

**PRAIRIE VIEW A&M UNIVERSITY
ROY G. PERRY COLLEGE OF ENGINEERING
DEPARTMENT OF MECHANICAL ENGINEERING
MCEG 3301 – Heat Transfer
FALL 2021**

- I. COURSE DESCRIPTION IN ABET FORMAT**
- II. ASSESSMENT RESULTS FOR CURRENT AND PAST THREE PERIODS**
 - Program Outcomes Assessed in Course**
 - Assessment Summary of Class Semester Averages for Outcomes**
 - End of Semester Course Outcomes Assessment Reports**
- III. TESTS**
 - Summary of Topics covered by Test**
 - Original Test Questions & Master Solution/grading scheme used**
(Summary of topics can be a separate page or include on the assignment itself)
 - Above Average (Graded original work)**
 - Average (Graded original work)**
 - Below Average (Graded original work)**
- IV. HOMEWORK ASSIGNMENTS**
 - Summary of Topics covered by Homework**
 - Original Homework Questions & Master Solution/grading scheme used**
 - Above Average (Graded original work)**
 - Average (Graded original work)**
 - Below Average (Graded original work)**
- V. PROJECT**
 - Summary of Topics Covered by Project**
 - Project Problem Statement and Grading Scheme**
 - Above Average (Graded original work)**
 - Average (Graded original work)**
 - Below Average (Graded original work)**

Course Syllabus in ABET Format

1. **Course number and name:** MCEG 3301 Heat Transfer
2. **Credits and contact hours:** 3 credit hours; 3 contact hours
3. **Instructor's or course coordinator's name:** Dr. Yuhao Xu
4. **Textbook:** Title: Fundamentals of Heat and Mass Transfer

Author: Theodore L. Bergman, Adrienne S. Lavine, Frank P. Incropera, and David P. DeWitt

Publisher, Year & Edition: John Wiley & Sons, 2011, 7th Edition
 - a. Other Supplemental material Fundamentals of Heat and Mass Transfer, Frank P. Incropera, David P. DeWitt, Theodore L. Bergman, and Adrienne S. Lavine, 6th Edition, John Wiley & Sons, 2006.
5. **Specific course information**
 - a. Brief description of the content of the course (catalog description) MCEG 3301 Heat Transfer: Credit 3 semester hours. Study of the fundamental modes of heat transfer, conduction, convection, and thermal radiation, separately and in combination. Theoretical, numerical, and design methods of analysis of steady, transient, single, and multidimensional problems will be emphasized.
 - b. Prerequisites or Co-requisites MATH 2320 Differential Equations
MCEG 3306 Fluid Mechanics
 - c. *Indicate whether a required, elective, or selected elective (as per definition provided) course in the program Required

6. Specific goals for the course

- a. Specific outcomes of instruction (e.g.,
The student will be able to explain the significance of current research about a particular topic.)

Students will be able to:

- (1) display and apply an understanding of the various ways or modes (conduction, convection, and radiation) in which heat can be transferred and relate these to other engineering requirements using the First Law of Thermodynamics Analysis.
- (2) evaluate, compare, and assess the significance of heat transfer in multi-mode applications.
- (3) determine the local rate of heat transfer and the relationship it has with thermodynamics properties, fluid flow properties, and other non-dimensional properties (e.g., the Reynolds and Biot numbers).
- (4) address heat transfer design requirements in processes or devices subject to specified or implied constraints.
- (5) apply engineering approximation, methodology, design, and analysis (empirical, analytical, and numerical) of both steady and transient problems in both simple and complex geometries.

- b. Explicitly indicate which of the student outcomes listed in **Criterion 3** or any other outcomes are addressed by the course.

The specific goals for the course-specific outcomes listed on the left column will address the following student learning outcomes:

Outcome 1

An ability to identify, formulate, and solve complex engineering problems by applying principles of engineering, science, and mathematics.

Outcome 2

An ability to apply engineering design to produce solutions that meet specified needs with consideration of public health, safety, and welfare, as well as global, cultural, social, environmental, and economic factors.

7 Brief list of topics to be covered

Introduction to heat transfer	(3 hours)
Introduction to conduction	(3 hours)
One-dimensional steady-state conduction	(3 hours)
Two-dimensional steady-state conduction	(3 hours)
Transient conduction	(3 hours)
Introduction to convection	(4.5 hours)
External convection	(4.5 hours)
Internal convection	(3 hours)
Free convection	(3 hours)
Heat exchangers	(3 hours)
Radiation processes and properties	(3 hours)
Radiation between surfaces	(3 hours)

***Required** courses are required of all students in the program, **elective** courses (often referred to as open or free electives) are optional for students, and **selected elective** courses are those for which students must take one or more courses from a specified group.

II ASSESSMENT RESULTS FOR CURRENT AND PAST THREE PERIODS

Program Outcomes Assessed in this course

Outcome 1

An ability to identify, formulate, and solve complex engineering problems by applying principles of engineering, science, and mathematics.

Outcome 2

An ability to apply engineering design to produce solutions that meet specified needs with consideration of public health, safety, and welfare, as well as global, cultural, social, environmental, and economic factors.

Assessment Summary of Class Semester Averages for Outcomes (Should be provided for Fall 2019 through Spring 2022 Assessment Period)

Outcomes	1	2
Semester	Semester Class Average (%)	Semester Class Average (%)
Fall 2021		77.7
Spring 2021	76.5	
Fall 2020		75.6
Fall 2019	67.4	
Comments on Trend of Data (Improving or deteriorating)	The class average in Spring 2021 is 76.5%, which is above the ideal semester class average of 75%.	The class average in Fall 2021 is 77.7%, which is above the ideal semester class average of 75%.

*Expected Semester Class Average = 70% (or as defined by the program)
Ideal Semester Class Average = 75% (or as defined by the program)

End of Semester Course Assessment Reports Fall 2021 and Last Three Assessment Periods.

Each summary contains the following Information

1. Performance Statistics

- Total number of students in class
- The class average performance (**stated as %**) in each course outcome assessed in this class and the **two previous times course was assessed**.
- The acceptable class average for the class in each outcome area as set by your Department.
- Percentage of students who scored below the expected average in each outcome.

2. Implementation Summary

- Brief summary of plans implemented during the semester based on last assessment report

3. Perceived Problems

- Instructor's **critical evaluation** of perceived problems that affected students' performance. **Realistic problems should be identified throughout the semester and summarized in this section of the report.**

4. New Plans for Addressing Problems

- Instructor's plans for addressing the perceived problems the next time the class is taught. **These should be specific enough for another instructor to implement. Most of these should be things that, you, the instructor can implement to help students to improve. Be creative.**

5. Overall Trend over last three assessment periods

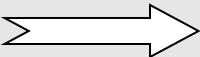
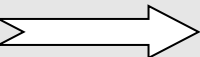
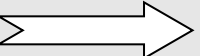
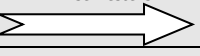
- Instructors summary of overall performance trend for each outcome measured over the last three periods.

6. Were Expectations Met?

- Answer **YES** if the **class average was above the expected average** and the **stated percentage of students** scored at or above the expected average, else answer NO.

END OF SEMESTER COURSE OUTCOME ASSESSMENT REPORT
MCEG 3301-P01 Heat Transfer
FALL 2021 SEMESTER

Report Prepared by: Dr. Yuhao Xu
 Report Date: December 20, 2021

Semester	Analysis Type	Number of Students	Outcome 1 Ability to identify, formulate, and solve complex engineering problems (CEP) by applying principles of engineering, science, and mathematics.				Outcome 2 Ability to apply engineering design to produce solutions that meet specified needs with consideration of public health, safety, and welfare, as well as global, cultural, social, environmental, and economic factors.			
			Expected/Acceptable Class Average/	Expected Percent of students at or above average	Class Average	% Students meeting Acceptable Average	Expected/Acceptable Class Average/	Expected Percent of students at or above average	Class Average	% Students meeting Acceptable Average
Fall 2021, P01	Direct	18					70	70	77.7	72.2
Spring 2021, Z02	Direct	11	70	70	76.5	72.7				
Fall 2020, Z01	Direct	27					70	70	75.6	70.4
Fall 2019, P01	Direct	22	70	70	67.4	54.5				
Fall 2018, P01	Direct	30	70	70	56.5	16.7				
Implementation Summary	Items implemented from previous report or after meeting previous instructor 		OUTCOME 1 WAS NOT ASSESSED IN FALL 2021.				Practical applications of relevant sections were discussed so that students could use the relevant knowledge in their design project. Students were given a sufficient amount of time to work on the project and write the report.			
Perceived Problems	Perceived problems are directly related to the PC and the sub items under them 		OUTCOME 1 WAS NOT ASSESSED IN FALL 2021.				Students' ability to perform a literature review and use the information found in the design project needs to be improved. Students' skills in writing a professional report need to be improved.			
Plans for Addressing Problems	Plans for address specific measures to address PC and their sub areas 		OUTCOME 1 WAS NOT ASSESSED IN FALL 2021.				The following plan is suggested: 1. Share relevant resources with students (e.g., those offered in the library) to help students with the literature review. 2. Explain the project requirements several times in class and encourage questions from students to ensure they understand the requirements.			
Overall Trend over Periods	Ascertain if there were improvements over previous semesters 		OUTCOME 1 WAS NOT ASSESSED IN FALL 2021.				The class average was 77.7%, which is above the expected class average. 72.2% of students had an average above the 70% level. The overall expectation was met in Outcome "2".			
Were Expectations Met?			OUTCOME 1 WAS NOT ASSESSED IN FALL 2021.				YES			

**Heat Transfer
FALL 2021**

Title of Assignment: TEST 1

TOPICS COVERED IN TEST 1

Test 1 covered the following topics:

1. Fourier's law
2. Newton's cooling law
3. Net radiation heat flux from surfaces
4. Surface irradiation and absorption
5. Surface energy balance
6. Thermal resistance network in one-dimensional steady-state heat transfer
7. The heat diffusion equation and its applications
8. Boundary conditions for the heat diffusion equation

**Heat Transfer
FALL 2021**

**SAMPLE TEST QUESTIONS, SOLUTION,
AND GRADING SCHEME FOR TEST 1**

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #1
October 7, 2021
8:00 AM – 9:20 AM

Print your name (Last name first)

Signature and Date

Score:

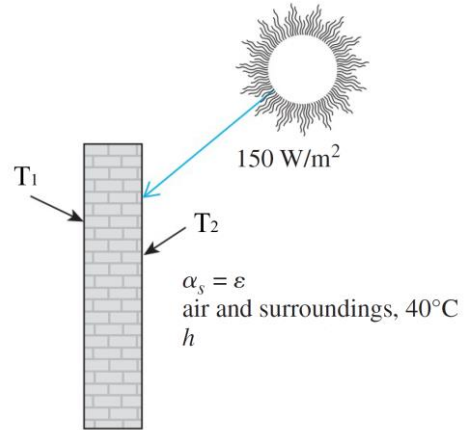
100

Instructions:

- Please stay on this page until you are asked to start the test.
- Only the required textbook outlined in the syllabus can be used in the test. Any other types of written materials, such as homework and quizzes, are not allowed.
- Make sure to write your answers clearly. Any unreadable answers will result in zero credit. **Place boxes around your answers.**
- Cell phones, iPads, laptops, and any other electronic devices should be turned off and put away during the test. Only pens, pencils, rulers, erasers, and calculators can be used in the test.
- You are not allowed to leave the classroom during the test.
- Academic dishonesty of any sort (plagiarism, cheating, fabrication, or facilitation) is prohibited under any circumstance. Any detected actions of dishonesty will cause the failure of the class directly.
- Writing your name in the space provided above signifies that you will abide by the Prairie View A&M University Code of Student Conduct. It also signifies you understand that **you will get a zero for this test if you have a cell phone or other electronic devices with you after the test starts.**

PROBLEM #1 (30 pts.)

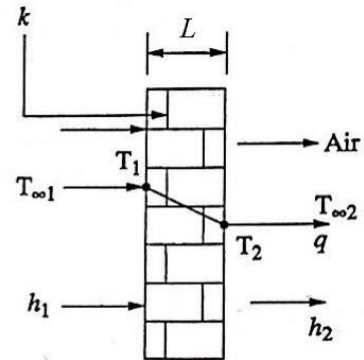
The inner and outer surfaces of a 20-cm-thick wall in summer are at $T_1 = 30^\circ\text{C}$ and $T_2 = 44^\circ\text{C}$, respectively. The outer surface of the wall exchanges heat by radiation with surrounding surfaces at 40°C , and convection with ambient air also at 40°C with a convection heat transfer coefficient of $10 \text{ W}/(\text{m}^2\cdot\text{K})$. Solar radiation is incident on the surface at a rate of $150 \text{ W}/\text{m}^2$. If both the emissivity and the solar absorptivity of the outer surface are 0.8, determine the thermal conductivity of the wall.



PROBLEM #2 (35 pts.)

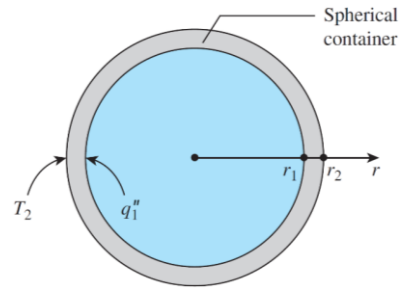
Consider a wall of a house that is 4 m high (H), 5 m wide (W), and 0.36 m thick (L). It is made from bricks with a thermal conductivity of $0.90 \text{ W}/(\text{m}\cdot\text{K})$. The inner surface of the wall is exposed to an airstream with $T_{\infty 1} = 25^\circ\text{C}$ and $h_1 = 12 \text{ W}/(\text{m}^2\cdot\text{K})$. The outer surface is exposed to an airstream with $T_{\infty 2} = -10^\circ\text{C}$ and $h_2 = 30 \text{ W}/(\text{m}^2\cdot\text{K})$. Assuming one-dimensional heat transfer and steady-state conditions,

- draw the thermal resistance network for this problem. Clearly label the heat transfer rate (or the heat flux, as appropriate), all temperatures, and corresponding thermal resistances on the network,
- calculate the heat flux through the wall, and
- determine the outside wall temperature T_2 .



PROBLEM #3 (35 pts.)

A spherical container, with an inner radius $r_1 = 1$ m and an outer radius $r_2 = 1.08$ m, has its inner surface subjected to a uniform heat flux of $q_1'' = 5.5$ kW/m². The outer surface of the container has a temperature $T_2 = 20^\circ\text{C}$, and the container wall thermal conductivity is $k = 1.8$ W/(m·K). Determine the following:



(a) Write the general form of the differential equation for heat conduction in this coordinate (Hint: This should be the general form of the heat equation with three coordinates in space and time, as well as the generation term).

(b) Simply the heat equation in (a) and express the simplified differential equation for heat conduction in the container wall under one-dimensional steady conditions, assuming constant thermal conductivity and no heat generation in the wall.

(c) How many boundary conditions do you need to solve the differential equation obtained in (b)? Provide the boundary condition(s).

(d) Obtain a general relation to express the variation of temperature in the container wall $T(r)$ by solving the differential equation obtained in (b) and applying boundary condition(s) obtained in (c). (Hint: Express $T(r)$ in terms of r , r_1 , r_2 , k , q_1'' , and T_2 .)

(e) Calculate the temperature at $r = r_1$ and $r = 1.05$ m using the expression obtained in (d).

#1

Perform an energy balance on the outer surface of the wall:

$$\begin{aligned} q_{\text{cond}}'' &= k \frac{T_2 - T_1}{L} \\ q_{\text{conv}}'' &= h(T_2 - T_{\infty}) \\ q_{\text{rad}}'' &= \epsilon \sigma (T_2^4 - T_{\text{surr}}^4) \\ q_{\text{solar}}'' &= \alpha_s q_{\text{solar, incident}}'' \end{aligned}$$

5 $q_{\text{solar}}'' = q_{\text{cond}}'' + q_{\text{conv}}'' + q_{\text{rad}}''$

$$\alpha_s q_{\text{solar, incident}}'' = k \frac{T_2 - T_1}{L} + h(T_2 - T_{\infty}) + \epsilon \sigma (T_2^4 - T_{\text{surr}}^4)$$

$$(0.8)(150 \frac{\text{W}}{\text{m}^2}) = k \frac{(44 - 30)^{\circ}\text{C}}{0.2 \text{ m}} + (10 \frac{\text{W}}{\text{m}^2 \cdot \text{K}})(44 - 40)^{\circ}\text{C}$$

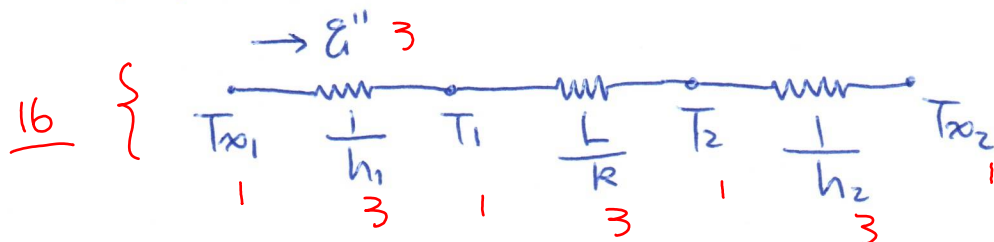
2 $+ (0.8)(5.67 \times 10^{-8} \frac{\text{W}}{\text{m}^2 \cdot \text{K}^4}) [(44 + 273 \text{ K})^4 - (40 + 273 \text{ K})^4]$

Solving for k gives:

3 $k = 0.819 \frac{\text{W}}{\text{m} \cdot \text{K}}$

#2.

(a) Thermal resistance network :



12

$$(b) \quad q'' = \frac{T_{\infty 1} - T_{\infty 2}}{\frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_2}} = \frac{25^\circ\text{C} - (-10^\circ\text{C})}{\frac{1}{12 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}} + \frac{0.36 \text{ m}}{0.9 \frac{\text{W}}{\text{m} \cdot \text{K}}} + \frac{1}{30 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}}} = 67.7 \frac{\text{W}}{\text{m}^2}$$

5+5 2

7

$$(c) \quad q'' = \frac{T_2 - T_{\infty 2}}{\frac{1}{h_2}} \quad 5$$

$$\Rightarrow T_2 = T_{\infty 2} + \frac{q''}{h_2} = -10^\circ\text{C} + \frac{67.7 \frac{\text{W}}{\text{m}^2}}{30 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}} = -7.7^\circ\text{C}$$

2

$$[\text{OR: } q'' = \frac{T_{\infty 1} - T_2}{\frac{1}{h_1} + \frac{L}{k}} \Rightarrow T_2 = T_{\infty 1} - q'' \left(\frac{1}{h_1} + \frac{L}{k} \right) = \dots = -7.7^\circ\text{C}]$$

#3

(a) For spherical coordinates, the heat conduction equation is:

$$\frac{1}{r^2} \frac{\partial}{\partial r} (kr^2 \frac{\partial T}{\partial r}) + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \phi} (k \frac{\partial T}{\partial \phi}) + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \theta} (k \sin \theta \frac{\partial T}{\partial \theta}) + \dot{q}''' = \rho c_p \frac{\partial T}{\partial t} \quad (2.29)$$

(b). For 1-D steady operations, with const. k and without $\dot{q}'''$,

the reduced equation is: $\frac{d}{dr} (r^2 \frac{dT}{dr}) = 0$

(c) Two (2) boundary conditions are needed. They are:

$$\left\{ \begin{array}{l} 2 \\ 4 \end{array} \right. \quad \begin{array}{l} r=r_1 : -k \frac{dT}{dr} \Big|_{r=r_1} = \dot{q}_1'' \\ r=r_2 : T(r_2) = T_2 \end{array}$$

(d). Integrate the differential equation twice with respect to r yields:

$$r^2 \frac{dT}{dr} = C_1, \text{ or } \frac{dT}{dr} = \frac{C_1}{r^2}$$

$$T(r) = -\frac{C_1}{r} + C_2 \quad \text{where } C_1 \text{ and } C_2 \text{ are arbitrary constants.}$$

Apply the boundary conditions:

$$r=r_1 : -k \frac{dT}{dr} \Big|_{r=r_1} = \dot{q}_1'' = -k \frac{C_1}{r_1^2} \rightarrow C_1 = -\dot{q}_1'' \frac{r_1^2}{k}$$

$$r=r_2 : T(r_2) = T_2 = -\frac{C_1}{r_2} + C_2 \rightarrow C_2 = T_2 + \frac{C_1}{r_2} = T_2 - \frac{\dot{q}_1''}{k} \frac{r_1^2}{r_2}$$

$$\text{So that } T(r) = -\frac{C_1}{r} + C_2 = \frac{\dot{q}_1''}{k} \frac{r_1^2}{r} + T_2 - \frac{\dot{q}_1''}{k} \frac{r_1^2}{r_2} = \dot{q}_1'' \frac{r_1^2}{k} \left(\frac{1}{r} - \frac{1}{r_2} \right) + T_2$$

$$(e) T(r=r_1=1\text{m}) = (5500 \frac{\text{W}}{\text{m}^2}) \frac{(1\text{m})^2}{1.8 \frac{\text{W}}{\text{m}\cdot\text{K}}} \left(\frac{1}{1\text{m}} - \frac{1}{1.08\text{m}} \right) + 20^\circ\text{C} = 246.3^\circ\text{C}$$

$$T(r=1.05\text{m}) = (5500 \frac{\text{W}}{\text{m}^2}) \frac{(1\text{m})^2}{1.8 \frac{\text{W}}{\text{m}\cdot\text{K}}} \left(\frac{1}{1.05\text{m}} - \frac{1}{1.08\text{m}} \right) + 20^\circ\text{C} = 100.8^\circ\text{C}$$

ABOVE AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #1
October 7, 2021
8:00 AM – 9:20 AM

Marcel, Tyler
Print your name (Last name first)

Tyler Marcel 10-7-21
Signature and Date

Score: _____
100

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17/30

PROBLEM #1 (30 pts.)

The inner and outer surfaces of a 20-cm-thick wall in summer are at $T_1 = 30^\circ\text{C}$ and $T_2 = 44^\circ\text{C}$, respectively. The outer surface of the wall exchanges heat by radiation with surrounding surfaces at 40°C , $T_s = 40^\circ\text{C}$ and convection with ambient air also at 40°C with a convection heat transfer coefficient of $10 \text{ W}/(\text{m}^2 \cdot \text{K})$. Solar radiation is incident on the surface at a rate of $150 \text{ W}/\text{m}^2$. If both the emissivity and the solar absorptivity of the outer surface are 0.8, determine the thermal conductivity of the wall.

$q_{\text{conv}} + q_{\text{rad}} = hA(T_s - T_{\text{wall}}) + \epsilon A \sigma (T_s^4 - T_{\text{wall}}^4)$
 $-q_{\text{cond}} = q_{\text{conv}} + q_{\text{rad}} = h(T_s - T_{\text{wall}}) + \epsilon \sigma (T_s^4 - T_{\text{wall}}^4)$
 $q_{\text{rad}} - q_{\text{cond}} - q_{\text{conv}} = 0$ *how about irradiation?*
 $\epsilon \sigma (T_s^4 - T_{\text{wall}}^4) - K \frac{T_1 - T_2}{L} - h(T_s - T_{\text{wall}}) = 0$
 $\epsilon \sigma (T_s^4 - T_{\text{wall}}^4) - h(T_s - T_{\text{wall}}) = K \frac{T_1 - T_2}{L}$
 $(0.8)(5.67 \times 10^{-8})(44^4 - 40^4) - 10(44 - 40) = K \frac{30 - 44}{0.2}$
 $0.0539 - 40 = K(-70)$
 $\frac{-39.95}{-70} = K$
 $K = 0.57 \text{ W}/\text{m} \cdot \text{K}$

$h = 10 \text{ W}/\text{m}^2 \cdot \text{K}$
 $\sigma = 5.67 \times 10^{-8} \text{ W}/\text{m}^2 \cdot \text{K}^4$
 $\epsilon = \alpha_s = 0.8$

$150 \text{ W}/\text{m}^2$
 $T_1 = 30^\circ\text{C}$
 303K
 $T_2 = 44^\circ\text{C}$
 317K
 $\alpha_s = \epsilon$
 $\text{air and surroundings, } 40^\circ\text{C}$
 h
 $L = 0.2 \text{ m}$

30/35

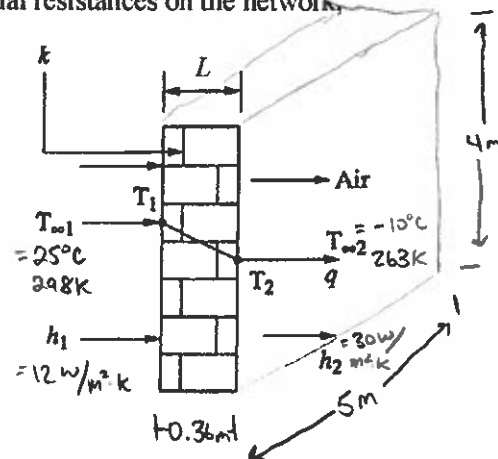
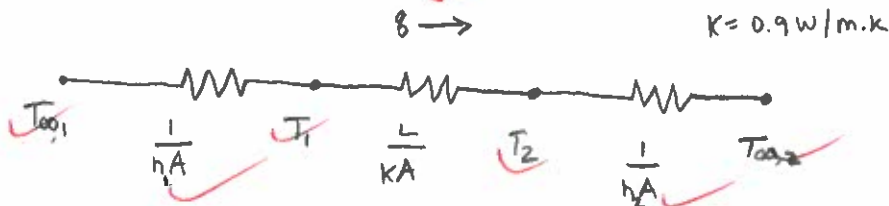
PROBLEM #2 (35 pts.)

Consider a wall of a house that is 4 m high (H), 5 m wide (W), and 0.36 m thick (L). It is made from bricks with a thermal conductivity of $0.90 \text{ W/(m}\cdot\text{K)}$. The inner surface of the wall is exposed to an airstream with $T_{\infty 1} = 25^\circ\text{C}$ and $h_1 = 12 \text{ W/(m}^2\cdot\text{K)}$. The outer surface is exposed to an airstream with $T_{\infty 2} = -10^\circ\text{C}$ and $h_2 = 30 \text{ W/(m}^2\cdot\text{K)}$. Assuming one-dimensional heat transfer and steady-state conditions,

(a) draw the thermal resistance network for this problem. Clearly label the heat transfer rate (or the heat flux, as appropriate), all temperatures, and corresponding thermal resistances on the network,

(b) calculate the heat flux through the wall, and

(c) determine the outside wall temperature T_2 .



$$= \frac{1}{(12)(20)} + \frac{0.36}{(0.9)(20)} + \frac{1}{(30)(20)}$$

$$q_x = \frac{T_{\infty 1} - T_{\infty 2}}{2 R_{t,c}}$$

$$R_{t,c} = 0.0258$$

$$q_x = \frac{298\text{K} - 263\text{K}}{0.0258} = 1356.59 \text{ W}$$

$$q'' = \frac{k}{L} (T_{s,1} - T_{s,2})$$

$$67.83 = \frac{0.9}{0.36} (298 - T_{s,2})$$

there are NO $T_{s,1}$, $T_{s,2}$ in this problem!

$$q'' = \frac{1356.59 \text{ W}}{20 \text{ m}^2}$$

$$q'' = 67.83 \text{ W/m}^2$$

$$27.13 = 298 - T_{s,2}$$

$$T_{s,2} = 298 - 27.13$$

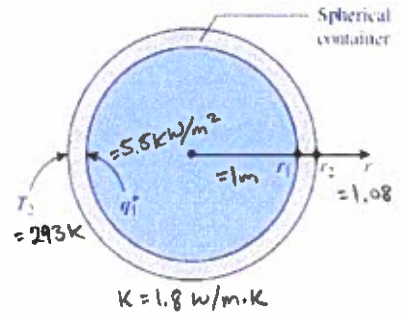
$$T_{s,2} = 270.87 \text{ K}$$

$$\nabla^2 T = -k \left(\frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial T}{\partial r} \right) + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \theta} \left(\sin \theta \frac{\partial T}{\partial \theta} \right) + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \phi} \left(\sin \theta \frac{\partial T}{\partial \phi} \right) \right) \quad \text{where, } q_r'' = -k \frac{\partial T}{\partial r}$$

11/35

PROBLEM #3 (35 pts.)

A spherical container, with an inner radius $r_1 = 1$ m and an outer radius $r_2 = 1.08$ m, has its inner surface subjected to a uniform heat flux of $q_1'' = 5.5$ kW/m². The outer surface of the container has a temperature $T_2 = 20^\circ\text{C}$, and the container wall thermal conductivity is $k = 1.8$ W/(m·K). Determine the following:



(a) Write the general form of the differential equation for heat conduction in this coordinate (Hint: This should be the general form of the heat equation with three coordinates in space and time, as well as the generation term).

(b) Simply the heat equation in (a) and express the simplified differential equation for heat conduction in the container wall under one-dimensional steady conditions, assuming constant thermal conductivity and no heat generation in the wall.

(c) How many boundary conditions do you need to solve the differential equation obtained in (b)? Provide the boundary condition(s).

(d) Obtain a general relation to express the variation of temperature in the container wall $T(r)$ by solving the differential equation obtained in (b) and applying boundary condition(s) obtained in (c). (Hint: Express $T(r)$ in terms of r , r_1 , r_2 , k , q_1'' , and T_2 .)

(e) Calculate the temperature at $r = r_1$ and $r = 1.05$ m using the expression obtained in (d).

$$a) \frac{1}{r^2} \frac{\partial}{\partial r} \left(k r^2 \frac{\partial T}{\partial r} \right) + \frac{1}{r^2 \sin^2 \theta} \frac{\partial}{\partial \phi} \left(k \frac{\partial T}{\partial \phi} \right) + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \theta} \left(k \sin \theta \frac{\partial T}{\partial \theta} \right) + \dot{q} = \rho c_p \frac{\partial T}{\partial t}$$

How did you get this?

$$b) T = \frac{C_1}{r} + C_2$$

$$T_1 = \frac{C_1}{r_1} + C_2$$

$$T_1 - T_2 = \frac{C_1}{r_1} + C_2 - \frac{C_1}{r_2} - C_2$$

$$c) \begin{aligned} T(r_1) &= T_1 \\ T(r_2) &= T_2 \end{aligned}$$

$$T_2 = \frac{C_1}{r_2} + C_2$$

$$T_1 - T_2 = C_1 \left(\frac{r_1 - r_2}{r_1 r_2} \right)$$

$$C_1 = \frac{T_1 - T_2}{\left(\frac{r_1 - r_2}{r_1 r_2} \right)}$$

$$T = \frac{T_1 - T_2 (r)}{\left(\frac{r_1 - r_2}{r_1 r_2} \right)} + C_2$$

wrong boundary conditions.

$$q'' = -k \frac{\partial T}{\partial r}$$

$$q'' = -k \frac{T_1 - T_2}{r_2 - r_1}$$

$$\frac{q''(r_2 - r_1)}{-k} + T_2 = T_1$$

$$\frac{(5.5)(1.05 - 1)}{-1.8} + 293 \text{ K} = T_1$$

$$T(r_1) = T_1 = 292.85 \text{ K}$$

$$\frac{q''(r_2 - r_1)}{-k} - T_1 = -T_2$$

$$\frac{(5.5)(1.05 - 1)}{-1.8} - 292.85 = -T_2$$

58/100

AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #1
October 7, 2021
8:00 AM – 9:20 AM

Dupre Ty

Print your name (Last name first)

Ty Dupre

Signature and Date

10-7-2021

Score:

100

Instructions:

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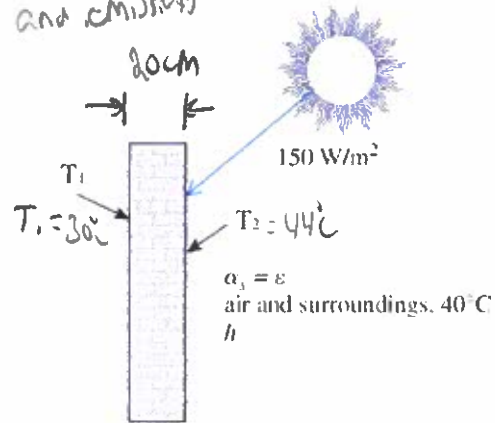
hwk chap 1
Energy balance

PROBLEM #1 (30 pts.)

The inner and outer surfaces of a 20-cm-thick wall in summer are at $T_1 = 30^\circ\text{C}$ and $T_2 = 44^\circ\text{C}$, respectively. The outer surface of the wall exchanges heat by radiation with surrounding surfaces at 40°C , and convection with ambient air also at 40°C with a convection heat transfer coefficient of $10 \text{ W}/(\text{m}^2 \cdot \text{K})$. Solar radiation is incident on the surface at a rate of $150 \text{ W}/\text{m}^2$. If both the emissivity and the solar absorptivity of the outer surface are 0.8, determine the thermal conductivity of the wall.

$T_1 = 30^\circ\text{C}$ $T_2 = 44^\circ\text{C}$ 0.8 absorptivity and emissivity
Thermal conductivity

$$k_x = - \frac{q_x''}{(\partial T / \partial x)} \quad \Delta x = 20 \text{ cm} \quad \Delta T = 44 - 30 = 14^\circ\text{C}$$



Convection heat transfer process

$$q'' = h(T_s - T_\infty) = 10 \text{ W}/(\text{m}^2 \cdot \text{K})(44 - 40) = 40 \text{ W}/\text{m}^2$$

$$q'' = -100 \text{ W}/\text{m}^2 \text{ convection}$$

Radiation

$$q''_{\text{rad}} = \frac{q}{A} = \epsilon F_0(T_s) - \alpha G = \epsilon \sigma (T_s^4 - T_{\text{sur}}^4)$$

Conduction

$$q''_x = k \frac{T_1 - T_2}{L} = k \frac{\Delta T}{L}$$

Surface energy balance

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = 0$$

$$q''_{\text{cond}} - q''_{\text{conv}} - q''_{\text{rad}} = 0$$

NOT complete,
how about irradiation?

find convection and radiation q''_x

then use energy balance to solve for conduction q''_x .

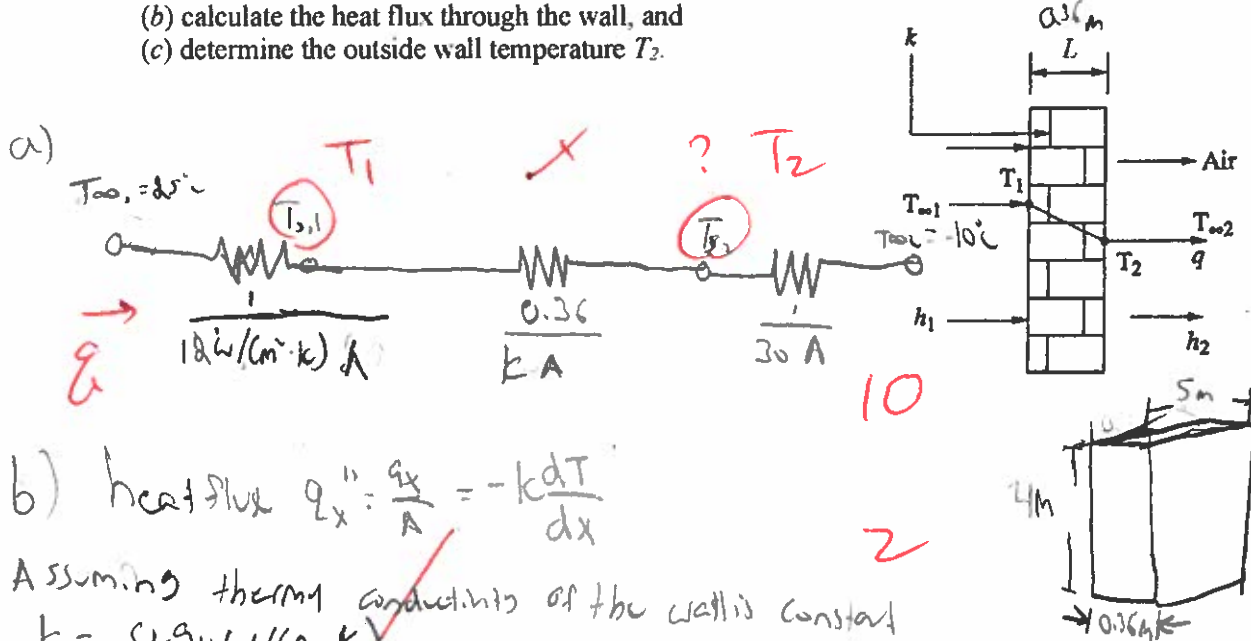
then solve for thermal conductivity

12/35

PROBLEM #2 (35 pts.)

Consider a wall of a house that is 4 m high (H), 5 m wide (W), and 0.36 m thick (L). It is made from bricks with a thermal conductivity of $0.90 \text{ W/(m}\cdot\text{K)}$. The inner surface of the wall is exposed to an airstream with $T_{\infty 1} = 25^\circ\text{C}$ and $h_1 = 12 \text{ W/(m}^2\cdot\text{K)}$. The outer surface is exposed to an airstream with $T_{\infty 2} = -10^\circ\text{C}$ and $h_2 = 30 \text{ W/(m}^2\cdot\text{K)}$. Assuming one-dimensional heat transfer and steady-state conditions,

- draw the thermal resistance network for this problem. Clearly label the heat transfer rate (or the heat flux, as appropriate), all temperatures, and corresponding thermal resistances on the network,
- calculate the heat flux through the wall, and
- determine the outside wall temperature T_2 .



b) heat flux $q_x'' = \frac{q_x}{A} = -k \frac{dT}{dx}$

Assuming thermal conductivity of the wall is constant
 $k = 0.90 \text{ W/(m}\cdot\text{K)}$

$$q_x = kA \frac{\Delta T}{\Delta x} = 0.9 (20 \text{ m}^2) \left(\frac{14^\circ\text{C}}{0.36 \text{ m}} \right) = 700 \text{ W}$$

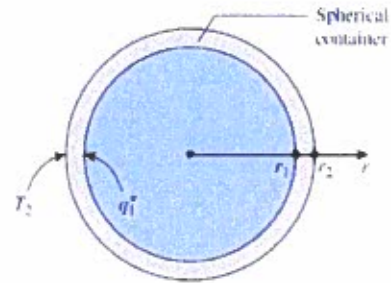
$$\text{heat flux} = q_x'' = \frac{q_x}{A} = \frac{700}{20} = 35 \text{ W/m}^2$$

c)

4/35

PROBLEM #3 (35 pts.)

A spherical container, with an inner radius $r_1 = 1$ m and an outer radius $r_2 = 1.08$ m, has its inner surface subjected to a uniform heat flux of $q_1'' = 5.5$ kW/m². The outer surface of the container has a temperature $T_2 = 20^\circ\text{C}$, and the container wall thermal conductivity is $k = 1.8$ W/(m·K). Determine the following:



Equation in book →

(a) Write the general form of the differential equation for heat conduction in this coordinate (Hint: This should be the general form of the heat equation with three coordinates in space and time, as well as the generation term).

Simplified equation →

(b) Simply the heat equation in (a) and express the simplified differential equation for heat conduction in the container wall under one-dimensional steady conditions, assuming constant thermal conductivity and no heat generation in the wall.

(c) How many boundary conditions do you need to solve the differential equation obtained in (b)? Provide the boundary condition(s).

(d) Obtain a general relation to express the variation of temperature in the container wall $T(r)$ by solving the differential equation obtained in (b) and applying boundary condition(s) obtained in (c). (Hint: Express $T(r)$ in terms of r, r_1, r_2, k, q_1'' , and T_2 .)

(e) Calculate the temperature at $r = r_1$ and $r = 1.05$ m using the expression obtained in (d).

✓ a) $\frac{1}{r^2} \frac{\partial}{\partial r} \left(1.8 \cdot 0.08 \frac{\partial T}{\partial r} \right) + \frac{1}{r^2 \sin^2 \theta} \frac{\partial}{\partial \theta} \left(1.8 \frac{\partial T}{\partial \theta} \right) + \frac{\partial}{\partial z} \left(1.8 \frac{\partial T}{\partial z} \right) \cdot \dot{q} = \rho c_p \frac{\partial T}{\partial t}$

$r_2 - r_1 = 0.08$ m

b) $\frac{\partial^2 T}{\partial r^2} + \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{k} = \frac{1}{\alpha} \frac{\partial T}{\partial t}$

c) $q_1'(0) = -k \frac{\partial T}{\partial x} \bigg|_{x=0} = q_1''$

33/100

BELOW AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #1
October 7, 2021
8:00 AM – 9:20 AM

Okezie, Chinonso
Print your name (Last name first)

 10/7/21
Signature and Date

Score:
100

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5/30

PROBLEM #1 (30 pts.)

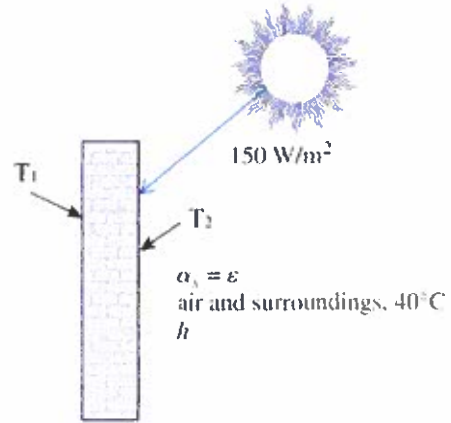
The inner and outer surfaces of a 20-cm-thick wall in summer are at $T_1 = 30^\circ\text{C}$ and $T_2 = 44^\circ\text{C}$, respectively. The outer surface of the wall exchanges heat by radiation with surrounding surfaces at 40°C , and convection with ambient air also at 40°C with a convection heat transfer coefficient of $10 \text{ W}/(\text{m}^2 \cdot \text{K})$. Solar radiation is incident on the surface at a rate of $150 \text{ W}/\text{m}^2$. If both the emissivity and the solar absorptivity of the outer surface are 0.8, determine the thermal conductivity of the wall.

$$T_1 = 30^\circ\text{C} \quad T_2 = 44^\circ\text{C}$$

$$\text{Solar radiation} = 150 \text{ W}/\text{m}^2$$

$$L = 20 \text{ cm}$$

$$\text{heat transfer coefficient} = 10 \frac{\text{W}}{(\text{m}^2 \cdot \text{K})}$$



$$q''_x = k \frac{\Delta T}{L} \quad \checkmark$$

5

$$L = 20 \text{ cm} = \frac{1 \text{ m}}{1 \times 10^2 \text{ cm}} = 0.2 \text{ m}$$

$$\Delta T = T_2 - T_1 = 44 - 30 = 14^\circ\text{C}$$

$$150 \frac{\text{W}}{\text{m}^2} = \frac{k (14^\circ\text{C})}{0.2 \text{ m}}$$

$$k = (150 \frac{\text{W}}{\text{m}^2}) \left(\frac{0.2 \text{ m}}{14^\circ\text{C}} \right) \approx 2.14 \text{ W}/\text{m} \cdot \text{K}$$

10/35

PROBLEM #2 (35 pts.)

Consider a wall of a house that is 4 m high (H), 5 m wide (W), and 0.36 m thick (L). It is made from bricks with a thermal conductivity of $0.90 \text{ W/(m}\cdot\text{K)}$. The inner surface of the wall is exposed to an airstream with $T_{\infty 1} = 25^\circ\text{C}$ and $h_1 = 12 \text{ W/(m}^2\cdot\text{K)}$. The outer surface is exposed to an airstream with $T_{\infty 2} = -10^\circ\text{C}$ and $h_2 = 30 \text{ W/(m}^2\cdot\text{K)}$. Assuming one-dimensional heat transfer and steady-state conditions,

(a) draw the thermal resistance network for this problem. Clearly label the heat transfer rate (or the heat flux, as appropriate), all temperatures, and corresponding thermal resistances on the network,

(b) calculate the heat flux through the wall, and

(c) determine the outside wall temperature T_2 .

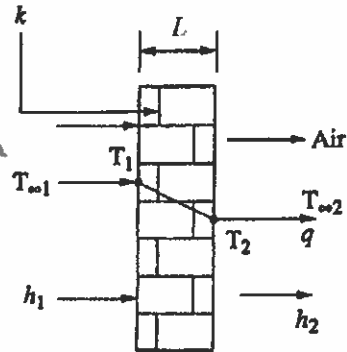
(a) No T provided!

Height = 4 m Width = 5 m thick $L = 0.36 \text{ m}$

$= 0.36 \times 10^3 \text{ mm}$

$k = 0.90 \text{ W/(m}\cdot\text{K)}$ $T_{\infty 1} = 25^\circ\text{C}$ $T_{\infty 2} = -10^\circ\text{C}$

$h_1 = 12 \text{ W/(m}^2\cdot\text{K)}$ $h_2 = 30 \text{ W/(m}^2\cdot\text{K)}$



$$R_{\text{conv}} = \frac{T_1 - T_{\infty 1}}{q} = \frac{1}{hA}$$

$$A = 4 \text{ m} \times 5 \text{ m} = 20 \text{ m}^2 = 20 \times 10^3 \text{ mm}^2$$

$$T_{\infty} = T_{\infty 2} - T_{\infty 1} = -10 - 25 = -35^\circ\text{C}$$

$$R_{\text{tot}} = \frac{1}{h_1 A} + \frac{L}{kA} + \frac{1}{h_2 A} = \frac{1}{12(20)} + \frac{0.36}{(0.90)(20)} + \frac{1}{(30)(20)} = \frac{1}{240} + 20 + \frac{1}{600}$$

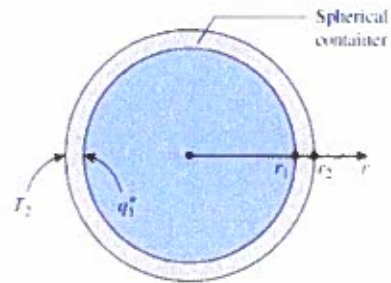
$$R_{\text{tot}} \approx 20.0058$$

$$q_x = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{tot}}} = \frac{25 + 10}{20.0058} \approx 1.75$$

PROBLEM #3 (35 pts.)

2/35

A spherical container, with an inner radius $r_1 = 1$ m and an outer radius $r_2 = 1.08$ m, has its inner surface subjected to a uniform heat flux of $q_1'' = 5.5$ kW/m². The outer surface of the container has a temperature $T_2 = 20^\circ\text{C}$, and the container wall thermal conductivity is $k = 1.8$ W/(m·K). Determine the following:



(a) Write the general form of the differential equation for heat conduction in this coordinate (Hint: This should be the general form of the heat equation with three coordinates in space and time, as well as the generation term).

(b) Simply the heat equation in (a) and express the simplified differential equation for heat conduction in the container wall under one-dimensional steady conditions, assuming constant thermal conductivity and no heat generation in the wall.

(c) How many boundary conditions do you need to solve the differential equation obtained in (b)? Provide the boundary condition(s).

(d) Obtain a general relation to express the variation of temperature in the container wall $T(r)$ by solving the differential equation obtained in (b) and applying boundary condition(s) obtained in (c). (Hint: Express $T(r)$ in terms of r, r_1, r_2, k, q_1'' , and T_2 .)

(e) Calculate the temperature at $r = r_1$ and $r = 1.05$ m using the expression obtained in (d).

$$r_1 = 1 \text{ m} \quad r_2 = 1.08 \text{ m} \quad T_2 = 20^\circ\text{C}$$
$$q_1'' = 5.5 \text{ kW/m}^2 \quad k = 1.8 \text{ W/(m}\cdot\text{K)}$$

$$T(r) = T_{s,1} - (T_{s,1} - T_{s,2}) \left[\frac{1 - (r_1/r)}{1 - (r_1/r_2)} \right]$$

$$q_r = \frac{4\pi k (T_{s,1} - T_{s,2})}{(1/r_1) - (1/r_2)}$$

$$\frac{4\pi k \Delta T}{(1/r_1) - (1/r_2)}$$

How?

$$\frac{1}{r} \frac{d}{dr} \left(k r \frac{dT}{dr} \right) = 0$$

$$q_r = -k(2\pi r L) \frac{dT}{dr}$$

$$\frac{1}{r^2} \frac{d}{dr} \left(r^2 \frac{dT}{dr} \right) = 0$$

$$\frac{k \Delta T}{r^2 \left[(1/r_1) - (1/r_2) \right]}$$

17/100

**Heat Transfer
FALL 2021**

Title of Assignment: TEST 2

TOPICS COVERED IN TEST 2

Test 2 covered the following topics:

1. The energy balance method to develop the finite-difference equations in two-dimensional, steady-state conduction
2. Transient conduction
3. Heat transfer from extended surfaces

**Heat Transfer
FALL 2021**

**SAMPLE TEST QUESTIONS, SOLUTION,
AND GRADING SCHEME FOR TEST 2**

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #2
November 11, 2021
8:00 AM – 9:20 AM

Print your name (Last name first)

Signature and Date

Score:

100

Instructions:

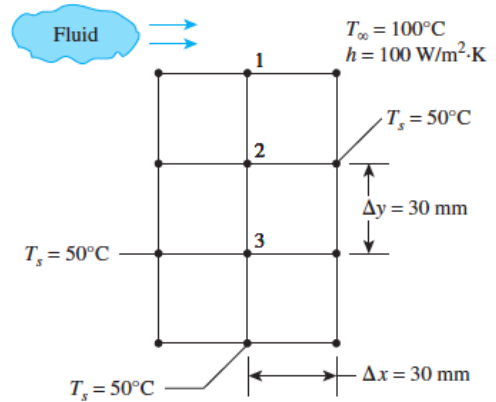
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PROBLEM #1 (25 pts.)

Consider a long bar of rectangular cross-section (60 mm by 90 mm on a side) and of thermal conductivity $1.2 \text{ W/(m}\cdot\text{K)}$. The top surface is exposed to a convection process with air at 100°C and a convection heat transfer coefficient of $100 \text{ W/(m}^2\cdot\text{K)}$, while the remaining surfaces are maintained at 50°C . Use a grid spacing of 30 mm, as shown in the figure on the right.

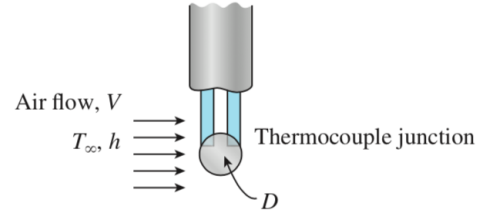
(a) Perform the energy balance for nodes #1 and #3, and write the nodal equations for these nodes.

(b) Write the final form of the finite difference expression for node #1.



PROBLEM #2 (30 pts.)

In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 36 \text{ W/(m}\cdot\text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 316 \text{ J/(kg}\cdot\text{K)}$. The hot air has a free stream temperature of 130°C , and the heat transfer coefficient between the free stream and the thermocouple junction is $239 \text{ W/(m}^2\cdot\text{K)}$. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple junction to reach 129°C .

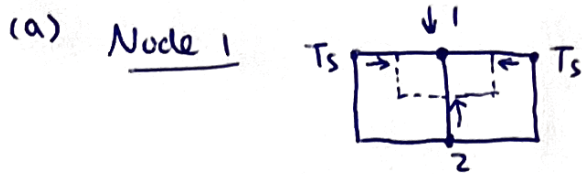


PROBLEM #3 (45 pts.)

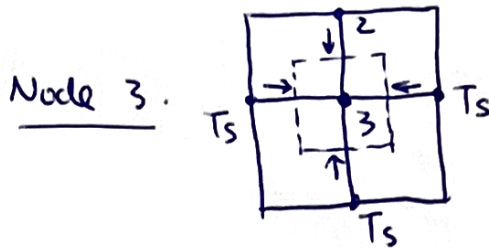
A very long rod, 26 mm in diameter, has one end maintained at 115°C. The surface of the rod is exposed to ambient air at 30°C with a convective heat transfer coefficient of 21 W/(m²·K). The rod is constructed from pure copper. Determine:

- (a) The temperature of the rod at 160 mm from the wall.
- (b) The rate of heat loss from the rod to the air.
- (c) The efficiency and effectiveness of the heat transfer.

#1



10 $k\left(\frac{\Delta y}{2} \cdot 1\right) \frac{T_5 - T_1}{\Delta x} + k\left(\frac{\Delta y}{2} \cdot 1\right) \frac{T_6 - T_1}{\Delta x} + k(\Delta x \cdot 1) \frac{T_2 - T_1}{\Delta y} + h(\Delta x \cdot 1)(T_\infty - T_1) = 0$... (1)



10 $k(\Delta x \cdot 1) \frac{T_2 - T_3}{\Delta y} + k(\Delta x \cdot 1) \frac{T_4 - T_3}{\Delta y} + k(\Delta y \cdot 1) \frac{T_5 - T_3}{\Delta x} + k(\Delta y \cdot 1) \frac{T_6 - T_3}{\Delta x} = 0$... (2)

(b) From Eq. (1):

$$(T_5 - T_1) + (T_6 - T_1) + \frac{h\Delta x}{k}(T_\infty - T_1) = 0$$

or, $(2 + \frac{h\Delta x}{k})T_1 - T_2 - T_5 - \frac{h\Delta x}{k}T_\infty = 0$

where $\frac{h\Delta x}{k} = \frac{(100 \frac{W}{m^2 \cdot K})(0.03m)}{1.2 \frac{W}{m \cdot K}} = 2.5$

$T_5 = 50^\circ C$, & $T_\infty = 100^\circ C$

so that, $4.5T_1 - T_2 = 300^\circ C$

#2.

$$5+2 \quad Bi = \frac{hL_c}{k} = \frac{(239 \frac{W}{m^2 \cdot K}) (\frac{0.001 m}{6})}{36 W/m \cdot K} = 0.001106 < 0.1$$

$$5 \quad \text{where } L_c = \frac{r_0}{3} = \frac{D}{6}$$

3 Since $Bi < 0.1$, the lumped system analysis is valid.

$$10 \quad \frac{T - T_\infty}{T_i - T_\infty} = \exp \left[- \frac{hA_s}{\rho C_p V} \cdot t \right]$$

$$\Rightarrow t = \frac{\rho C_p V}{hA_s} \ln \frac{T_i - T_\infty}{T - T_\infty} = \frac{\rho C_p (\frac{D}{6})}{h} \ln \frac{T_i - T_\infty}{T - T_\infty}$$

$$= \left(\frac{8500 \times 316 \times \frac{0.001}{6}}{239} \ln \frac{27 - 130}{129 - 130} \right) s$$

$$5 \quad = 8.7 s$$

#3

(a), very long rod, $\rightarrow$ treated as infinitely long rod.

and $T_b = 115^\circ\text{C}$, $T_\infty = 30^\circ\text{C}$, $h = 21 \text{ W/m}^2\cdot\text{K}$

12 { 2+5 Table 3.4, Case D : $\frac{\theta}{\theta_b} = \frac{T - T_\infty}{T_b - T_\infty} = e^{-mx}$

5 where $m = \sqrt{\frac{hP}{kA_c}}$

2 To find k : $T = \frac{115^\circ\text{C} + 30^\circ\text{C}}{2} = 72.5^\circ\text{C} = 345.5 \text{ K}$

3 From Table A.1, $k = 398.45 \frac{\text{W}}{\text{m}\cdot\text{K}}$

2 $P = \pi D = \pi(0.026 \text{ m}) = 0.08168 \text{ m}$

2 $A_c = \frac{\pi D^2}{4} = \frac{\pi}{4}(0.026 \text{ m})^2 = 5.309 \times 10^{-4} \text{ m}^2$

3 so that $m = \sqrt{\frac{hP}{kA_c}} = \sqrt{\frac{(21 \frac{\text{W}}{\text{m}^2\cdot\text{K}})(0.08168 \text{ m})}{(398.45 \frac{\text{W}}{\text{m}\cdot\text{K}})(5.309 \times 10^{-4} \text{ m}^2)}} = 2.848 \frac{1}{\text{m}}$

3 And $T = T_\infty + (T_b - T_\infty)e^{-mx}$
 $= 30^\circ\text{C} + (115^\circ\text{C} - 30^\circ\text{C})e^{-(2.848 \text{ m}^{-1})(0.16 \text{ m})} = 83.89^\circ\text{C}$

(b) $\dot{Q}_f = \sqrt{hPkA_c} \cdot (T_b - T_\infty) = 51.20 \text{ W}$

4+2 (c) Efficiency : $\eta_f = \frac{\dot{Q}_f}{hA_f(T_b - T_\infty)} = \frac{\sqrt{hPkA_c} \theta_b}{h \cdot P \cdot L \theta_b} = \sqrt{\frac{kA_c}{hP}} \cdot \frac{1}{L} = \frac{1}{mL}$

4
2 $L \rightarrow \infty$, so $\eta_f \rightarrow 0$

18 { Effectiveness: $\varepsilon_f = \frac{\dot{Q}_f}{hA_b(T_b - T_\infty)} = \frac{\sqrt{hPkA_c} \theta_b}{hA_c \theta_b} = \sqrt{\frac{Pk}{hA_c}} = 54.03$

4+2

ABOVE AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #2
November 11, 2021
8:00 AM – 9:20 AM

McFadden Darin
Print your name (Last name first)

Darin 11/11/21
Signature and Date

Score: _____
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PROBLEM #1 (25 pts.)

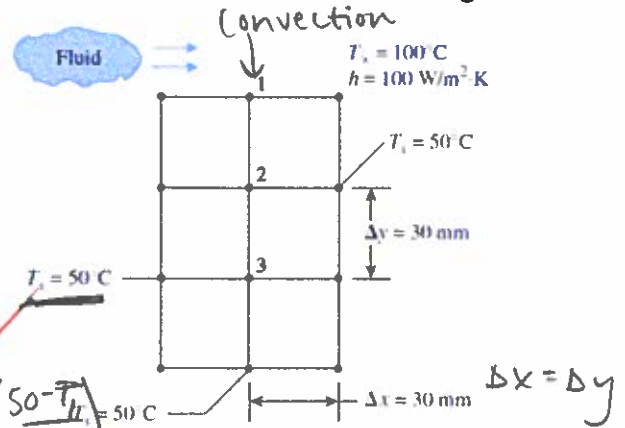
Chapter 4

18/25

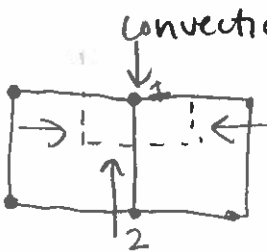
Consider a long bar of rectangular cross-section (60 mm by 90 mm on a side) and of thermal conductivity $1.2 \text{ W/(m} \cdot \text{K)}$. The top surface is exposed to a convection process with air at 100°C and a convection heat transfer coefficient of $100 \text{ W/(m}^2 \cdot \text{K)}$, while the remaining surfaces are maintained at 50°C . Use a grid spacing of 30 mm, as shown in the figure on the right.

(a) Perform the energy balance for nodes #1 and #3, and write the nodal equations for these nodes.

(b) Write the final form of the finite difference expression for node #1.



Node 1:



nodal equation

$$K \left(\frac{\Delta y}{2} \cdot 1 \right) \left(\frac{50 - T_1}{\Delta x} \right) + K \left(\frac{\Delta y}{2} \cdot 1 \right) \left(\frac{50 - T_1}{\Delta x} \right) + K (\Delta x \cdot 1) \left(\frac{50 - T_2}{\Delta y} \right) + K (\Delta x \cdot 1) \left(\frac{50 - T_4}{\Delta y} \right) + h \left(\Delta x \cdot \frac{\Delta y}{2} \cdot 1 \right) = 0$$

$$+ K (\Delta x \cdot 1) \left(\frac{50 - T_2}{\Delta y} \right) + K (\Delta x \cdot 1) \left(\frac{50 - T_4}{\Delta y} \right)$$

$$+ h \left(\Delta x \cdot \frac{\Delta y}{2} \cdot 1 \right) = 0$$

$$= (50 - T_1) (1 + 4K) + h \left(\Delta x \cdot \frac{\Delta y}{2} \cdot 1 \right) = 0$$

$$= (50 - T_1) (1 + 4K) + \frac{h}{2} (30)^2 = 0$$

$$50 - T_1 =$$

Node 3:

nodal equation

$$K (\Delta x \cdot 1) \left(\frac{T_2 - T_3}{\Delta y} \right) + K (\Delta y \cdot 1) \left(\frac{50 - T_3}{\Delta x} \right) + K (\Delta y \cdot 1) \left(\frac{50 - T_3}{\Delta x} \right) + K (\Delta x \cdot 1) \left(\frac{50 - T_3}{\Delta y} \right) = 0$$

$$= 0 = K (T_2 - T_3) + 3K (50 - T_3) = 0$$

$$= 1.2 (T_2 - T_3 + 3.6 (50 - T_3)) = 0$$

not needed

PROBLEM #2 (30 pts.)

In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 36 \text{ W/(m}\cdot\text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 316 \text{ J/(kg}\cdot\text{K)}$. The hot air has a free stream temperature of 130°C , and the heat transfer coefficient between the free stream and the thermocouple junction is $239 \text{ W/(m}^2\cdot\text{K)}$. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple junction to reach 129°C .

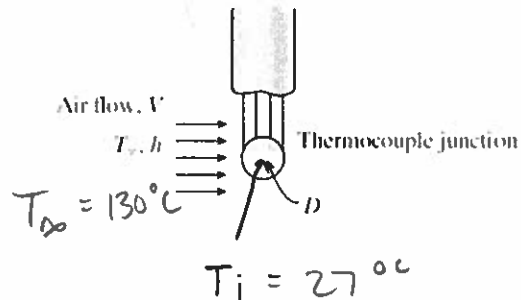
Given : $D = 1 \text{ mm} = .001 \text{ m}$

$$k = 36 \text{ W/m}\cdot\text{K}$$

$$\rho = 8500 \text{ kg/m}^3$$

$$c_p = 316 \text{ J/kg}\cdot\text{K}$$

$$h = 239 \text{ W/m}^2\cdot\text{K}$$



Check $Bi < .1 \rightarrow$ sphere

So that ... ?

$$Bi = \frac{h(r/3)}{k} = \frac{(239 \text{ W/m}^2\cdot\text{K}) \left(\frac{5 \times 10^{-4} \text{ m}}{3} \right)}{36 \text{ (W/m}\cdot\text{K)}}$$

$$Bi = .0011$$

$$Bi < .1$$

$$t = \frac{\rho V c_p}{h A_s} \ln \left(\frac{T_i - T_\infty}{T - T_\infty} \right) = \rho \left(\frac{\pi D^3}{6} \right) c_p \ln \left(\frac{T_i - T_\infty}{T - T_\infty} \right) / h (\pi D^2)$$

$$= \frac{8500 \text{ kg/m}^3 \left(\frac{\pi (.001)^3}{6} \right) (316 \text{ J/kg}\cdot\text{K})}{(239 \text{ W/m}^2\cdot\text{K}) (\pi) (.001)^2} \ln \left(\frac{27 - 130}{129 - 130} \right)$$

$$t = 8.68 \text{ seconds} = .144 \text{ hrs}$$

$$\frac{\rho D c_p}{6h} =$$

43/45

PROBLEM #3 (45 pts.) Infinite

A very long rod, 26 mm in diameter, has one end maintained at 115°C. The surface of the rod is exposed to ambient air at 30°C with a convective heat transfer coefficient of 21 W/(m²·K). The rod is constructed from pure copper. Determine:

- (a) The temperature of the rod at 160 mm from the wall.
- (b) The rate of heat loss from the rod to the air.
- (c) The efficiency and effectiveness of the heat transfer.

Given: $T_{\infty} = 30^{\circ}\text{C}$ $D = .026\text{ m}$
 $T_b = 115^{\circ}\text{C}$
 $h = 21\text{ W/m}^2\cdot\text{K}$
 pure copper

Assumption → infinitely long: Case D

Take average temperature: $\frac{T_b + T_{\infty}}{2} = \frac{115^{\circ}\text{C} + 30^{\circ}\text{C}}{2} = 72.5^{\circ}\text{C}$

$72.5^{\circ}\text{C} + 273 = 345.5$ K

pure copper from Table A.1

200	413
345.5	k
400	393

linear interpolation

27

$\frac{k - 413}{393 - 413} = \frac{345.5 - 200}{400 - 200}$

$k = 398.45\text{ W/m}\cdot\text{K}$

Temp of rod 160mm from wall:

$T(.16\text{ m}) = T_{\infty} + (T_b - T_{\infty})e^{-mx}$

$m = \sqrt{\frac{hP}{kA_c}} = \sqrt{\frac{h\pi D}{k(\pi D^2/4)}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(21\text{ W/m}^2\cdot\text{K})}{398.45(.026)}}$

$m = 2.8475\text{ 1/m}$ $T(.16\text{ m}) = 30^{\circ}\text{C} + (115^{\circ}\text{C} - 30^{\circ}\text{C})e^{-2.8475(.16)}$

rate of heat loss: unit

$T(.16\text{ m}) = 83.9^{\circ}\text{C}$

$q_f = \left(\sqrt{hPkA_c}\right)\theta_b = \left(\sqrt{hk(\pi D)(\pi D^2/4)}\right)\theta_b$

$b\ q_f = \left(\sqrt{\frac{hk\pi^2 D^3}{4}}\right)(T_b - T_{\infty}) = \sqrt{(21\text{ W/m}^2\cdot\text{K})(398.45\text{ W/m}\cdot\text{K})(\pi^2)(.026)}$

$q_f = 51.2\text{ Watts}$

$\times (115 - 30)$ efficiency on back

effectiveness $\epsilon_f = \sqrt{\frac{kP}{hA_c}} = \sqrt{\frac{k(\pi D)}{h(\pi D^2/4)}} = \sqrt{\frac{4k}{hD}} = \sqrt{\frac{4(398.45\text{ W/m}\cdot\text{K})}{21(.026)}} = 54.028 = \epsilon_f$

efficiency

$$\eta_f = \frac{q_f}{q_{\max}} = \frac{q_f}{h A_f \theta_b} = \frac{51.2 \text{ watts}}{21 \text{ W/m}^2 \cdot \text{K} \left(\pi \frac{(0.026)^2}{4} \right) (115 - 30)}$$

4

$\eta_f = 54.025$

~~91/100~~

AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #2
November 11, 2021
8:00 AM – 9:20 AM

Green, Guy
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Guy Green 11/11/2021
Signature and Date

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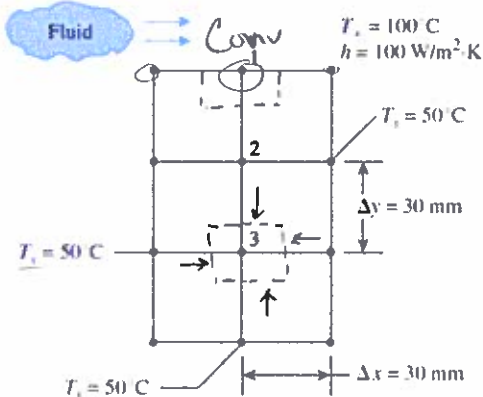
PROBLEM #1 (25 pts.)

Chapter 4

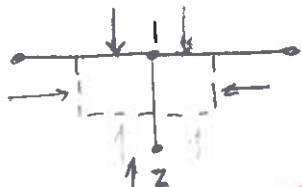
Consider a long bar of rectangular cross-section (60 mm by 90 mm on a side) and of thermal conductivity 1.2 W/(m·K). The top surface is exposed to a convection process with air at 100°C and a convection heat transfer coefficient of 100 W/(m²·K), while the remaining surfaces are maintained at 50°C. Use a grid spacing of 30 mm, as shown in the figure on the right.

(a) Perform the energy balance for nodes #1 and #3, and write the nodal equations for these nodes.

(b) Write the final form of the finite difference expression for node #1.



a) Node 1:



$$k \left(\frac{\Delta x}{2} \cdot l_m \right) \frac{T_{\infty} - T_1}{\Delta y} + k \left(\frac{\Delta y}{\Delta x} \cdot l_m \right) \frac{T_2 - T_1}{\Delta x} + k \left(\frac{\Delta y}{\Delta x} \cdot l_m \right) \frac{T_1 - T_1}{\Delta x} = 0$$

should be convection!

Node 3:

$$k \left(\Delta y \cdot l_m \right) \frac{T_s - T_3}{\Delta x} + k \left(\Delta x \cdot l_m \right) \frac{T_2 - T_3}{\Delta y} + k \left(\Delta x \cdot l_m \right) \frac{T_s - T_3}{\Delta y} = 0$$

b.)

8

NOT complete.

PROBLEM #2 (30 pts.)

In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 36 \text{ W/(m} \cdot \text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 316 \text{ J/(kg} \cdot \text{K)}$. The hot air has a free stream temperature of 130°C , and the heat transfer coefficient between the free stream and the thermocouple junction is $239 \text{ W/(m}^2 \cdot \text{K)}$. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple junction to reach 129°C .

Properties/Given

$$k = 36 \text{ W/m} \cdot \text{K}$$

$$\rho = 8500 \text{ kg/m}^3$$

$$c_p = 316 \text{ J/kg} \cdot \text{K}$$

$$h = 239 \text{ W/m}^2 \cdot \text{K}$$

$$T_i = 27^\circ\text{C}$$

$$T = 129$$

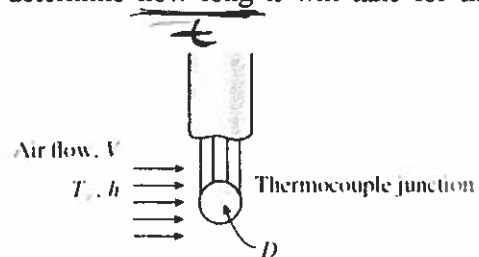
$$T_\infty = 130$$

$$D = 1 \text{ mm} = 0.001 \text{ m}$$

$$r = 5 \times 10^{-4} \text{ m}$$

$h = \text{conv}$

$k = \text{cond}$



$$\frac{\theta_i}{\theta} = \frac{T_i - T_\infty}{T - T_\infty}$$

$$\tau = \frac{\rho V c_p}{h A_s} \ln \frac{\theta_i}{\theta}$$

Schematic:

$$Bi = \frac{h L_c}{k} = L_c = \frac{r_0}{3}$$

$$= \frac{(239 \text{ W/m}^2 \cdot \text{K}) (5 \times 10^{-4} \text{ m})}{36 \text{ W/m} \cdot \text{K}}$$

$$= 0.001 < 0.1$$

$$\text{Sphere} = \frac{\frac{4}{3} \pi r^3}{\frac{4 \pi r^2}{6}} = \frac{D}{6}$$

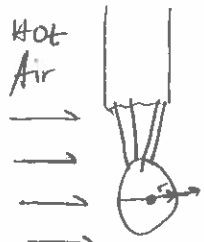
Biot number is the first thing to be calculated when determining which rule to use. With the calculated number being smaller than '0.1', the error using lump capacitance is a small chance.

$$\tau = \frac{\rho D c_p}{h(6)} \ln \left(\frac{T_i - T_\infty}{T - T_\infty} \right)$$

$$= \frac{(8500 \text{ kg/m}^3)(0.001 \text{ m})(316 \text{ J/kg} \cdot \text{K})}{239 \text{ W/m}^2 \cdot \text{K}(6)} \ln \left(\frac{27 - 130}{129 - 130} \right)$$

$$= 1.873(4.635)$$

$$\tau = 8.685$$



Conv.

T_∞

PROBLEM #3 (45 pts.)

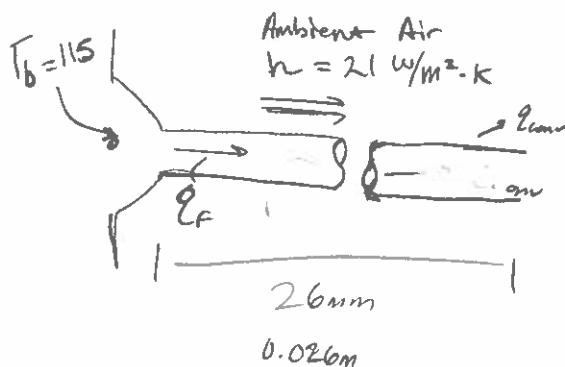
A very long rod, 26 mm in diameter, has one end maintained at 115°C . The surface of the rod is exposed to ambient air at 30°C with a convective heat transfer coefficient of $21 \text{ W}/(\text{m}^2 \cdot \text{K})$. The rod is constructed from pure copper. Determine:

- The temperature of the rod at 160 mm from the wall.
- The rate of heat loss from the rod to the air.
- The efficiency and effectiveness of the heat transfer.

$$\theta_L = \theta_b$$

$$D = 0.0$$

Schematic



Pure Copper Properties

Table A.1

$$T = \frac{T_b + T_a}{2}$$

$$= \frac{115 + 30}{2}$$

missing unit

$$T_{avg} = 72.5^\circ\text{C}$$

$$M = \sqrt{\frac{kA_c}{hP}}$$

$$M = 566.08 \times 10^{-3}$$

$$A_c = 1.062 \times 10^{-3}$$

$$\rho = 1358 \text{ kg}/\text{m}^3$$

$$C_p = 8933 \text{ J}/\text{kg} \cdot \text{K}$$

$$k = 493 \text{ W}/\text{m} \cdot \text{K}$$

$$\alpha = 117 \times 10^{-6}$$

Assumptions:

• Constant properties

• Steady state

No gen; 1-D

Solution:

a.) temperature of the rod @ 160 mm

$$\frac{\theta_L}{\theta_b} = \frac{\sinh mx + \sinh m(L-x)}{\sinh mL}$$

$$\frac{\theta_L}{\theta_b} = \sinh(21)$$

b.) heat loss $= q_F = \sqrt{hPkA_c} \theta_b$

$$M \frac{(\cosh mL - \theta_L/\theta_b)}{\sinh mL}$$

$$\eta_F = \frac{q_F}{q_{F,max}} + \epsilon_F = \frac{q_F}{hA_{c,b}\theta_b}$$

$$\frac{54}{100}$$

BELOW AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #2
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Ibeabuchi, Alvin

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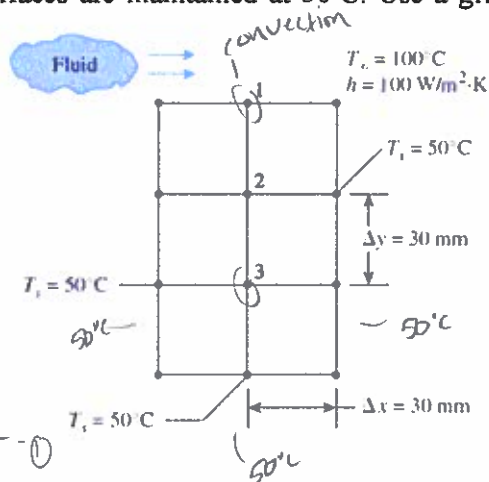
6/25

PROBLEM #1 (25 pts.)

Consider a long bar of rectangular cross-section (60 mm by 90 mm on a side) and of thermal conductivity 1.2 W/(m·K). The top surface is exposed to a convection process with air at 100°C and a convection heat transfer coefficient of 100 W/(m²·K), while the remaining surfaces are maintained at 50°C. Use a grid spacing of 30 mm, as shown in the figure on the right.

(a) Perform the energy balance for nodes #1 and #3, and write the nodal equations for these nodes.

(b) Write the final form of the finite difference expression for node #1.



$$T_{m-1,n} - 2T_{m,n} + T_{m+1,n} + T_{m,n-1} - 2T_{m,n} + T_{m,n+1}$$

$$= 0$$

$$h \Delta x \Delta y (T_{\infty} - T_1) + k \Delta x \Delta y \left(\frac{T_2 - T_1}{\Delta x} \right) + k \frac{\Delta x}{2} \left(\frac{50 - T_1}{\Delta x} \right) + \frac{k \Delta y}{2} (T_{\infty} - T_1) \dots = ?$$

$$h \Delta x \Delta y (T_{\infty} - T_1) + k \Delta x \Delta y \left(\frac{T_2 - T_1}{\Delta x} \right) + k \frac{\Delta x}{2} (50 - T_1) + \frac{k \Delta y}{2} (T_{\infty} - T_1) = 0$$

Node 3: $50 + T_2 + 50 + 50 - 4T_3 = 0$
 $T_3 = 0.25T_2 + 37.5$

Need to show your process!

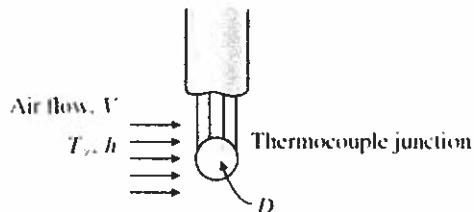
b.) $h \Delta x \Delta y (T_2 - T_1) + 50 - T_1 + T_2 - T_1 = 0$

Need to show the process of energy balance!

PROBLEM #2 (30 pts.)

In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 36 \text{ W/(m} \cdot \text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 316 \text{ J/(kg} \cdot \text{K)}$. The hot air has a free stream temperature of 130°C , and the heat transfer coefficient between the free stream and the thermocouple junction is $239 \text{ W/(m}^2 \cdot \text{K)}$. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple junction to reach 129°C .

$$\text{Biot \#} = \frac{h L_c}{k} = L_c = \frac{V}{A} = \frac{D}{6} = \frac{1 \text{ mm}}{6} = \frac{0.001}{6}$$



$$\frac{239 \times 0.001}{6 \times 36} = 0.0011$$

Since Biot # is less than 0.1, lumped capacitance method can be applied.

$$\frac{T - T_\infty}{T_i - T_\infty} = e^{-\left(\frac{hA}{\rho c_p V}\right)t}$$

$$\frac{129 - 27}{130 - 27} = e^{-\left(\frac{239 \times 6}{8500 \times 316 \times 0.001}\right) \times t}$$

$$t = 0.018275$$

PROBLEM #3 (45 pts.)

A very long rod, 26 mm in diameter, has one end maintained at 115°C. The surface of the rod is exposed to ambient air at 30°C with a convective heat transfer coefficient of 21 W/(m²·K). The rod is constructed from pure copper. Determine:

- The temperature of the rod at 160 mm from the wall.
- The rate of heat loss from the rod to the air.
- The efficiency and effectiveness of the heat transfer.

$t_o = 115^\circ\text{C}$ $t_a = 30^\circ\text{C}$ $h = 21 \text{ W/(m}^2\cdot\text{K)}$

$l = 160 \text{ mm}$

$k_{\text{copper}} \approx 398 \text{ W/mK}$

at what temperature?

Cross-sectional area = $\frac{\pi}{4} d^2 = A_s = \frac{\pi}{4} (0.026)^2$

Perimeter = πd

$\approx 5.6 \times 10^{-4} \text{ m}$

a.) $m = \sqrt{\frac{hP}{kA_c}} = \sqrt{\frac{21 \times \pi \times 0.026}{401 \times 0.026^2}}$

$m =$

$T = T_a + (T_b - T_a) e^{-mx}$

$q = \frac{hP}{kA_c} \left(\frac{1}{2} \right)$

$\frac{43}{100}$

**Heat Transfer
FALL 2021**

Title of Assignment: TEST 3

TOPICS COVERED IN TEST 3

Test 3 covered the following topics:

1. External convection: the flat plate in parallel flow
2. External convection: flow over a sphere
3. Transit conduction

**Heat Transfer
FALL 2021**

**SAMPLE TEST QUESTIONS, SOLUTION,
AND GRADING SCHEME FOR TEST 3**

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #3
December 2, 2021

Print your name (Last name first)

Signature and Date

Score:

100

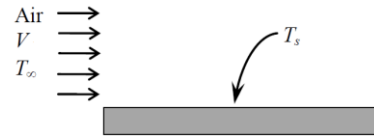
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PROBLEM #1 (40 pts.)

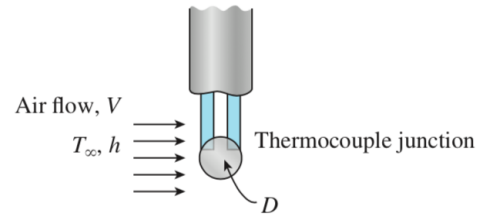
Air at $T_\infty = 20^\circ\text{C}$ and 1 atm flows over a 0.36-m-wide plate at $T_s = 60^\circ\text{C}$ at a velocity of $V = 2.0$ m/s. Using air properties of $\rho = 1.127$ kg/m³, $c_p = 1007$ J/(kg·K), $k = 0.02662$ W/(m·K), $\mu = 1.918 \times 10^{-5}$ kg/(m·s), and $Pr = 0.7255$, compute the following quantities:

- (a) Critical location of the plate associated with the critical Reynolds number;
- (b) Local friction coefficient at $x = 0.5$ m;
- (c) Local Nusselt number at $x = 0.5$ m;
- (d) Average convection heat transfer coefficient over $L = 0.5$ m of the plate; and
- (e) Rate of convective heat transfer over $L = 0.5$ m of the plate.



PROBLEM #2 (60 pts.)

In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 35 \text{ W/(m}\cdot\text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 320 \text{ J/(kg}\cdot\text{K)}$. The hot air has a free stream velocity and temperature of 3 m/s and 127°C , respectively. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple to reach 126°C . [Hint: The first step is to calculate the heat transfer coefficient between the free stream and the thermocouple junction.]



#1

$$3 \quad (a) \quad Re_{x,c} = \frac{\rho V x_c}{\mu} = 5 \times 10^5$$

$$5 \quad \Rightarrow x_c = \frac{Re_{x,c} \cdot \mu}{\rho \cdot V} = \frac{(5 \times 10^5) (1.918 \times 10^{-5} \text{ kg/m}\cdot\text{s})}{(1.127 \text{ kg/m}^3) (2.0 \text{ m/s})} = 4.25 \text{ m}$$

2 (b) since $x = 0.5 \text{ m} < x_c$, the flow at $x = 0.5 \text{ m}$ is in the laminar region.

$$5 \quad Re_x = \frac{\rho \cdot V \cdot x}{\mu} = \frac{(1.127 \text{ kg/m}^3) (2.0 \text{ m/s}) (0.5 \text{ m})}{(1.918 \times 10^{-5} \text{ kg/m}\cdot\text{s})} = 58,759$$

$$5 \quad C_{f,x} = 0.664 Re_x^{-1/2} = 0.664 (58,759)^{-1/2} = 0.00274$$

$$5 \quad (c) \quad Nu_x = 0.332 Re_x^{1/2} Pr^{1/3} = 0.332 (58,759)^{1/2} (0.7255) = 72.3$$

$$5 \quad (d) \quad \bar{Nu}_x = 0.664 Re_x^{1/2} Pr^{1/3} = 2 Nu_x = (2 \times 72.3) = 144.6$$

$$\bar{Nu}_x = \frac{\bar{h}_x \cdot x}{k}$$

$$5 \quad \Rightarrow \bar{h}_x = \bar{Nu}_x \cdot \frac{k}{x} = (144.6) \cdot \frac{0.02662 \text{ W/m}\cdot\text{K}}{0.5 \text{ m}} = 7.70 \frac{\text{W}}{\text{m}^2\cdot\text{K}}$$

$$5 \quad (e) \quad \dot{q} = h \cdot A_s \cdot (T_s - T_\infty) = (7.70 \frac{\text{W}}{\text{m}^2\cdot\text{K}}) (0.5 \times 0.36 \text{ m}^2) (60 - 20)^\circ\text{C} \\ = 55.44 \text{ W}$$

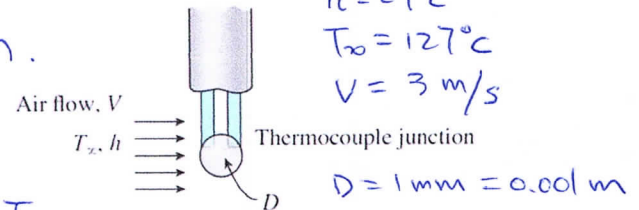
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* To evaluate the heat transfer coefficient h .

$$\overline{Nu}_D = 2 + (0.4 Re_D^{1/2} + 0.06 Re_D^{2/3}) Pr^{0.4} \left(\frac{\mu}{\mu_s}\right)^{1/4}$$

where μ_s is evaluated at T_s and others at T_∞ .



3 Table A.4. Air. $T_s = \frac{T_i + T_{\text{final}}}{2} = \frac{27^\circ\text{C} + 126^\circ\text{C}}{2} = 76.5^\circ\text{C} = 349.5 \text{ K} \approx 350 \text{ K}$,

1 $\rightarrow \mu_s = 208.2 \times 10^{-7} \text{ N}\cdot\text{s/m}^2$

$T_\infty = 127^\circ\text{C} = 400 \text{ K}$, $\rightarrow \mu = 230.1 \times 10^{-7} \text{ N}\cdot\text{s/m}^2$, $\nu = 26.41 \times 10^{-6} \text{ m}^2/\text{s}$

6 $k = 33.8 \times 10^{-3} \frac{\text{W}}{\text{m}\cdot\text{K}}$, $Pr = 0.690$.

6 $Re_D = \frac{V \cdot D}{\nu} = \frac{(3 \text{ m/s})(0.001 \text{ m})}{26.41 \times 10^{-6} \text{ m}^2/\text{s}} = 113.6$

10 $\overline{Nu}_D = 2 + (0.4 Re_D^{1/2} + 0.06 Re_D^{2/3}) Pr^{0.4} \left(\frac{\mu}{\mu_s}\right)^{1/4}$
 $= 2 + [0.4 (113.6)^{1/2} + 0.06 (113.6)^{2/3}] \cdot (0.690)^{0.4} \left(\frac{230.1 \times 10^{-7}}{208.2 \times 10^{-7}}\right)^{1/4}$
 $= 7.01$

8 $h = \frac{\overline{Nu}_D \cdot k_{\text{air}}}{D} = \frac{7.01 \times 33.8 \times 10^{-3} \frac{\text{W}}{\text{m}\cdot\text{K}}}{0.001 \text{ m}} = 236.94 \frac{\text{W}}{\text{m}^2\cdot\text{K}}$

* Transit analysis

10 $Bi = \frac{h \cdot L_c}{k_s} = \frac{h \cdot (\frac{D}{6})}{k_s} = \frac{236.94 \times \frac{0.001}{6}}{35} = 0.00113 < 0.1$

4 $\rightarrow$ lumped system analysis is valid.

10 $\frac{T - T_\infty}{T_i - T_\infty} = \exp\left[-\frac{h A_s}{\rho c_p V} t\right] \Rightarrow t = \frac{\rho c_p V}{h A_s} \ln \frac{T_i - T_\infty}{T - T_\infty} = \frac{\rho c_p (\frac{D}{6})}{h} \ln \frac{T_i - T_\infty}{T - T_\infty}$

2 and $t = \left(\frac{8500 \times 320 \times \frac{0.001}{6}}{236.94} \ln \frac{27 - 127}{126 - 127}\right) \text{ s} = 8.8 \text{ s}$

ABOVE AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #3
December 2, 2021

Yancy, Cori
Print your name (Last name first)

Cori Yancy 12/2/21
Signature and Date

Score: maybe 60?
100

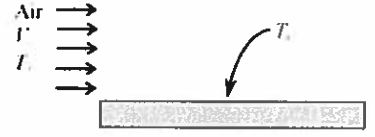
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PROBLEM #1 (40 pts.)

Air at $T_\infty = 20^\circ\text{C}$ and 1 atm flows over a 0.36-m-wide plate at $T_s = 60^\circ\text{C}$ at a velocity of $V = 2.0$ m/s. Using air properties of $\rho = 1.127$ kg/m³, $c_p = 1007$ J/(kg·K), $k = 0.02662$ W/(m·K), $\mu = 1.918 \times 10^{-5}$ kg/(m·s), and $Pr = 0.7255$, compute the following quantities:

- (a) Critical location of the plate associated with the critical Reynolds number;
(b) Local friction coefficient at $x = 0.5$ m;
(c) Local Nusselt number at $x = 0.5$ m;
(d) Average convection heat transfer coefficient over $L = 0.5$ m of the plate; and
(e) Rate of convective heat transfer over $L = 0.5$ m of the plate.



Given

$T_\infty = 20^\circ\text{C}$
@ 1 atm

$T_s = 60^\circ\text{C}$

$V = 2.0$ m/s

$\rho = 1.127$

$c_p = 1007$

$k = 0.02662$

$\mu = 1.918 \times 10^{-5}$

$Pr = 0.7255$

(a) $Re_{x,c} = \frac{\rho u_\infty x_c}{\mu}$

Use $Re_{x,c} \rightarrow$ to find x_c .

$= \frac{(1.127)(2.0)(0.36)}{1.918 \times 10^{-5}} \frac{(\text{kg/m}^3)(\text{m/s})(\text{m})}{\text{kg/(m·s)}}$

$Re_{x,c} = 42306.57$

$< 500,000$

laminar

(b)

$C_{f,x} = \frac{\tau_{s,x}}{\rho u_\infty^2/2} = 0.664 Re_x^{-1/2}$
 $= 0.003228$
 $= 0.10626$

$\tau_s = 0.332 \mu_\infty \sqrt{\rho \mu u_\infty}$
 $= 0.005988280$

$Re_x = ?$

(c)

$Nu_x = 0.332 Re_x^{1/2} Pr^{1/3}$
 $= 0.332 (42306.57)^{1/2} (0.7255)^{1/3}$

$Nu_x = 61.36$

(d)

$Nu_x = \frac{\bar{h}_x x}{k}$

$\bar{h}_x = \frac{k}{x} 0.664 Re_x^{1/2} Pr^{1/3}$

$= \frac{0.02662}{0.5} 0.664 (42306.57)^{1/2} (0.7255)^{1/3}$

$= 6.533$

(e)

$h =$ flow over sphere

PROBLEM #2 (60 pts.)

In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 35 \text{ W/(m}\cdot\text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 320 \text{ J/(kg}\cdot\text{K)}$. The hot air has a free stream velocity and temperature of 3 m/s and 127°C , respectively. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple to reach 126°C . [Hint: The first step is to calculate the heat transfer coefficient between the free stream and the thermocouple junction.]

44/60

$K = 35 \text{ W/(m}\cdot\text{K)}$

$\rho = 8500 \text{ kg/m}^3$

$c_p = 320 \text{ J/kg}\cdot\text{K}$

$u_\infty = 3 \text{ m/s}$

$T_\infty = 127^\circ\text{C}$

$T_i = 27^\circ\text{C}$

$T_f = 126^\circ\text{C}$



$h = ?$

$Nu_D = 2 + (0.4 Re_D^{1/2} + 0.06 Re_D^{1/3}) Pr^{0.4} \left(\frac{\mu}{\mu_s}\right)^{1/4}$

$(4.26 + 14.07299)$

$2 + (18.336) Pr^{0.4} \left(\frac{208.2 \times 10^{-7}}{230.1 \times 10^{-7}}\right)^{1/4} Re_D = \frac{VD}{\nu} = \frac{(3 \text{ m/s})(0.001 \text{ m})}{26.41 \times 10^{-6} \text{ m}^2/\text{s}} = 113.59$

$Pr \xrightarrow{A.4} 0.690$

$\nu \xrightarrow{A.4} 26.41$

$\mu_s \xrightarrow{A.4} 208.2 \times 10^{-7} \text{ unit?}$

$\mu \xrightarrow{A.4} 230.1 \times 10^{-7} \text{ unit?}$

$\frac{T_i + T_f}{2} = 349.5$

$Nu_D = 17.416$

$\bar{h} = \frac{Nu_D K}{D} = \frac{(17.416)(35 \text{ W/m}\cdot\text{K})}{(0.001 \text{ m})}$

$\bar{h} = 570 \text{ W/m}^2\cdot\text{K}$

$B = \frac{hL}{K} = \frac{h (0.001 \text{ m})}{35 \text{ W/m}\cdot\text{K}}$

lumped if $B < 0.1$

$t_a = \frac{\rho V c_p}{h A_s} \ln \frac{T_i - T_\infty}{T_a - T_\infty}$

not lumped

$F_0 = -\frac{1}{S^2} \ln \left[\frac{1}{C_1} \cdot \frac{T_0 - T_\infty}{T_i - T_\infty} \right]$

$t_w = F_0 r^2 / \alpha$

$Bi = \frac{h_w r_0}{K}$

Table 5.1

73/100

AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #3
December 2, 2021

Green, Gary

Print your name (Last name first)

[Signature]

Signature and Date

Dec. 2 2021

Score

100

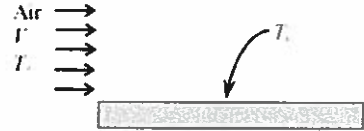
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PROBLEM #1 (40 pts.)

Air at $T_\infty = 20^\circ\text{C}$ and 1 atm flows over a 0.36-m-wide plate at $T_s = 60^\circ\text{C}$ at a velocity of $V = 2.0 \text{ m/s}$. Using air properties of $\rho = 1.127 \text{ kg/m}^3$, $c_p = 1007 \text{ J/(kg}\cdot\text{K)}$, $k = 0.02662 \text{ W/(m}\cdot\text{K)}$, $\mu = 1.918 \times 10^{-5} \text{ kg/(m}\cdot\text{s)}$, and $Pr = 0.7255$, compute the following quantities:

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- Rate of convective heat transfer over $L = 0.5 \text{ m}$ of the plate.



Given: $T_\infty = 20^\circ\text{C}$ 0.36 m wide $V = 2.0 \text{ m/s} = U_\infty$
 1 atm $T_s = 60^\circ\text{C}$

Properties of Air:
 $\rho = 1.127 \text{ kg/m}^3$

$c_p = 1007 \text{ J/kg}\cdot\text{K}$

$k = 0.02662 \text{ W/m}\cdot\text{K}$

$\mu = 1.918 \times 10^{-5} \text{ kg/m}\cdot\text{s}$

$Pr = 0.7255$

$Re_{crit} = \frac{\rho U_\infty x}{\mu} = 5 \times 10^5 = \frac{1.127 \text{ kg/m}^3 (2.0 \text{ m/s})}{1.918 \times 10^{-5}}$
 $= x_c = \frac{Re_{crit} \mu}{\rho U_\infty} = 4.25 \text{ m}$

b.) Local friction at $x = 0.5 \text{ m}$

$C_f = \frac{2}{Re_L} \left. \frac{\partial u}{\partial y} \right|_{y=0}$

$T_F = \frac{T_s - T_\infty}{2} = \frac{60 - 20}{2} = T_F = 20^\circ\text{C}$

$Re_L = \frac{VL}{\nu} = ?$

$Nu = \frac{hL}{k}$

Average

$\dot{Q} = \bar{h} A (T_s - T_\infty)$

$\bar{h} = \frac{1}{L} \int_0^L h dx = \bar{h}_m = \frac{1}{L} \int_0^L h_m dx$

30/60 **PROBLEM #2 (60 pts.)**

In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 35 \text{ W/(m}\cdot\text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 320 \text{ J/(kg}\cdot\text{K)}$. The hot air has a free stream velocity and temperature of 3 m/s and 127°C , respectively. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple to reach 126°C . [Hint: The first step is to calculate the heat transfer coefficient between the free stream and the thermocouple junction.]

Properties

$$k = 35 \text{ W/m}\cdot\text{K}$$

$$\rho = 8500 \text{ kg/m}^3$$

$$c_p = 320 \text{ J/kg}\cdot\text{K}$$

$$T_i = 27^\circ\text{C}$$

$$T = 126^\circ\text{C}$$

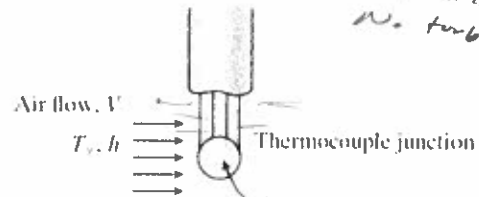
$$V = 3 \text{ m/s}$$

$$T_\infty = 127^\circ\text{C}$$

$$D = 0.001 \text{ m}$$

$$Pr = 0.762$$

at what temperature?



$$Bi = \frac{h \left(\frac{r}{3} \right)}{k}$$

Find h

$$\bar{h} = \frac{Nu_D k}{D}$$

$$Nu_D = 0.3 + \frac{0.62 Re_D^{1/2} Pr^{1/3}}{\left[1 + (0.4/Pr)^{1/4} \right]^{1/4}} \left[1 + \left(\frac{Re_D}{282,000} \right)^{5/8} \right]^{4/5}$$

wrong correlation

$$Nu_D = 0.3 + \frac{0.62 (992)^{1/2} (0.762)^{1/3}}{\left[1 + \left(\frac{0.4}{0.762} \right)^{1/4} \right]^{1/4}} \left[1 + \left(\frac{992}{282,000} \right)^{5/8} \right]^{4/5}$$

$$h = \frac{Nu_D k}{D} = \frac{16.40 (35 \text{ W/m}\cdot\text{K})}{0.001}$$

$$Re_D = \frac{VD}{\nu} = \frac{3(0.001 \text{ m})}{3.25 \times 10^{-6} \text{ m}^2/\text{s}} = 992$$

$$Re_D = 992$$

$$D = \frac{6 h T}{\rho c_p}$$

We have a case of flow over a sphere (not a cylinder)

$$Bi = \frac{h \left(\frac{r}{3} \right)}{k}$$

$$t = \frac{\rho D c_p}{6 h} \ln \Delta T = \frac{\rho D c_p}{6 h} \ln \frac{T_i - T_\infty}{T - T_\infty}$$

10

49/100

BELOW AVERAGE

Prairie View A&M University
Department of Mechanical Engineering
MCEG 3301 Heat Transfer, P01
Fall 2021
Test #3
December 2, 2021

Bellamy Liana

Print your name (Last name first)

[Signature]

12/2/2021

Signature and Date

Score: _____

100

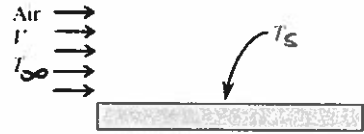
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- Rate of convective heat transfer over $L = 0.5$ m of the plate.



(a) use $Re_{x,c}$ to get x_c ,

Given

$T_\infty = 20^\circ\text{C}$

Air flow = 1 atm

.36

$T_s = 60^\circ\text{C}$

$V = 2.0$ m/s

$\rho = 1.127$ kg/m³

$c_p = 1007$ J/kg·K

$k = 0.02662$

$\mu = 1.918 \times 10^{-5}$ kg/m·s

a) $Re_x = \frac{\rho U_\infty x}{\mu} = \frac{1.127 \frac{\text{kg}}{\text{m}^3} (2.0 \text{ m/s}) (.36 \text{ m})}{1.918 \times 10^{-5} \frac{\text{kg}}{\text{m} \cdot \text{s}}}$

$= 211.53 \text{ m}^2$

b) $x = .5 \text{ m}$

$C_f = \frac{T_{s,x}}{\rho \frac{V^2}{2}} = \frac{60 (.5 \text{ m})}{1.127 \frac{\text{kg}}{\text{m}^3} \frac{2^2}{2}} = 26.619 \frac{\text{kg}}{\text{m}^2 \cdot \text{s}}$

c) $x = .5 \text{ m}$

$Nu = .332 Re^{1/2} Pr^{1/3}$
 $Re_L = \frac{U_\infty L}{\nu} = \frac{2.0 (.5)}{1.918 \times 10^{-5}} = 52140$
 $Nu = .332 (52140)^{1/2} (.7255)^{1/3} = 4.339$

d) $L = .5 \text{ m}$

$h = \frac{Nu k}{L} = \frac{4.339 (0.02662 \frac{\text{W}}{\text{m} \cdot \text{K}})}{.5} = 0.0298 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}$

$= 1.0016 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}$

Pr is given!

$= .7256 \frac{\text{J}}{\text{W} \cdot \text{s}}$

$Nu_L = .664 Re_L^{1/2} Pr^{1/3}$
 $= .664 (52140)^{1/2} (.7255)^{1/3}$

e) $L = .5$
 $q = h A_s (T_s - T_\infty) = ?$

2/60

PROBLEM #2 (60 pts.)

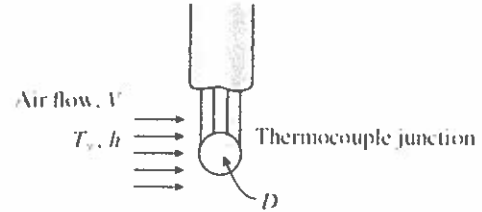
In an experiment, the temperature of a hot air stream is to be measured by a thermocouple whose junction can be approximated as a 1-mm-diameter sphere. The properties of the thermocouple junction are $k = 35 \text{ W/(m}\cdot\text{K)}$, $\rho = 8500 \text{ kg/m}^3$, and $c_p = 320 \text{ J/(kg}\cdot\text{K)}$. The hot air has a free stream velocity and temperature of 3 m/s and 127°C , respectively. If the initial temperature of the thermocouple junction is 27°C , determine how long it will take for the thermocouple to reach 126°C . [Hint: The first step is to calculate the heat transfer coefficient between the free stream and the thermocouple junction.]

Given
 $d = 1 \text{ mm}$
 $k = 35 \frac{\text{W}}{\text{m}\cdot\text{K}}$
 $\rho = 8500 \text{ kg/m}^3$
 $c_p = 320 \frac{\text{J}}{\text{kg}\cdot\text{K}}$

Thermo couple

$$T_i = 27^\circ\text{C}$$

$$T_\infty = 126^\circ\text{C}$$

Find: $t(\text{time})$ 

P.394

Equations

$$q'' = h(T_\infty - T_s)$$

$V = 3 \text{ m/s}$
 $T_s = 127^\circ$ } Free stream

2

27/100

**Heat Transfer
FALL 2021**

HOMEWORK 1

TOPICS COVERED IN HOMEWORK 1

1. Newton's law of cooling
2. Radiation exchange of a surface with the surroundings
3. Surface energy balance
4. Surface radiation and irradiation

**Heat Transfer
FALL 2021**

HOMEWORK 1 QUESTIONS

MCEG 3301 Heat Transfer, by Dr. Xu
Homework

• **I.**

1.47 Consider a carton of milk that is refrigerated at a temperature of $T_m = 5^\circ\text{C}$. The kitchen temperature on a hot summer day is $T_\infty = 30^\circ\text{C}$. If the four sides of the carton are of height and width $L = 200\text{ mm}$ and $w = 100\text{ mm}$, respectively, determine the heat transferred to the milk carton as it sits on the kitchen counter for durations of $t = 10\text{ s}$, 60 s , and 300 s before it is returned to the refrigerator. The convection coefficient associated with natural convection on the sides of the carton is $h = 10\text{ W/m}^2\cdot\text{K}$. The surface emissivity is 0.90 . Assume the milk carton temperature remains at 5°C during the process. Your parents have taught you the importance of refrigerating certain foods from the food safety perspective. Comment on the importance of quickly returning the milk carton to the refrigerator from an energy conservation point of view.

• **II.**

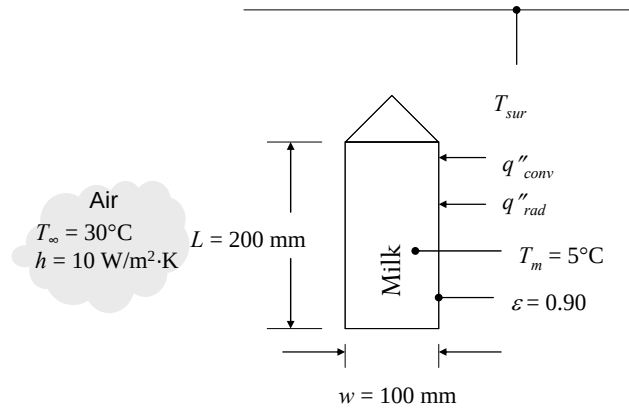
1.50 The emissivity of galvanized steel sheet, a common roofing material, is $\varepsilon = 0.13$ at temperatures around 300 K , while its absorptivity for solar irradiation is $\alpha_s = 0.65$. Would the neighborhood cat be comfortable walking on a roof constructed of the material on a day when $G_s = 750\text{ W/m}^2$, $T_\infty = 16^\circ\text{C}$, and $h = 7\text{ W/m}^2\cdot\text{K}$? Assume the bottom surface of the steel is insulated.

PROBLEM 1.47

KNOWN: Dimensions of a milk carton. Temperatures of milk carton and surrounding air. Convection heat transfer coefficient and surface emissivity.

FIND: Heat transferred to milk carton for durations of 10, 60, and 300 s.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible heat transfer from bottom surface of milk carton and from top surface since it is not in contact with cold milk, (2) Radiation is to large surroundings at the air temperature.

ANALYSIS: The area of the four sides is $A = 4L \times w = 4(0.2 \text{ m} \times 0.1 \text{ m}) = 0.08 \text{ m}^2$. Thus,

$$\begin{aligned} q &= (q_{\text{conv}} + q_{\text{rad}}) = hA(T_{\infty} - T_s) + \varepsilon\sigma A(T_{\text{sur}}^4 - T_s^4) \\ &= 10 \text{ W/m}^2 \cdot \text{K} \times 0.08 \text{ m}^2 (30^\circ\text{C} - 5^\circ\text{C}) + 0.90 \times 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 \times 0.08 \text{ m}^2 ((303 \text{ K})^4 - (278 \text{ K})^4) \\ &= 20.0 \text{ W} + 10.0 \text{ W} = 30.0 \text{ W} \end{aligned}$$

For a duration of 10 s,

$$Q = q\Delta t = 30.0 \text{ W} \times 10 \text{ s} = 300 \text{ J}$$

Similarly, $Q = 1800 \text{ J}$ and 9000 J for durations of 60 and 300 s, respectively.

COMMENTS: (1) The predicted heat transfer rates do not account for the fact that the milk temperature increases with time. If the increase in milk temperature were accounted for, the values of Q would be less than calculated. (2) If the coefficient of performance of the refrigerator is 2, $\text{COP} = Q/W = 2$, then the required work input to re-cool the milk after leaving it in the kitchen for 300 s is 4500 J. At an electricity price of \$0.18/kW·h, this would cost about \$0.0002, which is insignificant. Preventing bacterial growth is a more important reason to return the milk to the refrigerator promptly. (3) The analysis neglects condensation that might occur on the outside of the milk carton. Condensation would increase the rate of heat transfer to the milk significantly, increasing the importance of returning the milk to the refrigerator promptly.



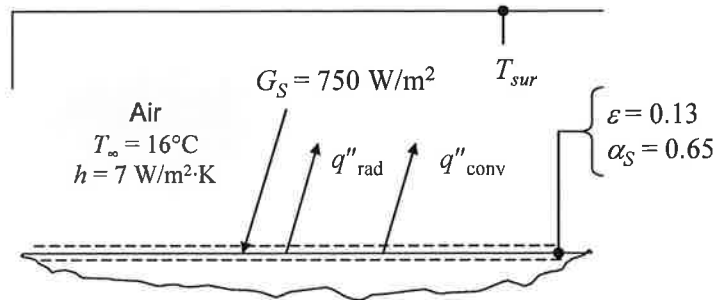
1.50

PROBLEM

KNOWN: Emissivity and solar absorptivity of steel sheet. Solar irradiation, air temperature and convection coefficient.

FIND: Temperature of the steel sheet to determine cat comfort.

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) Bottom surface of steel is insulated, (3) Radiation from the environment can be treated as radiation from large surroundings, with $\alpha = \epsilon$, (4) $T_{\text{sur}} = T_{\infty}$.

ANALYSIS: Performing a control surface energy balance on the top surface of the steel sheet gives (on a per unit area basis)

$$2 \quad \alpha_s G_s - q''_{\text{rad}} - q''_{\text{conv}} = 0$$

$$4 \quad \alpha_s G_s - \epsilon \sigma (T_s^4 - T_{\text{sur}}^4) - h(T_s - T_{\infty}) = 0$$

$$0.65 \times 750 \text{ W/m}^2 - 0.13 \times 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 (T_s^4 - (289 \text{ K})^4) - 7 \text{ W/m}^2 \cdot \text{K} (T_s - 289 \text{ K}) = 0$$

$$2562 \text{ W/m}^2 - 7.37 \times 10^{-9} \text{ W/m}^2 \cdot \text{K}^4 T_s^4 - 7 \text{ W/m}^2 \cdot \text{K} T_s = 0$$

- 4 Solving this equation for T_s using IHT or other software results in $T_s = 350 \text{ K} = 77^\circ\text{C}$. A temperature of 60°C is typically safe to touch without being burned. The steel sheet would be uncomfortably hot or even cause burning. <

COMMENTS: The individual heat flux terms are

$$\alpha_s G_s = 488 \text{ W/m}^2$$

$$q''_{\text{rad}} = \epsilon \sigma (T_s^4 - T_{\text{sur}}^4) = 59 \text{ W/m}^2$$

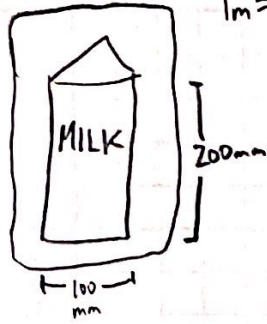
$$q''_{\text{conv}} = h(T_s - T_{\infty}) = 428 \text{ W/m}^2$$

None of these is negligible, although the radiation exchange with the surroundings is smaller than the solar radiation and convection terms.

ABOVE AVERAGE

10/10

1.47

 $T_m = 5^\circ\text{C}$ $T_\infty = 30^\circ\text{C}$ $t = 10\text{ s}$ $t = 60\text{ s}$ $t = 300\text{ s}$ $h = 10\text{ W/m}^2\cdot\text{K}$ $\epsilon = 0.90$ $q_{\text{conv}} + q_{\text{rad}}$

$$q_{\text{conv}} = hA(T_\infty - T_s)$$

$$q_{\text{rad}} = \epsilon A \sigma (T_{\text{surr}}^4 - T_s^4)$$

$$A = (0.2 \times 0.1)4$$

$$A = 0.08\text{ m}^2$$

$$q = \left(10 \frac{\text{W}}{\text{m}^2\cdot\text{K}}\right)(0.08\text{ m}^2)(303\text{ K} - 278\text{ K}) + (0.9)(0.08\text{ m}^2)(5.67 \times 10^{-8} \frac{\text{W}}{\text{m}^2\text{ K}^4})(303\text{ K}^4 - 278\text{ K}^4)$$

20 W

10.03 W

$$q = 30.03\text{ W}$$

For $t = 10\text{ s}$

$$Q = q \Delta t$$

$$= (30.03)(10)$$

$$= \boxed{300\text{ J}}$$

 $t = 60\text{ s}$

$$Q = (30)(60)$$

$$= \boxed{1800\text{ J}}$$

 $t = 300\text{ s}$

$$Q = (30)(300)$$

$$= \boxed{9000\text{ J}}$$

The milk temperature will increase w/ time even though the statement says assume T remains 5°C . So in order to conserve energy the milk must be returned before the temperature increases too much.

10/10

1.50

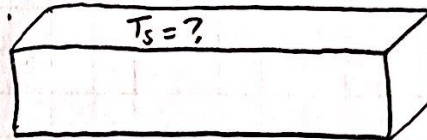
$$\epsilon = 0.13 \quad G_s = 750 \frac{\text{W}}{\text{m}^2}$$

$$T = 300\text{ K}$$

$$d_s = 0.65$$

$$T_\infty = 16^\circ\text{C} = 289\text{ K}$$

$$h = 7 \frac{\text{W}}{\text{m}^2\cdot\text{K}}$$

 $T_\infty = 289\text{ K}$

insulated

$$\alpha Q = q_{\text{conv}} + q_{\text{rad}}$$

$$\alpha Q = hA(T_s - T_\infty) + \epsilon \sigma (T_s^4 - T_{\text{surr}}^4) \leftarrow \text{move to one side w/ algebra, set } = 0$$

$$(0.65)(750 \frac{\text{W}}{\text{m}^2}) = (7 \frac{\text{W}}{\text{m}^2\cdot\text{K}})(T_s - 289\text{ K}) - (0.13)(5.67 \times 10^{-8} \frac{\text{W}}{\text{m}^2\text{ K}^4})(T_s^4 - 289\text{ K}^4) = 0$$

$$487.5 \frac{\text{W}}{\text{m}^2} - 7T_s + 2023 \frac{\text{W}}{\text{m}^2} - 7.37 \times 10^{-9} \frac{\text{W}}{\text{m}^2\text{ K}^4}(T_s^4 - 289\text{ K}^4) = 0$$

$$487.5 \frac{\text{W}}{\text{m}^2} - 7T_s + 2023 \frac{\text{W}}{\text{m}^2} - 7.37 \times 10^{-9} \frac{\text{W}}{\text{m}^2\text{ K}^4}T_s^4 + 51.41 \frac{\text{W}}{\text{m}^2} = 0$$

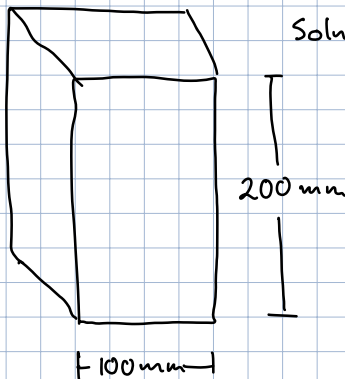
$$2561.91 \frac{\text{W}}{\text{m}^2} - 7T_s - 7.37 \times 10^{-9} T_s^4 = 0$$

$$\boxed{T_s = 350\text{ K}}$$

so it is too hot for the cat to walk on.

20/20

AVERAGE

① given: $T_m = 5^\circ\text{C}$ $T_\infty = 30^\circ\text{C}$ $L = 200\text{mm}$ $W = 100\text{mm}$ $t = 10, 60, 300$ $h = 10 \text{ W/m}^2\cdot\text{K}$ $\epsilon = .9$ assume: $A = 4LW \rightarrow 4 \text{ sides}$
 $= 4(.2)(.1) = .08\text{m}^2$ find: Q 10/10 T_∞, h Soln: $q = q_{\text{conv}} + q_{\text{rad}}$

$$= hA(T_\infty - T_s) + \epsilon \sigma A(T_{\text{sur}}^4 - T_s^4) \quad \checkmark$$

$$= (10 * .08 * (30 - 5)) + (1.9)(5.67E-8)(.08)(303^4 - 278^4)$$

$$= 20 + 10 = 30\text{W} \quad \checkmark$$

$$\text{i. } Q = q\Delta T = 30(10) = 300\text{J} = .3\text{kJ} \quad \checkmark$$

$$\text{ii. } Q = 30(60) = 1800\text{J} = 1.8\text{kJ} \quad \checkmark$$

$$\text{iii. } Q = 30(300) = 9000\text{J} = 9\text{kJ} \quad \checkmark$$

When the milk is out, the heat is transferred onto the carton from the surrounding areas and will cause it to go bad.

6/10

② $T_a = 16^\circ\text{C} = 289\text{K}$ $\rightarrow \epsilon \sigma (T_s^4 - T_{\text{surry}}^4)$

$\alpha G_s = \epsilon \sigma T_s^4 + h (T_s - T_a)$

2+2

$$\Rightarrow .65(750) = (.13)(5.67 \times 10^{-8}) T_s^4 + (7)(T_s - 289)$$

$$\Rightarrow 487.5 = (7.37 \times 10^{-9}) T_s^4 + 7(T_s) - 2023$$

$$\Rightarrow 2510.5 = (7.37 \times 10^{-9}) T_s^4 + 7(T_s)$$

$$T_s = 350 \Rightarrow (7.37 \times 10^{-9})(350\text{K})^4 + (7)(350\text{K}) = 2510.5$$

$T_s = \underline{344}$ unit?

$$(7.37 \times 10^{-9})(344) + (7)(344) = 2510.5$$

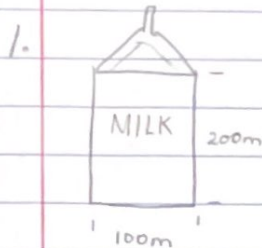
2

$$T_s = (344 - 273) = \underline{71}^\circ\text{C}$$

16/20

BELOW AVERAGE

Homework #1



Given:

$$T_m = 5^{\circ}\text{C}$$

$$h = 10 \text{ W/m}^2 \cdot \text{K}$$

$$T_{\infty} = 30^{\circ}\text{C}$$

$$A = L \times W$$

$$L = 200 \text{ mm}$$

$$\epsilon = 0.90$$

$$W = 100 \text{ mm}$$

$$\begin{aligned} A &= L \times W \\ &= (200 \text{ mm})(100 \text{ mm}) \\ &= (200 \text{ mm}) \end{aligned}$$

$$A = 200 \text{ mm}^2$$

$$4A = 4 \cdot (200 \text{ mm})^2$$

$$4A = 800 \text{ mm}^2$$

$$4A = 0.08 \text{ m}^2$$

$$\dot{q} = \dot{q}_{\text{conv}} + \dot{q}_{\text{rad}}$$

$$= hA(T_{\infty} - T_s) + \epsilon \sigma A(T_{\text{sur}}^4 - T_s^4)$$

$$= (10)(0.08)(30 - 5) + (0.9)(5.67 \times 10^{-8})(0.08)(303^4 - 278^4)$$

$$\dot{q} = 30 \text{ W}$$

$$t = 10^s$$

$$Q = \dot{q} \Delta t$$

$$= (30)(10)$$

$$Q = 300 \text{ J}$$

$$t = 60^s$$

$$Q = (30)(60)$$

$$Q = 1800 \text{ J}$$

$$t = 300^s$$

$$Q = (30)(300)$$

$$Q = 9000 \text{ J}$$

These calculations show

that the milk is not losing

temp. because we neglect

condensation, which would

increase the rate of heat transferred

from milk

4/10

Given:

$$T_R = 50^\circ\text{C} = 323\text{ K}$$

$$T_\infty = 30^\circ\text{C} = 303\text{ K}$$

$$\epsilon = 0.9 \quad \times \text{ wrong info}$$

$$W = 100\text{ mm} = 0.1\text{ m}$$

$$H = 0.2\text{ m}$$

$$h = 10\text{ W/m}^2\cdot\text{K}$$

$$\sigma = 5.64 \times 10^{-8}\text{ W/m}^2\cdot\text{K}^4$$

$$\begin{cases} A = L \times W \\ 4A = (0.1)(0.2) \times 4 \\ 4A = 0.08\text{ m}^2 \end{cases}$$

? not relevant
in this
problem.

4

$$Q_1 = hA(T_\infty - T_s) \quad ?$$

$$= (10)(0.08)(303\text{ K} - 273\text{ K})$$

$$Q_1 = 20\text{ W}$$

T_s is unknown!

$$Q_2 = \epsilon \sigma A (T_{\text{surf}}^4 - T_s^4)$$

$$= (0.9)(5.64 \times 10^{-8})(0.08)(303^4 - 273^4)$$

$$Q_2 = 10.026\text{ W}$$

$$Q_{\text{net}} = Q_1 + Q_2$$

$$= 20 + 10.026$$

$$Q_{\text{net}} = 30.026\text{ W}$$

You are solving a
wrong problem!

14/20

Heat Transfer
FALL 2021

HOMEWORK 2

TOPICS COVERED IN HOMEWORK 2

1. The heat diffusion equation and its applications
2. Boundary conditions for the heat diffusion equation
3. The resistance network in one-dimensional steady-state heat transfer and its practical applications

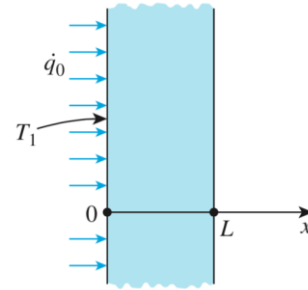
**Heat Transfer
FALL 2021**

HOMEWORK 2 QUESTIONS

MCEG 3301 Heat Transfer, by Dr. Xu
Homework

- **I.**

Consider a large plane wall of thickness $L = 0.4$ m, thermal conductivity $k = 2.8$ W/(m·K), and surface area $A = 15$ m². The left side of the wall at $x = 0$ is subjected to a net heat flux of $q_0 = 550$ W/m² while the temperature at that surface is measured to be $T_1 = 100^\circ\text{C}$. Assuming constant thermal conductivity and no heat generation in the wall, (a) express the differential equation and the boundary conditions for steady one-dimensional heat conduction through the wall; (b) obtain a relation for the variation of temperature in the wall by solving the differential equation; and (c) evaluate the temperature of the right surface of the wall at $x = 0.1$ m and $x = 0.25$ m.



- **II. 3.3a**

3.3 The rear window of an automobile is defogged by passing warm air over its inner surface.

- (a) If the warm air is at $T_{\infty,i} = 40^\circ\text{C}$ and the corresponding convection coefficient is $h_i = 30$ W/m²·K, what are the inner and outer surface temperatures of 4-mm-thick window glass, if the outside ambient air temperature is $T_{\infty,o} = -10^\circ\text{C}$ and the associated convection coefficient is $h_o = 65$ W/m²·K?

I.

a) Steady 1-D conduction through the wall;

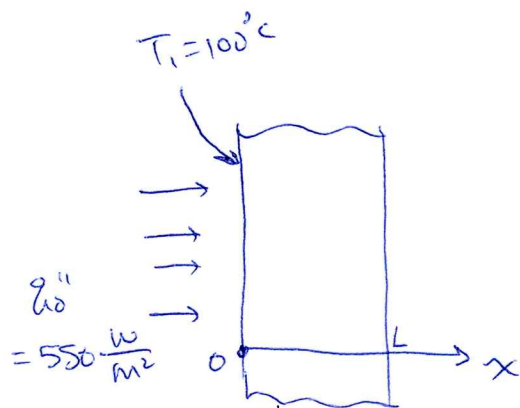
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$$\frac{d^2 T}{dx^2} = 0$$

BCs: $x=0, T(0) = T_1 = 100^\circ\text{C}$

2

$$x=0, -k \frac{dT}{dx} \Big|_{x=0} = q_0'' = 550 \frac{\text{W}}{\text{m}^2}$$



$$L = 0.4 \text{ m}$$

$$k = 2.8 \frac{\text{W}}{\text{m}\cdot\text{K}}$$

$$A = 15 \text{ m}^2$$

b) Integrating the diff. eq. twice with respect to x yields:

$$\frac{dT}{dx} = C_1$$

2

$$T(x) = C_1 x + C_2$$

Apply BCs: $T(0) = C_1 \cdot 0 + C_2 \Rightarrow C_2 = T_1 = 100^\circ\text{C}$

$$-k C_1 = q_0'' \Rightarrow C_1 = -\frac{q_0''}{k} = -\frac{550 \frac{\text{W}}{\text{m}^2}}{2.8 \frac{\text{W}}{\text{m}\cdot\text{K}}} = -196.4 \frac{^\circ\text{C}}{\text{m}}$$

So that $T(x) = C_1 x + C_2 = -\frac{q_0''}{k} x + T_1$

2

$$= -196.4 \frac{^\circ\text{C}}{\text{m}} \cdot x + 100^\circ\text{C}$$

c) $T(0.1) = (-196.4 \frac{^\circ\text{C}}{\text{m}})(0.1 \text{ m}) + 100^\circ\text{C} = 80.36^\circ\text{C} = 353.36 \text{ K}$

2

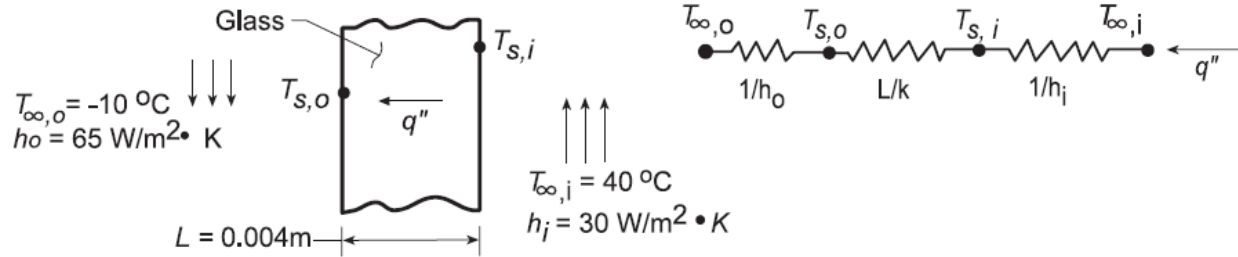
$$T(0.25) = (-196.4 \frac{^\circ\text{C}}{\text{m}})(0.25 \text{ m}) + 100^\circ\text{C} = 50.9^\circ\text{C} = 323.9 \text{ K}$$

PROBLEM 3.3

KNOWN: Temperatures and convection coefficients associated with air at the inner and outer surfaces of a rear window.

FIND: (a) Inner and outer window surface temperatures, $T_{s,i}$ and $T_{s,o}$, and (b) $T_{s,i}$ and $T_{s,o}$ as a function of the outside air temperature $T_{\infty,o}$ and for selected values of outer convection coefficient, h_o .

SCHEMATIC:



ASSUMPTIONS: (1) Steady-state conditions, (2) One-dimensional conduction, (3) Negligible radiation effects, (4) Constant properties.

PROPERTIES: Table A-3, Glass (300 K): $k = 1.4 \text{ W/m}\cdot\text{K}$.

ANALYSIS: (a) The heat flux may be obtained from Eqs. 3.11 and 3.12,

$$4 \quad q'' = \frac{T_{\infty,i} - T_{\infty,o}}{\frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_i}} = \frac{40^\circ\text{C} - (-10^\circ\text{C})}{\frac{1}{65 \text{ W/m}^2 \cdot \text{K}} + \frac{0.004 \text{ m}}{1.4 \text{ W/m}\cdot\text{K}} + \frac{1}{30 \text{ W/m}^2 \cdot \text{K}}}$$

$$2 \quad q'' = \frac{50^\circ\text{C}}{(0.0154 + 0.0029 + 0.0333) \text{ m}^2 \cdot \text{K/W}} = 969 \text{ W/m}^2.$$

Hence, with $q'' = h_i (T_{\infty,i} - T_{s,o})$, the inner surface temperature is

$$2 \quad T_{s,i} = T_{\infty,i} - \frac{q''}{h_i} = 40^\circ\text{C} - \frac{969 \text{ W/m}^2}{30 \text{ W/m}^2 \cdot \text{K}} = 7.7^\circ\text{C} \quad <$$

Similarly for the outer surface temperature with $q'' = h_o (T_{s,o} - T_{\infty,o})$ find

$$2 \quad T_{s,o} = T_{\infty,o} + \frac{q''}{h_o} = -10^\circ\text{C} + \frac{969 \text{ W/m}^2}{65 \text{ W/m}^2 \cdot \text{K}} = 4.9^\circ\text{C} \quad <$$

(b) Using the same analysis, $T_{s,i}$ and $T_{s,o}$ have been computed and plotted as a function of the outside air temperature, $T_{\infty,o}$, for outer convection coefficients of $h_o = 2, 65$, and $100 \text{ W/m}^2 \cdot \text{K}$. As expected, $T_{s,i}$ and $T_{s,o}$ are linear with changes in the outside air temperature. The difference between $T_{s,i}$ and $T_{s,o}$ increases with increasing convection coefficient, since the heat flux through the window likewise increases. This difference is larger at lower outside air temperatures for the same reason. Note that with $h_o = 2 \text{ W/m}^2 \cdot \text{K}$, $T_{s,i} - T_{s,o}$, is too small to show on the plot.

Continued ...

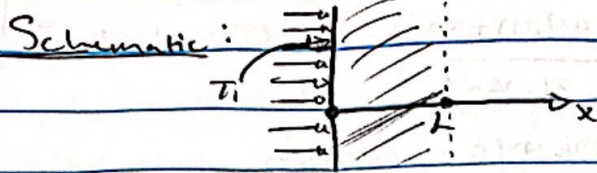
ABOVE AVERAGE

September 29, 2021

Homework 2

Garrett Warren

- I. Given:
- Thickness, $\lambda = 0.4 \text{ m}$
 - Thermal conductivity, $k = 2.8 \frac{\text{W}}{\text{mK}}$
 - Surface Area, $A = 15 \text{ m}^2$
 - $x = 0$ (left side of wall)
 - heat flux, $q_0'' = 550 \frac{\text{W}}{\text{m}^2}$
 - $T_1 = 100^\circ\text{C} = 373 \text{ K}$



- Assumptions:
- Constant Thermal conductivity (k)
 - no heat generation in the wall

Solution: a) Express the differential equation & the boundary conditions for steady 1D heat conduction through the wall.

($x=0$)

$$\frac{d^2 T}{dx^2} = 0$$

$$-k \frac{dT(0)}{dx} = q_0'' \rightarrow q_0'' = 550 \frac{\text{W}}{\text{m}^2}$$

$$T(0) = T_1 = 100^\circ\text{C}$$

b) Obtain a relation for the variation of temperature in the wall by solving the differential equation.

$$\frac{dT}{dx} = C_1$$

$$T(x) = C_1 x + C_2$$

$$\frac{-k C_1}{-k} = \frac{q_0''}{-k} \rightarrow C_1 = -\frac{q_0''}{k}$$

$$x=0 \rightarrow T(0) = C_1(0) + C_2 = T_1$$

$$C_2 = T_1$$

$$T(x) = -\frac{q_0''}{k} x + T_1$$

$$= -\frac{550 \frac{\text{W}}{\text{m}^2}}{2.8 \frac{\text{W}}{\text{mK}}} + 373 \text{ K}$$

$$T(x) = -\frac{550 \frac{\text{W}}{\text{m}^2}}{2.8 \frac{\text{W}}{\text{mK}}} + 373 \text{ K}$$

$$= -196.43x + 373 \text{ K}$$

c) Evaluate The temperature of the right surface of the wall at $x=0.1\text{m}$ and $x=0.25\text{m}$

$$T(x) = -196.43x + 373\text{K}$$

when $x=0.1$: $T(0.1) = -196.43(0.1) + 373\text{K}$

$$T = 353.36\text{K}$$

when $x=0.25$: $T(0.25) = -196.43(0.25) + 373\text{K}$

$$T = 323.89\text{K}$$

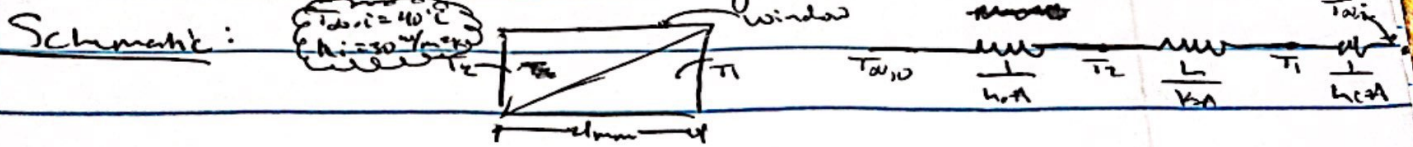
II Given: • subject: automobile • $L = 4\text{mm}$ window

• warm air, $T_{\infty,i} = 40^\circ\text{C}$ • $T_{\infty,o} = -10^\circ\text{C}$

convection

• coefficient, $h_i = 30 \frac{\text{W}}{\text{m}^2\text{K}}$ • $h_o = 65 \frac{\text{W}}{\text{m}^2\text{K}}$

Find: • The Inner & outer surface temperatures of 4-mm-thick window glass



Assumptions: • $k = 1.4 \frac{\text{W}}{\text{mK}}$

Solution: $T_{\infty,o} = -10^\circ\text{C} + 273 = 263\text{K}$

$T_{\infty,i} = 40^\circ\text{C} + 273 = 313\text{K}$

$L = 4\text{mm} \rightarrow 0.004\text{m}$

$$q = \frac{T_{\infty,i} - T_{\infty,o}}{\frac{1}{h_i A} + \frac{L}{k A} + \frac{1}{h_o A}} = \frac{T_2 - T_{\infty,o}}{\frac{1}{h_o A}}$$

$$\frac{T_{\infty,i} - T_{\infty,o}}{\frac{1}{h_i A} + \frac{L}{k A} + \frac{1}{h_o A}} = \frac{T_2 - T_{\infty,o}}{\frac{1}{h_o A}}$$

$$\frac{40 + 10}{\frac{1}{30} + \frac{0.004}{1.4} + \frac{1}{65}} = \frac{T_2 + 10}{\frac{1}{65}}$$

$$969.46 = \frac{T_2 + 10}{\frac{1}{65}}$$

$$14.9 = T_2 + 10$$

$$T_2 = 4.9^\circ\text{C} + 273\text{K} = \boxed{277.9\text{K}}$$

Outer Surface

$$\frac{T_{\infty,i} - T_{\infty,o}}{\frac{1}{h_i A} + \frac{L}{k A} + \frac{1}{h_o A}} = \frac{T_{\infty,i} - T_1}{\frac{1}{h_i A}}$$

$$\frac{40 + 10}{\frac{1}{30} + \frac{0.004}{1.4} + \frac{1}{65}} = \frac{40 - T_1}{\frac{1}{30}}$$

$$\frac{969.46}{30} = 40 - T_1$$

$$32.31 = 40 - T_1$$

$$T_1 = 7.69^\circ\text{C} + 273\text{K} = \boxed{280.69\text{K}}$$

Inner Surface

20/20

AVERAGE

6/10

① known: $L = 0.4m$
 $k = 2.8 \text{ W/m}\cdot\text{K}$
 $A = 15m^2$
 $q'' = 550 \text{ W/m}^2$
 $T_1 = 100^\circ\text{C}$

② Heat conduction through plane wall

$$\frac{d}{dx} \left(k \frac{dT}{dx} \right) = 0 \quad \checkmark$$

$$BC: \text{ i. at } x=0, T=T_1 \quad \checkmark$$

$$\text{ii. at } x=L, T=T_2 \quad \times \text{ wrong BC.}$$

$$\text{③ } q'' = k \frac{dT}{dx} \Rightarrow T(x) = C_1 x + C_2 \quad \checkmark$$

$$BC: \left. \begin{array}{l} x=0, T(0)=T_1 \\ T_1 = C_1(0) + C_2 \end{array} \right\} C_2 = T_1$$

$$\left. \begin{array}{l} x=L, T(L)=T_2 \\ T_2 = C_1(L) + T_1 \end{array} \right\} C_1 = \frac{T_2 - T_1}{L}$$

$$\Rightarrow T(x) = \frac{T_2 - T_1}{L} x + T_1$$

④ $x = 0.1m$ $x = 0.25m$

$$q'' = -kA \frac{T_2 - T_1}{x} \quad \text{⑤ } x=0.1 \Rightarrow 550 - 2.8(15) \left(\frac{T_2 - 100}{0.1} \right)$$

$$T_2 = 98.7^\circ\text{C}$$

$$\text{⑥ } x=0.25 \Rightarrow 550 - 2.8(15) \left(\frac{T_2 - 100}{0.25} \right)$$

$$T_2 = 96.7^\circ\text{C}$$

10/10

② $T_{\infty o} = -10^\circ\text{C}$
 $h_o = 65 \text{ W/m}^2\cdot\text{K}$
 $L = 4mm$
 $T_{\infty i} = 40^\circ\text{C}$
 $h_i = 30 \text{ W/m}^2\cdot\text{K}$

soln: $R = \frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_i}$, $q' = \frac{T_{\infty i} - T_{\infty o}}{R_{total}} \quad \left\} \quad q' = \frac{T_{\infty i} - T_{\infty o}}{\frac{1}{h_o} + \frac{L}{k} + \frac{1}{h_i}} \right.$

$$\Rightarrow q'' = \frac{50^\circ\text{C}}{(.0154 \text{ m}^2\cdot\text{K/W}) + (.0029 \text{ m}^2\cdot\text{K/W}) + (.0333 \text{ m}^2\cdot\text{K/W})}$$

$$= \frac{50^\circ\text{C}}{0.0516 \text{ m}^2\cdot\text{K/W}} = 968 \text{ W/m}^2 \quad \checkmark$$

$$\Rightarrow T_{s,i} = T_{a,i} - \frac{q''}{h_i} = (40^\circ\text{C}) - \frac{(968 \text{ W/m}^2)}{30 \text{ W/m}^2\cdot\text{K}} = \boxed{7.7^\circ\text{C}}$$

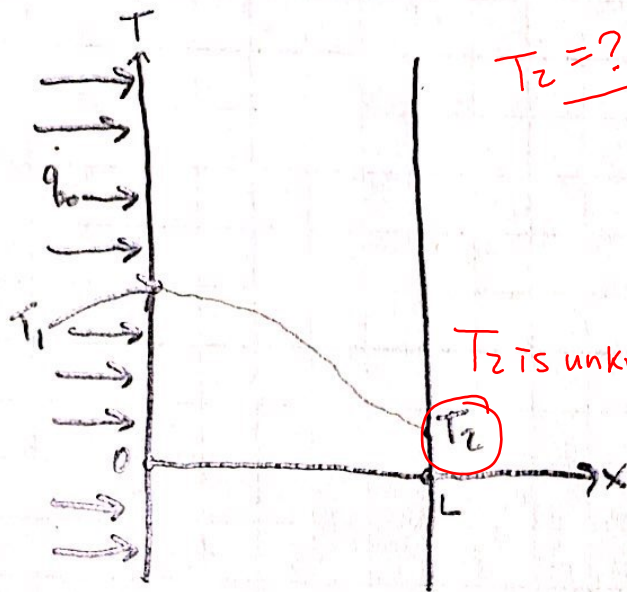
$$T_{s,o} = T_{a,o} + \frac{q''}{h_o} = (-10^\circ\text{C}) + \frac{(968 \text{ W/m}^2)}{65 \text{ W/m}^2\cdot\text{K}} = \boxed{4.9^\circ\text{C}}$$

16/20

BELOW AVERAGE

1) 4/10

$L = 0.4 \text{ m}$
 $k = 2.8 \text{ W/m}\cdot\text{K}$
 $A = 15 \text{ m}^2$
 $q_o'' = 550 \text{ W/m}^2$
 $T_1 = 100^\circ\text{C} = 373 \text{ K}$



$T_2 = ?$

a) $\frac{dT}{dx} = \frac{T_2 - T_1}{x_2 - x_1}$

$q_x'' = -k \frac{dT}{dx}$

$q_x'' = k \cdot \frac{T_1 - T_2}{L}$

$q_x = kA \frac{T_1 - T_2}{L}$

For $x=0$, $T=T_1$
 For $x=L$, $T=T_2$

b) $q_x'' = -k \frac{dT}{dx}$

$\frac{dT}{dx} = C_1$

$T = C_1 x + C_2$

For $x=0$, $T(0) = T_1$

$T_1 = C_1(0) + C_2$

$T_1 = C_2$

For $x=L$, $T(L) = T_2$

$T_2 = C_1(L) + T_1$

$C_1(L) = T_2 - T_1$

$C_1 = \frac{T_2 - T_1}{L}$

$T(x) = \frac{T_2 - T_1}{L} (x) + T_1$

wrong B.C.

c) For $x=0.1 \text{ m}$

$550 = (2.8)(15) \left(\frac{373 - T_2}{0.1} \right)$

$13.09 = \frac{373 - T_2}{0.1}$

$1.309 = 373 - T_2$

$T_2 = 371.69 \text{ K}$
 or
 98.69°C

For $x=0.25 \text{ m}$

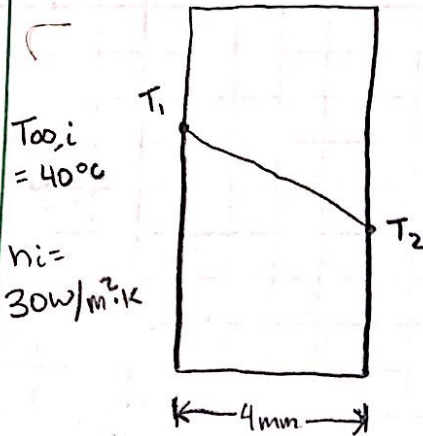
$550 = (2.8)(15) \left(\frac{373 - T_2}{0.25} \right)$

$13.09 = \frac{373 - T_2}{0.25}$

$3.273 = 373 - T_2$

$T_2 = 369.73 \text{ K}$
 or
 96.73°C

6/10



Assume $k = 0.8$

$$T_{\infty, o} = -10^{\circ}\text{C}$$

$$h_o = 65 \text{ W/m}^2 \cdot \text{K}$$

4

$$q = \frac{T_{\infty, i} - T_{\infty, o}}{\sum R_t}$$

$$= \frac{T_{\infty, i} - T_{\infty, o}}{\frac{1}{h_i} + \frac{L}{k} + \frac{1}{h_o}} = \frac{313 - 263}{\frac{1}{30} + \frac{.004}{0.8} + \frac{1}{65}}$$

$$q = 930.79 \text{ W/m}^2$$

wrong k value

$$q = h(T_s - T_{\infty})$$

$$q = h_i(T_{\infty, i} - T_1)$$

$$930.79 = 30(40 - T_1)$$

$$31.63 = 40 - T_1$$

$$T_1 = 8.97^{\circ}\text{C}$$

$$\text{or}$$

$$281.97 \text{ K}$$

$$q = h_o(T_2 - T_{\infty, o})$$

$$930.79 = 65(T_2 - (-10))$$

$$14.32 = T_2 + 10$$

$$T_2 = 4.32^{\circ}\text{C}$$

$$\text{or}$$

$$277.32 \text{ K}$$

10/20

**Heat Transfer
FALL 2021**

HOMEWORK 3

TOPICS COVERED IN HOMEWORK 3

1. The lumped capacitance method in transient conduction

**Heat Transfer
FALL 2021**

HOMEWORK 3 QUESTIONS

MCEG 3301 Heat Transfer, by Dr. Xu
Homework

- **I. 5.6**

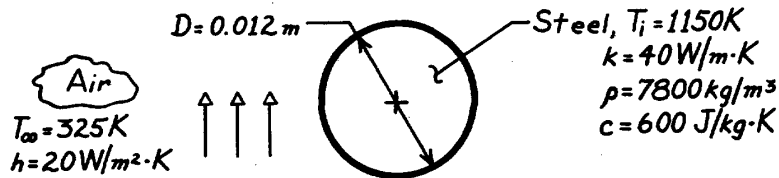
5.6 Steel balls 12 mm in diameter are annealed by heating to 1150 K and then slowly cooling to 400 K in an air environment for which $T_{\infty} = 325$ K and $h = 20$ W/m²·K. Assuming the properties of the steel to be $k = 40$ W/m·K, $\rho = 7800$ kg/m³, and $c = 600$ J/kg·K, estimate the time required for the cooling process.

PROBLEM 5.6

KNOWN: Diameter and initial temperature of steel balls cooling in air.

FIND: Time required to cool to a prescribed temperature.

SCHEMATIC:



ASSUMPTIONS: (1) Negligible radiation effects, (2) Constant properties.

ANALYSIS: Applying Eq. 5.10 to a sphere ($L_c = r_o/3$),

$$4 \quad Bi = \frac{hL_c}{k} = \frac{h(r_o/3)}{k} = \frac{20 \text{ W/m}^2 \cdot \text{K} (0.002 \text{ m})}{40 \text{ W/m} \cdot \text{K}} = 0.001.$$

Hence, the temperature of the steel remains approximately uniform during the cooling process, and the lumped capacitance method may be used. From Eqs. 5.4 and 5.5,

$$4 \quad t = \frac{\rho V c_p}{h A_s} \ln \frac{T_i - T_\infty}{T - T_\infty} = \frac{\rho (\pi D^3/6) c_p}{h \pi D^2} \ln \frac{T_i - T_\infty}{T - T_\infty}$$

$$t = \frac{7800 \text{ kg/m}^3 (0.012 \text{ m}) 600 \text{ J/kg} \cdot \text{K}}{6 \times 20 \text{ W/m}^2 \cdot \text{K}} \ln \frac{1150 - 325}{400 - 325}$$

$$2 \quad t = 1122 \text{ s} = 0.312 \text{ h}$$

<

COMMENTS: Due to the large value of T_i , radiation effects are likely to be significant during the early portion of the transient. The effect is to shorten the cooling time.

ABOVE AVERAGE

Home work #3

1.5.6.

Given

$$d = 12 \text{ mm}$$

$$T_i = 1150 \text{ K}$$

$$T = 400 \text{ K}$$

$$T_\infty = 325 \text{ K}$$

$$h = 20 \text{ W/m}^2 \cdot \text{K}$$

$$k = 40 \text{ W/m} \cdot \text{K}$$

$$\rho = 7800 \text{ kg/m}^3$$

$$C = 600 \text{ J/kg} \cdot \text{K}$$

sphere

Find?

$$t = ?$$

calculate characteristic length

$$L_c = \frac{V}{A_s} = \frac{r_o}{3} = \frac{6 \times 10^{-3}}{3} = 2 \times 10^{-3} \text{ m} = 0.002 \text{ m}$$

- biot number

$$Bi = \frac{h L_c}{k} = \frac{(20 \text{ W/m}^2 \cdot \text{K})(0.002 \text{ m})}{(40 \text{ W/m} \cdot \text{K})} = 0.001 < 0.1$$

We can do lump analysis

- lump analysis

$$\frac{T - T_\infty}{T_i - T_\infty} = \exp \left[- \left(\frac{h A_s}{\rho C_p V} \right) t \right]$$

$$\frac{T - T_\infty}{T_i - T_\infty} = \exp \left[- \left(\frac{h}{\rho L_c C_p} \right) t \right]$$

$$\frac{400 - 325}{1150 - 325} = \exp \left[- \left(\frac{(20 \text{ W/m}^2 \cdot \text{K})}{(7800 \text{ kg/m}^3)(0.002)(600)} \right) t \right]$$

$$\frac{1}{11} = \exp \left(- \frac{1}{468} t \right)$$

$$\ln \left(\frac{1}{11} \right) = - \frac{t}{468}$$

$$t = 2.39789 \times 468$$

$$t = 1122 \text{ sec}$$

or

$$t = 18.7 \text{ min}$$

10/10

AVERAGE

Guy Green
MKEG 3301
Dr. Xu T-TR 8-9:20

1.) Find the estimated time required for the cooling process

Assumptions:

- Steady-State
- No gen
- Use cr. Conv. + Condi.

Spherical (Steel balls)

Given:

$$d = 12 \text{ mm} \approx 1.2 \times 10^{-2}$$

$$T_i = 1150 \text{ K} \quad T = 400 \text{ K}$$

$$T_\infty = 325 \text{ K} \quad h = 20 \text{ W/m}^2 \cdot \text{K}$$

$$k = 40 \text{ W/m} \cdot \text{K} \quad \rho = 7800 \text{ kg/m}^3$$

$$C = 600 \text{ J/kg} \cdot \text{K}$$

⇒ Looking for t

Solution

$$t = \frac{\rho V C}{h A_s} \ln \frac{T_i - T_\infty}{T - T_\infty} = \frac{\rho D L}{6 h} \ln \frac{T_i - T_\infty}{T - T_\infty}$$

$$t = \frac{(7800 \text{ kg/m}^3)(1.2 \times 10^{-2} \text{ m})(600 \text{ J/kg} \cdot \text{K})}{6(20 \text{ W/m}^2 \cdot \text{K})} \ln \left(\frac{1150 \text{ K} - 325 \text{ K}}{400 \text{ K} - 325 \text{ K}} \right) \left[\frac{100}{1} \right]$$

$$t = 1122.2 \text{ s} \approx 18.7 \text{ min}$$

$$Bi = \frac{h(D/6)}{k} = \frac{(20 \text{ W/m}^2 \cdot \text{K})(1.2 \times 10^{-2} \text{ m})}{40 \text{ W/m} \cdot \text{K}}$$

$$Bi = 1 \times 10^{-5} < 0.1$$

wrong Bi

$$D = 1.2 \times 10^{-2} \text{ m}$$

8/10

BELOW AVERAGE

William Robinson

Homework

Given:

$$d = 12 \text{ mm} \rightarrow r = 6 \text{ mm}$$

$$T_i = 1150 \text{ K}$$

$$T_{\infty} = 325 \text{ K}$$

$$h = 20 \text{ W/m}^2 \cdot \text{K}$$

$$k = 40 \text{ W/m} \cdot \text{K}$$

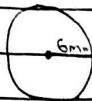
$$\rho = 7800 \text{ kg/m}^3$$

$$c = 600 \text{ J/kg} \cdot \text{K}$$

Schematic:

$$T_{\infty} = 325 \text{ K}$$

$$h = 20 \text{ W/m}^2 \cdot \text{K}$$



$$\rho = 7800 \text{ kg/m}^3$$

$$c = 600 \text{ J/kg} \cdot \text{K}$$

$$\checkmark \ln \left(\frac{T_s - T_{\infty}}{T_i - T_{\infty}} \right) = \left(\frac{hA}{\rho c V} \right) t$$

4

$$\frac{A}{V} = \frac{6}{d} = \frac{6}{12 \times 10^{-3}}$$

↑
need to compute
 B_c to
justify this eqn.

$$\ln \left(\frac{1150 - 325}{400 - 325} \right) = \left(\frac{20 \text{ W/m}^2 \cdot \text{K} \times 6 \text{ mm}}{7800 \text{ kg/m}^3 \times 600 \text{ J/kg} \cdot \text{K} \times 12 \times 10^{-3}} \right) t$$

$$= 2.136 \times 10^{-3} \text{ J}$$

2

$$t = 1122.61 \text{ sec} \checkmark$$

6/10

**Heat Transfer
FALL 2021**

HOMEWORK 4

TOPICS COVERED IN HOMEWORK 4

1. Internal convection: non-circular ducts
2. Free convection: vertical plates
3. Radiation processes and evaluating radiation properties, such as the spectral, hemispherical properties, the total, hemispherical properties, total irradiation, and the absorbed radiant flux
4. Radiation exchange between surfaces

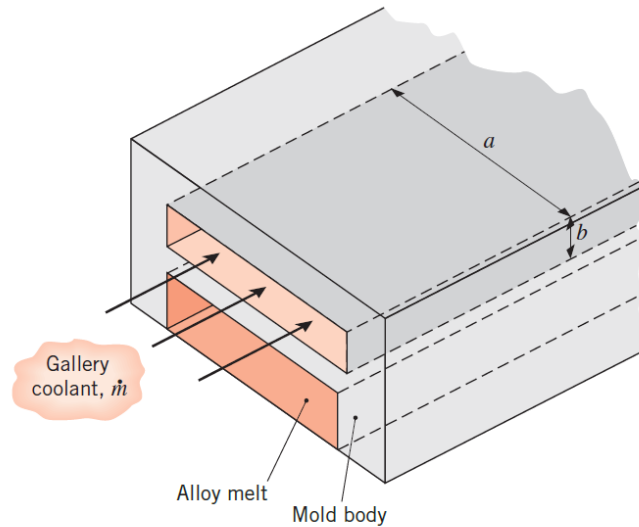
**Heat Transfer
FALL 2021**

HOMEWORK 4 QUESTIONS

MCEG 3301 Heat Transfer, by Dr. Xu
Homework

• **I. 8.89**

- 8.89** A coolant flows through a rectangular channel (*gallery*) within the body of a mold used to form metal injection parts. The gallery dimensions are $a = 90$ mm and $b = 9.5$ mm, and the fluid flow rate is 1.3×10^{-3} m³/s. The coolant temperature is 15°C, and the mold wall is at an approximately uniform temperature of 140°C.



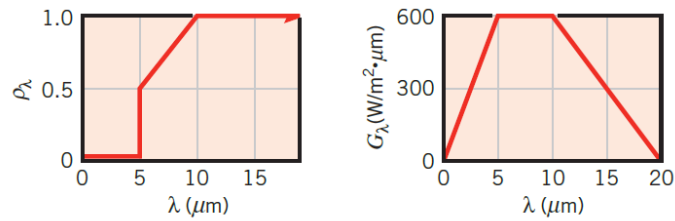
To minimize corrosion damage to the expensive mold, it is customary to use a heat transfer fluid such as ethylene glycol, rather than process water. Compare the convection coefficients of water and ethylene glycol for this application. What is the tradeoff between thermal performance and minimizing corrosion?

• **II. 9.6**

- 9.6** The heat transfer rate due to free convection from a vertical surface, 1 m high and 0.6 m wide, to quiescent air that is 20 K colder than the surface is known. What is the ratio of the heat transfer rate for that situation to the rate corresponding to a vertical surface, 0.6 m high and 1 m wide, when the quiescent air is 20 K warmer than the surface? Neglect heat transfer by radiation and any influence of temperature on the relevant thermophysical properties of air.

• **III. 12.49**

12.49 An opaque surface with the prescribed spectral, hemispherical reflectivity distribution is subjected to the spectral irradiation shown.



- Sketch the spectral, hemispherical absorptivity distribution.
- Determine the total irradiation on the surface.
- Determine the radiant flux that is absorbed by the surface.
- What is the total, hemispherical absorptivity of this surface?

• **IV. 13.55 [Parts (a), (b), and (c) only]**

13.55 Two concentric spheres of diameter $D_1 = 0.8$ m and $D_2 = 1.2$ m are separated by an air space and have surface temperatures of $T_1 = 400$ K and $T_2 = 300$ K.

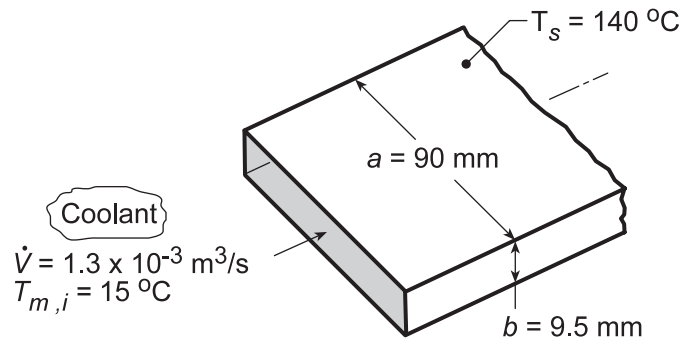
- If the surfaces are black, what is the net rate of radiation exchange between the spheres?
- What is the net rate of radiation exchange between the surfaces if they are diffuse and gray with $\varepsilon_1 = 0.5$ and $\varepsilon_2 = 0.05$?
- What is the net rate of radiation exchange if D_2 is increased to 20 m, with $\varepsilon_2 = 0.05$, $\varepsilon_1 = 0.5$, and $D_1 = 0.8$ m? What error would be introduced by assuming blackbody behavior for the outer surface ($\varepsilon_2 = 1$), with all other conditions remaining the same?

PROBLEM 8.89

KNOWN: Coolant flowing through a rectangular channel (gallery) within the body of a mold.

FIND: Convection coefficient when the coolant is process water or ethylene glycol.

SCHEMATIC:



ASSUMPTIONS: (1) Gallery can be approximated as a rectangular channel with a uniform surface temperature, (2) Fully developed flow conditions.

PROPERTIES: Table A.6, Water ($\bar{T}_m = (140 + 15)^\circ\text{C}/2 = 350\text{ K}$): $\rho = 974\text{ kg/m}^3$, $\mu = 365 \times 10^{-6}\text{ N}\cdot\text{s/m}^2$, $\nu = \mu/\rho = 3.749 \times 10^{-7}\text{ m}^2/\text{s}$, $k = 0.668\text{ W/m}\cdot\text{K}$, $\text{Pr} = 2.29$; Table A.5, Ethylene glycol ($\bar{T}_m = 350\text{ K}$): $\rho = 1079\text{ kg/m}^3$, $\nu = 3.17 \times 10^{-6}\text{ m}^2/\text{s}$, $k = 0.261\text{ W/m}\cdot\text{K}$, $\text{Pr} = 34.6$.

ANALYSIS: The characteristic length of the channel, the hydraulic diameter, Eq. 8.66, is $D_h = 4A_c/P$ where A_c is the cross-sectional flow area and P is the wetted perimeter. For our channel,

$$D_h = \frac{4(a \times b)}{2(a + b)} = \frac{4 \times 0.090\text{ m} \times 0.0095\text{ m}}{2(0.090 + 0.0095)\text{ m}} = 0.0172\text{ m}$$

For the *water* coolant, from the continuity equation, find the Reynolds number to characterize the flow

$$u_m = \frac{\dot{V}}{A_c} = \frac{1.3 \times 10^{-3}\text{ m}^3/\text{s}}{0.090\text{ m} \times 0.0095\text{ m}} = 1.52\text{ m/s}$$

$$\text{Re}_{Dh} = \frac{u_m D_h}{\nu} = \frac{1.52\text{ m/s} \times 0.0172\text{ m}}{3.749 \times 10^{-7}\text{ m}^2/\text{s}} = 69,736$$

Since the flow is turbulent, and assuming fully developed conditions, use the Dittus-Boelter correlation, Eq. 8.60, to estimate the convection coefficient,

$$\text{Nu}_{Dh} = \frac{h D_h}{k} = 0.023 \text{Re}_{Dh}^{0.8} \text{Pr}^{0.4} = 0.023 (69,736)^{0.8} (2.29)^{0.4} = 240$$

$$h_w = \frac{0.668\text{ W/m}\cdot\text{K}}{0.0172\text{ m}} \times 240 = 9326\text{ W/m}^2\cdot\text{K}$$

Repeating the calculations using properties for the *ethylene glycol* coolant, find

$$\text{Re}_{Dh} = 8,247 \quad \text{Nu}_{Dh} = 128 \quad h_{eg} = 1957\text{ W/m}^2\cdot\text{K}$$

Continued...

PROBLEM 8.89 (Cont.)

COMMENTS: (1) The convection coefficient for the *water* coolant is more than 4 times greater than that with the *ethylene glycol* coolant. The corrosion protection afforded by the latter coolant greatly compromises the thermal performance of the gallery. In such situations, it is useful to explore a compromise between corrosion protection and thermal performance by using an aqueous solution of ethylene glycol (50%-50%, for example).

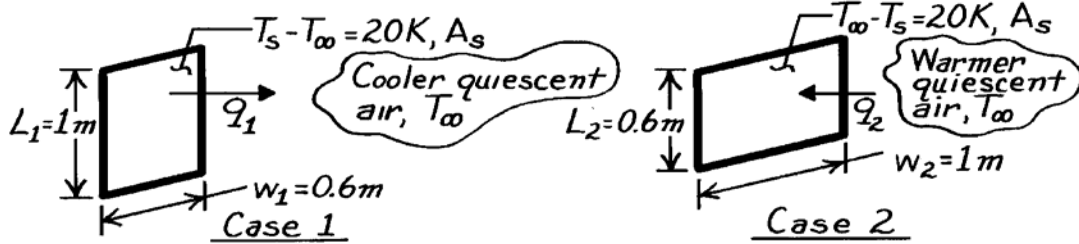
(2) Recognize that for the ethylene glycol coolant calculation the Reynolds number is slightly below the lower limit of applicability of the Dittus-Boelter correlation, and the Gnielinski correlation would be more accurate.

PROBLEM 9.6

KNOWN: Heat transfer rate by convection from a vertical surface, 1m high by 0.6m wide, to quiescent air that is 20K cooler.

FIND: Ratio of the heat transfer rate for the above case to that for a vertical surface that is 0.6m high by 1m wide with quiescent air that is 20K warmer.

SCHEMATIC:



ASSUMPTIONS: (1) Thermophysical properties independent of temperature; evaluate at 300K; (2) Negligible radiation exchange with surroundings, (3) Quiescent ambient air.

2 PROPERTIES: Table A-4, Air (300K, 1 atm): $\nu = 15.89 \times 10^{-6} \text{ m}^2/\text{s}$, $\alpha = 22.5 \times 10^{-6} \text{ m}^2/\text{s}$.

ANALYSIS: The rate equation for convection between the plates and quiescent air is

$$q = \bar{h}_L A_S \Delta T \quad (1)$$

where ΔT is either $(T_s - T_\infty)$ or $(T_\infty - T_s)$; for both cases, $A_S = Lw$. The desired heat transfer ratio is then

$$\frac{q_1}{q_2} = \frac{\bar{h}_{L1}}{\bar{h}_{L2}}. \quad (2)$$

To determine the dependence of $\bar{h}_L$ on geometry, first calculate the Rayleigh number,

$$\text{Ra}_L = g \beta \Delta T L^3 / \nu \alpha \quad (3)$$

and substituting property values at 300K, find,

$$\text{Case 1: } \text{Ra}_{L1} = 9.8 \text{ m/s}^2 (1/300\text{K}) 20\text{K} (1\text{m})^3 / 15.89 \times 10^{-6} \text{ m}^2/\text{s} \times 22.5 \times 10^{-6} \text{ m}^2/\text{s} = 1.82 \times 10^9$$

$$\text{Case 2: } \text{Ra}_{L2} = \text{Ra}_{L1} (L_2/L_1)^3 = 1.82 \times 10^9 (0.6\text{m}/1.0\text{m})^3 = 3.94 \times 10^8.$$

Hence, Case 1 is turbulent and Case 2 is laminar. Using the correlation of Eq. 9.24,

$$\overline{\text{Nu}}_L = \frac{\bar{h}_L L}{k} = C (\text{Ra}_L)^n \quad \bar{h}_L = \frac{k}{L} C \text{Ra}_L^n \quad (4)$$

where for Case 1: $C_1 = 0.10$, $n_1 = 1/3$ and for Case 2: $C_2 = 0.59$, $n_2 = 1/4$. Substituting Eq. (4) into the ratio of Eq. (2) with numerical values, find

$$\frac{q_1}{q_2} = \frac{(C_1/L_1) \text{Ra}_{L1}^{n_1}}{(C_2/L_2) \text{Ra}_{L2}^{n_2}} = \frac{(0.10/1\text{m}) (1.82 \times 10^9)^{1/3}}{(0.59/0.6\text{m}) (3.94 \times 10^8)^{1/4}} = 0.881 < \quad (5)$$

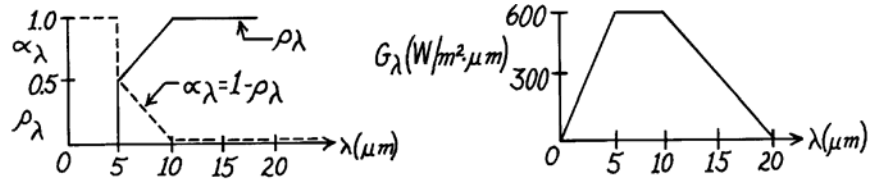
COMMENTS: Is this result to be expected? How do you explain this effect of plate orientation on the heat rates?

PROBLEM 12.49

KNOWN: An opaque surface with prescribed spectral, hemispherical reflectivity distribution is subjected to a prescribed spectral irradiation.

FIND: (a) The spectral, hemispherical absorptivity, (b) Total irradiation, (c) The absorbed radiant flux, and (d) Total, hemispherical absorptivity.

SCHEMATIC:



ASSUMPTIONS: (1) Surface is opaque.

ANALYSIS: (a) The spectral, hemispherical absorptivity, α_λ , for an opaque surface is given by Eq. 12.62,

$$\alpha_\lambda = 1 - \rho_\lambda$$

which is shown as a dashed line on the ρ_λ distribution axes.

(b) The total irradiation, G , follows from Eq. 12.19 which can be integrated by parts,

$$G = \int_0^\infty G_\lambda d\lambda = \int_0^{5\mu\text{m}} G_\lambda d\lambda + \int_{5\mu\text{m}}^{10\mu\text{m}} G_\lambda d\lambda + \int_{10\mu\text{m}}^{20\mu\text{m}} G_\lambda d\lambda$$

$$G = \frac{1}{2} \times 600 \frac{\text{W}}{\text{m}^2 \cdot \mu\text{m}} (5 - 0) \mu\text{m} + 600 \frac{\text{W}}{\text{m}^2 \cdot \mu\text{m}} (10 - 5) \mu\text{m} + \frac{1}{2} \times 600 \frac{\text{W}}{\text{m}^2 \cdot \mu\text{m}} \times (20 - 10) \mu\text{m}$$

$$G = 7500 \text{ W/m}^2.$$

(c) The absorbed irradiation follows from Eqs. 12.51 and 12.52 with the form

$$G_{\text{abs}} = \int_0^\infty \alpha_\lambda G_\lambda d\lambda = \alpha_1 \int_0^{5\mu\text{m}} G_\lambda d\lambda + G_{\lambda,2} \int_{5\mu\text{m}}^{10\mu\text{m}} \alpha_\lambda d\lambda + \alpha_3 \int_{10\mu\text{m}}^{20\mu\text{m}} G_\lambda d\lambda.$$

Noting that $\alpha_1 = 1.0$ for $\lambda = 0 \rightarrow 5 \mu\text{m}$, $G_{\lambda,2} = 600 \text{ W/m}^2 \cdot \mu\text{m}$ for $\lambda = 5 \rightarrow 10 \mu\text{m}$ and $\alpha_3 = 0$ for $\lambda > 10 \mu\text{m}$, find that

$$G_{\text{abs}} = 1.0 \left(0.5 \times 600 \text{ W/m}^2 \cdot \mu\text{m} \right) (5 - 0) \mu\text{m} + 600 \text{ W/m}^2 \cdot \mu\text{m} (0.5 \times 0.5) (10 - 5) \mu\text{m} + 0$$

$$G_{\text{abs}} = 2250 \text{ W/m}^2.$$

(d) The total, hemispherical absorptivity is defined as the fraction of the total irradiation that is absorbed. From Eq. 12.51,

$$\alpha = \frac{G_{\text{abs}}}{G} = \frac{2250 \text{ W/m}^2}{7500 \text{ W/m}^2} = 0.30.$$

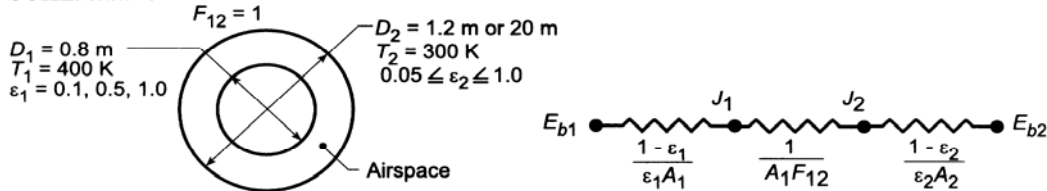
COMMENTS: Recognize that the total, hemispherical absorptivity, $\alpha = 0.3$, is for the given spectral irradiation. For a different G_λ , one would then expect a different value for α .

PROBLEM 13.55

KNOWN: Emissivities, diameters and temperatures of concentric spheres.

FIND: (a) Radiation transfer rate for black surfaces. (b) Radiation transfer rate for diffuse-gray surfaces, (c) Effects of increasing the diameter and assuming blackbody behavior for the outer sphere. (d) Effect of emissivities on net radiation exchange.

SCHEMATIC:



ASSUMPTIONS: (1) Blackbody or diffuse-gray surface behavior.

ANALYSIS: (a) Assuming blackbody behavior, it follows that $q_{ij} = A_i F_{ij} (J_i - J_j)$ where $J_i = \sigma T_i^4$ and $J_j = \sigma T_j^4$. Therefore,

$$2 \quad q_{12} = A_1 F_{12} \sigma (T_1^4 - T_2^4) = \pi (0.8 \text{ m})^2 (1) 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 [(400 \text{ K})^4 - (300 \text{ K})^4] = 1995 \text{ W.} <$$

(b) For diffuse-gray surface behavior, it follows from Eq. 13.26

$$4 \quad q_{12} = \frac{\sigma A_1 (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1 - \epsilon_2}{\epsilon_2} \left(\frac{r_1}{r_2} \right)^2} = \frac{5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 \pi (0.8 \text{ m})^2 [400^4 - 300^4] \text{ K}^4}{\frac{1}{0.5} + \frac{1 - 0.05}{0.05} \left(\frac{0.4}{0.6} \right)^2} = 191 \text{ W.} <$$

(c) With $D_2 = 20 \text{ m}$, it follows that

$$2 \quad q_{12} = \frac{5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 \pi (0.8 \text{ m})^2 [(400 \text{ K})^4 - (300 \text{ K})^4]}{\frac{1}{0.5} + \frac{1 - 0.05}{0.05} \left(\frac{0.4}{10} \right)^2} = 983 \text{ W.} <$$

With $\epsilon_2 = 1$, instead of 0.05, Eq. 13.21 reduces

$$2 \quad q_{12} = \sigma A_1 \epsilon_1 (T_1^4 - T_2^4) = 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 \pi (0.8 \text{ m})^2 0.5 [(400 \text{ K})^4 - (300 \text{ K})^4] = 998 \text{ W.} <$$

Continued ...

ABOVE AVERAGE

Homework #4

Lance Johnson

889

$$T_m = \frac{140 + 15}{2} = 77.5$$

Water

$$T_m = 350 \text{ K}$$

$$\rho = 974 \text{ kg/m}^3$$

$$\mu = 365 \times 10^{-6} \text{ N}\cdot\text{s/m}^2 \quad \dot{V} = 3.749 \times 10^{-7} \text{ m}^3/\text{s}$$

$$k = 0.668 \text{ W/m}\cdot\text{K}$$

$$Pr = 7.29$$

Ethylene glycol

$$T_m = 350 \text{ K}$$

$$\rho = 1079 \text{ kg/m}^3$$

$$\mu = 3.17 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

$$k = 0.261 \text{ W/m}\cdot\text{K}$$

$$Pr = 34.6$$

hydraulic diameters

$$D_h = \frac{4A_c}{P}$$

$$= \frac{4(a \times b)}{8(a+b)}$$

$$= \frac{4 \times 0.09 \times 0.0095}{2(0.09 + 0.0095)} = 0.0172 \text{ m}$$

Water coolant

$$U_m = \frac{\dot{V}}{A_c} = \frac{1.3 \times 10^{-3} \text{ m}^3/\text{s}}{0.090 \times 0.0095} = 1.52 \text{ m/s}$$

$$Re_{pt} = \frac{U_m D_h}{\nu} = \frac{1.52 \times 0.0172}{3.749 \times 10^{-7}} = 69736 > 2300 = \text{turbulent flow}$$

By Dittus-Boelter Correlation

$$Nu_{ph} = \frac{h D_h}{k} = 0.023 Re_{ph}^{0.8} Pr^{0.4}$$

$$= 0.023 \times (69736)^{0.8} (7.29)^{0.4}$$

$$= 240$$

$$h_w = \frac{0.668}{0.0172} \times 240$$

$$= 9396 \text{ W/m}^2\text{K}$$

For ethylene glycol

$$Re_{ph} = 8247$$

$$Nu_{ph} = 128$$

$$h_{eg} = 1957 \text{ W/m}^2\text{K}$$

When water is coolant, the convection coefficient is more than 4 times greater than ethylene glycol coolant.

Ethylene glycol supplies more corrosion protection but it weakens the thermal performance of the gallery

9.6

Can assume 1. Quiescent air is ambient

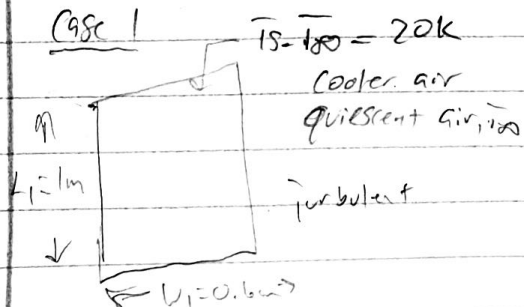
2. Radiation is negligible in exchange with surroundings

3. Thermophysical properties with temp can be neglected

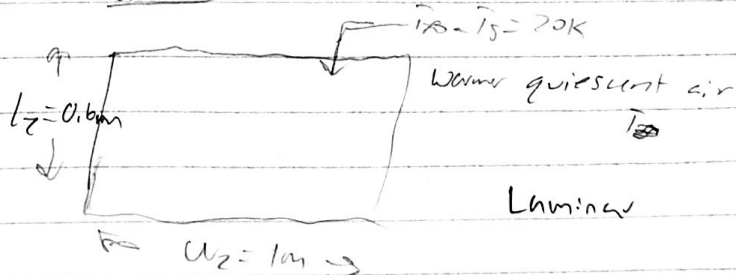
10/10

Air at 300K and 1 atm

Case 1



Case 2



$$A_s = L W$$

Rate equation for convection

$$q_s = h_2 A_s \Delta T$$

$$\Delta T = (T_s - T_\infty)$$

or $(T_\infty - T_s)$

heat transfer ratio

$$\frac{q_1}{q_2} = \frac{h_1}{h_2} \quad -D$$

$$RaL = g \beta \Delta T L^3 / \nu \alpha$$

$$RaL_1 = \frac{9.8 \times \left(\frac{1}{300}\right) \times 20 \times 1}{15.89 \times 10^{-6} \times 22.5 \times 10^{-6}}$$

$$RaL_1 = 1.82 \times 10^9$$

$$RaL_1 = 1.82 \times 10^9$$

$$\begin{aligned} RaL_2 &= RaL_1 \left(\frac{L_2}{L_1} \right)^3 \\ &= 1.82 \times 10^9 \left(\frac{0.6}{1} \right)^3 \\ &= 0.394 \\ &= 3.94 \times 10^8 \end{aligned}$$

$$NuL = \frac{\bar{h}_L L}{k} = C (RaL)^n$$

$$\text{or } \bar{h}_L = \frac{k}{L} C (RaL)^n \quad (2)$$

$$\begin{aligned} \text{Case 1: } g &= 0.10 \\ n_1 &= \frac{1}{3} \end{aligned}$$

$$\begin{aligned} \text{Case 2: } g &= 0.59 \\ n_2 &= \frac{1}{4} \end{aligned}$$

$$\frac{g_1}{g_2} = \frac{(g/L_1)(RaL_1)^{n_1}}{(g_2/L_2)(RaL_2)^{n_2}}$$

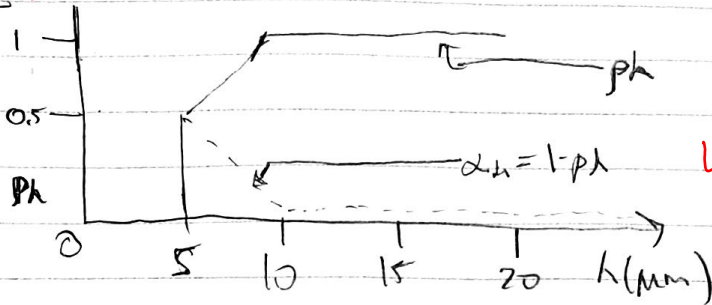
$$\begin{aligned} &= \frac{(0.10/1)(1.82 \times 10^9)^{1/3}}{(0.59/0.6)(3.94 \times 10^8)^{1/4}} \\ &= \frac{0.1 \times 1.2209291}{0.98333} \end{aligned}$$

$$\begin{aligned} &= \frac{0.12209291}{1.385398} = 0.0881 \times 10 \\ &= 0.881 \end{aligned}$$

12.49

10/10

a.)



dotted line indicate $\alpha_h = 1 - \rho_h$ is the hemispherical absorptivity distribution

$$b.) \quad G = \left[\frac{1}{2} \times (5-0) \times 600 \right] + \left[(10-5) \times 600 \right] + \left[\frac{1}{2} \times (20-10) \times 600 \right]$$

$$= 1500 + 3000 + 3000$$

$$= 7500 \text{ W/m}^2$$

$$c.) \quad \text{For } h = 0 \rightarrow 5 \text{ mm}$$

$$\alpha_1 = 1$$

$$\text{For } h = 5 \rightarrow 10 \text{ mm}$$

$$G_{L,2} = 600 \text{ W/m}^2 \cdot \text{mm}$$

$$\text{For } h > 10 \text{ mm}$$

$$\alpha_3 = 0$$

$$G_{abs} = \int_0^{\infty} \alpha_2 G_2 dh$$

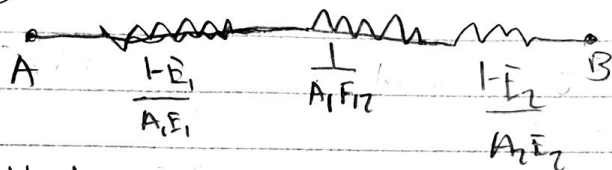
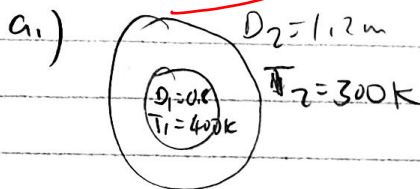
$$= \alpha_1 \int_0^5 G_2 dh + G_{L,2} \int_5^{10} \alpha_2 dh + \alpha_3 \int_{10}^{\infty} G_2 dh$$

$$= \left[1(0.5 \times 600) \times (5-0) \right] + \left[600(0.5 \times 0.5) \times (10-5) \right] + [0]$$

$$= 2250 \text{ W/m}^2$$

$$\begin{aligned} \text{cl.) } \alpha &= \frac{G_{\text{abs}}}{G} \\ &= \frac{7750}{7500} \\ &= 0.3 \end{aligned}$$

13.75 10/10



We know that

$$\begin{aligned} F_{11} + F_{12} &= 1 \\ F_{12} &= 1 \end{aligned}$$

$$\text{So, } q_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{A_1 \epsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1-\epsilon_2}{A_2 \epsilon_2}}$$

$$\begin{aligned} q_{12} &= \sigma(T_1^4 - T_2^4) A_1 \\ &= 5.67 \times 10^{-8} (400^4 - 300^4) \times \pi \times 0.8^2 \\ q_{12} &= 1995.04 \text{ W} \end{aligned}$$

b.) When $\epsilon_1 = 0.5$, $\epsilon_2 = 0.05$

$$\begin{aligned} q_{12} &= \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{A_1 \epsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1-\epsilon_2}{A_2 \epsilon_2}} \\ &= \frac{5.67 \times 10^{-8} (400^4 - 300^4)}{\frac{1-0.5}{(\pi \cdot 0.8)^2 \times 0.5} + \frac{1}{\pi \times 0.8^2} + \frac{1-0.05}{\pi (1.2)^2 \times 0.05}} \\ &= 191 \text{ W} \end{aligned}$$

c) If $\eta_2 = 20\text{m}$ with other parameters same

$$(q_{12})_1 = \frac{5.67 \times 10^{-8} (400^4 - 300^4)}{\frac{1-0.5}{\epsilon_1 (0.8)^2 \times 0.5} + \frac{1}{\epsilon_2 \times 0.8^2} + \frac{1-0.05}{\epsilon_1 (70)^2 \times 0.05}}$$
$$= 982.42 \text{ W}$$

$$\epsilon_2 = 1$$

$$(q_{12})_2 = \frac{5.67 \times 10^{-8} (400^4 - 300^4)}{\frac{1-0.5}{\epsilon_1 (0.8)^2 \times 0.5} + \frac{1}{\epsilon_1 \times 0.8^2} + \frac{1-1}{\epsilon_1 (70)^2}}$$
$$= 997.52 \text{ W}$$

$$\text{Error} = (q_{12})_2 - (q_{12})_1$$

$$= 15.1 \text{ W}$$

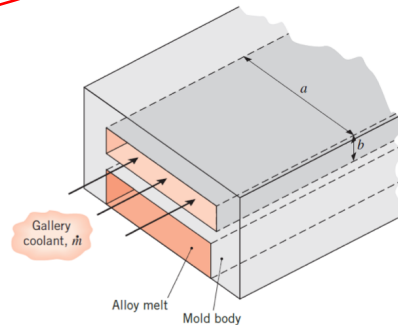
40/40

AVERAGE

• 1.8.89

8.89 A coolant flows through a rectangular channel (gallery) within the body of a mold used to form metal injection parts. The gallery dimensions are $a = 90$ mm and $b = 9.5$ mm, and the fluid flow rate is $1.3 \times 10^{-3} \text{ m}^3/\text{s}$. The coolant temperature is 15°C , and the mold wall is at an approximately uniform temperature of 140°C .

10/10



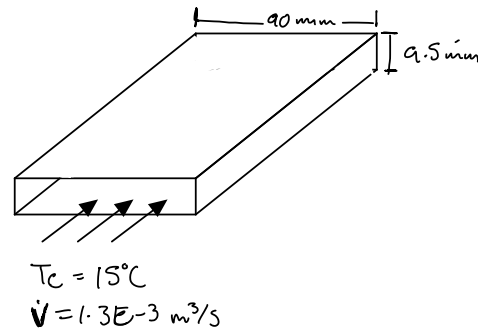
To minimize corrosion damage to the expensive mold, it is customary to use a heat transfer fluid such as ethylene glycol, rather than process water. Compare the convection coefficients of water and ethylene glycol for this application. What is the tradeoff between thermal performance and minimizing corrosion?

Known:

$$\begin{aligned} a &= 90 \text{ mm} \\ b &= 9.5 \text{ mm} \\ \dot{V} &= 1.3 \times 10^{-3} \text{ m}^3/\text{s} \\ T_c &= 15^\circ\text{C} \\ T_w &= 140^\circ\text{C} \end{aligned}$$

Find: Determine difference in thermal performance & minimizing corrosion between ethylene glycol & water.

Schematic:



Plan: - Find Film temp; $\frac{T_c + T_w}{2}$

- Obtain properties of water & ethylene glycol at T_{film} (ρ, k, μ, ν)
- Find cross-sectional area of channel; $A_c = a \cdot b$
- Find Perimeter of channel; $P = 2a + 2b$
- Find hydraulic diameter of channel; $D_h = \frac{4A_c}{P}$
- Find mean velocity of coolant; $u_m = \frac{\dot{V}}{A_c}$
- Find Reynolds # to find type of flow for each; $Re_b = \frac{u_m D_h}{\nu}$
- Depending on Re , Determine h
- Compare h for each

Soln:

$$- \frac{T_c + T_w}{2} = \frac{(15 + 273) + (140 + 273)}{2} \text{ K} = 350.5 \text{ K} \approx 350 \text{ K}$$

Water: at $T_f = 350 \text{ K}$, $\nu = \mu/\rho$

$$\begin{aligned} \rho &= 974 \text{ kg/m}^3 \\ k &= 0.668 \text{ W/m}\cdot\text{K} \\ Pr &= 2.29 \\ \mu &= 365 \times 10^{-6} \text{ N}\cdot\text{s/m}^2 \\ \nu &= 3.749 \times 10^{-7} \text{ m}^2/\text{s} \end{aligned}$$

Ethylene glycol:

$$\begin{aligned} \rho &= 1079 \text{ kg/m}^3 \\ k &= 0.261 \text{ W/m}\cdot\text{K} \\ Pr &= 34.6 \\ \nu &= 3.17 \times 10^{-6} \text{ m}^2/\text{s} \end{aligned}$$

$$A_c = a \cdot b = (90)(9.5) [\text{mm}^2] = 8.55 \times 10^{-4} \text{ m}^2$$

$$P = 2(0.09 + 0.0095) = 0.199 \text{ m}$$

$$D_h = \frac{4A_c}{P} = \frac{(4)(8.55 \times 10^{-4} \text{ m}^2)}{0.199 \text{ m}} = 0.0172 \text{ m}$$

$$u_m = \dot{V}/A_c = 1.3 \times 10^{-3} / 8.55 \times 10^{-4} = 1.52 \text{ m/s}$$

$$Re_{0,w} = \frac{(1.52)(0.0172)}{3.749 \times 10^{-7}} = 69736 > 2300 \text{ (turbulent)}$$

$$Re_{0,g} = \frac{(1.52)(0.0172)}{3.17 \times 10^{-6}} = 8247.3 > 2300 \text{ (turbulent)}$$

$$h_w = \frac{k}{D_h} 0.023 Re_0^{0.8} Pr^{0.4} = \frac{0.668}{0.0172} (0.023)(69736)^{0.8} (2.29)^{0.4} = 9326 \text{ W/m}^2$$

$$h_g = \frac{k}{D_h} 0.023 Re_0^{0.8} Pr^{0.4} = \frac{0.261}{0.0172} (0.023)(8247)^{0.8} (34.6)^{0.4} = 1956.6 \text{ W/m}^2$$

- Convection coefficient for water is greater but EG provides better corrosion protection.

II. 9.6

9.6 The heat transfer rate due to free convection from a vertical surface, 1 m high and 0.6 m wide, to quiescent air that is 20 K colder than the surface is known. What is the ratio of the heat transfer rate for that situation to the rate corresponding to a vertical surface, 0.6 m high and 1 m wide, when the quiescent air is 20 K warmer than the surface? Neglect heat transfer by radiation and any influence of temperature on the relevant thermophysical properties of air.

Assume: at 300K & 1 atm

$$V = 15.09 \times 10^{-6} \text{ m}^3/\text{s}$$

$$\alpha = 22.5 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\text{Soln: } Ra_{L2} = \frac{9.8(1/300)(20)(1)}{(15.09 \times 10^{-6})(22.5 \times 10^{-6})}$$

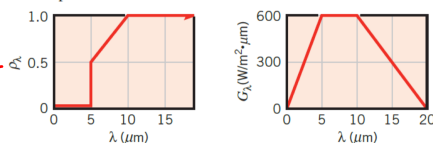
$$= 1.82 \times 10^9$$

$$Ra_{L2} = Ra_{L1} \left(\frac{L_2}{L_1} \right)^3 = 182 \times 10^9 \left(\frac{1}{1} \right)^3 = 3.94 \times 10^8$$

$$\left. \begin{array}{l} C_1 = 10, n_1 = 1/3 \\ C_2 = 0.59 \\ n_2 = 1/4 \end{array} \right\} \frac{q_1}{q_2} = \frac{(C_1/L_1)(Ra_{L1})^{n_1}}{(C_2/L_2)(Ra_{L2})^{n_2}} = \frac{(1/1)(1.82 \times 10^9)^{1/3}}{(0.59/1)(3.94 \times 10^8)^{1/4}} = \boxed{0.881}$$

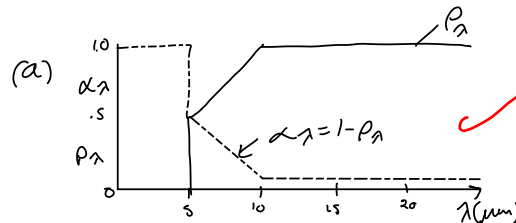
III. 12.49

12.49 An opaque surface with the prescribed spectral, hemispherical reflectivity distribution is subjected to the spectral irradiation shown.



- Sketch the spectral, hemispherical absorptivity distribution.
- Determine the total irradiation on the surface.
- Determine the radiant flux that is absorbed by the surface.
- What is the total, hemispherical absorptivity of this surface?

$$(a) \alpha = \frac{G_{abs}}{G} = \frac{2250}{7500} = \boxed{0.3}$$



$$(b) G = [(0.5)(5-0)(600)] + [(1.0-5)(600)] + [(0.5)(20-10)(600)]$$

$$= 7500 \text{ W/m}^2$$

$$(c) G_{abs} = \int_0^\infty \alpha_\lambda G_\lambda d\lambda$$

$$= \alpha_1 \int_0^5 G_\lambda d\lambda + G_{\lambda,2} \int_5^{10} \alpha_2 d\lambda + \alpha_3 \int_{10}^{20} G_\lambda d\lambda$$

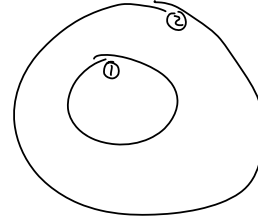
$$= 2250 \text{ W/m}^2$$

• IV. 13.55 [Parts (a), (b), and (c) only]

13.55 Two concentric spheres of diameter $D_1 = 0.8$ m and $D_2 = 1.2$ m are separated by an air space and have surface temperatures of $T_1 = 400$ K and $T_2 = 300$ K.

- If the surfaces are black, what is the net rate of radiation exchange between the spheres?
- What is the net rate of radiation exchange between the surfaces if they are diffuse and gray with $\epsilon_1 = 0.5$ and $\epsilon_2 = 0.05$?
- What is the net rate of radiation exchange if D_2 is increased to 20 m, with $\epsilon_2 = 0.05$, $\epsilon_1 = 0.5$, and $D_1 = 0.8$ m? What error would be introduced by assuming blackbody behavior for the outer surface ($\epsilon_2 = 1$), with all other conditions remaining the same?

Schematic



a) $Q_{1-2} = \sigma A_1 F_{12} (T_1^4 - T_2^4) = (5.67 \times 10^{-8}) (2.01) (1) (400^4 - 300^4) = 1994.4 \text{ W}$

b) $\epsilon_1 = 0.5, \epsilon_2 = 0.05$

$\Rightarrow Q_{1-2} = \frac{1994.4}{\left(\frac{1}{0.5}\right) \left(\frac{2.01}{4.523}\right) \left(\frac{1}{0.05} - 1\right)} = 190.9 \text{ W}$

c) $Q_{1-2} = \frac{1994.4}{\left(\frac{1}{0.5}\right) \left(\frac{2.01}{1256.0}\right) \left(\frac{1}{0.05} - 1\right)} = 982.27 \text{ W}$

what's the expression for this?

OK

with $\epsilon_2 = 1$, $Q_{1-2} = ?$

36/40

BELOW AVERAGE

Alvin Ibeabuchi:

ME6 3301

10/10

I. 8.89

$$\overline{T}_m = \frac{140 + 15}{2} = 147.5^\circ\text{C} \quad \text{Water} \quad \overline{T}_m = 350\text{K} \quad \rho = 974 \text{ kg/m}^3 \quad k = 365 \times 10^{-6} \frac{\text{W}}{\text{m}\cdot\text{K}}$$

$$\overline{T}_m = 350\text{K} \quad \rho = 974 \quad \nu = 3.749 \times 10^{-7} \text{ m}^2/\text{s} \quad k = 0.668 \text{ W/m}\cdot\text{K} \quad \text{Pr} = 2.24$$

Ethylene glycol

$$\overline{T}_m = 350\text{K} \quad \rho = 1075 \text{ kg/m}^3 \quad \nu = 3.17 \times 10^{-6} \text{ m}^2/\text{s}$$

$$k = 0.261 \text{ W/m}\cdot\text{K} \quad \text{Pr} = 34.6$$

$$\text{Hydraulic diameter } D_h = \frac{4A_c}{P} = \frac{4(2 \times 6)}{2(6) + 2(6)} = \frac{4 \times 0.09 \times 0.005}{2(0.009 + 0.005)} = 0.0192 \text{ m}$$

For water coolant, from continuity eq.

$$u_m = \frac{V}{A_c} = \frac{1.3 \times 10^{-3} \text{ m}^3/\text{s}}{0.009 \times 0.005} = 1.52 \text{ m/s}$$

$$\text{Re}_{ph} = \frac{u_m D_h}{\nu} = \frac{1.52 \times 0.0192}{3.749 \times 10^{-7}} = 6936$$

Flow is turbulent, fully developed

Dittus-Boelter Correlation:

$$\text{Nu}_{ph} = \frac{h D_h}{k} = 0.023 \text{ Re}_{ph}^{0.8} \text{ Pr}^{0.4} = 0.023 \times (6936)^{0.8} (2.24)^{0.4} = 246$$

$$h_w = \frac{0.668}{0.0192} \times 246 \quad k_w = 932.6 \text{ W/m}\cdot\text{K}$$

h_w ?

Ethylene glycol

$$\text{Re}_{ph} = 8247, \quad \text{Nu}_{ph} = 128 \quad h_{og} = 152 \text{ W/m}\cdot\text{K}$$

∴ The corrosion protection afforded by latter coolant greatly compromises the thermal performance of the gallery. In such situation, it is worth to explore a compromise between corrosion protection & thermal performance by using aqueous solution of ethylene glycol. 50-50%

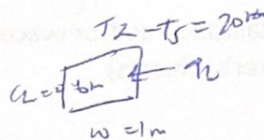
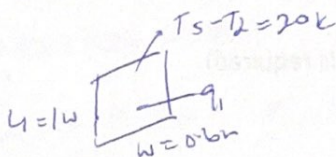
8/10

II 9.6

Vertical surface = 1 m height, 0.6 m wide, air $T_s = 250K$

Assume: Quiescent ambient air - Neglect radiation exchange w/ surroundings
- Thermophysical properties independent of temp.

Properties of air @ 300K & 1 atm: $\nu = 15.89 \times 10^{-6} \text{ m}^2/\text{s}$, $\alpha = 22.5 \times 10^{-6} \text{ m}^2/\text{s}$



$$A_s = LW$$

$$\text{Quiescent air: } q = h_2 A_s \Delta T \quad \Delta T = (T_s - T_\infty) \quad (T_\infty - T_s)$$

$$\text{Heat transfer rates} \Rightarrow q_1/q_2 = h_1/h_2 \quad - (1)$$

Rayleigh #

$$Ra_L = g \beta \Delta T L^3 / \nu \alpha \quad @ 300K \text{ substitute all}$$

$$Ra_{L1} = 9.8 \times (1/300) \times 20 \times 1 \quad \leftarrow ? \text{ what's the expression?}$$

$$15.89 \times 10^{-6} \times 22.5 \times 10^{-6} \Rightarrow Ra_{L1} = 1.82 \times 10^9$$

$$Ra_{L2} = Ra_{L1} \left[\frac{L_1}{L_2} \right]^3 = 1.82 \times 10^9 \left(\frac{0.6}{1} \right)^3 \quad Ra_{L2} = 3.94 \times 10^8$$

$$\frac{h_1}{h_2} = \frac{Ra_{L1}^{1/4}}{Ra_{L2}^{1/4}} \quad \text{or} \quad \frac{h_1}{h_2} = \frac{k}{L} C (Ra_L) \quad \frac{h_1}{h_2} = \frac{k}{L} C (Ra_L) \quad \checkmark$$

For case 1

$$C_1 = 0.10, n_1 = 1/3$$

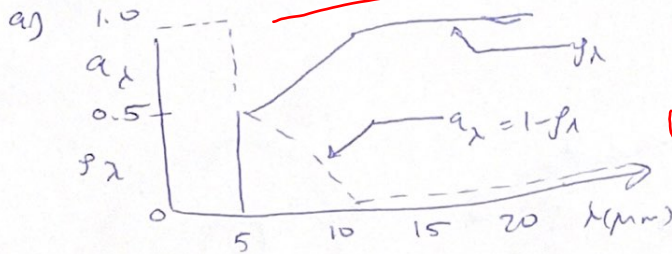
$$\text{case 2} \quad C_2 = 0.59, n_2 = 1/4$$

$$\Rightarrow q_1/q_2 = \frac{(C_1/L_1)(Ra_{L1})^{n_1}}{(C_2/L_2)(Ra_{L2})^{n_2}} = \frac{(0.10)(1.82 \times 10^9)^{1/3}}{(0.59/0.6)(3.94 \times 10^8)^{1/4}}$$

$$\frac{q_1}{q_2} = 0.881 \quad \checkmark$$

III. 12.49

10/10



b)

$$G = \left[\frac{1}{2} \times (5-0) \times 600 \right] + [(10-5) \times 600] + \left[\frac{1}{2} \times (20-10) \times 600 \right]$$

$$= 1500 + 3000 + 3000 = 7500 \text{ W/m}^2$$

c) For $\lambda = 0 \rightarrow 5 \mu\text{m}$ $a_1 = 1$

For $\lambda = 5 \rightarrow 10 \mu\text{m}$ $G_{\lambda,2} = 600 \text{ W/m}^2 \cdot \mu\text{m}$

For $\lambda > 10 \mu\text{m}$ $a_3 = 0$

$$G_{\text{abs}} = \int_0^\infty a_\lambda G_\lambda d\lambda = a_1 \int_0^5 G_{\lambda,1} d\lambda + G_{\lambda,2} \int_5^{10} d\lambda + a_3 \int_{10}^\infty G_\lambda d\lambda$$

$$= [1(0.5 \times 600) \times (5-0)] + [600(0.5 \times 5) \times (10-5)] + [0]$$

$$= 2250 \text{ W/m}^2$$

d)

$$a = \frac{G_{\text{abs}}}{G} = \frac{2250}{7500} = \boxed{0.3}$$

IV. 13.55

5/10

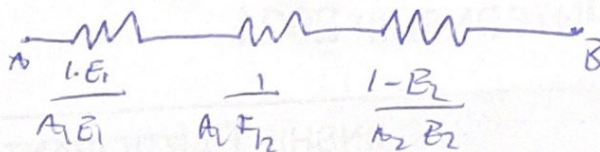
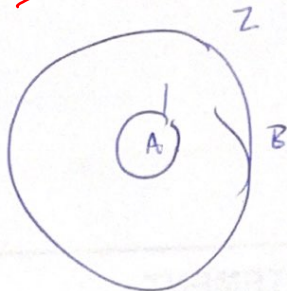
Given

$$D_1 = 6.8$$

$$D_2 = 1.2m$$

$$T_1 = 400K$$

$$T_2 = 300K$$



$$E_{11} + F_{12} = 1 \quad \therefore F_{12} = 1$$

$$q_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-E_1}{A_1 E_1} + \frac{1}{A_1 F_{12}} + \frac{1-E_2}{A_2 E_2}}$$

$$\frac{1-E_1}{A_1 E_1} + \frac{1}{A_1 F_{12}} + \frac{1-E_2}{A_2 E_2}$$

a) when $E_1 = E_2 = 1$ $\therefore q_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{A_1 F_{12}}}$

$$\frac{1-E_1}{A_1 E_1} + \frac{1}{A_1 F_{12}} + \frac{1-E_2}{A_2 E_2}$$

$$q_{12} = \sigma(T_1^4 - T_2^4) A_1$$

$$= 5.67 \times 10^{-8} (400^4 - 300^4) \times \pi \times 0.8^2$$

$$q_{12} = 1895.84 \text{ W}$$

b) when $\epsilon_1 = 0.5$, $\epsilon_2 = 0.05$

$$q_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{A_1 \epsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1-\epsilon_2}{A_2 \epsilon_2}}$$

$$\frac{1-\epsilon_1}{A_1 \epsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1-\epsilon_2}{A_2 \epsilon_2}$$

$$q_{12} = 5.67 \times 10^{-8} (400^4 - 300^4)$$

$$\frac{1-0.5}{(\pi \cdot 0.8)^2 \times 0.5} + \frac{1}{\pi \cdot 0.8^2} + \frac{1-0.05}{\pi (1.2)^2 \times 0.05}$$

$$q_{12} = 1891 \text{ W}$$

c) $(q_{12})_1 = 982.42 \text{ W}$

← How?

How?

$$\text{error} = (q_{12})_2 - (q_{12})_1 = 5.1 \text{ W}$$

when $\epsilon_2 = 1$

$$\frac{33}{40}$$

very difficult to recognize your hand-writing

**HEAT TRANSFER
SEMESTER PROJECT
FALL 2021**

**PROJECT ON
DESIGN OF THE COOLING PROCESS OF METAL PLATES TO
PREVENT THERMAL BURN**

TOPICS COVERED IN PROJECT

1. The applications of Prevention through Design (PtD)
2. Design of the cooling process of metal plates
3. Transit conduction analyses
4. The applications of convection transfer

**HEAT TRANSFER
SEMESTER PROJECT
FALL 2021**

**DETAILED PROJECT PROBLEM STATEMENT
AND REQUIREMENTS**

MCEG 3301 Heat Transfer Course Project

Fall 2021

Design of the Cooling Process of Metal Plates to Prevent Thermal Burn

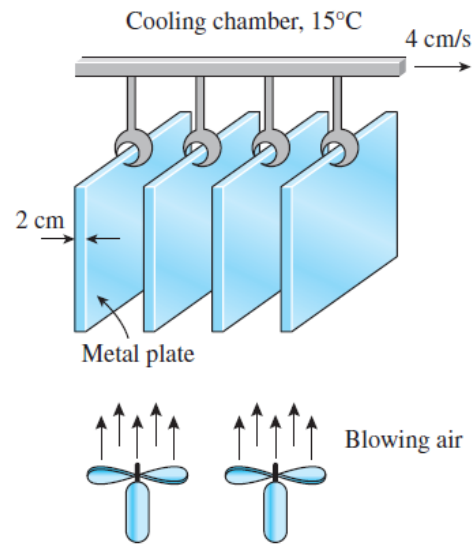
Due: November 18, 2021, at 8:00 am (CT)

Taskstream submission is required.

PROJECT DESCRIPTION

In a plant that processes metal plates, metal plates ($k = 180 \text{ W/m}\cdot\text{K}$, $\rho = 2800 \text{ kg/m}^3$, and $c_p = 885 \text{ J/kg}\cdot\text{K}$) with a thickness of 2 cm exiting an oven are conveyed through a 10-m long cooling chamber at a speed of 4 cm/s (see the figure below). The plates enter the cooling chamber at an initial temperature of 710°C . The air temperature in the cooling chamber is 15°C , and the plates are cooled with blowing air, and the convection heat transfer coefficient is given as a function of the air velocity $h = 33V^{0.8}$, where h is in $\text{W/m}^2\cdot\text{K}$, and V is in m/s.

To prevent any incident of thermal burn, it is necessary to design the cooling process such that the plates exit the cooling chamber at a relatively safe temperature. As a Thermal Engineer, you are asked to design such a cooling process to prevent any thermal burn when handling these metal plates and thus ensure workers' safety in this plant. Complete the following tasks for this design project.



(a) Perform a literature search on relevant temperatures related to thermal burn. Specifically, find out at what temperature the human skin can begin to feel discomfort and pain upon exposure. Moreover, identify relevant temperatures that may cause human skin to receive a first-degree burn injury and cause instantaneous damage to human skin in the current industrial standards. Provide a complete list of references from which you find this information.

(b) Based on your literature review results in (a), select a safe temperature T_{exit} such that the plates exit the cooling chamber at a relatively safe temperature to prevent any incident of thermal burn. Discuss and justify your selection.

(c) Perform relevant heat transfer calculation to study the effect of the air velocity on the temperature of the plates exiting the cooling chamber. Clearly list all the assumptions you use for your calculations. Show all steps in your calculations and type equations directly in Microsoft Word. Your final results may need to be presented in figures and/or tables.

(d) Determine the air velocity and the heat transfer coefficient such that the temperature of the plates exiting the cooling chamber is at or below T_{exit} you selected previously. Once the design is finalized, justify your assumptions made in (c).

REPORT REQUIREMENTS

A detailed report should be submitted for the project, and the following contents should be included:

1. (2%) A “Cover Page” with the course number and name, section number, semester, and your name. See the next page for a sample cover page.
2. (3%) A “Project Description” page that includes the original problem (as provided on the previous page of this document).
3. (5%) An “Abstract” page summarizes your design, calculations, and justifications.
4. (80%) A detailed presentation of all technical components of your design (i.e., Tasks a-d in the “Project Description” section).

SUBMISSION

5. (10%) Upload your report in Microsoft Word format (.doc or .docx), which includes all the contents described in the “Report Requirements” section, to both **Taskstream** and the **“Project Submission” portal on eCourses (Canvas)**. Moreover, submit a hard copy of the report (printed single-sided) by 8 am (CT) on November 18, 2021.

MECHANICAL ENGINEERING DEPARTMENT PRAIRIE VIEW A&M UNIVERSITY



**MCEG 3301-P01
Heat Transfer
Fall 2021**

COVER PAGE

**REPORT ON
“Design of the Cooling Process of Metal Plates
to Prevent Thermal Burn”**

**Submitted by
YOUR NAME**

**On
November 18, 2021**

**To
Dr. Yuhao Xu**

I, _____, declare that this report was entirely written by me. I understand that if any part of this report was copied from previous semester report, or copied from another student, or copied directly from the Internet (or any sources that are not authorized by the instructor), the score for the report will be 0%.

Signature_____ Date_____

MCEG 3301 Fall 2021 Project Solutions

(a) Key findings of the literature review:

The skin reaction to thermal burn is dependent upon the heat transfer rate from the hot medium to the skin. When exposed to temperatures above approximately 43°C, the human skin can begin to feel discomfort and pain; and damage can occur if the exposure is sufficiently long. Exposure to a temperature of about 48°C can cause human skin to receive a first-degree burn injury; and temperature above 70°C can cause instantaneous damage to human skin (*ASTM Standard C1055-03, 2009*).

(b) T_{exit} is selected to be 43°C (or any values below 48°C) based on the results in (a).

(c) Assumptions: 1 The thermal properties of metal plates are constant. 2 Convection heat transfer coefficient is uniform. 3 Heat transfer by radiation is negligible. 4 The Biot number is $Bi < 0.1$ so that the lumped system analysis is applicable (this assumption will be verified).

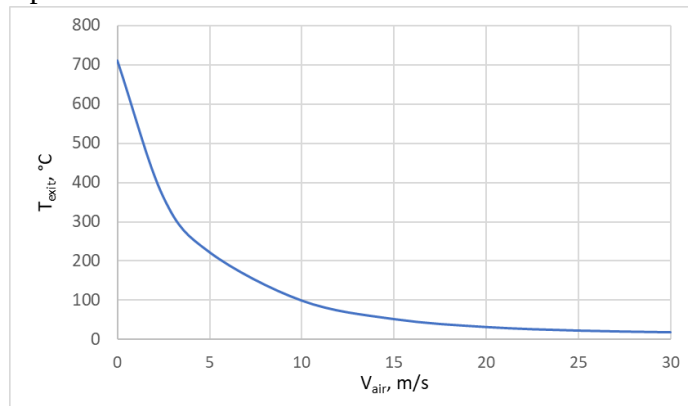
Characteristic length of the plate: $L_c = \frac{V}{A_s} = \frac{zL \cdot A}{zA} = L = \frac{20\text{ mm}}{2} = 10\text{ mm}$

The duration of cooling: $t = \frac{10\text{ m}}{0.04\text{ m/s}} = 250\text{ s}$

For lumped system: $\frac{T(t) - T_\infty}{T_i - T_\infty} = \exp\left[-\frac{hA_s}{\rho C_p V} \cdot t\right]$

or, $T(t) = T_\infty + (T_i - T_\infty) \exp\left[-\frac{hA_s}{\rho C_p V} \cdot t\right]$

Based on this equation, the effect of the air velocity on the temperature of the plates exiting the cooling chamber is plotted below.



(d) With $T(t) = T_{exit} = 43^\circ\text{C}$, V_{air} is computed to be 17.0 m/s.

At this air velocity, $h = 318.3\text{ W/m}^2\cdot\text{K}$.

$Bi = h \cdot L_c / k = 0.0177 < 0.1$. As such, the lumped system approximation is applicable.

ABOVE AVERAGE

**MECHANICAL ENGINEERING DEPARTMENT
PRAIRIE VIEW A&M UNIVERSITY**



**MCEG 3301-P01
Heat Transfer
Fall 2021**

COVER PAGE

**REPORT ON
“Design of the Cooling Process of Metal Plates
to Prevent Thermal Burn”**

**Submitted by
Darin McFadden**

**On
November 18, 2021**

**To
Dr. Yuhao Xu**

I, Darin McFadden, declare that this report was entirely written by me. I understand that if any part of this report was copied from previous semester report, or copied from another student, or copied directly from the Internet (or any sources that are not authorized by the instructor), the score for the report will be 0%.

Signature

Darin McFadden

Date

11/30/21

PROJECT DESCRIPTION

In a plant that processes metal plates, metal plates ($k = 180 \text{ W/m}\cdot\text{K}$, $\rho = 2800 \text{ kg/m}^3$, and $c_p = 885 \text{ J/kg}\cdot\text{K}$) with a thickness of 2 cm exiting an oven are conveyed through a 10-m long cooling chamber at a speed of 4 cm/s (see the figure below). The plates enter the cooling chamber at an initial temperature of 710°C . The air temperature in the cooling chamber is 15°C , and the plates are cooled with blowing air, and the convection heat transfer coefficient is given as a function of the air velocity $h = 33V^{0.8}$, where h is in $\text{W/m}^2\cdot\text{K}$, and V is in m/s.

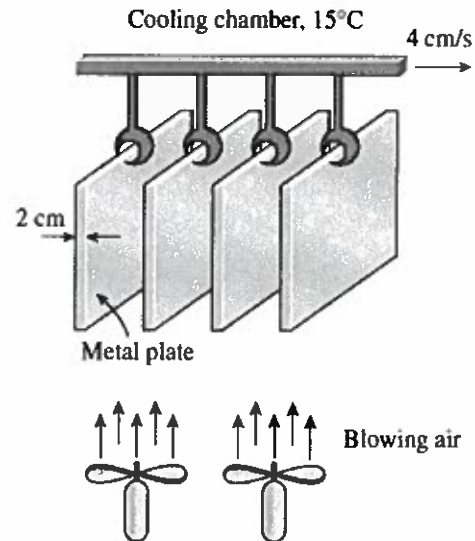
To prevent any incident of thermal burn, it is necessary to design the cooling process such that the plates exit the cooling chamber at a relatively safe temperature. As a Thermal Engineer, you are asked to design such a cooling process to prevent any thermal burn when handling these metal plates and thus ensure workers' safety in this plant. Complete the following tasks for this design project.

(a) Perform a literature search on relevant temperatures related to thermal burn. Specifically, find out at what temperature the human skin can begin to feel discomfort and pain upon exposure. Moreover, identify relevant temperatures that may cause human skin to receive a first-degree burn injury and cause instantaneous damage to human skin in the current industrial standards. Provide a complete list of references from which you find this information.

(b) Based on your literature review results in (a), select a safe temperature T_{exit} such that the plates exit the cooling chamber at a relatively safe temperature to prevent any incident of thermal burn. Discuss and justify your selection.

(c) Perform relevant heat transfer calculation to study the effect of the air velocity on the temperature of the plates exiting the cooling chamber. Clearly list all the assumptions you use for your calculations. Show all steps in your calculations and type equations directly in Microsoft Word. Your final results may need to be presented in figures and/or tables.

(d) Determine the air velocity and the heat transfer coefficient such that the temperature of the plates exiting the cooling chamber is at or below T_{exit} you selected previously. Once the design is finalized, justify your assumptions made in (c).



ABSTRACT

For the design of the cooling chamber, the blowing air velocity was set to $16.77 \frac{m}{s}$. The blowing air velocity as set to that value in order the human skin to not feel discomfort and pain upon exposure, by the metal plates leaving the chamber at 44 degrees Celsius. The Lumped Capacitance Method was assumed and used in the design and then justified by showing that the Biot number was less than .1. From the lumped capacitance method, the heat transfer coefficient was found to be $314.866 \text{ W/m}^2 \cdot \text{K}$.



(a) Literature Review:

According to the scholarly journal by Mark L. Smith, "Pediatric Burns", there are various temperatures at which one can experience a burn injury. Overall, thermal skin damage can start to occur at 44.0 degrees Celsius [1]. For adults, a "full thickness" burn can be experienced after only one second, if the temperature of the surface is 7, which is equivalent to 158.0 degrees Fahrenheit [1]. The degree of a burn is determined by the depth of the burn [1]. For example, first degree burns are of the likes of a sunburn while fourth degree burns, which are still normally grouped with third degree burns, the likes of a burn that extends through the skin possibly hitting muscle and bone [1]. According to the National Institute for Standards of Technology, the human skin can begin to feel discomfort and pain upon exposure at 111 degrees Fahrenheit [3]. The human skin can begin to experience a first degree burn at 118 degrees Fahrenheit [3]. ASTM C1055 (the Standard Guide for Heated System Surface Conditions that Produce Contact Burn Injuries) further enforces the previous articles' statement in regard to surface temperature burns and at which temperature they occur [2]. According to ASTM C1055, presented by MP Divakar, PhD, a Stack Design Automation Technical Editor for Electric Cooling online, the typical person can touch an object or surface at 140 degrees Fahrenheit for five seconds, at most, irreversible burn damage [2]. However, the goal is to not receive any burn damage, reversible or not. The ASTM C1055 table presented below shows that, just under the 140-degree Fahrenheit mark is a 111-degree Fahrenheit mark which indicates that up until that point, no skin damage will be done [2].

ASTM C1055 Table [2]

Sensation	Skin Color	Tissue Temperature		Process	Injury
		deg C	deg F		
Numbness	White	72 68	162	Protein Coagulation	Irreversible
	Mottled Red and White	84 60	140	Thermal Inactivation of Tissue Contents	Possibly Reversible
Maximum Pain	Bright Red	56			Reversible
Severe Pain	Light Red	52			
Threshold Pain		48			
Hot	Flushed	44	111		
		40			
		36			
Warm			93	Normal Metabolism	None
		32			

(b) Safe Temperature T_{exit} :

Based on the information obtained in a literature review, it is necessary to design the cooling process such that the plates exit the cooling chamber at a relatively safe temperature of 44 °C or less to prevent any hazard to workers handling the plates.

(c) Relevant Heat Transfer Calculations

Before the calculations were performed, the lumped system analysis was assumed. The following calculations were performed with the given information in the problem description. First the characteristic length (L_c) had to be determined. According to the textbook, for a symmetrically cooled plane wall of thickness $2L$, L_c would remain equal to the half wall thickness L . Being that the wall is 2 cm, the L_c is 1 cm or .01 m.

It was important to determine the time (t) it takes for a plate to leave the cooling chamber. It was used in the next step in determining the heat transfer coefficient. The time (t) was determined by the equation:

$$\frac{\text{distance of cooling chamber}}{\text{conveyor speed}} = t = \frac{10\text{m}}{.04\text{m/s}} = 250\text{s}$$

Next, Equation 5.6 from the sixth edition Introduction to Heat Transfer book was used to determine that value of the heat transfer coefficient (h). It is shown as follows:

$$\frac{\theta}{\theta_i} = \frac{T - T_\infty}{T_i - T_\infty} = \exp\left[-\frac{hA_s}{pVc}t\right]$$

$$h = -\frac{pcL_c}{t} \ln \left[\frac{T - T_\infty}{T_i - T_\infty} \right]$$

After calculating the heat transfer coefficient, the given air velocity formula from the problem description was derived to determine its value. The calculation to determine the air velocity was performed as shown:

$$V = \sqrt[0.8]{\frac{h}{33}}$$

Also, after calculating the heat transfer coefficient, the Biot number was calculated using Equation 5.10 to see if its resulting value was less than .1 in order to justify the assumption of using the lumped capacitance method. The calculation for the Biot number was performed as shown:

$$Bi = \frac{h * L_c}{k}$$

(d) Final Results and Justification

The time (t) was determined by the equation:

$$\frac{\text{distance of cooling chamber}}{\text{conveyor speed}} = t = \frac{10\text{m}}{.04\text{m/s}} = 250\text{s}$$

The value of the heat transfer coefficient (h) was determined by:

$$\frac{\theta}{\theta_i} = \frac{T - T_{\infty}}{T_i - T_{\infty}} = \exp\left[\left(-\frac{hA_s}{\rho V c}\right)t\right]$$
$$h = -\frac{\rho c L_c}{t} \ln \left[\frac{T - T_{\infty}}{T_i - T_{\infty}} \right] = -\frac{(2800 \frac{\text{kg}}{\text{m}^3})(885 \frac{\text{J}}{\text{kg} \cdot \text{K}})(\ln [\frac{44^{\circ}\text{C} - 15^{\circ}\text{C}}{710^{\circ}\text{C} - 15^{\circ}\text{C}}])}{250\text{s}}$$
$$= 314.866 \text{ W/m}^2 \cdot \text{K}$$

The calculation to determine the air velocity was performed as shown:

$$V = \sqrt[0.8]{\frac{h}{33}} = \sqrt[0.8]{\frac{314.866 \text{ W/m}^2 \cdot \text{K}}{33}} = 16.77 \frac{\text{m}}{\text{s}}$$

The calculation for the Biot number was performed as shown:

$$Bi = \frac{h * L_c}{k} = \frac{314.866 \text{ W/m}^2 \cdot \text{K} * .01\text{m}}{180 \frac{\text{W}}{\text{m}} \cdot \text{K}} = .01749$$

$$.01749 < .1$$

The lumped capacitance method assumed during the design is justified because the Biot number is less than .1.

cc), Texit vs Vair results missing

Submission, 10

11111

85/100

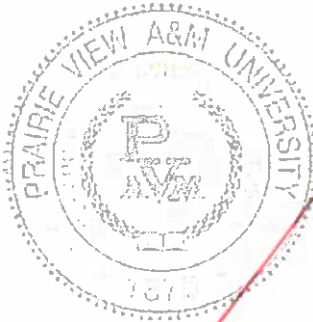
References

- [1] Smith, Mark L., M.D. "Pediatric Burns: Management of Thermal, Electrical, and Chemical Burns and Burn-Like Dermatologic Conditions." *Pediatric Annals*, vol. 29, no. 6, 2000, pp. 367-78. *ProQuest*,
<http://pvamu.idm.oclc.org/login?url=https://www.proquest.com/scholarly-journals/pediatric-burns-management-thermal-electrical/docview/217548001/se-2?accountid=7062>.
- [2] Divakar, MP. "Surface Temperatures of Electronics Products: Appliances vs. Wearables." *Electronics Cooling*, 1 Aug. 2019, <https://www.electronics-cooling.com/2016/09/surface-temperatures-of-electronics-products-appliances-vs-wearables/>.
- [3] "Tips to Be Fit: Different Burns Need Different Treatments." *The Philadelphia Tribune*, 27 Dec. 2018, https://www.phillytrib.com/lifestyle/health/tipstobefit/tips-to-be-fit-different-burns-need-different-treatments/article_5c66fff1-3646-5893-9372-58357d58014c.html.

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AVERAGE

**MECHANICAL ENGINEERING DEPARTMENT
PRAIRIE VIEW A&M UNIVERSITY**



**MCEG 3301-P01
Heat Transfer
Fall 2021**

COVER PAGE

**REPORT ON
“Design of the Cooling Process of Metal Plates
to Prevent Thermal Burn”**

Submitted by

Ethan Phillips

On

November 30, 2021

To

Dr. Yuhao Xu

I, Ethan Phillips, declare that this report was entirely written by me. I understand that if any part of this report was copied from previous semester report, or copied from another student, or copied directly from the Internet (or any sources that are not authorized by the instructor), the score for the report will be 0%.

Signature

Ethan Phillips

Date

11/30/21

MCEG 3301 Heat Transfer Course Project

Fall 2021

Design of the Cooling Process of Metal Plates to Prevent Thermal Burn

Due: November 18, 2021, at 8:00 am (CT)

Taskstream submission is required.

PROJECT DESCRIPTION

In a plant that processes metal plates, metal plates ($k = 180 \text{ W/m}\cdot\text{K}$, $\rho = 2800 \text{ kg/m}^3$, and $c_p = 885 \text{ J/kg}\cdot\text{K}$) with a thickness of 2 cm exiting an oven are conveyed through a 10-m long cooling chamber at a speed of 4 cm/s (see the figure below). The plates enter the cooling chamber at an initial temperature of 710°C . The air temperature in the cooling chamber is 15°C , and the plates are cooled with blowing air, and the convection heat transfer coefficient is given as a function of the air velocity $h = 33V^{0.8}$, where h is in $\text{W/m}^2\cdot\text{K}$, and V is in m/s .

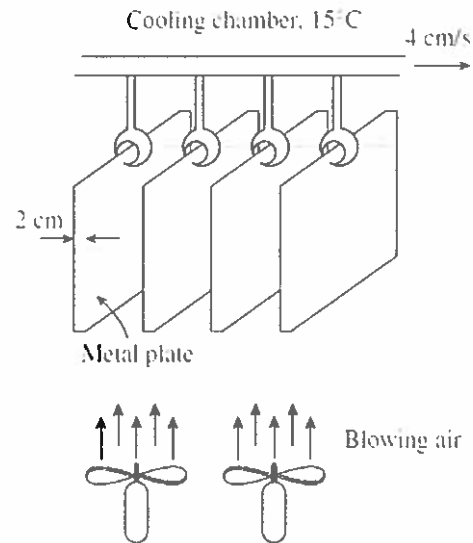
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(a) Perform a literature search on relevant temperatures related to thermal burn. Specifically, find out at what temperature the human skin can begin to feel discomfort and pain upon exposure. Moreover, identify relevant temperatures that may cause human skin to receive a first-degree burn injury and cause instantaneous damage to human skin in the current industrial standards. Provide a complete list of references from which you find this information.

(b) Based on your literature review results in (a), select a safe temperature T_{exit} such that the plates exit the cooling chamber at a relatively safe temperature to prevent any incident of thermal burn. Discuss and justify your selection.

(c) Perform relevant heat transfer calculation to study the effect of the air velocity on the temperature of the plates exiting the cooling chamber. Clearly list all the assumptions you use for your calculations. Show all steps in your calculations and type equations directly in Microsoft Word. Your final results may need to be presented in figures and/or tables.

(d) Determine the air velocity and the heat transfer coefficient such that the temperature of the plates exiting the cooling chamber is at or below T_{exit} you selected previously. Once the design is finalized, justify your assumptions made in (c).



ABSTRACT

The goal of this project is to find a safe departing temperature for metal plates that have been exposed to cooling temperatures. The relevant heat transfer calculation is performed to evaluate the influence of air velocity on the temperature of the plates departing the cooling chamber in order to determine this temperature. The link between air velocity and heat transfer coefficient is indicated in those calculations, such that the temperature of the plates exiting the cooling chamber is at or below the calculated exiting temperature.

Using information from the literature review, comprehensive analysis, and the project instructions' assumptions. Once the supplied values have been understood, inferences can be drawn. The report outlines the steps required to complete the computations and analysis in great detail. The equations technique was used to compute the p value, air velocity, and heat transfer coefficient. In order to attain the desired heat transfer coefficient, a comparison was done to see which value would be preferable for the air velocity. Once this has been found, the rest of the calculations can be performed. The corresponding heat transfer coefficient was estimated when the air velocity factor was determined to be 38.69 m/s.



CHAPTER 1: LITERATURE REVIEW

The skin is made up of three layers and is the biggest organ on the human body. The Epidermis layer is the visible outer layer of the skin. The Dermis layer, which contains hair follicles, blood vessels, and sweat glands, is the intermediate layer of skin. The Subcutis layer is the lowest layer of skin, and it is made up of a network of collagen and fat cells that helps to hold your body's heat while also acting as a shock absorber. Each layer of skin contributes to the protection of the human body from harm.

A burn is an injury to the skin caused by thermal, electrical, chemical, or electromagnetic radiation causing tissue destruction. Thermal burns occur when heat sources raise the temperature of the skin tissues, resulting in the death of tissue cells. Radiation burns are caused by prolonged exposure to the sun's UV rays or other forms of radiation. Strong chemicals that come into contact with the skin or eyes cause chemical burns. Electrical currents cause electrical burns.

According to the degree and depth of the burn, it is categorized into three. Only the epidermis, or outer layer of skin, is affected by a first-degree burn. The burn is reddened, unpleasant, dry, and blister-free. The epidermis and certain dermis layers of the skin are affected by a second-degree burn. The burned area is red, blistering, swollen, and painful. Both the epidermis and the dermis are destroyed in a third-degree burn. The burned area appears to be scorched or white. The nerve endings have been damaged, therefore there is no sensation in the area. According to the National Institute of Standards and Technology, human skin begins to feel uncomfortable at a temperature of 111 °F. At 118 °F, human skin will suffer first-degree burns. A second-degree burn can occur at a temperature of 131 °F.

How about instantaneous burn?

Plates are carried out of a cooling chamber, according to the project description. As a thermal engineer, you are expected to prevent any thermal burns. A thermal burn occurs between 40 and 50 degrees Celsius, according to research from the literature review. The safe temperature for the plate to exit this cooling chamber is 46 degrees Celsius. With this temperature, there will be a guarantee that the plates will depart the cooling chamber at a safe temperature, preventing any thermal burns.

10

+ 5

CHAPTER 2: HEAT TRANSFER ANALYSIS

Given:

$$k = 180 \frac{\text{W}}{\text{m} \cdot \text{K}}$$

$$\rho = 2800 \frac{\text{kg}}{\text{m}^3}$$

$$c_p = 885 \frac{\text{J}}{\text{kg} \cdot \text{K}}$$

$$T_i = 710^\circ \text{C}$$

$$T_\infty = 15^\circ \text{C}$$

$$h = 33 \text{ V}^{\cdot 8} \frac{\text{W}}{\text{m}^2 \cdot \text{K}}$$

$$T_{\text{exit}}: 45^\circ \text{C}, 46^\circ \text{C}, 47^\circ \text{C}$$

$$V_{\text{conveyor speed}} = \frac{4 \text{ cm}}{\text{s}}$$

$$L_{\text{conveyor length}} = 10 \text{ m}$$

Assumptions:

1. Heat transfer by radiation is not to be considered
2. Constant thermal properties of metal plate
3. Convection heat transfer coefficient is not to be considered
4. We are able to perform a Biot's number analysis and the value was less .1. This means that a lumped system analysis is appropriate.

Solution:

Biot's Number

$$Bi = \frac{hL_C}{k} = \frac{33 \text{ V}^{\cdot 8} \cdot .020}{180}$$

Charatistics Length

$$L_C = \frac{V}{A_S} = \frac{2L_t A}{2A} = L_t = 2 \text{ cm}$$

$$L_C = 2 \text{ cm} = 0.02 \text{ m}$$

Time of cooling

$$t = \frac{L_{\text{conveyor length}}}{V_{\text{conveyor speed}}} = \frac{10 \text{ m}}{\frac{4 \text{ cm}}{\text{s}}} = \frac{10 \text{ m}}{.04 \text{ m}} = 250 \text{ sec}$$

Lumped Capacitance Calculations

$$\frac{\rho V c_p}{h A_s} \ln \frac{T_i - T_\infty}{T - T_\infty} = t$$

$$\frac{\rho L c_p}{t} \ln \frac{T_i - T_\infty}{T_{exit} - T_\infty} = h$$

$$(h = 33V^{0.8})$$

$$\frac{\rho L c_p}{t} \ln \frac{T_i - T_\infty}{T_{exit} - T_\infty} = 33V^{0.8}$$

$$\frac{(2800 \text{ kg/m}^3)(0.02 \text{ m})(885 \text{ J/(kg} \cdot \text{K)})}{(250 \text{ s})} \cdot \ln \left(\frac{710 - 15}{T_{exit} - 15} \right) = 33V^{0.8}$$

T @ 45 °C

$$(198.24)(3.1427) = 33V^{0.8}$$

$$V_{a1} = 39.35 \frac{\text{m}}{\text{s}}$$

T @ 46 °C

$$(198.24)(3.1) = 33V^{0.8}$$

$$V_{a2} = 38.69 \frac{\text{m}}{\text{s}}$$

T @ 47 °C

$$(198.24)(3.08) = 33V^{0.8}$$

$$V_{a3} = 38.35 \frac{\text{m}}{\text{s}}$$

→ h = ? → Bi = ?

Table 2.1 Temperature vs Air Velocity Table

T_{exit}	V_a
45°C	39.35 m/s
46°C	38.69 m/s
47°C	38.35 m/s

need to have
more temps.
to get a trend.

CHAPTER 3: CONCLUSION

Air velocity values have an inverse relationship with the exit temperature of the metal plates, as shown in a calculation using lump capacity. When the heat transfer coefficient was determined, the existing temperature values from the literature review proved to be true. The design can withstand the limits and conditions that are imposed on it. Finally, the design, calculations, and analysis were all successful in completing the project.

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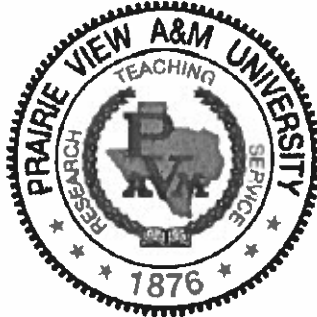
Number refs.

Submission: 10

78/100

BELOW AVERAGE

MECHANICAL ENGINEERING DEPARTMENT PRAIRIE VIEW A&M UNIVERSITY



**MCEG 3301-P01
Heat Transfer
Fall 2021**

COVER PAGE

**REPORT ON
“Design of the Cooling Process of Metal Plates
to Prevent Thermal Burn”**

**Submitted by
Alvin Ibeabuchi**

**On
November 30, 2021**

**To
Dr. Yuhao Xu**

I, Alvin Ibeabuchi, declare that this report was entirely written by me. I understand that if any part of this report was copied from previous semester report, or copied from another student, or copied directly from the Internet (or any sources that are not authorized by the instructor), the score for the report will be 0%.

Signature Alvin Ibeabuchi Date 11/30/21

MCEG 3301 Heat Transfer Course Project
Fall 2021

Design of the Cooling Process of Metal Plates to Prevent Thermal Burn

Due: November 30, 2021, at 8:00 am (CT)

Taskstream submission is required.

PROJECT DESCRIPTION

In a plant that processes metal plates, metal plates ($k = 180 \text{ W/m}\cdot\text{K}$, $\rho = 2800 \text{ kg/m}^3$, and $c_p = 885 \text{ J/kg}\cdot\text{K}$) with a thickness of 2 cm exiting an oven are conveyed through a 10-m long cooling chamber at a speed of 4 cm/s (see the figure below). The plates enter the cooling chamber at an initial temperature of 710°C . The air temperature in the cooling chamber is 15°C , and the plates are cooled with blowing air, and the convection heat transfer coefficient is given as a function of the air velocity $h = 33V^{0.8}$, where h is in $\text{W/m}^2\cdot\text{K}$, and V is in m/s.

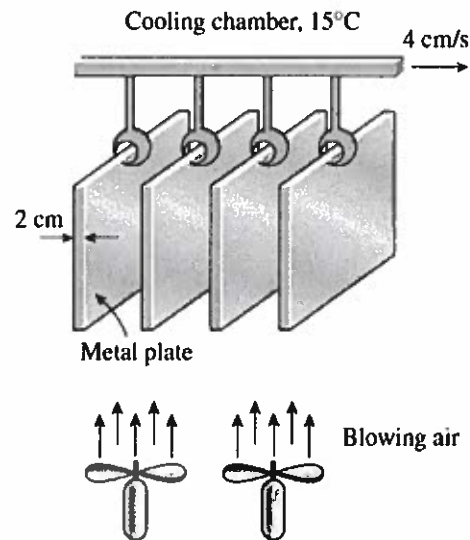
To prevent any incident of thermal burn, it is necessary to design the cooling process such that the plates exit the cooling chamber at a relatively safe temperature. As a Thermal Engineer, you are asked to design such a cooling process to prevent any thermal burn when handling these metal plates and thus ensure workers' safety in this plant. Complete the following tasks for this design project.

(a) Perform a literature search on relevant temperatures related to thermal burn. Specifically, find out at what temperature the human skin can begin to feel discomfort and pain upon exposure. Moreover, identify relevant temperatures that may cause human skin to receive a first-degree burn injury and cause instantaneous damage to human skin in the current industrial standards. Provide a complete list of references from which you find this information.

(b) Based on your literature review results in (a), select a safe temperature T_{exit} such that the plates exit the cooling chamber at a relatively safe temperature to prevent any incident of thermal burn. Discuss and justify your selection.

(c) Perform relevant heat transfer calculation to study the effect of the air velocity on the temperature of the plates exiting the cooling chamber. Clearly list all the assumptions you use for your calculations. Show all steps in your calculations and type equations directly in Microsoft Word. Your final results may need to be presented in figures and/or tables.

(d) Determine the air velocity and the heat transfer coefficient such that the temperature of the plates exiting the cooling chamber is at or below T_{exit} you selected previously. Once the design is finalized, justify your assumptions made in (c).



ABSTRACT



In this project the student must work as a thermal engineer for a plant that processes metal plates. The task was to prevent thermal burn to human skin by designing a cooling process when handling the metal plates to ensure workers' safety in the work place. To determine what is a safe temperature to prevent thermal burn, a literature search was conducted. This search was to include a relevant temperature that may cause a human to receive first degree burns. This was determined to be approximately 118 degrees Fahrenheit which converts to about 47.8 degrees Celsius. A safe temperature exit of 45 degrees Celsius was used to perform the needed calculations. These calculations include heat transfer calculations to study the effect of air velocity on the temperature of plates exiting the cooling chamber and the air velocity and heat transfer coefficient such that the temperature of the plates exiting the cooling chamber is at or below the temperature exit of 45 degrees Celsius.



a.) A literature search was performed to determine the relevant temperature related to thermal burn. The temperature the human skin can begin to feel discomfort and pain upon exposure was stated to be 111 degrees Fahrenheit or 44 degrees Celsius according to the National Institute for Standards and Technology. Moreover, the relevant temperature that may cause human skin to receive a first-degree burn injury and cause instantaneous damage to human skin in the current industrial standards was stated to be at 118 degrees Fahrenheit or 47.8 degrees Celsius.

b.) A safe exit temperature to prevent thermal burn is 45 degrees Celsius. This temperature was selected because it is below the temperature where human skin can receive thermal burns based on the findings from the literature search.

c.) Given Data:

$$k = 180 \text{ W/mK}$$

$$\rho = 2800 \text{ kg/m}^3$$

$$c_p = 880 \text{ J/kgK}$$

$$\text{thickness} = 2\text{cm} = 2 \times 10^{-2} \text{ m}$$

$$\text{distance} = 10\text{m}$$

$$v = 4 \text{ cm/s} = 40\text{mm/s}$$

$$\text{assume } L = 1\text{m}$$

Find value of the Biot Number. If it is less than 0.1 then the lumped system is applicable.

$$Bi = hL_e / k_p$$

$$b = h / \rho h_0 c_p \times L_e$$

Average temperature between T_{si} and T_{st}

$$T_{avg} = (T_{si} + T_{st}) / 2 = (710 + 45) / 2 = 377.5 \text{ degrees C}$$

Properties of air at 1atm

$$\nu = 1.847 \times 10^{-5} \text{ m}^2/\text{s}$$

$$Pr = 0.7210$$

$$k = 0.02772 \text{ W/mK}$$

$$\text{Reynolds Number, } Re = \rho L / \mu$$

$$Nu = (0.037 Re^{0.8} - 871) Pr^{1/3}$$

$$t = 10/40 \times 10^{-3} = 250 \text{ sec} \quad (4 \text{ cm/s} = 40 \text{ mm/s})$$

(c). T_{exit} vs V_{air}
results missing.

Surrounding (infinity) = 15 degrees C

$$T_{\text{avg}} = \frac{1}{2} (T_{\text{si}} + T_{\text{st}})$$

$$T - f = \frac{1}{2} (T_{\text{avg}} + T_{\text{st}})$$

$$K \times (T - f)$$

$$Pr = (T - f)$$

$$Rh_0 = (T - (T_{\text{avg}} + T_{\text{infinity}} / 2)), p = 101.3)$$

d.) Considering the biot number is less than 0.1 lumped heat analysis is applicable.

Characteristic length (L_c) is given by

$$L_c = V/A_s = 2LA / 2A = L = 2 / 2 \times 100 = 0.01 \quad \text{unit?} \quad 4$$

Time constant (t) given by

$$T = hA/pVC_p = h/pL_cC_p$$

$$\Rightarrow T = 33v^{0.8} / 2800 \times 0.01 \times 885 \quad / \text{sec} \text{ ----- (1)}$$

time duration of cooling

$$t = 10m / 4/100 \text{ m/s} = 250\text{sec} \quad \checkmark$$

$$(T_{\text{exit}} - T_{\text{infinity}} / T_i - T_{\text{infinity}}) = e^{-Tt} \quad \checkmark \quad \text{or} \quad 10$$

$$T = 1/t \ln [T_i - T_{\text{infinity}} / T - T_{\text{infinity}}]$$

$$= T = 1/250 \times \ln [710 - 15 / 45 - 15]$$

$$= T = 0.0126 \text{ sec}$$

Putting T in eq (1)

$$0.0126 = 33v^{0.8} / 2800 \times 0.1 \times 885$$

$$V = 16.59 \quad \text{unit?}$$

Heat transfer coefficient (h) given by

$$h = 33v^{0.8}$$

$$= 33 \times 16.59^{0.8}$$

$$h = 312.17 \text{ W/m}^2\text{K}$$

ok

$Bi = ?$

9

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Submission: 8
(no Taskstream Submission
by Jan on 11/30/21)

65/100