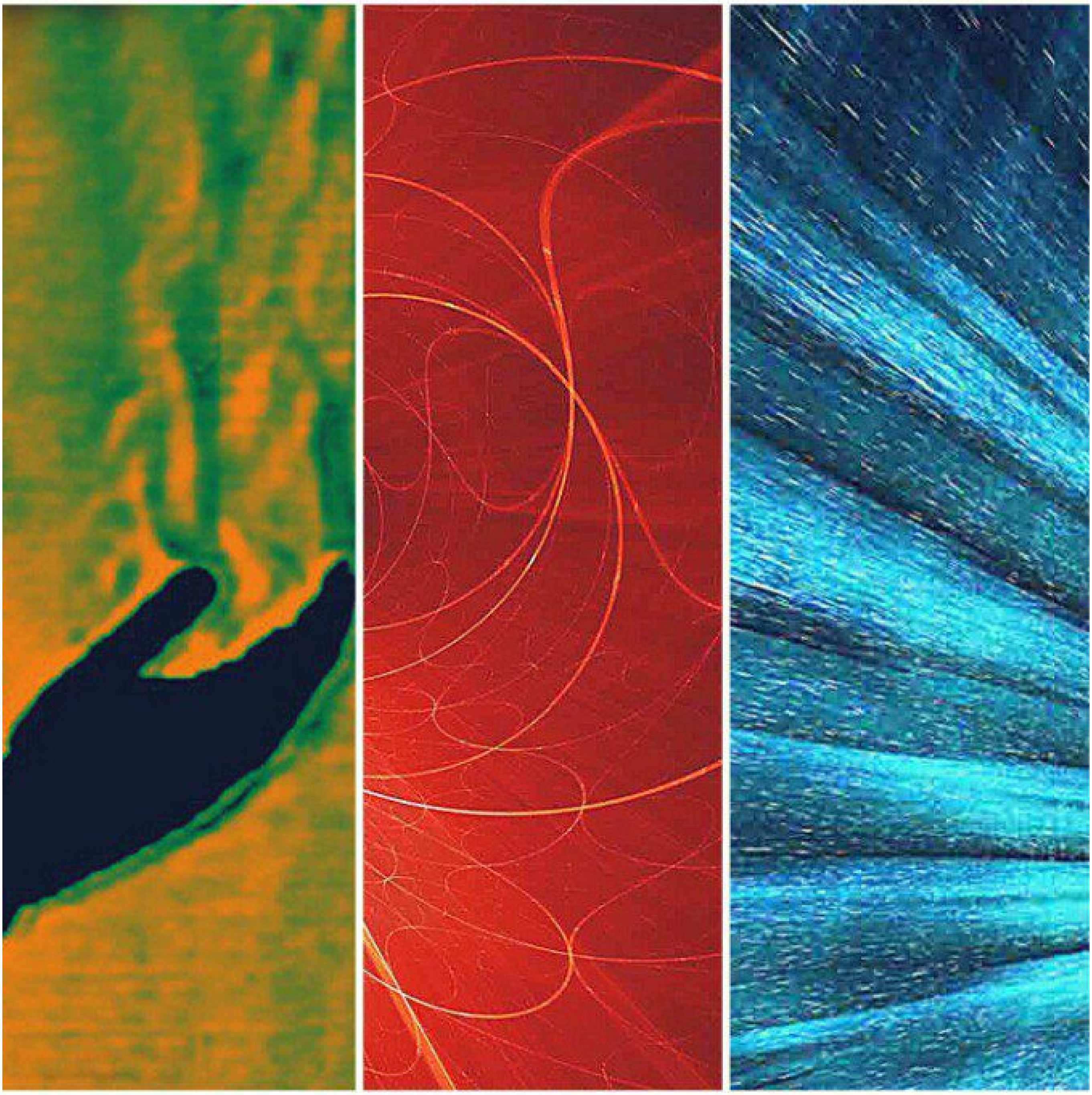
POLITEKNIK MALAYSIA
Tun Syed Nasir Syed Ismail



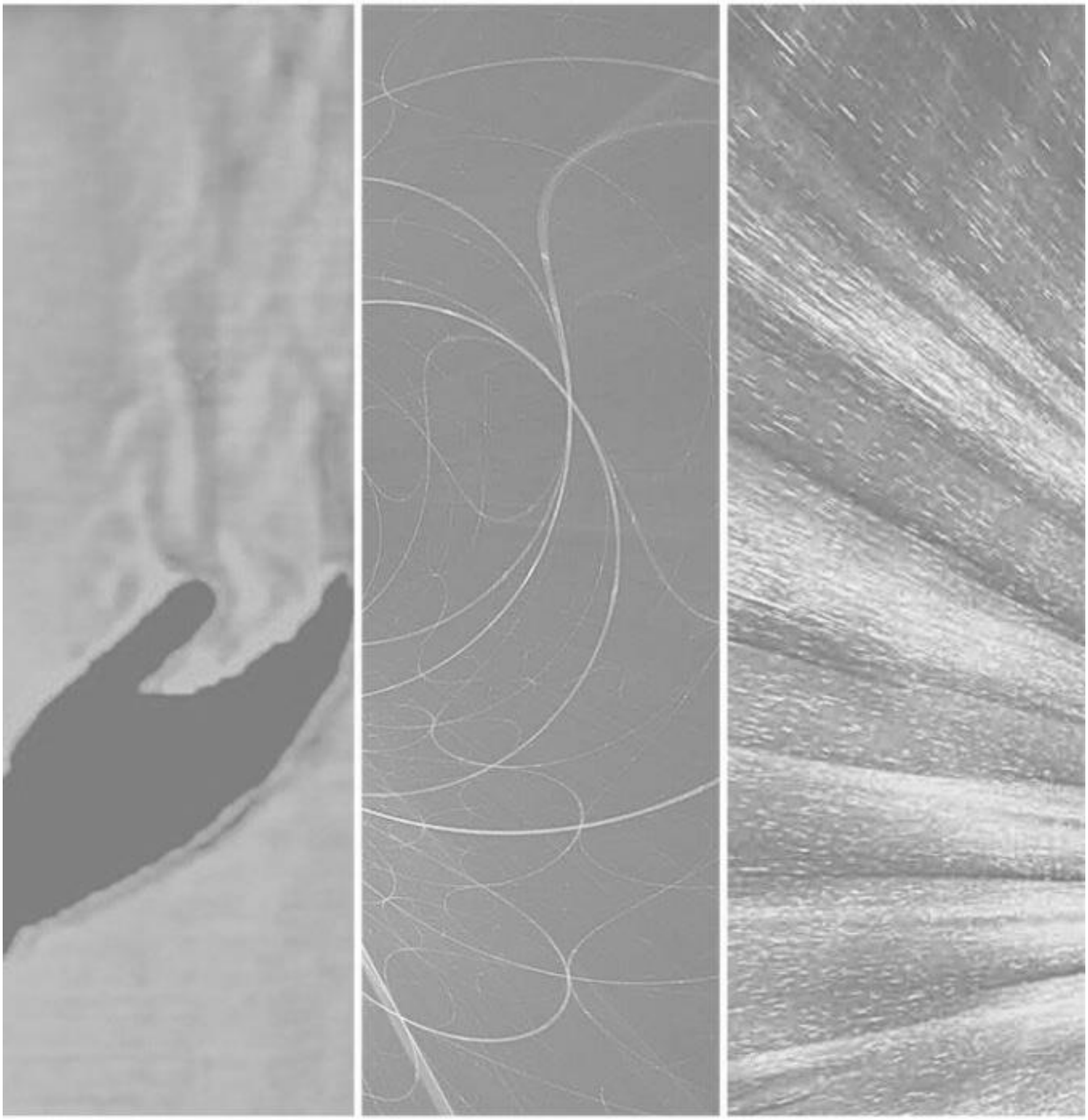
HEAT AND MASS TRANSFER

VOLUME 1

FOR CHEMICAL ENGINEERING POLYTECHNIC STUDENTS

BY:
NOORIZYAN BINTI ASIMAT
NOR SYUHADA BINTI MUSLIM
MOHD AZIM BIN MD JANIS

POLITEKNIK MALAYSIA
Tun Syed Nasir Syed Ismail



HEAT AND MASS TRANSFER

VOLUME 1

FOR CHEMICAL ENGINEERING POLYTECHNIC STUDENTS

BY:
NOORIZYAN BINTI ASIMAT
NOR SYUHADA BINTI MUSLIM
MOHD AZIM BIN MD JANIS

Copyright page

e ISBN 978-967-2736-02-8

9 7 8 9 6 7 2 7 3 6 0 2 8

FIRST PUBLISHED 2021

Editor

Fazliana Binti Abdullah

Illustrated by

Nor Syuhada Binti Muslim

Published in Malaysia by:
Politeknik Tun Syed Nasir Syed Ismail
Hab Pendidikan Tinggi Pagoh
KM 1 Jalan Panchor 84600 Pagoh, Johor
Tel: 06-974 2288 Faks: 06-974 2395
Website: www.ptsn.edu.my

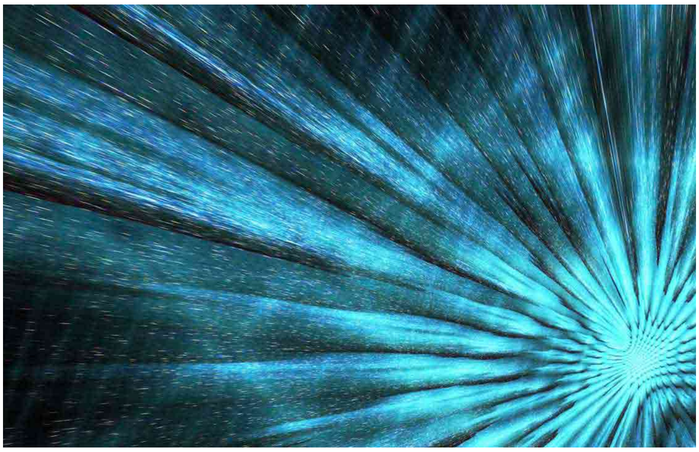
PREFACE

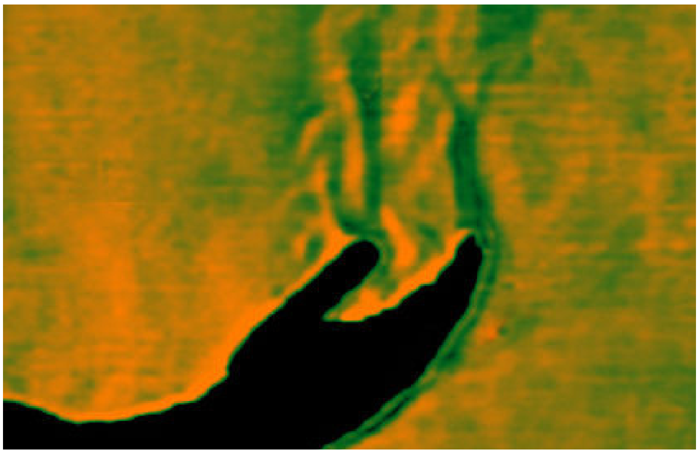
HEAT AND MASS TRANSFER emphasize on the principles of the Heat Transfer in steady state by conduction, convection and radiation. Principles of steady-state and transient heat conduction in solid are investigated. Laminar and turbulent boundary layer flows are treated, as well as condensation and boiling phenomena, thermal radiation, and radiation heat transfer between surfaces. Students will be exposed to the procedure for general problem solving and its application on heat exchanger. Student will be also exposed to the mass transfer mechanisms and develop relations for mass transfer rate for situations commonly encountered in practice.

In this ebook, Heat and Mass Transfer Volume 1 will focused in the Introduction and Basic Concepts, Steady State Heat Transfer in Conduction and Steady State Heat Transfer in Convection. This book will help polytechnic student to refer following the course outline. The main objective of this book to help student more understand the content of the subject referring to the course outline with the excercises and examples.

Last but not least we would like to express our gratitude to our family, colleagues, e-learning unit, UIDM, library , management of Politeknik Tun Syed Nasir Syed Ismail and also Jabatan Pendidikan Polteknik and Kolej Komuniti for giving the chance to publish this Heat And Mass Transfer Volume 1 ebook.

TABLE OF CONTENTS





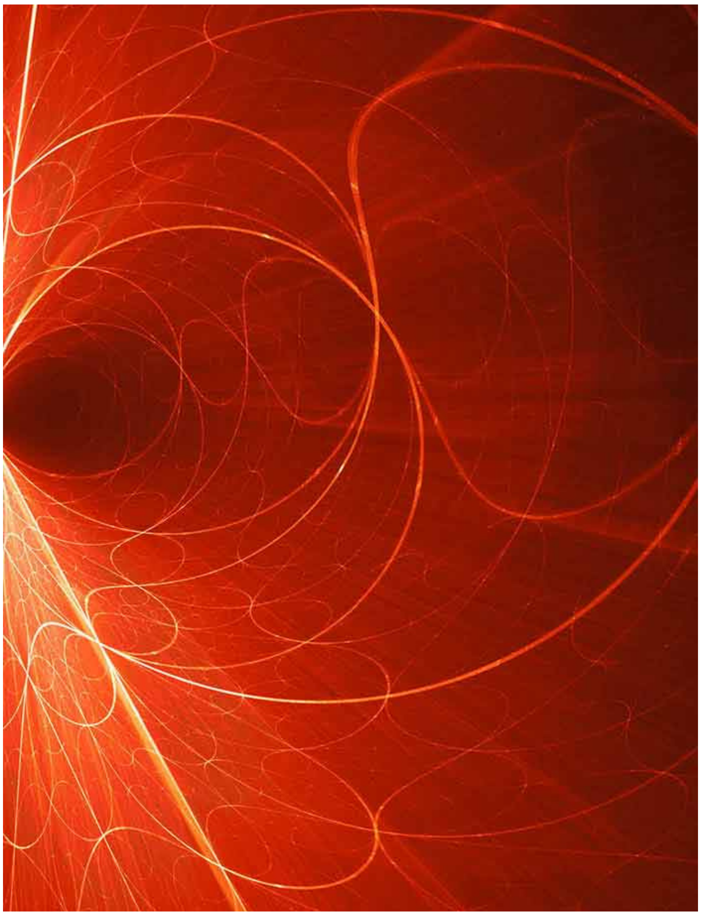
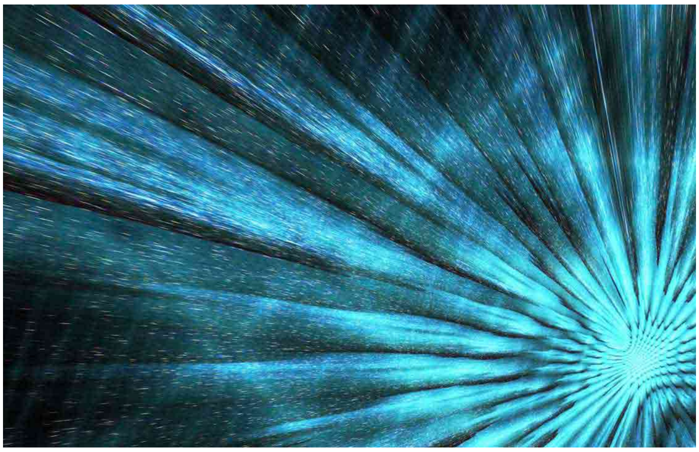
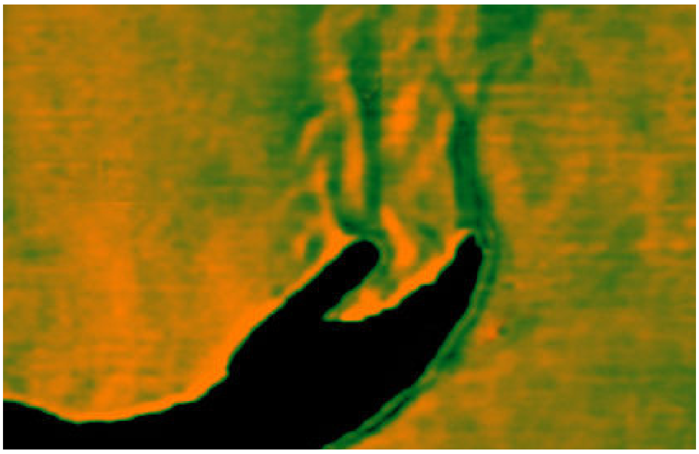
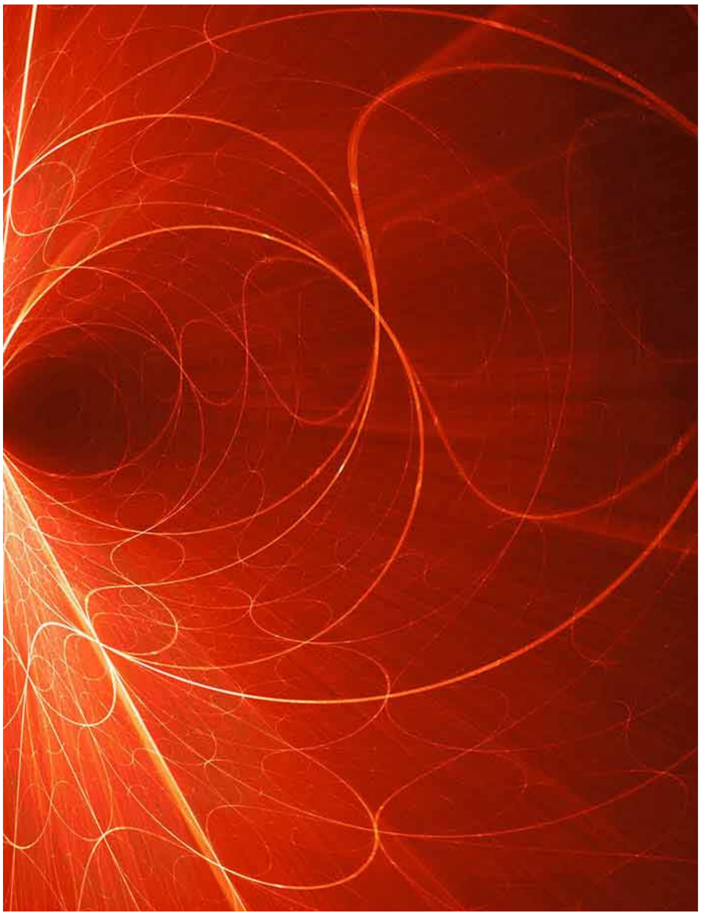


TABLE OF CONTENTS







CHAPTER 1

INTRODUCTION AND BASIC CONCEPTS

At the end of this chapter, you should be able to:

- Discuss the relationship between thermodynamics and heat transfer
- State energy and heat units.
- Apply the mechanisms of heat transfer.

1.1 INTRODUCTION

The science of thermodynamics deals with the amount of heat transfer as a system undergoes a process from one equilibrium state to another, and makes no reference to how long the process will take. But in engineering, we are often interested in the rate of heat transfer, which is the topic of the science of heat transfer.

There are three basic mechanisms of heat transfer, which are conduction, convection, and radiation. 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. Convection is the mode of heat transfer between a solid surface and the adjacent liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion. Radiation is the energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules.

1.1.1 THERMODYNAMIC AND HEAT TRANSFER

Heat always move from warmer place to cooler place. Hot object in cooler room will cool to room temperature. Cold objects in warmer room will heat up to room temperature. Energy transfer always from higher temperature medium to the lower temperature medium and the energy transfer stops when the two mediums reach the same temperature. Heat transfer is a physical process by which thermal energy is exchanged between material bodies or inside the same body as a result of a temperature difference. Heat transfer is the study of the mechanism and rate of this process.

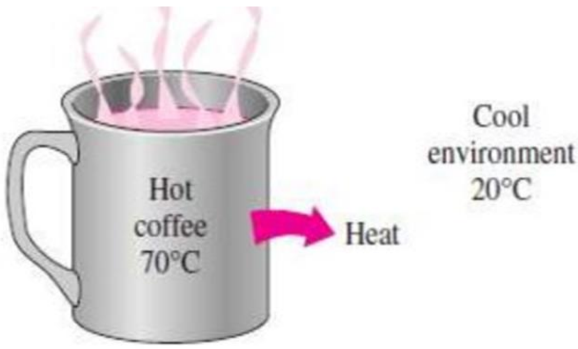


Figure 1.1 Mechanism of heat transfer

You may be wondering why we need to undertake a detailed study on heat transfer. After all, we can determine the amount of heat transfer for any system undergoing any process using a thermodynamic analysis alone. The reason is that thermodynamics is concerned with the amount of heat transfer as a system undergoes a process from one equilibrium state to another, and it gives no indication about how long the process will take. A thermodynamic analysis simply tells us how much heat must be transferred to realize a specified change of state to satisfy the conservation of energy principle. In practice we are more concerned about the rate of heat transfer (heat transfer per unit time) than we are with the amount of it. For example, we can determine the amount of heat transferred from a thermos bottle as the hot coffee inside cools from 90°C to 80°C by a thermodynamic analysis alone. But a typical user or designer of a thermos is primarily interested in how long it will be before the hot coffee inside cools to 80°C, and a thermodynamic analysis cannot answer this question. Determining the rates of heat transfer to or from a system and thus the times of cooling or heating, as well as the variation of the temperature, is the subject of heat transfer.

1.1.2 APPLICATION OF HEAT TRANSFER

Heat transfer is commonly encountered in engineering systems and other aspects of life. The human body is constantly rejecting heat to its surroundings, and human comfort is closely tied to the rate of this heat rejection. We try to control this heat transfer rate by adjusting our clothing to the environmental conditions.

Another commonly application is in household appliances such as electric and gas range, the heating equipment, air-conditioning system, the refrigerator and freezer, the water heater, the iron and even the computer or television. Heat transfer also applied in a lot of devices such as car radiators, solar collectors, various components of power plants and even spacecraft.

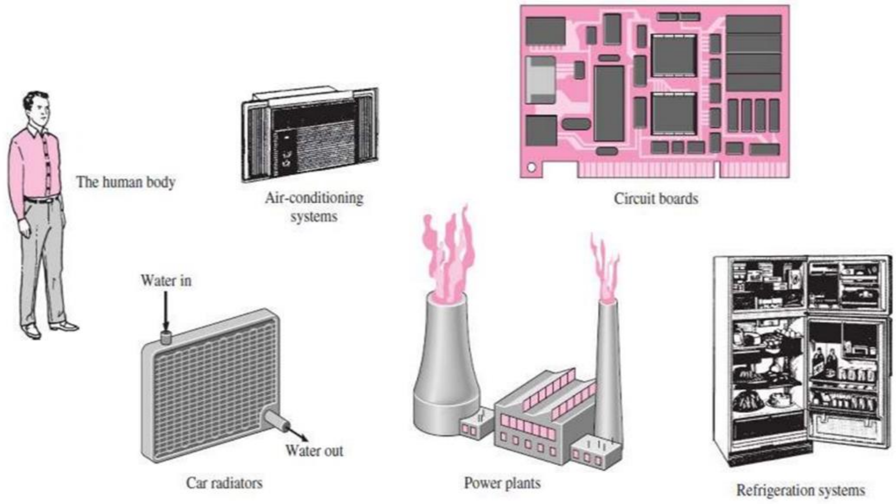


Figure 1.2 Example of application of heat transfer

1.2 ENERGY AND HEAT UNITS

Thermal energy is related to the temperature of matter. For a given material and mass, the higher the temperature, the greater its thermal energy. Heat transfer is a study of the exchange of thermal energy through a body or between bodies which occurs when there is a temperature difference. When two bodies are at different temperatures, thermal energy transfers from the one with higher temperature to the one with lower temperature.

1.2.1 ENERGY

Energy is the ability or capacity to do work on some form of matter. There are several forms of energy, including the following:

- **Potential energy** is the energy which a body possesses as a consequence of its position in a gravitational field
- **Kinetic energy** is the energy which a body possesses as a consequence of its motion (e.g., wind blowing across a wind generator). It is dependent upon an object's mass and velocity
- **Internal energy** is the total energy (potential and kinetic) stored in molecules.
- **Heat (or thermal) energy** is kinetic energy due to motion of atoms and molecules. It is energy that is in the process of being transferred from one object to another because of their temperature difference.

- **Radiant energy** is the energy that propagates through space or through material media in the form of electromagnetic radiation

1.2.2 HEAT

Heat is a form of energy which is passes from a body at higher temperature to a body at lower temperature. Heat is defined in physics as the transfer of thermal energy across a well-defined boundary around a thermodynamic system. Heat is total internal kinetic energy due to molecular motion in an object.

1.2.3 UNITS OF ENERGY AND HEAT

The international unit of energy is joule (J) or kilojoule (1 kJ = 1000 J). In the English system, the unit of energy is the British thermal unit (Btu), which is defined as the energy needed to raise the temperature of 1 lbm of water at 60°F by 1°F. The magnitudes of kJ and Btu are almost identical (1 Btu = 1.055056 kJ). Another well-known unit of energy is the calorie (1 cal = 4.1868 J), which is defined as the energy needed to raise the temperature of 1 gram of water at 14.5°C by 1°C.

1.3 MECHANISM OF HEAT TRANSFER

Heat can be transferred in three basic modes which are conduction, convection and radiation. All modes of heat transfer require the existence of a temperature difference. All modes are from the high-temperature medium to a lower-temperature one.

1.3.1 CONDUCTION

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. Conduction can take place in solids, liquids or gases. In gases and liquids conduction is due to the collisions and diffusion of the molecules during their random motion. In solids conduction is due to the combination of vibrations of the molecules in a lattice and the energy transport by free electrons.

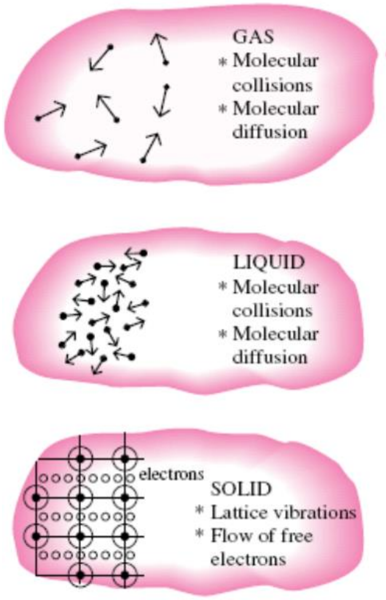


Figure 1.3 Motion of particles in gas liquid and solid

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. We know that wrapping a hot water tank with glass wool (an insulating material) reduces the rate of heat loss from the tank. The thicker the insulation, the smaller the heat loss. We also know that a hot water tank will lose heat at a higher rate when the temperature of the room housing the tank is lowered. Further, the larger the tank, the larger the surface area and thus the rate of heat loss.

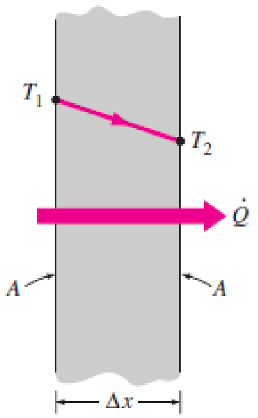
Consider steady heat conduction through a large plane wall of thickness $\Delta x = L$ and area A, as shown in Figure 1-4. The temperature difference across the wall is $\Delta T = T_2 - T_1$. Experiments have shown that the rate of heat transfer $\dot{Q}$ through the wall is *doubled* when the temperature difference ΔT across *halved* when the wall thickness L is doubled. Thus we conclude that *the rate of heat conduction through a plane layer is proportional to the temperature difference across the layer and the heat transfer area, but is inversely proportional to the thickness of the layer.* That is

$$Rate\ of\ heat\ conduction \propto \frac{(Area)(Temperature\ difference)}{Thickness}$$

or

$$\dot{Q}_{\text{cond}} = kA\frac{T_1 - T_2}{\Delta x} = -kA\frac{\Delta T}{\Delta x} \qquad \text{(W)}$$

(1-22)



Heat conduction through a large plane wall of thickness Δx and area A.

Figure 1.4 Heat conduction

where the constant of proportionality *k* is the **thermal conductivity** of the material, which is a *measure of the ability of a material to conduct heat.* In the limiting case of $\Delta x \rightarrow 0$, the equation above reduces to the differential form

$$\dot{Q}_{\text{cond}} = -kA\frac{dT}{dx} \qquad \text{(W)}$$

which is called **Fourier's law of heat conduction** after J. Fourier, who expressed it first in his heat transfer text in 1822. Here *dT/dx* is the **temperature gradient,** which is the slope of the temperature curve on a *T-x* diagram (the rate of change of *T* with *x*), at location *x.* The relation above indicates that the rate of heat conduction in a direction is proportional to the temperature gradient in that direction. Heat is conducted in the direction of decreasing temperature, and the temperature gradient becomes negative when temperature decreases with increasing *x.* The *negative sign* in equation is ensures that heat transfer in the positive *x* direction is a positive quantity.

Thermal Conductivity

Thermal Conductivity, k is the rate of heat transfer through a unit thickness of the material per unit area per unit temperature difference. The thermal conductivity of a material is a measure of the ability of the material to conduct heat. High value for thermal conductivity refer to good heat conductor while low value indicates poor heat conductor or insulator.

Table 1.1 The thermal conductivities of some material at room temperature

Material	k, W/m · °C*
Diamond	2300
Silver	429
Copper	401
Gold	317
Aluminum	237
Iron	80.2
Mercury (l)	8.54
Glass	0.78
Brick	0.72
Water (l)	0.613
Human skin	0.37
Wood (oak)	0.17
Helium (g)	0.152
Soft rubber	0.13
Glass fiber	0.043
Air (g)	0.026
Urethane, rigid foam	0.026

*Multiply by 0.5778 to convert to Btu/h · ft · °F.

***EXAMPLE 1.1* The Cost of Heat Loss through a Roof**

The roof of an electrically heated home is 6 m long, 8 m wide, and 0.25 m thick, and is made of a flat layer of concrete whose thermal conductivity is k = 0.8 W/m · °C (Fig. 1-5). The temperatures of the inner and the outer surfaces of the roof one night are measured to be 15°C and 4°C, respectively, for a period of 10 hours. Determine (*a*) the rate of heat loss through the roof that night and (*b*) the cost of that heat loss to the home owner if the cost of electricity is $0.08/kWh.

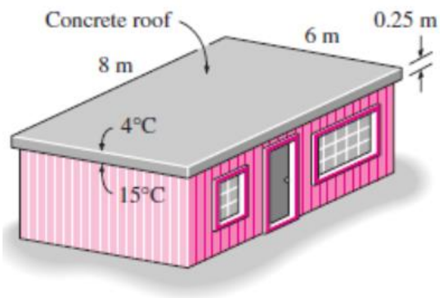


Figure 1.5 Heat loss through a roof

SOLUTION

The inner and outer surfaces of the flat concrete roof of an electrically heated home are maintained at specified temperatures during the night. The heat loss through the roof and its cost that night is to be determined.

Assumptions

Steady operating conditions exist during the entire night since the surface temperatures of the roof remain constant at the specified values.

Constant properties can be used for the roof.

Properties

The thermal conductivity of the roof is given to be $k = 0.8 W/m \cdot °C$.

Analysis

(a) Noting that heat transfer through the roof is by conduction and the area of the roof is

$A = 6 \text{ m} \times 8 \text{ m} = 48 \text{ m}^2$,

the steady rate of heat transfer through the roof is determined to be

$\dot{Q} = kA\frac{T_2 - T_1}{L} = (0.8 \text{ W/m} \cdot °\text{C})(48 \text{ m}^2)\frac{(15-4)°\text{C}}{0.28} =$ **1690 W = 1.69 kW**

(b) The amount of heat lost through the roof during a 10-hour period and its cost are determined from

$Q = \dot{Q}\Delta t = (1.69 \text{ kW})(10 \text{ h}) = 16.9 \text{ kWh}$

Cost = (Amount of energy)(Unit cost of energy)

= (16.9 kWh)($0.08/kWh) =**$1.35**

Discussion

The cost to the home owner of the heat loss through the roof that night was $1.35. The total heating bill of the house will be much larger since the heat losses through the walls are not considered in these calculations.

1.3.2 CONVECTION

Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion. The faster the fluid motion, the greater the convection heat transfer. In the absence of any bulk fluid motion, heat transfer between a solid surface and the adjacent fluid is by pure conduction. The presence of bulk motion of the fluid enhances the heat transfer between the solid surface and the fluid, but it also complicates the determination of heat transfer rates.

Consider the cooling of a hot block by blowing cool air over its top surface (Fig. 1–6). Energy is first transferred to the air layer adjacent to the block by conduction. This energy is then carried away from the surface by convection, that is, by the combined effects of conduction within the air that is due to random motion of air molecules and the bulk or macroscopic motion of the air that removes the heated air near the surface and replaces it by the cooler air.

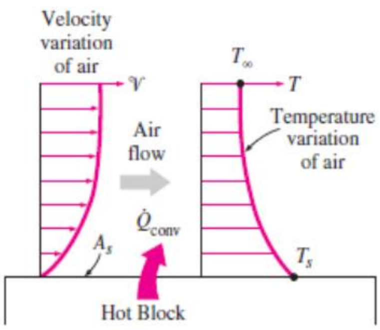


Figure 1.6 Heat transfer from hot surface to air by convection

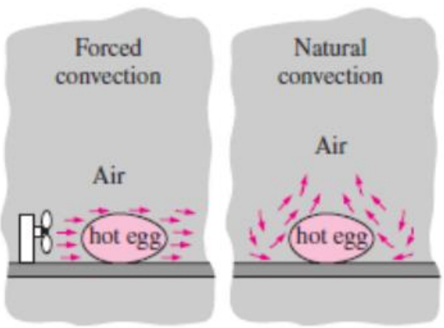


Figure 1.7 The cooling of a boiled egg by forced and natural convection.

Convection is called forced convection if the fluid is forced to flow over the surface by external means such as a fan, pump, or the wind. 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. 1–7). For example, in the absence of a fan, heat transfer from the surface of the hot block in Fig. 1–6 will be by natural convection since any motion in the air in this case will be due to the rise of the warmer (and thus lighter) air near the surface and the fall of the cooler (and thus heavier) air to fill its place. Heat transfer between the block and the surrounding air will be by conduction if the temperature difference between the air and the block is not large enough to overcome the resistance of air to movement and thus to initiate natural convection currents.

Heat transfer processes that involve change of phase of a fluid are also considered to be convection because of the fluid motion induced during the process, such as the rise of the vapor bubbles during boiling or the fall of the liquid droplets during condensation. Despite the complexity of convection, the rate of convection heat transfer is observed to be proportional to the temperature difference, and is conveniently expressed by Newton's law of cooling as

$$\dot{Q}_{conv} = hA_s\,(T_s - T_\infty) \qquad \text{(W)}$$

where h is the convection heat transfer coefficient in $W/m^2 \cdot °C$ or $Btu/h \cdot ft^2 \cdot °F$, as is the surface area through which convection heat transfer takes place, T_s is the surface temperature, and T_∞ is the temperature of the fluid sufficiently far from the surface. Note that at the surface, the fluid temperature equals the surface temperature of the solid.

The convection heat transfer coefficient h 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 the bulk fluid velocity. Typical values of h are given in Table 1.2.

Table 1.2 Typical values of convection heat transfer coefficient

Type of convection	h, W/m^2 · °C*
Free convection of gases	2–25
Free convection of liquids	10–1000
Forced convection of gases	25–250
Forced convection of liquids	50–20,000
Boiling and condensation	2500–100,000

*Multiply by 0.176 to convert to Btu/h · ft^2 · °F.

EXAMPLE 1.2 **Measuring Convection Heat Transfer Coefficient**

A 2-m-long, 0.3-cm-diameter electrical wire extends across a room at 15°C, as shown in Fig. 1-8. Heat is generated in the wire as a result of resistance heating, and the surface temperature of the wire is measured to be 152°C in steady operation. Also, the voltage drops and electric current through the wire are measured to be 60 V and 1.5 A, respectively. Disregarding any heat transfer by radiation, determine the convection heat transfer coefficient for heat transfer between the outer surface of the wire and the air in the room.

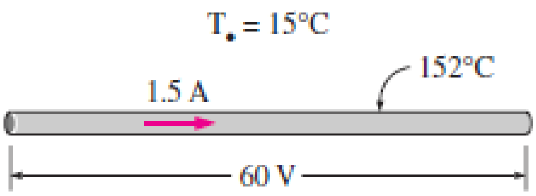


Figure 1.8 Surface of the wire

SOLUTION

The convection heat transfer coefficient for heat transfer from an electrically heated wire to air is to be determined by measuring temperatures when steady operating conditions are reached and the electric power consumed.

Assumptions

1 Steady operating conditions exist since the temperature readings do not change with time.

2 Radiation heat transfer is negligible.

Analysis

When steady operating conditions are reached, the rate of heat loss from the wire will equal the rate of heat generation in the wire as a result of resistance heating. That is,

$$\dot{Q} = \dot{E}_{\text{generated}} = VI = (60\text{ V})(1.5\text{ A}) = 90\text{ W}$$

The surface area of the wire is

$$A_s = \pi DL = \pi(0.003\text{ m})(2\text{ m}) = 0.01885\text{ m}^2$$

Newton's law of cooling for convection heat transfer is expressed as

$$\dot{Q}_{\text{conv}} = hA_s\,(T_s - T_\infty)$$

Disregarding any heat transfer by radiation and thus assuming all the heat loss from the wire to occur by convection, the convection heat transfer coefficient is determined to be

$$h = \frac{\dot{Q}_{\text{conv}}}{A_s(T_s - T_\infty)} = \frac{90\text{ W}}{(0.01885\text{ m}^2)(152 - 15)°\text{C}} = 34.9\text{ W/m}^2\cdot°\text{C}$$

Discussion

Note that the simple setup described above can be used to determine the average heat transfer coefficients from a variety of surfaces in air. Also, heat transfer by radiation can be eliminated by keeping the surrounding surfaces at the temperature of the wire.

1.3.3 RADIATION

Radiation is the energy emitted by matter in the form of electromagnetic waves (or photons) as a result of the changes in the electronic configurations of the atoms or molecules. Unlike conduction and convection, the transfer of energy by radiation does not require the presence of an intervening medium. In fact, energy transfer by radiation is fastest (at the speed of light) and it suffers no attenuation in a vacuum. This is how the energy of the sun reaches the earth.

In heat transfer studies we are interested in thermal radiation, which is the form of radiation emitted by bodies because of their temperature. It differs from other forms of electromagnetic radiation such as x-rays, gamma rays, microwaves, radio waves, and television waves that are not related to temperature. All bodies at a temperature above absolute zero emit thermal radiation.

Radiation is a volumetric phenomenon, and all solids, liquids, and gases emit, absorb, or transmit radiation to varying degrees. However, radiation is usually considered to be a surface phenomenon for solids that are opaque to thermal radiation such as metals, wood, and rocks since the radiation emitted by the interior regions of such material can never reach the surface, and the radiation incident on such bodies is usually absorbed within a few microns from the surface.

The maximum rate of radiation that can be emitted from a surface at an absolute temperature T_s (in K or R) is given by the Stefan–Boltzmann law as

$$\dot{Q}_{\text{emit, max}} = \sigma A_s T_s^4 \qquad \text{(W)}$$

where σ = 5.67 X 10^{-8} W/m^2 · K^4 or 0.1714 x 10-8 Btu/h · ft^2 · R^4 is the Stefan–Boltzmann constant. The idealized surface that emits radiation at this maximum rate is called a blackbody, and the radiation emitted by a blackbody is called blackbody radiation (Fig. 1-9). The radiation emitted by all real surfaces is less than the radiation emitted by a blackbody at the same temperature, and is expressed as

$$\dot{Q}_{\text{emit}} = \varepsilon \sigma A_s T_s^4 \qquad \text{(W)}$$

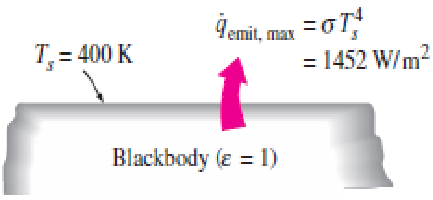


Figure 1.9 Blackbody

where ε is the emissivity of the surface. The property emissivity, whose value is in the range, is a measure of how closely a surface approximates a blackbody for which ε = 1. The emissivity of some surfaces are given in Table 1.3.

Table 1.3 Emissivity of some materials at 300K

Material	Emissivity
Aluminum foil	0.07
Anodized aluminum	0.82
Polished copper	0.03
Polished gold	0.03
Polished silver	0.02
Polished stainless steel	0.17
Black paint	0.98
White paint	0.90
White paper	0.92–0.97
Asphalt pavement	0.85–0.93
Red brick	0.93–0.96
Human skin	0.95
Wood	0.82–0.92
Soil	0.93–0.96
Water	0.96
Vegetation	0.92–0.96

Another important radiation property of a surface is its absorptivity α, which is the fraction of the radiation energy incident on a surface that is absorbed by the surface. Like emissivity, its value is in the range. A blackbody absorbs the entire radiation incident on it. That is, a blackbody is a perfect absorber (α= 1) as it is a perfect emitter.

In general, both ε and α of a surface depend on the temperature and the wavelength of the radiation. Kirchhoff's law of radiation states that the emissivity and the absorptivity of a surface at a given temperature and wavelength are equal. In many practical applications, the surface temperature and the temperature of the source of incident radiation are of the same order of magnitude, and the average absorptivity of a surface is taken to be equal to its average emissivity. The rate at which a surface absorbs radiation is determined from (Fig. 1-10)

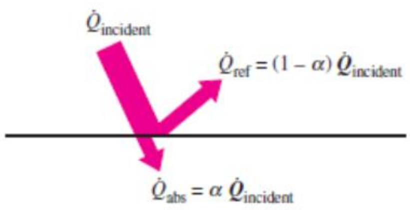


Figure 1.10 The absorption of radiation incident on an opaque surface of absorptivity

$$\dot{Q}_{\text{absorbed}} = \alpha \dot{Q}_{\text{incident}} \qquad \text{(W)}$$

where Q incident is the rate at which radiation is incident on the surface and is the absorptivity of the surface. For opaque (nontransparent) surfaces, the portion of incident radiation not absorbed by the surface is reflected back.

The difference between the rates of radiation emitted by the surface and the radiation absorbed is the net radiation heat transfer. If the rate of radiation absorption is greater than the rate of radiation emission, the surface is said to be gaining energy by radiation. Otherwise, the surface is said to be losing energy by radiation. In general, the determination of the net rate of heat transfer by radiation between two surfaces is a complicated matter since it depends on the properties of the surfaces, their orientation relative to each other, and the interaction of the medium between the surfaces with radiation.

When a surface of emissivity and surface area as at an absolute temperature T_s is completely enclosed by a much larger (or black) surface at absolute temperature T_{surr} separated by a gas (such as air) that does not intervene with radiation, the net rate of radiation heat transfer between these two surfaces is given by (Fig. 1-11)

$$\dot{Q}_{\text{rad}} = \varepsilon \sigma A_s \left(T_s^4 - T_{\text{surr}}^4\right) \qquad \text{(W)}$$

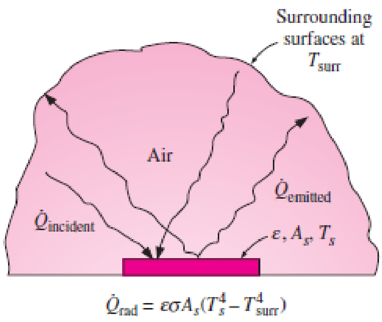


Figure 1.11 Radiation heat transfer between a surface and the surfaces surrounding it.

In this special case, the emissivity and the surface area of the surrounding surface do not have any effect on the net radiation heat transfer.

Radiation heat transfer to or from a surface surrounded by a gas such as air occurs parallel to conduction (or convection, if there is bulk gas motion) between the surface and the gas. Thus the total heat transfer is determined by adding the contributions of both heat transfer mechanisms. For simplicity and convenience, this is often done by defining a combined heat transfer coefficient $h_{combined}$ that includes the effects of both convection and radiation. Then the total heat transfer rate to or from a surface by convection and radiation is expressed as

$$\dot{Q}_{total} = h_{combined} A_s (T_s - T_\infty) \qquad \text{(W)}$$

Note that the combined heat transfer coefficient is essentially a convection heat transfer coefficient modified to include the effects of radiation. Radiation is usually significant relative to conduction or natural convection, but negligible relative to forced convection. Thus radiation in forced convection applications is usually disregarded, especially when the surfaces involved have low emissivity and low to moderate temperatures.

EXAMPLE 1.3 **Radiation Effect on Thermal Comfort**

It is a common experience to feel "chilly" in winter and "warm" in summer in our homes even when the thermostat setting is kept the same. This is due to the so called "radiation effect" resulting from radiation heat exchange between our bodies and the surrounding surfaces of the walls and the ceiling.

Consider a person standing in a room maintained at 22°C at all times. The inner surfaces of the walls, floors, and the ceiling of the house are observed to be at an average temperature of 10°C in winter and 25°C in summer. Determine the rate of radiation heat transfer between this person and the surrounding surfaces if the exposed surface area and the average outer surface temperature of the person are 1.4 m^2 and 30°C, respectively (Fig. 1-12).

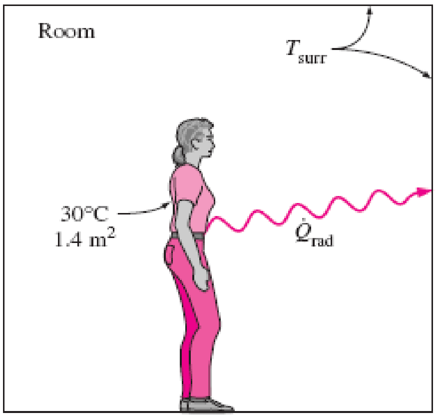


Figure 1.12 Schematic for Example 1.3

SOLUTION

The rates of radiation heat transfer between a person and the surrounding surfaces at specified temperatures are to be determined in summer and winter.

Assumptions

1 Steady operating conditions exist.

2 Heat transfer by convection is not considered.

3 The person is completely surrounded by the interior surfaces of the room.

4 The surrounding surfaces are at a uniform temperature.

Properties

The emissivity of a person is ε = 0.95 (Table 1.3).

Analysis

The net rates of radiation heat transfer from the body to the surrounding walls, ceiling, and floor in winter and summer are

$$\begin{aligned}\dot{Q}_{\text{rad, winter}} &= \varepsilon\sigma A_s (T_s^4 - T_{\text{surr, winter}}^4)\\ &= (0.95)(5.67 \times 10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(1.4\ \text{m}^2)\\ &\quad \times [(30 + 273)^4 - (10 + 273)^4]\ \text{K}^4\\ &= \mathbf{152\ W}\end{aligned}$$

$$\begin{aligned}\dot{Q}_{\text{rad, summer}} &= \varepsilon\sigma A_s (T_s^4 - T_{\text{surr, summer}}^4)\\ &= (0.95)(5.67 \times 10^{-8}\ \text{W/m}^2\cdot\text{K}^4)(1.4\ \text{m}^2)\\ &\quad \times [(30 + 273)^4 - (25 + 273)^4]\ \text{K}^4\\ &= \mathbf{40.9\ W}\end{aligned}$$

Discussion

Note that we must use absolute temperatures in radiation calculations. Also note that the rate of heat loss from the person by radiation is almost four times as large in winter than it is in summer, which explains the “chill” we feel in winter even if the thermostat setting is kept the same.

EXERCISES/TUTORIAL

1. How does the science of heat transfer differ from the science of thermodynamics

2. The inner and outer surfaces of a 0.5-cm thick 2-m by 2-m window glass in winter are 10°C and 3°C, respectively. If the thermal conductivity of the glass is 0.78 W/m·K, determine the amount of heat loss through the glass over a period of 5 h. What would your answer be if the glass were 1 cm thick?

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

4. Consider a person whose exposed surface area is 1.7 m^2, emissivity is 0.5, and surface temperature is 32°C. Determine the rate of heat loss from that person by radiation in a large room having walls at a temperature of (a) 300 K and (b) 280 K.

5. An inexpensive food and beverage container is fabricated from 25-mm-thick polystyrene (k = 0.023 W/m . K) and has interior dimensions of 0.8 m x 0.6 m x 0.6 m. Under conditions for which an inner surface temperature of approximately 2 ^oC is maintained by an ice-water mixture and an outer surface temperature of 20 ^oC is maintained by the ambient, what is the heat flux through the container wall?

6. You've experienced convection cooling if you've ever extended your hand out the window of a moving vehicle or into a flowing water stream. With the surface of your hand at a temperature of 30C, determine the convection heat flux for (a) a vehicle speed of 35 km/h in air at -5 ^oC with a convection coefficient of 40 W/m^2 . K and (b) a velocity of 0.2 m/s in a water stream at 10^oC with a convection coefficient of 900 W/m^2 .K.

7. A cartridge electrical heater is shaped as a cylinder of length L = 200 mm and outer diameter D =20 mm. Under normal operating conditions the heater dissipates 2 kW while submerged in a water flow that is at 20^oC and provides a convection heat transfer coefficient of h =5000 W/m^2 . K. Neglecting heat transfer from the ends of the heater, determine its surface temperature T_s

8. A 5-cm-external-diameter, 10-m-long hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 $W/m^2 \cdot °C$. Calculate the rate of heat loss from the pipe by natural convection, in W.

CHAPTER 2

PRINCIPLES OF STEADY STATE HEAT TRANSFER IN CONDUCTION

At the end of this chapter, you should be able to:

- Apply steady heat conductions in plane walls.
- Figure out the concept of thermal resistance.
- Apply steady conduction problems that involve multilayer plane walls, cylinders and spheres.

2.0 STEADY HEAT CONDUCTION IN PLANE WALLS

Consider steady heat conduction through the walls of a house during a winter day. Heat is continuously lost to the outdoors through the wall. Heat transfer through the wall is in the normal direction to the wall surface, and no significant heat transfer takes place in the wall in other directions (Fig. 2–1).

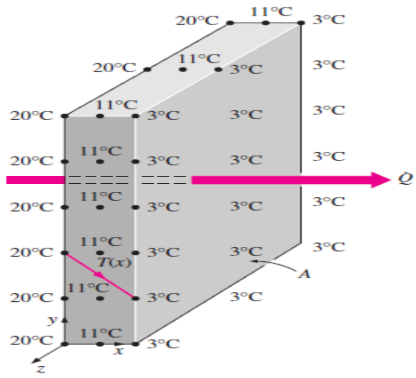


Figure 2.1 Heat transfer through a wall is one dimensional when the temperature of the wall varies in one direction only.

Noting that heat transfer is the only energy interaction involved in this case and there is no heat generation, the energy balance for the wall can be expressed as

$$\begin{pmatrix}\text{Rate of}\\ \text{heat transfer}\\ \text{into the wall}\end{pmatrix} - \begin{pmatrix}\text{Rate of}\\ \text{heat transfer}\\ \text{out of the wall}\end{pmatrix} = \begin{pmatrix}\text{Rate of change}\\ \text{of the energy}\\ \text{of the wall}\end{pmatrix}$$

or

$$\dot{Q}_{in} - \dot{Q}_{out} = \frac{dE_{wall}}{dt} \qquad (2\text{-}1)$$

But $dE_{wall}/dt = 0$ for *steady operation*, since there is no change in the temperature of the wall with time at any point. Therefore, the rate of heat transfer into the wall must be equal to the rate of heat transfer out of it. In other words, the rate of heat transfer through the wall must be constant, $\dot{Q}_{cond,wall} = constant$.

2.1 Fourier's Law of heat conduction

Consider a plane wall of thickness L and average thermal conductivity k. The two surfaces of the wall are maintained at constant temperatures of T_1 and T_2. For one-dimensional steady heat conduction through the wall, we have T(x). Then Fourier's law of heat conduction for the wall can be expressed as

$$\dot{Q}_{cond,wall} = -kA\frac{dT}{dx} \qquad \text{(W)}$$

where the rate of conduction heat transfer $\dot{Q}$cond wall and the wall area A are constant. Thus we have d_T/d_x = constant, which means that the temperature through the wall varies linearly with x. That is, the temperature distribution in the wall under steady conditions is a straight line (Fig. 2–2).

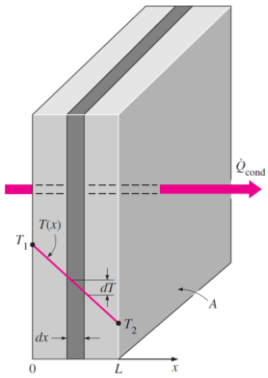


Figure 2.2 Under steady conditions, the temperature distribution in a plane wall is a straight line

After performing the integrations and rearranging gives

$$\dot{Q}_{cond,wall} = kA\frac{T_1-T_2}{L} \qquad \text{(W)}$$

Again, the rate of heat conduction through a plane wall is proportional to the average thermal conductivity, the wall area, and the temperature difference, but is inversely proportional to the wall thickness. Also, once the rate of heat conduction is available, the temperature T(x) at any location x can be determined by replacing T_2 in the equation by T, and L by x.

EXAMPLE 2.1 Heat Loss Through the Wall

Consider a 4-m-high, 6-m-wide, and 0.3-m-thick brick wall whose thermal conductivity is k = 0.8 W/m · °C . On a certain day, the temperatures of the inner and the outer surfaces of the wall are measured to be 14°C and 6°C, respectively. Determine the rate of heat loss through the wall on that day.

SOLUTION

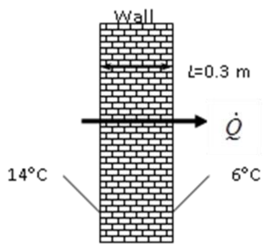
The two surfaces of a wall are maintained at specified temperatures. The rate of heat loss through the wall is to be determined.

Assumptions

1. Heat transfer through the wall is steady since the surface temperatures remain constant at the specified values.

2. Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors.

3. Thermal conductivity is constant.



Properties

The thermal conductivity is given to be k = 0.8 W/m^2°C.

Analysis The surface area of the wall and the rate of heat loss through the wall are

$$A = 4m\ x\ 6m = 24m^2$$

$$Q = kA\frac{T_1 - T_2}{L} = \left(\frac{0.8w}{m°C}\right)(24m^2)\frac{(14-6)°C}{0.3m} = 512W$$

2.2 THERMAL RESISTANCE CONCEPT

2.2.1 Thermal Resistance for Conduction, Convection and Radiation

Equation for heat conduction through a plane wall can be rearranged as

$$\dot{Q}_{cond,wall} = \frac{T_1 - T_2}{R_{wall}} \qquad \text{(W)}$$

where

$$R_{wall} = \frac{L}{kA} \qquad (^{o}\text{C/W})$$

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.

For convection heat transfer from a solid surface of area A_s and temperature T_s to a fluid whose temperature sufficiently far from the surface is T_∞, with a convection heat transfer coefficient h. Newton's law of cooling for convection heat transfer rate $\dot{Q}_{conv} = hA_s(T_s - T_\infty)$ can be rearranged as

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

Where

$$R_{conv} = \frac{1}{hA_s} \qquad (^{o}\text{C/W})$$

is the *thermal resistance* of the surface against heat convection, or simply the convection resistance of the surface (Fig. 2-3).

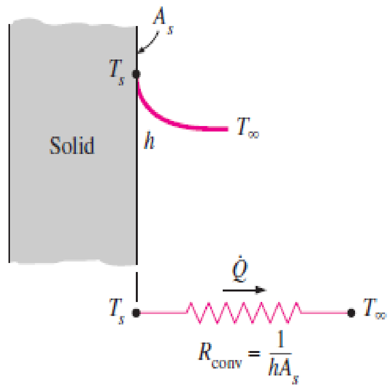


Figure 2.3 Schematic for convection resistance at a surface

Note that when the convection heat transfer coefficient is very large (h → ∞), the convection resistance becomes zero and Ts ≈ T∞. That is, the surface offers no resistance to convection, and thus it does not slow down the heat transfer process.

When the wall is surrounded by a gas, the *radiation effects*, which we have ignored so far, can be significant and may need to be considered. The rate of radiation heat transfers between a surface of emissivity ε and area A_s at temperature T_s and the surrounding surfaces at some average temperature T_{surr} can be expressed as

$$\dot{Q}_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{surr}^4) = h_{rad}A_s(T_s - T_{surr)} = \frac{T_s - T_{surr}}{R_{rad}} \qquad \text{(W)}$$

where

$$R_{rad} = \frac{1}{h_{rad}A_s} \qquad \text{(K/W)}$$

is the *thermal resistance* of a surface against radiation, or the radiation resistance, and

$$h_{rad} = \frac{\dot{Q}_{rad}}{A_s(T_s - T_{surr})} = \varepsilon\sigma(T_s^2 + T_{surr}^2)(T_s + T_{surr}) \qquad \text{(W/m}^2\text{.K)}$$

is the radiation heat transfer coefficient. Note that both T_s and T_{surr} must be in K in the evaluation of h_{rad}. The definition of the radiation heat transfer coefficient enables us to express radiation conveniently in an analogous manner to convection in terms of a temperature difference. But h_{rad} depends strongly on temperature while h_{conv} usually does not.

A surface exposed to the surrounding air involves convection and radiation simultaneously, and the total heat transfer at the surface is determined by adding (or subtracting, if in the opposite direction) the radiation and convection components. The convection and radiation resistances are parallel to each other, as shown in Fig. 2-4, and may cause some complication in the thermal resistance network. When $T_{surr} \approx T\infty$, the radiation effect can properly be accounted for by replacing h in the convection resistance relation by

$$h_{combined} = h_{conv} + h_{rad} \qquad (W/m2 \cdot K) \qquad (2\text{-}12)$$

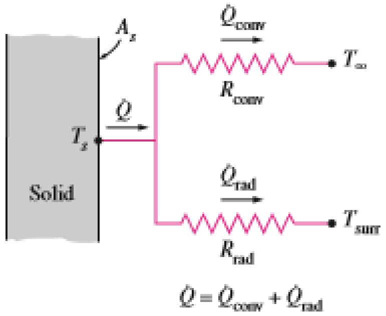
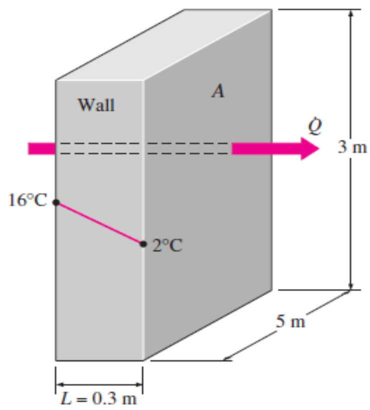


Figure 2.4 Schematic for convection and radiation resistance at a surface

EXAMPLE 2.2 **Heat Loss through a Wall**

Consider a 3-m-high, 5-m-wide, and 0.3-m-thick wall whose thermal conductivity is k = 0.9 W/m · °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.



SOLUTION

The two surfaces of a wall are maintained at specified temperatures. The rate of heat loss through the wall is to be determined.

Assumptions

1 Heat transfer through the wall is steady since the surface temperatures remain constant at the specified values.

2 Heat transfer is one dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors.

3 Thermal conductivity is constant.

Properties

The thermal conductivity is given to be *k* = 0.9 W/m · °C.

Analysis

Noting that the heat transfer through the wall is by conduction

A = 3 m x 5 m = 15 m²

$$\dot{Q} = kA\frac{T_1 - T_2}{L} = (0.9W/m℃)(15m^2)\frac{(16-2)℃}{0.3m} = 630W$$

We could also determine the steady rate of heat transfer through the wall by making use of the thermal resistance concept from

$$\dot{Q} = \frac{\Delta T_{wall}}{R_{wall}}$$

Where

$$R_{wall} = \frac{L}{kA} = \frac{0.3m}{(0.9W/m℃)(15m^2)} = 0.02222℃/W$$

Substituting, we get

$$\dot{Q} = \frac{(16-2)℃}{0.02222℃/W} = 630W$$

2.2.2 Analogy between thermal and electrical resistance concepts.

The equation for heat flow is analogous to the relation for electric current flow I, expressed as

$$I = \frac{V_1 - V_2}{R_e}$$

where $R_e = L/\sigma_e A$ is the electric resistance and V_1-V_2 is the *voltage difference* across the resistance (σ_e is the electrical conductivity). Thus, the rate of heat transfer through a layer corresponds to the electric current, the thermal resistance corresponds to electrical resistance, and the temperature difference corresponds to voltage difference across the layer

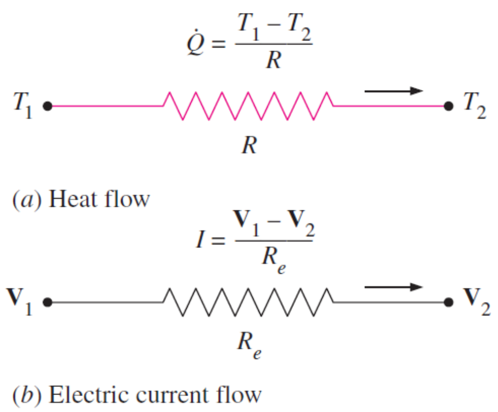


Figure 2.5 Analogy between thermal and electrical resistance concepts

2.2.3 Thermal Resistance Network

Consider steady one-dimensional heat flow through a plane wall of thickness L, area A, and thermal conductivity k that is exposed to convection on both sides to fluids at temperatures $T_{\infty1}$ and $T_{\infty2}$ with heat transfer coefficients h_1 and h_2, respectively, as shown in Fig. 2–6. Assuming $T_{\infty2} < T_{\infty1}$, the variation of temperature will be as shown in the figure.

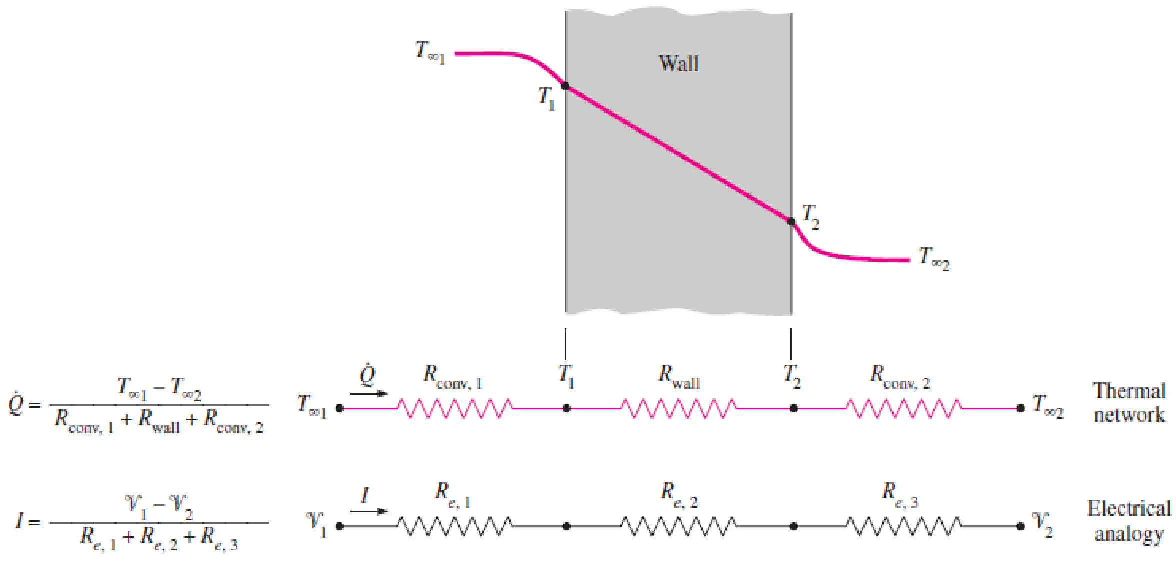


Figure 2.6 The thermal resistance network for heat transfer through a wall subjected to convection on both sides and the electrical analogy

Under steady conditions

$$\begin{pmatrix} Rate\ of \\ heat\ convection \\ into\ the\ wall \end{pmatrix} = \begin{pmatrix} Rate\ of \\ heat\ conduction \\ through\ the\ wall \end{pmatrix} = \begin{pmatrix} Rate\ of \\ heat\ convection \\ from\ the\ wall \end{pmatrix}$$

Or $$\dot{Q} = h_1 A(T_{\infty 1} - T_1) = kA\frac{T_1 - T_2}{L} = h_2 A(T_2 - T_{\infty 2})$$

Which can be rearranged as

$$\dot{Q} = \frac{T_{\infty 1} - T_1}{\frac{1}{h_1 A}} = \frac{T_1 - T_2}{\frac{L}{kA}} = \frac{T_2 - T_{\infty 2}}{\frac{1}{h_2 A}}$$

$$= \frac{T_{\infty 1} - T_1}{R_{conv,1}} = \frac{T_1 - T_2}{R_{wall}} = \frac{T_2 - T_{\infty 2}}{R_{conv,2}}$$

Adding the numerators and denominators yields

$$\boldsymbol{\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}}} \qquad \text{(W)}$$

Where

$$\boldsymbol{R_{total} = R_{conv,1} + R_{wall} + R_{conv,2} = \frac{1}{h_1 A} + \frac{L}{kA} + \frac{1}{h_2 A}} \qquad \text{(°C/W)}$$

EXAMPLE 2.3 **Heat Loss through a Single-Pane Window**

Consider a 0.8-m-high and 1.5-m-wide glass window with a thickness of 8 mm and a thermal conductivity of k = 0.78 W/m · °C. Determine the steady rate of heat transfers through this glass 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 outdoors is -10°C. Take the heat transfer coefficients on the inner and outer surfaces of the window to be h_1 = 10 W/m^2 · °C and h_2 = 40 W/m^2 · °C, which includes the effects of radiation.

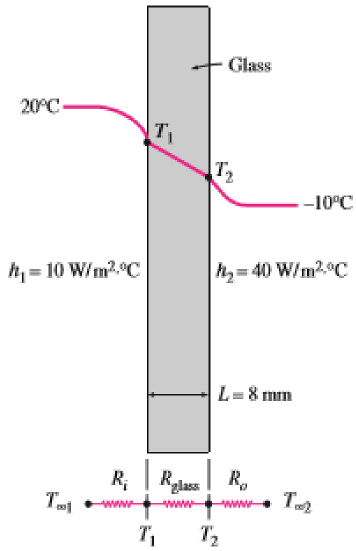


Figure 2.7 Thermal network resistance in single pane window

SOLUTION

Heat loss through a window glass is considered. The rate of heat transfers through the window and the inner surface temperature are to be determined.

Assumptions

1 Heat transfer through the window is steady since the surface temperatures remain constant at the specified values.

2 Heat transfer through the wall is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors.

3 Thermal conductivity is constant.

Properties

The thermal conductivity is given to be k = 0.78 W/m · °C.

Analysis

A = 0.8 m x 1.5 m = 1.2 m^2

$$R_i = R_{\text{conv},1} = \frac{1}{h_1 A} = \frac{1}{(10\ \text{W/m}^2\cdot{}^\circ\text{C})(1.2\ \text{m}^2)} = 0.08333^\circ\text{C/W}$$

$$R_{\text{glass}} = \frac{L}{kA} = \frac{0.008\ \text{m}}{(0.78\ \text{W/m}\cdot{}^\circ\text{C})(1.2\ \text{m}^2)} = 0.00855^\circ\text{C/W}$$

$$R_o = R_{\text{conv},2} = \frac{1}{h_2 A} = \frac{1}{(40\ \text{W/m}^2\cdot{}^\circ\text{C})(1.2\ \text{m}^2)} = 0.02083^\circ\text{C/W}$$

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

$$R_{\text{total}} = R_{\text{conv},1} + R_{\text{glass}} + R_{\text{conv},2} = 0.08333 + 0.00855 + 0.02083 = 0.1127^\circ\text{C/W}$$

Then the steady rate of heat transfer through the window becomes

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{\text{total}}} = \frac{[20-(-10)]^\circ\text{C}}{0.1127^\circ\text{C/W}} = \mathbf{266\ W}$$

Knowing the rate of heat transfer, the inner surface temperature of the window glass can be determined from

$$\dot{Q} = \frac{T_{\infty1} - T_1}{R_{\text{conv},1}} \longrightarrow T_1 = T_{\infty1} - \dot{Q}R_{\text{conv},1} = 20^\circ\text{C} - (266\ \text{W})(0.08333^\circ\text{C/W}) = \mathbf{-2.2^\circ C}$$

2.3 STEADY STATE HEAT TRANSFER THROUGH MULTILAYER PLANE WALLS, CYLINDERS AND SPHERES

2.3.1 Multilayer Plane Walls

Consider a plane wall that consists of two layers (such as a brick wall with a layer of insulation). The rate of steady heat transfer through this two-layer composite wall can be expressed as (Fig. 2.8)

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}}$$

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

$$R_{total} = R_{conv,1} + R_{wall,1} + R_{wall,2} + R_{conv,2}$$

$$= \frac{1}{h_1 A} + \frac{L_1}{k_1 A} + \frac{L_2}{k_2 A} + \frac{1}{h_2 A}$$

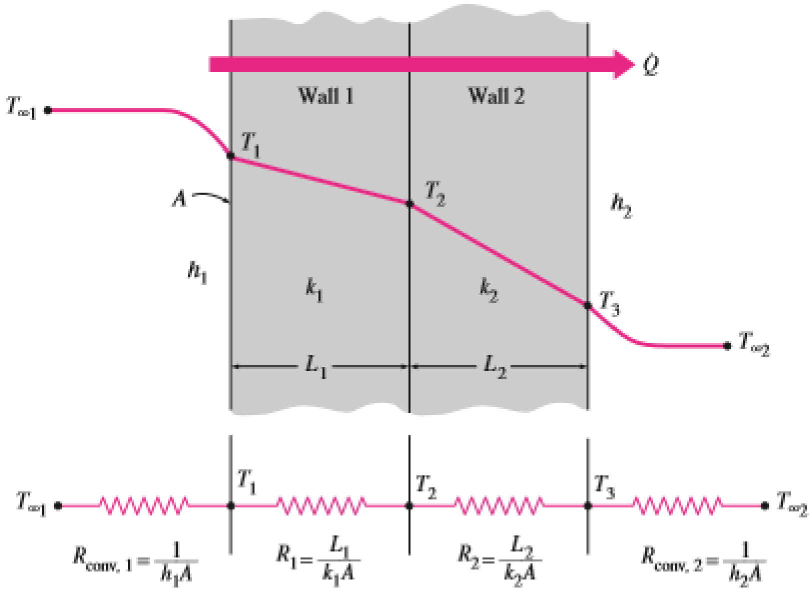


Figure 2.8 The thermal resistance network for heat transfer through a two-layer plane wall subjected to convection on both side

Note from the thermal resistance network that the resistances are in series, and thus the total thermal resistance is simply the arithmetic sum of the individual thermal resistances in the path of heat flow.

This result for the two-layer case is analogous to the single-layer case, except that an additional resistance is added for the additional layer. This result can be extended to plane walls that consist of three or more layers by adding an additional resistance for each additional layer.

When the fluid temperatures $T_{\infty 1}$ and $T_{\infty 2}$ for the two-layer case shown in Fig. 2.8 are available and $\dot{Q}$ is calculated from, the interface temperature T_2 between the two walls can be determined from (Fig. 2.9)

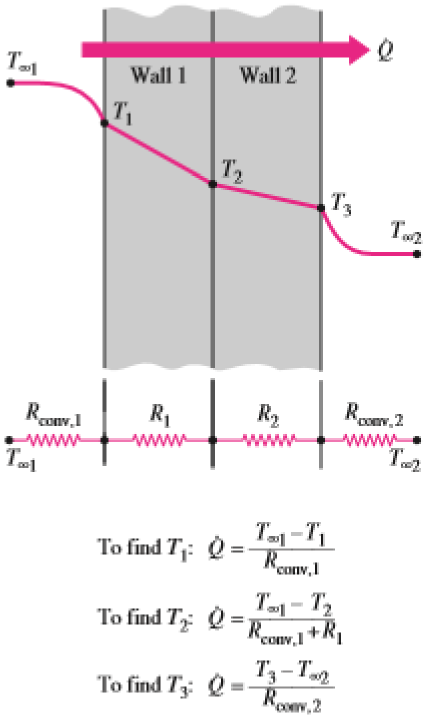


Figure 2.9 The evaluation of the surface and interface temperatures when $T_{\infty 1}$ and $T_{\infty 2}$ are given and $\dot{Q}$ is calculated

$$\dot{Q} = \frac{T\infty_1 - T_2}{R_{conv,1} - R_{wall,1}} = \frac{T\infty_1 - T_2}{\frac{1}{h_1 A} + \frac{L_1}{k_1 A}}$$

The thermal resistance concept is widely used in practice because it is intuitively easy to understand and it has proven to be a powerful tool in the solution of a wide range of heat transfer problems. But its use is limited to systems through which the rate of heat transfer $\dot{Q}$ remains constant; that is, to systems involving steady heat transfer with no heat generation (such as resistance heating or chemical reactions) within the medium.

EXAMPLE 2.4 **Heat Loss through Double-pane Windows**

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 outdoors is -10°C. Take the convection heat transfer coefficients on the inner and outer surfaces of the window to be h_1 = 10 W/m2 · °C and h_2 = 40 W/m2 · °C, which includes the effects of radiation.

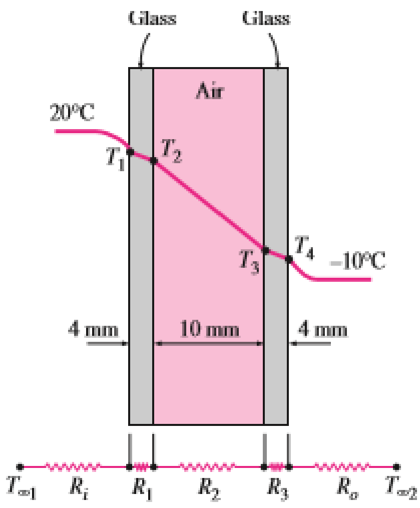


Figure 2.10 Heat loss through double pane windows

SOLUTION

A double-pane window is considered. The rate of heat transfers through the window and the inner surface temperature are to be determined.

Assumptions

1 Heat transfer through the window is steady since the surface temperatures remain constant at the specified values.

2 Heat transfer through the wall is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors.

3 Thermal conductivity is constant.

Properties

The thermal conductivity is given to be $k_1 = k_3$= 0.78 W/m · °C.

h_1 = 10 W/m2 · °C and h_2 = 40 W/m2 · °C

Analysis

A = 0.8m x 1.5m = 1.2m^2

$$R_i = R_{\text{conv, 1}} = \frac{1}{h_1 A} = \frac{1}{(10\ \text{W/m}^2 \cdot {}^\circ\text{C})(1.2\ \text{m}^2)} = 0.08333^\circ\text{C/W}$$

$$R_1 = R_3 = R_{\text{glass}} = \frac{L_1}{k_1 A} = \frac{0.004\ \text{m}}{(0.78\ \text{W/m} \cdot {}^\circ\text{C})(1.2\ \text{m}^2)} = 0.00427^\circ\text{C/W}$$

$$R_2 = R_{\text{air}} = \frac{L_2}{k_2 A} = \frac{0.01\ \text{m}}{(0.026\ \text{W/m} \cdot {}^\circ\text{C})(1.2\ \text{m}^2)} = 0.3205^\circ\text{C/W}$$

$$R_o = R_{\text{conv, 2}} = \frac{1}{h_2 A} = \frac{1}{(40\ \text{W/m}^2 \cdot {}^\circ\text{C})(1.2\ \text{m}^2)} = 0.02083^\circ\text{C/W}$$

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

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

Then the steady rate of heat transfer through the window becomes

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

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

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

Generalized Thermal Resistance Networks

The thermal resistance concept or the electrical analogy can also be used to solve steady heat transfer problems that involve parallel layers or combined series-parallel arrangements. Consider the composite wall shown in Fig. 2-11, which consists of two parallel layers.

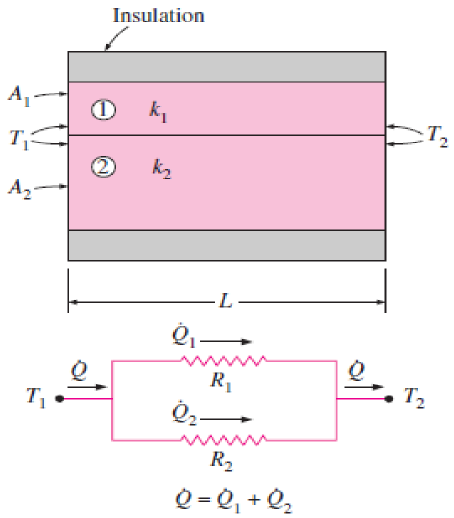


Figure 2.11 Thermal resistance network for two parallel layers

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

$$\dot{Q} = \dot{Q}_1 + \dot{Q}_2 = \frac{T_1 - T_2}{R_1} + \frac{T_1 - T_2}{R_2} = (T_1 - T_2)\left(\frac{1}{R_1} + \frac{1}{R_2}\right)$$

Utilizing electrical analogy, we get

$$\dot{Q} = \frac{T_1 - T_2}{R_{total}}$$

Where

$$\frac{1}{R_{total}} = \frac{1}{R_1} + \frac{1}{R_2} \rightarrow R_{total} = \frac{R_1 R_2}{R_1 + R_2}$$

since the resistances are in parallel.

Now consider the combined series-parallel arrangement shown in Fig. 2.12.

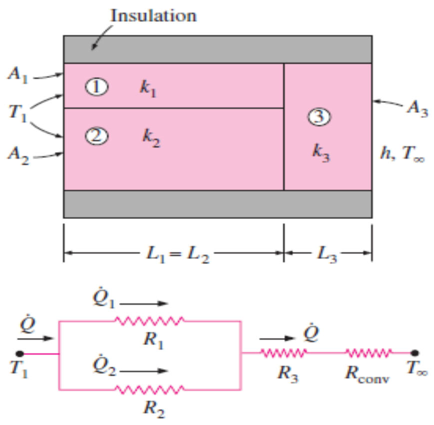


Figure 2.12 Thermal resistance network for combined series-parallel arrangement

The total rate of heat transfer through this composite system can again be expressed as

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{total}}$$

Where $$R_{total} = R_{12} + R_3 + R_{conv} = \frac{R_1 R_2}{R_1 + R_2} + R_3 + R_{conv}$$

And $$R_1 = \frac{L_1}{k_1 A_1} \qquad R_2 = \frac{L_2}{k_2 A_2} \qquad R_3 = \frac{L_3}{k_3 A_3} \qquad R_{conv} = \frac{1}{hA_3}$$

Once the individual thermal resistances are evaluated, the total resistance and the total rate of heat transfer can easily be determined from the relations above.

EXAMPLE 2.5 **Heat loss through composite wall**

Consider a 5-m-high, 8-m-long, and 0.22-m-thick wall whose representative cross section is as given in the figure. The thermal conductivities of various materials used, in W/m · °C, are $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, and $k_E = 35$. The left and right surfaces of the wall are maintained at uniform temperatures of 300°C and 100°C, respectively. Assuming heat transfer through the wall to be one-dimensional, determine the rate of heat transfer through the wall

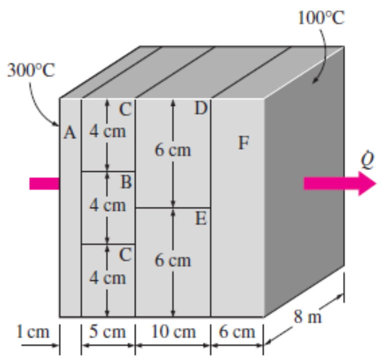


Figure 2.13 Composite wall

SOLUTION

A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfers through the wall, the interface temperatures, and the temperature drop across the section F are to be determined.

Assumptions

1 Heat transfer is steady since there is no indication of change with time.

2 Heat transfer through the wall is one-dimensional.

3 Thermal conductivities are constant.

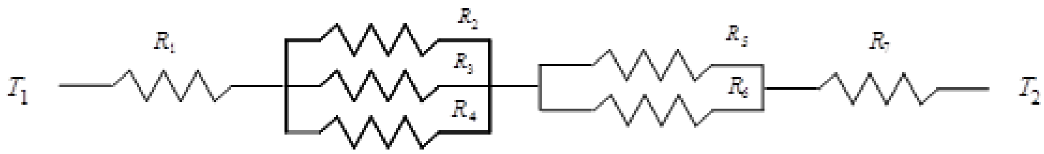
4 Thermal contact resistances at the interfaces are disregarded.

Properties

The thermal conductivities are given to be $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, $k_E = 35$ W/m·°C.

Analysis

(*a*) The representative surface area is 1m. The thermal resistance network and the individual



Then steady rate of heat transfer through entire wall becomes

$$R_1 = R_A = \left(\frac{L}{kA}\right)_A = \frac{0.01\text{ m}}{(2\text{ W/m.°C})(0.12\text{ m}^2)} = 0.04\text{ °C/W}$$

$$R_2 = R_4 = R_C = \left(\frac{L}{kA}\right)_C = \frac{0.05\text{ m}}{(20\text{ W/m.°C})(0.04\text{ m}^2)} = 0.06\text{ °C/W}$$

$$R_3 = R_B = \left(\frac{L}{kA}\right)_B = \frac{0.05\text{ m}}{(8\text{ W/m.°C})(0.04\text{ m}^2)} = 0.16\text{ °C/W}$$

$$R_5 = R_D = \left(\frac{L}{kA}\right)_D = \frac{0.1\text{ m}}{(15\text{ W/m.° C})(0.06\text{ m}^2)} = 0.11\text{ °C/W}$$

$$R_6 = R_E = \left(\frac{L}{kA}\right)_E = \frac{0.1\text{ m}}{(35\text{ W/m.°C})(0.06\text{ m}^2)} = 0.05\text{ °C/W}$$

$$R_7 = R_F = \left(\frac{L}{kA}\right)_F = \frac{0.06\text{ m}}{(2\text{ W/m.°C})(0.12\text{ m}^2)} = 0.25\text{ °C/W}$$

$$\frac{1}{R_{mid,1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06} \longrightarrow R_{mid,1} = 0.025\ °\text{C/W}$$

$$\frac{1}{R_{mid,2}} = \frac{1}{R_5} + \frac{1}{R_5} = \frac{1}{0.11} + \frac{1}{0.05} \longrightarrow R_{mid,2} = 0.034\ °\text{C/W}$$

$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 = 0.04 + 0.025 + 0.034 + 0.25 = 0.349\ °\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(300-100)°\text{C}}{0.349\ °\text{C/W}} = 572\ \text{W}\ \ (\text{for a } 0.12\ \text{m} \times 1\ \text{m section})$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (572\,\text{W})\frac{(5\,\text{m})(8\,\text{m})}{0.12\,\text{m}^2} = \mathbf{1.91 \times 10^5\ W}$$

2.3.2 Cylinders and Spheres

Consider a long cylindrical layer (such as a circular pipe) of inner radius r_1, outer radius r_2, length L, and average thermal conductivity k (Fig. 2.14).

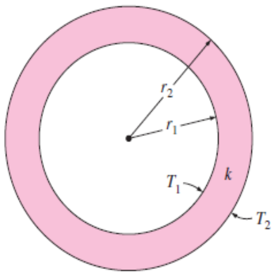


Figure 2-14 A long cylindrical pipe(or spherical shell) with specified inner and outer surface temperature T_1 and T_2

The two surfaces of the cylindrical layer are maintained at constant temperatures T_1 and T_2. There is no heat generation in the layer and the thermal conductivity is constant. For one-dimensional heat conduction through the cylindrical layer, we have T(r). Then Fourier's law of heat conduction for heat transfer through the cylindrical layer can be expressed as

$$\dot{Q}_{cond,cyl} = -kA\frac{dT}{dr} \qquad \text{(W)}$$

where $A = 2\pi rL$ is the heat transfer area at location *r.* Note that *A* depends on *r*, and thus it *varies* in the direction of heat transfer. Separating the variables in the above equation and substituting A = 2πrL and performing integrations give

$$\dot{Q}_{cond,cyl} = 2\pi Lk\frac{T_1 - T_2}{\ln(r_2/r_1)} \qquad \text{(W)}$$

Since $\dot{Q}_{cond,cyl} = constant$. This equation can be rearranged as

$$\dot{Q}_{cond,cyl} = \frac{T_1 - T_2}{R_{cyl}} \qquad \text{(W)}$$

Where

$$R_{cyl} = \frac{\ln({}^{r_2}/{r_1})}{2\pi Lk} = \frac{\ln(\frac{Outer\ radius}{inner\ radius})}{2\pi\ x\ Length\ x\ Thermal\ Conductivity}$$

is the *thermal resistance* of the cylindrical layer against heat conduction, or simply the conduction resistance of the cylinder layer.

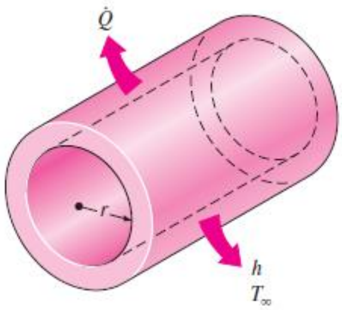


Figure 2.15 Heat is lost from a hot-water pipe to the air outside in the radial direction, and thus heat transfer from a long pipe is one-dimensional.

We can repeat the analysis above for a spherical layer by taking *A* = 4π*r*^{2} and the equation can be expressed as

$$\dot{Q}_{cond,sph} = 2\pi Lk\frac{T_1 - T_2}{R_{sph}}$$

Where

$$R_{sph} = \frac{r_2 - r_1}{4\pi r_1 r_2 k} = \frac{Outer\ radius - Inner\ radius}{4\pi(Outer\ radius)(Inner\ radius)(Thermal\ Conductivity)}$$

is the thermal resistance of the spherical layer against heat conduction, or simply the conduction resistance of the spherical layer.

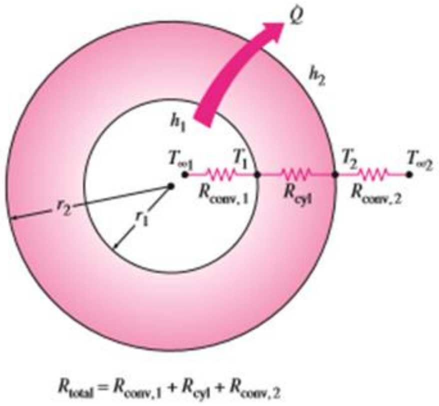


Figure 2.16 The thermal resistance network for cylinders (or spherical) shell subjected to convection from both the inner and the outer sides.

Now consider steady one-dimensional heat flow through a cylindrical or spherical layer that is exposed to convection on both sides to fluids at temperatures $T_{\infty 1}$ and $T_{\infty 1}$ with heat transfer coefficients h_1 and h_2, respectively, as shown in Fig. 2.16. The thermal resistance network in this case consists of one conduction and two convection resistances in series, just like the one for the plane wall, and the rate of heat transfer under steady conditions can be expressed as

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

Where for a cylindrical layer

$$R_{total} = R_{conv,1} + R_{cyl} + R_{conv,2}$$

$$= \frac{1}{(2\pi r_1)h_1} + \frac{\ln({}^{r_2}/_{r_1})}{2\pi L k} + \frac{1}{(2\pi r_2 L)h_2}$$

And for a spherical layer

$$R_{total} = R_{conv,1} + R_{sph} + R_{conv,2}$$

$$= \frac{1}{(4\pi r_1^2)h_1} + \frac{r_2 - r_1}{4\pi r_1 r_2 k} + \frac{1}{(4\pi r_2^2)h_2}$$

Note that A in the convection resistance relation $R_{conv} = 1/hA$ is the *surface area at which convection occurs.* It is equal to $A = 2\pi rL$ for a cylindrical surface and $A = 4\pi r^2$ for a spherical surface of radius *r.* Also note that the thermal resistances are in series, and thus the total thermal resistance is determined by simply adding the individual resistances, just like the electrical resistances connected in series.

EXAMPLE 2.6 Heat Loss Through an Insulated Steam Pipe

Steam at $T_{\infty 1}$ = 320°C flows in a cast iron pipe (k = 80 W/m · °C) whose inner and outer diameters are D_1 = 5 cm and D_2 = 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 $T_{\infty 1}$ = 5°C by natural convection and radiation, with a combined heat transfer coefficient of h_2 = 18 W/m^2 · °C. Taking the heat transfer coefficient inside the pipe to be h_1 = 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.

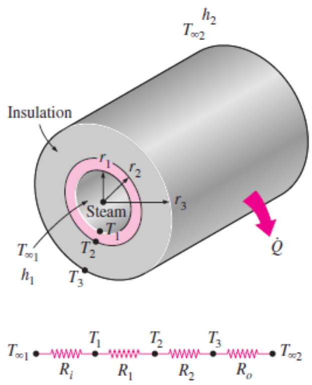


Figure 2.17 Heat Loss Through an Insulated Steam Pipe

SOLUTION

A steam pipe covered with glass wool insulation is subjected to convection on its surfaces. The rate of heat transfers per unit length and the temperature drops across the pipe and the insulation are to be determined.

Assumptions

1 Heat transfer is steady since there is no indication of any change with time.

2 Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction.

3 Thermal conductivities are constant.

4 The thermal contact resistance at the interface is negligible.

Properties

The thermal conductivities are given to be k = 80 W/m · °C for cast iron and k = 0.05 W/m · °C for glass wool insulation.

Analysis The thermal resistance network for this problem involves four resistances in series. Taking L = 1 m, the areas of the surfaces exposed to convection are determined 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

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

The heat loss for a given pipe length can be determined by multiplying the above quantity by the pipe length L. The temperature drops across the pipe and the insulation are determined to be

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

That is, the temperatures between the inner and the outer surfaces of the pipe differ by 0.02°C, whereas the temperatures between the inner and the outer surfaces of the insulation differ by 284°C.

EXERCISES/TUTORIAL

1. A storage chamber of interior dimensions 10m x 8m x 2.5m high has its inside maintained at the temperature of -20oC whilst the outside is at 25oC. The walls and ceiling of the chamber have three layers made of :

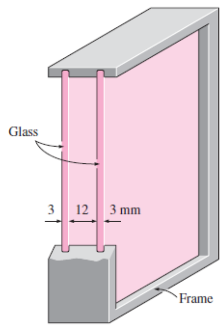
    60 mm thick board (k = 0.2 W/m°C) on the inside

    90mm thick insulation (k = 0.04 W/m°C) at mid

    240 mm thick concrete (k = 1.8 W/m°C) on the outside

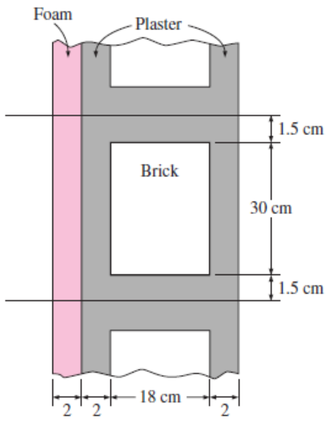
    Neglecting flow of heat through the floor, determine the rate at which heat can flow towards inside the chamber

2. Consider a 1.2-m-high and 2-m-wide double-pane window consisting of two 3-mm-thick layers of glass (k = 0.78 W/m · °C) separated by a 12-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 24°C while the temperature of the outdoors is -5°C. Take the convection heat transfer coefficients on the inner and outer surfaces of the window to be h_1 = 10 W/m^2 · °C and h_2 =25 W/m^2 · °C, and disregard any heat transfer by radiation.



3. A 4-m-high and 6-m-wide wall consists of a long 18-cm 30-cm cross section of horizontal bricks (k = 0.72 W/m · °C) separated by 3-cm-thick plaster layers (k = 0.22 W/m · °C). There are also 2-cm-thick plaster layers on each side of the wall, and a 2-cm-thick rigid foam (k = 0.026 W/m · °C) on the inner side of the wall. The indoor and the outdoor temperatures are 22°C and -4°C, and the convection heat transfer

coefficients on the inner and the outer sides are $h_1 = 10\ W/m^2 \cdot °C$ and $h_2 = 20\ W/m^2 \cdot °C$, respectively. Assuming one-dimensional heat transfer and disregarding radiation, determine the rate of heat transfer through the wall.



4. Steam at 320°C flows in a stainless steel pipe ($k = 15\ W/m \cdot °C$) whose inner and outer diameters are 5 cm and 5.5 cm, respectively. The pipe is covered with 3-cm-thick glass wool insulation ($k = 0.038\ W/m \cdot °C$). Heat is lost to the surroundings at 5°C by natural convection and radiation, with a combined natural convection and radiation heat transfer coefficient of $15\ W/m^2 \cdot °C$. Taking the heat transfer coefficient inside the pipe to be $80\ W/m^2 \cdot °C$, 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.

CHAPTER 3

PRINCIPLES OF STEADY STATE HEAT TRANSFER IN CONVECTION

At the end of this chapter, you should be able to:

- Determine the fundamentals of convection.
- Apply the external forced convection.

3.1 FUNDAMENTAL OF CONVECTION

3.1.1 Physical mechanisms of convection and its classification.

Conduction and convection are similar in that both mechanisms require the presence of a material medium but they are different in that convection requires the presence of a fluid motion. Heat transfer through a solid is always by conduction, since the molecules of a solid remain at relative positions. Heat transfer through a liquid or gas, however, can be conduction or convection in the presence of any bulk fluid motion. Heat transfer through a fluid is by convection in the presence of bulk fluid motion and by conduction in the absence of it. Therefore, conduction in a fluid can be viewed as the limiting case of convection, corresponding to the case of quiescent fluid in Figure 3.1

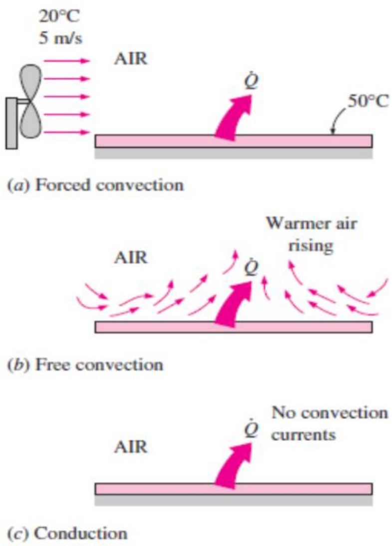


Figure 3.1 Heat transfer from a hot surface to the surrounding fluid by convection and conduction

The fluid motion enhances heat transfer, since it brings hotter and cooler chunks of fluid into contact, initiating higher rates of conduction at a greater number of sites in a fluid. Therefore, the rate of heat transfer through a fluid is much higher by convection than it is by conduction. In fact, the higher the fluid velocity, the higher the rates of heat transfer.

Consider the cooling of a hot iron block with a fan blowing air over its top surface, as shown in Figure 3.2

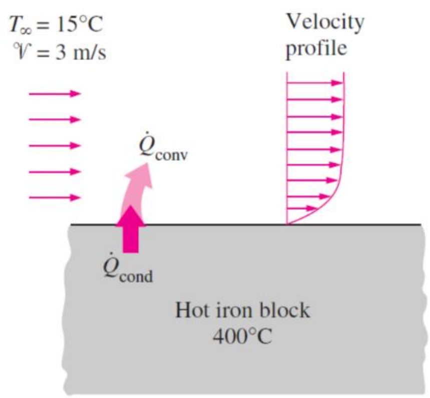


Figure 3.2 The cooling of a hot block by force convection

We know that heat will be transferred from the hot block to the surrounding cooler air, and the block will eventually cool. We also know that the block will cool faster if the fan is switched to a higher speed. Replacing air by water will enhance the convection heat transfer even more. Experience shows that convection heat transfer strongly depends on the fluid properties dynamic viscosity μ, thermal conductivity k, density ρ, and specific heat C_p, as well as the fluid velocity V. It also depends on the geometry and the roughness of the solid surface, in addition to the type of fluid flow (such as being streamlined or turbulent). Thus, we expect the convection heat transfer relations to be rather complex because of the dependence of convection on so many variables. This is not surprising, since convection is the most complex mechanism of heat transfer. The rate of convection heat transfer is observed to be proportional to the temperature difference and is conveniently expressed by Newton's law of cooling as

$$\dot{q}_{conv} = h(T_s - T_\infty) \quad \text{(W/m2)}$$

Or

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) \text{ (W)}$$

where

h = convection heat transfer coefficient, $W/m^2 \cdot °C$

A_s = heat transfer surface area, m^2

T_s = temperature of the surface, °C

T_∞ = temperature of the fluid sufficiently far from the surface, °C

Judging from its units, the convection heat transfer coefficient h can be defined as the rate of heat transfer between a solid surface and a fluid per unit surface area per unit temperature difference.

Classification of Fluid Flows

Convection heat transfer is closely tied with fluid mechanics, which is the science that deals with the behavior of fluids at rest or in motion, and the interaction of fluids with solids or other fluids at the boundaries. There are many ways to classify the fluid flow problems, and below we present some general categories.

A. Viscous Versus Inviscid Regions of 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, which is a measure of internal stickiness of the fluid. 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 greatly simplifies the analysis without much loss in accuracy. Such idealized flows of zero-viscosity fluids are called frictionless or **inviscid flows**.

B. Internal versus External Flow

The flow of an unbounded fluid over a surface such as a plate, a wire, or a pipe is external flow. The flow in a pipe or duct is internal flow if the fluid is completely bounded by solid surfaces. Water flow in a pipe, for example, is internal flow, and air flow over an exposed pipe during a windy day is external flow (Fig. 3.3).

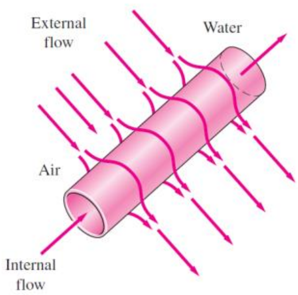


Figure 3.3 Internal flow of water in a pipe and the external flow of air over the same pipe.

C. Compressible versus Incompressible Flow

A flow is classified as being compressible or incompressible, depending on the density variation of the fluid during flow. Incompressibility is an approximation, and a flow is said to be incompressible if the density remains nearly constant throughout. Therefore, liquids are usually classified as incompressible substances. Gases, on the other hand, are highly compressible. However, gas flows can be treated as incompressible if the density changes are under about 5 percent, which is usually the case when the flow velocity is less than 30 percent of the velocity of sound in that gas (i.e., the Mach number of flow is less than 0.3). The velocity of sound in air at room temperature is 346 m/s. Therefore, the compressibility effects of air can be neglected at speeds under 100 m/s. Note that the flow of a gas is not necessarily a compressible flow.

D. Laminar versus Turbulent Flow

Some flows are smooth and orderly while others are rather chaotic. The highly ordered fluid motion characterized by smooth layer is called **laminar.** The flow of high-viscosity fluids such as oils at low velocities is typically laminar. The highly disordered fluid motion that typically occurs at high velocities characterized by velocity fluctuations is called **turbulent**. The flow of low-viscosity fluids such as air at high velocities is typically turbulent. The flow regime greatly influences the heat transfer rates and the required power for pumping. A flow that alternates between laminar and turbulent is called transitional.

E. Natural (or Unforced) versus Forced Flow

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 such as the buoyancy effect, which manifests itself as the rise of the warmer (and thus lighter) fluid and the fall of cooler (and thus denser) fluid.

F. Steady versus Unsteady Flow

The terms steady and uniform are used frequently in engineering, and thus it is important to have a clear understanding of their meanings. The term steady implies no change with time. The opposite of steady is unsteady, or transient. The term uniform, however, implies no change with location over a specified region.

Many devices such as turbines, compressors, boilers, condensers, and heat exchangers operate for long periods of time under the same conditions, and they are classified as steady-flow devices. During steady flow, the fluid properties can change from point to point within a device, but at any fixed point they remain constant.

G. 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 *V* varies in one, two, or three primary dimensions, respectively. A typical fluid flow involves a three-dimensional geometry and the velocity may vary in all three dimensions rendering the flow three-dimensional [V(x, y, z) in rectangular or V(r, ϴ, z) in cylindrical coordinates]. However, the variation of velocity in certain direction can be small relative to the variation in other directions, and can be ignored with negligible error. In such cases, the flow can be modeled conveniently as being one- or two-dimensional, which is easier to analyze.

3.1.2 The development of velocity and thermal boundary layers during flow over surfaces.

Velocity Boundary Layer

Consider the parallel flow of a fluid over a *flat plate,* as shown in Fig. 3.4. Surfaces that are slightly contoured such as turbine blades can also be approximated as flat plates with reasonable accuracy. The *x*-coordinate is measured along the plate surface from the *leading edge* of the plate in the direction of the flow, and *y* is measured from the surface in the normal direction. The fluid approaches the plate in the *x*-direction with a uniform upstream velocity of V, which is practically identical to the free-stream velocity u_∞ over the plate away from the surface (this would not be the case for cross flow over blunt bodies such as a cylinder).

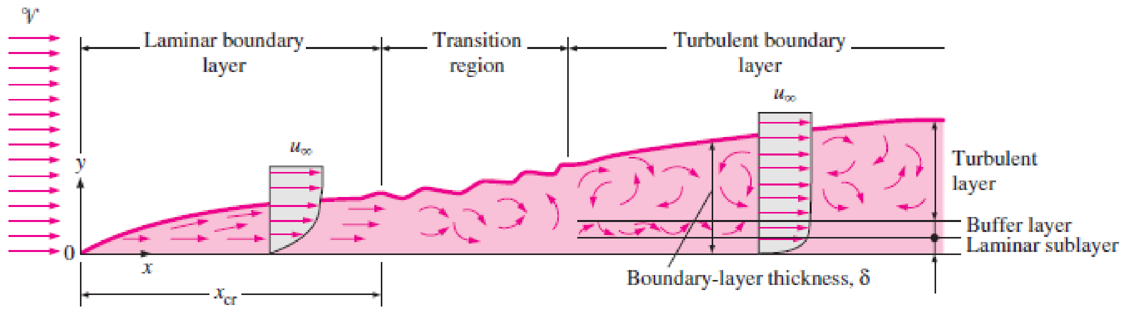


Figure 3.4 The development of the boundary layer for flow over a flat plate, and the different flow regimes

We can consider the fluid to consist of adjacent layers piled on top of each other. The velocity of the particles in the first fluid layer adjacent to the plate becomes zero because of the no-slip condition. This motionless layer slows down the particles of the neighboring fluid layer as a result of friction between the particles of these two adjoining fluid layers at different velocities. This fluid layer then slows down the molecules of the next layer, and so on. Thus, the presence of the plate is felt up to some normal distance δ from the plate beyond which the free-stream velocity u_∞ remains essentially unchanged. As a result, the *x*-component of the fluid velocity, *u,* will vary from 0 at $y = 0$ to nearly u_∞ at $y = \delta$ (Fig. 3.5)

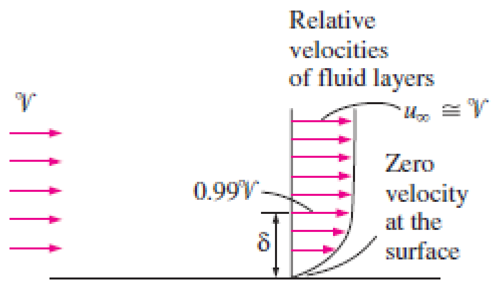


Figure 3.5 The development of a boundary layer on a surface is due to no-slip condition

The region of the flow above the plate bounded by δ in which the effects of the viscous shearing forces caused by fluid viscosity are felt is called the **velocity boundary layer.** The *boundary layer thickness, δ*, is typically defined as the distance *y* from the surface at which $u = 0.99u_\infty$.

The hypothetical line of $u = 0.99u_\infty$ divides the flow over a plate into two regions: the **boundary layer region,** in which the viscous effects and the velocity changes are significant, and the **inviscid flow region,** in which the frictional effects are negligible and the velocity remains essentially constant.

Thermal Boundary Layer

We have seen that a velocity boundary layer develops when a fluid flows over a surface as a result of the fluid layer adjacent to the surface assuming the surface velocity (i.e., zero velocity relative to the surface). Also, we defined the velocity boundary layer as the region in which the fluid velocity varies from zero to $0.99u_\infty$. Likewise, a *thermal boundary layer* develops when a fluid at a specified temperature flows over a surface that is at a different temperature, as shown in Fig. 3.6

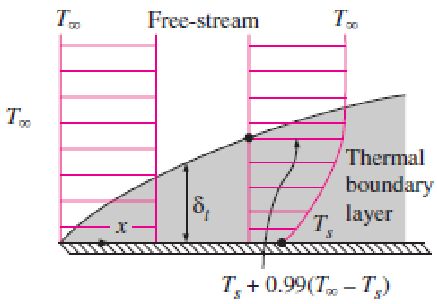


Figure 3.6 Thermal boundary layer on a flat plate (the fluid is hotter than the plate surface)

Consider the flow of a fluid at a uniform temperature of T_∞ over an isothermal flat plate at temperature T_s. The fluid particles in the layer adjacent to the surface will reach thermal equilibrium with the plate and assume the surface temperature T_s. These fluid particles will then exchange energy with the particles in the adjoining-fluid layer, and so on. As a result, a temperature profile will develop in the flow field that ranges from T_s at the surface to T_∞ sufficiently far from the surface. The flow region over the surface in which the temperature variation in the direction normal to the surface is significant is the **thermal boundary layer.** The *thickness* of the thermal boundary layer δ_t at any location along the surface is defined as *the distance from the surface at which the temperature difference* $T - T_s$ *equals* $0.99(T_\infty - T_s)$. Note that for the special case of $T_s = 0$, we have $T = 0.99T_\infty$ at the outer edge of the thermal boundary layer, which is analogous to $u = 0.99u_\infty$ for the velocity boundary layer.

3.1.3 Difference between Laminar and Turbulent Flows

A careful inspection of flow in a pipe reveals that the fluid flow is streamlined at low velocities but turns chaotic as the velocity is increased above a critical value, as shown in Figure 3.7. The flow regime in the first case is said to be laminar, characterized by smooth streamlines and highly-ordered motion, and turbulent in the second case, where it is characterized by velocity fluctuations and highly-disordered motion. The transition from laminar to turbulent flow does not occur suddenly; rather, it occurs over some region in which the flow fluctuates between laminar and turbulent flows before it becomes fully turbulent.

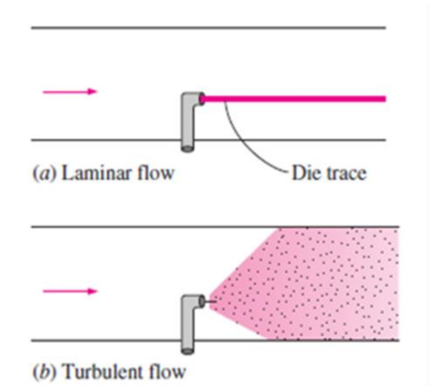


Figure 3.7 The behavior of coloured fluid injected into the flow in laminar and turbulent flows in a pipe

3.1.4 Reynolds, Prandtl and Nusselt numbers.

Reynolds Number

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.This ratio is called the Reynolds number, which is a dimensionless quantity, and is expressed for external flow as

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

where V is the upstream velocity (equivalent to the free-stream velocity for a flat plate),

L_c is the characteristic length of the geometry

$\nu = \mu/\rho$ is the kinematic viscosity of the fluid.

The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number value of the critical Reynolds number is $Re_{cr} = 5 \times 10^5$

Prandatl Number

The relative thickness of the velocity and the thermal boundary layers is best described by the dimensionless parameter Prandtl number, defined as

$$Pr = \frac{molecular\ diffusity\ of\ momentum}{molecular\ diffusity\ of\ heat} = \frac{v}{\alpha} = \frac{\mu c_p}{k}$$

The Prandtl numbers of fluids range from less than 0.01 for liquid metals to more than 100,000 for heavy oils (Table 3.1).

Table 3.1 Typical ranges of Prandtl numbers for common fluids

Fluid	Pr
Liquid metals	0.004–0.030
Gases	0.7–1.0
Water	1.7–13.7
Light organic fluids	5–50
Oils	50–100,000
Glycerin	2000–100,000

Note that the Prandtl number is in the order of 10 for water. The Prandtl numbers of gases are about 1, which indicates that both momentum and heat dissipate through the fluid at about the same rate. Heat diffuses very quickly in liquid metals ($Pr \le 1$) and very slowly in oils ($Pr \ge 1$) relative to momentum. Consequently, the thermal boundary layer is much thicker for liquid metals and much thinner for oils relative to the velocity boundary layer.

Nusselt Number

In convection studies, it is common practice to non-dimensionalize the governing equations and combine the variables, which group together into dimensionless numbers in order to reduce the number of total variables. It is also common practice to non-dimensionalize the heat transfer coefficient h with the Nusselt number, defined as

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

where k is the thermal conductivity of the fluid and L_c is the characteristic length.

Heat transfer through the fluid layer will be by convection when the fluid involves some motion and by conduction when the fluid layer is motionless.

Heat flux (the rate of heat transfer per unit time per unit surface area) in either case will be

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

And

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

Taking their ratio gives

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

Therefore, the Nusselt number represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer. The larger the Nusselt number, the more effective the convection. A Nusselt number of Nu = 1 for a fluid layer represents heat transfer across the layer by pure conduction.

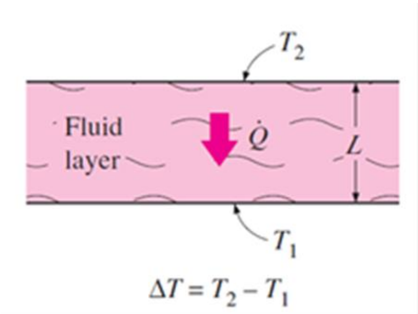


Figure 3.8 Heat transfer through a fluid layer of thickness L and temperature difference ΔT

3.2 EXTERNAL FORCED CONVECTION

Fluid flow over solid bodies frequently occurs in practice, and it is responsible for numerous physical phenomena such as the drag force acting on the automobiles, power lines, trees, and underwater pipelines; the lift developed by airplane wings; upward draft of rain, snow, hail, and dust particles in high winds; and the cooling of metal or plastic sheets, steam and hot water pipes, and extruded wires. Therefore, developing a good understanding of external flow and external forced convection is important in the mechanical and thermal design of many engineering systems such as aircraft, automobiles, buildings, electronic components, and turbine blades.

Friction and Pressure Drag

You may have seen high winds knocking down trees, power lines, and even trailers, and have felt the strong "push" the wind exerts on your body. You experience the same feeling when you extend your arm out of the window of a moving car. The force a flowing fluid exerts on a body in the flow direction is called drag. (Fig 3.9)

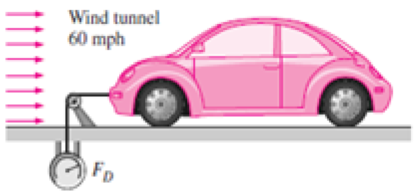


Figure 3.9 Schematic for measuring the drag force acting on a car in a wind tunnel

A stationary fluid exerts only normal pressure forces on the surface of a body immersed in it. A moving fluid, however, also exerts tangential shear forces on the surface because of the no-slip condition caused by viscous effects. Both of these forces, in general, have components in the direction of flow, and thus the drag force is due to the combined effects of pressure and wall shear forces in the flow direction. The components of the pressure and wall shear forces in the normal direction to flow tend to move the body in that direction, and their sum is called lift.

The drag force FD depends on the density ρ of the fluid, the upstream velocity V, and the size, shape, and orientation of the body, among other things. The drag characteristics of a body is represented by the dimensionless drag coefficient C_D defined as

Drag coefficient:
$$C_D = \frac{F_D}{\frac{1}{2}\rho \mathcal{V}^2 A}$$

where A is the frontal area (the area projected on a plane normal to the direction of flow) for blunt bodies—bodies that tends to block the flow. The frontal area of a cylinder of diameter D and length L, for example, is $A = L_D$. For parallel flow over flat plates or thin airfoils, A is the surface area. The drag coefficient is primarily a function of the shape of the body, but it may also depend on the Reynolds number and the surface roughness.

The friction drag is the component of the wall shear force in the direction of flow, and thus it depends on the orientation of the body as well as the magnitude of the wall shear stress τ_w. The friction drag is zero for a surface normal to flow, and maximum for a surface parallel to flow since the friction drag in this case equals the total shear force on the surface. Therefore, for parallel flow over a flat plate, the drag coefficient is equal to the friction drag coefficient, or simply the friction coefficient (Fig. 3.9). That is,

Flat plate: $C_D = C_{D,friction} = C_f$

Once the average friction coefficient C_f is available, the drag (or friction) force over the surface can be determined from Eq. 3-11. In this case A is the surface area of the plate exposed to fluid flow. When both sides of a thin plate are subjected to flow, A becomes the total area of the top and bottom surfaces. Note that the friction coefficient, in general, will vary with location along the surface.

Film Temperature

The fluid temperature in the thermal boundary layer varies from T_s at the surface to about T_∞ at the outer edge of the boundary. The fluid properties also vary with temperature, and thus with position across the boundary layer. In order to account for the variation of the properties with temperature, the fluid properties are usually evaluated at the so-called film temperature, defined as

$$T_f = \frac{T_s + T_\infty}{2}$$

which is the arithmetic average of the surface and the free-stream temperatures. The fluid properties are then assumed to remain constant at those values during the entire flow.

Parallel Flow Over Flat Plate

Consider the parallel flow of a fluid over a flat plate of length L in the flow direction, as shown in Figure 3.10 The x-coordinate is measured along the plate surface from the leading edge in the direction of the flow. The fluid approaches the plate in the x-direction with uniform upstream velocity V and temperature T_∞. The flow in the velocity boundary layer starts out as laminar, but if the plate is sufficiently long, the flow will become turbulent at a distance x_{cr} from the leading edge where the Reynolds number reaches its critical value for transition.

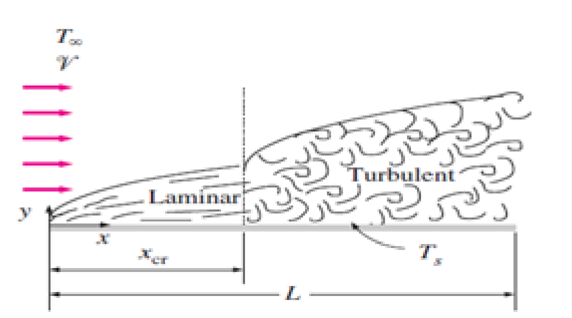


Figure 3.10 Laminar and turbulent regions of the boundary layer during flow over a flat plate

The transition from laminar to turbulent flow depends on the surface geometry, surface roughness, upstream velocity, surface temperature, and the type of fluid, among other things, and is best characterized by the Reynolds number. The Reynolds number at a distance x from the leading edge of a flat plate is expressed as

$$\mathrm{Re}_x = \frac{\rho \mathcal{V} x}{\mu} = \frac{\mathcal{V} x}{\nu}$$

Note that the value of the Reynolds number varies for a flat plate along the flow, reaching Re_L = VL /v at the end of the plate.

For flow over a flat plate, transition from laminar to turbulent is usually taken to occur at the critical Reynolds number of

$$\mathrm{Re}_{cr} = \frac{\rho \mathcal{V} x_{cr}}{\mu} = 5 \times 10^5$$

The actual value of the critical Reynolds number for a flat plate may vary from 10^5 to 3 x 10^6, depending on the surface roughness and the turbulence level, and the variation of pressure along the surface.

Friction Coefficient

The average friction coefficient over the entire plate is determined by

$$Laminar: \qquad C_f = \frac{1.33}{Re_L^{1/2}} \qquad Re_L < 5 \times 10^5$$

$$Turbulent: \qquad C_f = \frac{0.074}{Re_L^{1/5}} \qquad 5 \times 10^5 \leq Re_L \leq 10^7$$

The first relation gives the average friction coefficient for the entire plate when the flow is laminar over the entire plate. The second relation gives the average friction coefficient for the entire plate only when the flow is turbulent over the entire plate, or when the laminar flow region of the plate is too small relative to the turbulent flow region

In some cases, a flat plate is sufficiently long for the flow to become turbulent, but not long enough to disregard the laminar flow region. Note that we included the transition region with the turbulent region. Again taking the critical Reynolds number to be $Re_{cr} = 5 \times 10^5$ and the average friction coefficient over the entire plate is determined to be

$$C_f = \frac{0.074}{Re_L^{1/5}} - \frac{1742}{Re_L} \qquad 5 \times 10^5 \leq Re_L \leq 10^7$$

Heat Transfer Coefficient/ Average Nusselt Number

The average Nusselt number over the entire plate is

Laminar: $\quad Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} \quad$ $Re_L < 5 \times 10^5$

Turbulent: $\quad Nu = \frac{hL}{k} = 0.037 Re_L^{0.8} Pr^{1/3} \quad$ $0.6 \leq Pr \leq 60$

$5 \times 10^5 \leq Re_L \leq 10^7$

The first relation gives the average heat transfer coefficient for the entire plate when the flow is laminar over the entire plate. The second relation gives the average heat transfer coefficient for the entire plate only when the flow is turbulent over the entire plate, or when the laminar flow region of the plate is too small relative to the turbulent flow region.

In some cases, a flat plate is sufficiently long for the flow to become turbulent, but not long enough to disregard the laminar flow region so the average Nusselt number over the entire plate is determined to be

$$\mathrm{Nu} = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\mathrm{Pr}^{1/3} \qquad \begin{array}{l} 0.6 \le \mathrm{Pr} \le 60 \\ 5 \times 10^5 \le \mathrm{Re}_L \le 10^7 \end{array}$$

The constants in this relation will be different for different critical Reynolds numbers. Liquid metals such as mercury have high thermal conductivities, and are commonly used in applications that require high heat transfer rates. However, they have very small Prandtl numbers, and thus the thermal boundary layer develops much faster than the velocity boundary layer. It is desirable to have a single correlation that applies to all fluids, including liquid metals.

3.2.2 Calculation of the heat transfer associated with flow over a flat plate for laminar flow.

EXAMPLE 3.1 **Flow of Hot Oil Over a Flat Plate**

Engine oil at 60°C flows over the upper surface of a 5-m-long flat plate whose temperature is 20°C with a velocity of 2 m/s. Determine the total drag force and the rate of heat transfer per unit width of the entire plate.

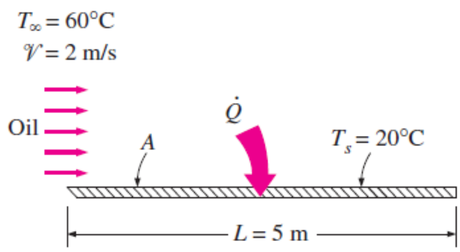


Figure 3.11 Hot oil over a flat plate

SOLUTION

Engine oil flows over a flat plate. The total drag force and the rate of heat transfer per unit width of the plate are to be determined.

Assumptions

1 The flow is steady and incompressible.

2 The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$.

Properties

The properties of engine oil at the film temperature of $T_f = (T_s + T_\infty)/2 = (20 + 60)/2 = 40°C$ are (Table A–13).

P = 876 kg/m3 Pr = 2870

k = 0.144 W/m .°C v = 242 x 10^{-6} m^2/s

Analysis

Noting that L = 5m, the Reynolds Number at the end of the plate is

$$Re_L = \frac{VL}{v} = \frac{(2m/s)(5m)}{2.485 \times 10^{-4}\ m^2/s} = 4.024 \times 10^4$$

Which is less than the critical Reynolds number. Thus we have laminar flow over the entire plate, the average friction coefficient is

$$C_f = 1.33 Re_L^{-0.5} = 1.33 \times (4.024 \times 10^4)^{-0.5} = 0.00663$$

Noting that the pressure drag is zero and thus $C_D = C_f$ for parallel flow over a flat plate, the drag force acting on the plate per unit width becomes

$$F_D = C_f A \frac{\rho V^2}{2} = 0.00663(5m \times 1m) \frac{\left(\frac{876kg}{m^3}\right)\left(\frac{2m}{s}\right)^2}{2} \left(\frac{1N}{1kg.\frac{m}{s^2}}\right) = 58.1N$$

Similarly, the Nusselt number is determined using the laminar flow relations for a flat plate,

$$Nu = \frac{hL}{k} = 0.664 Re_L^{0.5} Pr^{1/3} = 0.664 \times (4.024 \times 10^4)^{0.5} \times 2962^{1/3} = 1913$$

Then,

$$h = \frac{k}{L}\text{Nu} = \frac{0.144\ \text{W/m}\cdot{}^\circ\text{C}}{5\ \text{m}}(1918) = 55.2\ \text{W/m}^2\cdot{}^\circ\text{C}$$

And

$$\dot{Q} = hA_s(T_\infty - T_s) = (55.2\ \text{W/m}^2\cdot{}^\circ\text{C})(5 \times 1\ \text{m}^2)(60 - 20)^\circ\text{C} = \mathbf{11{,}040\ W}$$

EXAMPLE 3.2 **Cooling of a Hot Block by Forced Air at High Elevation**

The local atmospheric pressure in Denver, Colorado (elevation 1610 m), is 83.4 kPa. Air at this pressure and 20°C flows with a velocity of 8 m/s over a 1.5 m x 6m flat plate whose temperature is 140°C. Determine the rate of heat transfer from the plate if the air flows parallel to the

(a) 6-m-long side and

(b) the 1.5-m side.

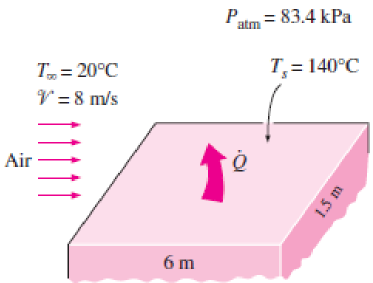


Figure 3.12 Hot Block

SOLUTION

The top surface of a hot block is to be cooled by forced air. The rate of heat transfer is to be determined for two cases.

Assumptions

1 Steady operating conditions exist.

2 The critical Reynolds number is $Re_{cr} = 5 \times 10^5$.

3 Radiation effects are negligible.

4 Air is an ideal gas.

Properties

The properties k, μ, C_p, and Pr of ideal gases are independent of pressure, while the properties ν and α are inversely proportional to density and thus pressure. The properties of air at the film temperature of $T_f = (T_s + T_\infty)/2 = (140 + 20)/2 = 80°C$ and 1 atm pressure are (Table A–15)

$k = 0.02953$ W/m · °C   Pr =0.7154

$\nu_{@\ 1atm} = 2.097 \times 10^{-5}\ m^2/s$

The atmospheric pressure in Denver is P = (83.4 kPa) / (101.325 kPa/atm) = 0.823 atm.

Then the kinematic viscosity of air in Denver becomes

$\nu = \nu_{@\ 1\ atm} / P = (2.097 \times 10^{-5}\ m^2/s)/0.823 = 2.548 \times 10^{-5}\ m^2/s$

(Refer to the notes from Table A-15)

Analysis

(a) When air flow is parallel to the long side, we have L = 6 m, and the Reynolds number at the end of the plate becomes

$$\text{Re}_L = \frac{\mathcal{V}L}{\nu} = \frac{(8\text{ m/s})(6\text{ m})}{2.548 \times 10^{-5}\text{ m}^2\text{/s}} = 1.884 \times 10^6$$

which is greater than the critical Reynolds number. Thus, we have combined laminar and turbulent flow, and the average Nusselt number for the entire plate is determined to be

$$\begin{aligned}\text{Nu} &= \frac{hL}{k} = (0.037\ \text{Re}_L^{0.8} - 871)\text{Pr}^{1/3} \\ &= [0.037(1.884 \times 10^6)^{0.8} - 871]0.7154^{1/3} \\ &= 2687\end{aligned}$$

Then

$$\begin{aligned}h &= \frac{k}{L}\text{Nu} = \frac{0.02953\text{ W/m}\cdot{}^\circ\text{C}}{6\text{ m}}(2687) = 13.2\text{ W/m}^2\cdot{}^\circ\text{C} \\ A_s &= wL = (1.5\text{ m})(6\text{ m}) = 9\text{ m}^2\end{aligned}$$

And

$$\dot{Q} = hA_s(T_s - T_\infty) = (13.2\text{ W/m}^2\cdot{}^\circ\text{C})(9\text{ m}^2)(140 - 20)^\circ\text{C} = 1.43 \times 10^4\text{ W}$$

b) When air flow is along the short side, we have L = 1.5 m, and the Reynolds number at the end of the plate becomes

$$\mathrm{Nu} = \frac{hL}{k} = 0.664\ \mathrm{Re}_L^{0.5}\ \mathrm{Pr}^{1/3} = 0.664 \times (4.71 \times 10^5)^{0.5} \times 0.7154^{1/3} = 408$$

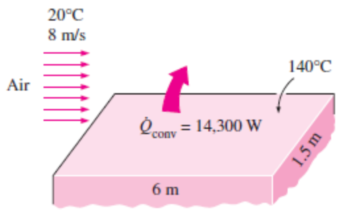
Then

$$h = \frac{k}{L}\mathrm{Nu} = \frac{0.02953\ \mathrm{W/m \cdot {}^\circ C}}{1.5\ \mathrm{m}}(408) = 8.03\ \mathrm{W/m^2 \cdot {}^\circ C}$$

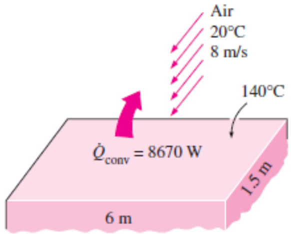
And

$$\dot{Q} = hA_s(T_s - T_\infty) = (8.03\ \mathrm{W/m^2 \cdot {}^\circ C})(9\ \mathrm{m^2})(140 - 20)^\circ\mathrm{C} = 8670\ \mathrm{W}$$

which is considerably less than the heat transfer rate determined in case (a).



(a) Flow along the long side



(b) Flow along the short side

EXERCISES/TUTORIAL

1. Engine oil at 80°C flows over a 6-m-long flat plate whose temperature is 30°C with a velocity of 3 m/s. Determine the total drag force and the rate of heat transfer over the entire plate per unit width.

2. The local atmospheric pressure in Denver, Colorado (elevation 1610 m), is 101.325 kPa. Air at this pressure and at 30°C flows with a velocity of 6 m/s over a 2.5-m X 8-m flat plate whose temperature is 120°C. Determine the rate of heat transfer from the plate if the air flows parallel to the (a) 8-m-long side and (b) the 2.5-m side.

3. During a cold winter day, wind at 55 km/h is blowing parallel to a 4-m-high and 10-m-long wall of a house. If the air outside is at 5°C and the surface temperature of the wall is 12°C, determine the r ate of heat loss from that wall by convection.

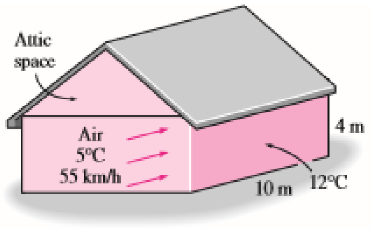


FIGURE P7–16

4. A long 8-cm-diameter steam pipe whose external surface temperature is 90°C passes through some open area that is not protected against the winds. Determine the rate of heat loss from the pipe per unit of its length when the airis at 1 atm pressure and 10°C and the wind is blowing across the pipe at a velocity of 50 km/h.

REFERENCE

Yunus A. Cengel & Afshin J. Ghajar(2015), Heat and Mass Transfer: Fundamentals & Application (Fifth Edition), McGraw Hill Education.

Bergman, Theodore L, Lavine, Adrienne S., Incropera, Frank P. Dewitt (2017), Incropera's Princiles of Heat and Mass Transfer (8th Edition), John Willey & Sons.)

D K Dixit (2016), Heat And Mass Transfer, McGraw Hill Education.

Karwa, Rejandra (2018), Heat And Mass Transfer, Springer Verlag (US)

Forthcoming (2018), Brig's Handbook of Methods & Research in Engineering Heat Transfer, Brig.

Dr. D.S Kumar (2015), Heat & Mass Transfer (9th Edition), S.K Kataria & Sons

APPENDIX

TABLE A–9

Properties of air at 1 atm pressure

Temp. T, °C	Density ρ, kg/m^3	Specific Heat c_p J/kg·K	Thermal Conductivity k, W/m·K	Thermal Diffusivity α, m^2/s	Dynamic Viscosity μ, kg/m·s	Kinematic Viscosity ν, m^2/s	Prandtl Number Pr
−150	2.866	983	0.01171	4.158×10^{-6}	8.636×10^{-6}	3.013×10^{-6}	0.7246
−100	2.038	966	0.01582	8.036×10^{-6}	1.189×10^{-6}	5.837×10^{-6}	0.7263
−50	1.582	999	0.01979	1.252×10^{-5}	1.474×10^{-5}	9.319×10^{-6}	0.7440
−40	1.514	1002	0.02057	1.356×10^{-5}	1.527×10^{-5}	1.008×10^{-5}	0.7436
−30	1.451	1004	0.02134	1.465×10^{-5}	1.579×10^{-5}	1.087×10^{-5}	0.7425
−20	1.394	1005	0.02211	1.578×10^{-5}	1.630×10^{-5}	1.169×10^{-5}	0.7408
−10	1.341	1006	0.02288	1.696×10^{-5}	1.680×10^{-5}	1.252×10^{-5}	0.7387
0	1.292	1006	0.02364	1.818×10^{-5}	1.729×10^{-5}	1.338×10^{-5}	0.7362
5	1.269	1006	0.02401	1.880×10^{-5}	1.754×10^{-5}	1.382×10^{-5}	0.7350
10	1.246	1006	0.02439	1.944×10^{-5}	1.778×10^{-5}	1.426×10^{-5}	0.7336
15	1.225	1007	0.02476	2.009×10^{-5}	1.802×10^{-5}	1.470×10^{-5}	0.7323
20	1.204	1007	0.02514	2.074×10^{-5}	1.825×10^{-5}	1.516×10^{-5}	0.7309
25	1.184	1007	0.02551	2.141×10^{-5}	1.849×10^{-5}	1.562×10^{-5}	0.7296
30	1.164	1007	0.02588	2.208×10^{-5}	1.872×10^{-5}	1.608×10^{-5}	0.7282
35	1.145	1007	0.02625	2.277×10^{-5}	1.895×10^{-5}	1.655×10^{-5}	0.7268
40	1.127	1007	0.02662	2.346×10^{-5}	1.918×10^{-5}	1.702×10^{-5}	0.7255
45	1.109	1007	0.02699	2.416×10^{-5}	1.941×10^{-5}	1.750×10^{-5}	0.7241
50	1.092	1007	0.02735	2.487×10^{-5}	1.963×10^{-5}	1.798×10^{-5}	0.7228
60	1.059	1007	0.02808	2.632×10^{-5}	2.008×10^{-5}	1.896×10^{-5}	0.7202
70	1.028	1007	0.02881	2.780×10^{-5}	2.052×10^{-5}	1.995×10^{-5}	0.7177
80	0.9994	1008	0.02953	2.931×10^{-5}	2.096×10^{-5}	2.097×10^{-5}	0.7154
90	0.9718	1008	0.03024	3.086×10^{-5}	2.139×10^{-5}	2.201×10^{-5}	0.7132
100	0.9458	1009	0.03095	3.243×10^{-5}	2.181×10^{-5}	2.306×10^{-5}	0.7111
120	0.8977	1011	0.03235	3.565×10^{-5}	2.264×10^{-5}	2.522×10^{-5}	0.7073
140	0.8542	1013	0.03374	3.898×10^{-5}	2.345×10^{-5}	2.745×10^{-5}	0.7041
160	0.8148	1016	0.03511	4.241×10^{-5}	2.420×10^{-5}	2.975×10^{-5}	0.7014
180	0.7788	1019	0.03646	4.593×10^{-5}	2.504×10^{-5}	3.212×10^{-5}	0.6992
200	0.7459	1023	0.03779	4.954×10^{-5}	2.577×10^{-5}	3.455×10^{-5}	0.6974
250	0.6746	1033	0.04104	5.890×10^{-5}	2.760×10^{-5}	4.091×10^{-5}	0.6946
300	0.6158	1044	0.04418	6.871×10^{-5}	2.934×10^{-5}	4.765×10^{-5}	0.6935
350	0.5664	1056	0.04721	7.892×10^{-5}	3.101×10^{-5}	5.475×10^{-5}	0.6937
400	0.5243	1069	0.05015	8.951×10^{-5}	3.261×10^{-5}	6.219×10^{-5}	0.6948
450	0.4880	1081	0.05298	1.004×10^{-4}	3.415×10^{-5}	6.997×10^{-5}	0.6965
500	0.4565	1093	0.05572	1.117×10^{-4}	3.563×10^{-5}	7.806×10^{-5}	0.6986
600	0.4042	1115	0.06093	1.352×10^{-4}	3.846×10^{-5}	9.515×10^{-5}	0.7037
700	0.3627	1135	0.06581	1.598×10^{-4}	4.111×10^{-5}	1.133×10^{-4}	0.7092
800	0.3289	1153	0.07037	1.855×10^{-4}	4.362×10^{-5}	1.326×10^{-4}	0.7149
900	0.3008	1169	0.07465	2.122×10^{-4}	4.600×10^{-5}	1.529×10^{-4}	0.7206
1000	0.2772	1184	0.07868	2.398×10^{-4}	4.826×10^{-5}	1.741×10^{-4}	0.7260
1500	0.1990	1234	0.09599	3.908×10^{-4}	5.817×10^{-5}	2.922×10^{-4}	0.7478
2000	0.1553	1264	0.11113	5.664×10^{-4}	6.630×10^{-5}	4.270×10^{-4}	0.7539

Note: For ideal gases, the properties c_p, k, μ, and Pr are independent of pressure. The properties ρ, ν, and α at a pressure P (in atm) other than 1 atm are determined by multiplying the values of ρ at the given temperature by P and by dividing ν and α by P.

Source: Data generated from the EES software developed by S. A. Klein and F. L. Alvarado. Original sources: Keenan, Chao, Keyes, Gas Tables, Wiley, 198; and Thermophysical Properties of Matter, Vol. 3: Thermal Conductivity, Y. S. Touloukian, P. E. Liley, S. C. Saxena, Vol. 11: Viscosity, Y. S. Touloukian, S. C. Saxena, and P. Hestermans, IFI/Plenun, NY, 1970, ISBN 0-306067020-8.