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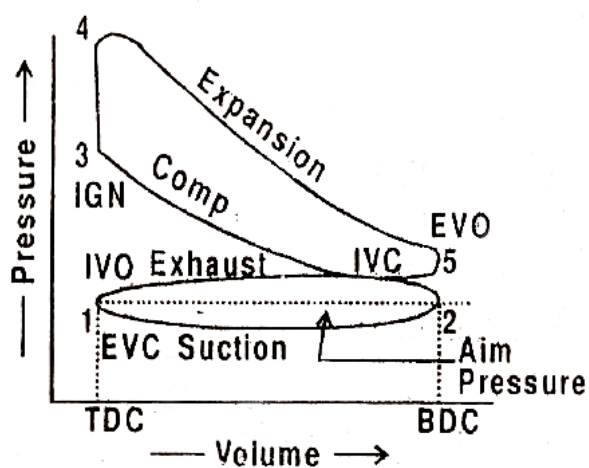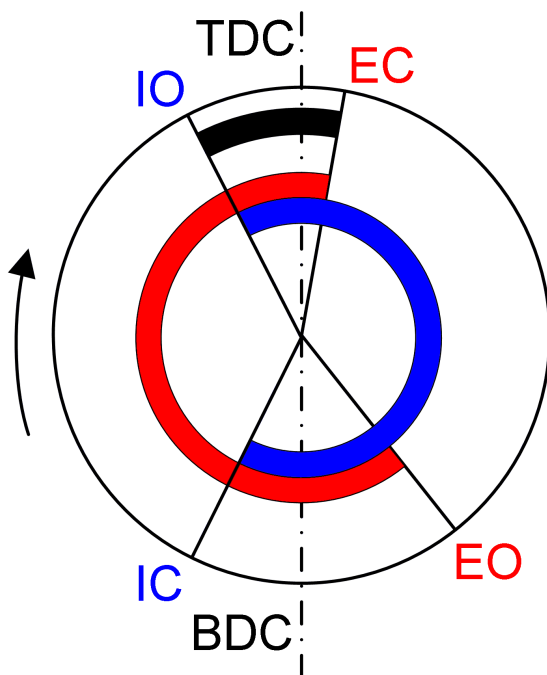
FUEL AIR CYCLES & ACTUAL AIR CYCLES

Indicator Diagram For A Four
Stroke Cycle Petrol Engine.

Course Contents

2.1 Fuel-Air cycle

2.1.1 Introduction

- The air cycle approximation of air standard theory has highly simplified assumptions. The air standard theory gives an estimate of engine performance which is much greater than the actual performance. For example the actual indicated thermal efficiency of a petrol engine of, say compression ratio 7:1, is of the order of 30% whereas the air standard efficiency is of the order of 54%.

- This large divergence is partly due to non-instantaneous burning and valve operation, incomplete combustion, etc. But the main reason of divergence is the oversimplification in using the values of the properties of the working fluid for cycle analysis.

- In the air cycle analysis it was assumed that the working fluid is nothing but air and this air was a perfect gas and had constant specific heats.

- In actual engine the working fluid is not air but a mixture of air, fuel and residual gases. Furthermore, the specific heats of the working fluid are not constant but increase as temperature rises, and finally, the products of combustion are subjected to dissociation at high temperature.

2.1.2 Factors considered for Fuel-Air cycle calculations

The following factors are taken into consideration while making fuel-air cycle calculations:

- **The actual composition of the cylinder gases:** The cylinder gases con-tains fuel, air, water vapour and residual gas. The fuel-air ratio changes during the operation of the engine which changes the relative amounts of CO_2, water vapour, etc.

- **The variation in the specific heat with temperature:** Specific heats increase with temperature except for mono-atomic gases. Therefore, the value of γ also changes with temperature.

- **The effect of dissociation:** The fuel and air do not completely combine chemically at high temperatures (above 1600 K) and this leads to the presence of CO, H_2, H and O_2 at equilibrium conditions.

- **The variation in the number of molecules:** The number of molecules present after combustion depends upon fuel-air ratio and upon the pressure and temperature after the combustion.

2.1.3 Assumptions made for Fuel-Air cycle analysis

- There is no chemical change in either fuel or air prior to combustion.

- Subsequent to combustion, the charge is always in chemical equilibrium.

- There is no heat exchange between the gases and the cylinder walls in any process, i.e. they are adiabatic. Also the compression and expansion processes are frictionless.

- In case of reciprocating engines it is assumed that fluid motion can be ignored inside the cylinder.

- With particular reference to constant- volume fuel-air cycle, it is also assumed that

- The fuel is completely vaporized and perfectly mixed with the air, and
- The burning takes place instantaneously at top dead centre (at constant volume).

2.1.4 Importance of Fuel-Air cycle

- The air-standard cycle analysis shows the general effect of only compression ratio on engine efficiency whereas the fuel-air cycle analysis gives the effect of variation of fuel-air ratio, inlet pressure and temperature on the engine performance. It will be noticed that compression ratio and fuel-air ratio are very important parameters of the engine while inlet conditions are not so important.
- The actual efficiency of a good engine is about 85 per cent of the estimated fuel-air cycle efficiency. A good estimate of the power to be expected from the actual engine can be made from fuel-air cycle analysis. Also, peak pressures and exhaust temperatures which affect the engine structure and design can be estimated reasonably close to an actual engine. Thus the effect of many variables on the performance of an engine can be understood better by fuel-air cycle analysis.

2.2 Variable Specific Heats

- All gases, except mono-atomic gases, show an increase in specific heat with temperature. The increase in specific heat does not follow any particular law. However, over the temperature range generally encountered for gases in heat engines (300 K to 2000 K) the specific heat curve is nearly a straight line which may be approximately expressed in the form

$$C_p = a_1 + K_1 T$$
$$C_v = b_1 + K_1 T$$
$$(2.1)$$

where a_1, b_1 and K_1 are constants. Now,

$$R = C_p - C_v = a_1 - b_1 \qquad (2.2)$$

where R is the characteristic gas constant.

- Above 1500 K the specific heat increases much more rapidly and may be expressed in the form

$$C_p = a_1 + K_1 T + K_2 T^2 \qquad (2.3)$$
$$C_v = b_1 + K_1 T + K_2 T^2 \qquad (2.4)$$

- In above equations if the term T^2 is neglected it becomes same as Eqn.2.1. Many expressions are available even upto sixth order of T (i.e. T^6) for the calculation of C_p and C_v.
- The physical explanation for increase in specific heat is that as the temperature is raised, larger fractions of the heat would be required to produce motion of the atoms within the molecules. Since temperature is the result of motion of the molecules, as a whole, the energy which goes into moving the atoms does not contribute to proportional temperature rise. Hence, more heat is required to raise the temperature

of unit mass through one degree at higher levels. This heat by definition is the specific heat. The values for C_p and C_v for air are usually taken as

$$C_p = 1.005 \text{ kJ/kg K}, \quad C_v = 0.717 \text{ kJ/kg K} \quad \text{at 300 K}$$

$$C_p = 1.345 \text{ kJ/kg K}, \quad C_v = 1.057 \text{ kJ/kg K} \quad \text{at 2000 K}$$

−   Since the difference between C_p and C_v is constant, the value of γ decreases with increase in temperature. Thus, if the variation of specific heats is taken into account during the compression stroke, the final temperature and pressure would be lower than if constant values of specific heat are used. This point is illustrated in Fig.2.1.
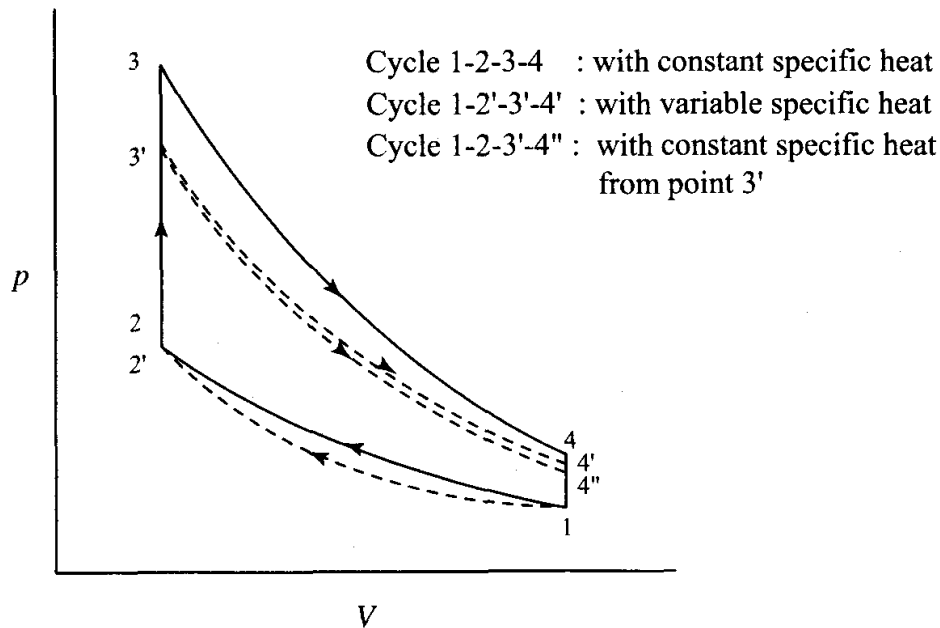

Cycle 1-2-3-4   : with constant specific heat
Cycle 1-2'-3'-4' : with variable specific heat
Cycle 1-2-3'-4" : with constant specific heat
                              from point 3'

Fig. 2.1 Loss of power due to variation of specific heat

−   With variable specific heats, the temperature at the end of compression will be 2', instead of 2. The magnitude of drop in temperature is proportional to the drop in the value of ratio of specific heats. For the process 1—>2, with constant specific heats

$$T_2 = T_1 \left(\frac{v_1}{v_2} \right)^{\gamma - 1} \tag{2.5}$$

with variable specific heats,

$$T_{2'} = T_1 \left(\frac{v_1}{v_{2'}} \right)^{k - 1} \tag{2.6}$$

where $k = \dfrac{C_p}{C_v}$. Note that $v_{2'} = v_2$ and $v_1/v_2 = v_1/v_{2'} = r$.

−   For given values of T_1, p_1 and r, the magnitude of T_2 depends on k. Constant volume combustion, from point 2 ' will give a temperature $T_{3'}$ instead of T_3. This is due to the fact that the rise in the value of C_v because of variable specific heat, which reduces the temperature as already explained.

−   The process, 2'-3' is heat addition with the variation in specific heat. From 3', if expansion takes place at constant specific heats, this would result in the process 3'-4"

whereas actual expansion due to variable specific heat will result in 3'-4' and 4' is higher than 4". The magnitude in the difference between 4' and 4 " is proportional to the reduction in the value of γ.

- Consider the process 3'-4"

$$T_{4"} = T_{3'} \left(\frac{v_3}{v_4} \right)^{k-1} \tag{2.7}$$

For the process 3'-4'

$$T_{4'} = T_{3'} \left(\frac{v_3}{v_4} \right)^{\gamma-1} \tag{2.8}$$

- Reduction in the value of k due to variable specific heat results in increase of temperature from $T_{4"}$ to $T_{4'}$.

2.3 Change of Internal energy and enthalpy during a process with variable specific heats

2.3.1 Change of Internal energy

- The small change in internal energy of a unit mass of a gas for small change in temperature (dT) is given by:

$$du = C_v \, dT$$

$$\therefore u_2 - u_1 = \int_{T_1}^{T_2} C_v \, dT$$

$$= \int_{T_1}^{T_2} (b + KT) \, dT$$

$$= \left[bT + K \frac{T^2}{2} \right]_{T_1}^{T_2} = b(T_2 - T_1) + \frac{K}{2}(T_2^2 - T_1^2)$$

$$= (T_2 - T_1) \left[b + K \frac{(T_2 + T_1)}{2} \right]$$

$$= (T_2 - T_1)(b + KT_m) \qquad \text{where, } T_m = \frac{T_1 + T_2}{2}$$

$$C_{vm} = b + KT_m \quad (C_{vm} \text{ mean specific heat at constant volume})$$

$$\therefore u_2 - u_1 = C_{vm}(T_2 - T_1) \tag{2.9}$$

2.3.2 Change of Enthalpy

- The small change in enthalpy of a unit mass of a gas for small change in temperature (dT) is given by:

$$dh = C_p \, dT$$

$$\therefore h_2 - h_1 = \int_{T_1}^{T_2} C_p \, dT$$

$$= \int_{T_1}^{T_2} (a + KT) \, dT$$

$$= \left[aT + K \frac{T^2}{2} \right]_{T_1}^{T_2} = a(T_2 - T_1) + \frac{K}{2}(T_2^2 - T_1^2)$$

$$= (T_2 - T_1) \left[a + K \frac{(T_2 + T_1)}{2} \right]$$

$$= (T_2 - T_1)(a + KT_m) \qquad \text{where, } T_m = \frac{T_1 + T_2}{2}$$

$$C_{pm} = a + KT_m \quad (C_{pm} \text{ mean specific heat at constant pressure})$$

$$\therefore h_2 - h_1 = C_{pm}(T_2 - T_1) \tag{2.10}$$

2.4 Isentropic expansion with variable specific heats

- Consider one kg of air, the heat transfer to a system using first law can be written as

$$dQ = du + dW$$

$$dQ = C_v \, dT + p \, dv$$

- For isentropic process, $dQ = 0$

$$\therefore C_v \, dT + p \, dv = 0$$

$$\therefore C_v \frac{dT}{T} + \frac{p}{T} \, dv = 0$$

$$\therefore C_v \frac{dT}{T} + R \frac{dv}{v} = 0 \qquad (\because pv = RT)$$

- Putting the values of R and C_v in the above equation, we get

$$\therefore (b + KT) \frac{dT}{T} + (a - b) \frac{dv}{v} = 0$$

- Integrating both sides we get

$$\therefore \quad \int (b + KT) \frac{dT}{T} + \int (a - b) \frac{dv}{v} = \text{constant}$$

$$\therefore \quad \int b \frac{dT}{T} + K \int dT + (a - b) \int \frac{dv}{v} = \text{constant}$$

$$\therefore \quad b \log_e T + KT + (a - b) \log_e v = \text{constant}$$

$$\therefore \quad \log_e T^b + \log_e e^{KT} + \log_e v^{(a-b)} = \text{constant}$$

$$\therefore \quad T^b e^{KT} v^{(a-b)} = \text{constant}$$

$$\therefore \quad T e^{\frac{K}{b}T} v^{(\frac{a}{b}-1)} = \text{constant} \tag{2.11}$$

$$\therefore \quad \frac{T}{v} e^{\frac{K}{b}T} v^{\frac{a}{b}} = \text{constant} \tag{2.12}$$

$$pv = RT \quad \Rightarrow \quad \frac{T}{v} = \frac{p}{R} = \frac{p}{a-b}$$

- Inserting the value of above equation in eq. 2.13.

$$\therefore \quad \frac{p}{a-b} e^{\frac{K}{b}T} v^{\frac{a}{b}} = \text{constant}$$

$$\therefore \quad pv^{\frac{a}{b}} e^{\frac{KT}{b}} = \text{constant} \tag{2.13}$$

2.5 Effect of variable specific heats on air standard efficiency of Otto and diesel cycle

2.5.1 Otto cycle

- The air standard efficiency of Otto cycle is given by

$$\eta = 1 - \frac{1}{r^{\gamma-1}}$$

$$Now, \; C_p - C_v = R$$

$$\therefore \quad \frac{C_p}{C_v} - 1 = \frac{R}{C_v}$$

$$\therefore \quad \gamma - 1 = \frac{R}{C_v} \qquad \left(\because \; \frac{C_p}{C_v} \right) \tag{2.14}$$

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

$$\therefore \quad 1 - \eta = (r)^{-\frac{R}{C_v}}$$

- Taking log on both sides, we have

$$\therefore \quad \log_e (1-\eta) = -\frac{R}{C_v} \log_e (r)$$

- Differentiating the above equation, we have

$$\therefore -\frac{1}{1-\eta} \frac{d\eta}{dC_v} = -R \log_e r \left(-\frac{1}{C_v^2} \right)$$

$$\therefore \frac{d\eta}{1-\eta} = -\frac{R}{C_v} \cdot \log_e r \cdot \frac{dC_v}{C_v}$$

$$\therefore \frac{d\eta}{\eta} = -\frac{1-\eta}{\eta} \cdot (\gamma - 1) \cdot \log_e r \cdot \frac{dC_v}{C_v} \tag{2.15}$$

- Negative sign indicates the decrease in efficiency with increase in C_v.
- The Eq. 2.15 gives the percentage variation in air standard efficiency of Otto cycle on account of percentage variation in C_v.

2.5.2 Diesel Cycle

- The air standard efficiency of diesel cycle is given by

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

$$\therefore 1 - \eta = \frac{1}{(r)^{\gamma-1}} \left[\frac{\rho^{\gamma} - 1}{\gamma(\rho - 1)} \right]$$

- Taking log on both sides, we get

$$\therefore \log(1 - \eta) = \log(\rho^{\gamma} - 1) - \log(r)^{\gamma-1} - \log \gamma - \log(\rho - 1)$$

$$\therefore \log(1 - \eta) = \log(\rho^{\gamma} - 1) - (\gamma - 1)\log r - \log \gamma - \log(\rho - 1)$$

- Differentiating the above equation with respect to γ

$$\therefore -\frac{1}{1 - \eta} \cdot \frac{d\eta}{d\gamma} = \frac{1}{\rho^{\gamma} - 1} \cdot \rho^{\gamma} \log_e \rho - \log_e r - \frac{1}{\gamma}$$

$$\therefore \frac{d\eta}{d\gamma} = (1 - \eta) \left[\log_e r - \frac{\rho^{\gamma} \log_e \rho}{\rho^{\gamma} - 1} \cdot + \frac{1}{\gamma} \right]$$

- Multiplying the above equation by $\dfrac{d\gamma}{\eta}$

$$\therefore \frac{d\eta}{\eta} = \left(\frac{1 - \eta}{\eta} \right) \left[\log_e r - \frac{\rho^{\gamma} \log_e \rho}{\rho^{\gamma} - 1} \cdot + \frac{1}{\gamma} \right] \cdot d\gamma \qquad (2.16)$$

- Eq. 2.14 is $\gamma - 1 = \dfrac{R}{C_v}$, differentiating this equation with respect to C_v

$$\therefore \frac{d\gamma}{dC_v} = -\frac{R}{C_v^2} \Rightarrow d\gamma = -\frac{R}{C_v} \cdot \frac{dC_v}{C_v}$$

$$d\gamma = -(\gamma - 1) \cdot \frac{dC_v}{C_v} \qquad (2.17)$$

- Inserting the value of Eq. 2.17 into Eq. 2.16, we get

$$\therefore \frac{d\eta}{\eta} = -\frac{1 - \eta}{\eta} \cdot (\gamma - 1) \left[\log_e r - \frac{\rho^{\gamma} \log_e \rho}{\rho^{\gamma} - 1} \cdot + \frac{1}{\gamma} \right] \cdot \frac{dC_v}{C_v} \qquad (2.18)$$

2.6 Dissociation

- Dissociation process can be considered as the disintegration of combustion products at high temperature.
- Dissociation can also be looked as the reverse process to combustion. During dissociation the heat is absorbed whereas during combustion the heat is liberated.
- In IC engines, mainly dissociation of CO_2 into CO and O_2 occurs, whereas there is a very little dissociation of H_2O.
- The dissociation of CO_2 into CO and O_2 starts commencing around 1000 °C and the reaction equation can be written as

$$2CO_2 + Heat \rightleftharpoons \qquad O_2$$
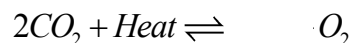
- Similarly, the dissociation of H_2O occurs at temperatures above 1300 °C and written as

$$2H_2O + Heat \rightleftharpoons \qquad O_2$$
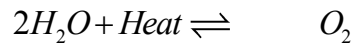
- The presence of CO and O_2 in the gases tends to prevent dissociation of CO_2; this is noticeable in a rich fuel mixture, which, by producing more CO, suppresses dissociation of CO_2.
- On the other hand, there is no dissociation in burnt gases of a lean fuel-air mixture. This is mainly due to the fact that temperature produced is too low for this phenomenon to occur.
- Hence, the maximum extent of dissociation occurs in the burnt gases of the chemically correct fuel-air mixture when the temperatures are expected to be high but decreases with the leaner and richer mixtures.
- In case of internal combustion engines heat transfer to the cooling medium causes a reduction in the maximum temperature and pressure. As the temperature falls during the expansion stroke the separated constituents recombine; the heat absorbed during dissociation is thus again released, but it is too late in the stroke to recover entirely the lost power. A portion of this heat is carried away by the exhaust gases.
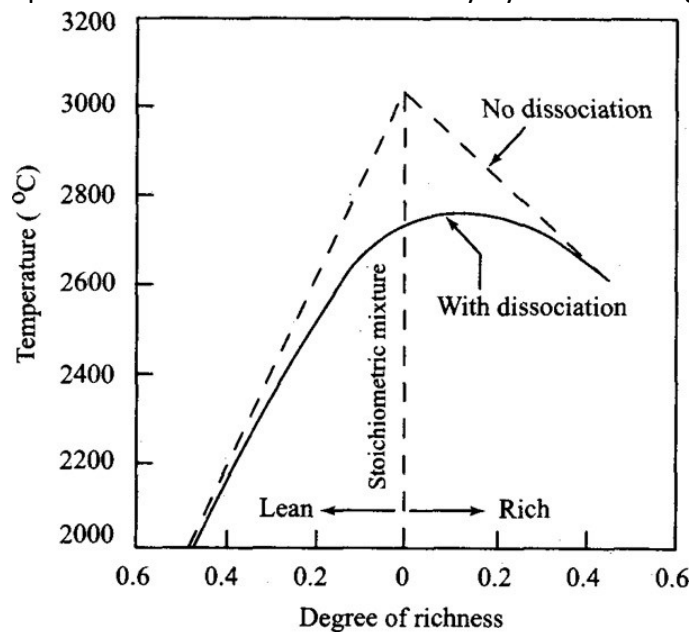

Fig. 2.2 Effect of dissociation on temperature

- Figure 2.2 shows a typical curve that indicates the reduction in the temperature of the exhaust gas mixtures due to dissociation with respect to air-fuel ratio. With no dissociation maximum temperature is attained at chemically correct air-fuel ratio.
- With dissociation maximum temperature is obtained when mixture is slightly rich. Dissociation reduces the maximum temperature by about 300 °C even at the chemically correct air-fuel ratio. In the Fig. 2.2, lean mixtures and rich mixtures are marked clearly.
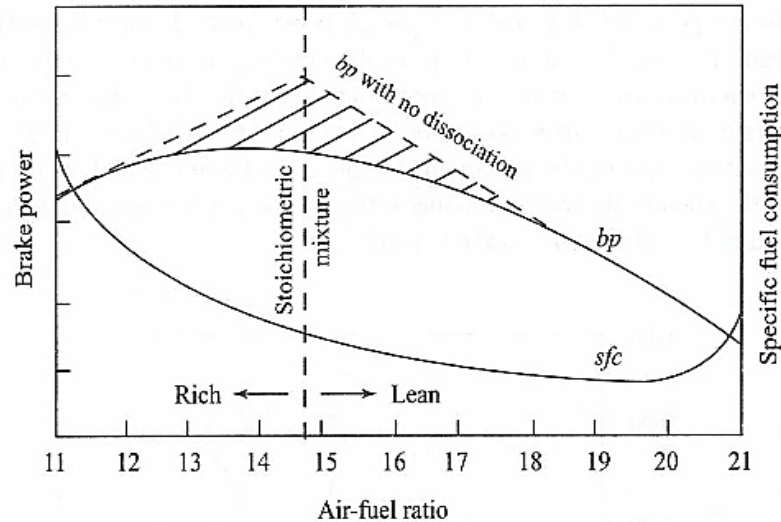
Fig. 2.3 Effect of dissociation on power

- The effect of dissociation on output power is shown in Fig.2.3 for a typical four-stroke spark-ignition engine operating at constant speed. If there is no dissociation the brake power output is maximum when the mixture ratio is stoichiometric.
- The shaded area between the brake power graphs shows the loss of power due to dissociation. When the mixture is quite lean there is no dissociation. As the air-fuel ratio decreases i.e., as the mixture becomes rich the maximum temperature raises and dissociation commences.
- The maximum dissociation occurs at chemically correct mixture strength. As the mixture becomes richer, dissociation effect tends to decline due to incomplete combus-tion.
- Dissociation effects are not so pronounced in a CI engine as in an SI engine. This is mainly due to

Fig. 2.4 Effect of dissosiation shown on a p-V diagram

  (i) the presence of a heterogeneous mixture and
  (ii) excess air to ensure complete combustion.
  Both these factors tend to reduce the peak gas temperature attained in the CI engine.
- Figure 2.4 shows the effect of dissociation on p-V diagram of Otto cycle. Because of lower maximum temperature due to dissociation the maximum pressure is also reduced and the state after combustion will be represented by 3' instead of 3. If there was no reassociation due to fall of temperature during expansion the expansion

process would be represented by 3'-4" but due to reassociation the expansion follows the path 3'-4'.

- By comparing with the ideal expansion 3-4, it is observed that the effect of dissociation is to lower the temperature and consequently the pressure at the beginning of the expansion stroke. This causes a loss of power and also efficiency. Though during recombining the heat is given back it is too late to contribute a convincing positive increase in the output of the engine.

2.7 Effect of operating variables

The effect of common engine operating variables on the pressure and temperature within the engine cylinder is better understood by fuel-air cycle analysis. The details are discussed in this section:

2.7.1 Compression Ratio

- The fuel-air cycle efficiency increases with the compression ratio in the same manner as the air-standard cycle efficiency, principally for the same reason (more scope of expansion work. This is shown in fig 2.5.
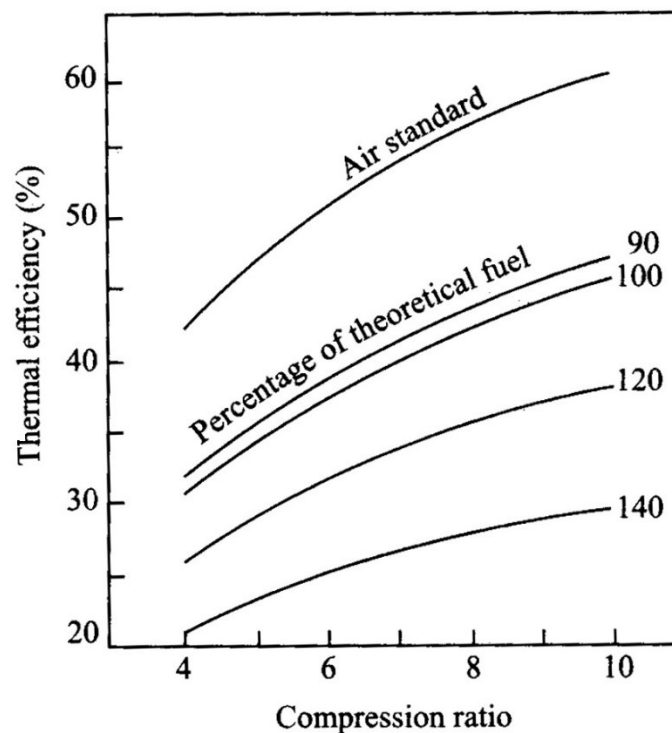

Fig. 2.5 Effect of compression ratio and mixture strength on efficiency

- The variation of indicated thermal efficiency with respect to the equivalence ratio for various compression ratios is given in fig 2.6. The equivalence ratio, ϕ, is defined as ratio of actual fuel-air ratio to chemically correct fuel-air ratio on mass basis.
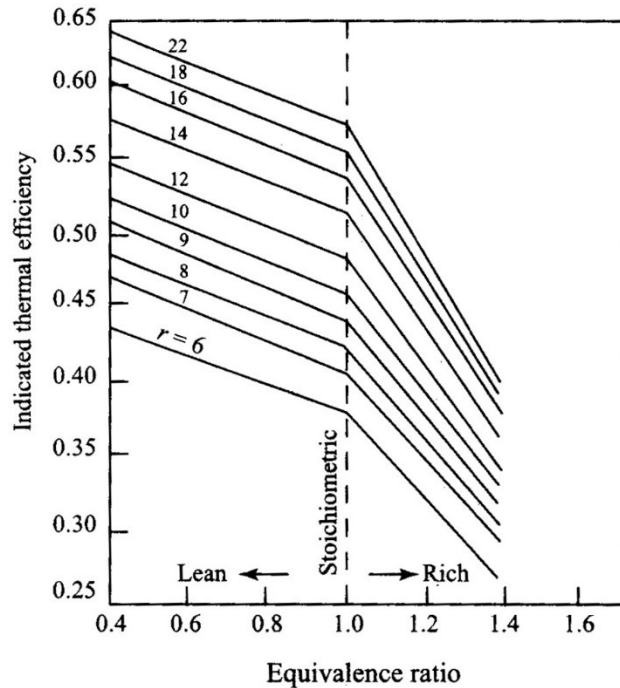
Fig. 2.6 Effect of mixture strength on thermal efficiency for various compression ratios

- The maximum pressure and temperature increase with compression ratio since the temperature, T_2, and pressure, p_2, at the end of compression are higher. However, it can be noted from the experimental results that the ratio of fuel-air cycle efficiency to air-standard efficiency is independent of the compression ratio for given equivalence ratio for the constant volume fuel-air cycle.

2.7.2 Fuel Air ratio

a) Efficiency

- As the mixture is made lean (less fuel) the temperature rise due to combustion will be lowered as a result of reduced energy input per unit mass of mixture. This will result in lower specific heat.
- Further, it will lower the losses due to dissociation and variation in specific heat. The efficiency is therefore, higher and, in fact, approaches the air-cycle efficiency as the fuel-air ratio is reduced as shown in Fig. 2.7.
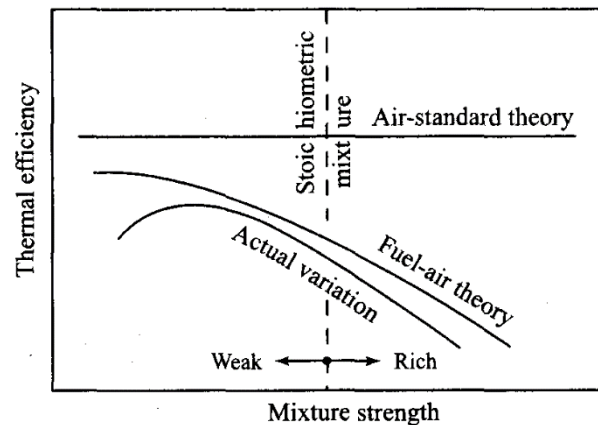

Fig. 2.7 Effect of mixture strength on thermal efficiency

b) Maximum Power

- Fig. 2.8 gives the cycle power as affected by fuel-air ratio. By air-standard theory maximum power is at chemically correct mixture, but by fuel-air theory maximum

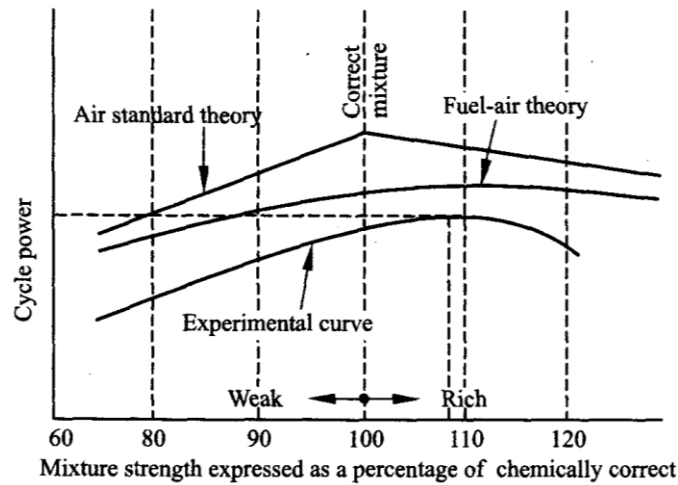power is when the mixture is about 10% rich. As the mixture becomes richer the efficiency falls rapidly.

Fig. 2.8 Effect of fuel-air ratio on power

- This is because in addition to higher specific heats and chemical equilibrium losses, there is insufficient air which will result in formation of CO and H_2 in combustibles, which represents a direct wastage of fuel.

c) **Maximum temperature**

- At a given compression ratio the temperature after combustion reaches a maximum when the mixture is slightly rich, i.e., around 6 % or so (F/A = 0.072 or A/F = 14:1) as shown in Fig. 2.9.
- At chemically correct ratio there is still some oxygen present at the point 3 because of chemical equilibrium effects a rich mixture will cause more fuel to combine with oxygen at that point thereby raising the temperature T_3. However, at richer mixtures increased formation of CO counters this effect.
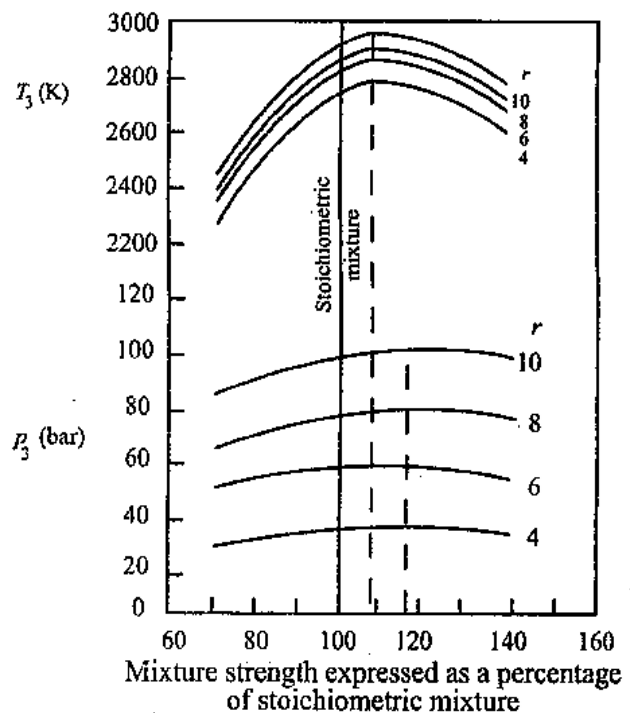
d) **Maximum Pressure**

- The pressure of a gas in a given space depends upon its temperature and

Fig. 2.9 Effect of equivalence ratio on T_3 and P_3

the number of molecules The curve of p_3, therefore follows T_3, but because of the increasing number of molecules p_3 does not start to decrease until the mixture is somewhat richer than that for maximum T_3 (at F/A = 0.083 or A/F = 12:1), i.e. about 20 per cent rich (Fig.2.9).

e) Exhaust Temperature

– The exhaust gas temperature, T_4 is maximum at the chemically correct mixture as shown in Fig. 2.10. At this point there is reassociation as the temperature decrease so heat will be released these heat cannot be used in engine cylinder so the exhaust gases carry these heat with them and it result in higher exhaust temperature.

– At lean mixtures, because of less fuel, T_3 is less and hence T_4 is less. At rich mixtures less sensible energy is developed and hence T_4 is less. That is, T_4 varies with fuel-air ratio in the same manner as T_3 except that maximum T_4 is at the chemically correct fuel-air ratio in place of slightly rich fuel-air ratio (6 %) as in case of T_3.
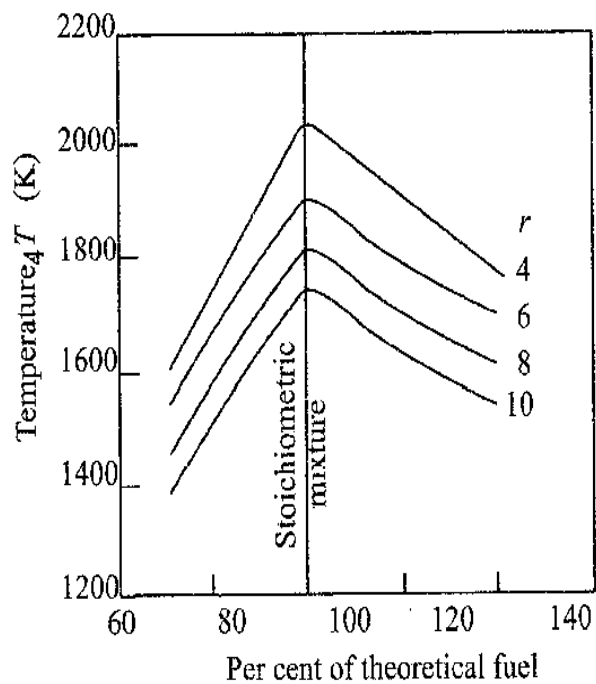

Fig. 2.10 Effect of fuel-air ratio on the exhaust gas temperature

– However, the behaviour of T_4 with compression ratio is different from that of T_3 as shown in Fig. 2.10 Unlike T_3, the exhaust gas temperature, T_4 is lower at high compression ratios, because the increased expansion causes the gas to do more work on the piston leaving less heat to be rejected at the end of the stroke. The same effect is present in the case of air-cycle analysis also.

2.8 Comparison of air standard and actual cycles

The actual cycles for internal combustion engines differ from air-standard cycles in many respects. These differences are mainly due to:

– The working substance being a mixture of air and fuel vapour or finely atomized liquid fuel in air combined with the products of combustion left from the previous cycle.

– The change in chemical composition of the working substance.

– The variation of specific heats with temperature.

– The change in the composition, temperature and actual amount of fresh charge because of the residual gases.

– The progressive combustion rather than the instantaneous combustion.

– The heat transfer to and from the working medium

– The substantial exhaust blowdown loss, i.e., loss of work on the expan-sion stroke due to early opening of the exhaust valve.

– Gas leakage, fluid friction etc., in actual engines.

Most of the factors listed above tend to decrease the thermal efficiency and power output of the actual engines. On the other hand, the analysis of the cycles while taking these factors into account clearly indicates that the estimated thermal efficiencies are not very different from those of the actual cycles.

2.9 Deviation of Actual cycle from Fuel-Air cycle

- Major deviation from of actual cycle from the Fuel air cycle is due to
  - Variation in Specific heats
  - Dissociation
  - Progressive combustion
  - Incomplete combustion of fuel
  - Time loss factor
  - Heat loss factor
  - Exhaust blowdown factor

2.9.1 Time losses

Time losses may be burning time loss and spark timings loss.

a) Burning time loss

- In theoretical cycle, the burning is assumed to be instantaneous but actually burning takes some time. The time required depends upon F:A ratio, fuel chemical structure and its ignition temperature. This also depends upon the flame velocity and the distance from the ignition point to the opposite side of combustion chamber.
- During combustion, there is always increase in volume. The time internal between the spark and complete burning of the charge is approximately 40° crank rotation.
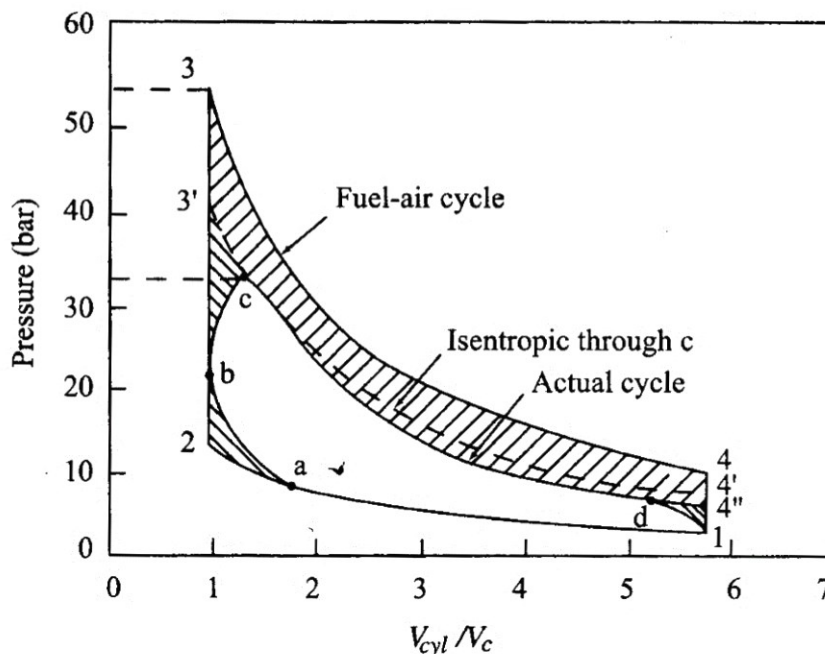

Fig. 2.11 Effect of time losses on p-V diagram

- The effect of time required for combustion; the maximum pressure is not produced when volume is minimum (v_c) as expected. It is produced some time after TDC. Therefore, the pressure rises from b to c as shown in Fig. 2.11.
- The point 3 represents the maximum pressure if the combustion should have taken place instantly. The difference in area of actual cycle and fuel-air cycle shows the loss of power as shown in Fig. 2.11. This loss of work is called burning time loss. This time loss is defined as the loss of power due to time required for mixing the fuel with air and for complete combustion.

b) Spark Timing Loss
- A definite time is required to start the burning of fuel after generating the spark in the cylinder. The effect of this, the maximum pressure is not reached at TDC and it reaches late during the expansion stroke. The time at which the burning starts is varied by varying the angle of advance (spark advance).
  - If the spark is given at T.D.C., the maximum pressure is low due to expansion of gases.
  - If the spark is advanced by 40° to start combustion at T.D.C., the combustion takes place at T.D.C. But the heat loss and the exhaust loss may be higher and again work obtained is not optimum.
- In the above two cases, the work area is less, and, therefore, power developed per cycle and efficiency are lower.
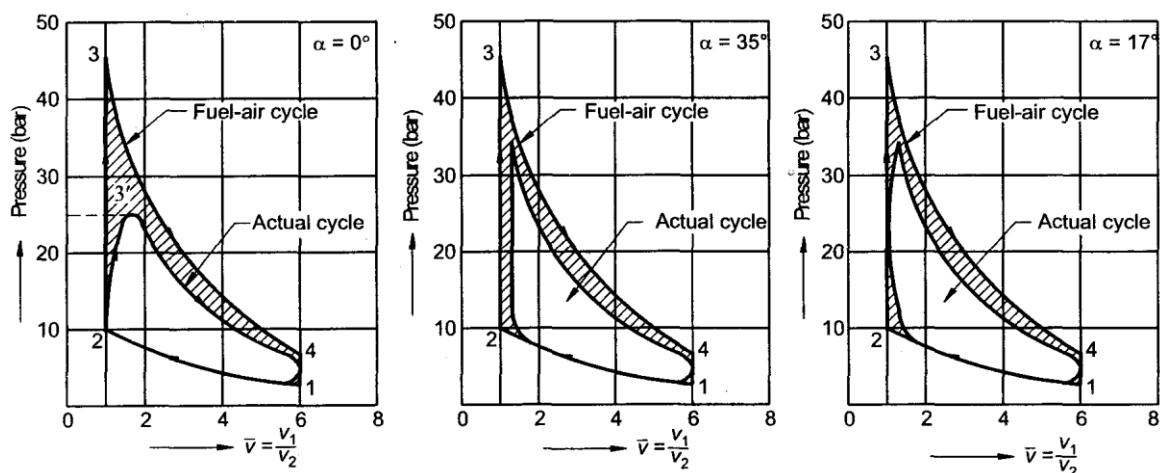

Fig. 2.12 Effects of angle of advance a on p - v diagram

- Thus for getting maximum work output, a moderate spark advance of 15° to 25° is the best.

c) Incomplete Combustion Loss
- The time loss always includes a loss due to incomplete combustion. It is impossible obtain perfect homogeneous air-fuel mixture. Fuel vapour, air, and residual gas is present in the cylinder before ignition takes place. Under these circumstances it is possible to have excess oxygen in one part of the cylinder and excess fuel in another part of it. Therefore, some fuel does not burn or burns partially. Both CO and O_2 will appear in the exhaust.

- It should be noted that it is necessary to use a lean mixture to eliminate fuel wastage while a rich mixture is required to utilize all the oxygen. Slightly leaner mixture will give maximum efficiency but too lean a mixture will burn slowly, increasing the losses or will not burn at all causing total waste. In the rich mixture some of the fuel will not get oxygen and will be completely wasted. Also, the flame speed in the rich mixture is low, thereby increasing the time losses and lowering the efficiency.

2.9.2 Direct heat loss

- During the combustion process and expansion process, the gases inside the engine cylinder are at a considerably higher temperature, so the heat is lost to the jacket cooling water or air. Some heat is lost to the lubricating oil where splash lubrication system is used for lubricating cylinder and piston.
- The loss of heat which takes place during combustion has the maximum effect, while that lost before the end of the expansion stroke has little effect, since it can do very small amount of useful work.
- During combustion and expansion, about 15% of the total heat is lost. Out of this, however, much is lost too late in the cycle to have done any useful work.
- In case all heat loss is recovered, about 20 percent of it may appear as useful work.

2.9.3 Exhaust blowdown loss

- At the end of exhaust stroke, the cylinder pressure is about 7 bar. If the exhaust valve is opened at B.D.C., the piston has to do work against high cylinder pressure costing part of the exhaust stroke. When the exhaust valve is opened too early entire part of the expansion stroke is lost.
- Thus, best compromise is that exhaust valve be opened 40° to 70° before B.D.C., thereby, reducing the cylinder pressure to halfway to atmosphere before the start of the exhaust stroke.
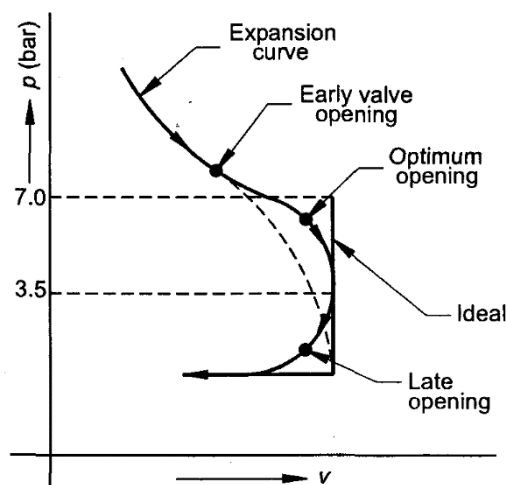

Fig. 2.13 Effect of blow down

2.9.4 Pumping losses

- In case of ideal cycles the suction and exhaust processes were assumed to be at atmospheric pressure. However some pressure differential is required to carry out the suction and exhaust processes between the fluid pressure and cylinder pressures.

Fig. 2.14 Pumping Loss

- During suction the cylinder pressure is lower them the fluid pressure in order to induct the fluid into the cylinder and the exhaust gases are expelled at a pressure higher than the atmospheric pressure.
- Therefore some work is done on the gases during suction and exhaust stroke. This work is called pumping work as shown in Fig. 2.14 by shaded area.
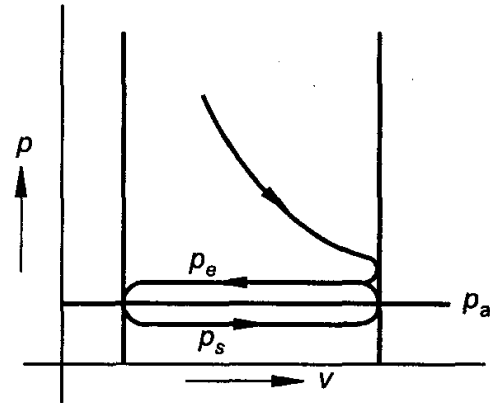
2.9.5 Rubbing Friction Losses

- The rubbing friction losses are caused due to
    - Friction between piston and cylinder walls
    - Friction in various bearings
    - Friction in auxiliary equipment such as pumps and fans.
- The piston friction increases rapidly with engine speed and to small extent by in-creases in m.e.p.
- The bearing and auxiliary friction also increase with engine speed.
- The engine efficiency is maximum at full load and reduces with the decrease in load. It is due to the fact that direct heat loss, pumping loss and rubbing friction loss increase at lower loads.

2.10 Valve and port timing diagrams

- The valve timing diagram shows the position of the crank when the various operations i.e., suction, compression, expansion exhaust begin and end.
- The valve timing is the regulation of the positions in the cycle at which the valves are set to at open and close
- The poppet valves of the reciprocating engines are opened and closed by cam mechanisms. The clearance between cam, tappet and valve must be slowly taken up and valve slowly lifted, at first, if noise and wear is to be avoided. For the same reasons the valve cannot be closed abruptly, else it will bounce on its seat. (Also, the cam contours should be so designed as to produce gradual and smooth changes in directional acceleration).
- Thus, the valve opening and closing periods are spread over a considerable number of crankshaft degrees. As a result, the opening of the valve must commence ahead of the time at which it is fully opened (i.e. before dead centres). The same reasoning applies for the closing time and the valves must close after the dead centres.

2.10.1 Valve timing diagram of 4-Stroke Petrol engine

 – The actual valve timings used for low speed and high speed engines are shown in Fig. 2.15 (a) and (b).

a) Inlet valve

 – The inlet valve opening occurs a few degrees prior to the arrival of the piston at TDC during the exhaust stroke. This is necessary to insure that the valve will be fully open and fresh charge starts to flow into the cylinder as soon as the piston starts to move down.

 – If the inlet valve is allowed to close at BDC, the cylinder would receive less charge than its capacity and the pressure of the charge at the end of suction stroke will be below atmosphere. To avoid this, the inlet valve is kept open for 40°-50° rotation of the crank after the suction stroke for high speed engine and 20° to 25° for low speed engine.

 – The kinetic energy of the charge produces a ram effect which packs more charge into the cylinder during this additional valve opening. Therefore, the inlet valve closing is delayed.

 – Higher the speed of the engine, the inlet valve closing is delayed longer to take an advantage of ram effect.
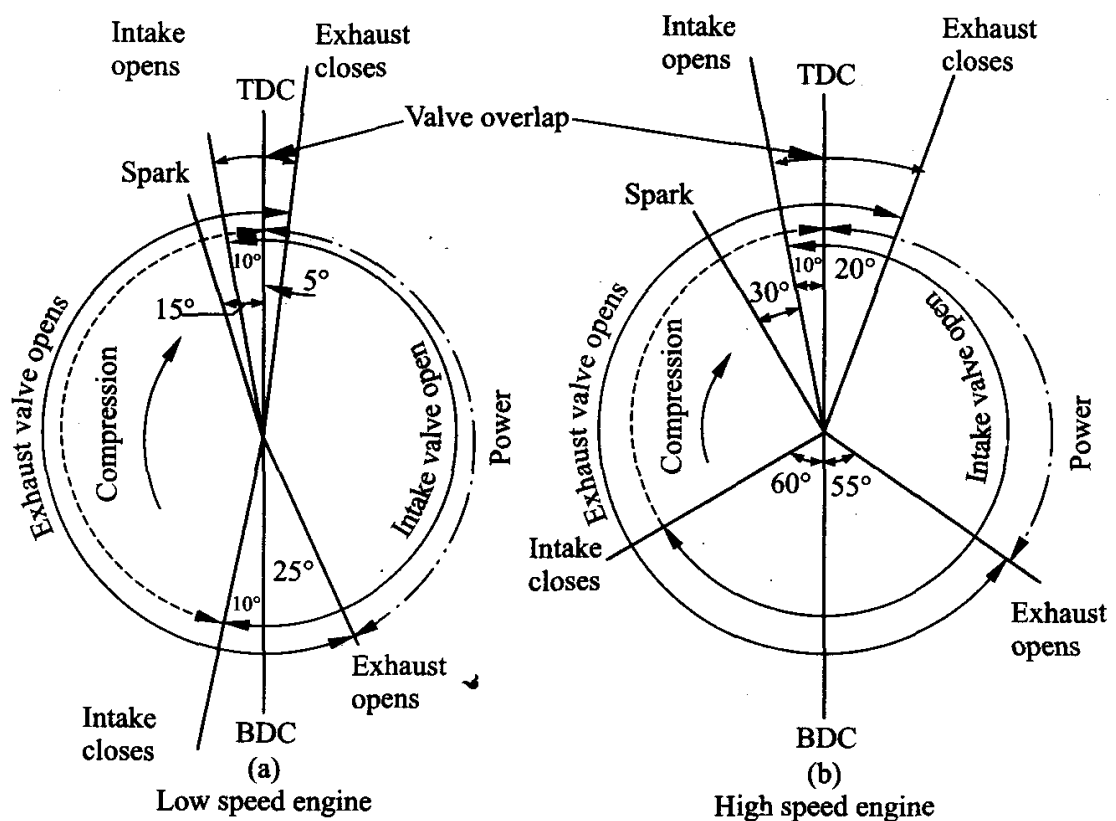

Fig. 2.15 Valve timing diagram for low and high speed 4-stroke SI engine

b) Exhaust valve

 – The exhaust valve is set to open before BDC (say about 25° before BDC in low speed engines and 55° before BDC in high speed engines).

- If the exhaust valve did not start to open until BDC, the pressures in the cylinder would be considerably above atmospheric pressure during the first portion of the exhaust stroke, increasing the work required to expel the exhaust gases. But opening the exhaust valve earlier reduces the pressure near the end of the power stroke and thus causes some loss of useful work on this stroke.
- However, the overall effect of opening the valve prior to the time the piston reaches BDC results in overall gain in output.
- The closing time of exhaust valve effects the volumetric efficiency. By closing the exhaust valve a few degrees after TDC (about 15° in case of low speed engines and 20° in case of high speed engines) the inertia of the exhaust gases tends to scavenge the cylinder by carrying out a greater mass of the gas left in the clearance volume. This results in increased volumetric efficiency.

c) Ignition
- Theoretically it is assumed that spark is given at the TDC and fuel burns instantaneously. However, there is always a time lag between the spark and ignition of the charge. The ignition starts some time after giving the spark, therefore it is necessary to produce the spark before piston reaches the TDC to obtain proper combustion without losses. The angle through which the spark is given earlier is known as "**Ignition Advance**" or "**Angle of Advance**".

d) Valve Overlap
- From the valve timing diagram it is obvious that there will a period when both the intake and exhaust valves are open at the same time. This is called **valve overlap** (say about 15° in low speed engine and 30° in high speed engines). This overlap should not be excessive otherwise it will allow the burned gases to be sucked into the intake manifold, or the fresh charge to escape through the exhaust valve.

2.10.2 Valve timing diagram of 4-Stroke Diesel engine
- The actual valve timing diagram of 4-Stroke Diesel cycle engine is shown in fig. 2.16. The various strokes are modified for similar reasons as explained in case of petrol engine.

Fuel Injection Timing
- The opening of fuel valve is necessary for better evaporation and mixing of the fuel. As there is always lag between ignition and supply of fuel, it is always necessary to supply the fuel little earlier.
- In case of diesel engine, the overlapping provided is sufficiently large compared with the petrol engine. More overlapping is not advisable in petrol engine because the mixture of air and petrol may pass out with the exhaust gases and it is highly uneconomical. This danger does not arise in case of diesel engine because only air is taken during the suction stroke.
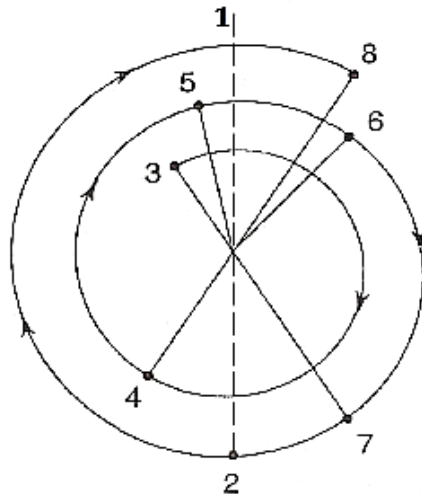
1 – TDC
2 – BDC
3 – Inlet Valve Opens
4 – Inlet Valve Closes
5 – Fuel Valve Opens
6 – Fuel Valve Closes
7 – Exhaust Valve Opens
8 – Exhaust Valve Closes

2.16 Valve Timing Diagram of 4-Stroke Diesel Cycle Engine

– The valve timing of diesel engine have to be adjusted depending upon the speed of the engine. The typical valve timings are as follows:
  – IV opens at 25^0 before TDC
  – IV closes at 30^0 after BDC
  – Fuel injection starts at 5^0 before TDC
  – Fuel injection closes at 25^0 after TDC
  – EV opens at 45^0 before BDC
  – EV closes at 15^0 after TDC

2.10.3 Port Timing Diagram of 2-stroke engine

– The port timing diagram for actual working of the two-stroke petrol and diesel engine is shown in Fig. 2.17. The port timing diagram is self-explanatory.
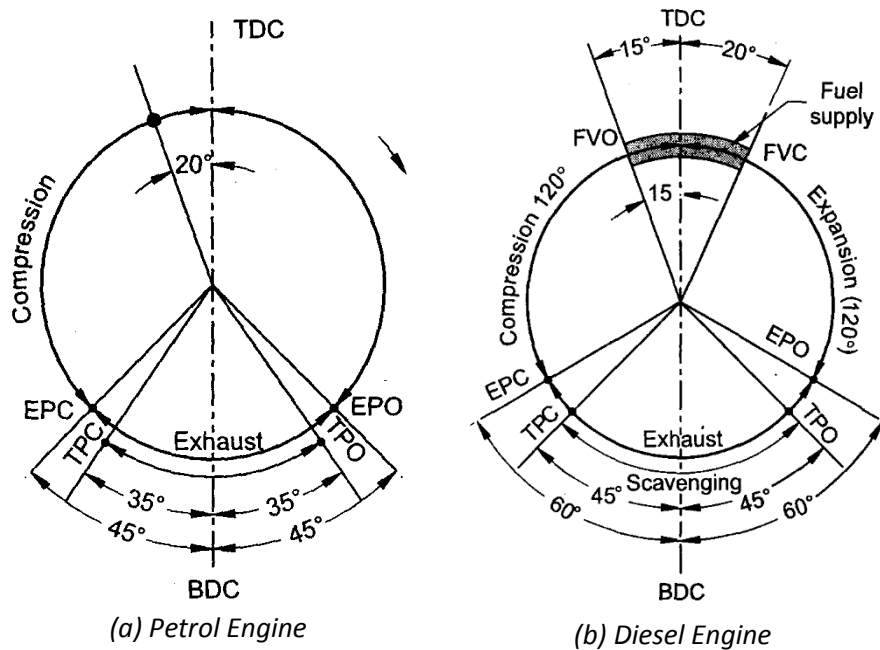

(a) Petrol Engine

(b) Diesel Engine

Fig. 2.17 Port Timing Diagram for 2-stroke Engine

Tutorial Questions

1	Give classification of IC Engines.
2	Distinguish between SI engines and CI engines?
3	Sketch and explain the valve timing diagram of a four stroke Otto cycle?
4	In what respect two stroke engines differs from 4-stroke engine Discuss?
5	Explain fuel injection system of an SI engine?
6	What are the different lubrication systems available for IC engines?
7	Discuss the importance of cooling system for an IC engines. Describe different cooling systems?
8	List out the properties of fuel for (i) SI engine (ii) CI engine.
9	Explain lubrication system for IC engines?
10	Explain cooling system for IC engines?

Assignment Questions

1	what is scavenging ? explain with sketches?
2	List the factors causes detonation and explain in detail?
3	Explain Magneto ignition system with a neat diagram?
4	Explain coil ignition system with a neat diagram?
5	What is Octane number? What is the role of Octane number in the performance of engine? For higher performance of engine which rated fuels are to be selected?