# INTERNAL COMBUSTION ENGINES

## Syllabus

1. Engine classification and engine components.
2. Air standard cycles.
3. Cycles having the Carnot cycle efficiency.
4. Comparison of Otto, Diesel and dual cycles.
5. Fuel- Air cycles; variation of specific heats.
6. Fuel- Air cycles; effect of engine variables.
7. Internal combustion engines fuels.
8. Combustion; basic chemistry.
9. Combustion; stoichiometry.
10. Combustion; exhausts gas analysis.
11. Combustion; Dissociation.
12. Combustion; internal energy, enthalpy of combustion and enthalpy of formation, and calorific value of fuels.
13. Real cycles.
14. Spark ignition engine and its components.
15. Compression ignition engine and its components.
16. Four stroke engine.
17. Two stroke engine
18. Scavenging in two - stroke engines.
19. Rotary engines; Gas turbine.
20. Rotary engines; Wankel engine.
21. Engine testing and performance; performance parameters.
22. Performance characteristics.
23. Testing and basic measurement of I.C. engines.

24. Supercharging and its effect on engine performance.
25. Turbo - charging.
26. I.C.E. fuel systems, carburetion, fuel injection.
27. Valves (ports) timing.
28. Firing order.

## References:

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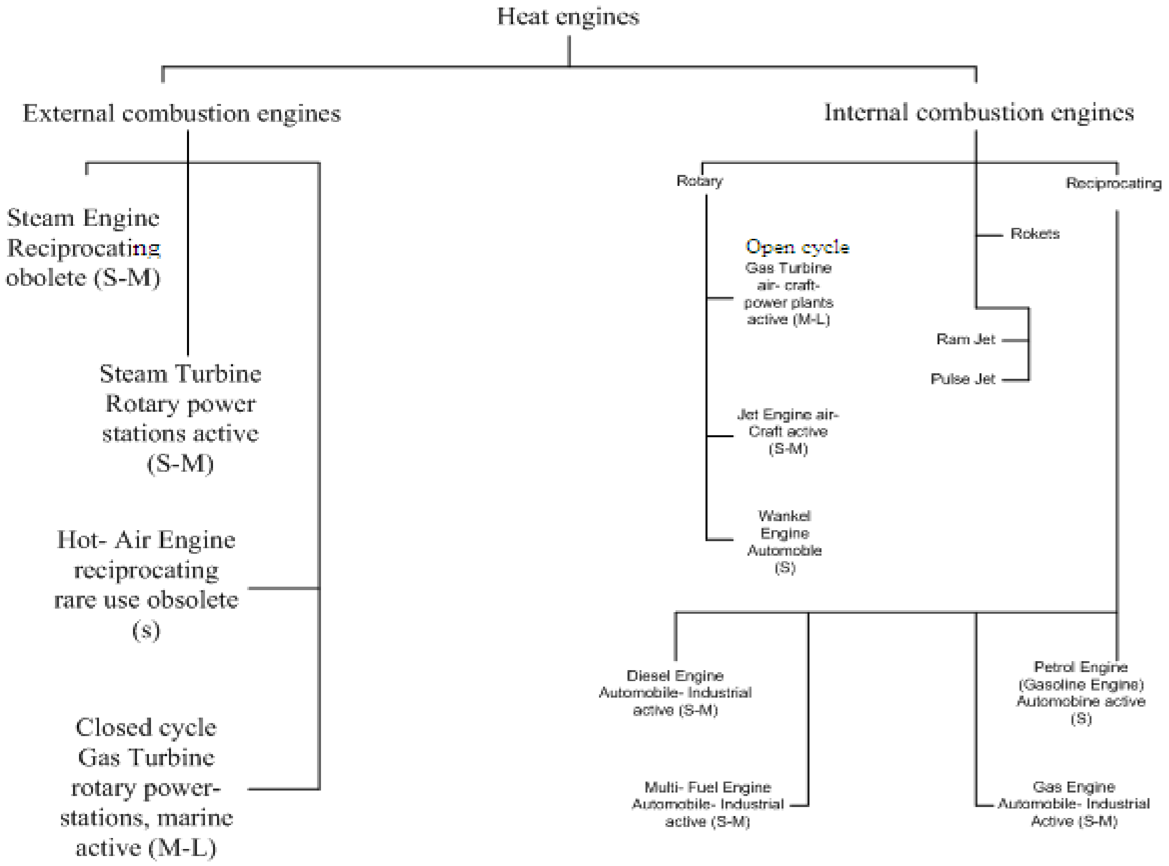
# *CHAPTER (1)*

# ENGINE CLASSIFICATION

## Classification of Heat Engines:



L: Large over 10000 KW
M: Medium between 1000- 10000 KW
S: Small under 1000 KW

*Figure (1-1): Engine Classification*

Heat engines can be classified as in figure (1-1); external combustion type in which the working fluid is entirely separated from the fuel- air mixture (ECE), and the internal - combustion (ICE) type, in which the working fluid consists of the products of combustion of the fuel- air mixture itself.

## Comparison between the Different Kinds:

### 1. The Reciprocation Piston Engine:

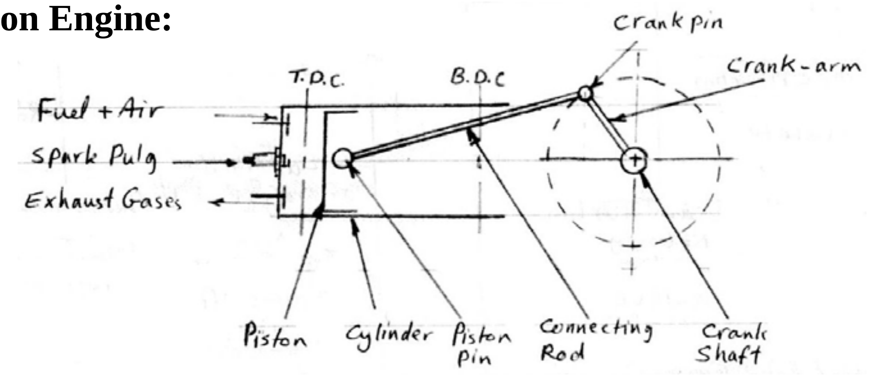


*Figure (1-2): Diagrammatic representation of reciprocating piston engine*

## 2. Open Cycle Gas Turbine:

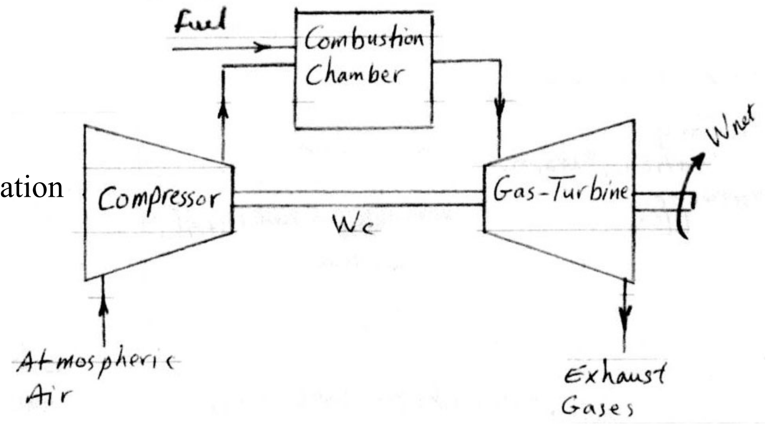


**Figure (1-3):** Diagrammatic representation of gas turbine

## 3. The Wankel Engine:

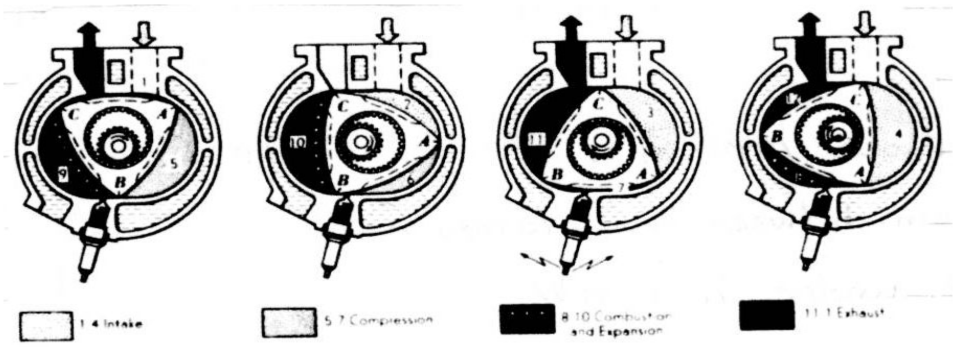


***Figure (1-4):*** *Wankel four- process cycle*

## 4. Steam Power Plant:

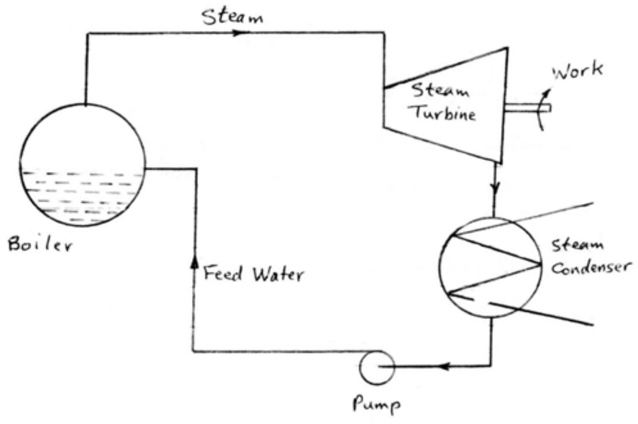


***Figure (1-5):*** *Diagrammatic representation of steam power plant*

The Reciprocating I.C.E. is one unit and does not need other devices, the efficiency of the engine is relatively high, and the fuel used is relatively expensive.

The gas turbine group needs a compressors, its weight is smaller than reciprocating I.C.E. of the same power, its efficiency is lower, the fuel relatively cheap, and it is suitable for air craft.

Rotary engine is a substitute for the reciprocating I.C.E. Wankel engine has a three lobe rotor which is driven eccentrically in a casing in such a way that there are three separate volumes trapped between the rotor and the casing. These volumes perform induction, compression, combustion, expansion and exhaust process in sequence. This design has a good power/volume ratio. Seal wear and heat transfer, were some of the initial development problems of the Wankel engine. These problems have now been largely solved.

The steam turbine is suitable for very large powers, its efficiency is reasonable; the fuel used in the boiler is cheap. The steam turbine needs a boiler, condenser and a continuous supply of water.

## Modern Developments of I.C.E.:

The modern I.C.E. is a product of research and developments extending over a long period of time many engines were proposed and tested, these include:

1. Stratified charge engine.
2. Dual Fuel and Multi- Fuel engines.
3. Sterling engine.
4. Free Piston engine.
5. Variable compression Ratio engine.
6. Combination of reciprocating engine with gas turbine.

## Advantages of I.C.E. over E.C.E.:

1. More mechanical simplicity and lower weight/power ratio.
2. They do not need auxiliary equipment, such as boiler & condenser.
3. They could be started and stopped in a short time.
4. Their thermal efficiency is higher than other heat engines.
5. Their initial cost is low.

These advantages make I.C.E. more suitable in the transport sector; motor cars, small ships, submarines, and small aircrafts.

## Types of Internal Combustion Engines:

I.C.E. can be divided into several groups according to different features as characteristics: operating cycles, method of charging the cylinder, fuel used, general design (position and number of cylinders, method of ignition, rotating speed, etc.), and method of cooling the engine.

## Fuel Used:

### 1. Liquid Fuels

a. Volatile liquid fuels: petrol, Alcohol, benzene. Fuel /Air mixture is usually ignited by a spark; Spark ignition.

b. Viscous liquid fuels: fuel oil, heavy and light diesel oil, gas-oil, bio-fuels. Usually combustion of fuel takes place due to its contact with high temperature compressed air (self-ignition); Compression ignition.

**2. Gaseous fuels:** Liquid Petroleum Gas ( LPG ), Natural gas ( NG ), Town gas, Blast Furnace gas; ignition usually by a spark.

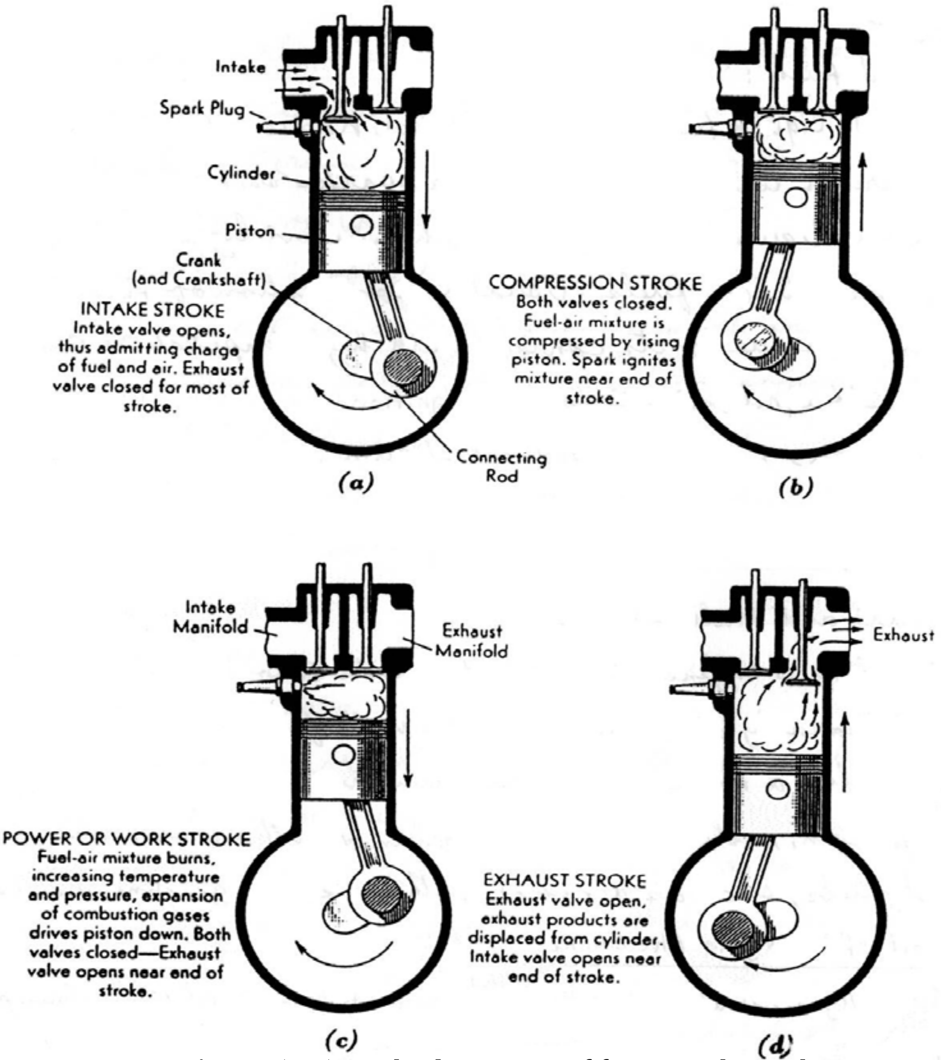
**3. Dual and Multi-Fuels:**

Dual fuel engines are operated with two types of fuels, either separately or mixed together. Multi-fuel engines could be operated by a mixture of more than two fuels, gaseous; such as: Hydrogen, methane, L.P.G. etc., combined with one or more of liquid fuels, such as alcohol, ethers, esters, gasoline, diesel etc...

## Method of Charging the Engine:

1. Naturally aspirated engine: Admission of charge at near atmospheric pressure.
2. Supercharged engine: Admission of charge at a pressure above atmospheric.

## Number of Piston Stroke to Complete the Thermodynamic Cycle:



INTAKE STROKE
Intake valve opens, thus admitting charge of fuel and air. Exhaust valve closed for most of stroke.

COMPRESSION STROKE
Both valves closed. Fuel-air mixture is compressed by rising piston. Spark ignites mixture near end of stroke.

POWER OR WORK STROKE
Fuel-air mixture burns, increasing temperature and pressure, expansion of combustion gases drives piston down. Both valves closed—Exhaust valve opens near end of stroke.

EXHAUST STROKE
Exhaust valve open, exhaust products are displaced from cylinder. Intake valve opens near end of stroke.

***Figure (1-6):*** *Cylinder events of four- stroke cycle*

Four stroke engines: the cycle of operation is completed in four strokes of the piston or two revolution of the crank shaft. Each stroke consists of $180^\circ$ of crank shaft rotation. The series of operations are shown in figure ( 1 – 6 ):

1. Two strokes engine: In two-stroke engine the cycle is completed in two strokes, i.e. one revolution of the crank shaft. The difference between two-stroke and four-stroke engines is in the method of filling the cylinder with the fresh charge and removing the products of combustion. The sequence of events shown in figure **(1 – 7):**

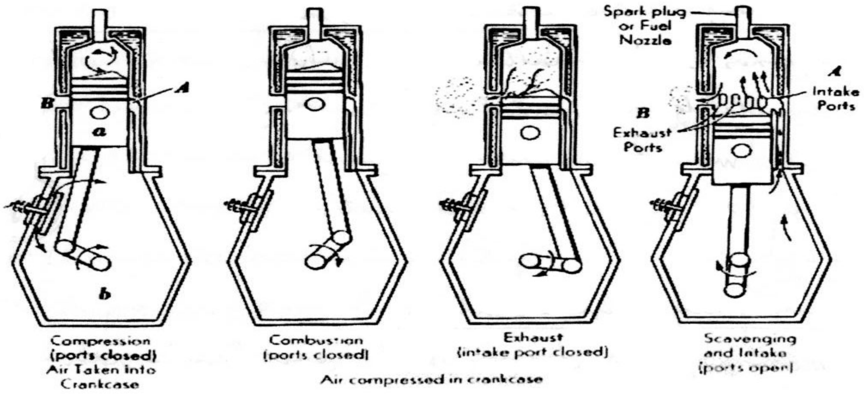
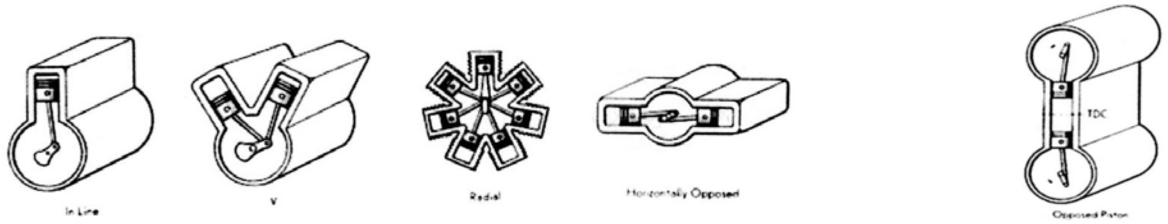


***Figure (1-7):*** *Cylinder events of two- stroke cycle*

## Cylinders Arrangement:

1. In- line engines: all cylinders are arranged linearly figure (1 – 8a).
2. "V" engines: cylinders are in two banks inclined at an angle to each other and with one crank-shaft figure (1 – 8b).
3. Radial engine: the radial engine is an engine with more than two cylinders in each row equally spaced around the crank shaft. Normally it is been used in air-crafts figure (1-8c).
4. Opposed cylinder banks located in the same plane on opposite sides of the crank-shaft figure (1 – 8d).
5. Opposed piston engine: when a single cylinder houses two pistons, each of which drives a separate crank shaft figure (1 – 8e).



a b c d e

***Figure (1-8):*** *Classification of engines by cylinder arrangement*

6. Double- acting engine: the fuel burns on both sides of the piston figure.
7. Cylinders may be vertical or horizontal, vertical engines needs smaller area, when area is available horizontal engines may be used, this would make all engine parts accessible and the operation and maintenance are easier.

## Classification by valves Location:

The most popular design is the overhead-valve design, there are also an under head valve engines and a combination of the two designs is also used see figure **( 1 – 9).**

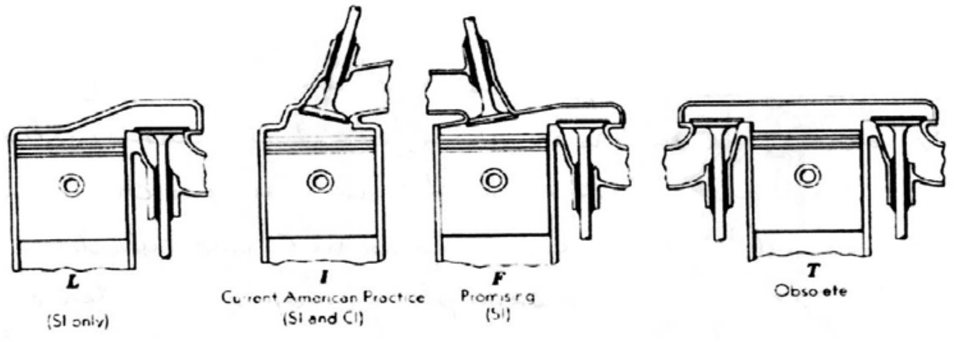


***Figure (1-9):*** *Classification of engines by valve location*

## Use of the Engine:

1. Marine engine: for propulsion of ships at sea.
2. Industrial engine: for power generation on land.
3. Automotive engine: for transport.

## Method of Cooling the Engine:

1. Water- cooled engines.
2. Air- cooled engines.

## The Continuous- Combustion Gas Turbine:

The main components of the gas turbine are; a compressor, a turbine and a combustion chamber, see figure (1 – 10). In operation, air is drawn into the compressor, compressed, and then passed, in part, through the combustion chamber. The high- temperature gases leaving the combustion chamber mix with the main body

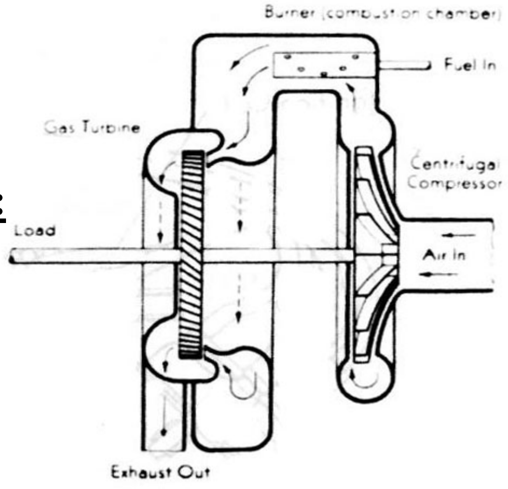
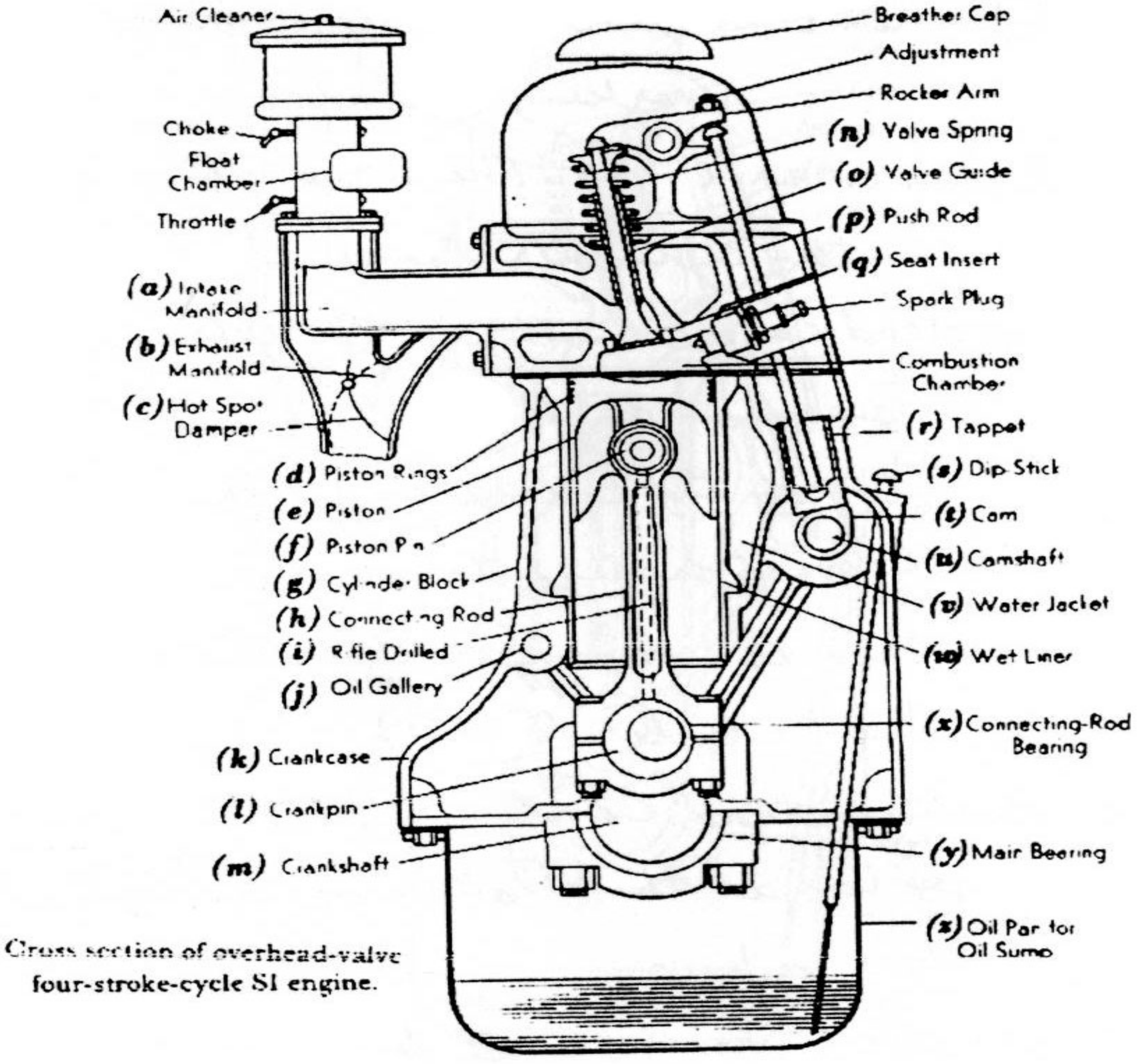


*Figure (1-10): Diagrammatic sketch of gas turbine* of air flowing around the combustor. This hot gas, with greatly increased volume, is led to a nozzle ring where the pressure is decreased and therefore the velocity is increased. The high- velocity gas is directed against the turbine wheel and the K.E. of the gas is utilized in turning the drive shaft, which also drives the air compressor.

## I.C.Engine Parts and Details:

The main components of the reciprocating internal combustion engine are shown in Figure (1-11). Engine parts are made of various materials and perform certain functions, some of which will be explained: cylinder block (g) it is integral with crank case (m), both are made of cast iron. The piston (e) reciprocates inside the cylinder, which include the combustion chamber.



Cross section of overhead-valve four-stroke-cycle SI engine.

*Figure (1-11): S.I.engine parts and details (over-head valve)*

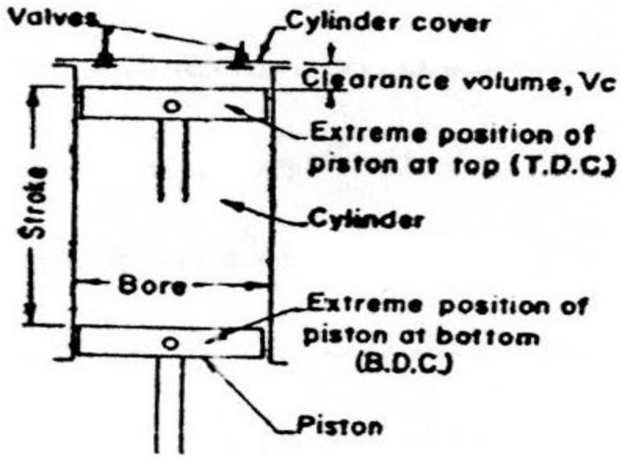
The piston is connected to the connecting rod (h) by piston pin (f). This end of the connecting rod is known as small end. The other end of the connecting rod called the big end is connected to the crank arm by crank pin (l).

Camshaft (u) makes the cam (t) to rotate and move up and down the valve rod through the tappet (r). Mainly each cylinder has two valves; one is admission or suction valve and the other is exhaust valve.

The ignition system consists of a battery, an ignition coil, a distributor with cam and breaker points, and spark plug for each cylinder. In diesel engines there is an injection system instead of ignition system.

## Internal Combustion Engines Terminology:

1. Cylinder bore (B): The nominal inner diameter of the working cylinder.
2. Piston area (A): the area of a circle diameter equal to the cylinder bore.
3. Top Dead Center (T.D.C.): the extreme position of the piston at the top of the cylinder. In the case of the horizontal engines this is known as the outer dead center (O.D.C.).



4. Bottom Dead Center (B.D.C.): the extreme position of the piston at the bottom of the cylinder. In horizontal engine this is known as the Inner Dead Center (I.D.C.).
5. Stroke: the distance between TDC and BDC is called the stroke length and is equal to double the crank radius (l).
6. Swept volume: the volume swept through by the piston in moving between TDC and is denoted by Vs:

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

Where d is the cylinder bore and l the stroke.

7. Clearance volume: the space above the piston head at the TDC, and is denoted by Vc:

Volume of the cylinder: $V = Vc + Vs$

8. Compression ratio: it is the ratio of the total volume of the cylinder to the clearance volume, and is denoted by (r)

$$r = \frac{V}{Vc} = \frac{Vc + Vs}{Vc}$$

9. Mean piston speed: the distance traveled by the piston per unit of time:

$$V_m = \frac{2lN}{60} m/s$$

Where l is the stroke in (m) and N the number of crankshaft revolution per minute (rpm).

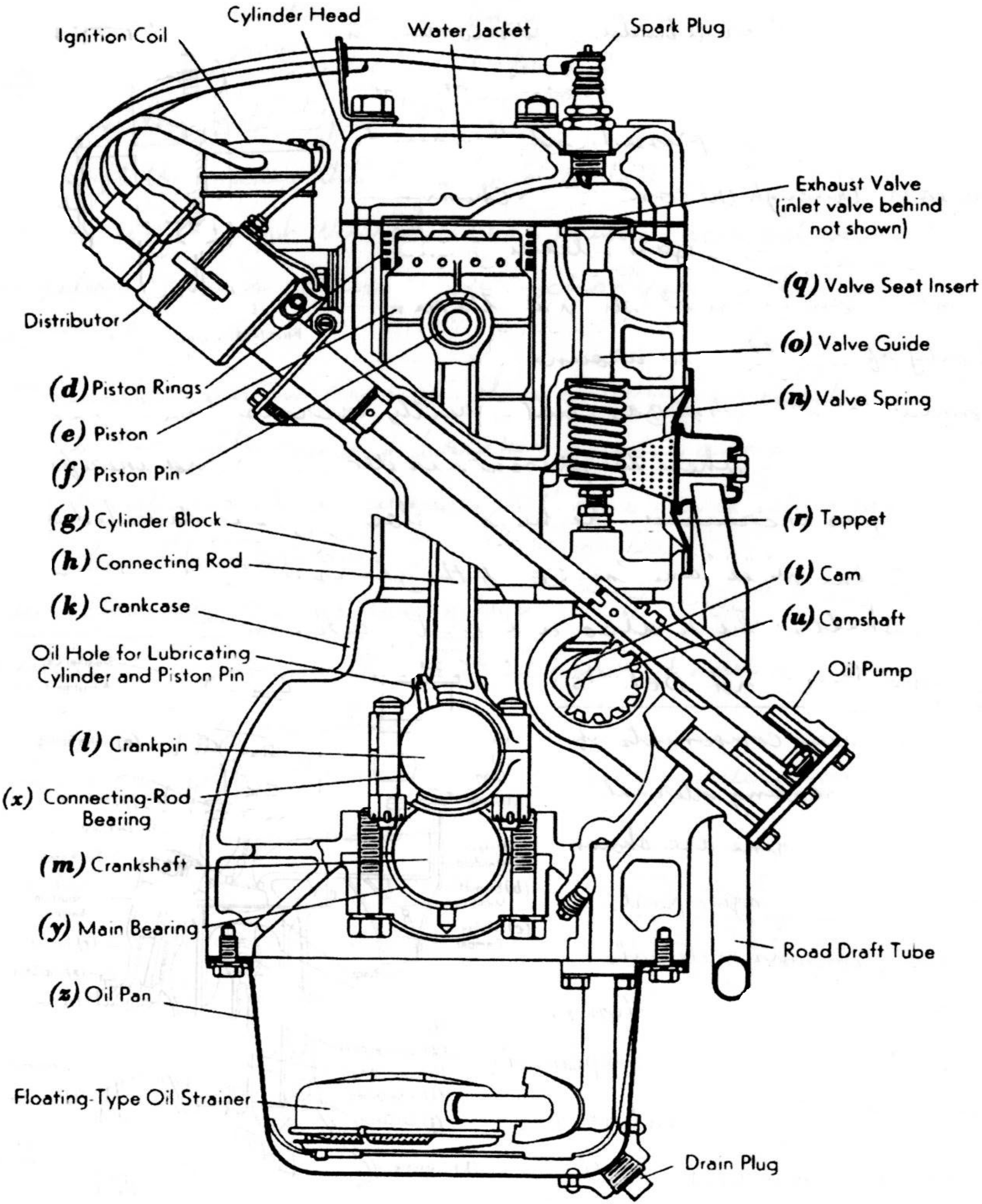


***Figure (1-12):*** *Piston and cylinder geometry ( under- head valve )*

## CHAPTER (2)

# AIR - STANDARD CYCLES

During every engine cycle, the medium changes sometimes it is a mixture of fuel and air or products of combustion, the specific heats and other properties of the medium change with temperature and composition.

The accurate study and analysis of I.C.E. processes is very complicated. To simplify the theoretical study "Standard Air Cycles" are introduced, these cycles are similar to the open cycles, but some simplifying assumptions are made:

1. Cylinder contains constant amount of air and it is treated as ideal gas.
2. The specific heats and other physical and chemical properties remain unchanged during the cycle.
3. Instead of heat generation by combustion, heat is transformed from external heat source.
4. The process of heat removal in the exhaust gases is represented by heat transfer from the cycle to external heat sink.
5. There is neither friction nor turbulence; all processes are assumed to be reversible.
6. No heat loss from the working fluid to the surroundings.
7. Cycles can be presented on any diagram of properties.

## <u>The Otto Cycle (constant Volume):</u>

These cycles is applied in petrol (or gasoline) engine, gas engine, and high speed diesel (oil) engine. The cycle is shown in Figure (1 – 13), and consists of the following processes:

1. Process 1 to 2 is isentropic compression;

$$Q_{12} = 0.$$

2. process 2 to 3 is reversible heat addition at constant volume

$$Q_{23} = C_v\,(T_3 - T_2)\ \text{kJ/kg}$$

3. process 3 to 4 is isentropic expression;

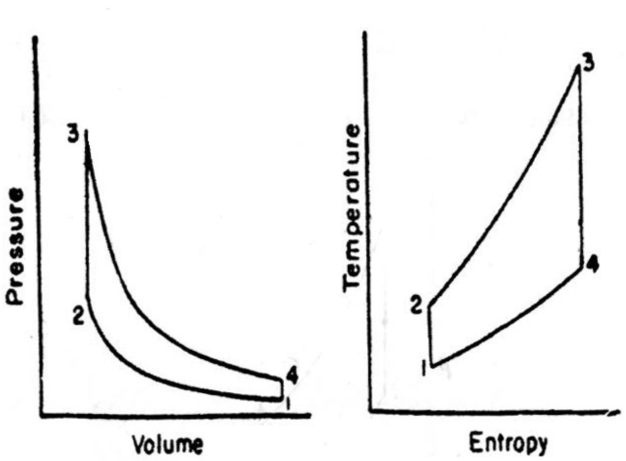


*Figure (1-13)*

$Q_{34} = 0$

*4. process 4 to 5 is reversible constant volume cooling*

$Q_{41} = C_v (T_4 - T_1)$ kj/ kg

This cycle is applied in 4- stroke and 2- stroke engines.

Work of Otto cycle = $W_{34} - W_{12}$

$$\eta = \frac{Q^{\uparrow} - Q^{\downarrow}}{Q^{\downarrow}} = 1 - \frac{Q^{\uparrow}}{Q^{\downarrow}}$$

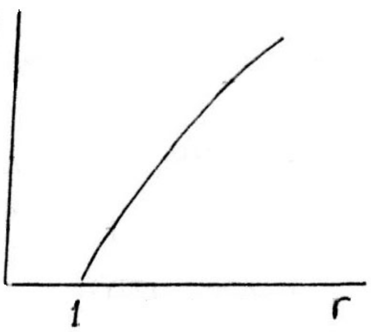
The thermal efficiency of the Otto cycle:

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

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

$T_3 = T_4 = r^{\gamma-1}$ and $T_2 = T_1 r^{\gamma-1}$

$$\therefore \eta = 1 - \frac{T_4 - T_1}{(T_4 - T_1) r^{\gamma-1}} = 1 - \frac{1}{r^{\gamma-1}}$$



η increased by increasing r

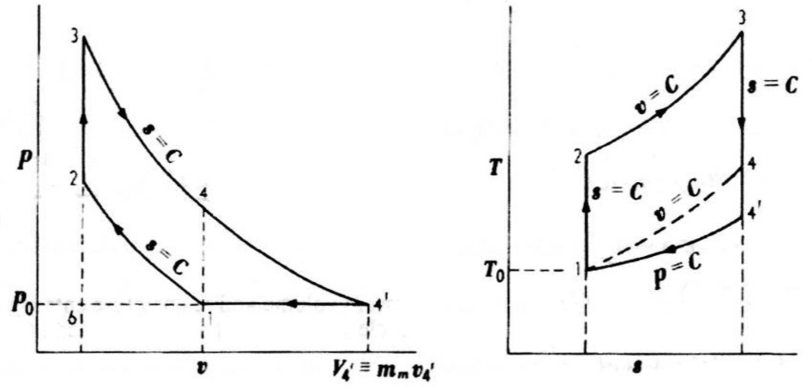
η increased by increasing γ

η independent on the heat added or load.

In modern petrol engines (r) reaches a value of 12.

To make use of that part of the energy in the exhaust gases, they may be expanded to atmospheric pressure in an exhaust gas turbine; the work of the cycle will be increased by the aria $144\bar{4}1$ as shown in figure (1-14). The new cycle $(1234\bar{4}1)$ is called **Atkinson cycle**, this cycle is applied in a combination of petrol engine and gas turbine, the turbine in usually used to drive a compressor to supercharge the engine.

**Figure (1-15):** *Air- standard complete- expansion cycle*



## The Diesel (or Constant Pressure) Cycle:

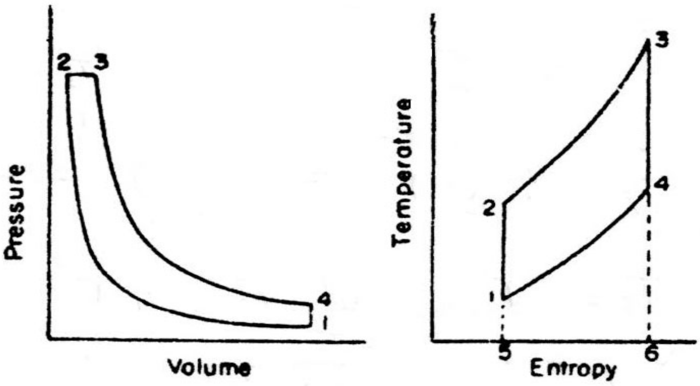


***Figure (1-16):*** *pressure volume and temperature entropy diagram of a cycle with constant pressure heat addition*

This cycle is the theoretical cycle for compression-ignition or diesel engine. For this cycle:

$Q_1 = C_p\,(T_3 - T_2)$
$Q_2 = C_v\,(T_4 - T_1)$

$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{1}{\gamma}\left(\frac{T_4 - T_1}{T_3 - T_2}\right) = 1 - \frac{T_1}{\gamma T_2}\left(\frac{\frac{T_4}{T_2} - 1}{\frac{T_3}{T_2} - 1}\right)$$

For isentropic compression and expansion:

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma-1} \text{ and, } \frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{\gamma-1}$$

For constant Pressure heat addition 2-3: $\frac{T_3}{T_2} = \frac{v_3}{v_2}$

Thus:

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}\left(\frac{v_3/v_2}{v_2/v_1}\right)^{\gamma-1} = \frac{v_3}{v_2}\left(\frac{v_3}{v_2}\right)^{\gamma-1} = \left(\frac{v_3}{v_2}\right)^{\gamma}, \qquad v_4 = v_1$$

By substitution:

$$\eta = 1 - \frac{1}{\gamma\left(\frac{v_1}{v_2}\right)^{\gamma-1}}\left(\frac{\left(\frac{v_3}{v_2}\right)^{\gamma} - 1}{\left(\frac{v_3}{v_2}\right) - 1}\right)$$

$\frac{v_1}{v_2} = r$ and $\frac{v_3}{v_2} = \beta$, then:

$$\eta = 1 - \frac{\beta^{\gamma} - 1}{(\beta - 1)\gamma\, r^{\gamma - 1}}$$

This equation shows that thermal efficiency depends not only on r but also on the cut-off ratio $\left(\beta = \frac{v_3}{v_2}\right)$ and the working medium properties γ. As β increase the work done per cycle increase but η decreases. When r increase more than 22, the increase in η is small, on the other hand, maximum pressure increases much and mass of the engine increases.

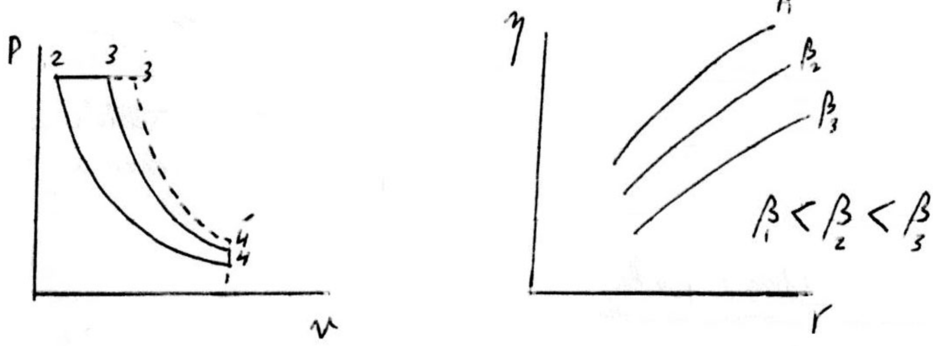


*Figure (1-17)*

## The Dual Cycle:

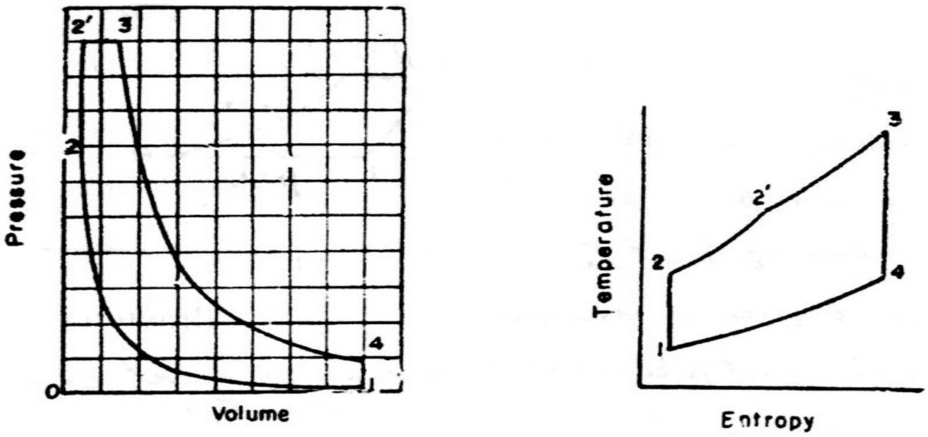


***Figure (1-18):*** *Pressure-Volume and Temperature-Entropy diagram of dual cycle*

The cycle is applied in medium speed and high speed diesel engines. The engine may be 4 or 2 strokes

$$Q_1 = Q_1' + Q_1''$$

$$= C_v\left(T_2' - T_2\right) + C_p\left(T_3 - T_{2'}\right)$$

$$Q_2 = C_v(T_4 - T_1)$$

$$\eta = 1 - \frac{Q_2}{Q_1}$$

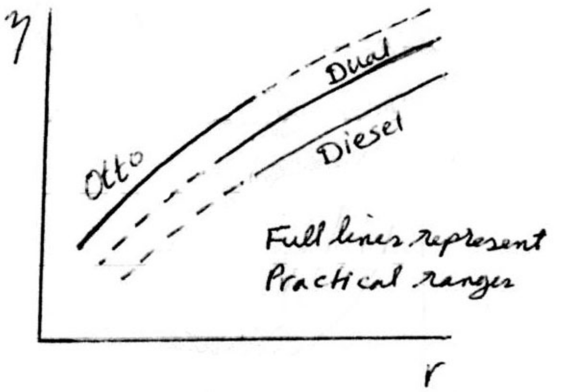
It can be proved that:

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{k\beta^{\gamma} - 1}{(k-1) + \gamma k(\beta - 1)}\right]$$

$$k = \frac{p_{2'}}{p_2} \text{ and } \beta = \frac{V_3}{V_{2'}}$$

When k = 1, then $p_{2'} = p_2$, we obtain diesel cycle. When β = 1, then $V_3 = V_{2'}$, we obtain Otto cycle. The indicated thermal efficiency of this cycle lies between that of the Otto and diesel.

*Figure (1-19)*



## Joule (or Brayton) Cycle:

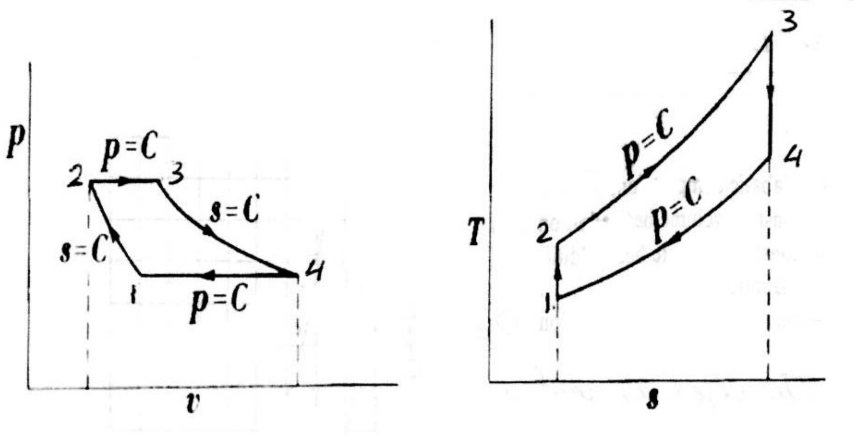


***Figure (1-20):*** *Brayton (Joule) air- standard cycle*

This cycle is used in gas turbine, it consists of:

ab: isentropic compression.

bc: constant pressure addition of heat $Q_1$.

cd: isentropic expansion.

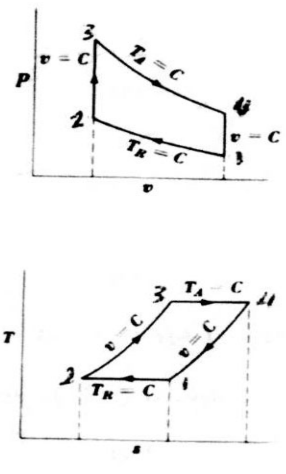
da: constant pressure rejection of heat $Q_2$.

$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{Cp(T_4 - T_1)}{Cp(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

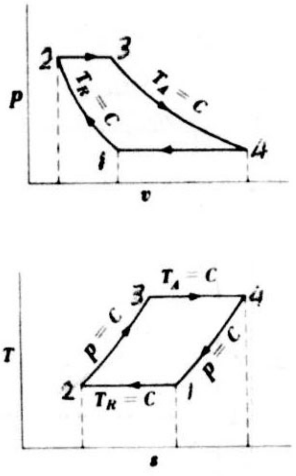
$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{\gamma-1} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \frac{1}{r^{\frac{\gamma-1}{\gamma}}}$$

## Regenerative Cycles:

1. Sterling Cycle: It consists of two isothermal; 12 & 34 and two constant volume 2-3 & 41. Heat is added in the constant 2 – 3 and also in the isothermal process 3 – 4. Not that the temperature of initial heat rejection is $T_4$ is higher than the initial temperature of heat addition $T_2$, therefore, it is possible to use a heat exchanger to utilize part of the rejected heat to the heat addition process. If the Stirling cycle could be perfectly regenerated, no heat would be necessary for process 23.



Sterling Cycle



Ericsson Cycle

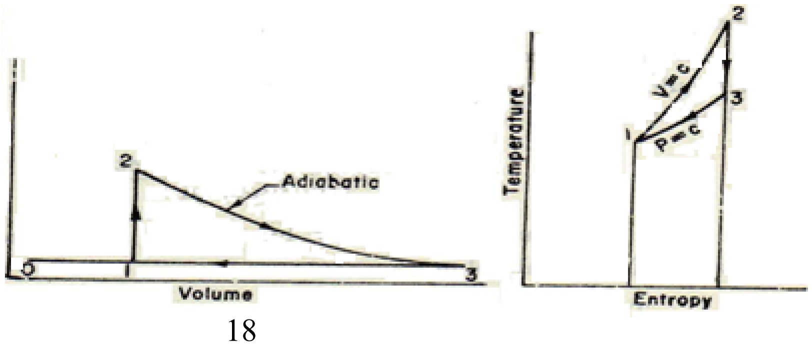
2. Ericsson Cycle: Same comments can be made for Ericsson cycle. Heat is added in the expansion at constant pressure 23 and at constant temperature 34. Heat is rejected in the compression process at constant pressure 41 and at constant temperature 12. Since 23 and 41 are parallel, then gas can be heated from 2 to 3 by cooling from 4 to 1.

## Lenoir Cycle:

$Q_{add} = C_v ( T_2 – T_1 )$

$Q_{rej} = C_p ( T_3 – T_1 )$

$\eta = 1 - \gamma\frac{(T3 - T1)}{(T2 - T1)}$

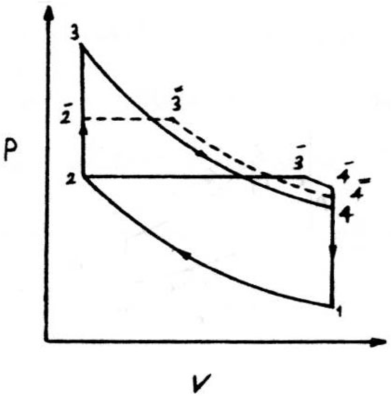


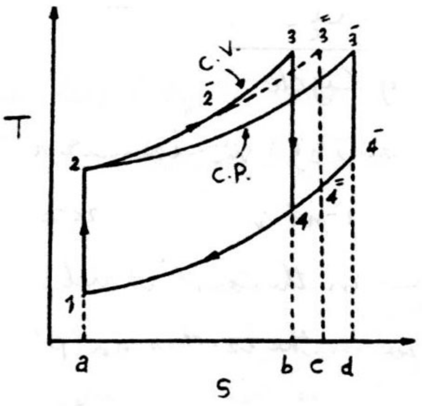
## Comparison of Otto, diesel and Dual Cycles:

The important parameters in cycle analysis are compression ratio, maximum pressure, maximum temperature, head input, work output, etc.

1. *Equal Compression Ratio and Equal Heat added:*

   The three cycles start from the same point (1); compression process is the same (12) for all cycles, equal heat added means:





$$area\ a23ba = area\ a2\bar{3}da = area\ a2\bar{2}\bar{\bar{3}}ca$$

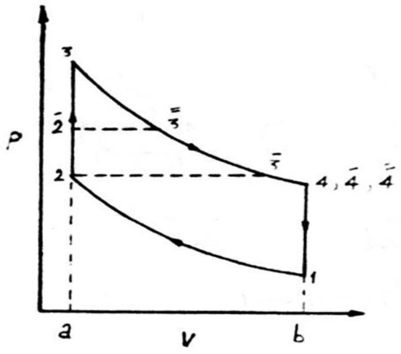
areas representing heat rejected are:

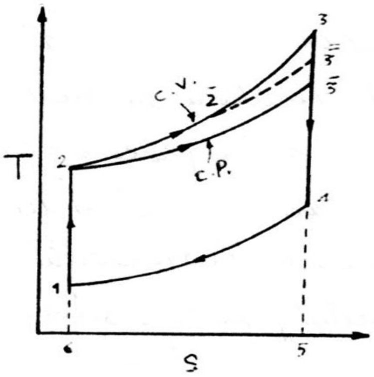
$$area\ a14ba < area\ a1\bar{\bar{4}}ca < area\ a1\bar{4}da$$

$$\eta = \frac{heat\ aded - heat\ rejected}{heat\ added}$$

$$\therefore \eta_{Otto} > \eta_{dual} > \eta_{diesel}$$

2. *Equal Compression Ratio and Heat rejected:*





Processes (12) and (41) are the same in all cycles.

Areas representing heat added are:

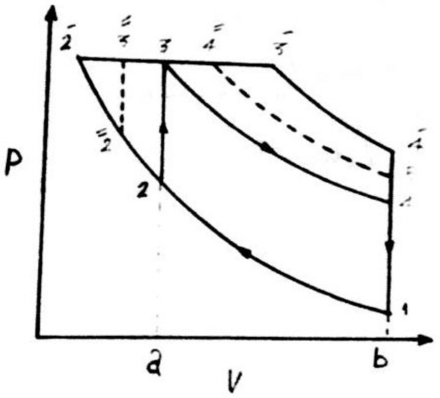
$area\,6235 > area\,62\bar{2}\bar{\bar{3}}5 > area\,62\bar{3}5$

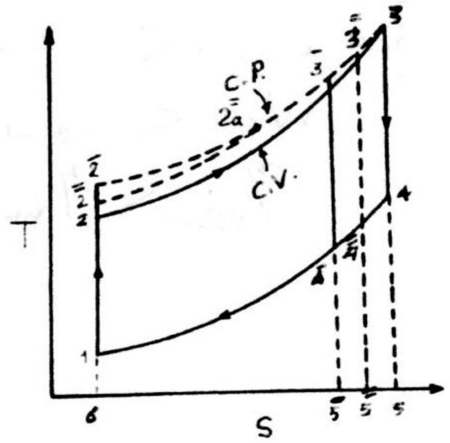
Heat rejected area is the same for the three cycles

$\therefore\ W_{Otto} > W_{dual} > W_{diesel}$, and

$\eta_{Otto} > \eta_{dual} > \eta_{diesel}$

3. *Equal Maximum Pressure and Equal Heat added:*





Point (1) is common in the three cycles.

Maximum pressure is the same, therefore, compression ratio are different as indicated in the diagram above.

Areas representing heat added: $6235 = 6\bar{2}\bar{\bar{2}}a\bar{\bar{3}}\bar{\bar{5}} = 6\bar{2}\bar{3}\bar{5}$

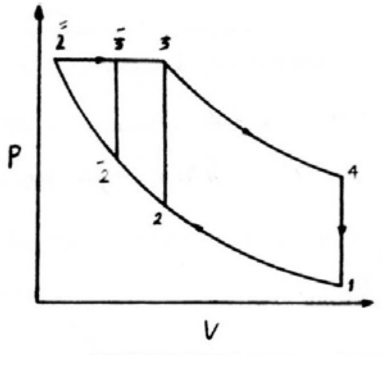
Areas representing heat rejected: $6145 > 61\bar{\bar{4}}\bar{\bar{5}} > 61\bar{4}\bar{5}$

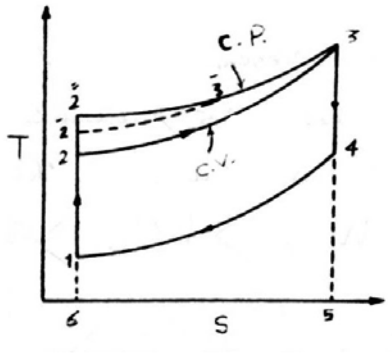
$\therefore\ W_{Otto} < W_{dual} < W_{diesel}$

i.e.

$\eta_{Otto} < \eta_{dual} < \eta_{diesel}$

4. *Equal Maximum Pressure and Maximum Temperature:*





Points 1, 3, 4 are common in the three cycles

Heat rejected is the same in the three cycles

Heat added is highest in diesel cycle:

Area $6235 < 6\bar{2}\bar{3}35 < 6\bar{\bar{2}}35$

$\therefore W_{Otto} < W_{dual} < W_{diesel}$

i.e.

$\eta_{Otto} < \eta_{dual} < \eta_{diesel}$

# *CHAPTER (3)*

# FUEL AND COMBUSTION

## ***Fuels:***

Fuel is any material when burnt will produce heat energy. Various fuels commonly used are as follows:

### 1. Solid Fuels:

Coal is the most important solid fuel; there are other types of solid fuels, such as: nuclear fuels, solid wastes (wood, sugar-cane). Coal is divided into groups according to their chemical and physical properties. An accurate chemical analysis for the fuel by mass is called the "ultimate analysis", which gives the percentage of the important elements present in the fuel. Another approximate analysis of fuel called the "proximate analysis" gives the percentage of moisture, volatile matter, and combustible solid (fixed carbon) and ash.

Ultimate Analysis of Coals

| Coal | C | H | O | N + S | Ash |
|---|---|---|---|---|---|
| Anthracite | 90.27 | 3.00 | 2.32 | 1.44 | 2.97 |
| Bituminous | 74.00 | 5.98 | 13.01 | 2.26 | 4.75 |
| Lignite | 56.52 | 5.72 | 31.89 | 1.62 | 4.25 |

### 2. Liquid Fuels:

Liquid fuels are widely used for I.C.E. Practically all liquid fuels have two basic combustible elements; carbon and hydrogen, present separately or in a combination called hydrocarbons, there are principal commercial types of liquid fuels:

*i. Petroleum and its Derivatives*

They are easy to handle, store, and burn and have nearly constant heating values. They are usually a mixture of hydrocarbons that may be represented by the molecular formula of the form $C_nH_m$, where m is a function of n that depends upon the family of the hydrocarbon.

| Family | Formula | Prefix/suffix | structure |
|---|---|---|---|
| **Paraffin** (alkanes) | $C_nH_{2n+2}$ | ane | Chain, saturated |
| **Olefin** | $C_nH_{2n}$ | ene,ylene | Chain, Unsaturated, one double- bond |
| **Diolefin** | $C_nH_{2n-2}$ | diene | Chain, Unsaturated, two double- bond |
| **Aromatic** | | | |
| Benzene | $C_nH_{2n-6}$ | | Ring, Unsaturated |
| Naphthalene | $C_nH_{2n-12}$ | | Double ring, Unsaturated |

ii. *Synthetic fuels*

Usually called synfuels, which are liquid and gases mainly produced from coal, oil shale, tar sands and also from various wastes and biomass.

iii. *Alcohols*

The alcohols are a partial oxidation product of petroleum, and are not found to any extent in the crude oil. The compounds are saturated, with a chain structure with the general formula R.OH. Here the radical R is the paraffin group attached to the hydroxyl radical OH. Alcohols are designated by the name of the radical:

$CH_3OH$: Methyl alcohol or Methanol.
$C_2H_5OH$: Ethyl alcohol or Ethanol.
$C_3H_7OH$: Propyl alcohol or Propanol.
$C_4H_9OH$: Butyl alcohol or Butanol.

### 3. Gaseous Fuels:

Gaseous fuels are may be either natural or manufactured; natural gas: is a mixture of components, consisting mainly of methane (60-98%) with small amount of other hydrocarbons. In addition it consists; $N_2$, $CO_2$, $H_2$ and traces of other gases. Its sulfur content ranges from very little (sweet) to larger amounts (sour). It is classified as associated or unassociated depending on whether it is associated with oil or not. It is stored as compressed natural gas (CNG) or as liquid (LNG) at pressure ranging between 70 to 210 kPa.

*Manufactured Gases Include:*

1. Liquid petroleum gas (LPG): It is mixture of propane and butane and some other light hydrocarbons. Propane and butane are the main constituent of LPG; their quantity varies with time of the year and place.
2. Coal gas: it is a by-product obtained during the destructive distillation of coal. Its main approximate composition is $CH_4$ = 25%, H = 53%, CO = 9%, N = 6%, $CO_2$ = 2% and other hydrocarbons ≈ 1%.
3. Coke oven Gas: It is produced during the manufacture of coke from new coal in a coke oven where the volatile matter is distilled off and the coke-oven separated from liquids and solids in the volatile matter by cooling and extraction. It consists about 50% $H_2$, about 30% methane($CH_4$), and the remainder of various other gases. Its heating value ranges between 14200 to21300 kJ/$m^3$.
4. Blast-furnace gas: it is produced as by-product from blast furnaces used in iron reduction process, has about 30%, CO, 2% $H_2$, 11% $CO_2$ and about 60% $N_2$.

There are other gaseous fuels of lesser importance like producer gas, water gas, and town gas.

**Advantages of gaseous fuels**:

1. Low emissions.
2. Mixing with air is uniform, so they need less excess air.
3. Economical compared to other types of fuels.

## Combustion of Fuels:

Combustion of fuel is accomplished by mixing fuel with air at elevated temperature:

*Fuel + Air →Products of Combustion + Heat*

The oxygen contained in the air unites chemically with carbon, hydrogen and other elements in fuel to produce heat. The amount of heat liberated during the combustion process depends on the amount of oxidation of the constituent of fuel and the nature of fuel.

In order that the combustion of fuel may take place with high efficiency, the following conditions must be fulfilled:

1. The amount of air supplied should be sufficient.
2. The air and fuel should be thoroughly mixed.
3. The temperature of the reactants should be high enough to ignite the mixture.
4. Sufficient time should be available to burn fuel completely.

## Combustion Chemistry:

### 1. Chemical Equation:

The chemical equation shows how the atoms of the reactants are arranged to form products. Before the chemical equation can be written it is necessary to know the number of atoms of elements in the molecules of the reactants and products. During combustion process the atoms are rearranged to form new molecules, and the total number of atoms of each element is unchanged. A chemical equation expresses the principle of the conservation of mass in terms of the conservation of atoms

*i- Combustion of Carbon*

$C + O_2 \rightarrow CO_2$

1 kmol C + 1 kmol $O_2 \rightarrow$ 1 kmol $CO_2$

12 kg C + 32 kg $O_2 \rightarrow$ 44 kg $CO_2$

If any of the reactants or products is in solid or liquid phase, the volume occupied by them can be neglected. It follows that:

0 vol. C + 1vol $O_2 \rightarrow$ 1vol. $CO_2$

If *insufficient* oxygen is present for all carbon to burn to carbon dioxide, some will burn to carbon monoxide, then:

$C + 1/2\ O_2 \rightarrow CO$

*ii- Combustion of Hydrogen*

$H_2 + 1/2\ O_2 \rightarrow H_2O$

1 kmol $H_2$ + 1/2 kmol $O_2 \rightarrow$ 1 kmol $H_2O$

2kg $H_2$ + 16 kg $O_2 \rightarrow$ 18kg $H_2O$

1 vol. $H_2$ + 1/2 vol. $O_2 \rightarrow$ 1 vol. $H_2O$ (vapour)

In most engineering combustion systems the necessary oxygen is obtained by mixing the fuel with air (except rockets) and it is necessary to use accurate and consistent analysis of air by mass and by volume. It is usual in combustion calculations to take air as 23.3% $O_2$, 76.7% $N_2$ by mass, and 21% $O_2$, 79% $N_2$ by volume. The small traces of other gases in dry air are included in the nitrogen, which is sometimes called "*atmospheric nitrogen*".

The moisture or humidity in atmospheric air varies over wide limits, depending on meteorological conditions. Its presence in most cases simply implies an additional amount of inert material.

The molar mass of $O_2$ can be taken as 32 kg/kmol, and that of $N_2$ as 28 kg/kmol and air 29 kg/kmol.

Since oxygen is accompanied by nitrogen when air is supplied for combustion, then this nitrogen should be included in the combustion equation, it will appear on both sides of the equation. With one mole of $O_2$ there are $\frac{79}{21} = 3.762$ moles of $N_2$, Hence:

$$\mathrm{C} + \mathrm{O}_2 + \frac{79}{21}\mathrm{N}_2 \rightarrow \mathrm{CO}_2 + \frac{79}{21}\mathrm{N}_2$$

Also

$$H_2 + \frac{1}{2}(O_2 + \frac{79}{21}N_2) \rightarrow H_2O + \frac{1}{2} \times \frac{79}{21}N_2$$

A frequently used quantity in the analysis of combustion process is the *air-fuelratio* A/F. it is defined as the ratio of the mass of air to the mass of fuel for a combustion process.

$$A/F = \frac{m_a}{m_f} = \frac{mass\,of\ air}{mass\,of\ \ fuel}$$

The mass m of a substance is related to the number of moles n through the relation: m = nM, where M is the *molar mass*. The reciprocal of A/F ratio is called the *fuel-air ratio*.

The minimum amount of air needed for the complete combustion of a fuel is called the stoichiometric or theoretical air. In actual combustion processes, it is common practice to use more air than the stoichiometric amount. The amount of extra air than the stoichiometric is called (***excess air***). Amount of air less than stoichiometric amount is called (***deficiency of air***). ***Equivalence ratio*** is the ratio of the actual fuel- air ratio to the stoichiometric fuel-air ratio. Sometimes this ratio is given in term of A/F ratio and called ***mixture strength***.

$$Mixture\,strength = \frac{Stoichiometric\ A/F\ Ratio}{Actual\ A/F\ Ratio}$$

$$Equivalence\,ratio = \frac{Actual\,(F/A)\,ratio}{Stoich.\ (F/A)\,ratio} = \phi$$

$$\phi = \frac{(\mathrm{F/A})\,\mathrm{actual}}{(\mathrm{F/A})\,\mathrm{stoich}} = \frac{(\mathrm{A/F})\,\mathrm{stoich}}{(\mathrm{A/F})\,\mathrm{actual}}$$

*Where:*

$\phi$ = 1: stoichiometric

$\phi$< 1: lean (week) mixture- excess of air.

$\phi$> 1: rich mixture- deficiency of air.

A general reaction equation of a hydrocarbon fuel for stoichiometric condition with air is given by:

$$C_nH_m + \left(n+\frac{m}{4}\right)O_2 + 3.762\left(n+\frac{m}{4}\right)N_2 \rightarrow nCO_2 + \frac{m}{2}H_2O + 3.762\left(n+\frac{m}{4}\right)N_2$$

The composition of a hydrocarbon fuel $C_nH_m$ are carbon and hydrogen, n and m can be determined for 1 kg of fuel as follows:

$$\frac{Weight\,of\ C\ in\ fuel}{Weight\,of\ \ fuel} = \frac{C}{1} = \frac{12n}{12n + 2.02\frac{m}{2}} = \frac{12n}{12n + m}$$

## Exhaust and Flue Gas Analysis:

The products of combustion are mainly gaseous. When a sample is taken for analysis it is usually cooled down to a temperature which is below the saturation temperature of the steam present. The steam content is therefore not included in the analysis, which is then quoted as the as the analysis of *dry products*. Since the products are gaseous, it is usual to quote the analysis by volume. An analysis which includes the steam in the exhaust is called a *wet analysis*.

**Example (3 – 1):**

Find the stoichiometric A/F ratio for the combustion of ethyl-alcohol ($C_2H_5OH$) in a petrol engine. Calculate the A/F ratios for 0.9 & 1.2 equivalence ratios(ϕ). Determine the wet and dry analyses by volume of the exhaust gas for each equivalence ratio.

**Solution:**

Combustion equation of ethyl-alcohol is:

$$C_2H_5OH + 3O_2 + 3\times\frac{79}{21}N_2 \rightarrow 2CO_2 + 3H_2O + 3\times\frac{79}{21}N_2$$

One mole of fuel has a mass of (2×12+16+6) =46 kg

Mass of air required for complete burning of one mole of fuel is:

$$\left(3\times 32 + 3\frac{79}{21}\times 28\right) = 412.0\,kg$$

$$\therefore \text{Stoichiometric A/F ratio} = \frac{412}{46} = 8.96/1$$

$$Equivalence\ ratio(\phi) = \frac{(A/F)stoich.}{(A/F)actual}$$

$$0.9 = \frac{8.96}{(A/F)_{actual}}$$

$$(A/F)_{actual} = 8.96/0.9 = 9.95$$

Volumetric A/F ratio = 3 × (1+3.762) =14.3

For ϕ = 0.9; air supplied is $= \frac{1}{0.9} = 1.11$ times as much air supplied for complete combustion, then: combustion equation becomes:

$$C_2H_5OH + 1.11\left(3O_2 + 3\times\frac{79}{21}N_2\right) \rightarrow 2CO_2 + 3H_2O + 0.11\times 3O_2 + 1.11\times 3\times\frac{79}{21}N_2$$

i.e. The total number of moles of products = 2+3+0.33+12.54 =17.87.

Hence wet analysis is:

Total dry moles = 2+0.33+12.54=14.87

Hence dry analysis is:

$$\frac{2}{17.87}\times 100 = 11.20\%\ CO_2; \qquad \frac{3}{17.87}\times 100 = 16.80\%\ H_2O$$

$$\frac{0.33}{17.87}\times 100 = 1.85\%\ O_2; \qquad \frac{12.54}{17.87}\times 100 = 70.20\%\ N_2$$

For $\phi = 1.2$:

$$\text{Actual A/F ratio} = \frac{8.96}{1.2} = 7.47:1$$

This means that $\frac{1}{1.2} = 0.834$ of the stoichiometric air is supplied. The combustion cannot be complete & is usual to assume that all the hydrogen is burned to $H_2O$, since $H_2$ atoms have a greater affinity for oxygen than C atoms. The carbon in the fuel will burn to CO and $CO_2$:

$$C_2H_5OH+0.834(3O_2+3\times 3.762N_2)\rightarrow aCO_2+bCO+3H_2O+0.834\times 3\times 3.762N_2$$

C balance: 2 = a + b

O balance: 1+2 × 0.834 × 3=2a+b+3

Subtracting the equations gives: a=1.004

and then: b =2 – 1.004 =0.996

i.e. The products are: 1.004 moles $CO_2$+ 0.996 moles CO+3 moles $H_2$+ 9.41 moles $N_2$

The total moles of products = 1.004+0.996 +3+9.41 =14.41

Hence wet analysis is:

$$\frac{1.004}{14.41}\times 100 = 6.97\%\ CO_2; \qquad \frac{0.996}{14.41}\times 100 = 6.91\%\ CO$$

$$\frac{3}{14.41}\times 100 = 20.8\%\ H_2O; \qquad \frac{9.41}{14.41}\times 100 = 65.3\%\ N_2$$

The total dry moles = 1.004+0.996+9.41=11.41

Hence dry analysis is:

$$\frac{1.004}{11.41}\times 100 = 8.80\%\ CO_2; \qquad \frac{0.996}{11.41}\times 100 = 8.73\%\ O_2$$

$$\frac{9.41}{14.41} = 82.47\%\ N_2$$

## Dissociation:

It is found that during adiabatic combustion the maximum temperature reached is lower than that expected on the bases of elementary calculation. One important reason for this is that the exothermic combustion process can be reversed to some extent if the temperature is high enough. The reversed process is an endothermic one, i.e. energy is absorbed. In a real process the reaction proceeds in both directions simultaneously and ***chemical equilibrium*** is reached when the rate of break-up of product molecules is equal to their rate of formation. This is represented, for the combustion of CO and $H_2$, by:

$$2CO+O_2 \Leftrightarrow 2CO_2 \text{ and } 2H_2+O_2 \Leftrightarrow 2H_2O$$

Both of these equations can take place simultaneously in the same combustion process. The proportions of the constituents adjust themselves to satisfy the equilibrium conditions and their actual values depend on the particular pressure and temperature.

Dissociation may be neglected at temperature below 1500K. As the gases cool down in the expansion stroke, dissociation is reduced and the recombination of the dissociated molecules gives up heat.

Let the gases A & B react to form the products C & D in the proportions dictated by the stoichiometric equation:

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

The proportion of gases A, B, C and D in the balanced mixture is determined by the dissociation ***equilibrium constant***, K which is defined as:

$$K = \frac{P_C^{\upsilon_c} P_D^{\upsilon_d}}{P_A^{\upsilon_a} P_B^{\upsilon_b}}$$

K is a function of temperature and its values are tabulated against temperature for each reaction equation.

**<u>Example (3 – 2):</u>**

The products from the combustion of a stoichiometric mixture of CO and $O_2$ are at a pressure of 1atm and a certain temperature. The products analysis shows that 35 per cent of each kmol if $CO_2$ is dissociated. Determine the equilibrium constant for this temperature, and hence find the percentage dissociation when the products are at the same temperature but compressed to 10 atmospheres.

**Solution:**

The combustion equation is:

$$CO + \frac{1}{2}O_2 \rightarrow (1-a)CO_2 + aCO + \frac{a}{2}O_2$$

*Where:*

a is the fraction of $CO_2$ dissociated

At 1 atm. the products consists of $0.65CO_2 + 0.35\ CO + 0.175\ O_2 = n_t$

i.e. total number of moles of products = 1.175

$$P_{Co_2} = \frac{n_{co_2}}{n_t} P = \frac{0.65}{1.175} = 0.553 \quad \text{atm}$$

$$P_{co} = \frac{n_{co}}{n_t} P = \frac{0.35}{1.175} = 0.298 \quad \text{atm}$$

$$P_{o_2} = \frac{n_{o_2}}{n_t} P = \frac{0.175}{1.175} = 0.149 \quad \text{atm}$$

Based on the equilibrium equation $CO + \frac{1}{2}O_2 \Leftrightarrow CO_2$

$$K = \frac{P_{co_2}}{P_{co}(P_{O2})^{1/2}} = \frac{0.553}{0.298\,(0.149)^{\frac{1}{2}}} = 4.81(atm)^{-\frac{1}{2}}$$

At pressure P, the partial pressures will be:

$$P_{co_2} = \frac{1-a}{1+\frac{a}{2}}P, \qquad P_{co} = \frac{a}{1+\frac{a}{2}}P, \qquad P_{o_2} = \frac{\frac{a}{2}}{1+\frac{a}{2}}P$$

And equilibrium constant will be:

$$K = \frac{\left(\frac{1-a}{1+\frac{a}{2}}\right)}{\left(\frac{a}{1+\frac{a}{2}}\right)\left(\frac{\frac{a}{2}}{1+\frac{a}{2}}\right)^{\frac{1}{2}}} \times \frac{P}{P \times P^{\frac{1}{2}}}$$

Since the temperature is unchanged, K will still equal $4.81(\text{atm})^{\frac{1}{2}}$. At 10 atm:

$$4.81 = \frac{(1-a)(2+a)^{\frac{1}{2}}}{a^{\frac{3}{2}}} \times \frac{1}{10^{\frac{1}{2}}}$$

Squaring and simplifying, we have:

$$230.4\,a^3 + 3a = 2$$

And the solution, by trial and error; a=0.185

Dissociation = 18.5 percent of the $CO_2$

This indicates that although K is independent of pressure of the reacting mixture, the actual fraction of the dissociated product usually varies with pressure.

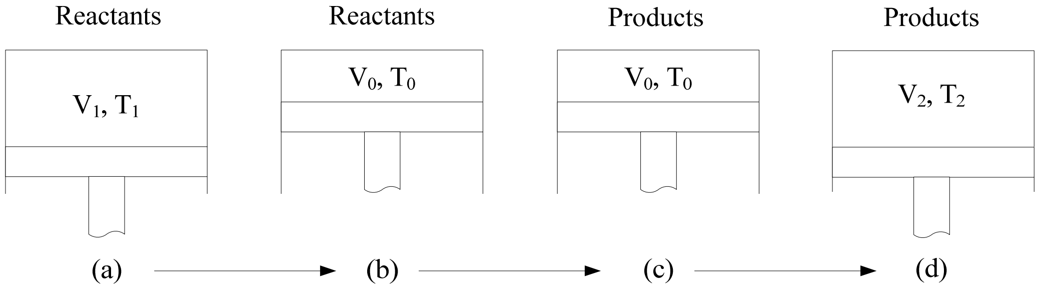
## Enthalpy of Formation $(\Delta h_f^o)$:

The enthalpy of formation of a chemical compound is; *the increase in enthalpy when a compound is formed from its constituent elements in their natural form and in a standard state.*

The natural state of the element is its most stable state. The normal form of oxygen ($O_2$) and hydrogen ($H_2$) are gaseous, so $(\Delta h_f^o)$ for these can be put equal to zero. The standard reference state(STP) is 25°C and 1 atm. pressure. Property values at the standard reference state are indicated by a superscript ($^o$) such as $h^o$, $u^o$, etc. It was found adequate to assume that the enthalpy of all elements is zero at the reference state. The enthalpy at any temperature and pressure is given by: $h_{PT} = \Delta h_f^o + h(T)$, h(T) is given in tables.

## First Law Applied to Combustion Process:

The first law of thermodynamics applies to any system, and the non-flow and steady- flow energy equation deduced from this law must be applicable to systems undergoing combustion processes. Since the internal energy is a function of state, a change in internal energy is independent of the path of the process. Consider a non-flow combustion process, starting with a mixture of fuel and air at state ($V_1$, $T_1$) and ending with products at state ($V_2$, $T_2$).



The change in internal energy between states 1 and 2 is:

$$U_2-U_1=U_{P2}-U_{R1}=(U_{P2}-U_{P0})+(U_{P0}-U_{R0})+(U_{R0}-U_{R1})$$

Where suffices R and P refer to reactants and products respectively, the first and third terms on the right – hand side represent changes of internal energy in processes not involving a chemical reaction, which can be evaluated by the methods applying to mixtures. The middle term has to be obtained from additional experimental information because it involves a chemical reaction.

Because b-c occurs at constant volume, no work is done and the non-flow energy equation becomes:

$$Q_{R_oP_o} = \left(U_{P_o} - U_{R_o}\right)$$

$\left(U_{P_o} - U_{R_o}\right)$Can there for be readily obtained by measuring the heat transferred in a constant – volume calorimetric process.

This term is called the "*internal energy of combustion at $T_o$*", or *the constant volume heat of combustion*, and it is symbolized by $\Delta U_0$, thus:

$$(U_{P_2} - U_{R_1}) = (U_{P_2} - U_{P_0}) + \Delta U_o + (U_{R_0} - U_{R_1})$$

$$U_{P_2} - U_{P_0} = \sum_P m_i c_{vi}(T_2 - T_o) = \sum_P m_i (u_{i2} - u_{io})$$

And

$$U_{R_0} - U_{R_1} = \sum_R m_i c_{vi}(T_o - T_1) = \sum_R m_i (u_{io} - u_{i1})$$

$m_i$ is the mass, and $c_{vi}$ the mean specific heat at constant volume of the constituent i.

Similar analysis can be made for a steady flow or constant pressure combustion process:

$$H_{P_2} - H_{R_1} = (H_{P_2} - H_{P_0}) + \Delta H_o + (H_{R_0} - H_{R_1})$$

Where $\Delta H_0$ = *enthalpy of combustion* at $T_o$, or the *constant pressure heat of combustion at $T_o$* and:

$$\Delta H_o = H_{P_o} - H_{R_o} \text{ ,and is always negative.}$$

$$H_{P_2} - H_{P_o} = \sum_P n_i (h_{i2} - h_{io}) \text{ ,to a mole basis}$$

$$= \sum_P m_i (h_{i2} - h_{io}) \text{ ,to a mass basis}$$

$$= \sum_P m_i c_{p_i}(T_2 - T_o) = (T_2 - T_o)\sum_P m_i c_{pi}$$

and

$$H_{R_o} - H_{R_1} = \sum_R n_i (h_{io} - h_{i1}) \text{ to a mole basis}$$

$$H_{R_o} - H_{R_1} = \sum_R m_i (h_{io} - h_{i1}) \text{ to a mass basis}$$

$$= \sum_R m_i c_{p_i}(T_o - T_1) = (T_o - T_1)\sum_R m_i c_{pi}$$

From the definition of the enthalpy of a perfect gas:

$H = U+PV = U + nR_oT$

$H_{Po}=U_{Po}+n_PR_oT_o$ and $H_{Ro}= U_{Ro}+ n_RR_oT_o$

$\therefore \Delta H_o = \Delta U_o + (n_p - n_R)R_oT_o$

If there is no change in number of moles (n) during the reaction, or if the reference temperature is absolute zero, then $\Delta H_0$ and $\Delta U_0$ will be equal.

**Example (3 – 3):**

Calculate $\Delta U_o$ in kJ/kg for the combustion of Benzene ($C_6H_6$) vapour at 25°C given that $\Delta H_0$= - 3169100 kJ/mole and the $H_2O$ is in the vapourphase.

**Solution:**

Combustion equation is: $C_6H_6+7\ 1/2\ O_2 \rightarrow 6CO_2+3H_2O$ (vap.)

$n_R = 1+7.5= 8.5,\ n_P = 6+3 = 9$

$$\Delta U_o = \Delta H_o - (n_p - n_R)R_oT_o$$

$$= -3169100-(1/2) \times 8.314 \times 298= -3170339 \text{ kJ/mol}$$

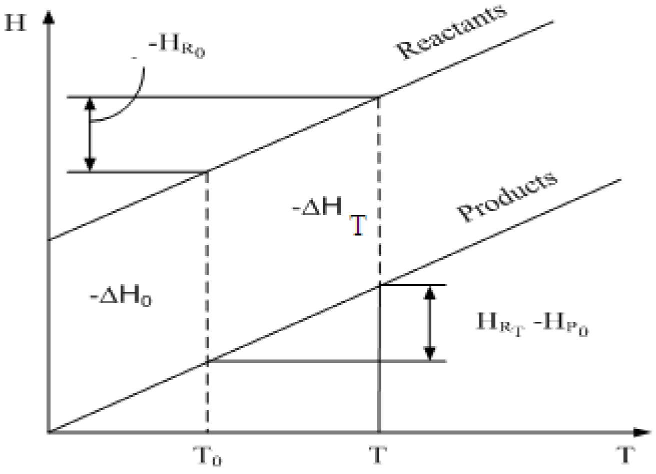
On mass basis, $\Delta U_o = \dfrac{-3170339}{(6\times 12+6\times 1)} = -40645$ kJ/kg

## Change in reference temperature:

Internal energy and enthalpy of combustion depend on the temperature at which the reaction occurs. This is due to the change in enthalpy and internal energy of the reactants and products with temperature.

It can be seen from the property diagram that the enthalpy of combustion at temperature T, $\Delta H_T$ can be obtained from $\Delta H_0$ at $T_o$ by the relationship:

$$-\Delta H_T = -\Delta H_o + (H_{R_T} - H_{R_o}) - (H_{P_T} - H_{P_o})$$



**Example (3 – 4):**

$\Delta H_0$For CO at 60$^o$C is given as -285200 kJ/mole. Calculate$\Delta H_T$ at 2500$^o$C given the enthalpies of the gases concerned in kJ/mole, are as follows:

| Gas | 60$^o$C | 2500$^o$C |
|---|---|---|
| CO | 9705 | 94080 |
| O | 9696 | 99790 |
| $CO_2$ | 10760 | 149100 |

**Solution:**

$$CO + \frac{1}{2} O_2 \rightarrow CO_2$$

$$-\Delta H_T = -\Delta H_o + (H_{R_T} - H_{R_0}) - (H_{P_T} - H_{P_o})$$

$$H_{R_0} = 1 \times 9705 + \frac{1}{2} \times 9696 = 14553 \quad kJ$$

$$\mathrm{H_{R_T}} = 1 \times 94080 + \frac{1}{2} \times 99790 = 143975 \quad \mathrm{kJ}$$

$$\mathrm{H_{P_0}} = 10760 \quad \mathrm{kJ} \quad \& \quad \mathrm{H_{P_T}} = 1 \times 149100 = 149100 \ \mathrm{kJ}$$

$$\therefore \Delta \mathrm{H_T} = -276282 \quad \mathrm{kJ/mole}$$

## Adiabatic flame temperature:

It is the temperature attained in combustion with no heat transfer, no work or changes in kinetic or potential energy; this is the maximum temperature that can be achieved.

For a given fuel and given pressure and temperature of the reactants, the maximum adiabatic flam temperature that can be achieved is with a stoichiometric mixture when complete combustion occurs.

**Example (3 – 5):**

Liquid heptane ($C_7H_{16}$) has an enthalpy of combustion of -44560 kJ/kg at 25$^o$C, when the water in the products is in the vapour phase. This fuel is to be burnt adiabatically in a steady stream of air in stoichiometric proportion. If the initial temperature of the fuel and air is 15$^o$C, estimate the temperature of the products. The specific heat of liquid heptane is 2.30kJ/kg K.

**Solution:**

The combustion equation for any hydrocarbon is:

$$C_nH_m+(n+\frac{m}{4})O_2+3.762(n+\frac{m}{4})N_2\rightarrow nCO_2+\frac{m}{2}H_2O+3.762(n+\frac{m}{4})N_2$$

i.e.

$C_7H_{16}$+11 $O_2$+41.382 $N_2$→ 7 $CO_2$+8 $H_2O$+41.382 $N_2$

The steady – flow energy equation:

$$\mathrm{H_R}+\Delta\mathrm{Q}=\mathrm{H_P}+\mathrm{W}$$

Since the process of combustion is adiabatic with no work, then:

$\mathrm{H_R}$=$\mathrm{H_P}$,ΔQ = 0 & W = 0

i.e. $(\mathrm{H_{Pt}}-\mathrm{H_{R15}})=0$

t represent products temperature. Expanding this equation we have:

$$(\mathrm{H_{Pt}}-\mathrm{H_{P25}})+\Delta\mathrm{H_{25}}+(\mathrm{H_{R25}}-\mathrm{H_{R15}})=0$$

$$\left(H_{R25}-H_{R15}\right)=\sum_R n_i\left(h_{i25}-h_{i15}\right)=n_{C7H16}+n_{O_2}\left(h_{25}-h_{15}\right)_{o_2}+n_{N_2}\left(h_{25}-h_{15}\right)_{N_2}$$

$$=\left\lfloor nC_p(25-15)\right\rfloor_{C_7H_{16}}+11(731.5-438.2)+41.382\times(728.4-437)$$

$$=100\times2.3\times10+3229+120581=17587kJ$$

$$\left(\mathrm{H_{pt}}-\mathrm{H_{p25}}\right)=\sum_{\mathrm{p}}\mathrm{n_1}\left(\mathrm{h_{it}}-\mathrm{h_{i25}}\right)$$

Assume T=1800 K, as a first estimate

$(\mathrm{H_{pt}}-\mathrm{H_{p25}})$ = 7 (80530 – 913.8) + 8 (63270 – 840.5) + 41.382 × 49750 – 728.4

= 7 × 79616 + 8 × 6249.5 + 41.382 × 49022

= 3085361.3 kJ

17636.5 – 100 × 44560 + 3085361.3 = -1354040

Assume T=2000 K

$(\mathrm{H_{pt}}-\mathrm{H_{p25}})$ =7(92610 – 913.8) + 8(73200 – 840.5) + 41.382 × (56920 – 728.4)

= 7 × 91696.2 + 8 × 72360 + 41.382 × 56192

= 3546073 kJ

It is clear the equation will not equal to zero, take t=2500

∴ $(\mathrm{H_{pt}}-\mathrm{H_{p25}})$ = 7(123335 – 913.8) + 8(98950 – 840.5) + 41.382 × (75280 – 728.4)

= 7 × 122421.4 + 8 × 98110.2 + 41.382 × 74551.6

$$= 7 \times 12242.1 + 8 \times 98110.2 + 41.382 \times 74551.6$$

$$= 4726921.2 \text{kJ}$$

| T | | ΔH |
|---|---|---|
| 1800 | − | 1354040 |
| 2000 | − | 894019 |
| 2500 | + | 282413.5 |

By linear interpolation, we have T=2380 K

## Calorific value of fuels:

The quantities $\Delta H_0$ and$\Delta U_0$ are approximated to, in fuel specification, by quantities called calorific values which are obtained by the combustion of the fuels in suitable apparatus. This may be of the constant volume type (e.g. *Bomb Calorimeter*) or constant pressure, steady flow type (e.g. *Boy's Calorimeter*).

1. **Higher (or Gross) Calorific value(H.C.V.) at constant volume**: It is the energy transferred at constant volume as heat to surroundings per unit quantity of fuel when burned at constant volume with the $H_2O$ product of combustion in the liquid phase,($Q_{gr,v}$). This approximate to -$\Delta U_0$at the reference temperature of 25°C with $H_2O$ in the liquid phase.

   i.e. $Q_{gr,\,v} = -\Delta U_O$

   If $H_2O$ products are in the vapour phase the energy released per unit quantity of fuel is called the **lower (or net) calorific value** (L.C.V).

2. **Higher calorific value at constant pressure**: It is the energy transferred at constant pressure as heat to the surrounding (cooling water) per unit quantity of fuel when burned at constant pressure with the $H_2O$ products of combustion in the liquid phase, $Q_{gr,p} = -\Delta H_0$ at the reference temperature of 25°C with the $H_2O$ in the liquid phase.

   *Lower calorific value (L.C.V) at constant pressure*; is the energy released with $H_2O$ products in the vapour phase.

$$Q_{gr,v} = Q_{net,v} + m_c v_{fg}$$

$$Q_{gr,p} = Q_{net,p} + m_c h_{fg}$$

And

$u_{fg}$ at 25°C for $H_2O$=2304.4 kJ/kg

$h_{fg}$ at 25°Cfor $H_2O$ =2441.8 kJ/kg

$m_c$: mass of condensate per unite quantity of fuel.

**Example (3 – 6):**

The calorific value $Q_{gr,v}$ of kerosene, whose ultimate analysis is 86 per cent carbon and 14 per cent hydrogen, was found by experiment to be 46890 kJ/kg. Calculate the other three calorific values.

**Solution:**

$$C_{7.167}H_{14}+10.667\ O_2 \rightarrow 7.167\ CO_2+7\ H_2O$$

On mass basis:

$$100\times 1\ C_{7.167}H_{14} + 32\times 10.667\ O_2 \longrightarrow 44\times 7.167\ CO_2 + 18\times 7\ H_2O$$

For one kg of fuel:

$$1\text{ kg fuel} + 3.41\text{ kg }O_2 \rightarrow 3.15\text{ kg }CO_2 + 1.26\text{ kg }H_2O$$

$Q_{net,v} = Q_{gr,v} – m_c\, u_{fg} = 46890 – 1.26 \times 2304 = 43987$ kJ/kg

The relation between $Q_{gr,v}$ and $Q_{gr,p}$ is identical with that between $-\Delta H_0$ and $-\Delta U_0$, i.e.

$Q_{gr,p} = Q_{gr,v} + R_oT_o(n_P-n_R)$

$= 46890 + 8.314\times 298\left(\frac{3.15}{44}-\frac{3.41}{32}\right)$, fuel & $H_2O$ are liquids

= 46890+87 = 46977 kJ/kg

And

$Q_{net,p} = Q_{gr,p} - m_ch_{fg} = 46977-1.26 \times 2442 = 43900$ kJ/kg

## *CHAPTER (4)*

# Fuel Air Cycle

The theoretical cycle based on the actual properties of the cylinder contents is called the fuel – air cycle. The fuel – air cycle take into consideration the following:

1- The actual composition of the cylinder contents.
2- The variation in the specific heat of the gases in the cylinder.
3- The dissociation effect.
4- The variation in the number of moles present in the cylinder as the pressure and temperature change.
5- No chemical changes in either fuel or air prior to combustion.
6- Combustion takes place instantaneously at top dead center.
7- All processes are adiabatic.
8- The fuel is mixed well with air.

### **Variation of specific heats:**

All gases, except mono-atomic gases, show an increase in specific heat as temperature increase. The specific heat may be written in the form:

$$C = a+bT+cT^2$$

where T is the absolute temperature and a, b and c are constants for any specific gas.

$C_p$ and $C_v$ increase with temperature but $\gamma = \frac{c_p}{c_v}$ decrease as the temperature increase. There are special tables and charts which gives the specific heat of different gases at different temperatures. Specific heats of a mixture of gases can be calculated if the constituents of the mixture are known, using the gas mixture relations.

If the variation of specific is taken into account during the compression stroke, the final pressure and temperature would be lower if constant value of specific heats is used ($\bar{2}$) as shown in Figure (4-1).When taking variable specific heat, end of combustion will be ($\bar{3}$) instead of 3. Expansion process would be $\overline{34}$ when assumed isentropic, but expansion taking variable specific heats into accounts is above $\overline{34}$ and represented by $\overline{\overline{34}}$. Thus it is seen that the effect of variation of specific heats is to deliver less work.

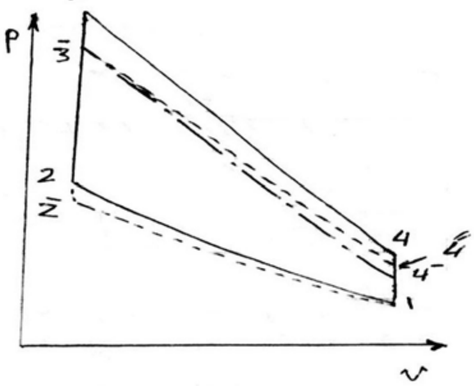


**Figure (4-1):** *Loss due to variation of specific heat*

## Dissociation effect:

The effect of dissociation is a suppression of a part of the heat release during combustion and the liberation of it as expansion proceeds, a condition similar to the effect produced by the variation of specific heats. However, the effect of dissociation is much smaller than that of change of specific heats.

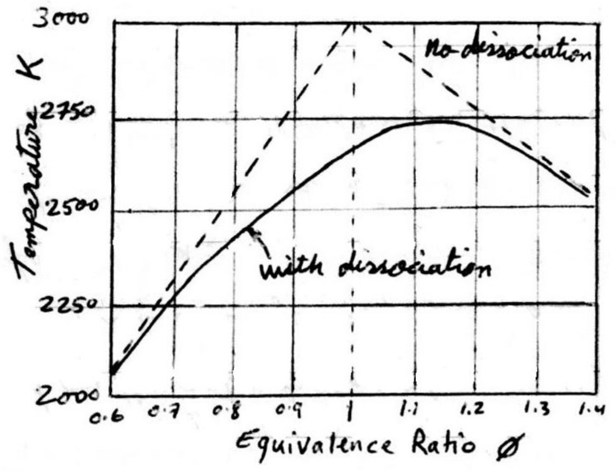


***Figure (4-2):*** *Effect of dissociation temperature at different $\phi$*

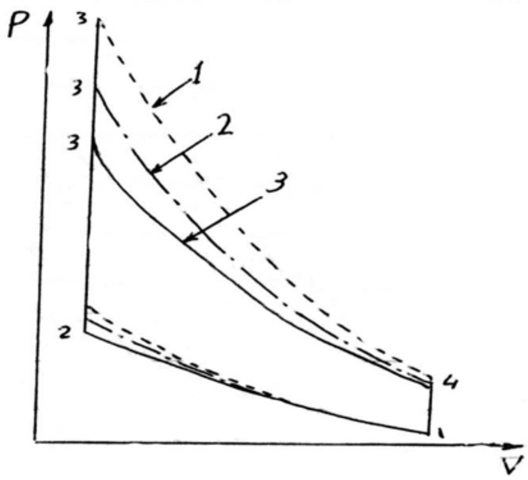


***Figure (4-3):*** *Comparison of air-Standard cycle with fuel – air cycle*

The effect of dissociation on combustion temperature is as shown in figure (4-2), the dotted line represents the maximum combustion temperature attained with no dissociation, and the full line is with dissociation. Dissociation reduces the maximum temperature by about 300°C.

## Comparison of P-V Diagram of Air-standard and Fuel – Air cycle for SI engine:

Diagram(4 – 3) shows three constant volume cycles:

1- Air – standard cycle.
2- Air – standard with variable specific heat.
3- Fuel – Air cycle with variable specific heat and dissociation.

## Example (4-1):

What is the percentage change in the efficiency of Otto cycle having a compression ratio of 7, if the specific heat at constant volume increases by 1%?

**Solution:**

$$\eta_{otto} = 1 - \frac{1}{r^{\gamma - 1}}$$

$$C_P - C_v = R$$

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

$$\eta = 1 - \frac{1}{r^{\frac{R}{C_v}}} \qquad \Rightarrow 1 - \eta = r^{-\frac{R}{C_v}}$$

$$\ln(1-\eta) = -\frac{R}{C_v} \ln r \qquad , r = \text{constant}$$

Differentiating:

$$\frac{1}{1-\eta} d\eta = \frac{-R}{C_v^2} \ln r \times d\, C_v$$

$$d\eta = -1(1-\eta)(\gamma - 1) \ln r \frac{dC_v}{C_v}$$

$$\frac{d\eta}{\eta} = -\frac{(1-\eta)}{\eta}(\gamma - 1) \ln r \times \frac{dC_v}{C_v}$$

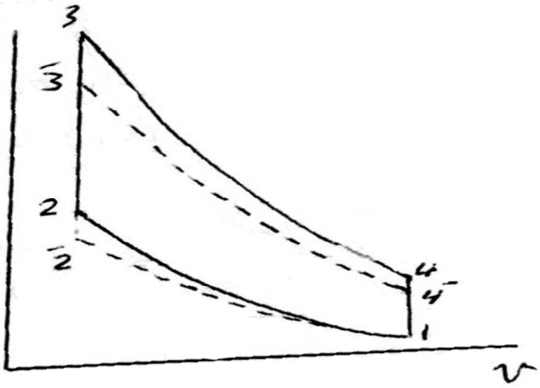
$$\eta = 1 - \frac{1}{7^{0.4}} = 0.54$$

$$\therefore \frac{d\eta}{\eta} = -\frac{(1-0.54)}{0.54} \times (1.4-1) \times \ln 7 = -0.663 \quad \text{(decrease)}$$

**Example (4-2):**

A petrol engine of compression ratio 6 uses a fuel of calorific value43950kJ/kg. The air – fuel ratio is 15:1. The temperature and pressure of the charge at the charge at the end of the suction stroke are 60°C and 1 bar. Determine the maximum pressure in the cylinder if the index of compression is 1.32 and the specific heat at constant volume is expressed by the expression; $C_v = 0.71 + 19 \times 10^{-5} T$ kJ/kg K, where T is the temperature in K.
Compare this value with that when constant specific heat $C_v = 0.72$ is used.

**Solution:**



$$p_1 v_1^n = p_2 v_2^n$$

$$p_2 = p_1 \left( \frac{v_1}{v_2} \right)^n = 1 \times 6^{1.32} = 10.645 \ bar$$

$$T_2 = T_1 \left( \frac{p_2 v_2}{p_1 v_1} \right) = 333 \times \frac{10.645}{6}$$

$$= 590.8 \text{K}$$

Mean specific heat $C_{v_{mean}} = 0.71 + 19 \times 10^{-5} \left[ \frac{T_2 + T_3}{2} \right]$

Assume 1 kg of air in the cylinder, heat added per kg air$= \frac{43950}{15}$

$Q = C_v \times$ mass of charge $\times(T_3-T_2)$

$$2930= [0.71+19 \times 10^{-5} (T_3+590.8)/2] \times\frac{16}{15} \times(T_3-590.8)$$

Solving we get $T_3$=3090 K

$$P_3 = P_2 \frac{T_3}{T_2} = 10.645\times \frac{3090}{590} = 55.75\text{bar}$$

For constant specific heat, $2930=0.72 \times\frac{16}{15}(T_3- 590)$

Solving we get $T_3$=4405 K

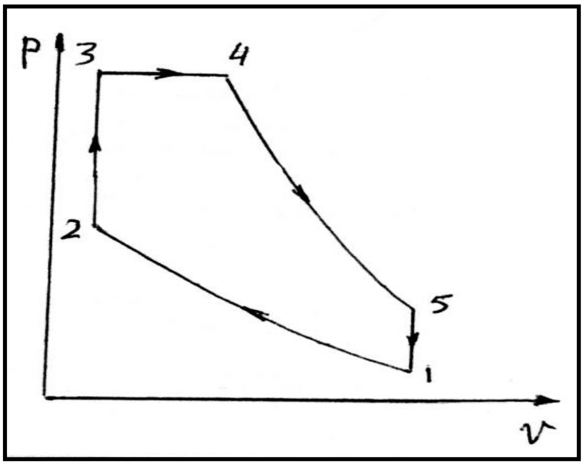
$$P_3=10.645 \times\frac{4405}{590} = 79.5 \text{ bar}$$

**Example (3 - 3):**

In an oil engine, working on dual combustion cycle, the temperature and pressure at the beginning of compression are 90°C and 1 bar. The compression ratio is 13:1. The heat supplied per kg of air is 1674 kJ, half of which is supplied at constant volume and half at constant pressure. Calculate (i) the maximum pressure in the cycle (ii) the percentage of stroke at which cut-off occurs.

Take $\gamma$ for compression 1.4, R= 0.293 kJ/kg K and $C_v$ for products of combustion $(0.71+12 \times 10^{-5}T)$

**Solution:**



$$(i)\ P_2 = P_1\left(\frac{v_1}{v_2}\right)^{\gamma} = 1\times(13)^{1.4} = 36.3 \text{ bar}$$

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{\gamma-1} = 363(13)^{0.4} = 1013K$$

$$Q_{12} = m\int_{T_2}^{T_3} C_v dT = m\int_{T_2}^{T_3}(0.71+12\times10^{-5}T)dT$$

Neglect mass of fuel;

$$837 = 1\left[0.71T+12\times10^{-5}\frac{T^2}{2}\right]_{T_2}^{T_3}$$

$$=0.71(T_3-1013)+\frac{12\times10^{-5}}{2}(T_3^2-1013^2)$$

$$=0.00006T_3^2+0.71T_3-1617.8$$

Solving we get $T_3$=1955.6 K

$$\therefore \text{Maximum pressure} = 36.3 \times\frac{1955.6}{1013} = 70.1 \text{ bar}$$

$(ii)\ C_p = C_v + R = 1.003 + 12\times10^{-5}T$

$$Q_{34} = m\int_{T_3}^{T_4} C_p dT = m\int_{T_3}^{T_4}(1.003 + 12\times10^{-5}T)dT$$

$$837 = 1\left[1.003\,T + 12\times10^{-5}\,\frac{T^2}{2}\right]_{1864}^{T_4}$$

$$837 = 1.003(T_4 - 1864) + 0.00006(T_4^2 - 1955.6^2)$$

$$= 1.003\,T_4 + 0.00006\,T_4^2 - 3028$$

$$0.00006\,T_4^2 + 1.003\,T_4 - 3865 = 0$$

Solving we get, $T_4$=2611 K

$$\frac{v_4}{v_3} = \frac{T_4}{T_3} = \frac{2611}{1955.6} = 1.335$$

$$\text{Cut – off} = v_4 - v_3 \qquad , \frac{v_1}{v_2} = 13$$

% of stroke at which cut off occurs = $\frac{v_4 - v_3}{v_1 - v_2}$

$$= \frac{1.335\,v_3 - v_3}{13\,v_2 - v_2} = \frac{(1.335-1)v_2}{12\ v_2} = 2.793\%, \qquad (V_3 = V_2)$$

**Example ( 3 – 4 ):**

A mixture of Heptane ( $C_7H_{16}$ ) and air which is 10% rich is initially at a pressure of 1 atm. and a temperature of 100°C, is compressed through a volumetric ratio of 6:1. It is then ignited and adiabatic combustion proceeds at constant volume. The maximum temperature reached is 2627°C and at this temperature the equilibrium constants are;

$$\frac{P_{H2O}.P_{CO}}{P_{CO2}P_{H2}} = 6.72 \qquad and \qquad \frac{P^2{}_{CO}.P_{O2}}{P^2{}_{CO2}} = 0.054\ atm.$$

If the constituent of the product gases are; $CO_2$, CO, $H_2O$, $H_2$, $O_2$ and $N_2$, show that approximately 30.2% of the carbon has burned incompletely. Pressures are measured in atmospheres.

**Solution:**

The stoichiometric combustion equation for a hydrocarbon fuel is:

$$C_nH_m + (n + m/4)[O_2 + 3.762\,N_2] \rightarrow n\,CO_2 + (m/2)\,H_2O + (n + m/4)\times 3.762\,N_2$$

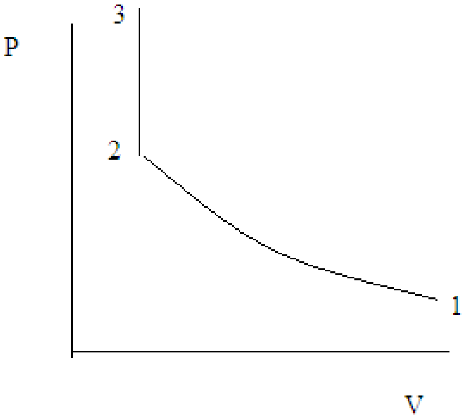
$$\therefore \qquad C_7H_{16} + 11(O_2 + 3.762\,N_2) \rightarrow 7\,CO_2 + 8\,H_2O + 41.36\,N_2$$

Actual combustion equation includes CO and H2 in the products also;

So; actual combustion equation may be written in the form:

$$C_7H_{16} + \frac{10}{11}( 11O_2 + 41.36\, N_2 ) \rightarrow a\, CO_2 + b\, CO + c\, H_2O + d\, H_2 + e\, O_2 + 37.6\, N_2$$

Atom balance:



$O_2 : a + b/2 + c/2 + e = 10$

$C : a + b = 7$

$H_2 : c + d = 8$

$n_1 = 1 + 10 + 37.6 = 48.6$

$n_3 = a + b + c + d + e + 37.6$

$P_1V_1 = n_1 R_oT_1$ and $P_3V_3 = n_3R_oT_3$

By division: $\frac{P_3}{n_3} = \frac{P_1}{n_1}\frac{V_1}{V_3}\frac{T_3}{T_1} = \frac{1}{48.6}\frac{6}{1}\frac{2900}{373} = 0.958$

$P_{CO_2} = \frac{a}{n_3} P_3$; $P_{CO} = (b/n_3) P_3$; $P_{H2O} = (c/n_3) P_3$; $P_{H2} = (d/n_3) P_3$; and $P_{O2} = (e/n_3) P_3$

$$\frac{P_{H2O}.P_{CO}}{P_{CO2}P_{H2}} = 6.72 \qquad \frac{P^2{}_{CO}.P_{O2}}{P^2{}_{CO2}} = 0.054\, atm.$$

$\therefore \frac{b.c}{a.d} = 6.72$ ……..(1) & $\frac{b^2}{a^2}\frac{e}{n_3} P_3 = 0.054$ ………(2)

The proportion of carbon incompletely burned is : $\frac{b}{7} = 0.302$

$\therefore$ b = 2.114 , then: a = 7 – 2.114 = 4.886

Substituting in (1), gives:

$$\frac{c}{d} = 6.72 \times \frac{a}{b} = 6.72 \times \frac{4.886}{2.114} = 15.5$$

C + d = 8, i.e. 15.5d + d = 8, then: d = 8/16.5 = 0.485

And c = 15.5 x 0.485 = 7.515

e = 10 – a – b/2 – c/2 = 10 – 4.886 – 2.114/2 – 7.515/2

= 0.297

Substitute in (2) gives:

$$\frac{b^2}{a^2}\frac{e}{n_3} P_3 = (\frac{2.114}{4.886})^2 \times 0.297 \times 0.958 = 0.0531$$

Which gives sufficient agreement to the 0.054 quoted, showing that approximately 30.2% of carbon was burned to CO.

### *CHAPTER (5)*

# The Actual Cycle

The actual cycle experienced by internal combustion engines is an open cycle with changing composition, actual cycle efficiency is much lower than the air standard efficiency due to various losses occurring in the actual engine. These losses are as follows:

1- Losses due to variation of specific heats with temperature: already discussed.
2- Losses due to dissociation: already discussed.
3- time losses:

In theoretical cycles the burning is assumed to be instantaneous. Whereas, in actual cycle, burning is completed in a definite interval of time. The effect of this time is that the maximum pressure will not be produced when the volume is minimum; but sometime after T.D.C., causes a reduction in the work produced.

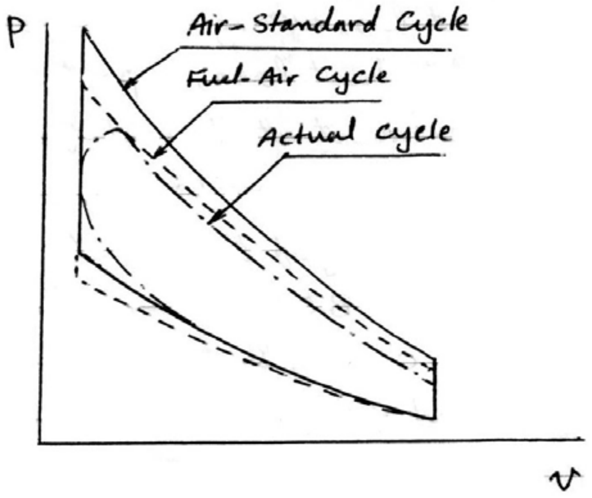


***Figure (5-1):*** *The effect of time losses on P-V diagram*

In order that the maximum pressure is not reached too late in the expansion stroke, the time at which burning starts is varied by varying the spark timing (spark advance).

### 4- incomplete combustion loss:

Fuel vapour, air, and residual gas are present in the cylinder, this makes it impossible to obtain perfect homogeneous mixture.. Therefore some fuel does not burn to $CO_2$ or partially burns to CO, and $O_2$ will appear in the exhaust. Energy release in actual engine is about 90 to 93% of fuel energy input.

### 5- direct heat loss:

During combustion process and subsequent expansion stroke, the heat flows from cylinder gases through cylinder walls and cylinder head into the water jacket or cooling fins. Some heat enters the piston head and flows through piston rings into the walls of the cylinder or carried away by the engine oil.

The heat loss during combustion and expansion does not represent a complete heat loss; a part of the heat loss would be rejected in the exhaust at the end of the expansion stroke.

### 6- exhaust blow down loss:

The opening of the exhaust value before B.D.C. reducing cylinder pressure, causing the roundness of the end of the P-V diagram, this means a reduction in the work done per cycle.

### 7- pumping losses:

Pumping loss is due to expelling the exhaust gases and the induction of the fresh charge. In naturally aspirated engine this would be a negative work.

### 8- Friction losses:

These losses are due to the friction between the piston and cylinder walls, the various bearings, and the friction in the auxiliary equipment, such as pumps, fans, etc…

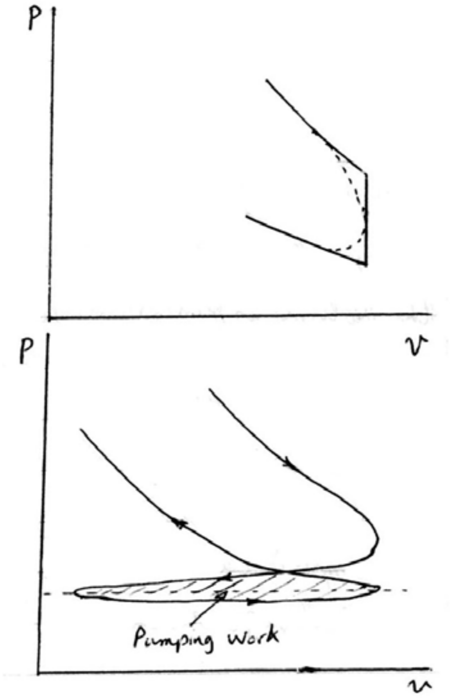


**Figure (5-2, 3):** *Effect of exhaust valve opening time and pumping*

## Effect of throttle opening:

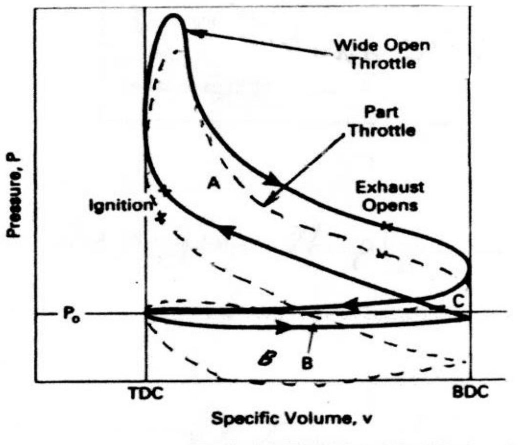


**Figure (5-4):** *Effect of throttle opening on the indicator diagram*

When a four- stroke SI engine is run at partially closed throttle, (throttle is a butterfly valve in the intake system), fuel supplied to the engine is reduced, and this would lead to less power output at part throttle opening as shown in figure (5-4). The upper loop represent positive work output (A) while the lower loop consisting of the exhaust and intake strokes is negative work (B). The more closed the throttle position, the lower will be the pressure during the intake stroke and the greater the negative pumping work.

## Effect of spark timing:

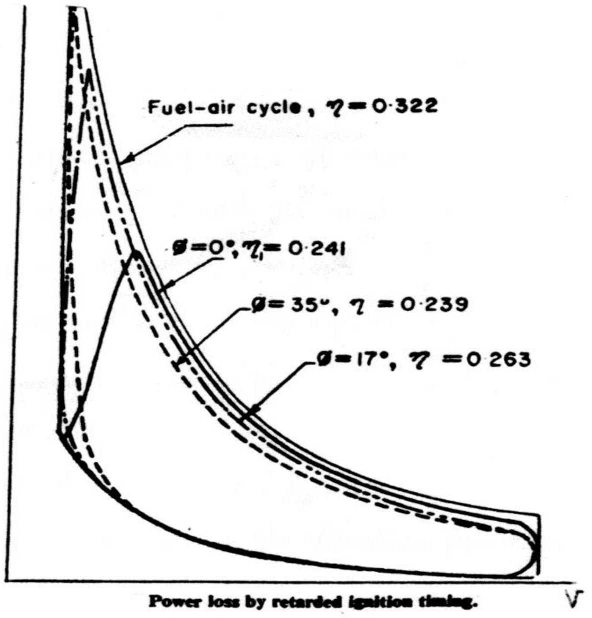


***Figure (5-5):*** *Power loss by retarded ignition timing*

The effect of spark timing is shown in figure (5-5), ϕ = 0 means spark timing at T.D.C, in this case the peak pressure is low and occurs nearly 40° after T.D.C., as spark timing is advanced to achieve combustion at TDC, additional work is required to compress the burning gases. Figure (5-5) shows the power loss by retarded ignition timing optimum loss in power. In actual practice a deliberate retard in spark from optimum may be done to avoid knocking and reduce exhaust emissions of HC and CO.

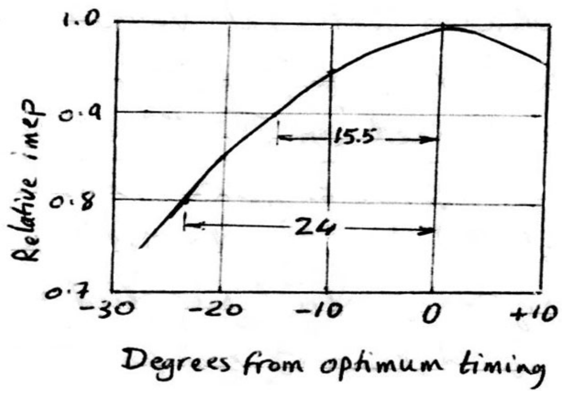


***Figure (5-6):*** *Variation of imep with spark timing*

# *CHAPTER (6)*

## ENGINE PERFORMANCE

The basic performance parameters of internal combustion engine (I.C.E) may be summarized as follows:

### 1. Indicated power (i.p.):

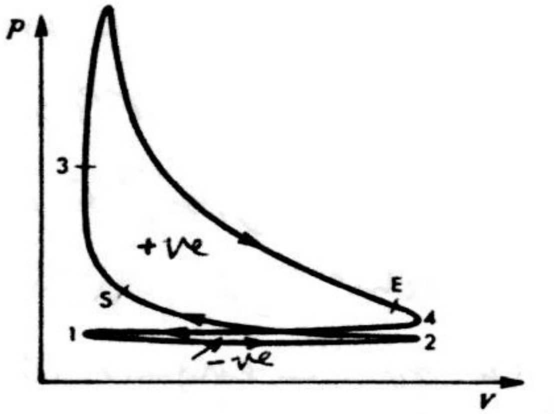


***Figure (6-1):*** *indicator diagram of SI engine*

It is the actual rate of work done by the working fluid on the piston. As its name implies, the i.p. can be determined from an "indicator diagram" as show in figure (6-1), by subtracting the pumping loop area (- ve) from the positive area of the main diagram. i.p. power could be estimated by performing a Morse test on the engine. The physical equation for the i.p. is:

$$\text{i.p.} = P_m LAN$$

where N is the number of machine cycles per unit times, which is 1/2 the rotational speed for a four- stroke engine, and the rotational speed for a two- stroke engine.

### 2. Brake power (b.p.):

This is the measured output of the engine. It is usually obtained by a power absorption device such as a brake or dynamometer which can be loaded in such a way that the torque exerted by the engine can be measured. The break power is given by:

$\text{b.p.} = 2\pi NT$

Where T is the torque

### 3. Friction power (f.p.) and Mechanical efficiency ($\eta_m$):

The difference between the i.p. and the b.p is the friction power (f.p.) and is that power required to overcome the frictional resistance of the engine parts,

$$\text{f.p.} = \text{i.p.} - \text{b.p.}$$

The mechanical efficiency of the engine is defined as: $\eta_m = \dfrac{\text{b.p.}}{\text{i.p.}}$

$\eta_m$ is usually between 80% and 90%

### 4. Indicated mean effective pressure (imep):

It is a hypothetical pressure which if acting on the engine piston during the working stroke would results in the indicated work of the engine. This means it is the height of a rectangle having the same length and area as the cycle plotted on a p- v diagram.

$$\text{imep } (P_i) = \frac{\text{Net area of the indicator diagram}}{\text{Swept volume}} \times \text{Indicator scale}$$

Consider one engine cylinder:

$$\text{Work done per cycle} = P_i \, AL$$

where: A = area of piston; L = length of stroke

$$\text{Work done per min.} = \text{work done per cycle} \times \text{active cycles per min.}$$

$$\text{i.p.} = P_i \, AL \times \text{active cycles/ min}$$

To obtain the total power of the engine this should be multiplied by the number of cylinder n, i.e.:

$$\text{Total i.p.} = P_i \, AL \, Nn/2 \qquad \text{for four- stroke engine}$$

and

$$= P_i ALNn \qquad \text{for Two- stroke engine}$$

### 5. Brake mean effective pressure (bmep) and brake thermal efficiency:

The bmep ($P_b$) may be thought of as that mean effective pressure acting on the pistons which would give the measured b.p., i.e.

$$\text{b.p.} = P_b \, AL \times \text{active cycles/ min}$$

The overall efficiency of the engine is given by the brake thermal efficiency, $\eta_{BT}$ i.e.

$$\eta_{BT} = \frac{\text{Brake power}}{\text{Energy supplied}}$$

$$\eta_{BT} = \frac{b.p.}{\dot{m}_f \times Q_{net}}$$

where $\dot{m}_f$ is the mass of fuel consumed per unit time, and $Q_{net}$ is the lower calorific value of the fuel.

### 6. Specific fuel consumption (s.f.c.):

It is the mass of fuel consumed per unit power output per hour, and is a criterion of economic power production.

$$sfc = \frac{\dot{m}_f}{b.p.} \; kg/kWh$$

Low values of s.f.c are obviously desired. Typical best values of bsfc for SI engines are about 270g/kW.h, and for C.I. engines are about 200g/kW.h.

### 7. Indicated thermal efficiency ($\eta_{IT}$):

It is defined in a similar way to $\eta_{BT}$

$$\eta_{IT} = \frac{i.p.}{\dot{m}_f \times Q_{net}}$$

Dividing $\eta_{BT}$ by $\eta_{IT}$ gives

$$\frac{\eta_{BT}}{\eta_{IT}} = \frac{\text{b.p.}}{\text{i.p.}} = \eta_m$$

$$\therefore \eta_{BT} = \eta_m \times \eta_{IT}$$

**8. Volumetric efficiency ($\eta_v$):**

Volumetric efficiency is only used with four- stroke cycle engines. It is defined as the ratio of the volume if air induced, measured at the free air conditions, to the swept volume of the cylinder:

$$\eta_v = \frac{\dot{v}}{v_s}$$

The air volume $(\dot{V})$may be refereed to N.T.P. to give a standard comparison.

The volumetric efficiency of an engine is affected by many variables such as compression ratio, valve timing, induction and port design, mixture strength, latent heat of evaporation of the fuel, heating of the induced charge, cylinder pressure, and the atmospheric conditions.

**Example 6-1:**

The peak pressure of a SI engine rotating at 1500 rpm occurs 0.003S after the spark, what will be the spark timing when peak pressure is at TDC. If the inlet valve opens at 10 degrees bTDC and closes at 45 degrees aBDC, how long the inlet valve opening period is in seconds.

**Solution:**

Number of revolutions per second $= \frac{1500}{60} = 25$ rev.

Number of revolutions between spark timing and TDC = 25 × 0.003 = 0.075 rev.

Crank shaft angle during this period = 0.075 × 360 = 27

i.e. spark must occurs 27 degree bTDC

inlet valve opening = 10 + 180 + 45 = 235 degrees

inlet valve opening time in seconds $= \frac{235}{360 \times 25} = 0.0265$.

**Example 6-2:**

In a four stroke single cylinder gas engine the indicated mean effective pressure is 0.46 $MN/m^2$, the brake power 9 kW, speed 250 rpm, mechanical efficiency, $\eta_m$ = 0.8, and bore to stroke ratio = 0.66. Calculate cylinder diameter and mean piston speed.

**Solution:**

$$\eta_m = \frac{bp}{i.p} \quad , \ \text{i.p} = \frac{9}{0.8} = 11.25 \ \text{ kw}$$

$$\text{i.p} = \frac{\text{P}_\text{i} LANn}{2}$$

$$LA = \frac{2 \times i.p}{P_i Nn} = \frac{2 \times 11.25}{0.46 \times 1000 \times 250 \times 1} = 0.01174 \quad \text{m}^3$$

$$\frac{d}{L} = 0.66 \quad , \ \text{L} = \frac{\text{d}}{0.66}$$

$$\therefore \frac{d}{0.66} \times \frac{\pi}{4} d^2 = 0.01174$$

$$\text{d}^3 = 0.009866$$

$$\text{d} = 0.2145 \ \text{ m}$$

Mean piston speed = $\frac{2LN}{60} = \frac{2 \times 0.2145 \times 250}{0.66 \times 60} = 2.71 \quad \text{m/s}$

**Example 6-3:**

A four stroke petrol engine delivers 35.75kW with a mechanical efficiency of 80%, the fuel consumption of the engine is 0.4 kg per brake power hour, and the A/F ratio is 14:1.The heating value of the fuel is 41870 kJ/kg. Find: (a) i.p, (b) f.p., (c) $\eta_{BT}$, (d) $\eta_{IT}$, (e) fuel consumption per hour, (f) air consumption per hour.

**Solution:**

a) $\eta_m = \frac{b.p}{i.p} \quad , \text{i.p} = \frac{35.75}{0.8} = 44.7 \ \text{ kW}$

b) f.p= i.p – b.p=44.7– 35.75=8.95kW

c) $\eta_{BT} = \frac{b.p}{Q_{added}} = \frac{35.75 \times 3600}{0.4 \times 35.75 \times 41870} = 0.215$

d) $\eta_{BT} = \eta_{IT} \times \eta_m$

$$\eta_{IT} = \frac{0.215}{0.8} = 0.2687$$

e) fuel consumption per hour = 0.4 ×35.75=14.32kg

f) air consumption per hour = 14.32 × 14=200.5kg

**Example 6-4:**

The air flow to a four cylinder four – stroke engine is 2.15 $\text{m}^3$/min. During a test on the engine the following data were recorded:

Bore 10.5cm; stroke 12.5cm; engine speed 1200 rpm, torque 150 N.m, fuel consumption 5.5 kg/h, calorific value of fuel, 43124 kJ/kg, ambient temperature and pressure are 20°C and 1.03 bars. Calculate:

1- The brake thermal efficiency.
2- The brakes mean effective pressure.
3- The volumetric efficiency.

**Solution:**

$$1\text{-}\ b.p.=\frac{2\pi NT}{60}=\frac{2\pi\times1200\times150}{60\times1000}=18.85\ \ \text{kW}$$

$$\eta_{BT}=\frac{b.p}{Q_{added}}=\frac{18.85\times3600}{5.5\times43124}=0.286$$

$$2\text{-}\ b.p=\frac{P_iLAN_n}{2}$$

$$\therefore P_i=\frac{2\times18.85\times4\times60}{0.125\times\pi\times(0.105)^2\times1200\times4}=435.4\quad\text{kPa}$$

$$3\text{-}\ \eta_v=\frac{\dot{V}}{V_s}$$

$$V_s=LA\frac{Nn}{2}=0.125\times\frac{\pi}{4}(0.105)^2\times\frac{1200\times4}{2}=2.6\quad m^3/\min$$

$$\therefore \eta_v=\frac{2.15}{2.6}=0.83$$

## Testing of Internal Combustion Engines:

There are a wide variety of engine tests, starting from simple fuel and air- flow measurements to taking of complicated injector needle lift diagram, swirl patterns and photographs of the combustion process, etc..

Here only certain basic tests and measurement will be considered.

**1-Measurement of speed:**

A wide variety of speed measuring devices are available they range from a mechanical tachometer to digital and triggered electrical tachometers.

The best method of measurement is to count the number of revolution in a given time; this could be done either mechanically or electrically.

**2-Fuel consumption measurement:**

The fuel consumption of an engine is measured by determining the volume flow in a given time interval, or to measure the time required for the consumption of a given volume (or mass) of fuel.

**3-Air consumption measurement:**

The measurement of the air flow in the engine intake is not an easy task, because of the cyclic nature of the engine which causes a pulsating air flow:

a) Air box method:

In this method the intake air is drawn from a large surge tank, and measurement of air flow into the surge tank is performed using a calibrated orifice or a flow nozzle (see fig.6-2).

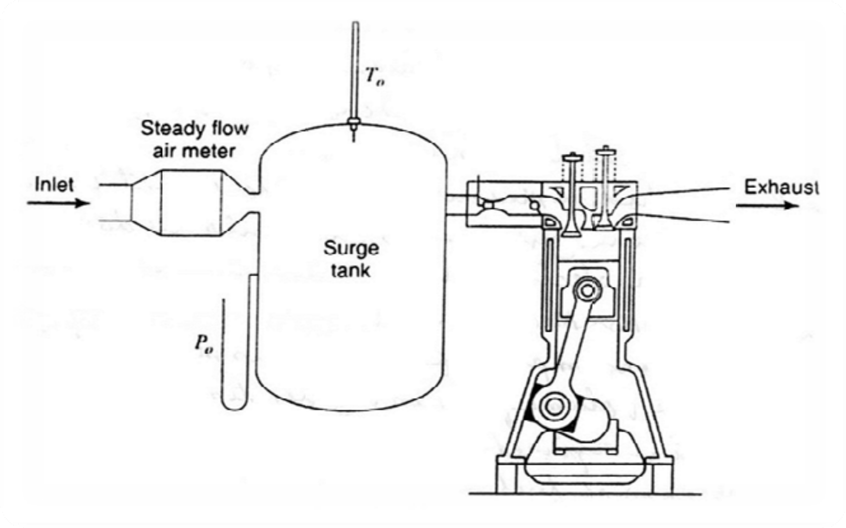


***Figure (6-2):*** *Testing equipment for measured of air consumption*

## b) Positive – Displacement meters:

Positive – displacement meters are very accurate, their working principle is shown in figure (6-3), as the impellers rotate, a fixed volume of air is alternately trapped between each impeller and the casing. This occurs four times for each complete revolution of both impellers.

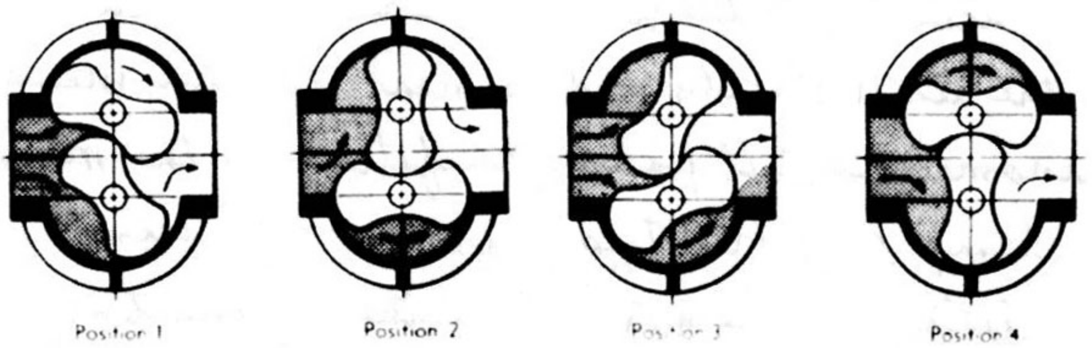


***Figure (6-3):****Rotary positive displacement meter*

## c) Viscous – flow air meter:

The meter is show diagrammatically in figure 6-4. It uses an element where viscous resistance is the principle source of pressure loss and kinetic effects are small. With the air box the flow is proportional to the square root of the pressure difference, while the viscous resistance is directly proportional to the air velocity and is measured by means of an inclined manometer. Felt pads are fitted in the manometer connections to damp out fluctuations, an additional damping vessel is fitted between the meter and the engine to increase the accuracy by reducing the effect of pulsations.

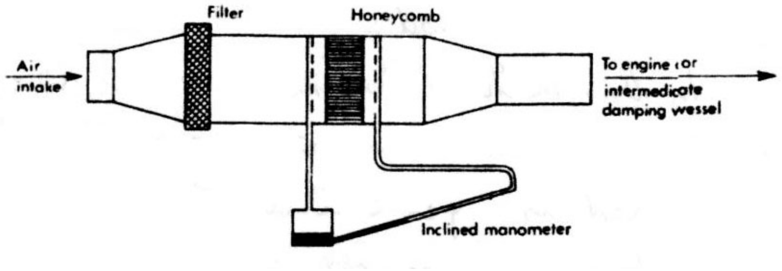


***Figure (6-4):*** *Viscous- flow air meter*

## 4-Measurement of engine torque and power:

Any apparatus that permits the measurement of torque and power of the engine is called a "***dynamometer***". There are many types of dynamometers; all operate on the principle illustrated in fig. (6-5). Here the rotor (a), driven by the engine to be tested, is couple (electrically, magnetically, hydraulically or by friction) to the stator (b). In one revolution of the shaft, the peripherally of the rotor moves through a distance($2\pi r$) against the coupling force f (drag force).

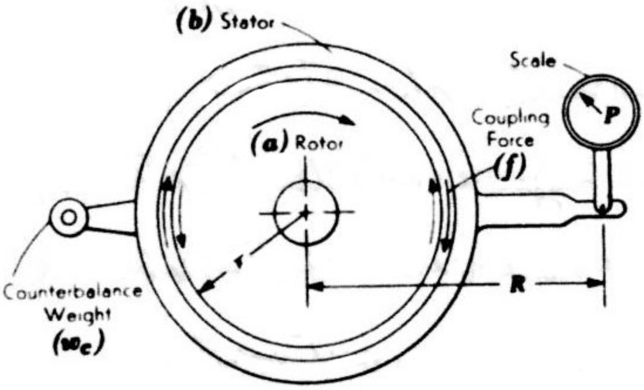


***Fig.6-5*** *The dynamometer principle*

Thus the work per revolution is:

$$\text{Work} = 2\pi r f$$

The external moment, which is the product of the reading p of the scale (could be a beam balance or weights) and the arm R, must just balance the turning moment, which is r×f;

$$r \times f = R \times P$$

$$\text{Work} = 2\pi RP$$

Work per minute $= 2\pi RPN$ (N is engine speed in rpm)

Power is defined as the time rate of doing work, i.e.

$$\text{Power} = 2\pi RPN$$

where R in meters and P in Newton, then

$$power = \frac{2\pi RPN}{1000 \times 60} \quad kW$$

## a) Fluid Dynamometers:

They absorb engine energy in water or oil pumped through orifices or dissipated with viscous losses in a rotor– stator combination. Large energy can be absorbed in this manner. Fluid brakes fall into two classes; the “friction" and the “agitator" type. In the friction type the coupling force arises from the viscous shearing of fluid between the rotor and stator, while in the agitator type the coupling force arises from the change in momentum of fluid as it is transported from rotor vanes to the stator vanes and back again.

Figure (6-6) illustrates Heenan – Froude hydraulic dynamometer. Here, the vanes of the rotor direct the water outward toward the stator vanes which redirect it back into the rotor. This highly turbulent process repeats itself again and again. The change of momentum experienced by the water as it changes direction is manifested as a reaction force on the stator housing.

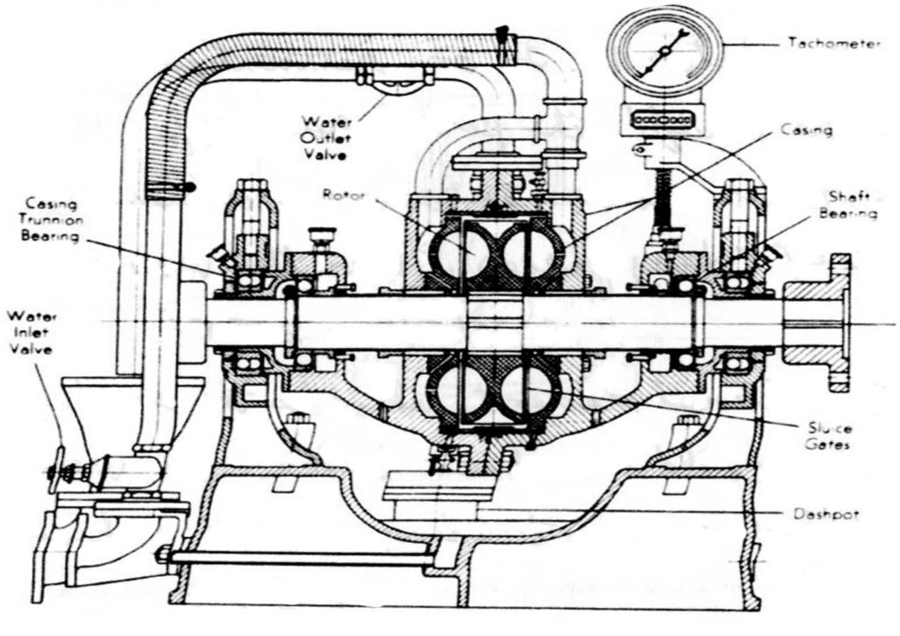


***Figure (6-6):*** *Cross-section through casing of Froud dynamometer*

## b) The eddy – current Dynamometer:

It consist of disk (d) which, driven by the engine under test, turns in a magnetic field, see figure (6-7). The strength of the field is controlled by varying the current through a series of coils (g) located on both sides of the disk (d).The revolving disk; act as a conductor cutting the magnetic field. Currents are induced in the disk and, since no external circuit exists, the induced current heats the disk. The temperature rise of the stator (a) is controlled by the flow of water in channels (h).

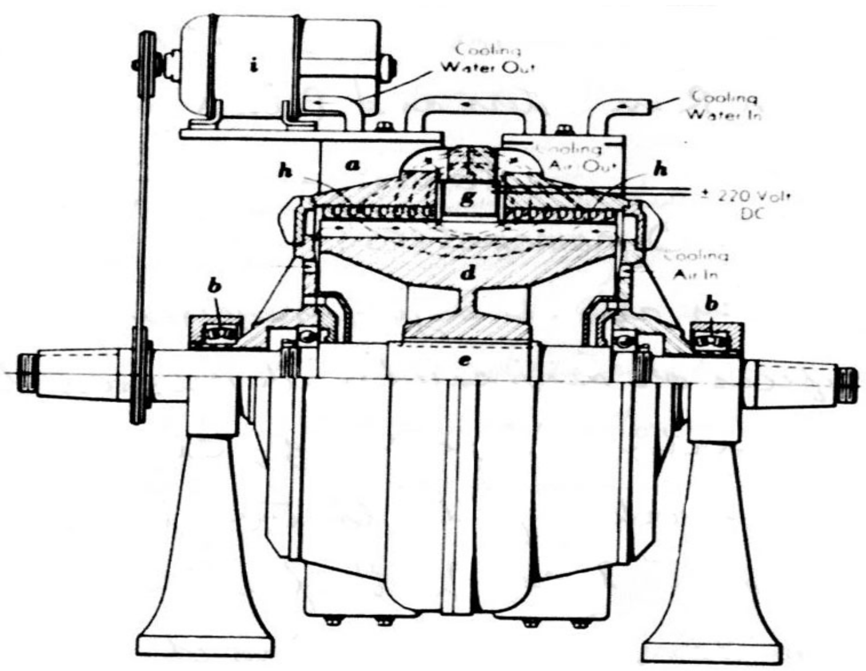
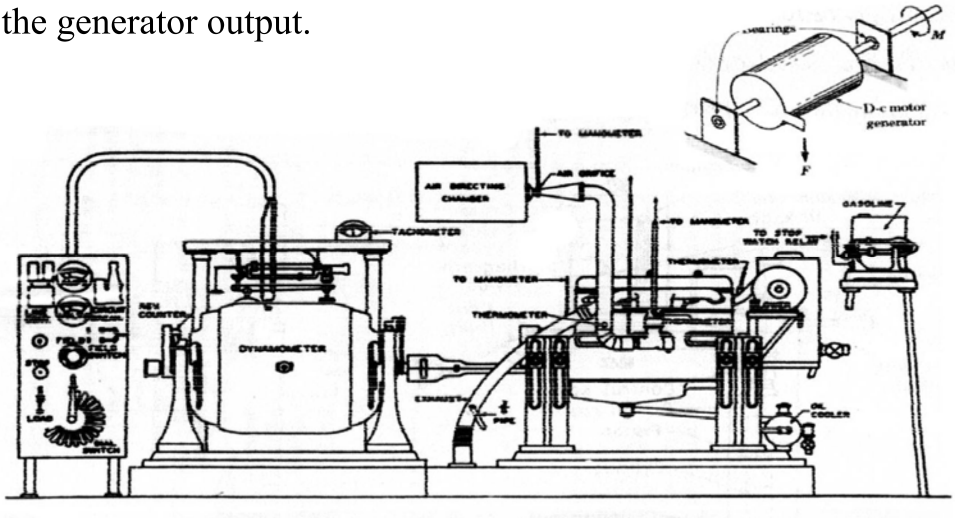


***Figure (6-7):*** *Cross- section of eddy- current dynamometer*

c) The electric dynamometer:

The electric dynamometer; as shown in fig.(6-8) can operate either as a motor to start and drive the engine at various speeds or as a generator to absorb the power output of the engine. The load is easily varied by changing the amount of resistances in the circuit connected to the generator output.

***Figure (6-8****): set- up of engine and dynamometer*



## 6-Measurment of engine indicated power:

There are two methods of finding the indicated power of an engine:
i-By taking the indicator diagram with the help of an indicator.
ii-By measuring b.p and f.p separately and adding the two.

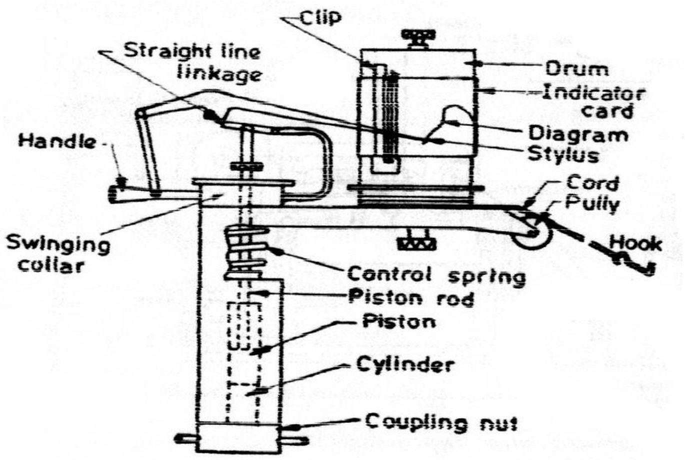
### i- Indicator Diagram:

The device which measures the variation of the pressure in the cylinder over the cycle is called an ***indicator*** and the plot (diagram) of such information obtained is called ***indicator diagram***. There are two types of indicator diagrams which can be taken from various indicators, these are:

1- Pressure – volume (p–v) plot.
2- Pressure – crank angle (p–$\theta$) plot.

There are number of indicators in use. However, only some representative types would show here:

a) Piston indicator (fig. 6-9)
b) Balance– Diaphragm (Farnborough balanced engine) indicator (fig. 6-10)



**Figure(6-9)** *Piston indicator*

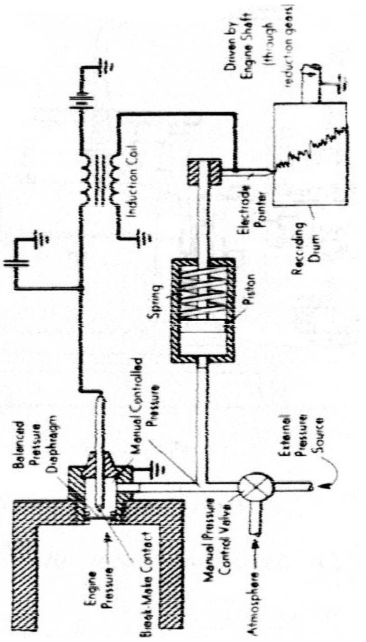


**Figure (6-10)**: *Schematic diagram of balanced- diaphragm type indicator*

## b) Transducers and electronic indicators:

In general, a transducer is any device which converts a non-electrical quantity into an electrical signal. Examples of quantities which can be converted to electrical signals are; displacement, velocity, acceleration, and force. The electrical properties of many materials change when the material is subjected to a mechanical deformation. This is the characteristics upon which all pressure transducers depend. Resistive (strain– gauge), capacitive, or piezoelectric elements are the most common types of pressure pickups for engine work.

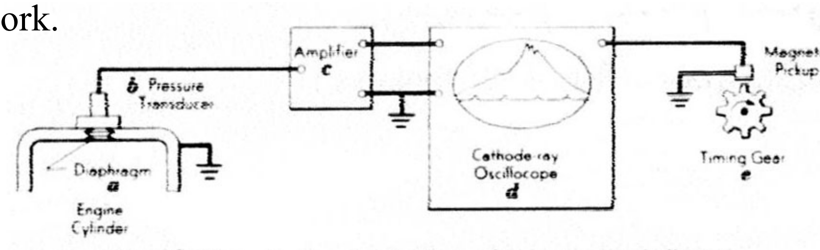
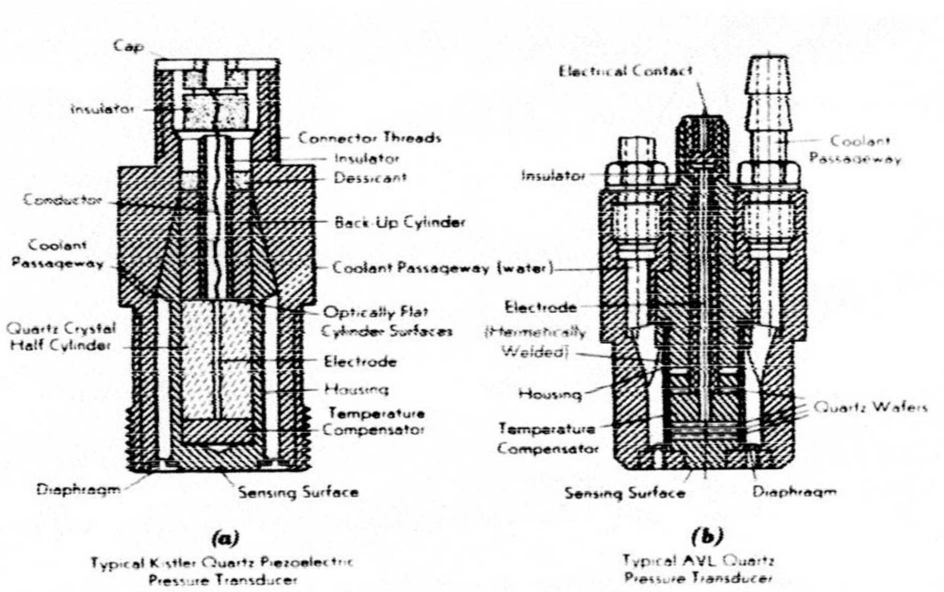


FIG. 5-14. Elements of an electrical instantaneous pressure indicator

**Figure (6-11)** *Elements of an electrical instantaneous pressure* transducer

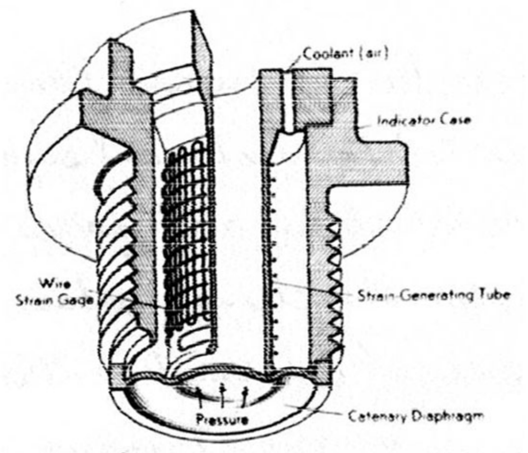
Fig. (6-11) shows a continuous – pressure system with a pressure pickup (b) (various types of pressure pickups are feasible). Fig. 6-12 shows different examples of pressure transducers, fig. (6-12 a and b) explain piezoelectric transducers, fig. (6-12b), indicate a strain – gauge transducer, while fig.6-12 d shows an electromagnetic pressure pickup.



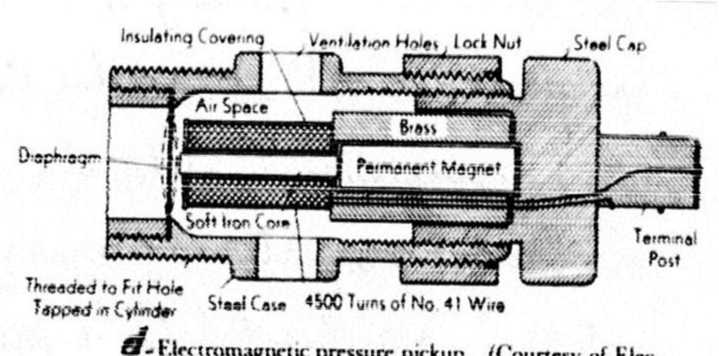
**(a)** Typical Kistler Quartz Piezoelectric Pressure Transducer

**(b)** Typical AVL Quartz Pressure Transducer

***Figure (6 – 12):*** *a & b*



**C**- Pictorial diagram of catenary diaphragm pressure pickup.



**d**- Electromagnetic pressure pickup. (Courtesy of Electro Products Laboratory, Chicago.)

***Figure (6-12****): c & d*

## 7-Pressure Diagrams for I.C engines:

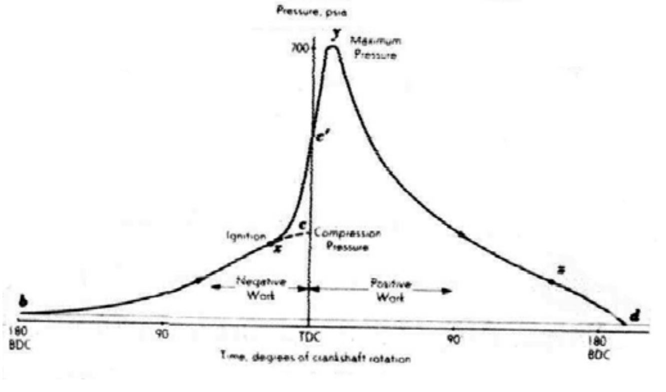


***Figure (6-13):*** *Typical p−t diagram for SI engine at wide- open throttle*

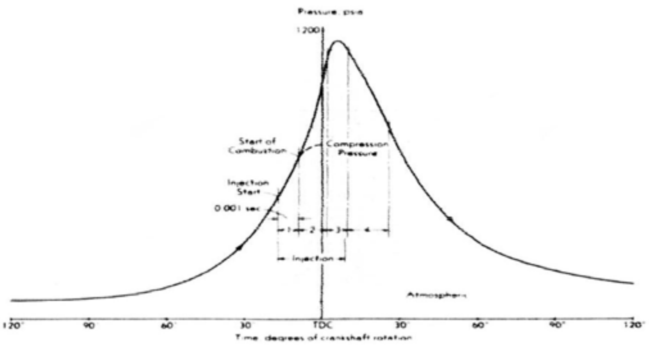


***Figure (6-14)*** *p-t diagram for mechanical- injection CI engine at full load*

## 7-Measurement of friction power (f.p):

The friction power is nearly constant at a given engine speed. Friction has a dominating effect on the performance of the engine. Frictional losses are dissipated to the cooling system as they appear in the form of heat.

Measurement of friction power is important for having better understanding on how the engine output can be increased. Methods of measuring the friction power are as follows:

i-Measurement of the i.p. and b.p. by the methods described previously for the engine at identical working conditions.

ii-Motoring test:

In this test; the engine is first run to measure the b.p at a given speed, then the fuel supply (or the spark) is cut-off and the dynamometer is converted to run as motor to drive the engine (motoring) at the same speed and keeping other parameters the same. The power supplied to the motor is measured which is a measure of the friction power (f.p).

The main objection to this method is that the engine is not firing, which leads to make running conditions are not similar. The pressure and temperature of cylinder contents, cylinder and piston surfaces are not the same.

iii-Morse test:

This test is only applicable to multi-cylinder engines. The engine is run at the required speed and the torque is measured. One cylinder is cut out, the speed falls because of the loss of power with one cylinder cut out, but is restored by reducing the load. The torque is measured again when the speed has reached its original value. If the values of i.p. of cylinders are denoted by $I_1$,$I_2$,$I_3$, and $I_4$(considering a four – cylinder engine), and the power losses in each cylinder are denoted by $L_1$,$L_2$, $L_3$ and $L_4$, then the value of b.p, B, at the test speed with all cylinders firing is given by:

$$B=(I_1-L_1)+(I_2-L_2)+(I_3-L_3)+(I_4-L_4)$$

If number 1 cylinder is cut out, then the contribution $I_1$ is lost; and if the losses due to that cylinder remain the same as when it is firing, then the b.p $B_1$ now obtained at the same speed is:

$$B_1= (0 -L_1)+(I_2-L_2)+(I_3-L_3)+(I_4-L_4)$$

Subtracting the second equation from the first given

$$B - B_1 = I_1$$

By cutting out each cylinder in turn the values $I_2$, $I_3$ and $I_4$ can be obtained, then:

$$I = I_1 + I_2 + I_3 + I_4$$

iv- Willan's line:

In this method gross fuel consumption versus b.p at a constant speed is plotted. The graph drawn is called the "Willan's line" and extrapolated back to cut the b.p axis at the point A. OA represent the power loss of the engine at this speed. The fuel consumption at zero b.p is given by OB; this would be equivalent to the power loss OA. This test is applicable to C.I. engines only.

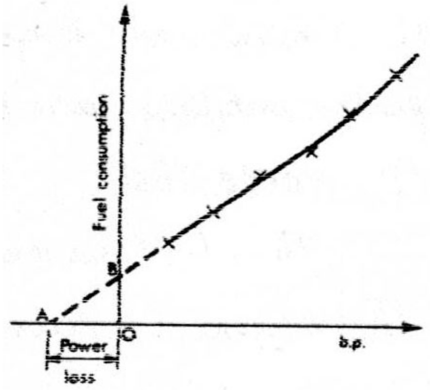


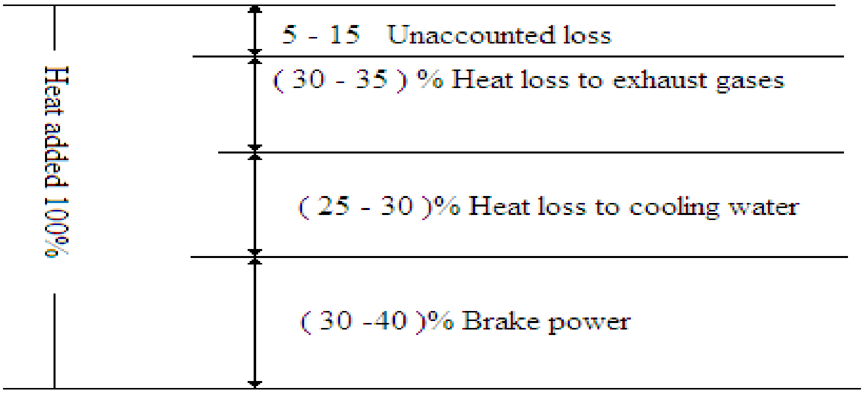
***Figure (6-15):*** *Willan's line*

## 8 – Heat balance of Engine:

The main components of the heat balance are:

1- Heat equivalent to the b.p of the engine.
2- Heat rejected to the cooling medium.
3- Heat carried away from the engine with the exhaust gases.
4- Unaccounted losses.

*Figure (6-16) Heat balance diagram ( or chart )*



The following table gives the approximate percentage values of various losses in SI and CI engines:

| Engine | % b.p | % heat to cooling water | % heat to exhaust gases | % unaccounted loss |
|---|---|---|---|---|
| S.I. | 21-28 | 12-27 | 30-55 | 0-15 |
| C.I. | 29-42 | 15-35 | 25-45 | 10-20 |

# Performance Characteristics:

The modern I.C engines have; higher ratios of power /weight than earlier types, increased values of (bmep) and thermal efficiency, and lower (s.f.c). At present time in the automotive field; the petrol engine is highly developed and flexible, but there is an increasing competition from the diesel engine. Brake thermal efficiencies of 25 to 35% are usual with S.I. engines and may reach 50% in diesel engines.

For comparing the performance of engines, a number of standards are available:

1-Specific fuel consumption (kg/kW.h).
2-Brake means effective pressure, bmep (kPa).
3-Specific weight (Weight of engine per kW, kg/kW)
4-Output per unit displacement kW per $m^3$)

Most of the performance factors are directly related to atmospheric conditions, so comparison between engines should be performed at similar atmospheric conditions.

The tests on I.C. engines can be divided into two types:

1-Variable – speed test.
2-Constant – speed test.

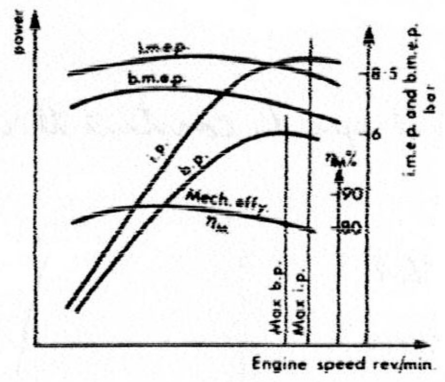
## 1-Variable – sped test:

Variable – speed tests can be divided into full – load tests, where maximum power and minimum s.f.c at each different speed are the objectives, and part – load tests to determine variation in the s.f.c.

### a) Full – load test with SI engine:

The throttle is fully opened and the lowest desired speed is maintained by brake load adjustment. The spark is adjusted to give maximum power at this speed. The test is started by the watch governing the fuel consumption, the test ended at the time the fuel- consumption test has been completed. During this interval of time, the average speed, brake load, temperatures, fuel weight … etc., are recorded, then load is adjusted for the next run at different speed.

After the completion of the test, the required results are calculated, and performance curves are drawn and a typical example is shown in fig. 5.15. The variation of volumetric efficiency with speed is indicated in fig. 5-16, and that of mechanical efficiency with speed in fig. 6-15.



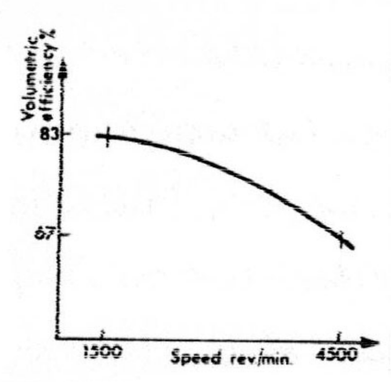


*Figure (6 – 15)* *Figure (6 – 16)*

### b) Part – load test:

To run a part – load test at variable speed, say $\frac{1}{2}$ load, power reading of half the maximum power at each speed are obtained by varying the throttle and brake setting.

## 2-Constant – speed test:

Constant – speed test is run with variable throttle from no load to full load in suitable steps of load to give smooth curves. Starting at zero load, the throttle is opened to give the desired speed. Then a load is put on the engine and the throttle is opened wider to maintain the same constant speed as before, and the second run is ready to start. The last run of the test is made at wide-open throttle. In a CI-engine test the last run would show smoke in the exhaust gas.

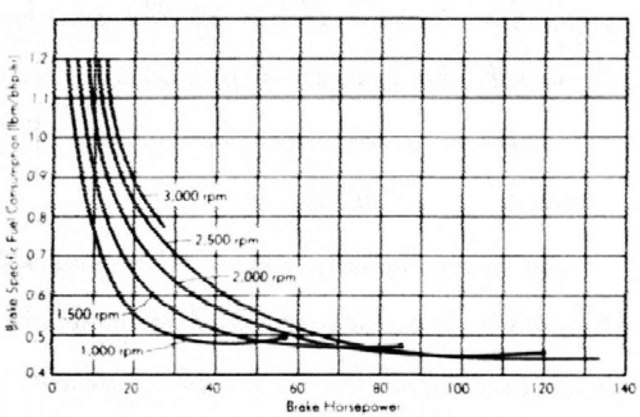


*Figure (6 – 17)* Constant speed, Variable throttle, test of automotive S.I. engine

## 3-Consumption loop test:

This test is carried out at constant speed, constant throttle opening, and constant ignition setting. The specific fuel consumption is plotted to a base of "bmep" and a "hook curve" is obtained. For a single cylinder at full throttle the curve is defined as in fig. 6-18.

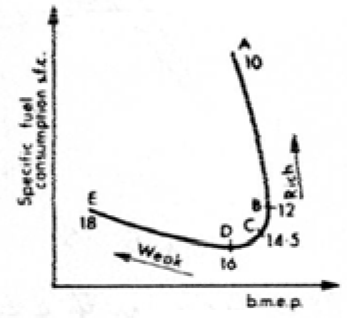


*Figure (6 – 18)*

The A/F ratio is a minimum at A(i.e. richest mixture). As the A/F ratio is increased the "bmep" increases until a maximum is reached at B (usually for an A/F ratio between 10/1 and 13/1). Further increase in A/F ratio produce a decrease in "bmep" with increasing economy until the position of maximum economy is reached at D. beyond D, for increasing A/F ratios, both "bmep" and consumption values are adversely affected. Near the point A the engine could be running unsteadily and there may be combustion of the mixture in the exhaust system. At E, with the weakest mixture, running will be unsteady and the combustion may be slow. Point C is the point of chemically correct A/F ratio.

For multi-cylinder engines the consumption loops are less distinct, but are generally similar to that for the single – cylinder engine. This is also true for tests made at part throttle opening. A series of reading obtained at different throttle positions at constant speed is shown in fig 6-19.

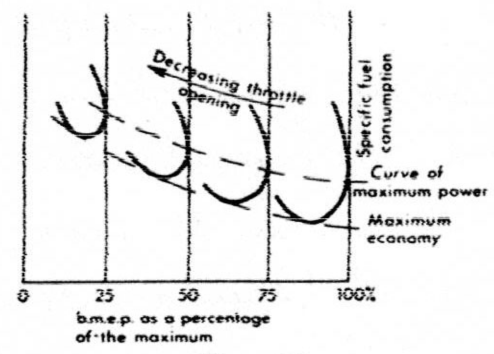


Fig.5 .19

## Example 6-5:

A four – cylinder petrol engine has a bore of 57mm and a stroke of 90mm. its rated speed is 2800 rpm and it is tested at this speed against a brake which has a torque arm of 0.356m. The net brake load is 155N and the fuel consumption is 6.74 l/h. The specific gravity of the petrol used is 0.735 and it has a lower calorific value of;44200 kJ/kg. A Morse test is carried out and the cylinders are cut out in the order 1,2,3,4, with corresponding brake load of 111,106.5, 104.2 and 111 N, respectively. Calculate for this speed, the engine torque, the bmep, the brake thermal efficiency, the specific fuel consumption, the mechanical efficiency and the imep.

Solution:

Torque T=RP=0.356 × 155 = 55.2 Nm

$$b.p = 2\pi\, NT = \frac{2\pi \times 2800 \times 55.2}{60 \times 10^3} = 16.2\ kw$$

$$bmep = \frac{b.p \times 2}{ALNn} = \frac{16.2 \times 2 \times 4 \times 60 \times 10^3}{\pi \times 0.057^2 \times 0.09 \times 2800 \times 4 \times 10^5} = 7.55 \ \text{ bar}$$

$$\eta_{BT} = \frac{b.p}{\dot{m}_f \times C.V} = \frac{16.2}{0.001377 \times 44200} = 0.266 \ \text{ or } \ 26.6\%$$

Where $\mathrm{m_f} = \dfrac{6.74}{3600} \times 1 \times 0.735 = 0.001377 \quad \text{kg/s}$

$$\mathrm{sfc} = \frac{\dot{\mathrm{m}}_\mathrm{f}}{\mathrm{b.p}} = \frac{0.001377 \times 3600}{16.2} = 0.306 \quad \text{kg/kW.h}$$

The indicated load for the engine is calculated by the Morse test method as:

$I=I_1+I_2+I_3+I_4$

and: $I_1= B-B_1=155 - 111=44$ N

$I_2= B-B_2=155-106.5=48.5$ N

$I_3= B-B_3=155-104.2=50.8$ N

$I_4= B-B_4=155 - 111=44$ N

$I=44+48.5+50.8+44=187.3$ N

$$\eta_M=\frac{b.p}{i.p}=\frac{155}{187.3}=0.828 \quad \text{or } 82.8\%$$

$$i.p=\frac{16.2}{0.828}=19.57 \quad \text{kw}$$

$$\text{bmep}=\eta_M\times imep$$

$$i.e \quad \text{imep}=\frac{7.55}{0.828}=9.12 \quad \text{bar}$$

---

## Solved problems – chapter (6):

**Ex.1**-31 six – cylinders SI engine operates on a four – stroke cycle and run at 3600 rpm. The compression ratio is 9.5 the length of connecting rode is 16.6cm, and the bore equal the stroke. Combustion ends at $20^o$ after TDC calculate: (1) Cylinder bore and stroke, (2) average piston speed, (3) clearance volume of one cylinder, (4) the distance piston has traveled from TDC at the end of combustion, (5) volume of the combustion chamber at the end of combustion.

**Solution**

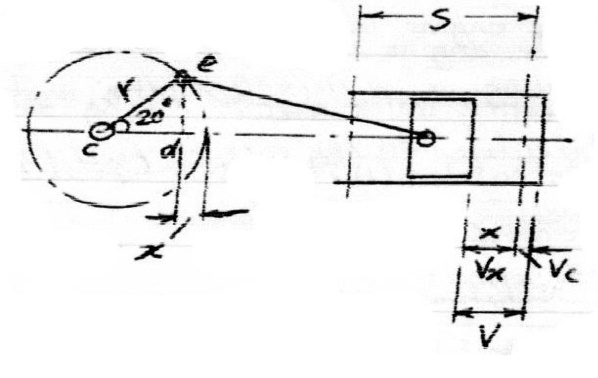
1- Volume of one cylinder, $V_s=\frac{3000}{6}=500 \text{ cc } =0.0005\text{m}^3=\frac{\pi}{4}B^2S$

$0.000637=B^3 \Rightarrow B=0.086m=8.6cm=S$

2-$V_m=\frac{2SN}{60}=\frac{2\times0.086\times3600}{60}=10.32 \text{ m/s}$

3- $r=\frac{V_s+V_c}{V_c}=9.5=\frac{0.0005+V_c}{V_c}$

i.e. $V_c=0.000059\text{m}^3=59 \text{ cm}^2$



4- Volume at any C.A. = $V_c+V_x$

$V=V_c+x\times\frac{\pi}{4}B^2$ (B=bore)

$5-\text{x}=\text{r}(1\text{-cos }\theta)$, $\quad$ $\text{r}=\frac{\text{S}}{2}=4.3$ cm

x = 4.3 (1-cos 20) = 0.26 cm, $\quad$ $V=59+\frac{\pi}{4}(9)^2\times0.26=75.54 \text{ cm}^3$

**Ex.2**- The engine in example 1 is connected to a dynamometer which gives a brake output torque of 205 Nm at 3600 rpm. At this speed air enters the cylinder at 85 kPa and 60$^o$C, and the mechanical efficiency of the engine is 85%. Calculate: (1) b.p, (2) i.p, (3) bmep, (4) imep, (5)fmep, (6) f.p, (7) engine specific volume.

Solution

$$1\text{-}\ b.p = 2\pi NT = 2\pi \times \frac{3600}{60} \times 205 = 77.3 \ \text{ kW}$$

$$2\text{-}\ i.p = \frac{b.p}{\eta_M} = \frac{77.3}{0.85} = 90.9 \ \text{kW}$$

$$3\text{-}\ bmep = \frac{b.p \times z}{LANn} = \frac{77.3 \times 60 \times 2}{0.0005 \times 3600 \times 6} = 859 \ \text{kPa}$$

$$4\text{-}\ imep = \frac{859}{0.85} = 1010.5 \ \text{ kPa}$$

$$5\text{-}\ fmep(P_f) = imep - bmep = 1010.5 - 859 = 151.57 \ \text{kPa}$$

$$6\text{-}\ f.p = P_f LAN \times \frac{n}{z} = 151.57 \times 0.0005 \times 3600 \times \frac{6}{2} \times \frac{1}{60} = 13.64 \ \text{kW}$$

$$7\text{-Engine specific volume} = \frac{\text{swept volume}}{\text{brake power}} = \frac{3\text{l}}{77.3} = 0.0388 \ \text{L/kW}$$

The inverse of the specific volume is = 25.8 kW/L

**Ex.3**-The engine in example 2 is running with A/F ratio =15,afuel of heating value;44000kJ/kg and a combustion efficiency of 97% calculate: (1) the rate of fuel flow. (2) $\eta_{BT}$ , (3) $\eta_{IT}$ ,(4) $\eta_V$ , and brake specific consumption.

**Solution:**

1- The clearance volume of the engine = 0.000059 m$^3$(example 1)

$$m_\alpha = \frac{PV_{BDC}}{RT} = \frac{P(V_c + V_s)}{RT} = \frac{85(kPa)(0.0005 + 0.000059)m^3}{0.287(kJ/kgK)(333K)} = 0.0005 \ \text{kg}$$

$$m_f = \frac{m_a}{\frac{A}{F}} = \frac{0.0005}{15} = 0.000033\,\text{kg}_\text{f} \text{ Per cylinder per cycle}$$

$$\dot{m}_f = (0.000033 \frac{kg}{cycle.cylinder})(6\,\text{cylinder})(\frac{3600}{60} rev/s)(\frac{1}{2} effective\,\text{cycle/cylinder})$$

$$= 0.006 \ \text{kg/s}$$

$$2\text{-}\ \eta_{BT} = \frac{\text{b.p}}{\dot{\text{m}}_\text{f} \times \text{C.V} \times \eta_\text{c}} = \frac{77.3}{0.006 \times 44000 \times 0.97} = 0.302 \ \text{ or } \ 30.2\%$$

$$3\text{-}\ \eta_{IT} = \frac{\eta_{BT}}{\eta_m} = \frac{0.302}{0.85} = 0.355 \ \text{ or } \ 35.5\%$$

$$4\text{-}\ \eta_V = \frac{m_a}{\rho_a V_s} = \frac{0.0005\,\text{kg}}{(P/RT)(0.0005\,\text{m}^3)} = \frac{1}{1.181} = 0.847 \qquad \text{or } \ 84.7\%$$

5- $bsfc = \frac{\dot{m}_f}{b.p} = \frac{0.006\,\frac{\text{kg}}{\text{s}}}{77.3\,\text{kw}} = 7.76\,\text{kg/kW.s} = 279\,\text{kg/kW.h}$

$$\rho_{\text{air}} = \frac{\text{P}}{\text{RT}} = \frac{1.013 \times 10^{2}}{0.287\,(15 \times 273)}$$

$$\eta_{\text{V}} = 0.76 = 76\,\%$$

**Ex.4**- A six-cylinder 4-stroke cycle petrol engine is to be designed to develop 300 kW of (b.p) at 2500 rpm the bore / stroke ratio is to be 1:1.25. Assuming $\eta_m$=83% and an indicated mean effective pressure of 9.5 bar, determine the required bore and stroke. If the compression ratio of the engine is to be 6.5 to 1, determine consumption of petrol in kg/h and in kg/bp.hr. Take the ratio of the indicated thermal efficiency of the engine to that of the constant volume air standard cycle as 0.55 and the calorific value of the petrol as; 44770kJ/kg.

**Solution**

b.p=300 kW

$$\eta_m = \frac{b.p}{i.p} \quad ; \qquad \text{i.p} = \frac{300}{0.83} = 361\ kW$$

$$P_i = 9.5\ \text{bar} \quad \& \quad \text{N} = 2500\ \text{rpm}$$

$$\text{i.p} = \frac{\text{P}_\text{i} \times L \times A \times N \times n}{60} \times \frac{1}{2} \quad (4 - strok\ \ engine\ )$$

$$1000 \times 361 = \frac{9.5 \times 10^{5} \times (LA) \times 2500 \times 6}{60 \times 2}$$

$$(LA) = 0.00304\ \ \text{m}^{3}$$

Let Diameter = D, $\therefore$ L=1.25D &A=$\frac{\pi}{4}$ D$^2$

$$1.25D \times \frac{\pi}{4} D^{2} = 0.00304$$

$$D^{3} = 0.003096 \quad \Rightarrow D = 0.146\ m$$

$$D = 14.6\ \text{cm} \quad \& \text{ L} = 1.25\text{D} = 18.25\ \text{cm}$$

**r = 6.5 & $\gamma$ = 1.4**

$$\eta_{a.s} = 1 - \frac{1}{r^{\gamma - 1}} = 1 - \frac{1}{6.5^{1.4-1}} = 52.6\%$$

$$\eta_r = \frac{\eta_{th}}{\eta_{A.S}} \times 100$$

$$55 = \frac{\eta_{th}}{52.6} \times 100$$

$$\eta_{th} = \frac{55 \times 52.6}{100} = 28.9\%$$

$$\eta_{th} = \frac{i.p \times 60}{\text{heat in fuel supplied /min}} \times 100$$

$\therefore$ Heat in fuel supplied /min = $\frac{361 \times 60}{0.289} = 74948\ \text{kJ}$

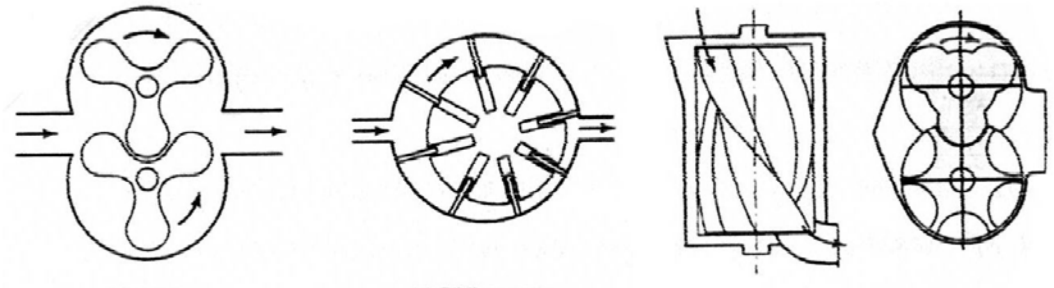
$\therefore$ Consumption of petrol in kg/h $= \frac{74948}{44770} = 100.4$ & $\frac{\text{kg}}{\text{kW.h}} = \frac{100.4}{300} = 0.33$

## *CHAPTER (7)*

# SUPERCHARGING

The power and efficiency of an internal combusting engine can be increased with the use of an air compression device such as a supercharger or turbocharger. Increasing the pressure and density of the inlet air will allow additional fuel to be induced into the cylinder, increasing the power produced by the engine. Spark ignition engines are knock limited, restricting the allowable compressor pressure increase, in many cases the compression ratio of a SI engine is reduced. Superchargers and turbochargers are used extensively on a wide range of diesel engines, since they are not knock limited.

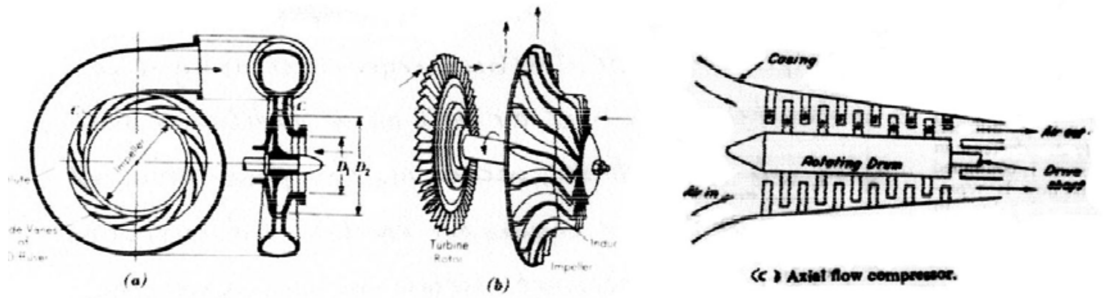
The types of compressors used on internal combustion engines are primarily of two types: positive displacement and dynamic. With a positive displacement compressor, a volume of gas is trapped, and compressed by movement of a compressor boundary element. Three types of positive displacement compressors are the roots, vane, and screw compressor, as shown in figure 7-1.



(a) Roots (b) Sliding vane (c) Lysholm screw

*Figure (7 – 1) Types of positive displacement compressors*

A dynamic compressor has a rotating element that adds tangential velocity to the flow which is converted to pressure in a diffuser. Two types of dynamic compressors and turbines are; radial (centrifugal) and axial as shown in figure 7-2.

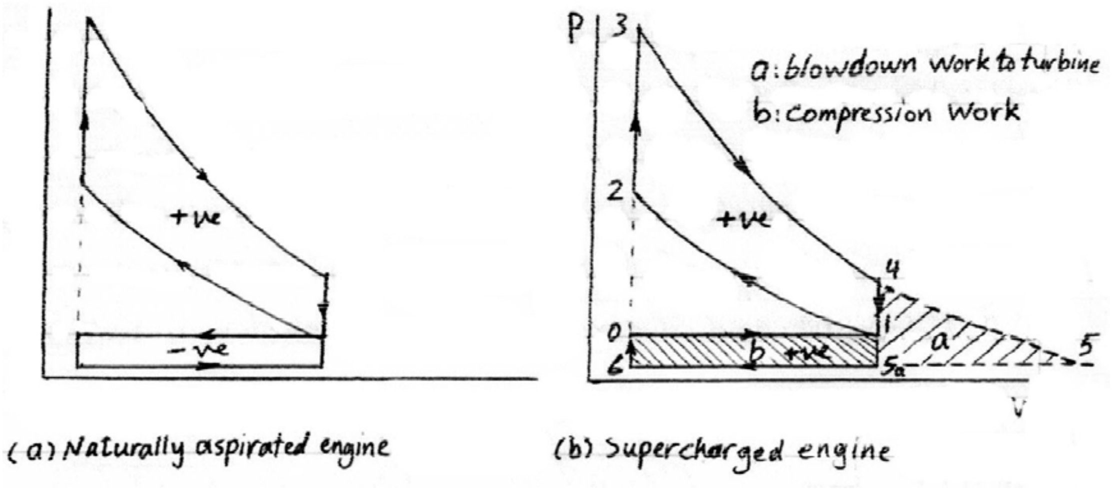


(a) (b) (c) Axial flow compressor.

(a) Radial centrifugal compressor with vaned diffuser and volute casing
(b) Radial impeller and axial turbine wheel.

*Figure (7 – 2)*

## 7-1 Thermodynamic Cycle with Supercharging:



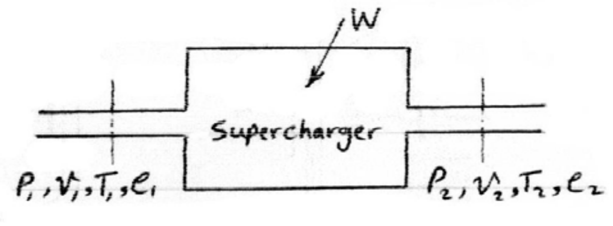
(a) Naturally aspirated engine (b) Supercharged engine

*Figure (7 – 3)*

The pumping loop of a supercharged engine is positive instead of negative. Hence to get the net indicated power (i.p), the power represented by pumping loop is to be added:

i.p=area 12341 + area 015a60

The power required for driving the supercharger can be calculated by considering the steady flow process as given in the figure. The air enters the supercharger at a pressure $p_1$ and has an internal energy $e_1$. The work supplied to the supercharger is W. The air leaves the supercharger at a pressure $P_2$ and has an internal energy $e_2$, then:



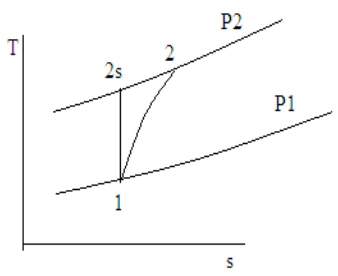
$$e_1+P_1V_1+W=e_2 + P_2V_2+Q$$

If we assume that the heat loss Q from the supercharger is zero, we get:

$$W=(e_2+P_2V_2)-(e_1+P_1V_1)$$
$$=h_2-h_1=C_p\,(T_2-T_1)$$

$T_2$ is the temperature at the end of compression in the supercharger, is given by:

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



$\eta_s$; is the adiabatic efficiency of the supercharger.

The power required to drive the supercharger is then equal to;

$$Wc=maC_pT_1\left\{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} \times \frac{1}{\eta_s}$$

This power can be supplied by a separate drive for the supercharger or by connecting the supercharger directly to the engine shaft or to gas turbine driven by the engine exhaust gases. In all cases the gain in the power output of the engine would be many times the power required to drive the compressor.

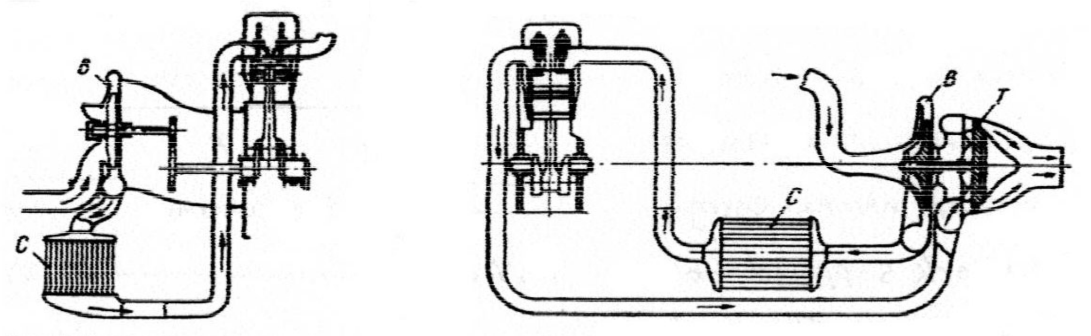


Fig. 6-4a Diagram of a mechanically supercharged engine: B—centrifugal blower, C—supercharging air cooler

Fig. 6-4b Diagram of a turbocharged engine: B—centrifugal blower; C—supercharging air cooler; T—gas turbine

Figure (7 – 4 a & b)

## 7-2 TURBOCHARGERS:

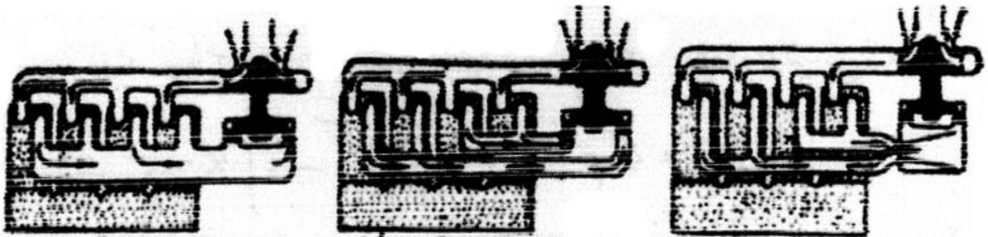
Turbochargers are made in all sizes to fit the smallest as well as the largest engines. Typical example is shown in fig.7-4.

In order to supply sufficient energy to the turbocharger the exhaust valve is opened much before the BDC as compared to naturally aspirated engine. This allows the exhaust gasses to escape at a higher pressure and temperature giving the turbocharger enough energy to drive the compressor.

### Methods of turbo charging:

Figure 7-5 shows various methods used for turbo charging of 4-stroke and 2-stroke diesel engines. The main types of turbo charging methods are:

a) **Constant pressure:**



(a) Constant pressure (b) Pulse (Buchi) (c) Pulse converter

Figure (7 – 5) Methods of turbo charging

The exhaust from various cylinders, discharge into a common manifold at pressures higher than atmospheric pressure and maintained at constant pressure during the whole cycle so that a pure reaction turbine can be used. This objective dictates a large exhaust manifold to absorb pressure fluctuations and therefore the kinetic energy in the exhaust blow down is dissipated.

**b) Pulse (or Buchi):**

In this system the objectives is to use the kinetic energy in the blow down process to drive the turbine, ideally, without increase in exhaust pressure. To accomplish this objective the exhaust lines must be small, and grouped to receive the exhaust from cylinders which are blowing down at different times. The turbine has separate inlets and nozzle segments for each exhaust pipe.

c) Pulse converter:

Pulse converter allows converting the K.E. in the blow down process into a pressure rise at the turbine by means of one or more diffusers. Ideally, the advantages of both the pulse system and the constant – pressure system are gained.

**Arrangement of exhaust manifolds:**

1-Constant pressure:

The exhaust piping system in this arrangement is not complex. The exhaust pipe diameter must be 1.4 times the piston diameter.

2-Pulse blow down:

With this system the exhaust system and turbo charger location must be carefully designed to avoid interference with the scavenging process, especially for two-stroke engines.

Figure 7-6 shows the exhaust pressure diagram for a 4-stroke diesel engine with a firing order 1-3-4-2, having two exhaust manifolds, Cylinders 1 and 4 exhausting in one manifold, and cylinders 2 and 3 into another. The pressure diagrams of the two manifolds are super-imposed.

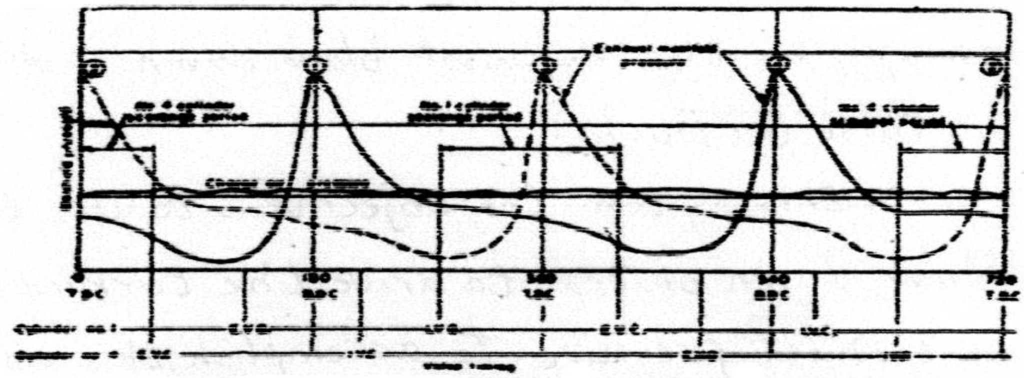


*Figure (7 – 6) Exhaust manifold pressure diagram for a 4 – Stroke diesel engine with firing order 1 – 3 – 4 – 2.*

The figure shows that during scavenge period the charge pressure is always more than the exhaust pressure. If the four cylinders are made to exhaust in a common exhaust, the exhaust pressure in cylinder 3 will be more than the charging air. That means a four cylinder engine requires two exhaust manifold, an eight cylinder engine require four exhaust manifolds and a five cylinder engine having equally spaced ranks require three manifold. Figure 7-7a shows the manifold arrangements used for four strokes in – line engines and figure 7-7b shows the manifold arrangements for 2-stroke engines.

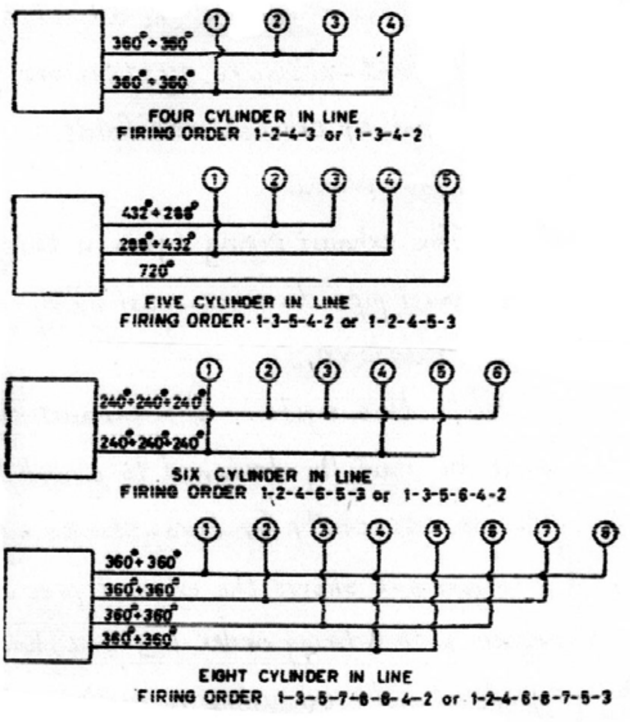


*Figure (7 – 7a) Typical exhaust arrangement and cylinder Combination for 4 – stroke in line engines using pulse Turbo charging.*

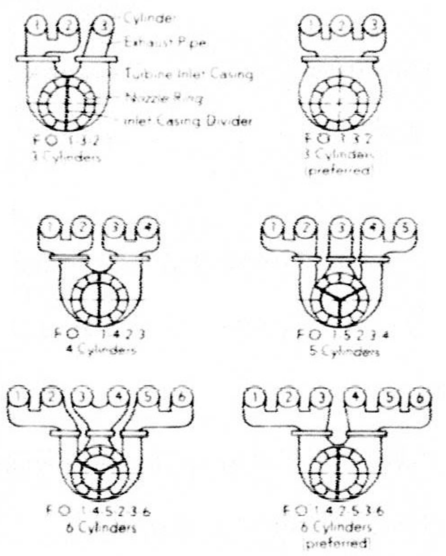


*Figure (7 – 7b) Typical arrangement for pulse turbocharged, two – stroke engines.*

## Example 7-1

A 4-stroke diesel engine is to be designed to operate with the following characteristics at sea level, where the ambient conditions are 103 kPa and 10°C.

b.p=260 kW, s.f.c=0.244 kg/kW.h,

Speed = 1500 rpm, volumetric efficiency =0.78.

A/F ratio = 17:1,

Calculate the required engine capacity and the anticipated bmep.

The engine is fitted with a supercharger so that it may be operated at an altitude of 2700m where the atmospheric pressure is 73kPa.The power taken by the supercharger is 8% of the total power produced by the engine and the temperature of the air leaving the supercharger is 32°C. The A/F ratio and thermal efficiency remain the same, so as the $\eta_v$.Calculate the increase of pressure at the supercharger to maintain the same net power output. Take R=0.287 kJ/kg.K.

## Solution:

**Naturally aspirated:**

Fuel consumption = 0.244 $\times$260=63.44 kg/h = 1.057 kg/min.

Air consumption = $1.057\times17=17.97$ kg/min.

$$\eta_v=\frac{\dot{V}}{V_s}=\frac{17.97\times0.287\times283}{1.03\times V_s\times750\times100} \qquad \text{(PV=mRT)}$$

$V_s = 0.0242\ \text{m}^3$

$$bmep=\frac{b.p.\times n}{LAN}=\frac{260\times2\times60}{0.0242\times1500}=859.5\ \text{kPa}$$

**Supercharged:**

Total power produced by the engine = $260+0.08\times260=280.8$ kW

This is by assuming ideal supercharging, i.e. all the power taken by the supercharger is added to the engine power.

Mass of fuel required per minute = $280.8\times\frac{0.244}{60}=1.142$kg/min

Mass of air/min. $=1.142\times17=19.41$kg/min.

Volume of air induced at supercharger outlet conditions = $\frac{mRT}{P}$

$$\dot{V}=\frac{19.41\times0.287\times305}{p_2}=\frac{1699}{p_2}\ \text{m}^3/\text{min}$$

$$\therefore 0.78=\frac{1699}{p_2\times0.0242\times750}$$

$$p_2=120\ \text{kPa}$$

Increase of pressure required = 120 – 73= 47kPa

Pressure ratio of supercharger = $\frac{120}{73}=1.644$

## Example 7-2:

A 4-stroke diesel engine of 3000 C.C. capacity develops 13.42 kW per $\text{m}^3$of free air induced per minute. When running at 3500 rev/min., it has a volumetric efficiency of 80 per cent referred to free air conditions of 1.03 bars and $27^{o}$C. It is proposed to boost the power of the engine by supercharging by a blower of pressure ratio 1.7 and isentropic efficiency of 75%. Assuming that at the end of induction the cylinders contains a volume of charge equal to the swept volume, at the pressure and temperature of the delivery from the blower, estimate the increase in brake power to be expected from the engine. Take overall mechanical efficiency as 80 per cent.

Solution:

Swept volume, $V_s=3000\times\frac{3500}{2}=5.25\times10^{6}$ C.C or 5.25 $\text{m}^3$/min

Naturally aspirated induced air volume = $0.8\times5.25=4.2\ \text{m}^3$/min.

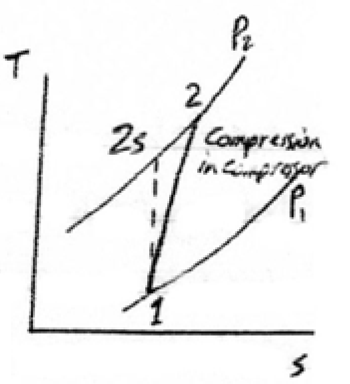
Blower delivery pressure = $1.7\times1.03=1.751$ bar

Temperature after isentropic compression = $T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 300(1.7)^{\frac{0.4}{1.4}} = 350\ K$

$$\eta_s = \frac{T_{2s} - T_1}{T_2 - T_1}$$

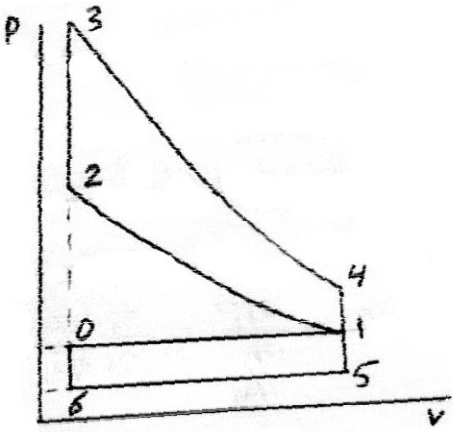
$$0.75 = \frac{350 - 300}{T_2 - 300}$$

$$T_2 = 300 + \frac{50}{0.75} = 366\ K$$



Blower delivery temperature = 366 – 273=93 K
The blower delivers; 5.25 $m^3$/min., at 1.751 bars and 366K, equivalent to a volume of:

$\frac{5.25\times1.751\times300}{1.03\times366} = 7.31\ \ m^3/\min.$ $\quad \frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2}$, (at the ambient conditions of; 1.03 bars and 300 K).



By assuming ideal supercharging; from the P-V diagram:
Increase in (i.p) due to increased induction pressure

$$= \frac{\Delta P \times V_s}{60} = (1.751 - 1.03)\times5.25\times\frac{100}{60}$$

$$= 6.31 \quad kW$$

Increase in induced volume = 7.31− 4.2=3.11 $m^3$/min
Increase in power from air induced = 13.42×3.11
=41.736 kW

Mass of air delivered by blower = $(1.751\times5.25\times10^2)/(0.287\times366)$
=8.75 kg/min.

Power required by the blower = $\dot{m}_a C_p T_1 \left\{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\}\times\frac{1}{\eta_s}$

$$= 8.75\times1.005\times366\ \left\{(1.7)^{0.286} - 1\right\}\times\frac{1}{0.75}\times\frac{1}{60}$$

$$= 11.72\ \text{kW}$$

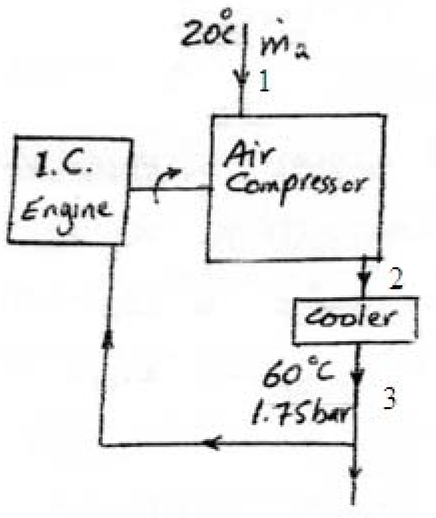
Total increase in b.p=41.736+6.31×0.8=46.78 kW
Net increase in b.p =46.78 − 11.72=35.06 kW

## **Example 7-3:**

Four – stroke oil engine is used to drive an air compressor, the air enters the compressor at 20°C and is delivered to a cooler which removes heat at the rate of 1340 kJ/min. The air leaves the cooler at 60°C and 1.75 bars. Part of this air flow is used to supercharge the engine which has a volumetric efficiency of 0.7 based on induction conditions of 60°C and 1.75 bars. The engine which has six cylinders of 90mm bore and 100mm stroke runs at 2000rpm and delivers an output torque of 150 Nm. The mechanical efficiency of the engine is 0.75. Determine:

1- The engine indicated mean effective pressure.
2- The air consumption in kg/min.
3- The air flow into the compressor in kg/min.

**Solution**



$$\text{b.p} = 2\pi\,N\,T = 2\pi \times 2000 \times 150 \times \frac{1}{60\times 1000}$$

$$= 31.42\ \text{kW} = W_c$$

$$\text{i.p} = \frac{\text{b.p}}{\eta_m} = \frac{31.42}{0.75} = 41.89\ \text{kW}$$

$$P_i = \frac{i.p \times z}{LAN \times n} = \frac{41.89 \times 2 \times 4 \times 60}{0.1 \times \pi \times (0.09)^2 \times 6 \times 2000 \times 100}$$

$$= 6.58\ \text{bar}$$

$$\text{Engine swept volume} = 6 \times \frac{\pi}{4}(0.09)^2 \times 0.1 \times \frac{2000}{2}$$

$$= 3.82\ \text{m}^3/\text{min}.$$

$$\eta_v = \frac{\dot V}{V_s} \qquad \dot V = 0.7 \times 3.82 = 2.674\,\text{m}^3/\text{min}$$

$$\text{Aspirated air mass flow into the engine} = \frac{PV}{RT} = \frac{1.75 \times 2.674 \times 10^2}{0.287 \times 333} = 4.9\ \text{kg/min}$$

Now, work done on air in compressor = gain in enthalpy of air in compressor.

$$W_c = \dot m_a\, C_{pa}\,(T_2 - T_1)$$

$$Q = \dot m a\, C_{pa}\,(T_2 - T_3)$$

$$\frac{W_c}{Q} = \frac{T2 - T1}{T2 - T3} = \frac{T2 - 293}{-333 + T2} = \frac{31.42 \times 60}{1340} = 1.407$$

$$\therefore T_2 = 431.17\ \text{K}$$

$$\frac{1340}{60} = 1.005\,\dot m_a \times 98.1 \quad \Rightarrow \dot m_a = 0.2265\ \text{kg/s}$$

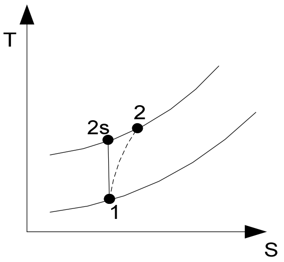
**Example 7-4**

Six-cylinder, 4.8 lit supercharged engine operating at 3500 rpmhas an overall volumetric efficiency of 158%. The supercharger has an isentropic efficiency of 92% and mechanical efficiency 87%.It is desired that air to be delivered to the cylinder at 65°C and 180 kPa, while ambient conditions are 23°C and 98 kPa.

Calculate: (a) Amount of after cooling needed.

(b) Engine power lost to run the supercharger

**Solution:**



$$\rho_{air} = \frac{P}{RT} = \frac{98}{0.287 \times 296} = 1.81\,\text{kg/m}^3$$

$$\eta_v = \frac{\dot{V}}{V_s}$$

$$\dot{V} = 1.58 \times \frac{4.8}{10^3} \times \frac{3500}{2 \times 60} = 0.2212\ \text{m}^3/s$$

$$\dot{m}_a = \dot{V}\rho_{air} = 0.2212 \times 1.81 = 0.4\ \text{kg/s}$$

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{2s} = 296 \times \left(\frac{180}{98}\right)^{0.286} = 352.15K$$

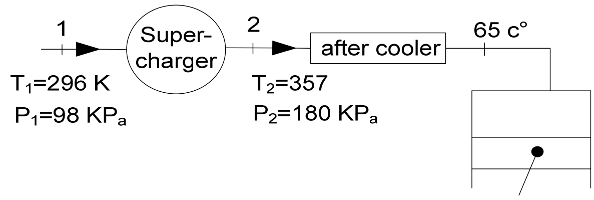
$$\eta_s = \frac{T_{2s} - T_1}{T_2 - T_1} \Rightarrow 0.92 = \frac{352.15 - 296}{T_2 - 296}$$

$$T_2 = 357\ \text{K} \quad \text{or}\ 84^{\text{o}}_{\text{C}}$$

a) The amount of after cooling needed to reduce air temperature from 84 to 65°C is:

$Q = m_a C_{pa}\,(T_2 - T_m) = 0.4 \times 1.005(84-65)$

$= 7.64$ kW



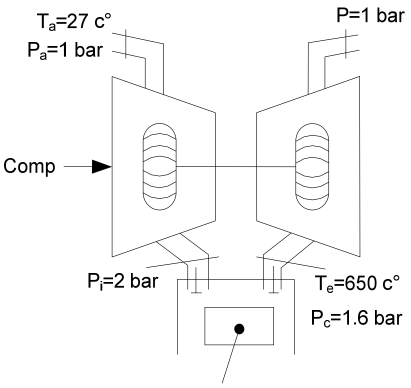
b) To find the engine power lost to drive the supercharger:

$$P = \frac{m_a C_{pa}(T_{out} - T_{in})}{\eta_m} = \frac{0.4 \times 1.005(357 - 296)}{0.87} = 28.2kW$$

**Example 7-5**: A diesel engine is fitted with a turbocharger. The engine is tested at constant speed of 500 r.p.m at atmospheric conditions of 1 bar and 27°C, the power output is 5000 kW, bmep is 15 bars and fuel consumption is 1250 kg/h an air intake manifold pressure is 2 bars, exhaust manifold pressure is 1.6 bars, and turbine inlet gas temperature is 650°C and leave the turbine at a pressure of 1 bar. The volumetric eff.of the engine is 100%. The air compressor of the turbocharger has an isentropic eff. of 70%. Calculate the isentropic eff. of the turbine. If the friction mean effective pressure is 1.5 bar, calculate the i.sfc of the engine. Assume Cp = 1.005 kJ/kg.K$\gamma$ = 1.4 for air, & $Cp_g$ = 1.05 kJ/ kg.K , $\gamma = 1\frac{1}{3}$ for exhaust gasses.

**Solution:**

$$bp = P_{bmep} \times \dot{V}_s$$

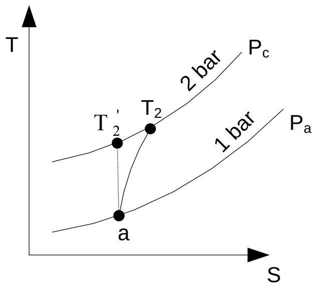


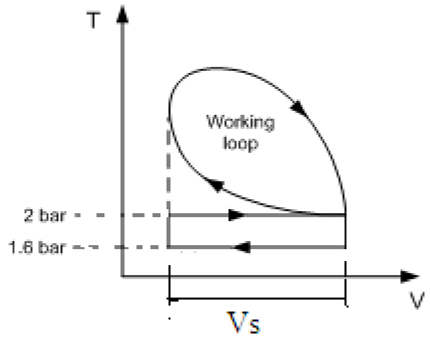
$$5000kW = 15\times100\frac{kN}{m^2}\times\dot{V}_s\frac{m^3}{\sec.}$$

$$\therefore \dot{V}_s = \frac{5000}{1500} = 3.333\frac{m^3}{\sec.} = \dot{V}_{air}$$

$$\eta_v = \frac{\dot{V}_{act}}{\dot{V}_s} = 100\%$$

At intake condition P = $P_1$ = 2bar and the temperature is:





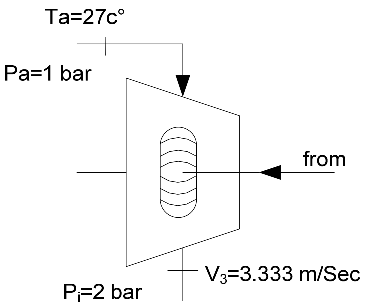
$$\frac{T_2'}{T_1} = \left(\frac{P_c}{P_a}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2' = (273+27)\left(\frac{2}{1}\right)^{\frac{0.4}{0.4}} \Rightarrow \qquad T_2' = 365.7k$$

$$0.7 = \frac{365.7-300}{T_2-300} \qquad \therefore T_2 = 393.86k$$

$$P_{comp} = 2\times100 = 200kpa, \quad T_c = 393.86k$$

$P_sV_s = \dot{m}_sRT_s, \Rightarrow \dot{m}_s = \frac{2\times100\times3.333}{0.286\times393.86} = 5.898\ kg/s$

Actual compressor (work) power is:



$Work_{comp.} = m_aC_{pa}(T_2-T_a)$

$= 5.898\times1.005(393.86-300) = 556.35$ kW

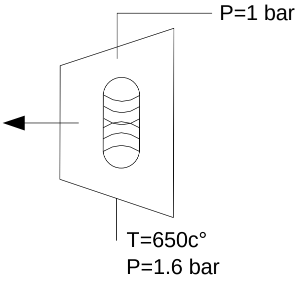
The power consumed by compressor actually developed by the turbocharger:

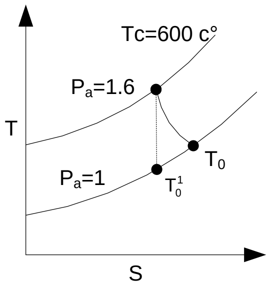
$Work_{comp.} = Work_{turbo}$

**For turbine:**

$$\frac{T_a'}{T_e} = \left(\frac{P_a}{P_e}\right)^{\frac{n-1}{n}} \qquad T_o^1 = \left(\frac{P_a}{P_e}\right)^{\frac{n-1}{n}}$$

$$T_0^1 = (650+273)\left(\frac{1}{1.6}\right)^{\frac{\frac{1}{3}}{1\frac{1}{3}}} = 820.7K$$





Isentropic turbine power is:

$$(W_t)_{isen} = \dot{m}_{ge} \times cp_{ge}(T_e - T_0'\ )$$

$$= (\mathrm{m_a} + m_f)cp_{ge}(T_e - T_0'\ ),\ ,\ \mathrm{m_a} = 5.898$$

$$\therefore \dot{m}g = \dot{m}_a + \dot{m}_f = 5.898 + \frac{1250}{3600} = 6.245 \ \text{kg/sec.}$$

$(W_t)_{isent} = 6.245 \times 1.05(923 - 820.7) = 668.8\,\text{kW}$, isentropic eff. Of turbine is

$$= \frac{\text{actual power}}{\text{Isentropic power}}$$

$$\therefore \eta_{is} = \frac{556.35}{668.8} = 83\%$$

$$b.p = i.p + p.p. - f.p$$

p.p.: is pumping power

p.p. $= \Delta p \times \dot{V}_s = (200-160) \times 3.333 = 133\ \text{kW}$

$$\text{f.p} = \text{P}_{\text{fmep}} \times \dot{\text{V}}\text{s} = 1.5 \times 100 \times 3.333 = 500\,\text{kW}$$

i.p = b.p – p.p. + f.p = 5000 – 133 + 500

= 5366.7 kW

i.sfc = $\dfrac{m_f}{Ip} = \dfrac{1250}{5366.7} = 0.233\,\text{kg/kW.hr}$

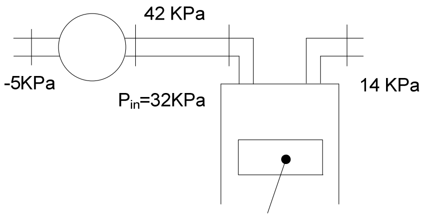
**Example7-6**:The compression ratio for a six cylinder, 4-stroke, supercharged C.I engine is 15:1. The bore is 130 mm and stroke 150 mm. The indicated thermal efficiency is 0.41 and the A/F is $\frac{18.5}{1}$. The lower heating value of the fuel is 44 MJ/kg. The intake and delivery pressure for the rotary blower are −5 kPa and 42 kPa gauge respectively. The blower mechanical eff. is 0.65. The average exhaust pressure is 14 kPa gauges and the average pressure during intake is 32 kPa gauges. The volumetric eff. for engine is 1.08. The mechanical friction power for engine is equivalent to 12.5% of the indicated power. The engine operates at 900 rpm and the atmospheric pressure and temperature are 100 kPa absolute and 25°C respectively. Determine:

a)Brake power for the engine.

b) Brake specific fuel consumption.

c) Brake thermal efficiency.

**Solution:**



$$V_s = (V_s \times n) = \frac{\pi}{4} b^2 L N n$$

$$= \frac{\pi}{4}\left(\frac{130}{1000}\right)^2 \times \frac{150}{1000} \times \frac{900}{2\times 60} \times 6 = 0.0869 \quad \mathrm{m^3/sec}.$$

$$\eta_v = \frac{\dot{V}_{air}}{\dot{V}_s} \Rightarrow \dot{V}_{air} = 1.08 \times 0.0896 = 0.096762$$

$$\rho_{air} = \frac{P}{RT} = \frac{100}{0.287(25+273)} = 1.169\,\mathrm{kg/m^3}$$

$$\therefore \dot{m}_a = \rho_{air} V_{air} = 1.169 \times 0.096762 = 0.11314\,\mathrm{kg/sec.}$$

$$\therefore \dot{m}_f = \frac{\dot{m}_a}{A/F} = \frac{0.11314}{18.5} = 6.116 * 10^{-3}\,\mathrm{kg/sec.}$$

$$\eta_{Ith} = \frac{i.p}{m_f * L.C.V} \qquad \therefore \mathrm{i.p} = 0.41 * 6.116 * 10^{-3} * 44000 = 110.324\,\mathrm{kW}$$

$$f.p_m = \frac{12.5}{100} \times i.p = \frac{12.5}{100} * 110.324 = 13.79\,\mathrm{kW}$$

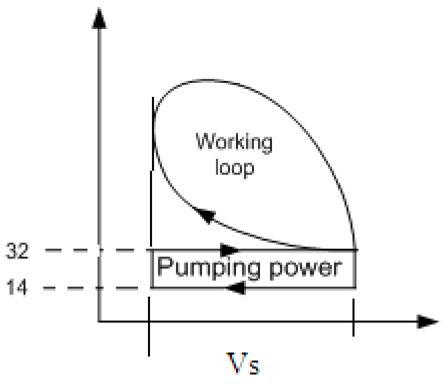
$$P_{blower} = \frac{\Delta P \ \dot{m}_a V_{air}}{\eta_m}$$

$$V_{air} = \frac{RT}{P} = \frac{0.287(25+273)}{100-5} = 0.9\,\mathrm{m^3/kg}$$

P= {[42-(-5)]×0.11314×0.9}/ 0.65 = 7.365 kW

Pumping power (p.p.) = $\Delta p \times \dot{V}_s$

p.p. = (32-14)×0.0896=1.613 kW



a) b.p = i.p+p.p.−f.p$_m$−P$_{blower}$

b.p =110.324+1.613 −13.79 −7.365

= 90.782 kW

b) bsfc = $\frac{6.11 \times 10^{-3}}{90.7} \times 3600$ = 0.24 kg/kW.hr

c) $$\eta_{bth} = \frac{b.p}{\dot{m}_f \times LCV} = \frac{90.782}{6.116 \times 10^{-3} \times 44000} = 0.3373 = 33.73\%$$

# CHAPTER (8)

## TWO – STROKE ENGINE

Both S.I. and C.I. engines can be designed so that one complete cycle of events in the cylinder is completed in two strokes instead of four. The difference between two-strokes and four stroke engines is in the method of filling the cylinder with fresh charge and removing the burned gases from the cylinder. The piston stroke in two strokes engine is longer because part of the compression and expansion strokes are used for the process of exhaust and induction. Fig. 8-1 shows a section of a common type of two stroke engine.

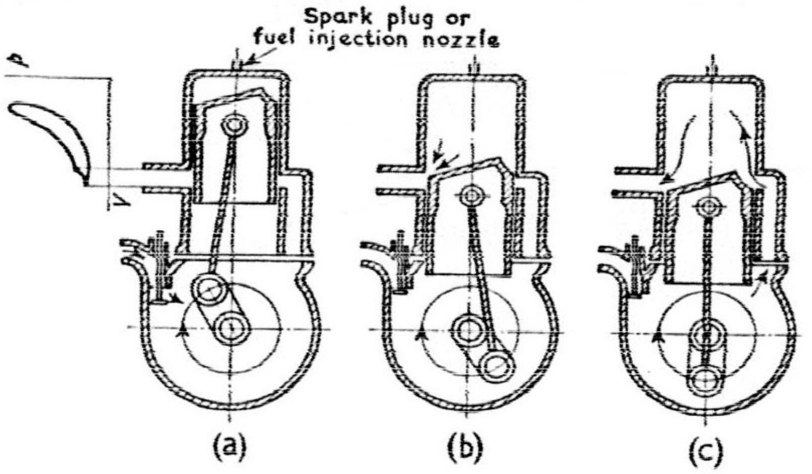


***Figure (8 – 1)*** *The two – stroke engine*

In Figure(8-1a) the piston is shown near the end of compression stroke. The upward motion has decompressed the crank – case and air has been admitted through the self-acting valve to the crank-case. During the expansion stroke the air in the crank-case is compressed, and near the end of this stroke (b) the exhaust port is uncovered to allow the hot gases to blow down the exhaust duct. Further movement of the piston uncovers the inlet port (c), and compressed air from the crank-case flows into the cylinder. The exhaust and inlet ports are open simultaneously for a short period so that the incoming air can assist in cleaning the cylinder from combustion products.

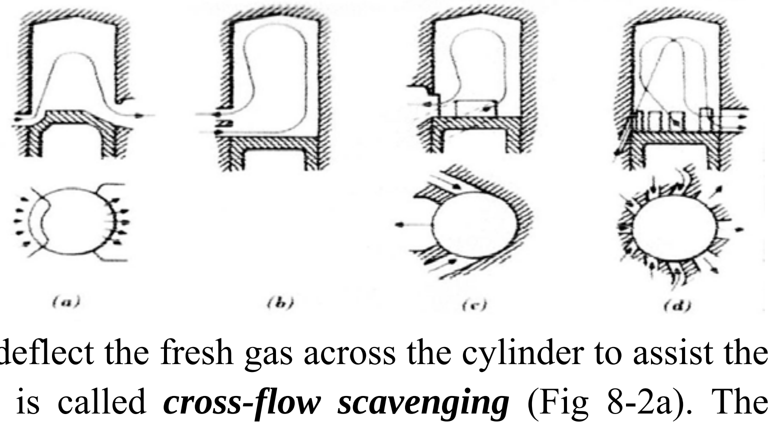


***Figure (8 – 2)***

*(a) Cross scavenging;*
*(b) MAN loop scavenging;*
*(c) Schnuerle loop scavenging;*
*(d) Curtiss loop scavenging.*

The piston can be shaped to deflect the fresh gas across the cylinder to assist the "scavenging" of the cylinder; this is called ***cross-flow scavenging*** (Fig 8-2a). The scavenging may be done by using two transfer ports (Fig 8-2c) which direct the incoming air up the cylinder; this is called ***inverted flow scavenging***. In ***loop scavenging*** the inlet and exhaust ports are on the same side (fig 8-2b). Ideal scavenging can be obtained by locating exhaust (or inlet) valve in the head, ***uniflow scavenging***.

For the same power output, more air is required in a two – stroke engine than in a four – stroke engine. This is because some of the air is lost in the overlap period of the scavenging process. Volumetric efficiency of the four- stroke engine is replaced by either delivery ratio or charging efficiency.

$$\text{Delivery ratio } \lambda_{dr} = \frac{m_{mi}}{V_s \rho_a} \equiv \text{scavenging ratio } \lambda = \frac{\dot{m}_{mi}}{\dot{m}_{cy}}$$

$$\text{Charging efficiency, } \eta_{CH} = \frac{m_{mt}}{V_s \rho_a} \equiv \text{scavenging efficiency } \lambda_{sc} = \frac{m_{mi}}{m_{cy}}$$

$$\text{Trapped efficiency } \eta_{tr} = \frac{\dot{m}_{mt}}{m_{mi}}$$

Where: $m_{mi}$ =mass of air-fuel mixture supplied

$m_{mt}$ = mass of air-fuel mixture trapped in cylinder after valves closes

$m_{cy}$ = mass of air-fuel mixture which could be trapped in cylinder volume

$V_s$= swept volume

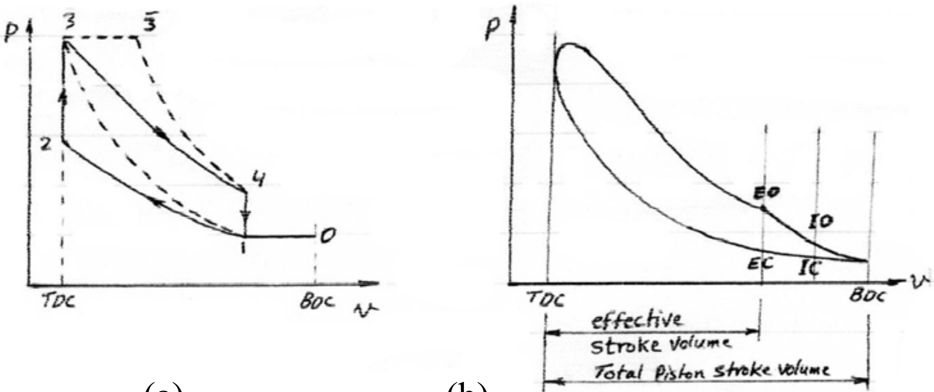
$\rho_a$=density of air at ambient conditions

Delivery ratio ($\lambda_{dr}$) is greater than charging efficiency ($\eta_{ch}$), because some of the air-fuel mixture supplied is lost though the exhaust port before it is closed. For engines that inject fuel after the values are closed, the mass of mixture should be replaced with mass of air.

The compression ratio of two- stroke engine is defined as :

$$C\bar{R} = \frac{\text{total volume above exhaust ports}}{\text{clearence volume}}$$

## Theoretical and actual cycle for two-stroke engine:



(a) (b)

***Figure (8 – 3)*** *(a) Theoretical p – v diagram; (b) Actual p – v diagram*

The m.e.p of the 2-stroke cycle engine may be measured on the basis of effective stroke or on the basis of total stroke. It is obvious the former is greater than the later.

## Valve Timing Diagram for 2-Stroke Engine:

The valve timing diagram for two-stroke engine is shown in fig.8-4; (a) petrol engine,(b) diesel engine,(c) gas engine. These timing diagrams are self-explanatory.

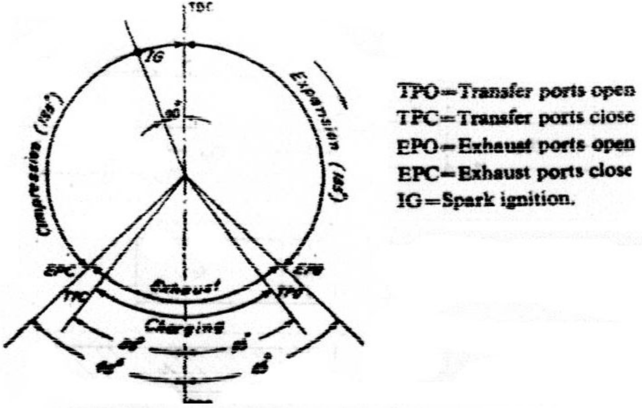
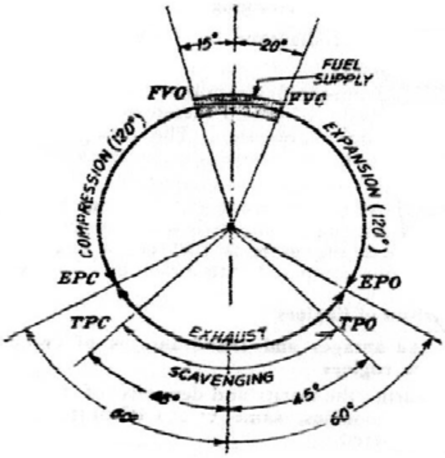


***Figure (8 – 4a)*** *Valve timing Diagram for 2 – stroke petrol engine*



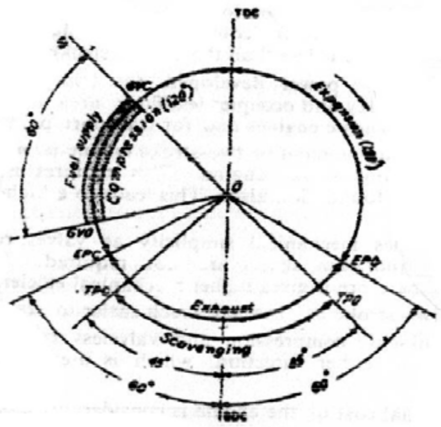


***Figure (8 – 4b)*** *Valve timing for two – stroke diesel engine*

***Figure (8 – 4c)*** *Valve timing diagram for 2 – stroke gas engine*

## Comparison of Two-Stroke and Four Stroke Engines:

### Advantages of two – stroke engine:

1. 2- stroke engine gives twice as many power strokes as a four stroke cycle engine at the same speed; therefore, the two-stroke engine should develop twice the power of four- stroke engine.
2. For the same power developed, the –stroke engine is much lighter.
3. Turning moment of 2-stroke engine is more uniform.
4. It provides mechanical simplicity.
5. Easier to start.
6. Initial cost is less.

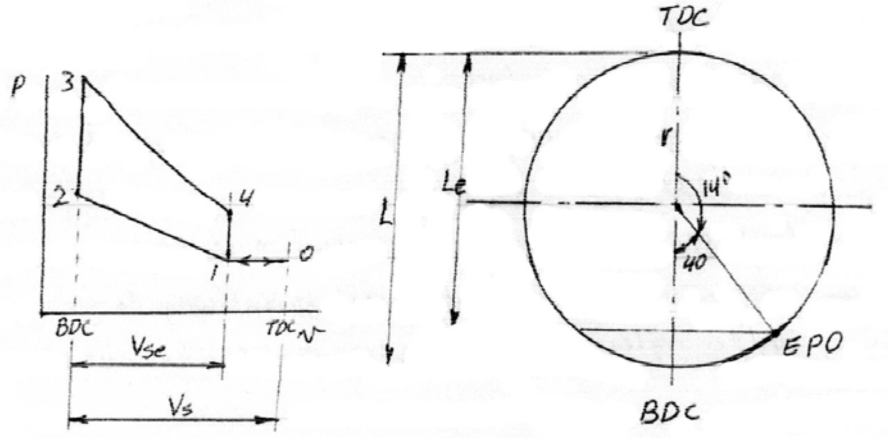
## Disadvantages:

1- Lower thermodynamic efficiency, because the effective compression ratio is less.
2- Portion of the fresh charge is escape through the exhaust port,therefore,sfc is higher.
3- The capacity of the cooling system must be higher.
4- Consumption of lubricating oil is sufficiently large.
5- Sudden release of burnt gases makes the exhaust more noisy.
6- The fresh charge is highly polluted by residuals from previous cycle.

## Example 8-1:

The stroke and diameter of the two stroke petrol engine are 14 cm and 10 cm respectively. The clearance volume is 157 $cm^3$/ If the exhaust ports open after $140^o$ after TDC, find the air standard efficiency of the cycle.

Solution:



Effective stroke = r +r cos $\theta$

Where r is crank radius which is $\frac{1}{2}$ stroke and $\theta$=$40^o$

Le=7+7 cos 40 = 12.43 cm

Effective stroke volume Ve = $\frac{\pi}{4} d^2 * Le = \frac{\pi}{4}(10)^2 * 12.43 = 975$ cm

Effective compression ratio = $\frac{V_{se}+V_c}{V_c} = \frac{975+157}{157} = 7.23$

$$\eta_a = 1-\frac{1}{r_e^{\gamma-1}} = 1-\frac{1}{7.23^{1.4-1}} = 1-\frac{1}{2.206} = 0.547$$

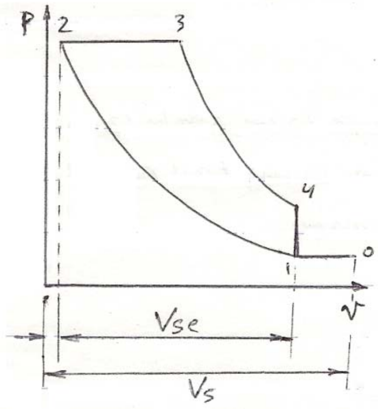
## Example 8-2:

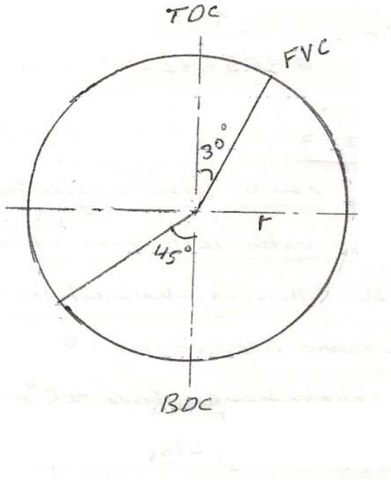
The diameter and stroke of two stroke diesel engine are 8 cm and 12 cm respectively. The clearance volume is 36.2 $cm^3$.The exhaust ports close after $45^o$ crank angle from BDC.

The fuel valve opens at TDC and closes $30^o$ crank angle after TDC. Find the air-standard efficiency of the cycle.

If the relative efficiency is 50% find the specific fuel consumption assuming the C.V. of the fuel used is; 42000 kJ/kg.

**Solution:**





$L_e = r\,(1+\cos\theta) = 6\,(1+\cos 45) = 10.242$ cm

$V_3 = (r - r\cos 30)\,A + V_c = 6\,(1-\cos 30)\frac{\pi}{4}\times 8^2 + 36.2$

$= 36.2 + 6.734 = 42.934\ \text{cm}^3$

$V_{se} = \frac{\pi}{4}d^2 \,.\, L_e = \frac{\pi}{4}\times 64 \times 10.242 = 515\text{cm}^3$

Effective compression ratio $= \frac{V_{se}+V_c}{V_c} = \frac{515+36.2}{36.2} = 15$

Cut- off ratio $= \frac{V_3}{V_2} = \frac{42.934}{36.2} = 1.186$

$$\eta_a = 1 - \frac{1}{r_e^{\gamma-1}}\left[\frac{1}{\gamma}\left(\frac{\left(\frac{V_3}{V_2}\right)^{\gamma}-1}{\left(\frac{V_3}{V_2}\right)^{\gamma}-1}\right)\right] = 1 - \frac{1}{1.4\times 15^{0.4}}\left(\frac{1.186^{1.4}-1}{1.186-1}\right)$$

$$= 1 - \frac{1}{4.136}\left(\frac{1.269-1}{0.186}\right) = 0.65$$

$$\eta_r = \frac{\eta_{IT}}{\eta_a}$$

$$\eta_{IT} = 0.5\times 0.65 = 0.325$$

$$\eta_{IT} = \frac{ip}{m_f \times CV}$$

$$m_f = \frac{ip}{0.325\times 42000}$$

$$sfc = \frac{3600}{0.325\times 42000} = 0.264 \quad \text{kg/kw.h}$$

## Example 8-3:

A 2-stroke diesel engine having bore 10 cm, stroke 12 cm, compression ratio 16:1, runs at 1500 rpm. During trail run of the engine, the following observations were made:

Atmospheric pressure 103 kPa

Atmospheric temperature 35°C

Air supplied by blower = 125 kg/hr

If the scavenging efficiency of the engine is 57.4%, calculate the scavenging ratio, trapped efficiency, and charging efficient.

## Solution:

Scavenging efficiency = $\frac{m_{mt}}{m_{cy}} = \eta_{sc}$

$m_{cy} = V_{cy} \times P_a$

$$V_{cy} = \frac{r}{r-1} V_s = \frac{16}{16-1} V_s$$

$$\eta_{CH} = \frac{m_{mt}}{V_s P_a}$$

$$\rho_a = \frac{P}{RT} = \frac{103}{0.287 \times 308} = 1.16 \quad \text{kg/m}^3$$

$$\dot{V}_S = \frac{\pi}{4} \times \left(\frac{10}{100}\right)^2 \times \frac{12}{100} \times \frac{1500}{60 \times 1} = 0.02356 \quad \text{m}^3/s \quad \text{or} \quad 1.4136 \, \text{m}^3/\text{min}.$$

$$\dot{V}_{cy} = \frac{16}{15} \times 0.02356 = 0.025133 \ \ \text{m}^3/s$$

$$\dot{m}_{mt} = \eta_{sc} \times m_{cy} = 0.574 \times 0.025133 \times 1.16 = 0.01673 \quad \text{kg/s}$$

Scavenging ratio $\lambda = \frac{m_{mi}}{m_{cy}} = \frac{125}{3600 \times 0.025133 \times 1.16} = 1.19$

Trapped efficiency $\eta_{trap} = \frac{m_{mt}}{m_{mi}} = \frac{0.01673 \times 3600}{125} = 0.4818$

$$\eta_{CH} = \frac{m_{mt}}{V_s P_a} = \frac{0.01673}{0.02356 \times 1.16} = 0.612$$

## Example 8-4:

A 2-stroke single cylinder diesel engine has a bore 125 mm, stroke 150 mm, compression ratio 15:1, runs on 1800 rpm, the atmospheric conditions are 300 k and 1 atm. The trapping efficiency is 60%, air / fuel ratio= 30:1, LCV = 43000 kJ/kg, imep = 4.36 bar and $\eta_{IT}$ =35%. Evaluate:

a) Scavenging ratio

b) charging efficiency

c) Delivery ratio

d) scavenging efficiency

**Solution:**

$$\dot{V}_s = \frac{\pi}{4}\left(\frac{125}{1000}\right)^2 \times \frac{150}{1000} \times \frac{1800}{60 \times 1} = 0.055\,\mathrm{m^3/s}$$

$$\dot{V}_{cy} = \frac{r}{r-1}V_s = \frac{15}{14} \times 0.055 = 0.0591\,\mathrm{m^3/s}$$

$$\rho_a = \frac{100}{0.287 \times 300} = 1.161$$

$$i.p = p_i \dot{V}_s = 4.36 \times 100 \times 0.055 = 24\,\mathrm{kW}$$

$$m_f = \frac{ip}{\eta_{IT} \times LCV} = \frac{24}{0.35 \times 43000} = 0.0016\,\mathrm{kg/s}$$

$$m_{mt} = m_f \times \frac{A}{F} = 0.0016 \times 30 = 0.048\,\mathrm{kg/s}$$

$$\eta_{trap} = \frac{m_{mt}}{m_{mi}} \Rightarrow m_{mi} = \frac{0.048}{0.6} = 0.08\,\mathrm{kg/s}$$

a) $\lambda = \dfrac{m_{mi}}{m_{cy}} = \dfrac{0.08}{0.0591 \times 1.16} = \dfrac{0.08}{0.0684} = 1.169$

b) $\eta_{CH} = \dfrac{m_{mt}}{V_s \rho_a} = \dfrac{0.048}{0.055 \times 1.16} = 0.7523$ or 75.23%

c) $\lambda_{dr} = \dfrac{m_{mi}}{V_s \rho_a} = \dfrac{0.08}{0.0638} = 1.25$

d) $\eta_{sc} = \dfrac{m_{mt}}{m_{cy}} = \dfrac{0.048}{0.0684} = 0.7$ or 70%

# *CHAPTER (9)*

# COMBUSTION IN I.C. ENGINES

## A- Combustion in SI Engines

Combustion in spark ignition engines normally begins at the spark pluck where the molecules in and around the spark discharge are activated to a level where reaction is self- sustaining. Combustion is SI engines may be one or more of the following types of combustion:

1- Normal combustion.
2- Abnormal combustion.
3- Uncontrolled combustion.

### 9-1 Normal combustion:

Once ignition has started the flame front expands across the chamber until it engulfs the whole mixture. Two stages may be distinguished during the normal combustion. The first stage, (AB) correspond to the time for the formation of the self propagation nucleus of the flame. This is mainly a chemical process and depends on the nature of mixture composition, temperature and turbulence.

The second stage (BC) corresponds to the propagation of the flame throughout the combustion chamber. The second stage begin at the point where first measurable rise of pressure can be seen on the indicator diagram. The rate of pressure rise is proportional to the rate of heat release because during this stage the combustion volume is nearly constant.

The spark occurs at the point A, there is a "**delay period**" between the occurrence of the spark and the noticeable pressure rise from that of motoring compression. This is a time delay which is independent of engine speed so that as the engine speed is increased the point A must occur earlier in the cycle to obtain the best position of the peak pressure.

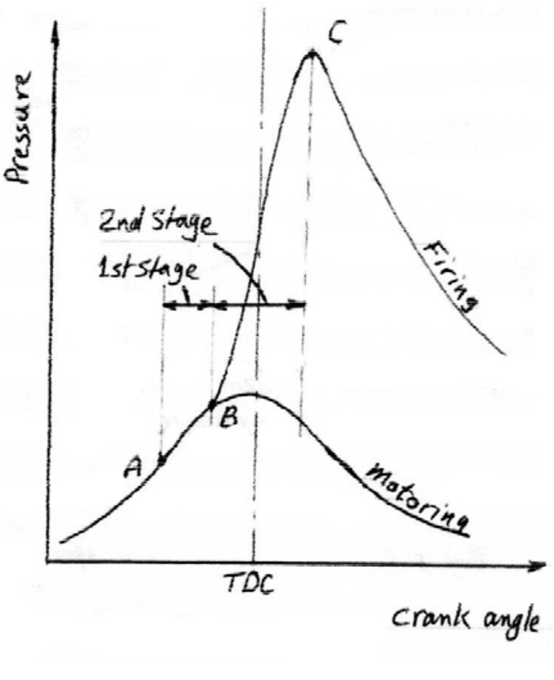


*Figure (9 – 1)*

Although the point C marks the completion of the flame travel, it does not follow that at this point the whole of the heat of the fuel has been liberated, some further chemical adjustments due to re-association, etc., and what is generally referred to as after – burning, will to a greater or less degree continue throughout the expansion stroke.

Effect of engine variables on first stage:

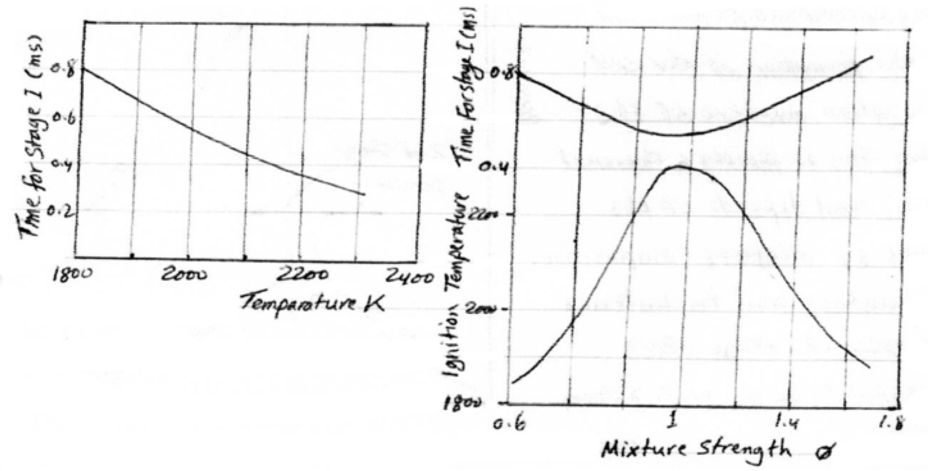
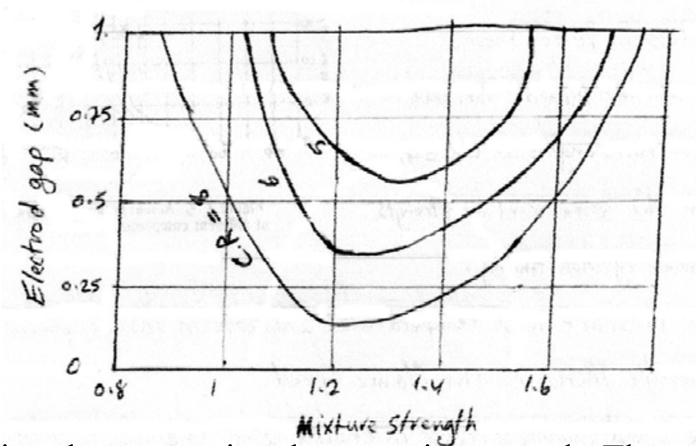


*Figure (9 – 2)*

Electrode gap: If the gap is too small, quenching of the flame nucleus may occur and the range of fuel / air ratio for the development of a flame nucleus is reduced.

*Figure (9 – 3)*

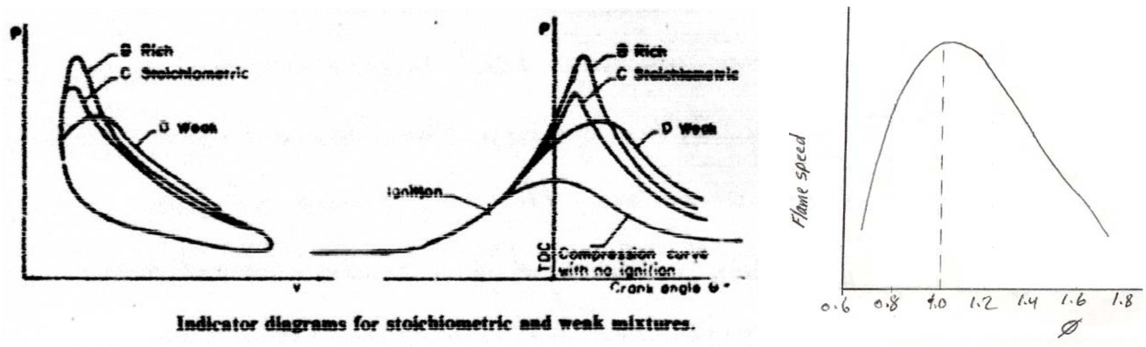


Turbulence: Turbulence is directly proportional to engine speed. Therefore increase in engine speed does not affect much ignition lag measured in millisecond.

Effect of engine variable on second stage:

There are several factors which affect the second stage (flame speed) such as:

1- fuel / air ratio:



**Indicator diagrams for stoichiometric and weak mixtures.**

*Figure (9 – 4)*

2- Compression Ratio:

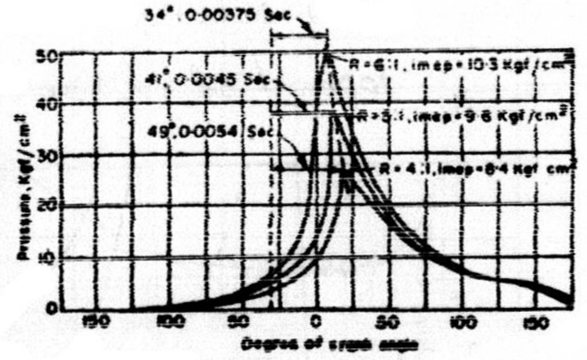


***Figure (9 – 5)** Actual indicator diagram at different compression ratios.*

Fig. 9-5 shows the increased speed of combustion with increase of compression ratio. These diagrams are for Ricardo variable compression ratio engine at CR=4, 5, 6with the same mixture strength and the same ignition timing.

3-Intake pressure and temperature: increase in intake pressure and temperature increases the flame speed.

4-Engine load: with increase in engine load the cycle pressure increases hence the flame speed increase.

5-Turbulence: Turbulence plays a very vital role in combustion phenomenon. The flame speed is very low in non-turbulent miniatures. A turbulent motion of the mixture intensifies the process of heat transfer and mixing of the burned and unburned portion in the flame front. These two factors cause the velocity of turbulent flame to increase.

## 9-2 Abnormal Combustion:

Normal combustion rarely occurs in a real engine without some trace of auto ignition appearing. After ignition, the flame front travels across the combustion chamber. The gas a heat of the flame front called the"**end gas** ". The end gas receives heat due to compression by expanding gases and by radiation from the advancing flame front, therefore, its temperature and density increases. If the temperature exceeds the self – ignition temperature and the un-burnt gas remains at or above this temperature for a period of time equal to/or greater the delay period, spontaneous ignition (or auto ignition) will occurs at various locations. Shortly after words an audible sound called **knock** appears.

If the end gas does not reach its self-ignition temperature, the combustion will be normal.

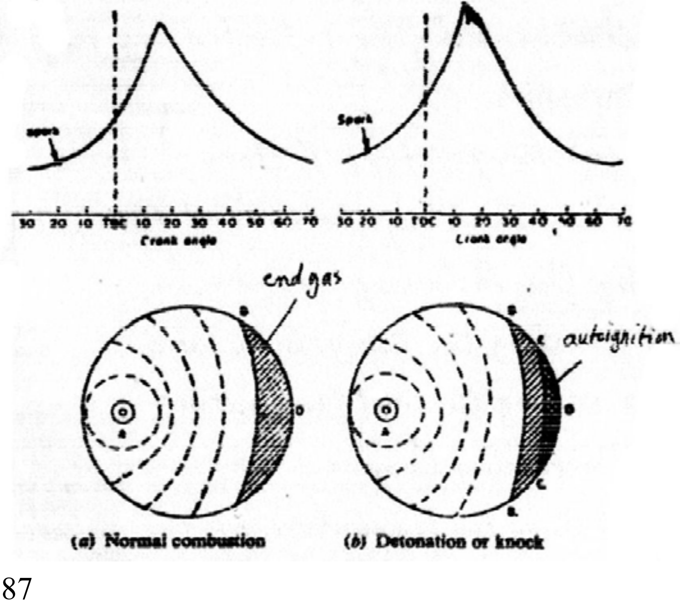


***Figure (9 – 6):** Combustion in SI engine.*

## Effect of Knock:

Knock has the following effects on engine operation:

1. Noise and Roughness.
2. Mechanical damage: increase in engine wear, cylinder head and valves may be pitted.
3. Carbon deposits.
4. Increase in heat transfer.
5. Decrease in power output and efficiency.
6. Pre-ignition: combustion Occurs before the spark.

## Effect of engine variables on Knock:

To prevent Knock in the S.I. engine the end gas should have:

A- Low temperature.
B- Low density.
C- Long ignition delay.
D- Non- reactive combustion.

When the engine conditions are changed, the effect of the change may be reflected by more than one of the above variables.

### A- Temperature factors:

The temperature of the unburned mixture is increased by the following factors:

1. Raising the compression ratio.
2. Supercharging.
3. Raising the inlet temperature.
4. Raising the coolant temp.
5. Increasing load.
6. Advancing the spark.
7. Raising the temperature of the cylinder and combustion chamber walls.

### B- Density factors:

Increasing density by any of the following methods, will increase the possibility of Knock:

1. Increasing load.
2. Increasing compression ratio.
3. Supercharging.
4. Advancing the spark.

### C- Time factors:

Increasing the time of exposure of the unburned mixture to auto-ignitions by any of the following factors will increase tendency to Knock:

1. Increasing the distance of the flame travel.

2. Decreasing the turbulence of mixture.
3. Decreasing the speed of the engine.

**D- Composition:**

The probability of Knock in S.I. engines is decreased by:

1. Increasing the octane rating of the fuel.
2. Either rich or lean mixtures.
3. Stratifying the mixture.
4. Increasing the humidity of the entering air.

## 9-3 Knock rating of S.I. Fuels:

The Knock rating of a gasoline is found by comparing the Knock of the fuel under test with that of a blend of primary reference fuels (PRF). These fuels are n-heptane ($C_7H_{16}$), which have a very low auto ignition reaction time and treated as fuel with octane number (ON) of O and 2,2,4- trimethyl pentane (iso- octane), which has inactive auto ignition reaction and is treated as ON equal to 100. The fuel is rated by the percent of iso - octane in the n- heptane and iso- octane mixture. The scale of octane rating is extended above 100 by adding Tetra- ethyl- lead (TEL) to iso- octane, and:

$$ON\left(above\ 100\right) = 100 + \frac{6.221\,T}{1 + 0.1619\,T + \sqrt{1 + 0.3238\,T - 0.001704\,T^2}}$$

Where T = ml TEL/ lt

There are several methods of Knock rating to suit the various matching conditions for different engines and operating variables, and in each of these methods, a standard engine built to exacting specifications must be run under prescribed operating conditions.

The standard engine used for either the research or motor method is the CFR (Co - operative Fuel research) engine.

Motor and research methods: the engine must first be calibrated under specified conditions, such as those indicated below:

| **Test method** | **Engine speed (rpm)** | **Coolant temp. (°C)** | **A/f ratio** | **Spark timing (bTDC)** |
|---|---|---|---|---|
| Research | 600 | 100 | Max | 13 |
| Motor | 900 | 100 | Max | 14-26 |

The sensitivity of a fuel to knock is measured by the difference in the two knock ratings; the greater the difference the greater the fuel sensitivity. The Octane rating of fuel is lower in the motor method than in the research method.

## Influence of fuel additive on knock:

It is the possible to raise the Octane number of fuel by refining methods alone, but these methods are in general expensive. These additives are used to raise ON of the fuel, control surface ignition, reduce spark plug fouling, resist gum formation, prevent rust, reduce carburetor icing, remove carburetor or injector deposits, minimize deposits in intake system, and prevent valve sticking. The most effective antiknock agents are lead alkyls. Tetraethyl lead (TEL), $(C_2H_5)_4$ Pb, was first introduced in 1923. Tetramethyl lead (TML), $(CH_3)_4$ Pb, was introduced in 1960. In 1959 manganese antiknock compound (methylcyclopentadienyl manganese tricarbonyl), MMT, was introduced.

The intereste in unleaded gasoline was started in 1970 because of the issue of legislation in many developing countries which control harmful pollutants. Lead has toxicological effect in the urban environment and the use of catalytic devices for emission control was introduced. The expanding use of unleaded fuels has increased interest in other methods of increasing the ON of gasoline, one of these methods is; the oxygenates (alcohols and ethers) as fuel additives to increase Octane rating.

## 9-4 Uncontrolled Combustion:

Under certain conditions the fuel- air mixture is ignited by hot spot in the cylinder. The hot spot might be the spark plug insulator or electrode, or combustion deposits etc…. When ignition occurs before the spark the phenomenon is called *pre-ignition*. When the phenomenon occurs after ignition is switched off it is called *running- on*. Combustion deposit ignition is called *rumbling*.

# B- Combustion in CI engines:

In compression ignition engine usually air is drawn during intake stroke. A compression ratio between 12 and 20 is used, so that temperature of the air near the end of compression stroke is quite high. Just before TDC, fuel is sprayed into the combustion chamber. This spray of fuel with the aid air movement distributes the fuel through the air. Owing to the high temperature of the air, the fuel ignites and burns almost as soon as introduced.

Since the cylinder contents are not homogeneous (heterogeneous). They vary from fuel, F/A combustion products mixture to air alone in different regions. These regions may exist close to each other. Because of the dependence of combustion on the process of the fuel finding oxygen, C.I. engines require excess air to a greater extent than do S.I. engines if high efficiency and low smoke levels are to be achieved.

## 9-5 stage of combustion in C.I. engine:

From the commencement of injection, the combustion process may be divided into several stages.

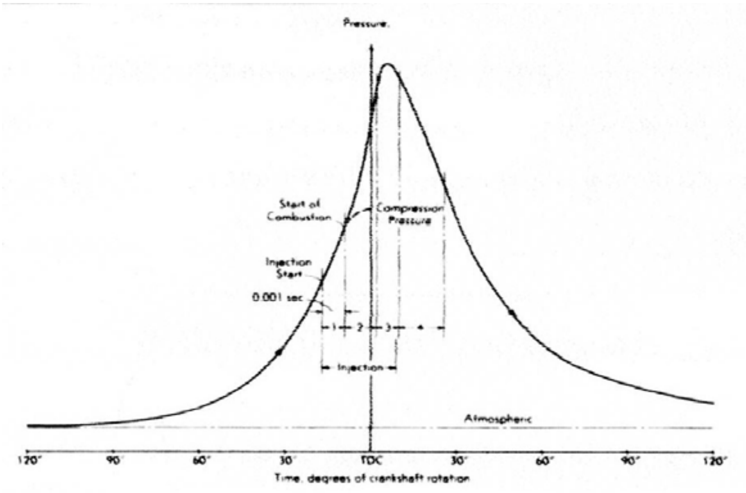


***Figure (9 – 7)*** *p – t diagram for C.I. engine at full load*

### 1. Delay Period:

During this period there is no visible pressure rise. The fuel is injected through nozzle, but does not ignite. There is a definite period for mixing of liquid fuel with air, vaporizing and mixing of fuel vapour with air, preflame reactions of fuel and then ignition. The delay period can be divided into two overlapping parts:

a- Physical delay: during which the cold fuel droplets are heated, vaporized, mixed with air and raised in temperature.
b- Chemical delay: during which chemical reaction proceeds very slowly and then accelerate until local inflammation or ignition takes place.

### 2. Rapid (uncontrolled) combustion:

During the delay period an appreciable amount of fuel has been prepared, fuel burns very rapidly, the velocity of burning is controlled by chemical kinetics. There is also partial combustion of fuel which continues to be injected during this period.

### 3. Controlled combustion (diffusion flame):

The rate of combustion in this stage is determined mainly by how rapidly the fuel vapour are heated and mixed with air. During this stage, the moving piston increase the volume of the cylinder and thus the end of this stage is to the right of TDC, this end is characterized by the point of maximum cycle temperature. This period is affected by: rate of injection of fuel- Turbulence in the cylinder- injection pressure.

### 4. Tail of combustion (after burning):

This stage commences from the point of maximum cycle temperature, the rate of heat release in this phase of combustion gradually drops to zero.

## 9-6 knock in the C.I. engine:

In C.I. engine, the fuel is injected into hot air and combustion begins with auto – ignition, if the first peak in the pressure rate diagram is high enough a knocking sound appears. This is due to very long delay period and a large amount of fuel would be prepared during the delay period.

If diesel knock is experienced in an engine a cure may be by means that either reduce the delay period or reduce the rate of injection during the delay period.

Knocking in C.I. engine is an auto ignition phenomenon same as in SI engine, but diesel knock requires cures diametrically opposite to those required for the SI engines.

**Knock rating of CI fuels:**

The knock rating of a diesel fuel is found by comparing the fuel under test in a special engine with primary reference fuels. These fuels are n- cetane (hexeadecane, $C_{16}H_{34}$) which has a low self ignition temperature and given cetane number of 100 and heptamethylnonane with a cetane number of 15. The cetane number for a blend is calculated by:

$$CN = \%\ \text{cetane} + 0.15 * \%\ \text{heptamethylnonane}.$$

In some text books the reference fuels are: cetane (100 CN) and $\alpha$- methylnaphthalene (CN = 0).

## 9-7 Firing order in multi- cylinder engines:

In multi- cylinder engines; the expansion strokes for the different pistons must be arranged to give suitable distribution of force, in this way the engine runs more quietly and smaller the flywheel would be. The crank angle between any two explosions, ensuring the best uniformity if crankshaft rotation should be as follows:

Four- stroke engines: $\phi = 720/ n$

Two- stroke engines: $\phi = 360/ n$

Where n is the number of cylinders.

Figure 9-8 shows diagrams of forces for individual cylinders and the summation force of an 8- cylinder four stroke engine.

The firing order is of more importance in multi- cylinder engines, because the exhaust valves remain open for some interval of crank motion, so two exhaust valves of two adjacent cylinders may open simultaneously. This overlapping will cause the exhaust of one of the two adjacent cylinders to 'below- over' into the other in which the exhaust stroke is nearly completing, thus interfering with the evacuation of the latter.

Blow- over can be minimized by using such a firing order that adjacent cylinders never fire in succession. The following table gives examples of the possible crank arrangements and firing order for four- stroke and two- stroke in line engines.

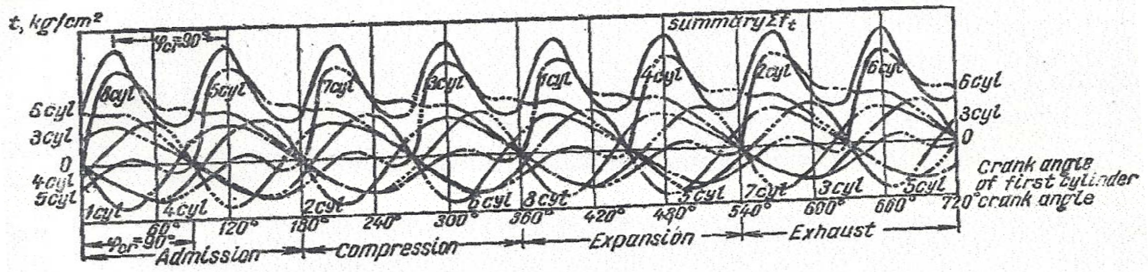
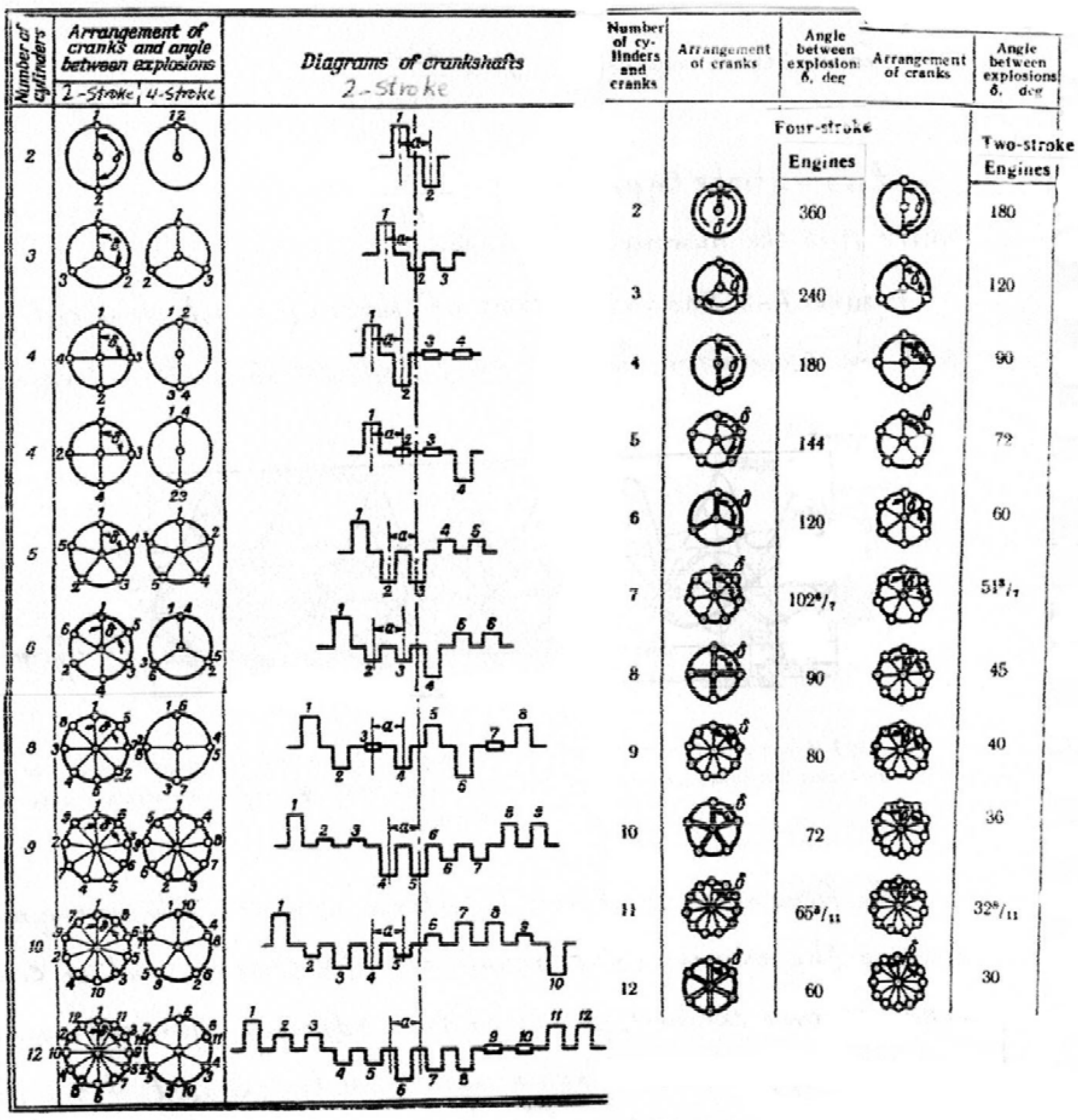


Fig. 8 Diagrams of tangential forces for individual cylinders and the total tangential force of an 8-cylinder four-stroke engine. Engine firing order 1-4-2-6-8-5-7-3; $\phi = \frac{720}{8} = 90°$—angular spacing of cranks



| Number of cy-linders and cranks | Arrangement of cranks | Angle between explosion δ, deg | Arrangement of cranks | Angle between explosions δ, deg |
|---|---|---|---|---|
| | | Four-stroke Engines | | Two-stroke Engines |
| 2 | | 360 | | 180 |
| 3 | | 240 | | 120 |
| 4 | | 180 | | 90 |
| 5 | | 144 | | 72 |
| 6 | | 120 | | 60 |
| 7 | | $102^{6}/_{7}$ | | $51^{3}/_{7}$ |
| 8 | | 90 | | 45 |
| 9 | | 80 | | 40 |
| 10 | | 72 | | 36 |
| 11 | | $65^{5}/_{11}$ | | $32^{8}/_{11}$ |
| 12 | | 60 | | 30 |

*Figure (9 – 9)*