Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

Learning package

Heat Transfer

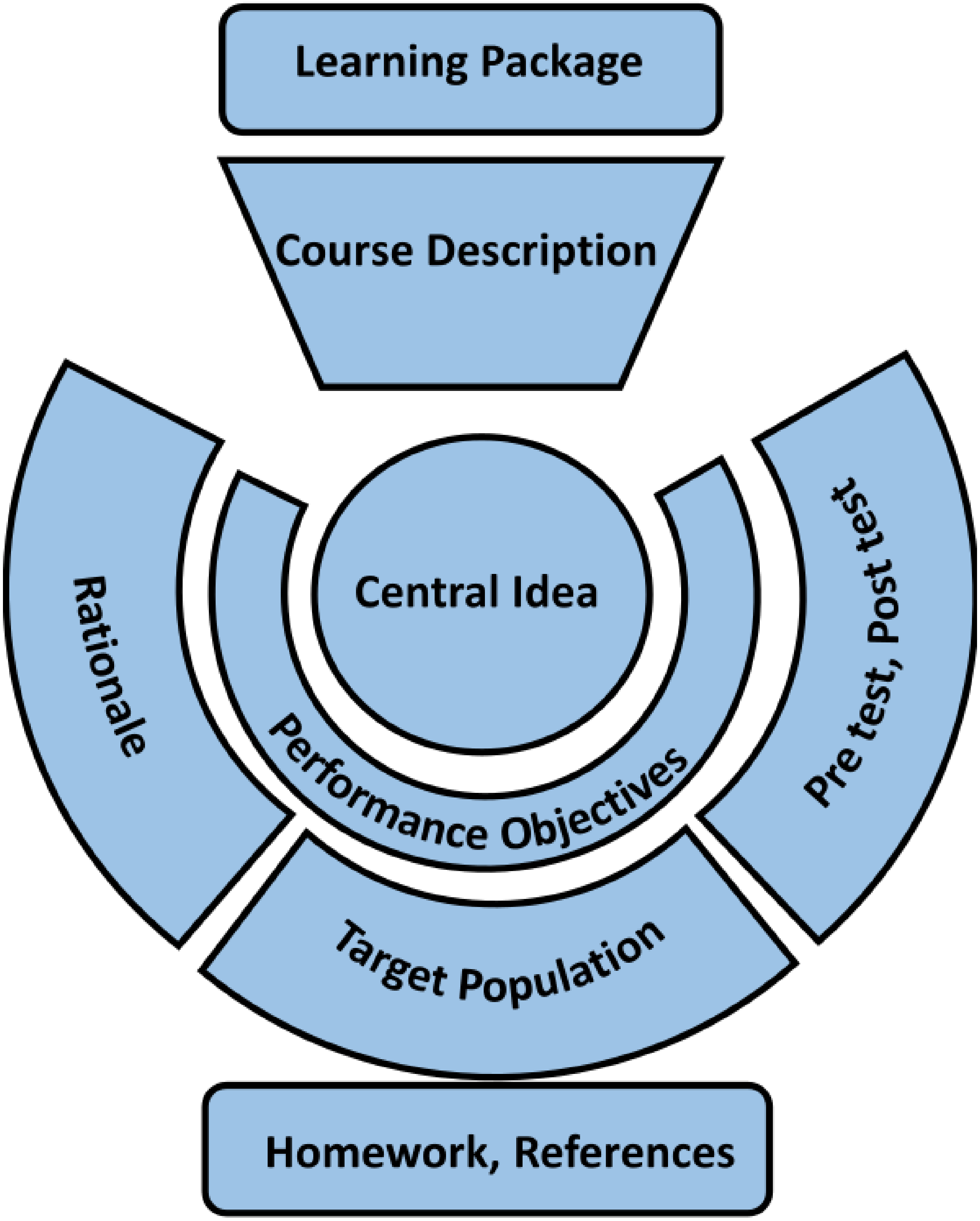
For

Second year students

By

Dr. Duna Tariq Yaseen
Assistant Professor
Dep. Of Power Techniques

2025


Learning Package
Course Description
Central Idea
Performance Objectives
Rationale
Pre test, Post test
Target Population
Homework, References

Course Description

Name The decision
Heat transfer
symbol The decision
Chapter/The year
Semester system/second stage
Date Preparation This Description
2025/5/9
.41 Shapes Attendance Available
My presence completely
Number of hours AcademicTotal Number Units (Total)
hours (theoretical + practical) per 60 semester
.43 Name Responsible The decision Academi
Name Dr. Duna Tariq Yassen
decision Goals The

Course objectives	1- Providing the student with basic skills in heat transfer calculations. 2- Identify the methods of heat transfer 3.Learn how to improve heat transfer and its applications -

Science and Education Strategy	
Strategy	- Midterm and final exams - Short daily exams - Homework - Reports - Interaction within the lecture - Continuing educational seminars. - Guidance and follow-up lectures. - Electronic seminars and workshops

week	**FIRST SEMESTER** Name of unit/course or topic	Teaching method	Evaluation method
3 - 1	Basic principles and importance of heat transfer.	Theoretical Practical +	N+A exam
5 - 4	The three kinds of heat transfer , conduction heat transfer, convection heat transfer , radiation heat transfer, examples	Theoretical Practical +	N+A exam
6-7	Conduction of heat transfer in the steady state conduction through a homogeneous plans wall	Theoretical Practical +	N+A exam
8	Conduction through a composite plans wall, heat resistance . conduction through a homogeneous cylinder wall	Theoretical Practical +	N+A exam
9	Conduction through a composite cylinder wall , influence of variable conductivity , examples	Theoretical Practical +	N+A exam
11-10	Heat transfer by convection , Reynolds concept of similarity of the flow of fluids and the viscosity , the most important dimensionless groups, examples	Theoretical Practical +	N+A exam
12-10	Heat transfer by free convection , heat transfer from vertical and horizontal surfaces , examples	Theoretical Practical +	N+A exam
13-14	Heat transfer by free convection from horizontal square plates , heat transfer proportion of air at atmospheric pressure and properties of water . examples	Theoretical Practical +	N+A exam
15	Heat transfer by forced convection , the heating of fluids in turbulent flow through pipes , examples	Theoretical Practical +	N+A exam
Second Semester			
1-3	Heat exchanger effectiveness ratio , examples	Theoretical Practical +	N+A exam
4-5	Heat transfer through fins , condensation and boiling heat transfer	Theoretical Practical +	N+A exam
6-8	Heat transfer by radiation , the concept of a perfect black body	Theoretical Practical +	N+A exam
9-10	Stefan – boltzmann’s law of total radiation , general equation for heat exchange by radiation between black surfaces , examples	Theoretical Practical +	N+A exam
11-12	Heat exchange by radiation between large parallel black plane , examples	Theoretical Practical +	N+A exam

13	Heat exchange by radiation between large parallel planes of different emissivity , examples	Theoretical Practical +	N+A exam
14	Heat conduction in series with convection and radiation, examples	Theoretical Practical +	N+A exam
15	Heat transfer through air space , examples		

References	
Required textbooks .1	
Main references .2 (sources)	1. FRANK P. INCROPERA) Fundamentals of heat transfer 2. Heat Transfer 10th – Holman 3. Yunus, heat transfer
A. Recommended books and references (scientific (.journals, reports, etc	Virtual Library of the Ministry of Higher Education and Scientific Research
,b. Electronic references websites	The Virtual Library of the Ministry of Higher Education - and Scientific Research The Institute's electronic library -

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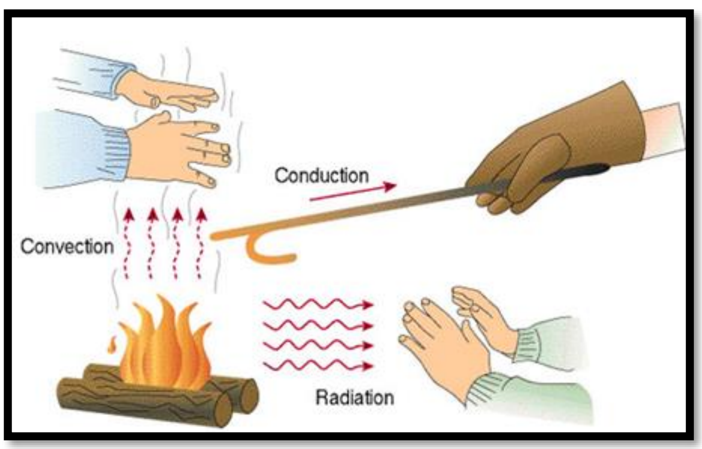
Learning package

In

Type of heat transfer

For

Students of Second Year



By
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2025

1/ Overview

1 / A –Target population :-

For students of Second year
Technological institute of Basra
Dep. Of Power Mechanics Techniques

1 / B –Rationale :-

1st , 2nd week

Understanding the (conduction, convection, Radiation).

1 / C –Central Idea :-

The student understands Basic modes of Heat Transfer

1 / D – Performance Objectives

Learn how to solve problems related to heat transfer modes.

https://youtu.be/FX_1h3XkN84
https://youtu.be/Me60Ti0E_rY

2/ Pretest :

Q 1	**1. Which of the following is a method of heat transfer?:**		
A	Conduction	B	Convection
C	Radiation	D	A,B and C
E	Reflection		
Q 2	**Which of the following is the worst conductor of heat?**		
A	air	B	Plastic
C	water	D	aluminum
E	wood		
Q 3	**The rate of Radiation heat transfer is expressed by**		
A	**Stefan–Boltzmann law**	B	Newton's law
C	Fourier's law	D	Kirchhoff's law
E	None of the above		
Q 4	**The thermal resistance for wall exposed by?**		
A	R_{wall}=KA/L	B	R_{wall}=LA/K
C	R_{wall}=L/KA	D	R_{wall}=A/KL
E	R_{wall}=K/AL		

<u>3/</u> <u>Heat transfer</u> <u>:-</u>

CHAPTER 1

1- Introduction:

Heat Transfer*:* **is that science which seeks to predict the energy transfer that maytake place between material bodes as a result of a temperature difference**.

Heat: the form of energy that can be transferred from one system to another as aresult of temperature difference.

We all know from experience that a cold canned drink left in a room warms up and a warm canned drink left in a refrigerator cools down. This is accomplished by the transfer of *energy* from the warm medium to the cold one. The energy transfer is always from the higher temperature medium to the lower temperature one, and the energy transfer stops when the two mediums reach the same temperature

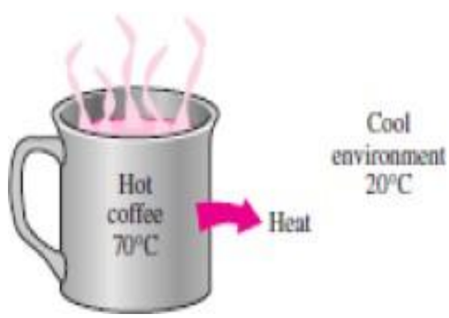
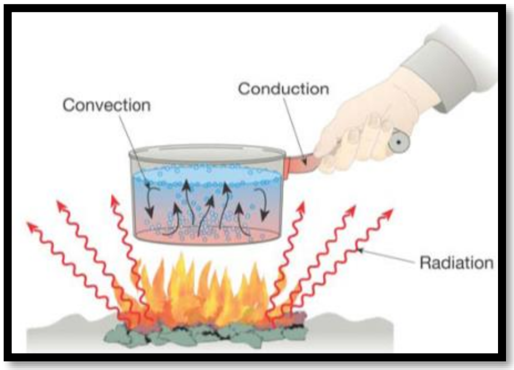
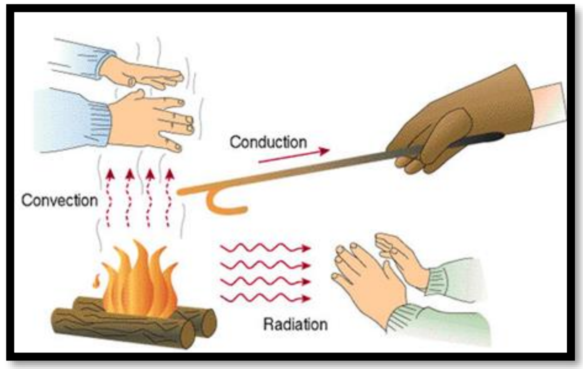


Fig. (1) Heat flows in the direction interface temperature *of decreasing temperature*

2.Modes of Heat Transfer:

There are three fundamental modes of heat transfer; *Conduction, Convection, and Radiation*





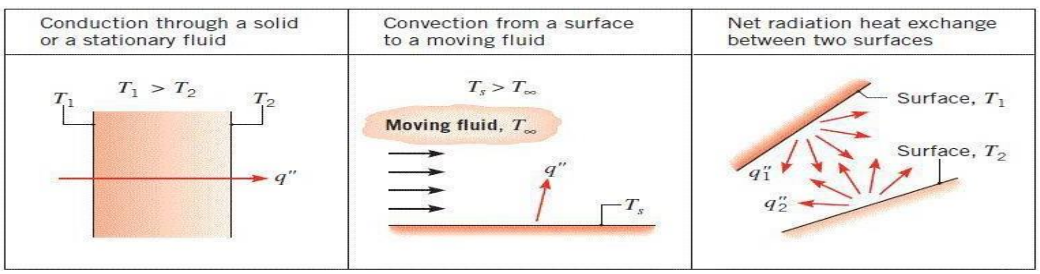
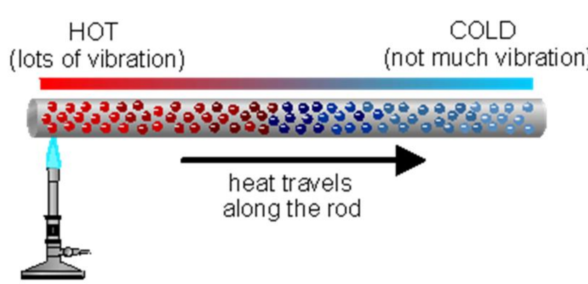


Fig.(2) heat transfer mode

2.1 **Conduction**: is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles.

- The rate of heat conduction through a medium depends on the geometry of the medium, its thickness, and the material of the medium, as well as the temperature difference across the medium.



Fourier's Law of Heat Conduction: - The law states that the rate of heat flow by conduction in any medium in any direction is proportional to the area normal to the direction of heat flow and also proportional to the temperature gradient in that direction.

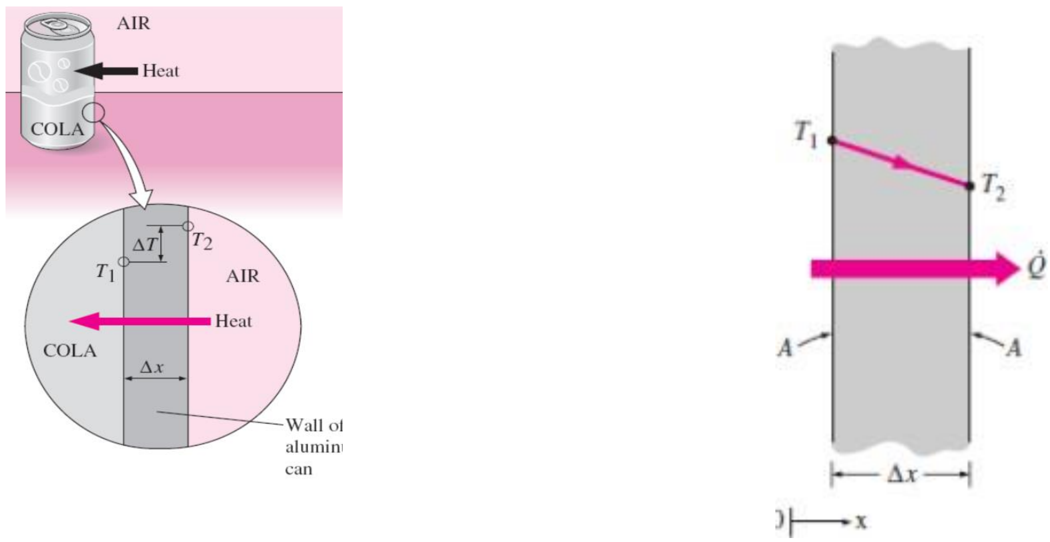


Fig.(3) Heat conduction through a plane wall of thickness Δx and area A.

$$0 = KA\frac{T1-T2}{\Delta X} = -KA\frac{\Delta T}{\Delta X} \qquad \text{WATT}$$

Where:

k: is the thermal conductivity of the material (W/mK)

A: is normal cross section area ($m2$)

ΔT: is temperature difference between surfaces (K)

Δx: is the wall thickness (m)

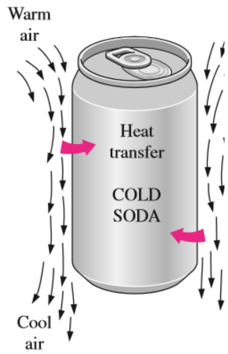
The negative sign ensures that heat transfer in the positive x direction is a posit quantity.

Thermal Conductivity: The thermal conductivity of a material is a measure of the ability of the material to conduct heat.

- ❖ A high value for thermal conductivity indicates that the material is a good heat conductor, and a low value indicates that the material is a poor heat conductor or insulator.
- ❖ The thermal conductivity value depends on the material.
- ❖ The thermal conductivity also depends somewhat on the temperature of the material.

2.2 **Convection**: is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion.

- Convection is called forced convection if the fluid is forced to flow over the surface by external means such as a fan or pump. In contrast, convection is called natural (or free) convection if the fluid motion is caused by buoyancy forces that are induced by density differences due to the variation of temperature in the fluid.



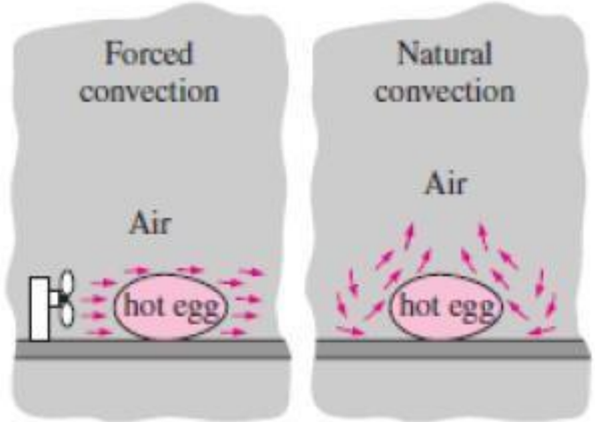


Fig.(4) The cooling of an egg by forced and natural convection

Newton's Equation:

$Q_{conv.}$=h A (T_S−T∞) (Watt)

Where:

h: is the convection heat transfer coefficient in W/m^2 °C

A: is the surface area through which convection heat transfer takes place in (m^2)

TS: is the surface temperature in (°C)

T∞ :is the temperature of the fluid sufficiently far from the surface

- *The convection heat transfer coefficient:* is not a property of the fluid. It is an experimentally determined parameter whose value depends on all the variables influencing convection such as the surface geometry, the nature of fluid motion, the properties of the fluid, and fluid velocity.

2.3 **Radiation:** is energy transport due to emission of electromagnetic waves or photons from a surface or volume. The radiation does not require a heat transfer medium, and can occur in a vacuum.

□ The heat transfer by radiation is proportional to the fourth power of the absolute material temperature.

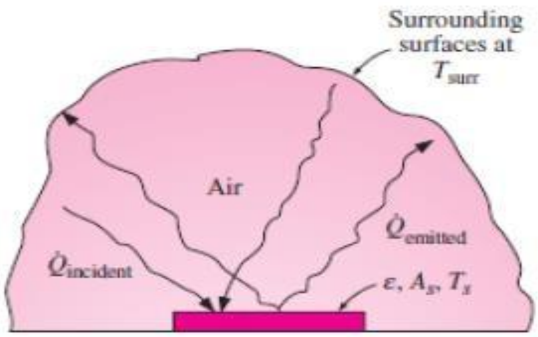


Fig.(5) Radiation heat transfer between a surface and the surfaces surrounding it.

Stefan–Boltzmann law:

$Q_{rad}=\varepsilon\sigma A_s\,(T^4{}_s - T^4{}_{surr})$ (Watt)

$\sigma=5.67\times10-8\ W/m^2.\ K^4$ is the Stefan–Boltzmann constant

ε is the emissivity of the material, its value depend on the material

T_s is the surface temperature

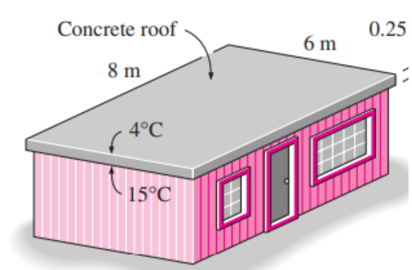
$Tsurr$ is the surrounding surface temperature

Example (1)/ One face of a copper plate 3 cm thick is maintained at 400◦C, and the other face is maintained at 100◦C. How much heat is transferred through the plate? the thermal conductivity for copper is 370W/m· ◦C .

Solution q/A= − k ΔT/Δx= −(370) (100−400)/0.03 = 3.7 MW/m2

***Example (2)**:* The roof of an electrically heated home is 6 m long, 8 m wide, an 0.25 m thick, and is made of a flat layer of concrete whose thermal conductivity is k 0.8 W/m · °C. The temperatures of the inner and the outer surfaces of the roof one night are measured to be 15°C and 4°C, respectively

Determine the rate of heat loss through the roof that night



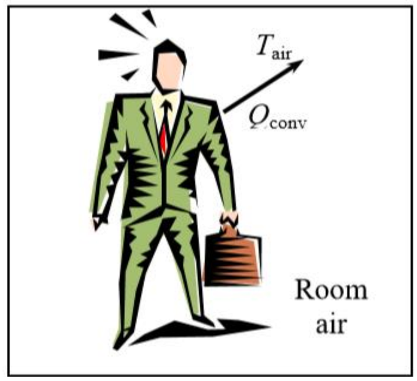
Solution: Noting that heat transfer through the roof is by conduction and the area of the roof A= 6 m x 8 m = 48 m2, the steady rate of heat transfer through the roof is determined to

$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.8\ \text{W/m}\cdot{}^\circ\text{C})(48\ \text{m}^2)\frac{(15-4)^\circ\text{C}}{0.25\ \text{m}} = \mathbf{1690\ W = 1.69\ kW}$$

Example (3)/ Air at 20◦C blows over a hot plate 50 by 75 cm maintained at 250◦C. The convection heat-transfer coefficient is 25 W/m^2 · ◦C. Calculate the heat transfer.

Solution/ $q = h\ A\ (T_s - T_\infty) = (25)\ (0.05)\ (0.75)\ (250 - 20) = 2.156\ kW$

Example (4)/ For heat transfer purposes, a standing man can be modeled as a 30-cm-diameter, 170-cm-long vertical cylinder with both the top and bottom surfaces insulated and with the side surface at an average temperature of 34°C. For a convection heat transfer coefficient of 15 W/m2 · °C, determine the rate of heat loss from this man by convection in an environment at 20°C.



Analysis The heat transfer surface area of the person is

$$A_s = \pi DL = \pi(0.3\text{ m})(1.70\text{ m}) = 1.60\text{ m}^2$$

Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{conv} = hA_s\Delta T = (15\text{W/m}^2\cdot{}^\circ\text{C})(1.60\text{m}^2)(34-20)^\circ\text{C} = \mathbf{336W}$$

Example (5)/ A long, cylindrical electrically heated rod, 2 cm in diameter, is installed in a vacuum furnace. The surface of the heating rod has an emissivity of 0.9 and is maintained at 1000 K, while the interior walls of the furnace are black and are at 800 K. Calculate the net rate at which heat is lost from the rod per unit length.

$$q_r = A\varepsilon\sigma(T_1^4 - T_2^4) = \pi D_1 L\varepsilon\sigma(T_1^4 - T_2^4)$$
$$= \pi(0.02\text{ m})(1.0\text{ m})(0.9)\left(5.67\times10^{-8}\frac{\text{W}}{\text{m}^2\text{K}^4}\right)(1000^4 - 800^4)(\text{K}^4)$$
$$= 1893\text{ W}$$

Example(6) Consider a person standing in a room maintained at 20°C at all times. The inner surfaces of the walls, floors, and ceiling of the house are observed to be at an average temperature of 12°C in winter and 23°C in summer. Determine the rates of radiation heat transfer between this person and the surrounding surfaces in both summer and winter if the exposed surface area, emissivity, and the average outer surface temperature of the person are 1.6 m2, 0.95, and 32°C, respectively

Analysis Noting that the person is completely enclosed by the surrounding surfaces, the net rates of radiation heat transfer from the body to the surrounding

(*a*) Summer: $T_{surr} = 23+273=296$

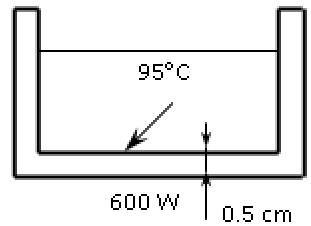
$$\dot{Q}_{rad} = \varepsilon\sigma A_s (T_s^4 - T_{surr}^4)$$
$$= (0.95)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.6\ \text{m}^2)[(32+273)^4 - (296\ \text{K})^4]\text{K}^4$$
$$= \mathbf{84.2\ W}$$

(*b*) Winter: $T_{surr} = 12+273= 285$ K

$$\dot{Q}_{rad} = \varepsilon\sigma A_s (T_s^4 - T_{surr}^4)$$
$$= (0.95)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1.6\ \text{m}^2)[(32+273)^4 - (285\ \text{K})^4]\text{K}^4$$
$$= \mathbf{177.2\ W}$$

Example(7): Water is boiling in a 25cm-diameter aluminum pan (k = 237 W/m · °C) at 95°C. Heat is transferred steadily to the boiling water in the pan through its 0.5cm-thick flat bottom at a rate of 800 W. If the inner surface temperature of the bottom of the pan is 108°C, determine (a) the boiling heat transfer coefficient on the inner surface of the pan, and (b) the outer surface temperature of the bottom of the pan.

Solution: (*a*) The boiling heat transfer coefficient is



$$A_s = \frac{\pi D^2}{4} = \frac{\pi (0.25\ \text{m})^2}{4} = 0.0491\ \text{m}^2$$

$$Q = hA_s(T_s - T_\infty)$$

$$h = \frac{Q}{A_s(T_s - T_\infty)} ,$$

$$= \frac{800\ \text{W}}{(0.0491\ \text{m}^2)(108-95)°\text{C}} = 1254\ \text{w/m}^2.°\text{c}$$

b) The outer surface temperature of the bottom of the pan is

$$Q = KA\frac{T_{s\,out} - \text{T inner}}{l}$$

$$T_{s\ outer} = T_{s\ inner} + \frac{QL}{KA} = 108 + \frac{(800)(0.005m)}{(237)(0.0491)} = 108.3\ \text{C}$$

Chapter contents:

CONDUCTION HEAT TEANSER
Convection HEAT TEANSER
Radiation HEAT TEANSER
COMINED HEAT TRANSFER SYSTEMS

4/ Post test :-

A vary large furnace contains hot gas and particles at (1900 K) heat is transferred from gases and particles by convection and radiation to interior surface where temperature is (1700 K) the rate of heat transfer by radiation is (1000 w/ m^2) while the unit surface conductance for convection on the interior surface is (20 w/ m^2.k) the furnace wail is (0.65 m) thick and its thermal conductivity (2.8 w/m.k) :

1- Draw the thermal circuit.
2- Determine the rate of heat flow through the wall.
3-Determine the outside surface temperature

5/ HomeWorks: -

Home work

1. Hot air at 80°C is blown over a 2m * 4m flat surface at 30°C. If the average convection heat transfer coefficient is 55 W/m2 · °C, determine the rate of heat transfer from the air to the plate, in kW.

https://youtu.be/R04tkiD6Hz4?si=KS7R_f08KhmGxK8Q

5. Primary sources:

1-Heat transfer	By : J.P. Holman
2. Heat and mass transfer	By: YunusA.Gengel

6. Suggested sources:

Fundamentals of heat and mass transfer.	By: Incropera
Heat transfer handbook	By: Bijan

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Department of Electronic Techniques

Learning package

In

Steady-State Conduction One Dimension

For

Second year students

By

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Assistant Professor
Dep. Of Power Mechanics Techniques

2025

1/ Overview

1 / A –Target population :-

For students of Second year
Technological institute of Basra
Dep. Of Power Mechanics Techniques

1 / B –Rationale :-

The student understands analysis heat transfer in Multilayer plane Walls ,cylinder and sphere)

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1 / C –Central Idea :-

3^{rd}- 6^{th} weeks

Understanding the (Heat transfer mechanism in Multilayer plane Walls ,cylinder and

1 / D – Performance Objectives

Students' ability to calculate the heat transfer through Multilayer plane Walls ,cylinder and sphere.

2/ Pretest :

The thermal resistance for wall cylinder is exposed by?		
$R_{cyl} = \frac{ln(r_2/r_1)}{2\pi LK}$	B	$R_{cyl} = \frac{ln(r_1/r_2)}{2\pi LK}$
$R_{cyl} = \frac{ln(r_2/r_1)}{2\pi rK}$	D	$R_{cyl} = \frac{\ln(r_2 - r_1)}{2\pi LK}$

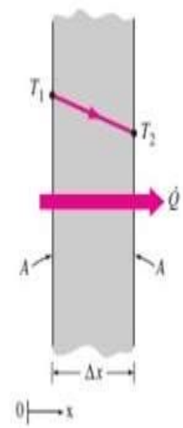
Heat is transferred through solid bodies by:		
Convection	B	Conduction
Radiation	D	All of the mentioned
Reflection		
Adding more insulation to a wall always		
increases the thermal resistance	B	decreases the thermal resistance
increases the rate heat transfer	D	increases the thermal conductivity
Everything mentioned is wrong		
Which of the following is the rate of heat flux transfer unit		
W/ m^2	B	J/ m^2
W/ s	D	W
J/ s		

3/ Scientific content:

Steady –State Heat Conduction in Plane Wall

The Fourier equation, for steady conduction through a constant area plane wall :

$$Q_{cond.} = kA\,\frac{T_1 - T_2}{\Delta x} = -kA\,\frac{\Delta T}{\Delta x}$$

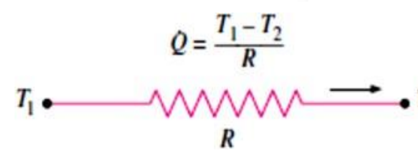


The Thermal Resistance:

Heat conduction through a plane wall can be rearranged as:

$$Q_{cond.} = \frac{T_1 - T_2}{R_{wall}} \qquad (w)$$

$$R_{wall} = \frac{\Delta x}{kA} \qquad (°C/w)$$



R wall: is the thermal resistance of the wall against heat conduction or simply the conduction resistance of the wall. Note that the thermal resistance of a medium depends on the geometry and the thermal properties of the medium

If more than one material is present, as in the multilayer wall shown in Fig.

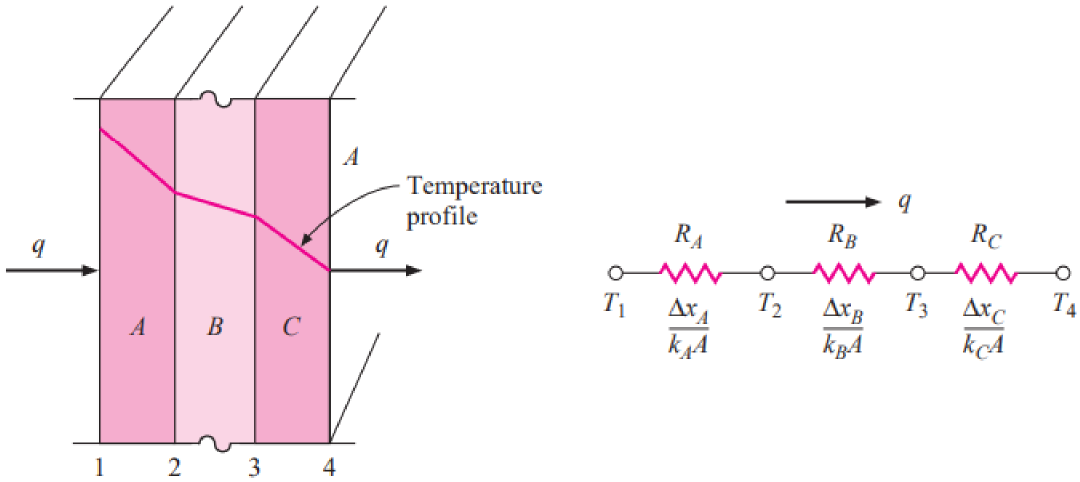


Fig.7 One-dimensional heat transfer through a composite wall

The analysis would proceed as follows:

- The temperature gradients in the three materials are shown, and the heat flow may be written

$$q = -k_A A \frac{T_2 - T_1}{\Delta x_A} = -k_B A \frac{T_3 - T_2}{\Delta x_B} = -k_C A \frac{T_4 - T_3}{\Delta x_C}$$

Note that the heat flow must be the same through all sections.

$$\text{Heat flow} = \frac{\text{thermal potential difference}}{\text{thermal resistance}}$$

$$q = \frac{T_1 - T_4}{\Delta x_A / k_A A + \Delta x_B / k_B A + \Delta x_C / k_C A}$$

$$q = \frac{\Delta T_{\text{overall}}}{\sum R_{\text{th}}}$$

- Consider the composite wall shown in Fig. (8), which consists of two parallel layers. The thermal resistance network, which consists of two parallel resistances, can be represented as shown in the figure.
- Noting that the total heat transfer is the sum of the heat transfers through each layer, we have

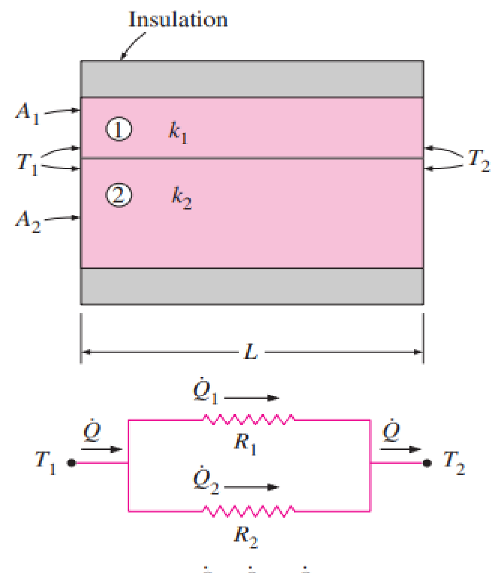


Fig.8 Thermal resistance network for two parallel layers

two parallel layer Series and parallel one-dimensional heat transfer through a composite wall

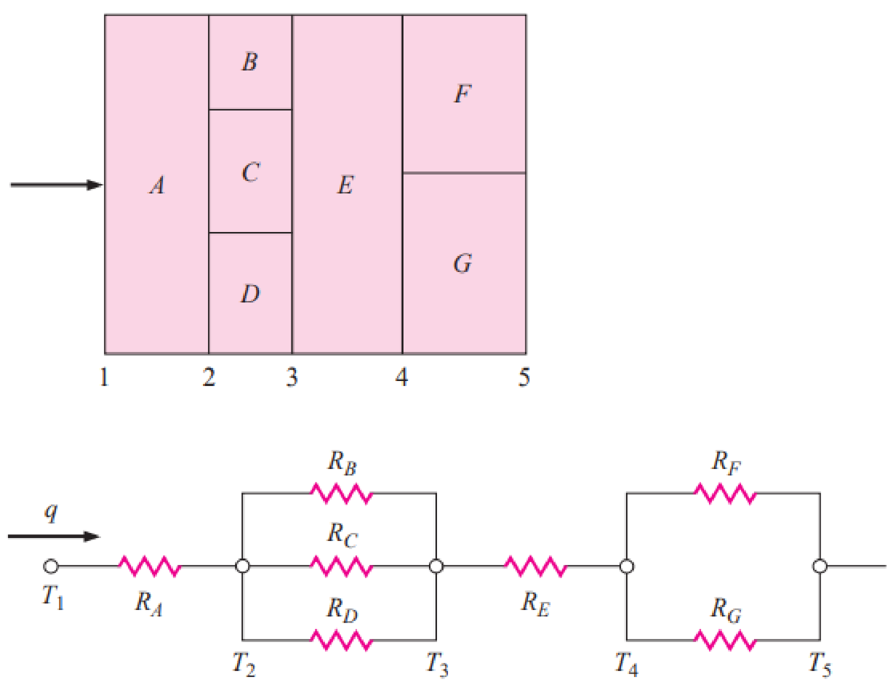
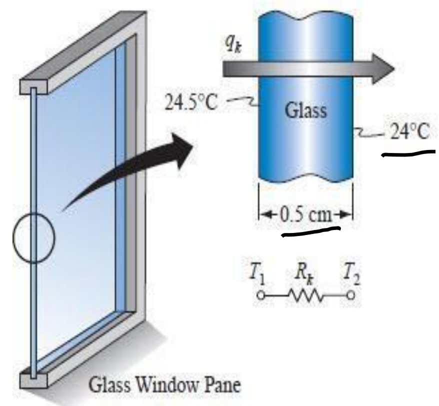


Fig 9. Composite wall in a series and parallel arrangement

Example(8): Calculate the thermal resistance and the rate of heat transfer through a pane of window glass (k = 0.81 W/m K) 1 m high, 0.5 m wide, and 0.5 cm thick, if the outer surface temperature is 24°C and the inner-surface temperature is 24.5°C.

Solution: A schematic diagram of the system is shown in Fig. Assume that steady state exists and that the temperature is uniform over the inner and outer surfaces. The thermal resistance to conduction *Rk* is:

$$R_k = \frac{L}{kA} = \frac{0.005\text{ m}}{0.81\text{ W/m K} \times 1\text{ m} \times 0.5\text{ m}} = 0.0123\text{ K/W}$$



The rate of heat loss from the interior to the exterior surface is:

$$q_k = \frac{T_1 - T_2}{R_k} = \frac{(24.5 - 24.0)^\circ\text{C}}{0.0123\text{ K/W}} = 40\text{ W}$$

Example(9): One side of a copper block 5 cm thick is maintained at 250◦C. The other side is covered with a layer of fiberglass 2.5 cm thick. The outside of the fiberglass is maintained at 35◦C, and the total heat flow through the copper- fiberglass combination is 44 kW. What is the area of the slab? $kCopper$=386 w/m°C $kfiberglass$=0.038 $w/\ m$°C

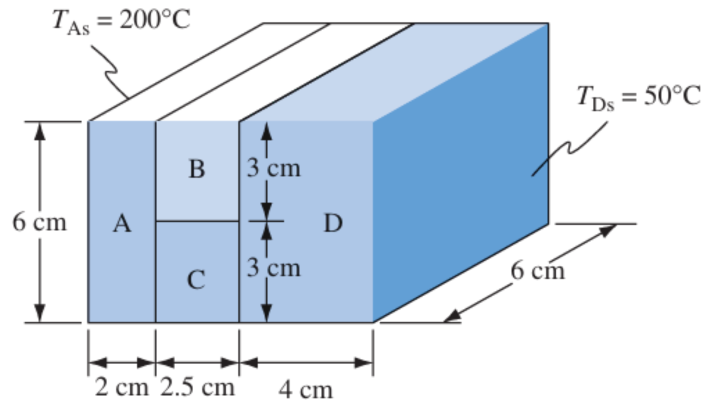
solution

$$q = \frac{\Delta T}{R_{total}} = \frac{\Delta T}{\frac{\Delta x_1}{k_1 A} + \frac{\Delta x_2}{k_2 A}} = \frac{\Delta T}{\frac{1}{A}\left[\frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2}\right]}$$

$$\frac{44{,}000}{A} = \frac{250 - 35}{\frac{0.05}{386} + \frac{0.025}{0.038}} \qquad A = 134.7\text{ m}^2$$

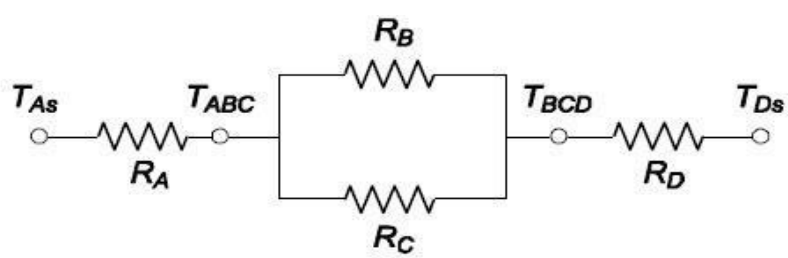
Example(10):

A section of a composite wall with the dimensions shown below has uniform temperatures of 200°C and 50°C over the left and right surfaces, respectively. If the thermal con ductivities of the wall materials are: $k_A = 70$ W/m K $k_B = 60$ W/m K $k_C = 40$ W/m K , and $k_D = 20$ W/m K, determine the rate of heat transfer through this section of the wall and the temperatures at the interfaces.



Solution:

The thermal circuit for the composite wall is



3. Each of these thermal resistances has a form given by

$$R_k = \frac{L}{A\,k}$$

$$R_A = \frac{L_A}{A_A\,k_A} = \frac{0.02\,\text{m}}{(0.06\,\text{m})(0.06\,\text{m})[70\,\text{W/(m K)}]} = 0.0794\ \text{K/W}$$

$$R_B = \frac{L_B}{A_B\,k_B} = \frac{0.025\,\text{m}}{(0.03\,\text{m})(0.06\,\text{m})[60\,\text{W/(m K)}]} = 0.2315\ \text{K/W}$$

$$R_C = \frac{L_C}{A_C\,k_C} = \frac{0.025\,\text{m}}{(0.03\,\text{m})(0.06\,\text{m})[40\,\text{W/(m K)}]} = 0.3472\ \text{K/W}$$

$$R_D = \frac{L_D}{A_D\,k_D} = \frac{0.04\,\text{m}}{(0.06\,\text{m})(0.06\,\text{m})[20\,\text{W/(m K)}]} = 0.5556\ \text{K/W}$$

The total thermal resistance of the wall section,

$$R_{\text{total}} = R_A + \frac{R_B\,R_C}{R_B + R_C} + R_D$$

$$R_{\text{total}} = 0.0794 + \frac{(0.2315)(0.3472)}{0.2315 + 0.3472} + 0.5556\ \text{K/W}$$

$$R_{\text{total}} = 0.7738\ \text{K/W}$$

The total rate of heat transfer through the composite wall is given by

$$q = \frac{\Delta T}{R_{\text{total}}} = \frac{200^\circ\text{C} - 50^\circ\text{C}}{0.7738\,\text{K/W}} = 194\ \text{W}$$

The average temperature at the interface between material *A* and materials *B* and *C* (*TABC*) can be determined by examining the conduction through material *A* alone

$$q_{ka} = \frac{T_{As} - T_{ABC}}{R_A} = q$$

Solving for T_{ABC}

$$T_{ABC} = T_{As} - q\,R_A = 200°\text{C} - (194\text{ W})\,(0.0794\text{ K/W}) = 185°\text{C}$$

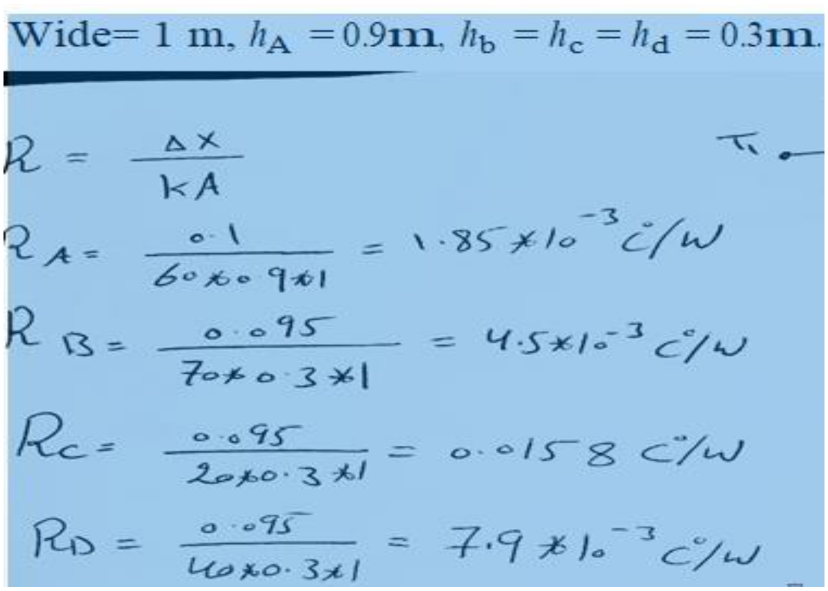
The average temperature at the interface between material *D* and materials *B* and *C* (*TBCD*) can be determined by examining the conduction through material *D* alone

$$q_{kD} = \frac{T_{BCD} - T_{Ds}}{R_D} = q$$

Solving for T_{BCD}

$$T_{BCD} = T_{Ds} + q\,R_D = 50°\text{C} + (194\text{ W})\,(0.5556\text{ K/W}) = 158°\text{C}$$

Example(11): Determine the rate of heat transfer through this section of the wall. And draw the thermal circuit. $k_A = 60$ W/m.k, , $k_b = 70$ w/m k, $k_c =$ 20 w/m.k, $k_d = 40$ w/m.k, $T_1 = 250$ °C, $T_2 = 100$°C, $\Delta X_A = 0.1$M, $\Delta X_B = \Delta X_C = \Delta X_D = 0.095$ m, Wide= 1 m, $h_A = 0.9$m, $h_b = h_c = h_d = 0.3$m.



Wide= 1 m, $h_A = 0.9$m, $h_b = h_c = h_d = 0.3$m.

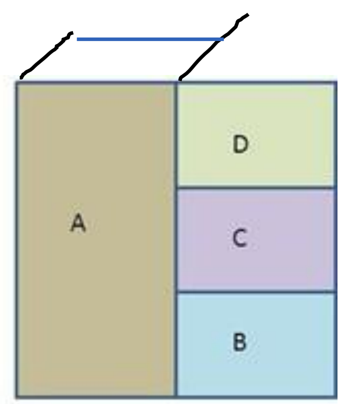
$R = \frac{\Delta X}{kA}$ T_1

$R_A = \frac{0.1}{60*0.9*1} = 1.85*10^{-3}$ °C/W

$R_B = \frac{0.095}{70*0.3*1} = 4.5*10^{-3}$ °C/W

$R_C = \frac{0.095}{20*0.3*1} = 0.0158$ °C/W

$R_D = \frac{0.095}{40*0.3*1} = 7.9*10^{-3}$ °C/W



4- Steady –State Heat Conduction in Cylinders:

Consider a long cylinder of inside radius *ri*, outside radius *ro*, and length *L*, such as the one shown in Figure(10) We expose this cylinder to a temperature differential *Ti* −*To* and ask what the heat flow will be.

For a cylinder with length very large compared to diameter, it may be assumed that the heat flows only in a radial direction.

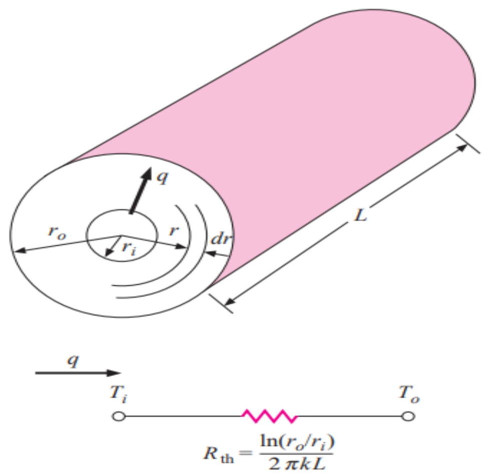


Fig 10. One-dimensionalheat flow heat flow through a hollo

Again, Fourier's law is used.
the area for heat flow in the cylinder is

$$A_r = 2\pi rL$$

so that Fourier's law is written

$$q_r = -kA_r \frac{dT}{dr}$$

$$q_r = -2\pi krL \frac{dT}{dr}$$

with the boundary conditions

$$T = T_i \qquad \text{at } r = r_i$$
$$T = T_o \qquad \text{at } r = r_o$$

The solution is:

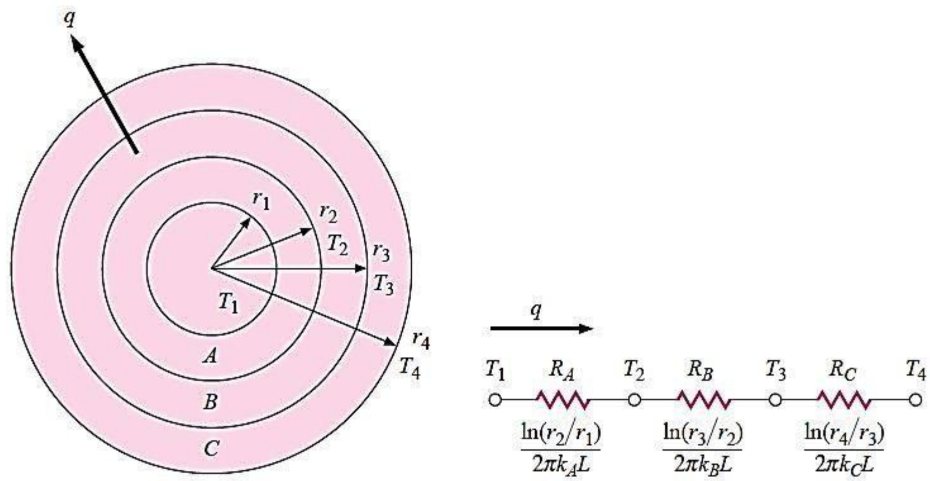
$$q = \frac{2\pi kL\,(T_i - T_o)}{\ln(r_o/r_i)}$$

$$R_{\text{th}} = \frac{\ln(r_o/r_i)}{2\pi kL}$$

and the thermal resistance in this case is

https://youtu.be/ucW5uylrmGM

❖ The thermal-resistance concept may be used for multiple-layer cylindrical walls just as it used for plane walls. For the three-layer system shown in Figure (11) the



solution is

fig.(11) one-dimensional heat flow through multiple cylindrical section

$$q = \frac{2\pi L\,(T_1 - T_4)}{\ln(r_2/r_1)/k_A + \ln(r_3/r_2)/k_B + \ln(r_4/r_3)/k_C}$$

Example(1): A stainless steel k = 14.4 W/(m.K) pipe carries water condensate from a condenser to a pump. The inside-wall temperature is 40 C, and the outside-wall temperature is 38°C. Determine the heat transfer through the pipe wall per unit length of pipe,***D2*** = 32.39cm, ***D1*** = 29.53 cm.

$$q_r = \frac{2\pi k L}{\ln(R_2/R_1)}(T_1 - T_2)$$

$$\frac{q_r}{L} = \frac{2\pi(14.4)}{\ln(32.39/29.53)}(40 - 38) \qquad \frac{q_r}{L} = 1\,957.4 \text{ W/m} = 1.96 \text{ kW/m}$$

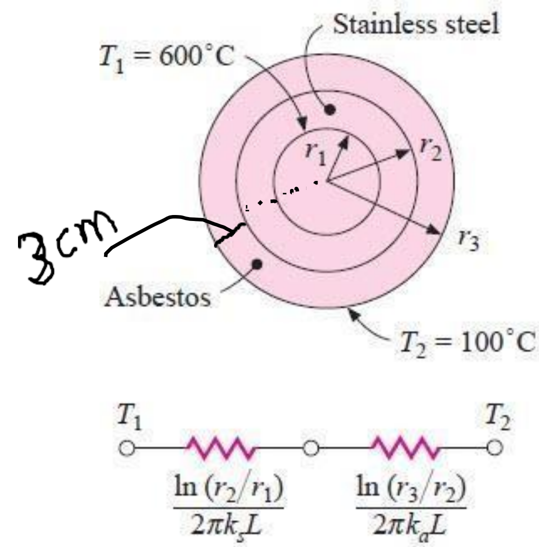
Example(2): A thick-walled tube of stainless steel [*k* =19 W/m.◦C] with 2 cm inner diameter (ID) and 4 cm outer diameter (OD) is covered with a 3-cm layer of asbestos insulation [*k* =0.2 W/m◦C]. If the inside wall temperature of the pipe is maintained at 600◦C, calculate the heat loss per meter of length. Also calculate the tube– insulation

Solution:

The heat flow is given by:

$$\frac{q}{L}=\frac{2\pi\,(T_1-T_2)}{\ln(r_2/r_1)/k_s+\ln(r_3/r_2)/k_a}$$

$$=\frac{2\pi\,(600-100)}{(\ln 2)/19+(\ln\frac{5}{2})/0.2}=680\ \mathrm{W/m}$$



this heat flow may be used to calculate the interface temperature between the outside tube wall and the insulation. we have

$$\frac{q}{L}=\frac{T_a-T_2}{\ln(r_3/r_2)/2\pi k_a}=680\ \mathrm{W/m}$$

Where Ta is the interface temperature, which may be obtained as

$Ta = 595.8^\circ C$

Example(3): A cylindrical radioactive waste piece (k_{rw} =20 W/m K) with radii equal to *r1*=0.4 m is stored in a cylindrical, stainless steel container (k_{ss} =15 W/m.k) of inner and outer radii equal to *r2* = 0.5 m and *r3* = 0.6 m is covered with a 5 cm layer of fiberglass $k_{fiberglass}$ = 0.038 w/ m.k. Calculate the amount of heat transferred if the temperature difference equal to 100°C. L=1m.

$$q=\frac{2\pi L\,\Delta T}{\ln(r_2/r_1)/k_{rw}+\ln(r_3/r_2)/k_{ss}+\ln(r_4/r_3)/k_{fiber}}$$

$$=\frac{2\pi\times 1\times 100}{\ln(0.5/0.4)/20+\ln(0.6/0.5)/15+\ln(0.65/0.6)/0.038}$$

$$q=294.88\,W$$

Example(4): hot steam pipe having an inside surfacetemperature of 250◦C an inside diameter of 8 cm and a wall thickness of 5.5 mm. It is covered with a 9-cm layer of insulation having k =0.5 W/m· ◦C, followed by a 4-cm layer of insulation having k =0.25W/m· ◦C. The outside temperature of the insulation is 20◦C. Calculate the heat lost per meter of length. Assume k =47 W/m· ◦C for the pipe.

solution

For 1 m length

$$R(\text{pipe}) = \frac{\ln(9.1/8)}{2\pi(47)} = 4.363 \times 10^{-4}$$

$$R(\text{ins}(1)) = \frac{\ln(27.1/9.1)}{2\pi(0.5)} = 0.3474$$

$$R(\text{ins}(2)) = \frac{\ln(35.1/27.1)}{2\pi(0.25)} = 0.8246$$

$$R(\text{tot}) = 1.172$$

$$q = \frac{\Delta T}{R} = \frac{250 - 20}{1.172} = 196.2 \ \frac{\text{W}}{\text{m}}$$

4-Post test

Example: 2 A thick-walled tube of stainless steel (k=19W/m^2.°C) with 2-cm inner diameter and 4-cm outer diameter is covered with a 3-cm layer of asbestos insulation (k=0.2 19W/m^2.°C). If the inside wall temperature of the pipe is maintained at 600°C and outside surface of the insulation temperature is 100°C, calculate the heat loss per meter of length, and also calculate the tube-insulation interface temperature.

Solution:

Figure example (2-6) shown the thermal network for this problem. The heat flow is given by

$$\frac{q}{L}=\frac{2\pi(T_1-T_2)}{\frac{\ln\left(\frac{r2}{r1}\right)}{ks}+\frac{\ln\left(\frac{r3}{r2}\right)}{k_{ins}}}=\frac{2\pi(600-100)}{\frac{\ln\left(\frac{2}{1}\right)}{19}+\frac{\ln\left(\frac{5}{2}\right)}{0.2}}=680W/m$$

The heat flow may be used to calculate the interface temperature between the outside tube wall and the insulation. We have

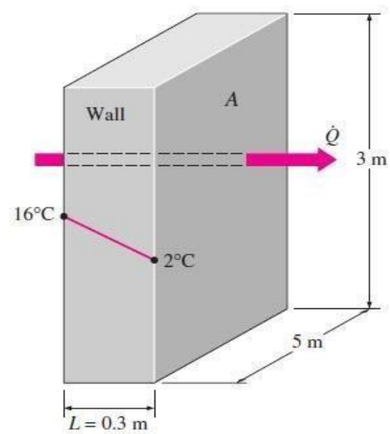
$$\frac{q}{L}=\frac{(T_1-T_a)}{\frac{\ln\left(\frac{r2}{r1}\right)}{2\pi ks}}=680W/m$$

T_a is interface temperature, which may be obtained as

T_a= 595.8°C.

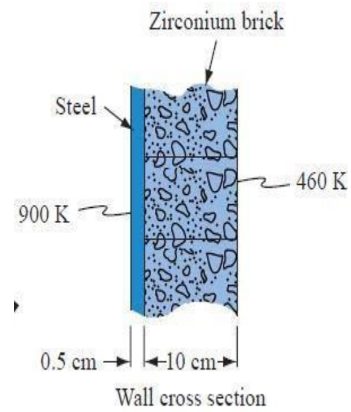
5/ HomeWorks:

1- Consider a 3m high, 5m wide, and 0.3m thick wall whose thermal conductivity is $k = 0.9\ \mathrm{W/m \cdot {}^\circ C}$ .On a certain day, the temperatures of the inner and the outer surfaces of the wall are measured to be 16°C and 2°C, respectively. Determine the rate of heat loss through the wall on that day.



2- Calculate the rate of heat loss from a furnace wall per unit area. The wall is constructed from an inner layer of 0.5cm thick steel ($k = 40$ W/m K) and an outer layer of 10 cm zirconium brick

($k = 2.5$ W/m K) as shown in Figure. The inner surface temperature is 900 K and the outside surface temperature is 460 K. What is the temperature at the interface?



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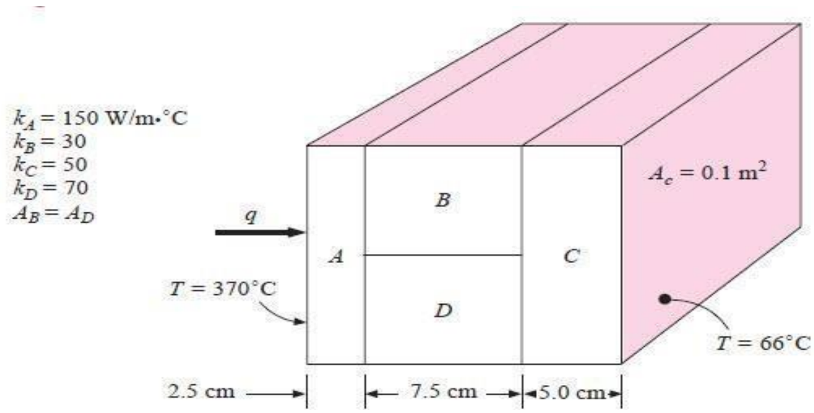
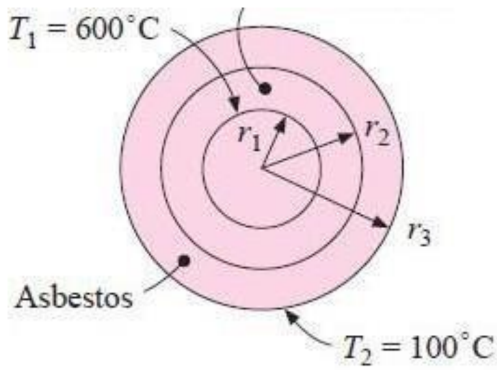
4 .Find the heat transfer per unit area through the composite wall in Figure Assume one-dimensional heat flow.



5. tube of aluminum ($k = 237$ W/m · °C) with 0.2m inner diameter (ID) and 0.3m outer diameter (OD) is covered with a 3-cm layer of asbestos insulation [k =0.2 W/m ◦C]. If the inside wall temperature of the pip is maintained at 600◦C, calculate the heat loss per meter of length. Also calculate the tube–insulation interface temperature.

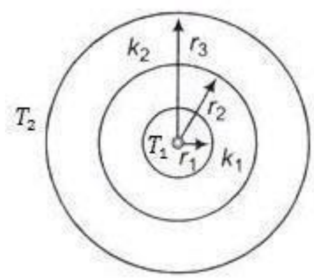


6. Determine the rate of heat transfer through this section of the wall. And draw the thermal circuit.

$k_1 = 60$ w/m. k, $k_2 = 70$ w/m.k

r1 = 0.4 m, r2 = 0.5 m, r3 = 0.6 m

T_1= 250 °C, T_2= 100°CL = 1 m



Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

Learning package

In

Convection and Conduction in Series for Plane wall and cylinder

For

Students of Second Year

By
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Assistant Professor
Dep. Of Power Mechanics Techniques
2025

1/ Overview

1 / A –Target population :-

For students of Second year
Technological institute of Basra
Dep. Of Power Mechanics Techniques

1 / B –Rationale :-

7th , 10th weeks

The student learns to analyze temperature distribution in a plane wall and cylinder with a heat generation system

1 / C –Central Idea :-

The student understands analysis one dimension heat transfer

1 / D – Performance Objectives

Learn how to solve one dimensional conduction problems with heat source

https://youtu.be/4G65iLk99wE?si=QKZwkS7f3pFZiksD

https://youtu.be/ZcO0yjdxIFQ?si=_a_xARj8OSls6bdI

2/ Pretest

Adding more insulation to a wall always		
decreases the thermal resistance	B	**decreases rate heat transfer**
decreases the thermal conductivity	D	**increases the rate heat transfer**
Everything mentioned is wrong		
The amount of heat flow through a body by conduction is		
directly proportional to the surface area of the body	B	**directly proportional to the temperature difference on the two faces**
dependent upon the material of the body	D	**inversely proportional to the thickness of the body**
all of the above		

2- The unit of thermal resistance is		
W/ $m C^0$	B	W/ $m^2 C^0$
C^o/ W	D	m/ $W C^0$
Wm/ C^0		
Which of the following is the best conductor of heat?		
air	B	plastic
water	D	aluminum
wood		

3/ Scientific content:-

CHAPTER 3

5. Convection and Conduction in Series for Plane wall:

In the preceding section we treated conduction through composite walls when the surface temperatures on both sides are specified. The more common problem is heat being transferred between two fluids of specified temperatures separated by a wall. In such a situation the surface temperatures are not known, but they can be calculated if the convection heat transfer coefficients on both sides of the wall are known.

- Newton's law for convection heat transfer rate

$$Q_{conv.} = h\,A\,(T_S - T_\infty)$$

Can be rearranged as

$$\dot{Q}_{conv} = \frac{T_s - T_\infty}{R_{conv}} \qquad (W)$$

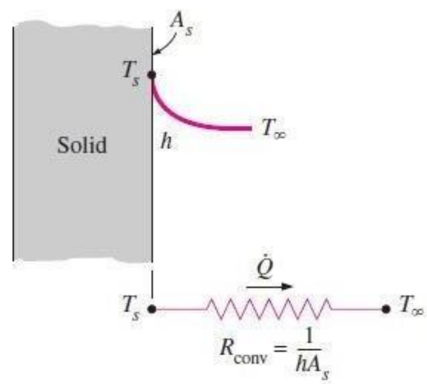


Fig.(12) Schematic for convection resistance at a surface.

- the thermal resistance for convection heat transfer iFigure (13) shows a situation in which heat is transferred between two fluids separated by a wall. According to the thermal network shown below.

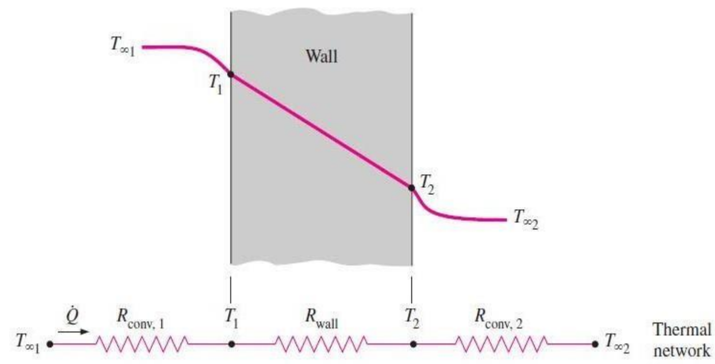


Fig.(13) The thermal resistance network for heat transfer through a plane wall

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{conv, 1}} + R_{\text{wall}} + R_{\text{conv, 2}}}$$

Example(13): A 0.1-m-thick brick wall (k = 0.7 W/m K) is exposed to a cold wind at 270 K through a convection heat transfer coefficient of 40 W/m2 K. On the other side is calm air at 330 K, with a natural-convection heat transfer coefficient of 10 W/m2 K. Calculate the rate of heat transfer per unit area (i.e., the heat flux).

Solution:

he three resistances are

$$R_1 = \frac{1}{\bar{h}_{c,\text{hot}} A} = \frac{1}{(10\ \text{W/m}^2\ \text{K})(1\ \text{m}^2)} = 0.10\ \text{K/W}$$

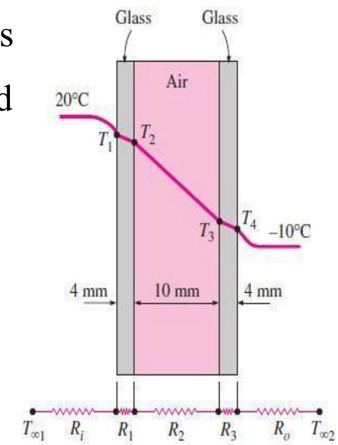
$$R_2 = \frac{L}{kA} = \frac{(0.1\ \text{m})}{(0.7\ \text{W/m K})(1\ \text{m}^2)} = 0.143\ \text{K/W}$$

$$R_3 = \frac{1}{\bar{h}_{c,\text{cold}} A} = \frac{1}{(40\ \text{W/m}^2\ \text{K})(1\ \text{m}^2)} = 0.025\ \text{K/W}$$

the rate of heat transfer per unit area is

$$\frac{q}{A} = \frac{\Delta T}{R_1 + R_2 + R_3} = \frac{(330 - 270)\ \text{K}}{(0.10 + 0.143 + 0.025)\ \text{K/W}} = 223.9\ \text{W}$$

__Example(14):__ Consider a 0.8-m-high and 1.5-m-wide double-pane window consisting of two 4-mm-thick layers of glass (k = 0.78 W/m · °C) separated by a 10- mm-wide stagnant air space (k = 0.026 W/m · °C). Determine the steady rate of heat transfer through this double-pane window and the temperature of its inner surface for a day during which the room is maintained at 20°C while the temperature of the outer doors is -10 C. take the convection heat transfer coefficient on the inner and outer surfaces of window tobe h1 = 10w/m^2.c, h2=40 w/m^2.°C



A =0.8 m *1.5 m =1.2 *m*2

The individual resistances are evaluated

Noting that all three resistances are in series, the total resistance is

$$
\begin{aligned}
R_{\text{total}} &= R_{\text{conv, 1}} + R_{\text{glass, 1}} + R_{\text{air}} + R_{\text{glass, 2}} + R_{\text{conv, 2}} \\
&= 0.08333 + 0.00427 + 0.3205 + 0.00427 + 0.02083 \\
&= 0.4332°\text{C/ W}
\end{aligned}
$$

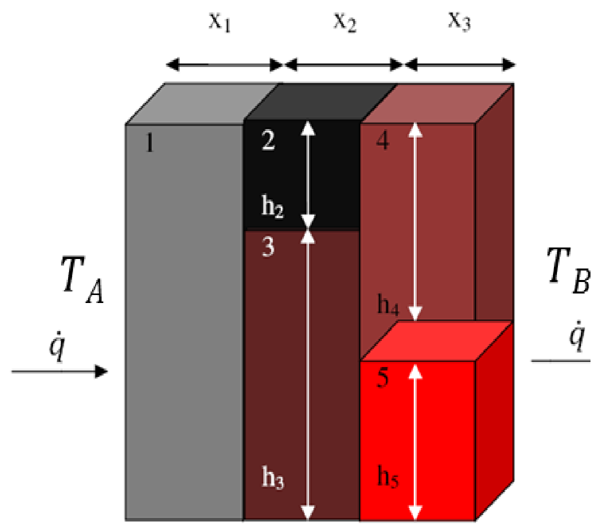
Then the steady rate of heat transfer through the window becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[20 - (-10)]°\text{C}}{0.4332°\text{C/ W}} = 69.2 \text{ W}$$

The inner surface temperature of the window in this case will be

$$T_1 = T_{\infty 1} - \dot{Q}R_{\text{conv, 1}} = 20°\text{C} - (69.2 \text{ W})(0.08333°\text{C/ W}) = 14.2°\text{C}$$

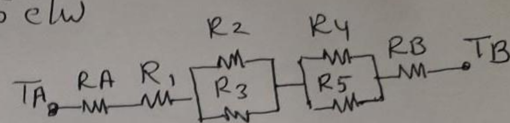
Example(15): Consider a composite wall containing 5-different materials as shown in the figure below. Draw the thermal circuit of the wall and calculate the rate of heat flow through the composite wall from the following data? $x_1 = 0.1m$, $x_2 = 0.2$ m , $x_3 = 0.15m$, $h_1 = 4m$, $h_2 = 1m$, $h_3 = 3m$, $h_4 = 2.5m$, $h_5 = 1.5m$, $k_1 = 15$ w/m °C , $k_2 = 25$ w/m°C , $k_3 = 30$ w/m °C, $k_4 = 20$ w/m°C , $k_5 = 35$ w/m °C , $T_A = 120°C$, $T_B = 50°C$, L=1m. Take the convection heat transfer coefficients on the inner and outer surfaces to be h_A= 10 W/m²·°C and h_B= 40 W/m²·°C.



$$R_{conv} = \frac{1}{hA}$$

$$R_A = \frac{1}{10*(4\times1)} = 0.025\ °C/W$$

$$R_B = \frac{1}{40*(4\times1)} = 6.25*10^{-3}\ °C/W$$



$$R_{cond} = \frac{\Delta x}{KA}$$

$$R_1 = \frac{0.1}{15*(4*1)} = 1.676*10^{-3}\ °C/W$$

$$R_2 = \frac{0.2}{25(1\times1)} = 8*10^{-3}\ °C/W$$

$$R_3 = \frac{0,2}{30(3*1)} = 2.2*10^{-3}\ °C/W$$

$$R_4 = \frac{0.15}{20(2.5*1)} = 3*10^{-3}\ °C/W$$

$$R_5 = \frac{0.15}{35(1.5*1)} = 2.86*10^{-3}\ °C/W$$

$$R_{mid_1} = \frac{R_2 R_3}{R_2+R_3} = 1.7*10^{-3}\ °C/W$$

$$R_{mid_2} = \frac{R_4 * R_5}{R_4+R_5} = 1.65*10^{-3}\ °C/W$$

$$R_{Total} = R_A+R_1+R_{mid_1}+R_{mid_2}+R_B = 0.036\ °C/W$$

$$\varphi = \frac{T_A-T_B}{R_{total}} = \frac{120-50}{0,036} = 1944.4\ W$$

5. Convection and Conduction in Series for a Cylindrical wall:

Now consider steady one-dimensional heat flow through a cylindrical that is exposed to convection on both sides to fluids at temperatures T_1 and T_2 with heat transfer coefficients $h1$ and $h2$, respectively, as shown in Fig. (3)

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}}$$

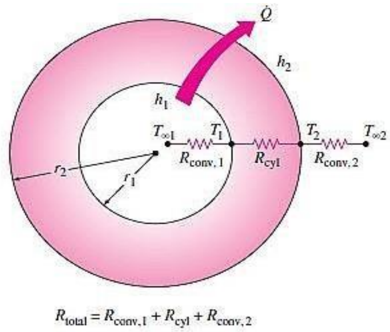


Fig.(14) The thermal resistance network for a cylindrical subjected to convection from both the inner and the outer side

Where:

$$R_{\text{total}} = R_{\text{conv},1} + R_{\text{cyl}} + R_{\text{conv},2} = \frac{1}{(2\pi r_1 L)h_1} + \frac{\ln(r_2/r_1)}{2\pi L k} + \frac{1}{(2\pi r_2 L)h_2}$$

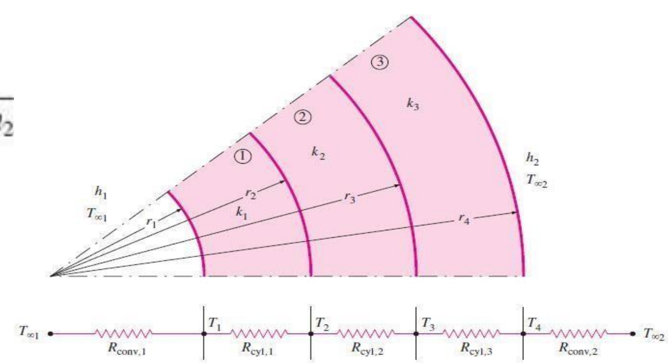


Fig.(15) The thermal resistance network for heat transfer through a three-layered composite cylinder subjected to convection on both sides

where R_{total} is the *total thermal resistance,* expressed as

$$R_{\text{total}} = R_{\text{conv},1} + R_{\text{cyl},1} + R_{\text{cyl},2} + R_{\text{cyl},3} + R_{\text{conv},2} = \frac{1}{h_1 A_1} + \frac{\ln(r_2/r_1)}{2\pi L k_1} + \frac{\ln(r_3/r_2)}{2\pi L k_2} + \frac{\ln(r_4/r_3)}{2\pi L k_3} + \frac{1}{h_2 A_4}$$

Example(16): A steel tube having k =46W/m· ◦C has an inside diameter of 3.0 cm and a tube wall thickness of 2 mm. A fluid flows on the inside of the tube producing a convection coefficient of 1500W/m^2 · ◦C on the inside surface, while a second fluid flows across the outside of the tube producing a convection coefficient of 197 W/m^2 · ◦C on the outside tube surface. The inside fluid temperature is 223◦C while the outside fluid temperature is 57◦C. Calculate the heat lost by the tube per meter of leng

$$\frac{1}{h_i A_i} = \frac{1}{(1500)\pi(0.03)} = 0.00709 \qquad r_0 = 0.015 + 0.002 = 0.017$$

$$\frac{\ln(r_0/r_i)}{2\pi k} = \frac{\ln(0.017/0.015)}{2\pi(46)} 0.000433$$

$$\frac{1}{h_0 A_0} = \frac{1}{(197)\pi(0.034)} = 0.0475$$

$$\sum R = 0.05502 \ \frac{°C \cdot m}{W}$$

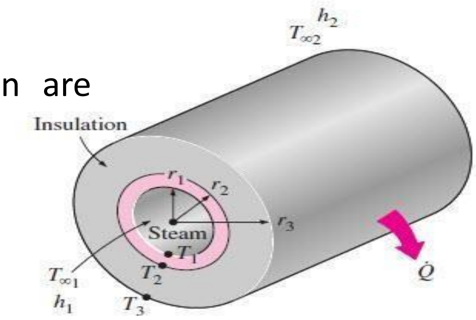
$$\frac{q}{L} = \frac{223 - 57}{0.05502} = 3017 \ W/m$$

Example(17): Steam at $T_{\infty 1}$ = 320°C flows in a cast iron pipe (k =80 W/m · °C) whose inner and outer diameters are D1 =5 cm and D2 =5.5 cm, respectively. The pipe is covered with 3-cm-thick glass wool insulation with k= 0.05 W/m·°C. Heat is lost to the surroundings at $T_{\infty 2}$ =5°C by natural convection, with a combined heat transfer coefficient of h2 =18 W/m^2·°C. Taking the heat transfer coefficient inside the pipe to be h1 =60 W/m^2·°C, determine the rate of heat loss from the steam per unit length of the pipe. Also determine the temperature drops across

the pipe shell and the insulation . take L= 1m

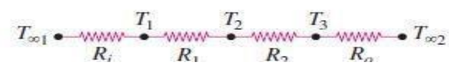
solution:

the area of the surface exposed to convection are determine to be



$$A_1 = 2\pi r_1 L = 2\pi(0.025\ \text{m})(1\ \text{m}) = 0.157\ \text{m}^2$$
$$A_3 = 2\pi r_3 L = 2\pi(0.0575\ \text{m})(1\ \text{m}) = 0.361\ \text{m}^2$$

Then the individual thermal resistances become



$$R_i = R_{\text{conv, 1}} = \frac{1}{h_1 A} = \frac{1}{(60\ \text{W/m}^2\cdot{}^\circ\text{C})(0.157\ \text{m}^2)} = 0.106^\circ\text{C/W}$$

$$R_1 = R_{\text{pipe}} = \frac{\ln(r_2/r_1)}{2\pi k_1 L} = \frac{\ln(2.75/2.5)}{2\pi(80\ \text{W/m}\cdot{}^\circ\text{C})(1\ \text{m})} = 0.0002^\circ\text{C/W}$$

$$R_2 = R_{\text{insulation}} = \frac{\ln(r_3/r_2)}{2\pi k_2 L} = \frac{\ln(5.75/2.75)}{2\pi(0.05\ \text{W/m}\cdot{}^\circ\text{C})(1\ \text{m})} = 2.35^\circ\text{C/W}$$

$$R_o = R_{\text{conv, 2}} = \frac{1}{h_2 A_3} = \frac{1}{(18\ \text{W/m}^2\cdot{}^\circ\text{C})(0.361\ \text{m}^2)} = 0.154^\circ\text{C/W}$$

Noting that all resistances are in series, the total resistance is determined to be

$$R_{\text{total}} = R_i + R_1 + R_2 + R_o = 0.106 + 0.0002 + 2.35 + 0.154 = 2.61^\circ\text{C/W}$$

Then the steady rate of heat loss from the steam becomes

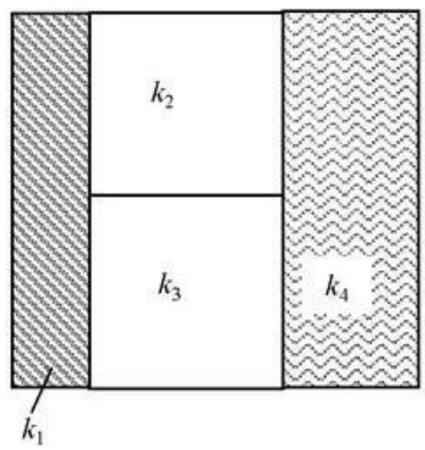
$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{(320-5)^\circ\text{C}}{2.61^\circ\text{C/W}} = 121\ \text{W} \qquad \text{(per m pipe length)}$$

The temperature drops across the pipe and the insulation are determine

$$\Delta T_{\text{pipe}} = \dot{Q}R_{\text{pipe}} = (121\ \text{W})(0.0002^\circ\text{C/W}) = 0.02^\circ\text{C}$$
$$\Delta T_{\text{insulation}} = \dot{Q}R_{\text{insulation}} = (121\ \text{W})(2.35^\circ\text{C/W}) = 284^\circ\text{C}$$

5/ HomeWorks:

1. Determine the rate of heat transfer through this section of the wall. And draw the thermal circuit.



k_1 = 60 w/m.^{0}c, k_2 = 30 w/m.°c,
k_3 = 20 w/m.°c, k_4 = 40 w/m.°c,
T_A = 250 °C, T_B = 100 °C, ΔX_1 = 0.01 m,
Δx_4 = 0.03 m, $\Delta x_2 = \Delta x_3$ = 0.05m,
Wide= 1 m, $h_1 = h_4$ = 0.8 m
$h_2 = h_3$ = 0.4 m, Take the convection heat transfer coefficients on the inner and outer surfaces to be h_A= 10 W/a^2·°C and h_B= 30 W/m^2·°C.

2. Steam at 320°C flows in a stainless steel pipe (k= 15 W/m · °C) whose inner and outer diameters are 5 cm and 5.5 cm, respectively. The pipe is covered with 3-cm-thick glass wool insulation (k =0.038 W/m·°C). Heat is lost to the surroundings at 5°C by natural convection, with natural convection heat transfer coefficient of 15 W/m^2·°C. Taking the heat transfer coefficient inside the pipe to be 80 W/m^2·°C, determine the rate of heat loss from the steam per unit length of the pipe. Also determine the temperature drops across the pipe shell and the insulation.

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

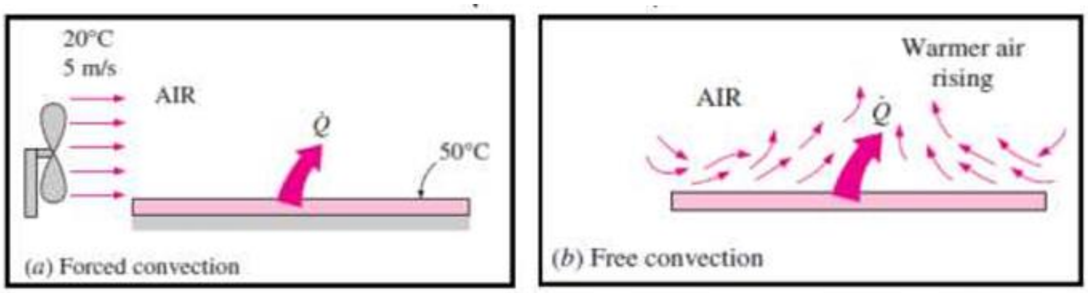
Learning package

In

Type of convection

For

Students of Second Year



(a) Forced convection (b) Free convection

By
Dr. Duna Tariq Yaseen
Assistant Professor
Dep. Of Power Mechanics Techniques
2025

1/ Overview

1 / A –Target population :-

For students of Second year
Technological institute of Basra
Dep. Of Power Mechanics Techniques

1 / B –Rationale :-

11 , 17^{th} week

The student understands analysis heat transfer with type of convection

1 / C –Central Idea :-

The student understands free and forced convection

1 / D – Performance Objectives

Learn how to solve problems related to heat transfer outside and inside tube.

2/ Pretest :

Complete the sentences with the correct answer

1- When two fluid layers move relative to each other, the frictional force between them is called ……………

(Viscous flow, Inviscid flow)

2-When the value of the Reynolds number is more significant than 10000, the flow is ……………

(Laminar, Turbulent, Transitional)

3- All fluid properties are to be evaluated at the film temperature, which is…………

(T_f = 1/2 (T_s + T_∞), (T_i + T_o) .

4- For flow in a circular tube, the Reynolds number is defined as……………

(a) $Re = \frac{\rho v d}{\mu}$,(b) $Re = \frac{\rho v L}{\mu}$, (c) $Re = \frac{vL}{\nu}$

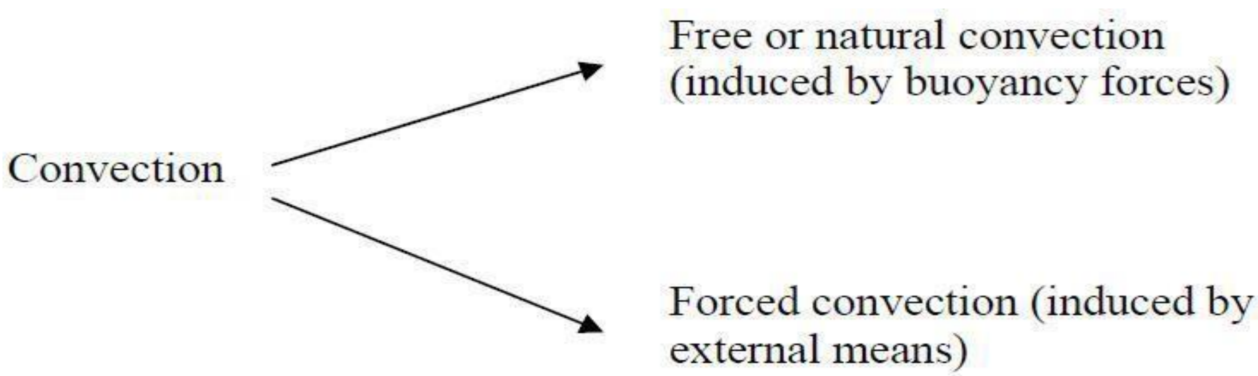
3/ Scientific content:-

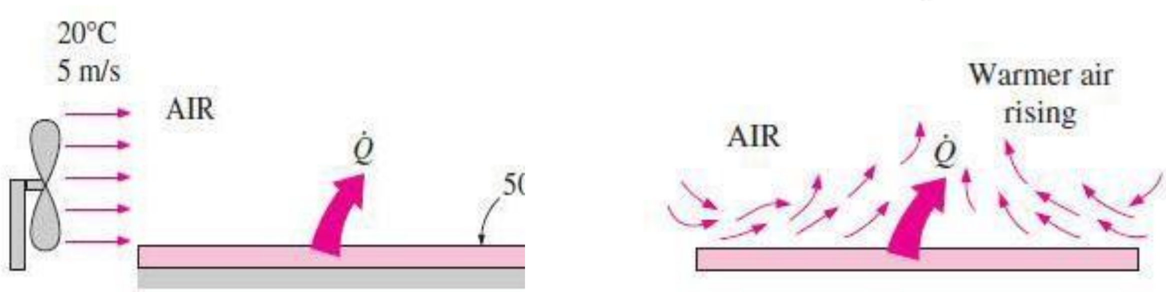
CHAPTER 5

Heat transfer by Convection

1. Introduction:

- Convection is energy transfer that takes place between a surface and a fluid moving over it when they are at different temperatures.
- Convection is usually the dominant form of heat transfer in liquids and gases.
- There are two type of convection





(a) Forced convection (b) Free convection

- All types of convection are governed by ***Newton's law of cooling*** "The rate of heat loss of a body is proportional to the difference in temperatures between the body and its surroundings" : $Q_{conv.} = h\ A\ (T_s - T_\infty)$

Where: h is the convection heat transfer coefficient in W/$m2$ °C

A: is the surface area through which convection heat transfer takes place in ($m2$)

TS is the surface temperature in (°C)

T∞ is the temperature of the fluid sufficiently far from the surface in (°C) 2

- ***Convection heat transfer coefficient depends on:***

1. The Fluid properties (density, thermal conductivity, specific heat and dynamic viscosity).

2. The geometry of the surface (flat plate, circular cylinder, sphere, etc...)

3. Fluid velocity.

4. The roughness of the solid surface.

5. The type of fluid flow (forced, natural, mixed (combined) convection as well as laminar, turbulent and transitional flows).

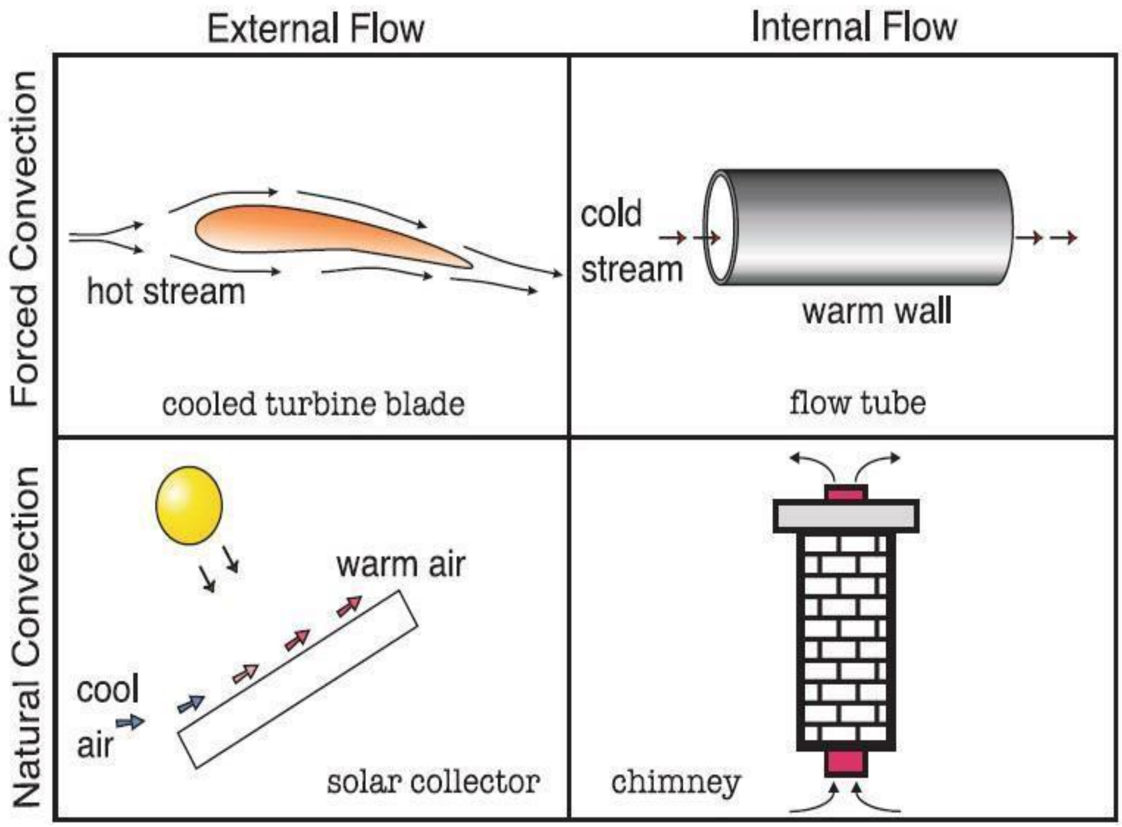
2. Common classifications of convection:

A. Based on geometry: External flow / Internal flow

B. Based on driving mechanism Natural convection / forced convection / mixed convection

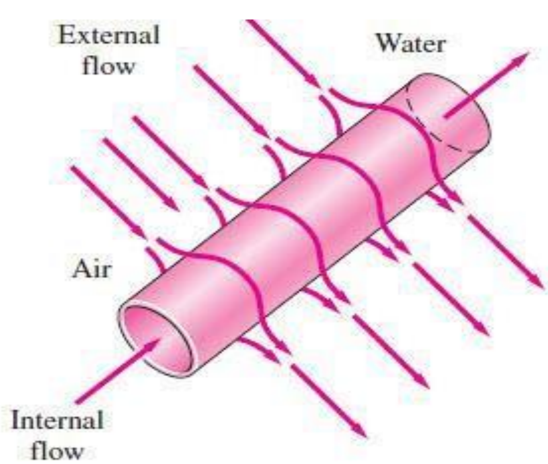
C. Based on number of phases Single phase / multiple phase

D. Based on nature of flow Laminar / turbulent



1. External flow / Internal flow:

A fluid flow is classified as being internal and external, depending on whether the fluid is forced to flow in a confined channel or over a surface. The flow of an unbounded fluid over a surface such as a plate, a wire, or pipe is external flow. The flow in a pipe or duct is internal flow if the fluid is completely bounded by solid surfaces.

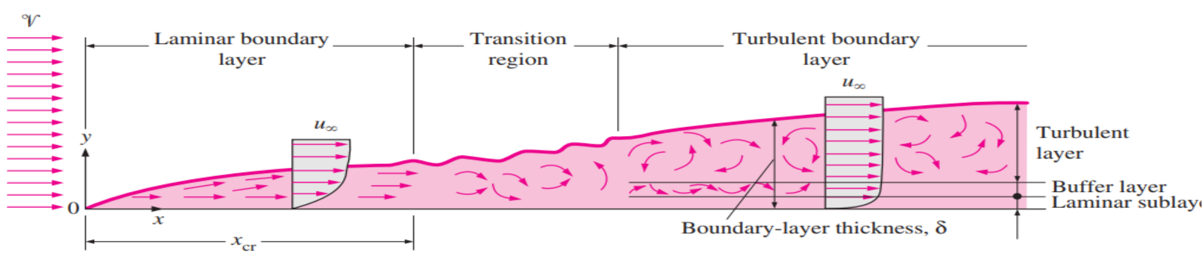


2. Natural convection / forced convection:

A fluid flow is said to be natural or forced, depending on how the fluid motion is initiated. In forced flow, a fluid is forced to flow over a surface or in a pipe by external means such as a pump or a fan. In natural flows, any fluid motion is due to a natural means .

3. Laminar / turbulent flow:

The highly ordered fluid motion characterized by smooth streamlines is called laminar. The highly disordered fluid motion that typically occurs at high velocities is called turbulent. The transition from laminar to turbulent flow depends on the surface geometry, surface roughness, free-stream velocity, surface temperature, and type of fluid, among other things.



4. Steady flow / Unsteady (Transient) Flow:

The term steady implies no change with time. The opposite of steady is unsteady, or transient. During steady flow, the fluid properties can change from point to point within a device, but at any fixed point they remain constant.

5. Compressible / Incompressible Flow:

A fluid flow is classified as being compressible or incompressible, depending on the density variation of the fluid during flow. The densities of liquids are essentially constant, and thus the flow of liquids is typically incompressible. Gases, on the other hand, are highly compressible.

6. One-, Two-, and Three-Dimensional Flows:

A flow field is best characterized by the velocity distribution, and thus a flow is said to be one-, two-, or three-dimensional if the flow velocity varies in one, two, or three primary dimensions.

7. Viscous / Inviscid Flow:

When two fluid layers move relative to each other, a friction force develops between them and the slower layer tries to slow down the faster layer. This internal resistance to flow is called the viscosity. Flows in which the effects of viscosity are significant are called viscous flows. The effects of viscosity are very small in some flows, and neglecting those effects are called inviscid flows.

Dimensionless groups:

Non- Dimension group are used to simplify analysis, and describe the physical situation without referring to units. A dimensionless quantity has no physical unit.

Reynolds Number:

Reynolds number is used in the study of fluid flows. It compares the relative strength of inertial and viscous effects. The value of the Reynolds number is defined as

$$Re = \frac{\text{Inertia forces}}{\text{Viscous}} = \frac{\mathcal{V} L_c}{\nu} = \frac{\rho \mathcal{V} L_c}{\mu}$$

where ρ(rho) is the density, μ(mu) is the absolute viscosity, V is the velocity of the flow, L is the length of the flow, and we define a parameter ν(nu) as the kinematic viscosity.

In boundary layer flow over a flat plate, experiments confirm that a laminar boundary layer will become unstable and turbulent after a certain length of flow. This instability occurs across different scales and with different fluids, usually when

$Re \approx 5\times10^5$

- ρ(rho) The density of a fluid is defined as the mass of the fluid over an infinitesimal volume
- ρ =mass/Volume=kg/m3
- μ(mu) is viscosity, is the property of a fluid, due to cohesion and interaction between molecules, which offers resistance to sheer deformation. ($N.sec/m2$)

- v(nu) is the kinematic viscosity, is defined as the ratio of dynamic viscosity to mass density.

$$v = \frac{\mu}{\rho} = \frac{m^2}{sec}$$

Prandtl number:

It is the ratio of the momentum diffusivity to the thermal diffusivity:

$$\Pr = \frac{v}{\alpha} = \frac{cp}{k}$$

Nusselt number:

this can be considered as the dimensionless heat transfer coefficient: it is the ratio of convection to pure conduction heat transfer

$$\mathrm{Nu} = \frac{hL_c}{k}$$

$$\dot{q}_{\text{conv}} = h\Delta T$$

$$\dot{q}_{\text{cond}} = k\frac{\Delta T}{L}$$

Taking their ratio gives

$$\frac{\dot{q}_{\text{conv}}}{\dot{q}_{\text{cond}}} = \frac{h\Delta T}{k\Delta T/L} = \frac{hL}{k} = \mathrm{Nu}$$

Example1: An average man has a body surface area of 1.8 m2 and the average surface temperature of the clothed person to be 30°C. The convection heat transfer coefficient for a clothed person walking in still air is expressed as

$h = 8.6\ v^{0.53}$ for $0.5 < V < 2$ m/s, where V is the walking velocity in m/s. Determine the rate of heat loss from an average man walking in still air at 10°C by convection at a walking velocity of (a) 0.5 m/s, (b) 1.0 m/s, (c) 1.5 m/s, and (d) 2.0 m/s.

Solution: The convection heat transfer coefficients and the rate of heat losses at different walking velocities are

(a) $h = 8.6\mathbf{V}^{0.53} = 8.6(0.5\,\text{m/s})^{0.53} = 5.956\ \text{W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (5.956\ \text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{214.4\ W}$

(b) $h = 8.6\mathbf{V}^{0.53} = 8.6(1.0\,\text{m/s})^{0.53} = 8.60\ \text{W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (8.60\ \text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{309.6\ W}$

(c) $h = 8.6\mathbf{V}^{0.53} = 8.6(1.5\,\text{m/s})^{0.53} = 10.66\ \text{W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (10.66\ \text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{383.8\ W}$

(d) $h = 8.6\mathbf{V}^{0.53} = 8.6(2.0\,\text{m/s})^{0.53} = 12.42\ \text{W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (12.42\ \text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{447.0\ W}$

Example2: Evaluate the Reynolds number for flow over a tube from the following data: D =6 cm, $U\infty$= 1.0 m/s, ρ = 300 kg/m3, μ = 0.04 N.s/m2.

solution : The Reynolds number

$$Re = \frac{U_\infty L}{\nu} = \frac{U_\infty L\rho}{\mu}$$

The Reynolds number based on the tube diameter is

$$Re = \frac{U_\infty D\rho}{\mu} = \frac{(1\,\text{m/s})(6\,\text{cm})(1\,\text{m}/(100\,\text{cm}))\left(300\,\text{kg/m}^3\right)}{\left(0.04(\text{Ns})/\text{m}^2\right)\left(\text{kg m}/(\text{s}^2\text{N})\right)} = 450$$

Example 3: A fluid flows at 5 m/s over a wide, flat plate 15 cm long. For each from the following list, calculate the Reynolds number at the downstream end of the plate. Indicate whether the flow at that point is laminar, transition, or turbulent. Assume all fluids are at 40°C. (a) air, (b) CO_2, (c) water, (d) engine oil.

for Air (ν_a) = 17.6×10^{-6} m^2/s
for CO_2 (ν_c) = 9.07×10^{-6} m^2/s
for Water (ν_w) = 0.658×10^{-6} m^2/s
for Engine Oil (ν_o) = 240×10^{-6} m^2/s

Solution: At 40°C, the kinematic viscosities of the given fluids are as follows

The Reynolds number

$$Re = \frac{U_\infty L}{\nu}$$

The transition from laminar to turbulent flow over a plate occurs at a Reynolds number of about $5 * 10^5$

For air

$$Re_L = \frac{(5\,\mathrm{m/s})(0.15\,\mathrm{m})}{17.6\times10^{-6}\,\mathrm{m^2/s}} = 4.3 \times 10^4 \text{ (Laminar)}$$

For CO_2

$$Re_L = \frac{(5\,\mathrm{m/s})(0.15\,\mathrm{m})}{9.07\times10^{-6}\,\mathrm{m^2/s}} = 8.3 \times 10^4 \text{ (Laminar)}$$

For water

$$Re_L = \frac{(5\,\mathrm{m/s})(0.15\,\mathrm{m})}{0.658\times10^{-6}\,\mathrm{m^2/s}} = 1.1 \times 10^6 \text{ (Turbulent)}$$

For engine oil

$$Re_L = \frac{(5\,\mathrm{m/s})(0.15\,\mathrm{m})}{240\times10^{-6}\,\mathrm{m^2/s}} = 3.1 \times 10^3 \text{ (Laminar)}$$

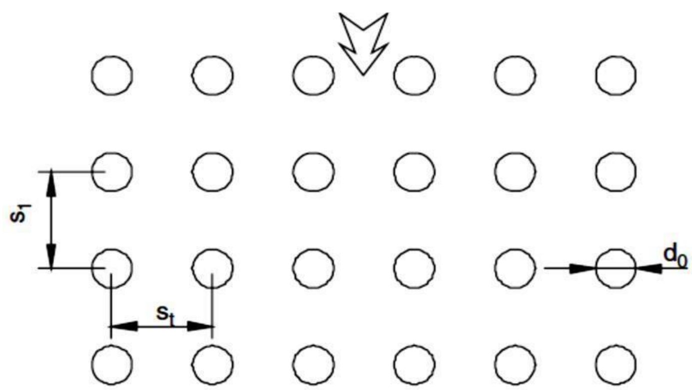
.

Chapter 6

Forced Convection outside a Tube Bank

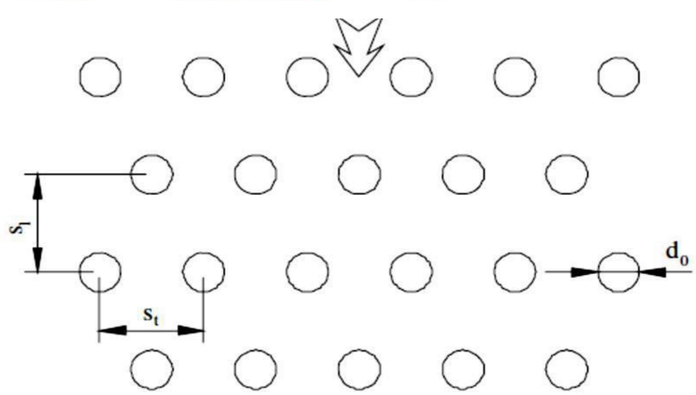
1. **Arrangement with in line tubes**

$$Nu = 0.26Re^{0.6}$$



2: Arrangement with staggered tubes

$$Nu = 0.33Re^{0.6}\,Pr^{1/3}$$



$$Re = \frac{v_{max} \; d}{v}$$

V_{max} is the maximum velocity experienced by the fluid as it passes through the bank(fluid mean velocity through smallest section area for free spaces between the tubes)

Example (1) Calculate the convection heat transfer coefficient for water flowing normally over a bank of tubes with outer diameter 0f (5 cm) the mean water velocity through the smallest section area for free spaces equal to 52cm /s, assume the mean temperature for the boundary layer is (50°C) , if (K=0.639W/m°C , pr=3.68 , v=0.00569 cm2/s)

Solution

$$Nu = 0.33 Re^{0.6} Pr^{1/3}$$

$$Nu = 0.33 \frac{(0.052 x 0.05)^{0.6}}{0.00569^{-4}} (3.68)^{1/3} = 318.5$$

N_u= hd/k

h=0.639×318.5/0.05 = 4070 w/m2°C

Heat Transfer to a Single Tube

$$Nu = 0.35 Re^{0.56}.$$

All physical properties are assigned at mean temperature

Example (2): Calculate the convection heat transfer coefficient for the air which flows normally over a tube (D=2.5 cm) the (Re= 8000) and the mean temperature for the tube surface is(84°C) , K=0.0305 w/m°C.

Solution:

$$Nu = 0.35Re^{0.56}.$$

$N_u = 0.35(8000)^{0.56} = 53.67$

$Nu = hD/K \quad = 53.67 \times 0.0305/0.025 = 65.49 \text{ w/m2°C}$

Example3: calculate the mean temperature and the convection heat transfer coefficient of water flowing inside a tube have(5cm) inner diameter. the water enters at(27°C) with mass flow rate of(2192 kg/hr) and leaves at (49°C) ,if at the mean temperature (k=0.6241 w/m°C , P_r= 4.618 , ρ=992.71 kg/m³ , ν = 0.006928 cm² /s)

Solution

$\dot{m} = \rho A V$, $V = \dfrac{\dot{m}}{\rho A}$

$$V = \frac{2129}{3600 \times 992.71 \times \pi \times (0.025)^2} = 0.312 \text{ m/s}$$

$$R_e = \frac{DV}{\nu} = \frac{0.05 \times 0.312}{0.006928 \times 10^{-4}} = 22517.321$$

$N = 0.023(Re)^{0.8}(Pr)^{0.4}$

$$N_U = 0.023(22517.321)^{0.8}(4.618)^{0.4} = 128.677$$

$$N_U = \frac{hD}{K}$$

$$h = \frac{K.N_U}{D} = \frac{128.677 \times 0.6241}{0.05} = 1606.146\, w$$

Internal Forced Convection Heat Transfer

Mechanism of Forced Convection:

- Forced convection: in which the motion in the fluid medium is generated by the application of an external force, e.g. by a pump, blower, etc.
- The rate of convection heat transfer is expressed by Newton's law of cooling

$Q_{conv.}$=h A (TS−T∞) (Watt)

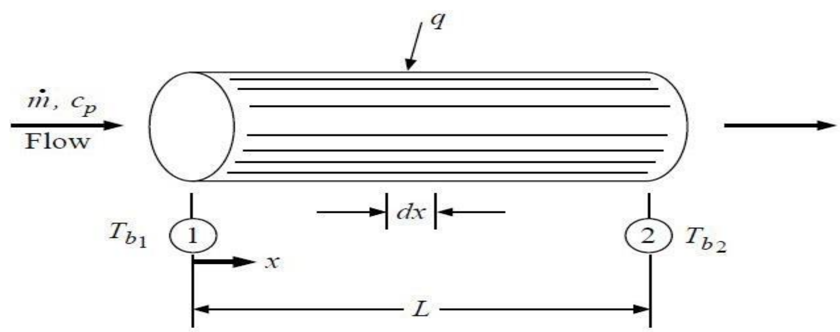
The convective heat transfer coefficient h strongly depends on the fluid properties and roughness of the solid surface, and the type of the fluid flow (laminar or turbulent).

The Bulk Temperature:

The bulk temperature represents energy average conditions. The total energy added can be expressed in terms of a bulk-temperature difference by

$$q = \dot{m} c_p (T_{b_2} - T_{b_1})$$

cp is reasonably constant over the length.



The total heat transfer can also be expressed as

$$q = hA(T_w - T_b)_{av}$$

Where *A* is the total surface area for heat transfer. Because both *Tw* and *Tb* can vary along the length of the tube.

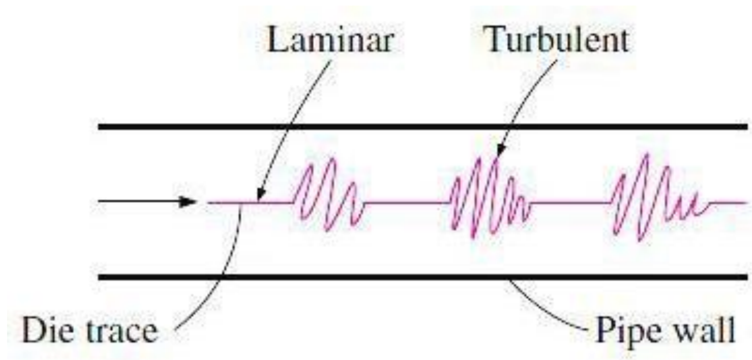
Laminar and Turbulent Flow in Tubes:

Flow in a tube can be laminar or turbulent, depending on the flow conditions. Fluid flow is streamlined and thus laminar at low velocities, but turns turbulent as the velocity is increased beyond a critical value.

For flow in a circular tube, the Reynolds number is defined as

$$\text{Re} = \frac{\rho \mathcal{V}_m D}{\mu} = \frac{\mathcal{V}_m D}{\nu}$$

$$\begin{array}{ll} \text{Re} < 2300 & \text{laminar flow} \\ 2300 \leq \text{Re} \leq 10{,}000 & \text{transitional flow} \\ \text{Re} > 10{,}000 & \text{turbulent flow} \end{array}$$



Empirical Relations for Pipe and Tube Flow:

- Fully developed turbulent flow in smooth tubes is that recommended by Dittus and Boelter:

$$Nu_d = 0.023\, Re_d^{0.8} Pr^n$$

$$n = \begin{cases} 0.4 & \text{for heating of the fluid} \\ 0.3 & \text{for cooling of the fluid} \end{cases}$$

The properties in this equation are evaluated at the average fluid bulk temperature.

Example1:

Water is to be heated from 15°C to 65°C as it flows through a 3 cm internal diameter 5 m long tube. The tube is equipped with a heater that provides uniform heating throughout the surface of the tube. The outer surface of the heater is well insulated. If the system is to provide hot water at a rate of 10 L/min, determine the power rating of the heater. Also, estimate the inner surface temperature of the pipe at the exit.

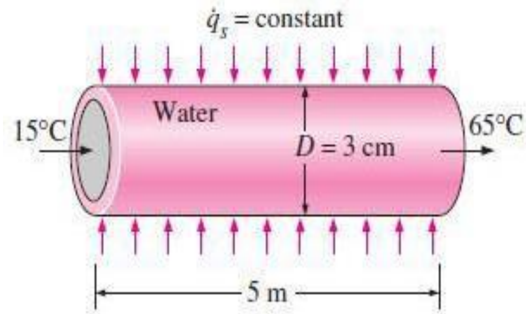
properties of water at bulk mean temperature of

$T_b = (T_i + T_e) / 2 = (15+65)/ 2 = 40$ °C are (table A.9).

$\rho = 992.1$ lg/m^3 cp = 4179j/kg.°C

k= o.631 w/m. °C Pr = 4.32, $\mu = 0.6 \times 10^{-3}$ kg/m.s

$\nu = \mu/\rho = 0.658 \times 10^{-6}$ m^2/s



Solution:

Analysis The cross sectional and heat transfer surface areas are

$$A_c = \tfrac{1}{4}\pi D^2 = \tfrac{1}{4}\pi(0.03\ \text{m})^2 = 7.069 \times 10^{-4}\ \text{m}^2$$
$$A_s = pL = \pi DL = \pi(0.03\ \text{m})(5\ \text{m}) = 0.471\ \text{m}^2$$

The volume flow rate of water is given as $\dot{V} = 10$ L/min $= 0.01$ m^3/min. Then the mass flow rate becomes

$$\dot{m} = \rho\dot{V} = (992.1\ \text{kg/m}^3)(0.01\ \text{m}^3/\text{min}) = 9.921\ \text{kg/min} = 0.1654\ \text{kg/s}$$

To heat the water at this mass flow rate from 15°C to 65°C, heat must be supplied to the water at a rate of

$$\begin{aligned}\dot{Q} &= \dot{m}C_p(T_e - T_i)\\ &= (0.1654\ \text{kg/s})(4.179\ \text{kJ/kg}\cdot{}^\circ\text{C})(65 - 15)^\circ\text{C}\\ &= 34.6\ \text{kJ/s} = 34.6\ \text{kW}\end{aligned}$$

The surface temperature T_s of the tube at any location can be determined from

$$\dot{q}_s = h(T_s - T_m) \quad \rightarrow \quad T_s = T_m + \frac{\dot{q}_s}{h}$$

where h is the heat transfer coefficient and T_m is the mean temperature of the fluid at that location. The surface heat flux is constant in this case, and its value can be determined from

$$\dot{q}_s = \frac{\dot{Q}}{A_s} = \frac{34.6\ \text{kW}}{0.471\ \text{m}^2} = 73.46\ \text{kW/m}^2$$

To determine the heat transfer coefficient, we first need to find the mean velocity of water and the Reynolds number:

$$\mathcal{V}_m = \frac{\dot{V}}{A_c} = \frac{0.010 \text{ m}^3/\text{min}}{7.069 \times 10^{-4} \text{ m}^2} = 14.15 \text{ m/min} = 0.236 \text{ m/s}$$

$$\text{Re} = \frac{\mathcal{V}_m D}{\nu} = \frac{(0.236 \text{ m/s})(0.03 \text{ m})}{0.658 \times 10^{-6} \text{ m}^2/\text{s}} = 10{,}760$$

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} = 0.023(10{,}760)^{0.8} (4.34)^{0.4} = 69.5$$

Then,

$$h = \frac{k}{D}\text{Nu} = \frac{0.631 \text{ W/m} \cdot {}^\circ\text{C}}{0.03 \text{ m}}(69.5) = 1462 \text{ W/m}^2 \cdot {}^\circ\text{C}$$

and the surface temperature of the pipe at the exit becomes

and the surface temperature of the pipe at the exit becomes

$$T_s = T_m + \frac{\dot{q}_s}{h} = 65^\circ\text{C} + \frac{73{,}460 \text{ W/m}^2}{1462 \text{ W/m}^2 \cdot {}^\circ\text{C}} = \mathbf{115^\circ C}$$

Example2: *Air at 200◦C is heated as it flows through a tube with a diameter of 2.54cm at a velocity of 10 m/s. Calculate the heat transfer per unit length of tube if a constant heat flux condition is maintained at the wall and the wall temperature is 20◦C above the air temperature, all along the length of the tube. How much would the bulk temperature increase over a 3-m length of the tube?* The properties of air at 200°C are; ρ=1.493 kg/m3

$$\mathrm{Pr} = 0.681$$
$$\mu = 2.57 \times 10^{-5}\ \mathrm{kg/m \cdot s}$$
$$k = 0.0386\ \mathrm{W/m \cdot {}^\circ C}$$
$$c_p = 1.025\ \mathrm{kJ/kg \cdot {}^\circ C}$$

Solution:

$$\mathrm{Re}_d = \frac{\rho u_m d}{\mu} = \frac{(1.493)(10)(0.0254)}{2.57 \times 10^{-5}} = 14{,}756$$

$$\mathrm{Nu}_d = \frac{hd}{k} = 0.023\,\mathrm{Re}_d^{0.8}\,\mathrm{Pr}^{0.4} = (0.023)(14{,}756)^{0.8}(0.681)^{0.4} = 42.67$$

$$h = \frac{k}{d}\mathrm{Nu}_d = \frac{(0.0386)(42.67)}{0.0254} = 64.85\ \mathrm{W/m^2 \cdot {}^\circ C} \qquad [11.42\ \mathrm{Btu/h \cdot ft^2 \cdot {}^\circ F}]$$

The heat flow per unit length is then $q = h(\pi D L)\,\Delta T$

1) $$\frac{q}{L} = h\pi d(T_w - T_b) = (64.85)\pi(0.0254)(20) = 103.5\ \mathrm{W/m} \qquad [107.7\ \mathrm{Btu/ft}]$$

We also have

$$\dot{m} = \rho u_m \frac{\pi d^2}{4} = (1.493)(10)\pi\frac{(0.0254)^2}{4}$$
$$= 7.565 \times 10^{-3}\ \mathrm{kg/s} \qquad [0.0167\ \mathrm{lb}_m/\mathrm{s}]$$

Over a 3 m length of the tube

2) $Q = q/L \times L = 103.5 \times 3 = 310.5\ Watt$

$Q = \dot{m}(Tw - Tb) = \dot{m}cp(\Delta Tb)$

$310.5 = 7.565 \times 10^{-3} \times 1025 \times \Delta Tb$

$\Delta Tb = 40.04$ °C

Example3: *Water at the rate of 3 kg/s is heated from 5 to 15◦C by passing it through a 5 cm ID copper tube. The tube wall temperature is maintained at 90◦C What is the length of the tube?*

$$q = (3)(4175)(15 - 5) = 125{,}850\ \mathrm{W} \quad \text{at } 10^\circ\mathrm{C} \qquad \mu = 1.31 \times 10^{-3}$$

$$k = 0.585 \qquad \mathrm{Pr} = 9.40 \qquad \mathrm{Re} = \frac{(0.05)(3)(4)}{\pi(0.05)^2(1.31 \times 10^{-3})} = 58{,}316$$

$$\bar{h} = \frac{0.585}{0.05}(0.023)(58{,}316)^{0.8}(9.4)^{0.4} = 4283\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot {}^\circ C}}$$

$$q = 125{,}85\underline{0} = (4283)\pi(0.05)L(90 - 10) \qquad L = 2.338\ \mathrm{m}$$

ex4: a liquid ammonia flows through a (0.025)m diameter smoth tube 2.5m long at arate of 0.4 kg/s. the ammonia enter at 10 °C and leave at 38 °C , and constant the average flux is   imposed on the tube wall. calculate the average wall temperature.

$$\dot{m} = 0.4 \ \text{kg/sec} \qquad T_{b_{avg}} = \frac{10+38}{2} = \underline{\underline{24}}\,°\text{C} \qquad \rho = 605.6$$

$$\nu = 0.355 \times 10^{-6} \qquad k = 0.515 \qquad c_p = 4840 \qquad \text{Pr} = 2.02$$

$$\text{Re} = \frac{(0.025)(0.4)(4)}{\pi(0.025)^2(0.355\times 10^{-6})(605.6)} = 1.0 \times 10^5$$

$$h = \frac{0.515}{0.025}(0.023)(1.0\times 10^5)^{0.8}(2.02)^{0.4} = 6277 \ \frac{\text{W}}{\text{m}^2 \cdot °\text{C}}$$

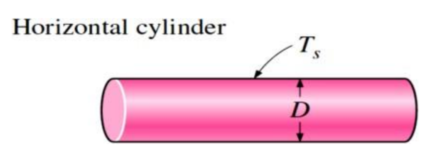
$$q = (0.4)(4840)(38-10) = 54{,}200 \ \text{W} = (6277)\pi(0.025)(2.5)(T_w - 24)$$

$$T_w = 68°\text{C}$$

Heat Transfer by Free Convection

Heat Transfer from Horizontal Cylinder Surface:

$$Ra_D = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}Pr = \frac{g\beta(T_s - T_\infty)D^3}{\nu\alpha}$$

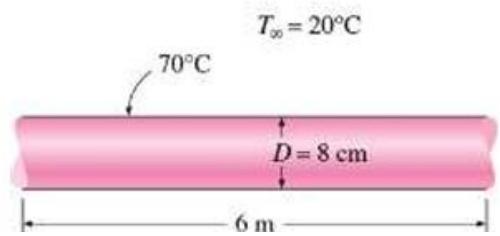


$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+(0.559/\text{Pr})^{9/16}\right]^{8/27}}\right\}^2$$

Example (3): *A 6 m long section of an 8 cm diameter horizontal hot water pipe shown in Figure passes through a large room whose temperature is 20°C. If the outer surface temperature of the pipe is 70°C, determine the rate of heat loss from the pipe by natural convection. The properties of air at the film temperature*

$T_f = 45+273= 318k$

Properties The properties of air at the film temperature of $T_f = (T_s + T_\infty)/2 = (70+20)/2 = 45^\circ C$



$k = 0.02699\ \text{W/m}\cdot{}^\circ\text{C}$ $\qquad$ $\text{Pr} = 0.7241$

$\nu = 1.749 \times 10^{-5}\ \text{m}^2/\text{s}$ $\qquad$ $\beta = \dfrac{1}{T_f} = \dfrac{1}{318\ \text{K}}$

$$\text{Ra}_D = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\text{Pr}$$

$$= \frac{(9.81\ \text{m/s}^2)[1/(318\ \text{K})](70 - 20\ \text{K})(0.08\ \text{m})^3}{(1.749 \times 10^{-5}\ \text{m}^2/\text{s})^2}(0.7241) = 1.869 \times 10^6$$

$$\text{Nu} = \left\{0.6 + \frac{0.387\ \text{Ra}_D^{1/6}}{[1 + (0.559/\text{Pr})^{9/16}]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(1869 \times 10^6)^{1/6}}{[1 + (0.559/0.7241)^{9/16}]^{8/27}}\right\}^2$$

$$= 17.40$$

Then,

$$h = \frac{k}{D}\text{Nu} = \frac{0.02699\ \text{W/m}\cdot{}^\circ\text{C}}{0.08\ \text{m}}(17.40) = 5.869\ \text{W/m}\cdot{}^\circ\text{C}$$

$$A_s = \pi DL = \pi(0.08\ \text{m})(6\ \text{m}) = 1.508\ \text{m}^2$$

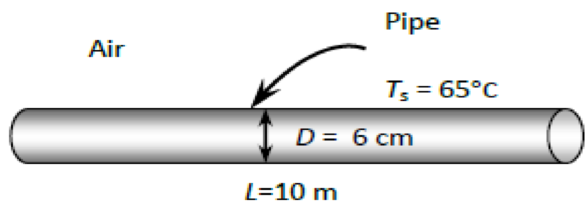
and

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.869\ \text{W/m}^2\cdot{}^\circ\text{C})(1.508\ \text{m}^2)(70 - 20)^\circ\text{C} = 443\ \text{W}$$

Example (4): *A 10 m long section of a 6 cm diameter horizontal hot water pipe passes through a large room whose temperature is 22°C. If the temperature of the outer surface of the pipe is 65°C, determine the rate of heat loss from the pipe by natural convection? The properties of air at the film temperature:*

k = 0.02688W/m.°C, υ = 1.735×10–5 m 2 /s, Pr = 0.7245

solution



Tf = (22+65)/2= 43.5 ºC , T_f= 43.5+273 = 316.5 K

$\beta = 1/T_f$

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\upsilon^2}\Pr$$
$$= \frac{(9.81\,\text{m/s}^2)(0.00316\text{K}^{-1})(65-22\,\text{K})(0.06\,\text{m})^3}{(1.735\times10^{-5}\,\text{m}^2/\text{s})^2}(0.7245) = 692{,}805$$

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+(0.559/\Pr)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(692{,}805)^{1/6}}{\left[1+(0.559/0.7245)^{9/16}\right]^{8/27}}\right\}^2 = 13.15$$

$$h = \frac{k}{D}Nu = \frac{0.02688\,\text{W/m.°C}}{0.06\,\text{m}}(13.15) = 5.893\,\text{W/m}^2.\text{°C}$$

$$A_s = \pi DL = \pi(0.06\,\text{m})(10\,\text{m}) = 1.885\,\text{m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.893\,\text{W/m}^2.\text{°C})(1.885\,\text{m}^2)(65-22)\text{°C} = \mathbf{477.6\,W}$$

Heat Transfer by Free Convection

Dimensionless groups:

1- The Grashof Number

The dimensionless parameter in the brackets represents the natural convection effects, and is called the **Grashof number** Gr,

$$\mathrm{Gr}_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}$$

where

g = gravitational acceleration, m/s^2
β = coefficient of volume expansion, 1/K ($\beta = 1/T$ for ideal gases)
T_s = temperature of the surface, °C
T_∞ = temperature of the fluid sufficiently far from the surface, °C
L_c = characteristic length of the geometry, m
ν = kinematic viscosity of the fluid, m^2/s

2- Rayleigh number

It is the product of the Grashof and Prandtl numbers:

$$\mathrm{Ra}_L = \mathrm{Gr}_L \Pr = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\Pr$$

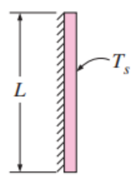
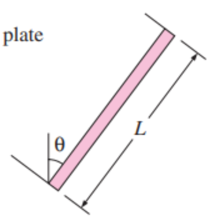
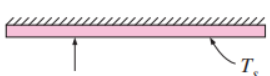
$$Pr = \frac{v}{\alpha}$$

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L^3}{\nu\alpha}$$

Heat Transfer from Vertical and Horizontal surfaces:

Simple relations for the average Nusselt number for various geometries are given in table below. Also given in this table are the characteristic lengths of the geometries and the ranges of Rayleigh number in which the relation is applicable. All fluid properties are to be evaluated at the film temperature Tf = 1/2 (Ts +T∞)

Empirical correlations for the average Nusselt number for natural convection over surfaces

Geometry	Characteristic length L_c	Range of Ra	Nu	
Vertical plate 	L	10^4–10^9 10^9–10^{13} Entire range	$Nu = 0.59Ra_L^{1/4}$ $Nu = 0.1Ra_L^{1/3}$ $Nu = \left\{0.825 + \frac{0.387Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}}\right\}^2$ (complex but more accurate)	(9-19) (9-20) (9-21)
Inclined plate 	L		Use vertical plate equations for the upper surface of a cold plate and the lower surface of a hot plate Replace g by $g\cos\theta$ for $Ra < 10^9$	
Horiontal plate (Surface area A and perimeter p) (*a*) Upper surface of a hot plate (or lower surface of a cold plate)  (*b*) Lower surface of a hot plate (or upper surface of a cold plate) 	A_s/p	10^4–10^7 10^7–10^{11} 10^5–10^{11}	$Nu = 0.54Ra_L^{1/4}$ $Nu = 0.15Ra_L^{1/3}$ $Nu = 0.27Ra_L^{1/4}$	(9-22) (9-23) (9-24)

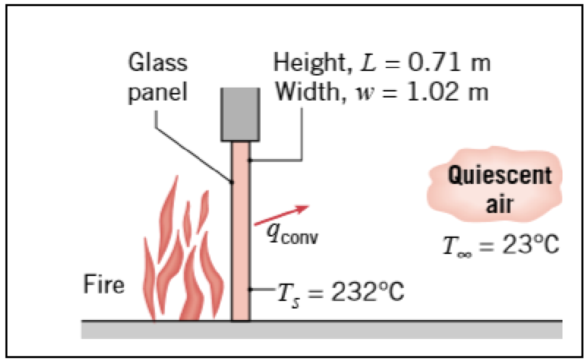
Example 1

A glass-door firescreen, used to reduce exfiltration of room air through a chimney, has a height of 0.71 m and a width of 1.02 m and reaches a temperature of 232°C. If the room temperature is 23°C, estimate the convection heat rate from the fireplace to the room

SOLUTION

Assumptions:

1. Screen is at a uniform temperature Ts.
2. Room air is quiescent



Properties: Table A.4, air (T_f = 400 K): $k = 33.8 \times 10^{-3}$ W/m · K, $\nu = 26.4 \times 10^{-6}$ m²/s, $\alpha = 38.3 \times 10^{-6}$ m²/s, $Pr = 0.690$, $\beta = (1/T_f) = 0.0025$ K^{-1}.

Analysis: The rate of heat transfer by free convection from the panel to the room is given by Newton's law of cooling

$$q = hA_s\,(T_s - T_\infty)$$

where h, may be obtained from knowledge of the Rayleigh number :

$$Ra_L = \frac{g\beta(T_s - T_\infty)L^3}{\alpha\nu}$$

$$= \frac{9.8\text{ m/s}^2 \times 1/400\text{ K}\,(232 - 23)^\circ\text{C} \times (0.71\text{ m})^3}{38.3 \times 10^{-6}\text{ m}^2/\text{s} \times 26.4 \times 10^{-6}\text{ m}^2/\text{s}} = 1.813 \times 10^9$$

$Nu_{L=}0.10\ (1.813 \times 10^9)^{1/3}$, $Nu_l = 147$

$$\bar{h} = \frac{\overline{Nu_L} \cdot k}{L} = \frac{147 \times 33.8 \times 10^{-3}\text{ W/m} \cdot \text{K}}{0.71\text{ m}} = 7.0\text{ W/m}^2 \cdot \text{K}$$

$$q = 7.0\text{ W/m}^2 \cdot \text{K}\,(1.02 \times 0.71)\text{ m}^2\,(232 - 23)^\circ\text{C} = 1060\text{ W}$$

Example 2

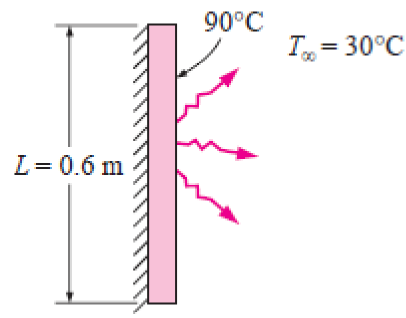
Consider a 0.6-m × 0.6-m thin square plate in a room at 30˚C. One side of the plate is maintained at a temperature of 90˚C, while the other side is insulated, as shown in Figure below. Determine the rate of heat transfer from the plate by natural convection if the plate is (a) vertical, (b) horizontal with hot surface facing up, and (c) horizontal with hot surface facing down.

Solution A hot plate with an insulated back is considered. The rate of heat loss by natural convection is to be determined for different orientations.

Assumptions 1 Steady operating conditions exist. 2 Air is an ideal gas. 3 The local atmospheric pressure is 1 atm.

Properties The properties of air at the film temperature of $T_f = (T_s + T_\infty)/2 = (90+30)/2 = 60$˚C and 1 atm are ;

$k = 0.02808$ W/m · ˚C	$\mathrm{Pr} = 0.7202$
$\nu = 1.896 \times 10^{-5}$ m²/s	$\beta = \dfrac{1}{T_f} = \dfrac{1}{333}$



(*a*) Vertical

Analysis :(a) Vertical. The characteristic length in this case is the height of the plate, which is L = 0.6 m. The Rayleigh number is;

$$\mathrm{Ra}_L = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2}\mathrm{Pr}$$
$$= \frac{(9.81\ \mathrm{m/s^2})[1/(333\ \mathrm{K})](90 - 30\ \mathrm{K})(0.6\ \mathrm{m})^3}{(1.896 \times 10^{-5}\ \mathrm{m^2/s})^2}(0.722) = 7.656 \times 10^8$$

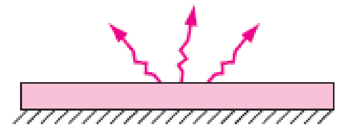
$$\mathrm{Nu} = 0.59\,\mathrm{Ra}_L^{1/4} = 98.14$$

$$h = \frac{K}{L}Nu = \frac{0.02808\ \mathrm{W/m.°C}}{0.6\ \mathrm{m}} \times 98.14 = 4.6\ \mathrm{W}/m^2\ .°\mathrm{C}$$

$$A_s = L^2 = (0.6\ \mathrm{m})^2 = 0.36\ \mathrm{m^2}$$

$$\dot{Q} = h\,A_S\,(T_s - T_\infty) = 4.6 \times 0.36\,(90 - 30) = 99.4\mathrm{W}$$

(b) Horizontal with hot surface facing up. The characteristic length and the Rayleigh number in this case are



(*b*) Hot surface facing up

$$L_c = \frac{A_s}{p} = \frac{L^2}{4L} = \frac{L}{4} = \frac{0.6\text{ m}}{4} = 0.15\text{ m}$$

$$\begin{aligned}\text{Ra}_L &= \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\,\text{Pr}\\ &= \frac{(9.81\text{ m/s}^2)[1/(333\text{ K})](90 - 30\text{ K})(0.15\text{ m})^3}{(1.896 \times 10^{-5}\text{ m}^2/\text{s})^2}(0.7202) = 1.196 \times 10^7\end{aligned}$$

$$\text{Nu} = 0.54\,\text{Ra}_L^{1/4} = 0.54(1.196 \times 10^7)^{1/4} = 31.76$$

Then,

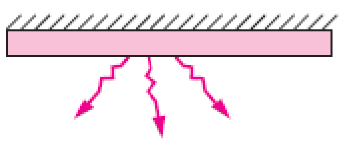
$$h = \frac{k}{L_c}\text{Nu} = \frac{0.0280\text{ W/m}\cdot{}^\circ\text{C}}{0.15\text{ m}}(31.76) = 5.946\text{ W/m}^2\cdot{}^\circ\text{C}$$

$$A_s = L^2 = (0.6\text{ m})^2 = 0.36\text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.946\text{ W/m}^2\cdot{}^\circ\text{C})(0.36\text{ m}^2)(90 - 30)^\circ\text{C} = \mathbf{128\text{ W}}$$

(c) Horizontal with hot surface facing down. The characteristic length, the heat transfer surface area, and the Rayleigh number in this case are the same as those determined in (b). But the natural convection Nusselt number is to be determined from



(*c*) Hot surface facing down

$$\text{Nu} = 0.27\,\text{Ra}_L^{1/4} = 0.27(1.196 \times 10^7)^{1/4} = 15.86$$

$$h = \frac{k}{L_c}\text{Nu} = \frac{0.02808\text{ W/m}\cdot{}^\circ\text{C}}{0.15\text{ m}}(15.86) = 2.973\text{ W/m}^2\cdot{}^\circ\text{C}$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (2.973\text{ W/m}^2\cdot{}^\circ\text{C})(0.36\text{ m}^2)(90 - 30)^\circ\text{C} = \mathbf{64.2\text{ W}}$$

4/ Post test :-

Q1

State heat definition

According to the heat definition, heat is a form of energy that can be transferred from one medium to another through various processes like conduction, convection and radiation.

Q2

What is convection?

Convection is the process of heat transfer by the bulk movement of molecules within fluids such as gases and liquids.

Q3

On what factors the heat-transfer coefficient (h) depends upon?

The value of heat-transfer coefficient (h) depends on:

- Density
- Viscosity
- Thermal conductivity
- Specific heat capacity

Q4

State True or False: Sea breeze is an example for natural convection.

TRUE.

Q5

What are the types of convection?

Types of convection are:

- Natural convection
- Forced convection

5/ HomeWorks:

1- Water at the rate of 0.8 kg/s is heated from 35 to 40◦C in a 2.5-cm-diameter tube whose surface is at 90◦C. How long must the tube be to accomplish this heating?

$\mu = 6.82 \times 10^{-4}$ $\rho = 993$ $k = 0.63$ $\Pr = 4.53$

2- Water is to be heated from 10°C to 80°C as it flows through a 2 cm internal diameter, 7 m long tube. The tube is equipped with a heater, which provides uniform heating throughout the surface of the tube. The outer surface of the heater is well insulated. If the system is to provide hot water at a rate of 8 L/min, determine the power rating of the resistance heater. Also, estimate the inner surface temperature of the pipe at the exit.

3- Water at the rate of 0.5 kg/s is forced through a smooth 2.5-cm-ID tube 15 m long. The inlet water temperature is 10◦C, and the tube wall temperature is 15◦C higher than the water temperature all along the length of the tube. What is the exit water temperature?

$\mu = 1.31 \times 10^{-3}$ $k = 0.585$ $\Pr = 9.4$

4- Water at the rate of 0.6 kg/s is heated from 35°C to 40°C in a 0.03 m diameter tube whose surface temperature is at 90°C. Determine the rate of heat transfer and how long must the tube be to accomplish this heating?

ρ = , 993kg/m^3, k= 0.63 W/m. C, μ = 6.82x10^{-4}kg/m.s , cp= 4221j/kg.ºC, Pr = 4.53

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For

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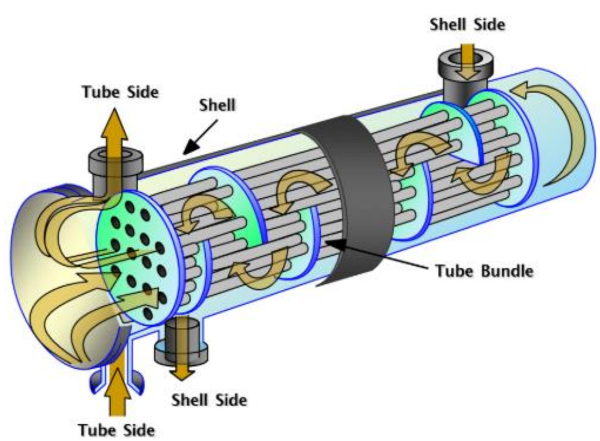

By
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Assistant Professor
Dep. Of Power Mechanics Techniques
2025

1/ Overview

1 / A –Target population :-

For students of Second year
Technological institute of Basra
Dep. Of Power Mechanics Techniques

1 / B –Rationale :-

18th , 24th week
The student understands fundamentals of convection in heat exchanger

1 / C –Central Idea :-

Understanding the type of heat exchangers

1 / D – Performance Objectives

Learn about the importance of heat exchangers and their applications.

https://youtu.be/r5qGAjuxSu8

2/ Pretest :

1- In ………both the hot and cold fluids enter the heat exchanger in the same direction. (Parallel flow, Counter flow, Opposite flow)

2- All fluid properties are to be evaluated at the film temperature, which is………….

($T_f = 1/2$ (Ts +T∞)), (Ti +To)

3- For flow in a circular tube, the Reynolds number is defined as……………

(a) ,(b) Re = ρvL/μ, (c) Re= vL/ν

3/ Scientific content:-

Chapter 7
HEAT EXCHANGERS

Heat exchangers are devices that facilitate the ***exchange of heat*** between ***two fluids*** that are at different temperatures while keeping them from mixing with each other. Heat exchangers are commonly used in practice in a wide range of applications, from heating and air-conditioning systems in a household, to chemical processing and power production in large plants. Heat exchangers differ from mixing chambers in that they do not allow the two fluids involved to mix. In a car radiator, for example, heat is transferred from the hot water flowing through the radiator tubes to the air flowing through the closely spaced thin plates outside attached to the tubes. *Heat transfer* in a *heat exchanger* usually involves *convection in each fluid and conduction* through the wall separating the two fluids.

TYPES OF HEAT EXCHANGERS

Different heat transfer applications require different types of hardware and different configurations of heat transfer equipment. one of the most important these equipment is a heat exchanger. The simplest type of heat exchanger consists of two concentric pipes of different diameters, as shown in figure -1, called the ***double-pipe*** heat exchanger. One fluid in a double-pipe heat exchanger flows through the smaller pipe while the other fluid flows through the annular space between the two pipes. Two types of flow arrangement are possible in a double-pipe heat exchanger according to ***flow directions*** namely ***parallel flow*** and ***counter flow.***

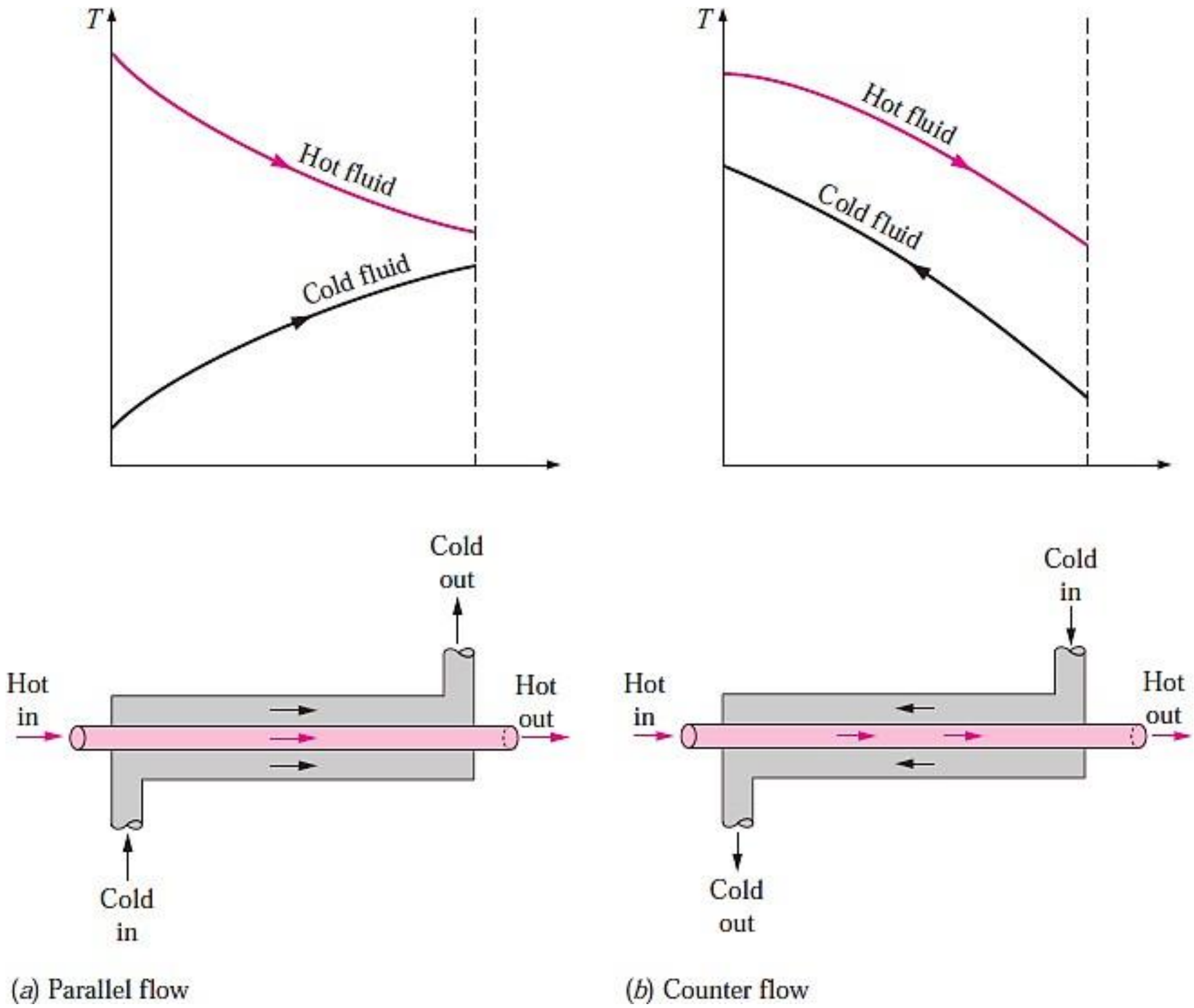


Figure 1: Different flow regimes and associated temperature profiles in a double-pipe heat exchanger.

- In ***parallel flow***, both the hot and cold fluids enter the heat exchanger at the same end and move in the ***same direction***.

- In ***counter flow***, on the other hand, the hot and cold fluids enter the heat exchanger at opposite ends and flow in ***opposite directions***.

Another type of heat exchanger, which is specifically designed to realize a large heat transfer surface area per unit volume, is the ***compact*** heat exchanger. Compact heat exchangers enable us to achieve high heat transfer rates between two fluids in a small volume, and they are commonly used in applications with strict limitations on the weight and volume of heat exchangers see figure 2.

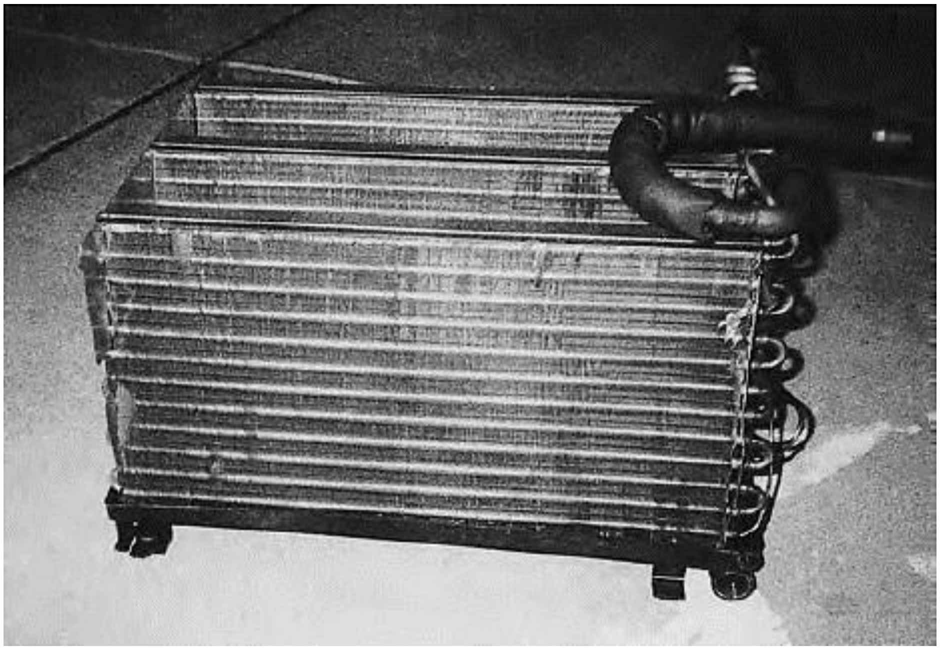


Figure 2: A gas-to-liquid compact heat exchanger for a residential air-conditioning system.

The large surface area in compact heat exchangers is obtained by attaching closely spaced thin plate or corrugated fins to the walls separating the two fluids. Compact heat exchangers are commonly used in gas-to-gas and gas-to liquid (or liquid-to-gas) heat exchangers to counteract the low heat transfer coefficient associated with gas flow with increased surface area. In a car radiator, which is a water-to-air compact heat exchanger, for example.

In compact heat exchangers, the two fluids usually move *perpendicular* to each other, and such flow configuration is called **cross-flow.** The cross-flow is further classified as ***unmixed and mixed flow**,* depending on the flow configuration, as shown in figure -3.

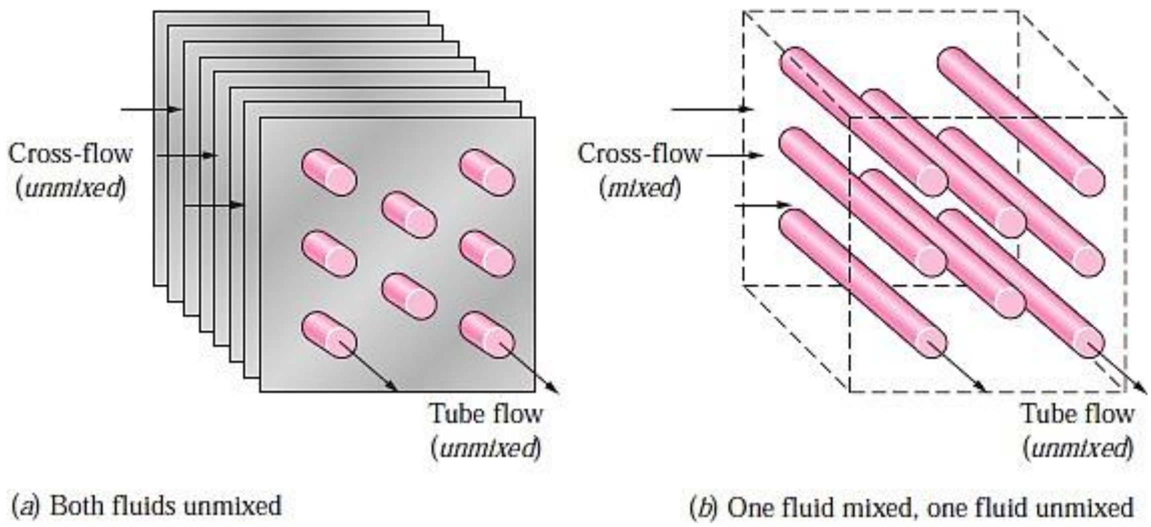


Figure 3 :Different flow configurations in cross-flow heat exchangers

In (a) the cross-flow is said to be unmixed since the plate fins force the fluid to flow through a particular inter fin spacing and prevent it from moving in the transverse direction (i.e., parallel to the tubes). The cross-flow in (b) is said to be mixed since the fluid now is free to move in the transverse direction. Both fluids are unmixed in a car radiator. The presence of mixing in the fluid can have a significant effect on the heat transfer characteristics of the heat exchanger.

Perhaps the most common type of heat exchanger in industrial applications is the ***shell-and-tube*** heat exchanger, shown in figure -4.

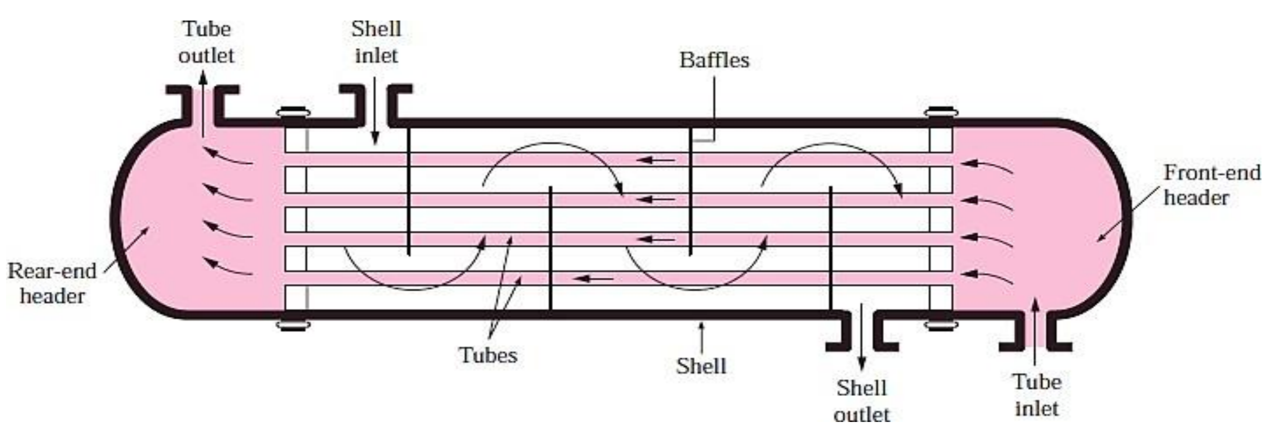
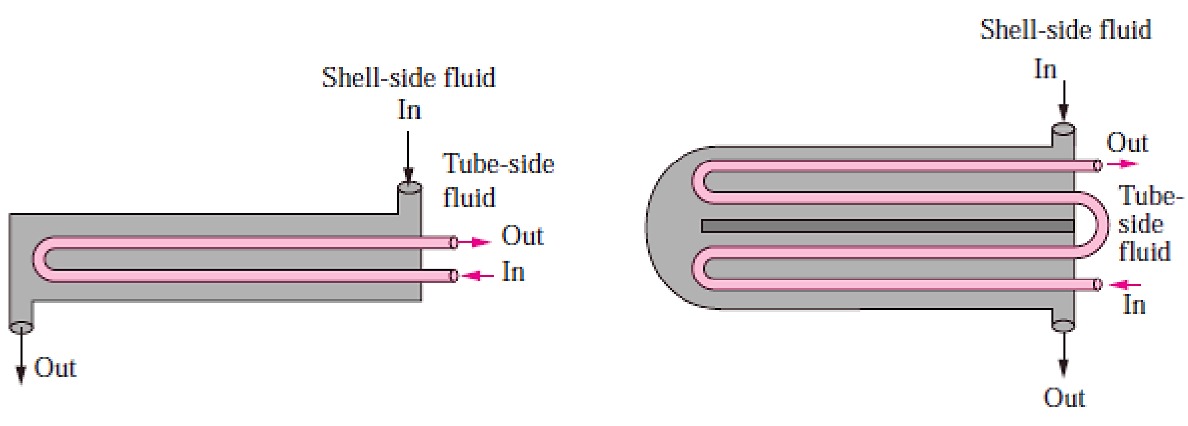


Figure 4 :The schematic of a shell-and-tube heat exchanger (one-shell pass and one-tube pass).

Shell-and-tube heat exchangers contain a large number of tubes (sometimes several hundred) packed in a shell with their axes parallel to that of the shell. Heat transfer takes place as one fluid flows inside the tubes while the other fluid flows outside the tubes through the shell. Baffles are commonly placed in the shell to force the shell-side fluid to flow across the shell to enhance heat transfer and to maintain uniform spacing between the tubes. Despite their widespread use, shell and-tube heat exchangers are not suitable for use in automotive and aircraft applications because of their relatively large size and weight.

Shell-and-tube heat exchangers are further classified according to the ***number of shell and tube passes*** involved. Heat exchangers in which all the tubes make one U- turn in the shell, for example, are called ***one-shell-pass and two tube-passes*** heat exchangers. Likewise, a heat exchanger that involves ***two passes in the shell*** and ***four passes in the tubes*** is called a ***two-shell-passes*** and ***four-tube-passes*** heat exchanger see (figure –5).



(*a*) One-shell pass and two-tube passes

(*b*) Two-shell passes and four-tube passes

Figure 5 :Multipass flow arrangements in shell and- tube heat exchangers

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https://youtu.be/GREWk6eOL5w?si=DhDrdvMfeZunN1uT
https://youtu.be/GREWk6eOL5w?si=J0LciwGwm973cLxl

THE OVERALL HEAT TRANSFER COEFFICIENT

A heat exchanger typically involves two flowing fluids separated by a solid wall. Heat is first transferred from the hot fluid to the wall by convection, through the wall by conduction, and from the wall to the cold fluid again by convection.
The thermal resistance network associated with this heat transfer process involves two convections and one conduction resistances, as shown in figure -6. Here the subscripts ***i*** and ***o*** represent the inner and outer surfaces of the inner tube.

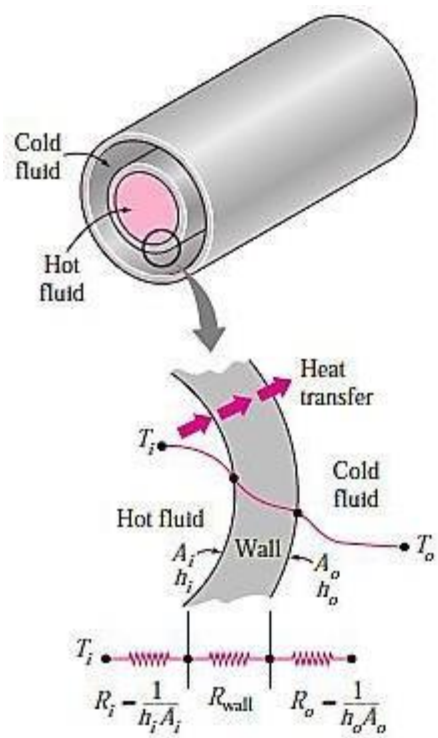


Figure 6: Thermal resistance network associated with heat transfer in a double-pipe heat exchanger.

For a double-pipe heat exchanger, we have $A_i = \pi D_i L$, $A_o = \pi D_o L$ and the *thermal resistance* of the tube wall in this case is

$$R_{\text{wall}} = \frac{\ln\left(\frac{Do}{Di}\right)}{2\pi kL} \quad (1)$$

where k is the thermal conductivity of the wall material and L is the length of the tube. Then the *total thermal resistance* becomes

$$R = R_{\text{wall}} = R_{\text{i}} + R_{\text{wall}} + R_{\text{o}} = \frac{1}{hiAi} + \frac{\ln\left(\frac{Do}{Di}\right)}{2\pi kL} + \frac{1}{hoAo} \quad (2)$$

The Ai is the area of the *inner surface* of the wall that separates the two fluids, and Ao is the area of the outer surface of the wall. In other words, Ai and Ao are surface areas of the separating wall wetted by the inner and the outer fluids, respectively. When one fluid flows inside a circular tube and the other outside of it, we have $Ai = \pi DiL$, $Ao = \pi DoL$ (see figure -7).

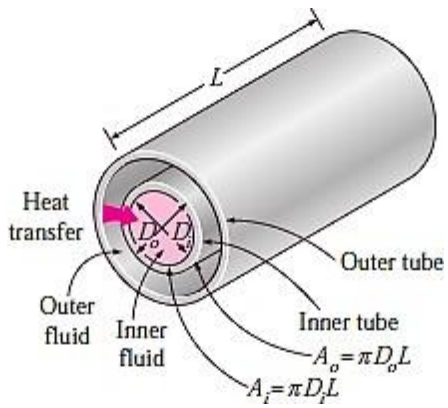


Figure 7: The two heat transfer surface areas associated with a double-pipe heat exchanger (for thin tubes, *Di* ≈ *Do* and thus *Ai* =*Ao*).

In the analysis of heat exchangers, it is convenient to combine all the thermal resistances in the path of heat flow from the hot fluid to the cold one into a single resistance R, and to express the rate of heat transfer between the two fluids as

$$Q = \Delta T/R = UA\Delta T = UiAi\Delta T = U_o\, A_o\, \Delta T \quad (3)$$

where U is the **overall heat transfer coefficient,** whose unit is W/$m2$°C, which is identical to the unit of the ordinary convection coefficient $\boldsymbol{h}$. Canceling ΔT, Eq. –3 reduces

$$\frac{1}{UAs} = \frac{1}{UiAi} = \frac{1}{UoAo}, \text{R}=\frac{1}{hiAi}+\text{R}_{\text{wall}}+\frac{1}{hoAo} \quad \ldots\ldots\ldots (4)$$

Perhaps you are wondering why we have two overall heat transfer coefficients Ui and Uo for a heat exchanger. The reason is that every heat exchanger has two heat transfer surface areas A_i and A_o, which, in general, are not equal to each other.

Note that $U_i A_i = U_o A_o$, but $U_i \neq U_o$ unless $A_i = A_o$. Therefore, the overall heat transfer coefficient U of a heat exchanger is meaningless unless the area on which it is based is specified. This is especially the case when one side of the tube wall is finned and the other side is not, since the surface area of the finned side is several times that of the unfinned side.

When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, as is usually the case, the thermal resistance of the tube is negligible ($Rwall$=0) and the inner and outer surfaces of the tube are almost identical (A_i=A_o=A_s). Then Eq. 4 for the overall heat transfer coefficient simplifies to

$$\frac{1}{U} = \frac{1}{hi} + \frac{1}{ho} = \ldots\ldots\ldots (5)$$

the heat transfer though the wall figure 8 expressed by equation 6

$$q = \frac{T_A - T_B}{1/h_1 A + \Delta x/kA + 1/h_2 A} \quad \ldots\ldots\ldots (6)$$

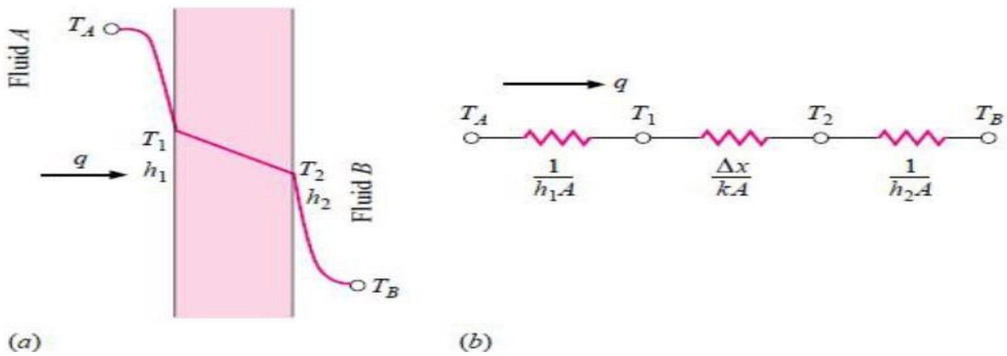


Figure 8:overall heat transfer through a plane wall

Figure 9 represented the double pipe heat exchanger

$$q = \frac{T_A - T_B}{\frac{1}{h_i A_i} + \frac{\ln(r_o/r_i)}{2\pi k L} + \frac{1}{h_o A_o}} \qquad \text{...(7)}$$

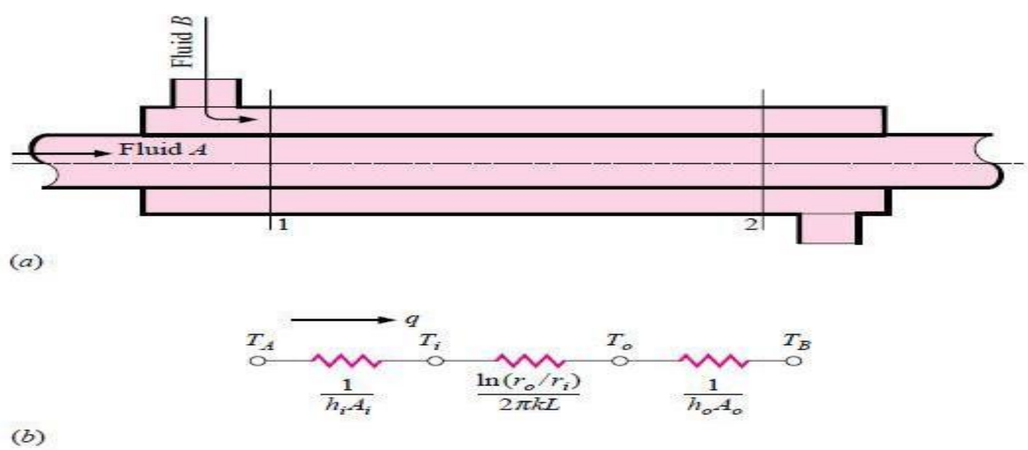


Figure -9 Double-pipe heat exchange: (*a*) schematic;
(*b*) thermal-resistance network for overall heat transfer

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{A_i \ln(r_o/r_i)}{2\pi k L} + \frac{A_i}{A_o}\frac{1}{h_o}}$$

$$U_o = \frac{1}{\frac{A_o}{A_i}\frac{1}{h_i} + \frac{A_o \ln(r_o/r_i)}{2\pi k L} + \frac{1}{h_o}}$$

Example 1:A heat exchanger consists of numerous rectangular channels, each 18 mm wide and 2.25 mm high. In an adjacent pair of channels, there are two streams: water k = 0.625 W/m K and air k = 0.0371 W/m K, separated by a 18 mm wide and 0.5 mm thick stainless steel plate of k = 16 W/m K. The fouling resistances for air and water are 2 x 10^{-4} m‘K/W and 5 x 10^{-4} m' K/W, respectively, and the Nusselt number given by Nu_{ph}= 5.95 where the subscript 'Da' refers to the hydraulic diameter.

a) Calculate the overall heat transfer coefficient ignoring both the thermal resistance of the separating wall and the two fouling resistances.

b) Calculate the overall heat transfer coefficient with these resistances. C) Which is the controlling heat transfer coefficient?

Hydraulic Diameter = 4 x Area / Wetted perimetre

$$D_h = 4\times\frac{2.25\times10^{-3}\times18\times10^{-3}}{2(2.25+18)\times10^{-3}} = 4\times10^{3}$$

$$h = \frac{Nu_D k}{D_h}$$

$$(a) \qquad h_{water} = \frac{5.95\times0.625}{4\times10^{-3}} = 930W/m^2K$$

$$h_{air} = \frac{5.95\times0.0371}{4\times10^{-3}} = 55.186W/m^2K$$

$$U = \left[\frac{1}{930}+\frac{1}{55.186}\right]^{-1} = 52.1 \quad W/m^2\,K$$

$$\text{b)} \quad U = \left[\frac{0.5\times10^{-3}}{16}+\frac{1}{930}+2\times10^{-4}+\frac{1}{55.186}+5\times10^{-4}\right]^{-1} = 50.2 \quad W/m^2\,K$$

C) the controlling heat transfer coefficient is the air heat transfer coefficient.

Example 2:

A heat exchanger tube of D = 20 mm diameter conveys 0.0983 kg/s of water (Pr = 4.3, k=0.632 W/m K, p = 1000 kg/m', u = 0.651 x 10 kg/ms) on the inside which is used to cool a stream of air on the outside where the external heat transfer coefficient has a value of h, = 100 W/m2K. Ignoring the thermal resistance of the tube walls, evaluate the overall heat transfer coefficient, U, assuming that the internal heat transfer coefficient is given by the Dittus-Boelter relation for fully developed turbulent pipe flow:

Solution:

$$\dot{m} = \rho V A$$

$$V = \frac{\dot{m}}{\rho A}$$

$$\mathrm{Re}_D = \frac{\rho V D}{\mu} = \frac{4\dot{m}}{\pi D \mu} = \frac{4 \times 0.0983}{\pi \times 0.02 \times 0.651 \times 10^{-3}} = 9613$$

$$Nu_D = 0.023 \times 9613^{0.8} \times 4.3^{0.4} = 63$$

$$Nu_D = \frac{hD}{k}$$

$$h = \frac{Nu_D k}{D} = \frac{63.3 \times 0.632}{0.02} = 2000 W / m^2 K$$

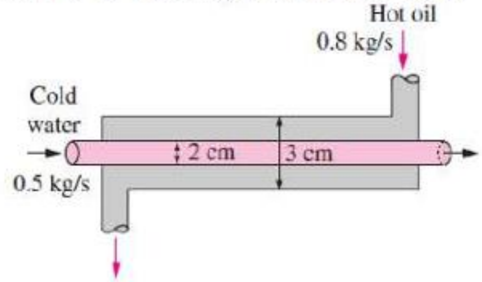
$$U = \left[\frac{1}{2000} + \frac{1}{100}\right]^{-1} = 95.2 W / m^2 K$$

Solved Problems

EXAMPLE -1- *Overall Heat Transfer Coefficient of a Heat Exchanger.*

Hot oil is to be cooled in a double-tube counter-flow heat exchanger. The copper inner tubes have a diameter of 2 cm and negligible thickness. The inner diameter of the outer tube (the shell) is 3 cm. Water flows through the tube at a rate of 0.5 kg/s, and the oil through the shell at a rate of 0.8 kg/s. Taking the average temperatures of the water and the oil to be 45°C and 80°C, respectively, determine the overall heat transfer coefficient of this heat exchanger given that $h_o = 75.2 W/m^2.K$ *and for the inner tube:*

$$Nu = \frac{h\ D}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}$$



Solution:

Properties The properties of water at 45°C are (Table A–9)

$$\rho = 990 \text{ kg/m}^3 \qquad \mathrm{Pr} = 3.91$$
$$k = 0.637 \text{ W/m}\cdot{}^\circ\text{C} \qquad \nu = \mu/\rho = 0.602 \times 10^{-6} \text{ m}^2\text{/s}$$

The properties of oil at 80°C are (Table A–16).

$$\rho = 852 \text{ kg/m}^3 \qquad \mathrm{Pr} = 490$$
$$k = 0.138 \text{ W/m}\cdot{}^\circ\text{C} \qquad \nu = 37.5 \times 10^{-6} \text{ m}^2\text{/s}$$

The overall heat transfer coefficient U can be determined from:

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o}$$

For the inner tube:

$$\mathcal{V}_m = \frac{m}{\rho A_c} = \frac{\dot{m}}{\rho(\frac{1}{4}\pi D^2)} = \frac{0.5 \text{ kg/s}}{(990 \text{ kg/m}^3)[\frac{1}{4}\pi(0.02 \text{ m})^2]} = 1.61 \text{ m/s}$$

$$\mathrm{Re} = \frac{V_m\,D}{\nu} = \frac{(1.61)(0.02)}{0.602 * 10^{-6}} = 53490$$

$$Nu = \frac{h\ D}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} = 0.023(53{,}490)^{0.8}(3.91)^{0.4} = 240.6$$

$$\rightarrow \mathrm{h_i} = 7663 \text{ W/m}^2\cdot{}^\circ\text{C}$$

Then the overall heat transfer coefficient for this heat exchanger becomes

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{7663 \text{ W/m}^2\cdot{}^\circ\text{C}} + \frac{1}{75.2 \text{ W/m}^2\cdot{}^\circ\text{C}}} = \mathbf{74.5\ W/m^2\cdot{}^\circ C}$$

ANALYSIS OF HEAT EXCHANGERS

In upcoming sections, we will discuss the two methods used in the analysis of heat exchangers. Of these, the *log mean temperature difference* (or LMTD) method is best suited for the first task and the *effectiveness–NTU* method for the second task as just stated. But first we present some general considerations.

The *first law of thermodynamics* requires that the rate of heat transfer from the hot fluid be equal to the rate of heat transfer to the cold one figure 13-12. That is,

$Q = m_C\, Cp_C(T_{C,\,OUT} - T_{c,\,in})$

and

$Q = m_h\, Cp_h(T_{h,in} - T_{h,out})$

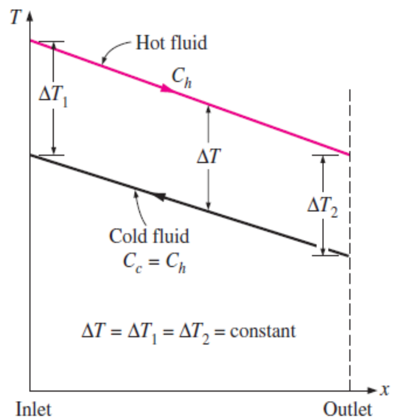


FIGURE 13–12
Two fluids that have the same mass flow rate and the same specific heat experience the same temperature change in a well-insulated heat exchanger.

In heat exchanger analysis, it is often convenient to combine the product of the *mass flow rate* and the *specific heat* of a fluid into a single quantity. This quantity is called the **heat capacity rate** and is defined for the hot and cold fluid streams as figure 13-13

$$C_h = \dot{m}_h C_{ph} \qquad \text{and} \qquad C_c = \dot{m}_c C_{pc}$$

$$\dot{Q} = C_c(T_{c,\,\text{out}} - T_{c,\,\text{in}})$$

$$\dot{Q} = C_h(T_{h,\,\text{in}} - T_{h,\,\text{out}})$$

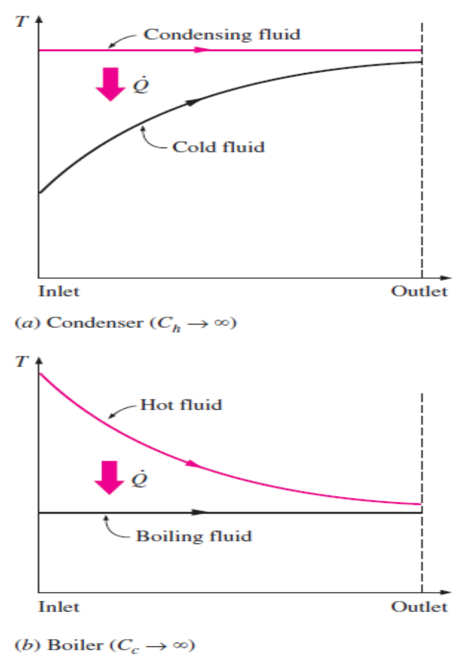


Figure 13- 13:Heat capacity rate

Two special types of heat exchangers commonly used in practice are *condensers* and *boilers*. One of the fluids in a condenser or a boiler undergoes a phase-change process, and the rate of **heat** transfer is expressed as $Q \cdot = m \cdot hfg$ where $m \cdot$ is the rate of evaporation or condensation of the fluid and hfg is the enthalpy of vaporization of the fluid at the specified temperature or pressure. An ordinary fluid absorbs or releases a large amount of heat essentially at constant temperature during a phase-change process, as shown in Figure 13-13 . The heat capacity rate of a fluid during a phase-change process must approach infinity since the temperature change is practically zero. That is,

$C = m \cdot Cp \rightarrow \infty$ when $\Delta T \rightarrow 0$, so that the heat transfer rate

$Q \cdot = m \cdot Cp\, \Delta T$ is a finite quantity. Therefore, in heat exchanger analysis, a condensing or boiling fluid is conveniently modeled as a fluid whose heat capacity rate is *infinity*.

The rate of heat transfer in a heat exchanger can also be expressed in an analogous manner to Newton's law of cooling as $Q = UAs\, \Delta Tm$

THE LOG MEAN TEMPERATURE DIFFERENCE METHOD

The **log mean temperature difference,** which is the suitable form of the average temperature difference for use in the analysis of heat exchangers. Here ΔT_1 and ΔT_2 represent the temperature difference between the two fluids at the two ends (inlet and outlet) of the heat exchanger fig 13-14 .

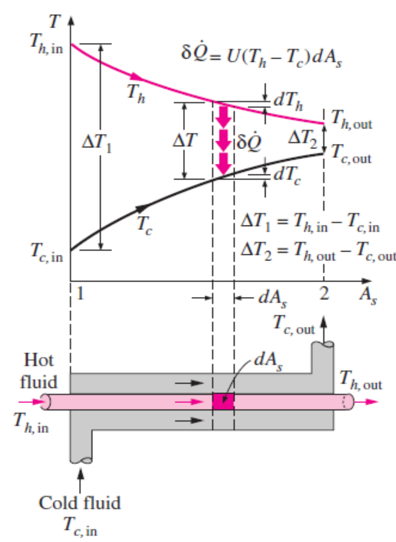


figure 13-14

Counter-Flow Heat Exchangers

The relation above for the log mean temperature difference is developed using a parallel-flow heat exchanger, but we can show by repeating the analysis above for a counter-flow heat exchanger that is also applicable to counter flow heat exchangers.

But this time, ΔT_1 and ΔT_2 are expressed as shown in Figure 13-15.

Multipass and Cross-Flow Heat Exchangers:

Use of a Correction Factor

The log mean temperature difference ΔT_{lm} relation developed earlier is limited to parallel-flow and counter-flow heat exchangers only. Similar relations are also developed for **cross-flow** and **multipass shell-and-tube** heat exchangers, but the resulting expressions are too complicated because of the complex flow conditions.

$\Delta T_{lm} = F\ \Delta T_{lm,\ cf}$

where F is the **correction factor,** which depends on the *geometry* of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. The correction factor F for common cross-flow and shell-and-tube heat exchanger configurations is given in Figure 13-18 versus two temperature ratios P and R defined as .where the subscripts 1 and 2 represent the *inlet* and *outlet,* respectively. Note that for a shell-and-tube heat exchanger, T and t represent the *shell-* and *tube-side* temperatures, respectively

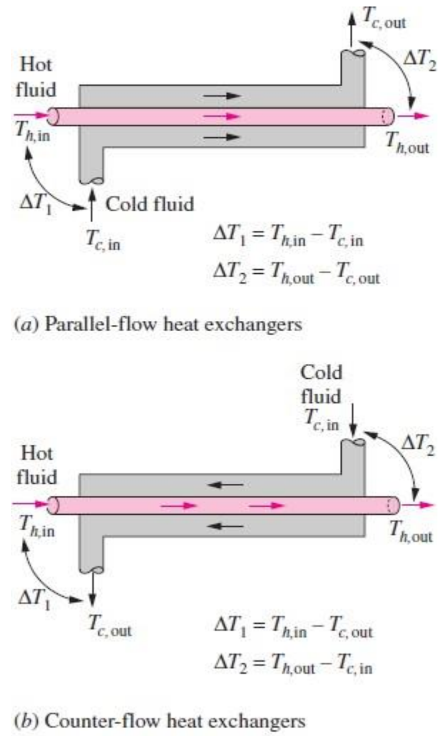
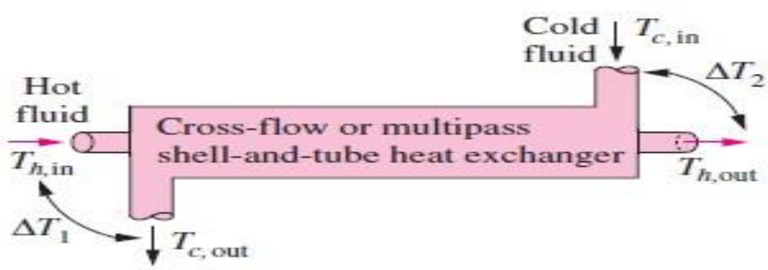


FIGURE 13–15
The ΔT_1 and ΔT_2 expressions in parallel-flow and counter-flow heat exchangers.



Heat transfer rate:

$$\dot{Q} = UA_s F \Delta T_{lm,CF}$$

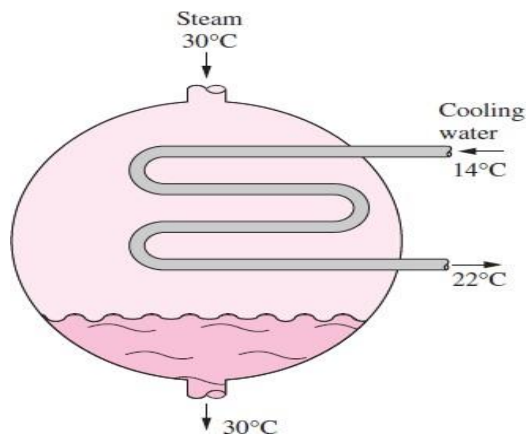
where $$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out}$$
$$\Delta T_2 = T_{h,out} - T_{c,in}$$

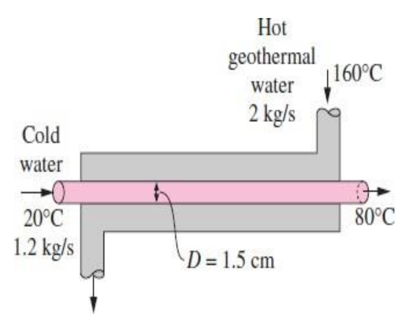
and $F = \ldots$ (Fig. 13–18)

FIGURE 13–17
The determination of the heat transfer rate for cross-flow and multipass shell-and-tube heat exchangers using the correction factor.

EXAMPLE 2 **The Condensation of Steam in a** Steam in the condenser of a power plant is to be condensed at a temperature of 30°C with cooling water from a nearby lake, which enters the tubes of the condenser at 14°C and leaves at 22°C. The surface area of the tubes is 45 m2, and the overall heat transfer coefficient is 2100 W/m2 · °C. Determine the mass flow rate of the cooling water needed and the rate of condensation of the steam in the condenser.

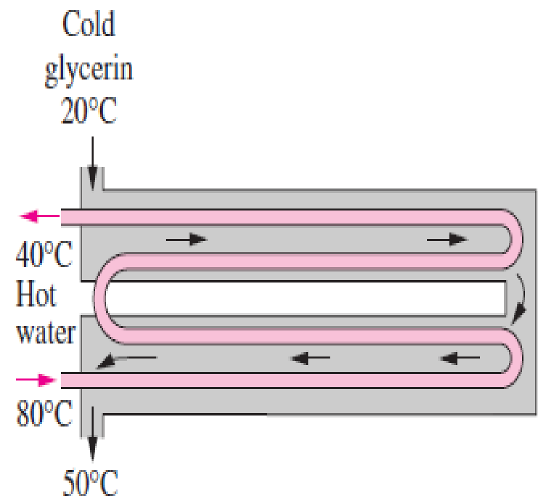


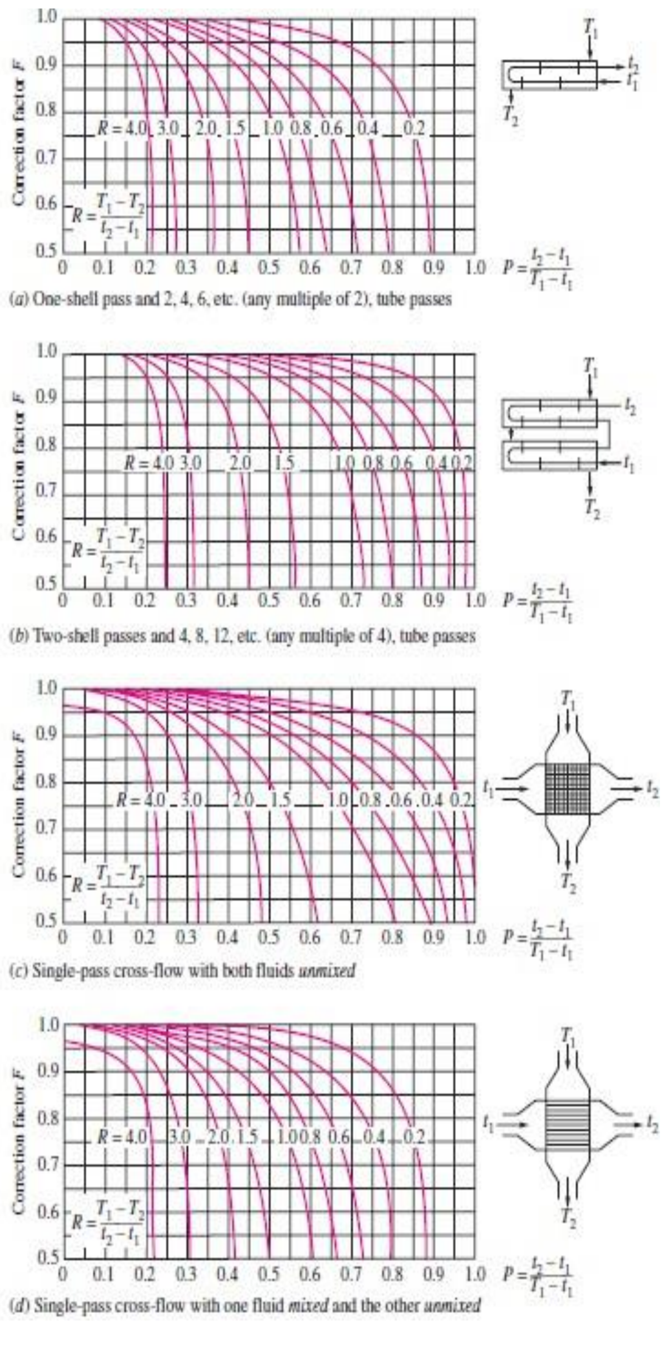
EXAMPLE 3 Heating Water in a Counter-Flow Heat Exchanger
A counter-flow double-pipe heat exchanger is to heat water from 20°C to 80°C at a rate of 1.2 kg/s. The heating is to be accomplished by geothermal water available at 160°C at a mass flow rate of 2 kg/s. The inner tube is thin-walled and has a diameter of 1.5 cm. If the overall heat transfer coefficient of the heat exchanger is 640 W/m2 · °C, determine the length of the heat exchanger required to achieve the desired heating.



EXAMPLE 4 Heating of Glycerin in a Multipass Heat Exchanger

A 2-shell passes and 4-tube passes heat exchanger is used to heat glycerin from 20°C to 50°C by hot water, which enters the thin-walled 2-cm-diameter tubes at 80°C and leaves at 40°C . The total length of the tubes in the heat exchanger is 60 m. The convection heat transfer coefficient is 25 W/m2 · °C on the glycerin (shell) side and 160 W/m2 · °C on the water (tube) side. Determine the rate of heat transfer in the heat exchanger (*a*) before any fouling occurs and (*b*) after fouling with a fouling factor of 0.0006 m2 · °C/ W occurs on the outer surfaces of the tubes.





(*a*) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes

(*b*) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes

(*c*) Single-pass cross-flow with both fluids *unmixed*

(*d*) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

FIGURE 13–18

Correction factor F charts for common shell-and-tube and cross-flow heat exchangers (from Bowman, Mueller, and Nagle, Ref. 2).

THE EFFECTIVENESS–NTU METHOD

The **effectiveness–NTU method,** which greatly simplified heat exchanger analysis. This method is based on a dimensionless parameter called the **heat transfer effectiveness** ε, defined as

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}} \quad \text{(13-29)}$$

The *actual* heat transfer rate in a heat exchanger can be determined from an energy balance on the hot or cold fluids and can be expressed as

$$\dot{Q} = C_c(T_{c,\text{ out}} - T_{c,\text{ in}}) = C_h(T_{h,\text{ in}} - T_{h,\text{ out}}) \quad \text{(13-30)}$$

To determine the maximum possible heat transfer rate in a heat exchanger, we first recognize that the *maximum temperature difference* in a heat exchanger is the difference between the *inlet* temperatures of the hot and cold fluids. That is,

$\Delta T\text{max} = T_h\text{, in} - T_c\text{, in}$

The maximum possible heat transfer rate in a heat exchanger is

$Q\text{ max} = C\text{min}(T_h\text{, in} \_ T_c\text{, in})$

where Cmin is the smaller of $C_h = mhCph$ and $Cc = m_cCp_c$. This is further clarified by the following example. See fig

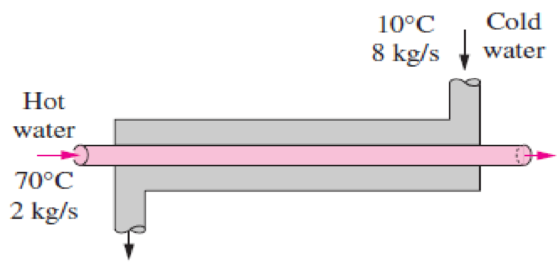
***EXAMPLE* 5**: Upper Limit for Heat Transfer in a Heat Exchanger

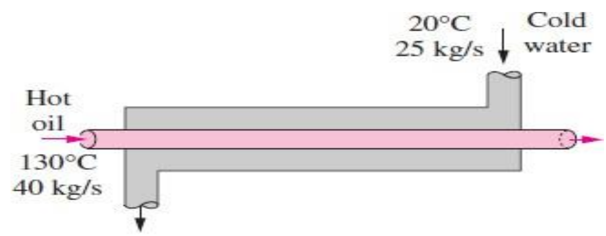
Cold water enters a counter-flow heat exchanger at 10°C at a rate of 8 kg/s, where it is heated by a hot water stream that enters the heat exchanger at 70°C at a rate of 2 kg/s. Assuming the specific heat of water to remain constant at Cp = 4.18 kJ/kg · °C, determine the maximum heat transfer rate and the outlet temperatures of the cold and the hot water streams for this limiting case.

The determination of Q max requires the availability of the *inlet temperature* of the hot and cold fluids and their *mass flow rates,* which are usually specified. Then, once the effectiveness of the heat exchanger is known, the actual heat transfer rate Q can be determined from

$Q \cdot = \varepsilon\, Q_{\text{max}} = \varepsilon C_{min}\, (T_h\text{, in} - T_c\text{, in})$

The effectiveness of a heat exchanger depends on the *geometry* of the heat exchanger as well as the *flow arrangement.* Therefore, different types of heat exchangers have different effectiveness relations. Below we illustrate the development of the effectiveness ε relation for the double-pipe *parallel-flow* heat exchanger.





$$C_c = \dot{m}_c C_{pc} = 104.5 \text{ kW/°C}$$
$$C_h = \dot{m}_c C_{ph} = 92 \text{ kW/°C}$$
$$C_{\min} = 92 \text{ kW/°C}$$
$$\Delta T_{\max} = T_{h,\text{in}} - T_{c,\text{in}} = 110\text{°C}$$
$$\dot{Q}_{\max} = C_{\min}\,\Delta T_{\max} = 10{,}120 \text{ kW}$$

Effectiveness relations of the heat exchangers typically involve the *dimensionless* group *UAs* /*C*min. This quantity is called the **number of transfer units NTU** and is expressed as Effectiveness

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{UA_s}{(\dot{m}C_p)_{\min}} \qquad \text{(13-39)}$$

where U is the overall heat transfer coefficient and A_s is the heat transfer surface area of the heat exchanger. Note that NTU is proportional to A_s. Therefore, for specified values of U and $C_{\min}$, the value of NTU *is a measure of the heat transfer surface area* A_s. Thus, the larger the NTU, the larger the heat exchanger.

In heat exchanger analysis, it is also convenient to define another dimensionless quantity called the **capacity ratio** ***c*** as

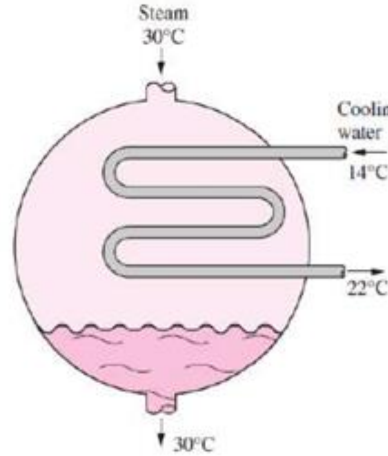
$$c = \frac{C_{\min}}{C_{\max}} \qquad \text{(13-40)}$$

It can be shown that the effectiveness of a heat exchanger is a function of the number of transfer units NTU and the capacity ratio c. That is,

$$\varepsilon = \text{function}\,(UA_s/C_{\min},\, C_{\min}/C_{\max}) = \text{function}\,(\text{NTU},\, c)$$

EXAMPLE –2- The Condensation of Steam in a Condenser.

Steam in the condenser of a power plant is to be condensed at a temperature of 30°C with cooling water from a nearby lake, which enters the tubes of the condenser at 14°C and leaves at 22°C. The surface area of the tubes is 45 m2, and the overall heat transfer coefficient is 2100 W/m2· °C. Determine the mass flow rate of the cooling water needed and the rate of condensation of the steam in the condenser.



Solution:

Properties: The heat of vaporization of water at 30°C is h_{fg}=2431 kJ/kg and the specific heat of cold water at the average temperature of 18°C is Cp=4184J/kg · °C.

The condenser can be treated as a counter-flow heat exchanger since the temperature of one of the fluids (the steam) remains constant. The temperature difference between the steam and the cooling water at the two ends of the condenser is:

$$\Delta T_1 = T_{h,\text{in}} - T_{c,\text{out}} = (30 - 22)°\text{C} = 8°\text{C}$$

$$\Delta T_2 = T_{h,\text{out}} - T_{c,\text{in}} = (30 - 14)°\text{C} = 16°\text{C}$$

That is, the temperature difference between the two fluids varies from 8°C at one end to 16°C at the other. The proper average temperature difference between the two fluids is the logarithmic mean temperature difference (not the arithmetic), which is determined from:

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{8 - 16}{\ln(8/16)} = 11.5°\text{C}$$

Then the heat transfer rate in the condenser is determined from:

$$\dot{Q} = UA_s\,\Delta T_{\text{lm}} = (2100\ \text{W/m}^2\cdot°\text{C})(45\ \text{m}^2)(11.5°\text{C}) = \mathbf{1.087 \times 10^6\ W = 1087\ kW}$$

The cooling water will gain practically all of it, since the condenser is well insulated. The mass flow rate of the cooling water and the rate of the condensation of the steam are determined from:

$$\dot{Q} = [\dot{m}C_p(T_{\text{out}} - T_{\text{in}})]_{\text{cooling water}} = (\dot{m}h_{fg})_{\text{steam}}$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}}{C_p(T_{\text{out}} - T_{\text{in}})}$$

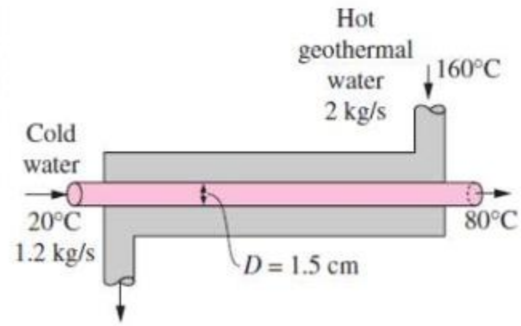
$$= \frac{1{,}087\ \text{kJ/s}}{(4.184\ \text{kJ/kg}\cdot°\text{C})(22 - 14)°\text{C}} = \mathbf{32.5\ kg/s}$$

$$\dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1{,}087\ \text{kJ/s}}{2431\ \text{kJ/kg}} = \mathbf{0.45\ kg/s}$$

Therefore, we need to circulate about 72 kg of cooling water for each 1 kg of steam condensing to remove the heat released during the condensation process.

EXAMPLE –3- Heating Water in a Counter-Flow Heat Exchanger.

A counter-flow double-pipe heat exchanger is to heat water from 20°C to 80°C at a rate of 1.2 kg/s. The heating is to be accomplished by geothermal water available at 160°C at a mass flow rate of 2 kg/s. The inner tube is thin-walled and has a diameter of 1.5 cm. If the overall heat transfer coefficient of the heat exchanger is 640 W/m2· °C, determine the length of the heat exchanger required to achieve the desired heating.



Solution:

Properties: We take the specific heats of water and geothermal fluid to be 4.18and 4.31 kJ/kg · °C, respectively.

The rate of heat transfer in the heat exchanger can be determined from:

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{water} = (1.2 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot °\text{C})(80 - 20)°\text{C} = 301 \text{ kW}$$

Noting that all of this heat is supplied by the geothermal water, the outlet temperature of the geothermal water is determined to be:

$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{geothermal} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}C_p}$$

$$= 160°\text{C} - \frac{301 \text{ kW}}{(2 \text{ kg/s})(4.31 \text{ kJ/kg} \cdot °\text{C})}$$

$$= 125°\text{C}$$

Knowing the inlet and outlet temperatures of both fluids, the logarithmic mean temperature difference for this counter-flow heat exchanger becomes:

$$\Delta T_1 = T_{h,\,in} - T_{c,\,out} = (160 - 80)°\text{C} = 80°\text{C}$$

$$\Delta T_2 = T_{h,\,out} - T_{c,\,in} = (125 - 20)°\text{C} = 105°\text{C}$$

And:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{80 - 105}{\ln(80/105)} = 92.0°\text{C}$$

Then the surface area of the heat exchanger is determined to be

$$\dot{Q} = UA_s\,\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\,\Delta T_{lm}} = \frac{301{,}000 \text{ W}}{(640 \text{ W/m}^2 \cdot °\text{C})(92.0°\text{C})} = 5.11 \text{ m}^2$$

To provide this much heat transfer surface area, the length of the tube must be:

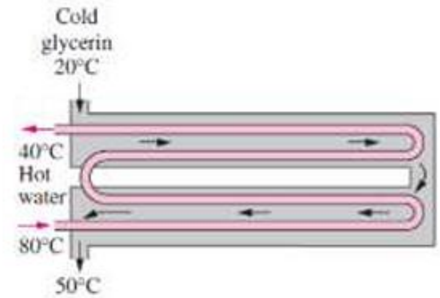
$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{5.11 \text{ m}^2}{\pi(0.015 \text{ m})} = 108 \text{ m}$$

Discussion:

The inner tube of this counter-flow heat exchanger (and thus the heat exchanger itself) needs to be over 100 m long to achieve the desired heat transfer, which is impractical. In cases like this, we need to use a plate heat exchanger or a multi-pass shell-and-tube heat exchanger with multiple passes of tube bundles.

EXAMPLE –4- Heating of Glycerin in a Multi-pass Heat Exchanger

A 2-shell passes and 4-tube passes heat exchanger is used to heat glycerin from 20°C to 50°C by hot water, which enters the thin-walled 2-cm-diameter tube sat 80°C and leaves at 40°C. The total length of the tubes in the heat exchanger is 60 m. The convection heat transfer coefficient is 25 W/m2·°C on the glycerin (shell) side and 160 W/m2· °C on the water (tube) side. Determine the rate of heat transfer in the heat exchanger



Solution:

The tubes are said to be thin-walled, and thus it is reasonable to assume the inner and outer surface areas of the tubes to be equal. Then the heat transfer surface area becomes:

$$A_s = \pi DL = \pi(0.02 \text{ m})(60 \text{ m}) = 3.77 \text{ m}^2$$

The rate of heat transfer in this heat exchanger can be determined from:

$$\dot{Q} = UA_s F \, \Delta T_{\text{lm, }CF}$$

Where: F is the correction factor and $\Delta T_{lm,CF}$ is the log mean temperature difference for the counter-flow arrangement. These two quantities are determined from:

$$\Delta T_1 = T_{h,\text{ in}} - T_{c,\text{ out}} = (80 - 50)°\text{C} = 30°\text{C}$$

$$\Delta T_2 = T_{h,\text{ out}} - T_{c,\text{ in}} = (40 - 20)°\text{C} = 20°\text{C}$$

$$\Delta T_{\text{lm, }CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{30 - 20}{\ln(30/20)} = 24.7°\text{C}$$

And:

F =0.91 from figures.

(a) In the case of no fouling, the overall heat transfer coefficient U is determined from:

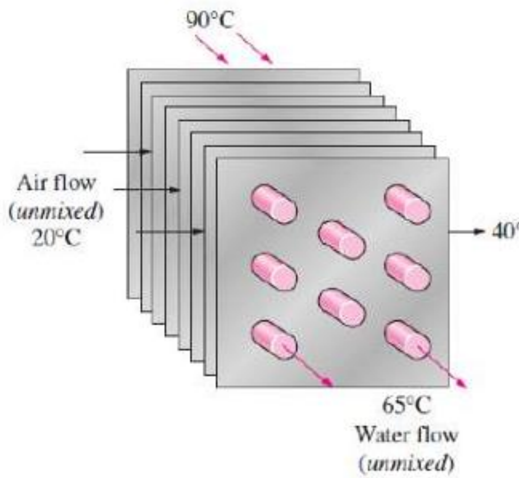
$$U = \frac{1}{\dfrac{1}{h_i} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{160 \text{ W/m}^2 \cdot °\text{C}} + \dfrac{1}{25 \text{ W/m}^2 \cdot °\text{C}}} = 21.6 \text{ W/m}^2 \cdot °\text{C}$$

Then the rate of heat transfer becomes:

$$\dot{Q} = UA_s F \, \Delta T_{\text{lm, }CF} = (21.6 \text{ W/m}^2 \cdot °\text{C})(3.77\text{m}^2)(0.91)(24.7°\text{C}) = \mathbf{1830 \text{ W}}$$

EXAMPLE-5- <u>Cooling of an Automotive Radiator</u>

A test is conducted to determine the overall heat transfer coefficient in an automotive radiator that is a compact cross-flow water-to-air heat exchanger with both fluids (air and water) unmixed. The radiator has 40 tubes of internal diameter 0.5 cm and length 65 cm in a closely spaced plate-finned matrix. Hot water enters the tubes at 90°C at a rate of 0.6 kg/s and leaves at 65°C. Air flows across the radiator through the interfin spaces and is heated from 20°C to 40°C. Determine the overall heat transfer coefficient Ui of this radiator based on the inner surface area of the tubes.



Solution:

Properties:The specific heat of water at the average temperature of (90 □65)/2 =77.5°C is 4.195 kJ/kg·°C.

The rate of heat transfer in this radiator from the hot water to the air is determined from an energy balance on water flow:

$$\dot{Q} = [\dot{m}C_p(T_{in} - T_{out})]_{water} = (0.6 \text{ kg/s})(4.195 \text{ kJ/kg} \cdot °\text{C})(90 - 65)°\text{C} = 62.93 \text{ kW}$$

The tube-side heat transfer area is the total surface area of the tubes, and is determined from:

$$A_i = n\pi D_i L = (40)\pi(0.005 \text{ m})(0.65 \text{ m}) = 0.408 \text{ m}^2$$

Knowing the rate of heat transfer and the surface area, the overall heat transfer coefficient can be determined from:

$$\dot{Q} = U_i A_i F \, \Delta T_{\text{lm},\, CF} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \, \Delta T_{\text{lm},\, CF}}$$

Where: F is the correction factor and $\Delta T_{lm,CF}$ is the log mean temperature difference for the counter-flow arrangement. These two quantities are determined from:

$$\Delta T_1 = T_{h,\text{ in}} - T_{c,\text{ out}} = (90 - 40)°\text{C} = 50°\text{C}$$

$$\Delta T_2 = T_{h,\text{ out}} - T_{c,\text{ in}} = (65 - 20)°\text{C} = 45°\text{C}$$

$$\Delta T_{\text{lm},\, CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{50 - 45}{\ln(50/45)} = 47.6°\text{C}$$

And F=0.97 from charts.

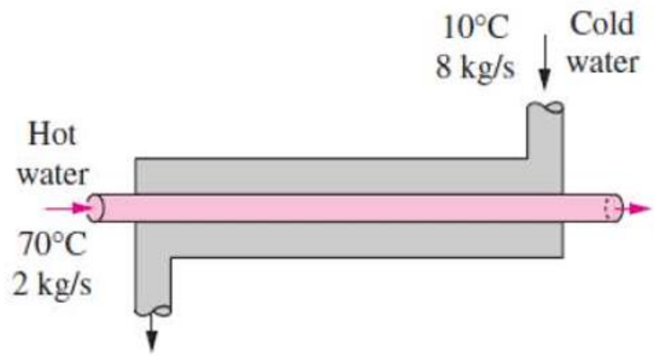
Substituting, the overall heat transfer coefficient Ui is determined to be:

$$U_i = \frac{\dot{Q}}{A_i F \, \Delta T_{\text{lm},\, CF}} = \frac{62{,}930 \text{ W}}{(0.408 \text{ m}^2)(0.97)(47.6°\text{C})} = \mathbf{3341 \text{ W/m}^2 \cdot °\text{C}}$$

Note that the overall heat transfer coefficient on the air side will be much lower because of the large surface area involved on that side.

EXAMPLE –6- Upper Limit for Heat Transfer in a Heat Exchanger.

Cold water enters a counter-flow heat exchanger at 10°C at a rate of 8 kg/s, where it is heated by a hot water stream that enters the heat exchanger at 70°C at a rate of 2 kg/s. Assuming the specific heat of water to remain constant at Cp=4.18 kJ/kg·°C, determine the maximum heat transfer rate and the outlet temperatures of the cold and the hot water streams for this limiting case.



Solution:

Properties: The specific heat of water is given to be Cp=4.18 kJ/kg · °C.

The heat capacity rates of the hot and cold fluids are determined from:

$$C_h = \dot{m}_h C_{ph} = (2\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C}) = 8.36\ \text{kW/}^\circ\text{C}$$

And:

$$C_c = \dot{m}_c C_{pc} = (8\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C}) = 33.4\ \text{kW/}^\circ\text{C}$$

Therefore:

$$C_{\min} = C_h = 8.36\ \text{kW/}^\circ\text{C}$$

Which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate is determined to be:

$$\begin{aligned}\dot{Q}_{\max} &= C_{\min}(T_{h,\,\text{in}} - T_{c,\,\text{in}}) \\ &= (8.36\ \text{kW/}^\circ\text{C})(70 - 10)^\circ\text{C} \\ &= \mathbf{502\ kW}\end{aligned}$$

That is, the maximum possible heat transfer rate in this heat exchanger is 502kW. This value would be approached in a counter-flow heat exchanger with a very large heat transfer surface area.

The maximum temperature difference in this heat exchanger is:

$$\Delta T_{\max} = T_{h,\,\text{in}} - T_{c,\,\text{in}} = (70 - 10) = 60^\circ\text{C}$$

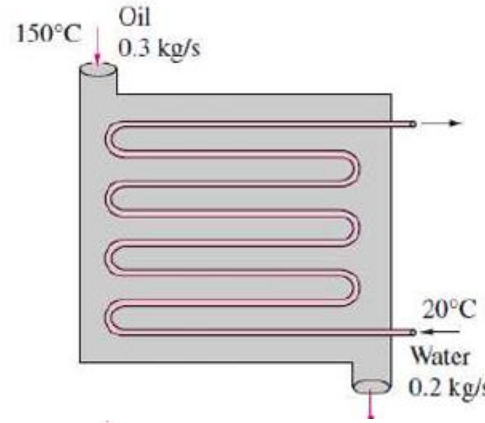
Therefore, the hot water cannot be cooled by more than 60°C (to 10°C) in this heat exchanger, and the cold water cannot be heated by more than 60°C (to 70°C), no matter what we do. The outlet temperatures of the cold and the hot streams in this limiting case are determined to be:

$$\dot{Q} = C_c(T_{c,\,\text{out}} - T_{c,\,\text{in}}) \longrightarrow T_{c,\,\text{out}} = T_{c,\,\text{in}} + \frac{\dot{Q}}{C_c} = 10^\circ\text{C} + \frac{502\ \text{kW}}{33.4\ \text{kW/}^\circ\text{C}} = \mathbf{25^\circ C}$$

$$\dot{Q} = C_h(T_{h,\,\text{in}} - T_{h,\,\text{out}}) \longrightarrow T_{h,\,\text{out}} = T_{h,\,\text{in}} - \frac{\dot{Q}}{C_h} = 70^\circ\text{C} - \frac{502\ \text{kW}}{8.38\ \text{kW/}^\circ\text{C}} = \mathbf{10^\circ C}$$

EXAMPLE -7- -Cooling Hot Oil by Water in a MultipassHeatExchanger

Hot oil is to be cooled by water in a 1-shell-pass and 8-tube-passes heat exchanger. The tubes are thin-walled and are made of copper with an internal diameter of 1.4 cm. The length of each tube pass in the heat exchanger is 5 m, and the overall heat transfer coefficient is 310 W/m2· °C. Water flows through the tubes at a rate of 0.2 kg/s, and the oil through the shell at a rate of 0.3 kg/s. The water and the oil enter at temperatures of 20°C and 150°C, respectively. Determine the rate of heat transfer in the heat exchanger and the outlet temperatures of the water and the oil.



Solution:

The outlet temperatures are not specified, and they cannot be determined from an energy balance. The use of the LMTD method in this case will involve tedious iterations, and thus the $\varepsilon - NTU$ method is indicated. The first step in the $\varepsilon - NTU$ method is to determine the heat capacity rates of the hot and cold fluids and identify the smaller one:

$$C_h = \dot{m}_h C_{ph} = (0.3 \text{ kg/s})(2.13 \text{ kJ/kg} \cdot {}^\circ\text{C}) = 0.639 \text{ kW/}^\circ\text{C}$$

$$C_c = \dot{m}_c C_{pc} = (0.2 \text{ kg/s})(4.18 \text{ kJ/kg} \cdot {}^\circ\text{C}) = 0.836 \text{ kW/}^\circ\text{C}$$

Therefore,

$C_{\min} = C_h = 0.639 kW/{}^oC$ And: $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{0.639}{0.836} = 0.764$

Then the maximum heat transfer rate is determined to be:

$$\dot{Q}_{\max} = C_{\min}(T_{h,\text{ in}} - T_{c,\text{ in}})$$
$$= (0.639 \text{ kW/}^\circ\text{C})(150 - 20)^\circ\text{C} = 83.1 \text{ kW}$$

That is, the maximum possible heat transfer rate in this heat exchanger is 83.1kW. The heat transfer surface area is:

$$A_s = n(\pi DL) = 8\pi(0.014 \text{ m})(5 \text{ m}) = 1.76 \text{ m}^2$$

Then the NTU of this heat exchanger becomes:

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(310 \text{ W/m}^2 \cdot {}^\circ\text{C})(1.76 \text{ m}^2)}{639 \text{ W/}^\circ\text{C}} = 0.853$$

The effectiveness of this HX corresponding to (c =0.764&NTU=0.853 is determined from Figs to be:

$$\varepsilon = 0.47$$

We could also determine the effectiveness from the relations in table. Then the actual rate of heat transfer becomes:

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.47)(83.1 \text{ kW}) = \mathbf{39.1 \text{ kW}}$$

Finally, the outlet temperatures of the cold and the hot fluid streams are determined to be:

$$\dot{Q} = C_c(T_{c,\text{ out}} - T_{c,\text{ in}}) \longrightarrow T_{c,\text{ out}} = T_{c,\text{ in}} + \frac{\dot{Q}}{C_c}$$

$$\dot{Q} = C_h(T_{h,\text{ in}} - T_{h,\text{ out}}) \longrightarrow T_{h,\text{ out}} = T_{h,\text{ in}} - \frac{\dot{Q}}{C_h}$$

$$= 150^\circ\text{C} - \frac{39.1 \text{ kW}}{0.639 \text{ kW/}^\circ\text{C}} = \mathbf{88.8^\circ C}$$

Ministry of high Education and Scientific Research
Southern Technical University
Technological institute of Basra
Department of Electronic Techniques

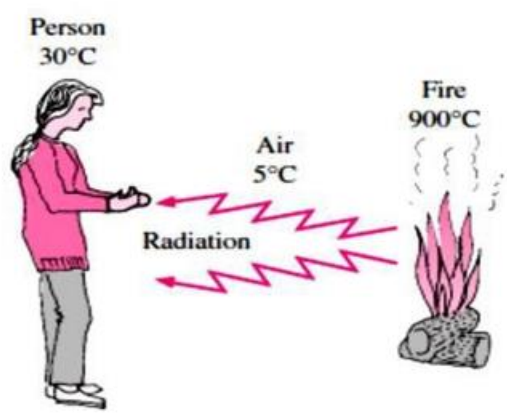
Learning package

In

Fundamentals of Radiation

For

Students of Second Year



By

Dr. Duna Tariq Yaseen
Assistant Professor
Dep. Of Power Mechanics Techniques

2025

1/ Overview

1 / A –Target population :-

For students of Second year
Technological institute of Basra
Dep. Of Power Mechanics Techniques

1 / B –Rationale :-

25^{th} , 27^{th} weeks

The student understands fundamentals of radiation

1 / C –Central Idea :-

Understanding the (Absorptivity , Reflectivity , and Transmissivity)

1 / D – Performance Objectives

Learn (Radiation Mechanism, Physical properties, Gray body, construction of a blackbody)

2/ Pretest :

1-heat transfer mechanism that it does not require the presence of a material medium to take place
a) Conduction
b) Convection
c) Radiation
d) no-of them
2- The amount of radiation mainly depends upon the__
a) nature of the body
b) temperature of the body
c) type of surface of the body
d) all of these heat transfer through an evacuated space can occur only by_____
a) Radiation
b) Convection
c) Conduction
d) No-of them
3- _______is defined as a perfect emitter and absorber of radiation.
a) Blackbody
b) Real body
c) Reflected body
d) No-of them
4- The emissivity for black bodies is __
a) 1
b) 0
c) 0.5
d) 3

3/ Scientific content:-

chapter 8

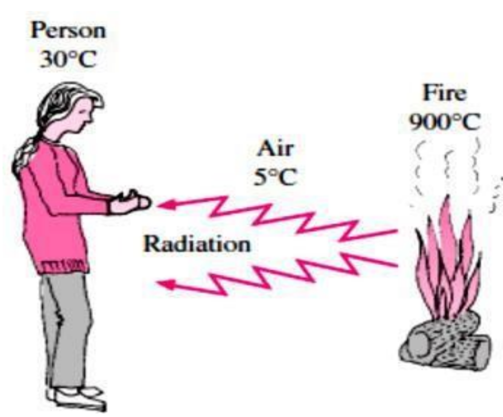
Heat Radiation

1. Introduction

Radiation differs from the other two heat transfer mechanisms in that it does not require the presence of a material medium to take place. In fact, energy transfer by radiation in a *vacuum*. Also, radiation transfer occurs in solids as well as liquids and gases. For example, the energy of the sun reaches the earth by radiation.

You will recall that heat transfer by conduction or convection takes place in the direction of decreasing temperature; that is, from a high-temperature medium to a lower-temperature one. It is interesting that radiation heat transfer can occur between two bodies separated by a medium colder than both bodies.

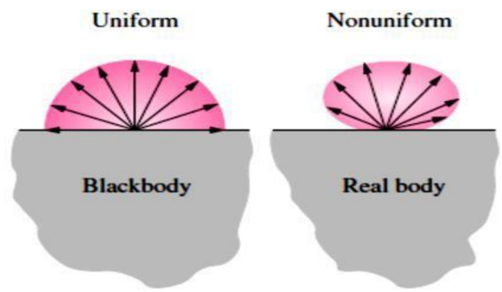
Most materials such as metals, wood, and bricks, are *opaque* to thermal radiation,



and radiation is considered to be a surface phenomenon for such materials. Some other materials, such as glass and water, allow visible radiation to penetrate to considerable depths before any significant absorption takes place. Therefore, materials can exhibit different behavior at different wavelengths, and the dependence on wavelength is an important consideration in the study of radiative properties such as emissivity, absorptivity, reflectivity, and transmissivity of materials

2. Blackbody Radiation

A **blackbody** is defined as *a perfect emitter and absorber of radiation.* At a specified temperature and wavelength, no surface can emit more energy than a blackbody. A blackbody absorbs *all* incident radiation, regardless of wavelength and direction. Also, a blackbody emits radiation energy uniformly in all directions per unit area normal to direction of emission.



The radiation energy emitted by a blackbody is $Eb = A_s \, \sigma \, T^4$

Where As is surface area, $\sigma = 5.67 \times 10^{-8} \, Wm^2.K^4/$ is the ***Stefan–Boltzmann constant*** and T is the absolute temperature of the surface in ***K***. This Equation is known as the **Stefan–Boltzmann law** and E_b is called the **blackbody emissive power.** A small blackbody at absolute temperature T enclosed by a much larger blackbody at absolute temperature Ts will transfer a **net heat flow** of, $Q = A_s \, \sigma \, (T^4 - T^4{}_s)$

3. Gray Body Radiation Heat Transfer

Bodies that emit less thermal radiation than a blackbody have surface emissivity ε less than 1. If the surface emissivity is independent of wavelength, then the body is called a "gray" body.

$$Eg = A_s \, \sigma \, \varepsilon \, T^4$$

The net heat transfer from a small gray body at absolute temperature T with surface emissivity ε to a much larger enclosing gray (or black) body at absolute temperature Ts is given by, $Q = A_s \, \sigma \, \varepsilon \, (T^4 - T^4{}_s)$

4. Emissivity

The **emissivity** of a surface represents the ratio of the radiation emitted by the surface at a given temperature to the radiation emitted by a blackbody at the same temperature. The emissivity of a surface is denoted by ε, and it varies between zero and one, $0 \leq \varepsilon \leq 1$. Emissivity is a measure of how closely a surface approximates a blackbody, for which $\varepsilon = 1$.

Example (1): A Cylindrical black body of 25 cm radius and 3 cm length, the cylinder surface temperature is 1100°C. Calculate the rate of thermal energy emitted from the outside surface.

Solution:

$$Eb = As\sigma T^4 = 0.0471 \times 5.67 \times 10^{-8} \times (1373)^4 = 94904.3W$$

$$As = \pi\, D\, L = 3.14 \times 2 \times 25/100 \times 3/100 = 0.0471\ m^2$$

$$T = 1100 + 273 = 1373\ K$$

Example (2): A Cylindrical gray body of 30 cm radius and 10 cm length with surface temperature of 1300°C, Calculate the rate of emitted thermal energy from the cylinder. The emissivity for the cylinder is 0.85.

Solution:

$$Eg = As\ \sigma\ \varepsilon\ T^4 = 0.1884 \times 5.67 \times 10^{-8} \times 0.85 \times (1573)^4 = 55590\ W$$

$$As = \pi\, D\, L = 3.14 \times (2 \times 0.3) \times 0.1 = 0.1884\ m^2, T = 1300 + 273 = 1573\ K$$

Example (3): The parallel infinite black plane surfaces are maintained at 1387°C, 987°C. Determine the net rate of radiant interchange between the surfaces per unit area.

Solution: $T = 1387 + 273 = 1660$, $Ts = 987 + 273 = 1260\ K$

$$Q = A_s\, \sigma\, (T^4 - T^4_s) = 1 \times 5.67 \times 10^{-8} \times (1660^4 - 1260^4) = 287631\ W$$

Example (4): *Calculate the radiant flux density from a black surface at 400°C?*

Solution: $E_b/A_s = \sigma T^4 = 5.67 \times 10^{-8} \times (400+273)^4 = 11631.7\ w/m^2$

Example (5): A gray body ($\varepsilon = 0.8$) emits the same amount of heat as a black body at 1075 K. Find out the required temperature of the gray body?

Solution:

$Q_b = Q_g$

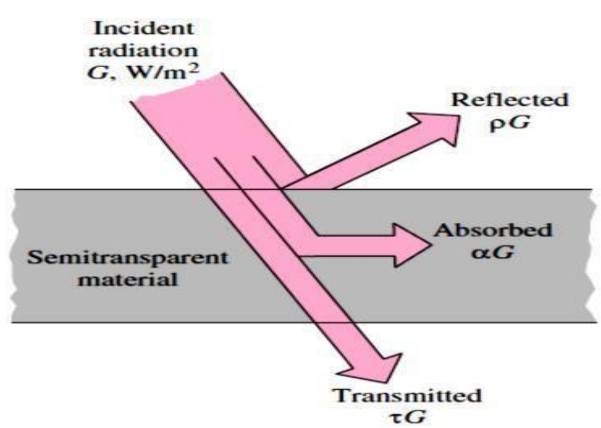
$\cancel{A_s}\cancel{\sigma}\, T^4{}_b = \cancel{A_s}\cancel{\sigma}\, \underline{\underline{\varepsilon}}\, T^4{}_g$

$1075^4 = 0.8\, T^4{}_g$

$T_g = 1136.67\ K$

5. Absorptivity, Reflectivity, and Transmissivity

The radiation flux *incident on a surface* is called **irradiation** and is denoted by *G*. When radiation strikes a surface, part of it is absorbed, part of it is reflected, and the remaining part, is transmitted, as illustrated in Figure below.



The fraction of irradiation absorbed by the surface is called the **absorptivity** $\boldsymbol{\alpha}$, *the fraction reflected by the surface* is called the **reflectivity ρ**, and *the fraction transmitted* is called the **transmissivity τ**. That is, *Absorptivity:* $\alpha = \dfrac{\text{Absorbed radiation}}{\text{Incident radiation}} = \dfrac{G_{abs}}{G}$, $0 \le \alpha \le 1$

Reflectivity: $\rho = \dfrac{\text{Reflected radiation}}{\text{Incident radiation}} = \dfrac{G_{ref}}{G}$, $0 \le \rho \le 1$

Transmissivity: $\tau = \dfrac{\text{Transmitted radiation}}{\text{Incident radiation}} = \dfrac{G_{tr}}{G}$, $0 \le \tau \le 1$

where G is the radiation energy incident on the surface, and Gabs, Gref, and Gtr are the absorbed, reflected, and transmitted portions of it, respectively.

$G_{abs}+G_{ref} + G_{tr} = G$

Dividing each term of this relation by G yields

$$\alpha + \rho + \tau = 1$$

Example (6): A surface receives radiation energy at rate of 120 w/m2 absorb 80 w/m2 and transmit 10w/m2 and reflect the rest. Calculate (a) absorptivity (b) reflectivity (c) transmissivity.

Solution:

$a)$ $\alpha = G_{abs}/G$ = 80/120 =0.667

a) $G_{abs}+G_{ref} + G_{tr} = G$

$80+ G_{ref}+10 = 120$, $G_{ref} = 30$ w/m^2

$$p = \frac{G_{ref}}{G} = \frac{30}{120} = 0.25$$

$$\tau = \frac{10}{120} = 0.083$$

Example (7): *The incident thermal radiation energy on gray body, is 300 w/m2 absorbed, 200w/m2 reflect, and 60 w/m2 transmitted. Calculate the absorptivity .*

$G_{abs}+G_{ref} + G_{tr} = G$

G=300+200+60=560 w/m^2

$$\propto= \frac{G_{abs}}{G} = \frac{300}{560} = 0.536$$

Example (8): Electromagnetic waves strikes a hot body which has a reflectivity of 0.59 and a transmissivity of 0.032 and the absorbed flux be 60 w/m2. Determine the rate of incident flux.

$\alpha+\rho+\tau=1$, $\alpha+0.59+0.032=1$, $\alpha= 0.373$

$\propto= \frac{G_{abs}}{G}$,

$$G = \frac{G_{abs}}{\propto} \cdot \frac{60}{0.373} = 160.86 w/m^2$$

5/ HomeWorks:

1. 220 w/m2of radiant energy is absorbed by a convex surface, 90 w/m2is reflected and 40 w/m2 is transmitted through it, what is the values of absorptivity, reflectivity, and transmissivity?

2. A surface receives radiation energy at rate of 100 w/m2, absorb 60 w/m2 and transmit 10w/m2and reflect the rest. Calculate absorptivity and reflectivity.

3. A black body emits the same amount of heat as a gray body (ε= 0.65), the absolute temperature of gray body is 205°C. Find out the required temperature of the black body?

6/References :-

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6. Suggested sources:

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