...ducing all above mentioned modifications.

### 22.5.3. Valve Time Diagram for 2-Stroke Petrol Engine

The valve-timing diagram for actual working of the two-stroke petrol engine is shown in Fig. 22.15. The valve timing diagram is self-explanatory.

### 22.5.4. Working of two-Stroke Diesel Engine

Except that fuel is admitted near the TDC, the working of two-stroke cycle diesel engine is similar to that of the two-stroke petrol engine.

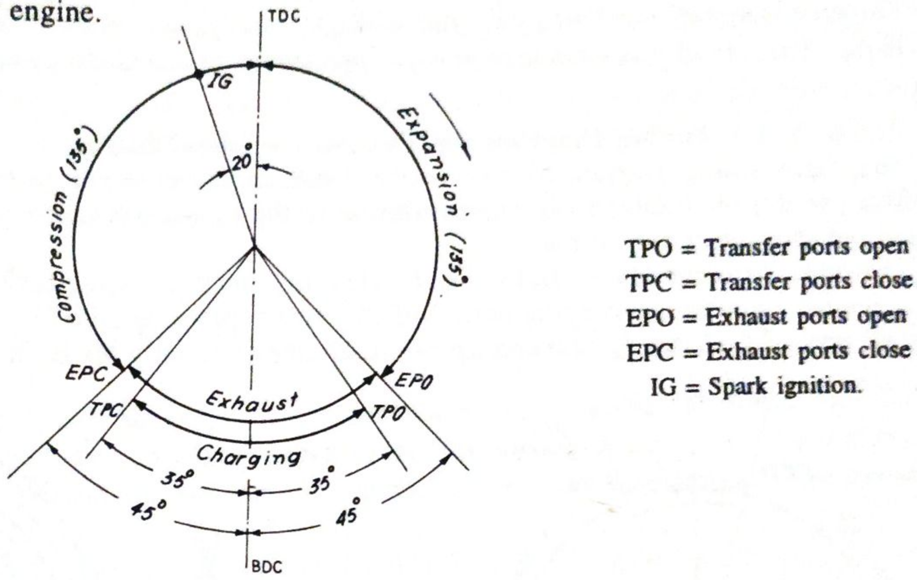


**Fig. 22.15.** Valve Timing Diagram for 2-stroke petrol engine.

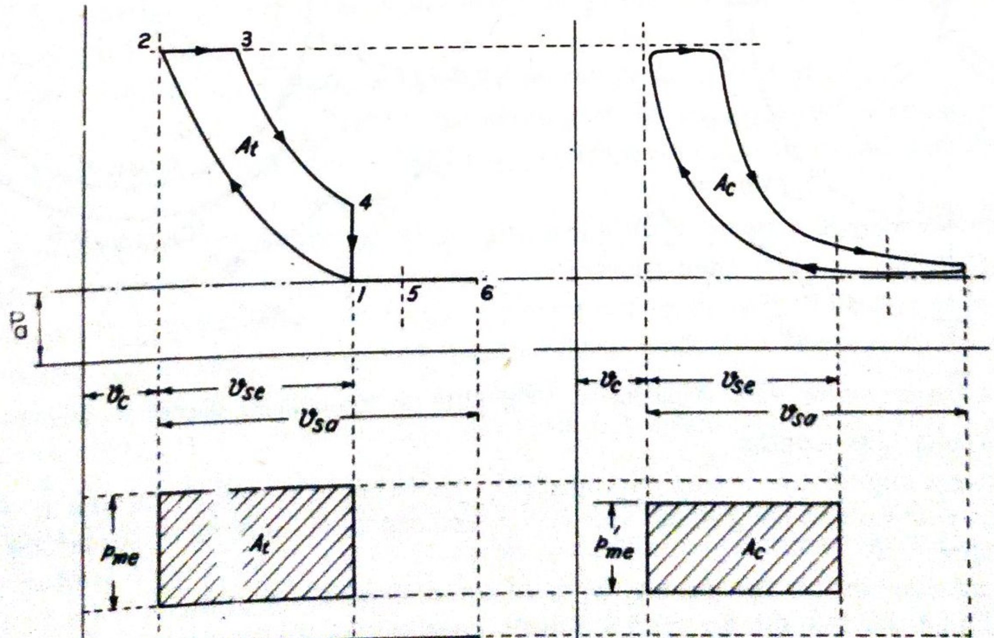


**Fig. 22.16.** Theoretical and actual *p-v* diagrams for two-stroke diesel engine.

### 22.5.5. Theoretical and Actual p-v Diagram for 2-Stroke Diesel Engine

The theoretical and actual *p-v* diagrams are shown in Figs. 22.16 (*a*) and 22.16 (*b*). The theoretical and actual mean effective pressure of the cycle on effective and actual length bases are also shown in Figs. 22.16 (*a*) and (*b*).

The engine works on Diesel cycle, therefore, the air standard efficiency of the cycle is given by

$$\eta_a = 1 - \frac{1}{R_{ce}^{\gamma - 1}}\left[\frac{\rho^{\gamma} - 1}{\rho - 1} \cdot \frac{1}{\gamma}\right] \quad \text{... (22.7)}$$

where $R_{ce}$ (effective compression ratio) $= \dfrac{v_{ce} + v_c}{v_c}$

The procedure for calculating the value of $R_{ce}$ is exactly same as described in two-stroke petrol engine.

**Note.** The cut-off is always mentioned in percentage of actual stroke and not on the basis of effective stroke.

### 22.5.6. Valve Timing Diagram for Two-Stroke Diesel Engine

The valve timing diagram for two-stroke diesel engine is shown in Fig. 22.17.

The valve timing diagram is exactly similar to the two-stroke petrol engine except the positions of opening and closing of fuel valve.

The other difference between two is, the charging and scavenging period of two-stroke diesel engine (90°) is greater than the scavenging period of two-stroke petrol engine (70°). This is because, there is no danger of loss of fuel during scavenging of diesel engine as only air is charged.

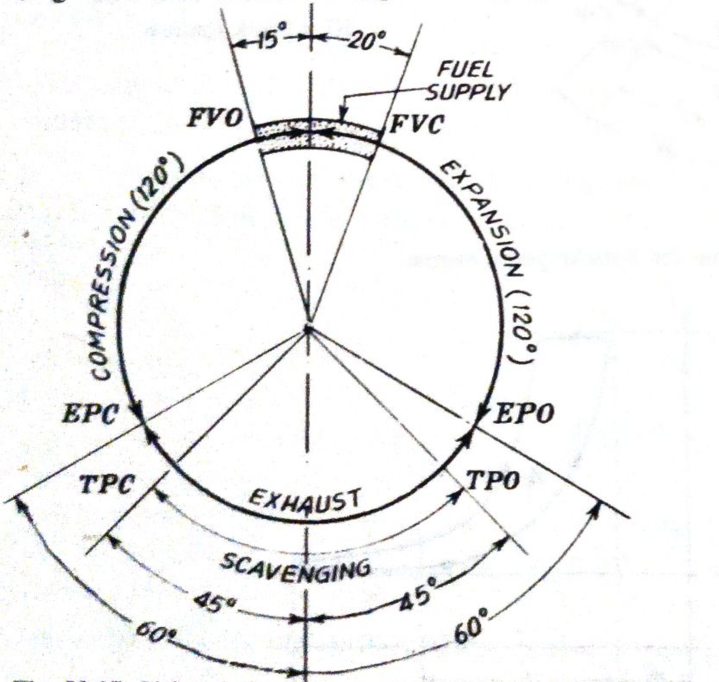


**Fig. 22.17.** Valve timing diagram for two stroke diesel engine.

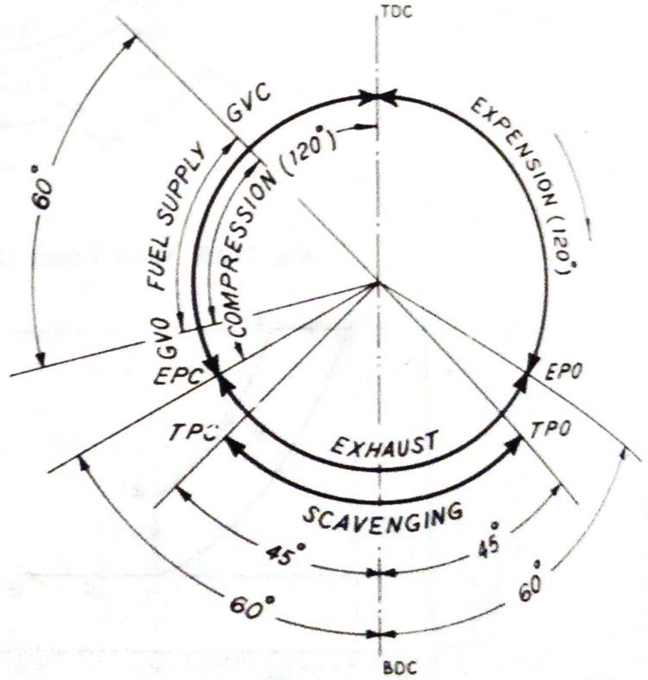


**Fig. 22.17** (*a*) Valve timing diagram for 2-stroke gas engine.

### 22.5.7. Two-Stroke Gas Engine

The two-stroke gas engine also works on Otto-cycle. Its working is exactly similar to the working of two-stroke petrol engine except, the gaseous fuel is supplied separately instead of supplying the air-fuel mixture during charging.

The valve timing diagram for the gas engine is shown in Fig. 22.17 (*a*).

It is obvious from the diagram that the gas valve opens after starting the compression of the air therefore it is necessary to supply the gaseous fuel at higher pressure. The pressure of supplied gas to the cylinder should be above the pressure of the air in the cylinder.

The fuel valve of two-stroke gas engine is kept open for more rotation of the crank angle compared with the diesel engine. This is necessary because the gaseous fuel is low in calorific value and high specific volume compared with diesel oil.

## 22.6. COMPARISON OF ENGINES

### 22.6.1. Advantages and Disadvantages of Two-Stroke Engines over Four-Stroke Engines

For comparing the merits and demerits of two-stroke engines over four-stroke engines, same values of output and speed of the engines are considered.

**Advantages**

1. A two-stroke engine gives twice as many power strokes as a four-stroke cycle engine at the same engine speed, therefore, the two-stroke engine of the same size should develop twice the power of a four-stroke engine. In practice, the actual power developed by two-stroke engine is about 1.7 to 1.8 times the power developed by four-stroke engine of the same dimensions and speed. This is because, some of the power is used for compressing the charge in the crank case and effective stroke is less than the actual stroke.
2. For the same power developed, the two-stroke engine is much lighter, less bulky and occupies less floor area and so is more suitable for use in marine engines and for transport purposes.
3. It provides mechanical simplicity as valves, rocker arms, push-rods, cam and cam shafts are not required. The friction loss is also less therefore it gives higher mechanical efficiency.
4. The two-stroke engines are much easier to start.
5. A crank-case compression and valveless type two-stroke engine can run in either direction, which is useful in marine applications.
6. The initial cost of the engine is considerably less.
7. The weight/H.P. ratio is considerably less.

**Disadvantages**

1. The thermodynamic efficiency of an engine is only dependent on the compression ratio. The effective compression ratio in the case of two-stroke engine is less than that for four-stroke engine for the same stroke (actual) and clearance volume. So the thermodynamic efficiency of two-stroke cycle is always less than four-stroke cycle engine.
2. The actual efficiency of the two-stroke cycle is less than four-stroke cycle engine because greater overlapping of the ports is necessary in two-stroke engine for effective scavenging. A portion of fresh charge in the case of S.I. engine always escapes unused through the exhaust ports, therefore, the specific fuel consumption is usually higher.
3. The power strokes per minute are twice the power stroke of four-stroke cycle engines, the capacity of the cooling system used must be higher. The cooling of the engine also presents difficulty as quantity of heat removed per minute is large. Due to the firing in each revolution, the piston is likely to get overheated and oil cooling of the piston is necessary.
4. The consumption of lubricating oil is sufficiently large because of high operating temperatures.
5. Sudden release of the burnt gases makes the exhaust more noisy.
6. A two-stroke petrol engine with crank case compression 50 to 60% of the swept volume filled with fresh charge while four-stroke petrol engine contains 80 to 95%. This is because, the space occupied by rotating parts in the crank case prevents a full charge being sucked in.
7. The scavenging is not complete particularly in high speed engines as very short time is available for exhaust and hence the fresh charge is highly polluted. This polluting can be reduced using opposed piston two-stroke diesel engine which provides undirectional scavenging.
8. The turning movement of 2-stroke engine is more non-uniform compared with 4-stroke engine, so it requires heavier flywheel and strong foundation.

It is possible to get expected performance of two-stroke engine only if the engine is built for a large output and good design.

Due to the excessive loss of combustible mixture with exhaust gases and resultant high fuel consumption, two-stroke cycle S.I. engines are not widely used except in the case of outboard motors, scooters, motor cycles and light vehicles. The current practice is that large two-stroke engines are generally C.I. engines. It can be said that two-stroke C.I. engines are better for slow and moderate speeds.

The use of two-stroke opposed piston C.I. engine is justified for marine installations where the engine room is short.

**22.6.2. Power of Engine**

1. **Indicated Horse Power (I.H.P.).** The actual power developed inside the cylinder is known as indicated power. If the net work developed per cycle is considered at constant mean pressure throughout the stroke of engine, instead of the actual varying pressure, then the work done per stroke is given by

$W$ = Force × Distance travelled

= (Mean pressure × area of piston) × (stroke of engine) = $P_m A \cdot L$.

If the number of working strokes per sec is $n$, then the I.P. given by

$$\text{I.P.} = \frac{P_m LAn}{1000} \text{ kW where } p_m = N/m^2,\ L = m,\ A = m^2 \quad \text{... (22.8)}$$

If the engine is a single cylinder, 4-stroke engine and running at $N$ r.p.m. then it gives one power stroke for every two revolutions, therefore, number of power strokes per minute are $N/2$.

$$\therefore \quad n = \frac{N}{2} \text{ for four-stroke engines} \quad \text{... (22.9)}$$

If the engine is a single cylinder, 2-stroke engine and running at $N$ r.p.m. then it gives one power stroke for every revolution, therefore number of power strokes per minute are $N$.

$$\therefore \quad n = N \text{ for two-stroke engine.}$$

If $p_m$ is the theoretical mean effective pressure then I.P. calculated is the theoretical I.P. of the engine. If $p_m$ is the actual mean effective pressure then I.P. calculated is the actual I.P. of the engine.

2. **Brake Power (B.P.).** A portion of the power developed inside the cylinder is lost in friction between the piston and cylinder, friction between the crank-shaft and main bearings, and friction between gears and valve mechanisms. Some of the power is used for running the fuel pumps, lubricating oil pumps, water-circulating pumps and governor. The net power available at the crank-shaft for doing useful work is always less than the actual I.P. of the engine. The net power available at the crank-shaft for doing useful work is known as brake power (B.P.). This available power is about 20 to 30% less than the normal.

The part of the I.P. lost (which is not available for external work) by different ways as mentioned above is known as frictional horse power (F.P.). the relation between I.P. and B.P. is given by

$$\text{I.P.} - \text{F.P.} = \text{B.P.} \quad \text{... (22.10)}$$

3. **Brake Mean Effective Pressure.** If the power available at the crank shaft per cycle is represented on a $p$-$v$ diagram, in the form of rectangular as shown in Fig. 19.18, then it is always less than the area on $p$-$v$ diagram which refers to the power developed inside the cylinder.

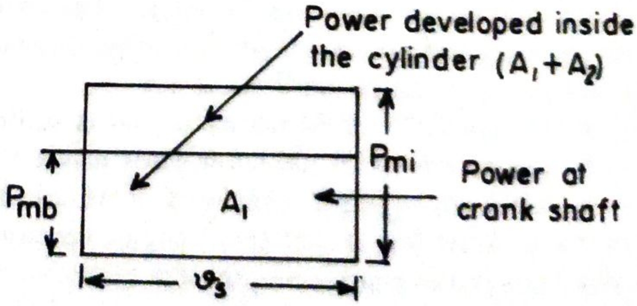


Fig. 22.18.

Referring to Fig. 22.18, $p_{mb}$ represents the brake mean effective pressure and $p_{mi}$ represents indicated mean effective pressure.

The brake horse power can be given by the formula as follows :

$$\text{B.P.} = \frac{p_{mb}LAN}{1000} \, kW \quad \text{... (22.11)}$$

Thus the brake mean effective pressure gives an indication of the power developed by the engine.

**22.6.3. Engine-Efficiencies**

(*a*) **Mechanical Efficiency.** The ratio of B.P. to I.P. is known as mechanical efficiency.

$$\therefore \quad \eta_m = \frac{\text{B.P.}}{\text{I.P.}} = \frac{\text{I.P.} - \text{F.P.}}{\text{I.P.}} \quad \text{... (22.12)}$$

$$= \frac{\text{B.P.}}{\text{B.P.} + \text{F.P.}} \quad \text{... (22.13)}$$

The mechanical efficiency depends on the following factors.

1. Method of cooling the engine, 2. Method of lubricating the engine,
3. Design of the engine, 4. Accuracy used in manufacturing the parts,
5. Alignment of different engine parts during assembly. 6. Load on the engine.

Referring to Fig. 22.18, the mechanical efficiency can also be given as

$$\eta_m = \frac{\text{power available at crank per power stroke}}{\text{power developed inside the engine per power stroke}}$$

$$= \frac{A_1}{A_1 + A_2} = \frac{v_s \times p_{mb}}{v_s \times p_{mi}} = \frac{p_{mb}}{p_{mi}} \quad \text{... (22.14)}$$

When there is no load on the engine or no useful work is taken from the crankshaft, then B.P. = 0, therefore at no load condition, $\eta_m$ is zero. The power developed inside the cylinder at no load condition is just sufficient to overcome the losses mentioned earlier under the head F.P. The frictional horse power increases with speed and is essentially constant with load.

(*b*) **Thermal Efficiency**

The brake thermal efficiency is the ratio of B.P. (output) to the heat energy of fuel supplied during the same interval of time (per minute)

$$\eta_{bt} = \frac{\text{Power developed/hr}}{\text{Energy supplied/hr}} = \frac{\text{B.P. (in kW)} \times 3600}{m_{fb} \times C.V.}$$

where $m_{fb}$ is fuel supplied per hour and C.V. is calorific valve of fuel in kJ/kg

$$= \frac{3600}{\dfrac{m_{fb}}{BP} \times CV} = \frac{3600}{C_b \times CV}$$

where $C_b$ is known as specific fuel consumption and its units are kg/kWh

The brake thermal efficiency is also known as overall thermal efficiency.

The indicated thermal efficiency is the ratio of I.P. to the heat of fuel supplied during the same interval of time (per hour).

$$\therefore \quad \eta_{it} = (\text{I.P. (in kW)} \times 3600) \times \frac{1}{m_{fi} \times \text{C.V.}}$$

$$= \frac{632.32}{\left(\dfrac{m_{fi}}{\text{I.P.}}\right) \times \text{C.V.}} = \frac{3600}{C_i \times CV} \quad \text{... (22.16)}$$

where $C_i$ is the fuel consumption per kW per hour on *IP* basis and it is known as specific fuel consumption on I.P. basis.

(*c*) **Relative Efficiency.** The ratio of indicated thermal efficiency to the air-standard efficiency is known as relative efficiency and it is given by

$$\eta_r \text{ (Relative efficiency)} = \frac{\eta_{it}}{\eta_a} \quad \text{... (22.17)}$$

If the indicated thermal efficiency is calculated on theoretical I.P. basis then $\eta_r$ is also on theoretical I.P. basis. If $\eta_{it}$ is calculated on actual I.P. basis then $\eta_r$ is also on actual I.P. basis.

Many times, the relative efficiency is also calculated on B.P. basis, then it is given by

$$\eta_r \text{ (Relative efficiency on B.P. basis)} = \frac{\eta_{bt}}{\eta_a} \quad \text{... (22.18)}$$