Lecture Note on
Subject: THERMAL ENGINEERING-II
Code TH – 4
Branch: Mechanical Engineering
Name of Faculty: - Er Litu Behera

Syllabus

1. Performance of I.C engine

1.1 Define mechanical efficiency, Indicated thermal efficiency, Relative Efficiency, brake thermal efficiency overall efficiency Mean effective pressure &specific fuel consumption.

1.2 Define air-fuel ratio & calorific value of fuel.

1.3 Work out problems to determine efficiencies & specific fuel consumption.

2. Air Compressor

2.1 Explain functions of compressor & industrial use of compressor air

2.2 Classify air compressor & principle of operation.

2.3 Describe the parts and working principle of reciprocating Air compressor.

2.4 Explain the terminology of reciprocating compressor such as bore, stroke, pressure ratio free air delivered &volumetric efficiency.

2.5 Derive the work done of single stage & two stage compressor with and without clearance. 2.6 Solve simple problems (without clearance only)

3. Properties of Steam

3.1 Difference between gas & vapours.

3.2 Formation of steam.

3.3 Representation on P-V, T-S, H-S, & T-H diagram.

3.4 Definition & Properties of Steam.

3.5 Use of steam table & mollier chart for finding unknown properties.

3.6 Non flow & flow process of vapour.

3.7 P-V, T-S & H-S, diagram.

3.8 Determine the changes in properties & solve simple numerical.

4. Steam Generator

4.1 Classification & types of Boiler.

4.2 Important terms for Boiler.

4.3 Comparison between fire tube & Water tube Boiler.

4.4 Description & working of common boilers (Cochran, Lancashire, Babcock & Wilcox Boiler)
4.5 Boiler Draught (Forced, induced & balanced)
4.6 Boiler mountings & accessories.

5. Steam Power Cycles

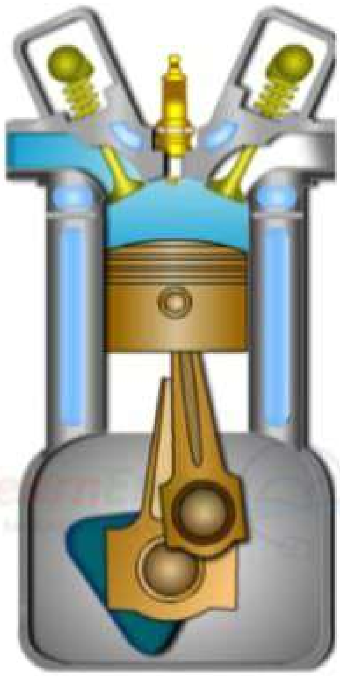
5.1 Carnot cycle with vapour.
5.2 Derive work & efficiency of the cycle.
5.3 Rankine cycle.
5.3.1 Representation in P-V, T-S & h-s diagram.
5.3.2 Derive Work & Efficiency.
5.3.3 Effect of Various end conditions in Rankine cycle.
5.3.4 Reheat cycle & regenerative Cycle.
5.4 Solve simple numerical on Carnot vapour Cycle & Rankine Cycle.

6. Heat Transfer

6.1 Modes of Heat Transfer (Conduction, Convection, Radiation).
6.2 Fourier law of heat conduction and thermal conductivity (k).
6.3 Newton's laws of cooling.
6.4 Radiation heat transfer (Stefan, Boltzmann & Kirchhoff's law) only statement, no derivation & no numerical problem.
6.5 Black body Radiation, Definition of Emissivity, absorptivity, & transmissibility.

Introduction

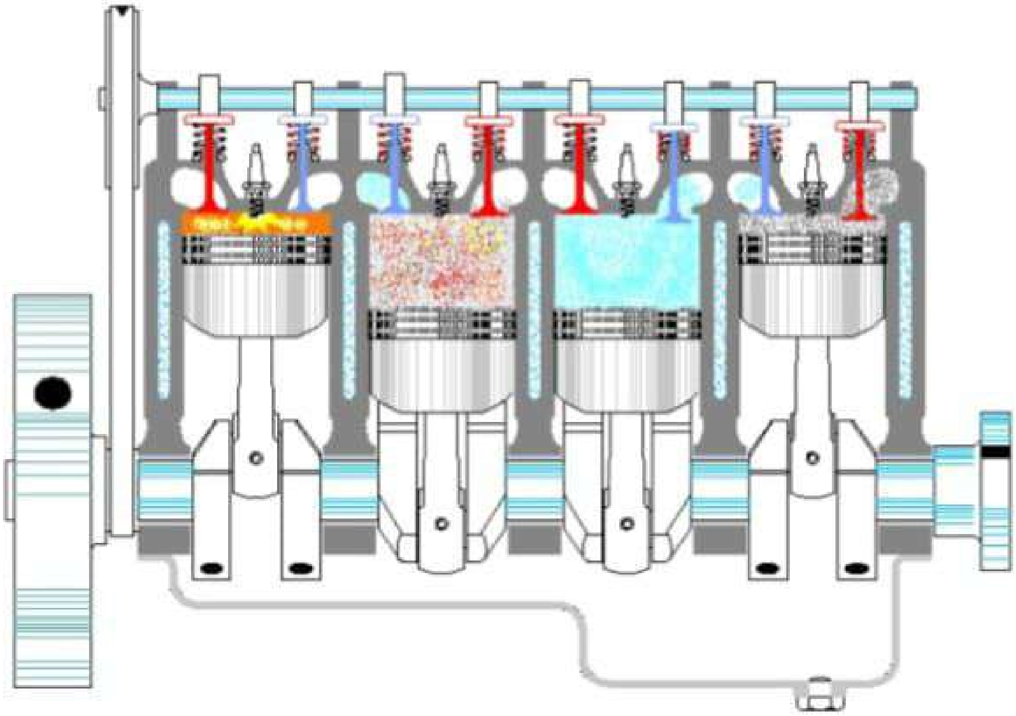
- The basic task in the design and development of engines is to reduce the cost and improve the efficiency and power output.
- In order to achieve the above task, the 'development engineer' has to compare the engine developed with other- engines in terms of its output and efficiency.
- Towards this end he has to test the engine and make measurements of relevant parameters that reflect the performance of the engine.
- I.C. engine generally operates within a useful- range of speed.



- Some engines are made to run at fixed speed by means of speed governor, which is its rated speed.
- The performance of the engine depends on the inter-relationship between the power developed, speed and the specific fuel consumption at each operating condition within the useful range of speed and load.

The following factors are to be considered in evaluating the performance of an engine

- Maximum power or torque available at each speed within the useful range of speed.
- The range of power output at constant speed for stable operation of the engine. The different speeds should be related at equal intervals within the useful speed range.
- Brake specific fuel consumption at each operating condition within the useful range of operation.
- Reliability and durability of the engine for the given range of operation.



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Purpose of testing an I.C. Engine

In general the purpose or significance of testing an I.C. engine is to determine the following,

- To determine the information, which can not be obtained by calculations.
- To confirm the data used in design, the validity of which may be doubtful.
- To satisfy the customer regarding the performence of the engine.

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IC Engine testing

Some important terms as per ISI Standard,

- **Speed:** The speed of an engine is the mean speed of its crank shaft in revolutions per minute (RPM), except in case of 'free piston' engines where the speed is the number of cycles per minute , of the reciprocating components.

- **Standard operating conditions:** The following are the standard operating conditions:

  - Mean barometric pressure: It is taken as 736 mm of mercury (Hg).
  - Intake air temperature : It is taken as 300K or 27°C .

- **Mean effective pressure and torque:** Mean effective pressure, p_m, is defined as a hypothetical pressure which is thought to be acting on the piston throughout the power stroke.

$$p_m = \frac{\text{net area of indicator diagram in mm}^2 \times \text{spring constant}}{\text{length of indicator diagram in mm}}$$

- The torque is related to mean effective pressure by the relation,

$$bp = \frac{2\pi N T}{60}$$

$$ip = \frac{p_m L A N K}{60}$$

By equation (i),

$$\frac{2\pi N T}{60} = \left(\text{bemp } A L \frac{NK}{60}\right)$$

$$T = \frac{(\text{bemp } A L K)}{2\pi}$$

- Thus the torque and the mean effective pressure are related by the engine size.

- A large engine products more torque for the same mean effective pressure, For this reason, torque is not the measure of the ability of an engine to utilize its displacement fro producing power from fuel.

- A large engine products more torque for the same mean effective pressure, For this reason, torque is not the measure of the ability of an engine to utilize its displacement fro producing power from fuel.

- It is the mean effective pressure which gives an indication of engine displacement utilization for this conversion.
- Higher the mean effective pressure, higher will be the power developed by the engine for a given displacement.
- Therefore, it is not possible to compare engines on the basic of either power or torque.
- Mean effective pressure is the true indication of the relative performance of different engines.
- **Indicated power:** It is the total power developed in the working cylinder by the gases on the combustion side of the working pistons.
- **Brake power:** It is the total power measured at the driving shaft.
- **Friction power:** It is the power consumed in frictional resistance.
- **Mechanical efficiency:** It is the ratio of break power to the indicated power.

$$\eta_{mech} = \frac{b.p}{i.p}$$

i.p is always greater than, b.p

- **Fuel consumption:** The quantity of fuel consumed by the engine per unit time of the stated power and under stated operating conditions.
- **Specific fuel consumption:** It is the quantity of fuel consumed per unit of power per unit of time. It is generally expressed in gms of fuel consumed per kW hr or B.H.P./bp.
- **Indicated thermal efficiency:** It is the ratio of heat equivalent of i.p to the heat energy of the fuel supplied

$$\text{Indicated thermal } \eta = \frac{3600 \times i.p}{W_f \times C}$$

Where

W_f = weight of fuel consumed per hour in kg

C = Calorific value of the fuel in kJ/kg i.p in kW

- **Brake thermal efficiency:** It is the ratio of heat equivalent of b.p to the heat energy of the fuel supplied.

$$\text{Brake thermal } \eta = \frac{3600 \times b.p}{W_f \times C}$$

- **Volumetric efficiency**: It is the ratio of actual volume of the charge admitted during the suction stroke at N.T.P to the swept volume of the piston.

$$\therefore \quad \eta_{vol.} = \frac{\text{volume of charge admitted at N.T.P during suction stroke}}{\text{Swept volume of the piton}}$$

- **Heat balance sheet:** The complete record of heat supplied and heat rejected during a certain time (say one minute) by an I.C. engine is entered in a tabulated form known as heat balance sheet.

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Mean effective pressure

- The algebraic sum of the mean pressures acting on the surface of the piston during each stroke over one complete cycle.

Indicated Mean Effective Pressure (P_{im})

- It may be defined as, the constant pressure acting over the full length of the stroke and capable of producing the same amount of work, as is actually produced during the complete cycle of the engine.
- It is generally denoted by 'P_{im}' or i.m.e.p.
- As, the pressure in the cylinder varies throughout the cycles and the variation can be expressed with respect to the volume or crank angle rotation to obtain p-V or p-θ diagrams, respectively.
- However, such a continuous variation does not readily lend itself to simple mathematical analysis in the computation of indicated power.
- If an average pressure for one cycle can be used, then the computations become far less difficult.
- Refering figure, as the piston moves back and forth between TDC and BDC, the process lines on the p-V diagram indicated the successive states of the working fluid through the cycle.

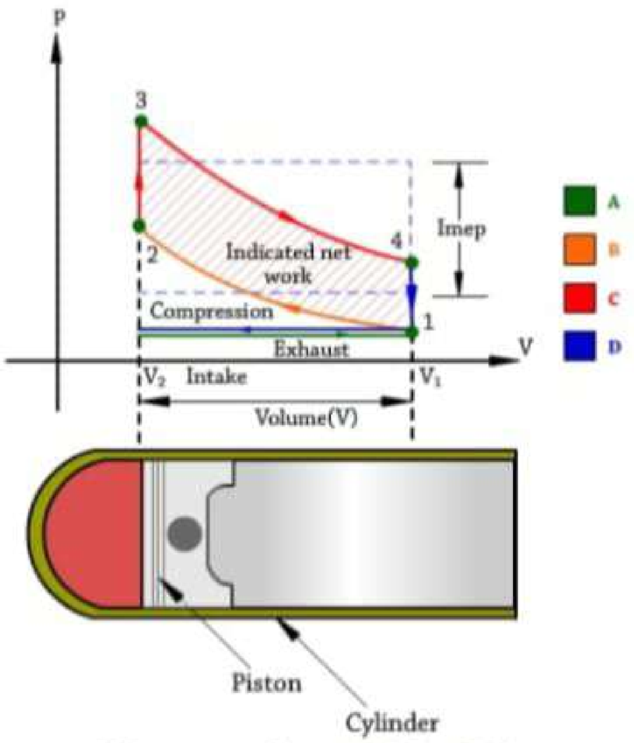

Fig 1: p-V diagram for an ideal four-stroke cycle engine

- The indicated network of the cycle is represented by the area 1-2-3-4 enclosed by the process lines for that cycle.
- If the area of rectangle A-B-C-D equals, the area 1-2-3-4, the vertical distance between the horizontal lines AB and CD respectively gives the 'indicated mean effective pressure', imep.
- It is a mean value expressed in N/m^2, which when multiplied by the displacement volume or swept volume, Vs gives the 'same indicated net work' as is actually produced with the varying pressures.

$$p_{im} \times (V_1 - V_2) = \text{Net work of cycle}$$

$$p_{im} = \frac{\text{Network of cycle}}{(V_1 - V_2)}$$

- It is obtained from indicater diagram drawn with the help of engine indicator.
- The value of the area measured, when divided by the piston displacement and multiplied by the spring number of the indicator, results in the mean ordinate or indicated mean effective pressure, p_{im}.

$$p_{im} = \frac{\text{Area of indicator diagram} \times \text{Spring number}}{\text{Length of the indicator diagram}}$$

$$p_{im} = \frac{a \times s}{l}$$

Where,

P_{im} = Mean Effective Pressure in N/m^2

If, a = Area of indicator diagram in cm^2 or mm^2

L = length of the indicator diagram in cm or mm

And, s = Spring number in N/m^2 per cm or N/m^2 / mm

Then $p_{im} = \frac{a \times s}{l} N/m^2$

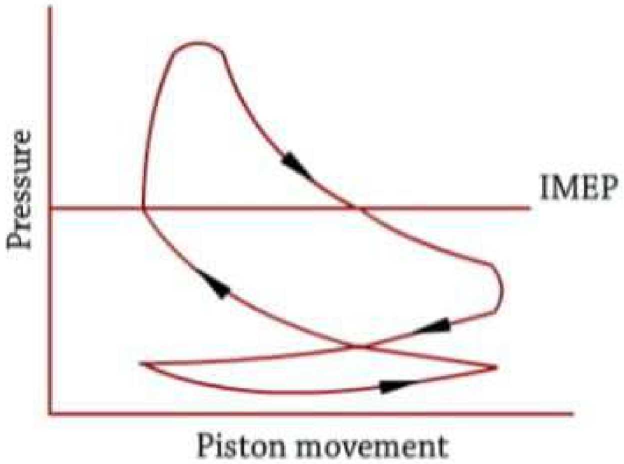

Fig 2: Indicated mean effective pressure

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Indicated power

- **Definition:** The power developed inside the engine cylinder due to combustion of fuel is defined as indicated power.

$$\text{Indicated power, I.P} = \left(\frac{p_m L A n}{60}\right) K$$

Where,

p_m = Indicated mean effective pressure (I.M.E.P) in N/m^2

L = Length of stroke (m)

A = Area of the inside cylinder (or) Bore area in m^2

n = Number of working / power / explosions strokes per minute

n = N = for 2 stroke

$n = \frac{N}{2}$ = for four stroke.

K = No.of cylinders

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Piston indicator

- Measurement of I.P mainly involves evaluation of indicated mean effective pressure.
- It is found from p-V diagram which is obtained from an indicator diagram.
- Indicator diagram is experimentally obtained using an indicator fitted to an engine cylinder.
- Main types of engine indicators are,
  - Piston indicators
  - Balanced diaphragm type indicator (Ex: Farmborough Indicator)
  - Electronic indicator

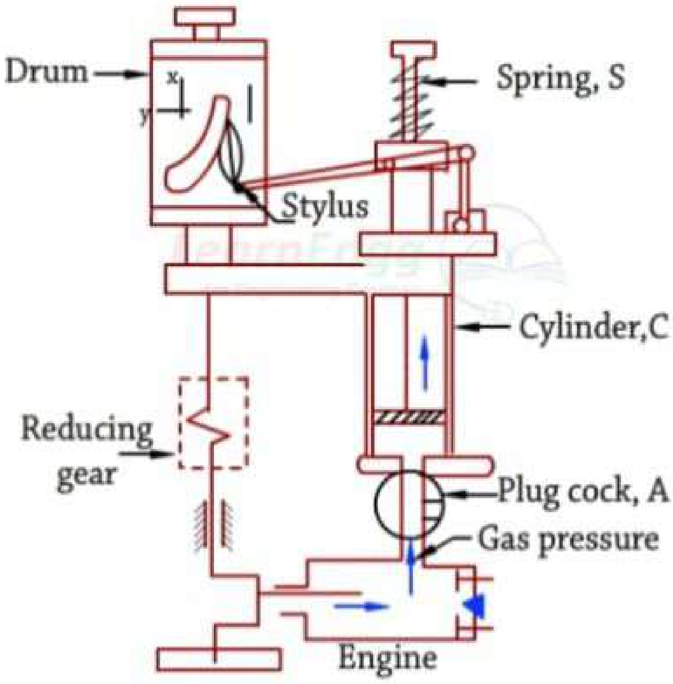

Fig 3: Piston indicator

- A cylinder C which is connected to an engine cylinder through a plug cock. A piston D of this cylinder is attached to the top of a tension spring S with the help of a push rod.
- A magnifying mechanical linkage at the end of which a stylus is connected. The stylus traces diagram on the paper wrapped around a drum. The drum is directly driven by the engine crankshaft.
- With the help of a 3 Way cock plug A cylinder C can be connected either to atmosphere or to engine cylinder.

- When it is connected to the engine cylinder gas pressure forces the piston 'D' up against tension spring S.
- The movement of this spring is magnified by a link mechanism and transmitted to the stylus.
- The stylus moves parallel to the drum axis.
- Movement of stylus is proportional to gas pressure.
- The rotational of drum will be proportional to piston displacement.
- p-V diagram is obtained on the paper wrapped around this drum.
- It is an average of number of cycles.

$$\text{IMEP} = \frac{\text{area of indicator diagram}}{\text{length of indicator diagram}} \times \text{spring scale}$$

- The first method is the one, which is more commonly used.

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Brake power

- The brake power (B.P.) i.e. the power obtained at the engine flywheel is measured with the help of dynamometers, a brief description of some of the types being given here.

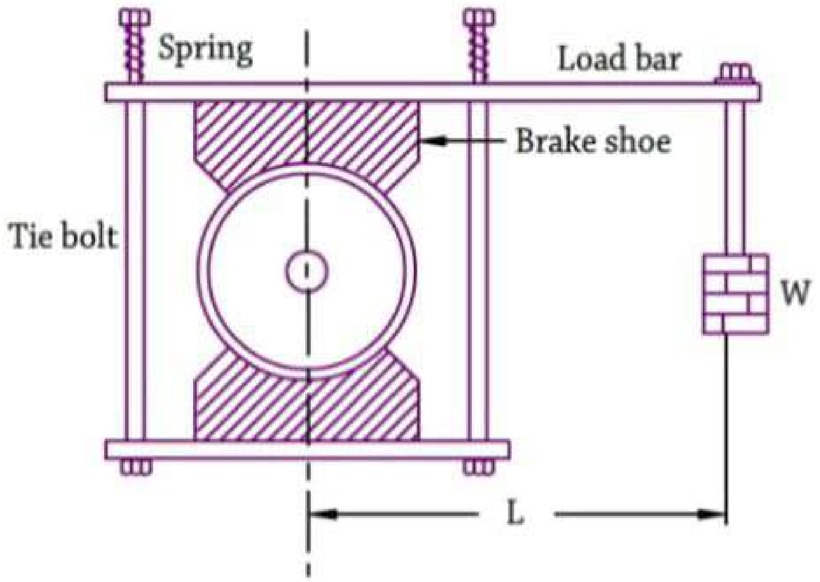

Fig 4: Prony brake

- In case of prony brake, let

W = Brake load in newtons,

l = Length of arm in metres, and

N = Speed of the engine in r.p.m.

$\therefore$ Brake power of the engine,

$$B.P = \frac{\text{Torque in N} - \text{m} \times \text{Angle turned in radians through 1 revolution}}{60} \times \text{R.P.M. watts}$$

$$= \frac{T \times 2\pi N}{60}$$

$$= \frac{Wl \times 2\pi N}{60} \text{ watts}$$

Rope brake

- In case of rope brake, let

W = Dead load in newtons,

S = Spring balance reading in newtons,

D = Diameter of brake drum in meters,

d = Diameter of the rope in meters, and

N = Speed of the engine in r.p.m.

$\therefore$ Brake power of the engine,

$$B.P = \frac{(W - S)\,\pi DN}{60} \text{ watts} \quad \text{[without considering diameter (d) of the rope]}$$

$$= \frac{(W - S)\,\pi(D + d)N}{60} \text{ watts} \quad \text{[Considering diameter (d) of the rope]}$$

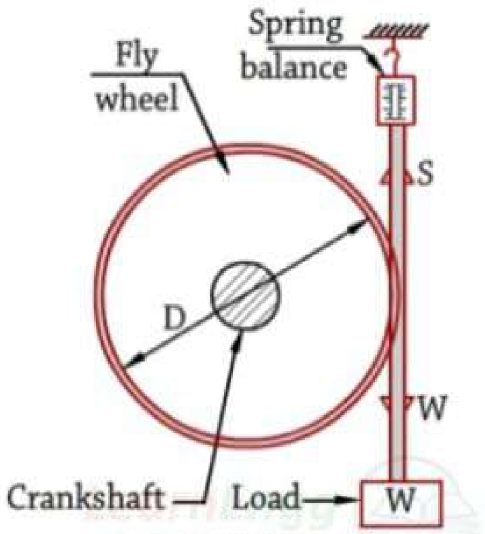

Fig 5: Rope brake dryamometer

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Frictional power

- Some of the indicated power developed in engine cylinder is lost in overcoming the friction at various engine parts and remaining power is obtained as brake power at crank shaft.
- This type of power lost due to friction is defined as friction power.

$$\therefore \quad \text{Frictional power} = \text{Indicated power} - \text{Brake power}$$

- Frictional Power (FP) is given by,

$$FP = IP - BP$$

- It is measured in Kilo Watts (kW).

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Mechanical efficiency

- It may be defined as the ratio of the power obtained at the crank shaft, i.e. brake power (bp) to the indicated power (ip).
- Thus mechanical efficiency,

$$(\eta_m) = \frac{bp}{ip}$$

- Mechanical efficiency takes into account the mechanical losses in an engine.
- Mechanical losses of an engine may be further subdivided into the following groups.
- Friction losses as in case of pistons, bearings, gears, valve mechanism.
- With the development in the bearing design and materials improvement in gears etc., these losses are usually limited from 7 to 9 percent of the indicated power output.
- Power is absorbed by engine auxiliaries such as fuel pump, lubricating oil pump, water circulating pump, radiator magneto and distributor, electric generator for battery charging, radiator fan etc. These losses may account for 3 to 8 percent of the indicated output.
- Ventilating or faning action of the flywheel. This loss is usually below 4 percent of the indicated output.
- Work of charging the cylinder with fresh charge and discharging the exhaust gases during the exhaust stroke. In case of two stroke engines the power absorbed by the scavenging pump etc. These losses may account for 2 to 6 percent of the indicated power output.
- In general, the mechanical efficiency of engines varies from 65 to 85 %.

Indicated thermal efficiency

- It may be defined as the ratio of heat converted into indicated work to the heat energy supplied by the fuel, during a specified period of time.
- Indicated thermal efficiency,

$$\eta_{ith} = \frac{\text{Heat equivalent to Ip per minute}}{\text{Heat energy supplied by fuel per minute}}$$

$$= \frac{Ip \times 60}{M_f \times CV}$$

Where, M_f = Mass of fuel supplied to the engine in kJ/kg.

CV = lower calorific value of the fuel in kJ/kg.

Brake thermal efficiency (η_{bth})

- It may be defined as the ratio of heat equivalent to brake power (bp) to the heat energy supplied by the fuel during a specific period of time .
- Brake thermal efficiency,

$$\eta_{bth} = \frac{\text{Heat equivalent to Bp per minute}}{\text{Heat energy supplied by fuel per minute}}$$

$$= \frac{Bp \times 60}{M_f \times CV}$$

Where, M_f = Mass of fuel supplied to the engine in kJ/kg.

CV = lower calorific value of the fuel in kJ/Kg.

- In modern engines, an indicated thermal efficiency of almost 28 percent is desirable with gas and gasoline spark ignition engines having a moderate compression ratio and as high 36 percent or even more with high compression ratio diesel engines, i.e. CI engines.

Brake thermal efficiency may be obtained from other efficiencies.

$$\eta_m = \frac{\eta_{bth}}{\eta_{ith}} = \frac{B\,P}{I\,P}$$

$$\eta_{bth} = \eta_m \times \eta_{ith}$$

Relative efficiency

- It is the ratio of actual thermal efficiency to air standard efficiency of the engine. It is sometimes referred as efficiency ratio. It is expressed as

$$\eta_{Relative} = \frac{\text{Brake thermal efficiency}}{\text{Air standard efficiency}}$$

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Measurement of fuel consumption

- Two glass vessels of 100cc and 200cc capacity are connected in between the engine and main fuel tank through two, three-way cocks.
- When one is supplying the fuel to the engine, the other is being filled.
- The time for the consumption of 100 or 200cc fuel is measured with the help of stop watch.
- When fuel rate is to be measured, the valve is closed so that fuel is consumed from the burette.
- The time for a known value of fuel consumption can be measured and fuel consumption rate can be calculated.

$$\text{Fuel consumption kg/hr} = \frac{X_{CC} \times \text{Sp.gravity of fuel}}{1000 \times t}$$

Where,

t = time of fuel consumtion, kg/m^3 sec

X_{cc} = volume of fuel consumtion, c.c

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Specific fuel consumption

- It is amount of the fuel consumed per hour per unit power developed by the engine.

Indicated specific fuel consumption

- This is defined as the mass of fuel consumption per hour in order to produce an indicated power of one kilo watt.

- Thus, indicated specific fuel consumption = isfc $= \dfrac{\text{Fuel consumed in kg/hr}}{\text{I.P. of the engine}}$ kg/kwhr

Brake specific fuel consumption

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

- Thus, Brake specific fuel consumption = bsfc $= \dfrac{\text{Fuel consumed in kg/hr}}{\text{B.P. of the engine}}$ kg / kwhr

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Air-Fuel Ratio (A/F)

- It is the ratio between the mass of the air and mass of the fuel supplied to the engine. It is expressed as,

$$A/F = \frac{\text{Mass flow rate of air}}{\text{Mass flow rate of fuel}}$$

- Theoretically, the correct (stoichiometric) air– fuel ratio is 15. But the combustion of air–fuel mixture can take place in A/F ratio ranges from 12 to 19 for petrol engines and 20 to 60 in Diesel engines.

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Calorific value of fuels

- The calorific value or heat value of a fuel is defined as the total quantity of heat liberated by the complete combustion of unit quantity of fuel.
- The calorific value of solid or liquid fuels is expressed in kJ/kg.
- The calorific value of gaseous fuels is expressed in kJ/m^3.
- Thus the calorific value of fuel may be expressed by two values,
    - Higher (gross) calorific value (H.C.V)
    - Lower (net) calorific value (L.C.V)

Higher calorific value (HCV)

- Higher calorific value or gross calorific value is defined as the total amount of heat liberated when unit quantity of the fuel is completely burnt and the products of combustion are cooled to the initial temperature at which fuel and air are supplied, usually taken as 15° C.

Lower calorific value (LCV)

- Lower calorific value or net calorific value is defined as the amount of heat liberated by complete combustion of unit quantity of fuel without the condensation of steam formed.
- It is given by the difference between HCV and the latent heat absorbed by the water vapour,

$$LCV = HCV - m_s L \text{ kJ/kg}$$

Where

m_s = mass of water vapour formed per kg of fuel burnt = 9 H_2 ;
H_2 = Percentage of hydrogen by mass

L = Latent heat of water vapour, corresponding to 15°C
= 2466 kJ/kg

- Thus the above equation can be written as,

$$LCV = HCV - m_s\, 2466 \text{ kJ/kg}$$

$$LCV = HCV - 9H_2 \times 2466 \text{ kJ/kg}$$

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Volumetric efficiency

- It is the ratio of actual volume of charge admitted during the suction stroke at NTP to the swept volume of the piston.

- It may also be defined as the ratio of actual mass of air drawn into the engine during a given period of time to the theoretical mass which should have been drawn in during that same period of time, based upon the total piston displacement of the engine and the temperature as well as pressure of the surrounding atmosphere.

$$\eta_{vol} = \frac{m_{act}}{m_{th}}$$

Here,

m_{act} is the mass of actual air drawn $(= V_s \times \rho)$

m_{th} is the mass of the theoretical air, i.e. the air drawn at environmental condition $(= \rho_a \times n \times V_s)$

Where,

ρ_a is the density of air

V_s is the swept volume

n is the number of intake strokes per min. $= \frac{N}{2}$ for 4 stroke engines

$= N$ for 2 stroke engines

- The volumetric efficiency is measure of the success, with which the air supply and thus the charges induced into the engine cylinder.

- It is very important parameter, since it indicates the breathing capacity of the engine.

- The actual mass is a measured quantity.

- The theoretical mass is computed from the geometry of the cylinder, the number of cylinders and the speed of the engine in conjunction with the density of the surrounding atmosphere.

- Volumetric efficiency for a naturally aspirated engine is generally about 75%.

Factors affecting volumetric efficiency

- Volumetric efficiency is a measure of the breathing ability of an engine.
- It depends on the following factors,
  - Engine Speed: It increases with speed up to certain limit and decreases thereafter.
  - Compression Ratio: It decreases with increase in compression ratio.
  - Mixture Strength: It is max for lean mixtures but decreases with richness of the mixture.
  - Inlet Air Temperature: It decreases with increase in temperature.
  - Cooling Water Temperature: Slightly increases with decrease in temperature of cooling water.

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Example 1

- The following data were recorded a test on an oil engine,

  Speed of the engine = 1000 rpm,

  Load on the brake = 1000 N,

  Length of the arm = 750 mm

  Determine

  (i) Brake torque

  (ii) B.P. of the engine.

Given data

Speed of the engine (N) = 1000 rpm

Load on the brake (W) = 1000 N

Length of the arm (l) = 750 mm

To find

Brake torque

B.P. of the engine

Solution

Brake torque,

$$T = W \times l$$

$$= 1000 \times 0.75$$

$$T = 750 \text{ N-m}$$

Brake power of the engine,

$$\text{B.P.} = \frac{(2\pi N T)}{60} \text{ Watts}$$

$$= \frac{(2\pi \times 1000 \times 750)}{60}$$

$$= \frac{4712388.98}{60}$$

$$= 78{,}539.81 \text{ Watts}$$

$$B.P = 78.54 \text{ KW}$$

Result

- Brake torque = 750 N-m
- Brake Power of the engine = 78.54 kW

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Example 2

- A rope brake has rope wheel diameter 600 mm and the diameter of rope is 5 mm the dead load on the brake is 210 N and spring balance reads 30 N if the engine make 450 rpm find the brake power developed.

Given data

Rope wheel diameter (D) = 600 mm = 0.6 m

Diameter of rope (d) = 5 mm = 0.005 m

Dead load on the brake (W) = 210 N

Spring balance reads (S) = 30 N

Speed (N) = 450 rpm

To find

Brake power developed = ?

Solution

$$\text{Brake power(B.P)} = \frac{(W-S)\,\pi\,(D+d)\,N}{60}\ \text{Watts}$$

$$= \frac{(210-30)\times\pi\times(0.6+0.005)\times450}{60}$$

$$= \frac{180\times\pi\times0.605\times450}{60}$$

$$= \frac{153953.748}{60}$$

$$= 2565.89\ \text{Watts}$$

$$\text{Brake power} = 2.57\ \text{kW}$$

Result

- Brake power developed = 2.57 kW

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Example 3

- A single cylinder 4 stroke diesel engine has a bore of 110 mm, stroke 120 mm, the indicated mean effective pressure is 375 kN/m^2. calculate the indicated power at a crank speed of 60 rpm.

Given data

Bore, d = 110 mm = 0.11 m

Piston area (A) = $(\pi/4)(0.11)^2 = 0.0095\ m^2$

Stroke (l) = 120 mm = 0.12 m

Indicated mean effective pressure (i.m.e.p.) $P_m = 375\ kN/m^2$

Speed (N) = 60 rpm

To find

Indicated power at a crank = ?

Solution

$$I.P = \frac{P_m LA\left(\frac{N}{2}\right)}{60}$$

$$= \frac{375 \times 0.12 \times 0.0095 \times 60}{2 \times 60}$$

$$= \frac{25.65}{120}$$

Indicated power = 0.21375 kW

Result

- Indicated power at a crank = 0.21375 kW

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Example 4

- In a laboratory experiment the following observations were noted during the test of 4 stroke diesel engine

Area of indicator diagram = 420 mm^2

Length of indicator diagram = 62 mm

Spring number = 1.1 bar/min

Diameter of piston = 100 mm

Length of stroke = 150 mm

Engine speed = 450 rpm

Determine

(i) Indicated mean effective pressure

(ii) Indicated power

Given data

Area of indicator diagram (a) = 420 mm^2

Length of indicator diagram (l) = 62 mm

Spring number, (s) = 1.1 bar/mm

Diameter of piston (d) = 100 mm

Length of stroke (L) = 150 mm

Engine speed (N) = 450 rpm

To find

Indicated mean effective pressure = ?

Indicated power = ?

Solution

Area of piston,

$$A = \left(\frac{\pi}{4}\right) \times 0.1^2$$

$$= 0.785 \times 0.01$$

$$A = 7.853 \times 10^{-3}\ m^2$$

$$n = \frac{N}{2} = \text{No. of power strokes per min}$$

$$= \frac{450}{2}$$

$$n = 225$$

(a)

$$P_m = \frac{(a.s)}{l}$$

$$= \frac{(420 \times 1.1)}{62}$$

$$P_m = 7.45\ \text{bar}$$

(b)

$$I.P = \frac{(100\, P_m\ LAn)}{60}$$

$$= \frac{(100 \times 7.45 \times 0.15 \times 7.885 \times 10^{-3} \times 225)}{60}$$

$$= \frac{198.25}{60}$$

$$I.P = 3.30\,\text{kW}$$

Result

- Indicated mean effective pressure = 7.45 bar
- Indicated power = 3.30 kW

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Example 1

- The following observations were made during a test on a single cylinder two stroke cycle oil engine. Cylinder dimensions 20 cm bore, 26 cm stroke speed 350 rpm, effective diameter of brake wheel is 1.2 m. Net load on brake is 480 N, indicated mean effective pressure is 0.28 N/m². Fuel oil consumption is 3.6 kg/hr. Calorific value of fuel is 42000 kJ/kg. Calculate IP, BP, mechanical efficiency, brake thermal efficiency and specific fuel consumption in kg per BP hour.

Given data

Stroke length (L) = 26 cm = 0.26m

Engine speed (N) = 350 rpm

= 350 / 60 = 5.83 rps

Here n = N = 5.83 rps (since engine is 2 – stroke engine)

Effective brake wheel diameter (D) = 1.2 m

R = 1.2/2 =0.6 m

Net load on brake, (w-s) = 480N

No. of cylinders (k) = 1

Bore dia (d) = 20 cm = 0.2 m

Fuel consumption m_f = 3.6kg / hr = 3.6/3600 = 0.001 kg/sec

Calorific value CV = 42000 kJ / kg

i.m.e.p = 0.28 N/m² = $0.28 \times 1000 = 280 kN/m^2$

To find

Indicated power = ?

Brake power = ?

Mechanical efficiency = ?

Brake thermal efficiency = ?

Specific fuel consumption = ?

Solution

Hence area of cross section of cylinder

$$A = \frac{\pi}{4}(0.2)^2$$

$$= 0.78 \times 0.04$$

$$A = 0.03141$$

(i) Indicated power (IP)

$$= Pm \times L \times A \times n \times k$$

$$= 280 \times 0.26 \times 0.03141 \times 5.83 \times 1$$

$$IP = 13.33 \text{ kW}$$

(ii) Brake power

$$BP = 2\pi \times N \times (W - S) \times R$$

$$= 2\pi \times 5.83 \times 0.48 \times 0.6$$

Brake power, BP = 10.54 kW

(iii) Mechanical efficiency (η_{mech})

$$= BP / IP$$

$$= 10.54 / 13.33$$

$$\eta_{mech} = 0.7912$$

Mechanical efficiency = 79.12 %

(iv) Brake thermal efficiency

$$\eta_{bth} = \frac{BP}{(m_f \times CV)}$$

$$= \frac{10.54}{(0.001 \times 42000)}$$

Brake thermal efficiency (η_{bth}) = 0.25 or 25 %

(v) Specific fuel consumption

As we have determine specific fuel consumption per BP hour.

Specific fuel consumption

$$= \frac{\text{Mass of fuel consumed per hour}}{BP}$$

$$= \frac{3.6}{10.54}$$

$$= 0.341 \text{ kg/hr / BP}$$

$$\text{spfc} = 0.341 \text{ kg / BP} - \text{hr}$$

Result

- Indicated power = 13.33 kW
- Mechanical efficiency = 79.12 %
- Brake thermal efficiency = 25 %
- Specific fuel consumption = 0.341 kg per BP hour

-- PAGE END --

Example 2

- A single-cylinder four-stroke diesel engine gave the following results while running on full load :

Area of indicator card = 300 mm^2

Length of diagram = 40 mm

Spring constant = 1 bar/mm

Speed of the engine = 400 r.p.m.

Load on the brake = 370 N

Spring balance reading = 50N

Diameter of brake drum = 1.2 m

Fuel consumption = 2.8 kg/h

Calorific value of fuel = 41800 kJ/kg

Diameter of the cylinder = 160 mm

Stroke of the piston = 200 mm

Calculate (i) Indicated mean effective pressure, (ii) Brake power and brake mean effective pressure (iii) Brake specific fuel consumption, Brake thermal and indicated thermal efficiencies.

Given data

Area of indicator card = 300 mm^2

Length of diagram = 40 mm

Spring constant = 1 bar/mm

Speed of the engine (N) = 400 r.p.m.

Load on the brake (W) = 370 N

Spring balance reading (S) = 50N

Diameter of brake drum (D_b) = 1.2 m

Fuel consumption (m_f) = 2.8 kg/h

Calorific value of fuel (C) = 41800 kJ/kg

Diameter of the cylinder (D) = 160 mm = 0.16 m

Stroke of the piston (L) = 200 mm = 0.2 m

To find

Indicated mean effective pressure

Brake power and brake mean effective pressure

Brake specific fuel consumption,

Brake thermal and indicated thermal efficiencies.

Solution

(i) Indicated mean effective pressure

$$p_{mi} = \frac{\text{Area of indicator diagram on card} \times \text{spring constant}}{\text{Length of diagram}}$$

$$= \frac{300 \times 1}{40}$$

$$p_{mi} = 7.5 \text{ bar}$$

(ii) Indicated power,

$$\text{I.P} = K\left(\frac{P_m \ L \ A \ n}{60}\right)$$

$$= \frac{1 \times 7.5 \times 0.2 \times \pi/4 \times 0.16^2 \times \left(400 \times \frac{1}{2}\right) \times 100}{60}$$

$$= \frac{1.5 \times 0.785 \times 0.0256 \times 400 \times 0.5 \times 100}{60}$$

$$= 10.048 \text{ (or) } 10.05 \text{ kW}$$

(iii) Brake power,

$$\text{B.P} = \frac{(W - S) \ \pi \ D_b \ N}{60 \times 1000}$$

$$= \frac{(370 - 50) \ \pi \times 1.2 \times 400}{60 \times 1000}$$

$$= \frac{482548.63}{60000}$$

$$B.P = 8.04 \text{ kW}$$

Also,

$$B.P = \frac{n\, p_{mb} \times LANk \times 100}{60}$$

$$8.04 = \frac{1 \times p_{mb} \times 0.2 \times \pi/4 \times 0.16^2 \times 400 \times \frac{1}{2} \times 100}{60}$$

$$p_{mb} = \frac{8.04 \times 6 \times 4 \times 2}{0.2 \times \pi \times 0.16^2 \times 400 \times 10}$$

$$p_{mb} = 6 \text{ bar}$$

(iv) Brake specific fuel consumption

$$b.s.f.c = \text{fuel consumption per B.P. hour}$$

$$= \frac{2.8}{8.04}$$

$$= 0.348 \text{ kg/ B.P. hour}$$

(v) Brake thermal efficiency,

$$\eta_{thb} = \frac{B.P}{m_f \times C.V}$$

$$= \frac{8.04}{\frac{2.8}{3600} \times 41800}$$

$$= \frac{8.04}{7.77 \times 10^{-4} \times 41800}$$

$$= \frac{8.04}{32.51}$$

$$\eta_{thb} = 0.2473 \text{ or } 24.73\ \%$$

(vi) Indicated thermal efficiency,

$$\eta_{th\,I} = \frac{I.P}{m_f \times C.V}$$

$$= \frac{10.05}{\frac{2.8}{3600} \times 41800}$$

$$= \frac{10.05}{7.77 \times 10^{-4} \times 41800}$$

$$= 0.3091 \text{ or } 30.91\ \%$$

Result

- Indicated mean effective pressure = 7.5 bar
- Brake power = 8.004 kW
- Brake mean effective pressure = 6 bar
- Brake specific fuel consumption = 0.348 kg/B.P hour
- Brake thermal efficiency = 24.73 %
- Indicated thermal efficiency = 30.91 %

-- PAGE END --

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$= 350 / 60 = 5.83$ rps

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Effective brake wheel diameter (D) = 1.2 m

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Net load on brake, (w-s) = 480N

No. of cylinders (k) = 1

Bore dia (d) = 20 cm = 0.2 m

Fuel consumption $m_f = 3.6\text{kg} / \text{hr} = 3.6/3600 = 0.001$ kg/sec

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$$\text{i.m.e.p} = 0.28\ N/m^2 = 0.28 \times 1000 = 280 kN/m^2$$

To find

Indicated power = ?

Brake power = ?

Mechanical efficiency = ?

Brake thermal efficiency = ?

Specific fuel consumption = ?

Solution

Hence area of cross section of cylinder

$$A = \frac{\pi}{4}(0.2)^2$$

$$= 0.78 \times 0.04$$

$$A = 0.03141$$

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$$= 280 \times 0.26 \times 0.03141 \times 5.83 \times 1$$

$$IP = 13.33 \text{ kW}$$

(ii) Brake power

$$BP = 2\pi \times N \times (W - S) \times R$$

$$= 2\pi \times 5.83 \times 0.48 \times 0.6$$

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$$\eta_{mech} = 0.7912$$

Mechanical efficiency = 79.12 %

(iv) Brake thermal efficiency

$$\eta_{bth} = \frac{BP}{(m_f \times CV)}$$

$$= \frac{10.54}{(0.001 \times 42000)}$$

Brake thermal efficiency (η_{bth}) = 0.25 or 25 %

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As we have determine specific fuel consumption per BP hour.

Specific fuel consumption

$$= \frac{\text{Mass of fuel consumed per hour}}{\text{BP}}$$

$$= \frac{3.6}{10.54}$$

$$= 0.341 \text{ kg/hr / BP}$$

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- Brake thermal efficiency = 25 %
- Specific fuel consumption = 0.341 kg per BP hour

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Example 2

- A single-cylinder four-stroke diesel engine gave the following results while running on full load :

  Area of indicator card = 300 mm^2

  Length of diagram = 40 mm

  Spring constant = 1 bar/mm

  Speed of the engine = 400 r.p.m.

  Load on the brake = 370 N

  Spring balance reading = 50N

  Diameter of brake drum = 1.2 m

  Fuel consumption = 2.8 kg/h

  Calorific value of fuel = 41800 kJ/kg

  Diameter of the cylinder = 160 mm

  Stroke of the piston = 200 mm

  Calculate (i) Indicated mean effective pressure, (ii) Brake power and brake mean effective pressure (iii) Brake specific fuel consumption, Brake thermal and indicated thermal efficiencies.

Given data

Area of indicator card = 300 mm^2

Length of diagram = 40 mm

Spring constant = 1 bar/mm

Speed of the engine (N) = 400 r.p.m.

Load on the brake (W) = 370 N

Spring balance reading (S) = 50N

Diameter of brake drum (D_b) = 1.2 m

Fuel consumption (m_f) = 2.8 kg/h

Calorific value of fuel (C) = 41800 kJ/kg

Diameter of the cylinder (D) = 160 mm = 0.16 m

Stroke of the piston (L) = 200 mm = 0.2 m

To find

Indicated mean effective pressure

Brake power and brake mean effective pressure

Brake specific fuel consumption,

Brake thermal and indicated thermal efficiencies.

Solution

(i) Indicated mean effective pressure

$$p_{mi} = \frac{\text{Area of indicator diagram on card} \times \text{spring constant}}{\text{Length of diagram}}$$

$$= \frac{300 \times 1}{40}$$

$$p_{mi} = 7.5 \text{ bar}$$

(ii) Indicated power,

$$\text{I.P} = K\left(\frac{P_m \ L \ A \ n}{60}\right)$$

$$= \frac{1 \times 7.5 \times 0.2 \times \pi/4 \times 0.16^2 \times \left(400 \times \frac{1}{2}\right) \times 100}{60}$$

$$= \frac{1.5 \times 0.785 \times 0.0256 \times 400 \times 0.5 \times 100}{60}$$

$$= 10.048 \text{ (or) } 10.05 \text{ kW}$$

(iii) Brake power,

$$\text{B.P} = \frac{(W - S) \ \pi \ D_b \ N}{60 \times 1000}$$

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Also,

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$$p_{mb} = \frac{8.04 \times 6 \times 4 \times 2}{0.2 \times \pi \times 0.16^2 \times 400 \times 10}$$

$$p_{mb} = 6 \text{ bar}$$

(iv) Brake specific fuel consumption

$$b.s.f.c = \text{fuel consumption per B.P. hour}$$

$$= \frac{2.8}{8.04}$$

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(v) Brake thermal efficiency,

$$\eta_{thb} = \frac{B.P}{m_f \times C.V}$$

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$$= \frac{8.04}{32.51}$$

$$\eta_{thb} = 0.2473 \text{ or } 24.73\ \%$$

(vi) Indicated thermal efficiency,

$$\eta_{th\,I} = \frac{I.P}{m_f \times C.V}$$

$$= \frac{10.05}{\frac{2.8}{3600} \times 41800}$$

$$= \frac{10.05}{7.77 \times 10^{-4} \times 41800}$$

$$= 0.3091 \text{ or } 30.91\,\%$$

Result

- Indicated mean effective pressure = 7.5 bar
- Brake power = 8.004 kW
- Brake mean effective pressure = 6 bar
- Brake specific fuel consumption = 0.348 kg/B.P hour
- Brake thermal efficiency = 24.73 %
- Indicated thermal efficiency = 30.91 %

-- PAGE END --

Example 1

- The air flow a four cylinder four stroke oil engine is measured by means of a 5 cm diameter orifice, having a coefficient of discharge of 0.6. During a test on the engine the following data were recorded:

  Bore, 10.5 cm;

  Engine speed, 1200 rev/min;

  Brake torque, 147 Nm;

  Fuel consumption, 5.5 kg/hr

  Calorific value of fuel 43100 kJ/hr;

  Head across orifice, 5.7 cm of water;

  Ambient temperature and pressure are 200c and 1.013 bar respectively.

  Calculate

  (i) The thermal efficiency on b.p. basis

  (ii) The brake mean effective pressure

  The volumetric efficiency based on free air conditions.

  Given data

  Orifice diameter = 5 cm

  Coefficient of discharge = 0.6

  Bore = 10.5 cm

  Engine speed = 1200 rev/min

  Brake torque = 147 Nm

  Fuel consumption = 5.5 kg/hr

  Calorific value of fuel = 43100 kJ/hr

  Head across orifice = 5.7 cm of water

  Ambient temperature = 20°C

  Ambient pressure = 1.013 bar

  To find

Thermal efficiency on brake power basis

Brake mean effective pressure

Solution

Brake power

$$= 2\pi\ NT$$

$$= 2\ \pi\left(\frac{1200}{60}\right) \times 147 \times \frac{1}{10^{-3}}$$

$$= 2\ \pi \times 20 \times 147 \times \frac{1}{10^{-3}}$$

$$= 18.5\ \text{kW}$$

Thermal efficiency

$$= \frac{\text{b.p} \times 3600}{m_f \times CV}$$

$$\eta_t = \frac{18.5 \times 3600}{5.5 \times 43100}$$

$$= \frac{66600}{237050}$$

$$\eta_t = 0.2809 \text{ or } 28.1\ \%$$

Brake mean effective pressure

$$\text{bmep} = \frac{\text{b.p}}{\text{LAN} \times \text{No of cylinders}}$$

$$= \frac{18.5 \times 10^3}{0.125 \times (\pi / 4)(0.105)^2 \times \left(\frac{1200}{(2 \times 60)}\right) \times 4} \times \frac{1}{10^5}$$

$$= \frac{18.5 \times 10^3}{0.125 \times 0.78 \times 0.0110 \times 10 \times 4} \times \frac{1}{10^5}$$

$$= \frac{18.5 \times 10^3}{0.0429}$$

Brake mean effective pressure = 4.31 bar

Result

- Thermal efficiency on brake power = 28.1%
- Brake mean effective pressure = 4.31 bar

-- PAGE END --

Example 2

- A four cylinder, four stroke cycle engine, 82.5 mm bore × 130 mm stroke develops 28 kW while running at 1500 r.p.m and using a 20 percent rich mixture. If the volume of the air in the cylinder when measured at 15.5°C and 762 mm of mercury is 70 percent of the swept volume, the theoretical air fuel ratio is 14.8, heating value of petrol used is 45980 kJ/kg and the mechanical efficiency of the engine is 90%, find

  (i) The indicated thermal efficiency

  (ii) The brake mean effective pressure

  Take R = 287 N-m/kg K.

Given data

Stroke, l = 130 mm

Diameter, = 82.5

Temperature of the air (T) = 15.5°C

Swept volume = 70%

Mechanical efficiency of the engine = 90%,

Calorific value – 45980 kJ/kg

Theoretical air fuel ratio is 14.8

R = 287 N-m/kg K

To find

Indicated thermal efficiency = ?

Brake mean effective pressure = ?

Solution

Indicated thermal efficiency, $\eta_{th\,I}$

Swept volume,

$$V_s = \frac{\pi}{4} D^2 L$$

$$= \frac{\pi}{4} \times 0.0825^2 \times 0.13$$

$$= 0.000690 \text{ m}^3$$

Volume of air drawn in

$$= \frac{70}{100} \times 0.000690$$

$$= 0.0004865 \text{ m}^3$$

$$p = \frac{762}{760} \times 1.0132$$

$$p = 1.015 \text{ bar}$$

$$V = 0.0004865 \text{ m}^3$$

$$R = 287 \text{ N-m /kg K}$$

$$T = 15.5 + 273 = 288.5 \text{ K}$$

$$m = \text{Mass of air}$$

$$m = \frac{pV}{RT}$$

$$= \frac{1.015 \times 10^5 \times 0.0004865}{287 \times 288.5}$$

$$m = 0.000596 \text{ kg}$$

Theoretical mass of air used per minute

$$= 0.000596 \times \frac{1500}{2} \times 4$$

$$= 0.000596 \times 750 \times 4$$

$$= 1.788 \text{ kg}$$

Theoretical air fuel ratio = 14.8

Theoretical mass of fuel used/min

$$= \frac{1.788}{14.8}$$

$$= 0.1208 \text{ kg/min}$$

When using 20% rich mixture, then

m_f = mass of fuel burnt/sec

$$= \frac{0.1208}{60} \times \frac{120}{100}$$

$$m_f = 0.002416 \text{ kg/s}$$

$$\eta_{mech} = \frac{B.P}{I.P} = \frac{28}{I.P}$$

$$0.9 = \frac{28}{I.P}$$

$$0.9 = \frac{28}{I.P}$$

$$I.P = \frac{28}{0.9}$$

$$IP = 31.11 \text{ kW}$$

(i) Indicated thermal efficiency

$$\eta_{th\,I} = \frac{I.P}{m_f \times C}$$

$$= \frac{31.11}{0.002416 \times 45980}$$

$$= \frac{31.11}{111.08}$$

$$\eta_{th\,I} = 0.28 \text{ or } 28\ \%$$

(ii) Brake mean effective pressure, p_{mb}

$$B.P = \frac{n \times p_{mb}\, LAN\, k \times 10}{6}$$

$$28 = \frac{4 \times p_{mb} \times 0.13 \times \pi / 4 \times 0.0825^2 \times 1500 \times \frac{1}{2} \times 10}{6}$$

$$p_{mb} = \frac{28 \times 6 \times 4 \times 2}{4 \times 0.13 \times \pi \times 0.0825^2 \times 1500 \times 10}$$

$$p_{mb} = \frac{1344}{166.78}$$

$$p_{mb} = 8.058\ (\text{or})\ 8.06 \text{ bar}$$

Result

- Indicated thermal efficiency = 28 %
- Brake mean effective pressure = 8.06 bar

-- PAGE END --

Example 1

- The following particulars were obtained in a trial on a four stroke gas engine.

  Duration of trial = 1 hour

  Revolutions = 14000

  Number of missed cycle = 500

  Net brake load = 1470 N

  Mean effective pressure = 7.5 bar

  Gas consumption = 20000 litres

  L.C.V. of gas at supply condition = 21 kJ/litre

  Cylinder diameter = 250 mm

  Stroke = 400 mm

  Effective brake circumference = 4 m

  Compression radio = 6.5 : 1

  Calculate:

  (i) Indicated power

  (ii) Brake power

  (iii) Mechanical efficiency

  (iv) Indicated thermal efficiency

  (v) Relative efficiency

Given data

Duration of trial = 1 hour = 60 min

Revolutions = 14000 / hour

Number of missed cycle = 500

Net brake load = 1470 N

Mean effective pressure = 7.5 bar = $7.5 \times 100 = 7500 \text{ kN/m}^2$

Gas consumption = 20,000 litres

L.C.V. of gas at supply condition = 21 kJ/litre

Cylinder diameter = 250 mm

Stroke = 400 mm

Effective brake circumference = 4 m

Compression radio = 6.5 : 1

To find

Indicated power = ?

Brake power = ?

Mechanical efficiency = ?

Indicated thermal efficiency = ?

Relative efficiency = ?

Solution

Speed,

$$N = \frac{14000}{60}$$

$$= \frac{700}{3} = 233.3 \text{ r.p.m} = 3.888 \text{ rps}$$

Number of missed cycle per hour = 500

$$\text{Number of missed cycle per sec} = \frac{500}{3600}$$

$$\text{Number of actual fires / sec} = \left[\frac{3.888}{2} - \frac{500}{3600}\right]$$

$$= 1.8051$$

$$W - S = 1470 \text{ N}$$

$$p_m = 7500 \text{ kN/m}^2$$

$$V_g = \frac{20000}{3600}$$

$$V_g = 5.55 \text{ litres/s}$$

$$D = 250 \text{ mm} = 0.25 \text{ m}$$

$$L = 400 \text{ mm} = 0.4 \text{ m}$$

$$\pi D_b = 4 \text{ m}, r = 6.5, n = 1$$

(i) Indicated power, I.P

$$I.P = \frac{p_m \, L \, A \, N \, k \times 100}{60}$$

$$I.P = \frac{7500 \times 0.4 \times \pi / 4 \times 0.25^2 \times (6500/60)}{60}$$

$$= \frac{7.5 \times 0.4 \times 0.78 \times 0.0625 \times 108.33 \times 100}{60}$$

$$I.P = 26.40 \text{ kW}$$

Brake power, B.P

$$B.P = \frac{(W - S) \, \pi \, D_b \, N}{60 \times 1000}$$

$$= \frac{1470 \times 4 \times (700/3)}{60 \times 1000}$$

$$= \frac{1470 \times 4 \times 233.33}{60 \times 1000}$$

$$= \frac{1372000}{60000}$$

$$B.P = 22.86 \text{ kW}$$

Mechanical efficiency, η_{mech}

$$\eta_{mech} = \frac{B.P}{I.P}$$

$$= \frac{22.86}{26.59}$$

$$\eta_{mech} = 0.859 \text{ (or) } 85.9\,\%$$

Indicated thermal efficiency, $\eta_{th\,I}$

$$\eta_{th\,I} = \frac{I.P}{V_g \times C}$$

$$= \frac{26.59}{5.5 \times 21}$$

$$\eta_{th\,I} = 0.23 \text{ or } 23\,\%$$

Relative efficiency,

$$\eta_{relative} = \frac{\eta_{thermal}}{\eta_{air\ standard}}$$

But, $\eta_{air\ standard} = 1 - \frac{1}{(r)^{\gamma-1}}$

$$= 1 - \frac{1}{(6.5)^{1.4-1}}$$

$$= 1 - \frac{1}{(6.5)^{0.4}}$$

$$= 1 - 0.472$$

$$\eta_{air\ standard} = 0.527 \text{ or } 52.7\,\%$$

$$\eta_{relative} = \frac{0.23}{0.527}$$

$$\eta_{relative} = 0.436 \text{ or } 43.6\,\%$$

Result

- Indicated power = 26.59 kW
- Brake power = 22.86 kW
- Mechanical efficiency = 85.9 %
- Indicated thermal efficiency = 23 %
- Relative efficiency = 43.6 %

-- PAGE END --

Example 2

- Air consumption for a four-stroke petrol engine is measured by means of a or circular orifice of diameter 3.2 cm. The coefficient of discharge for the orifice is 0.62 and the pressure acorss the orifice is 150 mm of water. The barometer reads 760 mm of Hg. Temperature of air in the room is 20°C. The piston displacement volume is 0.00178 m^3. The compression ratio is 6.5. The fuel consumption is 0.135 kg/min of calorific value 43900 kJ/kg. The brake power developed at 2500 r.p.m is 28 kW. Determine :

(i) The volumetric efficiency on the basis of air alone.

(ii) The air-fuel ratio.

(iii) The brake mean effective pressure.

(iv) The relative efficiency on the brake thermal efficiency basis.

Given data

Circular orifice of diameter = 3.2 cm.

Coefficient of discharge for the orifice = 0.62 and the

Pressure acorss the orifice = 150 mm of water.

Barometer reads 760 mm of Hg.

Temperature of air in the room = 20°C.

Piston displacement volume = 0.00178 m^3

Compression ratio = 6.5

Fuel consumption = 0.135 kg/min

Calorific value 43900 kJ/kg.

Brake power developed at 2500 r.p.m is 28 kW.

To find

Volumetric efficiency on the basis of air alone.

Air-fuel ratio.

Brake mean effective pressure.

Relative efficiency on the brake thermal efficiency basis.

Solution

Diameter of circular orifice, d = 3.2 cm = 0.032 m

Coefficient of discharge, Cd = 0.62

Pressure across orifice, hw = 150 mm of water

Temperature of air in the room = 20°C

Piston displacement =0.00178 m^3

Compression ratio, r = 6.5

Fuel consumption = 0.135 kg/min

Calorific value of fuel, CV = 43900 kJ/kg

Brake power ,B.P = 28 kW

Speed = 2500 r.p.m

$k = \frac{1}{2}$ .... for four stroke cycle

Volumetric efficiency on the basis of air alone

$$p\,V = m\,R\,T$$

$$\frac{m}{V} = \frac{p}{R\,T}$$

$$= \frac{1.0132 \times 10^5}{287 \times (20 + 273)}$$

$$\frac{m}{V} = 1.2\,\text{kg/m}^3$$

Also, 150 mm of H_2O,

$$= \frac{150}{1000} \times 1000 = 150\,\text{kg/m}^2$$

Thus head of air column causing flow,

$$H = \frac{150}{1.2} = 125\,\text{m}$$

Thus air flow through the orifice = Air consumption

$$= C_d \times A \times \sqrt{2gH}$$

$$= 0.62 \times \frac{\pi}{4} \times (0.032)^2 \times \sqrt{2 \times 9.81 \times 125}$$

$$= 0.0247 \text{ m}^3/\text{s}$$

Therefore, air consumption per stroke

$$= \frac{0.0247 \times 60}{\left(\frac{2500}{2}\right)}$$

$$= 0.001185 \text{ m}^3$$

Volumetric efficiency,

$$\eta_{vol} = \frac{\text{Air consumption of stroke}}{\text{Piston displacement}}$$

$$= \frac{0.001185}{0.00178}$$

$$\eta_{vol} = 0.665 \text{ (or) } 66.5\%$$

Air fuel ratio

Mass of air drawn into the cylinder per min

$$= 0.0247 \times 60 \times 1.2$$

$$= 1.778 \text{ kg/min}$$

Air fuel ratio

$$= \frac{1.778}{0.135} = 13.67 : 1$$

Brake mean effective pressure, p_{mb}

$$\text{B.P} = \frac{p_{mb} \text{ L A N k} \times 100}{60}$$

$$28 = \frac{1 \times p_{mb} \times 0.00178 \times 2500 \times \frac{1}{2} \times 100}{60}$$

$$p_{mb} = \frac{28 \times 60 \times 2}{0.00178 \times 2500 \times 100}$$

$$p_{mb} = 7.55 \text{ bar}$$

Relative efficiency,

$$\eta_{\text{air standard}} = 1 - \frac{1}{(r)^{\gamma - 1}}$$

$$= 1 - \frac{1}{(6.5)^{1.4 - 1}}$$

$$= 0.527 \text{ or } 52.7\%$$

Brake thermal efficiency,

$$\eta_{\text{th b}} = \frac{B.P}{m_f \times C}$$

$$= \frac{28}{\frac{0.135}{60} \times 43900}$$

$$\eta_{\text{th. B}} = 0.2835 \text{ or } 28.35\ \%$$

$$\eta_{\text{relative}} = \frac{\eta_{\text{thermal}}}{\eta_{\text{airstandard}}}$$

$$= \frac{0.2835}{0.527}$$

$$\eta_{\text{relative}} = 0.5379 \text{ or } 53.79\ \%$$

Result

- Volumetric efficiency = 66.5 %
- Air-fuel ratio = 13.67 : 1
- Brake mean effective pressure = 7.55 bar
- Relative efficiency on the brake = 52.7 %
- Thermal efficiency = 28.35 %

-- PAGE END --

Heat balance sheet

- Energy can only be converted from one form to another form.

    Eg., Heat energy to mechanical energy

- Heat balance sheet shows how the total heat liberated by combustion fuel is utilized and how much energy is lost.
- It is a statement showing the account of heat supplied and heat utilized.
- It is a statement that gives information governing the performance of engine.
- Heat balance is done on second basis or minute basis or hour basis. (Time basis).
- Heat supplied to the engine is given by $(Q_s) = m_f \times C_v$

    where, m_f = mass of fuel

    C_V = calorific value of fuel

- Heat utilised by the engine :
    - Heat equivalent to B.P.
    - Heat carried away by cooling water.
    - Heat carried away by exhaust gases.
    - Unaccounted heat.

Heat equivalent to B.P

- Heat equivalent of B.P kW = B.P × 60 kJ/min.

Heat carried away by cooling water

- Heat carried away by cooling water = $m_w C_w(T_2 - T_1)$ kJ/min.

  Where,

  m_w = mass of cooling water in kg,

  C_w = specific heat of water

  $(T_2 - T_1)$ = rise in temperature of cooling water

Heat carried away by exhaust gases

- Heat carried away by exhaust gases = $m_g C_{pg} (T_g - T_r)$ (kJ/min.) or (kJ/sec).

  Where, mg is the mass of exhaust gases in kg/min. or kg/sec

T_g = Temperature of burnt gases coming out of the engine.

T_r = Room temperature.

C_{pg} = Sp. Heat of exhaust gases in (kJ/kg-K)

M_g = Mass of air (m_a) + Mass of fuel (m_f) = $m_a + m_f$

$$= m_f + \left(\frac{m_a}{m_f} + 1\right)$$

$$= m_f\,(AFR + 1)$$

Where AFR = Air fuel ratio

Some of heat is lost by convection or radiation, as well as

- Due to leakages of gases,
- Part of the power is used to run accessories,
- Such as lubricating pump, cam shaft, water circulating pump.
- These cannot be measured precisely and so known as unaccounted losses or energy to surroundings.
- Un accounted losses are equal to the difference between heat supplied and sum of a,b,c.

heat unaccounted

$$G = Qs - (a+b+c)$$

- The results of the above calculations are tabulated in a table and this table is known as "Heat Balance Sheet".
- It is generally practice to represent the heat distribution as percentage of heat supplied. This is also tabulated in the same heat balance sheet.

Heat input per minute	kJ	%	Heat utilised or energy distribution per minute	kJ	%
Heat supplied by the combustion on of fuel	Q_s	100%	(a) Heat equivalent of B.P (b) Heat carried by cooling water (c) Heat carried by exhaust gasses (d) Heat unaccounted		
Total	Q_s	**100%**			**100%**

-- PAGE END --

Example 1

- Results obtained in a full load test performed on an oil engine are as follows :-

  Brake power, B. P = 24 kW ;

  Fuel consumption = 0.128 kg/min

  Water circulating the cylinder = 5.9 kg/min;

  Temperature rise of cooling water = 49.5°c;

  Temperature of exhaust gases = 387.8°c.

  Temperature of engine room = 18.4°c

  Air fuel ratio = 20

  Calorific value of oil = 45200 kJ/kg

  Specific heat of exhaust gas = 1.05kJ/kg° c

  Specific heat of water = 4.2 kJ/ kg° c

  Prepare heat balance sheet in kJ/min.

Given data

Brake power, B. P = 24 kW ;

Fuel consumption = 0.128 kg/min

Water circulating the cylinder = 5.9 kg/min;

Temperature rise of cooling water = 49.5°c;

Temperature of exhaust gases = 387.8°c.

Temperature of engine room = 18.4°c

Air fuel ratio = 20

Calorific value of oil = 45200 kJ/kg

Specific heat of exhaust gas = 1.05kJ/kg° c

Specific heat of water = 4.2 kJ/ kg° c

To find

Heat balance sheet

Solution

Heat supplied to the engine,

$$Q_s = mf \times Cv$$

$$= 0.128 \times 45200$$

$$Q_s = 5785.6 \text{ kJ/min}$$

(a) Heat energy equivalent to B.P

$$= 24 \times 60$$

$$= 1440 \text{ kJ/min}$$

(b) Heat carried away by cooling water

$$= m_w C (t_2 - t_1)$$

$$= 5.9 \times 4.2 \times 49.5$$

$$= 1226.6 \text{ kJ/min}$$

(c) Heat carried away by exhaust gases

$$= m_f (AFR+1) C_g (t_g - t_r)$$

$$(\because\ m_g = m_f(AFR+1)\ )$$

$$= 0.128 (20 + 1) 1.05(387.8 - 18.4)$$

$$= 1042.56 \text{ kJ/min}$$

(d) Heat un accounted or energy to surroundings

$$= Q_s - (a+b+c)$$

$$= 5785.6(1440 + 1226.6 + 1042.56)$$

$$= 2076.44 \text{ kJ/min}$$

Heat input	kJ/ min	%	Heat utilised or energy distribution	kJ	%
Heat supplied by the fuel	5785.6	100%	(a) Energy to B.P	1440	24.88
			(b) Heat carried by cooling water	1226.6	21.19
			(c) Heat carried by exhaust gasses	1042.56	18.01
			(d) Heat unaccounted	2076.44	35.92
Total	**5785.6**	**100%**		**5785.60**	**100%**

-- PAGE END --

Morse test

- It is the method of determining indicated power (ip) of each cylinder individually, of a multi cylinder IC engine, without the use of an indicator and thus computing the 'total ip of the engine' by summing up ip of all the cylinders.
- This method is adopted to calculate ip of high speed engines, i.e. where the indicator method is unsuitable.

-- PAGE END --

Testing of constant speed IC engines for general purposes according to IS: 1600-1960

Introduction:

- This code applies to testing of constant speed reciprocating internal combustion engines of the following types used for general purposes.
  - Compression ignition engines.
  - Carburettor type engines, and
  - Gas engines.
- This code is not applicable to pressure charged engines, engines for road or rail traction, engines for ships propulsion or for marine auxiliaries and engines for aircraft propulsion or aircraft auxiliaries.

General requirements for tests

- The manufacturer shall supply the performance characteristics of the engine prior to the commencement of the tests.
- The engine shall be tested as offered to the purchaser.
- All parts shall be in stock and all parts essential for engine operation should be included.
- Accessories used on the engine under test shall be listed.

Preparation for tests

- The engine shall be completely stripped and examined physically so that design features and also the condition of the various parts may be noted before tests are, commenced.
- After the physical examination the dimensions of the main working parts, listed below shall be checked and recorded.
- Cylinder head.
- Valves, valve seats, valve springs and valve guides.
- Cylinder liner.
- Piston assembly.
- Connecting rod small end big end bearings and connecting rod bolts.
- Crankshaft, including bearings and journals &

- Governer springs.

Preliminary run

- The engine shall be subjected to a preliminary run of 49 hours at rated speed under operating temperatures as specified by the manufacture, in non stop cycle of 7 hours each, conforming to the following cycle, the period of each run being a minimum of one cycle:

Load	Running time (hr)
25% of rated load	$1\frac{1}{2}$
50% of rated load	2
75% of rated load	$1\frac{1}{2}$
100% of rated load	2

- During the preliminary run, special attention shall be paid to engine vibration and quiteness.
- The oil pressure shall be checked from time to time.
- Oil, coolant and fuel leaks shall be rectified and faculty components replaced as may be found necessary.
- A complete record of such attention and running time of components changed shall be kept.

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Test procedure

Engine adjustment:

- The distributor, carburetor or the fuel pump rack, as the case may be set as its nominal specified value at idling in contrast to its manual adjustments for maximum power at each speed.

Temperature:

- The temperature of the inlet air shall be measured at the entrance of the induction system.

Number of runs:

- In every test, a sufficient number of runs shall be made throughout the speed range. A run shall be made at the lowest steady at which the engine operates.

Duration of runs:

- Performance data shall be obtained under stabilized operating conditions. Durations of the experimental run depends upon two principles:
  - No data shall be taken until load, speed and temperature have been stabilized.
  - Recorded data shall be average sustained values maintained over a period of at least one minute, with no significant change occuring during that time.

Power test:

- For all power tests with results to be plotted versus speed, a single series of stabilized runs at ascending speeds is sufficient.
- This series of runs should progress continuously, from the lowest to the maximum.
- If the engine requires to be idled between runs to avoid excessively high temperature, sufficient time should for the engine to reach its stabilized condition before taking readings.
- The brake load recorded should be steady and constant throughout the run.

Engine speed:

- Engine speed should be held constant as possible by means of applied dynamometer load at wide open throttle or by throttle adjustment at part load.

Friction power:

- The friction power test shall, if possible, follow immediately after the power test.

- If this is not possible, the test shall be conducted under condition similar to those for the power test.

Fuel consumption:

- Fuel consumption shall be measured simultaneously with brake power.
- The fuel consumption measurement shall not be started un-till the engine is stabilized.

-- PAGE END --

Frequently Asked Questions

- What is the need of engine performance test ?
- what are the various tests conducted to evaluate the performence of I.C.engines ?

-- PAGE END --

Frequently Asked Questions

- Define the following terms:
    - Indicated power
    - Brake power
    - Frictional power

-- PAGE END --

Frequently Asked Questions

- Define the following,
  - Mechanical Efficiency
  - Indicated Thermal Efficiency
  - Brake Thermal Efficiency

-- PAGE END --

Frequently Asked Questions

- Define specific fuel consumption based on indicated power.
- Define specific fuel consumption based on brake power.

-- PAGE END --

Frequently Asked Questions

- Define volumetric efficiency.
- Write factors effecting the volumetric efficiency.

-- PAGE END --

Frequently Asked Questions

- The following observations were made during a trial on 2-stroke engine for half an hour when it was running at 300 rpm

    Stroke: 300 mm

    bore: 200 mm

    MEP: 6 bar

    reading of spring balances : 1390 N and 90 N

    mean circumference of break drum : 3600 mm

    fuel consumed: 4.35 kg

    C.V of fuel: 44000 kJ/kg

    Determine I.P, B.P

-- PAGE END --

Frequently Asked Questions

- The following particulars refer to a single cylinder of engine having cylinder diameter 250 mm, stroke 400 mm and working as 4-stroke cycle

  Speed: 250 rpm

  gross MEP: 7.25 bar

  pumping MEP: 0.75 bar

  net load : 1080 N

  effective wheel diameter : 1.6

  Determine (a) B.P (b) mechanical efficiency

-- PAGE END --

Frequently Asked Questions

- The following observations were made during a test on a single cylinder two stroke cycle oil engine. Cylinder dimensions 150 mm bore, 200 mm stroke speed 500 rpm, effective diameter of brake wheel is 1.5 m. Net load on brake is 500N, indicated mean effective pressure is 0.3 N / m^2. Fuel oil consumption is 3.5 kg/hr. Calorific value of fuel is 42000 kJ/kg. Calculate IP, BP, mechanical efficiency, brake thermal efficiency and specific fuel consumption in kg per BP hour.

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Frequently Asked Questions

- The air flow a four cylinder four stroke oil engine is measured by means of a 40 mm diameter orifice, having a coefficient of discharge of 0.6. During a test on the engine the following data were recorded:

  (i) Bore, 120 mm;

  (ii) Engine speed, 1500 rev/min;

  (iii) Brake torque, 150 Nm;

  (iv) Fuel consumption, 6 kg/hr

  (v) Calorific value of fuel 45000 kJ / kg;

  (iv) Head across orifice, 60 mm of water;

  Ambient temperature and pressure are $20^{\circ}c$ and 1.013 bar respectively.

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Frequently Asked Questions

- A single cylindrical oil engine working on 4-stroke cycle has bore of 100 mm and 150 mm and runs at 750 r.p.m. The mean effective pressure is 6 bar. It consumed 15 cc of fuel in 28 seconds. The diesel oil used is having a C.V of 42,000 kJ/kg and specific gravity of 0.85. The brake wheel diameter is 980 mm and rope diameter is 20 mm. The load on brake drum is 150 N and spring balances reads 20 N.

Calculate

(i) I.P

(ii) B.P

(iii) Mechanical efficiency

(iv) Indicated thermal efficiency

(v) Brake thermal efficiency

-- PAGE END --

Frequently Asked Questions

- Prepare an heat balance sheet using the following results obtained during performance test of I.C engine.

  B.P=25 kW; m_f = 0.281 kg/min; mw=4.8 kg/min; $(t_2 - t_1)$=46.8°c; t_g =391.8°c; t_{room} =190°c; m_g = 2.69 kg/min; C_v of oil =46200 kJ/kg°c ; sp. Heat of gases =1.05 kJ/kg°c ; sp. Heat of water = 4.2 kJ/kg°c.

-- PAGE END --

Frequently Asked Questions

- A four stroke gas engine has a cylinder diameter of 25 cm and stroke 45 cm. The effective diameter of the brake is 1.6 m. The observations made in a test of the engine were as follows. Duration of test = 40 min; Total number of revolutions = 8080; Total number of explosions = 3230; Net load on the brake = 80 kg; mean effective pressure = 5.7 bar; Volume of gas used = 7 m^3; Pressure of gas indicated in meter = 136 mm of water (gauge); Atmospheric temperature = 15°C; Calorific value of gas = 17 MJ/m^3 at NTP; Temperature rise of cooling water = 40°C; Cooling water supplied = 170 kg. Draw up a heat balance sheet and find the indicated thermal efficiency and brake thermal efficiency. Assume atmospheric pressure to be 760 mm of mercury.

-- PAGE END --

Frequently Asked Questions

- Define Morse test.
- Explain the testing of constant speed IC engines for general purposes.

-- PAGE END --

Air compressor

- Compression of air and vapour plays an important role in engineering fields.

Fig 1: Air compressor

- Compression of air is mostly used since it is easy to transmit air compared with vapour.

- The source of air compressor is air ,which can be received from the atmosphere.
- A machine which receives air, compresses (increase in pressure) and then stores.
- The functions of a compressor is to take a definite quantity of fluid (usually gas, and most often air) and deliver it at a required pressure.

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Applications of compressed air

- Compressed air has wide applications in industries as well as in commercial equipment.
- Air refrigeration and cooling of large buildings
- Driving pneumatic tools in shops like drills, riveters, screw drivers, etc.
- Driving air motors in mines, where electric motors and IC engines cannot be used because of fire risks due to the presence of inflammable gases, etc.

Fig 2: Applications of compressed air

- Cleaning purposes
- Blast furnaces
- Spray painting and spraying fuel in Diesel engines
- Hard excavation work, turning, boring, mining, etc.
- Starting of heavy-duty diesel engines
- Operating air brakes in buses, trucks and trains etc.
- Inflating automobile and aircraft tyres
- Supercharging internal combustion engines

- Conveying solid and powder materials in pipelines
- Process industries
- Operating lifts, hoists, cranes and to operate pumps etc.
- Pump sets for oil and gas transmission line
- Automobile suspension system.

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Working principle of a compressor

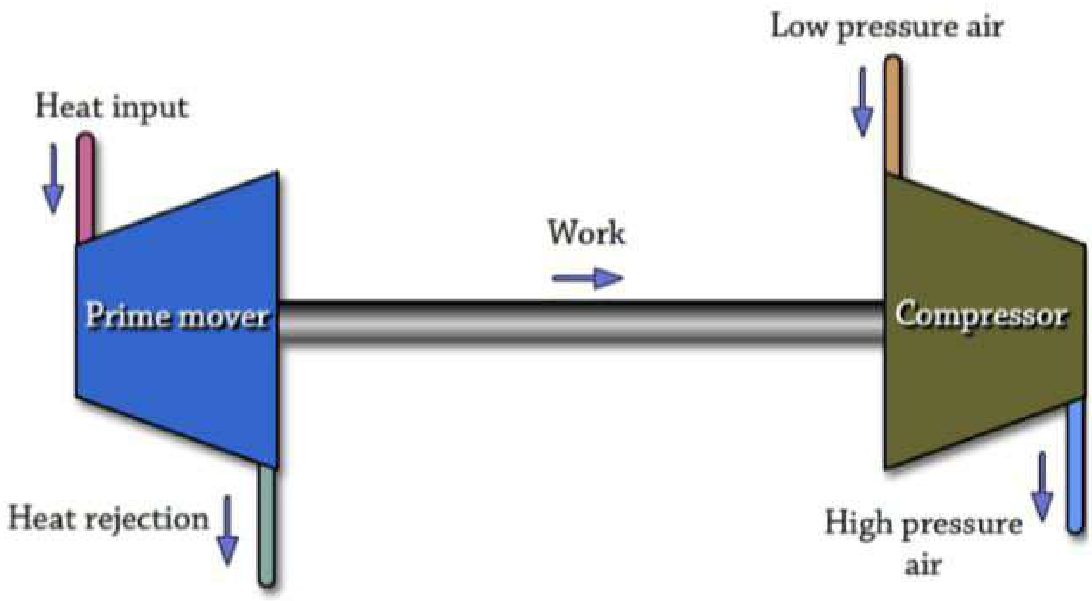

Fig 3: Air compressor

- The compression process requires work input (air).
- A compressor is driven by a prime mover (commonly electric motor).
- Air from the atmosphere enters into the compressor.
- Entered air is compressed to a high pressure air.
- Then it is delivered to a storage vessel (reservoir).
- From the reservoir it can be conveyed to the desired place through pipe lines.

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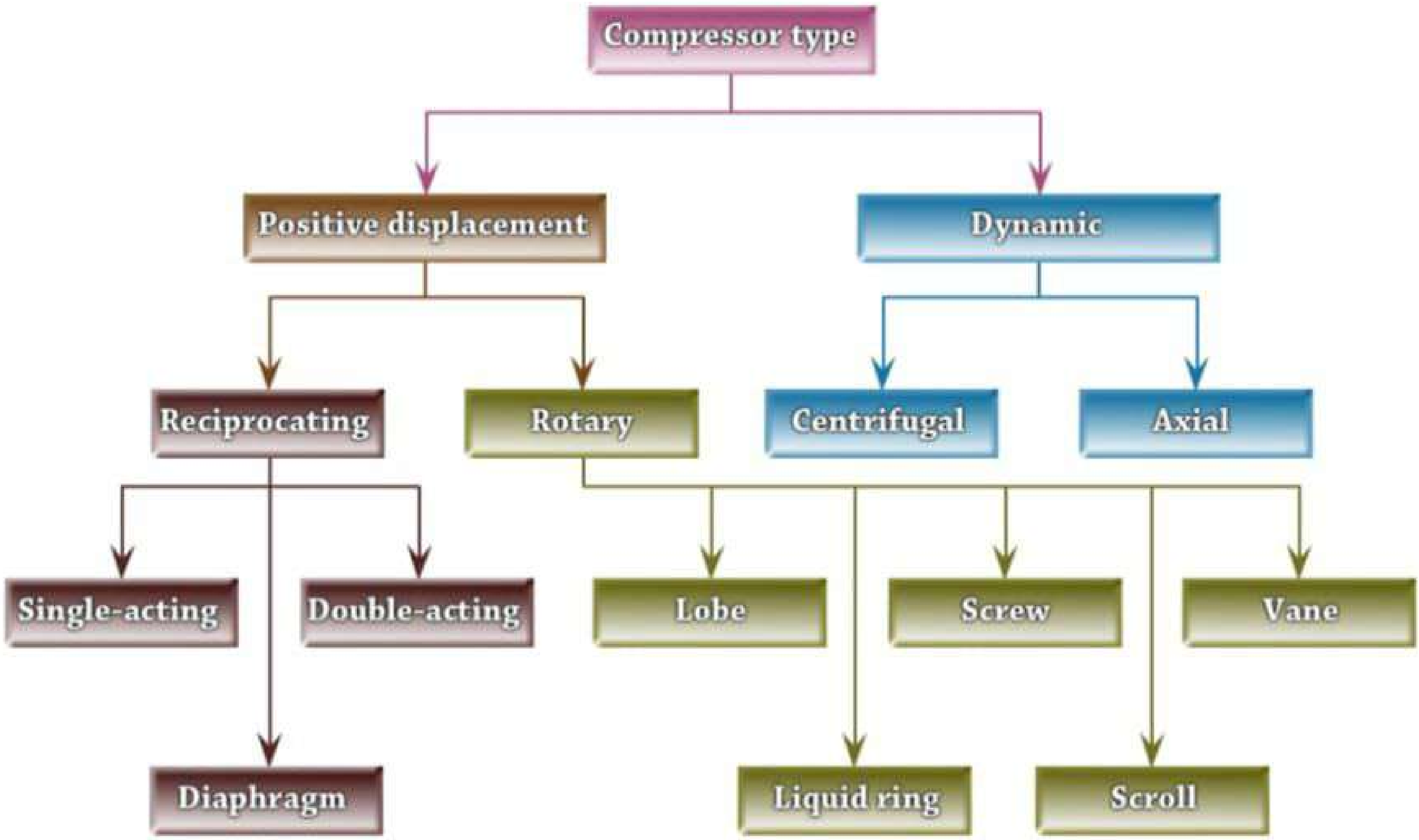
Classification of compressors



According to design and principle of operation

- Positive displacement compressors
  - Reciprocating compressors
  - Rotary compressors
    - Roots blower, vane blower
- Non positive displacement compressors
  - Rotary compressors of centrifugal and axial flow compressors

According to number of stages

- Single stage compressors
  - Delivery pressure up to 5 bar.
- Multistage compressors.
  - Delivery pressure above 5 bar.

According to pressure limit

- Low pressure compressors

- Delivery pressure is less than 10 bar.

- Medium pressure compressors
  - Delivery pressure 10 to 80 bar.
- High pressure compressors
  - Delivery pressure 80 to 100 bar.
- Super high pressure compressors
  - Delivery pressure above 100 bar.

According to pressure rise limit

- Fans: pressure ratio is 1 to 1.1.
- Blowers: pressure ratio is 1.1 to 2.5.
- Compressors: pressure ratio is above 2.5.

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Classification of reciprocating and rotary compressors

Classification of reciprocating compressors

- Single acting compressors
  - In which suction, compression and delivery of air takes place on one side of the piston.
- Double acting compressor
  - In which suction, compression and delivery of air takes place on both sides of the piston.

Classification of rotary compressors

- Steady flow compressors
  - In which there is a continuous steady flow of air.
    - Centrifugal and axial flow compressor
- Displacement compressors
  - In which the air is trapped in between two sets of engaging surfaces.
    - Root blower, vane blower

Important terms used in air compressor

- Single stage compressor
  - In single stage compressor, the compression of air from the initial pressure to the final pressure is carried out in one cylinder only.
- Multi stage compressor
  - When the compression of air from initial pressure to the final pressure is carried in more than one cylinder, then the compressor is known as multistage compressor.
- Single acting compressor
  - In single acting reciprocating compressor, the suction, compression and delivery of the air takes place on one side of the piston only. Such compressors would have one delivery strokes per revolution of the crank shaft.
- Double acting compressor

- In double acting compressor, the suction, compression and delivery of the air takes place on both side of the piston. Such compressors would have two delivery strokes per revolution of the crank shaft.

- Compression ratio
  - It is the ratio of absolute discharge pressure to the absolute inlet pressure.
- Free air delivered (F.A.D)
  - It is the actual volume of air delivered at the stated pressure, reduced to intake pressure and temperature on to STP conditions.
- Displacement of the compressor
  - The swept volume of the piston in the cylinder is known as displacement of the compressor.
- Actual capacity of the compressor
  - The actual free air delivered by the cylinder per cycle or per minute is known as actual capacity of the compressor.
- Volumetric efficiency
  - The ratio of actual free air delivered by the compressor per stroke to the displacement of the compressor is known as volumetric efficiency.
- Average piston speed
  - The average speed at which the piston reciprocates inside the cylinder per minute is called average piston speed.
- Swept volume
  - It is the volume of air sucked by the compressor during its suction stroke.
- Mean effective pressure
  - Air pressure on the compressor piston keeps on changing with the movement of the piston in the cylinder. The mean effective pressure of the compressor is found out mathematically by dividing the work done per cycle to the stroke volume.

Reciprocating air compressor

Construction

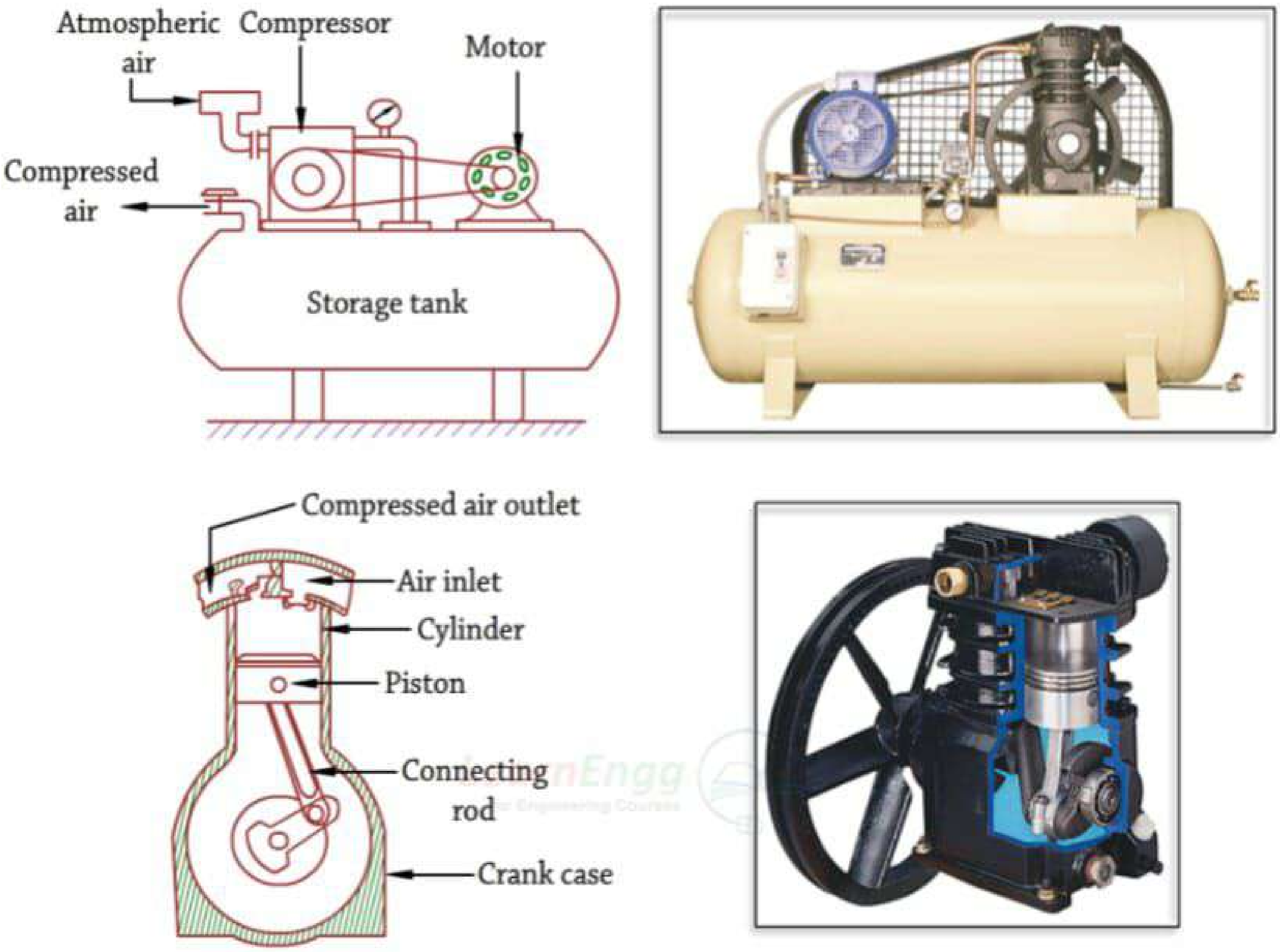
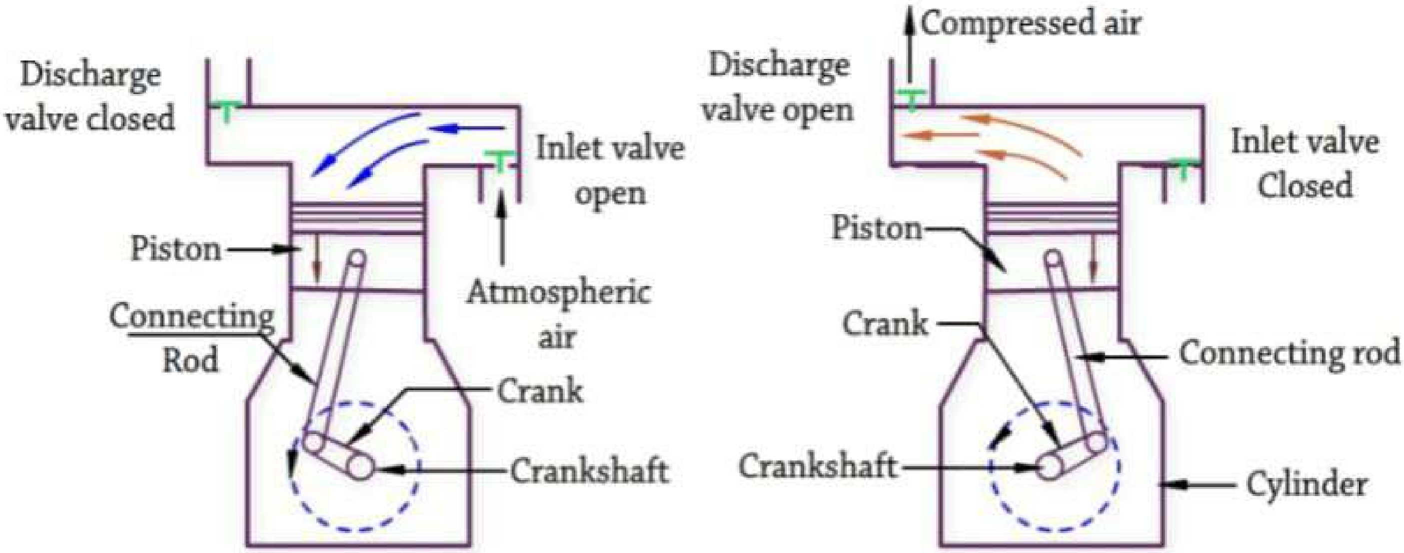


Fig 1: Reciprocating air compressor

- It consists of a cylinder in which piston reciprocates by means of an external source of work.
- The cylinder cover accommodates two valves, one is intake valve and the other discharge valve.
- Both valves are opened and closed automatically due to pressure difference.

Working



(a) Piston moving down, air flows into cylinder through inlet valve

(b) Piston moving up, air flows into receiver at a desired pressure through discharge valve

Fig 2

- When the piston moves downwards, the pressure in the cylinder is made less than the atmospheric pressure, the inlet valve opens and the air flows into the cylinder.
- During the return stroke (upward movement) of the piston, the pressure in the cylinder will rise, and the inlet valve closes when the pressure in the cylinder is just above that of the atmospheric pressure.
- The pressure of air steadily increases to a desired level (greater than the receiver pressure), the discharge valve opens and compressed air is discharged into a receiver from which the high pressure air is dispensed to required purpose.
- At the end of upward stroke, a small volume of air at high pressure is left in the clearance volume.
- As the piston moves down on the next stroke this air expands and pressure falls just below the atmosphere.
- Then the inlet valve opens and air flows into the cylinder, and the cycle is repeated.

Disadvantages

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

Applications

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

Work done and power required by a single stage compressor

- A reciprocating air compressor, the air is first sucked, compressed and then delivered.
- So there are three different operations of the compressor.
- Thus we that work is done on the piston during the suction of the air.
- Similarly, work is done by the piston during compression as well as delivery of the air.
- A little consideration will show, that the work done by a reciprocating air compressor is mathematically equal to the work done by the compressor during suction.
- The following two important cases of work done :
  - When there is no clearance volume in the cylinder, and
  - When there is some clearance volume.

Workdone in single stage single acting reciprocating air compressor neglecting clearance volume

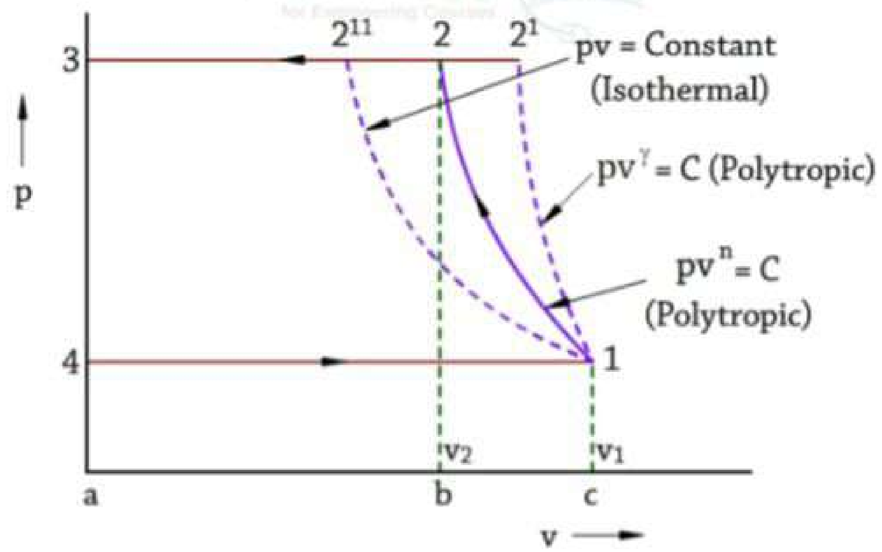


Fig 3: Theoretical p-v diagram for single acting compressor without clearance

Process: 4 - 1

- Represents the suction of air at pressure p_1. During this operation inlet valve remains open.

Process: 1 - 2

- Represents the compression of air polytropically. During this operation inlet and delivery valves remain closed.

Process: 2 - 3

- Represents the discharge of air to the receiver at pressure p_2. The delivery valve remains open during this period.
- We know that, Work done/cycle is equal to area of the pV diagram
- Area of the pv diagram 4 - 1 - 2 - 3 - 4 = Area of a - b - 2 - 3 - a + Area 1 - 2 - b - C - 1 - Area of a - c - 1 - 4 - a

$$= p_2 v_2 + \left[\frac{-(p_1 V_1 - p_2 V_2)}{n-1}\right] - p_1 V_1$$

Negative sign indicates work done on the system

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

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

$$= \frac{n}{n-1}(p_2 V_2 - p_1 V_1) \quad \text{...Eq.1}$$

$$= \frac{n}{n-1} mR(T_2 - T_1) \quad \text{...Eq.2}$$

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

We know that pressure and temperature relation in polytropic process.

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

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

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

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

Substituting the value of V_2 / V_1 in equation

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

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

$$= \frac{n}{n-1} \times p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \quad \text{...Eq.3}$$

$$W = \frac{n}{n-1} \times m R T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1 \right] \quad \text{...Eq.4}$$

Workdone/kg of air

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

Power required

- If N is number of cycles per minute (or) r.p.m of crank shaft.
- Then power required to drive the compressor,

$$P = W \times \frac{N}{60}$$

Note

- When the air is compressed adiabatically ($p v^{\gamma}$ = Constant)

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

Where,

γ - Adiabatic index

- When air is compressed isothermally ($pV = C$)

$$W/cycle = p_2 V_2 + p_1 V_1 \log_e\left(\frac{V_1}{V_2}\right) - p_1 V_1$$

$$\because p_1 V_1 = p_2 V_2$$

$$W/cycle = p_1 V_1 \log_e\left(\frac{V_1}{V_2}\right) \quad \text{or}$$

$$= p_1 V_1 \log_e\left(\frac{p_2}{p_1}\right)$$

- If the compressor is double acting then power required to drive the compressor,

$$P = W \times \frac{2N}{60}$$

-- PAGE END --

Example 1

- A single stage reciprocating air compressor is required to compress $60 m^3$ of air from 1 bar abs to 8 bar abs. Find the work to be supplied; If law of compression is $pV^{1.25}$ = Constant

Given data

Volume of air compressed, $V_1 = 60\ m^3$

Initial pressure, $p_1 = 1$ bar $= 100\ kN/m^2$

Final pressure, $p_2 = 8$ bar $= 800\ kN/m^2$

Index of compression , $n = 1.25$

To Find

Work to be supplied

Solution

Work to be supplied

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

$$= \frac{1.25}{1.25-1} \times 100 \times 60 \left[\left(\frac{8}{1} \right)^{\frac{1.25-1}{1.25}} - 1 \right]$$

$$W = 15471.49\ kJ$$

Result

- Work to be supplied, $W = 15471.49$ kJ

-- PAGE END --

Example 2

- Find the amount of work required to compress and discharge $1m^3$ of air at 15°C and 1bar to 7 bar abs. When the compression is isothermal. Take R= 0.29 kJ/kg K.

Given data

Initial pressure, p_1 = 1 bar = 100 kN/m^2

Initial temperature, T_1 = 15+273 = 288 K

Volume of air comp, V_1= $1m^3$

To Find

Work required, W

Solution

Mass of air,

$$m = \frac{p_1 V_1}{RT_1}$$

$$= \frac{100 \times 1}{0.29 \times 288}$$

$$= 1.197\,kg$$

Work required,

$$W = mRT_1 \log e\left[\frac{p_2}{p_1}\right]$$

$$= 1.197 \times 0.29 \times 288 \log_e\left[\frac{7}{1}\right]$$

$$W = 194.54\ kJ$$

Result

- Work required, W = 194.54 kJ

-- PAGE END --

Example 3

- A single stage air compressor has an effective swept volume of $5m^3$/min and delivers to a receiver at a pressure of 6.5 bar. The index of compression n=1.25, and the temperature at the end of suction stroke is 35°C and pressure is 1.03 bar. Calculate:

  (i) The mass of air compressed per min

  (ii) The temperature at the end of compression

  (iii) The power required to the compressor

  (iv) Take R = 0.287 kJ/kg K

Given data

Volume of air sucked, $V_1 = 5\ m^3/min$

Initial pressure, $p_1 = 1.03\ bar = 103\ kN/m^2$

Initial temperature $T_1 = 35 + 273 = 308\ K$

Find (delivery)pressure, $p_2 = 6.5\ bar$

To Find

Mass of air compressed per min

Temperature at the end of compression

Power required to the compressor

Solution

Mass of air, m

$$p_1V_1 = mRT_1$$

$$m = \frac{p_1V_1}{RT_1}$$

$$= \frac{103 \times 5}{0.287 \times 308}$$

$$m = 5.83 kg/min$$

Temperature at the end of compression, T_2

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

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} 308\times\left[\left(\frac{6.5}{1.03}\right)^{\frac{1.25-1}{1.25}}-1\right]$$

$$= 445.2\,\text{K}$$

$$T_2 = 172.2^\circ\,\text{C}$$

Power required to the compressor, P :

$$P = \frac{\oint W.N}{60}$$

When $\oint W.N$ = work done/min

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

Where V_1= Volume of air sucked/min

Work done/min

$$= \frac{1.25}{1.25-1}\times 103\times 5\left[\left(\frac{6.5}{1.03}\right)^{\frac{1.25-1}{1.25}}-1\right]$$

$$= 1147.14\ \text{kJ/min}$$

$$\therefore \text{Power, } P = 1147.14/60$$

$$P = 19.12\ \text{kW}$$

Result

- Mass of air, m = 5.83 kg/min
- Temperature at the end of compression, T_2 = 172.2°C
- Power required to the compressor, P = 19.12 kW

-- PAGE END --

Efficiencies of compressor

- The effective performance of a compressor is determined by its efficiencies. Defnitions of these efficiencies are presented below.

Compressor efficiency

- Isothermal power is used as a standard for comparing reciprocating compressor, and is often referred as theoretical power of a compressor.
- The power obtained from actual indicator card of a compressor is known as indicator power.
- The compressor efficiency is defined as the ratio of isothermal power (i.e power required by the compressor if the process of compression follows the law pV = Constant) to the indicated power of the compressor.

$$\text{Compressor efficiency} = \frac{\text{Isothermal power}}{\text{Indicated power}}$$

$$\eta_{comp} = \frac{P_{iso}}{I.P}$$

Isothermal efficiency

- Isothermal efficiency of a compressor is defined as the ratio of isothermal power to the shaft power (brake power).

$$\text{Isothermal efficiency} = \frac{\text{Isothermal power}}{\text{Shaft power}}$$

$$\eta_{iso} = \frac{P_{iso}}{B.P}$$

- Shaft power or brake power is the power supplied to compressor from the prime mover and is obtained by conducting brake test.

Adiabatic efficiency

- It is defined as the ratio of adiabatic power (I.P of compressor when the compression follows the law, pV^{γ} = Constant) to the brake power.

$$\eta_{adia} = \frac{P_{adia}}{B.P}$$

Mechanical efficiency

- It is the ratio of indicated power of the compressor to the brake power.

$$\eta_{mech} = \frac{I.P}{B.P}$$

Overall efficiency

- The ratio of the indicated power of the compressor to the indicated power of the prime mover (source of power to drive the compressor) is known as overall efficiency.

$$\text{Overall efficiency} = \frac{\text{I.P of compressor}}{\text{IP of primemover}}$$

Volumetric efficiency

- Volumetric efficiency is defined as the ratio of the actual volume of air drawn per stroke at STP to the piston displacement.

$$\text{Volumetric efficiency} = \frac{\text{Actual volume of air drawn per stroke at STP}}{\text{Stroke volume}}$$

- The actual volume of air drawn into the compressor, measured at STP conditions is called free air.
- The volume of free air delivered will be less than the swept or stroke volume of the piston.
- The volume of free air compressed and delivered per minute is known as capacity of compressor.
- Thus the volumetric efficiency may be expressed as

$$\text{Volumetric efficiency} = \frac{\text{Compressor capacity}}{\text{Piston displacement}}$$

-- PAGE END --

Example 4

- A double acting reciprocating compressor running at 250 RPM has a bore 250 mm and a stroke 360 mm. The air is drawn into a cylinder at a pressure of 1 bar and temperature 35°C. The volumetric efficiency of compressor was 75%. Determine the power required if the delivery pressure is 5 bar and the compression follows the law $pV^{1.3}$ = const.

Given data

Speed, N = 250 RPM

Bore, d = 0.25 m

Stroke length, l = 0.36 m

Suction pressure, p_1 = 1 bar

Delivery pressure, p_2 = 5 bar

Volumetric efficiency, ηv = 75%

To Find

Power required, P

Solution

Swept volume/cycle

$$= \frac{\pi(d)^2}{4} \times 1$$

$$= \frac{\pi(0.25)^2}{4} \times 0.36$$

$$= 0.01766 m^3/cycle$$

$$\text{Volumetric efficiency} = \frac{\text{Volume of air admitted at STP}}{\text{Stroke(or swept) volume}}$$

∴Volume of air admitted at STP, V_c

$$= \eta v \times Vs$$

$$= 0.75 \times 0.01766$$

$$V_0 = 0.01325 \ m^3$$

Let V_1 be the volume of air admitted at suction conditions i.e., at p_1 = 100 kN/m^2 and T_1 = 308 K.

Using the equation,

$$\frac{p_1 V_1}{T_1} = \frac{p_o V_o}{T_o}$$

At STP,

$$P_o = 101.3\ kN/m^2, \text{ and}$$

$$T_o = (15 + 273)$$

$$T_o = 288\ K$$

$$V_1 = \frac{101.3 \times 0.01325}{100} \times \frac{308}{288}$$

$$V_1 = 0.0143\,m^3/\text{cycle}$$

Workdone per cycle,

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

$$= \frac{1.3}{1.3-1} \times 100 \times 0.0143 \left[\left(\frac{5}{1} \right)^{\frac{1.3-1}{1.3}} - 1 \right]$$

$$\oint W = 2.787\,kJ/\text{cycle}$$

Power required,

$$P = \frac{\oint W.2N}{60}$$

$$= \frac{2.787 \times 2 \times 250}{60}$$

$$P = 23.22\,kW$$

Result

- Power required, P = 23.22 kW

-- PAGE END --

Single stage air compressor with clearance

- In practice, all reciprocating compressors must have a clearance volume at the end of the delivery stroke.
- At the end of the delivery stroke, some high pressure air remains in the clearance, and this will expand down to suction pressure before a fresh air can be induced.

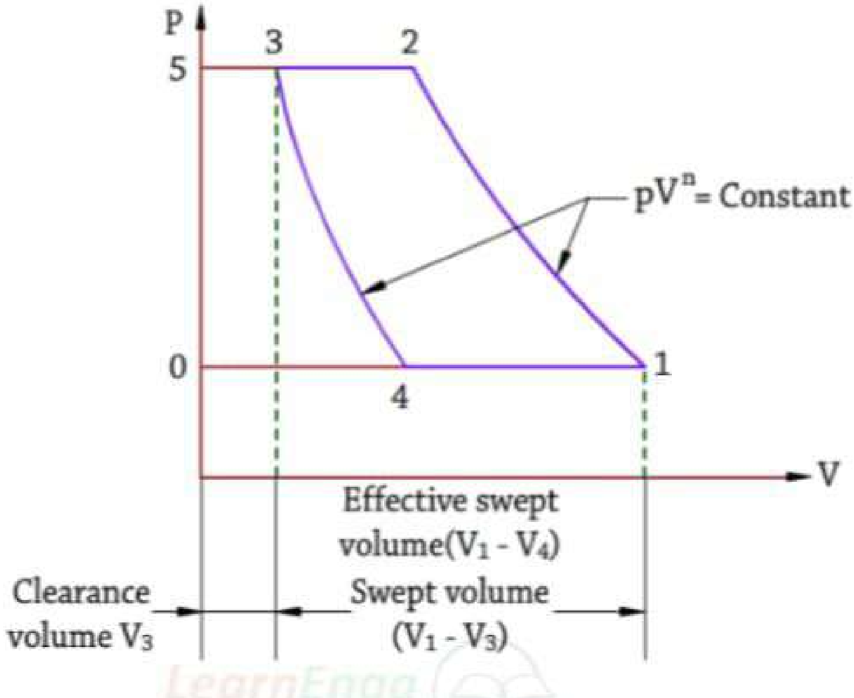


Fig 1

- p-V diagram for a single acting, single-stage reciprocating air compressor, having clearance volume, V_3 and the piston displacement $(V_1 - V_3)$.
- The clearance ratio (k) may be expressed as a ratio of the clearance volume to the piston displacement.
- Thus the clearance ratio is given as.

$$\text{Clearance ratio, } k = \left[\frac{V_3}{V_1 - V_3}\right]$$

Or percentage of clearance volume

$$= \left[\frac{V_3}{V_1 - V_3}\right] \times 100$$

$$= (k \times 100)$$

Volumetric efficiency

$$= \frac{V_1 - V_4}{V_1 - V_3}$$

Now

$$\left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} = \frac{V_4}{V_3} \qquad (\because\ p_3 = p_2 \text{ and } p_4 = p_1)$$

$$= \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \qquad \because V_4 = V_3\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

Volumetric efficiency,

$$= \frac{V_1 - V_3\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}}{V_1 - V_3}$$

$$\eta_v = \frac{V_1 - V_3\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}}{V_3 \cdot \left[\frac{V_1 - V_3}{V_3}\right]}$$

$$= \frac{V_1 - V_3\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}}{\frac{V_3}{k}} \qquad \text{or}$$

$$\eta_v = \frac{kV_1}{V_3} - k\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

Again

$$V_1 = V_3\left(1 + \frac{1}{k}\right) \qquad \left(\because k = \frac{V_3}{V_1 - V_3}\right)$$

$$\therefore \quad \eta_v = \frac{k.V_3\left(1 + \frac{1}{k}\right)}{V_3} - k\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

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

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

- From this expression it is clear that if $p_2 = p_1$, the volumetric efficiency is 100%.
- In practice, the air that is sucked in during suction stroke differ from STP conditions. Considering p_o and T_o are pressure and temperature of air at STP. Using the relation,

$$\frac{p_0 V_0}{T_0} = \frac{p_1 V'_1}{T_1}$$

$$V_o = \left[\frac{p_1 . T_o}{p_o T_1}\right] . V'_1$$

Where

$$V'_1 = \text{effective stroke volume}$$

$$= (V_1 - V_4)$$

$$\eta_v = \frac{V_o}{V_s}$$

$$= \left[\frac{p_1 . T_o}{p_o T_1}\right] \left[\frac{V_1 - V_4}{V_1 - V_3}\right]$$

$$\eta_v = \left[\frac{p_1 . T_o}{p_o T_1}\right] \left[1 + k - k\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]$$

Expression for work done:

- The net WD per cycle = area 01250 – 04350

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

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

$$\therefore \quad \oint W = \frac{n}{n-1} p_1 \text{ (effective stroke vol.)} \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Then

$$IP = \frac{\oint WN}{60} \text{ for single acting compressor}$$

$$= \frac{\oint W 2N}{60} \text{ for double acting compressor}$$

Where,

N = RPM of compressor

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Factors influencing the volumetric efficiency

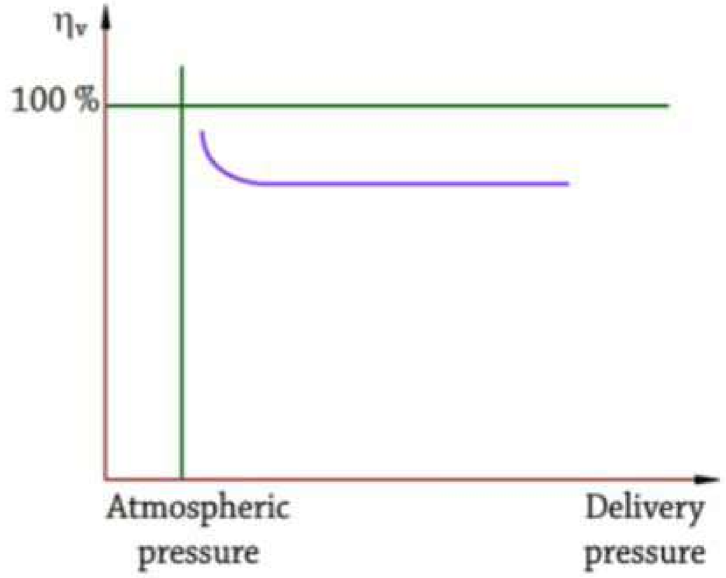

Fig 2: Effect of delivery pressure on volumetric efficiency

- Figure shows the effect of delivery pressure on volumetric efficiency.
- An increase in delivery pressure reduces the effective swept volume, there will be a corresponding decrease in volumetric efficiency.
- The volumetric efficiency decreases rapidly with increase in delivery pressure at first, and then more slowly for later high pressure.
- Leakage of air past the piston will also decrease the amount of air delivered, and as the delivery pressure is increased, more air will leak past the piston during the compression process.
- Frictional effects in the air itself and turbulence in air passages give rise to a reduction in the suction pressure in the cylinder and subsequently volumetric efficiency decreases.
- The air drawn into cylinder at a lower temperature than the expanded air left in the clearance will decrease the volumetric efficiency.
- The incoming air is heated, and thus expands and allows less quantity of air into cylinder.
- As a result volumetric efficiency is decreased.

-- PAGE END --

Example 1

- An air compressor with a clearance 5% of the stroke, draws in air 0.965 bar, compresses it according to law $pV^{1.25}$ = constant and delivers at 4.14 bar. Find the volumetric efficiency, if the temperature of air within the cylinder during the suction stroke is 30°C.

Given data

STP conditions,

$p_o = 101.3\ kN/m^2$

$T_o = (15+273) = 288K$

Suction conditions,

$p_1 = 0.965\ bar = 96.5\ kN/m^2$

$T_1 = (30 + 273) = 303\ K$

$n = 1.25$

$p_2 = 4.14\ bar$

Clearance ratio, $k = \left(\frac{5}{100}\right)$

$k = 0.05$

To Find

Volumetric efficiency

Solution

Volumetric efficiency, η_v

$$= \left[1+k-k\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right]\left[\frac{p_1 . T_o}{p_o T_1}\right]$$

$$= \left[1+0.05-0.05\left(\frac{4.14}{0.965}\right)^{\frac{1}{1.25}}\right]\left[\frac{96.5\times 288}{101.3\times 303}\right]$$

$$= 0.8897 \times 0.9054$$

$$\eta_v = 80.5\%$$

Example 2

- A single acting single stage compressor has a cylinder of 200 mm diameter and 300 mm stroke. It runs at a speed of 500RPM. The air is taken in at standard atmospheric pressure and temperature. The compression pressure is 6 bar abs. The clearance volume is 5% of the stroke volume. The index of compression and expansion is 1.3. Determine

  (i) The volumetric efficiency, and

  (ii) The brake power required to drive the compressor, if the mechanical efficiency is 80 percent.

Given data

Diameter, d = 200mm

Stroke length, l = 300mm

Compression pressure, p_2 = 6bar

Clearance volume = 5%

n = 1.3

To Find

Volumetric efficiency

Brake power

Solution

Intake conditions,

Pressure,

$$p_o = 101.3 \text{ kN/m}^2$$

$$= 1.013 \text{ bar}$$

Temperature

$$= (15 + 273)$$

$$= 288 \text{ K}$$

Stroke volume,

$$V_s = (V_1 - V_3)$$

$$= \frac{\pi d^2}{4} \times l$$

$$= \frac{\pi (0.2)^2}{4} \times 0.3$$

$$V_s = 0.00942 m^3$$

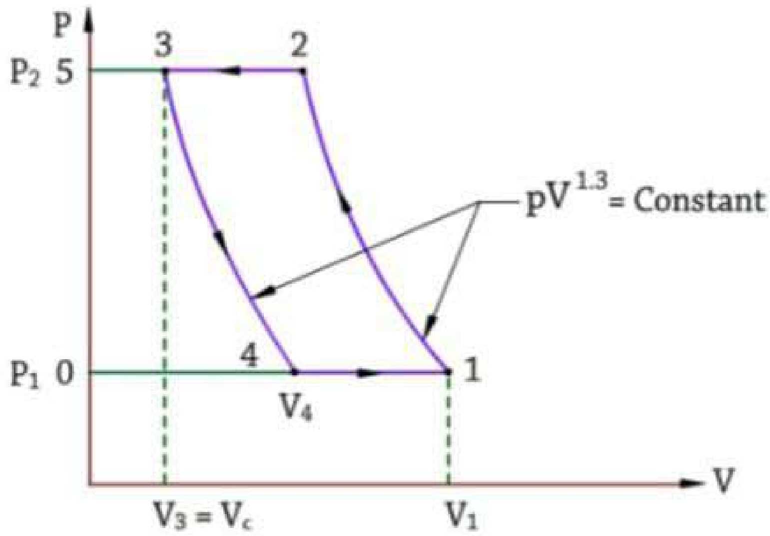

Fig 3

Clearance volume,

$$V_3 = 0.05 \, (V_1 - V_3)$$

$$= 0.05 \times 0.00942$$

$$V_3 = 0.000471 m^3$$

Cylinder volume,

$$V_1 = V_c + V_s$$

$$V_1 = 0.00989 \; m^3$$

Considering polytropic expansion, (3- 4)

$$V_4 = V_3 \cdot \left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} \qquad \left(\because \frac{p_3}{p_4} = \frac{p_2}{p_1}\right)$$

$$= 0.000471 \cdot \left(\frac{6}{1.013}\right)^{\frac{1}{1.3}}$$

$$V_4 = 0.00185 m^3$$

Volumetric efficiency,

$$= \frac{V_1 - V_4}{V_1 - V_3}$$

$$= \frac{0.00989 - 0.00185}{0.00989 - 0.000471}$$

$$= \frac{0.00804}{0.00942}$$

$$= 0.853$$

Volumetric efficiency,

$$= 85.3\%$$

Work required / cycle

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

$$= \frac{1.3}{1.3-1} \times 101.3(0.00804)\left[\left(\frac{6}{1.013}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= (3.529 \times 0.5076)$$

$$= 1.79 \text{kJ/cycle}$$

$$\therefore IP = \frac{\int W \cdot N}{60}$$

$$= \frac{1.79 \times 500}{60}$$

$$I.P = 14.93 \text{kW}$$

Mechanical efficiency,

$$= 80\%$$

$$\eta_m = \frac{IP}{BP}$$

$$BP = \frac{14.93}{0.8}$$

$= 18.66 kW$

Result

Volumetric efficiency = 85.3%

Brake power = 18.66 kW

-- PAGE END --

Multi-stage air compressor

Fig 1: Multi-stage air compressor

- The compression of air in single-stage has many disadvantages and its use is limited where low delivery pressure is required.
- Multi-stage compression is more efficient and mostly employed for rising the high pressure.

Construction

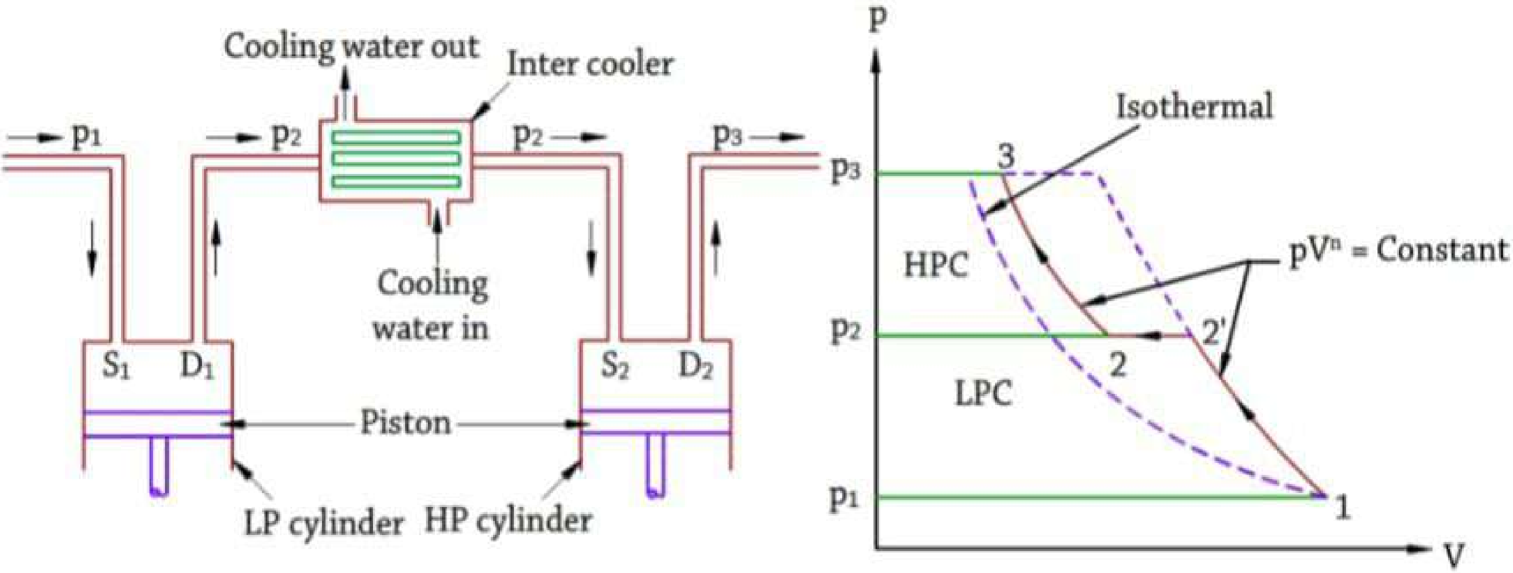

Fig 2: Multi stage air compressor

- In a multi stage air compressor, compression of air takes place in more than one cylinder.
- It consist of a

- Low pressure cylinder (LP)
- High pressure cylinder (HP)
- An intercooler

- Both the pistons (in LP & HP cylinders) are driven by a prime mover through a common shaft.

Working

- Atmospheric air at (p_1) is taken into the low pressure cylinder. It is compressed to a high pressure (p_2).
- This pressure is intermediate between intake pressure (p_1) and delivery pressure (p_3). Hence this is known as intermediate pressure.
- The air from low pressure cylinder is then passed into an intercooler.
- In the intercooler, the air is cooled at constant pressure by circulating cold water.
- The cooled air from the intercooler is then taken into the high pressure cylinder.
- In the high pressure cylinder, air is further compressed to the final delivery pressure (p_3) and supplied to the storage tank.

Advantages of multistage compressor over single stage compressor

- The work done per kg of air can be reduced by introducing an intercooler between the two stages for the same delivery pressure.
- Better mechanical balance can be achieved.
- No leakage and better lubrication.
- More volumetric efficiency.
- High delivery pressure.
- Simple construction of LP cylinder.

Disadvantages

- More than one cylinder is required.
- An intercooler is required. This increases initial cost. Also space required is more.
- Continuous flow of cooling water is required.
- Complicated in construction.

Intercooler

- An intercooler is a simple heat exchanger.
- It exchanges the heat of compressed air from the LP compressor to the circulating water before the air enters the HP compressor.
- It consists of a number of special metal tubes connected to corrosion resistant plates at both ends.
- The entire nest of tubes is covered by an outer shell.

Working

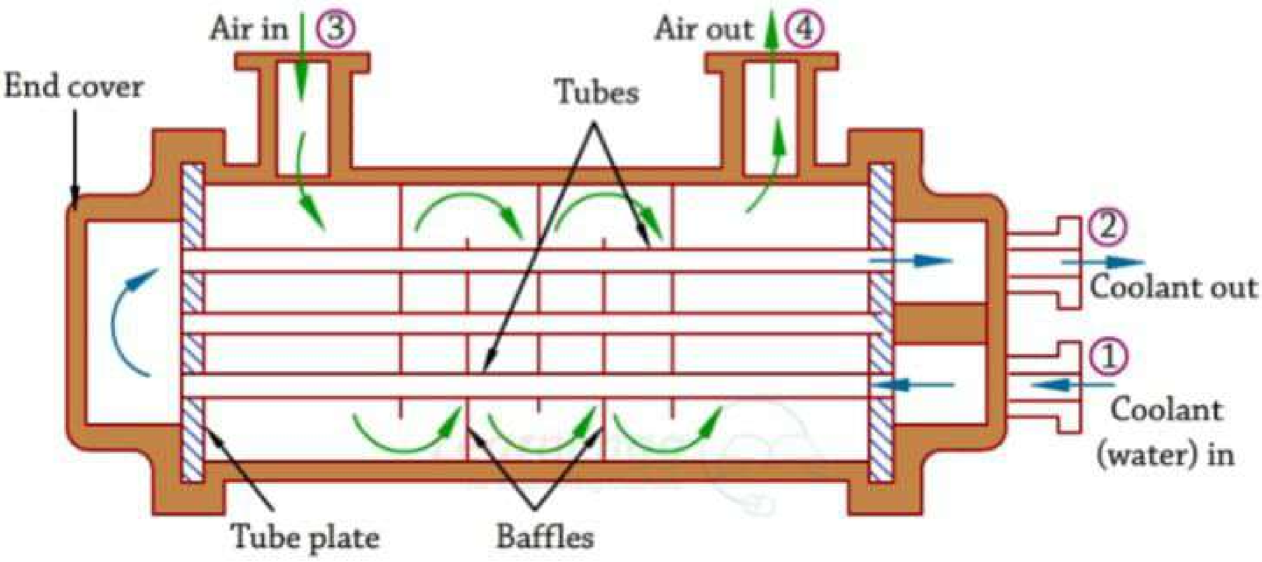

Fig 3: Inter cooler

- Cold water enters the bottom of the intercooler through water inlet (1) and flows into the bottom tubes.
- Then they pass through the top tubes and leaves through the water outlet (2) at the top.
- Air from LP compressor enters through the air inlet (3) of the intercooler and passes over the tubes.
- While passing over the tubes, the air is cooled (by the cold water circulated through the tubes).
- This cold air leaves the intercooler through the air outlet (4).
- Baffle plates are provided in the intercooler to change the direction of air.
- This provides a better heat transfer from air to circulating water.

Uses of intercooler

- Better lubrication due to lower temperature.
- Saving of work input is up to 20% can be achieved.
- It reduces the cost of compression.
- The low pressure cylinder may be lighter.
- It reduces the leakage loss.

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Work required for multi-stage air compressor

- The work required for multi-stage compression is less, for the same inlet conditions and the same delivery (exit) pressure than the single stage compression.
- The general arrangement of the cylinders in 2-stage air compressor is shown in figure 4.

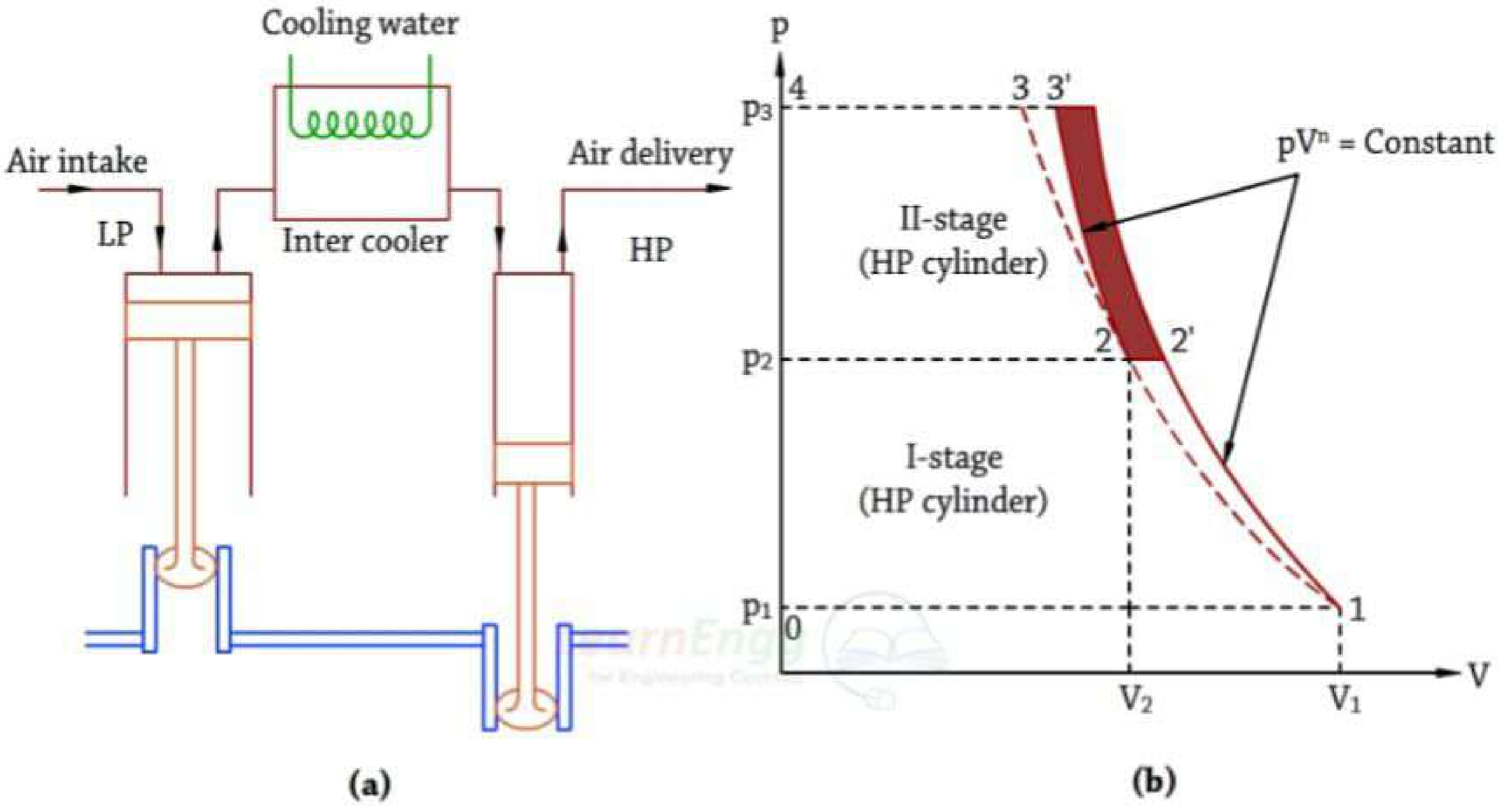

Fig 4: Multi-stage air compressor

- In L.P cylinder, the air is compressed to intermediate pressure, p_2 according to the law pV^n = constant.
- Work required per cycle in L-P cylinder.

$$W_1 = \frac{n}{n-1} \cdot p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

- Similarly, the air is compressed in 2nd stage to increase pressure from p_2 to delivery pressure p_3.
- Work required per cylinder in H-P cylinder,

$$W_2 = \frac{n}{n-1} \cdot p_2 V_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{n-1}{n}} - 1 \right]$$

- Total work required,

$$W = \frac{n}{n-1}\left[p_1V_1\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} + p_2V_2\left\{\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right\}\right] \quad \text{...Eq.1}$$

- If the intercooling is perfect, the point 2' will be on the isothermal line i.e., at point 2.

Then

$$p_1V_1 = p_2V_2$$

  - Substituting this in equation 1

$$W = \frac{n}{n-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right]$$

- If the compression is adiabatic, then the total work required.

$$W = \frac{\gamma}{\gamma-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} + \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} - 2\right]$$

$$\text{Power required} = \frac{W \cdot N}{60}$$

Where

N = Speed of compressor in RPM

Heat rejected in intercooler

- Referring to the pV diagram let T_1, T_2' and T_2 be the absolute temperatures of the air at points 1, 2' and 2.

- Let,

  - m = mass of air compressed

  - C_p = specific heat of air at constant pressure

Then

Heat rejected in intercooler= $m.C_p\,(T_2' - T_2)$

- If inter cooling is perfect, $T_2 = T_1$

Conditions for maximum efficiency

- The effect of various intermediate pressure on the work requirement is shown in figure 5.

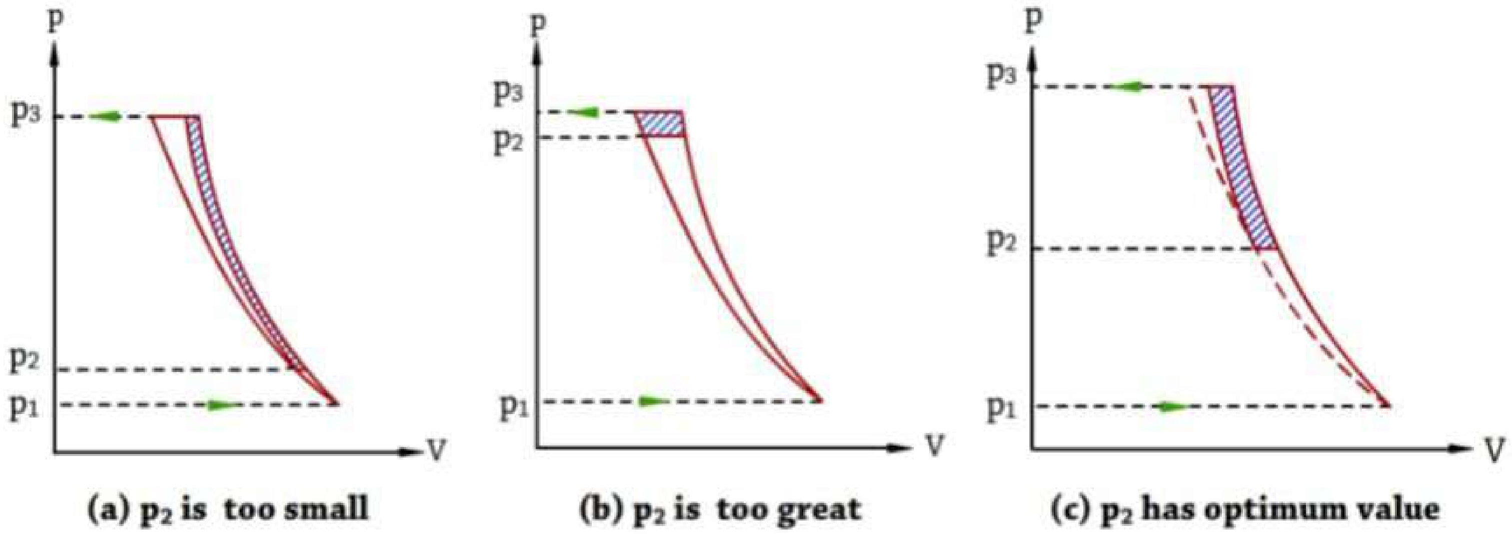

Fig 5: Condition for maximum efficiency

- In figure 5(a), the intermediate pressure is quite low relative to p_3.
- In figure 5(b), the intermediate pressure is high.
- In both cases the amount of work saved is small as compared to that shown in figure 5(c). It shows that there exists an optimum value of p_2 for which the work required will be a minimum i.e., work saved will be a maximum.
- To determine the optimum value for intermediate pressure, the following assumptions are made,
  - The indices of compression in each cylinder are equal .i.e., compression in each cylinder follows the law pV^n = constant.
  - Intercooling is complete and perfect i.e., the air is cooled back to initial temperature T_1 in the intercooler.
  - There is no pressure loss in intercooler.
  - Suction and delivery of air take place at constant pressure.
  - The effects of clearance are neglected.
- The work required per cycle for 2-stage air compressor with perfect intercooling is given by the equation.

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

- For fixed intake and delivery pressures, and for a constant value of n, the variable for a given compressor is p_2.
- The optimum value of p_2 can be obtained by differentiating this equation in terms of p_2 and equating to zero.
- Nothing that n, p_1 and V_1 are constant. Thus $\frac{n}{n-1} p_1 V_1$ is constant.
  - Let,

$$\frac{n}{n-1} = a$$

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

$$W = a\,\text{constant} \times \left[p_2^{\,a} \cdot p_1^{\,-a} + p_3^{\,a} \cdot p_2^{\,-a} - 2 \right]$$

  - Differentiating the above equation,

$$\frac{dW}{dp_2} = a \cdot p_2^{\,a-1} \cdot p_1^{\,-a} - a p_2^{\,-a-1} \cdot p_3^{\,a}$$

  - For minimum work

$$\frac{dW}{dp_2} = 0$$

$$a \cdot p_2^{\,a-1} \cdot p_1^{\,-a} - a p_2^{\,-a-1} \cdot p_3^{\,a} = 0$$

$$a \cdot p_2^{\,a-1} \cdot p_1^{\,-a} = a p_2^{\,-a-1} \cdot p_3^{\,a}$$

  - Dividing throughout by a, and re-arranging the terms

$$\frac{p_2^{\,a}}{p_1^{\,a} \cdot p_2} = \frac{p_3^{\,a}}{p_2^{\,a} \cdot p_2}$$

  - Multiplying throughout by p_2, and taking the 'a' root throughout.

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} \qquad \text{...Eq.2}$$

- Hence, for maximum efficiency the pressure ratio in each cylinder is the same.
- From equation 2, the optimum value of p_2 is given by the equation,

$$p_2 = \sqrt{p_1 \cdot p_3}$$

- Hence for maximum efficiency, the intermediate pressure is the geometric mean of the suction and delivery pressure.
- From equations 1 and 2 it is clear that the workdone in each stage (i.e., in cylinder) is the same.
- Workdone per cycle,

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

$$W = \frac{2 \cdot n}{n-1} mRT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

- The above conditions are also applicable for multi-stage compressor.

- For multi-stage compressor, the workdone per cycle is given by the equation,

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

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

Where

m = No. of stages

Alternate expression

- For maximum efficiency,

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

$$p_2 = \sqrt{p_1 \cdot p_3}$$

Therefore

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

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

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

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

$$W = \frac{2n}{n-1} \cdot mRT_1 \left[\left(\frac{p_3}{p_1} \right)^{\frac{n-1}{2n}} - 1 \right]$$

- For three stage,

$$W = \frac{3n}{n-1} \cdot p_1 V_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$$

$$W = \frac{3n}{n-1} \cdot mRT_1 \left[\left(\frac{p_4}{p_1} \right)^{\frac{n-1}{3n}} - 1 \right]$$

- For m-stages,

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \frac{p_m + 1}{p_m}$$

- Workdone per cycle

$$W = \frac{mn}{n-1} \cdot p_1 V_1 \left[\left(\frac{p_m + 1}{p_m} \right)^{\frac{n-1}{mn}} - 1 \right]$$

$$W = \frac{mn}{n-1} \cdot mRT_1 \left[\left(\frac{p_m + 1}{p_m} \right)^{\frac{n-1}{mn}} - 1 \right]$$

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Condition for minimum work in two stage compressor

- The intercooling is perfect.
- The compressed air is cooled back to the initial temperature T_1 in the intercooler at constant pressure.
- The intermediate pressure,

$$p_2 = \sqrt{p_1 \cdot p_3}$$

-- PAGE END --

Example 1

- Find the minimum energy required to compress one kg of air from 15°C and 1 bar to 40 bar in 2-stage compressor. The law of compression is $pV^{1.25}$ = constant, and intercooling is perfect.

Given data

Initial pressure, p_1 = 1 bar

Initial temperature, T_1 = 15+273 = 288K

Mass of air, m = 1kg

Final pressure, p_3 = 40 bar

Law of compression, $pV^{1.25}$ = Constant

For minimum work (energy)

$$p_2 = \sqrt{p_1 \times p_3}$$

$$= \sqrt{1 \times 40}$$

$$p_2 = 6.324 \text{ bar}$$

To Find

Energy required

Solution

Energy required per cycle,

$$W = \frac{2 \cdot n}{n-1} mRT_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

Assume, R=0.287 kJ/kgK for air

$$W = \frac{2 \times 1.25}{1.25 - 1} \times 1 \times 0.287 \times 288 \left[\left(\frac{6.324}{1} \right)^{\frac{1.25-1}{1.25}} - 1 \right]$$

$$= 826.56 \times 0.4461$$

$$W = 368.728 \text{ kJ}$$

Note

Energy required per cycle can also be obtained by using the equation.

$$W = \frac{2 \cdot n}{n-1} mRT_1 \left[\left(\frac{p_3}{p_1} \right)^{\frac{n-1}{2n}} - 1 \right]$$

$$W = \frac{2 \times 1.25}{1.25 - 1} \times 1 \times 0.287 \times 288 \left[\left(\frac{40}{1} \right)^{\frac{1.25-1}{2 \times 1.25}} - 1 \right]$$

$$= 826.56 \times 0.4461$$

$$W = 368.728 \, kJ$$

Result

- Energy required per cycle = 368.73 kJ

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Example 2

- In a two-stage air compressor the pressures are atmospheric 1 bar, inter cooler 7.5 bar; delivery 42.5 bar. Assuming complete intercooling to the original temperature of 288 K and the compression follows the $pV^{1.3}$ = constant. Find the workdone in compressing 1 kg of air; Assume R = 0.287 kJ/kgK

Given data

Initial pressure, $p_1 = 1$ bar

Inter cooler pressure, $p_2 = 7.5$bar

Final pressure, $p_3 = 42.5$ bar

Initial temperature, $T_1 = 288$K

Mass of air, m = 1kg

Law of compression, $pV^{1.3}$ = Constant

R = 0.287 kJ/kgK

To Find

Work done, W

Solution

Work done for 2-stage compression with complete intercooling.

$$W = \frac{n}{n-1} mRT_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2 \right]$$

$$= \frac{1.3}{1.3-1} \times 1 \times 0.287 \times 288 \left[\left(\frac{7.5}{1}\right)^{\frac{1.3-1}{1.3}} + \left(\frac{42.5}{7.5}\right)^{\frac{1.3-1}{1.3}} - 2 \right]$$

$$= 358.176\,(1.592 + 1.492 - 2)$$

$$W = 388.26\,\text{kJ}$$

Result

- Work done = 388.26 kJ

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Example 3

- A 3-stage air compressor is used to compress air from 101.3 kN/m² to 3600 kN/m². The compression follows the law $pV^{1.25} = C$. The temperature of air at inlet of compressor is 300 K. Neglecting clearance and assuming complete intercooling, find the indicated power required to deliver 15 m³ of air per minute measured at inlet conditions. Take R = 0.287 kJ/kgK.

Given data

Initial pressure, $p_1 = 101.3\ kN/m^2$

Delivery pressure, $p_4 = 3600\ kN/m^2$

$n = 1.25$

Initial temperature, $T_1 = 300\ K$

Volume of air delivered, $V_1 = 15\ m^3/min$

$R = 0.287\ kJ/kgK$

To Find

Indicated power

Solution

Work d

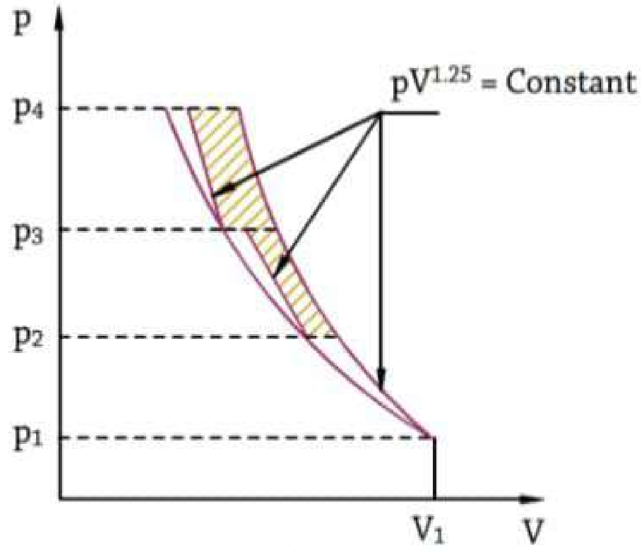

Fig 1

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

Where,

V_1 = Volume of air admitted per minute

$$W/\min = \frac{3\times1.25}{1.25-1}\times101.3\times15\left[\left(\frac{3600}{101.3}\right)^{\frac{1.25-1}{3\times1.25}}-1\right]$$

$$=22792.5\times0.2687$$

$$=6124.35\,\text{kJ}/\min$$

$$\therefore \text{Indicated power} = \frac{6124.35}{60}$$

$$=102.07\text{kW}$$

Alternate solution

When intercooling is perfect, with optimum intermediate pressures, the workdone in each cylinder is same.

$\therefore$ For 3-stages, workdone per minute

$$W = \frac{3\cdot n}{n-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right]$$

Where,

p_2 = Intermediate pressure between 1st and 2nd stage.

For optimum conditions,

$$\frac{p_2}{p_1}=\frac{p_3}{p_2}=\frac{p_4}{p_3}=Z=\left[\frac{p_4}{p_1}\right]^{\frac{1}{3}}$$

$$\text{In general, for 'm' stages}, Z=\left(\frac{p_m+1}{p_1}\right)^{\frac{1}{m}}$$

For three-stage compressor, the intermediate pressure can be obtained from the equations

$$p_2=\sqrt[3]{(p_1)^2\cdot p_4}$$

$$p_3=\sqrt[3]{p_1(p_4)^2}$$

Where,

p_1 = Intake pressure

p_4 = Delivery pressure

$$p_2 = \sqrt[3]{(101.3)^2 \times 3600}$$

$$= 333.05\,\text{kN/m}^2$$

$$p_3 = \sqrt[3]{101.3 \times (3600)^2}$$

$$= 1095\,\text{kN/m}^2$$

Also, workdone per min

$$W/\text{min} = \frac{3 \times 1.25}{1.25 - 1} \times 101.3 \times 15\left[\left(\frac{333.05}{101.3}\right)^{\frac{1.25-1}{1.25}} - 1\right]$$

$$= 22792.5 \times 0.2687$$

$$= 6124.35\,\text{kJ/min}$$

$$\therefore \text{Indicated power} = \frac{6124.35}{60}$$

$$= 102.07\,\text{kW}$$

Result

- Indicated power = 102.07 kW

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Rotary compressors

Fig 1: Rotary compressors

- Reciprocating compressors are not suitable for high speed operation.
- Because of large inertia forces, the reciprocating compressors need to operate at slow speed which limits the capacity of the machine.
- Rotary compressors do not possess any reciprocating parts.
- In rotary compressors, the air is trapped between two surfaces which are in rotation and compressed to moderate pressure.

Salient features of rotary compressors

- They supply large quantity of air.
- The delivery pressure is relatively low.
- They run at much higher speeds.
- They can be directly coupled to the prime mover (steam or gas turbine).

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Types of rotary compressors

- Rotary compressors are broadly classified into two types:
  - Positive displacement compressor
    - Roots blower
    - Vane blower
    - Lysholm compressor
    - Screw compressor
  - Non-positive displacement (steady flow) compressor
    - Radial centrifugal compressor
    - Axial flow compressor

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Positive displacement compressor

- In positive displacement compressor, the air is trapped and compressed in the reduced space between engaging surfaces.
- The change in pressure is either by the back flow of air or both by squeezing action and back flow action.

Roots blower air compressor

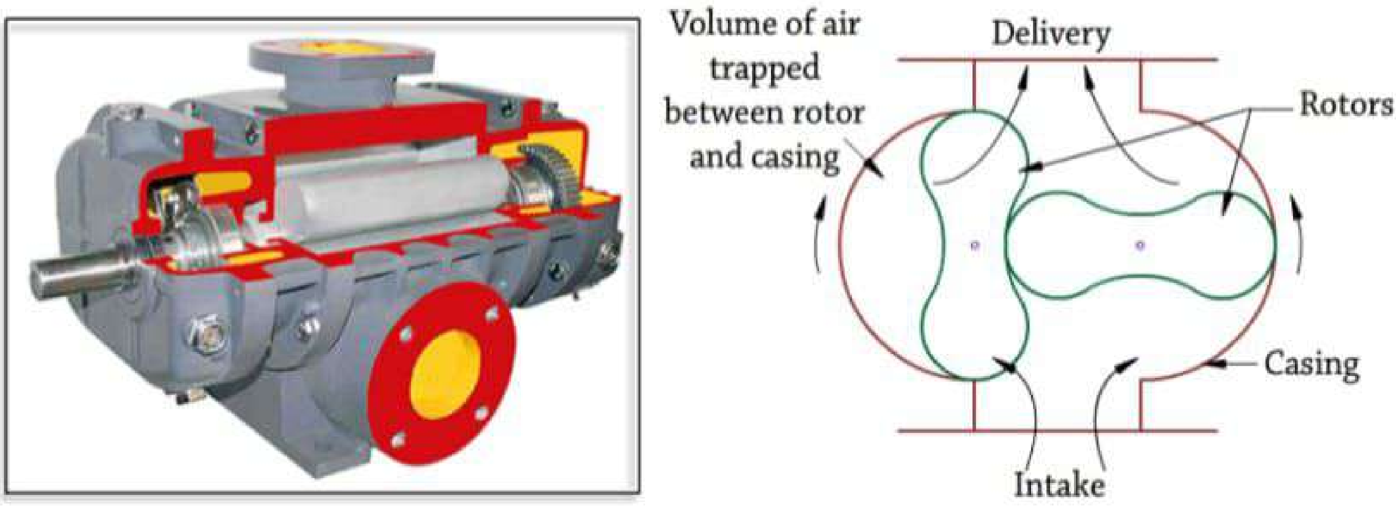

Fig 2: Roots blower air compressor

Construction and working

- The unit consists of two rotors each having two lobes and are enclosed in a casing.
- The two rotors are driven by a pair of equal spur gears and revolved in opposite directions.
- There is a small clearance between the rotors and outer casing.
- This clearance reduces the wear, but forms a leakage path which has an adverse effect on the performance of the blower.
- As the rotor rotates further, air is trapped between the rotor and casing.
- This air is transferred to the delivery side at constant pressure.
- A further rotation of the rotor open this air to the receiver and air flows back from the receiver since air is at high pressure in receiver.
- The air induced is compressed irreversibly by the air from the receiver to the delivery pressure p_2, and then delivery begins.

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Vane blower compressor

Construction

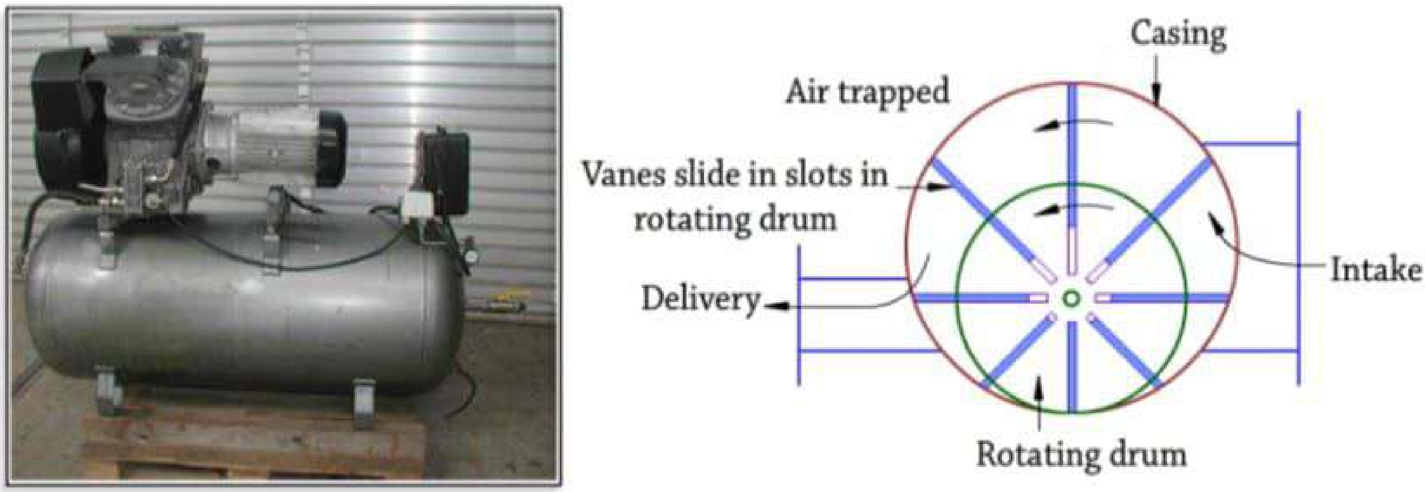

Fig 3: Vane blower compressor

- It consists of a circular casing in which a drum rotates.
- The center of drum is eccentric to the center of casing.
- The slots are cut in the drum to accommodate the straight blades (vanes).
- During the rotation the vanes remains in contact with the casing due to centrifugal force.

Working

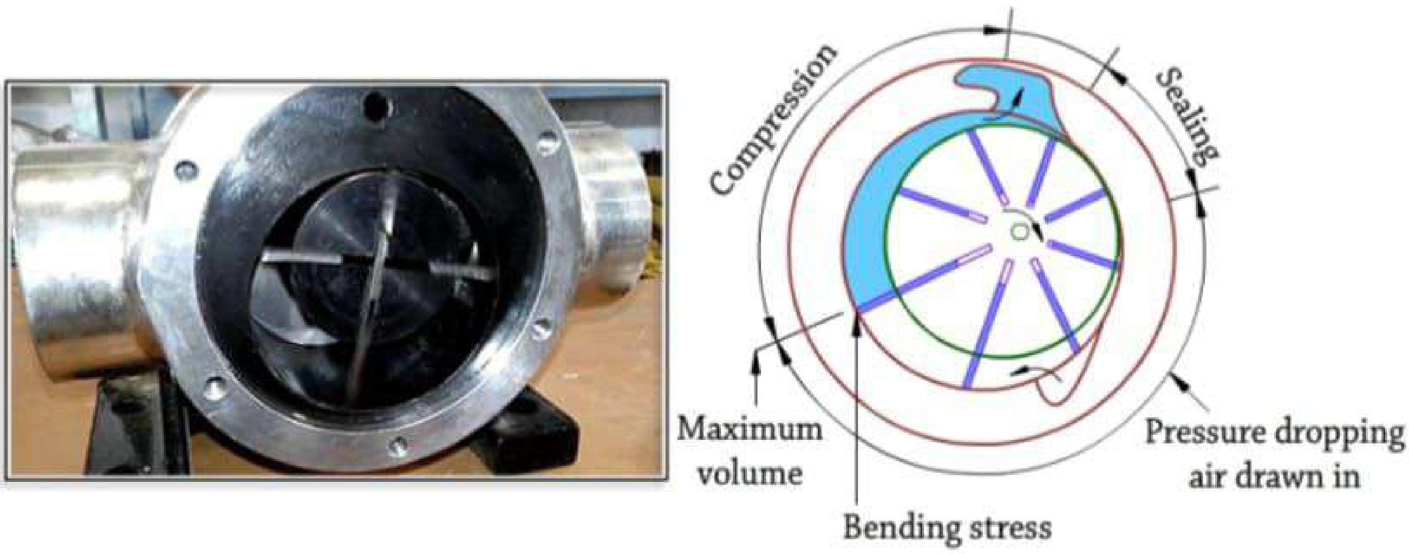

Fig 4

- As the drum rotates, certain volume of air is trapped between vanes, drum and casing.

- This air is partially compressed due to decrease in volume between the drum and casing at the delivery side.
- Further compression can be obtained irreversibly by back-flow of air from the receiver as in case of roots blower.
- The delivery begins when the vanes uncovers the delivery passage.
- The vane blower requires less energy compared to roots blower.

Uses

- These are used for supercharging I.C engines.
- Scavenging in 2-stroke engines.
- Used as a compressor in gas turbine unit.

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Comparison of reciprocating and rotary compressor

Factors	Reciprocating compressor	Rotary compressor
Capacity	Free air discharged is about 250-300 m^3/min	Free air discharged is about 2000 - 3000 m^3/min
Delivery pressure	Maximum delivery pressure ranges from 800 to 1000 bars	Maximum delivery pressure is about 10 bars
Suitability	Suitable for low discharge of air at high pressure	Suitable for large discharges at low pressures
Compression	Isothermal compression can be achieved due to low speed	Adiabatic compressions can be achieved due to high speed
Air supply	Air supply is intermittent	Air supply is continuous
Operational speed	Low, cannot be directly coupled to prime mover	High, can be directly coupled to prime mover
Quality of air delivered	Not clean, generally contaminated with lubricating oil	Relatively more clean
Lubrication system	Complicated	Simple
Balancing	Major problem due to cyclic vibrations	Balancing is perfect as compres is free from vibrations

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Non-positive displacement (steady flow) compressors

- In non-positive displacement compressors, the compression occurs by transfer of kinetic energy from a rotor.
- The centrifugal (radial) compressor and axial flow compressor are two,basic types of non-positive displacement compressors.

Centrifugal (radial flow) compressor

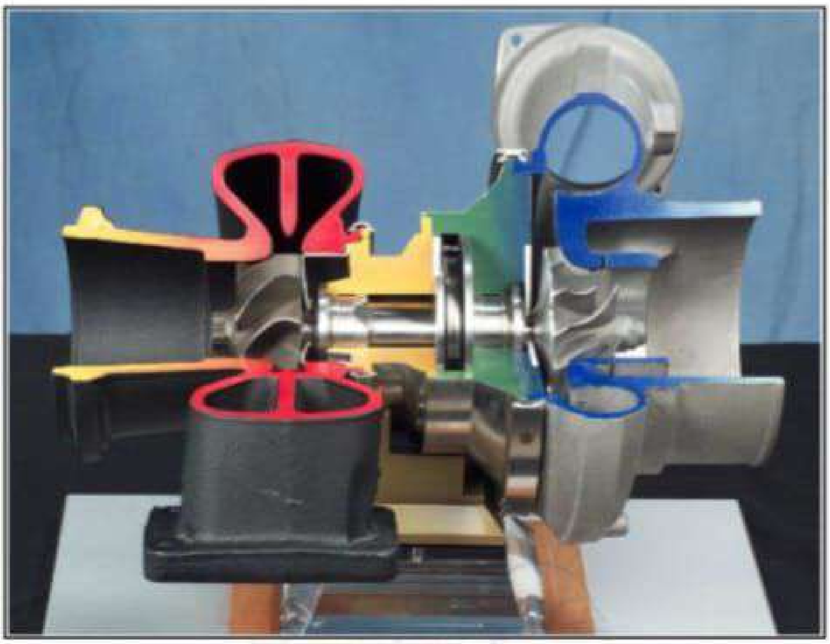

Fig 1: Centrifugal compressor

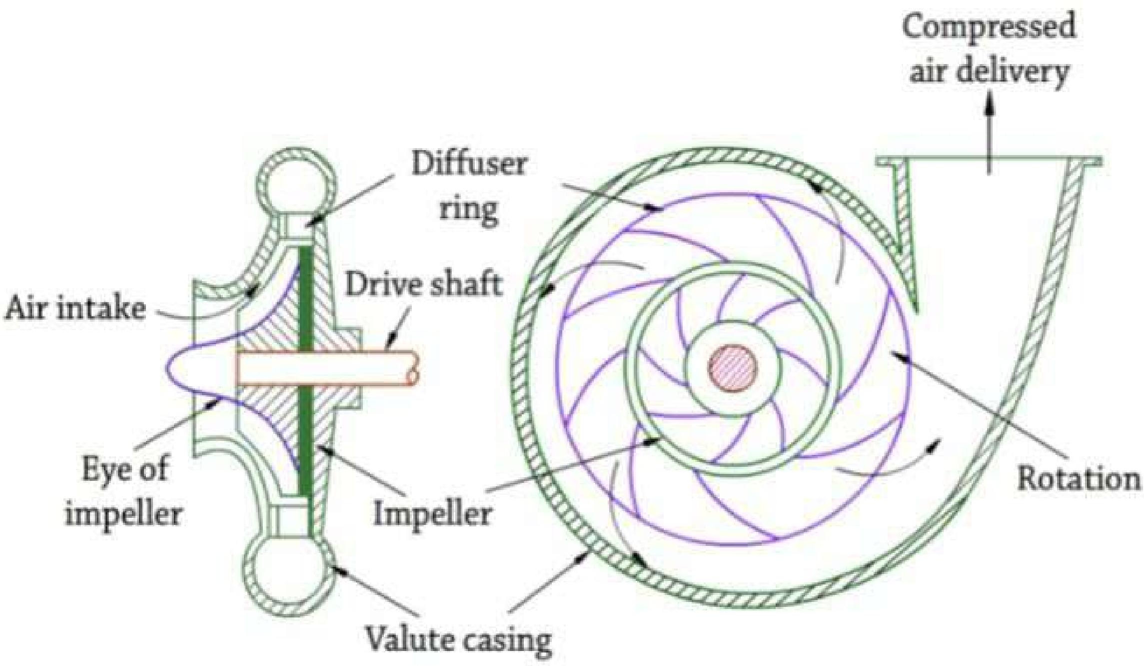

Fig 2: Centrifugal compressor

Construction and working

- Centrifugal compressor consists of a rotating impeller surrounded by a diffuser.

- The impellor and diffuser are concentric, and are enclosed in volute casing.
- The clearance between casing and diffuser gradually increases towards the delivery side.
- The air enters through the eye of an impellor.
- Due to centrifugal force air flows radially outwards with high velocity.
- As the air moving outward, more air flows into the impeller, creating a continuous air flow.
- The air at high velocity passes into a diffuser where the kinetic energy acquired is converted into pressure energy.
- The air at high pressure is delivered to the receiver through the divergent passage of casing.
- This type of compressor is a continuous flow device, and deals with large quantities of air at moderate pressures.
- The pressure ratios between 4:1 to 6:1 may be-achieved.

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Axial flow compressor

- Axial flow compressors are integral to the design of large gas turbines such as jet engines, high speed ship engines, and small scale power stations, etc.,

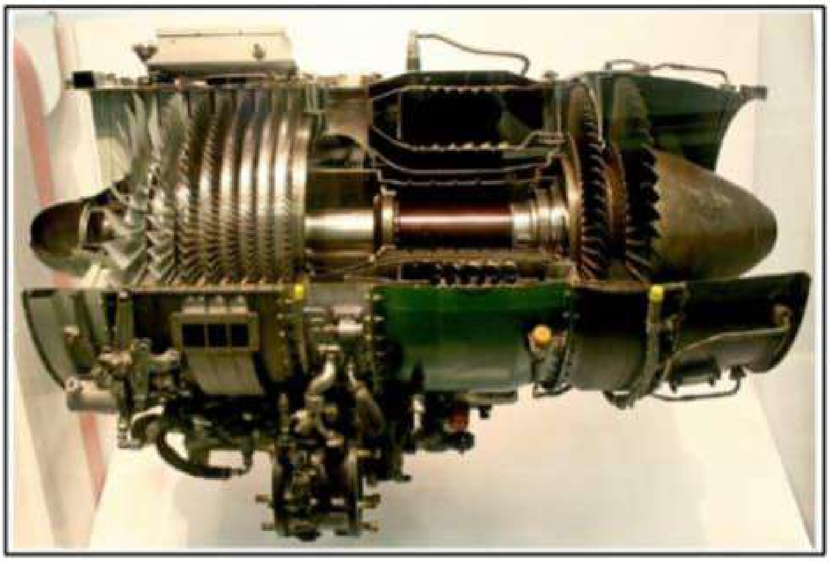

Fig 3: Axial flow compressor

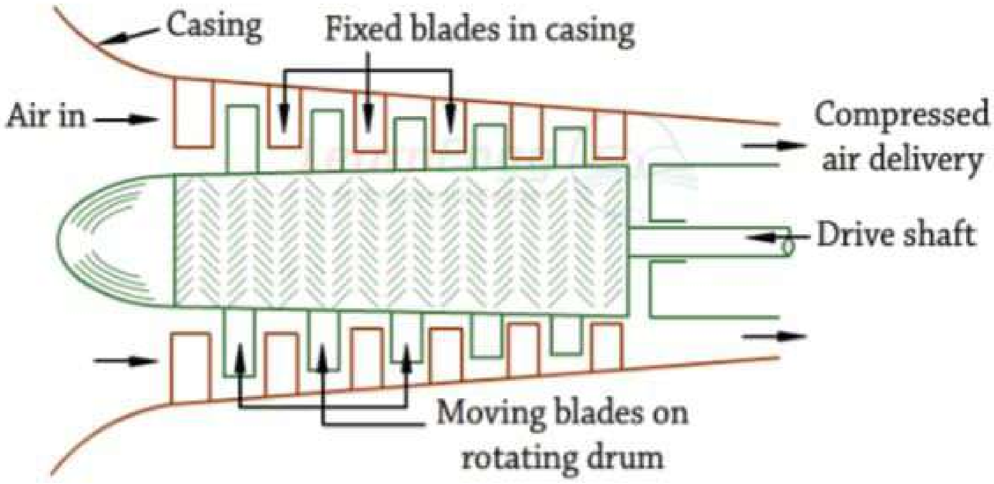

Fig 4: Axial flow compressor

Construction and working

- The axial flow compressor consists a casing and a central drum which is driven by prime mover.
- The blades fitted to the casing are stationary where as the moving blades are attached to the drum.
- Fixed and moving blades are arranged alternately.
- A pair of fixed and moving blades is considered to be one stage of compressor.
- The flow of air is essentially axial, and pressure increases in both fixed and moving blades.

- Fixed blades directs air flow into the moving blades and the kinetic energy of air is converted into pressure energy due to diffuser action in the fixed blades.
- In moving blades the pressure of air is rises further due to decrease in relative kinetic energy of the air.
- Finally the air at high pressure leads to delivery pipe from the casing.
- Pressure ratio of 10:1 or more can be achieved with axial flow compressor.

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Comparison between centrifugal and axial flow compressors

Factors	Centrifugal compressor	Axial flow compressor
Direction of flow	Air flows radially i.e., perpendicular to the axis of compressor	Air flows axially i.e., parallel to the axis of compressor
Operating principle	Pressure increases in the diffuser	Pressure increases in the fixed and moving blades
Pressure ratio per stage	4:1	12:1
Flexibility of operation	More	Less
Frontal area	More	Less
Starting torque	Low	High
Application	Used in I.C engines for supercharging and for compressing refrigerants.	Used in large gas turbine units.

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Frequently Asked Questions

- What are the uses of compressed air ?
- How do you classify air compressors ?
- Explain the following terms:
  - Pressure ratio
  - Compressor capacity
  - Swept volume
  - Mean effective pressure

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Frequently Asked Questions

- Explain the working of single stage reciprocating air compressor with a neat line diagram.
- Write down the work done and power required by a single stage compressor using p-V diagram.

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Frequently Asked Questions

- Draw a p-V diagram for reciprocating air compressor with clearance volume and write the formula for work done.
- List out the factors effecting the volumetric efficiency.

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Frequently Asked Questions

- What is the function of intercooler ?
- What are the advantages of multistage compressors ?
- Draw the line diagram of an intercooler and explain the working.
- List out the uses of intercooler.
- Write down the assumptions made to get the maximum efficiency.
- Derive the expression for 2–stage compression with perfect inter cooler.

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Frequently Asked Questions

- Determine the minimum work required to compress 1 kg of air from 1 bar abs and 27 °C to 9 bar abs in 2 stages. The law of compression is $pV^{1.35} = c$ and inter cooling is complete. If the air was compressed in one stage between the same pressure limits, what is the percentage saving in work by compressing it in two stages. Assume: R = 0.287 kJ/kgK.

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Frequently Asked Questions

- How do you classify the rotary air compressors ?
- Explain the working principle of the following with a neat sketches
  - Roots blower
  - Vane blower
- Differentiate between reciprocating and rotary compressors.

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Frequently asked questions

- Describe centrifugal compressor with a neat sketch and explain its working?
- Describe an axial flow compressor with a neat sketch.

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