SCHOOL OF ELECTRICAL AND ELECTRONICS

DEPARTMENT OF ELECTRICAL AND ELECTRONICS

SMEA1405 – Applied Thermodynamics

UNIT – I – FIRST LAW OF THERMODYNAMICS – SMEA1405

I. INTRODUCTION TO THERMODYNAMICS

Thermodynamics may be defined as follows:

- **Thermodynamics is an axiomatic science which deals with the relations among heat, work and properties of system which are in equilibrium. It describes state and changes in state of physical systems.**

Or

- **Thermodynamics is the science of the regularities governing processes of energy conversion.**

Or

- **Thermodynamics is the science that deals with the interaction between energy and material systems.**

Thermodynamics, basically entails four laws or axioms known as Zeroth, First, Second and Third law of thermodynamics.

- The First law throws light on concept of internal energy.
- The Zeroth law deals with thermal equilibrium and establishes a concept of temperature.
- The Second law indicates the limit of converting heat into work and introduces the principle of increase of entropy.
- The Third law defines the absolute zero of entropy.

Thermodynamic System: A system is a finite quantity of matter or a prescribed region of space (Refer Fig. 2.2)

Boundary. The actual or hypothetical envelope enclosing the system is the boundary of the system. The boundary may be fixed or it may move, as and when a system containing a gas is compressed or expanded. The boundary may be real or imaginary.
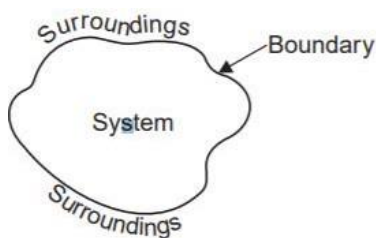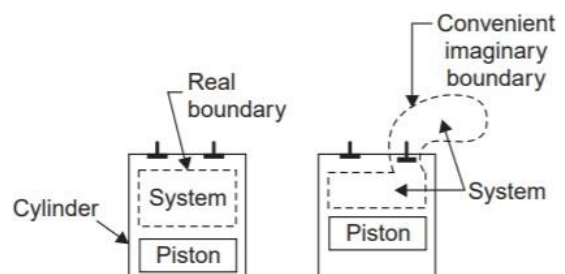

Fig. 1. Description of a system

Fig. 2. Real and Imaginary Boundary

Closed System

If the boundary of the system is impervious to the flow of matter, it is called a closed system. An example of this system is mass of gas or vapour contained in an engine cylinder, the boundary of which is drawn by the cylinder walls, the cylinder head and piston crown. Here the boundary is continuous and no matter may enter or leave.
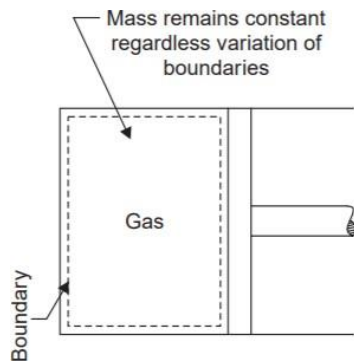

Fig. 3. Closed system

Fig. 4. Open system

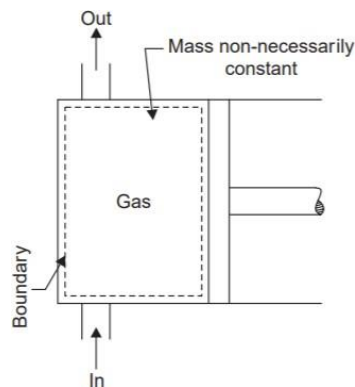## Open System

An open system is one in which matter flows into or out of the system. Most of the engineering systems are open.

Isolated System

An isolated system is that system which exchanges neither energy nor matter with any other system or with environment.

Phase. A phase is a quantity of matter which is homogeneous throughout in chemical composition and physical structure.

Homogeneous System

A system which consists of a single phase is termed as homogeneous system. Examples: Mixture of air and water vapour, water plus nitric acid and octane plus heptane.

Heterogeneous System

A system which consists of two or more phases is called a heterogeneous system. Examples: Water plus steam, ice plus water and water plus oil.

Macroscopic and microscopic points of view

S.No	Macroscopic approach	Microscopic approach
1	In this approach a certain quantity of matter is considered without taking into account the events occurring at molecular level. In other words, this approach to thermodynamics is concerned with gross or overall behaviour. This is known as classical thermodynamics.	The approach considers that the system is made up of a very large number of discrete particles known as molecules. These molecules have different velocities and energies. The values of these energies are constantly changing with time. This approach to thermodynamics which is concerned directly with the structure of the matter is known as statistical thermodynamics.
2	The analysis of macroscopic system requires simple mathematical formulae.	The behaviour of the system is found by using statistical methods as the number of molecules is very large. So advanced statistical and mathematical methods are needed to explain the changes in the system.
3	The values of the properties of the system are their average values. These properties like pressure and temperature can be measured very easily. The changes in properties can be felt by our senses	The properties like velocity, momentum, impulse, kinetic energy, force of impact etc. which describe the molecule cannot be easily measured by instruments. Our senses cannot feel them.
4	In order to describe a system only a few properties are needed.	Large number of variables are needed to describe a system. So, the approach is complicated.

Pure substance

A pure substance is one that has a homogeneous and invariable chemical composition even though there is a change of phase. In other words, it is a system which is (a) homogeneous in composition, (b) homogeneous in chemical aggregation. Examples: Liquid, water, mixture of liquid water and steam, mixture of ice and water. The mixture of liquid air and gaseous air is not a pure substance.

Thermodynamic equilibrium

A system is in thermodynamic equilibrium if the temperature and pressure at all points are same; there should be no velocity gradient; the chemical equilibrium is also necessary.

Thus, for attaining a state of thermodynamic equilibrium the following three types of equilibrium states must be achieved:

1. **Thermal equilibrium.** The temperature of the system does not change with time and has same value at all points of the system.

2. **Mechanical equilibrium**. There are no unbalanced forces within the system or between the surroundings. The pressure in the system is same at all points and does not change with respect to time.

3. **Chemical equilibrium.** No chemical reaction takes place in the system and the chemical composition which is same throughout the system does not vary with time.

Properties of systems

A property of a system is a characteristic of the system which depends upon its state, but not upon how the state is reached. There are two sorts of property:

1. **Intensive properties**. These properties do not depend on the mass of the system. Examples: Temperature and pressure.

2. **Extensive properties.** These properties depend on the mass of the system. Example: Volume. Extensive properties are often divided by mass associated with them to obtain the intensive properties. For example, if the volume of a system of mass m is V, then the specific volume of matter within the system is $\frac{V}{m} = v$ which is an intensive property.

State

State is the condition of the system at an instant of time as described or measured by its properties. Or each unique condition of a system is called a state.

Process

A process occurs when the system undergoes a change in a state or an energy transfer at a steady state.

Non-flow process

A process may be non-flow in which a fixed mass within the defined boundary is undergoing a change of state. Example: A substance which is being heated in a closed cylinder undergoes a non-flow process. Closed systems undergo non-flow processes.

Flow process

A process may be a flow process in which mass is entering and leaving through the boundary of an open system. In a steady flow process mass is crossing the boundary from surroundings at entry, and an equal mass is crossing the boundary at the exit so that the total mass of the system remains constant.

Quasi-static process. Quasi means 'almost'. A quasi-static process is also called a reversible process. This process is a succession of equilibrium states and infinite slowness is its characteristic feature.

Cycle

Any process or series of processes whose end states are identical is termed a cycle.. Fig. 5 shows such a cycle in which a system commencing at condition '1' changes in pressure and volume through a path 123 and returns to its initial condition '1'.
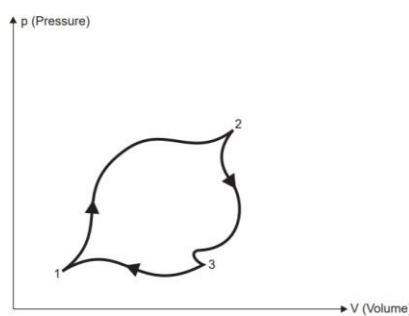

Fig.5. Cycle

Point function

When two properties locate a point on the graph (co-ordinate axes) then those properties are called as point function. Examples. Pressure, temperature, volume etc.

$$\int_1^2 dV = V_2 - V_1 \text{ (an } exact \text{ } differential\text{)}.$$

Path function

There are certain quantities which cannot be located on a graph by a point but are given by the area or so, on that graph. In that case, the area on the graph, pertaining to the particular process, is a function of the path of the process. Such quantities are called path functions. Examples. Heat, work etc. Heat and work are inexact differentials. Their change cannot be written as difference between their end states.

$$\text{Thus } \int_1^2 \delta Q \neq Q_2 - Q_1 \text{ and is shown as } _1Q_2 \text{ or } Q_{1-2}$$

$$\text{Similarly } \int_1^2 \delta W \neq W_2 - W_1, \text{ and is shown as } _1W_2 \text{ or } W_{1-2}$$

Zeroth law of thermodynamics

Zeroth law of thermodynamics states that if two systems are each equal in temperature to a third, they are equal in temperature to each other.

Energy

Energy is a general term embracing energy in transition and stored energy. The stored energy of a substance may be in the forms of mechanical energy and internal energy (other forms of stored energy may be chemical energy and electrical energy). Part of the stored energy may take the form of either potential energy (which is the gravitational energy due to height above a chosen datum line) or kinetic energy due to velocity. The balance part of the energy is known as internal energy. In a non-flow process usually, there is no change of potential or kinetic energy and hence change of mechanical energy will not enter the calculations. In a flow process, however, there may be changes in both potential and kinetic energy and these must be taken into account while considering the changes of stored energy. Heat and work are the forms of energy in transition. These are the only forms in which energy can cross the boundaries of a system. Neither heat nor work can exist as stored energy.

Work

Work is said to be done when a force moves through a distance. If a part of the boundary of a system undergoes a displacement under the action of a pressure, the work done W is the product of the force (pressure × area), and the distance it moves in the direction of the force. Fig. 6 (a) illustrates this with the conventional piston and cylinder arrangement, the heavy line defining the boundary of the system. Fig. 6 (b) illustrates another way in which work might be applied to a system. A force is exerted by the paddle as it changes the momentum of the fluid, and since this force moves during rotation of the paddle work is done.
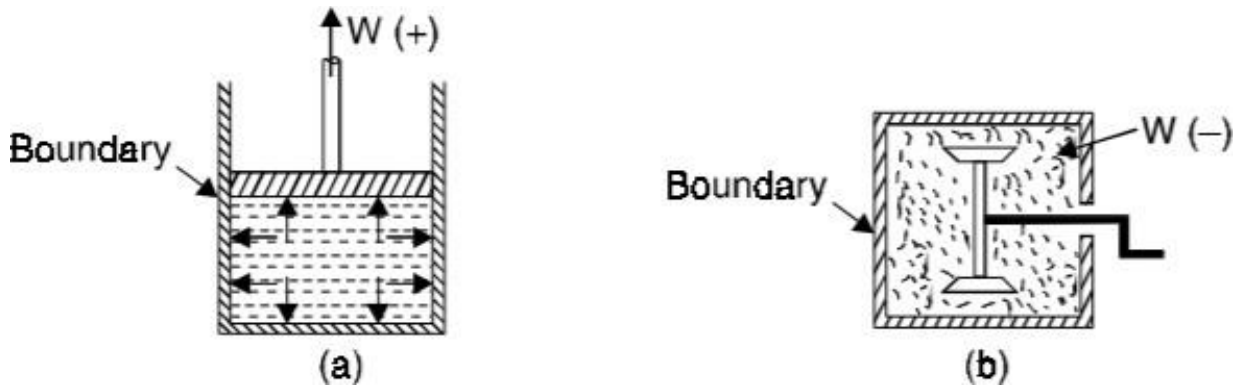
Fig. 6. Work Transfer

Internal energy (ΔU).

It is the heat energy stored in a gas. If a certain amount of heat is supplied to a gas the result is that temperature of gas may increase or volume of gas may increase thereby doing some external work or both temperature and volume may increase; but it will be decided by the conditions under which the gas is supplied heat. If during heating of the gas the temperature increases its internal energy will also increase.

Joule's law of internal energy states that internal energy of a perfect gas is a function of temperature only. In other words, internal energy of a gas is dependent on the temperature change only and is not affected by the change in pressure and volume.

First law of thermodynamics

When a system undergoes a thermodynamic cycle then the net heat supplied to the system from the surroundings is equal to net work done by the system on its surroundings.

$$\oint dQ = \oint dW$$

Application of first law to a process

When a process is executed by a system, the change in stored energy of the system is numerically equal to the net heat interactions minus the network interaction during the process.

$$\therefore \quad E_2 - E_1 = Q - W$$
$$\therefore \quad \Delta E = Q - W \qquad [\text{or } Q = \Delta E + W]$$
$$\int_1^2 d(Q - W) = \Delta E = E_2 - E_1$$

where E represents the total internal energy

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Application of first law of thermodynamics to non-flow or Closed system

Reversible Constant Volume (or Isochoric) 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. Fig. 7 shows the system and states before and after the heat addition at constant volume.
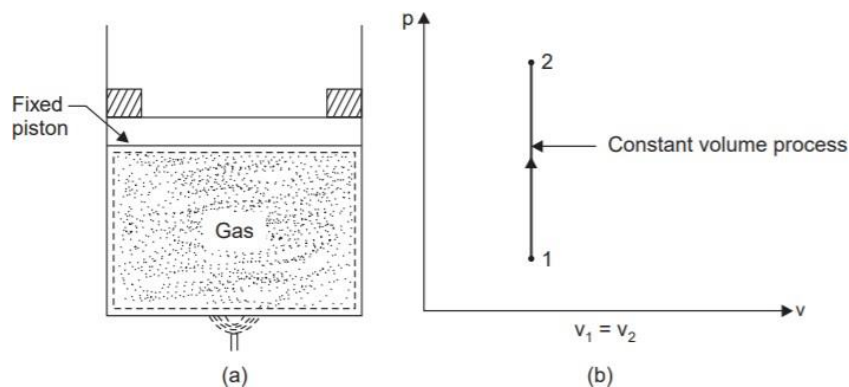

Fig. 7 Reversible constant volume process.

Considering mass of the working substance unity and applying first law of thermodynamics to the process,

$$Q = (u_2 - u_1) + W$$

The work done
$$W = \int_1^2 p\,dv = 0 \text{ as } dv = 0.$$

$$\therefore \quad Q = (u_2 - u_1) = c_v(T_2 - T_1)$$

where c_v = Specific heat at constant volume.

For mass, m, of working substance

$$Q = U_2 - U_1 = mc_v(T_2 - T_1)$$

Reversible Constant Pressure (or Isobaric) Process (p = constant).

It can be seen from Fig. 7 (b) that when the boundary of the system is inflexible as in a constant volume process, then the pressure rises when heat is supplied. Hence for a constant pressure process, the boundary must move against an external resistance as heat is supplied; for instance, a gas [Fig. 8 (a)] in a cylinder behind a piston can be made to undergo a constant pressure process. Since the piston is pushed through a certain distance by the force exerted by the gas, then the work is done by the gas on its surroundings. Fig. 8 shows the system and

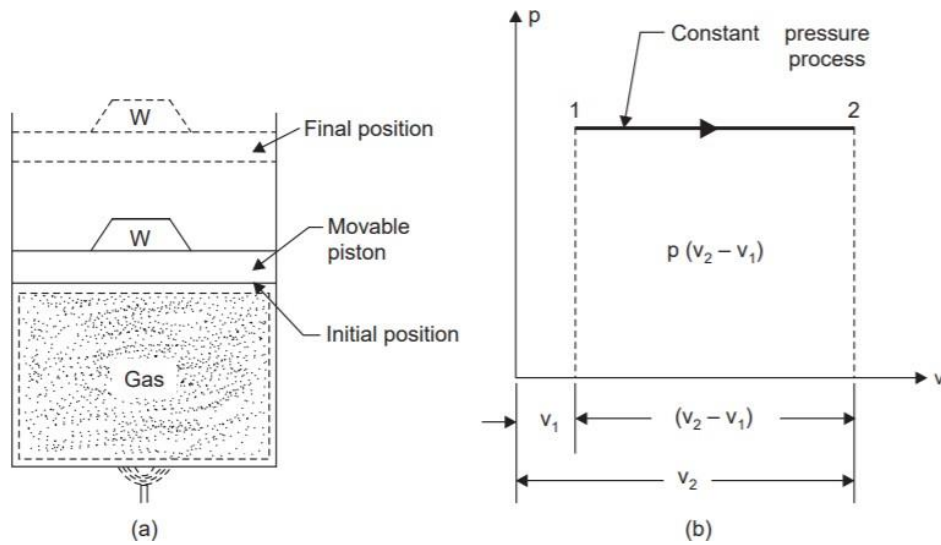states before and after the heat addition at constant pressure.

Fig. 8 Reversible constant pressure process.

Considering unit mass of working substance and applying first law of thermodynamics to the process,

$$Q = (u_2 - u_1) + W$$

The work done, $W = \int_1^2 pdv = p(v_2 - v_1)$

$\therefore \qquad Q = (u_2 - u_1) + p(v_2 - v_1) = u_2 - u_1 + pv_2 - pv_1$

$$= (u_2 + pv_2) - (u_1 + pv_1) = h_2 - h_1$$

or $\qquad Q = h_2 - h_1 = c_p \, (T_2 - T_1)$

where h = Enthalpy (specific), and

c_p = Specific heat at constant pressure.

For mass, m, of working substance

$$Q = H_2 - H_1 = mc_p \, (T_2 - T_1)$$

Reversible Temperature (or Isothermal) Process (pv = constant, T = constant) :

A process at a constant temperature is called an isothermal process. When a working substance in a cylinder behind a piston expands from a high pressure to a low pressure there is a tendency for the temperature to fall. 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 working substance continuously during the process. Fig. 9 shows the system and states before and after the heat addition at constant temperature.
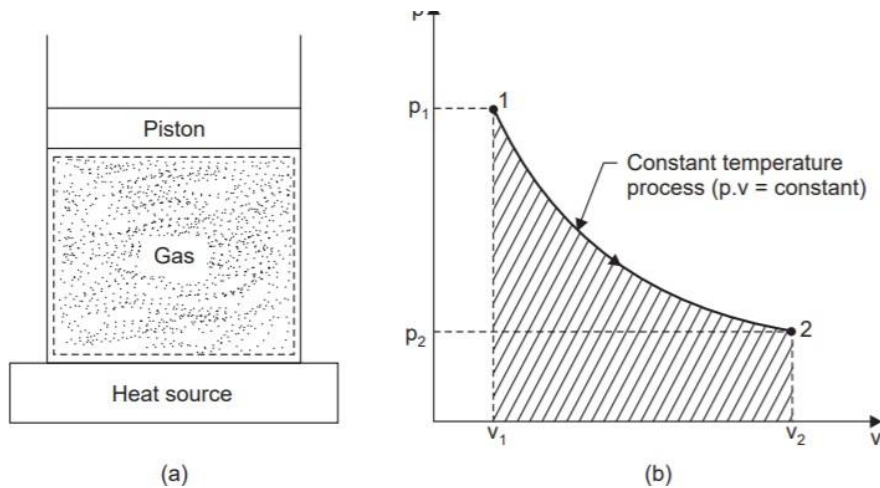
Fig. 9. Reversible isothermal process.

Considering unit mass of working substance and applying first law to the process,

$$Q = (u_2 - u_1) + W$$
$$= c_v\,(T_2 - T_1) + W$$
$$= 0 + W \qquad\qquad\qquad [\because\ T_2 = T_1]$$

The work done, $W = \int_1^2 pdv$

In this case pv = constant or $P = \dfrac{C}{v}$ (where C = constant)

$$\therefore \qquad W = \int_{v_1}^{v_2} C\,\frac{dv}{v} = C\big[\log_e v\big]_{v_1}^{v_2} = C\log_e \frac{v_2}{v_1}$$

The constant C can either be written as $p_1 v_1$ or as $p_2 v_2$, since
$$p_1 v_1 = p_2 v_2 = \text{constant},\ C$$

i.e., $$W = p_1 v_1 \log_e \frac{v_2}{v_1}\ \text{ per unit mass of working substance}$$

or $$W = p_2 v_2 \log_e \frac{v_2}{v_1}\ \text{ per unit mass of working substance}$$

$$\therefore \qquad Q = W = p_1 v_1 \log_e \frac{v_2}{v_1}$$

For mass, m, of the working substance
$$Q = p_1 V_1 \log_e \frac{V_2}{V_1}$$

or $$Q = p_1 V_1 \log_e \frac{p_1}{p_2} \qquad \left[\because\ \frac{V_2}{V_1} = \frac{p_1}{p_2}\right]$$

Reversible Adiabatic Process (pv^{γ} = constant) :

An adiabatic process is one in which no heat is transferred to or from the fluid during the process. Such a process can be reversible or irreversible.

Considering unit mass of working substance and applying first law to the process,

$$Q = (u_2 - u_1) + W$$
$$O = (u_2 - u_1) + W$$
$$W = (u_1 - u_2)\ \text{for any adiabatic process}$$

In an adiabatic expansion, the work done W by the fluid is at the expense of a reduction in the internal energy of the fluid. Similarly, in an adiabatic compression process all the work done on the fluid goes to increase the internal energy of the fluid. For an adiabatic process to take place, perfect thermal insulation for the system must be available.
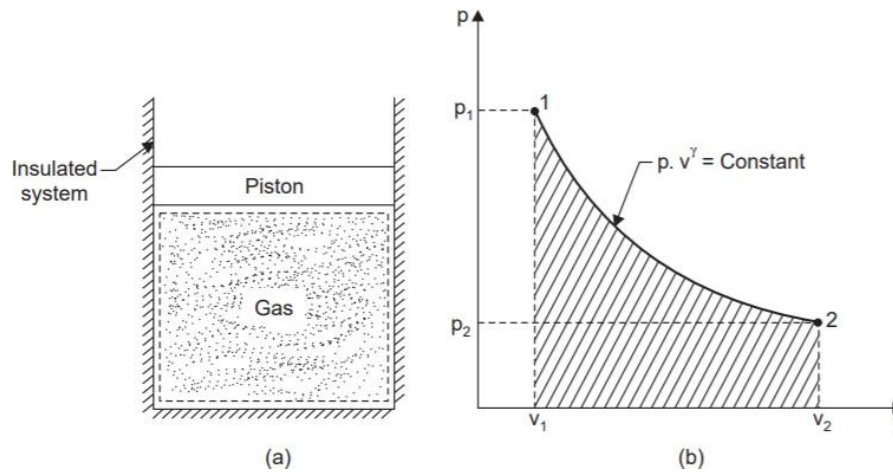

Fig. 9. Reversible adiabatic process

Polytropic Reversible Process (pv^n = constant) :

It is found that many processes in practice approximate to a reversible law of form pv^n = constant, where n is a constant. Both vapours and perfect gases obey this type of law closely in many non-flow processes. Such processes are internally reversible.

$$\text{Work done,} \quad W = \frac{p_1 v_1 - p_2 v_2}{n - 1}$$

$$W = \frac{R(T_1 - T_2)}{n - 1}$$

Summary of Processes for Perfect Gas (Unit mass)

Process	Index n	Heat added	$\int_1^2 pdv$	p, v, T relations	Specific heat, c
Constant pressure	$n = 0$	$c_p(T_2 - T_1)$	$p(v_2 - v_1)$	$\dfrac{T_2}{T_1} = \dfrac{v_2}{v_1}$	c_p
Constant volume	$n = \infty$	$c_v(T_2 - T_1)$	0	$\dfrac{T_1}{T_2} = \dfrac{p_1}{p_2}$	c_v
Constant temperature	$n = 1$	$p_1 v_1 \log_e \dfrac{v_2}{v_1}$	$p_1 v_1 \log_e \dfrac{v_2}{v_1}$	$p_1 v_1 = p_2 v_2$	∞
Reversible adiabatic	$n = \gamma$	0	$\dfrac{p_1 v_1 - p_2 v_2}{\gamma - 1}$	$p_1 v_1^{\gamma} = p_2 v_2^{\gamma}$ $\dfrac{T_2}{T_1} = \left(\dfrac{v_1}{v_2}\right)^{\gamma - 1}$ $= \left(\dfrac{p_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}$	0
Polytropic	$n = n$	$c_n(T_2 - T_1)$ $= c_v\left(\dfrac{\gamma - n}{1 - n}\right)$ $\times (T_2 - T_1)$ $= \dfrac{\gamma - n}{\gamma - 1} \times \text{work}$ done (non–flow)	$\dfrac{p_1 v_1 - p_2 v_2}{n - 1}$	$p_1 v_1^{n} = p_2 v_2^{n}$ $\dfrac{T_2}{T_1} = \left(\dfrac{v_1}{v_2}\right)^{n - 1}$ $= \left(\dfrac{p_2}{p_1}\right)^{\frac{n - 1}{n}}$	$c_n = c_v\left(\dfrac{\gamma - n}{1 - n}\right)$

Problems:

Problem 1: In an internal combustion engine, during the compression stroke the heat rejected to the cooling water is 50 kJ/kg and the work input is 100 kJ/kg. Calculate the change in internal energy of the working fluid stating whether it is a gain or loss.

Solution. Heat rejected to the cooling water, $Q = -50$ kJ/kg
(–ve sign since heat is rejected)
Work input, $\qquad\qquad\qquad W = -100$ kJ/kg
(–ve sign since work is supplied to the system)
Using the relation, $\qquad\qquad Q = (u_2 - u_1) + W$
$$-50 = (u_2 - u_1) - 100$$
$$u_2 - u_1 = -50 + 100 = 50 \text{ kJ/kg}$$
Hence, **gain in internal energy = 50 kJ/kg. (Ans.)**

Problem 2: Air enters a compressor at 105 Pa and 25°C having volume of 1.8 m3/kg and is compressed to 5 × 105 Pa isothermally. Determine: (i) Work done; (ii) Change in internal energy ; and (iii) Heat transferred.

Solution. Initial pressure of air, $\quad p_1 = 10^5$ Pa
Initial temperature of air, $\qquad\qquad T_1 = 25 + 273 = 298$ K
Final pressure of air, $\qquad\qquad\quad p_2 = 5 \times 10^5$ Pa
Final temperature of air, $\qquad\qquad T_2 = T_1 = 298$ K (isothermal process)
Since, it is a closed steady state process, we can write down the first law of thermodynamics

$$Q = (u_2 - u_1) + W \text{ ......per kg}$$

(*i*) *For isothermal process :*

$$W_{1-2} = \int_1^2 p \cdot dv = p_1 v_1 \log_e\left(\frac{p_1}{p_2}\right)$$

— Boundary

— Piston

Air

← W

Q
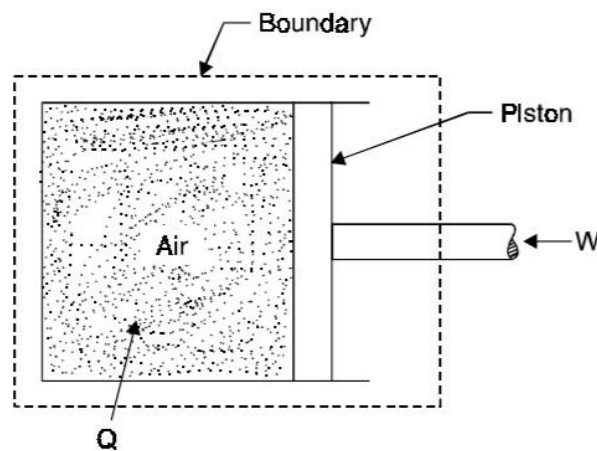
$p_1 v_1 = p_2 v_2$ for isothermal process

$\therefore \qquad\qquad W_{1-2} = -10^5 \times 1.8 \log_e\left(\dfrac{1 \times 10^5}{5 \times 10^5}\right)$

$$= -2.897 \times 10^5 = -289.7 \text{ kJ/kg.}$$

(– ve sign indicates that the work is supplied to the air)
$\therefore$ **Work done on the air = 289.7 kJ/kg. (Ans.)**
(*ii*) Since temperature is constant,
$\therefore \qquad\qquad\qquad u_2 - u_1 = 0$
$\therefore$ **Change in internal energy = zero. (Ans.)**
(*iii*) Again, $\qquad\qquad\qquad Q_{1-2} = (u_2 - u_1) + W$
$\qquad\qquad\qquad\qquad\quad = 0 + (-289.7) = -289.7$ kJ
(– ve sign indicates that heat is lost from the system to the surroundings)
$\therefore$ **Heat rejected = 289.7 kJ/kg. (Ans.)**

Problem 3: **A fluid system, contained in a piston and cylinder machine, passes through a complete cycle of four processes. The sum of all heat transferred during a cycle is – 340 kJ. The system completes 200 cycles per min. Complete the following table showing the method for each item, and compute the net rate of work output in kW**

Process	Q (kJ/min)	W (kJ/min)	ΔE (kJ/min)
1—2	0	4340	—
2—3	42000	0	—
3—4	– 4200	—	– 73200
4—1	—	—	—

Solution. Sum of all heat transferred during the cycle = – 340 kJ.

Number of cycles completed by the system = 200 cycles/min.

Process 1—2 :

$$Q = \Delta E + W$$
$$0 = \Delta E + 4340$$
$$\therefore \quad \Delta E = -4340 \text{ kJ/min.}$$

Process 2—3 :

$$Q = \Delta E + W$$
$$42000 = \Delta E + 0$$
$$\Delta E = 42000 \text{ kJ/min.}$$

Process 3—4 :

$$Q = \Delta E + W$$
$$-4200 = -73200 + W$$
$$\therefore \quad W = 69000 \text{ kJ/min.}$$

Process 4—1 :

$$\sum_{cycle} Q = -340 \text{ kJ}$$

The system completes 200 cycles/min

$$\because \quad Q_{1-2} = Q_{2-3} + Q_{3-4} + Q_{4-1} = -340 \times 200 = -68000 \text{ kJ/min}$$
$$0 + 42000 + (-4200) + Q_{4-1} = -68000$$
$$Q_{4-1} = -105800 \text{ kJ/min.}$$

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

$$\Delta E_{1-2} + \Delta E_{2-3} + \Delta E_{3-4} + \Delta E_{4-1} = 0$$
$$-4340 + 42000 + (-73200) + \Delta E_{4-1} = 0$$
$$\therefore \quad \Delta E_{4-1} = 35540 \text{ kJ/min.}$$
$$\therefore \quad W_{4-1} = Q_{4-1} - \Delta E_{4-1}$$
$$= -105800 - 35540 = -141340 \text{ kJ/min}$$

The completed table is given below :

Process	Q(kJ/min)	W(kJ/min)	ΔE(kJ/min)
1—2	0	4340	– 4340
2—3	42000	0	42000
3—4	– 4200	69000	– 73200
4—1	– 105800	– 141340	35540

Since

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

Rate of work output = – 68000 kJ/min $= -\dfrac{68000}{60}$ kJ/s or kW

$$= 1133.33 \text{ kW.} \quad \textbf{(Ans.)}$$

Problem 4: **0.44 kg of air at 180°C expands adiabatically to three times its original volume and during the process, there is a fall in temperature to 15°C. The work done during the process is 52.5 kJ. Calculate cp and cv**

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Solution:

Mass of air, $m = 0.44$ kg

Initial temperature, $T_1 = 180 + 273 = 453$ K

$$\text{Ratio} = \frac{V_2}{V_1} = 3$$

Final temperature, $T_2 = 15 + 273 = 288$ K

Work done during the process, $W_{1-2} = 52.5$ kJ

$$c_p = ?, \ c_v = ?$$

For adiabatic process, we have

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

$$\frac{288}{453} = \left(\frac{1}{3}\right)^{\gamma - 1} \qquad \text{or} \qquad 0.6357 = (0.333)^{\gamma-1}$$

or   Taking log on both sides, we get

$$\log_e (0.6357) = (\gamma - 1) \log_e (0.333)$$
$$- 0.453 = (\gamma - 1) \times (- 1.0996)$$

∴ $\gamma = \dfrac{0.453}{1.0996} + 1 = 1.41$

Also, $\dfrac{c_p}{c_v} = \gamma = 1.41$

Work done during adiabatic process,

$$W_{1-2} = \frac{mR(T_1 - T_2)}{\gamma - 1}$$

∴ $52.5 = \dfrac{0.44\, R(453 - 288)}{(1.41 - 1)}$

∴ $R = \dfrac{52.5\,(1.41 - 1)}{0.44\,(453 - 288)} = 0.296$

∴ $c_p - c_v = 0.296$ $[\because \ R = c_p - c_v]$

Also $\dfrac{c_p}{c_v} = 1.41$ or $c_p = 1.41\, c_v$

∴ $1.41\, c_v - c_p = 0.296$

or $c_v = 0.722$ **kJ/kg K.**   **(Ans.)**

and $c_p = 1.018$ **kJ/kg K.**   **(Ans.)**

Problem 5: **0.1 m3 of an ideal gas at 300 K and 1 bar is compressed adiabatically to 8 bar. It is then cooled at constant volume and further expanded isothermally so as to reach the condition from where it started. Calculate : (i) Pressure at the end of constant volume cooling. (ii) Change in internal energy during constant volume process. (iii) Net work done and heat transferred during the cycle. Assume cp = 14.3 kJ/kg K and cv = kJ/kg K**

Solution. *Given :* $V_1 = 0.1$ m³ ; $T_1 = 300$ K ; $p_1 = 1$ bar ; $c_p = 14.3$ kJ/kg K ;
$c_v = 10.2$ kJ/kg K.

Refer to Fig. 4.24.

(*i*) **Pressure at the end of constant volume cooling, p_3 :**

$$\gamma = \frac{c_p}{c_v} = \frac{14.3}{10.2} = 1.402$$

Characteristic gas constant,

$$R = c_p - c_v = 14.3 - 10.2 = 4.1 \text{ kJ/kg K}$$

Considering *process 1-2*, we have :

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

$$V_2 = V_1 \times \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} = 0.1 \times \left(\frac{1}{8}\right)^{\frac{1}{1.402}} = 0.0227 \text{ m}^3$$

Also, $\dfrac{T_2}{T_1} = \left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\dfrac{8}{1}\right)^{\frac{1402-1}{1402}} = 1.815$

$$T_2 = T_1 \times 1.815 = 300 \times 1.815 = 544.5 \text{ K}$$

Considering *process 3–1*, we have

$$p_3 V_3 = p_1 V_1$$

$\therefore \qquad p_3 = \dfrac{p_1 V_1}{V_3} = \dfrac{1 \times 0.1}{0.0227} = 4.4 \text{ bar.}$ (**Ans.**) $(\because \ V_3 = V_2)$

(Figure 4.24: p–V diagram showing process with points 1, 2, 3; axes p(bar) with values 8, p_3, 1 and V(m³) with $V_2 = V_3$ and 0.1. Curves labelled $pV^\gamma = C$, $pV = C$, and $V = C$.)

Fig. 4.24

(*ii*) **Change in internal energy during constant volume process, $(U_3 - U_2)$:**

Mass of gas, $\qquad m = \dfrac{p_1 V_1}{R T_1} = \dfrac{(1 \times 10^5) \times 0.1}{(4.1 \times 1000) \times 300} = 0.00813$ kg

$\therefore$ Change in internal energy during *constant volume process 2–3*,

$$U_3 - U_2 = m c_v (T_3 - T_2)$$
$$= 0.00813 \times 10.2 \,(300 - 544.5) \qquad (\because \ T_3 = T_1)$$
$$= -20.27 \text{ kJ.} \text{ (Ans.)}$$

(– ve sign means *decrease* in internal energy)

• During constant volume cooling process, temperature and hence internal energy is reduced. This *decrease in internal energy equals to heat flow to surroundings since work done is zero.*

(*iii*) **Net work done and heat transferred during the cycle :**

$$W_{1-2} = \frac{p_1 V_1 - p_2 V_2}{\gamma - 1} = \frac{mR(T_1 - T_2)}{\gamma - 1}$$

$$= \frac{0.00813 \times 4.1(300 - 544.5)}{1.402 - 1} = -20.27 \text{ kJ}$$

$W_{2-3} = 0$ \qquad\qquad ... since volume remains constant

$$W_{3-1} = p_3 V_3 \log_e \left(\frac{V_1}{V_3}\right) = p_1 V_1 \log_e \left(\frac{p_3}{p_1}\right) \qquad (\because \ p_3 V_3 = p_1 V_1)$$

$$= (1 \times 10^5) \times 0.1 \times \log_e \left(\frac{4.4}{1}\right)$$

$$= 14816 \text{ Nm (or J) or } 14.82 \text{ kJ}$$

$\therefore$ **Net work done** $= W_{1-2} + W_{2-3} + W_{3-1}$
$$= (-20.27) + 0 + 14.82 = -5.45 \text{ kJ}$$

–ve sign indicates that work has been done *on the system.* (**Ans.**)

For a cyclic process : $\oint \delta Q = \oint \delta W$

$\therefore$ Heat transferred during the complete cycle = – **5.45 kJ**

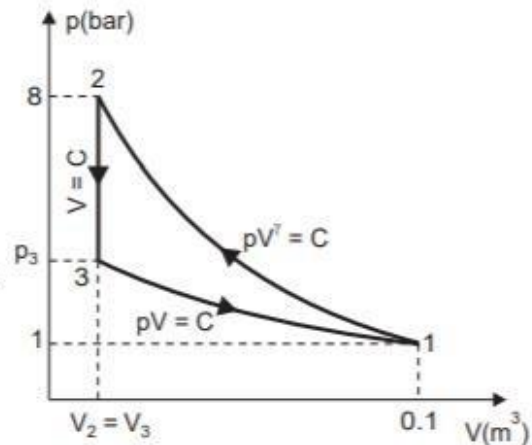–ve sign means heat has been *rejected i.e., lost from the system.* (**Ans.**)

APPLICATION OF FIRST LAW TO STEADY FLOW PROCESS

Steady Flow Energy Equation (S.F.E.E.)

In many practical problems, the rate at which the fluid flows through a machine or piece of apparatus is constant. This type of flow is called steady flow.

Assumptions : The following assumptions are made in the system analysis : (i) The mass flow through the system remains constant. (ii) Fluid is uniform in composition. (iii) The only interaction between the system and surroundings are work and heat. (iv) The state of fluid at any point remains constant with time. (v) In the analysis only potential, kinetic and flow energies are considered.

Fig. 10. shows a schematic flow process for an open system. An open system is one in which both mass and energy may cross the boundaries. A wide interchange of energy may take place within an open system. Let the system be an automatic engine with the inlet manifold at the first state point and exhaust pipe as the second point. There would be an interchange of chemical energy in the fuel, kinetic energy of moving particles, internal energy of gas and heat transferred and shaft work within the system. From Fig. 4.30 it is obvious that if there is no variation of flow of mass or energy with time across the boundaries of the system the steady flow will prevail. The conditions may pass through the cyclic or non-cyclic changes within the system. As a result, the mass entering the system equals the mass leaving, also energy entering the system equals energy leaving.
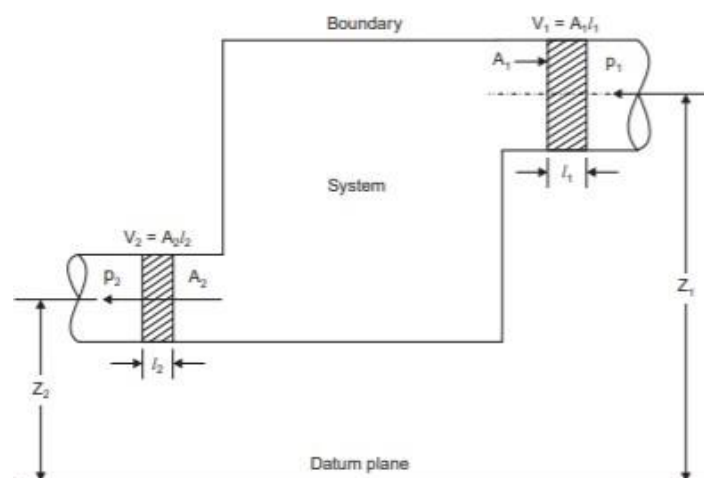

Fig.10. Open system

The steady flow equation can be expressed as follows :

$$u_1 + \frac{C_1^2}{2} + Z_1g + p_1v_1 + Q = u_2 + \frac{C_2^2}{2} + Z_2g + p_2v_2 + W$$

$$(u_1 + p_1v_1) + \frac{C_1^2}{2} + Z_1g + Q = (u_2 + p_2v_2) + \frac{C_2^2}{2} + Z_2g + W$$

$$h_1 + \frac{C_1^2}{2} + Z_1g + Q = h_2 + \frac{C_2^2}{2} + Z_2g + W \qquad [\because$$

If Z_1 and Z_2 are neglected, we get

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

where, Q = Heat supplied (or entering the boundary) per kg of fluid,

W = Work done by (or work coming out of the boundary) 1 kg of fluid,

C = Velocity of fluid , Z = Height above datum, p = Pressure of the fluid, u = Internal energy per kg of fluid, and pv = Energy required for 1 kg of fluid.

This equation is applicable to any medium in any steady flow. It is applicable not only to rotary machines such as centrifugal fans, pumps and compressors but also to reciprocating machines such as steam engines. In a steady flow the rate of mass flow of fluid at any section is the same as at any other section. Consider any section of cross-sectional area A, where the fluid velocity is C, the rate of volume flow past the section is CA. Also, since mass flow is volume flow divided by specific volume,

$$\therefore \qquad \text{Mass flow rate, } \dot{m} = \frac{CA}{v}$$

(where v = *Specific volume* at the section)

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

With reference to Fig. 4.30.

$$\therefore \qquad \dot{m} = \frac{C_1 A_1}{v_1} = \frac{C_2 A_2}{v_2}$$

Problem 6. In a gas turbine unit, the gases flow through the turbine is 15 kg/s and the power developed by the turbine is 12000 kW. The enthalpies of gases at the inlet and outlet are 1260 kJ/kg and 400 kJ/kg respectively, and the velocity of gases at the inlet and outlet are 50 m/s and 110 m/s respectively. Calculate : (i) The rate at which heat is rejected to the turbine, and (ii) The area of the inlet pipe given that the specific volume of the gases at the inlet is 0.45 m3/kg

Solution. Rate of flow of gases, $\dot{m} = 15$ kg/s

Volume of gases at the inlet, $v = 0.45$ m³/kg

Power developed by the turbine, $P = 12000$ kW

∴ Work done, $W = \dfrac{12000}{15} = 800$ kJ/kg

Enthalpy of gases at the inlet, $h_1 = 1260$ kJ/kg

Enthalpy of gases at the oulet, $h_2 = 400$ kJ/kg

Velocity of gases at the inlet, $C_1 = 50$ m/s

Velocity of gases at the outlet, $C_2 = 110$ m/s.

Gases in   Gases out

Gas Turbine

W_{out}

Q_{out} Boundary
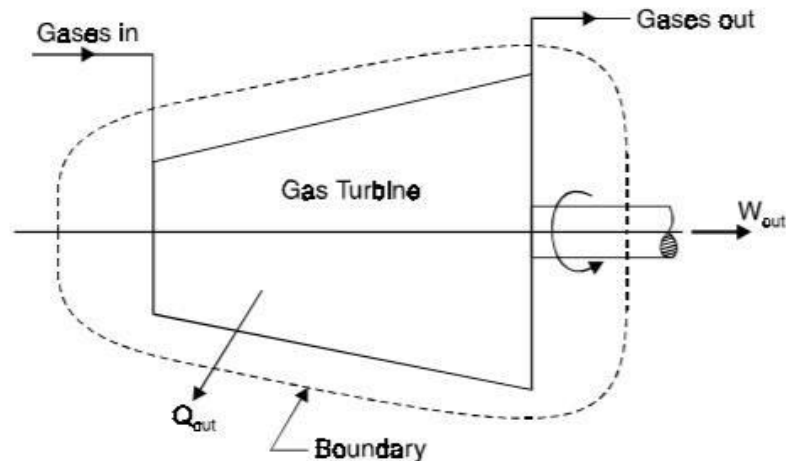
(*i*) **Heat rejected, Q :**

Using the flow equation,

$$h_1 + \frac{C_1^2}{2} + Q = h_2 + \frac{C_2^2}{2} + W \qquad \text{...(i)} \ [\because \ Z_1 = Z_2]$$

Kinetic energy at inlet $= \dfrac{C_1^2}{2} = \dfrac{50^2}{2}$ m²/s² $= \dfrac{50^2 \ \text{kg m}^3}{2 \ \text{s}^2 \text{kg}} = 1250$ Nm/kg $= 1.25$ kJ/kg

Kinetic energy at outlet $= \dfrac{C_1^2}{2} = \dfrac{110^2}{2 \times 1000} = 6.05$ kJ/kg

Substituting these values in eqn. (*i*), we get

$$1260 + 1.25 + Q = 400 + 6.05 + 800$$

∴ $\qquad\qquad\qquad Q = -55.2$ kJ/kg

Heat rejected $\qquad\qquad = +55.2$ kJ/kg $= 55.2 \times 15$ kJ/s $= \textbf{828 kW.}$ **(Ans.)**

(*ii*) **Inlet area, A :**

Using the relation,

$$\dot{m} = \frac{CA}{v}$$

∴ $\qquad\qquad A = \dfrac{v\dot{m}}{C} = \dfrac{0.45 \times 15}{50} = \textbf{0.135 m}^2.$ **(Ans.)**

Problem 7. In an air compressor air flows steadily at the rate of 0.5 kg/s through an air compressor. It enters the compressor at 6 m/s with a pressure of 1 bar and a specific volume of 0.85 m3/kg and leaves at 5 m/s with a pressure of 7 bar and a specific volume of 0.16 m3/kg. The internal energy of the air leaving is 90 kJ/kg greater than that of the air entering. Cooling water in a jacket surrounding the cylinder absorbs heat from the air at the rate of 60 kJ/s. Calculate : (i) The power required to drive the compressor ; (ii) The inlet and output pipe cross-sectional areas. Solution. Air flow rate through the compressor, m = 0.5 kg/s Velocity of air at the inlet to compressor, C1 = 6 m/s Velocity

of air at the outlet of compressor, C2 = 5 m/s Pressure of air at the inlet to the compressor, p1 = 1 bar
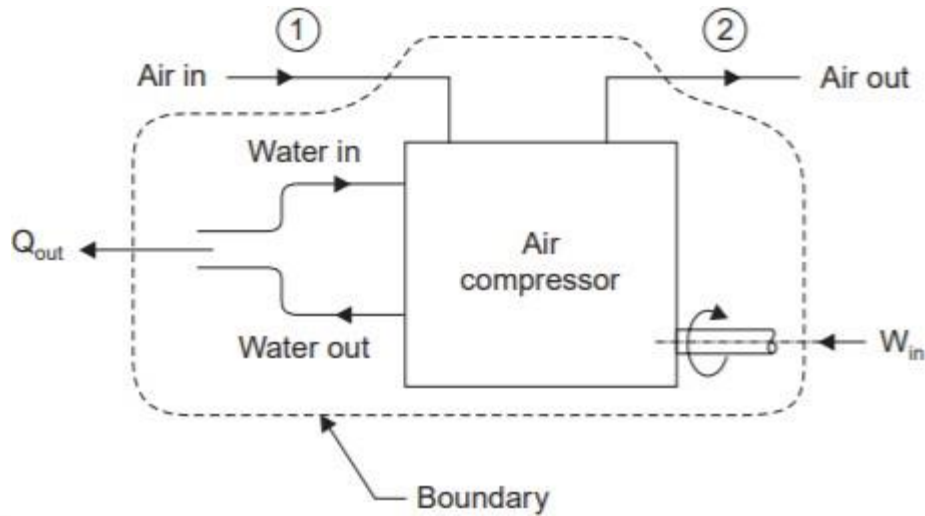
Pressure of air at outlet to the compressor, p_2 = 7 bar

Specific volume of air at inlet to the compressor, v_1 = 0.85 m³/kg

Specific volume of air at outlet to the compressor, v_2 = 0.16 m³/kg

Difference of internal energy at the outlet and inlet of the compressor,

$$(u_2 - u_1) = 90 \text{ kJ/kg}$$

Heat rejected by air (to cooling water),

$$Q = -\frac{60}{0.5} = -120 \text{ kJ/kg}.$$

(i) Power required to drive the compressor :

Using the steady flow energy equation,

$$u_1 + \frac{C_1^2}{2} + p_1 v_1 + Q = u_2 + \frac{C_2^2}{2} + p_2 v_2 + W$$

∴ $$W = (u_1 - u_2) + \left(\frac{C_1^2}{2} - \frac{C_2^2}{2}\right) + (p_1 v_1 - p_2 v_2) + Q$$

$$= -90 + \frac{1}{1000}\left(\frac{6^2}{2} - \frac{5^2}{2}\right) + \frac{10^5}{1000}(1 \times 0.85 - 7 \times 0.16) + (-120)$$

$$= -90 + 0.0055 - 27 - 120 = -237 \text{ kJ/kg (app.).}$$

(Note that the change in kinetic energy is negligibly small in comparison with the other terms).

i.e., Work input required = 237 kJ/kg = 237 × 0.5 kJ/s = 118.5 kW

Hence, **power required to drive the compressor = 118.5 kW.** **(Ans.)**

(ii) Inlet and outlet pipe cross-sectional areas, A_1 and A_2 :

Using the relation,

$$\dot{m} = \frac{CA}{v}$$

∴ $$A_1 = \frac{\dot{m} v_1}{C_1} = \frac{0.5 \times 0.85}{6} \text{ m}^2 = 0.0708 \text{ m}^2$$

i.e., Inlet pipe cross-sectional area, A_1 = **0.0708 m².** **(Ans.)**

Again, $$A_2 = \frac{\dot{m} v_2}{C_2} = \frac{0.5 \times 0.16}{5} \text{ m}^2 = 0.016 \text{ m}^2.$$

i.e., Outlet pipe cross-sectional area, A_2 = **0.016 m².** **(Ans.)**

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UNIT – II – SECOND LAW OF THERMODYNAMICS– SMEA1405

II. SECOND LAW OF THERMODYNAMICS AND ITSAPPLICATIONS

Limitations of first law of thermodynamics and introduction to second law

It has been observed that energy can flow from a system in the form of heat or work. The first law of thermodynamics sets no limit to the amount of the total energy of a system which can be caused to flow out as work. A limit is imposed, however, as a result of the principle enunciated in the second law of thermodynamics which states that heat will flow naturally from one energy reservoir to another at a lower temperature, but not in opposite direction without assistance. This is very important because a heat engine operates between two energy reservoirs at different temperatures. Further the first law of thermodynamics establishes equivalence between the quantity of heat used and the mechanical work but does not specify the conditions under which conversion of heat into work is possible, neither the direction in which heat transfer can take place. This gap has been bridged by the second law of thermodynamics.

Performance of heat engines and reversed heat engines

A heat engine is used to produce the maximum work transfer from a given positive heat transfer. The measure of success is called the thermal efficiency of the engine and is

$$\text{Thermal efficiency, } \eta_{th} = \frac{W}{Q_1}$$

defined by the ratio :

where, W = Net work transfer from the engine, and Q_1 = Heat transfer to engine.

For a reversed heat engine acting as a refrigerator when the purpose is to achieve the maximum heat transfer from the cold reservoir, the measure of success is called the co-efficient of performance (C.O.P.). It is defined by the ratio :

$$\text{Co-efficient of performance, } (C.O.P.)_{ref.} = \frac{Q_2}{W}$$

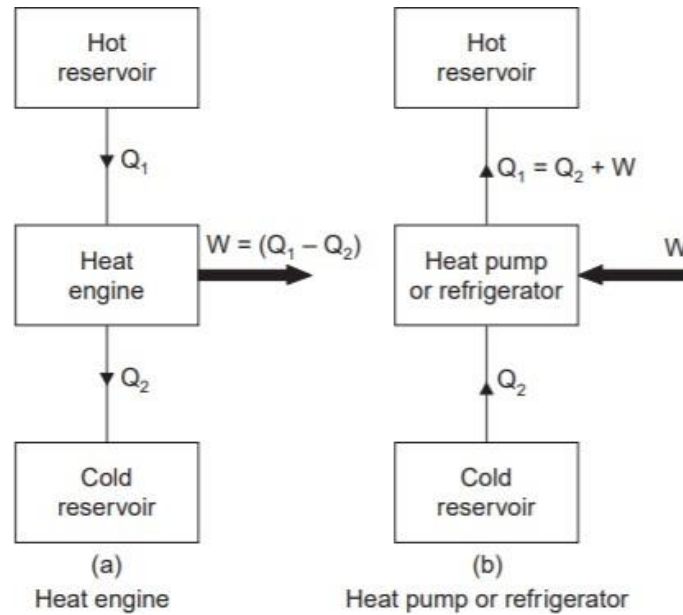where, Q_1 = Heat transfer to hot reservoir, and W = Net work transfer to the heat pump.

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Fig.1

In all the above three cases application of the first law gives the relation $Q_1 - Q_2 = W$, and this can be used to rewrite the expressions for thermal efficiency and co-efficient of performance solely in terms of the heat transfers.

$$\eta_{th} = \frac{Q_1 - Q_2}{Q_1}$$

$$(C.O.P.)_{ref} = \frac{Q_2}{Q_1 - Q_2}$$

$$(C.O.P.)_{heat\ pump} = \frac{Q_1}{Q_1 - Q_2}$$

It may be seen that η_{th} is always less than unity and $(C.O.P.)_{heat\ pump}$ is always greater thanunity.

Reversible processes:

A reversible process should fulfill the following conditions :

      1. The process should not involve friction of any kind.

      2. Heat transfer should not take place with finite temperature difference.

      3. The energy transfer as heat and work during the forward process should be identically equal to energy transfer as heat and work during the reversal of the process.

      4. There should be no free or unrestricted expansion.

      5. There should be no mixing of the fluids. 6. The process must proceed in a series ofequilibrium states.

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Some examples of ideal reversible processes are :

        (i) Frictionless adiabatic expansion or compression ;

        (ii) Frictionless isothermal expansion or compression ;

        (iii) Condensation and boiling of

liquids.Some examples of irreversible

processes are :

(i)   Combustion process;

(ii)   Mixing of two fluids ;

(iii)   All processes involving friction;

(iv)   Flow of electric current through a resistance ;

(v)   Heat flow from a higher temperature to lower temperature.

Reversible processes are preferred because the devices which produce work such as engines and turbines, reversible process of the working fluid delivers more work than the corresponding irreversible processes. Also in case of fans, compressors, refrigerators and pumps less power input is required when reversible processes are used in place of corresponding irreversible ones. In thermodynamic analysis concept of reversibility, though hypothetical, is very important because a reversible process is the most efficient process. Only reversible processes can be truely represented on property diagrams. Thermodynamic reversibility can only be approached but can never be achieved. Thus the main task of the engineer is to design the system which will evolve approximate reversible processes.

Statements of second law of thermodynamics:

The second law of thermodynamics has been enunciated meticulously by Clausius, Kelvin and Planck in slightly different words although both statements are basically identical.Each statement is based on an irreversible process. The first considers transformation of heat between two thermal reservoirs while the second considers the transformation of heat into work.

Clausius Statement:

"It is impossible for a self-acting machine working in a cyclic process unaided by any external agency, to convey heat from a body at a lower temperature to a body at a higher temperature"

Kelvin-Planck Statement:

"It is impossible to construct an engine, which while operating in a cycle produces no other effect except to extract heat from a single reservoir and do equivalent amount of work".

Equivalence of Clausius Statement to the Kelvin-Planck Statement:

Consider a higher temperature reservoir T_1 and low temperature reservoir T_2. Fig. shows a heat pump which requires no work and transfers an amount of Q_2 from a low temperature to a higher temperature reservoir (in violation of the Clausius statement). Let an amount of heat Q_1 (greater than Q_2) be transferred from high temperature reservoir to heat engine which devolops a net work, $W = Q_1 - Q_2$ and rejects Q_2 to the low temperature reservoir. Since there is no heat interaction with the low temperature, it can be eliminated. The combined system of the heat engine and heat pump acts then like a heat engine exchanging heat with a single reservoir, which is the violation of the Kelvin-Planck statement.
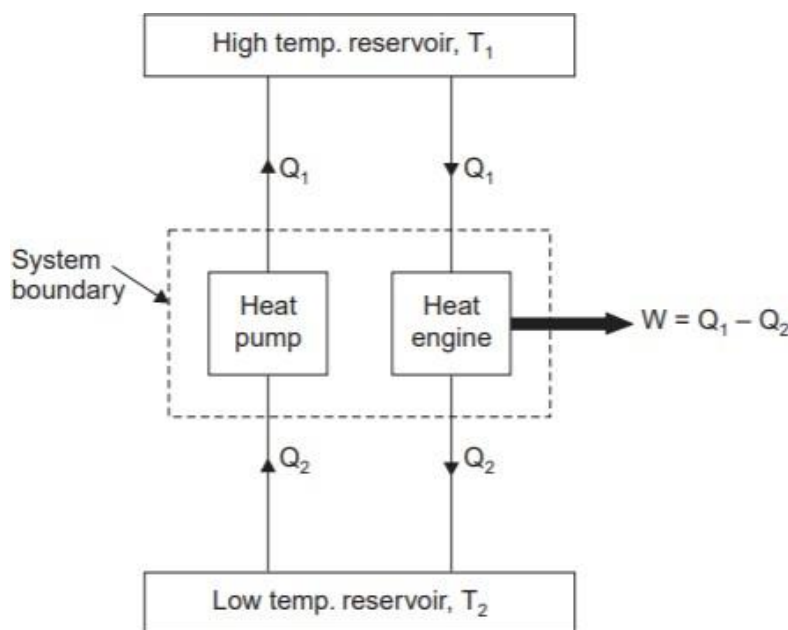

Fig. 2 Equivalence of Clausius statement to Kelvin-Planck statement

Clausius inequality:

When a reversible engine uses more than two reservoirs the third or higher numbered

reservoirs will not be equal in temperature to the original two. Consideration of expression for efficiency of the engine indicates that for maximum efficiency, all the heat transfer should

take place at maximum or minimum reservoir temperatures. Any intermediate reservoir used will, therefore, lower the efficiency of the heat engine. Practical engine cycles often involve continuous changes of temperature during heat transfer. A relationship among processes in which these sort of changes occur is necessary. The ideal approach to a cycle in which temperature continually changes is to consider the system to be in communication with a large number of reservoirs in procession. Each reservoir is considered to have a temperature differing by a small amount from the previous one. In such a model it is possible to imagine that each reservoir is replaced by a reversible heat engine in communication with standard reservoirs at same temperature T_0. Fig. shows one example to this substitution.
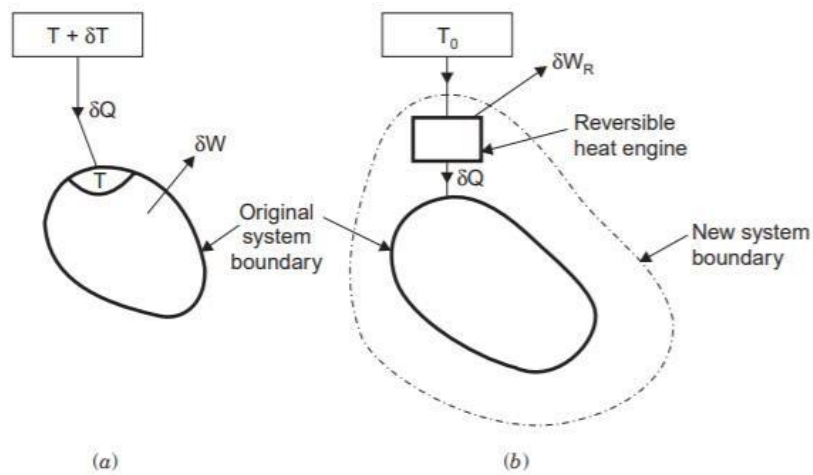

Fig.3

The system to which the heat transfer is effected is neither concerned with the source of energy it receives nor with the method of transfer, save that it must be reversible. Associated with the small heat transfer dQ to the original system is a small work transfer dW and for thissystem the first law gives,

$$\sum_{cycle} (\delta Q - \delta W) = 0$$

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Now consider the engine replacing the reservoirs and apply the second law to the new system in Fig. (b). If the new system is not a perpetual motion machine of second kind, no positive work transfer is possible with a single reservoir.

Therefore,
$$\sum_{cycle} (\delta W - \delta W_R) \leq 0$$

But by the definition of thermodynamic temperature in equation

$$\frac{\delta W_R}{\delta Q} = \frac{\delta Q_0 - \delta Q}{\delta Q} = \frac{T_0 - T}{T}$$

$$T_0 \sum_{cycle} \left(\frac{\delta Q}{T}\right) \leq 0 \text{ but } T_0 \neq 0 \text{ and therefore ;}$$

$$\sum_{cycle} \left(\frac{\delta Q}{T}\right) \leq 0$$

This is known as Clausius inequality.

Carnot cycle:

The cycle was first suggested by a French engineer Sadi Carnot in 1824 which works on reversible cycle and is known as Carnot cycle. Any fluid may be used to operate the Carnot cycle which is performed in an engine cylinder the head of which is supposed alternatively to be perfect conductor or a perfect insulator of a heat. Heat is caused to flow into the cylinder by the application of high temperature energy source to the cylinder head during expansion, and to flow from the cylinder by the application of a

(a)
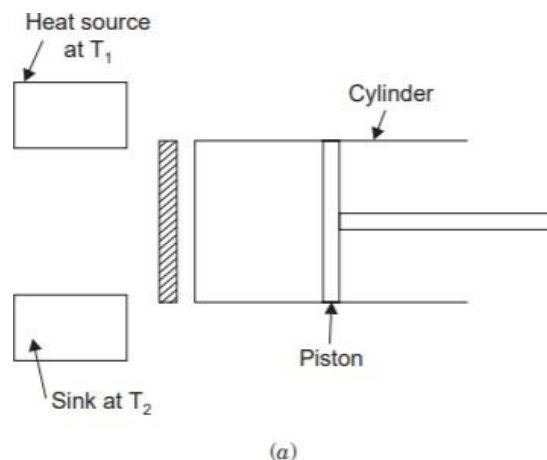
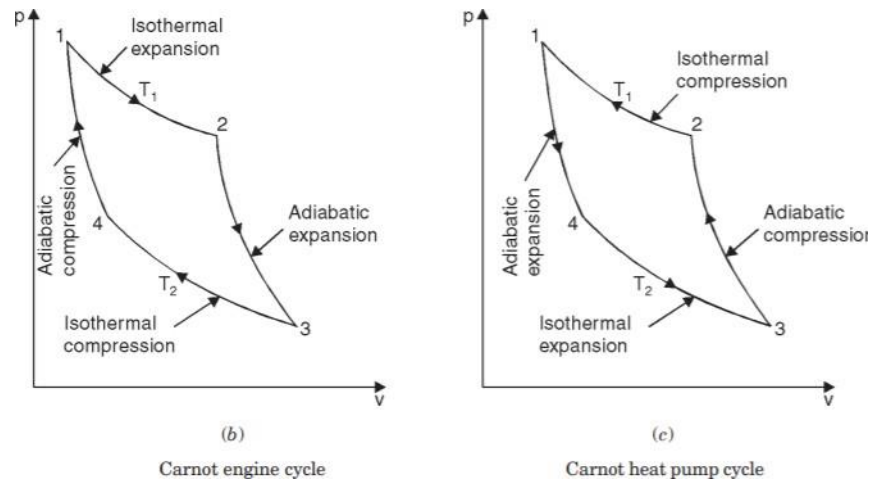lower temperature energy source to the head during compression.

Fig.4

Fig.6

The assumptions made for describing the working of the Carnot engine are as follows :

   (i) The piston moving in a cylinder does not develop any friction during motion.

   (ii) The walls of piston and cylinder are considered as perfect insulators of heat.

   (iii)The cylinder head is so arranged that it can be a perfect heat conductor or perfect heatinsulator.

   (iv) The transfer of heat does not affect the temperature of source or sink.

   (v) Working medium is a perfect gas and has constant specific heat.

      (vi)Compression and expansion are reversible.Following are the four stages of Carnot cycle :

Stage 1. (Process 1-2). Hot energy source is applied. Heat Q_1 is taken in whilst the fluid expands isothermally and reversibly at constant high temperature T_1.

Stage 2. (Process 2-3). The cylinder becomes a perfect insulator so that no heat flow takes place. The fluid expands adiabatically and reversibly whilst temperature falls from T_1 to T_2.

Stage 3. (Process 3-4). Cold energy source is applied. Heat Q_2 flows from the fluid whilst it is compressed isothermally and reversibly at constant lower temperature T_2.

Stage 4. (Process 4-1). Cylinder head becomes a perfect insulator so that no heat flow occurs. The compression is continued adiabatically and reversibly during which temperature is raisedfrom T_2 to T_1. The work delivered from the system during the cycle is represented by the enclosed area of the cycle. Again for a closed cycle, according to first law of the thermodynamics the work obtained is equal to the difference between the heat supplied by thesource (Q_1) and the heat rejected to the sink (Q_2).

$$\therefore \qquad W = Q_1 - Q_2$$

Also, thermal efficiency, $\eta_{th} = \dfrac{\text{Work done}}{\text{Heat supplied by the source}} = \dfrac{Q_1 - Q_2}{Q_1}$

$$= 1 - \frac{Q_2}{Q_1}\left(= 1 - \frac{T_2}{T_1}\right) \qquad \begin{bmatrix} \because & Q_1 = m\,c_p\,T_1 \\ & Q_2 = m\,c_p\,T_2 \\ \text{where, } m = \text{mass of fluid.} \end{bmatrix}$$

Such an engine since it consists entirely of reversible processes, can operate in the reverse direction so that it follows the cycle shown in Fig. (b) and operates as a heat pump. Q2 is being taken in at the lower temperature T2 during the isothermal expansion (process 4 -3) and heat Q1 is being rejected at the upper temperature T1 (process 2 -1). Work W will be needed to drive the pump. Again, the enclosed area represents this work which is exactly equal to that flowing from it when used as engine.

The Carnot cycle cannot be performed in practice because of the following reasons :

1. It is impossible to perform a frictionless process.

2. It is impossible to transfer the heat without temperature potential.

3. Isothermal process can be achieved only if the piston moves very slowly to allow heat transfer so that the temperature remains contant. Adiabatic process can be achieved only if the piston moves as fast as possible so that the heat transfer is negligible due to very short time available. The isothermal and adiabatic processes take place during the same stroke therefore the piston has to move very slowly for part of the stroke and it has to move very fast during remaining stroke. This variation of motion of the piston during the same stroke is not possible.

Carnot's theorem:

"It states that of all engines operating between a given constant temperature source and a given constant temperature sink, none has a higher efficiency than a reversible engine".
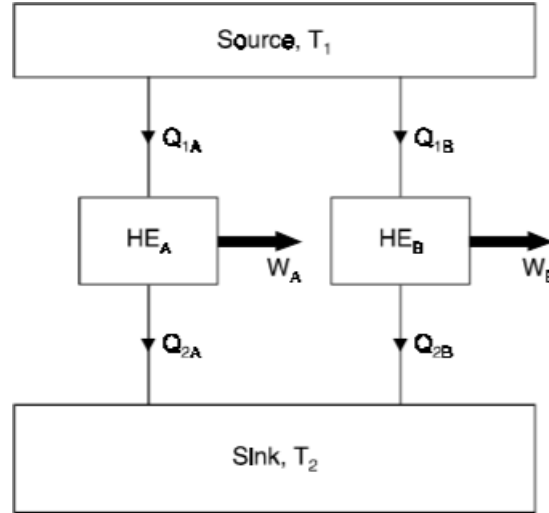
Fig. 7Two cyclic heat engines HEA and HEB operating between the same source andsink, of which HEB is reversible.

HE$_A$ and HE$_B$ are the two engines operating between the given source at temperature T$_1$ and the given sink at temperature T$_2$.

Let HE$_A$ be any heat engine and HE$_B$ be any reversible heat engine. We have to prove that efficiency of HE$_B$ is more than that of HE$_A$. Let us assume that $\eta_A > \eta_B$. Let the rates of working of the engines be such that,

$$Q_{1A} = Q_{1B} = Q_1$$
$$\eta_A > \eta_B$$
$$\frac{W_A}{Q_{1A}} > \frac{W_B}{Q_{1B}}$$
$$W_A > W_B$$

Now, let HE$_B$ be reversed. Since HE$_B$ is a reversible heat engine, the magnitudes of heat and work transfer quantities will remain the same, but their directions will be reversed, as shown in Fig. Since W$_A$ > W$_B$, some part of W$_A$ (equal to W$_B$) may be fed to drive the reversed heat engine ∃H$_B$. Since Q$_{1A}$ = Q$_{1B}$ = Q$_1$, the heat discharged by ∃H$_B$ may be supplied to HE$_A$. Thesource may, therefore, be eliminated. The net result is that HE$_A$ and ∃H$_B$ together constitute aheat engine which, operating in a cycle produces network W$_A$ – W$_B$ while exchanging heat with a single reservoir at T$_2$. This violates the Kelvin-Planck statement of the second law. Hence the assumption that $\eta_A > \eta_B$ is wrong.
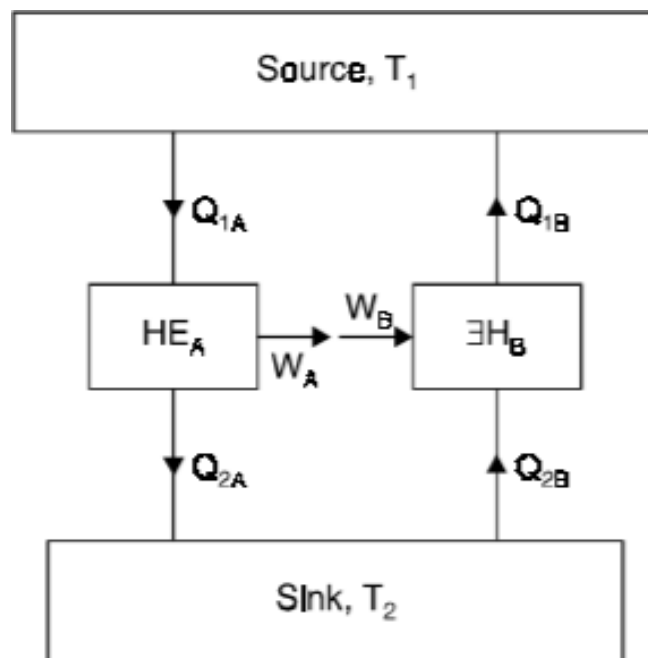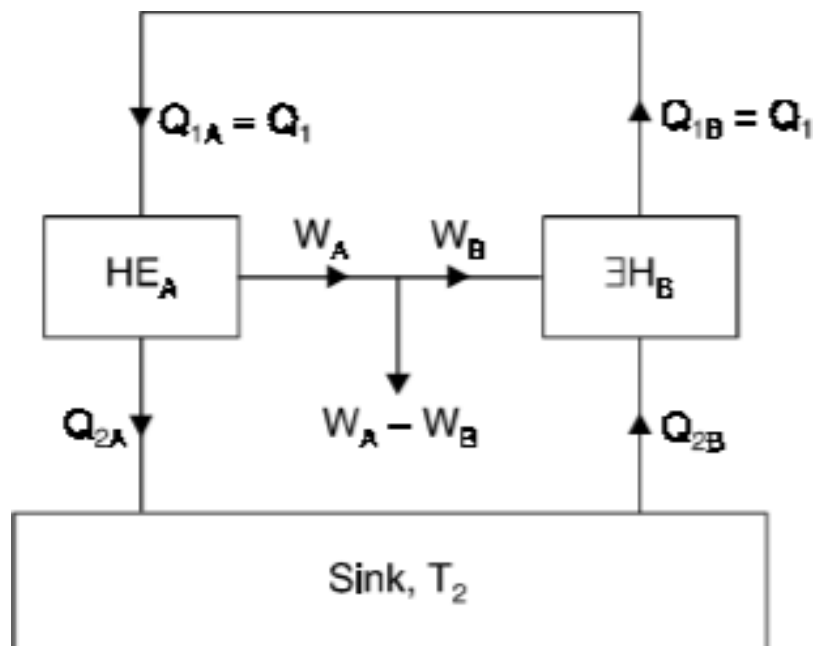
Fig. HEB is reversed

Fig.8 HEA and ∃HB together violate the Kelvin-Planck statement.

Therefore,

$$\eta_B \geq \eta_A.$$

Corollary of carnot's theorem:

''The efficiency of all reversible heat engines operating between the same temperature levelsis the same''.

Let both the heat engines HE_A and HE_B be reversible. Let us assume $\eta_A > \eta_B$. if HE_B is reversed to run say, as a heat pump using some part of the work output (W_A) of engine HE_A, we see that the combined system of heat pump HE_B and engine HE_A, becomes a PMM2. So η_A cannot be greater than η_B. Similary, if we assume $\eta_B > \eta_A$ and reverse the engine HE_A, we observe that η_B cannot be greater than η_A $\therefore$ $\eta_A = \eta_B$. Since the efficiencies of all reversible engines operating between the same heat reservoirs are the same, the efficiency of a reversible engine is independent of the nature or amount of the working substanceundergoing the cycle.

Efficiency of the reversible heat engine:

The efficiency of a reversible heat engine in which heat is received solely at T_1 is found to be,

$$\eta_{rev.} = \eta_{max} = 1 - \left(\frac{Q_2}{Q_1}\right)_{rev.} = 1 - \frac{T_2}{T_1}$$

$$\eta_{rev.} = \frac{T_1 - T_2}{T_1}$$

From the above expression, it may be noted that as T_2 decreases and T_1 increases, theefficiency of the reversible cycle increases.

Since η is always less than unity, T_2 is always greater than zero and + ve. The C.O.P. of arefrigerator is given by,

$$(C.O.P.)_{ref.} = \frac{Q_2}{Q_1 - Q_2} = \frac{1}{\frac{Q_1}{Q_2} - 1}$$

For a *reversible refrigerator*, using

$$\frac{Q_1}{Q_2} = \frac{T_1}{T_2}$$

$$(C.O.P.)_{rev.} = \frac{1}{\frac{T_1}{T_2} - 1}$$

$\therefore$ $$[(C.O.P.)_{ref.}]_{rev.} = \frac{T_2}{T_1 - T_2}$$

Similarly, for a *reversible heat pump*

$$[(C.O.P.)_{heat\ pump}]_{rev.} = \frac{T_1}{T_1 - T_2}$$

Problem 1: A heat engine receives heat at the rate of 1500 kJ/min and gives an output of 8.2kW. Determine : (i) The thermal efficiency ; (ii) The rate of heat rejection.

Solution. Heat received by the heat engine,

$$Q_1 = 1500 \text{ kJ/min}$$

$$= \frac{1500}{60} = 25 \text{ kJ/s}$$

Work output, $W = 8.2$ kW $= 8.2$ kJ/s.

(i) Thermal efficiency, $\eta_{th} = \dfrac{W}{Q_1}$
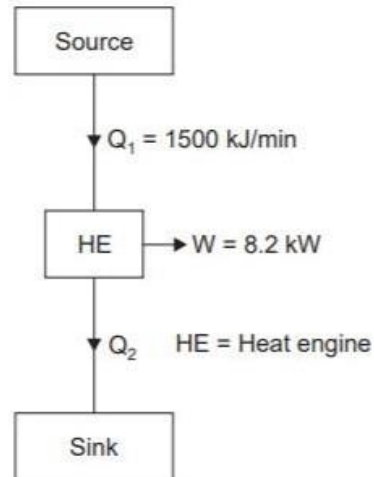
$$= \frac{8.2}{25} = 0.328 = 32.8\%$$

Hence, **thermal efficiency = 32.8%. (Ans.)**

(ii) Rate of heat rejection,

$$Q_2 = Q_1 - W = 25 - 8.2$$
$$= 16.8 \text{ kJ/s}$$

Hence, **the rate of heat rejection = 16.8 kJ/s.**

Source

$Q_1 = 1500$ kJ/min

HE → W = 8.2 kW

Q_2 HE = Heat engine

Sink

Problem 2: Find the co-efficient of performance and heat transfer rate in the condenser of a refrigerator in kJ/h which has a refrigeration capacity of 12000 kJ/h when power input is 0.75kW.

Solution:

Solution.
Refrigeration capacity, $Q_2 = 12000$ kJ/h
Power input, $W = 0.75$ kW $(= 0.75 \times 60 \times 60$ kJ/h$)$
Co-efficient of performance, C.O.P. :
Heat transfer rate :

$$(\text{C.O.P.})_{refrigerator} = \frac{\text{Heat absorbed at lower temperature}}{\text{Work input}}$$

$$\therefore \quad \text{C.O.P.} = \frac{Q_2}{W} = \frac{12000}{0.75 \times 60 \times 60} = 4.44$$
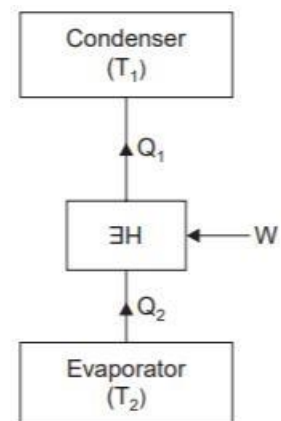
Hence   **C.O.P. = 4.44.   (Ans.)**
Hence transfer rate in condenser $= Q_1$
According to the first law
$Q_1 = Q_2 + W = 12000 + 0.75 \times 60 \times 60 = 14700$ kJ/h
Hence, **heat transfer rate = 14700 kJ/h.   (Ans.)**

Condenser
(T_1)

Q_1

HE ← W

Q_2

Evaporator
(T_2)

Problem 3: A domestic food refrigerator maintains a temperature of – 12°C. The ambient airtemperature is 35°C. If heat leaks into the freezer at the continuous rate of 2 kJ/s determine the least power necessary to pump this heat out continuously.

Solution. Freezer temperature,
$$T_2 = -12 + 273 = 261 \text{ K}$$
Ambient air temperature,
$$T_1 = 35 + 273 = 308 \text{ K}$$
Rate of heat leakage into the freezer = 2 kJ/s

Least power required to pump the heat :

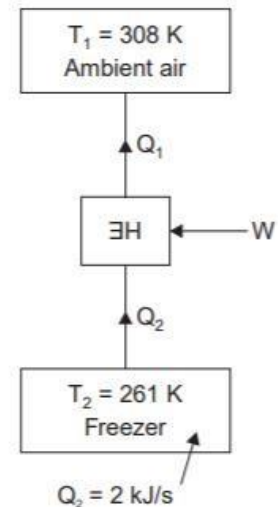The refrigerator cycle removes heat from the freezer at the same rate at which heat leaks into it (Fig. 5.12).

For minimum power requirement
$$\frac{Q_2}{T_2} = \frac{Q_1}{T_1}$$

∴ $Q_1 = \dfrac{Q_2}{T_2} \times T_1 = \dfrac{2}{261} \times 308 = 2.36 \text{ kJ/s}$

∴ $W = Q_1 - Q_2$
$$= 2.36 - 2 = 0.36 \text{ kJ/s} = 0.36 \text{ kW}$$

Hence, *least power required to pump the heat continuously*
$$= \textbf{0.36 kW.} \quad \textbf{(Ans.)}$$

(Diagram:)
T₁ = 308 K Ambient air — Q₁ — HE ← W — Q₂ — T₂ = 261 K Freezer — Q₂ = 2 kJ/s

Problem 4: An inventor claims that his engine has the following specifications :Temperature limits   750°C and 25°C

Power developed......75 kW

Fuel burned per hour..... 3.9 kg

Heating value of the fuel..... 74500 kJ/kg

State whether his claim is valid or not

Solution. Temperature of source, $T_1 = 750 + 273 = 1023 \text{ K}$
Temperature of sink, $T_2 = 25 + 273 = 298 \text{ K}$

We know that the thermal efficiency of Carnot cycle is the maximum between the specified temperature limits.

Now, $\eta_{carnot} = 1 - \dfrac{T_2}{T_1} = 1 - \dfrac{298}{1023} = 0.7086 \quad \text{or} \quad 70.86\%$

The actual thermal efficiency claimed,

$$\eta_{thermal} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{75 \times 1000 \times 60 \times 60}{3.9 \times 74500 \times 1000} = 0.9292 \quad \text{or} \quad 92.92\%.$$

Since $\eta_{thermal} > \eta_{carnot}$, therefore claim of the inventor is **not valid** (or possible). **(Ans.)**

Problem 5: A cyclic heat engine operates between a source temperature of 1000°C and asink temperature of 40°C. Find the least rate of heat rejection per kW net output of the engine ?

Solution. Temperature of source,

$$T_1 = 1000 + 273 = 1273 \text{ K}$$

Temperature of sink,

$$T_2 = 40 + 273 = 313 \text{ K}$$

Least rate of heat rejection per kW net output :

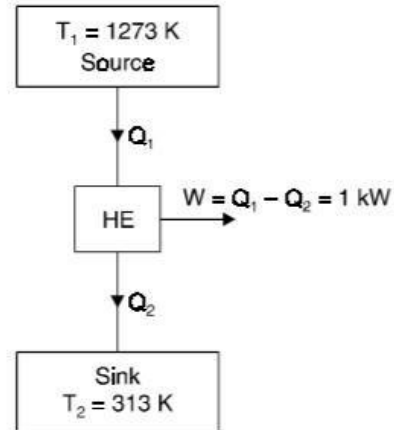For a reversible heat engine, the rate of heat rejection will be minimum (Fig. 5.13)

$$\eta_{max} = \eta_{rev.} = 1 - \frac{T_2}{T_1}$$

$$= 1 - \frac{313}{1273} = 0.754$$

Now

$$\frac{W_{net}}{Q_1} = \eta_{max} = 0.754$$

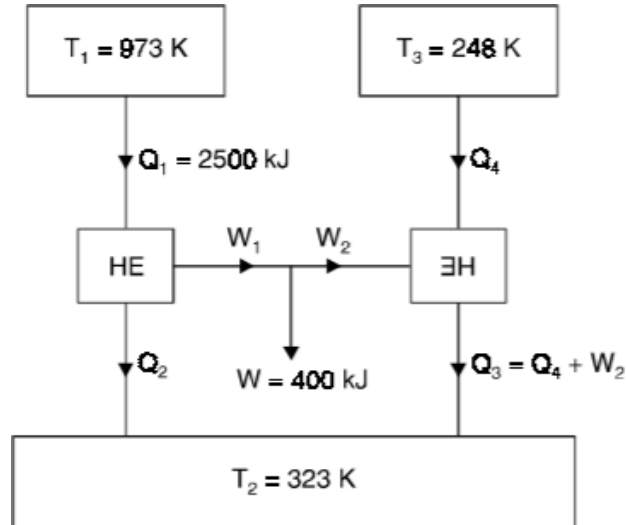$$\therefore \quad Q_1 = \frac{W_{net}}{0.754} = \frac{1}{0.754} = 1.326 \text{ kW}$$

Now

$$Q_2 = Q_1 - W_{net} = 1.326 - 1 = 0.326 \text{ kW}$$

Hence, **the least rate of heat rejection = 0.326 kW. (Ans.)**

Problem 6: A reversible heat engine operates between two reservoirs at temperatures 700°C and 50°C. The engine drives a reversible refrigerator which operates between reservoirs at temperatures of 50°C and – 25°C. The heat transfer to the engine is 2500 kJ and the net workoutput of the combined engine refrigerator plant is 400 kJ. (i) Determine the heat transfer to the refrigerant and the net heat transfer to the reservoir at 50°C ; (ii) Reconsider (i) given that the efficiency of the heat engine and the C.O.P. of the refrigerator are each 45 per cent of their maximum possible values.

Solution:

```
┌──────────────┐          ┌──────────────┐
│ T₁ = 973 K   │          │ T₃ = 248 K   │
└──────────────┘          └──────────────┘
        │                         │
        ▼ Q₁ = 2500 kJ            ▼ Q₄
   ┌──────┐  W₁    W₂    ┌──────┐
   │  HE  │ ───► ───►    │  ƎH  │
   └──────┘              └──────┘
        │       │               │
        ▼ Q₂    ▼ W = 400 kJ    ▼ Q₃ = Q₄ + W₂
┌──────────────────────────────────────────┐
│            T₂ = 323 K                      │
└──────────────────────────────────────────┘
```

Temperature, $T_1 = 700 + 273 = 973$ K

Temperature, $T_2 = 50 + 273 = 323$ K

Temperature, $T_3 = -25 + 273 = 248$ K

The heat transfer to the heat engine, $Q_1 = 2500$ kJ

The network output of the combined engine refrigerator plant,

$$W = W_1 - W_2 = 400 \text{ kJ.}$$

(i) Maximum efficiency of the heat engine cycle is given by

$$\eta_{max} = 1 - \frac{T_2}{T_1} = 1 - \frac{323}{973} = 0.668$$

Again, $\quad \dfrac{W_1}{Q_1} = 0.668$

∴ $\quad W_1 = 0.668 \times 2500 = 1670 \text{ kJ}$

$$(\text{C.O.P.})_{max} = \frac{T_3}{T_2 - T_3} = \frac{248}{323 - 248} = 3.306$$

Also, $\quad \text{C.O.P.} = \dfrac{Q_4}{W_2} = 3.306$

Since,
$$W_1 - W_2 = W = 400 \text{ kJ}$$
$$W_2 = W_1 - W = 1670 - 400 = 1270 \text{ kJ}$$
∴
$$Q_4 = 3.306 \times 1270 = 4198.6 \text{ kJ}$$
$$Q_3 = Q_4 + W_2 = 4198.6 + 1270 = 5468.6 \text{ kJ}$$
$$Q_2 = Q_1 - W_1 = 2500 - 1670 = 830 \text{ kJ.}$$

Heat rejection to the 50°C reservoir
$$= Q_2 + Q_3 = 830 + 5468.6 = \textbf{6298.6 kJ.} \quad \textbf{(Ans.)}$$

(*ii*) Efficiency of actual heat engine cycle,
$$\eta = 0.45 \; \eta_{max} = 0.45 \times 0.668 = 0.3$$
∴
$$W_1 = \eta \times Q_1 = 0.3 \times 2500 = 750 \text{ kJ}$$
∴
$$W_2 = 750 - 400 = 350 \text{ kJ}$$

C.O.P. of the actual refrigerator cycle,

$$\text{C.O.P.} = \frac{Q_4}{W_2} = 0.45 \times 3.306 = 1.48$$

∴
$$Q_4 = 350 \times 1.48 = \textbf{518 kJ.} \quad \textbf{(Ans.)}$$
$$Q_3 = 518 + 350 = 868 \text{ kJ}$$
$$Q_2 = 2500 - 750 = 1750 \text{ kJ}$$

Heat rejected to 50°C reservoir
$$= Q_2 + Q_3 = 1750 + 868 = \textbf{2618 kJ.} \quad \textbf{(Ans.)}$$

Problem 7: A reversible heat pump is used to maintain a temperature of 0°C in a refrigerator when it rejects the heat to the surroundings at 25°C. If the heat removal rate from the refrigerator is 1440 kJ/min, determine the C.O.P. of the machine and work input required. (ii) If the required input to run the pump is developed by a reversible engine which receives heat at 380°C and rejects heat to atmosphere, then determine the overall C.O.P. of the system.

Solution. (i) Temperature, T1 = 25 + 273 = 298 K Temperature, T2 = 0 + 273 = 273 K

(a) Single system   (b) Combined system
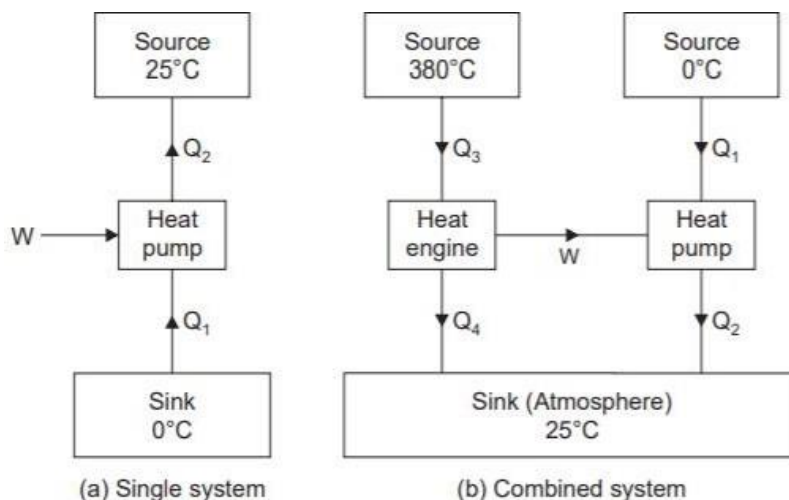
Heat removal rate from the refrigerator,

$$Q_1 = 1440 \text{ kJ/min} = 24 \text{ kJ/s}$$

Now, co-efficient of performance, for reversible heat pump,

$$\text{C.O.P.} = \frac{T_1}{T_1 - T_2} = \frac{298}{(298 - 273)} = \textbf{11.92.} \quad \textbf{(Ans.)}$$

$\therefore \qquad (\text{C.O.P.})_{ref.} = \dfrac{T_2}{T_1 - T_2} = \dfrac{273}{298 - 273} = 10.92$

Now, $\qquad 10.92 = \dfrac{Q_1}{W} = \dfrac{24}{W}$

$\therefore \qquad W = 2.2 \text{ kW}$

i.e., **Work input required** $\quad = \textbf{2.2 kW.} \quad \textbf{(Ans.)}$

$$Q_2 = Q_1 + W = 24 + 2.2 = 26.2 \text{ kJ/s}$$

(*ii*) Refer Fig. 5.15 (*b*).

The overall C.O.P. is given by,

$$\text{C.O.P.} = \frac{\text{Heat removed from the refrigerator}}{\text{Heat supplied from the source}}$$

$$= \frac{Q_1}{Q_3}$$

For the reversible engine, we can write

$$\frac{Q_3}{T_3} = \frac{Q_4}{T_4}$$

or $\qquad \dfrac{Q_4 + W}{T_3} = \dfrac{Q_4}{T_4}$

or $\qquad \dfrac{Q_4 + 2.2}{(380 + 273)} = \dfrac{Q_4}{(25 + 273)}$

or $\qquad \dfrac{Q_4 + 2.2}{653} = \dfrac{Q_4}{298}$

or $\qquad 298(Q_4 + 2.2) = 653\, Q_4$

or $\qquad Q_4(653 - 298) = 298 \times 2.2$

or $\qquad Q_4 = \dfrac{298 \times 2.2}{(653 - 298)} = 1.847 \text{ kJ/s}$

$\therefore \qquad Q_3 = Q_4 + W = 1.847 + 2.2 = 4.047 \text{ kJ/s}$

Substituting this value in eqn. (*i*), we get

$$\text{C.O.P.} = \frac{24}{4.047} = \textbf{5.93.} \quad \textbf{(Ans.)}$$

If the purpose of the system is to supply the heat to the sink at 25°C, then

$$\textbf{Overall C.O.P.} = \frac{Q_2 + Q_4}{Q_3} = \frac{26.2 + 1.847}{4.047} = \textbf{6.93.} \quad \textbf{(Ans.)}$$

Unit -3 - POWER CYCLES AND INTERNAL COMBUSTION ENGINES – SMEA1405

DEFINITION OF A CYCLE

A cycle is defined as a repeated series of operations occurring in a certain order. It may be repeated by repeating the processes in the same order. The cycle may be of imaginary perfect engine or actual engine. The former is called ideal cycle and the latter actual cycle. In ideal cycle all accidental heat losses are prevented and the working substance is assumed to behave like a perfect working substance.

AIR STANDARD EFFICIENCY

To compare the effects of different cycles, it is of paramount importance that the effect of the calorific value of the fuel is altogether eliminated and this can be achieved by considering air (which is assumed to behave as a perfect gas) as the working substance in the engine cylinder. The efficiency of engine using air as the working medium is known as an "Air standard efficiency". This efficiency is often called ideal efficiency. The actual efficiency of a cycle is always less than the air-standard efficiency of that cycle under ideal conditions. This is taken into account by introducing a new term "Relative efficiency" which is defined as the ratio of Actual thermal efficiency to Air standard efficiency.

The analysis of all air standard cycles is based upon the following assumptions:

1. The gas in the engine cylinder is a perfect gas i.e., it obeys the gas laws and has constant specific heats.

2. The physical constants of the gas in the cylinder are the same as those of air at moderate temperatures i.e., the molecular weight of cylinder gas is 29. $C_p = 1.005$ kJ/kg-K, $C_v = 0.718$ kJ/kg-K.

3. The compression and expansion processes are adiabatic and they take place without internal friction, i.e., these processes are isentropic.

4. No chemical reaction takes place in the cylinder. Heat is supplied or rejected by bringing a hot bodyor a cold body in contact with cylinder at appropriate points during the process.

5. The cycle is considered closed with the same 'air' always remaining in the cylinder to repeat the cycle.

CONSTANT VOLUME OR OTTO CYCLE

This cycle is so named as it was conceived by 'Otto'. On this cycle, petrol, gas and many types of oil engines work. It is the standard of comparison for internal combustion engines.

Figs. 1 (a) and (b) shows the theoretical p-V diagram and T-s diagrams of this cycle respectively.

- The point 1 represents that cylinder is full of air with volume V_1, pressure P_1 and absolute temperature T_1.

- Line 1-2 represents the adiabatic compression of air due to which P_1, V_1 and T_1 change to P_2, V_2 and T_2 respectively.

- Line 2-3 shows the supply of heat to the air at constant volume so that P_2 and T_2 change to P_3 and T_3 (V_3 being the same as V_2).

- Line 3-4 represents the adiabatic expansion of the air. During expansion P_3, V_3 and T_3 change to a final value of P_4, V_4 or V_1 and T_4, respectively.

- Line 4-1 shows the rejection of heat by air at constant volume till original state (point 1) reaches. Consider 1 kg of air (working substance):
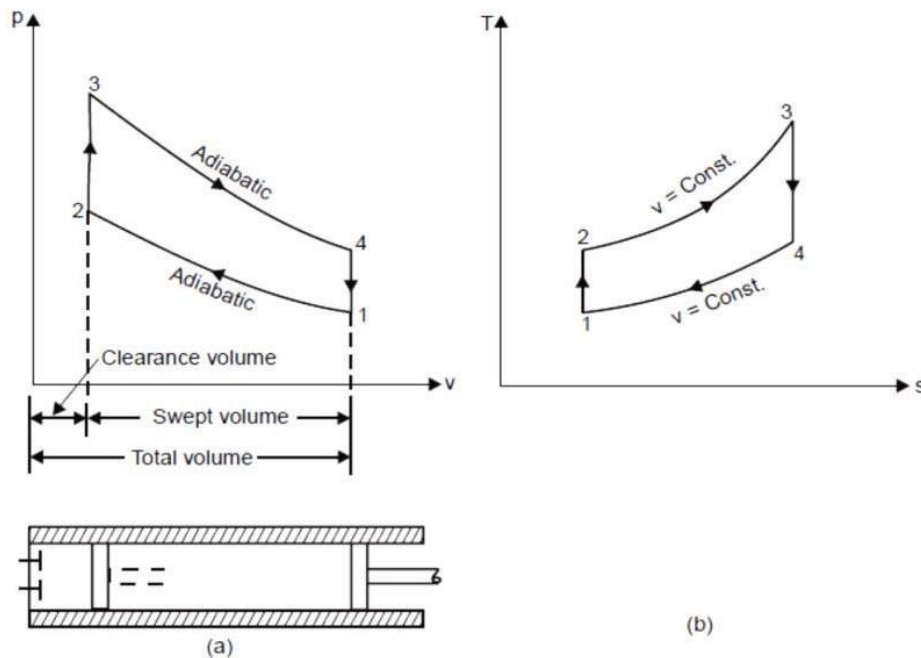

Fig. 1

Heat supplied at constant volume $= c_v(T_3 - T_2)$.

Heat rejected at constant volume $= c_v(T_4 - T_1)$.

But, work done $=$ Heat supplied $-$ Heat rejected

$$= c_v(T_3 - T_2) - c_v(T_4 - T_1)$$

$\therefore$ Efficiency $= \dfrac{\text{Work done}}{\text{Heat supplied}} = \dfrac{c_v(T_3 - T_2) - c_v(T_4 - T_1)}{c_v(T_3 - T_2)}$

$$= 1 - \dfrac{T_4 - T_1}{T_3 - T_2}$$

Let compression ratio, $r_c \, (= r) = \dfrac{v_1}{v_2}$

and expansion ratio, $r_e \, (= r) = \dfrac{v_4}{v_3}$

(These *two ratios are same* in this cycle)

As $\dfrac{T_2}{T_1} = \left(\dfrac{v_1}{v_2}\right)^{\gamma-1}$

Then, $T_2 = T_1 \cdot (r)^{\gamma-1}$

Similarly, $\dfrac{T_3}{T_4} = \left(\dfrac{v_4}{v_3}\right)^{\gamma-1}$

or $T_3 = T_4 \cdot (r)^{\gamma-1}$

Inserting the values of T_2 and T_3 in equation (*i*), we get

$$\eta_{otto} = 1 - \dfrac{T_4 - T_1}{T_4 \cdot (r)^{\gamma-1} - T_1 \cdot (r)^{\gamma-1}} = 1 - \dfrac{T_4 - T_1}{r^{\gamma-1}(T_4 - T_1)}$$

$$= 1 - \dfrac{1}{(r)^{\gamma-1}}$$

This expression is known as the air standard efficiency of the Otto cycle. It is clear from the above expression that efficiency increases with the increase in the value of r, which means we can have maximum efficiency by increasing r to a considerable extent, but due to practical difficulties its value is limited to about 8. The net work done per kg in the Otto cycle can also be expressed in terms of p, v. If p is expressed in bar i.e., 10^5 N/m², then work done

$$W = \left(\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right) \times 10^2 \text{ kJ}$$

$$\frac{p_3}{p_4} = r^\gamma = \frac{p_2}{p_1}$$

$$\frac{p_3}{p_2} = \frac{p_4}{p_1} = r_p$$

where r_p stands for *pressure ratio*.

and
$$v_1 = r v_2 = v_4 = r v_3 \qquad \left[\because \frac{v_1}{v_2} = \frac{v_4}{v_3} = r \right]$$

$\therefore$
$$W = \frac{1}{\gamma - 1} \left[p_4 v_4 \left(\frac{p_3 v_3}{p_4 v_4} - 1 \right) - p_1 v_1 \left(\frac{p_2 v_2}{p_1 v_1} - 1 \right) \right]$$

$$= \frac{1}{\gamma - 1} \left[p_4 v_4 \left(\frac{p_3}{p_4 r} - 1 \right) - p_1 v_1 \left(\frac{p_2}{p_1 r} - 1 \right) \right]$$

$$= \frac{v_1}{\gamma - 1} \left[p_4 (r^{\gamma - 1} - 1) - p_1 (r^{\gamma - 1} - 1) \right]$$

$$= \frac{v_1}{\gamma - 1} \left[(r^{\gamma - 1} - 1)(p_4 - p_1) \right]$$

$$= \frac{p_1 v_1}{\gamma - 1} \left[(r^{\gamma - 1} - 1)(r_p - 1) \right]$$

Mean effective pressure (p_m) is given by :

$$p_m = \left[\left(\frac{p_3 v_3 - p_4 v_4}{\gamma - 1} - \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} \right) \div (v_1 - v_2) \right] \text{ bar}$$

$$p_m = \frac{\left[\dfrac{p_1 v_1}{\gamma - 1} (r^{\gamma - 1} - 1)(r_p - 1) \right]}{(v_1 - v_2)}$$

$$= \frac{\dfrac{p_1 v_1}{\gamma - 1} [(r^{\gamma - 1} - 1)(r_p - 1)}{v_1 - \dfrac{v_1}{r}}$$

$$= \frac{\frac{p_1 v_1}{\gamma - 1}[(r^{\gamma - 1} - 1)(r_p - 1)]}{v_1\left(\frac{r - 1}{r}\right)}$$

$$\mathbf{P_m} = \frac{p_1 r[(r^{\gamma - 1} - 1)(r_p - 1)]}{(\gamma - 1)(r - 1)}$$

MEP may be thought of as the average pressure acting on a piston during different portions of its cycle. It is the ratio of the work done to stoke volume of the cycle

CONSTANT PRESSURE OR DIESEL CYCLE

This cycle was introduced by Dr. R. Diesel in 1897. It differs from Otto cycle in that heat is supplied at constant pressure instead of at constant volume. Fig.2 shows the p-v and T-s diagrams of this cycle respectively.

This cycle comprises of the following operations:
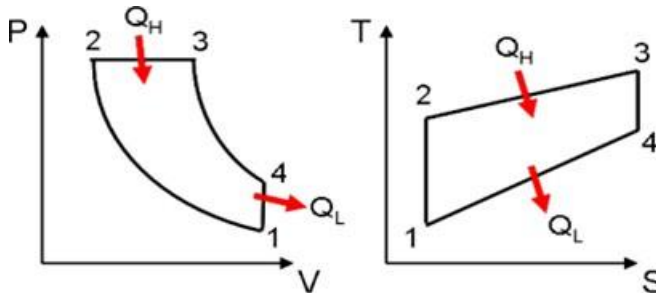
Fig. 2 P-V and T-S diagrams of Ideal Diesel Cycle

(i) 1-2 ...Adiabatic compression.

(ii) 2-3... Addition of heat at constant pressure.

(iii) 3-4... Adiabatic expansion.

(iv) 4-1... Rejection of heat at constant volume.

Point 1 represents that the cylinder is full of air. Let P_1, V_1 and T_1 be the corresponding pressure, volume and absolute temperature. The piston then compresses the air adiabatically (i.e., $pV^r = $ constant) till the values become P_2, V_2 and T_2 respectively (at the end of the stroke) at point 2. Heat is then added from a hot body at a constant pressure. During this addition of heat let volume increases from V_2 to V_3 and temperature T_2 to T_3, corresponding to point 3. This point (3) is

called the point of cut-off. The air then expands adiabatically to the conditions P4, V4 and T4 respectively corresponding to point 4. Finally, the air rejects the heat to the cold body at constant volume till the point 1 where it returns to its original state

Consider 1 kg of air.

Heat supplied at constant pressure $= c_p(T_3 - T_2)$

Heat rejected at constant volume $= c_v(T_4 - T_1)$

Work done $=$ Heat supplied − heat rejected

$\qquad\qquad = c_p(T_3 - T_2) - c_v(T_4 - T_1)$

$\therefore \qquad \eta_{\text{diesel}} = \dfrac{\text{Work done}}{\text{Heat supplied}}$

$$= \dfrac{c_p(T_3 - T_2) - c_v(T_4 - T_1)}{c_p(T_3 - T_2)}$$

$$= 1 - \dfrac{(T_4 - T_1)}{\gamma(T_3 - T_2)} \qquad ...(i) \quad \left[\because \dfrac{c_p}{c_v} = \gamma \right]$$

Let compression ratio, $\quad r = \dfrac{v_1}{v_2}$, and cut-off ratio, $\quad \rho = \dfrac{v_3}{v_2}$ *i.e.,* $\dfrac{\text{Volume at cut-off}}{\text{Clearance volume}}$

Now, during *adiabatic compression 1-2,*

$$\dfrac{T_2}{T_1} = \left(\dfrac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} \quad \text{or} \quad T_2 = T_1 \cdot (r)^{\gamma-1}$$

During *constant pressure process 2-3,*

$$\dfrac{T_3}{T_2} = \dfrac{v_3}{v_2} = \rho \quad \text{or} \quad T_3 = \rho \cdot T_2 = \rho \cdot T_1 \cdot (r)^{\gamma-1}$$

During *adiabatic expansion 3-4*

$$\dfrac{T_3}{T_4} = \left(\dfrac{v_4}{v_3}\right)^{\gamma-1}$$

$$= \left(\dfrac{r}{\rho}\right)^{\gamma-1} \qquad \left(\because \dfrac{v_4}{v_3} = \dfrac{v_1}{v_3} = \dfrac{v_1}{v_2} \times \dfrac{v_2}{v_3} = \dfrac{r}{\rho} \right)$$

$\therefore \qquad T_4 = \dfrac{T_3}{\left(\dfrac{r}{\rho}\right)^{\gamma-1}} = \dfrac{\rho \cdot T_1 (r)^{\gamma-1}}{\left(\dfrac{r}{\rho}\right)^{\gamma-1}} = T_1 \cdot \rho^{\gamma}$

By inserting values of T_2, T_3 and T_4 in eqn. (*i*), we get

$$\eta_{\text{diesel}} = 1 - \dfrac{(T_1 \cdot \rho^{\gamma} - T_1)}{\gamma(\rho \cdot T_1 \cdot (r)^{\gamma-1} - T_1 \cdot (r)^{\gamma-1})} = 1 - \dfrac{(\rho^{\gamma} - 1)}{\gamma(r)^{\gamma-1}(\rho - 1)}$$

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It may be observed that eqn. for efficiency of diesel cycle is different from that of the Otto cycle only in bracketed factor. This factor is always greater than unity, because r > 1. Hence for a given compression ratio, the Otto cycle is more efficient.

Mean effective pressure p_m is given by :

$$p_m = \frac{p_1 v_1 r^{\gamma-1} [\gamma(\rho-1) - r^{1-\gamma}(\rho^\gamma - 1)]}{(\gamma-1)v_1 \left(\dfrac{r-1}{r}\right)}$$

$$\mathbf{p_m} = \frac{p_1 r^\gamma [\gamma(\rho-1) - r^{1-\gamma}(\rho^\gamma - 1)]}{(\gamma-1)(r-1)} .$$

DUAL COMBUSTION CYCLE

This cycle (also called the limited pressure cycle or mixed cycle) is a combination of Otto and Diesel cycles, in a way, that heat is added partly at constant volume and partly at constant pressure ; the advantage of which is that more time is available to fuel (which is injected into the engine cylinder before the end of compression stroke) for combustion. Because of lagging characteristics of fuel this cycle is invariably used for diesel and hot spot ignition engines.

The dual combustion cycle (Fig 3) consists of the following operations :

(i) 1-2—Adiabatic compression

(ii) 2-3—Addition of heat at constant volume

(iii) 3-4—Addition of heat at constant pressure

(iv) 4-5—Adiabatic expansion

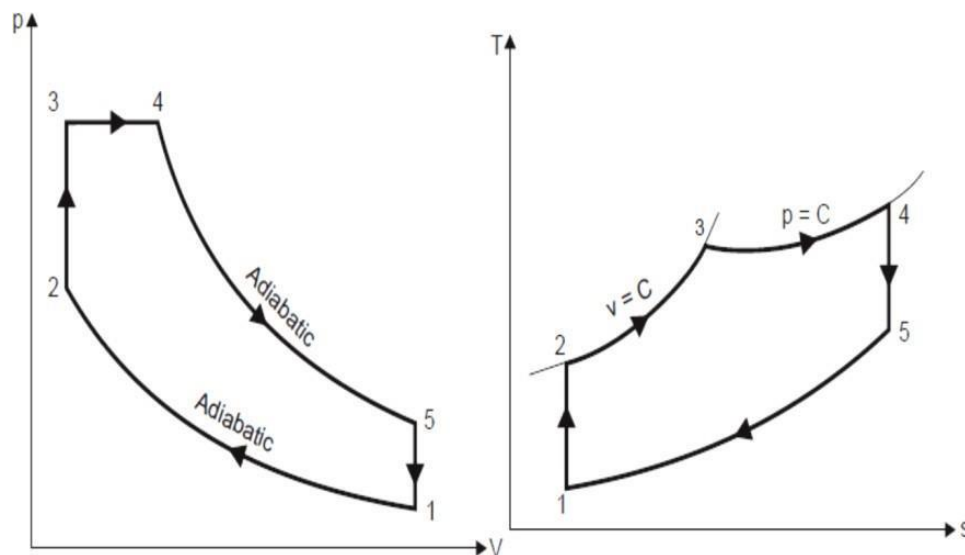(v) 5-1—Rejection of heat at constant volume.

8

Fig.3

Consider 1 kg of air.

Total heat supplied = Heat supplied during the operation 2-3

+ heat supplied during the operation 3-4

$= c_v(T_3 - T_2) + c_p(T_4 - T_3)$

Heat rejected during operation 5-1 $= c_v(T_5 - T_1)$

Work done = Heat supplied – heat rejected

$= c_v(T_3 - T_2) + c_p(T_4 - T_3) - c_v(T_5 - T_1)$

$$\eta_{\text{dual}} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{c_v(T_3 - T_2) + c_p(T_4 - T_3) - c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}$$

$$= 1 - \frac{c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)}$$

$$= 1 - \frac{c_v(T_5 - T_1)}{(T_3 - T_2) + \gamma(T_4 - T_3)} \qquad ...(i) \qquad \left(\because \ \gamma = \frac{c_p}{c_v}\right)$$

Compression ratio, $\qquad r = \dfrac{v_1}{v_2}$

During *adiabatic compression process 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (r)^{\gamma-1} \qquad\qquad ...(ii)$$

During *constant volume heating process,*

$$\frac{p_3}{T_3} = \frac{p_2}{T_2}$$

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = \beta, \quad \text{where } \beta \text{ is known as } \textbf{pressure or explosion ratio.}$$

$$T_2 = \frac{T_3}{\beta} \qquad\qquad ...(iii)$$

During *adiabatic expansion process,*

$$\frac{T_4}{T_5} = \left(\frac{v_5}{v_4}\right)^{\gamma-1}$$

$$= \left(\frac{r}{\rho}\right)^{\gamma-1} \qquad\qquad ...(iv)$$

$$\left(\because \ \frac{v_5}{v_4} = \frac{v_1}{v_4} = \frac{v_1}{v_2} \times \frac{v_2}{v_4} = \frac{v_1}{v_2} \times \frac{v_3}{v_4} = \frac{r}{\rho}, \rho \text{ being the cut-off ratio}\right)$$

During *constant pressure heating process,*

$$\frac{v_3}{T_3} = \frac{v_4}{T_4}$$

$$T_4 = T_3 \frac{v_4}{v_3} = \rho \, T_3 \qquad\qquad ...(v)$$

Putting the value of T_4 in the eqn. (iv), we get

$$\frac{\rho T_3}{T_5} = \left(\frac{r}{\rho}\right)^{\gamma-1} \quad \text{or} \quad T_5 = \rho \cdot T_3 \cdot \left(\frac{\rho}{r}\right)^{\gamma-1}$$

Putting the value of T_2 in eqn. (ii), we get

$$\frac{\dfrac{T_3}{\beta}}{T_1} = (r)^{\gamma-1}$$

$$T_1 = \frac{T_3}{\beta} \cdot \frac{1}{(r)^{\gamma-1}}$$

Now inserting the values of T_1, T_2, T_4 and T_5 in eqn. (i), we get

$$\eta_{\text{dual}} = 1 - \frac{\left[\rho \cdot T_3 \left(\dfrac{\rho}{r}\right)^{\gamma-1} - \dfrac{T_3}{\beta} \cdot \dfrac{1}{(r)^{\gamma-1}}\right]}{\left[\left(T_3 - \dfrac{T_3}{\beta}\right) + \gamma(\rho T_3 - T_3)\right]} = 1 - \frac{\dfrac{1}{(r)^{\gamma-1}}\left(\rho^{\gamma} - \dfrac{1}{\beta}\right)}{\left(1 - \dfrac{1}{\beta}\right) + \gamma(\rho-1)}$$

$$\eta_{\text{dual}} = 1 - \frac{1}{(r)^{\gamma-1}} \cdot \frac{(\beta \cdot \rho^{\gamma} - 1)}{[(\beta-1) + \beta\gamma(\rho-1)]}$$

$$P_m = \frac{p_1(r)^{\gamma}[\beta(\rho-1) + (\beta-1) - r^{1-\gamma}(\beta\rho^{\gamma} - 1)]}{(\gamma-1)(r-1)}$$

COMPARISON OF OTTO, DIESEL AND DUAL COMBUSTION CYCLES

Following are the important variable factors which are used as a basis for comparison of the cycles:

- Compression ratio.
- Maximum pressure
- Heat supplied
- Heat rejected
- Net work

Some of the above mentioned variables are fixed when the performance of Otto, Diesel and dual combustion cycles is to be compared.

Efficiency Versus Compression Ratio

Fig shows the comparison for the air standard efficiencies of the Otto, Diesel and Dual combustion cycles at various compression ratios and with given cut-off ratio for the Diesel and Dual combustion cycles. It is evident from the Fig that the air standard efficiencies increase with the increase in the compression ratio. For a given compression ratio Otto cycle is the most efficient while the Diesel cycle is the least efficient.

Note. The maximum compression ratio for the petrol engine is limited by detonation. In their respective ratio ranges, the Diesel cycle is more efficient than the Otto cycle.

For the Same Compression Ratio and the Same Heat Input

A comparison of the cycles (Otto, Diesel and Dual) on the p-v and T-s diagrams for the same compression ratio and heat supplied is shown in the Fig.4
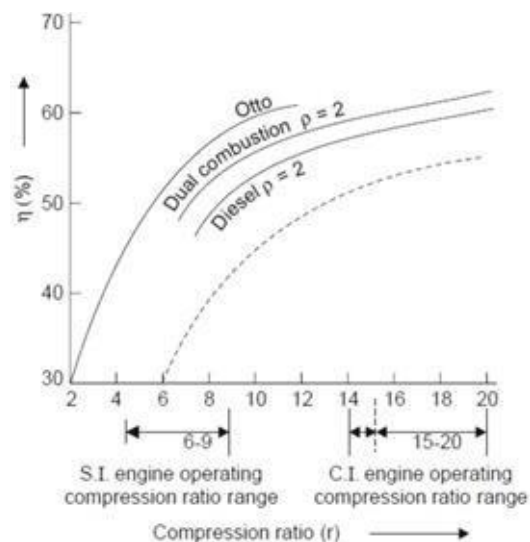
Fig.4
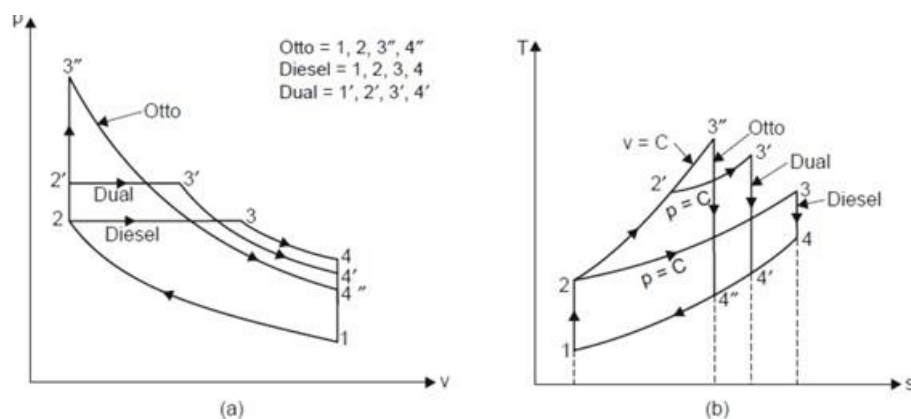
Fig.5

We know that,
$$\eta = 1 - \frac{\text{Heat rejected}}{\text{Heat supplied}} \qquad \qquad ...(13.13)$$

Since all the cycles reject their heat at the same specific volume, process line from state 4 to 1, the quantity of heat rejected from each cycle is represented by the appropriate area under the line 4 to 1 on the T-s diagram. As is evident from the eqn. the cycle which has the least heat rejected will have the highest efficiency. Thus, Otto cycle is the most efficient and Diesel cycle is the least efficient of the three cycles.

$$\eta otto > \eta dual > \eta diesel$$

For Constant Maximum Pressure and Heat Supplied

Fig. 6 shows the Otto and Diesel cycles on p-v and T-s diagrams for constant maximum pressure and heat input respectively.
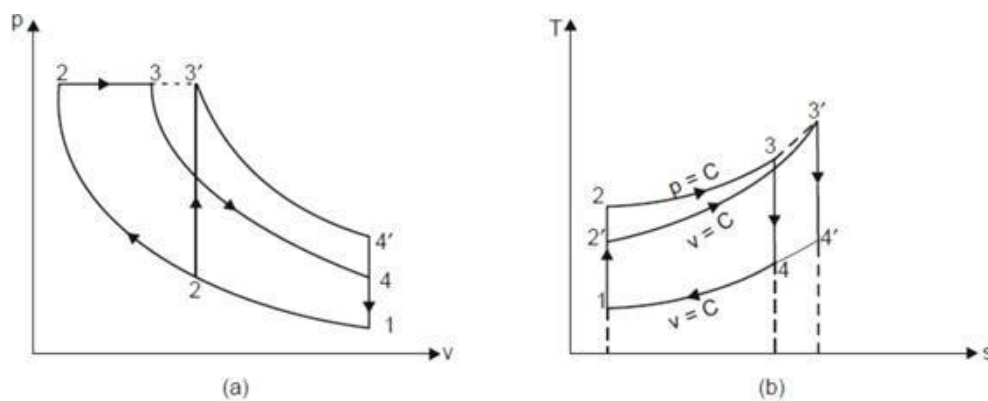
Fig.6

For the maximum pressure the points 3 and 3′ must lie on a constant pressure line.

On T-s diagram the heat rejected from the Diesel cycle is represented by the area under the line 4 to 1 and this area is less than the Otto cycle area under the curve 4′ to 1 ; hence the Diesel cycle is more efficient than the Otto cycle for the condition of maximum pressure and heat supplied

1. An engine working on the Otto cycle is supplied with air at 0.1 MPa, 350C. The compression ratio is 8. Heat supplied is 2100 kJ/kg. Calculate the maximum pressure and temperature of the cycle, the cycle efficiency, and the mean effective pressure. (For air, cp = 1.005kJ/kgK, cv = 0.718 kJ/kgK, and R = 0.287 kJ/kgK)

Data given:

- Otto Cycle
- $p_1 = 0.1$ MPa $= 0.1 \times 10^6$ N/m^2
- $T_1 = 35^0C = 35 + 273 = 308$ K
- $r_c = 8$
- $q_{in} = q_{2\text{-}3} = 2100$ kJ/kg $= 2100 \times 10^3$ J/kgK
- $c_p = 2100$ kJ/kg $= 2100 \times 10^3$ J/kgK
- $c_p = 1.005$ kJ/kgK $= 1.005 \times 10^3$ J/kgK
- $c_v = 0.718$ kJ/kgK $= 0.718 \times 10^3$ J/kgK
- $R = 0.287$ kJ/kgK $= 0.287 \times 10^3$ J/kgK

To calculate:

(i)   Maximum pressure (p_3)

(ii) Maximum temperature (T_3)

(iii) Cycle efficiency (η)

(iv) MEP (p_m)

p-V diagram

Solution:

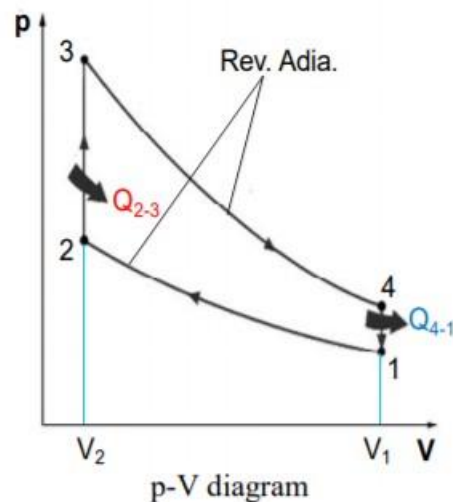Given that: $r_c = \dfrac{v_1}{v_2} = 8$

Cycle efficiency:

$$\eta = 1 - \frac{1}{r_c^{\gamma-1}} = 1 - \frac{1}{8^{1.4-1}} = 0.5647 (56.47\%)$$

Also, Cycle efficiency:

$$\eta = \frac{w_{net}}{q_{2-3}} \Rightarrow w_{net} = \eta \times q_{2-3}$$

$$\therefore w_{net} = 0.5647 \times 2100 \times 10^3 = 1185.87 \times 10^3 \, J/kg$$

At State 1:

$$p_1 v_1 = RT_1 \Rightarrow v_1 = \frac{RT_1}{p_1} = \frac{0.287 \times 10^3 \times 308}{0.1 \times 10^6}$$

$$\therefore v_1 = 0.88 m^3 / kg$$

Given that:

$$r_c = \frac{v_1}{v_2} = 8 \Rightarrow v_2 = \frac{v_1}{8} = \frac{0.88}{8}$$

$$\therefore v_2 = 0.11 m^3 / kg$$

Mean effective pressure:

$$p_m = \frac{w_{net}}{v_1 - v_2} = \frac{1185.87 \times 10^3}{0.88 - 0.11} = 1540.09 \times 10^3 \, N / m^2$$

Given that: $c_p = 1.005$ kJ/kgK, $c_v = 0.718$ kJ/kgK

$$\gamma = \frac{c_p}{c_v} = \frac{1.005}{0.718} = 1.4$$

For isentropic process 1-2:

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = \left(r_c\right)^{\gamma-1} \Rightarrow T_2 = T_1 r_c^{\gamma-1}$$

$$\therefore T_2 = 308 \times 8^{1.4-1} = 707.6 K (434.6^0 C)$$

and

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = \left(r_c\right)^{\gamma} \Rightarrow p_2 = p_1 r_c^{\gamma}$$

$$\therefore p_2 = 0.1 \times 10^6 \times 8^{1.4} = 1.84 \times 10^6 \, N / m^2$$

Given that: $q_{2-3} = 2100 \times 10^3$ J/kg

$$q_{2-3} = c_v \left(T_3 - T_2\right) \Rightarrow 2100 \times 10^3 = 0.718 \times 10^3 \left(T_3 - 707.6\right)$$

$$\therefore T_3 = 3632.39 K (3359.39^0 C)$$

For constant volume process 2-3:

$$\frac{p_3}{p_2} = \frac{T_3}{T_2} \Rightarrow p_3 = p_2 \left(\frac{T_3}{T_2}\right) = 1.84 \times 10^6 \left(\frac{3632.39}{707.6}\right)$$

$$\therefore p_3 = 9.46 \times 10^6 \, N / m^2$$

2. An engine of 250 mm bore and 375 mm stroke works on constant volume cycle. The clearance volume is 0.00263 m3 . The initial pressure and temperature are 1 bar and 500C. If the maximum pressure is 25 bar, determine (i) the air standard efficiency of the cycle and (ii) the mean effective pressure.
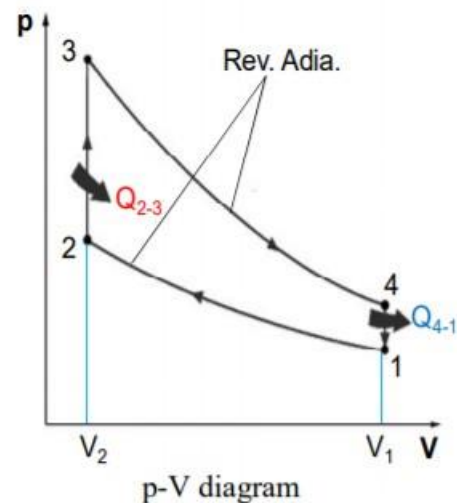
Data given:

Constant Volume Cycle (Otto Cycle)

- $d = 250$ mm $= 250 \times 10^{-3}$ m

- $L = 375$ mm $= 375 \times 10^{-3}$ m

- $v_2 = v_3 = 0.00263$ m^3

- $p_1 = 1$ bar $= 1 \times 10^5$ N/m^2

- $T_1 = 50^0C = 50 + 273 = 323$ K

- $p_3 = 25$ bar $= 25 \times 10^5$ N/m^2

For air, assume that:

- $c_p = 1.005$kJ/kgK $= 1.005 \times 10^3$ J/kgK

- $c_v = 0.718$ kJ/kgK $= 0.718 \times 10^3$ J/kgK

- $R = 0.287$ kJ/kgK $= 0.287 \times 10^3$ J/kgK

- $\gamma = 1.4$

p-V diagram

To determine:

(i) Air standard efficiency (η_{Otto})

(ii) Mean effective pressure (p_m)

T-s diagram

Solution:

Stroke volume, V_s:

$$V_s = \frac{\pi d^2 L}{4} = \frac{\pi \times \left(250 \times 10^{-3}\right)^2 \times 375 \times 10^{-3}}{4} = 0.0184 m^3$$
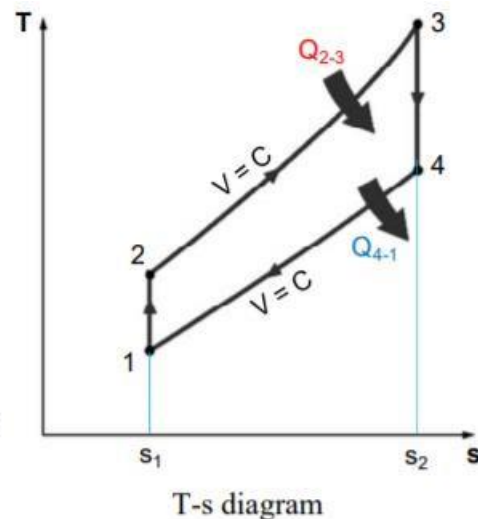
Total volume, V:

$$V = V_c + V_s = 0.00263 + 0.0184 = 0.02103 m^3$$

Compression ratio, r_c:

$$r_c = \frac{V_1}{V_2} = \frac{0.02103}{0.00263} = 7.996$$

Air standard efficiency, η:

$$\eta = 1 - \frac{1}{r_c^{\gamma-1}} = 1 - \frac{1}{7.996^{1.4-1}} = 0.5646(56.46\%)$$

For isentropic process 1-2:

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (r_c)^{\gamma} \Rightarrow p_2 = p_1 r_c^{\gamma}$$

$$\therefore p_2 = 1 \times 10^5 \times 7.996^{1.4} = 18.37 \times 10^5 \, N/m^2$$

Given that: $p_3 = 25 \times 10^5 \, N/m^2$

Explosion ratio, α:

$$\alpha = \frac{p_3}{p_2} = \frac{25}{18.37} = 1.36$$

Mean effective pressure, p_m:

$$p_m = \frac{p_1 r_c (\alpha - 1)\left(r_c^{\gamma-1} - 1\right)}{(\gamma - 1)(r_c - 1)}$$

$$p_m = \frac{1 \times 10^5 \times 7.996(1.36 - 1)\left(7.996^{1.4-1} - 1\right)}{(1.4 - 1)(7.996 - 1)}$$

$$\therefore p_m = 1.334 \times 10^5 \, N/m^2$$

3. The compression ratio of an air standard diesel cycle is 16. The temperature and pressure at the beginning of isentropic compression are 150C and 0.1 MPa respectively. During the constant pressure process, the heat is added until the temperature reaches 14800C. Determine (i) the cutoff ratio (ii) the heat supplied per kg of air, (iii) the cycle efficiency, and (iv) the m.e.p

Data given:

- Diesel Cycle

- $r_c = (v_1 / v_2) = 16$

- $T_1 = 15^0C = 15 + 273 = 288$ K

- $p_1 = 0.1$ MPa $= 0.1 \times 10^6 \, N/m^2$

- $T_3 = 1480^0C = 1480 + 273 = 1753$ K

For air, assume that

- $c_p = 1.005 \text{kJ/kgK} = 1.005 \times 10^3 \text{ J/kgK}$

- $c_v = 0.718 \text{ kJ/kgK} = 0.718 \times 10^3 \text{ J/kgK}$

- $R = 0.287 \text{ kJ/kgK} = 0.287 \times 10^3 \text{ J/kgK}$

- $\gamma = 1.4$

To calculate:

(i) Cut-off ratio (v_3/v_2)

(ii) Heat supplied per kg of air ($q_{2\text{-}3}$)

(iii) Cycle efficiency (η)

(iv) MEP (p_m)

Solution:

For isentropic process 1-2:

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = \left(r_c\right)^{\gamma-1} \Rightarrow T_2 = T_1 r_c^{\gamma-1}$$

$$\therefore T_2 = 288 \times 16^{1.4-1} = 873.05 K \, (600.05^0 C)$$

For constant pressure process 2-3:

$$Cut - off \ \ ratio: \rho = \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{1753}{873.05} = 2.01$$

Heat Supplied, $q_{2\text{-}3}$:

$$q_{2-3} = c_p \left(T_3 - T_2\right) = 1.005 \times 10^3 \times \left(1753 - 873.05\right)$$

$$\therefore q_{2-3} = 884.35 \times 10^3 \, J \, / \, kg$$

For Isentropic expansion process 3-4:

$$\frac{T_4}{T_3} = \left(\frac{\rho}{r_c}\right)^{\gamma-1} \Rightarrow T_4 = T_3 \left(\frac{\rho}{r_c}\right)^{\gamma-1} = 1753 \times \left(\frac{2.01}{16}\right)^{1.4-1}$$

$$\therefore T_4 = 764.56 K \, (491.56^0 C)$$

Heat rejected, $q_{4\text{-}1}$:

$$q_{4-1} = c_v \left(T_4 - T_1\right) = 0.718 \times 10^3 \times \left(764.56 - 288\right)$$

$$\therefore q_{4-1} = 342.17 \times 10^3 \, J \, / \, kg$$

Cycle efficiency, η:

$$\eta = 1 - \frac{q_{4-1}}{q_{2-3}} = 1 - \frac{342.17}{884.35} = 0.6131(61.31\%)$$

At State 1:

$$p_1 v_1 = RT_1 \Rightarrow v_1 = \frac{RT_1}{p_1} = \frac{0.287 \times 10^3 \times 288}{0.1 \times 10^6} = 0.827 m^3 / kg$$

Mean effective pressure, p_m:

$$p_m = \frac{w_{net}}{v_s} = \frac{q_{2-3} - q_{4-1}}{v_1 - v_2} = \frac{q_{2-3} - q_{4-1}}{v_1 \left(1 - \frac{1}{r_c}\right)} = \frac{884.35 - 342.17}{0.827 \left(1 - \frac{1}{16}\right)}$$

$$\therefore p_m = 6.993 \times 10^5 \, N / m^2$$

INTERNAL COMBUSTION ENGINES

INTRODUCTION

As the name implies or suggests, the internal combustion engines (briefly written as IC engines) are those engines in which the combustion of fuel takes place inside the engine cylinder. These are petrol, diesel, and gas engines.

CLASSIFICATION OF IC ENGINES

The internal combustion engines may be classified in many ways, but the following are important from the subject point of view

1. According to the type of fuel used

   (a) Petrol engines. (b) Diesel engines or oil engines, and (c) Gas engines.

2. According to the method of igniting the fuel

   (a) Spark ignition engines (briefly written as S.1. engines), (b) Compression ignitionengines (briefly written as C.I. engines), and (c) Hot spot ignition engines

3. According to the number of strokes per cycle

   (a) Four stroke cycle engines, and (b) Two stroke cycle engines.

4. According to the cycle of operation

   (a) Otto. cycle (also known as constant volume cycle) engines, (b) Diesel cycle (also known as constant pressure cycle) engines, and (c) Dual combustion cycle (also known as semi- diesel cycle) engines.

5. According to the speed of the engine

   (a) Slow speed engines, (b) Medium speed engines, (c) High speed engines.

6. According to the cooling system

   (a) Air-cooled engines. (b) Water-cooled engines. (c) Evaporative cooling engines.

7. According to the method of fuel injection

   (a) Carburetor engines, (b) Air injection engines, (c) Airless or solid injection
      engines.

8. According to the number of cylinders

   (a) Single cylinder engines (b) Multi-cylinder engines.

9. According to the arrangement of cylinders

   (a) Vertical engines, (b) Horizontal engines, (c) Radial engines, (d) In-line multi-cylinder
      engines, (e)V-type multi-cylinder engines, (j) Opposite-cylinder engines, (g) Opposite-
      piston engines

MAIN COMPONENTS OF IC ENGINES
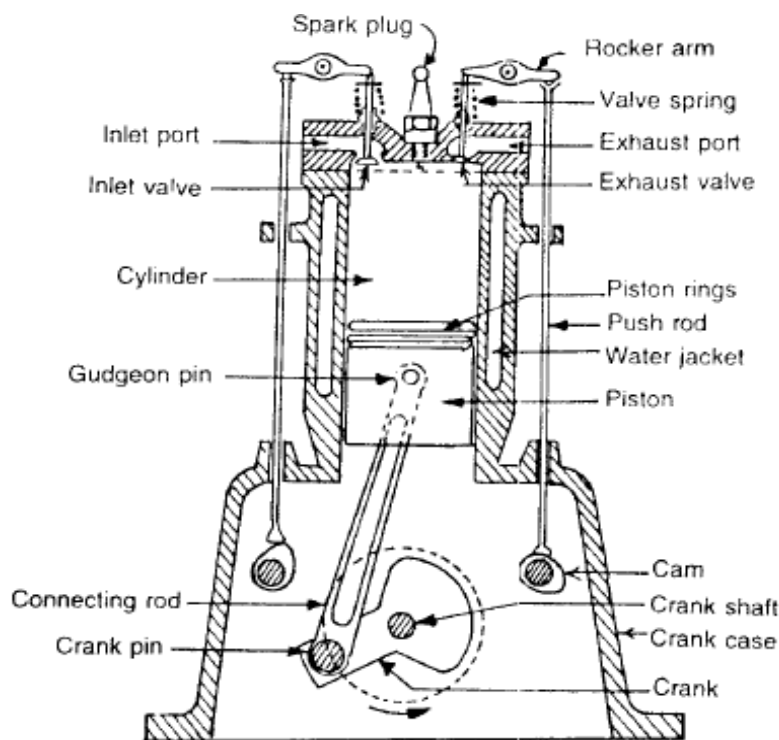
Fig.7

As a matter of fact, an IC engine consists of hundreds of different parts, which are important for its proper working. The description of all these parts is beyond the scope. However, the main components, which are important from academic point of view, are shown and are discussed below:

1. Cylinder. It is one of the most important part of the engine, in which the piston moves

to and fro in order to develop power. Generally, the engine cylinder has to withstand a high pressure (more than 50 bar) and temperature (more than 2000°C). Thus the materials for an engine cylinder should be such that it can retain sufficient strength at such a high pressure and temperature. For ordinary engines, the cylinder is made of ordinary cast iron. But for heavy duty engines, it is made of steel alloys or aluminium alloys. In case of multiple cylinder engines, the cylinders are cast in one block known as cylinder block.Sometimes, a liner or sleeve is inserted into the cylinder, which can be replaced when worn out. As the material required for liner is comparatively small, it can be made of alloy cast iron having long life and sufficient resistance to rapid wear and tear to the fast moving reciprocating parts.

2. Cylinder head: It is fitted on one end of the cylinder, and acts as a cover to close the cylinder bore. Generally, the cylinder head contains inlet and exit valves for admitting fresh charge and exhausting the burnt gases. In petrol engines, the cylinder head also contains a spark plug for igniting the fuel-air mixture, towards the end of compression stroke. But in diesel engines, the cylinder head contains nozzle (*i.e.* fuel valve) for injecting the fuel into the cylinder. The cylinder head is, usually, cast as one piece and bolted to one end of the cylinder. Generally, the cylinder block and cylinder head are made from the same material. A copper or asbestos gasket is provided between the engine cylinder and cylinder head to make an air-tight joint.

3. Piston: It is considered as the heart of an I.C. engine, whose main function is to transmit the force exeI1ed by the burning of charge to the connecting rod. The pistons are generally made of aluminium alloys which are light in weight. They have good heat conducting property and also greater strength at higher temperatures.

4. Piston rings: These are circular rings and made of special steel alloys which retain elastic properties even at high temperatures. The piston rings are housed in the circumferential grooves provided on the outer surface of the piston. Generally, there are two sets of rings mounted for the piston. The function of the upper rings is to provide air tight seal to prevent leakage of the burnt gases into the lower portion. Similarly, the function of the lower rings is to provide effective seal to prevent leakage of the oil into the engine cylinder.

5. Connecting rod: It is a link between the piston and crankshaft, whose main function is to transmit force from the piston to the crankshaft. Moreover, it converts reciprocating motion of the piston into circular motion of the crankshaft, in the working stroke. The upper (*i.e.* smaller) end of the connecting rod is fitted to the piston and the lower (*i.e.* bigger) end to the crank. The special steel alloys or aluminium alloys are used for the manufacture of connecting rods. A special care is required for the design and manufacture of connecting rod, as it is subjected to alternatively compressive and tensile stresses as well as bending stresses.

6. Crankshaft: It is considered as the backbone of an l.c. engine whose function is to convert the reciprocating motion of the piston into the rotary motion with the help of connecting rod. This shaft contains one or more eccentric portions called cranks. That part of the crank, to which bigger end of the connecting rod is fitted, is called crank pin.It has been experienced that too many main bearings create difficulty of correct alignment. Special steel alloys are used for the manufacture of crankshaft. A special care is required for the design and manufacture of crankshaft.

7. Crank case: It is a cast iron case, which holds the cylinder and crankshaft of an I.c. engine. It also serves as a sump for the lubricating oil. The lower portion of the crank case is known as bed plate, which is fixed with the help of bolts.

8. Flywheel: It is a big wheel, mounted on the crankshaft, whose function is to maintain its speed constant. It is done by storing excess energy during the power stroke, which is returned during other strokes.

FOUR STROKE CYCLE PETROL ENGINE

It is also known as Otto cycle. It requires four strokes of the piston to complete one cycle of operation in the engine cylinder. The four strokes of a petrol engine sucking fuel-air mixture (petrol mixed with proportionate quantity of air in the carburetor known as charge) are described below:

1. Suction or charging stroke: In this stroke, the inlet valve opens and charge is sucked into the cylinder as the piston moves downward from top dead centre *(T.D.C.)*. It continues till the piston reaches its bottom dead centre *(B.D.* C.) as shown in (a).

2. Compression stroke: In this stroke, both the inlet and exhaust valves are closed and the charge is compressed as the piston moves upwards from *B.D.* C. to *TD.* C. As a result of compression, the pressure and temperature of the charge increases considerably (the actual values depend upon the compression ratio). This completes one revolution of the crank shaft. The compression stroke is shown in (b).

(*a*) Suction or charging stroke.   (*b*) Compression stroke.   (*c*) Expansion or working stroke.   (*d*) Exhaust stroke.
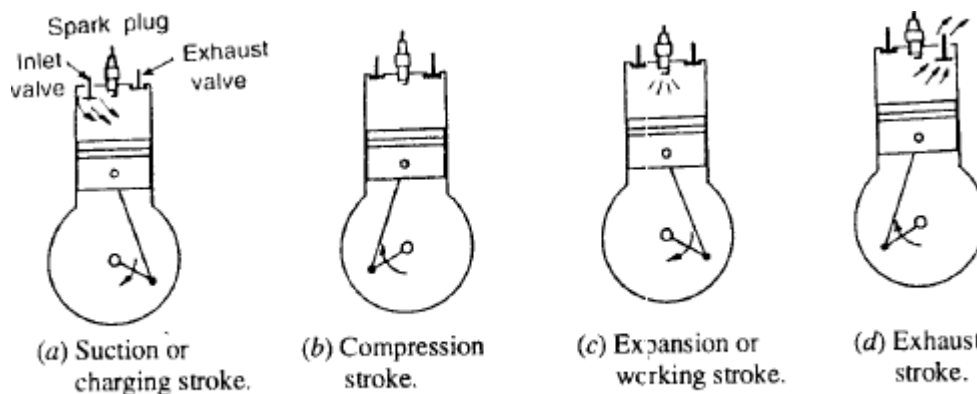
Four-stroke cycle petrol engine.

Fig.8

3. **Expansion or working stroke** Shortly before the piston reaches *T.D.*C. (during compression stroke), the charge is ignited with the help of a spark plug. It suddenly increases the pressure and temperature of the products of combustion but the volume, practically, remains constant. Due to the rise in pressure, the piston is pushed down with a great force. The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy produced is transformed into mechanical work. It may be noted that during this working stroke, as shown in *(c),* both the valves are closed and piston moves from *T.D.C.* to *B.D.C*

4. **Exhaust stroke:** In this stroke, the exhaust valve is open as piston moves from *B.D.C.* to *T.D.*C. This movement of the piston pushes out the products of combustion, from the engine cylinder and are exhausted through the exhaust valve into the atmosphere, as shown in *(d).* This completes the cycle, and the engine cylinder is ready to suck the charge again.

FOUR-STROKE CYCLE DIESEL ENGINE

It is also known as *compression ignition engine* because the ignition takes p\ace due to the heat produced in the engine cylinder at the end of compression stroke. The four strokes of a diesel engine sucking pure air are described below:

1. Suction or charging stroke: In this stroke, the inlet valve opens and pure air is sucked into the cylinder as the piston moves downwards from the top dead centre *(TDC).* It continues till the piston reaches its bottom dead centre *(BDC)* as shown *(a).*

2. **Compression stroke:** In this stroke, both the valves are closed and the air is compressed as the piston moves upwards from *BDC* to *TDC.* As a result of compression, pressure and temperature of the air increases considerably (the actual value depends upon the compression ratio). This completes one revolution of the crank shaft. The compression stroke is shown in *(b).*
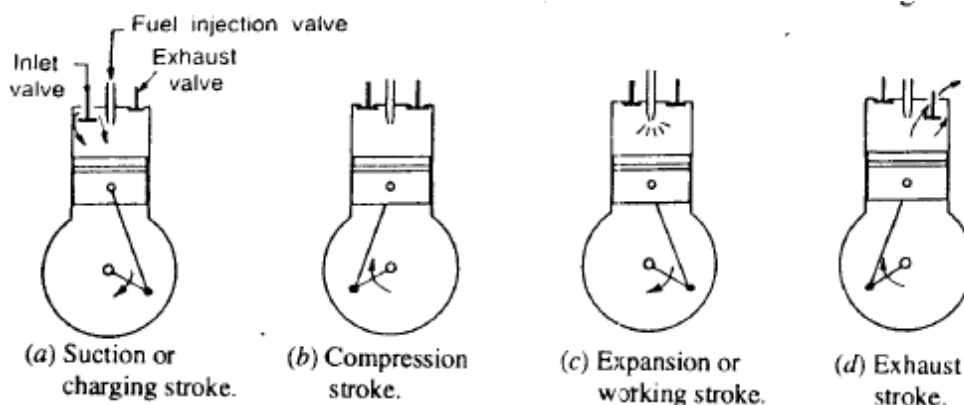

(a) Suction or charging stroke.   (b) Compression stroke.   (c) Expansion or working stroke.   (d) Exhaust stroke.

Fig.9

3. Expansion or working stroke: Shortly before the piston reaches the *TDC* (during the compression stroke), fuel oil is injected in the form of very fine spray into the engine cylinder, through the nozzle, known as fuel injection valve. At this moment temperature of the compressed

air is sufficiently high to ignite the fuel. It suddenly increases the pressure and temperature of the products of combustion. The fuel oil is continuously injected for a fraction of the revolution. The fuel oil is assumed to be burnt at constant pressure. Due to increased pressure, the piston is pushed down with a great force. The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy is transformed into mechanical work. It may be noted that during this working stroke, both the valves are closed and the piston moves from *TDC* to *BDC*.

4. Exhaust stroke: In this stroke, the exhaust valve is open as the piston moves from *BDC* to *TDC*. This movement of the piston pushes out the products of combustion from the engine cylinder through the exhaust valve into the atmosphere. This completes the cycle and the engine cylinder is ready to suck the fresh air again.

TWO-STROKE CYCLE PETROL ENGINE

A two-stroke cycle petrol engine was devised by Duglad Clerk in I RHO. In this cycle, the suction, compression, expansion and exhaust takes place during two strokes of the piston. It means that there is one working stroke after every revolution of the crank shaft. A two stroke engine has ports instead of valves. All the four stages of a two stroke petrol engine are described below:

1. Suction stage: In this stage, the piston, while going down towards *BDC,* uncovers both the transfer port and the exhaust port The fresh fuel-air mixture flows into the engine cylinder from the crank case, as shown *(a).*

2. Compression stage: In this stage, the piston, while moving up, first covers the transfer port and then exhaust port. After that the fuel is compressed as the piston moves upwards as shown *(b).* In this stage, the inlet port opens and fresh fuel-air mixture enters into the crank case.

3. Expansion stage: Shortly before this piston reaches the *TDC* (during compression stroke), the charge is ignited with the help of a spark plug. It suddenly increases the pressure and temperature of the products of combustion. But the volume, practically, remains constant. Due to rise in the pressure, the piston is pushed downwards with a great force as shown in *(c).* The hot burnt gases expand due to high speed of the piston. During this expansion, some of the heat energy produced is transformed into mechanical work.
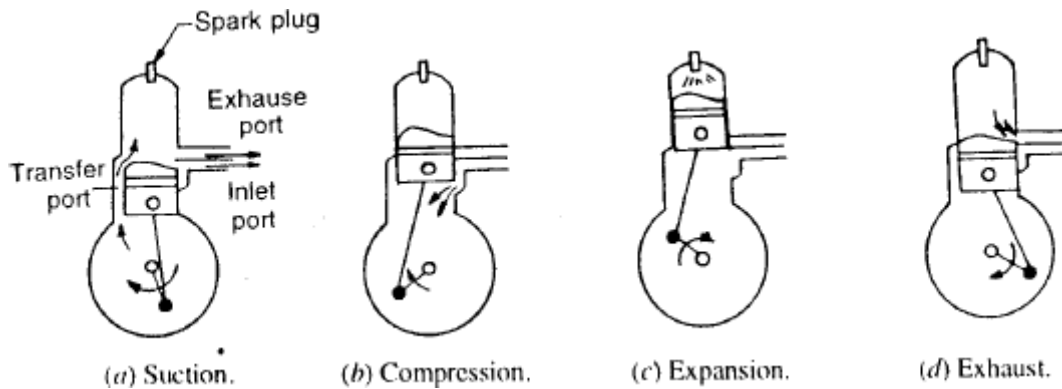
Fig.10

1. Exhaust stage: In this stage, the exhaust port is opened as the piston moves downwards. The products of combustion, from the engine cylinder are exhausted through the exhaust port into the atmosphere, as shown *(d).* This completes the cycle and the engine cylinder is ready to suck the charge again

TWO STROKE AND FOUR STROKE CYCLE ENGINE

In a two-stroke engine, the working cycle is completed in two strokes of the piston or one revolution of the crankshaft. This is achieved by carrying out the suction and compression processes in one stroke (or more precisely in inward stroke), expansion and exhaust processes in the second stroke (or more precisely in outward stroke). In a four-stroke engine, the working cycle is completed in four-strokes of the piston or two-revolutions of the crankshaft. This is achieved by carrying out suction, compression, expansion and exhaust processes in each stroke. It will be interesting to know that from the thermodynamic point of view, there is no difference between two-stroke and four-stroke cycle engines. The difference is purely mechanical.

Advantages and Disadvantage of Two-stroke over Four-stroke Cycle Engines

Advantages

1. A two stroke cycle engine gives twice the number of power strokes than the four stroke cycle engine at the same engine speed. Theoretically, a two-stroke cycle engine should develop twice the power as that of a four-stroke cycle engine. But in actual practice, a two-stroke cycle engine develops 1.7 to 1.8 times greater value for slow speed engines the power developed by four-stroke cycle engine of the same dimensions and speed. This is due to lower compression ratio and effective stroke being less than the theoretical stroke.

2. For the same power developed, a two-stroke cycle engine is lighter, less bulky and occupies less floor area. Thus it makes, a two-stroke cycle engine suitable for marine engines and other light vehicles.

3. As the number of working strokes in a two-stroke cycle engine are twice than the

four-stroke cycle engine, so the turning moment of a two-stroke cycle engine is more uniform. Thus it makes a two-stroke cycle engine to have a lighter flywheel and foundations. This also leads to a higher mechanical efficiency of a two-stroke cycle engine.

4. The initial cost of a two-stroke cycle engine is considerably less than a four-stroke cycle engine.

5. The mechanism of a two-stroke cycle engine is much simpler than a four-stroke cycle engine.

6. The two-stroke cycle engines are much easier to start.

Disadvantages

1. Thermal efficiency of a two-stroke cycle engine is less than that a four-stroke cycle engine, because a two-stroke cycle engine has less compression ratio than that of a four-stroke cycle engine.

2. Overall efficiency of a two stroke cycle engine is also less than that of a four-stroke cycle engine because in a two-stroke cycle, inlet and exhaust ports remain open simultaneously for some time. In spite of careful design, a small quantity of charge is lost from the engine cylinder.

3. In case of a two-stroke cycle engine, the number of power strokes is twice as those of a four-stroke cycle engine. Thus the capacity of the cooling system must be higher. Beyond a certain limit, the cooling capacity offers a considerable difficulty. Moreover, there is a greater wear and tear in a two-stroke cycle engine.

4. The consumption of lubricating oil is large in a two-stroke cycle engine because of high operating temperature.

5. The exhaust gases in a two-stroke cycle engine create noise, because of short time available for their exhaust.

COMPARISON OF PETROL AND DIESEL ENGINES (SI and CI Engines)

Following points are important for the comparison of petrol engines and diesel engines:

Table.1

Petrol Engines	Diesel Engines
• A petrol engine draws a mixture of petrol and air during suction stroke.	• A diesel engine draws only air during suction stroke
• The carburetor is employed to mix air and petrol in the required proportion and to supply it to the	• The injector or atomizer is employed to inject the fuel at the end of compression stroke.

engine during suction stroke	• Pressure at the end of compression is about 35 bar.
• Pressure at the end of compression is about 10 bar	• The fuel is injected in the form of fine spray. The temperature of the compressed air (about 600"C at a pressure of about *35*bar) is sufficiently high to ignite the fuel.
• The charge *(i.e.* petrol and air mixture) is ignited with the help of spark plug	
• The combustion of fuel takes place approximately at constant volume. In other words, it works on Otto cycle	• The combustion of fuel takes place approximately at constant pressure. In other words. It works on Diesel cycle.
• A petrol engine has compression ratio approximately from 6 to 10.	• A diesel engine has compression ratio approximately from 15 to 25.
• The starting' is easy due to low compression ratio.	• The starting is little difficult due. to high compression ratio.
• As the compression ratio is low, the petrol engines are lighter and cheaper.	• As the compression ratio is high. the diesel engine;; are heavier and costlier.
• The running cost of a petrol engine is high because of the higher cost of petrol.	• The running cost of diesel engine is low because of the lower cost of diesel.
• The maintenance cost is less.	• The maintenance cost is more.
• The thermal efficiency is up to about 26%.	• The thermal efficiency is up to about 40%
• Overheating trouble is more due to low thermal efficiency.	• Overheating trouble is less due to high thermal efficiency
• These are high speed engines.	• These are relatively low speed engines.
• The petrol engines arc generally employed in light duty vehicles such as scooters, motorcycles, cars. These are also used in aero planes	• The diesel engines are generally employed in heavy duty vehicles like buses. trucks, and earth moving machines etc.

Performance Calculation: Engine performance is an indication of the degree of success of the engine performs its assigned task, i.e. the conversion of the chemical energy contained in the fuel into the useful mechanical work. The performance of an engine is evaluated on the basis of the following : (a) Specific Fuel Consumption. (b) Brake Mean Effective Pressure. I Specific Power Output. (d) Specific Weight. (e) Exhaust Smoke and Other Emissions. The particular application of the engine decides the relative importance of these performance parameters. For Example : For an aircraft engine specific weight is more important whereas for an industrial engine specific fuel consumption is more important. For the evaluation of an engine performance few more parameters are chosen and the effect of various operating conditions, design concepts and modifications on these parameters are studied. The basic performance parameters are the following : (a) Power and Mechanical Efficiency. (b) Mean Effective Pressure and Torque. I Specific Output. (d) Volumetric Efficiency. (e) Fuel-air Ratio. (f) Specific Fuel Consumption. (g) Thermal Efficiency and Heat Balance. (h) Exhaust Smoke and Other Emissions. (i) Specific Weight.

Power and Mechanical Efficiency The main purpose of running an engine is to obtain mechanical power. • Power is defined as the rate of doing work and is equal to the product of force and linear velocity or the product of torque and angular velocity. Thus, the measurement of power involves the measurement of force (or torque) as well as speed. The force or torque is measured with the help of a dynamometer and the speed by a tachometer. The power developed by an engine and measured at the output shaft is called the brake power (bp) and is given by

$$BP = 2\Pi NT/60$$

Where, T is torque in N-m and N is the rotational speed in revolutions per minute. The total power developed by combustion of fuel in the combustion chamber is, however, more than the BP and is called indicated power (IP). Of the power developed by the engine, i.e. IP, some power is consumed in overcoming the friction between moving parts, some in the process of inducting the air and removing the products of combustion from the engine combustion chamber.

Indicated Power It is the power developed in the cylinder and thus, forms the basis of evaluation of combustion efficiency or the heat release in the cylinder.

$$I.P = PmLANK/60$$

Where,
Pm = Mean effective pressure, N/m^2, L = Length of the stroke, m, A = Area of the piston, m^2, N = Rotational speed of the engine, rpm (It is N/2 for four stroke engine), and k = Number of cylinders. Thus, we see that for a given engine the power output can be measured in terms of mean effective pressure. The difference between the IP and BP is the indication of the power lost in the mechanical components of the engine (due to friction) and forms the basis of mechanical efficiency; which is defined as follows:

Mechanical efficiency = BP/IP

The difference between ip and bp is called friction power (FP). FP = IP − Bp Mechanical efficiency = BP/(BP+FP)

Mean Effective Pressure and Torque Mean effective pressure is defined as a hypothetical/average pressure which is assumed to be acting on the piston throughout the power stroke. Therefore,

$$Pm=60XI.P/LANk$$

where, Pm = Mean effective pressure, N/m^2, IP = Indicated power, Watt, L = Length of the stroke, m, A = Area of the piston, m^2, N = Rotational speed of the engine, rpm (It is N/2 for four stroke engine), and k = Number of cylinders. If the mean effective pressure is based on BP it is called the brake mean effective pressure(Pm), and if based on IP it is called indicated mean effective pressure (imep). Similarly, the friction mean effective pressure (fmep) can be defined as,

$$fmep = imep - bmep$$

The torque is related to mean effective pressure by the relation

$$B.P=2\Pi nT/60$$

$$I.P=PmLANk/60$$

$$2\Pi nT/60=[bmep.A.L.(Nk/60)]$$

or, $$T=(bmep.A.L.k)/2\pi$$

Thus, the torque and the mean effective pressure are related by the engine size. A large engine produces more torque for the same mean effective pressure. For this reason, torque is not the measure of the ability of an engine to utilize its displacement for producing power from fuel. It is the mean effective pressure which gives an indication of engine displacement utilization for this conversion. Higher the mean effective pressure, higher will be the power developed by the engine for a given displacement. Again we see that the power of an engine is dependent on its size and speed. Therefore, it is not possible to compare engines on the basis of either power or torque. Mean effective pressure is the true indication of the relative performance of different engines.

Specific Output Specific output of an engine is defined as the brake power (output) per unit of piston displacement and is given by,
Specific output=B.P/A.L

$$Constant = bmep \times rpm \bullet$$

The specific output consists of two elements – the bmep (force) available to work and the speed with which it is working. • Therefore, for the same piston displacement and bmep an engine operating at higher speed will give more output. It is clear that the output of an engine can be increased by increasing either speed or bmep. Increasing speed involves increase in the mechanical stress of various engine parts whereas increasing bmep requires better heat release and more load on engine cylinder.

Fuel-Air Ratio (F/A)
Fuel-air ratio (F/A) is the ratio of the mass of fuel to the mass of air in the fuel-air mixture. Air-fuel ratio (A/F) is reciprocal of fuel-air ratio. Fuel-air ratio of the mixture affects the combustion phenomenon in that it determines the flame propagation velocity, the heat release in the combustion chamber, the maximum temperature and the completeness of combustion. Relative fuel-air ratio is defined as the ratio of the actual fuel-air ratio to that of the stoichiometric fuel-air ratio required to burn the fuel supplied. Stoichiometric fuel-air ratio is the ratio of fuel to air is one in which case fuel is completely burned due to minimum quantity of air supplied.

Relative fuel-air ratio, =(Actual Fuel- Air ratio)/(Stoichiometric fuel-Air ratio)

Indicated Specific Fuel Consumption: This is defined as the mass of fuel consumption per hour in order to produce an indicated power of one kilo watt.

$$\text{Thus, indicated specific fuel consumption} = isfc = \frac{3600\ m_f}{IP}\ kg/kWh$$

Brake Specific fuel consumption:- This defined as the mass of fuel consumed per hour, in order to develop a brake power of one kilowatt.

$$\text{Thus, brake specific fuel consumption} = bsfc = \frac{3600\ m_f}{BP}\ kg/kWh$$

Thermal Efficiency : There are two definitions of thermal efficiency as applied to IC engines. One is based on indicated power and the other on brake power. The one based on indicated power is called as *'indicated thermal efficiency''*, and the one based on brake power is known as *"brake thermal efficiency"*.

Indicated thermal efficiency is defined as the ratio of indicated power to the energy available due to combustion of the fuel.

$$\text{Or} \quad ith = \frac{ip}{m \times CV}$$

Similarly brake thermal efficiency is defined as the ratio of brake power to energy available due to combustion of the fuel.

$$\text{Or} \quad bth = \frac{bp}{m \times CV}$$

Mechanical Efficiency: Mechanical efficiency takes into account the mechanical losses in an engine. The mechanical losses include (i) frictional losses, (ii) power absorbed by engine auxillaries like fuel pump, lubricating oil pump, water circulating pump, magneto and distributor, electric generator for battery charging, radiator fan etc., and (iii) work requited to charge the cylinder with fresh charge and work for discharging the exhaust gases during the exhaust stroke. It is defined as the ratio of brake power to indicated power. Thus

$$mech = \frac{bp}{ip}$$

Volumetric efficiency: Volumetric efficiency is the ratio of the actual mass of air drawn into the cylinder during a given period of time to the theoretical mass which should have been drawn in during the same interval of time based on the total piston displacement, and the pressure and temperature of the surrounding atmosphere.

Thus
$$v = \frac{V_a}{V_{th}}$$

where n is the number of intake strokes per minute and V_S is the stroke volume of the piston.

Example 1:- The following observations have been made from the test of a fourcylinder, two – stroke petrol engine. Diameter of the cylinder = 10 cm; stroke = 15cm; speed = 1600 rpm; Area of indicator diagram = 5.5 cm^2; Length of the indicator diagram = 55 mm; spring constant = 3.5 bar/cm; Determine the indicated power ofthe engine.

Given:- d = 0.1 m; L = 0.15 m ; No. of cylinders = K = 4; N = 1600 rpm; n = N (two – stroke); a = 5.5 cm^2; length of the diagram = l_d = 5.5. cm; spring constant = k_S = 3.5 bar/cm ;
To find: indicated power, ip.

Solution: Indicated mean effective pressure = pim = $\dfrac{a\,k_S}{l_d}$

or   pim $= \dfrac{5.5 \times 3.5}{5.5} = 3.5 \text{ bar} = 3.5 \times 10^5 \text{ N/m}^2$

Indicated power = ip = $\dfrac{pim\ LAnK}{60{,}000} = \dfrac{3.5 \times 10^5 \times 0.15 \times (3.14\ /4) \times 0.1^2 \times 1600 \times 4}{60{,}000}$

= 43.98 kW

Example 2:- A gasoline engine (petrol engine) working on Otto cycle consumes 8 litres of petrol per hour and develops 25 kW. The specific gravity of petrol is 0.75 andits calorific value is 44,000 kJ/kg. Determine the indicated thermal efficiency of the engine

Given:- Volume of fuel consumed/hour = v/t = 8 x 10^3 / 3600 cc/s ;

ip = 25 kW; CV = 44,000 kJ/kg;

Specific gravity of petrol = s = 0.75
To find: ith

$$\text{Solution: Mass of fuel consumed} = m = \frac{vs}{1000\,t} = \frac{0.75 \times 8 \times 10^3}{1000 \times 3600} = 1.67 \times 10^{-3} \text{ kg/s.}$$

$$\text{Indicated thermal efficiency} = \eta_{ith} = \frac{ip}{m_f\,CV} = \frac{25}{1.67 \times 10^{-3} \times 44000}$$

$$= 0.3402 = 34.02\,\%.$$

Example 3:- The bore and stroke of a water cooled, vertical, single-cylinder, four stroke diesel engine are 80 mm and 110 mm respectively. The torque is 23.5 N- m. Calculate the brake mean effective pressure. What would be the mean effective pressure and torque if the engine rating is 4 kW at 1500 rpm?

Given:- Diameter = d = 80 x10^{-3} = 0.008 m ; stroke = L = 0.110 m; T = 23.5 N-m;

To find (i) bmep ; (ii) bmep if bp = 4 kW and N= 1500 rpm.

Solution: (i) Relation between brake power (bp) and brake mean effective pressure (bmep) is given by

$$bp = \frac{2\pi\,NT}{60,000} = \frac{(bmep)LAn}{60,000}$$

Hence bmep = $(2\pi\,NT) / (LAn) = (2\pi\,NT) / \{(L\,d^2\,\pi/4)\,N/2\}$

$$= \frac{16T}{d^2 L} = \frac{16 \times 23.5}{0.08^2 \times 0.11} = 5.34 \times 10^5 \text{ N}/m^2 = 5.34 \text{ bar}$$

(ii) when bp = 4 kw and N = 1500 rpm, we have

$$bmep = \frac{60,000\,bp}{LAn} = \frac{60,000 \times 4}{0.110 \times (3.14\,/4) \times 0.08^2 \times (1500/2)}$$

$= 5.79 \times 10^5$ N/m^2 = 5.79 bar.

Also bp $= 2\pi$ NT / 60,000 or T $= \dfrac{60,000 \ bp}{2\pi \ N} = \dfrac{60,000 \times 4}{2 \times \pi \times 1500} = 25.46$ N – m.

Example 4:-Find the air fuel ratio of a four stroke, single cylinder, air cooled enginewith fuel consumption time for 10 cc is 20.4 s and air consumption time for 0.1 m^{3}is 16.3 s. The load is 7 N at the speed of 3000 rpm. Find also the brake specific fuelconsumption in kg/kWh and brake thermal efficiency. Assume the density of air as 1.175 kg/m^3 and specific gravity of the fuel to be 0.7. The lower heating value of the fuel is 43MJ/kg and the dynamometer constant is 5000.

Given:- v= 10 cc ; t = 20.4 s ; V_a = 0.1 m^3; t_a = 16.3 s ; W = 7 N ; N = 3000 rpm;

$_a$ = 1.175 kg/m^3; s = 0.7 ; CV = 43 x 10^3 kJ/kg; Dynamometer constant = C = 5000.

To find:- (i) m_a / m_f ; (ii) bsfc ; (iii)bth

Solution: (i) Mass of air consumed = m_a = $\dfrac{0.1 \times 1.175}{16.3}$ = 7.21 x 10^{-3} kg/s.

Mass of fuel consumed = m_f = $\dfrac{v \ s}{1000 \ t}$ = $\dfrac{10 \times 0.7}{1000 \times 20.4}$ = 0.343 x 10^{-3} kg/s

Air fuel ratio = $\dfrac{m_a}{m_f}$ = $\dfrac{7.21 \times 10^{-3}}{0.343 \times 10^{-3}}$ = 21

(ii) Brake power = bp = WN / C = $\dfrac{7 \times 3000}{5000}$ = 4.2 kW

bsfc = $\dfrac{m_f \times 3600}{bp}$ = $\dfrac{0.343 \times 10^{-3} \times 3600}{4.2}$ = 0.294 kg/kWh

(iii) bith = $\dfrac{bp}{m_f \ CV}$ = $\dfrac{4.2}{0.343 \times 10^{-3} \times 43 \times 10^3}$ = 0.2848 = 28.48

UNIT – 4 – AIR COMPRESSORS – SMEA1405

Introduction:

Compression of air and vapour plays an important role in engineering fields. Compression of air is mostly used since it is easy to transmit air compared with vapour.

Uses of compressed air:

The applications of compressed air are listed below:

1) It is used in gas turbines and propulsion units.
2) It is used in striking type pneumatic tools for concrete breaking, clay or rock drilling, chipping,   caulking, riveting etc.
3) It is used in rotary type pneumatic tools for drilling, grinding, hammering etc.
4) Pneumatic lifts and elevators work by compressed air.
5) It is used for cleaning purposes
6) It is used as an atomiser in paint spray and insecticides spray guns.
7) Pile drivers, extractors, concrete vibrators require compressed air.
8) Air-operated brakes are used in railways and heavy vehicles such as buses and lorries.
9) Sand blasting operation for cleaning of iron castings needs compressed air.
10) It is used for blast furnaces and air-operated chucks.
11) Compressed air is used for starting I.C.engines and also super charging them.

Working principle of a compressor:
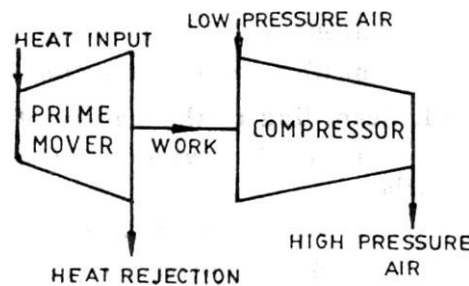
Fig:4.1 Air Compressor

A line diagram of a compressor unit is shown in fig:4.1. The compression process requires work input. Hence a compressor is driven by a prime mover. Generally, an electric motor is used as prime mover. Air from atmosphere enters into the compressor It is compressed to a high pressure. Then, this high pressure air is delivered to a storage vessel (reservoir). From the reservoir, it can be conveyed to the desired place through pipe lines.

Some of the energy supplied by the prime mover is absorbed in work done against friction. Some portion of energy is lost due to radiation and coolant. The rest of the energy is maintained within the high pressure air delivered.

Classification of compressors:

Air compressors may be classified as follows:

According to design and principle of operation:
    (a) Reciprocating compressors in which a piston reciprocates inside the cylinder.
    (b) Rotary compressors in which a rotor is rotated.

According to number of stages:
   (a) Single stage compressors in which compression of air takes place in one cylinder only.
   (b) Multi stage compressors in which compression of air takes place in more than one cylinder.
According to pressure limit:
   (a) Low pressure compressors in which the final delivery pressure is less than 10 bar,
   (b) Medium pressure compressor in which the final delivery pressure is 10 bar to 80 bar and
   (c) High pressure compressors in which the final delivery pressure is 80 to 100 bar.

According to capacity:
   (a) Low capacity compressor (delivers $0.15m^3$ /s of compressed air),
   (b) Medium capacity compressor (delivers $5m^3$ /s of compressed air) and
   (c) High capacity compressor (delivers more than $5m^3$ /s of compressed air).
According to method of cooling:
   (a) Air cooled compressor (Air is the cooling medium) *and*
   (b) Water cooled compressor (Water is the cooling medium).
According to the nature of installation:
   (a) Portable compressors (can be moved from one place to another).
   (b) Semi-fixed compressors and
   (c) Fixed compressors (They are permanently installed in one place).
According to applications:
   (a) Rock drill compressors (used for drilling rocks),
   (b) Quarrying compressors (used in quarries),
   (c) Sandblasting compressors (used for cleaning of cast iron) and
   (d) Spray painting compressors (used for spray painting).
According to number of air cylinders
   (a) Simplex - contains one air cylinder
   (b) Duplex - contains two air cylinders
   (c) Triplex - contains three air cylinders

Reciprocating compressors may be classified as follows:

   (a) Single acting compressors in which suction, compression and delivery of air (or gas) take place on one side of the piston.
   (b) Double acting compressors in which suction, compression and delivery of air (or gas) take place on both sides of the piston.

Single stage reciprocating air compressor:

   In a single stage compressor, the compression of air (or gas) takes place in a single cylinder. A schematic diagram of a single stage, single acting compressor is shown in fig:4.2.

Construction: It consists of a piston which reciprocates inside a cylinder. The piston is connected to the crankshaft by means of a connecting rod and a crank. Thus, the rotary movement of the crankshaft is converted into the reciprocating motion of the piston. Inlet and outlet valves (suction and delivery valves) are provided at the top of the cylinder.
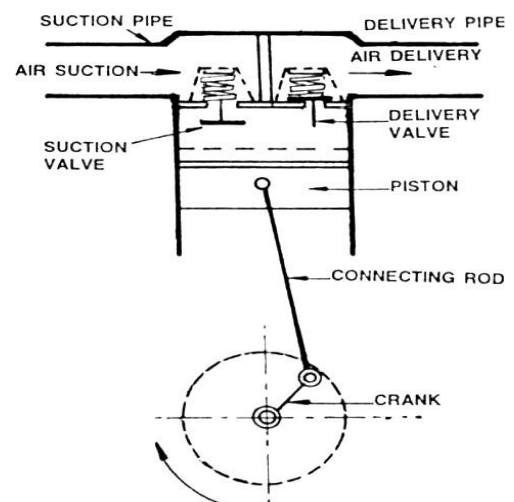

Fig :4.2 Single stage reciprocating Air Compressor

3

Working: When the piston moves down, the pressure inside the cylinder is reduced. When the cylinder pressure is reduced below atmospheric pressure, the inlet valve opens. Atmospheric air is drawn into the cylinder till the piston reaches the bottom dead centre. The delivery valve remains closed during this period. When the piston moves up, the pressure inside the cylinder increases. The inlet valve is closed, since the pressure inside the cylinder is above atmospheric. The pressure of air inside the cylinder is increased steadily. The outlet valve is then opened and the high pressure air is delivered through the outlet valve in to the deliverypipe line.

At the top dead centre of the piston, a small volume of high pressure air is left in the clearance space. When the piston moves down again, this air is expanded and pressure reduces, Again the inlet valve opens and thus the cycle is repeated.

Disadvantages
1. Handling of high pressure air results in leakage through the piston.
2. Cooling of the gas *is not effective.*
3. Requires a stronger cylinder to withstand high delivery pressure.

Applications: It is used in places where the required pressure ratio is small.

Compression processes:
The air may be compressed by the following processes.
(a) Isentropic or adiabatic compression,
(b) Polytropic compression and
(c) Isothermal compression

(a) **Isentropic(or)adiabatic compression***:*
In internal combustion engines, the air (or air fuel mixture) is compressed isentropically. By isentropic compression, maximum available energy in the gas is obtained.
(b) Polytropic compression:
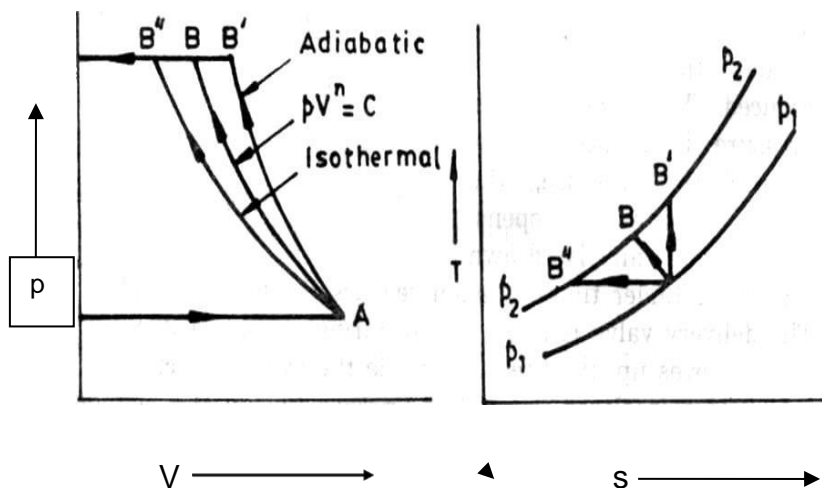
Fig: 4.3 Compression processes
A-B": Isothermal; A-B: Polytropic; A-B': Isentropic

The compression follows the law pV^n = Constant. This type of compression may be used in Bell-Coleman cycle of refrigeration.

(c) Isothermal compression:

When compressed air (or gas) is stored in a tank, it loses its heat to the surroundings. It attains the temperature of surroundings after some time. Hence, the overall effect of this compression process is to increase the pressure of the gas keeping the temperature constant. Thus isothermal compression is suitable ifthe compressed air (or gas) is to be stored.

Power required for driving the compressor:

The following assumptions are made in deriving the power required to drive the compressor.

1. There is no pressure drop through suction and delivery valves.

2. Complete compression process takes place in one cylinder.

3. There is no clearance volume in the compressor cylinder.

4. Pressure in the suction line remains constant. Similarly, pressure in the delivery line remains constant.

5. The working fluid behaves as a perfect gas.

6. There is no frictional losses.

The cycle can be analysed for the three different case of compression. Work required can be obtained fromthe p - V diagram.

Let,

$p_1 = $ Pressure of the air (kN/m^2),
before compressionV $_1 = $ Volume of
the air (m^3), before compression

$T_1 = $ Temperature of the air *(K),* before compression

p_2, V_2 and T_2 be the corresponding values after compression.

m - Mass of air induced or delivered by the cycle (kg).

N - Speed in RPM.

Polytropic Compression
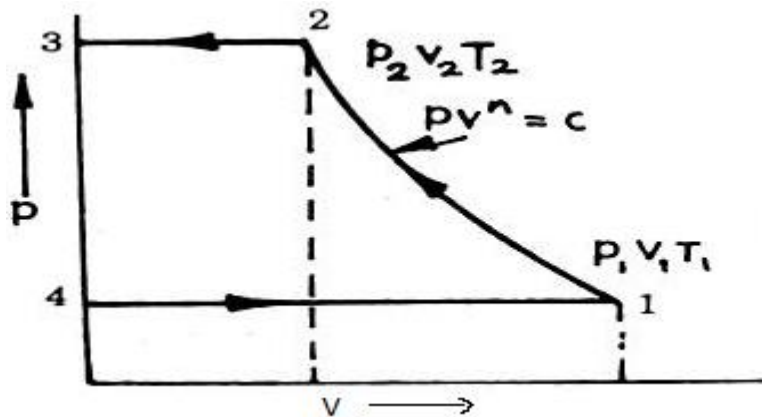
Fig:3.4 Polytropic compression
(Compression follows pV^n = Constant)

Let n= Index of polytropic compression

Net work done on air/cycle is given by

W = Area 1-2-3-4-1

 = Work done during compression (1-2) + Work done during air delivery (2-3) - Work done during suction (4-1).

$$W = \frac{p_2 v_2 - p_1 v_1}{n-1} + p_2 v_2 - p_1 v_1$$

$$W = \frac{p_2 v_{2-} p_1 + (n-1)p_2 v_2 - \ (n-1)p_1 v_1}{n-1}$$

$$= \frac{np_2 v_2 - np_1 v_1}{n-1} \ = \left(\frac{n}{n-1}\right) p_2 v_2 - p_1 v_1$$

We know that, $p_1 V_1 = m RT_1$ & $p_2 V_2 = m RT_2$

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

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

For polytropic process, $\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$

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

$W = \frac{n}{n-1}$ p_1 $V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$ kJ/cycle

Indicated power (or) Power required, P = W x N , kW for single acting reciprocating compressor;

= W x 2N, kW for double acting reciprocating compressor.

3.7.2 Isentropic compression

Compression follows, pV^{γ} = Constant

Let γ = Index of isentropic compression
Net work done on air/cycle is given by
W = Area 1-2-3-4-1
= Work done during compression (1-2) + Work done during air delivery (2-3) - Work done during suction (4-1).

$$W = \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} + p_2 v_2 - p_1 v_1$$

$$W = \frac{p_2 v_{2-} p_1 + (\gamma - 1)p_2 v_2 - \ (\gamma - 1)p_1 v_1}{\gamma - 1}$$

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

We know that, $p_1 V_1 = m RT_1$ & $p_2 V_2 = m RT_2$

$W = \frac{\gamma}{\gamma - 1}$ m R (T_2 - T_1)

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

For isentropic process, $\dfrac{T_2}{T_1} = \left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$

Therefore, $W = \dfrac{\gamma}{\gamma-1} \, m \, R \, T_1 \left[\left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$ kJ/cycle

$$W = \dfrac{\gamma}{\gamma-1} \, p_1 \, V_1 \left[\left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ kJ/cycle}$$

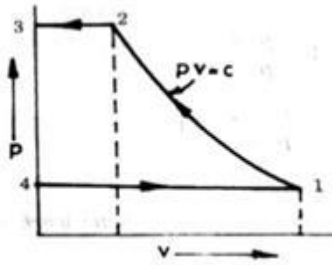## 3.7.3 Isothermal Compression

Compression follows, pV= Constant

Fig: 3.5 Isothermal Compression

Isothermal Work input, W = Area 1-2-3-4-1 = area under 1-2 + area under 2-3 - area under 4-1

$$W = p_1 V_1 \, ln\left(\dfrac{V_1}{V_2}\right) + p_2 V_2 - p_1 V_1$$

But $p_1 V_1 = p_2 V_2$

$W = p_1 V_1 \, ln\left(\dfrac{V_1}{V_2}\right) \qquad and \qquad \dfrac{V_1}{V_2} = \dfrac{p_2}{p_1}$

Therefore, W = $p_2 V_2 \, ln\left(\dfrac{p_2}{p_1}\right)$ kJ/cycle

3.8 Isothermal efficiency: Isothermal efficiency is defined as the ratio of isothermal work input to the actual work input. This is used for comparing the compressors.

Isothermal efficiency, $\eta_{iso} = \dfrac{Isothermal work input}{Actual work output}$

3.9 Adiabatic efficiency: Adiabatic efficiency is defined as the ratio of adiabatic work input to the actual work input. This is used for comparing the compressors.

Adiabatic efficiency, $\eta_{adia} = \dfrac{Adiabatic work input}{Actual work output}$

3.10 Mechanical efficiency:
The compressor is driven by a prime mover. The power input to the compressor is the shaft power (brake power) of the prime mover. This is also known as brake power of the compressor.

Mechanical efficiency is defined as the ratio of indicated power of the compressor to the power input to the compressor.

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$$\eta_m = \frac{\text{Indicated power of compressor}}{\text{Power input}}$$

Indicated Power, IP $= \frac{p_m l \, aNk}{60}$,

     where, p_m = mean effective pressure, kN/m²
     l = length of stroke of piston, m
     a = area of cross section of cylinder,m²
     N= crank speed in rpm, and
     K=number of cylinders

3.11 Clearance and clearance volume:

     When the piston reaches top dead centre (TDC) in the cylinder, there is a dead space between piston top and the cylinder head. This space is known as clearance space and the volume occupied by this space is known as clearance volume, V_c.

The clearance volume is expressed as percentage of piston displacement. Its value ranges from 5% - 10% of swept volume or stroke volume (V_s).The p - V diagram for a single stage compressor, considering clearance volume is shown in fig. . At the end of delivery of high pressure air (at point 3), a small amount of high pressure air at p_2 remains in the clearance space. This high pressure air which remains at the clearance space when the piston is at TDC is known as remnant air. It is expanded polytropically till atmospheric pressure ($p_4=p_1$) is reached. The inlet valve is opened and the fresh air is sucked into the cylinder. The suction of air takes place for the rest of stroke (upto point 1). The volume of air sucked is known as effective suction volume ($V_1 - V_4$). At point 1, the air is compressed polytropically till the delivery pressure (p_2) is reached. Then the delivery valve is opened and high pressure air is discharged into the receiver. The delivery of air continues till the piston reaches its top dead centre, then the cycle is repeated.

3.11.1 Effect of clearance volume:

The following are the effects of clearance space.
1. Suction volume (volume of air sucked) is reduced.
2. Mass of air is reduced.
3. If clearance volume increases, heavy compression is required.
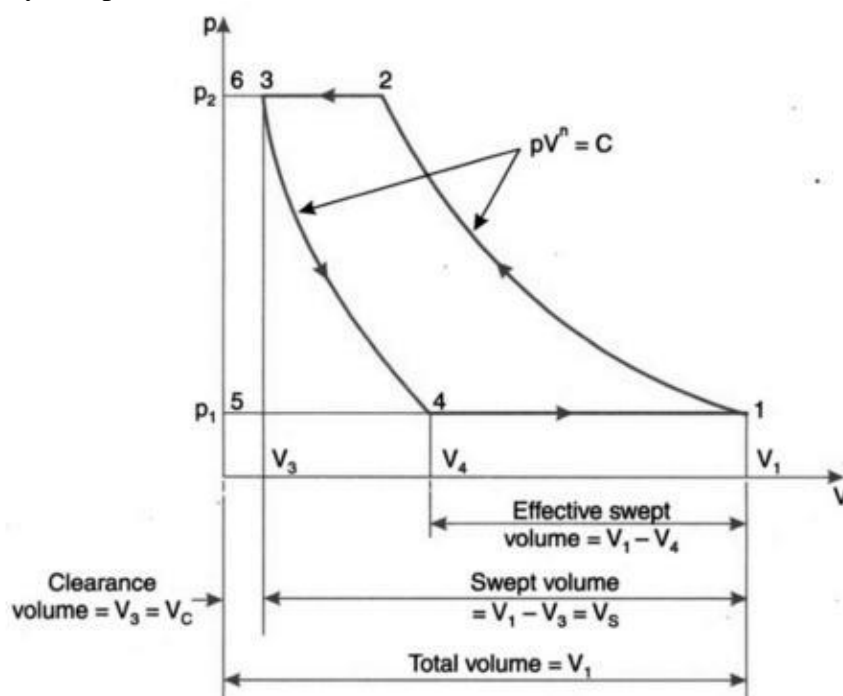4. Heavy compression increases mechanical losses

Fig: 3.6 p-V diagram with clearance volume

3.11.3 Work input considering clearance volume:

Assuming the expansion (3-4) and compression (1-2) follow the law p V^n = C,
Work input per cycle is given by,
W = Area (1-2-3-6-5-4-1)- Area (3-6-5-4 -3)
W = Workdone during compression - Work done during expansion

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} p_4 V_4 \left[\left(\frac{p_3}{p_4}\right)^{\frac{n-1}{n}} - 1 \right]$$

But, $p_3 = p_2$ and $p_4 = p_1$
therefore

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} p_1 V_4 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

$$W = \frac{n}{n-1} p_1 (V_1 - V_4) \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \text{ kJ/cycle}$$

V_1-V_4 is called as effective suction volume.

3.12 Volumetric efficiency:
The clearance volume in a compressor reduces the intake capacity of the cylinder. This leads to a term called volumetric efficiency.
The volumetric efficiency is denned as the volume of free air sucked into the compressor per cycle to the stroke volume of the cylinder,the volume measured at the intake pressure and temperature or at standard atmospheric conditions,(p_s =101.325 kN/m^2 and T_s = 288K)

Volumetric efficiency, η_{vol} = $\dfrac{\text{Volume of free air taken in per cycle}}{\text{Stroke volume of the cylinder}}$

$$= \frac{\text{Effective suction volume}}{\text{Swept volume}} = \frac{(V1 - V4)}{(V1 - V3)} = \frac{V_1 - V_4}{V_s}$$

Clearance ratio: Clearance ratio is defined as, the ratio of clearance volume to swept volume. It is denoted by the letter C.

Clearance ratio,C = $\dfrac{Clearance\ volume}{Swept\ volume} = \dfrac{V_c}{V_s} = \dfrac{V_c}{V_1 - V_3}$

Pressure ratio, R_p = $\dfrac{Delivery\ pressure}{Suction\ pressure} = \dfrac{p_2}{p_1} = \dfrac{p_3}{p_4}$

3.12.1 Expression for Volumetric efficiency
Let the compression and expansion follows the law, pV^n =Constant.

Clearance ratio, C = $\dfrac{Clearance volume}{Swept volume} = \dfrac{V_c}{V_s} = \dfrac{V_3}{V_1 - V_3}$

V_1-V_3 = $\dfrac{V_3}{C}$------------(1)
V_1 = $\dfrac{V_3}{C}$ + V_3
V_1 = $V_3 \left(\dfrac{1}{C} + 1\right)$-------------------- (2)

We know that, Pressure ratio, R_p = $\dfrac{Delivery\ pressure}{Suction\ pressure} = \dfrac{p_2}{p_1} = \dfrac{p_3}{p_4}$

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By polytropic expansionprocess 3-4:

$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^n$$

$$\frac{V_4}{V_3} = \left(\frac{p_3}{p_4}\right)^{1/n} = \left(R_p\right)^{\frac{1}{n}}$$

Therefore, $V_4 = V_3\left(R_p\right)^{\frac{1}{n}}$---------------- (3)

Volumetric efficiency, η_{vol} = $\dfrac{\text{Effective suction volume}}{\text{Swept volume}} = \dfrac{(V_1 - V_4)}{(V_1 - V_3)}$ ------------ (4)

Using equations 1,2 and 3 in 4,

$$\eta_{vol} = \frac{V_3\left(\frac{1}{c}+1\right) - V_3[R_p]^{1/n}}{\frac{V_3}{c}} = \frac{V_3\left\{\left(\frac{1}{c}+1\right) - [R_p]^{1/n}\right\}}{V_3\left(\frac{1}{c}\right)} = \frac{\left\{\left(\frac{1}{c}+1\right) - [R_p]^{1/n}\right\}}{\left(\frac{1}{c}\right)} = C\left[\left(\frac{1}{c}+1\right) - [R_p]^{1/n}\right]$$

$$\eta_{vol} = 1 + C - C[R_p]^{1/n} = 1 + C - C\left[\frac{p_2}{p_1}\right]^{1/n}$$

3.13 Multi-stage air compressor:

In a multi stage air compressor, compression of air takes place in more than one cylinder. Multi stage air compressor is used in places where high pressure air is required. Fig. shows the general arrangement of a two-stage air compressor. It consists of a low pressure (L.P) cylinder, an intercooler and a high pressure (H.P) cylinder. Both the pistons (in L.P and H.P cylinders) are driven by a single prime mover through a common shaft.

Atmospheric air at pressure p_1 taken into the low pressure cylinder is compressed to a high pressure (p_2). This pressure is intermediate between intake pressure (p_1) and delivery pressure p_3). Hence this is known as intermediate pressure.

The air from low pressure cylinder is then passed into an intercooler. In the intercooler, the air is cooled at constant pressure by circulating cold water. The cooled air from the intercooler is then taken into the high pressure cylinder. In the high pressure cylinder, air is further compressed to the final delivery pressure (p_3) and supplied to the air receiver tank.
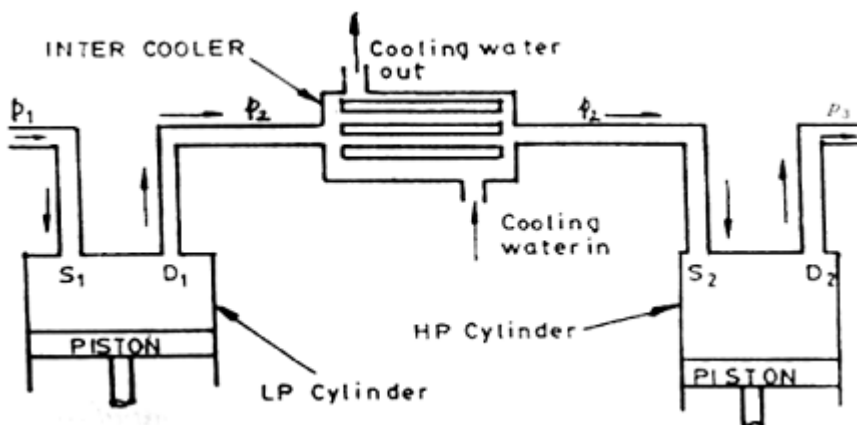

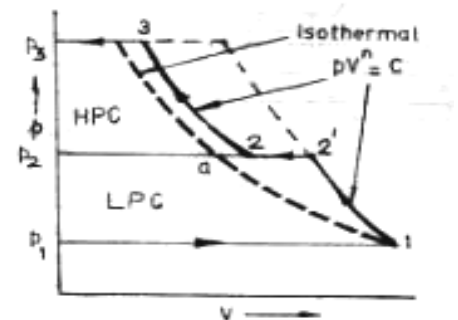Fig: 3.7   Multistage compressor (Two stage)   Fig:3.8pV diagram of two stage compressor

1

Advantages:

1. Saving in work input: The air is cooled in an intercooler before entering the high pressure cylinder. Hence less power is required to drive a multistage compressor as compared to a single stage compressor for delivering same quantity of air at the same delivery pressure.

2. Better balancing: When the air is sucked in one cylinder, there is compression in the other cylinder. This provides more uniform torque. Hence size of the flywheel is reduced.

3. No leakage and better lubrication: The pressure and temperature ranges are kept within desirablelimits. This results in a) Minimum air leakage through the piston of the cylinder and b) effective lubrication due to lower temperature.

4. More volumetric efficiency: For small pressure range, effect of expansion of the remnant air (high pressure air in the clearance space) is less. Thus by increasing number of stages, volumetric efficiency is improved.

5. High delivery pressure: The delivery pressure of air is high with reasonable volumetric efficiency.

6. Simple construction of LP cylinder: The maximum pressure in the low pressure cylinder is less. Hence, low pressure cylinder can be made lighter in construction.

7. Cheaper materials: Lower operating temperature permits the use of cheaper materials for construction.

Disadvantages:
1. More than one cylinder is required.
2 An intercooler is required. This increases initial cost. Also space required is more.
3. Continuous flow of cooling water is required.
4. Complicated in construction.

3.14 Intercoolers:

An intercooler is a simple heat exchanger. It exchanges the heat of compressed air from the LP compressor to the circulating water before the air enters the HP compressor. It consists of a number of special metal tubes connected to corrosion resistant plates at both ends. The entire nest of tubes is covered by an outer shell
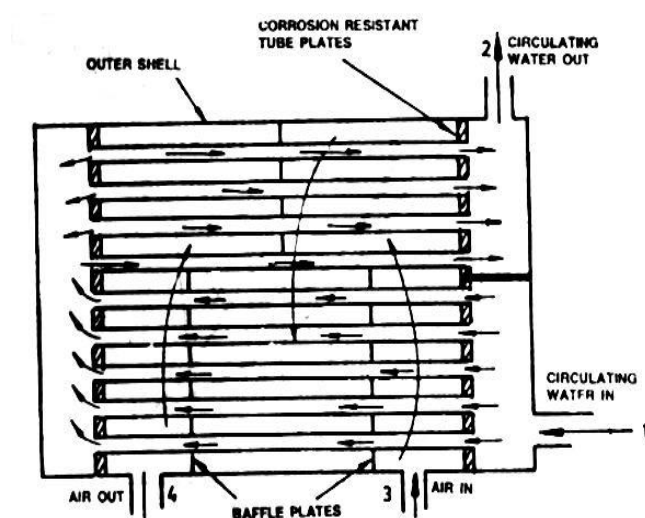
Working: Cold water enters the bottom of the intercooler through water inlet (1) and flows into the bottom tubes. Then they pass through the top tubes and leaves through the water outlet (2) atthe top. Air from LP compressor enters through the air inlet (3) of the intercooler and passes over the tubes. While passing over the tubes, the air is cooled (by the cold water circulated throughthe tubes). This cold air leaves the intercooler through the air outlet (4). Baffle plates are provided in the intercooler to change the direction of air. This provides a better heat transfer from air to the circulating water.

Fig:3.9 Intercooler

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3.15 Work input required in multistage compressor:

The following assumptions are made for calculating the work input in multistage compression.
1. Pressure during suction and delivery remains constant in each stage.
2. Intercooling takes place at constant pressure in each stage.
3. The compression process is same for each stage.
4. The mass of air handled by LP cylinder and HP cylinder is same.
5. There is no clearance volume in each cylinder.
6 There is no pressure drop between the two stages, i.e., exhaust pressure of one stage is equal to the suction pressure of the next stage.
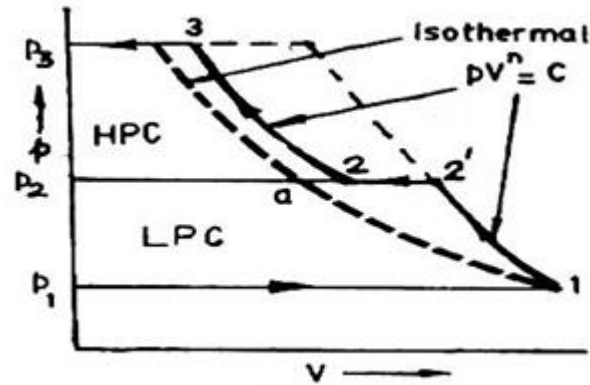
Fig:3.10 Two Stage compression

Work required to drive the multi-stage compressor can be calculated from the area of the p - V diagram .
Let, p_1, V_1 and T_1 be the condition of air entering the LP cylinder.
P_2, V_2 and T_2 be the condition of air entering the HP cylinder.
p_3 be the final delivery pressure of air.

Then,
Total work input = Work input for LP compressor + Work input for HP compressor.

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1 \right] \text{ kJ/cycle}$$

$$W = \frac{n}{n-1} m R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} m R T_2 \left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1 \right] \text{kJ/cycle}$$

If intercooling is perfect, $T_2 = T_1$, therefore,

$$W = \frac{n}{n-1} m R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] + \frac{n}{n-1} m R T_1 \left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1 \right] \text{kJ/cycle}$$

$$W = \frac{n}{n-1} m R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2 \right] \text{kJ/cycle}$$

Or

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2 \right] \text{kJ/cycle}$$

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3.16 Condition for maximum efficiency (or) Condition for minimum work input (or) To prove that for minimum work input the intermediate pressure of a two-stage compressor with perfect intercooling is the geometric mean of the intake pressure and delivery pressure (or) To prove $p_2 = \sqrt{p_1 p_3}$

Work input for a two-stage air compressor with perfect intercooling is given by,

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 2 \right] \text{ kJ/cycle}$$

If the initial pressure (p_1) and final pressure (p_3) are fixed, the value of intermediate pressure (p_2) can be determined by differentiating the above equation of work input in terms of p_2 and equating it to zero.

Let, $\frac{n}{n-1} p_1 V_1 = k \ (constant)$ and $\frac{n-1}{n} = a$

then,

$$W = k \left[\left(\frac{p_2}{p_1} \right)^a + \left(\frac{p_3}{p_2} \right)^a - 2 \right]$$

or

$$W = k(p_2^a p_1^{-a} + p_3^a p_2^{-a} - 2) \quad \text{-------------------- (1)}$$

Differentiating the above equation (1) with respect to p_2 and equating it to zero,

$$\frac{dW}{dp_2} = k \ a \ p_2^{a-1} p_1^{-a} + k \ (-a) p_3^a p_2^{-a-1} = 0$$

$$k \ a \ \frac{p_2^a}{p_2 p_1^a} - k \ a \ p_3^a \frac{1}{p_2^a \ p_2} = 0$$

or

$$\frac{k \ a \ p_2^a}{p_2 \ p_1^a} = \frac{k \ a \ p_3^a}{p_2 \ p_2^a}$$

$$\left(\frac{p_2}{p_1} \right)^a = \left(\frac{p_3}{p_2} \right)^a$$

or

$$\frac{p_2}{p_1} = \frac{p_3}{p_2}$$

$\Rightarrow \quad p_2^2 = p_1 p_3$

or

Intermediate pressure, $\mathbf{p_2 = \sqrt{p_1 p_3}}$

Thus for maximum efficiency the intermediate pressure is the geometric mean of the initial and final pressures.

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3.17 Minimum work input for multistage compression with perfect intercooling:

Work input for a two-stage compressor with perfect intercooling is given by

$$W = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2 \right] \text{----------------- (1)}$$

Work input will be minimum if $\dfrac{p_2}{p_1} = \dfrac{p_3}{p_2}$ ----------------------------(2)

$$p_2^2 = p_1\, p_3$$

Dividing both sides by p_1^2 ,

$$\left(\frac{p_2}{p_1}\right)^2 = \frac{p_3}{p_1}\frac{p_2}{p_1} \Longrightarrow \left(\frac{p_3}{p_1}\right)^{1/2} \text{----------------------------- (3)}$$

From (2), $\dfrac{p_3}{p_2} = \dfrac{p_2}{p_1} = \left(\dfrac{p_3}{p_1}\right)^{1/2}$ ------------------------------------- (4)

Substituting the equation (4) in equation (1), work input for a two stage compressor,

$$W_{min} = \frac{n}{n-1} p_1 V_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{1}{2}\left[\frac{n-1}{n}\right]} + \left(\frac{p_3}{p_1}\right)^{\frac{1}{2}\left[\frac{n-1}{n}\right]} - 2 \right]$$

$$= \frac{n}{n-1} p_1 V_1 \left[2 \left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 2 \right]$$

$$W_{min} = \frac{2n}{n-1} p_1 V_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1 \right]$$

or

$$W_{min} = \frac{2n}{n-1} mRT_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1 \right]$$

For a three stage compressor,

$$W_{min} = \frac{3n}{n-1} p_1 V_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{3n}} - 1 \right]$$

or

$$W_{min} = \frac{3n}{n-1} mRT_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{3n}} - 1 \right]$$

Generally, the minimum work input for a multistage reciprocating air compressor with x number of stages is given by,

$$W_{min} = \frac{xn}{n-1} p_1 V_1 \left[\left(\frac{p_{x+1}}{p_1}\right)^{\frac{n-1}{xn}} - 1 \right]$$

Minimum work input required for a two stage reciprocating air compressor with perfect intercooling is given by,

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$$W_{min} = \frac{2n}{n-1} p_1 V_1 \left[\left(\frac{p_3}{p_1} \right)^{\frac{n-1}{2n}} - 1 \right] \, kJ$$

But, from equation (4), $\left(\frac{p_3}{p_1} \right)^{1/2} = \frac{p_2}{p_1}$

Therefore,

$$W_{min} = \frac{2n}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \, kJ$$

So, for maximum efficiency ie., for minimum work input, the work required for each stage is same.

For maximum efficiency, the following conditions must be satisfied:

1. The air is cooled to the initial temperature between the stages (Perfect cooling between stages).

2. In each stage, the pressure ratio is same. $\left(\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \cdots \right)$

3. The work input for each stage is same.

Rotary compressors:

Rotary compressors have a rotor to develop pressure. They are classified as
(1) Positive displacement compressors and (2) Non positive displacement (Dynamic) compressors

In positive displacement compressors, the air is trapped in between two sets of engaging surfaces. The pressure rise is obtained by the back flow of air (as in the case of Roots blower) or both by squeezing action and back flow of air (as in the case of vane blower). Example: (1) Roots blower, (2) Vane blower, (3) Screw compressor.

In dynamic compressors, there is a continuous steady flow of air. The air is not positively contained within certain boundaries. Energy is transferred from the rotor of the compressor to the air. The pressure rise is primarily due to dynamic effects.

Example: (1) Centrifugal compressor, (2) Axial flow compressor.

Roots blower:

The Roots blower is a development of the gear pump.
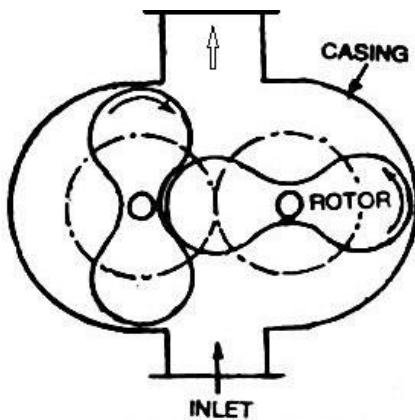
Fig:4.11 Roots blower

Construction: It consists of two lobed rotors placed

in separate parallel axis of a casing as shown in fig:4.11. The two rotors are driven by a pair of gears (which are driven by the prime mover) and they revolve in opposite directions. The lobes of the rotor are of cycloid shape to ensure correct mating. A small clearance of 0.1 mm to 0.2 mm is provided between the lobe and casing. This reduces the wear of moving parts.

Working: When the rotor is driven by the gear, air is trapped between the lobes and the casing. the trapped air moves along the casing and discharged into the receiver. There is no increase in pressure since the flow area from entry to exit remains constant. But, when the outlet is opened, there is a

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back flow of high pressure air in the receiver. This creates the rise in pressure of the air delivered. These types of blowers are used in automobiles for supercharging.

Vane blower:

Construction: A vane blower consists of (1) a rotor, (2) vanes mounted on the rotor, (3) inlet and outlet ports and (4) casing. The rotor is placed eccentrically in the outer casing. Concentric vanes (usually 6 to 8 nos.) are mounted on the rotor. The vanes are made of fiber or carbon. Inlet suction area is greater than outlet delivery area.
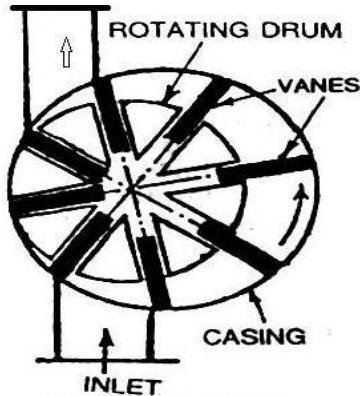
Fig: 4.12 Vane blower

Working: When the rotor is rotated by the prime mover, air is entrapped between two consecutive vanes. This air is gradually compressed due to decreasing volume between the rotor and the outer casing. This air is delivered to the receiver. This partly compressed air is further increased in pressure due to the back flow of high pressure air from the receiver.

Advantages: 1. Very simple and compact, 2. High efficiency 3. Higher speeds are possible

Centrifugal compressor

Construction: It consists of an impeller, a casing and a diffuser. The impeller consists of a number of blades or vanes, is mounted on the compressor shaft inside the casing. The impeller is surrounded by the casing.
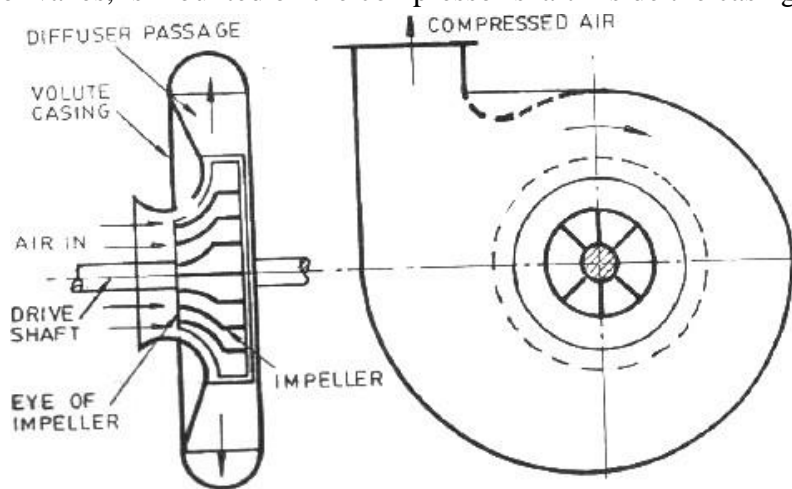
Fig: 4.13 Centrifugal compressor   **Fig: 4.14 Pressure – velocity Plot**
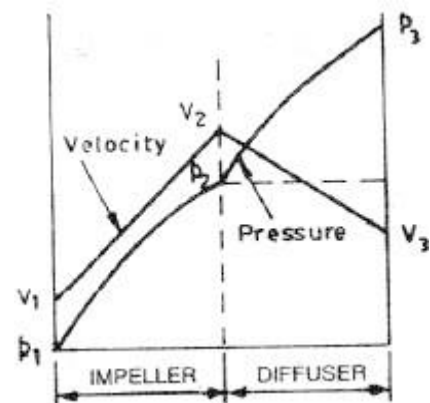
Working: In this compressor air enters axially and leaves radially. When the impeller rotates, air enters axially through the eye of the impeller with a low velocity. This air moves over the impeller vanes. Then, it flows radially outwards from the impeller. The velocity and pressure increases in the impeller. The air then enters the diverging passage known as diffuser. In the diffuser, kinetic energy is converted into pressure energy and the pressure of the air further increases. It is shown in fig:4.14. Finally, high pressure air is delivered to the receiver. Generally half of the total pressure rise takes place in the impeller and the other half in the diffuser.

Applications: Centrifugal compressors are used for low pressure units such as for refrigeration, supercharging of internal combustion engines, etc.

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Axial flow compressor

In this air compressor, air enters and leaves axially.

Construction: It consists of two sets of blades: Rotor blades and stator blades. The blades are so arranged that the unit consists of adjacent rows of rotor blades and stator blades as shown in fig:4.15. The stator blades are fixed to the casing. The rotor blades are fixed on the rotating drum. The drum is rotated by a prime mover through a driving shaft. Single stage compressor consists of a row of rotor blades followed by a row of stator blades. Compression of air takes place in each pair of blades (one rotor blade and one stator blade). Hence there are many stages of compression in this type of compressor.

Working: When the switch is switched on, the prime mover rotates the drum. Air enters through the compressor inlet and passes through the rotor and stator blades. While passing through the blades, the air is compressed between the blades. The air is also compressed between the casing and the blades. The air flow passage area is gradually reduced from the inlet to the outlet of the compressor. This increases the pressure of the air considerably. Finally, high pressure air is delivered to the receiver.
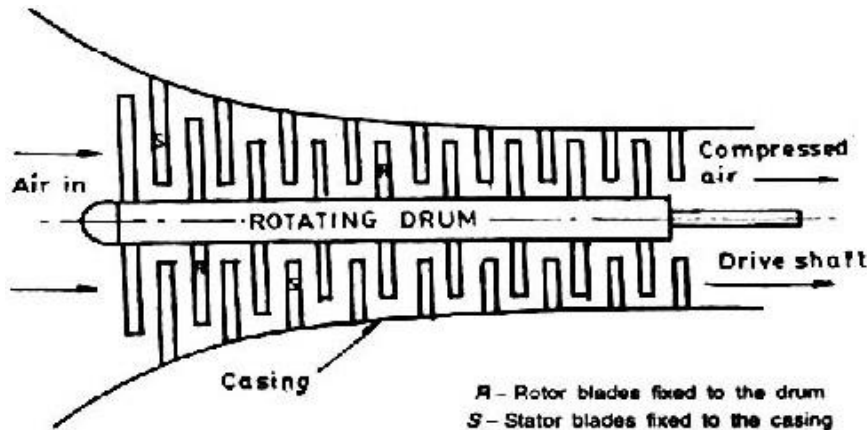
Fig:4.15 Axial flow compressor

Applications:
1. They are widely used in high pressure units such as industrial and marine gas turbine plants,
2. They are most suitable for aircraft work (Jet propulsion) since they require less frontal area.

Comparison of Reciprocating and Rotary compressors

Reciprocating compressors	Rotary compressors
1. It is suitable for low rates of flow. Flow rate is limited to m³/s	It is suitable for large rates of flow. Flow rate can be as large as 50 m³/s.
2. It is used for high pressure rise. It can compress fluids up to 1000 bar.	It is used for medium pressure rise. The pressure rise is limited to 10 bar.
3. It cannot be coupled to turbines or I.C. engines.	It can be directly coupled to turbines or high speed internal combustion engines due to their higher speeds.
4. The flow of air is intermittent.	It gives uniform delivery of air.
5. The criterion of thermodynamic efficiency is isentropic.	The criterion of thermodynamic efficiency is isothermal.
6. Due to sliding parts it requires more lubrication.	No sliding parts. Hence needs lesser lubrication. It gives clean supply of air.
7. Maintenance cost is high because of large number of reciprocating parts.	Maintenance cost is less.
8. Complicated construction. It has more number of parts.	Simple in construction. It has less number of parts.
9. Torque is not uniform.	Uniform torque.

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Free Air Delivery(FAD): It is the volume of air drawn into a compressor from the atmosphere. After compression and cooling the air is returned to the original temperature but it is at a higher pressure. Suppose atmospheric conditions are p, T_a and V_a (the FAD) and the compressed conditions are p, V and T.

Applying the gas law we have

$$\frac{pV}{T} = \frac{p_a V_a}{T_a}$$

$$V_a = \frac{pVT_a}{Tp_a} = F.A.D.$$

1. A Single cylinder, single acting air compressor has cylinder diameter 160mm and stroke length 300mm. It draws air into its cylinder at pressure of 100kpa at 27°C. The air is then compressed to a pressure of 650kpa. If the compressor runs at a speed of 2 rev/sec, Determine.
i)   **Mass of air compressed per cycle**
ii)   **Work required per cycle**
iii)   **Power required to derive the compressor in kW**
Assume the compression process follows PV = constant.

Given data:
        D = 160mm = 0.16m
        L = 300mm = 0.3m
        P_1= 100kpa
        T_1= 27°C= 27+ 273= 300K
        P_2= 650kpa
        N= 2rev/sec = 120rpm
        PV^γ= C γ= 1.4

Solution:
Work done during Isothermal Compression (PV = C)
        W = mRT_1ln [P_2/ P_1]
        W = P_1V_1 ln [P_2/ P_1]   [PV = mRT]
We know that,
        Vs=(π/4)D^2L =(π/4) *$(0.16)^2$*0.3
        Vs=6.03X$10^{-3}$$m^3$ = V_1 [clearance volume is neglected]
        Vs=6.03X$10^{-3}$$m^3$
Substituting V_1 in work done equation
        W=100 X 6.03 X10^{-3} X ln [650/100]
        W= 1.13kJ
Power = [W*N/60] = 1.13*120/60

P = 2026kW
We know that,
        P_1V_1= mRT_1
        m = P_1V_1/R T_1 = [(100*6.03x10^{-3})/(0.287*300)]
        m = 0.007kg
Result:
i.   m = 0.007kg
ii.   W=1.13kJ
iii.   P = 2.26 kW

2. A Single cylinder, single acting reciprocating air compressor with a bore of 12cm and stroke of 16cm runs at 410rpm. At the beginning of compression, the pressure and temperature in the cylinder are 0.98bar and 40°C. the delivery pressure is 6bar. The index of compression is 1.32. the clearance is 6% of stroke volume. Determine the volume of air delivered referred to 1bar and 20°C. what is the power required?

Given data:
D = 12cm = 0.12m
L= 16cm = 0.16m
N = 410rpm
P_1= 0.98 bar= 98kpa
T_1= 40°C = 313K
P_2= 6bar = 600kpa
N = 1.32
Vc=6%=0.06Vs
Po=1bar=100kpa
To=20°C=293K

Solution:
We know that,
$\quad$ Vs=$(\pi/4)D^2L$ =$(\pi/4)$ *$(12)^2$*16
$\quad$ Vs= $0.0018m^3$

$\quad$ We know that,
$\quad$ V_1=Vc+Vs
$\quad$ V_1= 0.06Vs+Vs
$\quad$ V_1=1.06x0.0018
$\quad$ V_1=1.908x10^{-3} m^3

Work done on the single stage compressor with clearance volume
$\quad$ W= [n/n-1] P_1 V_1 [$(P_2/P_1)^{(n-1/n)}$-1]
We know that,
$\quad$ $P_3V_3^n$= $P_4V_4^n$
$\quad$ $[V_4/V_3]^n$=[P_3/P_4]
$\quad$ $[V_4/V_3]^n$=[P_2/P_1]
$\quad$ $[V_4/V_c]^n$=[P_2/P_1]
$\quad$ $[V_4/V_c]$=[P_2/P_1]$^{1/n}$
$\quad$ V_4 = V_c x[P_2/P_1]$^{1/n}$
$\quad\quad$ =0.06xVs$[600/98]^{1/1.32}$
$\quad\quad$ =0.06x0.0018x$[600/98]^{1/1.32}$
$\quad$ V_4=4.26x$10^{-4}m^3$
We know that,
$\quad$ Va =V_1- V_4 = 1.908x10^{-3}- 4.26x10^{-4}
$\quad$ Va = 0.00148 m^3
Substituting Va value in work done equation
$\quad$ W=[1.32/1.32-1]x98x0.00148[$(600/98)^{1.32-1/1.32}$-1]
$\quad$ W = 0.329 kJ
$\quad$ Power = WxN/60 = (0.329x410)/60
$\quad$ P = 2.25 kW
We know that,
$\quad$ PoVo/To = P_2V_d/T_2
$\quad$ Vo = To/Po x P_2V_d/T_2
We know that,
T_2/T_1=[P_2/P_1]$^{n-1/n}$
T_2=T_1x[P_2/P_1]$^{n-1/n}$

$T_2 = 313 \times [600/98]^{1.32-1/1.32}$

$T_2 = 485.6K$

$[V_2/V_1]^n = P_1/P_2$

$V_2/V_1 = [P_1/P_2]^{1/n}$

$V_2 = V_1 [P_1/P_2]^{1/n}$

$V_2 = 1.908 \times 10^{-3} [98/600]^{1/1.32}$

$V_2 = 0.00048 \ m^3$

We know that,

$V_d = V_2 - V_3 = V_2 - V_c = 0.00048 - (0.06 \times 0.0018)$

$V_{d=} 0.000372 m^3$

Sub, T_o, P_o, P_2, T_2, V_d values in ..(1)

$V_o = (293/100) \times (600/485.6) \times 0.000372$

$V_o = 0.0013 \ m^3$

Result:

P = 2.25 kW

V_o = 0.0013 m^3

3. A single stage reciprocating compressor receives air at 25m³/min at 1 bar, 15°C and discharges it at 15 bar. Assume the value of n for compression as 1.35 and volumetric efficiency as 0.75. determine i) theoretical power required ii) piston displacement per min ii) maximum air temperature. [Dec 2003]

Given data:

V_a = 24m3/min

P_2 = 15 bar = 1500kpa

P_1 = 1 bar = 100kpa

N = 1.35

T_1 = 15°C

η_{vol} = 0.75

Solution :

work done on the single stage compressor with clearance volume,

$W = n/n-1 P_1 V_a [(P_2/P_1)^{n-1/n} -1]$

W = 9816.04 KJ/min = 163.6Kj/s

P = 163.6 KW

We know that ,

$\eta_{vol} = V_a/ V_s$

0.75 = 25/ V_s

V_s = 33.33 m3/min

We know that,

$T_2/T_1 = [P_2/P_1]^{n-1/n}$

$T_2 = T_1 \times [P_2/P_1]^{n-1/n}$

$T_2 = 288 \times [1500/100]^{1.35-1/1.35}$

$T_2 = 581.17K$

Result:

P = 163.6 KW

V_s = 33.33 m3/min

T_2 = 581.17K

4. A single stage reciprocating air compressor takes 1 m³ of air per minute at 1bar and 15°C and delivers it at 7bar. The law of compression is $PV^{1.3}$== constant. Calculate the indicated power neglect clearance. If the speed of compressor is 300rpm and stroke to bore ratio is 1.5, calculate the cylinder dimensions. Find the power required if the mechanical efficiency of compressor is 85% and motor transmission efficiency is 90%

Given data:
V_1=1 m³/min
P_1= 1bar=100kpa

T_1=15°C=288K
P_2=7bar = 700kpa
N=300rpm
L/D=1.5
η_{mech}=85%
motor efficiency= 90%
$PV^{1.3}$= C

Solution:

We know that, Work done during polytropic compression
W = (n/n-1)P_1 V_a [(P_2/P_1) $^{n-1/n}$ -1]
W = (1.3/1.3-1)x100x 1x[(700/100) $^{1.3-1/1.3}$-1]
= 244.6kJ/min

Indicated Power = 4.07kW
We know that,
Stroke volume, Vs= V_1= $(\pi/4)D^2L$
1/300=$(\pi/4)xD^2x1.5D$
1/300=$(\pi/4)x1.5D^3$
D = 0.141m
L= 1.5x0.141
L = 0.212m
We know that,
η_{mech}= (Indicated power/ Power input)
Power input = 4.07/0.85
Power input = 4.79kW
Motor efficiency = power input/ motor power
Motor power = 4.79/ 0.90
Motor power = 5.32kW

Result:
Indicated power = 4.07kWPower input = 4.79kW Motor power = 5.32kW

5. The free air delivered of a single cylinder single stage reciprocating air compressor 2.5 m³/min. The ambient air is at STP conditions and delivery pressure is 7bar. The clearance volume is 5% of the stroke volume and law of compression and expansion is PV$^{1.25}$=C. if L= 1.2D and the compressor runs at 150rpm, determine the size of the cylinders.

Given data:
Va = 2.5 m³/min= 0.04166 m³/sec
For STP condition, the pressure and temperature are
V_1=1 m³/min
P_1= 1.013bar=101.3kpa
T_1=15°C=288K
P_2=7bar = 700kpa
N=150rpm
L=1.2D
Vc=5%Vs = 0.05Vs
PV$^{1.25}$= C
n = 1.25

Solution:
The mass of free air delivered per second is given by
m_a=PV/RT =(1.013x10⁵x0.04166)/(287x288)=0.051kg/sec
We know that,

Work done, W = (n/n-1)P V_a [(P_2/P_1) $^{n-1/n}$ -1]
W = m_aRT(n/n-1) [(P_2/P_1) $^{n-1/n}$ -1]
W = 0.054x0.287x288x (1.25/1.25-1) [(700/101.3) $^{1.25-1/1.25}$ -1]
W = 9.95kW
We know that,
Indicated power, IP =PmLAN/1000
Pm = (n/n-1) P_1xη $_{vol}$[(P_2/P_1) $^{n-1/n}$ -1]
But, η $_{vol}$=1+C-C (P_2/P_1) $^{1/n}$
Where   C = Vc/Vs
η $_{vol}$=1+ (Vc/Vs) – (Vc/Vs) (P_2/P_1) $^{1/n}$
η $_{vol}$=1+ (0.05) – (0.05) (700/101.3) $^{1/1.25}$
η $_{vol}$= 0.815
Substituting Pm value in eqn (2)
Pm = (1.25/1.25-1) x1 x 0.815 x [(700/101.3) $^{1.25-1/1.25}$ -1]
Pm =1.923bar
Substituting Pm value in eqn (1)
Indicated Power IP (or) work out put

1.95   = [1.923x10⁵ x 1.2 D x (π/4)D² x 150/60] / 1000

D = 0.28m
Result:
L = 1.2 D = 1.2x 0.28 = 0.336m
 D = 0.28 m
L = 0.336m

6. A single stage double acting compressor has a free air delivery (FAD) of 14m³/min measured at 1.013bar and 15°C. the pressure and temperature in the cylinder during induction are 0.95bar and 32°C respectively. The delivery pressure is 7bar and index of compression and expansion, n=1.3. the clearance volume is 5% of the swept volume. Calculate the indicated power required and the volumetric efficiency.

Given data:
$V_0 = 14 \, m^3/min = 0.233 \, m^3/sec$
$P_1 = 0.95 \, bar = 95 \, kpa$
$P_2 = 7 \, bar = 700 \, kpa$
$T_1 = 32°C = 305K$
$T_0 = 15°C = 288K$
$P_0 = 1.013 \, bar = 101.3 \, kpa$
$V_c = 5\% \, V_s = 0.05 \, V_s \quad V_c/V_s = 0.05$
$n = 1.3$

Solution:
Volumetric efficiency, $\eta_{vol} = 1 + C - C \, (P_2/P_1)^{1/n}$
$\quad \eta_{vol} = 1 + (V_c/V_s) - (V_c/V_s)(P_2/P_1)^{1/n}$
$\quad \eta_{vol} = 1 + (0.05) - (0.05)(700/95)^{1/1.3}$
$\quad \eta_{vol} = 0.818 = 81.8 \, \%$
We know that,
$\quad P_o \, V_o / T_o = P_1 \, V_1 / T_1$

$\quad 101.3 \times 0.233 / 288 = 95 \times V_a / 305$
$\quad V_a = 0.263 \, m^3/sec$

Work done or power,
$\quad P = (n/n-1) P_1 \, V_a \, [\, (P_2/P_1)^{n-1/n} - 1]$
$\quad P = (1.3/1.3-1) \times 95 \times 0.263 \, [\, (700/95)^{1.3-1/1.3} - 1]$
$\quad P = 63.39 \, kW$
Result:
$\quad \eta_{vol} = 81.8 \, \%$
$\quad$ Indicated power $P = 63.39 \, kW$

7. A single cylinder single acting reciprocating compressor takes in 6m³/min of air at 1bar and 15°C and compresses into 6 bar. Calculate the saving in the power required when the compression process in changed from adiabatic compression to isothermal compression

Given data:
$V_1 = 6 \, m^3/min$
$P_1 = 1 \, bar = 100 \, kpa$
$T_1 = 15° \, C = 288K$
$P_2 = 6 \, bar = 600 \, kpa$

Solution:
Work done during isothermal compression (pv=c)
$W = P_1 \, V_1 \ln[P_2/P_1]$
$\quad = 100 \times 6 \times \ln[600/100]$
W = 1075.5 kJ/min

Power , P = 17.91kW
Work done during adiabatic process
$W = [\gamma/\gamma-1] P_1 \, V_1 \, [(P_2/P_1)^{(\gamma-1/\gamma)} - 1]$
$W = [1.4/1.4-1] \times 100 \times 6 \times [(600/100)^{(1.4-1/1.4)} - 1]$

W= 1403.87 kJ/min
P = 23.39kW
Saving power = 23.39-17.91
Saving power = 5.48 kW

8. Air is to be compressed in a single stage reciprocating compressor from 1.013bar and 15°C to 7bar. Calculate the indicated power required for a free air delivery of 0.3 m^3/min, when the compression process is i) Isentropic ii) polytropic with (n=1.45)

Given data:
P_1=1.013bar =101.3kpa
T_1=15°C = 288K
P_2=7bar=700kpa
Vo=0.3m^3/min
n=1.25

solution:
we know that, PoVo/To = P_1V_1/T_1
V_1= [PoVo/To] X [T_1/ P_1] .... (1)
We know that, at atmospheric condition the pressure and temperature are
$\qquad$ Po = 101.3kpa
$\qquad$ To = 298 K
Substituting To,Po,Vo, P_1, V_1 values in eqn (1)
$\qquad$ V_1 =[(101.3x0.3)/298]x[288/101.3]
$\qquad$ **V_1=0.289m^3/min**
Work done duringisentropic Compression
$\qquad$ W= [γ/γ-1] P_1 V_1 [(P_2/ P_1)$^{(γ-1/γ)}$-1]
$\qquad$ W= [1.4/1.4-1] *101.3*0.289* [(700/ 101.3)$^{(1.4-1/1.4)}$-1]
$\qquad$ W= 75.53kJ/min
$\qquad$ W= 1.25kJ/s
$\qquad$ **P_{Iso}=1.25kW**
Work done during polytropic compression
$\qquad$ W= [n/n-1] P_1 V_1 [(P_2/ P_1)$^{(n-1/n)}$-1]
$\qquad$ W= [1.25/1.25-1] x101.3x 0.289x [(700/ 101.3)$^{(1.25-1/1.25)}$-1]
$\qquad$ W = 69.08kJ/min
$\qquad$ **P_{poly}=1.15kW**
Result:
P_{Iso}=1.25kW P_{poly}=1.15kW

9. Air enters a single stage double acting air compressor at 100kpa and 29°C. the compression ratio is 6:1. The speed of compression in 550rpm. The volume rate measured at suction condition is 5 m3/min. find the motor power required if the mechanical efficiency is 90%. If the volumetric efficiency is 80%. Find swept volume of cylinder.

Given data:
P_1=100kpa
T_1=29°C =302K
N=550rpm
V_1=5 m3/min
Compression ratio = 6:1
n = 1.3
η vol=80%
η max=90%

Solution:

Compression ratio =(total cylinder volume)/(clearance volume)= V_1/V_c

V_1/V_c =6

$5/V_c$ =6

V_c =0.833 m3/min

We know that,

$V_1= V_c+ V_s$

$5= 0.833+ V_s$

V_s=4.167 m3/kg

Work done on the single stage compressor with clearance volume

$W = (n/n-1)P_1 V_a [(P_2/P_1)^{n-1/n} -1]$ ……….(1)

Volumetric efficiency, $\eta_{vol}=1+C-C(P_2/P_1)^{1/n}$

$C= V_c/ V_s$

$\eta_{vol}=1+ (V_c/ V_s)- (V_c/ V_s) (P_2/P_1)^{1/n}$

$\eta_{vol}=1+ (0.833/ 4.167)- (0.833/ 4.167) (P_2/100)^{1/1.3}$

P_2= 247.03kpa

We know that,

$\eta_{vol}= V_a/ V_s$

$0.8 = V_a/ 4.167$

V_a=3.33 m^3/min

Applying V_a, P_2 values in eqn (1)

$W = [1.3/1.3-1]x100x 3.3 [(247.03/100)^{1.3-1/1.3} -1]$

$W = 334.87kJ/min$

W = 5.58kW

We know that,

Mech. efficiency =(power output of compressor)/(power supplied to compressor)

$0.9 = (5.58)/$ (power supplied to compressor)

Power Supplied To Compressor =6.2kW

Result:

V_s=4.167 m3/kg

Power Supplied To Compressor =6.2kW

10. A single stage single acting compressor delivers 15m^3 of free air per minute from 1bar to 8 bar. The speed of compressor is 300rpm. Assuming that compression and expansion follow the law PV$^{1.3}$ = constant and clearance is 1/16 th of swept volume, find the diameter and stroke of the compressor. Take L/D=1.5, the temperature and pressure of air at the suction are same as atmospheric air [Nov 2004]

Given data:

V_0=15 m^3/min

P_1= 1bar=100kpa

P_2=8bar = 800kpa

N=300rpm

L=1.5D

PV$^{1.3}$= C

n = 1.3

L/D = 1.5

Solution:

We know that the volumetric efficiency

$\eta_{vol}=1 – (Vc/Vs)[(P_2/P_1)^{1/n}-1]$

$\eta_{vol} = 1 - (1/16)[(8/1)^{1/1.3} - 1]$

$\eta_{vol} = 0.753 = 75.3\%$

We know that, free air delivered

$V_a = V_s \times \eta_{vol} \times 300$

$15 = V_s \times 0.753 \times 300$

$V_s = 0.0664 \ m^3$

Stroke volume = $0.0664 \ m^3$

We know that,

$V_s = (\pi/4)D^2 L = 0.0664$

$(\pi/4)D^2 \times 1.5D = 0.0664$

D = 0.3834 m

We know that,

L/D = 1.5

L = 1.5 × 0.3834

L = 0.5751m

Unit -5 - HEAT TRANSFER & REGRIGERATION – SMEA1405

Introduction

Heat transfer is the science that seeks to predict the energy transfer that may take place between material bodies as a result of a temperature difference. Thermodynamics deals with systems in equilibrium; it may be used to predict the amount of energy required to change a system from one equilibrium state to another; it may not be used to predict how fast a change will take place since the system is not in equilibrium during the process. Heat transfer supplements the first and second principles of thermodynamics by providing additional experimental rules which may be used to establish energy-transfer rates.

As an example of the different kinds of problems that are treated by thermodynamics and heat transfer, consider the cooling of a hot steel bar that is placed in a pail of water. Thermodynamics may be used to predict the final equilibrium temperature of the steel bar-water combination. Thermodynamics will not tell us how long it takes to reach this equilibrium condition or what the temperature of the bar will be after a certain length of time before the equilibrium condition is attained. Heat transfer may be used to predict the temperature of both the bar and the water as a function of time.

Basic Mechanisms of Heat Transfer

Heat transfer may occur by anyone or more of the three basic mechanisms of heat transfer: conduction, convection, and radiation.

1. Conduction Heat transfer. In conduction, heat can be conducted through solids, liquids, andgases. The heat is conducted by the *transfer of the energy of motion between adjacent molecules*. In a gas the "hotter" molecules, which have greater energy and motions, impart energy to the adjacent molecules at lower energy levels. This type of transfer is present to some extent in all solids, gases, or liquids in which a temperature gradient exists. In conduction, energy can also be transferred by "free" electrons, which is quite important in metallic solids. **Examples** of heat transfer mainly by conduction are heat transfer through walls of exchangers or a refrigerator, heat treatment of steel forgings, freezing of the ground during the winter, and so on.

Fourier's Law of Heat Conduction

The basic rate transfer process equation for processes such as momentum transfer, heat transfer, mass transfer and electric current is as follows:

This equation states what we know intuitively: that in order to transfer a property

$$\text{rate of a transfer process} = \frac{\text{driving force}}{\text{resis}\tan ce} \quad \text{----------} \quad (1)$$

such as heat or mass, we need a driving force to overcome a resistance.

The transfer of heat by conduction also follows this basic equation and is written as Fourier's law for heat conduction in fluids or solids:

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

---------------- (2)

where q_x is the heat-transfer rate in the x direction in watts (W), A is the cross-sectional area normal to the direction of flow of heat in m^2, T is temperature in K, x is distance in m, and k is the thermal conductivity in W/m·K in the SI system. The quantity q_x / A is called the heat flux in W/m^2. The quantity dT / dx is the temperature gradient in the x direction. The minus sign in eq. (2) is required because if the heat flow is positive in a given direction, the temperature decreases in this direction.

Fourier's law, eq.(2), can be integrated for the case of steady-state heat transfer through a flat wall of constant cross-sectional area A, where the inside temperature is T1 at point 1 and T2 at point 2, a distance of $x_2 x_1$ m away. Rearranging eq. (2),

$$\frac{q_x}{A}\int_{x_1}^{x_2} dx = -k\int_{T_1}^{T_2} dT$$

-------(3)

Integrating, assuming that k is constant and does not vary with temperature and dropping the subscript x on q_x for convenience,

$$\frac{q}{A} = \frac{k}{x_2 - x_1}(T_1 - T_2)$$

------- (4)

2. Convection Heat Transfer. The transfer of heat by convection implies the transfer of heat bybulk transport and mixing of macroscopic elements of warmer portions with cooler portions of a gas or liquid. It also often refers to the energy exchange between a solid surface and a fluid. A distinction must be made between forced-convection heat transfer, where a fluid is forced to flow past a solid surface by a pump, fan, or other mechanical means, and natural or free convection, where warmer or cooler fluid next to the solid surface causes a circulation because of a density difference resulting from the temperature differences in the fluid. Examples of heat transfer by convection are loss of heat from a car radiator where the air is being circulated by a fan, cooking of foods in a vessel being stirred, cooling of a hot cup of coffee by blowing over the surface, and so on.

Convective Heat-Transfer Coefficient

It is well known that a hot piece of material will cool faster when air is blown or forced past the object. When the fluid outside the solid surface is in forced or natural convective motion, we express the rate of heat transfer from the solid to the fluid or vice versa, by the following equation:

$$q = hA\,(T_w - T_f) \qquad \text{------------ (5)}$$

where q is the heat-transfer rate in W, A is the area in m^2, T_w is the temperature of the solid surface in K, T_f is the average or bulk temperature of the fluid flowing past in K, and h is the convective heat-transfer coefficient in $W/m^2.K$.

The coefficient h is a function of the system geometry, fluid properties, flow velocity, and temperature difference. In many cases, empirical correlations are available to predict this coefficient, since it often cannot be predicted theoretically. Since we know that when a fluid flows past a surface there is a thin, almost stationary layer or film of fluid adjacent to the wall which presents most of the resistance to heat transfer, we often call the coefficient h a film coefficient.

3. Radiation Heat Transfer:- Radiation differs from heat transfer by conduction and convectionin that **no physical medium is needed** for its propagation. Radiation is the transfer of energy through space by means of **electromagnetic waves** in much the same way as electromagnetic light waves transfer light. The same laws that govern the transfer of light govern the radiant transfer of heat. Solids and liquids tend to absorb the radiation being transferred through them, so that radiation is important primarily in transfer through space or gases. The most important example of radiation is the transport of heat to the earth from the sun. Other examples are cooking of food when passed below red-hot electric heaters, heating of fluids in coils of tubing inside a combustion furnace, and so on.

The rate of energy emitted by a black body is proportional to the fourth power of the absolute temperature of the body and directly proportional to its surface area. Thus

$$q_{emitted} = \sigma A T^4$$

Where σ is the proportionality constant and is called the Stefan-Boltzmann constant with the value of 5.669×10^{-8} $W/m^2.K^4$. The equation is called the Stefan-Boltzmann law of thermal radiation, and it applies only to blackbodies.

CONDUCTION HEAT TRANSFER

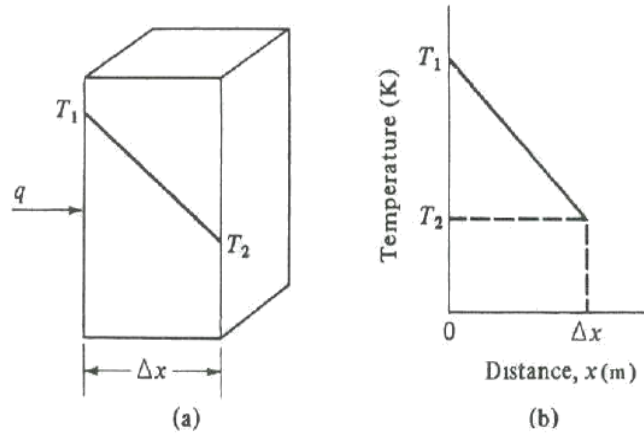**One-dimensional steady-state conduction of heat through some simple geometries**

Fig. 3.1 Heat conduction in a flat wall: (a) geometry of wall, (b) temperature plot.

Conduction Through a Flat Slab or Wall

Consider a flat slab or wall (Fig. 3.1) where the cross-sectional area A and k in are constant,

The eq. $\dfrac{q}{A} = \dfrac{k}{x_2 - x_1}(T_1 - T_2)$ can be rewrite as

$$\frac{q}{A} = \frac{k}{\Delta x}(T_1 - T_2)$$

where $\Delta x = x_2 - x_1$.

The above indicates that if T is substituted for T_2 and x for x_2, the temperature varies linearly with distance, as shown in Fig.3.1(b).

If the thermal conductivity is not constant but varies linearly with temperature, then substituting $k = a = bT$ into the above equation and integrating,

$$\frac{q}{A} = \frac{a + b\dfrac{T_1 + T_2}{2}}{\Delta x}(T_1 - T_2) = \frac{k_m}{\Delta x}(T_1 - T_2)$$

where

$$k_m = a + b\frac{T_1 + T_2}{2}$$

This means that the mean value of k (i.e., k_m) to use in

$\dfrac{q}{A} = \dfrac{k}{\Delta x}(T_1 - T_2)$ is the value of k evaluated at the linear average of T_1 and T_2.

The rate of a transfer process equals the driving force over the resistance and the equation $\dfrac{q}{A} = \dfrac{k}{\Delta x}(T_1 - T_2)$ can be rewritten in that form as:

$$q = \frac{T_1 - T_2}{\Delta x / kA} = \frac{T_1 - T_2}{R} = \frac{driving \quad force}{resis \tan ce}$$

where $R = \Delta x / kA$ A and is the resistance in K/W.

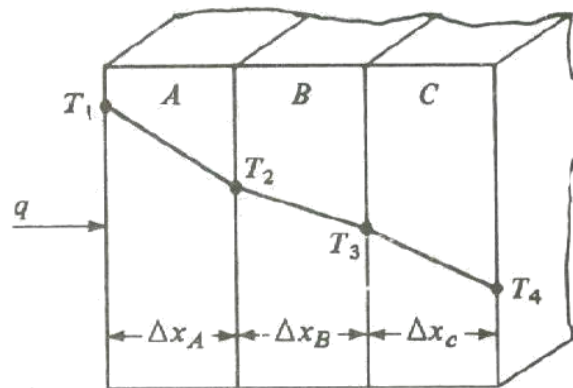# CONDUCTION HEAT TRANSFER THROUGH A COMPOSITE PLANE WALL

Fig. 4.1 Heat flow through multilayer wall

Consider the heat flow through composite wall made of several materials of different thermal conductivities and thicknesses. An example is a wall of a cold storage, constructed of different layers of materials of different insulating properties. All materials are arranged in series in the direction of heat transfer, as shown in the above Figure.

The thickness of the walls are x_1, x_2, and x_3 and the thermal conductivites of the walls are K_1, K_2, and K_3, respectively. The temperatures at the contact surfaces are T_2, T_3, and T_4.

From Fourier's Law,

$$\frac{q}{A} = -K\,A\frac{dT}{dx}$$

This may be written as $\Delta T = T_2 - T_1 = -\dfrac{q \cdot \Delta x}{KA}$

$$T_1 - T_2 = \frac{x_1}{K_1\,A} \cdot q$$

$$T_2 - T_3 = \frac{x_2}{K_2\,A} \cdot q$$

$$T_4 - T_3 = \frac{x_3}{K_3\,A} \cdot q$$

Total temperature difference, $\Delta T = \Delta T_1 + \Delta T_2 + \Delta T_3$

$$T_1 - T_4 = q \cdot \left[\frac{x_1}{K_1 A} + \frac{x_2}{K_2 A} + \frac{x_3}{K_{31} A} \right]$$

where, $T_1 - T_4$ = thermal potential responsible for heat flow. The

$\left[\dfrac{x_1}{K_1 A} + \dfrac{x_2}{K_2 A} + \dfrac{x_3}{K_{31} A} \right]$ is known as the total thermal resistance of the

composite ass. It is similar to the electrical resistance in series.

The thermal circuit for multilayer rectangular system is shown in the following figure.
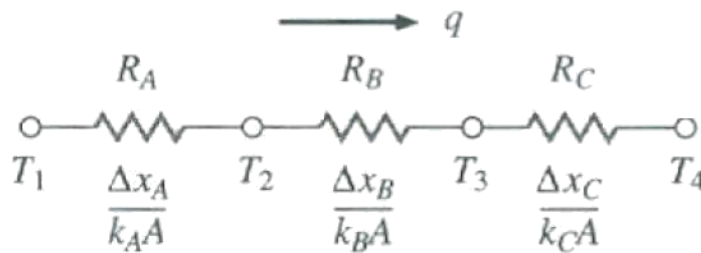
Fig 4.2 Electrical analog of one dimensional heat transfer through composite wall

$$q = \frac{T_1 - T_4}{R_1 + R_2 + R_3}$$

Problem Heat Loss Through an Insulating Wall

Calculate the heat loss per m² of surface area for an insulating wall composed of 25.4 -mm-thick fiber insulating board, where the inside temperature is 352.7 K and the outside temperature is 297.1 K. The thermal conductivity of fiber insulating board is 0.048 W/m.K

Solution:

The thickness $x_2 - x_1 = 0.0254$ m.

Substituting into the eq.

$$\frac{q}{A} = \frac{k}{x_2 - x_1}(T_1 - T_2) = \frac{0.048}{0.0254}(352.7 - 297.1)$$

$$= 105.1 \text{ W/m}^2$$

1. Calculate the rate of heat loss through the vertical walls of a boiler furnace of size 4 m by 3 m by 3 m high. The walls are constructed from an inner fire brick wall 25 cm thick of thermal conductivity 0.4 W/mK, a layer of ceramic blanket insulation of thermal conductivity 0.2 W/mK and 8 cm thick, and a steel protective layer of thermal conductivity 55 W/mK and 2 mm thick. The inside temperature of the fire brick layer was measured at 600° C and the temperature of the outside of the insulation 60⁰ C. Also find the interface temperature of layers.

Given:

Composite Wall

l= 4m   b= 3m   h= 3m

Area of rectangular wall lb = 4x3 = 12m²

$L_1 = 25$ cm ⎤ Fire brick

$k_1 = 0.4$ W/mK ⎦

$L_2 = 0.002$m ⎤ Steel

$k_2 = 54$ W/mK ⎦

$L_3 = 0.08$ m ⎤ insulation

$k_1 = 0.2$ W/mK ⎦

$T_{1=}600^0$ C

$T_2 = 60^0$ C
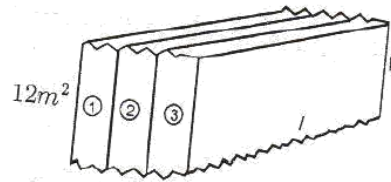
Find

(i) Q   (ii) $(T_3 - T_4)$

Solution
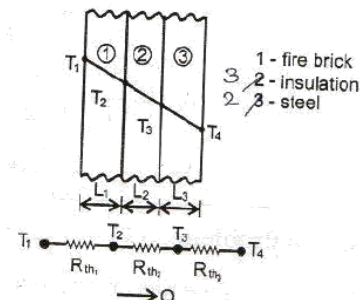
We know that,

$$Q = \frac{(\Delta T)_{overall}}{\Sigma R_{th}}$$

Here

(ΔT) overall $= T_1 - T_4$

And $\Sigma R_{th} = R_{th1} + R_{th2} + R_{th3}$

$$R_{th1} = \frac{L_1}{k_1 A} = \frac{0.25}{0.4x12} = 0.0521 \text{K/W}$$

$$R_{th2} = \frac{L_2}{k_2 A} = \frac{0.08}{0.2x12} = 0.0333 \text{K/W}$$

$$R_{th3} = \frac{L_3}{k_3 A} = \frac{0.002}{54x12} = 0.0000031 \text{K/W}$$

35

$$Q = \frac{T_1 - T_4}{R_{th1} + R_{th2} + R_{th3}}$$

$$= \frac{600 - 60}{0.0521 + 0.0000031 + 0.0333}$$

$$Q = 6320.96 \ W$$

(i)   To find temperature drop across the steel layer (T_2 - T_3)

$$Q = \frac{T_2 - T_3}{R_{th3}}$$

$T_3 - T_4 =$ $Q \times R_{th2}$

         $=$ 6320.96×0.0000031

$T_3 - T_4 =$ $0.0196 \ K$.

2. **A spherical container of negligible thickness holding a hot fluid at 140^0 and having an outer diameter of 0.4 m is insulated with three layers of each 50 mm thick insulation of $k_1 = 0.02$: $k_2 = 0.06$ and $k_3 = 0.16$ W/mK. (Starting from inside). The outside surface temperature is 30^0C. Determine (i) the heat loss, and (ii) Interface temperatures of insulating layers.**

    Given:

    OD $=$ 0.4 m

    r_1 $=$ 0.2 m

    r_2 $=$ r_1 + thickness of 1st insulation

         $=$ 0.2+0.05

    r_2 $=$ 0.25m

    r_3 $=$ r_2 + thickness of 2nd insulation

         $=$ 0.25+0.05

    r_3 $=$ 0.3m

    r_4 $=$ r_3 + thickness of 3rd insulation

         $=$ 0.3+0.05

    r_4 $=$ 0.35m

    T_{hf} $=$ 140° C, $T_{cf} = 30^\circ$ C,

    k_1 $=$ 0.02 W/mK

    k_2 $=$ 0.06 W/mK

    k_3 $=$ 0.16 W/mK.

Find (i) Q (ii) T_2, T_3

Solution

$$Q = \frac{(\Delta T)_{overall}}{\Sigma R_{th}}$$

$\Delta T = T_{hf} - T_{cf}$

$\Sigma R_{th} = R_{th1} + R_{th2} + R_{th3}$

$$R_{th1} = \frac{r_2 - r_1}{4\pi k_1 r_2 r_1} = \frac{(0.25 - 0.20)}{4\pi \, x0.02x0.25x0.2} = 3.978^\circ \, C/W$$

$$R_{th2} = \frac{r_3 - r_2}{4\pi k_2 r_3 r_2} = \frac{(0.30 - 0.25)}{4\pi \, x0.06x0.3x0.25} = 0.8842^\circ \, C/W$$

$$R_{th1} = \frac{r_4 - r_3}{4\pi k_3 r_4 r_3} = \frac{(0.35 - 0.30)}{4\pi \, x0.16x0.35x0.30} = 0.23684^\circ \, C/W$$

$$Q = \frac{140 - 30}{0.0796 + 0.8842 + 0.23684}$$

$$Q = 21.57 \, W$$

To find interface temperature (T$_2$, T$_3$)

$$Q = \frac{T_2 - T_3}{R_{th1}}$$
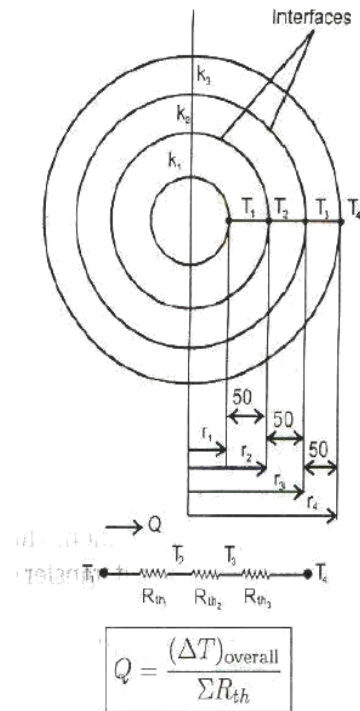
$$T_2 = T_1 - [Q \times R_{th1}]$$

$$= 140 - [91.62 \times 0.0796]$$

$$T_2 = 54.17^0 C$$

$$Q = \frac{T_2 - T_3}{R_{th1}}$$

$$T_3 = T_2 - [Q \times R_{th2}]$$

$$= 132.71 - [91.62 \times 0.8842]$$

$$T_3 = 35.09^\circ \, C$$

15. A furnace walls made up of three layers , one of fire brick, one of insulating brick and one of red brick. The inner and outer surfaces are at 870° C and 40° C respectively. The respective co- efficient of thermal conduciveness of the layer are 1.0, 0.12 and 0.75 W/mK and thicknesses are 22 cm, 7.5, and 11 cm. assuming close bonding of the layer at their interfaces, find the rate of heat loss per sq.meter per hour and the interface temperatures.

Given

Composite wall (without convection)

$L_1 = 22 \times 10^{-2}$ m

$k_1 = 1$ W/mK

$L_2 = 7.5 \times 10^{-2}$ m

$k_2 = 0.12$ W/mK

$L_3 = 11 \times 10^{-2}$ m

$k_3 = 0.75$ W/mK

$T_1 = 870°$ C

$T_4 = 40°$ C

Find

(i)   Q / hr   (ii) T_2, T_3

Solution

We know that,

$$Q = \frac{(\Delta T) overall}{\Sigma Rth}$$

Here

(ΔT) overall = $T_1 - T_4$

= 870 – 40

= 830 °C

And $\Sigma R_{th} = R_{th1} + R_{th2} + R_{th3}$

(assume A = 1 m^2)

$R_{th1} = \frac{L1}{k1A} = \frac{22 \times 10^{-2}}{1 \times 1} = 22 \times 10^{-2}$ K/W

$R_{th2} = \frac{L2}{k2A} = \frac{7.5 \times 10^{-2}}{0.12 \times 1} = 0.625$ K/W

$R_{th3} = \frac{L3}{k3A} = \frac{11 \times 10^{-2}}{0.75 \times 1} = 0.1467$ K/W

$$Q = \frac{T1 - T4}{Rth1 + Rth2 + Rth3}$$

$$= \frac{870 - 40}{0.9917}$$

Q = 836.95   W/m^2

Q = 3.01X 10^5 J/h

REFRIGERATION CYCLES & REFRIGERANTS

INTRODUCTION

For specific applications, efficiencies of both living and non-living beings depend to a great extent on the physical environment. The nature keeps conditions in the physical environment in the dynamic state ranging from one extreme to the other. Temperature, humidity, pressure and air motion are some of the important environment variables that at any location keep changing throughout the year. Adaptation to these many a times unpredictable variations are not possible and thus working efficiently is not feasible either for the living beings or the non-living ones. Thus for any specific purpose, control of the environment is essential. Refrigeration and air-conditioning is the subject which deals with the techniques to control the environments of the living and non-living subjects and thus provide them comforts to enable them to perform better and have longer lives.

DEFINITIONS

Refrigeration:

Refrigeration is defined as a method of reducing the temperature of a system below that of the surroundings and maintaining it at the lower temperature by continuously extracting the heat from it.
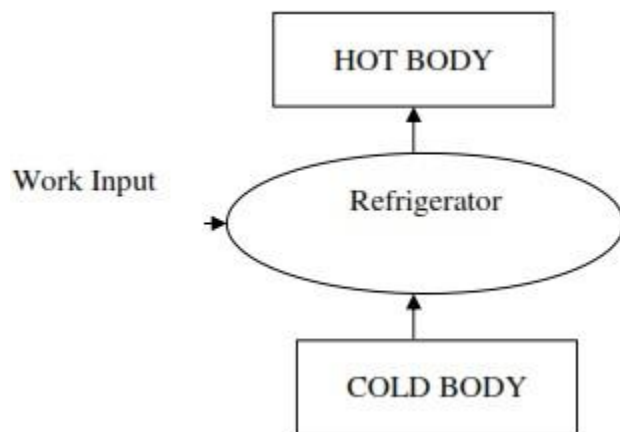

Fig 5.1

The principle of refrigeration is based on second law of thermodynamics. It states that heat does not flow from a low temperature body to a high temperature body without the help of an external work. In refrigeration process, the heat is continuously removed from a system at lower temperature and transfers it to the surroundings at a higher temperature. This operation according to second law of thermodynamics can only be performed by the aid of the external work. Therefore in a refrigeration system, power is to be supplied to remove heat continuously from the

refrigerator to keep it cool at a temperature less than the surroundings. The refrigeration cycle is based on reversible Carnot cycle.

Refrigeration effect:

The rate at which the heat is absorbed in a cycle of from the interior space to be cooled is called refrigeration effect. It is defined as the quantity of heat removed to the time taken. It is also called as the capacity of a refrigerator.

Ton of Refrigeration (or) Unit of Refrigeration (TR):

The standard unit of refrigeration is *ton refrigeration* or simply *ton* denoted by TR. It is equivalent to the rate of heat transfer needed to produce 1 ton (2000 lbs) of ice at 32 ^{0}F from water at 32 ^{0}F in one day, i.e., 24 hours. The enthalpy of solidification of water from and at 32 ^{0}F in British thermal unit is 144 Btu/lb. Thus

$$1 \ TR = \frac{2000\,lb \times 144\,Btu/lb}{24\,hr}$$
$$= 12000 \ Btu/hr = 200 \ Btu/min$$

In general, 1 TR means 200 Btu of heat removal per minute. Thus if a refrigeration system is capable of cooling at the rate of 400 Btu/min, it is a 2 ton machine. A machine of 20 ton rating is capable of cooling at a rate of $20 \times 200 = 4000$ Btu/min. This unit of refrigeration is currently in use in the USA, the UK and India. In many countries, the standard MKS unit of kcal/hr is used. In the MKS it can be seen that

$$1 \ TR = 12000 \ Btu/hr = \frac{12000}{3.968} = 3024.2 \ kcal/hr$$
$$= 50.4 \ kcal/min \ \approx 50 \ kcal/min$$

If Btu ton unit is expressed into SI system, it is found to be **210 kJ/min or 3.5 kW.**

Co-efficient of Performance (COP):

The Co-efficient of Performance is defined as the ratio of heat absorbed in a system to the work supplied.

The theoretical Coefficient of Performance (Carnot), (COP a standard measure of refrigeration efficiency of an ideal refrigeration system) depends on two key system temperatures: evaporator temperature T_e and condenser temperature T_c

COP is given as: $COP_{Carnot} = T_e / (T_c - T_e)$

This expression also indicates that higher COP$_{Carnot}$ is achieved with higher evaporator temperatures and lower condenser temperatures. But COP is only a ratio of temperatures, and does not take into account the type of compressor. Hence the COP normally used in industry is calculated as follows:

$$COP = \frac{\text{Cooling effect (kW)}}{\text{Power input to compressor (kW)}}$$

Where the cooling effect is the difference in enthalpy across the evaporator and expressed as kW.

Ice making capacity:

It is the ability of the refrigeration system to make ice. In other words, it is the capacity of refrigeration system to remove heat from water to make ice.

Relative COP:

It is the ratio of actual COP to the theoretical COP of a refrigerator. Actual COP is measured during a test and theoretical COP is obtained by applying the laws of thermodynamics.

REFRIGERATOR AND HEAT PUMP

The vapor compression refrigeration cycle is a common method for transferring heat from a low temperature to a high temperature.
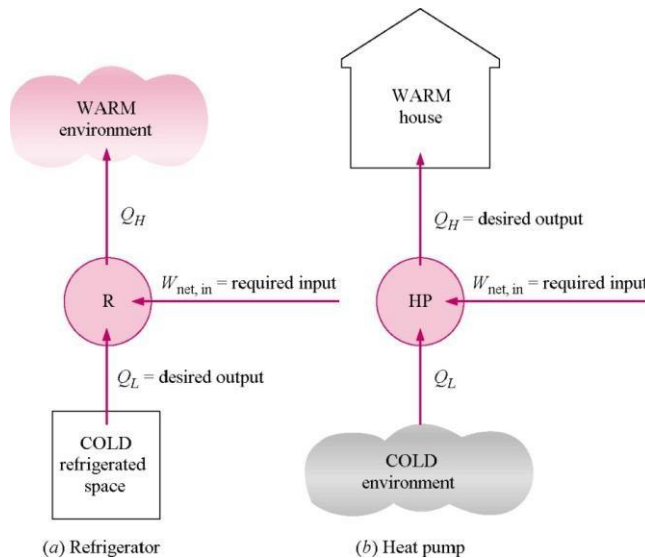

Fig 5.2

The above figure shows the objectives of refrigerators and heat pumps. The purpose of a refrigerator is the removal of heat, called the cooling load, from a low temperature medium. The purpose of a heat pump is the transfer of heat to a high temperature medium, called the heating load. When we are interested in the heat energy removed from a low temperature space, the

device is called a refrigerator. When we are interested in the heat energy supplied to the high temperature space, the device is called a heat pump. In general, the term "heat pump" is used to describe the cycle as heat energy is removed from the low temperature space and rejected to the high temperature space.

The performance of refrigerators and heat pumps is expressed in terms of *coefficient of performance* (COP), defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

Both COP_R and COP_{HP} can be larger than 1. Under the same operating conditions, the COPs are related by

$$COP_{HP} = COP_R + 1$$

TYPES OF REFRIGERATION

Refrigeration is classified as based on working substance used

- Air refrigeration system (Bell-Coleman cycle)

- Water refrigeration system

- Ice refrigeration system

- Refrigeration by special fluid (low boiling point fluids – Refrigerants) (Reversed Carnot cycle)

    o Vapour compression refrigeration system (VCR)

    o Vapour absorbtion refrigeration system (VAR)

    o Vapour adsorbtion refrigeration system and etc.,

Simple Vapour Compression Refrigeration System (VCR)

It consists of the following essential parts:

Compressor

The low pressure and temperature vapour refrigerant from evaporator is drawn into the compressor through the inlet or suction valve A, where it is compressed to a high pressure and temperature. This high pressure and temperature vapour refrigerant is discharged into the condenser through the delivery or discharge valve B.

Condenser

The condenser or cooler consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.
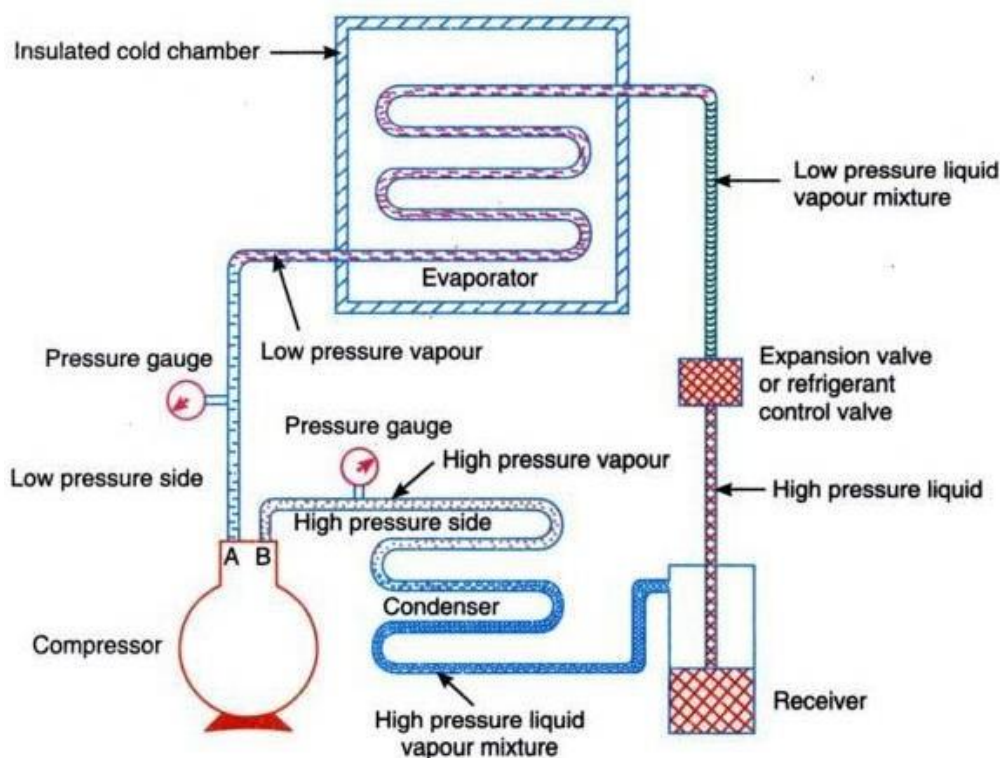

Fig 5.3

Receiver

The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from where it is supplied to the evaporator through the expansion valve or refrigerant control valve.

Expansion Valve

It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator at the low pressure and temperature

Evaporator

An evaporator consists of coils of pipe in which the liquid-vapour. Refrigerant at low pressure and temperature is evaporated and changed into vapour refrigerant at low pressure and temperature. In evaporating, the liquid vapour refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which is to be cooled.

The Simple Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle has four components: evaporator, compressor, condenser, and expansion (or throttle) valve. The most widely used refrigeration cycle is the *vapor-compression refrigeration cycle.* In an ideal or simple vapor-compression refrigeration cycle, the refrigerant enters the compressor as a saturated vapor and is cooled to the saturated liquid state in the condenser. It is then throttled to the evaporator pressure and vaporizes as it absorbs heat from the refrigerated space.

The ideal vapor compression cycle consists of four processes.

Ideal Vapor-Compression Refrigeration Cycle

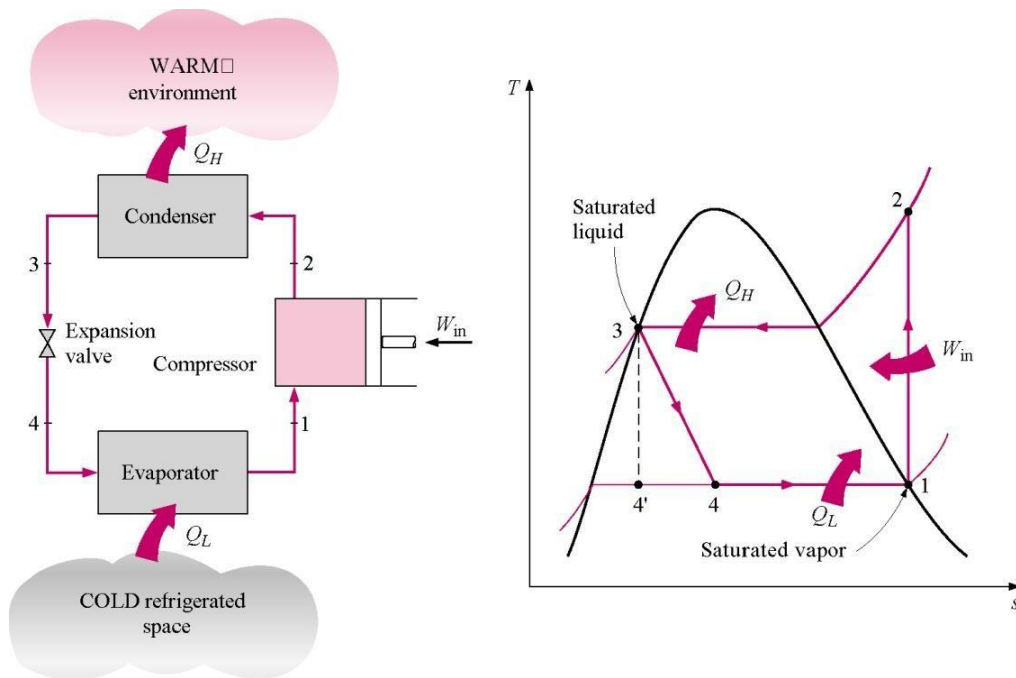| Process	Description
1-2	Isentropic compression
2-3	Constant pressure heat rejection in the condenser
3-4	Throttling in an expansion valve
4-1	Constant pressure heat addition in the evaporator

Fig 5.4

The P-h diagram is another convenient diagram often used to illustrate the refrigeration cycle.
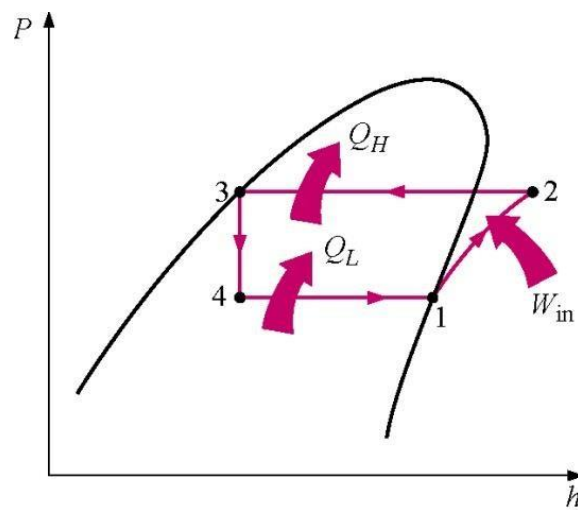

Fig 5.5

The ordinary household refrigerator is a good example of the application of this cycle.
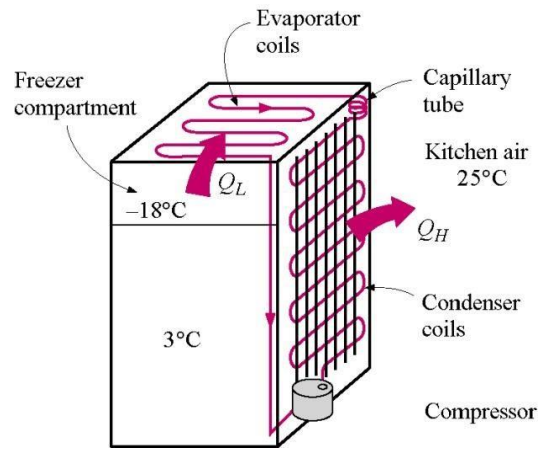
Fig 5.6

Results of First and Second Law Analysis for Steady-Flow

Component	Process	First Law Result
Compressor	s = Const.	$W_{in} = m(h_2 - h_1)$
Condenser	P = Const.	$Q_H = m(h_2 - h_3)$
Throttle Valve	$\Delta s > 0$	$h_4 = h_3$
	$W_{net} = 0$	
	$Q_{net} = 0$	
Evaporator	P = Const.	$Q_L = m(h_1 - h_4)$

$$COP_R = \frac{Q_L}{W_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_{HP} = \frac{Q_H}{W_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1}$$

Methods to enhance the COP of simple vapour compression refrigeration system

1. Cycle with dry saturated vapour after compression,
2. Cycle with wet vapour after compression,
3. Cycle with superheated vapour after compression,
4. Cycle with superheated vapour before compression, and
5. Cycle with undercooling or subcooling of refrigerant.

Theoretical Vapour Compression Cycle with Dry Saturated Vapour after Compression

A vapour compression cycle with dry saturated vapour after compression is shown in the following Figures (a) and (b) respectively. At point 1, let T_1, p_1, S_1 be the temperature, pressure and entropy of the vapour refrigerant respectively. The four processes of the cycle are as follows:
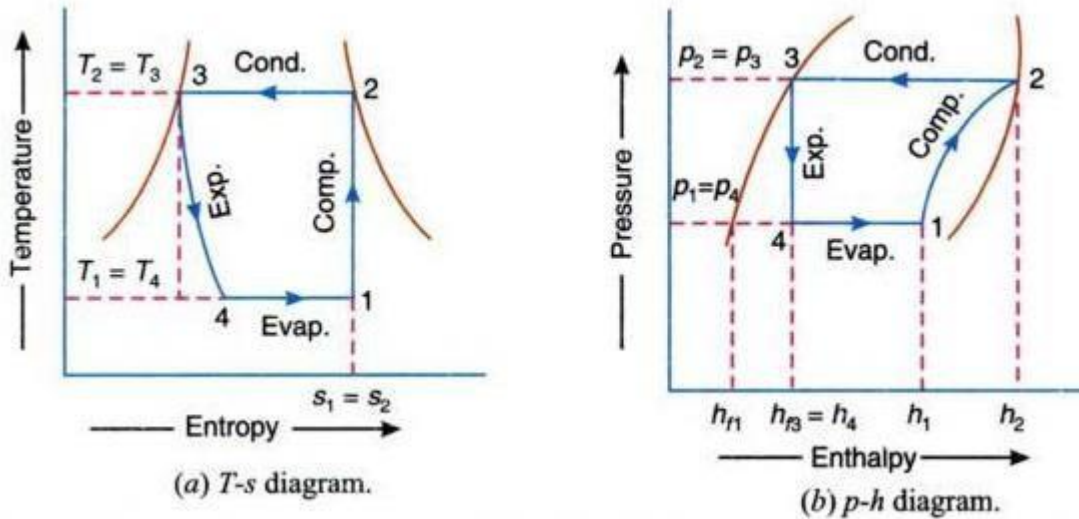

Fig 5.7

Compression Process

The vapour refrigerant at low pressure $p1$ and temperature$T1$ is compressed isentropic ally to dry saturated vapour as shown by the vertical line 1-2 on the T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rise from $p1$ to $p2$ and $T1$ to $T2$ respectively.

The work done during isentropic compression per kg of refrigerant is given by

$w = h2 - h1$

where $h1$ = Enthalpy of vapour refrigerant at temperature $T1$, i.e. at suction of the compressor, and

$h2$ = Enthalpy of the vapour refrigerant at temperature $T2$. i.e. at discharge of the compressor.

Condensing Process

The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure p2 and temperature T2 as shown by the horizontal line 2-3 on T-s and p-h diagrams. The vapour refrigerant is changed into liquid refrigerant. The refrigerant, while passing through the condenser, gives its latent heat to the surrounding condensing medium.

Expansion Process

The liquid refrigerant at pressure p3 = p2 and temperature T3 = T2, is expanded by throttling process through the expansion valve to a low pressure p4 = p1 and Temperature T4 = T1 as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on p-h diagram. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

Vaporizing Process

The liquid-vapour mixture of the refrigerant at pressure p4 = p1 and temperature T4 = T1 is evaporated and changed into vapour refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporization from the medium (air, water or brine) which, is to be cooled, This heat which is absorbed by the refrigerant is called refrigerating effect and it is briefly written as R_E. The process of vaporization continues up to point 1 which is the starting point and thus the cycle is completed.

We know that the refrigerating effect or the heat absorbed or extracted by the liquid-vapour refrigerant during evaporation per kg of refrigerant is given by

$$R_E = h_1 - h_4 = h_1 - hf_3$$

where hf3 = Sensible heat at temperature T3, i.e. enthalpy of liquid refrigerant leaving the condenser.

It may be noticed from the cycle that the liquid-vapour refrigerant has extracted heat during evaporation and the work will be done by the compressor for isentropic compression of the high pressure and temperature vapour refrigerant.

Coefficient of performance, C.O.P. = (Refrigerating effect)/(Work done)

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{h_1 - h_{f3}}{h_2 - h_1}$$

Theoretical Vapour Compression Cycle with Wet Vapour after Compression

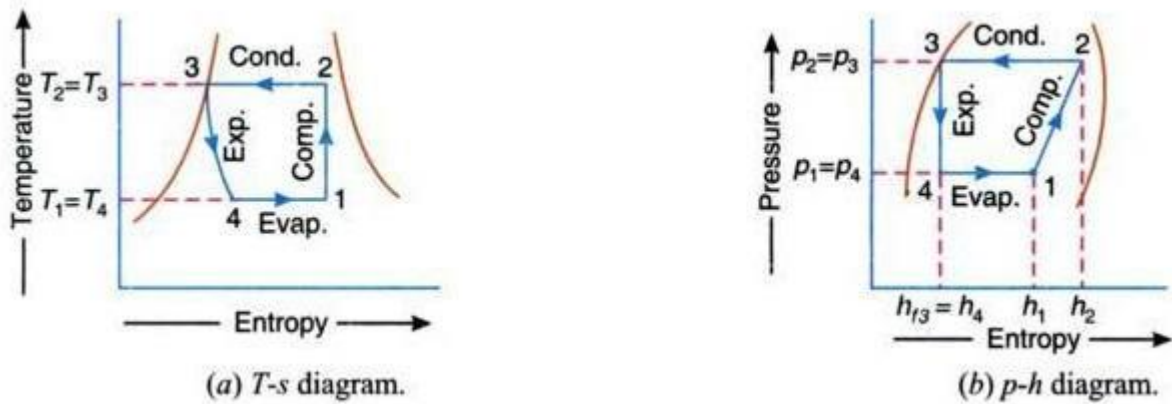(a) T-s diagram.　　　　(b) p-h diagram.

Fig 5.8

From the above figure, the end point of compression is lies in the region of wet (liquid and vapour). The enthalpy and entropy at this point is calculated by following formula

$$h_2 = h_{f2} + x_2 \, h_{fg2}$$

$$s_2 = s_{f2} + \frac{x_2 \, h_{fg2}}{T_2}$$

Theoretical Vapour Compression Cycle with Superheated Vapour after Compression

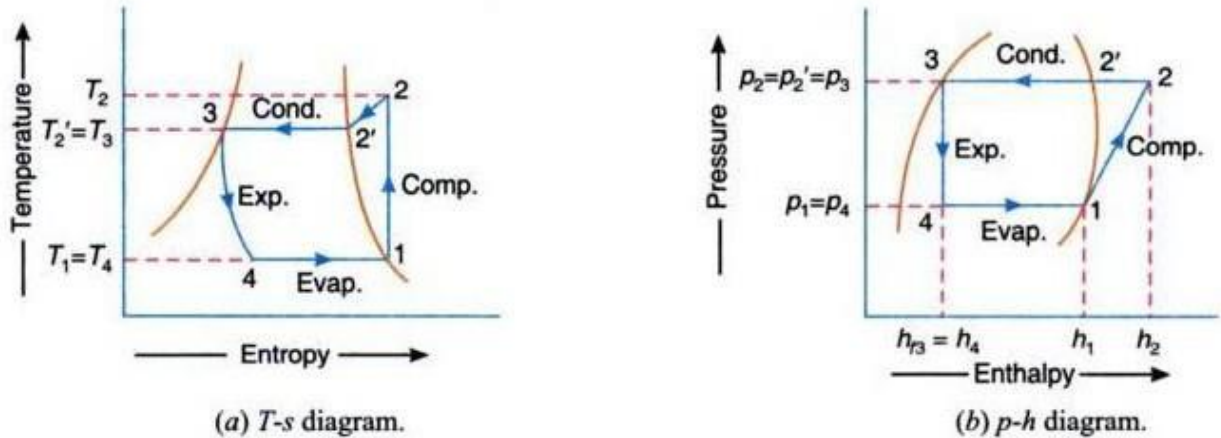(a) T-s diagram.　　　　(b) p-h diagram.

Fig 5.9

From the above figure, the end point of compression is lies in the region of superheated vapour. The enthalpy and entropy at this point is calculated by following formula

$$h_2 = h_{2'} + c_p \times \text{Degree of superheat} = h_{2'} + c_p \, (T_2 - T_{2'})$$

$$s_2 = s_{2'} + 2.3 \, c_p \, \log \left(\frac{T_2}{T_{2'}} \right)$$

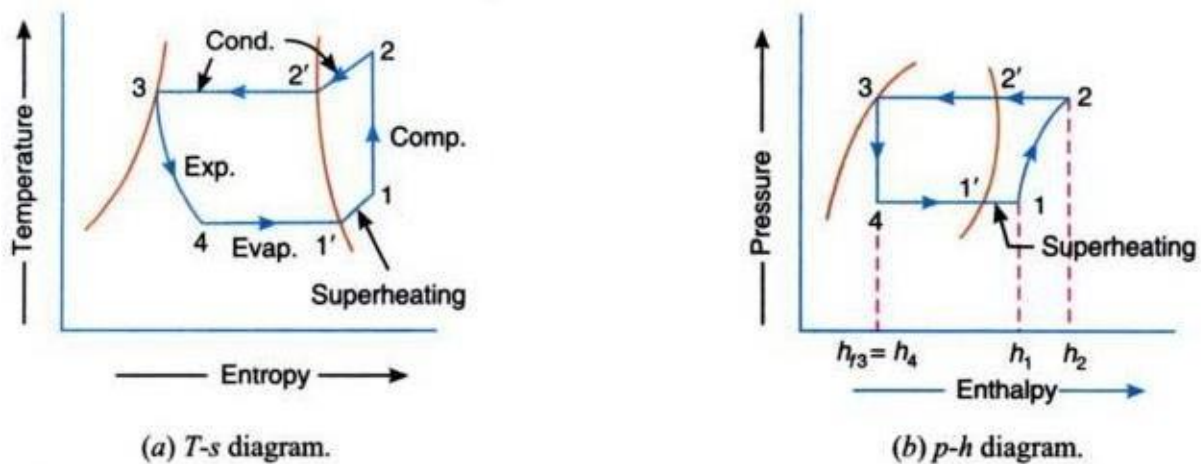**Theoretical Vapour Compression Cycle with Superheated Vapour before Compression**

(a) T-s diagram.

(b) p-h diagram.

Fig 5.10

Theoretical Vapour Compression Cycle with Sub-Cooling or Under cooling ofRefrigerant

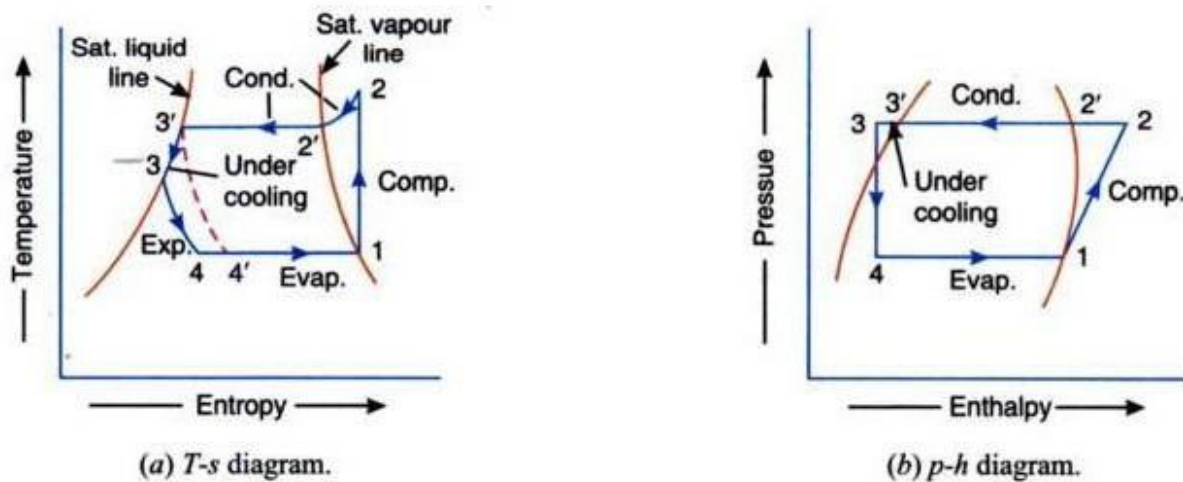(a) T-s diagram.

(b) p-h diagram.

Fig 5.11

$$h_{f3} = h_{f3'} - c_p \times \text{Degree of undercooling}$$

The process 3-3', cooling of the refrigerant temperature below its saturation temperature value is called **sub cooling or under cooling** at the end of condensation process.

REFRIGERANTS

The working agent in a refrigerating system that absorbs carries or releases heat from the place to be cooled or refrigerated can be termed as a refrigerant. This heat transfer generally takes place through a phase change of the refrigerant. A more complete definition of a refrigerant could be given as follows:

"Refrigerant is the fluid used for heat transfer in a refrigerating system that absorbs heat during evaporation from the region of low temperature and pressure, and releases heat during condensation at a region of higher temperature and pressure."

Primary and secondary refrigerants

Primary refrigerants are those which can be directly used for the purpose of refrigeration. If the refrigerant is allowed to flow freely into the space to be refrigerated and there is no danger of possible harm to human beings, then primary refrigerants are used. The refrigerants used in home refrigerators like Freon-12 are primary refrigerants.

On the other hand, there may be certain situations in which we cannot allow the refrigerant to come in direct contact with the items being refrigerated, and then the refrigerant used is termed as a secondary refrigerant. As for example, we cannot allow a toxic refrigerant to be used for air conditioning in residential buildings. There are some refrigerants which are highly inflammable and so their direct use is forbidden for safety reasons. Again, it may so happen that if direct refrigeration, such as in cooling a big cold storage, is allowed, then the amount of refrigerant required may be so large that its cost becomes prohibitively high. These are some typical situations for which we favor the use of secondary refrigerants. Water and brine solutions are common examples of secondary refrigerants.

DESIRABLE PROPERTIES OF REFRIGERANTS

The vast number of refrigerants available in the market today allows us to choose a refrigerant depending upon the operating conditions of the refrigeration system. As such, there is no refrigerant that can be advantageously used under all operating conditions and in all types of refrigeration systems. In spite of that, we can state certain desirable properties that a refrigerant should posses. These properties can be divided into favorable thermodynamic, chemical and physical properties:

Thermodynamic   Properties

Critical Temperature and Pressure

The critical temperature of the refrigerant should be as high as possible above the condensing temperature in order to have a greater heat transfer at a constant temperature. If this is not taken care of, then we will have excessive power consumption by the refrigeration system. The critical pressure should be moderate and positive. A very high pressure will make the system heavy and bulky whereas in case of very low pressures, there is a possibility of air leaking into the refrigerating system.

Specific Heat

The specific heat of the liquid should be as small as possible. This ensures that the irreversibilities associated with throttling are small and there is greater subcooling of the liquid. On the other hand, the specific heat of vapor should be high to have less superheating of the vapor.

Enthalpy of Vaporization

This should be as large as possible to minimize the area under superheat and the area reduction due to throttling. Also, the higher value of enthalpy of vaporization lowers the required flow rate per ton of refrigeration.

Conductivity

The conductivity of the refrigerant should be as high as possible so that the size of the evaporator and condenser is manageable. From this viewpoint, ammonia has a better conductivity than that of R12 or R22 and is more suitable than the latter. But, ammonia is toxic and this does not allow its use in home refrigeration systems.

Evaporator and Condenser Pressure

Both the evaporator and condenser pressures need to be above atmospheric pressure otherwise there is a possibility of air leaking into the system. Presence of air drastically reduces the capacity of the refrigeration system. Also, due to presence of moisture in air, acids or other corrosive compounds may form and this may affect the tubing of the refrigeration system.

Compression Ratio

The compression ratio needs to be as small as possible otherwise the leakage of refrigerant occurs across the piston. Also, the volumetric efficiency is affected.

Freezing Point

It should be as low as possible or else there will be a possibility of blockage of passages during flow of fluid through evaporator.

Volume of Refrigerant Handled Per Ton of Refrigeration

This should be as small as possible in order to have a small size of the compressor. The type of compressor is decided by this value. For refrigerants like R12, R500, R22 etc., a reciprocating compressor is suitable. For others like R11 and water, a centrifugal compressor is required to handle the large volume.

Coefficient of Performance

The Coefficient of performance or COP has a direct bearing on the running cost of the refrigeration system. Higher the magnitude of COP, lower will be the running cost. Since, the COP of any refrigeration system is limited by the Carnot COP, for large operating pressures a multi-stage refrigeration system should be employed. CO_2 has a very low COP. Hence, it is not suitable for use as a refrigerant.

Density

The density of the refrigerant should be as large as possible. In reciprocating compressors, the pressure rise is accomplished by squeezing the entrapped fluid inside the piston-cylinder assembly. Hence, density decides the size of the cylinder. Again in centrifugal compressors pressure rise is related to the density of the vapor. A high value of density results in high pressure rise.

Compression Temperature

Whenever a refrigerant gets compressed, there is a rise in the temperature of the refrigerant resulting in the heating of the cylinder walls of the compressor. This necessitates external cooling of the cylinder walls to prevent volumetric and material losses. Refrigerants having lowest compression temperatures are thus better than others.

Chemical Properties Chemical

Stability and Inertness

It should be chemically stable for the operating ranges of temperature. Also, it should not react with the materials of the refrigeration system or with which it comes into contact. Further, it should be chemically inert and must not undergo polymerization reactions at either the lower or higher ranges of temperatures.

Action on Rubber or Plastics

Rubber and plastics are used extensively in the refrigeration system. These materials are mostly used in the seals and gaskets of the refrigeration system. They help to prevent the leakage of the refrigerant and ensure the smooth functioning of the compressor. The refrigerant should not react with them or else there might be leakage of refrigerant from the system or loss of functioning of the compressor.

Flammability

The refrigerant should be inert and not catch fire when subjected to high temperatures. From this viewpoint CO_2 is the most suitable as it is not only non-flammable, but also acts as a fire-extinguisher. Ethane, butane, isobutene are highly undesirable as they catch fire quickly.

Effect on Oil

The refrigerant should not react with the lubricating oil else, there is a possibility of loss of lubricating action due to either thickening or thinning of the oil. It should not be soluble in the oil else there will be reduction in the viscosity of the lubricating oil.

Effect on Commodity

If the refrigerant is directly used for chilling, then it should not affect the commodity kept in the conditioned space. Also, in case where direct cooling is not employed, the refrigerant should still not affect the commodity if there is any leakage.

Toxicity

The refrigerant used in air conditioning, food preservation etc. should not be toxic as they will come into contact with human beings.

Physical Properties

Leakage and Detection

Since pressures higher than atmospheric are usually employed in refrigeration systems, there is a possibility of leakage of refrigerants after long period of operation. It is desirable to detect this leak early else the system would operate under reduced capacity or stop functioning altogether. Hence, it is desirable that the refrigerant has a pungent smell so that its leakage can be detected immediately.

Miscibility with Oil

The refrigerant should not be miscible with the oil else the lubricating strength will be reduced.

Viscosity

It should be as small as possible to ensure that the pressure drop in the system is as small as possible. A low viscosity refrigerant will require less energy for its circulation through the refrigeration system.

Safety Criteria

Under safety criteria, we consider the toxicity, flammability, action on perishable food and formation of explosive compound on exposure to air. An ideal refrigerant should be non-toxic, non-flammable, have no effect on food products and should not react with atmospheric air. No refrigerant satisfy these criteria fully. We can therefore, group refrigerants into different sub-groups based on their flammability and toxicity levels.

Economic Criteria

Apart from the thermodynamic, chemical, physical and safety criteria, there is another criterion by which we judge an ideal refrigerant. The economic criterion takes into account the cost of the refrigerant, the availability and supply levels of the refrigerant, cost of storage and handling. We discuss each of these in detail below.

Cost of Refrigerant

The cost of the refrigerant has a big impact on the overall cost of the refrigeration system. Hence, its cost should be as low as possible. From this viewpoint, ammonia and water are ideally suited, but their low thermodynamic and chemical properties restrict their use in all types of refrigeration systems. Particularly, for flooded type evaporator or condenser, the refrigerant amount required is high and their cost needs to be factored in while making the initial investments.

Availability and Supply

The refrigerant should be easily available in the market and in abundant quantity. This ensures that the cost of the refrigerant is not prohibitive. An abundant and free supply of the refrigerant ensures that refrigeration systems will be designed specifically for use with them.

Storage and Handling

The refrigerant should be such that it can be conveniently stored and handled during transportation and charging. It should be stored in as small a pressure vessel as possible. Also, if we have to handle a toxic or flammable refrigerant, then the cost involved will be higher compared to handling and storage cost of non-toxic and non-flammable refrigerant.

VAPOUR ABSORPTION SYSTEM

Introduction

In a vapour absorption system the refrigerant is absorbed on leaving the evaporator, the absorbing medium being a solid or liquid. In order that the sequence of events should be continuous it is necessary for the refrigerant to be separated from the absorbent and subsequently

condensed before being returned to the evaporator. The separation is accomplished by the application of direct heat in a 'generator'. The solubility of the refrigerant and absorbent must be suitable and the plant which uses ammonia as the refrigerant and water as absorbent will be described.

Simple Vapour Absorption System

Refer Fig. 5.12 for a simple absorption system. The solubility of ammonia in water at low temperatures and pressures is higher than it is at higher temperatures and pressures. The ammonia vapour leaving the evaporator at point 2 is readily absorbed in the low temperature hot solution in the absorber. This process is accompanied by the rejection of heat. The ammonia in water solution is pumped to the higher pressure and is heated in the generator. Due to reduced solubility of ammonia in water at the higher pressure and temperature, the vapour is removed from the solution. The vapour then passes to the condenser and the weakened ammonia in water solution is returned to the absorber.
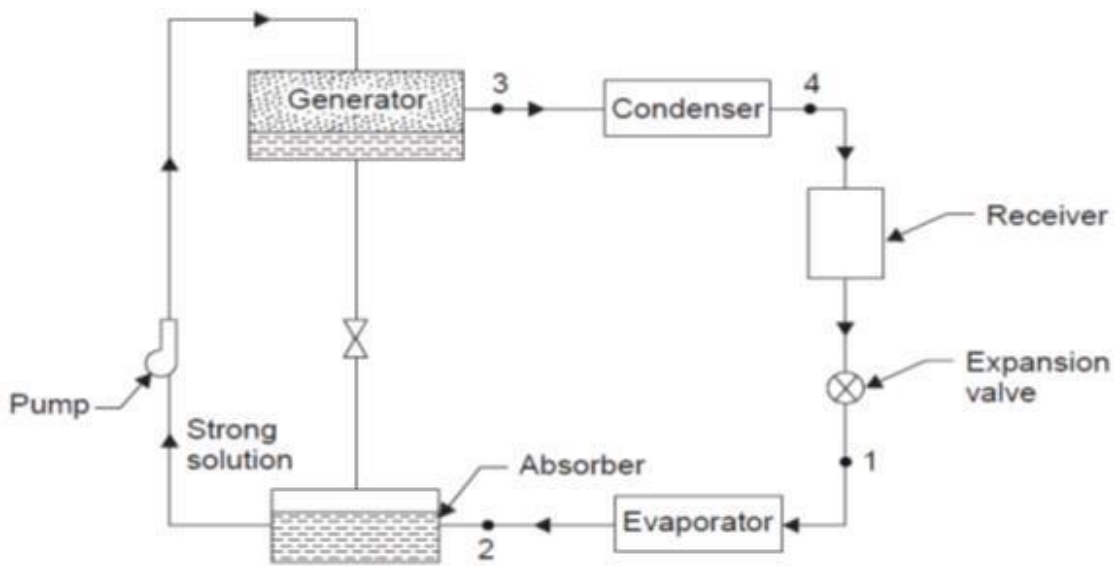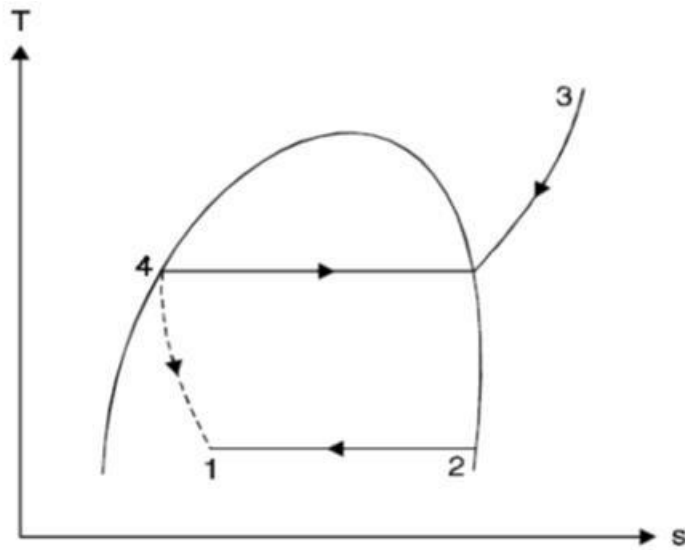

Fig 5.12

Simple vapour absorption system—T-s diagram.

Fig 5.13

In this system the work done on compression is less than in vapour compression cycle (since pumping a liquid requires much less work than compressing a vapour between the same pressures) but a heat input to the generator is required. The heat may be supplied by any convenient form e.g. steam or gas heating.

Practical Vapour Absorption System

Refer Fig. 5.14. Although a simple vapour absorption system can provide refrigeration yet its operating efficiency is low. The following accessories are fitted to make the system more practical and improve the performance and working of the plant.

1. Heat exchanger. 2. Analyser. 3. Rectifier.

1. **Heat exchanger.** A heat exchanger is located between the generator and the absorber.

The strong solution which is pumped from the absorber to the generator must be heated ; and the weak solution from the generator to the absorber must be cooled. This is accomplished by a heat exchanger and consequently cost of heating the generator and cost of cooling the absorber are reduced.

2. **Analyser.** An analyser consists of a series of trays mounted above the generator. Its main function is to remove partly some of the unwanted water particles associated with ammonia vapour going to condenser. If these water vapours are permitted to enter condenser they may enter the expansion valve and freeze ; as a result the pipe line may get choked.
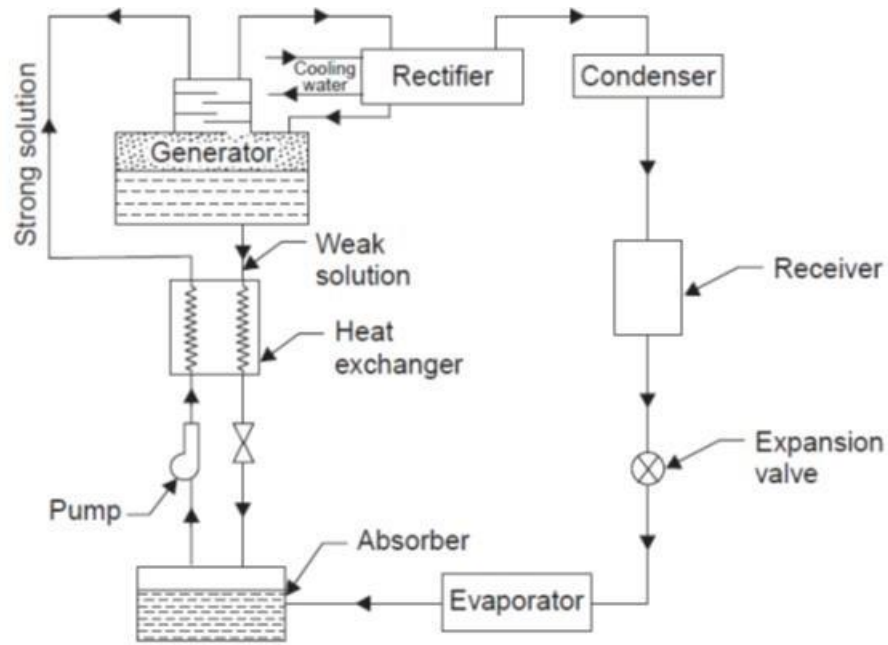
Fig 5.14

3. **Rectifier.** A rectifier is a water-cooled heat exchanger which condenses water vapour and some ammonia and sends back to the generator. Thus final reduction or elimination of the percentage of water vapour takes place in a rectifier. The co-efficient of performance (C.O.P.) of this system is given by :

$$\text{C.O.P.} = \frac{\text{Heat extracted from the evaporator}}{\text{Heat supplied in the generator} + \text{Work done by the liquid pump}} \; .$$

1. A 5 tonne refrigerator plant uses RR as refrigerant. It enters the compressor at -5°C as saturated vapour. Condension takes place at 32°C and there is no under cooling of refrigerant liquid. Assuming isentropic compression,determine COP of the plant, mass flow of refrigerant, power required to run the compressor in kw. The properties of R-12 are given table.

T(°C)	P(bar)	Enthalpy(kw/kg)		Entropy(KJ/kgk)
		hf	hg	Sg
32	7.85	130.5	264.5	1.542
-5	2.61	-	249.3	1.557

Solution:

Beginning of compression in dry and end of compression is superheated. So the P-h and T-S diagrams are

From table, at point 1

$T_1 = -5°C = 268K$

$hg_1 = 2493kJ/Kg$, $Sg_1 = 1.557 KJ/Kgk$

At point 2

$T_2 = 32°C = 305k$, $hf_2 = 130.5 KJ/Kg$, $hg_2 = 264.5$, $Sg_2 = 1.542 KJ/KgK$

From ph diagram, At point eqn(1) (dry).

At -5°C, i.e at 268k

$hg_1 = 249.3 KJ/Kg = hg$

$h_1 = 249.3 KJ/Kg$

At 32°C, i.e at 305K

$hg_2 = 264.5 KJ/Kg = h_2'$

$h_2' = 264.5 KJ/Kg$

Entropy is constant during the compression process so,

$S_1 = S_2$

From T- S diagram

At point (1) dry,

$S_1 = S_g$ at -5°C

$S_1 = S_{g_1} = 1.557 KJ/Kgk$

$S_1 = S_2 = 1.557 KJ/Kgk$

At point (2) (super heated)

$S_1 = S_2' + C_p \ln (T_2/T_2')$

$1.557 = S_2' + 1.884 \ln(T_2/305)$ -------- (1)

$S_2' = S_g$ at 32°C.

$S_{g_2} = 1.542 = S_2$

$S_2' = 1.542 KJ/Kg \ k$

$1.884 \ln (T_2/305) = 0.015$

$T_2 = 307.44 k$

For super heated vapour the enthalpy is

$h_2 = h_2' + C_p(T_2 - T_1')$

$h_2 = 264.5 + 1.884 (307.44 - 305)$

$h_2 = 269.1 \ KJ/Kg$

From P-h diagram, we know that,

$h_3 = h_4$

$h_3 = h_f$ at 32°C

$h_{f \, 2} = 130.5 = h_f$

We Know that,

COP = Refrigeration effect / Work done = $(h_1 - h_4) / (h_2 - h_1)$

$= (2493 - 130.5)/(269.1 - 249.3) = 6$

Refrigeration effect = m× (h1- h4)

$$m= (2\times210)/ (249.3-130.5)$$

$$m = 8.84 \text{Kg/min}$$

Work done = Refrigeration effect/ cop

$$= (2\times210)/6 = 175 \text{ KJ/min}$$

Power = 2.92kw.

2. A refrigerator works between -7°C and 27°C the vapour is dry at the end of adiabatic compression. Assuming there is no under cooling determine (i) cop (ii) power of the compressor to remove a heat load of 12140KJ/hr.The properties of refrigerant are given in

T(°C)	sensible Heat (h_f)	Latent heat(h_{fg}) KJ/Kgk)	Entropy of liquid (KJ/Kgk)	Entropy of vapour Sg (KJ/Kgk)
-7	-29.3	1297.9	-0.109	4.748
27	1117.23	1172.3	0.427	4.333

table.

Solution:

The vapour is dry at end of compression i.e, beginning of compression is wet and of compression is dry saturated.

At point (1)

T_1=-7°C=266k, h_{fg1}=1297.9 KJ/Kg, S_{fl}=-0.109 KJ/KgK

h_{fl}=-29.3 KJ/Kg, S_{fg1}=4.478 KJ/KgK

At point (2)

T_2=27°C=300k, h_{fg2}=1172.3 KJ/Kg, S_{f2}= 0.427KJ/KgK

$h_{f2}=117.23$ KJ/Kg, $S_{fg2}=4.333$KJ/KgK

We point $S_1=S_2$

At point (1) (wet)

$S_1=S_{wet}=S_{f1}+x_1+S_{fg1}$

$S_1=-0.109+x_1(S_{fg1}-S_{f1})$ $(S_{fg}=S_g-S_f)$

$S_1=-0.109+x_1(4.857)$

At point (2) (dry)

$S_2=S_{g2}=4.33$KJ/KgK

$S_2=4.33$KJ/KgK

$S_1=S_2$ So, $4.33=-0.109+x_1(4.857)$

Dryness fraction

$x_1=0.913$

At point (1) (wet)

$h_1=h_{f1}+x_1\times h_{fg1}$

$h_1=-29.3+0.913\times1297.3$

$h_1=1156.3$ KJ/Kg

At point (2) (dry)

$h_2=h_{f2}+h_{g2}$

$h_2=117.23+1172.3$

$h_2=1289.53$ KJ/Kg

From P-h diagram

$h_3=h_4$

$h_3=h_{f2}$

$h_3=1172.3$ KJ/Kg

$h_4 = 117.23$ KJ/Kg

COP = $(h_1 - h_4)/(h_2 - h_1)$ = (1156.3 - 117.23) / (1289.53 - 1156.3) = 7.7

Work done = Heat removed/ COP
= 12140/7.7

Power

= 0.43 KJ/hr

3. Comparison between vapour compression and vapour absorption systems.

Compression systems	Absorption systems
Work operated	Heat operated
High COP	Low COP
Performance very sensitive to evaporator temperatures.	Performance not very sensitive to evaporator temperatures.
System COP reduces considerably at part loads.	COP does not reduce significantly with load.
Liquid at the exit of evaporator may damage compressor.	Presence of liquid at evaporator exit is not a serious problem.
Performance of sensitive to evaporator superheat.	Evaporator superheat is not very important.
Many moving parts	Very few moving parts
Regular maintenance required	Very low maintenance required
Higher noise and vibration	Less noise and vibration.

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