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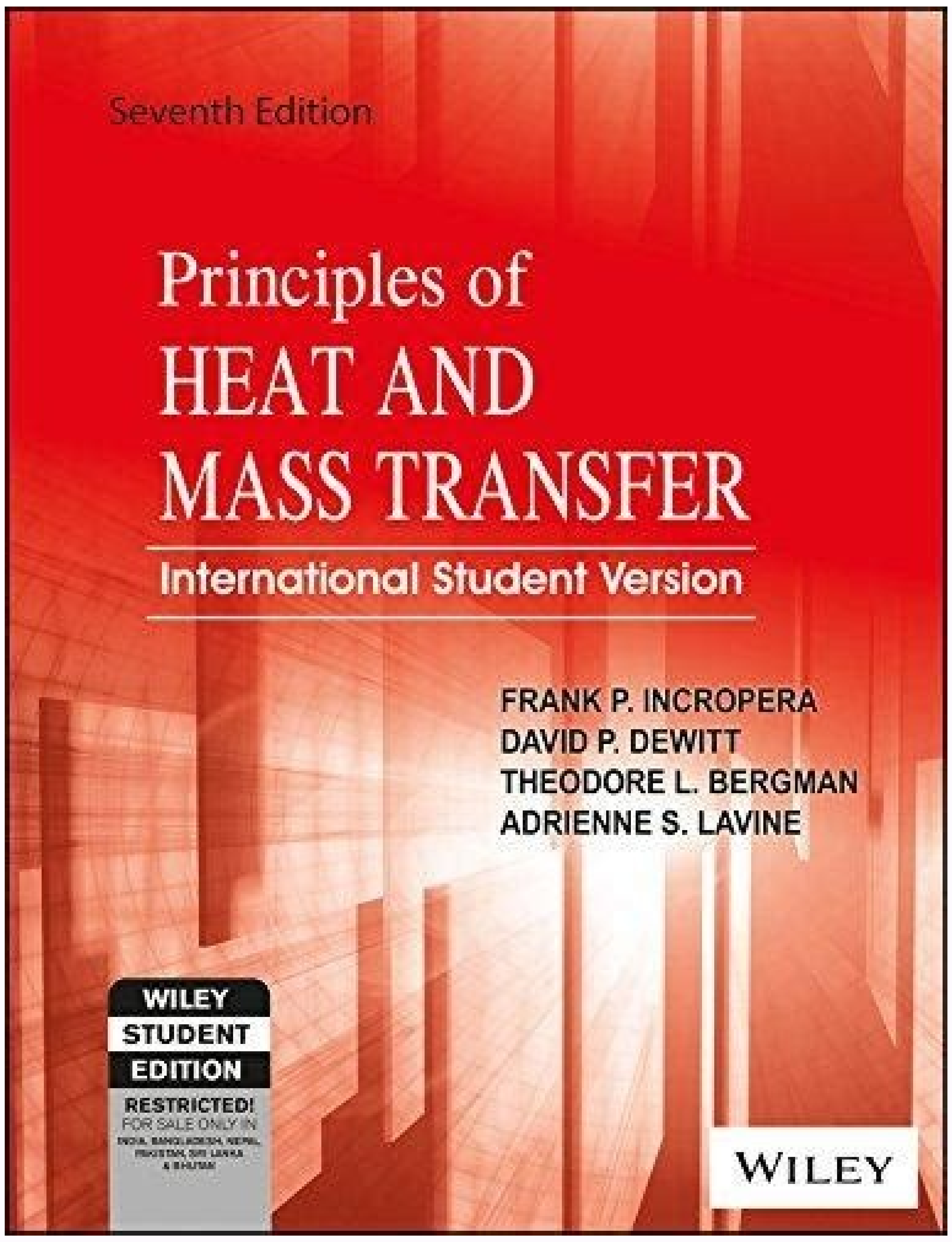
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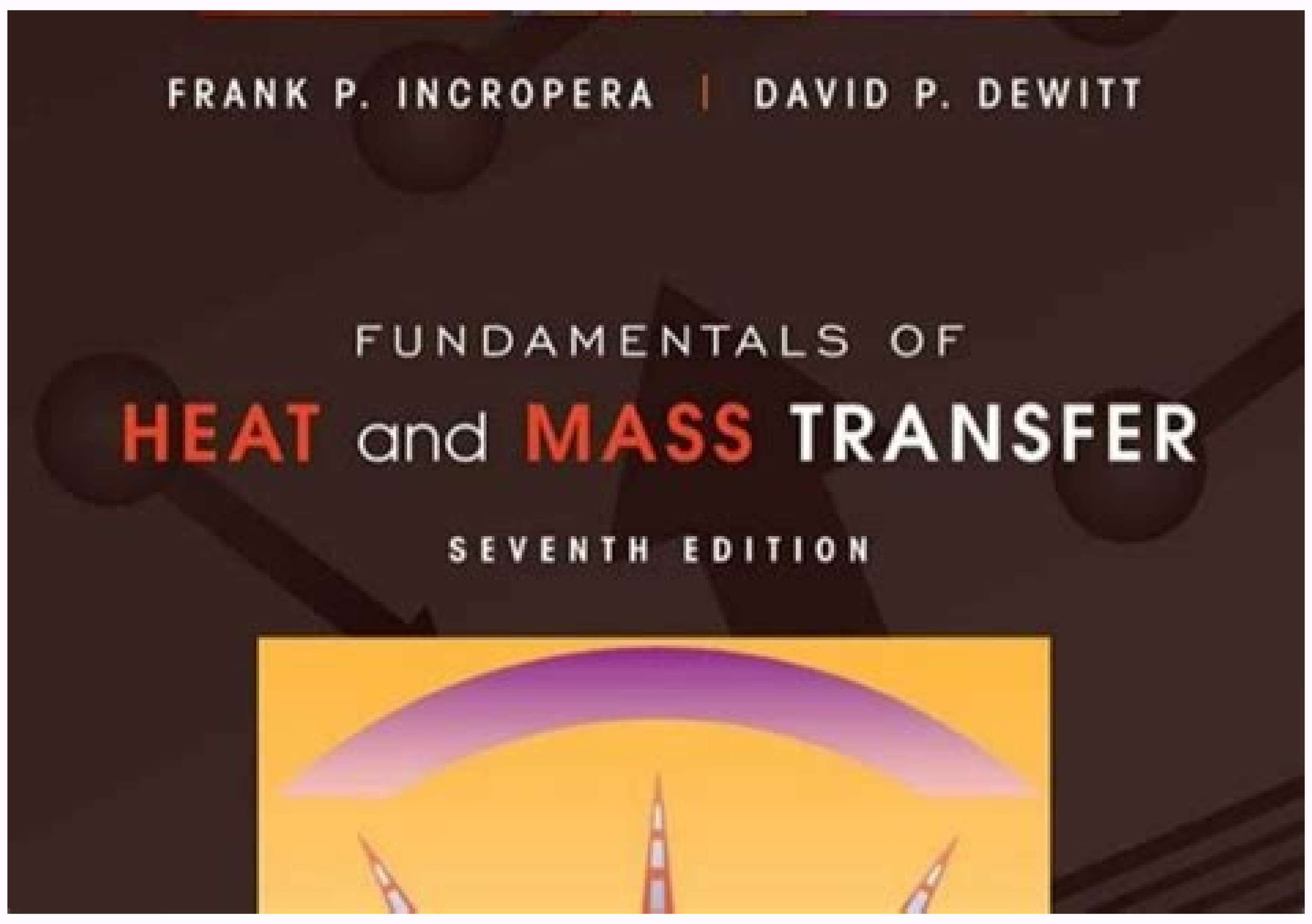
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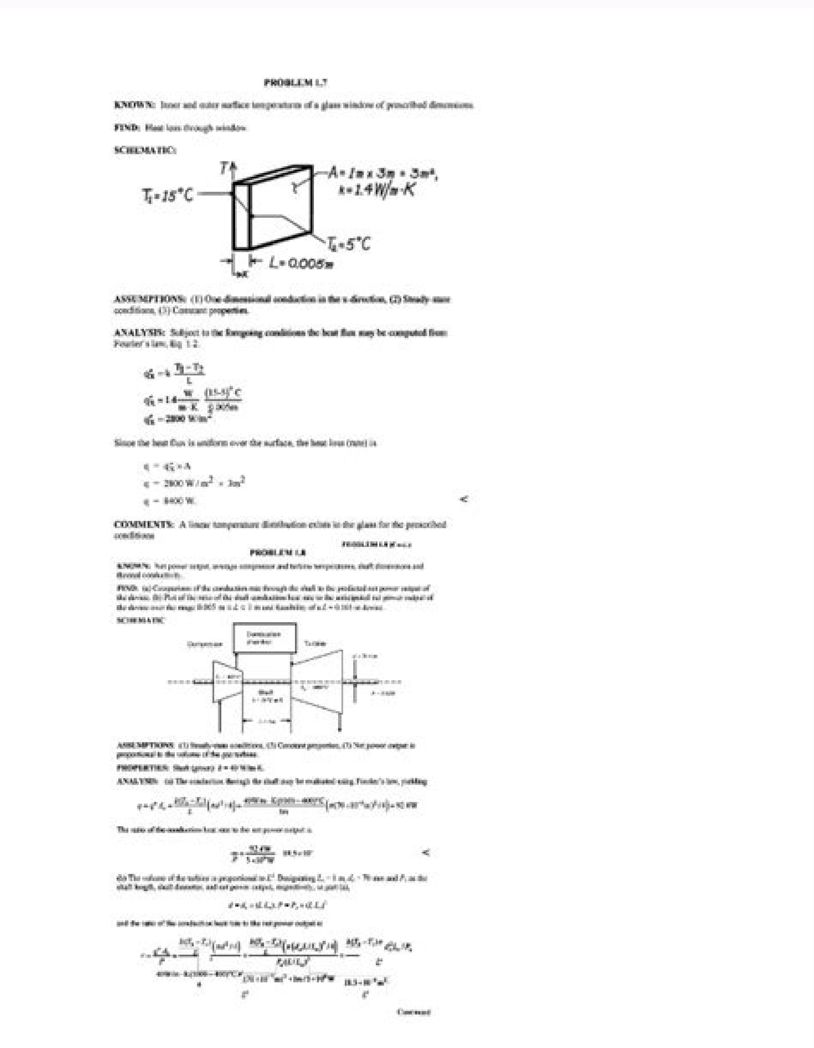

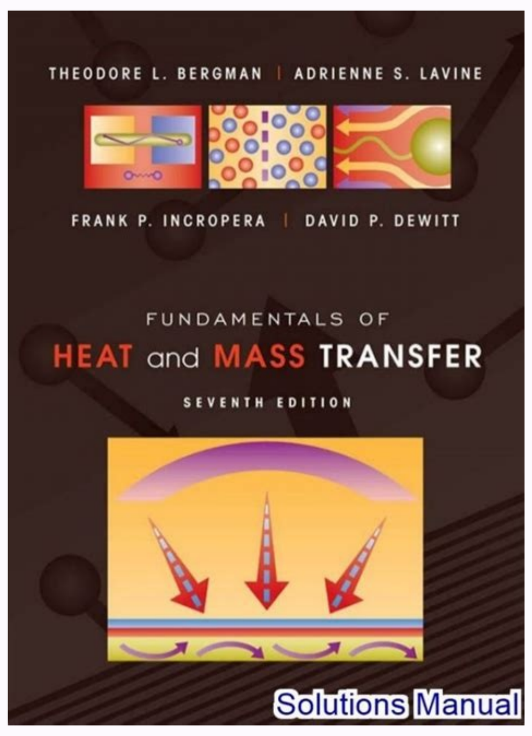
THEODORE L. BERGMAN | ADRIENNE S. LAVINE
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FUNDAMENTALS OF
HEAT and MASS TRANSFER
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Solutions Manual

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This manual may not be sold, distributed or used by students or other third parties. No part of this manual may be reproduced, displayed, or distributed in any form or by any means, electronic or otherwise, without the prior written consent of McGraw-Hill Education. Course Hero uses AI to try to automatically extract content from you and other documents displayed to help you learn, such as B. in search results to enrich documents and more. This preview shows 1-12. p. of 12 pages. The requested URL was not found on this server. Additionally, a 404 Not Found error was encountered when attempting to use ErrorDocument to handle the request. Apache/2.4.41 (Ubuntu) server on e2shi.jhu.edu port 443 Want more? Advanced embedding information, examples and help! You are reading a free preview. Pages 6 to 11 are not shown in this preview. KNOWN PROBLEM 1.2: Wall thickness and thermal conductivity. heat flowon one side and the temperature of both surfaces. FIND: whether stationary conditions exist. SCHEME: ASSUMPTIONS: (1) One-dimensional conductivity, (2) Constant properties, (3) No internal energy generation. ANALYSIS: Under steady-state conditions, the energy balance of the specified control volume is 2inoutcond12 ()/12 W/mK(50C30C)/0.01 m24,000 W/m q q q k T T L â²â² â²â² â²â ²= = = = Â° Â Heat flow to the left side only 20 W/m 2, conditions are not constant. < NOTES: If the same heat flux is maintained until steady-state conditions are reached, the steady-state temperature difference across the wall is Î TÂ Â = 2/20 W/m0.01 m/12 W/mK0. 0167 KÂ q L kÂ â²â²Â \u003d Â â \u003d Â which is much less than the specified temperature difference of 20 °C. q â= 20 W/m 2 L = 10 mm k = 12 W/m K T 1 = 50 °C T 2 = 30 °C q ≈ conditional PROBLEM 1.3 KNOWN: Internal surface temperature and thermal conductivity of a concrete wall. IDENTIFICATION: Heat losses by conduction through the wall as a function of the temperature of the external surface ranging from -15 to 38 °C. SCHEME: ASSUMPTIONS: (1) One-dimensional thermal conductivity in x-direction, (2) steady-state conditions, (3) constant properties. ANALYSIS: From Fourier's law, if x qÂ â²â² and k are constant, it is clear that the gradient x dTdxqkÂ â²â²= is constant and therefore the temperature distribution is linear. The heat flow must be constant under one-dimensional steady-state conditions; and k is approximately constant if it depends weakly on temperature. Heat flow and heat emission at the outside wall temperature T 2 = -15 °C are = â = o o . (1) 22xx qqA133.3Wm20m2667W â²â²= Ã= Ã= . (2) < combination of Eq. According to equations (1) and (2), the heat release rate qB x B can be determined for the temperature range of the outer surface -15 ÷ T 2 ÷ 38 °C with different thermal properties of the wall.k. For a concrete wall, the heat loss k = 1 W / m K varies linearly from +2667 W to -867 W and is equal to zero at the same internal and external surface temperatures. The size of the heat flux increases with the increase in thermal conductivity. NOTES: Without stationary conditions and constant k, the temperature distribution on a flat wall would not be linear. -20-10010203040 ambient temperature, T2 (C) -1500-500500150025003500 € € Thermal conductivity of the wall, k = 1.25 W/m.Kk = 1 W/m.K, concrete wall = 0.75 W/m.K External side surface -20 -10010203040 Ambient temperature, T2(C) -1500-500505013-50050501 Height aa, concrete wall = 0.75 W/m.K Out 1. PROBLEM 1.1. KNOWN: Thermal conductivity, thickness and temperature difference in a sheet of rigid extruded insulation. FIND: (a) the heat flux through the 2 m × 2 m insulation sheet and (b) the rate of heat transfer through the sheet. SCHEME: ASSUMPTIONS: (1) one-dimensional conductivity in the x direction, (2) stationary conditions, (3) constant properties. ANALYSIS: from 1.2. the heat flux of the equation is 1 2 x T - TdT q = -k = k dx L â²â² Solution, " x W 10 K q = 0.029 Ã m K 0.02 m² x 2 W q = 14.5 m â²â² < The heat capacity is 2 x x 2 W q = q A = 14.5 x 4 m = 58 W m â²â² â < NOTES: (1) Remember the important difference between heat flux (W/m2) and heat output (W). (2) Direction heat flow from hot to cold (3) Note that the temperature difference can be expressed in Kelvin or Celsius = 10 °C qcond A = 4 m2 T2T1 k = 0.029 × W m H x D = 20 mm T1 T2 = 10° C Fundamentals Heat and Mass Transfer Issue 7 Incroper Solutions Guide Full load:This is only an example. Download all chapters at: testbankreal.com 2. 1.2. KNOWN PROBLEM: Wall thickness and thermal conductivity. Heat flux applied to one surface and temperature of both surfaces. FIND: Whether stationary conditions exist. SCHEME: ASSUMPTIONS: (1) one-dimensional conductivity, (2) constant properties, (3) no internal energy generation. ANALYSIS: In steady state, the energy balance of the given control volume 2 in out cond 1 2( )/ 12 W/m K(50 C 30 C)/0.01 m 24,000 W/mq q q k T T Lâ²â² â²â² â²â²= = â = â Â° â Â° = Since the heat flux on the left is only 20 W/m2, the conditions are not uniform. < REMARKS: If the same heat flux is maintained until steady state conditions are reached, then the steady state temperature difference across the wall is ÎT = 2 / 20 W/m 0.01 m/12 W/m K 0.0167 Kq l kâ²â² = Ã â = which is much smaller than the specified temperature difference of 20°C. q' = 20 W/m2 L = 10 mm k = 12 W/m KT1 = 50°C T2 = 30°C q'¹cond 3. KNOWN PROBLEM 1.3: Temperature and thermal conductivity of the inner surface of a concrete wall. INVESTIGATION: Heat loss by conduction through the wall as a function of the outside surface temperature in the range of -15 to 38°C. DIAGRAM: ASSUMPTIONS: (1) one-dimensional conductivity in x-direction, (2) stationary conditions, (3) constant properties. ANALYSIS: If xqâ²â² and k are constant, it is clear from Fourier's law that the gradient xdT dx q kâ²â²= is a constant and therefore the temperature distribution is linear. The heat flow must be constant in the one-dimensional steady state; and k is approximately constant if it depends weakly on temperature. Heat flow and heat return at outside wall temperature T2 = -15°C is ( ) 21 2 x 25 CdT T T q k k 1 W m K 133.3 W m dx L 0.30 m â ââ â ²â² = â = = â = o o . (1) 2 2 x x q q A 133.3 W m 20 m 2667 W²²²= Ã = Ã = . (2) < union of Eq. (1) and (2), the heating rate qx varintended for the temperature range of the external surface, -15 ÷ T2 ÷ 38°C, with different thermal conductivity of the wall, k. For a concrete wall, k = 1 W/m²K, heat losses change linearly from + 2667 W to -867 W and is zero when the temperature of the inner and outer surfaces are the same. The magnitude of the heat velocity increases with increasing thermal conductivity. NOTE: Without stationary conditions and constant k, the temperature distribution in a flat wall would not be linear. -20 -10 0 10 20 30 40 Ambient temperature, T2 (C) -1500 -500 500 1500 2500 3500 Heat loss, qx(W) Thermal conductivity of the wall, k = 1.25 W/m.K k = 1 W/m.K , wall concrete k = 0.75 W/mK External surface -20 -10 0 10 20 30 40 Ambient temperature, T2 (C) -1500 -500 500 1500 2500 3500 Heat loss, qx(W) Wall thermal conductivity, k = 1 .25 W/m.K k = 1 W/m.K, concrete wall k = 0.75 W/m.K external surface 4. PROBLEM 1.4 KNOWN: Dimensions, thermal conductivity and surface temperatures of the concrete slab. Gas furnace efficiency and natural gas costs. FIND: Daily Cost of Heat Loss. SCHEME: ASSUMPTIONS: (1) steady state, (2) one-dimensional line, (3) constant properties. ANALYSIS: The rate of heat loss by conduction through the slab is () () 1 2 T T 7 C q k LW 1.4 W/m K 11 m 8 m 4312 W t 0.20 m â Â° = = â Ã= < Daily cost the mass of gas that must be burned to compensate for heat loss is ( ) ( )g d 6 f qC 4312W USD 0.02/MJ Ct 24 h/d 3600 s/h USD 8.28/d 0.9 10 J / MJÎ· Ã= Î = Ã= Ã < NOTES: Losses can be reduced by installing a floor covering with a layer of insulation between it and the concrete. 5. PROBLEM 1.5 KNOWN: Thermal conductivity and wall thickness. heat flow through the wall. steady state conditions. FIND: Temperature gradient value in K/m in °C/m. SCHEME: L = 20 mm k = 2.3 W/m K qâx = 10 W/m2 x ASSUMPTIONS: (1) One-dimensional line, (2) Constant properties. ANALYSIS: Under stationary conditions " 2 10 W/m 4.35 K/m 4.35 C/m 2.3K xdT q dx k = â = â = â = â Â° â < Since the units of K here represent a temperature difference, and the temperature difference is the same in units of K and °C, the value of the temperature gradient is the same in both units. NOTES: A negative temperature gradient means that the temperature decreases as x increases, which corresponds to a positive heat flow in the x direction. 6. 1.6. KNOWN PROBLEM: Heat flow and surface temperature related to a wooden board of a certain thickness. FIND: Thermal conductivity, k, wood. CHART: ASSUMPTIONS: (1) one-dimensional conductivity in the x direction, (2) stationary conditions, (3) constant properties. ANALYSIS: Using the above assumptions, thermal conductivity can be determined from the equation of Fourier's law. 1.2. Permutation, ( ) L W 0.05 m k=q 40 T T m 40-20 C x 21 2 â²â² = â o k = 0.10 W/m Kâ < NOTES: Please note that temperature units are °C in temperature calculations or K can be used interchangeably for differences. 7. PROBLEM 1.7. KNOWN: The temperature of the inner and outer surface of window panes with given dimensions. DEFINITION: Heat loss through a window. CHART: ASSUMPTIONS: (1) one-dimensional conductivity in the x direction, (2) stationary conditions, (3) constant properties. ANALYSIS. Under the above conditions, the heat flow can be calculated according to Fourier's law, equation 1.2. ( ) T T q k L 15-5 CW q 1.4 m K 0.005 m q 2800 W/m. 1 2 x x 2 x â²â² = â²â² = â â²â² = o Since the heat flow over the surface is uniform, the heat loss (coefficient) is q = qx A q = 2800 W/m2 3 m2 ≈ q = 8400 W. < NOTES: In the data under these conditions, the glass has a linear temperature distribution. 8. PROBLEM 1.8. KNOWN: net power, average compressor and turbine temperatures, shaft dimensions and thermal conductivity. FIND: (a) Comparison of the shaft conductivity with the expected net power output of the device, (b) Shaft conductance coefficient graphratio to the expected useful power of the device in the range 0.005 m ≤ L ≤ 1 m and the probability of the device L = 0.005 m. turbines. PROPERTIES: Shaft (data): k = 40 W/mâK ANALYSIS: (a) The conductivity of the shaft can be estimated using Fourier's law, which gives ( ) ( )2 3 2( ) 40 W/m K (1000 400) C " / 4 (70 10 m) /4 92.4W 1m h c c k T T q q A d L Ï Ï ââ â â Â° = = = Ã = The ratio of heat conduction speed to useful power is 6 6 92.4 W 18.5 10 5 10 W q r P = = = Ã Ã < (b) The turbine volume is proportional to L3, which is La = 1 m , da = 70 mm and Pa as the shaft length, shaft diameter and useful power in part (a), d = da Ã (L /La) P = Pa Ã (L/La)3, and the ratio of heat conduction rate to useful power is ( ) ( )( )22 2 3 2 3 2 6 6 2 2 2 ( ) ( ) ( ) / 4 / / 4 / " 4 (/) 40W/m K(1000 400) C (70 10 m) 1m/5 10 W 18.5 10 m4= h c h c a a a c a a k T T k T T k T d d L L d L P q A Lrâ Pâ P P P L = = = = â â Â° Ã Ã Ã Ã = Continued L = 1 m d = 70 mm P = 5 MW Turbine Compressor shaft Th = 1000Â Tc = 400 °C Combustion chamber L = 1 m d = 70 mm P = 5 MW Turbine Compressor shaft Th = 1000 °C Tc = 400 °C Combustion chamber k = 40 W/m·K PROBLEM 9 1.8 (Step by Step) The relationship between shaft ratio and useful power is shown below. For L = 0.005 m = 5 mm, the shaft control to useful power ratio is 0.74. The concept of a very small turbine is not feasible because it is unlikely to be able to withstand a large temperature difference between the compressor and the turbine. < NOTES: (1) The thermodynamic analysis does not take into account the effects of heat transfer and therefore only makes sense if heat transfer can be safely neglected, as in the case of the shaft in part (a). (2) Successful miniaturizationDevices are often limited by heat transfer effects that must be overcome through innovative design. Waveguide power to net power ratio 0 0.2 0.4 0.6 0.8 1 L (m) 0.0001 0.001 0.01 0.1 1 r 10. KNOWN PROBLEM 1.9: Width, height, thickness and thermal conductivity single glass and the air gap of double-glazed windows. Representative winter glass surface and airspace temperatures. FIND: Heat loss through single and double glazed windows. DIAGRAM: ASSUMPTIONS: (1) One dimensional conductivity through glass or air, (2) steady state, (3) air trapped in a double window is stagnant (negligible buoyant movement). ANALYSIS: According to Fourier's law, heat losses result from a single panel: ( )T T 35 C21 2q k A 1.4 W/m K 2m 19600 Wg g L 0.005m â = â â = o < double panel: ( )T T 25 C21 2q k A 0.024 2 m 120 Wa a L 0.010 m π = = = o < NOTES: Single pane losses are unacceptable and will remain excessive even when the glass thickness is doubled to accommodate the air space. The main advantage of the double-glazed construction is the low thermal conductivity of the air (~60x lower than in the case of glass). With a constant outside air temperature, the use of a double-glazed design would also increase the temperature of the glass surface exposed to the indoor (inside) air. 11. PROBLEM 1.10 KNOWN: Freezer compartment dimensions. Inner and outer surface temperatures. FIND: The thickness of polystyrene insulation required to keep the heat load below a certain value. SCHEME: ASSUMPTIONS: (1) Perfectly insulated ground, (2) One-dimensional conduction through 5 walls with an area of A = 4m 2 , (3) Stationary conditions, (4) Constant properties. ANALYSIS: Using Fourier's law, the equation. 1.2, the rate of heating is q = q A = k T L Totalâ²â²â Î Solving for L and finding that Total = 5ÂW 2, find L = 5 k T W q 2 Î ( ) ( )5 0.03 W/m K 35 - - 10C 4m L = 500W 2â¡ â¤Ã â â€ â€¦ about L = 0.054m = 54mm. < COMMENTS:will lead to a local deviation from one-dimensional thermal conductivity and somewhat large heat losses. 12. 1.11. KNOWN PROBLEM: Heat flux on one flat wall surface and air temperature and convection coefficient on the other side. The temperature of the surface exposed to convection. FIND: If stable conditions exist. If not, then the temperature is rising or falling. SCHEMATIC: ASSUMPTIONS: (1) One-dimensional conductivity, (2) No generation of internal energy. ANALYSIS: The law of conservation of energy for the control volume around the wall gives st v dE E E E E dt = â +& & & [ ]st v 2 2 2 ( ) ( ) 20 W/m 20 W/m K(50 C 30 C) 380 W /m s dE q A hA T T q h T T A dt A A â â â²â² â²â²= â â = â = â â Â° â Â° = ââ¡ â¤â£ â¦ Since dEst/d , the system is not in equilibrium. < Since dEst/dt < 0, the stored energy decreases, so the wall temperature also decreases. < COMMENTS: When the convective surface temperature drops to 31°C, qin = qout and dEst/dt = 0 and the wall reaches stationary conditions. qv = 20 W/m2 Ts = 50 °C h = 20 W/m2 K Tv = 30 °C Air qconv 13. PROBLEM 1.12. KNOWN: Dimensions of food/beverage containers and their thermal conductivity. Internal and external surface temperature. WE FIND: The heat flow through the container wall and the total heat load. SCHEMATIC: ASSUMPTIONS: (1) stationary conditions, (2) negligible heat transfer through the bottom wall, (3) uniform surface temperature and one-dimensional conduction through the remaining walls. ANALYSIS: From Fourier's law equation. 1.2 is the heat flow ( )0.023 W/m K 20 2 CT T 22 1q k 16.6 W/m L 0.025 m â ââ â²â² = = = o heat load ( )q q A q H 2W 2W W Wtot 1 â² 1 2 â²â²â¡ â= Ã = + + Ãâ£ â¦ ( ) ( )2q 16.6 W/m 0.6 m 1.6 m 1.2 m 0.8 m 0.6 m 35.9 m 35 .9 m Container corner: + + Ã = â £ â¦ and edges cause local deviationsconductivity, which increases the heat load. However, the influence of H, W1, W2 >> L is negligible. 14. 1.13. KNOWN PROBLEM: A masonry wall of known thermal conductivity has a thermal index equal to 80% of the heat transfer coefficient of a composite wall of recommended thermal conductivity and thickness. FIND: wall thickness. SCHEMATIC: ASSUMPTIONS: (1) both walls exposed to the same surface temperature, (2) one-dimensional conductivity, (3) steady-state conditions, (4) constant properties. ANALYSIS: Under steady conditions, the flow of heat conduction through a one-dimensional wall is given by Fourier's law, Eq. 1.2, â²â²q = k TL Î where ÎT denotes the difference in surface temperatures. Since ÎT is the same for both surfaces, it follows that L = L k k q q1 2 1 2 2 1 â â²â² â²â² . With heat fluxes related as â²â² = â²â²q 0.8 q1 2 L = 100 mm 0.75 W/m K 0.25 W/m K 1 0.8 = 375 mm.1 â â Ã < NOTES: I do not know the temperature difference at on both sides of the walls, we do not find the value of the heat coefficient. 15. 1.14. KNOWN ASSIGNMENT: Expression of the variable thermal conductivity of the wall. Constant heat flow. Temperature at x = 0. FIND: Expression for temperature gradient and temperature distribution. SCHEMATIC: ASSUMPTIONS: (1) One-dimensional conductivity. ANALYSIS: The heat flux is given by Fourier's law and is known to be constant, so x dT q k constant dx â²â² = â = Solving for the temperature gradient and substituting the expression for k gives x xq qdT dx k ax b â²â² â²â² = â = â + < The expression can be integrated to find the temperature distribution as: xqdT dx dx dx ax b â²â ² = â +â« â« Since the constant xqâ²â² = , we can integrate the right-hand side to find ( ) xq T ln ax b c a â²â² = â + + where c is the integration constant. Using the known condition that T = T1 at x = 0, we can solve for c. Continuedâ¦ qâ x k = ax + b T1 16. Problem 1.14 (Continued) 1 x 1 x 1 T(x 0) T q ln b c T a q c T ln b a = = â²â² â + == + Therefore, the temperature distribution is given by ( )x x 1 q q T ln ax b T ln b a a â²â² â²â² = â + + + < x 1 q b T ln a ax b â ²â² = + + < NOTES: Temperature distribution Many situations non-linear, for example, if the thermal conductivity varies in space or is a function of temperature. Non-linear temperature distributions can also arise when there is internal energy production or when non-stationary conditions are present. 17.1.15. KNOWN PROBLEM: Thickness, diameter and temperature of the inner surface of the bottom of the pot used to boil water. The rate of heat transfer in the pan. CONCLUSION: temperature of the outer surface of the vessel for aluminum and copper bottoms. SCHEME: ASSUMPTIONS: (1) One-dimensional stationary conduction through the bottom of the dish. ANALYSIS: According to Fourier's law, the rate of heat transfer due to heat conduction through the bottom of the vessel is T T1 2q kA L ≈ = So qL T T1 2 kA = + where ( )22 2A D / 4 0.2 m /4 0.0314 m T 110 C 110.24 C1 2390 W/m K 0.0314 m = + = o o < NOTES: Although the temperature drop at the bottom is slightly greater for aluminum (due to its lower thermal conductivity), it is small enough that not significant for both materials. To a good approximation, the bottom can be considered isothermal at T = 110 °C, which is a desirable property for pots and pans. 18.1.16. KNOWN ISSUE: Chip dimensions and thermal conductivity. The power is distributed over one surface. DETECTED: Drop in chip temperature. SCHEME: ASSUMPTIONS: (1) steady-state conditions, (2) constant properties, (3) uniform heat dissipation, (4) negligible rear and side heat losses, (5) one-dimensional thermal conductivity of the chip. ANALYSIS: All electrical energy dissipated on the back surface of the chip is carried by the chip. Thus, from Fourier's law P = q = kA T t O or ( ) t P 0.001 m 4 W T = kW 150 W/m K 0.005 m2 ∞ Γ O = ∞ OT = 1.1 C.o < NOTES. At constant P, the temperature drop across the crystal decreases with increasing k and W and decreasing t. 19. 1.17. KNOWN ISSUE: Heat flow and convective heat transfer coefficient of water. Saturation temperature and coefficient of convective heat transfer of a boiling dielectric fluid. FIND: The temperature of the upper surface of the plate when water boils. Is it planned to reduce the surface temperature with a dielectric liquid. DIAGRAM: Tsat,d = 52°C q" = 20 × 105 W/m2 Tsat,w = 100°C hw = 20,000 W/m2 K hd = 3000 W/m2 K ASSUMPTION: Uniform conditions. PROPERTIES: Tsat ,w = 100°C at p = 1 atm ANALYSIS: According to the description of the problem, Newton's law of cooling for the boiling process can be expressed as sat( )sq h T Tâ²â² = â So sat /st T q hâ²â²= + When the liquid is water, 5 2 , saturated ., 3 2 20 10 W/m / 100 C 200 C 20 10 W/m K s w wT T q h Ã â²â²= + = Â° + = Â° Ã â If a dielectric fluid is used, 5 2 , sat , 3 2 20 10 W/m / 52 C 719 C 3 10 W/m K s d d dT T q h Ã â²â²= + = Â° + = Â° Ã â Thus, the approach suggested by the technician will not lead to a decrease in surface temperature < NOTES: (1) Although the dielectric fluid has a lower saturation temperature, which is more than offset by the lower heat transfer coefficient associated with the dielectric fluid, the surface temperature of the dielectric coolant exceeds the melting point of many metals such as alloys aluminum and indium, alum (2) Liquids are dielectrics, but are used in applications such as immersion cooling of electronic components where an electrically conductive liquid such as water cannot be used. 20. 1.18. KNOWN ISSUE: The arm experiences convective heat exchange with moving air and water. FIND: Determine which state is colder. Compare these results with a heat loss of 30 W/m2 under normal room conditions. DIAGRAM: ASSUMPTIONS: (1) The temperature on the surface of the hand is uniform, (2)ratio is the same throughout the hand and (3) negligible exchange of radiation between the hand and the environment in the case of airflow. ANALYSIS: The hand will be cooler because the position causes more heat loss. Heat loss can be determined by Newton's law of cooling, Eq. 1.3a, written as ( )sq h T Tââ²â² = â For air flow: ( )2 2 airq 40W m K 30 5 K 1400W mâ²â² â¡ â ¤= â â â =â£ â¦ < For water flow: ( ) 2 2 waterq 900 W m K 30 10 K 18,000 W mâ²â² = â â = < COM : The heat loss of the hand in the water flow is an order of magnitude higher than in the air flow at a given temperature and convection speed. In contrast, heat loss in a normal room environment is only 30 W/m2, which is 400 times less than air flow loss. A hand surrounded by a room would feel comfortable; in currents of air and water, as you probably know from experience, the hand would become uncomfortably cold because the heat loss is too great. Fundamentals of Heat and Mass Transfer 7th Edition Incropera Solutions Manual Full Download: This is only sample Download all chapters from: testbankreal.com testbankreal.com