha t L2 = \theta$ o, wall = $\tau = \alpha t$ 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 - $\lambda 1 \tau = (1.2728)e^{-(1.5421)}(0.03) \approx 1$ Ti - T∞ = (0.0077 ft 2 / h)(10 / 60 h) (0.4 / 12 ft) 2 = 1156 > 0.2 . 4 The surface of the tank is at the same temperature as the water temperature. kPa)(600 m3) m = = 747.9 kg RT (0.287 kPa.m3 / kg.K)(10 + 273.15 K) 22°C 10°C AIR Noting that the pressure in the house as it is heated from 10 to 22°C is determined to be Q = mC p (T 2 - T1) = (747.9 kg)(1.007 kJ/kg. 2 Specific heats of beef carcass and air are constant. 2 Heat transfer to the water is negligible. 5 The convection resistance inside the pipe is 0°C. Analysis This is a transient heat conduction problem, and the temperature of the pipe is negligible so that the inner surface temperature of the pipe is negligible. between the potato and the surroundings decreases. ° C)(0147. A small value of NTU (NTU < 5) indicates more opportunities for heat transfer will not increase no matter how much we extend the length of the tube. $2 W = 1959 W/m^2 q \& = As 0.0001021 m^2 As = 2 Power$ Transistor 0.2 W (c) The surface temperature of the transistor can be determined from Q& 0. 3-13C The window, and thus the heat transfer rate will be smaller relative to the one which consists of a single 8 mm thick glass sheet. 5-31 4 + 273) 4 - Tsurr] = 83.6 W Chapter 5 Numerical Methods in Heat Conduction 5-39 "PROBLEM 5-39" "GIVEN" t_pipe=0.004 "[m]" k=52 "[W/m-C]" epsilon=0.8 D_o_pipe=0.10 "[m]" t_flange=0.01 "[m]" D_o_flange=0.20 "[m]" T_steam=200 "[C], parameter to be varied" h_i=180 "[W/m^2-C]" T_infinity=8 "[C]" "h=25 [W/m^2-C], parameter to be varied" T_surr=290 "[K]" DELTAx_1=t_pipe "the distance between nodes 0 and 1" DELTAx_2=t_flange "nodal spacing along the flange" L=(D_o_flange D o pipe)/2 M=L/DELTAx 2+2 "Number of nodes" t=2*t flange "total thixkness of the flange" The values of radii at the nodes and between the nodes and between the nodes [m]" r 3=0.07 "[m]" r 3=0.07 "[m]" r 3=0.07 "[m]" r 3=0.07 "[m]" r 3=0.07 [m]" r 3=0.0 [m]" r 45=0.085 "[m]" r 56=0.095 "[m]" "Using the finite difference method, the five equations for the unknown temperatures at 7 nodes are determined to be h i*(2*pi*t*r 0)*(T 1-T 0)/DELTAx 1+k*(2*pi*t*r 0)*(T 1-T 0)/DELTAx 1=0 "node 0" k*(2*pi*t*r 0)*(T 1-T 0)/DELTAx 1=0 "node 0" k*(2* $(DELTAx 2/2)*(h*(T infinityT 1)+epsilon*sigma*(T surr^4-(T 1+273)^4))=0$ "node 1" k*(2*pi*t*r 23)*(T 2-T 3)/DELTAx 2+k*(2*pi*t*r 23)*(T 1-T 2)/DELTAx 2+k*(2*pi*t*r 23)*(T
3T 2)/DELTAx 2+k*(2*pi*t*r 34)*(T 3T $(T 4T 3)/DELTAx 2+2*2*pi*t*r 3*DELTAx 2+k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 3)+epsilon*sigma*(T surr^4(T 3+273)^4))=0$ "node 3" k*(2*pi*t*r 45)*(T 4-T 5)/DELTAx 2+k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 3)+epsilon*sigma*(T surr^4(T 3+273)^4))=0 "node 3" k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 3)+epsilon*sigma*(T surr^4(T 3+273)^4))=0 "node 3" k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 3)+epsilon*sigma*(T surr^4(T 3+273)^4))=0 "node 3" k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 3)+epsilon*sigma*(T surr^4(T 3+273)^4))=0 "node 3" k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 4)+epsilon*sigma*(T surr^4(T 3+273)^4))=0 "node 3" k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 4)+epsilon*sigma*(T surr^4(T 3+273)^4))=0 "node 3" k*(2*pi*t*r 45)*(T 5T 4)/DELTAx 2+(h*(T infinity-T 4)+epsilon*sigma*(T surr^4(T 3+273)^4))=0 $(T_6T_5)/DELTAx_2+2*2*pi*t*r_5*DELTAx_2+2*2*pi*t*r_5)*(T_infinity_T_5)+epsilon*sigma*(T_surr^4(T_5+273)^4))=0$ "node 5" 5-32 Chapter 5 Numerical Methods in Heat Conduction k*(2*pi*t*r_6)*(T_infinity_T_6)+epsilon*sigma*(T_surr^4(T_5+273)^4))=0 "node 6" T_tip=T_6 "(c)' Q dot =Q dot 1+Q dot 2+Q dot 3+Q dot 4+Q dot 5+Q dot 6 "where" Q dot 1=h*2*2*pi*t*(r 1+r 12)/2*DELTAx 2/2*(T 1T infinity)+epsilon*sigma*2*2*pi*t*(r 1+r 12)/2*DELTAx 2/2*(T 1 Q dot 3=h*2*2*pi*t*r 3*DELTAx 2*(T 3T infinity)+epsilon*sigma*2*2*pi*t*r 4*DELTAx 2*(T 4T infinity)+epsilon*sigma*2*2*pi*t*r 4*DELTAx $2*((T 4+273)^4-T surr^4)$ Q dot 5=h*2*2*pi*t*r 5*DELTAx $2*((T 5+273)^4-T surr^4)$ Q dot 4=h*2*2*pi*t*r 4*DELTAx $2*((T 5+273)^4-T surr^4)$ Q dot 5=h*2*2*pi*t*r 5*DELTAx $2*((T 5+273)^4-T surr^4)$ Q dot 5=h*2*2*pi*t*r 5*DELTAx 2*((T 5Q dot 6=h*2*(2*pi*t*(r 56+r 6)/2*(DELTAx 2/2)+2*pi*t*r 6)*((T 6+273)^4T surr^4) Tsteam [C] 150 160 170 180 190 200 210 220 230 240 250 260 270 280 290 300 Ttip [C] 84.42 89.57 94.69 99.78 104.8 109.9 114.9 119.9 124.8 129.7 134.6 139.5 144.3 149.1 153.9 158.7 Q [W] 60.83 65.33 69.85 74.4 78.98 83.58 88.21 92.87 97.55 102.3 107 111.8 116.6 121.4 126.2 131.1 h [W/m2.C] 15 20 25 30 35 40 45 50 55 60 Ttip [C] 126.5 117.6 109.9 103.1 97.17 91.89 87.17 82.95 79.14 75.69 Q [W] 68.18 76.42 83.58 89.85 95.38 100.3 104.7 108.6 112.1 115.3 5-33 Chapter 5 Numerical Methods in Heat Conduction 160 140 150 130 120 tem perature 130 110 120 100 heat 110 90 100 80 90 70 80 140 160 180 200 220 240 260 280 Q [W] T tip [C] 140 60 300 T steam [C] 130 120 120 110 100 100 90 tem perature 90 80 80 70 70 15 20 25 30 35 40 2 45 h [W /m -C] 5-34 50 55 60 60 Q [W] T tip [C] heat 110 Chapter 5 Numerical Methods in Heat Conduction 5-40 Using an equation solver or an iteration method, the solutions of the following systems of algebraic equations are determined to be as follows: (a) 3x1 - x2 + x3 = 3 4x1 - 2x2 + x"ANALYSIS" "(a)" 3*x 1a-x 2a+3*x 3a=0 -x 1a+2*x 2a+x 3a=3 2*x 1a-x 2a-x 3a=2 "(b)" 4*x 1b-2*x 2b-2+0.5*x 3b=-2 x 1b-3-x 2b+-x 3b=-11.964 x 1b+x 2b+x 3b=-3 5-41 Using an equation solver or an iteration method, the solutions of the following systems of algebraic equations are determined to be as follows: (a) 3x1 - 2x2 - x3 + x4 = 6x14*x 4a=-6 "(b)" $3*x 1b+x 2b^2+2*x 3b=8-x 1b^2+3*x 2b+2*x 3b=-6.293 2*x 1b-x 2b^4+4*x 3b=-12 5-35$ Chapter 5 Numerical Methods in Heat Conduction 5-42 Using an equation solver or an iteration method, the solutions of the following systems of algebraic equations are determined to be as follows: (a) $4x1 - x^2 + 2x^3 + x^4 = -6$ (b) x1 $3x^2 - x^3 + 4x^4 = -12x^1 + x^24 - 2x^3 + x^4 = 1x^{2} + 2x^{2} - 2x^{3} + x^{4} = -12x^{2} + 5x^{4} = -3 - x^{1} + 2x^{2} + 5x^{4} = 5 - x^{1} + x^{24} + 5x^{3} = 102x^{2} - 4x^{3} - 3x^{4} = 2$ Solution: $x_1 = -0.744$, $x_2 = -8$, $x_3 = -7.54$, $x_4 = -12x^{1} + x^{24} - 2x^{3} + x^{4} = -12x^{1} + x^{24} + 5x^{3} = 102x^{2} - 4x^{3} - 3x^{4} = 2$ Solution: $x_1 = -0.744$, $x_2 = -8$, $x_3 = -7.54$, $x_4 = -12x^{1} + x^{24} + 5x^{3} = 102x^{2} - 4x^{3} - 3x^{4} = 2$ Solution: $x_1 = -0.744$, $x_2 = -8$, $x_3 = -7.54$, $x_4 = -12x^{1} + x^{24} - 2x^{3} + x^{4} = -12x^{1} + x^{24} + 5x^{3} = -12x^{1} + x^{24} + 5x^{2} + 5x^{2}$ $2*x_1b+x_2b^4+2*x_3b+x_4b=1x_1b^2+4*x_2b+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3b^2+2*x_3$ formulation of a general interior node is given in its g& l 2 simplest form as Tleft + Ttop + Tright + Tbottom - 4Tnode + node = 0 : k (a) Heat transfer is steady, (b) heat transfer is steady, (c) there is heat generation in the medium, (d) the nodal spacing is constant, and (e) the thermal conductivity of the medium is constant. It is likely to occur at places with high ceilings. 5-21 Chapter 5 Numerical Methods in Heat Conductivity and emissivity are given to be k = 15.1 W/m·°C and $\varepsilon = 0.8$. Analysis The nodal spacing is given to be $\Delta x = 1.5$ cm. Assumptions 1 The apples are spherical in shape with a diameter of 9 cm. The R - values of 100-mm face brick and a 20-mm air space between the wall and the bricks various layers are 0.075 and 0.170 m2.°C/W, respectively. 6-5C Nusselt number is the dimensionless convection heat transfer through a fluid layer. 4-103 Chickens are to be cooled by chilled water in an immersion chiller. 2 The temperature in the board and along the fins varies in one direction only (normal to the board). Therefore, the flow is laminar. Then the dimensionless temperature at the center of the plane wall is determined from θ o, wall = 2 2 T 0 - T ∞ = A1 e - λ 1 τ = (1.0050)e - (0.164) $(5.424) = 0.869 \text{ Ti} - T \infty$ We repeat the same calculations for the long cylinder, Bi = hr0 (40 W/m 2.°C)(0.04 m) = 0.01455 k (110 W/m.°C) $\lambda 1 = 0.01455 \text{ k}$ (110 W/m.°C) $\lambda 1 = 0.00455 \text{ k}$ (110 W/m.°C) (110 W/ 35 s/h (0.05 m) 2 = 0.536 > 0.2 and thus the assumption of τ > 0.2 for the applicability of the one-term approximate solution is verified. The velocity and temperature, the rate of heat transfer, and the mechanical power wasted in oil are to be determined. Properties The thermal conductivity is given to be k = 0.2 for the applicability of the one-term approximate solution is verified. $237 \text{ W/m} \cdot \text{C}$. Analysis The electrical power consumed by the heater and converted to heat is W&e = VI = (110 V)(0.6 A) = 66 W Q The rate of heat flow through each sample is W& 66 W Q&e = = = 33 W 2 2 3 cm Then the thermal conductivity of the sample becomes A = $\pi D 2 4 = \pi (0.04 \text{ m}) 2 3 \text{ cm} = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33
W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T QL Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T Q&e = 0.001257 \text{ m} 2 4 \& (33 W)(0.03 \text{ m}) \Delta T Q&e = 0.001257 \text$ $kA \rightarrow k = = 78.8 \text{ W}/\text{m}$. °F)(1 ft) Ro = Rconv, 2 = 1 1 1 = = 0.0955 h. °C) = 0.32 m k L 7-82E The thickness of flat R-20 insulation in English units is to be determined if the water in the pipe will completely freeze during a cold night. – (10° F) 0.475 – 0.0203(10 mph $+ 0.304 \ 10 \ \text{mph} = -9^\circ \text{FV} = 20 \ \text{mph}$: Tequiv = 914. 3 We the temperature of the room remains constant during this process. 4 m Q& = Sk (T1 - T2) = (7.68 \text{ m})(0.55 \text{ W/m.}^\circ\text{C})(18 - 0)^\circ\text{C} = 76 \text{ W} \ 3-138 \ T2 = 0^\circ\text{C} \ D = 1.4 \text{ m} Chapter 3 Steady Heat Conduction 3-177 A cylindrical tank containing liquefied natural gas (LNG) is placed at the center of a square solid bar. 3 The thermal properties of the apples are constant. The energy balance for this system E -E = 1in424 out 3 1 424 3 Net energy transfer by heat, work, and mass Change in internal, kinetic, potential, etc. Substituting these values into the one-term solution gives $\theta 0 = To - T \infty 2 = A1e - \lambda 1\tau \rightarrow Ti - T \infty$ $T = 78^{\circ}F 40 - 25$. The change in the drag force and the rate of heat transfer are to be determined when the free-stream velocity of the fluid is doubled. The fraction of heat lost from the glass cover by radiation is to be determined when the free-stream velocity of the fluid is doubled. The fraction of heat lost from the glass cover by radiation is to be determined. tubes is equal to the temperature of steam. The center and surface temperatures of the apples, and the amount of heat transfer from each apple in 1 h are to be determined. \times 10 - 6 m 2 / s , $\Delta x = 0.25$ m, where k = 0.61 W/m.°C, $\alpha = k / \rho C = 015$ and $\Delta t = 15$ min = 900 s. ° F)(0.0967 ft) (247 Btu / h.ft. Analysis We determine the temperature at a depth of x = 0.3 m in 3 h using the analytical solution, $(x T (x, t) - Ti = erfc | | Ts - Ti (2 \alpha t) Kiln wall Substituting, () T (x, t) - 2 0.3 m | | = erfc | -5 2 42 - 5 | 2 (0.23 \times 10 m/s)(3 h \times 3600 s/h) | | () = erfc (0.952) = 0.1782 30 cm 42°C T (x, t) = 9.1 °C 2°C 0 x which is greater than the initial temperature of 2°C. We know that the$ boundary nodes are more restrictive than the interior nodes, and thus we examine the formulations of the boundary nodes 0 and 6 only. Outside surface, 12 km/h (summer) 2. 2-26 For a medium in which the heat conduction equation is given by 1 $\partial \left(2 \partial T \right) 1 \partial T |r| = r 2 \partial r \langle \partial r \rangle$ and thus we examine the formulations of the boundary nodes 0 and 6 only. Outside surface, 12 km/h (summer) 2. 2-26 For a medium in which the heat conduction equation is given by 1 $\partial \left(2 \partial T \right) 1 \partial T |r| = r 2 \partial r \langle \partial r \rangle$ no heat generation, and (d) the thermal conductivity is constant. $\times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 3600 \text{ s} / \text{h})$ (0.02 m) 2 (0124. Review Problems 2-120 A small hot metal object is allowed to cool in an environment by convection. The total rate of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the actual amount of heat transfer from the ball is to be determined. °C)(163 - 4.5)°C = 2080 \text{ kJ Then the ac becomes sin($\lambda 1$) - $\lambda 1 \cos(\lambda 1) \sin(3.0372)$ - (3.0372) cos(3.0372) Q = 0.783 = 1 - 30 o, sph = 1 - 3(0.65) 3 Qmax (3.0372) 3 $\lambda 1$ Q = 0.783 Qmax = (0.783)(2080 k]) = 1629 kJ (d) The cooking time for medium-done rib is determined to be θ 0, sph = t = 2 2 T 0 - T ∞ 71 - 163 = A1e - $\lambda 1$ T \rightarrow = (1.9898)e - (3.0372) T \rightarrow T = 0.1336 4.5 - 163 Ti - $T \propto \tau ro 2$ (0.1336)(0.08603 m) 2 = 10,866 s = 181 min \cong 3 hr \alpha (0.91× 10 - 7 m 2/s) This result is close to the listed value of 3 hours and 20 minutes. 5-73 Chapter 5 Numerical Methods in Heat Conduction 5-84 A uranium plate initially at a uniform temperature is subjected to insulation on one side and convection on the other. 5-18 Chapter 5 Numerical Methods in Heat Conduction 5-30E A large plate lying on the ground is subjected to convection and radiation. As a result, the drag coefficient suddenly drops. The temperature difference between the two sides of the circuit board is to be determined from Equation. 2-77 to be () T – T2 $\begin{bmatrix} \beta \end{bmatrix}$ T – T2 Q& cylinder = 2\pi k ave L 1 = 2\pi k 0 $\begin{bmatrix} 1 + T22 + T172 + T12 \end{bmatrix}$ L 1 ln(r2 / r1) 3 $\begin{bmatrix} 1 \\ 1 \end{bmatrix}$ ln(r2 / r1) Discussion We would obtain the same result if we substituted the given k(T) relation into the second part of Eq. 2-77, and performed the indicated integration. The heat transfer coefficient is higher in turbulent flow. h. 4 m 1 + 0.252. Therefore, the window will will be fogged at all times. The mass of the potato is $4 \text{ m} = \rho \text{V} = \rho \, \text{Irr} 3$ Ts Rtowel Rconv $3 \, \text{T} \propto 4 = (62.2 \, \text{lbm/ft} 3) \, \pi (1.5 / 12 \, \text{ft}) 3 = 0.5089 \, \text{lbm} (0.998 \, \text{Btu} / \text{lbm})$. The rate of heat transfer from the tank is to be determined. Properties The thermal conductivity and emissivity of copper are given to be $k = 386 \text{ W/m} \cdot \text{°C}$ and $\epsilon = 0.7$. Analysis The individual resistances are Ai = π Di L = π (0.04 m)(15 m) = 1885. Analysis The individual resistances are Ai = π Di L = π (0.04 m)(15 m) = 1885. cracks/openings are not considered. 2 Heat transfer is one-dimensional since the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness, and the thermal conditions on both sides of the roof area is large relative to its thickness. the stated assumptions and observations, the energy balance becomes E -E 1in424out 3 = Net energy transfer by heat, work, and mass ΔE system 1 424 3 Change in internal, kinetic, potential, etc. Analysis The conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in
conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion factors for W, m, and K are given in conversion f the Stefan-Boltzmann constant in the desired units, $\sigma = 5.67 \text{ W} / \text{m} 2$. 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.ft 2. °F. Vapor Analysis The rate of heat transfer to the oxygen tank is As = $\pi D 2 = \pi (4 \text{ m}) 2 = 50.27 \text{ m} 2 Q_{\&} = hAs (Ts - Tair) = (25 \text{ W/m} .°C)(50.27 \text{ m})[20 - (-183)]°C = 255,120 \text{ W} 2 2 \text{ Air } 20° 1 \text{ atm Liquid O2 Then the rate of evaporation of liquid oxygen in the tank is determined Q_{\&} to be -183°C Q_{\&} . 4 Thermal contact resistances at the interfaces are to be considered. q = 1000 W Analysis A 1000 W$ 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 in nuclear fuel rods. 3-14C Convection heat transfer through the wall is expressed as $Q_{\&}$ = hAs (Ts - T ∞). 2-9 $\partial (\partial T) \partial 2T$ || = - || k || = - k $\partial y \langle \partial y \rangle \partial y 2$ / Chapter 2 Heat Conduction Equation 2-29 We consider a thin ring shaped volume element of width Δz and thickness Δr 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 canstant 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 $\theta 0 = 2$ To $-T \infty = A1e - \lambda 1\tau \rightarrow Ti - T \infty - 30^{\circ}C$ 14 / Meat 7°C 2 -18 - (-30) = 1239. 5-114 Chapter 5 Numerical Methods in 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 during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period is determined by adding the heat transfer during a specified time period. T0 + T0i -1) / 2 - Tin] Δt 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 heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and the heat conduction heat transfer Q& is constant and transfer Q& is const because of the constant agitation of the engine block. = 4.201 ° C / W The steady rate 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 y 3 or (y') 2, (2) any products 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 y 5-102C As the step size is decreased, the discretization error decreases but the round-off error increases. Applying the boundary conditions give $C1 = -q\& 0 \rightarrow x = L$: 4] - kC1 = q& 0 $\rightarrow x = L$: 4] - kC1 = h[T2 - T ∞] + $\varepsilon\sigma$ [(T2 + 273) 4 - Tsurr Eliminating the constant C1 from the two relations above gives the following expression for the outer surface temperature T2, 4] - kC1 = h[T2 - T ∞] + $\varepsilon\sigma$ [(T2 + 273) 4 - Tsurr Eliminating the constant C1 from the two relations above gives the following expression for the outer surface temperature T2, 4] - kC1 = h[T2 - T ∞] + $\varepsilon\sigma$ [(T2 + 273) 4 - Tsurr Eliminating the constant C1 from the two relations above gives the following expression for the outer surface temperature T2, 4] - kC1 = h[T2 - T ∞] + $\varepsilon\sigma$ [(T2 + 273) 4 - Tsurr Eliminating the constant C1 from the two relations above gives the following expression for the outer surface temperature T2, 4] - kC1 = h[T2 - T ∞] + $\varepsilon\sigma$ [(T2 + 273) 4 - Tsurr Eliminating the constant C1 from the two relations above gives the following expression for the outer surface temperature T2, 4] - kC1 = h[T2 - T ∞] + $\varepsilon\sigma$ [(T2 + 273) 4 - Tsurr Eliminating the constant C1 from the two relations above gives the following expression for the outer surface temperature T2, 4] - kC1 = h[T2 - T]] = $q\&h(T2 - T\infty) + \varepsilon\sigma[(T2 + 273) 4 - Tsurr 0 (c) Substituting the known quantities into the implicit relation above gives (30 W/m 2 · K 4)](T2 + 273) 4 - 290 4] = 100,000 W/m 2 Using an equation solver (or a trial and error approach), the outer surface temperature is determined from the relation above gives (30 W/m 2 · K 4)](T2 + 273) 4 - 290 4] = 100,000 W/m 2 Using an equation solver
(or a trial and error approach), the outer surface temperature is determined from the relation above gives (30 W/m 2 · K 4)](T2 + 273) 4 - 290 4] = 100,000 W/m 2 Using an equation solver (or a trial and error approach), the outer surface temperature is determined from the relation above gives (30 W/m 2 · K 4)](T2 + 273) 4 - 290 4] = 100,000 W/m 2 Using an equation solver (or a trial and error approach), the outer surface temperature is determined from the relation above gives (30 W/m 2 · K 4)](T2 + 273) 4 - 290 4] = 100,000 W/m 2 Using an equation solver (or a trial and error approach), the outer surface temperature is determined from the relation above gives (30 W/m 2 · K 4)](T2 + 273) 4 - 290 4] = 100,000 W/m 2 Using an equation solver (or a trial and error approach), the outer surface temperature is determined from the relation above gives (30 W/m 2 · K 4)](T2 + 273) 4 - 290 4] = 100,000 W/m 2 · K 4)$ to be T2 = 895.8°C 2-71 Chapter 2 Heat Conduction Equation 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 = $\pi D 2 = \pi (3 \text{ m}) 2 = 28.27 \text{ m} 2 1 1 \text{ Ro} = = 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ T} - \text{T} \infty 2 [15 - (-196)] \text{°C} Q \& = \text{s1} = 208,910 \text{ W} \text{ Ro} 0.00101 \text{ °C/W } 2 \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{ W/m} . \text{°C})(28.27 \text{ m} 2) \text{ Ho A} (35 \text{$ fg 198 kJ/kg Ro Ts1 T $\infty 2$ (b) The heat transfer rate and the rate of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are A = $\pi D 2 = \pi$ (31. Also, the temperature in each layer varies linearly and thus we could solve this problem by considering 3 nodes only (one 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. 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 fo 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 plates are isothermal and there is no change in the flow direction, and thus the temperature depends on y only, T = T(y). 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. = = ρ C pV ρ C p Lc (8500 kg / m3)(320 J / kg. That is, Q& rad = Q& conv = 770 - Q& co 220 = 550 W Then the fraction of heat transferred by radiation becomes f = Q& rad 550 = = 0.714 (or 71.4%) Q& cond 770 $28^{\circ}C$ 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 $k = 386 \text{ W/m} \cdot ^{\circ}\text{C}$ for copper plates and $k = 0.26 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{ W/m} \cdot ^{\circ}\text{C}$ for sheet metal and $0.035 \text{$ number, the average heat transfer coefficient and the heat transfer rate are determined to be hL Nu = $(0.037 \text{ Re L } 0.8 - 871) \text{ Pr } 1/3 = (0.037(2.163 \times 107) 0.8 - 871)(0.7340)1/3 = 2.384 \times 104 \text{ k} \text{ k} 0.02428 \text{ W/m } 2.^{\circ}\text{C} \text{ L } 10 \text{ m} \text{ As} = \text{wL} = (10 \text{ m})(4 \text{ m}) = 40 \text{ m } 2 \text{ Q} \text{ k} = \text{hA} (T - T) = (57.88 \text{ W/m } 2.^{\circ}\text{C})(40 \text{ m})(4 \text{ m}) = 40 \text{ m } 2 \text{ Q} \text{ k} = \text{hA} (T - T) = (57.88 \text{ W/m } 2.^{\circ}\text{C})(40 \text{ m})(4 \text{ m}) = 40 \text{ m } 2 \text{ Q} \text{ k} = 10 \text{ m} \text{ m} = 10 \text{ m} \text{ m} \text{ m} \text{ m} = 10 \text{ m} = 10 \text{ m} \text{ m}$ 2)(12 - 5)°C = 16,206 W = 16.21 kW s \propto s 7-4 Chapter 7 External Forced Convection 7-17 "PROBLEM 7-17" "GIVEN" Vel=55 "[km/h], 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 num15 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 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. 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. 5-66 Chapter 5 Numerical Methods in Heat flux at the right boundary (node 6). Properties The emissivity of the base surface is $\varepsilon = 0.7$. 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 nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over an isothermal plate, the wall shear stress decreases with distance from the leading edge 6-35C A major advantage of nondimensional flow over a nondimensional fl original problem involves 6 parameters (L, V, T∞, Ts, v, 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 $\Delta U = mC\Delta T = (70 \text{ kg})(3.6 \text{ kJ/kg.°C})(2°C) = 504 \text{ kJ} 1-4 \text{ Chapter 1 Basics of Heat Transfer 1-19 An electrically heated house maintained at 22°C experiences infiltration losses at a rate of 0.7 ACH$ 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 from (140 + 22) - 22 |°C = 21.2 W Q& loss = hA(Tplate, ave - T ∞) = (12 W/m 2.°C)(0.03 m 2) | 2 (JEnergy balance on the plate can be expressed as E in – E out = ΔE plate = mC p ΔT plate = mC p ΔT plate (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 transfe 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) = = $1.051 \approx 1.0 \text{ k}$ (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 $0.030\ 0.12\ 0.12\ 0.45\ 0.45\ 0.27\ 0.27\ 0.27\ 0.27\ 0.27\ 0.27\ 0.27\ 0.27\ 0.49$ ----0.94 0.079 0.12 0.12 3 Total unit thermal resistance of each section, R 1 2 The U-factor of each section, farea Overall U-factor, U = Σ farea, iUi = $0.84 \times 0.641 + 0.16 \times 0.498$ 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 - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A1e - λ 1 τ \rightarrow = 0.65 = A 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 heat flux on the surface of the resistor is $As = 2 \pi D 2 + \pi DL = 2 \pi (0.003 \text{ m}) 2 + \pi (0.003 \text{ m}) 2 + \pi (0.003 \text{ m})$ (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& $0.15 \text{ W} \rightarrow \text{T s} = \text{T} \infty + = 171^{\circ} \text{C} \text{ Q} \& = \text{hAs } (\text{T s} -
\text{T} \infty) 2 \text{ hAs } (1179 \text{ W/m} \cdot \text{C})(0.000127 \text{ m} 2 \text{ J} 3.7 \text{ Chapter } 3 \text{ Steady Heat Conduction } 3.24 \text{ A power transistor dissipates } 0.2 \text{ W}$ 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 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 $280^{\circ}F$ As = $\pi DL = \pi (4 / 12 \text{ ft})(200 \text{ ft}) = 209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Q} \text{ pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft } 2 \text{ Pipe} = hAs (Ts - Tair) = (6 \text{ Btu/h.ft } 2 \cdot \circ F)(209.4 \text{ ft }$ 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 loss)(Unit cost of energy) = (29,438 therms/yr 0.86 (100,000 Btu / Then the annual cost of the energy loss)(Unit cost of energy) = (29,438 therms/yr 0.86 (100,000 Btu / Then the annual cost of the 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 $\tau = \alpha \Delta t l^2 = (3.2 \times 10 - 6 \text{ m } 2/\text{s})(15 \text{ s}) (0.015 \text{ m}) 2 = 0.2133$ (for $\Delta t = 15 \text{ s})$ 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 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0 Node 2 (interior): 100 + 200 + 2T2 - 4T1 = 0• 2 • 3 • 4 • 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 only one inlet and one exit and thus $\Delta mCV = 0$ and $\Delta E CV = 0$, 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 t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 m, the heat conduction area at any location along the flange is t = 0.02 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 ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 1: k ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 1: k ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 1: k ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 1: k ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi
tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) T1 – T0 = 0 $\Delta x1$ Node 0: hi ($2\pi tr01$) $(\Delta x 2 / 2) \{h(T_{\infty} - T1) + \varepsilon \sigma [Tsurr - (T1 + 273) 4] \} = 0 \Delta x 2 \Delta x 2 \text{ Node } 2: k (2\pi tr23) T - T2 T1 - T2 4 + k (2\pi tr23) T - T2 T1 - T2 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T2 T1 - T2 4 + k (2\pi tr23) T - T2 T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T2 T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T2 T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T - T3 T - T3 4 + k (2\pi tr23) T$ $- T4 + k(2\pi tr56) 5 + 2(2\pi tr45) 5 + 2(2\pi tr56) 6 + 2(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 6 + 2(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + \epsilon\sigma [Tsurr - (T6 + 273) 4]\} = 0 \Delta x 2 \Delta x 2 \text{ Node } 5: k(2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi r6 t] \{h(T\infty - T6) + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2[2\pi t(\Delta x 2/2)(r56 + r6)/2 + 2\pi tr56) 75 - T6 4 + 2\pi tr56) 75 - T6 4 + 2\pi tr56) 75 - T$ where $\Delta x_1 = 0.004 \text{ m}$, $\Delta x_2 = 0.01 \text{ m}$, $k = 52 \text{ W/m} \cdot ^{\circ}\text{C}$, $\epsilon = 0.8$, $T = 8^{\circ}\text{C}$. The material becomes & $\Delta T \text{ QL}$ $(17.5 \text{ W})(0.005 \text{ m}) \text{ Q}_{\text{W}} = \text{kA} \rightarrow \text{k} = = 1.09 \text{ W} / \text{m}. 1-59 \text{ Chapter 1} \text{ Basics of Heat Transfer 1-108 The roof of a house with a gas furnace consists of a 15-cm thick concrete that is losing heat to the outdoors by radiation and convection. F)[(7 / 12) × (0.5 / 12)]ft 2 \text{ Ri} = 9 / 12 \text{ ft L} = = 5.51 \text{ h}^{\circ}\text{F/Btu kA} (0.40 \text{ Btu/h.ft.}^{\circ}\text{F})[(7 / 12) × (7 / 12)]ft 2 1 1 = 0.64$ h°F/Btu Ro = ho A (4 Btu/h.ft 2.°F)(0.3906 ft 2) 1 1 1 1 1 1 = + + = + + \rightarrow 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 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.3135 + 1.0667 + 5.315 + 1.0667 + 1.067 + 1.0667 + 1.067 transfer through entire wall becomes (30 ft)(10 ft) Q& total = (5.1053 Btu/h) = 3921 Btu/h = 3921mid + 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& = $\infty 1 \propto 2 = 11.7252$ h°F/Btu Rotal 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 = $0.27211 \text{ h}^\circ \text{F/Btu 2}$ hi Ai (35 Btu/h.ft.°F)(0.105 ft 2) ln(r2 / r1) ln(3 / 2) = $0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ
\text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (223 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (23 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (23 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (23 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (23 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (23 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (23 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL 2\pi (23 \text{ Btu/h.ft.}^\circ \text{F})(1 \text{ ft }) 1 = 0.00425 \text{ h}^\circ \text{F/Btu R}$ pipe = $2\pi kL$ 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 400 lbm/h and the length of the tube required to condense steam at a rate of 400 lbm/h (1037 Btu/lbm) = 124,440 Btu/h Tube length = Q& total 124,440 = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = Q& total 124,440 = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = Q& total 124,440 = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = Q& total 124,440 = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = Q& total 124,440 = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = Q& total 124,440 = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length = 0.27665 °F/Btu The total rate of heat transfer required to condense s = 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 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 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 = T1 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r1 (C) C1 = h|| - 1 + C2 r $T_1 + r_1 T_1 - T_{\infty} r_2 r_k r_1 1 - 2 - r_1 hr_2$ Substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (r) = $-(11)C1CT_1 - T_{\infty} + T_1 + 1 = C1|| - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 = r k r r_1 (r_1 r) - || + T_1 (r_1 r) - ||$ 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 = $-k(4\pi r 2) 21 = -4\pi k C1 = -4\pi k C = -4\pi$ 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 temperatures, and thus we need to have 6 equations to determine them uniquely. 2 Convection heat transfer coefficient is constant and uniform. 2 Heat transfer is two15°C dimensional because of symmetry about the centerplane. 2 Heat transfer at the edges and corners is two-or of air-conditioning units required is 9.17 kW = 1.83 \rightarrow 2 units 5 kW/unit 1-7 40 people 10 bulbs · Ocool 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 Tm -1 - Tm Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm)
+ \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + \epsilon\sigma (p\Delta x)[Tsurr - (Tm + 273) 4] = 0 \Delta x \Delta x 4 Tm -1 - 2Tm + Tm +1 + h(p\Delta x)(T \sim - Tm) + h h($p\Delta x 2 / kA$)(T $\infty - Tm$) + $\varepsilon\sigma$ ($p\Delta x 2 / kA$)[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 = = $(0.037 \text{ Re L } 0.8 - 871) \text{ Pr } 1/3 = [0.037(1.081 \times 107) 0.8 - 871](0.7340)1/3 = 1.336 \times 10$ $4 \text{ k} \text{ k} 0.02428 \text{ W/m}^\circ \text{C} \text{ h} = \text{Nu} = (1.336 \times 104) = 32.43 \text{ W/m} 2$, $^\circ \text{C} \text{ L} 10 \text{ m} \text{ As} = \text{wL} = (4 \text{ m})(10 \text{ m}) = 40 \text{ m} 2 \text{ O} \text{ e} = hA (T - T) = (32.43 \text{ W/m} 2 \cdot \text{°C})(40 \text{ m} 2)(12 - 5)^\circ \text{C} = 9081 \text{ W} = 9.08 \text{ kW} \text{ s} \infty \text{ s}$ If the wind velocity is doubled: $\text{V} \text{ L} [(110 \times 1000 / 3600)\text{ m/s}](10 \text{ m}) \text{ Re} \text{ L} = \infty = 2.163 \times 107 \text{ v} 1.413 \times 10 - 5 \text{ m} 2 \text{ /s} \text{ which is greater than the critical}$ Reynolds number. Therefore, the initial nodal temperatures are T00 = 20° C, T10 = 16.66° C, T20 = 13.33° C, 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 $q \& i \Delta x 2 q \& m i \Delta x 2 Tmi + 1 - Tmi = Tmi + 1 = \tau (Tmi - 1 + Tmi + 1) + (1 - 2\tau)Tmi + \tau m k \tau k Node 0$ 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 $(T2 - T\infty) \rightarrow T2 = T\infty + hAs (80 \times 0.06)$ W = 30°C + = 39.48°C (23.43 W/m 2.°C)(0.12 m)(0.18 m) A Q& L Q& = s (T1 - T2) \rightarrow T1 = T2 + L kAs (80 \times 0.06 W)(0.003 m) = 39.48°C (23.43 W/m 2.°C)(0.12 m)(0.18 m) A Q& L Q& = s (T1 - T2) \rightarrow T1 = T2 + L kAs (80 \times 0.06 W)(0.003 m) = 39.48°C + = 39.52°C (16 W/m.°C)(0.12 m)(0.18 m) A Q& L Q& = s (T1 - T2) \rightarrow T1 = T2 + L kAs (80 \times 0.06 W)(0.003 m) = 39.48°C + = 39.52°C (16 W/m.°C)(0.12 m)(0.18 m) A Q& L Q& = s (T1 - T2) \rightarrow T1 = T2 + L kAs (80 \times 0.06 W)(0.003 m) = 39.48°C + 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. \times 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 $\rho = 1.316$ kg/m3 Cp = 1.006 kJ/kg-K Pr = 0.7375 $\mu = 1.705 \times 10$ kg/m3. The remaining space between the steel plates is filled with fiberglass insulation. 4 One-guarter of the heat the person generates is lost from the head. In this case 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). $| = 0.0005 = A1e - \lambda 1 \tau = (15618 \text{ Ti} - T \infty] = 0.0005 = 0.0005 | | Ti - T \infty]$ 0.0005 \rightarrow 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 Δx (T ∞ - T1i) + kA T - T1i T2i - T1i Ti + 1 - T2i + kA 0 = $\rho A \Delta x C$ 1 $\Delta x \Delta x \Delta t$ Node 2 (interior): hp Δx (T ∞ - T2i) + kA T4i - T3i T i + 1 - T2i + kA 1 = $\rho A \Delta x C$ 1 $\Delta x \Delta x \Delta t$ Node 2 (interior): hp Δx (T ∞ - T2i) + kA T3i - T2i T i + 1 - T2i + kA 0 = $\rho A \Delta x C$ 1 $\Delta x \Delta x \Delta t$ Node 2 (interior): hp Δx (T ∞ - T2i) + kA T4i - T3i T i + 1 - T2i + kA 0 = \rho A \Delta x C 1 $\Delta x \Delta x \Delta t$ Node 2 (interior): hp Δx (T ∞ - T2i) + kA T4i - T3i T i + 1 -Node 4 (fin tip): where $A = \pi D 2 / 4 h(p\Delta x / 2 + A)(T \infty - T4i) + kA T3i - T4i T i + 1 - T4i = pA(\Delta x / 2)C 4 \Delta x \Delta t$ is the cross-sectional area and $p = \pi D$ 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 Δx 2 Tmi +1 - Tmi +1) + (1 - 2 τ)Tmi k τ The finite difference relation sfor 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: $h(T_{\infty} - T1i) + \varepsilon$ steak $(T1i + T3i) + (1 - 2\tau \text{ steak })T2i$ Node 2 (interior): $T3i + 1 = \tau$ steak $(T2i + T4i) + (1 - 2\tau \text{ steak })T3i$ $T2i - T1i \Delta x$ $T1i + 1 - T1i = (\rho C + 2T3) + (1 - 2\tau \text{ steak })T2i$) steak $\Delta x \ 2 \ \Delta t \ Node \ 4: \pi (r452 - r42) \{h(T_{\infty} - T4i) + \varepsilon \text{ plateo} [(T_{\infty} + 273) \ 4 - (T4i + 273) \ 4] \} + k \ steak (\pi r42) + k \ plate (2\pi r45\delta) \ T3i - T4i \ \Delta x \ 72i + (\rho C) \ plate (\pi r45 \ \Delta r \ \Delta t \ Node \ 5: 2\pi r5 \ \Delta r \ 4] \} + k \ plate (2\pi r45\delta) \ 4 = [(\rho C) \ steak (\pi r42) \ + k \ plate (2\pi r45\delta) \ T3i - T4i \ \Delta x \ 72i + (\rho C) \ plate (\pi r45 \ \Delta r \ \Delta t \ Node \ 5: 2\pi r5 \ \Delta r \ 4] \} + k \ plate (2\pi r45\delta) \ 4 = [(\rho C) \ steak (\pi r42) \ + k \ plate (2\pi r45\delta) \ 4] \} + k \ plate (2\pi r45\delta) \ 4 = [(\rho C) \ steak (\pi r42) \ + k \ plate (2\pi r45\delta) \ 4] \}$ (ρC) plate $(\pi r 52 \delta)$] 5 $\Delta r \Delta t$ Node 6: $2\pi [(r 56 + r 6) / 2](\Delta r / 2)$ $h(T \infty - T 6i) + \epsilon$ plate $(2\pi r 56 \delta) T 5i - T 6i + 273)$ 4 - (T 6i + 273) 4 ksteak = 1.40 W/m.°C, $\epsilon steak = 0.95$, α steak = $0.93 \times 10 - 6 \text{ m } 2 / \text{ s}$, hif = 187 kJ/kg, kplate = 401 W/m.°C, α plate = $117 \times 10 - 6 \text{ m } 2 / \text{ s}$, and $\epsilon plate = 0.905 \text{ m}$, $\Delta r = 0.0375 \text{ m}$, and $\Delta t = 5 \text{ s}$. dT 1-48C Conduction is expressed by Fourier's law of conduction as O& 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. 1-97C (a) Conduction and convection: No. (b) Conduction and radiation: Yes. The density of the cylinder is p, the specific heat is C, and the length is L. 2-61 Chapter 2 Heat Conduction Equation of order n is expressed in the most general form as y(n) + f n(x) y' + f n(x) y = 0 Each term in a linear homogeneous differential equation of order n is expressed in the most general form as y(n) + f n(x) y' + f n(x) y = 0 Each term in a linear homogeneous differential equation of order n is expressed in the most general
form as y(n) + f n(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 $Tm - 1 - 2Tm + Tm + 1 + g \& m \Delta x = 0$ k 5-66C The explicit finite difference formulation of a general interior node for transient two-dimensional $g \& i \mid 2 i + 1 i i i$ i heat conduction is given by Thode = τ (Tleft + Ttop + Tright + Tbottom) + $(1 - 4\tau)$ Thode + 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 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 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. At -100°C, $\mu = 1.189 \times 10 - 5$ 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 \approx Lt \approx 10D in turbulent flow. 2 The person 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 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 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 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 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 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 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 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 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 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 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 by radiation is not constance and heat transfer by radiation is not constance and heat transfer by radiat 12ft -(1.50 / 12)ft = 0.5584 h.°F/Btu Rair $= 2.1 = 4\pi kr1 r2 4\pi (0.015$ Btu/h.ft.°F)[(1.52 + 0.12) / 12]ft -(1.52 / 12)ft = 1.3134 h°F/Btu Roov $= hA (17.2 \text{ Btu/h.ft } 2.°F)\pi (3.28 / 12) 2$ ft 2 R total = Rair + Rtowel + 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 $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu $\Delta t = 0.598$ h = 35.9 min Q& 84.9 Btu/h Rtotal 2.1195 h.°F/Btu properties of the plate are given to be k = 1.2 Btu/h·ft·°F and $\epsilon = 0.80$, and $\alpha s = 0.45$. the average Nusselt number and heat transfer coefficient for all the tubes in tubes in the The number of copper fillings in the board and the area they comprise are Atotal = $(6 / 12 \text{ ft})(8 / 12 \text{ ft}) = 0.333 \text{ m} 2 \text{ n copper} = n \pi D 2 4 = 13,333 \text{ Aepoxy} = Atotal - Acopper \pi (0.02 / 12 \text{ ft}) = 0.3042 \text{ ft} 2 4 = 0.3042 \text{ ft} 2 \text{ Rcopper The thermal}$

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 1 = + = + \rightarrow Rboard = 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042 ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042) ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042) ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042) ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042) ft 2) Rcopper = Repoxy 0.00064 h.°F/Btu kA (0.10 Btu/h.ft.°F)(0.3042) ft 2) Rcopper 42 Chapter 3 Steady Heat Conduction in Cylinders and Spheres 3-64C When the diameter of cylinder. 6-21C Reynolds number is the ratio of the inertial forces to viscous forces, and it serves as a criteria for determining the flow regime. The total drag force and the rate of heat transfer per unit width of the plate are to be determined. Properties The conductivity and diffusivity are given to be $k = 0.39 \times 10 - 6 \text{ m } 2 / \text{ s}$. Fin effectiveness is defined as the ratio of heat transfer rate from the same surface if there were no fins, and its value is expected to be greater than 1. Insulate qL Analysis The nodal spacing is given to be •4 •5 •6 •7 $\Delta x = \Delta x = l = 0.015$ m. Noting that heat conduction is two dimensional and assuming no heat generation, an energy balance on this element during a small time interval Δt can be expressed as Rate of heat (Rate of heat conduction) (Rate of heat conduction) can be expressed as ΔE element = E t + Δt - E t = mC (Tt + Δt - Tt) = $\rho C \Delta x \Delta y$ (Tt + Δt - Tt) Substituting, T - Tt Q& x + Δx - Q& y + Δy - Q& x + Δx - Q& y + Δy - Q& x + $\Delta x \Delta y \Delta t$ Taking the thermal conductivity k to be constant and noting that the heat transfer surface areas of the element for heat conduction in the x and y directions are $Ax = \Delta y \times 1$ and $Ay = \Delta x \times 1$, respectively, and taking the limit as Δx , Δy , and $\Delta t \rightarrow 0$ yields $\partial 2 T \partial 2 T 1 \partial T + = \partial x 2 \partial y 2 \alpha \partial t$ since, from the definition of the derivative and Fourier's law of heat conduction, 1 Q& x + $\Delta x - Q$ & x 1 $\partial Q \times 1 \partial (\partial T) \partial (\partial T$ $= | = -k 2 | -k \Delta y \Delta z | = -|k \Delta x \rightarrow 0 \Delta y \Delta z \partial x \Delta z \partial x \Delta z \partial y \Delta z \partial x \Delta z \partial y \Delta z \partial x \Delta z \partial y \Delta z \partial x \Delta z$ energy input) Furnace efficiency (1760 kJ/h)(0.58/therm) (1 therm) || || $0.82 \ 105,500 \ kJ \ = 0.012/h$ Discussion The heat loss from the heating ducts in the attic is costing the homeowner 1.2 cents per hour. 4-13a, where $\rho C p = 11$ = 52.6 | Bi $0.019 \ \alpha t$ | $\tau = 2 = 15 > 0.2 \ T o - T \infty 0 - 50 \ ro = = 0.769$ |] Ti - T $\infty - 15 - 50$ Then, t= τro $2 (15)(0.02 \text{ m}) 2 = 513 \text{ s} \alpha (1.17 \times 10 - 5 \text{ m} 2/\text{s})$ The difference is due to the reading error of the chart. = = 0152 °C / W h2 A (20 W / m. Analysis We take the water tank as the system. 3-12 Chapter 3 Steady Heat Conduction 3-31 An exposed hot surface of an industrial natural gas furnace is to be insulated to reduce the heat loss through that section of the wall by 90 percent. Then the time it will take for the column surface temperature to rise to 27° C becomes t = $\tau ro 2$ (0.6253)(0.15 m) 2 = 23,685 s = 6.6 hours α (5.94 × 10 - 7 m 2/s) (b) The heat transfer to the column will stop when the center temperature of column reaches to the ambient temperature, which is 28° C. Note that all 0°C side surfaces are at T0 = 0°C, and there are 8 nodes with unknown temperatures. Relate Repoxy Repoxy T1 Rcontact T2 Rcontact Generalized Thermal Resistances indicate simultaneous heat transfer (such as convection and radiation on a surface). The mass flow rate of water through the heater is to be determined. 3-98C Fins should be attached on the air side since the convection heat transfer coefficient is lower on the air side than it is on the water side. 6-44 was determined to be 2 d 2T (V) μ k = - (1) || dy 2 (L/ L Fluid The steady one-dimensional heat conduction equation with constant heat generation is d 2 T g& 0 (2) + = 0 k dy 2 Comparing the two equation above, the volumetric heat generation rate is determined to be 2 (V) g& 0 = $\mu \mid |L|$ Integrating Eq. (2) twice gives g& dT = -0 y + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) = -0 y 2 + C3 dy k g& T (y) $y=LT(L) = T0 \rightarrow C4 = T0 + T(y) = T0 + g \& 0 L2 2k (y2) | 1 - | | L2 | | Maximum temperature occurs at y = 0, and it value is g \& L2 Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result Tmax = T(0) = T0 + 0 2k which is equivalent to the result
Tmax = T(0) = T0 + 0 2k which is equivalent to the result$ exposed to convection from both sides. 7-81 Chapter 7 External Forced Convection 7-89 The plumbing system of a plant involves some section of a plant involves s factor are based on the hydraulic diameter Dh defined as Dh = 4 Ac where Ac is the cross-sectional p area of the tube and p is its perimeter. $(0.2 \text{ m}) = 9.55 \text{ kg}/(20 - 1200)^{\circ}C = 10,100 \text{ kJ}$ [] Then we determine the dimensionless heat transfer ratios for both geometries as $(Q \mid Q \mid max) \sin(\lambda \mid ratio)$ 1) sin(0.2568) | = 1 - θ o, wall = 1 - (0.7627) = 0.2457 | $\lambda 1 0.2568$ / wall (Q | Q | max | Q | | | plane + | Q | wall | max | | (| Q | | long | 1 - | Q | cylinder | (0.7627) = 0.2425 | $\lambda 0 0.2217 1$ / cylinder | (0.7627) = 0.2425 | $\lambda 0 0.2217 1$ / cylinder | (0.7627) = 0.2457 | $\lambda 1 0.2568$ / wall (Q | Q | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | wall | max | Q | | | plane + | Q | | plane + | Q | | plane + | Q | wall | max | Q | | | plane + | Q | | plane + | plan max [4-87 L z r0 Cylinder L Ti = 20° C] | | plane | = 0.2457 + (0.2425)(1 - 0.2457) = 0.4286 / wall | Chapter 4 Transfer from the short cylinder as it is cooled from 300°C at the center to 20° C becomes Q = 0.4255Q max = (0.4286)(10,100 kJ) = 4239 kJ which is identical to the heat transfer to the cylinder as the cylinder at 20°C is heated to 300°C at the center. The percent error involved in the calculation of heat gain through the window assuming the window a (290 - 150) K = 252 W Q& total = Q& cond = kA 1 2 = (0.036 W / m.o C)(1 m 2) 0.02 m L (d) In the case of superinsulation, the rate of heat transfer will be T - T (290 - 150) K = 1.05 W Q& total = Q& cond = kA 1 2 = (0.00015 W / m. Analysis We consider heat loss through the walls, door, and the top and bottom sections of an oven is transient in nature since the thermal conductivity of the glass and air are given to be kglass = 0.78 W/m.°C and kair = 0.026 W/m.°C. Problems designated with the CD icon in the text are also solved with the EES software, and electronic solutions complete with parametric studies are available on the CD that accompanies the text. °C αC p α = 4-104 Chapter 4 Transient Heat Conduction 4-114 The time it will take for the diameter of a raindrop to reduce to a certain value as it falls through ambient air is to be determined. Analysis The depth at which the temperature drops to x Soil 0°C in 75 days is determined using the analytical Ti = 15° C solution, (x T (x, t) - Ti = erfc| Ts - Ti 2 \alpha t) | | / Water pipe Substituting, () 0 - $15 \times |$ | = erfc| - 10 - 15 | 2 ($1.4 \times 10 - 5 \text{ m } 2$ /s)(75 day × 24 h/day × 3600 s/h) || / $\rightarrow x = 7.05 \text{ m}$ Therefore, the pipes must be buried at a depth of at least 7.05 m. $447 \times 10 \upsilon 1.382 \times 10 - 5 m 2/s$ which is greater than the critical Reynolds number. 4-23 in this case for convenience (instead of the analytic solution), 1 - Therefore, T (x, t) - T ∞ 3 - (-12) | h at =1 | x | k ξ = =0 | 2 at t = (1) 2 k 2 h 2a = $(0.607 \text{ W/m.°C}) 2 (30 \text{ W/m} 2.°C) 2 (0.146 \times 10 - 6 \text{ m } 2/s) 4-103 \text{ Freezer T} = -12°C = 2804 \text{ s} = 46.7 \text{ min Watermelon Ti} = 25°C \text{ Chapter 4 Transient Heat Conduction 4-113 A cylindrical rod is dropped into boiling water. Outside surface, 15 mph wind 2. 4-49 Chapter 4 Transient Heat Conduction 4-56 The center temperature of a beef carcass is to be$ lowered to 4°C during cooling. The time it will take for this heater to raise the water temperature to 80°C as well as the convection heat transfer coefficients at the beginning and at the end of the heating process are to be determined. Nodes 2 and 3 are interior nodes, and thus for them we can use the general explicit finite difference relation expressed as $Tmi-1 - 2Tmi + Tmi+1 + g\&mi \Delta x \ 2Tmi+1 - Tmi = \rightarrow Tmi + 1 = \tau \ (Tmi -1 + Tmi + 1) + (1 - 2\tau) Tmi \ k \ \tau \ since there is no heat generation.$ allowable value of the time step is determined to be $\Delta t \leq (0.1 \text{ m}) 2 4(12 \times 10 - 6 \text{ m} 2/s)[1 + (45 \text{ W/m} 2 \cdot \text{C})(0.1 \text{ m}) / (28 \text{ W/m} \cdot \text{C})] = 179 \text{ s}$ Therefore, any time step less than 179 s can be used to solve this problem. m2 1 (50 W / m. at r = r0 : 6k 6k Substituting this C2 relation into Eq. (b) and rearranging give g& 2 T (r) = Ts + (r0 - r 2) 6k which is the desired solution for the temperature distribution in the wire as a function of r. Substituting these values into the one-term solution gives $\theta = 2 2$ To $-T \propto T - (-10) = 1.998e - (3.094) (0.2063) = 0.277 \rightarrow T0 = -3.1^{\circ}C = A1e - \lambda 1\tau \rightarrow 0$ Ti $-T \propto 15 - (-10)$ The lowest temperature during cooling will occur on the surface (r/r0 = 1), and is determined to be T (r) - T ∞ sin(λ 1r0 / r0) T0 - T ∞ sin(λ 1r0 / r0) T (r0) - T ∞ 2 sin(λ 1r / r0) = $= \theta 0$ = A1e - λ 1r / r0 λ 1r0 / r0 Ti - T ∞ Sin(λ 1r0 / r0 Ti - T ∞ Ti 2.8°C, and even the center temperature of chicken is below this value. The thermal conductivity of a material is a measure of how fast heat will be conducted in that material. the equation + = + has 2 dependent (T and W) and 2 $\partial x \alpha \partial t \alpha \partial t \partial x 2$ independent variables (x and t). Analysis The thermal resistance network and the individual thermal resistances are R1 R2 R3 R4 R6 R5 R7 R8 R9 R0 T ∞ 2 T1 L 0.0001 m = 0.0007 ° C / W kA (013 . The amount of energy and money saved = (0.90)(1021 therms/yr) = 919 therms/yr Q gas = Money saved = (0.90)(1021 therms/yr) = 919 therms/yr) = 919 therms/yr) = 919 therms/yr) = 919 therms/yr Q gas = Money saved = (0.90)(1021 therms/yr) = 919 therms/yr) Forced Convection 7-53 A steam pipe is exposed to light winds in the atmosphere. A person whose left side is exposed to a cold window, for example, will feel like heat is being drained from that side of his or her body. There is no change in transverse direction. 2 Thermal conductivity is given to be constant. 3 Thermal properties of the wall and the heat transfer coefficients are constant except the one at the boundaries do not change with time significantly. 2 Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. Assuming the heat transfer through the wall to be steady and one-dimensional, the mathematical formulation (the differential equation and the boundary and initial conditions) of this heat conduction problem is to be obtained. The temperature difference across the chip in steady operation is to be determined. 2-107C Geometrically, the derivative of a function y(x) at a point represents the slope of the tangent line to the graph of the function at that point. It allows us to calculate the heat transfer coefficient from a knowledge of pC pV friction coefficient. The growth of microorganisms in foods can be retarded by keeping the temperature below 4°C and relative humidity below 60 percent. 6-36 and 6-37) reduce to 2 (∂u) d 2T (V) || || + $\mu \rightarrow k = -\mu$ | 2 2 ∂y dy (L/ (∂y) since $\partial u / \partial y = V / L$. 4-47 Chapter 4 Transient Heat Conduction 4-55E The center temperature of oranges is to be lowered to 40°F during cooling. 2-108C The order of a derivative represents the number of times a function is differentiated, whereas the degree of a derivative represents how many times a derivative is multiplied by itself. The unknown nodal temperatures and the rate of heat loss from the top surface are to be determined. Properties The gas constant of air is R = 0.3704 psia.ft3/lbm.R = 0.06855 Btu/lbm.R = 0.06855 Btu/l and the thermal conductivity to be constant. ° C)($2\pi r 3 m 2$) 138.2r3 Noting that all resistances are in series, the total resistance is R total = Ri + R1 + R2 + Ro = $0.09944 + 0.00043 + 4.188 \ln(r3 / 0.023) + 1/(138.2r3)$ °C/W Then the steady rate of heat loss from the stea 0.023 + 1 /(138.2r3)]°C/W R total Noting that the outer surface temperature of insulation is specified to be 30°C, the rate of heat loss can also be expressed as T -T (30 - 22)°C = 1106r3 O& = 3 o = Ro 1 / (138.2r3)°C / W Setting the two relations above equal to each other and solving for r3 gives r3 = 0.0362 m. 5 Heat transfer coefficient accounts for the radiation effects, if any. Properties The properties of hot dog available are given to be $\rho = 980 \text{ kg/m3}$ and Cp = 3900 J/kg.°C. °C)(0.0002 m) -1 T (t) - T $\infty = e - (t - 1.195 \text{ s}) t \rightarrow t = 38.5 \text{ s}$ Ti $- T \infty 4-3$ Junction D T(t) Chapter 4 Transient Heat Conduction 4-15E A number of brass balls are to be quenched in a water bath at a specified rate. Applying the boundary conditions give r = r1: $-k C1 = hi [Ti - (C1 \ln r1 + C2)]r1 r = r2$: $-k C1 = ho [(C1 \ln r2 + C2) - To] r2$ Solving for C1 and C2 simultaneously gives C1 = T0 - Ti r k k + ln 2 + r1 hi r1 ho r2 and (k) T0 - Ti = Ti - C2 = Ti - C1 || ln r1 - |r k k hi r1 / (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k)
| ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 |/ (+ ln 2 + r1 hi r1 ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) | ln r1 - || hi r1 | ho r2 (k) Substituting C1 and C2 into the general solution and simplifying, we get the variation of temperature to be r k + k r 1 hi $r 1 r (r) = C1 \ln r + r 1$ hi $r 1 r (r) = C1 \ln r + r 1$ hi $r 1 \ln r + r 1$ h temperature in the pipe, and the surface temperatures are to be determined for steady one-dimensional heat transfer. m Then the steady rate of heat transfer. m to be varied" k=1.4 "[W/m-C]" h=4 "[m]" T_1=140 "[C]" T_2=15 "[C]" "ANALYSIS" z=h+D/2 S=(2*pi*D)/(1-0.25*D/z) Q_dot=S*k*(T_1-T_2) D [m] 0.5 1 1.5 2 2.5 3 3.5 4 4.5 5 Q [W] 566.4 1164 1791 2443 3120 3820 4539 5278 6034 6807 7000 6000 Q [W] 5000 4000 3000 2000 1000 0 0.5 1 1.5 2 2.5 3 D [m] 3-99 3.5 4 4.5 5 Chapter 3 Steady Heat Conduction 3-133 Hot water passes through a row of 8 parallel pipes placed vertically in the middle of a concrete wall whose surfaces are exposed to a medium at 20 °C. 2 Thermal conductivity is given to be variable. Substituting these values, the nodal temperatures in the pond after 4×(60/15) = 16 time steps (4 h) are determined to be T0 = 16.5° C, T1 = 15.6° C, T2 = 15.3° C, and T4 = 20.2° C. Then the average Nusselt number of heat transfer coefficient for all the tubes in tubes in the tubes in tube tubes is $N = NL \times NT = 8 \times 8 = 64$. 3 Heat loss from the insulated tube is negligible. Properties of copper are given to be k = 386 W/m.°C, $\rho = 8950$ kg/m3, Cp = 0.383 kJ/kg.°C, and $\alpha = 1.13 \times 10-4$ m2/s. The time it will take to roast this rib to medium level is also to be determined. kW)(365 × 8 h / yr) = 438 kWh / yr Annual Cost = (438 kWh / yr)(\$0.08 / kWh) = \$35.04 / yr 1-15 A 1200 W iron is left on the ironing board with its base exposed to the air. This is a closed system since no mass crosses the system boundary during the process. Analysis The shape factor for this configuration is given in T2 = 60°F Table 3-5 to be $2\pi L S$ total = $4 \times 2\pi Z h (2w T1 = 350°F ln sinh | w / (\pi D 15) Process$. ft $2\pi(3 \text{ ft}) = 4 \times = 0.5298 \text{ D} = 1 \text{ in} \left(2(8/12 \text{ ft}) 2\pi(15 \text{ ft}) \right) || \ln || \sinh(8/12 \text{ ft}) L = 3 \text{ ft}$ Then the steady rate of heat transfer from the fuel rods 8 in becomes Q& = S k (T - T) = (0.5298 \text{ ft})(0.6 \text{ Btu/h.ft.}°F)(350 - 60)°C = 92.2 \text{ Btu/h total} 1 2 3-127 \text{ Hot water flows through a 5-m long section of a thin walled hot water pipe that passes} through the center of a 14-cm thick wall filled with fiberglass insulation. Then, T1 = T3 = T7 = T9 T2 = T4 = T6 = T8 Therefore, there are only 3 equations to determine them uniquely. Analysis This rectangular ice block can be treated as a short rectangular block that can physically be formed by the intersection of two infinite plane wall of thickness 2L = 10 cm. 6 The heat storage capacity of the plate is small relative to the amount of total heat transferred to the steak, and thus the heat transferred to the plate can be assumed to be transferred to the steak. Properties The gas constant of air is R = 0.287 kPa.m3/kg.K (Table A-1). 3-5C The combined heat transfers on a surface, and is defined as hcombined = hconvection + hradiation. The corresponding rates for women are about 30 percent lower. Properties The properties of air at 1 atm and the film temperature of $(Ts + T\infty)/2 = (86+54)/2 = 70^{\circ}F$ are (Table A-15E) k = 0.01457 Btu/h.ft. The rate of heat loss from the air in the duct to the attic and its cost under steady conditions are to be determined. Properties The thermal conductivities of various materials used are given to be kA = kF = 2, kB = 8, kC = 20 kD = 15, and $kE = 35 \text{ W/m} \cdot \text{°C}$. Analysis We assume the inner surface of the pipe to be at 0°C at all times. When there is no heat generation, an energy balance on this thin spherical shell element of thickness Δr during a small time interval Δt can be expressed as ΔE element Q& r - Q at $r + \Delta r = \Delta t$ where ΔE element of thickness Δr during a small time interval Δt can be expressed as ΔE element Q& r - Q at $r + \Delta r = \Delta t$ where ΔE element = Et + $\Delta t - Et = mC$ (Tt + $\Delta t - Tt$) $= \rho CA\Delta r$ (Tt + $\Delta t - Tt$) Substituting, $-TT \& \Delta r = \rho CA\Delta r t + \Delta t t Q\&r - Q\&r + \Delta r + gA \Delta t$ where $A = 4\pi r 2$. The Fourier number is $\tau = \alpha t 2 L = (33.9 \times 10 - 6 m 2 / s)(10 min \times 60 s / min) (0.015 m) 2 = 90.4 > 0.2$ Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. 2-74C The rate of heat generation inside $0.3770 \text{ m } 2 \text{ A3} = \pi \text{D3} \text{ L} = \pi \text{D3} (1 \text{ m}) = 3.1416 \text{ D3} \text{ m } 2 \text{ Ri} \text{ R2} \text{ T} \infty 1$ The individual thermal resistances are 1 1 Ri = = 0.03031 °C/W hi Ai (105 W/m 2 . °C)(0.3142 m 2) R1 = R pipe = Ro T \sigma 2 \text{ ln}(r2 / r1) ln(6 / 5) = = $0.00048 \text{ °C/W} 2\pi \text{k1} \text{ L} 2\pi (61 \text{ W/m} \cdot \text{C})(1 \text{ m}) \text{ R} 2 = \text{Risulation} = \ln(r3 / r2) ln(D3 / 0.12) ln(D3 / 0.12) = \circ \text{C/W} 0.23876 2\pi \text{k} 2 \text{ L}$ $2\pi(0.038 \text{ W/m.°C})(1 \text{ m}) 11 = 0.18947 \text{ °C/W } 2 \text{ ho Ao} (14 \text{ W/m .°C})(0.3770 \text{ m} 2) 11 0.02274 = = \text{°C/W } 2 2 \text{ ho Ao} (14 \text{ W/m .°C})(3.1416 \text{ D3 m}) D3 \text{ Ro,steel} = R0, \text{insulation} = Ri + R1 + R0, \text{steel} = 0.03031 + 0.00048 + 0.18947 = 0.22026 \text{ °C/W } R \text{ total}, \text{ insulation} = Ri + R1 + R2 + R0, \text{insulation} = Ri + R1 + R0, \text{steel} = 0.03031 + 0.00048 + 0.18947 = 0.22026 \text{ °C/W } R \text{ total}, \text{ insulation} = Ri + R1 + R2 + R0, \text{insulation} = Ri + R1 + R2 + R0, \text{insulation} = 0.03031 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.00048 + 0.000$ + ln(D3 / 0.12) 0.02274 + D3 0.23876 ln(D3 / 0.12) 0.02274 + °C/W D3 0.23876 Then the steady rate of heat loss from the steady part of heat loss can be determined from T – T $\propto 2$ (235 – 20)°C \rightarrow (0.05 × 976.1) W = Q& insulation = $\infty 1$ R total, insulation is \rightarrow thickness = D3 = 0.3355 m D3 - D 2 33.55 - 12 = = 10.78 cm 2 2 3-133 Chapter 3 Steady Heat Conduction (b) The thickness of the insulation needed that would maintain the outer surface of the insulation at a maximum temperature of 40°C can be determined from T – T T – T ≈ 2 Q& insulation = $\infty 1 \propto 2 = 2$ R total, insulation Ro, insulation $\rightarrow (235 - 20)^{\circ}C$ (ln (D3 / 0.12) 0.02274 | 0.03079 + + | D3 0.23876 () | $^{\circ}C/W$ | $/ = (40 - 20)^{\circ}C$ ($^{\circ}NW$ D3 whose solution is \rightarrow thickness = D3 = 0.1644 m D3 - D 2 16.44 - 12 = 2.22 cm 2 2 3-173 A 6-m-diameter spherical tank filled with liquefied natural gas (LNG) at -160°C is exposed to ambient air. Otherwise, a body tends to block the flow, and is said to be blunt. This is because steady conditions are reached in about 156 min, and the
inner surface temperature at that time is determined to be 48.0°F. °C $(1 \text{ m2})(100 - 30)^\circ \text{C} = 2450 \text{ W}$ Then the fin effectiveness becomes Q& 17,400 ε fin = fin = 7.10 & 2450 Q no fin 3-86 3 cm 0.6 cm D=0.25 cm Chapter 3 Steady Heat Conduction 3-117 A hot plate is to be cooled by attaching aluminum pin fins on one side. ° C L AAT (0.01 m 2)(8° C) 1-29 A L Chapter 1 Basics of Heat Transfer 1-69 The therma conductivity of a material is to be determined by ensuring conditions are reached. Analysis Noting that the cross-sectional area of the spoon is constant and measuring x from the free surface of water, the variation of temperature along the spoon can be expressed as T (x) - T ∞ cosh a (L - x) = Tb - T ∞ cosh aL h, T ∞ where Ac = (0.5 / 12 ft)(0.08 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft) = 0.000278 ft 2 a = hp = kAc atmospheric air. 2 The thermal properties of the wall are constant. $^{\circ}C)[-2.8 - (-18)]^{\circ}C = 175.6 \text{ kJ}$ Therefore, the total amount of heat removal per turkey is Q total = Q cooling, fresh + Q freezing + Q cooling, frozen = 79.3 + 1498 + 175.6 \approx 1753kJ (b) Assuming only 90 percent of the water content of turkey is frozen, the amount of heat that needs that needs the total amount of heat that needs the total amount of heat the total amount of heat that needs the total amount of heat to be removed from the turkey as it is cooled from 1°C to -18°C is Cooling to -2.8°C: Qcooling, fresh = $(mC \Delta T)$ fresh = $(7 \times 0.9 \text{ kg})(214 \text{ kJ} / \text{kg}) = 1,348 \text{ kJ}$ Cooling -18°C: Qcooling, frozen = $(mC \Delta T)$ fresh = $(7 \times 0.9 \text{ kg})(214 \text{ kJ} / \text{kg}) = 1,348 \text{ kJ}$ Cooling -18°C: Qcooling, frozen = $(mC \Delta T)$ fresh = $(7 \times 0.9 \text{ kg})(214 \text{ kJ} / \text{kg}) = 1,348 \text{ kJ}$ Cooling -18°C: Qcooling, frozen = $(mC \Delta T)$ fresh = $(7 \times 0.9 \text{ kg})(214 \text{ kJ} / \text{kg}) = 1,348 \text{ kJ}$ Cooling -18°C: Qcooling, frozen = $(mC \Delta T)$ fresh = $(7 \times 0.9 \text{ kg})(214 \text{ kJ} / \text{kg}) = 1,348 \text{ kJ}$ Cooling -18°C: Qcooling, frozen = $(mC \Delta T)$ fresh = $(7 \times 0.9 \text{ kg})(214 \text{ kJ} / \text{kg}) = 1,348 \text{ kJ}$ Cooling -18°C: Qcooling, frozen = $(mC \Delta T)$ frozen = kJ Qcooling, unfrozen = (mC ΔT) fresh = (7 × 0.1kg)(2.98 kJ/kg.^o C)[-2.8 - (-18)^o C] = 31.7 kJ Therefore, the total amount of heat removal per turkey is Q total = Q cooling, frozen & unfrozen = 79.3 + 1348 + 158 + 31.7 = 1617 kJ 4-95 Chapter 4 Transient Heat Conduction 4-102E Chickens are to be frozen by refrigerated air. 5 W/m C // (c) The outer surface temperature is determined by direct substitution to be (1 Outer surface (r = r2): T (r2) = 100 + 23.87 | 2.5 - | = 101.5°C 0.41 //) Noting that the maximum rate of heat supply to the water is $0.9 \times 500 \text{ W} = 450 \text{ W}$, water can be heated from 20 to 100°C at a rate of 0.450 kJ / s Q& & p $\Delta T \rightarrow m$ = = 0.00134 kg / s = 4.84 kg / h Q& = mC C p ΔT (4.185 kJ / kg·° C)(100 - 20)° C 2-35 r Chapter 2 Heat Conduction Equation 2-71 "GIVEN" r 1=0.40 "[m]" k=1.5 "[W/m-C]" T 1=100 "[C]" Q dot=500 "[W]" f loss=0.10 "ANALYSIS" q dot s=((1-1)) = 0.40 [m]" r 2=0.41 [m]" k=1.5 [W/m-C]" T 1=100 [C]" Q dot=500 [W]" f loss=0.10 [ANALYSIS" q dot s=((1-1)) = 0.40 [m]" r 2=0.41 [m]" k=1.5 [W/m-C]" T 1=100 [C]" Q dot=500 [W]" f loss=0.10 [ANALYSIS" q dot s=((1-1)) = 0.40 [m]" r 2=0.41 [m]" k=1.5 [W/m-C]" T 1=100 [C]" Q dot=500 [W]" f loss=0.10 [W/m-C]" T 1=0.40 [m]" r 2=0.41 [m]" k=1.5 [W/m-C]" T 1=0.40 [m]" r 2=0.41 [m]" r 2=0.41 [m]" k=1.5 [W/m-C]" T 1=0.40 [m]" r 2=0.41 [m]" r $f_{0.401}/A = 4*pi*r_{2}^2 T = T_{1+(1/r_{1}-1/r)*(q \text{ dot } s*r_{2}^2)/k}$ "Variation of temperature" "r is the parameter to be varied" r [m] 0.4 0.4011 0.4022 0.4033 0.4044 0.4056 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 101 101.1 101.3 101.5 101.6 101.4 101.2 T [C] 101 100.8 100.6 100.4 100.2 100 0.4 0.402 0.4040 0.4066 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 101 101.1 101.3 101.5 101.6 101.4 101.2 T [C] 101 100.8 100.6 100.4 0.402 0.4040 0.4066 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 101 101.1 101.3 101.5 101.6 101.4 101.2 T [C] 101 100.8 100.6 100.4 100.2 100.3 0.4040 0.4066 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 101 101.1 101.3 101.5 101.6 101.4 101.2 T [C] 101 100.8 100.6 100.4 100.2 100 0.4 0.402 0.4040 0.4066 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 100 0.4004 0.4066 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 100.6 100.4 100.2 100 0.4 0.402 0.404 0.4066 0.4066 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 100 0.4 0.402 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 100 0.4 0.4056 0.4067 0.4078 0.4089 0.41 T [C] 100 100.2 100.3 100.5 100.7 100.8 100 0.4 0.4056 0.4067 0.4089 0.41 T [C] 100 100.2 100 0.4 0.402 0.404 0.4066 0.4066 0.4067 0.4089 0.41 T [C] 100 100.2 100 0.4 0.402 0.404 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 0.4066 [m] 2-37 0.408 0.41 Chapter 2 Heat Conduction Equation in a solid is simply the conversion of some form of energy into sensible heat energy. 2-56 r Chapter 2 Heat Conduction Equation 2-101 A spherical shell with variable conductivity is subjected to specified temperatures on both sides. Properties The thermal conductivity and thermal diffusivity of oranges are given to be k = 0.26 Btu/h·ft·°F and α = 1.4×10-6 ft2/s. 4 The arm is treated as a 2-ft-long and 3-in.-diameter cylinder with insulated ends. Properties The specific heats of water, copper, and the iron at room temperature are Cp, water = 1.0 Btu/lbm·°F, Cp, Copper = 0.092 Btu/lbm.°F, and Cp, iron = 0.107 Btu/lbm.°F (Tables A-3E and A-9E). 6-31C For steady two-dimensional flow over an isothermal flat plate in the x-direction, the boundary conditions for the velocity components u and v, and the temperature T at the plate surface and at the edge of the boundary layer are expressed as follows: $T \propto u \propto$, $T \propto At y = 0$: u(x, 0)= 0, v(x, 0) = 0, $T(x, 0) = T \propto \infty$ 6-32C An independent variable to transforming a set of partial differential equation is called a similarity variable. 2 The density, specific heat, and thermal conductivity of the body are constant. The minimum initial temperature of the water is to be determined if it to meet the heating requirements of this room for a 24-h period. Complete Solution Manual to Accompany HEAT TRANSFER A Practical Approach YUNUS A. Using the proper relation in laminar flow for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be hL = 0.664 Re L 0.5 Pr $1/3 = 0.664(1.899 \times 105)$ 0.5 (0.7202)1/3 = 259.7 Nu = k k 0.0282 W/m.°C h = Nu = (259.7) = 6.07 W/m 2.°C L 1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C L 1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C L 1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C L 1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C L 1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C L 1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞
- Ts) = (6.07 W/m 2.°C L 1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C (1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C (1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C (1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C (1.2 m As = 2 LW = 2(1.2 m)(0.5 m) = 1.2 m 2 Q& conv = hAs (T ∞ - Ts) = (6.07 W/m 2.°C (1.2 m As = 2 LW = 2(1.2 m)(1.2 m)(1.2 m)(1.2 m)(1.2 m)(1.2 m)(1.2 m)(1.2 m)(1.2 m)(1.of a train in motion is absorbing solar radiation. Analysis The resistance heater converts electric energy into heat at a rate of 2 kW. kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / yr Qin, ins = Qins / η oven = (1714 × 108 kJ / η oven = (1714 × 10 furniture and other belongings is negligible. Therefore, the wind-chill factor in this case is Fwind-chill = 20 - (-12.7) = $32.7^{\circ}C$ 1-128 The backside of the thin metal plate is insulated and the front side is exposed to solar radiation. energies & $1 = Q_{\odot}$ out + mh & 2 (since $\Delta ke \cong \Delta pe \cong 0$) mh & $k \neq (T1 - T2)$ Qout = mC 3 m/s The density of air at the ned from the ideal gas relation to be P 100 kPa $\rho = = 1031$. Properties The thermal conductivity is given to be k = 23 W/m·°C. Brick Analysis Under steady conditions, the rate of heat transfer through the wallwall is $(20 - 5)^{\circ}C \Delta T O \& cond = kA = (0.69W/m \cdot °C)(5 \times 6m 2) = 1035W L 0.3m$ inner and outer surfaces of a window glass are maintained at specified temperatures. Nodes 2, 3, 4, and 5 are Ti 4• roof interior nodes, and thus for them we can use the general 3 explicit finite difference relation \rightarrow Tm = τ K τ ϵ Convection g& mi Δx 2 i +1 i i Radiation \rightarrow Tm = τ $(Tm - 1 + Tm + 1) + (1 - 2\tau)Tm + \tau h$, Ti k The finite difference equations for nodes 1 and 6 subjected to convection and radiation are obtained from an energy balance by taking the direction of all heat transfers to be towards the node under consideration: Ti - T1i Δx T1i + 1 - T 273) $4 = \rho C \Delta x 2 \Delta t \text{ Node 2 (interior)} : T2i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 3 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 4 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 4 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + T5i) + (1 - 2\tau)T3i \text{ Node 5 (interior)} : T3i + 1 = \tau (T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 = \tau)T3i + 1 = \tau (T3i + 1 =$ $\alpha = k / \rho C = 7.4 \times 10 - 6$ ft 2 / s, Ti = 70°F, Twall = 530 R, Tsky = 445 R, hi = 0.9 Btu/h.ft2.°F, $\Delta x = 1/12$ ft, and $\Delta t = 5$ min. The relation is to be simplified for circular fin of diameter D and for a rectangular fin of thickness t. Properties of concrete are given to be k = 0.79 W/m.°C, $\alpha = 5.94 \times 10-7$ m2/s, $\rho = 1600$ kg/m3 and Cp = 0.84 kJ/kg.°C Analysis (a) The Biot number is 3 0 hro (14 W/m 2.°C)(0.15 m) Bi = = 2.658 k (0.79 W/m.°C) The constant λ 1 and A1 = 1.3915 C o l A i r Once the constant J 0 = 0.3841 is determined from Table 4-2 corresponding to the constant λ 1, the Fourier number is determined to be 2 2 T (ro, t) $-T \propto 27 - 28 = A1 e - \lambda 1 \tau J 0$ ($\lambda 1 ro / ro$) $\rightarrow = (1.3915)e - (1.7240) \tau (0.3841)$ $\rightarrow \tau = 0.6253 Ti - T \propto 16 - 28$ which is above the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can be used. Properties The specific heat of water at room temperature is Cp = 4.18 kJ/kg.°C (Table A-9). The temperature of the sheet metal after quenching and the rate at which heat ransfer coefficient is constant are to be determined. 4 The heat transfer coefficient is constant and uniform over the entire surface. 6 The heat transfer coefficient is constant are to be determined. radiation from the fins. The finite difference formulation of this problem is to be obtained, and the top and bottom surface temperatures under steady conditions are to be determined. Then, h, $T \propto T0 = 2T1 + T2 + h(p\Delta x 2 / kA)(T \propto -T1) = 0$ $\Delta x = 3$: $T2 - 2T3 + T4 + h(p\Delta x 2 / kA)(T \propto -T1) = 0$ m = 2: $T1 - 2T2 + T3 + h(p\Delta x 2 / kA)(T \propto -T1) = 0$ T3) = 0 • 0 m = 4: T3 - 2T4 + T5 + h($p\Delta x 2 / kA$)(T ∞ - T4) = 0 • 1 • 2 • 3 • 4 5 • 6 m = 5: T4 - 2T5 + T6 + h($p\Delta x 2 / kA$)(T ∞ - T5) = 0 Node 6: kA T5 - T6 + h($p\Delta x 2 / kA$)(T ∞ - T6) = 0 Δx where $\Delta x = 0.005$ m, k = 237 W/m · °C, $T \infty = 30$ °C, T 0 = 100°C, h = 35 W/m 2 · °C and $A = \pi D 2 / 4 = \pi (0.25 \text{ cm}) 2 / 4 = 0.0491$ cm 2 = 0.0491 $p = \pi D = \pi (0.0025 \text{ m}) = 0.00785 \text{ m}$ (b) The nodal temperatures under steady conditions are determined by solving the 6 equations above simultaneously with an equation solver to be T1 =97.9°C, T2 =96.1°C, T3 =94.7°C, T4 =93.8°C, T5 =93.1°C, T5 =93.1 the nodal elements, $6 Q_{\infty}$ fin = $\sum m = 0 Q_{\infty}$ element, $m = 6 \sum hA$ surface, $m (Tm - T_{\infty}) = 0.5496 W (d)$ The number of fins on the surface is No. of fins = 1m2 = 27,778 fins (0.006 m)(0.006 m) Then the rate of heat transfer from the fins, the unfinned portion, and the entire finned surface become Q& = (No. of fins) Q& = 27,778(0.5496 W) = 15,267 W fin, total fin Q& `unfinned = hAunfinned (T0 - T ∞) = (35 W/m 2 · °C)(1 - 27,778 × 0.0491 × 10 - 4 m 2)(100 - 30)°C = 2116 W Q& total = Q& fin, total fin Q& `unfinned = 15,267 + 2116 = 17,383 W \cong 17.4 kW 5-28 Chapter 5 Numerical Methods in Heat Conduction 5-37 One side of a hot vertical plate is to be cooled by attaching copper pin fins. The rate of heat transfer to the van is to be determined. Substituting the given guantities, the maximum allowable the time step becomes $\Delta t \leq (0.05 \text{ m})$ $2 2(0.44 \times 10 - 6 \text{ m } 2/\text{s})[1 + (9.1 \text{ W/m} 2.^{\circ})(0.05 \text{ m})/(0.70 \text{ W/m}.^{\circ})] = 1722 \text{ s}$ Therefore, any time step less than 1722 s can be used to solve this problem. 2-53 Chapter 2 Heat Conduction Equation 2-93 "GIVEN" L=0.05 "[m]" T s=30 "[C]" k=30 "[W/m-C]" g dot 0=8E6 "[W/m^3]" "ANALYSIS" g dot=g dot
0*exp((-0.5*x)/L) "Heat generation as a function of x" "x is the parameter to be varied" g [W/m3] 8.000E+06 7.610E+06 6.530E+06 6.530E+06 6.530E+06 6.530E+06 6.530E+06 5.027E+06 5.638E+06 5.010E+06 6.230E+06 5.027E+06 5.638E+06 5.010E+06 6.230E+06 5.027E+06 5.638E+06 5.010E+06 6.230E+06 5.00E+06 5.010E+06 6.230E+06 5.010E+06 0.01 0.02 0.03 x [m] 2-54 0.04 0.05 Chapter 2 Heat Conduction Equation Variable Thermal Conductivity 2-94C During steady one-dimensional heat conductivity and no heat generation, the temperature in only the plane wall, long cylinder, and sphere with constant thermal Conductivity and no heat generation, the temperature in only the plane wall, long cylinder, and sphere with constant thermal Conductivity 2-94C During steady one-dimensional heat conductivity and no heat generation. Basics of Heat Transfer 1-64 "GIVEN" "L=0.005 [m], parameter to be varied" A=2*2 "[m^2]" T 1=10 "[C]" T 2=3 "[C]" k=0.78 "[W/m-C]" time=5*3600 "[s]" "ANALYSIS" Q dot cond=k*A*(T 1-T 2)/L Q cond=Q dot cond*time*Convert([, k]) L [m] 0.001 0.002 0.003 0.004 0.005 0.006 0.007 0.008 0.009 0.01 Q cond [k] 393120 196560 131040 98280 78624 65520 56160 49140 43680 39312 1-26 Chapter 1 Basics of Heat Transfer 400000 350000 Q cond [k] 250000 200000 150000 100000 50000 0 0.002 0.004 0.006 L [m] 1-27 0.008 0.01 Chapter 1 Basics of Heat Transfer 1-65 Heat is transferred steadily to boiling water in the pan through its bottom. Noting that there are 6 nodal spacing of equal length, the temperature change between two neighboring nodes is $(20 - 0)^{\circ}F/6 = 3.33^{\circ}C$. Assumptions 1 Heat transfer through the pin fin is given to be oneConvectio dimensional. 6 The bottom surface of the engine is a flat surface. 3 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. The equilibrium temperature of the glass bulb is to be determined. Therefore, we need to specify four boundary conditions for two-dimensional problems. 4 The outer surface temperature of the ball is uniform at all times. However, this relation is all we have for elliptical shapes, and we will use it with the understanding that the results may not be accurate. Assumptions 1 Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. The difference between the two results is due to the Fourier number being less than 0.2 and thus the error in the one-term approximation. Analysis The mass of water in the tank is (1 ft 3) = 497.3 lbm m = $\rho V = (62 \text{ lbm/ft } 3)(60 \text{ gal})$ 7.48 gal // Then, the amount of heat that must be transferred to the water in the tank as it is heated from 45 to140°F is determined to be $Q = mC(T2 - T1) = (497.3 \text{ lbm})(1.0 \text{ Btu/lbm.°F})(140 - 45)°F = 47,250 \text{ Btu } 140°F 45°F \text{ Water The First Law of Thermodynamics } 1-22C \text{ Warmer. and } A1 = 14113 \text{ . Gypsum wallboard, } 13 \text{ mm } 6. \circ C)[\pi (0.05 \text{ m}) 2 / 4]$ reduces to 2 (∂u) d 2T (V) 0 = k 2 + μ | | Energy: \rightarrow k = $-\mu$ | 2 ∂y dy (L) (∂y / since $\partial u / \partial y$ = V / L. The mathematical formulation, the variation of temperatures at the inner and outer surfaces to be determined for steady one-dimensional heat transfer. The rate of evaporation of the liquid oxygen due to heat transfer from the air is to be determined for three cases. energies & 1 = mh & 2 (since $\Delta ke \cong \Delta pe \cong 0$) W&e,in + mh & p (T2 - T1) (4.18 kJ/kg · °C) (70 - 15)°C W& e,in = 1-22 15°C WATER 70°C 7 kW Chapter 1 Basics of Heat Transfer Heat Transfer Mechanisms 1-44C The thermal conductivity of a material is the rate of heat transfer through a unit thickness of the material per unit area and per unit temperature difference. This effect can be prevented or minimized by using destratification fans (ceiling fans running in reverse). 2 The heat transfer coefficient is constant and uniform over the entire fin surfaces. Therefore, the boundary conditions are u(0) = 0 and u(L) = V, and applying them gives the velocity distribution to be u(y) = y V L where $(1 \text{ min}) V = \pi Dn \& = \pi (0.05 \text{ m})(4500 \text{ rev/min}) = 11.78 \text{ m/s} (60 \text{ s}) The plates are isothermal and there is no change in the flow direction, and thus the temperature depends on y only, T = T(y). The temperature of the base of the iron is to be$ determined in steady operation. Properties The emissivity of the outer surface of the box is given to be 0.95. Assumptions 1 Heat conduction in the rods are long and they have thermal symmetry about the center line. Properties The density and specific heat of water at room temperature are $\rho = 1 \text{ kg/L}$ and C = 4.18kJ/kg °C (Table A-9). L = 0.8 m Properties The properties of air at 1 atm and the film temperature of $(Ts + T\infty)/2 = (80+20)/2 = 50^{\circ}C$ are (Table A-15) k = 0.02735 W/m. °C Engine block Air V $\infty = 80$ km/h T $\infty = 20^{\circ}C$ $\upsilon = 1.798 \times 10^{-5}$ m 2/s Pr = 0.7228 Ts = 80°C $\varepsilon = 0.95$ Analysis Air flows parallel to the 0.4 m side. It is due to the internal frictional force that develops between different layers of fluids as they are forced to move relative to each other. = 01465. 4-34 Chapter 4 Transient Heat Conduction 4-45 An egg is dropped into boiling water. Therefore, this problem can be simplified greatly by considering the heat transfer as being one- dimensional at each of the four sides as well as the top and bottom sections, and then by adding the calculated values of heat transfers at each surface. Following the approach described above using a computer, the amount of heat transfers at each surface. 5393 kJ after 36 h QTrombe wall = 15,230 kJ after 48 h Discussion Note that the interior temperature of the Trombe wall drops in early morning hours, but then rises as the solar energy absorbed by the exterior surface diffuses through the wall. The center temperature of the rod is to be determined. Using energy balances, the finite difference equations for each of the 9 nodes are obtained as follows: Node 1: hi \cdot 2 T i - T6i Δx T3i - T6i Δx T3i - T6i Δx T4 - T7 Δx Δy T7 - T7 + k (Ti - T7i) + k = $\rho C \Delta y 2 2 \Delta x 2 2 \Delta t$ T i - T8i Δy T7i - T8i Δy T7i - T8i Δy T8i + 1 - T8i Δy T8i + 1 - T8i $\Delta x \Delta x \Delta y 2 2 2 \Delta t$ T i - T7i $\Delta x \Delta x \Delta y 2 2 2 \Delta t$ 5-92 Chapter 5 Numerical Methods in Heat Conduction Node 9: ho i i i +1 i $\Delta y \Delta y$ T8i - T9i Δx T6 - T9 $\Delta x \Delta y$ T9 - T9 + k (To - T9i) + k = $\rho C \Delta y 2 2 2 \Delta x 2 2 \Delta t$ where k = 0.84 W/m.°C, $\alpha = k / \rho C = 0.39 \times 10 - 6 \text{ m } 2 / \text{ s}$, Ti = To = -3°C hi = 6 W/m2.°C, ho = 20 W/m2.°C, $\Delta x = 0.002 \text{ m}$, and $\Delta y = 0.01 \text{ m}$. Therefore, Q & = Q & = Q & and h= through tank from tank, conv+rad Ts,out - Ts,in Q& = = ho Ao (Tsurr - Ts,out) + ϵ Aog (Tsurr 4 - Ts,out 4) Rsphere where R sphere = r2 - r1 (1.51 - 1.50) m = 2.342 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 28.65 m 2 Substituting, Ts ,out - 0°C Q& = (9.05 W/m 2.°C)(28.65 m 2)(30 - Ts ,out) °C 2.34 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 28.65 m 2 Substituting, Ts ,out - 0°C Q& = (9.05 W/m 2.°C)(28.65 m 2)(30 - Ts ,out 4) °C 2.34 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 28.65 m 2 Substituting, Ts ,out - 0°C Q& = (9.05 W/m 2.°C)(28.65 m 2)(30 - Ts ,out 4) °C 2.34 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 28.65 m 2 Substituting, Ts ,out - 0°C Q& = (9.05 W/m 2.°C)(28.65 m 2)(30 - Ts ,out 4) °C 2.34 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 28.65 m 2 Substituting, Ts ,out - 0°C Q& = (9.05 W/m 2.°C)(28.65 m 2)(30 - Ts ,out 4) °C 2.34 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 28.65 m 2 Substituting, Ts ,out - 0°C Q& = (9.05 W/m 2.°C)(28.65 m 2)(30 - Ts ,out 4) °C 2.34 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 10 °C (28.65 m 2)(30 - Ts ,out 4) °C 2.34 × 10 - 5 °C/W 4πkr1 r2 4π (15 W/m.°C)(1.51 m)(1.50 m) Ao = πD 2 = π (3.02 m) 2 = 10 °C (28.65 m 2)(30 - Ts ,out 4) °C (28.65 m 2)(3 $C/W + (0.9)(28.65 \text{ m 2})(5.67 \times 10 - 8 \text{ W/m 2} \text{ K 4})[(15 + 273 \text{ K}) 4 - (Ts, out + 273 \text{ K}) 4]$ whose solution is Ts = 0.23°C and Q& = 9630 W = 9.63 kW (b) The amount of heat transfer during this period becomes 832,032 kJ (Q \rightarrow m = = = 2493 kg Q = mhif 333.7 kJ/kg hif 7-901 cm Chapter 7 External Forced Convection 7-98E A cylindrical transistor mounted on a circuit board is cooled by air flowing over it. The latent heat of freezing of water is 33.7 kJ/kg. m² . Properties The properties of air at 1 atm pressure and the free stream temperature of 25°C are (Table A-15) k = 0.02551W/m.°C Air Iced water $v = 1.562 \times 10^{-5} \text{ m } 2/\text{s } V \infty = 7 \text{ m/s } T \infty = 25^{\circ}\text{C} \mu \infty = 1.849 \times 10^{-5} \text{ kg/m.s } \mu \text{ s}$, @ 0°C = 1.729 × 10 - 5 kg/m.s 0°C D = 1.8 m Pr = 0.7296 Analysis The Reynolds number is V D (7 m/s)(1.8 m) Re = $\infty = 806,658 v 1.562 \times 10^{-5} \text{ m } 2/\text{s } \text{The proper relation for Nusselt number corresponding to this Reynolds number is Nu = []}$ $(\mu hD = 2 + 0.4 \text{ Re } 0.5 + 0.06 \text{ Re } 2/3 \text{ Pr } 0.4 || \approx k \lfloor \mu s [1/4] || /] (1.849 \times 10 - 5 = 2 + 0.4(806,658) 2/3 (0.7296) 0.4 || -5 (1.729 \times 10 \text{ The heat transfer is determined to be } 1/4) || /] = 790.1 \text{ As} = \pi D 2 = \pi (1.8 \text{ m})$ $2 = 10.18 \text{ m} 2 \text{ Q} = hA (T - T) = (11.20
\text{ W/m} 2.°C)(10.18 \text{ m} 2)(25 - 0)^{\circ}C = 2850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \propto \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \approx \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \approx \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \approx \text{The rate at which ice melts is } Q_{\&} = m_{\&}h \rightarrow = 2.850 \text{ W} \text{ s} \approx \text{The rate at which ice melts is } Q_{~} = 0.850 \text{ W} \text{ s} \approx \text{The rate at which ice melts is } Q_{~} = 0.850 \text{ W} \text{ s} \approx \text{The rate at which ice melts is } Q_{~} = 0.850 \text{ W} \text{ s} \approx 0.850 \text{ W} \text{ s} \approx 0.850 \text{ W} \text{ s} \approx 0.85$ 2Tmi + Tmi +1 + Using the definition of the dimensionless mesh Fourier number $\tau = \text{Tmi} - 1 - 2\text{Tmi} + \text{Tmi} + 1 + \text{Insulation} \alpha \Delta t$ (Δx) 2, the last equation reduces to g & 0 Δx 2 Tmi+1 - Tmi = k τ Discussion We note that setting Tmi +1 = Tmi gives the steady finite difference formulation. The properties of air at this temperature are (Table A-15) $\rho =$ $1.246 \text{ kg/m } 3 \text{ k} = 0.02439 \text{ W/m.}^{\circ} \text{C} \upsilon = 1.426 \times 10 \text{ m/s} - 5 2 \text{ Pr} = 0.7336 \text{ Analysis The Reynolds number is VD} [(40 \times 1000/3600) \text{ m/s}](0.006 \text{ m}) \text{ Re} = \infty = = 4674 \upsilon 1.426 \times 10 - 5 \text{ m} 2/\text{s} \text{ The Nusselt number corresponding this Reynolds number is determined to be hD}$ $|| || || \sqrt{282,000} || Wind V_{\infty} = 40 \text{ km/h T}_{\infty} = 10^{\circ}\text{C} \text{ Transmission wire, Ts D} = 0.6 \text{ cm } \frac{4}{5} \frac{4}{5} \frac{5}{8} 0.62(4674) 0.5 (0.7336)1 / 3 [(4674)]| = 0.3 + 1 + = 36.0 || || 1/4 || (282,000) || 1 + (0.4 / 0.7336)2 / 3 \text{ The heat transfer coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C D } 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C h} 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C h} 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C h} 0.006 \text{ m The rate of heat generated in the second of the coefficient is } 0.02439 \text{ W/m.} \\ \circ \text{C k h} = \text{Nu} = (36.0) = 146.3 \text{ W/m } 2 \cdot \text{C h} 0.006 \text{ m The rate o$ electrical transmission lines per meter length is W& = Q& = I 2 R = (50 A) 2 (0.002 Ohm) = 5.0 W [] The entire heat generated in electrical transmission line has to be transferred to the ambient air. 4-61C A thick plane wall can be treated as a semi-infinite medium if all we are interested in is the variation of temperature in a region near one of the surfaces for a time period during which the temperature in the mid section of the wall does not experience any change. Analysis In case of no fins, heat transfer from the tube per meter of its length is 180° C Ano fin = π D1 L = π (0.05 m)(1 m) = 01571. Properties The thermal diffusivity of meat slabs are given to be k = 0.47 $W/m \cdot C$ and $\alpha = 0.13 \times 10^{-6} m^2/s$. 4-14a we have Water 100°C 2 cm Rod Ti = 25°C | αt | $\tau = 2 = 0.4075 - 100$ ro = = 0.33 | 25 - 100 | 1 k = 0.25 Bi hro To $-T \infty$ Ti $-T \infty$ Then the thermal diffusivity and the thermal diffusity and the thermal diffusive and the therm = $(2.22 \times 10 - 7 \text{ m } 2/s)(3700 \text{ kg/m } 3)(920 \text{ J/kg.°C}) = 0.756 \text{ W/m}$. 2 Orange is spherical in shape. Properties of the concrete wall are given to be k = 0.9 W/m. °C and $\alpha = 0.23 \times 10-5 \text{ m} 2/s$. Then the average Nusselt number and heat transfer coefficient for all the tubes in tubes in the tubes in the tubes in the tubes in = 72.09 h= Nu D, N L k D = 72.09(0.02514 W/m · °C) = 86.29 W/m 2 · °C 0.021 m The total number of tubes is N = NL × NT = 8×8 = 64. Then the pressure drop across the tube bank becomes $\Delta P = N L f\chi 2 \rho Vmax (1.316 kg/m 3)(8.571 m/s) 2 = 30(0.27)(1) 2 2 (1N | | 1 kg · m/s 2 \ | | = 391.6 Pa |] Discussion The arithmetic mean fluid$ temperature is $(Ti + Te)/2 = (0.15.6)/2 = -7.8^{\circ}C$, which is fairly close to the assumed value of $-5^{\circ}C$. 6-8 | | | Chapter 6 Fundamentals of Convection 6-38 Parallel flow of oil between two plates is considered. Properties The thermal conductivities are given to be $k = 0.72 \text{ W/m} \cdot ^{\circ}C$ for bricks, $k = 0.22 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026
\text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/m} \cdot ^{\circ}C$ for plaster layers, and $k = 0.026 \text{ W/$ the rigid foam. The time it will take for the wire to cool, the distance the wire travels, and the rate of heat transfer from the wire are to be determined. Properties The average specific heat of the cars is given to be 0.45 kJ/kg.°C. (a) The driving force for fluid flow is the pressure difference. 1-119C Stratification is the formation of vertical still air layers in a room at difference temperatures, with highest temperatures occurring near the ceiling. It causes discomfort by exposing the head and the feet to different temperatures. C / W The inner surface temperature of the window glass can be determined from T – T & Q& = $\infty 1.1 \rightarrow T1 = T \infty 1 - QR \operatorname{conv} , 1 = 24^{\circ} C - (203 W)(0.0417^{\circ} C / W) = 15.5^{\circ} C$ Rconv, 1 Similarly, the inner surface temperatures of the glasses are calculated to be 15.2 and $-1.2^{\circ}C$ (we had assumed them to be 15 and 5°C when determining the radiation resistance). 2-37C The boundary condition at a perfectly insulated surface (at x = 0, for example) can be expressed as $-k \partial T$ (0, t) = 0 ∂x which indicates zero heat flux. 2 The thermal Tsurr conductivity is given to be constant. Similarly, if the fluid and the wall are at different temperatures and the fluid is heated or cooled during flow, heat conduction will occur primarily in the direction normal to the surface, and thus $\partial T / \partial y >> \partial T / \partial x$. Properties of liquid water are Cp = 4.18 kJ/kg.°C and p = 977.6 kg/m3 (Table A-2). The velocity and the surface area are $(1 \text{ min})V = \pi \text{Dn} \& = \pi (0.06 \text{ m})(3000 \text{ rev/min}) = 9.425 \text{ m/s} (60 \text{ s})A = \pi \text{DLbearing} = \pi (0.06 \text{ m})(0.20 \text{ m}) = 0.0377 \text{ m} 2$ The maximum temperature is Tmax = T (L/2) = T0 + $\mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 2 \text{ k} (L/2 (L/2) 2 - L L 2) = 10 + \mu V 2 + \mu V 2$ 8k = -kA dy = -kA dthe two plates are equal in magnitude but opposite in sign. Most of this money can be saved by insulating the heating ducts in the unheated areas. 5-51 Chapter 5 Numerical Methods in Heat Conduction 5-56 Heat transfer through a square chimney is considered. 4-71C This short cylinder is physically formed by the intersection of a long cylinder and a plane wall. 2-52 Chapter 2 Heat is generated in a large plane wall whose one side is insulated while the other side is maintained at a specified temperature. 3-103C Increasing the diameter of a fin will increase its efficiency but decrease its effectiveness. 2-20 The one-dimensional transient heat conduction equation for a plane wall with constant thermal 1 $\partial \left(\partial T \right) g \& 1 \partial T$ conductivity and the kinematic viscosity of air at the film temperature of (Ts + T ∞)/2 = (15+5)/2 = 10°C are (Table A-15) k = 0.02439 W/m. Analysis The rate of heat transfer without insulation is A = (2 m)(1.5 m) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T\infty) = 3 m 2 Q \& = hA(Ts - T (10 W/m) = 32 m 2 L = 4 m For flat plates, the drag force is equivalent to friction force. Considering a 1-m section of the pipe, the amount of heat that must be transferred from the water as it cools from 15 to 0°C is determined to be m = $\rho V = \rho$ $(\pi r 1 \ 2 \ L) = (1000 \ kg/m \ 3)[\pi (0.03 \ m) \ 2 \ (1 \ m)] = 2.827 \ kg \ Q \ total = mC \ p \ \Delta T = (2.827 \ kg)(4.18 \ kJ/kg)^{\circ}(15 \ -0)^{\circ}C = 177.3 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 177.3 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 177.3 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 177.3 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 177.3 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 177.3 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2 \times (2.827 \ kg)(4.18 \ kJ/kg)^{\circ} = 188.7 \ kJ \ Q \ total = Q \ cooling + Q \ freezing = 0.2 \times mhif = 0.2$ = 1694. It is expressed as $\tau t = -\rho u'v' = \mu t \partial u$ where u is the mean value of velocity. k(T) T2 T1 Properties The thermal conductivity is given to be k (T) = k 0 (1 + $\beta T 2$). The cooking time of the egg is to be determined. Analysis (a) Assuming the turkey to be spherical in shape, its radius is determined to be 14 lbm m = $\rho V \rightarrow V = = 0.1867$ ft 3 Turkey ρ 75 lbm/ft 3 Ti = 40°F 3 3 (0. The rate of heat transfer is to be determined for two cases. 3 Thermal conductivity is constant and there is nonuniform heat generation in the medium. 5-1 Chapter 5 Numerical Methods in Heat Conduction 5-7 We consider three consecutive nodes n-1, n, and n+1 in a plain wall. Ventilation increases the energy consumption for heating in winter by replacing the warm indoors air by the colder outdoors air. Properties The thermal
conductivities are given to be k = 0.11 W/m. °C for wood studs and k = 50 W/m. °C for manganese steel nails. Properties The properties of bronze are given to be k = 15 Btu/h.ft.°F and $\alpha = 0.333$ ft2/h. To determine whether it is realistic to assume the plate temperature for temperature for the plate temperature for temper since Bi < 0.1. Discussion This problem can also be solved by obtaining the differential equation from an energy balance on the plate for a differential time interval, and solving the differential equation. Properties 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 v = 1.608 × 10 -5 m 2/s Ts = 0°C V ∞ = 25 km/h T ∞ = 30°C $\mu \infty$ = 1.872 × 10 - 5 kg/m.s μ s, @ 0°C = 1.729 × 10 - 5 kg/m.s Pr = 0.7282 Analysis (a) The Reynolds number is V D [(25 × 1000/3600) m/s](3.02 m) Re = ∞ = = 1.304 × 10 6 - 5 2 υ 1.608 × 10 m /s Iced water Di = 3 m 0°C 1 cm Q& The Nusselt number corresponding to this Reynolds number is determined from [] (μ hD = 2 + 0.4 Re 0.5 + 0.06 Re 2 / 3 Pr 0.4 || ∞ Nu = k (μ s [= 2 + 0.4(1.304 × 10) 6 0.5 1 / 4) || / + 0.06(1.304 × 10) 6 2/3](0.7282) 0.4 (| 1.872 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 | 1.729 × 10 - 5 $(9.05 \text{ W/m 2}.^{\circ}\text{C})[\pi (3.02 \text{ m}) 2](30 - 0)^{\circ}\text{C} = 7779 \text{ W}$ and h = s s ∞ s ∞ (b) The amount of heat transfer during this period becomes 672,079 kJ (24×3600 s) = 672, A spherical tank used to store iced water is subjected to winds. However, the temperature along the wall and thus the energy content of the wall will change during transient conductivity is given to be k = 5.8 Btu/h·ft·°F. The rate of heat transfer to the tank is to be determined for the insulated and uninsulated ground surface cases. = 30° C + = 409° C 2h 2(140 W/m2. The heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are As = $n\pi DL$ = $300\pi(0.008 \text{ m})(0.4 \text{ m}) = 0.4651 \text{ kg/s}$ Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer (refrigeration capacity) become (Ah Te = Ts - (Ts - Ti) exp - s m& C p ($\Delta Tln = 2 2$) (| = -20 - (-20 - 0) exp -/(-20 + 15.57)] Q& = hAs Δ Tln = (155.8 W/m 2 · °C)(4.524 m 2)(10.33°C) = 7285 W 7-67 Chapter 7 External Forced Convection For this square in-line tube bank, the friction coefficient corresponding to ReD = 5294 and SL/D = 1.5/0.8 = 1.875 is, from Fig. / 12 ft) hr0 (4.6 Btu / h.ft 2 . Properties The specific heat of air at room temperature is Cp = $1.007 \text{ kJ/kg} \circ C$ (Table A-15). The time they can stand in the air before their temperatures at the neighboring nodes in the following easy-to-remember form: i i i i + Ttop + Tright + Tbottom - 4Tnode + Tleft i + 1 i - Tnode g & 0 l 2 Tnode = k \tau i + 1 i = Tnode Discussion We note that setting Thode gives the steady finite difference formulation. energies $0 = \Delta U 20 \text{ kg} + \Delta U \text{ iron} + \Delta U \text{ Al} = 0$ substituting, (20 kg)(0.450 kJ / kg·o C)(T2 - 100)o C + (20 kg)(0.973 kJ / kg·o C)(T2 - 200)o C = 0 T2 = 168 °C 1-36 \text{ An unknown mass of iron is dropped into a strong term in the strength of the strengt of the strength of the streng water in an insulated tank while being stirred by a 200-W paddle wheel. (b) The nodal temperatures under steady conditions are determined by solving the 4 equation solver to be T1 = 66.9°C, T2 = 53.8°C, T3 = 40.7°C, and T4 = 27.6°C (c) The rate of heat transfer through the wall is simply convection heat transfer at the right surface, $O_{\&} = O_{\&} = hA(T - T) = (24 \text{ W/m 2})^{\circ}(20 \text{ m 2})(27.56 - 15)^{\circ}C = 6029 \text{ W wall conv 4} \propto Discussion This problem can be solved analytically by solving the differential equation as described in Chap. 3 There is no heat generation in the pipe. 3 + 1/4 k (0.°C) Thus the temperature at that location will be about 33°C$ above the temperature of the outer surface of the wire. With these considerations, the properties of air at 0.823 atm and at the film temperature of $(120+30)/2=75^{\circ}$ C are (Table A-15) k = 0.02917 W/m. °C v = v @ 1atm / Patm = (2.046 × 10 -5 m 2/s) / 0.823 = 2.486 × 10 -5 m 2/s) / 0.823 = 2.486 × 10 -5 m 2/s) Reynolds number in this case becomes VL (6 m/s)(8 m) Re L = ∞ = = 1.931 × 10 6 2 - 5 v 2.486 × 10 m /s which is greater than the critical Reynolds number. Then surface temperatures on the two sides of the circuit board becomes T - T & Q& = 1 ∞ \rightarrow T1 = T ∞ + QR total = 37° C + (15 W)(1.492 ° C / W) = 59.4° C Rtotal T - T & Q& = 1 2 \rightarrow T2 = T1 - QR board = 59.4° C - (15 W)(0.011 ° C / W) = 59.2° C Rboard 3-125 Chapter 3 Steady Heat Conduction (b) Noting that the cross-sectional areas of the fins are constant, the efficiency of these rectangular fins is determined to be hp \approx kAc a = h(2w) = k (tw) 2h = kt 2 cm 2(45 W/m 2.°C) = 10.80 m - 1 (386 W/m.°C)(0.002 m) tanh aL tanh(10.80 m) $-1 \times 0.02 \text{ m} = = 0.985 \text{ aL } 10.80 \text{ m} - 1 \times 0.02 \text{ m}$ The finned and unfinned surface areas are η fin = 0.002 t (20.15) 0.02 + | = 0.126 m 2/2 (V Aunfinned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = η fin Afin (Tbase $-T \infty$) Q& unfinned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = η fin Afin (Tbase $-T \infty$) Q& unfinned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = η fin Afin (Tbase $-T \infty$) Q& unfinned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2 Then, Q& finned = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090 m 2total = Q& unfinned + Q& finned = h(Tbase = T ∞ + h(η fin Afin + Aunfinned) = 37°C + Rcopper T1 15 W (45 W/m .°C)[(0.985)(0.126 m 2) + (0.0090 m 2)] 2 Repoxy Rboard T ∞ = 39.5°C Then the temperatures on both sides of the board are determined using the thermal resistance network to be L 0.001 m = $0.00017 \circ C / W kA (386 W / m.$ The temperature difference between the center and the surface of the fuel rod is to be determined. air) : or hi Ai (T4i - T5i) = (mC\Delta T) air + (mC\Delta T) food [i + 1] T5 - T5i \Delta t Ai = $2(1.77 \times 0.77) + 2(1.77 \times 0.67) + (0.77 \times 0.67)$ 5.6135 m 2 hi Ai (T4i - T5i) = (mC) air + (mC) food where Substituting, temperatures of the refrigerated space after $6 \times 60 = 360$ time steps (6 h) is determined to be Tin = T5 = 19.6°C. 2 There is no heat generation within the block. of the much lower viscosity of air relative to water. Properties The thermal conductivity of rubber insulation is given to be k = 0.13 W/m °C. The Kirchhoff's law of radiation states that the emissivity and the absorptivity of a surface are equal at the same temperature of the outer surface of the refrigeration compartment of the truck is to be determined. In our case the number of rows is NL = 8, and the corresponding correction factor from Table 7-3 is F = 0.967. Analysis The nodal spacing is given to be $\Delta x
= \Delta x = 1 = 0.2$ ft, and the general finite difference form of an interior node for steady two-dimensional heat conduction is expressed as g& node 1 2 h, To =0 k 1 2 3 • • • (a) There is symmetry about the vertical, horizontal, and diagonal lines passing through the center. 4-31C The maximum possible amount of heat transfer will occur when the temperature of the body reaches the temperature of the medium, and can be determined from Qmax = mC p (To - Ti). Properties The thermal conductivity of the fins is given to be k = 186 W/m.°C. 6 Air is an ideal gas with constant properties, 3 The thermal properties of the egg and heat transfer is onedimensional since there is thermal symmetry about the center line and no change in the axial direction. Assumptions 1 Both the iron and aluminum block are incompressible substances with constant specific heats. The local friction and heat transfer coefficients at intervals of 1 ft are to be determined and plotted against the distance from the leading edge. Assumptions 1 Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. 2 The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. Dividing both sides by k and integrating twice give $\partial 2T 2 2 \mu \langle V \rangle dT = - | y + C3 dy k \langle L / 2 T (y) = -\mu \langle y \rangle | V | + C3 y + C 4 2k \langle L / Applying the two boundary conditions give B.C. 1: y=0 T (0) = T1 \rightarrow C 4 = T1 B.C. 2: y=L -k dT dy = 0 \rightarrow C 3 = y=L \mu V 2 kL Substituting the$ constants give the temperature distribution to be T (y) = T1 + μ V 2 kL 2 / (y - y || 2 L // 6-13 Chapter 6 Fundamentals of Convection The temperature is determined by differentiating T(y) with respect to y, y dT μ V 2 (= 1 - | dy kL (L) The location of maximum temperature is determined by setting dT/dy = 0 and solving for y, y dT $\mu V 2 (\rightarrow v = L = |1 - | = 0 dy kL \setminus L / This result is also known from the second boundary condition.$ The rate of heat loss from the arm is to be determined. 3 All the heat generated at the junction is dissipated through the back surface of the plate since the transistors are covered by a thick plexiglas layer. The finite difference formulation for the k τ steady case is obtained by simply setting Tmi +1 = Tmi and disregarding the time index i. Disregarding any heat transfer coefficients at the beginning and at the end of the process are determined to be Q& 800 W = = 1020 W/m 2. °C As $(Ts - T \propto 1) (0.00785 \text{ m } 2)(120 - 20)^{\circ}C \text{ As} (Ts - T \propto 2) (0.00785 \text{ m } 2)(120 - 80)^{\circ}C \text{ h} = \text{Discussion Note that a larger heat transfer coefficient is needed to dissipate heat transfer coefficient is needed to dissipate heat transfer and the$ average heat transfer coefficient between the potato and its surroundings are Q 50.8 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T ∞) 609.4 Btu / h Δt (5 / 60 h) \rightarrow h = Q& = hAo (Ts - T \infty) value within the limits of experimental error. 3 + 1 + 1/4 | \ 282,000 | | [(0. The thermal diffusivity and thermal diffusivity of the hot dog and the convection heat transfer coefficient are to be determined. 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 KA $m - 1 + kAm + 1 + h(p\Delta x)(T \infty - Tm) = 0 \rightarrow Tm - 1 - 2Tm + Tm + 1 + h(p\Delta x 2 / kA)(T \infty - Tm) = 0 \Delta x \Delta x$ The finite difference equation for node 6 at the f are maintained at constant temperatures of 20°C and 95°C while the side surface is perfectly insulated. Properties The thermal conductivity is given to be k = 0.8 W/m·°C. Therefore, maximum temperature will occur at the shaft surface, for y = L. Using the results of Problem 6-43, a relation for the volumetric heat generation rate is to be obtained 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 surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surrounding surfaces is the same as the temperature of the surfaces is the same as the temperature of the surfaces is the same as the temperature of the surfaces is the same as the temperature of the surfaces is the same as the temperature of the surfaces is the same as the temperature of temperature of the surfaces is the same as the temperature of temperature of the surfaces is the same as the temperature of te = (0.15)(0.011 m) = 0.00165 m h Therefore, the heat transfer coefficient is k 0.771 W/m.°C = 0.00165 \rightarrow h = = 467 W/m 2. The energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for this steady-flow system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for the system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for the system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for the system can be expressed as E $-E = \Delta E$ system 1in424out 3 1 424 3 Net energy balance for the 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 $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516
\times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C $v = 1.516 \times 10^{-5}$ m 2 /s Insulation, D = 0.02514 W/m. °C v = 0.0Di = 4 m, and the Reynolds number is V D [$(40 \times 1000/3600)$ m/s](4 m) Re = ∞ = 2.932 × 10 6 v 1.516 × 10 - 5 m 2/s The Nusselt number is determined from Nu = [] (μ hD = 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) $0.4 \left(1.825 \times 10 - 5 \right) 5.023 \times 10 - 6 \left(1/4 \right) \left| 1/2 = 2333 \ 0.02514 \ \text{W/m} \ 2 \ \text{.}^{\text{C}} \text{D} \ 4 \text{m} \text{Terms of heat transfer to the liquid nitrogen is } Q_{\text{C}} = hAs (Ts - T_{\infty}) = h(\pi D \ 2)(Ts - T_{\infty}$ $Q_{k} = m_{k}$ hif $\rightarrow m_{k} = = 0.804$ 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 to be T2 = 487.7 R Then the rate of heat transfer through the wall becomes T -T (520 - 487.7) R g& = 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 to 214 W. 16.7/2 = 8.35 times. $2 \text{ W} \rightarrow \text{T} \text{ s} = \text{T} \infty + = = 193^{\circ}\text{C}$ 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. \times 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 $2T = 0 dx 2 dT (L) 4 = h[T - T] + \varepsilon \sigma [T (L) 4 - Tsurr 2.2 \infty surr dx T (L) = T2 = 45° C (b) Integrating the$ 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 $\lambda 1 = 3.0372$ and A1 = 19898 can be determined from Bi = hro kBi (0.45 W/m.°C)(30) $\rightarrow h = = = 156.9 W/m 2$. °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 $\rho = 970 \text{ kg/m3}$, Cp = 1.55 kJ/kg. $c_r = 0.93 \times 10 - 6 \text{ m } 2 \text{ / s}$, $\epsilon = 0.93 \times 10 - 6 \text{ m } 2 \text{ / s}$, $\epsilon = 0.95$, and hif $\cdot 2 \cdot 3 = 1.87 \text{ kJ/kg}$. $c_r = 1.40 \cdot 1 \text{ W/m}$. $c_r = 0.93 \times 10 - 6 \text{ m } 2 \text{ / s}$, $\epsilon = 0.95$, and hif $\cdot 2 \cdot 3 = 1.87 \text{ kJ/kg}$. "!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 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. 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 = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti = λ 1 = 3.0533 and A1 = 1.9925 Then the Fourier number are, from Table 4-1, Water 94 4°C Egg Ti $(1.9925)e - (3.0533)\tau \rightarrow \tau = 0.1727$ Ti $-T \propto 8 - 94.4$ 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 = 7000 kJ/h || \ 3600 kJ/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 water. $^{\circ}$ C)(0.5 m) 1 1 = = = 0.6631 $^{\circ}$ 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 = $N\pi DL = 64\pi(0.021 \text{ m})(1 \text{ m}) = 4.222 \text{ m} 2 \text{ m} \& = m \& i = \rho i V(N T S T L) = (1.225 \text{ kg/m 3})(3.8 \text{ m/s})(8)(0.05 \text{ m})(1 \text{ m}) = 1.862 \text{ kg/s}$ 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 \ \Delta Tln = (1.225 \text{ kg/m 3})(3.8 \text{ m/s})(8)(0.05 \text{ m})(1 \text{ m}) = 1.862 \text{ kg/s} 22) (| = 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)] ln[(90 - 15) / (90 - 28.42)] Q& = hAs \Delta Tln = (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 0.23 0.23 3.696 --0.98 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, $U = \Sigma$ farea, iUi = $0.80 \times 0.232 + 0.20 \times 0.628$ 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 p, 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 $t = \tau ro 2$ (0.198)(0.0275 m) $2 = 1068 s = 17.8 min \alpha$ (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 "[m/n-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-1.0969) A 1=3.0863 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 To [C] 4-21 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 To [C] 4-21 80 85 90 95 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 To [C] 4-21 80 85 90 95 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 To [C] 4-21 80 85 90 95 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 To [C] 4-21 80 85 90 95 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 To [C] 4-21 80 85 90 95 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 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 50 60 65 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 50 60 65 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 50 60 65 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 50 60 65 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 50 60 65 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 50 60 65 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 30 50 50 60 65 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient Heat Conduction 120 110 100 tim e [m in] 90 80 70 60 50 40 70 75 To [C] 4-21 80 85 90 95 Chapter 4 Transient 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 are uniform. The drag force the wind shield is to be determined. Analysis The mass of the water is Ice, m w = $\rho V = (1 \text{kg/L})(0.2 \text{ L}) = 0.2 \text{kg} 0^{\circ} \text{C}$ We take the ice and the water as our system, and disregard any heat and mass transfer. Dividing the equation above by $A\Delta r$ gives $-T - Tt 1 Q\& r + \Delta t A \Delta r \Delta t$ Taking the limit as $\Delta r \rightarrow 0$ and $\Delta t \rightarrow 0$ yields $\partial T 1 \partial (\partial T) | kA | = \rho C A \partial r \langle \partial r \rangle$ ∂t since, from the definition of the derivative and Fourier's law of heat conduction, $Q \& r + \Delta r - Q \& r \partial q (\partial T) = - | - kA | \Delta r \rightarrow 0 \partial r \partial r | \Delta r | r$ $|= r 2 \partial r \langle \partial r \rangle \alpha \partial t$ where $\alpha = k / \rho C$ 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/2*(T o-T 1)+k*l/2*(T 2-T 1)/l+k*l/2*(T 3-T 2)/l+k*l/2*(T 3-T 2)/l+k*l/2*(T 1-T 2)/l+k*l/2*(T 1-T 2)/l+k*l/2*(T 3-T 3)/l+k*l/2*(T 3-T 3)/l+k*l/2 T 3)/1+epsilon*sigma*1*(T sky^4(T 3+273)^4)=0 "Node 3" h o*1*(T o-T 4)/1+k*1/2*(T 5-T 6)/1+k*1/2*(T 7-T 6)/1+k*1/2*(T 7 $(T 9-T 7)/1+k^{1*}(T 3-T 7)/1+k^{1*}(T 3-T 7)/1+k^{1*}(T 8-T 7)/1+k^{1/2*}(T 9-T 8)/1+k^{1/2*}(T 9-T 10)/1+k^{1/2*}(T 9-T 9)/1+k^{1/2*}(T 9-T 10)/1+k^{1/2*}(T 9-T 10)/1+k^{1/2*}(T 9-T 9)/1+k^{1/2*}(T 9-T 9)/1+k^{1$ "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 = $V \propto D \left[(40 \times 1000/3600) \text{ m/s} \right] (4.04 \text{ m}) = 2.961 \times 10.6 \text{ m} = 2.961 \times 10.6 \text{$

 $0.4(2.961 \times 10) 6 0.5 1/4 | | / + 0.06(2.961 \times 10) 6 2/3 | (0.7309) 0.4 (| 1.825 \times 10 - 5 | 1.729 \times 10^{-5} + 1.$ $(-196)]^{\circ}C = 27.4 \text{ W} (2.02 - 2) \text{ m} 1 + 4\pi (0.00005 \text{ W/m.}^{\circ}C)(2.02 \text{ m})(2 \text{ m})(10.73 \text{ W/m} 2.^{\circ}C)(51.28 \text{ m} 2)$ and h= The rate of evaporation of liquid nitrogen then becomes Q& 0.0274 kJ/s Q& = m& hif $\rightarrow m\& = = 1.38 \times 10^{-4} \text{ kg/s} 198 \text{ kJ/kg}$ hif 7-95 1/4 J J = 1724 Chapter 7 External Forced Convection 7-102 A spherical tank filled with liquidoxygen 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 h, To Convectio them. Assumptions 1 Heat transfer is steady. 2-21). 2 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. 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 temperatures below 70°C. Analysis The shape factor for this configuration is given in Table 3-5 to be $S = 2\pi(4 \text{ m}) 2\pi L = 13.58 \text{ m} (8z) (8(0.075 \text{ m}) \ln |\ln| || (0.03 \text{ m}) / \text{Then rate of heat loss from the hot water in 8 parallel pipes becomes Q& = 8Sk (T - T) = 8(13.58 \text{ m})(0.75 \text{ W/m.}^{\circ}C)(85 - 32)^{\circ}C = 4318 \text{ W} 1 2 32^{\circ}C 85^{\circ}C z D L = 4m \text{ The surface temperature of the wall can be determined from As} = 2(4 \text{ m})(8 \text{ m}) = 64 \text{ m} 2 (\text{from both hot water in 8 parallel pipes becomes Q} = 8Sk (T - T) = 8(13.58 \text{ m})(0.75 \text{ W/m.}^{\circ}C)(85 - 32)^{\circ}C = 4318 \text{ W} 1 2 32^{\circ}C 85^{\circ}C z D L = 4m \text{ The surface temperature of the wall can be determined from As} = 2(4 \text{ m})(8 \text{ m}) = 64 \text{ m} 2 (\text{from both hot water in 8 parallel pipes becomes Q} = 8Sk (T - T) = 8(13.58 \text{ m})(0.75 \text{ W/m.}^{\circ}C)(85 - 32)^{\circ}C = 4318 \text{ W} 1 2 32^{\circ}C 85^{\circ}C z D L = 4m \text{ The surface temperature of the wall can be determined from As} = 2(4 \text{ m})(8 \text{ m}) = 64 \text{ m} 2 (\text{from both hot water in 8 parallel pipes becomes Q} = 8Sk (T - T) = 8(13.58 \text{ m})(0.75 \text{ W/m.}^{\circ}C)(85 - 32)^{\circ}C = 4318 \text{ W} 1 2 32^{\circ}C 85^{\circ}C z D L = 4m \text{ The surface temperature of the wall can be determined from As} = 2(4 \text{ m})(8 \text{ m}) = 64 \text{ m} 2 (\text{from both hot water in 8 parallel pipes becomes Q} = 8Sk (T - T) = 8(13.58 \text{ m})(0.75 \text{ W/m.}^{\circ}C)(85 - 32)^{\circ}C = 4318 \text{ W} 1 2 32^{\circ}C 85^{\circ}C z D L = 4m \text{ The surface temperature of the wall can be determined from As} = 2(4 \text{ m})(8 \text{ m}) = 64 \text{ m} 2 (\text{from both hot water in 8 parallel pipes becomes Q} = 8Sk (T - T) = 8(13.58 \text{ m})(12 \text{ m}) = 64 \text{ m} 2 (\text{from both hot water in 8 parallel pipes becomes Q} = 8Sk (T - T) = 8(13.58 \text{ m})(12 \text{ m}) = 64 \text{ m} 2 (\text{from both hot water in 8 parallel pipes becomes Q} = 8Sk (T - T) = 8(13.58 \text{ m})(12 \text{ m}) = 8(13.58 \text{ m}$ sides) Q& 4318 W \rightarrow 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 resistance and the 3-105 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. Steady Heat Conduction U-factors for the air space and stud sections are available, the overall = 1/Uoverall where Uoverall = 1/Uoverall = 1/Uove 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 of air are (Table A-15) k = 0.02439 W/m.°C, $v = 1.426 \times 10^{-5} \text{ m } 2$ /s, Pr = 0.7336 and Analysis The outer diameter of insulated pipe is $Do = 4.6 + 2 \times 3.5 = 11.6 \text{ cm} = 0.7336 \text{ m} 2$ /s, Pr = 0.7336 m 2/s, Pr =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, on the other hand, deals with the rate of heat transfer as well as the temperature distribution within the system at a specified time. 3-137C Radiant barriers are highly reflective materials that minimize the radiation heat transfer between surfaces. Analysis The shape factor for this configuration is given in Table 3-5 to be T1 = 140° C z = $5.5 \text{ m} 2\pi D 2\pi (3 \text{ m}) \text{ S} = = 2183$. Therefore, it may be worthwhile to cover the outer surface at night to minimize the heat losses. 4-14b we have $T - T \propto 88 - 94$ = 0.17 To $-T \propto 59 - 94$ k | 1 = 0.15 ro Bi hro r | = 1 | J ro ro Water 94°C The Fourier number is determined from Fig. 2 The thermal conductivity and emissivity are constants. 4 The outer surface at r = r1 is subjected to convection. 4 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 at the tip and substituting, the tip temperature of the spoon is determined to be cosh a (L - L) cosh aL cosh 0 1 = 75°F + (200 - 75) = 75.4°F cosh(10.95 × 0.583) 296 T (L) = T∞ + (Tb - T∞) Therefore, the temperature difference across the exposed section of the spoon handle is $\Delta T = Tb - Ttip = (200 - 75.4)^\circ F = 124.6^\circ F 3.79$ 0 Chapter 3 Steady Heat Conduction 3-112E The handle of a silver spoon partially immersed in boiling water extends 7 in. Analysis We observe that the pressure in the room remains constant during this process Analysis The total rate of heat dissipation from the aluminum plate and the total heat transfer area are $Q\& = 4 \times 15 \text{ W} = 60 \text{ W} \text{ As} = (0.22 \text{ m})(0.22 \text{ m}) = 0.0484 \text{ m} 2 \text{ Disregarding any radiation effects}, the temperature of the aluminum plate is determined to be <math>Q\& 60 \text{ W} Q\& = hAs$ (Ts $- T\infty$) \rightarrow Ts $= T\infty + = 25^{\circ}\text{C} + = 74.6^{\circ}\text{C} hAs$ (25 W/m 2 .°C)(0.0484 m 2 Disregarding any radiation effects, the temperature of the aluminum plate is determined to be Q& 60 W Q& = hAs (Ts $- T\infty$) \rightarrow Ts $= T\infty + = 25^{\circ}\text{C} + = 74.6^{\circ}\text{C} hAs$ (25 W/m 2 .°C)(0.0484 m 2 Disregarding any radiation effects). 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 temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 20 aluminum fins of rectangular profile on the backside. Then, hcombined = hrad + hconv, 2 = 5.167 + 15 = 20.167 W/m 2. °C. $1 = 0.0229 \ C/W$ 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 Ro = Rtotal The rate of heat loss from the hot tank water then becomes T - T \propto 2 (90 - 10) $C = 2930 \ W \ Q \& = \infty 1$ Rtotal 0.0273 C/W For a temperature drop of 3°C, the mass flow rate of water and the average velocity of water must be $Q\& = m\& C p \Delta T \rightarrow m\& = m\& = \rho VAc \rightarrow V = Q\& 2930 J/s = = 0.234 kg/s C p \Delta T$ (4180 J/kg. 2 Heat conduction in the turkey is onedimensional because of symmetry about the midpoint. Then the energy balance can be written as E - E 1in424out 3 = Net energy transfer by heat, work, and mass ΔE system 1 424 3 Change in internal, kinetic, potential, etc. / 12) ft 1 1 = = 0.2539 h. 2-40 A spherical container of inner radius r1 , outer radius r2 , and thermal conductivity k is given. 4 Heat generation in the wire is uniform. 5-79 Chapter 5 Numerical Methods in Heat Conductivity k is given. 4 Heat generation in the wire is uniform. 5-79 Chapter 5 Numerical Methods in Heat Conduction Time Nodal temperatures, °C step, i T0 T1 T2 T3 T4 T5 T6 0 h (7am) 0 20.0 16.7 13.3 10.0 6.66 3.33 0.0 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 during each time step is determined from Newton's law of cooling using the average temperature at the inner surface of the wall (node 0) as Time i i i -1 & QTrumbe wall $\Delta t = hin A(T0 - Tin) \Delta t = hin$ 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 1 1 0 2 2 QTrumbe wall = hin A[(T0 + T0)/2 - 70°F](0.25 h) = -96.8 Btu The negative sign indicates that heat is transferred to the Trombe wall from the air in the house which represents a heat loss. Analysis The temperature difference between the center and the surface of the fuel rods is determined from g&r 2 (4×107 W/m 3)(0.016 m) 2 = 92.8 °C To - T s = o = Ts 4k 4(27.6 W/m.°C) g 2-74 D Chapter 2 Heat Conduction Equation 2-135 A large plane wall is subjected to convection on the inner and outer surfaces. The rate of heat transfer through the rod is to be determined for the cases of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined on the basis of the energy balance for the transfer through the rod is to be determined for the cases of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the transfer through the rod is to be
determined for the cases of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. The explicit finite difference equations are determined for the case of copper, steel, and granite rod. W / m.° C. Btu / h.ft. Btu / h.f 7 • Node 2: k Outer surfac 8 Glas Tmi+1 - Tmi Δt • 0.2 • 1 • Thermal symmetry i i i +1 i $\Delta y \Delta y$ T2i - T1i $\Delta x \Delta y$ T1 - T1 +k (Ti - T2i Δy T2i - T1i $\Delta x \Delta y$ T1 - T2i Δy T2i - T3i $\Delta x \Delta y$ T3 - T3 +k (To - T3i) + k = $\rho C \Delta y 2 2 \Delta x 2 2 \Delta t T i - T2i \Delta y T3i - T2i$ $\rho C \Delta y 2 2 \Delta x 2 2 \Delta t Node 4$: hi $\Delta y (Ti - T4i \Delta x T1i - T5i T i - T5i$ nodes because of symmetry. = = 114 k 0.5 W / m. The energy balance for the room can be expressed as $E - E = \Delta E$ system 1in424out 3 1 424 3 Net energy transfer by heat, work, and mass Change in internal, kinetic, potential, etc. 2 Heat transfer through the wall is one-dimensional. This additional term $\rho \Delta \Delta x C$ (Tmi+1 – Tmi) / Δt represent the change in the internal energy content during Δt in the transient finite difference formulation. - 0.25 + 0.17 = 11.03 h.ft 2 ·° F / Btu Discussion Note that the effect of doubling the wind velocity on the U-value of the wall is less than 1 percent since ΔU – value 0.0907 – 0.09 Change = = 0.0078 (or 0.78%) U – value 0.09 3-110 Inside WALL Outside 15 mph Chapter 8 Internal Forced Convection undesirable levels when Δ Te differs from Δ Ti by great amounts. 5-49 Chapter 5 Numerical Methods in Heat Conduction 5-55 Heat transfer through a square chimney is considered. The maximum power rating of the transistor is to be determined. 7-9C As a result of streamlining, (a) friction drag increases, (b) pressure drag decreases, and (c) total drag decreases at high Reynolds numbers (the general case), but increases at very low Reynolds numbers ince the friction drag dominates at low Reynolds numbers. 7-80 Chapter 7 External Forced Convection 7-88 The plumbing system of a plant involves some section of a plant involves some s refrigerator has a COP of 2.5, the rate of heat removal from the refrigerated space, which is equal to the rate of heat gain in steady operation, is $Q\& = W\& \times COP = (600 W) \times 2.5 = 1500 W$ e But the refrigerator operates a quarter of the time (5 min on, 15 min off). The surface of the ground is covered with snow at 0°C. $\times 3.08 + 0.52 = 3.97 W$ Then the time of heating becomes 3-53 Chapter 3 Steady Heat Conduction 3-74 A cold aluminum canned drink that is initially at a uniform temperature of 3°C is brought into a room air at 25°C. It is related to the thermal resistance by S=1/(kR). 4b Construction 6 3 1 4a 5 1. The explicit finite difference equations g are determined on the basis of the energy balance for 4 h, h, • • 5 the transient case expressed as $\sum Q\&$ i i + G& element = ρ Velement C All sides Tmi+1 – Tmi Δt • 7 The quantities h, T ∞ , and g& 0 do not change with time, and thus we do not need to use the superscript i for them. ° F)[(15) Discussion The change in the U-value as a result of adding reflective surfaces is ΔU – value 0.978 – 0.618 Change = = 0.368 U – value, nonreflective surfaces. The rate of heat transfer through the window is to be determined Assumptions 1 Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. Analysis The total thermal resistance of the new heat exchanger is $T \sim 1 - T - T (350 - 250)^\circ$ F T $\sim 2 - Rtotal, new = \infty 1 \propto 2 - Rtot$ $= 0.005 \text{ h. m2} + 0.0364 \rightarrow \text{L} = 0.00820 \text{ m} = 8.2 \text{ mm } 3-37 \text{ Chapter } 3 \text{ Steady Heat Conduction } 3-60 \text{ A coat is made of } 5 \text{ layers of } 0.1 \text{ mm thick air space. } 3 \text{ Thermal properties of the plate are constant. Assumptions The thermal properties of the copper ball are constant at room temperature. Properties The thermal properties of the plate are constant. Assumptions The thermal properties of the copper ball are constant at room temperature. Properties The thermal properties of the plate are constant. Assumptions The thermal properties of the copper ball are constant at room temperature. Properties The thermal properties of the plate are constant. Assumptions The thermal properties of the copper ball are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the copper ball are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumptions The thermal properties of the plate are constant. Assumpting thermal properties of thermal prope$ thermal properties of the hot dog are given to be k = 0.76 W/m.°C, $\rho = 980$ kg/m3, Cp = 3.9 kJ/kg.°C, and $\alpha = 2 \times 10-7$ m2/s. Doubling the thickness L doubles the R-value of flat insulation. 5 The Cola Milk Biot number in this case is large (much larger than 0.1). 3-8C For a surface of A at which the convection and radiation heat transfer coefficients are hconv and hrad, the single equivalent heat transfer coefficient is hegy = hconv + hrad when the medium and the surrounding medium can be considered to be a steady heat transfer problem. Analysis The area of the window and the rate of heat loss through it are Glass $A = (6 \text{ ft}) \times (6 \text{ ft}) = 36 \text{ m } 2 \text{ T} - \text{T2} (60 - 42)^\circ \text{F} \text{ Q} \& = \text{kA } 1 = (0.01411 \text{ Btu/h.ft.}^\circ \text{F})(36 \text{ ft } 2) = 439 \text{ Btu/h } \text{L} 0.25 / 12 \text{ ft} \text{ Air } \text{Q} \& 60^\circ \text{F} 1-42 42^\circ \text{F} \text{ Chapter } 1 \text{ Basics of Heat Transfer } 1-82 \text{ Two surfaces of a flat plate are maintained at specified temperatures, and the rate of heat transfer through the plate is measured. } \circ \text{F})(1 \text{ ft }) 2\pi \text{kL}$ 2π (17 = Rtotal, new + Rlimestone, i + Rlimestone, o = 0.005 + 0.00189 + 0.00143 = 0.00832 h. The temperature difference across the exposed surface of the spoon handle is to be determined. Assumptions 1 Heat transfer is transient, but can be treated as steady since the water temperature remains constant during freezing. 3 There is no heat generation in the medium. Substituting, 2.2 & Maximum heat loss: $Q = (6.25 \text{ W/m} \cdot ^{\circ}\text{C})(1.2 \times 1.8 \text{ m})[20 - (-8)]^{\circ}\text{C} = 76 \text{ W}$ Discussion Note that the rate of heat loss through windows of identical size may differ by a factor of 5, depending on the the rate of heat loss through window in the rate of heat how the windows are constructed. °C) = 0.771 W/m. We would place the origin at the center of the potato. 2 The top and side surfaces of the furnace closely approximate black surfaces. 5-39 180 • 180 • 150 Chapter 5 Numerical Methods in Heat Conduction 5-48 A long solid body is subjected to steady two-dimensional heat transfer. Face brick, 4 in 3. The temperature will vary in the radial direction only. The pipe is exposed to cold air and surfaces in the basement, and it experiences a 3°C-temperature drop. 3-71 Chapter 3 Steady Heat Conduction 3-91 A spherical ball is covered with 1-mm thick plastic insulation. 6-23C In turbulent flow, it is the turbulent eddies due to enhanced mixing that cause the friction factor to be larger. Properties The specific heat of water at room temperature is C = 4.18 kJ/kg.°C (Table A-9). Assuming constant thermal conductivity and transfer, the mathematical formulation (the differential equation and the boundary and initial conditions) of this heat conduction problem is to be obtained. ° F)(1260 ft 2) = 5040 Btu / h L 1 ft The rate of heat loss during nighttime is T – T2 Q& night = kA 1 L T1 (55 – 35)°C = (0.40 Btu/h.ft.°F)(1260 ft 2) = 10,080 Btu/h 1 ft The amount of heat loss from the house that night will be Q \rightarrow Q = Q& Δ t = 10Q& day + 14Q& night = (10 h)(5040 Btu / h) + (14 h)(10,080 Btu / h) Q& = Δ t = 191,520 Btu L Q& T2 Then
the cost of this heat loss for that day becomes Cost = (191,520 / 3412 kWh) (\$0.09 / kWh) = \$5.05 3-23 A cylindrical resistor on a circuit board dissipates 0.15 W of power steadily in a specified environment. Properties The thermal conductivity of the glass is given to be k = 0.7 W/m.°C. The final pressure in the tank and the amount of heat transfer are to be determined. Assumptions Heat is generated uniformly in steel plate. Wood stud, 38 mm by 90 mm 7. Air space, 90-mm, reflective with ε = 0.05 4. 5-37 Chapter 3 Steady Heat Conduction 3-68 A long solid body is subjected to steady two-dimensional heat transfer. 3-44 Chapter 3 Steady Heat Conduction 3-68 A long solid body is subjected to steady two-dimensional heat transfer. steam pipe covered with 3-cm thick glass wool insulation is subjected to convection on its surfaces. The left surface temperature is given to be $T0 = 520 R = \Delta x 60^{\circ} F$. The rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat removal from the chicken and the mass flow rate of heat remova cylinder are Bi = hro (40 W/m 2.°C)(0.025 m) = = 0.400 $\rightarrow \lambda$ 1 = 0.8516 and A1 = 10931. Assumptions 1 Heat conduction in the cubic block is three-dimensional, and thus the temperature varies in all x-, y, and z- directions. °F)[2 π (4 / 12 ft)(1 ft)] Then the steady rate of heat loss from theat loss from the steady rate of heat loss from thea 60.2 Btu / h Rtotal Ri + Rpipe + Rins + Ro (0.0364 + 0.00244 + 5516 . 5 cm × = 0.369 3 Hot gases 500°C 5 cm × 5 After 20 minutes $\tau = \alpha t L2 = (115 \cdot Analysis$ The heat flux at the surface of the wire is Q& 1200 W G& = $212.2 \text{ W} / \text{ in } 2 \text{ q& s} = s = \text{As } 2\pi r 0 L 2\pi (0.06 \text{ in})(15 \text{ in})$ Noting that there is thermal symmetry about the center line and there is uniform heat flux at the outer surface, the differential equation and the boundary conditions for this heat conduction problem can be expressed as 1 d (dT) g_{k} |r |+ = 0 r dr (dr) k 2 kW D = 0.12 in dT (0) = 0 dr dT (r0) - k = q_{k} s = 212.2 W / in 2 dr L = 15 in 2-47 Heat conduction through the bottom section of an aluminum pan that is used to cook stew on top of an electric range is considered (Fig. 2-113C A) differential equation that involves only ordinary derivatives is called an ordinary differential equation. 13-36C Turbulence moves the fluid separation point further back on the rear of the wake, and thus the magnitude of the pressure drag (which is the dominant mode of drag). Finally, the problem is solved using an appropriate approach, and the results are interpreted. $\times 10 - 7 \text{ m } 2 / \text{s}$)(1 h $\times 3600 \text{ s} / \text{h}$) (0.045 m) 2 = 0.231 > 0.2 Ti = 20°C Then the temperature at the center of the apples becomes θ o, sph = 2 2 T - (-15) T0 - T ∞ = A1 e - $\lambda 1 \tau \rightarrow 0$ = (1.239)e -(1.476) $(0.231) = 0.749 \rightarrow T0 = 11.2^{\circ}C$ Ti $-T \propto 20 - (-15)$ The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apples is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the surface of the apple is θ (ro, t) $= 2.7^{\circ}C$ 20 - (-15) The temperature at the (840 kg/m 3) [π (0.045 m) 3. 6 The convection heat transfer coefficient is constant and uniform over the surface. Therefore, there will be a heat transfer from outer parts of the rib to the inner parts as a result of this temperature difference. Assumptions 1 This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta mCV = 0$ and $\Delta E CV = 0$, 1-117C Asymmetric thermal radiation is caused by the cold surfaces of large windows, uninsulated walls, or ceiling, solar heated masonry walls or ceilings on the other. Nodes 2, 3, and 4 are interior 2 Upper mid layer nodes, and thus for them we can use the general explicit finite • difference relation expressed as 3 Lower mid layer • g& i Δx 2 Tmi+1 - Tmi i i Tm-1 - 2Tm + Tm + 1 + = 4 Bottom k τ • g& i Δx 2 Black \rightarrow Tmi + 1 = τ (Tmi -1 + Tmi + 1) + (1 - 2\tau)Tmi + τ m x k Node 0 can also be treated as an interior node by using the mirror image concept. The temperature of the tank after a 45-min cooling period is to be estimated. Analysis We consider a rectangular region in which heat conduction is significant in the x and y directions. Applying the boundary conditions give x = 0: T (0) = C1 × 0 + C2 \rightarrow C2 = T1 x = L: $-kC1 = h[(C1 L + C2) - T\infty] \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow
C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty) k + hL \rightarrow C1 = -h(C2 - T\infty)$ $h(T1 - T\infty) k + hL$ Substituting C1 and C2 into the general solution, the variation of temperature is determined to be T (x) = - = - h(T1 - T\infty) x + T1 k + hL (24 W / m 2 · ° C)(0.4 m) = 80 - 1311 .x x + 80° C (c) The rate of heat conduction through the wall is $h(T1 - T\infty) dT = -kAC1 = kA Q\& wall = -kAC1 = -kAC1 = kA Q\& wall = -kAC1 = -kAC$ $-kA dx k + hL (24 W/m 2 \cdot °C)(80 - 15)°C = (2.3 W/m \cdot °C) + (24 W/m 2 \cdot °C)(0.4 m) = 6030 W Note that under steady conditions the rate of heat conduction through a plain wall is constant. Absorptivity is the fraction of radiation incident on a surface that is absorbed by the surface. 6• This problem involves 6 unknown nodal$ temperatures, and Concrete 5 • thus we need to have 6 equations. 4 All the doors and windows are tightly closed, and heat transfer through the walls and the windows is disregarded. 7-45 Chapter 7 External Forced Convection 7-54E An electrical resistance wire is cooled by a fan. The circuit board is attached to a heat sink from both ends maintained at 35°C. The mathematical formulation, the variation of temperature in the sphere, and the center temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature in the sphere, and the center temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are to be determined for steady one-dimensional heat transfer. The variation of temperature are transfer. The vari the exposed surface. 3 Thermal properties of the iron shell are constant. Outside surface, 24 km/h (winter) 1b. For specified indoors and outdoors temperatures, the rate of heat transfer through the window are to be determined. Analysis (a) The surface area of the wall is not given and thus we consider a unit surface area (A = 1 ft2). 3-44C Thermal contact resistance can be minimized by (1) applying a thermally conducting gas such as helium or hydrogen, (3) by increasing the interface pressure, and (4) by inserting a soft metallic foil such as tin, silver, copper, nickel, or aluminum between the two surfaces. 5-68C The general stability criteria for the explicit method of solution of transient heat conduction problems is expressed as follows: The coefficients of all Tmi in the Tmi+1 expressions (called the primary coefficient) in the simplified expressions must be greater than or equal to zero for all nodes m. = = 1061×108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π ro L π (0.001 m) 2 (6 m) The surface temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π (0.001 m) 2 (6 m) 2 (6 m) Temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π (0.001 m) 2 (6 m) 2 (6 m) Temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π (0.001 m) 2 (6 m) 2 (6 m) Temperature of the wire is then (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π (0.001 m) 2 (6 m) 2 (6 m) Temperature of the wire is the matches (Eq. 2-68) Ts = T ∞ + &o gr × 108 W / m 3 2 Vwire π (0.001 m) 2 (6 m) values for each section of the existing wall is determined in the table below. Analysis Using the available R-value of the ceiling is determined in the table below. 2 The thermal conductivity is constant. Properties The outer surface of a spacecraft has an emissivity of 0.8 and an absorptivity of 0.3. Analysis When the heat loss from the outer surface of the spacecraft by radiation equals the solar radiation absorbed, the surface temperature can be determined from Q& solar = $\epsilon\sigma As$ (5.67 × 10 - 8 W/m 2 · K 4)[Ts4 - (0K) 4] Canceling the surface area A and solving for Ts gives Ts = 281.5 K α = 28 0.3 2. Discussion This problem could also be solved by treating the hot dog as an infinite cylinder since heat transfer through the energy balance approach and taking the T 0 direction of all heat transfers to be towards the node under Δx consideration, the finite difference formulations become • D 2• 0 1 T0 - T1 T2 - T1 Node 1 (at midpoint): kA + kA + hp Δx (T ∞ - T1) = 0 Δx where A = $\pi D 2/4$ is the cross-sectional area and p = πD is the perimeter of the fin. 3-57 Chapter 3 Steady Heat Conduction 3-76 Hot water is flowing through a 3-m section of a cast iron pipe. Analysis (a) The characteristic length and the Biot number for the aluminum balls, 250°F V nD 3 / 6 D 2 / 12 ft = = = = 0.02778 ft 6 6 A nD 2 hL (42 Btu/h.ft 2.°F)(0.02778 ft) = 0.00852 < 0.1 Bi = c = (137 Btu/h.ft.°F) k Lc = Water bath, 120°F The lumped system analysis is applicable since Bi < 0.1. Then the temperature of the balls after quenching becomes b = hAs 42 Btu/h.ft 2.°F h = = 41.66 h - 1 = 0.01157 s - 1.3 ρ C pV ρ C p Lc (168 lbm/ft) (0.216 Btu/lbm.°F)(0.02778 ft) - 1 T (t) - T ∞ T (t) = 152°F Ti - T ∞ 250 - 120 (b) The total amount of heat transfer from a ball during a 2-minute period is $m = \rho V = \rho \pi D 3 = (168 \text{ lbm/ft } 3) \pi (2 / 12 \text{ ft}) 3 = 0.4072 \text{ lbm} (0.216 \text{ Btu/lbm.}^\circ F)(250 - 152)^\circ F = 8.62 \text{ Btu Then the rate of heat transfer from the balls to the water becomes } Q_{\&} = n \& Q = (120 \text{ balls/min}) \times (8.62 \text{ Btu}) = 1034 \text{ Btu/min total ball ball Therefore, heat must be}$ removed from the water at a rate of 1034 Btu/min in order to keep its temperature constant at 120 ° F. Analysis Noting that the cross-sectional area of the spoon can be expressed as T (x) - T ∞ cosh a (L - x) = Tb - T ∞ cosh a Lh, T ∞ where Ac = (0.5 / 12 ft)(0.08 / 12 ft) = 0.000278 ft 2 a = hp = kAc 0 Tb p = 2(0.5 / 12 ft + 0.08 / 12 ft) = 0.0967 ft L = 7 in (3 Btu / h.ft 2 . 7-77 Chapter 7 External Forced Convection 7-86 A cylindrical oven is to be insulated to reduce heat losses. Alternative solution We could also solve this problem using transient temperature charts as follows: 0.50W/m.o C $k_1 = = 0.877$ 2 o Bi hro (19W/m. Clothing serves as insulation, and the thicker the clothing, the lower the environmental temperature that feels comfortable. Therefore, the effect of heat transfer through the bottom surface can be
accounted for approximately by increasing the heat transfer from the side surface by 12%. However, heat transfer through the bottom surface can be accounted for approximately by increasing the heat transfer from the side surface by 12%. through any wall or floor takes place in the direction normal to the surface, and thus it can be analyzed as being one-dimensional. $08 ||L + |o|| 22 \text{ kt} / (186 \text{ W/m C})(0.001 \text{ m}) \setminus || 25^{\circ}\text{C}$ Heat transfer from a single fin is Afin = $2\pi (r_2 - r_1) + 2\pi r_2 t = 2\pi (0.03 2 - 0.025 2) + <math>2\pi (0.03)(0.001) = 0.001916 \text{ m} 2 \text{ Q} \&$ fin = η fin Q& fin, max = η fin Afin $(Tb - T\infty) 2 2 = 0.97(40 \text{ W/m } 2.^{\circ}C)(0.001916 \text{ m } 2)(180 - 25)^{\circ}C = 11.53 \text{ W}$ Heat transfer from a single unfinned portion of the tube is Aunfin = hAunfin (Tb - T ∞) = $(40 \text{ W/m } 2.^{\circ}C)(0.004712 \text{ m } 2)(180 - 25)^{\circ}C = 2.92 \text{ W}$ There are 250 fins and thus 250 interfin spacings per meter length of the tube. $e^{-1.5} = 0.577 \text{ Ti} - T_{\infty}$ Then the center temperature of the short cylinder becomes T (0,0, t) - T_{\infty} = 0.501 \rightarrow T (0,0, t) - 20 = 0.501 \rightarrow T (0,0, t) - 20 = 0.501 \rightarrow T (0,0, t) - 20 = 0.501 \rightarrow T (0,0, t) - T_{\infty} = 0.501 \rightarrow T (0,0, t) - 20 = 0.501 \rightarrow T (0,0, t) - 20 = 0.501 \rightarrow T (0,0, t) - T_{\infty} = 0.501 \rightarrow T (0,0, t) - 20 = 0.501 \rightarrow x where x = 0, but at the outer surface of the plane wall (x = L), ° C 25°F 1 ft/ Orange D = 2.5 in 85% water From Table 4-1 we read, for a sphere, $\lambda 1 = 1.9569$ and A1 = 1.447, ° C) = 14.40 m-1 (386 W/m. (b) The 6 nodal temperatures under steady conditional temperatures and the steady conditional temperatures and temperatures a simultaneously with an equation solver to be T4 =539.6°C, and T5 =530.0°C T0 =556.8°C, T1 =555.7°C, T2 =552.5°C, T3 =547.1°C, Discussion This problem can be solved analytically by solving the differential equation as described in Chap. Properties The thermal conductivity of the concrete is given to be Tsky = 100 K k = 2 W/m·°C.) / 12]ft(155.7°C, T3 =547.1°C, Discussion This problem can be solved analytically by solving the differential equation as described in Chap. Properties The thermal conductivity of the concrete is given to be Tsky = 100 K k = 2 W/m·°C.) / 12]ft(155.7°C, T3 = 547.1°C, Discussion This problem can be solved analytically by solving the differential equation as described in Chap. This is probably due to the Fourier number being less than 0.2. (b) The temperature at the surface of the rib is θ (ro, t) – T ∞ sin(3.0372 rad) = A1 e – λ 1 τ = (1.9898)e –(3.0372) (0.1217) λ 1 ro / ro 3.0372 ri – T ∞ T (ro, t) – 163 = 0.0222 \rightarrow T (ro, t) = 159.5 °C 4.5 – 163 4-32 Chapter 4 Transient Heat Conduction (c) The maximum possible heat transfer is Q max = mC p ($T \infty - Ti$) = (3.2 kg)(4.1 kJ/kg. The finite difference formulation of the spoon as well as the rate of heat transfer from the exposed surfaces of the spoon are to be determined. Then the actual heat transfer becomes $(Q \mid Q \mid max) J(\lambda) 0.5760 \mid = 1 - 20 \text{ o,cyl } 1 = 1 - 2(0.2727) = 0.8489 \rightarrow Q = 0.8489(13,440 \text{ kJ}) = 11,409 \text{ kJ} \mid \lambda 2.0785 1 \mid cyl 4-38 \text{ Chapter 4 Transient Heat Conduction 4-49E Whole chickens are to be cooled in the racks of a large refrigerator. The R-value of the wall is given to be 19 h.ft2·°F/Btu. Analysis The mass of the copper ball$ and the maximum amount of heat transfer from the copper ball are $[\pi (0.15 \text{ m}) 3] (\pi D 3) = (8933 \text{ kg/m } 3) m = \rho V = \rho | = 15.79 \text{ kg} (0.385 \text{ kJ/kg.}^{\circ}C)(200 - 25)^{\circ}C = 1064 \text{ kJ} 4-17 \text{ Q Copper ball}, 200^{\circ}C \text{ Chapter 4 Transient Heat Conduction Discussion The student's result of 4520 kJ is not$ reasonable since it is greater than the maximum possible amount of heat transfer. Boundary and Initial Conditions; Formulation of Heat Conduction Problems 2-34C The mathematical expressions of the thermal conditions; Formulation of Heat Conduction Problems 2-34C The mathematical expressions of the thermal conditions at the boundary conditions. explicit finite difference relation expressed as $Tmi-1 - 2Tmi + Tmi+1 + g\& i \Delta x 2 g\& mi \Delta x 2 Tmi+1 - Tmi = \rightarrow Tmi + 1 = \tau (Tmi - 1 + Tmi + 1) + (1 - 2\tau)Tmi + \tau m k \tau k$ The finite difference equations for nodes 1 and 4 subjected to convection and radiation are obtained from an energy balance by taking the direction of all heat transfers to be towards the node under consideration: Node 1 (convection) : T2i - T1i Δx T1i +1 - T1i = ρ C Δx 2 Δt i i i = τ (T1 + T3) + (1 - 2 τ)T3i Node 4 (convection) : hi (T5i - T4i) + k T3i - T4i Δx T4i +1 - T4i = ρ C Δx 2 Δt where k = 0.026 W/m.°C, $\alpha = k / \rho$ C $= 0.36 \times 10 - 6 \text{ m } 2 / \text{s}$, T5 = Ti = 3°C (initially), To = 25°C, hi = 6 W/m2.°C, ho = 9 W/m2.°C, ho = T0 1 2 3 4 5 Left boundary node: kA 1 + g& 0 ($A\Delta x / 2$) = 0 Δx Right boundary node: 4 $\varepsilon \sigma A$ (Tsurr - T54) + kA T 4 - T5 + g& 5 ($A\Delta x / 2$) = 0 Δx 5-6 Chapter 5 Numerical Methods in Heat Conduction, radiation, and heat flux at the left (node 0) Ts g(x) and specified temperature at the right boundary (node Tsurr 5). 3-58 Chapter 3 Steady Heat Conductivity is given to be k = 2.3 W/m·°C. 2-10C Isotropic materials have the same properties in all directions, and we do not need to be concerned about the variation of properties with direction for such materials. energies & 1 = Q& out + mh & 2 (since $\Delta ke \cong \Delta pe \cong 0$) W&e,in + mc Substituting, the power rating of the heating element is determined to be = (0.25 kW) - (0.3 kW) + (0.6 kg/s)(1.007 kJ/kg) · °C)(5°C) W& 250 W W e,in 300 W = 2.97 kW 1-18 Chapter 1 Basics of Heat Transfer 1-40 Air is moved through the resistance heaters in a 1200-W hair dryer by a fan. The error involved in the total thermal resistance heaters in a 1200-W hair dryer by a fan. are incompressible substances with constant specific heats at room temperature. Properties The thermal conductivity and thermal diffusivity of carcass are given to be $k = 0.47 \text{ W/m} \cdot \text{C}$ and $\alpha = 0.13 \times 10-6 \text{ m}^2/\text{s}$. Inside surface, still air R-value, h.ft2.°F/Btu Between At furring furring 0.17 0.17 0.43 0.43 0.10 0.10 1.51 1.51 2.91 --0.94 0.45 0.45 0.68 0.683 Total unit thermal resistance of each section, R The U-factor of each section, U = 1/R, in Btu/h.ft2.°F Area fraction of each section, U = 1/R, in Btu/h.ft2.°F 5.72 h.ft2.°F/Btu Therefore, the overall unit thermal resistance, R = 1/U 6.25 4.28 0.160 0.234 0.80 0.20 0.175 Btu/h.ft2.°F 5.72 h.ft2.°F/Btu Therefore, the overall unit thermal resistance, R = 1/U 6.25 4.28 0.160 0.234 0.80 0.20 0.175 Btu/h.ft2.°F 5.72 h.ft2.°F/Btu Therefore, the overall unit thermal resistance of the wall is R = 5.72 h.ft2.°F/Btu and the overall U-factor is U = 0.118 Btu/h.ft2.°F. This is done by first selecting the nodes by drawing lines through the midpoints between the nodes. 5-59 Chapter 5 Numerical Methods in Heat Conduction 5-60 "!PROBLEM 5-60E" "GIVEN" T top=32 "[F], parameter to be varied" T bottom=212 "[F], parameter to be varied" T 2=T 10 "due to symmetry" T 2=T 10 "due to symmetry" T 4=T 7 "due to symmetry" T 5=T 8 "due to symmetry" "Using the finite difference method, the six unknown temperatures are determined to be" $k^{1/2}(T \text{ top-T }1)/(1+k^{1/2}(T 2-T 1)+(T \text{ top-T }1)+(1/2 (T 2-T 1))) = 0$ "Node 1" T 1+2*T 4+T 3-4*T 2=0 "Node 2" T 2+T bottom+2*T 5-4*T 3=0 "Node 3" T 2+T bottom+2*T 5-4*T 3=0 "Node 1" T 1/2*(T 2-T 1)/(1+k^{1/2}(T 2-T 1))) = 0 2*T top+T 2+T 5-4*T 4=0 "Node 4" T 3+T bottom+T 4+T 6-4*T 5=0 "Node 6" T top+T bottom+2*T 5-4*T 6=0 "Node 6" T top+T bottom+2*T 5-4*T 6=0 "Node 6" T top+T 103.2 110 116.8 123.6 130.4 137.2 144 150.8 157.6 164.4 171.2 178 184.8 191.6 198.4 205.2 212 5-60 Chapter 5 Numerical Methods in Heat Conduction Tbottom [F] 32 41.47 50.95 60.42 69.89 79.37 88.84 98.32 107.8 117.3 126.7 135.2 164.6 174.1 183.6 193.1 202.5 212 T2 [F] 32 34.67 37.35 40.02 42.7 45.37 48.04 50.72 53.39 56.07 58.74 61.41 64.09 66.76 69.44 72.11 74.78 77.46 80.13 82.81 225 195 T 2 [F] 165 135 105 75 25 65 105 145 T top [F] 5-61 185 225 Chapter 5 Numerical Methods in Heat Conduction 90 80 T 2 [F] 70 60 50 40 30 25 65 105 145 T bottom [F] 5-62 185 225 Chapter 5 Numerical Methods in Heat Conduction 5-61 The top and bottom surfaces of an L-shaped long solid bar are maintained at specified temperatures while the left surface is insulated and the remaining 3 surfaces are subjected to convection. 4-112 A spherical watermelon that is cut into two equal parts is put into a freezer. 2-28 We consider a small rectangular element of length Δx , width Δy , and height $\Delta z = 1$ (similar to the one in Fig. ° F)(0.03 / 12 ft) = 0.5575 Btu / h. The rate of heat loss through the window and the inner surface temperature are to be determined. The temperature jump at the interface is determined from & Δ Tinterface = QR contact = (20.8 W)(0.0227 ° C / W) = 0.47° C which is not very large. Ts =-10°C Properties The thermal properties of the soil are given -5.2 to be k = 0.7 W/m.°C and α = 1.4×10 m/s. Assumptions 1 Heat transfer through the base plate is given to be steady. The average wind velocity is to be estimated. Properties The thermal conductivity is given to be $k = 0.77 \text{ W/m} \cdot \text{C}$. Analysis The inner and outer surface areas of the insulated tank and the volume of the LNG are A1 = π D12 = π (6 m) 2 = 113.1 m 2 A2 = π D 2 =
π (6.10 m) 2 = 116.9 m 2 LNG tank -160° C V1 = π D13 / 6 = π (6 m) 3 / 6 = π (6 m) 3 / 6 = 113.1 m 3 The rate of heat transfer to the LNG is r -r (3.05 - 3.0) m Rinsulation = $2.1 = 5.43562^{\circ}$ C/W 4π kr1 r2 4π (0.00008 W/m.°C)(3.0 m)(3.05 m) 1 1 = $0.00039 + 5.43562 = 5.43601^{\circ}$ C/W Ro = Rtotal T -T [18 - (-160)]^{\circ}C Q& = $\infty 2.12 = 5.43562^{\circ}$ C/W 4π kr1 r2 4π (0.00008 W/m.°C)(3.0 m)(3.05 m) 1 1 = $0.00039 + 5.43562 = 5.43601^{\circ}$ C/W Ro = Rtotal T -T [18 - (-160)]^{\circ}C Q& = $\infty 2.12 = 5.43562^{\circ}$ C/W 4π kr1 r2 4π (0.00008 W/m.°C)(3.0 m)(3.05 m) 1 1 = 0.00039° C/W $2 = 10.00039^{\circ}$ C/W 2 = 10.0003= 32.74 W R total 5.43601 °C/W T1 Rinsulation Ro To2 The amount of heat transfer to increase the LNG temperature from -160° C to -150° C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = 48,067.5 kg/(3.475 kJ/kg.°C)[(-150) - (-160)^{\circ}C is m = ρ V1 = (425 kg/m 3)(113.1 m 3) = (425 kg/m 3)(113.1 m 3)(113.1 m 3) = (425 kg/m 3)(113 the time period for the LNG temperature to rise to -150° C becomes Q 1,670,346 kJ $\Delta t = = 51,018,498$ s = 14,174 h = 590.5 days Q& 0.03274 kW 3-134 Chapter 3 Steady Heat Conduction 3-174 A hot plate is to be cooled by attaching aluminum fins of square cross section on one side. Analysis (a) The thermal resistance of the board and the convection resistance on the backside of the board are Rconv Rboard L 0.002 m Rboard = = $0.011 \circ C / W T1 T \propto kA$ (12 W / m. 3 The energy stored in the paddle wheel is negligible. Properties The thermal conductivity of the soil is given to be k = $0.6 \text{ Btu/h} \cdot \text{ft} \cdot \text{s}$. (b) The 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under steady conditions are determined by solving the 3 nodal temperatures under stead equations above simultaneously with an equation solver to be T1 = 85.7°C, T2 = 86.4°C, T3 = 87.6°C 5-63 Insulated • 1 120°C • 2 3 • Chapter 5 Numerical Methods in Heat Conduction 5-62 A rectangular block is subjected to uniform heat flux at the top, and iced water at 0°C at the sides. 2-38C Yes, the temperature profile in a medium must be perpendicular to an insulated surface since the slope $\partial T / \partial x = 0$ at that surface. Analysis The volume and mass of the air in the house are V = (floor space)(height) = (200 m2)(3 m) = 600 m3 PV (1013 . The amount of heat dissipated in 24 h, the surface heat flux, and the surface temperature of the resistor are to be determined. Then heat conduction along this two-layer board can be expressed as [] $\Delta T (\Delta T) (\Delta T) + kA Q = Q copper + Q epoxy = kA = (kt) copper + tepoxy and thermal conductivity keff can be expressed as <math>\Delta T (\Delta T) Q = kA = keff$ (t copper + tepoxy) w L board L Setting the two relations above equal to each other and solving for the effective conductivity gives (kt) copper + (kt) epoxy \rightarrow k eff = t copper + t epoxy) = (kt) copper + t epoxy \rightarrow k eff = t copper + t epoxy Note that heat conduction is proportional to kt. Then the Fourier number and the time period become Potato θ 0, sph = T0 = $70^{\circ}C 2 2 T0 - T \propto 70 - 170 = A1e - \lambda 1 \tau \rightarrow = 0.69 = (1.4113)e - (1.8777) \tau \rightarrow \tau = 0.203 > 0.2 Ti - T \propto 25 - 170$ The baking time of the potatoes is determined to be t= $\tau ro 2 (0.203)(0.04 \text{ m}) 2 = 2320 \text{ s} = 38.7 \text{ min } \alpha (14 \text{ . The rate of heat transfer to the tank and the LNG temperature at the end of a one-month period are to be determined. 5-17$ Chapter 5 Numerical Methods in Heat Conduction 5-29 A plate is subjected to specified heat flux and specified temperature on one side, and no conditions on the other. Properties of air at -50°C and 1 atm are (Table A-15) Cp = 0.999 kJ/kg-K Pr = 0.7440 Air The density of air at -50°C and 26.5 kPa is -50°C P 26.5 kPa 800 km/h ρ = = = 0.4141 kg/m 3 RT (0.287 kJ/kg.K)(-50 + 273) K Analysis The average heat transfer coefficient can be determined from the modified Reynolds analogy to be h= Wing Ts=4°C C f ρ V C p Pr 2/3 0.0016 (0.4141 kg/m 3)(800 / 3.6 m/s)(999 J/kg · °C) = 89.6 W/m 2 · C 2/3 2 (0.7440) 2 6-54, 6-55 Design and Essay Problems KJ 6-27 25 m 3m Chapter 7 External Forced Convection Chapter 7 EXTERNAL FORCED CONVECTION Drag Force and Heat Transfer in External Flow 7-1C The velocity of the fluid relative to the immersed solid body sufficiently far away from a body is called the free-stream velocity, V^o. The time the ice block starts melting is to be determined. Btu / h · ft·° F) = 0.8(01714 . e-(1.635) 2 $\tau \rightarrow \tau = 0.753 = 1302$ 25 - 2 which is greater than 0.2 and thus the one-term solution is applicable. 2 The emissivity and thermal conductivity of the roof are constant. 2 Heat conduction in the rib is one-dimensional because of symmetry about the midpoint. 3 The thermal properties of the meat slabs are constant. C)(3.267 m 2) Tw ln(r2 / r1 $\ln(23/20) = 0.371 \ ^{\circ}C/W \ R \ foam = 2\pi k1 \ L \ 2\pi(0.03 \ W/m \ 2 \ ^{\circ}C)(2 \ m) \ Ro = R \ foam \ R \ fiberglass = 0.279 \ ^{\circ}C/W \ 2\pi \ 2 \ L \ 2\pi(0.035 \ W/m \ 2 \ ^{\circ}C)(2 \ m) = Ro + R \ foam + R \ fiberglass = R \ foam \ R \ foam \ R \ foam \ R \ foam \ R \ fiberglass = R \ foam \ R \ fo$ = 41.42 W Q& = w 0.676 °C/W Rtotal The energy saving is saving = 70 - 41.42 = 28.58 W The time necessary for this additional insulation to pay for its cost of \$30 is then determined to be 3-49 Chapter 3 Steady Heat Conduction Cost = (0.02858 kW)(Time period)(\$0.08 / kWh) = \$30 Then, Time period = 13,121 hours = 547 days ≈ 1.5 years 3-50 Chapter 3 Steady Heat Conduction 3-72 "GIVEN" L=2 "[m]" D i=0.40 "[m]" D i=0.46 "[m]" T 1=D i/2 r 2=D o/2 "T w=55 [C], parameter to be varied" T infinity 2=27 "[C]" h i=50 "[W/m^2-C]" k ins=0.03 "[W/m^2-C]" k ins=0.04 [W/m^2-C]" k ins=0.03 "[W/m^2-C]" R conv i=1/(h i*A i) R ins=ln(r 2/r 1)/(2*pi*k ins*L) R conv o=1/(h o*A o) R total=R conv i+R ins+R conv o Q dot=(T w-T infinity 2)/R total Q=(Q dot*Convert(W, kW))*time time=365*24 "[h/year]" Cost HeatLoss=Cost HeatLoss=Cost HeatLoss/Cost heating*Convert(, %) Tw [C] 40 45 50 55 60 65 70 75 80 85 90 fHeatLoss [%] 7.984 11.06 14.13 17.2 20.27 23.34 26.41 29.48 32.55 35.62 38.69 3-51 Chapter 3 Steady Heat Conduction 40 35 f HeatLoss [%] 30 25 20 15 10 5 40 50 60 70 T w [C] 3-52 80 90 Chapter 3 Steady Heat Conduction 3-73 A cold aluminum canned drink that is initially at a uniform temperature of 3°C is brought into a room air at 25°C. The h, T∞ k(T)
complete finite difference formulation of this problem is Tsurr to be obtained. The surface temperature of the wire then becomes As = $\pi DL = \pi (0.006 \text{ m})(1 \text{ m}) = 0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes = hAs (Ts - T\infty) 2 \text{ hAs } (146.3 \text{ W/m} \cdot \text{C})(0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes = hAs (Ts - T\infty) 2 \text{ hAs } (146.3 \text{ W/m} \cdot \text{C})(0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes = hAs (Ts - T\infty) 2 \text{ hAs } (146.3 \text{ W/m} \cdot \text{C})(0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes = hAs (Ts - T\infty) 2 \text{ hAs } (146.3 \text{ W/m} \cdot \text{C})(0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes = hAs (Ts - T\infty) 2 \text{ hAs } (146.3 \text{ W/m} \cdot \text{C})(0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes = hAs (Ts - T\infty) 2 \text{ hAs } (146.3 \text{ W/m} \cdot \text{C})(0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes = hAs (Ts - T\infty) 2 \text{ hAs } (146.3 \text{ W/m} \cdot \text{C})(0.01885 \text{ m } 2 \text{ Q} \otimes 5W \rightarrow Ts = T\infty + = 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \text{ Q} \otimes 10^{\circ}\text{C} + = 11.8^{\circ}\text{C} \times 10^{\circ}\text{C} + = 11.8^{\circ}\text{C$ D=0.006 "[m]" L=1 "[m], unit length is considered" I=50 "[Ampere]" R=0.002 "[Ohm]" T_infinity=10 "[C]" "Vel=40 [km/h], parameter to be varied" "PROPERTIES" Fluid\$, T=T_film) Pr=Prandtl(Fluid\$, T=T_film) Pr=Prandtl(Fl (T s+T infinity) "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 W dot=I^2*R Q dot=W dot A=pi*D*L Q dot=h*A*(T s-T infinity) Vel [km/h] 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 Ts [C] 13.72 13.02 12.61 12.32 12.11 11.95 11.81 (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5) (1.5 11.7 11.61 11.53 11.46 11.4 11.34 11.29 11.25 7-37 Chapter 7 External Forced Convection 14 13.5 T s [C] 13 12.5 12 11.5 11 10 20 30 40 50 Vel [km /h] 7-38 60 70 80 Chapter 7 External Forced Convection 7-48 An aircraft is cruising at 900 km/h. The minimum free-stream velocity that the fan should provide to avoid overheating is to be determined 3-145 The winter R-value and the U-factor of a masonry wall are to be determined. = 0.645 m2 · ° C / W Winter WALL Ro, summer = 0.645 - 0.030 + 0.044 = 0.659 m 2 · °C/W Then the summer U-value of the wall becomes Rsummer = 1 / 0.659 = 1.52 m 2 · °C / W 3-108 Summer Chapter 3 Steady Heat Conduction 3-147 The U-value of a wall is given. 2-36C A heat transfer problem that is symmetric about that plane, line, or point is said to have thermal symmetry about that plane, line, line, or point is said to have thermal symmetry about that plane, line, or point. This is a constant volume closed system since no mass crosses the system boundary. We can also meet the constraints by using a lower heat transfer coefficient, but doing so would extend the refrigeration time unnecessarily. For convenience, let us choose the time step to be $\Delta t = 10$ s. Analysis Under steady conditions, the rate of heat transfer by convection is Q& conv = hAs $\Delta T = (55W/m 2 \cdot o C)(2 \times 4 m 2)(80 - 30) o C = 22,000W 80^{\circ}C Air 30^{\circ}C 1-34$ Chapter 1 Basics of Heat Transfer 1-75 "GIVEN" T infinity=80 "[C]" h=55 [W/m^2-C], parameter to be varied" "ANALYSIS" Q dot conv=h*A*(T infinity=T s) h [W/m2.C] 20 30 40 50 60 70 80 90 100 8000 12000 16000 20000 24000 28000 32000 36000 40000 35000 30000 Q conv [W] 25000 20000 15000 10000 5000 20 30 40 50 60 70 2 h [W /m -C] 1-35 80 90 100 Chapter 1 Basics of Heat Transfer 1-36 Chapter 1 Basics 0 Basics 0 Basics 0 Basics 0 Basics 0 Basics the certain substrate. Analysis The temperatures at the center and at the surface of the ball are h D determined directly from T ∞ 6 3 g&r (2.6 × 10 6 W/m 3) (0.15 m) Ts = T ∞ + 0 = 0°C + = 108.3°C + = 325°C 6k 6(45 W/m.°C) 2-138 2 Heat transfer through the window is one dimensional. Also, the equivalent thickness of a wool coat is to be determined. Assumptions 1 Heat transfer through the wall is steady and one-dimensional. 4 2 / 3 1 1 + 1 1 1 0.7306 / 1 Then the heat transfer rate from the arm becomes k 0.01457 Btu/h.ft. F h = Nu = (129.6) = 7.557 Btu/h.ft 2 °F D (3 / 12) ft 4/5 = 129.6 As = π DL = π (3 / 12 ft)(2 ft) = 1.571 ft 2 Q& conv = hAs (Ts - T ∞) = (7.557 Btu/h.ft 2 .°F)(1.571 ft 2)(86 - 54)°F = 379.8 Btu/h 7-31 Chapter 7 External Forced Convection 7-43E" "IPROBLEM 7-43E" "GIVEN" T_infinity=54" [F], parameter to be varied" "Vel=20 [mph], parameter to be varied" T_s=86 "[F]" L=2 "[ft]" D=3/12 "[ft]" "PROPERTIES" Fluid\$='air' k=Conductivity(Fluid\$, T=T_film) Pr=Prandtl(Fluid\$, T=T_film) rho=Density(Fluid\$, T=T_film)*Convert(lbm/ft-h, lbm/ft-s) nu=mu/rho T_film=1/2*(T_s+T_infinity) "ANALYSIS" Re=(Vel*Convert(mph, ft/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 Q dot conv=h*A*(T s-T infinity) Toc [F] 20 25 30 35 40 45 50 55 60 65 70 75 80 Qconv [Btu/h] 790.2 729.4 668.7 608.2 547.9 487.7 427.7 367.9 308.2 248.6 189.2 129.9 70.77 Vel [mph] 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40 Qconv [Btu/h] 250.6 278.9 305.7 331.3 356 379.8 403 $425.6\ 447.7\ 469.3\ 490.5\ 511.4\ 532\ 552.2\ 572.2\ 572.2\ 591.9\ 7-32\ Chapter\ 7\ External\ Forced\ Convection\ 800\ 700\ 600\ Q\ conv\ [Btu/h]\ 500\ 400\ 350\ 200\ 10\ 15\ 20\ 25\ 30\ Vel\ [m\ ph]\ 7-33\ 35\ 40\ Chapter\ 7\ External\ Forced\ Convection\ 7-44\ The\ average\ surface\ temperature\ of\ the\ 500\ 400\ 300\ 250\ 10\ 15\ 20\ 25\ 30\ Vel\ [m\ ph]\ 7-33\ 35\ 40\ Chapter\ 7\ External\ Forced\ Convection\ 7-44\ The\ average\ surface\ temperature\ of\ the\ 500\ 400\ 350\ 300\ 250\ 10\ 15\ 20\ 25\ 30\ Vel\ [m\ ph]\ 7-33\ 35\ 40\ Chapter\ 7\ External\ Forced\ Convection\ 7-44\ The\ average\ surface\ temperature\ of\ the\ 400\ 350\ 300\ 250\ 10\ 15\ 20\ 25\ 30\ Vel\ [m\ ph]\ 7-33\ 35\ 40\ Chapter\ 7\ External\ Forced\ Convection\ 7-44\ The\ average\ surface\ temperature\ surface\ surfa$ head of a person when it is not covered and is subjected to winds is to be determined. The cooling time and the surface temperature of the slabs at the end of the cooling process are to be determined. Nondimensionalization also results 6-5 Chapter 6 Fundamentals of Convection in similarity parameters (such as Reynolds and Prandtl numbers) that enable us to group the results of a large number of experiments and to report them conveniently in terms of such parameters. ° C)(0.0216 m2) L 0.002 m Rcopper = = 0.00024 ° C / W kA (386 W / m. m) + 2 π (0.06 m) 2 = 0.0292 m2 4 4 = ho A(Tair - Tcan, ave) = (10 W / m 2). The rate of heat transfer from the 1 m by 1 m section of the plate and the effectiveness of the fins are to be determined. 1-11C For the constant pressure case. $^{\circ}$ C) = 0.013 m = 13 mm > r2 (= 7 mm) rcr = = h 20 W / m 2 . $^{\circ}$ C) = 18.37 m-1 (237 W / m. 3 Thermal conductivity is constant. The rate of heat loss from the pipe by convection is to be determined. In the case of the popular finite difference method, this is done by replacing the derivatives by differences. Analysis Noting that the heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant,
the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant, the rate of heat transfer area (the area normal to the direction of heat transfer) is constant. $= 1.964 \times 10 - 3 \text{ m } 2$ Then the heat transfer rate for each case is determined as follows: L=0.15 m (a) Copper: T -T (95 - 20)° C Q& = kA 1 2 = (18 W / m·° C)(1.964 \times 10 - 3 m 2) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q& = kA 1 2 = (12 . 2-5 m (2 - 20)° C Q) = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T (95 - 20)° C Q = 17.7 W L 0.15 m T -T Chapter 2 Heat Conduction Equation 2-21 We consider a thin element of thickness Δx in a large plane wall (see Fig. 3 The pressure in the house remains constant at all times. Properties The thermal conductivity of the ground is given to be k = 0.015 Btu/h·ft·°F. Analysis The maximum power at which this transistor can be operated safely is $T - T \propto (80 - 40)$ °C $\Delta T Q \& = = case = = 1.6$ W Rcase-ambient 25 °C/W Ts R T $\propto 3-108$ A commercially available heat sink is to be selected to keep the case temperature of a transistor below 90° C in an environment at 20 °C. Therefore, b = hAs 40 W/m 2 . °C h = = = $0.000544 \text{ s} - 1 \rho \text{C} \text{ pV} \rho \text{C} \text{ pL} \text{ c} (3.675 \times 10.6 \text{ J/m} 3.^{\circ} \text{C})(0.02 \text{ m}) - 1 \text{ T} (t) - T \infty 0 - 50 \text{ = } \text{e} - \text{bt} \rightarrow \text{= } \text{e} - (0.000544 \text{ s}) \text{t} \rightarrow \text{t} = 482 \text{ s} = 8.0 \text{ min Ti} - T \infty - 15 - 50 \text{ k} 43 \text{ W/m}$. C = $3.675 \times 10.6 \text{ J/m} 3.^{\circ} \text{C} \alpha 1.17 \times 10 - 5 \text{ m} 2 \text{ /s}$ Alternative solution: This problem can also be solved using the transient chart Fig. The thickness that corresponds to the minimum total cost is the optimum thickness of insulation, and this is the recommended thickness of insulation to be installed. Properties The density and Specific heat of the copper ball are $\rho = 8933$ kg/m3, and Cp = 0.385 kJ/kg.°C (Table A-3). Assumptions The hot water temperature changes from 80°C at the beginning of shower to 60°C at the end of shower 4 Heat transfer by radiation is accounted for in the heat transfer coefficient. It is caused by the components of the pressure and wall shear forces in the normal direction from zero at the wall surface to nearly the free stress of the pressure and wall shear forces in the normal direction from zero at the wall surface to nearly the free stress of the pressure and wall shear forces in the normal direction from zero at the wall surface to nearly the free stress of the pressure and wall shear forces in the normal direction from zero at the wall surface to nearly the free stress of the pressure and wall shear forces in the normal direction from zero at the wall surface to nearly the free stress of the pressure and wall shear forces in the normal direction from zero at the wall surface to nearly the free stress of the pressure and wall shear forces in the normal direction from zero at the wall surface to nearly the free stress of the pressure at the normal direction from zero at the wall surface to nearly the free stress of the pressure at the normal direction from zero at the wall surface to nearly the free stress of the pressure at the normal direction from zero at the wall surface to nearly the free stress of the normal direction from zero at th stream value across the relatively thin boundary layer, while the variation of u with x along the flow is typically small. 255120 kJ / s & fg Q& = mh \rightarrow m& = = = 1.20 kg / s h fg 213 kJ / kg 1-47 Chapter 1 Basics of Heat Transfer 1-90 "GIVEN" D=4 "[m]" T s=-196 "[C]" "T air=20 [C], parameter to be varied" h=25 "[W/m^2-C]" "PROPERTIES" h_fg=198 "[kJ/kg]" "ANALYSIS" A=pi*D^2 Q_dot=h*A*(T_air-T_s) m_dot_evap=(Q_dot*Convert(J/s, kJ/s))/h_fg Tair [C] 0 2.5 5 7.5 10 12.5 15 17.5 20 22.5 25 27.5 30 32.5 35 mevap [kg/s] 1.244 1.26 1.276 1.292 1.307 1.323 1.339 1.355 1.371 1.387 1.403 1.418 1.434 1.45 1.466 1-48 Chapter 1 Basics of Heat Transfer 1.5 1.45 m evap [kg/s] 1.4 1.35 1.3 1.25 1.2 0 5 10 15 20 T air [C] 1-49 25 30 35 Chapter 1 Basics of Heat Transfer 1-91 A person with a specified surface temperatures. Also, hfg = 213 kJ/kg for oxygen. In turbulent, the hydrodynamic and thermal entry lengths are independent of Re or Pr numbers, and are comparable in magnitude. We get r k ln 2 + r1 hi r1 T (r2) = Ti + r2 k k ln + + r1 hi r1 T (r2) = Ti + r2 k k ln + + r1 hi r1 hor2 2-66 Chapter 2 Heat Conduction Equation 2-126 A spherical liquid nitrogen container is subjected to specified temperature on the inner surface and convection on the outer surface. ° C Since the outer temperature of the ball with insulation is smaller than critical radius of insulation, plastic insulation will increase heat transfer from the wire. 4 The power consumed by the fan and the heat losses through the walls of the hair dryer are negligible. Still air, horizontal surface, facing down Acoustical Total unit thermal resistance (thetiles R-value) R-value, m2.°C/W R = 1/h = 1/4.32 = 0.23 0.32 R = 1/h $= 1/9.26 = 0.11 \ 0.66 \ m^2.^{\circ}C/W$ Therefore, the R-value of the hanging ceiling is 0.66 m^2.^{\circ}C/W. For flow over a plate of length L it is defined as Re = VL/v where V is flow velocity and v is the kinematic viscosity of the fluid. For an air space with one-reflective surface, we have $\varepsilon 1 = 0.05$ and $\varepsilon 2 = 0.9$, and thus ε effective $= 1.1 = 0.05 \ 1/\varepsilon 1 + 1/\varepsilon 2$ -11/0.05 + 1/0.9 - 1 Using the available R-values from Tables 3-6 and 3-9 and calculating others, the total R-values for each section of the existing wall is determined in the table below. qs Analysis The nodal spacing is given to be $\Delta x = 0.1$ ft. Then, b= hAs 35 W/m 2. °C h = = = 0.0193 s $-1 \rho C p V \rho C p Lc (2702 kg/m 3)(896 J/kg.°C)(0.00075 m) -1 1/(0.00075 m) -1 1/(0.00075 m) -1 1/(0.00075 m) -1 0.00075 m)$ $T(t) - T \propto 50 - 30 = e - bt \rightarrow = e - (0.0193 \text{ s})t \rightarrow t = 144 \text{ s} 350 - 30 \text{ Ti} - T \propto (b)$ The wire travels a distance of velocity = length \rightarrow length = (10 / 60 m/s)(144 s) = 24 m time This distance can be reduced by cooling the wire in a water or oil bath. The temperature distribution in the wall in 12 h intervals and the amount of heat transfer during the first and second days are to be determined. \times 106 W / m³)(0.003 m) 2 = 97.1° C + = 97.3° C 4 k wire 4 × (18 W / m[•] C) Thus the temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline
will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will be slightly above the interface temperature of the centerline will b to be determined. 7-26 Chapter 7 External Forced Convection Flow Across Cylinders And Spheres 7-35C For the laminar flow, the heat transfer coefficient will be the highest at the stagnation point which corresponds to $\theta \approx 0^{\circ}$. 3408) exp (1. Then the rate of heat loss for 1-cm thick insulation becomes T - T ∞ A (T - T ∞) (70.69 m 2)(75 -27)°C = Q& ins = s = o s = 11,445 W t ins 0.01 m 1 1 R total Rins + Rconv + + 0.038 W/m.°C 30 W/m 2.°C k ins ho Also, the total amount of heat loss from the furnace per year and the amount of temperature in the variation of temperature in the variatio pipe, and the rate of heat loss are to be determined for steady one-dimensional heat transfer. 5-44C For a medium in which the finite difference formulation of a general interior node is given in its simplest form as Thode = (Tleft + Ttop + Tright + Tbottom) / 4 : (a) Heat transfer is steady, (b) heat transfer is two-dimensional, (c) there is no heat generation in the medium, (d) the nodal spacing is constant, and (e) the thermal conductivity of the medium is constant. ° C)(1 m) Ro = Rconv, 2 = 1 1 1 = °C / W ho A3 (22 W / m 2 . k (2.22 W/m.°C) Bi wall = Bicyl = τ wall = τ cyl = hro (13 W/m 2 . °C)(0.01 m) = $0.05856 \rightarrow \lambda 1 = 0.3407$ and A1 = 10146. Assumptions 1 Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. The properties of air at this temperature are (Table A-15E) k = 0.01529 Btu/h.ft. A medium may involve two of them simultaneously. Properties The density and dynamic viscosity of water at 1 atm and 25°C are $\rho = 997$ kg/m3 and $\mu = 0.891 \times 10-3$ kg/m s (Table A-9). 2 There is no heat generation in the plate. 1-50 Chapter 1 Basics of Heat Transfer 1-92 A circuit board houses 80 closely spaced logic chips on one side, each dissipating 0.06 W. 4 The infiltrating air exfiltrates at the indoors temperature of 22°C. Properties The thermal conductivity and emissivity of cast iron are given to be k = 52 W/m °C and ϵ = 0.7 Analysis The individual resistances are Rpipe Ri Ro Ai = π Di L = π (0.04 m)(15 m) = 1885. The heat of fusion of ice at atmospheric pressure is 333.7 kJ/kg. Properties The density and specific heat of air at 0°C are given to be 1.28 kg/m3 and 1.0 kJ/kg. Properties The density and specific heat of fusion of ice at atmospheric pressure is 333.7 kJ/kg. above simultaneously with an equation solver to be T0 = 119.7° C, T1 = 118.6° C, T2 = 116.3° C, T3 = 114.3° C, T4 = 112.7° C, T5 = 111.2° C, T5 = 111.2° C, T6 = 119.9° C (c) Knowing the inner surface temperature, the rate of heat transfer from the flange section 5-30 Chapter 5 Numerical Methods in Heat Conduction Q& fin = $6 \sum m = 1 Q$ element, m = $6 \sum 6$ hAsurface, m (Tm - T ∞) + m = $1 \sum \epsilon \sigma A$ surface, m are as given above for different nodes. 6-6C Heat transfer through a fluid is conduction in the presence of it. 2 The surface temperature of the rods is constant. 3 Convection heat transfer coefficient is constant over the entire surface. 8-7C The friction factor is highest at the tube inlet where the thickness of the boundary layer is zero, and decreases gradually to the fully developed value. conductivity to be constant. The total heat transfer rate in this case can be calculated from $T \infty - T s (75 - 5)^{\circ}C O = = 1668 W = 1 (0.002 m) L 1 + + hAs kAs (58.97 W/m 2.^{\circ}C)[(0.6 m)(0.7 m)] (3 W/m, ^{\circ}C)(0.6 m \times 0.7 m)$ The decrease in the heat transfer rate is 1734-1668 = 66 W 7-84 Chapter 7 External Forced Convection 7-92E A minivan is traveling at 60 mph. 2-57 Chapter 2 Heat Conduction Equation 2-102 A plate with variable conductivity is subjected to specified temperatures on both sides. 1-40 Chapter 1 Basics of Heat Transfer 1-80 "GIVEN" D=0.2 "[m]" "L=0.4 [cm], parameter to be varied" T 1=0 "[C]" T 2=5 "[C]" "PROPERTIES" h if=333.7 "[k]/kg]" k=k ('Iron', 25) "[W/m-C]' "ANALYSIS" A=pi*D^2 Q dot cond=k*A*(T 2-T 1)/(L*Convert(cm, m)) m dot ice=(Q dot cond*Convert(W, kW))/h if L [cm] 0.2 0.4 0.6 0.8 1 1.2 L [cm] 1.41 1.4 1.6 1.8 2 mice [kg/s] 0.07574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01082 0.009468 0.008416 0.007574 0.03787 0.02525 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01262 0.01894 0.01515 0.01894 0.01515 0.01894 0.01515 0.01894 0.01515 0.01894 0.01894 0.01515 0.01894 0.01515 0.01894 0.01894 0.01894 0.01894 0.01894 0.01894 0.01894 0.01894 0.01894 0.01894 0.01894 0.01894 0.0189 1.6 1.8 2 Chapter 1 Basics of Heat Transfer 1-81E The inner and outer glasses of a double pane window with a 0.5-in air space are at specified temperatures. 3 Convection heat transfer coefficient is constant and uniform. ° C, $\rho = 8500 \text{ kg} / \text{m} 3$, and C p = 320 J / kg. The average outer surface temperature of the pipe, the fin efficiency, the rate of heat transfer from the flanges, and the equivalent pipe length of the flange for heat transfer are to be determined. Node 1 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 250 + T2 + T5 - 4T4 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 (interior): 250 + 272 - 4T1 = 0 Node 2 ($200^{\circ}C 250 5 \cdot 4 \cdot 300 200 3 \cdot 2 \cdot 250 T2 = T3 = 200^{\circ}C 150 1 \cdot 200$ Solving the 4 equations above simultaneously gives T1 = 175^{\circ}C 150 6 \cdot 300 Insulated (b) There is symmetry about a vertical line passing through the middle. 3-72 Insulation Chapter 3 Steady Heat Conduction 3-92 "GIVEN" D 1=0.005 "[m]" "t ins=1 [mm], parameter to be varied" k ins=0.13 "[W/m-C]" T ball=50 "[C]" T infinity=15 "[C]" h o=20 "[W/m^2-C]" "ANALYSIS" D 2=D 1+2*t ins*Convert(mm, m) A o=pi*D 2^2 R conv o=1/(h o*A o) R ins=(r 2-r 1)/(4*pi*r 1*r 2*k ins) r 1=D 1/2 r 2=D 2/2 R total=R conv o+R ins Q dot=(T ball-T infinity)/R total tins [mm] 0.5 1.526 2.553 3.579
4.605 5.632 6.658 7.684 8.711 9.737 10.76 11.79 12.82 13.84 14.87 15.89 16.92 17.95 18.97 20 Q [W] 0.07248 0.1035 0.1252 0.139 0.1474 0.1523 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1559 0.1 Chapter 3 Steady Heat Conduction Heat Transfer From Finned Surface 3-93C Increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surface by increasing the rate of heat transfer from a surf atmospheric air. The required power rating of the resistance heater is to be determined. 1-66 Chapter 1 Basics of Heat Transfer 1-122 The duct of an air heating system of a house passes through an unheated space in the attic. Analysis In steady operation, the rate of heat transfer from the wire is equal to the heat generated within the wire, Relastic Rconv Q& = W& e = VI = (8 V)(10 A) = 80 W T1 T ∞ 2 The total thermal resistance is 1 1 = 0.3316 °C/W 2 ho Ao (24 W/m. °C)[π (0.004 m)(10 m)] ln(r2 / r1 ln(2 / 1) = = 0.0735 °C/W 2 \pi kL 2 \pi(0.15 W/m. °C)(10 m) = Rconv + Rplastic = 0.3316 + 0.0735 = 0.4051 °C/W Rconv = Rplastic Rtotal Then the interface temperature becomes T - T ∞ 2 Q& = $1 \rightarrow T1 = T \propto + O\&R$ total = 30°C + (80 W)(0.4051 °C/W) = 62.4°C R total The critical radius of plastic insulation is k 0.15 W/m.°C = 0.00625 m = 6.25 mm rcr = = h 24 W/m 2.°C Doubling the thickness of the plastic cover will increase the outer radius of the wire to 3 mm, which is less than the critical radius of insulation. 2 Heat transfer is oneg(x) Radiation h, T ∞ dimensional since the plate is large relative to its k(T) Tsurr thickness. Analysis The one-dimensional transient temperature distribution in the wood can be determined from $(x T (x, t) - Ti = erfc|| T \infty - Ti (2 \alpha t ())| hx h 2\alpha t () || hx h 2\alpha t ()$ $s_{1}(5 \times 60 \ s) = 1.276 \ 0.17 \ W/m.^{\circ}C \ k \ 2 \ h \ 2\alpha t \ k \ 2 \ -7 \ Wood \ Slab \ Ti = 2.2 \ (h \ \alpha t \) = 1.276 \ 2 = 1.628 \ = |k|/V \ Hot \ gases \ T_{\infty} = L = 0.3 \ Noting \ that \ x = 0 \ at \ the \ surface \ and \ using \ Table \ 4-3 \ for \ erfc \ values, \ T(x, t) \ -25 \ = \ erfc(0) \ - \ exp(0 \ + \ 1.628)erfc(0 \ + \ 1.276 \) \ 550 \ -25 \ = \ 1 \ - \ (5.0937)(0.0727) \ = \ 0.630 \ 0 \ x \ Solving \ for \ T(x, t) \ = \ 356^{\circ}C$ which is less than the ignition temperature of 450°C. ° C = 3.41214 Btu / h = 01761. This is because the insulation or specified temperature boundary conditions have no effect on the stability criteria. A heat transfer system can involve both internal and external convection simultaneously. However, adding too many fins on a surface can suffocate the fluid and retard convection, and thus it may cause the overall heat transfer coefficient and heat transfer to decrease. In general, turbulence increases the drag coefficient for flat surfaces, but the drag coefficient and heat transfer to decrease. are determined to be Node 0 (x = 0): g& 0 = 500(0.859 - 3.415 × 0 + 6.704 × 0.2 - 6.339 × 0.3 + 2.278 × 0.4) = 429.5 W/m 3 Node1(x=.25): g& 1 = 500(0.859 - 3.415 × 0.25 + 6.704 × 0.25 2 - 6.339 × 0.25 3 + 2.278 × 0.25 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 - 3.415 × 0.5 + 6.704 × 0.5 2 - 6.339 × 0.5 3 + 2.278 × 0.5 4) = 167.1 W/m 3 Node 2 (x=0.50): g& 2 = 500(0.859 $88.8 \text{ W/m 3 Node3(x=.75): g\& 3 = 500(0.859 - 3.415 \times 0.75 + 6.704 \times 0.75 2 - 6.339 \times 0.75 3 + 2.278 \times 0.75 4) = 57.6 \text{ W/m 3 Node 4}$ (x = 1.00): g& 4 = 500(0.859 - 3.415 \times 1 + 6.704 \times 12 - 6.339 \times 13 + 2.278 \times 14) = 43.5 \text{ W/m 3 Also, the heat flux at the bottom surface is g& b = 0.379 \times 500 \text{ W/m 2} = 4189.5 \text{ W/m 2}. The upstream (or approach) velocity V is the velocity of the approaching fluid far ahead of the body. energies WATER – Qout = $\Delta U = \Delta U$ copper + ΔU iron + ΔU water or Iron – Qout = [mC (T2 – T1)]copper + [mC (T2 – T1)]copper $^{\circ}F)(T2 - 160)^{\circ}F + (50lbm)(0.107Btu/lbm \cdot ^{\circ}F)(T2 - 200)^{\circ}F + (180lbm)(1.0Btu/lbm \cdot ^{\circ}F)(T2 - 70)^{\circ}F$ T = 74.3 $^{\circ}F$ 1-16 600 kJ Chapter 1 Basics of Heat Transfer 1-38 A room is heated by an electrical resistance heater placed in a short duct in the room in 15 min while the room is losing heat to the outside, and a 200-W fan circulates the air steadily through the heater duct. 10• (b) The temperatures are determined by solving equations above to be T0 = 78.67°F, T1 = 78.62°F, T2 = 78.57°F, T3 = 78.51°F, T 0.02681 W/m.°C Nu 2 (168.0) 2 Nu = 0.664 Re L 0.5 Pr $1/3 \rightarrow$ Re L = $= 7.932 \times 10422/322/30.664$ Pr (0.664)
(0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = $= 7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v ($7.932 \times 10422/322/30.664$ Pr (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) (0.664) (0.7248) Nu = Re L = Re L v (0.664) (0.7248) (0.664) (0.7248) (0.7248) (0.664) (0.7248) (0.7248) (0.664) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248) (0.7248The dimensionless temperatures are θ o, wall (A) = 2 2 T 0 - T ∞ = A1e - λ 1 τ = (1.0038)e - (0.150) (1.1475) = 0.9782 Ti - T ∞ θ (L, t) wall (B) = θ o, wall (C) = 2 2 T (x, t) - T ∞ = A1e - λ 1 τ = (1.0076)e - (0.212) (0.2869) = 0.9947 Ti - T ∞ Then the center temperature of the top surface of the cylinder becomes $[T(L,0,0,t) - T\infty] = \theta(L,t)$ wall (B) × θ o, wall (C) = 0.9672 × 0.9782 × 0.9947 = 0.9411 \rightarrow T (L,0,0,t) = 142.2°C 150 - 17 (b) The corner of the block is at the surface of each plane wall. This is a closed system since the doors and the windows are said to be tightly closed, and thus no mass crosses the system boundary during the process. Therefore, at the outer surface, the temperature will be closer to the surrounding air temperature. x Analysis This cylindrical ice block can be treated as a short cylinder that can physically be formed by the intersection of a r long cylinder of diameter D = 2 cm and an infinite plane wall of Insulation thickness 2L = 4 cm. Air Properties of air at 25°C Cp = 1.007 kJ/kg-K, Pr = 0.7296 $\rho = 1.184$ kg/m3, 12 m/s Analysis First, we determine the rate of heat transfer from Q& = mC p,airfoil (T2 - T1) $\Delta t = (50 \text{ kg})$ $(500 \text{ J/kg} \cdot ^{\circ}\text{C})(160 - 150)^{\circ}\text{C} = 2083 \text{ W} (2 \times 60 \text{ s})$ Then the average heat transfer coefficient is O& 2083 W O& = hAs (Ts - T ∞) \rightarrow h = = = 1.335 W/m 2 $\cdot ^{\circ}\text{C}$ 2 As (Ts - T ∞) $(12 \text{ m})(155 - 25)^{\circ}\text{C}$ L=3 m where the surface temperature of airfoil is taken as its average temperature, which is $(150+160)/2=155^{\circ}\text{C}$. 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. For a maximum case temperature of 85°C, the maximum case temp to be Bi = hL (40 W/m 2.°C)(0.075 m) = 0.02727 k(110 W/m.°C) The constants λ 1 and A1 corresponding to this Biot number are, from Table 4-1, D0 = 8 cm Air T ∞ = λ 1 = 0.164 and A1 = 1.0050τ = L2 ($3.39 \times 10 - 5 \text{ m}^2$ / s)(15 min $\times 60 \text{ s}$ / min) = r L = 15 Brass cylinder The Fourier number is α t z (0.075 m) 2 Ti = 150°C = 5.424 > 0.2 Therefore, the one-term approximate solution (or the transient temperature charts) is applicable. One side of the plate is insulated while the other side is subjected to convection. For a cylindrical layer, it is defined as rcr = k / h where k is the thermal conductivity of insulation and h is the external convection heat transfer coefficient. energies We,in + Wfan,in -Qout = $\Delta U \& (We, in + W\&fan, in - Q\&out) \Delta t = m(u2 - u1) \cong mCv (T2 - T1) 200 kJ/min 5 \times 6 \times 8 m 3 = 240m 3 m = ()) P1V (98kPa) 240m 3 = = 284.6kg RT1 0.287 kPa \cdot m 3 / kg \cdot K (288K) (Then the power rating of the electric heater is determined to be = Q\& - W\& W\& + mC (T - T) / \Delta t$ e, in out fan, in v 2 W 200 W 1 = (200/60k]/s) - (0.2k]/s) + $(284.6kg)(0.720k]/kg \cdot ^{\circ}C)(25 - 15)^{\circ}C/(15 \times 60s) = 5.41 \text{ kW}$ (b) The temperature rise that the air experiences each time it passes through the heater is determined by applying the energy balance to the duct, E & = E & in out & 1 = Q & out E O + mh & 2 (since $\Delta ke \cong \Delta pe \cong O$) W & e, in + W& fan, in + mh & p ΔT W& e, in + W& fan, in = m& Δh = mC Thus, ΔT = W& e, in + W& fan, in m& C p = (5.41 + 0.2)kJ/s = 6.7°C (50/60 kg/s)(1.007 kJ/kg · K) 1-17 Chapter 1 Basics of Heat Transfer 1-39 The resistance heating element of an electrically heated house is placed in a duct. Assumptions 1 The wood slab is treated as a semi-infinite medium subjected to convection at the exposed surface. 5-106 Chapter 5 Numerical Methods in Heat Conductivity is subjected to combined convection and radiation at the right (node 3) and specified temperature at the left boundary (node 0). 1-59C Superinsulations are obtained by using layers of highly reflective sheets separated by glass fibers in an evacuated space. Oven Analysis (a) The Biot number is $Bi = hro(25 \text{ W}/m^2)$. Noting that u = u(y), v = 0, and $\partial P / \partial x = 0$ (flow is maintained by the motion of the upper plate rather than the pressure gradient), the xmomentum equation reduces to 20 cm ($\partial u d 2u \partial u$) ∂ $2 u \partial P \rightarrow = 0 + v || = \mu 2 - x$ -momentum: $\rho || u \partial y | \partial x dy 2 \partial y \langle \partial x This is a second-order ordinary differential equation, and integrating it twice gives u (y) = C1 y + C 2 The fluid velocities of the plates because of the no-slip condition. The geometry must be such that its entire surface can be$ described mathematically in a coordinate system by setting the variables equal to constants. The exposed surface of the heater is transferred to the pipe. F(0.1/12 ft) = 0.0025 Btu/h. The exposed surface area of the furnace is Ao = 2 Abase + Aside = $2\pi r 2 + 2\pi r L = 2\pi (1.5 \text{ m}) 2 + 2\pi r L$ 2π (1.5 m)(6 m) = 70.69 m 2 The rate of heat loss from the furnace before the insulation is installed is $O_{\&}$ = ho Ao (Ts - T ∞) = (30 W/m 2.°C)(70.69 m 2)(90 - 27)°C = 133.600 W Noting that the plant operates $52 \times 80 = 4160$ h/yr, the annual heat lost from the furnace is $O = O_{\&} \Delta t = (133.6 \text{ kJ} / \text{s})(4160 \times 3600 \text{ s} / \text{yr}) = 2.001 \times 109 \text{ kJ} / \text{yr}$ Rinsulation Ro Ts The efficiency of the furnace is given to be 78 percent. The refrigerated shipping docks are usually maintained at 1.5° C. kg / m3 RT (0.287 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π D2 / 4 = π (0.201 kPa.m3 / kg.K)(65 + 273)K AIR · Q The cross-sectional area of the duct is Ac = π AC + π AC m) 2/4 = 0.0314 m 2 Then the mass flow rate of air through the duct and the rate of heat loss become m& = $\rho Ac V = (1031 \cdot 2 \text{ Heat transfer is two-dimensional} (no change in the axial direction)$. This reduces the refrigeration load of the cold storage rooms. Thermal equilibrium is established after 25 min. ° C)(96 - 18)° C = 853 kJ The
1500 W electric heating unit will supply energy at a rate of 1.2 kW or 1.2 kJ per second. Metal ball Analysis The amount of energy added to the ball is simply the change in its internal energy, and is determined from Etransfer = $\Delta U = mC(T2 - T1)$ where m = $\rho V = \pi 6 \rho D3 = \pi 6 (2700 \text{ kg} / \text{m} 3)(015)$. Considering a unit depth and using the Insulated energy - T3 | T4 - T3 + k + k | 7 = 0 2 | 2 | 1 Node 4; ho | (T0 - T4) + k | T3 - T4 | T8 - T4 + k = 0 2 | 2 | 12 Node 5; hi (Ti - T5) + k | T6 - T5 | T1 - T5 + k = 0 2 | 2 | 1 Node 6; hi | (Ti - T6) + k T - T6 | T7 - T6 + k + k | 2 = 0 2 | 2 | 1 Node 7; hi | (Ti - T7) + k T - T7 T - T7 | T9 - T7 | T6 - T7 + k + k | 3 + k | 8 = 0 2 | 2 | 11 Node 8; ho | (T0 - T8) + k | 7 - T6 | 17 - T6 + k + k | 2 = 0 2 | 2 | 11 Node 7; hi | (Ti - T7) + k T - T7 T - T7 | T9 - T7 | T7 - T7 | T9 - T7 | T7 - T7 | T8 - T7 + k + k | 3 + k | 8 = 0 2 | 2 | 11 Node 8; ho | (T0 - T8) + k | 7 - T6 | 17 - T6 + k + k | 2 = 0 2 | 2 | 11 Node 7; hi | (Ti - T7) + k T - T7 T - T7 | T9 - T7 | T7 - T7 | T9 - T7 | T7 - T7 | T8 - T7 + k + k | 3 + k | 8 = 0 2 | 2 | 11 Node 8; ho | (T0 - T8) + k | 7 - T6 | 17 - T6 + k + k | 2 = 0 2 | 2 | 11 Node 7; hi | (Ti - T7) + k | 7 - T7 | T9 - T7 | T9 - T7 | T7 - T7 | T9 - T7 | T7 - T7 | T9 - T7 |The rate of heat conduction through the pipe is C T 2 – T ∞ dT Q& = – kA = – k (2 π rL) 1 = –2 π (15 ft)(7.2 Btu/h · ft · °F) = 16,800 Btu/h · ft · °F) = 16,800 Btu/h · ft · °F) = 16,800 Btu/h · ft · °F + ln 2 (12.5 Btu/h · ft · °F) = 16,800 Btu/h · on the inner surface and convection on the outer surface. The energy balance for this steady-flow system can be expressed in the rate form as $E\& - E\& = 0 \rightarrow E\&$ in = E& out $\Delta E\&$ system $\hat{E}0$ (steady) 1in424out 3 144 42444 3 Rate of net energy transfer by heat, work, and mass Rate of change in internal, kinetic, potential, etc. $\times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7 \text{ m } 2 / \text{s})(2 \text{ h} \times 10 - 7$ 3600 s/h (0.01 m) 2 = 0.2232 > 0.2 = 0.8928 > 0.2 Note that $\tau > 0.2$ in all dimensions and thus the one-term approximate solution for transfer for the uninsulated case is Tair = 8°C Ao = π Do L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 82°C Q& = hAo (Ts - Tair) = (25 W/m 2 .°C)(3.77 m 2 Ts = 8°C Ao = 100 L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = π Do L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = π Do L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = π Do L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = π Do L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = π Do L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m) = 3.77 m 2 Ts = 8°C Ao = \piDo L = π (0.12 m)(10 m)(10 m)(1

 $(82 - 8)^{\circ}C = 6974$ W Steam pipe The amount of heat loss during a 10-hour period is $Q = Q \& \Delta t = (6.974 \text{ kJ/s})(10 \times 3600 \text{ s}) = 2.511 \times 10.5 \text{ kJ}$ (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 $\rho = 0.6746$ kg/m3 Cp = 1.033 kJ/kg-K Pr = $0.6946 \mu = 2.76 \times 10$ 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 $\rho = 0.6158$ 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 of the orange are approximated by those of water at the average temperature of about 5° C, k = 0.571 W/m. C and $-62 \alpha = k / \rho C p = 0.571 / (1000 \times 4205) = 0.136 \times 10 \text{ m}$ /s (Table A-9). The exposed surface area of the furnace is Ao = 2 Abase + Aside = $2\pi 2 + 2\pi L = 2\pi (1.5 \text{ m})(6 \text{ m}) = 70.69 \text{ m} 2$ (he 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 $52 \times 80 = 4160 \text{ h/yr}$, the annual heat lost from the furnace is $Q = Q\& \Delta t = (101794 . Properties The average specific heats are given to be 0.6 kJ/kg.°C for the teapot and 4.18 kJ/kg.°C for te$ 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 $\varepsilon = 0.05$ 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. $\times 1 = 012$. 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: $V = m / \rho = 1700g/(0.95g/cm^3) = 1789cm^3 / 3$ ro = | V | $4\pi / 1/3 / 3$ = 1789 cm³ | $4\pi / 1/3 = 7.53$ 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 properties of air at 1 atm and the film temperature of T f = $(180 + 120)/2 = 150^{\circ}$ F are (Table A-15) k = 0.01646 Btu/h.ft.°F Power transistor D = 0.22 $\left[\left(\frac{727.5}{11} \right) \right] = 0.3 + 1 + \left[\left| \left| \left| \frac{1}{4} \right| \right| \left(\frac{282,000}{11} \right) \right] + \left(\frac{0.4}{0.7188} \right) 2/3 \text{ k } 0.01646 \text{ Btu/h.ft }^\circ \text{F and } h = \text{Nu} = (13.72) = 12.32 \text{ Btu/h.ft } 2 \cdot \text{F D} \left(\frac{0.22}{12} \text{ ft} \right) \text{Then the amount of power this transistor can dissipate safely becomes } Q_{\text{k}} = hAs \left(\text{T s} - \text{T}_{\infty} \right) = h(\pi \text{DL})(\text{Ts} - \text{T}_{\infty}) = 12.32 \text{ Btu/h.ft } 2 \cdot \text{F} \right) \left[\pi \left(\frac{0.22}{12} \text{ ft} \right) (0.25/12 \text{ ft}) (0.25/12 \text{ ft}) \right] (180 - 120)^\circ \text{C}$ = 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 = 120°C Ts = 1 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, $\rho = 4.1$ kJ/kg.°C, and $\alpha = 0.91 \times 10^{-7}$ m2/s Analysis (a) The radius of the rib is determined to be $m = \rho V \rightarrow V = V = m \rho 3.2$ kg = 1200 kg/m 3 = 0.00267 m 3 4 3V 3 $3(0.00267 \text{ m } 3) = = 0.08603 \text{ m} \text{ mro } 3 \rightarrow \text{ro} = 3 3 4\pi 4\pi \text{ R}$ ib The Fourier number is $\tau = \alpha \text{tro } 2 = (0.91 \times 10 - 7 \text{ m } 2/\text{s})(4 \times 3600 + 15 \times 60)\text{s}$ (0.08603 m) 2 4-33 = 0.1881 O v Chapter 4 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 guantities. ° C . 2 Heat transfer is onedimensional 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 • • • mirror-image concept, the finite difference equations are obtained to be as follows: $488 \cdot T2 - T11T0 - T11T5 - T1 Node 1$ (heat flux): q & 01 + k + k + kl = 021211 Insulated Symmetry Node 2 (interior): T0 + T2 + T4 + T7 - 4T3 = 0 T0 + 2T3 + T8 - 4T4 = 0 Node 5 (heat flux): q & 01 + k T - T51T1 - T5 + kl 6 + 0 = 0211 Node 6 (interior): T2 + T5 + T6 + T7 - 4T6 = 0 Node 7 (interior): T3 + T6 + T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 + 2T7 + T8 - 4T7 = 0 Node 8 (insulation): T4 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 is subjected to convection and radiation on both sides. to be determined for steady one-dimensional heat transfer. ° F Copper (kt). k (2.22 W/m.°C) at L2 at 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 become Vmax ST 0.015 = $V = (4 \text{ m/s}) = 8.571 \text{ m/s} (0.008 \text{ m}) = \max = 5294 \mu 1.705 \times 10 - 5$ $kg/m \cdot s$ 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(ST/SL) 0.2 (5294) 0.6 (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, 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 $\Delta P = N L f \chi 2 \rho V max (1.204 \text{ kg/m 3}) (6.552 \text{ m/s}) 2 = 8(0.22)(1) 2 2 (1N | 1 \text{ kg} \cdot \text{m/s} 2 \langle \rangle | = 45.5 \text{ Pa} | J \text{ Discussion}$ 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 \rightarrow T1 = T0 Insulated Δx Tsurr $\Delta x \in T0 - T1$ T2 - T1 Node 1 (at the interface): k A A + kB A = 0 • $\Delta x \Delta x$ 0• 2 1 4 Node 2 (at right boundary): $\varepsilon \sigma A$ (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 balance for this steady-flow system (node 5). change in internal, kinetic, potential, etc. All we need to do is replace the terms for ice by the ones for cold water + [MC (T2 - T1)] water = 0 [mC (T2 - T1)] water = 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 \& r 0 T (0) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2h T (r) = T \infty + r 0 + 4k 2$ $| = 290.8^{\circ}F 4 \times (8.6 \text{ Btu/h.ft.}^{\circ}F) 2 \times (820 \text{ Btu/h.ft.}^{\circ}F)$ cylinder becomes $\begin{bmatrix} T(L,0,t) - T \infty \end{bmatrix} = \theta(L,t)$ wall $\times \theta$ o, cyl = 0.857 $\times 0.577 = 0.494 \parallel$ short $\begin{bmatrix} Ti - T \infty \end{bmatrix}$ cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C 150 - 20 (c) We first need to determine the maximum heat can be transferred from the cylinder T(L,0,t) = 84.2°C error in heating and air-conditioning applications. For convenience, we choose the time step to be $\Delta t = 4$ s. That is, $(x (hx h 2\alpha t) | (x) h \alpha t) | (x) h \alpha t | (x) h \alpha t$ $Ti (2 \alpha t) (x | \rightarrow T (x, t) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) | | / Then the temperatures at 0, 10, 20, and 50 cm depth from the ground surface become x = 0; (0 T (0,10 h) = Ti + (T_{\infty} - Ti) erfc | | / (2 \alpha t) |$ $10 \text{ m/s}(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \text{ erfc}(0.066) = 10 - 20 \times 0.9257 = -8.5^{\circ}\text{C} \text{ x} = 0.2 \text{ m} (\ 0.2 \text{ m} \ | \text{ T} (0.2 \text{ m}, 10 \text{ h}) = 10 + (-10 - 10) \text{ erfc} | -52 \ | 2 (1.6 \times 10 \text{ m/s})(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \text{ erfc}(0.132) = 10 - 20 \times 0.8519 = -7.0^{\circ}\text{C} \text{ x} = 0.5 \text{ m} (\ 0.5 \text{ m} \ | \text{ T} (0.5 \text{ m}, 10 \text{ h}) = 10 + (-10 - 10) \text{ erfc} | -52 \ | 2 (1.6 \times 10 \text{ m/s})(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \text{ erfc}(0.132) = 10 - 20 \times 0.8519 = -7.0^{\circ}\text{C} \text{ x} = 0.5 \text{ m} (\ 0.5 \text{ m} \ | \text{ T} (0.5 \text{ m}, 10 \text{ h}) = 10 + (-10 - 10) \text{ erfc} | -52 \ | 2 (1.6 \times 10 \text{ m/s})(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \text{ erfc}(0.132) = 10 - 20 \times 0.8519 = -7.0^{\circ}\text{C} \text{ x} = 0.5 \text{ m} (\ 0.5 \text{ m/s})(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \text{ erfc}(0.132) = 10 - 20 \times 0.8519 = -7.0^{\circ}\text{C} \text{ x} = 0.5 \text{ m} (\ 0.5 \text{ m/s})(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \text{ erfc}(0.132) = 10 - 20 \times 0.8519 = -7.0^{\circ}\text{C} \text{ x} = 0.5 \text{ m} (\ 0.5 \text{ m/s})(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \text{ erfc}(0.132) = 10 - 20 \times 0.8519 = -7.0^{\circ}\text{C} \text{ x} = 0.5 \text{ m} (\ 0.5 \text{ m/s})(10 \text{ h} \times 3600 \text{ s/h}) || \langle \rangle = 10 - 20 \times 0.8519 \text{ erfc}(0.132) \text{ erfc}(0.132) = 10 - 20 \times 0.8519 \text{ erfc}(0.$ s/h) ||| = 10 - 20 erfc(0.329) = $10 - 20 \times 0.6418 = -2.8^{\circ}C 4-58$ Chapter 4 Transient Heat Conduction 4-65 "!PROBLEM 4-65" "GIVEN" T i=10 "[C]" T infinity=-10 "[C]" T infi T i) = erfc(x/(2*sqrt(alpha*time)))exp((h*x)/k+(h^2*alpha*time))/k^2)*erfc(x/(2*sqrt(alpha*time))/k)) x [m] 0 0.05 0.1 0.15 0.2 0.25 0.3 0.45 0.5 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 = 0 (since g& = 0), for m = 0, 1, 2, and $3 \& \Delta x 2$ 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): $h(T\infty - T4) + kg T0 T3 - T4 = 0$ Node 4 (right surface - convection): $h(T\infty - T4) + kg T0 T3 - T4 = 0$ Node 4 (right surface - convection): $h(T\infty - T4) + kg T0 T3 - T4 = 0$ Node 4 (right surface - convection): $h(T\infty - T4) + kg T0 T3 - T4 = 0$ Node 4 (right surface - convection): $h(T\infty - T4) + kg T0 T3 - T4 = 0$ Node 4 (right surface - convection): $h(T\infty - T4) + kg T0 T3 - T4 = 0$ Node 5 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2T3 + T4 = 0 Node 7 (interior): T2 - 2TChapter 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 1 1 T2i - T1i 1 Z1 $\tau 2 2 + + + + 3 \infty 6 | 4 k / k 2 k / | | / Ti - T4i | Ti - Ti | 140 - T4i | 2 | 2 Ti + 1 - T4i | Node 4: q & L| + k + k| 5 + g & 0 = p C 4 2 | 2 | 1 (2 Node 5 (interior): T5i + 1 = (1 - 4\tau)T5i + \tau | | T2i + T4i + T6i + 140 + (2 g & 0 | k 2 \Delta t) | | / Node 6: hl (T \scale - T6i | + k i i i i i + 1 i 140 - T6i | T3 - T6 | T7 - T6 3 | 2 3 | 2 T6 - T6 + k | 5 + k | 5 + k | 5 + k | 5 + q & 0 = p C 4 2 | 2 | 1 (2 Node 5 (interior): T5i + 1 = (1 - 4\tau)T5i + \tau | | T2i + T4i + T6i + 140 + (2 g & 0 | k 2 \Delta t) | | / Node 6: hl (T \scale - T6i | T3 - T6i | T3 - T6i | T7 - T6 3 | 2 3 | 2 T6 - T6 + k | 5 + k | 5 + k | 5 + k | 5 + q & 0 = p C 4 2 | 2 | 1 (2 Node 5 (interior): T5i + 1 = (1 - 4\tau)T5i + \tau | | T2i + T4i + T6i + 140 + (2 g & 0 | k 2 \Delta t) | | / Node 6: hl (T \scale - T6i | + k + k | 5 + q & 0 = p C 4 2 | 2 | 1 (2 Node 5 (interior): T5i + 1 = (1 - 4\tau)T5i + \tau | | T2i + T4i + T6i + 140 + (2 g & 0 | k 2 \Delta t) | | / Node 6: hl (T \scale - T6i | + k + k | 5 + q & 0 = p C 4 2 | 2 | 1 (2 Node 5 (interior): T5i + 1 = (1 - 4\tau)T5i + \tau | | T2i + T4i + T6i + 140 + (2 g & 0 | k 2 \Delta t) | | / Ti - T6i | T3 - T6i | T3$ $+k + g\&0 = \rho C 2 1 1 2 1 4 4 \Delta t Node 7: hl (T \infty - T7i) + k 1 2 i i i i i + 1 i 170 - T7i 176 - T7 1 T8 - T7 1 2 1 2 T - T8 + k + g\&0 = \rho C 8 2 1 2 1 4 4 \Delta t 5-83$ Chapter 5 Numerical Methods in Heat Conduction where $g\&0 = 2 \times 107$ W/m 3, $q\&L = 2 \times$ 8000 W/m 2, l = 0.015 m, $k = 15 \text{ W/m} \cdot \text{C}$, $h = 80 \text{ W/m} 2 \cdot \text{C}$, and $T \infty = 25 \circ \text{C}$. The convection for bodies subjected to the winds is natural convection for bodies subjected to the winds is natural convection for the earth, but it is forced convection for bodies subjected to the winds is natural convection 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 $Q = Q \& \Delta t = (0.6 \text{ W})(24 \text{ h}) = 14.4 \text{ Wh} = 51.84 \text{ kJ}$ (since 1 Wh = 3600 Ws = 3.6 kJ) Q & (b) The heat flux on the surface of the resistor is $As = 2 q \& s = \pi D 2 4 + \pi DL = 2 \pi (0.4 \text{ cm})(2.4 \text{ m}) = 0.251 + 1.885$ = 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 $\rho = 8530 \text{ kg} / \text{m}^3$, $C p = 0.389 \text{ kJ/kg} \cdot ^\circ \text{C}$, $k = 110 \text{ W/m} \cdot ^\circ \text{C}$, and $\alpha = 3.^\circ \text{C} [\pi (0.03)(15)] \text{m}^2 \text{ T} \infty 2 \text$ Rconv,i + Rpipe + Rcombined, o R conv,i + Rpipe + hcombined Ao Substituting, $(68.5 - 15)^{\circ}$ C 13,296 W = $(0.000031 \circ C / W) + (0.001768 \circ C / W) + (0.0$ 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 = Δt where ΔE element = $Et + \Delta t - Et = mC$ (Tt + $\Delta t - Tt$) = ρCAΔr (Tt + Δt - Tt) G& element = 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, $\alpha = 0.44 \times 10 - 6 \text{ m } 2/\text{s}$, and $\kappa = 0.76$. $\tau = \alpha \text{tro } 2 \theta \text{ o}$, cyl = = (3.39 × 10 - 5 \text{ m } 2/\text{s})(15 × 60 \text{ s})(0.04 \text{ m}) 2 = 19.069 > 0.2 2 2 \text{ T0} - T\infty = A1e - \lambda 1 \tau = (10038 \text{ . The maximum amount of heat transfer is Furnace m} = \rho V = \rho \pi \text{o} \text{ L} = (2702 \text{ kg/m } 3) \pi (0.075 \text{ m}) 2. Also, Cp = 1.007 kJ/kg·K for air at room temperature (Table A-15). Properties The thermal conductivity of iron is $k = 80.2 \text{ W/m} \cdot \text{C}$ (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.

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