# SATHYABAMA
## INSTITUTE OF SCIENCE AND TECHNOLOGY

**DEPARTMENT OF Electrical and Electronics Engineering**

**COURSE MATERIAL**

**Subject Name: Applied Thermal Engineering UNIT I Subject Code: SME1208**

CONCEPTS - CONCEPT OF CONTINUUM, MACROSCOPIC APPROACH, THERMODYNAMIC SYSTEMS - CLOSED, OPEN AND ISOLATED. PROPERTY, STATE, PATH AND PROCESS, QUASI-STATIC PROCESS, WORK, MODES OF WORK, ZEROTH LAW OF BASIC THERMODYNAMICS - CONCEPT OF TEMPERATURE AND HEAT. CONCEPT OF IDEAL GAS. FIRST LAW OF THERMODYNAMICS - APPLICATION TO CLOSED AND OPEN SYSTEMS, INTERNAL ENERGY, SPECIFIC HEAT CAPACITIES, ENTHALPY, STEADY FLOW PROCESS WITH REFERENCE TO VARIOUS THERMAL EQUIPMENTS.

**BASIC DEFINITION -**Thermodynamics is the field of science which deals with the relationship among heat, work and properties of system which are in equilibrium with one another. It is a characteristic of the system. The system is identified by some quantise like temperature, pressure, volume etc.

**CLASSIFICATIONS OF THERMODYNAMICS** I.

Classical Thermodynamics –Macroscopic Approach II.

Statistical Thermodynamics -Microscopic Approach

**Macroscopic Approach**-Macroscopic Approach is the study of thermodynamics which does not require any knowledge of the behaviour of individual particles of the substance is called classical thermodynamics

**Microscopic Approach**- Microscopic Approach is the study of thermodynamics based on the behaviour of a large numbers of individual particles is called statistical thermodynamics.

**INTENSIVE AND EXTENSIVE PROPERTIES**

**Intensive Properties -**The properties are independent on mass of the system or independent on size of the system.

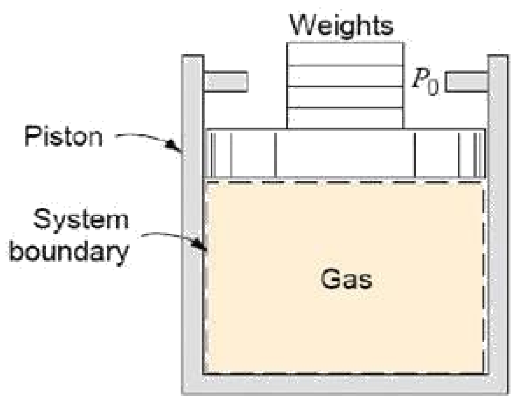
Example: Pressure, Temperature, Specific volume etc.

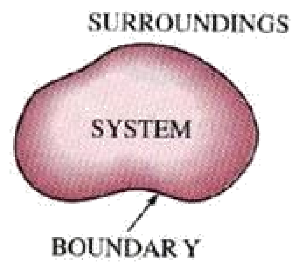
**Extensive properties**- The properties are dependent of the mass of the system or extensive properties are those which vary directly with the size or extent of the system.

Example: Total volume, total energy

**SYSTEM SURROUNDINGS AND BOUNDARY**

In thermodynamics we confine our attention to a particular part of the universe which we call our system. A system is a region of space containing a quantity of matter whose behavior is being investigated. This quantity of matter is separated from the surroundings by a boundary, which may be a physical boundary like walls of a vessel, or some imaginary surface enveloping the region. The term surroundings is restricted to those portions of the matter external to the system, which are affected by changes occurring within the system. Before any thermodynamic analysis is attempted, it is necessary to define the boundary of the system because it is across the boundary that work, heat and mass are said to be transferred.




System, surroundings, and boundary.

**Surrounding** - The rest of the universe outside our system is call the surroundings.

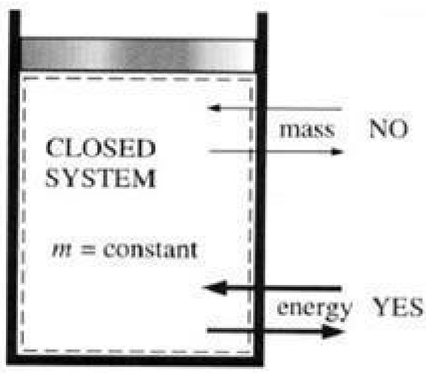
**Boundary**-The system and the surroundings are separated by a boundary or a wall. They may, in general, exchange energy and matter, depending on the nature of the wall.

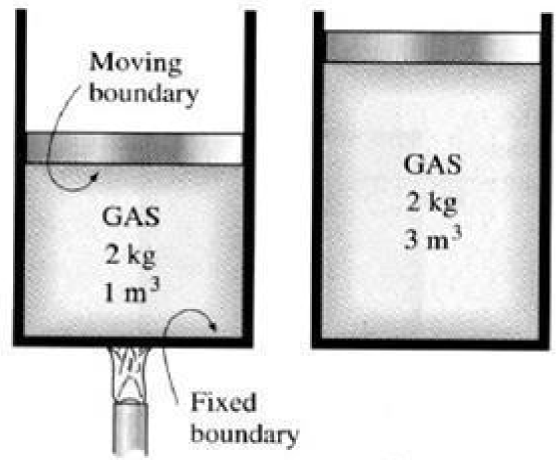
## TYPES OF THERMODYNAMICS SYSTEMS

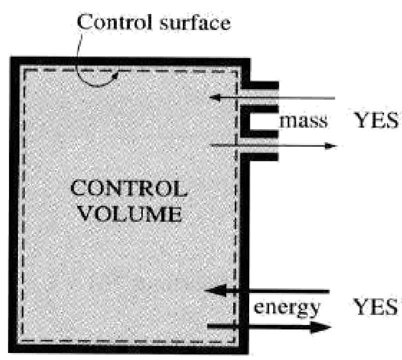
**Closed system** – When the same matter remains within the region throughout the process under investigation it is called closed system. In this case, only heat and work cross the boundary mass does not transfer,

**Open system**- An open system is a region in space defined by a boundary across which the matter may flow in addition to work and heat, it meant both mass and energy can transfer

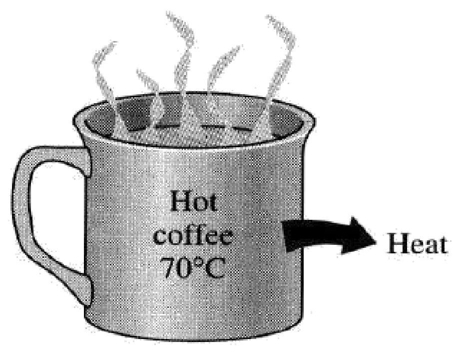
**Isolated system**-The isolated system is one in which there is no interaction between the system and surroundings. There is no mass or energy transfer across the system boundary.




Both mass and energy can cross the boundaries of a control volume.




Examples of open system- flow nozzles, steam turbine, boiler etc.,

Examples of closed system- mixer of water and steam in a closed vessel, a gas expanding in a cylinder by displacing a piston. Hence, for a closed system, boundary need not be fixed; it may contract or expand to accommodate any change in volume undergone by a fixed quantity of fluid.

Examples of isolated system -entire universe of the system

## EQUILIBRIUM OF THE SYSTEM

When a system is said to be in equilibrium, it involves no change with time. Also it can be defined as a system is said to be in equilibrium, if it does not tend to undergo any change of state on its own accord.

When a system is in Thermodynamic equilibrium, it should be satisfy the following three conditions.

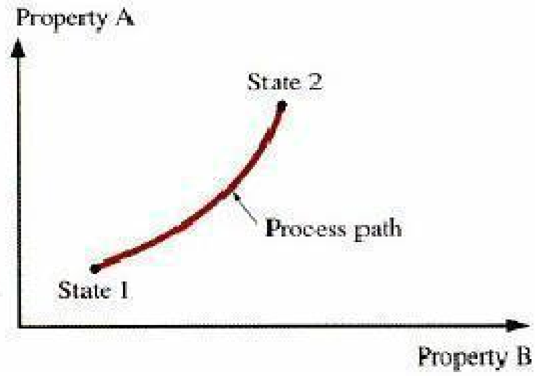
**(a) Mechanical Equilibrium** – Pressure remains constant. No pressure difference exists between any two points within the system (Neglecting gravitational effects) and between the system and surroundings, so that it is mechanically balanced.

**(b)Thermal Equilibrium** – Temperature remains constant. There should not be any temperature difference within the system, so that the system is thermally balanced.

**(c) Chemical Equilibrium** – There is no chemical reaction of the system. No chemical reaction is taking place, so that it is chemically balanced.

## STATE, PROPERTY, PATH, PROCESS

Every system has certain characteristics by which its physical condition may be described, e.g. volume temperature, pressure etc. Such characteristics are called properties of the system. These are all macroscopic in nature. When all the properties of a system have definite values, the system is said to exist at a definite state. Any operation in which one or more properties of a system change is called a change of state. The succession of states passed through during a change of state is called the path of the change of state. When the path is completely specified, the change of state is called a process, e.g. a constant pressure process. The value of property does not depend on process through which the fluid is passed. The change in the value of property depends on the initial and final states of the system. Pressure, specific volume and temperature are some examples of basic properties. Three more properties- internal energy, enthalpy and entropy emerge as a consequence of First and Second Laws of Thermodynamics. From these six properties, only two may be selected to determine the state of a closed system in thermodynamic equilibrium and the remaining four values are then fixed. Care must be taken to see that the two properties are independent of each other, i.e. it must be possible to vary one of these properties without changing the other For example, when a liquid is in contact with its vapour in a closed vessel it is found that the temperature at which the liquid and vapour in equilibrium is always associated with a particular pressure and one cannot change one without the other. Pressure and temperature cannot be used to determine the state of such systems. However, pressure and specific volume may be used to define the state of such system. It follows that the initial and final states of any closed system can be located as points on a diagram using two properties as coordinates. Properties may be of two types. Intensive (Intrinsic) properties are independent of mass of the system, e.g. pressure, temperature, etc. Extensive properties are related to the mass of the system, e.g. volume, energy etc. A process that eventually returns to its initial state is called a cyclic process.

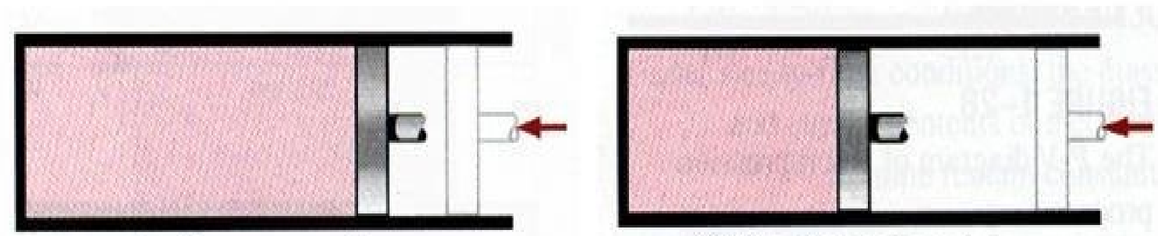



**ZEROTH LAW OF THERMODYNAMICS.** Zeroth law of thermodynamics states that when two systems are separately in thermal equilibrium with a third system , then they themselves are in thermal equilibrium with each other.

**QUASI - STATIC PROCESS**

The process is said to be quasi-static, it should proceed infinite slow and follow continuous serious of equilibrium states. Therefore, the quasi - static process may be a reversible process.

A process is said to the reversible, it should trace the same path in the reverse direction when the process is reversed, and it is possible only when the system passes through a continuous series of equilibrium state if a system does not pass through continuous equilibrium state, then the system is said to be irreversible.



(*a*) Slow compression (quasi-equilibrium)

(*b*) Very fast compression (non-quasi-equilibrium)

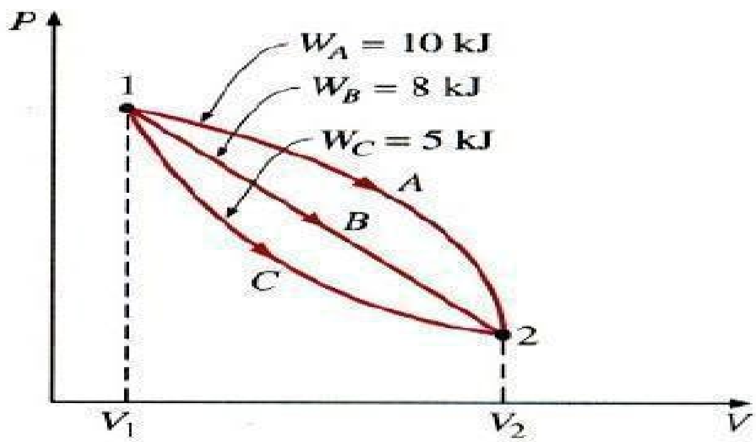
**POINT AND PATH FUNCTION OF THERMODYNAMICS SYSTEMS**

**Point function**:

The quantity which is independent on the process or path followed by the system is known as point function. Ex: Pressure, volume, temperature etc

**Path function:**

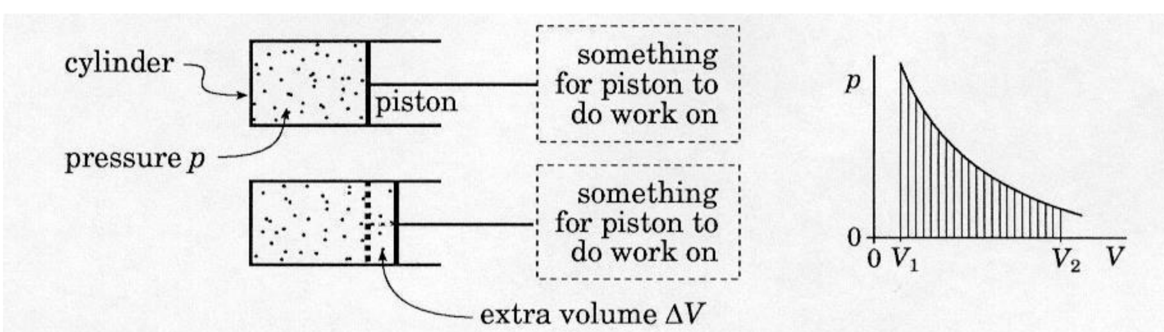
The quantity which is dependent on the process or path followed by the system is known as path function. Ex: Heat transfer, Work transfer.




**WORK DONE BY A GAS**

Let us consider a closed system where a part of the boundary is allowed to move under such conditions that the external restraining force is infinitesimally smaller than the force produced by the pressure of the system. The area of the piston is A and the pressure of the fluid at any instant is p. If p is assumed to be constant during an infinitesimal movement of the piston over a distance dl, the work done by the fluid in moving the external force pA through this distance is pA dl. But A.dl is dV, the infinitesimal change of volume, therefore

$$dW = pdV$$




Force acting on the piston = Pressure X Area

= pA

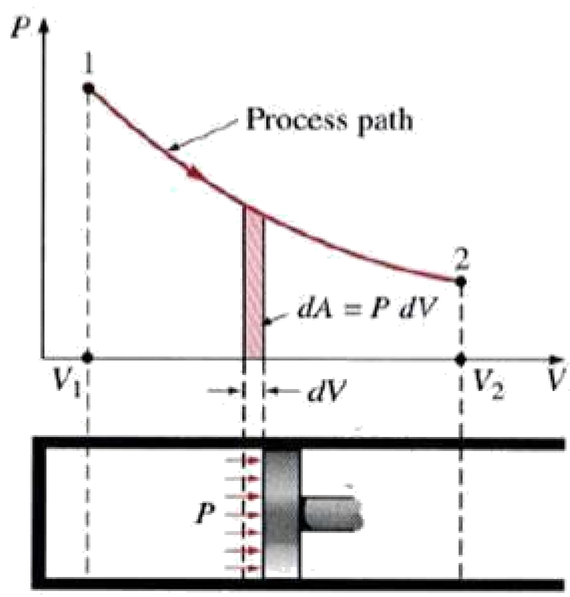
Therefore, Work done= Force X distance

= pA X dx

= pdV

where dV - change in volume.

if the expansion occurs from pressure p1 to a pressure p2 in such a way that the restraining force is changed continuously, then the total work done can be found out by summing up all the increments of work p dV, i.e. W = ∫ p dV




The area under the process curve on a *P*-*V* diagram represents the boundary work.

## Work done by a rubber band

When we increase the length of a rubber band by a small length $\Delta x$ by applying a pull $F$, then: Work done *on* band = $F\Delta x$

We could equally well write this as

Work done *by* band = $-F\Delta x$

To see the point of this, suppose we allow the band to contract, exerting a pull $F$. In this case $\Delta x$ is negative so the work done by the band is positive, which makes sense.

If we stretch the band a lot, then $F$ will change significantly during the stretching. In this case we add together all the bits of work, $F\Delta x$, which means adding together the areas of all the narrow strips under the graph, from the initial extension $x_1$ up to the final extension, $x_2$. So:

Work done on (or by) band = area under Force-extension graph

A positive amount of work is done *on* the band if $x$ is increasing; a positive amount of work is done *by* the band if $x$ is decreasing.

When a gas exerting a pressure $p$ expands by a small volume $\Delta V$, then: Work done by gas = $p\Delta V$

[This is, in fact, just a more convenient way of writing $F\Delta x$, in which $F$ is the force on the piston and $\Delta x$ is the distance it moves.]

We could equally well write

Work done on gas = $-p\Delta V$

If we push the piston in a little way $\Delta V$ is negative so a positive amount of work is done on the gas – as expected, since we‘ve had to do the pushing .If the gas expands a lot, then $p$ will change significantly. In this case we add together all the bits of work, $p\Delta V$, which means adding together the areas of all the narrow strips under the graph (see diagram), from the initial volume $V_1$ up to the final volume, $V_2$. So:

Work done by gas = area under pressure - volume graph

**A positive amount of work is done by the gas if V is increasing; a positive amount of work is done on the gas if V is decreasing.**

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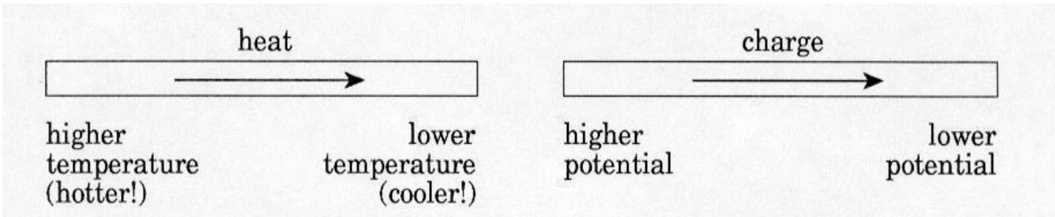
## Heat Energy

Heat is energy flowing from a region of higher temperature to a region of lower temperature because of the temperature difference.




Compare with the flow of charge due to a potential difference…

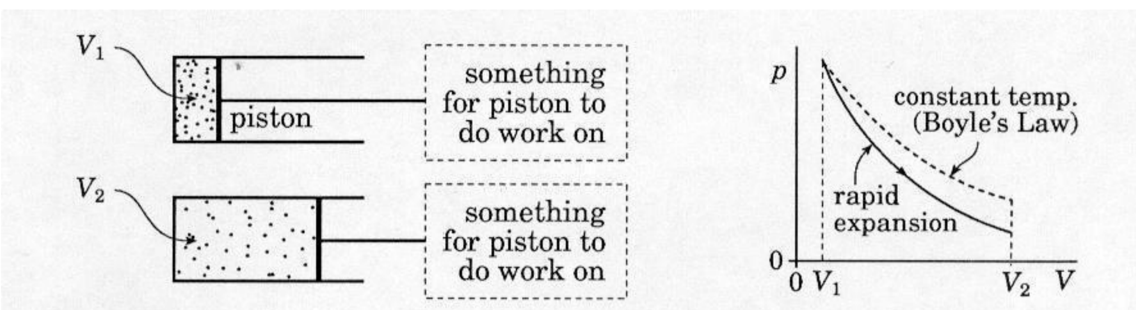
The formula used to calculate heat transfer; $Q = m\, C_v(T_2-T_1)$

Where Q= heat transfer
m= mass of the body
$T_2-T_1$ = Temperature Difference

If there is no temperature difference, no heat will flow. [In the electrical case, if there is no potential difference, no charge will flow.]

It takes time for a finite amount of heat to flow, though the rate of flow is greater the greater the temperature difference (other things being the same).

**Example 1: Rapid expansion of an ideal gas**




| Gain in system's internal energy | = | Net heat flowing into system | + Net work done on system |
|---|---|---|---|

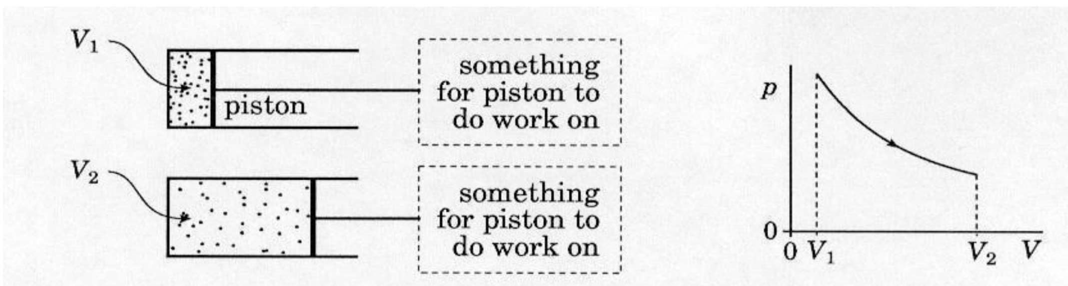
Heat-If the expansion is really rapid the heat flow will be (almost) zero.

Work- As the gas expands work is done by the gas. The last term in the equation is negative. Another way of seeing this is to write the equation as:

Gain in system‘s internal energy = Net heat flowing into system _ Net work done by system

Whichever way we look at it, because the heat term is zero, the right hand side of the equation is negative, so the gain in internal energy of the gas is negative, i.e. the internal energy decreases.

**Example 2: Slow expansion of an ideal gas**




Here it takes time to flow of heat. For each small increase in volume the gas temperature will drop a little and heat will flow in from the surroundings – limiting further temperature drop. If the expansion is really slow (and the cylinder walls conduct heat well) the temperature drop is negligible, so the expansion is isothermal.

The energy stored by the gas is internal energy. It isn‘t heat and it isn‘t work. Its decrease could result in heat being given out or in work being done (or both). Heat and work are both energy in transit.

$\Delta U$ is a change in a property of the system, its internal energy, $U$. A positive value of $\Delta U$ means an increase in $U$; a negative value means a decrease in $U$.

Q is heat entering the system from (hotter) surroundings. A negative value of $Q$ means heat leaving the system (to cooler surroundings). We don‘t have a ‗Δ‘ in front of $Q$, because heat flow is not a change in heat. It‘s energy in transit. It is not a function of the system‘s state.

W is work done by the system. A negative value of $W$ means a positive amount of work done on the system. We don‘t have a ‗Δ‘ in front of $W$ for the same reason as for $Q$; namely, $W$ is not a change in work. It‘s energy in transit. Work is not a function of the system‘s state.

## Specific heat capacity at constant volume and at constant pressure

Specific heat at constant volume of a substance is the amount of heat added to rise the temperature of unit mass of the given substance by 1 degree at constant volume

From first law for a stationary closed system undergoing a process $dQ = pdV + dU$ or $dq = pdv + du$

For a constant volume process

$$dQ = dU \text{ or } dq = du$$

or $du = C_V dT$

Similarly specific heat at constant pressure is the quantity of heat added to rise the temperature of unit mass of the given substance by 1 degree at constant pressure

$$\text{where } dQ = pdV + dU$$
$$= pdV + d(H - PV)$$
$$dQ = pdV + dH - Vdp - pdV$$
$$dQ = dH - Vdp$$

For a constant pressure process $dp = 0$ Hence $dQ = dH$ or $dq = dh$

or $dh = C_p dT$

For solids and liquids, constant volume and constant pressure processes are identical and hence, there will be only one specific heat.

The difference in specific heats $C_p - C_V = R$

The ratio of sp. heat $C_p/C_v =$

Since h and u are properties of a system, $dh = C_p dT$ and $du = C_V dT$, for all processes.

## FIRST LAW OF THERMODYNAMICS

First law of thermodynamics states that when system undergoes a cyclic process, net heat transfer is equal to work transfer.

In a cyclic process the system is taken through a series of processes and finally returned to its original state. The end state of a cyclic process is identical with the state of the system at the beginning of the cycle. This is possible if the energy level at the beginning and end of the cyclic process are also the same. In other words, the net energy change in a cyclic process is zero.

Consider a system undergoing a cycle consisting of two processes A & B .Net energy change

$$\Delta E_A + \Delta E_B = 0$$

$$(Q_A - W_A) + (Q_B - W_B) = 0$$

$$Q_A - Q_B = W_A - W_B$$

(or) $$\oint dQ = \oint dW$$

Hence for a cyclic process algebraic sum of heat transfers is equal to the algebraic sum of work transfer.

**First Law of Thermodynamics for a Change of State**

The expression $(\Sigma\delta Q)$ cycle = $(\Sigma\delta W)$cycle applies to a system undergoing a cycle. But if the system undergoes a change of state during which both heat and work transfers are involved, the net energy transfer will be stored or accumulated within the system. If Q is the amount of heat transferred to the system and W is the amount of work transferred from the system during the process, the net energy transfer (Q-W) will be stored in the system. This stored energy is known as internal energy or simply energy of the system. Hence, the First Law for a change of state is the net energy transfer during a process involving heat and work transfers is equal to the change in increase in energy of the system.ie $Q - W = \Delta E$

**COROLLARIES FIRST LAW OF THERMODYNAMICS.**

**Corollaries I**

There exists a property of a closed system such that a change in its value is equal to the difference between the heat supplied and the work done during any change of state.

**Corollaries II**

The internal energy of a closed system remains unchanged system is isolated from its surrounding.

**Corollaries III**

A perpetual motion machine of first kind is impossible.

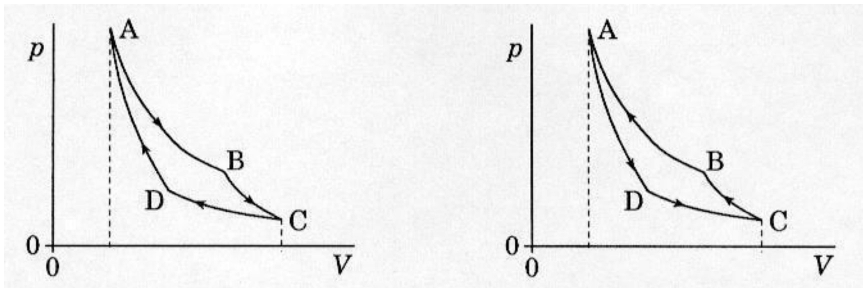
In an isolated system, there is no change in internal energy.

For any isolated system, there is no heat, work and mass transfer. $Q = W = 0$

According to the first law of thermodynamics,

$$Q = W + \Delta U$$
$$\Delta U = 0$$

## CYCLIC PROCESSES




Suppose an ideal gas is taken through the cycle of changes ABCDA shown above on the left. Irrespective of exactly what is going on in the individual stages (AB, BC, CD, DA), or the exact shapes of the curves, we can draw some general conclusions…

When the gas has undergone one cycle and is back at A, its internal energy is the same as it was originally, that is $\Delta U = 0$. This is because internal energy is a function of state, and the gas is back in the same state as originally.

Over the cycle as a whole, the gas has done a positive amount of work. This is because the area under ABC represents the gas doing a positive amount of work. The area under CDA represents work being done on the gas, but this area is smaller.

Thus applying the First Law, we see that for the cycle as a whole, there has been a net flow of heat into the gas.

## THE INTERNAL ENERGY OF A SYSTEM

The internal energy, U, of a system is the sum of the potential and kinetic energies of its particles.

Internal energy of a gas is the energy stored in a gas due to its molecular interactions. It is also defined as the energy possessed by a gas at a given temperature

There are two ways, then, that the total energy inside a system may change—heat and/or work. We use the term internal energy for the total energy inside a system, and the symbol U. Q and W will stand for heat and work, respectively. Energy conservation gives us the First ―Law‖ of Thermodynamics:

$$\Delta U = Q + W$$

.

Now, we have to be careful with the algebraic signs. In this case, Q is positive as the heat entering the system and W is positive as the work done on the system. So a positive Q and a positive W both cause an increase of internal energy, U.

## ENTHALPY OF THE SYSTEM

The total energy required to create a system of particles at sea level air pressure would include the expansive work done in displacing the air. It is sum of internal energy and flow energy.

We define the enthalpy to be $H \equiv U + PV$. The enthalpy is useful when a change takes place in a system while pressure is constant.

$$\Delta H = \Delta U + P\Delta V = Q + W + P\Delta V = Q + (-P\Delta V) + W_{other} + P\Delta V$$

$$\Delta H = Q + W_{other}$$

Now, if no other work is done, then $\Delta H = Q$ exactly. In practice, tables of measured enthalpies for various processes, usually chemical reactions or phase transitions are compiled. The text mentions the enthalpy of formation for liquid water. Evidently, when oxygen and hydrogen gases are combined to form a mole of liquid water, the change in enthalpy is -286 kJ. In other words, burning hydrogen at constant pressure releases this much energy.

$$dW = -p\,dV$$

We define $H \equiv U + pV$ as the *enthalpy* of the system, and $h = u + pv$ is the specific enthalpy. In particular, for a constant pressure process,

$$\Delta Q = \Delta H$$

## TERMODYNAMICS PROCESSESS

## CONSTANT VOLUME PROCESS

For a constant volume process, work can only be done by some method of churning the liquid. There cannot be any $\int pdV$ work as no external force has been moved through a distance. Hence, W must be zero or negative. For a constant volume process, unless otherwise stated, work done is taken as zero. Thus energy equation for a constant volume process is usually written as

$$Q = u_2 - u_1$$

$$W = 0,\ Q = \Delta u$$

If, in addition to work being zero, it is stipulated that the heat is transferred by virtue of an infinitesimally small temperature difference, then the process is reversible and the equation can be written in differential form as $dQ = du$

## CONSTANT PRESSURE PROCESS

A closed system undergoing at constant process is shown in Fig. The fluid is enclosed in a cylinder by a piston on which rests a constant weight. If heat is supplied, the fluid expands and work is being done by the system in overcoming the constant force; if the heat is extracted, the fluid contracts and work is done on the system by the constant force. In the general case Q – W = ($u_2$ – $u_1$)

If no paddle work is done on the system, and the process is

reversible, dQ – p dv = du

Since p is constant, this can be integrated to give

Q – p (v2 – v1) = (u2 – u1)

A further simplification is for constant pressure process if a new property is introduced

Since p is constant, p dv is identical to d(pv). Thus energy equation becomes

dQ – d(pv) = du (or)

dQ = d(u + pv) = dh

where

h = u + pv, known as enthalpy.

Since enthalpy is a combination of properties u, p and v, it itself is a

property. Hence h = u + pv

and for any mass of fluid m, in a state of equilibrium H =

U + pV

Using this derived property, the energy equation for a reversible constant pressure process becomes

dQ = dh (or) in the integrated

form Q = (h2 – h1)

Thus heat added in a reversible constant pressure process is equal to increase of enthalpy, whereas it is equal to increase of internal energy in the reversible constant

## ISOTHERMAL & ADIABATIC PROCESSES

Imagine a system in thermal contact with its environment. The environment is much larger than the system of interest, so that heat flows into or out of the system has no effect on the temperature of the environment. We speak of the system being in contact with a heat bath so that no matter what happens to the system, its temperature remains constant. If such a system is compressed slowly enough, its temperature is unchanged during the compression. The system is compressed isothermally. As an example, consider an ideal gas:

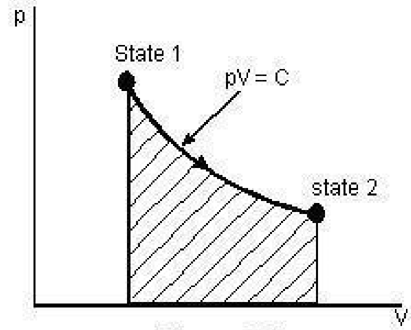



Figure 2.7

$$W_{1\,2} = \int_1^2 p\,dV$$

$$= \int_1^2 C\,dV \text{ where C=pV}$$

$$= c\int_1^2 \frac{1}{v}\,dV$$

$$= C \ln (V_2/V_1)$$

$$= p_1 v_1 \ln(V_2/V_1) \text{ (or) } p_2 v_2 \ln (V_2/V_1)$$

=

= On the other hand, the compression may be so fast that no heat is exchanged with the environment (or, the system is isolated from the environment) so that Q = 0. Such a process is adiabatic. Naturally, the temperature of the system will increase.

An isotherm is a curve of constant temperature, T, indicates the direction that the system is changing with time on the PV diagram. An arrow For an ideal gas, an isotherm is parabolic, since $P \propto V^1$ ; that's a special case. A curve along which Q = 0 is called an adiabatic. Work done for isothermal process; W= $P_1V_1 \ln(V_2/V_1)$ Work done for adiabatic process= $(P_1V_1-P_2V_2)/(\gamma-1)$

## POLYTROPIC PROCESS

Generally all processes are polytropic process, by changing the index of ‗n' either expansion or compression can get number of process. Then the process name is changed.

Any process can be represented by the general form $pV^n$ [redacted] of **n**, the process differs as given below; For other values of n, the process is known as polytropic process. Figure 2.8 shows the polytropic process of various possible polytropic index ‗n' on p-V coordinates. Expression for displacements work for a polytropic process can be obtained as follows:

$$1\ 2\ \int_1^2 PdV$$

$$= \int_1^2 \frac{C}{V^n}\, dV$$

where C = $pV^n$

$$= C\int_1^2 V^{-n}\, dV$$

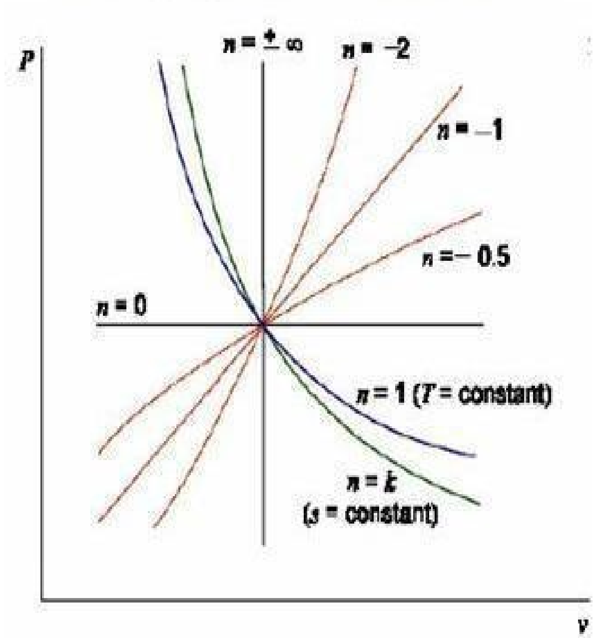
$$= C\left[\frac{V^{-n+1}}{-n+1}\right]_1^2$$

$$\left[\frac{CV_2^{-n+1} - CV_1^{-n+1}}{-n+1}\right]_1^2$$

$$= \left[\frac{p_2V_2^nV_2^{-n+1} - p_1V_1^nV_1^{-n+1}}{-n+1}\right] \quad \text{since } C = p_1V_1^n = p_2V_2^n$$

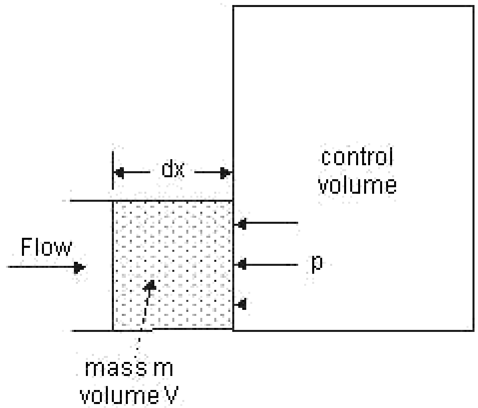
$$= \left[\frac{p_2V_2 - p_1V_1}{-n+1}\right]$$

*Isobaric process: P=const, n=0*
*Isothermal process: T=const, n=1*
*Isometric process: v=const, n=∞;*
*Isentropic process: s=const, n=k*




**APPLICATION OF FIRST LAW OF THERMODYNAMICS** The processes undergone in a closed system - Non-flow process The processes undergone in an open system - Flow process. **FLOW ENERGY PROCESS**

Flow energy is defined as the energy required to move a mass into the a control volume against a pressure. Consider a mass of volume V entering into a control volume as given in the Figure




The Flow energy = Work done in moving the mass

= Force X distance

= pA X dx

= p X (Adx)

= pV

Therefore, Enthalpy = Internal energy + Flow energy

**First Law of Thermodynamics for a Control Volume**

Mass simultaneously entering and leaving the system is a very common phenomenon in most of the engineering applications. Control volume concept is applied to these devices by assuming suitable control surfaces.

To analyze these control volume problems, conservation of mass and energy concepts are to be simultaneously considered.

Energy may cross the control surface not only in the form of heat and work but also by total energy associated with the mass crossing the boundaries. Hence apart from kinetic, potential and internal energies, flow energy should also be taken into account.

**Conservation of mass**

$$\begin{bmatrix} \textit{Total mass} \\ \textit{entering the} \\ \textit{control volume} \end{bmatrix} + \begin{bmatrix} \textit{Total mass} \\ \textit{leaving the} \\ \textit{control volume} \end{bmatrix} = \begin{bmatrix} \textit{Net change in the} \\ \textit{mass content of the} \\ \textit{control volume} \end{bmatrix}$$

SATHYABAMA

INSTITUTE OF SCIENCE AND TECHNOLOGY

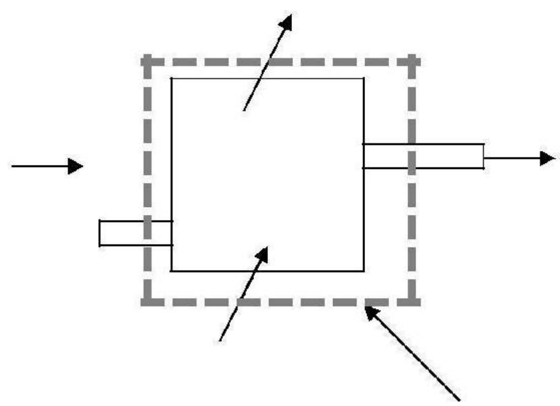
**DEPARTMENT OF Electrical and Electronics Engineering**

**COURSE MATERIAL**

**Subject Name: Applied Thermal Engineering** **UNIT I** **Subject Code: SME1208**

**Conservation of energy**

$$\begin{bmatrix} \textit{Net energy crossing the} \\ \textit{boundary in the} \\ \textit{form of heat} \\ \textit{and work} \end{bmatrix} + \begin{bmatrix} \textit{Total energy} \\ \textit{associated with the} \\ \textit{mass entering} \\ \textit{the control volume} \end{bmatrix} - \begin{bmatrix} \textit{Total energy} \\ \textit{associated with the} \\ \textit{mass leaving} \\ \textit{the control volume} \end{bmatrix} = \begin{bmatrix} \textit{Net change} \\ \textit{in the enrgy} \\ \textit{content of the} \\ \textit{control volume} \end{bmatrix}$$


Control Surface

As a rate equation, it becomes

$$\sum_{in} m \left[ h + \frac{C^2}{2} + Zg \right]_{in} - \sum_{out} m \left[ h + \frac{C^2}{2} + Zg \right]_{out} = [\Delta E]_{CV}$$

**The Steady-state Flow Process**

When a flow process is satisfying the following conditions, it is known as a steady flow process.

1. The mass and energy content of the control volume remains constant with time.
2. The state and energy of the fluid at inlet, at the exit and at every point within the control volume are time independent.
3. The rate of energy transfer in the form of work and heat across the control surface is constant with time.

Therefore for a steady flow process

$$\sum m_{in} = \sum m_{out}$$

Also

$$[\Delta E_{CV}] = 0$$

$$\sum_{in} m \left[ h + \frac{C^2}{2} + Zg \right]_{in} - \sum_{out} m \left[ h + \frac{C^2}{2} + Zg \right]_{out} = 0$$

For problem of single inlet stream and single outlet stream

$$\left[ (h_2 - h_1) + \left( \frac{C_2^2 - C_1^2}{2} \right) + (Z_2 - Z_1) g \right]$$

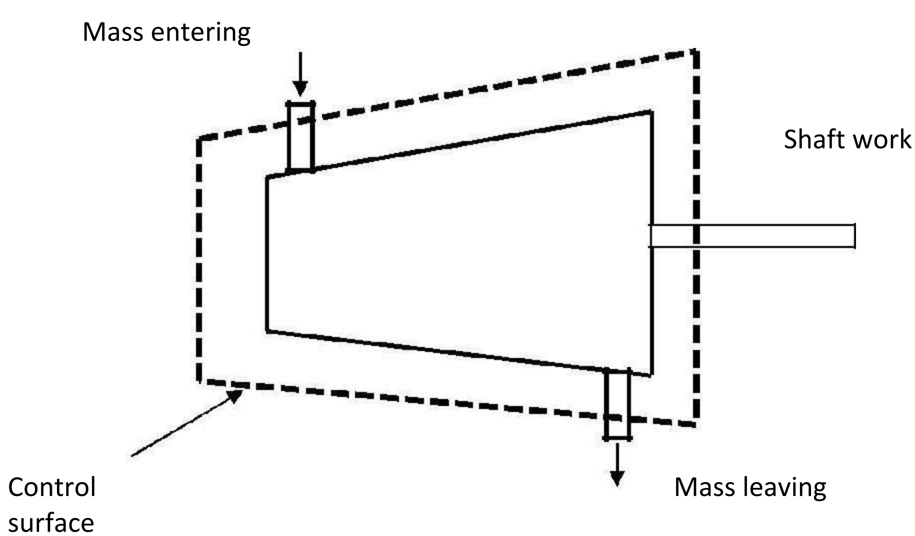
This equation is commonly known as steady flow energy equation (SFEE).

**Application of SFEE**

SFEE governs the working of a large number of components used in many engineering practices. In this section a brief analysis of such components working under steady flow conditions are given and the respective governing equations are obtained.

**Turbines**

Turbines are devices used in hydraulic, steam and gas turbine power plants. As the fluid passes through the turbine, work is done on the blades of the turbine which are attached to a shaft. Due to the work given to the blades, the turbine shaft rotates producing work.

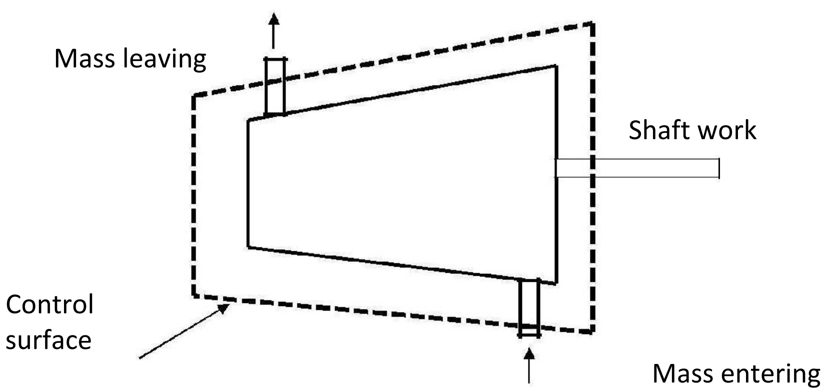


1. Changes in kinetic energy of the fluid are negligible 2.
Changes in potential energy of the fluid are negligible.

$$\left[Q - W\right] = m\left[(h_2 - h_1)\right]$$

## Compressors

Compressors (fans and blowers) are work consuming devices, where a low-pressure fluid is compressed by utilising mechanical work. Blades attached to the shaft of the turbine imparts kinetic energy to the fluid which is later converted into pressure energy.




## General Assumptions

1. Changes in the kinetic energy of the fluid are negligible
2. Changes in the potential energy of the fluid are negligible

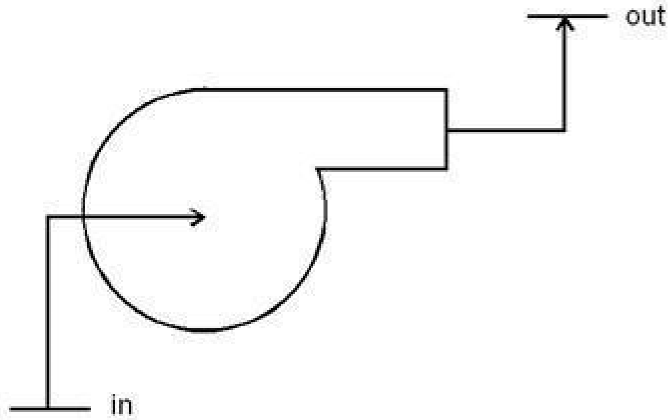
## Governing Equation

Applying the above equations SFEE becomes

$$[Q - W] = m[(h_2 - h_1)]$$

## Pumps

Similar to compressors pumps are also work consuming devices. But pumps handle incompressible fluids, whereas compressors deal with compressible fluids.




**General Assumptions**

1. No heat energy is gained or lost by the fluids;
2. Changes in kinetic energy of the fluid are negligible.

**Governing Equation**

$$[-W] = m[(h_2 - h_1) + (Z_2 - Z_1)g]$$

As the fluid passes through a pump, enthalpy of the fluid increases, (internal energy of the fluid remains constant) due to the increase in pv (flow energy). Increase in potential energy of fluid is the most important change found in almost all pump applications.

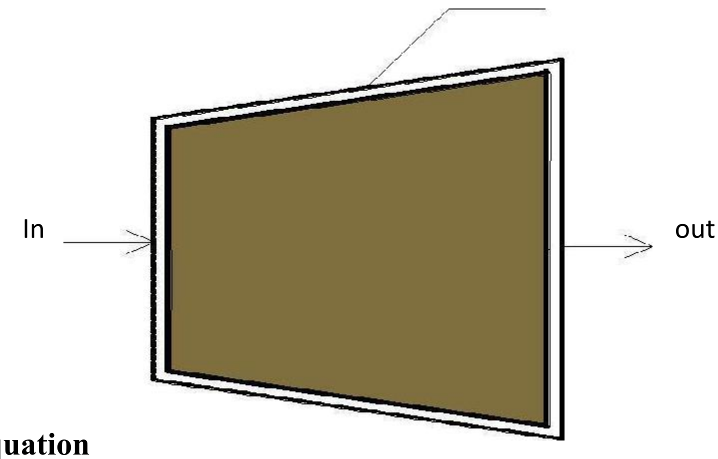
**Nozzles**

Nozzles are devices which increase the velocity of a fluid at the expense of pressure. A typical nozzle used for fluid flow at subsonic* speeds is shown in Figure.

**General Assumptions**

1. In nozzles fluids flow at a speed which is high enough to neglect heat lost or gained as it crosses the entire length of the nozzle. Therefore, flow through nozzles can be regarded as adiabatic. That is = 0.
2. There is no shaft or any other form of work transfer to fluid or from the fluid; that is = 0.
3. Changes in the potential energy of the fluid are negligible.

Control surface




**Governing Equation**

$$\left[ (h_2 - h_1) + \left( \frac{C_2^2 - C_1^2}{2} \right) \right] = 0$$

$$\left( \frac{C_2^2 - C_1^2}{2} \right) = (h_1 - h_2)$$

**Diffusers**

Diffusers are (reverse of nozzles) devices which increase the pressure of a fluid stream by reducing its kinetic energy.

**General Assumptions**

Similar to nozzles, the following assumptions hold good for diffusers.

1. Heat lost or gained as it crosses the entire length of the nozzle. Therefore, flow through nozzles can be regarded as adiabatic. That is $Q = 0$
2. There is no shaft or any other form of work transfer to the fluid or from the fluid; that is = 0.
3. Changes in the potential energy of the fluid are negligible

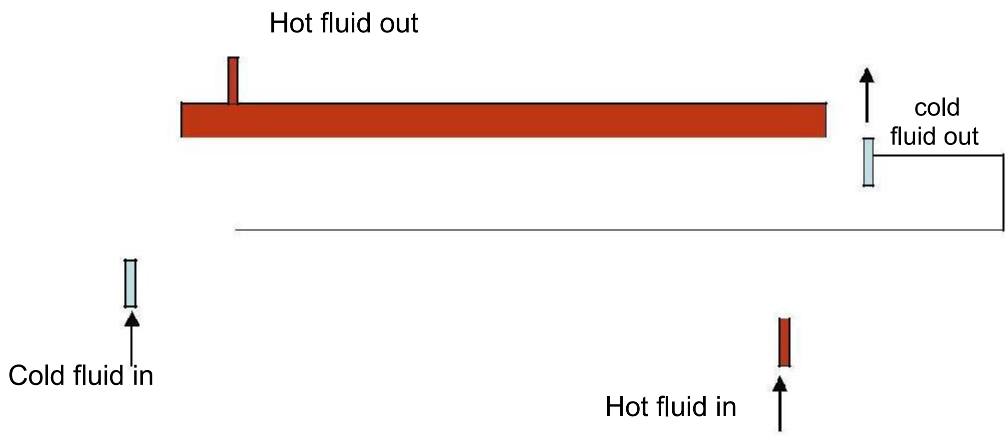
**Governing Equation**

$$\left[(h_2 - h_1) + \left(\frac{C_2^2 - C_1^2}{2}\right)\right] = 0$$

$$(h_2 - h_1) = \left(\frac{C_1^2 - C_2^2}{2}\right)$$

## Heat Exchangers

Devices in which heat is transferred from a hot fluid stream to a cold fluid stream are known as heat exchangers.




## General Assumptions

1. Heat lost by the hot fluid is equal to the heat gained by the cold fluid.
2. No work transfer across the control volume.
3. Changes in kinetic and potential energies of both the streams are negligible

## Governing Equation

For both hot and cold streams

$$- h )]$$

2 1

As per the assumption,

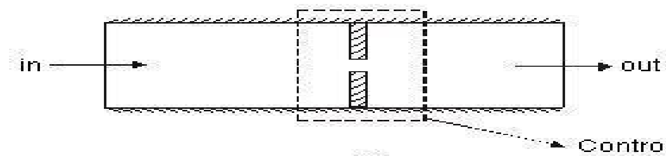
$$= \quad -Q_{hot} = Q_{cold}$$

The negative sign in the LHS is to represent that heat is going out of the system.

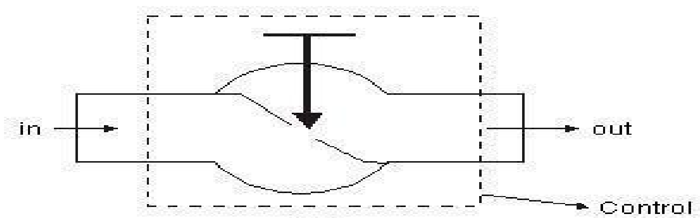
$$m_h(h_1 - h_2) = m_{c2}(h - h_1)$$

### Throttling

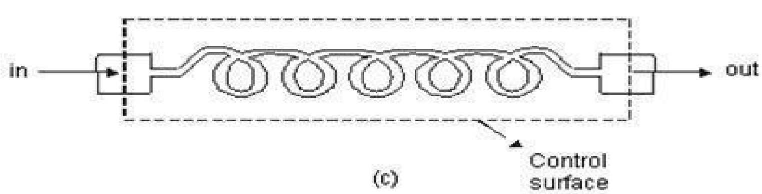
A throttling process occurs when a fluid flowing in a line suddenly encounters a restriction in the flow passage. It may be Plate with a small hole as shown in (a), Valve partially closed as shown in (b) Capillary tube which is normally found in a refrigerator as shown in (c)Porous plug as shown in (d)



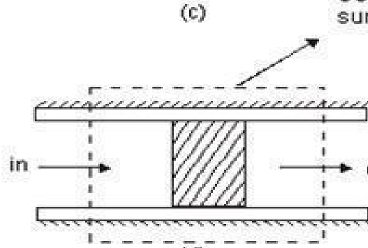
(a)



(b)



(c)



(d)

## QUESTIONS
## PART-A

1.What is meant by thermodynamics system? How do you classify it?
2. What is meant by closed system? Give an example.
3. Define open system. Give an example
4. When a system is said to be in‖ Thermodynamic equilibrium —?
5. Define Zeroth law and first law of thermodynamics.
6. State corollaries first law of thermodynamics.
7. Prove that for an isolated system, there is no change in internal energy.
8. What is meant by open and closed cycle?
9. What is meant by reversible and irreversible process?
10. What is meant by point and path function?

.

## PART -B

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

| Process | Q (kJ/min) | W (kJ/min) | ΔE (kJ/min) |
|---|---|---|---|
| a–b | 0 | 2,170 | — |
| b–c | 21,000 | 0 | — |
| c–d | –2,100 | — | –36,600 |
| d–a | — | — | — |

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

3. The internal energy of a certain substance is given by the following equation

$$U=3.56PV_{3}+84$$

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

(a) If the expansion is quasi static, find Q, $\Delta U$, and W for the process.

(b) In another process the same system expands according to the same pressure volume relationship as in part (a), and from the same initial state to the same final state as in part (a), but the heat transfer in this case is 30kJ. Find the work transfer for this process.
(C) Explain the difference in work transferin part (a) and (b)

4. Air flows steadily at the rate of 0.5kg/s through an air compressor, entering 7m/s velocity, 100kPa pressure and 0.95 $m^3$/kg volume, and leaving at 5m/s, 700k Pa and 0.19 $m^3$/kg. The internal energy of the air leaving is 90kJ/kg greater than that of the air entering. Cooling water in the compressor jacket absorbs heat from the air at the rate of 58kW. (a) Compute the rate of shaft work input to the air in kW. (b) Find the ratio of the inlet pipe diameter to outlet diameter

5. Air at a temperature of 15 $^0$C passes through a heat exchanger at a velocity of 30m/s where its temperature is raised to 800 $^0$C. It then enters a turbine with the same velocity of 30m/s and expands untill the temperature falls to 650 $^0$C. On leaving the turbine, the air is taken at a velocity of 60m/s to a nozzle where it expands untill the temperature has fallen to 500 $^0$C. If the air flow rate is 200kg/s, calculate

(a) the rate of heat transferto the air in the heat exchanger,
(b) the power output from the turbine assuming no heat loss and
(c) the velocity at exit from the nozzle, assuming no heat loss.

Take the enthalpy of the air as $h=C_pT$,
where $C_p$ is the specific heat equal to 1.005 kJ/kgK and T the temperature.

## Solutions

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

| *Process* | *Q (kJ/min)* | *W (kJ/min)* | *ΔE (kJ/min)* |
|---|---|---|---|
| a–b | 0 | 2,170 | — |
| b–c | 21,000 | 0 | — |
| c–d | –2,100 | — | –36,600 |
| d–a | — | — | — |

*Solution* Process *a–b*:

$$Q = \Delta E + W$$

$$0 = \Delta E + 2170$$

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

Process *b–c*:

$$Q = \Delta E + W$$

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

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

Process *c–d*:

$$Q = \Delta E + W$$

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

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

Process *d–a*:

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

The system completes 100 cycles/min.

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

$$0 + 21{,}000 - 2{,}100 + Q_{da} = -17{,}000$$

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

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

$\therefore$ $$\Delta E_{a-b} + \Delta E_{b-c} + \Delta E_{c-d} + \Delta E_{d-a} = 0$$

$$-2{,}170 + 21{,}000 - 36{,}600 + \Delta E_{d-a} = 0$$

$\therefore$ $$\Delta E_{d-a} = 17{,}770 \text{ kJ/min}$$

$\therefore$ $$W_{d-a} = Q_{d-a} - \Delta E_{d-a}$$

$$= -35{,}900 - 17{,}770$$

$$= -53{,}670 \text{ kJ/min}$$

The table becomes

| *Process* | $Q$ (kJ/min) | $W$ (kJ/min) | $\Delta E$ (kJ/min) |
|---|---|---|---|
| a–b | 0 | 2,170 | – 2,170 |
| b–c | 21,000 | 0 | 21,000 |
| c–d | – 2,100 | 34,500 | – 36,600 |
| d–a | – 35,900 | – 53,670 | 17,770 |

Since

Rate of work output $$\sum_{\text{cycle}} Q = \sum_{\text{cycle}} W$$

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

$$= -283.3 \text{ kW}$$

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

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

$$Q = \Delta U + W$$

or

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

Here

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

$$= 0.105\ (0.15 - 0.30)\ \text{MJ}$$

$$= -15.75\ \text{kJ}$$

$$Q_{1-2} = -37.6\ \text{kJ}$$

∴ Substituting in equation (1)

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

$$\therefore \qquad U_2 - U_1 = -21.85\ \text{kJ} \qquad \textit{Ans.}$$

The internal energy of the gas decreases by 21.85 kJ in the process.

3. The internal energy of a certain substance is given by the following equation U=3.56PV + 84 where U is given in kJ/kg, P is in kPa and V is in $m^3$/kg.

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

(a) If the expansion is quasi static, find Q, $\Delta U$, and W for the process.

(b) In another process the same system expands according to the same pressure volume relationship as in part (a), and from the same initial state to the same final state as in part (a), but the heat transfer in this case is 30kJ. Find the work transfer for this process.

(C) Explain the difference in work transferin part (a) and (b)

Solution: **(a)**

$$u = 3.56\,pv + 84$$

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

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

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

$\therefore$ $$V_2 = V_1\left(\frac{p_1}{p_2}\right)^{1/1.2} = 0.22\left(\frac{5}{1}\right)^{1/1.2}$$

$$= 0.22 \times 3.83 = 0.845 \text{ m}^3$$

$\therefore$ $$\Delta U = 356\,(1 \times 0.845 - 5 \times 0.22) \text{ kJ}$$

$$= -356 \times 0.255 = -91 \text{ kJ}$$

**(a)**

$$u = 3.56\,pv + 84$$

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

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

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

$\therefore$ $$V_2 = V_1\left(\frac{p_1}{p_2}\right)^{1/1.2} = 0.22\left(\frac{5}{1}\right)^{1/1.2}$$

$$= 0.22 \times 3.83 = 0.845 \text{ m}^3$$

$\therefore$ $$\Delta U = 356\,(1 \times 0.845 - 5 \times 0.22) \text{ kJ}$$

$$= -356 \times 0.255 = -91 \text{ kJ}$$

(b) Here $Q = 30$ kJ

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

$\therefore$ $$W = Q - \Delta U$$

$$= 30 - (-91)$$

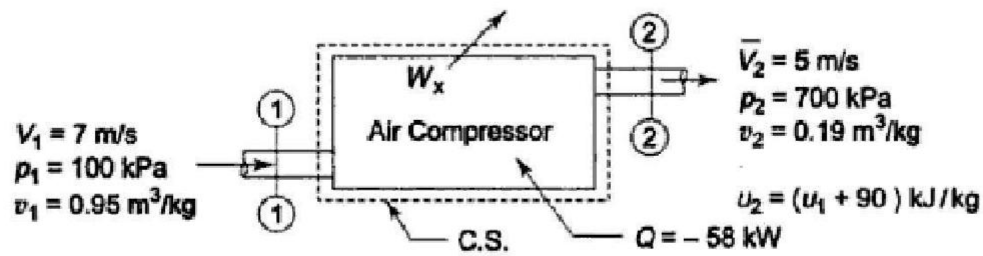
$$= 121 \text{ kJ}$$

*Ans.* (b)

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

4. Air flows steadily at the rate of 0.5kg/s through an air compressor, entering 7m/s velocity, 100kPa pressure and 0.95 $m^3$/kg volume, and leaving at 5m/s, 700k Pa and 0.19 $m^3$/kg. The internal energy of the air leaving is 90kJ/kg greater than that of the air entering. Cooling water in the compressor jacket absorbs heat from the air at the rate of 58kW. (a) Compute the rate of shaft work input to the air in kW. (b) Find the ratio of the inlet pipe diameter to outlet diameter

Solution: Figure shows the detail of the problem




(a) Writing the steady flow energy equation, we have

$$w\left(u_1 + p_1v_1 + \frac{V_1^2}{2} + Z_1g\right) + \frac{đQ}{d\tau}$$

$$= w\left(u_2 + p_2v_2 + \frac{V_2^2}{2} + Z_2g\right) + \frac{đW_x}{d\tau}$$

$$\therefore \frac{đW_x}{d\tau} = -w\left[(u_2 - u_1) + (p_2v_2 - p_1v_1) + \frac{V_2^2 - V_1^2}{2} + (Z_2 - Z_1)g\right] + \frac{đQ}{d\tau}$$

$$\therefore \quad \frac{đW_x}{d\tau} = -0.5\,\frac{kg}{s}\left[90\,\frac{kJ}{kg} + (7\times 0.19 - 1\times 0.95)100\,\frac{kJ}{kg} + \frac{(5^2 - 7^2)\times 10^{-3}}{2}\,\frac{kJ}{kg} + 0\right] - 58\text{ kW}$$

$$= -0.5\,[90 + 38 - 0.012]\text{ kJ/s} - 58\text{ kW}$$

$$= -122\text{ kW}$$

Rate of work input is 122 kW.

(b) From mass balance, we have

$$w = \frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2}$$

$$\therefore \quad \frac{A_1}{A_2} = \frac{v_1}{v_2} \cdot \frac{V_2}{V_1} = \frac{0.95}{0.19} \times \frac{5}{7} = 3.57$$

$$\therefore \quad \frac{d_1}{d_2} = \sqrt{3.57} = 1.89$$

5. Air at a temperature of $15^0$C passes through a heat exchanger at a velocity of 30m/s where its temperature is raised to $800^0$C. It then enters a turbine with the same velocity of 30m/s and expands untill the temperature falls to $650^0$C. On leaving the turbine, the air is taken at a velocity of 60m/s to a nozzle where it expands untill the temperature has fallen to $500^0$C. If the air flow rate is 200kg/s, calculate

(a) the rate of heat transferto the air in the heat exchanger,

(b) the power output from the turbine assuming no heat loss and

(c) the velocity at exit from the nozzle, assuming no heat loss. Take the enthalpy of the air as $h=C_pT$, where $C_p$ is the specific heat equal to 1.005 kJ/kgK and T the temperature.

Solution:

A shown in figure write the S.F.E.E for the heat exchanger and eliminating the terms not relevant

$$w\left(h_1 + \frac{V_1^2}{2} + Z_1 g\right) + Q_{1-2} = w\left(h_2 + \frac{V_2^2}{2} + Z_2 g\right) + W_{1-2}$$

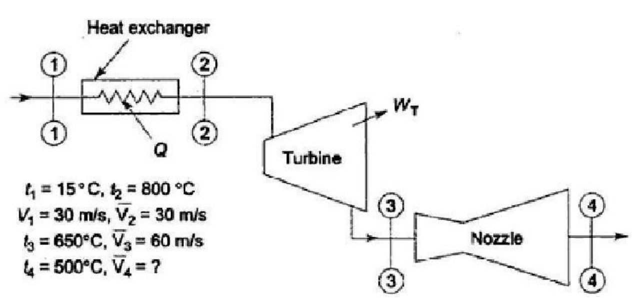
$$\therefore \quad wh_1 + Q_{1-2} = wh_2$$

$$\therefore \quad Q_{1-2} = w(h_2 - h_1) = wc_p(t_2 - t_1)$$

$$= 2 \times 1.005\,(800 - 15)$$

$$= 2.01 \times 785$$

$$= 1580 \text{ kJ/s}$$



Energy equation for the turbine gives

$$w\left(\frac{V_2^2}{2} + h_2\right) = wh_3 + w\frac{V_3^2}{2} + W_T$$

$$\frac{V_2^2 - V_3^2}{2} + (h_2 - h_3) = W_T/w$$

$$\frac{(30^2 - 60^2) \times 10^{-3}}{2} + 1.005\,(800 - 650) = W_T/w$$

$$\therefore \quad \frac{W_T}{w} = -1.35 + 150.75$$

$$= 149.4 \text{ kJ/kg}$$

$$\therefore \quad W_T = 149.4 \times 2 \text{ kJ/s}$$

$$= 298.8 \text{ kW}$$

## SECOND LAW OF THERMODYNAMICS - KELVIN'S AND CLAUSIUS STATEMENTS OF SECOND LAW. REVERSIBILITY AND IRREVERSIBILITY. CARNOT THEOREM, CARNOT CYCLE, REVERSED CARNOT CYCLE, EFFICIENCY, COP, CLAUSIUS INEQUALITY, CONCEPT OF ENTROPY, ENTROPY OF IDEAL GAS, PRINCIPLE OF INCREASE OF ENTROPY.

## INTRODUCTION

In this chapter the idea of cycle efficiency in introduced and the Second Law is then stated and distinguished from the First Law. Formal definition of a reversible process is made and its implications both for non-flow and steady-flow processes are discussed

A process is said to the reversible, it should trace the same path in the reverse direction when the process is reversed, and it is possible only when the system passes through a continuous series of equilibrium state if a system does not pass through continuous equilibrium state, then the system is said to be irreversible.

The direction of spontaneous change for a ball bouncing on a floor. On each bounce some of its potential energy is degraded into the thermal motion of the atoms of the floor, and that energy disperses into the atoms of the floor. The reverse has never been observed to take place. The reverse, if it occurs, does not violate the1st Law as long as the energy is conserved Recall also that only a small amount of thermal energy is required to make the ball jump very high. Hence, the first Law only states that the net work cannot be produced during a cycle without some supply of heat. However, First Law never says that some proportions of heat supplied to an engine must be rejected. Hence, as per the First Law, cycle efficiency can be unity, which is impossible in practice. All that First Law states that net work cannot be produced during a cycle without some supply of heat, i.e. that a perpetual motion machine of the first kind is impossible So, the 1st Law is not enough. Something is missing! What is missing? A law that can tell us about the direction of spontaneous change. The Second Law of Thermodynamics tells us about the directionality of the process. We need it to ensure that systems we design will work. As we will see later, ENTROPY is a property that we have invented (like internal energy) that will allow us to apply the 2nd Law quantitatively.

## REVERSIBLE AND IRREVERSIBLE PROCESSES

A process is said to the reversible, it should trace the same path in the reverse direction when the process is reversed, and it is possible only when the system passes through

a continuous series of equilibrium state if a system does not pass through continuous equilibrium state, then the system is said to be irreversible

In the course of this development, the idea of a completely reversible process is central, and we can recall the definition, ``a process is called completely reversible if, after the process has occurred, both the system and its surroundings can be wholly restored by any means to their respective initial states''. Especially, it is to be noted that the definition does not, in this form, specify that the reverse path must be identical with the forward path. If the initial states can be restored by any means whatever, the process is by definition completely reversible. If the paths are identical, then one usually calls the process (of the system) reversible, or one may say that the state of the system follows a reversible path. In this path (between two equilibrium states 1 and 2), (i) the system passes through the path followed by the equilibrium states only, and (ii) the system will take the reversed path 2 to 1 by a simple reversal of the work done and heat added.

Reversible processes are idealizations not actually encountered. However, they are clearly useful idealizations. For a process to be completely reversible, it is necessary that it be quasi-static and that there be no dissipative influences such as friction and diffusion. The precise (necessary and sufficient) condition to be satisfied if a process is to be reversible is the second part of the Second Law.

The criterion as to whether a process is completely reversible must be based on the initial and final states. In the form presented above, the Second Law furnishes a relation between the properties defining the two states, and thereby shows whether a natural process connecting the states is possible.

## SECOND LAW OF THERMODYNAMICS

Kelvin –Planck statement: It is impossible to construct an engine working on a cyclic process which converts all the heat energy supplied to it into equivalent amount of useful work.

Clausius statement: Heat cannot flow from cold reservoir to hot reservoir without any external aid. But heat can flow from hot reservoir to cold reservoir without any external aid.

## HEAT ENGINE, HEAT PUMP, REFRIGERATOR

**ENERGY RESERVOIRS**: Thermal energy reservoirs (TER) is defined as a large body of infinite heat capacity, which is capable of absorbing or rejecting an unlimited quantity of heat without suffering appreciable changes in its thermodynamic coordinates.

**SOURCE:** TER from which heat is transferred to the system operating in a heat engine cycle.

**SINK**: TER in which heat is rejected from the cycle during a cycle.

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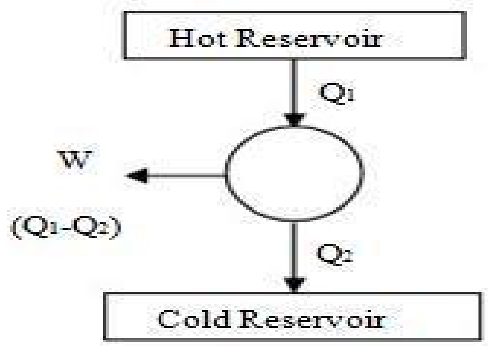
**DEPARTMENT OF Electrical and Electronics Engineering**

**COURSE MATERIAL**

**Subject Name: Applied Thermal Engineering** **UNIT II** **SUBJECTCODE: SME1208**

## HEAT ENGINE

A heat engine is a device which is used to convert the thermal energy into mechanical energy. Heat supplied is input work done is the output. Hence efficiency of the engine



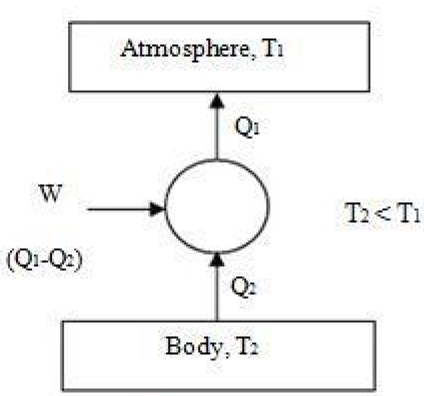

Heat engine

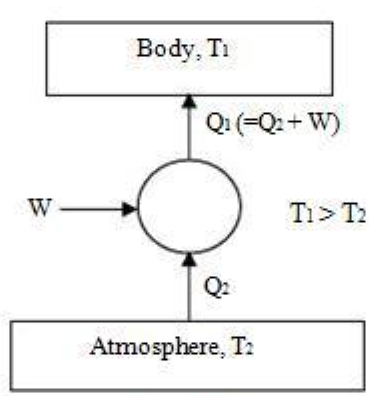
$$\text{Efficiency of the engine} = \frac{\text{Work output}}{\text{Heat supplied}}$$

€=(T1-T2)/T1

**REFRIGERATOR**: A device which operating in a cycle maintains a body at a temperature lower than the temperature of the surroundings

**HEAT PUMP**: Heat pump is a device which operating in a cycle process maintains the temperature of a hot body at a temperature of a hot body at a temperature higher that the temperature of surrounding. Different between heat pump and refrigeration states that heat pump is a device which operating in a cycle process maintains the temperature of a hot body at a temperature of a hot body at a temperature higher that the temperature of surrounding.



Coefficient of performance is defined as the ratio of heat of heat extracted of rejected to work input.

$$COP = \frac{\text{Heat extracted or rejected}}{\text{Work input}}$$

Expression for COP of heat pump and a refrigerator

COP for heat pump:

$$COPHP = \frac{\text{Heat rejected}}{\text{Work input}} = \frac{T2}{T2 - T1}$$

COP for refrigerator:

$$COP\ Ref = \frac{\text{Heat extracted}}{\text{Work input}} = \frac{T2}{T2 - T1}$$

The Second Law states that some heat must be rejected during the cycle and hence, the cycle efficiency is always less than unity Thus the First Law states that net work cannot be greater than heat supplied, while the Second Law goes further and states that it must be less than heat supplied. If energy is to be supplied to a system in the form of heat, the system must be in contact with a reservoir whose temperature is higher than that of the fluid at some point in the cycle. Similarly, if heat is to be rejected, the system must be at some time

be in contact with a reservoir of lower temperature than the fluid. Thus Second Law implies that if a system is to undergo a cycle and produce work, it must operate between two reservoirs of different temperatures. A machine which will work continuously, while exchanging heat with only single reservoirs, is known as a perpetual motion machine of the second kind (PMM II); such a machine contradicts Second Law. It is now possible to see why a ship could not be driven by an engine using the ocean as a source of heat, or why a power station could not be run using the atmosphere as a source of heat. They are impossible because there is no natural sink of heat at a lower temperature than the atmosphere or ocean, and they would therefore be PMM II.

It should be noted that Second Law does not restrict that work cannot be continuously and completely converted to heat. In fact, a fluid in a closed vessel may have work done on it and the heat thus generated is allowed to cross the boundary. The rates of work and heat may be made equal and the internal energy of the system remaining constant. An important consequence of

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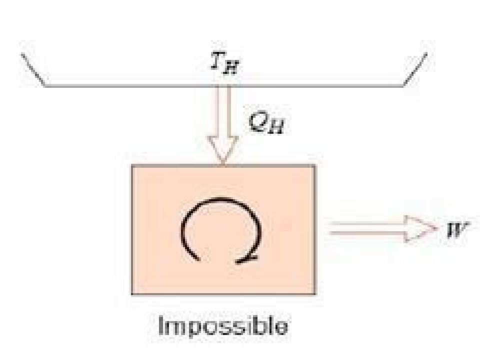
Second law is work is a more valuable form of energy transfer than heat as heat can never be transformed continuously and completely to work, whereas work can always be transformed continuously and completely to heat.

The following statements summarise the obvious consequences of the Second Law:
a) If a system is taken through a cycle and produces work, it must be exchanging heat with at least two reservoirs at different temperatures,
b) If a system is taken through a cycle while exchanging heat with one reservoir, the work done must be zero or negative,
c) Since heat can never be continuously and completely converted into work whereas work can always be continuously and completely converted into heat, work is more valuable form of energy transfer than heat.

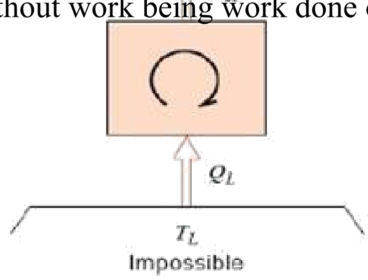
## THE KEIVIN-PLANK'S STATEMENT OF THE SECOND LAW

It is impossible to construct a system, It is impossible to construct a system, which will operate in a cycle, extract heat from a reservoir and do an equivalent



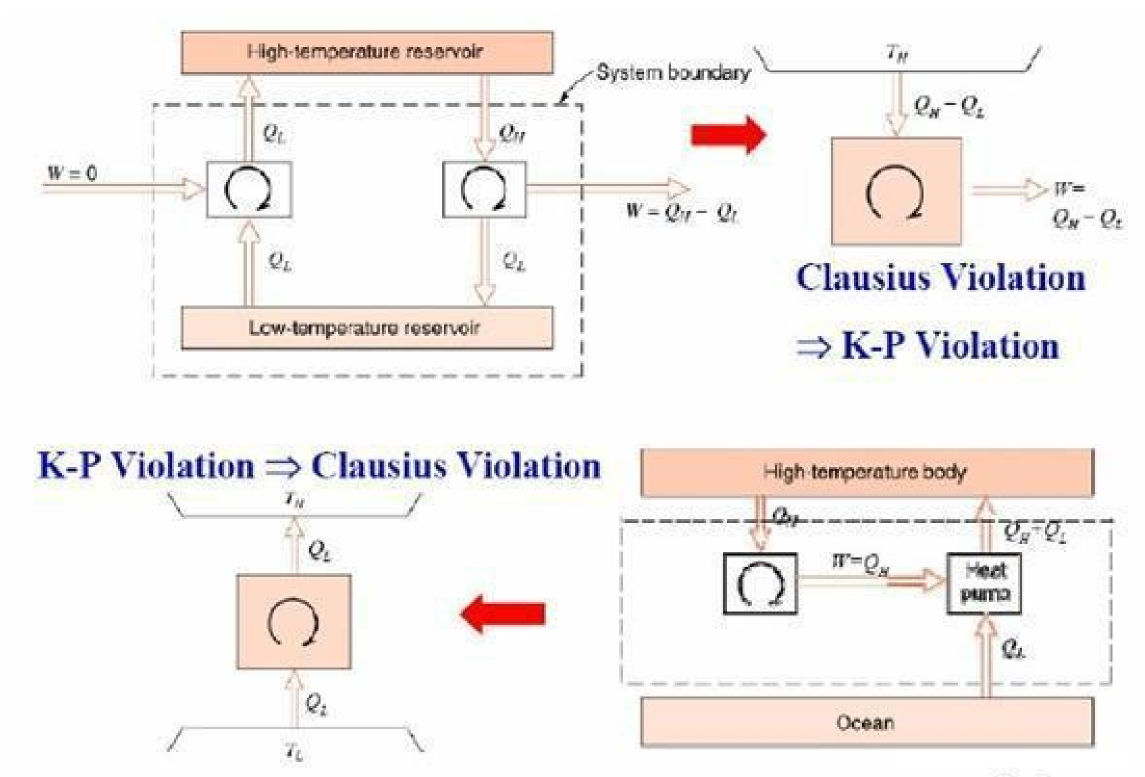

## THE CLAUSIUS STATEMENT OF THE SECOND LAW

It is impossible to construct a system, which will operate in a cycle and transfer heat from a cooler to a hotter body without work being work done on the system by the surrounding



## EQUIVALENCE KELVIN PLANK'S AND CLAUSIUS STATEMENTS.

Proof: Suppose the converse of the Clausius' proposition is true. The system can be represented by a heat pump for which W = 0. If it takes Q units of heat from the cold reservoir, it must deliver Q units to the hot reservoir to satisfy the First Law. A heat engine could also be operated between the two reservoirs; let it be of such a size that it delivers Q units of heat to the cold reservoir while performing W units of work. Then the First Law states that the engine must be supplied with (W + Q) units of heat from the hot reservoir. In the combined plant, the cold reservoir becomes superfluous because the heat engine could reject its heat directly to the heat pump. The combined plant represents a heat engine extracting (W + Q) – Q = W units of heat from a reservoir and delivering an equivalent amount of work. This is impossible according to Kelvin-Plank's statement of Second Law. Hence converse of Clausius' statement is not true and the original proposition must be true.



**CARNOT'S THEOREM**.

No heat engine operating in a cycle process between two fixed temperatures can be more efficient that a reversible engine operating between the same temperature limits.

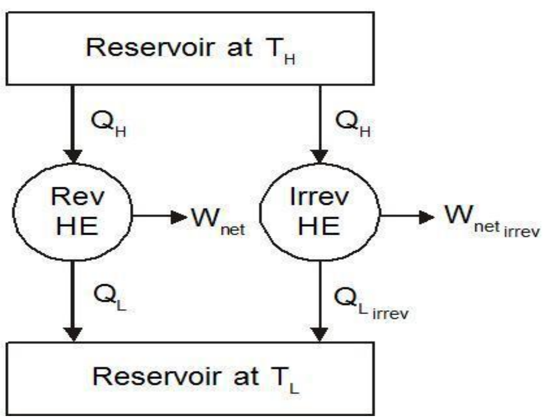
**COROLLARIES OF CORNOT THEOREM**

i. All the reversible engines operating between the two given thermal reservoir with fixed temperature have the same efficient.

ii.The efficient of any reversible heat engine operating between two reservoir is independent of the nature of the working fluid and depends only on the temperature of the reservoirs.

**CLAUSIUS INEQUALITY**

Consider two heat engines operating between two reservoirs kept at temperature $T_H$ and $T_L$ as shown in the Figure. Of the two heat engines, one is reversible and the other is irreversible.




For the reversible heat engine it has already been proved that

$$\frac{Q_H}{Q_L} = \frac{T_H}{T_L}$$

$$\frac{Q_H}{T_H} - \frac{Q_L}{T_L} = 0$$

$$\oint \left(\frac{dQ}{T}\right)_{rev} = 0$$

As discussed earlier, the work output from the irreversible engine should be less than that of the reversible engine for the same heat input $Q_H$. Therefore $Q_{L,Irrev}$ will be greater than $Q_{L,Rev}$. Let us define

$$Q_{L,Irrev} = Q_{L,Rev} + dQ$$

then

$$\oint \left(\frac{dQ}{T}\right)_{Irrev} = \frac{Q_H}{T_H} - \frac{Q_{L,Irev}}{T_L}$$

$$= \frac{Q_H}{T_H} - \frac{Q_{L,rev}}{T_L} - \frac{dQ}{T_L}$$

$$= 0 - \frac{dQ}{T_L}$$

$$< 0$$

By combining this result with that of a reversible engine we get

$$\oint \left(\frac{dQ}{T}\right)_{Irrev} \le 0$$

This is known as Clausius inequality.

The following conditions of clausius inequality

$\oint dQ/T \le 0$ is known as inequality of clausius.

If 1. $\oint dQ/T = 0$, the cycle is reversible.

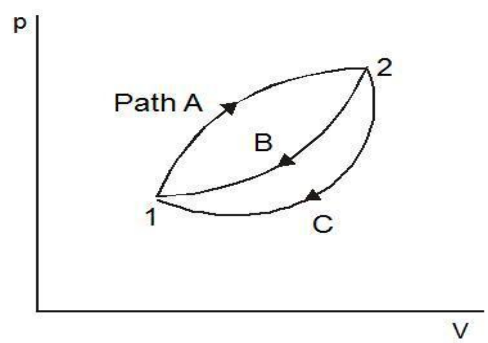
2. $\oint dQ/T < 0$, the cycle is irreversible and possible.

3. ƒ dQ/T > 0,the cycle is impossible.

## ENTROPY

The measure of irreversibility when the energy transfer takes place within the system or between system and surrounding is called as change of entropy. It is simply known as unaccounted heat loss.The change entropy of the system with respect to ambient conditions or any other standard reference conditions is known as absolute entropy.

Clausius inequality forms the basis for the definition of a new property known as entropy.




Consider a system taken from state 1 to state 2 along a reversible path A as shown in Figure. Let the system be brought back to the initial state 1 from state 2 along a reversible path B. Now the system has completed one cycle. Applying Clausius inequality we get

$$\oint \frac{dQ}{T} = 0$$

$$\int_1^2 \left(\frac{dQ}{T}\right)_A + \int_2^1 \left(\frac{dQ}{T}\right)_B = 0$$

Instead of taking the system from state2 to state1 along B, consider another reversible path C. Then for this cycle 1-A-2-C-1, applying Clausius inequality:

$$\oint \frac{dQ}{T} = 0$$

$$\int_1^2 \left(\frac{dQ}{T}\right)_A + \int_2^1 \left(\frac{dQ}{T}\right)_C = 0$$

Comparing, Hence, it can be concluded that the quantity is a point function, independent of the path followed. Therefore it is a property of the system

**Principle of increasing entropy**

Applying Clausius inequality,
For any infinitesimal process undergone by a system, change in

entropy $dS \geq dQ/T$
For reversible, $dQ = 0$ hence $dS = 0$
For irreversible, $dS > 0$

Consider a system interacting with its surroundings. Let the system and its surroundings are included in a boundary forming an isolated system. Since all the reactions are taking place within the combined system, whenever a process occurs entropy of the universe (System plus surroundings) will increase if it is irreversible and remain constant if it is reversible. Since all the processes in practice are irreversible, entropy of universe always increases

ie., $(\Delta s)_{universe} > 0$

**QUESTION**
**PART -A**

1.State the Kelvin –Planck statement of second law of Thermodynamics.
2. State the Clausius statement of second law of Thermodynamics
3. Write the two statement of second law of Thermodynamics

4. State Cornot's Theorem.
5. What are the corollaries of cornot theorem?
6. Define – PMM of second kind?
7. What is different between heat pump and refrigeration?
8. What is meant by heat engine?
9. Define the term COP?
10. Define change of entropy .How is entropy compared with heat transfer and absolute temperature?
11. Define the term source, sink and heat reservoir.
12. Why the performance of refrigerator and heat pump are given in terms of C.O.P and not in terms of efficiency?
13. What is meant by principle of increase of entropy?
14. What do you mean by "clausius inequality"?

## PART-B

1.A cyclic heat engine operates between a source temperature of $800^0$C and a sink temperature of $30^0$C. What is the least rate of heat rejection per kW net output of the engine?

2. A reversible heat engine operates between two reservoirs at temperatures of $600^0$C and $40^0$C. The engine drives a reversible refrigerator which operates between reservoirs at temperatures of $40^0$C and $-20^0$C. The heat transfer to the heat engine is 2000kJ and the net work output of the combined engine transfer plant is 360kJ.

(a) Evaluate the heat transfer to the refrigerant and the net heat transfer to the reservoir at $40^0$C.

(b)Reconsider (a) given that the efficiency of the heat engine and the COP of the refrigerator are each 40% of their maximum possible values.

3. It is proposed that solar energy be used to warm a large collector plate. This energy would, in turn, be transferred as heat to a fluid within a heat engine, and engine would reject energy as heat to the atmosphere. Experiments indicate that about 1880 kJ/$m^2$ h of energy can be collected when the plate is operating at $90^0$C. Estimate the minimum collector area that would be required for a plant producing 1kW of useful shaft power. The atmospheric temperature may be assumed to be $20^0$C.

4. One kg of ice at -50$^0$C is exposed to the atmosphere which is at 20$^0$C. The ice melts and comes into thermal equilibrium with the atmosphere. (a) Determine the entropy increases of the universe . (b) what is the minimum amount of work necessary to convert the water back into ice ar -5$^0$C? $C_p$ of ice is 2.093 kJ/kgK and the latent heat of fusion of ice is 333.3 kJ/kg.

5. A fluid undergoes a reversible adiabatic compression from 0.5 MPa, 0.2 $m^3$ to 0.05$m^3$ $^{1.3}$=constant.
Determine the change in enthalpy, internal energy andaccordingtothelaw,pv
entropy, and the heat transfer and work transfer during the process.

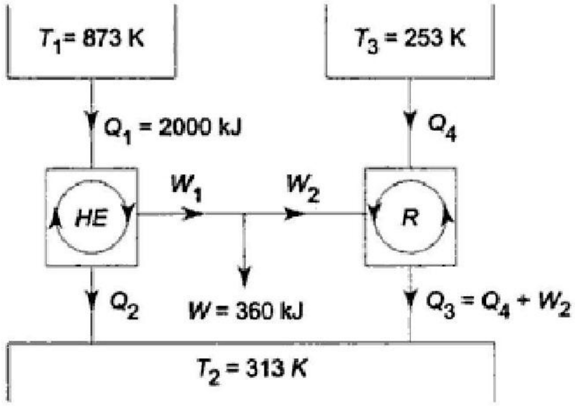
6. Calculate the available energy in 40kg of water at 75$^0$C with respect to the surrounding at 500 C, the pressure of water being 1atm.

7. Air enters a compressor at 1 bar, 30$^0$C, which is also the state of environment. It leaves at 3.5 bar, 141$^0$ and 90m/s. Neglecting inlet velocity and P.E. effect, determine (a) whether the compression is adiabatic or polytropic, (b) If not adiabatic, the polytropic index, (c) the isothermal efficiency, (d) the minimum work input and irreversibiity and (e) second law efficiency. Take $C_p$ of air =1.0035kJ/kgK

## Solutions

1.A cyclic heat engine operates between a source temperature of 800$^0$C and a sink temperature of 30$^0$C. What is the least rate of heat rejection per kW net output of the engine?

Solution: For a reversible engine, the rate of heat rejection will be minimum Solution: (a) Maximum efficiency of the heat engine cycle is given by




$$\eta_{max} = 1 - \frac{T_2}{T_1} = 1 - \frac{313}{873} = 1 - 0.358 = 0.642$$

Again $\frac{W_1}{Q_1} = 0.642$

$\therefore$ $W_1 = 0.642 \times 2000 = 1284$ kJ

Maximum COP of the refrigerator cycle

$$(\text{COP})_{max} = \frac{T_3}{T_2 - T_3} = \frac{253}{313 - 253} = 4.22$$

Also $\text{COP} = \frac{Q_4}{W_2} = 4.22$

Since $W_1 - W_2 = W = 360$ kJ

$\therefore$ $W_2 = W_1 - W = 1284 - 360 = 924$ kJ

$\therefore$ $Q_4 = 4.22 \times 924 = 3899$ kJ

$\therefore$ $Q_3 = Q_4 + W_2 = 924 + 3899 = 4823$ kJ

$Q_2 = Q_1 - W_1 = 2000 - 1284 = 716$ kJ

Heat rejection to the 40°C reservoir

$= Q_2 + Q_3 = 716 + 4823 = 5539$ kJ *Ans.* (a)

(b) Efficiency of the actual heat engine cycle

$\eta = 0.4\ \eta_{max} = 0.4 \times 0.642$

$\therefore$ $W_1 = 0.4 \times 0.642 \times 2000$

$= 513.6$ kJ

$\therefore$ $W_2 = 513.6 - 360 = 153.6$ kJ

COP of the actual refrigerator cycle

$$COP = \frac{Q_4}{W_2} = 0.4 \times 4.22 = 1.69$$

Therefore

$$Q_4 = 153.6 \times 1.69 = 259.6 \text{ kJ} \qquad \textit{Ans.} \text{ (b)}$$
$$Q_3 = 259.6 + 153.6 = 413.2 \text{ kJ}$$
$$Q_2 = Q_1 - W_1 = 2000 - 513.6 = 1486.4 \text{ kJ}$$

Heat rejected to the 40°C reservoir

$$= Q_2 + Q_3 = 413.2 + 1486.4 = 1899.6 \text{ kJ} \qquad \textit{Ans.} \text{ (b)}$$

. It is proposed that solar energy be used to warm a large collector plate. This energy would, in turn, be transferred as heat to a fluid within a heat engine, and engine would reject energy as heat to the atmosphere. Experiments indicate that about 1880 kJ/m$^2$ h of energy can be collected when the plate is operating at 90$^0$C. Estimate the minimum collector area that would be required for a plant producing 1kW of useful shaft power. The atmospheric temperature may be assumed to be 20$^0$C.

Solution: The maximum efficiency for the heat engine operating between the collector plate temperature and the atmospheric temperature is

$$\eta_{max} = 1 - \frac{T_2}{T_1} = 1 - \frac{293}{363} = 0.192$$

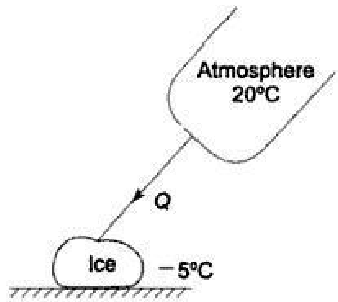
The efficiency of any actual heat engine operating between these temperatures would be less than this efficiency.

$$\therefore \qquad Q_{min} = \frac{W}{\eta_{max}} = \frac{1 \text{ kJ/s}}{0.192} = 5.21 \text{ kJ/s}$$
$$= 18{,}800 \text{ kJ/h}$$

$\therefore$ Minimum area required for the collector plate

$$= \frac{18{,}800}{1880} = 10 \text{ m}^2 \qquad \textit{Ans.}$$

4. One kg of ice at -50$^0$C is exposed to the atmosphere which is at 20$^0$C. The ice melts and comes into thermal equilibrium with the atmosphere. (a) Determine the entropy increases of the universe . (b) what is the minimum amount of work necessary to convert the water back into ice ar -5$^0$C? Cp of ice is 2.093 kJ/kgK and the latent heat of fusion of ice is 333.3 kJ/kg.




= Heat absorbed in solid phase + Latent heat
+ Heat absorbed in liquid phase
$= 1 \times 2.093 \times [0 - (-5)] + 1 \times 333.3 + 1 \times 4.187 \times (20 - 0)$
$= 427.5$ kJ

Entropy change of the atmospher.

$$(\Delta S)_{atm} = -\frac{Q}{T} = -\frac{427.5}{293} = -1.46 \text{ kJ/K}$$

Entropy change of the system (ice) as it gets heated from −5°C to 0°C

$$(\Delta S_I)_{system} = \int_{268}^{273} mc_p \frac{dT}{T} = 1 \times 2.093 \ln \frac{273}{268} = 2.093 \times 0.0186$$
$$= 0.0389 \text{ kJ/K}$$

Entropy change of the system as ice melts at 0°C to become water at 0°C

$$(\Delta S_{II})_{system} = \frac{333.3}{273} = 1.22 \text{ kJ/K}$$

Entropy change of water as it gets heated from 0°C to 20°C

$$(\Delta S_{III})_{system} = \int_{273}^{293} mc_p \frac{dT}{T} = 1 \times 4.187 \ln \frac{293}{273} = 0.296 \text{ kJ/K}$$

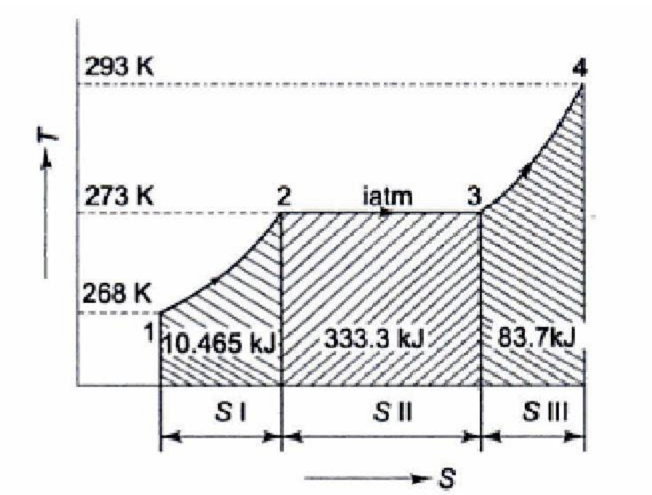
Total entropy change of ice as it melts into water

$$(\Delta S)_{total} = \Delta S_I + \Delta S_{II} + \Delta S_{III}$$
$$= 0.0389 + 1.22 + 0.296$$
$$= 1.5549 \text{ kJ/K}$$

The entropy-temperature diagram for the system at -5$^0$C converts to water at 20$^0$C is shown in fig.

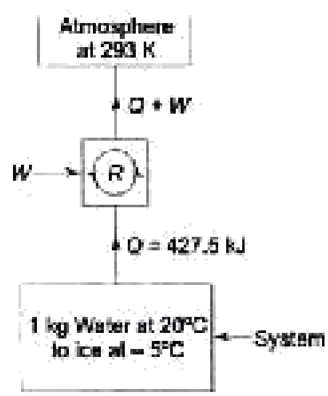
$\therefore$ **Entropy increase of the universe**

$$(\Delta S)_{univ} = (\Delta S)_{system} + (\Delta S)_{atm}$$
$$= 1.5549 - 1.46 = 0.0949 \text{ kJ/K} \qquad \textit{Ans.} \text{ (a)}$$




(b) To convert 1 kg of water at 20°C to ice at −5°C, 427.5 kJ of heat have to be removed from it, and the system has to be brought from state 4 to state 1 (Fig. Ex. 7.3.2). A refrigerator cycle, as shown in Fig. Ex. 7.3.3, is assumed to accomplish this.

The entropy change of the system would be the same, i.e. $S_4 - S_1$, with the only difference that its sign will be negative, because heat is removed from the system (Fig. Ex. 7.3.2).



$$(\Delta S)_{system} = \underset{\text{(negative)}}{S_1 - S_4}$$

The entropy change of the working fluid in the refrigerator would be zero, since it is operating in a cycle, i.e.,

$$(\Delta S)_{ref} = 0$$

The entropy change of the atmosphere (positive)

$$(\Delta S)_{atm} = \frac{Q + W}{T}$$

∴ Entropy change of the universe

$$(\Delta S)_{univ} = (\Delta S)_{system} + (\Delta S)_{ref} + (\Delta S)_{atm}$$

$$= (S_1 - S_4) + \frac{Q + W}{T}$$

By the principle of increase of entropy

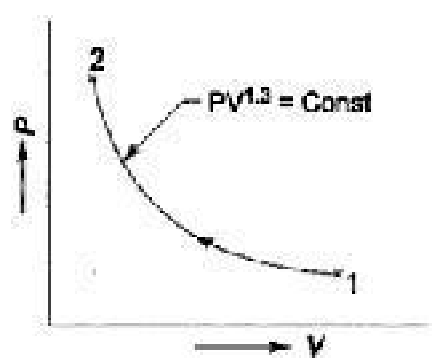
$$(\Delta S)_{\text{univ or isolated system}} \geq 0$$

$$\therefore \qquad \left[(S_1 - S_4) + \frac{Q + W}{T}\right] \geq 0$$

5. A fluid undergoes a reversible adiabatic compression from 0.5 MPa, 0.2 $m^3$ to 0.05$m^3$ according to the law, $pv^{1.3}$=constant. Determine the change in enthalpy, internal energy and entropy, and the heat transfer and work transfer during the process.

Solution:

$$TdS = dH - VdP$$



$$dH = Vdp$$

$$p_1 = 0.5 \text{ MPa}, V_1 = 0.2 \text{ m}^3$$

$$V_2 = 0.05 \text{ m}^3, p_1V_1^n = p_2V_2^n$$

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

$$= 0.5 \times \left(\frac{0.20}{0.05}\right)^{1.3} \text{ MPa}$$

$$= 0.5 \times 6.061 \text{ MPa}$$

$$= 3.0305 \text{ MPa}$$

$$p_1V_1^n = pV^n$$

$$\therefore \quad V = \left(\frac{p_1V_1^n}{p}\right)^{1/n}$$

$$\int_{H_1}^{H_2} dH = \int_{p_1}^{p_2} V dp$$

$$H_2 - H_1 = \int_{p_1}^{p_2}\left[\left(\frac{p_1V_1^n}{p}\right)^{1/n}\right]dp$$

$$= (p_1V_1^n)^{1/n}\left(\frac{p_1^{1-1/n} - p_1^{1-n/n}}{1 - 1/n}\right)$$

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

$$= \frac{1.3(3030.5 \times 0.05 - 500 \times 0.2)}{1.3 - 1}$$

$$= 223.3 \text{ kJ}$$

$$H_2 - H_1 = (U_2 + p_2V_2) - (U_1 + p_1V_1)$$

$$= (U_2 - U_1) + (p_2V_2 - p_1V_1)$$

$$U_2 - U_1 = (H_2 - H_1) - (p_2V_2 - p_1V_1)$$

$$= 223.3 - 51.53$$

$$= 171.77 \text{ kJ} \qquad \textit{Ans.}$$

$$S_2 - S_1 = 0 \qquad \textit{Ans.}$$

$$Q_{1-2} = 0 \qquad \textit{Ans.}$$

$$Q_{1-2} = U_2 - U_1 + W_{1-2}$$

$$W_{1-2} = U_1 - U_2 = -171.77 \text{ kJ} \qquad \textit{Ans.}$$

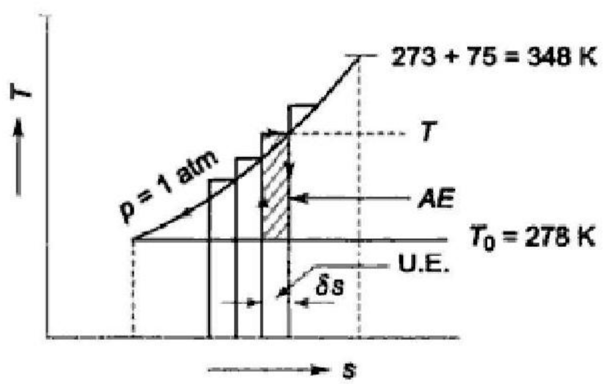
6. Calculate the available energy in 40kg of water at $75^0$C with respect to the surrounding at $50^0$C, the pressure of water being 1atm.

Solution:

If the water is cooled at a constant pressure of 1 atm from $75^0$C to $5^0$C (as shown in fig) the heat given up may be used as a source for a series of carnot engines each using the surrounding as a sink. It is assumed that the amount of energy received by any engine is small relative to that in the source and the temperature of the source doesnot change while heat is being exchanged with the engine.

Let us consider that the source has fallen to temperature T, at which level there operates a carnot engine which takes in heat at this temperature and rejects heat at $T_0$=278K. If del S is change in entropy water, the work obtainable is

$$\delta W = -m(T - T_0)\delta s$$




where $\delta s$ is negative.

$$\delta W = -40(T - T_0)\frac{c_p \delta T}{T}$$

$$\therefore \qquad = -40 c_p \left(1 - \frac{T_0}{T}\right)\delta T$$

With a very great number of engines in the series, the total work (maximum) obtainable when the water is cooled from 348K to 278K would be

$$W_{(max)} = \text{A.E.} = -\lim \sum_{348}^{278} 40c_p\left(1-\frac{T_0}{T}\right)\delta T$$

$$= \int_{278}^{348} 40\, c_p\left(1-\frac{T_0}{T}\right)dT$$

$$= 40c_p\left[(348-278) - 278 \ln \frac{348}{278}\right]$$

$$= 40 \times 4.2\ (70-62)$$

$$= 1340 \text{ kJ} \qquad \textit{Ans.}$$

$$Q_1 = 40 \times 4.2\ (348-278)$$

$$= 11{,}760 \text{ kJ}$$

$$\text{U.E.} = Q_1 - W_{(max)}$$

$$= 11{,}760 - 1340 = 10{,}420 \text{ kJ}$$

7. Air enters a compressor at 1 bar, $30^0$C, which is also the state of environment. It leaves at 3.5 bar, $141^0$ and 90m/s. Neglecting inlet velocity and P.E. effect, determine (a) whether the compression is adiabatic or polytropic, (b) If not adiabatic, the polytropic index, (c) the isothermal efficiency, (d) the minimum work input and irreversibiity and (e) second law efficiency. Take $C_p$ of air =1.0035kJ/kgK

Solution:
(a) After isentropic compression

$$\frac{T_{2s}}{T_1} = \left[\frac{p_2}{p_1}\right]^{(\gamma-1)/\gamma}$$

$$T_{2s} = 303\ (3.5)^{0.286} = 433.6 \text{ K} = 160.6°\text{C}$$

Since this temperature is higher than the given temperature of $141^0$C, there is heat loss to the surroundings. The compression cannot be adiabatic. It must be polytropic.

(b) $$\frac{T_2}{T_1} = \left[\frac{p_2}{p_1}\right]^{(n-1)/n}$$

$$\frac{141 + 273}{30 + 273} = 1.366 = \left(\frac{3.5}{1}\right)^{(n-1)/n}$$

$$\log 1.366 = \frac{n-1}{n} \log 3.5$$

$$1 - \frac{1}{n} = \frac{0.135}{0.544} = 0.248$$

$$n = 1.32978 = 1.33$$

(c) Actual work of compression

$$W_a = h_1 - h_2 - \frac{V_2^2}{2} = 1.0035\,(30 - 141) - \frac{90^2}{2} \times 10^{-3}$$
$$= -115.7 \text{ kJ/kg}$$

Isothermal work

$$W_T = \int_1^2 v\,dp - \frac{V_2^2}{2} = -RT_1 \ln \frac{p_2}{p_1} - \frac{V_2^2}{2}$$

$$= -0.287 \times 303 \ln (3.5) - \frac{90^2}{2} \times 10^{-3}$$

$$= -113 \text{ kJ/kg}$$

Isothermal efficiency:

$$\eta_T = \frac{W_T}{W_a} = \frac{113}{115.7} = 0.977 \text{ or } 97.7\% \qquad \textit{Ans.}$$

(d) Decrease in availability or exergy:

$$\psi_1 - \psi_2 = h_1 - h_2 - T_0(s_1 - s_2) + \frac{V_1^2 - V_2^2}{2}$$

$$= c_p(T_1 - T_2) - T_0\left[R \ln \frac{p_2}{p_1} - c_p \ln \frac{T_2}{T_1}\right] - \frac{V_2^2}{2}$$

$$= 1.0035\,(30 - 141)$$

$$- 303\left[0.287 \ln 3.5 - 1.0035 \ln \frac{414}{303}\right] - \frac{90^2}{2000}$$

$$= -101.8 \text{ kJ/kg}$$

Minimum work input $= -101.8$ kJ/kg *Ans.*

Irreversibility,

$$I = W_{rev} - W_a$$

$$= -101.8 - (-115.7)$$

$$= 13.9 \text{ kJ/kg} \qquad \textit{Ans.}$$

(e) Second law efficiency,

$$\eta_{II} = \frac{\text{Minimum work input}}{\text{Actual work input}} = \frac{101.8}{115.7}$$

$$= 0.88 \text{ or } 88\% \qquad \textit{Ans.}$$

# SATHYABAMA
**INSTITUTE OF SCIENCE AND TECHNOLOGY**

**DEPARTMENT OF Electrical and Electronics Engineering**

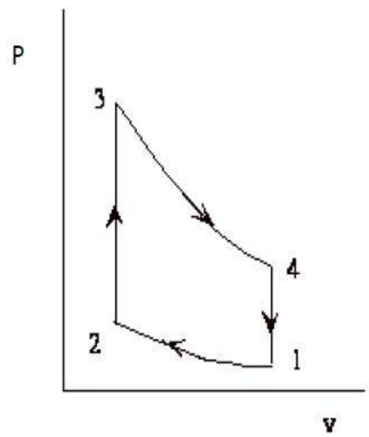
**COURSE MATERIAL**

**Subject Name: Applied Thermal Engineering UNIT III Subject Code: SME1208**

**AIR STANDARD CYCLES - OTTO, DIESEL AND DUAL CYCLES. DERIVATION OF EXPRESSION FOR AIR STANDARD EFFICIENCY AND MEAN EFFECTIVE PRESSURE. IC ENGINES- INTRODUCTION-CLASSIFICATION, COMPARISON BETWEEN FOUR STROKE AND TWO STROKE, PERFORMANCE TESTING ON INTERNAL COMBUSTION ENGINES, PERFORMANCE CURVES.**

## The Otto Cycle

The Otto cycle, which was first proposed by a Frenchman, Beau de Rochas in 1862, was first used on an engine built by a German, Nicholas A. Otto, in 1876. The cycle is also called a constant volume or explosion cycle. This is the equivalent air cycle for reciprocating piston engines using spark ignition. Figures 1 and 2 show the P-V and T-s diagrams respectively.



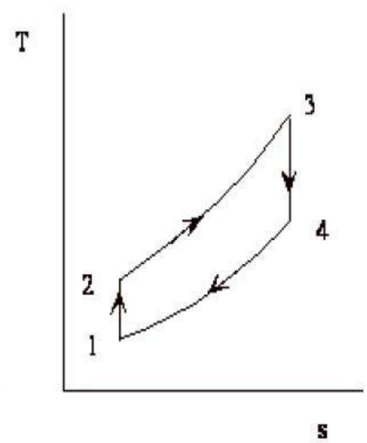




Fig.1: P-V Diagram of Otto Cycle. Fig.2: T-S Diagram of Otto Cycle.

At the start of the cycle, the cylinder contains a mass M of air at the pressure and volume indicated at point 1. The piston is at its lowest position. It moves upward and the gas is compressed isentropically to point 2. At this point, heat is added at constant volume which raises the pressure to point 3. The high pressure charge now expands isentropically, pushing the piston down on its expansion stroke to point 4 where the charge rejects heat at constant volume to the initial state, point 1.

The isothermal heat addition and rejection of the Carnot cycle are replaced by the constant volume processes which are, theoretically more plausible, although in practice, even these processes are not practicable.

The heat supplied, $Q_s$, per unit mass of charge, is given by

$$c_v(T_3 - T_2) \qquad (1)$$

the heat rejected, $Q_r$ per unit mass of charge is given by

$$c_v(T_4 - T_1) \qquad (2)$$

and the thermal efficiency is given by

$$\eta_{th} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$= 1 - \frac{T_1}{T_2}\left\{\frac{\left(\frac{T_4}{T_1} - 1\right)}{\left(\frac{T_3}{T_2} - 1\right)}\right\} \qquad (3)$$

$$\text{Now } \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \frac{T_4}{T_3}$$

$$\text{And since } \frac{T_1}{T_2} = \frac{T_4}{T_3} \quad \textit{we have} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

Hence, substituting in Eq. 3, we get, assuming that r is the compression ratio $V_1/V_2$

$$\eta_{th} = 1 - \frac{T_1}{T_2}$$

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

$$= 1 - \frac{1}{r^{\gamma-1}} \qquad (4)$$

In a true thermodynamic cycle, the term expansion ratio and compression ratio are synonymous. However, in a real engine, these two ratios need not be equal because of the valve timing and therefore the term expansion ratio is preferred sometimes.

Equation 4 shows that the thermal efficiency of the theoretical Otto cycle increases with increase in compression ratio and specific heat ratio but is independent of the heat added (independent of load) and initial conditions of pressure, volume and temperature.

Figure 3 shows a plot of thermal efficiency versus compression ratio for an Otto cycle. It is seen that the increase in efficiency is significant at lower compression ratios. This is also seen in Table 1 given below.

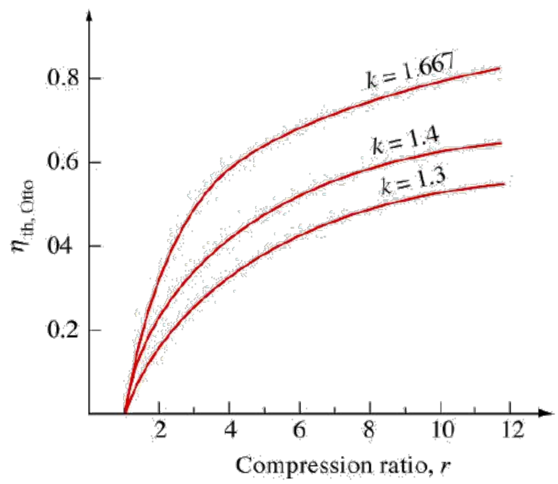



Fig.3: variation of efficiency with compression ratio

Table1: compression ratio and corresponding thermal efficiency for Otto cycle

| R | η |
|---|---|
| 1 | 0 |
| 2 | 0.242 |
| 3 | 0.356 |
| 4 | 0.426 |
| 5 | 0.475 |
| 6 | 0.512 |
| 7 | 0.541 |
| 8 | 0.565 |
| 9 | 0.585 |
| 10 | 0.602 |
| 16 | 0.67 |
| 20 | 0.698 |
| 50 | 0.791 |

From the table it is seen that if:

CR is increased from 2 to 4, efficiency increase is 76%

CR is increased from 4 to 8, efficiency increase is only 32.6%

CR is increased from 8 to 16, efficiency increase is only 18.6%

**Mean effective pressure and air standard efficiency**

It is seen that the air standard efficiency of the Otto cycle depends only on the compression ratio. However, the pressures and temperatures at the various points in the cycle and the net work done, all depend upon the initial pressure and temperature and the heat input from point 2 to point 3, besides the compression ratio.

A quantity of special interest in reciprocating engine analysis is the mean effective pressure. Mathematically, it is the net work done on the piston, W, divided by the piston displacement volume, $V_1 - V_2$. This quantity has the units of pressure. Physically, it is that constant pressure which, if exerted on the piston for the whole outward stroke, would yield work equal to the work of the cycle. It is given by

$$mep = \frac{W}{V_1 - V_2}$$

$$= \frac{\eta Q_{2-3}}{V_1 - V_2} \qquad (5)$$

where $Q_{2\text{-}3}$ is the heat added from points 2 to 3.

Work done per kg of air

$$W = \frac{P_3V_3 - P_4V_4}{\gamma - 1} - \frac{P_2V_2 - P_1V_1}{\gamma - 1} = mepV_s = P_m(V_1 - V_2)$$

$$mep = \frac{1}{(V_1 - V_2)}\left[\frac{P_3V_3 - P_4V_4}{\gamma - 1} - \frac{P_2V_2 - P_1V_1}{\gamma - 1}\right] \qquad (5A)$$

The pressure ratio $P_3/P_2$ is known as explosion ratio $r_p$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\gamma} = r^{\gamma} \Rightarrow P_2 = P_1 r^{\gamma},$$

$$P_3 = P_2 r_p = P_1 r^{\gamma} r_p,$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{\gamma} = P_1 r^{\gamma} r_p \left(\frac{V_2}{V_1}\right)^{\gamma} = P_1 r_p$$

$$\frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = r$$

$$\therefore V_s = V_c(r-1)$$

Substituting the above values in Eq 5A

$$mep = P_1 \frac{r(r_p - 1)(r^{\gamma-1} - 1)}{(r-1)(\gamma-1)} \qquad , \qquad \text{Now}$$

$$V_1 - V_2 = V_1\left(1 - \frac{V_2}{V_1}\right)$$

$$= V_1\left(1 - \frac{1}{r}\right) \qquad (6)$$

Here r is the compression ratio, $V_1/V_2$

**Diesel Cycle**

This cycle, proposed by a German engineer, Dr. Rudolph Diesel to describe the processes of his engine, is also called the constant pressure cycle. This is believed to be the equivalent air cycle for the reciprocating slow speed compression ignition engine. The P-V and T-s diagrams are shown in Figs 4 and 5 respectively.

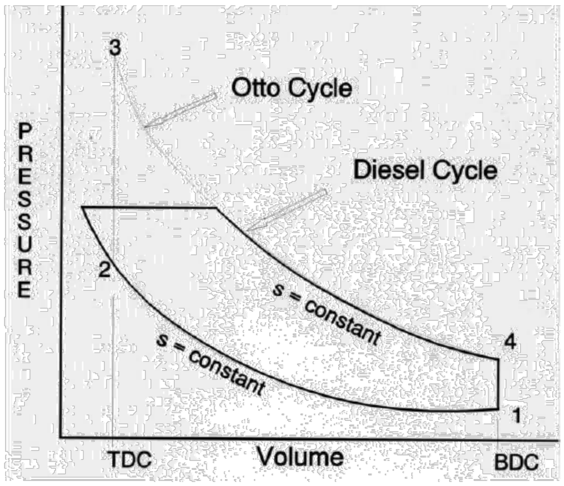



Fig.4: P-V Diagram of Diesel Cycle.

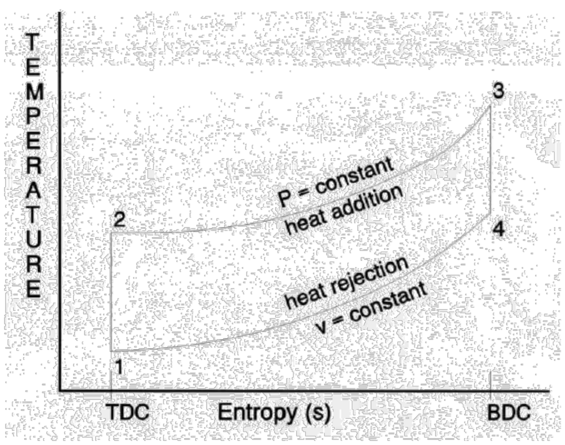



Fig.5: T-S Diagram of Diesel Cycle.

The cycle has processes which are the same as that of the Otto cycle except that the heat is added at constant pressure.

The heat supplied, $Q_s$ is given by

$$c_p(T_3 - T_2) \qquad (22)$$

whereas the heat rejected, $Q_r$ is given by

$$c_v(T_4 - T_1) \qquad (23)$$

and the thermal efficiency is given by

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

$$= 1 - \frac{1}{\gamma}\left\{\frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)}\right\} \qquad (24)$$

From the T-s diagram, Fig. 5, the difference in enthalpy between points 2 and 3 is the same as that between 4 and 1, thus

$$\Delta s_{2-3} = \Delta s_{4-1}$$

$$\therefore\ c_v \ln\left(\frac{T_4}{T_1}\right) = c_p \ln\left(\frac{T_3}{T_2}\right)$$

$$\therefore\ \ln\left(\frac{T_4}{T_1}\right) = \gamma \ln\left(\frac{T_3}{T_2}\right)$$

$$\therefore\ \frac{T_4}{T_1} = \left(\frac{T_3}{T_2}\right)^{\gamma} \text{ and } \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \frac{1}{r^{\gamma-1}}$$

Substituting in eq. 24, we get

$$\eta_{th} = 1 - \frac{1}{\gamma}\left(\frac{1}{r}\right)^{\gamma-1}\left[\frac{\left(\frac{T_3}{T_2}\right)^{\gamma} - 1}{\frac{T_3}{T_2} - 1}\right] \qquad (25)$$

$$\text{Now } \frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c = cut - off \ ratio$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_c^{\gamma} - 1}{\gamma(r_c - 1)}\right] \qquad (26)$$

When Eq. 26 is compared with Eq. 8, it is seen that the expressions are similar except for the term in the parentheses for the Diesel cycle. It can be shown that this term is always greater than unity.

Now $r_c = \frac{V_3}{V_2} = \frac{V_3}{V_4}\bigg/\frac{V_2}{V_1} = \frac{r}{r_e}$ where r is the compression ratio and $r_e$ is the expansion ratio

Thus, the thermal efficiency of the Diesel cycle can be written as

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\left(\frac{r}{r_e}\right)^{\gamma} - 1}{\gamma\left(\frac{r}{r_e} - 1\right)}\right] \qquad (27)$$

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Thus Otto cycle engines have compression ratios in the range of 7 to 12 while diesel cycle engines have compression ratios in the range of 16 to 22.

$$mep = \frac{1}{V_s}\left[P_2(V_3 - V_2) + \frac{P_3V_3 - P_4V_4}{\nu - 1} - \frac{P_2V_2 - P_1V_1}{\nu - 1}\right] \qquad (29)$$

The pressure ratio $P_3/P_2$ is known as explosion ratio $r_p$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{\nu} = r^{\nu} \Rightarrow P_2 = P_1 r^{\nu},$$

$$P_3 = P_2 = P_1 r^{\nu}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{\nu} = P_1 r^{\nu}\left(\frac{V_2}{V_1}\right)^{\nu} = P_1 r_c^{\ \nu}$$

$$V_4 = V_1, V_2 = V_c,$$

$$\frac{V_1}{V_2} = \frac{V_c + V_s}{V_c} = r$$

$$\therefore V_s = V_c(r-1)$$

Substituting the above values in Eq 29 to get Eq (29A)

In terms of the cut-off ratio, we can obtain another expression for mep/$p_1$ as follows

$$mep = P_1 \frac{\gamma r^{\gamma}(r_c - 1) - r(r_c^{\gamma} - 1)}{(r-1)(\gamma - 1)} \qquad (29A)$$

**The Dual Cycle**

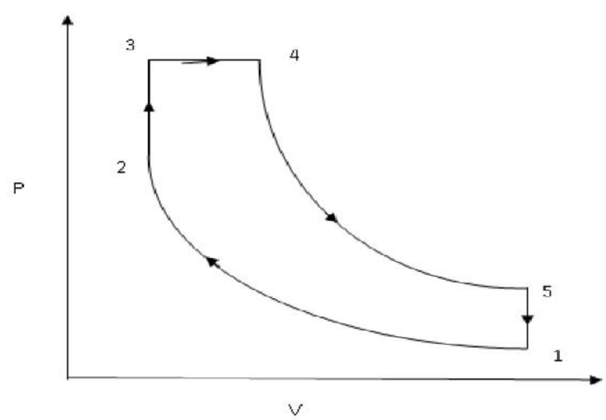



Fig.6: P-V Diagram of Dual Cycle.

Process 1-2: Reversible adiabatic compression.

Process 2-3: Constant volume heat addition.

Process 3-4: Constant pressure heat addition.

Process 4-5: Reversible adiabatic expansion.

Process 5-1: Constant volume heat reject

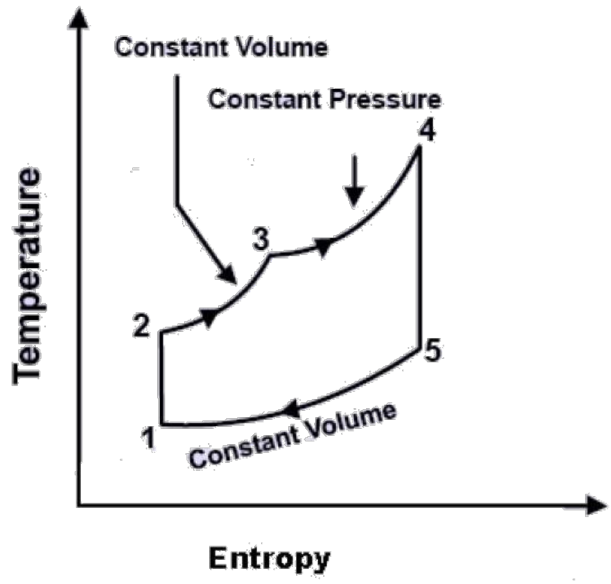



Fig.7: T-S Diagram of Carnot Cycle.

The cycle is the equivalent air cycle for reciprocating high speed compression ignition engines. The P-V and T-s diagrams are shown in Figs.6 and 7. In the cycle, compression and expansion processes are isentropic; heat addition is partly at constant volume and partly at constant pressure while heat rejection is at constant volume as in the case of the Otto and Diesel cycles.

The heat supplied, $Q_s$ per unit mass of charge is given by

$$c_{v}(T_3 - T_2) + c_{p}(T_{3'} - T_2) \qquad (32)$$

whereas the heat rejected, $Q_r$ per unit mass of charge is given by

$$c_{v}(T_4 - T_1)$$

and the thermal efficiency is given by

$$\eta_{th} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2) + c_p(T_{3'} - T_2)} \qquad (33A)$$

$$= 1 - \left\{ \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right) + \gamma T_3\left(\frac{T_{3'}}{T_3} - 1\right)} \right\} \qquad (33B)$$

$$= 1 - \frac{\frac{T_4}{T_1} - 1}{\frac{T_2}{T_1}\left(\frac{T_3}{T_2} - 1\right) + \frac{\gamma T_3}{T_2}\frac{T_2}{T_1}\left(\frac{T_{3'}}{T_3} - 1\right)} \qquad (33C)$$

From thermodynamics

$$\frac{T_3}{T_2} = \frac{p_3}{p_2} = r_p \qquad (34)$$

the explosion or pressure ratio and

$$\frac{T_{3'}}{T_3} = \frac{V_{3'}}{V_3} = r_c \qquad (35)$$

the cut-off ratio.

$$\text{Now, } \frac{T_4}{T_1} = \frac{p_4}{p_1} = \frac{p_4}{p_{3'}}\frac{p_{3'}}{p_3}\frac{p_3}{p_2}\frac{p_2}{p_1}$$

$$\text{Also } \frac{p_4}{p_{3'}} = \left(\frac{V_{3'}}{V_4}\right)^\gamma = \left(\frac{V_{3'}}{V_3}\frac{V_3}{V_4}\right)^\gamma = \left(r_c\frac{1}{r}\right)^\gamma$$

$$\text{And } \frac{p_2}{p_1} = r^\gamma$$

$$\text{Thus } \frac{T_4}{T_1} = r_p r_c^\gamma$$

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

Therefore, the thermal efficiency of the dual cycle is

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_p r_c^{\gamma} - 1}{(r_p - 1) + \gamma r_p (r_c - 1)}\right] \qquad (36)$$

We can substitute the value of η from Eq. 36 in Eq. 14 and obtain the value of $mep/p_1$ for the dual cycle.

In terms of the cut-off ratio and pressure ratio, we can obtain another expression for $mep/p_1$ as follows:

$$\frac{mep}{p_1} = \frac{\gamma r_p r^{\gamma}(r_c - 1) + r^{\gamma}(r_p - 1) - r(r_p r_c^{\gamma} - 1)}{(r-1)(\gamma - 1)} \qquad (37)$$

**PART-A (2 Marks)**

**1. Define mean effective pressure of Otto cycle. [May/Jun 2011, 2012]**

The mean pressure developed during one cycle of operation is called as mean effective pressure. In other words it is the ratio of work done to the swept volume.
Mean Effective Pressure {Pm] = Work done/Swept volume.

**2. Mention the thermodynamic processes involved in Diesel cycle. [May/Jun 2011]**

1. Isentropic compression. 2. Constant pressure heat addition.
3. Isentropic expansion, and 4. Constant volume heat rejection.

**3. What is a thermodynamic cycle? [Oct 1997]**

Thermodynamic cycle is defined as the series of processes performed on the system, so that the system attains its original state.

**4. What are the assumptions made for air standard cycle analysis? [Nov 2002, May 2003, Apr 2005 & May 2013]**

1. The working medium is a perfect gas throughout i.e., it follows the law pν =mRT
2. The working medium does not undergo any chemical change throughout the cycle.
3. The compression and expansion processes are reversible adiabatic i.e., There are no loss or gain of entropy.
4. Kinetic and potential energies of the working fluid are neglected.
5. The operation of the engine is frictionless.
6. Heat is supplied and rejected in a reversible manner.

**5. Mention the various processes of the Brayton cycle. [Oct 1996]**

1. Isentropic compression,
2. Constant pressure heat supplied,
3. Isentropic expansion, and
4. Constant pressure heat rejection.

**6. Mention the various processes of dual cycle. [Apr 1996]**
1. Isentropic compression,
2. Constant volume heat addition,
3. Constant pressure heat addition,
4. Isentropic expansion, and
5. Constant volume heat rejection.

**7. Define air standard cycle efficiency. [Oct 1996, & Oct 1997]**
Air standard efficiency is defined as the ratio of work done by the cycle to the heat supplied to the cycle.

**8. Which cycle is more efficient with respect to the same compression ratio? [Oct 1995]** For the same compression ratio, Otto cycle is more efficient than diesel cycle.

**9. For the same compression ratio and heat supplied, state the order of decreasing air standard efficiency of Otto, diesel and dual cycle. [Apr 1998, Oct 1998]**
ή Otto > ή Dual > ή Diesel

**10. Name the factors that affect air standard efficiency of Diesel cycle. [Apr 1997]** Compression ratio and cut-off ratio

**11. What is the effect of cut-off ratio on the efficiency of diesel cycle when the compression ratio is kept constant? [Apr 2003]**
When cut-off ratio of diesel cycle increases, the efficiency of cycle is decreased when compression ratio is kept constant and vice versa.

**12. What are all the modifications are carried out in Brayton cycle? Why?**
In Brayton we incorporate [i] Regenerator [ii] Reheater and [iii] Intercooler, because of increasing thermal efficiency of the cycle.

**13. Mention the thermodynamic processes involved in otto cycle.**
1. Isentropic compression. 2. Constant volume heat addition.
3. Isentropic expansion, and 4. Constant volume heat rejection.

**14. What is the range of compression ratio for otto and diesel cycles?** Otto cycle 6 to 8
Diesel cycle 12 to 16

**15. What is compression ratio (Nov 2010 & May 2014)**
It is the ratio of volume when the piston is at BDC to the volume when the piston is at TDC

**PART-B (16 Marks)**

**1. A six cylinder petrol engine has a compression ratio of 5:1. The clearance volume of each cylinder is 110CC. It operator on the four stroke constant volume cycle and the indicated efficiency ratio referred to air standard efficiency is 0.56. At the speed of 2400 rpm. It consumer 10kg of fuel per hour. The calorific value of fuel is 44000KJ/kg. Determine the average indicated mean effective pressure. [Apr 1995]**

**Given data:**

$r = 5$
$V_c = 110CC$
$\eta_{relative} = 0.56$

$Z = 6$

**Solution:**

**Compression ratio:**

$r = V_s + V_c/V_c$ $\rightarrow 5 = V_s + 110/110 \rightarrow V_s = 440CC = 44 x 10^{-6}$ 3

**Air standard efficiency:**

$\eta = 1- 1 / ( ^{-1} ) = 47.47\%$ $( = 1.4)$

**Relative efficiency:**

$\eta_{relative} = \eta_{actual} / \eta_{air-standard} \rightarrow 0.56 = \eta_{actual}/47.47$

$\eta_{actual} = 26.58\%$

**Actual efficiency** = work output/ head input

$W = 32.49kw.$

**The net work output:**

$W = P_m \times V_s \times N/60 \times Z \rightarrow 32.49\ 10^3 = P_m \times 440 \times 10^{-6} \times 1200/60 \times 6$ $P_m = 6.15$ bar

**2. One kg of air taken through, a) Otto cycle, b) Diesel cycle initially the air is at 1 bar and 290k. The compression ratio for both cycles is 12 and heat addition is 1.9 MJ in each cycle. Calculate the air standard efficiency and mean effective pressure for both the cycles.**

**Given data:**

$P_1 = 1$ bar = 100KN/ m³
$T_1 = 290K$
$R = 12$
$Q_s = 1.9MJ = 1900KJ$

**Solution:**

**a) Otto cycle:**

**For process 1-2: isentropic compression:**

$P_2/P_1 = ( V_1/V_2) \rightarrow P_2 = P_1$ x
$P_2 = 3242.3$kN/m2
$T_2/T_1 = ( V_1/V_2)$ $-1 \rightarrow T_2 = T_1$ x $( V_1/V_2)$ $-1 = 290$ x $(12)1.4\text{-}1$
$T_2 = 783.55$K

**Heat supplied:**

$Q = m$ x $C_v ( T_3 - T_2 )$
$1900 = 1$ x $0.718$ x $(T_3 - 783.55)$
$T_3 = 3429.79$K

**For process 2-3 : constant volume process**

$P_3/P_2 = T_3/ T_2 \rightarrow P_3 = P_2$ x $T_3/ T_2 = 3242.3$ x $3429.79/783.55$
$P = 14196.7$KN/m2

**Air standard efficiency:**
$\eta = 1- 1 / ($ $^{-1}) = 0.6298$
$\eta = 62.98\%$

**Pressure ratio,** $K = P_3/P_2 = 14196.7/32423 = 4.378$

**Mean effective pressure,**

$Pm = p_1 r ( k\text{-}1/$ $- 1) ( ^{-1}\text{-}1/r\text{-}1) = 100$ x $12 ( 4.378\text{-}1/1.4) [ ( 12^{1.4\text{-}1}\text{-}1/12\text{-}1)]$

**b) Diesel cycle:**

**Consider 1-2 isentropic compression process:**

$T_2 = (V_1/V_2)^{-1}$ x $T_1 = ( r )^{-1}$ x $T_1 = (12)^{1.4-1}$ x $290$
$T_2 = 783.56$K

**Consider 2-3 constant pressure heat addition:**

$Q_s = C_p ( T_3 - T_2 )$
$1.9$ x $10^3 = 1.005$ x $( T_3 - 783.56 )$
$T_3 = 2674$ K.

**Cut off ratio:**

$P = V_3/V_2 = T_3/T_2 = 2674/783.56 = 3.413$

**Air standard efficiency:**

$\eta = 1\text{-}1/ (r)^{-1} \{ P^{-1}/p\text{-}1\} = 1\text{-}1/ 1.4(12)^{1.4-1} \{3.413^{1.4-1}/3.413\text{-}1\}$ $\eta = 49.86\%$

**Mean effective pressure:**

$P_m = P_1 r [ (p\text{-}1) - r^{1-} ( p -1)/ ( -1) ( r\text{-}1 )]$
$100$ x $(12)^{1.4} [ 1.4 (3.413\text{-}1) - (12)^{1.4-1} ( 3.413^{1.4}\text{-}1)]/(1.4\text{-}1)$
$(12\text{-}1)$ $P_m = 1241$KN/m$^2$

**3.An air standard dual cycle has a compression ratio of 16 and compression begins at 1 bar and 50°C. The maximum pressure is 70 bar. The heat transformed to air at constant pressure is equal to heat transferred at constant volume. Find the temperature at all cardial points, cycle efficiency and mean effective pressure take Cp= 1.005KJ/kgK, Cv = 0.718KJ/kgK. [May 2003]**

**Given data:**

$R = 16$

$P_1 = 1$ bar

$T_1 = 50°C = 323K$

$P_3 = 70$ bar

$Q_{s1} = Q_{s2}$

$C_p = 1.005$kJ/kgk

$C_v = 0.718$kJ/kgk

**Solution:**

**Specific volume,**

$V_1 \quad RT_1/P_1 = 287 \times 323/1 \times 10^5$

$V_1 = 0.92701 m^3/kg$

$V_2 = 0.05794 m^3/kg$

**1-2 isentropic compression process:**

$P_2 = (r) \quad x\ P_1 = (16)^{1.4} \times 1 = 48.5$

bar $T_2 = (r)^{-1} \times T_1 = (16)^{1.4-1} \times 323$

$T_2 = 979K$

**2-3 constant volume heat addition process:**

$T_3 = (P_3/P_2) \times T_2 = 70/48.5 \times 979$

$T_3 = 1413K$

$Q_{s1} = Cv\ (T_3-T_2);\ 0.718(1413 – 979)$

$Q_{s1} = 311.612KJ/kg$

**3-4 constant pressure heat addition**:

$Q_{s1} = Q_{s2} = C_p\ (T_4 – T_3)$

$311.62 = 1.005\ (T_4 – 1413)$

$T_4 = 1723K$

$V_4 = T_4/T_3 \times V_3 = 1723/1413 \ X\ 0.05794$

$V_4 = 0.070652 m^3/kg$

**Expansion ratio:**

$r_e = V_4/V_1 = 0.70652/0.92701 = 0.06215$

**4-5 isentropic expansion process:**

$P_5 = (r) \times P_4 = (0.076215)^{1.4} \times 70$
$P_5 = 1.9063$ bar
$T_5 = (r)^{-1} \times T_4$
$= (0.076215)^{1.4-1} \times 1723$
$= 567$ K

**Cut off ratio,**

P=V4/V3
= 0.00652/0.05744
P = 1.2194

**Pressure ratio** (K) $= (P_3/P_2) = (70/48.5)$

$^{-1}$ [ ( kp -1)/ (k-1 + K (p-1)]

K = 1.4433

**The cycle efficiency:**

$\eta = 1 - 1/(r)$
$\eta = 66.34\%$

**Net heat supplied to the cycle:**

$Q_s = Q_{s1} + Q_{s2}$
$= 311.612 + 311.612$
$= 623.224$ KJ/kg

**The mean effective pressure:**

$P_m = W/V_1 - V_2 = 413.45/(0.92701 - 0.05794)$
$P_m = 4.75$ bar

**4. In a air standard dual cycle, the compression ratio is 12 and the maximum pressure and temperature of the cycle are 1 bar and 300K. haet added during constant pressure process is upto 3% of the stroke. taking diameter as 25cm and stroke as 30cm, determine.**

**1) The pressure and temperature at the end of compression**
**2) The thermal efficiency**
**3) The mean effective pressure**

**Take , Cp =1.005KJ/kgk Cv =0.118KJ/kgk , = 1.4 [Nov 2004]**

**Given data:**

$P_1 = 1$ bar
R = 12
$T_1 = 300$K
K = 3% of Vs = 0.03Vs
$P_3 = 70$ bar
D = 25 cm
L = 30cm

**Solution :**

**Specific volumes,:**

$V_1$ $RT_1/P_1$ = 287 x 300/ 1 x $10^5$
= 0.861 m3 /kg
$V_3 = V_2 = V_1/r$ = 0.861/12
= 0.07175$m^3$/kg
$V_4 - V_3 = 0.03\ (V_1 - V_2)$
$V_4$ = 0.0954275 $m^3$/kg

**Cut off ratio:**

P = $V_4/V_3$ = 0.054275/0.07175
P = 1.33

**1-2 isentropic compression process:**

$P_2$ = ( r) x P1 = $(12)^{1.4}$ x 1
= 32.423 bar
$V_2 = (r)^{-1} \times T_1 = (12)^{1.4-1} \times 300$
$T_2$ = 810.57K.

**2-3 constant volume heat addition process**

$P_3/T_3 = P_2/T_2$
$T_3 = (P_3/P_2) \times T_2$ = ( 70 / 32.423) x 810.57
$T_3$ = 1750K

**3-4 constant pressure heat addition process:**

$T_4 = (V_4/V_3) \times T_3$ = ( 0.0954275 / 0.07175 ) x1750
$T_4$ = 2327.5 K

**Pressure ratio, K** = $(P_3/P_2)$ = 70/32.423 = 2.159

**Net heat supplied to the cycle:**

$Q_S = C_v\ (T_3 - T_2) + C_p\ (T_4 - T_3)$
= 0.718 ( 1750 -810.57 ) + 1.005( 2327.5-1759)
= 1254.9 KJ/kg

**Efficiency of the cycle:**

$\eta = 1 - 1/(r)^{-1}$ [ ( K x P -1)/(k-1) + K (p-1)]
$\eta$ = 61.92%

**Net workdone of the cycle:**

W = $\eta \times Q_s$
= 0.6192 x 1254.9
= 777.1 KJ/kg

**Mean effective pressure,**

$P_m = W/ V_1 - V_2$
= 777.1/ 0.361 – 0.07115
= 984.6 Kpa

$P_m = 9.846$ bar

**5.The compression ratio of a dual cycle is 10. The pressure and temperature at the beginning of the cycle are 1 bar and 27$^{o}$C. the maximum pressure of the cycle is limited to 70 bar and heat supplied is limited to 1675KJ/kg fair find thermal efficiency.**

**Given data:**
V=10
$P_1 = 1$ bar
$T_1 = 27^{o}C = 300K$
$P_3 = 70$ bar
$Qs = 1675$ KJ/kg
**Solution:**
**Specific volumes:**
$V_1 = RT_1/P_1 = 287 \times 300/1 \times 10^5$
$V_2 = V_1/r$
$= 0.861/10$
**1-2 isentropic compression process:**
$P_1 = (r) \quad x\ P_1 = (10)^{1.4} \times 1 = 25.12$ bar
$T_1 = (r)^{-1} \times T_1 = (10)^{1.4-1} \times 300 = 753.57K$
**2-3 constant volume heat addition process:**
$T_3 = (P_3/P_2)\ x = (70 / 25.12) \times 753.37 = 2100K$

**Total heat supplied to the cycle:**
$Q_s = C_v (T_3 - T_2) + C_p (T_4 - T_3)$
$1675 = 0.718 (2100 - 753.57) + 1.005 (T4 - 2100)$
$T_4 = 2804.6$ K

**Cut off ratio:**
$P = V_4/V_3 = T_4/T_3 =$
$2804.6/2300\ P = 1.3356$

**Pressure ratio:**
$K = P3/P2 = 70/25.12 = 2.787$

**Efficiency of the cycle:**
$\eta = 1 - 1/(r)^{-1} [(K \times P - 1)/(k-1) + K (p-1)]$
$\eta = 59.13\%$

**6. In an air standard diesel cycle, the pressure and temperature of air at the beginning of cycle are 1 bar x 40$^{0}$C. The temperatures before and after the heat supplied are 400$^{0}$C and 1500$^{0}$C. Find the air standard efficiency and mean effective pressure of the cycle. What is the power output if it makes 100 cycles / min?**

**Given data:**
$P_1 = 1$ bar = 100KN/m2
$T1 = 40^{0}C = 313K$
$T2 = 400^{0}C = 673K$
$T3 = 1500^{0}C = 1773K$

**Solution :**
**1-2 isentropic compression:**
$T_2/T_1 = ( r )^{-1}$ **Compression ratio :**
$r = V_1/V_2 = ( T_2/T_1)^{1/ -1}$
$= ( 673/313)^{1/1.4-1}$
$= 6.779$
**2-3 constant pressure heating:**
$V_2/T_2 = V_3/T_3$
**Cut off ration, P** $= V_3/V_2 = T_3/T_2 = 1773/673 = 2.634$

**Efficiency :**
$\eta = 1- 1/ ( r )^{-1}( p -1/p-1)$
$= 0.4142\%$

**Mean effective pressure:**
$P_m = P_1 r [ (p-1) –r^{1-} ( p -1)]/( -1) ( r -1)$
$=100 x ( 6.779)^{1.4} [ 1.4 (2.634-1) – (6.779)^{1-1.4} ( 2.634^{1.4}-1)]/(1.4-1) x( 6.779-1)$ $P_m = 597.77KN/m^2$

**Heat supplied :**
$m x C_p ( T_3 – T_2)$
$= 1 x 1.005 ( 1773 -673)$
$Q_s = 1105.5 KJ /kg$

**Work done :**
$\eta x Q_s = 0.4142 x 1105.5$
$= 457.89KJ/kg$

= W x cycle /min =457.89 x 100 =45 x $10^{-3}$ KJ/kg-min
= 763.16Kj/kg-sec
= 763.16W/kg

**7. In a brayton cycle, the air enters the compressor at 1 bar and 25$^o$C. the pressure of air leaving the compressor is 3 bar and temperature at turbine inlet is 650$^o$C. determine per kg of aire, i) cycle efficiency ii) heat supplied to air iii) work input iv) heat rejected in the cooler and v) temperature of air leaving the turbine. [May 2003]**

**Given data:**
$P_1 = 1$ bar
$T_1 = 25^oC$

$T_3 = 650^oC$
$P_2 = 3$ bar

**Solution :**
**Consider the process 1-2 adiabatic compression:**
$T_2/T_1 = (P_2/P_1)^{-1/}$
$T_2 = (P_2/P_1)^{-1/}$ x T1 $T_2$
$= (3/1)^{1.4-1/1.4}$ x 298
**3-4 adiabatic expansion:**
$T_4/T_3 = (P_4/P_3)^{-1/}$
$T_4 = (P_4/P_3)^{-1/}$ x 923 = 674.3k

**Air standard efficiency :**
$\eta = 1- 1/(R_p)^{-1/} = 1- 1/(3)^{1.4-1/1.4} = 0.2694$ $\eta$
= 26.94%

**Heat supplied $Q_s$** = $C_p$ ( $T_3$ – $T_2$) = 1.005 ( 923 – 408) = 517.575 KJ/kg

**Heat rejected $Q_R$** = $C_p$ ($T_4$ – $T_1$) = 1.008 ( 673.4 – 298) = 377.277KJ/kg

**Compressor work $W_C$** = $C_p$ ( $T_2$ – $T_1$) = 1.005 x ( 408 – 298) = 110.55Kj/kg

**Similarly for expander,:**
$W_e = C_p$ x ( $T_3$ – $T_4$ ) = 1.005 (923 – 6734)
$W_e$ = 250.848 – 110.55 = 140.288Kj/kg
**Temperature of air leaving the turbine** = 673.4K

**8. In an air standard brayton cycle, the air enter the compressor at 1 bar and 15$^o$C. The pressure leaving the compressor is 5 bar the maximum temperature in the cycle 900$^o$C. Find the following .**

a) **Compressor and expander work per kg of air. b) the cycle efficiency .**

**If an ideal regenerator is incorporated into the cycle, determine the percentage change in efficiency.**

**Given data:**
$P_1$ = P4 = 1 bar = 100KN /m2
$T_1 = 15^oC$ =288k
$P_2$ = P3 = 5 bar = 500Kn /m2
$T_3 = 900^oC$ =1173k

**Solution:**
**1-2 isentropic compression:**
$T_2/T_1 = ( p_2/P_1)^{-1/}$ ; $T_2 = (P_2/P_1)^{-1/}$ x $T_1$ = 456k

**Consider the process 3-4 isentropic expansion:**

$T_4/T_3 = ( P_4/P_3)^{-1/}$ : $T_4 = (P_4/P_3)^{-1/} \times T_3 = 740.6k$

Work done by the compressor when it operates isentropic ally is given by

**Compressor work** $W_c = C_p ( T_2 – T_1 ) = 1.005(456 – 288) = 168.756KJ$

**For expander** $W_e = C_p ( T_3 –T_4 ) = 1.005(1173 – 740.6) = 434.34KJ$

**Air standard efficiency :**

$\eta = 1- 1/(R_p)^{-1/} = 1- 1/(5)^{1.4-1/1.4} = 36.86\%$

**When ideal regenerator is incorporated:**

$T_3 = T_5 \times T_2 = T_6$

**Heat supplied** $Q_s = C_p ( T_4 –T_3)$

**rejected** $Q_R = C_p (T_6 –T_1)$

$T_1 = 288k$

$T_2 =T_6 = 456k$

$T_3 = T_5 =740.6k$

$T_4 = 1173k$

$Q_s = 1.005 ( 1173 -790.6) = 434.56 KJ/kg$

$Q_R = 1.005 ( 456 -288 ) =186.84 KJ/kg$

**Efficiency :**

$\eta = 1- Q_R/Q_s = 168.84/434.56 =0.6114 = 61.14\%$

**%change in efficiency :** = 61.14 – 36.86/61.14 = 39.71%

**9. A closed cycle ideal gas plant operates temperature limited of 800°C and 30°C and produces a power of 100Kw.The plant is designed such that there is no need for a regenerator. A fuel of calorific value 45000KJ/kg is used. Calculated the mass flow rate of air through the plant and the rate of fuel combustion take $C_p$ = 1 KJ/kgk and = 1.4**

**Given data:**

T1 = 30°C = 303k ,T3 = 800°C KJ/kg

P = 100Kw ,Cp = 1 KJ /kgk , = 1.4

**Solution:**

**For maximum net work done:**

$T_4 = T_2 =$ T1 xT3 = 1073 x 303 = 570.2k

**Net work done**

$W_{net} = C_p [ ( T_3- T_4 ) - ( T_2 – T_1 ) = 235.6 KJ/kg$

**Total power development**

$P = m_a \times W_{net} = 100/ 235.6 = 0.4244$kg/sec

**Heat supply to the system:**

$m_f \times C_v = m_a \times C_p \times (T_3 - T_2)$

$m_f = m_a \times C_p \times (T_3 - T_2) / C_v$ $0.4244 \times 1 (1073 - 570.2)/45.000 = 4.742 \times 10^{-3}$ kg/s

## INTERNAL COMBUSTION ENGINES

### PRINCIPLES OF OPERATION OF IC ENGINES:

### FOUR-STROKE CYCLE DIESEL ENGINE

In four-stroke cycle engines there are four strokes completing two revolutions of the crankshaft. These are respectively, the suction, compression, power and exhaust strokes. In Fig. 3, the piston is shown descending on its suction stroke. Only pure air is drawn into the cylinder during this stroke through the inlet valve, whereas, the exhaust valve is closed. These valves can be operated by the cam, push rod and rocker arm. The next stroke is the compression stroke in which the piston moves up with both the valves remaining closed. The air, which has been drawn into the cylinder during the suction stroke, is progressively compressed as the piston ascends. The compression ratio usually varies from 14:1 to 22:1. The pressure at the end of the compression stroke ranges from 30 to 45 kg/cm2. As the air is progressively compressed in the cylinder, its temperature increases, until when near the end of the compression stroke, it becomes sufficiently high (650-80O oC) to instantly ignite any fuel that is injected into the cylinder. When the piston is near the top of its compression stroke, a liquid hydrocarbon fuel, such as diesel oil, is sprayed into the combustion chamber under high pressure (140-160 kg/cm2), higher than that existing in the cylinder itself. This fuel then ignites, being burnt with the oxygen of the highly compressed air.

During the fuel injection period, the piston reaches the end of its compression stroke and commences to return on its third consecutive stroke, viz., power stroke. During this stroke the hot products of combustion consisting chiefly of carbon dioxide, together with the nitrogen left from the compressed air expand, thus forcing the piston downward. This is only the working stroke of the cylinder. During the power stroke the pressure falls from its maximum combustion value (47-55kg/cm2), which is usually higher than the greater value of the compression pressure (45kg/cm2), to about 3.5-5 kg/cm2 near the end of the stroke. The exhaust valve then opens, usually a little earlier than when the piston reaches its lowest point of travel. The exhaust gases are swept out on the following upward stroke of the piston. The exhaust valve remains open throughout the whole stroke and closes at the top of the stroke.

The reciprocating motion of the piston is converted into the rotary motion of the crankshaft by means of a connecting rod and crankshaft. The crankshaft rotates in the main bearings, which are set in the crankcase. The flywheel is fitted on the crankshaft in order to smoothen out the uneven torque that is generated in the reciprocating engine.

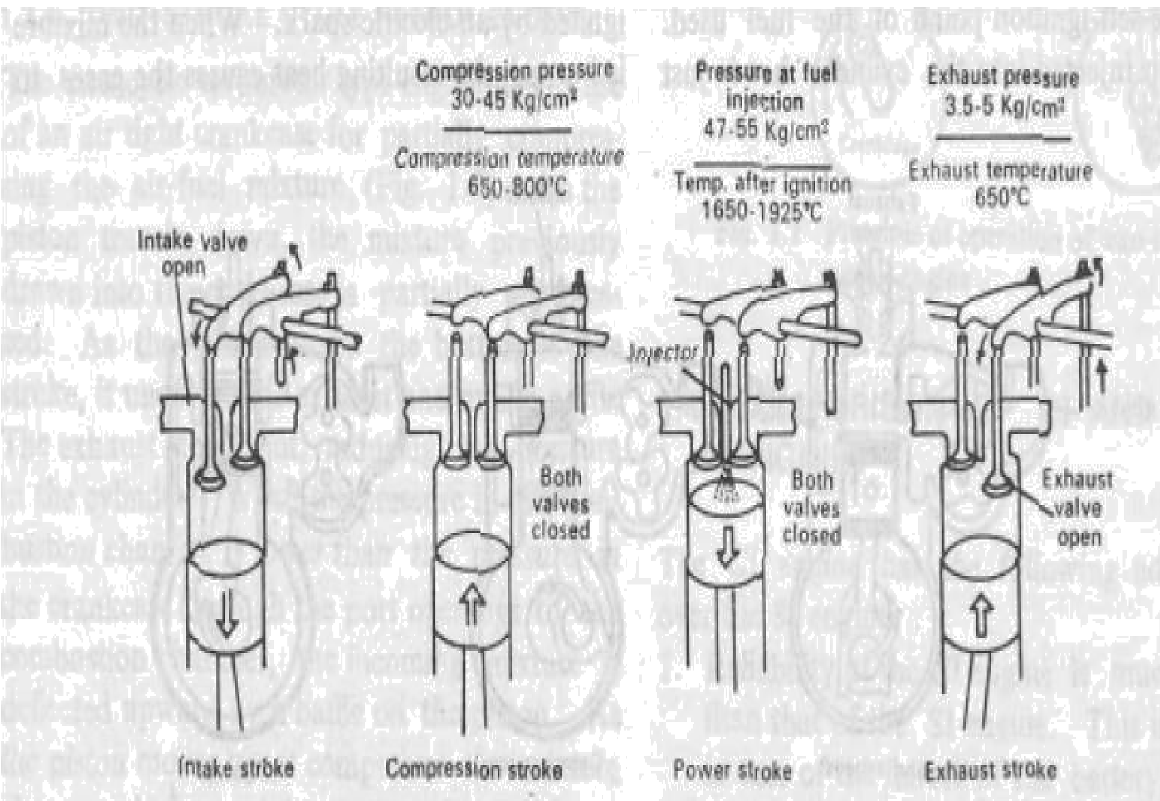



Fig. 3. Principle of four-stroke engine

## FOUR-STROKE CYCLE S-I ENGINE - PRINCIPLE OF OPERATION

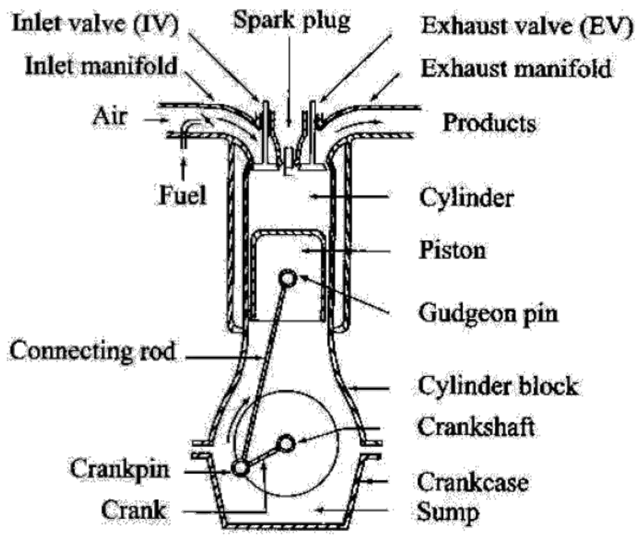



Fig: cross section of a SI Engine

In Four-stroke cycle engine, the cycle of operation is completed in four-strokes of the piston or two revolutions of the crankshaft. Each stroke consists of 180°, of crankshaft rotation and hence a cycle consists of 720°of crankshaft rotation. The series of operations of an ideal four-stroke. SI engine are as follows (see Fig.2.1 & 2.2)

**1. Suction stroke**

Suction stroke 0-1 starts when the piston is at top dead centre and about to move downwards. The inlet valve is open at this time and the exhaust valve is closed. Due to the suction created by the motion of the piston towards bottom dead centre, the charge consisting of fresh air mixed with the fuel is drawn into the cylinder. At the end of the suction stroke the inlet valve closes.

**2. Compression stroke.**

The fresh charge taken into the cylinder during suction stroke is compressed by the return stroke of the piston 1-2. During this stroke both inlet and exhaust valves remain closed. The air which occupied the whole cylinder volume is now compressed into clearance volume. Just before the end of the compression stroke the mixture is ignited with the help of an electric spark between the electrodes of the spark plug located in combustion chamber wall. Burning takes place when the piston is almost at top dead centre. During the burning process the chemical energy of the fuel is converted into sensible energy, producing a temperature rise of about 2000°C, and the pressure is also considerably increased.

**3. Expansion or power stroke.**

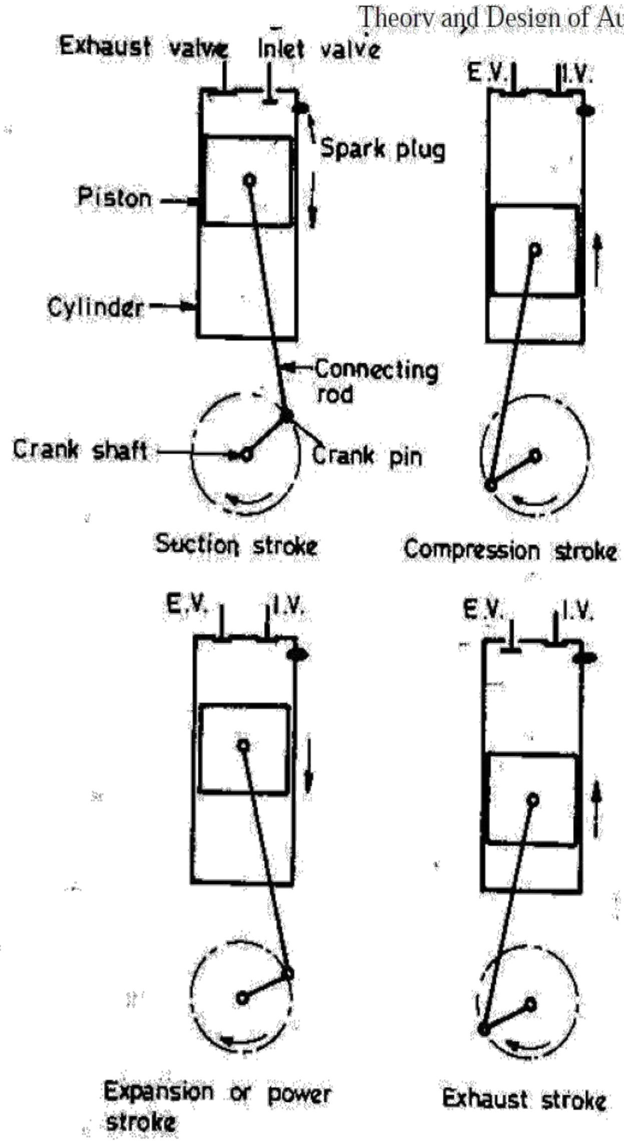
Due to high pressure the burnt gases force the piston towards bottom dead centre, stroke 3-4, and both the inlet and exhaust valves remaining closed. Thus power is obtained during this stroke. Both pressure and temperature decrease during expansion.

**4. Exhaust stroke.**

At the end of the expansion stroke the exhaust valve opens, the inlet valve remaining closed, and the piston is moving from bottom dead centre to top dead centre sweeps out the burnt gases from the cylinder, stroke 4-0. The exhaust valve closes at the end of the exhaust stroke and some 'residual' gases remain in the cylinder.

Each cylinder of a four-stroke engine completes the above four operations in two engine revolutions. One revolution of the crankshaft occurs during the suction and compression strokes, and second revolution during the power and exhaust strokes. Thus for one complete cycle, there is only one power stroke while the crankshaft turns by two revolutions. Most of the spark-ignition internal combustion engines are of the four-stroke type. They are most popular for passenger cars and small aircraft applications.

Theory and Design of Automotive Engines




| *Stroke* | *Valve position* |
|---|---|
| Suction stroke. | Suction valve open.<br>Exhaust valve closed. |
| Compression stroke. | Both valves closed. |
| Expansion stroke. | Both valves closed. |
| Exhaust stroke. | Exhaust valve open.<br>Suction valve closed. |

Fig.2.1-The four-stroke spark-ignition (SI) engine cycle (Otto cycle or constant volume cycle)

## Two Stroke Petrol Engine:

Fig. 2.1 shows a two stroke petrol engine. It has no valves but consists of inlet or induction port (IP), exhaust port (EP), and a third port called the transfer port (TP). Referring to the fig. 2.1 (a) let the piston be nearing the completion of its compression stroke. The ignition starts due the spark given by the spark plug and the piston is pushed down (fig. (b) and (c) ) performing the working strokes and in doing so the air fuel mixture already drawn from the inlet port in the previous stroke is compressed to a pressure of about 1.4 bar. When about $4/5^{th}$ of this stroke is completed the exhaust port (EP) is uncovered slightly and some of the burning gases escape to the atmosphere. Immediately afterwards as the exhaust port is uncovered by the further downward movement of the piston, the transfer port which is only very slightly lower than exhaust port is also uncovered as shown in fig.(d) and a charge of compressed fuel air mixture enters the cylinder and further pushes out the burnt gases out of the exhaust port. The top of the piston is made of a particular shape that facilitates the deflection of fresh charge upwards and thus avoids its escape along with the exhaust gases. This process is known as scavenging.

After reaching the bottom dead center when the piston moves up, it first closes the inlet port, then transfer port and then exhaust port. The charge of fuel which previously entered the cylinder is now compressed. Simultaneously there is a fall of pressure in the crank case, creating a partial vacuum. When the piston is nearing the upward movement, the inlet port opens and a fresh charge of air fuel mixture from the carburetor enters the crank case. After the ignition of the charge, the piston moves down for the power stroke and the cycle is repeated as before.

## Two Stroke Diesel Engine:

In a two stroke cycle C.I. engine all the operations are exactly the same as those in S.I. engine except that in this case only air is taken in instead of air fuel mixture and the fuel is injected at the end of compression stroke, a fuel injector being fitted instead of a spark plug.

## Applications of Internal Combustion Engines:

The main applications of I.C. engines are:

1. Four stroke petrol engine- light vehicles such as cars, jeeps, motor bikes, and small generating sets etc.
2. Two stroke petrol engines- very light vehicles such as scooters, mopeds, three wheelers and portable crop sprayers etc.

3. Four stroke diesel engine- diesel power plants, heavy vehicles such as trucks, buses, road rollers, tractors, diesel locomotives and water pumps.
4. Two stroke diesel engine- mainly used in marine engines where lesser weight is the main consideration.

**Comparison between 2- stroke and 4- stroke cycle engine**

| S. No. | Criteria of comparison | 2- stroke cycle engine | 4- stroke cycle engine |
|---|---|---|---|
| 1. | Power stroke | One working stroke in each cylinder per revolution of crankshaft | One working stroke in each cylinder per two revolutions of crankshaft |
| 2. | Weight and size | Lighter and compact for same power | Heavier and larger |
| 3. | Turning moment | Even and more uniform | Less uniform |
| 4. | Flywheel size | Smaller | Larger |
| 5. | Construction | Simpler and easy to manufacture | More complicated due to valve mechanism |
| 6. | Moving parts | Few in number | More |
| 7. | Mechanical efficiency | More due to lesser moving parts | Lesser |
| 8. | Thermal efficiency | Lesser because a part of air fuel mixture goes as waste with the exhaust gases | More |
| 9. | Noise | More | Lesser |
| 10. | Wear and tear | More due to smaller size for the same power | Less |
| 11. | Scavenging | Required | Most efficient |
| 12. | Fuel consumption | More | Less |

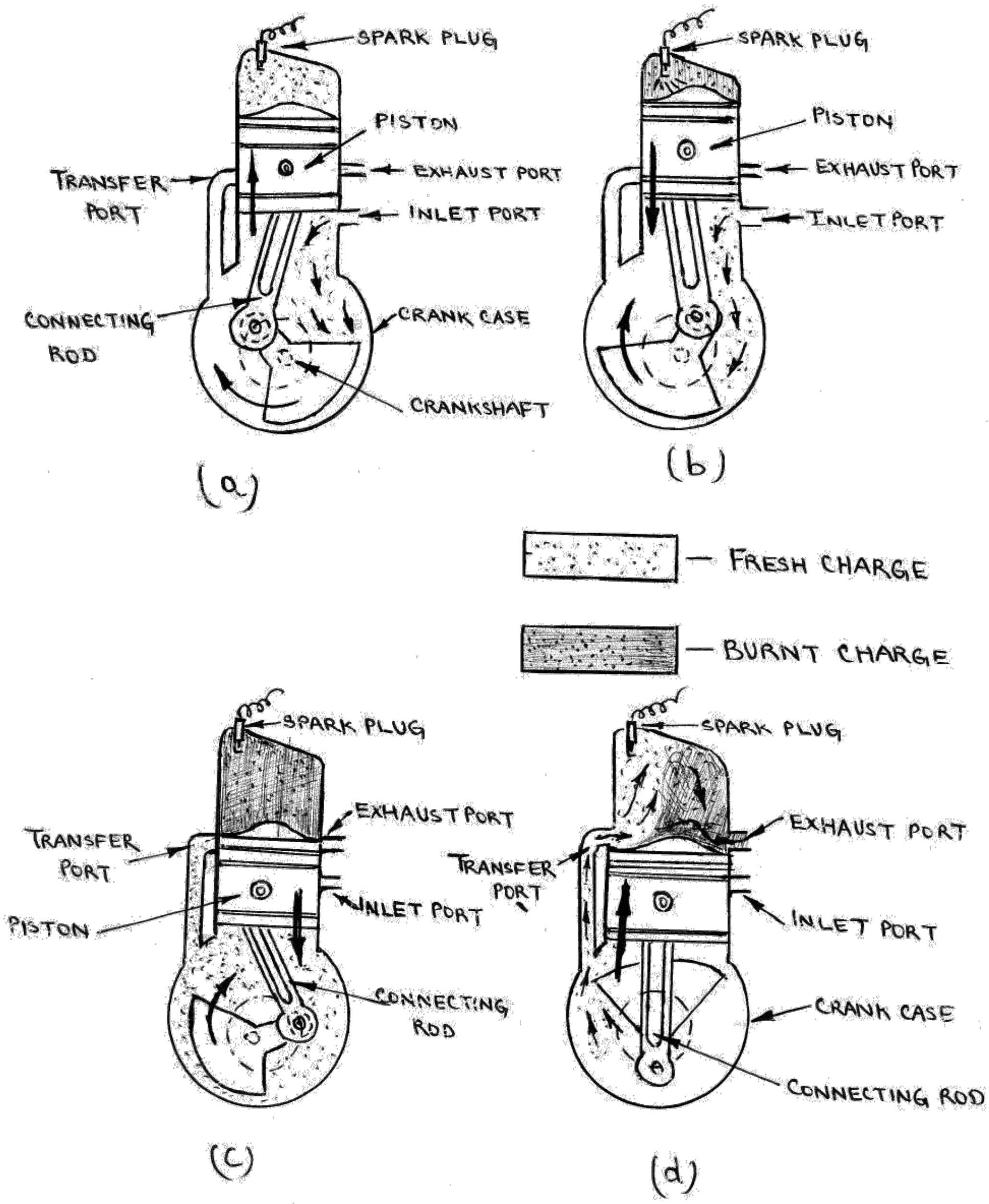



Fig.4.1 Two Stroke Petrol Engine

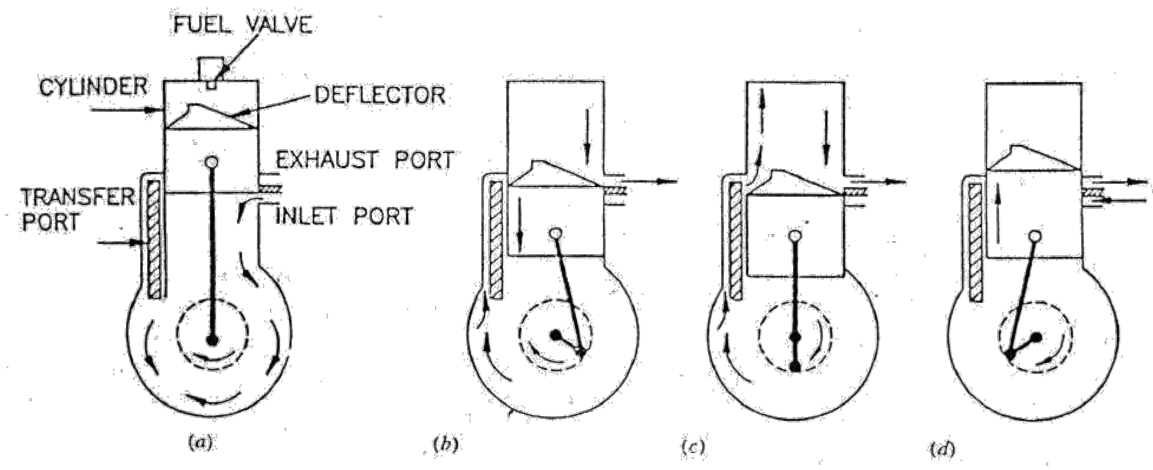



Fig.4.2 Two Stroke Diesel Engine

**PART-A (2 Marks)**

**1. Name the basic thermodynamic cycles of the two types of internal combustion reciprocating engines. [Apr 2003]**
Otto cycle in S.I. engines and diesel cycle in C.I. engines.

**2. Define the terms as applied to reciprocating I.C. engines. "Mean effective pressure" and "Compression ratio". [Nov 2004]**
Mean effective pressure: It is defined as the algebraic sum of the mean pressure acting on the piston during one complete cycle. Compression ratio: Same as previous question.

**3.What is meant by highest useful compression ratio? [Apr 1997]**
The compression ratio which gives maximum efficiency is known as highest useful compression ratio

**4. Why compression ratio of petrol engines is low while diesel engines have high compression ratio? [Oct 1998]**
Since fire point of petrol is less as compared to diesel, petrol engine has low compression ratio.

**5.Compare the thermal efficiency of petrol engines with diesel engines. [Apr 2000]**
Thermal efficiency of diesel engine is greater than petrol engine. This is due to high compression ratio.

**6. Why the actual cycle efficiency is much lower than the air standard cycle efficiency? List and explain the major losses in an actual engine. [Nov 2003]**

Theoretically, the compression and expansion are followed adiabatically. Butt in actual cycle it is not so. Because of the heat and pressure losses are involved. Actual area on p-v diagram per cycle is less than theoretical because of lower pressure rise and pumping losses. Major losses in an actual engine: Heat rejected to the cooling water Heat carried away by exhaust gas Heat loss due to radiation.

**7.What is splash lubrication? [May/Jun 2011]**
In this system, oil is stored in the crank case. A small scoop is attached with the big end of connecting rod. When the crank is rotated, the scoop dips in the oil and splashes the oil. The oil is splashed on cylinder wall, connecting rod ends and valve mechanisms.

**8.Mention different types of fuel injection systems in C.I. Engines. [May/Jun 2011]**
a] Air injection system
b] Airless or solid injection
[i] Common rail system [ii]
Individual pump system.

**9.What do you mean by scavenging in I.C. Engines? Apr 2003 & May/June 2013]**
The process of removing the burnt gases from the combustion chamber of engine cylinder is known as Scavenging.

**10. Define Cetane number. [Apr 2003]**
The property that quantities the ignition delay is called as Cetane number

**11. What is the purpose of a thermostat in an engine cooling system? [Apr 2003]**
A thermostat valve is used in the water-cooling system to regulate the circulation of water in system to maintain the normal working temperature of the engine parts during the different operating conditions.

**12. Explain exhaust blow down in case of IC engines. [Nov 2003]**
The opening of the exhaust valve during the expansion stroke itself, before the piston reaches BDC which enables the exhaust gas to leave under pressure is known as exhaust blow down.

**13. List out the effects of detonation. [Nov/Dec 2011]**
The impact on the engine components and structures may cause failure and creates undesirable noise which is always objectionable.
The lack of control of combustion process leads to pre ignition and local over-heating.
Therefore, piston may be changed by overheating.
The pressure differences in the combustion chamber cause the gas to vibrate and scrub the chamber walls causing increased loss of heat to the coolant.
Detonation results in increased carbon deposits on the wall of the cylinder.

Due to increase in the rate of heat transfer, the power output as well as efficiency of the engine will decrease.

**14. What do we feel the necessity of cooling an IC engine? [May/June 2012]**
When the air-fuel mixture is ignited and combustion takes place at about 25000C for producing power inside an engine the temperature of the cylinder, cylinder head, piston and value, continuous to raise when the engine runs. It these parts are not cooled by some means then they likely to get damaged and even melted. The piston may cease inside the cylinder. To prevent this, the temperature of the parts around the combustion, chamber is maintained as 2000C to 2500C. Too much of cooling will lower the thermal efficiency of the engine. Hence the purpose of cooling is to keep the engine at its most efficient operating temperature at all engine speeds and all driving conditions.

**15. What catalytic converter does? [May/June 2013]**
The diesel engine catalytic convertor is a pure oxidation catalytic converter. It oxidizes HC and CO into water and CO2. It cannot reduce NO2.

**PART-B (12 Marks)**

**1. Comparison of four stroke and two stroke engines. [May 2011]**

| **S.No** | **Four stroke cycle engine** | **Two stroke engine** |
|---|---|---|
| 1 | For every two revolutions of the crank shaft, there is one power stroke i.e after every four piston strokes. | For every one revolution of crank shaft, there is one power stroke i.e after every two piston strokes. |
| 2 | For same power more space is required. | For the same power less space is required. |
| 3 | Valves are required inlet and outlet valves | Ports are made in the cylinder walls inlet, exhaust and transfer port. |
| 4 | As the valves move frequently lubrication is essential. | Arrangement of ports reduce wear and tear and lubrication is not very essential. |
| 5 | These engines are water cooled, making it complicated in design and difficulty to maintain. | These engines are generally air cooled, simple in design and easy to maintain. |
| 6 | The air fuel mixture is completely utilized thus efficiency is higher. | As inlet and outlet port open simultaneous some times fresh charge escapes with the exhaust gases are not always completely removed. |

**2. Comparison of petrol and diesel engines.**

| S.No | Petrol engine | Diesel engine |
|---|---|---|
| 1 | The exhaust is less noisy | The exhaust is noisy due to short time available for exhaust. |
| 2 | Petrol and air admitted into the cylinder during suction stroke. | Air alone is admitted into the cylinder during suction stroke. |
| 3 | Fuel ignition: by spark plug- spark ignition (SI Engine) | By the compressed hot air compression ignition (CI Engine). |
| 4 | Cycle of operation: otto cycle. | Diesel cycle |
| 5 | Compression ratio is low (6 to 8) | High (16 to 18) |
| 6 | Fuel admission through carburetor | Through Fuel injector |
| 7 | Engine speed: can up to 5000rpm | 3500 rpm |
| 8 | Engine starting in cold condition is easy. | Greater cranking effort is required to overcome the higher compression ratio, due to the cold air in the combustion chamber. |
| 9 | Engine life : less than 60000 km | More than 150000 km |

**Table**: Factors tending to reduce knocking in SI and CI engine

| S.No. | Factors | SI Engine | CI Engine |
|---|---|---|---|
| 1 | Self ignition temperature of fuel | High | Low |
| 2 | Time lag or delay period for fuel | Long | Short |
| 3 | Compression ratio | Low | High |

| | | | |
|---|---|---|---|
| 4 | Inlet temperature | Low | High |
| 5 | Inlet pressure | Low | High |
| 6 | Combustion chamber wall temperature | Low | High |
| 7 | Speed | High | Low |
| 8 | Cylinder size | Small | Large |

It is also clear from the table and discussion that a good CI engine fuel is a bad SI engine fuel and a good SI engine is bad CI engine fuel. In other words diesel oil has low self ignition temperature and short time lag where as petrol have high self ignition temperature and along ignition lag. In terms of fuel rating diesel oil has high cetane number (40–60) and low octane number (about30) and petrol has high octane number (80–90) and low cetane number (18).

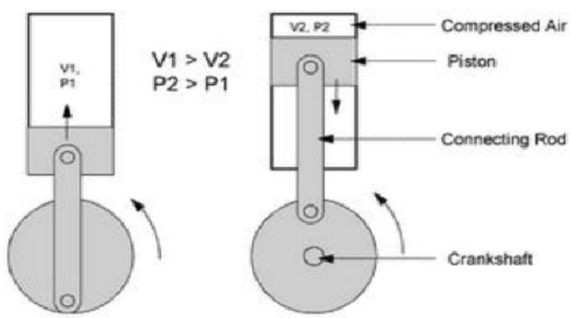
**POSITIVE DISPLACEMENT COMPRESSOR - RECIPROCATING AIR COMPRESSOR, WORK DONE, VOLUMETRIC EFFICIENCY, EFFECT OF CLEARANCE VOLUME FOR QUALITATIVE TREATMENT- ROTARY COMPRESSORS - VANE TYPE, ROOTS BLOWER-CENTRIFUGAL COMPRESSOR.**

**Positive Displacement compressors:**

**Reciprocating Compressor:**

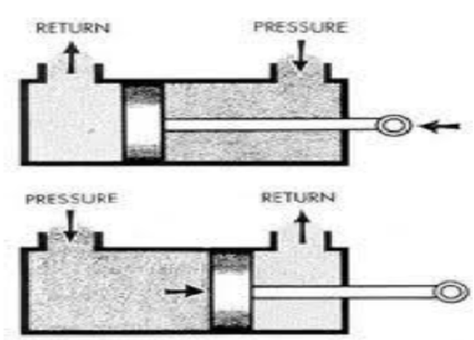
**Single-Acting Reciprocating compressor:**

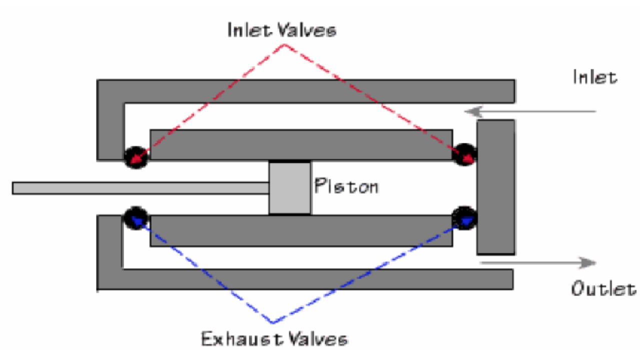
These are usually reciprocating compressors, which has piston working on air only in one direction. The other end of the piston is often free or open which does not perform any work. The air is compressed only on the top part of the piston. The bottom of the piston is open to crankcase and not utilized for the compression of air.




**Double acting compressor:**

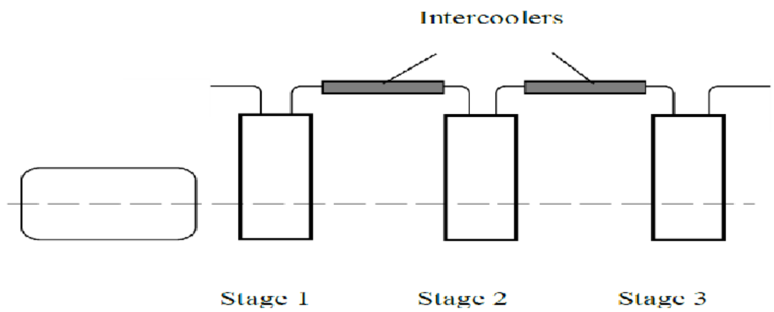
These compressors are having two sets of suction/intake and delivery valves on both sides of the piston. As the piston moves up and down, both sides of the piston is utilized in compressing the air. The intake and delivery valves operate corresponding to the stroke of the compressor. The compressed air delivery is comparatively continuous when compared to a single-acting air compressor. Thus both sides of the pistons are effectively used in compressing the air.



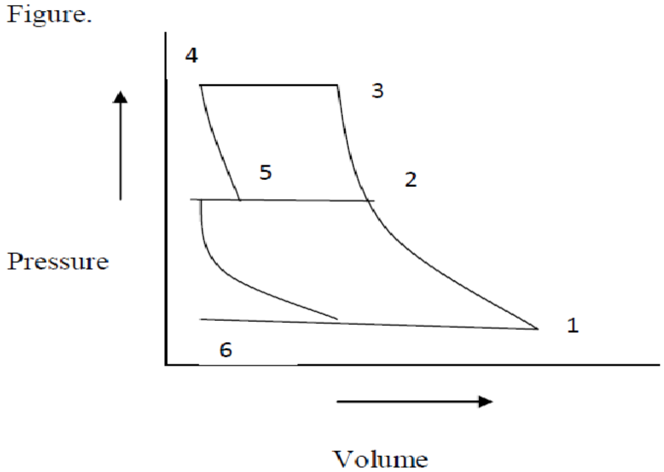




Double Acting Compressor

**Multistage Compression:**




Multistage compression refers to the compression process completed in more than one stage i.e., a part of compression occurs in one cylinder and subsequently compressed air is sent to subsequent cylinders for further compression. In case it is desired to increase the compression ratio of compressor then multi-stage compression becomes inevitable. If we look at the expression for volumetric efficiency then it shows that the volumetric efficiency decreases with increase in pressure ratio. This aspect can also be explained using p-V representation shown in Figure.

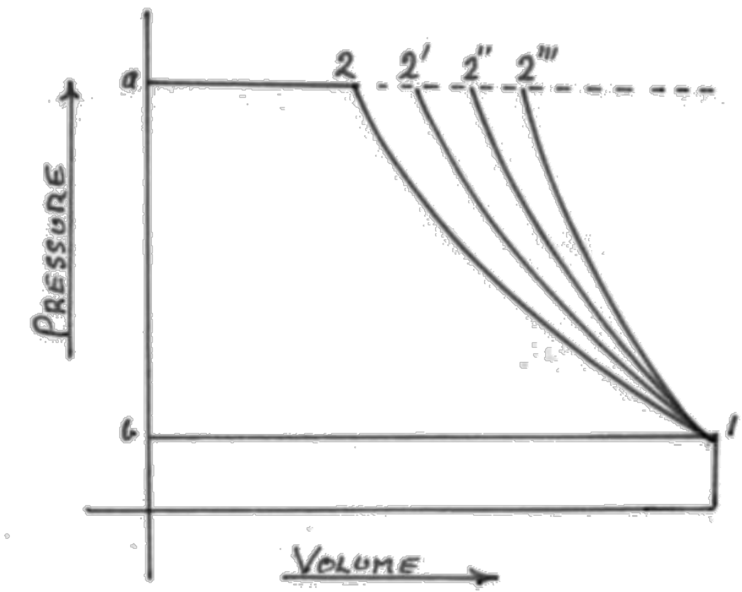



A multi-stage compressor is one in which there are several cylinders of different diameters. The intake of air in the first stage gets compressed and then it is passed over a cooler to achieve a temperature very close to ambient air. This cooled air is passed to the intermediate stage where it is again getting compressed and heated. This air is again passed over a cooler to achieve a temperature as close to ambient as possible. Then this compressed air is passed to the final or the third stage of the air compressor where it is compressed to the required pressure and delivered to the air receiver after cooling sufficiently in an after-cooler.

**Advantages of Multi-stage compression:**

1. The work done in compressing the air is reduced, thus power can be saved
2. Prevents mechanical problems as the air temperature is controlled
3. The suction and delivery valves remain in cleaner condition as the temperature and vaporization of lubricating oil is less
4. The machine is smaller and better balanced
5. Effects from moisture can be handled better, by draining at each stage
6. Compression approaches near isothermal
7. Compression ratio at each stage is lower when compared to a single-stage machine
8. Light moving parts usually made of aluminum, thus less cost and better maintenance

**Work done in a single stage reciprocating compressor without clearance volume:**




Air enters compressor at pressure $p_1$ and is compressed upto $p_2$. Compression work requirement can be estimated from the area below the each compression process. Area on p-V diagram shows that work requirement shall be minimum with isothermal process 1-2''. Work requirement is maximum with process 1-2 ie., adiabatic process. As a designer one shall be interested in a compressor having minimum compression work requirement. Therefore, ideally compression should occur isothermally for minimum work input. In practice it is not possible to have isothermal compression because constancy of temperature during compression can not be realized. Generally, compressors run at substantially high speed while isothermal compression requires compressor to run at very slow speed so that heat evolved during compression is dissipated out and temperature remains constant. Actually due to high speed running of compressor the compression process may be assumed to be near adiabatic or polytropic process following law of compression as $Pv^n=C$ with of 'n' varying between 1.25 to 1.35 for air. Compression process following three processes is also shown on T-s diagram in Fig.16.4. it is thus obvious that actual compression process should be compared with isothermal compression process. A mathematical parameter called isothermal efficiency is defined for quantifying the degree of deviation of actual compression process from ideal compression process. Isothermal efficiency is defined by the ratio is isothermal work and actual indicated work in reciprocating compressor.

$$\text{Isothermal efficiency} = \frac{\text{Isothermal work}}{\text{Actual indicated Work}}$$

Practically, compression process is attempted to be closed to isothermal process by air/water cooling, spraying cold water during compression process. In case of multistage compression process the compression in different stages is accompanied by intercooling in between the stages. $P_2$ $V_2$.

Mathematically, for the compression work following polytropic process, $PV^n$=C. Assuming negligible clearance volume the cycle work done.

$$Wc = \text{Area on p-V diagram}$$

$$Wc = \left[p_2V_2 + \left(\frac{p_2V_2 - p_1V_1}{n-1}\right)\right] - p_1V_1$$

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

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

$$= \left(\frac{n}{n-1}\right)(p_1V_1)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

$$= \left(\frac{n}{n-1}\right)(mRT_1)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right]$$

$$= \left(\frac{n}{n-1}\right)(mR)(T_2 - T_1)$$

In case of compressor having isothermal compression process, n = 1, ie., $p_1V_1 = p_2V_2$

$$W_{iso} = p_2V_2 + p_1V_1 \ln r - p_1V_1$$

$$W_{iso} = p_1V_1 \ln r, \qquad where, r = \frac{V_1}{V_2}$$

In case of compressor having adiabatic compression process,

$$W_{adiabatic} = \left(\frac{\gamma}{\gamma - 1}\right)(mR)(T_2 - T_1) \quad \text{(Or)}$$

$$W_{adiabatic} = (mC_p)(T_2 - T_1)$$

$$W_{adiabatic} = (m)(h_2 - h_1)$$

$$\eta_{iso} = \frac{p_1 V_1 \ln r}{\left(\frac{n}{n-1}\right)(p_1 V_1)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

The isothermal efficiency of a compressor should be close to 100% which means that actual compression should occur following a process close to isothermal process. For this the mechanism be derived to maintain constant temperature during compression process. Different arrangements which can be used are:

(i) Faster heat dissipation from inside of compressor to outside by use of fins over cylinder. Fins facilitate quick heat transfer from air being compressed to atmosphere so that temperature rise during compression can be minimized.

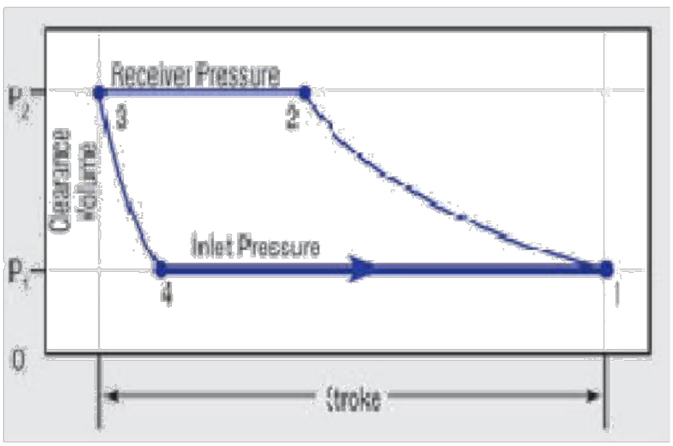
(ii) Water jacket may be provided around compressor cylinder so that heat can be picked by cooling water circulating through water jacket. Cooling water circulation around compressor regulates rise in temperature to great extent.

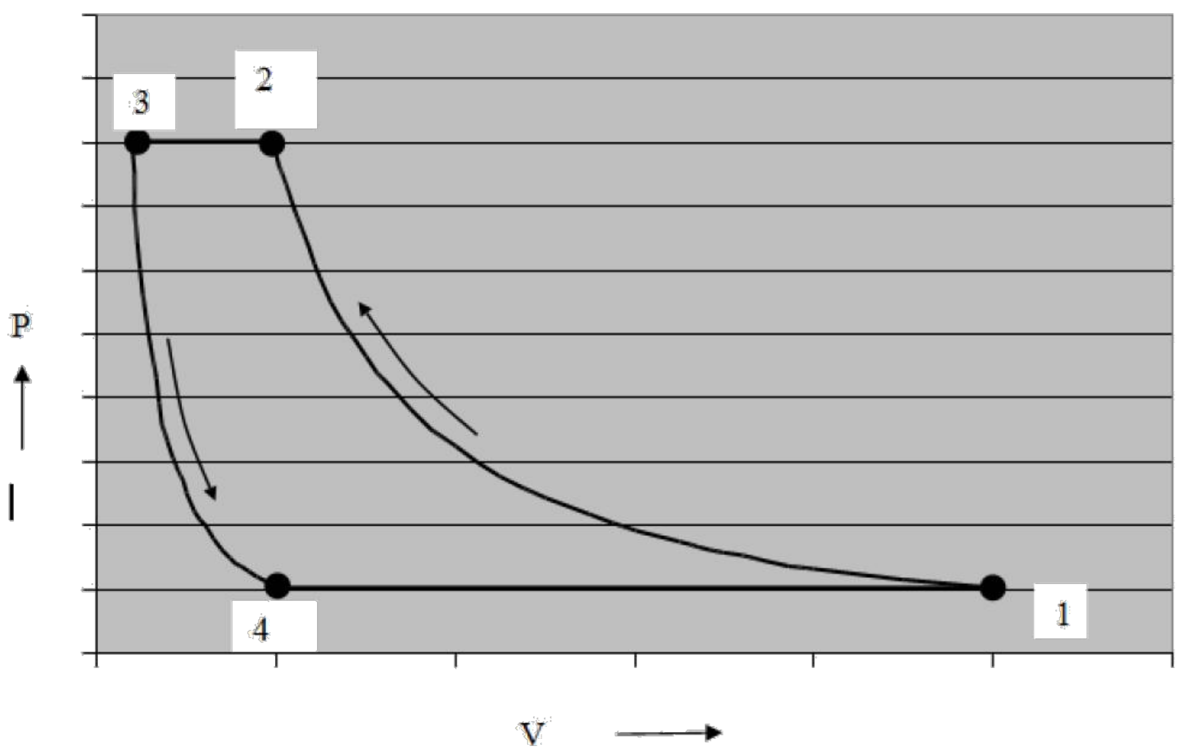
(iii) The water may also be injected at the end of compression process in order to cool the air being compressed. This water injection near the end of compression process requires special arrangement in compressor and also the air gets mixed with water and needs to be separated out before being used. Water injection also contaminates the lubricant film inner surface of cylinder and may initiate corrosion etc, The water injection is not popularly used.

(iv) In case of multistage compression in different compressors operating serially, the air leaving one compressor may be cooled upto ambient state or somewhat high temperature before being injected into subsequent compressor. This cooling of fluid being compressed between two consecutive compressors is called intercooling and is frequently used in case of multistage compressors.

**Work done in a single stage reciprocating compressor with clearance volume:**

Considering clearance volume: With clearance volume the cycle is represented on Figure. The work done for compression of air polytropically can be given by the are a enclosed in cycle 1-2-3-4. Clearance volume in compressors varies from 1.5% to 35% depending upon type of compressor.



$W_{c,\text{with CV}}$ = Area 1234

$$=\left(\frac{n}{n-1}\right)(p_1V_1)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)}-1\right]-\left(\frac{n}{n-1}\right)(p_4V_4)\left[\left(\frac{p_3}{p_4}\right)^{\frac{n-1}{n}}-1\right]$$

Here $P_1 = P_4$, $P_2 = P_3$

$$=\left(\frac{n}{n-1}\right)(p_1V_1)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)}-1\right]-\left(\frac{n}{n-1}\right)(p_1V_4)\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]$$

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

In the cylinder of reciprocating compressor ($V_1$-$V_4$) shall be the actual volume of air delivered per cycle. $V_d = V_1 - V_4$. This ($V_1 - V_4$) is actually the volume of air in hated in the cycle and delivered subsequently.

$$W_{c,withCV}=\left(\frac{n}{n-1}\right)(p_1V_d)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)}-1\right]$$

If air is considered to behave as perfect gas then pressure, temperature, volume and mass can be inter related using perfect gas equation. The mass at state 1 may be given as $m_1$ mass at state 2 shall be m1, but at state 3 after delivery mass reduces to $m_2$ and at state 4 it shall be $m_2$.

So, at state 1, $p_1V_1 = m_1RT_1$

at state 2, $p_2V_2 = m_1RT_2$

at state 3, $p_3V_3 = m_2RT_3$ or $p_2V_3 = m_2RT_3$

at state 4, $p_4V_4 = m_2RT_4$ or $p_1V_4 = m_2RT_4$

Ideally there shall be no change in temperature during suction and delivery i.e., $T_4 = T_1$ and $T_2 = T_3$ from earlier equation

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

Temperature and pressure can be related as,

$$\left(\frac{p_2}{p_1}\right)^{\frac{(n-1)}{n}} = \frac{T_2}{T_1} \quad \text{and} \quad \left(\frac{p_4}{p_3}\right)^{\frac{(n-1)}{n}} = \frac{T_4}{T_3} \quad \Longrightarrow \quad \left(\frac{p_1}{p_2}\right)^{\frac{(n-1)}{n}} = \frac{T_4}{T_3}$$

Substiuting

$$W_{c,withCV} = \left(\frac{n}{n-1}\right)(m_1RT_1 - m_2RT_4)\left[\frac{T_2}{T_1} - 1\right]$$

Substituting for constancy of temperature during suction and delivery.

$$W_{c,withCV} = \left(\frac{n}{n-1}\right)(m_1RT_1 - m_2RT_1)\left[\frac{T_2 - T_1}{T_1}\right]$$

Or

$$W_{c,withCV} = \left(\frac{n}{n-1}\right)(m_1 - m_2)R(T_2 - T_1)$$

Thus $(m_1-m_2)$ denotes the mass of air sucked or delivered. For unit mass of air delivered the work done per kg of air can be given as,

$$W_{c,withCV} = \left(\frac{n}{n-1}\right)R(T_2 - T_1) \quad \text{per kg of air}$$

Thus from above expressions it is obvious that the clearance volume reduces the effective swept volume i.e., the mass of air handled but the work done per kg of air delivered remains unaffected.

From the cycle work estimated as above the theoretical power required for running compressor shall be, For single acting compressor running with N rpm, power input required, assuming clearance volume.

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

For double acting compressor, Power

$$Powerrequired = \left[\left(\frac{n}{n-1}\right)\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] p_1(V_1 - V_4)\right](2N)$$

**Volumetric Efficiency:**

Volumetric efficiency of compressor is the measure of the deviation from volume handling capacity of compressor. Mathematically, the volumetric efficiency is given by the ratio of actual volume of air sucked and swept volume of cylinder. Ideally the volume of air sucked should be equal to the swept volume of cylinder, but it is not so in actual case. Practically the volumetric efficiency lies between 60 to 90%.

Volumetric efficiency can be overall volumetric efficiency and absolute volumetric efficiency as given below.

$$\text{Overall volumetric efficiency} = \frac{\text{Volume of free air sucked in cylinder}}{\text{Swept volume of LP cylinder}}$$

$$(\text{Volumetric efficiency})_{\text{freeaircondition}} = \frac{\text{Volume of free air sucked in cylinder}}{(\text{Swept volume of LP cylinder})_{freeaircondition}}$$

Here free air condition refers to the standard conditions. Free air condition may be taken as 1 atm or 1.01325 bar and 15°C or 288K. consideration for free air is necessary as otherwise the different compressors can not be compared using volumetric efficiency because specific volume or density of air varies with altitude. It may be seen that a compressor at datum level (sea level) shall deliver large mass than the same compressor at high altitude.

This concept is used for giving the capacity of compressor in terms of 'free air delivery' (FAD). "Free air delivery is the volume of air delivered being reduced to free air conditions". In case of air the free air delivery can be obtained using perfect gas equation as,

$$\frac{p_a V_a}{T_a} = \frac{p_1(V_1 - V_4)}{T_1} = \frac{p_2(V_2 - V_3)}{T_2}$$

Where subscript a or $p_a$, $V_a$, $T_a$ denote properties at free air conditions

$$V_a = \frac{p_1 T_a}{p_a} \frac{p_1(V_1 - V_4)}{T_1} = \text{FAD per cycle}$$

This volume $V_a$ gives 'free air delivered' per cycle by the compressor.

Absolute volumetric efficiency can be defined, using NTP conditions in place of free air conditions.

$$\eta_{vol} = \frac{FAD}{Sweptvolume} = \frac{V_a}{(V_1 - V_2)} = \frac{p_1 T_a (V_1 - V_4)}{p_a T_1 (V_1 - V_3)}$$

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1}\right)\left\{\frac{(V_s + V_c) - V_4}{V_s}\right\}$$

Here $V_s$ is the swept volume = $V_1 - V_3$ and $V_c$ is the clearance volume = $V_3$

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1}\right)\left\{1 + \left(\frac{V_c}{V_s}\right) - \left(\frac{V_4}{V_s}\right)\right\}$$

Here $\frac{V_4}{V_s} = \frac{V_4}{V_c} \cdot \frac{V_c}{V_s} = \left(\frac{V_4}{V_3} \cdot \frac{V_c}{V_s}\right)$

Let the ratio of clearance volume to swept volume be given by C. = $\frac{V_c}{V_s}$

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1}\right)\left\{1 + C - C\left(\frac{V_4}{V_3}\right)\right\}$$

$$\eta_{vol} = \left(\frac{p_1 T_a}{p_a T_1}\right)\left\{1 + C - C\left(\frac{p_2}{p_1}\right)^{1/n}\right\}$$

Volumetric efficiency depends on ambient pressure and temperature, suction pressure and temperature, ratio of clearance to swept volume, and pressure limits. Volumetric efficiency increases with decrease in pressure ratio in compressor.

**Mathematical analysis of multistage compressor is done with following assumptions:**

(i) Compression in all the stages is done following same index of compression and there is no pressure drop in suction and delivery pressures in each stage. Suction and delivery pressure remains constant in the stages.

(ii) There is perfect intercooling between compression stages.

(iii) Mass handled in different stages is same i.e, mass of air in LP and HP stages are same.

(iv) Air behaves as perfect gas during compression.

From combined p-V diagram the compressor work requirement can be given as,

Work requirement in LP cylinder, $W_{LP} = \left(\frac{n}{n-1}\right) P_1 V_1 \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}} - 1 \right\}$

Work requirement in HP cylinder, $W_{HP} = \left(\frac{n}{n-1}\right) P_2 V_2 \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}} - 1 \right\}$

For perfect intercooling, $p_1 V_1 = p_2 V_2'$ and

$$W_{HP} = \left(\frac{n}{n-1}\right) P_2 V_{2'} \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}} - 1 \right\}$$

Therefore, total work requirement, $W_c = W_{LP} + W_{HP}$, for perfect inter cooling

$$W_C = \left(\frac{n}{n-1}\right) \left[ P_1 V_1 \left\{ \left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}} - 1 \right\} + P_2 V_{2'} \left\{ \left(\frac{P_{2'}}{P_2}\right)^{\frac{n-1}{n}} - 1 \right\} \right]$$

$$=\left(\frac{n}{n-1}\right)\left[P_1V_1\left\{\left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}}-1\right\}+P_1V_1\left\{\left(\frac{P_{2'}}{P_2}\right)^{\frac{n-1}{n}}-1\right\}\right]$$

$$W_C=\left(\frac{n}{n-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}+\left(\frac{P_{2'}}{P_1}\right)^{\frac{n-1}{n}}-2\right]$$

**Minimum work required in two stage compressor:**

Minimum work required in two stage compressor can be given by

$$W_{C,\min}=\left(\frac{n}{n-1}\right)P_1V_1\cdot 2\left\{\left(\frac{P_2}{P_1}\right)^{\frac{(n-1)}{n}}-1\right\}$$

For *I* number of stages, minimum work,

$$W_{C,\min}=i\cdot\left(\frac{n}{n-1}\right)P_1V_1\left\{\left(\frac{P_{i+1}}{P_i}\right)^{\frac{(n-1)}{n\cdot i}}-1\right\}$$

It also shows that for optimum pressure ratio the work required in different stages remains same for the assumptions made for present analysis. Due to pressure ration being equal in all stages the temperature ratios and maximum temperature in each stage remains same for perfect intercooling.

If the actual volume sucked during suction stroke is $V_1$, $V_2$, $V_3$. . . . for different stages they by perfect gas law, $P_1 V_1 = RT_1$, $P_2 V_2 = RT_2$, Pc, $V_3 = RT_3$

For perfect intercooling

$P_1 V_1 = RT_1$, $p_2 V_2 = RT_1$, $p_3$, $V_3 = RT_1$

$P_1 V_1 = P_2 V_2 = RT_2$, $P_3$, $V_3 =$ . . . . .

If the volumetric efficiency of respective stages in $\eta_{V_1}, \eta_{V_2}, \eta_{V_3}, \ldots$

Then theoretical volume of cylinder1, $V_{1,th} = \frac{V_1}{\eta_{V_1}}; V_1 = \eta_{V_1} \cdot V_{1,th}$

Cylinder 2, $V_{2,th} = \frac{V_1}{\eta_{V_1}}; V_2 = \eta_{V_2} \cdot V_{2,th}$

Cylinder 3, $V_{3,th} = \frac{V_3}{\eta_{V_3}}; V_3 = \eta_{V3} \cdot V_{3,th}$

Substituting,

$$P_1 \cdot \eta_{V_1} \cdot V_{1,th} = P_2 \cdot \eta_{V_2} \cdot V_{2,th} = P_3 \cdot \eta_{V_3} \cdot V_{3,th} = \ldots$$

Theoretical volumes of cylinder can be given using geometrical dimensions of cylinder as diameters $D_1$, $D_2$, $D_3$ . . . and stroke lengths $L_1$, $L_2$, $L_3$ . . . .

Or $$V_{1,th} = \frac{\pi}{4} \cdot D_1^{\,2} \cdot L_1$$

$$V_{2,th} = \frac{\pi}{4} \cdot D_2^{\,2} \cdot L_2$$

$$V_{3,th} = \frac{\pi}{4} \cdot D_3^{\,2} \cdot L_3$$

Or $$P_1 \cdot \eta_{V_1} \cdot \frac{\pi}{4} \cdot D_1^{\,2} \cdot L_1 = P_2 \cdot \eta_{V2} \cdot \frac{\pi}{4} \cdot D_2^{\,2} \cdot L_2$$

$$= P_3 \cdot \eta_{V3} \cdot \frac{\pi}{4} \cdot D_3^{\,2} \cdot L_3 = \ldots$$

$$P_1 \cdot \eta_{V1} \cdot \frac{\pi}{4} \cdot D_1^{\,2} \cdot L_1 = P_2 \cdot \eta_{V2} \cdot \frac{\pi}{4} \cdot D_2^{\,2} \cdot L_2$$

$$= P_3 \cdot \eta_{V3} \cdot D_3^{\,2} \cdot L_3 = \ldots$$

If the volumetric efficiency is same for all cylinders, i.e $\eta_{V1} = \eta_{V2} = \eta_{V3} = \ldots$ and stroke for all cylinder is same i.e. $L_1 = L_2 = L_3 = \ldots$

Then, $D_1^2 P_1 = D_2^2 P_2 = D_3^2 P_3 = \ldots$

These generic relations may be used for getting the ratio of diameters of cylinders of multistage compression.

Energy balance: Energy balance may be applied on the different components constituting multistage compression.

For LP stage the steady flow energy equation can be written as below:

$$m \cdot h_1 + W_{LP} = m \cdot h_2 + Q_{LP}$$

$$Q_{LP} = W_{LP} - m(h_2 - h_1)$$

$$Q_{LP} = W_{LP} - m \cdot C_p (T_2 - T_1)$$

For intercooling (Fig. 5.5) between LP and HP stage steady flow energy equation shall be;

$$m \cdot h_2 = m \cdot h_{2'} + Q_{Int}$$

$$Q_{Int} = m(h_2 - h_{2'})$$

$$Q_{Int} = m \cdot C_p (T_2 - T_{2'})$$

For HP stage (Fig.5.5) the steady flow energy equation yields.

$$m \cdot h_{2'} + W_{HP} = m \cdot h_{3'} + Q_{HP}$$

$$Q_{HP} = W_{HP} + m(h_{2'} - h_{3'})$$

$$Q_{HP} = W_{HP} + m \cdot C_p (T_{2'} - T_{3'}) = W_{HP} - m \cdot C_p (T_{3'} - T_{2'})$$

In case of perfect intercooling and optimum pressure ratio, $T_{2'} = T_1$ and $T_2 = T_{3'}$

Hence for these conditions,

$$Q_{LP} = W_{LP} - m \cdot C_p (T_2 - T_1)$$

$$Q_{Int} = m \cdot C_p (T_2 - T_1)$$

$$Q_{HP} = W_{HP} - m \cdot C_p (T_2 - T_1)$$

Total heat rejected during compression shall be the sum of heat rejected during compression and heat extracted in intercooler for perfect intercooling.

Heat rejected during compression for polytropic process $= \left(\frac{\gamma - n}{\gamma - 1}\right) \times Work$

**Sample Problems:**

1. A single stage single acting air compressor is used to compress air from 1.013 bar and 25° C to 7 bar according to law PV1.3 = C.The bore and stroke of a cylinder are 120mm and 150mm respectively. The compressor runs at 250 rpm .If clearance volume of the cylinder is 5% of stroke volume and the mechanical efficiency of the compressor is 85%, determine volumetric efficiency, power, and mass of air delivered per minute.

2. A two stage singe acting air compressor compresses 2m3 air from 1 bar and 20° C to 15 bar. The air from the low pressure compressor is cooled to 25° C in the intercooler. Calculate the minimum power required to run the compressor if the compression follows PV1.25=C and the compressor runs at 400 rpm.

3. A single stage single acting air compressor is used to compress air from 1 bar and 22° C to 6 bar according to the law PV1.25 = C. The compressor runs at 125 rpm and the ratio of stroke length to bore of a cylinder is 1.5. If the power required by the compressor is 20 Kw, determine the size of the cylinder.

**PART-A (2 Marks)**

**1.Classify the various types of air-compressors. [Dec 2003]** a.According to the design and principle of operation.
i] Reciprocating compressors.
ii] Rotary compressors.
b. According to the action
i] Single acting compressors.
ii] Double acting compressors.
c.According to the number of stages
i] Single stage compressors.
ii] Multistage compressors.
d. According to the pressure limit
i] Low pressure compressors.
ii] Medium pressure compressors.

iii] High pressure compressors.
e.According to the capacity
i] Low capacity compressors [Volume delivered 0.15 m3/s or less].
ii] Medium capacity compressors [Volume delivered 0.15m3/s to 5m3/s].
iii] High capacity compressors [Volume delivered is above 5m3/s].

**2. Indicate the applications of reciprocating compressors in industry. [Nov 2004]**
The applications of compressed air are as follows:
1. Pneumatic brakes, 2. Pneumatic drills, 3. Pneumatic jacks, 4. Pneumatic lifts, 5. Spray painting, 6. Shop cleaning, 7. Injecting fuel in diesel engines, 8. Supercharging internal combustion engines, 9. Refrigeration, and air conditioning systems.

**3. What are the advantages of multi stage compression with inter cooling over single stage compression for the same pressure ratio? [May 2003, Nov 2002 & Apr 2004]**
a.The work done per kg of air is reduced in multistage compression with inter cooler as compared with single stage compression for the same delivery pressure. b.It improves the volumetric efficiency for the given pressure ratio.
c.The size of the cylinders [i.e. high pressure and low pressure] may be adjusted to suit the volume and the pressure of the air.
d.It reduces the leakage loss considerably.
e. It gives more uniform torque and hence, a smaller size flywheel is required.
f. It provides effective lubrication because of lower operating temperature.
g.It reduces the cost of the compressor.

**4. What is meant by free air delivered? [Dec 2003, Nov/Dec 2011]**
The free air delivered is the actual volume delivered at the stated pressure reduced to intake pressure and temperature and expressed in m3/min.

**5. What for inter cooling is used in compressors? [Nov/Dec 2010]**
An inter cooling is a simple heat exchanger. It exchanges the heat of compressed air from the low-pressure compressor to the circulating water before the air enters to the high pressure compressor. The purpose of inter cooling is to minimize the work of compression.

**6. Define mechanical efficiency and isothermal efficiency of a reciprocating compressor. [May/June 2011]**
**Mechanical Efficiency**: Mechanical efficiency is defined as the ratio between brake power to the indicated power. Mechanical Efficiency = Brake power/Indicated power.
**Isothermal Efficiency**: Isothermal efficiency is defined as the ratio between isothermal work to the actual work of the compressor. Isothermal Efficiency = Isothermal work/Actual work.

**7. What is the difference between perfect inter cooling and imperfect inter cooling? [May/June 2011]**

**Perfect cooling**: When the temperature of air leaving the inter cooler is equal to the original atmospheric air temperature, then inter cooling is known as perfect inter cooling.
**Imperfect Inter cooling**: When the temperature of air leaving the inter cooling is more than original atmospheric air temperature, then inter cooling is known as imperfect inter cooling

**8. What factors limit the delivery pressure in a reciprocating compressor? [Oct 1997]**
a. To obtain high delivery pressure, the size of the cylinder will be large.

b. Temperature of air.

**9. Why clearance is necessary and what is its effect on the performance of reciprocation compressor? [Apr 1998 & Oct 1999]**
When the piston reaches top dead center in the cylinder, there is a dead space between piston top and cylinder head. This space is known as clearance space and the volume occupied by this space is known as clearance volume.

**10. Give the expression for work done for a two-stage compressor with perfect inter cooling. [May/June 2013]**
W = [2n/n-1]p1V1{[p3/p1][n-1/2n] – 1}

**11. What are the factors that affect the volumetric efficiency of a reciprocating compressor? [Apr 1998]**
a. Clearance volume. b. Compression ratio.

**12. Discuss the effect of clearance upon the performance of an air compressor. [May 2003]**
The volumetric efficiency of air compressor increases with decrease in clearance of the compressor. The free air delivered by the compressor is increased by decreasing the clearance volume.

**13. Give two merits of rotary compressor over reciprocating compressor. [Apr1998]**
a. Rotary compressor gives uniform delivery of air when compared to reciprocating compressor.
b. Rotary compressors are small in size for the same discharge as compared with reciprocating compressors.
c. Lubricating system is more complicated in reciprocating compressor where as it is very simple in rotary compressor.

**14. Give two examples for positive displacement rotary compressor?**
1. Roots blower 2. Van blower

**15. List out the application of compressed air [May/June 2012]**
Compressed air is mostly used in pneumatic brakes, pneumatic drills, pneumatic jacks, neumatic lifts, spray painting, shop cleaning, injecting fuel in diesel engines, supercharging, internal combustion engines, refrigeration and air conditioning systems

**16. Define volumetric efficiency of a reciprocating compressor. [June 2013]**
Volumetric efficiency is defined as the ratio of volume of free air sucked into the compressor per cycle to the stroke volume of the cylinder.
$\eta vol$ = Volume of free air taken per cycle / Stroke volume of the cylinder.

**PART-B (12 Marks)**

**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. [NOV 2002]**

**Given data:**

$D = 160mm = 0.16m$
$L = 300mm = 0.3m$
$P_1 = 100kpa$
$T_1 = 27°C = 27 + 273 = 300K$
$P_2 = 650kpa$
$N = 2rev/sec = 120rpm$
$PV^{\gamma} = C \quad \gamma = 1.4$

**Solution:**
**Work done during Isothermal Compression** (PV = C)

$W = mRT_1 \ln [P_2/ P_1]$
$W = P_1V_1 \ln [P_2/ P_1] \qquad [PV = mRT]$

We know that,

$Vs=(\pi/4)D^2L =(\pi/4) *(0.16)^2*0.3$
$Vs=6.03X10^{-3}m^3 = V_1 \qquad$ [clearance volume is neglected]
$Vs=6.03X10^{-3}m^3$

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

$P_1V_1= mRT_1$

$m = P_1V_1/R\,T_1 = [(100*6.03\times10^{-3})/(0.287*300)]$ $m = 0.007kg$

**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? [May 2004]**

**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,

$Vs=(\pi/4)D^2L =(\pi/4) *(12)^2*16$

$Vs= 0.0018m^3$

We know that,

$V_1=Vc+Vs$

$V_1= 0.06Vs+Vs$

$V_1=1.06\times0.0018$

$V_1=1.908\times10^{-3}\ m^3$

**Work done on the single stage compressor with clearance volume**

$W= [n/n-1]\ P_1\ V_1\ [(P_2/\ P_1)^{(n-1/n)}-1]$

We know that,

$P_3V_3^n= P_4V_4^n$

$[V_4/\ V_3]^n=[\ P_3/\ P_4]$

$[V_4/\ V_3]^n=[\ P_2/\ P_1]$

$[V_4/\ V_c\ ]^n=[\ P_2/\ P_1]$

$[V_4/V_c]=[P_2/P_1]^{1/n}$
$V_4 = V_c \times [P_2/P_1]^{1/n}$
$=0.06 \times Vs[600/98]^{1/1.32}$
$=0.06 \times 0.0018 \times [600/98]^{1/1.32}$
$V_4=4.26 \times 10^{-4} m^3$
We know that,
$Va = V_1 - V_4 = 1.908 \times 10^{-3} - 4.26 \times 10^{-4}$
$Va = 0.00148\ m^3$
Substituting Va value in work done equation
$W=[1.32/1.32-1] \times 98 \times 0.00148[(600/98)^{1.32-1/1.32}-1]$ W = 0.329 kJ
Power = WxN/60 = (0.329x410)/60
P = 2.25 kW
We know that,
$PoVo/To = P_2V_d/T_2$
$Vo = To/Po \times P_2V_d/T_2$
We know that,
$T_2/T_1=[P_2/P_1]^{n-1/n}$
$T_2=T_1 \times [P_2/P_1]^{n-1/n}$

$T_2=313 \times [600/98]^{1.32-1/1.32}$
$T_2=485.6K$
$[V_2/V_1]^n= P_1/P_2$
$V_2/V_1=[P_1/P_2]^{1/n}$
$V_2 = V_1 [P_1/P_2]^{1/n}$
$V_2 = 1.908 \times 10^{-3} [98/600]^{1/1.32}$
$V_2 = 0.00048\ m^3$
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, To, Po, $P_2$,$T_2$, $V_d$ values in ..(1)
Vo= (293/100)x (600/485.6)x0.000372
$Vo = 0.0013\ m^3$

**Result:**

P = 2.25 kW $^{n-1/n}$ Vo = 0.0013 $m^3$

**3. A single stage reciprocating compressor receives air at 25m$^3$/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
$\eta_{vol}$ = 0.75

**Solution :**

work done on the single stage compressor with clearance volume,
$W = n/n\text{-}1 P_1 V_a [ ( P_2/P_1)^{n-1/n} -$
$1]$ 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,

$T_2/T_1=[ P_2/P_1]$ $^{1.3-1/1.3}$-1]
$T_2= T_1 x[ P_2/P_1]^{n-1/n}$
$T_2= 288 x[ 1500/100]^{1.35-1/1.35}$
$T_2$= 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 $m^3$ 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 $m^3$/min
$P_1$= 1bar=100kpa
$T_1$=15°C=288K
$P_2$=7bar = 700kpa
N=300rpm
L/D=1.5
$\eta_{mech}$=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)
= 244.6kJ/min

**Indicated Power = 4.07kW**
We know that,
Stroke volume, $V_s= V_1= (\pi/4)D^2L$
$1/300=(\pi/4)xD^2x1.5D$
$1/300=(\pi/4)x1.5D^3$
D = 0.141m
L= 1.5x0.141
L = 0.212m
We know that,
$\eta_{mech}$= (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 = 4.07kW
Power input = 4.79kW
Motor power = 5.32kW

**5. The free air delivered of a single cylinder single stage reciprocating air compressor 2.5 $m^3$/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. [Dec 2003]**

**Given data:**
$V_a = 2.5\ m^3/min= 0.04166\ m^3/sec$
For STP condition, the pressure and temperature are
$V_1=1\ m^3/min$
$P_1$= 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
$m_a=PV/RT$
$=(1.013x10^5x0.04166)/(287x288)=0.051kg/sec$ We know that,

$W = m_aRT(n/n-1) [ ( P_2/P_1)^{n-1/n} -1]$
$W = 0.054x0.287x288x (1.25/1.25-1) [ ( 700/101.3)^{1.25-1/1.25} -1]$ $W = 9.95kW$

We know that,
Indicated power, IP =PmLAN/1000
$Pm = (n/n-1) P_1x\eta_{vol}[ ( P_2/P_1)^{n-1/n} -1]$
But, $\eta_{vol}=1+C-C (P_2/P_1)^{1/n}$
Where $C = Vc/Vs$
$\eta_{vol}=1+ (Vc/Vs) – (Vc/Vs) (P_2/P_1)^{1/n}$
$\eta_{vol}=1+ (0.05) – (0.05) (700/101.3)^{1/1.25}$
$\eta_{vol}= 0.815$

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**

$1.95 = [1.923x10^5 x 1.2 D x (\pi/4)D^2 x 150/60] / 1000$

$D = 0.28m$

**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 14m$^3$/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. [Apr 2004]**

**Given data:**

$V_0$=14m$^3$/min = 0.233 m$^3$/sec
$P_1$= 0.95bar=95kpa
$P_2$= 7bar = 700kpa
$T_1$=32°C=305K
$T_0$=15°C=288K
$P_0$=1.013bar = 101.3kpa
Vc=5%Vs=0.05Vs Vc/Vs= 0.05
n = 1.3

**Solution:**

**Volumetric efficiency**, $\eta_{vol}=1+C-C\,(P_2/P_1)^{1/n}$

$\eta_{vol}=1+(Vc/Vs)-(Vc/Vs)\,(P_2/P_1)^{1/n}$

$\eta_{vol}=1+(0.05)-(0.05)\,(700/95)^{1/1.3}$

$\eta_{vol}= 0.818 = 81.8\ \%$

We know that,

$P_o V_o/T_o = P_1 V_1 / T_1$

$101.3 \times 0.233/288 = 95 \times V_a /305$

$V_a = 0.263 m^3/sec$

**Work done or power,**

$P = (n/n-1)P_1 V_a\,[\,(P_2/P_1)^{n-1/n} -1]$

$P = (1.3/1.3-1) \times 95 \times 0.263\,[\,(700/95)^{1.3-1/1.3} -1]$

P = 63.39 kW

**Result:**

$\eta_{vol}= 81.8\ \%$

Indicated power P = 63.39 kW

**7. A single cylinder single acting reciprocating compressor takes in 6m$^3$/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 = 100kpa
$T_1$= 15° C = 288K

$P_2$= 6bar =600kpa

**Solution:**

Work done during isothermal compression (pv=c)
W= $P_1 V_1 \ln[P_2/ P_1]$
= 100*6*ln[600/100]
**W = 1075.5kJ/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= $[1.4/1.4-1] *100*6* [(600/ 100)^{(1.4-1/1.4)}-1]$
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=0.3$m^3$/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
Po = 101.3kpa
To = 298 K
Substituting To,Po,Vo, $P_1$, $V_1$ values in eqn (1)
$V_1$ =[(101.3x0.3)/298]x[288/101.3]
**$V_1$=0.289$m^3$/min**
Work done duringisentropic Compression
W= $[\gamma/\gamma-1] P_1 V_1 [(P_2/ P_1)^{(\gamma-1/\gamma)}-1]$
W= $[1.4/1.4-1] *101.3*0.289* [(700/ 101.3)^{(1.4-1/1.4)}-1]$
W= 75.53kJ/min
W= 1.25kJ/s

**$P_{Iso}$=1.25kW**

Work done during polytropic compression

$W= [n/n-1]\ P_1\ V_1\ [(P_2/\ P_1)^{(n-1/n)}-1]$

$W= [1.25/1.25-1]\ x101.3x\ 0.289x\ [(700/\ 101.3)^{(1.25-1/1.25)}-1]$

W = 69.08kJ/min

**$P_{poly}$=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

$\eta_{vol}$=80%

$\eta_{max}$=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)
$W = [1.3/1.3-1] x 100 x 3.3 [ ( 247.03/100)^{1.3-1/1.3} -$
$1]$ $W = 334.87 kJ/min$
**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
$Va = Vs \times \eta_{vol} \times 300$
$15 = Vs \times 0.753 \times 300$
**$Vs = 0.0664\ m^3$**

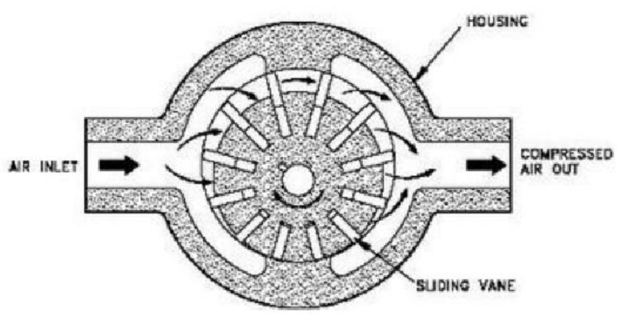
Stroke volume = 0.0664 $m^3$
We know that,
$V_s = (\pi/4)D^2L = 0.0664$
$(\pi/4)D^2 \times 1.5D = 0.0664$
**D = 0.3834 m**
We know that,
$L/D = 1.5$
$L = 1.5 \times 0.3834$
**L = 0.5751m**

## Rotary compressors:

### Vane Type compressor:

The rotary slide vane-type, as illustrated in Figure, has longitudinal vanes, sliding radially in a slotted rotor mounted eccentrically in a cylinder. The centrifugal force carries the sliding vanes against the cylindrical case with the vanes forming a number of individual longitudinal cells in the eccentric annulus between the case and rotor. The suction port is located where the longitudinal cells are largest. The size of each cell is reduced by the eccentricity of the rotor as the vanes approach the discharge port, thus compressing the air.

This type of compressor, looks and functions like a vane type hydraulic pump. An eccentrically mounted rotor turns in a cylindrical housing having an inlet and outlet. Vanes slide back and forth in grooves in the rotor. Air pressure or spring force keeps the tip of these vanes in contact with the housing. Air is trapped in the compartments formed by the vanes and housing and is compressed as the rotor turns.




**Non-Positive displacement compressors or Dynamic compressor:**

**Centrifugal Compressor:**

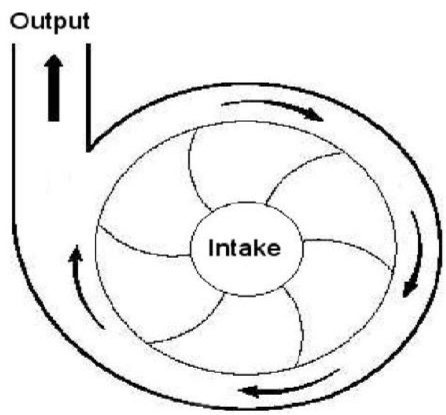
The centrifugal air compressor is a **dynamic** compressor which depends on transfer of energy from a **rotating impeller** to the air.

Centrifugal compressors produce high-pressure discharge by converting angular momentum imparted by the rotating impeller (dynamic displacement). In order to do this efficiently, centrifugal compressors rotate at higher speeds than the other types of compressors. These types of compressors are also designed for higher capacity because flow through the compressor is continuous.

Adjusting the inlet guide vanes is the most common method to control capacity of a centrifugal compressor. By closing the guide vanes, volumetric flows and capacity are reduced.

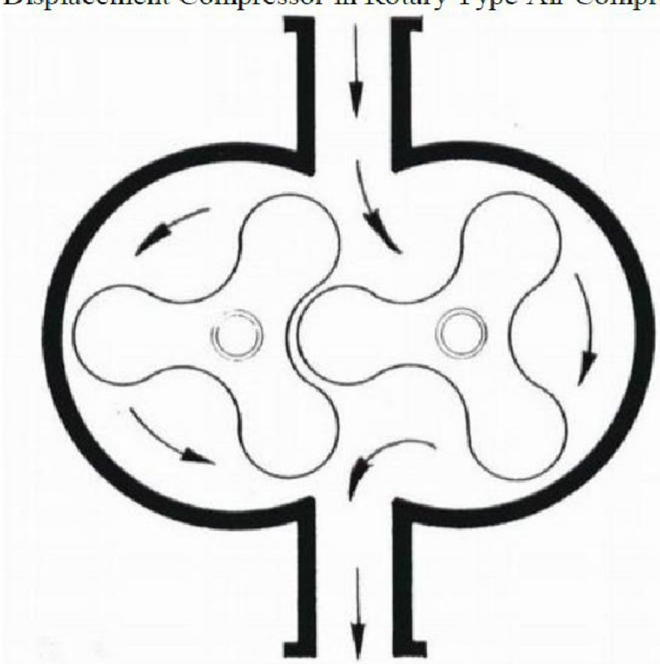
The centrifugal air compressor is an oil free compressor by design. The oil lubricated running gear is separated from the air by shaft seals and atmospheric vents.

The centrifugal air compressor is a dynamic compressor which depends on a rotating impeller to compress the air. In order to do this efficiently, centrifugal compressors must rotate at higher speeds than the other types of compressors. These types of compressors are designed for higher capacity because flow through the compressor is continuous and oil free by design.




Impeller

**Roots Blower Compressor:**
This type is generally called as blower. The discharge air pressure obtained from this type of machine is very low. The Discharge Pressure of 1 bar can be obtained in Single Stage and pressure of 2.2 bar is obtained from Stage. The discharge pressure achieved by two rotors which have separate parallel axis and rotate in opposite directions. This is the example of Positive Displacement Compressor in Rotary Type Air Compressor.



**HEAT TRANSFER-MODES OF HEAT TRANSFER- FOURIER LAW OF CONDUCTION, ONE DIMENSIONAL STEADY STATE CONDUCTION HEAT TRANSFER IN COMPOSITE WALLS. FOR QUALITATIVE TREATMENT- CONVECTION AND RADIATION VAPOUR COMPRESSION REFRIGERATION CYCLE, CALCULATION OF COEFFICIENT OF PERFORMANCE.**

## 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, and gases. 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

2.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:

$$\text{rate of a transfer process} = \frac{\text{driving force}}{\text{resistance}} \quad \text{-----------} \quad (1)$$

This equation states what we know intuitively: that in order to transfer a property 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:

$$\frac{q_x}{A} = -k\frac{dT}{dx} \quad \text{---------------} \quad (2)$$

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 $x2\ x1$ m away. Rearranging eq. (2),

$$\frac{q_x}{A}\int_{x_1}^{x_2} dx = -k \int_{T_1}^{T_2} dT \qquad \text{-------(3)}$$

Integrating, assuming that k is constant and does not vary with temperature and dropping the subscript $x$ on $q_x$ for convenience, 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 $x2\ x1$ m away. Rearranging eq. (2),

$$\frac{q_x}{A}\int_{x_1}^{x_2} dx = -k\int_{T_1}^{T_2} dT \qquad \text{-------(3)}$$

Integrating, assuming that k is constant and does not vary with temperature and dropping the subscript $x$ on $q_x$ for convenience,

$$\frac{q}{A} = \frac{k}{x_2 - x_1}(T_1 - T_2) \qquad \text{-------- (4)}$$

**2. Convection Heat Transfer.** The transfer of heat by convection implies the transfer of heat by bulk 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 convection in 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 $\sigma$ is the proportionality constant and is called the Stefan-Boltzmann constant with the value of 5.669 x $10^{-8}$ 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

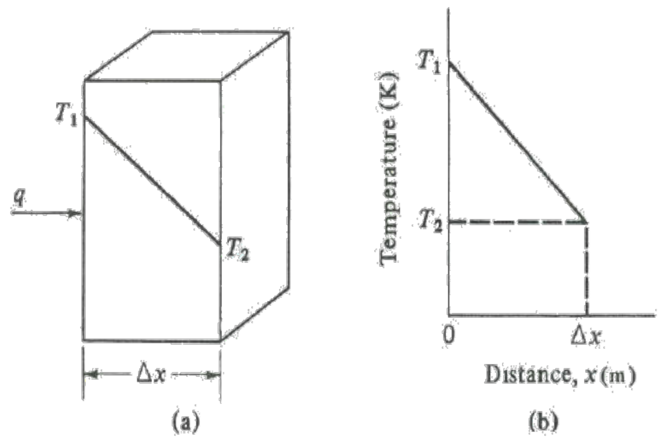



Fig. 3.1 Heat conduction in a flat wall: (a) geometry of wall, (b) temperature plot.

### 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. $\frac{q}{A} = \frac{k}{x_2 - x_1}(T_1 - T_2)$ 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,

$$\frac{q}{A}=\frac{a+b\frac{T_1+T_2}{2}}{\Delta x}(T_1-T_2)=\frac{k_m}{\Delta x}(T_1-T_2)$$

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 $\frac{q}{A}=\frac{k}{\Delta x}(T_1-T_2)$ 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 $\frac{q}{A}=\frac{k}{\Delta x}(T_1-T_2)$ 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

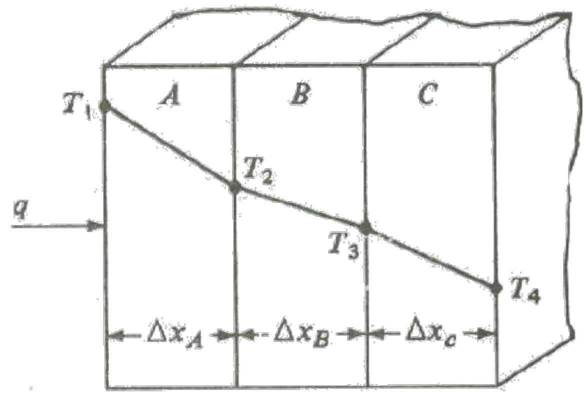



Fig. 4.1 Heat flow through multilayer wall

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 $\Delta T = T_2 - T_1 = -\frac{q \cdot \Delta x}{KA}$

$$T_1 - T_2 = \frac{x_1}{K_1\,A} \cdot q$$

$$T_2 - T_3 = \frac{x_2}{K_2\,A} \cdot q$$

$$T_4 - T_3 = \frac{x_3}{K_3\,A} \cdot q$$

Total temperature difference, $\Delta T = \Delta T_1 + \Delta T_2 + \Delta T_3$

$$T_1 - T_4 = q.\left[\frac{x_1}{K_1 A} + \frac{x_2}{K_2 A} + \frac{x_3}{K_{31} A}\right]$$

where, $T_1 - T_4 =$ thermal potential responsible for heat flow. The

$\left[\frac{x_1}{K_1 A} + \frac{x_2}{K_2 A} + \frac{x_3}{K_{31} A}\right]$ 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.

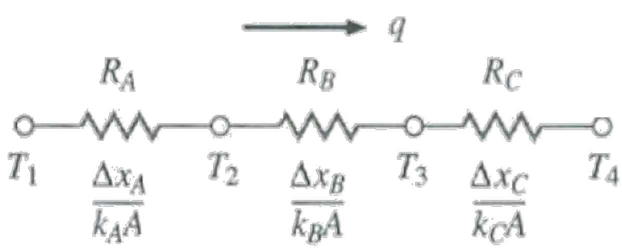



Fig 4.2 Electrical analog of one dimensional heat transfer through composite wall

$$q = \frac{T_1 - T_4}{R_1 + R_2 + R_3}$$

**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 W/m²

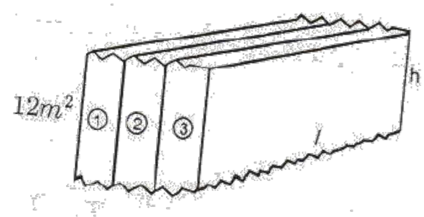
1. **Calculate the rate of heat loss through the vertical walls of a boiler furnace of size 4 m by 3 m by 3 m high. The walls are constructed from an inner fire brick wall 25 cm thick of thermal conductivity 0.4 W/mK, a layer of ceramic blanket insulation of thermal conductivity 0.2 W/mK and 8 cm thick, 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 60$^0$ C. Also find the interface temperature of layers.**

**Given:**

**Composite Wall**

**l= 4m b= 3m h= 3m**

**Area of rectangular wall lb = 4x3 = 12m$^2$**




$L_1$ = 25 cm, $k_1$ = **0.4 W/mK** } Fire brick

$L_2$ =0.002m, $k_2$ = **54 W/mK** } Steel

$L_3$ = 0.08 m, $k_1$ = **0.2 W/mK** } insulation

**$T_1$ = 600$^0$ C**

**$T_2$ = 60$^0$ C**

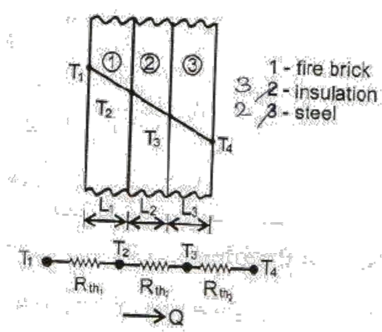
**Find**

(i) Q (ii) ($T_3$–$T_4$)

**Solution**

We know that,

$$Q = \frac{(\Delta T)_{overall}}{\Sigma R_{th}}$$




Here

$(\Delta T)$ overall = $T_1 - T_4$

And $\Sigma R_{th} = R_{th1} + R_{th2} + R_{th3}$

$$R_{th1} = \frac{L_1}{k_1 A} = \frac{0.25}{0.4x12} = 0.0521 K/W$$

$$R_{th2} = \frac{L_2}{k_2 A} = \frac{0.08}{0.2x12} = 0.0333 K/W$$

$$R_{th3} = \frac{L_3}{k_3 A} = \frac{0.002}{54x12} = 0.0000031 K/W$$

$$Q = \frac{T_1 - T_4}{R_{th1}+R_{th2}+R_{th3}}$$

$$= \frac{600 - 60}{0.0521 + 0.0000031 + 0.0333}$$

$Q = 6320.96$ W

(i) To find temperature drop across the steel layer ($T_2$ - $T_3$)

$$Q = \frac{T_2 - T_3}{R_{th3}}$$

$T_3$- $T_4$ = $Q\times R_{th2}$

= $6320.96\times 0.0000031$

$T_3$- $T_4$ = 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$^0$C. 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 1$^{st}$ insulation

= 0.2+0.05

$r_2$ = 0.25m

$r_3$ = $r_2$ + thickness of 2$^{nd}$ insulation

= 0.25+0.05

$r_3$ = 0.3m

$r_4$ = $r_3$ + thickness of 3$^{rd}$ insulation

= 0.3+0.05

$r_4$ = 0.35m

$T_{hf}$ = 140° C, $T_{cf}$ = 30° 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**

$$Q = \frac{(\Delta T)_{overall}}{\Sigma R_{th}}$$

$\Delta T = T_{hf} - T_{cf}$

$\Sigma R_{th} = R_{th1} + R_{th2} + R_{th3}$

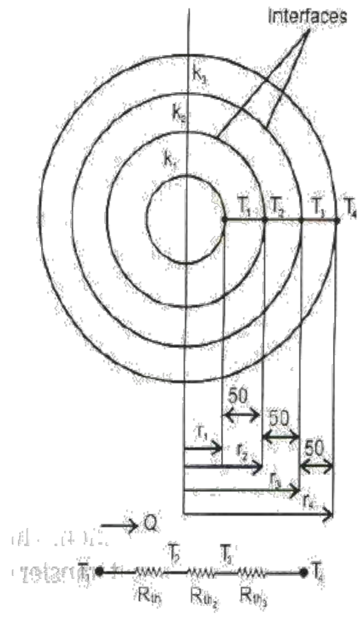
$$R_{th1} = \frac{r_2 - r_1}{4\pi k_1 r_2 r_1} = \frac{(0.25-0.20)}{4\pi\, x 0.02 x 0.25 x 0.2} = 3.978^\circ\, C/W$$

$$R_{th2} = \frac{r_3 - r_2}{4\pi k_2 r_3 r_2} = \frac{(0.30-0.25)}{4\pi\, x 0.06 x 0.3 x 0.25} = 0.8842^\circ\, C/W$$

$$R_{th1} = \frac{r_4 - r_3}{4\pi k_3 r_4 r_3} = \frac{(0.35-0.30)}{4\pi\, x 0.16 x 0.35 x 0.30} = 0.23684^\circ\, C/W$$

$$Q = \frac{140 - 30}{0.0796 + 0.8842 + 0.23684}$$

$Q = 21.57\ W$




$$Q = \frac{(\Delta T)_{overall}}{\Sigma R_{th}}$$

To find interface temperature ($T_2$, $T_3$)

$$Q = \frac{T_2 - T_3}{R_{th1}}$$

$T_2 = T_1 - [Q \text{ x } R_{th1}]$

$= 140 - [91.62 \times 0.0796]$

$T_2 = 54.17^0 C$

$$Q = \frac{T_2 - T_3}{R_{th1}}$$

$T_3 = T_2 - [Q \times R_{th2}]$

$= 132.71 - [91.62 \times 0.8842]$

$T_3 = 35.09^\circ\ C$

**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

$(\Delta T)$ overall = $T_1$ $T_4$

= 870 – 40

= 830 °C

And $\Sigma R_{th} = R_{th1} + R_{th2} + R_{th3}$

(assume A = 1 $m^2$ )

$$R_{th1} = \frac{L1}{k1A} = \frac{22 \times 10-2}{1 \times 1} = 22 \times 10^{-2} \text{ K/W}$$

$$R_{th2} = \frac{L2}{k2A} = \frac{7.5 \times 10-2}{0.12 \times 1} = 0.625 \text{ K/W}$$

$$R_{th3} = \frac{L3}{k3A} = \frac{11 \times 10-2}{0.75 \times 1} = 0.1467 \text{ K/W}$$

$$Q = \frac{T1 - T4}{Rth1 + Rth2 + Rth3}$$

$$= \frac{870 - 40}{0.9917}$$

Q = 836.95 W/$m^2$

Q = $3.01 \times 10^5$ J/h

# VAPOUR COMPRESSION REFRIGERATION SYSTEM

## FUNDAMENTALS OF REFRIGERATION

The first mechanical refrigerators for the production of ice appeared around the year 1860. In 1880 the first ammonia compressors and insulated cold stores were put into use in the USA. Electricity began to play a part at the beginning of this century and mechanical refrigeration plants became common in some fields: e.g. breweries, slaughter-houses, fishery, ice production, for example. After the Second World War the development of small hermetic refrigeration compressors evolved and refrigerators and freezers began to take their place in the home. Today, these appliances are regarded as normal household necessities.

Refrigeration is the process of removing heat from an area or a substance and is usually done by an artificial means of lowering the temperature, such as the use of ice or mechanical refrigeration.

Mechanical Refrigeration is defined as a mechanical system or apparatus so designed and constructed that, through its function, heat is transferred from one substance to another. Since refrigeration deals entirely with the removal or transfer of heat, some knowledge of the nature and effects of heat is necessary for a clear understanding of the subject.

## Common Refrigerants

Today, there are three specific types of refrigerants used in refrigeration and air-conditioning systems:

1. Chlorofluorocarbons or CFCs, such as R-11, R-12, and R-114

2.Hydro chlorofluorocarbons or HCFCs, such as R-22 or R-123

3. Hydro fluorocarbons or HFCs, such as R-134a. All these refrigerants are "halogenated," which means they contain chlorine, fluorine, bromine, astatine, or iodine.

Refrigerants, such as Dichlorodifluoromethane (R-12), Mono chloro difluoromethane (R-22), and Refrigerant 502 (R-502), are called primary refrigerants because each one changes its state upon the application or absorption of heat, and, in this act of change, absorbs and extracts heat from the area or substance.

The primary refrigerant is so termed because it acts directly upon the area or substance, although it may be enclosed within a system. For a primary refrigerant to cool, it must be placed in a closed system in which it can be controlled by the pressure imposed upon it. The refrigerant can then absorb at the temperature ranges desired. If a primary refrigerant were used without being controlled, it would absorb heat from most perishables and freeze them solid.

Secondary Refrigerants are substances, such as air, water, or brine. Though hot refrigerants in themselves, they have been cooled by the primary refrigeration system; they pass over and around the areas and substances to be cooled; and they are returned with their heat load to the primary refrigeration system. Secondary refrigerants pay off where the cooling effect must be moved over a long distance and gastight lines cost too much.

Refrigerants are classified into groups. The National Refrigeration Safety Code catalogs all refrigerants into three groups:

Group I – safest of the refrigerants, such as R-12, R-22, and R-502

Group II – toxic and somewhat flammable, such as R-40 (Methyl chloride) and R-764 (Sulfur dioxide)

Group III – flammable refrigerants, such as R-170 (Ethane) and R-290 (Propane).

R-12 Dichlorodifluoromethane (CC12 F2) Dichlorodifluoromethane, commonly referred to as R-12, is colorless and odorless in concentrations of less than 20 percent by volume in air. In higher concentrations, its odor resembles that of carbon tetrachloride. It is nontoxic, noncorrosive, nonflammable, and has a boiling point of -21.7°F (-29°C) at atmospheric pressure.

**Required Properties of Ideal Refrigerant**

1) The refrigerant should have low boiling point and low freezing point.
2) It must have low specific heat and high latent heat. Because high specific heat decreases the refrigerating effect per kg of refrigerant and high latent heat at low temperature increases the refrigerating effect per kg of refrigerant.
3) The pressures required to be maintained in the evaporator and condenser should be low enough to reduce the material cost and must be positive to avoid leakage of air into the system.
4) It must have high critical pressure and temperature to avoid large power requirements.
5) It should have low specific volume to reduce the size of the compressor.
6) It must have high thermal conductivity to reduce the area of heat transfer in evaporator and condenser.
7) It should be non-flammable, non-explosive, non-toxic and non-corrosive.
8) It should not have any bad effects on the stored material or food, when any leak develops in the system.
9) It must have high miscibility with lubricating oil and it should not have reacting properly with lubricating oil in the temperature range of the system.
10) It should give high COP in the working temperature range. This is necessary to reduce the running cost of the system.

**Coefficient of Performance (COP)**

The performance of refrigerators and heat pumps is expressed in terms of coefficient of performance (COP), defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

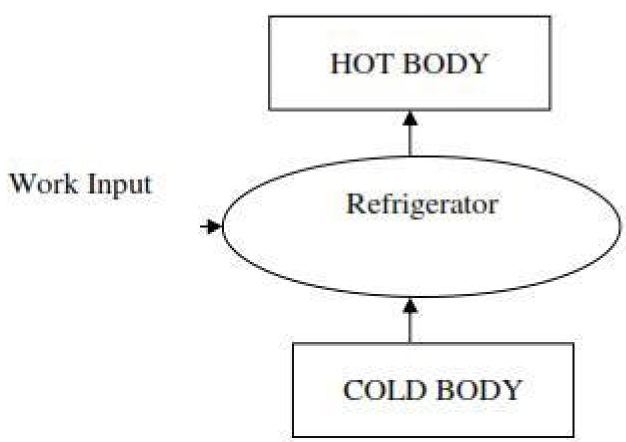
## 1.1 INTRODUCTION

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.

## 1.2 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.



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

$$1 \text{ TR} = \frac{2000\,\text{lb} \times 144\,\text{Btu/lb}}{24\,\text{hr}}$$
$$= 12000 \text{ Btu/hr} = 200 \text{ Btu/min}$$

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 × 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

$$1 \text{ TR} = 12000 \text{ Btu/hr} = \frac{12000}{3.968} = 3024.2 \text{ kcal/hr}$$
$$= 50.4 \text{ kcal/min} \approx 50 \text{ kcal/min}$$

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.

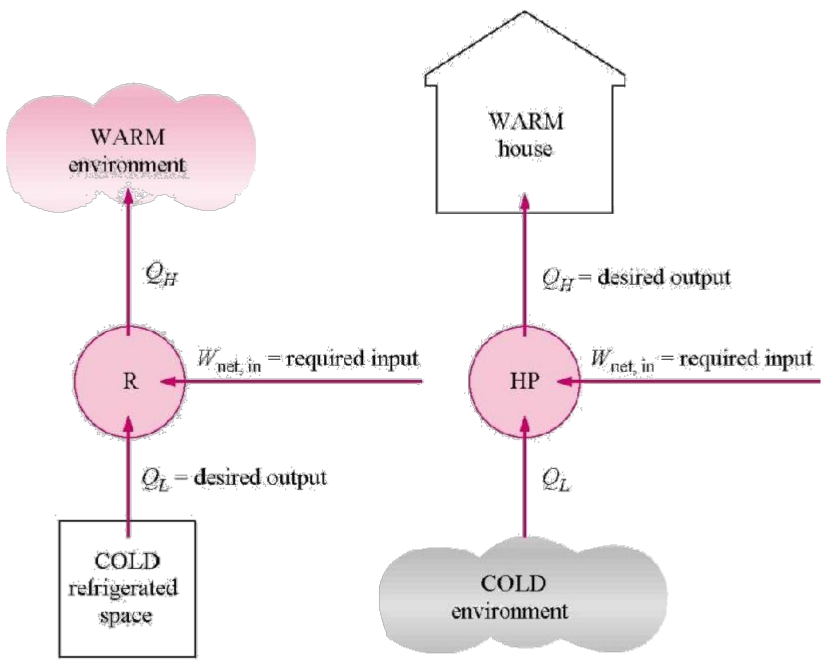
**<u>1.3 PROBLEMS</u>**

1) A refrigeration system produces 40 kg/hr of ice at $0^{O}C$ from water at $25^{O}C$. Find the refrigeration effect per hour and TR. If it consumes 1 kW of energy to produce the ice, find the COP. Take latent heat of solidification of water at $0^{O}C$ as 335 kJ/kg and specific heat of water 4.19 kJ/kg$^{O}$C.

2) 200 kg of ice at – $10^{O}C$ is placed in a bunker to cool some vegetables. 24 hours later the ice has melted into water at $5^{O}C$. What is the average rate of cooling in kJ/hr and TR provided by the ice? Assume Specific heat of ice, *Cp ice* = 1.94 kJ/kg$^{O}$C Specific heat of water, *cp,w* = 4.1868 kJ/kg$^{O}$C Latent heat of fusion of ice at $0^{O}C$, *L* = 335 kJ/kg.

## 1.4 REFRIGERATOR AND HEAT PUMP

The vapor compression refrigeration cycle is a common method for transferring heat from a low temperature to a high temperature.



(*a*) Refrigerator (*b*) Heat pump

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

$$COP_R = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}}$$

$$COP_{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}}$$

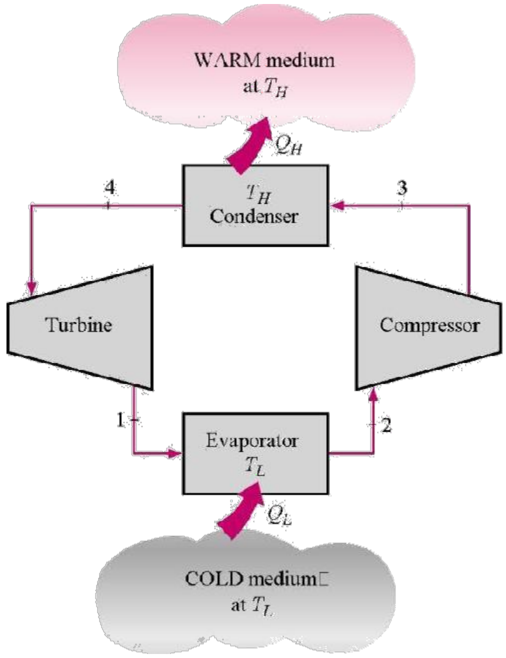
Both $COP_R$ and $COP_{HP}$ can be larger than 1. Under the same operating conditions, the

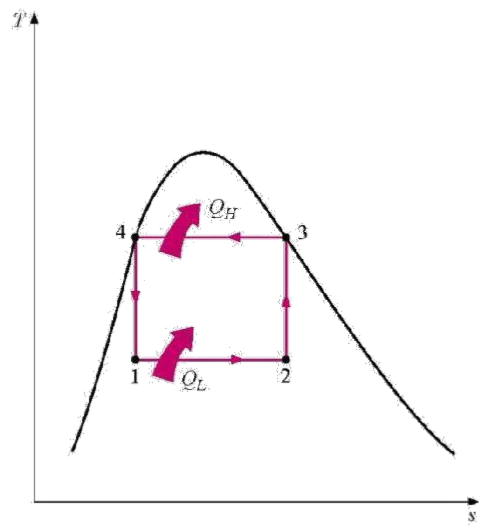
COPs are related by

$$COP_{HP} = COP_R + 1$$

## 1.5 REVERSED CARNOT CYCLE- REFRIGERATOR AND HEAT PUMP

Shown below is the cyclic refrigeration device operating between two constant temperature reservoirs and the T-s diagram for the working fluid when the reversed Carnot cycle is used. Recall that in the Carnot cycle heat transfers take place at constant temperature. If our interest is the cooling load, the cycle is called the Carnot refrigerator. If our interest is the heat load, the cycle is called the Carnot heat pump.




The standard of comparison for refrigeration cycles is the *reversed Carnot cycle.* A refrigerator or heat pump that operates on the reversed Carnot cycle is called a *Carnot refrigerator* or a *Carnot heat pump,* and their COPs are

$$COP_{R,Carnot} = \frac{1}{T_H / T_L - 1} = \frac{T_L}{T_H - T_L}$$

$$COP_{HP,Carnot} = \frac{1}{1 - T_L / T_H} = \frac{T_H}{T_H - T_L}$$

## 1.6 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)
  - Vapour compression refrigeration system (VCR)
  - Vapour absorbtion refrigeration system (VAR)
- Vapour adsorbtion refrigeration system and etc.,

Refrigeration by special fluid ( low boiling point fluids – Refrigerants) ( Reversed Carnot cycle)

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## 1.7 Simple Vapour Compression Refrigeration System (VCR)

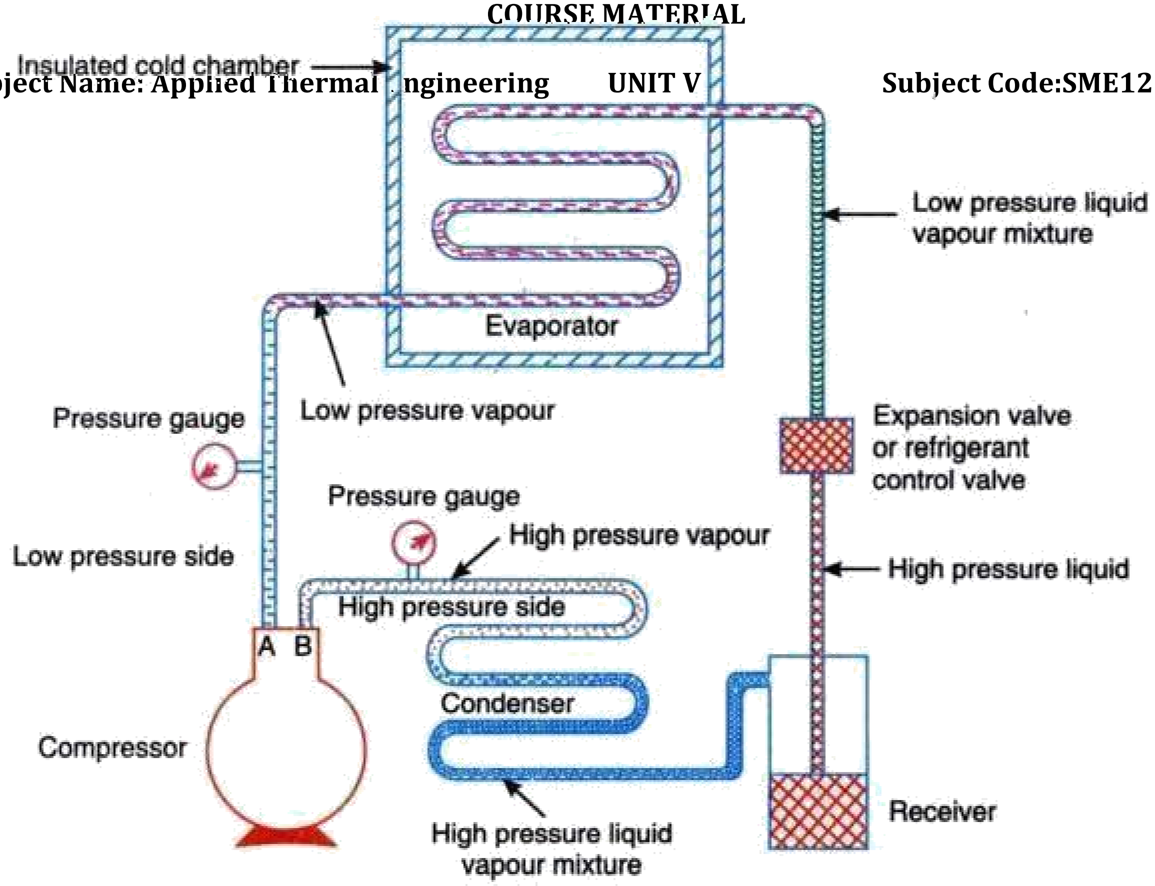
It consists of the following essential parts:

**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.




### 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.

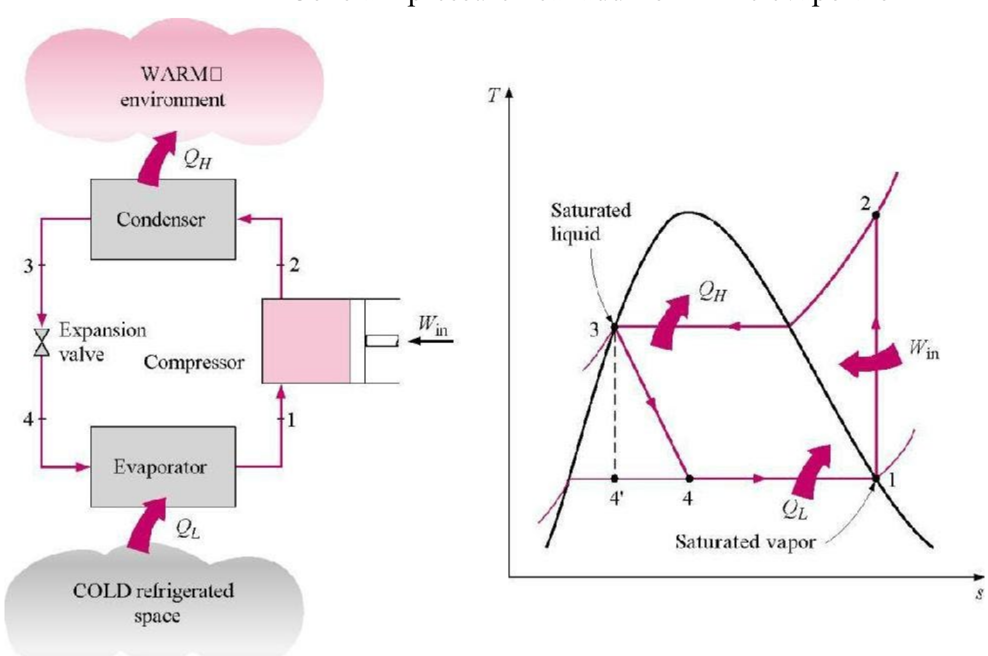
**The Simple Vapor Compression Refrigeration Cycle**

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.

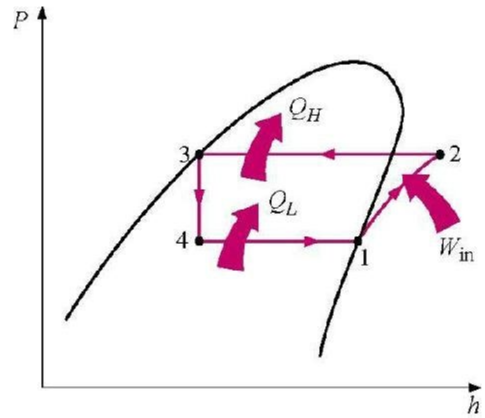
The ideal vapor compression cycle consists of four processes.

Ideal Vapor-Compression Refrigeration Cycle

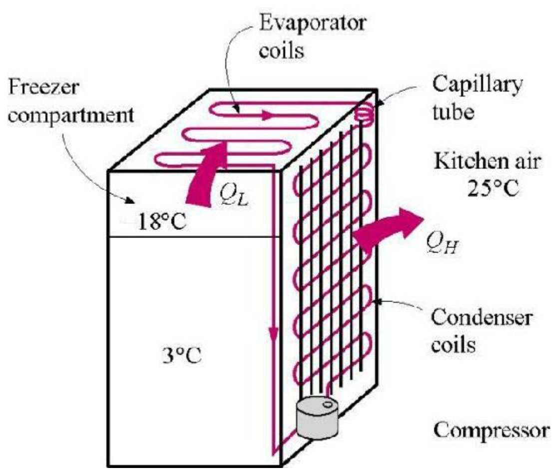
| Process | Description |
|---|---|
| 1-2 | Isentropic compression |
| 2-3 | Constant pressure heat rejection in the condenser |
| 3-4 | Throttling in an expansion valve |
| 4-1 | Constant pressure heat addition in the evaporator |



The P-h diagram is another convenient diagram often used to illustrate the refrigeration cycle.




The ordinary household refrigerator is a good example of the application of this cycle.




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. | |
| Throttle Valve | $\Delta s > 0$ | $h_4 = h_3$ |
| | | $Q_{net} = 0$ |
| Evaporator | P = Const. | $Q_L = m(h_1 - h_4)$ |

$$COP_R = \frac{Q_L}{W_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_{HP} = \frac{Q_H}{W_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1}$$

**PART-B**

**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. [Nov 2002]**

| 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 , $Sg_1$=1.557KJ/Kgk

At point 2

$T_2$=32°C=305k, $hf_2$=130.5KJ/Kg, $hg_2$=264.5,

$Sg_2$=1.542KJ/KgK From ph diagram, At point eqn(1) (dry).

At -5°C, i.e at 268k

$hg_1$=249.3KJ/Kg=hg

$h_1$=249.3KJ/Kg

At 32°C, i.e at 305K

$hg_2$=264.5KJ/Kg=$h_2$'

$h_2$'=264.5KJ/Kg

Entropy is constant during the compression process

so, $S_1$=$S_2$

From T- S diagram

At point (1) dry,

$S_1$=Sg at -5°C

$S_1=S_{g1}=1.557$KJ/Kgk

$S_1=S_2=1.557$KJ/Kgk

At point (2) (super heated)

$S_1=S_2' + Cp \ln (T_2/T_2')$

$1.557=S_2'+1.884 \ln(T_2/305)$ --------

(1) $S_2'=Sg$ at 32°C.

$Sg_2=1.542=S_2$

$S_2'=1.542$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

$h_{f2}=130.5= h_f$

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= (2×210)/ (249.3-

130.5) m = 8.84Kg/min

Work done = Refrigeration effect/ cop

= (2×210)/6 = 175 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 table. [May 2003]**

| **T(°C)** | **sensible Heat (hf)** | **Latent heat(hfg) 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 |

**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, $S_{f1}$=-0.109

KJ/KgK $h_{f1}$=-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, $S_{fg2}$=4.333KJ/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)

$S_2=S_{g2}$=4.33KJ/KgK

$S_2$=4.33KJ/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. Air enters the compressor of air craft system at 100kpa, 277k and is compressed to 300kpa with an isentropic efficiency of 72%. After being cooled to 328k and air expands is 100kpa and an $\eta_{Isen}$=78% the load is 3 tons and find COP, power, mass flow rate.**

**[May 2005]**

**Given data:**

$P_1$= 100kpa, $T_3$=38k

$T_1$= 277k, $P_4$=100kpa

$P_2$= 300kpa, $\eta_T$=78%

$\eta_c$= 72%

**Solution:**

process 1-2 Isentropic compression

$T_2= (P_2/P_1)^{\gamma-1/\gamma} \times T_1$

$T_2= (300/100)^{1.4-1/1.4}$

$\eta_c=(T_2-T_1)/(T_2'-T_1)$

0.72= (379.14-277)/ ($T_2'$-

277) $T_2$=418.86k

Process 3-4 isentropic compression

$T_3/T_4= (P_3/P_4)^{\gamma-1/\gamma}$

$328/T_4= (300/100)^{\gamma-1/\gamma}$

$T_4$=239.64k

$\eta_t=(T_3-T_4')/(T_3-T_4)$

0.78= (328-$T_4'$)/ (328-239.64)

$T_4'$= 259.08k

$COP = (T_1 - T_4')/(T_2' - T_1)$

$COP = (277-259.08)/(418.86-277)$

$=0.17$ 1 tonne= 3.5kw of heat

3tonne= 3×3.5=10.5kw

**Energy balance**.

Heat energy absorbed by $I_{ce}$=Heat rejected by

$$\text{air} = m \times C_p \times (T_1 - T_4')$$

$$10.5 = m_a \times 1.005 \times (277-259.08)$$

**Mass of air,** $m_a = 0.583$ Kg/sec.

**Power,** $P = m_a \times C_{pa} \times (T_2' - T_1)$

$= 0.583 \times 1.005 \times (418.86-277)$

$= 83.12$kw

**1. An ammonia refrigerator process 20tons of ice per day from and at 0°C.The condensation and evaporation takes at 20°C and -20°C respectively the temperature of the vapour at the end of Isentropic compression is 50°C and there is no under cooling of the liquid.COP=70% of theoretical COP. Determine (i) Rate of $NH_3$ circulation (ii) size of compressor, N=240rpm,L=D, $\eta_{vol}$=80%. Take Laten heat of $I_{cc}$=335kJ/Kg, $C_p$= 2.8 kJ/Kg, $V_{s1}=0.624m^3$kg. Use the following properties of ammonia.**

| Sat.Temp(°C) | Enthalpy(kJ/Kg) | | Entropy(kJ/Kgk) | | |
|---|---|---|---|---|---|
| | $h_f$ | $h_g$ | $S_f$ | $S_g$ | |
| **20** | **274.98** | **1461.58** | **1.0434** | **5.0919** | |
| **20** | **89.72** | **1419.05** | **0.3682** | **5.6204** | **[Apr 2003]** |

**Given data:** 20 tons of Icc per day at °C

$T_3$= 20°C

$T_1$=-20°C

$T_3$=50°C

COP =70% of theoretical cop

N=240rpm

L=D

$h_v$=80%

Latent heat of Ice=335KJ/Kg

$C_p$=2.8KJ/Kgk

$V_{s1}$=0.624$m^3$/kg

**Solution:-**

The refrigeration effect=20×3.5=77.55kw

$h_1$=1419.05KJ/Kg

$h_{g2}$=1461.58KJ/Kg at 20°C

$h_{f3}$=274.98 KJ/Kg

$h_2=h_{g2}+C_p(T_2-20)$

$h_2$=1461.58+2.8(50-20)

$h_2$=1545.58 KJ/Kg

COP= $(h_1-h_{f3})/(h_1-h_2)$

= (1419.05-274.98)/(1545.58-

1419.05) =9.04.