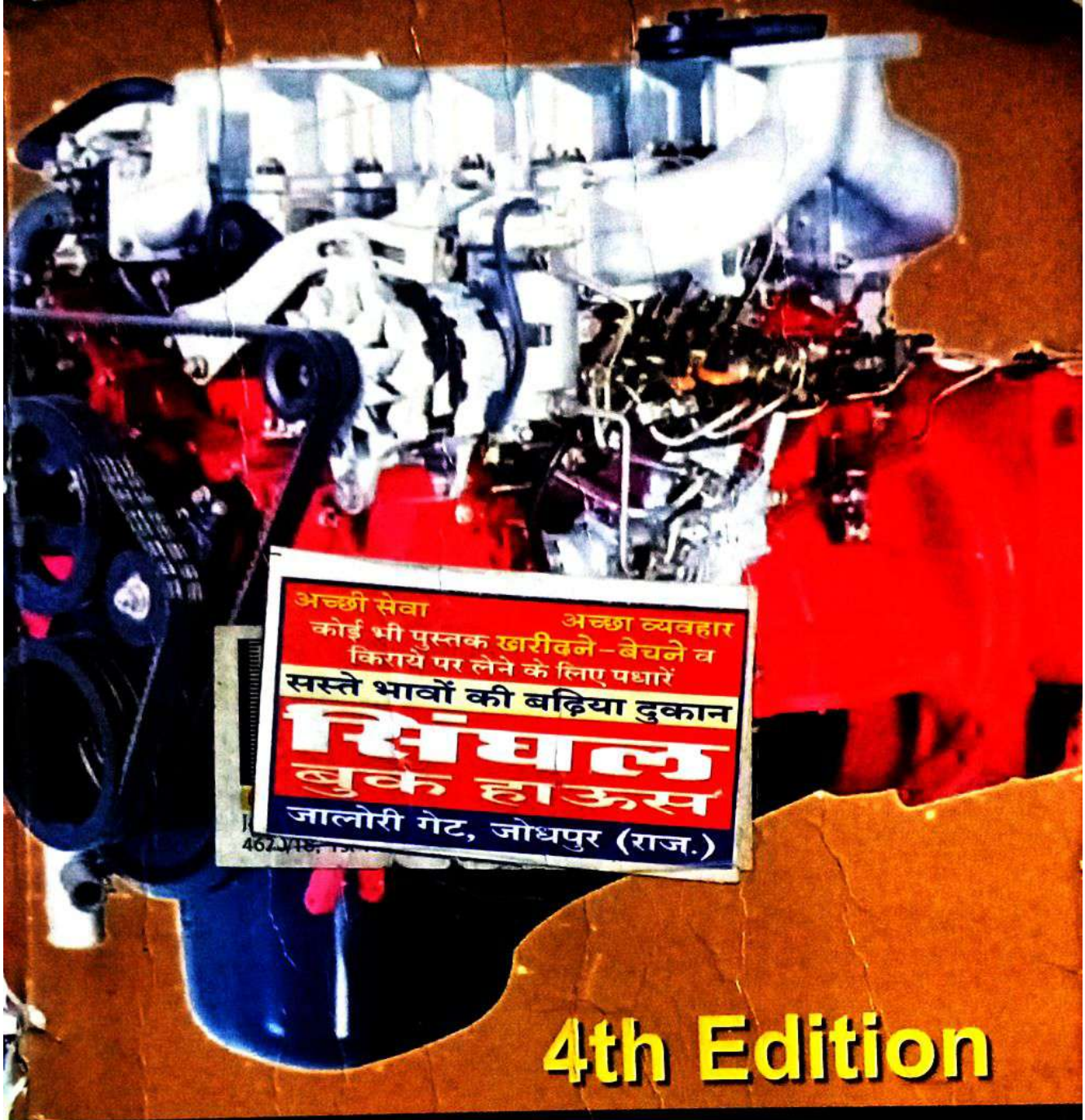

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Ph.D.,
M.ASME, M. ASCE, M. IASH, LM. TSI
Former Associate Professor
Department of Mechanical Engineering
M.B.M. Engineering College
JNV University
Jodhpur - 342 001 (India)

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Kapil Gupta

1

Basic Concepts Related to Thermodynamics

1.1. INTRODUCTION

Thermodynamics, like other physical sciences, is based on observations of nature. It is a discipline that involves the formulation of a number of concepts derived from the common experience.

The subject of thermodynamics, is of interest to students of all branches of engineering, in fact to all mankind, as it deals with the most basic processes occuring in nature.

The first law of thermodynamics, is simply an expression of the conservation of energy principle, and it asserts that energy is a thermodynamic property. The second law of thermodynamics asserts that energy has quality as well as quantity, and actual processes occur in the direction of decreasing quality of energy.

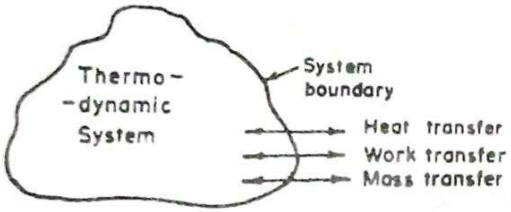
The first and second laws of thermodynamics emerged simultaneously in the 1850s primarily out of the works of William Rankine, Rudolph Clausius, and Lord Kelvin (formerly William Thomson). The term *thermodynamics* was first used in publication of Lord Kelvin in 1849. The first thermodynamic text book was written in 1859 by William Rankine a professor at the University of Glasgow.

1.2. BASIC DEFINITIONS

The study of any science begins with definitions of words used to state the fundamental principles. In this chapter we build a vocabulary for thermodynamics wherein each new terms already defined or terms that we accept verbally undefined. In this book, terms that we arbitrarily accept as verbally undefined are time, length, temperature, mass and force.

1.2.1. System, boundary and surroundings

A thermodynamic system, or simply a system, is defined as quantity of matter or a region in space chosen for study. The quant of matter or region of space must be within a prescribed bounda This boundary may be deformable or rigid, it may even be imagina Everything outside a system boundary is referred to as t surroundings (Fig. 1.1). Usually, the term surroundings is restrict to those things outside the system that in some way influence t system.



Possible interactions with Surroundings.

Fig. 1.1. System, boundary and surrounding

1.2.2 Closed system

Any system in which mass does not cross the system boundar but energy may cross the system boundary (Fig. 1.2). Closed system also known as control mass.

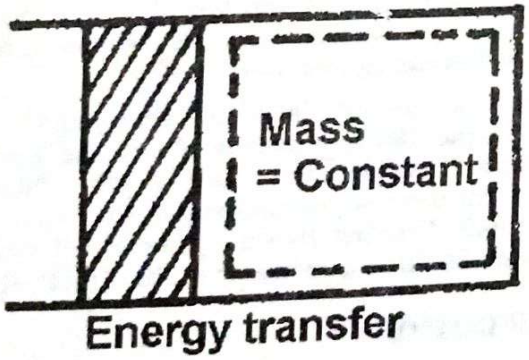


Fig. 1.2. Closed system

1.2.3. Open System

Any system in which both mass and energy may cross th system boundary (Fig. 1.3).

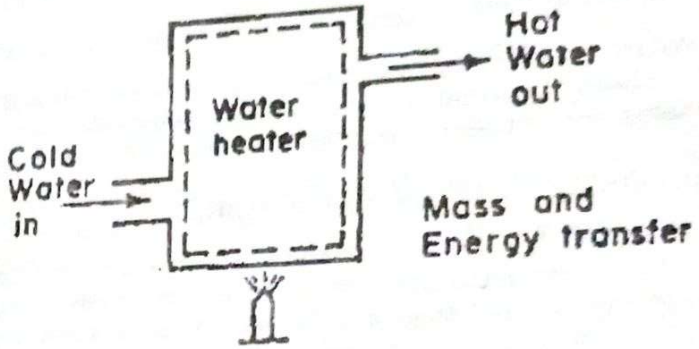


Fig. 1.3. Open system

.2.4. Control Volume and Control Surface

Some authors call an open system a control volume and its oundary a control surface. These two terms are completely nterchangeable with open system and boundary.

.2.5. Non-flow and flow processes

The processes undergone by the fluid in a closed system are escribed as non-flow processes, whereas those undergone by the fluid n an open system are referred to as flow processes.

.2.6. Isolated system

Any system in which neither mass nor energy crosses the system oundary (Fig. 1.4).

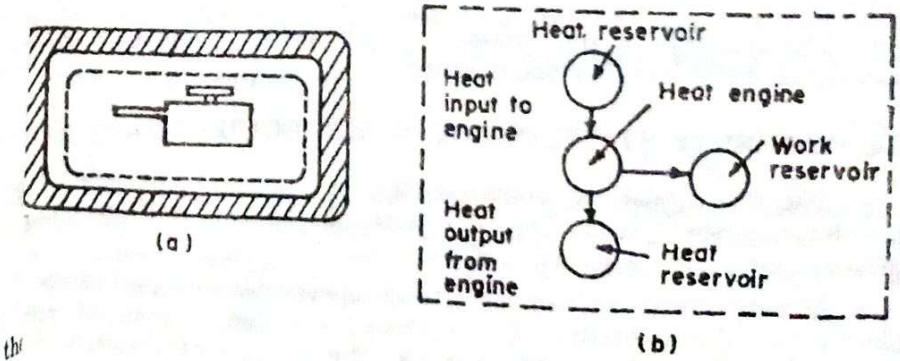


Fig. 1.4. Isolated system

1.2.7. Adiabatic system

A system which is thermally insulated from its surrounding called an adiabatic system. It can, however, exchange work wit surroundings. If it does not, it becomes an isolated system.

1.3. MACROSCOPIC AND MICROSCOPIC

Systems can be studied from macroscopic or a microscopic of view. The macroscopic approach to thermodynamics is conce with the gross or overall behaviour. This is sometimes called clas thermodynamics.

The microscopic approach to thermodynamics, known statistical thermodynamics is concerned directly with the structu matter. At the present state of development, statis thermodynamics is not as general and fruitful in its applicability classical thermodynamics.

The vast mazority of engineering problems can be solved continuum concepts. There are two other technical terms that express these ideas :

1.3.1. Macroscopic system analysis

The analysis of the systems at the continuum level molecular dimensions and time scales do not enter into the analy This is the domain of classical and nonequilibrium thermodynamic

1.3.2. Microscopic system analysis

The analysis of the systems at the atomic level. This is domain of statistical thermodynamics.

1.4. PROPERTY, STATE, PROCESS, AND EQUILIBRIUM

The distinguishing characteristics of a system are called properties of the system. They are quantities that we must speci give a macroscopic description of a system.

A property is either a directly observable or an indire observable characteristic of a system. Any combination of characteristics, such as, for example, the product of pressure volume, is also a property. That is, we can obtain new propertie defining them in terms of other properties.

Thermodynamic properties can be placed in two general classes : ntensive and extensive.

.4.1. Intensive and extensive properties

Intensive Properties: Any thermodynamic property of a ystem that is independent of the system mass. Examples are ressure, temperature, density and velocity. They are not additive.

Extensive Properties: Any thermodynamic property of a ystem that depends on the mass of the system. Examples are the ystem mass itself, volume, and total energy. They are additive.

Most extensive properties can be converted into intensive roperties by dividing the extensive property by the system mass or he number of moles in the system. Intensive properties created in his way are called specific properties. For example, the total volume f a system divided by the total mass of the system is the intensive roperty called specific volume, and the total volume divided by the otal number of moles of the system is the intensive property called nolar specific volume.

Extensive properties are symbolized by uppercase (capital) etters. Intensive properties are symbolized by lowercase letter. For xample specific volume is given as

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

where v is specific volume, V is volume and m is mass of the system.

Density is defined as the mass of a substance divided by its olume, or the mass per unit volume. Thus $\rho = \frac{1}{v}$ and it is an ntensive property.

An easy way to determine whether a property is intensive or xtensive is to divide the system into three parts with a partition, as hown in Fig. 1.5. Each part will have the same value of intensive roperties as the original system, but one third the value of the xtensive properties.

.4.2. State

In any pure working fluid, only two independent properties are ecessary to define completely the state of the fluid. Since any two ndependent properties suffice to define the state of a system, it is ossible to represent the state of a system by a point situated on a iagram of properties (Fig. 1.6).

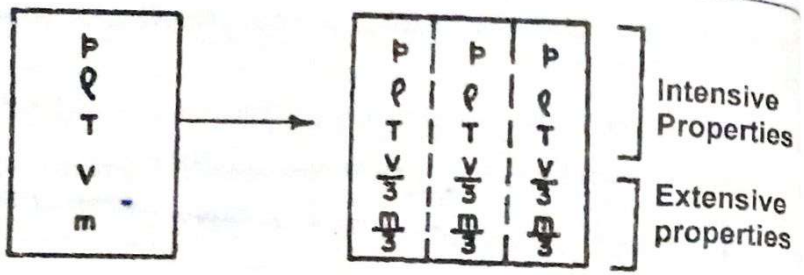


Fig. 1.5. Criteria to differentiate intensive and extensive properties

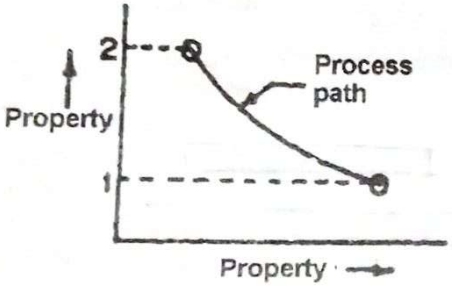


Fig. 1.6. A Process between state 1 and state 2 and the process path.

1.4.3. Equilibrium

Thermodynamics deals with equilibrium states. The equilibrium implies a state of balance. In an equilibrium state there are no unbalanced potentials (or driving forces) within the system other words : A system is said to be in thermodynamic equilibrium does not have the capacity to spontaneously change its state after has been isolated.

1.4.4. Thermodynamic process

Any change that a system undergoes from one equilibrium st to another is called a process, and the series of states through which system passes during a process is called the path of the process. describe a process completely, one should specify the initial and fi states of the process, as well as the path it follows, and the in actions with the surroundings (Fig 1.7).

1.4.5. Quasi-static process

When a process proceeds in such a manner that the syst remain infinitesimally close to an equilibrium state at all times, it called a quasi-static, or quasi-equilibrium process. A quasi-sta

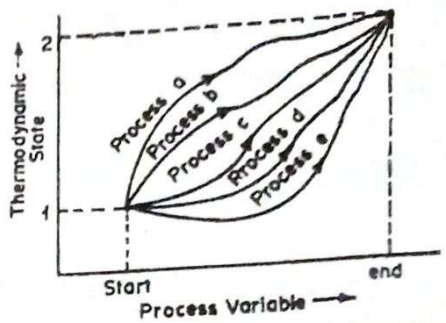


Fig. 1.7. Five different process paths that change the state of the system from 1 to 2

process can be viewed as a sufficiently slow process which allows the system to adjust itself internally so that properties in one part of the system do not change any faster than those at other parts (Fig 1.8). For a quasi-static process, the path followed by the system may be represented by a succession of equilibrium states. If there are finite departures from equilibrium, the process is non-quasi-static.

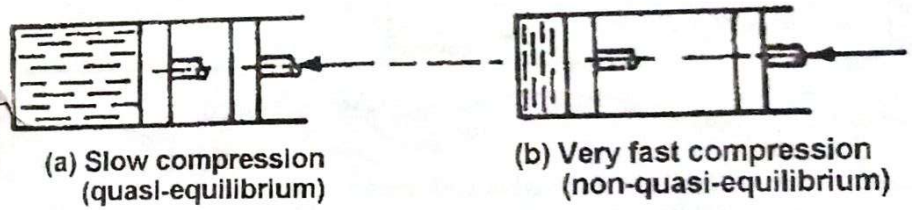


Fig. 1.8. Quasi equilibrium and non quasi-equilibrium compression process.

A quasi-static process is ~~an ideal~~ process. It is approximately realized by making the change very slowly. All real processes are not quasi-static because they take place with finite differences of pressure, temperature, and so on, between system and surroundings.

1.4.6. Reversible and irreversible processes

A thermodynamic process is said to be reversible if at any time during the process, both the system and surroundings can be returned to their initial states without any evidence of any change remaining in the system or its surroundings.

An **internally reversible** process is visualized as one in which the matter inside the system can be returned to its original state without any change from its original state, even though some change

is left in the surroundings. Similarly, an **externally reversibl** process is visualized as one in which the surroundings are returned their original state but a change is left within the system. **A proces is said to be completely reversible if it can be interrupted a any stage, and means can be found to restore both the syste and its surroundings to their respective initial states withou leaving a trace in either the system or its surroundings (Fi 1.9(a)).**

If changes are left in either the system or th surroundings after the process occurs, the process is called a irreversible process (Fig. 1.9(b)).

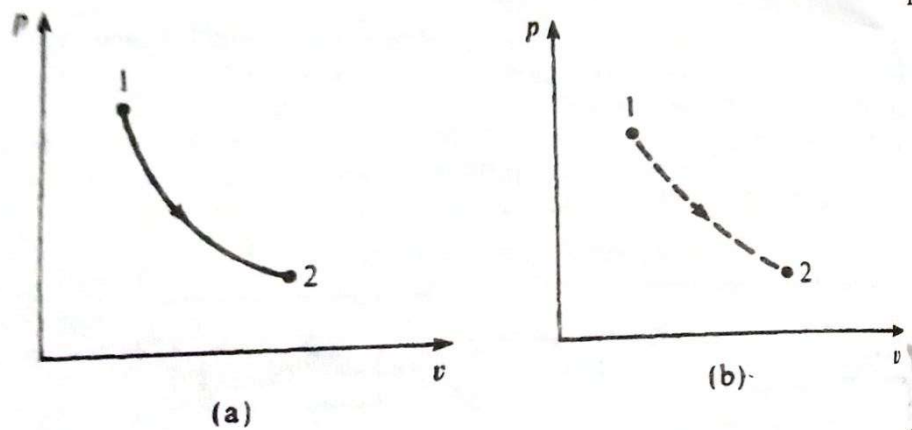


Fig. 1.9. Reversible and Irreversible process.
(a) Reversible process (b) Irreversible process.

Real processes are irreversible because they do not occur reversibly for both the system and surroundings. The necessar conditions for a reversible process are the same as those for the quasi equilibrium process. For the completely reversible or quasi equilibrium process, both a system and its surroundings must pas through a series of equilibrium states, and therefore the changes i state must occur very slowly.

Reversible processes actually do not occur in nature. They ar merely idealizations of actual processes. Reversible processes can b approximated by actual devices, but they can never be achieved. Tha is, all the processes occuring in nature are irreversible.

Reversible processes can be viewed as theoretical limits for th corresponding irreversible ones. Some processes are more irreversibl than others. We may never be able to have a reversible process, bu may certainly approach it.

.4.7. Thermodynamic Cycle

A thermodynamic cycle is a sequence of processes that begins nd ends at the same state. At the conclusion of a cycle all properties ave the same values they had at the beginning. Consequently, over he cycle the system experiences no net change of state. [Fig 1.10]. ycles that are repeated periodically play prominent roles in many eas of application.

Thermodynamic cycle can be defined with the help of athematics. Mathematically, this is, $\oint dx = 0$ where x is any roperty and the symbol $\oint$ indicates integration around a cycle.

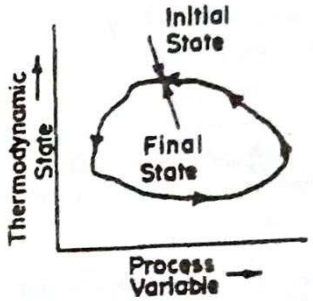


Fig. 1.10. Thermodynamic cycle

.4.8. Point function

When two properties locate a point on the graph (coordinate xes) then those properties are called as point function. For example, ressure, temperature, volume etc.

$$\int_1^2 dV = V_2 - V_1 \quad \text{[An exact differential]}$$

.4.9. Path function

Certain quantities cannot be located on the graph by a point but e given by the area or so, on that graph. In that case, the area on e graph, pertaining to the particular process, is a function of the ath of the process. Such quantities are called path functions. xamples of path function are heat, work etc.

Neither δQ nor δW are exact differential and therefore any tegration of the elemental quantities of work or heat which appear uring a change from state 1 to state 2 must be written as

$$\int_1^2 \delta W = W_{12} \text{ or } {}_1W_2 \text{ or } W$$

$$\int_1^2 \delta Q = Q_{12} \text{ or } {}_1Q_2 \text{ or } Q$$

1.4.10. Phase and pure substance

The term phase refers to a quantity of matter that homogeneous throughout in both chemical composition and phy structure. Homogeneity in physical structure means that the ma is all solid, or all liquid, or all vapour (or equivalently all gas). example, a mixture of water vapour and liquid water is a substance. On the other hand, air is not really a pure subst because when it is cooled sufficiently some of its components condense into their liquid state, thus changing the composition of remaining gases.

We further define a homogeneous substance as anything contains a single physical phase. Air at normal atmospheric condit is a homogeneous substance, but being a mixture of various gases not a pure substance. A mixture of liquid water and ice on the o hand is not a homogeneous substance, but it is a pure substanc pure substance that is also a homogeneous substance is calle simple substance. Liquid water is an example of a pure substance.

Pure substances must be chemically uniform, but need n consist of a single chemical species. For example, a homogene mixture of uniform chemical composition can often be treated as were a single phase of a pure substance. Air in its gaseous stat often treated as a pure substance even though it does not satisfy general definition of a pure substance.

1.5. PRESSURE

Pressure is the force exerted by a fluid per unit area. We sp of pressure only when we deal with a gas or a liquid. The counter- of pressure in solids is stress. Pressure can be viewed a compressive stress. Thus, absolute zero pressure corresponds t level of zero stress.

The pressure at a given point is the same in all directions. pressure in a fluid increases with depth as a result of the weight the fluid. This is due to the fluid at lower levels carrying more wei that the fluid at upper levels. The pressure varies in the verti direction as a result of gravitational effects, but there is no variat in the horizontal direction.

Since pressure is defined as force per unit area, it has the unit of newtons per square meter $\left(\frac{N}{m^2}\right)$, which is called a Pascal (Pa). That is,

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

The pressure unit pascal is too small for pressure encountered in practice; therefore kilopascal, Megapascal and bar are commonly used.

$$1 \text{ kPa} = 10^3 \text{ Pa}$$
$$1 \text{ MPa} = 10^6 \text{ Pa}$$
$$1 \text{ bar} = 10^5 \text{ Pa}$$

Atmospheric pressure varies with location on the earth, a standard reference can be defined and used to express other pressures

$$1 \text{ standard atmosphere (atm.)} = 101325 \text{ Pa}$$
$$= 101.325 \text{ kPa}$$
$$= 1.01325 \text{ bar.}$$

Most pressure measuring devices, however, are calibrated to read zero in the atmosphere, and so they indicate the difference between the absolute pressure and the local atmospheric pressure. This difference is called the gauge pressure. Pressures below atmospheric pressure are called vacuum pressure (Negative gauge pressure) and are measured by vacuum gauge which indicate the difference between the atmospheric pressure and the absolute pressure.

∴ Absolute presssure = Atomospheric pressure ± Gauge pressure

In thermodynamic relations and tables, absolute pressure is almost always used. Often the letter "a" (for absolute pressure) and "g" (for gauge pressue) are added to pressure units in order to clarify what is meant.

1.5.1. Gauge pressure

The pressure which is above the atmospheric pressure is known as gauge pressure. The atmospheric pressure is a zero gauge pressure. (Fig 1.11)

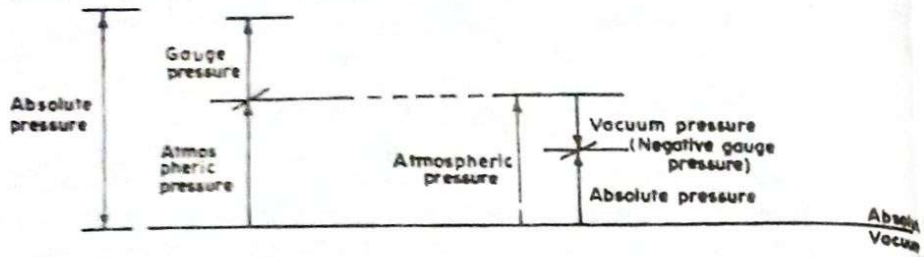


Fig. 1.11. Absolute, gauge and vacuum pressures.

1.5.2. Vacuum pressure

The pressure which is below the atmospheric pressure is know as vacuum pressure. (Fig 1.11)

1.5.3. Manometer

Manometer is a device which measures either gauge pressure vacuum pressure.

1.5.4. Barometer

The atmospheric pressure is measured by a device called barometer, thus the atmospheric pressure is often called th barometric pressure.

1.6. TEMPERATURE SCALES

The temperature scale used in the S.I. units today are th Celsius scale (also called the Centigrade scale). The ice and stea points are assigned the values of 0°C and 100°C, respectively.

A more useful temperature scale in thermodynamics is th absolute temperature scale. As the name implies, there are n negative temperatures on an absolute temperature scale, and th lowest attainable temperature is absolute zero. The absolu temperature scale in the S.I. is the Kelvin scale. The temperature un on this scale is the **Kelvin which is designated by K without th degree symbol.**

The Celsius temperature scale has the same magnitude as th Kelvin. Thus temperature differences are identical on both scales However, the zero point on the Celsius scale is shifted to 273.15 K, a shown by the following relationship between the Celsius temperatur and Kelvin temperature

$$t\,(0\,^{\circ}C) = T\,(K) - 273.15$$

From this it can be seen that on the Celsius scale the triple point of water is 0.01°C and that 0K corresponds to - 273.15°C.

The Celsius scale is defined so that the temperature at the ice point, 273.15K, is 0°C, and the temperature at the steam point, 373.15K, is 100°C. Accordingly, there are 100 Celsius degrees in this interval of 100 Kelvins, a correspondance that is contrived through the selection of 273.16K as the triple point temperature. Observe that since the temperatures at the ice and steam points are experimental values subject to revision in light of more precise determinations, the only Celsius temperature that is fixed by definition is that at the Triple point of water.

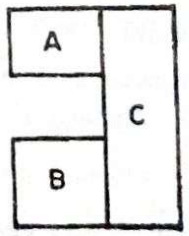
Finally, when making engineering calculations, it is common to round off the last number to 273. This is frequently done in subsequent sections of the text.

1.7. ZEROTH LAW OF THERMODYNAMICS

When two bodies are in thermal equilibrium with a third body, the two bodies are in thermal equilibrium with each other, and all three bodies are said to be at the same temperature.

Let us consider the three thermodynamic system, A, B, and C [Fig 1.12]. If system A is in thermal equilibrium with system C and system B is in thermal equilibrium with system C, then system A is in thermal equilibrium with system B.

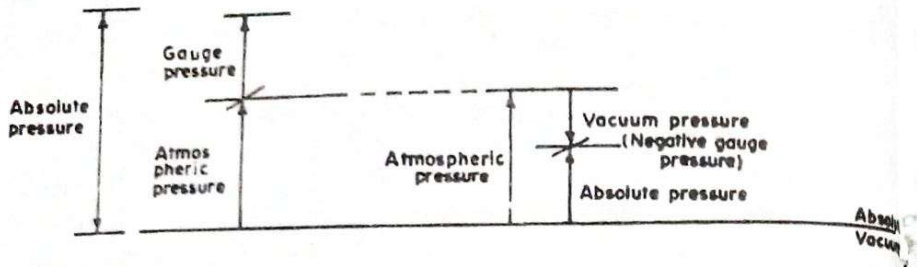
Because the principle is not derivable from other laws, and because it precedes the first and second laws of thermodynamics in the logical presentation of thermodynamics, it is called the zeroth law of thermodynamics. This zeroth law of thermodynamics was enunciated only very recently by R.H. Fowler in the year 1931.



If $T_A = T_C$ and $T_B = T_C$

Then $T_A = T_B$

Fig. 1.12. Zeroth law of thermodynamics



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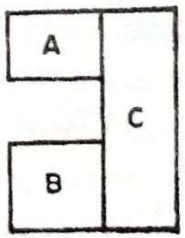

If $T_A = T_C$ and $T_B = T_C$

Then $T_A = T_B$

Fig. 1.12. Zeroth law of thermodynamics

1.8. WORK

In mechanics, work is defined as the product of a force and distance moved in the direction of the force.

In thermodynamics, work is identified only when it crosses the boundary of the system. Otherwise no work will be done by the system or on the system. It is a form of energy. It is a transient phenomenon. Systems do not posses work, in otherwords, work is not a property of the system.

Engineeing thermodynamics is concerned mainly with systems whose equilibrium states are described in terms of P, V and T coordinates. For such systems,

$$W = \int p\,dV$$

$$= \int (\text{Intensive property}) \times (\text{change in extensive property})$$

Since Work is a path function, cannot be integrated until the pressure volume path is known.

The magnitude of work during a process will be written as

$$W = \int_1^2 \delta W = W_{1-2} \text{ or } {}_1W_2$$

and cannot be written as

$$W = \int_1^2 \delta W \neq W_2 - W_1$$

The restrictions under which the expression $\delta W = pdV$ holds are

(1) the system is closed,

(2) there are no viscous effects within the system,

(3) all property values has the same value on all boundaries of the system

(4) path is assumed to be reversible

(5) effects due to electric, magnetic & gravity are not considered (or neglected or no effect).

1.9. HEAT

Heat is energy transferred across a system boundary by virtue of a temperature difference across the boundary.

The definition implies the important fact that heat, like work, is a transient quantity which can be identified only as it crosses the

boundary of a system which is interacting with the surroundings or with other systems. Heat is not a property possesed by a system, and its differential is, therefore, not exact. For this reason, if Q denotes heat, its differential will be written as δQ in this text, and the total heat transferred during a process in which a system passes from state 1 to state 2 will be written as

$$Q = \int_1^2 \delta Q = Q_{1-2} \text{ or } {}_1Q_2$$

and cannot be

$$Q = \int_1^2 \delta Q \neq Q_2 - Q_1$$

It means that heat is a path function.

If during a path or process, the system is surrounded by an enclosure impermeable to heat, the process is termed adiabatic. Thus, during an adiatatic process, the heat transfer, Q = 0.

1.10. SIGN CONVENTION FOR WORK AND HEAT

When the system does work on the surroundings, i.e. energy is transferred in the form of work from the system to the surroundings, the work done by the system is taken to be positive, (Fig 1.13).

Work output of system = + W

Similarly, when the surroundings do work on the system i.e. energy is transferred in the form of work from the surroundings to the system, the work done on the system is taken to be negative (Fig 1.13)

Work input to system = – W

When heat is transferred from the surroundings to the system (as the result of an excess of temperature of the surroundings over the system) the heat received by the system is taken to be positive (Fig 1.13).

Heat received by the system = + Q

Conversely, when heat is transferred from the system to the surroundings, the heat given by the system is taken to be negative (Fig. 1.13)

Heat given up by system = – Q.

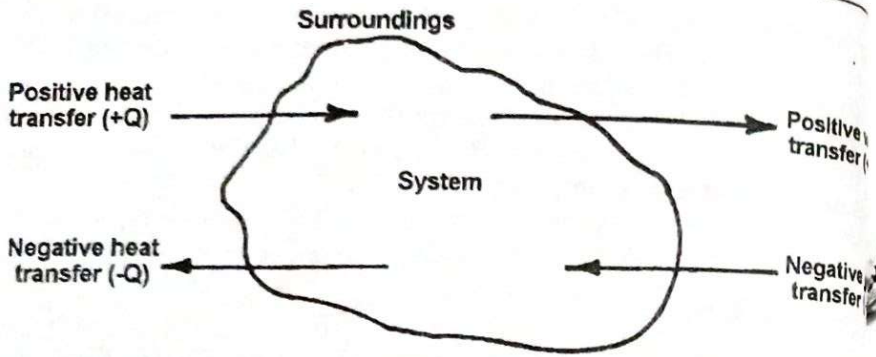

Fig. 1.13. Sign convention for work and heat.

1.11. COMPARISON OF WORK AND HEAT

Similarities

(1) Systems posses energy, but not work or heat.

(2) Both are path functions and inexact differentials.

(3) Both are associated with a process, not a state. Ur properties, work or heat has no meaning at a state.

(4) Both are recognized at the boundaries of the system as cross them. That is, both work and heat are bou phenomenon.

Dissimilarties

(1) The sole effect external to the system could be reduce rise of a weight but in the case of a heat transfer o effects are also observed.

(2) Temperature difference is needed in heat transfer.

(3) In a stable system there cannot be work transfer, howe there is no restriction for the transfer of heat.

1.12. INTERNAL ENERGY

In engineering thermodynamics the change in the total energy a system is considered to be made up of three macrosc contributions. One is the change in kinetic energy, associated with motion of the system as a whole relative to an external coordin frame. Another is the change in gravitational potential ener associated with the position of the system as a whole in the earl gravitational field. All other energy changes are lumped together the internal energy of the system.

The symbol E denotes the total energy of a system.

E = Kinetic energy + Potential energy + Internal energy.

$$= E_{kin.} + E_{Pot.} + U$$

The symbol U, denotes the internal energy. We do not know to determine the absolute values of U. We can, however, determine changes in U and, fortunately, this is all we need in the solution of practical problems. Internal energy is an extensive property of the system. The specific internal energy is symbolized by u.

It has been demonstrated mathematically and experimentally by Joule in 1843 that for an ideal gas the internal energy is a function of the temperature only, That is,

$$u = u(T)$$

For real gases internal energy is not a function of temperature only, but Joule did not detect a temperature change of the water, because the mass (or, to state the matter completely, the product of mass and specific heat) of the water was so great compared to that of the air that the temperature rise of the water was too small to be detect by his instruments. Joule later ran other experiments with different equipment to show that **for real gases internal energy is not a function of temperature alone**. However, in the ranges of pressure and temperature where pv = RT is sufficiently accurate for real gases, Joules law holds with the same order of accuracy.

1.13. ENTHALPY – A COMBINATION PROPERTY

In many thermodynamic analyses the sum of internal energy U and the product of pressure P and volume V appears. Because the sum (PV+U) occurs so frequently in subsequent discussions. For the sake of simplicity and convenience, this combination is defined as a new property, enthalpy and given the symbol H.

$$H = PV + U. \qquad \text{(kJ)}$$

Here, $pV = f_1(T)$ and $U = f_2(T)$,

Therefore, $H = f_3(T) = f(T)$.

Thus enthalpy, like internal energy, is a function of temperature alone for the perfect gas.

Enthalpy is an extensive property. Specific enthalpy h is given by

$$h = pv + u \qquad \left(\frac{kJ}{kg}\right)$$

Values of enthalpy are usually listed in tables of propertie Enthalpy is particularly useful in dealing with problems involvi flows.

1.14. GAS AND VAPOUR

Gas and vapour are often used as synonymous words. Th vapour phase of a substance is customarily called a gas when it above the critical temperature. Vapour usually implies a gas which not far from a state of condensation.

1.15. BOYLE'S AND CHARLE'S LAW

In 1662 Robert Boyle, an Englishman, observed during h experiments with a vacuum chamber that the pressure of gases inversely proportional to their volume. About 150 years later, Charles and J. Gay Lussac, Frenchmen, experimentally determine that at low pressures the volume of a gas is proportional to it temperature.

That is, $PV = \text{constant}$ (Boyle's law)

$$\frac{V}{T} = \text{constant} \quad \text{(Charle's law)}$$

1.16. THE IDEAL GAS EQUATION OF STATE

Any equation that relates the pressure, temperature, an specific volume of a substance is called an equation of state

$$pv = RT \qquad (1.1)$$

Where the R is called the characteristic gas constant. Equatio (1.1) is called the ideal gas equation of state, or simply the ideal ga relation, and a gas which obeys this relation is called an ideal gas. I this equation, p is the absolute pressure, T is the absolut temperature, v is the specific volume.

The characteristic gas constant R is different for each gas and i determined from

$$R = \frac{R_u}{M} \qquad (1.2)$$

Where R_u is the universal gas constant and M is the molar mas (also called molecular weight) of the gas. Constant R_u is the same fo all substances and its value is

usual tables of properties

$$R_u = 8.314 \frac{kJ}{kgmol.K}$$

1.16.1. Characteristic gas constant

The characteristic gas constant, R for any gas can be found when the molar mass is known. For example oxygen of molar mass 2 kg/kg mole, the characteristic gas constant

$$R = \frac{R_u}{M}$$

$$= \frac{8.314}{32} = 0.2598 \frac{kJ}{kg\,K}$$

1.17. SPECIFIC HEAT

The specific heat is defined as the energy required to raise the temperature of a unit mass of a substance by one degree. In general, this energy will depend on how the process is executed. In thermodynamics, we are interested in two kinds of specific heats. Specific heat at constant volume c_v and specific heat at constant pressure c_p.

Physically, the specific heat at constant volume c_v can be viewed s the energy required to raise the temperature of the unit mass of a substance by one degree as the volume is maintained constant. The energy required to do the same as the pressure is maintained constant s the specific heat at constant pressure c_p. The specific heat at constant pressure c_p is always greater than c_v because at constant pressure the the system is allowed to expand and the energy for this expansion work must also be supplied to the system.

Assuming a constant volume process and the definition of c_v

$$c_v = \left(\frac{\partial u}{\partial T}\right)_{\text{Constant volume}} \qquad (1.7)$$

$$\text{or,} \quad u_2 - u_1 = c_v(T_2 - T_1) = Q$$

Assuming a constant pressure process and the definition of c_p

$$c_p = \left(\frac{\partial h}{\partial T}\right)_{\text{Constant pressure}} \qquad (1.8)$$

$$\text{or,} \quad h_2 - h_1 = c_p(T_2 - T_1) = Q$$

Equations (1.7) and (1.8) are the defining equations for c_v and c_p nd their interpretation is given as follows.

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v$$ = the change in internal energy with temperature at constant volume.

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p$$ = the change in enthalpy with temperature at constant pressure.

Note that c_v and c_p are expressed in terms of other propen equation (1.9) should be replaced by the universal gas constant R_u. thus they must be properties themselves. Like any other property, that is, specific heats of substance depend on the state which, in genera specified by two independent, intensive properties.

1.18. RELATIONSHIP BETWEEN c_p AND c_v

The specific enthalpy of a perfect gas is written as

$$h = pv + u$$

and for a perfect gas, $pv = RT$

$$u = c_v T$$

$$h = c_p T$$

Putting in the above equation

$$c_p T = RT + c_v T$$

Therefore $$c_p = R + c_v$$

or, $$c_p - c_v = R$$

The ratio of two specific heat is written as That is

$$\frac{c_p}{c_v} = \gamma$$

γ is aloways greater than unity, γ is about 1.4 for diatomic g (C_0), (H_2), (N_2), (O_2). For monatomic gases such as Argon (A) helium, γ is about 1.6. For triatomic gases (CO_2), (SO_2), γ is about For some hydrocarbons the value of γ is quite law e.g. Ethane (C_2H = 1.22 & isobutane (C_4H_{10}), γ = 1.11.

By combining equations (1.9) and (1.10) c_p and c_v are obta interms of R and γ

$$c_p = \frac{R\gamma}{\gamma - 1}$$

and $$c_v = \frac{R}{\gamma - 1}$$

R = 8.314

Mol

1.19. MOLAR HEATS – VOLUMETRIC OR MOLECULAR

The molar heat is that required to raise the temperature of one mole by 1°C.

when the specific heats are given on molar basis, R in the

$$C_p = C_v + R_u \quad (1.13)$$

where $$C_p = Mc_p \quad (1.14)$$

and $$C_v = Mc_v \quad (1.15)$$

1.20. VARIATION IN SPECIFIC HEAT

Regnault's law states that the specific heats, (c_p and c_v) remain constant for all temperatures and pressures. This is not strictly true as the specific heats are found to change with temperature and has values approximating to

$$c_p = a + KT \quad \text{and} \quad c_v = b + KT$$

Where a, b and K are constants.

In problems specific heats are taken as constant unless otherwise mentioned.

1.21. JOULE'S LAW

Joule's law states that the internal energy of a perfect gas is a function of the absolute temperature only.

Joule's law states that u = f (T), hence it follows that the internal energy varies linearly with absolute temperature.

POINTS TO NOTE

1. All the thermodynamic properties are point functions.
2. A path function is not a point function.
3. A closed system is also known as control mass.
4. A open system is also known as control volume.
5. A boundary is also known as control surface.
6. Intensive properties are not a function of mass.
7. Extensive properties are function of mass.

8. All specific quantities are intensive properties.
9. A boundary is a surface in space that has zero thickness and mass.
10. In the pV = RT equation p is in absolute pressure and T absolute temperature.
11. Absolute pressure
    = Atmospheric pressure ± Gauge pressure.
12. T = °C + 273
13. 1 kilogram mole is known as Avogadro's number and has value 6.022×10^{26}.
14. A mole is defined as 10^{-3} kilogram moles.
15. (i) 1 N = (1 kg) (1 m/s^2) = 1 kg m/s^2,
    (ii) 1 Nm = 1 J = 1 kg m^2/s^2
    (iii) 1 J/kg = 1 m^2/s^2
16. Both heat and work are
    (i) Path functions,
    (ii) boundary phenomena and
    (iii) transient phenomena.
17. 1 bar = 750 mm of Hg = 10.2 m of water
18. For conversion purpose always use 1 mm of Hg = $\frac{1}{750}$ bar and 1 m of water = 9.81 kPa
19. 1 mm of Hg = 1 torr = 133.4 Pa
20. 1 atm = 760 mm of Hg = 1.01325 bar
21. It is often necessary to use the absolute pressure rather than the gauge or negative gauge (vacuum) in thermodynamics.
22. Use R_u = 8.314 kJ/kg mole K
23. For air
    c_p = 1.005 kJ/kgK, c_v = 0.718 kJ/kgK and R = 0.287 kJ/kgK

EXERCISES

A. Theory

1.1. Define the following terms
(a) Thermodynamics; (b) System; (c) Closed system;
(d) Open system; (e) Isolated system; (f) Adiabatic system.

1.2. Define Engineering thermodynamics. Give some applications

1.3. Discuss the limitations of engineering thermodynamics?

1.4. Draw a flow chart on application of engineering thermodynamics.

1.5. Define system, boundary and surroundings.

1.6. Define control mass, control volume and control surface.

1.7. Define macroscopic and microscopic system analysis.

1.8. Define property, state process and equilibrium.

1.9. Define intrinsic and extrinsic properties.

1.10. Define the following terms.
(a) Intensive property and
(b) Extensive property

1.11. Which of the following are extensive properties
(a) Volume; (b) Density; (c) Mass; (d) Temperature;
(e) Work and ; (f) Heat.

1.12. Are the following properties intensive or extensive?
(a) Internal energy; (b) Specific internal energy;
(c) Volume; (d) Specific volume;
(e) Enthalpy; (f) Specific enthalpy;
(g) Temperature and (h) Pressure.

1.13. Differentiate between microscopic and macroscopic points of view of thermodynamics.

1.14. Differentiate between homogenous & hetrogenous systems.

1.15. What do you understand by pure substance?

1.16. How state is defined in a thermodynamic cycle?

1.17. What is thermodynamic equilibrium? Explain the significance of quasi - static process.

1.18. Define : Reversible, internally reversible, and externally reversible process.

1.19. Define completely reversible process.

1.20. Differentiative between Revsersible and Irrevsible Process.

1.21. Define pressure, write all the units of pressure.

1.22. Differentiate between gauge pressure, vacuum pressure, atmospheric pressure and absolute pressure?

1.23. Define zeroth law of thermodynamics.

1.24. Write the value of atmospheric pressure in bar, mm of mercury and m of water.

1.25. What are the various forms of work ? How can these be obtained.

1.26. Explain the difference between heat and internal energy.

1.27. Define enthalpy. Show that enthalpy is a thermodynamic property.

1.28. Define thermodynamic work. What is positive and negative work?

1.29. Define Heat. What is positive and negative heat?

1.30. State Boyle's and Charle's laws.

1.31. Differentiate between universal gas constant and characteristic gas constant.

1.32. State Avogadro's hypothesis and prove that $MR = R_u$ has the same value for all gases.

1.33. Define kg mole; molecular heat.

1.34. Differentiate between gas and vapour.

1.35. Define specific heat. Why gases have two specific heats ?

1.36. Prove the formula $c_p - c_v = R$.

1.37. Prove $c_p = \dfrac{R\gamma}{\gamma - 1}$ where $\gamma = \dfrac{c_p}{c_v}$

1.38. Define Regnaults law.

1.39. Define Joule's law.

1.40. Write the units of universal gas constant and characteristic gas constant.

1.41. At N.T.P., find the volume of 1 Mole.

1.42. Calculate the volume of 1 Mole at 1 bar.

1.43. Differentiate with examples between heat and work. What are the similar features in their definition.

1.44. Are work and heat properties of system? Explain.

2

The First Law of Thermodynamics

2.1. INTRODUCTION

Chapter 1 covered work and heat. From 1840 to 1848, an Englishman named James Prescott Joule conducted his famous experiments that established a relationship between work and heat and paved the way to the formulation of the first law of thermodynamics. The first law of thermodynamics, also known as the conservation of energy principle, provides a sound basis for studying the relationships among the various forms of energy and energy interactions. The first law cannot be proved mathematically, but no process in nature is known to have violated the first law, and this should be taken as sufficient proof.

A major consequence of the first law is the existence and the definition of the property total energy E. In other words, strictly speaking, there is no definition for energy in thermodynamics until the first law of thermodynamics is considered. Thus, in the classical statements of the first law principle, the basic concepts work and heat are used without being defined in reference to energy.

Below we define the first law of thermodynamics for closed systems.

2.2. THE FIRST LAW OF THERMODYNAMICS

When a system undergoes a thermodynamic cycle then the net heat supplied to the system from its surroundings is equal to the net work obtained from the system to surroundigns. Mathematically, the first law may be written as

$$\oint \delta Q = \oint \delta W \tag{2.1}$$

The symbol δ has been used to indicate both Q and W inexact differentials.

2.3. JOULES CONSTANT (J)

If different units are used for heat and work then we proportional instead of equal in the definition of first law thermodynamics. The proportinality factor, called the mechan equivalent of heat or Joules constant, denoted by the symbol J required, so that the **first law may be written** as

$$J \oint \delta Q = \oint \delta W \quad (2$$

J is constant and it depends upon the units of Q and W

If SI units are used for both heat and work, then it is necessary to use J in the equation (2.2)

2.4. COROLLARIES OF FIRST LAW OF THERMODYNAMICS

The first law of thermodynamics, in the form of equation (2. has a number of important consequences which are stated in the for of corollaries. These are propositions (3 in number) which are deduc from the first law and are often proved by "contradiction".

2.4.1. Corollary 1

There exists a property of a closed system called energy su that a change in its value is equal to the difference between the h supplied and the work done during any change of state. In oth words, there exists a point function such that

$$\delta Q - \delta W = \Delta E = E_2 - E_1$$

Proof :– Let a system change its state by some process A fr state 1 to state 2 (Fig 2.1), and by some process B from state 2 to st 1. Then by the first law, we may write

$$\int_1^2 (\delta Q - \delta W)_A + \int_2^1 (\delta Q - \delta W)_B = 0 = \int (\delta Q - \delta W)_{AB} \quad (2$$

Let process C be any other process by which the system m change in state from 2 to 1. Then for the cycle 1 A 2 C 1 we likewise write

$$\int_1^2 (\delta Q - \delta W)_A + \int_2^1 (\delta Q - \delta W)_C = 0 = \int (\delta Q - \delta W)_{AC} \quad (2$$

Comparing equations (2.3) and (2.4) one concludes that

$$\int_1^2 (\delta Q - \delta W)_B = \int_1^2 (\delta Q - \delta W)_C \quad (2.5)$$

The processes B and C were chosen arbitrarily, but note that the value of the integral $\int_1^2 (\delta Q - \delta W)$ is determined solely by the end states and does not depend on the process. A thermodynamic quantity whose value is determined only by the end states is a thermodynamic property. The integral of (δQ – δW) for any process transforming the system from state 1 to state 2 thus defines a change in thermodynamic property. This property is referred to as energy and is designated E. It also follows that incremental change in E (dE) is equal to the quantity (δQ – δW). Thus

$$\delta Q - \delta W = dE$$

$$\text{or,} \quad Q_{12} - W_{12} = \Delta E = E_2 - E_1 \quad (2.6)$$

Energy E, is composed of internal, kinetic and potential energy. Since, in a closed system, kinetic and potential energy effects generally play a negligible part, equation (2.6) may be simplified still further to the form known as the non flow energy equation (NFEE)

$$Q_{12} = W_{12} + \Delta U = W_{12} + (U_2 - U_1) \quad (2.7)$$

The property U is called the internal energy of the system. This property U is the outcome of the first law of thermodynamics. Physically, an increase in U is associated with a rise in temperature of the system, although this correspondence does not always exist.

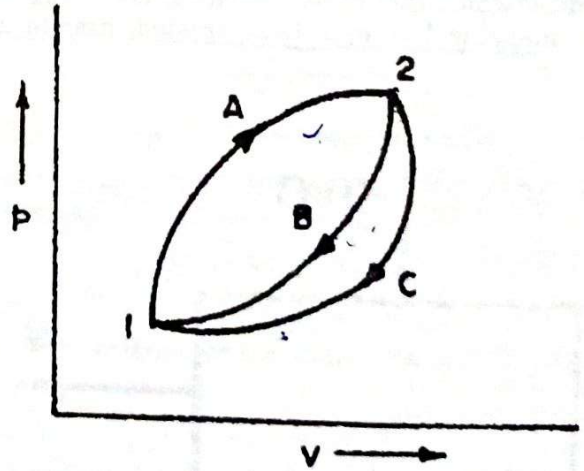

Fig. 2.1. Cyclic process between two fixed end states.

2.4.2. Corollary 2, Conservation of energy

The internal energy of a closed system remains unchanged if the system is isolated from its surroundings.

Proof : By definition of an isolated system, Q = W = 0 Therefore $\Delta U = U_2 - U_1 = 0$.

$$Q = 0, \; W = 0, \; \Delta U = 0$$

This corollary is often called the law of conservation of energy.

Law of conservation of energy is less general than the first law as it does not state the energy changes when Q and W are not zero.

2.4.3. Corollary 3, Perpetual motion machine of first kind.

A perpetual motion machine of first kind (pmm I) is impossible.

A perpetual motion machine is a device which produces continuous supply of work without requiring any interaction from surroundings or other systems. Such a device would, in effect, create energy. Equation (2.4) states that, if a net amount of heat is not supplied during a cycle, no net amount of work can be obtained (Fig. 2.2), That is

$$\oint \delta W = 0 \quad \text{if } \delta Q = 0$$

The existence of a **pmm** would therefore violate the first law of thermodynamics (which cannot be proved). It is an experimental fact that no such machine has ever been created, despite attempts to invent one.

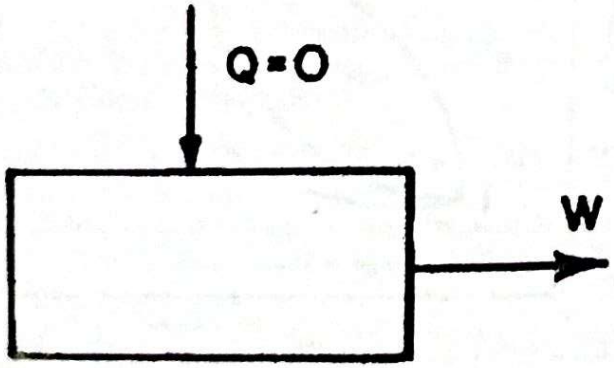

Fig. 2.2. PMM-I kind

2.4.4. Energy balanced examples

The first law of thermodynamics as stated by equation (2.1) applies to a clsoed system undergoing a thermodynamic cycle. However, all heat engines and refrigerators/heat pumps comprise of a series of open system components. The working substance in them also goes through cyclic changes of states, and always returns to the initial state. Further, when a boundary is drawn around the cyclic equipment as a whole, as shown in Figs. 2.3 and 2.4, these reduce to closed systems with mass transfer neither entering nor leaving the boundary. For all such cyclic equipment, equation (2.1) can be expressed in the form of an energy balance.

$$\sum Q = \sum W \tag{2.8}$$

It is seen that the energy balance refers to cycle analysis of the complete cyclic equipment as a closed system. Whereas each individual equipment separately requires a component analysis as an open system by itself.

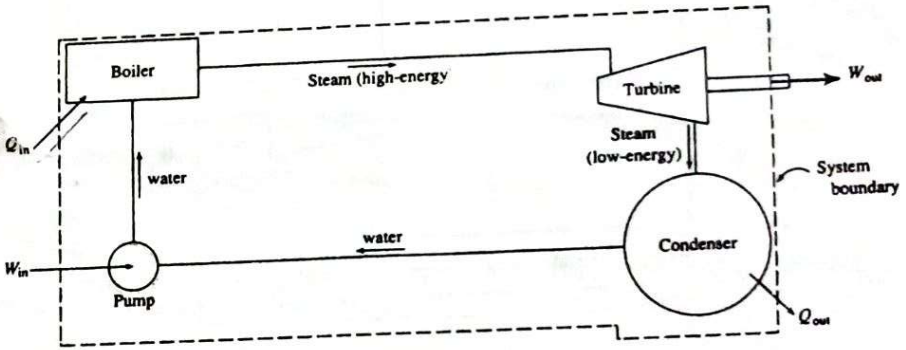

Fig. 2.3. Steam power plant cycle

The energy balance for the simple steam power plant cycle in Fig 2.4 is given as

$$Q_B - Q_C = W_T - W_P \tag{2.9}$$

Where $Q_B = Q_f - Q_{gases}$

The energy balance for the simple refrigeration cycle in Fig. 2.5 is given as

$$Q_E - Q_C = W \tag{2.10}$$

The basis of the first law of thermodynamics is the law of conservation of energy which, for a system, can be expressed as :

Net energy added to a system = Energy in – Energy out

= Increase in stored energy

But, for a system undergoing a cycle, there is no change in the state of the system so that the stored energy is zero. Hence :

Energy in = Energy out

The first law of thermodynamics as enunciated by equation (2.1) or by the energy balance equation (2.8) states the same fact in the context of heat and work interactions between system and surroundings.

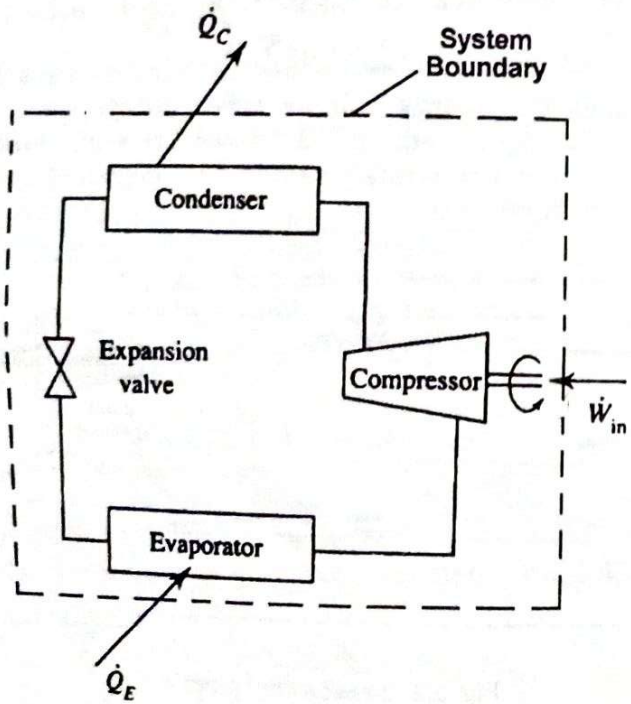

Fig. 2.4. Simple refrigeration cycle

2.5. REVERSIBLE NON FLOW PROCESSES

The following assumptions are used for the non flow processes :

(1) Reversible

(2) Displacement work is used

(3) Kinetic and Potential energy effects are negligible

All processes apply to mass m of working fluid contained within the system boundaries.

The non flow energy equation is always satisfied, when applied to a non flow proess; irrespective of the nature of vapours, a perfect gas, non perfect gas, a liquid, a solid, in fact any conglomeration of non reacting matter imaginable.

2.5.1. Constant volume process (V = Constant)

In a constant volume process the working substance is contained in a **rigid vessel**, hence the boundaries of the system are immovable and no work can be done on or by the system, other than paddle wheel work input. It will be assumed that constant volume implies zero work unless stated otherwise.

$$W_{12} = \int_1^2 p dV = 0 \qquad [dV = 0]$$

$$\int_1^2 dU = U_2 - U_1 = m\, c_v\, (T_2 - T_1) = \Delta U$$

From the non flow energy equation

$$Q_{12} = W_{12} + \Delta U$$

$$Q_{12} = O + (U_2 - U_1)$$

Therefore,

$$Q_{12} = m\, c_v\, (T_2 - T_1) \qquad (2.11)$$

All the heat transferred or supplied in a constant volume process goes to increasing the internal energy.

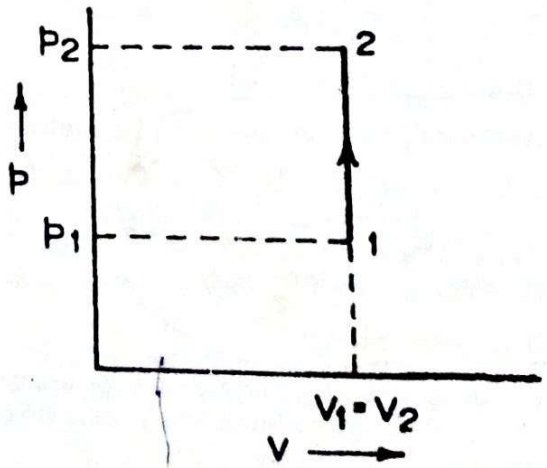

Fig. 2.5. Constant volume process

In the p–V diagram the constant volume process is graphi presented by a straight line parallel to the p-axis (Fig 2.5)

Using the p-V-T relation,

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

But here, $V_1 = V_2$

Therefore, $\frac{p_1}{p_2} = \frac{T_1}{T_2}$

i.e. in the constant volume process the pressure of gas is dire proportional to the absolute temperature.

2.5.2. Constant pressure process (p = Constant)

When the boundary of the system is inflexible as in cons volume process, then the pressure rises when heat is supplied. H for a constant pressure process the boundary must move agains external resistance as heat is supplied; for instance a fluid cylinder behind a piston can be made to undergo a constant press process. Since the piston is pushed through a certain distance by th force exerted by the fluid, then work is done by the fluid o surroundings.

$$W_{12} = \int_1^2 p\,dV = p\int_1^2 dV = p(V_2 - V_1)$$

$$\int_1^2 dU = U_2 - U_1 = m\,c_v(T_2 - T_1) = \Delta U$$

From the non flow energy equation

$$Q_{12} = W_{12} + (U_2 - U_1)$$

$$Q_{12} = p(V_2 - V_1) + (U_2 - U_1)$$

$$= (p_2 V_2 + U_2) - (p_1 V_1 + U_1)$$

$$= H_2 - H_1 = m\,c_p(T_2 - T_1)$$

= Change in enthalpy

In the p-V diagram the constant pressure process is graph presented by a straight line parallel to the V- axis (Fig 2.6).

Using p-V-T relation

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

But here, $p_1 = p_2$

Therefore, $\frac{V_1}{T_1} = \frac{V_2}{T_2}$

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \qquad (2.14)$$

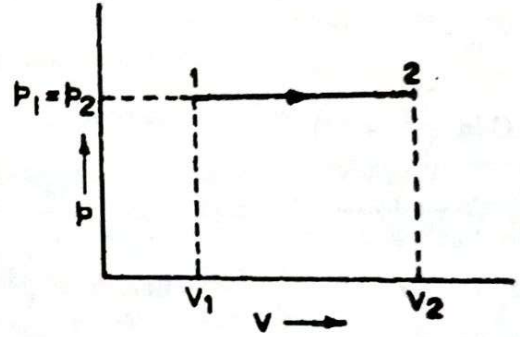

Fig. 2.6. Constant pressure process.

2.5.3. Constant temperature or isothermal process (T=Constant)

If a system be put in contact with a constant temperature and the heat and work be so proportioned that the temperature of the system remains the same. The process then becomes a constant temperature or isothermal process. In an isothermal expansion heat must be added continuously in order to keep the temperature at the initial value. Similarly in an isothermal compression heat must be removed from the fluid continuously during the process.

For the perfect gas only, the internal energy U is a function of temperature only. The change in the internal energy in the isothermal process $\Delta U = 0$, since $\Delta T = 0$. Hence U is constant, and all heat imparted to the gas during the process is used for the work of expansion.

The work done by a perfect gas in expanding from state 1 to state 2 isothermally and reversibly is given by the shaded area on Fig.2.7.

$$W_{12} = \int_1^2 p\,dV$$

But, $$p\,V = p_1 V_1 = p_2 V_2 = C$$

Therefore $$P = \frac{C}{V}$$

Hence, $$W_{12} = C \int_1^2 \frac{dV}{V}$$

$$C\,ln\,\frac{V_2}{V_1} = p_1 V_1\,ln\,\frac{V_2}{V_1} \qquad (2.$$

Appling the non flow energy equation

$$Q_{12} = W_{12} + (U_2 - U_1)$$

$$= C\,ln\,\frac{V_2}{V_1} + 0$$

$$= p_1 V_1\,ln\,\frac{V_2}{V_1}$$

$$= p_1 V_1\,ln\,r \qquad \left(\text{where } r = \frac{V_2}{V_1}\right)$$

$$= m\,RT_1\,ln\,r \qquad (2.$$

Therefore for an isothermal process for a perfect gas, the h flow is equivalent to work done in an isothermal process for a perf gas only. The equation for this process is derived from the equation state

$$pV = \text{Constant}$$

In the p-V diagram, Fig 2.7, the isothermal line, representi the process, is a section of an equilateral hyperbola.

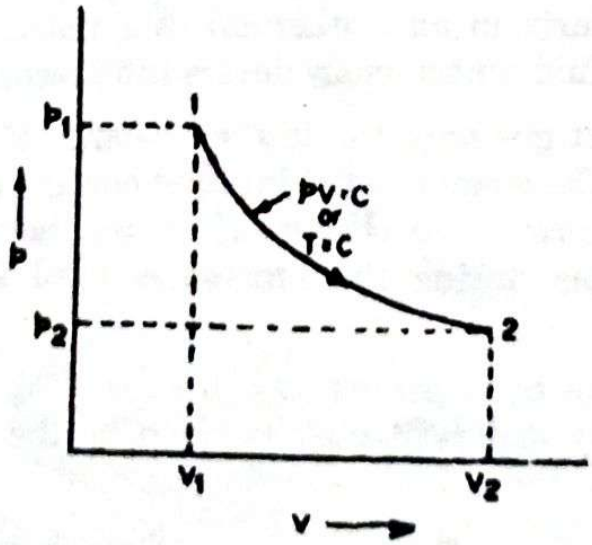

Fig. 2.7. Isothermal process.

2.5.4. Hyperbolic process (pV = C)

For perfect gas, $\frac{pV}{T}$ = constant. Therefore, when pV= Constant, T is also constant, Thus for a perfect gas hyperbolic expansion is also a constant temperature expansion.

The work done and heat supplied in this case are the same as those in the case of Isothermal expansion discussed in article 2.5.3.

It is to be noted that the isothermal and hyperbolic processes are identical only in the case of a perfect gas and not for a vapour. For example in the wet steam hyperbolic expansion is not equal to isothermal. It will be discussed in the properties of steam chapter.

2.5.5. Adiabatic process (pV^{γ} = C)

An adiabatic is the process in which there is no heat transfer between the working fluid and the surroundings (Q = 0 and δQ = 0) such a process can be reversible or irreversible.

From the non flow energy equation

$$Q_{12} = W_{12} + U_2 - U_1$$

Here, $Q_{12} = 0$

Therefore $$W_{12} = U_1 - U_2 \qquad (2.17)$$

Equation (2.14) is true for an adiabatic process whether the process is reversible or not. For an adiabatic process to take place, **perfect thermal insulation** for the system must be available.

$$W_{12} = U_1 - U_2 = m\,c_v(T_1 - T_2)$$

We know that $$c_v = \frac{R}{\gamma - 1}$$

Therefore, $$W_{12} = \frac{m\,R}{\gamma - 1}(T_1 - T_2)$$

$$\boxed{W_{12} = \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}} \qquad (2.18)$$

A reversible adiabatic process for a perfect gas is shown on a p-V diagram in Fig 2.9. The work done is given by the shaded area and this area can be evaluated by integration.

$$W_{12} = \int_1^2 p\,dV$$

$$pV^{\gamma} = p_1 V_1^{\gamma} = p_2 V_2^{\gamma} = C$$

$$W_{12} = C\int_1^2 \frac{dV}{V^{\gamma}}$$

$$= C\left[\frac{V^{-\gamma+1}}{-\gamma+1}\right]_1^2$$

$$= C\left(\frac{V_2^{-\gamma+1} - V_1^{-\gamma+1}}{1-\gamma}\right)$$

$$= \frac{p_2 V_2^{\gamma} V_2^{-\gamma+1} - p_1 V_1^{\gamma} V_1^{-\gamma+1}}{1-\gamma}$$

$$= \frac{p_2 V_2 - p_1 V_1}{1-\gamma}$$

$$= \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$$

This is the same expression obtained before in equation (2.18)

For a perfect gas, a law relating p and V may be obtained for a reversible adiabatic process, by considering the properties of an ideal gas.

Since,

$$H = pV + U$$

$$dH = pdV + Vdp + dU$$

$$pdV + dU = dH - Vdp \qquad [\delta Q = pdV + dU]$$

$$\delta Q = dH - Vdp \qquad [\delta Q = 0]$$

$$0 = dH - Vdp$$

$$dH = Vdp \qquad (2.19)$$

From non flow energy equation in differential form

$$\delta Q = \delta W + dU$$

$$0 = pdV + dU \qquad [\delta Q = 0]$$

$$dU = -pdV \qquad (2.20)$$

Dividing (2.19) by (2.20) we obtain

$$\frac{dH}{dU} = -\frac{Vdp}{pdV}$$

$$\frac{m\,c_p\,dT}{m\,c_v\,dT} = -\frac{Vdp}{pdV}$$

$$\gamma = -\frac{Vdp}{pdV}$$

$$\gamma\,pdV + Vdp = 0$$

Dividing the above expression by p-V and integrating it, we obtain the equation for the adiabatic process :

$$pV^{\gamma} = \text{Constant} \qquad (2.21)$$

In the p-V diagram (Fig 2.8) the adiabatic line is an unequilateral hyperbola

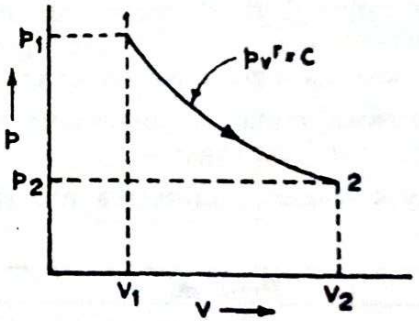

Fig. 2.8. Adiabatic process.

The relation between the parameters p and V in this process is obtained from equation (2.21)

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} \qquad (2.22)$$

The following equation of state can be written for two points of the process under consideration

$$p_1 V_1 = m R T_1 \quad \text{and} \quad p_2 V_2 = m R T_2$$

whence, $$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

or, $$\frac{p_2}{p_1} = \frac{V_1 T_2}{V_2 T_1}$$

Substituting the ratio of the pressures from equation (2.22), we obtain

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma - 1} \qquad (2.23)$$

Solving equations (2.22) and (2.23), we determine the relation between P and T.

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(\gamma - 1)/\gamma} \qquad (2.24)$$

2.5.6. Polytropic process ($pV^n = C$)

Processes are referred to as polytropic if they satisfy the equation $pV^n = C$, where n is the polytropic exponent which has a constant numerical value for a given polytropic process.

Polytropic processes make it possible to investigate diverse phenomena taking place in heat engines.

The proviously considered processes are special case of the polytropic processes.

process	Equation	Value of n
Constant pressure	$pv^0 = C$ or, $p = C$	$n = 0$
Constant volume	$p^1/_{\pm\infty} V = C$ or, $V = C$	$n = \pm\infty$
Hyperbolic and Isothermal	$pV = C$ and $T = C$	$n = 1$
Adiabatic	$pV^\gamma = C$	$n = \gamma$

The relation between the parameters and the expression of work in the polytropic process are similar to those in the adiabatic process, since the equation for the polytropic process is similar to that for the adiabatic process in which the exponent γ was replaced with exponent n, namely

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^{n} \qquad (2.25a)$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1} \qquad (2.25b)$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(n-1)/n} \qquad (2.25c)$$

The same relates to the expression for work

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{n - 1}$$

and the internal energy is given by

$$\Delta U = m\, c_v (T_2 - T_1)$$

From the non flow energy equation

$$Q_{12} = W_{12} + \Delta U$$

$$= \frac{p_1 V_1 - p_2 V_2}{n - 1} + m\, c_v (T_2 - T_1) \qquad (2.26)$$

$$= \frac{m R (T_1 - T_2)}{n - 1} + m\, c_v (T_2 - T_1)$$

$$= m (T_2 - T_1) \left[c_v - \frac{R}{n - 1}\right]$$

$$= m (T_2 - T_1) \left[c_v - \frac{(c_p - c_v)}{n - 1}\right]$$

$$= m (T_2 - T_1) \left[\frac{n c_v - c_v - c_p + c_v}{n - 1}\right]$$

$$= m (T_2 - T_1) \left[\frac{c_p - n c_v}{1 - n}\right] \qquad (2.27)$$

$$= m c_n (T_2 - T_1)$$

Where,

$$c_n = \frac{c_p - n c_v}{1 - n} = \frac{c_v (\gamma - n)}{1 - n} \qquad (2.28)$$

c_n is often called the polytropic specific heat.

The value of c_n will vary even if c_p and c_v are constants as its

variation linked with the value of n. The possible variation of n c between zero and infinity. Observe that c_n is negative when $\gamma > n$. A negative specific heat means that heat is rejected by the gas though its temperature rises; or that heat is supplied to the gas though its temperature decreases(i.e. there is some cooling du compression and heating during expansion). Fig 2.9 shows effec varying n on p-V plane. Expansions or compressions are imagine commence from point 0. Note that all positive values of n give cu in the second and fourth quadrants on the p-V diagram. It is usef remember that isentropic curve on a p-V plane is steeper than isothermal curve.

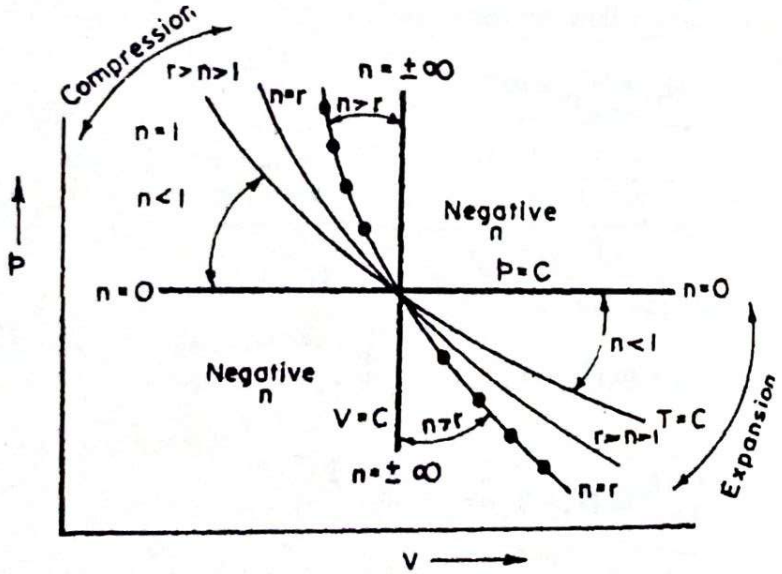

Fig. 2.9. Effects of varying n.

If the initial and final volumes and pressures are known for expansion of the form $pV^n = C$, the value of n can be obtained fro logarithms of these values

Now, $p_1V_1^n = p_2V_2^n$

Taking log on both the sides

$$\log p_1 + n \log V_1 = \log p_2 + n \log V_2$$

$$\text{or, } n(\log V_2 - \log V_1) = \log p_1 - \log p_2$$

$$\text{or, } n \log\left(\frac{V_2}{V_1}\right) = \log\left(\frac{p_1}{p_2}\right)$$

$$\text{or } n = \frac{\log(p_1/p_2)}{\log(V_2/V_1)} \tag{2.29}$$

2.6. IRREVERSIBLE PROCESSES

Most processes occuring in a cylinder behind a piston can be assumed to be internally reversible to a close approximation. Certain processes cannot be assumed to be internally reversible due to friction, turbulence, temperature gradients, and so on. The important cases will be dissused briefly.

2.6.1. Constant internal energy process or free expansion process or unresisted expansion

The expansion of gas against vacuum is called constant internal energy process.

The constant internal energy be represented by a line labbled U= constant as in Fig 2.10. The first law would then give

$$\delta Q = \delta W = pdV$$

The most important practical example of a constant internal energy process, however, happens to be irreversible.

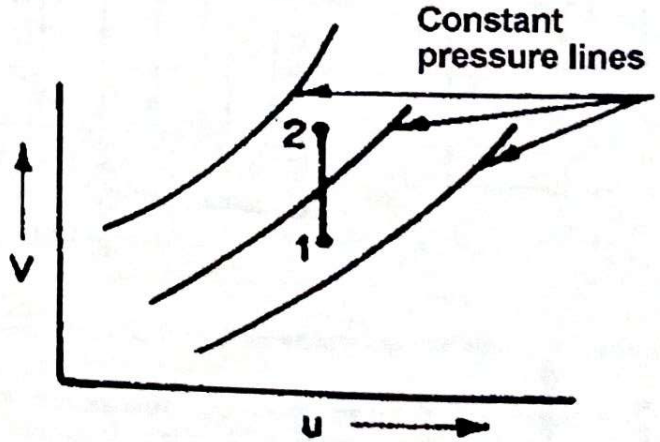

Fig. 2.10. Constant internal energy.

able 2.1. Process relations for perfect gas

Process	Constant pressure	Constant volume	Constant temprature	Adiabactic	Polytropic
Exponent in $pV^n = C$	$n = 0$	$n = \infty$	$n = 1$	$n = \gamma$	$n = n$
p, V, T relations	$\frac{T_2}{T_1} = \frac{V_2}{V_1}$	$\frac{T_2}{T_1} = \frac{p_2}{p_1}$	$p_1 V_1 = p_2 V_2$	$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$ $\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1}$ $\left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma}$	$p_1 V_1^n = p_2 V_2^n$ $\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1}$ $\left(\frac{p_2}{p_1}\right)^{(n-1)/n}$
Heat added	$m c_p (T_2 - T_1)$	$m c_v (T_2 - T_1)$	$= p_1 V_1 \ln r$	0	$m c_n (T_2 - T_1)$ $= m c_v \left(\frac{\gamma - n}{1 - n}\right)$ $\times (T_2 - T_1)$
$\int_1^2 p\,dv$	$p_1 (V_2 - V_1)$	0	$p_1 V_1 \ln r$	$\frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$	$\frac{p_1 V_1 - p_2 V_2}{n - 1}$
$\int_1^2 v\,dp$	0	$V_1 (p_1 - p_2)$	$p_1 V_1 \ln r$	$\frac{\gamma}{\gamma - 1} (p_1 V_1 - p_2 V_2)$	$\frac{n}{n - 1} (p_1 V_1 - p_2 V_2)$

Consider a perfectly insulated ($\delta Q = 0$) rigid container divided into two compartments as shown in Fig 2.11(a). One compartment is filled with gas; the other compartment is evacuated.

When the diaphragm is punctured (or removed), the gas will expand freely into that part of the container originally evacuated. The specific volume of the gas will increase; its pressure and temperature will probably decrease. This change of state, however, takes place without the displacement of the boundary against an external resistance, so that the work done is zero ($\delta w = 0$) although $pdV \neq 0$ ($\delta W \neq pdV$) The process, however, is still one of constant internal energy, since

$$dU = \delta Q - \delta W = 0 \tag{2.30}$$

But it is an irreversible process because, apart from the initial and final states, the remaining states are not equilibrium states. The pressure, for example, at any intermediate stage in the process is not uniform throughout the system. Thus, no points can be located on a diagram to represent the intermediate states, and, consequently, no smooth curve may be drawn to represent the process.

By providing a large number of partitions in the exhausted space, a number of intermediate states may be plotted and these will trace out a line of constant internal energy as in Fig 2.11(b).

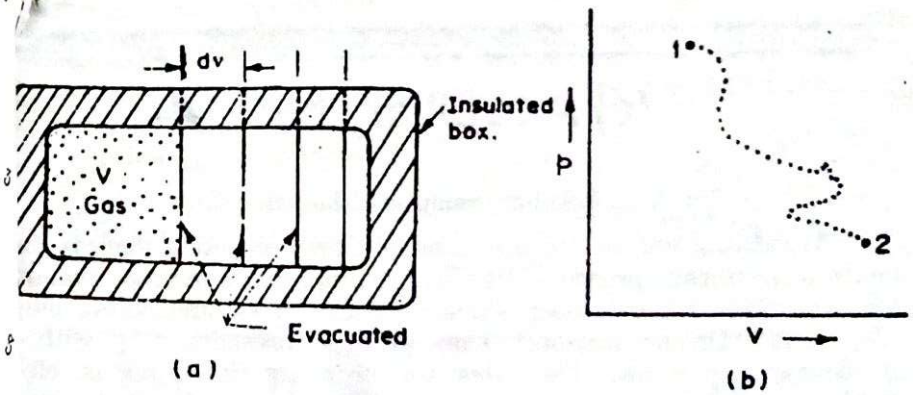

Fig. 2.11. Free expansion process.

A free expansion process cannot be reversed exactly along the original path of change without introducing the work and heat quantities which played no part in the original process.

2.6.2. Throttling process (isoenthalpic process)

A flow of fluid is said to be throttled when there is restriction to the flow, when the velocities before and either equal or negligibly small, and there is a negligible loss to the surroundings.

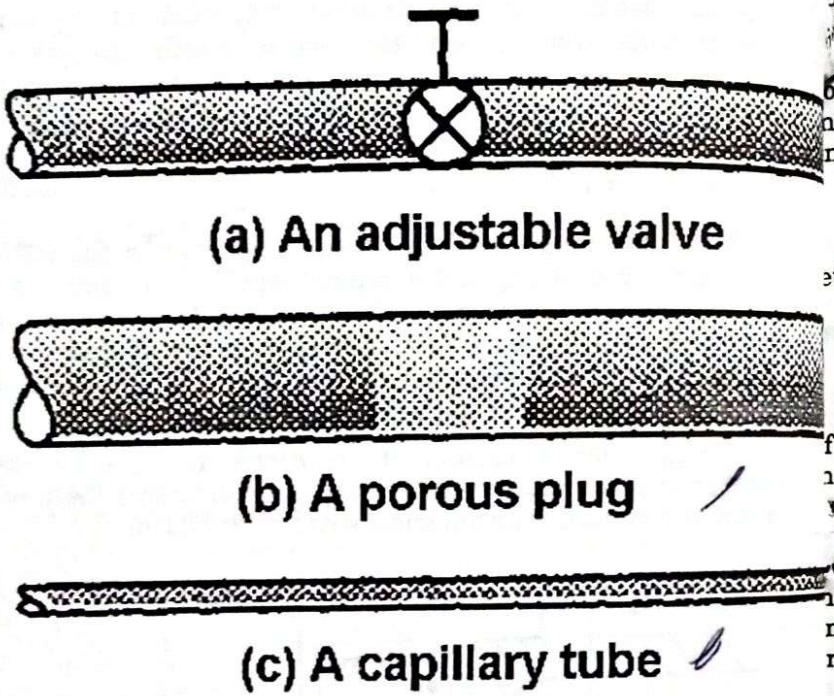

Fig. 2.12. Familiar examples of thorttling valve.

Throttling valves are any kind of flow restricting devices t cause a significant pressure drop in the fluid. Some familiar exam are, an orifice, a partly open valve, capillary tubes, and porous pl (Fig. 2.12). Unlike turbines, they produce pressure drop with involving any work. The pressure drop in the fluid is of accompanied by a large drop in temperature, and for that rea throttling devices are commonly used in refrigeration and conditioning applications. The magnitude of the temperature drop sometimes, the temperature rise) during a throttling process governed by a property called the Joule-Thomson coefficient

$$\mu_J = \left(\frac{\partial T}{\partial p}\right)_h \qquad (2.$$

A positive μ_J means that the temperature of the gas drops uring throttling; a negative μ_J means that the temperature rise uring throttling. For $\mu_J = 0$, temperature remains constant.

Throttling valves are usually small devices, and the flow hrough them may be assumed to be adiabatic, since their is neither ufficient time nor large enough area for any effective heat transfer to ke place. Also, there is no work done, and the change in potential ergy, if any, is very small. Even though the exit velocity is often onsiderably higher than the inlet velocity, in many cases, the ncrease in kinetic energy is insignificant. Then the conservation of nergy equation for this single stream steady flow device reduces to

$$h_2 \equiv h_1 \qquad (2.32)$$

To gain some insight into how throttling affects fluid properties, et us express equation (2.32) as follows

$$h_1 = h_2$$

$$u_1 + p_1 V_1 = u_2 + p_2 V_2$$

Thus the final outcome of a throttling process depends on which f the two quantities increases during the process. If the flow energy ncreases during the process ($p_2 V_2 >> p_1 V_1$), it can do so at the xpense of the internal energy. As a result, internal energy decreases, ich is usually accompanied by a drop in temperature. If the product V decreases, the internal energy and the temperature of a fluid will ncrease during a throttling process. In the case of ideal gas h = h (T), nd thus the temperature has to remain constant during a throttling rocess (Fig 2.13).

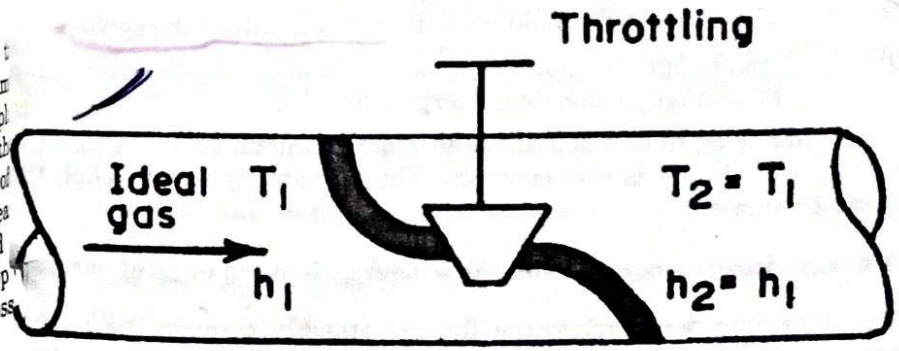

Fig. 2.13. Throttling process.

2.6.3. Adiabatic mixing

The mixing process is highly irreversible due to the amount of eddying and churning of the fluid that takes place. discussed in detail in the next article.

2.7. FIRST LAW ANALYSIS OF STEADY FLOW

A large number of engineering devices such as nozzles, turbi and compressors operate for long periods of time under the sa conditions and they are classified as steady flow devices.

Processes involving steady flow devices can be represen reasonably well by a somewhat idealized process, called the ste flow process. A steady flow process can be defined as a process dur which a fluid flows through a control volume steadily. That is, fluid properties can change from point to point within the con volume, but at any fixed point they remain the same during the en process (steady means no change with time).

2.7.1. Steady flow energy equation (SFEE)

Assumptions :

(1) Mass rate of flow into and out of the control volume equal and constant with respect to time; i.e.

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

(2) State of fluid at any given location does not vary with time

(3) Rates of work and heat transfer crosses the boundary constant.

(4) The mass of the fluid remain constant within the system.

(5) No change in chemical composition of the system and he no change in chemical energy is involved.

For the state of a substance to remain constant implies that energy stored by it is also constant. Thus for steady flow through control volume :

Energy entering control volume = Energy leaving control volum

Consider some substance flowing steadily through the cont volume of Fig 2.14. Let this control volume include devices wh effect heat and work transfers.

Specific energies	Energy in at 1–1	Energy out at 2–2
Work of introduction (Flow work or flow energy)	$p_1 V_1$	$p_2 V_2$
Internal energy	u_1	u_2
Kinetic energy	$\frac{C_1^2}{2}$	$\frac{C_2^2}{2}$
Potential energy	gZ_1	gZ_2
Heat transffered to the system	q	—
Energy transferred out of the system in the form of shaft work	—	w

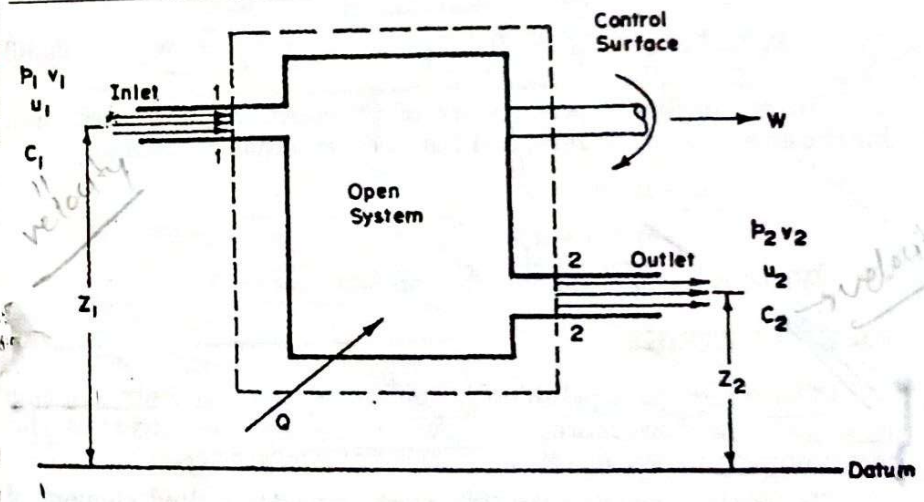

Fig. 2.14. Steady flow process.

Writing the steady flow energy equation for the flow of unit mass so that heat and work transfer, both are per unit mass. Then,

$$p_1 v_1 + u_1 + \frac{C_1^2}{2} + gZ_1 + q = p_2 v_2 + u_2 + \frac{C_2^2}{2} + gZ_2 + w \quad (2.33)$$

It is essential to observe consistency of units in substituting in the above equation.

Now, specific enthalpy $h = pv + u$

Therefore

$$h_1 + \frac{C_1^2}{2} + gZ_1 + q = h_2 + \frac{C_2^2}{2} + gZ_2 + w \quad (2.34)$$

Specific enthalpy can also be written as

$$h = c_p T$$

Therefore,

$$c_p T_1 + \frac{C_1^2}{2} + gZ_1 + q = c_p T_2 + \frac{C_2^2}{2} + gZ_2 + w \quad (2.3$$

For gases and vapours the variations of potential energy negligible.

Therefore,

$$p_1 v_1 + u_1 + \frac{C_1^2}{2} + q = p_2 v_2 + u_2 + \frac{C_2^2}{2} + w \quad (2.3$$

In a non flow process, work of introduction (Flow work) a kinetic energy terms is zero, and the above equation reduces to

$$u_1 + q = u_2 + w$$

$$q = w + u_2 - u_1$$ Non flow process

Which is the same as that for non flow process.

2.8. FLOW WORK

The energy required to push fluid into or out of a control volu is called the flow work, or flow energy and is necessary maintaining a continuous flow through a control volume.

To obtain a relation for flow work, consider a fluid element volume V as shown in Fig. 2.15(a). The fluid immediately upstrea will force this fluid element to enter the control volume; thus it can regarded as an imaginary piston. The fluid element can be chosen be sufficiently small so that it has uniform properties throughout.

If the fluid pressure is p and the cross-sectional area of the flu element is A (Fig. 2.15b), the force applied on the fluid element by t imaginary piston is

$$F = pA$$

To push the entire fluid element into the control volume, t force must act through a distance L. Thus the work done in pushi the fluid element across the boundary (i.e. the flow work) is -

$$W_{flow} = FL = pAL = pV \quad (kJ)$$

$$w_{flow} = pv \quad (kJ/kg) \quad (2.37)$$

The flow work relation is the same whether the fluid is pushed into our out of the control volume (Fig. 2.15c).

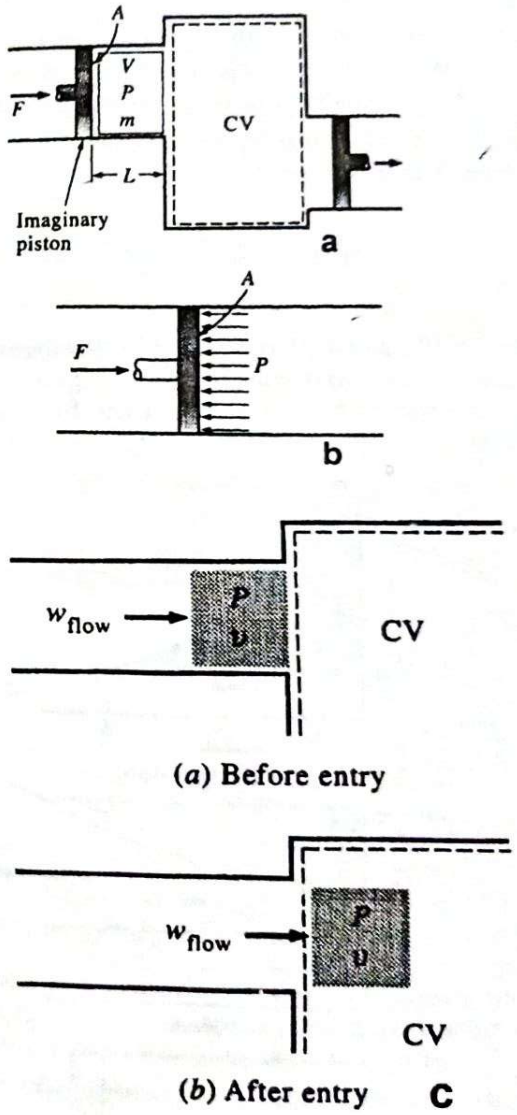

Fig. 2.15. Flow work

2.8.1. Work done in a flow process

A process undergone by a closed system of fixed mass is referred to as a non flow process. A change of state of substance taking place while flowing through a control volume in an open system is referred to as a flow process.

The work done in a non flow process in a closed system is called non flow work. The work done in a flow process in an open system is equal to the non flow work plus the flow work.

The non flow work done by the system in a piston cylinder arrangement from 1 to 2 is given by

$$W_{non\ flow} = \int_1^2 pdV$$

Now consider the same piston and cylinder arrangement to be with inlet and exhaust valves, and working as a engine. This becomes an open system. The process taking place on the fluid as shown in Fig. 2.16 are as follows :

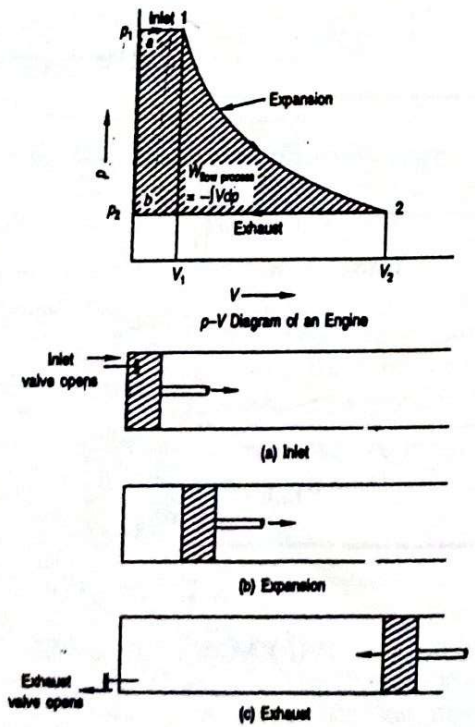

Fig. 2.16. Work done in a flow process.

(a) Inlet

$$W_{flow} = \int_a^1 pdV = p_1 V_1$$

(b) Expansion

$$W_{non\ flow} = \int_1^2 pdV$$

$$= \frac{p_1 V_1 - p_2 V_2}{n-1}$$

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(c) Exhaust

$$W_{flow} = \int_2^b pdV = -p_2 V_2$$

$$\therefore W_{flow\ process} = \int_a^1 pdV + \int_1^2 pdV + \int_2^b pdV$$

$$= p_1 V_1 + \frac{p_1 V_1 - p_2 V_2}{n-1} - p_2 V_2$$

$(p_1V_1 - p_2V_2)\left(\frac{n}{n-1}\right)$

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

= Non flow work + Flow work

= Area a - 1 - 2 - b - a

$$= -\int_1^2 Vdp$$

$$W_{FR} = -m\int_1^2 v\,dp$$ Area

$$W_{flow\ process} = (p_1 v_1 - p_2 v_2) + \int_1^2 pdv$$

$$= \int_1^2 v\,dp$$

In the case of a compressor, reverse operations would be taking place comprising of suction, compression and discharge as shown in Fig. 2.17 and work done would again be given by $-\int v\,dp$.

Work done for a steady flow process ($w = -\int v\,dp$).

From the first law of thermodynamics, we have

$$\delta q - \delta w = du$$

For a reversible process

$$\delta w = pdv$$

Hence, $\delta q - pdv = du$

or, $\delta q = pdv + du$ (2.3

Also, from the definition of enthalpy

$$h = u + pv$$

$$dh = du + d(pv)$$

$$= du + pdv + vdp$$ (2.3

From the steady flow energy equation (2.34), when changes kinetic and potential energies are neglected

$$q - w = \Delta h$$

In the differential form

$$\delta q - \delta w = dh$$

or, $\delta w = \delta q - dh$ (2.40

Substituting the values of δq and dh from equations (2.38) a (2.39) in equation (2.40)

$$\delta w = pdv + du - du - pdv - vdp$$

$$= -vdp$$

or, $w = -\int vdp$ (2.4

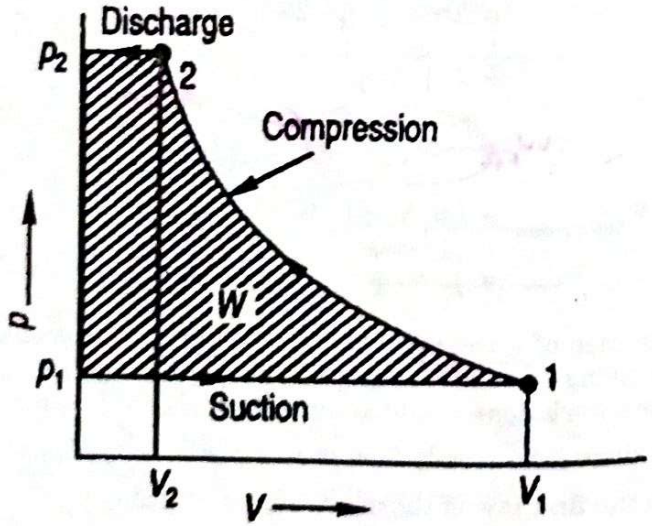


Fig. 2.17. Work done in a reciprocating compressor.

2.8.2. Non flow work versus flow work

In summary now we can say that the work done in a non flow process is given by

$$w_{non\ flow} = \int_1^2 pdv = \frac{p_1 v_1 - p_2 v_2}{n-1}$$

as shown by the integration of the p-v path along v-axis in Fig. 2.18(a). And the work done in a flow process is given by

$$w_{flow\ process} = -\int_1^2 v\,dp$$

$$= \frac{n}{n-1}\left(p_1 v_1 - p_2 v_2\right)$$

as shown by the integration of the path along p-axis in Fig. 2.18(b).

It must be pointed out here that it is not necessary for the machine to be a reciprocating piston and cylinder type only with suction and discharge valves. The expression $-\int v\,dp$ applies to all open system including rotary machines with or without valves such as turbine or a centrifugal or an axial flow compressors.

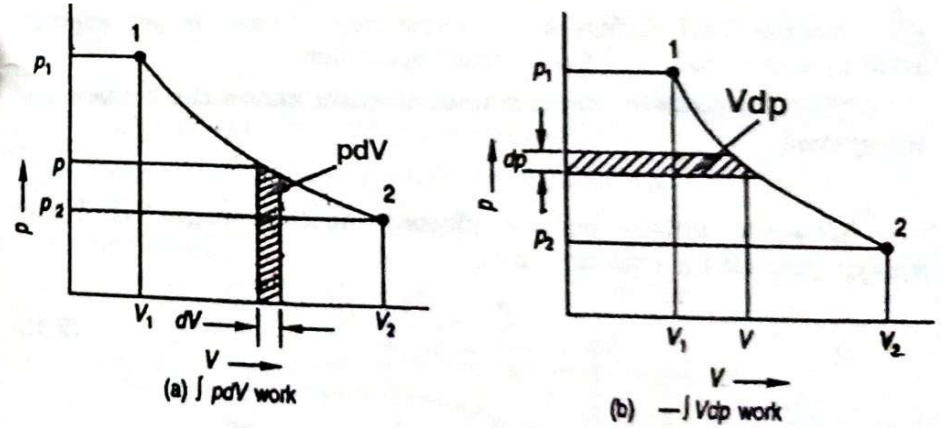


Fig. 2.18. Non flow work and flow work.

2.9. APPLICATION OF THE STEADY FLOW ENERGY EQUATION FOR ENGINEERING DEVICES

(a) Nozzles and diffusers :

A nozzle is a device that increases the velocity of a fluid at the expense of pressure. (Fig 2.19).

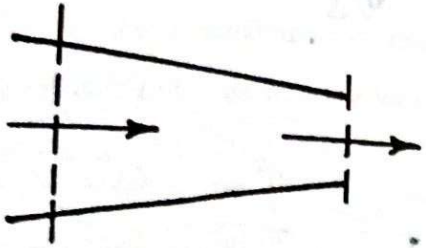


Fig. 2.19. Nozzle

A diffuser is a device that increases the pressure of a fluid by slowing it down (Fig 2.20).

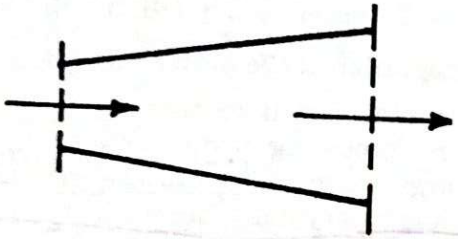


Fig. 2.20. Diffuser

Nozzles and diffusers are commonly utilized in jet engines rockets, space craft, and even garden sprinkler.

There is neither heat nor work transfer across the boundary of the system

$$q = w = 0$$

However, kinetic energy effects cannot be neglected. Hence applying the SFEE equation (2.41)

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2} \qquad (2.42)$$

or,

$$C_2^2 = C_1^2 + 2(h_1 - h_2)$$

For the nozzle $C_2 >> C_1$, the SFEE is given by

$$C_2^2 = 2(h_1 - h_2) \qquad (2.43)$$

For the diffuser $C_1 >> C_2$, the SFEE is given by

$$C_1^2 = 2(h_2 - h_1) \qquad (2.44)$$

If heat transfer is considering in the nozzle the SFEE is given by

$$h_1 = h_2 + \frac{C_2^2}{2} + q \qquad (2.45)$$

b) Turbine and compressor

A turbine is a device for extracting work from a flowing fluid expanding from a high pressure to low pressure (Fig 2.21).

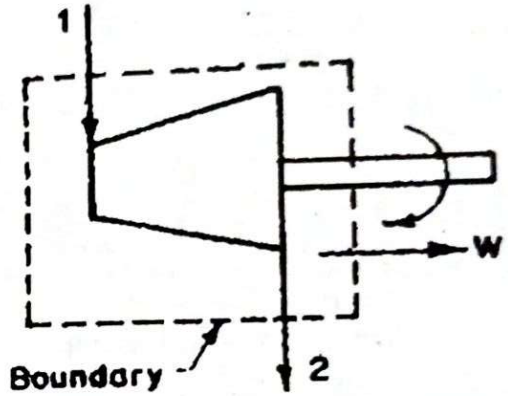


Fig. 2.21. Turbine

A rotary compressor may be regarded as a reversed turbine, doing work on a fluid to raise its pressure (Fig 2.22).

Assuming adiabatic flow for both the cases i.e. $q = 0$

The change in potential energy is usually very small and is normally neglected.

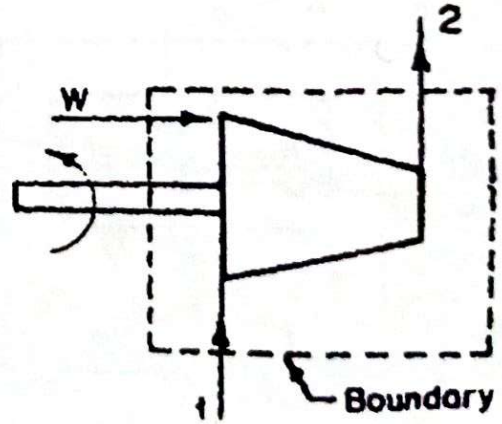


Fig. 2.22. Compressor

The change in velocity is usually very small relative change in enthalpy, and thus it is often disregarded.

For the turbine SFEE reduces in the form given below

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2} + w$$

For the compressor SFEE is given by

$$h_1 + \frac{C_1^2}{2} + w = h_2 + \frac{C_2^2}{2}$$

Considering the heat transfer for the turbine the SFE written as

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2} + w + q$$

and for the compressor

$$h_1 + \frac{C_1^2}{2} + w = h_2 + \frac{C_2^2}{2} + q$$

The velocities involved in the turbine and compr considered.

(c) Process involving heat transfer but no work transfer

Boilers, condensers and evaporators have as their main fun the transfer of heat to or from a steady flowing medium. Negle potential and kinetic energy effects and putting w = 0 (Figs. 2.23 and 2.25).

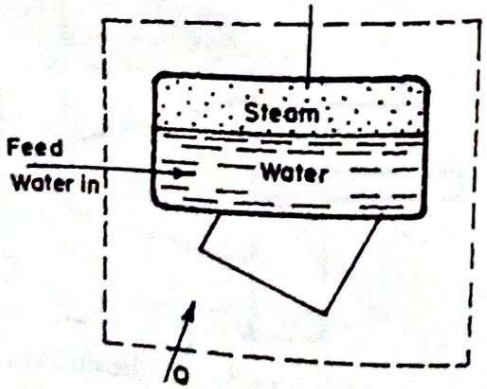


Fig. 2.23. Boiler

$$h_1 \pm q = h_2 \tag{2.50}$$

$$\pm q = h_2 - h_1$$

Positve sign for boiler and negative sign for condenser.

For boiler, $h_1 + q = h_2$ (2.51)

For condenser, $h_1 = h_2 + q$ (2.52)

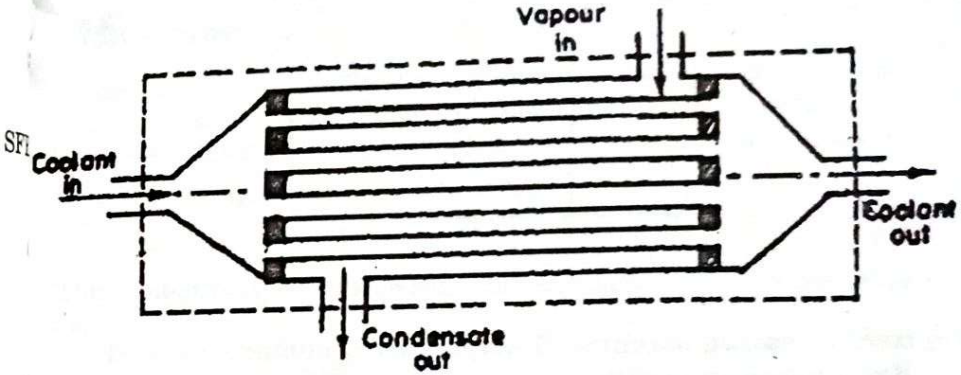


Fig. 2.24. Condenser.

Fig. 2.25 shows the evaporator of a refrigerating installation. Freon enters a coil (in contact with the air in a refrigerator cabinet) as a liquid and leaves as vapour. It is to be noted that this is the reverse of the process occuring in a condenser. Inlet and outlet velocities are small. There is no shaft work and no elevation change. The steady flow energy equation becomes simply.

$$h_1 + q = h_2 \tag{2.53}$$

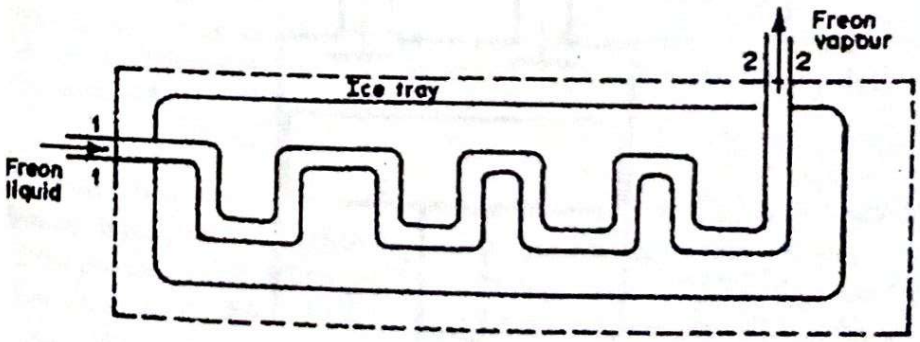


Fig. 2.25. Evaporator

(d) Insulated duct

Consider the adiabatic flow of a fluid in an insulated pipe o duct, as shown in Fig. 2.26. Assuming adiabatic process (q = 0 Potential energy term neglected. No work transfer from the system The SFEE yields.

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2} \quad (2.5$$

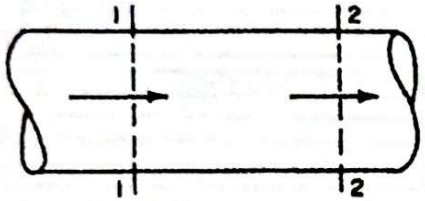


Fig. 2.26. Insulated duct.

(e) Reciprocating machine [Reciprocating compressor and Reciprocating engine]

A steady flow may be assumed for receiprocating machine (Fi 2.27). Some heat transfer, however, does occur in a reciprocatin machines because

(i) the relative low speed of flow and

(ii) the large cylinder wall area.

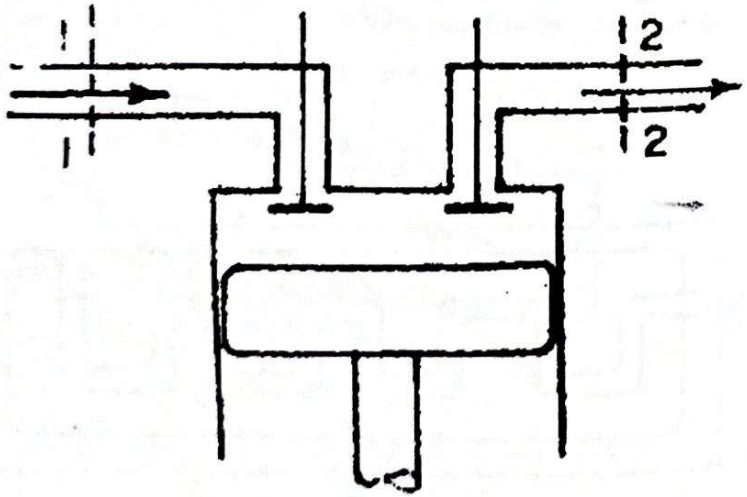


Fig. 2.27. Reciprocating machine

The SFEE for reciprocating machines is

$$h_1 + q = h_2 + w \quad (2.55)$$

Assuming a adiabatic flow.

$$h_1 = h_2 + w$$

$$\text{or, } w = h_1 - h_2 \quad (2.56)$$

Throttle valve

There is neither heat nor work transfer i.e. (q = w = 0) (see ticle 2.6.2). Furthermore, both the initial and final velocities are ually low, so that kinetic energy effects may be neglected (Fig. 2.28).

Hence applying SFEE equation (2.36).

$$h_1 = h_2 \quad (2.57)$$

Thus, inspite of pressure drop ($p_1 - p_2$) the enthalpy remains ıstant.

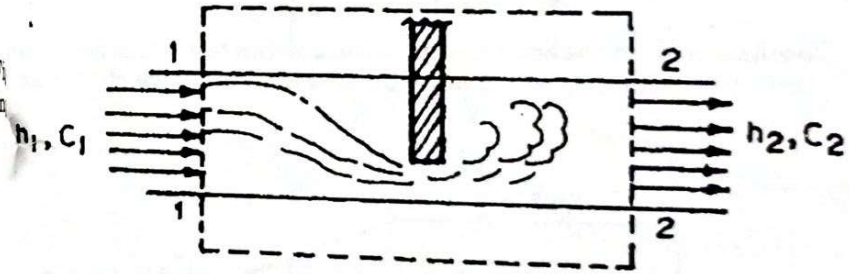


Fig. 2.28. Throttle valve

Adiabatic mixing

The conservation of mass principle for a mixing requires that the ı of the incoming mass flow rates equal the mass flow rate of the ;oing mixture.

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Consider two streams of fluid mixing as shown in Fig 2.29. Let 'treams have mass flow rate $\dot{m}_1$ and $\dot{m}_2$ and temperatures T_1 and ,et the resulting mixed stream have a temperature T_3. There is no flow to or from the fluid (q = 0), and no work is done (w = 0), e from the flow equation. We have, neglect changes in kinetic gy.

$$H_1 + H_2 = H_3$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3$$

$$\dot{m}_1 c_p T_1 + \dot{m}_2 c_p T_2 = \dot{m}_3 c_p T_3$$

$$\dot{m}_1 T_1 + \dot{m}_2 T_2 = (\dot{m}_1 + \dot{m}_2) T_3$$

Examples of this process is discuss below.

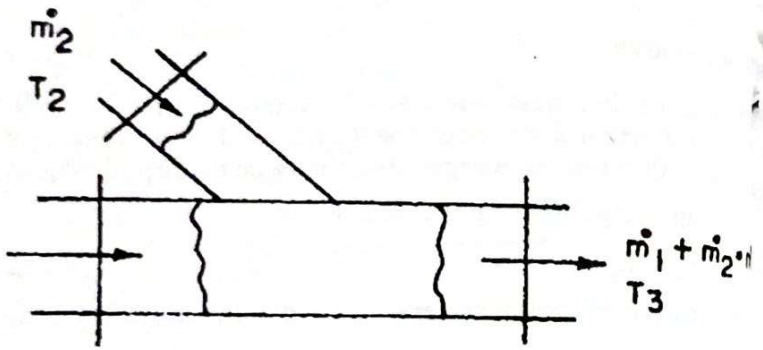


Fig. 2.29. Adiabatic mixing

(i) Combustion chamber : Imagine the combustion chamber of turbine or the furnace of a steam power plant to be shown 2.30

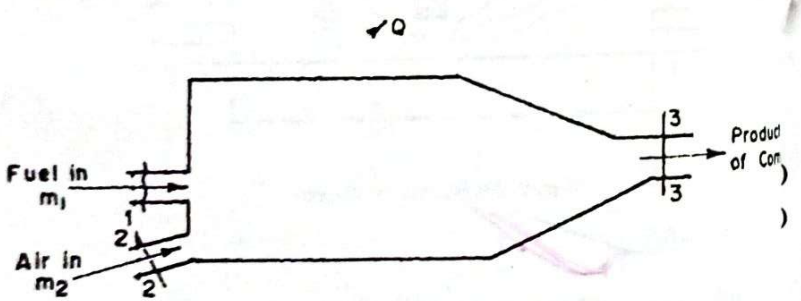


Fig. 2.30. Combustion chamber

Fuel enters at section 1 at a rate $\dot{m}_1$, air enters at section 2 at a rate $\dot{m}_2$, and combustion products leave at section 3 at a rate $\dot{m}_3$. The material balance is

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Neglecting velocity changes, the energy balance is

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_1 q_f = \dot{m}_3 h_3 + q$$

Where q_f is the chemical energy or heat released by the combustion of a unit mass of fuel. Equations (2.61) and (2.62) are to be solved simultaneously.

(ii) Dehumidifier : The effect of kinetic and potential energies are neglected. No work transfer. The energy balance in Fig 2.31 is

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 = \dot{m}_3 h_3 + q$$

The q is the heat removed.

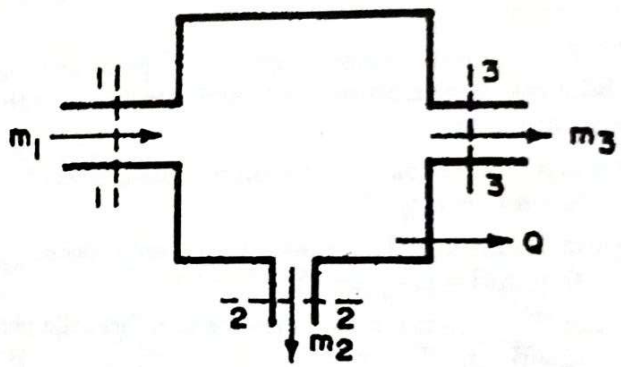


Fig. 2.31. Dehumidifier

POINTS TO NOTE

Work throughout with absolute pressures and temperatures.

In all cases assume given pressures to be absolute unless stated "gauge" or "vacuum" (negative gauge pressure).

Absolute pressure = Atmpospheric pressure ± Gauge pressure

Absolute temperature T = t°C + 273. (K)

Universal gas constant R_u = 8.314 kJ/kgmol K.

Universal gas constant and characteristic gas constant is related by a relation R_u = MR. where M is the molecular mass.

The characteristic gas equation pV=m RT. If R is in kJ/kg K then pressure must be converted from bar or MPa or Pa into kPa.

(7) The 1 bar = $\frac{100}{9.81}$ m of water ≈ 10.2 m of water = 750 m mercury must be used for conversion purposes.

(8) The atmospheric pressure is 1.01325 bar = 760 mm of 10.33 m of water.

(9) In the characteristic gas equation pV = m R T, mass should be invariably calculated.

(10) Work done and Heat transfer, Enthalpy etc. should be for calculated mass.

(11) It is customary to consider the work done during expansi be positive and hence the work done during compressio be negative.

(12) N.T.P. (normal temperature and pressure) and (standard temperature and pressure) are both use reference data.

(13) Q and Δ U are equal in constant volume process

$$Q = \Delta U = m\, c_v\, (T_2 - T_1)$$

(14) Q and ΔH are equal in constant pressure process.

$$Q = \Delta H = m\, c_p\, (T_2 - T_1)$$

(15) Q and W are equal in isothermal and hyberbolic process

$$Q = W = p_1 V_1 \ln r$$

(16) For polytropic process always use the basic equatio solving problems

$$Q = \frac{p_1 V_1 - p_2 V_2}{n-1} + m\, c_v\, (T_2 - T_1)$$

$$Q = \frac{m R (T_1 - T_2)}{n-1} + m\, c_v\, (T_2 - T_1)$$

(17) Use the relation for polytropic process

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

put n = γ for adiabatic process in the above equation.

(18) γ is a ratio of c_p / c_v

(19) The index "n" differs from the ratio of specific heats (γ).

ILLUSTRATIVE EXAMPLES

2.1. A perfect gas for which the ratio of specific heats is 1.4 occupies a volume of 0.06 m^3 at 10 bar and 500 K. The gas undergoes a expansion to 0.3 m^3. Find the heat absorbed or rejected by the gas for each of the following methods of expansion.

(a) Constant pressure; (b) Isothermal; (c) Hyperbolic
(d) According to the law $pV^{1.125}$=C and
(e) Adiabatic. For gas R = 0.25 kJ/kgK.

R = 0.0821 KJ

Solution :

$$p_1 V_1 = m R T_1$$

$$10 \times 100 \times 0.06 = m \times 0.25 \times 500 \quad \therefore \quad m = 0.48 \text{ kg}$$

$$c_v = \frac{R}{\gamma - 1} = \frac{0.25}{0.4} = 0.625 \text{ kJ/kg K}$$

$$c_p = R + c_v = 0.25 + 0.625 = 0.875 \text{ kJ/kg K}$$

(a) Constant pressure (p = Constant)

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$

$$\therefore T_2 = T_1 \times \frac{V_2}{V_1} = 500 \times \frac{0.3}{0.06} = 2500 \text{ K}$$

W & Q path fn

$$W_{12} = p\,(V_2 - V_1)$$

$$= 10 \times 100\,(0.3 - 0.06) = 240 \text{ kJ}$$

$Q = m c_p \Delta T$

ΔU = first law

$$\Delta U = m\, c_v\, (T_2 - T_1)$$

$$= 0.48 \times 0.625\,(2500 - 500) = 600 \text{ kJ}$$

$$\therefore Q_{12} = W_{12} + \Delta U$$ → point function

$$= 240 + 600 = 840 \text{ kJ}$$

Ans

[Check $Q_{12} = \Delta H = m\, c_p\, (T_2 - T_1)$

$$= 0.48 \times 0.875\,(2500 - 500) = 840 \text{ kJ}]$$

(b) Isothermal (T = Constant)

$$Q_{12} = W_{12} = p_1 V_1 \ln \frac{V_2}{V_1}$$

$$= 10 \times 100 \times 0.06 \ ln \ \frac{0.3}{0.06} = 96.56 \text{ kJ}$$

(c) Hyperbolic and Isothermal process are same in perfect gas.

(d) $pv^{1.25} = C, \quad p_1 V_1^{1.125} = p_2 V_2^{1.125}$

$$\therefore \quad p_2 = p_1 \left(\frac{V_1}{V_2}\right)^{1.125} = 10\left(\frac{0.06}{0.3}\right)^{1.125} = 1.635$$

$$T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{1.125-1}$$

$$= 500\left(\frac{0.06}{0.3}\right)^{0.125} = 408.88 \text{ K}$$

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{n-1}$$

$$= \frac{10 \times 100 \times 0.06 - 1.635 \times 100 \times 0.3}{1.125 - 1} = 87.47 \text{ kJ}$$

$$\Delta U = m \ c_v (T_2 - T_1)$$

$$= 0.48 \times 0.625 \ (408.88 - 500) = -27.33 \text{ kJ.}$$

$$Q_{12} = W_{12} + \Delta U$$

$$= 87.47 + (-27.33) = 60.14 \text{ kJ}$$

(e) Adiabatic process $\left(p V^{\gamma} = C\right)$

$$p_1 v_1^{\gamma} = p_2 v_2^{\gamma}$$

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^{\gamma}$$

$$= 10\left(\frac{0.06}{0.3}\right)^{0.4} = 1.05 \text{ bar}$$

$$T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{\gamma - 1}$$

$$= 500\left(\frac{0.06}{0.3}\right)^{.4} = 262.65 \text{ K}$$

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$$

$$= \frac{100(10 \times .06 - 1.05 \times 0.3)}{1.4 - 1} = 71.204 \text{ kJ}$$

$$\Delta U = m \ c_v (T_2 - T_1)$$

$$= 0.48 \times 0.625 \ (262.65 - 500) = -71.204 \text{ kJ}$$

$$Q_{12} = W_{12} + \Delta U$$

$$= 71.204 - 71.204 = 0$$ Ans.

2.2. 2.57 kg of a gas for which the ratio of specific heats is 1.35 occupies a volume of 0.15 m^3 at 10 bar and 500 K. The gas undergoes a expansion to 0.6 m^3. Determine the heat absorbed or rejected by the gas for each of the following methods of expansion.

(a) Constant pressure,

(b) Hyperbolic,

(c) According to the law $PV^{1.32}$=C and (d) Adiabatic.

Solution :

$$p_1 V_1 = m R T_1$$

$$10 \times 100 \times 0.15 = 2.57 \times R \times 500$$

$$\therefore R = 0.1167 \text{ kJ/kg K}$$

$$c_v = \frac{R}{\gamma - 1} = \frac{0.1167}{1.35 - 1} = 0.333 \text{ kJ/kg K}$$

(a) p = constant $\quad \frac{V_1}{T_1} = \frac{V_2}{T_2}$

$$\therefore T_2 = T_1 \frac{V_2}{V_1} = 500 \times \frac{0.6}{0.15} = 2000 \text{ K}$$

$$W_{12} = p (V_2 - V_1)$$

$$= 10 \times 100 (0.6 - 0.15) = 450 \text{ kJ}$$

$$\Delta U = m \ c_v (T_2 - T_1)$$

$$= 2.57 \times 0.333\ (2000-500) \quad = 1283.7 \text{ kJ}$$

$$Q_{12} = W_{12} + \Delta U$$

$$= 450 + 1283.7 \quad = 1733.7 \text{ kJ}$$

(b) $pV = C$

$$Q_{12} = W_{12} = p_1 V_1 \ ln \frac{0.6}{0.15}$$

$$= 10 \times 100 \times 0.15 \ ln \frac{0.6}{0.15}$$

$$= 207.94 \text{ kJ}$$

(c) According to $pV^{1.32} = C$

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

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^n$$

$$= 10 \left(\frac{0.15}{0.6}\right)^{1.32} = 1.605 \text{ bar}$$

$$T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{n-1}$$

$$= 500 \left(\frac{0.15}{0.6}\right)^{1.32-1} = 320.85 \text{ kJ}$$

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{n-1}$$

$$= \frac{100\ [10 \times 0.15 - 1.604 \times 0.6]}{1.32-1} = 168 \text{ kJ}$$

$$\Delta U = m\ c_v\ (T_2 - T_1)$$

$$= 2.57 \times 0.333\ (320.85 - 500) \quad = -153.31 \text{ kJ}$$

$$Q_{12} = W_{12} = \Delta U$$

$$= 167.94 - 153.54 \quad = 14.39 \text{ kJ}$$

(d) Adiabatic process $\left(p V^{\gamma} = C\right)$

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma} \qquad \text{Here, } \gamma = 1.35$$

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^{\gamma}$$

$$= 10 \left(\frac{0.15}{0.6}\right)^{1.35} = 1.5389 \text{ bar}$$

$$T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{\gamma-1} = 500 \left(\frac{0.15}{0.6}\right)^{1.35-1}$$

$$= 500\ (0.25)^{.35} \quad = 307.78 \text{ K}$$

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$$

$$= \frac{100\ [10 \times 0.15 - 1.5389 \times 0.6\ ]}{1.35-1} = 164.75 \text{ kJ}$$

$$\Delta U = m\ c_v\ (T_2 - T_1)$$

$$= 2.57 \times 0.333\ (307.78 - 500) \quad = -164.75 \text{ kJ}$$

$$Q_{12} = W_{12} + \Delta U$$

$$= 164.75 - 164.75 \quad = 0$$ Ans

2.3. Derive the following expressions for a polytropic process

(a) $W = \dfrac{m R T_1}{n-1}\left[1 - \left(r_p\right)^{\frac{n-1}{n}}\right]$ where $r_p = \dfrac{p_2}{p_1}$

(b) $W = \dfrac{p_1 V_1}{n-1}\left[1 - \dfrac{1}{(r_v)^{n-1}}\right]$ where $r_v = \dfrac{v_2}{v_1}$

(c) $Q = W\left[\dfrac{\gamma - n}{\gamma - 1}\right]$

Solution :

(a) We know that for a polytropic process

$$W = \frac{p_1 V_1 - p_2 V_2}{n-1}$$

$$= \frac{m R T_1 - m R T_2}{n-1} = \frac{m R T_1}{n-1}\left[1 - \frac{T_2}{T_1}\right]$$

We know that

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

and $p_1 V_1^n = p_2 V_2^n$

Therefore,

$$\frac{T_2}{T_1} = \frac{p_2 V_2}{p_1 V_1}$$

$$= \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}$$

$$= \left(\frac{p_2}{p_1}\right)\left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

Substituting in equation (1)

$$W = \frac{m R T_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$= \frac{m R T_1}{n-1}\left[1 - (r_p)^{\frac{n-1}{n}}\right]$$

(b) We know that for a polytropic process

$$W = \frac{p_1 V_1 - p_2 V_2}{n-1}$$

$$= \frac{p_1 V_1}{n-1}\left[1 - \frac{p_2 V_2}{p_1 V_1}\right]$$

$$W = \frac{p_1 V_1}{n-1}\left[1 - \frac{m R T_2}{m R T_1}\right] = \frac{p_1 V_1}{n-1}\left[1 - \frac{T_2}{T_1}\right]$$

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

$$= \frac{p_1 V_1}{n-1}\left[1 - \frac{1}{\left(\frac{V_2}{V_1}\right)^{n-1}}\right] = \frac{p_1 V_1}{n-1}\left[1 - \frac{1}{(r_v)^{n-1}}\right] \quad \text{Proved.}$$

(c) We know that a polytropic process for the non flow energy equation

$$Q = W + \Delta U$$

$$= \frac{p_1 V_1 - p_2 V_2}{n-1} + m c_v (T_2 - T_1)$$

We know that

$$c_v = \frac{R}{\gamma - 1} \quad \text{and} \quad p_1 v_1 = m R T_1$$

Substituting in equation (1)

$$Q = \frac{p_1 V_1 - p_2 V_2}{n-1} + \frac{m R}{\gamma - 1}(T_2 - T_1)$$

$$= \frac{p_1 V_1 - p_2 V_2}{n-1} - \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$$

$$= \frac{p_1 V_1 - p_2 V_2}{n-1}\left[1 - \frac{n-1}{\gamma - 1}\right]$$

$$= W\left[\frac{\gamma - n}{\gamma - 1}\right] \quad \text{Proved}$$

1.4 Nitrogen at 100°C and 6 bar expands in such a way that it can be approximated by a polytropic process with n = 1.25. Calculate the work done, the change in internal energy, the change in enthalpy and the heat transfer if the final pressure is 1.25 bar. Take $\gamma = 1.4$.

Solution :

$$R = \frac{R_u}{M} = \frac{8.314}{28} = 0.297 \text{ kJ/kg K}$$

$$c_v = \frac{R}{\gamma - 1} = \frac{0.297}{0.4} = 0.7425 \text{ kJ/kg K}$$

$$c_p = \gamma c_v = 1.4 \times 0.7425$$
$$= 1.0395 \text{ kJ/kg K}$$

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 373 \left(\frac{1.25}{6}\right)^{\frac{0.25}{1.25}}$$

$$= 373\,(0.73072) = 272.559 \text{ K}$$

$$p_1 v_1 = R\,T_1$$

m = 1 kg

$$600 \times v_1 = 0.297 \times 373$$

$$\therefore v_1 = 0.1846 \text{ m}^3/\text{kg}$$

$$\text{and } v_2 = \frac{R\,T_2}{p_2} = \frac{0.297 \times 272.559}{125} = 0.6476 \text{ m}^3/\text{kg}$$

$$w = \frac{p_1 v_1 - p_2 v_2}{n-1} = \frac{600 \times 0.1846 - 125 \times 0.6476}{0.25}$$

$$= \frac{110.76 - 80.95}{0.25} = \frac{29.81}{0.25} = 119.24 \text{ kJ/kg} \quad \text{Ans}$$

$$u_2 - u_1 = c_v\,(T_2 - T_1) = 0.7425\ (272.559 - 373)$$
$$= 0.7425 \times (100.441) = -74.577 \text{ kJ/kg} \quad \text{An}$$

$$q = w + (u_2 - u_1) = 119.24 - 74.577$$
$$= 44.6625 \text{ kJ/kg} \quad \text{An}$$

$$h_2 - h_1 = c_p\,(T_2 - T_1)$$
$$= 1.0395\ (272.559 - 373) = 1.0395\,(-100.441)$$
$$= -104.4084 \text{ kJ/kg} \quad \text{An}$$

2.5. An ideal gas with a molecular weight 6.5 kg/kgmol compressed in a reversible manner from 690 kPa and 277 to a final specific volume of 0.47 m^3/kg according to relatio $p = 561 + 200\,v + 100\,v^2$, where p is the pressure in kPa, v the specific volume in m^3/kg. The specific heat at consta volume is 0.887 kJ/kgK. Determine

(a) the work, (b) the heat,
(c) the final temperature, and (d) the initial specific volume.

Solution :

$$R_u = MR$$

$$\therefore R = \frac{R_u}{M} = \frac{8.3143}{6.5} = 1.279 \text{ kJ/kg K}$$

1 kg

$$p_1 v_1 = R\,T_1$$

$$\therefore v_1 = \frac{R\,T_1}{p_1} = \frac{1.279 \times 277}{690} = 0.5135 \text{ m}^3/\text{kg} \quad \text{Ans}$$

$$W_{12} = \int_1^2 p\,dv = \int [561 + 200\,v + 100\,v^2]\,dv$$

$$= \left[561\,v + \frac{200\,v^2}{2} + \frac{100\,v^3}{3}\right]_{v_1 = 5135}^{v_2 = 0.47} = -29.73 \text{ kJ/kg} \quad \text{Ans}$$

$$p_2 = 561 + 200\,v_2 + 100\,v^2$$
$$= 561 + 200\,(0.47) + 100\,(0.47)^2 = 677.1 \text{ k Pa}$$

$$p_2\,v_2 = R\,T_2 \therefore T_2 = \frac{p_2 v_2}{R} \quad \frac{677.1 \times .47}{1.279} = 248.8 \text{ K} \quad \text{Ans}$$

$$\Delta u = c_v\,(T_2 - T_1) = 0.837\,(248.8 - 277) = -23.6 \text{ kJ/kg}$$

$$q_{12} = w_{12} + \Delta u = -29.73 - 23.6 = -53.33 \text{ kJ/kg} \quad \text{Ans}$$

2.6. A certain fluid at 10 bar is contained in a cylinder behind a piston, the initial volume being 0.05 m^3. Calculate the work done by the fluid when it expands reversibly according to a law

$$p = \frac{A}{V^2} - \frac{B}{V}$$

to a final volume of 0.1 m^3 and a final pressure of 1 bar, where A and B are constants. **(JNVU BE I, 1987)**

Solution :

$$p = \frac{A}{V^2} - \frac{B}{V} \quad (1)$$

Initial condition

$$10 = \frac{A}{(0.05)^2} - \frac{B}{0.05}$$

$$10 = 400\,A - 20\,B$$

Final condition

$$1 = \frac{A}{(0.1)^2} - \frac{B}{0.1}$$

$$1 = 100\,A - 10B$$

Solving equations (2) and (3), we get

$$A = \frac{8}{200} \text{ and } B = \frac{6}{20}$$

Substituting in equation (1)

$$p = \frac{8}{200\,V^2} - \frac{6}{20\,V}$$

Therefore,

$$W = \int p\,dV$$

$$= \int \left(\frac{8}{200\,V^2} - \frac{6}{20\,V}\right) dV$$

$$= \frac{8}{200}\left[-\frac{1}{V}\right]_{0.05}^{0.1} - \frac{6}{20}\,[ln\,V]_{0.05}^{0.1}$$

$$= \frac{8}{200}\left(\frac{1}{0.1} - \frac{1}{0.05}\right) - \frac{6}{20}\,ln\left(\frac{0.1}{0.05}\right)$$

$$= \frac{-8}{200} \times (-10) - \frac{6}{20}\,(ln\,2)$$

$$= 0.4 - \frac{6}{20} \times (0.693)$$

$$= 0.4 - 0.2079 = 0.1921$$

$$= 0.1921 \times 10^5 \text{ NM} = 19210 \text{ J}$$

2.7. One kg of a certain fluid is contained in a cylinder at an initial pressure of 20 bar. The fluid is allowed to expand reversibly behind a piston according to a law pV^2 = constant until the volume is doubled. The fluid is then cooled reversibly at constant pressure until the piston regains its original position; heat is then supplied reversibly with the piston firmly locked in position until the pressure rises to the original value of 20 bar. Calculate the net work done by the fluid, for an initial volume of 0.05 m³. **[JNVU BE I, 1984]**

Solution :

Process 1 – 2

$$p_1 V_1^{\,2} = p_2 V_2^{\,2}$$

therefore,

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^2 \quad \frac{20}{4} = 5 \text{ bar}$$

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{n-1}$$

$$= \frac{20 \times 100 \times 0.05 - 5 \times 100 \times 0.1}{2-1}$$

$$= 100 - 50 \quad = 50 \text{ kJ}$$

$$W_{23} = p_2 \left(V_3 - V_2\right)$$

$$= 5 \times 100\,(0.05 - 0.1) \quad = -25 \text{ kJ}$$

$$W_{31} = 0$$

Net work done

$$W_{net} = W_{12} + W_{23} + W_{31}$$

$$= 50 - 25 + 0 \quad = 25 \text{ kJ}$$

Ans.

2.8. One kg of fluid expands reversibly according to a linear law from 4.2 bar to 1.4 bar. The initial and final volumes are 0.004 m³ and 0.02 m³ respectively. The fluid is then cooled reversibly at constant pressure and finally compressed reversibly according to a law pv = constant back to the initial condition of 4.2 bar and 0.004 m³. Calculate the work done in each process stating whether it is done on or by the fluid and calculate the net work of the cycle. Sketch the cycle on a p-v diagram. **[JNVU BE I, 1985]**

Solution :

Process 1 to 2 linear law (Fig. 2.32)

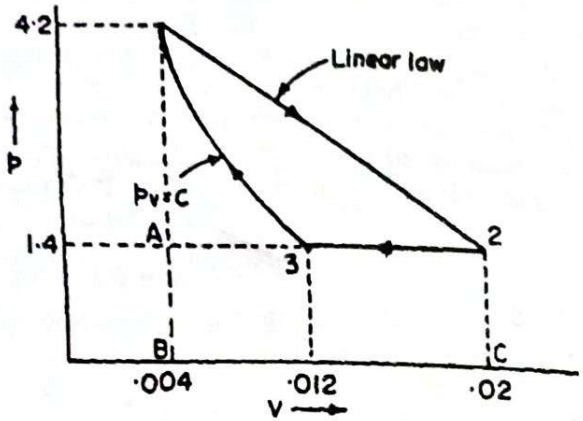


Fig. 2.32. p-V diagram P. 2.8.

W_{12} = Area under the line 1-2

= Area of triangle 1A2 + Area of rectangle 2 A B C

= Area of trapezium 1 B C 2

$$= \frac{\text{Sum of the parallel lines}}{2} \times \text{Height}$$

$$= \left(\frac{4.2 + 1.4}{2}\right)(0.02 - 0.004) \quad = 0.0448$$

= 4480 J [work done by the fluid]

Process 2-3 Constant pressure

$p_3 = p_2 = 1.4$ bar

V_3 is calculated from process 3-1

$$V_3 = \frac{p_1 V_1}{p_3} = \frac{4.2 \times 0.004}{1.4} = 0.012 \text{ m}^3$$

$$W_{2\text{-}3} = p_2 (V_3 - V_2)$$

$$= 1.4 \times 10^5 (0.012 - 0.02)$$

= – 1120 J [work done on the fluid]

$$W_{31} = p_1 V_1 \ln \frac{V_1}{V_3}$$

$$= 4.2 \times 10^5 \times 0.004 \ln \frac{0.004}{0.012}$$

= – 1845.6 J [work done on the fluid] Ans

Net work done $W = W_{12} + W_{23} + W_{31}$

$$= 4480 + (-1120) + (-1845.6)$$

= 1514.3 J Ans

2.9. A quantity of gas occupies a volume of 0.3 m^3 at a pressure of 1 bar and a temperature of 20°C. The gas is compressed isothermally to a pressure of 5 bar and then expanded adiabatically to its initial volume. Determine for this quantity of gas

(a) the heat received or rejected (state which) during the compression,

(b) the change of internal energy during the expansion,

(c) the mass of the gas.

Assume $\gamma = 1.4$ and $c_p = 1$ kJ/kg K.

Solution :

For process 1 to 2 (Fig. 2.33)

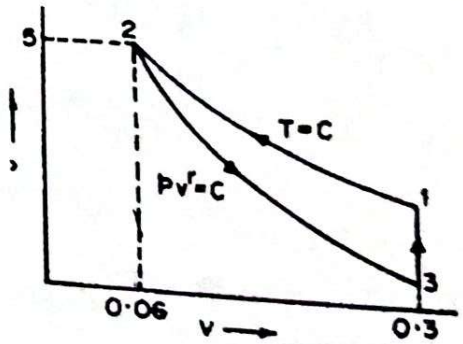


Fig. 2.33. p-V diagram P. 2.9.

$$p_1 V_1 = p_2 V_2$$

$$V_2 = V_1 \frac{p_1}{p_2}$$

$$= 0.3 \times \frac{1}{5} = 0.06 \text{ m}^3$$

$$Q = W = p_1 V_1 \ln r$$

$$= 1 \times 100 \times 0.3 \ln \frac{1}{5} = -48.3 \text{ kJ} \quad \text{[Rejected]}$$

For adibatic process 2 to 3

$$p_2 V_2^{\gamma} = p_3 V_3^{\gamma}$$

$$\therefore p_3 = p_2 \left(\frac{V_2}{V_3}\right)^{\gamma} = 5 \times \left(\frac{0.06}{0.3}\right)^{1.4} = 0.526 \text{ bar}$$

$$Q = W + \Delta U$$

$$0 = W + \Delta U$$

$$\Delta U = -W = -\frac{p_2 V_2 - p_3 V_3}{\gamma - 1} = \frac{p_3 V_3 - p_2 V_2}{\gamma - 1}$$

$$= \frac{5 \times 0.06 - 0.526 \times 0.3}{1.4 - 1}$$

$$= -35.5 \text{ kJ} \quad \text{(Loss of internal energy)}$$

We know that

$$c_p = \frac{R\gamma}{\gamma - 1}$$

$$R = \frac{c_p(\gamma - 1)}{\gamma} = \frac{1(1.4 - 1)}{1.4} = 0.286 \text{ kJ/kgK}$$

$$p_1 V_1 = m R T_1$$

$$1 \times 100 \times 0.3 = m \times 0.286 \times 293$$

$$m = 0.358 \text{ kg}$$

2.10. 0.09 m³ of a fluid at 0.7 bar are compressed reversibly pressure of 3.5 bar according to a law pv^n = Constant. fluid is then heated reversibly at a constant volume until pressure is 4 bar; the specific volume is then 0.5 m³/kg reversible expansion according to a law PV^2= const restores the fluid to its initial state. Calculate the mass fluid present, the value of n in the first process and the work done on or by the fluid in the cycle. Sketch the cycle a P-V diagram **[JNVU BE I, 1987 (ATKT), 1997 (ATK**

Solution :

For process 3 to 1 (Fig. 2.34)

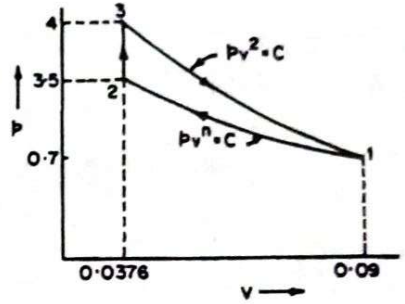


Fig. 2.34. p-V diagram P. 2.10.

$$p_3 V_3^2 = p_1 V_1^2$$

$$V_3 = \sqrt{\frac{p_1}{p_3}} \times V_1$$

$$= \sqrt{\frac{0.7}{4}} \times 0.09 \qquad = 0.0376 \text{ m}^3$$

$$m v_3 = V_3 \qquad m \times 0.5 = 0.0376$$

$$m = 0.0753 \text{ kg}$$ Ans.

For process 1 to 2

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

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^n$$

Taking log on both the sides

$$\log \frac{p_2}{p_1} = n \log \frac{V_1}{V_2}$$

$$\log \frac{3.5}{0.7} = n \log \frac{0.09}{0.0376}$$

$$n = 1.846 \approx 1.85$$ Ans

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{n-1} = \frac{0.7 \times 100 \times 0.09 - 3.5 \times 100 \times 0.037}{1.846 - 1}$$

$$= -8.108 \text{ kJ}$$

$$W_{23} = 0$$

$$W_{31} = \frac{p_3 V_3 - p_1 V_1}{n-1} = \frac{4 \times 100 \times 0.0376 - 0.7 \times 100 \times 0.09}{2-1}$$

$$= 8.74 \text{ kJ}$$

Net work done $= W_{12} + W_{23} + W_{31}$

$= -8.108 + 0 + 8.74 = 0.632$ kJ

2.11. 1 kg of Nitrogen undergoes the following sequence processes, an adiabatic expansion in which the volu doubles, a constant presure process in which the volume reduced to its initial value and a constant volu compression back to the initial state. The Nitrogen is initi at 150°C and 5 bar, calculate the net work done and net h transfer on the gas. Take $\gamma = 1.4$.

Solution :

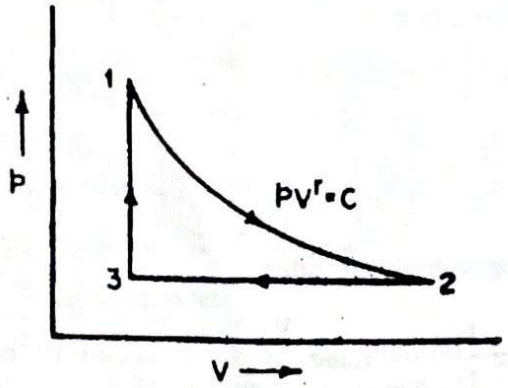


Fig. 2.35. p-V diagram P. 2.11.

$$R = \frac{R_u}{M} = \frac{8.314}{28} = 0.297 \text{ kJ/kg K}$$

$$c_p = \frac{R\gamma}{\gamma - 1} = \frac{0.297 \times 1.4}{0.4} = 1.0395 \text{ kJ/kg K}$$

$$c_v = \frac{R}{\gamma - 1} = \frac{0.297}{0.4} = 0.7425 \text{ kJ/kg K}$$

$$p_1 V_1 = m R T_1$$

$$5 \times 100\, V_1 = 1 \times 0.297 \times 423$$

$$V_1 = 0.251 \text{ m}^3 = V_3$$

$$V_2 = 2 V_1 = 2 \times 0.251 = 0.502 \text{ m}^3$$

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^{\gamma} = 5 \left(\frac{1}{2}\right)^{1.4} = 1.894 \text{ bar}$$

$$W_{12} = \frac{p_1 V_1 - p_2 V_2}{\gamma - 1} = \frac{5 \times 100 \times 0.251 - 1.894 \times 100 \times 0.502}{1.4 - 1}$$

$$= 76.053 \text{ kJ}$$

$$W_{23} = p_2 (V_3 - V_2) = 1.894 \times 100\,(0.251 - 0.502)$$

$$= -47.54 \text{ kJ}$$

$$W_{31} = 0$$

Net work

$$W = W_{12} + W_{23} + W_{31} = 76.053 - 47.54 + 0$$

$$= +28.5136 \text{ kJ}$$ Ans

$$T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = 423\,(0.5)^{0.4} = 320.6\text{K}$$

$$T_3 = T_2 \frac{V_3}{V_2} = 30.6\,(0.5) = 160.3 \text{ K}$$

$$Q_{12} = 0$$

$$Q_{23} = m\, c_p (T_3 - T_2)$$

$$= 1 \times 1.0395\,(160.3 - 320.6) = -166.63 \text{ kJ}$$

$$Q_{31} = m\, c_v (T_1 - T_3)$$

$$= 1 \times 0.7425\,(423 - 160.3) = 195.054 \text{ kJ}$$

Net heat

$$Q_{net} = Q_{12} + Q_{23} + Q_{31}$$

$$= 0 - 166.63 + 195.0524 = 28.424 \text{ kJ}$$ Ans

EXERCISES

A. Theory

2.1. Define first law of thermodynamics and explain the perp motion machine of the first kind.

2.2. State and explain the law of conservation of energy.

2.3. Define internal energy and prove that it is a property o system.

2.4. Write and explain three corollaries of the first la thermodynamics.

2.5. State the first law of thermodynamics and prove that non flow process, it leads to the energy equation Q = W + Explain the difference between total energy and int energy.

2.6. State the first law of thermodynamics for (a) a closed sy undergoing a cycle; (b) a closed system undergoing a pro (c) a steady flow process.

2.7. Show that the first law of thermodynamics requires th an ideal gas with a constant specific heat ratio c_p/c_v = undergoing a polytropic process ($pv^n = c$), that
(a) n must be greater than γ for $T_2 < T_1$ when there is transfer from the gas, and
(b) n must be less than γ for $T_2 < T_1$ when there is a transfer to the gas.

2.8. Show that polytropic specific heat is given by the expre $c_n = c_v\left(\frac{\gamma-n}{n-1}\right)$ where γ is the specific heat ratio.

2.9. Derive an expression for polytropic specific heat. What is significance of its negative sign.

2.10. Prove the relation $c_p - c_v = R$.

2.11. State and explain Jule's law of internal energy for an gas.

2.12. Prove that γ in pv^γ = Constant for the reversible adia expansion of a gas, is a ratio of specific heats at cons pressure and constant volume.

2.13. Derive the general energy equation for steady flow pro Explain the physical significance of the several terms o equations.

2.14. Derive an expression for emptying and filling process.

2.15. What is a non steady flow process ? Derive an expression for (a) filling a vessel, (b) emptying a vessel.

2.16. What is a throttling process ? What is the difference between free expansion and throttling process ?

2.17. State all the assumptions made for the steady flow process.

2.18. Derive the steady flow energy equation taking into account the change in the kinetic energy and Potential energy. State all the assumptions made.

2.19. Prove that for adiabatic process $pv^\gamma = c$.

2.20. Prove that the work interaction in a steady flow process is given by $-\int v\,dp$. State clearly the assumptions, if any.

2.21. Explain fully the significance of $\int_1^2 p\,dv$ in case of steady flow system and non flow system.

2.22. Prove that work done im a steady flow process is given by $\int_2^1 v\,dp$ when volume changes from v_1 to v_2 .

2.23. Let the entrance condition of an adiabatic nozzle be denoted by (1) and the exit condition be denoted by (2) For given p_1, v_1, C_1 and p_2 with fluid property given by $u = A + B\,pv$, show that

$$\frac{C_2^{\,2} - C_1^{\,2}}{2} = \frac{n\,p_1 v_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

where $n = \frac{1+B}{B}$

2.24. Show that the rate of change of heat transfer with volume for a gas having variable specific heats is given by

$$\frac{dQ}{dV} = p\left[(1-n)\frac{b+\alpha T}{a-b} + 1\right]$$

For a polytropic non flow process where $c_p = a + \alpha T$, $c_v = b + \alpha T$ and n is the polytropic index.

2.25. For a polytropic process, prove that

$$Q = \left(\frac{c_p - n\,c_v}{n-1}\right)(T_1 - T_2) \quad \text{or,} \quad Q = c_n\,(T_1 - T_2)$$

where c_n = Polytropic specific heat. **[JNVU BE I, 1988]**

2.26. Show that the rate of heat interchange per unit change of volume which occurs when a quantity of gas is compressed or expanded according to the law $pV^n = C$ is given by

$$\frac{dQ}{dV} = \frac{\gamma - n}{\gamma - 1}\, p$$ **[JNVU BE II (M...]**

2.27. Develop from first principles, the following expressions for polytropic process.

(a) $W = \dfrac{p_1 v_1}{n-1}\left[1 - \dfrac{1}{(r_v)^{n-1}}\right]$

(b) $\Delta H = \dfrac{\gamma}{\gamma - 1}(p_2 v_2 - p_1 v_1)$

(c) $\Delta u = \dfrac{1}{\gamma - 1}(p_2 v_2 - p_1 v_1)$

(d) $\dfrac{T_2}{T_1} = \left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \left(\dfrac{v_1}{v_2}\right)^{n-1}$

(e) $Q = \dfrac{\gamma - n}{1-n}\,\dfrac{mR}{\gamma - 1}(T_1 - T_2) = m\,c_n(T_2 - T_1)$

2.28. Show that when gas is compressed in a cylinder by a piston according to the law $PV^n = C$ = Constant, the heat rejected, increase in internal energy and work done in compression are in the ratio

$$(\gamma - n) : (n - 1) : (\gamma - 1)$$

2.29. The pressure and specific volume of a fluid are related by a law of the form $pv^\gamma = c$, and the internal energy is given by $u = a + b\,(pv)$, where a and b are constants. If the fluid expands in a non flow isentropic manner, determine the value of the index γ

$$\left[\gamma = \frac{b+1}{b}\right]$$ **[JNVU BE I...]**

2.30. (a) Explain $\int V\,dp$ for steady flow process. **[JNVU BE I, 1999, ...]**

(b) Apply steady flow energy equation to (i) Adiabatic process, (ii) Throttling process, (iii) Nozzle, (iv) Turbine, (v) Steam engine, (vi) Water pump and (vii) Any heat transfer equipment.

2.31. Prove the following for a closed system.

(a) For a constant volume process

$$Q = m\,c_v(T_2 - T_1) = \Delta U$$

(b) For a constant pressure process

$$Q = m\,c_p(T_2 - T_1) = \Delta H$$

(c) For a Hyperbolic process

$$Q = W = p_1 V_1 \ln \frac{V_2}{V_1}$$

(d) For an adiabatic process

$$W = \frac{p_1 V_1 - p_2 V_2}{\gamma - 1}$$

3. Unsolved Problems

2.1. A perfect gas for which the ratio of specific heats is 1.4, occupies a volume of 3.0 m^3 at 1.03 bar and 27°C. The gas undergoes a compression to 0.06 m^3. Find the heat absorbed or rejected by the gas for each of the following methods of compression (i) Constant pressure, (ii) hyperbolic (iii) Adiabatic. Take R = 0.287 kJ/kg K **[MREC, 2001]**

2.2. A fluid system undergoes a non flow frictionless process from $v_1 = 0.12\ m^3$ to $v_2 = 0.04\ m^3$ in accordance with

$$v = \frac{4.5}{P-2}$$

where p is in bar and v is in m^3. During the process, the fluid system rejects 40 kJ of heat, determine change in enthalpy and internal energy **[RU 2001]**

2.3. A cylinder contains 2 kg of working fluid. During the period the piston moves out, the temperature of the system falls from 95°C to 40°C and net heat rejected through the cylinder walls amounts to 20 kJ. The specific internal energy and temperature are related to each other by the equation

$$u = \left(u_o + \frac{t}{1.25}\right)\text{kJ}$$

where u_o is constant and t is temperature in °C. Estimate the work done. **[RU 2000]**

3

The Second Law of Thermodynamics and Entropy

3.1. INTRODUCTION

Chapter 2 considered the first law of thermodynamics and observed that Joule's experiment showed an equivalence between heat and work. This equivalence is quantitative and means only that heat and work are alternate methods of transferring energy. No indication was given of the relative values of heat and work.

In this chapter, we will first discuss the limitations of the first law and we introduce the second law of thermodynamics, which asserts that processes occur in a certain direction and that energy has quality as well as quantity. A process cannot take place unless it satisfies both the first and second laws of thermodynamics. Second law of thermodynamics, like the first law, is also base on observations in nature.

The first law of thermodynamics leads to the definition of a very useful property : stored energy, E. The second law also leads to the definition of a very useful property : entropy, S. unlike energy, the term entropy is not used in everyday conversation but this new property is very important because it is at the core of the second law of thermodynamics. This chapter also discusses several uses of entropy.

3.2. LIMITATIONS OF FIRST LAW OF THERMODYNAMICS

The first law is a versatile and valuable tool. However, there are energy conversion phenomena that cannot be explained using the first law alone. The first law expresses the relationship between heat and

work, and it allows us to define stored energy. But it does not allo to predict the extent of an energy conversion nor does it ind whether a proposed conceived energy process is possible. It ca written as :

(1) It does not give any information regarding the directio flow of heat and work. It does not give any condition u which these transfer takes place.

(2) The conversion of work into heat in a cyclic process may always be complete, the conversion of heat into work only be partial. In other words all work can be converte heat, but all heat cannot be converted into work.

3.3. DEFINITIONS AND TERMINOLOGY

In the following, the definitions and terminology to be use introducing the second law of thermodynamics are outlined.

(a) **Heat reservoir :** A body which is large enough or is b replenished continuously so that its temperature does change when heat is added or removed is a heat reservoi practice, large bodies of water such as oceans, lakes, and ri as well as the atmospheric air can be modeled accurate thermal energy storage capabilities of thermal masses.

(b) **Heat source :–** A reservoir that supplies energy in the form of heat is called a source (Fig. 3.1).

(c) **Heat sink :–** A reservoir that absorbs energy in the form heat is called a sink (Fig. 3.1).

(d) **Heat engine :–** A heat engine is a device that operates i cycle and produces net positive work while heat transfer oc across the boundaries of the device.

Heat engines differ considerably from one another, but all ca characterized by the following (Fig 3.2)

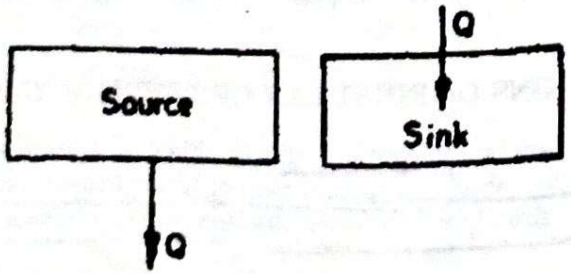


Fig. 3.1. Heat source and sink

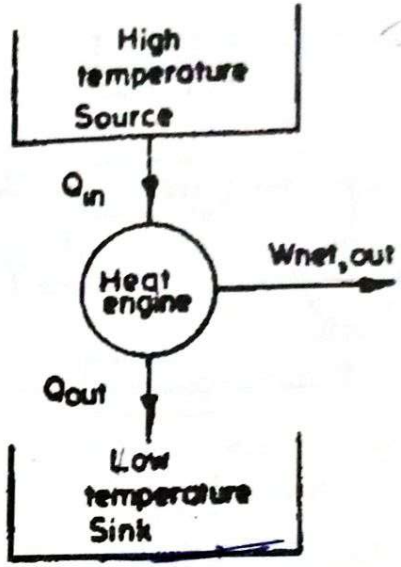


Fig. 3.2. Heat engine

(1) They receive heat from a high temperature source (nuclear reactor, solar energy, oil furnace, etc.).

(2) They convert part of this heat to work (usually in the form of a rotating shaft).

(3) They reject remaining waste heat to a low temperature sink (the atmosphere, rivers, etc).

(4) They operate on a cycle.

The heat engine is used in a broader sense as work producing device. The steam power plant device best fits into the definition of a heat engine. Heat transfer to the system occurs in the boiler work is produced by the turbine, heat transfer occurs from the system in the condenser, and a work input is required to compress the liquid to the operating pressure of the boiler. The schematic of a basic steam power plant is shown in Fig 3.3.

(e) Cycle efficiency or efficiency of a engine or thermal efficiency

The cycle efficiency of a heat engine is defined as a ratio of net work done to heat supplied. This is also known as thermal efficiency.

$$\eta = \frac{\text{Net work output}}{\text{Heat supplied}}$$

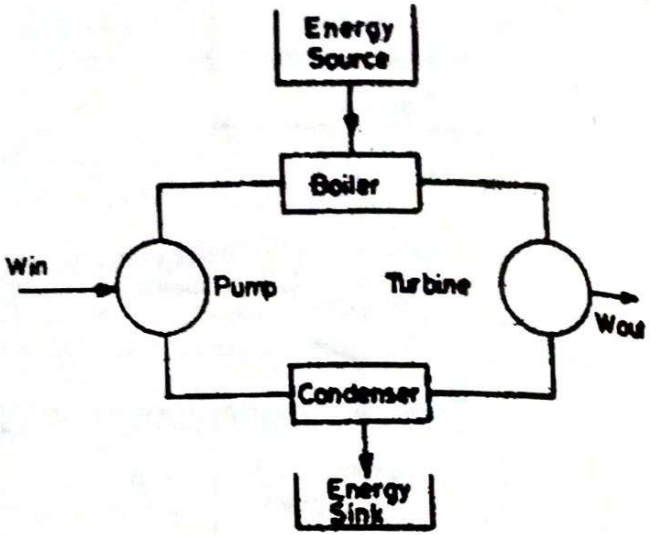


Fig. 3.3. Schematic of steam power plant

$$= \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}}$$

$$= \text{Thermal efficiency}$$

$$= \frac{Q_1 - Q_2}{Q_1} = \eta_{th}$$

3.4. THE SECOND LAW OF THERMODYNAMICS

Much experience, including many deliberate experiments, led to the second law of thermodynamics. This law has been stat many different forms, some of which appear at first to bear no rel whatsoever to each other. However, if any one of the statements c second law is accepted as a postulate, all the other statement ca proved from this starting point. None of the statements, though be derived from any other law of nature.

There are two classical statements of the second law, know the **Kelvin-Planck statement and the Clausius statement.**

3.4.1. Kelvin-Planck statement

It is impossible to construct a system which operate in cycle, extract heat from a single reservoir and do equivalent amount of work on the surroundings.

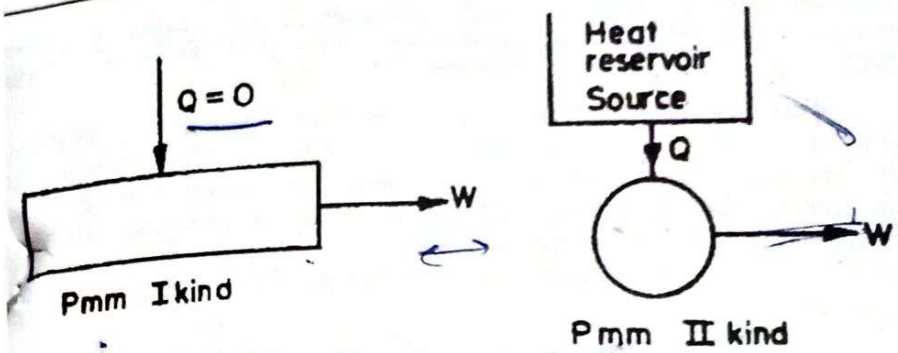


Fig. 3.4. Perpetnal motion machines

The above statement pertains more directly to heat engine.

Heat engine under ideal conditions, must reject some heat to a low temperature sink in order to complete the cycle. That is a heat engine must exchange heat with a low temperature sink as well as a high temperature source to keep operating. Thus, work can be done by the transfer of heat only if there are two temperature levels. This implies that it is impossible to build a heat engine that has a thermal efficiency of 100%. It is to be noted that the impossibility of having 100% efficient heat engine is not due to friction or other dissipative fects.

3.4.2. Clausius statement

It is impossible to construct a system which will operate in a cycle and transfer heat from a cooler to a hotter body without work being done on the system by the surroundings.

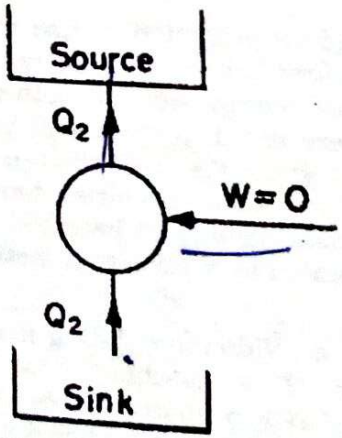


Fig. 3.5. Clausius statement.

In other words, heat cannot flow from a l temperature sink to a higher temperature source without help of an external work (Fig. 3.5).'

The Clausius statement does not imply that the cyclic d that transfers heat from a sink to a source is impossible to cons In fact, this is precisely what a common household refrigerator It simply states that a refrigerator will not operate unless compressor is driven by an external power source, such as an electri motor. This also implies that the coefficient of performance is al less than infinity.

Three observations should be made about these two stateme

(1) Both are negative statements.

(2) Impossibility of constructing a perpetual - motion mach the second kind.

(3) Both are equivalent.

The first observation is that both are negative statements of course impossible to prove a negative statement. However, we say that the second law of thermodynamics (like every other la nature) rests on experimental evidences. Every relevant experi that has been conducted either directly or indirectly verifies second law, and no experiment has ever been conducted contradicts the second law. The basis of the second law is therefo experimental evidence.

3.5 PERPETUAL MOTION MACHINES

Any perpetual motion machines may be classified as one of kinds.

(a) **PMM I kind** : A perpetual motion machine of the first violates the first law. Such a machine, shown in Fig 3 would create energy out of nothing. In other wo machine operating in a cycle and producing a greate work output than the net amount of heat put int machine. Many such machines have been propose several of them have been patented, but none of actually operated as a perpetual motion machine, know kind. ent.

(b) **PMM II kind** : Without violating the first law, to conceive of a machine that would r ate in continuously from a single source, and cor d do into work. Such a machine is shown in Fig

The machine receives heat from a single heat reservoir and converts all of the heat into work. It does not create energy, but, as a heat engine, it would convert heat entirely into work and would have an efficiency of 100%. It would, however, violate the second law of thermodynamics; therefore, it is called a perpetual motion machine of the second kind.

PMM III kind : A perpetual motion machine of the third kind would have no friction, and thus would run indefinitely but produce no work.

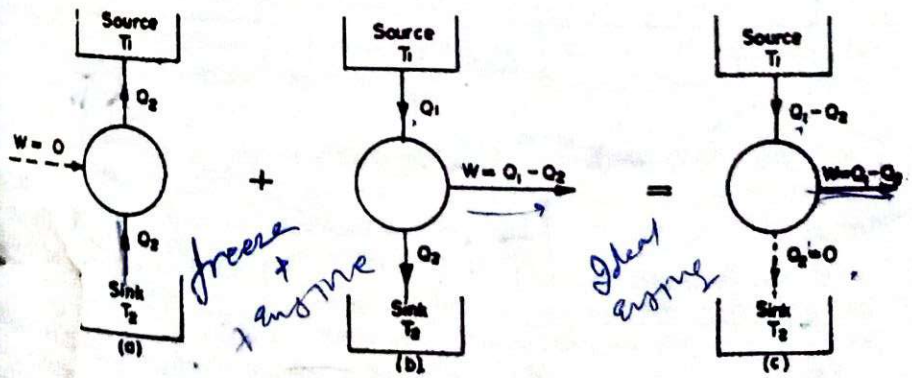
3.6 EQUIVALENCE OF THE TWO STATEMENTS

Any device that violates the Kelvin-Planck statement also violates the Clausius statement, and vice versa. This can be demonstrated as follows :

3.6.1. Violation of Clausius statement leading to violation of Kelvin-Planck statement.

Assume, contrary to the corollary, that a system can be constructed which will operate in a cycle and will transfer heat from a low temperature reservoir (sink) to a high temperature reservoir (source) without receiving work from the surroundings. This is shown in Fig 3.6 (a).

Construct a heat engine which operates between the two reservoirs, and takes in Q_1 from high temperature reservoir, delivers $(Q_1 - Q_2)$ of work to the surroundings, and rejects Q_2 to the lower temperature reservoir. This is shown in Fig 3.6 (b). The heat engine is legitimate, and violates no law.



.6. Violation of Clausius statement leads to a violation of Kelvin-Planck statement

Fig 3.6 (a) and Fig 3.6 (b) coupled together. The Q_2 transfer into and out of the lower temperature reservoir cancel each other, there is no net heat transfer at T_2. Fig 3.6 (a) and Fig 3.6 (b) coupled together constitute a device which now operates continuously from single heat reservoir at T_1, and which converts heat completely into work. The result is shown in Fig. 3.6 (c), but this is a violation of second law (in the form of a Kelvin-Planck statement), hence violation of the Clausius statement implies the violation of Kelvin-Planck statement.

3.6.2. Violation of Kelvin-Planck statement leading to violation of Clausius statement.

Assume, contrary to Kelvin-Planck's statement, that it possible to have an engine which operates from a single heat reservoir and completely converts the heat it receives into work. This is shown in Fig 3.7 (a), where a heat engine takes in Q_1 from the heat reservoir (source) at temperature T_1, and converts this into $W = Q_1$ of work (the thermal efficiency of 100%).

This work is now supplied to a refrigerator that removes heat the amount of Q_2 from the low temperature reservoir at T_2 and rejects heat in the amount of $Q_1 + Q_2$ to the high temperature reservoir at shown in Fig 3.7 (b). The refrigerator is perfectly legitimate, violates no law.

Now, let the heat engine in Fig 3.7 (a) be coupled with the refrigerator in Fig 3.7 (b) by having the output work of the engine drive the refrigerator. Thus, the combination of these two devices can be viewed as a refrigerator, as shown in Fig 3.7 (c), that transfers heat

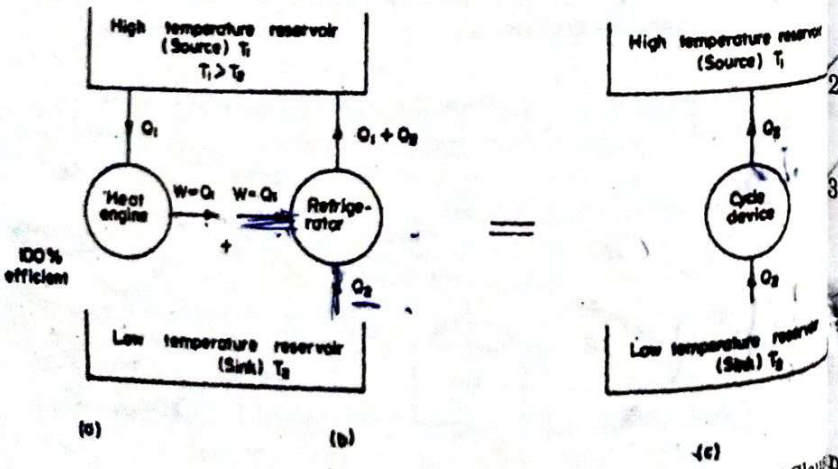


Fig. 3.7. Violation of Kelvin-Planck statement leads to a violation of Clausius statement

in an amount of Q_2 from the lower temperature to a higher temperature one without requiring any input from outside. This is clearly a violation of the Clausius statement. Therefore, a violation of the Kelvin-Planck's statement results in the violation of the Clausius statement.

It is seen that a violation of the Kelvin-Planck statement leads to a violation of the Clausius statement, and a violation of the Clausius statement leads to a violation of the Kelvin-Planck statement. The two statements are therefore completely equivalent.

3.7. REVERSIBLE AND IRREVERSIBLE PROCESSES

The second law renders certain processes impossible. The processes that are possible fall into two categories : reversible processes which bring about changes in a system and its surroundings that can be completely erased; irreversible processes which involve changes in the system and its surroundings that are permanent. These processes are defined as follows :

A process is reversible with respect to a system and its surroundings if the system and the surroundings can be completely restored to their respective initial state by reversing the direction of the process. A process is irreversible if it does not fulfil these condition.

3.7.1. Characteristics of reversible and irreversible processes

1) A reversible process must be such that, after it has occured, the system and the surroundings can be made to traverse, in the reverse order, the states they passed through during the original process. All energy transformations of the original process would be unchanged in form or magnitude.

2) The direction of a reversible process can be changed by making infinitesimal changes in the conditions that control it.

3) During a reversible process, the system and the surroundings must each at all times be in states of equilibrium or infinitesimally close to states of equilibrium; that is, the process must be a qusi-equilibrium one.

4) A reversible process must involve no friction, unrestrained expansion, mixing, heat transfer across a finite temperature difference, or inelastic deformation.

A reversible process must meet each of the conditions listed. If any one of the conditions is not met, the process is irreversible. Thus

we can apply these conditions to test any process for reve without having to go through the process of reasoning all t from a second law statement.

3.7.2. Examples of reversible and irreversible processes

Isothermal and adiabtic processes are the only processe are truely reversible. But even adiabatic process becomes irre when internal friction due to friction between the molecul cylinder surface or between molecules and molecules are ge An example of this is turbines and rotary compressors where friction due to high speed.

All actual processes are irreversible. Reversible processe occur. Nevertheless, reversible processes are extremely use serve as standards of comparison since they are often the l case of actual processes. Many actual processes are difficult to completely; therefore, an engineer must frequently base an ana design on reversible processes and then adjust the results t them to actual processes. This is similar to the use of ideal sy the analysis or design of actual systems.

The criterion of irreversibility based on the second law follows :

A process is irreversible if a perpetual motion machine second kind results from its being reversible.

The concept of reversible process is also important bec permits the definition of a highly useful property, entropy.

Reversible processes alone can be truely represented diagrams. In practice thermodynamic reversibility can be appr but cannot be achieved. Reversible process provides approximation for the evaluation of actual processes.

The number of irreversible processes commonly encount engineering are listed below.

(i) Relative motion with friction.

(ii) Free expansion of gases.

(iii) Combustion processes.

(iv) Diffusion of two unlike fluids into each other.

(v) Plastic deformation.

(vi) Shock wave in gas flow.

(vii) Hydraulic jump in water flow.

The concept of reversiblity is a powerful one that allows an pper limit for the efficiency of performance of heat engines and efrigerators to be established. Such a concept allowed Sadi Carnot to evelop the Carnot cycle, which turned out to be an important oundation for the eventual formulation of the second law of hermodynamics.

CARNOT CYCLE

Imaginary

A) The Carnot cycle is a hypothetical cycle developed by Carnot for either a heat engine or a reversed heat engine.

B) The Carnot cycle is a reversible cycle because it consists of two isothermal and two adiabatic processes.

C) All the processes involved in the cycle are reversible, thereby providing the best possible device that one could construct.

D) Results from the cycle analysis can be used to determine the maximum efficiency of performance possible for either a heat engine or reversed heat engine.

The following assumptions will be used for the derivation of the fficiency of a Carnot cycle applied to a non-flow cycle, i.e. one carried ut in a closed system, such as a cylinder sealed by a frictionless ston.

a) A hot source, i.e. heat reservoir of infinite capacity at the maximum constant temperature T_1.

b) A cold sink, i.e. heat reservoir of infinite capacity at the minimum constant temperature T_2.

c) Working fluid is perfect gas.

d) A frictionless piston sliding in a perfectly sealed cylinder, both piston and cylinder being perfect insulators.

e) A perfectly conducting cylinder head.

f) Since all processes have to be fully reversible, it follows that they must be performed infinitely slowly.

Carnot recognized that the conditions for converting heat into ork with maximum efficiency were as follows.

1) The cycle must consist of a series of processes each of which reversible with regard to both internal and external actions.

2) All heat received occurs at the maximum cycle temperature and all heat rejected occurs at the minimum cycle temperature.

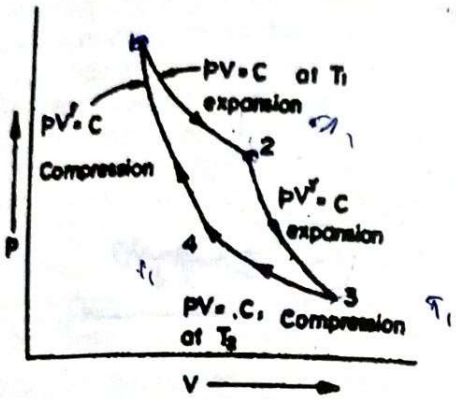


Fig. 3.8. Carnot cycle

For the analysis of Carnot cycle applied to a heat e consider the cylinder to contain one kg of air (gas) at temperat (Fig. 3.8).

Process 1-2

Isothermal expansion of the working fluid at the temperat the source T_1 accompanied by heat absorption from the source.

Process 2-3

Adiabatic expansion with temperature drop of working from T_1 to T_2, the temperature of the sink.

Process 3-4

Isothermal ~~expansion~~ compression at the temperature of the sin accompanied by the heat rejected to the sink.

Process 4-1

Adiabatic compression with temperature rise of working from T_2 to T_1.

The four processes applied for unit mass of working fluid become.

1-2 Isothermal heat supplied.

$$Q_{12} = R\,T_1\,ln\,\frac{v_2}{v_1}$$

2-3 Adiabatic expansion.

$$\frac{T_1}{T_2} = \left(\frac{v_3}{v_2}\right)^{\gamma-1} \tag{3.2}$$

3-4 Isothermal heat rejection.

$$Q_{34} = R\,T_2\,ln\,\frac{v_3}{v_4} \tag{3.3}$$

4-1 Adiabatic compression.

$$\frac{T_1}{T_2} = \left(\frac{v_4}{v_1}\right)^{\gamma-1} \tag{3.4}$$

From equations (3.2) and (3.4)

$$\frac{v_3}{v_2} = \frac{v_4}{v_1}$$

$$\text{or,}\quad \frac{v_3}{v_4} = \frac{v_2}{v_1} \tag{3.5}$$

Putting in equation (3.3)

$$Q_{34} = R\,T_2\,ln\,\frac{v_2}{v_1} \tag{3.6}$$

But the efficiency of the Carnot cycle can be written as

$$\eta_{Carnot} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat Supplied}}$$

$$= \frac{R\,T_1\,ln\,\frac{v_2}{v_1} - R\,T_2\,ln\,\frac{v_2}{v_1}}{R\,T_1\,ln\,\frac{v_2}{v_1}}$$

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

3.8.1. Limitations of Carnot cycle

(1) Adiabatic process can only be achieved if the piston mov very fast so that heat transfer is negligible due to short time available (Q =0). Isothermal process can only be achieve the piston moves very slowly to allow heat transfer so temperature remains constant (T=C).

The Carnot cycle consists of two isothermal and two adiaba For a stroke length one isothermal and one adiabatic process wi means that for part of the stroke the piston should move very slo and remaining part very fast.

This means that variable speed in a stroke is not possible.

(2) The work obtained per cycle is very small and may not able to overcome the friction of the reciprocating parts.

3.9. HEAT PUMPS, REFRIGERATORS, AND AIR CONDITIONERS

When a heat engine is run thermodynamically backwar becomes a heat pump, a refrigerator or an air conditioner, depend on your point of view. Fig 3.9 shows that when a heat engin thermodynamically reversed the directions of all the energy flows reversed. Thus, a work input W_{in} will cause a thermal energy tran Q_2 from a low temperature reservoir, and a thermal energy tran Q_1 to a high temperature reservoir. Consequently, the backw running heat engine appears to "pump" heat from a low tempera reservoir to a high temperature reservoir. However, since hea really a thermal energy transport phenomenon and not a fluid, somewhat misleading to refer to it as being "pumped". Yet common practice in the heating, ventilating and air conditio (HVAC) industry to refer to these devices as heat pumps when are used to provide a thermal energy transfer to a warm environ from a cold environment.

$$\eta_{HP} = \frac{Q_1}{Q_1 - Q_2} = \frac{\text{Desired energy result}}{\text{Required energy input}}$$

The energy conversion efficiency of a heat pump is al greater than 100%. No basic laws are violated here; this is simpl

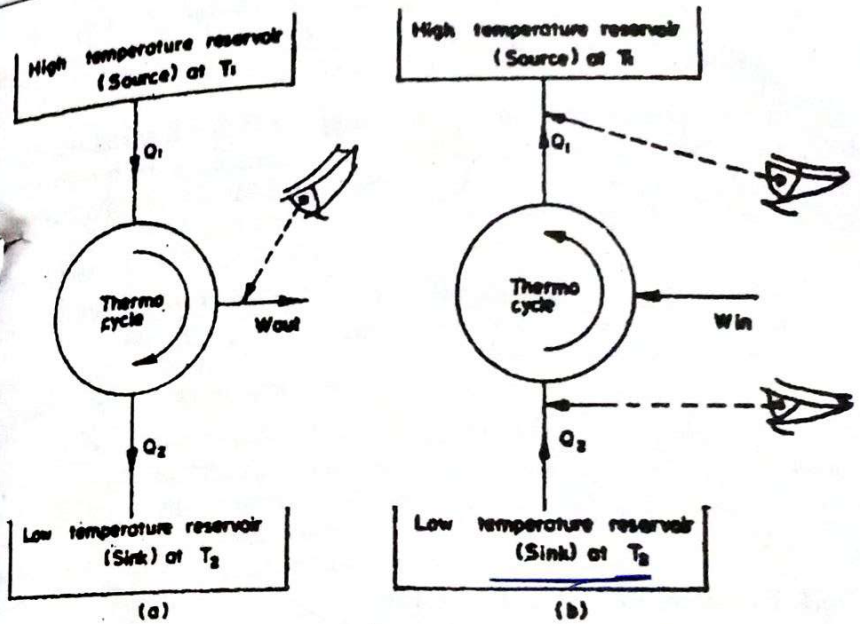


Fig. 3.9. (a) Heat engine (b) thermodynamically reversed heat engine (heat pump, refrigerator, air conditioner)

sult of the way in which the general efficiency formula is structured. It is true that in the case of a heat pump you get more thermal energy out of it than energy put into it. This makes a heat pump much more attractive for domestic heating than, say, a purely resistive electrical heater. Electrical heaters convert all of their input electrical energy directly into thermal energy and therefore have energy conversion efficiencies of 100%, whereas most heat pumps have energy conversion efficiencies far in excess of 100% for the same electrical energy input.

To eliminate the confusion that would arise in the general public by quoting efficiencies in excess of 100%, the HVAC industry uses the less suggestive phrase coefficient of performance (COP) to express this concept. The COP is simply the pure efficiency number before it is converted into a percentage (i.e. the COP of a heat pump with an energy conversion efficiency of 350% is 3.5).

$$COP_{HP} = \frac{\text{Desired heating effect}}{\text{Requried energy input}}$$

$$COP_{HP} = \frac{Q_1}{Q_1 - Q_2} \qquad (3.8)$$

$$COP_{HP} = \frac{T_1}{T_1 - T_2}$$

If the removal of heat Q_2 from a space is the desired result, backward running heat engine, then the engine is called a refrigerator when food is stored in the cooled space and an air conditioner when people occupy the cooled space.

$$COP_R = \frac{\text{Desired Refrigerating effect (Cooling effect)}}{\text{Required energy input}}$$

$$= \frac{Q_2}{Q_1 - Q_2}$$

$$COP_R = \frac{T_2}{T_1 - T_2}$$

3.9.1. Relation between two COPs

$$COP_{HP} = \frac{T_1}{T_1 - T_2}$$

$$= \frac{T_1 - T_2 + T_2}{T_1 - T_2}$$

$$= \frac{T_1 - T_2}{T_1 - T_2} + \frac{T_2}{T_1 - T_2}$$

$$= 1 + \frac{T_2}{T_1 - T_2}$$

$$= 1 + COP_R$$

Therefore, $COP_{HP} = COP_R + 1$

3.9.2. Actual and reversible refrigerators

No refrigerator can have a higher COP than a reversible refrigerator operating between the same temperature limits.

The COP of actual and reversible refrigerators operating between the same temperature limits can be compared as follows:

Reversible Carnot	Actual refrigerator	Remarks	
COP_R	> COP	Irreversible (Possible)	
	= COP	Reversible	(3.13)
	< COP	Impossible	

A similar relation can be obtained for heat pumps by replacing all values of $COP_{R(actual)}$ in equation (3.13) by $COP_{HP\,(actual)}$.

The COP of a reversible refrigerator or heat pump is the maximum theoretical value for the specified temperature limis. Actual refrigerators or heat pumps may approach these values as their design are improved, but they can never reach them.

As a final note, the COP of both the refrigerators and the heat pumps decrease as T_2 decreases. That is, it requires more work to absorb heat from lower temperature media. As the temperature of the refrigerated space approaches zero, the amount of work required to produce a finite amount of refrigeration approaches infinity and COP_R approaches zero.

3.10. INEQUALITY OF CLAUSIUS

In many real cycles, heat is received and rejected during processes that involve a continuous change in the temperature of the working fluid. Such a cycle can be considered reversible if the sources and sinks are assumed to consist of an infinite number of reservoirs that differ infinitesimally from one another in temperature. At any instant of time during the heating or cooling processes, heat is exchanged from the system and the source or sink with only on infinitesimal difference in temperature between the reservoir and the fluid in the system. This is equivalent to considering a heat engine that is operating between many different reservoirs. A corollary of the second law known as the Clausius inequity (or Inequility of Clausius) deals with such a heat engine operating between many different reservoirs. This corollary can be stated as follows :

'Whenever a system undergoes a cycle, the integral around the cycle of $\delta Q/T$ is less than zero for an irreversible cycle, and equal to zero for a reversible cycle'. In the mathematical form expressed as

$$\oint \frac{\delta Q}{T} \leq 0$$

Inequility of Clausius has been derived by using Carnot principles. Carnot principle is a consequence of the second law, thus Inequility of Clausius is also regarded as a corollary of the second law.

Note : It is to be noted that the Inequility of Clausius pro test for the reversibility of a cycle. If

$\int \frac{\delta Q}{T} = 0$, the cycle is reversible

$\int \frac{\delta Q}{T} < 0$, it is irreversible and possible.

$\int \frac{\delta Q}{T} > 0$, the cycle is impossible because it violate Kelvin-Planck statement of the second la thermodynamics.

3.11. ENTROPY

Inequility of Clausius discussed in the article 3.10 form basis for the definition of a new property called entropy. Entro derived concept in the same way that internal energy for a syste derived concept. The existence of entropy turns out to be espe useful in the analysis of non cyclic processes. The entropy extensive property.

According to Clausius definition, the integral of δQ a reversible process defines a change in a thermodyn property. This property is called entropy and is nor designated by S.

Since entropy is a thermodynamic property, a small chan entropy is designated by the exact differential dS. This quanti be expressed as

$$dS = \left(\frac{\delta Q}{T}\right)_{rev}$$

The change in entropy for a system as it undergoes a ch state from 1 to 2 can be obtained by integrating equation (3.24 any thermodynamic path for a reversible process to yield

$$S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T}\right)_{rev}$$

To integrate $\delta Q/T$, the mathematical relationship betwe and T must be known for the reversible process chosen.

The specific entropy is written as

$$s_2 - s_1 = \frac{1}{m}\int_1^2 \left(\frac{\delta Q}{T}\right)_{rev}$$

Be careful to note that equations (3.14) to (3.16), which entropy, are for a closed system of fixed mass m only. The su "rev" is a reminder that equations (3.14) to (3.16) hold true reversible cycle and a reversible process respectively.

'There exists a property (denoted by S) of a system such that a change in its value is equal to

$$S_2 - S_1 = \int_1^2 \frac{\delta Q}{T}$$

For any reversible process undergone by the system between states 1 and 2. This property is called entropy.'

Let a system undergo a reversible process from state 1 to state 2 along path A, and let the cycle be completed along path B, which is also reversible. (Fig 3.10).

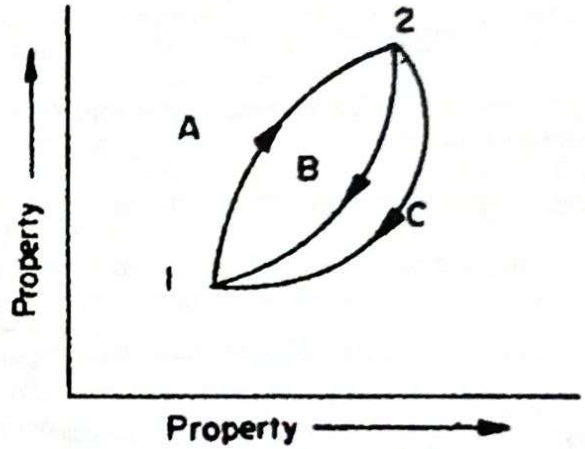


Fig. 3.10. Entropy is a property of a substance

Since this is a reversible cycle we can write for 1-A-2-B-1

$$\oint \left(\frac{\delta Q}{T}\right)_{rev} = 0 = \int_1^2 \left(\frac{\delta Q}{T}\right)_A + \int_2^1 \left(\frac{\delta Q}{T}\right)_B \quad (3.17)$$

Now, consider another cycle, which has the same initial process, but let the cycle be completed along path C. For this cycle (1-A-2-C-1) we can write

$$\oint \left(\frac{\delta Q}{T}\right)_{rev} = 0 = \int_1^2 \left(\frac{\delta Q}{T}\right)_A + \int_2^1 \left(\frac{\delta Q}{T}\right)_C \quad (3.18)$$

Subtracting the equation (3.18) from the equation (3.17), we have

$$\int_2^1 \left(\frac{dQ}{T}\right)_B = \int_2^1 \left(\frac{dQ}{T}\right)_C$$

Since the ($\delta Q/T$) is the same for all reversible paths bet states 2 and 1, we conclude that this quantity is independent of path and is a function of the end states only, and is therefo property. This property is called entropy.

Equation (3.15) was developed for a reversible process, change in entropy between state 1 and state 2 must be the s whether the process is reversible or irreversible, in view of the that entropy is a property. The definition given by the equation (3.14) i.e. incremental change of entropy (dS) for a system in terms of δQ for a reversible process, however, provides a theoretical basis fo computation of changes of the property between specified equilib states. For an irreversible process, the mathematical relatio between δQ and T, if any, does not provide any basis to determine property change except that the amount of change of the propert must be greater than δQ/T.

3.12. TEMPERATURE ENTROPY DIAGRAM

The $\frac{\delta Q}{T} = 0$ for a reversibles adiabatic process. For any reversible process $\frac{\delta Q}{T} \neq 0$ (Polytropic process)

This result can be applid to all working substances.

$$ds = \frac{\delta Q}{T}$$

or, $$\delta Q = T\,ds$$

or for any reversible process

$$Q = \int_1^2 T\,ds$$

This equation is analogous to the work equation for reversible process.

$$W = \int_1^2 p\,dv$$

Thus, as there is a diagram on which areas represent work in a reversible process, there is also a diagram on which represent heat flow in a reversible process. These diagrams are p–v and the T-s diagrams respectively, as shown in Figs 3.11(a) 3.11(b). For a reversible process 1-2 in Fig 3.11(a), the shaded $\int_1^2 p\,dv$, represents work done; for a reversible process 1-2 in 3.11(b), the shaded area $\int_1^2 T\,ds$, represents heat flow. Therefore great use of the property entropy is that it enables a diagram drawn on which areas represent heat flow in a reversible process the next section the T-s diagram will be considered.

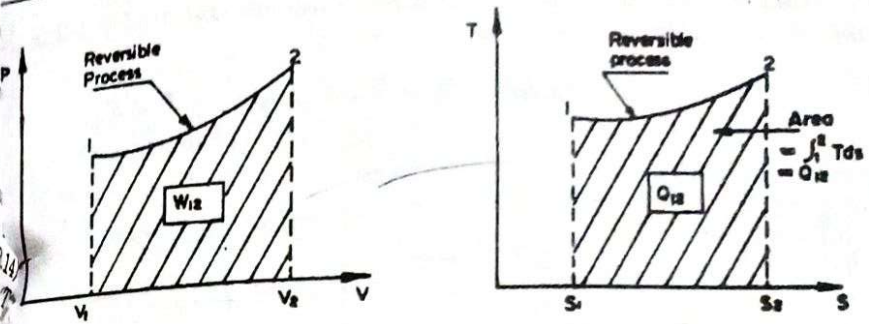


Fig. 3.11. Anology between work done and heat transfer reversible process.

If a path is drawn between two end states as a full curve, the rocess connecting these points is understood to be reversible. If a ath represented by a dashed line, then it is understood that the rocess between the end state is irreversible. In such a case, the areas nder the curve have no significance as to work and heat. The reason r this distinction is that the system, in going from state (1) to state) irreversible, may pass through some equilibrium states which are sconnected, in contrast to a reversible path where in all of its points e equilibrium points, differing, say in pressure and temperature, ly infinitesimally from the surroundings.

For a cyclic transformation the first law gives $\oint \delta W = \oint \delta Q$. nder reversible conditions, this becomes.

$$\oint p\,dv = \oint T\,ds \qquad (3.21)$$

Equation (3.21) states that the area within the closed curve on e p-v diagram equals that within the corresponding closed curve on e T-s diagram when both areas are measured in the same units.

3.12.1. Carnot cycle on T-s diagram

Carnot cycle up to here, it has only represented on the p-v gram. With the help of the definition of entropy it is represented on th p-v and T-s diagrams. The isothermal processes 1-2 and 3-4 are, course, represented by horizontal lines on the T-s diagram. As for reversible adiabatics, δQ = 0 (Tds), and, since T ≠. 0, ds = 0. Thus, ersible adiabatic processes are also isentropic (constant entropy) ocesses. Fig 3.12 depicts the Carnot cycle on both the p-v and T-s grams.

The efficiency of the Carnot cycle is most easily obtained with help of T-s diagram as compared to the p–v diagram. Area

enclosed within 1-2-3-4 is the net work, whereas area under 1-2 heat added to the engine. Thus,

$$\eta = \frac{\oint \delta W}{Q} = \frac{(T_1 - T_2)\ (s_2 - s_1)}{T_1\ (s_2 - s_1)} = \frac{T_1 - T_2}{T_1}$$

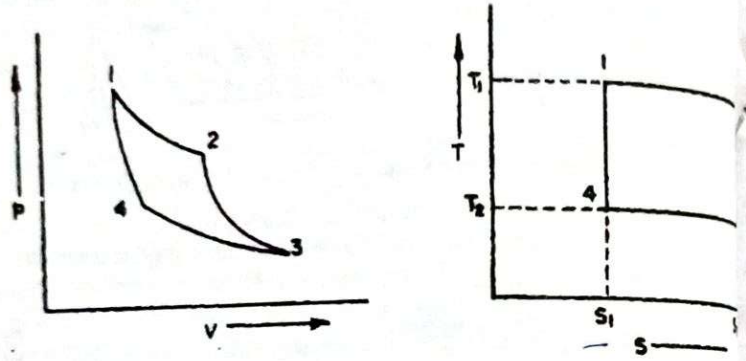


Fig. 3.12. Carnot cycle on p-V and T-s diagram.

3.13. ISENTROPIC PROCESS

There are three mechanisms which can cause the entropy system to change : heat transfer, mass flow, and irreversibilities.

Irreversibilities such as friction, fast expansion or compr and heat transfer through a finite temperature difference cause the entropy to increase. Thus the entropy of a closed cannot decrease during an adiabatic process. Entropy gen during a process is due to irreversibilities, and S_{gen} = reversible process.

If a process involves no heat transfer (adiabatic) irreversibilities within the system (internally reversible), the of a fixed mass must remain constant during that process. process is called an internally reversible adiabatic or is process.

3.14. THE ENTROPY CHANGE OF IDEAL GASES

An expression for the entropy change of an ideal gas obtained by employing the property relations for ideal gases.

$$du = c_v\, dT,$$

$$dh = c_p\, dT,$$

$$p = \frac{RT}{v}$$

and $c_p - c_v = R$ in the non flow energy equation. From the first law of thermodynamics.

$$\delta q = pdv + c_v\, dT$$

According to definition

$$ds = \frac{\delta q}{T} = \frac{pdv}{T} + \frac{c_v dT}{T}$$

$$ds = R\frac{dv}{v} + c_v\frac{dT}{T} \qquad (3.22)$$

The entropy change for a process is obtained by integrating this relation (3.22) between the end states

$$s_2 - s_1 = R\, ln\frac{v_2}{v_1} + c_v\, ln\frac{T_2}{T_1} \qquad (3.23)$$

Also, using $\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}$ in the equation (3.41)

$$s_2 - s_1 = R\, ln\frac{v_2}{v_1} + c_v\, ln\frac{p_2 v_2}{p_1 v_1}$$

$$= R\, ln\frac{v_2}{v_1} + c_v\, ln\frac{p_2}{p_1} + c_v\, ln\frac{v_2}{v_1}$$

$$= (R + c_v)\, ln\frac{v_2}{v_1} + c_v\, ln\frac{p_2}{p_1}$$

$$\Delta s = c_p\, ln\frac{v_2}{v_1} + c_v\, ln\frac{p_2}{p_1} \qquad (3.24)$$

Eliminating v_2/v_1 from equation (3.24) a new another relation for polytropic process can be obtained as

$$s_2 - s_1 = c_p\, ln\frac{T_2}{T_1} - R\, ln\frac{p_2}{p_1} \qquad (3.25)$$

The equations (3.23), (3.24) and (3.25) are used for polytropic process ($pv^n = c$).

3.14.1. Constant volume process (V = C)

Putting $ln\frac{v_2}{v_1} = 0$

The general equation becomes

$$s_2 - s_1 = c_v \, ln \frac{T_2}{T_1} = c_v \, ln \frac{p_2}{p_1}$$

3.14.2. Constant pressure process (p = C)

$$\text{Putting, } ln \frac{p_2}{p_1} = 0$$

The general equation becomes

$$s_2 - s_1 = c_p \, ln \frac{T_2}{T_1} = c_p \, ln \frac{v_2}{v_1}$$

Note :- The slope of the lines v = constant in the T-s diagr steeper than that of the lines p = constant (Fig 3.13 (a) and (b)).

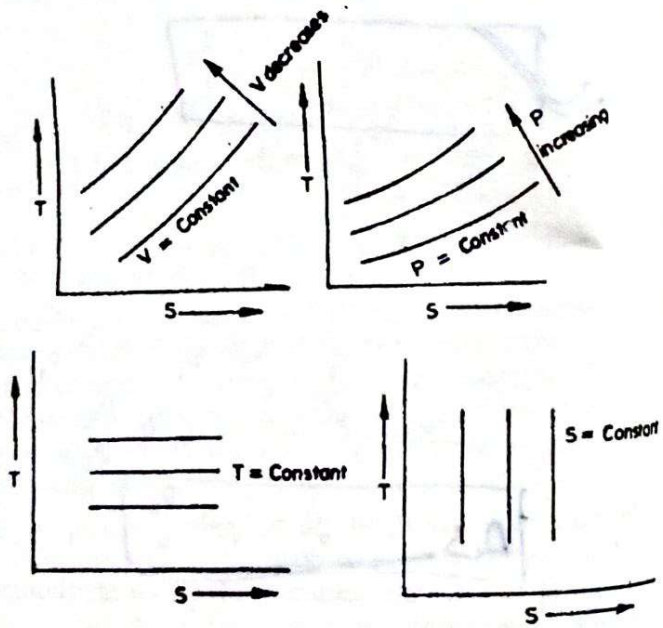


Fig. 3.13. Processes on T-s diagram. (a) Constant volume process, (b) Constant pressure process (c) Constant temperature process. (d) Isentropic process.

Consider the T-ds relation involving change in volume.

$$Tds = pdv + du$$

For the constant volume process dv = 0. Further, for the gas, $du = c_v \, dT$

Hence, we get

$$TdS = c_v \, dT$$

and for the slope of the constant volume line, we get

$$\left(\frac{dT}{ds}\right)_v = \frac{T}{c_v}$$

Now consider the T-ds relation involving change in pressure, i.e.,

$$Tds = dh - vdp$$

$h = pv + u$

For the constant pressure process dp = 0. Further, for the ideal gas, $dh = c_p \, dT$

Hence, we have

$$Tds = c_p \, dT$$

and for the slope of the constant pressure line, we get

$$\left(\frac{dT}{ds}\right)_p = \frac{T}{c_p}$$

Since $c_p > c_v$, we see that the slope of the constant volume line is steeper than the slope of the constant pressure line.

3.13. Constant temperature process (T = C)

The general equation becomes

$$\text{Putting, } ln \frac{T_2}{T_1} = 0$$

The general equation becomes

$$s_2 - s_1 = R \, ln \frac{v_2}{v_1} = R \, ln \frac{p_1}{p_2} \qquad (3.28)$$

Also, $\delta q = Tds$

$$\text{or,} \quad s_2 - s_1 = \int_1^2 \frac{\delta q}{T} \qquad (3.29)$$

Note : Isothermal are horizontal lines in the T-s diagram; [Fig.3.13(c)].

3.14. Adiabatic process (s = Constant)

Put $s_2 - s_1 = 0$ in the any of the general equation (3.23), (3.24) and (3.25)

$$O = c_p \ln \frac{v_2}{v_1} + c_v \ln \frac{p_2}{p_1}$$

or, $pv^{\gamma} = c$

The isentropic or adiabatic (reversible) lines are vert [Fig 3.13 (d)].

POINTS TO NOTE

(1) Entropy calculations must be made with absolute te i.e. Kelvin (°C+273).

(2) Read the question carefuly to see if the entropy required for one kg or m kg.

(3) All proofs are dependent upon the basic relationship

(4) All the equations given in the text give the change per kg of gas for a reversible process.

(5) The student very often has initial trouble with the ca use $\log_{10}$ or $\log_e = \ln$. Use in the change in entropy rela

(6) Use the basic equation for polytropic process. [(3.23), (3.25)].

(7) In reversible adiabatic expansion or compressi entropy remains constant.

(8) COP is defined for refrigerator, air conditioner and h In refrigeration cycle the object is the abstraction of a cold body whereas in heat pump cycle the obje delivery of heat to a body at higher temperature.

(9) Efficiency of a reversible cycle is maximum.

(10) COP of a reversible cycle is maximum.

ILLUSTRATIVE EXAMPLES

3.1. A power cycle operating between two reservoir energy Q_1 by heat transfer from a hot reservoir at T and rejects energy Q_2 by heat transfer to a cold re T_2 = 400K. For each of the following cases determine the cycle operates reversibly, irreversibly, or is impo

(a) Q_1 = 1000 kJ, W = 850 kJ

(b) Q_1 = 2000 kJ, Q_2 = 400 kJ

(c) W = 1600 kJ, Q_2 = 500 kJ

(d) Q_1 = 1000 kJ, η = 30 %

ution:

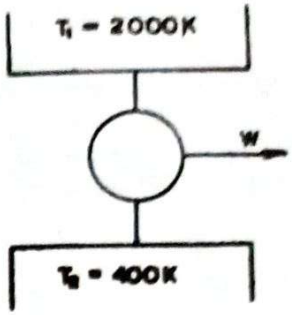


Fig. 3.14. Power cycle P. 3.1.

(a) $\eta_{max} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}$

$\eta_{max} = 1 - \frac{400}{2000} = 0.8 = 80\ \%$

Using the given data

$\eta = \frac{W}{Q_1} = \frac{850}{1000} = 0.85 = 85\ \%$

$\eta > n_{max}$ (Impossible) Ans

(b) $\eta = \frac{W}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = \frac{2000 - 400}{200} = 0.8 = 80\ \%$

$\eta = \eta_{max}$ (Reversible) Ans.

(c) $\eta = \frac{W}{Q_1}$

$W = Q_1 - Q_2$

$1600 = Q_1 - 500$

$Q_1 = 2100$ kJ

$\therefore \eta = \frac{1600}{2100} = 0.762 = 76.2\ \%$

$\eta < \eta_{max}$ (Irreversible) Ans

(d) $\eta = 30\%$ given

$\eta < \eta_{max}$ (Irreversible)

3.2. The data listed below are claimed for a power cycle between 727°C and 127°C. For each case, determine principles of thermodynamics would be violated.

(a) $Q_1 = 300$ kJ, $W = 160$ kJ, $Q_2 = 14$
(b) $Q_1 = 300$ kJ, $W = 180$ kJ, $Q_2 = 12$
(c) $Q_1 = 300$ kJ, $W = 170$ kJ, $Q_2 = 14$

Solution:

(a) $$\eta_{max} = \frac{T_1 - T_2}{T_1}$$

$$\eta_{max} = 1 - \frac{T_2}{T_1} = 1 - \frac{400}{1000} = 0.6 = 60\%$$

$$\eta = \frac{W}{Q_1} = \frac{160}{3000} = 0.533 = 53.3\%$$

Thus, the second law of thermodynamics is satisfied as energy balance ($W = Q_1 - Q_2 = 160$ kJ) satisfied.

(b) $W = Q_1 - Q_2 = 300 - 120 = 180$ kJ

Satisfy the energy balance.

$$\eta = \frac{180}{300} = 0.6 = 60\%$$

The cycle is reversible ($\eta = \eta_{max}$). For such a cycle the sec is also satisfied.

(c) $W = Q_1 - Q_2 = 300 - 140 = 160$ kJ

Accordingly, the conservation of energy principle with th data ($W = 170$ kJ) is not satisfied.

3.3. A refrigeration cycle operating between two res receives energy Q_2 from a cold reservoir at $T_2 = 25$ rejects energy Q_1 to a hot reservoir at $T_1 = 300$K. For the following cases determine whether the cycle op reversibly, irreversibly or is impossible.

(a) $Q_2 = 1000$ kJ, $W = 400$ kJ
(b) $Q_2 = 2000$ kJ, $Q_1 = 2200$ kJ
(c) $Q_1 = 3000$ kJ, $W = 500$ kJ
(d) $W = 400$ kJ, $COP = 6$

Solution:

(a) $$COP_{max} = \frac{T_2}{T_1 - T_2} = \frac{250}{300 - 250} = 5$$

$$COP = \frac{Q_2}{W} = \frac{1000}{400} = 2.5$$

Accordingly as $COP < COP_{max}$; the cycle is irreversible. Ans.

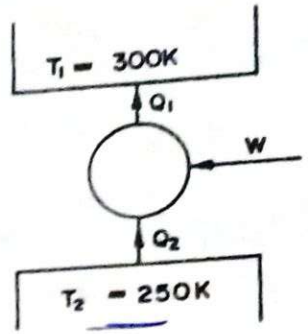


Fig. 3.15. Refrigeration cycle P. 3.3

(b) $$COP = \frac{Q_2}{W} = \frac{2000}{Q_1 - Q_2} = \frac{2000}{200} = 10$$

As $COP > COP_{max}$ this cycle is impossible Ans.

(c) $Q_2 = Q_1 - W = 3000 - 500 = 2500$ kJ

$$COP = \frac{2500}{500} = 5$$

$COP = COP_{max}$ (Reversible) Ans.

(d) $COP = 6$ given

As $COP > COP_{max}$ (Impossible) Ans.

3.4. An inventor claims to have developed a heat engine which operates between 27°C and 327°C and produces a 10 kW of power. The heat supplied to the engine is 25 kW. Comment on the claim of the inventor. **[JNVU BE I, 1975]**

Solution :

$T_1 = 327 + 273 = 600$ K

$T_2 = 27 + 273 = 300$ K

$$\eta_{carnot} = \frac{T_1 - T_2}{T_1}$$

$$= \frac{600 - 300}{600} = \frac{300}{600} = 0.5 = 50\%$$

$$\eta_{Actual} = \frac{\text{Power obtained}}{\text{Heat supplied}}$$

$$= \frac{10}{25} = 0.4$$

$$= 40\%$$

The efficiency of the Carnot engine is greater than the effi of the actual engine hence the claim of the inventor is justified.

3.5. An inventor proposes an engine that operates betwee 27°C warm surface layer of the ocean and a 10°C layer metres down. The inventor claims that the engine pro 100kW by pumping 20 kg/s of sea water. Is this possible?

Solution :

The maximum temperature drop for the seawater is 17°C maximum rate of heat transfer from the high temperature wa then

$$Q_1 = \dot{m}\, c_{pw}\, (t_1 - t_1)$$
$$= 20 \times 4.18\ (27 - 10)$$
$$= 1421.2 \text{ kJ/s}$$

$$\eta_{\text{proposed engine}} = \frac{W}{Q_1} = \frac{100}{1421.2} = 0.07036 = 7.036\%$$

The efficiency of the Carnot engine operating between the two temperatures is

$$\eta_c = \frac{T_1 - T_2}{T_1}\ 1 - \frac{T_2}{T_1} = 1 - \frac{287}{300} = 1 - 0.9433 = 0.0567 = 5.67$$

The proposed engine's efficiency exceeds that of a Carnot en hence the inventors claim is impossible.

3.6. Calculate the thermal efficiency of a internal combus engine that burns fuel at the rate of 4.11 kg/h whil produces 11.43 kW of output power. The calorific value of fuel is 42.5 MJ/kg. Calculate also the rate of energy tran to the surroundings from the engine.

olution :

$P = 11.43$ kW

$m_f = 4.11$ kg/h, C.V. = 42.5 MJ/kg

$$\text{Thermal efficiency} = \frac{3600}{\frac{\dot{m}_f}{P} \times \text{C.V.}}$$

$$\eta_{th} = \frac{3600}{\frac{4.11}{11.43} \times 42500} \times 100\% = 23.55\%$$

$$Q_s = \dot{m}_f \times \text{C.V.} = \frac{4.11 \times 42500}{3600} = 48.52 \text{ kW}$$

$$W = Q_s - Q_r$$
$$11.43 = 48.52 - Q_r$$
$$Q_r = 48.52 - 11.43$$
$$= 37.0908 \text{ kW}$$ Ans.

A domestic refrigerator consumes 98 W of electric power on average. The average rate of heat transfer to the refrigerant circuit inside the cabinet is 163W. Calculate the COP of the fridge and the rate of heat rejection.

lution :

Q_{in} = Net refrigerating effect = 163 W

W = Work input = 98 W

$$COP_R = \frac{Q_{in}}{W} = \frac{163}{98} = 1.6632$$ Ans.

$$W = Q_{out} - Q_{in}$$
$$98 = Q_{out} - 163$$
$$Q_{out} = 98 + 163 = 261 \text{ W}$$ Ans.

3.8. A heat engine operates on a Carnot cycle with an effi 25%. What COP would a refrigerator operating on th cycle have? The low temperature is 0°C.

Solution :

$$T_2 = 273 + 0 = 273 \text{ K}$$

$$\eta = \frac{T_1 - T_2}{T_1}$$

$$0.25 = 1 - \frac{T_2}{T_1} = 1 - \frac{273}{T_1}$$

$$\frac{273}{T_1} = 0.75$$

$$T_1 = 364 \text{ K}$$

$$COP_R = \frac{T_2}{T_1 - T_2} = \frac{273}{364 - 273} = \frac{273}{91} = 3$$

3.9. Calculate the required power input to a refrigrator if operates with COP of 1.75 and provides a cooling effect of kW. Determine the rate of heat transfer to the surroun is not justified.

Solution :

Net refrigerating effect = 41 kW

$$COP_R = 1.75$$

$$\therefore COP_R = \frac{\text{Net refrigerating effect}}{\text{Work input}}$$

$$1.75 = \frac{41}{W}$$

$$W = 23.4285 \text{ kW}$$

$$W = Q_{sourroundigns} - Q_{in}$$

$$23.4285 = Q_s - 41$$

$$Q_s = 64.4285 \text{ kW}$$

3.10. An inventor claims to have developed a refrigerating machine which operates between –23°C and 27°C and consumes 1 kW of power. The machine gives a refrigerating effect of 27 MJ/h. Comment on the claim of the inventor. **[JNVU BE I, 1976]**

Solution :

$$T_1 = 27 + 273 = 300 \text{ K}$$

$$T_2 = 23 + 273 = 250 \text{ K}$$

$$COP_R = \frac{T_2}{T_1 - T_2}$$

$$= \frac{250}{50}$$

$$= 5$$

$$COP_{Actual} = \frac{\text{Net refrigerating effect}}{\text{Power input}}$$

$$= \frac{27 \times 1000}{\frac{3600}{1}}$$

$$COP_{Actual} = 7.5$$

$\therefore$ COP_{Actual} is greater than the COP_R hence the inventor claim is not justified.

3.11. An inventor claims to have developed a heat pump which operates between –23°C and 27°C and consumes 1 kW of power. The machine gives a heating effect of 18 MJ/h. Comment on the claim of the inventor.

Solution :

$$T_1 = 27 + 273 = 300 \text{ K}$$

$$T_2 = -23 + 273 = 250 \text{ K}$$

$$COP_{HP} = \frac{T_1}{T_1 - T_2}$$

$$= \frac{300}{50} = 6$$

$$COP_{Actual} = \frac{\text{Net heating effect}}{\text{Power input}}$$

$$= \frac{18 \times 1000}{\frac{3600}{1}}$$

$$= 5$$

$\therefore$ COP_{HP} is greater than the COP_{Actual} hence the inventors c is justified.

3.12. Two Carnot refrigerators operate in series between reservoirs maintained at 27°C and 207°C, respectively. energy output by the first refrigerator is used as heat e input to the second refrigerator. If the COPs of the refrigerators are the same, what should the interme temperature be? **[JNVU BE I, 19**

Solution :

$$T_3 = 27 + 273 = 300 \text{ K}$$

$$T_1 = 207 + 273 = 480 \text{ K}$$

$$COP_1 = \frac{T_3}{T_2 - T_3}$$

$$= \frac{300}{T_2 - 300}$$

$$COP_2 = \frac{T_2}{T_1 - T_2}$$

$$= \frac{T_2}{480 - T_2}$$

As per question $COP_1 = COP_2$

$$\frac{300}{T_2 - 300} = \frac{T_2}{480 - T_2}$$

$$144000 - 300\,T_2 = T_2^{\,2} - 300\,T_2$$

$$T_2^{\,2} = 144000$$

$$T_2 = 379.473 \text{ K}$$

$$= 106.473°\text{C}$$

3.13. The Carnot engines operate in series between two reservoirs maintained at 327°C and 27°C respectively. The energy rejected by the first engine is input into the second engine. If the first engine's efficiency is 20% greater than the second engines efficiency, calculate the intermediate temperature. **[JNVU BE I, 1980]**

Solution :

$$T_1 = 327 + 273 = 600 \text{ K}$$

$$T_3 = 27 + 273 = 300 \text{ K}$$

$$T_2 = ?$$

$$\eta_1 = 1 - \frac{T_2}{600} \quad (1)$$

$$\text{and } \eta_2 = 1 - \frac{300}{T_2} \quad (2)$$

$$\text{Given that } \eta_1 = \eta_2 + 0.2\,\eta_2 = 1.2\,\eta_2 \quad (3)$$

$$1.2\,\eta_2 = 1 - \frac{T_2}{600}$$

$$1.2\left(1 - \frac{300}{T_2}\right) = 1 - \frac{T_2}{600}$$

$$1.2 - \frac{360}{T_2} = 1 - \frac{T_2}{600}$$

$$\frac{360}{T_2} - \frac{T_2}{600} = 0.2$$

$$360 \times 600 - T_2^2 = 0.2 \times 600\,T_2$$

$$T_2^2 + 120\,T_2 - 216000 = 0$$

$$T_2 = \frac{-120 \pm \sqrt{(120)^2 - 4 \times 1 \times (-216000)}}{2 \times 1}$$

$$= -60 \pm \sqrt{\frac{14400 + 864000}{4}}$$

$$= -60 \pm \sqrt{\frac{878400}{4}}$$

Internal Combustion Engines

10.1 INTRODUCTION

All heat engines convert heat into work by the expansion or increase in volume of a working fluid into which heat has been introduced by combustion of a fuel either external to the engine, as in steam engine, or internally by the burning of a combustible mixture in the engine itself, a process giving rise to the phrase internal combustion (IC) engine.

Internal combustion engines date back to 1876 when Otto first developed the spark ignition (SI) engine and 1892 when Diesel invented the compression ignition (CI) engine. The last thirty years have seen an explosive growth in engine research and development as the issues of air pollution, fuel cost, and market competitiveness have become increasingly important.

There are at present three basic types of mass produced automotive engines : the gasoline - piston engine, the Diesel engine, and the Wankel engine. The gas turbine has, outside its traditional applications in aviation and other areas, been used commercially only in trucks and buses. Among other reasons, high cost and a relatively high output of nitrogen oxides have, so far, prevented wider acceptance of it. While Diesel engines are used in some cars, they are also, because of high cost and a fairly low power to weight ratio, mostly confined to heavy-duty equipment.

Car powered by steam and electricity are still in the experimental stage, and their future appears uncertain at present. Compared with ordinary steam engines, the type of unit needed for modern cars with their high output, light weight, and compact size

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would be very expensive and complicated. Also, the condensers that are required to recycle the used steam are still too heavy and bulky to be practical. Low efficiency, high fuel consumption and maintenance cost, and safety risks are other problems associated with steam engine. The electric motor would offer great advantages if it were possible to store the necessary energy in some device comparable in size, weight and energy content to automotive fuel tanks. A lead-acid battery weighs approximately one hundred times as much as the quantity of gasoline required to produce an equivalent amount of energy ! So far no suitable substitute battery has been developed. The claim that electrical cars are "clean" overlooks the fact that the necessary generating plants for recharging the batteries are contributory sources of pollution (sulphur oxides, heat pollution of waterways, radio active waste material etc.). My discussion will therefore be confined to those internal combustion engines currently being mass produced.

Table 10.1 shows the most common applications of IC engines, the predominant type of engine used in each classification listed, and the approximate engine power range in each type of service.

10.2. CLASSIFICATION OF IC ENGINES

There are many different types of internal combustion engines. They can be classified on following considerations.

1. Basic engine design : Reciprocating engines (cylinder arrangement : e.g. in line, V, radial, opposed), rotary engines (Wankel and other geometries)

2. Fuel : Gasoline (Petrol), fuel oil (Diesel fuel), natural gas, liquid petroleum gas, alcohols (methanol, ethanol), hydrogen, dual fuel.

3. Method of ignition : Petrol and gas engines are the spark ignition engines (SI engines) The Diesel oil engines are the compression ignition engines (CI engines).

4. Method of mixture preparation : Carburetion, fuel injection intake ports or intake manifold, fuel injection into the engine cylinder.

5. Working cycle : Four stroke cycle : The cycle is completed in four strokes (i.e. two revolution of the crankshaft), whereas when the cycle is completed in two strokes (i.e. one revolution of the crankshaft) the engine is called the two stroke cycle engine.

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6. Intake system : Four stroke cycle : naturally aspirated (admitted atmospheric air), supercharged (admitting precompressed fresh mixture), and turbocharged (admitting fresh mixture compressed in a compressor driven by an exhaust turbine), two stroke cycle : crankcase scavenged, supercharged, and turbocharged.

7. Method of cooling : Water cooled, air cooled, uncooled (other than by natural convection and radiation).

8. Method of load control : Throttling of fuel and air flow together so mixture composition is essentially unchanged, control of fuel flow alone, and combination.

9. Valve or port design and location : Over head (or I-head) valves, underhead (or L-head) valves, rotary valves, cross scavenged porting (inlet and exhaust ports on opposite sides of cylinder at one end), loop scavenged porting (inlet and exhaust ports on same side of cylinder at one end), through-or uniflow scavenged (inlet and exhaust ports or valves at different ends of cylinder).

10. Combustion chamber design : Open chamber (many designs : e.g. disc, wedge, hemisphere, bowl-in-piston), divided chamber (small and large auxiliary chambers; many designs : e.g. swirl chambers, pre chambers).

11. Operating cycle : The cycles commonly used are constant volume or Otto cycle (mostly for petrol and gas engines). Diesel cycle (mostly low speed oil engines) and mixed or limited pressure cycle or Dual cycle (mostly high speed oil engines.)

12. Speed : Engines having rpm above 900 are called high speed engines and less than 400 rpm are called slow speed engines.

13. Single acting or double acting : I.C. engines for road transport are always single acting; such engines are too small to warrant the complication of the piston cooling devices which would be needed if hot gases were in contact with both sides of the piston. Most reciprocating marine diesel engines are double acting to achieve more power output from a given cylinder volume.

14. Method of starting : Engines are classified according to how an engine is started, i.e. by the compressed air, inertia, cartridge or by an electric motor.

15. Method of governing : Engines are classified according to the method of governing - hit and miss, quality and quantity governed engines.

16. Application : Automobile, truck, locomotive, light aircraft, marine, portable power system, power generation.

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10.3. IC ENGINE AND ITS COMPONENTS

The essential parts and the job that each performs are as follows:

1. **A piston** sliding in a **cylinder**. The piston has two jobs; first, to compress the air charge, second, to receive the pressure of the gases while they are burning and expanding.

2. **A cylinder head** which closes the top end of the cylinder so as to make a confined space in which to compress the air and confine the gases while they are burning and expanding.

3. **Valves or ports** to admit the air and to discharge the spent or exhaust gases.

4. **Connecting rod** to transmit force in either direction between the piston and the crank on the crank shaft.

5. **Crankshaft** and the **"main" bearings** which support the crankshaft and permit it to rotates.

6. A supporting structure to hold the cylinder, crankshaft and main bearings in firm relation to each other. This structure is usually made up of two parts, called frame and bedplate.

7. **Fuel injection pump** to force the oil into the cylinder; and also fuel injection nozzle to break up the oil into a fine spray as it enters the cylinder.

8. **Camshaft,** driven by the crankshaft, to operate the fuel injection pump and also to open the valves (in engines which uses valves).

9. **Flywheel,** to store up surplus energy on the power stroke and to return that energy when the piston is being pushed upward on the compression stroke.

10. **Governor** or throttle, to regulate the amount of fuel supplied each stroke, and thus control the engine speed and power.

11. **Blower,** to force air into the cylinder of two stroke engines.

12. Miscellaneous parts, such as piping to supply air and remove exhaust gases lubricating system to lubricate the moving parts, water jacket to cool the cylinder, etc.

In addition to the above nomenclature the following are the standard terminology used in internal combustion (IC) engines. (Fig. 10.1).

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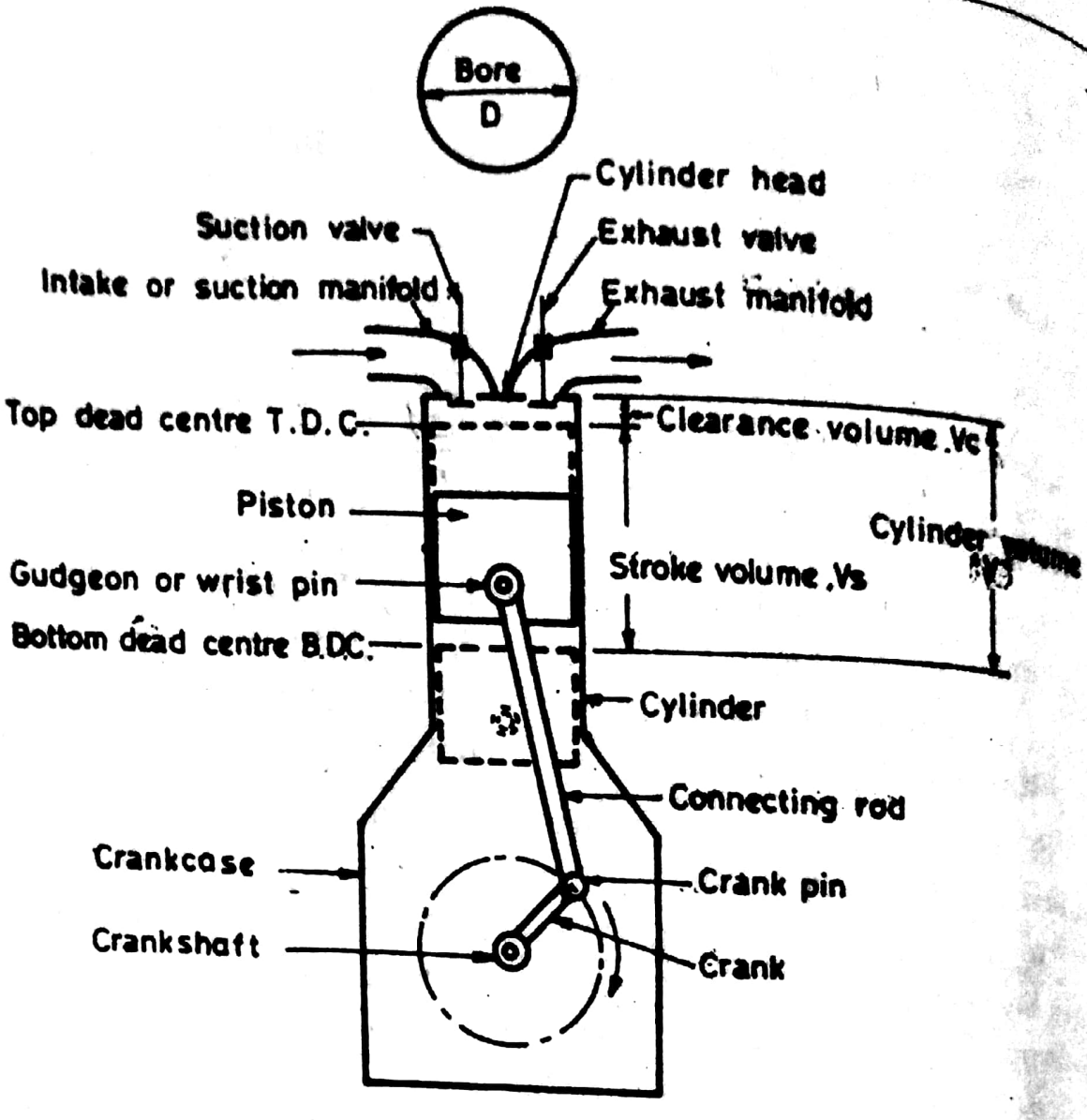

Fig. 10.1. Defintion sketch.

1. **Cylinder bore (D).** The nominal inner diameter of the working cylinder.

2. **Piston area (A)** : The area of a circle of diameter D (cylinder bore) is known as piston area.

3. **Stroke (L)** : The nominal distance through which a working piston moves between two successive reversals of its direction of motion.

4. **Dead centre** : The position of the working piston and the moving parts which are mechanically connected to it at the moment when the direction of the piston motion is reversed (at either end point of the stroke).

(a) Bottom dead centre (BDC) : Dead centre when the piston is nearest to the crankshaft. In horizontal engines it is also called **outer dead centre (ODC)** [Fig. 10.2].

(b) Top dead centre (TDC) : Dead centre when the position is farthest from the crankshaft. In horizontal engines it is also called inner dead centre (IDC) [Fig. 10.2].

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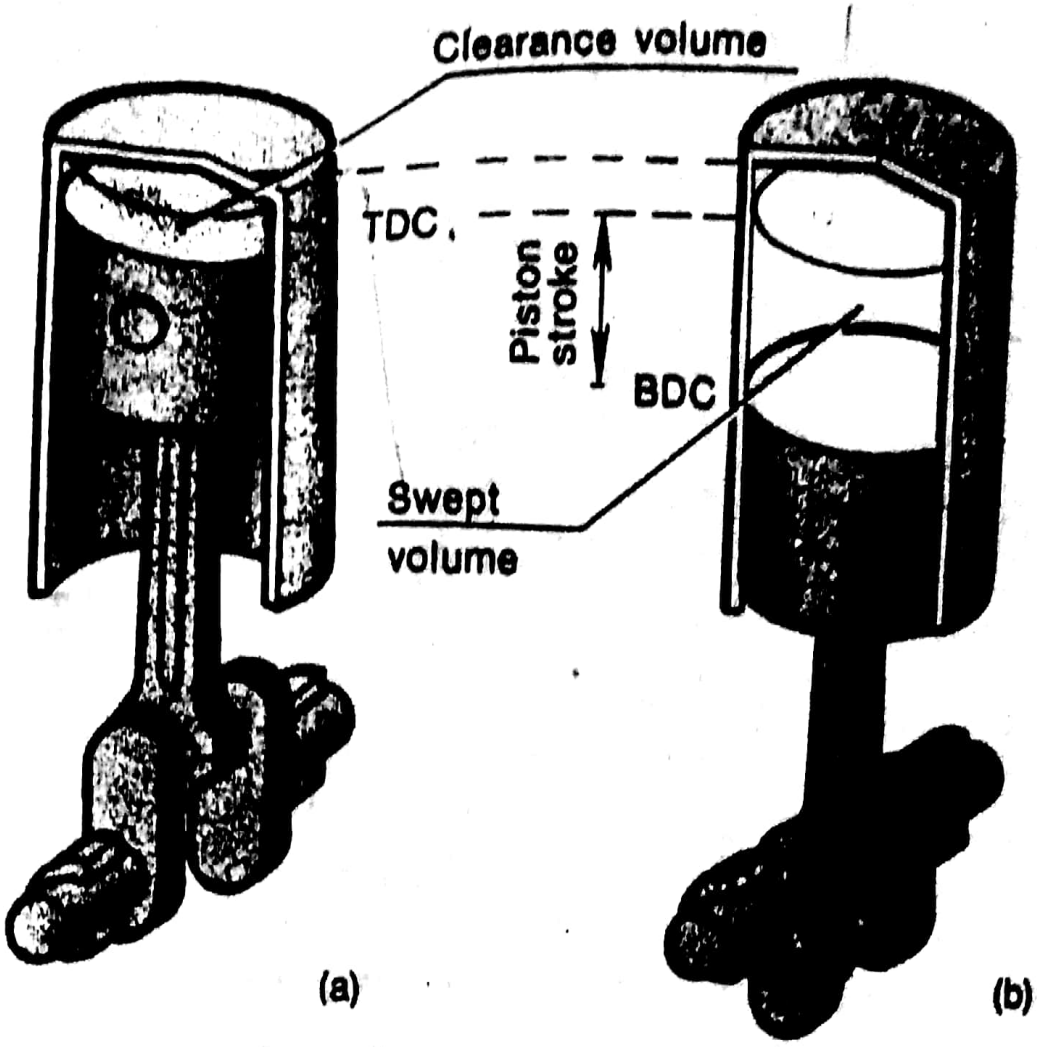

Fig. 10.2. Piston positions at dead centres.

5. **Clearance volume (V_c)** : The nominal volume of the space on the combustion side of the piston at top dead centre [Fig. 10.2].

6. **Displacement volume or swept volume (V_s)** : The nominal volume generated by the working piston when travelling from one dead centre to next one, calculated as the product of piston area and stroke length [Fig. 10.2].

$$V_s = A L$$

7. **Cylinder volume** : The sum of Swept volume and clearance volume.

$$V = V_s + V_C$$

8. **Compression ratio (r)** : The numerical value of the cylinder volume ($V = V_s + V_C$) is divided by the clearance volume

$$r = \frac{V_s + V_C}{V_C}$$

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10.4.1. Four stroke Petrol engine (SI engine) cycle

The four stroke cycle principle can be described as follows (Fig. 10.3) and p-v diagram (Fig 10.4).

Suction stroke (Intake stroke) : During the suction stroke (intake stroke), the piston travels downward in the cylinder from TDC to BDC, creating a partial vacuum in the cylinder. Atmospheric pressure forces air through the carburettor where it mixes with petrol and enters the cylinder through the intake manifold and the open intake valve. The exhaust valve remains closed during this stroke and the crankshaft turns through half a revolution. The camshaft turns through one fourth of a revolution (Fig 10.3a). The engine operating temperature some where between 82°C and 116°C.

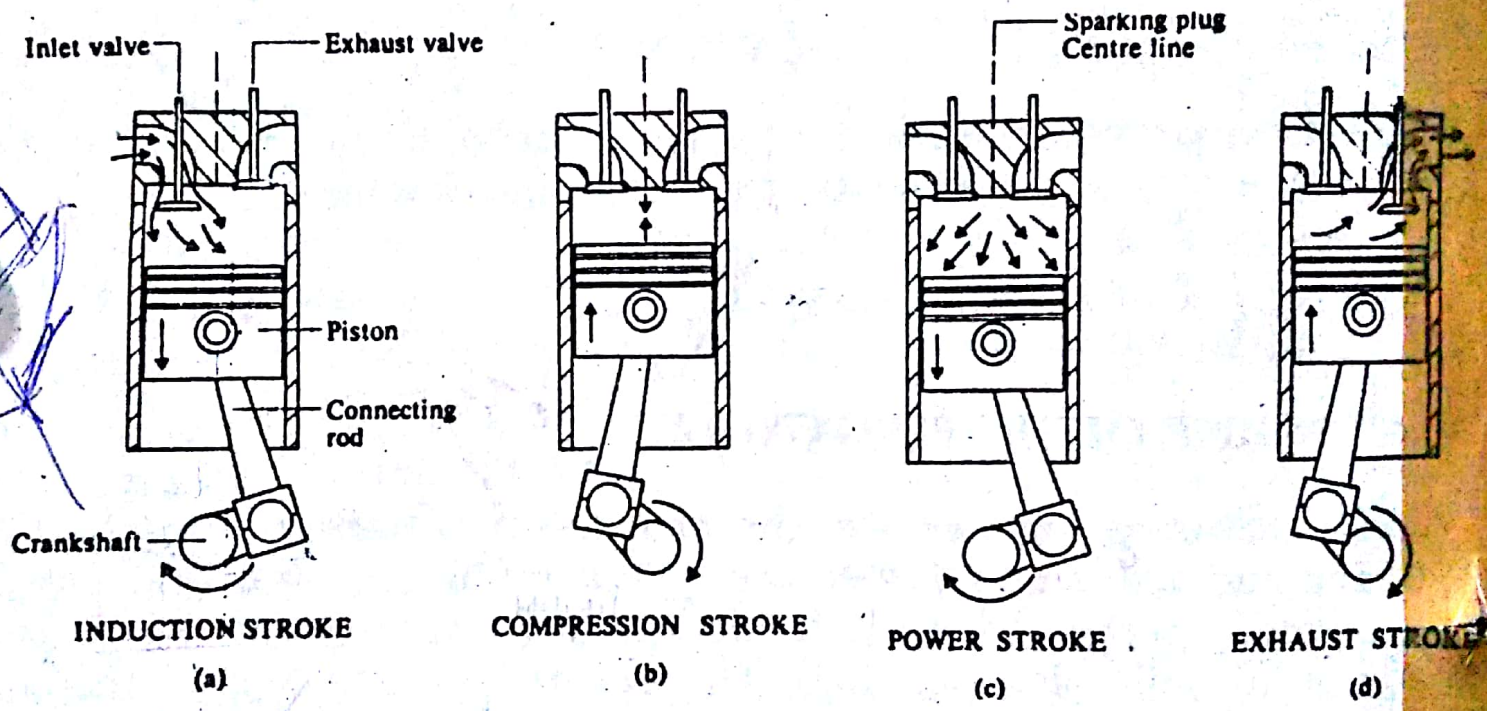

Fig. 10.3. Working of four stroke petrol engine.

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The compression stroke: During the compression stroke (Fig. 10.3b) a further half revolution of the crankshaft (one fourth of crankshaft) causes the piston to moves upwards from BDC to TDC. Since both valves are closed, the mixture is now compressed into the small cavity remaining above the piston. This cavity is called the combustion or firing chamber. The compression pressure may reach anywhere between 827 kPa and 1241 kPa. The friction between the compressed gas molecules causes a further temperature rise of several hundred degrees. This prepares the gas for firing.

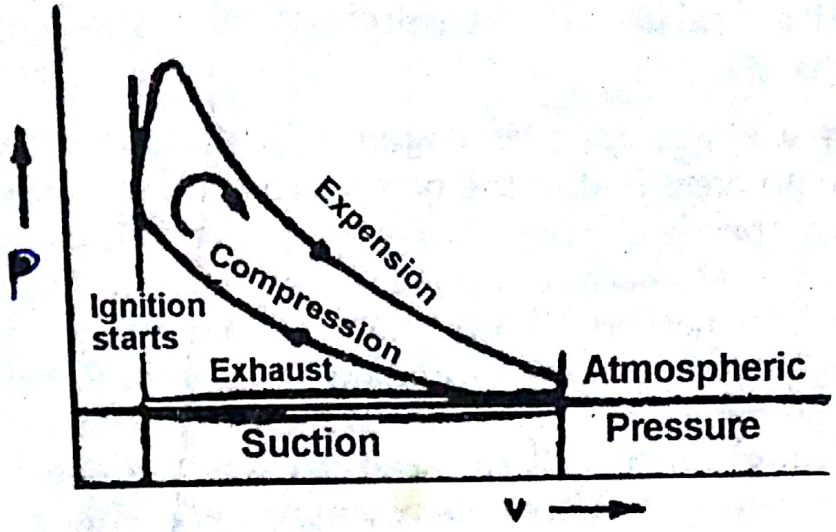

Fig. 10.4. p-V diagram of four stroke petrol engine.

The power stroke or firing stroke or working stroke or expansion stroke : The piston now approaches TDC, and both valves remain closed. At this point a carefully timed spark from the spark plug ignites the first "layer" of compressed gas, which in turn ignites the next "layer", and this one, the next, and so on. This results in a rapid but controlled chain reaction until all "layers" of the entire "charge" are ignited. The very high temperatures, which momentarily may approach 2800°C, causes the confined gases, including the relatively inert (chemically inactive) nitrogen, to expand at an extremely fast rate. The resulting high pressure pushes the piston downwards (Fig 10.3c) the chemical energy of the fuel is first converted to the heat energy of combustion, which in turn is converted to mechanical energy at the piston. The connecting rod transmits the firing pressure acting against the piston to the crankshaft. In this manner, the reciprocating motion of the piston is changed to the rotary motion of the crankshaft. During the power stroke the crankshaft completes a further half revolution (camshaft complete one fourth revolution).

The exhaust stroke : The crankshaft has now rotated one and one half revolution (camshaft rotated three fourth of revolution), and the cylinder is now filled with burnt gases which must be removed.

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The exhaust valve opens and the piston moves from BDC to TDC, forcing the burnt gases out of the cylinder. This is known as the exhaust stroke. During this stroke the intake valve remains closed. The crankshaft has completed two revolutions. The piston is at TDC (Fig. 10.3d) and the engine is ready to repeat the cycle of operation.

Since the crankshaft makes half a revolution during each stroke, the four strokes of one complete cycle equal two revolutions of the crankshaft. **Because each valve opens only once during the four stroke cycle, the camshaft makes only one revolution. Therefore, the ratio of crankshaft to camshaft speed (revolutions) is 2:1.**

Obviously a single cylinder engine will run rather unevenly at lower rpm. The problem is that the power stroke is the only lift stroke; it is followed by the three dead strokes, namely, the exhaust stroke, the intake stroke, and the compression stroke. This is the main reason why a flywheel is needed. While most of the mechanical energy released by the power stroke is available to do useful work, some of the energy is stored in the flywheel. During the dead strokes this energy, called kinetic energy, is released to carry the engine from one power stroke to the next without any appreciable change in engine speed.

10.4.2. Four stroke Diesel engine (CI engine) cycle

In a CI engine, air and fuel oil vapour compressed enough, the heat so generated would be sufficient to ignite the mixture without the need for an electric spark. The air could be compressed to approximately 4100 kPa and into which fuel oil was injected at approximately 8270 kPa. This two pressures generated sufficient heat to ignite the mixture of air and fuel vapour and therefore force the piston down its cylinder. The fuel employed in this type of engine is less volatile than petrol, although of a similar chemical nature.

The actual construction of a Diesel engine is very similar to that of a petrol engine, except that the Diesel engine does not have a carburettor or sparking plug. The sparking plugs are replaced by an injector through which a pump forces a very fine spray of fuel.

Referring to Fig. 10.5, the operation of the four stroke Diesel cycle (CI engine) is as follows and p-v diagram (Fig. 10.6).

Induction stroke : The inlet valve opens and the piston moves downward, a depression is created in the cylinder. Atmospheric pressure outside the cylinder forces air through the open inlet valve into the cylinder, as in Fig 10.5a. Once the piston has reached the end of the induction stroke the inlet valve is closed.

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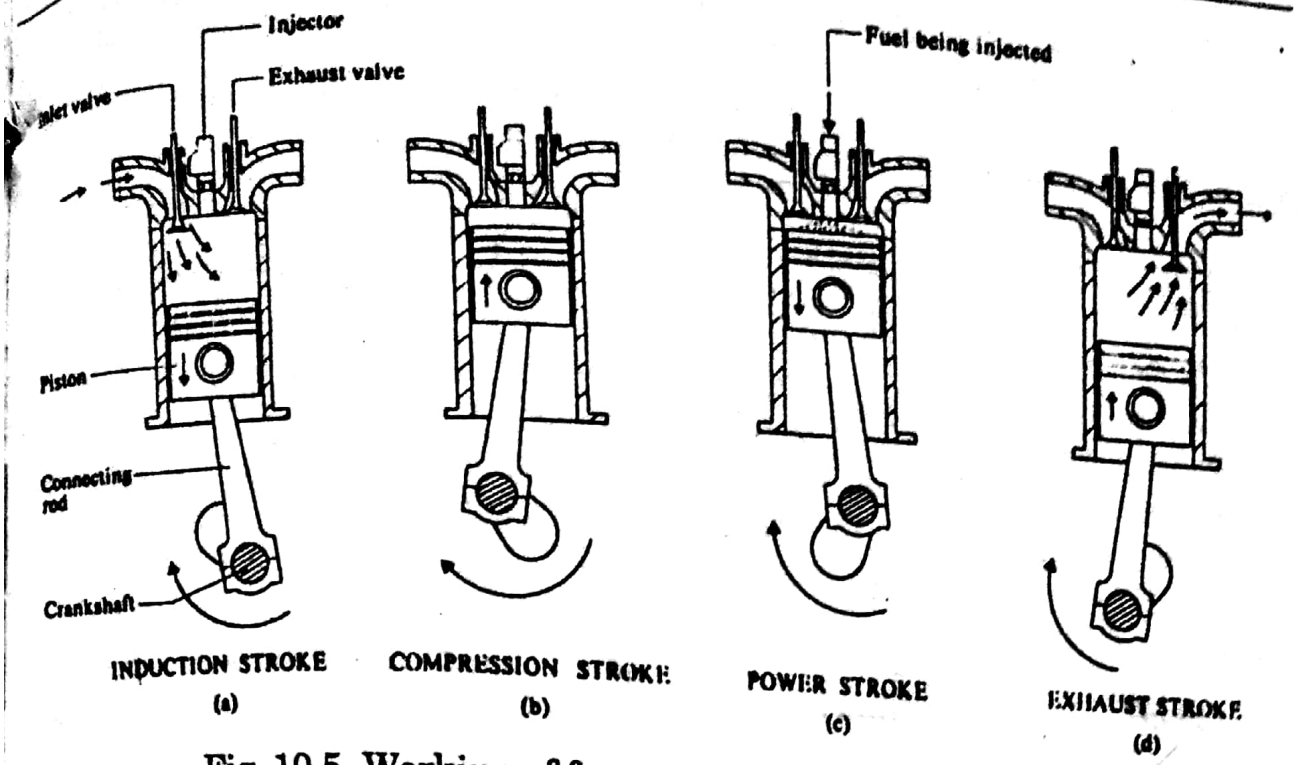

Fig. 10.5. Working of four stroke Diesel engine cycle.

Compression stroke : The piston moves upward, both inlet and exhaust valves remain closed, and the air trapped in the cylinder is compressed, as indicated in Fig 10.5b. As a result of the high pressure the air temperature within the cylinder will be about 800°C. A measured quantity of atomised fuel is injected into the cylinder just before the piston reaches the top of its stroke.

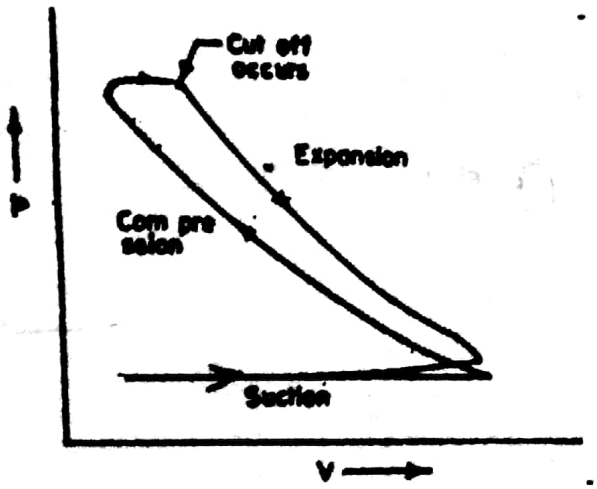

Fig. 10.6. p-V diagram of four stroke Diesel engine.

Power stroke : The fuel injected has a self ignition temperature in the region of 400°C, therefore, being introduced into a temperature of 800°C, it ignites and begins to burn. The expansion of the burning gases forces the piston down on its power stroke; as shown in Fig. 10.5c.

Exhaust stroke : As the piston nears the end of its downward stroke, the exhaust valve opens. The exhaust gases are forced out of the cylinder as the piston moves upward on its exhaust stroke as illustrated in Fig 10.5d. When the piston reaches the top of its travel,

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the exhaust valve is closed and the inlet valve opens and another cycle of operations begins.

10.4.3. Two stroke SI engine cycle

The conventional three port two stroke engine does not employ mechanically operated valves. Instead, ports are formed in the cylinder walls which are covered and uncovered by the movement of the piston. In this type of engine there are only three moving parts : the piston, connecting rod and crankshaft. The cycle of operation is completed in two strokes of the piston or one revolution of the crank shaft. The general arrangement of the simple two stroke engine is shown in different positions in Fig 10.7, and the operations are as follows :

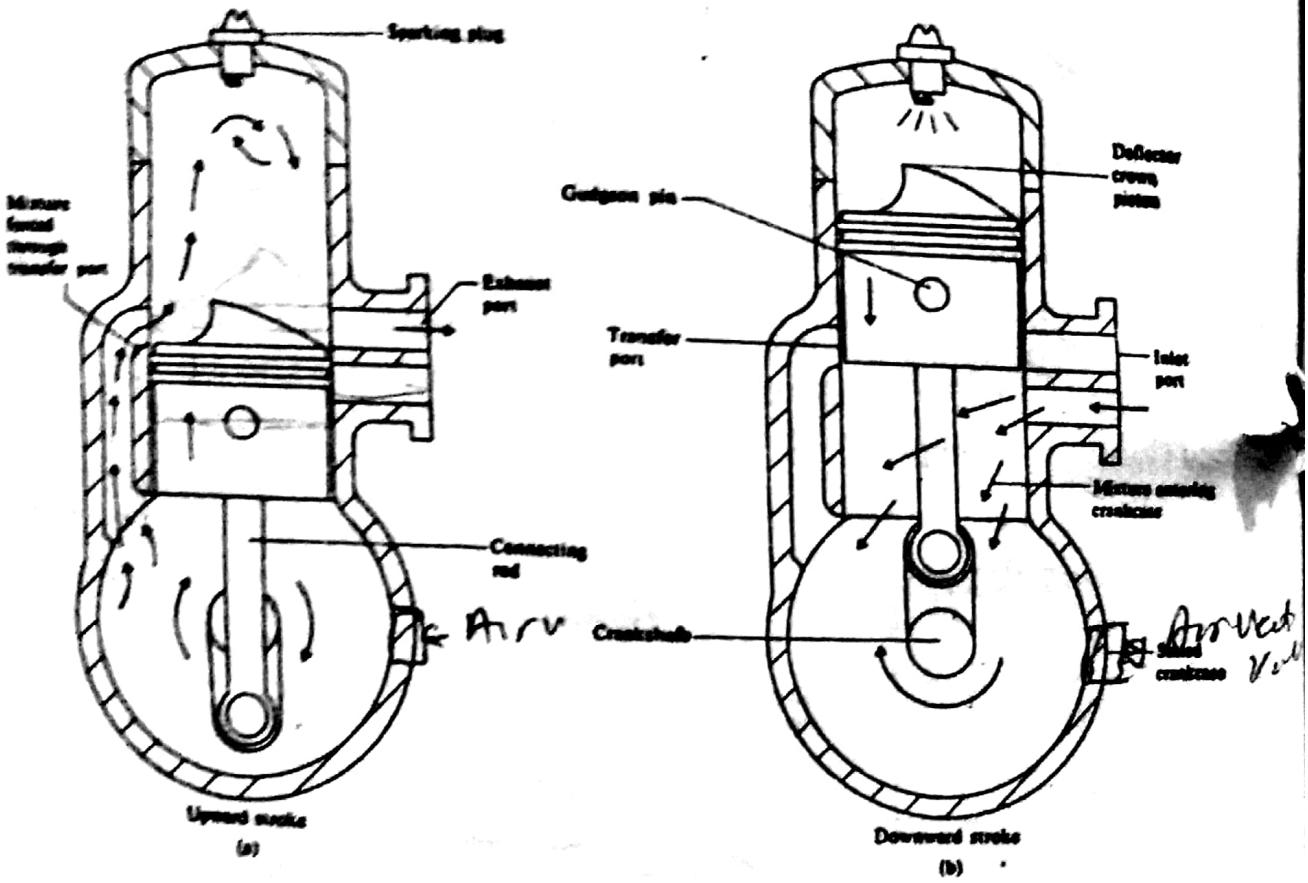

Fig. 10.7. Working of two stroke petrol engine.

The upward stroke : When the piston is at the bottom of its stroke the inlet port is closed, but the exhaust and transfer ports are open. On one side of the cylinder exhaust gases are leaving and at the opposite side of the cylinder a fresh charge of mixture is entering from the crankcase. The piston is provided with a deflector head or crown which directs the fresh charge of mixture to the top of the cylinder and the exhaust gases towards the exhaust port. As the piston moves up the cylinder it 'cuts off' or closes both the transfer and exhaust ports. The continued upward movement of the piston creates a depression in the crankcase and as the inlet port is uncovered and air

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and petrol mixture from the carburettor flows into the crankcase. At the same time the mixture above the piston is being compressed.

As the piston nears the end of its upward stroke, a spark occurs at the spark plug which ignites the compressed mixture of air and fuel in the combustion chamber.

The downward stroke : The high temperature produced by the burning mixture causes an increase in pressure followed by expansion, which forces the piston down the cylinder on its effective power stroke. Fresh mixture is still flowing into the crankcase via the inlet port but as the piston descends this port is closed and the mixture in the air tight crankcase is compressed. At the same time the exhaust port above the piston is beginning to open allowing the burnt gases to escape to the exhaust system.

As the transfer port has also been uncovered by the piston, the compressed mixture from the crankcase can flow through the transfer passage into the cylinder. The crown of the piston deflects the mixture towards the top of the cylinder, scavenging the exhaust gases. Again the piston moves up the cylinder, closes transfer and exhaust ports and compresses the new charge of mixture in the combustion chamber. The cycle of operation is continuously repeated while the engine is running, thereby completing the cycle of events in 360° of crank shaft rotation.

An ideal p-v diagram for a Otto cycle engine is drawn in Fig. 10.8a. A plot of crankcase pressure is included. Starting at point 1, with both ports closed, the cycle passes through compression, combustion, and expansion, arriving at point 4. Now the ports are uncovered, cylinder pressure drops to crankcase pressure, crankcase gas passes into the cylinder through the transfer passage, mixing occurs in the cylinder, and some gases leave the cylinder. This complicated set of events in the cycle ends at point 5, when cylinder and crankcase pressures are both atmospheric. With further piston movement, more gas transfers from crankcase to cylinder until BDC is reached at point 6. On the return to point 1, the crankcase opens, the transfer passage closes, and a portion of the gases in the cylinder is displaced into the exhaust port and out of the engine.

Fig 10.8b is shown a sketch of an indicator card taken from a two stroke cycle engine cylinder. The various events, as they occur are indicated. The reduction in pressure in the expansion line as exhaust port opens is to be noted. Note also the similarity of shape between this and the ideal p-v diagram of a constant volume cycle or Otto cycle.

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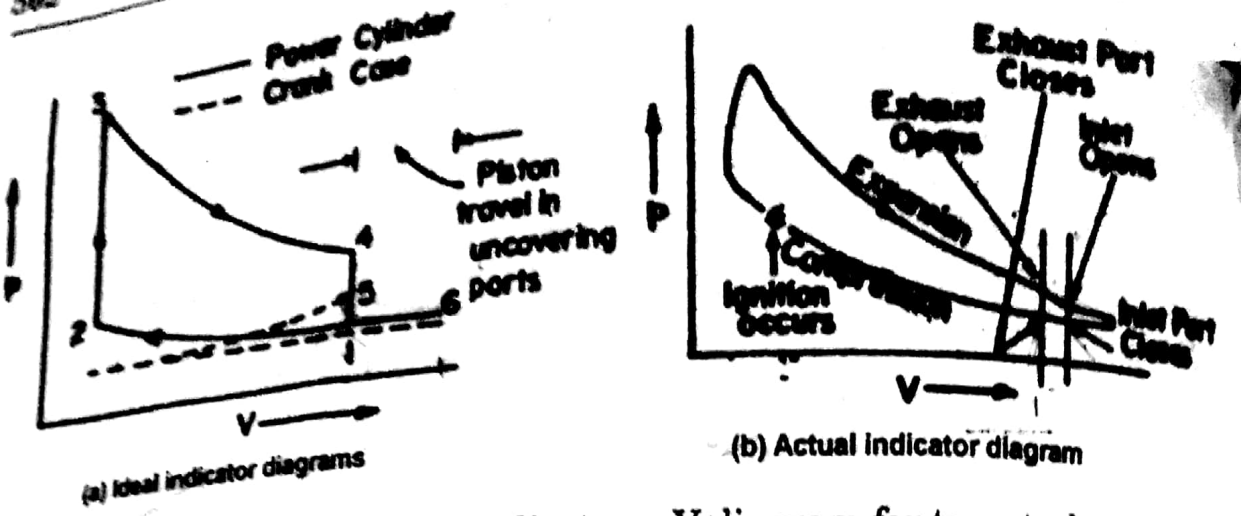

Fig. 10.8. Ideal and actual indicator p-V diagram for two stroke petrol engine.

10.4.4 Two stroke CI engine cycle

The **disadvantages of the two stroke petrol engine are** : (a) No pressure lubrication, (b) Exhaust emission is high (c) Sensitive to mixture adjustments, (d) Low thermal efficiency.

Due to no pressure lubrication, and the oil must either be mixed with the petrol or be supplied directly to the engine by a metering pump. Exhaust emission is high, particularly of unburned hydrocarbons (oil and gasoline). The engine is very sensitive to mixture adjustments, and the wrong mixture can cause fouled spark plugs, rough engine-idle. The exhaust system may clog up with carbon deposits, and proper lubrication requires close attention. The theoretical claim that the power-per-engine displacement is doubled is not possible with ordinary designs, partly due to scavenging and transfer problems. The low thermal efficiency, which is due to incomplete scavenging of the exhaust gases and to unburnt mixture flowing out of the combustion chamber with the exhaust gases.

Diesel engines that work on the two stroke cycle are equipped with regular exhaust valves in the cylinder head and several intake ports in the cylinder. This arrangement also permits the use of a pressurized lubrication system with a conventional oil pan because the air is forced into the cylinder by a blower or supercharges and not by means of crankcase precompression.

Cycle of operation : The general construction and layout of a two stroke Diesel engine is shown in Fig 10.9. It can be seen that the cylinder is surrounded by an air chest or box, which is kept charged with air under pressure by an engine driven blower. Ports air under pressure by an engine driven blower. Ports lead from the air chest into the cylinder. Poppet type exhaust valves are located in the cylinder

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head and those are operated by conventional cams, push rods, rocker arms, etc. and are closed by springs. As the ports are uncovered by the piston the incoming air, as well as filling the cylinder, assists in disposing of the gases past the open exhaust valves at the top of the cylinder. The cycle of operation of a two stroke CI engine is described as follows.

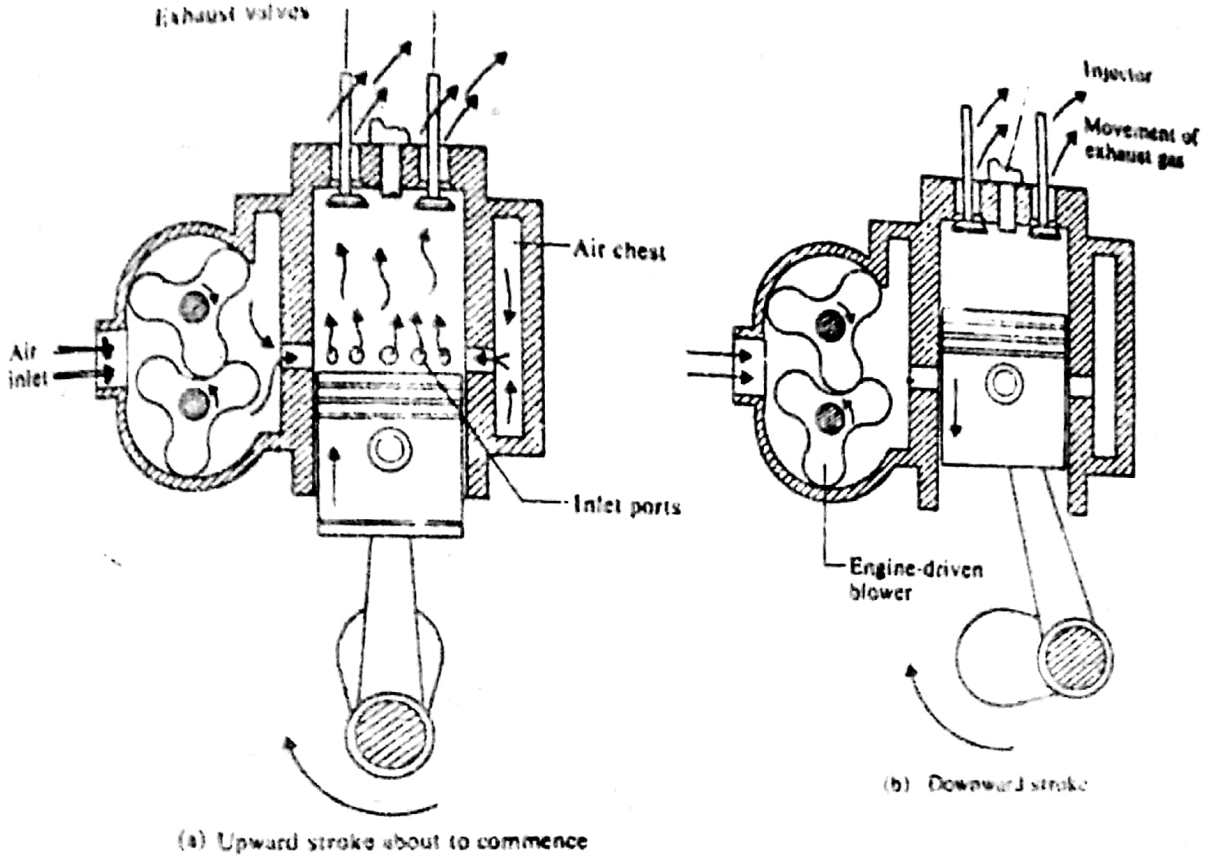

Fig. 10.9. Working of two stroke Diesel engine.

The upward stroke : Only air is allowed to pass into the cylinder while the inlet ports are uncovered, at the same time the exhaust valves are open. This combination of uncovered inlet ports and open exhaust valves permits air to be blown through the cylinder, thereby removing exhaust gas and filling the cylinder with a fresh charge of air, as indicated in Fig 10.9. As the piston advances up the cylinder the inlet ports are covered, the exhaust valves are closed and the air is compressed into the lesser volume of the combustion chamber where the air temperature rises to approximately 540°C where self ignition of fuel will occur. Just before the piston reaches the top of its stroke, fuel is injected into the cylinder and the combustion occurs. The heat developed from the compression of air in the cylinder ignites the fuel and the expanding gases force the piston downward on its effective power stroke.

The downward stroke : Before the piston reaches the end of its downward or power stroke, the exhaust valves open, allowing the burnt exhaust gases to escape, as shown in Fig 10.9. Shortly afterwards the piston uncovers the air inlet ports, allowing the

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pressurized air in the air chest to remove the remaining exhaust gases and charge the cylinder enabling the whole cycle to start all over again.

Note : In the Fig. 10.7 instead of spark plug use injector for the working of diesel engine. Only air is allowed to pass into the crankcase while the inlet ports are uncovered. As the piston advances up the cylinder the inlet ports are covered and the air is compressed into the lesser volume of the combustion chamber. Just before the piston reaches the top of its stroke, fuel is injected into the cylinder and the combustion occurs.

10.5. VALVE TIMING DIAGRAMS

Theoretically, the opening and closing events of the valves would occur at the crankshaft dead centre positions coincident with the beginning and closing of the piston strokes, but the time necessary for opening and closing the valves and the effects of high speeds upon the flow of gases require that these events be displaced somewhat from the dead centre points. The intake and exhaust periods of the cycle are therefore not truly intake and exhaust strokes, since each includes 50 or more degrees of crank rotation in excess of a full piston stroke, and the two periods actually overlap to some extent. The most desirable timing of the valve operation is determined experimentally for each engine.

10.5.1. Valve timing terms

Valve timing is expressed in degrees of crankshaft rotation as shown in the valve timing diagram.

Valve lead : Valve lead is the number of degrees a valve opens before TDC or BDC (btdc or bbdc).

Valve Lag : Valve lag is the number of degrees a valve closes after BDC or TDC (abdc or atdc).

Valve overlap : Valve overlap is the total number of degrees the intake and exhaust valves of a given cylinder are open at the same time.

Total valve opening : Total valve opening is the total number of degrees that a given valve stays open.

Total valve opening

= Valve lead + 180° (the equivalent of one stroke) + Valve lag.

On same high performance engines, total valve opening may be

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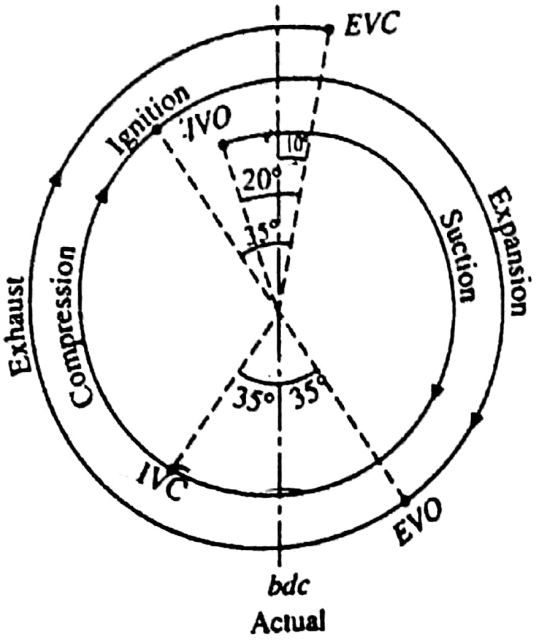
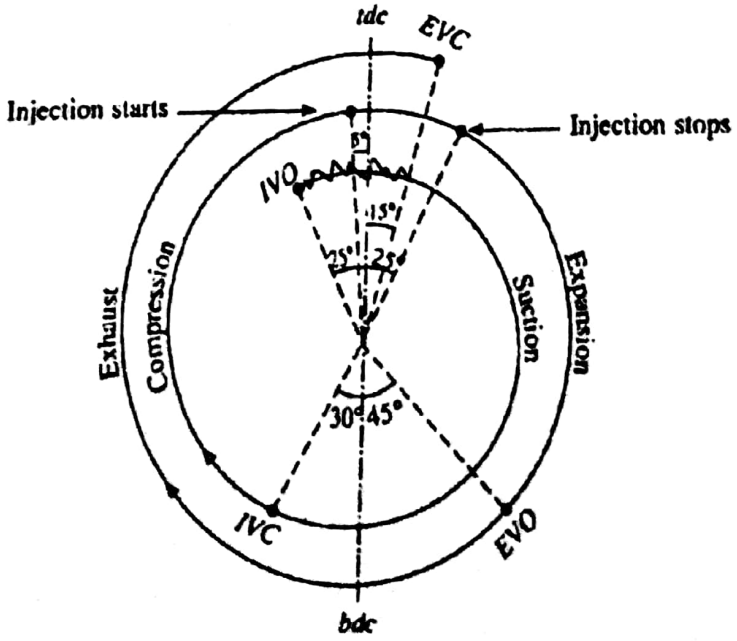
300° or more. This represents a gain of 120° in valve opening over an engine without any valve lead or lag.

10.5.2. Valve timing diagram for four stroke engine

Fig 10.10 shows all four valve events, inlet opening, inlet closure, exhaust opening and exhaust closure, to be displaced appreciably from the top and bottom dead centre points, where they would theoretically occur. **Opening events are earlier and closing events later to allow for time required to open and close the valves without excessive inertia loading or slamming.** Ramps are provided on



Four Stroke SI engine



Four Stroke CI engine

Fig. 10.10. Valve timing diagram for four stroke engine.

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each cam to start the valve lift gradually and to slow down the valve return motion slightly before the seat is contacted. Of still greater significance, however, are the advance in opening timing and delay in closure for the purpose of utilizing the inertia of the flowing gases and other practical consideration.

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0.6. COMPARISON OF TWO STROKE AND FOUR STROKE ENGINES

Two stroke engines have no valves but only ports, but in two troke diesel engines conventional exhaust valves are provided. The wo stroke engine developed greater output from the same size of ngine. In the two stroke diesel engines, there is no loss of fuel with xhaust gases as the intake charge in diesel engine is air only.

A disadvantage common to all two stroke engines, petrol as well s diesel, is greater cooling and lubrication requirements due to one ower stroke in each revolution of crankshaft. Due to higher mperature, consumption of lubricating oil is also high.

able 10.2. Comparison of two and four stroke engines.

o.	Points	Two stroke engine	Four stroke engine
	Strokes/ working cycle	The cycle is completed in two strokes of the piston or in one revolution of the crank shaft. Thus one power stroke is obtained in each revolution of the crankshaft.	The cycle is completed in four strokes of the piston or in two revolution of the crank shaft. Thus one power stroke is obtained in every two revolution of the crank-shaft.
2	Valves	Generally no valves. Valves are replaced by inlet, exhaust and transfer ports. In two stroke diesel engines, conventional exhaust valves are provides.	Inlet and exhaust valves. It contains valve mechanism.
3	Fly wheel	Turning moment is uniform hence lighter flywheel is needed	Turning moment is not uniform hence heavier flywheel is needed.
4	Moving parts	Less in number.	More in number.
	Wear and tear	Greater cooling and lubrication requirements due to one power stroke in one revolution, therefore lesser rate of wear and tear.	Lesser cooling and lubrication requirements due to one power stroke in two revolution, therefore greater rate of wear and tear.
	Volumetric efficincy	Volumetric efficiency less due to lesser time for induction.	Volumetric efficiency more due to more time of induction

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7	Mechanical efficiency	Lower	Higher
8	Thrmal efficiency & part load efficiency	Thermal efficiency lower, part load efficiency lesser than four stroke engine.	Thermal efficiency higher, part load efficiency better than two stroke engine.
9	Scavenging	Less efficient. Scavenging losses are more.	More efficient Scavenging losses are less.
10	Cost	Initial cost is less due to light weight and absent of valve mechanism.	Higher in cost due to heavy weight and complicated valve mechanism.
11	Uses	Two stroke (air cooled) petrol engines used in very small size only : Scooters, motor cycles, mopeds etc. (Lubricating air mixed with petrol). Two stroke diesel engines used in very large sizes. Used in ship propulsion because of light weight and compactness. It is generally used where low cost, compactness and light weight important.	Cars, buses, trucks tractors- efficiency is more important. Here Honda motor cycle-better fuel economy. Aeroplanes, Industrial engines and power generation etc.

10.7 CARBURETTOR

10.7.1. Definition

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The term carburetion may conveniently be applied to the whole process of supplying to the engine continuously a suitable combustible mixture of vaporized fuel and air. The process of preparing the mixture is called carburetion. This complicated process is achieved in the induction system, which is shown in Fig 10.12.

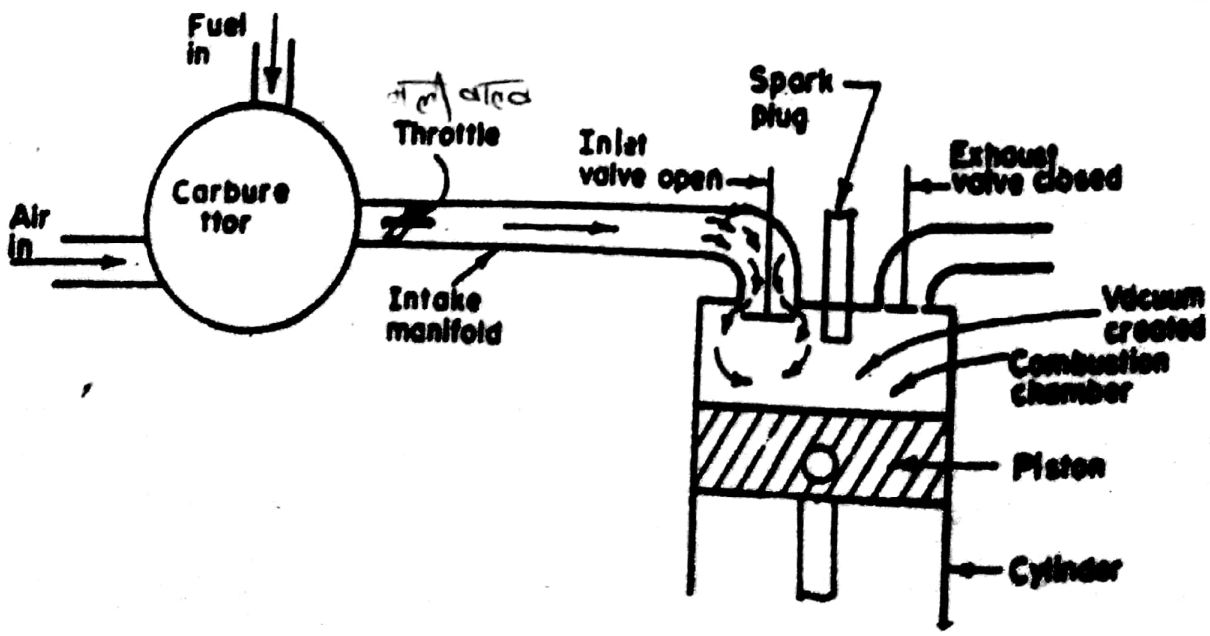

Fig. 10.12. Induction system.

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7.4 Action of the simple carburettor

The Fig 10.13 shows the action of the simple carburettor. When e engine is stationary, atmospheric pressure acting on the surface of e fuel in the float chamber is equal to that in the spray tube outlet, erefore under this condition the fuel in the chamber will remain atic as shown in Fig. 10.13(a). However, when the engine crankshaft rotated and air is drawn through the venturi, an imbalance of essure between the venturi and float chamber will result. Owing to

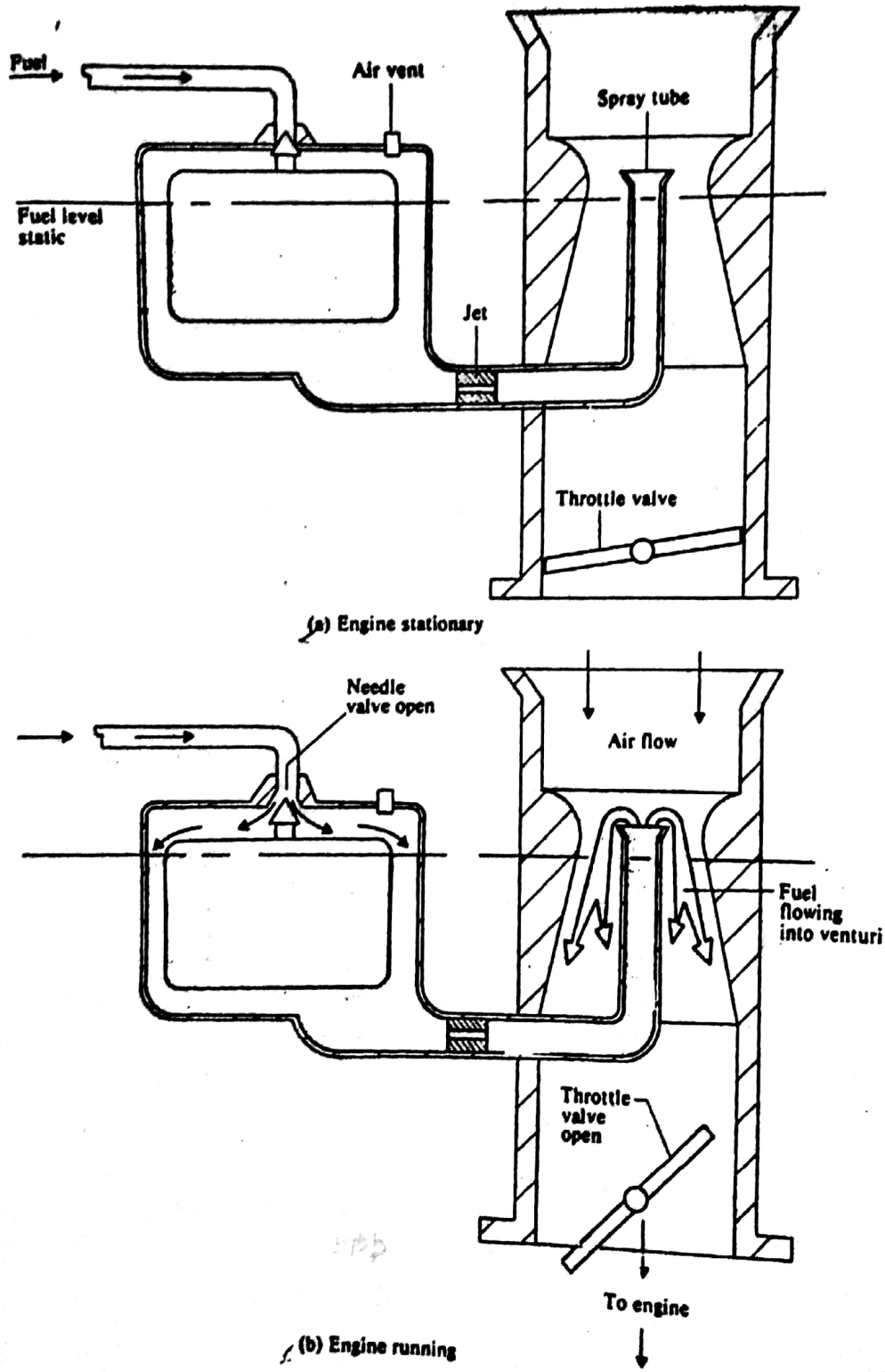

Fig. 10.13. Operation of the simple carburettor.

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the speed of the air flowing through the carburettor, the pressure will fall in the venturi tube and as the fuel in the float chamber remains at atmospheric pressure, this higher pressure will cause petrol in the chamber to be forced out through the spray tube into the venturi as indicated in Fig 10.13(b). The difference between the pressure in the venturi and atmospheric pressure is termed the 'pressure drop' across the carburettor. Petrol will start to flow from the spray tube and into the air stream. Fuel which emerges from the spray tube will break down into very small droplets which will mix with the air passing through the carburettor to form a very fine spray or vapour which will be drawn into the engine cylinder.

The combustible mixture of the air and petrol vapour is drawn into the engine at a rate which depends upon the engine speed and the position of the throttle valve.

If the air speed is increased by opening the throttle valve, the pressure difference between the float chamber and the spray tube will also increase, so drawing more fuel from the spray tube. Conversely, by reducing the air speed through the choke tube by closing the throttle valve, the pressure difference will be reduced and the amount of fuel being delivered will also be reduced.

The main jet of the carburettor is located between the float chamber and the spray tube. The main jet is calibrated so that it passes a fixed quantity of fuel to the spray tube. The required air/fuel ratio can be obtained by employing a size of jet which allows the correct amount of fuel to flow and combine with the air passing through the choke tube.

10.7.3 Limitations of the simple carburettor

The main defects or disadvantages or limitations of the simple carburettor are considered in the following :

(1) No provision is made for the cold starting of the engine as the depression in the venturi drops still further as a result of the lower cranking speed imposed by the starter. For cold starting purposes a rich mixture is required, but the simple carburettor is unable to supply such a mixture as it is not equipped with a starting device or choke.

(2) At very low engine speed there is insufficient air flow through the carburettor venturi to create a depression. The result is that no fuel flows from the float chamber, therefore the engine will not be idle.

(3) As engine speed increases, air pressure and therefore density decreases but the quantity of fuel will increase as a result of the greater pressure being exerted upon it. Therefore, the mixture

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strength will become progressively richer as engine speed increases.

(4) Similarly, as engine speed decreases, the air becomes more dense and the pressure acting on the fuel effectively becomes less. The mixture strength therefore becomes progressively weaker as engine speed decreases.

Because of the above defects the simple carbuettor is of little practical value on the variable speed engine used in modern vehicles. To prevent the mixture strength from becoming progressively richer as engine speed increases some, type of 'compensation' or 'correction' system must be employed within the corburettor.

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10.8 FUEL SUPPLY AND INJECTION SYSTEM

One of the most important parts of the CI engine is its injection system. Engine performance depends on a proper functioning of this injection system, which must supply, meter, inject, and atomize the

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fuel. In fact, it plays such an important part that it can be considered the heart of the CI engine.

10.8.1 Fuel injection pump

In the fuel pump system there are two ports in the barrel, both of which connect to a common fuel gallery feeding all the pumping elements. This gallery is supplied with clean fuel by the fuel feed lift pump and a suitable filter assembly. The plunger moves vertically in the barrel at a constant stroke, but the pumping action starts during the middle stage of the upward stroke of the plunger and not when the plunger just commences to rise. To enable the pump to vary the quantity of fuel delivered per stroke, a helix is formed in the plunger, which communicates with the plunger crown either by drilling in the centre of the plunger or a slot machined in the side. External means are provided so that the plunger can be rotated in its barrel during operation and thereby control the amount of fuel injected into each cylinder. Generally, the plunger assembly is operated by a cam through the medium of a tappet and returned by a coil spring. Fig. 10.14 illustrates the basic constructional detail of a single pumping element.

10.8.2. Sequence of operations

The system of operation of the pump element which comprises the plunger and barrel, is shown in Fig. 10.15. When the plunger is at the bottom of its stroke, fuel from the common gallery will flow through the spill port and through the inlet port of the barrel filling the chamber above and also the machined portion forming the helix of the plunger (Fig. 10.15a). As the plunger rises, it will reach a point where both inlet and spill ports are effectively cut off as shown in Fig. 10.15b. This is known as the spill cut off position and is the theoretical commencement of injection. In a fuel system which has been primed the barrel and pipes leading from the pump to the injectors, are filled with fuel. The fuel which is trapped above the plunger is forced towards the spring loaded delivery valve which is mounted on top of the pump barrel. The pressure exerted by the rising plunger upon the fuel causes this to lift the delivery valve and to enter the pipe line which connects the pump to the injector. As the pipe line is already full of fuel, the extra fuel which is being pumped in by the plunger, causes a rise in pressure throughout the line and lifts the injector valve. [Fig. 10.15c]. This action allows fuel to be sprayed into the combustion chamber of the engine. At this point in the operation fuel is being pumped into the line by the plunger at one end, and an

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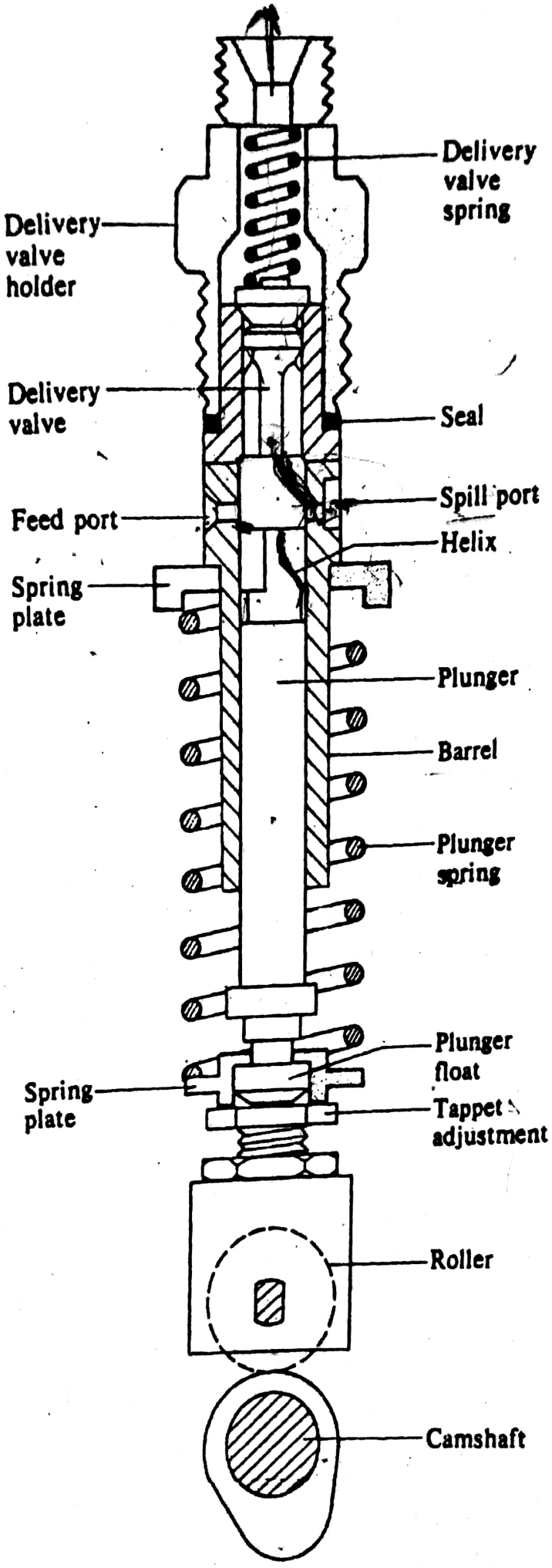

Fig. 10.14. Basic constructional detail of a pumping element.

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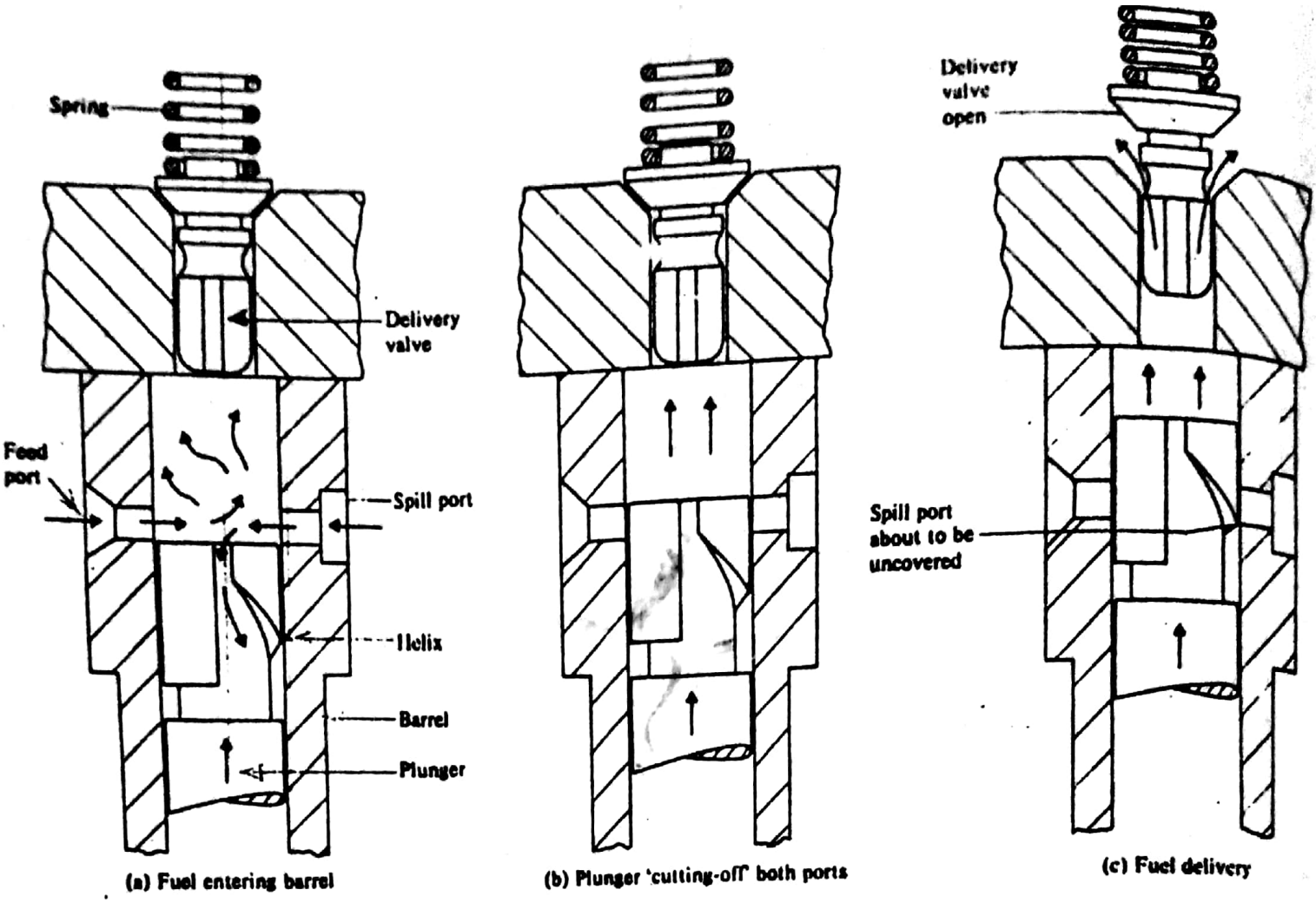

Fig. 10.15. Sequence of operations of fuel pump.

equal quantity is being pushed out through the injector valve at the other. Fuel is pumped until the helical edge of the plunger uncovers the spill port. Injection of fuel then ceases but the plunger continues to move in the upward direction as the delivery valve 'snaps' down on to its seat. As the plunger moves up fuel returns to the gallery of the injection pump via the spill port in the barrel. This spilling action continues until the plunger reaches the top of its stroke. Once the plunger has reached the end of its upward stroke a spring returns it to the bottom and fuel will flow through spill and inlet ports once again.

The stroke of the plunger is constant, therefore any variation of fuel delivery is effected by rotating the plunger, thus causing the helix to uncover the spill port sooner or later depending upon rotation. Immediately the helix uncovers the spill port, the fuel at high pressure above the plunger spills back to the common gallery, the delivery valve is returned to its seat by spring pressure, and injection stops.

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10.8.6 Injector

The function of an injector to receive correct amount of fuel at the correct time so that the fuel in a condition in which it can be effectively burnt. The injector may be considered as a type of automatic valve which:

(1) Ensures injection occurs at the correct pressure.

(2) Instantaneously stops injection as soon as injection pump pressure is reduced.

(3) Breaks up the fuel into a very fine spray, and forms the correct spray pattern to ensure thorough intermixing of fuel and air.

10.8.7 Working

A complete injector is shown in Fig 10.18 and it consists of a nozzle and nozzle holder which is located in the cylinder head and clamped to form a gas tight seal. Adequate cooling of the injector is ensured by careful design and positioning in the cylinder head. To prevent overheating, it is essential that the correct injector is fitted and joint washers are replaced each time the injector is removed. The injector nozzle consists of a valve and seat, the valve being held against its seat by a spindle and adjustable spring assembly. On top of the valve spring there is an adjustable screw, which ensures that the effective force exerted by the spring can be accurately adjusted. There is always some leakage of oil which is taken away through leak off connection. In fact, this amount of oil leak lubricates the valve and body.

Fuel is delivered to the injector body through a pipe line and fuel connections. The fuel then passes through holes which are formed in the nozzle holder body to an annular groove or gallery above the nozzle valve seat. At this point the shoulder of the valve is larger in diameter than the actual conical seating section of the nozzle valve, so that fuel, delivered under high pressure from the fuel injection pump, acting on the difference in area between the valve shoulder and the nozzle valve seat attempts to lift the valve in the nozzle body. When fuel pressure exceeds the preset spring tension, the nozzle valve lifts off its seating and fuel is forced through the hole(s) in the lower end of the nozzle body in the form of a finely atomised spray. At the end of injection the spring returns the nozzle valve to its seat. The loading on the tension spring is adjustable and determines at what pressure injection commences. Injection pressure is usually between 10.342 MPa and 18.099 MPa. It should be noted, however, that the spring tension only determines the pressure at which injection begins; pressure may increase upto approximately 70 MPa depending upon engine speed and load factors.

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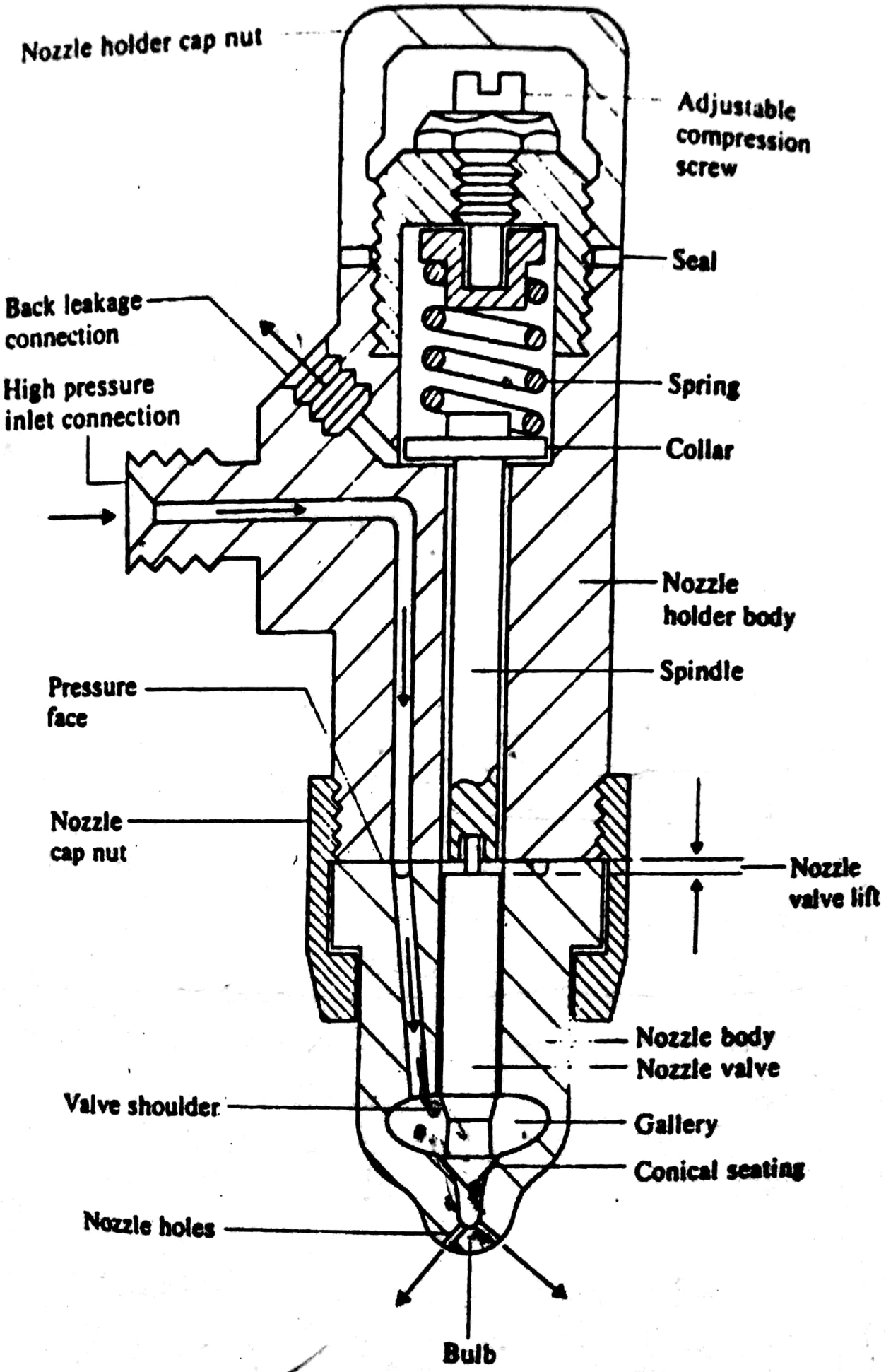

Fig. 10.18. The injector.

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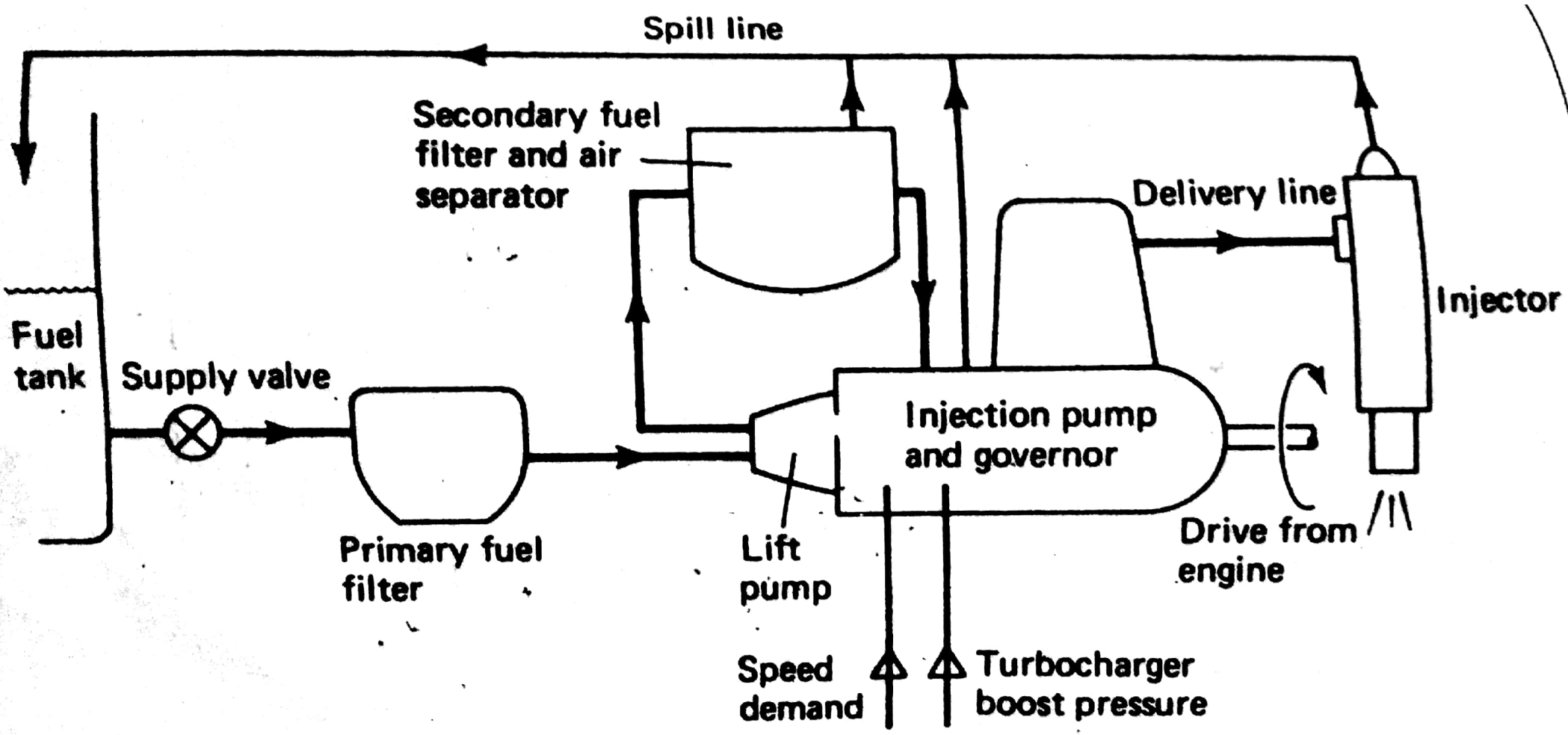

Fig. 10.19. Line diagram of fuel injection system for a CI engine.

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10.9. THE IGNITION SYSTEM

The internal combustion engine derives its power from the controlled burning of an air and fuel mixture, under compression, in the engine cylinder. In order to ignite this combustible mixture, at precisely the right moment, the engine must be provided with on ignition system. In the case of a spark ignition engine, the air/fuel mixture is ignited by a spark which 'jumps' across the gap between the electrodes in the sparking plug. The voltage required to bring about the electrical discharge across the sparking plug electrodes varies between 5000 and 30000 volts. The means for providing in the vehicles electrical system. The spark must also be provided at the correct time in the cycle of operation of the engine.

10.9.1. Battery Ignition System (Coil Ignition System)

A conventional ignition system consists of the following main components.

1. Battery : provides the source of supply of the low tension current.

2. Ignition coil : this acts as a transformer and converts low tension current into high tension at the spark plugs.

3. Distributor : a mechanism which distributes the high voltage to each of the cylinders in turn and in their correct order of firing.

4. Contact breaker points : function as a rotary switch to produce the high voltage that is required from the ignition coil.

5. Condenser (or capacitor) : this is connected in parallel to the contact breaker points and assists in preventing sparking at the points.

6. Spark plugs : the high tension current is passed through the central electrodes of the spark plug and the spark passes or jumps across the gap at the instant required.

10.9.2. The Ignition System

A conventional coil ignition (battery ignition) system is shown in Fig 10.20. The coil is a with primary or LT (Low Tension) winding of about 200 turns, and a secondary or HT (High Tension) winding of about 20 000 turns of fine wire, all wrapped round an iron core.

The HT output is shown in Fig 10.21. Initially 9 kV is needed to ionise the gas sufficiently before the spark jumps with a voltage drop of 2kV. As engine speed increases, the dwell period becomes shorter

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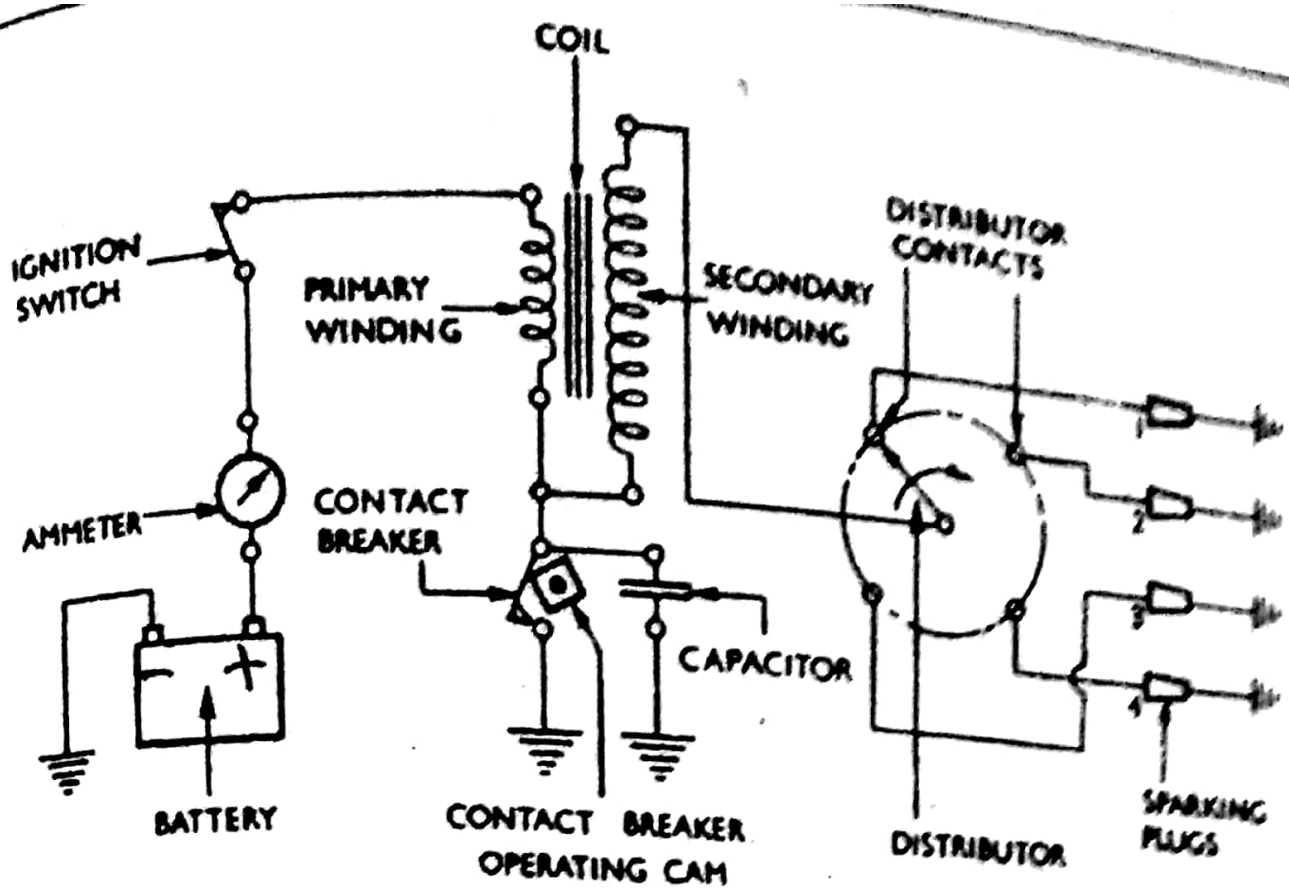

Fig. 10.20. Battery ignition system.

and the spark energy will be reduced. Such a system cam produce up to 400 sparks per second beyond this the spark energy becomes too low and this leads to misfiring. For higher spark rates, twin coil/contact breaker systems or alternatively electronic systems can be used. The current through the contact breaker can be reduced by using it to switch the base of a transister that controls the current to the LT winding, this prolong the life of the contact breaker. The contact breaker can be replaced by making use of opto-electronic, inductive or magnetic switching. However, all these systems use the coil in the same way as the contact breaker, but are less prone to wear and maladjustment. Without the mechanical limitations of a contact breaker, a coil ignition system can produce up to 800 sparks per second.

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10.9.4. Magneto ignition system

In a magneto ignition system there is no need to use a battery since a current is induced in the LT winding by the changing magnetic field. Again a voltage is induced in the HT winding, as before, it is significant only when dI/dt is large at the instant when the contact breaker opens. The air gap between the rotating magnet and the iron core of the coil should be as small as possible, so that the path for the magnetic flux has as low an impedance as possible. Magneto ignition is best suited to engines that are independent of a battery.

A magneto supplies the ignition voltage for the spark discharge independent of a battery or generator. Magneto ignition is commonly used in small four stroke and two stroke engines. Fig 10.23 illustrates the system and its operation. A time varying magnetic flux ϕ_o is set up in the ignition armature as the rotating permanent magnets on the pole wheel generate a current in the closed primary winding. This primary current generates an additional flux ϕ_I, giving a resultant flux $\varphi_R = \varphi_o + \varphi_I$. To generate the ignition voltage, the primary current flow is interrupted and the flux collapses rapidly from φ_R to φ_o, producing a high voltage pulse in the winding which is connected to the spark plug electrode. The current can be interrupted with contact breakers (breaker - triggered magneto) or with a transistor (semiconductor magneto). Since the flux generated by the rotating pole wheel depends on engine speed, the magnitude of the ignition voltage varies with speed.

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10.9.5. Comparison between battery and magneto ignition system

The operation of the magneto ignition system is exactly same as that of battery ignition system. The magneto is best at high speeds, and therefore, is widely used for sports and racing cars, aircrafts etc. In comparison to the coil ignition system magneto system is more costly but highly reliable. However, due to the poor starting characteristics and due to the fact that voltage generated is effected with the changes in the spark timing, almost invariably the coil ignition system is preferred to the magneto system. The coil ignition system requires more maintenance than the magneto system. It is also heavier than the magneto system.

Table 10.3. gives the comparison of the two systems.

S.No.	Points	Battery Ignition system	Magneto ignition system
1	Current supply source	Battery	Magnets
2	Wiring	Complicated	Simple
3	Space	More space	less space
4	Spark produced at the starting	Low speeds are quite strong	Low speeds are weak
5	Starting	Easy	Difficult
6	Spark strength	Decreases with the increase of speeds	Increases with the increase of speeds
7	Battery	In case of battery gets discharged the engine cannot be started	Due to absence of battery, there is no such difficulty
8	Cost	Less costly	More costly
9	Maintenance	Periodical maintenance is small	powerful voltage at high speed causes burning of the electrodes
10	Uses	Mostly employed in petrol cars and buses	More maintenance. Used in racing cars, scooter, motor cycles etc.

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Table 10.4. Comparison of SI and CI engines

S.No.	Points	SI engines	CI engines
1	Thermodynamic cycle	Ottocycle - (Heat supplied at constant volume)	For slow speed Diesel cycle - (Heat supplied at constant pressure) For high speed dual cycle - (Heat supplied partly at constant volume partly at constant pressure).
2	Combustion	Spark ignition	Compression ignition
3	Air fuel ratio	10 to 1 - 20 to 1	18 to 1 - 100 to 1
4	Fuel	Petrol High self ignition temperature High octane number	Diesel Low self ignition temperature High cetane number
5	Compression ratio	Upto 11. Upper limit of compression ratio fixed by antiknock quality of fuel (average value 7 to 9)	12 to 24. Upper limit of compression ratio is limited by thermal and mechanical stresses (average valve 15 to 18)
6	Speed (a) operating (b) piston	 High speed (2000 to 6000 rpm) High 16 m/s	 Low speed (400 rpm) Medium speed (400 to 1200 rpm) High speed (1200 to 3500 rpm) Upto 11 m/s
7	Fuel supply method	Cheap carburettor	Expensive injection
8	Control of Power	Quantity governing by throttle	Quality governing by rack
9.	Ignition	Ignition system with spark plug in the combustion chamber	Ignition due to high temperature caused by high compression of air
10.	Operating pressure (a) Compression pressure (b) Maximum pressure (c) bmep	 700 kPa to 1500 kPa 4.5 MPa to 6 MPa 1144 kPa	 3 MPa to 5 MPa 6 MPa to 12 MPa 2190 kPa
11	Calorific value	44 MJ/kg	42 MJ/kg
12	Starting	Easy, Low cranking effort	Difficult, high cranking effort
13	(a) Specific fuel consumption (b) Fuel economy (i) Full load (ii) Part load	0.27 kg/kWh Costly fuel, low density high calorific value. Less calories per litre Medium Poor	0.2 kg/kWh Cheap fuel, high density calorific value slightly low More calories per litre Good Good
14	Mass per unit power	Low (0.54 to 4.76 kg/kW)	High (3.4 to 13.6 kW)
15	Power per unit displacement	High (29.4 kW/litre)	Low (14.7 kW/litre)

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16	Supercharging	Limited by detonation. Used only in aircraft engines.	Widely used. Limited by blower power and mechanical and thermal stresses.
17	Cost of running	High	Low
18	Maintenance cost	Minor maintenance	Major overhaul required but less frequently.
19.	Fuel safety	Volatile fuel, more fire hazard.	Less volatile, less fire hazard. Safe in marine and other confined installations.
20	Odour and smoke	Less objectionable.	More objectionable.
21	Noise and vibration	Less	More idle noise mazor problem.
22.	Two stroke operation	Less suitable, fuel loss in scavenging. But small two stroke engines are used in scooters, motor cycles, mopeds due to simplicity and cheap.	More suitable, no fuel loss in scavenging.
	High powers	No	Yes
	Uses	Two stroke engines scooters, motor cycle, mopeds due to cheapter and simple engine passanger cars, small mobile, applications air crafts.	Buses, trucks, Locomotives, stationary generating plants. Heavy duty equipment like tractors, earth moving machinery ships.

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