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Mechanical Engineering Series

# Engineering Thermodynamics

Third Edition

**P K NAG**

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# Preface to the Third Edition

The response to the second (previous) edition of this book has been overwhelming with its 25th reprint since 1995 exhausted. *Engineering Thermodynamics* is a basic subject so much so that there is hardly any scope for major revision. Simple precise statements of basic underlying principles have always been emphasized. However, in this edition, certain portions have been updated, some new material like chemical exergy and fuel cell have been added, and a good number of new problems have been included, of which some are solved while others are given as exercise. Chapter 19—Gas Compressors is new chapter that has been compiled by taking away the relevant matter furnished in Chapter 10 of the earlier edition and making additions to it.

The Solution Manual providing solutions to the chapter-end exercises given in the book is available for the teachers.

The author hopes that this new edition will continue to remain a popular book on the subject as it has always been since its initial publication in 1981, about a quarter of a century ago. He would appreciate any suggestion towards improvement or detection of errors from the readers.

**P K NAG**

# Preface to the First Edition

The purpose of this book is to provide a mature approach to the basic principles of classical thermodynamics which is one of the few core subjects for the undergraduate students of almost all branches of engineering. The system-surroundings interactions involving work and heat transfer with associated property changes, and the system-control volume approaches of the first law have been emphasized. The second law has been elaborated upon in considerable detail. Except for some physical explanations, a statistical or microscopic analysis of the subject has not been made.

The first eight chapters of the book are devoted to a thorough treatment of the basic principles and concepts of classical thermodynamics. The second law and entropy have been introduced using the concept of heat engine. Chapters 9 and 10 present the properties of substances. Chapter 11 gives the general thermodynamic relationships among properties. A detailed analysis of power and refrigeration cycles is given in Chapters 12 to 14. Chapter 15 deals with air-water vapour mixtures, and reactive systems are analyzed in Chapter 16. To increase the utility of the book, Chapters 17 and 18 dealing with compressible fluid flow and heat transfer respectively, have been added.

Many illustrative examples are solved and many problems are provided in each chapter to aid comprehension and to stimulate the interest of the students.

Throughout the text SI units have been used. Tables and charts given in the Appendix are also in SI units.

This book is based mainly on the lecture notes prepared for classes on the subject at IIT Kharagpur. I am grateful to the authors of the books that I used in preparing the notes, a list of which is given in the bibliography. I am thankful to my colleagues in the mechanical engineering department of IIT Kharagpur, for many stimulating discussions, and for encouragement. I am indebted to all those who have helped in the preparation of the book.

I would very much appreciate criticisms, suggestions for improvement, and detection of errors from my readers, which will be gratefully acknowledged.

**P K NAG**

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The author is indebted to his erstwhile colleagues at IIT for many positive interactions, some of whom were his teachers during his M.Tech course, viz., Prof. C N Lakshminarayan, Late Prof. H R Narayan, Prof. Jayanand H. Hiranandani, Prof. H V Rao, Prof. B G Ghosh, Prof. A K Roy, Late Prof. P K G Pannikar, Prof. Darshan Lal, Prof. N S Murthy and Prof. S K Som.

**P K NAG**

# 1

## CHAPTER

# Introduction

Thermodynamics is the science of energy transfer and its effect on the physical properties of substances. It is based upon observations of common experience which have been formulated into thermodynamic laws. These laws govern the principles of energy conversion. The applications of the thermodynamic laws and principles are found in all fields of energy technology, notably in steam and nuclear power plants, internal combustion engines, gas turbines, air conditioning, refrigeration, gas dynamics, jet propulsion, compressors, chemical process plants, and direct energy conversion devices.

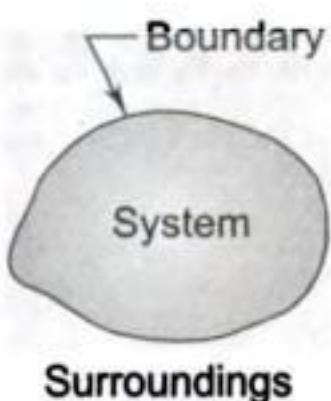
### 1.1 MACROSCOPIC VERSUS MICROSCOPIC VIEWPOINT

There are two points of view from which the behaviour of matter can be studied: the macroscopic and the microscopic. In the macroscopic approach, a certain quantity of matter is considered, without the events occurring at the molecular level being taken into account. From the microscopic point of view, matter is composed of myriads of molecules. If it is a gas, each molecule at a given instant has a certain position, velocity, and energy, and for each molecule these change very frequently as a result of collisions. The behaviour of the gas is described by summing up the behaviour of each molecule. Such a study is made in *microscopic or statistical thermodynamics*. *Macroscopic thermodynamics* is only concerned with the effects of the action of many molecules, and these effects can be perceived by human senses. For example, the macroscopic quantity, pressure, is the average rate of change of momentum due to all the molecular collisions made on a unit area. The effects of pressure can be felt. The macroscopic point of view is not concerned with the action of individual molecules, and the force on a given unit area can be measured by using, e.g. a pressure gauge. These macroscopic observations are completely independent of the assumptions regarding the nature of matter. All the results of classical or macroscopic thermodynamics can, however, be derived from the microscopic and statistical study of matter.

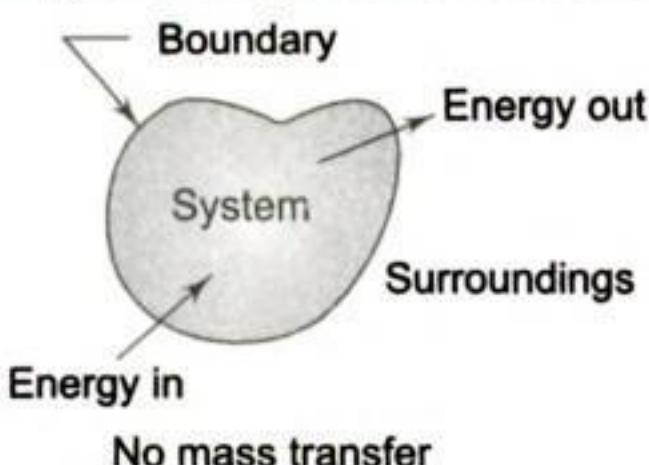
## 1.2 THERMODYNAMIC SYSTEM AND CONTROL VOLUME

A thermodynamic *system* is defined as a quantity of matter or a region in space upon which attention is concentrated in the analysis of a problem. Everything external to the system is called the *surroundings* or the *environment*. The system is separated from the surroundings by the system boundary (Fig. 1.1). The boundary may be either *fixed* or *moving*. A system and its surroundings together comprise a *universe*.

There are three classes of systems: (a) closed system, (b) open system, and (c) isolated system. The *closed system* (Fig. 1.2) is a system of fixed mass. There is

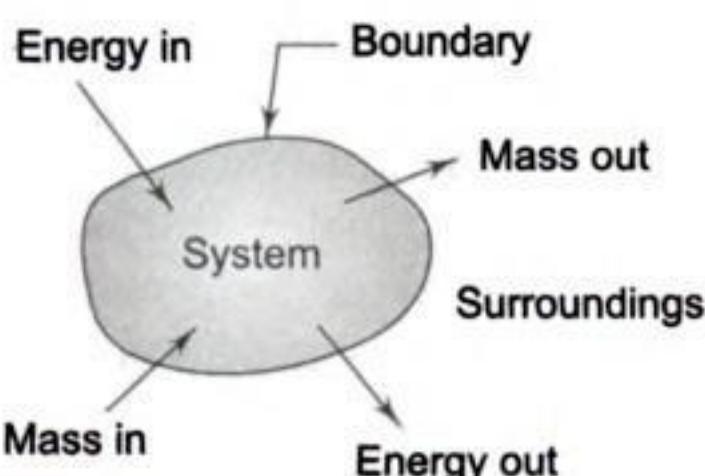


**Fig. 1.1** A Thermodynamic System

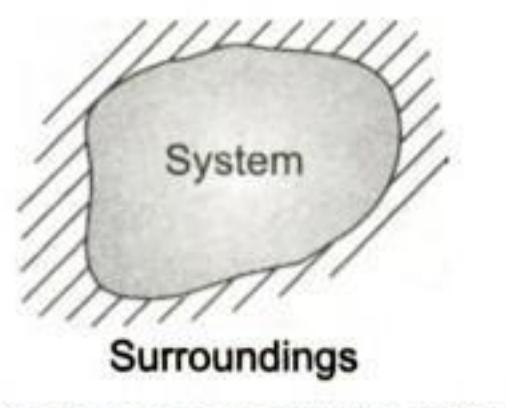


**Fig. 1.2** A Closed System

no mass transfer across the system boundary. There may be energy transfer into or out of the system. A certain quantity of fluid in a cylinder bounded by a piston constitutes a closed system. The *open system* (Fig. 1.3) is one in which matter crosses the boundary of the system. There may be energy transfer also. Most of the engineering devices are generally open systems, e.g. an air compressor in which air enters at low pressure and leaves at high pressure and there are energy transfers across the system boundary. The *isolated system* (Fig. 1.4) is one in which there is no interaction between the system and the surroundings. It is of fixed mass and energy, and there is no mass or energy transfer across the system boundary.



**Fig. 1.3** An Open System

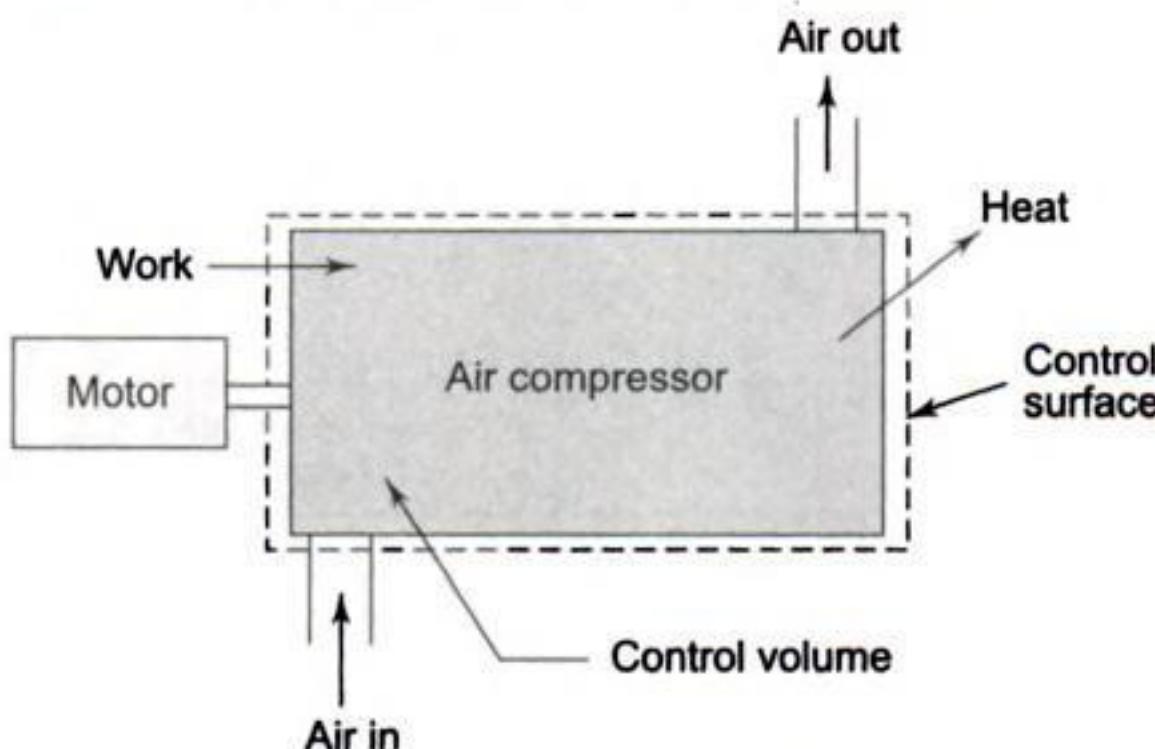


No mass or energy transfer

**Fig. 1.4** An Isolated System

If a system is defined as a certain quantity of matter, then the system contains the same matter and there can be no transfer of mass across its boundary. However, if a system is defined as a region of space within a prescribed boundary, then matter can cross the system boundary. While the former is called a closed system, the latter is an open system.

For thermodynamic analysis of an open system, such as an air compressor (Fig. 1.5), attention is focussed on a certain volume in space surrounding the compressor, known as the *control volume*, bounded by a surface called the *control surface*. Matter as well as energy crosses the control surface.



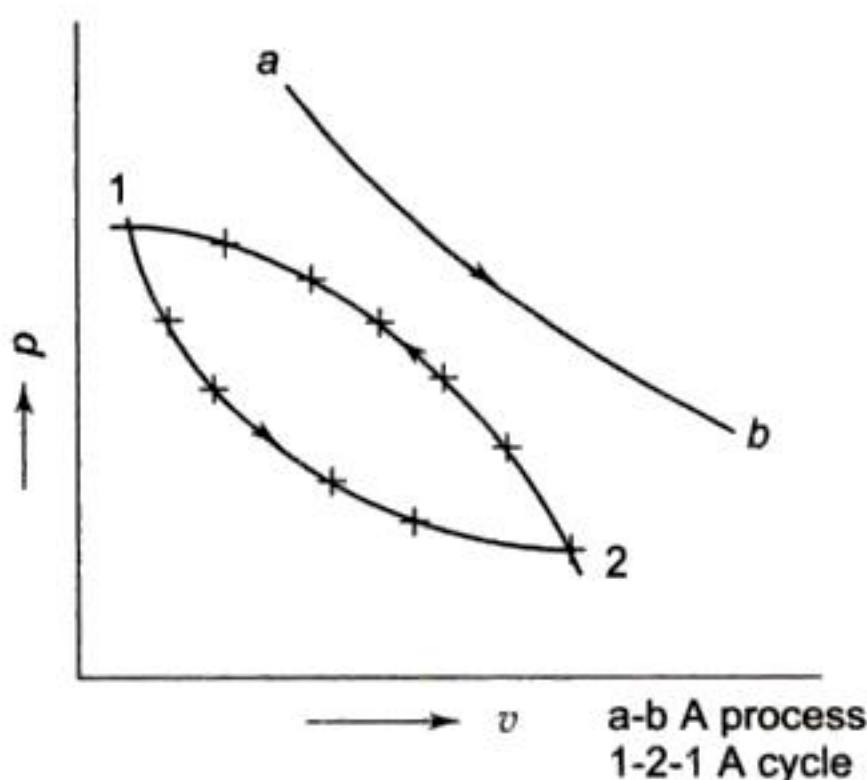
**Fig. 1.5 Control Volume and Control Surface**

A closed system is a system closed to matter flow, though its volume can change against a flexible boundary. When there is matter flow, then the system is considered to be a volume of fixed identity, the control volume. There is thus no difference between an open system and a control volume.

### 1.3 THERMODYNAMIC PROPERTIES, PROCESSES AND CYCLES

Every system has certain characteristics by which its physical condition may be described, e.g. volume, temperature, pressure, etc. Such characteristics are called *properties* of the system. These are all *macroscopic* in nature. When all the properties of a system have definite values, the system is said to exist at a definite *state*. Properties are the coordinates to describe the state of a system. They are the state variables of the system. Any operation in which one or more of the properties of a system changes is called a *change of state*. The succession of states passed through during a change of state is called the *path* of the change of state. When the path is completely specified, the change of state is called a *process*, e.g. a constant pressure process. A thermodynamic *cycle* is defined as a series of state changes such that the final state is identical with the initial state (Fig. 1.6).

Properties may be of two types. *Intensive properties* are independent of the mass in the system, e.g. pressure, temperature, etc. *Extensive properties* are related to mass, e.g. volume, energy, etc. If mass is increased, the values of the extensive properties also increase. Specific extensive properties, i.e. extensive properties per unit mass, are intensive properties, e.g. specific volume, specific energy, density, etc.



**Fig. 1.6** A Process and a Cycle

## 1.4 HOMOGENEOUS AND HETEROGENEOUS SYSTEMS

A quantity of matter homogeneous throughout in chemical composition and physical structure is called a *phase*. Every substance can exist in any one of the three phases, viz. solid, liquid and gas. A system consisting of a single phase is called a *homogeneous system*, while a system consisting of more than one phase is known as a *heterogeneous system*.

## 1.5 THERMODYNAMIC EQUILIBRIUM

A system is said to exist in a state of *thermodynamic equilibrium* when no change in any macroscopic property is registered, if the system is isolated from its surroundings.

An isolated system always reaches in course of time a state of thermodynamic equilibrium and *can never depart from it spontaneously*.

Therefore, there can be no *spontaneous change in any macroscopic property* if the system exists in an equilibrium state. Thermodynamics studies mainly the properties of physical systems that are found in equilibrium states.

A system will be in a state of thermodynamic equilibrium, if the conditions for the following three types of equilibrium are satisfied:

- (a) Mechanical equilibrium
- (b) Chemical equilibrium
- (c) Thermal equilibrium

In the absence of any unbalanced force within the system itself and also between the system and the surroundings, the system is said to be in a state of *mechanical equilibrium*. If an unbalanced force exists, either the system alone or both the system and the surroundings will undergo a change of state till mechanical equilibrium is attained.

If there is no chemical reaction or transfer of matter from one part of the system to another, such as diffusion or solution, the system is said to exist in a state of *chemical equilibrium*.

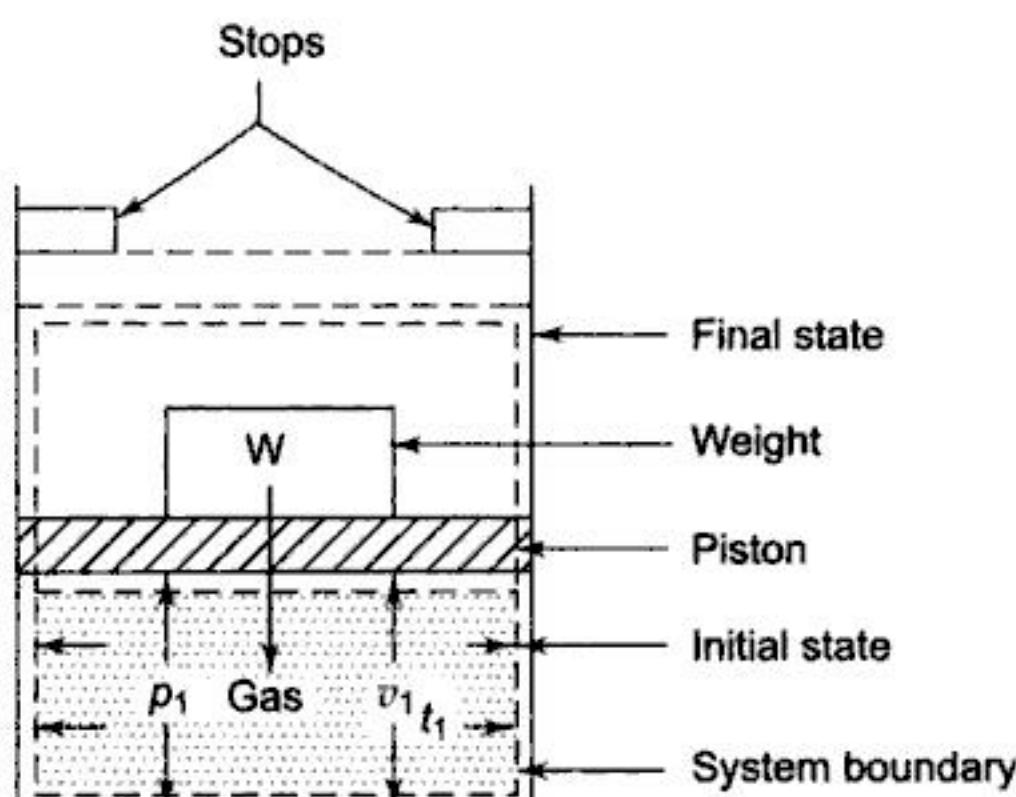
When a system existing in mechanical and chemical equilibrium is separated from its surroundings by a diathermic wall (diathermic means 'which allows heat to flow') and if there is no spontaneous change in any property of the system, the system is said to exist in a state of *thermal equilibrium*. When this is not satisfied, the system will undergo a change of state till thermal equilibrium is restored.

When the conditions for any one of the three types of equilibrium are not satisfied, a system is said to be in a *nonequilibrium state*. If the nonequilibrium of the state is due to an unbalanced force in the interior of a system or between the system and the surroundings, the pressure varies from one part of the system to another. There is no single pressure that refers to the system as a whole. Similarly, if the nonequilibrium is because of the temperature of the system being different from that of its surroundings, there is a nonuniform temperature distribution set up within the system and there is no single temperature that stands for the system as a whole. It can thus be inferred that when the conditions for thermodynamic equilibrium are not satisfied, the states passed through by a system cannot be described by thermodynamic properties which represent the system as a whole.

Thermodynamic properties are the macroscopic coordinates defined for, and significant to, only thermodynamic equilibrium states. Both classical and statistical thermodynamics study mainly the equilibrium states of a system.

## 1.6 QUASI-STATIC PROCESS

Let us consider a system of gas contained in a cylinder (Fig. 1.7). The system initially is in an equilibrium state, represented by the properties  $p_1$ ,  $v_1$ ,  $t_1$ . The weight on the piston just balances the upward force exerted by the gas. If the weight is removed, there will be an unbalanced force between the system and the surroundings, and under gas pressure, the piston will move up till it hits the stops. The system again comes to an equilibrium state, being described by the properties  $p_2$ ,  $v_2$ ,  $t_2$ . But the intermediate states passed through by the system are nonequilibrium states which cannot be described by thermodynamic coordinates.



**Fig. 1.7**

Transition between Two Equilibrium States by an Unbalanced Force

Figure 1.8 shows points 1 and 2 as the initial and final equilibrium states joined by a dotted line, which has got no meaning otherwise. Now if the single weight on the piston is made up of many very small pieces of weights (Fig. 1.9), and these weights are removed one by one very slowly from the top of the piston, at any instant of the upward travel of the piston, if the gas system is isolated, *the departure of the state of the system from the thermodynamic equilibrium state will be infinitesimally small*. So every state passed through by the system will be an

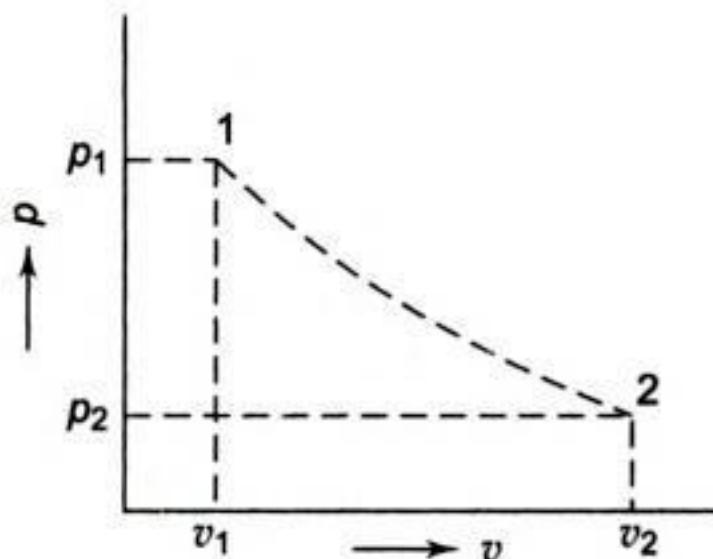


Fig. 1.8 Plot Representing the Transition between Two Equilibrium States

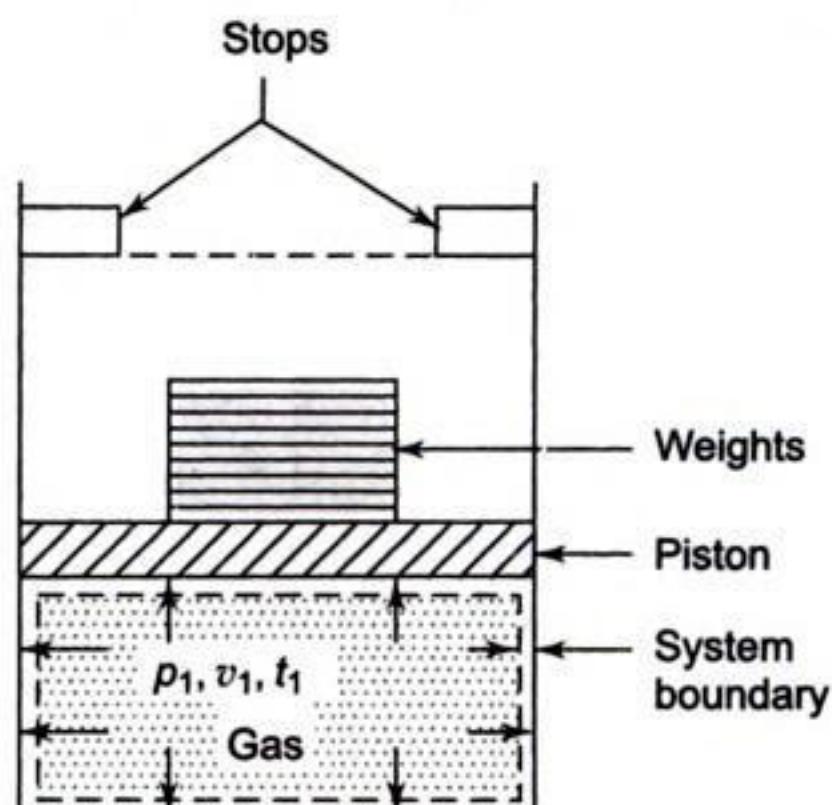


Fig. 1.9 Infinitely Slow Transition of a System by Infinitesimal Force

equilibrium state. Such a process, which is but a locus of all the equilibrium points passed through by the system, is known as a *quasi-static process* (Fig. 1.10), 'quasi' meaning 'almost'. *Infinite slowness is the characteristic feature of a quasi-static process*. A quasi-static process is thus a succession of equilibrium states. A quasi-static process is also called a *reversible process*.

## 1.7 PURE SUBSTANCE

A *pure substance* is defined as one that is homogeneous and invariable in chemical composition throughout its mass. The relative proportions of the chemical elements constituting the substance are also constant. Atmospheric air, steam-water mixture and combustion products of a fuel are regarded as pure substances. But the mixture of air and liquid air is not a pure substance, since the relative proportions of oxygen and nitrogen differ in the gas and liquid phases in equilibrium.

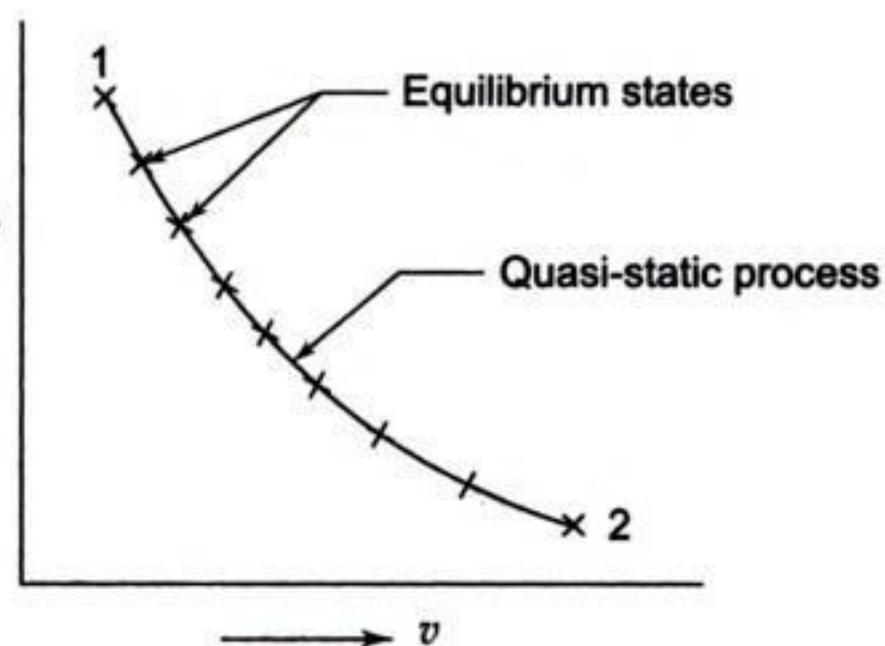


Fig. 1.10 A Quasi-Static Process

The state of a pure substance of given mass can be fixed by specifying two independent intensive properties, provided the system is in equilibrium. This is known as the '*two-property rule*'. The state can thus be represented as a point on thermodynamic property diagrams. Once any two properties of a pure substance are known, other properties can be determined from the available thermodynamic relations.

## 1.8 CONCEPT OF CONTINUUM

From the macroscopic viewpoint, we are always concerned with volumes which are very large compared to molecular dimensions. Even a very small volume of a system is assumed to contain a large number of molecules so that statistical averaging is meaningful and a property value can be assigned to it. Disregarding the behaviour of individual molecules, matter is here treated as continuous. Let us consider the mass  $\delta m$  in a volume  $\delta V$  surrounding the point P (Fig. 1.11). The ratio  $\delta m/\delta V$  is the average mass density of the system within the volume  $\delta V$ . We suppose that at first  $\delta V$  is rather large, and is subsequently shrunk about the point P. If we plot  $\delta m/\delta V$  against  $\delta V$ , the average density tends to approach an asymptote as  $\delta V$  increases (Fig. 1.12). However, when  $\delta V$  becomes so small as to contain relatively few molecules, the average density fluctuates substantially with time as molecules pass into and out of the volume in random motion, and so it is impossible to speak of a definite value of  $\delta m/\delta V$ . The smallest volume which may be regarded as continuous is  $\delta V'$ . The density  $\rho$  of the system at a point is thus defined as

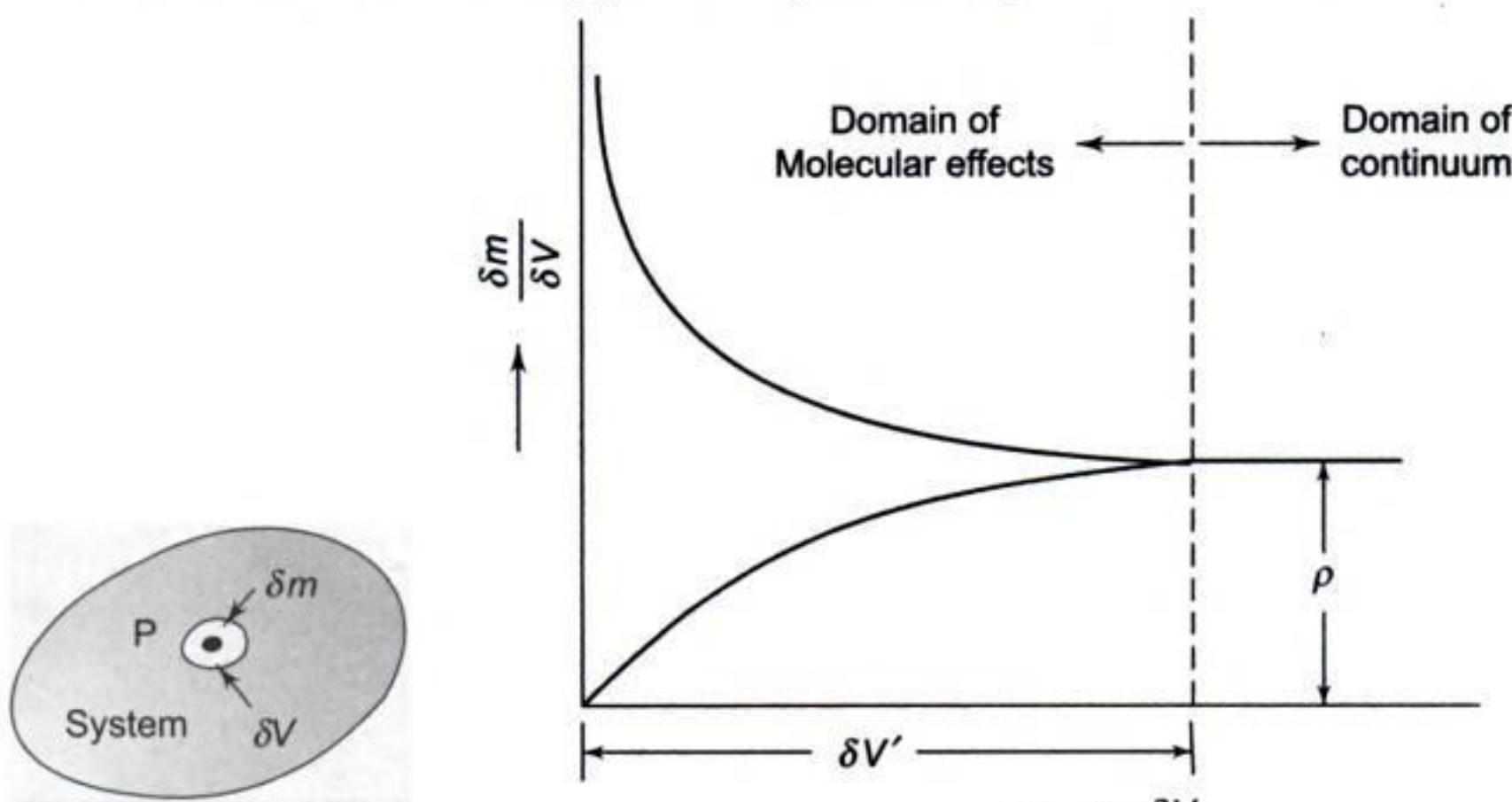


Fig. 1.11

Fig. 1.12

Definition of the Macroscopic Property, Density

$$\rho = \lim_{\delta V \rightarrow \delta V'} \frac{\delta m}{\delta V} \quad (1.1)$$

Similarly, the fluid velocity at a point P is defined as the instantaneous velocity of the centre of gravity of the smallest continuous volume  $\delta V'$ .

The concept of continuum loses validity when the mean free path of the molecules approaches the order of magnitude of the dimensions of the vessel, as, for instance, in highly rarefied gases encountered in high vacuum technology, in rocket flights at high altitudes and in electron tubes. In most engineering applications, however, the assumption of a continuum is valid and convenient, and goes hand in hand with the macroscopic point of view.

## 1.9 THERMOSTATICS

The science of thermodynamics deals with systems existing in thermodynamic equilibrium states which are specified by properties. Infinitely slow quasi-static processes executed by systems are only meaningful in thermodynamic plots. The name ‘thermodynamics’ is thus said to be a misnomer, since it does not deal with the dynamics of heat, which is nonquasi-static. The name ‘thermostatics’ then seems to be more appropriate. However, most of the real processes are dynamic and nonquasi-static, although the initial and final states of the system might be in equilibrium. Such processes can be successfully dealt with by the subject. Hence, the term ‘thermodynamics’ is not inappropriate.

## 1.10 UNITS AND DIMENSIONS

In the present text, the SI (System International) system of units has been used. The basic units in this system are given in Table 1.1.

**Table 1.1 System: Basic Units**

Quantity	Unit	Symbol
Length (L)	Metre	m
Mass (M)	Kilogram	kg
Time (t)	Second	s
Amount of substance	Mole	mol
Temperature (T)	Kelvin	K
Electric current	Ampere	A
Luminous intensity	Candela	cd
Plane angle	Radian	rad
Solid angle	Steradian	sr

The dimensions of all other quantities are derived from these basic units which are given in Table 1.2.

**Table 1.2** SI System: Derived Units

<i>Quantity</i>	<i>Unit</i>	<i>Symbol</i>	<i>Alternative unit</i>	<i>In basic units</i>
Force (F)	Newton	N		$\text{kg m/s}^2$
Energy (E)	Joule	J	Nm	$\text{kg m}^2/\text{s}^2$
Power	Watt	W	J/s	$\text{kg m}^2/\text{s}^3$
Pressure	Pascal	Pa	N/m <sup>2</sup>	$\text{kg}/(\text{ms}^2)$
Frequency	Hertz	Hz		s <sup>-1</sup>
Electric charge	Coulomb	C		As
Electric potential	Volt	V	W/A = J/C	$\text{kg m}^2/(\text{s}^3 \text{ A})$
Capacitance	Farad	F	C/V	$\text{s}^4 \text{ A}^2/(\text{kg m}^2)$
Electrical resistance	Ohm	$\Omega$	V/A	$\text{kg m}^2/(\text{s}^3 \text{ A}^2)$
Magnetic flux	Weber	Wb	Vs	$\text{kg m}^2/(\text{s}^2 \text{ A})$
Magnetic flux density	Tesla	T	Wb/m <sup>2</sup>	$\text{kg}/(\text{s}^2 \text{ A})$
Inductance	Henry	H	Wb/A	$\text{kg m}^2/(\text{s}^2 \text{ A}^2)$

It is often convenient and desirable to use multiples of various units, the standard list of which is given in Table 1.3.

**Table 1.3** SI System: Standard Multipliers

<i>Factor</i>	<i>Prefix</i>	<i>Factor</i>	<i>Prefix</i>
$10^{12}$	tera, T	$10^{-3}$	milli, m
$10^9$	giga, G	$10^{-6}$	micro, $\mu$
$10^6$	mega, M	$10^{-9}$	nano, n
$10^3$	kilo, k	$10^{-12}$	pico, p

### 1.10.1 Force

The force acting on a body is defined by Newton's second law of motion. The unit of force is the newton (N). A force of one newton produces an acceleration of  $1 \text{ ms}^{-2}$  when applied to a mass of 1 kg.

$$1 \text{ N} = 1 \text{ kg m/s}^2$$

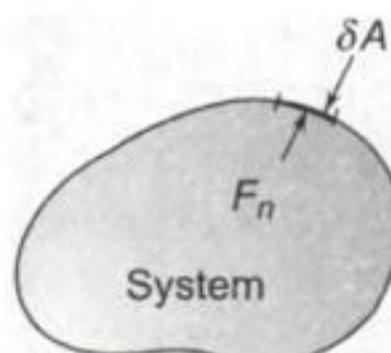
The weight of a body (W) is the force with which the body is attracted to the centre of the earth. It is the product of its mass (m) and the local gravitational acceleration (g), i.e.

$$W = mg$$

The value of g at sea level is  $9.80665 \text{ m/s}^2$ . The mass of a substance remains constant with elevation, but its weight varies with elevation.

### 1.10.2 Pressure

Pressure is the normal force exerted by a system against unit area of the bounding surface. If  $\delta A$  is a small area and  $\delta A'$  is the smallest area from continuum consideration, and  $\delta F_n$  is the component of force normal to  $\delta A$  (Fig. 1.13), the pressure  $p$  at a point on the wall is defined as

**Fig. 1.13** Definition of Pressure

$$p = \lim_{\delta A \rightarrow \delta A'} \frac{\delta F_n}{\delta A}$$

The pressure  $p$  at a point in a fluid in equilibrium is the same in all directions.

The unit for pressure in the SI system is the *pascal* (Pa), which is the force of one newton acting on an area of  $1\text{ m}^2$ .

$$1\text{ Pa} = 1\text{ N/m}^2$$

The unit of pascal is very small. Very often kilo-pascal (kPa) or mega-pascal (MPa) is used.

Two other units, not within the SI system of units, continue to be widely used. These are the *bar*, where

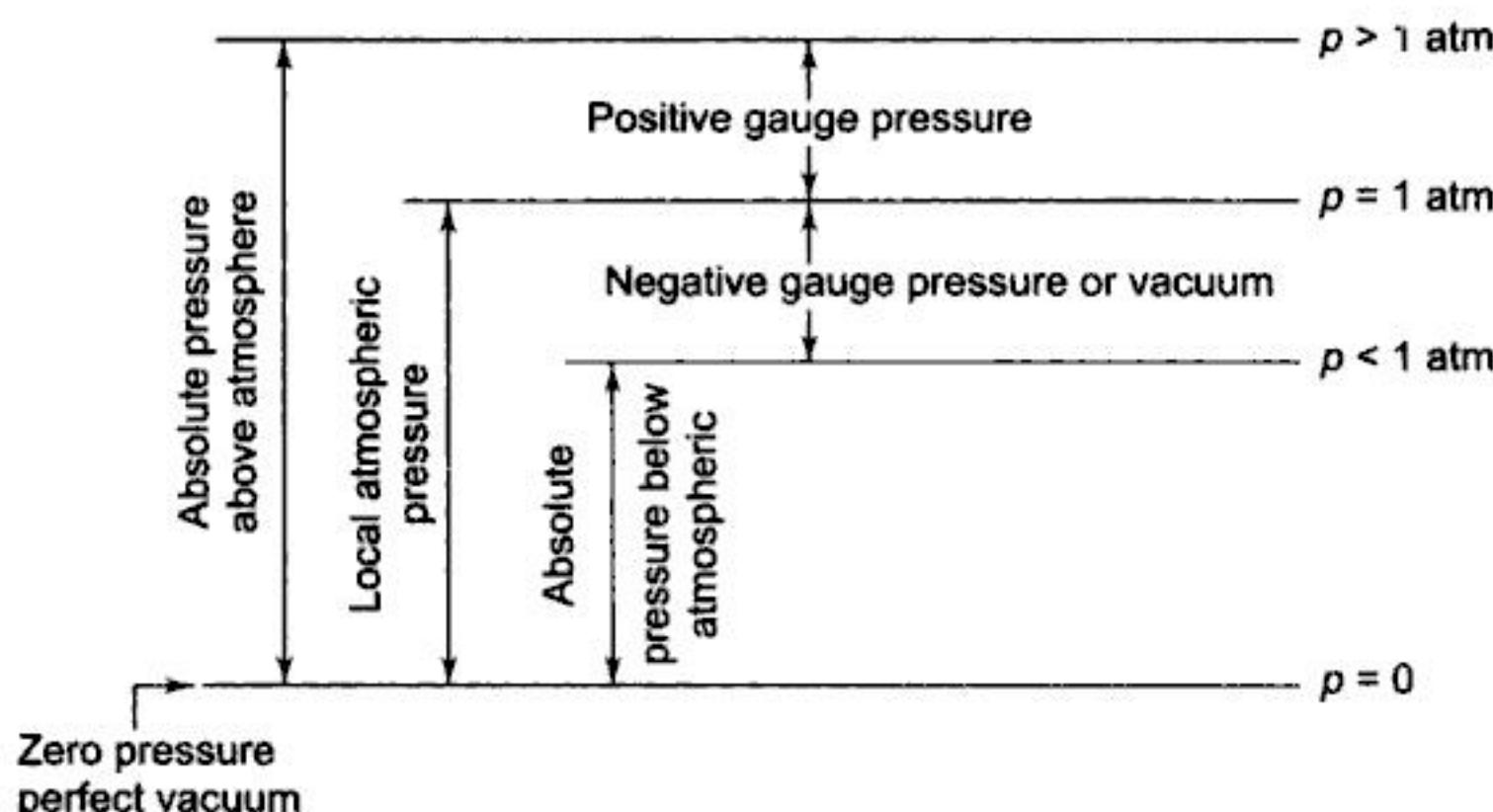
$$1\text{ bar} = 10^5\text{ Pa} = 100\text{ kPa} = 0.1\text{ MPa}$$

and the standard atmosphere, where

$$1\text{ atm} = 101.325\text{ kPa} = 1.01325\text{ bar}$$

Most instruments indicate pressure relative to the atmospheric pressure, whereas the pressure of a system is its pressure above zero, or relative to a perfect vacuum (Fig. 1.14). The pressure relative to the atmosphere is called *gauge pressure*. The pressure relative to a perfect vacuum is called *absolute pressure*.

$$\text{Absolute pressure} = \text{Gauge pressure} + \text{Atmospheric pressure}$$



**Fig. 1.14 Relationship between Pressures**

When the pressure in a system is less than atmospheric pressure, the gauge pressure becomes negative, but is frequently designated by a positive number and called *vacuum*. For example, 16 cm vacuum will be

$$\frac{76 - 16}{76} \times 1.013 = 0.80\text{ bar}$$

Thus,

$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}}$$

Figure 1.15 shows a few pressure measuring devices. Figure (a) shows the Bourdon gauge which measures the difference between the system pressure inside the tube and atmospheric pressure. It relies on the deformation of a bent hollow tube of suitable material which, when subjected to the pressure to be measured on the inside (and atmospheric pressure on the outside), tends to unbend. This moves a pointer through a suitable gear-and lever mechanism against a calibrated scale. Figure (b) shows an open U-tube indicating gauge pressure, and Fig. (c) shows an open U-tube indicating vacuum. Figure (d) shows a closed U-tube indicating absolute pressure. If  $p$  is atmospheric pressure, this is a *barometer*. These are called U-tube manometers.

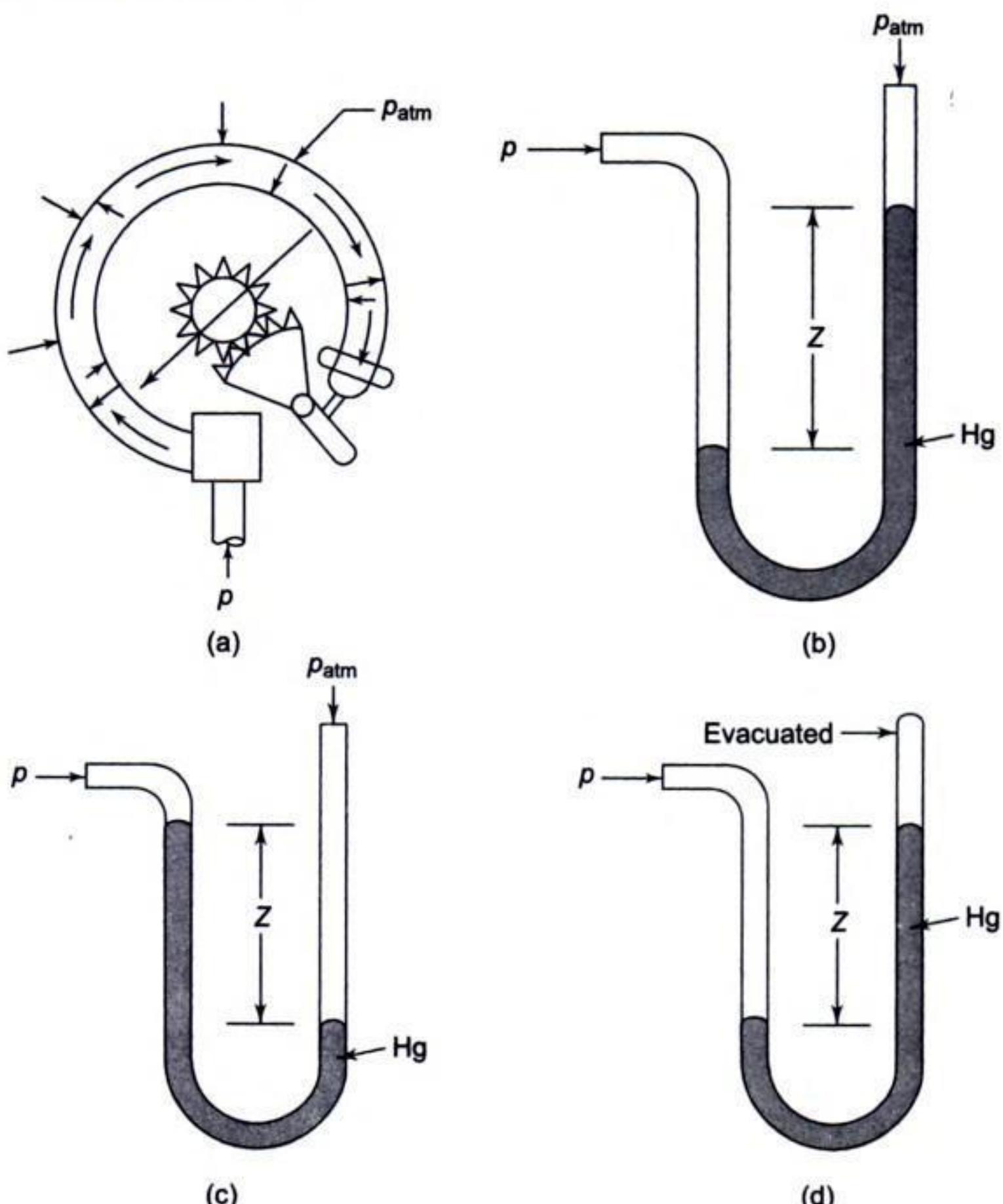


Fig. 1.15 Pressure Gauges (a) Bourdon Gauge (b) Open U-tube Indicating Gauge Pressure (c) Open U-tube Indicating Vacuum (d) Closed U-tube Indicating Absolute Pressure

If  $Z$  is the difference in the heights of the fluid columns in the two limbs of the U-tube (Fig. (b) and Fig. (c)],  $\rho$  the density of the fluid and  $g$  the acceleration due to gravity, then from the elementary principle of hydrostatics, the gauge pressure  $p_g$  is given by

$$p_g = Z\rho g \left[ m \cdot \frac{\text{kg}}{\text{m}^3} \frac{\text{m}}{\text{s}^2} \right] = Z\rho g \text{ N/m}^2$$

If the fluid is mercury having  $\rho = 13,616 \text{ kg/m}^3$ , one metre head of mercury column is equivalent to a pressure of 1.3366 bar, as shown below

$$\begin{aligned} 1 \text{ m Hg} &= Z \rho g = 1 \times 13616 \times 9.81 \\ &= 1.3366 \times 10^5 \text{ N/m}^2 = 1.3366 \text{ bar} \end{aligned}$$

The manometer is a sensitive, accurate and simple device, but it is limited to fairly small pressure differentials and, because of the inertia and friction of the liquid, is not suitable for fluctuating pressures, unless the rate of pressure change is small. A diaphragm-type pressure transducer along with a cathode ray oscilloscope can be used to measure rapidly fluctuating pressures. The unit of 1 mm Hg pressure is called *torr*, so that

$$1 \text{ mm Hg} = 1 \text{ torr} = 133 \text{ Pa}$$

Figure 1.16 shows a typical U-tube manometer, one end of which is connected to the vessel the pressure of which is to be measured and the other end is open to atmosphere exerting pressure  $p_{\text{atm}}$ . The manometric fluid may be mercury, water, alcohol, oil, etc. The pressure along the horizontal line AB is the same in both the limbs of the manometer, so that

$$p + \rho_1 g z_1 = p_{\text{atm}} + \rho_2 g z_2$$

If the vessel contains a gas of density  $\rho_1$  and since the density of the manometric fluid, (say Hg),  $\rho_2 \gg \rho_1$ ,

$$p - p_{\text{atm}} = \rho_2 g z_2 = p_{\text{gauge}}$$

### 1.10.3 Specific Volume and Density

Volume ( $V$ ) is the space occupied by a substance and is measured in  $\text{m}^3$ . The specific volume ( $v$ ) of a substance is defined as the volume per unit mass and is measured in  $\text{m}^3/\text{kg}$ . From continuum consideration the specific volume at a point is defined as

$$v = \lim_{\delta V' \rightarrow 0} \frac{\delta V}{\delta m}$$

where  $dV'$  is the smallest volume for which the system can be considered a continuum.

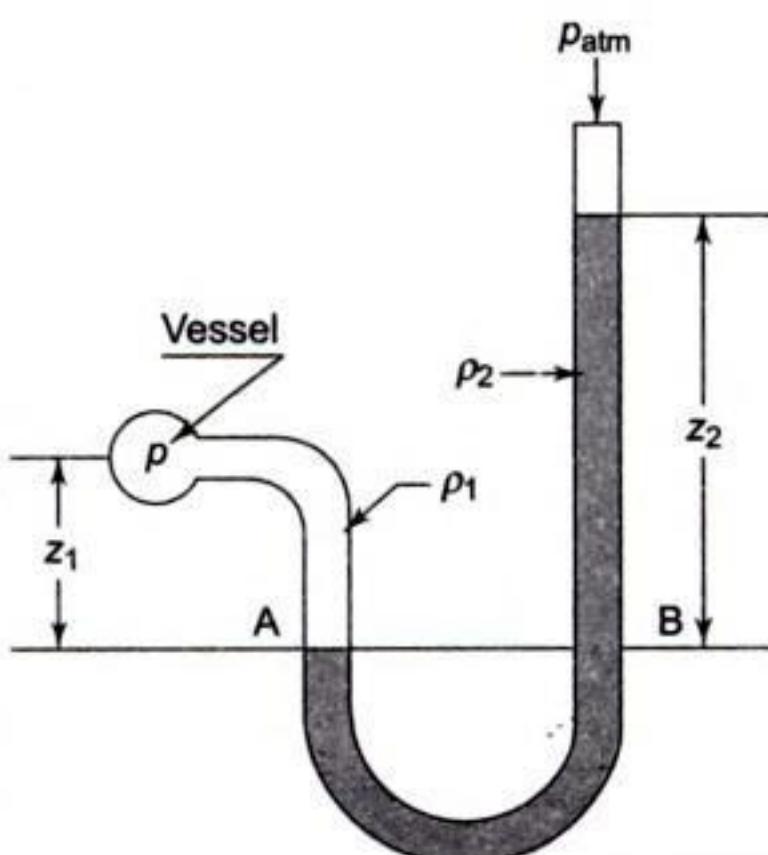


Fig. 1.16

Pressure Measurement  
by a Manometer

Density ( $\rho$ ) is the mass per unit volume of a substance, which has been discussed earlier, and is given in  $\text{kg/m}^3$ .

$$\rho = \frac{m}{v}$$

In addition to  $\text{m}^3$ , another commonly used unit of volume is the litre (l).

$$1 \text{ l} = 10^{-3} \text{ m}^3$$

The specific volume or density may be given either on the basis of mass or in respect of mole. A *mole* of a substance has a mass numerically equal to the molecular weight of the substance. One g mol of oxygen has a mass of 32 g and 1 kg mol (or kmol) of nitrogen has a mass of 28 kg. The symbol  $\bar{v}$  is used for molar specific volume ( $\text{m}^3/\text{kmol}$ ).

#### 1.10.4 Energy

Energy is the capacity to exert a force through a distance, and manifests itself in various forms. Engineering processes involve the conversion of energy from one form to another, the transfer of energy from place to place, and the storage of energy in various forms, utilizing a working substance.

The unit of energy in the SI system is Nm or J (joule). The energy per unit mass is the specific energy, the unit of which is  $\text{J/kg}$ .

#### 1.10.5 Power

The rate of energy transfer or storage is called power. The unit of power is watt (W), kilowatt (kW) or megawatt (MW).

$$1 \text{ W} = 1 \text{ J/s} = 1 \text{ Nm/s}$$

$$1 \text{ kW} = 1000 \text{ W}$$

### Solved Examples

**Example 1.1** The pressure of gas in a pipe line is measured with a mercury manometer having one limb open to the atmosphere (Fig. 1.17). If the difference in the height of mercury in the two limbs is 562 mm, calculate the gas pressure. The barometer reads 761 mm Hg, the acceleration due to gravity is  $9.79 \text{ m/s}^2$ , and the density of mercury is  $13,640 \text{ kg/m}^3$ .

**Solution** At the plane AB, we have

$$p = p_0 + \rho g z$$

Now

$$p_0 = \rho g z_0$$

where  $z_0$  is the barometric height,  $\rho$  the density of mercury and  $p_0$  the atmospheric pressure.

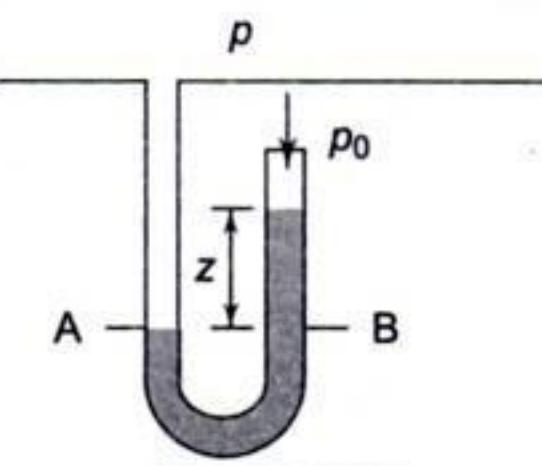


Fig. 1.17

Therefore

$$\begin{aligned} p &= \rho g (z + z_0) \\ &= 13,640 \text{ kg/m}^3 \times 9.79 \text{ m/s}^2 (0.562 + 0.761) \text{ m} \\ &= 177 \times 10^3 \text{ N/m}^2 = 177 \text{ kPa} = 1.77 \text{ bar} = \mathbf{1.746 \text{ atm}} \end{aligned}$$

**Example 1.2** A turbine is supplied with steam at a gauge pressure of 1.4 MPa. After expansion in the turbine the steam flows into a condenser which is maintained at a vacuum of 710 mm Hg. The barometric pressure is 772 mm Hg. Express the inlet and exhaust steam pressures in pascals (absolute). Take the density of mercury as  $13.6 \times 10^3 \text{ kg/m}^3$ .

**Solution** The atmospheric pressure  $p_0$

$$\begin{aligned} &= \rho g z_0 = 13.6 \times 10^3 \text{ kg/m}^3 \times 9.81 \text{ m/s}^2 \times 0.772 \text{ m} \\ &= 1.03 \times 10^5 \text{ Pa} \end{aligned}$$

Inlet steam pressure

$$\begin{aligned} &= [(1.4 \times 10^6) + (1.03 \times 10^5)] \text{ Pa} \\ &= 15.03 \times 10^5 \text{ Pa} = \mathbf{1.503 \text{ MPa}} \end{aligned}$$

Condenser pressure

$$\begin{aligned} &= (0.772 - 0.710) \text{ m} \times 9.81 \text{ m/s}^2 \times 13.6 \times 10^3 \text{ kg/m}^3 \\ &= 0.827 \times 10^4 \text{ Pa} = \mathbf{8.27 \text{ kPa}} \end{aligned}$$

**Example 1.3** Convert the following readings of pressure to kPa, assuming that the barometer reads 760 mm of Hg.

- (a) 40 cm Hg vacuum
- (b) 90 cm Hg gauge
- (c) 1.2 m of  $\text{H}_2\text{O}$  gauge

**Solution** (a)

$$\begin{aligned} p_{\text{vacuum}} &= h \rho_g = (40 \times 10^{-2}) \times (13.6 \times 10^3) \times 9.8 \\ &= 53.31 \times 10^3 \text{ N/m}^2 = \mathbf{53.31 \text{ kPa}} \end{aligned}$$

$$\begin{aligned} p_{\text{absolute}} &= p_{\text{atm}} - p_{\text{vac}} \\ &= (760 - 400) \times 9.8 \times 13.6 \times 10^3 \\ &= 48 \times 10^3 \text{ N/m}^2 = \mathbf{48 \text{ kPa}} \end{aligned}$$

Also,

$$p_{\text{abs}} = 101.325 - 53.31 = 48.015 \text{ kPa}$$

(b)

$$\begin{aligned} p_{\text{gauge}} &= h \rho g \\ &= (90 \times 10^{-2}) \times (13.6 \times 10^3) \times 9.8 \\ &= 120 \times 10^3 \text{ N/m}^2 = \mathbf{120 \text{ kPa}} \end{aligned}$$

Since

$$p_{\text{atm}} = 760 \text{ mm Hg} = 101.325 \text{ kPa}$$

$$\begin{aligned} p_{\text{abs}} &= p_{\text{gauge}} + p_{\text{atm}} \\ &= 120 + 101.325 = \mathbf{221.325 \text{ kPa}} \end{aligned}$$

(c) 
$$P_{\text{gauge}} = h \rho g = 1.2 \text{ m} \times 1000 \frac{\text{kg}}{\text{m}^3} \times 9.81 \frac{\text{m}}{\text{s}^2}$$
  

$$= 11.772 \text{ kPa}$$

$\therefore P_{\text{abs}} = 11.772 + 101.325 = 113.097 \text{ kPa}$

**Example 1.4** The pressure and specific volume of the atmosphere are related according to the equation  $p v^{1.4} = 2.5 \times 10^5$ , where  $p$  is in  $\text{N/m}^2$  and  $v$  in  $\text{m}^3/\text{kg}$ . What depth of atmosphere is required to produce a pressure of 1.033 bar at the earth's surface? Assume  $g = 9.81 \text{ m/s}^2$ .

**Solution** Let  $H$  be the depth of the atmosphere required to produce the pressure of 1.033 bar at the earth's surface. Considering an element of length  $dh$  (Fig. 1.18), by force balance:

$$A(p + dp) = mg + pA$$

$$pA + Adp = \rho Adhg + pA$$

or, 
$$dh = \frac{1}{g} \nu dp$$

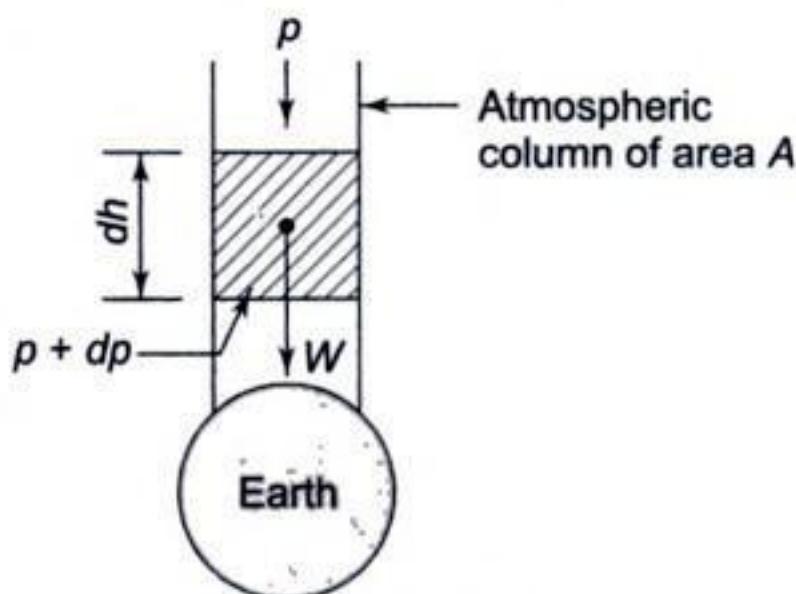


Fig. 1.18

or, 
$$\int_0^H dh = \int_0^{1.033 \times 10^5} \frac{1}{g} \left( \frac{2.5 \times 10^5}{p} \right)^{1/4} dp$$

$$H = \frac{1}{g} (2.5 \times 10^5)^{0.714} \int_0^{1.033 \times 10^5} p^{-0.714} dp$$

$$= 731.274 \frac{(1.033 \times 10^5)^{0.286}}{0.286} = 69,462 \text{ m} = 69.462 \text{ km}$$

## Review Questions

- 1.1 What do you understand by macroscopic and microscopic viewpoints?
- 1.2 Is thermodynamics a misnomer for the subject?
- 1.3 How does the subject of thermodynamics differ from the concept of heat transfer?
- 1.4 What is the scope of classical thermodynamics?
- 1.5 What is a thermodynamic system?
- 1.6 What is the difference between a closed system and an open system?
- 1.7 An open system defined for a fixed region and a control volume are synonymous. Explain.
- 1.8 Define an isolated system.
- 1.9 Distinguish between the terms 'change of state', 'path', and 'process'.
- 1.10 What is a thermodynamic cycle?

- 1.11 What are intensive and extensive properties?
- 1.12 What do you mean by homogeneous and heterogeneous systems?
- 1.13 Explain what you understand by thermodynamic equilibrium.
- 1.14 Explain mechanical, chemical and thermal equilibrium.
- 1.15 What is a quasi-static process? What is its characteristic feature?
- 1.16 What is the concept of continuum? How will you define density and pressure using this concept?
- 1.17 What is vacuum? How can it be measured?
- 1.18 What is a pressure transducer?

## Problems

- 1.1 A pump discharges a liquid into a drum at the rate of  $0.032 \text{ m}^3/\text{s}$ . The drum, 1.50 m in diameter and 4.20 m in length, can hold 3000 kg of the liquid. Find the density of the liquid and the mass flow rate of the liquid handled by the pump.  
Ans. 12.934 kg/s

- 1.2 The acceleration of gravity is given as a function of elevation above sea level by

$$g = 980.6 - 3.086 \times 10^{-6} H$$

where  $g$  is in  $\text{cm/s}^2$ , and  $H$  is in cm. If an aeroplane weighs 90,000 N at sea level, what is the gravity force upon it at 10,000 m elevation? What is the percentage difference from the sea-level weight? Ans. 89,716.4 N, 0.315%

- 1.3 Prove that the weight of a body at an elevation  $H$  above sea-level is given by

$$W = \frac{mg}{g_0} \left( \frac{d}{d + 2H} \right)^2$$

where  $d$  is the diameter of the earth.

- 1.4 The first artificial earth satellite is reported to have encircled the earth at a speed of 28,840 km/h and its maximum height above the earth's surface was stated to be 916 km. Taking the mean diameter of the earth to be 12,680 km, and assuming the orbit to be circular, evaluate the value of the gravitational acceleration at this height.

The mass of the satellite is reported to have been 86 kg at sea-level. Estimate the gravitational force acting on the satellite at the operational altitude.

Ans.  $8.9 \text{ m/s}^2$ ; 765 N

- 1.5 Convert the following readings of pressure to kPa, assuming that the barometer reads 760 mm Hg:

(a) 90 cm Hg gauge, (b) 40 cm Hg vacuum, (c) 1.2 m  $\text{H}_2\text{O}$  gauge, (d) 3.1 bar.

- 1.6 A 30 m high vertical column of a fluid of density  $1878 \text{ kg/m}^3$  exists in a place where  $g = 9.65 \text{ m/s}^2$ . What is the pressure at the base of the column?

Ans. 544 kPa

- 1.7 Assume that the pressure  $p$  and the specific volume  $v$  of the atmosphere are related according to the equation  $pv^{1.4} = 2.3 \times 10^5$ , where  $p$  is in  $\text{N/m}^2$  abs and  $v$  is in  $\text{m}^3/\text{kg}$ . The acceleration due to gravity is constant at  $9.81 \text{ m/s}^2$ . What is the depth of atmosphere necessary to produce a pressure of 1.0132 bar at the earth's surface? Consider the atmosphere as a fluid column.

Ans. 64.8 km

- 1.8 The pressure of steam flowing in a pipeline is measured with a mercury manometer, shown in Fig. 1.19. Some steam condenses into water. Estimate the steam pressure in kPa. Take the density of mercury as  $13.6 \times 10^3 \text{ kg/m}^3$ , density of water as  $10^3 \text{ kg/m}^3$ , the barometer reading as 76.1 cm Hg, and  $g$  as  $9.806 \text{ m/s}^2$ .

- 1.9 A vacuum gauge mounted on a condenser reads 0.66 m Hg. What is the absolute pressure in the condenser in kPa when the atmospheric pressure is 101.3 kPa?

Ans. 13.3 kPa

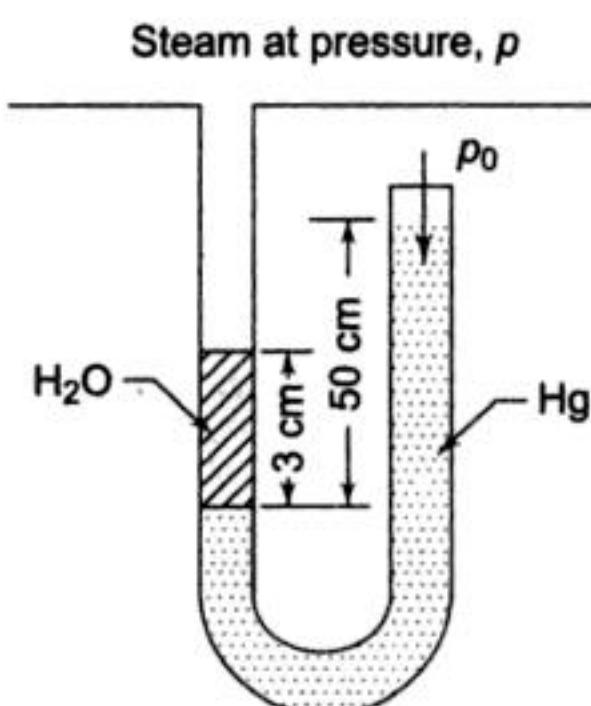


Fig. 1.19

# 2

## CHAPTER

# Temperature

### 2.1 ZEROTH LAW OF THERMODYNAMICS

The property which distinguishes thermodynamics from other sciences is temperature. One might say that temperature bears as important a relation to thermodynamics as force does to statics or velocity does to dynamics. Temperature is associated with the ability to distinguish hot from cold. When two bodies at different temperatures are brought into contact, after some time they attain a common temperature and are then said to exist in thermal equilibrium.

*When a body A is in thermal equilibrium with a body B, and also separately with a body C, then B and C will be in thermal equilibrium with each other.*

This is known as the *zeroth law of thermodynamics*. It is the basis of temperature measurement.

In order to obtain a quantitative measure of temperature, a reference body is used, and a certain physical characteristic of this body which changes with temperature is selected. The changes in the selected characteristic may be taken as an indication of change in temperature. The selected characteristic is called the *thermometric property*, and the reference body which is used in the determination of temperature is called the *thermometer*. A very common thermometer consists of a small amount of mercury in an evacuated capillary tube. In this case the extension of the mercury in the tube is used as the thermometric property.

There are five different kinds of thermometer, each with its own thermometric property, as shown in Table 2.1.

### 2.2 MEASUREMENT OF TEMPERATURE—THE REFERENCE POINTS

The temperature of a system is a property that determines whether or not a system is in thermal equilibrium with other systems. If a body is at, say,  $70^{\circ}\text{C}$ , it will be  $70^{\circ}\text{C}$ , whether measured by a mercury-in-glass thermometer, resistance thermometer or constant volume gas thermometer. If  $X$  is the thermometric property, let us arbitrarily choose for the temperature common to the thermometer and *to all systems in thermal equilibrium with it* the following *linear* function of  $X$ :

**Table 2.1 Thermometers and Thermometric Properties**

<i>Thermometer</i>	<i>Thermometric property</i>	<i>Symbol</i>
1. Constant volume gas thermometer	Pressure	<i>p</i>
2. Constant pressure gas thermometer	Volume	<i>V</i>
3. Electrical resistance thermometer	Resistance	<i>R</i>
4. Thermocouple	Thermal e.m.f.	<i>ε</i>
5. Mercury-in-glass thermometer	Length	<i>L</i>

$\theta(X) = aX$ , where  $a$  is an arbitrary constant.

If  $X_1$  corresponds to  $\theta(X_1)$ , then  $X_2$  will correspond to

$$\frac{\theta(X_1)}{X_1} \cdot X_2$$

that is

$$\theta(X_2) = \frac{\theta(X_1)}{X_1} \cdot X_2 \quad (2.1)$$

Two temperatures on the linear  $X$  scale are to each other as the ratio of the corresponding  $X$ 's.

### 2.2.1 Method Used Before 1954

The thermometer is first placed in contact with the system whose temperature  $\theta(X)$  is to be measured, and then in contact with an arbitrarily chosen standard system in an easily reproducible state where the temperature is  $\theta(X_1)$ . Thus

$$\frac{\theta(X_1)}{\theta(X)} = \frac{X_1}{X} \quad (2.2)$$

Then the thermometer at the temperature  $\theta(X)$  is placed in contact with another arbitrarily chosen standard system in another easily reproducible state where the temperature is  $\theta(X_2)$ . It gives

$$\frac{\theta(X_2)}{\theta(X)} = \frac{X_2}{X} \quad (2.3)$$

From Eqs. (2.2) and (2.3)

$$\frac{\theta(X_1) - \theta(X_2)}{\theta(X)} = \frac{X_1 - X_2}{X}$$

or 
$$\theta(X) = \frac{\theta(X_1) - \theta(X_2)}{X_1 - X_2} \cdot X \quad (2.4)$$

If we assign an arbitrary number of degrees to the temperature interval  $\theta(X_1) - \theta(X_2)$ , then  $\theta(X)$  can be calculated from the measurements of  $X$ ,  $X_1$  and  $X_2$ .

An easily reproducible state of an arbitrarily chosen standard system is called a fixed point. Before 1954, there were two fixed points: (a) the ice point, the temperature at which pure ice coexisted in equilibrium with air-saturated water at one atmosphere pressure, and (b) the steam point, the temperature of equilibrium between pure water and pure steam at one atmosphere pressure. The temperature interval,  $\theta(X_1) - \theta(X_2)$ , between these two fixed points was chosen to be 100 degrees.

The use of two fixed points was found unsatisfactory and later abandoned, because of (a) the difficulty of achieving equilibrium between pure ice and air-saturated water (since when ice melts, it surrounds itself only with pure water and prevents intimate contact with air-saturated water), and (b) extreme sensitiveness of the steam point to the change in pressure.

### 2.2.2 Method in Use After 1954

Since 1954 only one fixed point has been in use, viz. the triple point of water, the state at which ice, liquid water and water vapour coexist in equilibrium. The temperature at which this state exists is arbitrarily assigned the value of 273.16 degrees Kelvin, or 273.16 K (the reason for using Kelvin's name will be explained later). Designating the triple point of water by  $\theta_t$ , and with  $X_t$  being the value of the thermometric property when the body, whose temperature  $\theta$  is to be measured, is placed in contact with water at its triple point, it follows that

$$\theta_t = aX_t$$

$$\therefore a = \frac{\theta_t}{X_t} = \frac{273.16}{X_t}$$

Therefore  $\theta = aX = \frac{273.16}{X_t} \cdot X$

or  $\theta = 273.16 \frac{X}{X_t}$  (2.5)

The temperature of the triple point of water, which is an easily reproducible state, is now the *standard fixed point of thermometry*.

### 2.3 COMPARISON OF THERMOMETERS

Applying the above principle to the five thermometers listed in Table 2.1, the temperatures are given as follows:

- |                                       |                                    |
|---------------------------------------|------------------------------------|
| (a) Constant volume gas thermometer   | $\theta(P) = 273.16 \frac{P}{P_t}$ |
| (b) Constant pressure gas thermometer | $\theta(V) = 273.16 \frac{V}{V_t}$ |
| (c) Electric resistance thermometer   | $\theta(R) = 273.16 \frac{R}{R_t}$ |

(d) Thermocouple

$$\theta(\varepsilon) = 273.16 \frac{\varepsilon}{\varepsilon_t}$$

(e) Liquid-in-glass thermometer

$$\theta(L) = 273.16 \frac{L}{L_t}$$

If the temperature of a given system is measured simultaneously with each of the five thermometers, it is found that there is considerable difference among the readings. The smallest variation is, however, observed among different gas thermometers. That is why a gas is chosen as the standard thermometric substance.

## 2.4 IDEAL GAS

It has been established from experimental observations that the  $p - v - T$  behaviour of gases at a low pressure is closely given by the following relation

$$p\bar{v} = \bar{R}T \quad (2.6)$$

where  $\bar{R}$  is the universal gas constant, 8.3143 J/mol K and  $\bar{v}$  is the molar specific volume,  $\text{m}^3/\text{gmol}$ . (see Sec. 10.3.). Dividing Eq. (2.6) by the molecular weight  $\mu$ ,

$$pv = RT \quad (2.7)$$

where  $v$  is specific volume, in  $\text{m}^3/\text{kg}$ , and  $R$  is the characteristic gas constant. Substituting  $R = \bar{R}/\mu$  J/kg K, we get in terms of the total volume  $V$  of gas,

$$PV = n\bar{R}T$$

$$PV = mRT \quad (2.8)$$

where  $n$  is the number of moles and  $m$  is the mass of the gas. Equation (2.8) can be written for two states of the gas,

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2} \quad (2.9)$$

Equation (2.6), (2.7) or (2.8) is called the *ideal gas equation of state*. At very low pressure or density, all gases and vapours approach ideal gas behaviour.

## 2.5 GAS THERMOMETERS

A schematic diagram of a constant volume gas thermometer is given in Fig. 2.1. A small amount of gas is enclosed in bulb  $B$  which is in communication via the capillary tube  $C$  with one limb of the mercury manometer  $M$ . The other limb of the mercury manometer is open to the atmosphere and can be moved vertically to adjust the mercury levels so that the mercury just touches lip  $L$  of the capillary. The pressure in the bulb is used as a thermometric property and is given by

$$p = p_0 + \rho_M Zg$$

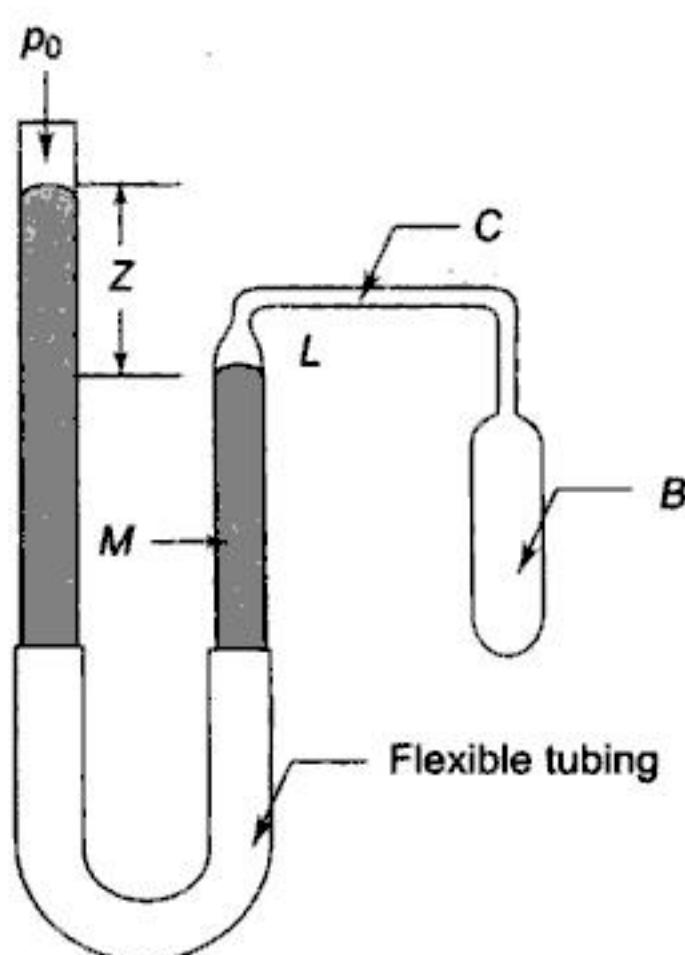
where  $p_0$  is the atmospheric pressure  $\rho_M$  is the density of mercury.

When the bulb is brought in contact with the system whose temperature is to be measured, the bulb, in course of time, comes in thermal equilibrium with the system.

The gas in the bulb expands, on being heated, pushing the mercury downward. The flexible limb of the manometer is then adjusted so that the mercury again touches the lip  $L$ . The difference in mercury level  $Z$  is recorded and the pressure  $p$  of the gas in the bulb is estimated. Since the volume of the trapped gas is constant, from the ideal gas equation,

$$\Delta T = \frac{V}{R} \Delta p \quad (2.10)$$

i.e. the temperature increase is proportional to the pressure increase.



**Fig. 2.1** Constant Volume Gas Thermometer

In a constant pressure gas thermometer, the mercury levels have to be adjusted to keep  $Z$  constant, and the volume of gas  $V$ , which would vary with the temperature of the system, becomes the thermometric property.

$$\therefore \Delta T = \frac{p}{R} \Delta V \quad (2.11)$$

i.e. the temperature increase is proportional to the observed volume increase. The constant volume gas thermometer is, however, mostly in use, since it is simpler in construction and easier to operate.

## 2.6 IDEAL GAS TEMPERATURE

Let us suppose that the bulb of a constant volume gas thermometer contains an amount of gas such that when the bulb is surrounded by water at its triple point, the pressure  $p$ , is 1000 mm Hg. Keeping the volume  $V$  constant, let the following procedure be conducted:

- (a) Surround the bulb with steam condensing at 1 atm, determine the gas pressure  $p$  and calculate

$$\theta = 273.16 \frac{p}{1000}$$

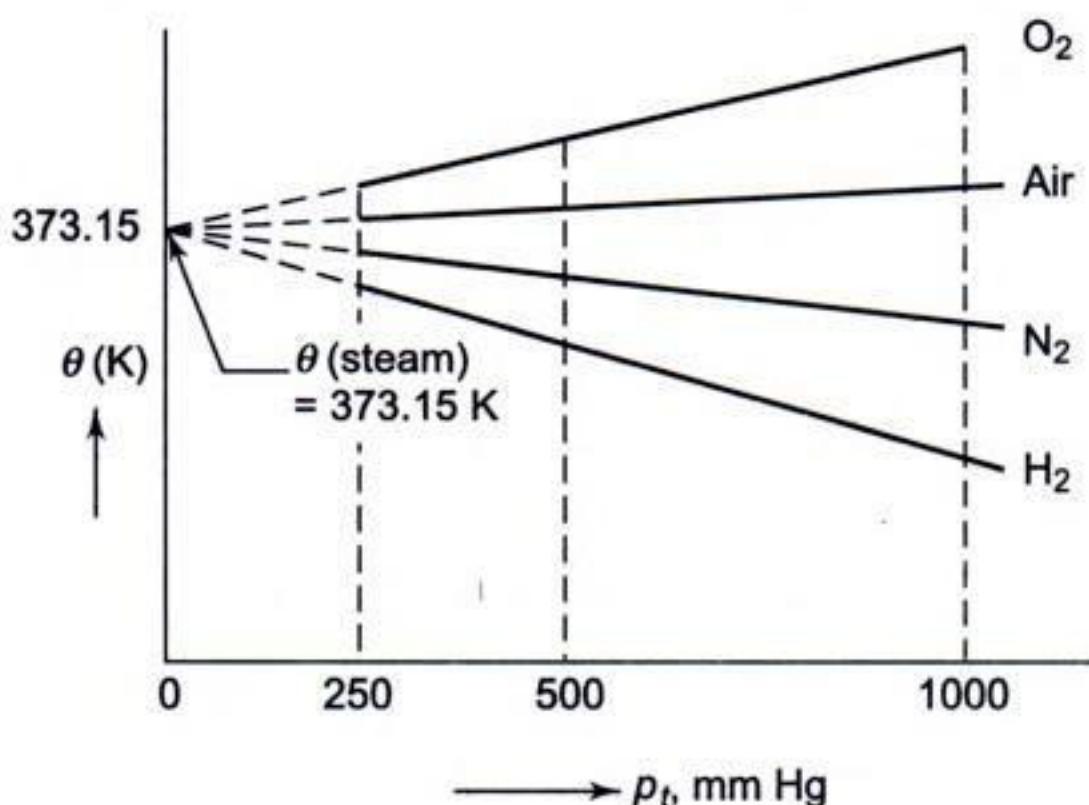
- (b) Remove some gas from the bulb so that when it is surrounded by water at its triple point, the pressure  $p$ , is 500 mm Hg. Determine the new value of  $p$  and then  $\theta$  for steam condensing at 1 atm.

$$\theta = 273.16 \frac{p}{500}$$

- (c) Continue reducing the amount of gas in the bulb so that  $p_t$  and  $p$  have smaller and smaller values, e.g.  $p_t$  having, say, 250 mm Hg, 100 mm Hg, and so on. At each value of  $p_t$ , calculate the corresponding  $\theta$ .
- (d) Plot  $\theta$  vs.  $p_t$  and extrapolate the curve to the axis where  $p_t = 0$ . Read from the graph

$$\lim_{p_t \rightarrow 0} \theta$$

The graph, as shown in Fig. 2.2, indicates that although the readings of a constant volume gas thermometer depend upon the nature of the gas, *all gases indicate the same temperature as  $p_t$  is lowered and made to approach zero.*



**Fig. 2.2** Ideal Gas Temperature for Steam Point

A similar series of tests may be conducted with a constant pressure gas thermometer. The constant pressure may first be taken to be 1000 mm Hg, then 500 mm Hg, etc. and at each value of  $p$ , the volumes of gas  $V$  and  $V_t$  may be recorded when the bulb is surrounded by steam condensing at 1 atm and the triple point of water, respectively. The corresponding value of  $\theta$  may be calculated from

$$\theta = 273.16 \frac{V}{V_t}$$

and  $\theta$  vs.  $p$  may be plotted, similar to Fig. 2.2. It is found from the experiments that all gases indicate the same value of  $\theta$  as  $p$  approaches zero.

Since a real gas, as used in the bulb, behaves as an ideal gas as pressure approaches zero (which would be explained later in Chapter 10), the *ideal gas temperature T* is defined by either of the two equations

$$T = 273.16 \lim_{p_t \rightarrow 0} \frac{p}{p_t}$$

$$p_t \rightarrow 0$$

$$= 273.16 \lim_{p_t \rightarrow 0} \frac{V}{V_t} \quad (2.12)$$

$p \rightarrow 0$

where  $\theta$  has been replaced by  $T$  to denote this particular temperature scale, the *ideal gas temperature scale*.

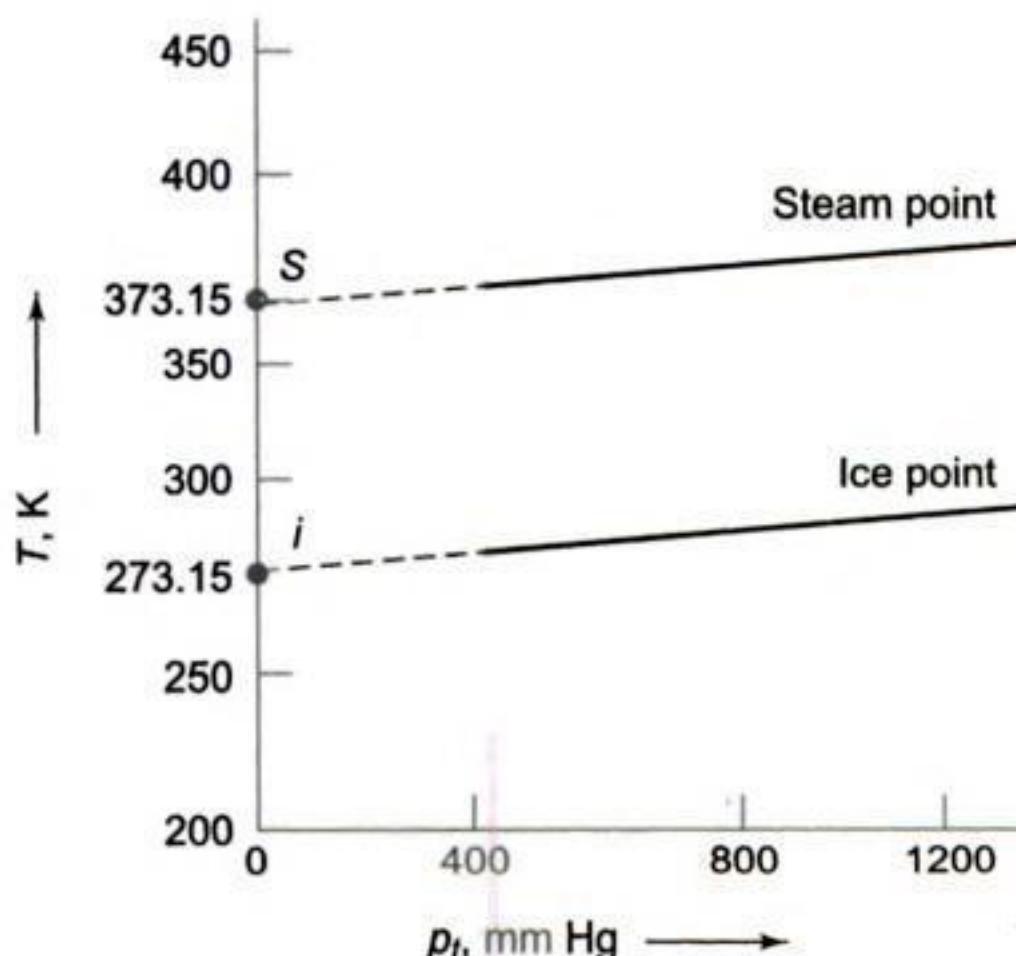


Fig. 2.3 Steam-point and Ice-point from Constant Volume Gas Thermometer

If  $p_s$  and  $p_t$  are the measured pressures at the steam point and the triple point respectively, one gets the value of the steam point temperature  $T_s$  as

$$T_s = 273.16 \lim_{p_t \rightarrow 0} \frac{p_s}{p_t} \quad (2.13)$$

which is equal to 373.15 K.

Similarly, the temperature  $T_i$  at the ice point is

$$T_i = 273.16 \lim_{p_t \rightarrow 0} \frac{p_i}{p_t} \quad (2.14)$$

which is equal to 273.15 K (Fig. 2.3).

Alternatively, if the ratio  $p_s/p_i$  is plotted against  $p_t$  with different gases in the bulb, one gets different curves as in Fig. 2.2. However, when extrapolated to zero pressure all curves converge, and the ratio  $p_s/p_i$  tends to a constant value giving

$$\frac{T_s}{T_i} = \lim_{p_t \rightarrow 0} \frac{p_s}{p_i} = 1.366099 \quad (2.15)$$

This value may be considered as a *universal constant*.

One may now decide to have a certain number of divisions between the steam point and the ice point, say 100 as in the Kelvin and Celsius scales so that

$$T_s - T_i = 100 \quad (2.16)$$

Solving Eqs. (2.15) and (2.16),

$$T_s = 373.15 \text{ K} \text{ and } T_i = 273.15 \text{ K}$$

## 2.7 CELSIUS TEMPERATURE SCALE

The Celsius temperature scale employs a degree of the same magnitude as that of the ideal gas scale, but its zero point is shifted, so that the Celsius temperature of the triple point of water is 0.01 degree Celsius or  $0.01^\circ\text{C}$ . If  $t$  denotes the Celsius temperature, then

$$t = T - 273.15^\circ$$

Thus the Celsius temperature  $t_s$  at which steam condenses at 1 atm. pressure

$$\begin{aligned} t_s &= T_s - 273.15^\circ \\ &= 373.15 - 273.15 = 100.00^\circ\text{C} \end{aligned}$$

Similar measurements for ice points show this temperature on the Celsius scale to be  $0.00^\circ\text{C}$ . The only Celsius temperature which is fixed by definition is that of the triple point.

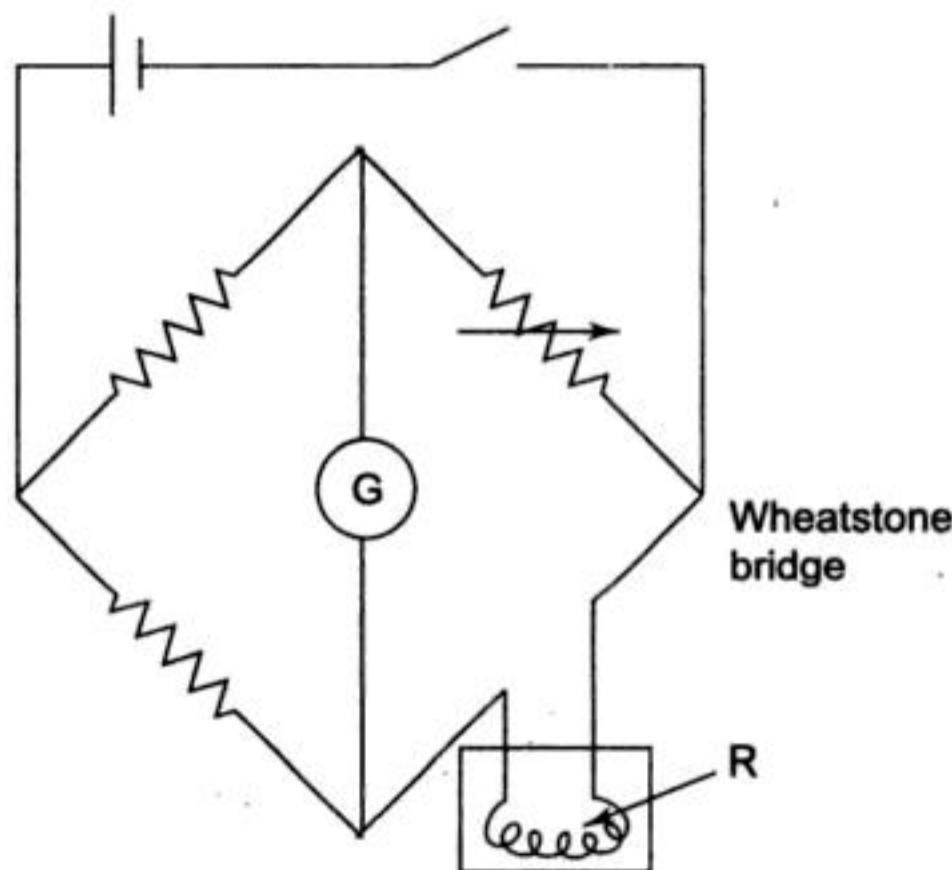
## 2.8 ELECTRICAL RESISTANCE THERMOMETER

In the resistance thermometer (Fig. 2.4) the change in resistance of a metal wire due to its change in temperature is the thermometric property. The wire, frequently platinum, may be incorporated in a Wheatstone bridge circuit. The platinum resistance thermometer measures temperature to a high degree of accuracy and sensitivity, which makes it suitable as a standard for the calibration of other thermometers.

In a restricted range, the following quadratic equation is often used

$$R = R_0 (1 + At + Bt^2)$$

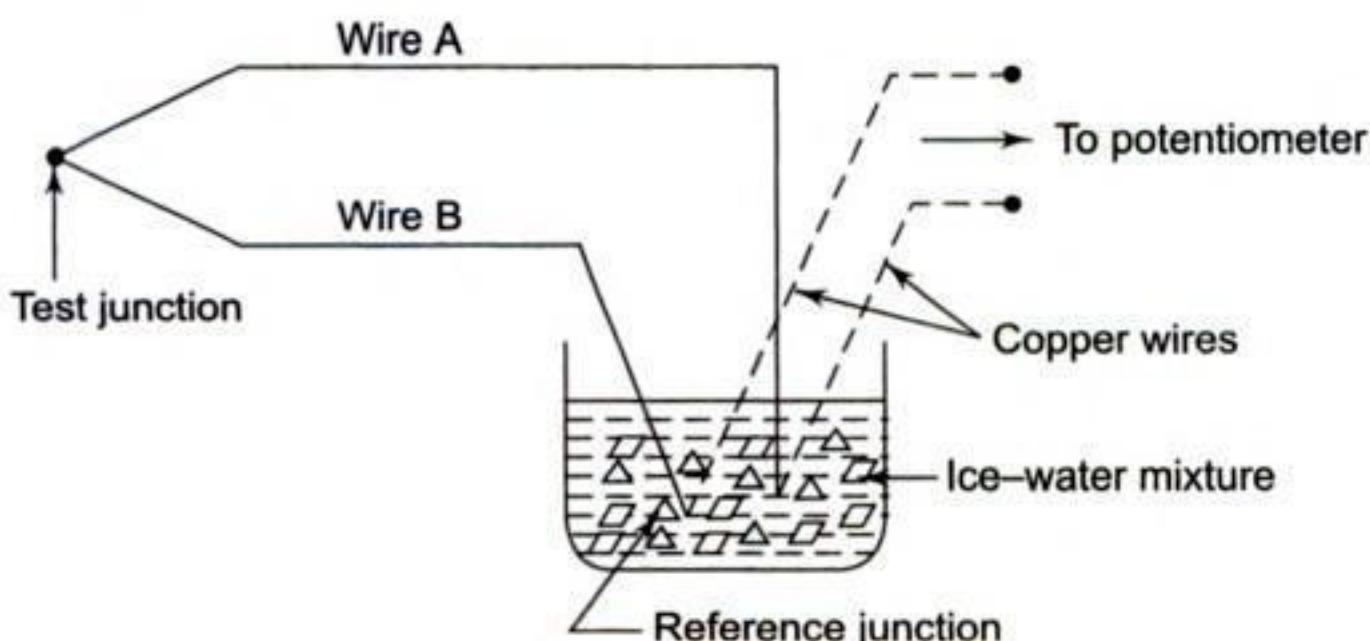
where  $R_0$  is the resistance of the platinum wire when it is surrounded by melting ice and  $A$  and  $B$  are constants.



**Fig. 2.4** Resistance Thermometer

## 2.9 THERMOCOUPLE

A thermocouple circuit made up from joining two wires A and B made of dissimilar metals is shown in Fig. 2.5. Due to the Seebeck effect, a net e.m.f. is generated in the circuit which depends on the difference in temperature between the hot and cold junctions and is, therefore, a thermometric property of the circuit. This e.m.f. can be measured by a microvoltmeter to a high degree of accuracy. The choice of metals depends largely on the temperature range to be investigated, and copper-constantan, chromel-alumel and platinum-platinum-rhodium are typical combinations in use.



**Fig. 2.5 Thermocouple**

A thermocouple is calibrated by measuring the thermal e.m.f. at various known temperatures, the reference junction being kept at 0°C. The results of such measurements on most thermocouples can usually be represented by a cubic equation of the form

$$\varepsilon = a + bt + ct^2 + dt^3$$

where  $\varepsilon$  is the thermal e.m.f. and the constants  $a, b, c$  and  $d$  are different for each thermocouple.

The advantage of a thermocouple is that it comes to thermal equilibrium with the system, whose temperature is to be measured, quite rapidly, because its mass is small.

## 2.10 INTERNATIONAL PRACTICAL TEMPERATURE SCALE

An international temperature scale was adopted at the Seventh General Conference on Weights and Measures held in 1927. It was not to replace the Celsius or ideal gas scales, but to provide a scale that could be easily and rapidly used to calibrate scientific and industrial instruments. Slight refinements were incorporated into the scale in revisions adopted in 1948, 1954, 1960 and 1968. The international practical scale agrees with the Celsius scale at the defining fixed points listed in Table 2.2. The temperature interval from the oxygen point to the gold point is divided into three main parts, as given below.

**Table 2.2 Temperatures of Fixed Points**

	Temperature °C
Normal boiling point of oxygen	– 182.97
Triple point of water (standard)	+ 0.01
Normal boiling point of water	100.00
Normal boiling point of sulphur	444.60
(Normal melting point of zinc-suggested as an alternative to the sulphur point)	419.50
Normal melting point of antimony	630.50
Normal melting point of silver	960.80
Normal melting point of gold	1063.00

**(a) From 0 to 660°C** A platinum resistance thermometer with a platinum wire whose diameter must lie between 0.05 and 0.20 mm is used, and the temperature is given by the equation

$$R = R_0 (1 + At + Bt^2)$$

where the constants  $R_0$ , A, and B are computed by measurements at the ice point, steam point, and sulphur point.

**(b) From –190 to 0°C** The same platinum resistance thermometer is used, and the temperature is given by

$$R = R_0 [1 + At + Bt^2 + C(t - 100)t^3]$$

where  $R_0$ , A and B are the same as before, and C is determined from a measurement at the oxygen point.

**(c) From 660 to 1063°C** A thermocouple, one wire of which is made of platinum and the other of an alloy of 90% platinum and 10% rhodium, is used with one junction at 0°C. The temperature is given by the formula

$$\varepsilon = a + bt + ct^2$$

where a, b, and c are computed from measurements at the antimony point, silver point, and gold point. The diameter of each wire of the thermocouple must lie between 0.35 and 0.65 mm.

An optical method is adopted for measuring temperatures higher than the gold point. The intensity of radiation of any convenient wavelength is compared with the intensity of radiation of the same wavelength emitted by a black body at the gold point. The temperature is then determined with the help of Planck's law of thermal radiation.

### Solved Examples

**Example 2.1** Two mercury-in-glass thermometers are made of identical materials and are accurately calibrated at 0°C and 100°C. One has a tube of

constant diameter, while the other has a tube of conical bore, ten per cent greater in diameter at 100°C than at 0°C. Both thermometers have the length between 0 and 100 subdivided uniformly. What will be the straight bore thermometer read in a place where the conical bore thermometer reads 50°C?

**Solution** The volume of mercury in the tube at  $t^\circ\text{C}$ ,  $V_t$ , is given by

$$V_t = V_0 [1 + \beta(t - t_0)]$$

where  $V_0$  is the volume of mercury at 0°C,  $\beta$  is the coefficient of volume expansion of mercury, and  $t_0$  is the ice point temperature which is 0°C. The volume change of glass is neglected.

Therefore  $V_t - V_0 = \beta V_0 t$

The temperature  $t$  is thus a linear function of volume change of mercury ( $V_t - V_0$ ).

Therefore  $\Delta V_{0-100} = \beta V_0 \cdot 100$

$$\Delta V_{0-50} = \beta V_0 \cdot 50$$

$$\therefore \frac{\Delta V_{0-50}}{\Delta V_{0-100}} = \frac{1}{2}$$

i.e. at 50°C, the volume of mercury will be half of that at 100°C, for the straight bore thermometer [Fig. 2.6(a)].

But if the bore is conical [Fig. 2.6 (b)], mercury will fill up the volume ACDB, which is less than half of the mercury volume at 100°C, i.e. volume AEFB. Let  $t$  be the true temperature when mercury rises half the length of the conical tube (the apparent temperature being 50°C). Let EA and FB be extended to meet at G. Let  $l$  represent the length of the thermometers and  $l'$  the vertical height of the cone ABG, as shown in the figure. Now,

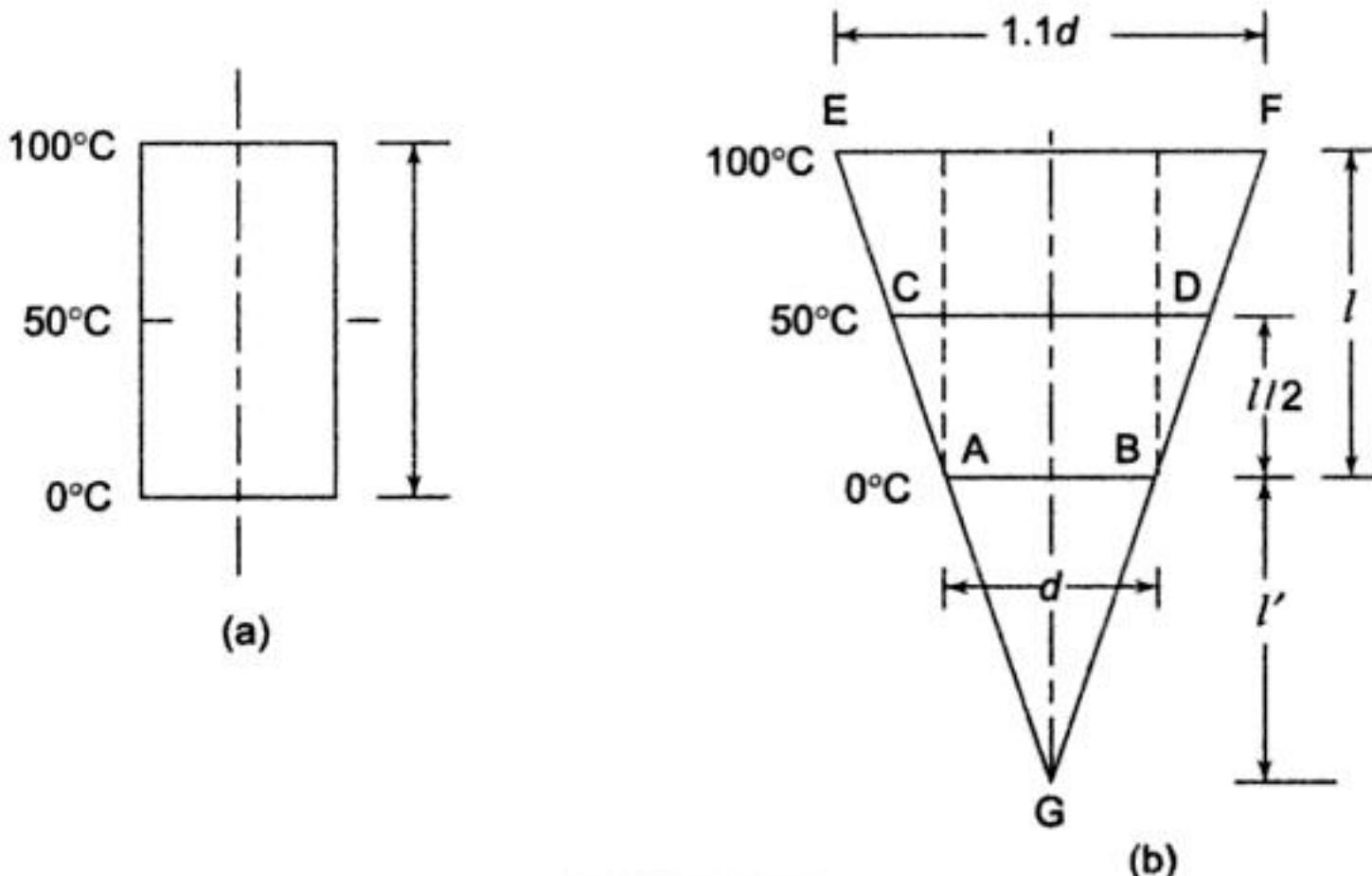


Fig. 2.6

$$\frac{l'}{l+l'} = \frac{d}{1.1d} = \frac{1}{1.1}$$

$$l' = 10$$

and  $\frac{l'}{l'+l/2} = \frac{d}{CD}$

$$\therefore CD = \frac{10.5}{10} d = 1.05d$$

Again

$$\Delta V_{0-100} = V_0 \cdot \beta \cdot 100$$

$$\Delta V_{0-t} = V_0 \beta t$$

$$\frac{\Delta V_{0-t}}{\Delta V_{0-100}} = \frac{t}{100}$$

or  $\frac{\text{Volume ACDB}}{\text{Volume AEFB}} = \frac{t}{100}$

or  $\frac{\frac{1}{3}\frac{\pi}{4}(1.05d)^2 \times 10.5l - \frac{1}{3}\frac{\pi}{4}d^2 \cdot 10l}{\frac{1}{3}\frac{\pi}{4}(1.1d)^2 \times 11l - \frac{1}{3}\frac{\pi}{4}d^2 \cdot 10l} = \frac{t}{100}$

or  $\frac{1.05 \times 1.05 \times 10.5 - 10}{1.1 \times 1.1 \times 11 - 10} = \frac{t}{100}$

$\therefore t = \frac{1.58}{3.31} \times 100 = 47.7^\circ\text{C}$

**Example 2.2** The e.m.f. in a thermocouple with the test junction at  $t^\circ\text{C}$  on gas thermometer scale and reference junction at ice point is given by

$$\varepsilon = 0.20 t - 5 \times 10^{-4} t^2 \text{ mV}$$

The millivoltmeter is calibrated at ice and steam points. What will this thermometer read in a place where the gas thermometer reads  $50^\circ\text{C}$ ?

**Solution** At ice point, when  $t = 0^\circ\text{C}$ ,  $\varepsilon = 0 \text{ mV}$

At steam point, when  $t = 100^\circ\text{C}$ ,  $\varepsilon = 0.20 \times 100 - 5 \times 10^{-4} \times (100)^2 = 15 \text{ mV}$

At  $t = 50^\circ\text{C}$ ,  $\varepsilon = 0.20 \times 50 - 5 \times 10^{-4} (50)^2 = 8.75 \text{ mV}$

When the gas thermometer reads  $50^\circ\text{C}$ , the thermocouple will read

$$\frac{100}{15} \times 8.75, \text{ or } 58.33^\circ\text{C}$$

**Example 2.3** A platinum resistance thermometer has a resistance of 2.8 ohm at 0°C and 3.8 ohm at 100°C. Calculate the temperature when the resistance indicated is 5.8 ohm.

**Solution** Let  $R = R_0 (1 + \alpha t)$   
where  $R_0$  is the resistance at 0°C.

Therefore,  $R_0 = 2.8$  ohm

$$R_{100} = 3.8 \text{ ohm} = 2.8 (1 + \alpha \times 100)$$

$$\therefore \alpha = \left( \frac{3.8}{2.8} - 1 \right) \times 10^{-2} = 0.357 \times 10^{-2}$$

When  $R = 5.8$  ohm,

$$5.8 = 2.8 (1 + 0.357 \times 10^{-2} t)$$

or,

$$t = \left( \frac{5.8}{2.8} - 1 \right) \times \frac{100}{0.357} = 300^\circ\text{C}.$$

## Review Questions

- 2.1 What is the zeroth law of thermodynamics?
- 2.2 Define thermometric property.
- 2.3 What is a thermometer?
- 2.4 What is a fixed point?
- 2.5 How many fixed points were used prior to 1954? What are these?
- 2.6 What is the standard fixed point in thermometry? Define it.
- 2.7 Why is a gas chosen as the standard thermometric substance?
- 2.8 What is an ideal gas?
- 2.9 What is the difference between the universal gas constant and a characteristic gas constant?
- 2.10 What is a constant volume gas thermometer? Why is it preferred to a constant pressure gas thermometer?
- 2.11 What do you understand by the ideal gas temperature scale?
- 2.12 How can the ideal gas temperature for the steam point be measured?
- 2.13 What is the Celsius temperature scale?
- 2.14 What is the advantage of a thermocouple in temperature measurement?
- 2.15 How does the resistance thermometer measure temperature?
- 2.16 What is the need of the international practical temperature scale?

## Problems

- 2.1 The limiting value of the ratio of the pressure of gas at the steam point and at the triple point of water when the gas is kept at constant volume is found to be 1.36605. What is the ideal gas temperature of the steam point? Ans. 100°C

- 2.2 In a constant volume gas thermometer the following pairs of pressure readings were taken at the boiling point of water and the boiling point of sulphur, respectively:

Water b.p.	50.0	100	200	300
Sulphur b.p.	96.4	193	387	582

The numbers are the gas pressures, mm Hg, each pair being taken with the same amount of gas in the thermometer, but the successive pairs being taken with different amounts of gas in the thermometer. Plot the ratio of  $S_{\text{b.p.}} : H_2O_{\text{b.p.}}$  against the reading at the water boiling point, and extrapolate the plot to zero pressure at the water boiling point. This gives the ratio of  $S_{\text{b.p.}} : H_2O_{\text{b.p.}}$  on a gas thermometer operating at zero gas pressure, i.e., an ideal gas thermometer. What is the boiling point of sulphur on the gas scale, from your plot? Ans. 445°C

- 2.3 The resistance of a platinum wire is found to be 11,000 ohms at the ice point, 15.247 ohms at the steam point, and 28.887 ohms at the sulphur point. Find the constants A and B in the equation

$$R = R_0 (1 + At + Bt^2)$$

and plot  $R$  against  $t$  in the range 0 to 660°C.

- 2.4 When the reference junction of a thermocouple is kept at the ice point and the test junction is at the Celsius temperature  $t$ , and e.m.f.  $e$  of the thermocouple is given by the equation

$$e = at + bt^2$$

where  $a = 0.20 \text{ mV/deg}$ , and  $b = -5.0 \times 10^{-4} \text{ mV/deg}^2$

- (a) Compute the e.m.f. when  $t = -100^\circ\text{C}, 200^\circ\text{C}, 400^\circ\text{C}$ , and  $500^\circ\text{C}$ , and draw graph of  $e$  against  $t$  in this range.
- (b) Suppose the e.m.f.  $e$  is taken as a thermometric property and that a temperature scale  $t^*$  is defined by the linear equation.

$$t^* = a' e + b'$$

and that  $t^* = 0$  at the ice point and  $t^* = 100$  at the steam point. Find the numerical values of  $a'$  and  $b'$  and draw a graph of  $e$  against  $t^*$ .

- (c) Find the values of  $t^*$  when  $t = -100^\circ\text{C}, 200^\circ\text{C}, 400^\circ\text{C}$ , and  $500^\circ\text{C}$ , and draw a graph of  $t^*$  against  $t$ .
- (d) Compare the Celsius scale with the  $t^*$  scale.

- 2.5 The temperature  $t$  on a thermometric scale is defined in terms of a property  $K$  by the relation

$$t = a \ln K + b$$

where  $a$  and  $b$  are constants.

The values of  $K$  are found to be 1.83 and 6.78 at the ice point and the steam point, the temperatures of which are assigned the numbers 0 and 100 respectively. Determine the temperature corresponding to a reading of  $K$  equal to 2.42 on the thermometer. Ans. 21.346°C

- 2.6 The resistance of the windings in a certain motor is found to be 80 ohms at room temperature ( $25^\circ\text{C}$ ). When operating at full load under steady state conditions, the motor is switched off and the resistance of the windings, immediately measured again, is found to be 93 ohms. The windings are made of copper whose resistance at temperature  $t^\circ\text{C}$  is given by

$$R_t = R_0 [1 + 0.00393 t]$$

where  $R_0$  is the resistance at  $0^\circ\text{C}$ . Find the temperature attained by the coil during full load.

Ans.  $70.41^\circ\text{C}$

- 2.7 A new scale  $N$  of temperature is divided in such a way that the freezing point of ice is  $100^\circ\text{N}$  and the boiling point is  $400^\circ\text{N}$ . What is the temperature reading on this new scale when the temperature is  $150^\circ\text{C}$ ? At what temperature both the Celsius and the new temperature scale reading would be the same?

Ans.  $550^\circ\text{N}, -50^\circ\text{C}$ .

- 2.8 A platinum wire is used as a resistance thermometer. The wire resistance was found to be 10 ohm and 16 ohm at ice point and steam point respectively, and 30 ohm at sulphur boiling point of  $444.6^\circ\text{C}$ . Find the resistance of the wire at  $500^\circ\text{C}$ , if the resistance varies with temperature by the relation.

$$R = R_0 (1 + \alpha t + \beta t^2)$$

Ans. 31.3 ohm

# 3

## CHAPTER

# Work and Heat Transfer

A closed system and its surroundings can interact in two ways: (a) by work transfer, and (b) by heat transfer. These may be called *energy interactions* and these bring about changes in the properties of the system. Thermodynamics mainly studies these energy interactions and the associated property changes of the system.

### 3.1 WORK TRANSFER

Work is one of the basic modes of energy transfer. In mechanics the action of a force on a moving body is identified as work. A force is a means of transmitting an effect from one body to another. But a force itself never produces a physical effect except when coupled with motion and hence it is not a form of energy. An effect such as the raising of a weight through a certain distance can be performed by using a small force through a large distance or a large force through a small distance. The product of force and distance is the same to accomplish the same effect. In mechanics work is defined as:

*The work is done by a force as it acts upon a body moving in the direction of the force.*

The action of a force through a distance (or of a torque through an angle) is called *mechanical work* since other forms of work can be identified, as discussed later. The product of the force and the distance moved parallel to the force is the magnitude of mechanical work.

In thermodynamics, work transfer is considered as occurring between the system and the surroundings. *Work is said to be done by a system if the sole effect on things external to the system can be reduced to the raising of a weight.* The weight may not actually be raised, but the net effect external to the system would be the raising of a weight. Let us consider the battery and the motor in Fig. 3.1 as a system. The motor is driving a fan. The system is doing work upon the surroundings. When the fan is replaced by a pulley and a weight, as shown in Fig. 3.2, the weight may be raised with the pulley driven by the motor. The sole effect on things external to the system is then the raising of a weight.

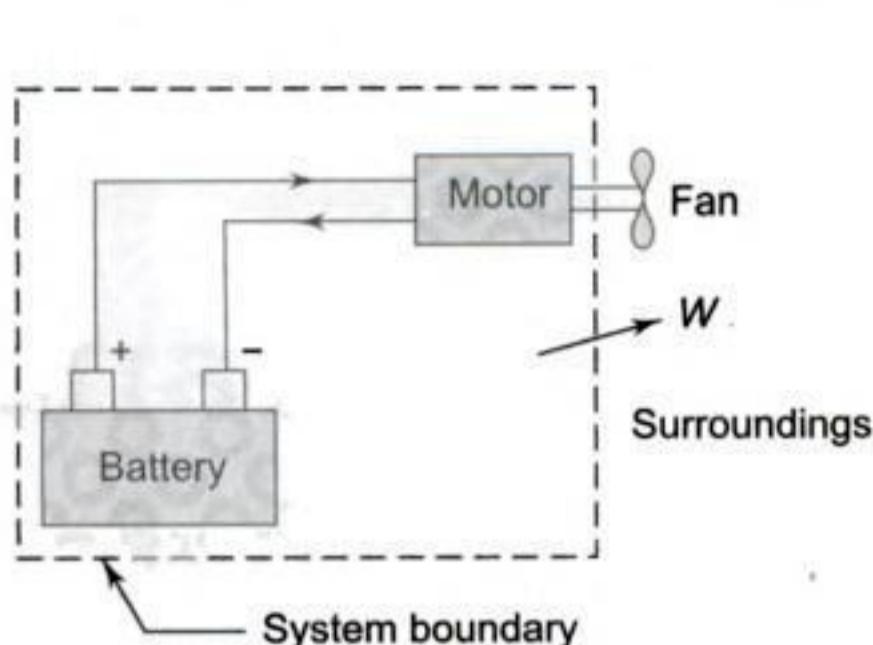


Fig. 3.1 *Battery-motor System Driving a Fan*

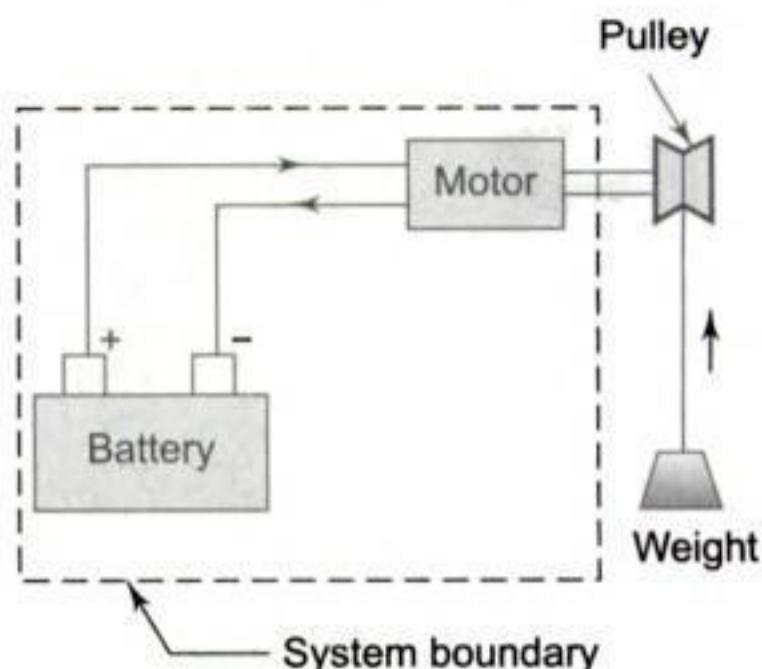
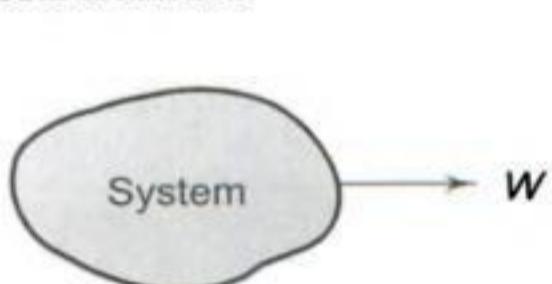
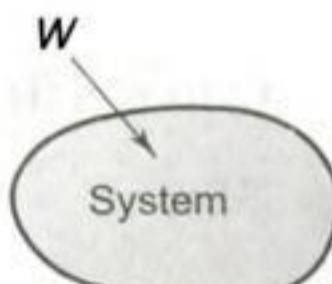


Fig. 3.2 *Work Transfer from a System*

*When work is done by a system, it is arbitrarily taken to be positive, and when work is done on a system, it is taken to be negative (Fig. 3.3). The symbol W is used for work transfer.*



(a)  $W$  is positive



(b)  $W$  is negative

Fig. 3.3 *Work Interaction between a System and the Surroundings*

The unit of work is N.m or Joule [1 Nm = 1 Joule]. The rate at which work is done by, or upon, the system is known as *power*. The unit of power is J/s or watt.

Work is one of the forms in which a system and its surroundings can interact with each other. There are various types of work transfers which can get involved between them.

### 3.2 $pdV$ -WORK OR DISPLACEMENT WORK

Let the gas in the cylinder (Fig. 3.4) be a system having initially the pressure  $p_1$  and volume  $V_1$ . The system is in thermodynamic equilibrium, the state of which is described by the coordinates  $p_1$ ,  $V_1$ . The piston is the only boundary which moves due to gas pressure. Let the piston move out to a new final position 2, which is also a thermodynamic equilibrium state specified by pressure  $p_2$  and volume  $V_2$ . At any intermediate point in the travel of the piston, let the pressure be  $p$  and the volume  $V$ . This must also be an equilibrium state,

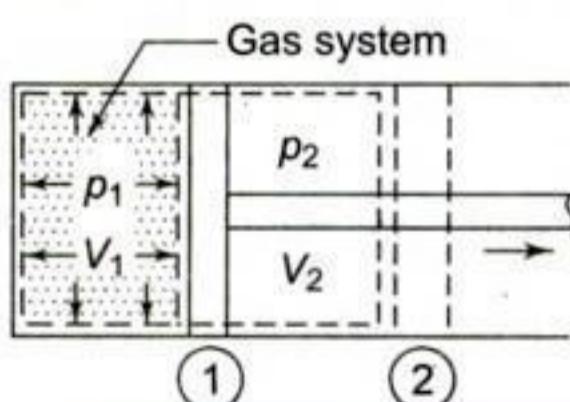


Fig. 3.4  *$pdV$  Work*

since macroscopic properties  $p$  and  $V$  are significant only for equilibrium states. When the piston moves an infinitesimal distance  $dl$ , and if 'a' be the area of the piston, the force  $F$  acting on the piston  $F = p.a.$  and the infinitesimal amount of work done by the gas on the piston

$$dW = F \cdot dl = padl = pdV \quad (3.1)$$

where  $dV = adl =$  infinitesimal displacement volume. The differential sign in  $dW$  with the line drawn at the top of it will be explained later.

When the piston moves out from position 1 to position 2 with the volume changing from  $V_1$  to  $V_2$ , the amount of work  $W$  done by the system will be

$$W_{1-2} = \int_{V_1}^{V_2} pdV$$

The magnitude of the work done is given by the area under the path 1–2, as shown in Fig. 3.5. Since  $p$  is at all times a thermodynamic coordinate, all the states passed through by the system as the volume changes from  $V_1$  to  $V_2$  must be equilibrium states, and the path 1–2 *must be quasi-static*. The piston moves infinitely slowly so that every state passed through is an equilibrium state. *The integration  $\int pdV$  can be performed only on a quasi-static path.*

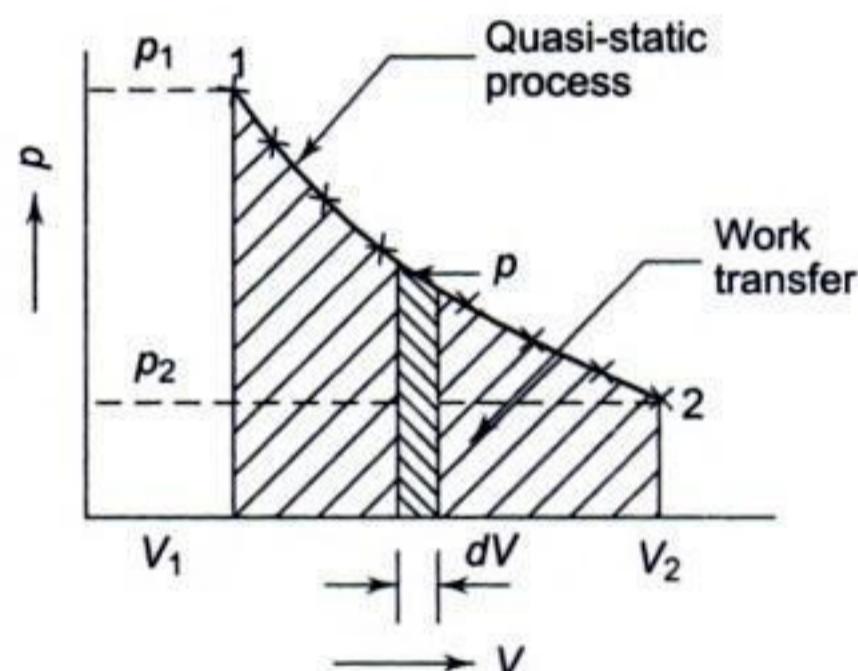


Fig. 3.5 Quasi-Static  $pdV$  Work

### 3.2.1 Path Function and Point Function

With reference to Fig. 3.6, it is possible to take a system from state 1 to state 2 along many quasi-static paths, such as A, B or C. Since the area under each curve represents the work for each process, the amount of work involved in each case is not a function of the end states of the process, and it depends on the path the system follows in going from state 1 to state 2. For this reason, work is called a *path function*, and  $dW$  is an *inexact or imperfect differential*.

Thermodynamic properties are *point functions*, since for a given state, there is a definite value for each property. The change in a thermodynamic property of a system in a change of state is independent of the path the system follows during the change of state, and depends only on the initial and final states of the system. The differentials of point functions are *exact or perfect differentials*, and the integration is simply

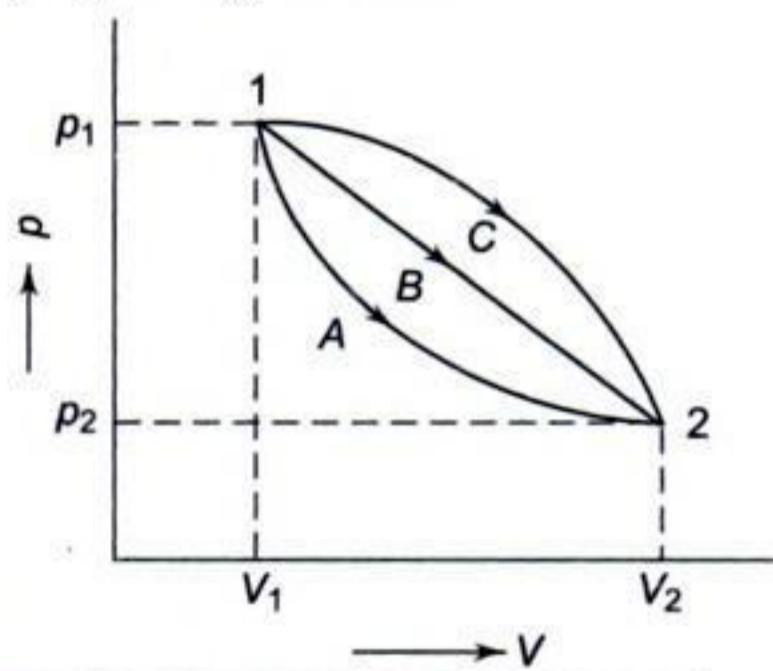


Fig. 3.6 Work—A Path Function

$$\int_{V_1}^{V_2} dV = V_2 - V_1$$

The change in volume thus depends only on the end states of the system irrespective of the path the system follows.

On the other hand, work done in a quasi-static process between two given states depends on the path followed.

$$\int_1^2 dW \neq W_2 - W_1$$

Rather,

$$\int_1^2 dW = W_{1-2} \quad \text{or} \quad _1 W_2$$

To distinguish an inexact differential  $dW$  from an exact differential  $dV$  or  $dp$  the differential sign is being cut by a line at its top.

From Eq. (3.1),

$$dV = \frac{1}{p} dW \quad (3.2)$$

Here,  $1/p$  is called the *integrating factor*. Therefore, an inexact differential  $dW$  when multiplied by an integrating factor  $1/p$  becomes an exact differential  $dV$ .

For a cyclic process, the initial and final states of the system are the same, and hence, the change in any property is zero, i.e.

$$\oint dV = 0, \quad \oint dp = 0, \quad \oint dT = 0 \quad (3.3)$$

where the symbol  $\oint$  denotes the cyclic integral for the closed path. Therefore, *the cyclic integral of a property is always zero*.

### 3.2.2 $p dV$ -Work in Various Quasi-Static Processes

(a) Constant pressure process (Fig. 3.7) (isobaric or isopiestic process)

$$W_{1-2} = \int_{V_1}^{V_2} p dV = p(V_2 - V_1) \quad (3.4)$$

(b) Constant volume process (Fig. 3.8) (isochoric process)

$$W_{1-2} = \int p dV = 0 \quad (3.5)$$

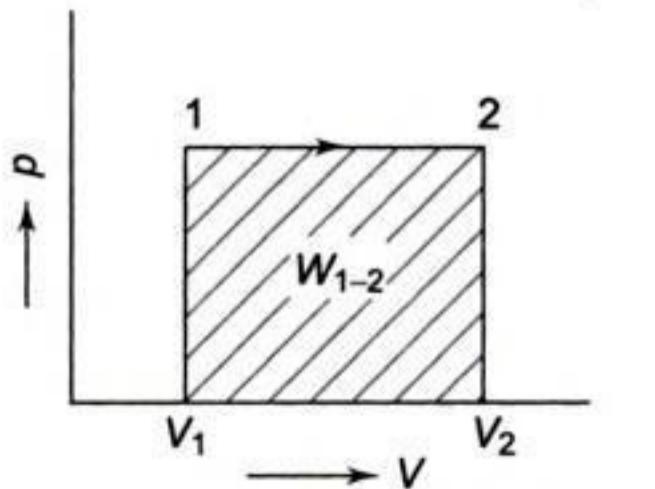


Fig. 3.7 Constant Pressure Process

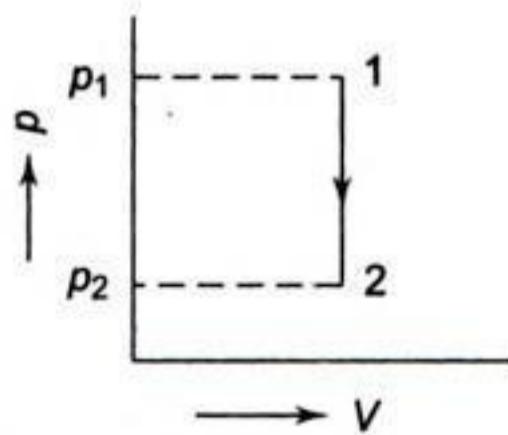


Fig. 3.8 Constant Volume Process

(c) Process in which  $pV = C$  (Fig. 3.9)

$$\therefore W_{1-2} = \int_{V_1}^{V_2} p dV \quad pV = p_1 V_1 = C$$

$$p = \frac{(p_1 V_1)}{V}$$

$$W_{1-2} = p_1 V_1 \int_{V_1}^{V_2} \frac{dV}{V} = p_1 V_1 \ln \frac{V_2}{V_1}$$

$$= p_1 V_1 \ln \frac{p_1}{p_2} \quad (3.6)$$

(d) Process in which  $pV^n = C$ , where  $n$  is a constant (Fig. 3.10).

$$pV^n = p_1 V_1^n = p_2 V_2^n = C$$

$$\therefore p = \frac{(p_1 V_1^n)}{V^n}$$

$$\therefore W_{1-2} = \int_{V_1}^{V_2} p dV = \int_{V_1}^{V_2} \frac{p_1 V_1^n}{V^n} \cdot dV = (p_1 V_1^n) \left[ \frac{V^{-n+1}}{-n+1} \right]_{V_1}^{V_2}$$

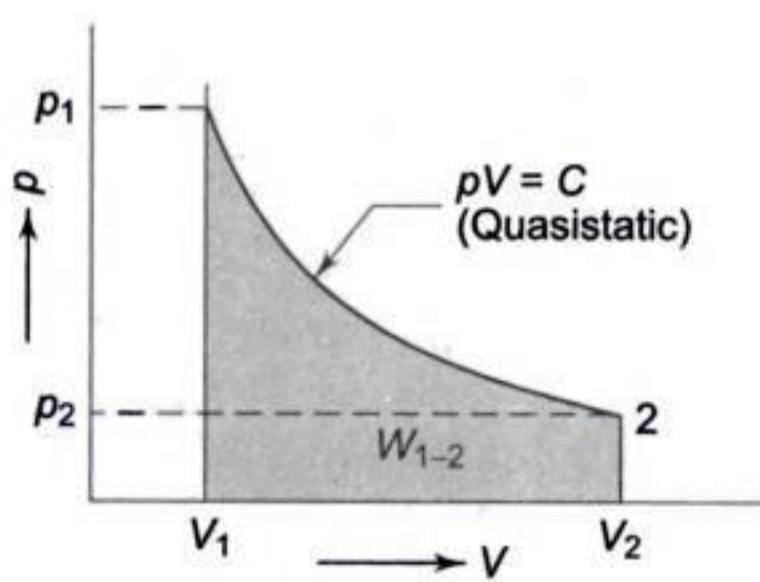


Fig. 3.9      *Process in Which  
pV = Constant*

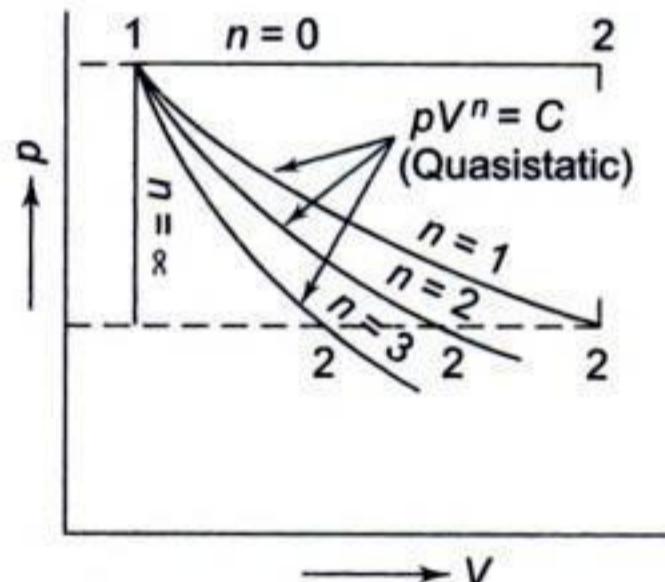


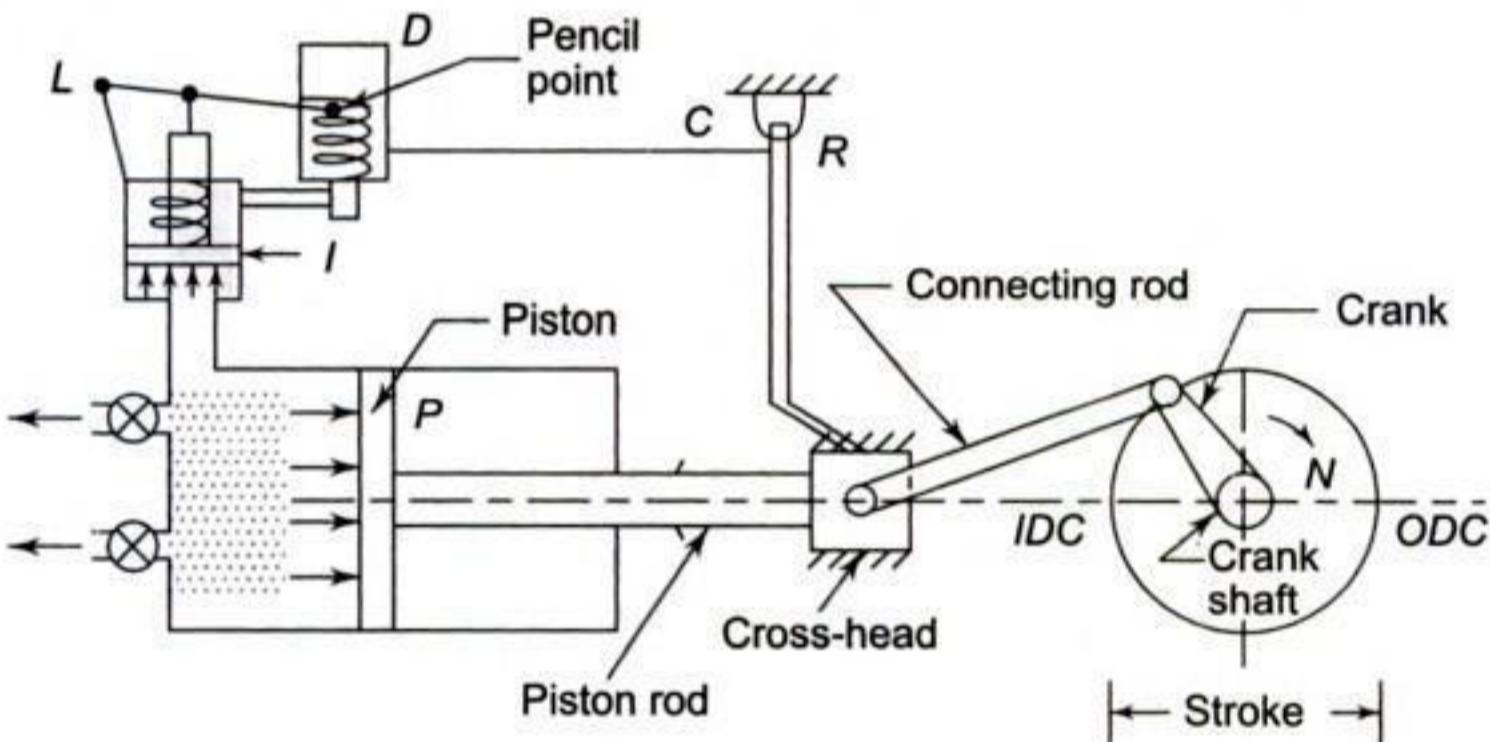
Fig. 3.10      *Process in which  
pV^n = Constant*

$$= \frac{p_1 V_1^n}{1-n} (V_2^{1-n} - V_1^{1-n}) = \frac{p_2 V_2^n \times V_2^{1-n} - p_1 V_1^n \times V_1^{1-n}}{1-n}$$

$$= \frac{p_1 V_1 - p_2 V_2}{n-1} = \frac{p_1 V_1}{n-1} \left[ 1 - \left( \frac{p_2}{p_1} \right)^{n-1/n} \right] \quad (3.7)$$

### 3.3 INDICATOR DIAGRAM

An indicator diagram is a trace made by a recording pressure gauge, called the indicator, attached to the cylinder of a reciprocating engine. This represents the work done in one engine cycle. Figure 3.11 shows a typical engine indicator.



**Fig. 3.11** Engine Indicator

The same gas pressure acts on both the engine piston  $P$  and the indicator piston  $I$ . The indicator piston is loaded by a spring and it moves in direct proportion to the change in pressure. The motion of the indicator piston causes a pencil held at the end of the linkage  $L$  to move upon a strip of paper wrapped around drum  $D$ . The drum is rotated about its axis by cord  $C$ , which is connected through a reducing motion  $R$  to the piston  $P$  of the engine. The surface of drum  $D$  moves horizontally under the pencil while the pencil moves vertically over the surface and a plot of pressure upon the piston vs. piston travel is obtained.

Before tracing the final indicator diagram, a pressure reference line is recorded by subjecting the indicator to the atmosphere and tracing a line at a constant pressure of one atmosphere.

The area of the indicator diagram represents the magnitude of the net work done by the system in one engine cycle. The area under the path 1–2 represents work done by the system and the area under the path 2–1 represents work done upon the system (Fig. 3.12). The area of the diagram,  $a_d$ , is measured by means of a planimeter, and the length of the diagram,  $l_d$ , is also measured. The *mean effective pressure* (m.e.p.)  $p_m$  is defined in the following way

$$p_m = \frac{a_d}{l_d} \times K$$

where  $K$  is the indicator spring constant ( $\text{N}/\text{cm}^2 \times \text{cm travel}$ ). Work done in one engine cycle

$$= (p_m \cdot A) L$$

where

$A$  = cross-sectional area of the cylinder

$$= \frac{\pi}{4} D^2, \text{ where } D \text{ is the cylinder diameter}$$

and

$L$  = stroke of piston, or length of cylinder.

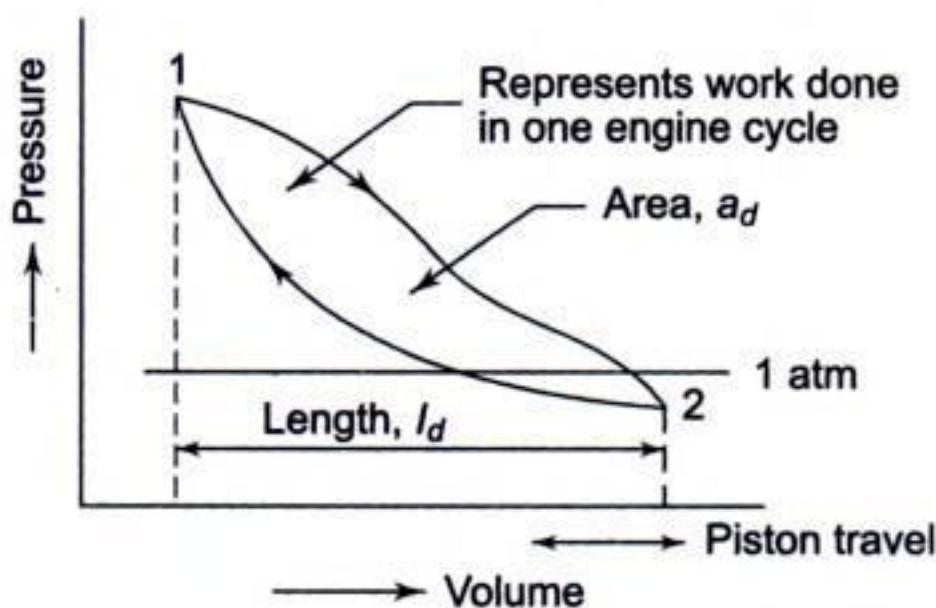


Fig. 3.12 Indicator Diagram

Let  $N$  be the revolutions per minute (r.p.m.) of the crankshaft. In a two-stroke cycle, the engine cycle is completed in two strokes of the piston or in one revolution of the crankshaft. In a four-stroke cycle, the engine cycle is completed in four strokes of the piston or two revolutions of the crankshaft.

For a two-stroke engine, work done in one minute =  $p_m A L N$ , and for a four-stroke engine, work done in one minute =  $p_m A L N / 2$ .

The power developed inside the cylinder of the engine is called *indicated power* (IP),

$$\therefore IP = \frac{p_m A L \left( N \text{ or } \frac{N}{2} \right) n}{60} \text{ kW} \quad (3.8)$$

where  $p_m$  is in kPa and  $n$  is the number of cylinders in the engine.

The power available at the crankshaft is always less than this value (IP) due to friction, etc. and is called the *brake power* (BP) or *shaft power* (SP). If  $w$  is the angular velocity of the crankshaft in radian/sec, then

$$BP = T \omega \quad (3.9)$$

where  $T$  is the torque transmitted to the crankshaft in mN.

$$\therefore BP = \frac{2\pi TN}{60} \quad (3.10)$$

where  $N$  is the number of revolutions per minute (rpm).

The mechanical efficiency of the engine,  $\eta_{\text{mech}}$ , is defined as

$$\eta_{\text{mech}} = \frac{BP}{IP} \quad (3.11)$$

An engine is said to be *double-acting*, if the working fluid is made to work on both sides of the piston. Such an engine theoretically develops twice the amount of work developed in a single-acting engine. Most reciprocating steam engines are double-acting, and so are many marine diesel engines. Internal combustion engines for road transport are always single-acting.

### 3.4 OTHER TYPES OF WORK TRANSFER

There are forms of work other than  $pdV$  or displacement work. The following are the additional types of work transfer which may get involved in system-surroundings interactions.

**(a) Electrical Work** When a current flows through a resistor (Fig. 3.13), taken as a system, there is work transfer into the system. This is because the current can drive a motor, the motor can drive a pulley and the pulley can raise a weight.

The current flow,  $I$ , in amperes, is given by

$$I = \frac{dC}{d\tau}$$

where  $C$  is the charge in coulombs and  $\tau$  is time in seconds. Thus  $dC$  is the charge crossing a boundary during time  $d\tau$ . If  $E$  is the voltage potential, the work is

$$\begin{aligned} dW &= E \cdot dC \\ &= EI d\tau \\ \therefore W &= \int_1^2 EI d\tau \end{aligned} \tag{3.12}$$

The electrical power will be

$$\dot{W} = \lim_{d\tau \rightarrow 0} \frac{dW}{d\tau} = EI \tag{3.13}$$

This is the rate at which work is transferred.

**(b) Shaft Work** When a shaft, taken as the system (Fig. 3.14), is rotated by a motor, there is work transfer into the system. This is because the shaft can rotate a pulley which can raise a weight. If  $T$  is the torque applied to the shaft and  $d\theta$  is the angular displacement of the shaft, the shaft work is

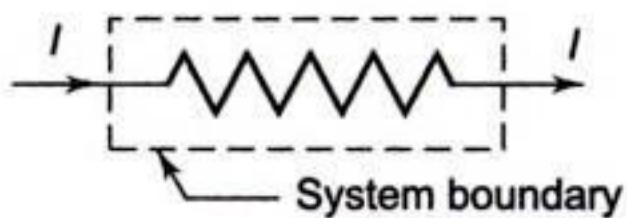


Fig. 3.13 Electrical Work

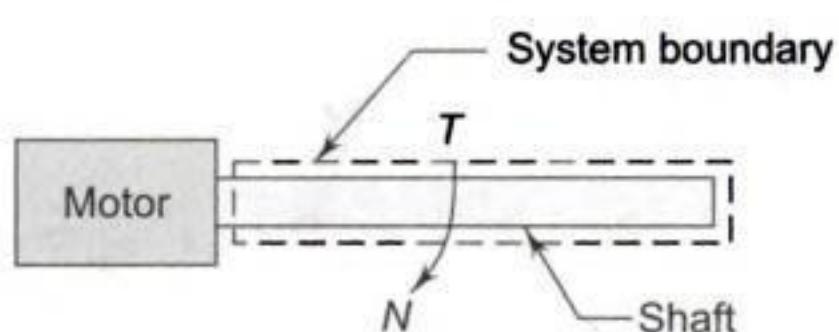


Fig. 3.14 Shaft Work

$$W = \int_1^2 T d\theta \quad (3.14a)$$

and the shaft power is

$$W = \int_1^2 T \frac{d\theta}{d\tau} = T\omega \quad (3.15b)$$

where  $\omega$  is the angular velocity and  $T$  is considered a constant in this case.

**(c) Paddle-Wheel Work or Stirring Work** As the weight is lowered, and the paddle wheel turns (Fig. 3.15), there is work transfer into the fluid system which gets stirred. Since the volume of the system remains constant,  $\int pdV = 0$ . If  $m$  is the mass of the weight lowered through a distance  $dz$  and  $T$  is the torque transmitted by the shaft in rotating through an angle  $d\theta$ , the differential work transfer to the fluid is given by

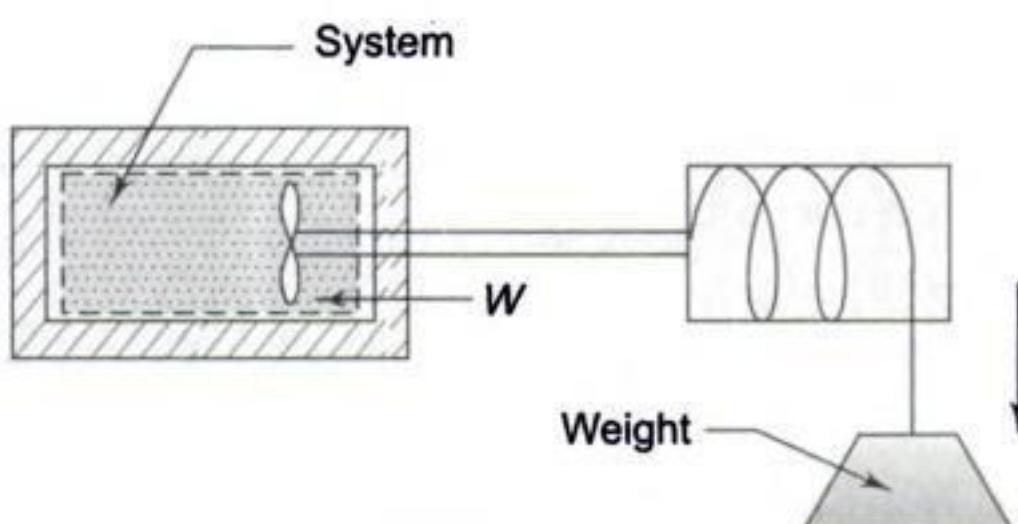


Fig. 3.15 Paddle-Wheel Work

$$dW = mgdz = Td\theta$$

and the total work transfer is

$$W = \int_1^2 mgdz = \int_1^2 W' dz = \int_1^2 Td\theta \quad (3.15)$$

where  $W'$  is the weight lowered.

**(d) Flow Work** The flow work, significant only in a flow process or an open system, represents the energy transferred across the system boundary as a result of the energy imparted to the fluid by a pump, blower or compressor to make the fluid flow across the control volume. Flow work is analogous to displacement work. Let  $p$  be the fluid pressure in the plane of the imaginary piston, which acts in a direction normal to it (Fig. 3.16). The work done on this imaginary piston by the external pressure as the piston moves forward is given by

$$dW_{\text{flow}} = p dV, \quad (3.16)$$

where  $dV$  is the volume of fluid element about to enter the system.

$$\therefore dW_{\text{flow}} = pv dm \quad (3.17)$$

where  $dV = v dm$

Therefore, flow work at inlet (Fig. 3.16),

$$(dW_{\text{flow}})_{\text{in}} = p_1 v_1 dm_1 \quad (3.18)$$

Equation (3.18) can also be derived in a slightly different manner. If the normal pressure  $p_1$  is exerted against the area  $A_1$ , giving a total force  $(p_1 A_1)$  against the piston, in time  $d\tau$ , this force

moves a distance  $\mathbf{V}_1 d\tau$ , where  $\mathbf{V}_1$  is the velocity of flow (piston). The work in the time  $d\tau$  is  $p_1 A_1 \mathbf{V}_1 d\tau$ , or the work per unit time is  $p_1 A_1 \mathbf{V}_1$ . Since the flow rate

$$\omega_1 = \frac{A_1 \mathbf{V}_1}{v_1} = \frac{dm_1}{d\tau}$$

the work done in time  $d\tau$  becomes

$$(dW_{\text{flow}})_{\text{in}} = p_1 v_1 dm_1$$

Similarly, flow work of the fluid element leaving the system is

$$(dW_{\text{flow}})_{\text{out}} = p_2 v_2 dm_2 \quad (3.19)$$

The flow work per unit mass is thus

$$dW_{\text{flow}} = pv \quad (3.20)$$

It is the displacement work done at the moving system boundary.

**(e) Work Done in Stretching a Wire** Let us consider a wire as the system. If the length of the wire in which there is a tension  $\mathcal{F}$  is changed from  $L$  to  $L + dL$ , the infinitesimal amount of work that is done is equal to

$$dW = -\mathcal{F} dL$$

The minus sign is used because a positive value of  $dL$  means an expansion of the wire, for which work must be done on the wire, i.e. negative work. For a finite change of length,

$$W = - \int_1^2 \mathcal{F} dL \quad (3.21)$$

If we limit the problem to within the elastic limit, where  $E$  is the modulus of elasticity,  $s$  is the stress,  $\epsilon$  is the strain, and  $A$  is the cross-sectional area, then

$$\mathcal{F} = sA = E\epsilon A, \text{ since } \frac{s}{\epsilon} = E$$

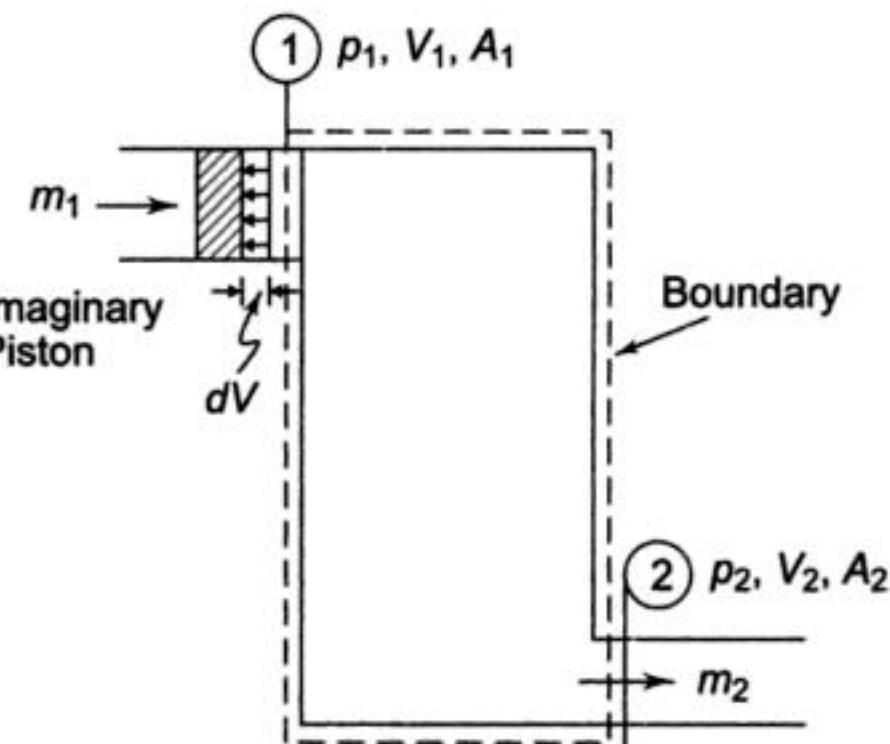


Fig. 3.16 Flow Work

$$d\varepsilon = \frac{dL}{L}$$

$$dW = -\mathcal{F} dL = -E\varepsilon AL d\varepsilon$$

$$\therefore W = -A\varepsilon L \int_1^2 \varepsilon d\varepsilon = -\frac{AEL}{2} (\varepsilon_2^2 - \varepsilon_1^2) \quad (3.22)$$

**(f) Work Done in Changing the Area of a Surface Film** A film on the surface of a liquid has a surface tension, which is a property of the liquid and the surroundings. The surface tension acts to make the surface area of the liquid a minimum. It has the unit of force per unit length. The work done on a homogeneous liquid film in changing its surface area by an infinitesimal amount  $dA$  is

$$dW = -\sigma dA$$

where  $\sigma$  is the surface tension (N/m).

$$\therefore W = - \int_1^2 \sigma dA \quad (3.23)$$

**(g) Magnetization of a Paramagnetic Solid** The work done per unit volume on a magnetic material through which the magnetic and magnetization fields are uniform is

$$dW = -HdI$$

and

$$W_{1-2} = - \int_{I_1}^{I_2} H dI \quad (3.24)$$

where  $H$  is the field strength, and  $I$  is the component of the magnetization field in the direction of the field. The minus sign provides that an increase in magnetization (positive  $dI$ ) involves negative work.

The following equations summarize the different forms of work transfer:

Displacement work  
(compressible fluid)  $W = \int_1^2 p dV$

Electrical work  $W = \int_1^2 E dC = \int_1^2 EI d\tau$

Shaft work  $W = \int_1^2 T d\theta$

Surface film  $W = - \int_1^2 \sigma dA \quad (3.25)$

Stretched wire

$$W = - \int_1^2 \mathcal{F} dL$$

Magnetised solid

$$W = - \int_1^2 H dI$$

It may be noted in the above expressions that the work is equal to the integral of the product of an intensive property and the change in its related extensive property. These expressions are valid only for infinitesimally slow quasi-static processes.

There are some other forms of work which can be identified in processes that are not quasi-static, for example, the work done by shearing forces in a process involving friction in a viscous fluid.

### 3.5 FREE EXPANSION WITH ZERO WORK TRANSFER

Work transfer is identified only at the boundaries of a system. It is a boundary phenomenon, and a form of energy in transit crossing the boundary. Let us consider a gas separated from the vacuum by a partition (Fig. 3.17). Let the partition be removed. The gas rushes to fill the entire volume. The expansion of a gas against vacuum is called *free expansion*. If we neglect the work associated with the removal of partition, and consider the gas and vacuum together as our system (Fig. 3.17a), there is no work transfer involved here, since no work crosses the system boundary, and hence

$$\int_1^2 dW = 0, \quad \text{although} \quad \int_1^2 pdV \neq 0$$

If only the gas is taken as the system (Fig. 3.17b), when the partition is removed there is a change in the volume of the gas, and one is tempted to calculate the work from the expression  $\int_1^2 pdV$ . However, this is not a quasi-static process, although the initial and final end states are in equilibrium. Therefore, the work cannot be calculated from this relation. The two end states can be located on the  $p$ - $V$  diagram and these are joined by a dotted line (Fig. 3.17c) to indicate that the process had occurred. However, if the vacuum space is divided into a large number of small volumes by partitions and the partitions are removed one by one slowly (Fig. 3.17d), then every state passed through by the system is an equilibrium state and the work done can then be estimated from the relation  $\int_1^2 pdV$  (Fig. 3.17e).

Yet, in free expansion of a gas, there is no resistance to the fluid at the system boundary as the volume of the gas increases to fill up the vacuum space. Work is done by a system to overcome some resistance. Since vacuum does not offer any resistance, there is no work transfer involved in free expansion.

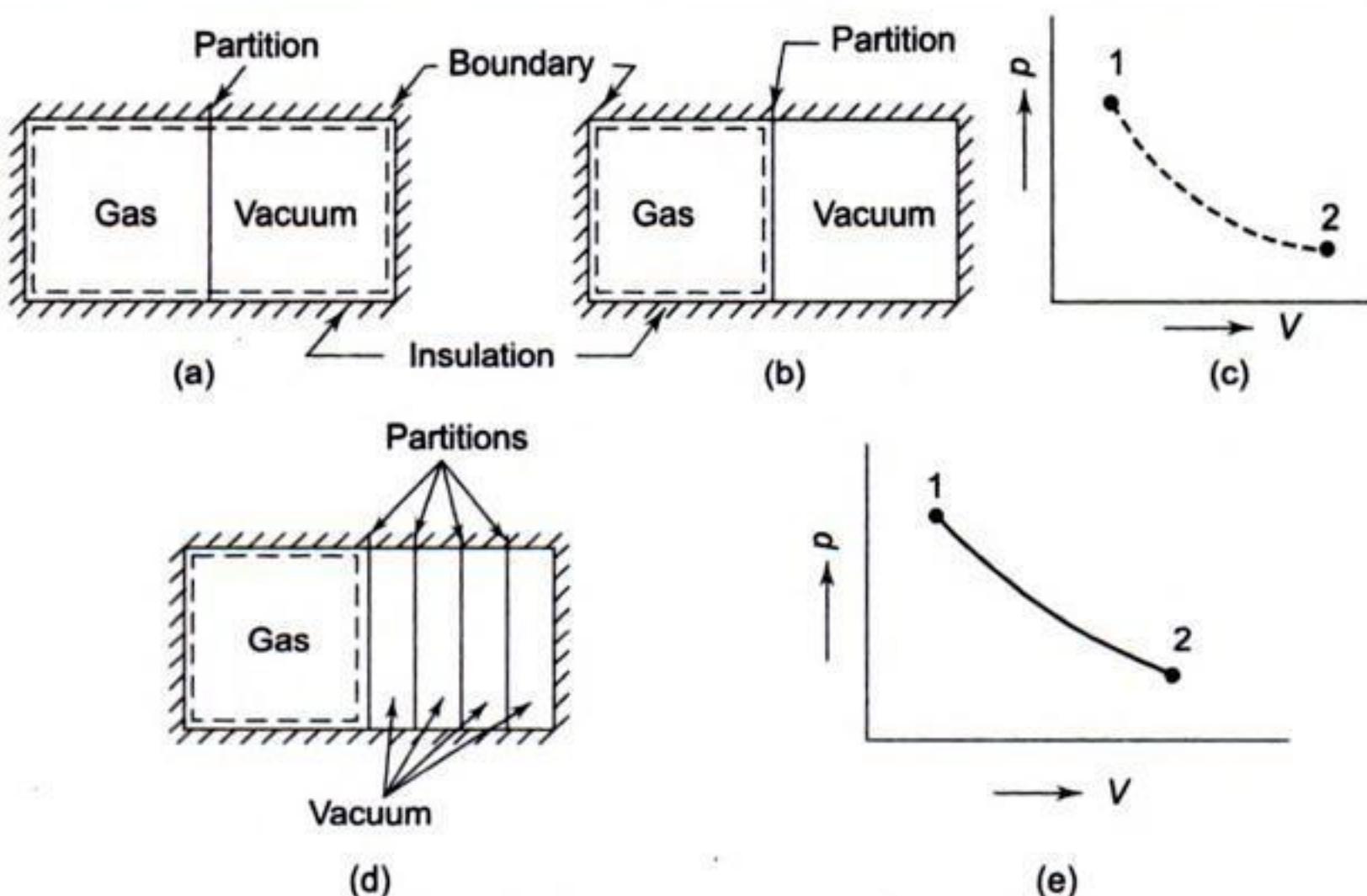


Fig. 3.17 Free Expansion

### 3.6 NET WORK DONE BY A SYSTEM

Often different forms of work transfer occur simultaneously during a process executed by a system. When all these work interactions have been evaluated, the total or net work done by the system would be equal to the algebraic sum of these as given below

$$W_{\text{total}} = W_{\text{displacement}} + W_{\text{shear}} + W_{\text{electrical}} + W_{\text{stirring}} + \dots$$

### 3.7 HEAT TRANSFER

Heat is defined as the form of energy that is transferred across a boundary by virtue of a temperature difference. The temperature difference is the ‘potential’ or ‘force’ and heat transfer is the ‘flux’.

The transfer of heat between two bodies in direct contact is called *conduction*. Heat may be transferred between two bodies separated by empty space or gases by the mechanism of *radiation* through electromagnetic waves. A third method of heat transfer is *convection* which refers to the transfer of heat between a wall and a fluid system in motion.

The direction of heat transfer is taken from the high temperature system to the low temperature system. *Heat flow into a system is taken to be positive, and heat flow out of a system is taken as negative* (Fig. 3.18). The symbol  $Q$  is used for heat transfer, i.e. the quantity of heat transferred within a certain time.

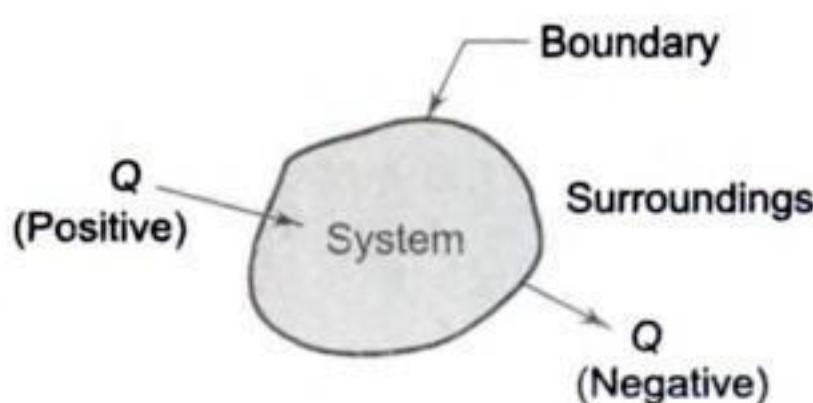


Fig. 3.18 Direction of Heat Transfer

Heat is a form of energy in transit (like work transfer). It is a boundary phenomenon, since it occurs only at the boundary of a system. Energy transfer by virtue of temperature difference only is called heat transfer. All other energy interactions may be termed as work transfer.

Heat is not that which inevitably causes a temperature rise. When heat is transferred to an ice-and-water mixture, the temperature does not rise until all the ice has melted. When a temperature rise in a system occurs, it may not be due to heat transfer, since a temperature rise may be caused by work transfer also. Heat, like work, is not a conserved quantity, and is not a property of a system.

A process in which no heat crosses the boundary of the system is called an *adiabatic process*.

Thus, an adiabatic process is one in which there is only work interaction between the system and its surroundings.

A wall which is impermeable to the flow of heat is an *adiabatic wall*, whereas a wall which permits the flow of heat is a *diathermic wall*.

The unit of heat is Joule in S.I. units.

The rate of heat transfer or work transfer is given in kW or W.

### 3.8 HEAT TRANSFER—A PATH FUNCTION

Heat transfer is a *path function*, that is, the amount of heat transferred when a system changes from state 1 to state 2 depends on the intermediate states through which the system passes, i.e. its path. Therefore  $dQ$  is an inexact differential, and we write

$$\int_1^2 dQ = Q_{1-2} \quad \text{or} \quad _1Q_2$$

The displacement work is given by

$$W_{1-2} = \int_1^2 dW = \int_1^2 pdV$$

It is valid for a quasi-static process, and the work transfer involved is represented by the area under the path on  $p-v$  diagram (Fig. 3.19a). Whenever there is a difference in pressure, there will be displacement work. The pressure difference is the cause and work transfer is the effect. The work transfer is equal to the integral of the product of the intensive property,  $p$  and the differential change in the extensive property,  $dV$ .

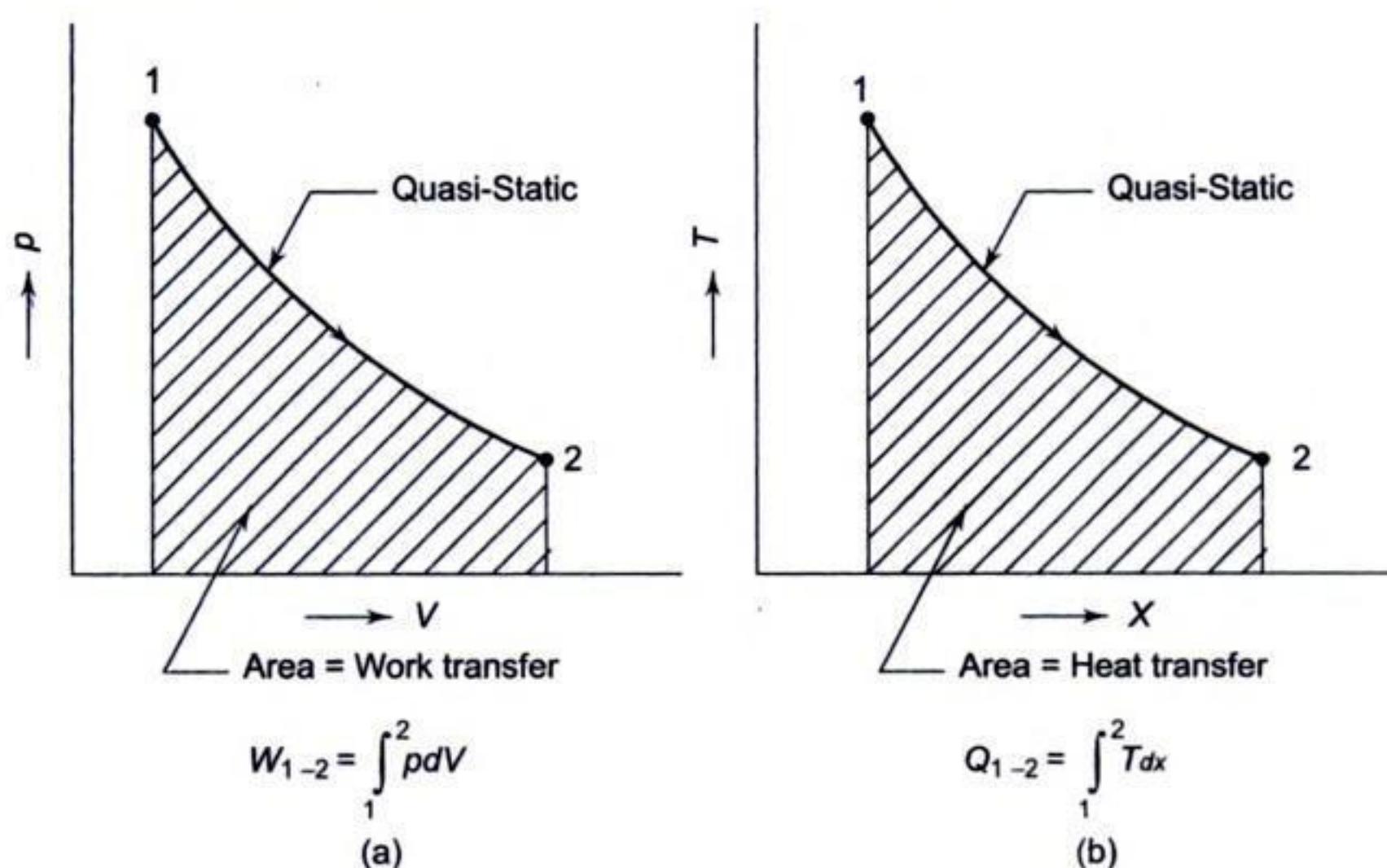


Fig. 3.19 Representation of Work Transfer and Heat Transfer in Quasi-Static Processes on  $p$ - $v$  and  $T$ - $x$  Coordinates

Likewise, whenever there is a difference in temperature, there will be heat flow. The temperature difference is the cause and heat transfer is the effect. Just like displacement work, the heat transfer can also be written as the integral of the product of the intensive property  $T$  and the differential change of an extensive property, say  $X$  (Fig. 3.19b).

$$Q_{1-2} = \int_1^2 dQ = \int_1^2 TdX \quad (3.26)$$

It must also be valid for a quasi-static process only, and the heat transfer involved is represented by the area under the path 1–2 in  $T$ - $X$  plot (Fig. 3.19b). Heat transfer is, therefore, a path function, i.e. the amount of heat transferred when a system changes from state 1 to state 2 depends on the path the system follows (Fig. 3.19b). Therefore  $dQ$  is an inexact differential. Now,

$$dQ = T dX$$

where  $X$  is an extensive property and  $dX$  is an exact differential.

$$\therefore dX = \frac{1}{T} dQ \quad (3.27)$$

To make  $dQ$  integrable, i.e. an exact differential, it must be multiplied by an integrating factor which is, in this case,  $1/T$ . The extensive property  $X$  is yet to be defined. It has been introduced in Chapter 7 and it is called ‘entropy’.

### 3.9 SPECIFIC HEAT AND LATENT HEAT

The *specific heat* of a substance is defined as the amount of heat required to raise a unit mass of the substance through a unit rise in temperature. The symbol  $c$  will be used for specific heat.

$$\therefore c = \frac{Q}{m \cdot \Delta t} \text{ J/kg K}$$

where  $Q$  is the amount of heat transfer (J),  $m$ , the mass of the substance (kg), and  $\Delta t$ , the rise in temperature (K).

Since heat is not a property, as explained later, so the specific heat is qualified with the process through which exchange of heat is made. For gases, if the process is at constant pressure, it is  $c_p$ , and if the process is at constant volume, it is  $c_v$ . For solids and liquids, however, the specific heat does not depend on the process. An elegant manner of defining specific heats,  $c_v$  and  $c_p$ , in terms of properties is given in Secs 4.5 and 4.6.

The product of mass and specific heat ( $mc$ ) is called the *heat capacity* of the substance. The capital letter  $C$ ,  $C_p$  or  $C_v$ , is used for heat capacity.

The latent heat is the amount of heat transfer required to cause a phase change in unit mass of a substance *at a constant pressure and temperature*. There are three phases in which matter can exist: solid, liquid, and vapour or gas. The *latent heat of fusion* ( $l_{fu}$ ) is the amount of heat transferred to melt unit mass of solid into liquid, or to freeze unit mass of liquid to solid. The *latent heat of vaporization* ( $l_{vap}$ ) is the quantity of heat required to vaporize unit mass of liquid into vapour, or condense unit mass of vapour into liquid. The *latent heat of sublimation* ( $l_{sub}$ ) is the amount of heat transferred to convert unit mass of solid to vapour or vice versa.  $l_{fu}$  is not much affected by pressure, whereas  $l_{vap}$  is highly sensitive to pressure.

### 3.10 POINTS TO REMEMBER REGARDING HEAT TRANSFER AND WORK TRANSFER

- (a) Heat transfer and work transfer are the *energy interactions*. A closed system and its surroundings can interact in two ways: by heat transfer and by work transfer. Thermodynamics studies how these interactions bring about property changes in a system.
- (b) The same effect in a closed system can be brought about either by heat transfer or by work transfer. Whether heat transfer or work transfer has taken place depends on what constitutes the system.
- (c) Both heat transfer and work transfer are boundary phenomena. Both are observed at the boundaries of the system, and both represent energy crossing the boundaries of the system.
- (d) It is wrong to say '*total heat*' or '*heat content*' of a closed system, because heat or work is not a property of the system. Heat, like work, cannot be stored by the system. Both heat and work are the energy in transit.

- (e) Heat transfer is the energy interaction due to temperature difference only. All other energy interactions may be termed as work transfer.
- (f) Both heat and work are path functions and inexact differentials. The magnitude of heat transfer or work transfer depends upon the path the system follows during the change of state.

## Solved Examples

**Example 3.1** Gas from a bottle of compressed helium is used to inflate an inelastic flexible balloon, originally folded completely flat to a volume of  $0.5 \text{ m}^3$ . If the barometer reads 760 mm Hg, what is the amount of work done upon the atmosphere by the balloon? Sketch the system before and after the process.

**Solution** The firm line  $P_1$  (Fig. 3.20) shows the boundary of the system before the process, and the dotted line  $P_2$  shows the boundary after the process. The displacement work

$$W_d = \int_{\text{Balloon}} pdV + \int_{\text{Bottle}} pdV = p \Delta V + 0$$

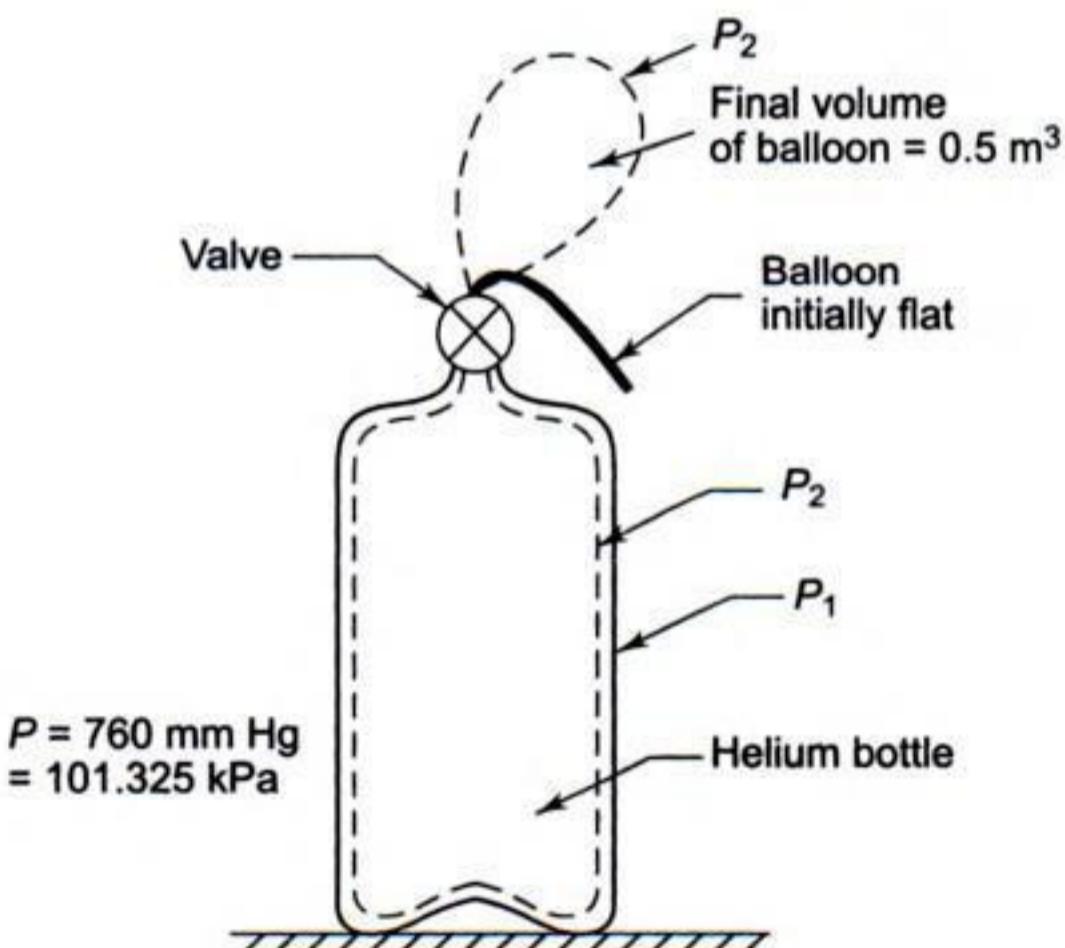


Fig. 3.20

$$= 101.325 \frac{\text{kN}}{\text{m}^2} \times 0.5 \text{ m}^3 = 50.66 \text{ kJ}$$

This is positive, because work is done by the system. Work done by the atmosphere is  $-50.66 \text{ kJ}$ . Since the wall of the bottle is rigid, there is no  $pdV$ -work involved in it.

It is assumed that the pressure in the balloon is atmospheric at all times, since the balloon fabric is light, inelastic and unstressed. If the balloon were elastic and

stressed during the filling process, the work done by the gas would be greater than 50.66 kJ by an amount equal to the work done in stretching the balloon, although the displacement work done by the atmosphere is still  $-50.66 \text{ kJ}$ . However, if the system includes both the gas and the balloon, the displacement work would be 50.66 kJ, as estimated above.

**Example 3.2** When the valve of the evacuated bottle (Fig. 3.21) is opened, atmospheric air rushes into it. If the atmospheric pressure is 101.325 kPa, and  $0.6 \text{ m}^3$  of air (measured at atmospheric conditions) enters into the bottle, calculate the work done by air.

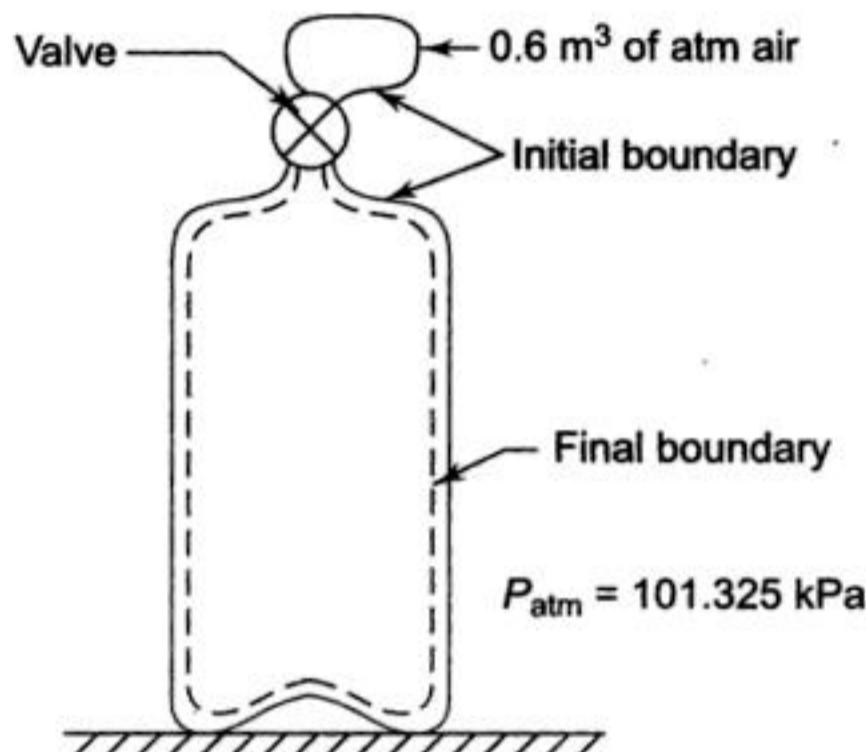


Fig. 3.21

**Solution** The displacement work done by air

$$\begin{aligned} W_d &= \int_{\text{Bottle}} pdV + \int_{\text{Free-air boundary}} pdV \\ &= 0 + p \Delta V \\ &= 101.325 \text{ kN/m}^2 \times 0.6 \text{ m}^3 = 60.8 \text{ kJ} \end{aligned}$$

Since the free-air boundary is contracting, the work done by the system is negative ( $\Delta V$  being negative), and the surroundings do positive work upon the system.

**Example 3.3** A piston and cylinder machine containing a fluid system has a stirring device in the cylinder (Fig. 3.22). The piston is frictionless, and it is held down against the fluid due to the atmospheric pressure of 101.325 kPa. The stirring device is turned 10,000 revolutions with an average torque against the fluid of 1.275 mN. Meanwhile the piston of 0.6 m diameter moves out 0.8 m. Find the net work transfer for the system.

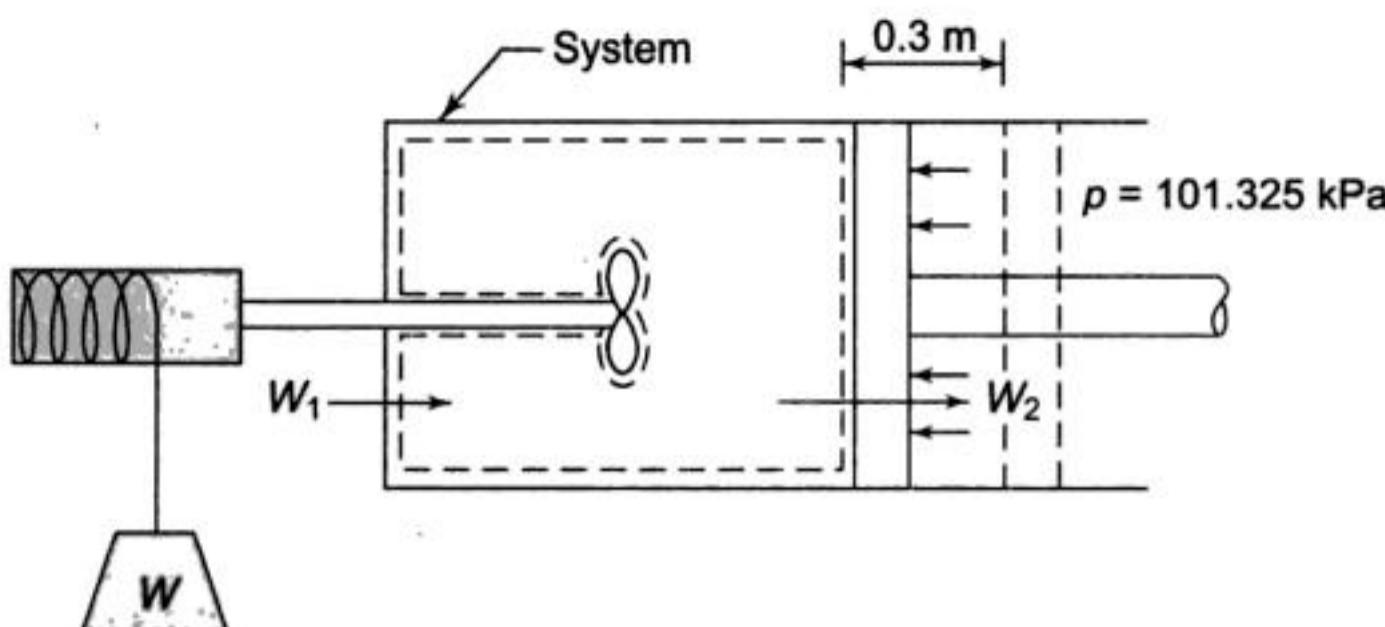


Fig. 3.22

**Solution** Work done by the stirring device upon the system (Fig. 3.22)

$$\begin{aligned} W_1 &= 2\pi TN \\ &= 2\pi \times 1.275 \times 10,000 \text{ Nm} = 80 \text{ kJ} \end{aligned}$$

This is negative work for the system.

Work done by the system upon the surroundings

$$\begin{aligned} W_2 &= (pA) \cdot L \\ &= 101.325 \frac{\text{kN}}{\text{m}^2} \times \frac{\pi}{4} (0.6)^2 \text{ m}^2 \times 0.80 \text{ m} = 22.9 \text{ kJ} \end{aligned}$$

This is positive work for the system. Hence, the net work transfer for the system

$$W = W_1 + W_2 = -80 + 22.9 = -57.1 \text{ kJ}$$

**Example 3.4** The following data refer to a 12-cylinder, single-acting, two-stroke marine diesel engine:

Speed—150 rpm

Cylinder diameter—0.8 m

Stroke of piston—1.2 m

Area of indicator diagram— $5.5 \times 10^{-4} \text{ m}^2$

Length of diagram—0.06 m

Spring value—147 MPa per m

Find the net rate of work transfer from the gas to the pistons in kW.

**Solution** Mean effective pressure,  $p_m$ , is given by

$$\begin{aligned} p_m &= \frac{a_d}{l_d} \times \text{spring constant} \\ &= \frac{5.5 \times 10^{-4} \text{ m}^2}{0.06} \times 147 \frac{\text{MPa}}{\text{m}} = 1.35 \text{ MPa} \end{aligned}$$

One engine cycle is completed in two strokes of the piston or one revolution of the crank-shaft.

∴ Work done in one minute

$$= p_m LAN$$

$$= 1.35 \times \frac{\pi}{4} (0.8)^2 \times 1.2 \times 150 = 122 \text{ MJ}$$

Since the engine is single-acting, and it has 12 cylinders, each contributing an equal power, the rate of work transfer from the gas to the piston is given by

$$W = 122 \times 12 \text{ MJ/min}$$

$$= 24.4 \text{ MJ/s}$$

$$= 24.4 \text{ MW} = 24,400 \text{ kW}$$

**Example 3.5** It is required to melt 5 tonnes/h of iron from a charge at 15°C to molten metal at 1650°C. The melting point is 1535°C, and the latent heat is 270 kJ/kg. The specific heat in solid state is 0.502 and in liquid state (29.93/atomic weight) kJ/kg K. If an electric furnace has 70% efficiency, find the kW rating needed. If the density in molten state is 6900 kg/m<sup>3</sup> and the bath volume is three times the hourly melting rate, find the dimensions of the cylindrical furnace if the length to diameter ratio is 2. The atomic weight of iron is 56.

**Solution** Heat required to melt 1 kg of iron at 15°C to molten metal at 1650°C

$$\begin{aligned} &= \text{Heat required to raise the temperature from } 15^\circ\text{C to } 1535^\circ\text{C} + \text{Latent heat} + \text{Heat required to raise the temperature from } 1535^\circ\text{C to } 1650^\circ\text{C} \\ &= 0.502 (1535 - 15) + 270 + 29.93 (1650 - 1535)/56 \\ &= 763 + 270 + 61.5 = 1094.5 \text{ kJ/kg} \end{aligned}$$

$$\text{Melting rate} = 5 \times 10^3 \text{ kg/h}$$

So, the rate of heat supply required

$$= (5 \times 10^3 \times 1094.5) \text{ kJ/h}$$

Since the furnace has 70% efficiency, the rating of the furnace would be

$$= \frac{\text{Rate of heat supply per second}}{\text{Furnace efficiency}}$$

$$= \frac{5 \times 10^3 \times 1094.5}{0.7 \times 3600} = 217 \times 10^3 \text{ kW}$$

$$\text{Volume needed} = \frac{3 \times 5 \times 10^3}{6900} \text{ m}^3 = 2.18 \text{ m}^3$$

If  $d$  is the diameter and  $l$  the length of the furnace

$$\frac{\pi}{4} d^2 l = 2.18 \text{ m}^3$$

or  $\frac{\pi}{4} d^2 \times 2d = 2.18 \text{ m}^3$

$$\therefore d = 1.15 \text{ m}$$

and  $l = 2.30 \text{ m}$

**Example 3.6** If it is desired to melt aluminium with solid state specific heat 0.9 kJ/kg K latent heat 390 kJ/kg, atomic weight 27, density in molten state 2400 kg/m<sup>3</sup> and final temperature 700°C, find out how much metal can be melted per hour with the above kW rating. Other data are as in the above example. Also, find the mass of aluminium that the above furnace will hold. The melting point of aluminium is 660°C.

**Solution** Heat required per kg of aluminium

$$\begin{aligned} &= 0.9 (660 - 15) + 390 + \frac{29.93}{27} (700 - 660) \\ &= 580.5 + 390 + 44.3 = 1014.8 \text{ kJ} \end{aligned}$$

$$\text{Heat to be supplied} = \frac{1014.8}{0.7} = 1449.7 \text{ kJ/kg}$$

With the given power, the rate at which aluminium can be melted

$$= \frac{217 \times 10^3 \times 3600}{1449.7} \text{ kg/h} = 5.39 \text{ tonnes/h}$$

Mass of aluminium that can be held in the above furnace

$$= 2.18 \times 2400 \text{ kg} = 5.23 \text{ tonnes}$$

## Review Questions

- 3.1 How can a closed system and its surroundings interact? What is the effect of such interactions on the system?
- 3.2 When is work said to be done by a system?
- 3.3 What are positive and negative work interactions?
- 3.4 What is displacement work?
- 3.5 Under what conditions is the work done equal to  $\int_1^2 p dV$ ?
- 3.6 What do you understand by path function and point function? What are exact and inexact differentials?
- 3.7 Show that work is a path function, and not a property.
- 3.8 What is an indicator diagram?
- 3.9 What is mean effective pressure? How is it measured?

- 3.10 What are the indicated power and the brake power of an engine?
- 3.11 How does the current flowing through a resistor represent work transfer?
- 3.12 What do you understand by flow work? Is it different from displacement work?
- 3.13 Why does free expansion have zero work transfer?
- 3.14 What is heat transfer? What are its positive and negative directions?
- 3.15 What are adiabatic and diathermic walls?
- 3.16 What is an integrating factor?
- 3.17 Show that heat is a path function and not a property.
- 3.18 What is the difference between work transfer and heat transfer?
- 3.19 Does heat transfer inevitably cause a temperature rise?

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## Problems

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- 3.1 (a) A pump forces  $1 \text{ m}^3/\text{min}$  of water horizontally from an open well to a closed tank where the pressure is  $0.9 \text{ MPa}$ . Compute the work the pump must do upon the water in an hour just to force the water into the tank against the pressure. Sketch the system upon which the work is done before and after the process.  
Ans.  $5400 \text{ kJ/h}$
- (b) If the work done as above upon the water had been used solely to raise the same amount of water vertically against gravity without change of pressure, how many metres would the water have been elevated? Ans.  $91.74 \text{ m}$
- (c) If the work done in (a) upon the water had been used solely to accelerate the water from zero velocity without change of pressure or elevation, what velocity would the water have reached? If the work had been used to accelerate the water from an initial velocity of  $10 \text{ m/s}$ , what would the final velocity have been?  
Ans.  $42.4 \text{ m/s}; 43.6 \text{ m/s}$
- 3.2 The piston of an oil engine, of area  $0.0045 \text{ m}^2$ , moves downwards  $75 \text{ mm}$ , drawing in  $0.00028 \text{ m}^3$  of fresh air from the atmosphere. The pressure in the cylinder is uniform during the process at  $80 \text{ kPa}$ , while the atmospheric pressure is  $101.325 \text{ kPa}$ , the difference being due to the flow resistance in the induction pipe and the inlet valve. Estimate the displacement work done by the air finally in the cylinder.  
Ans.  $27 \text{ J}$
- 3.3 An engine cylinder has a piston of area  $0.12 \text{ m}^2$  and contains gas at a pressure of  $1.5 \text{ MPa}$ . The gas expands according to a process which is represented by a straight line on a pressure-volume diagram. The final pressure is  $0.15 \text{ MPa}$ . Calculate the work done by the gas on the piston if the stroke is  $0.30 \text{ m}$ .  
Ans.  $29.7 \text{ kJ}$
- 3.4 A mass of  $1.5 \text{ kg}$  of air is compressed in a quasi-static process from  $0.1 \text{ MPa}$  to  $0.7 \text{ MPa}$  for which  $pv = \text{constant}$ . The initial density of air is  $1.16 \text{ kg/m}^3$ . Find the work done by the piston to compress the air.  
Ans.  $251.62 \text{ kJ}$

- 3.5 A mass of gas is compressed in a quasi-static process from 80 kPa, 0.1 m<sup>3</sup> to 0.4 MPa, 0.03 m<sup>3</sup>. Assuming that the pressure and volume are related by  $pV^n = \text{constant}$ , find the work done by the gas system.
- Ans. -11.83 kJ
- 3.6 A single-cylinder, double-acting, reciprocating water pump has an indicator diagram which is a rectangle 0.075 m long and 0.05 m high. The indicator spring constant is 147 MPa per m. The pump runs at 50 rpm. The pump cylinder diameter is 0.15 m and the piston stroke is 0.20 m. Find the rate in kW at which the piston does work on the water.
- Ans. 43.3 kW
- 3.7 A single-cylinder, single-acting, 4 stroke engine of 0.15 m bore develops an indicated power of 4 kW when running at 216 rpm. Calculate the area of the indicator diagram that would be obtained with an indicator having a spring constant of  $25 \times 10^6$  N/m<sup>3</sup>. The length of the indicator diagram is 0.1 times the length of the stroke of the engine.
- Ans. 505 mm<sup>2</sup>
- 3.8 A six-cylinder, 4-stroke gasoline engine is run at a speed of 2520 RPM. The area of the indicator card of one cylinder is  $2.45 \times 10^3$  mm<sup>2</sup> and its length is 58.5 mm. The spring constant is  $20 \times 10^6$  N/m<sup>3</sup>. The bore of the cylinders is 140 mm and the piston stroke is 150 mm. Determine the indicated power, assuming that each cylinder contributes an equal power.
- Ans. 243.57 kW
- 3.9 A closed cylinder of 0.25 m diameter is fitted with a light frictionless piston. The piston is retained in position by a catch in the cylinder wall and the volume on one side of the piston contains air at a pressure of 750 kN/m<sup>2</sup>. The volume on the other side of the piston is evacuated. A helical spring is mounted coaxially with the cylinder in this evacuated space to give a force of 120 N on the piston in this position. The catch is released and the piston travels along the cylinder until it comes to rest after a stroke of 1.2 m. The piston is then held in its position of maximum travel by a ratchet mechanism. The spring force increases linearly with the piston displacement to a final value of 5 kN. Calculate the work done by the compressed air on the piston.
- Ans. 3.07 kJ
- 3.10 A steam turbine drives a ship's propeller through an 8 : 1 reduction gear. The average resisting torque imposed by the water on the propeller is  $750 \times 10^3$  mN and the shaft power delivered by the turbine to the reduction gear is 15 MW. The turbine speed is 1450 rpm. Determine (a) the torque developed by the turbine, (b) the power delivered to the propeller shaft, and (c) the net rate of working of the reduction gear.
- Ans. (a)  $T = 98.84$  km N, (b) 14.235 MW, (c) 0.765 MW
- 3.11 A fluid, contained in a horizontal cylinder fitted with a frictionless leakproof piston, is continuously agitated by means of a stirrer passing through the cylinder cover. The cylinder diameter is 0.40 m. During the stirring process lasting 10 minutes, the piston slowly moves out a distance of 0.485 m against the

atmosphere. The net work done by the fluid during the process is 2 kJ. The speed of the electric motor driving the stirrer is 840 rpm. Determine the torque in the shaft and the power output of the motor.

Ans. 0.08 mN, 6.92 W

- 3.12 At the beginning of the compression stroke of a two-cylinder internal combustion engine the air is at a pressure of 101.325 kPa. Compression reduces the volume to 1/5 of its original volume, and the law of compression is given by  $pV^{1.2} = \text{constant}$ . If the bore and stroke of each cylinder is 0.15 m and 0.25 m, respectively, determine the power absorbed in kW by compression strokes when the engine speed is such that each cylinder undergoes 500 compression strokes per minute.

Ans. 17.95 kW

- 3.13 Determine the total work done by a gas system following an expansion process as shown in Fig. 3.23.

Ans. 2.253 MJ

- 3.14 A system of volume  $V$  contains a mass  $m$  of gas at pressure  $p$  and temperature  $T$ . The macroscopic properties of the system obey the following relationship:

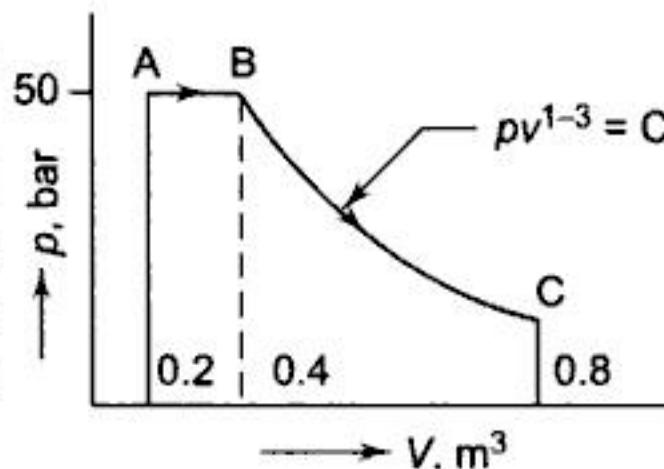


Fig. 3.23

where  $a$ ,  $b$ , and  $R$  constants.

Obtain an expression for the displacement work done by the system during a constant-temperature expansion from volume  $V_1$  to volume  $V_2$ . Calculate the work done by a system which contains 10 kg of this gas expanding from  $1\text{ m}^3$  to  $10\text{ m}^3$  at a temperature of 293 K. Use the values  $a = 15.7 \times 10\text{ Nm}^4$ ,  $b = 1.07 \times 10^{-2}\text{ m}^3$ , and  $R = 0.278\text{ kJ/kg-K}$ .

Ans. 1742 kJ

- 3.15 If a gas of volume  $6000\text{ cm}^3$  and at pressure of  $100\text{ kPa}$  is compressed quasistatically according to  $pV^2 = \text{constant}$  until the volume becomes  $2000\text{ cm}^3$ , determine the final pressure and the work transfer.

Ans. 900 kPa, -1.2 kJ

- 3.16 The flow energy of  $0.124\text{ m}^3/\text{min}$  of a fluid crossing a boundary to a system is 18 kW. Find the pressure at this point.

Ans. 8709 kPa

- 3.17 A milk chilling unit can remove heat from the milk at the rate of  $41.87\text{ MJ/h}$ . Heat leaks into the milk from the surroundings at an average rate of  $4.187\text{ MJ/h}$ . Find the time required for cooling a batch of 500 kg of milk from  $45^\circ\text{C}$  to  $5^\circ\text{C}$ . Take the  $c_p$  of milk to be  $4.187\text{ kJ/kgK}$ .

Ans. 2h 13 min

- 3.18 680 kg of fish at  $5^\circ\text{C}$  are to be frozen and stored at  $-12^\circ\text{C}$ . The specific heat of fish above freezing point is 3.182, and below freezing point is 1.717 kJ/kgK. The freezing point is  $-2^\circ\text{C}$ , and the latent heat of fusion is 234.5 kJ/kg. How much heat must be removed to cool the fish, and what per cent of this is latent heat?

Ans. 186.28 MJ, 85.6%

# 4

## CHAPTER

# First Law of Thermodynamics

### 4.1 FIRST LAW FOR A CLOSED SYSTEM UNDERGOING A CYCLE

The transfer of heat and the performance of work may both cause the same effect in a system. Heat and work are different forms of the same entity, called energy, which is conserved. Energy which enters a system as heat may leave the system as work, or energy which enters the system as work may leave as heat.

Let us consider a closed system which consists of a known mass of water contained in an adiabatic vessel having a thermometer and a paddle wheel, as shown in Fig. 4.1. Let a certain amount of work  $W_{1-2}$  be done upon the system by the paddle wheel. The quantity of work can be measured by the fall of weight which drives the paddle wheel through a pulley. The system was initially at

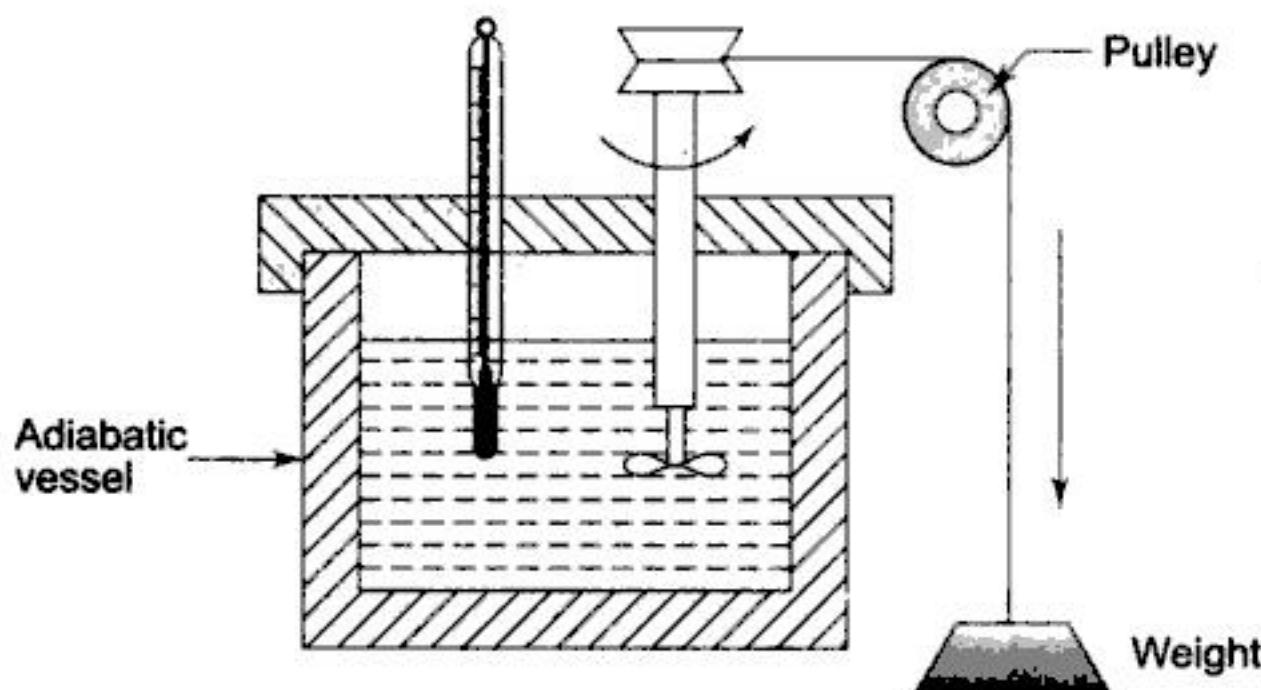


Fig. 4.1 Adiabatic Work

temperature  $t_1$ , the same as that of atmosphere, and after work transfer let the temperature rise to  $t_2$ . The pressure is always 1 atm. The process 1–2 undergone

by the system is shown in Fig. 4.2 in generalized thermodynamic coordinates  $X, Y$ . Let the insulation now be removed. The system and the surroundings interact by heat transfer till the system returns to the original temperature  $t_1$ , attaining the condition of thermal equilibrium with the atmosphere. The amount of heat transfer  $Q_{2-1}$  from the system during this process, 2–1, shown in Fig. 4.2, can be estimated. The system thus executes a cycle, which consists of a definite amount of work input  $W_{1-2}$  to the system followed by the transfer of an amount of heat  $Q_{2-1}$  from the system. It has been found that this  $W_{1-2}$  is always proportional to the heat  $Q_{2-1}$ , and the constant of proportionality is called the Joule's equivalent or the *mechanical equivalent of heat*. In the simple example given here, there are only two energy transfer quantities as the system performs a thermodynamic cycle. If the cycle involves many more heat and work quantities, the same result will be found. Expressed algebraically.

$$(\Sigma W)_{\text{cycle}} = J (\Sigma Q)_{\text{cycle}} \quad (4.1)$$

where  $J$  is the Joule's equivalent. This is also expressed in the form

$$\oint dW = J \oint dQ$$

where the symbol  $\oint$  denotes the cyclic integral for the closed path. This is the *first law for a closed system undergoing a cycle*. It is accepted as a *general law of nature*, since no violation of it has ever been demonstrated.

In the S.I. system of units, both heat and work are measured in the derived unit of energy, the Joule. The constant of proportionality,  $J$ , is therefore unity ( $J = 1 \text{ Nm/J}$ ).

The first law of thermodynamics owes much to J.P. Joule who, during the period 1840–1849, carried out a series of experiments to investigate the equivalence of work and heat. In one of these experiments, Joule used an apparatus similar to the one shown in Fig. 4.1. Work was transferred to the measured mass of water by means of paddle wheel driven by the falling weight. The rise in the temperature of water was recorded. Joule also used mercury as the fluid system, and later a solid system of metal blocks which absorbed work by friction when rubbed against each other. Other experiments involved the supplying of work in an electric current. In every case, he found the same ratio ( $J$ ) between the amount of work and the quantity of heat that would produce identical effects in the system.

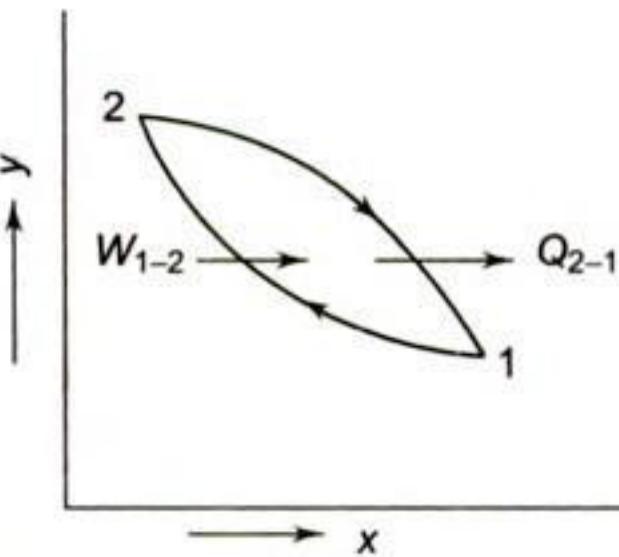


Fig. 4.2

*Cycle Completed by a System with Two Energy Interactions: Adiabatic Work Transfer  $W_{1-2}$  Followed by Heat Transfer  $Q_{2-1}$*

Prior to Joule, heat was considered to be an invisible fluid flowing from a body of higher calorie to a body of lower calorie, and this was known as the *caloric theory of heat*. It was Joule who first established that heat is a form of energy, and thus laid the foundation of the first law of thermodynamics.

## 4.2 FIRST LAW FOR A CLOSED SYSTEM UNDERGOING A CHANGE OF STATE

The expression  $(\sum W)_{\text{cycle}} = (\sum Q)_{\text{cycle}}$  applies only to systems undergoing cycles, and the algebraic summation of all energy transfer across system boundaries is zero. But if a system undergoes a change of state during which both heat transfer and work transfer are involved, the *net* energy transfer will be stored or accumulated within the system. If  $Q$  is the amount of heat transferred to the system and  $W$  is the amount of work transferred from the system during the process (Fig. 4.3), the net energy transfer ( $Q - W$ ) will be stored in the system. Energy in storage is neither heat nor work, and is given the name *internal energy* or simply, the *energy* of the system.

Therefore 
$$Q - W = \Delta E$$

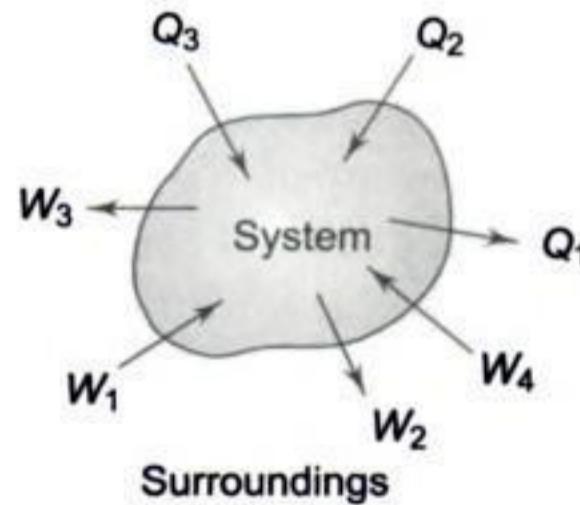
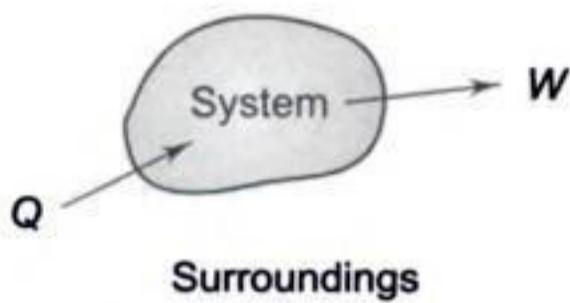
where  $\Delta E$  is the increase in the energy of the system

or 
$$Q = \Delta E + W \quad (4.2)$$

Here  $Q$ ,  $W$ , and  $\Delta E$  are all expressed in the same units (in joules). Energy may be stored by a system in different modes, as explained in Article 4.4.

If there are more energy transfer quantities involved in the process, as shown in Fig. 4.4, the first law gives

$$(Q_2 + Q_3 - Q_1) = \Delta E + (W_2 + W_3 - W_1 - W_4)$$



**Fig. 4.3 Heat and Work Interactions of a System with Its Surroundings in a Process**

**Fig. 4.4 System-surroundings Interaction in a Process Involving Many Energy Fluxes**

Energy is thus conserved in the operation. The first law is a particular formulation of the principle of the conservation of energy. Equation (4.2) may also be considered as the definition of energy. This definition does not give an absolute value of energy  $E$ , but only the change of energy  $\Delta E$  for the process. It can, however, be shown that the energy has a definite value at every state of a system and is, therefore, a property of the system.

### 4.3 ENERGY—A PROPERTY OF THE SYSTEM

Consider a system which changes its state from state 1 to state 2 by following the path *A*, and returns from state 2 to state 1 by following the path *B* (Fig. 4.5). So the system undergoes a cycle. Writing the first law for path *A*

$$Q_A = \Delta E_A + W_A \quad (4.3)$$

and for path *B*

$$Q_B = \Delta E_B + W_B \quad (4.4)$$

The processes *A* and *B* together constitute a cycle, for which

$$(\Sigma W)_{\text{cycle}} = (\Sigma Q)_{\text{cycle}}$$

or

$$W_A + W_B = Q_A + Q_B$$

or

$$Q_A - W_A = W_B - Q_B \quad (4.5)$$

From Eqs (4.3), (4.4), (4.5), it yields

$$\Delta E_A = -\Delta E_B \quad (4.6)$$

Similarly, had the system returned from state 2 to state 1 by following the path *C* instead of path *B*

$$\Delta E_A = -\Delta E_C \quad (4.7)$$

From Eqs (4.6) and (4.7)

$$\Delta E_B = \Delta E_C \quad (4.8)$$

Therefore, it is seen that the change in energy between two states of a system is the same, whatever path the system may follow in undergoing that change of state. If some arbitrary value of energy is assigned to state 2, the value of energy at state 1 is fixed independent of the path the system follows. Therefore, energy has a definite value for every state of the system. Hence, it is a *point function and a property of the system*.

The energy *E* is an extensive property. The *specific energy*,  $e = E/M$  (J/kg), is an intensive property.

The cyclic integral of any property is zero, because the final state is identical with the initial state.  $\oint dE = 0$ ,  $\oint dV = 0$ , etc. So for a cycle, the Eq. (4.2) reduces to Eq. (4.1).

### 4.4 DIFFERENT FORMS OF STORED ENERGY

It was stated at the beginning that thermodynamics is the science of energy transfer and its effect on the physical properties of a substance. Energy, as we know, is the capacity of doing work. In thermodynamics, energy can be in two forms: (i) Energy in transit, (ii) Energy in storage. Work and heat interactions are the forms of energy

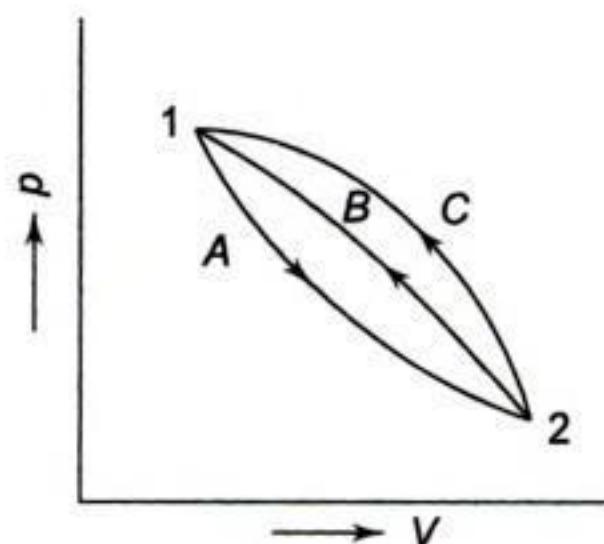


Fig. 4.5 Energy—A Property of a System

in transit, observed at the boundaries of a system. They are not properties of a system. They are path functions, their magnitudes depending upon the path the system follows during a change of state. Energy in storage, called internal energy, is a point or state function and hence a property of a system.

The symbol  $E$  refers to the total energy stored in a system. Basically there are two modes in which energy may be stored in a system:

- Macroscopic energy mode
- Microscopic energy mode

The macroscopic energy mode includes the macroscopic kinetic energy and potential energy of a system. Let us consider a fluid element of mass  $m$  having the centre of mass velocity  $\bar{V}$  (Fig. 4.6). The macroscopic kinetic energy  $E_K$  of the fluid element by virtue of its motion is given by

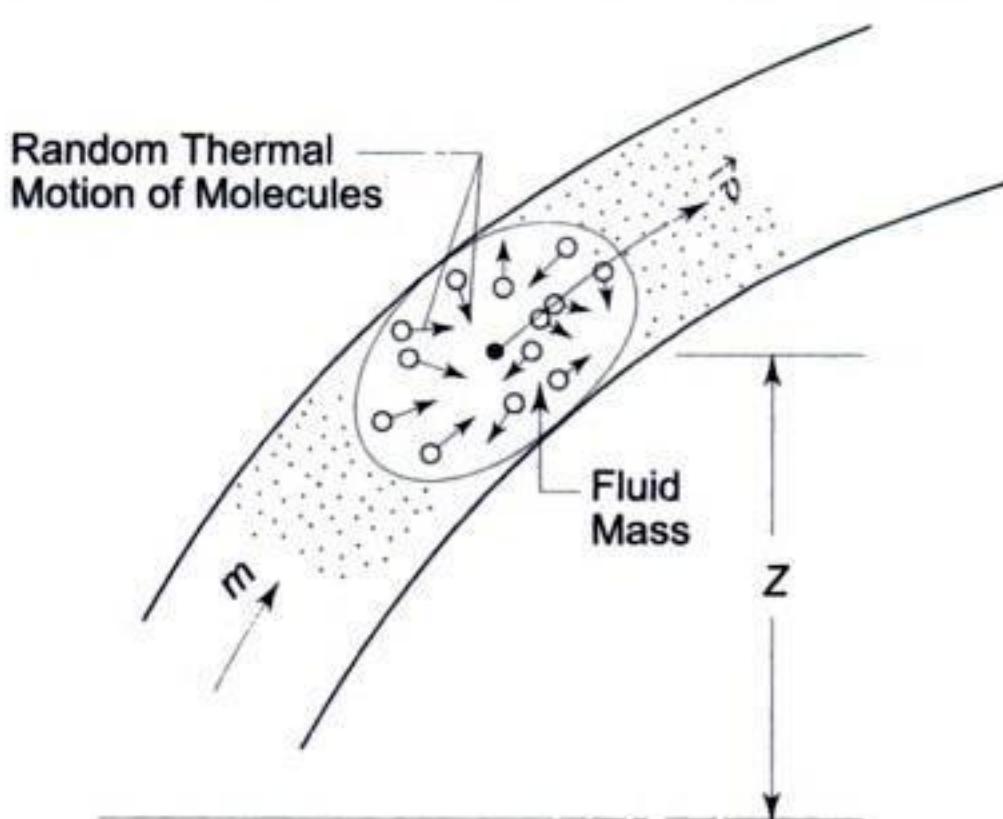
$$E_K = \frac{m \bar{V}^2}{2}$$

If the elevation of the fluid element from an arbitrary datum is  $z$ , then the macroscopic potential energy  $E_p$  by virtue of its position is given by

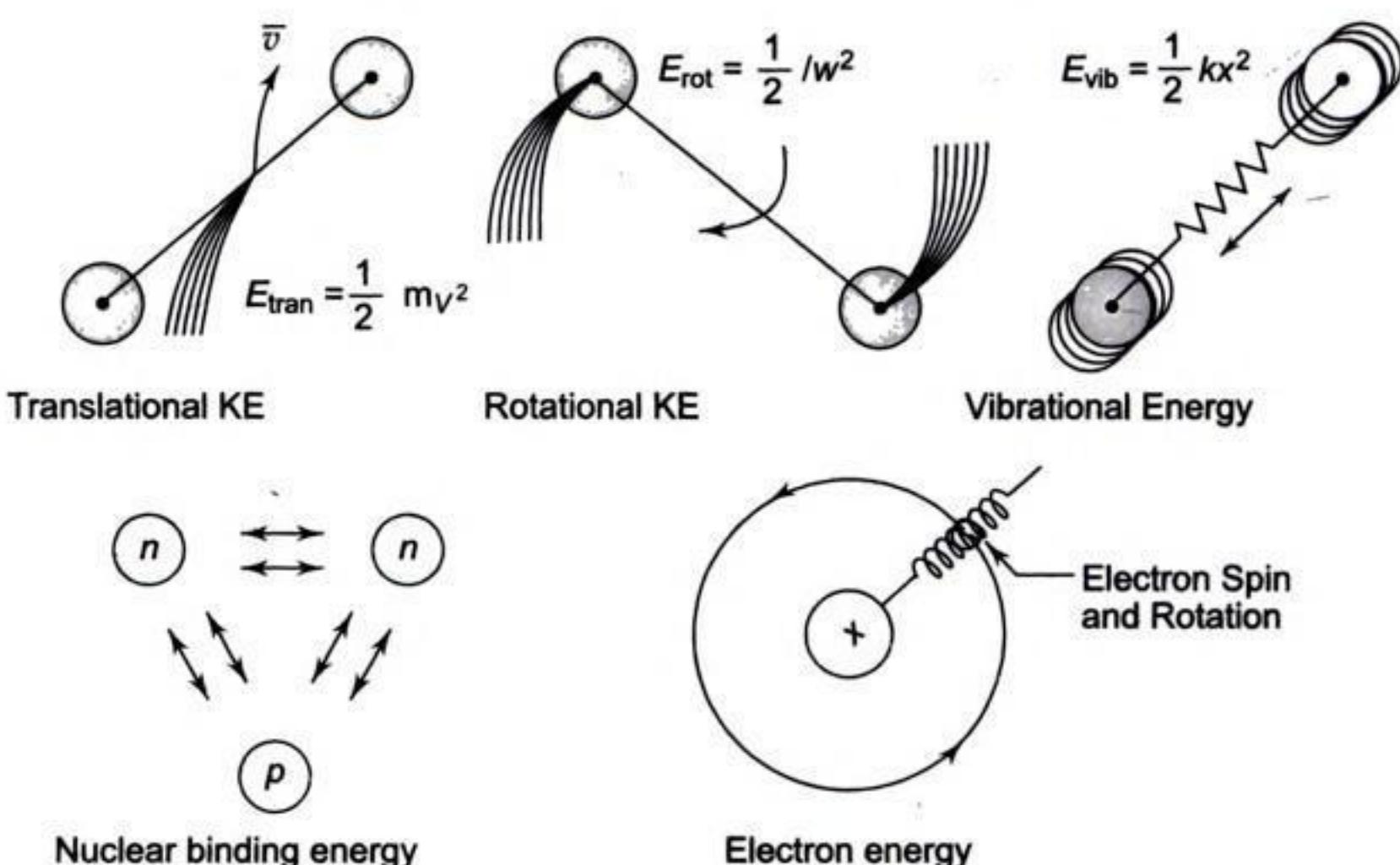
$$E_p = mgz$$

The microscopic energy mode refers to the energy stored in the molecular and atomic structure of the system, which is called the *molecular internal energy* or *simply internal energy*, customarily denoted by the symbol  $U$ . Matter is composed of molecules. Molecules are in random thermal motion (for a gas) with an average velocity  $\bar{v}$ , constantly colliding with one another and with the walls (Fig. 4.6). Due to a collision, the molecules may be subjected to rotation as well as vibration. They can have translational kinetic energy, rotational kinetic energy, vibrational energy, electronic energy, chemical energy and nuclear energy (Fig. 4.7). If  $\epsilon$  represents the energy of one molecule, then

$$\epsilon = \epsilon_{\text{trans}} + \epsilon_{\text{rot}} + \epsilon_{\text{vib}} + \epsilon_{\text{chem}} + \epsilon_{\text{electronic}} + \epsilon_{\text{nuclear}} \quad (4.9)$$



**Fig. 4.6** Macroscopic and Microscopic Energy



**Fig. 4.7** Various Components of Internal Energy Stored in a Molecule

If  $N$  is the total number of molecules in the system, then the total internal energy

$$U = N\epsilon \quad (4.10)$$

In an ideal gas there are no intermolecular forces of attraction and repulsion, and the internal energy depends only on temperature. Thus

$$U = f(T) \text{ only} \quad (4.11)$$

for an ideal gas

Other forms of energy which can also be possessed by a system are magnetic energy, electrical energy and surface (tension) energy. In the absence of these forms, the total energy  $E$  of a system is given by

$$E = \underbrace{E_K + E_P}_{\text{macro}} + \underbrace{U}_{\text{micro}} \quad (4.12)$$

where  $E_K$ ,  $E_P$ , and  $U$  refer to the kinetic, potential and internal energy, respectively. In the absence of motion and gravity.

$$E_K = 0, \quad E_P = 0$$

$$E = U$$

and Eq. (4.2) becomes

$$Q = \Delta U + W \quad (4.13)$$

$U$  is an extensive property of the system. The specific internal energy  $u$  is equal to  $U/m$  and its unit is J/kg.

In the differential forms, Eqs (4.2) and (4.13) become

$$dQ = dE + dW \quad (4.14)$$

$$dQ = dU + dW \quad (4.15)$$

where

$$dW = dW_{pdV} + dW_{\text{shaft}} + dW_{\text{electrical}} + \dots,$$

considering the different forms of work transfer which may be present. When only  $pdV$  work is present, the equations become

$$dQ = dE + pdV \quad (4.16)$$

$$dQ = dU + pdV \quad (4.17)$$

or, in the integral form

$$Q = \Delta E + \int pdV \quad (4.18)$$

$$Q = \Delta U + \int pdV \quad (4.19)$$

## 4.5 SPECIFIC HEAT AT CONSTANT VOLUME

The specific heat of a substance at constant volume  $c_v$  is defined as the rate of change of specific internal energy with respect to temperature when the volume is held constant, i.e.

$$c_v = \left( \frac{\partial u}{\partial T} \right)_v \quad (4.20)$$

For a constant-volume process

$$(\Delta u)_v = \int_{T_1}^{T_2} c_v \cdot dT \quad (4.21)$$

The first law may be written for a closed stationary system composed of a unit mass of a pure substance

$$Q = \Delta u + W$$

or

$$dQ = du + dW$$

For a process in the absence of work other than  $pdV$  work

$$dW = pdV$$

$$\therefore dQ = dU + pdV \quad (4.22)$$

When the volume is held constant

$$(Q)_v = (\Delta u)_v$$

$$\therefore (Q)_v = \int_{T_1}^{T_2} c_v \cdot dT \quad (4.23)$$

Heat transferred at constant volume increases the internal energy of the system.

If the specific heat of a substance is defined in terms of heat transfer, then

$$c_v = \left( \frac{\partial Q}{\partial T} \right)_u$$

Since  $Q$  is not a property, this definition does not imply that  $c_v$  is a property of a substance. Therefore, this is not the appropriate method of defining the specific heat, although  $(dQ)_v = du$ .

Since  $u$ ,  $T$ , and  $v$  are properties,  $c_v$  is a property of the system. The product  $mc_v = C_v$  is called the *heat capacity at constant volume* (J/K).

## 4.6 ENTHALPY

The enthalpy of a substance,  $h$ , is defined as

$$h = u + pv \quad (4.24)$$

It is an intensive property of a system (kJ/kg).

Internal energy change is equal to the heat transferred in a constant volume process involving no work other than  $pdV$  work. From Eq. (4.22), it is possible to derive an expression for the heat transfer in a constant pressure process involving no work other than  $pdV$  work. In such a process in a closed stationary system of unit mass of a pure substance

$$dQ = du + pdv$$

At constant pressure

$$pdv = d(pv)$$

$$\therefore (dQ)_p = du + d(pv)$$

$$\text{or } (dQ)_p = d(u + pv)$$

$$\text{or } (dQ)_p = dh \quad (4.25)$$

where  $h = u + pv$  is the *specific enthalpy*, a property of the system.

Heat transferred at constant pressure increases the enthalpy of a system.

For an ideal gas, the enthalpy becomes

$$h = u + RT \quad (4.26)$$

Since the internal energy of an ideal gas depends only on the temperature Eq. (4.11), the enthalpy of an ideal gas also depends on the temperature only, i.e.

$$h = f(T) \text{ only} \quad (4.27)$$

Total enthalpy  $H = mh$

Also  $H = U + pV$

and  $h = H/m$  (J/kg)

## 4.7 SPECIFIC HEAT AT CONSTANT PRESSURE

The specific heat at constant pressure  $c_p$  is defined as the rate of change of enthalpy with respect to temperature when the pressure is held constant

$$c_p = \left( \frac{\partial h}{\partial T} \right)_p \quad (4.28)$$

Since  $h$ ,  $T$  and  $p$  are properties, so  $c_p$  is a property of the system. Like  $c_v$ ,  $c_p$  should not be defined in terms of heat transfer at constant pressure, although  $(dQ)_p = dh$ .

For a constant pressure process

$$(\Delta h)_p = \int_{T_1}^{T_2} c_p dT \quad (4.29)$$

The first law for a closed stationary system of unit mass

$$dQ = du + pdv$$

Again

$$h = u + pv$$

∴

$$\begin{aligned} dh &= du + pdv + vdp \\ &= dQ + vdp \end{aligned}$$

∴

$$dQ = dh - vdp$$

∴

$$(dQ)_p = dh$$

or

$$(Q)_p = (\Delta h)_p$$

∴ From Eqs. (4.19) and (4.20)

$$(Q)_p = \int_{T_1}^{T_2} c_p \, dT$$

$c_p$  is a property of the system, just like  $c_v$ . The *heat capacity at constant pressure*  $C_p$  is equal to  $mc_p$  (J/K).

## 4.8 ENERGY OF AN ISOLATED SYSTEM

An isolated system is one in which there is no interaction of the system with the surroundings. For an isolated system,  $dQ = 0$ ,  $dW = 0$ .

The first law gives

$$dE = 0$$

or

$$E = \text{constant}$$

*The energy of an isolated system is always constant.*

## 4.9 PERPETUAL MOTION MACHINE OF THE FIRST KIND—PMM1

The first law states the general principle of the conservation of energy. *Energy is neither created nor destroyed, but only gets transformed from one form to another.* There can be no machine which would continuously supply mechanical work without some other form of energy disappearing simultaneously (Fig. 4.8). Such a *fictitious machine* is called a *perpetual motion machine of the first kind*, or in brief, PMM1. A PMM1 is thus impossible.

The converse of the above statement is also true, i.e. there can be no machine which would continuously consume work without some other form of energy appearing simultaneously (Fig. 4.9).

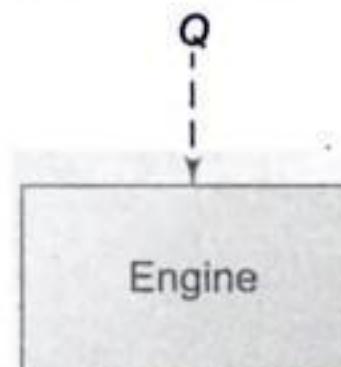


Fig. 4.8 A PMM1

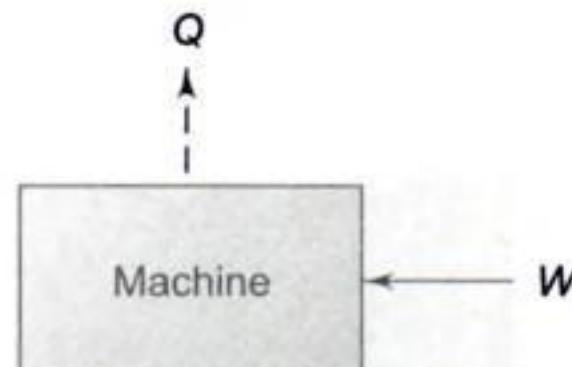


Fig. 4.9 The Converse of PMM1

## Solved Examples

**Example 4.1** A stationary mass of gas is compressed without friction from an initial state of  $0.3 \text{ m}^3$  and  $0.105 \text{ MPa}$  to a final state of  $0.15 \text{ m}^3$  and  $0.105 \text{ MPa}$ , the pressure remaining constant during the process. There is a transfer of  $37.6 \text{ kJ}$  of heat from the gas during the process. How much does the internal energy of the gas change?

**Solution** First law for a stationary system in a process gives

$$Q = \Delta U + W$$

or

$$Q_{1-2} = U_2 - U_1 + W_{1-2} \quad (1)$$

Here

$$\begin{aligned} W_{1-2} &= \int_{V_1}^{V_2} pdV = p(V_2 - V_1) \\ &= 0.105 (0.15 - 0.30) \text{ MJ} = -15.75 \text{ kJ} \\ Q_{1-2} &= -37.6 \text{ kJ} \end{aligned}$$

∴ Substituting in Eq. (1)

$$-37.6 \text{ kJ} = U_2 - U_1 - 15.75 \text{ kJ}$$

$$\therefore U_2 - U_1 = -21.85 \text{ kJ}$$

The internal energy of the gas decreases by  $21.85 \text{ kJ}$  in the process.

**Example 4.2** When a system is taken from state *a* to state *b*, in Fig. 4.10 along path *acb*,  $84 \text{ kJ}$  of heat flow into the system, and the system does  $32 \text{ kJ}$  of work. (a) How much will the heat that flows into the system along path *adb* be, if the work done is  $10.5 \text{ kJ}$ ? (b) When the system is returned from *b* to *a* along the curved path, the work done on the system is  $21 \text{ kJ}$ . Does the system absorb or liberate heat, and how much of the heat is absorbed or liberated? (c) If  $U_a = 0$  and  $U_d = 42 \text{ kJ}$ , find the heat absorbed in the processes *ad* and *db*.

**Solution**

$$Q_{acb} = 84 \text{ kJ}$$

$$W_{acb} = 32 \text{ kJ}$$

We have

$$Q_{acb} = U_b - U_a + W_{acb}$$

$$\therefore U_b - U_a = 84 - 32 = 52 \text{ kJ}$$

$$\begin{aligned} (a) \quad Q_{adb} &= U_b - U_a + W_{adb} \\ &= 52 + 10.5 = 62.5 \text{ kJ} \end{aligned}$$

$$\begin{aligned} (b) \quad Q_{b-a} &= U_a - U_b + W_{b-a} \\ &= -52 - 21 = -73 \text{ kJ} \end{aligned}$$

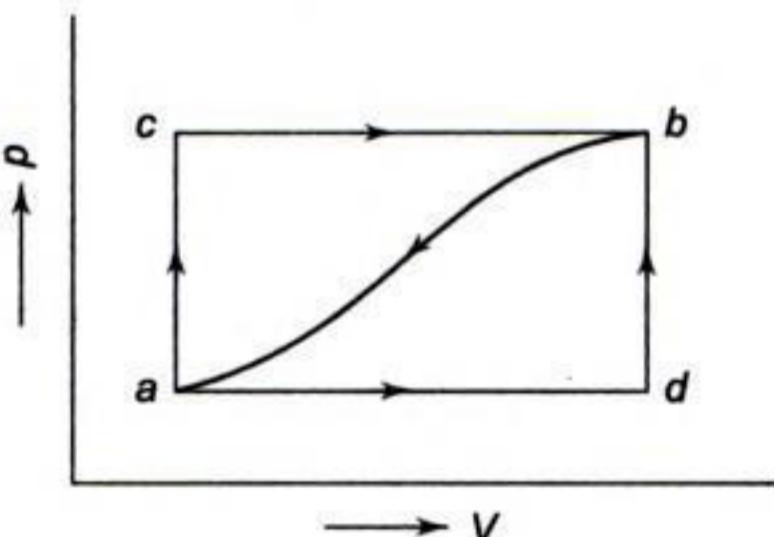


Fig. 4.10

The system liberates 73 kJ of heat

$$(c) \quad W_{adb} = W_{ad} + W_{db} = W_{ad} = 10.5 \text{ kJ}$$

$$\therefore Q_{ad} = U_d - U_a + W_{ad}$$

$$= 42 - 0 + 10.5 = 52.5 \text{ kJ}$$

$$\text{Now } Q_{adb} = 62.5 \text{ kJ} = Q_{ad} + Q_{db}$$

$$\therefore Q_{db} = 62.5 - 52.5 = 10 \text{ kJ}$$

**Example 4.3** A piston and cylinder machine contains a fluid system which passes through a complete cycle of four processes. During a cycle, the sum of all heat transfers is  $-170 \text{ kJ}$ . The system completes 100 cycles per min. Complete the following table showing the method for each item, and compute the net rate of work output in kW.

Process	$Q$ (kJ/min)	$W$ (kJ/min)	$\Delta E$ (kJ/min)
$a-b$	0	2,170	—
$b-c$	21,000	0	—
$c-d$	-2,100	—	-36,600
$d-a$	—	—	—

**Solution** Process  $a-b$ :

$$Q = \Delta E + W$$

$$0 = \Delta E + 2170$$

$$\therefore \Delta E = -2170 \text{ kJ/min}$$

Process  $b-c$ :

$$Q = \Delta E + W$$

$$21,000 = \Delta E + 0$$

$$\therefore \Delta E = 21,000 \text{ kJ/min}$$

Process  $c-d$ :

$$Q = \Delta E + W$$

$$-2100 = -36,600 + W$$

$$\therefore W = 34,500 \text{ kJ/min}$$

Process  $d-a$ :

$$\sum_{\text{cycle}} Q = -170 \text{ kJ}$$

The system completes 100 cycles/min.

$$\therefore Q_{ab} + Q_{bc} + Q_{cd} + Q_{da} = -17,000 \text{ kJ/min}$$

$$0 + 21,000 - 2100 + Q_{da} = -17,000$$

$$\therefore Q_{da} = -35,900 \text{ kJ/min}$$

Now  $\oint dE = 0$ , since cyclic integral of any property is zero.

$$\begin{aligned} \therefore \Delta E_{a-b} + \Delta E_{b-c} + \Delta E_{c-d} + \Delta E_{d-a} &= 0 \\ -2170 + 21,000 - 36,600 + \Delta E_{d-a} &= 0 \\ \therefore \Delta E_{d-a} &= 17,770 \text{ kJ/min} \\ \therefore W_{d-a} &= Q_{d-a} - \Delta E_{d-a} \\ &= -35,900 - 17,770 = -53,670 \text{ kJ/min} \end{aligned}$$

The table becomes

Process	$Q$ (kJ/min)	$W$ (kJ/min)	$\Delta E$ (kJ/min)
$a-b$	0	2170	-2170
$b-c$	21,000	0	21,000
$c-d$	-2100	34,500	-36,600
$d-a$	-35,900	-53,670	17,770

Since

$$\sum_{\text{cycle}} Q = \sum_{\text{cycle}} W$$

Rate of work output

$$= -17,000 \text{ kJ/min} = -283.3 \text{ kW}$$

**Example 4.4** The internal energy of a certain substance is given by the following equation

$$u = 3.56 pv + 84$$

where  $u$  is given in kJ/kg,  $p$  is in kPa, and  $v$  is in m<sup>3</sup>/kg.

A system composed of 3 kg of this substance expands from an initial pressure of 500 kPa and a volume of 0.22 m<sup>3</sup> to a final pressure 100 kPa in a process in which pressure and volume are related by  $pv^{1.2} = \text{constant}$ .

- If the expansion is quasi-static, find  $Q$ ,  $\Delta U$ , and  $W$  for the process.
- In another process the same system expands according to the same pressure-volume relationship as in part (a), and from the same initial state to the same final state as in part (a), but the heat transfer in this case is 30 kJ. Find the work transfer for this process.
- Explain the difference in work transfer in parts (a) and (b).

**Solution**

$$(a) u = 3.56 pv + 84$$

$$\Delta u = u_2 - u_1 = 3.56 (p_2 v_2 - p_1 v_1)$$

$$\therefore \Delta U = 3.56 (p_2 V_2 - p_1 V_1)$$

$$\text{Now } p_1 V_1^{1.2} = p_2 V_2^{1.2}$$

$$\begin{aligned} \therefore V_2 &= V_1 \left( \frac{p_1}{p_2} \right)^{1/1.2} = 0.22 \left( \frac{5}{1} \right)^{1/1.2} \\ &= 0.22 \times 3.83 = 0.845 \text{ m}^3 \end{aligned}$$

$$\begin{aligned} \therefore \Delta U &= 356 (1 \times 0.845 - 5 \times 0.22) \text{ kJ} \\ &= -356 \times 0.255 = -91 \text{ kJ} \end{aligned}$$

For a quasi-static process

$$\begin{aligned} W &= \int pdV = \frac{p_2 V_2 - p_1 V_1}{1-n} \\ &= \frac{(1 \times 0.845 - 5 \times 0.22) 100}{1-1.2} \\ &= 127.5 \text{ kJ} \end{aligned}$$

$$\therefore \begin{aligned} Q &= \Delta U + W \\ &= -91 + 127.5 = 36.5 \text{ kJ} \end{aligned}$$

(b) Here  $Q = 30 \text{ kJ}$

Since the end states are the same,  $\Delta U$  would remain the same as in (a).

$$\begin{aligned} \therefore W &= Q - \Delta U \\ &= 30 - (-91) = 121 \text{ kJ} \end{aligned}$$

(c) The work in (b) is not equal to  $\int pdV$  since the process is not quasi-static.

**Example 4.5** A fluid is confined in a cylinder by a spring-loaded, frictionless piston so that the pressure in the fluid is a linear function of the volume ( $p = a + bV$ ). The internal energy of the fluid is given by the following equation

$$U = 34 + 3.15 pV$$

where  $U$  is in kJ,  $p$  in kPa, and  $V$  in cubic metre. If the fluid changes from an initial state of 170 kPa, 0.03 m<sup>3</sup> to a final state of 400 kPa, 0.06 m<sup>3</sup>, with no work other than that done on the piston, find the direction and magnitude of the work and heat transfer.

**Solution** The change in the internal energy of the fluid during the process.

$$\begin{aligned} U_2 - U_1 &= 3.15 (p_2 V_2 - p_1 V_1) \\ &= 315 (4 \times 0.06 - 1.7 \times 0.03) \\ &= 315 \times 0.189 = 59.5 \text{ kJ} \end{aligned}$$

Now

$$\begin{aligned} p &= a + bV \\ 170 &= a + b \times 0.03 \\ 400 &= a + b \times 0.06 \end{aligned}$$

From these two equations

$$\begin{aligned} a &= -60 \text{ kN/m}^2 \\ b &= 7667 \text{ kN/m}^5 \end{aligned}$$

Work transfer involved during the process

$$W_{1-2} = \int_{V_1}^{V_2} pdV = \int_{V_1}^{V_2} (a + bV) dV$$

$$\begin{aligned}
 &= a(V_2 - V_1) + b \frac{V_2^2 - V_1^2}{2} \\
 &= (V_2 - V_1) \left[ a + \frac{b}{2}(V_1 + V_2) \right] \\
 &= 0.03 \text{ m}^3 \left[ -60 \text{ kN/m}^2 + \frac{7667}{2} \frac{\text{kN}}{\text{m}^5} \times 0.09 \text{ m}^3 \right] \\
 &= \mathbf{8.55 \text{ kJ}}
 \end{aligned}$$

Work is done by the system, the magnitude being 8.55 kJ.

∴ Heat transfer involved is given by

$$\begin{aligned}
 Q_{1-2} &= U_2 - U_1 + W_{1-2} \\
 &= 59.5 + 8.55 = \mathbf{68.05 \text{ kJ}}
 \end{aligned}$$

68.05 kJ of heat flow into the system during the process.

**Example 4.6** A stationary fluid system goes through a cycle shown in Fig. 4.11 comprising the following processes:

- (i) Process 1–2 isochoric heat addition of 235 kJ/kg;
- (ii) Process 2–3 adiabatic expansion to its original pressure with loss of 70 kJ/kg in internal energy;
- (iii) Process 3–1 isobaric compression to its original volume with heat rejection of 200 kJ/kg.

Prepare a balance sheet of energy quantities and find the overall changes during the cycle.

**Solution**

$$\begin{aligned}
 Q_{1-2} &= 235 \text{ kJ/kg}, W_{1-2} = 0, u_2 - u_1 \\
 &= Q_{1-2} - W_{1-2} = 235 \text{ kJ/kg}
 \end{aligned}$$

$$\begin{aligned}
 Q_{2-3} &= 0, u_3 - u_2 = -70 \text{ kJ/kg}; W_{2-3} \\
 &= Q_{2-3} - (u_3 - u_2) = 0 - (-70) \\
 &= 70 \text{ kJ/kg}
 \end{aligned}$$

$$\begin{aligned}
 Q_{3-1} &= -200 \text{ kJ/kg}, u_1 - u_3 = (u_1 - u_2) - (u_3 - u_2) = -235 + 70 \\
 &= -165 \text{ kJ/kg}
 \end{aligned}$$

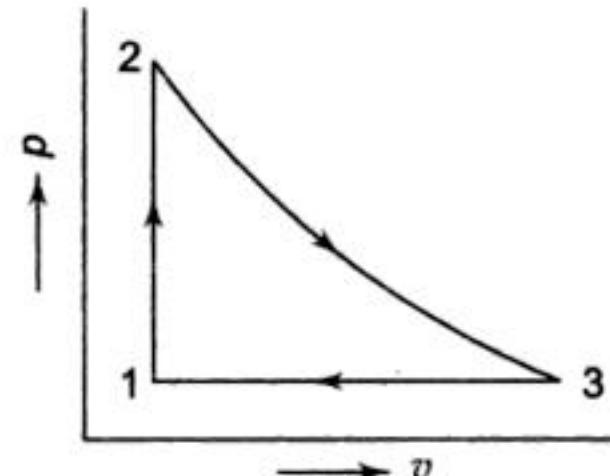
$$W_{3-1} = Q_{3-1} - (u_1 - u_3) = -200 - (-165) = -35 \text{ kJ/kg}$$

$$\sum_{\text{cycle}} Q = Q_{1-2} + Q_{2-3} + Q_{3-1} = 235 + 0 - 200 = \mathbf{35 \text{ kJ/kg}}$$

$$\sum_{\text{cycle}} W = W_{1-2} + W_{2-3} + W_{3-1} = 0 + 70 - 35 = \mathbf{35 \text{ kJ/kg}}$$

∴

$$\sum_{\text{cycle}} Q = \sum_{\text{cycle}} W$$



**Fig. 4.11**

This cycle is known as *Lenoir cycle*, and the operation of pulse jet approximates to it.

## Review Questions

- 4.1 State the first law for a closed system undergoing a cycle.
- 4.2 What was the contribution of J.P. Joule in establishing the first law?
- 4.3 What is the caloric theory of heat? Why was it rejected?
- 4.4 Which is the property introduced by the first law?
- 4.5 State the first law for a closed system undergoing a change of state.
- 4.6 Show that energy is a property of a system.
- 4.7 What are the modes in which energy is stored in a system?
- 4.8 Define internal energy. How is energy stored in molecules and atoms?
- 4.9 What is the difference between the standard symbols of  $E$  and  $U$ ?
- 4.10 What is the difference between heat and internal energy?
- 4.11 Define enthalpy. Why does the enthalpy of an ideal gas depend only on temperature?
- 4.12 Define the specific heats at constant volume and at constant pressure.
- 4.13 Why should specific heat not be defined in terms of heat transfer?
- 4.14 Which property of a system increases when heat is transferred: (a) at constant volume, (b) at constant pressure?
- 4.15 What is a PMM1? Why is it impossible?

## Problems

- 4.1 An engine is tested by means of a water brake at 1000 rpm. The measured torque of the engine is 10000 mN and the water consumption of the brake is  $0.5 \text{ m}^3/\text{s}$ , its inlet temperature being  $20^\circ\text{C}$ . Calculate the water temperature at exit, assuming that the whole of the engine power is ultimately transformed into heat which is absorbed by the cooling water. Ans.  $20.5^\circ\text{C}$
- 4.2 In a cyclic process, heat transfers are  $+ 14.7 \text{ kJ}$ ,  $- 25.2 \text{ kJ}$ ,  $- 3.56 \text{ kJ}$  and  $+ 31.5 \text{ kJ}$ . What is the net work for this cyclic process? Ans.  $17.34 \text{ kJ}$
- 4.3 A slow chemical reaction takes place in a fluid at the constant pressure of  $0.1 \text{ MPa}$ . The fluid is surrounded by a perfect heat insulator during the reaction which begins at state 1 and ends at state 2. The insulation is then removed and  $105 \text{ kJ}$  of heat flow to the surroundings as the fluid goes to state 3. The following data are observed for the fluid at states 1, 2 and 3.

<i>State</i>	$V(\text{m}^3)$	$t(\text{ }^\circ\text{C})$
1	0.003	20
2	0.3	370
3	0.06	20

For the fluid system, calculate  $E_2$  and  $E_3$ , if  $E_1 = 0$

Ans.  $E_2 = - 29.7 \text{ kJ}$ ,  $E_3 = - 110.7 \text{ kJ}$

- 4.4 During one cycle the working fluid in an engine engages in two work interactions: 15 kJ to the fluid and 44 kJ from the fluid, and three heat interactions, two of which are known: 75 kJ to the fluid and 40 kJ from the fluid. Evaluate the magnitude and direction of the third heat transfer. Ans. - 6 kJ
- 4.5 A domestic refrigerator is loaded with food and the door closed. During a certain period the machine consumes 1 kW h of energy and the internal energy of the system drops by 5000 kJ. Find the net heat transfer for the system. Ans. - 8.6 MJ
- 4.6 1.5 kg of liquid having a constant specific heat of 2.5 kJ/kg K is stirred in a well-insulated chamber causing the temperature to rise by 15°C. Find  $\Delta E$  and  $W$  for the process. Ans.  $\Delta E = 56.25 \text{ kJ}$ ,  $W = -56.25 \text{ kJ}$
- 4.7 The same liquid as in Problem 4.6 is stirred in a conducting chamber. During the process 1.7 kJ of heat are transferred from the liquid to the surroundings, while the temperature of the liquid is rising to 15°C. Find  $\Delta E$  and  $W$  for the process. Ans.  $\Delta E = 54.55 \text{ kJ}$ ,  $W = 56.25 \text{ kJ}$
- 4.8 The properties of a certain fluid are related as follows
- $$u = 196 + 0.718 t$$
- $$pv = 0.287 (t + 273)$$
- where  $u$  is the specific internal energy (kJ/kg),  $t$  is in °C,  $p$  is pressure (kN/m<sup>2</sup>), and  $v$  is specific volume (m<sup>3</sup>/kg). For this fluid, find  $c_v$  and  $c_p$ . Ans. 0.718, 1.005 kJ/kg K
- 4.9 A system composed of 2 kg of the above fluid expands in a frictionless piston and cylinder machine from an initial state of 1 MPa, 100°C to a final temperature of 30°C. If there is no heat transfer, find the net work for the process. Ans. 100.52 kJ
- 4.10 If all the work in the expansion of Problem 4.9 is done on the moving piston, show that the equation representing the path of the expansion in the  $pv$ -plane is given by  $pv^{1.4} = \text{constant}$ .
- 4.11 A stationary system consisting of 2 kg of the fluid of Problem 4.8 expands in an adiabatic process according to  $pv^{1.2} = \text{constant}$ . The initial conditions are 1 MPa and 200°C, and the final pressure is 0.1 MPa. Find  $W$  and  $\Delta E$  for the process. Why is the work transfer not equal to  $\int pdV$ ? Ans.  $W = 217.35$ ,  $\Delta E = -217.35 \text{ kJ}$ ,  $\int pdV = 434.4 \text{ kJ}$
- 4.12 A mixture of gases expands at constant pressure from 1 MPa, 0.03 m<sup>3</sup> to 0.06 m<sup>3</sup> with 84 kJ positive heat transfer. There is no work other than that done on a piston. Find  $\Delta E$  for the gaseous mixture. Ans. 54 kJ
- The same mixture expands through the same state path while a stirring device does 21 kJ of work on the system. Find  $\Delta E$ ,  $W$ , and  $Q$  for the process. Ans. 54 kJ, -21 kJ, 33 kJ
- 4.13 A mass of 8 kg gas expands within a flexible container so that the  $p-v$  relationship is of the form  $pv^{1.2} = \text{const}$ . The initial pressure is 1000 kPa and the initial volume is 1 m<sup>3</sup>. The final pressure is 5 kPa. If specific internal energy of the gas decreases by 40 kJ/kg, find the heat transfer in magnitude and direction. Ans. + 2615 kJ
- 4.14 A gas of mass 1.5 kg undergoes a quasi-static expansion which follows a relationship  $p = a + bV$ , where  $a$  and  $b$  are constants. The initial and final pressures

are 1000 kPa and 200 kPa respectively and the corresponding volumes are 0.20 m<sup>3</sup> and 1.20 m<sup>3</sup>. The specific internal energy of the gas is given by the relation

$$u = 1.5 pv - 85 \text{ kJ/kg}$$

where  $p$  is the kPa and  $v$  is in m<sup>3</sup>/kg. Calculate the net heat transfer and the maximum internal energy of the gas attained during expansion.

Ans. 660 kJ, 503.3 kJ

- 4.15 The heat capacity at constant pressure of a certain system is a function of temperature only and may be expressed as

$$C_p = 2.093 + \frac{41.87}{t + 100} \text{ J}/^\circ\text{C}$$

where  $t$  is the temperature of the system in °C. The system is heated while it is maintained at a pressure of 1 atmosphere until its volume increases from 2000 cm<sup>3</sup> to 2400 cm<sup>3</sup> and its temperature increases from 0°C to 100°C. (a) Find the magnitude of the heat interaction. (b) How much does the internal energy of the system increase?

Ans. (a) 238.32 J (b) 197.79 J

- 4.16 An imaginary engine receives heat and does work on a slowly moving piston at such rates that the cycle of operation of 1 kg of working fluid can be represented as a circle 10 cm in diameter on a  $p$ - $v$  diagram on which 1 cm = 300 kPa and 1 cm = 0.1 m<sup>3</sup>/kg. (a) How much work is done by each kg of working fluid for each cycle of operation? (b) The thermal efficiency of an engine is defined as the ratio of work done and heat input in a cycle. If the heat rejected by the engine in a cycle is 1000 kJ per kg of working fluid, what would be its thermal efficiency?

Ans. (a) 2356.19 kJ/kg, (b) 0.702

- 4.17 A gas undergoes a thermodynamic cycle consisting of three processes beginning at an initial state where  $p_1 = 1$  bar,  $V_1 = 1.5$  m<sup>3</sup> and  $U_1 = 512$  kJ. The processes are as follows:

- (i) Process 1–2: Compression with  $pV = \text{constant}$  to  $p_2 = 2$  bar,  $U_2 = 690$  kJ
- (ii) Process 2–3:  $W_{23} = 0$ ,  $Q_{23} = -150$  kJ, and
- (iii) Process 3–1:  $W_{31} = +50$  kJ. Neglecting KE and PE changes, determine the heat interactions  $Q_{12}$  and  $Q_{31}$ .

Ans. 74 kJ, 22 kJ

- 4.18 A gas undergoes a thermodynamic cycle consisting of the following processes: (i) Process 1–2: Constant pressure  $p = 1.4$  bar,  $V_1 = 0.028$  m<sup>3</sup>,  $W_{12} = 10.5$  kJ, (ii) Process 2–3: Compression with  $pV = \text{constant}$ ,  $U_3 = U_2$ , (iii) Process 3–1: Constant volume,  $U_1 - U_3 = -26.4$  kJ. There are no significant changes in KE and PE. (a) Sketch the cycle on a  $p$ - $V$  diagram. (b) Calculate the net work for the cycle in kJ. (c) Calculate the heat transfer for process 1–2 (d) Show that  $\sum Q = \sum W$ .

$\underset{\text{cycle}}{\text{cycle}}$

Ans. (b) – 8.28 kJ, (c) 36.9 kJ

- 4.19 A fluid contained in a cylinder receives 150 kJ of mechanical energy by means of a paddle wheel, together with 50 kJ in the form of heat. At the same time, a piston in the cylinder moves in such a way that the pressure remains constant at 200 kN/m<sup>2</sup> during the fluid expansion from 2m<sup>3</sup> to 5m<sup>3</sup>. What is the change in internal energy, and in enthalpy?

Ans. – 400 kJ, + 200 kJ

# 5

## CHAPTER

# First Law Applied to Flow Processes

### 5.1 CONTROL VOLUME

For any system and in any process, the first law can be written as

$$Q = \Delta E + W$$

where  $E$  represents all forms of energy stored in the system.

For a pure substance,

$$E = E_K + E_P + U$$

where  $E_K$  is the K.E.  $E_P$  the P.E. and  $U$  the residual energy stored in the molecular structure of the substance.

$$\therefore Q = \Delta E_K + \Delta E_P + \Delta U + W \quad (5.1)$$

When there is mass transfer across the system boundary, the system is called an open system. Most of the engineering devices are open systems involving the flow of fluids through them.

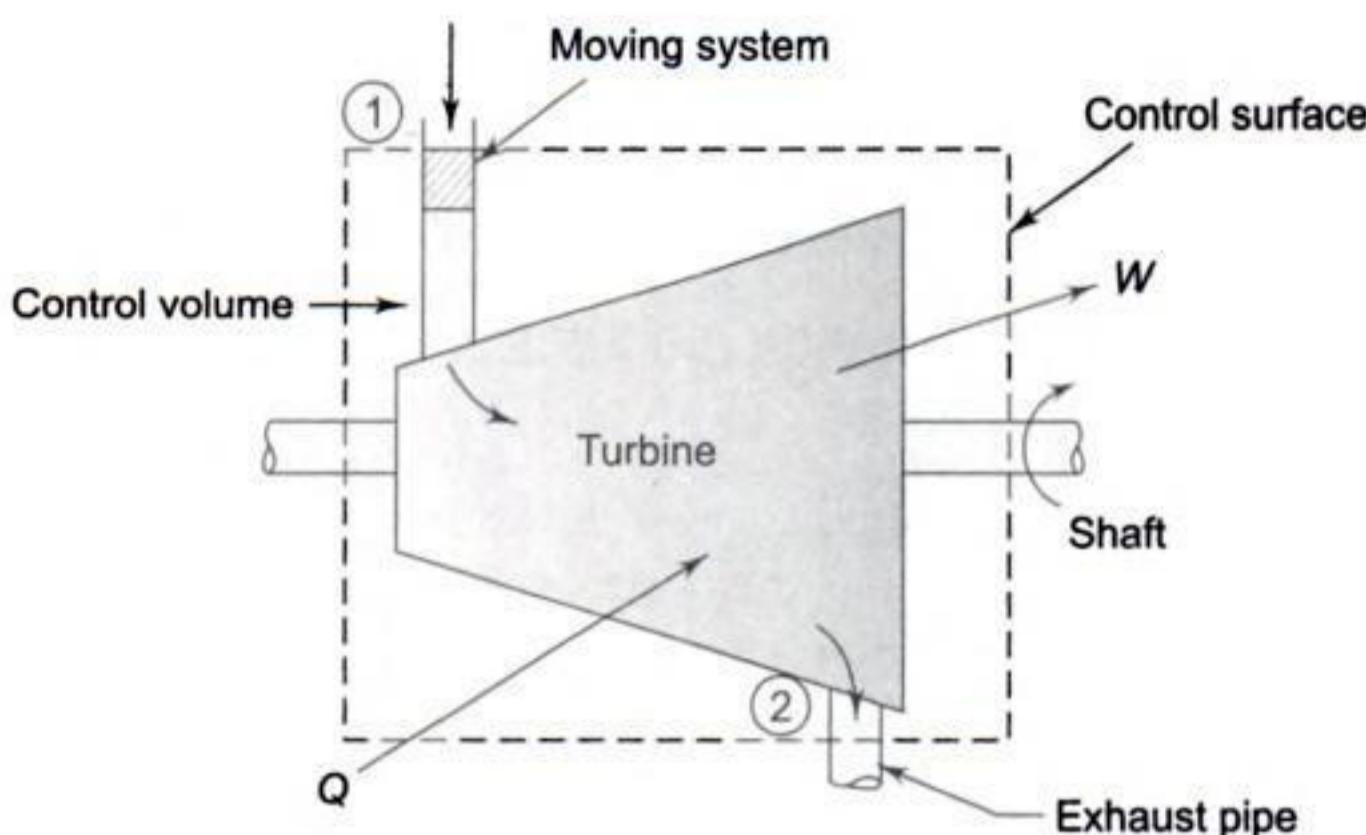
Equation (5.1) refers to a system having a particular mass of substance, and it is free to move from place to place.

Consider a steam turbine (Fig. 5.1) in which steam enters at a high pressure, does work upon the turbine rotor, and then leaves the turbine at low pressure through the exhaust pipe.

If a certain mass of steam is considered as the thermodynamic system, then the energy equation becomes

$$Q = \Delta E_K + \Delta E_P + \Delta U + W$$

and in order to analyze the expansion process in turbine the moving system is to be followed as it travels through the turbine, taking into account the work and heat interactions all the way through. This method of analysis is similar to that of Lagrange in fluid mechanics.



**Fig. 5.1** Flow Process Involving Work and Heat Interactions

Although the system approach is quite valid, there is another approach which is found to be highly convenient. Instead of concentrating attention upon a certain quantity of fluid, which constitutes a moving system in flow processes, attention is focused upon a certain fixed region in space called a *control volume* through which the moving substance flows. This is similar to the analysis of Euler in fluid mechanics.

To distinguish the two concepts, it may be noted that while the system (closed) boundary usually changes shape, position and orientation relative to the observer, the control volume boundary remains fixed and unaltered. Again, while matter usually crosses the control volume boundary, no such flow occurs across the system boundary.

The broken line in Fig. 5.1 represents the surface of the control volume which is known as the *control surface*. This is the same as the system boundary of the open system. The method of analysis is to inspect the control surface and account for all energy quantities transferred through this surface. Since there is mass transfer across the control surface, a mass balance also has to be made. Sections 1 and 2 allow mass transfer to take place, and  $Q$  and  $W$  are the heat and work interactions respectively.

## 5.2 STEADY FLOW PROCESS

As a fluid flows through a certain control volume, its thermodynamic properties may vary along the space coordinates as well as with time. If the rates of flow of mass and energy through the control surface change with time, the mass and energy within the control volume also would change with time.

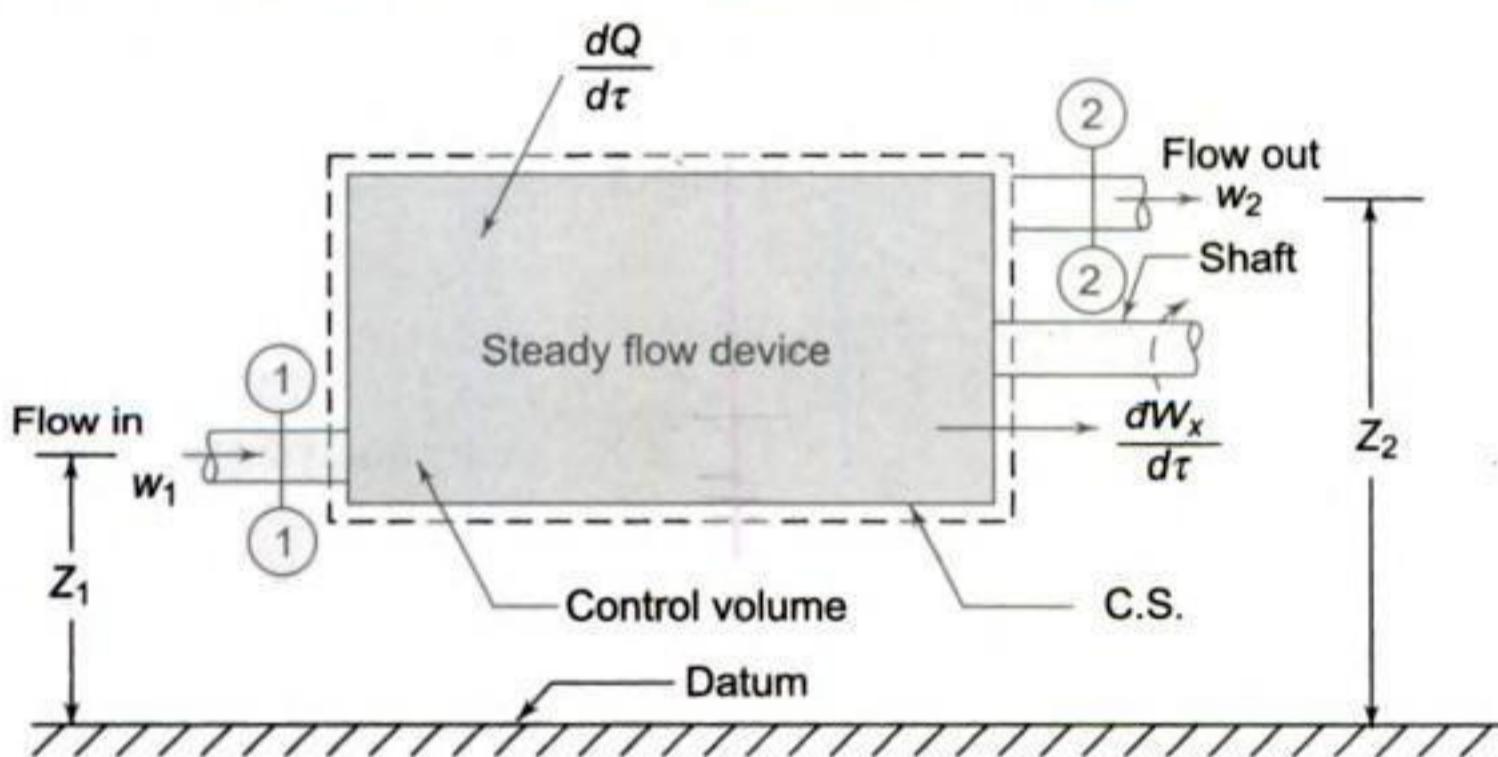
'Steady flow' means that the rates of flow of mass and energy across the control surface are constant.

In most engineering devices, there is a constant rate of flow of mass and energy through the control surface, and the control volume in course of time attains a

steady state. At the steady state of a system, any thermodynamic property will have a fixed value at a particular location, and will not alter with time. Thermodynamic properties may vary along space coordinates, but do not vary with time. 'Steady state' means that the state is steady or invariant with time.

### 5.3 MASS BALANCE AND ENERGY BALANCE IN A SIMPLE STEADY FLOW PROCESS

In Fig. 5.2, a steady flow system has been shown in which, one stream of fluid enters and another stream leaves the control volume. There is no accumulation of mass or energy within the control volume, and the properties at any location within the control volume are steady with time. Sections 1.1 and 2.2 indicate, respectively, the entrance and exit of the fluid across the control surface. The following quantities are defined with reference to Fig. 5.2



**Fig. 5.2** Steady Flow Process

$A_1, A_2$ —cross-section of stream,  $\text{m}^2$

$w_1, w_2$ —mass flow rate,  $\text{kg/s}$

$p_1, p_2$ —pressure, absolute,  $\text{N/m}^2$

$v_1, v_2$ —specific volume,  $\text{m}^3/\text{kg}$

$u_1, u_2$ —specific internal energy,  $\text{J/kg}$

$\mathbf{V}_1, \mathbf{V}_2$ —velocity,  $\text{m/s}$

$Z_1, Z_2$ —elevation above an arbitrary datum,  $\text{m}$

$\frac{dQ}{d\tau}$ —net rate of heat transfer through the control surface,  $\text{J/s}$

$\frac{dW_x}{d\tau}$ —net rate of work transfer through the control surface,  $\text{J/s}$

exclusive of work done at Sections 1 and 2 in transferring the fluid through the control surface.

$\tau$ —time,  $\text{s}$ .

Subscripts 1 and 2 refer to the inlet and exit sections.

### 5.3.1 Mass Balance

By the conservation of mass, if there is no accumulation of mass within the control volume, the mass flow rate entering must equal the mass flow rate leaving, or

$$w_1 = w_2$$

or

$$\frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2} \quad (5.2)$$

This equation is known as the *equation of continuity*.

### 5.3.2 Energy Balance

In a flow process, the work transfer may be of two types: the *external work* and the *flow work*. The external work refers to all the work transfer across the control surface other than that due to normal fluid forces. In engineering thermodynamics the only kinds of external work of importance are *shear* (shaft or stirring) *work* and *electrical work*. In Fig. 5.2 the only external work occurs in the form of shaft work,  $W_x$ . The flow work, as discussed in Sec. 3.4, is the displacement work done by the fluid of mass  $dm_1$  at the inlet Section 1 and that of mass  $dm_2$  at the exit Section 2, which are  $(-p_1 v_1 dm_1)$  and  $(+p_2 v_2 dm_2)$  respectively. Therefore, the total work transfer is given by

$$W = W_x - p_1 v_1 dm_1 + p_2 v_2 dm_2 \quad (5.3)$$

In the rate form,

$$\begin{aligned} \frac{dW}{d\tau} &= \frac{dW_x}{d\tau} - p_1 v_1 \frac{dm_1}{d\tau} + p_2 v_2 \frac{dm_2}{d\tau} \\ \text{or } \frac{dW}{d\tau} &= \frac{dW_x}{d\tau} - w_1 p_1 v_1 + w_2 p_2 v_2 \end{aligned} \quad (5.4)$$

Since there is no accumulation of energy, by the conservation of energy, the total rate of flow of all energy streams entering the control volume must equal the total rate of flow of all energy streams leaving the control volume. This may be expressed in the following equation

$$w_1 e_1 + \frac{dQ}{d\tau} = w_2 e_2 + \frac{dW}{d\tau}$$

Substituting for  $\frac{dW}{d\tau}$  from Eq. (5.4)

$$\begin{aligned} w_1 e_1 + \frac{dQ}{d\tau} &= w_2 e_2 + \frac{dW_x}{d\tau} - w_1 p_1 v_1 + w_2 p_2 v_2 \\ \therefore w_1 e_1 + w_1 p_1 v_1 + \frac{dQ}{d\tau} &= w_2 e_2 + w_2 p_2 v_2 + \frac{dW_x}{d\tau} \end{aligned} \quad (5.5)$$

where  $e_1$  and  $e_2$  refer to the energy carried into or out of the control volume with unit mass of fluid.

The specific energy  $e$  is given by

$$e = e_k + e_p + u$$

$$= \frac{\mathbf{V}^2}{2} + Zg + u \quad (5.6)$$

Substituting the expression for  $e$  in Equation (5.5)

$$\begin{aligned} w_1 \left( \frac{\mathbf{V}_1^2}{2} + Z_1 g + u_1 \right) + w_1 p_1 v_1 + \frac{dQ}{d\tau} \\ = w_2 \left( \frac{\mathbf{V}_2^2}{2} + Z_2 g + u_2 \right) + w_2 p_2 v_2 + \frac{dW_x}{d\tau} \\ \text{or} \\ w_1 \left( h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g \right) + \frac{dQ}{d\tau} \\ = w_2 \left( h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g \right) + \frac{dW_x}{d\tau} \end{aligned} \quad (5.7)$$

where  $h = u + pv$ .

And, since  $w_1 = w_2$ , let  $w = w_1 = w_2 = \frac{dm}{d\tau}$

Dividing Equation (5.7) by  $\frac{dm}{d\tau}$

$$\begin{aligned} h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g + \frac{dQ}{dm} \\ = h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g + \frac{dW_x}{dm} \end{aligned} \quad (5.8)$$

Equations (5.7) and (5.8) are known as *steady flow energy equations* (S.F.E.E.), for a single stream of fluid entering and a single stream of fluid leaving the control volume. All the terms in Equation (5.8) represent energy flow per unit mass of fluid (J/kg), whereas the terms in Equation (5.7) represent energy flow per unit time (J/s). The basis of energy flow per unit mass is usually more convenient when only a single stream of fluid enters and leaves a control volume. When more than one fluid stream is involved the basis of energy flow per unit time is more convenient.

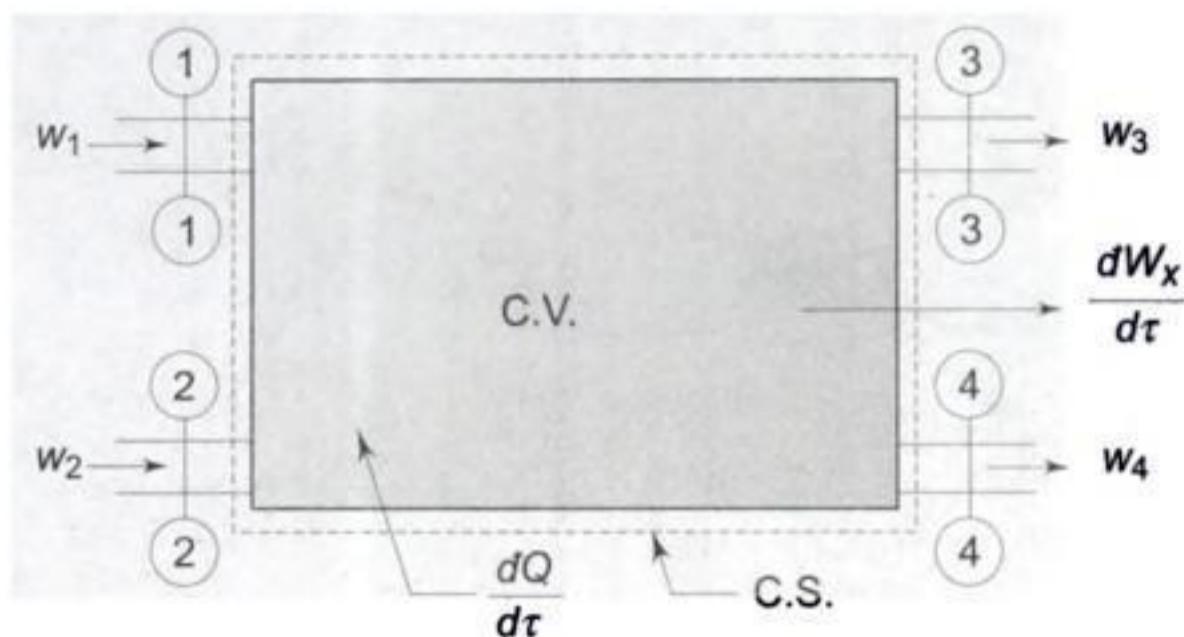
Equation (5.8) can be written in the following form,

$$Q - W_x = (h_2 - h_1) + \frac{\bar{\mathbf{V}}_2^2 - \bar{\mathbf{V}}_1^2}{2} + g(Z_2 - Z_1) \quad (5.9)$$

where  $Q$  and  $W_x$  refer to energy transfer per unit mass. In the differential form, the SFEE becomes

$$dQ - dW_x = dh + \bar{\mathbf{V}}d\bar{\mathbf{V}} + gdZ \quad (5.10)$$

When more than one stream of fluid enters or leaves the control volume (Fig. 5.3), the mass balance and energy balance for steady flow are given as follows.



**Fig. 5.3 Steady Flow Process Involving Two Fluid Streams at the Inlet and Exit of the Control Volume**

### Mass balance

$$w_1 + w_2 = w_3 + w_4 \quad (5.11)$$

$$\frac{A_1 \mathbf{V}_1}{v_1} + \frac{A_2 \mathbf{V}_2}{v_2} = \frac{A_3 \mathbf{V}_3}{v_3} + \frac{A_4 \mathbf{V}_4}{v_4} \quad (5.12)$$

### Energy balance

$$\begin{aligned} w_1 \left( h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g \right) + w_2 \left( h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g \right) + \frac{dQ}{dt} \\ = w_3 \left( h_3 + \frac{\mathbf{V}_3^2}{2} + Z_3 g \right) + w_4 \left( h_4 + \frac{\mathbf{V}_4^2}{2} + Z_4 g \right) + \frac{dW_x}{dt} \end{aligned} \quad (5.13)$$

The steady flow energy equation applies to a wide variety of processes like pipe line flows, heat transfer processes, mechanical power generation in engines and turbines, combustion processes, and flows through nozzles and diffusors. In certain problems, some of the terms in steady flow energy equation may be negligible or zero. But it is best to write the full equation first, and then eliminate the terms which are unnecessary.

## 5.4 SOME EXAMPLES OF STEADY FLOW PROCESSES

The following examples illustrate the applications of the steady flow energy equation in some of the engineering systems.

### 5.4.1 Nozzle and Diffusor

A nozzle is a device which increases the velocity or K.E. of a fluid at the expense of its pressure drop, whereas a diffusor increases the pressure of a fluid at the expense of its K.E. Figure 5.4 shows a nozzle which is insulated. The steady flow energy equation of the control surface gives

$$h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g + \frac{dQ}{dm} = h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g + \frac{dW_x}{dm}$$

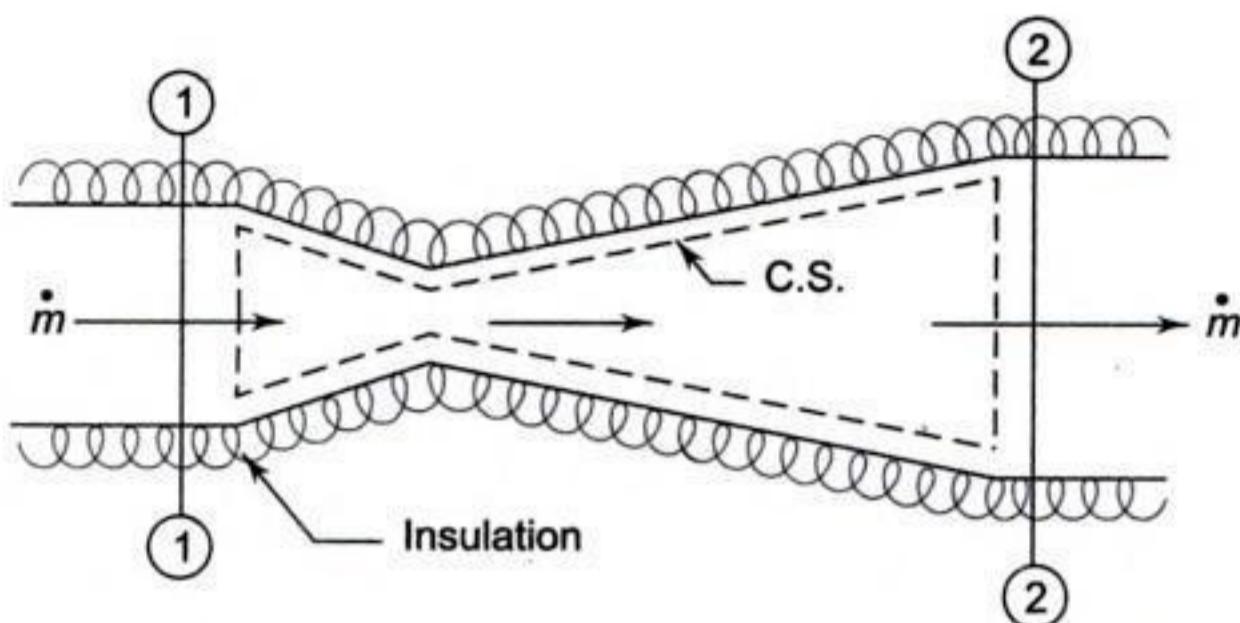


Fig. 5.4 Steady Flow Process Involving Two Fluid Streams at the Inlet and Exit of the Control Volume

Here  $\frac{dQ}{dm} = 0$ ,  $\frac{dW_x}{dm} = 0$ , and the change in potential energy is zero. The equation reduces to

$$h_1 + \frac{\mathbf{V}_1^2}{2} = h_2 + \frac{\mathbf{V}_2^2}{2} \quad (5.14)$$

The continuity equation gives

$$w = \frac{A_1 \mathbf{V}_1}{v_1} = \frac{A_2 \mathbf{V}_2}{v_2} \quad (5.15)$$

When the inlet velocity or the 'velocity of approach'  $\mathbf{V}_1$  is small compared to the exit velocity  $\mathbf{V}_2$ , Eq. (5.14) becomes

$$h_1 = h_2 + \frac{\mathbf{V}_2^2}{2}$$

or

$$\mathbf{V}_2 = \sqrt{2(h_1 - h_2)} \text{ m/s}$$

where  $(h_1 - h_2)$  is in J/kg.

Equations (5.14) and (5.15) hold good for a diffusor as well.

#### 5.4.2 Throttling Device

When a fluid flows through a constricted passage, like a partially opened valve, an orifice, or a porous plug, there is an appreciable drop in pressure, and the flow is said to be throttled. Figure 5.5 shows the process of throttling by a partially opened valve on a fluid flowing in an insulated pipe. In the steady-flow energy Equation (5.8),

$$\frac{dQ}{dm} = 0, \quad \frac{dW_x}{dm} = 0$$

and the changes in P.E. are very small and ignored. Thus, the S.F.E.E. reduces to

$$h_1 + \frac{\mathbf{V}_1^2}{2} = h_2 + \frac{\mathbf{V}_2^2}{2}$$

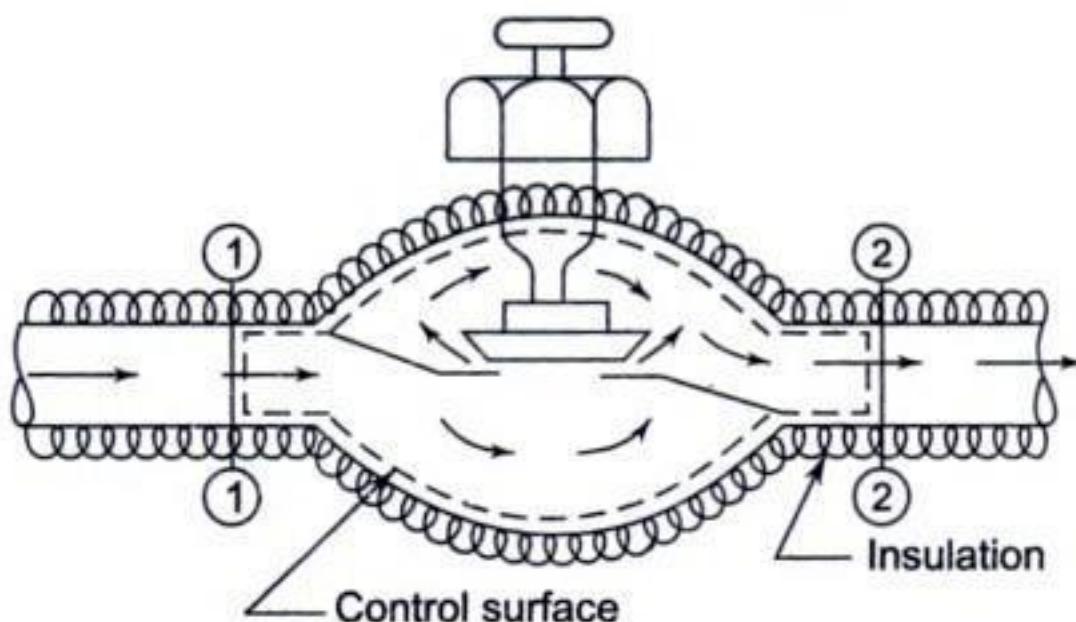


Fig. 5.5 Flow Through a Valve

Often the pipe velocities in throttling are so low that the K.E. terms are also negligible. So

$$h_1 = h_2 \quad (5.16)$$

or the enthalpy of the fluid before throttling is equal to the enthalpy of the fluid after throttling.

### 5.4.3 Turbine and Compressor

Turbines and engines give positive power output, whereas compressors and pumps require power input.

For a turbine (Fig. 5.6) which is well insulated, the flow velocities are often small, and the K.E. terms can be neglected. The S.F.E.E. then becomes

$$h_1 = h_2 + \frac{dW_x}{dm}$$

or  $\frac{W_x}{m} = (h_1 - h_2)$

It is seen that work is done by the fluid at the expense of its enthalpy.

Similarly, for an adiabatic pump or compressor, work is done upon the fluid and  $W$  is negative. So the S.F.E.E. becomes

$$h_1 = h_2 - \frac{W_x}{m}$$

or  $\frac{W_x}{m} = h_2 - h_1$

The enthalpy of the fluid increases by the amount of work input.

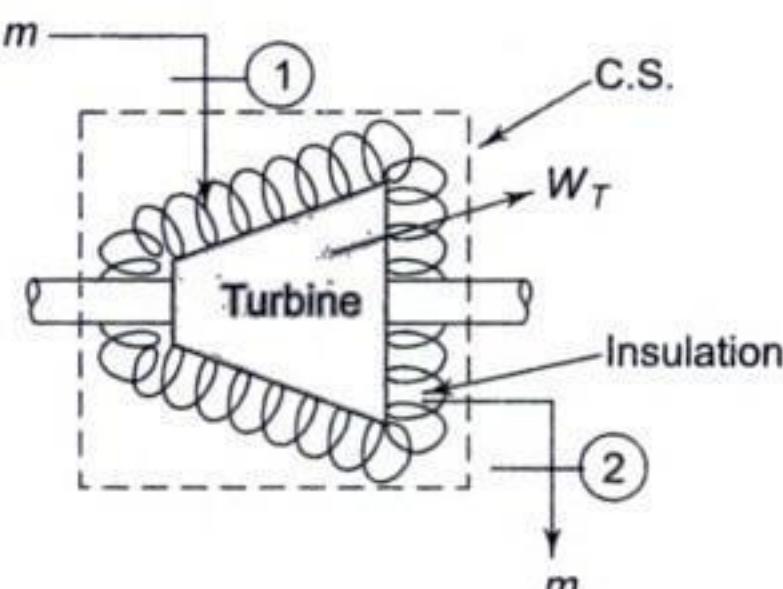


Fig. 5.6 Flow Through a Turbine

### 5.4.4 Heat Exchanger

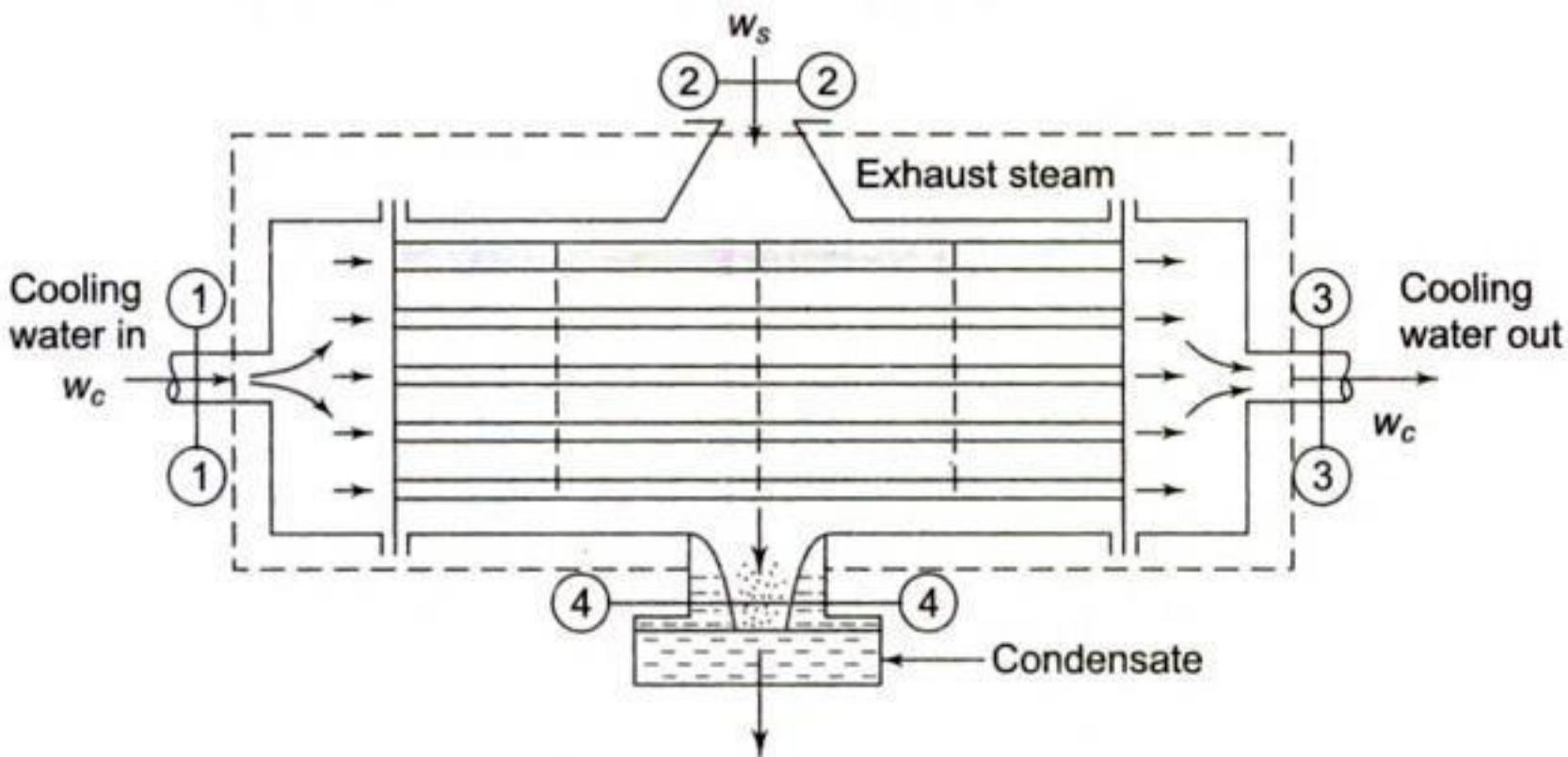
A heat exchanger is a device in which heat is transferred from one fluid to another. Figure 5.7 shows a steam condenser, where steam condensers outside the

tubes and cooling water flows through the tubes. The S.F.E.E. for the C.S. gives

$$w_c h_1 + w_s h_2 = w_c h_3 + w_s h_4$$

or  $w_s (h_2 - h_4) = w_c (h_3 - h_1)$

Here the K.E. and P.E. terms are considered small, there is no external work done, and energy exchange in the form of heat is confined only between the two fluids, i.e. there is no external heat interaction or heat loss.



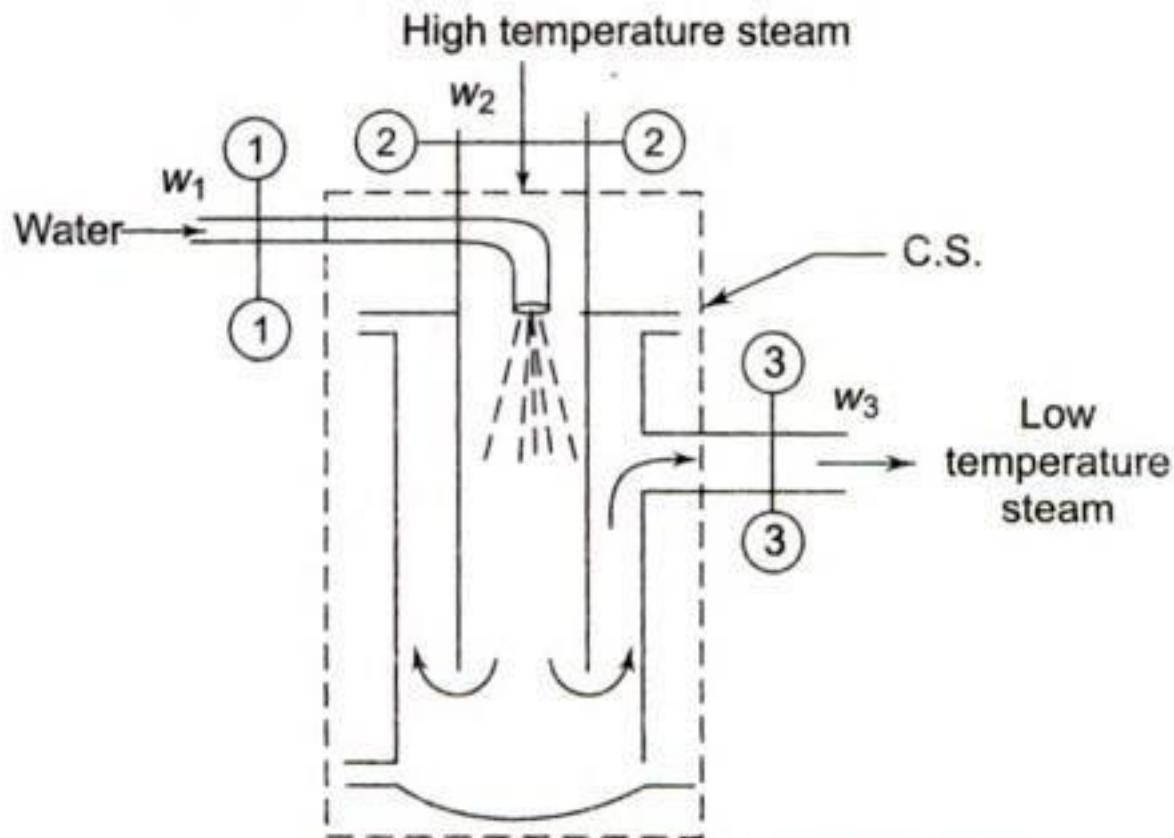
**Fig. 5.7** Steam Condenser

Figure 5.8 shows a steam desuperheater where the temperature of the superheated steam is reduced by spraying water. If  $w_1$ ,  $w_2$ , and  $w_3$  are the mass flow rates of the injected water, of the steam entering, and of the steam leaving, respectively, and  $h_1$ ,  $h_2$ , and  $h_3$  are the corresponding enthalpies, and if K.E. and P.E. terms are neglected as before, the S.F.E.E. becomes

$$w_1 h_1 + w_2 h_2 = w_3 h_3$$

and the mass balance gives

$$w_1 + w_2 = w_3$$



**Fig. 5.8** Steam Desuperheater

## 5.5 COMPARISON OF S.F.E.E. WITH EULER AND BERNOULLI EQUATIONS

The steady flow energy equation (Eq. 5.8) can be written as

$$\frac{dQ}{dm} = (h_2 - h_1) + \frac{\bar{\mathbf{V}}_2^2 - \bar{\mathbf{V}}_1^2}{2} + (Z_2 - Z_1) g + \frac{dW_x}{dm}$$

In the differential form the S.F.E.E. becomes

$$dQ = dh + \bar{\mathbf{V}} d\bar{\mathbf{V}} + gdZ + dW_x \quad (5.17)$$

where  $dQ$  and  $dW_x$  refer to unit mass of the substance. Since  $h = u + pv$  and  $dQ = du + pdv$  (for a quasi-static path involving only  $pdv$ -work), Eq. (5.17) can be written as

$$du + pdv = du + pdv + vdp + \bar{\mathbf{V}} d\bar{\mathbf{V}} + gdZ + dW_x$$

For an inviscid frictionless fluid flowing through a pipe

$$vdp + \bar{\mathbf{V}} d\bar{\mathbf{V}} + gdZ = 0 \quad (5.18)$$

This is the *Euler equation*. If we integrate between two Sections 1 and 2 of the pipe

$$\int_1^2 vdp + \int_1^2 \bar{\mathbf{V}} d\bar{\mathbf{V}} + \int_1^2 gdZ = 0$$

For an incompressible fluid,  $v = \text{constant}$

$$\therefore v(p_2 - p_1) + \frac{\bar{\mathbf{V}}_2^2}{2} - \frac{\bar{\mathbf{V}}_1^2}{2} + g(Z_2 - Z_1) = 0 \quad (5.19)$$

Since the specific volume  $v$  is the reciprocal of the density  $\rho$ , we have

$$\frac{p_1}{\rho} + \frac{\bar{\mathbf{V}}_1^2}{2} + Z_1g = \frac{p_2}{\rho} + \frac{\bar{\mathbf{V}}_2^2}{2} + Z_2g \quad (5.20)$$

or  $\frac{p}{\rho} + \frac{\bar{\mathbf{V}}^2}{2} + Zg = \text{constant} \quad (5.21)$

This is known as the *Bernoulli equation*, which is valid for an inviscid incompressible fluid. It can also be expressed in the following form

$$\Delta \left( p\mathbf{v} + \frac{\bar{\mathbf{V}}^2}{2} + gZ \right) = 0 \quad (5.22)$$

where  $v$  is constant and  $\Delta (\dots)$  means ‘increase in ...’

The S.F.E.E. as given by Eq. (5.8) or Eq. (5.17) can be written with  $(u + pv)$  substituted for  $h$ , as follows:

$$Q - W_x = \Delta \left( u + pv + \frac{\bar{\mathbf{V}}^2}{2} + gZ \right) \quad (5.23)$$

A comparison of Eqs (5.22) and (5.23) shows that they have several terms in common. However, while the Bernoulli equation is restricted to frictionless in-

compressible fluids, the S.F.E.E. is not, and is valid for viscous compressible fluids as well. The Bernoulli equation is, therefore, a special limiting case of the more general steady flow energy equation.

## 5.6 VARIABLE FLOW PROCESSES

Many flow processes, such as filling up and evacuating gas cylinders, are not steady. Such processes can be analyzed by the control volume technique. Consider a device through which a fluid is flowing under non-steady state conditions (Fig. 5.9). The rate at which the mass of fluid within the control volume is accumulated is equal to the net rate of mass flow across the control surface, as given below

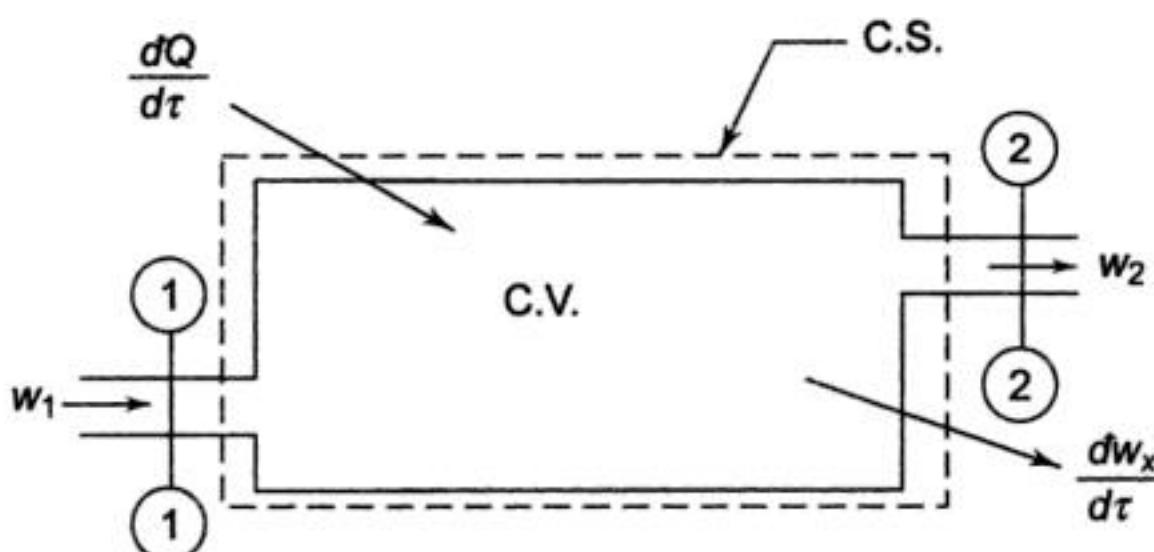


Fig. 5.9 Variable Flow Process

$$\frac{dm_V}{d\tau} = w_1 - w_2 = \frac{dm_1}{d\tau} - \frac{dm_2}{d\tau} \quad (5.24)$$

where  $m_V$  is the mass of fluid within the control volume at any instant.

Over any finite period of time

$$\Delta m_V = \Delta m_1 - \Delta m_2 \quad (5.25)$$

The rate of accumulation of energy within the control volume is equal to the net rate of energy flow across the control surface. If  $E_V$  is the energy of fluid within the control volume at any instant

Rate of energy increase = Rate of energy inflow – Rate of energy outflow

$$\begin{aligned} \frac{dE_V}{d\tau} &= w_1 \left( h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g \right) + \frac{dQ}{d\tau} \\ &\quad - w_2 \left( h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g \right) - \frac{dW_x}{d\tau} \end{aligned} \quad (5.26)$$

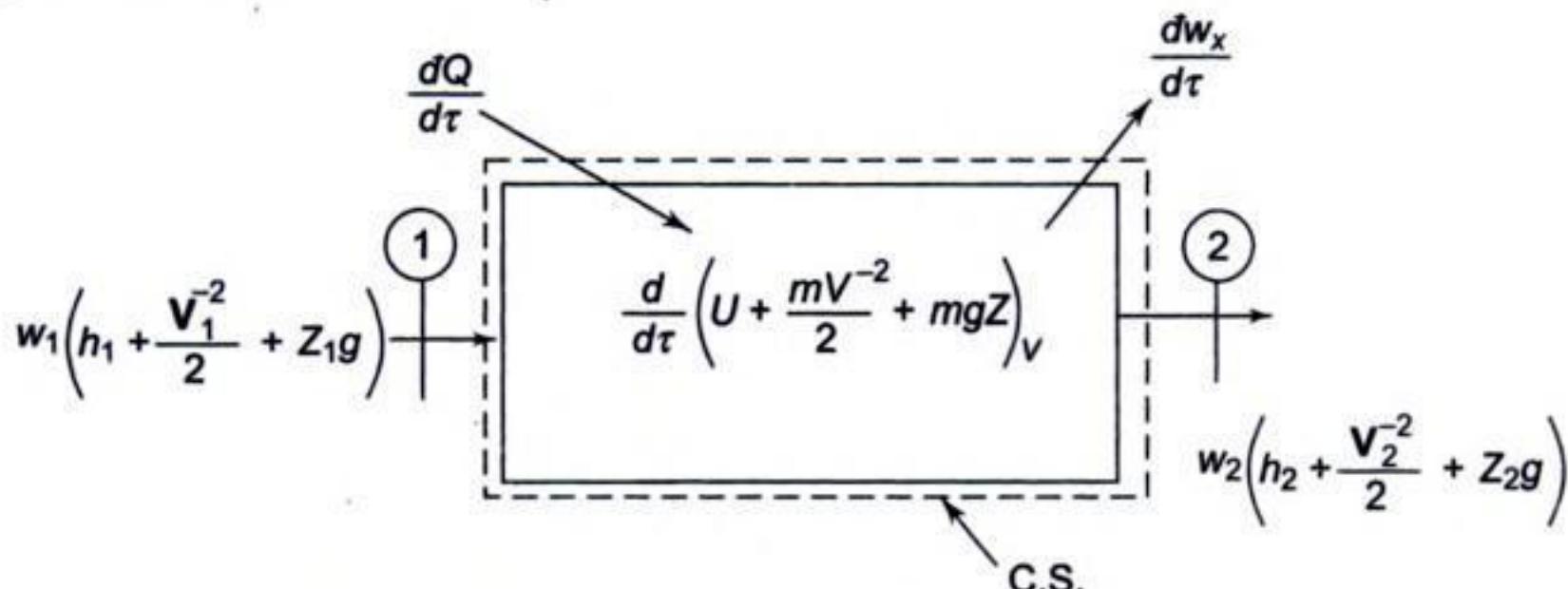
Now

$$E_V = \left( U + \frac{mV^2}{2} + mgZ \right)_V$$

where  $m$  is the mass of fluid in the control volume at any instant.

$$\therefore \frac{dE_V}{d\tau} = \frac{d}{d\tau} \left( U + \frac{mV^2}{2} + mgZ \right)_V \\ = \left( h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g \right) \frac{dm_1}{d\tau} + \frac{dQ}{d\tau} \\ - \left( h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g \right) \frac{dm_2}{d\tau} - \frac{dW_x}{d\tau} \quad (5.27)$$

Figure 5.10 shows all these energy flux quantities. For any finite time interval, Eq. (5.27) becomes



**Fig. 5.10 Energy Fluxes in an Unsteady System**

$$\Delta E_V = Q - W_x + \int \left( h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g \right) dm_1 \\ - \int \left( h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g \right) dm_2 \quad (5.28)$$

Equation (5.26) is the *general energy equation*. For steady flow,

$$\frac{dE_V}{d\tau} = 0,$$

and the equation reduces to Eq. (5.7). For a closed system  $w_1 = 0$ ,  $w_2 = 0$ , then from Eq. (5.26),

$$\frac{dE_V}{d\tau} = \frac{dQ}{d\tau} - \frac{dW_x}{d\tau}$$

or

$$dE_V = dQ - dW_x \quad \text{or,} \quad dQ = dE + dW_x$$

as obtained earlier.

## 5.7 EXAMPLE OF A VARIABLE FLOW PROBLEM

Variable flow processes may be analyzed either by the system technique or the control volume technique, as illustrated as follows.

Consider a process in which a gas bottle is filled from a pipeline (Fig. 5.11). In the beginning the bottle contains gas of mass  $m_1$  at state  $p_1, t_1, v_1, h_1$  and  $u_1$ . The valve is opened and gas flows into the bottle till the mass of gas in the bottle is  $m_2$  at state  $p_2, t_2, v_2, h_2$  and  $u_2$ . The supply to the pipeline is very large so that the state of gas in the pipeline is constant at  $p_p, t_p, v_p, h_p, u_p$ , and  $\mathbf{V}_p$ .

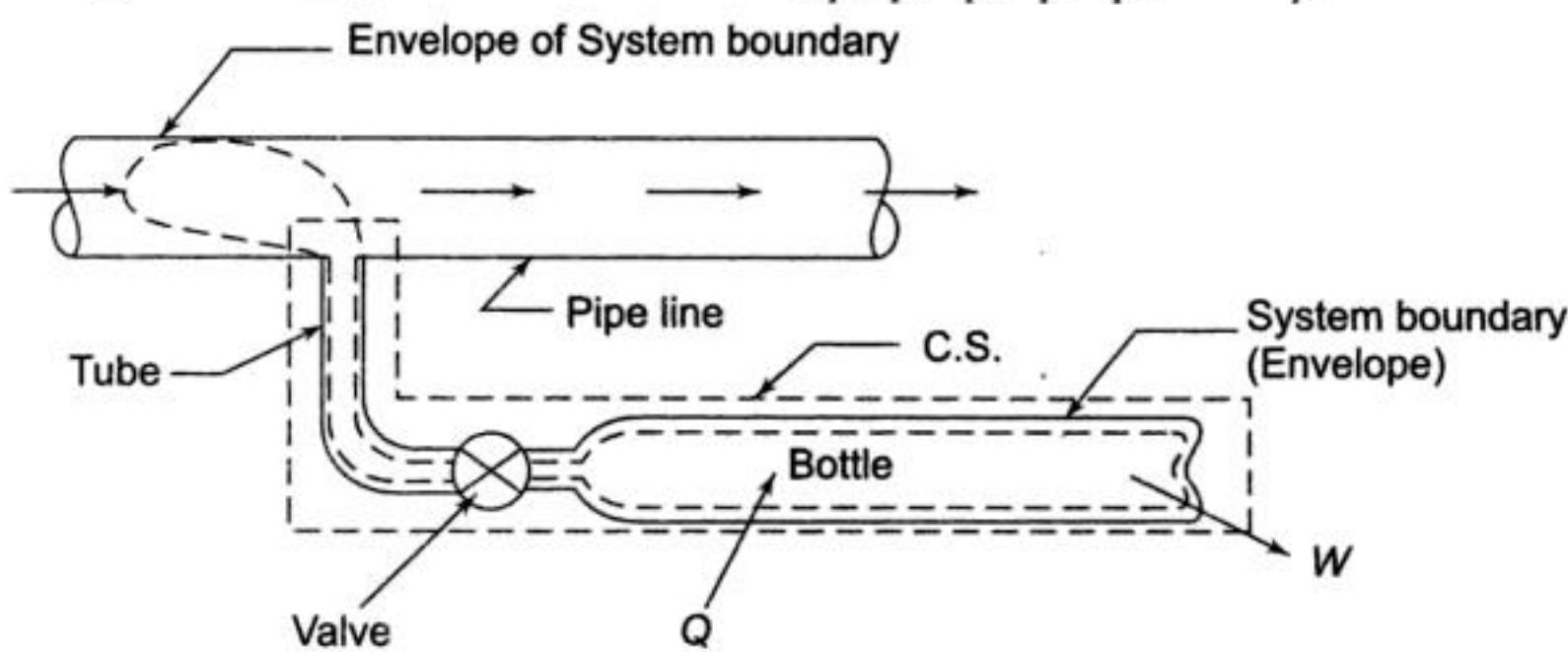


Fig. 5.11 Bottle-filling Process

### System Technique

Assume an envelope (which is extensible) of gas in the pipeline and the tube which would eventually enter the bottle, as shown in Fig. 5.11.

Energy of the gas before filling

$$E_1 = m_1 u_1 + (m_2 - m_1) \left( \frac{\mathbf{V}_p^2}{2} + u_p \right)$$

where  $(m_2 - m_1)$  is the mass of gas in the pipeline and tube which would enter the bottle.

Energy of the gas after filling

$$E_2 = m_2 u_2$$

$$\Delta E = E_2 - E_1 = m_2 u_2 - \left[ m_1 u_1 + (m_2 - m_1) \left( \frac{\mathbf{V}_p^2}{2} + u_p \right) \right] \quad (5.29)$$

The P.E. terms are neglected. The gas in the bottle is not in motion, and so the K.E. terms have been omitted.

Now, there is a change in the volume of gas because of the collapse of the envelope to zero volume. Then the work done

$$W = p_p(V_2 - V_1) = p_p[0 - (m_2 - m_1)v_p]$$

$$= -(m_2 - m_1)p_p v_p$$

$\therefore$  Using the first law for the process

$$Q = \Delta E + W$$

$$= m_2 u_2 - m_1 u_1 - (m_2 - m_1) \left( \frac{\mathbf{V}_p^2}{2} + u_p \right) - (m_2 - m_1)p_p v_p$$

$$= m_2 u_2 - m_1 u_1 - (m_2 - m_1) \left( \frac{\mathbf{V}_p^2}{2} + h_p \right) \quad (5.30)$$

which gives the energy balance for the process.

### Control Volume Technique

Assume a control volume bounded by a control surface, as shown in Fig. 5.11. Applying the energy Eq. (5.27) to this case, the following energy balance may be written on a time rate basis

$$\frac{dE_V}{d\tau} = \frac{dQ}{d\tau} + \left( h_p + \frac{\mathbf{V}_p^2}{2} \right) \frac{dm}{d\tau}$$

Since  $h_p$  and  $\mathbf{V}_p$  are constant, the equation is integrated to give for the total process

$$\Delta E_V = Q + \left( h_p + \frac{\mathbf{V}_p^2}{2} \right) (m_2 - m_1)$$

Now

$$\Delta E_V = U_2 - U_1 = m_2 u_2 - m_1 u_1$$

$$\therefore Q = m_2 u_2 - m_1 u_1 - \left( h_p + \frac{\mathbf{V}_p^2}{2} \right) (m_2 - m_1)$$

This equation is the same as Eq. (5.30).

## 5.8 DISCHARGING AND CHARGING A TANK

Let us consider a tank discharging a fluid into a supply line (Fig. 5.12). Since  $dW_x = 0$  and  $dm_{in} = 0$ , applying first law to the control volume,

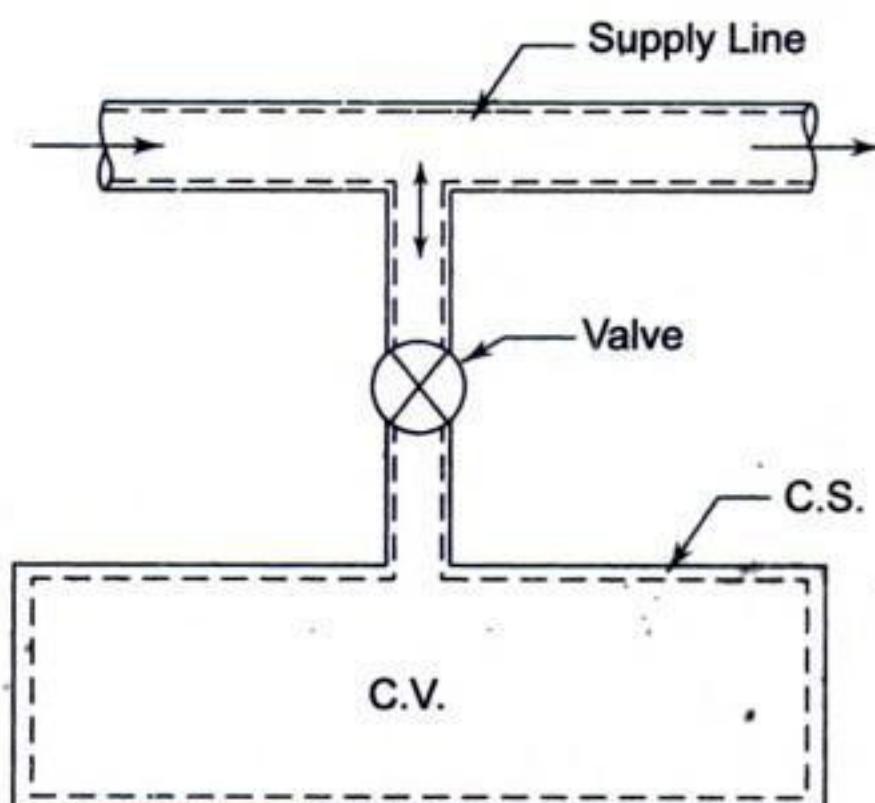


Fig. 5.12 Charging and Discharging a Tank

$$dU_V = \cancel{dQ} + \left( h + \frac{V^2}{2} + gz \right)_{\text{out}} dm_{\text{out}} \quad (5.31)$$

Assuming K.E. and P.E. of the fluid to be small and  $\cancel{dQ} = 0$

$$d(mu) = hdm$$

$$mdu + udm = udm + pv dm$$

$$\frac{dm}{m} = \frac{du}{pv} \quad (5.32)$$

Again

$$V = vm = \text{const.}$$

$$vdm + mdv = 0$$

or

$$\frac{dm}{m} = -\frac{dv}{v} \quad (5.33)$$

From Eqs (5.32) and 5.33),

$$\frac{du}{pv} = -\frac{dv}{v}$$

$$d(u + pv) = 0$$

or

$$\cancel{dQ} = 0$$

which shows that the process is adiabatic and quasi-static.

For charging the tank

$$\int (hdm)_{\text{in}} = \Delta U_V = m_2 u_2 - m_1 u_1 \quad (5.34)$$

$$m_p h_p = m_2 u_2 - m_1 u_1$$

where the subscript  $p$  refers to the constant state of the fluid in the pipeline. If the tank is initially empty,  $m_1 = 0$ .

$$m_p h_p = m_2 u_2$$

Since

$$m_p = m_2$$

$$h_p = u_2$$

If the fluid is an ideal gas, the temperature of the gas in the tank after it is charged is given by

$$c_p T_p = c_v T_2$$

or

$$T_2 = \gamma T_p \quad (5.35)$$

### Solved Examples

**Example 5.1** Air flows steadily at the rate of 0.5 kg/s through an air compressor, entering at 7 m/s velocity, 100 kPa pressure, and  $0.95 \text{ m}^3/\text{kg}$  volume, and

leaving at 5 m/s, 700 kPa, and  $0.19 \text{ m}^3/\text{kg}$ . The internal energy of the air leaving is 90 kJ/kg greater than that of the air entering. Cooling water in the compressor jackets absorbs heat from the air at the rate of 58 kW. (a) Compute the rate of shaft work input to the air in kW. (b) Find the ratio of the inlet pipe diameter to outlet pipe diameter.

**Solution** Figure 5.13 shows the details of the problem.

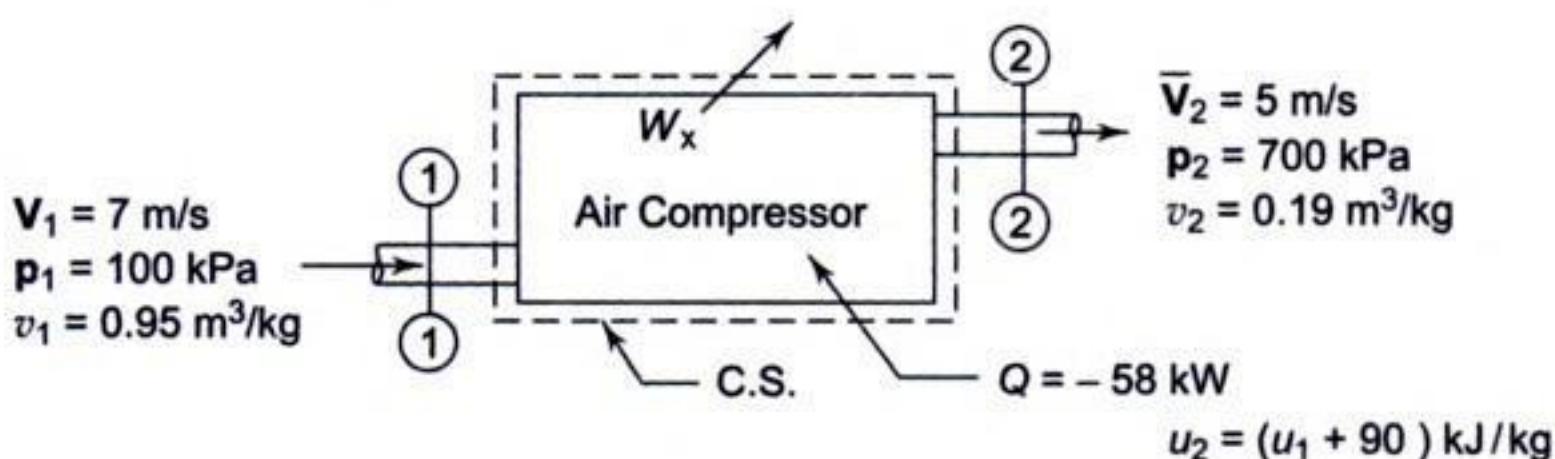


Fig. 5.13

(a) Writing the steady flow energy equation, we have

$$w \left( u_1 + p_1 v_1 + \frac{V_1^2}{2} + Z_1 g \right) + \frac{dQ}{dt} = w \left( u_2 + p_2 v_2 + \frac{V_2^2}{2} + Z_2 g \right) + \frac{dW_x}{dt}$$

$$\therefore \frac{dW_x}{dt} = -w \left( (u_2 - u_1) + (p_2 v_2 - p_1 v_1) + \frac{V_2^2 - V_1^2}{2} + (Z_2 - Z_1)g \right) \frac{dQ}{dt}$$

$$\therefore \frac{dW_x}{dt} = -0.5 \frac{\text{kg}}{\text{s}} \left[ 90 \frac{\text{kJ}}{\text{kg}} + (7 \times 0.19 - 1 \times 0.95) 100 \frac{\text{kJ}}{\text{kg}}$$

$$+ \frac{(5^2 - 7^2) \times 10^{-3}}{2} \frac{\text{kJ}}{\text{kg}} + 0 \right] - 58 \text{ kW}$$

$$= -0.5 [90 + 38 - 0.012] \text{ kJ/s} - 58 \text{ kW}$$

$$= -122 \text{ kW}$$

Rate of work input is **122 kW**.

(b) From mass balance, we have

$$w = \frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2}$$

$$\therefore \frac{A_1}{A_2} = \frac{v_1}{v_2} \cdot \frac{V_2}{V_1} = \frac{0.95}{0.19} \times \frac{5}{7} = 3.57$$

$$\therefore \frac{d_1}{d_2} = \sqrt{3.57} = 1.89$$

**Example 5.2** In a steady flow apparatus, 135 kJ of work is done by each kg of fluid. The specific volume of the fluid, pressure, and velocity at the inlet are  $0.37 \text{ m}^3/\text{kg}$ , 600 kPa, and 16 m/s. The inlet is 32 m above the floor, and the discharge pipe is at floor level. The discharge conditions are  $0.62 \text{ m}^3/\text{kg}$ , 100 kPa, and 270 m/s. The total heat loss between the inlet and discharge is 9 kJ/kg of fluid. In flowing through this apparatus, does the specific internal energy increase or decrease, and by how much?

**Solution** Writing the steady flow energy equation for the control volume, as shown in Fig. 5.14.

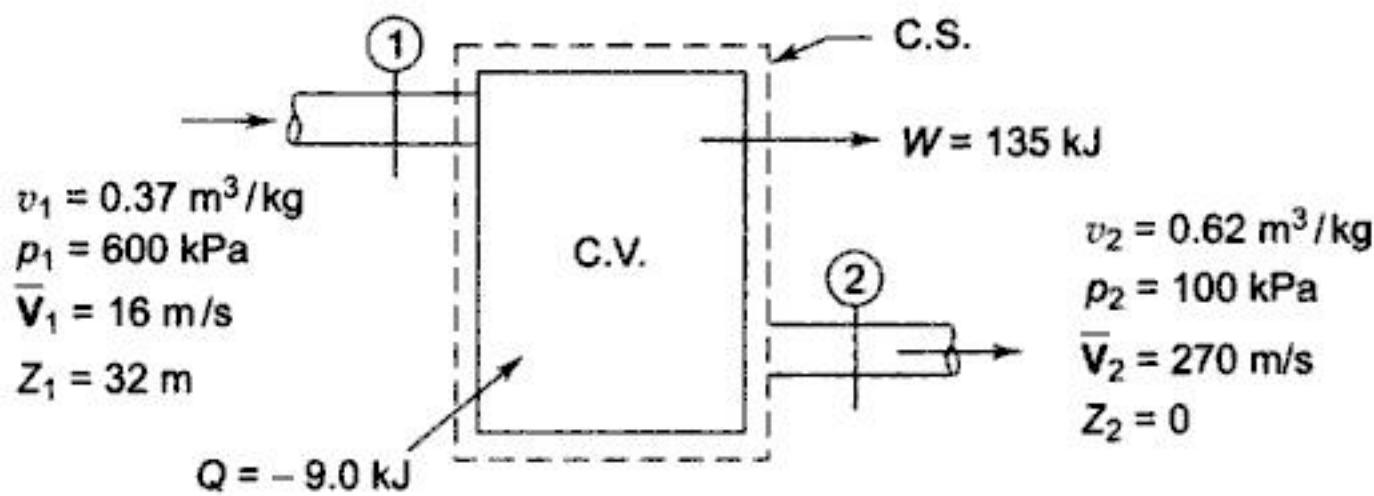


Fig. 5.14

$$\begin{aligned}
 u_1 + p_1 v_1 &= \frac{V_1^2}{2} + Z_1 g + \frac{dQ}{dm} = u_2 + p_2 v_2 + \frac{V_2^2}{2} + Z_2 g + \frac{dW_x}{dm} \\
 \therefore u_1 - u_2 &= (p_2 v_2 - p_1 v_1) + \frac{V_2^2 - V_1^2}{2} + (Z_2 - Z_1)g + \frac{dW_x}{dm} - \frac{dQ}{dm} \\
 &= (1 \times 0.62 - 6 \times 0.37) \times 10^2 + \frac{(270^2 - 16^2) \times 10^{-3}}{2} \\
 &\quad + (-32 \times 9.81 \times 10^{-3}) + 135 - (-9.0) \\
 &= -160 + 36.45 - 0.314 + 135 + 9 \\
 &= 20.136 \text{ kJ/kg}
 \end{aligned}$$

Specific internal energy decreases by **20.136 kJ**.

**Example 5.3** In a steam power station, steam flows steadily through a 0.2 m diameter pipeline from the boiler to the turbine. At the boiler end, the steam conditions are found to be:  $p = 4 \text{ MPa}$ ,  $t = 400^\circ\text{C}$ ,  $h = 3213.6 \text{ kJ/kg}$ , and  $v = 0.073 \text{ m}^3/\text{kg}$ . At the turbine end, the conditions are found to be:  $p = 3.5 \text{ MPa}$ ,  $t = 392^\circ\text{C}$ ,  $h = 3202.6 \text{ kJ/kg}$ , and  $v = 0.084 \text{ m}^3/\text{kg}$ . There is a heat loss of 8.5 kJ/kg from the pipeline. Calculate the steam flow rate.

**Solution** Writing the steady flow energy equation for the control volume as shown in Fig. 5.15.

$$h_1 + \frac{V_1^2}{2} + Z_1 g + \frac{dQ}{dm} = h_2 + \frac{V_2^2}{2} + Z_2 g + \frac{dW_x}{dm}$$

Here, there is no change in datum, so change in potential energy will be zero.

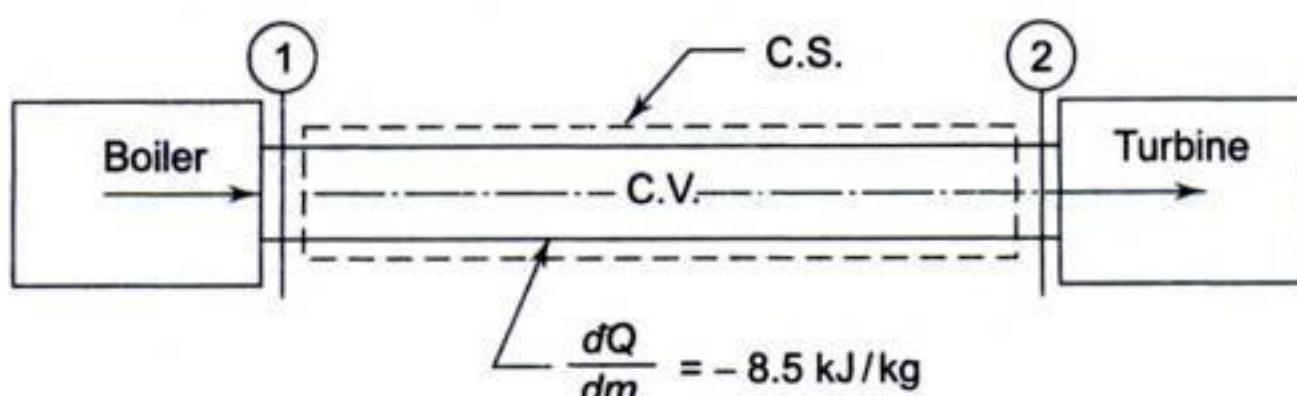


Fig. 5.15

Now

$$\frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2}$$

$$\therefore V_2 = \frac{A_1 V_1}{v_1} \cdot \frac{v_2}{A_2} = \frac{v_2}{v_1} \cdot V_1 = \frac{0.084}{0.073} V_1 = 1.15 V_1$$

and

$$\frac{dW_x}{dm} = 0$$

$$h_1 + \frac{V_1^2}{2} + \frac{dQ}{dm} = h_2 + \frac{V_2^2}{2}$$

$$\begin{aligned} \therefore \frac{(V_2^2 - V_1^2) \times 10^{-3}}{2} &= h_1 - h_2 + \frac{dQ}{dm} \\ &= 3213.6 - 3202.6 + (-8.5) = 2.5 \text{ kJ/kg} \end{aligned}$$

$$V_1^2 (1.15^2 - 1^2) = 5 \times 10^3$$

$$V_1^2 = 15,650 \text{ m}^2/\text{s}^2$$

$$V_1 = 125.1 \text{ m/s}$$

$$\therefore \text{Mass flow rate } w = \frac{A_1 V_1}{v_1} = \frac{\frac{\pi}{4} \times (0.2)^2 \text{ m}^2 \times 125.1 \text{ m/s}}{0.073 \text{ m}^3/\text{kg}} = 53.8 \text{ kg/s}$$

**Example 5.4** A certain water heater operates under steady flow conditions receiving 4.2 kg/s of water at 75°C temperature, enthalpy 313.93 kJ/kg. The water is heated by mixing with steam which is supplied to the heater at temperature 100.2°C and enthalpy 2676 kJ/kg. The mixture leaves the heater as liquid water at temperature 100°C and enthalpy 419 kJ/kg. How much steam must be supplied to the heater per hour?

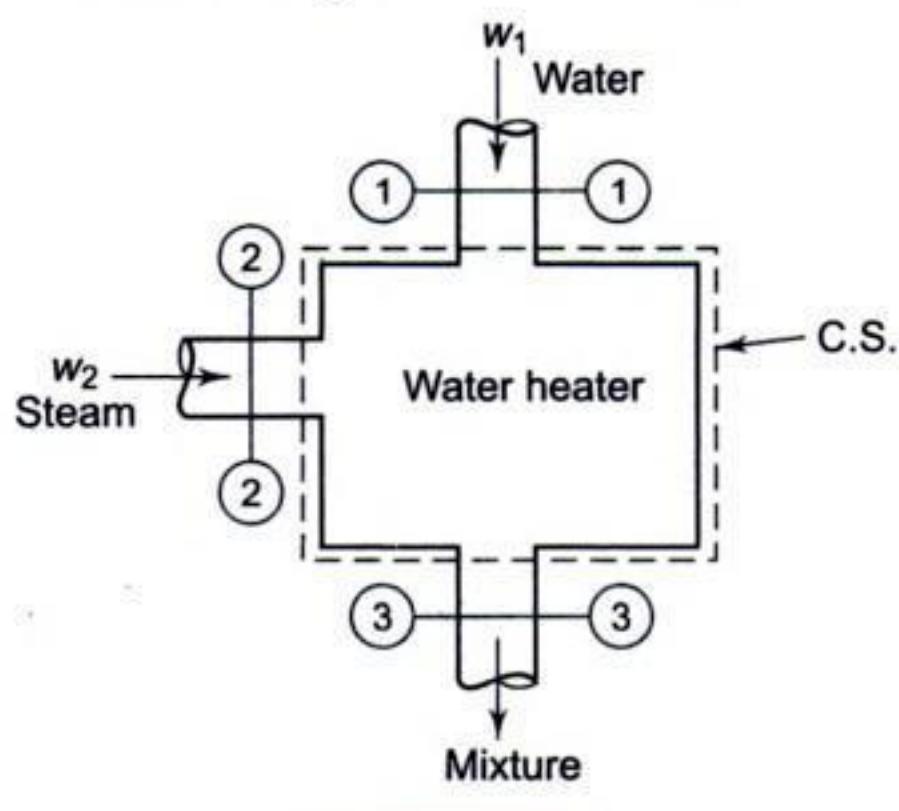


Fig. 5.16

**Solution** By mass balance across the control surface (Fig. 5.16)

$$w_1 + w_2 = w_3$$

By energy balance

$$\begin{aligned} & w_1 \left( h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g \right) + \frac{dQ}{d\tau} + w_2 \left( h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g \right) \\ &= w_3 \left( h_3 + \frac{\mathbf{V}_3^2}{2} + Z_3 g \right) + \frac{dW_x}{d\tau} \end{aligned}$$

By the nature of the process, there is no shaft work. Potential and kinetic energy terms are assumed to balance zero. The heater is assumed to be insulated. So the steady flow energy equation reduces to

$$w_1 h_1 + w_2 h_2 = w_3 h_3$$

$$4.2 \times 313.93 + w_2 \times 2676 = (4.2 + w_2) 419$$

∴

$$w_2 = 0.196 \text{ kg/s} = 705 \text{ kg/h}$$

**Example 5.5** Air at a temperature of 15°C passes through a heat exchanger at a velocity of 30 m/s where its temperature is raised to 800°C. It then enters a turbine with the same velocity of 30 m/s and expands until the temperature falls to 650°C. On leaving the turbine, the air is taken at a velocity of 60 m/s to a nozzle where it expands until the temperature has fallen to 500°C. If the air flow rate is 2 kg/s, calculate (a) the rate of heat transfer to the air in the heat exchanger, (b) the power output from the turbine assuming no heat loss, and (c) the velocity at exit from the nozzle, assuming no heat loss. Take the enthalpy of air as  $h = c_p t$ , where  $c_p$  is the specific heat equal to 1.005 kJ/kg K and  $t$  is the temperature.

**Solution** As shown in Fig. 5.17, writing the S.F.E.E. for the heat exchanger and eliminating the terms not relevant,

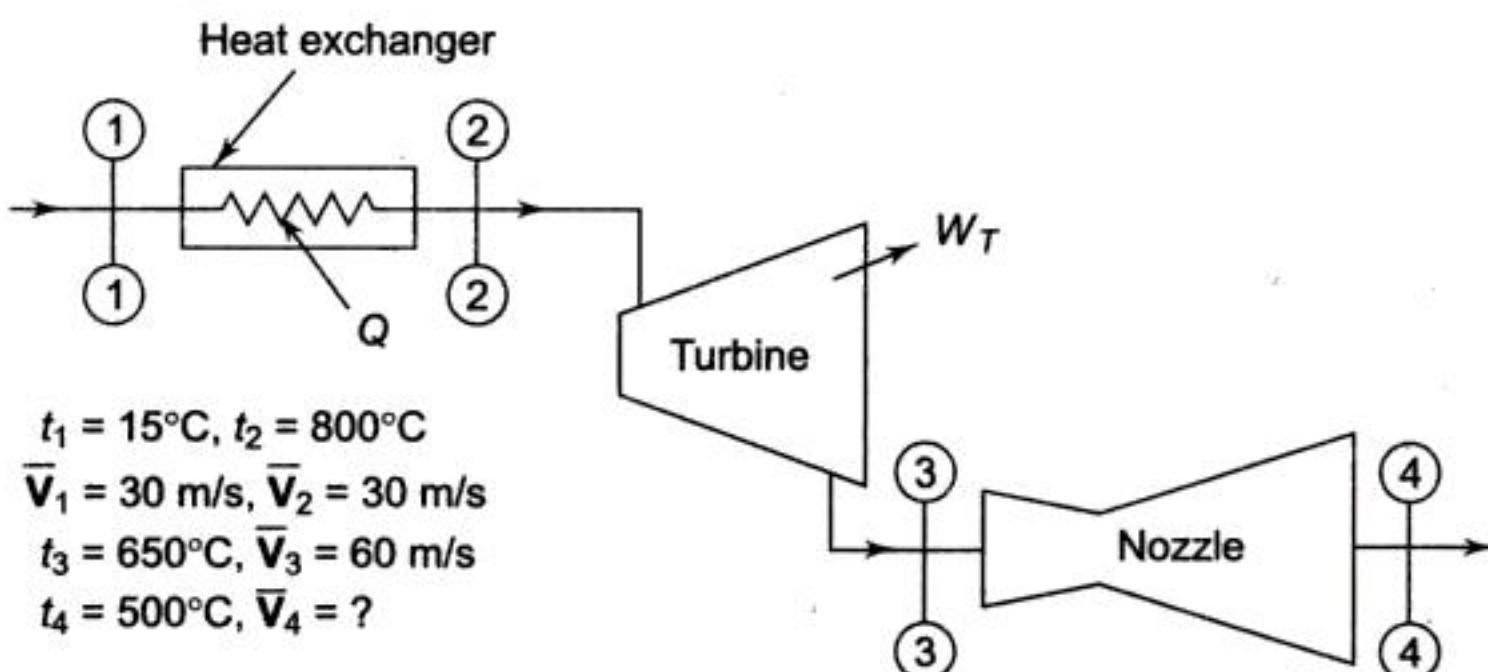


Fig. 5.17

$$w \left( h_1 + \frac{\mathbf{V}_1^2}{2} + Z_1 g \right) + Q_{1-2} = w \left( h_2 + \frac{\mathbf{V}_2^2}{2} + Z_2 g \right) + W_{1-2}$$

∴

$$wh_1 + Q_{1-2} = wh_2$$

$$\therefore Q_{1-2} = w(h_2 - h_1) = w c_p(t_2 - t_1)$$

$$= 2 \times 1.005 (800 - 15)$$

$$= 2.01 \times 785$$

$$= \mathbf{1580 \text{ kJ/s}}$$

Energy equation for the turbine gives

$$w \left( \frac{\mathbf{V}_2^2}{2} + h_2 \right) = wh_3 + w \frac{\mathbf{V}_3^2}{2} + W_T$$

$$\frac{\mathbf{V}_2^2 - \mathbf{V}_3^2}{2} + (h_2 - h_3) = W_T/w$$

$$\frac{(30^2 - 60^2) \times 10^{-3}}{2} + 1.005 (800 - 650) = W_T/w$$

$$\therefore \frac{W_T}{w} = 1.35 + 150.75$$

$$= 149.4 \text{ kJ/kg}$$

$$\therefore W_T = 149.4 \times 2 \text{ kJ/s}$$

$$= \mathbf{298.8 \text{ kW}}$$

Writing the energy equation for the nozzle

$$\frac{\mathbf{V}_3^3}{2} + h_3 = \frac{\mathbf{V}_4^2}{2} + h_4$$

$$\frac{\mathbf{V}_4^2 - \mathbf{V}_3^2}{2} = c_p (t_3 - t_4)$$

$$\mathbf{V}_4^2 - \mathbf{V}_3^2 = 1.005 (650 - 500) \times 2 \times 10^3$$

$$= 301.50 \times 10^3 \text{ m}^2/\text{s}^2$$

$$\mathbf{V}_4^2 = 30.15 \times 10^4 + 0.36 \times 10^4$$

$$= 30.51 \times 10^4 \text{ m}^2/\text{s}^2$$

$\therefore$  Velocity at exit from the nozzle

$$\mathbf{V}_4 = 554 \text{ m/s}$$

**Example 5.6** In a gas turbine the gas enters at the rate of 5 kg/s with a velocity of 50 m/s and enthalpy of 900 kJ/kg and leaves the turbine with a velocity of 150 m/s and enthalpy of 400 kJ/kg. The loss of heat from the gases to the surroundings is 25 kJ/kg. Assume for gas  $R = 0.285 \text{ kJ/kg K}$  and  $c_p = 1.004 \text{ kJ/kg K}$  and the inlet conditions to be at 100 kPa and 27°C. Determine the power output of the turbine and the diameter of the inlet pipe.

**Solution** Steady flow energy equation for the C.V. gives

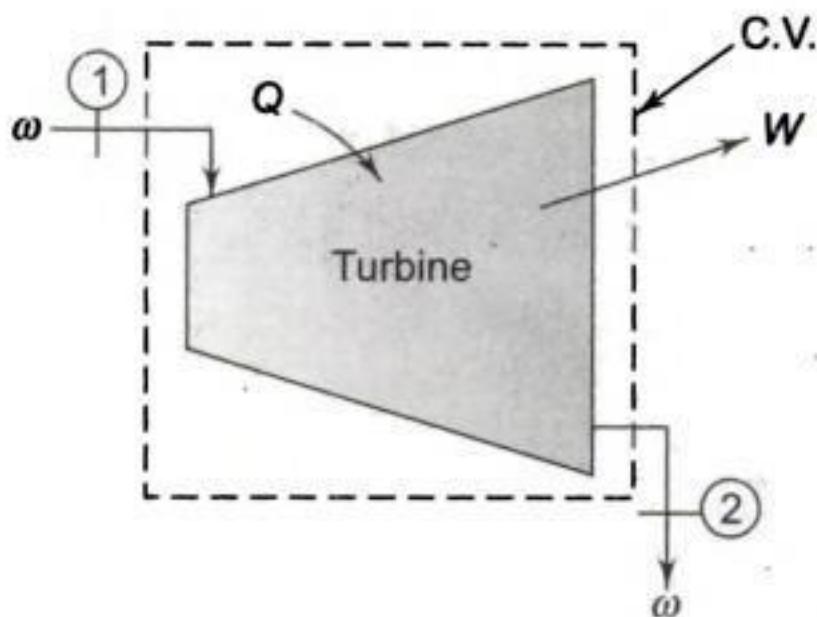


Fig. 5.18

$$w \left( h_1 + \frac{V_1^2}{2} + gZ_1 \right) + \frac{dQ}{d\tau} = w \left( h_2 + \frac{V_2^2}{2} + gZ_2 \right) + \frac{dW}{d\tau}$$

$$\therefore \frac{dW}{d\tau} = w \left[ (h_1 - h_2) + \frac{V_1^2 - V_2^2}{2} \right] + \frac{dQ}{d\tau}$$

$$= 5 \left[ \left( 900 - 400 + \frac{50^2 - 150^2}{2} \times 10^{-3} \right) \right] - 25 \times 5$$

$$= 2325 \text{ kW}$$

Using ideal gas equation of state at pipe inlet,

$$p_1 \dot{V}_1 = w_1 R T_1$$

$$\dot{V} = \text{volume flow rate at inlet} = \frac{5 \times 0.285 \times 300 \times 10^3}{100 \times 10^3} \\ = 4.275 \text{ m}^3/\text{s}$$

$$\text{Inlet area, } A_1 = \frac{4.275}{50}$$

$$= 0.086 \text{ m}^2 = \frac{\pi}{4} D_1^2$$

$$\therefore D_1 = 0.33 \text{ m or } 33 \text{ cm}$$

**Example 5.7** The air speed of a turbojet engine in flight is 270 m/s. Ambient air temperature is  $-15^\circ\text{C}$ . Gas temperature of outlet of nozzle is  $600^\circ\text{C}$ . Corresponding enthalpy values for air and gas are respectively 260 and 912 kJ/kg. Fuel-air ratio is 0.0190. Chemical energy of the fuel is 44.5 MJ/kg. Owing to incomplete combustion 5% of the chemical energy is not released in the reaction. Heat loss from the engine is 21 kJ/kg of air. Calculate the velocity of the exhaust jet.

**Solution** Energy equation for the turbojet engine (Fig. 5.19) gives

$$w_a \left( h_a + \frac{\mathbf{V}_a^2}{2} \right) + w_f E_f + Q = w_g \left( h_g + \frac{\mathbf{V}_g^2}{2} + E_g \right)$$

$$\left( 260 + \frac{270^2 \times 10^{-3}}{2} \right) + 0.0190 \times 44500 - 21$$

$$= 1.0190 \left( 912 + \frac{\mathbf{V}_g^2 \times 10^{-3}}{2} + 0.05 \frac{0.019}{1.019} \times 44500 \right)$$

$$260 + 36.5 + 845 - 21 = 1.019 \left( 912 + \frac{\mathbf{V}_g^2 \times 10^{-3}}{2} + 42 \right)$$

$$\therefore \frac{\mathbf{V}_g^2}{2} = 156 \times 10^3 \text{ m}^2/\text{s}^2$$

$$\mathbf{V}_g = \sqrt{3.12} \times 100 \text{ m/s}$$

Velocity of exhaust gas,  $\mathbf{V}_g = 560 \text{ m/s}$

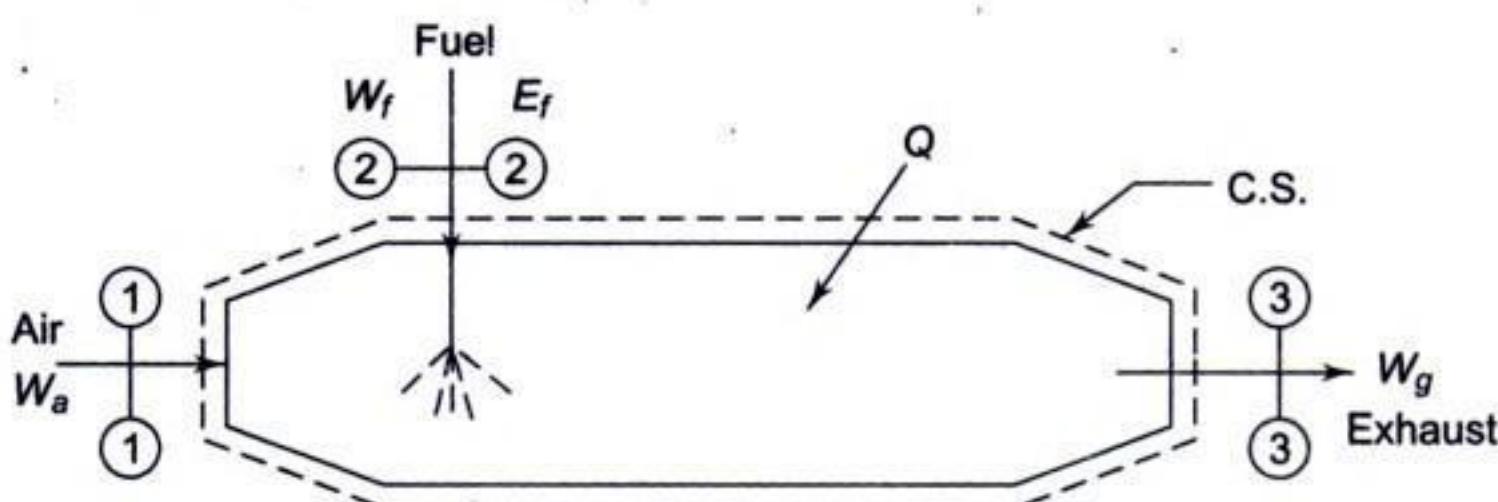


Fig. 5.19

**Example 5.8** In a reciprocating engine, the mass of gas occupying the clearance volume is  $m_c$  kg at state  $p_1, u_1, v_1$  and  $h_1$ . By opening the inlet valve,  $m_f$  kg of gas is taken into the cylinder, and at the conclusion of the intake process the state of the gas is given by  $p_2, u_2, v_2, h_2$ . The state of the gas in the supply pipe is constant and is given by  $p_p, u_p, v_p, h_p, \mathbf{V}_p$ . How much heat is transferred between the gas and the cylinder walls during the intake process?

**Solution** Let us consider the control volume as shown in Fig. 5.20. Writing the energy balance on a time rate basis

$$\frac{dE_V}{d\tau} = \frac{dQ}{d\tau} - \frac{dW}{d\tau} + \left( h_p + \frac{\mathbf{V}_p^2}{2} \right) \frac{dm}{d\tau}$$

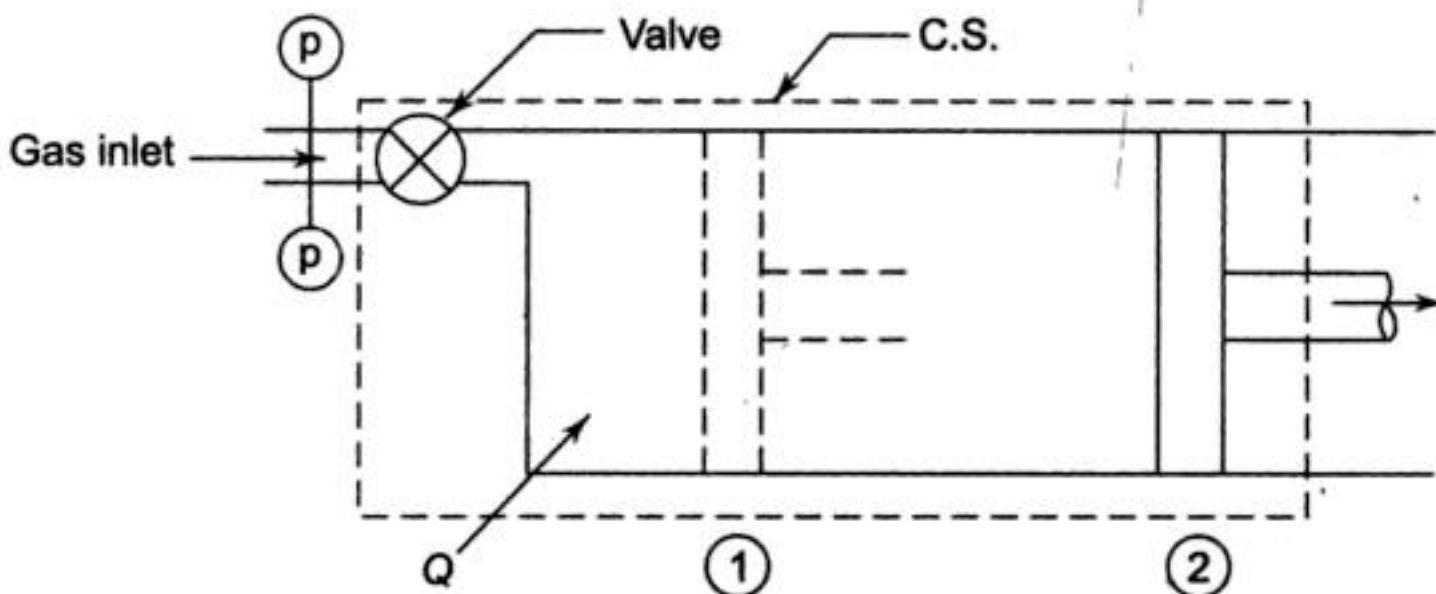


Fig. 5.20

With  $h_p$  and  $\mathbf{V}_p$  being constant, the above equation can be integrated to give for the total process

$$\Delta E_V = Q - W + \left( h_p + \frac{\mathbf{V}_p^2}{2} \right) m_f$$

Now

$$\Delta E_v = U_2 - U_1 = (m_e + m_f) u_2 - m_e u_1$$

$$\therefore Q = (m_e + m_f) u_2 - m_e u_1 - m_f \left( h_p + \frac{\mathbf{V}_p^2}{2} \right) + W$$

**Example 5.9** The internal energy of air is given by

$$u = u_0 + 0.718 t$$

where  $u$  is in kJ/kg,  $u_0$  is any arbitrary value of  $u$  at  $0^\circ\text{C}$ , kJ/kg, and  $t$  is the temperature in  $^\circ\text{C}$ . Also for air,  $pv = 0.287(t + 273)$ , where  $p$  is in kPa and  $v$  is in  $\text{m}^3/\text{kg}$ .

A mass of air is stirred by a paddle wheel in an insulated constant volume tank. The velocities due to stirring make a negligible contribution to the internal energy of the air. Air flows out through a small valve in the tank at a rate controlled to keep the temperature in the tank constant. At a certain instant the conditions are as follows: tank volume  $0.12 \text{ m}^3$ , pressure  $1 \text{ MPa}$ , temperature  $150^\circ\text{C}$ , and power to paddle wheel  $0.1 \text{ kW}$ . Find the rate of flow of air out of the tank at this instant.

**Solution** Writing the energy balance for the control volume as shown in Fig. 5.21

$$\frac{dE_V}{d\tau} = \frac{dW}{d\tau} - (h_p) \frac{dm}{d\tau}$$

Since there is no change in internal energy of air in the tank,

$$h_p \cdot \frac{dm}{d\tau} = \frac{dW}{d\tau}$$

where  $h_p = u + pv$ .

Let  $u = 0$  at  $t = 0$  K =  $-273^{\circ}\text{C}$

$$u = u_0 + 0.718 t$$

$$0 = u_0 + 0.718 (-273)$$

$$u_0 = 0.718 \times 273 \text{ kJ/kg}$$

At  $t^{\circ}\text{C}$

$$u = 0.718 \times 273 + 0.718 t$$

$$= 0.718 (t + 273) \text{ kJ/kg}$$

$$h_p = 0.718 (t + 273) + 0.287 (t + 273)$$

or  $h_p = 1.005 (t + 273)$

At  $150^{\circ}\text{C}$

$$\begin{aligned} h_p &= 1.005 \times 423 \\ &= 425 \text{ kJ/kg} \end{aligned}$$

$$\therefore \frac{dm}{d\tau} = \frac{1}{h_p} \frac{dW}{d\tau}$$

$$= \frac{0.1 \text{ kJ/s}}{425 \text{ kJ/kg}} = 0.236 \times 10^{-3} \text{ kg/s}$$

$$= 0.845 \text{ kg/h}$$

This is the rate at which air flows out of the tank.

**Example 5.10** A well-insulated vessel of volume  $V$  contains a gas at pressure  $p_0$  and temperature  $t_0$ . The gas from a main at a uniform temperature  $t_1$  is pumped into the vessel and the inflow rate decreases exponentially with time according to  $\dot{m} = \dot{m}_0 e^{-a\tau}$ , where  $a$  is a constant. Determine the pressure and temperature of the gas in the vessel as a function of time. Neglect the K.E. of the gas entering the vessel and assume that the gas follows the relation

$$pv = RT, \text{ where } T = t + 273$$

and its specific heats are constant.

- (i) If the vessel was initially evacuated, show that the temperature inside the vessel is independent of time.
- (ii) Determine the charging time when the pressure inside the vessel reaches that of the main.

**Solution** Since the vessel is well-insulated,  $\dot{Q} = 0$  and there is no external work transfer,  $\dot{W} = 0$ . Therefore,

$$\frac{dE_V}{d\tau} = h_1 \frac{dm}{d\tau} m = h_1 \dot{m}_0 e^{-a\tau}$$

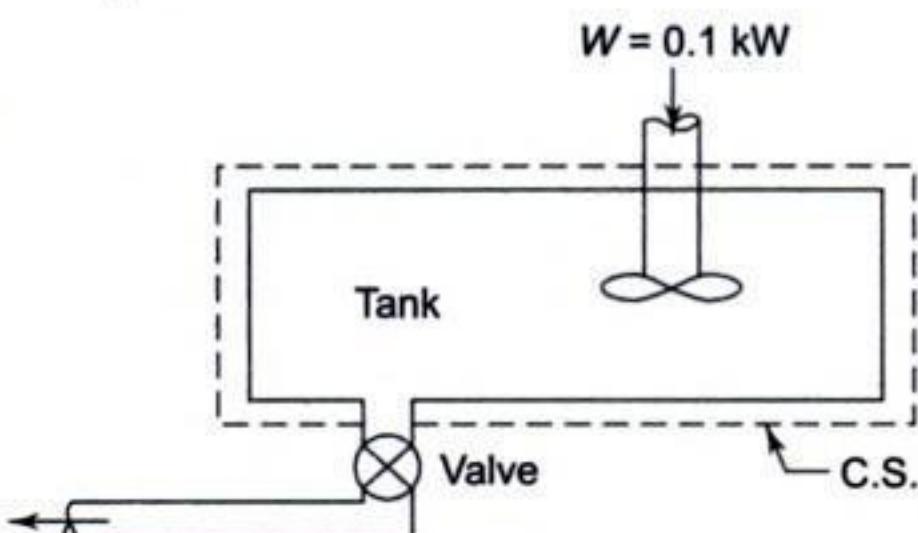


Fig. 5.21

where  $h_1$  is the enthalpy of the gas in the main

On integration,

$$E = E_0 + \frac{h_1 \dot{m}_0}{a} (1 - e^{-a\tau})$$

where  $E_0$  is the initial energy of the vessel at the beginning of the charging process, i.e.  $E = E_0$  at  $\tau = 0$ . Neglecting K.E. and P.E. changes, by energy balance

$$Mu = M_0 u_0 + \frac{\dot{m}_0}{a} (1 - e^{-a\tau}) (u_1 + p_1 v_1) \quad (1)$$

Again,

$$\frac{dm}{d\tau} = \dot{m}_0 e^{-a\tau}$$

On integration,

$$M = M_0 + \frac{\dot{m}_0 (1 - e^{-a\tau})}{a} \quad (2)$$

where  $M_0$  is the initial mass of the gas. Eliminating  $M$  from Eqs. (1) and (2),

$$\left\{ M_0 + \frac{\dot{m}_0 (1 - e^{-a\tau})}{a} \right\} u - M_0 u_0$$

$$= \frac{\dot{m}_0}{a} (1 - e^{-a\tau}) (u_1 + RT_1)$$

$$M_0 c_v (T - T_0) = \frac{\dot{m}_0}{a} (1 - e^{-a\tau}) \{ c_v (T_1 - T) + RT_1 \}$$

$$\therefore T = \frac{M_0 c_v T_0 + \frac{\dot{m}_0}{a} (1 - e^{-a\tau}) c_p T_1}{\left\{ M_0 + \frac{\dot{m}_0}{a} (1 - e^{-a\tau}) \right\} c_v}$$

$$p = \frac{MRT}{V} = \frac{R}{Vc_v} \left\{ M_0 c_v T_0 + \frac{\dot{m}_0}{a} (1 - e^{-a\tau}) c_p T_1 \right\}$$

$$= p_0 + \frac{\dot{m}_0 R}{aV} (1 - e^{-a\tau}) \gamma T_1$$

The above two equations show the temperature and pressure of the gas in the vessel as functions of time.

- (i) If  $M_0 = 0$ ,  $T = \gamma T_1$ , i.e. the temperature inside the vessel becomes independent of time and is equal to  $\gamma T_1$  throughout the charging process.

- (ii) The charging process will stop when pressure inside the vessel reaches that of the main. The charging time can be found by setting  $p = p_1$  in the pressure relation

$$p_1 - p_0 = \frac{\dot{m}_0 R \gamma T_1}{aV} - \frac{\dot{m}_0 R}{aV} e^{-a\tau} \gamma T_1$$

By rearrangement,

$$\begin{aligned} e^{a\tau} &= \frac{\dot{m}_0 R \gamma T_1 / (aV)}{\left( \frac{\dot{m}_0 R \gamma T_1}{aV} \right) - (p_1 - p_0)} \\ \therefore \tau &= \frac{1}{a} \ln \left[ 1 - (p_1 - p_0) \frac{aV}{\dot{m}_0 R \gamma T_1} \right] \end{aligned}$$

## Review Questions

- 5.1 Explain the system approach and the control volume approach in the analysis of a flow process.
- 5.2 What is a steady flow process?
- 5.3 Write the steady flow energy equation for a single stream entering and a single stream leaving a control volume and explain the various terms in it.
- 5.4 Give the differential form of the S.F.E.E.
- 5.5 Under what conditions does the S.F.E.E. reduce to Euler's equation?
- 5.6 How does Bernoulli's equation compare with S.F.E.E.?
- 5.7 What will be the velocity of a fluid leaving a nozzle, if the velocity of approach is very small?
- 5.8 Show that the enthalpy of a fluid before throttling is equal to that after throttling.
- 5.9 Write the general energy equation for a variable flow process.
- 5.10 What is the system technique in a bottle-filling process?
- 5.11 Explain the control volume technique in a variable flow process.

## Problems

- 5.1 A blower handles 1 kg/s of air at 20°C and consumes a power of 15 kW. The inlet and outlet velocities of air are 100 m/s and 150 m/s respectively. Find the exit air temperature, assuming adiabatic conditions. Take  $c_p$  of air is 1.005 kJ/kg-K. Ans. 28.38°C
- 5.2 A turbine operates under steady flow conditions, receiving steam at the following state: pressure 1.2 MPa, temperature 188°C, enthalpy 2785 kJ/kg, velocity 33.3 m/s and elevation 3 m. The steam leaves the turbine at the following state: pressure 20 kPa, enthalpy 2512 kJ/kg, velocity 100 m/s, and elevation 0 m. Heat is lost to the surroundings at the rate of 0.29 kJ/s. If the rate of steam flow through

the turbine is 0.42 kg/s, what is the power output of the turbine in kW?

Ans. 112.51 kW

- 5.3 A nozzle is a device for increasing the velocity of a steadily flowing stream. At the inlet to a certain nozzle, the enthalpy of the fluid passing is 3000 kJ/kg and the velocity is 60 m/s. At the discharge end, the enthalpy is 2762 kJ/kg. The nozzle is horizontal and there is negligible heat loss from it. (a) Find the velocity at exist from the nozzle. (b) If the inlet area is 0.1 m<sup>2</sup> and the specific volume at inlet is 0.187 m<sup>3</sup>/kg, find the mass flow rate. (c) If the specific volume at the nozzle exit is 0.498 m<sup>3</sup>/kg, find the exit area of the nozzle.

Ans. (a) 692.5 m/s, (b) 32.08 kg/s (c) 0.023 m<sup>2</sup>

- 5.4 In an oil cooler, oil flows steadily through a bundle of metal tubes submerged in a steady stream of cooling water. Under steady flow conditions, the oil enters at 90°C and leaves at 30°C, while the water enters at 25°C and leaves at 70°C. The enthalpy of oil at  $t^\circ\text{C}$  is given by

$$h = 1.68 t + 10.5 \times 10^{-4} t^2 \text{ kJ/kg}$$

What is the cooling water flow required for cooling 2.78 kg/s of oil?

Ans. 1.47 kg/s

- 5.5 A thermoelectric generator consists of a series of semiconductor elements (Fig. 5.22), heated on one side and cooled on the other. Electric current flow is produced as a result of energy transfer as heat. In a particular experiment the current was measured to be 0.5 amp and the electrostatic potential at (1) was 0.8 volt above that at (2). Energy transfer as heat to the hot side of the generator was taking place at a rate of 5.5 watts. Determine the rate of energy transfer as heat from the cold side and the energy conversion efficiency.

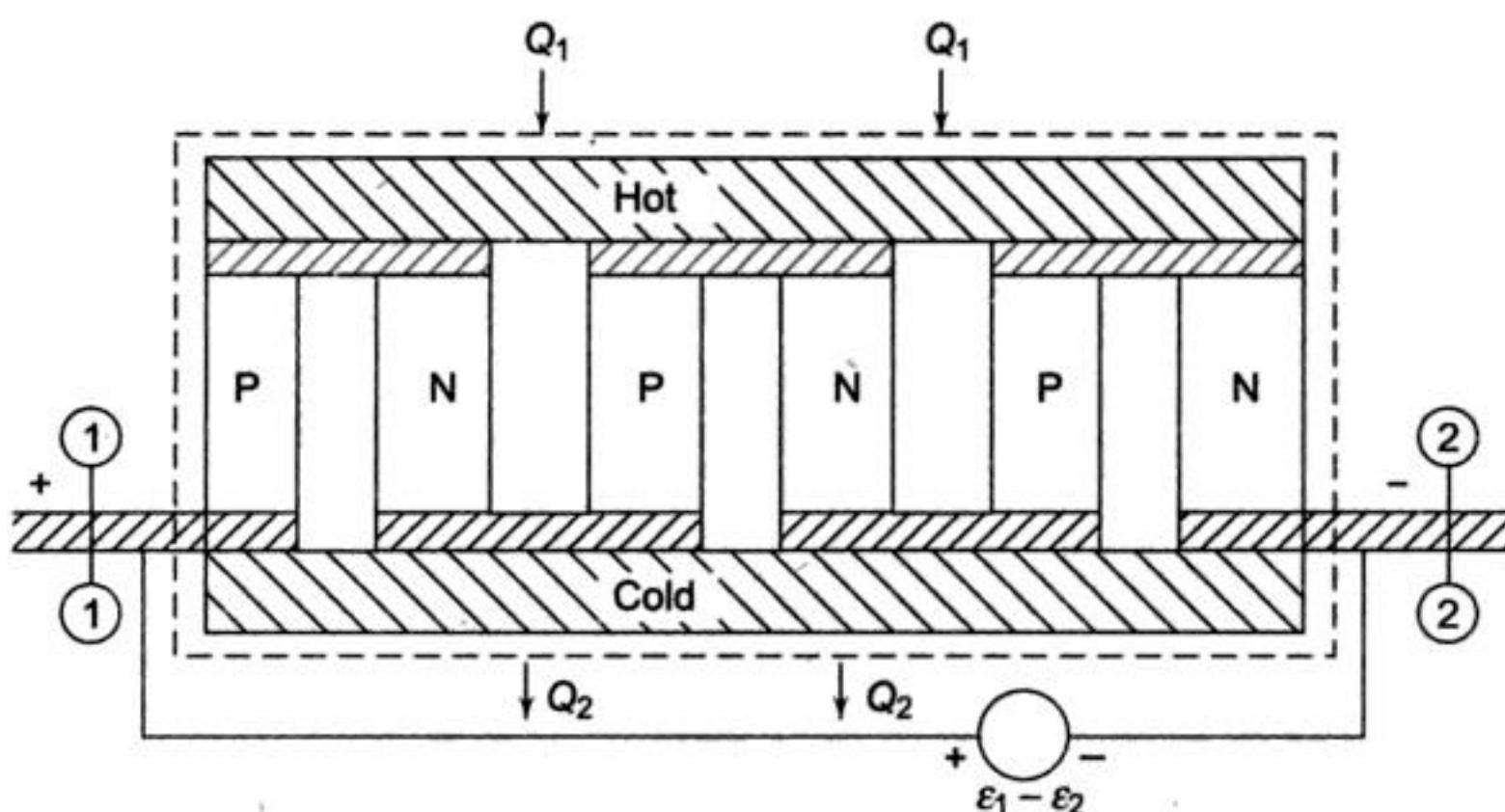


Fig. 5.22

Ans.  $Q_2 = 5.1$  watts,  $\eta = 0.073$

- 5.6 A turbocompressor delivers 2.33 m<sup>3</sup>/s at 0.276 MPa, 43°C which is heated at this pressure to 430°C and finally expanded in a turbine which delivers 1860 kW. During the expansion, there is a heat transfer of 0.09 MJ/s to the surroundings. Calculate the turbine exhaust temperature if changes in kinetic and potential energy are negligible.

Ans. 157°C

- 5.7 A reciprocating air compressor takes in  $2 \text{ m}^3/\text{min}$  at  $0.11 \text{ MPa}$ ,  $20^\circ\text{C}$  which it delivers at  $1.5 \text{ MPa}$ ,  $111^\circ\text{C}$  to an aftercooler where the air is cooled at constant pressure to  $25^\circ\text{C}$ . The power absorbed by the compressor is  $4.15 \text{ kW}$ . Determine the heat transfer in (a) the compressor, and (b) the cooler. State your assumptions. Ans.  $-0.17 \text{ kJ/s}$ ,  $-3.76 \text{ kJ/s}$
- 5.8 In a water cooling tower air enters at a height of  $1 \text{ m}$  above the ground level and leaves at a height of  $7 \text{ m}$ . The inlet and outlet velocities are  $20 \text{ m/s}$  and  $30 \text{ m/s}$  respectively. Water enters at a height of  $8 \text{ m}$  and leaves at a height of  $0.8 \text{ m}$ . The velocity of water at entry and exit are  $3 \text{ m/s}$  and  $1 \text{ m/s}$  respectively. Water temperatures are  $80^\circ\text{C}$  and  $50^\circ\text{C}$  at the entry and exit respectively. Air temperatures are  $30^\circ\text{C}$  and  $70^\circ\text{C}$  at the entry and exit respectively. The cooling tower is well insulated and a fan of  $2.25 \text{ kW}$  drives the air through the cooler. Find the amount of air per second required for  $1 \text{ kg/s}$  of water flow. The values of  $c_p$  of air and water are  $1.005 \text{ J/kg K}$  and  $4.187 \text{ kJ/kg K}$  respectively. Ans.  $3.16 \text{ kg/s}$
- 5.9 Air at  $101.325 \text{ kPa}$ ,  $20^\circ\text{C}$  is taken into a gas turbine power plant at a velocity of  $140 \text{ m/s}$  through an opening of  $0.15 \text{ m}^2$  cross-sectional area. The air is compressed heated, expanded through a turbine, and exhausted at  $0.18 \text{ MPa}$ ,  $150^\circ\text{C}$  through an opening of  $0.10 \text{ m}^2$  cross-sectional area. The power output is  $375 \text{ kW}$ . Calculate the net amount of heat added to the air in  $\text{kJ/kg}$ . Assume that air obeys the law  $pv = 0.287(t + 273)$ , where  $p$  is the pressure in  $\text{kPa}$ ,  $v$  is the specific volume in  $\text{m}^3/\text{kg}$ , and  $t$  is the temperature in  $^\circ\text{C}$ . Take  $c_p = 1.005 \text{ kJ/kg K}$ . Ans.  $150.23 \text{ kJ/kg}$
- 5.10 A gas flows steadily through a rotary compressor. The gas enters the compressor at a temperature of  $16^\circ\text{C}$ , a pressure of  $100 \text{ kPa}$ , and an enthalpy of  $391.2 \text{ kJ/kg}$ . The gas leaves the compressor at a temperature of  $245^\circ\text{C}$ , a pressure of  $0.6 \text{ MPa}$ , and an enthalpy of  $534.5 \text{ kJ/kg}$ . There is no heat transfer to or from the gas as it flows through the compressor. (a) Evaluate the external work done per unit mass of gas assuming the gas velocities at entry and exit to be negligible. (b) Evaluate the external work done per unit mass of gas when the gas velocity at entry is  $80 \text{ m/s}$  and that at exit is  $160 \text{ m/s}$ . Ans.  $143.3 \text{ kJ/kg}$ ,  $152.9 \text{ kJ/kg}$
- 5.11 The steam supply to an engine comprises two streams which mix before entering the engine. One stream is supplied at the rate of  $0.01 \text{ kg/s}$  with an enthalpy of  $2952 \text{ kJ/kg}$  and a velocity of  $20 \text{ m/s}$ . The other stream is supplied at the rate of  $0.1 \text{ kg/s}$  with an enthalpy of  $2569 \text{ kJ/kg}$  and a velocity of  $120 \text{ m/s}$ . At the exit from the engine the fluid leaves as two streams, one of water at the rate of  $0.001 \text{ kg/s}$  with an enthalpy of  $420 \text{ kJ/kg}$  and the other of steam; the fluid velocities at the exit are negligible. The engine develops a shaft power of  $25 \text{ kW}$ . The heat transfer is negligible. Evaluate the enthalpy of the second exit stream. Ans.  $2402 \text{ kJ/kg}$
- 5.12 The stream of air and gasoline vapour, in the ratio of  $14 : 1$  by mass, enters a gasoline engine at a temperature of  $30^\circ\text{C}$  and leaves as combustion products at a temperature of  $790^\circ\text{C}$ . The engine has a specific fuel consumption of  $0.3 \text{ kg/kWh}$ . The net heat transfer rate from the fuel-air stream to the jacket cooling water and to the surroundings is  $35 \text{ kW}$ . The shaft power delivered by the engine is  $26 \text{ kW}$ . Compute the increase in the specific enthalpy of the fuel-air stream, assuming the changes in kinetic energy and in elevation to be negligible. Ans.  $-1877 \text{ kJ/kg}$  mixture

- 5.13 An air turbine forms part of an aircraft refrigerating plant. Air at a pressure of 295 kPa and a temperature of 58°C flows steadily into the turbine with a velocity of 45 m/s. The air leaves the turbine at a pressure of 115 kPa, a temperature of 2°C, and a velocity of 150 m/s. The shaft work delivered by the turbine is 54 kJ/kg of air. Neglecting changes in elevation, determine the magnitude and sign of the heat transfer per unit mass of air flowing. For air, take  $c_p = 1.005 \text{ kJ/kg K}$  and the enthalpy  $h = c_p t$ . Ans. + 7.96 kJ/kg
- 5.14 In a turbomachine handling an incompressible fluid with a density of  $1000 \text{ kg/m}^3$  the conditions of the fluid at the rotor entry and exit are as given below
- |                    | Inlet    | Exit     |
|--------------------|----------|----------|
| Pressure           | 1.15 MPa | 0.05 MPa |
| Velocity           | 30 m/s   | 15.5 m/s |
| Height above datum | 10 m     | 2 m      |
- If the volume flow rate of the fluid is  $40 \text{ m}^3/\text{s}$ , estimate the net energy transfer from the fluid as work. Ans. 60.3 MW
- 5.15 A room for four persons has two fans, each consuming 0.18 kW power, and three 100 W lamps. Ventilation air at the rate of  $80 \text{ kg/h}$  enters with an enthalpy of 84 kJ/kg and leaves with an enthalpy of 59 kJ/kg. If each person puts out heat at the rate of  $630 \text{ kJ/h}$  determine the rate at which heat is to be removed by a room cooler, so that a steady state is maintained in the room. Ans. 1.92 kW
- 5.16 Air flows steadily at the rate of  $0.4 \text{ kg/s}$  through an air compressor, entering at 6 m/s with a pressure of 1 bar and a specific volume of  $0.85 \text{ m}^3/\text{kg}$ , and leaving at 4.5 m/s with a pressure of 6.9 bar and a specific volume of  $0.16 \text{ m}^3/\text{kg}$ . The internal energy of the air leaving is 88 kJ/kg greater than that of the air entering. Cooling water in a jacket surrounding the cylinder absorbs heat from the air at the rate of 59 W. Calculate the power required to drive the compressor and the inlet and outlet cross-sectional areas.
- Ans.  $45.4 \text{ kW}, 0.057 \text{ m}^2, 0.0142 \text{ m}^2$
- 5.17 Two streams of air, one at 1 bar, 27°C and velocity of 30 m/s and the other at 5 bar, 227°C and 50 m/s velocity, mix in equal proportion in a chamber from which heat at the rate of  $100 \text{ kJ/kg}$  is removed. The mixture is then passed through an adiabatic nozzle. Find the velocity of the stream issuing out of the nozzle. The temperature of air leaving the nozzle is 27°C, and its  $c_p = 1.005 \text{ kJ/kg K}$ .
- Ans.  $51.96 \text{ m/s}$
- 5.18 Steam flowing in a pipeline is at a steady state represented by  $p_p, t_p, u_p, v_p, h_p$  and  $\mathbf{V}_p$ . A small amount of the total flow is led through a small tube to an evacuated chamber which is allowed to fill slowly until the pressure is equal to the pipeline pressure. If there is no heat transfer, derive an expression for the final specific internal energy in the chamber, in terms of the properties in the pipeline.
- 5.19 The internal energy of air is given, at ordinary temperatures, by

$$u = u_0 + 0.718 t$$

where  $u$  is in kJ/kg,  $u_0$  is any arbitrary value of  $u$  at 0°C, kJ/kg, and  $t$  is temperature in °C.

Also for air,  $pv = 0.287(t + 273)$   
where  $p$  is in kPa and  $v$  is in  $\text{m}^3/\text{kg}$ .

- (a) An evacuated bottle is fitted with a valve through which air from the atmosphere, at 760 mm Hg and 25°C, is allowed to flow slowly to fill the bottle. If no heat is transferred to or from the air in the bottle, what will its temperature be when the pressure in the bottle reaches 760 mm Hg?

Ans. 144.2°C

- (b) If the bottle initially contains 0.03 m<sup>3</sup> of air at 400 mm Hg and 25°C, what will the temperature be when the pressure in the bottle reaches 760 mm Hg?

Ans. 71.6°C

- 5.20 A pressure cylinder of volume  $V$  contains air at pressure  $p_0$  and temperature  $T_0$ . It is to be filled from a compressed air line maintained at constant pressure  $p_1$  and temperature  $T_1$ . Show that the temperature of the air in the cylinder after it has been charged to the pressure of the line is given by

$$T = \frac{\gamma T_1}{1 + \frac{p_0}{p_1} \left( \gamma \frac{T_1}{T_0} - 1 \right)}$$

- 5.21 A small reciprocating vacuum pump having the rate of volume displacement  $V_d$  is used to evacuate a large vessel of volume  $V$ . The air in the vessel is maintained at a constant temperature  $T$  by energy transfer as heat. If the initial and final pressures are  $p_1$  and  $p_2$  respectively, find the time taken for the pressure drop and the necessary energy transfer as heat during evacuation. Assume that for air,  $pV = mRT$ , where  $m$  is the mass and  $R$  is a constant, and  $u$  is a function of  $T$  only.

$$\left[ \text{Ans. } t = \frac{V}{V_d} \ln \frac{p_1}{p_2}; \quad Q = (p_1 - p_2)V \right]$$

[Hint:  $dm = -p(V_d \cdot dt)/(RT) = V dp/(RT)$ ].

- 5.22 A tank containing 45 kg of water initially at 45°C has one inlet and one exit with equal mass flow rates. Liquid water enters at 45°C and a mass flow rate of 270 kg/h. A cooling coil immersed in the water removes energy at the rate of 7.6 kW. The water is well mixed by a paddle wheel with a power input of 0.6 kW. The pressures at inlet and exit are equal. Ignoring changes in KE and PE, find the variation of water temperature with time.

Ans.  $T = 318 - 22 [1 - \exp(-6t)]$

- 5.23 A rigid tank of volume 0.5 m<sup>3</sup> is initially evacuated. A tiny hole develops in the wall, and air from the surroundings at 1 bar, 21°C leaks in. Eventually, the pressure in the tank reaches 1 bar. The process occurs slowly enough that heat transfer between the tank and the surroundings keeps the temperature of the air inside the tank constant at 21°C. Determine the amount of heat transfer.

Ans. - 50 kJ

# 6

## CHAPTER

# Second Law of Thermodynamics

## 6.1 QUALITATIVE DIFFERENCE BETWEEN HEAT AND WORK

The first law of thermodynamics states that a certain energy balance will hold when a system undergoes a change of state or a thermodynamic process. But it does not give any information on whether that change of state or the process is at all feasible or not. The first law cannot indicate whether a metallic bar of uniform temperature can spontaneously become warmer at one end and cooler at the other. All that the law can state is that if this process did occur, the energy gained by one end would be exactly equal to that lost by the other. *It is the second law of thermodynamics which provides the criterion as to the probability of various processes.*

Spontaneous processes in nature occur only in one direction. Heat always flows from a body at a higher temperature to a body at a lower temperature, water always flows downward, time always flows in the forward direction. The reverse of these never happens spontaneously. The spontaneity of the process is due to a finite driving potential, sometimes called the ‘force’ or the ‘cause’, and what happens is called the ‘flux’, the ‘current’ or the ‘effect’. The typical forces like temperature gradient, concentration gradient and electric potential gradient, have their respective conjugate fluxes of heat transfer, mass transfer, and flow of electric current. These transfer processes can never spontaneously occur from a lower to a higher potential. This directional law puts a limitation on energy transformation other than that imposed by the first law.

Joule’s experiments (Article 4.1) amply demonstrate that energy, when supplied to a system in the form of work, can be completely converted into heat (work transfer → internal energy increase → heat transfer). But the complete conversion of heat into work in a cycle is not possible. *So heat and work are not completely interchangeable forms of energy.*

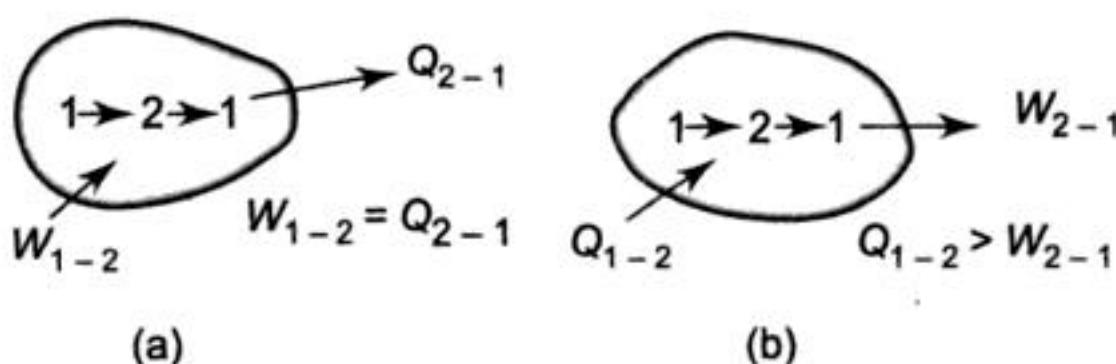
When work is converted into heat, we always have

$$W \rightarrow Q$$

but when heat is converted into work in a complete closed cycle process

$$Q \xrightarrow{\rightarrow} W$$

The arrow indicates the direction of energy transformation. This is illustrated in Fig. 6.1. As shown in Fig. 6.1(a), a system is taken from state 1 to state 2 by work transfer  $W_{1-2}$ , and then by heat transfer  $Q_{2-1}$  the system is brought back from state 2 to state 1 to complete a cycle. It is always found that  $W_{1-2} = Q_{2-1}$ . But if the system is taken from state 1 to state 2 by heat transfer  $Q_{1-2}$ , as shown in Fig. 6.1(b), then the system cannot be brought back from state 2 to state 1 by work transfer  $W_{2-1}$ . Hence, heat cannot be converted completely and continuously into work in a cycle. Some heat has to be rejected. In Fig. 6.1(b),  $W_{2-1}$  is the work done and  $Q_{3-1}$  is the heat rejected to complete the cycle. This underlies the work of Sadi Carnot, a French military engineer, who first studied this aspect of energy transformation (1824). Work is said to be a *high grade energy* and heat a *low grade energy*. *The complete conversion of low grade energy into high grade energy in a cycle is impossible.*



**Fig. 6.1 Qualitative Distinction between Heat and Work**

## 6.2 CYCLIC HEAT ENGINE

For engineering purposes, the second law is best expressed in terms of the conditions which govern the production of work by a thermodynamic system operating in a cycle.

A heat engine cycle is a thermodynamic cycle in which there is a net heat transfer to the system and a net work transfer *from* the system. The system which executes a heat engine cycle is called a *heat engine*.

A heat engine may be in the form of mass of gas confined in a cylinder and piston machine (Fig. 6.2a) or a mass of water moving in a steady flow through a steam power plant (Fig. 6.2b).

In the cyclic heat engine, as represented in Fig. 6.2(a), heat  $Q_1$  is transferred to the system, work  $W_E$  is done by the system, work  $W_c$  is done upon the system, and then heat  $Q_2$  is rejected from the system. The system is brought back to the initial state through all these four successive processes which constitute a heat engine cycle. In Fig. 6.2(b) heat  $Q_1$  is transferred from the furnace to the water in the

boiler to form steam which then works on the turbine rotor to produce work  $W_T$ , then the steam is condensed to water in the condenser in which an amount of heat  $Q_2$  is rejected from the system, and finally work  $W_p$  is done on the system (water) to pump it to the boiler. The system repeats the cycle.

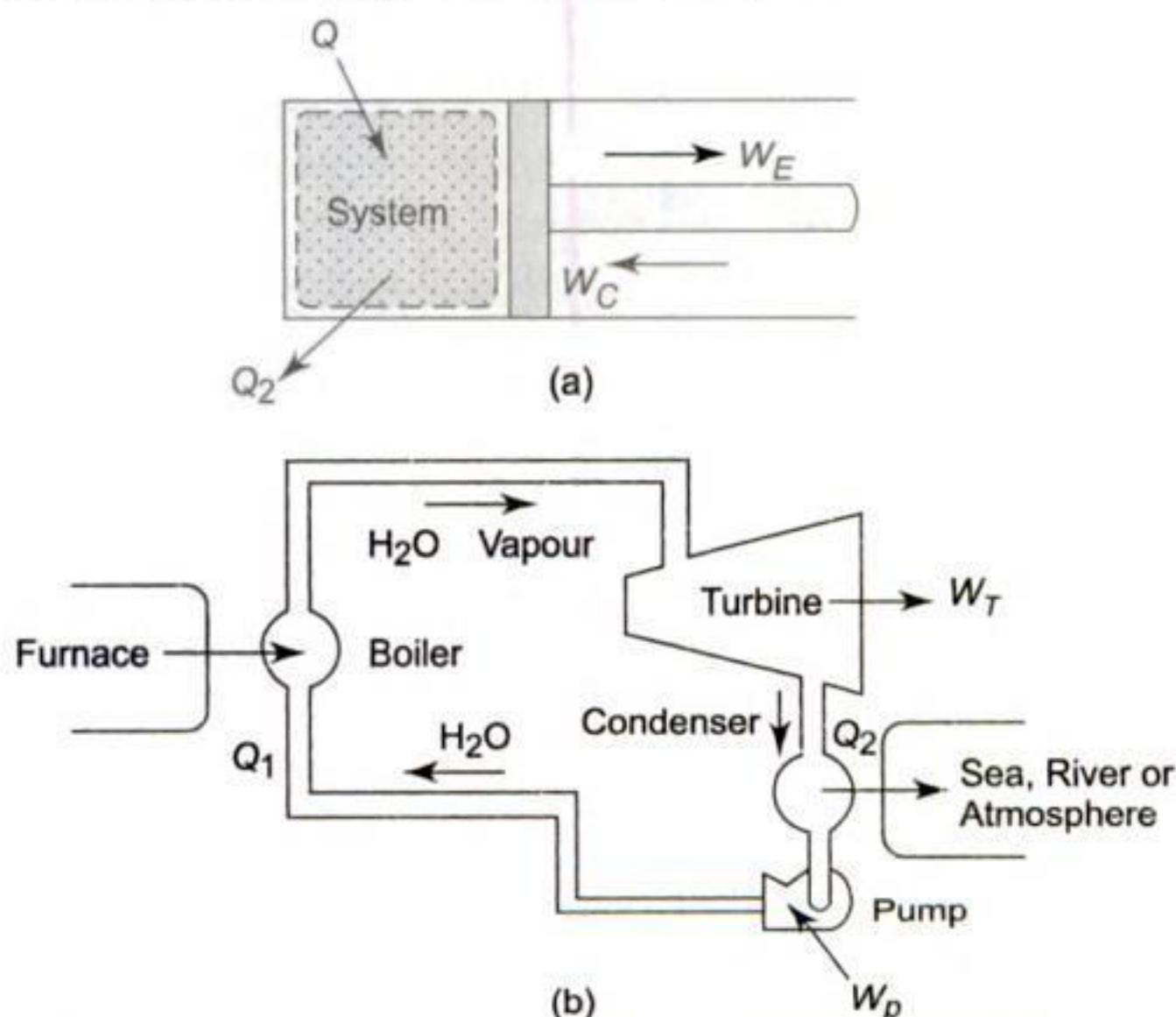


Fig. 6.2 *Cycle Heat Engine*

(a) *Heat Engine Cycle Performed by a Closed System Undergoing Four Successive Energy Interactions with the Surroundings*

(b) *Heat Engine Cycle Performed by a Steady Flow System Interacting with the Surroundings as Shown*

The net heat transfer in a cycle to either of the heat engines

$$Q_{\text{net}} = Q_1 - Q_2 \quad (6.1)$$

and the net work transfer in a cycle

$$W_{\text{net}} = W_T - W_P \quad (6.2)$$

$$\text{(or } W_{\text{net}} = W_E - W_C \text{)}$$

By the first law of thermodynamics, we have

$$\sum_{\text{cycle}} Q = \sum_{\text{cycle}} W$$

$$\therefore Q_{\text{net}} = W_{\text{net}}$$

$$\text{or } Q_1 - Q_2 = W_T - W_P \quad (6.3)$$

Figure 6.3 represents a cyclic heat engine in the form of a block diagram indicating the various energy interactions during a cycle. Boiler ( $B$ ), turbine ( $T$ ), condenser ( $C$ ), and pump ( $P$ ), all four together constitute a heat engine. A heat engine is here

a certain quantity of water undergoing the energy interactions, as shown, in cyclic operations to produce net work from a certain heat input.

The function of a heat engine cycle is to produce work continuously at the expense of heat input to the system. So the net work  $W_{\text{net}}$  and heat input  $Q_1$  referred to the cycle are of primary interest. The efficiency of a heat engine or a heat engine cycle is defined as follows:

$$\eta = \frac{\text{Net work output of the cycle}}{\text{Total heat input to the cycle}} \\ = \frac{W_{\text{net}}}{Q_1} \quad (6.4)$$

From Eqs (6.1), (6.2), (6.3) and (6.4),

$$\eta = \frac{W_{\text{net}}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{Q_1 - Q_2}{Q_1}$$

$$\eta = 1 - \frac{Q_2}{Q_1} \quad (6.5)$$

This is also known as the *thermal efficiency* of a heat engine cycle. A heat engine is very often called upon to extract as much work (net) as possible from a certain heat input, i.e. to maximize the cycle efficiency.

### 6.3 ENERGY RESERVOIRS

A thermal energy reservoir (TER) is defined as a large body of infinite heat capacity, which is capable of absorbing or rejecting an unlimited quantity of heat without suffering appreciable changes in its thermodynamic coordinates. The changes that do take place in the large body as heat enters or leaves are so very slow and so very minute that all processes within it are quasi-static.

The thermal energy reservoir  $\text{TER}_H$  from which heat  $Q_1$  is transferred to the system operating in a heat engine cycle is called the *source*. The thermal energy reservoir  $\text{TER}_L$  to which heat  $Q_2$  is rejected from the system during a cycle is the *sink*. A typical source is a constant temperature furnace where fuel is continuously burnt, and a typical sink is a river or sea or the atmosphere itself.

A mechanical energy reservoir (MER) is a large body enclosed by an adiabatic impermeable wall capable of storing work as potential energy (such as a raised weight or wound spring) or kinetic energy (such as a rotating flywheel). All processes of interest within an MER are essentially quasi-static. An MER receives and delivers mechanical energy quasi-statically.

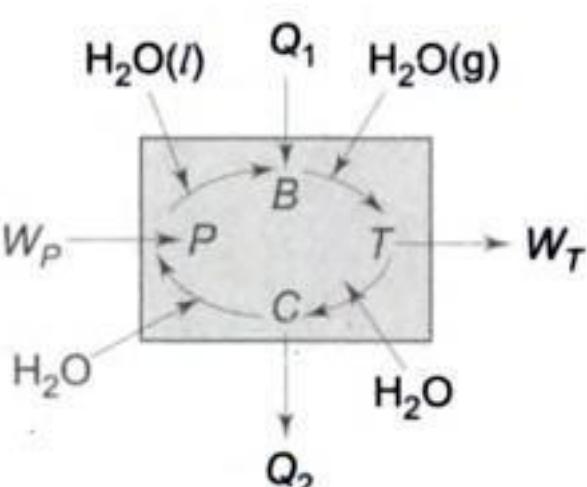
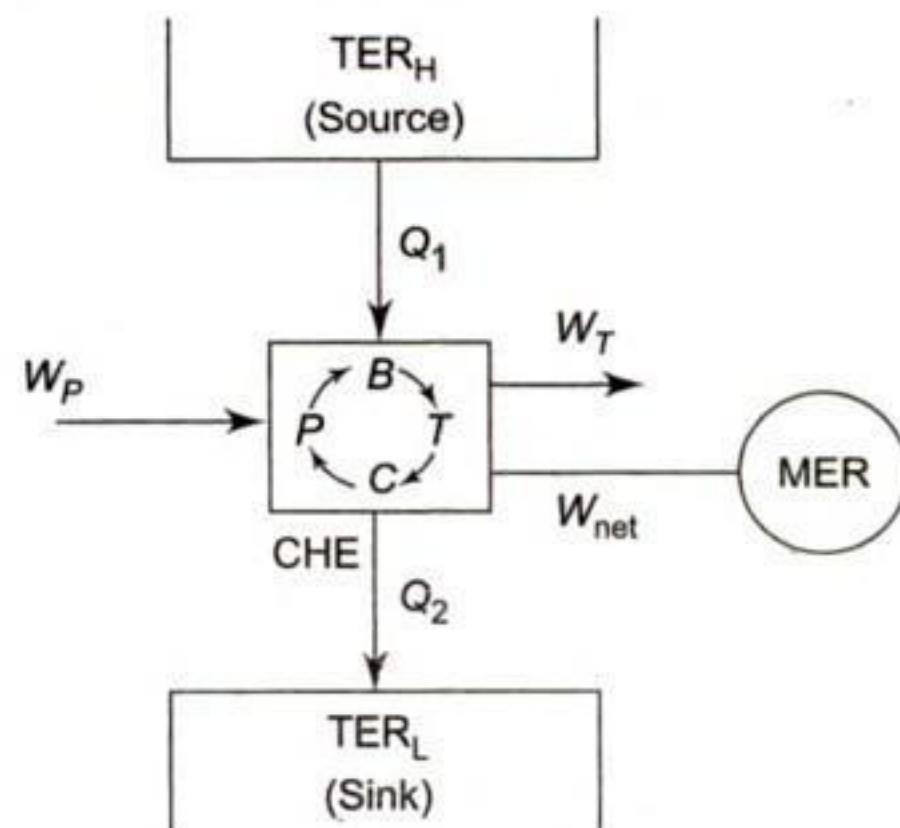


Fig. 6.3

Cyclic Heat Engine with Energy Interactions Represented in a Block Diagram

Figure 6.4 shows a cyclic heat engine exchanging heat with a source and a sink and delivering  $W_{\text{net}}$  in a cycle to an MER.



**Fig. 6.4** Cyclic Heat Engine (CHE) with Source and Sink

#### 6.4 KELVIN-PLANCK STATEMENT OF SECOND LAW

The efficiency of a heat engine is given by

$$\eta = \frac{W_{\text{net}}}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

Experience shows that  $W_{\text{net}} < Q_1$ , since heat  $Q_1$  transferred to a system cannot be completely converted to work in a cycle (Article 6.1). Therefore,  $\eta$  is less than unity. A heat engine can never be 100% efficient. Therefore,  $Q_2 > 0$ , i.e. there has always to be a heat rejection. To produce net work in a thermodynamic cycle, a heat engine has thus to exchange heat with two reservoirs, the source and the sink.

The *Kelvin-Planck statement* of the second law states: *It is impossible for a heat engine to produce net work in a complete cycle if it exchanges heat only with bodies at a single fixed temperature.*

If  $Q_2 = 0$  (i.e.  $W_{\text{net}} = Q_1$ , or  $\eta = 1.00$ ), the heat engine will produce net work in a complete cycle by exchanging heat with only one reservoir, thus violating the Kelvin-Planck statement (Fig. 6.5). Such a heat engine is called a *perpetual motion machine of the second kind*, abbreviated to PMM2. A PMM2 is impossible.

A heat engine has, therefore, to exchange heat with two thermal energy reservoirs at two different temperatures to produce net work in a complete cycle (Fig. 6.6). So long as there is a difference in temperature, motive power (i.e. work) can be produced. If the bodies with which the heat engine exchanges heat are of finite heat capacities, work will be produced by the heat engine till the temperatures of the two bodies are equalized.

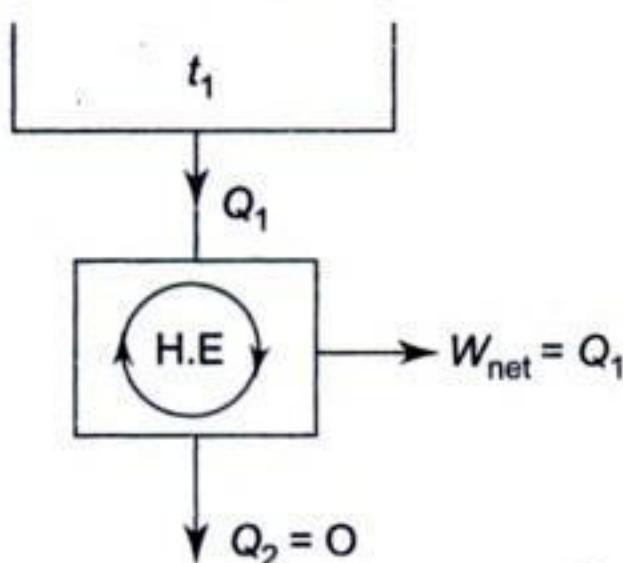


Fig. 6.5 A PMM2

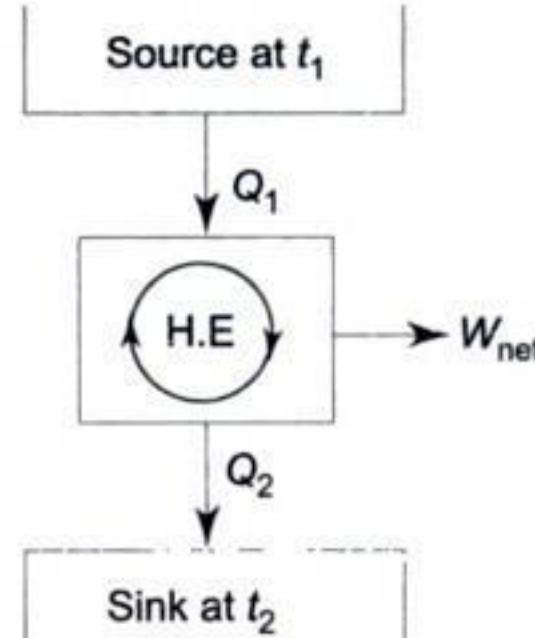


Fig. 6.6 Heat Engine Producing Net Work in a Cycle by Exchanging Heat at Two Different Temperatures

## 6.5 CLAUSIUS' STATEMENT OF THE SECOND LAW

Heat always flows from a body at a higher temperature to a body at a lower temperature. The reverse process never occurs spontaneously.

Clausius' statement of the second law gives: *It is impossible to construct a device which, operating in a cycle, will produce no effect other than the transfer of heat from a cooler to a hotter body.*

Heat cannot flow of itself from a body at a lower temperature to a body at a higher temperature. Some work must be expended to achieve this.

## 6.6 REFRIGERATOR AND HEAT PUMP

A *refrigerator* is a device which, operating in a cycle, maintains a body at a temperature lower than the temperature of the surroundings. Let the body A (Fig. 6.7) be maintained at  $t_2$ , which is lower than the ambient temperature  $t_1$ . Even though A is insulated, there will always be heat leakage  $Q_2$  into the body from the surroundings by virtue of the temperature difference. In order to maintain body A at the constant temperature  $t_2$ , heat has to be removed from the body at the same rate at which heat is leaking into the body. This heat ( $Q_2$ ) is absorbed by a working fluid, called the refrigerant, which evaporates in the evaporator  $E_1$  at a temperature lower than  $t_2$  absorbing the latent heat of vaporization from the body A which is cooled or refrigerated (Process 4–1). The vapour is first compressed in the compressor  $C_1$  driven by a motor which absorbs work  $W_C$  (Process 1–2), and is then condensed in the condenser  $C_2$  rejecting the latent heat of condensation  $Q_1$  at a temperature higher than that of the atmosphere (at  $t_1$ ) for heat transfer to take place (Process 2–3). The condensate then expands adiabatically through an expander (an engine or turbine) producing work  $W_E$ , when the temperature drops to a value lower than  $t_2$  such that heat  $Q_2$  flows from the body A to make the refrigerant evaporate (Process 3–4). Such a cyclic device of flow through  $E_1-C_1-C_2-E_2$  is called a *refrigerator*. In a refrigerator cycle, attention is

concentrated on the body  $A$ .  $Q_2$  and  $W$  are of primary interest. Just like efficiency in a heat engine cycle, there is a performance parameter in a refrigerator cycle, called the *coefficient of performance*, abbreviated to COP, which is defined as

$$\text{COP} = \frac{\text{Desired effect}}{\text{Work input}} = \frac{Q_2}{W}$$

$$\therefore [\text{COP}]_{\text{ref}} = \frac{Q_2}{Q_1 - Q_2} \quad (6.6)$$

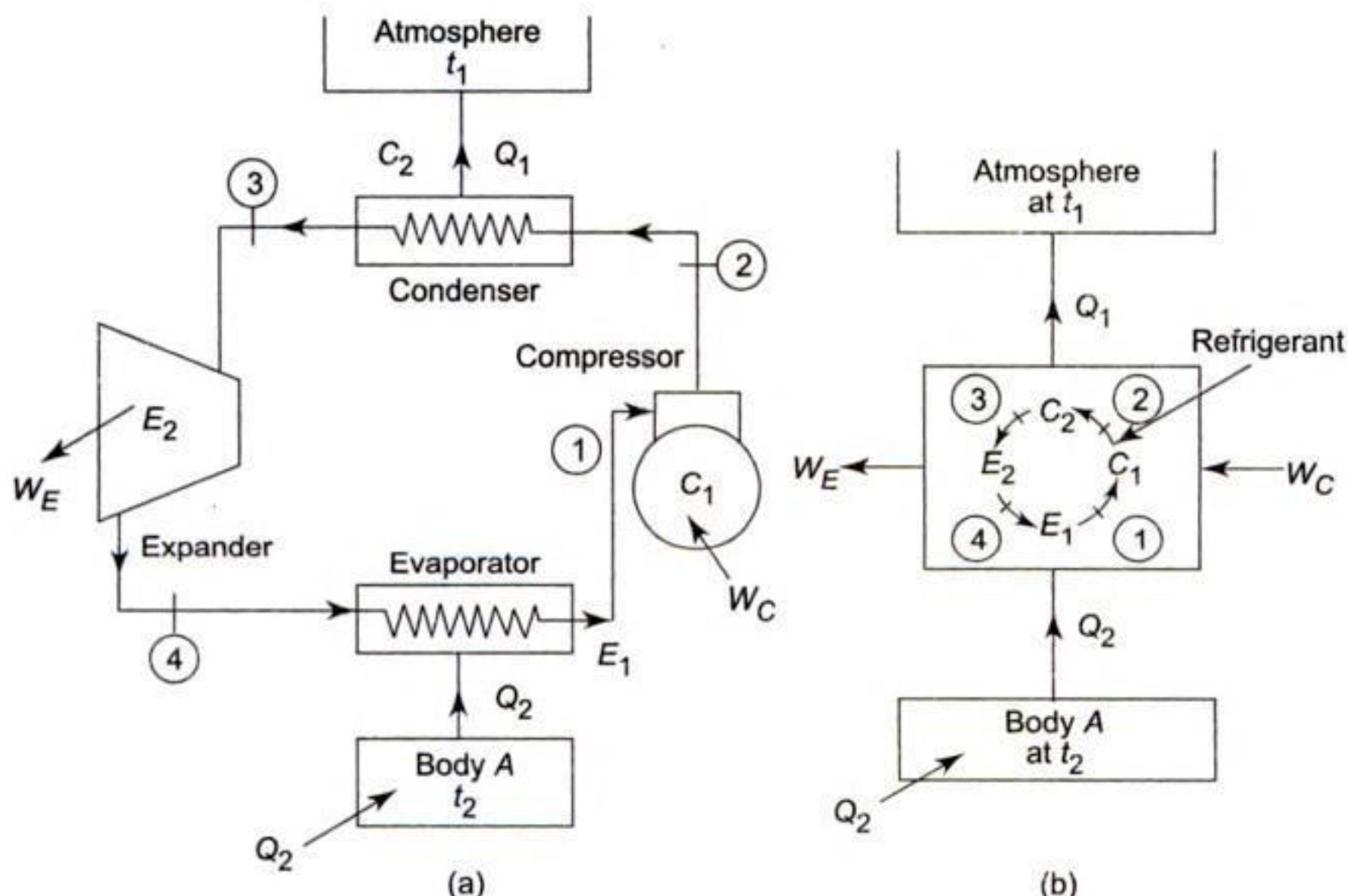


Fig. 6.7

A Cyclic Refrigeration Plant

A *heat pump* is a device which, operating in a cycle, maintains a body, say  $B$  (Fig. 6.8), at a temperature higher than the temperature of the surroundings. By virtue of the temperature difference, there will be heat leakage  $Q_1$  from the body to the surroundings. The body will be maintained at the constant temperature  $t_1$ , if heat is discharged into the body at the same rate at which heat leaks out of the body. The heat is extracted from the low temperature reservoir, which is nothing but the atmosphere, and discharged into the high temperature body  $B$ , with the expenditure of work  $W$  in a cyclic device called a heat pump. The working fluid operates in a cycle flowing through the evaporator  $E_1$ , compressor  $C_1$ , condenser  $C_2$  and expander  $E_2$ , similar to a refrigerator, but the attention is here focussed on the high temperature body  $B$ . Here  $Q_1$  and  $W$  are of primary interest, and the COP is defined as

$$\text{COP} = \frac{Q_1}{W}$$

$$\therefore [\text{COP}]_{H.P.} = \frac{Q_1}{Q_1 - Q_2} \quad (6.7)$$

From Eqs. (6.6) and (6.7), it is found that

$$[\text{COP}]_{H.P.} = [\text{COP}]_{\text{ref}} + 1 \quad (6.8)$$

The COP of a heat pump is greater than the COP of a refrigerator by unity. Equation (6.8) expresses a very interesting feature of a heat pump. Since

$$\begin{aligned} Q_1 &= [\text{COP}]_{H.P.} W \\ &= [\text{COP}_{\text{ref}} + 1] W \end{aligned} \quad (6.9)$$

$Q_1$  is always greater than  $W$ .

For an electrical resistance heater, if  $W$  is the electrical energy consumption, then the heat transferred to the space at steady state is  $W$  only, i.e.  $Q_1 = W$ .

A 1 kW electric heater can give 1 kW of heat at steady state and nothing more. In other words, 1 kW of work (high grade energy) dissipates to give 1 kW of heat (low grade energy), which is thermodynamically inefficient.

However, if this electrical energy  $W$  is used to drive the compressor of a heat pump, the heat supplied  $Q_1$  will always be more than  $W$ , or  $Q_1 > W$ . Thus, a heat pump provides a thermodynamic advantage over direct heating.

For heat to flow from a cooler to a hotter body,  $W$  cannot be zero, and hence, the COP (both for refrigerator and heat pump) cannot be infinity. Therefore,  $W > 0$ , and  $\text{COP} < \infty$ .

## 6.7 EQUIVALENCE OF KELVIN-PLANCK AND CLAUSIUS STATEMENTS

At first sight, Kelvin-Planck's and Clausius' statements may appear to be unconnected, but it can easily be shown that they are virtually two parallel statements of the second law and are equivalent in all respects.

The equivalence of the two statements will be proved if it can be shown that the violation of one statement implies the violation of the second, and vice versa.

(a) Let us first consider a cyclic heat pump  $P$  which transfers heat from a low temperature reservoir ( $t_2$ ) to a high temperature reservoir ( $t_1$ ) with no other effect, i.e. with no expenditure of work, violating Clausius statement (Fig. 6.9).

Let us assume a cyclic heat engine  $E$  operating between the same thermal energy reservoirs, producing  $W_{\text{net}}$  in one cycle. The rate of working of the heat engine is such that it draws an amount of heat  $Q_1$  from the hot reservoir equal to that discharged by the heat pump. Then the hot reservoir may be eliminated and the heat  $Q_1$  discharged by the heat pump is fed to the heat engine. So we see that the heat pump  $P$  and the heat engine  $E$  acting together constitute a heat engine operating in cycles and producing net work while exchanging heat only with one body at a single fixed temperature. This violates the Kelvin-Planck statement.

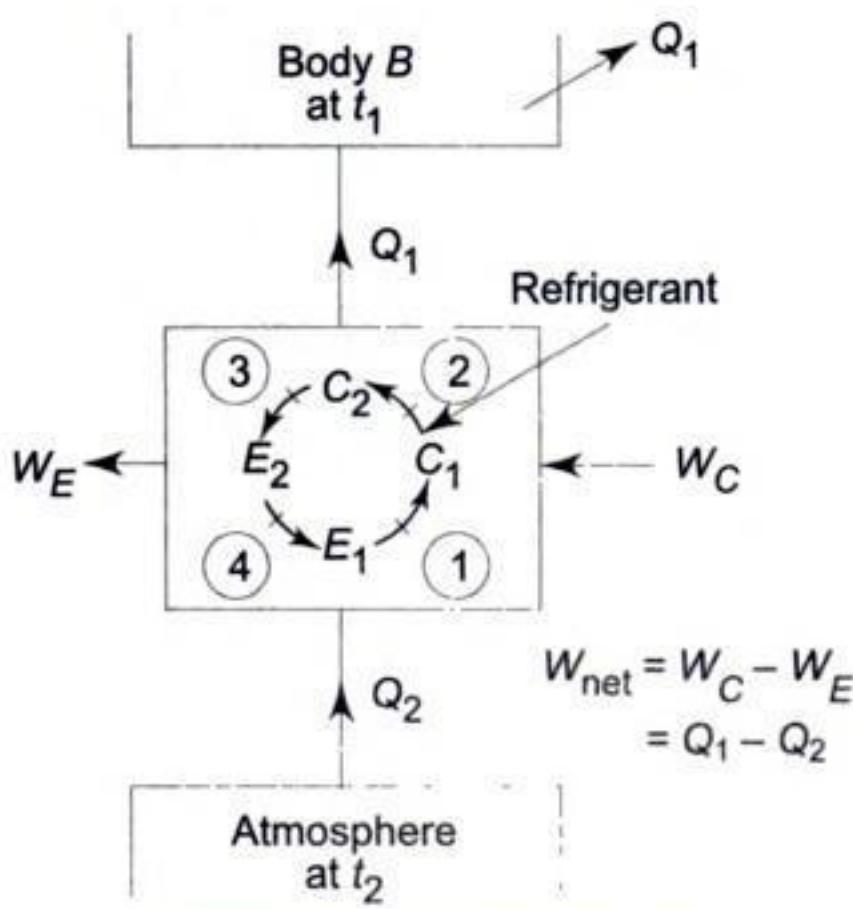


Fig. 6.8 A Cyclic Heat Pump

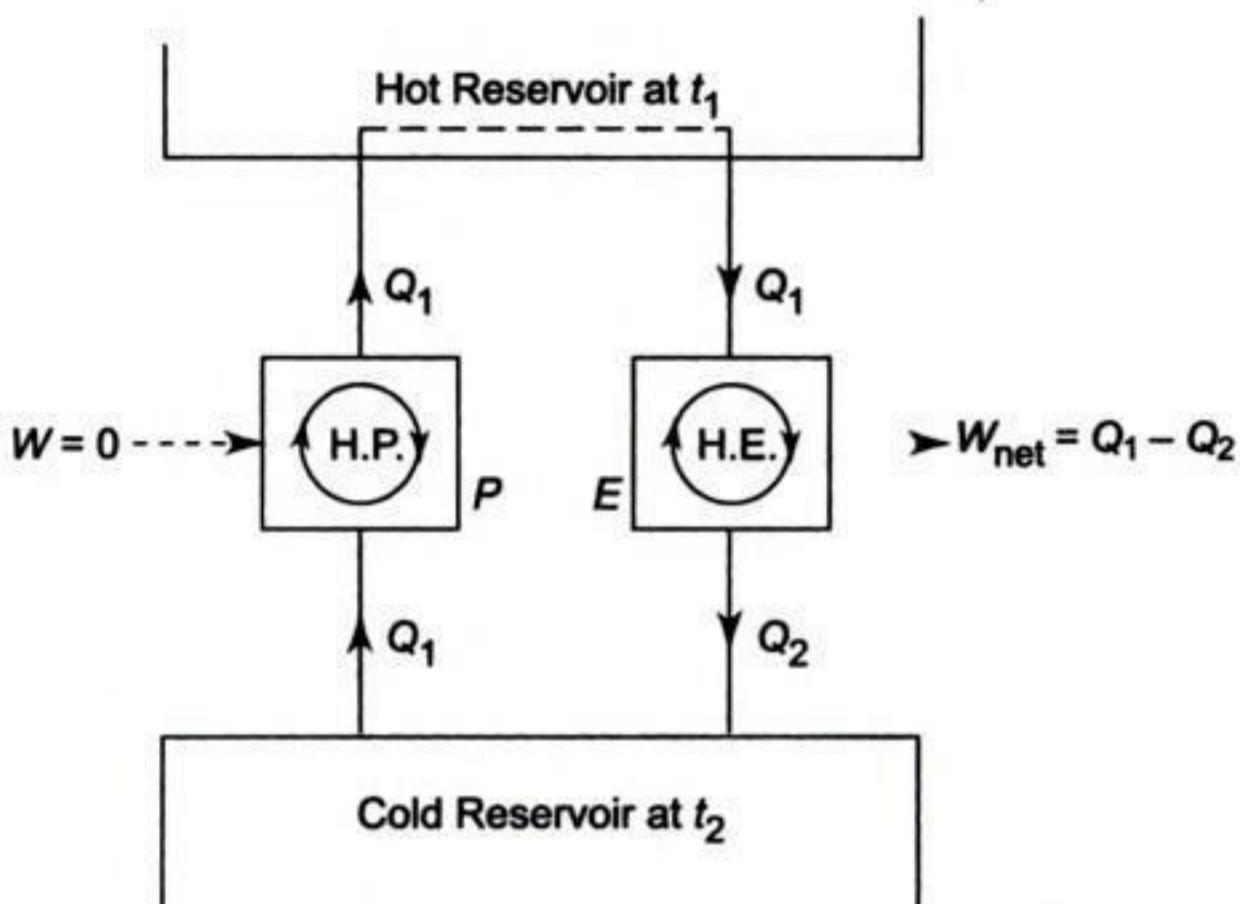


Fig. 6.9 Violation of the Clausius Statement

(b) Let us now consider a perpetual motion machine of the second kind ( $E$ ) which produces net work in a cycle by exchanging heat with only one thermal energy reservoir (at  $t_1$ ) and thus violates the Kelvin-Planck statement (Fig. 6.10).

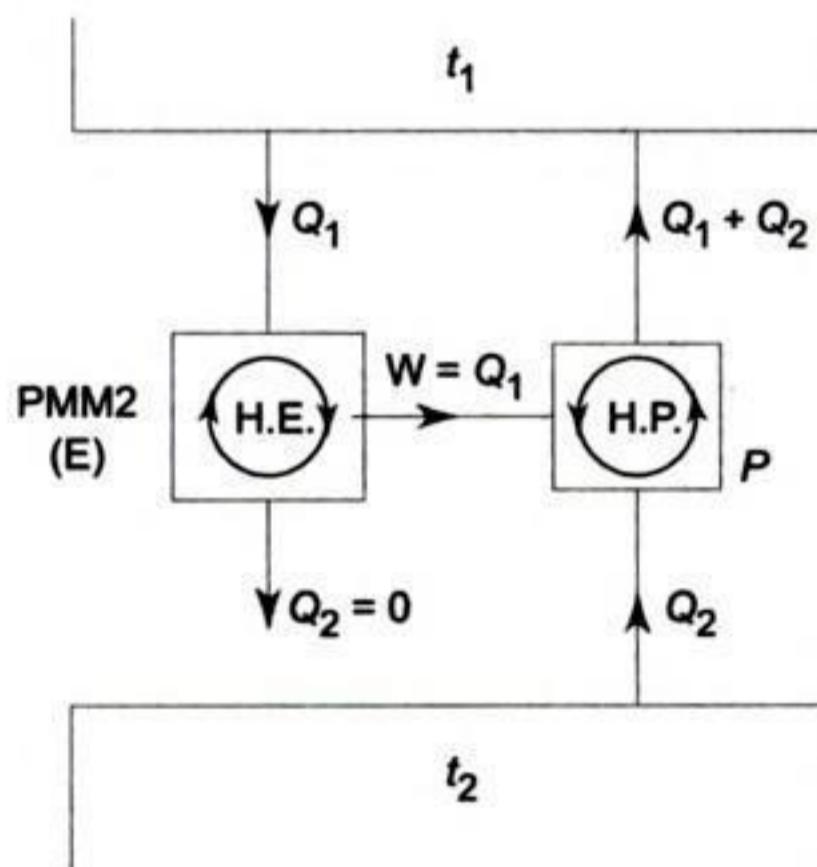


Fig. 6.10 Violation of the Kelvin-Planck Statement

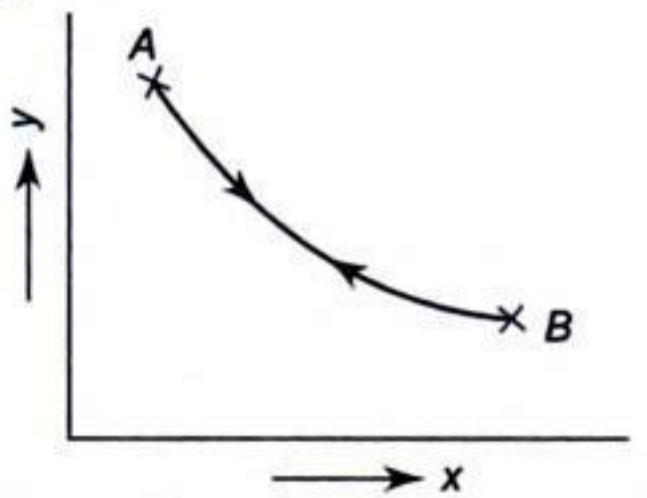
Let us assume a cyclic heat pump ( $P$ ) extracting heat  $Q_2$  from a low temperature reservoir at  $t_2$  and discharging heat to the high temperature reservoir at  $t_1$  with the expenditure of work  $W$  equal to what the PMM2 delivers in a complete cycle. So  $E$  and  $P$  together constitute a heat pump working in cycles and producing the sole effect of transferring heat from a lower to a higher temperature body, thus violating the Clausius statement.

## 6.8 REVERSIBILITY AND IRREVERSIBILITY

The second law of thermodynamics enables us to divide all processes into two classes:

- (a) Reversible or ideal process.
- (b) Irreversible or natural process.

A *reversible* process is one which is performed in such a way that at the conclusion of the process, both the system and the surroundings may be restored to their initial states, without producing any changes in the rest of the universe. Let the state of a system be represented by *A* (Fig. 6.11), and let the system be taken to state *B* by following the path *A*–*B*. If the system and also the surroundings are restored to their initial states and no change in the universe is produced, then the process *A*–*B* will be a reversible process. In the reverse process, the system has to be taken from state *B* to *A* by following the same path *B*–*A*. A reversible process should not leave any trace or relic behind to show that the process had ever occurred.



**Fig. 6.11** A Reversible Process

A *reversible process* is carried out infinitely slowly with an infinitesimal gradient, so that every state passed through by the system is an equilibrium state. So a reversible process coincides with a quasi-static process.

Any natural process carried out with a finite gradient is an irreversible process. A reversible process, which consists of a succession of equilibrium states, is an idealized hypothetical process, approached only as a limit. It is said to be an *asymptote to reality*. All spontaneous processes are irreversible.

Time has an important effect on reversibility. If the time allowed for a process to occur is infinitely large, even though the gradient is finite, the process becomes reversible. However, if this time is squeezed to a finite value, the finite gradient makes the process irreversible.

## 6.9 CAUSES OF IRREVERSIBILITY

The irreversibility of a process may be due to either one or both of the following:

- (a) Lack of equilibrium during the process.
- (b) Involvement of dissipative effects.

### 6.9.1 Irreversibility due to Lack of Equilibrium

The lack of equilibrium (mechanical, thermal or chemical) between the system

and its surroundings, or between two systems, or two parts of the same system, causes a spontaneous change which is irreversible. The following are specific examples in this regard:

**(a) Heat Transfer through a Finite Temperature Difference**

A heat transfer process approaches reversibility as the temperature difference between two bodies approaches zero. We define a reversible heat transfer process as one in which heat is transferred through an infinitesimal temperature difference. So to transfer a finite amount of heat through an infinitesimal temperature difference would require an infinite amount of time, or infinite area. All actual heat transfer processes are through a finite temperature difference and are, therefore, irreversible, and greater the temperature difference, the greater is the irreversibility.

We can demonstrate by the second law that heat transfer through a finite temperature difference is irreversible. Let us assume that a source at  $t_A$  and a sink at  $t_B$  ( $t_A > t_B$ ) are available, and let  $Q_{A-B}$  be the amount of heat flowing from A to B (Fig. 6.12). Let us assume an engine operating between A and B, taking heat  $Q_1$  from A and discharging heat  $Q_2$  to B. Let the heat transfer process be reversed, and  $Q_{B-A}$  be the heat flowing from B to A, and let the rate of working of the engine be such that

$$Q_2 = Q_{B-A}$$

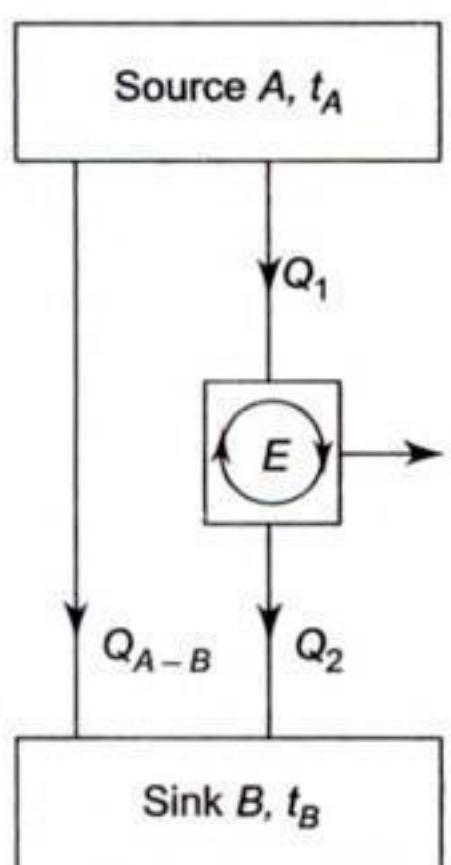


Fig. 6.12 Heat Transfer Through a Finite Temperature Difference

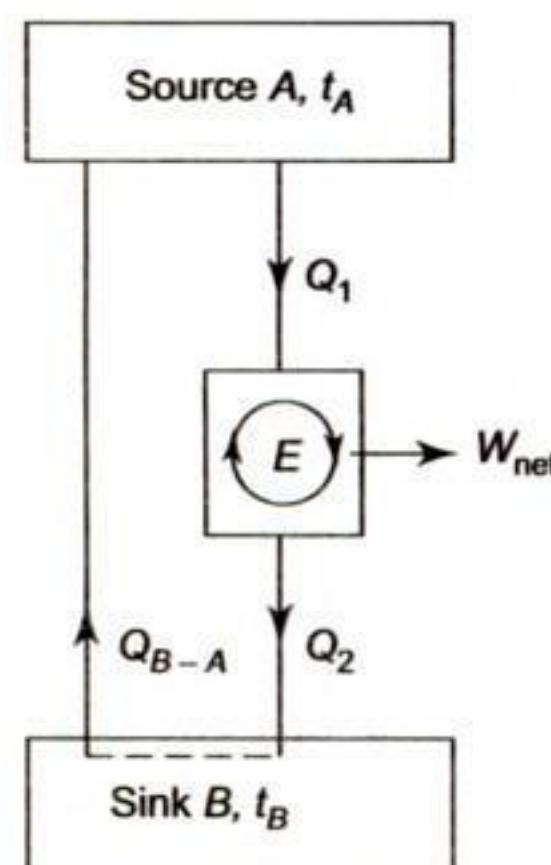


Fig. 6.13 Heat Transfer Through a Finite Temperature Difference is Irreversible

(Fig. 6.13). Then the sink B may be eliminated. The net result is that E produces network  $W$  in a cycle by exchanging heat only with A, thus violating the Kelvin-Planck statement. So the heat transfer process  $Q_{A-B}$  is irreversible, and  $Q_{B-A}$  is not possible.

**(b) Lack of Pressure Equilibrium within the Interior of the System or between the System and the Surroundings**

When there exists a difference in pressure between the system and the surroundings, or within the system itself, then both the system and its surroundings or the system alone, will undergo a change of state which will cease only when mechanical equilibrium is established. The reverse of this process is not possible spontaneously without producing any other effect. That the reverse process will violate the second law becomes obvious from the following illustration.

**(c) Free Expansion**

Let us consider an insulated container (Fig. 6.14) which is divided into two compartments *A* and *B* by a thin diaphragm. Compartment *A* contains a mass of gas, while compartment *B* is completely evacuated. If the diaphragm is punctured, the gas in *A* will expand into *B* until the pressures in *A* and *B* become equal. This is known as free or unrestrained expansion. We can demonstrate by the second law, that the process of free expansion is irreversible.

To prove this, let us assume that free expansion is reversible, and that the gas in *B* returns into *A* with an increase in pressure, and *B* becomes evacuated as before (Fig. 6.15). There is no other effect. Let us install an engine (a machine, not a cyclic heat engine) between *A* and *B*, and permit the gas to expand through the engine from *A* to *B*. The engine develops a work output *W* at the expense of the internal energy of the gas. The internal energy of the gas (system) in *B* can be restored to its initial value by heat transfer *Q* ( $= W$ ) from a source. Now, by the use of the reversed free expansion, the system can be restored to the initial state of high pressure in *A* and vacuum in *B*. The net result is a cycle, in which we observe that net work output *W* is accomplished by exchanging heat with a single reservoir. This violates the Kelvin-Planck statement. Hence, free expansion is irreversible.

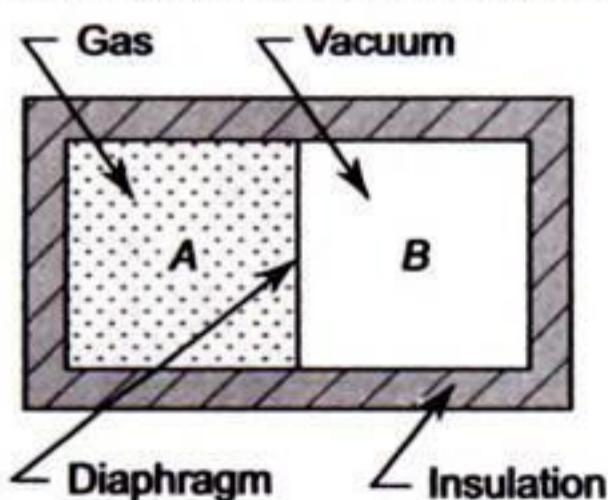


Fig. 6.14 Free Expansion

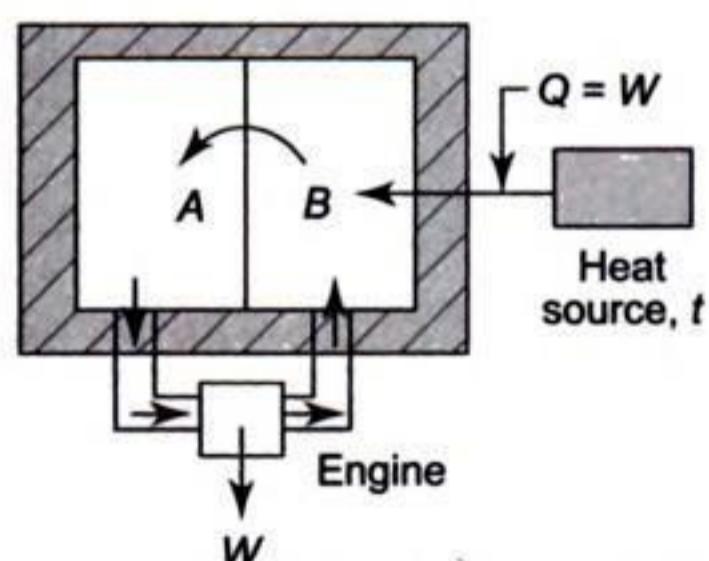


Fig. 6.15 Second Law Demonstrates that Free Expansion is Irreversible

The same argument will hold if the compartment *B* is not in vacuum but at a pressure lower than that in compartment *A* (case b).

### 6.9.2 Irreversibility due to Dissipative Effects

The irreversibility of a process may be due to the *dissipative effects* in which work is done without producing an equivalent increase in the kinetic or potential

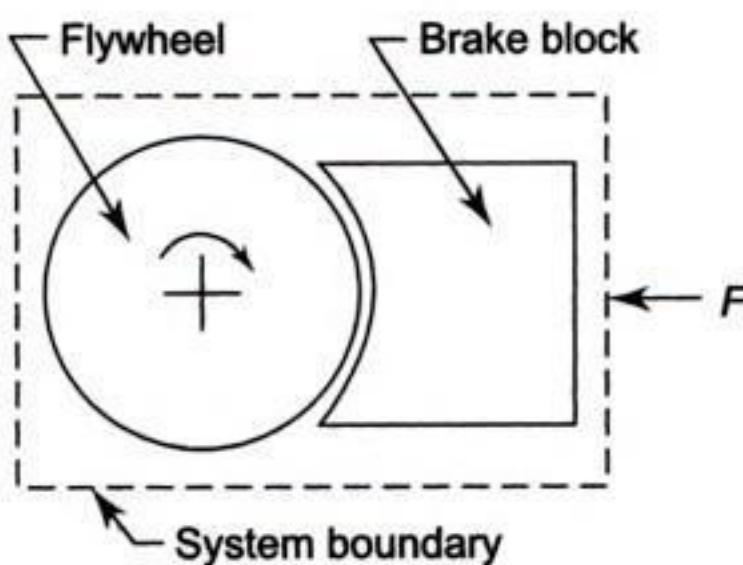
energy of any system. The transformation of work into molecular internal energy either of the system or of the reservoir takes place through the agency of such phenomena as friction, viscosity, inelasticity, electrical resistance, and magnetic hysteresis. These effects are known as dissipative effects, and work is said to be dissipated.

### (a) Friction

Friction is always present in moving devices. Friction may be reduced by suitable lubrication, but it can never be completely eliminated. If this were possible, a movable device could be kept in continual motion without violating either of the two laws of thermodynamics. The continual motion of a movable device in the complete absence of friction is known as *perpetual motion of the third kind*.

That friction makes a process irreversible can be demonstrated by the second law. Let us consider a system consisting of a flywheel and a brake block (Fig. 6.16). The flywheel was rotating with a certain rpm, and it was brought to rest by applying the friction brake. The distance moved by the brake block is very small, so work transfer is very nearly equal to zero. If the braking process occurs very rapidly, there is little heat transfer. Using suffix 2 after braking and suffix 1 before braking, and applying the first law, we have

$$\begin{aligned} Q_{1-2} &= E_2 - E_1 + W_{1-2} \\ 0 &= E_2 - E_1 + 0 \\ \therefore E_2 &= E_1 \end{aligned} \quad (6.10)$$



**Fig. 6.16 Irreversibility due to Dissipative Effect like Friction**

The energy of the system (isolated) remains constant. Since the energy may exist in the forms of kinetic, potential, and molecular internal energy, we have

$$U_2 + \frac{mV_2^2}{2} + mZ_2g = U_1 + \frac{mV_1^2}{2} + mZ_1g$$

Since the wheel is brought to rest,  $V_2 = 0$ , and there is no change in P.E.

$$U_2 = U_1 + \frac{mV_1^2}{2} \quad (6.11)$$

Therefore, the molecular internal energy of the system (i.e., of the brake and the wheel) increases by the absorption of the K.E. of the wheel. The reverse process, i.e., the conversion of this increase in molecular internal energy into K.E. within the system to cause the wheel to rotate is not possible. To prove it by the second law, let us assume that it is possible, and imagine the following cycle with three processes:

**Process A** Initially, the wheel and the brake are at high temperature as a result of the absorption of the K.E. of the wheel, and the flywheel is at rest. Let the flywheel now start rotating at a particular rpm at the expense of the internal energy of the wheel and brake, the temperature of which will then decrease.

**Process B** Let the flywheel be brought to rest by using its K.E. in raising weights, with no change in temperature.

**Process C** Now let heat be supplied from a source to the flywheel and the brake, to restore the system to its initial state.

Therefore, the processes A, B, and C together constitute a cycle producing work by exchanging heat with a single reservoir. This violates the Kelvin-Planck statement, and it will become a PMM2. So the braking process, i.e. the transformation of K.E. into molecular internal energy, is irreversible.

### (b) Paddle-Wheel Work Transfer

Work may be transferred into a system in an insulated container by means of a paddle wheel (Fig. 6.17) which is also known as stirring work. Here work transferred is dissipated adiabatically into an increase in the molecular internal energy of the system. To prove the irreversibility of the process, let us assume that the same amount of work is delivered by the system at the expense of its molecular internal energy, and the temperature of the system goes down (Fig. 6.18). The system is brought back to its initial state by heat transfer from a source. These two processes together constitute a cycle in which there is work output and the system exchanges heat with a single reservoir. It becomes a PMM2, and hence the dissipation of stirring work to internal energy is irreversible.

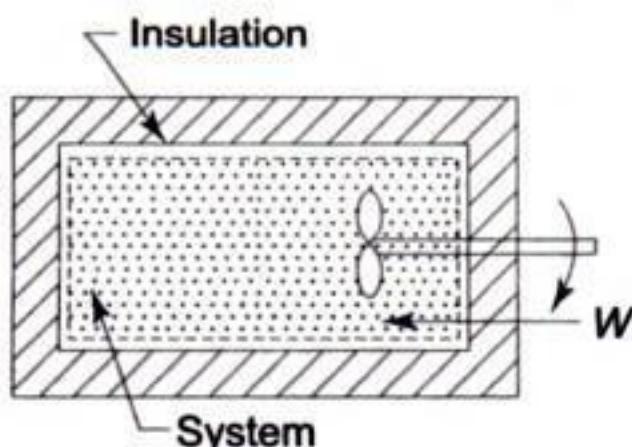


Fig. 6.17 Adiabatic Work Transfer

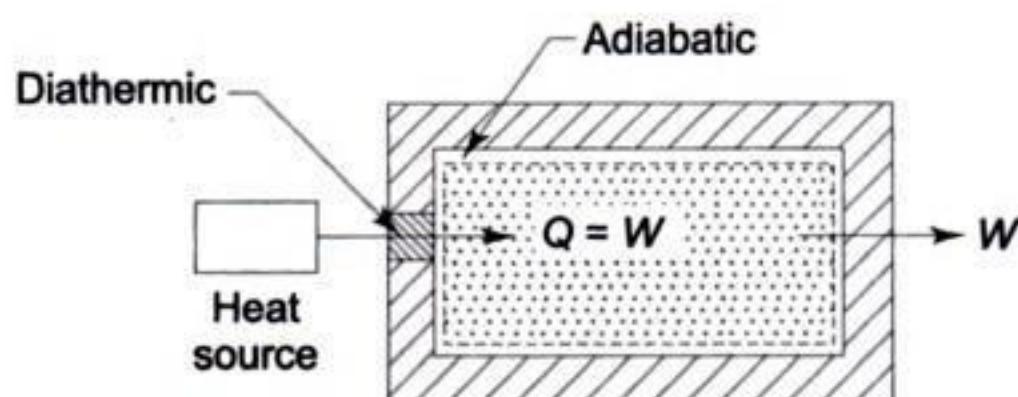


Fig. 6.18 Irreversibility Due to Dissipation of Stirring Work into Internal Energy

### (c) Transfer of Electricity through a Resistor

The flow of electric current through a wire represents work transfer, because the current can drive a motor which can raise a weight. Taking the wire or the resistor as the system (Fig. 6.19) and writing the first law

$$Q_{1-2} = U_2 - U_1 + W_{1-2}$$

Here both  $W_{1-2}$  and  $Q_{1-2}$  are negative.

$$W_{1-2} = U_2 - U_1 + Q_{1-2} \quad (6.12)$$

A part of the work transfer is stored as an increase in the internal energy of the wire (to give an increase in its temperature), and the remainder leaves the system as heat. At steady state, the internal energy and hence the temperature of the resistor become constant with respect to time and

$$W_{1-2} = Q_{1-2} \quad (6.13)$$

The reverse process, i.e. the conversion of heat  $Q_{1-2}$  into electrical work  $W_{1-2}$  of the same magnitude is not possible. Let us assume that this is possible. Then heat  $Q_{1-2}$  will be absorbed and equal work  $W_{1-2}$  will be delivered. But this will become a PMM2. So the dissipation of electrical work into internal energy or heat is irreversible.

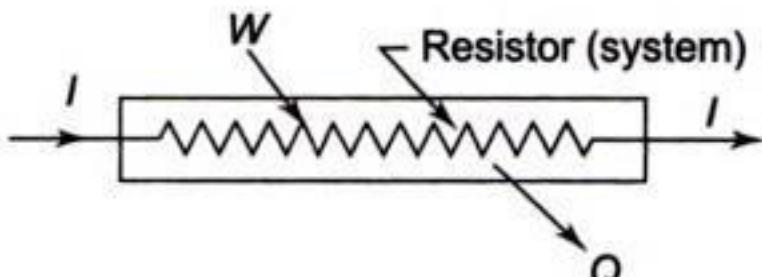


Fig. 6.19

*Irreversibility Due to Dissipation of Electrical Work into Internal Energy*

## 6.10 CONDITIONS FOR REVERSIBILITY

A natural process is irreversible because the conditions for mechanical, thermal and chemical equilibrium are not satisfied, and the dissipative effects, in which work is transformed into an increase in internal energy, are present. For a process to be reversible, it must not possess these features. If a process is performed quasi-statically, the system passes through states of thermodynamic equilibrium, which may be traversed as well in one direction as in the opposite direction. *If there are no dissipative effects, all the work done by the system during the performance of a process in one direction can be returned to the system during the reverse process.*

A process will be reversible when it is performed in such a way that the system is at all times infinitesimally near a state of thermodynamic equilibrium and in the absence of dissipative effect of any form. Reversible processes are, therefore, purely ideal, limiting cases of actual processes.

## 6.11 CARNOT CYCLE

A reversible cycle is an ideal hypothetical cycle in which all the processes constituting the cycle are reversible. Carnot cycle is a reversible cycle. For a stationary

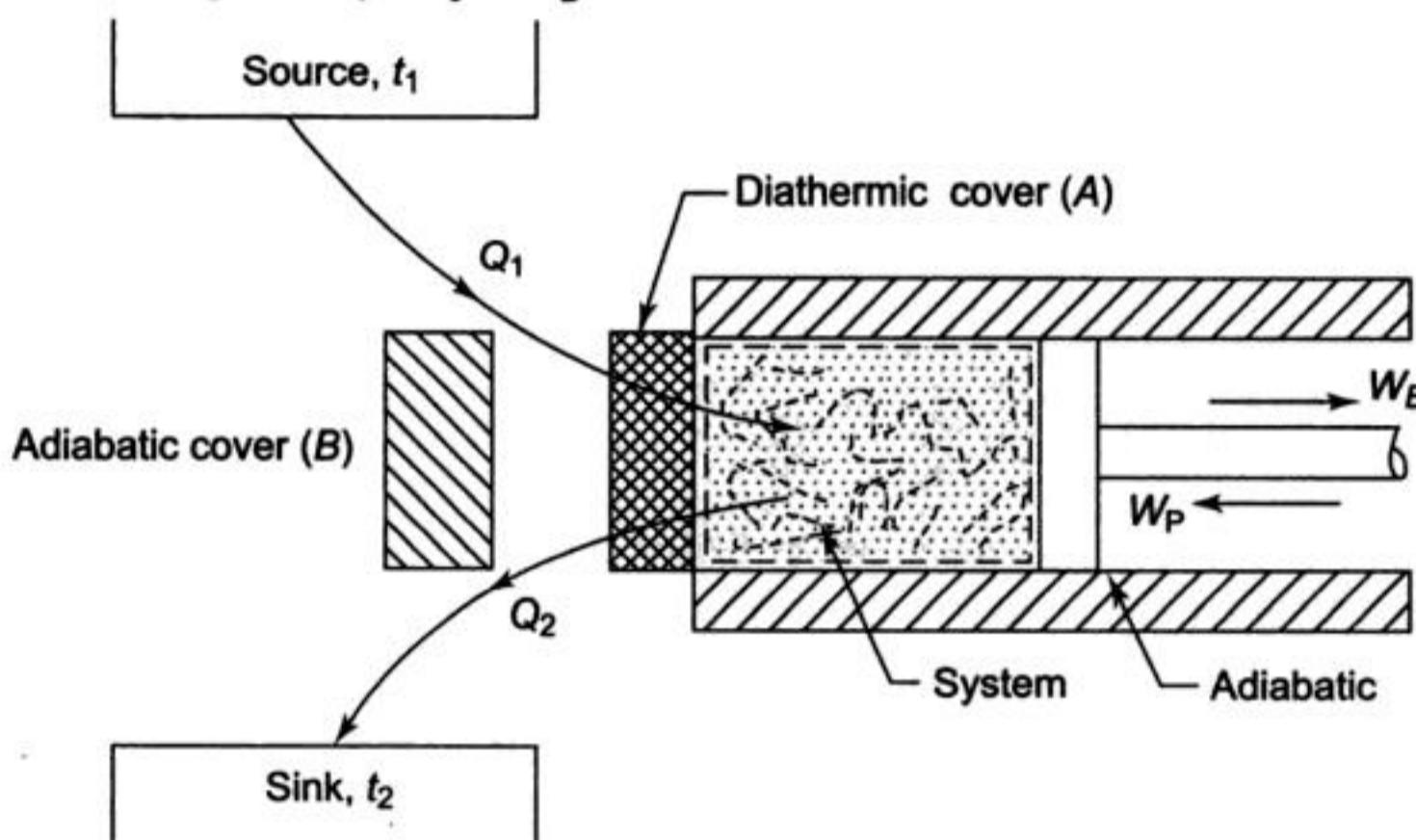
system, as in a piston and cylinder machine, the cycle consists of the following four successive processes (Fig. 6.20):

(a) A reversible isothermal process in which heat  $Q_1$  enters the system at  $t_1$  reversibly from a constant temperature source at  $t_1$  when the cylinder cover is in contact with the diathermic cover  $A$ . The internal energy of the system increases.

From First law,

$$Q_1 = U_2 - U_1 + W_{1-2} \quad (6.14)$$

(for an ideal gas only,  $U_1 = U_2$ )



**Fig. 6.20** Carnot Heat Engine—Stationary System

(b) A reversible adiabatic process in which the diathermic cover  $A$  is replaced by the adiabatic cover  $B$ , and work  $W_E$  is done by the system adiabatically and reversibly at the expense of its internal energy, and the temperature of the system decreases from  $t_1$  to  $t_2$ .

Using the first law,

$$0 = U_3 - U_2 + W_{2-3} \quad (6.15)$$

(c) A reversible isothermal process in which  $B$  is replaced by  $A$  and heat  $Q_2$  leaves the system at  $t_2$  to a constant temperature sink at  $t_2$  reversibly, and the internal energy of the system further decreases.

From the first law,

$$-Q_2 = U_4 - U_3 - W_{3-4} \quad (6.16)$$

only for an ideal gas,  $U_3 = U_4$

(d) A reversible adiabatic process in which  $B$  again replaces  $A$ , and work  $W_P$  is done upon the system reversibly and adiabatically, and the internal energy of the system increases and the temperature rises from  $t_2$  to  $t_1$ .

Applying the first law,

$$0 = U_1 - U_4 - W_{4-1} \quad (6.17)$$

Two reversible isotherms and two reversible adiabatics constitute a Carnot cycle, which is represented in  $p-v$  coordinates in Fig. 6.21.

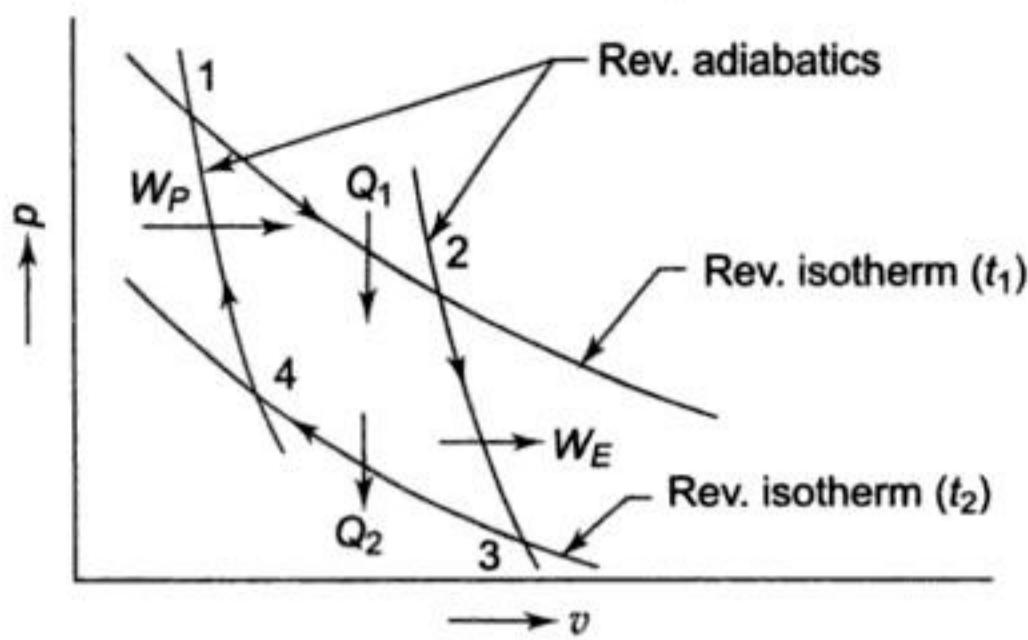


Fig. 6.21 Carnot Cycle

Summing up Eqs. (6.14) to (6.17),

$$Q_1 - Q_2 = (W_{1-2} + W_{2-3}) - (W_{3-4} + W_{4-1})$$

$$\text{or } \sum_{\text{cycle}} Q_{\text{net}} = \sum_{\text{cycle}} W_{\text{net}}$$

A cyclic heat engine operating on the Carnot cycle is called a Carnot heat engine.

For a steady flow system, the Carnot cycle is represented as shown in Fig. 6.22. Here heat  $Q_1$  is transferred to the system reversibly and isothermally at  $t_1$  in the heat exchanger  $A$ , work  $W_T$  is done by the system reversibly and adiabatically in the turbine  $(B)$ , then heat  $Q_2$  is transferred from the system reversibly and isothermally at  $t_2$  in the heat exchanger  $(C)$ , and then work  $W_p$  is done upon the system reversibly and adiabatically by the pump  $(D)$ . To satisfy the conditions for the Carnot cycle, there must not be any friction or heat transfer in the pipelines through which the working fluid flows.

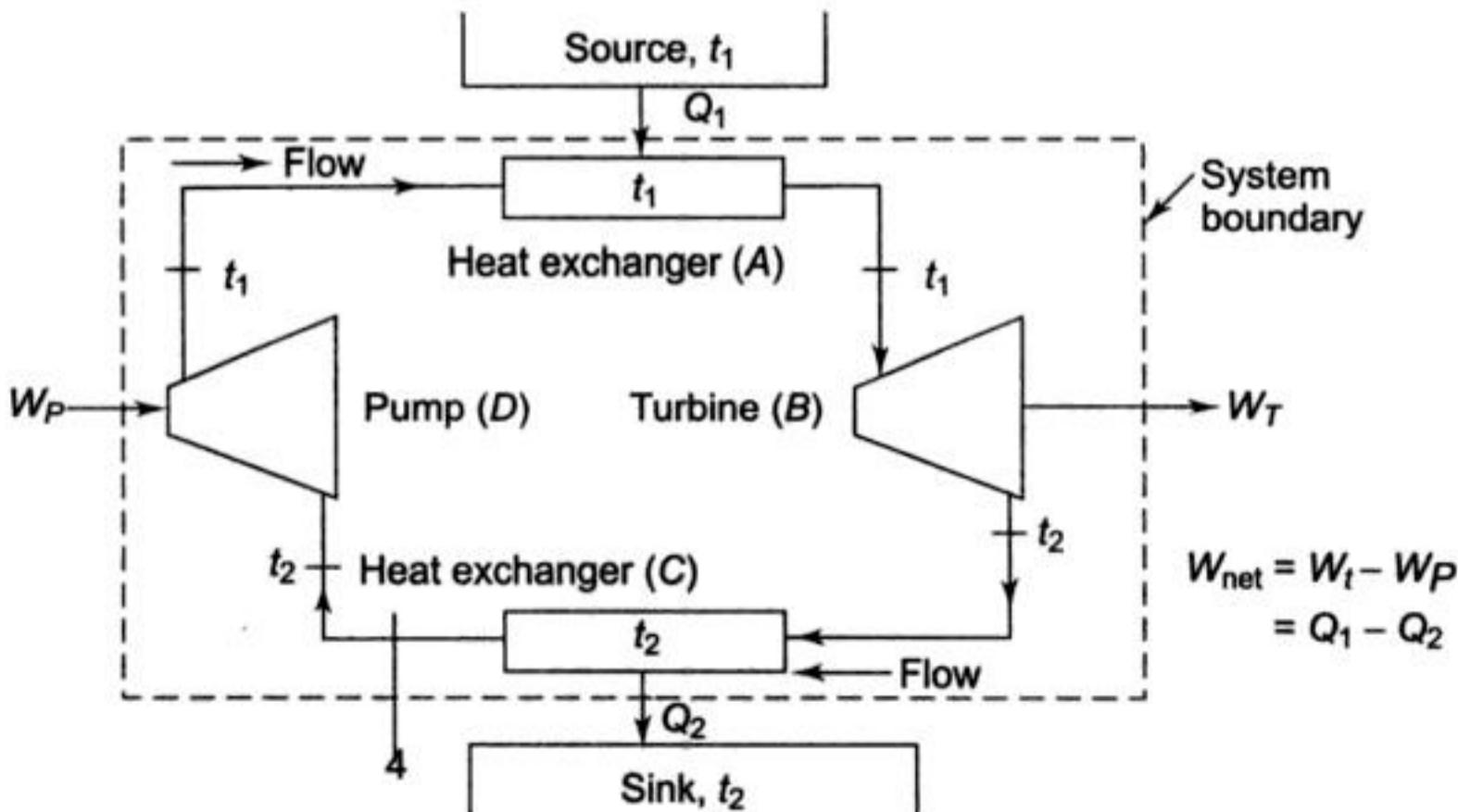


Fig. 6.22 Carnot Heat Engine—Steady Flow System

## 6.12 REVERSED HEAT ENGINE

Since all the processes of the Carnot cycle are reversible, it is possible to imagine that the processes are individually reversed and carried out in reverse order. When a reversible process is reversed, all the energy transfers associated with the process are reversed in direction, but remain the same in magnitude. The reversed Carnot cycle for a steady flow system is shown in Fig. 6.23. The reversible heat engine and the reversed Carnot heat engine are represented in block diagrams in Fig. 6.24. If  $E$  is a reversible heat engine (Fig. 6.24a), and if it is reversed (Fig. 6.24b), the quantities  $Q_1$ ,  $Q_2$  and  $W$  remain the same in magnitude, and only their directions are reversed. The reversed heat engine  $\exists$  takes heat from a low temperature body, discharges heat to a high temperature body, and receives an inward flow of network.

The names *heat pump* and *refrigerator* are applied to the reversed heat engine, which have already been discussed in Sec. 6.6, where the working fluid flows through the compressor ( $B$ ), condenser ( $A$ ), expander ( $D$ ), and evaporator ( $C$ ) to complete the cycle.

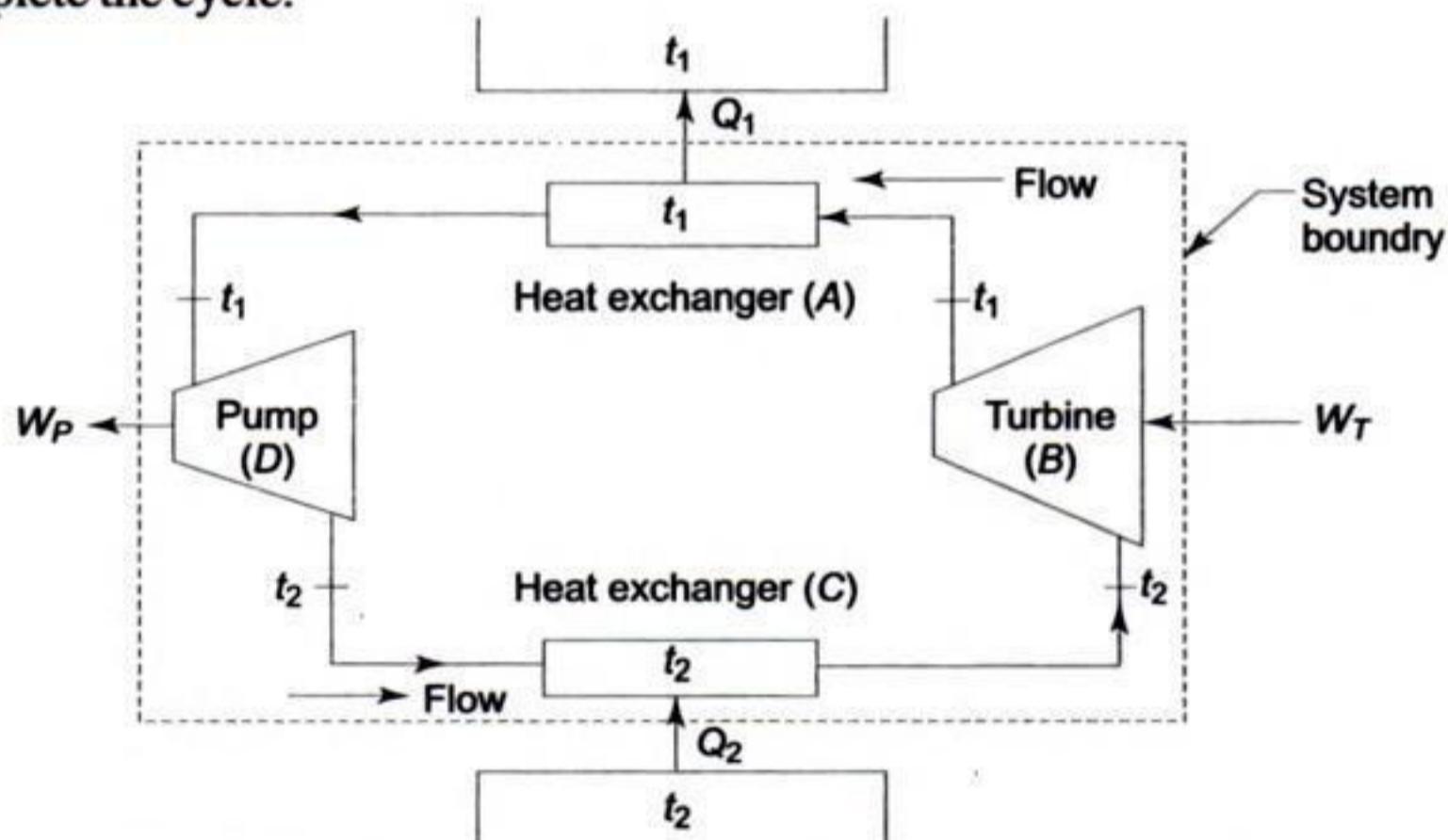


Fig. 6.23 Reversed Carnot Heat Engine—Steady Flow Process

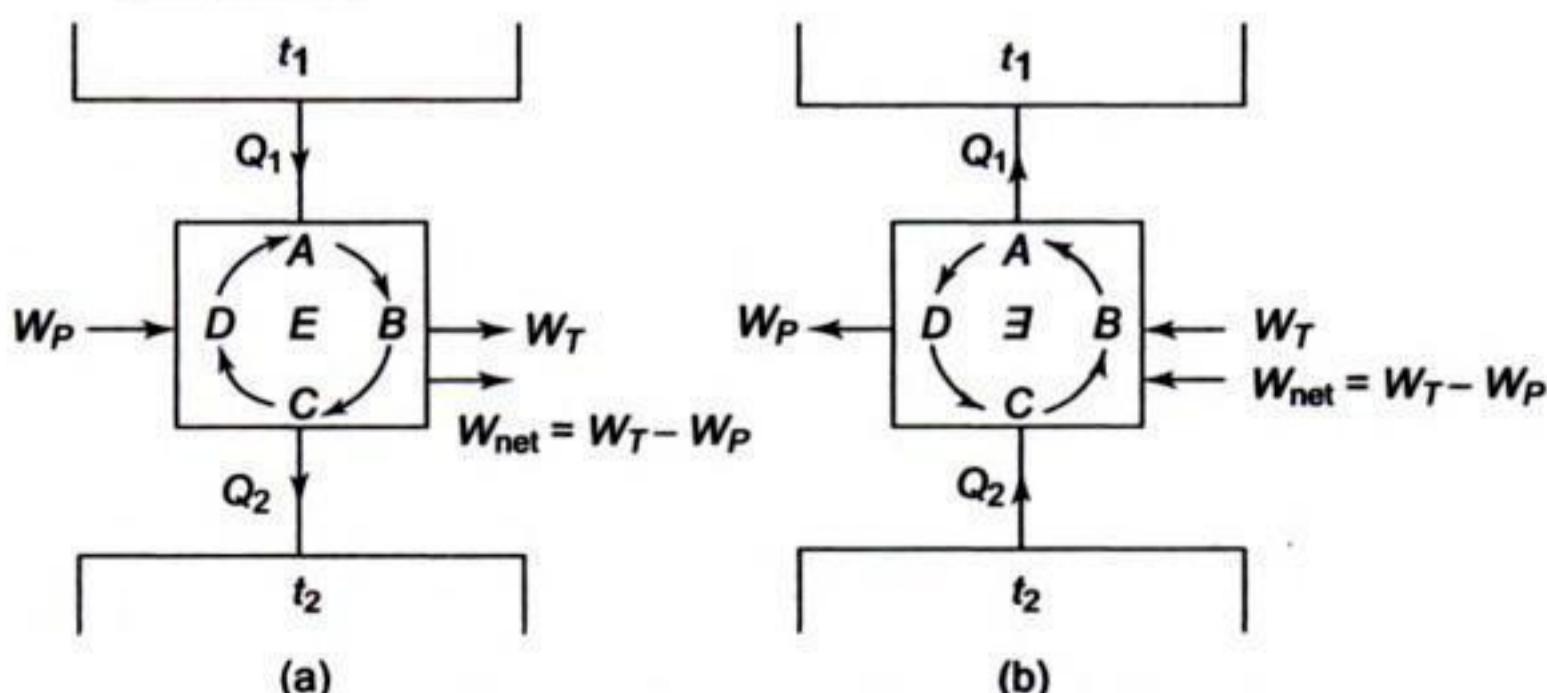


Fig. 6.24 Carnot Heat Engine and Reversed Carnot Heat Engine Shown in Block Diagrams

### 6.13 CARNOT'S THEOREM

*It states that of all heat engines operating between a given constant temperature source and a given constant temperature sink, none has a higher efficiency than a reversible engine.*

Let two heat engines  $E_A$  and  $E_B$  operate between the given source at temperature  $t_1$  and the given sink at temperature  $t_2$  as shown in Fig. 6.25.

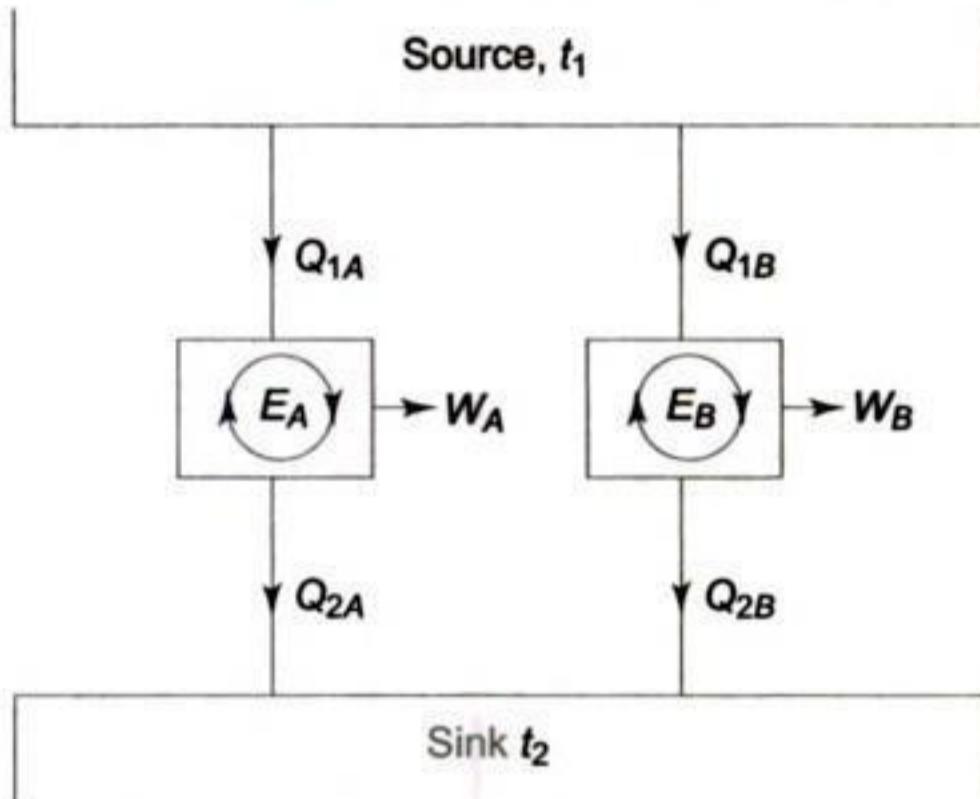


Fig. 6.25 Two Cyclic Heat Engines  $E_A$  and  $E_B$  Operating between the Same Source and Sink, of which  $E_B$  is Reversible

Let  $E_A$  be *any* heat engine and  $E_B$  be *any reversible* heat engine. We have to prove that the efficiency of  $E_B$  is more than that of  $E_A$ . Let us assume that this is not true and  $\eta_A > \eta_B$ . Let the rates of working of the engines be such that

$$Q_{1A} = Q_{1B} = Q_1$$

Since

$$\eta_A > \eta_B$$

$$\frac{W_A}{Q_{1A}} > \frac{W_B}{Q_{1B}}$$

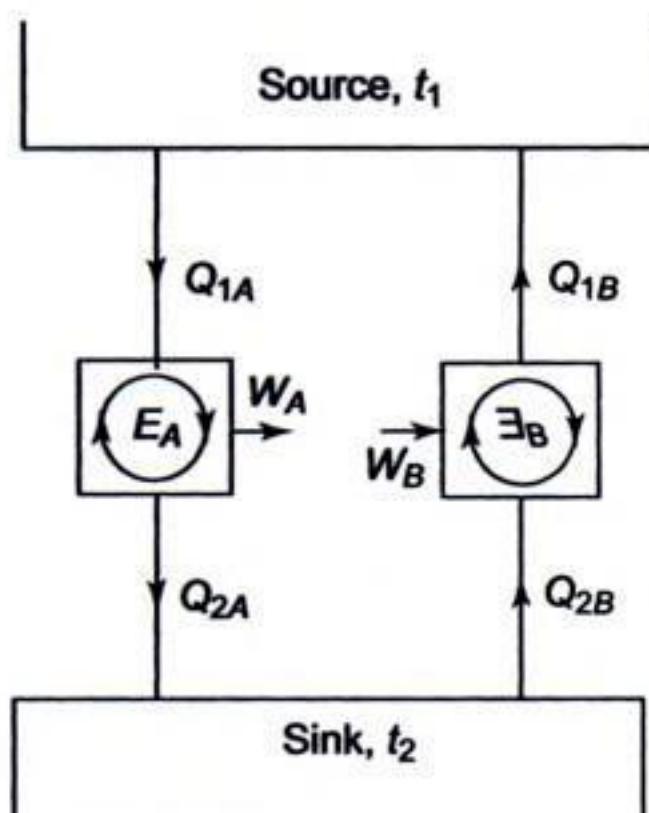
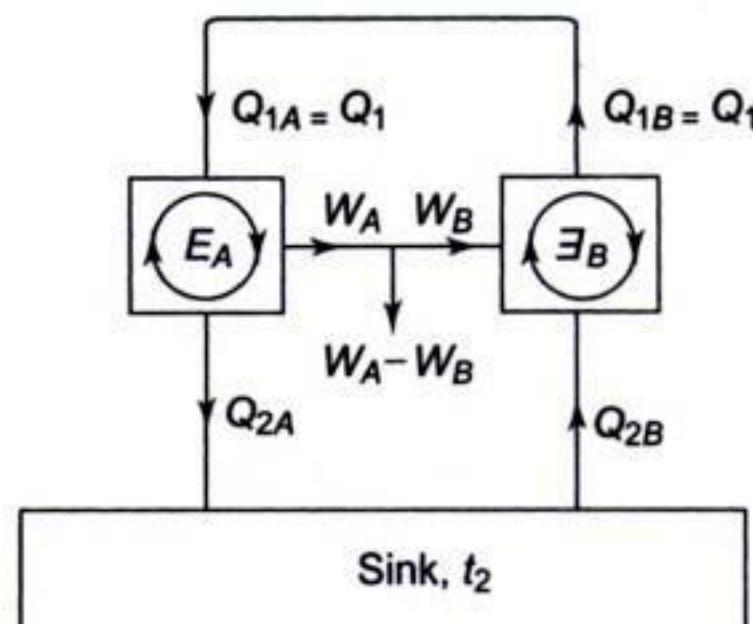
$$\therefore W_A > W_B$$

Now, let  $E_B$  be reversed. Since  $E_B$  is a reversible heat engine, the magnitudes of heat and work transfer quantities will remain the same, but their directions will be reversed, as shown in Fig. 6.26. Since  $W_A > W_B$ , some part of  $W_A$  (equal to  $W_B$ ) may be fed to drive the reversed heat engine  $\exists_B$ .

Since  $Q_{1A} = Q_{1B} = Q_1$ , the heat discharged by  $\exists_B$  may be supplied to  $E_A$ . The source may, therefore, be eliminated (Fig. 6.27). The net result is that  $E_A$  and  $\exists_B$  together constitute a heat engine which, operating in a cycle, produces net work  $W_A - W_B$ , while exchanging heat with a single reservoir at  $t_2$ . This violates the Kelvin-Planck statement of the second law. Hence the assumption that  $\eta_A > \eta_B$  is wrong.

Therefore

$$\eta_B \geq \eta_A$$

Fig. 6.26  $E_B$  is ReversedFig. 6.27  $E_A$  and  $E_B$  together Violate the K-P Statement

## 6.14 COROLLARY OF CARNOT'S THEOREM

The efficiency of all reversible heat engines operating between the same temperature levels is the same.

Let both the heat engines  $E_A$  and  $E_B$  (Fig. 6.25) be reversible. Let us assume  $\eta_A > \eta_B$ . Similar to the procedure outlined in the preceding article, if  $E_B$  is reversed to run, say, as a heat pump using some part of the work output ( $W_A$ ) of engine  $E_A$ , we see that the combined system of heat pump  $E_B$  and engine  $E_A$ , becomes a PMM2. So  $\eta_A$  cannot be greater than  $\eta_B$ . Similarly, if we assume  $\eta_B > \eta_A$  and reverse the engine  $E_A$ , we observe that  $\eta_B$  cannot be greater than  $\eta_A$ .

Therefore  $\eta_A = \eta_B$

Since the efficiencies of all reversible heat engines operating between the same heat reservoirs are the same, *the efficiency of a reversible engine is independent of the nature or amount of the working substance undergoing the cycle*.

## 6.15 ABSOLUTE THERMODYNAMIC TEMPERATURE SCALE

The efficiency of any heat engine cycle receiving heat  $Q_1$  and rejecting heat  $Q_2$  is given by

$$\eta = \frac{W_{\text{net}}}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} \quad (6.18)$$

By the second law, it is necessary to have a temperature difference  $(t_1 - t_2)$  to obtain work for any cycle. We know that the efficiency of all heat engines operating between the same temperature levels is the same, and it is independent of the working substance. Therefore, for a reversible cycle (Carnot cycle), the efficiency will depend solely upon the temperatures  $t_1$  and  $t_2$ , at which heat is transferred, or

$$\eta_{\text{rev}} = f(t_1, t_2) \quad (6.19)$$

where  $f$  signifies some function of the temperatures. From Equations (6.18) and (6.19)

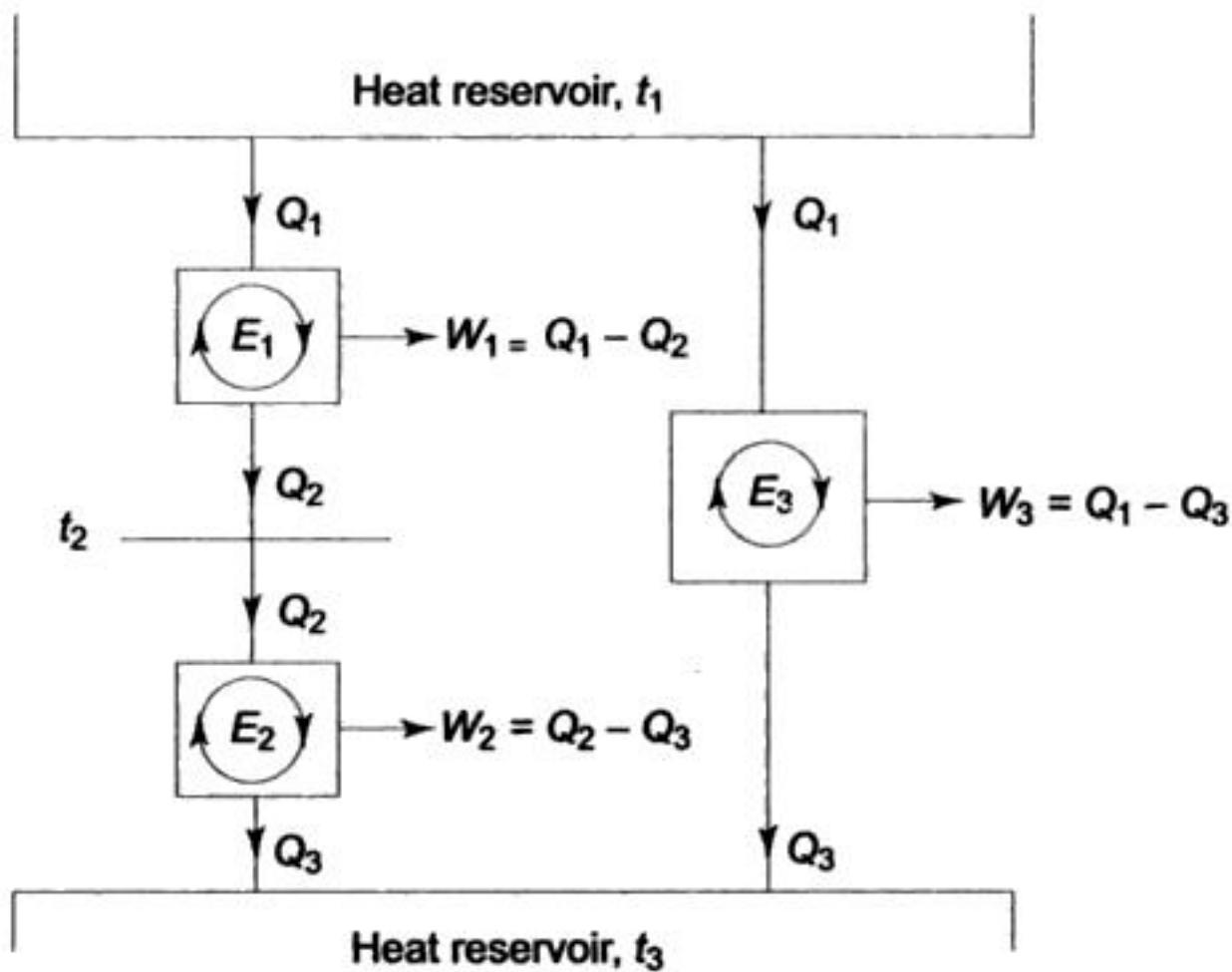
$$1 - \frac{Q_2}{Q_1} = f(t_1, t_2)$$

In terms of a new function  $F$

$$\frac{Q_1}{Q_2} = F(t_1, t_2) \quad (6.20)$$

If some functional relationship is assigned between  $t_1$ ,  $t_2$  and  $Q_1/Q_2$ , the equation becomes the definition of a temperature scale.

Let us consider two reversible heat engines,  $E_1$  receiving heat from the source at  $t_1$ , and rejecting heat at  $t_2$  to  $E_2$  which, in turn, rejects heat to the sink at  $t_3$  (Fig. 6.28).



**Fig. 6.28 Three Carnot Engines**

Now

$$\frac{Q_1}{Q_2} = F(t_1, t_2); \quad \frac{Q_2}{Q_3} = F(t_2, t_3)$$

$E_1$  and  $E_2$  together constitute another heat engine  $E_3$  operating between  $t_1$  and  $t_3$ .

∴

$$\frac{Q_1}{Q_3} = F(t_1, t_3)$$

Now

$$\frac{Q_1}{Q_2} = \frac{Q_1/Q_3}{Q_2/Q_3}$$

or

$$\frac{Q_1}{Q_2} = F(t_1, t_2) = \frac{F(t_1, t_3)}{F(t_2, t_3)} \quad (6.21)$$

The temperatures  $t_1$ ,  $t_2$  and  $t_3$  are arbitrarily chosen. The ratio  $Q_1/Q_2$  depends only on  $t_1$  and  $t_2$ , and is independent of  $t_3$ . So  $t_3$  will drop out from the ratio on the right in equation (6.21). After it has been cancelled, the numerator can be written as  $\phi(t_1)$ , and the denominator as  $\phi(t_2)$ , where  $\phi$  is another unknown function. Thus

$$\frac{Q_1}{Q_2} = F(t_1, t_2) = \frac{\phi(t_1)}{\phi(t_2)}$$

Since  $\phi(t)$  is an arbitrary function, the simplest possible way to define the *absolute thermodynamic temperature T* is to let  $\phi(t) = T$ , as proposed by Kelvin. Then, by definition

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2} \quad (6.22)$$

The absolute thermodynamic temperature scale is also known as the *Kelvin scale*. Two temperatures on the Kelvin scale bear the same relationship to each other as do the heats absorbed and rejected respectively by a Carnot engine operating between two reservoirs at these temperatures. The Kelvin temperature scale is, therefore, independent of the peculiar characteristics of any particular substance.

The heat absorbed  $Q_1$  and the heat rejected  $Q_2$  during the two reversible isothermal processes bounded by two reversible adiabatics in a Carnot engine can be measured. In defining the Kelvin temperature scale also, the triple point of water is taken as the standard reference point. For a Carnot engine operating between reservoirs at temperatures  $T$  and  $T_t$ ,  $T_t$  being the triple point of water (Fig. 6.29), arbitrarily assigned the value 273.16 K,

$$\frac{Q}{Q_t} = \frac{T}{T_t}$$

$$\therefore T = 273.16 \frac{Q}{Q_t} \quad (6.23)$$

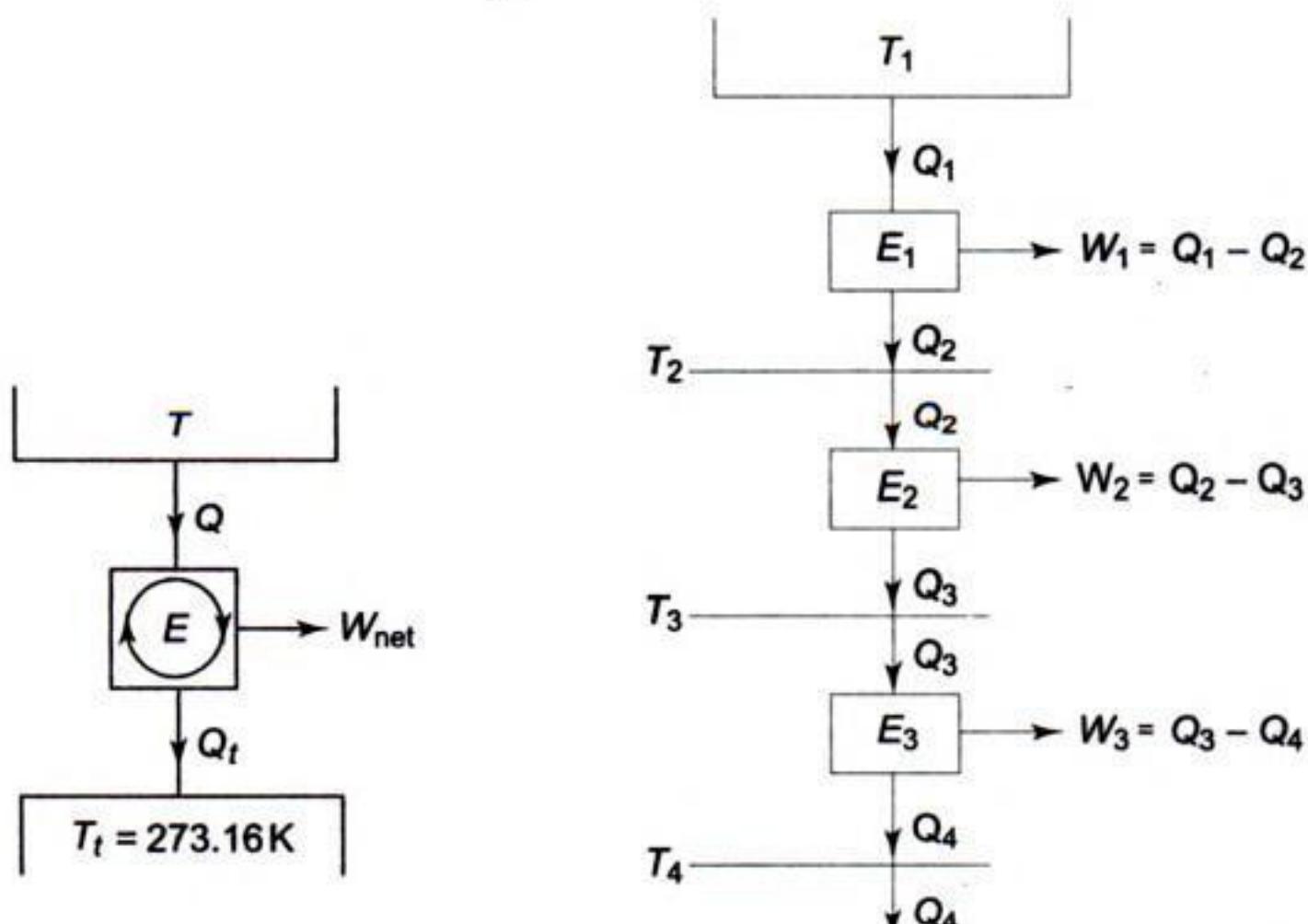


Fig. 6.29 Carnot Heat Engine with Sink at Triple Point of Water

Fig. 6.30 Heat Engines Operating in Series

If this equation is compared with the equations given in Article 2.3, it is seen that *in the Kelvin scale, Q plays the role of thermometric property*. The amount of heat supply  $Q$  changes with change in temperature, just like the thermal emf in a thermocouple.

That the absolute thermodynamic temperature scale has a definite zero point can be shown by imagining a series of reversible engines, extending from a source at  $T_1$  to lower temperatures (Fig. 6.30).

Since

$$\frac{T_1}{T_2} = \frac{Q_1}{Q_2}$$

$$\therefore \frac{T_1 - T_2}{T_2} = \frac{Q_1 - Q_2}{Q_2}$$

or  $T_1 - T_2 = (Q_1 - Q_2) \frac{T_2}{Q_2}$

Similarly

$$T_2 - T_3 = (Q_2 - Q_3) \frac{T_3}{Q_3}$$

$$= (Q_2 - Q_3) \frac{T_2}{Q_2}$$

$$T_3 - T_4 = (Q_3 - Q_4) \frac{T_2}{Q_2}$$

and so on.

If  $T_1 - T_2 = T_2 - T_3 = T_3 - T_4 = \dots$ , assuming equal temperature intervals

$$Q_1 - Q_2 = Q_2 - Q_3 = Q_3 - Q_4 = \dots$$

or  $W_1 = W_2 = W_3 = \dots$

Conversely, by making the work quantities performed by the engines in series equal ( $W_1 = W_2 = W_3 = \dots$ ), we will get

$$T_1 - T_2 = T_2 - T_3 = T_3 - T_4 = \dots$$

at equal temperature intervals. A scale having one hundred equal intervals between the steam point and the ice point could be realized by a series of one hundred Carnot engines operating as in Fig. 6.30. Such a scale would be independent of the working substance.

If enough engines are placed in series to make the total work output equal to  $Q_1$ , then by the first law the heat rejected from the last engine will be zero. By the second law, however, the operation of a cyclic heat engine with zero heat rejection cannot be achieved, although it may be approached as a limit. When the heat rejected approaches zero, the temperature of heat rejection also approaches zero as a limit. *Thus it appears that a definite zero point exist on the absolute temperature scale but this point cannot be reached without a violation of the second law.*

Thus any attainable value of absolute temperature is always greater than zero. This is also known as the *Third Law of Thermodynamics* which may be stated as follows: *It is impossible by any procedure, no matter how idealized, to reduce any system to the absolute zero of temperature in a finite number of operations.*

This is what is called the Fowler-Guggenheim statement of the third law. The third law itself is an independent law of nature, and not an extension of the second law. The concept of heat engine is not necessary to prove the non-attainability of absolute zero of temperature by any system in a finite number of operations.

## 6.16 EFFICIENCY OF THE REVERSIBLE HEAT ENGINE

The efficiency of a reversible heat engine in which heat is received solely at  $T_1$  is found to be

$$\eta_{\text{rev}} = \eta_{\max} = 1 - \left( \frac{Q_2}{Q_1} \right)_{\text{rev}} = 1 - \frac{T_2}{T_1}$$

or

$$\eta_{\text{rev}} = \frac{T_1 - T_2}{T_1}$$

It is observed here that as  $T_2$  decreases, and  $T_1$  increases, the efficiency of the reversible cycle increases.

Since  $\eta$  is always less than unity,  $T_2$  is always greater than zero and positive.

The COP of a refrigerator is given by

$$(\text{COP})_{\text{refr}} = \frac{Q_2}{Q_1 - Q_2} = \frac{1}{\frac{Q_1}{Q_2} - 1}$$

For a reversible refrigerator, using

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2}$$

$$[\text{COP}_{\text{refr}}]_{\text{rev}} = \frac{T_2}{T_1 - T_2} \quad (6.24)$$

Similarly, for a reversible heat pump

$$[\text{COP}_{H.P.}]_{\text{rev}} = \frac{T_1}{T_1 - T_2} \quad (6.25)$$

## 6.17 EQUALITY OF IDEAL GAS TEMPERATURE AND KELVIN TEMPERATURE

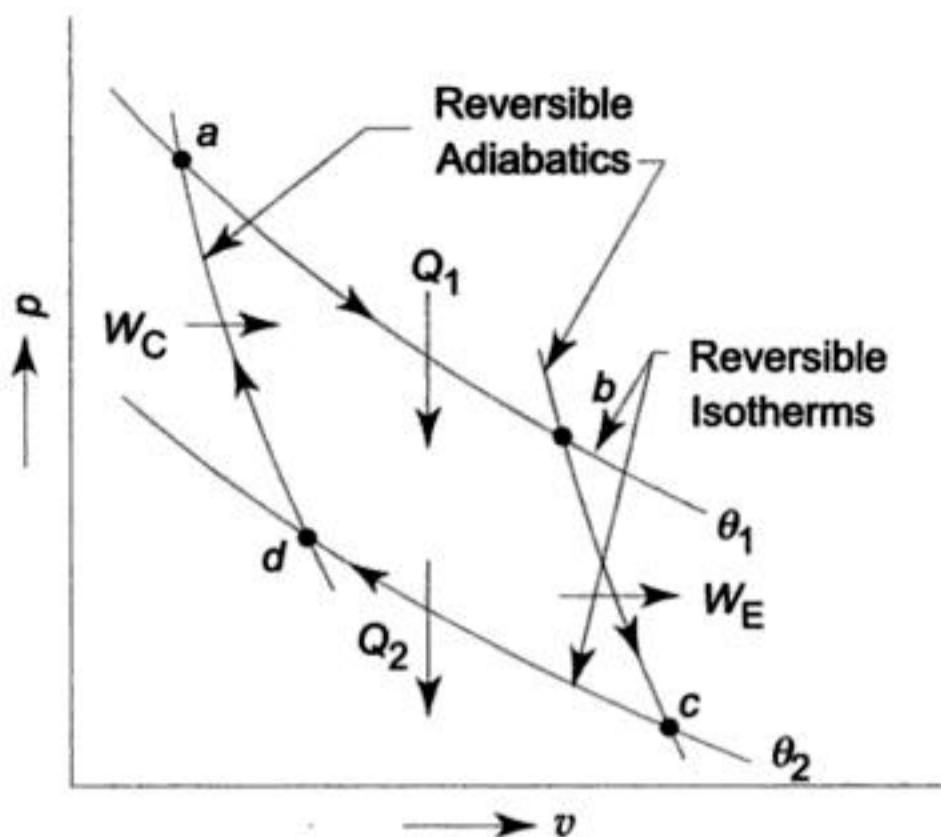
Let us consider a Carnot cycle executed by an ideal gas, as shown in Fig. 6.31.

The two isothermal processes  $a-b$  and  $c-d$  are represented by equilateral hyperbolas whose equations are respectively

$$pV = nR \theta_1$$

and

$$pV = nR \theta_2$$



**Fig. 6.31** Carnot Cycle of an Ideal Gas

For any infinitesimal reversible process of an ideal gas, the first law may be written as

$$dQ = C_v d\theta + pdV$$

Applying this equation to the isothermal process  $a-b$ , the heat absorbed is found to be

$$Q_1 = \int_{V_a}^{V_b} pdV = \int_{V_a}^{V_b} \frac{nR\theta_1}{V} dV = nR\theta_1 \ln \frac{V_b}{V_a}$$

Similarly, for the isothermal process  $c-d$ , the heat rejected is

$$Q_2 = nR\theta_2 \ln \frac{V_c}{V_d}$$

$$\therefore \frac{Q_1}{Q_2} = \frac{Q_1 \ln \frac{V_b}{V_a}}{Q_2 \ln \frac{V_c}{V_d}} \quad (6.26)$$

Since the process  $b-c$  is adiabatic, the first law gives

$$-C_v d\theta = pdV = \frac{nR\theta}{V} dV$$

$$\frac{1}{nR} \int_{\theta_2}^{\theta_1} C_v \frac{d\theta}{\theta} = \ln \frac{V_c}{V_b}$$

Similarly, for the adiabatic process  $d-a$

$$\frac{1}{nR} \int_{\theta_2}^{\theta_1} C_v \frac{d\theta}{\theta} = \ln \frac{V_d}{V_a}$$

$$\therefore \ln \frac{V_c}{V_b} = \ln \frac{V_d}{V_a}$$

or  $\frac{V_c}{V_b} = \frac{V_d}{V_a}$

or  $\frac{V_b}{V_a} = \frac{V_c}{V_d}$  (6.27)

Equation (6.26) thus reduces to

$$\frac{Q_1}{Q_2} = \frac{\theta_1}{\theta_2} \quad (6.28)$$

Kelvin temperature was defined by Eq. (6.22)

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2}$$

If  $\theta$  and  $T$  refer to any temperature, and  $\theta_t$  and  $T_t$  refer to the triple point of water,

$$\frac{\theta}{\theta_t} = \frac{T}{T_t}$$

Since  $\theta_t = T_t = 273.16$  K, it follows that

$$\theta = T \quad (6.29)$$

The Kelvin temperature is, therefore, numerically equal to the ideal gas temperature and may be measured by means of a gas thermometer.

## 6.18 TYPES OF IRREVERSIBILITY

It has been discussed in Sec. 6.9 that a process becomes irreversible if it occurs due to a finite potential gradient like the gradient in temperature or pressure, or if there is dissipative effect like friction, in which work is transformed into internal energy increase of the system. Two types of irreversibility can be distinguished:

- (a) Internal irreversibility
- (b) External irreversibility

The *internal irreversibility* is caused by the internal dissipative effects like friction, turbulence, electrical resistance, magnetic hysteresis, etc. within the system. The *external irreversibility* refers to the irreversibility occurring at the system boundary like heat interaction with the surroundings due to a finite temperature gradient.

Sometimes, it is useful to make other distinctions. If the irreversibility of a process is due to the dissipation of work into the increase in internal energy of a system, or due to a finite pressure gradient, it is called *mechanical irreversibility*. If the process occurs on account of a finite temperature gradient, it is *thermal irreversibility*, and if it is due to a finite concentration gradient or a chemical reaction, it is called *chemical irreversibility*.

A heat engine cycle in which there is a temperature difference (i) between the source and the working fluid during heat supply, and (ii) between the working fluid and the sink during heat rejection, exhibits external thermal irreversibility. If

the real source and sink are not considered and hypothetical reversible processes for heat supply and heat rejection are assumed, the cycle can be reversible. With the inclusion of the actual source and sink, however, the cycle becomes externally irreversible.

### Solved Examples

**Example 6.1** A cyclic heat engine operates between a source temperature of  $800^{\circ}\text{C}$  and a sink temperature of  $30^{\circ}\text{C}$ . What is the least rate of heat rejection per kW net output of the engine?

**Solution** For a reversible engine, the rate of heat rejection will be minimum (Fig. 6.32).

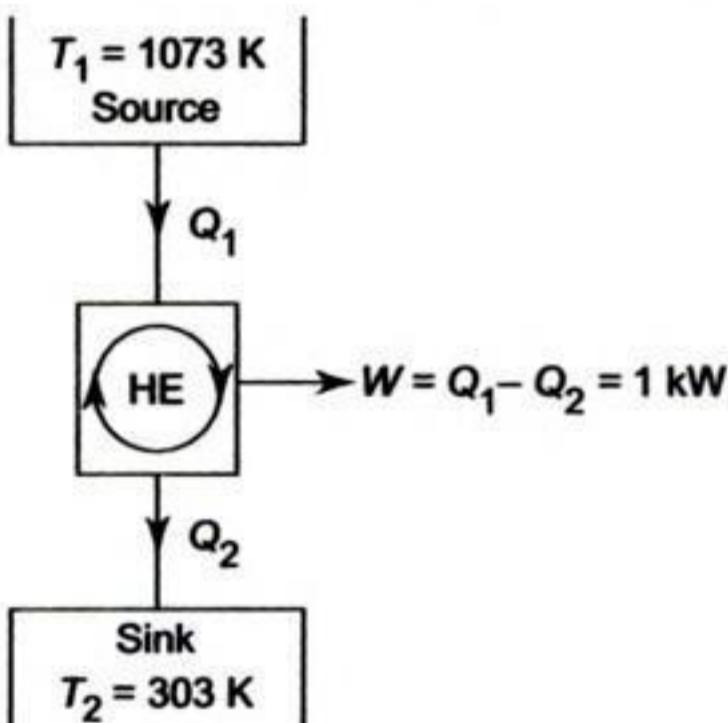


Fig. 6.32

$$\begin{aligned}\eta_{\max} &= \eta_{rev} = 1 - \frac{T_2}{T_1} \\ &= 1 - \frac{30 + 273}{800 + 273} \\ &= 1 - 0.282 = 0.718\end{aligned}$$

Now

$$\frac{W_{\text{net}}}{Q_1} = \eta_{\max} = 0.718$$

$$\therefore Q_1 = \frac{1}{0.718} = 1.392 \text{ kW}$$

Now

$$Q_2 = Q_1 - W_{\text{net}} = 1.392 - 1 = 0.392 \text{ kW}$$

This is the least rate of heat rejection.

**Example 6.2** A domestic food freezer maintains a temperature of  $-15^{\circ}\text{C}$ . The ambient air temperature is  $30^{\circ}\text{C}$ . If heat leaks into the freezer at the continuous rate of  $1.75 \text{ kJ/s}$  what is the least power necessary to pump this heat out continuously?

**Solution** Freezer temperature,

$$T_2 = -15 + 273 = 258 \text{ K}$$

Ambient air temperature,

$$T_1 = 30 + 273 = 303 \text{ K}$$

The refrigerator cycle removes heat from the freezer at the same rate at which heat leaks into it (Fig. 6.33).

For minimum power requirement

$$\frac{Q_2}{T_2} = \frac{Q_1}{T_1}$$

$$\therefore Q_1 = \frac{1.75}{2.8} \times 303 = 2.06 \text{ kJ/s}$$

$$\therefore W = Q_1 - Q_2 \\ = 2.06 - 1.75 = 0.31 \text{ kJ/s} = 0.31 \text{ kW}$$

**Example 6.3** A reversible heat engine operates between two reservoirs at temperatures of  $600^\circ\text{C}$  and  $40^\circ\text{C}$ . The engine drives a reversible refrigerator which operates between reservoirs at temperatures of  $40^\circ\text{C}$  and  $-20^\circ\text{C}$ . The heat transfer to the heat engine is  $2000 \text{ kJ}$  and the net work output of the combined engine-refrigerator plant is  $360 \text{ kJ}$ .

(a) Evaluate the heat transfer to the refrigerant and the net heat transfer to the reservoir at  $40^\circ\text{C}$ .

(b) Reconsider (a) given that the efficiency of the heat engine and the COP of the refrigerator are each 40% of their maximum possible values.

**Solution** (a) Maximum efficiency of the heat engine cycle (Fig. 6.34) is given by

$$\eta_{\max} = 1 - \frac{T_2}{T_1} = 1 - \frac{313}{873} = 1 - 0.358 = 0.642$$

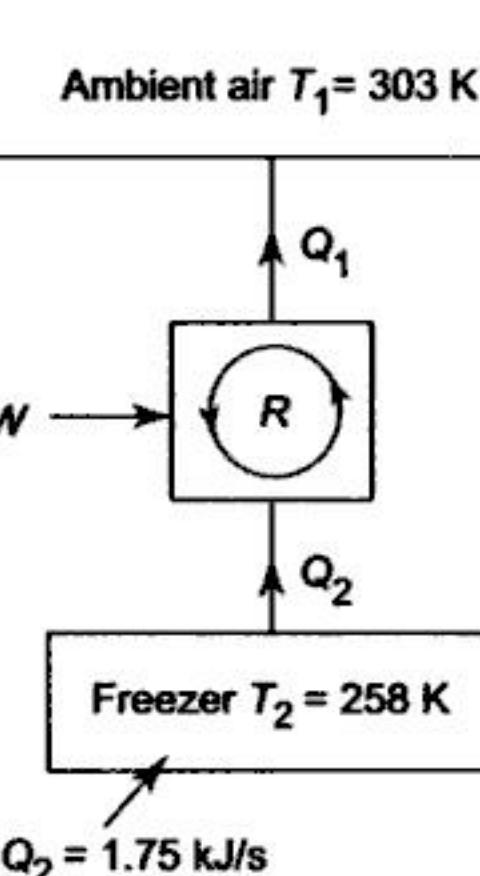
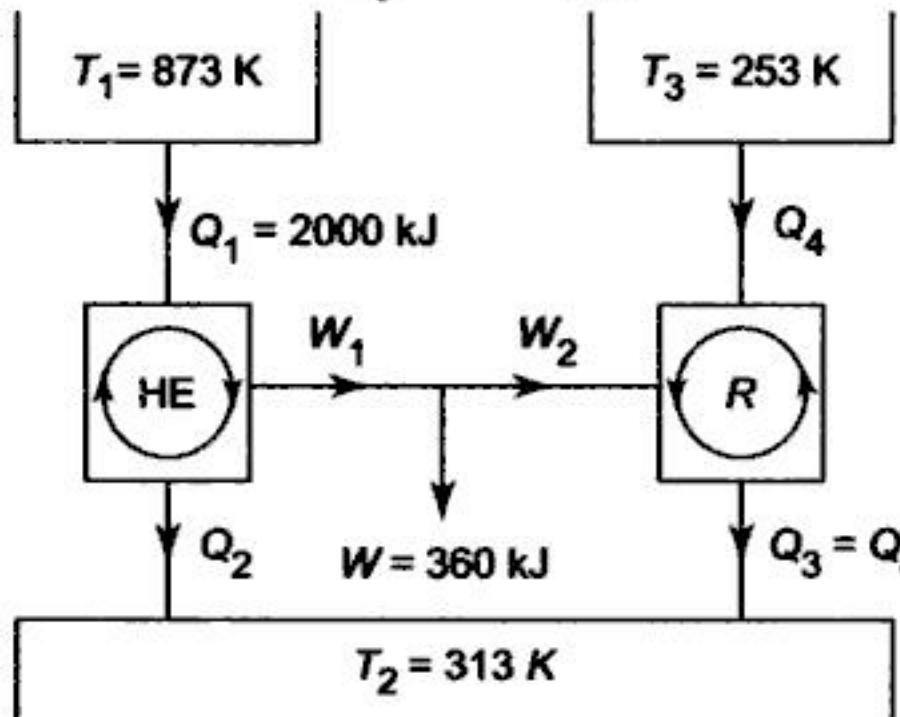


Fig. 6.33

Fig. 6.34

Again

$$\frac{W_1}{Q_1} = 0.642$$

$$\therefore W_1 = 0.642 \times 2000 = 1284 \text{ kJ}$$

Maximum COP of the refrigerator cycle

$$(\text{COP})_{\max} = \frac{T_3}{T_2 - T_3} = \frac{253}{313 - 253} = 4.22$$

Also

$$\text{COP} = \frac{Q_4}{W_2} = 4.22$$

Since

$$W_1 - W_2 = W = 360 \text{ kJ}$$

$$\therefore W_2 = W_1 - W = 1284 - 360 = 924 \text{ kJ}$$

$$\therefore Q_4 = 4.22 \times 924 = 3899 \text{ kJ}$$

$$\therefore Q_3 = Q_4 + W_2 = 924 + 3899 = 4823 \text{ kJ}$$

$$Q_2 = Q_1 - W_1 = 2000 - 1284 = 716 \text{ kJ}$$

Heat rejection to the 40°C reservoir

$$= Q_2 + Q_3 = 716 + 4823 = 5539 \text{ kJ}$$

(b) Efficiency of the actual heat engine cycle

$$\eta = 0.4 \quad \eta_{\max} = 0.4 \times 0.642$$

$$\therefore W_1 = 0.4 \times 0.642 \times 2000 \\ = 513.6 \text{ kJ}$$

$$\therefore W_2 = 513.6 - 360 = 153.6 \text{ kJ}$$

COP of the actual refrigerator cycle

$$\text{COP} = \frac{Q_4}{W_2} = 0.4 \times 4.22 = 1.69$$

Therefore

$$Q_4 = 153.6 \times 1.69 = 259.6 \text{ kJ}$$

$$Q_3 = 259.6 + 153.6 = 413.2 \text{ kJ}$$

$$Q_2 = Q_1 - W_1 = 2000 - 513.6 = 1486.4 \text{ kJ}$$

Heat rejected to the 40°C reservoir

$$= Q_2 + Q_3 = 413.2 + 1486.4 = 1899.6 \text{ kJ}$$

**Example 6.4** Which is the more effective way to increase the efficiency of a Carnot engine: to increase  $T_1$ , keeping  $T_2$  constant; or to decrease  $T_2$ , keeping  $T_1$  constant?

**Solution** The efficiency of a Carnot engine is given by

$$\eta = 1 - \frac{T_2}{T_1}$$

If  $T_2$  is constant

$$\left( \frac{\partial \eta}{\partial T_1} \right)_{T_2} = \frac{T_2}{T_1^2}$$

As  $T_1$  increases,  $\eta$  increases, and the slope  $\left( \frac{\partial \eta}{\partial T_1} \right)_{T_2}$  decreases (Fig. 6.35). If  $T_1$  is constant,

$$\left( \frac{\partial \eta}{\partial T_2} \right)_{T_1} = -\frac{1}{T_1}$$

As  $T_2$  decreases,  $\eta$  increases, but the slope  $\left( \frac{\partial \eta}{\partial T_2} \right)_{T_1}$  remains constant (Fig. 6.36).

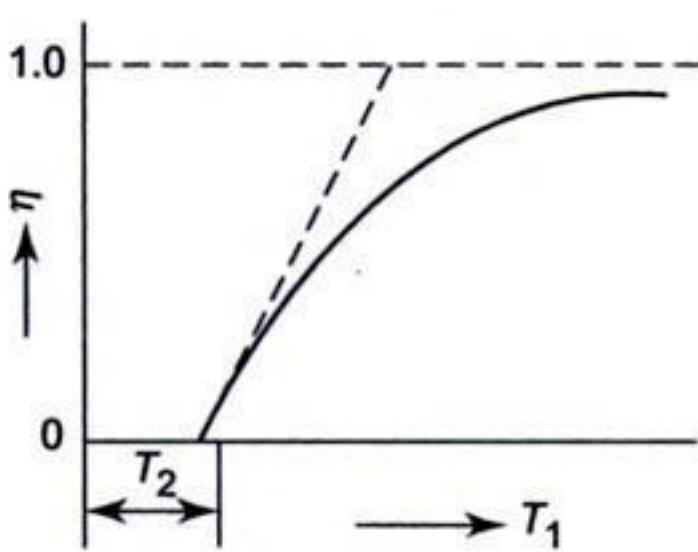


Fig. 6.35

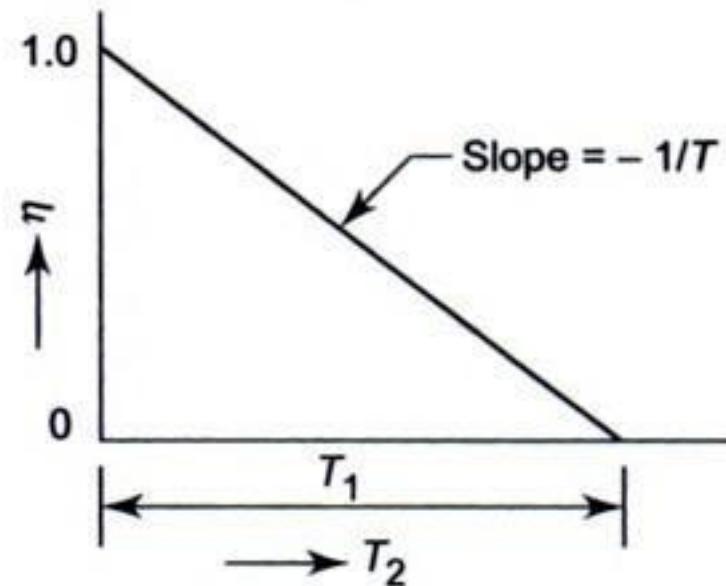


Fig. 6.36

Also

$$\left( \frac{\partial \eta}{\partial T_1} \right)_{T_2} = \frac{T_2}{T_1^2} \text{ and } \left( \frac{\partial \eta}{\partial T_2} \right)_{T_1} = -\frac{T_1}{T_2^2}$$

Since

$$T_1 > T_2, \left( \frac{\partial \eta}{\partial T_2} \right)_{T_1} > \left( \frac{\partial \eta}{\partial T_1} \right)_{T_2}$$

So, the more effective way to increase the efficiency is to decrease  $T_2$ . Alternatively, let  $T_2$  be decreased by  $\Delta T$  with  $T_1$  remaining the same

$$\eta_1 = 1 - \frac{T_2 - \Delta T}{T_1}$$

If  $T_1$  is increased by the same  $\Delta T$ ,  $T_2$  remaining the same

$$\eta_2 = 1 - \frac{T_2}{T_1 + \Delta T}$$

Then

$$\eta_1 - \eta_2 = \frac{T_2}{T_1 + \Delta T} - \frac{T_2 - \Delta T}{T_1}$$

$$= \frac{(T_1 - T_2)\Delta T + (\Delta T)^2}{T_1(T_1 + \Delta T)}$$

Since  $T_1 > T_2, (\eta_1 - \eta_2) > 0$

The more effective way to increase the cycle efficiency is to decrease  $T_2$ .

**Example 6.5** Kelvin was the first to point out the thermodynamic wastefulness of burning fuel for the direct heating of a house. It is much more economical to use the high temperature heat produced by combustion in a heat engine and then to use the work so developed to pump heat from outdoors up to the temperature desired in the house. In Fig. 6.37 a boiler furnishes heat  $Q_1$  at the high temperature  $T_1$ . This heat is absorbed by a heat engine, which extracts work  $W$  and rejects the waste heat  $Q_2$  into the house at  $T_2$ . Work  $W$  is in turn used to operate a mechanical refrigerator or heat pump, which extracts  $Q_3$  from outdoors at temperature  $T_3$  and rejects  $Q'_2$  (where  $Q'_2 = Q_3 + W$ ) into the house. As a result of this cycle of operations, a total quantity of heat equal to  $Q_2 + Q'_2$  is liberated in the house, against  $Q_1$  which would be provided directly by the ordinary combustion of the fuel. Thus the ratio  $(Q_2 + Q'_2)/Q_1$  represents the heat multiplication factor of this method. Determine this multiplication factor if  $T_1 = 473$  K,  $T_2 = 293$  K, and  $T_3 = 273$  K.

**Solution** For the reversible heat engine (Fig. 6.37)

$$\frac{Q_2}{Q_1} = \frac{T_2}{T_1}$$

$$\therefore Q_2 = Q_1 \left( \frac{T_2}{T_1} \right)$$

$$\text{Also } \eta = \frac{W}{Q_1} = \frac{T_1 - T_2}{T_1}$$

$$\text{or } W = \frac{T_1 - T_2}{T_1} \cdot Q_1$$

For the reversible heat pump

$$\text{COP} = \frac{Q'_2}{W} = \frac{T_2}{T_2 - T_3}$$

$$\therefore Q'_2 = \frac{T_2}{T_2 - T_3} \cdot \frac{T_1 - T_2}{T_1} \cdot Q_1$$

$\therefore$  Multiplication factor (M.F.)

$$= \frac{Q_2 + Q'_2}{Q_1} = \frac{Q_1 \frac{T_2}{T_1} + Q_1 \cdot \frac{T_2}{T_2 - T_3} \cdot \frac{T_1 - T_2}{T_1}}{Q_1}$$

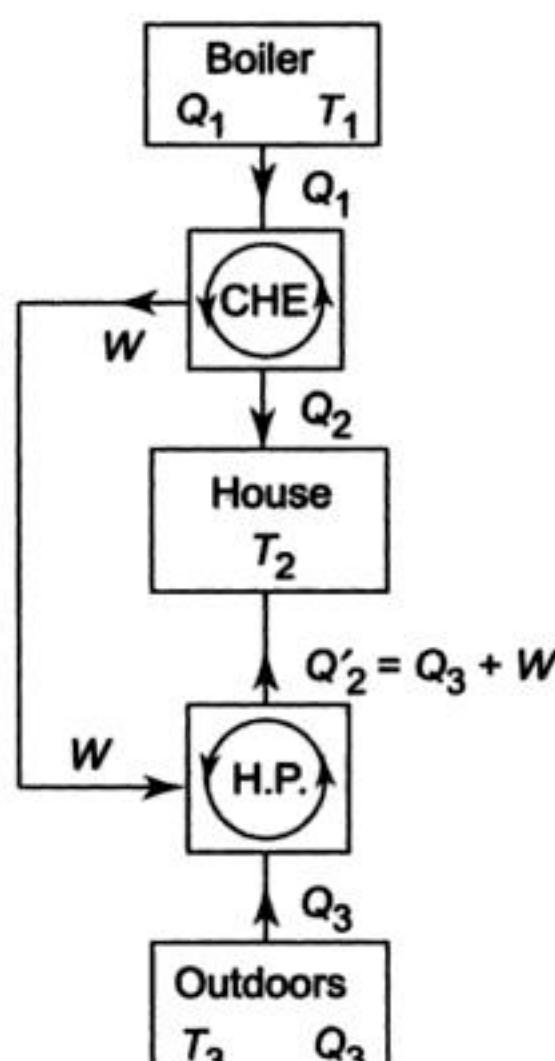


Fig. 6.37

or

$$\text{M.F.} = \frac{T_2^2 - T_2 T_3 + T_2 T_1 - T_2^2}{T_1(T_2 - T_3)}$$

or

$$\text{M.F.} = \frac{T_2(T_1 - T_3)}{T_1(T_2 - T_3)}$$

Here

$$T_1 = 473 \text{ K}, T_2 = 293 \text{ K} \text{ and } T_3 = 273 \text{ K}$$

$$\therefore \text{M.F.} = \frac{293(473 - 273)}{473(293 - 273)} = \frac{2930}{473} = 6.3$$

which means that every kg of coal burned would deliver the heat equivalent to over 6 kg. Of course, in an actual case, the efficiencies would be less than Carnot efficiencies, but even with a reduction of 50%, the possible savings would be quite significant.

**Example 6.6** It is proposed that solar energy be used to warm a large collector plate. This energy would, in turn, be transferred as heat to a fluid within a heat engine, and the engine would reject energy as heat to the atmosphere. Experiments indicate that about  $1880 \text{ kJ/m}^2 \text{ h}$  of energy can be collected when the plate is operating at  $90^\circ\text{C}$ . Estimate the minimum collector area that would be required for a plant producing 1 kW of useful shaft power. The atmospheric temperature may be assumed to be  $20^\circ\text{C}$ .

**Solution** The maximum efficiency for the heat engine operating between the collector plate temperature and the atmospheric temperature (Fig. 6.38) as follows:

$$\eta_{\max} = 1 - \frac{T_2}{T_1} = 1 - \frac{293}{363} = 0.192$$

The efficiency of any actual heat engine operating between these temperatures would be less than this efficiency.

$$\begin{aligned} \therefore Q_{\min} &= \frac{W}{\eta_{\max}} = \frac{1 \text{ kJ/s}}{0.192} = 5.21 \text{ kJ/s} \\ &= 18,800 \text{ kJ/h} \end{aligned}$$

$\therefore$  Minimum area required for the collector plate

$$= \frac{18,800}{1880} = 10 \text{ m}^2$$

**Example 6.7** A reversible heat engine in a satellite operates between a hot reservoir at  $T_1$  and a radiating panel at  $T_2$ . Radiation from the panel is proportional to its area and to  $T_2^4$ . For a given work output and value of  $T_1$  show that the

area of the panel will be minimum when  $\frac{T_2}{T_1} = 0.75$ .

Determine the minimum area of the panel for an output of 1 kW if the constant of proportionality is  $5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$  and  $T_1$  is 1000 K.

**Solution** For the heat engine (Fig. 6.38), the heat rejected  $Q_2$  to the panel (at  $T_2$ ) is equal to the energy emitted from the panel to the surroundings by radiation. If  $A$  is the area of the panel,  $Q_2 \propto AT_2^4$ , or  $Q_2 = KAT_2^4$ , where  $K$  is a constant.

Now,

$$\eta = \frac{W}{Q_1} = \frac{T_1 - T_2}{T_1}$$

or

$$\frac{W}{T_1 - T_2} = \frac{Q_1}{T_1} = \frac{Q_2}{T_2} = \frac{KAT_2^4}{T_2}$$

$$= KAT_2^3$$

∴

$$A = \frac{W}{KT_2^3(T_1 - T_2)} = \frac{W}{K(T_1 T_2^3 - T_2^4)}$$

For a given  $W$  and  $T_1$ ,  $A$  will be minimum when

$$\frac{dA}{dT_2} = -\frac{W}{K} (3T_1 T_2^2 - 4T_2^3) \cdot (T_1 T_2^3 - T_2^4)^{-2} = 0$$

Since  $(T_1 T_2^3 - T_2^4)^{-2} \neq 0$ ,  $3T_1 T_2^2 = 4T_2^3$

∴

$$\frac{T_2}{T_1} = 0.75 \text{ Proved.}$$

∴

$$A_{\min} = \frac{W}{K(0.75)^3 T_1^3 (T_1 - 0.75 T_1)}$$

$$= \frac{W}{K \frac{27}{256} T_1^4} = \frac{256 W}{27 K T_1^4}$$

Here

$$W = 1 \text{ kW}, K = 5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4, \text{ and } T_1 = 1000 \text{ K}$$

∴

$$A_{\min} = \frac{256 \times 1 \text{ kW} \times \text{m}^2 \text{ K}^4}{27 \times 5.67 \times 10^{-8} \text{ W} \times (1000)^4 \text{ K}^4}$$

$$= \frac{256 \times 10^3}{27 \times 5.67 \times 10^{-8} \times 10^{12}} \text{ m}^2 = 0.1672 \text{ m}^2$$

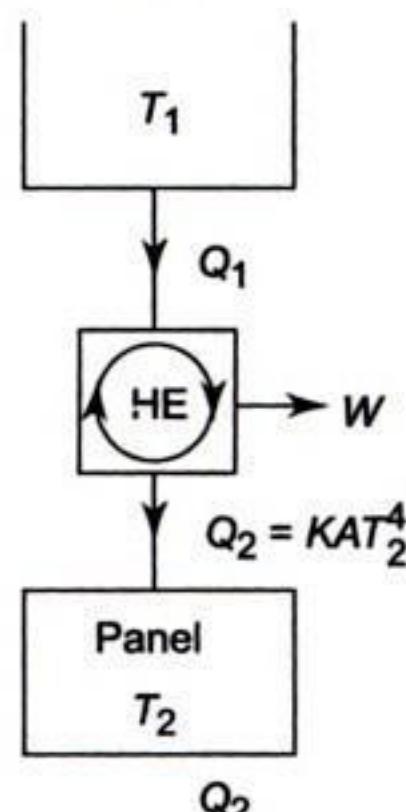


Fig. 6.38

## Review Questions

- 6.1 What is the qualitative difference between heat and work? Why are heat and work not completely interchangeable forms of energy?
- 6.2 What is a cyclic heat engine?
- 6.3 Explain a heat engine cycle performed by a closed system.
- 6.4 Explain a heat engine cycle performed by a steady flow system.
- 6.5 Define the thermal efficiency of a heat engine cycle. Can this be 100%?
- 6.6 Draw a block diagram showing the four energy interactions of a cyclic heat engine.

- 6.7 What is a thermal energy reservoir? Explain the terms 'source' and 'sink'.
- 6.8 What is a mechanical energy reservoir?
- 6.9 Why can all processes in a TER or an MER be assumed to be quasi-static?
- 6.10 Give the Kelvin-Planck statement of the second law.
- 6.11 To produce net work in a thermodynamic cycle, a heat engine has to exchange heat with two thermal reservoirs. Explain.
- 6.12 What is a PMM2? Why is it impossible?
- 6.13 Give the Clausius' statement of the second law.
- 6.14 Explain the operation of a cyclic refrigerator plant with a block diagram.
- 6.15 Define the COP of a refrigerator.
- 6.16 What is a heat pump? How does it differ from a refrigerator?
- 6.17 Can you use the same plant as a heat pump in winter and as a refrigerator in summer? Explain.
- 6.18 Show that the COP of a heat pump is greater than the COP of a refrigerator by unity.
- 6.19 Why is direct heating thermodynamically wasteful?
- 6.20 How can a heat pump upgrade low grade waste heat?
- 6.21 Establish the equivalence of Kelvin-Planck and Clausius statements.
- 6.22 What is a reversible process? A reversible process should not leave any evidence to show that the process had ever occurred. Explain.
- 6.23 How is a reversible process only a limiting process, never to be attained in practice?
- 6.24 All spontaneous processes are irreversible. Explain.
- 6.25 What are the causes of irreversibility of a process?
- 6.26 Show that heat transfer through a finite temperature difference is irreversible.
- 6.27 Demonstrate, using the second law, that free expansion is irreversible.
- 6.28 What do you understand by dissipative effects? When is work said to be dissipated?
- 6.29 Explain perpetual motion of the third kind.
- 6.30 Demonstrate using the second law how friction makes a process irreversible.
- 6.31 When a rotating wheel is brought to rest by applying a brake, show that the molecular internal energy of the system (of the brake and the wheel) increases.
- 6.32 Show that the dissipation of stirring work to internal energy is irreversible.
- 6.33 Show by second law that the dissipation of electrical work into internal energy or heat is irreversible.
- 6.34 What is a Carnot cycle? What are the four processes which constitute the cycle?
- 6.35 Explain the Carnot heat engine cycle executed by: (a) a stationary system, and (b) a steady flow system.
- 6.36 What is a reversed heat engine?
- 6.37 Show that the efficiency of a reversible engine operating between two given constant temperatures is the maximum.
- 6.38 Show that the efficiency of all reversible heat engines operating between the same temperature levels is the same.
- 6.39 Show that the efficiency of a reversible engine is independent of the nature or amount of the working substance going through the cycle.
- 6.40 How does the efficiency of a reversible cycle depend only on the two temperatures at which heat is transferred?



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- 6.5 A household refrigerator is maintained at a temperature of  $2^{\circ}\text{C}$ . Every time the door is opened, warm material is placed inside, introducing an average of 420 kJ, but making only a small change in the temperature of the refrigerator. The door is opened 20 times a day, and the refrigerator operates at 15% of the ideal COP. The cost of work is Rs. 2.50 per kWh. What is the monthly bill for this refrigerator? The atmosphere is at  $30^{\circ}\text{C}$ . Ans. Rs. 118.80
- 6.6 A heat pump working on the Carnot cycle takes in heat from a reservoir at  $5^{\circ}\text{C}$  and delivers heat to a reservoir at  $60^{\circ}\text{C}$ . The heat pump is driven by a reversible heat engine which takes in heat from a reservoir at  $840^{\circ}\text{C}$  and rejects heat to a reservoir at  $60^{\circ}\text{C}$ . The reversible heat engine also drives a machine that absorbs 30 kW. If the heat pump extracts 17 kJ/s from the  $5^{\circ}\text{C}$  reservoir, determine (a) the rate of heat supply from the  $840^{\circ}\text{C}$  source, and (b) the rate of heat rejection to the  $60^{\circ}\text{C}$  sink. Ans. (a) 47.61 kW; (b) 34.61 kW
- 6.7 A refrigeration plant for a food store operates as a reversed Carnot heat engine cycle. The store is to be maintained at a temperature of  $-5^{\circ}\text{C}$  and the heat transfer from the store to the cycle is at the rate of 5 kW. If heat is transferred from the cycle to the atmosphere at a temperature of  $25^{\circ}\text{C}$ , calculate the power required to drive the plant. Ans. 0.56 kW
- 6.8 A heat engine is used to drive a heat pump. The heat transfers from the heat engine and from the heat pump are used to heat the water circulating through the radiators of a building. The efficiency of the heat engine is 27% and the COP of the heat pump is 4. Evaluate the ratio of the heat transfer to the circulating water to the heat transfer to the heat engine. Ans. 1.81
- 6.9 If 20 kJ are added to a Carnot cycle at a temperature of  $100^{\circ}\text{C}$  and 14.6 kJ are rejected at  $0^{\circ}\text{C}$ , determine the location of absolute zero on the Celsius scale. Ans.  $-270.37^{\circ}\text{C}$
- 6.10 Two reversible heat engines *A* and *B* are arranged in series, *A* rejecting heat directly to *B*. Engine *A* receives 200 kJ at a temperature of  $421^{\circ}\text{C}$  from a hot source, while engine *B* is in communication with a cold sink at a temperature of  $4.4^{\circ}\text{C}$ . If the work output of *A* is twice that of *B*, find (a) the intermediate temperature between *A* and *B*, (b) the efficiency of each engine, and (c) the heat rejected to the cold sink. Ans.  $143.4^{\circ}\text{C}$ , 40% and 33.5%, 80 kJ
- 6.11 A heat engine operates between the maximum and minimum temperatures of  $671^{\circ}\text{C}$  and  $60^{\circ}\text{C}$  respectively, with an efficiency of 50% of the appropriate Carnot efficiency. It drives a heat pump which uses river water at  $4.4^{\circ}\text{C}$  to heat a block of flats in which the temperature is to be maintained at  $21.1^{\circ}\text{C}$ . Assuming that a temperature difference of  $11.1^{\circ}\text{C}$  exists between the working fluid and the river water, on the one hand, and the required room temperature on the other, and assuming the heat pump to operate on the reversed Carnot cycle, but with a COP of 50% of the ideal COP, find the heat input to the engine per unit heat output from the heat pump. Why is direct heating thermodynamically more wasteful? Ans. 0.79 kJ/kJ heat input
- 6.12 An ice-making plant produces ice at atmospheric pressure and at  $0^{\circ}\text{C}$  from water. The mean temperature of the cooling water circulating through the condenser of the refrigerating machine is  $18^{\circ}\text{C}$ . Evaluate the minimum electrical work in kWh required to produce 1 tonne of ice. (The enthalpy of fusion of ice at atmospheric pressure is 333.5 kJ/kg). Ans. 6.11 kWh

- 6.13 A reversible engine works between three thermal reservoirs, *A*, *B* and *C*. The engine absorbs an equal amount of heat from the thermal reservoirs *A* and *B* kept at temperatures  $T_A$  and  $T_B$  respectively, and rejects heat to the thermal reservoir *C* kept at temperature  $T_C$ . The efficiency of the engine is  $\alpha$  times the efficiency of the reversible engine, which works between the two reservoirs *A* and *C*. Prove that

$$\frac{T_A}{T_B} = (2\alpha - 1) + 2(1 - \alpha) \frac{T_A}{T_C}$$

- 6.14 A reversible engine operates between temperatures  $T_1$  and  $T$  ( $T_1 > T$ ). The energy rejected from this engine is received by a second reversible engine at the same temperature  $T$ . The second engine rejects energy at temperature  $T_2$  ( $T_2 < T_1$ ). Show that (a) temperature  $T$  is the arithmetic mean of temperatures  $T_1$  and  $T_2$  if the engines produce the same amount of work output, and (b) temperature  $T$  is the geometric mean of temperatures  $T_1$  and  $T_2$  if the engines have the same cycle efficiencies.

- 6.15 Two Carnot engines *A* and *B* are connected in series between two thermal reservoirs maintained at 1000 K and 100 K respectively. Engine *A* receives 1680 kJ of heat from the high-temperature reservoir and rejects heat to the Carnot engine *B*. Engine *B* takes in heat rejected by engine *A* and rejects heat to the low-temperature reservoir. If engines *A* and *B* have equal thermal efficiencies, determine (a) the heat rejected by engine *B*, (b) the temperature at which heat is rejected by engine *A*, and (c) the work done during the process by engines, *A* and *B* respectively. If engines *A* and *B* deliver equal work, determine (d) the amount of heat taken in by engine *B*, and (e) the efficiencies of engines *A* and *B*.

Ans. (a) 168 kJ, b) 316.2 K, (c) 1148.7, 363.3 kJ, (d) 924 kJ, (e) 45%, 81.8%.

- 6.16 A heat pump is to be used to heat a house in winter and then reversed to cool the house in summer. The interior temperature is to be maintained at 20°C. Heat transfer through the walls and roof is estimated to be 0.525 kJ/s per degree temperature difference between the inside and outside. (a) If the outside temperature in winter is 5°C, what is the minimum power required to drive the heat pump? (b) If the power output is the same as in part (a), what is the maximum outer temperature for which the inside can be maintained at 20°C?

Ans. (a) 403 W, (b) 35.4°C.

- 6.17 Consider an engine in outer space which operates on the Carnot cycle. The only way in which heat can be transferred from the engine is by radiation. The rate at which heat is radiated is proportional to the fourth power of the absolute temperature and to the area of the radiating surface. Show that for a given power output and a given  $T_1$ , the area of the radiator will be a minimum when

$$\frac{T_2}{T_1} = \frac{3}{4}$$

- 6.18 It takes 10 kW to keep the interior of a certain house at 20°C when the outside temperature is 0°C. This heat flow is usually obtained directly by burning gas or oil. Calculate the power required if the 10 kW heat flow were supplied by operating a reversible engine with the house as the upper reservoir and the outside surroundings as the lower reservoir, so that the power were used only to perform work needed to operate the engine.

Ans. 0.683 kW

- 6.19 Prove that the COP of a reversible refrigerator operating between two given temperatures is the maximum.
- 6.20 A house is to be maintained at a temperature of  $20^\circ\text{C}$  by means of a heat pump pumping heat from the atmosphere. Heat losses through the walls of the house are estimated at  $0.65 \text{ kW}$  per unit of temperature difference between the inside of the house and the atmosphere. (a) If the atmospheric temperature is  $-10^\circ\text{C}$ , what is the minimum power required to drive the pump? (b) It is proposed to use the same heat pump to cool the house in summer. For the same room temperature, the same heat loss rate, and the same power input to the pump, what is the maximum permissible atmospheric temperature? Ans.  $2 \text{ kW}, 50^\circ\text{C}$ .
- 6.21 A solar-powered heat pump receives heat from a solar collector at  $T_h$ , rejects heat to the atmosphere at  $T_a$ , and pumps heat from a cold space at  $T_c$ . The three heat transfer rates are  $Q_h$ ,  $Q_a$ , and  $Q_c$  respectively. Derive an expression for the minimum ratio  $Q_h/Q_c$ , in terms of the three temperatures.  
If  $T_h = 400 \text{ K}$ ,  $T_a = 300 \text{ K}$ ,  $T_c = 200 \text{ K}$ ,  $Q_c = 12 \text{ kW}$ , what is the minimum  $Q_h$ ? If the collector captures  $0.2 \text{ kW/m}^2$ , what is the minimum collector area required?  
Ans.  $26.25 \text{ kW}, 131.25 \text{ m}^2$
- 6.22 A heat engine operating between two reservoirs at  $1000 \text{ K}$  and  $300 \text{ K}$  is used to drive a heat pump which extracts heat from the reservoir at  $300 \text{ K}$  at a rate twice that at which the engine rejects heat to it. If the efficiency of the engine is 40% of the maximum possible and the COP of the heat pump is 50% of the maximum possible, what is the temperature of the reservoir to which the heat pump rejects heat? What is the rate of heat rejection from the heat pump if the rate of heat supply to the engine is  $50 \text{ kW}$ ? Ans.  $326.5 \text{ K}, 86 \text{ kW}$
- 6.23 A reversible power cycle is used to drive a reversible heat pump cycle. The power cycle takes in  $Q_1$  heat units at  $T_1$  and rejects  $Q_2$  at  $T_2$ . The heat pump abstracts  $Q_4$  from the sink at  $T_4$  and discharges  $Q_3$  at  $T_3$ . Develop an expression for the ratio  $Q_4/Q_1$  in terms of the four temperatures.  
Ans.  $\frac{Q_4}{Q_1} = \frac{T_4(T_1 - T_2)}{T_1(T_3 - T_4)}$
- 6.24 Prove that the following propositions are logically equivalent: (a) A PMM2 is impossible, (b) A weight sliding at constant velocity down a frictional inclined plane executes an irreversible process.
- 6.25 A heat engine receives half of its heat supply at  $1000 \text{ K}$  and half at  $500 \text{ K}$  while rejecting heat to a sink at  $300 \text{ K}$ . What is the maximum thermal efficiency of the heat engine? Ans. 55%
- 6.26 A heat pump provides  $3 \times 10^4 \text{ kJ/h}$  to maintain a dwelling at  $23^\circ\text{C}$  on a day when the outside temperature is  $0^\circ\text{C}$ . The power input to the heat pump is  $4 \text{ kW}$ . Determine the COP of the heat pump and compare it with the COP of a reversible heat pump operating between the reservoirs at the same two temperatures.  
Ans. 2.08, 12.87
- 6.27 When the outside temperature is  $-10^\circ\text{C}$ , a residential heat pump must provide  $3.5 \times 10^6 \text{ kJ}$  per day to a dwelling to maintain its temperature at  $20^\circ\text{C}$ . If the electricity costs  $\text{Rs } 2.10$  per kWh, find the minimum operating cost for each day of operation. Ans.  $\text{Rs } 208.83$

# 7

## CHAPTER

# Entropy

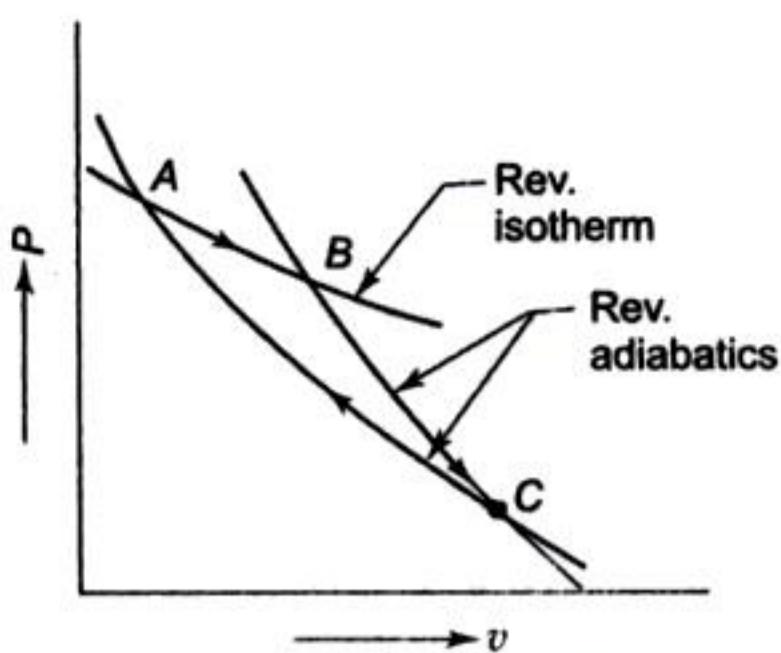
### 7.1 INTRODUCTION

The first law of thermodynamics was stated in terms of cycles first and it was shown that the cyclic integral of heat is equal to the cyclic integral of work. When the first law was applied for thermodynamic processes, the existence of a property, the internal energy, was found. Similarly, the second law was also first stated in terms of cycles executed by systems. When applied to processes, the second law also leads to the definition of a new property, known as entropy. If the first law is said to be the law of internal energy, then second law may be stated to be the law of entropy. In fact, *thermodynamics is the study of three E's, namely, energy, equilibrium and entropy.*

### 7.2 TWO REVERSIBLE ADIABATIC PATHS CANNOT INTERSECT EACH OTHER

Let it be assumed that two reversible adiabatics  $AC$  and  $BC$  intersect each other at point  $C$  (Fig. 7.1). Let a reversible isotherm  $AB$  be drawn in such a way that it intersects the reversible adiabatics at  $A$  and  $B$ . The three reversible processes  $AB$ ,  $BC$ , and  $CA$  together constitute a reversible cycle, and the area included represents the net work output in a cycle. But such a cycle is impossible, since net work is being produced in a cycle by a heat engine by exchanging heat with a single reservoir in the process  $AB$ , which violates the Kelvin-Planck statement of the second law. Therefore, the assumption of the intersection of the reversible adiabatics is wrong. *Through one point, there can pass only one reversible adiabatic.*

Since two constant property lines can never intersect each other, it is inferred that a reversible adiabatic path must represent some property, which is yet to be identified.



**Fig. 7.1 Assumption of Two Reversible Adiabatics Intersecting Each Other**

### 7.3 CLAUSIUS' THEOREM

Let a system be taken from an equilibrium state  $i$  to another equilibrium state  $f$  by following the reversible path  $i-f$  (Fig. 7.2). Let a reversible adiabatic  $i-a$  be drawn through  $i$  and another reversible adiabatic  $b-f$  be drawn through  $f$ . Then a reversible isotherm  $a-b$  is drawn in such a way that the area under  $i-a-b-f$  is equal to the area under  $i-f$ . Applying the first law for

Process  $i-f$

$$Q_{i-f} = U_f - U_i + W_{if} \quad (7.1)$$

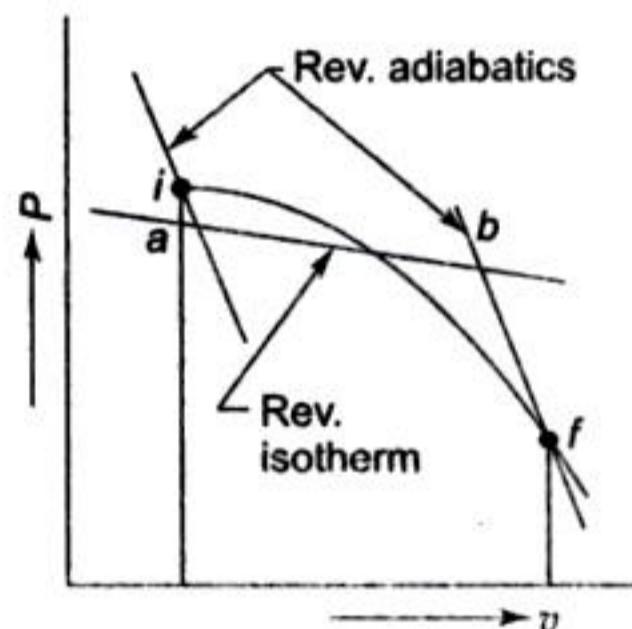
Process  $i-a-b-f$

$$Q_{iabf} = U_f - U_i + W_{iabf} \quad (7.2)$$

Since

$$W_{if} = W_{iabf}$$

∴ From Eqs. (7.1) and (7.2)



**Fig. 7.2 Reversible Path Substituted by Two Reversible Adiabatics and a Reversible Isotherm**

$$\begin{aligned} Q_{if} &= Q_{iabf} \\ &= Q_{ia} + Q_{ab} + Q_{bf} \end{aligned}$$

Since

$$Q_{ia} = 0 \text{ and } Q_{bf} = 0$$

$$Q_{if} = Q_{ab}$$

Heat transferred in the process  $i-f$  is equal to the heat transferred in the isothermal process  $a-b$ .

Thus any reversible path may be substituted by a reversible zigzag path, between the same end states, consisting of a reversible adiabatic followed by a reversible isotherm and then by a reversible adiabatic, such that the heat transferred during the isothermal process is the same as that transferred during the original process.



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The cyclic integral of  $dQ/T$  for a reversible cycle is equal to zero. This is known as *Clausius' theorem*. The letter  $R$  emphasizes the fact that the equation is valid only for a reversible cycle.

## 7.4 THE PROPERTY OF ENTROPY

Let a system be taken from an initial equilibrium state  $i$  to a final equilibrium state  $f$  by following the reversible path  $R_1$  (Fig. 7.4). The system is brought back from  $f$  to  $i$  by following another reversible path  $R_2$ . Then the two paths  $R_1$  and  $R_2$  together constitute a reversible cycle. From Clausius' theorem

$$\oint_{R_1 R_2} \frac{dQ}{T} = 0$$

The above integral may be replaced as the sum of two integrals, one for path  $R_1$  and the other for path  $R_2$

$$\int_{R_1}^f \frac{dQ}{T} + \int_{R_2}^i \frac{dQ}{T} = 0$$

or  $\int_{R_1}^f \frac{dQ}{T} = - \int_{R_2}^i \frac{dQ}{T}$

Since  $R_2$  is a reversible path

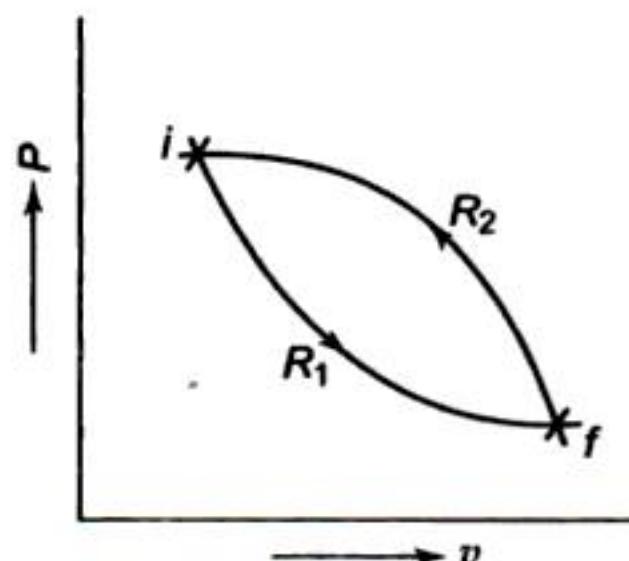


Fig. 7.4

Two Reversible Paths  $R_1$  and  $R_2$  Between Two Equilibrium States  $i$  and  $f$

$$\int_{R_1}^f \frac{dQ}{T} = \int_{R_2}^i \frac{dQ}{T}$$

Since  $R_1$  and  $R_2$  represent any two reversible paths,  $\int_R^f \frac{dQ}{T}$  is independent of the reversible path connecting  $i$  and  $f$ . Therefore, there exists a property of a system whose value at the final state  $f$  minus its value at the initial state  $i$  is equal to  $\int_R^f \frac{dQ}{T}$ . This property is called entropy, and is denoted by  $S$ . If  $S_i$  is the entropy at the initial state  $i$ , and  $S_f$  is the entropy at the final state  $f$ , then

$$\int_R^f \frac{dQ}{T} = S_f - S_i \quad (7.4)$$

When the two equilibrium states are infinitesimally near

$$\frac{dQ_R}{T} = dS \quad (7.5)$$

where  $dS$  is an exact differential because  $S$  is a point function and a property. The subscript  $R$  in  $dQ$  indicates that heat  $dQ$  is transferred reversibly.



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For reversible isothermal heat transfer (Fig. 7.7),  $T = \text{constant}$ .

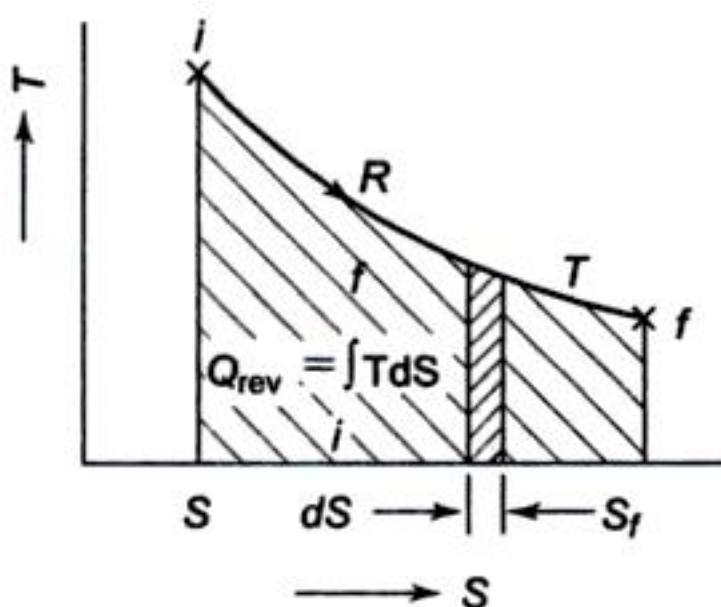


Fig. 7.6 Area Under a Reversible Path on the T-s Plot Represents Heat Transfer

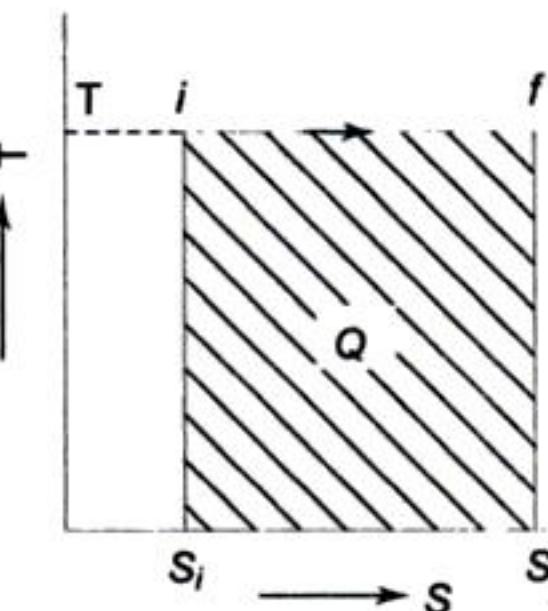


Fig. 7.7 Reversible Isothermal Heat Transfer

$$\therefore Q_{\text{rev}} = T \int_i^f dS = T(S_f - S_i)$$

For a reversible adiabatic process,  $dS = 0$ ,  $S = C$  (Fig. 7.8).

The *Carnot cycle* comprising two reversible isotherms and two reversible adiabatics forms a rectangle in the T-S plane (Fig. 7.9). Process 4–1 represents reversible isothermal heat addition  $Q_1$  to the system at  $T_1$  from an external source, process 1–2 is the reversible adiabatic expansion of the system producing  $W_E$  amount of work, process 2–3 is the reversible isothermal heat rejection from the system to an external sink at  $T_2$ , and process 3–4 represents reversible adiabatic compression of the system consuming  $W_c$  amount of work. Area 1234 represents the net work output per cycle and the area under 4–1 indicates the quantity of heat added to the system  $Q_1$ .

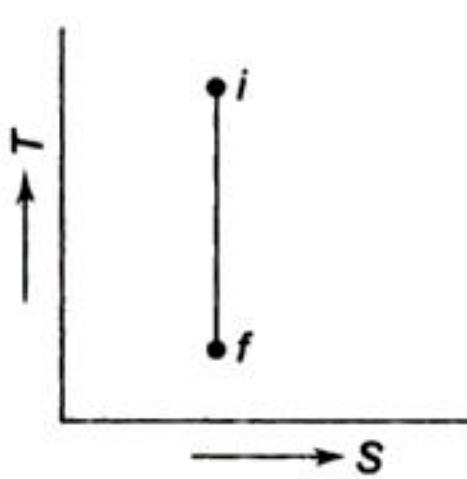


Fig. 7.8 Reversible Adiabatic is Isentropic

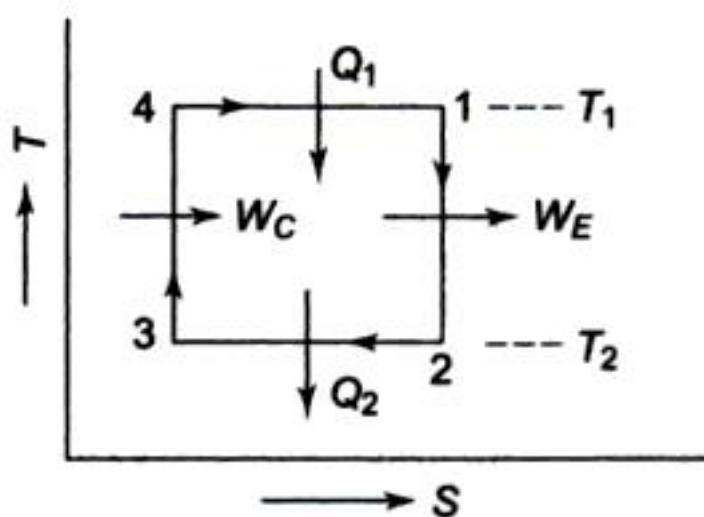


Fig. 7.9 Carnot Cycle

$$\begin{aligned} \therefore \eta_{\text{Carnot}} &= \frac{Q_1 - Q_2}{Q_1} = \frac{T_1(S_1 - S_4) - T_2(S_2 - S_3)}{T_1(S_1 - S_4)} \\ &= \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1} \end{aligned}$$

and

$$W_{\text{net}} = Q_1 - Q_2 = (T_1 - T_2)(S_1 - S_4)$$



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## 7.8 ENTROPY PRINCIPLE

For any infinitesimal process undergone by a system, we have from Eq. (7.10) for the total mass

$$dS \geq \frac{dQ}{T}$$

For an isolated system which does not undergo any energy interaction with the surroundings  $dQ = 0$ .

Therefore, for an isolated system

$$dS_{\text{iso}} \geq 0 \quad (7.17)$$

For a reversible process,

$$dS_{\text{iso}} = 0$$

or

$$S = \text{constant}$$

For an irreversible process

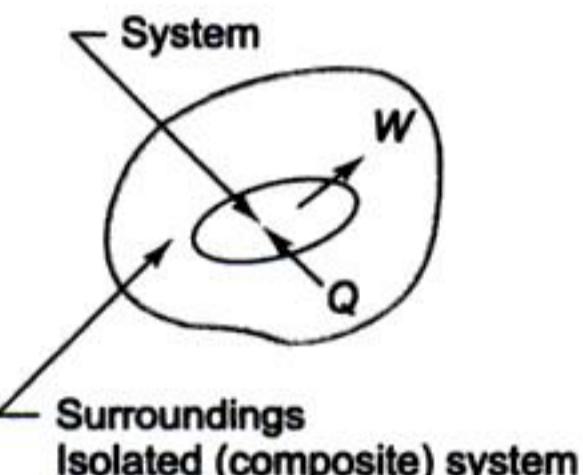
$$dS_{\text{iso}} > 0$$

It is thus proved that *the entropy of an isolated system can never decrease*. It always increases and remains constant only when the process is reversible. This is known as the *principle of increase of entropy*, or simply the *entropy principle*. It is the quantitative general statement of second law from the macroscopic viewpoint.

An isolated system can always be formed by including any system and its surroundings within a single boundary (Fig. 7.12). Sometimes the original system which is then only a part of the isolated system is called a 'subsystem'.

The system and the surroundings together (the universe or the isolated system) include everything which is affected by the process. For all possible processes that a system in the given surroundings can undergo

$$dS_{\text{univ}} \geq 0$$



or  $dS_{\text{sys}} + dS_{\text{surr}} \geq 0$

(7.18)

**Fig. 7.12 Isolated System**

Entropy may decrease locally at some region within the isolated system, but it must be compensated by a greater increase of entropy somewhere within the system so that the net effect of an irreversible process is an entropy increase of the whole system. The entropy increase of an isolated system is a measure of the extent of irreversibility of the process undergone by the system.



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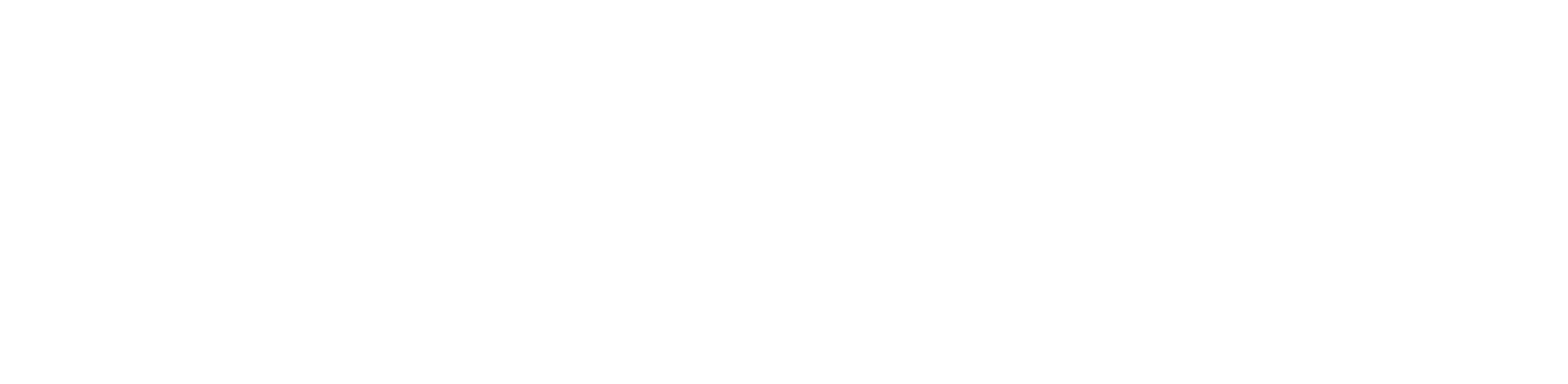
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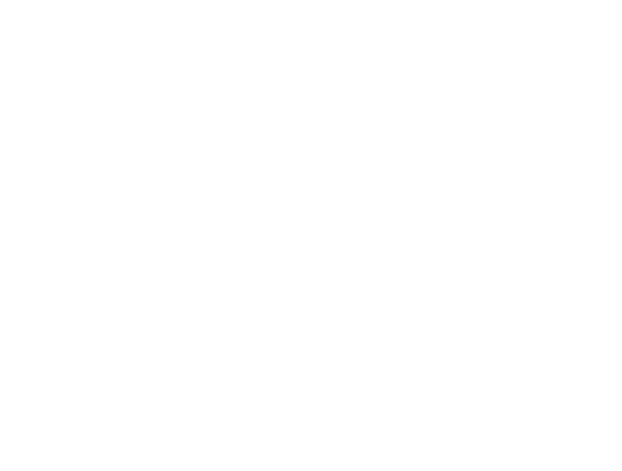
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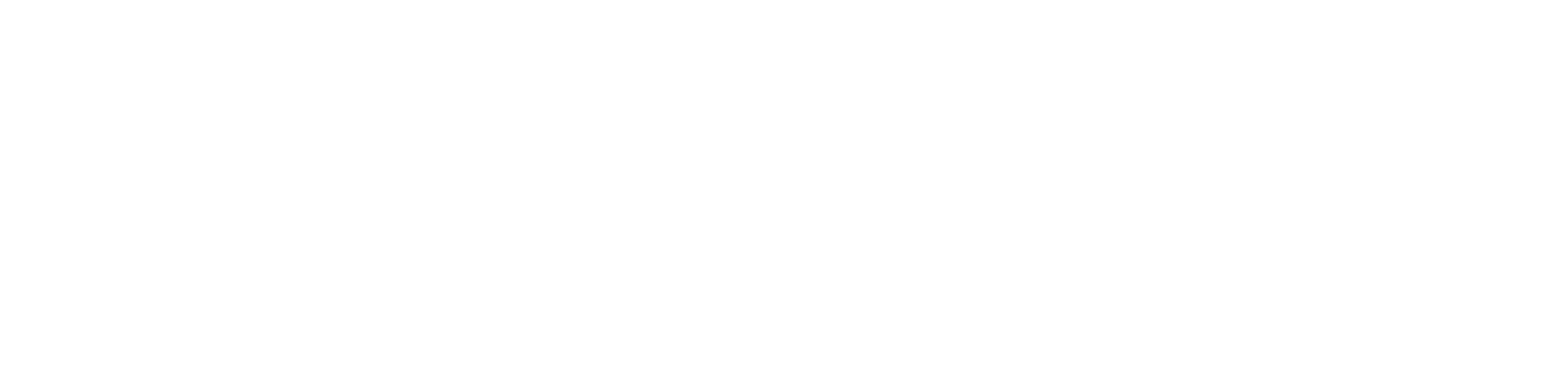
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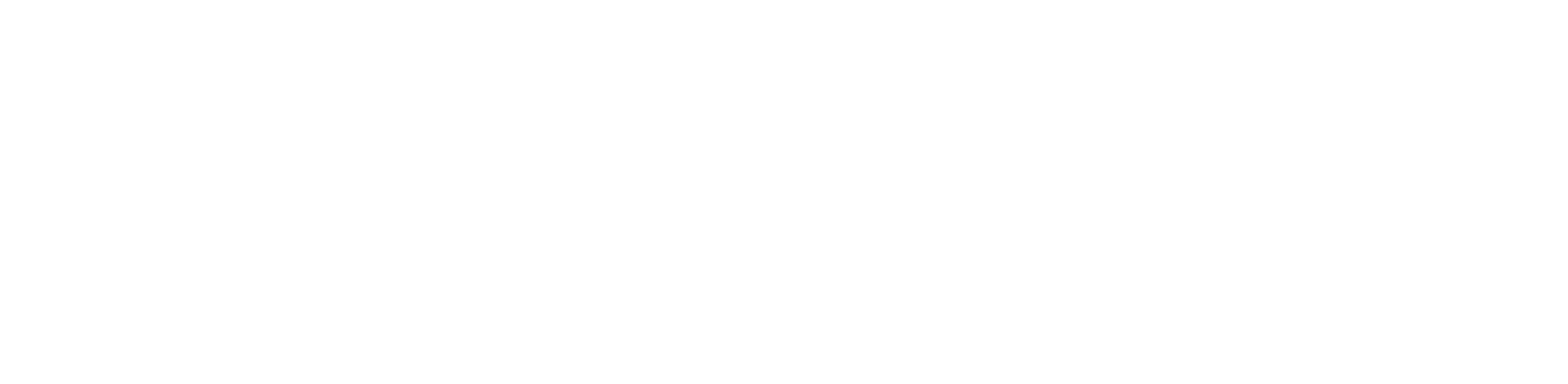
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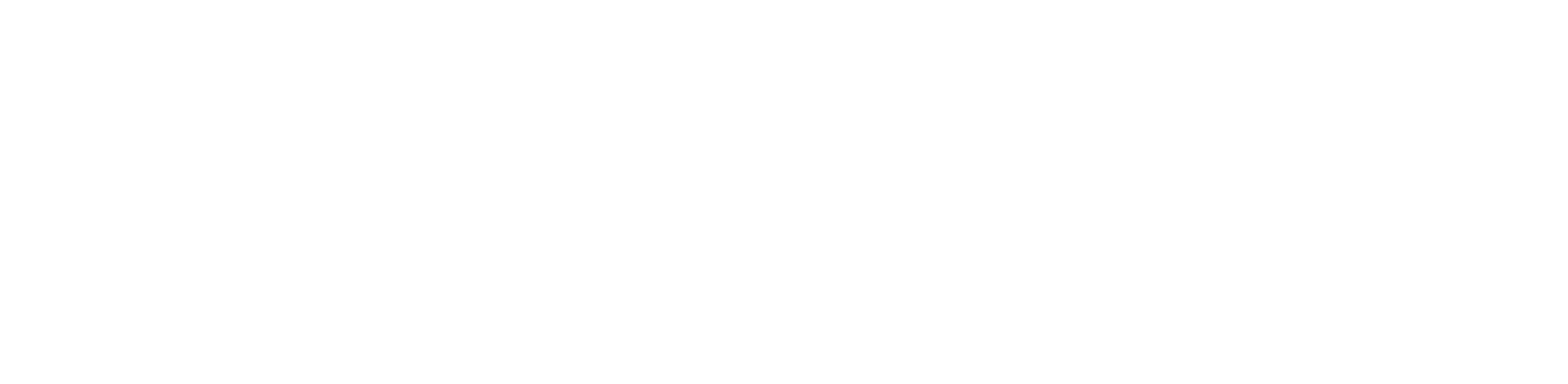
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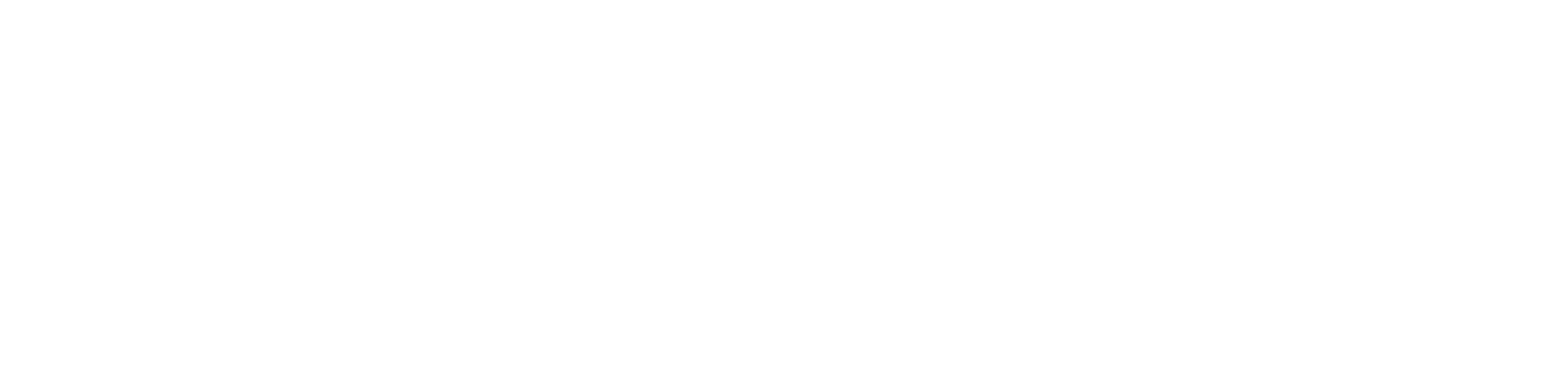
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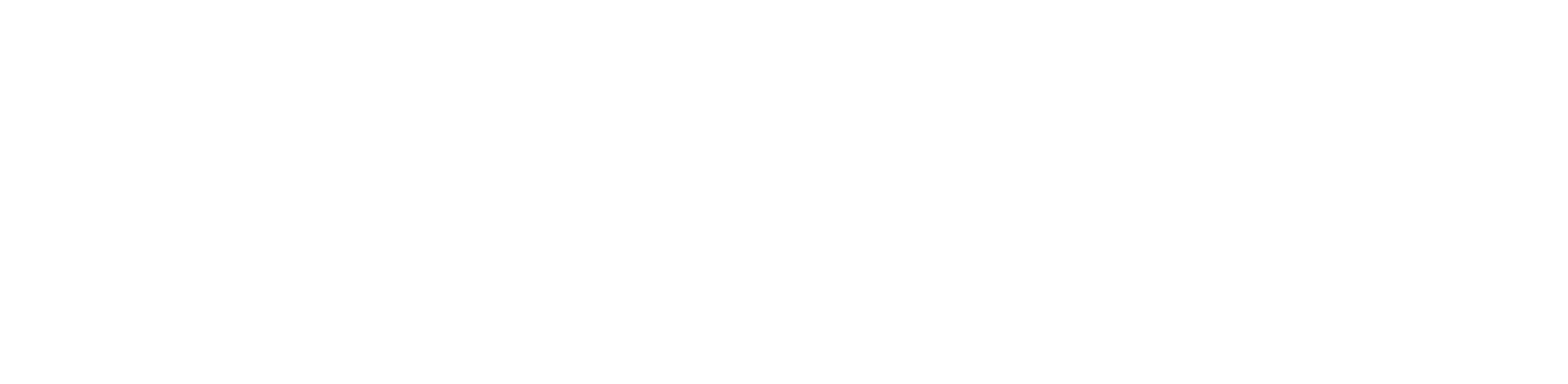
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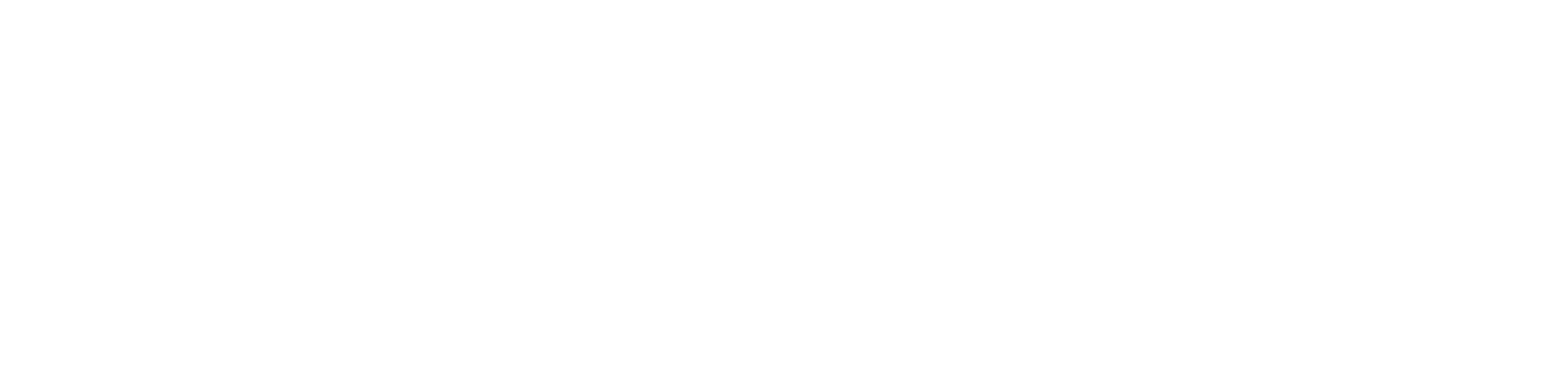
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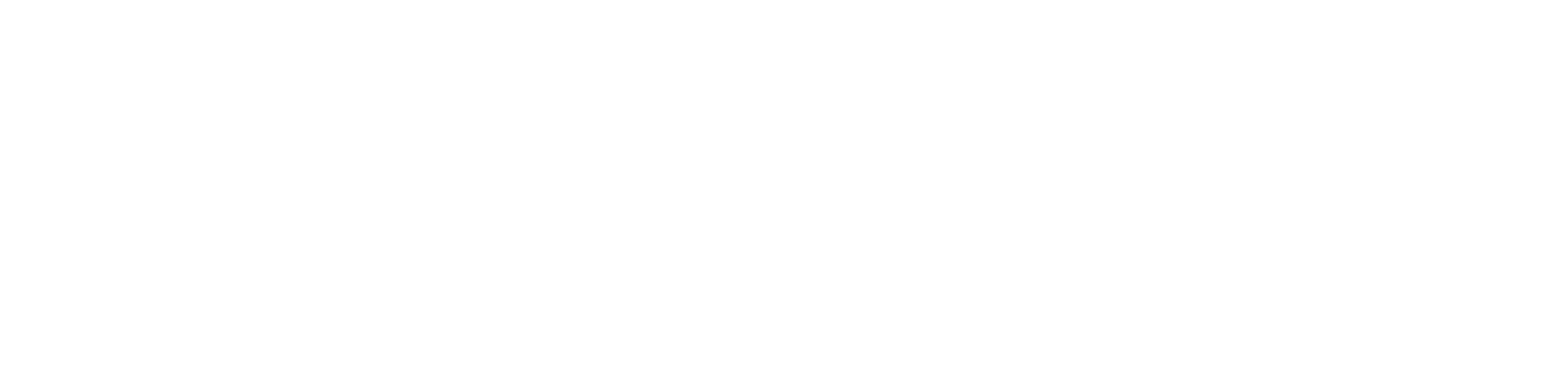
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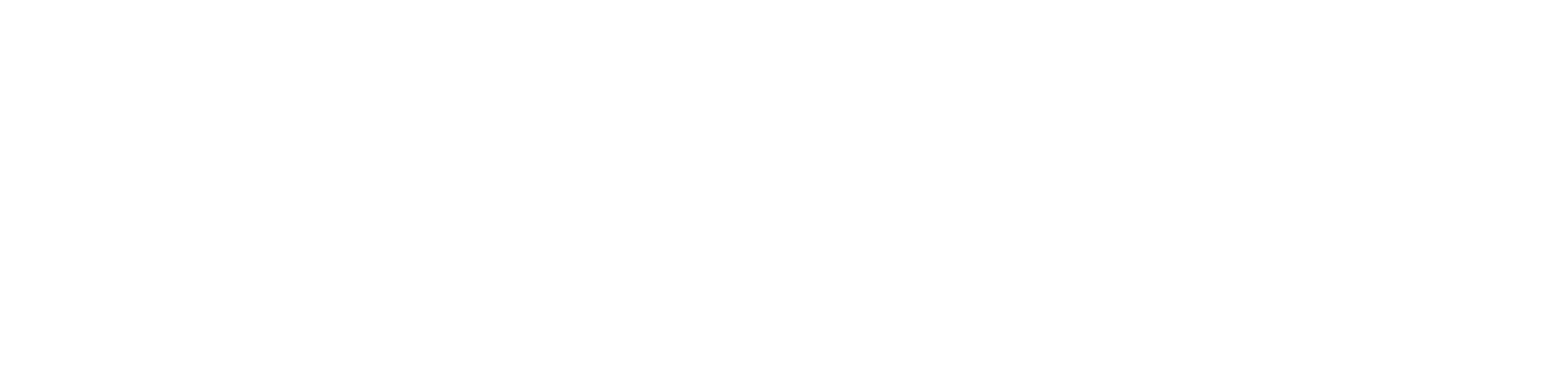
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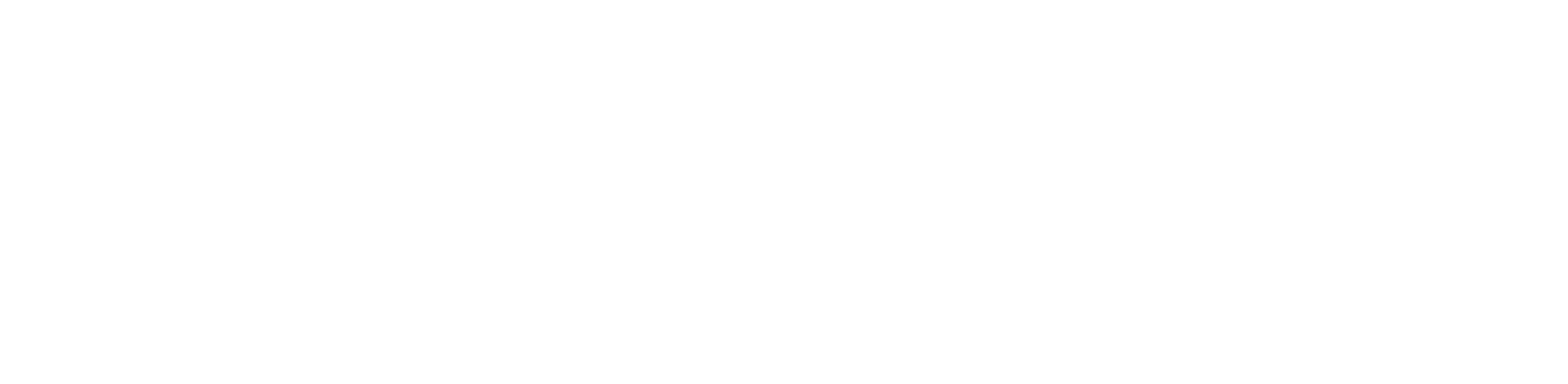
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