Republic of Iraq

Ministry of Higher Education and Scientific Research

Southern Technical University

Department of Power Mechanics

Lecture No1

INTRODUCTION

By

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Lecture No: 1 **(INTRODUCTION)**

1.1 General

The internal combustion engine (I.C.E.) is a heat engine that converts chemical energy in a fuel to mechanical energy, usually made available on a rotating shaft. Internal combustion engines have been in use for more than a century and have undergone tremendous changes in design, materials used and operating characteristics. Never once during their long history of development have they lost their importance as the planet ́s most widely used prime movers. It is a well-known fact that I.C.E. is one of the most fuel-efficient power producing units in use today. Hence, the importance of this type of engine has been long recognized, particularly in heavy duty engine applications.

1.2 Basic Engine Nomenclature

The following terms and abbreviations are commonly used in engine technology literature. These should be learned to assure maximum understanding of the following lectures.

1. Cylinder bore (B). The nominal inner diameter of the working cylinder.
2. Stroke (L). The nominal distance through which a working piston moves between two successive reversals of its direction of motion.
3. Connecting rod (R). Rod connecting the piston with rotating crank shaft.
4. Crankshaft: Rotating shaft through which engine work output is supplied to external systems.
5. Piston. The cylindrical shaped mass that reciprocated back and forth in the cylinder as shown in Fig.(1), transmitting the pressure forces in the combustion chamber to the rotating crankshaft.
6. Piston area (A). The area of a circle of diameter equal to the cylinder bore.

7. Top dead center (TDC). When the position of the piston and moving parts is farthest to the crank shaft.

8. Bottom dead center (BDC). When the position of the piston and moving parts is nearest to the crank shaft.

9. Displacement volume (swept volume) (stroke volume) (V_d). The nominal volume generated by the working piston when traveling from one dead center to the next one, calculated as the product of piston area and stroke.

10. Clearance volume (V_c). The nominal volume of the space on the combustion side of the piston at (TDC).

11. Cylinder volume (V). the sum of the displacement volume and clearance volume $V=V_d+V_c$

12. Crankcase: part of the engine bock surrounding the rotating crankshaft

13. Connecting rod bearing: bearing where connecting rod fastens to crankshaft

14. Valves: used to allow flow into and out of the cylinder at the proper time in the cycle.

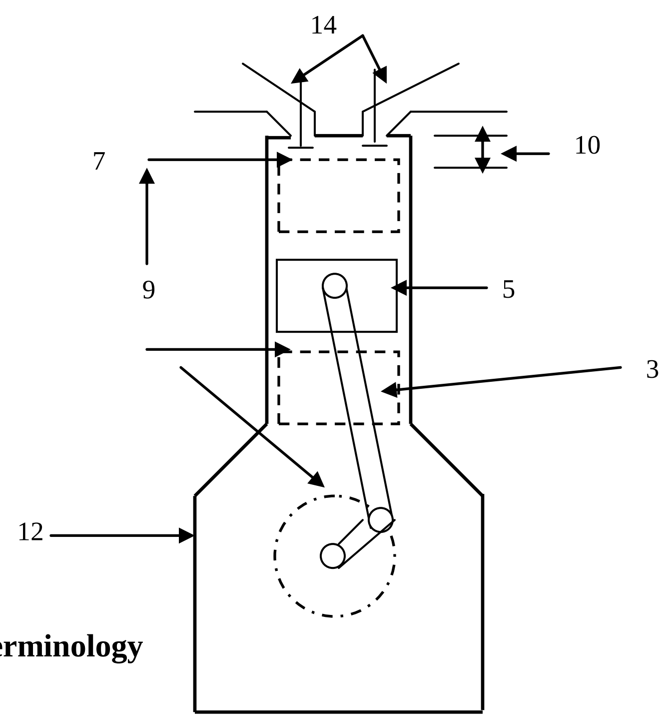


Fig.1 Engine Terminology

1.3 I.C. Engine Classifications

Internal combustion engines can be classified in a number of different ways:

1. Type of ignition:

(a) Spark Ignition Engine (S.I.E.): An engine in which the combustion process is started by use of a spark plug.

(b) Compression Ignition Engine (C.I.E.) also called Diesel Engine: An engine in which the combustion process is started when the air-fuel mixture self-ignites due to high temperature in the combustion chamber caused by high compression.

2. Engine Cycle:

(a) Four Stroke Cycle (b) Two Stroke Cycle

3. Valve Location: see fig.(1.)

(a) Valves in head (overhead valve) also called I Head engine.

(b) Valves in block (flat head), also called L Head engine. Some historic engines with valves in block had the intake valve on one side of the cylinder and the exhaust valve on the other side.

(c) F head engine in which intake valve in head and one in block.

4. Basic Design

(a) Reciprocating. Engine has one or more cylinder in which pistons reciprocate back and forth.

(b) Rotary. Engine is made of a block (stator) built around a large non-concentric rotor and crankshaft.

5. Position and number of Cylinders.

(a) Single Cylinder

(b) In-line. Cylinders are positioned in a straight line, one behind other along the length of crankshaft as shown in fig.(1.2).

(c) V Engine. Two banks of cylinders at an angle with each other along a single crankshaft

(d) Opposed Cylinder Engine. Two banks of cylinders opposite each other on a single crankshaft.

(e) W engine. Same as V engine except with three banks of cylinders on the same crankshaft.

(f) Opposed Piston Engine. Two pistons in each cylinder which combustion chamber in the center between the pistons.

(g) Radial Engine. Engine with pistons positioned in a circular plane a rounded the central crankshaft.

6. Air Intake Process.

(a) Naturally Aspirated. No intake pressure boost system.

(b) Supercharged. Intake air pressure increased with the compressor driven-off of the crank shaft.

(c) Turbocharged. Intake air pressure increased with the turbine-compressor driven by the engine exhaust gases

(d) Crankcase Composed. Two stroke cycle engine which uses crankcase as the intake air compressor.

7. Method of Fuel Input for SI Engines.

(a) Carbureted. (b) Multipoint Port Fuel Injection. (c) Throttle Body Fuel Injection.

8. Fuel Used.

(a) Gasoline. (b) Diesel oil or Fuel Oil. (c) Gas, natural gas, Methane.

(d) LPG. (e) Alcohol-Ethyl, Methyl.

(f) Dual Fuel. There a number of engines that use a combination of two or more fuels such as CI engines use a combination of methane and diesel fuel.

(g) Gasohol. Common fuel consisting of 90 % gasoline and 10 % alcohol

9. Application.

(a) Automobile, Truck, Bus. (b) Locomotive. (c) Stationary. (d) Marine.

(e) Aircraft. (f) Small Portable.

10. Type of Cooling.

(a) Air Cooled. (b) Liquid Cooled, Water Cooled.

1.4 Basic Engine Cycles

Most internal combustion engines, both spark ignition and compression ignition operate on either a four stroke or two stroke cycles. These basic cycles are fairly standard for all engines, with only slight variation found in individual designs.

1.4.1 Four Stroke Cycle

In four stroke cycle there is one working cycle per two revolutions. The sequence of operation for the four stroke cycle is as follows:

1**. Intake stroke:** the piston travels TDC to BDC with intake valve open and exhaust valve closed as shown in Fig.(2). This creates an increasing volume in the combustion chamber which in turn crates a vacuum. The resulting pressure differential through the intake system from atmosphere pressure on the outside to the vacuum on the inside causes air to be pushed into the cylinder.

2. **Compression stroke:** when the piston reaches BDC, the intake valve closes and the piston travel back to TDC with all valves closed. This compress air-fuel mixture raising both temperature and pressure in the cylinder.

3. Combustion: combustion of the fuel air mixture occurs in very short time with finite length with the piston near TDC. Its starts near the end of compression stroke slightly bTDC and lasts in to power stroke slightly aTDC. Combustion changes the composition of the gas mixture to exhaust products and increases both temperature and pressure in the cylinder to a very high peak value.

4. Expansion stroke or Power stroke: with al valves closed, the high pressure created by the combustion process pushes the piston away from TDC. This is the stroke which produces the work output of the engine cycle. As the piston travel from TDC to BDC, cylinder volume increased, causing pressure and temperature to drop.

5. Exhaust Blow down: late in the power stroke, the exhaust valve is opened and exhaust blow down occurs. Pressure and temperature in the cylinder is still high relative to the surroundings at this point, and pressure differential is created through the exhaust system which is open to the atmospheric pressure. This pressure differential causes much of the hot exhaust gas to be pushed out of the cylinder and through the exhaust system when the piston near BDC.

6. Exhaust stroke: by the time the piston reaches BDC exhaust blow down is complete, but the cylinder is still full of exhaust gases at approximately atmospheric pressure. With the exhaust valve remaining open, the piston now travels from BDC to TDC in the exhaust stroke as shown in Fig.(2). This pushes most of the exhaust gases out of the cylinder into the exhaust system.

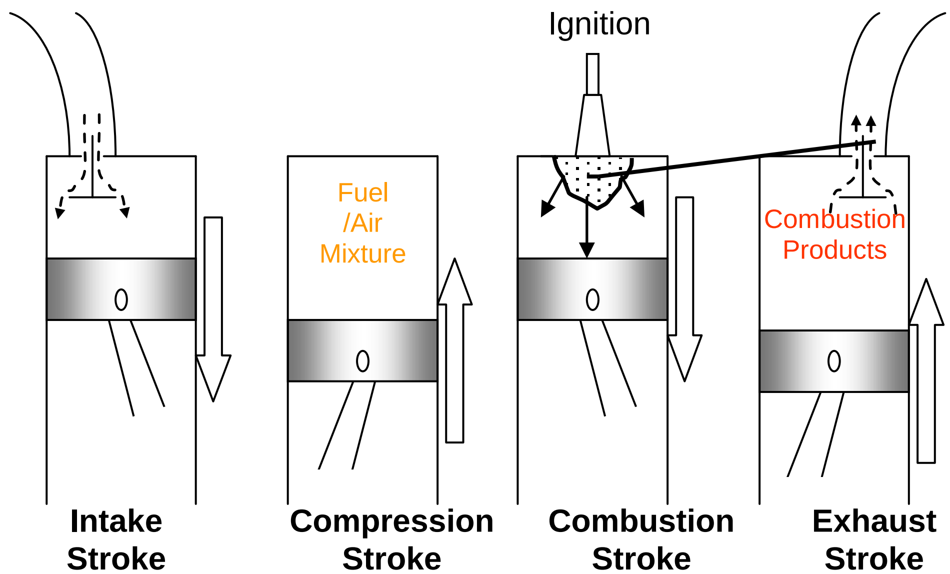
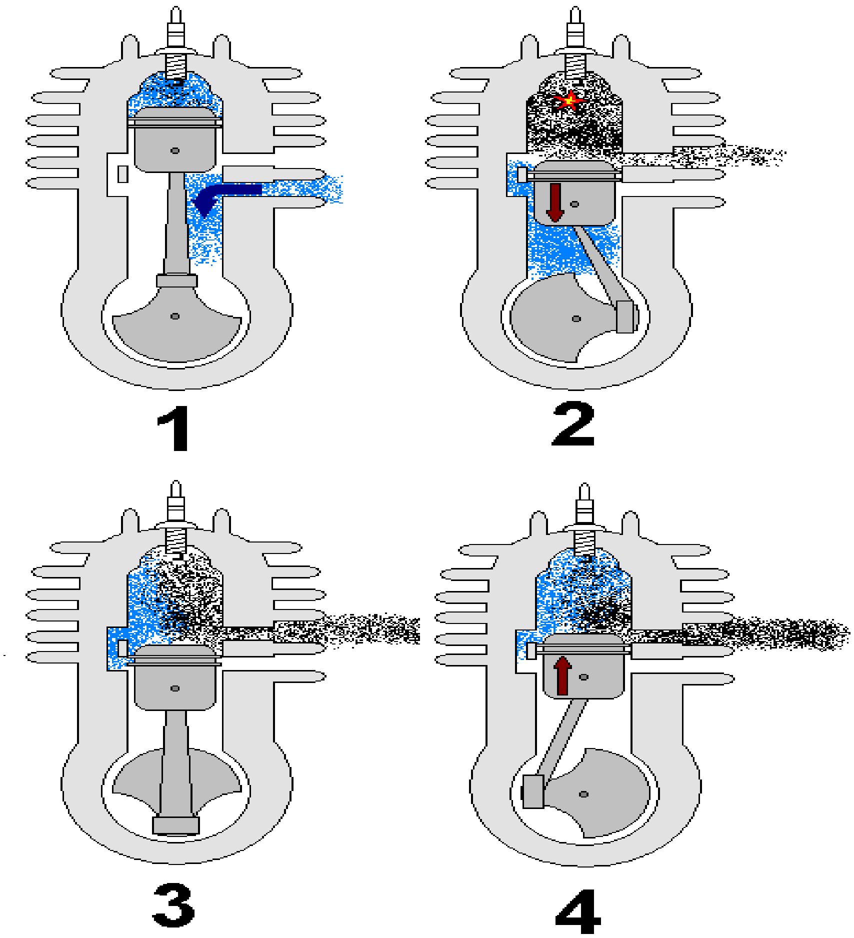


Fig.(2) Principle operation of four stroke cycle

1.4.2 Two Stroke Cycle

The difference between four and two stroke engines is in the method of filling the cylinder with the fresh charge and removing the burned gases from the cylinder. In 4-stroke engine these operations are performed by the engine piston during the suction and exhaust respectively. In 2-stroke engine suction is accomplished by air compressed in crank case or by blower. The induction of compressed air removed the products of combustion through exhaust ports without the use of piston for these two operations see Fig.3(a-b)



(a)

(b)

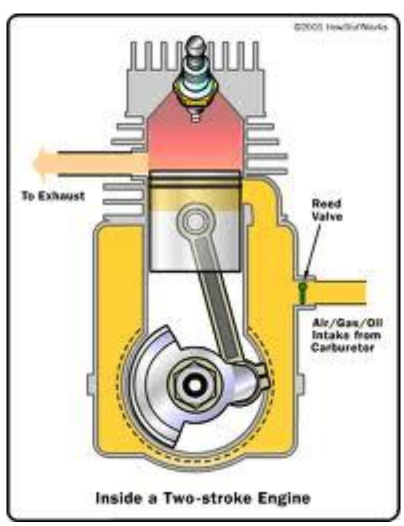


Fig.3(a-b) The two Stroke Cycle Gasoline Powered Spark Ignition Engine

Lecture No: 2 (OPERATING CHARACTERISTIC)

This lecture examines the operating characteristic of reciprocating internal combustion engines. Theses includes the mechanical output parameters of work, torque, and power; the input requirements of air, fuel and combustion efficiencies.

2.1 ENGINE PARAMETERES

For an engine of bore B shown in fig.(2.1), crank radius a, stroke length S, turning at an engine speed of N(rpm)

$S = 2a$

Average piston speed is: $U_p = 2S\,N$

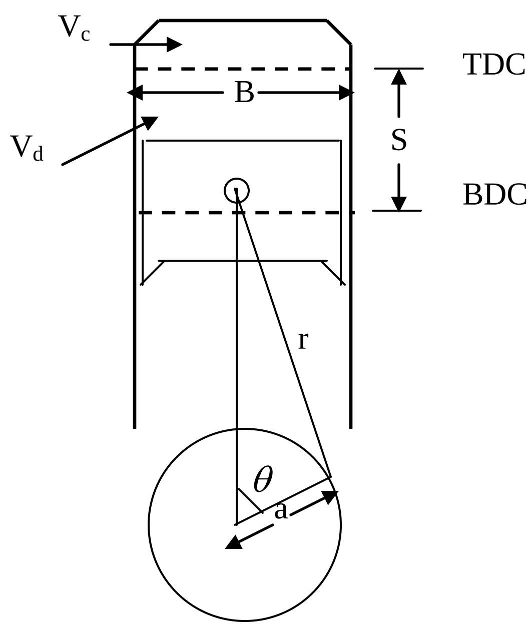


Fig.(2.1) Piston and Cylinder geometry

Swept Volume or (stroke volume) or (displacement volume) can be given for one cylinder engine: $V_d = (\pi / 4) B^2 S$

For an engine with N_c cylinders: $V_d = N_c (\pi / 4) B^2 S$

Engine displacement volume can be given in m^3, cm^3, in^3, and most commonly in liters (L).

$1L = 10^{-3} m^3 = 10^3 cm^3 \approx 61 in^3$

The compression ratio of an engine is defined as: $\varepsilon = \dfrac{V_{BDC}}{V_{TDC}} = \dfrac{(V_c + V_d)}{V_c}$

Modern SI engines have compression ratios of 8 to 11, while CI engines have compression ratios of in the range 12 to 24.

The cross section area of cylinder and the surface area of a flat-topped piston are each given by: $A_p = (\pi / 4) B^2$

Ex1: An automobile has a three liter SI V6 engine that operates on a four stroke cycle at 3600 rpm. The compression ratio is 9.5, and the engine is square (B=S). Calculate:

1. Cylinder bore and stroke length.
2. Average piston speed.
3. Clearance volume of one cylinder.

Solution:

1. For one cylinder, with (B=S)

$$V_d = V_{total} / 6 = 3L / 6 = 0.5L = 0.0005 m^3 = (\pi / 4) B^2 S = (\pi / 4) B^3 \rightarrow B = 0.0860 m$$
$$\therefore B = 8.6 cm = S$$

2. Average piston speed

$$U_p = 2 S N = 2 \times 0.0860 \times \left(\frac{3600}{60} \right) = 10.32 m/s$$

3. Clearance volume of one cylinder

$$\varepsilon = (V_d + V_c)/V_c \rightarrow 9.5 = (0.0005 \ + V_c)/V_c$$

$$\therefore V_c = 0.000059 m^3 = 59 cm^3$$

2.2 WORK

Work is the output of any heat engine, and in a reciprocating IC engine this work is generated by the gases in the combustion chamber of the cylinder. Work is the result of force acting through a distance. Force due to gas pressure on the moving piston generates the work in an IC engine.

$$W = \int F\,dx = \int P A_p\,dx$$

Where P=pressure in combustion chamber.

A_p=area against which the pressure acts (piston force).

x=piston distance.

$A_p\,dx = dV$ where dV is the differential volume displaced by the piston, so work done can be written:

$$W = \int P\,dV$$

Because engines are often multicylinder, it is convenient to analysis engine cycles per unit mass of gas within the cylinder

$$w = W/m \quad \& \quad v = V/m$$

$$\therefore w = \int P\,dv$$

It is important to distinguish between indicated work and brake work. Indicated work gives the work inside combustion chamber, but work delivered by the crank shaft called brake or effective work which is less than indicated work due to mechanical friction and parasitic loads of the engine.

$$w_f = w_i - w_b$$

The ratio of brake work at the crank shaft to indicated work in the combustion chamber defines the mechanical efficiency of an engine:

$$\eta_m = W_b / W_i = w_b / w_i$$

2.3 MEAN EFFECTIVE PRESSURE

Mean effective pressure is a good parameter to compare engines for design because it's dependent of engine size and /or speed. If torque is used for engine comparison, a large engine will always look better. If power is used as the engine comparison, speed becomes very important.

Various mean effective pressure can be defined by using different work terms. If brake work is used, **brake mean effective pressure** is obtained:

$bmep = W_b / \Delta V = W_b / V_d$

And if indicated work is used, **indicated mean effective pressure** is obtained: $imep = W_i / \Delta V = W_i / V_d$

Friction mean effective pressure is $fmep = W_f / V_d$

Also $fmep = imep - bmep$ & $\eta_m = bmep / imep$

2.4 TORQUE AND POWER

Torque is a good indicator of an engine's ability to do work. It's defined as force acting at a moment distance. Torque is related to work by:

$T = (bmep) V_d / 2\pi n$ Where n: number of revolutions per cycle.

For a two stroke cycle engine which take one revolution per cycle:

$T = (bmep) V_d / 2\pi$ two stroke cycle $n = 1$.

For a four stroke cycle engine which take two revolution per cycle:

$T = (bmep) V_d / 4\pi$ Four stroke cycle $n = 2$.

In these equations $bmep$ is used because torque is measured from the output crankshaft.

Power is defined as the rate of work of the engine. Depending on the definition of *mep* is used power can be defined as indicated or brake power respectively.

$W_i^{\cdot} = (imep) A_p U_p / 2n$ & $W_b^{\cdot} = (bmep) A_p U_p / 2n$

$W_i^{\cdot} = (imep) A_p U_p / 2$ Two stroke cycle.

$W_b^{\cdot} = (bmep) A_p U_p / 2$

$W_i^{\cdot} = (imep) A_p U_p / 4$ Four stroke cycle.

$W_b^{\cdot} = (bmep) A_p U_p / 4$

Also $\eta_m = \dfrac{W_b^{\cdot}}{W_i^{\cdot}}$ and $W_f^{\cdot} = W_i^{\cdot} - W_b^{\cdot}$ where $W_f^{\cdot}$ friction power.

Power is measured in kW, but horse power is still common

$1hp = 0.7457\,kW$

Specific power (SP)$= W_b^{\cdot} / A_p$

Output power per displacement (OPD) $= W_b^{\cdot} / V_d$ & Engine specific volume (BSV) $= V_d / W_b^{\cdot}$

Ex2: The engine in example 1 gives a brake torque reading of 205 N.m at 3600 rpm. At this speed air enters the cylinders at 85 kPa and 60 ^{o}C, and the mechanical efficiency of the engine is 85% calculate:

1.brake power 2. indicated power 3. *bmep* 4. *imep* 5. *fmep*

6. power lost to friction 7. brake work 8. brake specific power

9. brake output per displacement 10. engine specific volume.

Solution:

1. $\dot{W}_b = 2\pi NT = (2\pi radians/rev)(3600/60 rev/\sec)(205 N.m)$

$= 77.3 kW = 104 hp$

2. $\dot{W}_i = \dfrac{\dot{W}_b}{\eta_m} = \dfrac{77.3}{0.85} = 90.9 kW = 122 hp$

3. $bmep = 4\pi T/V_d = (4\pi radians/cycle)(205 N.m)/(0.003 m^3/cycle)$
$= 859 kPa$

4. $imep = \dfrac{bmep}{\eta_m} = \dfrac{859}{0.85} = 1010 kPa$

5. $fmep = imep - bmep = 1010 - 859 = 151 kPa$

6. To find friction power lost:

$A_p = (\pi/4)B^2 = (\pi/4)(0.086)^2 = 0.00581 m^2$ for one cylinder

$\dot{W}_i = (fmep)A_p U_p/4 \quad = (1/4)(151)(0.00581)(10.32)\times(6cyl) = 13.6 kW = 18 hp$
Or, it can found from

$\dot{W}_f = \dot{W}_i - \dot{W}_b = 90.9 - 77.3 = 13.6 kW$

7. Brake work for one cylinder for one cycle is $W_b = (bmep)V_d$

$= (859 kPa)(0.0005 m^3) = 0.43 kJ$

It can be assumed the gas entering the cylinders at BDC is air:

$m_a = PV_{BDC}/RT = P(V_d + V_c)/RT$
$= 85\times(0.0005 + 0.000059)/(0.287)(333) = 0.00050 kg$

Brake specific work per unit mass:

$w_b = W_b/m_a = 0.43/0.00050 = 860 kJ/kg$

8. $BSP = \dot{W}_b/A_p = 77.3/\left[(\pi/4)(0.086)^2\times 6\right] = 2220 kW/m^2$

9. $BOPD = \dot{W}_b/V_d = 77.3/3 = 25.8 Kw/L$

10. $BSV = V_d/\dot{W}_b = 1/BOPD = 1/25.8 = 0.0388 L/kW$

2.5 AIR-FUEL RATIO AND FUEL-AIR RATIO

Energy input to an engine Q_{in} comes from the combustion of a hydrocarbon fuel. Air is used to supply the oxygen needed for this chemical reaction. For combustion reaction to occur, the proper relative amount of air (oxygen) and fuel must be present.

Air-fuel ratio (AF) and fuel-air ratio (FA) are parameters used to describe the mixture ratio

$AF = m_a / m_f = \dot{m}_a / \dot{m}_f$

$FA = m_f / m_a = \dot{m}_f / \dot{m}_a$

2.6 SPECFIC FUEL CONSUMPTION

Specific fuel consumption is defined by: $\dot{m}_f / \dot{W}$

Where $\dot{m}_f$ & $\dot{W}$ are mass flow rate of fuel into engine and engine power respectively.

Indicated power gives **indicated specific fuel consumption** $isfc = \dot{m}_f / \dot{W}_i$

Brake power gives **brake specific fuel consumption** $bsfc = \dot{m}_f / \dot{W}_b$

It also we have: $\eta_m = \dot{W}_b / \dot{W}_i = \left(\dot{m}_f / \dot{W}_i\right) / \left(\dot{m}_f / \dot{W}_b\right) = (isfc)/(bsfc)$

2.7 ENGINE EFFICIENCES

the time available foe the combustion process of an engine is very brief, and not all fuel molecules may find an oxygen molecule with which to combine, or the local temperature may not favor a reaction. Combustion efficiency η_c is defined to account for the fraction of fuel which burns and has values in the range 0.95-0.98 when an engine is operating properly. For one engine cycle in one cylinder, the heat added is:

$Q_{in} = m_f Q_{HV} \eta_c$ Or $\dot{Q}_{in} = \dot{m}_f Q_{HV} \eta_c$

And the thermal efficiency is:

$\eta_t = W / Q_{in} = \dot{W} / \dot{Q}_{in} = \dot{W} / \dot{m}_f Q_{HV} \eta_c$

Where Q_{HV} heating value of fuel

Thermal efficiency can be given as indicated or brake, depending on whether indicated power or brake power is used.

$\eta_m = (\eta_t)_b / (\eta_t)_i$

Fuel conversion efficiency is defined as:

$\eta_f = W / m_f Q_{HV} = \dot{W} / \dot{m}_f Q_{HV}$

$\eta_f = 1/(sfc) Q_{HV}$

2.8 VOLUMETRIC EFFICIENCY

Volumetric efficiency is the ratio of actually intake volume to the stroke volume. Volumetric efficiency is defined as:

$\eta_v = m_a / \rho_a . V_d$

$\eta_v = n\dot{m}_a / \rho_a . V_d N$

Where m_a = mass of air into the engine (or cylinder) for one cycle.

$\dot{m}_a$ = steady state flow of air into the engine.

ρ_a = air density evaluated at atmospheric conditions.

N = engine speed.

n =number of revolution per cycle.

Ex3: the engine in example 2 is running with an air-fuel ratio AF =15, a fuel heating value of 44,000 kJ/kg, and a combustion efficiency of 97%. Calculate:

1. Rate of fuel flow into engine.
2. Brake & indicated thermal efficiencies.
3. Volumetric efficiency.
4. Brake specific fuel consumption.

Solution: from **Example 2** the mass of air in one cylinder for one cycle is 0.00050 kg

1. $m_f = m_a / AF = 0.00050/15 = 0.000033 kg$ of fuel per cylinder per cycle.

Therefore, the rate of fuel flow into the engine is:

$\dot{m}_f = (0.000033) \times (6cyl) \times (3600/60) \times (1cycle/2rev) = 0.0060 kg/\sec$

2. $(\eta_t)_b = \dot{W}_b / \dot{m}_f Q_{HV} \eta_c = (77.3)/(0.0060)(44000)(0.97) = 0.302 = 30.2\%$

Or using: $(\eta_t)_b = W / m_f Q_{HV} \eta_c = (0.43)/(0.000033)(44000)(0.97) = 0.302$

$\therefore (\eta_t)_i = (\eta_t)_b / \eta_m = 0.302/0.85 = 0.355 = 35.5\%$

3. $\eta_v = m_a / \rho_a . V_d = (0.00050)/(1.181)(0.0005) = 0.847 = 84.7\%$

4. $bsfc = \dot{m}_f / \dot{W}_b = (0.0060)/(77.3) = 7.76 \times 10^{-5} kg/kW.\sec$

Ex4: A two cylinder I.C.E. working on 2-stroke cycle principle to develop 41.8 I.hp at 1000 rpm. If the mean effective pressure is 6 kgf/cm^2. Find the necessary bore and stroke of the piston assuming stroke 1.5 times the bore.

Ex5: A six cylinder I.C.E, 12 cm by 15 cm has a piston speed of 480 m/min. it develops 60 bhp and has mechanical efficiency of 75%. The mean effective pressure is 4.42 kgf/cm^2. The specific fuel consumption is 0.25 kg per bhp. Hour and the calorific value of fuel is 10,000 kcal/kg. Determine:

a. Whether this is a two stroke or four stroke cycle engine.
b. The thermal efficiency based on brake horse power.

Ex6: 4-stroke, single cylinder gas engine has a bore of 146 mm and a stroke of 280 mm. at 475 rpm and full load the net load no the friction brake is 433 N, and the torque arm is 0.45 m. the indicator diagram gives a net area of 578 mm^2 and a length of 70 mm with a spring rating of 0.815 bar/mm. calculate the indicated power, brake power, and mechanical efficiency.

Lecture No: 3 (AIR STANDRD CYCLES)

3.1 INTRODUCTION

The engine cycle analysis is an important tool in the design and study of I.C.E. a thermodynamic cycle is defined as series of processes through which the working fluid progresses, which will eventually return the fluid to its original state. In other wards a thermodynamic cycle implies a closed system with no exchange of matter with the surrounding.

3.2 AIR STANDARD CYCLES

Air standard cycles assume that the working fluid in the engine is always an ideal gas; namely, pure air with constant specific heats and that:

(a) A fixed mass of air is the working fluid thought the entire cycle, i.e. there are no intake and exhaust processes.

(b) The combustion process is replaced by a heat transfer process from an external source

(c) The cycle is complete by heat transfer to the surroundings until the air temperature and pressure to the initial conditions.

The three cycles of great practical importance in the analysis of piston engine performance are:

3.2.1 Constant Volume or Otto Cycle.

This cycle consists of four processes as shown in the (P-V) and (T-S) diagrams respectively. The first law of thermodynamic is applied to each process.

Process (1-2) Isentropic Compression Process

$$\Delta q_{1-2} = \Delta u_{1-2} + \Delta w_{1-2}$$

Where Δq heat transfer to the wall kJ/kg.

Δu Change in the internal energy kJ/kg.

Δw Work down to the system kJ/kg.

Since this process is adiabatic hence, there is no heat transfer to the cylinder walls and the work down during compression is :

$$\Delta w_{1-2} = w_c = \Delta u_{1-2} = c_v (T_2 - T_1) \qquad \text{P(1-2)}$$

For adiabatic process

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

Where ε is the compression ratio,

$$\varepsilon = \frac{V_{cylinder}}{V_c} = \frac{V_1}{V_2}$$

In Otto cycle ε has a value lies between (4-9) and γ specific heat ratio

Process (2-3) Heat Addition Process

$$Q_{2-3} = m_m.\Delta q_{2-3}$$

$$\Delta q_{2-3} = q_{in} = \Delta u_{2-3} + \Delta w_{2-3}$$

At constant volume there is no work down, hence equation above became

$$\Delta q_{2-3} = c_v (T_3 - T_2)$$

$$Q_{2-3} = Q_{in} = m_f Q_{HV} \eta_c = m_m c_v (T_3 - T_2) \qquad \textbf{P(2-3-4)}$$

$$\therefore m_f Q_{HV} \eta_c = (m_a + m_f) c_v (T_3 - T_2) \qquad \div m_f$$

$$\rightarrow Q_{HV} \eta_c = (AF + 1) c_v (T_3 - T_2)$$

Where $AF = \dfrac{m_a}{m_f}$ is air to fuel ratio & η_c combustion efficiency

$T_3 = T_{\max}$; $P_3 = P_{\max}$

Process (3-4) Isentropic Expansion Process

$$\Delta q_{3-4} = \Delta u_{3-4} + \Delta w_{3-4}$$

Since this process is adiabatic hence, there is no heat transfer to the cylinder walls and the work down during expansion is:

$$\Delta w_{3-4} = \Delta u_{3-4} = c_v (T_3 - T_4)$$

Process (4-1) Heat Rejection Process **P(4-1)**

$$Q_{out} = Q_{4-1} = m_m . \Delta q_{4-1}$$

$$\Delta q_{4-1} = q_{out} = \Delta u_{4-1} + \Delta w_{4-1}$$

At constant volume there is no work down, hence equation above became

$$\Delta q_{4-1} = c_v (T_4 - T_1)$$

Also for adiabatic expansion;

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1}$$

The thermal efficiency of Otto cycle is defined as:

$$(\eta_t)_{Otto} = \frac{q_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_v (T_4 - T_1)}{c_v (T_3 - T_2)} = 1 - \frac{T_1\left(\dfrac{T_4}{T_1} - 1\right)}{T_2\left(\dfrac{T_3}{T_2} - 1\right)}$$

We have $\dfrac{T_2}{T_1} = \left(\dfrac{V_1}{V_2}\right)^{\gamma-1} = \varepsilon^{\gamma-1}$

Note the following simplification $$\frac{\frac{T_4}{T_1}}{\frac{T_3}{T_2}} = \frac{\frac{T_2}{T_1}}{\frac{T_3}{T_4}}$$

$\therefore (\eta_t)_{Otto} = 1 - \frac{1}{\varepsilon^{\gamma-1}}$; **Hence the thermal efficiency is function of ε only**

Ex1: Show that the efficiency of ideal Otto cycle depends on compression ratio only?

Ex2: in an ideal constant volume cycle the pressure and temperature at the beginning of compression are 97 kN/m^2 and 50^o respectively, compression ratio 5:1. The heat supplied during the cycle is 930 kJ/kg of working fluid. Determine (a) the maximum temperature attained in the cycle, (b) the thermal efficiency of the cycle and (c) the work down during the cycle/kg of working fluid.

Solution: assume the working fluid is air hence $\gamma = 1.4, c_v = 0.717 kJ/kg.K$

(a) $T_2 = T_1.\left(V_1/V_2\right)^{\gamma-1} = 323 \times 5^{1.4-1} = 323 \times 5^{0.4} = 323 \times 1.903 = 615K$

For constant volume heating

$$Q_{in} = m.c_v.(T_3 - T_2)$$
$$\therefore 920 = 1 \times 0.717(T_3 - 618) \rightarrow T_3 = T_{\max} = 1910K$$

(b) Thermal efficiency

$$(\eta_t)_{Otto} = 1 - \frac{1}{\varepsilon^{\gamma-1}} = 1 - \frac{1}{5^{1.4-1}} = 0.475 = 47.5\%$$

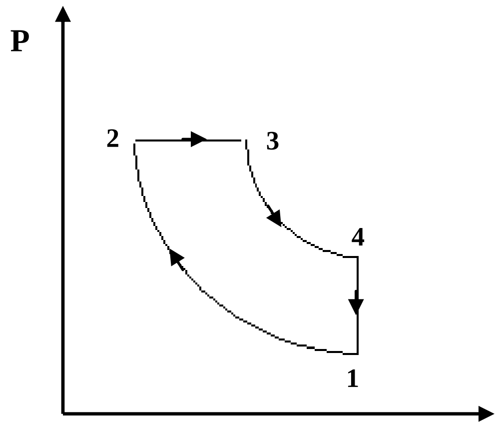
(c) Work down/kg/cycle=heat received/kg/cycle$\times \eta_t$

$$= 930 \times 0.475 = 422kJ$$

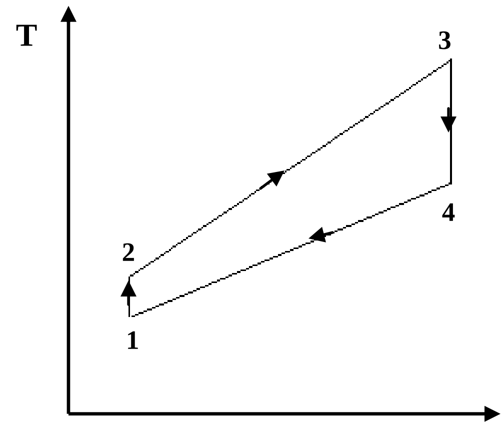
Ex3: Show that for an Otto cycle $\frac{T_3}{T_2} = \frac{T_4}{T_1}$

3.2.2 Constant pressure or Diesel cycle

This cycle consists of four processes as shown in the (P-V) and (T-S) diagrams respectively. The first law of thermodynamic is applied to each process.



(P-V) Diagram **V**



(T-S) Diagram **S**

Process (1-2) Isentropic Compression Process

$$\Delta q_{1-2} = \Delta u_{1-2} + \Delta w_{1-2}$$

Where Δq heat transfer to the wall kJ/kg.

Δu Change in the internal energy kJ/kg.

Δw Work down to the system kJ/kg.

Since this process is adiabatic hence, there is no heat transfer to the cylinder walls and the work down during compression is :

$$\Delta w_{1-2} = w_c = \Delta u_{1-2} = c_v (T_2 - T_1)$$

For adiabatic process P(1-2)

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

Where ε is the compression ratio,

$$\varepsilon = \frac{V_{cylinder}}{V_c} = \frac{V_1}{V_2}$$

In diesel engine ε has a value lies between (12-24) and γ specific heat ratio

Process (2-3) Heat Addition Process

$$Q_{2-3} = m_m.\Delta q_{2-3}$$

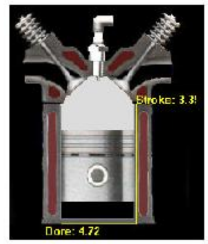
$$\Delta q_{2-3} = q_{in} = \Delta u_{2-3} + \Delta w_{2-3}$$

At constant pressure there is no work down, hence equation became

$$\Delta q_{2-3} = c_p (T_3 - T_2)$$

Another important parameter is the cut-off ratio which is

P(2-3-4)



Equal to $\rho = \frac{V_3}{V_2} = \frac{T_3}{T_2}$

Process (3-4) Isentropic Expansion Process

$$\Delta q_{3-4} = \Delta u_{3-4} + \Delta w_{3-4}$$

Since this process is adiabatic hence, there is no heat transfer to the cylinder walls and the work down during expansion is:

$$\Delta w_{3-4} = \Delta u_{3-4} = c_v (T_3 - T_4)$$

Also for adiabatic expansion;

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{\rho}{\varepsilon}\right)^{\gamma-1}$$

Process (4-1) Heat Rejection Process

P(4-1)

$$Q_{out} = Q_{4-1} = m_m.\Delta q_{4-1}$$

$$\Delta q_{4-1} = q_{out} = \Delta u_{4-1} + \Delta w_{4-1}$$

At constant volume there is no work down, hence equation above became

$$\Delta q_{4-1} = c_v (T_4 - T_1)$$

The thermal efficiency of diesel engine is given by;

$$(\eta_t)_{Diesel} = \frac{q_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_v(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{\gamma . T_2\left(\frac{T_3}{T_2} - 1\right)}$$

$$= 1 - \frac{1}{\varepsilon^{\gamma-1}} . \frac{\left(\frac{T_4}{T_1} - 1\right)}{\gamma\left(\frac{T_3}{T_2} - 1\right)} = 1 - \frac{1}{\varepsilon^{\gamma-1}} . \frac{\left(\frac{\rho . T_1 . \varepsilon^{\gamma-1} . \rho^{\gamma-1}}{T_1 . \varepsilon^{\gamma-1}} - 1\right)}{\gamma\left(\frac{\rho . T_2}{T_2} - 1\right)} = 1 - \frac{1}{\varepsilon^{\gamma-1}} . \frac{(\rho^{\gamma} - 1)}{\gamma(\rho - 1)}$$

$\therefore (\eta_t)_{Diesel} = 1 - \frac{1}{\varepsilon^{\gamma-1}} . \frac{(\rho^{\gamma} - 1)}{\gamma(\rho - 1)}$ which is function of (ε, ρ) only.

Ex4: Show that the efficiency of ideal Diesel cycle depend on compression ratio and cut-off ratio only?

Ex5: An ideal diesel cycle has a diameter of 15 cm and a stroke 20cm. the clearance volume is 10 percent of the swept volume. Determine the compression ratio and the air standard efficiency of the engine if the cut-off takes place at 6 percent of the stroke.

Solution:

$$V_d = \frac{\pi}{4} . B^2 . L = \frac{\pi}{4}(15)^2 . 20 = 3540 \text{ cm}^3$$

$$\therefore V_c = 0.1 \times V_d = 0.1 \times 3540 = 354 \text{ cm}^3$$

Total cylinder volume $V_1 = V_c + V_d = 3540 + 354 = 3894 \, \text{cm}^3$

$$\therefore \varepsilon = \frac{V_1}{V_2} = \frac{V_1}{V_c} = \frac{3894}{354} = 11$$

cut-off ratio, $\rho = \frac{V_3}{V_2} = \frac{V_2 + (V_3 - V_2)}{V_2} = \frac{354 + 0.06 \times 3540}{354} = 1.6$

$$\therefore (\eta_t)_{Diesel} = 1 - \frac{1}{\varepsilon^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}\right] = 1 - \frac{1}{11^{0.4}}\left[\frac{1.6^{1.4}-1}{1.4(1.6-1)}\right] = 0.57 = 57\%$$

3.2.3 Dual Cycle or Limited Pressure Cycle

This cycle consists of five processes as shown in the (P-V) and (T-S) diagrams respectively. In this cycle part of heat addition is at constant volume and remainder at constant pressure. The remaining processes are same as in Otto or Diesel cycles. The first law of thermodynamic is applied to each process.

Process (1-2) Isentropic Compression Process

$$\Delta q_{1-2} = \Delta u_{1-2} + \Delta w_{1-2}$$

Where Δq heat transfer to the wall kJ/kg.

Δu Change in the internal energy kJ/kg.

Δw Work down to the system kJ/kg.

Since this process is adiabatic hence, there is no heat transfer to the cylinder walls and the work down during compression is :

$$\Delta w_{1-2} = w_c = \Delta u_{1-2} = c_v (T_2 - T_1)$$

For adiabatic process

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

Process (2-3) Heat Addition at Constant Volume

$$Q_{2-3} = m_m . \Delta q_{2-3}$$

$$\Delta q_{2-3} = q_{in} = \Delta u_{2-3} + \Delta w_{2-3}$$

At constant volume there is no work down, hence equation above became

$$\Delta q_{2-3} = c_v (T_3 - T_2)$$

$$Q_{2-3} = Q_{in} = m_f Q_{HV} \eta_c = m_m c_v (T_3 - T_2)$$

$$\therefore m_f Q_{HV} \eta_c = (m_a + m_f) c_v (T_3 - T_2) \qquad \div m_f$$

$$\rightarrow Q_{HV} \eta_c = (AF + 1) c_v (T_3 - T_2)$$

Process (3-4) Heat Addition at Constant Pressure

$$Q_{3-4} = m_m . \Delta q_{3-4}$$

$$\Delta q_{3-4} = q_{in} = \Delta u_{3-4} + \Delta w_{3-4}$$

At constant pressure there is no work down, hence equation became

$$\Delta q_{3-4} = c_p (T_4 - T_3)$$

Another important parameters are the cut-off ratio which is

Equal to $\rho = \dfrac{V_4}{V_3} = \dfrac{T_4}{T_3}$

And Degree of pressure increase $\nu = \dfrac{P_3}{P_2} = \dfrac{T_3}{T_2}$

Process (4-5) Isentropic Expansion Process

$$\Delta q_{4-5} = \Delta u_{4-5} + \Delta w_{4-5}$$

Since this process is adiabatic hence, there is no heat transfer to the cylinder walls and the work down during expansion is:

$$\Delta w_{4-5} = \Delta u_{4-5} = c_v (T_4 - T_5)$$

Also for adiabatic expansion;

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5} \right)^{\gamma - 1} = \left(\frac{\rho . V_3}{V_1} \right)^{\gamma - 1} = \left(\frac{\rho}{\varepsilon} \right)^{\gamma - 1}$$

Process (5-1) Heat Rejection Process

$$Q_{out} = Q_{5-1} = m_m . \Delta q_{5-1}$$

$$\Delta q_{5-1} = q_{out} = \Delta u_{5-1} + \Delta w_{5-1}$$

At constant volume there is no work down, hence equation above became

$\Delta q_{5-1} = c_v (T_5 - T_1)$

The thermal efficiency of dual cycle is given by;

$$(\eta_t)_{Diesel} = \frac{q_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}}$$

$$= 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_v(T_5 - T_1)}{c_v(T_3 - T_2) + c_p(T_4 - T_3)} = 1 - \frac{T_1\left(\frac{T_5}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right) + \gamma.T_3\left(\frac{T_4}{T_3} - 1\right)}$$

$$= 1 - .\frac{T_1\left(\frac{T_5}{T_1} - 1\right)}{T_2\left(\frac{\nu.T_2}{T_2} - 1\right) + \gamma.T_2.\nu\left(\frac{T_4}{T_3} - 1\right)} = 1 - \frac{1}{\varepsilon^{\gamma-1}}.\frac{\left(\frac{T_4\left(\frac{V_4}{V_5}\right)^{\gamma-1}}{T_1} - 1\right)}{(\nu - 1) + \gamma\nu\left(\frac{\rho.T_3}{T_3} - 1\right)}$$

$$= 1 - \frac{1}{\varepsilon^{\gamma-1}}.\frac{\left(\rho.\nu.T_1.\varepsilon^{\gamma-1}.\left(\frac{\rho}{\varepsilon}\right)^{\gamma-1} - 1\right)}{T_1} \times \frac{1}{(\nu - 1) + \gamma\nu\left(\frac{\rho.T_3}{T_3} - 1\right)}$$

$$= 1 - \frac{1}{\varepsilon^{\gamma-1}}.\frac{(\rho^{\gamma}\nu - 1)}{(\nu - 1) + \gamma\nu(\rho - 1)}$$

Ex5: Drive an expression for the efficiency of dual cycle in terms of ε, ρ, ν

Ex6: a dual combustion cycle has an adiabatic compression ratio of 15:1. The conditions at the commencement of compression are 1 kg f/cm^2, 25 c^o and 0.1 m^3. The maximum pressure of the cycle is 65 kg f/cm^2 and the maximum temperature of the cycle is 1500 c^o. if cv=0.17 and $\gamma = 1.4$, calculate the pressure, volume and temperature at the corners of the cycle and the thermal efficiency of the cycle.

Solution:

$P_1 = 1$ kg f/cm^2, $T_1 = 25 + 273 = 298K$, $V_1 = 0.1m^3$

$$V_2 = \left(0.1/15\right) = 0.0660m^3$$

$$P_2 = P_1 \times \left(V_1/V_2\right)^{\gamma} = 1 \times (15)^{1.4} = 44.2kgf/cm^2$$

Lecture No: 4 (COMBUSTION)

4.1 Introduction

It's important to provide a useful starting point for a definition of combustion as"rapid oxidation generating heat, or both light and heat". For our purpose, we will restrict the definition to include only rapid oxidation portion. Combustion transforms energy stored in chemical bonds of fuel to heat that cam be utilized in a variety of ways. There are many thousands of different hydrocarbon fuel components which consist mainly of hydrogen and carbon but may also contain oxygen. Nitrogen, and or sulfur, etc. the maximum amount of chemical energy that can be released (heat) from the fuel is when it reacts (combusts) with a **stoichiometric** amount of oxygen. Stoichiometric oxygen (sometimes called theoretical oxygen) is just enough to convert all carbon in fuel to CO_2 and all hydrogen to H_2O with no oxygen left over. The balanced chemical equation of the simplest hydrocarbon fuel, methane CH_4 burning with stoichiometric oxygen is:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

It takes two moles of oxygen to react with 1 moles of fuel and this gives one mole of carbon dioxide and two moles of water vapor. If isooctane the fuel component, the balanced stoichiometric combustion with oxygen would be:

$$C_8H_{18} + 12.5O_2 \rightarrow 8CO_2 + 9H_2O$$

One kg mole of a substance has a mass in kilogram equal in number to the molecular weight (molar mass) of that substance

$m = N.M$ Where:

m: mass

N: number of moles.

M: molecular weight.

The components of the left side of a chemical reaction equation which are present before the reaction are called reactants, while the components on the right side of the equation are called products or exhaust. Air is used as the source of oxygen to react with fuel. On the **mole basis** atmospheric air is made up about:

78 % nitrogen by mole, 21 % oxygen & 1% argon. To simplify calculations without causing any large error, the neutral argon in air is assumed to be combined with neutral nitrogen, and atmospheric air then can be modeled as being made up **of 21% oxygen and 79% nitrogen by volume or by mole.** For example stoichiometric combustion of methane with air is:

$$CH_4 + 2O_2 + 2(3.76)N_2 \rightarrow CO_2 + 2H_2O + 2(3.76)N_2$$

And isooctane with air is:

$$C_8H_{18} + 12.5O_2 + 12.5(3.76)N_2 \rightarrow 8CO_2 + 9H_2O + 12.5(3.76)N_2$$

On the **mass basis** atmospheric air is made up about:

77 % nitrogen by mole, 23 % oxygen.

Combustion can occur, within limits, when more than stoichiometric air is present (lean) or (weak) but when less than stoichiometric air is present (rich) for a given amount of fuel. For example if methane is burned with 150 % stoichiometric air, the excess oxygen ends up in the products:

$$CH_4 + 3O_2 + 3(3.76)N_2 \rightarrow CO_2 + 2H_2O + 3(3.76)N_2 + O_2$$

if isooctane is burned with 80% stoichiometric air, there not enough oxygen to convert all the carbon to CO_2 and carbon monoxide CO is found in the products:

$$C_8H_{18} + 10O_2 + 10(3.76)N_2 \rightarrow 3CO_2 + 9H_2O + 5CO + 10(3.76)N_2$$

Carbon monoxide CO is colorless, odorless, and poisonous gas which can be further burned to form CO_2. **It's formed in any combustion process when there is deficiency of oxygen**. The unburned fuel ends up as pollution in the exhaust of the engine. For actual combustion in an engine, the equivalence ratio is a measure of the fuel-air mixture relative to stoichiometric conditions. It's defined as:

$$\phi = \frac{(m_f / m_a)_{act}}{(m_f / m_a)_{stoich}}$$

Combustion

Lean (weak) $\phi < 1$	Stoichiometric $\phi = 1$	Rich $\phi > 1$
Products of combustion	Products of combustion	Products of combustion
CO_2, H_2O, N_2, O_2	CO_2, H_2O, N_2	CO_2, H_2O, N_2, CO

4.2 Examples:

Ex1: Calculate the stoichiometric (A/F) ratio for the combustion of sample of dry anthracite of the following composition by mass:

C : 90 % , H_2 : 3 % **,** O_2 : 2.5 %, N_2 : 1 %, S : 0.5 % , ash : 3 %. Also determine the (A/F) ratio and the dry and wet analysis of the products of combustion by volume when 20 % excess air is supplied.

Ex2: The analysis of supply of coal gas is:

H_2 : 49.4 % , CO : 18 % , CH_4 : 20 % , C_4H_8 : 2.5 % , O_2 : 0.4 % ,N_2: 6.2 %

CO_2 : 4 %. Calculate the stoichiometric (A/F) ratio find also the wet and dry analysis of the products of the combustion if the actual mixture is 20 % lean

EX3: Mixture of heavy fuel oil (black fuel) and air is supplied to the boiler. If the heavy fuel consists of:

C : 86.1 % , H_2 : 12.7 % **,** O_2 : 1.2 %, N_2 : 0.5 %. Determine (A/F) ratio and products of combustion when:

a- stoichiometric reaction

b- 20% weak

c- 10% rich

Ex4: Liquid fuel (C_7H_{16}) is burned with 10 % excess air assuming complete combustion calculate mass of air supplied per kg of fuel and the volumetric analysis of dry products of combustion.

Ex5: A liquid fuel was found to contain the following proportions by weight of combustible elements: C : 84.5 % , H_2 : 14 % .its burnt completely with 22 times its weight of air**.** Determine the probable percentage analysis by weight of the product of combustion including ash.

Ex6: The coal supplied to a boiler has the following analysis carbon 84 percent, hydrogen 6 percent and reminder ash. When the rate of combustion is 3.650 kg of copal per hour, the air supplied is 205 m^3/sec measured at 14.7 N/m^2 and 85 ^oc. Calculate (a) the percentage excess air supplied. (b) the percentage analysis by volume of the dry flue gases assuming that the combustion is complete. Air contains 23.1% oxygen.

Ex7: A gaseous fuel has the following analysis by volume: H_2 : 52 % , CO : 16 % , CH_4 : 20 % , O_2 : 2 % ,N_2: 7 % , CO_2 : 3 % . Determine:

(a) The volume of air required for complete combustion of 10 m^3 of gas

(b) The volumetric analysis of the dry products of combustion with 30 percent excess air.

Air contains 20.9 percent by volume of O_2.

Ex8: In the carburetor of an engine fuel of composition C_6H_6 is mixed with air and after wards burned in the engine. Calculate (a) the weight of air required per kg of fuel for complete combustion. (b) if the amount of air supplied per kg of fuel is 5% less than in (a) find (A/F) ratio and the analysis of products by volume.

Ex9: The dry exhaust gases from a petrol engine contain 2 % by volume of CO and negligible O_2. If the fuel contains 84 % C, and 16 % H_2 by weight, all hydrogen is burned find:

(a) The weight of air supplied per kg of fuel.

(b) The proportion of C burnt to CO_2.

Ex10: During a boiler trial a petrol analysis of the flue gases showed 13.2 % CO_2 , 3.2 % O_2 by volume. Some CO was probably present but its percentage was not measured. The analysis of the coal burnt was 88 % C , 4.4 % H_2 , and 7.6 % ash. The moisture in the fuel was nil. Assuming all the carbon and hydrogen have been burnt, estimate:

(a) The complete volumetric analysis of the composition of the dry flue gases. (b) The weight of air supplied per kg of fuel.

4.3 CHEMICAL EQUILIBRIEM

If a general chemical reaction is represented by:

$$\upsilon_a A + \upsilon_b B \rightarrow \upsilon_c C + \upsilon_d D$$

Where A and B represent all reactant species, whether one or two or more,

C and D represent all products, regardless of the number,

$\upsilon_a, \upsilon_b, \upsilon_c, and\, \upsilon_d$ represent the stoichiometric coefficients of A,B,C, and D

Equilibrium composition for this reaction can be found if one knowns the chemical constant:

$$K_e = \left[\left(N_C^{\upsilon_c} . N_D^{\upsilon_d}\right) / \left(N_A^{\upsilon_a} . N_B^{\upsilon_b}\right)\right]\left(P / N_t\right)^{\Delta\upsilon}$$

Where $\Delta\upsilon = \upsilon_C + \upsilon_D - \upsilon_A - \upsilon_B$

N_i=number of moles of component i at equilibrium.

N_t= total number of moles at equilibrium.

P= total absolute pressure in unit of atmosphere.

Equilibrium constant for many reactions can be found in thermodynamic textbooks or chemical handbooks, tabulated in logarithmic form. Equilibrium constant very dependent on temperature, changing many orders of magnitude over the temperature range experienced in an IC engine.

4.4 COMBUSTION TEMPERARURE

Heat liberated in combustion reaction of a hydrocarbon fuel with air is the difference between the total enthalpy of the products and the total enthalpy of the reactants. This is called heat of reaction, heat of combustion, or enthalpy of reaction and is given by;

$$Q = \sum_P N_i h_i - \sum_R N_i h_i$$

where N_i= number of moles of component i

$h_i = (h^o{}_f)_i + \Delta h_i$

$(h^o_f)_i$ & Δh_i can be found in most thermodynamics textbooks.

Q will be negative, meaning that heat is given up by the reacting gases. Q_{HV} is the negative of the heat of reaction for one unit of fuel, and thus it's a positive number. It's calculated assuming both reactants and products are at 25°c. Care must be taken when using heat values. Higher heating value is used when water in the exhaust products is in the liquid state, and lower heating value is used when water in products is vapor. The difference is the heat of vaporization of the water:

$$Q_{HHV} = Q_{LHV} + \Delta h_{vap}$$

For engine analysis, lower heating value is the logical value to use.

4.5 ADIABATIC FLAME TEMPERRATURE

An estimation of the maximum temperature reached in an IC engine can be obtained by calculating the adiabatic flame temperature of the input air-fuel mixture. This is done by setting Q=0

$$Q = \sum_P N_i h_i - \sum_R N_i h_i$$

$$\rightarrow 0 = \sum_P N_i h_i - \sum_R N_i h_i$$

$$\therefore \sum_P N_i h_i = \sum_R N_i h_i$$

Assuming that the inlet conditions of the reactants are known, its necessary to find the temperature of the products such that this equation will be satisfied. Adiabatic flame temperature is the ideal theoretical maximum temperature that can be obtained for a given fuel and air mixture. The actual peak temperature in an engine cycle will be several hundred degrees less than this. There is some heat loss even in the very short time of one cycle, combustion efficiency is less than 100% and some components dissociate at the high engine temperatures. All theses factors contribute to make the actual peak engine temperature less than adiabatic flame temperature.

4.6 SOME COMMON HYDROCARBON COMPONETS

Carbon atoms form four bonds in molecular structures while hydrogen has one bond. A number of different families of hydrocarbon molecules have been identified a few of the more common ones are describe:

1. Paraffins (allkane) $\mathbf{C_nH_{2n+2}}$ CH_4 , C_3H_8
2. Olefins $\mathbf{C_nH_{2n}}$ C_2H_4 , C_3H_6
3. Diolefins $\mathbf{C_nH_{2n-2}}$ C_7H_{12} , C_8H_{14}
4. Cycloparaffins $\mathbf{C_nH_{2n}}$ C_4H_8 , C_5H_{10}
5. Aromatics $\mathbf{C_nH_{2n-6}}$ C_6H_6 , C_7H_8

Lecture No: 5 (MIXTURE PREPARATION IN S.I.E.)

5.1 CARBURETION

In the SI engine a combustible fuel-air mixture is prepared outside the engine cylinder. The process of preparing this mixture is called carburetion. This complicated process is achieved in the induction system, which is shown in **Fig.(5.1)**. The carburetor, a device which atomises the fuel and mix it with air. During the suction stroke vacuum is created in the cylinder which causes the air to flow through the carburetor and the fuel to be sprayed from the fuel jets.

Four important factors which significantly affect the process of carburetion are:

1. The time available for the preparation of the mixture.
2. The temperature of the incoming air of the intake manifold.
3. The quality of fuel supplied.
4. The design of the induction system and combustion chamber.

5.1.1 A SIMPLE OR ELEMENTARY CARBURETOR

Fig.(5.2) shows a simple carburetor. It consists of a float, float chamber, nozzle with metering orifice, venture and throttle valve. The float and a needle valve system maintains a constants height of petrol in the float chamber. If the amount of fuel in the float chamber below the designed level, the float lowers, there by opening the of fuel supply valve. When the designed level has been reached, the float closes the needle valve, thus stopping additional fuel flow from the supply system.

5.1.2 COMPLETE CARBURETOR

In order to satisfy the demands of an engine under all conditions of operation the following additional systems are added to the simple carburetor:

1. Main Metering System: the main metering system of a carburetor is designed to supply a nearly constant basic fuel-air ratio over a wide range of speeds and loads. This mixture corresponds approximately to best economy at full throttle (A/F ratio ≈ 15.6). Since a simple carburetor tends to enrich the mixture at high speeds automatic compensating devices are incorporated in the main metering system to correct this tendency. These devices are:

(a) Use of a compensating jet that allows an increasing flow of air through a fuel passage as the mixture flow increases.

(b) Use of emulsion tube for air bleeding. In this device the emphasis is on air bleeding alone.

(c) Use of tapered metering pin that is arranged to be to be moved in and out of the main or auxiliary fuel orifice either manually or by means of some automatic mechanism changing the quantity of fuel drawn into the air charge.

(d) Changing the position of jet in the venturi. The suction action is highest at the venture throat, therefore, by raising the venturi, the nozzle relatively moves to points with smaller suction and the flow of fuel is decreased.

(e) Use of an auxiliary air valve or port that automatically admits additional air as mixture flow increases.

2. Idling System: An example of idling jet are shown in **Fig.(5.3)**. It consists of a small fuel line from the float chamber to a point a little on the engine side of the throttle. This line contains a fixed fuel orifice. When the throttle is practically closed, the full manifold suction operates on the outlet of this jet. In addition, the very high velocity past the throttle plate increases the suction locally.

3. Power Enrichment or Economizer System:

Such a device provides a rich uneconomical mixture at high load demand without interring with economical operation in the normal power range. Its provides a large orifice opening to the main jet as the throttle is opened beyond a certain limit as shown in **Fig.(5.4)**

4. Acceleration Pump System: when its desire to accelerate the engine rapidly, a simple carburetor will not provide the required rich mixture. Rapid opening of the throttle will be immediately followed by an increased air flow, but the inertia of the liquid fuel will cause at least a momentarily lean mixture just when richness is desired for power. To overcome this deficiency an acceleration pump is provide which is shown in **Fig.(5.5)**. The pump consists of a spring-loaded plunger. A linkage mechanism is provided so that when the throttle is rapidly open the plunger moves into the cylinder and forces an additional jet of fuel into venturi. The plunger is raised again against the spring force when the throttle is partly closed.

5. Choke: During cold starting period, at low cranking speed and before the engine has warmed up, mixture much richer than usual mixture must be supplied simply because large fraction of the fuel will remain liquid even in the cylinder and only the vapour fraction can provide a combustible mixture with the air. The most common means of obtaining this rich mixture is by the use of chock, which is a butterfly type of valve placed between the entrance to the carburetor and the venturi throat as shown in **Fig.(5.6)**. By partially closing the chock, a large pressure drop can be produced at the venturi throat than would normally result from the amount of air flowing through the venturi. This strong suction at the throat will draw large quantities of fuel from the main nozzle and supply a sufficiently rich mixture.

5.1.3 CARBURETOR TYPES

There are basically two types of carburetors, open choke and constant vacuum type. In the former, the main air orifice, known as the choke tube or venturi, is of fixed dimensions, and metering is effected by varying the pressure drop across it. In the case of constant vacuum type the area of air passage is varied automatically, while the pressure drop is kept approximately constant. The important examples of open choke type are Zenith, Solex, Carter and Stromberg S.U. carburetor is of constant vacuum type. Carburetors may be up draught, horizontal, and down drought [see Fig.(5.7a,b, and c) respectively. The down draught has the advantage that the mixture is assisted by gravity in its passage in to the engine induction tract, and at the same time the carburetor is usually reasonably accessible.

5.1.4 DESCIPTION OF SOME TYPES OF CARBURETORS

1. Solex Carburetor: The solex carburetor is famous for easy starting, good performance, and reliability. It’s used in Fiat and Jeep Cars. **Fig.(5.8)** shows a schematic arrangement of a solex carburetor. The solex carburetor has a conventional float in the float chamber. For normal running, the fuel is provided by the main jet and the air by the choke tube or venturi. The unique feature of this carburetor is the Bi-starter for cold starting. The starter valve is in the form of a flat disc with holes of different sizes. These holes connect the starter petrol jet and starter air jet sides to the passage which opens into a hole just below the throttle valve. Depending upon the position of the starter lever either bigger or smaller holes come opposite the passage. The starter lever, which rotates the starter valve, is operated from the dashboard control by means of a flexible cable.

2. S.U. Carburetor: This carburetor is an example of constant vacuum type with automatically variable choke, which differs completely from them being constant choke type. **Fig.(5.9)** shows the diagrammatic view of the basic components of a horizontal S.U. carburetor. The carburetor has a conventional system of float chamber which feeds fuel into a vertical channel in which is situated the jet sleeve. The sleeve bears a number of holes in its side so that the will enter the sleeve and thus stand at the same level as in the float chamber. The unique feature of the S.U. carburetor is that it has only one jet. There is no separate idling jet or acceleration pump. Since a constant high air velocity across the jet is maintained even under idling condition, the necessity for a separate idling jet is obviated. The S.U. carburetor is used in many British cars.

5.1.5 CALCULATION OF THE AIR-FUEL RATIO FOR A SIMPLE CARBURETOR

The air fuel ratio will be calculated for a simple carburetor with fuel nozzle tip h meters above the fuel level in the float chamber as shown in **Fig.(5.10)**. The air from atmosphere is induced through the carburetor by the pressure difference across it when the piston moves on its induction stroke. The velocity of air increased as it passes through the venturi and reaches maximum at the venturi throat as it is the minimum area in the induction passage. The pressure also changes and is minimum at the venturi throat. The density of air is not same at the inlet to the carburetor (point1) and venturi throat (point2). The calculation of exact air mass flow involves taking this density change or compressibility of the air into account.

Let

$A_1 =$ Air flows area at inlet, m^2

$A_2 =$ Air flow area at venturi throat, m^2

$A_f =$ Area of the fuel nozzle, m^2

$C_1 =$ Air velocity at inlet, m/s.

$C_2 =$ Air velocity at venturi throat, m/s.

$C_f =$ Fuel velocity at discharge from nozzle, m/s.

$m_a =$ mass flow of air, kg/s.

$m_f =$ mass flow of fuel, kg/s.

$p_1 =$ air pressure at inlet (atmospheric pressure) kgf/m^2.

$p_2 =$ air pressure at throat (atmospheric pressure) kgf/m^2.

$h_1 =$ specific enthalpy of air at inlet kcal/kg.

$h_2 =$ specific enthalpy of air at venturi kcal/kg.

$T_1 =$ absolute temperature of air at inlet K.

$T_2 =$ absolute temperature of air at venturi throat K.

$v_1 =$ specific volume of air at inlet m^3/kg.

$v_2 =$ specific volume of air at venturi throat m^3/kg.

$\rho =$ density of air kg/ m^3

$\rho_f =$ density of fuel kg/ m^3

$Cd_a =$coefficient of discharge of venturi

$Cd_f =$coefficient of discharge for fuel orifice

$h =$height of nozzle tip above fuel level

Calculation: Air-Flow Exact Calculation

Applying the steady flow energy equation to entrance (section 1) and venturi throat (section 2) for 1 kg of air flow.

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

where q and w are work and heat transfer respectively

$q = 0, w = 0 \; \& \; C_1 = 0$

$C_2 = \sqrt{2gJ(h_1 - h_2)}$ Substituting $h_1 = c_p T_1 \; \& \; h_2 = c_p T_2$

$$C_2 = \sqrt{2gJc_p T_1\left(1 - \frac{T_2}{T_1}\right)}$$

As the flow process between the atmospheric and venturi throat is isentropic;

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore C_2 = \sqrt{2gJc_p T_1\left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)} = C_2 = \sqrt{2gJ\frac{\gamma}{\gamma-1}RT_1\left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)}$$

$$C_2 = \sqrt{2gc_p T_1\left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)} \qquad Jc_p = R\left(\frac{\gamma}{\gamma-1}\right)$$

Now the mass flow of air is constant from inlet to venturi throat and is given by:

$$m_a = A_1 C_1 \rho_1 = A_2 C_2 \rho_2$$

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

$$v_2 = v_1 \left(\frac{p_1}{p_2}\right)^{\left(\frac{1}{\gamma}\right)} = \frac{RT_1}{p_1}\left(\frac{p_1}{p_2}\right)^{\left(\frac{1}{\gamma}\right)}$$

$$m_a = \frac{A_2 p_1}{RT_1}\left(\frac{p_2}{p1}\right)^{\left(\frac{1}{\gamma}\right)} \sqrt{2gJc_p T_1\left(1-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right)}$$

$$= \frac{A_2 p_1}{RT_1}\sqrt{2gJc_p T_1\left(\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right)}$$

Substituting g=9.81, J=427, c_p=0.24 R=29.27 and $\gamma = 1.4$

$$m_a = 1.53\frac{A_2 p_1}{\sqrt{T_1}}\sqrt{\left(\left(\frac{P_2}{P_1}\right)^{1.43} - \left(\frac{P_2}{P_1}\right)^{1.71}\right)} = 1.53\frac{A_2 p_1}{\sqrt{T_1}}\phi \qquad where$$

$$\phi = \sqrt{\left(\left(\frac{P_2}{P_1}\right)^{1.43} - \left(\frac{P_2}{P_1}\right)^{1.71}\right)}$$

The above equation gives theoretical mass flow. The actual flow is obtained by multiplying it by the coefficient of discharge of the venturi

$$\therefore (m_a)_{actual} = C_d(1.53 * \frac{A_2 p_1}{\sqrt{T_1}}\phi)$$

C_d & A_2 are constant for a given venturi hence;

$$\therefore (m_a)_{actual} = const * \frac{p_1}{\sqrt{T_1}}\phi$$

An alternative expression for $(m_a)_{actual}$ is:

$$(m_a)_{actual} = C_d A_2 p_1 \sqrt{2gJc_p T_1 \left(\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}} \right)}$$

Fuel Flow Calculation:

Now to find the air-fuel ratio, we have to calculate fuel flow. As the fuel is incompressible, applying Bernoulli's theorem, we get:

$$\frac{p_1}{\rho_1} = \frac{p_2}{\rho_f} + \frac{C_f^2}{2g} h \qquad or$$

$$C_f = \sqrt{2g\left(\frac{p_1 - p_2}{\rho_f}.h\right)}$$

$$Now, m_f = A_f C_f \rho_f$$

$$\& \left(m_f\right)_{actual} = C_{df} A_f \sqrt{2g\rho_f \left(p_1 - p_2 - h\rho_f\right)}$$

Air fuel ratio:

Thus the air-fuel ratio, m_a/m_f is given by:

$$\frac{m_a}{m_f} = 1.53 \times \frac{C_d}{C_{df}} \times \frac{A_2}{A_f} \times \frac{p_1.\phi}{\sqrt{2g\rho_f \left(p_1 - p_2 - h\rho_f\right)}} \qquad \text{or}$$

$$\frac{m_a}{m_f} = \frac{p_1.\sqrt{\left(\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}} \right)}}{\sqrt{T_1\left(p_1 - p_2 - h\rho_f\right)}}$$

This is the exact expression taking into account the effect of compressibility (variation of density) of air. However, the effect of compressibility of air is small for the velocities of air encountered.

Calculation of air-fuel ratio when neglecting compressibility. When air is considered is compressible, Bernoulli's theorem is applicable to air flow also Hence assuming C_1=0 we get:

$$\frac{p_1}{\rho_a} = \frac{p_2}{\rho_a} + \frac{C_2^2}{2g} \quad or$$

$$C_2 = \sqrt{2g\left(\frac{p_1 - p_2}{\rho_a}\right)}$$

$$Now, m_a = A_2 C_2 \rho_a$$

$$\& \left(m_a\right)_{actual} = C_d A_2 \sqrt{2g\rho_a\left(p_1 - p_2\right)}$$

$$\frac{m_a}{m_f} = \frac{C_d}{C_{df}} \times \frac{A_2}{A_f} \times \sqrt{\frac{\rho_a}{\rho_f} \times \frac{p_1 - p_2}{p_1 - p_2 - h.\rho_f}} \quad if \quad h = 0 \rightarrow$$

$$\frac{m_a}{m_f} = \frac{C_d}{C_{df}} \times \frac{A_2}{A_f} \times \sqrt{\frac{\rho_a}{\rho_f}}$$

$$\frac{m_a}{m_f} \alpha \sqrt{\frac{\rho_a}{\rho_f}}$$

5.1.6 Examples

Ex1: A simple jet carburetor is required to supply 6 kg of air per minute and 0.45 kg of fuel of density 740 kg/m^3. The air is initially at 1.03 kgf/cm^2 and 27 ^{o}C. Calculate the throat diameter of the chock for a flow velocity of 92 m/s. velocity coefficient =0.8. if the pressure drop across the fuel metering orifice is 0.75 of that of choke, calculate orifice diameter assuming C_d =0.06

Solution:

$$C_2 = C_d \sqrt{2gcJc_pT_1\left[1-\left(\frac{P_2}{P_1}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right]}$$

$$92 = 0.8\sqrt{2\times 9.81\times 427\times 0.24\times 300\left[1-\left(\frac{P_2}{P_1}\right)^{0.4/1.4}\right]}$$

$$\rightarrow 1-\left(\frac{P_2}{P_1}\right)^{0.286} = \left(\frac{92}{0.8}\right)^2\times\frac{1}{2\times 9.81\times 427\times 0.24\times 300}$$

$$\rightarrow \left(\frac{P_2}{P_1}\right)^{0.286} = 0.978 \qquad \therefore \frac{P_2}{P_1} = 0.925$$

$$P_2 = 1.03\times 0.925 = 0.953 \qquad kgf/cm^2$$

$$P_1v_1^{\gamma} = P_2v_2^{\gamma}$$

$$v_2 = v_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{1.4}} = \frac{RT_1}{P_1}\left(\frac{1}{0.925}\right)^{\frac{1}{1.4}} = \frac{29.27\times 300}{1.03\times 10^4}(1.08)^{1/1.4}$$

$$= 0.825\times 1.0565 = 0.9 \qquad m^3/kg$$

$$\therefore throatarea \qquad A_2 = \frac{m_a\times v_2}{C_2} = \frac{6\times 0.9}{60\times 92}\times 10^4 = 9.79 \qquad m^2$$

$$\therefore d_2 = 3.52 \qquad cm$$

Pressure drop at venturi $= 1.03 - 0.953 = 0.077 \qquad kgf/cm^2$

Pressure drop at jet $= 0.75\times 0.077 = 0.0577 \qquad kgf/cm^2$ Now

$$m_f = A_fC_f\sqrt{2g\rho_f\left(P-P_2\right)} = A_fC_f\sqrt{2\times 9.81\times 0.74\times 10^3\times 0.0577\times 10^4}$$

$$m_f = A_fC_f\times 2.9\times 10^3 \rightarrow A_f = \frac{m_f}{C_f\times 2.9\times 10^3} = \frac{0.48}{60}\times\frac{1}{0.6}\times\frac{10^4}{29\times 10^3} = 0.0431 \qquad cm^2$$

$$\therefore d_f = 2.34 \qquad mm$$

Ex2: A 10 ×12 cm four cylinder**,** four stroke engine running at 2000 rpm has a carburetor venturi with a 3 cm throat. Determine the suction at throat assuming volumetric efficiency of the engine to be 70 %**.** Assume air density to be 1.2 kg/m^3 and coefficient of air flow 0.8.

Solution:

Stroke volume $= V_d = \frac{\pi}{4}(10)^2 \times 12 \times 10^3 \times 4 = 0.00377 \quad m^3$

Actual volume $= \eta_v \times 0.00377 = 0.7 \times 0.00377$

Actual volume sucked/second $= V_d \times \frac{N}{n} \times \frac{1}{60} = 0.044 \quad m^3$

$$= 0.7 \times 0.00377 \times \frac{2000}{2} \times \frac{1}{60} = 0.044 \quad m^3/s$$

Mass of air/s $= 0.044 \times 1.2 = 0.0528 \quad kg/s$

As the initial temperature and pressure is not given, the problem is solved by approximate method (neglecting compressibility of the air).

$$m_a = C_d \times A_2 \sqrt{2g\rho_a \Delta P_a}$$

$$0.0528 = 0.8 \times \frac{\pi}{4}(3)^2 \times 10^{-4} \sqrt{2 \times 9.81 \times 1.2 \times \Delta P_a}$$

$$\therefore \Delta P_a = \left(\frac{0.0528 \times 4}{0.8 \times \pi \times 9 \times 10^{-4}}\right)^2 \times \frac{1}{2 \times 9.81 \times 1.2} \times 10^{-4} = 0.0344 \quad kgf/cm^2$$

Ex3: an experimental four stroke petrol engine of 1710 cm^3 capacity is to develop maximum power at 5400 rpm. The volumetric efficiency at this speed is assumed to be 70 percent and the air-fuel is 13:1. Two carburetors are to be fitted and it is expected that at peak power the air speed at the choke will be 107 m/s. the coefficient of discharge for the venturi is assumed to be 0.85 and that of main petrol jet 0.66. An allowance should be made for the emulsion tube, the diameter of which can be taken as ½.5 of the choke diameter. The petrol surface is 6mm below the choke at this engine condition. Calculate the sizes of a suitable choke and main jet. The specific gravity of petrol is 0.75. Atmospheric pressure and temperature are 1.03 kgf/cm^2 and 27 ^{o}c respectively.

Solution:

Whenever initial temperature and pressure of air is given, the problem is solved by taking compressibility of air into account (exact method).

Volumetric efficiency $\eta_v = \frac{V_a}{V_d} \rightarrow V_a = \eta_v \times V_d = 0.7 \times 1710 = 1197 cm^3$

Actual volume sucked/second $= V_a \times \frac{5400}{2} \times \frac{1}{60} = 0.0539 m^3$

Volume of air through each carburetor, $V_1 = \frac{0.0539}{2} 0.02695\ m^3/s$

Mass flow of air, $m = \frac{pv}{RT} = \frac{1.03 \times 10^4 \times 0.02695}{29.27 \times 300} = 0.0316\ kg/s$

The air velocity at throat C_2, assuming initial velocity is zero is given by floe equation: $C_2 = \sqrt{2g(h_1 - h_2)} = \sqrt{2gc_p(T_1 - T_2)} = \sqrt{2gc_p T_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\lambda}}\right)}$

$$\therefore 107 = \sqrt{2 \times 9.81 \times 427 \times 0.24 \times 300 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{1.4-1}{1.4}}\right)}$$

$$\therefore \frac{p_2}{p_1} = 0.9495 \rightarrow pressure\ \ at\ throat = 0.9495 \times 1.03 = 0.979\,\text{kgf/cm}^2$$

now

$$p_1 v_1^{\gamma} = p_2 v_2^{\gamma} \therefore volume\ flow\, at\ choke$$

$$V_2 = 0.02895 \times \left(\frac{1}{0.9495}\right)^{1/1.4} = 0.0279 m^3/s$$

Ex4: the venturi of a simple carburetor has a throat diameter of 20 mm and the coefficient of air flow is 0.85. The fuel orifice has a diameter of 1.25 mm and the coefficient of the fuel flow is 0.66. The petrol surface is 5 mm below the throat. Find (a) The air-fuel ratio for a pressure drops of 0.07 N/m^2 when the nozzle tip is neglected. (b) The air-fuel ratio when the nozzle tip is taken into account (c) the minimum velocity of air or critical air velocity required to start the fuel flow when nozzle tip is provided. Take density of air and fuel as 1.2 and 750 kg/m^3 respectively.

Ex5: a carburetor in which the float chamber is vented to atmosphere is tested in the factory without an air cleaner. The main metering system of the carburetor is found to give an air-fuel ratio of 14 at the sea level conditions. The pressure drop at the venturi throat is 0.85 N/m^2. The atmospheric pressure is 1 N/m^2. The carburetor is tested again when an air cleaner is fitted at the inlet of the carburetor. The pressure drop to air cleaner is found to be 30 mm of mercury when the air flow at sea level condition is 250 kg per hour. Assuming zero nozzle lip and constant coefficient of flow, calculate (a) the throat pressure when the air cleaner is fitted, and (b) air-flow ratio with air cleaner fitted.

Ex6: determine the air/fuel ratio supplied at 5000 m altitude by a carburetor which is adjusted to give an air fuel ratio of 14:1 at sea level where air temperature is 27 C and pressure 1.03 N/m^2.

Ex7: find the maximum possible air velocity and corresponding throat area of the choke tube for a carburetor with a jet area of 2 mm^2 used on a petrol engine consuming 5 kg of petrol per hour. The atmospheric pressure is 1.03 N/m^2 and temperature 27 C. the fuel-air ratio by weight is 1:14, assume C_d =0.8

5.2 PETROL INJECTION

Some of the above disadvantages of a carburetor may be avoided by introducing the fuel by injection rather than carburetion. In petrol injection the fuel may be injected during the suction stroke in the inlet manifold at low pressures. There are two important methods of petrol injection which are: **(a) continuous injection and (b) timed injection.**

(a) **Continuous Port Injection**: in this system fuel is sprayed at low pressure continuously into the air supply. The amount of fuel is governed by the air throttle opening, increasing as throttle is opened. No timing device is used. For example the G.M.C. fuel injection system supplies fuel continuously to 8 nozzles each located in inlet port and aimed to inlet valve. The operation of such system shown in **Fig.(5.11)** conventional throttle (D) controls the flow of flow of air entering through a large radial entry, annular venturi (E). a very small vacuum is created at the throat of the venturi (M) and increases with increase with increase in air flow. This air metering signal is sent to fuel meter and the fuel pressure is increase with increase in air flow to maintain essentially a constant fuel-air ratio. The continuous injection method has the advantage of promoting efficient atomization of the fuel and uniform pressure and uniform mixture strength in all cylinders.

Also the evaporation effect of the fuel cools the compressed charge and gives a higher volumetric efficiency. This method requires only one fuel injection pump and one injector.

(b) Timed Injection System: the timed injection system is similar to the system used in high speed diesel engines. It's divided to:

1. Low pressure single pump and distributor system: this system employs a single plunger or gear pump delivering fuel at low pressure to a rotating distributor which supplies each cylinder with fuel, in turn at the correct time
2. Multi pump: it employs a pump having a separate plunger for each cylinder and usually a high injection nozzle pressure. A measured quantity of fuel is delivered into each cylinder during the induction stroke, but at definite time and over a definite period in this stroke.

5.2.1 ADVANTAGE OF PETROL INJECTION

The main **advantages** of petrol injection are:

1. Increased volumetric efficiency and hence increased power and torque, due to the absence of any restriction such as venturies and other metering in the air passage.
2. Better distribution of mixture to each cylinder and hence lower specific fuel consumption.

3. Lower mixture temperatures in the engine cylinders, despite the increase in power developed and hence possibility of employing higher compression ratios.

4. Freedom from blowbacks and icing.

5. Better starting and acceleration.

6. Engines fitted with petrol injection system can be used in any tilt position which will cause surge trouble in carburetors.

5.2.2 DISADVANTAGE OF PETROL INJECTION

The **disadvantages** of fuel injection as against conventional carburetion are as follows:

1. Higher initial costs due to complicated component assembly.
2. Increased service problem.
3. Weight of petrol injection system is more than that of carburetors.
4. Injection system generates more noise.

Lecture No: 6 (FUEL INJECTION IN DIESEL ENGINE)

6.1 INTRODUCTION

In diesel engine air is drawn into the cylinder during suction stroke and compressed to a very high pressure, thus raising air temperature to a value required to ignite the injected fuel into the cylinder. Fuel is injected into the cylinder at the end of the compression stroke, thus requiring a high injection pressure. During the process of injection the fuel is broken into a fine spray of very small droplets. These droplets take heat from the hot compressed air. The surfaces of these droplets form vapour, which in turn mixes with air to form a fuel-air mixture.

6.2 REQUIREMENTS OF A DIESEL INJECTION SYSTEM

1. The fuel should be introduced into the combustion chamber within a precisely defined period of the cycle.
2. The amount of the fuel injected per cycle should be metered very accurately.
3. The quantities of the fuel metered should vary to meet changing speed and load requirements.
4. The injected fuel must be broken into very fine droplets i.e. good atomization should be obtained.
5. The beginning and the end of injection should be sharp i.e. there should not be any dribbling or after-injection.
6. The weight and size of fuel injection system must be minimum.

6.3 TYPES OF INJECTION SYSTEM

Diesel injection systems can be divided into two basic types. They are:

(1) Air injection, and (2) Solid injection

6.3.1 Air Injection: the fuel is metered and pumped to the fuel valve by a camshaft driven the fuel pump. The fuel valve is opened by means of mechanical linkage operated by the camshaft which controls the timing of injection. The fuel valve is also connected to a high pressure line fed by a multistage compressor which supplies air at a pressure about 60 to 70 kgf/cm^2. When the fuel valve is opened, the blast sweeps along with it the fuel and a well-atomized fuel spray is sent to the combustion chamber.

The advantages and disadvantages of air injection system are discussed below:

1. Good atomization obtained. A high imep can be attained as rapid combustion results due to good mixing of fuel and air.
2. Heavy and viscous fuels, which are cheaper can also be injected.
3. The fuel pump is required to develop only a small pressure.

Disadvantages

1. It’s complicated and expensive since it’s required a high pressure multistage compressor.
2. Separate mechanical linkage is required to time the operation of the fuel valve.
3. Due to the compressor and the linkage the bulk of engine increase. This also results in reduced (bhp) due to power losses.
4. The fuel in the combustion chamber burns very near to the injection nozzle which many times lead to overheating and burning the valve and its seat
5. The fuel valve seating requires a considerable skill.
6. In the case of sticking of the fuel valve, the system becomes quite dangerous due to the presence of high pressure air.

6.3.2 Solid Injection: injection of fuel directly into the combustion chamber without primary atomization is termed as solid injection. This is also called airless mechanical injection. Every solid injection must have:

(1) A pressurizing unit (the pump) and (2) An atomizing unit (injector).

The different types of solid injection systems vary in the manner of operation and control of these two basic elements. The main types of modern fuel injection systems are:

1. The individual pump and injector (jerk pump system). In this system there is a separate metering and compression pump is used for each cylinder. The pump which metres the fuel also times the injection. A jerk pump is a reciprocating fuel pump which meters the fuel and also furnishes the injection pressure. Jerk pump is universally used for medium and high speed diesel engines.

2. Common rail system. In this system for each cylinder there is a separate metering and timing element which is connect to an automatic injector injecting fuel into the cylinder. Also it's requires that the nozzle for different cylinder must be accurately matched to insure good fuel distribution between various cylinder.

3. Distributor system. Fig.(6.1) shows a schematic diagram of a distributor system. In this system the pump which pressurizes the fuel also meters and times it. The fuel pump after metering the required amount of fuel supplies it to a rotating distributor at the correct time. To supply each cylinder, the number of injection stroke per cycle for the pump is equal to the number of cylinders. The advantages of this system are shown;

Advantage: (1) small cost (2) uniform distribution (3) one metering element

6.4 FUEL PUMP

A large number of fuel pump designs have been developed by various manufactures. It’s not possible to discuss all of them. Therefore only one type of fuel pump will be discussed. Fig.(6.2) shows a complete injection system.

Operation: when the plunger is at bottom of its stroke the fuel flows through the inlet port into the barrel and fills the space above the plunger and also the vertical groove and the space below the helix. When the plunger starts moving up, a certain amount of fuel goes out of the fuel chamber through the ports until the plunger reaches the position (b) and closes the ports. On further upwards movement of the plunger the trapped fuel is compressed and is forced out through the delivery valve to the pipe leading to the injector which immediately injects the fuel into the combustion chamber. The injection process continues till the end of the upward stroke of the plunger when the lower end of the helix uncovers the spill port

6.4 INJECTION NOZZLES

The main requirements of an injector nozzle are as follows:

(1) To inject fuel at sufficiently high pressure so that the fuel enters the cylinder with a high velocity (less penetration).

(2) The penetration should not be high so as to impinge on cylinder walls. This may result in poor staring.

(3) The fuel supply and cut off should be rapid. There should not be any dribbling.

6.4.1 Types of Nozzles

The type of nozzle used is greatly dependent on the type of combustion chamber in use. The mixing of fuel and air depends on the relative velocity between them, which in turn is greatly affected by the nature of the air movement in the combustion chamber. The injection nozzles may be classified as open or close type

Open type: has the fuel orifice, or orifices and the part of the fuel passageway open to the burner or cylinder pressure at all types. Open burner are cheap but less efficient and are rarely used, one example being opposite piston two stroke Junkers diesel engines.

Closed type: the advantages of a closed nozzle as compared to those of an open nozzle lie in its avoidance of pressure drop and in its control of injection pressure also the needle can not be blocked by deposits this nozzle is proffered in practice.

6.5 QUANTITY OF FUEL PER CYCLE

The quantity of fuel injected per cycle depends upon the amount of air available (displacement volume) and the load of the engine. The fuel is supplied into the combustion chamber through the nozzle holes. The velocity of the fuel through nozzle orifice can be given by:

$$V = C\sqrt{2g\Delta P_N}$$

V : velocity of fuel.

C : flow coefficient of orifice

ΔP_N : pressure difference between injection and cylinder pressure

The volume of the fuel injected per second Q is given by:

$$Q = \left(\frac{\pi}{4}d^2 \times n\right) \times V \times \left(\frac{\theta}{360} \times \frac{1}{N}\right) \times \left(\frac{N}{60}\right)$$

d : diameter of orifice, n : number of hole, θ : injection duration

N : number of revolution per minute

Lecture No: 7 (IGNITION)

7.1 INTRODUCTION

Ignition is merely prerequisite of combustion and is considered from the stand point of the beginning of the combustion process that it initiates. Ignition has no degree: either the combustion of the medium is initiated or it is not. Ignition process is intimately connected with the initiation of combustion and not with behavior of the combustion wave. The ignition process must add necessary energy for starting and sustaining burning of the fuel till combustion take place.

7.2 BASIC IGNITION SYSTEMS

The basic ignition systems in use are:

1. Conventional (Battery) ignition system.
2. Magneto ignition system.
3. Electronic ignition system.

Battery and magneto ignition systems differ only in the source of electrical and all other system component being similar.

7.2.1 Conventional (Battery) Ignition System (C.I.S.)

The essential elements of a battery ignition system are shown in fig.(7.). they are battery, ignition switch, ignition coil, battery resistor, distributor housing, the breaker points, cam, condenser, rotor, and advance mechanism, spark plug, and low and high tension wiring.

(1) The coil: The ignition coil consists of two coils one primary and the other secondary. The primary winding is connected to the battery through an ignition switch and the contact breaker. The secondary winding is connected to spark plugs through the distributor. A typical ignition coil has 100-200 number of turns in primary winding and about 20,000 turns in the secondary winding.

(2) The resistor or ballast: a resistor is provided in series with the primary winding to regulate primary current. For starting purposes this resistor is by passed so that more current can flow in the primary circuit.

(3) The condenser: the purpose of the condenser is to interrupt the primary current as quickly as possible and so cause a rapid collapse of the flux field.

(4) The cam: a cam rotating at camshaft speed operates breaker points and causes the breaker points to open and close. When the ignition switch is on and the breaker points are closed, current flows from the battery, through the primary winding and builds up a magnetic field. In the growing process, the magnetic field cuts the primary winding and induced a back e.m.f. which a poises the battery current and therefore delays the building process of the field itself. Thus time (A,B,C) is required to obtain max current and field strength. During the interval of time (AC), the distributor rotor is revolving and approaching a terminal leading to a spark plug. When the points open, the magnetic field collapse with consequent flow of (primary and secondary winding) that charges the capacitance of the two circuits. The voltage raises at the spark plug until it reaches the value that cam breakdown the sparkplug CD. Once the high resistance of the air-gap is overcome and reduced by ionization, the voltage falls as the arc is established. Most electrical energy stored in the magnetic field is dissipated to initiate combustion. When the point open, the field strength falls sharply while the condenser is charging (CE) and discharging (EF).

Advantage of conventional ignition system

1. Simple to maintenance. 2. Its suitable for low and medium speed

3. Inexpensive

7.2.2 Magneto Ignition System (M.I.S.)

Magneto is a special type of electric generator. It's mounted on the engine and replaces all the components of the coil ignition system except the spark plug. A magneto when rotating by the engine is capable of producing very high voltage and does not need a battery as a source of external energy. Fig.(7.) shows the schematic diagram of magneto ignition system.

Its does not require a battery, its used in many air graft engines for racing and industrial engine. In this design the distributor finger is replaced by carbon brash and secondary winding is moved to location near or at spark plug. When the beaker points are open and closed with the cam, primary current in the primary winding is produced which causes high voltage in the secondary circuit.

Table of comparison between **Conventional** ignition system and **Magneto** ignition system

Conventional ignition system	**Magneto** ignition system
1. Battery is required the engine can not start when battery is discharge. 2. Primary current from the battery. 3. Good spark at low speed. 4. Starting of engine is easier. 5. Occupies more space. 6. Mostly employed in petrol cars and buses.	1. No battery is needed and hence no problem of battery discharges. 2. Primary current from the magnetic. 3. Spark is poor at low speed and auxiliary help may be required. 4. Engine starting is rather difficult. 5. Occupies less space. 6.Used in motor cycles and racing cars.