where $p_x/p_1$ is the pressure ratio in each stage. In terms of overall pressure ratio, it becomes

$$W_c = \frac{2nRT_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{2n}} - 1\right] \qquad (19.21)$$

Heat rejected in the intercooler,

$$Q_{bc} = c_p\,(T_x - T_1)\ \text{kJ/kg}$$

Let us now consider compression efficiencies and imperfect intercooling. As shown in Fig. 19.11, an ideal gas is compressed from the initial state $p_1$, $T_1$ to $p_x$. It is then cooled at constant pressure to $T_y$ and then compressed from $p_x$, $T_y$ to $p_2$. Given $p_1$, $T_1$, $T_y$ and $p_2$, it is desired to find the value of $p_x$ which gives minimum work. Let the adiabatic compression efficiencies of the two stages be respectively $\eta_{C_1}$ and $\eta_{C_2}$; then the work of compression is

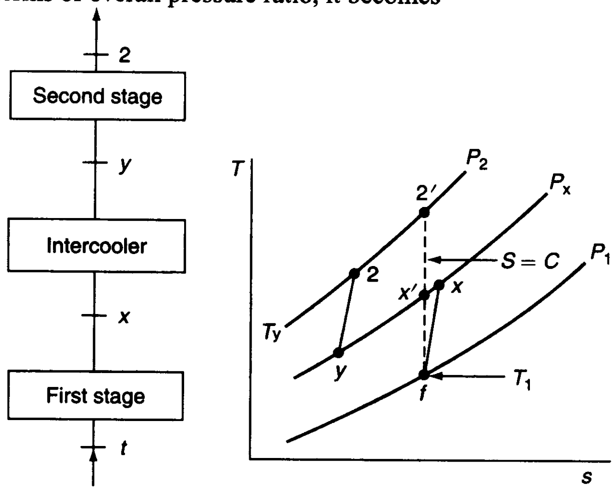



Fig. 19.11 T-s plot of two-stage compression process

$$W_c = W_1 + W_2 = \frac{1}{\eta_{C_1}}\frac{\gamma}{\gamma-1}RT_1\left[\left(\frac{p_x}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + \frac{1}{\eta_{C_2}}\frac{\gamma}{\gamma-1}RT_y\left[\left(\frac{p_2}{p_y}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

But $$\left(\frac{p_x}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_x'}{T_1} \quad\text{and}\quad \left(\frac{p_2}{p_y}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_2}{p_x}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2'}{T_x'}$$

$$W_c = \frac{\gamma R}{\gamma-1}\left[\frac{T_1}{\eta_{C_1}}\left(\frac{T_x'}{T_1} - 1\right) + \frac{T_y}{\eta_{C_2}}\left(\frac{T_2'}{T_x'} - 1\right)\right]$$

Taking the derivative with respect to $T_x'$ and setting it equal to zero (noting that $T_1$, $T_2'$, and $T_y$ are constant),

$$\frac{dW_c}{dT_x'} = 0 = \frac{\gamma R}{\gamma-1}\left[\frac{1}{\eta_{C_1}} + \frac{T_yT_2'}{\eta_{C_2}}\left(-\frac{1}{(T_x')^2}\right)\right]$$

Then, $$(T_x')^2 = \frac{\eta_{C_1}}{\eta_{C_2}}T_y\,T_2' \quad\text{and}\quad \frac{T_x'}{T_1} = \sqrt{\frac{\eta_{C_1}}{\eta_{C_2}}\cdot\frac{T_y}{T_1}\cdot\frac{T_2'}{T_1}}$$

for minimum work. Now $\dfrac{T_x'}{T_1} = \left(\dfrac{p_x}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$ and $\dfrac{T_2'}{T_1} = \left(\dfrac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$

Therefore, for minimum work in two-stage compression of an ideal gas with intercooling to a fixed temperature $T_y$,

$$\frac{p_x}{p_1} = \sqrt{\left(\frac{\eta_{C_1}}{\eta_{C_2}}\frac{T_y}{T_1}\right)^{\frac{\gamma}{\gamma-1}}\cdot\frac{p_2}{p_1}} \qquad (19.22)$$

For the special case of $T_y = T_1$ and $\eta_{C_1} = \eta_{C_2}$, which is often taken as a standard of comparison, the requirement for minimum work is

$$\frac{p_x}{p_1} = \sqrt{\frac{p_2}{p_1}} \tag{19.23}$$

as obtained earlier in Eq. (19.18).

Also for this special case the condition of minimum work is the condition of equal work in the two stages.

When, three stages of equal efficiencies are used, with intercooling to the initial temperature at two points as shown in Fig. 19.12, the condition of minimum work, and of equal division of work among stages is

$$\frac{p_{x1}}{p_1} = \frac{p_{x2}}{p_{x1}} = \frac{p_2}{p_{x2}} = \left(\frac{p_2}{p_1}\right)^{1/3} \tag{19.24}$$

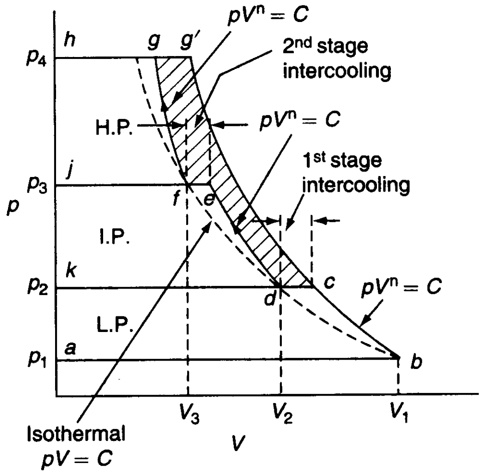



Fig. 19.12 Three-stage compression with perfect intercooling

Thus for 3-stage compressor the optimum pressure ratio per stage can be written as

$$\frac{p_x}{p_1} = \left(\frac{p_d}{p_s}\right)^{\frac{1}{3}}$$

where $p_d$ and $p_x$ are the discharge and suction pressure respectively.

For N-stage compressor, the optimum pressure ratio per stage is

$$\frac{p_x}{p_1} = \left(\frac{p_d}{p_s}\right)^{\frac{1}{N}} \tag{19.25}$$

The minimum work of compression for N-stages is then

$$W_c = \frac{NnRT_1}{n-1}\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{Nn}} - 1\right] \tag{19.26}$$

In the case of gas compression, the desirable idealized process is, however, a reversible isothermal process, and the isothermal efficiency is given by

$$\eta_t = \frac{W_t}{W_c} = \frac{p_1 v_1 ln \frac{p_2}{p_1}}{W_c} \tag{19.27}$$

The indicator diagram for a three-stage compressor with perfect intercooling is shown is Fig. 19.12. The air delivered from the first stage at $c$ is cooled to point $d$ on the curve $pV = C$ (show dotted) before admission to the second stage cylinder. Again the air is cooled from $e$ to $f$ after compression in the second stage and before admission to the third stage cylinder. The shaded area represents the saving of work resulting from such intercooling since without intercooling the compression curve would have followed the path $bcg'$.

Advantages of multi-stage compression are:

(i) Improved overall volumetric efficiency. If all compression were done in one cylinder the gas in the clearance volume would expand to a large volume before the new intake could begin. This results in a very low volumetric efficiency. By cooling the gas between the stages a much higher efficiency can be obtained.
(ii) A reduction in work required per stroke, and therefore the total driving power.
(iii) Size and strength of cylinders can be adjusted to suit volume and pressure of gas.
(iv) Multi-cylinders give more uniform torque and better mechanical balance thus needing smaller flywheel.
(v) Since the maximum temperature reached during the compression process is greatly reduced by intercooling lubrication difficulties and explosion hazards are lessened.
(vi) Leakage losses are reduced considerably.

Practice appears to indicate that the economical value of pressure ratio per stage is in the range of 3 to 5. For compression of atmospheric air, single-stage machines are often used up to 550 kPa discharge pressure, two-stage from 350 kPa to 2.1. MPa, three stage from 2.1 MPa to 7.0 MPa, and four or more stages for higher pressures.

## 19.6 AIR MOTORS

Compressed air is used in a wide variety of applications where the use of electric motors or S.I. engines is not permissible and where high safety is required to be met as in mining applications. Pneumatic breakers, picks, spades, rammers, vibrators, riveters, etc. form a range of hand tools having wide use in construction work.

Basically the cycle in the reciprocating expander is the reverse of that in the reciprocating compressor (Fig.19.13). Air is supplied from an air receiver (process 5-1) in which air is at approximately ambient air temperature. The state 1 is the point of cut-off. Air expands in the cylinder according to the law $pv^n$ = constant (where $n \cong 1.3$) pushing the piston out and doing work against the surroundings till the point of release at 2. There is blowdown of air from 2 to 3 and air is exhausted from 3 to 4. The compression of trapped or cushion air begins at 4 and ends a 5 according to $pV^n$ = constant and the compressed air is again admitted to repeat the cycle.

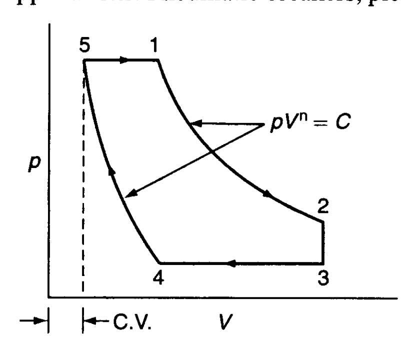



Fig. 19.13 p-V diagram for an air motor

## 19.7 ROTARY COMPRESSOR

Rotary compressors are used where large quantities of air or gas are needed at releatively low pressure. They may be classified into two main types.

(a) **Displacement compressors** in which air is compressed by being trapped in the reducing space formed by two sets of engaging surfaces.
(b) **Steady-flow compressors** in which compression occurs by the transfer of kinetic energy from a rotor.

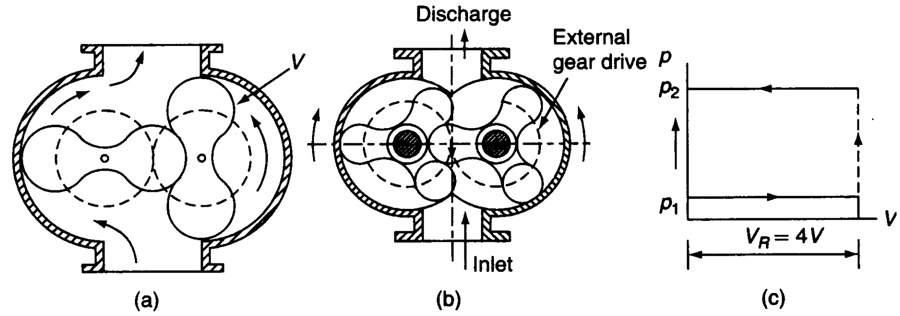
Rotary positive displacement machines are generally uncooled and the compression is largely adiabatic. There are two types: Roots blower and vane-type compressor.

**Roots blower** is an extension of the idea of a gear pump, popular in engines for pumping oil. There are two lobes on each rotor, and their shape is of cycloidal or involute form (Fig.19.14). One of the lobes is connected to the drive and the second is gear driven from the first, the two rotating in opposite directions. Very small clearances between the lobes and between the casing and the lobes are provided to prevent leakages and to reduce wear. Four times the volume between the casing and one side of the rotor will be displaced in each revolution of the driving shaft. As each side of each lobe faces its side of the casing a volume of gas $V$, at pressure $p_1$ is displaced towards the delivery side at constant pressure. A further rotation of the rotor opens this volume to the receiver, and the gas flows back from the receiver, since this gas is at a higher pressure. The gas induced is compressed irreversibly by that from the receiver to the pressure $p_2$, and then delivery begins. This process is carried out four times per revolution of the driving shaft.

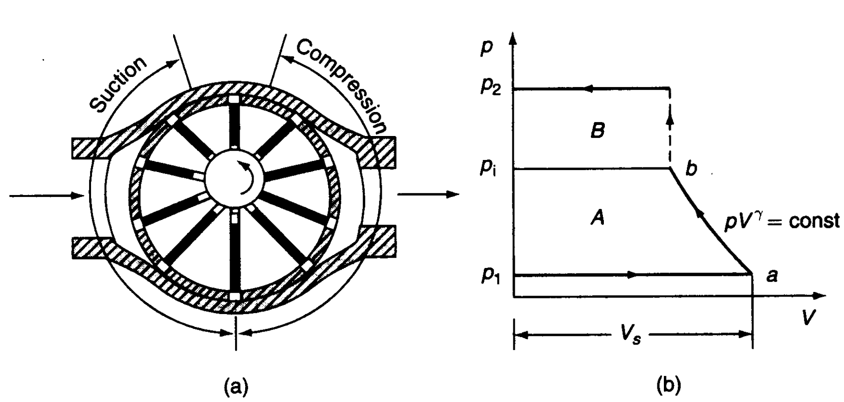
The $p$-$V$ diagram for this machine is shown in Fig. 19.14 (c), in which the pressure rises irreversibly from $p_1$ to $p_2$ at constant volume.

For pressure ratios of 1.2,1.6 and 2.0, the Roots blower efficiency becomes 0.945,0.84 and 0.765 respectively, which show that the efficiency decreases as the pressure ratio increases.

*Vane type compressor* (Fig. 19.15) consists of a rotor mounted eccentrically in the body and supported by ball-and-roller bearings at the ends. The rotor is slotted to take the blades which are of a non-metallic material, usually fibre or carbon. As each blade moves past the inlet passage, compression begins due to decreasing volume between the rotor and casing. Delivery begins with the arrival of each blade at the delivery passage. The $p$-$V$ diagram is shown in Fig. 19.15 (b) where $V_s$ is the induced volume at state $p_1$, $T_1$. Compression occurs to the pressure $p_i$ isentropically. At this pressure the displaced gas is opened to the receiver and the gas flowing back from the receiver raises the pressure irreversibly to $p_2$. The work input is given by the sum of the areas $A$ and $B$. Comparing the areas of Figs19.14 (c) and 19.15 (b) it is seen that for a given air flow and given pressure ratio the vane type requires less work input than the Roots blower.




*Roots blower with (a) two lobe rotor, (b) three lobe rotor, and (c) P-V diagram*




*Vane-type positive displacement compressor with p-v diagram*

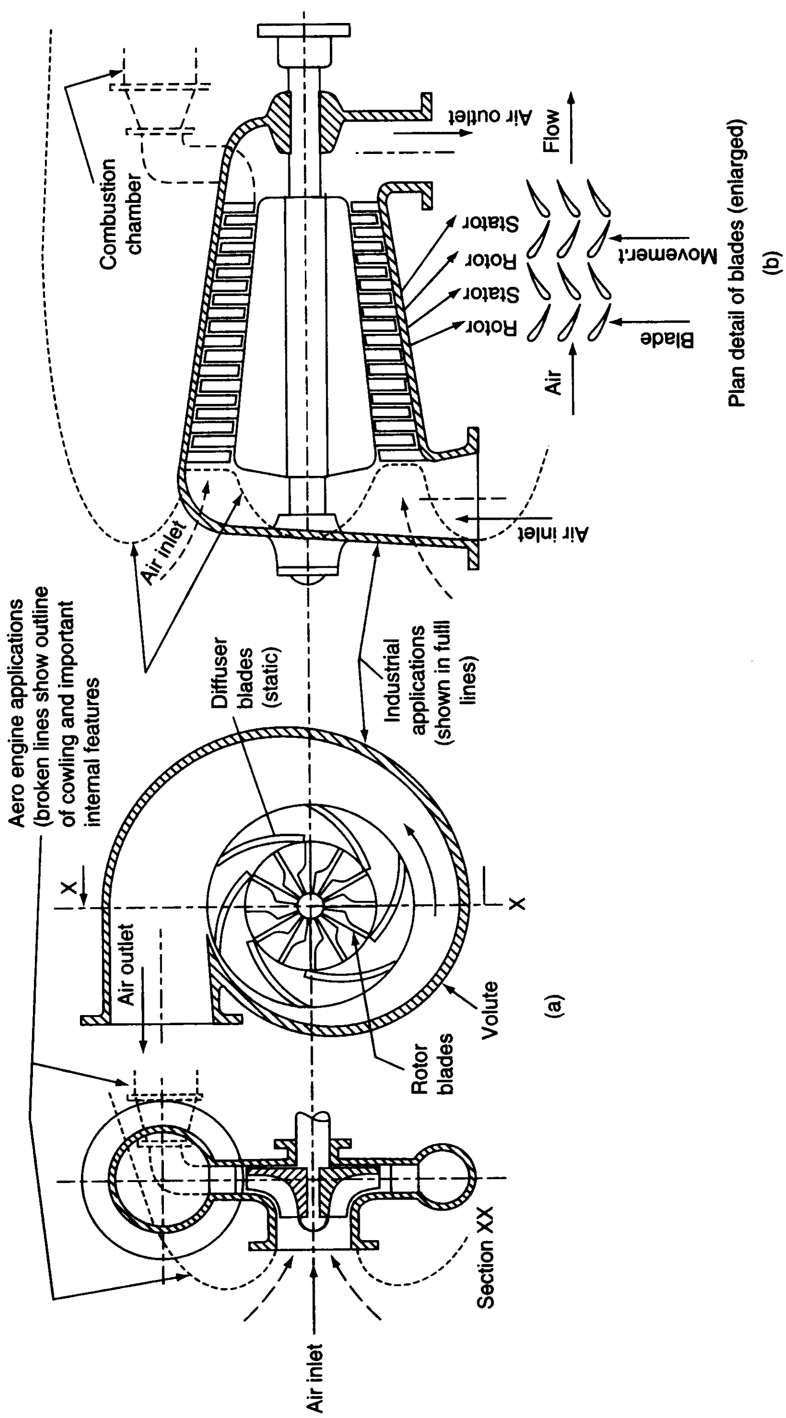



Fig. 18.16 (a) Centrifugal compressor, (b) axial-flow compressor

**Steady flow compressors** are of two types: centrifugal and axial flow, the diagrammatic sketches of which are shown in Fig. 19.16. In the **centrifugal compressor**, rotation of the impeller causes the air to be compressed by centrifugal action into the reduced volume at the tips of the compressor. A diffuser is fitted in which a part of the K.E. (Velocity) of the air is converted into internal energy so that there is a rise of pressure in both the impeller and the diffuser. The typical pressure ratio is about 1.4 to 1, but multistage arrangements can provide delivery pressures upto 10 atm.

In the **axial-flow compressor** the blades are arranged in much the same way as in a reaction turbine and the flow of air is along the axis of the compressor. The velocity of the air is changed during its passage through the sets of blades, and pressure lift occurs both in the moving blade passages and in the fixed blade passages which act as a diffuser. One set of moving and fixed blades represent a stage and a typical pressure ratio per stage is about 1.2. Multi-stage arrangements achieve higher delivery pressures with high efficiency.

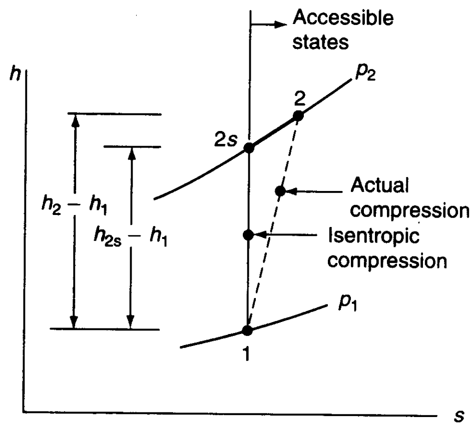



Fig. 19.17 Comparison of actual and isentropic compressions

For uncooled rotary compressors having a necessarily high-speed action, the compression process is regarded as adiabatic while the ideal process is isentropic (reversible and adiabatic) (Fig.19.17). In an irreversible adiabatic process the extra work which flows into the system to overcome friction increases the temperature and hence enthalpy of the gas. From S.F.E.E. neglecting P.E. and K.E. terms, the work input is

$$W_c = h_2 - h_1 = c_p (T_2 - T_1) \tag{19.28}$$

where $T_2$ is the final temperature for the actual compression and the isentropic efficiency $\eta_s$ is given by

$$\eta_s = \frac{W_s}{W_c} = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{19.29}$$

### 19.7.1 Centrifugal Compressor

The centrifugal compressor consists of an impeller with a series of curved radial vanes (Fig. 19.16a) housed in a stationary casing. The impeller imparts a high velocity to the air which flows through fixed divergent passages, in which air is decelerated with a consequent increase in static pressure. The main components of the compressor are (i) inlet pipe, (ii) impeller, (iii) diffuser (vaneless or with vanes), (iv) volute casing, and (v) outlet pipe. The air is sucked in near the hub called the eye through the inlet pipe in the axial direction and is turned through an angle in the impeller. The impeller rotates at a very high speed (15000 to 30000 rpm) and imparts the air a very high velocity. The static pressure of air increases from the eye to the tip of the impeller. Air leaving the impeller tip flows through diffuser passages (scroll) which convert the kinetic energy to pressure energy. The delivery pressure developed in a single stage compressor is of the order of 10 to 150 kPa gauge. Multi-stage compressors can develop discharge pressures as high as 1000 kPa.

The compressor may have single inlet or double inlet (Fig. 19.18). In a double inlet impeller having an eye on either side, air is drawn in on both sides (Fig. 19.18b). The impeller is subjected to approximately equal forces in the axial direction. About half the pressure rise occurs in the impeller vanes and the other half occurs in the diffuser passages.

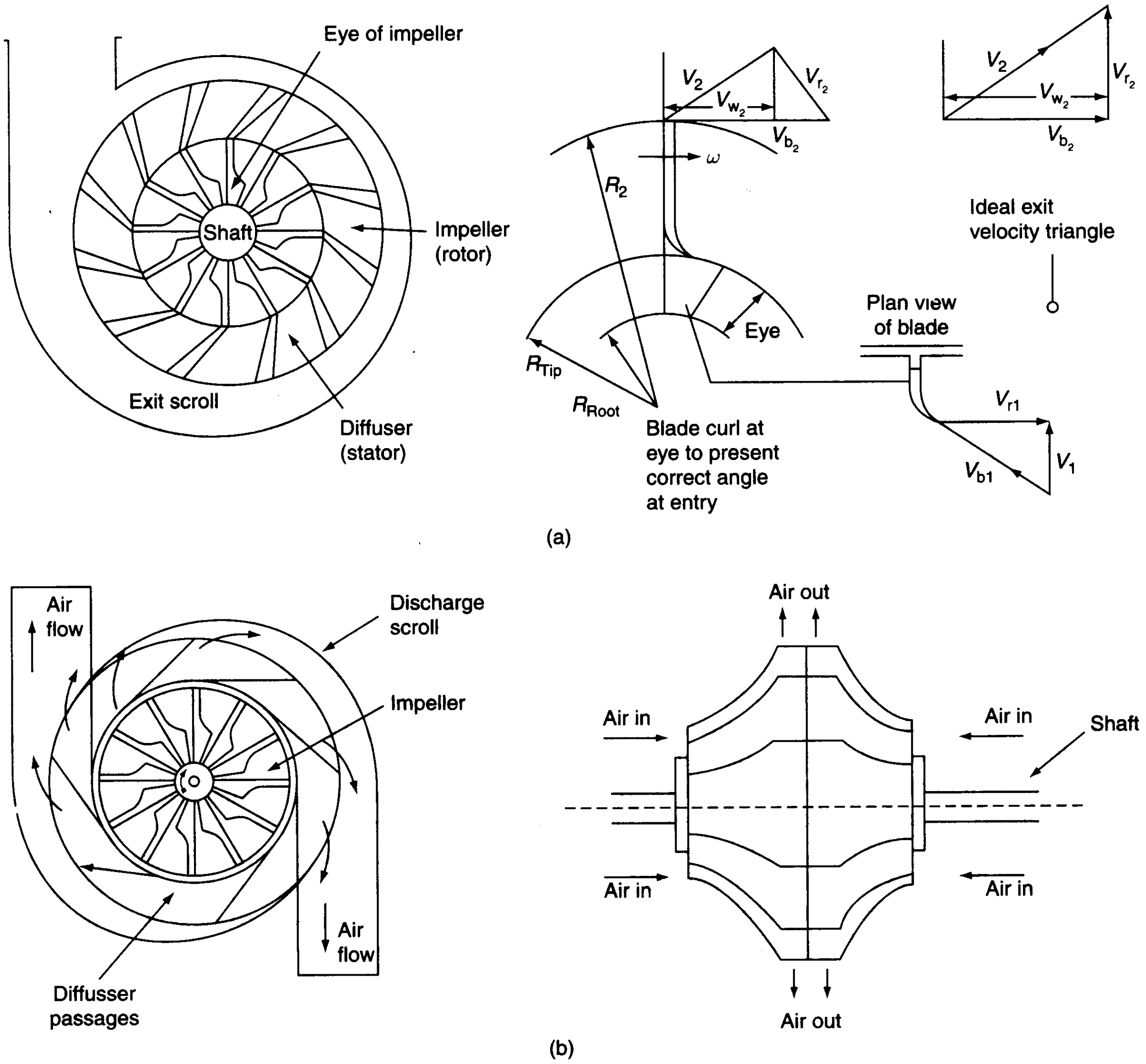



**Fig. 19.18** (a) *The centrifugal compressor (b) Doulble-sided impeller of a centrifugal compressor*

The velocity diagrams at inlet and outlet of the impeller are shown in Fig. 19.19. The air enters the impeller with an absolute velocity $V_1 = V_{f1}$ in the axial direction and $V_{b1}$ is the peripheral velocity at the inlet. Thus, the air will be entering the blades at an angle $\beta_1$, the blade angle at inlet with relative velocity $V_{r1}$. The air will be leaving the blades in the radial direction ($\beta_2 = 90°$) with relative velocity $V_{r2}$, and with a n absolute velocity at outlet $V_2$. There is no velocity of whirl at inlet ($V_{\omega 1} = 0$) (Fig.19.19a), the theoretical work required to drive the compressor would be

$$W_c = \rho Q\, V_{\omega_2} V_{b_2} = \rho Q\, V_{\omega_2}^2 = \rho V_{b_2}^2 \tag{19.30}$$

where $Q$ is the volume flow rate of air ($m^3/s$).

The fluid leaving the impeller is directed into the fixed diffuser vanes.

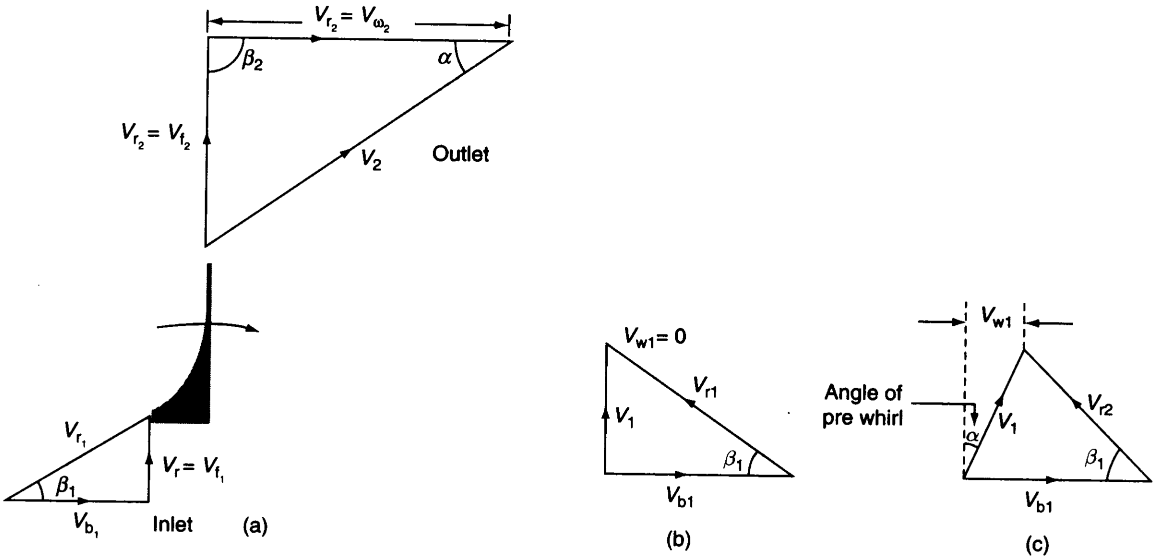



**Fig. 19.19** *Blade velocity diagrams at blade inlet of a (a) Velocity triangles centrifugal compressor (b) without, and (c) with pre-whirl*

***(a) Pre-whirl*** The rise in the static pressure of the fluid flowing through the centrifugal compressor is caused by the impeller running at a very high speed. The profiles of the impeller blades are designed in such a way that the absolute velocity of the fluid at the inlet is axial and the fluid enters the blades without shock. The velocity triangle at the inlet is a right-angled triangle ($V_{\omega 1} = 0$).

When the peripheral velocity of the blades is very high, the relative velocity at the inlet is much higher and the flow through the passage may be supersonic leading to the formation of shock waves. In order to reduce the relative velocity at the inlet, the fluid is given a pre-whirl so that the flow remains subsonic through the flow passage, shock waves do not form and the losses are minimum. The pre-whirl is achieved by providing guide vanes at the impeller inlet, without reducing the mass-flow rate. The velocity triangle at the inlet gets modified (Fig. 19.19b), and the work input to the compressor is given by

$$W_c = \rho Q V_{b_2} \left( V_{\omega_2} V_{b_2} - V_{\omega_1} V_{b_1} \right) \tag{19.31}$$

That is, the pre-whirl reduces the work input of the compressor by $V_{\omega_1} V_{b_1}$. The inlet velocity of the impeller eye is inclined by an angle, known as *pre-whirl*, $\alpha$.

At exit from the impeller, the flow is in the radial direction and the blade velocity $V_{b_2}$ is larger, since the radius of the impeller is larger at the outlet.

The blade velocity diagram is shown in Fig. 19.20 (a) being the case of radially inclined blades, and (b) being that of blades inclined backward at an angle $\beta_2$.

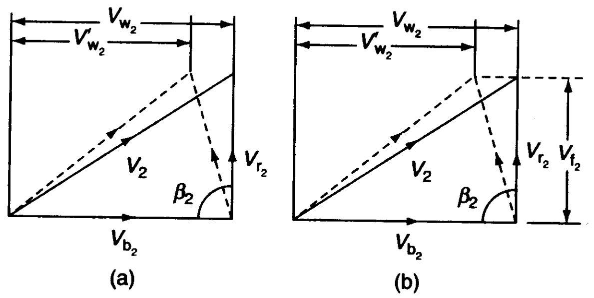



**Fig. 19.20** *Blade velocity diagrams at blade outlet of a centrifugal compressor for (a) radially inclined and (b) backward inclined blading*

**(b) *Slip Factor*** The inertia of the air trapped between the impeller blades, however, causes, actual whirl velocity $v'_{\omega_2}$ to be less than $v_{\omega_2}$. It is known as *slip* $(v_{\omega_2} - v'_{\omega_2})$.

$$\text{Slip factor, } \sigma = \frac{V'_{\omega_2}}{V_{\omega_2}} = \frac{V'_{\omega_2}}{V_{b_2} - V_{f2} \cot \beta_2} \tag{19.32}$$

$$\text{Power input, } W_c = \dot{m}\left(V_{b2} \cdot V'_{\omega 1} - V_{b_1} V_{\omega_1}\right) \tag{19.33}$$

The *slip factor* $\sigma$ is defined as the ratio of work done (when the number of blades are finite) to the work done under ideal conditions (when the number of blades are infinite). Thus

$$\sigma = \frac{V'_{\omega_2} V_{b_2}}{V_{\omega_2} V_{b_2}} = \frac{V'_{\omega 2}}{V_{\omega 2}}$$

The smaller the slip factor, the lower the pressure ratio achieved. The slip factor may be increased by increasing the number of vanes in the impeller, but this has the adverse effect of making the eye more solid.

Because of skin friction, heat losses and leakage around the vanes, the actual work input is greater than the theoretical value.

The work input to the compressor is then

$$W_c = Z\dot{m}\sigma\left(V'_{\omega_2} V_{b_2} - V_{\omega_1} V_{b_1}\right) \tag{19.34}$$

where Z in greater than 1.

The steady-flow energy equation applied to a control volume around the adiabatic compressor gives

$$\dot{W}_c = \dot{m} c_p (T_2 - T_1) \tag{19.35}$$

The *isentropic efficiency* of the compressor is defined by (Fig. 19.21)

$$\eta_s = \frac{(\dot{W}_c)_{\text{ideal}}}{(W_c)_{\text{actual}}} = \frac{T_{2s} - T_1}{T_2 - T}$$

and

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}},$$

$r_p$ being the pressure ratio.

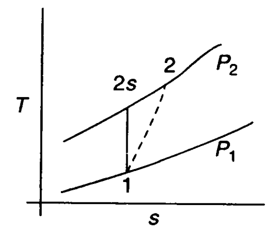



Fig. 19.21

$$\therefore \quad W_c = \frac{\dot{m} c_p T_1}{\eta_s}\left[(r_p)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

From Eq. (19.34) if $V_{\omega_1} = 0, Z = 1, V_{b_2} = V_{\omega_2}$,

$$r_p = 1 + \eta_s\left[1 + \eta_s\left(\frac{\sigma V_{b_2}^2}{c_p T_1}\right)\right] \tag{19.36}$$

For values of $T_1 = 289$ K, $\eta_s = 0.8$ and $V_{b_2}$ limited to 440 m/s by stress conditions, it is found that the pressure ratio $r_p$ achieved by a centrifugal air compressor will not exceed 4 to 1. This means that the centrifugal compressor has limited application and when high pressure ratios are required axial flow machines must be used.

**(c) *Influence of Impeller Blade Shape*** The various shapes of blades are shown in Fig.19.22. It is seen from the velocity triangles that $V_{\omega_2}$ is maximum for forward curved blades ($\beta_2 > 90°$) and $V_{\omega_2}$ is minimum for backward curved blades ($\beta_2 < 90°$). Since the power required is directly proportional to $V_{w_2}$, the power input to the radial blades will be between the two cases. From the efficiency point of view, backward curved blades are the best, requiring the minimum work input.

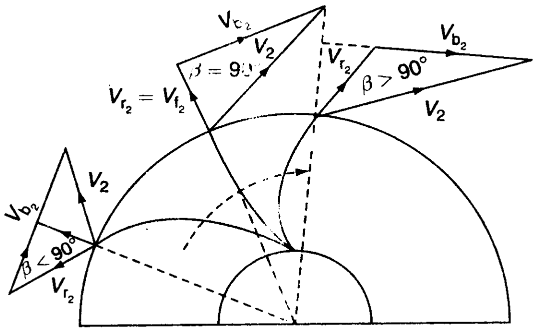


Fig. 19.22 *Influence of blade shape*

***(d) Stagnation Values in Centrifugal Compressors*** Since very high velocity is imparted to the air flowing through the compressor, the total or stagnation properties of air are significant. Let us assume that air is entering the compressor with velocity $V_1$, pressure $p_1$ and temperature $T_1$. The stagnation temperature is

$$T_{01} = T_1 + \frac{V_1^2}{2c_p}$$

and stagnation pressure is

$$p_{01} = p_1 \left( \frac{T_{01}}{T_1} \right)^{\frac{\gamma}{\gamma-1}}$$

For isentropic compression from $p_{01}$ to $p_{02}$ (Fig. 19.23)

$$\frac{T'_{02}}{T_{01}} = \left( \frac{p_{02}}{p_{01}} \right)^{\frac{\gamma-1}{\gamma}}$$

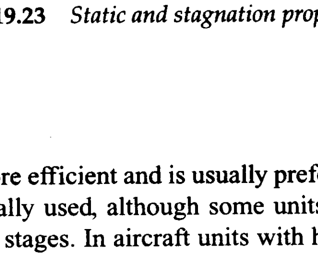


Fig. 19.23 *Static and stagnation properties*

The isentropic efficiency of the compressor is

$$\eta_c = \eta_s = \frac{T'_{02} - T_{01}}{T_{02} - T_{01}}$$

The power input becomes

$$W_c = \dot{m}\, c_p \left( T_{02} - T_{01} \right)$$

## 19.7.2 Axial Flow Compressor

For larger units with high pressure ratios the axial flow compressor is more efficient and is usually preferred. For industrial and large gas turbine plants axial compressors are normally used, although some units may employ two or more centrifugal compressors with intercooling between stages. In aircraft units with higher pressure ratios, the advantage of the smaller diameter axial flow compressor can offset the disadvantage of the increased length and weight compared with an equivalent centrifugal compressor. However, centrifugal compressors are cheaper to produce, more robust, less prone to icing problems at high altitudes, and have a wider operating range than the axial flow type.

An axial flow compressor (Figs 19.16b, 19.24a) is similar to an axial flow turbine with a succession of moving blades on the rotor shaft and fixed blades arranged around the stator casing (Fig. 19.24b). Air flows axially through the moving and fixed blades, with stationary guide blades provided at entry to the first row of moving blades.

The work input to the rotor shaft is transferred by the moving blades to the air, thus accelerating it. The blades are arranged so that spaces between blades form diffuser passages, and hence the velocity of the air relative to the blades is decreased as the air passes through them, and there is a rise in pressure. The air is then further diffused in the stator blades which also form diffusing passages (Fig. 19.24a). One

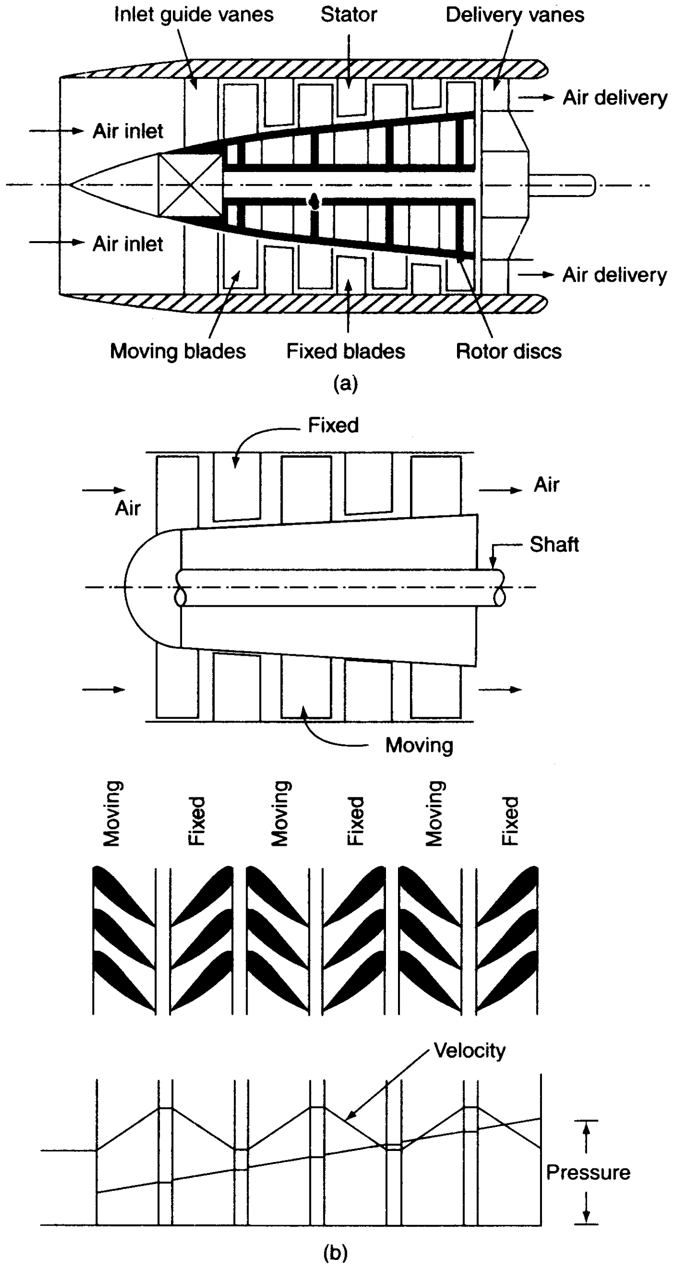
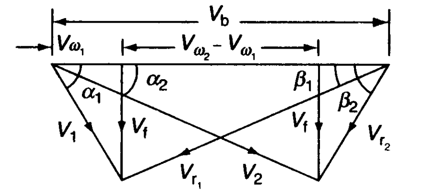



**Fig. 19.24** (*a*) *Axial flow compressor* (*b*) *velocity and pressure variation in axial flow compressor*

row of moving blades along with a row of fixed blades constitutes a stage, and the pressure ratio developed in a stage is about 1.2. There can be a large number of stages (5 to 14) with a constant work input per stage.

Figure 19.24(b) shows the pressure and velocity variation of the fluid flowing through several stages of the axial flow compressor. The pressure continuously rises through the successive stages of the compressor, but the absolute velocity increases in the rotor blades and decreases in the stator blade passages.



For $R = 0.5$, $\alpha_1 = \beta_1$; $\alpha_2 = \beta_2$

**Fig. 19.25** *Velocity triangles for the rotor and stator of an axial flow compressor*

An equal temperature rise in the moving and fixed blades is usually maintained. The axial velocity of air ($V_f$) is also kept constant throughout the compressor. Typical blade sections of an axial flow compressor are shown in Fig. 19.26(a) and the corresponding velocity diagrams in Fig. 19.26(b).

The air coming out of the fixed blades of the previous stage enters the moving blades with relative velocity $V_{r1}$ and comes out with relative velocity $V_{r2}$ and absolute velocity $V_2$. The peripheral speed $V_b$ and the velocity of flow $V_f$ is constant at the inlet and outlet. From the velocity triangles (Fig. 19.25).

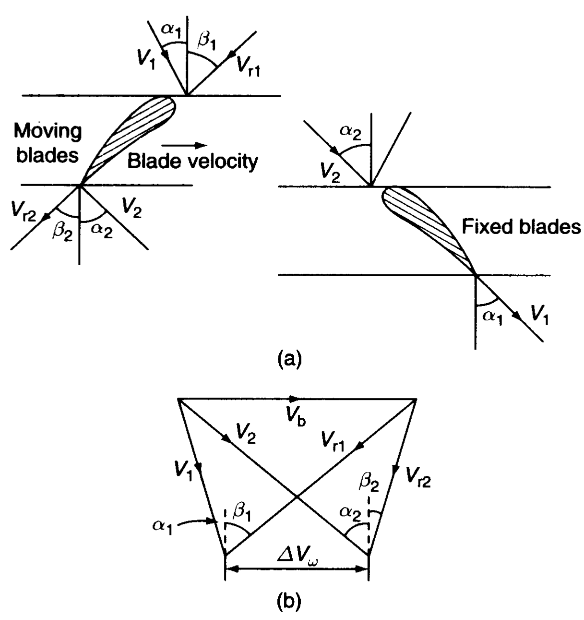



**Fig. 19.26** *(a) Typical blade sections (b) blade velocity diagrams for an axial-flow compressor*

$$V_b = V_f(\cot\beta_1 + \cot\alpha_1) = V_f(\cot\beta_2 + \cot\alpha_2)$$

Power input for a mass flow rate of $\dot{m}$ kg/s,

$$W = \Delta V_\omega V_b = \left(V_{\omega_2} - V_{\omega_1}\right)V_b = V_f(\cot\alpha_2 - \cot\alpha_1)V_b \tag{19.37}$$

The degree of reaction $R$ is defined as the ratio

$$R = \frac{\text{Enthalpy rise in rotor}}{\text{Enthalpy rise in the stage}} = \frac{\Delta h_r}{\Delta h_r + \Delta h_s} = \frac{\Delta T_r}{\Delta T_r + \Delta T_s} \tag{19.38}$$

where $\Delta T_r$ and $\Delta T_s$ are the temperature rises in the rotor and stator respectively.

From the SFEE for the rotor, we have

$$h_1 + \frac{V_1^2}{2} + W = h_2 + \frac{V_2^2}{2}$$

neglecting any heat loss,

$$\therefore \quad \Delta h_r = h_2 - h_1 = W - \frac{V_2^2 - V_1^2}{2}$$

where $W$ is the work done on 1 kg of air.

$$\therefore \quad R = \frac{W - \left(V_2^2 - V_1^2\right)/2}{W} = 1 - \frac{V_2^2 - V_1^2}{2W}$$

$$= 1 - \frac{V_2^2 - V_1^2}{2V_b(\cot\alpha_2 - \cot\alpha_1)V_f} = 1 - \frac{V_f^2(\text{cosec}^2\alpha_2 - \text{cosec}^2\alpha_1)}{2V_bV_f(\cot\alpha_2 - \cot\alpha_1)}$$

$$= 1 - \frac{V_f}{2V_1} \cdot \frac{(\cot^2\alpha_2 - \cot^2\alpha_1)}{(\cot\alpha_2 - \cot\alpha_1)}$$

$$= 1 - \frac{V_f}{2V_b}(\cot\alpha_1 + \cot\alpha_2) \tag{19.39}$$

Also, $R = \frac{V_f}{2V_b}(\tan\beta_1 + \tan\beta_2)$

For $R = 0.5, \frac{V_b}{V_f} = \cot\beta_1 + \cot\alpha_1$

$\therefore$ $\beta_2 = \alpha_1, \alpha_2 = \beta_1$

and $V_1 = V_{r_2}, V_2 = V_{r_1}$

For 50% reaction blading, with equal enthalpy or temperature rise in the rotor and stator, the velocity triangles are symmetrical.

Blades are usually of twisted section designed according to free vortex theory (see Cohen et al.).

Due to non-uniformity of velocity profiles in the blade passages, the work that can be put into a given blade is less than that given by the ideal diagram. It is taken care of by introducing a *work done factor, y*, defined as

Work done factor, $y = \frac{\text{Actual power input}}{\dot{m}\Delta V_\omega \cdot V_b}$

which is about 0.85 for a compressor stage.

***(c) Polytropic Efficiency*** The compression process between the stages of an axial flow compressor is never isentropic, through it can be nearly adiabatic due to its high speed of rotation.

In Fig. 19.27, for a 4-stage compression, the pressure ratios for the stages are $p_2/p_1, p_3/p_2, p_4/p_3$ and $p_5/p_4$. If the entire process of compression would have been isentropic, the state points will follow the line 1–6. But for polytropic compression, the actual path followed is 1-2′-3′-4′-5′. The isentropic efficiency for the second stage, e.g., is

$$\eta_{st} = \frac{T_3 - T_2'}{T_3' - T_2'}$$

while the *polytropic or internal* efficiency

$$\eta_{int} = \eta_{pol.} = \frac{T_6 - T_1}{T_5' - T_1}$$

The polytropic efficiency will always be less than the average stage efficiency because the sum $[(h_2 - h_1) + (h_3 - h_{2'})h + (h_4 - h_{3'}) + (h_5 - h_{4'})$ is greater than $(h_6 - h_1)$.

For a polytropic process,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \quad \text{or} \quad T = \text{const.} \times p^{\frac{n-1}{n}}$$

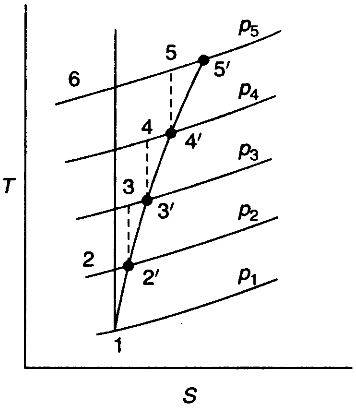


**Fig. 19.27** *Polytropic compression in a 4-stage axial compression*

or,
$$\frac{dT}{T} = \frac{n-1}{n}\frac{dp}{p}.$$

For an isentropic process,
$$\frac{dT}{T} = \frac{\gamma-1}{\gamma}\frac{dp}{p}.$$

Let us assume that when the compression process is isentropic the temperature rise is $dT$ and $dT'$ is the temperature rise for a polytropic process, then the stage or isentropic efficiency

$$\eta_s = \frac{dT}{dT'} = \frac{\gamma-1}{\gamma} \times \frac{n}{n-1}$$

$$\therefore \qquad \frac{n-1}{n} = \frac{\gamma-1}{\eta\gamma}$$

For a given compressor, let the total pressure rise be given by $p_r = p_5/p_1$, and the actual temperature rises from $T_1$ to $T_{5'}$, then internal or polytropic efficiency is given by

$$\eta_{\text{int.}} = \eta_{\text{pol.}} = \frac{T_6 - T_1}{T_{5'} - T_1} = \frac{(T_6/T_1 - 1)}{(T_{5'}/T_1 - 1)} = \frac{(p_r)^{\frac{\gamma-1}{\gamma}} - 1}{(p_r)\eta_s^{\frac{\gamma-1}{\gamma}} - 1}$$

## 19.8 BLOWERS AND FANS

*Fans* are used to handle large volume flows of air with a *pressure ratio varying between* 1.0 *and* 1.1. They run at slow speed, compared to compressors, and with or without casing. They are either centrifugal or axial flow type.

*Blowers* have *pressure ratios varying between* 1.1 to 4.0 and their speeds are higher than fans. Blowers may have more than one stage of compression and are widely used in power plants, workshops and for ventilation. The principles of operation of blowers and fans are similar to those of compressors.

### Solved Examples

**Example 19.1**

*A single cylinder reciprocating compressor has a bore of* 120 *mm and a stroke of* 150 *mm, and is driven at a speed of* 1200 *rpm. It is compressing* $CO_2$ *gas from a pressure of* 120 *kPa and a temperature of* 20°C *to a temperature of* 215°C. *Assuming polytropic compression with n* = 1.3. *no clearance and volumetric efficiency of* 100%, *calculate* (*a*) *pressure ratio,* (*b*) *indicated power,* (*c*) *shaft power, with a mechanical efficiency of* 80%, (*d*) *mass flow rate.*

*If a second stage of equal pressure ratio were added, calculate* (*e*) *the overall pressure ratio and* (*f*) *the bore of the second stage cylinder, if the same stroke was maintained.*

*Solution* (a)
$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\frac{n}{n-1}} = \left(\frac{488}{298}\right)^{\frac{1.3}{0.3}} = 8.48$$

(b)
$$V_l = V_S = \frac{\pi}{4}(0.12)^2 \times 0.15 = 0.0017 \text{ m}^3$$

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

$$\therefore \quad W = \frac{1.3}{0.3} \times 120 \times 10^3 \times 0.0017 \left[ (8.48)^{\frac{0.3}{1.3}} - 1 \right] = 563.6 \text{ J}$$

$$\text{Indicated power} = \frac{563.6 \times 1200}{60} \times 10^{-3} = 11.27 \text{ kW}$$

(c) Shaft power $= \dfrac{11.27}{0.8} = 14.1$ kW

(d) Volume flow rate $\dot{V} = 0.0017 \times \dfrac{1200}{60}$

$$= 0.034 \text{ m}^3/\text{s}$$

$$\therefore \quad \dot{m} = \frac{p_1 \dot{V}_1}{RT_1}$$

$$\frac{120 \times 10^3 \times 0.034}{(8314/44) \times 298} = 0.0725 \text{ kg/s}$$

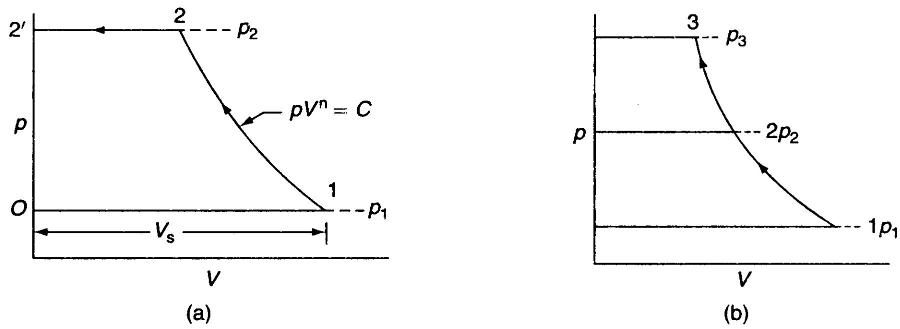



Fig. Ex. 19.1

(e) If a second stage were added with the same pressure ratio, the overall pressure ratio would be

$$\frac{p_3}{p_1} = \frac{p_3}{p_2} \times \frac{p_2}{p_1} = 8.48 \times 8.48 = 71.9$$

(f) Volume delivered per cycle is $V_2$

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

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

$$= \left( \frac{1}{8.48} \right)^{\frac{1}{1.3}} \times 0.0017 = 0.00033 \text{ m}^3$$

The second stage would have a swept volume of 0.00033 m$^3$. Since stroke is the same,

$$\frac{\pi}{4} d^2 \times 0.15 = 0.00033 \text{ m}^3$$

$$\therefore \quad d = 0.053 \text{ m} = 53 \text{ mm}$$

**Example 19.2**

*A single stage reciprocating compressor takes in air at a pressure of 96 kPa and a temperature of 305 K. The air is compressed to a pressure of 725 kPa and delivered to a reservoir. The clearance volume of the compressor is 5 percent of the swept volume. Both the compression and expansion processes may be represented by a reversible relation $pv^{1.3}$ = constant. Determine the volumetric efficiency referred to atmospheric conditions of 101.3 kPa and 292 K.*

*Solution*

$$\eta_{vol.} = \left[1 + C - C\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}\right] \times \frac{p_1}{p_a} \times \frac{T_a}{T_1}$$

$$= \left[1 + 0.05 - 0.05\left(\frac{725}{96}\right)^{\frac{1}{1.3}}\right] \times \frac{96}{101.3} \times \frac{292}{305}$$

**Example 19.3**

*A single stage reciprocating air compressor has a swept volume of 2000 cm³ and runs at 800 rpm. It operates on a pressure ratio of 8, wih a clearance of 5% of the swept volume. Assume NTP room conditions and at inlet (p = 101.3 kPa, t = 15°C), and polytropic compression and expansion with n = 1.25. Calculate (a) indicated power, (b) volumetric efficiency, (c) mass flow rate, (d) FAD, (e) isothermal efficiency, (f) the actual power needed to drive the compressor, if mechanical efficiency is 0.85.*

$= 0.737$ or 73.7% *Ans.*

*Solution* $p_1 = 101.3$ kPa, $p_2 = 8p_1 = 810.4$ kPa

$T_1 = 288$ K, $V_S = 2000$ cm³

$V_3 = V_c = 0.05\, V_S = 100$ cm³

$V_1 = V_c + V_s = 2100$ cm³

$p_3 V_3^n = p_4 V_4^n$

$$\therefore \quad V_4 = \left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} \cdot V_3 = (8)^{\frac{1}{1.25}} \times 100 = 528 \text{ cm}^3$$

$V_1 - V_4 = 2100 - 528 = 1572$ cm³

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

$$= \frac{1.25}{0.25} \times 101.3 \times 10^3 \times 1572 \times 10^{-6}\left[(8)^{\frac{0.25}{1.25}} - 1\right] = 411 \text{ J}$$

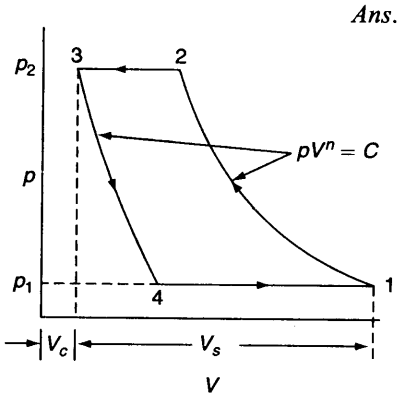


Fig. Ex. 19.3

(a) Indicated power $= \dfrac{411 \times 800 \times 10^{-3}}{60} = 5.47$ kW

(b) Volumetric efficiency $= \dfrac{1572}{2000} \times 100 = 78.6\%$

(c) Mass of air compressed per cycle

$$m = \frac{pV}{RT} = \frac{101.3 \times 10^3 \times 1572 \times 10^{-6}}{287 \times 288} = 1.93 \times 10^{-3}\,\text{kg}$$

$\therefore$ Mass flow rate $= 1.93 \times 10^{-3} \times 800 = 1.54$ kg/min

(d) FAD = Free air delivery $= 1572 \times 10^{-6} \times 800 = 1.26$ m$^3$/min

(e) $W_t = p_1(V_1 - V_4)\ln\dfrac{p_2}{p_1} = 101.3 \times 10^3 \times 1572 \times 10^{-6} \ln 8 = 331$ J

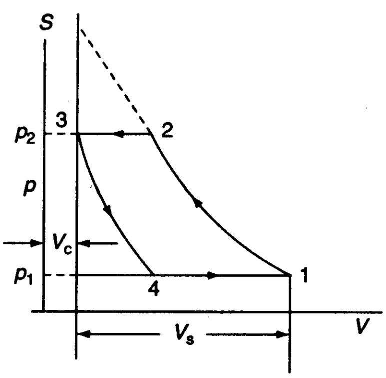
$$\eta_{\text{isothermal}} = \frac{0.331 \times 800}{5.47 \times 60} = 80.7\%$$

(f) Input power $= \dfrac{5.47}{0.85} = 6.44$ kW

## *Example* 19.4

*A single cylinder double-acting reciprocating compressor compresses* 3 *kg/min of air from* 1 *bar,* 300 *K to* 6 *bar. The clearance volume is such that the maximum pressure at the end of compression cannot exceed* 15 *bar. Calculate the power input, the volumetric efficiency and the cylinder dimensions, when the compressor runs at* 300 *rpm, the index of compression and expansion is equal to* 1.3 *and stroke/bore ratio is* 1.5.

*Solution* As shown in Fig. Ex. 19.4, as the pressure ratio $(p_2/p_1)$ increases, the volume of air trapped in the clearance space reduces the actual quantity of air taken inside the cylinder and the volumetric efficiency decreases. When the delivery pressure is 15 bar, the volumetric efficiency would be zero. The power input is




[redacted]

$$W_c = \frac{n}{n-1}\dot{m}RT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{0.3} \times \frac{3}{60} \times 0.287 \times 300 \times \left[(6)^{\frac{0.3}{1.3}} - 1\right]$$

$= 9.55$ kW **Ans.**

Now, $\quad p_1 V_1^n = p_5 V_5^n$

$$\therefore \quad V_5 = V_c = \left(\frac{p_1}{p_5}\right)^{\frac{1}{n}} V_1 = \left(\frac{1}{15}\right)^{\frac{1}{1.3}} V_1 = 0.1245\, V_1$$

$$\frac{V_1}{V_c} = \frac{1}{0.1245} = 8.032$$

$$\frac{V_s}{V_c} = 8.032 - 1 = 7.032$$

$$p_3 V_3^{\,n} = p_4 V_4^{\,n}$$

$$\therefore \quad V_4 = V_3 \left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} = V_c (6)^{\frac{1}{1.3}} = 3.968\, V_c$$

Volume of air taken in (F.A.D.) at 1 bar, 15°C

$$= V_1 - V_4 = (8.032 - 3.968) V_c \times \frac{288}{300} = 3.9\, V_c$$

$$\eta_{vol} = \frac{V_1 - V_4}{V_s} = \frac{3.9\, V_c}{7.032\, V_c} = 0.5548 \text{ or } 55.48\%$$ *Ans.*

The compressor handles 3 kg/min and it is double-acting and runs at 300 rpm.

Volume of air taken in on one side of the piston per cycle at 1 bar, 288 K,

$$= \frac{3 \times 287 \times 288}{2 \times 10^5 \times 300} = 4.1328 \times 10^{-3} \text{ m}^3$$

$$V_s = \frac{4.1328 \times 10^{-3}}{0.5548} = 7.45 \times 10^{-3} \text{ m}^3 = \frac{\pi}{4} d^2 \times 1.5d$$

$$\therefore \quad d = 0.1848 \text{ m} \quad \text{and} \quad l = 0.2772 \text{ m}$$ *Ans.*

### *Example* 19.5

*A single acting reciprocating compressor with cylinder of 15 cm diameter and 18 cm stroke has a clearance volume of 4% of swept volume. It takes in air at 1 bar, 25°C and delivers at 8 bar while running at 1200 rpm. The actual power input is 18 kW. Estimate (a) the power required to drive the unit, (b) the isothermal efficiency, and (c) the mechanical efficiency when the mass flow rate in 4 kg/min.*

*Solution* Specific volume of air entering the compressor at 1 bar, 25°C is

$$v = \frac{RT}{p} = \frac{0.287 \times 298}{100} = 0.855 \text{ m}^3/\text{kg}$$

Volume flow rate handled, $\dot{V} = 0.855 \times 4 = 3.42 \text{ m}^3/\text{min}$

Swept volume $= \frac{\pi}{4} d^2 lN = \frac{\pi}{4} \times (0.15)^2 \times 0.18 \times 1200 = 3.84 \text{ m}^3/\text{min}$

$$\therefore \quad \eta_{vol} = \frac{3.42}{3.84} = 0.89$$

Now, $\eta_{vol} = 1 + C - C\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$

$$1+0.04-0.04\times 8^{1/n}=0.89$$

$$8^{1/n}=\frac{0.15}{0.04}=3.75$$

$$\frac{1}{n}\ln 8=\ln 3.75$$

$\therefore$ $$n=1.573$$

$$\frac{T_2}{T_1}=\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}=(8)^{\frac{0.573}{1.573}}=\frac{T_2}{298}=8^{0.364}$$

$$T_2=298\times 2.13=635.2\text{ K.}$$

Power required $$W_c=\frac{n}{n-1}\dot{m}R(T_2-T_1)=\frac{1.573}{0.573}\times\frac{4}{60}\times 0.287\times(635.2-298)$$

$$=17.71\text{ kW}$$

$\therefore$ $$\eta_{mech}=\frac{17.71}{18}=0.984\text{ or }98.4\%$$ *Ans.*

$$W_{isothermal}=\dot{m}RT_1\ln\frac{p_2}{p_1}=\frac{4}{60}\times 0.287\times 298\ln 8=11.85\text{ kW}$$

$$\eta_{iso.}=\frac{11.85}{18}=0.6585\quad\text{or}\quad 65.85$$ *Ans.*

**Example 19.6**

*A single-stage reciprocatirg compressor has two double-acting cylinders each having 40-cm diameter and 50-cm stroke. The pisto ι rod diameter is 5-cm and the speed is 300 rpm. The inlet condition of air is at 1 bar, 15°C. The deli∙ery pressure is 7.5 bar. If the volumetric efficiency is 80%, mechanical efficiency is 95% and the isoth∙rmal efficiency is 70%, determine the power required to drive the compressor.*

*Solution* Swept volume of the cylinder (head end) $=\frac{\pi}{4}\times(0.4)^2\times 0.5=0.063\text{ m}^3$

Swept volume on the crank side $=\frac{\pi}{4}\left[(0.4)^2-(0.05)^2\right]\times 0.5=0.0619\text{ m}^3$

Swept volume/min for both cylinders = (0.063 + 0.0619) × 2 × 300 = 74.94 $m^3$/min

Actual volume handled by the compressor $=74.94\times\eta_{vol}=74.94\times 0.8=59.95\text{ m}^3\text{/min}$

Isothermal work $=p_1\dot{V}_1\ln\frac{p_2}{p_1}$

$=59.95\times 10^5\times\ln 7.5=120.79$ J/min

Input power $=\frac{120.79}{0.7}=172.56$ kW *Ans.*

Power required to drive the compressor $=\frac{\text{power input}}{\eta_{mech}}=\frac{172.56}{0.95}=181.64$ kW *Ans.*

**Example 19.7**

*A two-stage air compressor with perfect intercooling takes in air at 1 bar pressure and 27°C. The law of compression in both the stages is $pv^{1.3}$ = constant. The compressed air is delivered at 9 bar from the H.P. cylinder to an air receiver. Calculate, per kilogram of air, (a) the minimum work done and (b) the heat rejected to the intercooler.*

*Solution* The minimum work required in a two-stage compressor is given by Eq. (19.20),

$$W_c = \frac{2nRT_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{n-1/n} - 1\right]$$

$$= \frac{2\times1.3\times0.287\times300}{0.3}\left[(3)^{0.3/1.3} - 1\right]$$

$$= 26 \times 0.287 \times 100 \times 0.287 = 214.16 \text{ kJ/kg}$$

$$p_2 = \sqrt{p_1p_3} = \sqrt{1\times9} = 3 \text{ bar}$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{n-1/n} = 3^{0.3/1.3} = 1.28856$$

$$T_2 = 386.56 \text{ K}$$

Heat rejected to the intercooler

$$= 1.005\,(386.56 - 300) = 86.99 \text{ kJ/kg}$$

**Example 19.8**

*A single-acting two-stage air compressor deals with 4 m³/min of air at 1.013 bar and 15°C with a speed of 250 rpm. The delivery pressure is 80 bar. Assuming complete intercooling, find the minimum power required by the compressor and the bore and stroke of the compressor. Assume a piston speed of 3 m/s, mechanical efficiency of 75% and volumetric efficiency of 80% per stage. Assume the polytropic index of compression in both the stages to be n = 1.25 and neglect clearance.*

*Solution* $p_2 = \sqrt{p_1p_4} = \sqrt{1.013\times80} = 9$ bar

Minimum power required by the compressor

$$\dot{W} = \frac{2n}{n-1}p_1\dot{V}_1\left[\left(\frac{p_2}{p_1}\right)^{n-1/n} - 1\right]\times\frac{1}{\eta_{mech}}$$

$$= \frac{2\times1.25}{0.25}\times\frac{1.013\times100}{0.75}\times\frac{4}{60}\left[\left(\frac{9}{1.013}\right)^{0.25/1.25} - 1\right]$$

$$= \frac{1013\times4}{45}\times 0.548 = 49.34 \text{ kW}$$

If $L$ be the stroke length of the piston.

$$2L\frac{N}{60} = 3 \text{ m/s}$$

$$L = \frac{90 \times 100}{250} = 36 \text{ cm}$$

Effective LP swept volume = 4/250 = 0.016 m$^3$

$$\frac{\pi}{4}(D_{LP})^2 \times 0.36 \times \eta_{vol} = 0.016$$

$$D_{LP} = \sqrt{\frac{0.016 \times 4}{\pi \times 0.36 \times 0.8}}$$

$$= 0.266 \text{ m} = 26.6 \text{ cm}$$

$$\frac{p_1 V_1}{T_1} = \frac{p_3 V_3}{T_3} \qquad \therefore \qquad \frac{V_3}{V_1} = \frac{p_1}{p_3}$$

$\therefore$

$$\frac{\frac{\pi}{4} D^2_{HP} L}{\frac{\pi}{4} D^2_{LP} L} = \frac{1.013}{9}$$

$$D_{HP} = 0.266\sqrt{\frac{1.013}{9}} = 0.892 \text{ m} = 8.92 \text{ cm}$$

**Example 19.9**

*A gas is compressed in a two-stage reciprocating compressor from* 1 *bar,* 300 *K to* 9 *bar. Assuming perfect intercooling, estimate the compressor work and the total heat transfer. Take* $R$ = 0.287 *kJ/kg K,* $c_p$ = 1.042 *kJ/kg K and* $n$ = 1.3.

*Solution* Inter cooler pressure, $p_i = \sqrt{p_1 p_4} = \sqrt{1 \times 9} = 3$ bar

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 300(3)^{0.286} = 386.57 \text{ K}$$

$$W_c = \frac{2n}{n-1} mR(T_2 - T_1)$$

$$= \frac{2 \times 1.3}{0.3} \times 1 \times 0.287(386.67 - 300) = 222.83 \text{ kJ/kg}$$

For eacy cylinder, $W_c$ = 111.415 kJ/kg

Assuming perfect intercooling, heat transfer in the intercooler,

$$Q = mc_p (T_2 - T_1) = 1 \times 1.042 \text{ } (386.57 - 300) = 90.206 \text{ kJ/kg}$$

LP compressor: SFEE gives

$$h_1 + W_c = h_2 + Q$$

$$Q = h_1 - h_2 + W_c$$

$$= 1.042 \text{ } (300 - 386.57) + 111.415$$

$$= 21.209 \text{ kJ/kg}$$

Similary, for HP compressor, $Q' = 21.209$ kJ/kg

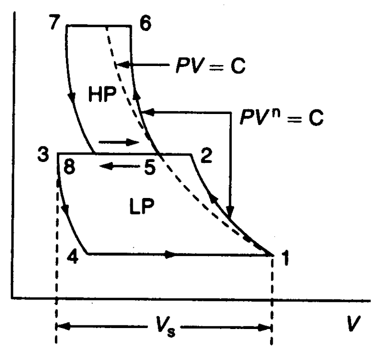
$\therefore$ total heat transfer to the surroundings

$$= 90.206 + 2 \times 21.209 = 132.624 \text{ kJ/kg} \qquad \textit{Ans.}$$

### *Example* 19.10

*The intake conditions of a single-acting two-stage air compressor running at 300 rpm are 0.98 bar and 305 K. The delivery pressure is 20 bar. The intermediate pressure is 5 bar and the clearance volume of the low-pressure compressor is 4% of the stroke volume. The compressor delivers 3 m³/min at 1 bar, 25°C. Determine (a) the power required to drive the compressor, (b) the low pressure cylinder dimensions if L = D, and (c) isothermal efficiency when the intercooling is perfect and the index n = 1.3 for compression and expansion in both the cylinders.*

*Solution* The P-V diagram has been shown in Fig. Ex. 19.10. For the LP cylinder,




$$\frac{V_3}{V_s} = 0.04$$

$$V_3 = 0.04\, V_s$$

$$V_1 = 1.04\, V_s$$

$$V_4 = V_3\left(\frac{p_3}{p_4}\right)^{\frac{1}{n}} = 0.04\, V_s\left(\frac{5}{0.98}\right)^{\frac{1}{1.3}} = 0.14\, V_s$$

Volume of air taken in at 0.98 bar, 305 K $= V_1 - V_4$

$$= 1.04\, V_s - 0.14\, V_s$$

$$= 0.90\, V_s$$

$\therefore$ Volume of air taken at free air conditions

$$= 0.90\, V_s \times \frac{298}{305} \times \frac{0.98}{1.00} = 0.861\, V_s$$

$\therefore$ Volumetric efficiency $= \dfrac{V_1 - V_4}{V_s} = 0.861$ or $86.1\%$.

Mass of air delivered, $\dot{m} = \dfrac{p\dot{V}}{RT} = \dfrac{3 \times 10^5}{287 \times 298 \times 60} = 5.846 \times 10^{-2}$ kg/s

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

$$\therefore \quad T_2 = 305\left[\frac{5}{0.98}\right]^{\frac{0.3}{1.3}} = 444.24 \text{ K}$$

For perfect intercooling, $T_1 = T_5 = 305$ K

$$T_6 - T_5\left(\frac{p_6}{p_5}\right)^{\frac{n-1}{n}} = 305\left(\frac{20}{5}\right)^{\frac{0.3}{1.3}} = 420 \text{ K}$$

Power required to drive the compressor

$$\dot{W}_c = \frac{n}{n-1}\dot{m}R\left[(T_2 - T_1) + (T_6 - T_5)\right]$$

$$= \frac{1.3}{0.3} \times 5.846 \times 10^{-2} \times 0.287\left[(444.24 - 305) + (420 - 305)\right]$$

$$= 18.484 \text{ kW}$$ *Ans.* (a)

Mass of air delivered per stroke $= 5.846 \times 10^{-2} \times \frac{60}{300} = 1.1692 \times 10^{-2}$ kg

Volume of free air per stroke $= \frac{1.1692 \times 10^{-2} \times 287 \times 298}{10^5} = 10^{-2}$ m$^3$

Stroke volume $= \frac{\pi}{4} d^2 l = \frac{10^{-2}}{0.861} = 1.61 \times 10^{-2}$ m$^3$

$$\therefore \quad d^3 = \frac{1.61 \times 10^{-2} \times 4}{\pi}$$

$$\therefore \quad d = l = 0.2735 \text{ m} = 27.35 \text{ cm}$$ *Ans.* (b)

Isothermal power required

$$= \dot{m}RT_1 \ln\frac{p_2}{p_1} = 5.846 \times 10^{-2} \times 0.287 \times 305 \ln\frac{20}{0.98} = 15.433 \text{ kW}$$

$\therefore$ isothermal efficiency $= \frac{15.433}{18.484} = 0.835$ or 83.5% *Ans.* (c)

### *Example* 19.11

*A multi-stage air compressor takes in air at* 1 *bar,* 298 *K and delivers at* 36 *bar. The maximum temperature in any stage is not to exceed* 390 *K. If the law of compression and expansion is* $pv^{1.3}$ = *constant, find the number of stages for minimum power input. Estimate the power required. What would have been the power required for single-stage compression? What would be the maximum temperature in any-stage?*

*Solution* The optimum pressure ratio per stage given by Eq. (19.25),

$$\frac{p_2}{p_1} = \left(\frac{p_d}{p_s}\right)^{\frac{1}{N}},$$

where $N$ is the number of stages.

Since the maximum temperature should not exceed 390 K,

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\frac{n}{n-1}} = \left(\frac{390}{298}\right)^{\frac{1.3}{0.3}} = 3.205 \text{ or } 4 \text{ stages}$$ *Ans.*

Corresponding intermediate pressures are

$$p_2 = (36)^{\frac{1}{4}} = 2.45 \text{ bar}$$

$$P_3 = (36)^{0.5} = 6.0 \text{ bar}$$

$$P_4 = (36)^{0.75} = 14.7 \text{ bar}$$

Total power required (Eq. 19.26)

$$W_c = \frac{N_n R T_1}{n-1}\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{N_n}} - 1\right]$$

$$= \frac{4\times1.3\times0.287\times298}{0.3}\left[\left(\frac{36}{1}\right)^{\frac{0.3}{4\times1.3}} - 1\right]$$

$$= 1482.45\left[(2.45)^{0.23} - 1\right]$$

$$= 339.3 \text{ kW/kg air.}$$

For a single-stage compressor, the power required

$$W_c = \frac{n}{n-1} mRT_1\left[\left(\frac{p_d}{p_s}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3}{0.3}\times1\times0.287\times298\left[(36)^{\frac{0.3}{1.3}} - 1\right]$$

$$= 370.61\ (2.28 - 1) = 474.4 \text{ kW}$$ *Ans.*

Maximum temperature in any stage

$$T = T_1(2.45)^{\frac{0.3}{1.3}} = 298\times(2.45)^{0.23} = 366.2 \text{ K.}$$

**Example 19.12**

*The pressure and temperature of air supplied to an air engine are 700 kPa 38°C respectively. Cut-off takes place at 0.4 of the stroke. Expansion follows the law $pv^{1.3} = c$ (constant) to the release point, which is at the end of the outstroke. The pressure then falls to the constant back pressure of 112 kPa. Neglect the effect of clearance and assuming that the area of the actual indicator diagram is 0.85 of that outlined above, determine the indicated output if the air mass is 1.25 kg.*

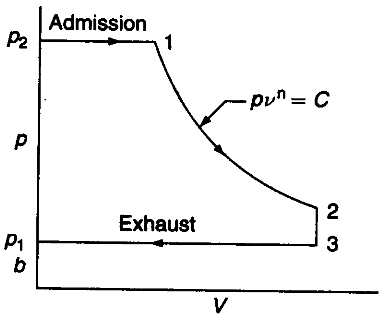
*Solution* $V_1 = 0.4\,V_2, \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1}$

$$= (2.5)^{0.3} = 1.316$$

$$\therefore \quad T_2 = \frac{273+38}{1.316} = 236.3 \text{ K}$$

$$p_1V_1^{1.3} = p_2V_2^{1.3}$$

$$\therefore \quad p_2 = p_1\left(\frac{V_1}{V_2}\right)^{1.3} = 700(0.4)^{1.3} = 212.7 \text{ kPa}$$

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$$v_2 = \frac{mRT_2}{p_2} = \frac{1.25 \times 0.287 \times 236.3}{212.7} = 0.3986 \text{ m}^3$$

$$v_2 = \frac{V_2}{m} = \frac{0.3986}{1.25} = 0.3188 \text{ m}^3/\text{kg}$$

Area of indicator diagram

$$= p_1 v_1 + \frac{p_1 v_1 - p_2 v_2}{n-1} - p_3 v_3 = RT_1 + \frac{R(T_1 - T_2)}{n-1} - p_3 v_3$$

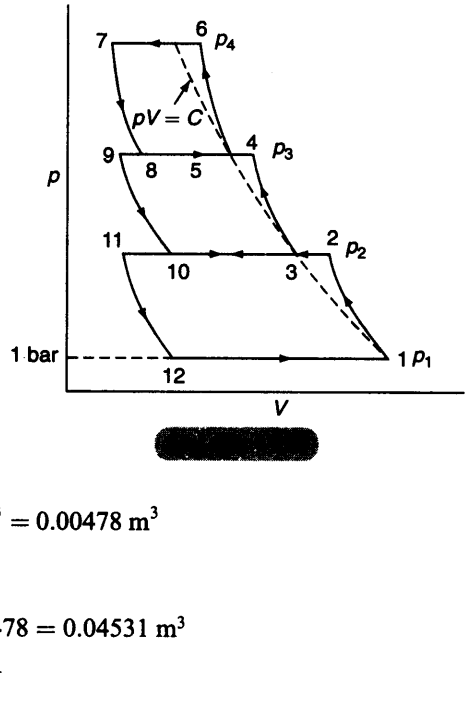
$$= 0.287 \times 311 + \frac{0.287(311 - 236.3)}{0.3} - 112 \times 0.3188 = 125 \text{ kJ/kg}$$

Indicated output $= 125 \times 0.85 = 106.26$ kJ/kg $= 106.26 \times 1.25 = 132.82$ kJ

### *Example* 19.13

*A three-stage single-acting reciprocating air compressor has a low-pressure (L.P.) cylinder of* 450 *mm bore and* 300 *mm stroke. The clearance volume of the l.p. cylinder is* 5% *of the swept volume. Intake pressure and temperature are* 1 *bar and* 18°C *respectively, while the delivery pressure is* 15 *bar. Intermediate pressures are ideal and intercooling is perfect. The compression and expansion index can be taken as* 1.3. *Estimate (a) the intermediate pressures, (b) the effective swept volume of L.P. cylinder, (c) the temperature and volume of air delivered per stroke at* 15 *bar, and (d) the work done per kg of air. Take* $R = 0.29$ *kJ/kg K.*

*Solution*

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(a) $$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} = \left(\frac{p_4}{p_1}\right)^{1/3} = (15)^{1/3} = 2.466$$

$$p_2 = 2.466 \text{ bar}$$

$$p_3 = 2.466 \times 2.466 = 6.081 \text{ bar}$$

(b) Swept volume of L.P. cylinder (Fig. Ex. 19.13)

$$= V_1 - V_{11} = \frac{\pi}{4} \times (0.45)^2 \times 0.3 = 0.0477 \text{ m}^3$$

$$V_{11} = 0.05 \times 0.0477 = 0.00239 \text{ m}^3$$

$$\therefore \quad V_1 = 0.0477 + 0.00239 = 0.05009 \text{ m}^3$$

$$p_{11} V_{11}^{1.3} = p_{12} V_{12}^{1.3}$$

$$V_{12} = V_{11} \times \left(\frac{p_{11}}{p_{12}}\right)^{1/1.3} = 0.00239 \times (2.466)^{1/1.3} = 0.00478 \text{ m}^3$$

Effective swept volume of the L.P. cylinder is

$$V_1 - V_{12} = 0.05009 - 0.00478 = 0.04531 \text{ m}^3$$

$$T_5 = T_3 = T_1 = 291 \text{ K}$$

$$\frac{T_6}{T_5} = \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}}$$

$$\therefore \quad T_6 = 291 \times (2.466)^{\frac{0.3}{1.3}} = 358.5 \text{ K}$$

$$\therefore \quad t_6 = \text{delivery temperature} = 85.5°\text{C}$$

$$\frac{p_4(V_6 - V_7)}{T_6} = \frac{p_1(V_1 - V_{12})}{T_1}$$

$$\therefore \quad V_6 - V_7 = \frac{p_1}{p_6} \cdot \frac{T_6}{T_1} \cdot (V_1 - V_{12}) = \frac{1}{15} \times \frac{358.5}{291} \times 0.04531 = 0.00372 \text{ m}^3$$

$$\text{Work per kg air} = \frac{3nRT_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{3 \times 1.3 \times 0.29 \times 291}{0.3}\left[(2.466)^{\frac{0.3}{1.3}} - 1\right] = 254.3 \text{ kJ}$$

### *Example* 19.14

*For a Roots blower, the inlet pressure is 1.013 bar and the pressure ratio is 1.5 to 1. The induced volume of air is 0.03 m³/rev. Estimate the work input. What would be the work input for a vane-type compressor if the internal compression takes place through half the pressure range.*

*Solution* For the Roots blower (Fig. Ex. 19. 14(a))

$$p_1 = 1.013 \text{ bar}$$

$$p_2 = 1.5 \times 1.013 = 1.52 \text{ bar}$$

$$V_s = 0.03 \text{ m}^3/\text{rev.}$$

$$\therefore \quad \text{Work done} = (p_2 - p_1)V_s$$

$$= (1.52 - 1.013) \times 100 \times 0.03 = 1.52 \text{ kJ/rev}$$

For the vane-type compressor (Fig. Ex. 19.14(b))

$$p_i = \frac{(1.5 \times 1.013) + 1.013}{2} = 1.266 \text{ bar}$$

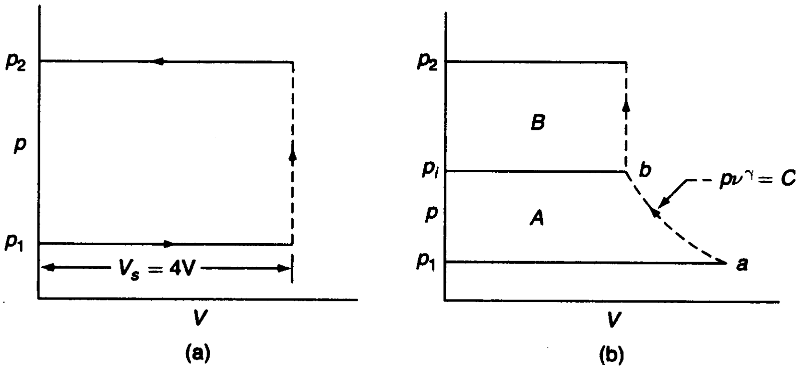
$$\text{Work required} = \text{Area } A + \text{Area } B$$

$$\text{Area } A = \frac{\gamma}{\gamma - 1} p_1 V_s \left[\left(\frac{p_i}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= \frac{1.4}{0.4} \times 1.013 \times 100 \times 0.03 \left[\left(\frac{1.266}{1.013}\right)^{\frac{0.4}{1.4}} - 1\right] = 0.70 \text{ kJ/rev}$$

$$\text{Area } B = (p_2 - p_i)\, V_b$$

$$p_1 V_a^{\gamma} = p_i V_b^{\gamma}$$




$$V_b = V_a\left(\frac{p_1}{p_i}\right)^{\frac{1}{\gamma}} = 0.03\left(\frac{1.013}{1.266}\right)^{\frac{1}{1.4}} = 0.0256 \text{ m}^3$$

Area $B = (1.52 - 1.266) \times 100 \times 0.0256 = 0.65$ kJ/rev

$\therefore$ Worked required $= 0.70 + 0.65 = 1.35$ kJ/rev.

(Compared to 1.52 kJ/rev. for the Roots blower)

**_Example_ 19.15**

*A Roots blower supplies air at the rate of 1 kg/s. The pressure ratio of the blower is 2:1 with an intake pressure and temperature of 1 bar and 70°C respectively. Find the power required to drive the blower. Take $R = 0.29$ kJ/kg K.*

*If a vane pump having the same air flow, pressure ratio and intake conditions has the volume reduced to 0.7 of the intake volume before delivering the air, estimate the power required.*

*Solution* $$\dot{V} = \frac{\dot{m} R T_1}{p_1} = \frac{1 \times 0.29 \times 343}{100} = 0.995 \text{ m}^3\text{/s}$$

$\therefore$ Power required by the Roots blower

$$= \dot{V}(p_2 - p_1) = 0.995 \times 100 = 99.5 \text{ kW}$$

For the vane compressor, $p_1 V_1^\gamma = p_2 V_2^\gamma$

$$\therefore \quad p_2 = p_1\left(\frac{V_1}{V_2}\right)^\gamma = 1 \times \left(\frac{1}{0.7}\right)^{1.4} = 1.65 \text{ bar}$$

$$\dot{V}_2 = 0.7\,\dot{V}_1 = 0.7 \times 0.995 = 0.696 \text{ m}^3\text{/s}$$

$$\text{Power required} = \frac{\gamma}{\gamma - 1} p_1 \dot{V}_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + \dot{V}_2(p_3 - p_2)$$

$$= \frac{1.4}{0.4} \times 100 \times 0.995 \left[ \left( \frac{1.65}{1} \right)^{\frac{0.4}{1.4}} - 1 \right]$$

$$+ 0.696\,(2 - 1.65) \times 100 = 78\text{kW}$$

**Example 19.16**

*A gas turbine utilizes a two-stage centrifugal compressor. The pressure ratios for the first and second stages are 2.5 to 1 and 2.1 to 1 respectively. The flow of air is 5 kg/s, this air being drawn at 1.013 bar, and 10°C. If the temperature drop in the intercooler is 50°C and the isentropic efficiency is 85% for each stage, calculate (a) the actual temperatures at the end of eacn stage (b) the total compressor power. Take $\gamma = 1.4$ and $c_p = 1.005$ kJ/kg K.*

*Solution* Stage I

$$\frac{T_{2s}}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} \quad \text{(Fig. Ex. 19.16)}$$

$$T_{2s} = 283(2.5)^{0.4/1.4} = 367.7\text{ K}$$

$$T_2 - T_3 = 50°\text{C}$$

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

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

$$= 283 + \frac{367.7 - 283}{0.85} = 382.65\text{ K}$$

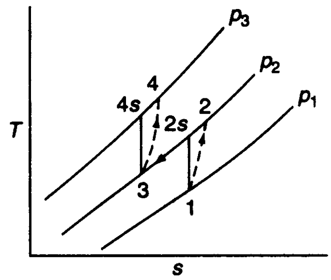
Stage II $T_3 = T_2 - 50 = 382.65 - 50 = 332.65$ K

$$\frac{T_{4s}}{T_3} = \left( \frac{p_3}{p_2} \right)^{\frac{\gamma - 1}{\gamma}}$$

$\therefore$ $T_{4s} = 332.65 \times (2.1)^{0.4/1.4} = 411.19$ K

$$T_4 - T_3 = \frac{T_{4s} - T_3}{n_s}$$

$$= \frac{411.19 - 322.65}{0.85} = 104.17\text{ K}$$

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$\therefore$ $T_4 = 104.17 + 332.65 = 436.82$ K

$$\text{Total compressor power} = \dot{m}_a c_p [(T_2 - T_1) + (T_4 - T_3)]$$

$$= 5 \times 1.005\,[(382.65 - 283) + (436.82 - 332.65)]$$

$$= 1024.2\text{ kW}$$

**Example 19.17**

*A rotary compressor is used to supercharge a petrol engine. The static pressure ratio across the rotor is 2.5 : 1. Static inlet pressure and temperature are 0.6 bar and 5°C respectively. The air-fuel ratio is 13:1 and the engine consumes 0.04 kg fuel/s. For the air-fuel mixture take $\gamma$ = 1.39 and $c_p$ = 1.005 kJ/kg K. The isentropic efficiency of the compressor is 84%. Estimate the power required to drive the compressor. Taking the exit velocity from the compressor as 120 m/s and assuming that the mixture is adiabatically brought to rest in the engine cylinders, estimate the stagnation temperature and pressure of the mixture at the beginning of the compression stroke in the engine cylinder.*

*Solution*

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (2.5)^{\frac{0.39}{1.39}} = 1.294$$

$$T_{2s} = 278 \times 1.294 = 359.7 \text{ K}$$

$$T_2 - T_1 = \frac{T_{2s} - T_1}{\eta_s} = \frac{359.7 - 278}{0.84} = 97.3 \text{ K}$$

$$\therefore \quad T_2 = 97.3 + 278 = 375.3 \text{ K}$$

Mass of air-fuel mixture = 0.04 × (13+1) = 0.56 kg/s

Power to drive the compressor = $\dot{m}_g \, c_p \, (T_2 - T_1)$

$$= 0.56 \times 1.005 \times 97.3 = 54.8 \text{ kW}$$

Stagnation temperature

$$T_{02} = T_2 + \frac{V_2^2}{2c_p} = 375.3 + \frac{120^2}{2 \times 1.005 \times 1000}$$

$$= 382.46 \text{ K} = 109.46°\text{C}$$

The temperature in the engine cylinder is 109.46°C

$$p_{02} = p_2 \times \left(\frac{T_{02}}{T_2}\right)^{\frac{\gamma}{\gamma-1}} = 0.6 \times 2.5 \times \left(\frac{382.46}{375.3}\right)^{0.36} = 1.605 \text{ bar or } 160.5 \text{ kPa}$$

**Example 19.18**

*A centrifugal compressor running at 10,000 rpm delivers 1.2 m³/s of free air. The pressure and temperature at the inlet are 1 bar and 27°C. The pressure ratio is 5, the blades are radial at outlet. The velocity of flow is 60 m/s and is constant throughout, and the slip factor is 0.9. Calculate (i) the pressure of air at outlet, (ii) power input, (iii) the impeller diameter and blade angle at inlet, and (iv) diffuser inlet angle. Take the isentropic efficiency as 0.85 and the impeller diameter at inlet as half of that at outlet.*

*Solution*

$$T_{2s} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (273+27)5^{\frac{0.4}{1.4}} = 475.36 \text{ K}$$

$$\eta_s = \frac{T_2 - T_1}{T_{2s} - T_1} = 0.85$$

$$\therefore \qquad T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_s} = 300 + \frac{175.36}{0.85} = 506.3\,\text{K} = 233.3\ \text{C} \qquad \textit{Ans. (i)}$$

$$\dot{m} = \frac{p\dot{v}}{RT} = \frac{100 \times 1.2}{0.287 \times 288} = 1.452\,\text{kg/s}$$

Power input, $\dot{W}_c = \dot{m}c_p(T_2 - T_1)$

$$= 1.452 \times 1.005 \times (506.3 - 300) = 301\ \text{kW} \qquad \textit{Ans. (ii)}$$

Since the blades are radial at outlet

$$\dot{W}_c = \dot{m}\,\sigma\,V_{b_2}^2$$

$$V_{b_2} = \left[\frac{301 \times 10^3}{1.452 \times 0.9}\right]^{\frac{1}{2}} = 480\,\text{m/s} = \frac{\pi DN}{60}$$

$\therefore$ Diameter of the impeller

$$D = \frac{480 \times 60}{\pi \times 10{,}000} = 0.9166\,\text{m} \qquad \textit{Ans. (iii)}$$

$$V_{b_1} = \frac{V_{b_2}}{2} = \frac{480}{2} = 240\,\text{m/s}$$

$$\tan\beta_1 = \frac{V_f}{V_{b_1}} = \frac{60}{240} = 0.25$$

$$\therefore \qquad \beta_1 = \text{blade inlet angle} = 14° \qquad \textit{Ans. (iii)}$$

From the outlet velocity triangle (Fig. 19.19),

$$\tan\alpha = \frac{V_f}{\sigma V_{b_2}} = \frac{60}{0.9 \times 480}$$

$$\therefore \qquad \alpha = 7.9° \qquad \textit{Ans. (iv)}$$

**Example 19.19**

*The following data comprise the design specification for a single-sided centrifugal air compressor. Calculate the total head pressure ratio that should be achieved, the required power at the input shaft and the inlet angles of the blades at the root and tip of the impeller eye.*

*Data: Rotational speed = 264 rps; slip factor = 0.91; Impeller diameter = 0.482 m; Impeller eye tip diameter = 0.306 m; Impeller eye root diameter = 0.153 m; Uniform axial inlet velocity = 138 m/s; Air mass-flow rate = 9.1 kg/s; Inlet air stagnation temperature = 294 K; Total head isentropic efficiency = 0.80; Mechanical efficiency = 0.98.*

*The compression process is adiabatic, and take air as an ideal gas with $c_p$ = 1.006 kJ/kg K and $\gamma$ = 1.4.*

*Solution* Power input, $\dot{W}_c = \dot{m}\sigma V_{b_2}^2$

where $V_{b_2} = 2\pi R_2 N$

$$\therefore \qquad \dot{W}_c = \frac{9.1 \times 0.91}{10^3}\left(2\pi \times \frac{0.482}{2} \times 264\right) = 1320 \text{ kW}$$

External power required $= \dfrac{1320}{0.98} = 1350$ kW *Ans.*

Now, $\dot{m}c_p \Delta T_t = \dot{W}_c$

$\therefore$ Total head temperature rise, $\Delta T_t = \dfrac{1320}{9.1 \times 1.006} = 144.5$ K

$\therefore$ Ideal temperature rise $= 144.5 \times \eta_s = 144.5 \times 0.8 = 115.5$ K

$$\therefore \qquad (T_2)_i = 294 + 115.5 = 409.5 \text{ K} = T_{02}$$

Now,
$$\frac{p_{02}}{p_{01}} = \left(\frac{T_{02}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{409.5}{294}\right)^{3.5} = 3.18$$
*Ans.*

Blade angles: At the eye, tip velocity $= 2\pi \times 264 \times \dfrac{0.306}{2} = 254$ m/s

Eye root velocity $= 127$ m/s

From velocity triangle, $\beta_1 = \tan^{-1}\left(\dfrac{V_f}{V_b}\right) = \tan^{-1}\left(\dfrac{138}{254}\right) = 28°31'$

= blade inlet angle at the eye tip

$$\beta_2 = \tan^{-1}\left(\frac{138}{127}\right) = 47°23'$$

= blade inlet angle at the eye root *Ans.*

### *Example* 19.20

*A centrifugal compressor running at 16000 rpm takes in air at 17°C and compresses it through a pressure ratio of 4 with an isentropic efficiency of 82%. The blades are radially inclined and the slip factor is 0.85. Guide vanes at inlet give the air an angle of pre-whirl of 20° to the axial direction. The mean diameter of the impeller eye is 200 mm and the absolute air velocity at inlet is 120 m/s. Calculate the impeller tip diameter. Take $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$.*

*Solution* With reference to Fig. Ex. 19.20(a),

$$\frac{T_{2s}}{T_1} = (4)^{\frac{1.4-1}{1.4}} = 4^{0.286} = 1.487$$

$$T_{2s} = 290 \times 1.487 = 431 \text{ K}$$

$$\Delta T_s = T_{2s} - 1 = 431 - 290 = 141 \text{ K}$$

$$\Delta T = 141/0.82 = 171.95 \text{K}$$

$$W_c = mc_p \Delta T = 1 \times 1.005 \times 171.95 = 172.81 \text{ kJ/kg}$$

Absolute air velocity at inlet (Fig. Ex. 19.20(b)),

$$V_1 = 120 \text{ m/s}$$

$$V_{b_1} = \frac{\pi d_1 N}{60} = \frac{\pi \times 0.2 \times 16{,}000}{60} = 167.55 \text{ m/s}$$

Pre-whirl angle = 20°

$$V_{\omega_1} = V_1 \sin 20° = 120 \sin 20° = 41.04 \text{ m/s}$$

At exit of the vanes (Fig. Ex.19.20(c)),

$V_{\omega_2} = V_{b_2}$ (since the blades are radially inclined)

slip factor $\sigma = \dfrac{V'_{\omega_2}}{V_{\omega_2}} = 0.85$

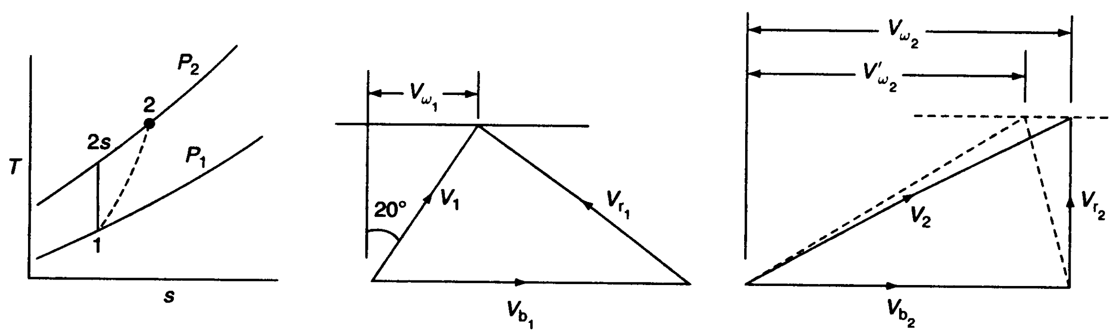
$$V'_{\omega_2} = 0.85 V_{b_2}$$

Power input per kg $V'_{\omega_2} V_{b_2} - V_{\omega_1} V_{b_1}$

$$172.81 \times 10^3 = 0.85\, V_{b_2}^2 - 167.55 \times 41.05$$

$$\therefore \quad V_{b_2} = 459.78 \text{ m/s} = \frac{\pi d_2 \times 16000}{60}$$

Tip diameter $d_2$ = 0.5488 m = 549 mm *Ans.*




(a) (b) (c)

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**Example 19.21**

*A multi-stage axial flow compressor delivers 2.5 kg/s of air. The inlet conditions are 1 bar, 300 K. Estimate (i) the delivery pressure, (ii) the number of stages and (iii) the internal efficiency of the compressor, when the stage efficiency is 0.88, the power input is 600 kW, the stage pressure ratio is the same throughout and the temperature rise in the first stage is 21 °C.*

*Solution* The stage efficiency for polytropic compression is

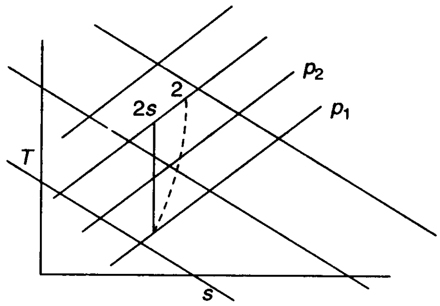


Fig. Ex. 19.21

$$\eta_s = \frac{\gamma - 1}{\gamma} \times \frac{n}{n-1}$$

$$0.88 = \frac{1.4-1}{1.4} \times \frac{n}{n-1}$$

$$\therefore \quad \frac{n-1}{n} = 3.08$$

Now, $W_c = mc_p(T - T_1)$,
where $T$ is the actual temperature of air at the end of compression Fig. Ex. 19.21.

$$\therefore \quad T = \frac{W_c}{mc_p} + T_1 = \frac{600}{2.5 \times 1.005} + 300 = 538.81 \text{ K}$$

If the delivery pressure is denoted by $p$, then

$$\frac{p}{p_1} = \left(\frac{T}{T_1}\right)^{\frac{n}{n-1}} = \left(\frac{538.81}{300}\right)^{3.08} = 6.07$$

$\therefore$ delivery pressure, $p = 6.07 \times 1 = 6.07$ bar *Ans.* (i)

The stage efficiency is given by (for the first stage)

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

$$T_{2s} - T_1 = \eta_s (T_2 - T_1) = 0.88 \times 21 = 18.48$$

$$\therefore \quad T_2 = 318.48 \text{ K}$$

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{318.48}{300}\right)^{3.5} = 1.2327$$

$p_1$ = Pressure ratio for the first stage

Taking the same pressure ratio for each stage,

$$\left(\frac{p_2}{p_1}\right)^N = 6.07$$

where $N$ is the number of stages.

$$N \ln\left(\frac{p_2}{p_1}\right) = \ln 6.07$$

$$N = \frac{\ln 6.07}{\ln 1.2327} = 8.619$$

i.e., 9 stages *Ans.* (ii)

Temperature at the end of isentropic compression

$$T_s = T_1 \left(\frac{p}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 300(6.07)^{0.286} = 502.47 \text{ K}$$

$$\therefore \quad \text{Internal efficiency} = \frac{T_s - T_1}{T_1 - T_1} = \frac{502.47 - 300}{538.81 - 300} = 0.85$$ *Ans.* (iii)

**Example 19.22**

*An axial flow compressor having an impeller of mean diameter 0.5 m rotates at 15000 rpm. The velocity of flow is constant at 230 m/s, and the velocity of whirl at inlet is 80 m/s. The inlet pressure and temperature are 1 bar and 300 K. The stage efficiency is 0.88. The pressure ratio through the stage is 1.5. Calculate the (i) fluid deflection angle, (ii) the power input, and (iii) the degree of reaction.*

*Solution*

$$V_b = \text{peripheral speed } = \frac{\pi DN}{60}$$

$$= \frac{\pi \times 0.5 \times 15000}{60} = 392.75 \text{ m/s}$$

Temperature at the end of isentropic compression

$$T_s = T_1\left(\frac{p}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 300 \times (1.5)^{0.286} = 336.85 \text{ K}$$

Actual temperature $T = T_1 + \dfrac{T_s - T_1}{\eta_s} = 300 + \dfrac{36.85}{0.88} = 341.87$ K

Work input, $\quad W_c = c_p(T - T_1) = 1.005 \times 41.87 = 42.08$ kJ/kg *Ans.* (ii)

$$\therefore \quad \left(V_{\omega_2} - V_{\omega_1}\right)V_b = 42.08$$

$$V_{\omega_2} = 80 + \frac{42.08 \times 10^3}{392.75} = 187.14 \text{ m/s}$$

From velocity triangles at inlet and outlet of blades

$$\tan\beta_1 = \frac{V_f}{V_b - V_{\omega_1}} = \frac{220}{392.75 - 80}$$

$$\therefore \quad \beta_1 = 35.12°$$

and $$\tan\beta_2 = \frac{V_f}{V_b - V_{\omega_2}} = \frac{220}{392.75 - 187.14} = \frac{220}{205.61}$$

$$\beta_2 = 46.94°$$

$\therefore$ fluid deflection angle $= \beta_2 - \beta_1 = 46.94 - 35.12 = 11.82°$ *Ans.* (i)

Degree of reaction, $$R = 1 - \frac{V_f(\cot\alpha_1 + \cot\alpha_2)}{2V_b} = 1 - \frac{V_{\omega_1} + V_{\omega_2}}{2V_b}$$

$$= 1 - \frac{80 + 187.14}{2 \times 392.75} = 0.66 \text{ or } 66\%$$ *Ans.* (iii)

**Example 19.23**

*An axial flow fan delivering 5m³/s has the mean impeller diameter of 1.0 m and the hub diameter of 0.6 m. It rotates at 600 rpm and develops a theoretical head equal to 35 mm of water. Determine the blade angles at the tip and at the hub. Assume that the velocity of flow is independent of radius and the energy input per unit. The length of the blade is constant. Take the density of air as 1.2 kg/m³ and the density of water as 1000 kg/m³.*

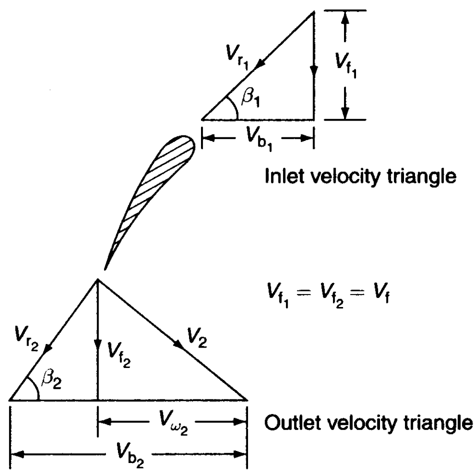


Fig. Ex. 19.23

*Solution* Velocity of flow, $V_f = \dfrac{5 \times 4}{\pi\left(1^2 - 0.6^2\right)} = 9.95$ m/s

$$V_b \text{ at blade tip } = \frac{\pi \times 1.0 \times 600}{60} = 31.42 \text{ m/s}$$

$$V_b \text{ at the hub } = \frac{\pi \times 0.6 \times 600}{60} = 18.85 \text{ m/s}$$

From Fig. Ex. 19.23,

$$\tan\beta_2 = \frac{V_{f_2}}{V_{b_2} - V_{\omega_2}}; \text{ Head developed by the fan}$$

$$= \frac{V_{b_2} \cdot V_{\omega_2}}{g}$$

Now, H = 35 mm of water

$$= \frac{0.035 \times 10^3}{1.2} = 29.17 \text{ m of air}$$

$$V_{\omega_2} \text{ at the tip} = \frac{29.17 \times 9.81}{31.42} = 9.11 \text{ m/s}$$

$$V_{\omega_2} \text{ at the hub} = \frac{29.17 \times 9.81}{18.85} = 15.18 \text{ m/s}$$

$$\beta_{tip} = \tan^{-1}\frac{V_f}{V_{b_2} - V_{\omega_2}} = \tan^{-1}\frac{9.95}{31.42 - 9.11} = 24°$$

$$\beta_{hub} = \tan^{-1}\frac{9.95}{18.85 - 15.18} = 69.75° \qquad \textit{Ans.}$$

### *Example* 19.24

*A centrifugal blower running at* 9000 *rpm compresses* 6 *m*$^3$*/s of air from* 1 *bar,* 25°*C to* 2.2 *bar. The compression index is* 1.33. *The velocity of flow is* 75 *m/s and is constant throughout the impeller. The impeller blade angles at the inlet and outlet are* 30° *and* 55° *respectively. Estimate the speed of the impeller and its width at inlet and outlet. The outlet diameter of the impeller is* 0.75 *m.*

*Solution* The outlet temperature of air of the impeller

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 298 \times \left(\frac{2.2}{1}\right)^{\frac{0.33}{1.33}} = 362.39 \text{ K}$$

Power input to the impeller

$$V_{\omega_2} V_{b_2} = c_p(T_2 - T_1) = 1.005\,(362.39 - 298) = 64.71 \text{ kJ/kg}$$

From outlet velocity triangle

$$\tan\beta_2 = \frac{V_f}{V_{b_2} - V_{\omega_2}} = \frac{75}{V_{b_2} - V_{\omega_2}} = \tan 55° = 1.428$$

$$\therefore \quad V_{b_2} - V_{\omega_2} = 52.51 \text{ m/s}$$

$$V_{\omega_2} + V_{b_2} = \left[\left(V_{b_2} - V_{\omega_2}\right)^2 + 4 V_{b_2} \cdot V_{\omega_2}\right]$$

$$= \left[(52.51)^2 + 4 \times 64.71 \times 10^3\right] = 511.43$$

$$V_{b_2} - V_{\omega_2} = 52.41$$

Solving,

$$V_{b_2} = 281.97 \text{ m/s}, \; V_{\omega_2} = 229.46 \text{ m/s}$$

$$V_{b_2} = \frac{\pi D_2 N}{60} = 281.97$$

$$\therefore \quad N = \frac{281.97 \times 60}{\pi \times 0.75} = 7180 \text{ rpm}$$ *Ans.*

$$V_{b_1} = \frac{\pi D_1 \times 7180}{60} = \frac{V_f}{\tan 30°} = \frac{75}{0.5773} = 129.9$$

$$\therefore \quad D_1 = \text{impeller diameter at inlet} = 0.345 \text{ m}$$ *Ans.*

Discharge at inlet

$$Q = \pi D_1 \, b_1 \, V_f = 6 \text{ m}^3\text{/s}$$

$$\therefore \quad b_1 = \frac{6}{\pi \times 0.345 \times 75} = 7.38 \text{ cm}$$ *Ans.*

Discharge at outlet when the pressure is 2.2 bar and temperature 362.39 K,

$$Q = 6 \times \frac{1}{2.2} \times \frac{362.39}{298} = 3.316 \text{ m}^3\text{/s} \quad = \pi D_2 \, b_2 \, V_f$$

$$\therefore \quad b_2 = \frac{3.316}{\pi \times 0.75 \times 75} = 1.88 \text{ cm}$$ *Ans.*

## Review Questions

19.1 What is the function of a compressor? What are the different types of compressors?

19.2 What are the main applications of compressors?

19.3 Derive the expressions for the reversible work of compression if the compression process is (a) adiabatic, (b) polytropic, and (c) isothermal.

19.4 Which of the compression processes needs minimum work and which the maximum work?

19.5 Define (a) the adiabatic efficiency and (b) the isothermal efficiency.

19.6 Show that for a single-stage reciprocating compressor the work of compression remains the same irrespective of its derivation on the basis of (a) a steady flow system and (b) a closed system.

19.7 Define the volumetric efficiency of a compressor. On what factors does it depend?

19.8 Discuss how the volumetric efficiency varies with the clearance and the pressure ratio. For a given pressure ratio and the polytropic index, find the maximum clearance when the volumetric efficiency is reduced to zero.

19.9 What is the maximum pressure ratio attainable with a reciprocating compressor for a given clearance?

19.10 What is the need of staging the compression process?

19.11 Show that the optimum intermediate pressure of a two-stage reciprocating compressor for minimum work is the geometric mean of the suction and discharge pressures.

19.12 Explain how does the use of intermediate pressure for minimum work result in equal pressure ratios in the two-stages of compression, equal discharge temperatures, and equal work for the two stages.

19.13 What do you mean by perfect intercooling? What is the amount of heat rejected in the intercooler?

19.14 What is the function of an aftercooler?

19.15 Explain the advantages of multistage compression?

19.16 Explain the operation of an air motor.

19.17 What are rotary compressors? How does the Roots blower operate? What is the power input to it? Define Roots blower efficiency.

19.18 What is a vane type compressor? Briefly explain its operation.

19.19 How does the pressure rise in a centrifugal compressor? Where is it used?

19.20 How does an axial flow compressor operate?

19.21 How does the pressure rise occur in a centrifugal compressor?

19.22 Why is pre-whirl provided at the vanes of a centrifugal compressor? Draw the velocity triangles at inlet and exit of the vanes and explain.

19.23 Explain what you understand by slip factor.

19.24 Explain the mechanism of flow in an axial flow compressor. Show the pressure and velocity profiles in a multi-stage compressor.

19.25 What is the degree of reaction? Draw the velocity triangles for R=50%.

19.26 Define work done factor. What is polytropic efficiency?

19.27 What is the difference between a blower and a fan? How does a compressor differ?

## Problems

19.1 A 4-cylinder single stage air compressor has a bore of 200 mm and a stroke of 300 mm and runs at 400 rpm. At a working pressure of 620 kPa (g) it delivers 3.1 m$^3$ of air per min. at 270°C. Calculate (a) mass flow rate, (b) free air delivery (FAD), (c) effective swept volume, (d) volumetric efficiency. Take free air conditions at inlet as 101.3 kPa, 21°C. *Ans.* (a) 0.239 kg/s, (b) 0.199 m$^3$s, (c) 0.0299 m$^3$ (d) 79.2 %

19.2 0.2 m$^3$ of air at 20°C and 100 kPa is compressed according to the relation $pv^{1.3}$ = constant by the piston in an engine cylinder that has a compression ratio of 6. Heat is then added while the pressure remains the same, until the piston returns to its original position. Calculate (a) mass of air, (b) pressure at the end of compression, (c) final temperature, (d) network transfer in the combined process. *Ans.* (a) 0.238 kg, (b) 1.027 MPa, (c) 2736°C, (d) 123.7 kJ

19.3 A single stage single-acting air compressor deals with 90 m$^3$/h of air at 101.325 kPa and 15°C. The pressure and temperature during the suction stroke remain constant at 98 kPa and 40°C respectively, $n$= 1.22. The air is delivered at 735 kPa, $R_a$ = 0.287 kJ/kg K. Find (a) the power needed to drive the compressor if the mechanical efficiency is 0.85, (b) the swept volume if the speed is 120 rpm. Take the volumetric efficiency as 0.78.

19.4 Find the stroke, piston diameter, and indicated power of a single-acting air compressor, which operates under the following conditions: volume of F.A.D. at 101.325 kPa and 15°C = 105 m$^3$/min; pressure and temperature at the beginning of compression, 98 kPa and 30°C; discharge pressure is 40 kPa; speed 220 rpm: $n$ = 1.25; stroke = bore and the clearance volume is 6% of the swept volume.

19.5 A two-stage single-acting compressor compresses air from 101.325 kPa and 15°C to a pressure of 162 kPa. Calculate the work done per kg of air delivered, and the heat transferred to the intercooler if the intercooling is ideal.

If the volumetric efficiency is expected to be 85% for the L.P. stage and the speed is 400 rpm, calculate a suitable diameter for the L.P. cylinder for an air delivery of 2.5 kg/min if the stroke-bore ratio is unity. Assume the polytropic index to be 1.3 and state all the assumptions made. For air, $R$ = 0.287 kJ/kg K and $c_p$ = 1.005 kJ/kg K.

19.6 Air flows steadily into a compressor at a temperature of 17°C and a pressure of 1.05 bar and leaves at a temperature of 247°C and a pressure of 6.3 bar. There is no heat transfer to or from the air as it flows through the compressor; changes in elevation and velocity are negligible. Evaluate the external work done per kg of air, assuming air as an ideal gas for which $R$ = 0.287 kJ/kg K and

$\gamma = 1.4$. Evaluate the minimum external work required to compress the air adiabatically from the same initial state to the same final pressure and the isentropic efficiency of the compressor.

*Ans.* –225 kJ/kg, –190 kJ/kg, 84.4%

19.7 A slow-speed reciprocating air compressor with a water jacket for cooling approximates a quasi-static compression process following a path $pv^{1.3}$ = const. If air enters at a temperature of 20°C and a pressure of 1 bar, and is compressed to 6 bar at a rate of 1000 kg/h, determine the discharge temperature of air, the power required and the heat transferred per kg. *Ans.* 443 K, 51.82 kW, 36 kJ/kg

19.8 A single-acting two-stage reciprocating air compressor with complete intercooling delivers 6 kg/min at 15 bar pressure. Assume an intake condition of 1 bar and 15°C and that the compression and expansion processes are polytropic with $n = 1.3$. Calculate: (a) the power required, (b) the isothermal efficiency. *Ans.* (a) 26.15 kW (b) 85.6%

19.9 A two-stage air compressor receives 0.238 m³/s of air at 1 bar and 27°C and discharges it at 10 bar. The polytropic index of compression is 1.35. Determine (a) the minimum power necessary for compression, (b) the power needed for single-stage compression to the same pressure, (c) the maximum temperature for (a) and (b), and (d) the heat removed in the intercooler.

*Ans.* (a) 63.8 kW, (b) 74.9 kW, (c) 404.2 K, 544.9 K, (d) 28.9 kW

19.10 A single-acting air compressor has a cylinder of bore 15 cm and the piston stroke is 25 cm. The crank speed is 600 rpm. Air taken from the atmosphere (1 atm, 27°C) is delivered at 11 bar. Assuming polytropic compression $pv^{1.25}$ = const., find the power required to drive the compressor, when its mechanical efficiency is 80%. The compressor has a clearance volume which is 1/20th of the stroke volume. How long will it take to deliver 1 m³ of air at the compressor outlet conditions. Find the volumetric efficiency of the compressor.

*Ans.* 12.25 kW, 3.55 min, 72%

19.11 A multistage air compressor compresses air from 1 bar to 40 bar. The maximum temperature in any stage is not to exceed 400 K. (a) If the law of compression for all the stages is $pv^{1.3}$ = const., and the initial temperature is 300 K, find the number of stages for the minimum power input. (b) Find the intermediate pressures for optimum compression as well as the power needed. (c) What is the heat transfer in each of the intercooler?

*Ans.* (a) 3 (b) 3.48 bar, 12.1 bar, 373.1 kJ/kg (c) 100.5 kJ/kg

19.12 A single stage reciprocating air compressor has a swept volume of 2000 cm³ and runs at 800 rpm. It operates on a pressure ratio of 8, with a clearance of 5% of the swept volume. Assume NTP room conditions, and at inlet, and polytropic compression and expansion with $n = 1.25$. Calculate (a) the indicated power, (b) volumetric efficiency, (c) mass flow rate, (d) the free air delivery FAD, (e) isothermal efficiency, (f) actual power needed to drive the compressor, if the mechanical efficiency is 0.85.

*Ans.* (a) 5.47 kW (b) 78.6% (c) 1.54 kg/min (d) 1.26 m³/min, (e) 80.7% (f) 6.44 kW

19.13 A two-stage single-acting reciprocating compressor takes in air at the rate of 0.2 m³/s. Intake pressure and temperature are 0.1 MPa and 16°C respectively. The air is compressed to a final pressure of 0.7 MPa. The intermediate pressure is ideal, and intercooling is perfect. The compression index is 1.25 and the compressor runs at 10 rps. Neglecting clearance, determine (a) the intermediate pressure, (b) the total volume of each cylinder, (c) the power required to drive the compressor, (d) the rate of heat absorption in the intercooler.

*Ans.* (a) 0.264 MPa (b) 0.0076 m³ h.p. and 0.02 m³ l.p. cylinder (c) 42.8 kW (d) 14.95 kW

19.14 A 3-stage single-acting air compressor running in an atmosphere at 1.013 bar and 15°C has a free air delivery of 2.83 m³/min. The suction pressure and temperature are 0.98 bar and 32°C respectively. The delivery pressure is to be 72 bar. Calculate the indicated power required, assuming complete intercooling, $n = 1.3$ and that the compressor is designed for minimum work. What will be the heat loss to the intercoolers? *Ans.* 25.568 kW, 13.78 kW

19.15 A two-stage single-acting air compressor delivers 0.07 m³ of free air per sec (free air condition 101.325 kN/m² and 15°C). Intake conditions are 95 kN/m² and 22°C. Delivery pressure from the compressor is 1300 kN/m². The intermediate pressure is ideal and there is perfect intercooling. The compression index is 1.25 in both cylinders. The overall mechanical and electrical efficiency is 75%. Neglecting clearance, determine (a) the energy input to the driving motor, (b) the heat transfer rate in the intercooler, (c) the percentage saving in work by using a two-stage intercooled compressor instead of a single-stage compressor. Take $R = 0.287$ kJ/kg K and $c_p = 1.006$ kJ/kg K.

*Ans.* (a) 29.1 kW, (b) 6.99 kW, (c) 13%

19.16 Air at 1.013 bar and 15°C is to be compressed at the rate of 5.6 $m^3$/min to 1.75 bar. Two machines are considered: (a) the Roots blower, and (b) a sliding vane rotary compressor. Compare the powers required, assuming for the vane type that internal compression takes place through 75% of the pressure rise before delivery takes place, and that the compressor is an ideal uncooled machine.

*Ans.* 6.88 kW, 5.71 kW

19.17 A rotary air compressor has an inlet static pressure and temperature of 100 kN/$m^2$ and 20°C respectively. The compressor has an air mass flow rate of 2 kg/s through a pressure ratio of 5:1. The isentropic efficiency of compression is 85%. Exit velocity from the compressor is 150 m/s. Neglecting change of velocity through the compressor, determine the power required to drive the compressor. Estimate the stagnation temperature and pressure at exit from the compressor.

*Ans.* 404 kW, 232.2°C, 541.5 kN/$M^2$

19.18 The cylinder of an air motor has a bore of 63.5 mm and stroke of 114 mm. The supply pressure is 6.3 bar, the supply temperature 24°C, and the exhaust pressure is 1.013 bar. The clearance volume is 5% of the swept volume and the cut-off ratio is 0.5. The air is compressed by the returning piston after it has travelled through 0.95 of its stroke. The law of compression and expansion is $pv^{1.3}$ = constant. Calculate (a) the temperature at the end of expansion, (b) the indicated power of the motor which runs at 300 rpm, and (c) the air supplied per minute.

*Ans.* (a) – 28.4°C, (b) 0.749 kW, (c) 0.42 kg/min.

19.19 A single-acting two-stage air compressor runs at 300 rpm and delivers 510 $m^3$/h with intake air at 1 bar, 15°C. The delivery pressure is 40 bar. Calculate for minimum work input (i) the optimum pressure ratio for each stage, (ii) the theorectical power input of each stage when the intercooling is perfect and the index of compression is $n = 1.3$, (iii) the amount of heat rejected from each cylinder and the intercooler, and (iv) the swept volumes of the L.P. and H.P. cylinders, if their volumetric efficiencies are 0.9 and 0.85 respectively.

*Ans.* (i) 6.324 bar, (ii) 65.14 kW, (iii) 6.358 kW, 26.214 kW, (iv) 0.026 $m^3$, 0.0045 $m^3$

19.20 A centrifugal compressor has a pressure ratio of 4 with an isentropic efficiency 0.82 when running at 16000 rpm. It takes in air at 1 bar, 17°C. Guide vanes at the inlet give the air a pre-whirl of 20° to the axial direction and the mean diameter of the eye is 200 mm, the absolute air velocity at the inlet is 120 m/s. At the exit, the blades are radially inclined and the impeller tip diameter is 550 mm. Calculate the slip factor of the compressor.

*Ans.* 0.847

19.21 A centrifugal compressor compresses at the rate of 2k g/s from 1 bar, 20°C to a total pressure at outlet equal to 5 bar. The velocity of air at inlet is 150 m/s and the compressor runs at 20,000 rpm. The isentropic efficiency is 0.8 and the slip factor is 0.9. Calculate (i) the change in total temperature, (ii) the impeller diameter at outlet and inlet if the hub diameter is 10 cm, and (iii) the power input.

*Ans.* (i) $T_{02} - T_{01} = 200$ K, (ii) $D_2$ =0.45 m, $D_1 = 0.156$ m, (iii) 402 kW

19.22 The mean diameter of the rotor of an axial flow compressor is 0.55 m and it rotates at 16000 rpm. The velocity of flow, 230 m/s, is constant and the velocity of whirl at the inlet is 85 m/s. The air inlet condions are 1 bar, 305 K. The stage efficiency is 0.9. The pressure ratio through the stage is 1.55. Calculate (i) the fluid deflection angle, (ii) the work input, and (iii) the degree of reaction.

*Ans.* (i) 8.23°, (ii) 45.48 kJ/kg, (iii) 0.708

19.23 An axial-flow compressor stage has a mean blade velocity of 210 m/s, stagnation temperature rise 28 K, the axial velocity of flow is constant and is equal to 150 m/s. The degree of reaction is 0.5. Determine the appropriate air angles, the stagnation pressure ratio of the stage, if the isentropic efficiency of the stage is 0.85 and the stagnation temperature at the inlet is 520 K.

*Ans.* Outlet angle 75.78°, Inlet angle 41.09°; Pressure ratio =1.169

19.24 A centrifugal fan delivers 9 $m^3$/s of air while running at 1200 rpm. The impeller diameter at the inlet and outlet are 0.6 m and 0.8 m respectively. The blades are curved backwards and the blade outlet angle is 70°. The width of the blade at the outlet is 0.12 m. The volute casing recovers 35% of the outlet velocity head, while the impeller losses are 20% of the outlet velocity head. Determine (i) the pressure rise at outlet, (ii) manometric efficiency, when the air enters the impeller in the axial direction at 16 m/s.

*Ans.* (i) Net pressure rise = 109.13 m of air (ii) 54%

C H A P T E R

# 20 Internal Combustion Engines

## 20.1 INTERNAL AND EXTERNAL COMBUSTION ENGINES

The mechanism in which the energy released by the combustion of fuel is converted to mechanical or shaft work is called a heat engine. The heat engine can be an internal or external combustion engine. In an *internal combustion engine* (*IC engine*), the combustion of fuel occurs inside the engine cylinder itself and the products of combustion are the working fluid which by virtue of its high internal energy with high temperature and pressure pushes the piston outward to do work. In an *external combustion engine,* on the other hand, the fuel is burnt outside the engine or turbine and the heat released is utilized to heat an intermediate fluid like water as in a steam power plant, which by virtue of its high enthalpy, does work on the turbine blades to produce mechanical or shaft work.

An IC engine has high overall efficiency, is compact with a high ratio of maximum output to its weight and bulk, has greater mechanical simplicity to operate, and lower initial cost. Its wide ranging applications include (i) transport vehicles in roads (cars, trucks, buses, bikes etc), rail locomotive, aircraft, marine propulsion, (ii) as a prime mover for electric generator, welding sets, grinders, pumps, compressors and blowers, fork lifts etc., (iii) agricultural machinery like harvesters, thrashers, pump sets, (iv) earth moving equipment like dumpers, excavators and so on. Because of their multifarious activities, IC engines play a very important role in industry.

## 20.2 CLASSIFICATION OF IC ENGINES

IC engines are of two types—reciprocating and rotary. Rotary IC engines are the open-cycle gas turbines which have been discussed in the next chapter. *Reciprocating IC engines* can be classified on the following basis (Fig. 20.1):

***(a) Thermodynamic Cycle***

*(i) Otts or Constant Volume Cycle* The energy released during the combustion of fuel occurs at constant volume.

*(ii) Diesel or Constant Pressure Cycle* Energy is released by combustion at constant pressure.

*(iii) Dual or Limited Pressure Cycle* Energy is released by combustion of fuel partly at constant volume and partly at constant pressure.

Energy is rejected at constant volume in all the three cycles.

***(b) Number of Strokes Per Cycle*** Reciprocating IC engines use a piston which moves back and forth inside a cylinder. The distance travelled by the piston inside the cylinder from one extreme end (top dead centre or tdc) to the other (bottom dead centre or bdc) is called one *stroke.*

*(i) 4-stroke Engine* The engine cycle is completed in four strokes of the piston

*(ii) 2-stroke Engine* The engine cycle is completed in two strokes of the piston

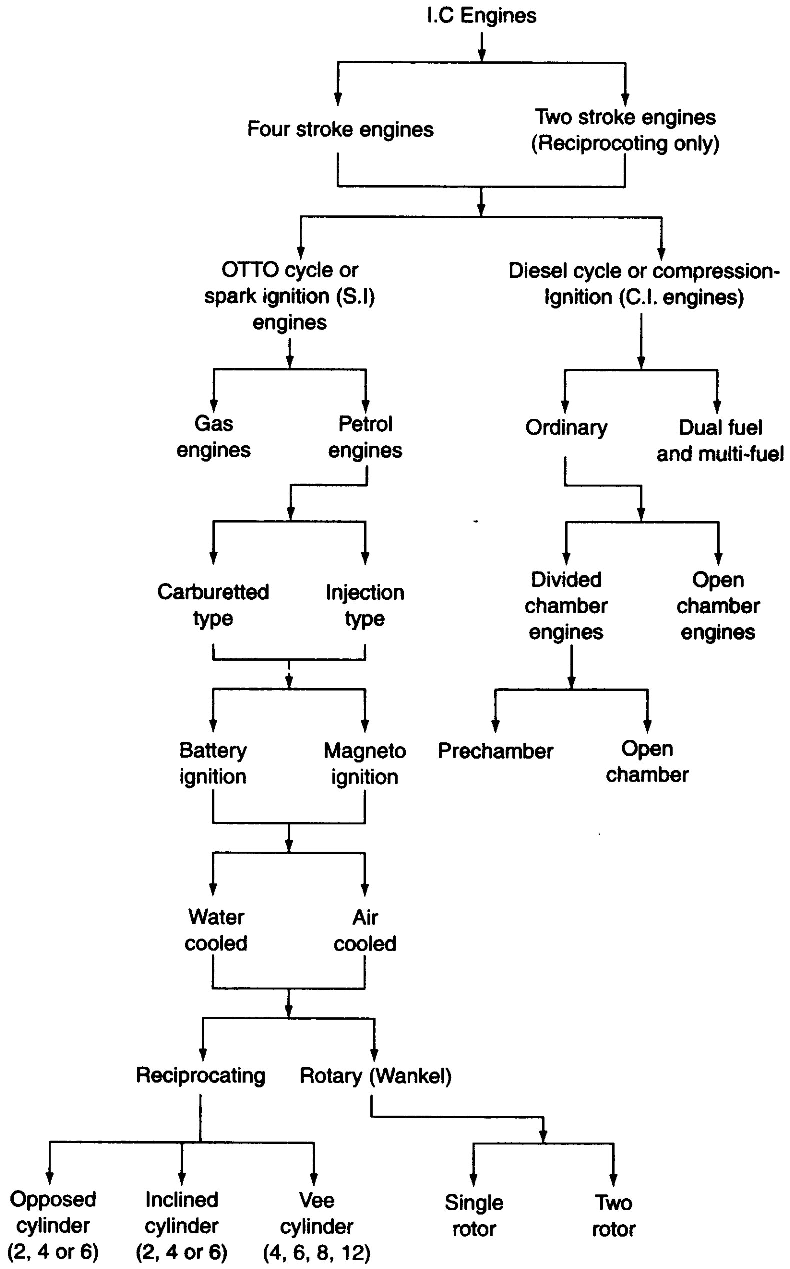



**Fig. 20.1** *Classification of IC engines*

### *(c) Ignition System*

*(i) Spark Ignition (S.I.) Engines* A homogeneous mixture of air and fuel vapour is supplied to the engine and the combustion is initiated by a spark plug.

(*ii*) *Compression Ignition* (*C.I.*) *Engines* Air sucked inside the cylinder is compressed to a higher pressure and temperature (this temperature exceeds the self-ignition temperature, S.I.T., of the fuel), the fuel is injected into the cylinder in the form of fine spray and the mixture automatically burns inside the cylinder.

The engine cycle of both the SI and CI engines can be completed in 2 strokes or 4 strokes of the piston.

**(*d*) *Fuels Used***

(*i*) *Petrol Engine* The engine uses petrol or gasoline as the source of energy

(*ii*) *Oil or Diesel Engine* Uses diesel oil as fuel

(*iii*) *Gas Engines* Use gaseous fuels like CNG, LPG, producer gas, etc.

(*iv*) *Multi-fuel Engines* Use gasoline or diesel oil for starting the engine and kerosene or biogas as their primary fuel

**(*e*) *Cooling Systems***

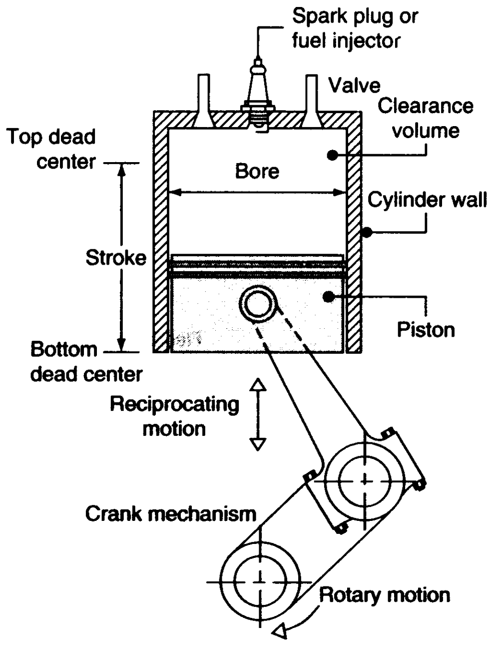
(*i*) *Water Cooled* Cylinder walls are cooled by circulating water

(*ii*) *Air Cooled* Cylinder walls are cooled by blowing atmospheric air over the hot surfaces. Motor cycles, Scooters, aircraft and a few small four wheelers have the air-cooling system.

**(*f*) *Multicylinder Engines*** The power output of an engine is directly proportional to its speed. When the reciprocating masses of the piston and the connecting rod are accelerated and decelerated, inertia forces develop and they put a limit on the speed of the engine. As such, smaller cylinders are used to reduce the inertia forces per cylinder and the forces in one cylinder can easily be balanced by a suitable arrangement of other cylinders. Thus, multicylinder engines are invariably used in 4-wheelers and other large output engines.

## 20.3 AN OVERVIEW OF RECIPROCATING ENGINES

The reciprocating engine, basically a piston-cylinder device, has a wide range of applications. It is the powerhouse of the vast majority of automobiles, trucks, light aircraft, ships, electric power generators and so on. The basic components of such an engine are shown in Fig. 20.2. The piston reciprocates in the cylinder between two fixed positions called the top dead centre (TDC)-the position of the piston when it forms the smallest volume in the cylinder-and the **bottom dead centre (BDC)**-the position of the piston when if forms the largest volume in the cylinder. The distance between the TDC and the BDC is the largest distance that the piston can travel in one direction, and it is called the **stroke** of the engine. The diameter of the piston is called the **bore**. The air or air-fuel mixture is drawn into the cylinder through the **intake valve,** and the combustion products are expelled from the cylinder through the **exhaust valve.**




*Nomenclature for reciprocating piston–cylinder engine*

The minimum volume formed in the cylinder when the piston is at TDC is called the **clearance volume**. The volume displaced by the piston as it moves between TDC and BDC is called the **displacement volume.** The ratio of the maximum volume formed in the cylinder to the minimum (clearance) volume is called the **compression ratio** $r_k$ of the engine

$$r_k = V_{max}/V_{min} = V_{BDC}/V_{TDC}$$

Note that $r_k$ is a **volume ratio** and should not be confused with the pressure ratio.

Another term frequently used in regard to reciprocating engines is the **mean effective pressure (mep).** It is a fictitions pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle.

$$W_{net} = \text{m.e.p.} \times \text{piston area} \times \text{stroke} = \text{m.e.p.} \times \text{displacement volume}$$

or,
$$\text{m.e.p., } p_m = (W_{net})/[V_{max} - V_{min}] \text{ (k Pa)}$$

The mean effective pressure can be used as a parameter to compare the performance of reciprocating engines of equal size. The engine with a larger value of m.e.p. will deliver more net work per cycle and thus will perform better.

Reciprocating engines are classified as **spark-ignition** (SI) **engines** or **compression-ignition** (CI) **engines**, depending on how the combustion process in the cylinder in initiated. In SI engines, also called petrol or gasoline engines, the combustion of the air-fuel mixture is initiated by a spark plug. In CI engines, also called diesel engines, the air-fuel mixture is self-ignited as a result of compressing the mixture above its self-ignition temperature.

## 20.4 DESCRIPTION OF AN SI ENGINE

The different parts of an SI engine including its main components are shown in Fig. 20.3. The cylinder block is made of cast iron because of its good wear resistance and low cost. The cooling water passages are provided in the block during casting. The piston is made of aluminum in small engines and cast steel in bigger engines. The space between the cylinder head and the piston crown is called the *combustion chamber.* The piston is connected to the crankshaft through a connecting rod made of forged steel. The connecting rod-crank assembly converts the reciprocating motion of the piston to rotary motion. The crankshaft is supported in bearings mounted in the crank case which is sealed at the bottom with a pressed-steel or cast-aluminium oil pan acting as an oil reservoir for the lubrication system.

The piston rings do not permit the high pressure gases to escape through the gap between the cylinder wall and the piston and also scraps the excess oil from the cylinder walls and allows the oil to return to the oil pan. The valves made of chrome – nickel alloy are usually poppet valves and are operated by a cam. The camshaft is made of forged steel and is driven by the crankshaft through gears.

## 20.5 OPERATING PRINCIPLES

The reciprocating internal combustion engine remain by far the most common form of engine or prime mover. The two main types of internal combustion engines are: Spark Ignition (SI) engines where the fuel is ignited by a spark; and compression ignition (CI) engines where the rise in temperature and pressure during compression is sufficient to cause spontaneous ignition of the fuel.

### 20.5.1 4-Storke Spark Ignition Engine

The working principles of a typical 4-stroke spark ignition or petrol engine can be described as follows:

***(a) Suction Process*** Let us consider that the piston is at the top dead centre (tdc) and the crankshaft is being rotated so that the piston moves downwards (from tdc to bdc) and the inlet valve opens to admit a homogeneous

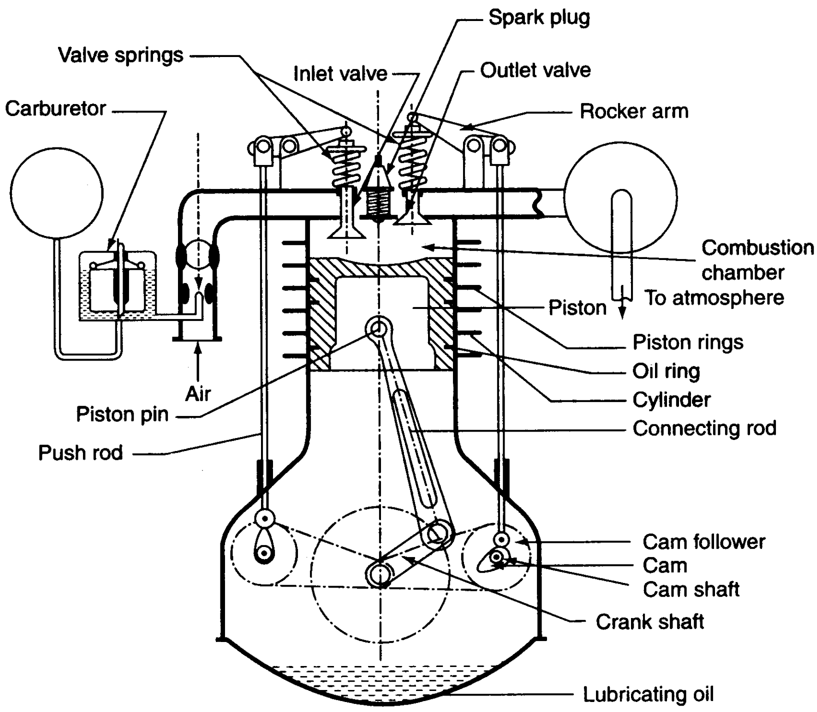



Fig. 20.3 Main components of a spark ignition engine

mixture of air and fuel inside the engine cylinder, Fig 20.4(a). During the suction process, the pressure inside the cylinder is lower than the ambient pressure by an amount that depends upon the speed of the engine.

***(b) Compression Process*** The fresh charge admitted during the suction process mixes with the residual gases present inside the cylinder and the mixture is compressed to a higher pressure when the piston describes the return stoke, i.e., moves from the bdc to tdc, while the valves, inlet and outlet, are closed. The pressure at the end of the compression process, Fig 20.4(b), depends upon the compression ratio, varying between 6 and 11, and its magnitude is about 0.6 to 0.9 MPa.

***(c) Ignition and Expansion Process*** Near the end of the compression stroke, there is an electric discharge across the spark plug, between 10 and 40 crank angle degrees before tdc, and that initiates the combustion process. The fuel burns and the combustion process is completed within a few milliseconds. It is assumed that the combustion process takes place at constant volume and the liberated heat energy rapidly increases the pressure and temperature of the working fluid present inside the cylinder. The products of combustion exert pressure at the piston crown and the piston is forced to descend downwards, from tdc to bdc, Fig. 20.4(c). During this expansion process, both the valves are closed.

***(d) Exhaust Process*** The exhaust valve opens near the end of the expansion process and the burned gases are forced out of the cylinder by the piston moving from bdc to tdc, Fig. 20.4(d). The exhaust valve communicates with the muffler or the silencer through which the burned gases are released to the atmosphere. During the exhaust process, the pressure inside the engine cylinder is a little more than the ambient pressure. The cycle is completed in four strokes of the piston or two revolutions of the crankshaft. The theoretical and actual pressure-volume diagram is shown in Fig. 20.5.

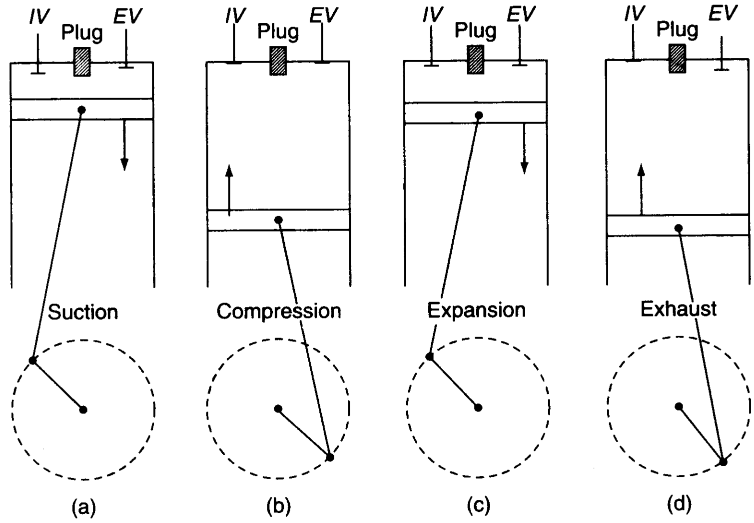



**Fig. 20.4** *Sequence of operation on a 4-stroke SI engine*

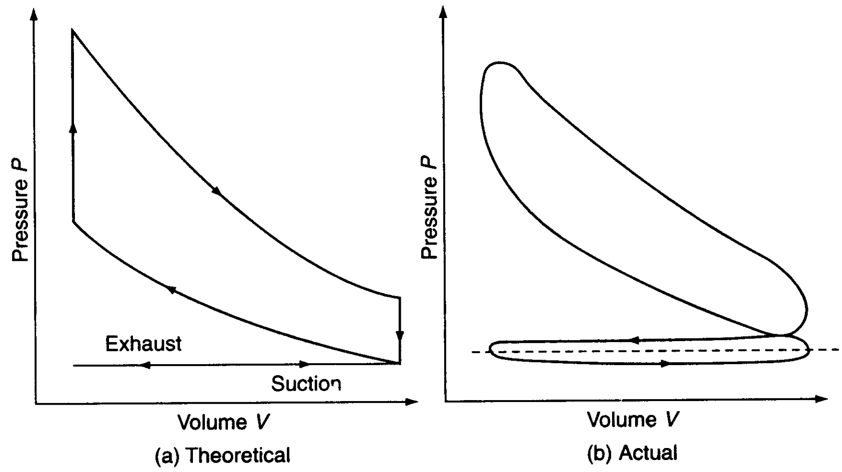
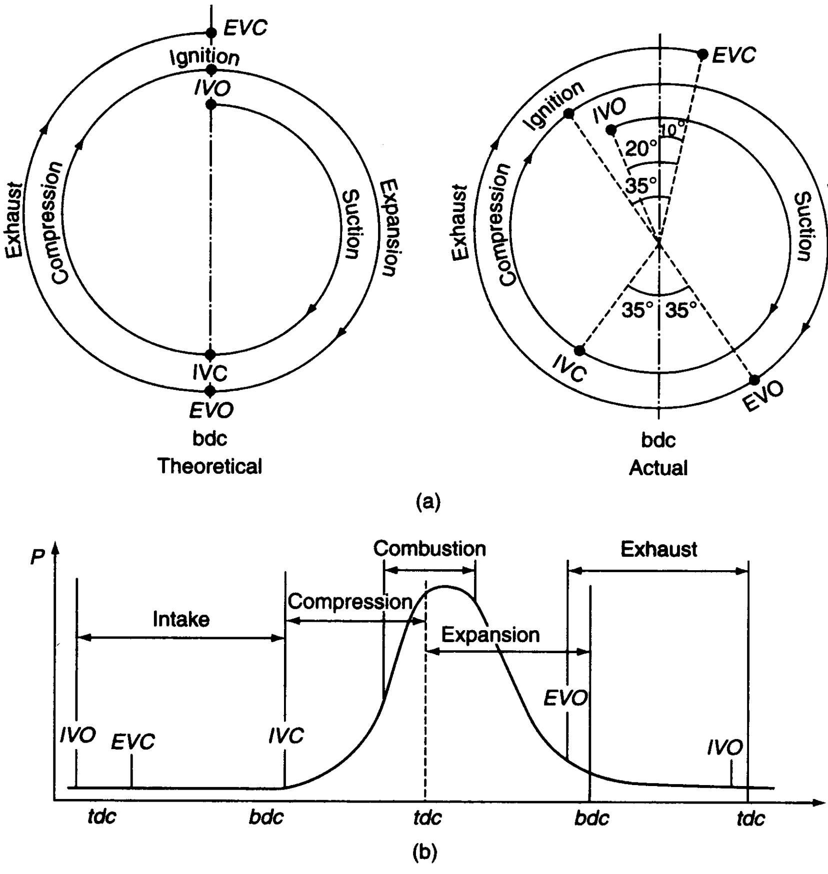



**Fig. 20.5** *Typical pressure-volume diagram for SI engine*

Otto, a German engineer, using the principles of Alphonse Beau de Rochas, described above, built a 4-stroke engine that became highly successful and is known as Otto Cycle.

The inlet and outlet valves are opened and closed by cam mechanism. The clearance between the cam, tappet and valve has to be taken up slowly and in order to avoid noise and wear, the valve is lifted slowly. Similarly the valve cannot be closed suddenly because it will bounce on its seat. Therefore, the opening and closing periods of valves are spread over a few crank angles. Figure 20.6(a) shows the valve-timing diagram for a 4-stroke SI engine.

Crank angle is also an independent and useful variable for analyzing the performance of an IC engine because the engine processes occupy almost 180 crank-angle intervals over a wide range of engine speeds. The sequence of events that take place inside the engine cylinder of 4-stroke SI engine is shown in Fig. 20.6(b). This figure also shows the valve timing.



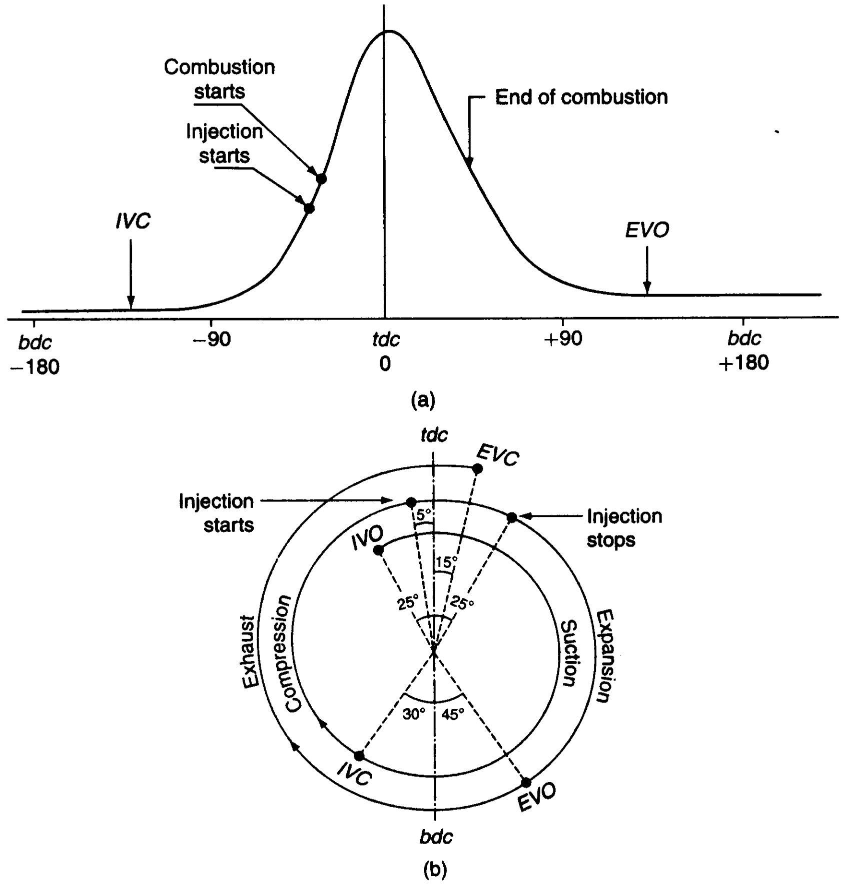

(a) *Valve timing diagram for a 4-stroke SI engine* (b) *Sequence of events in 4-stroke SI engine-pressure vs crank angle*

### 20.5.2 4-Stroke Compression Ignition Engine

In 1892, Dr Rudolf Diesel, a German engineer, designed and fabricated an engine capable of running at higher compression ratios and using cheaper fuels. The cycle named after him is known as 'Diesel cycle'. The different processes are:

**(*a*) *Suction Process*** Air alone is admitted inside the cylinder through the inlet valve while the piston moves from tdc to bdc. The inlet valve starts opening at about 30 before tdc so that it is fully open before the piston describes the suction process.

**(*b*) *Compression Process*** The piston compresses the air admitted earlier while moving from bdc to tdc and during this process both the valves should remain closed. The compression ratio varies from 12:1 to 24:1 and the air is compressed to a very high pressure, about 4 MPa, and as such the temperature at the end of the compression stroke is about 800 K, much higher than the self ignition temperature of the fuel. The fuel usually a light fuel oil, is injected directly into the engine cylinder nearly at the end of compression stroke. The injection starts at about 20 deg before tdc and continues during a part of the expansion process.




(a) *Sequence of events during compression, combustion and expansion in 4-stroke CI engine: pressure-crank angle* (b) *Valve timing diagram for a diesel engine*

***(c) Combustion and Expansion Process*** The liquid fuel enters the engine cylinder in the form of a fine spray and after entering the cylinder, it atomises, evaporates and mixes with heated air to form a combustible mixture. Since the temperature of air is much more than the self ignition temperature of the fuel, the fuel autoignites and the flame rapidly spreads through the combustion chamber. The products of combustion push the piston downwards from tdc to bdc, and the expansion stroke completed.

***(d) Exhaust Process*** The exhaust valve starts opening at about 30° before the bdc (before the completion of the expansion process) and a substantial part of the working fluid leaves the cylinder during the expansion process itself. During the exhaust process, the piston moves from the bdc to tdc and the gases are pushed out of the cylinder and the exhaust valve close at about 30 deg after tdc. The sequence of events that take place inside the engine cylinder of a 4-stroke CI engine is shown in Fig. 20.7.

The basic difference between the operation of the two types of engines lies in the method of injecting the fuel inside the engine and the system that initiates the combustion process and these lead to considerable variations in their operation. These are given in a tabular form, Table 20.1.

**Table 20.1**
*Comparison of SI and CI Engines*

| Criteria | SI Engine | CI engine |
|---|---|---|
| 1. Working cycle | Otto cycle | Diesel cycle |
| 2. Fuel used | Petrol/gasoline, should have a higher self ignition temperature for better efficiency | Diesel, should have lower self ignition temperature for better efficiency |
| 3. Suction process | A homogeneous air-fuel mixture prepared by the carburetor is sucked inside the cylinder | Only air is sucked inside, fuel is directly injected in the combustion chamber at high pressure by a fuel pump and injector. No carburetor |
| 4. Ignition & combustion process | An ignition system with a spark plug is required. Combustion takes place at constant volume | Fuel autoignites at high temperature inside the engine. Combustion at constant pressure, no spark plug required |
| 5. Compression ratio and engine weight | Lower (6 to 11) lighter engine, power output per unit weight high | Higher (12 to 24), heavier engine, power output per unit weight low |
| 6. Speed | Higher because of lighter engine | Lower because of heavier engine |
| 7. Efficiency | Lower because of lower compression ratio | Higher because of higher compression ratio |

## 20.6 THE TWO-STROKE ENGINE

A four-stroke engine requires two revolutions of crankshaft to complete the four basic processes-suction, compression, expansion and exhaust. Douglas Clerk, in 1878, developed a two-stroke engine to produce a greater output without changing the engine dimensions. He simplified the valve mechanism, reduced the proportion of the idle strokes and found that a 2-stroke engine can be applied to both SI and CI engines.

Let us consider the piston at the tdc, as shown in Fig. 20.8. For a SI engine, the clearance volume will be occupied by a homogeneous air-fuel mixture at a higher pressure and the spark plug will initiate the combustion process. For a CI engine, the clearance volume will be occupied by compressed air only and the fuel would be sprayed

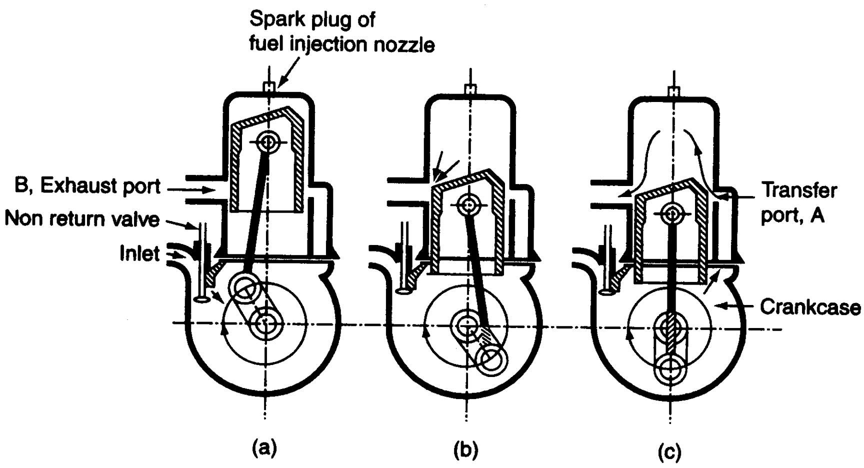



Fig. 20.8 Working of a 2-stroke engine

inside the cylinder. After combustion, the released heat energy will raise the pressure and temperature of the working fluid and the piston would be pushed downwards describing the power stroke. Near the end of the downward motion, the piston uncovers an opening/port, provided in the cylinder wall at B and most of the products of combustion would escape to the atmosphere through the exhaust manifold.

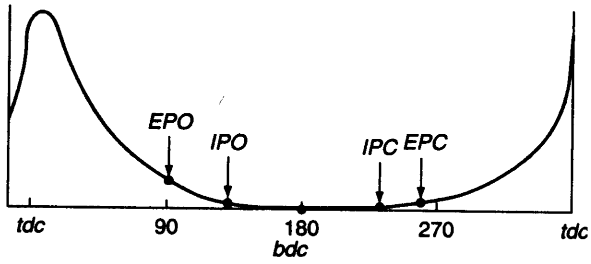



Fig. 20.9 *Sequence of operation in 2-stroke engine pressure—crank angle*

While the piston continues to descend downwards, it uncovers the transfer/inlet port A and either air (in CI engines) or air-fuel mixture (in SI engines) enters the engine cylinder. The entering fluid drives the exhaust gases out of the cylinder through the exhaust port B. This is an example of cross-scavenging. The deflector provided at the piston crown prevents the incoming charge from passing straight across the cylinder to the exhaust manifold.

During the upward movement of the piston (from bdc to tdc), the two ports A and B are covered and the charge admitted inside the cylinder is compressed whereas the crankcase sucks in the fresh charge through the reed spring inlet valve. This charge is compressed during the downward movement of the piston and is transferred to the main cylinder through the transfer port A during the last part of the downward stroke and in the beginning of the upward stroke. Thus, the cycle is completed in one revolution of the crankshaft or two-strokes of the piston. The sequence of operations that takes place inside the engine cylinder of a 2-stroke engine is shown in Fig. 20.9, and Fig. 20.10 shows the ideal and actual *p-v* diagrams for a two stroke engine along with the port opening diagram.

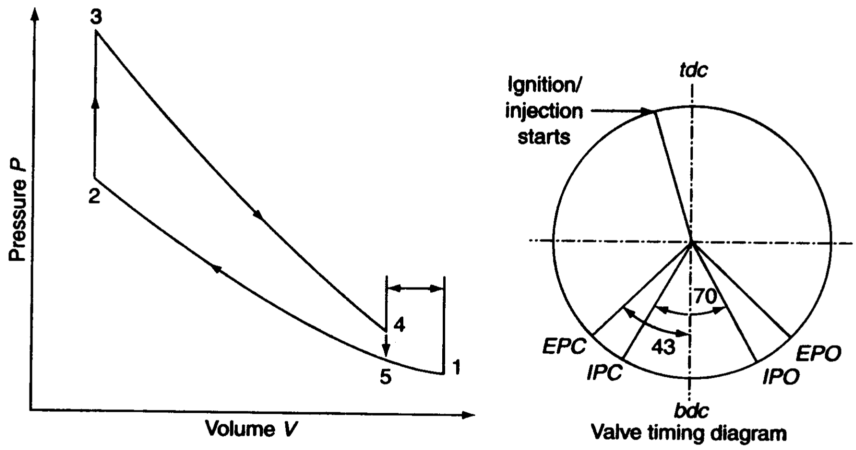



Fig. 20.10 *Pressure—volume and valve timing diagram for 2-stroke engine*

It is clear from the working of a 2-stroke engine that some freshly inducted working fluid is lost through the exhaust port during the scavenging process and therefore, the power output of a 2-stroke engine per unit displaced volume is less than twice the power output of an equivalent 4-stroke engine running at the same speed. The relative merits and demerits of 4-stroke and 2-stroke engines have been put in a tabular form in Table 20.2.

## 20.7 ENGINE PERFORMANCE

The power output of an engine is measured with the help of a brake or dynamometer, and is called '*brake power*' or '*shaft power*', which is given by

**Table 20.2**

*Comparison of 4-Stroke and 2-Stroke Engines*

| Criteria | Four-stroke engine | Two-stroke engine |
|---|---|---|
| 1. Thermodynamic cycle | Cycle is completed in 4-stroke or 2 revolutions of the crankshaft. One power stroke in 2 revolutions | Cycle is completed in 2-stroke or 1 revolution of the crankshaft. One power stroke in one revolution |
| 2. Thermal efficiency | Higher at normal and part loads | Lower at normal and part loads |
| 3. Volumetric efficiency | Higher because of more time for induction | Lower because less time for induction |
| 4. [illegible] | [illegible] | [illegible] |
| 5. Turning moment diagram | Less uniform because of one power stroke in 2 revolution, heavier fly-wheel is required | More uniform because of one power stroke in each revolution, lighter fly-wheel is required |
| 6. [illegible] | [illegible] | [illegible] |
| 7. Valve mechanism | Required, design gets complicated | Not required, design is simplified |
| 8. Initial cost | High | Low |
| 9. Use | Heavy duty vehicles where efficiency is more important | Light vehicles where compactness and weight are more important |
| 10. Effect on environment | Less | More polluting gases |

$$\text{BP (or bp)} = \frac{2\pi TN}{60} \qquad (20.1)$$

where $T$ is torque developed in Nm and N is the r.p.m. of the shaft.

The power output developed inside the engine cylinder is measured by obtaining the $p$–$v$ diagram with the help of an indicator and is called '*indicated power*' which is given by

$$ip \text{ or IP} = \frac{\text{IMEP} \times L \times A \times \left(N \text{ or } \dfrac{N}{2}\right)}{60}$$

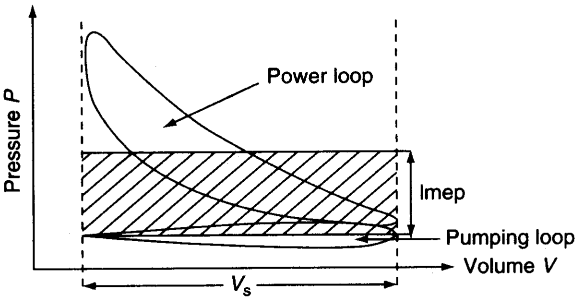



**Fig. 20.11** *Indicated mean effective pressure*

where IMEP (or imep) = *indicated mean effective pressure*, which is defined as the hypothetical constant pressure which when applied to each piston during the expansion stroke, would give the measured power (Fig. 20.11). Here, $A$ is the cross-sectional area of the cylinder, $L$ is the stroke length, $N$ is the rpm.

In the expression, for a 2-stroke engine $N$ cycles are completed and for a 4-stroke engine $\frac{N}{2}$ cycles are completed per minute.

The difference between the IP and BP is the power absorbed in mechanical friction, in driving the auxiliaries of the engine and the pumping power. This difference is called the *friction power* or

$$IP - BP = FP \tag{20.2}$$

The *mechanical efficiency* is defined as

$$\eta_{mech} = \frac{BP}{IP} \tag{20.3}$$

The brake power can also be expressed as

$$BP = \frac{BMEP \times L \times A \times \left(N \text{ or } N/2\right)}{60} \tag{20.4}$$

where BMEP is the *brake mean effective pressure.*

The BMEP may be considered as that mep acting on the pistons which would give the measured bp. (or BP) if the engine were frictionless.

The volumetric efficiency, $\eta_{vol}$ is a measure of the effectiveness of the suction and exhaust process of the engine. It is defined as the ratio of the volume of air sucked inside per cycle per cylinder at the ambient condition to the stroke volume of the cylinder, or

$$\eta_{vol} = \frac{\text{volume of air inducted at ambient condition}}{\text{stroke volume}}$$

$$= \frac{\text{mass of air inducted per cycle (or minute)}}{\text{mass of air occupying the stroke volume at ambient conditions}}$$

$$= \frac{\dot{m}_a \text{ kg/min}}{\rho_a \times V_s \times \left(N \text{ or } \dfrac{N}{2}\right)} \tag{20.5}$$

where $V_s$ is the stroke volume.

$$\dot{m}_a = \eta_{vol} \times \rho_a \times \frac{\pi}{4} D^2 L \left(N \text{ or } \frac{N}{2}\right)$$

The fuel burning role

$$\dot{m}_f = \dot{m}_a \times \frac{F}{A}$$

where F/A is the fuel-air ratio.

The thermal efficiency of an IC engine is expressed as the ratio of power output and rate of energy input in the form of fuel. It is defined on the basis of indicated or brake output. The *indicated thermal efficiency* is defined as

$$\eta_{i.th} = \frac{IP}{\dot{m}_f \times HV} \tag{20.6}$$

where HV is the heating value of the fuel. Similarly, the *brake thermal* efficiency is given by

$$\eta_{br.th} = \frac{BP}{\dot{m}_f \times HV} \tag{20.7}$$

Thus,

$$\eta_{mech} = \frac{BP}{IP} = \frac{BMEP}{IMEP} = \frac{\eta_{i.th}}{\eta_{br.th}} \tag{20.8}$$

From Eq. (20.6), $\quad I.P. = \dot{m}_f \times HV \times \eta_{i.th}$

or, $\quad B.P. = \eta_{vol} \times \rho_a \times \frac{\pi}{4} D^2 L n \times \left(N \text{ or } \frac{N}{2}\right) \times HV \times \eta_{i.th} \times \eta_{mech}$

or, $$\text{B.P.} = \eta_{vol} \times \eta_{mech} \times \eta_{i.th} \times \rho_a \times \frac{\pi}{4} D^2 L\, n \times \left( N \text{ or } \frac{N}{2} \right) \times \frac{F}{A} \times \text{HV} \tag{20.9}$$

where $n$ = number of cylinders.

Thus, $$\text{B.P.} \propto \rho_a$$

Power developed by the engine is directly proportional to the density of air. As density of air decreases, power output of the engine decreases. At high altitudes the air density is less, so the power output is also less. A supercharger is often used to supply compressed air to the engine to improve the engine output at high altitudes.

The power output of an IC engine depends greatly upon the amount of charge which can be induced into the cylinder. This is called the breathing capacity of the engine and is expressed by the *volumetric efficiency*. The volumetric efficiency with normal aspiration is seldom above 80% and to improve it, supercharging is used. The $\eta_{vol}$ depends on compression ratio, valve timing, induction and port design, air-fuel ratio, enthalpy of vaporization of fuel, cylinder temperature and the atmospheric conditions.

The specific fuel consumption (s.f.c) of an engine can be expressed either on indicated power or brake power basis. Indicated specific fuel consumption (isfc) is the fuel consumption per unit IP or

$$isfc = \dot{m}_f / \text{IP kg/kwh}$$

Similarly, $$bsfc = \dot{m}_f / \text{BP kg/kwh} \tag{20.10}$$

The *bsfc* is the mass-flow rate of fuel consumed per unit shaft output and is the criterion of economical power production.

The rate of work done by the gas on the piston is evaluated from the indicator diagram obtained from the engine (Fig. 20.11). The $i_{mep}$ is given by

$$imep = \frac{\text{net area of indicator diagram}}{\text{length of diameter}} \times \text{Constant}$$

where the constant depends on the scales of the recorder and net area = area of power loop-area of pumping loop.

Then $$\text{IP or } ip = \frac{imep \times LA(N/2 \text{ or } N)n}{60}$$

where $N$ = rpm and $n$ = number of cylinders.

Brake power (*bp*) is the output of the engine at the shaft measured by a dynamometer. Absorption dynamometers which are more common can be (a) friction-type like prony brake, (b) hydraulic, and (c) electrical, where

$$bp = \frac{2\pi TN}{60}$$

where $T$ is the torque measured.

## 20.8 AIR STANDARD CYCLES

Internal combustion engines are noncyclic heat engines. The working fluid, the fuel–air mixture, undergoes permanent chemical change due to combustion, and the products of combustion after doing work are thrown out of the engine, and a fresh charge is taken. So the working fluid does not undergo a complete thermodynamic cycle.

To simplify the analysis of IC engines, air standard cycles are conceived. In an air standard cycle, a certain mass of air operates in a complete thermodynamic cycle where heat is added and rejected with external heat reservoirs, and all the processes in the cycle are reversible. Air is assumed to behave as an ideal gas, and its specific heats are constant.

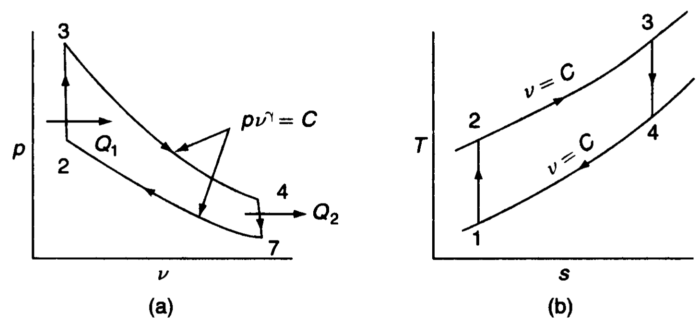



Fig. 20.12 Otto cycle

### 20.8.1 Otto Cycle

The Otto cycle is the air standard cycle of the SI engine, the operation of which has been explained in Fig. 20.5. It consists of two reversible constant volume processes in which heat is supplied and rejected and two reversible adiabatics in which the working fluid, air, is expanded and compressed (Fig. 20.12).

$$\eta_{\text{Otto}} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{mc_v(T_4 - T_1)}{mc_v(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

$$\text{Process } 1-2, \quad \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}, \quad \text{Process } 3\text{–}4, \quad \frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$

$$\therefore \quad \frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \text{or,} \quad \frac{T_3}{T_2} = \frac{T_4}{T_1}, \quad \text{or,} \quad \frac{T_4 - T_1}{T_3 - T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma-1}$$

$$\therefore \quad \eta_{\text{Otto}} = 1 - \left(\frac{v_2}{v_1}\right)^{\gamma-1} = 1 - \frac{1}{r_k^{\gamma-1}} \tag{20.11}$$

where $r_k = \dfrac{v_1}{v_2}$ is called the compression ratio.

The efficiency of the air standard Otto cycle is thus a function of the compression ratio only. The higher the compression ratio, the higher the efficiency. It is independent of the temperature levels at which the cycle operates. The compression ratio cannot, however, be increased beyond a certain limit, because of a noisy and destructive combustion phenomenon, known as detonation. It also depends upon the fuel, the engine design, and the operating conditions.

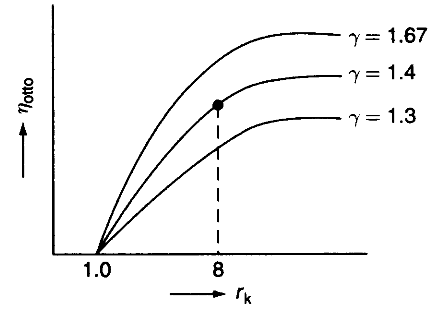



Fig. 20.13 Effect of $r_k$ and $\gamma$ on Otto cycle Efficiency

Figure 20.13 shows the effect of compression ratio and the specific heat ratio on the efficiency of Otto cycle. For an air standard cycle, air is the working fluid, $\gamma = 1.4$.

We can observe that the thermal efficiency curve is rather steep at low compression ratios but flattens our starting with a $r_k$ of about 8. Therefore, the increase in thermal efficiency with the compression ratio is not that pronounced at high compression ratios. Also, when high compression ratios are used, the temperature of the air fuel mixture rises above the self-ignition temperature of the fuel when the mixture ignites without the spark, causing an early and rapid combustion of the fuel ahead of the flame front, followed by almost instantaneous burning of the remaining mixture. This premature ignition of the fuel, called **autoignition**, produces an audible noise, which is called **engine knock or detonation.** This auto-ignition hurts performance and can cause engine damage, thus setting the upper limit of the compression ratio that can be used in SI engines.

Improvement of the thermal efficiency of gasoline engines by utilizing higher compression ratios (upto 12) without facing the auto-ignition problem has been made possible by using gasoline blends that have good antiknock characteristics, such as gasoline mixed with tetraethyl lead. Tetraethyl lead has been added to gasoline since the 1920s because it is the cheapest method of raising the **octane rating**, which is a measure of the engine knock resistance of a fuel. Leaded gasoline, however, has a very undesirable side effect as it forms compounds during the combustion process that are hazardous to health and pollute the environment. Most cars made since 1975 have been designed to use unleaded gasoline, and the compression ratios had to be lowered to avoid engine knock. As a result the thermal efficiency of car engines has somewhat decreased. However, owing to improvement in other areas like reduction in overall automobile weight, improved aerodynamic design etc, today's cars have better fuel economy.

The second parameter affecting the thermal efficiency of the Otto cycle is the specific heat ratio $\gamma$. For a given $r_k$, the use of a monatomic gas such as argon or helium as the working fluid yields the highest thermal efficiency (Fig. 20.13). The specific heat ratio $\gamma$ and thus the thermal efficiency of the Otto cycle decreases as the molecules of the working fluid get larger. At room temperature it is 1.4 for air, 1.3 for carbon dioxide and 1.2 for ethane. The working fluid in actual engines contains larger molecules such as $CO_2$, and $\gamma$ decreases with temperature, because of which the thermal efficiencies of the actual engines are lower than those of the Otto cycle and vary from 25 to 30 percent.

The ratio of brake thermal efficiency and air standard efficiency is called *relative efficiency.*

$$\eta_{rel} = \frac{\eta_{br.th}}{\eta_{air\ std.}}$$

### 20.8.2 Work Output

The net work output for an Otto cycle (Fig. 20.13) can be expressed

$$W_{net} = \frac{p_3V_3 - p_4V_4}{\gamma - 1} - \frac{p_2V_2 - p_1V_1}{\gamma - 1}$$

Now,

$$\frac{v_1}{v_2} = \frac{V_1}{V_2} = r_k, \quad \text{or} \quad V_1 = V_2 r_k = V_4$$

$$\frac{p_2}{p_1} = \frac{p_3}{p_4} = \left(\frac{V_1}{V_2}\right)^{\gamma} = r_k^{\gamma}$$

$$\frac{p_3}{p_2} = \frac{p_4}{p_1} = r_p \text{ (say)}$$

$$W_{net} = \frac{p_1V_1}{\gamma - 1}\left(\frac{p_3V_3}{p_1V_1} - \frac{p_4V_4}{p_1V_1} - \frac{p_2V_2}{p_1V_1} + 1\right)$$

$$= \frac{p_1V1}{\gamma - 1}\left(\frac{r_p r_k^{\gamma}}{r_k} - r_p - \frac{r_k^{\gamma}}{r_k} + 1\right) = \frac{p_1V_1}{\gamma - 1}\left(r_p r_k^{\gamma-1} - r_p - r_k^{\gamma-1} + 1\right)$$

$$W_{net} = \frac{p_1V_1}{\gamma - 1}(r_p - 1)\left(r_k^{\gamma-1} - 1\right) \tag{20.12}$$

### 20.8.3 Mean Effective Pressure

The mean effective pressure (m.e.p) of the cycle is given by

$$p_m = \frac{\text{Net work output}}{\text{Swept volume}}$$

where swept volume $= V_1 - V_2 = V_2 (r_k - 1)$

$$\therefore \quad p_m = \frac{\frac{p_1 V_1}{\gamma - 1}(r_p - 1)(r_k^{\gamma-1} - 1)}{V_2 (r_k - 1)}$$

$$= \frac{p_1 r_k (r_p - 1)(r_k^{\gamma-1} - 1)}{(\gamma - 1)(r_k - 1)} \quad (20.13)$$

Thus, it is seen that the net work output is directly proportional to the pressure ratio $r_p$. For given values of $r_k$ and $\gamma$, $p_m$ increases with $r_p$. For an Otto cycle, an increase in $r_k$ leads to an increase in $p_m$, $W_{net}$ and cycle efficiency.

### 20.8.4 Diesel Cycle

The limitation on compression ratio in the S.I. engine can be overcome by compressing air alone, instead of the fuel-air mixture, and then injecting the fuel into the cylinder in spray form when combustion is desired. The CI engine, first proposed by Rudolph Diesel in the 1890s, is very similar to the SI engine, differing mainly in the method of initiating combustion. In SI engines, a mixture of air fuel is compressed during compression stroke, and the compression ratios are limited by the onset of autoignition or engine knock. In CI engines, only air is compressed during the compression stroke. Therefore, diesel engines can operate at much higher compression ratios, typically between 12 and 24. The spark plug and carburettor (for mixing fuel and air) are replaced by a fuel injector in diesel engines. The temperature of air after compression must be high enough so that the fuel sprayed into the hot air burns spontaneously. The rate of burning can, to some extent, be controlled by the rate of injection of fuel. An engine operating in this way is called a *compression ignition (C.I.) engine.* The sequence of processes in the elementary operation of a C.I. engine, shown in Fig. 20.14 is as follows.

*Process* 1–2 *intake* The air valve is open. The piston moves out admitting air into the cylinder at constant pressure.

*Process* 2–3 *Compression* The air is then compressed by the piston to the minimum volume with all the valves closed.

*Process* 3–4 *Fuel injection and combustion* The fuel valve is open, fuel is sprayed into the hot air, and combustion takes place at constant pressure.

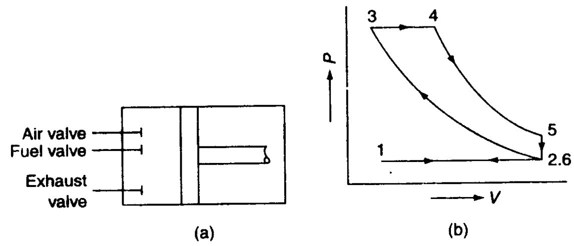



Fig. 20.14 (a) C.I. Engine (b) indicator diagram

*Process* 4–5 *Expansion* The combustion products expand, doing work on the piston which moves out to the maximum volume.

*Process* 5–6 *Blowdown* The exhaust valve opens, and the pressure drops to the initial pressure.

*Process* 6–1 *Exhaust* With the exhaust valve open, the piston moves towards the cylinder cover driving away the combustion products from the cylinder at constant pressure.

The above processes constitute an engine cycle, which is completed in four strokes of the piston or two revolutions of the crankshaft.

Figure 20.15 shows the air standard cycle called the Diesel cycle, corresponding to the CI engine as described above. The cycle is composed of two reversible adiabaties, one reversible isobar and one reversible isochore.

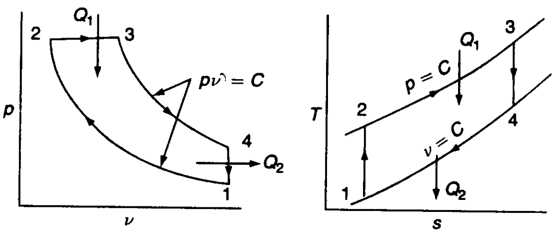



Fig. 20.15 Diesel cycle

Air is compressed reversibly and adiabatically in process 1–2. Heat is then added to it from an external source reversibly at constant pressure in process 2–3. Air then expands reversibly and adiabatically in process 3–4. Heat is then rejected reversibly at constant volume in process 4–1 and the cycle repeats itself.

For $m$ kg of air,

heat supplied $$Q_1 = Q_{2-3} = mc_p(T_3 - T_2)$$

heat rejected $$Q_2 = Q_{4-1} = mc_v(T_4 - T_1)$$

$$\therefore \qquad \eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{1}{\gamma}\cdot\frac{T_4 - T_1}{T_3 - T_2} \tag{20.14}$$

Compression ratio, $r_k = \frac{V_1}{V_2} = \frac{v_1}{v_2}$; Expansion ratio, $r_e = \frac{V_4}{V_3} = \frac{v_4}{v_3}$

Cut-off ratio, $$r_c = \frac{V_3}{V_2} - \frac{v_3}{v_2}$$

Now, $$r_k = r_e \cdot r_c$$

Also, $$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{\gamma-1} = \frac{1}{r_e^{\gamma-1}}; \qquad T_4 = T_3\frac{r_c^{\gamma-1}}{r_k^{\gamma-1}} \tag{20.15}$$

$$\frac{T_2}{T_3} = \frac{p_2v_2}{p_3v_3} = \frac{1}{r_c}; \qquad T_2 = \frac{T_3}{r_c} \tag{20.16}$$

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma-1} = \frac{1}{r_k^{\gamma-1}}; \qquad \therefore\ T_1 = \frac{T_3}{r_c}\cdot\frac{1}{r_k^{\gamma-1}} \tag{20.17}$$

$\therefore$ substituting $T_1$, $T_2$ and $T_4$ in Eq. (20.14)

$$\eta_{\text{Diesel}} = 1 - \frac{1}{\gamma}\cdot\frac{1}{r_k^{\gamma-1}}\cdot\frac{r_c^{\gamma} - 1}{r_c - 1} \tag{20.18}$$

As $r_c > 1$, $\eta_{\text{Diesel}} < 1 - \frac{1}{r_k^{\gamma-1}}$, i.e., $\eta_{\text{Diesel}} < \eta_{\text{Otto}}$ for the same compresion ratio, $r_k$.

## 20.9 FUELS FOR IC ENGINES

IC engine fuels are either in gaseous or liquid forms. *Gaseous fuels* can mix with air and distribute homogeneously to the various cylinders in a multicylinder engine. They can burn completely and leave relatively little combustion deposits compared with other fuels.

However, in vehicular engines, large containers are required for storing and carrying the fuel in the vehicles. The gases are thus compressed to high pressures and stored in thick containers. In stationary engines, they can be located close to the gas source. The gaseous fuels mostly used are natural gas (CNG), coke oven gas, blast furnace gas, refinery gas, producer gas and liquefied petroleum gas (LPG).

Liquid fuels are mostly derived from crude petroleum which consists of a large number of hydrocarbons having different molecular structures and boiling range. Generally, the compounds are classified into five groups — paraffins ($C_n H_{2n+2}$), olefins ($C_nH_{2n}$), diolefins ($C_nH_{2n-2}$), all being chain structure, hapthenes ($C_nH_{2n}$ ring), and aromatics ($C_nH_{2n-6}$ ring). No individual fuel belongs to a single group only, for example, petrol is generally a blend of paraffin, napthene and aromatic. A fuel tends to exhibit the characteristics of the hydrocarbon which forms a major part of it: The anti-knock quality of an SI engine fuel is worst in the paraffin group, improves gradually with the groups stated in the order above, and is best in the aromatic group. But with CI engines the suitability of fuels are in the reverse order, i.e., paraffin is the best and aromatic the worst. Boiling point temperature of a fuel depends on the number of atoms in the molecular structure and increases with the number of atoms. Fuels with fewer atoms in the molecule tend to be more volalile. The heating value of a fuel increases as the ratio of hydrogen to carbon atoms in the molecule increases. Therefore, paraffin has the greatest and aromatic has the least calorific value.

Crude petroleum, as such, is rarely used as fuel, and it is processed in a refinery to get the desirable products. Fractional distillation (separation of compounds depending on their boiling points) is used to separate crude petroleum, and fractions separated with increasing boiling points are in the order: petrol, naptha, kerosene, fuel and gas oil, lubricating oil, residue.

SI engines need liquid fuels with high volatility (vaporizing ability) for carburetion (mixing the fuel with air before admission to the engine), but CI engines fuels are from the fractions of kerosene, and fuel and gas oil range. CI engine fuels are less volatile, more viscous and heavier than petrol.

*Fuel-Air/Air-Fuel Ratio* The relative proportions of fuel and air in the engine are very important for combustion and efficiency of the engine. This is expressed either as a ratio of the mass of mass to that of air (F/A) or vice versa (A/F). A mixture that contains just enough air for complete combuation of the fuel in the mixture is called chemically correct or stoichiometric fuel–air ratio. A mixture having more fuel than this is a rich mixture and a mixture that contains less fuel is called a lean mixture. The ratio actual F/A ratio to stoichiometrice F/A ratio is called the ***equivalence radio***, $\phi$ so that

$$\phi = \frac{\text{Actual F/A ratio}}{\text{Stoichiometric F/A ratio}}$$

When $\phi > 1$ it is a rich mixture. If $\phi = 1$, it is chemically correct mixture.

## 20.9.1 SI Engine Fuels

The important qualities to be considered for a *SI* engine fuel are the following:

***(i) Volatility*** It is one of the most important properties of *SI* engine fuel and it influences the operation and maintenance of the engine. For ease in engine starting, good acceleration, proper distribution of air-fuel mixture in the cylinders, avoiding crankcase dilution (i.e. dilution of the lubricating oil in the crankcase by liquid petrol droplets) and washing of the cylinder lubricating oil, it is necessary to have high volatility, i.e. low boiling-off temperature of the fuel. Again, a high volatility can stop the flow of fuel to the engine by forming vapour locks in the fuel passages. Therefore, actual selection of a fuel is a compromise between requirements.

***(ii) Gum Deposits*** Some components (hydrocarbons) of a fuel have a tendency to form gum due to oxidation during storage. This gum causes undesirable deposits in the inlet valves, piston rings, carburetor and other engine parts. Therefore, for smooth operation of *SI* engines, the gum content and the tendency to form gum during storage of a fuel should be within reasonable limit.

***(iii) Sulphur Content*** Any sulphur present in the fuel tends to form corrosive compounds ($H_2SO_4$) which will attack the engine parts and damage them. Therefore, sulphur content in a fuel must be within permissible limit.

***(iv) Anti-knock Quality*** Under certain conditions, *SI* engine fuel (petrol) tends to cause engine knock due to sudden rise in pressure during combustion, accompanied by a hammer like sound (detonation). This kind of engine knock occurs towards the end of the pressure rise *after ignition*, and depends on the chemical composition and molecular structure of the fuel, and other factors such as engine load, speed, spark ignition timing, air-fuel ratio, temperature in the later stages of combustion (this temperature depends upon compression ratio and a given fuel will have increasing tendency to knock with increase in compression ratio).

The anti-knock quality of a fuel is expressed by a number (*Octane number*). The usual method used to determine the anti-knock quality of a *SI* engine fuel consists in testing the given fuel in a *CFR* (Co-operative Fuel Research) test engine with variable compression, and comparing it with a certain mixture whose knocking property is known. Iso-octane practically does not have any knocking tendency and it is arbitrarily assigned a rating of 100 octane number; normal heptane knocks worse than any known petrol and is given a rating of 0 octane number. These two can be mixed to match the knocking characteristic of any given fuel, and such a mixture is designated by the percentage of iso-octane (by volume) in the mixture. The octane number of a fuel is the percentage by volume of iso-octane in a mixture of iso-octane and normal heptane which produces the same knock intensity as the given fuel. Knocking (detonation) limits the power output and efficiency of *SI* engines. The higher the octane number of a fuel the greater will be its resistance to knock. Ordinary petrol can be made less likely to knock by adding small quantities of certain compound such as tetraethyl lead. However, now its use is discouraged due to severe environmental pollution.

## 20.9.2 CI Engine Fuels

Commonly used fuels in *CI* engines are high-speed diesel, light-diesel and heavy-diesel oils. The main qualities to be considered in the selection of *CI* engine fuels are:

***(i) Ignition*** It is a measure of the ability of the fuel to ignite promptly after it is injected into the cylinder. The ideal fuel should ignite immediately after injection, but in reality there is a certain lapse of time between the beginning of injection and ignition of the fuel. This time lag is known as *ignition lag* or *delay*, and if the ignition lag is large, the whole amount of the fuel may be injected before it ignites and when it ignites it burns suddenly with excessive pressure rise, producing a *combustion knock*. This combustion knock will be less with fuels having short ignition lag, because, the fuel will begin to burn soon after injection and hence there is less chance of the fuel to accumulate and burn suddenly.

The ignition quality or ignition lag of *CI* engine fuels is measured in terms of *Cetane number*. The ignition quality of a given fuel is compared in a *CFR* test engine with a reference fuel. Cetane $C_{16}H_{34}$ has very good ignition quality and is arbitrarily given a rating of 100 cetane number, and it is mixed with alphamethyl naphthalene $C_{11}H_{10}$ which has very poor ignition quality and assigned 0 cetane number. The percentage of cetane (by volume) in a mixture of cetane and alphamethyl naphthalene, which has the same ignition quality as the given fuel, is called the cetane number. Thus a fuel with cetane number 44 has the same ignition quality as a mixture of 44% of cetane and 56% of alphamethyl naphthalene. Adding certain compounds like amyl nitrate improves the ignition quality of a fuel.

***(ii) Volatility*** Though *CI* engine fuels are less volatile than *SI* engine fuels, it should be volatile enough to promote good mixing and combustion to ensure a clean exhaust.

***(iii) Viscosity*** *CI* engine fuels are more viscous than *SI* engine fuels, but they should be fluid enough to flow through the fuel system and filters.

***(iv) Impurities*** Sulphur and solid particle contents in the fuel should be within permissible limit to prevent formation of corrosive compounds, engine wear, and clogging of fuel injectors.

***(v) Flash point*** The flash point, i.e. the temperature at which the fuel gives off sufficient vapour to form a combustible mixture with air, should be sufficiently low to prevent fire hazard.

## 20.10 COMBUSTION IN SI ENGINES

Chemically correct or stoichiometric air-fuel ratio (*AFR*) is the ratio as obtained from the chemical reaction equation. The actual *AFR* during operation is, however, likely to be different from this ratio. *SI* engines generally operate on a limited range of *AFR* (8–20), beyond which the mixture is either too rich or too lean. There is no single *AFR* which is best for all conditions; for example the *AFR* for maximum power is not the same as the *AFR* for maximum economy (i.e. the power for which fuel consumption per kWh is minimum). The best power *AFR* is richer and the best economy *AFR* is leaner than the stoichiometric ratio (Fig. 20.16).

Combustion in *SI* engines may be of two types-normal combustion, and abnormal combustion. During *normal combustion* the spark plug starts the burning of the mixture and a flame-front spreads in all directions from the spark through compressed charge until the whole mixture (charge) is burned. The speed with which the flame spreads is called the rate of flame propagation or flame speed and it influences the combustion and knocking in the engine. The flame speed depends on the following factors:

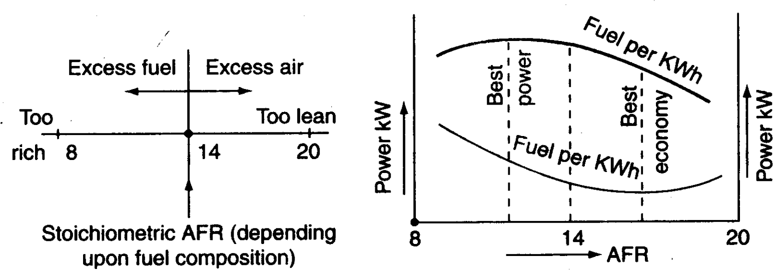
***(i) Turbulence*** Increased turbluence in the charge increases the flame speed due to increased mixing of the burning and unburned particles in the charge. Again, increase in engine speed increases turbulence.

***(ii) Air-fuel Ratio*** Highest flame speed occurs at an *AFR* somewhat richer than the stoichiometric *AFR*.

***(iii) Other Factors*** Pressure, temperature, and humidity of the intake charge, amount of residual gas (charge dilution), compression ratio, spark timing; the effect of these factors are, however, small.

The rate at which the pressure rises in the cylinder, during the combustion process, is of primary interest to both the designer and operator. This *rate of pressure rise* exerts considerable influence on the peak pressures encountered, the power produced, and the smoothness with which the forces are transmitted to the piston. The rate of pressure rise is dependent upon the *mass rate of combustion of the mixture* in the cylinder.

A convenient and generally used graphical presentation depicting the pressure rise in the cylinder is known as the **pressure-crank angle** or the **pressure-time** ($p$-$t$) diagram. Such a diagram is shown in Fig. 20.17. In addition to showing the rate of pressure rise, the $p$-$t$ diagram indicates, more clearly than the $p$-$V$ diagram, the events occurring near TDC.



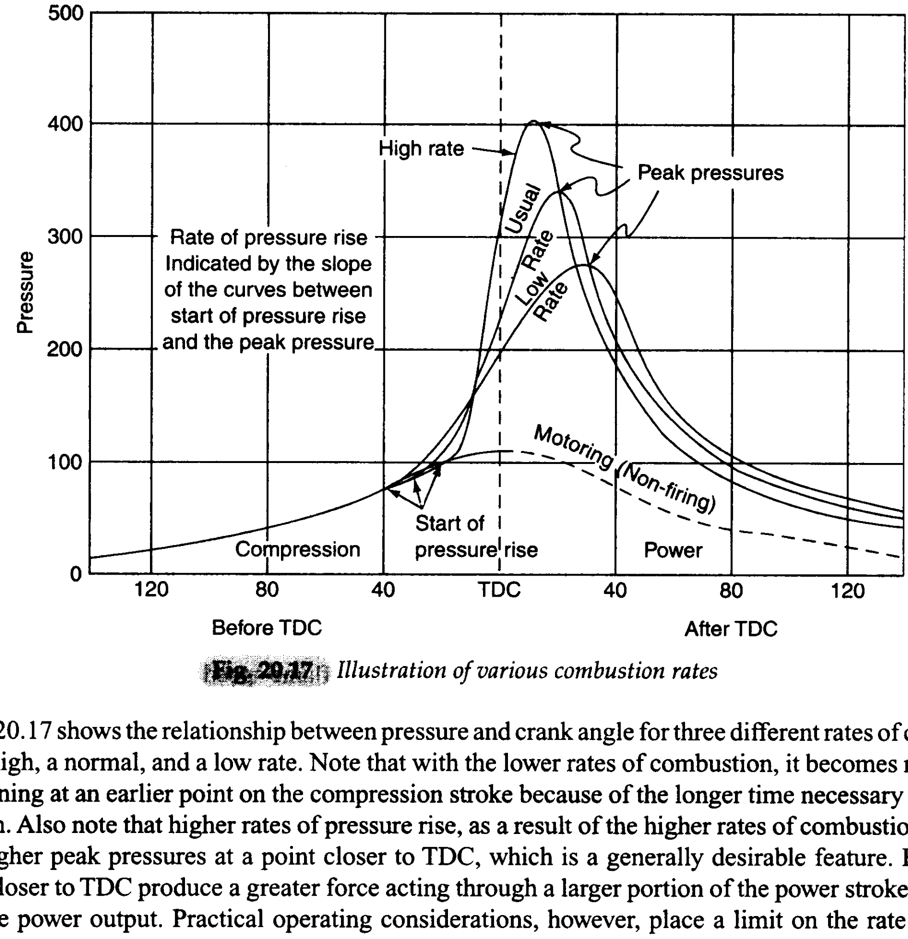



Fig. 20.17 Illustration of various combustion rates

Figure. 20.17 shows the relationship between pressure and crank angle for three different rates of combustion, namely, a high, a normal, and a low rate. Note that with the lower rates of combustion, it becomes necessary to initiate burning at an earlier point on the compression stroke because of the longer time necessary to complete combustion. Also note that higher rates of pressure rise, as a result of the higher rates of combustion, generally produce higher peak pressures at a point closer to TDC, which is a generally desirable feature. Higher peak pressures closer to TDC produce a greater force acting through a larger portion of the power stroke, and hence, increase the power output. Practical operating considerations, however, place a limit on the rate of pressure rise. If the rate is too high, the forces exerted on the piston tend to become "impactive," causing rough, or "jerky" operation. Also, if the peak pressures become excessive, they tend to create a situation conducive to an undesirable occurrence known as detonation. A compromise between these opposing factors is obviously necessary. This is accomplished by designing and operating the engine in such a manner that approximately one-half of the pressure rise has taken place as the piston reaches TDC. The result is a peak pressure reasonably close to the beginning of the power stroke, yet maintaining smooth engine operation.

## 20.10.1 Abnormal Combustion

The discussion thus far has been concerned with normal combustion. Under certain engine operating conditions, however, the air-fuel mixture has inherent characteristics conducive to various forms of abnormal combustion. The more important types of abnormal combustion, to be discussed in this article, are pre-ignition and detonation.

**(1) *Preignition*** If some portion of the boundary of the combustion chamber, such as a spark plug, exhaust valve, or carbon particle, becomes overheated under certain operating conditions, it is possible for this part to act in the same manner as the regular spark, and ignite the adjacent fresh combustible charge. An entirely distinct flame front is thus produced, and the process is termed **preignition**. Such a condition is undesirable since combustion becomes both erratic and uncontrollable. Moreover, preignition tends to raise the temperatures and pressures in the chamber which cause the temperature of the "hot spot" to rise further, and encourage still earlier preignition on succeeding cycles. The cumulative effect not only tends to raise peak pressures and encourage the possibility of detonation, but also tends to cause the peak pressures to occur progressively earlier in the cycle. In fact, preignition may advance these peak pressures to such a point that they occur before the piston reaches TDC on the compression stroke. In such a case, the peak pressure in those cylinders which are preigniting will oppose piston movement during the last part of the compression stroke, thus decreasing total output as well as causing rough engine operation. Preignition may also cause damage, through burning, to those engine parts which are subjected to the extreme temperatures.

***(2) Detonation*** A combustible mixture of fuel and air, under certain conditions of temperature, pressure, and density, has the faculty of igniting without the assistance of an initiating flame or spark. Such an event is known as auto-ignition. It is comparable to the more familiar layman's term of "spontaneous combustion."

In SI engines, the main "actor" in the auto-ignition phenomenon is *the last portion of the unburned charge* in the combustion chamber. As the normal flame front proceeds across the chamber, it raises the pressure and temperature of the remaining portion of the unburned charge. Under certain conditions of pressure, temperature, and density of the unburned charge, this charge may auto-ignite and burn almost instantaneously, thus releasing energy at a much greater rate than during the normal combustion process. The extremely rapid release of energy causes pressure differentials of considerable magnitude in the combustion chamber which give rise to radical vibrations of the gaseous products, producing an audible knock. This condition is known as detonation.

Detonation is a most important aspect in the operation of SI engines, since it is *the major factor limiting the compression ratio* of an engine. The violent pressure fluctuations accompanying detonation can cause severe damage to the engine, and *sustained detonation cannot be tolerated.* In the operation of automobile engines, sonic warnings are given in the form of an audible "pinging" sound, and operating conditions can be eased to prevent damage. In a SI aircraft engine, however, the engine noises override any detonation sounds and thus increase the inherent danger of detonation damage.

Although little is actually known concerning the processes which occur during detonation, it is desirable to follow through the events which are believed to take place.

It is known that, in order to auto-ignite, the last unburned portion of the charge must reach and remain for a definite amount of time above a certain critical temperature which is dependent upon conditions of pressure and density of the unburned charge. Once these conditions are reached, a "preparation" phase commences, followed by the "actual burning" phase. The "Preparation" phase is known as the ignition delay.

Figure 20.18(a) represents a normal flame front travelling across a combustion chamber from $A$ toward $D$, and increasing the pressure, temperature, and density of the unburned charge (area $BB'D$). If this unburned charge does not reach its critical temperature for auto-ignition, it will not auto-ignite, and the flame front $BB'$ will proceed on through the unburned charge to point $D$ in an orderly manner. The $p$-$t$ diagram for such normal combustion is illustrated in Fig. 20.18(b).

Refer now to Fig. 20.18(c) If the unburned charge (area $BB'D$) reaches and remains above its critical conditions for auto-ignition, there is a possibility of detonation. In essence, a "race" develops between the flame front and the ignition delay. If the flame front can proceed from $BB'$ to $D$ and consume the unburned charge in a normal manner, prior to completion of the ignition delay period, there will be no detonation. If, however, the flame front is able to proceed only as far as, say $CC'$, during the ignition delay period, then the remaining portion of the unburned charge (area $CC'D$) will detonate. Figure 20.18(d) is a $p$-$t$ diagram showing the extreme pressure fluctuations occurring during detonation. Note that *detonation occurs, in a SI engine, near the end of combustion.*

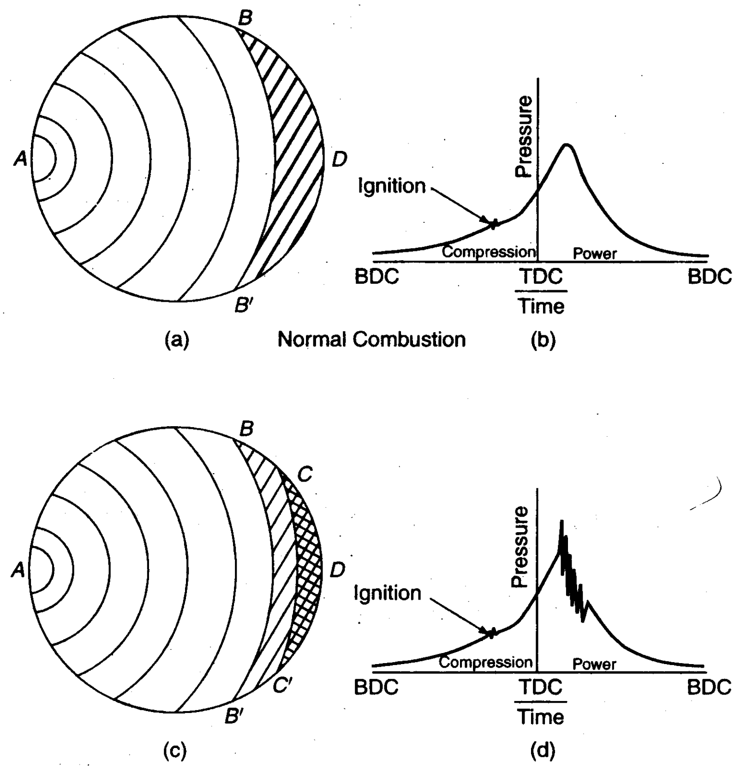



**Fig. 20.18** *Schematic presentation of the principles of normal and detonating combustion processes*

In summary, if the unburned charge does not reach its critical temperature, there can be no detonation. If the ignition delay period is longer than the time required for the flame front to burn through the unburned charge, there can be no detonation. But if the critical temperature is reached and maintained, and the ignition delay is shorter than the time it takes for the flame front to burn through the unburned charge, then the charge will detonate. Consequently, in order to inhibit detonation, *a high critical temperature for auto-ignition, and a long ignition delay, are desirable qualities in SI engine fuels.*

## 20.10.2 SI Engine combustion chamber

Figure 20.19 shows some combustion chambers with valve and spark plug locations. The main considerations in designing a good combustion chamber for SI engines are the following:

(i) *High volumetric efficiency* The incoming charge should enter with a high velocity with minimum pressure loss.

(ii) Combustion be completed near the TDC, which would give high pressure to the piston for high output

(iii) *Turbulence*-Correct degree of turbulence should be generated to promote rapid combustion

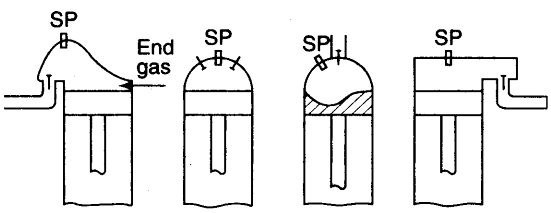



**Fig. 20.19** *SI engine combustion chambers*

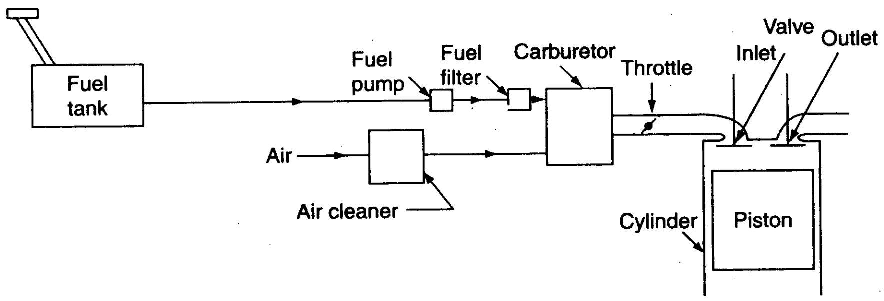



**Fig. 20.20** *Fuel and air induction system*

(iv) Satisfactory cooling of exhaust valve head
(v) Good scavenging of hot exhaust gases.

SI engines now having compression ratios of 9 or more use overhead valves as shown.

### 20.10.3 Ignition Systems in SI Engines

In SI engines the fuel system consists of a fuel supply tank, fuel pump, filters, lines, carburetor and intake pipe or manifold as shown in Fig. 20.20. The fuel pump may not be used if the fuel tank is placed at a higher level. The carburetor is the most important component; it prepares the correct air–fuel mixture and delivers this mixture to each cylinder through a ducting called intake manifold.

## 20.11 SIMPLE FLOAT CARBURETOR

Figure 20.21 shows a simple carburetor which consists of a venturi, a fuel nozzle with a metering orifice, a fuel tank with a float, a throttle valve and a choke. The fuel is pumped or flows by gravity to the float chamber. The fuel level is maintained at a constant level in the chamber with the help of a fuel supply valve controlled by the float. When air flows through the intake pipe, a vacuum is created at the venturi throat, which draws the fuel out through the discharge nozzle like a stream of fuel droplets, because of the pressure difference between the atmospheric pressure in the float chamber and the low pressure at the throat.

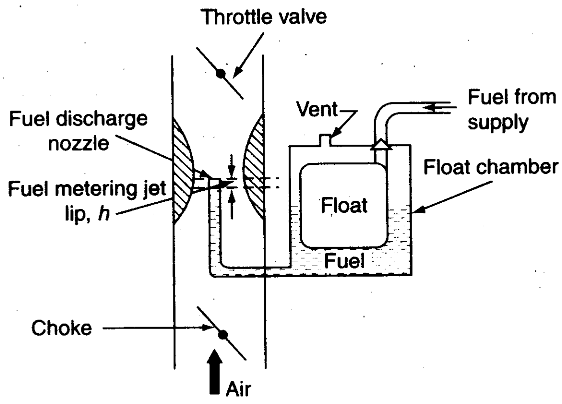



**Fig. 20.21** *Simple float-type carburetor*

As stated earlier, the carburetor must provide air–fuel mixture with appropriate air–fuel ratio for different operating conditions of the engine, which are (i) starting and warming, (ii) idling, (iii) part load, (iv) full load, and (v) acceleration.

During starting and warming up, the engine speed is low, the quantity of air flowing through the venturi is low and the vacuum created at the venturi throat is insufficient to draw fuel into the air stream. When the speed of the engine increases, the quantity of air flowing through the venturi increases and this creates a large vacuum at the throat. Therefore, a proportionately larger amount of fuel is sprayed into the air stream. Thus, a simple carburetor has a tendency to supply a rich mixture ($\phi > 1$) at higher speeds of the engine and a weak mixture ($\phi < 1$) at lower speeds.

During idling, the engine operates without any load and with a nearly closed throttle. As the throttle valve is almost closed, the air flow through the venturi is reduced, and it reduces also the fuel flow due to less pressure difference between the float chamber and the fuel discharge nozzle. The exhaust gas is drawn back into the cylinder along with a fraction of fresh charge when the throttle is closed. The overall mixture will contain a high percentage of exhaust gas, and in order to have combustion inside the engine, the carburetor must supply an increasingly rich mixture. There is an air bleed arrangement so that fuel can flow to the intake manifold when the throttle is almost closed (Fig. 20.22).

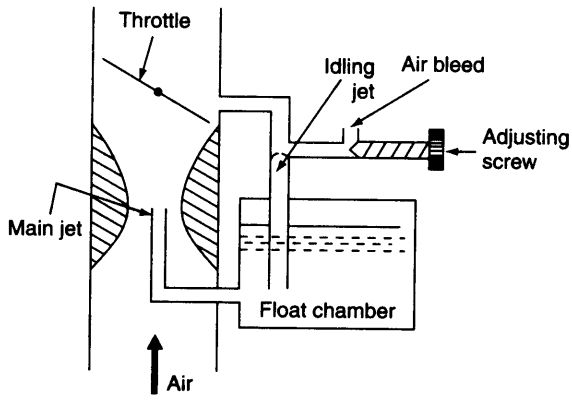



Fig. 20.22 *Idling jet in a carburetor*

When the engine is operating at part load with the throttle partly opened, maximum economy is the objective and therefore the air–fuel ratio for maximum efficiency should be selected, which would require a lean mixture by supplying extra air. When the throttle is fully opened, the carburetor must supply a rich mixture for maximum output with the supply of extra fuel.

In order to obtain a smooth and rapid acceleration of the engine, extra quantity of fuel is momentarily required to supply a richer mixture. An accelerating pump discharges a jet of fuel through a special jet in the carburetor. It is actuated by a lever from the throttle linkage.

When the engine is started under cold conditions by cranking, the speed of the engine is very low and a very rich mixture is required, which is done with the help of choke closing the air passage.

Thus a complete carburetor must have the following systems: (i) idling system, (ii) the fuel metering system, (iii) power/fuel enrichment system, (iv) accelerating system, and (v) choke.

### 20.11.1 S.I. Engine Fuel Injection

An engine running under varying operating conditions with a good exhaust control requires a complex carburetor, which is often replaced by a fuel injection system. The latter can be of two types, viz, single-point injection and multi-point injection. In a single-point injection system, a simple injector injects fuel just upstream of the throttle valve, serving all cylinders. It is difficult to achieve proper distribution of fuel to all the cylinders. In a multi-point injection system, the injectors are located immediately before the inlet valve accommodated in the cylinder head.

### 20.11.2 Flow Equations in a Carburetor

***(a) Air Flow*** By applying steady flow energy equation between section 1 (atmospheric condition) and venturi throat, section 2, we have

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

Since $V_2 << V_1$ and $h = c_p T$ for air,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2c_p(T_1 - T_2)} = \sqrt{2c_p T_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]},$$

assuming isentropic expansion.

The mass flow late of air $\dot{m}_a$ is obtained as

$$\dot{m} = \rho_2 A_2 V_2 = \frac{p_2}{RT_2} \cdot A_2 \times \sqrt{2c_p T_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}$$

***(b) Fuel Flow*** Since gasoline is incompressible, Bernoulli's equation is used between sections 1 and 2, which gives

$$\frac{p_1}{\rho_f g} = \frac{p_2}{\rho_f g} + \frac{V_2^2}{2g} + h$$

where $h$ is the lip.

and
$$V_f = \sqrt{2g \frac{p_1 - p_2}{\rho_f} - h}$$

where $\rho_f$ is the density of fuel.

The mass flow rate of fuel $\dot{m}_f$ is given by $\dot{m}_f = C_d \rho_f A_f V_f$

where $C_d$ is the coefficient of discharge of the fuel orifice.

### 20.11.3 Ignition System in SI Engines

The ignition system is an essential element in the operation of *SI* engines. The two commonly used systems are

(i) Battery ignition system
(ii) Magneto ignition system

Figure 20.23 shows the essential components of a battery ignition system for a four cylinder engine. A voltage of the order of 10000 to 20000 volts is required at the spark plug to overcome the resistance at the spark gap and to release sufficient energy to start the ignition of the charge. The ignition coil, which is essentially an induction coil, consists of two separate insulated windings (primary and secondary) on an iron core. The primary coil consists of a small number of thick wire while the secondary coil consists of a large number of thin wire. The primary coil is energized by a 6–12 V storage battery and the high voltage current is induced in the secondary coil at each break at the contact breaker which is operated by a special cam. The number of lobes in this cam is equal to the number of cylinders in the engine. The high voltage current from the secondary is led to the spark plugs in turn at correct times by the distributor. Both the cam and the distributor are driven by the engine cam shaft. The purpose of the condenser is to facilitate the collapse of the magnetic field and to prevent arcing (sparking) at the breaker points.

The magneto ignition system works on the same principle as the battery ignition system except that no battery is used; the magnetic field in the core of the primary and secondary coils is produced by a rotating permanent magnet. The magneto ignition system is light, compact and requires less maintenance; therefore, it is suitable for applications where storage batteries prove heavy and bulky, e.g. in aircraft and scooter engines. The disadvantage with magneto system is that at starting and at low speeds the voltage developed (which depends upon engine speed) is low.

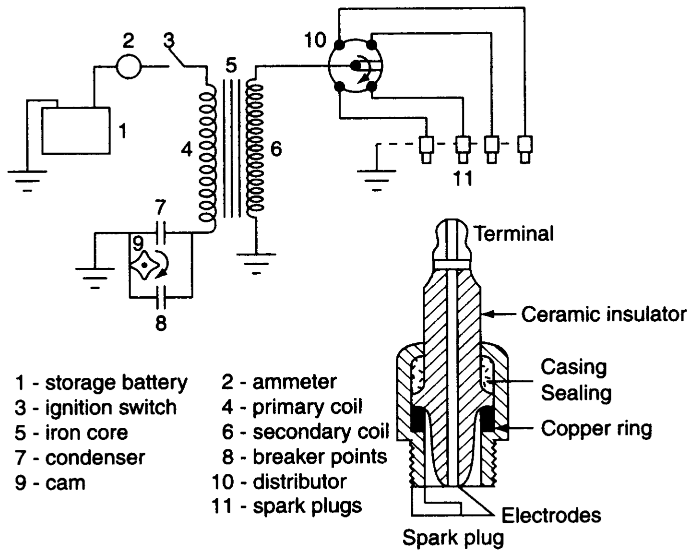



**Fig. 20.23** *Battery ignition system and spark plug*

## 20.12 FUEL INJECTION IN CI ENGINES

The fuel injection in CI engines consists of a fuel tank, filters, lines, fuel pump and fuel injector. Its function is to supply correct quantity of fuel and inject it at the correct time without after dribbling, atomize the fuel properly, and ensure that the fuel spray penetrates the desired areas of the combustion chambers. CI engine fuel injection system may be of two types: air injection and airless or solid injection.

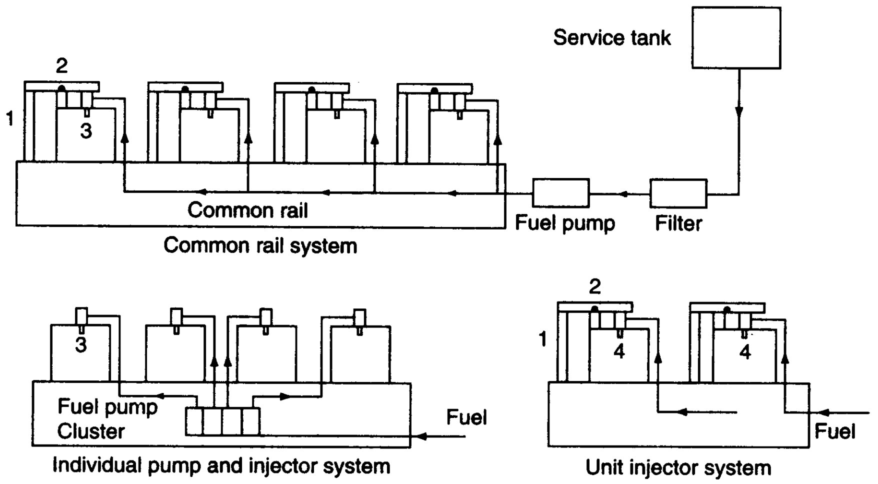
In air-injection, the fuel is injected into the cylinder by means of compressed air at about 7 MPa. Though this system was used in early years, it is seldom used now. Advantages of this system are good atomization and distribution, and possibility of using high viscosity, less expensive fuel. Disadvantages are complication of the engine with high pressure multi-stage air compressor which absorbs a part of engine power.

Now-a-days the airless or solid injection is used, and this system consists of two main parts-a high pressure fuel pump (15–30 MPa), and a fuel injector. Depending upon the arrangement of the fuel pumps and injectors, solid injection system may be classified as: *common rail system, unit injector system, individual pump and injector system* (Fig. 20.24). Of these, the latter two are mostly used.

In common rail system a single pump supplies fuel to a high pressure header or common rail from which the high pressure fuel goes to injectors in each cylinder. The injectors are operated by cams through push rods and rocker arms at correct sequence and time. In unit injector system the fuel pump and the injector or nozzle are combined into a single unit. Each cylinder is provided with one unit injector which is operated by cams as shown. In individual pump and nozzle system each cylinder has one pump and one nozzle or injector, but they are separated from each other. The injector is placed on the cylinder and the pump is placed on the side of the engine. The fuel pumps may be placed separately near each cylinder or in a cluster as shown.

### 20.12.1 Fuel Pump

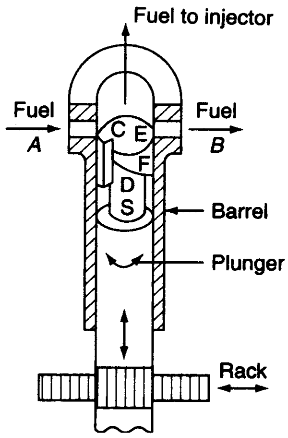
The fuel pump plunger is actuated by a cam and produces the required fuel pressure to open the injector valve at correct time. The amount of fuel injected depends upon the *effective stroke* of the plunger which is controlled by a rack. The fuel injector contains a delivery valve which is actuated by the fuel pressure. Figure 20.25 shows a schematic diagram of a fuel pump.




1 - Push rod; 2 - rocker arm; 3 - injector and fuel pump

*Fuel injection system*

The pump plunger has a constant stroke as determined by the cam lift. The plunger can be rotated inside the barrel with the rack to control the amount of fuel delivered by the pump to the injector. The vertical groove $C$ is connected to the helical groove $D$ and the space $S$ as shown. During upward stroke of the plunger it covers the ports $A$ and $B$, and fuel above the plunger and in the space $S$ is raised to high pressure. This high pressure fuel is delivered to the injector until the edge $F$ (on the helical edge) of the plunger uncovers the port $B$; at that instant the pressure developed is released and the fuel goes out through the overflow port $B$. The effective stroke of the pump is $EF$, because only during this movement of the plunger the fuel is delivered, although the plunger stroke may be more (constant and equal to cam lift). The effective stroke $EF$ of the pump can be changed with the rack through the fuel control lever, and hence the amount of fuel injected to match the engine load.




*Fuel pump*

### 20.12.2 Fuel Injector

Figure 20.26(a) shows a schematic diagram of a fuel injector. Injection of fuel begins when the valve $V$ is lifted from its seat against the spring force by the high pressure oil from the fuel pump acting on the annular area $A$.

Fuel injectors may be of single-orifice (20.26b), multi-orifice (20.26c) and pintle (20.26d) type. A single-orifice nozzle or injector gives good penetration but requires high injection pressure for fine atomization. A multi-orifice nozzle gives better atomization but get easily clogged. Pintle type nozzles can be designed to produce various spray-cone angles and are less likely to be clogged.

### 20.12.3 Combustion in CI Engines

In $SI$ engines a homogenous mixture of air and fuel (produced by the carburettor) within the combustible range maintains the combustion. In $CI$ engines the air-fuel mixture is not homogeneous and the $AFR$ in the various parts of the combustion chamber is different. As the combustion chamber contains compressed air at a temperature above the ignition temperature of the fuel, combustion occurs at many points within the chamber. As the fuel is injected, each minute droplet produced after atomization by the injector is enveloped by its own vapour, and after a small interval, combustion begins at the surface of this envelope. As stated in Section 20.5, the fuel is not injected all at once but continues over a number of degrees of crank angles (upto about 35 degree, depending upon speed and size of engine). The first droplets of fuel entering the cylinder come in contact with air whose temperature is only a little above the ignition temperature and hence the beginning of the burning process takes a little time. The droplets which enter later find the air already heated to a temperature much above the ignition temperature due to burning of the earlier droplets and begin to burn almost immediately as they enter the cylinder, but the last droplets to enter find some difficulty in burning as much of the oxygen in the air has been consumed. Therefore, to ensure proper combustion of the fuel, sufficient mixing of the fuel and air is necessary by dispersion of fuel and turbulence of the air.

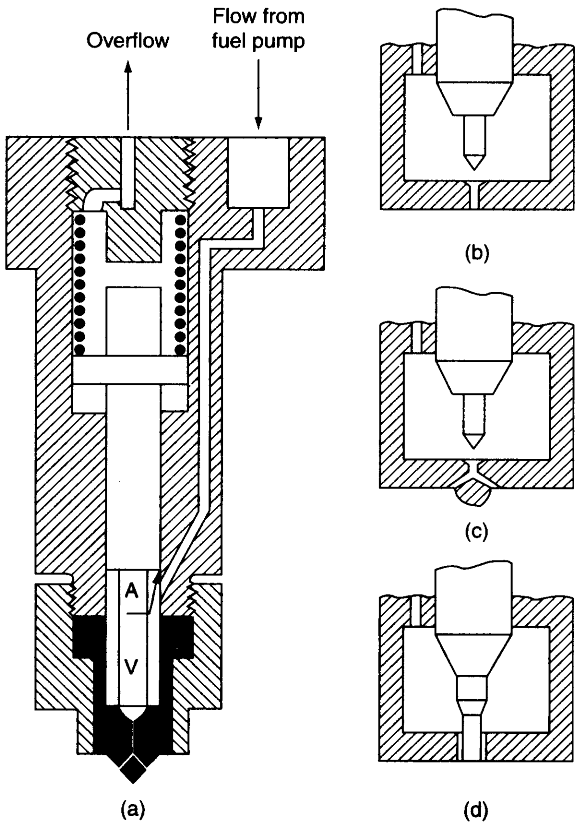


Fig. 20.26 Fuel injector

In *SI* engines, a too lean air-fuel mixture cannot support combustion, but in a *CI* engine burning can take place in a mixture which is infinitely lean. A *CI* engine begins to give out black exhaust (due to incomplete combustion) when the *AFR* is close to the stoichiometric ratio, because, in certain parts of the combustion chamber the *AFR* is too rich for complete combustion, and therefore, it is necessary to operate a *CI* engine with an *AFR* which is more (i.e. leaner) than the stoichiometric ratio.

As stated earlier, in *CI* engines the fuel does not ignite immediately on injection. The period between start of injection and start of combustion (ignition) is called *ignition delay* or *ignition lag,* and during this period there is no pressure rise in the cylinder due to combustion (Fig. 20.27).

Fuel injection in *CI* engines takes place over a certain time interval and as the initial droplets go through the delay period, additional droplets continue to enter the cylinder. If the ignition delay of the fuel is short, the first droplets will begin to burn soon after injection and a very small amount of fuel will accumulate in the combustion chamber.

This will give normal combustion with smooth pressure variation in the cylinder. But if the ignition delay of the fuel is long, a considerable part of the fuel injected will accumulate and when ignition begins it will suddenly burn, causing rapid pressure rise and pressure fluctuations. This will result in vibrations and audible knocks in a phenomenon similar to *SI* engine detonation (knocks); but, while in *SI* engines knocks due to detonation occur near the end of combustion, in *CI* engines knocks (called combustion or diesel knocks) due to detonation occur at the beginning of combustion.

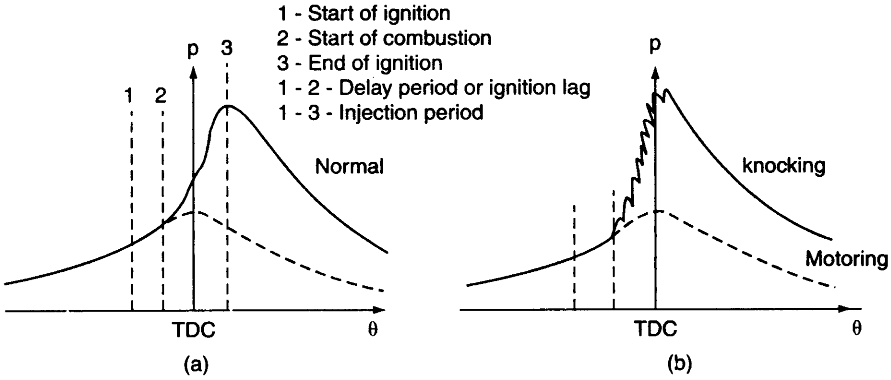



**Fig. 20.27** *Normal and abnormal combustion in CI engines*

From the above we see that ignition delay of a fuel plays an important part in smooth combustion in a *CI* engine. The ignition delay of a *CI* engine fuel may be reduced by adding certain additives to the fuel. Factors which influence ignition delay are:

***(i) Compression Ratio*** Increase in compression ratio reduces ignition lag; a higher pressure increases density resulting in closer contact of the molecules which reduces the time of reaction when fuel is injected.

***(ii) Inlet Air Temperature*** An increase in inlet temperature increases the air temperature after compression and hence decreases the ignition lag.

***(iii) Coolant Temperature*** Increase in coolant temperature increases cylinder air temperature and thus reduces ignition lag.

***(iv) Engine Speed*** Increase in speed increases turbulence and this reduces the ignition lag.

### 20.12.4 CI Engine Combustion Chamber

In *SI* engine the mixing of the fuel and air is done in the carburetor outside the cylinder, but in a *CI* engine this mixing is done in the combustion chamber. Therefore, in *CI* engines the combustion chamber must be designed to provide for this mixing of fuel and air. In general, combustion chambers in *CI* engines may be of two types-non-turbulent type, and turbulent type (Fig. 20.28).

In non-turbulent type combustion chambers the mixing of the fuel and air depends only to a small extent upon the turbulence created by the chamber. Mixing is mainly achieved by using orifice type nozzles working under high pressure. The fuel is highly atomized and spreads throughout the combustion chamber at high velocities and thereby promotes good mixing. In this type of combustion

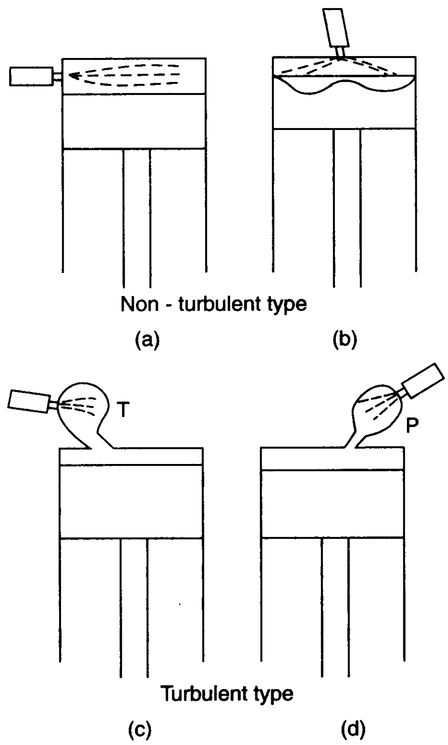



**Fig. 20.28** *CI Engine combustion chambers*

chamber (Fig. 20.28a and b), the surface to volume ratio is small and hence heat loss through the walls is also small, and consequently, starting is easier. But the use of orifice type nozzles with small openings increases the possibility for frequent clogging. Use of air swirl produced by designed cavities in the piston crown improves performance. This type of combustion chambers are suitable for low speed stationary engines.

In turbulent type, the combustion chamber produces the desired air-fuel mixing through turbulence (Fig. 20.28c and d). With turbulent type combustion chambers, pintle type nozzles working under lower pressure are used, and therefore, the possibility of clogging is less. Due to greater turbulence and larger surface to volume ratio, heat loss is more with consequent starting difficulties. This type of combustion chamber is suitable for variable speed operation due to greater turbulence, which enables ready response to any change in the fuel injection rate. Figure 20.28c shows a turbulent type combustion chamber with a turbulence chamber *T*. During compression, almost all the air is compressed into the turbulence chamber *T*. Figure 20.28d shows a chamber with pre-combustion chamber *P*; during compression about one-third of the air is forced into the small pre-combustion chamber. Combustion produces high pressure in *T* and *P*, which force the burning mixture out, and thereby producing high turbulence to promote good mixing and combustion.

## 20.13 COOLING SYSTEM OF IC ENGINES

The temperature of the gases inside the cylinder may be as high as 2750°C. If there is no external cooling, the cylinder walls and piston will tend to assume the average temperature of the gases which may be of the order of 1000° to 1500°C. The cooling of the engine is necessary for the following reasons.

(a) The lubricating oil used determines the maximum engine temperature that can be used. This temperature varies from 160°C to 200°C. Above these temperatures the lubricating oil deteriorates very rapidly and may evaporate and burn damaging the piston and cylinder surfaces. Piston seizure due to overheating may also occur.
(b) The strength of the materials used for various engine parts decreases with increase in temperature. Local thermal stresses can develop due to uneven expansion of various parts, often resulting in cracking.
(c) High engine temperatures may result in very hot exhaust valve, giving rise to pre-ignition and detonation or knocking.
(d) Due to high cylinder head temperature, the volumetric efficiency and hence power output of the engine are reduced.

Following are the two methods of cooling the engine.

(i) Air cooling
(ii) Water cooling

Air cooling is used in small engines, where fins are provided to increase heat transfer surface area.

Big diesel engines are always water cooled. The cylinder and its head are enclosed in a water jacket which is connected to a radiator. Water flowing in the jacket carries away the heat from the engine and becomes heated. The hot water then flows into the radiator and gets cooled by rejecting heat to air from the radiator walls. Cooled water is again circulated in the water jacket.

Various methods used for circulating the water around the cylinder are the following.

(a) *Thermosiphon cooling* In this method water flow is caused by density difference (Fig. 20.29). The rate of circulation is however slow and insufficient.
(b) *Forced cooling by pump* In this method a pump, taking power from the engine, forces water to circulate, ensuring engine cooling under all operating conditions. There may be overcooling which may cause low temperature corrosion of metal parts due to the presence of acids.

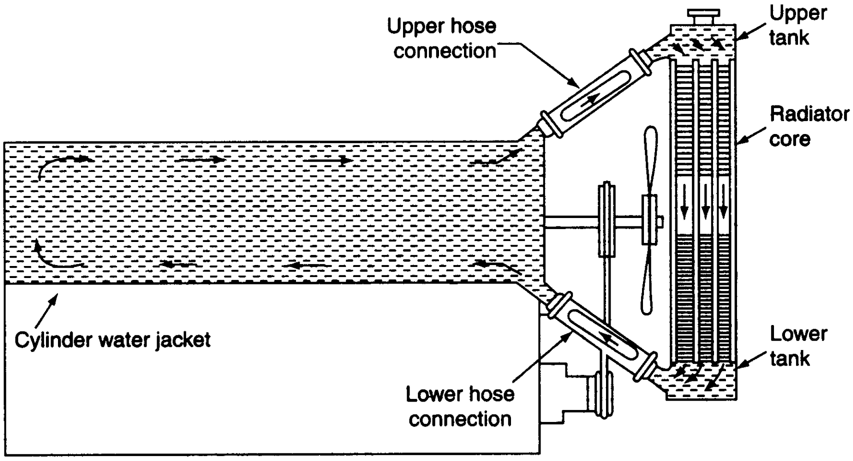



Fig. 20.29 *Thermosiphon cooling*

(c) *Thermostat cooling* This is a method in which a thermostat maintains the desired temperature and protects the engine from getting overcooled (Fig. 20.30).

(d) *Pressurized water cooling* In this method a higher water pressure, 1.5 to 2 bar, is maintained to increase heat transfer in the radiator. A pressure relief valve is provided against any pressure drop or vacuum.

(e) *Evaporative cooling* In this method water is allowed to evaporate absorbing the latent heat of evaporation from the cylinder walls. The cooling circuit is such that the coolant is always liquid and the steam flashes in a separate vessel (Fig. 20.31).

## 20.14 LUBRICATING SYSTEM

Lubrication is the flow of oil between two surfaces having relative motion. The following are the important functions of a lubricating system.

(a) *Lubrication* To keep moving parts sliding freely past each other, thus reducing engine friction and wear.

(b) *Cooling* To keep the surfaces cool by taking away a part of the heat caused by friction.

(c) *Cleaning* To keep the bearings and piston rings clean of the products of wear as well as of combustion by washing them away.

(d) *Sealing* To form a good seal between the piston rings and cylinder walls.

(e) *Reducing noise* To reduce the noise of the engine by absorbing vibration.

Various lubrication systems used for IC engines may be classified in the following manner.

1. Mist lubrication system
2. Wet sump lubricating system
3. Dry sump lubricating system

**1. *Mist Lubrication System*** This system is used for two-stroke cycle engines which employ crankcase compression. Crankcase lubrication is thus not suitable in these engines.

**2. *Wet Sump Lubricating System*** The bottom part of the crankcase, called sump, contains the lubricating oil which is pumped to various parts of the engine. There can be three types of wet sump lubrication system.

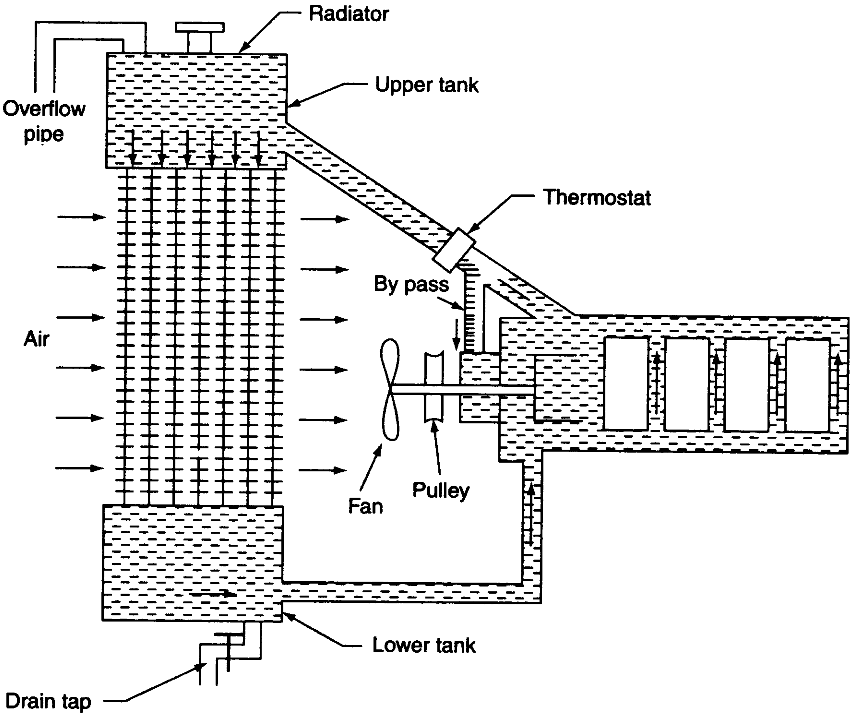



**Fig. 20.30** *Thermostat cooling*

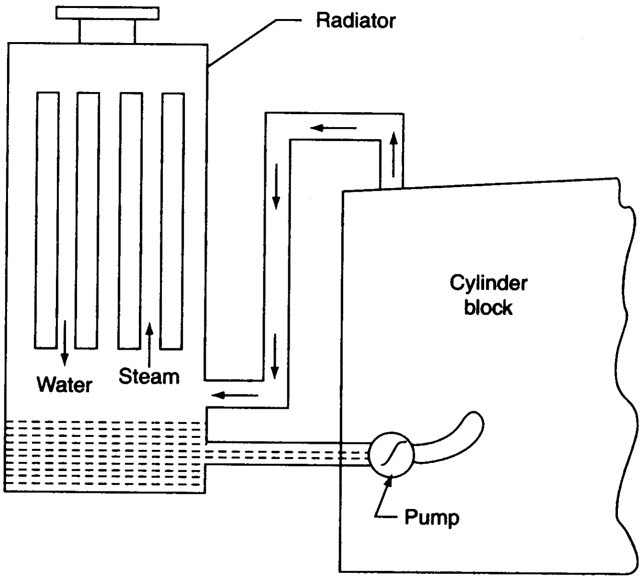



**Fig. 20.31** *Evaporative cooling*

*Splash System* It is used for small four-stroke stationary engines (Fig. 20.32). The oil level in the sump is maintained in a way that when the connecting rod's big end is at its lowest position the drippers at that end strike the oil in the troughs which are supplied with oil from the sump by an oil pump. Due to striking of the drippers, oil splashes over various engine parts like crankpin bearings, piston rings, piston pins, etc. Excess oil drips back to the sump.

*Modified Splash System* The splash system is not sufficient if the bearing loads are high. For such cases, the modified splash system is used, where the main and camshaft bearings are lubricated by oil under pressure pumped by an oil pump. The other engine parts are lubricated by splash as shown in Fig. 20.32.

*Full Pressure System* An oil pump is used to lubricate all parts of the engine (Fig. 20.33). Oil is pumped to the main bearings of crankshaft and camshaft at pressures varying between 1.5 and 4 bar. Drilled passages are used to lubricate connecting rod end bearings.

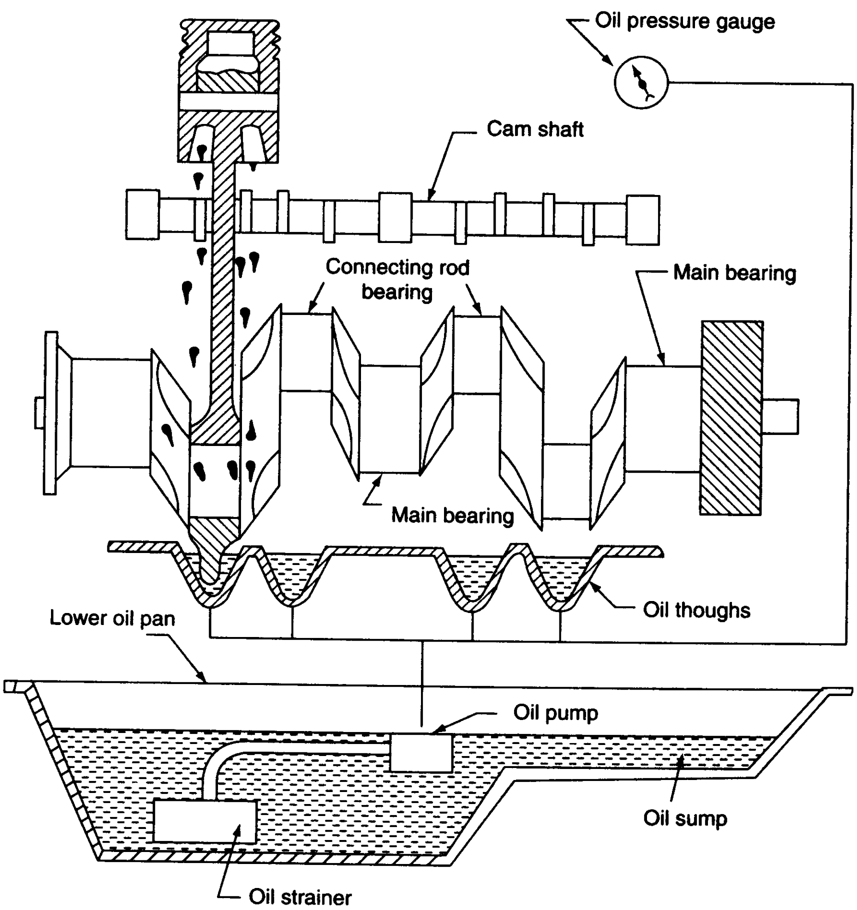



**Fig. 20.32** *Splash lubrication system*

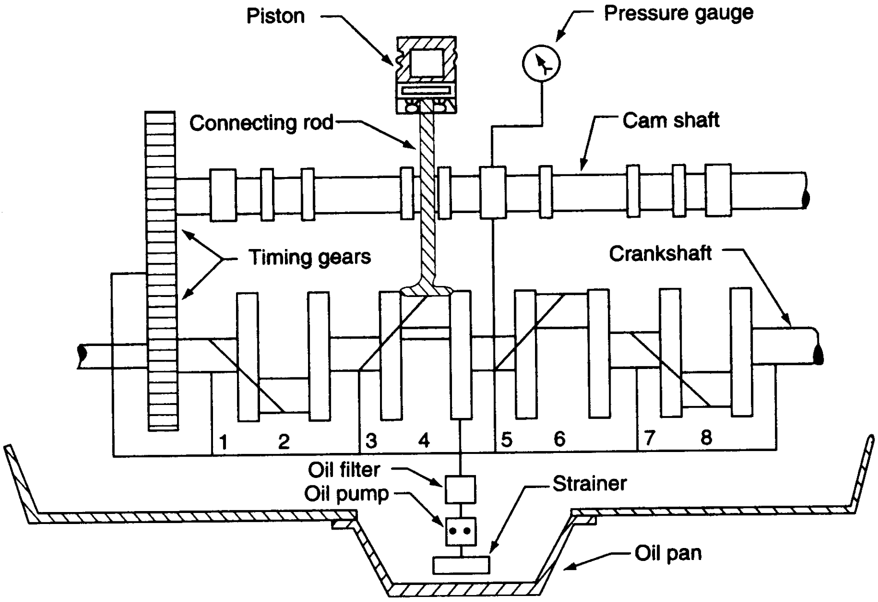



Fig. 20.33 *Full pressure lubrication system*

A gear pump (Fig. 20.34) submerged in the oil and driven by the camshaft draws oil from the sump through a strainer. A pressure relief valve is provided on the delivery side to prevent excessive pressure.

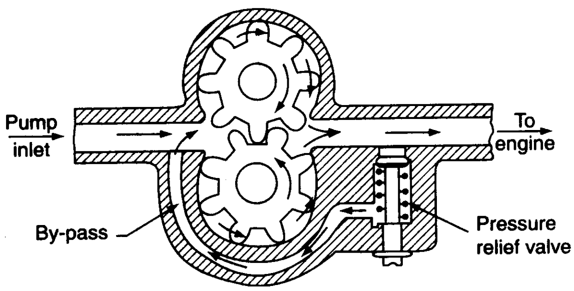



Fig. 20.34 *Gear oil pump with relief valve*

**3. *Dry Sump Lubrication System*** Oil from the sump is carried to a separate storage tank outside the engine cylinder. The oil from the sump is pumped through filters to the storage tank (Fig. 20.35). Oil from the storage tank is pumped to the engine cylinder through oil cooler. Oil pressure varies from 3 to 8 bar. Dry sump system is generally used for high capacity engines.

From the pump all the oil used for lubrication usually passes through an oil filter before it reaches the engine bearings. The bearings are machined to a very close tolerance and are likely to be damaged if any foreign material is allowed to enter the lubrication line. Filter arrangement may be of the following two types.

(i) Full flow type, where all the oil is filtered before it is fed to the bearings, as in dry sump lubrication (Fig. 20.35).

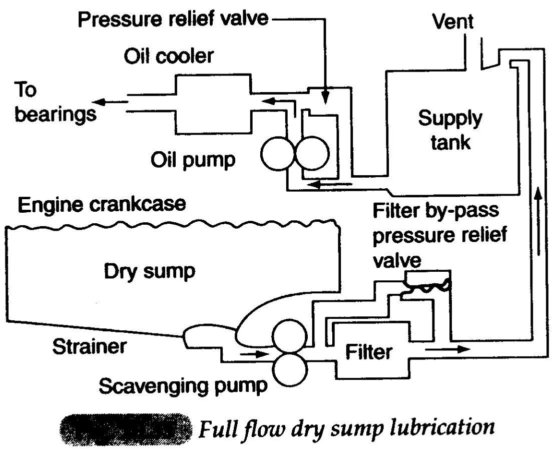
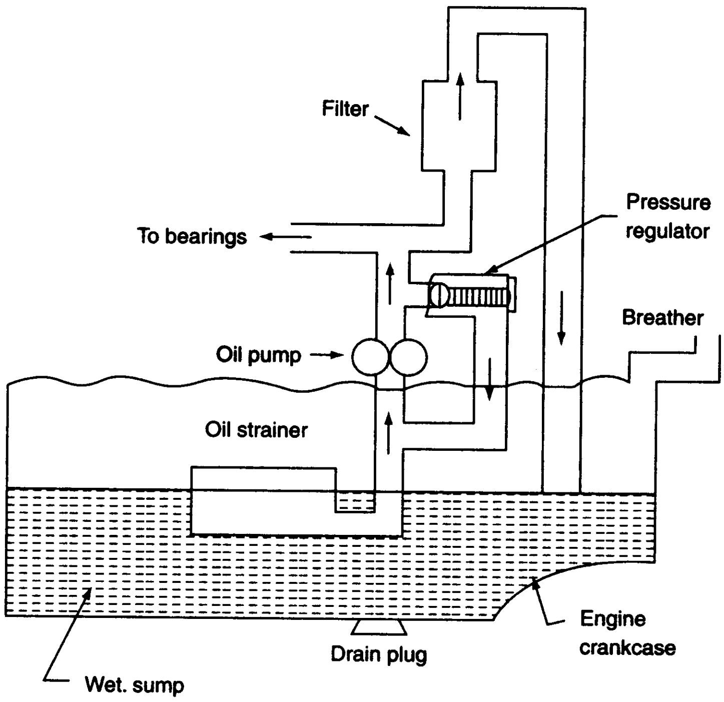



Fig. 20.35 *Full flow dry sump lubrication*




*By-pass type wet sump lubrication*

(ii) By-pass type, where only a small part of the oil is passed through the filter and the rest is directly supplied to the bearings, as in wet sump lubrication system (Fig. 20.36).

## 20.15 STARTING OF ENGINE

Following are the three common methods of starting an engine.

(i) By an auxiliary engine, which is mounted close to the main engine and drives the latter through a clutch and gears.

(ii) By using an electric motor, in which a storage battery of 12 to 36 volts is used to supply power to an electric motor that derives the engine.

(iii) By compressed air system, in which compressed air at about 17 bar supplied from an air tank is admitted to a few engine cylinders making them work like reciprocating air motors to run the engine shaft. Fuel is admitted to the remaining cylinders and ignited in the normal way causing the engine to start. The compressed air system is commonly used for starting large diesel engines employed for stationary power plant service.

## 20.16 PERFORMANCE CHARACTERISTICS

It was stated in Art. 20.7 that

$$fp = ip - bp \quad \text{and} \quad \eta_M = bp/ip.$$

The *fp* is very nearly constant at a given engine speed. If the load is decreased giving lower values of *bp*, then the variation in $\eta_M$ with *bp* is as shown in Fig. 20.37.

The Morse test can be used to measure the *ip* of multi-cylinder engines. The engine, say having four cylinders, is run at the required speed and the torque is measured. One cylinder is cut out by disconnecting the injector of a CI engine (or by shorting the spark plug of an SI engine). The speed falls because of the loss of power with one cylinder cut out, but is restored by reducing the load. When the speed has reached the original value, the torque is again measured. It is repeated by cutting out other cylinders one by one. If the values of *ip* of the cylinders are denoted by $I_1$, $I_2$, $I_3$ and $I_4$ and the power losses in each cylinder are denoted by $L_1$, $L_2$, $L_3$ and $L_4$, then the value of *bp*, *B*, at the test speed with all cylinders firing is given by

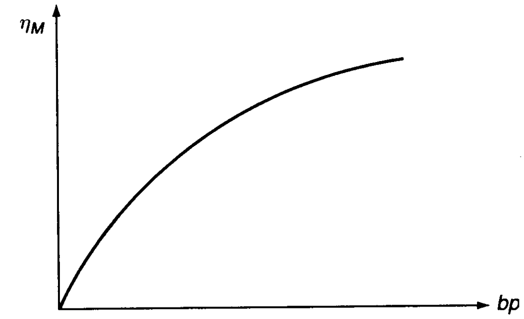


Fig. 20.37 Variation of mechanical efficiency

$$B = (I_1 - L_1) + (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4) \tag{20.19}$$

If number 1 cylinder is cut out, then the contribution $I_1$ is lost. If the losses due to that cylinder remain the same as when it was firing, then the *bp*, $B_1$, obtained at the same speed is

$$B_1 = (0 - L_1) + (I_2 - L_2) + (I_3 - L_3) + (I_4 - L_4) \tag{20.20}$$

By subtracting Eq. (20.20) from Eq. (20.19), we get

$$B - B_1 = I_1$$

By cutting out each cylinder in turn the values of $I_2$, $I_3$ and $I_4$, can be similarly obtained. Therefore, the total indicated power of the engine is

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

At a constant engine speed if the load is decreased in steps and the corresponding *bp* and fuel consumption are measured and plotted, a graph called the "Willan's Line" is obtained as shown in Fig. 20.38.

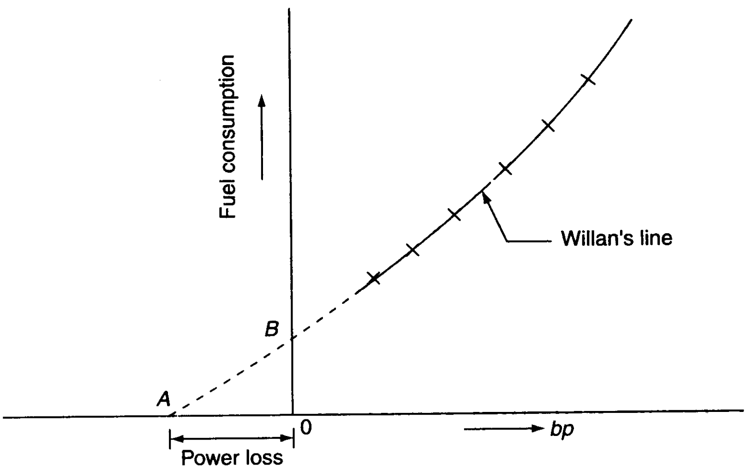


Fig. 20.38 Fuel consumption against brake power for a CI engine

The extrapolated reading $OA$ is the power loss of the engine at that speed and the fuel consumption $OB$ at zero $bp$ is equivalent to the power loss $OA$.

The $bp$ of an engine can be obtained accurately and conveniently using a dynamometer, which is

$$bp = \eta_M \times ip = \eta_M \times \frac{P_i AL \frac{N}{2} n}{60}$$

Since $\eta_M$ and $p_i$ are difficult to obtain they may be combined and replaced by a brake mean effective pressure (bmep), $p_b$, i.e.

$$bp = \frac{p_b AL \frac{N}{2} n}{60} \tag{20.21}$$

where $p_b = \eta_M \times p_i$

The bmep may be considered as that mep acting on the pistons which would give the measured $bp$ if the engine were frictionless ($fp = 0$). Again,

$$\frac{p_b AL \frac{N}{2} n}{60} = \frac{2\pi TN}{60}$$

$$p_b = KT$$

where $K$ is a constant.

Therefore, bmep is directly proportional to the engine torque and is independent of the engine speed.

***Energy balance*** Energy supplied by the fuel

$\dot{m}_f \times CV = bp +$ the heat transferred to cooling water ($Q_c$) + the energy of the exhaust referred to inlet condition ($Q_{ex}$) + unaccounted energy losses which include losses by radiation and convection.

Energy to exhaust is $\quad (Q_{ex}) = (\dot{m}_a + \dot{m}_f) h_e - \dot{m}_a h_a$

where $\dot{m}_a$ and $\dot{m}_f$ are the air and fuel mass flow rates, $h_e$ is the enthalpy of the exhaust gas (dry exhaust + steam), reckoned from 0°C, and $h_a$ is the enthalpy of air at inlet, reckoned from 0°C.

For a diesel engine at full load, typical values are: to $bp$ 35%, to cooling water 20%, to exhaust 35%, and to radiation, etc. 10%. The heat to the jacket cooling water and exhaust can be utilized in industries which have heating loads as space heating and hot water systems.

Engine characteristics of power, imep, bmep and mechanical efficiency against engine speed are shown in Fig. 20.39. The difference between the $ip$ and the $bp$ at any speed is the $fp$ which increases with speed. Both $ip$ and $bp$ curves show maximum values, but they occur at different speeds. After the maximum both $ip$ and $bp$ fall because of the reduction in volumetric efficiency with speed (Fig. 20.40).

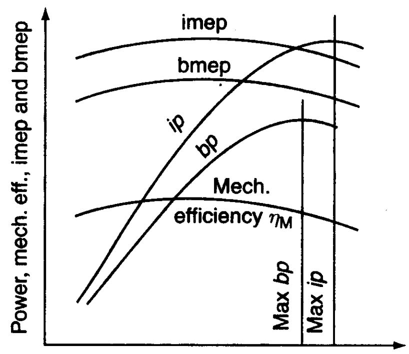



**Fig. 20.39** *Engine characteristics of power, imep, bmep, and mechanical efficiency against engine speed*

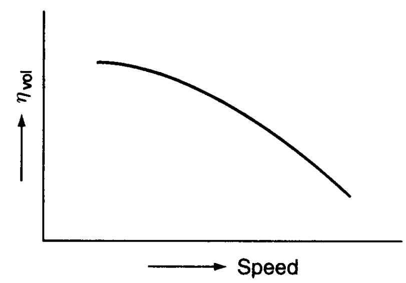
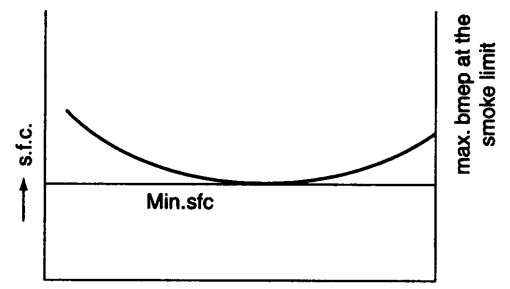



**Fig. 20.40** *Variation of volumetric efficiency with speed*

The limiting condition of fuel supply to a CI engine is the smoke limit when black smoke appears in the exhaust. It occurs at air-fuel ratios of about 16/1. Figure 20.41 shows sfc varying with bmep. A minimum sfc and hence a maximum brake thermal efficiency occur at part load (i.e. less than the maximum bmep).




*Specific fuel consumption against bmep for a CI engine*

## 20.17 SUPERCHARGING

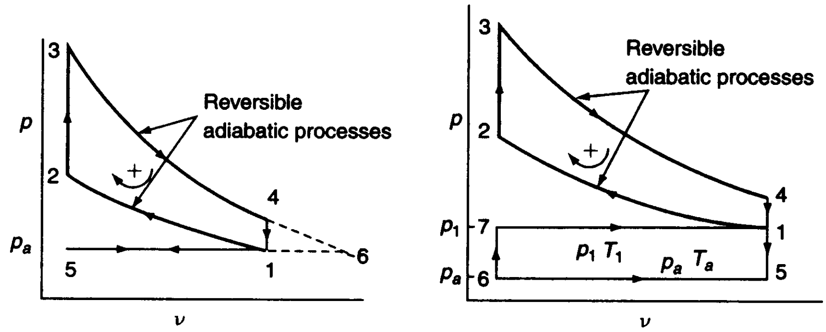
The power output of an engine is affected by the reduction in volumetric efficiency at increased engine speed (Fig. 20.40). The object of supercharging is to increase the volumetric efficiency above that obtained with normal aspiration. Supercharging of air by a compressor or blower increases its density and also its mass flow rate which permits the burning of more fuel and thus augments the output of the engine.

The increase in pressure and temperature of the intake air in diesel engine reduces ignition delay, lowers the rate of pressure rise and makes the combustion process better, quieter and smoother. There is a decrease in the exhaust gas temperature due to high expansion ratio as well as *A/F* ratio. Increased reliability, durability and better fuel consumption are some other benefits of supercharging. In a spark ignition engine, however, supercharging promotes knocking tendency, which leads to the use of lower compression ratio and hence leads to less efficiency.

The main features of supercharging are illustrated in *p–V* diagrams for the idealized constant volume four-stroke cycle in Fig. 11.42 and the plant line diagrams in Fig. 20.43. Figure 20.42(a) shows the normally aspirated cycle with line 5–1 representing both the inlet and exhaust strokes at about ambient air pressure $p_a$. The supercharged cycle is shown in Fig. 20.42(b), where $p_i$, $T_i$ refer to the engine inlet condition. Two ways of supercharging, viz. (a) mechanical supercharging with a blower driven by the engine itself and (b) turbocharging, where a gas turbine driven by the engine exhaust drives the compressor (centrifugal), are illustrated in Fig. 20.43.

The *T–s* diagram for a turbocharger is shown in Fig. 20.44. The compressor power input is,

$$\dot{W}_c = \dot{m}_a C p_a T_a \left[ (p_i / p_a)^{(\gamma_a - 1)/\gamma_a} - 1 \right] \eta_c$$




*Pressure-volume diagram for a four-stroke CI engine*
*(a) Without supercharging and (b) With supercharging*

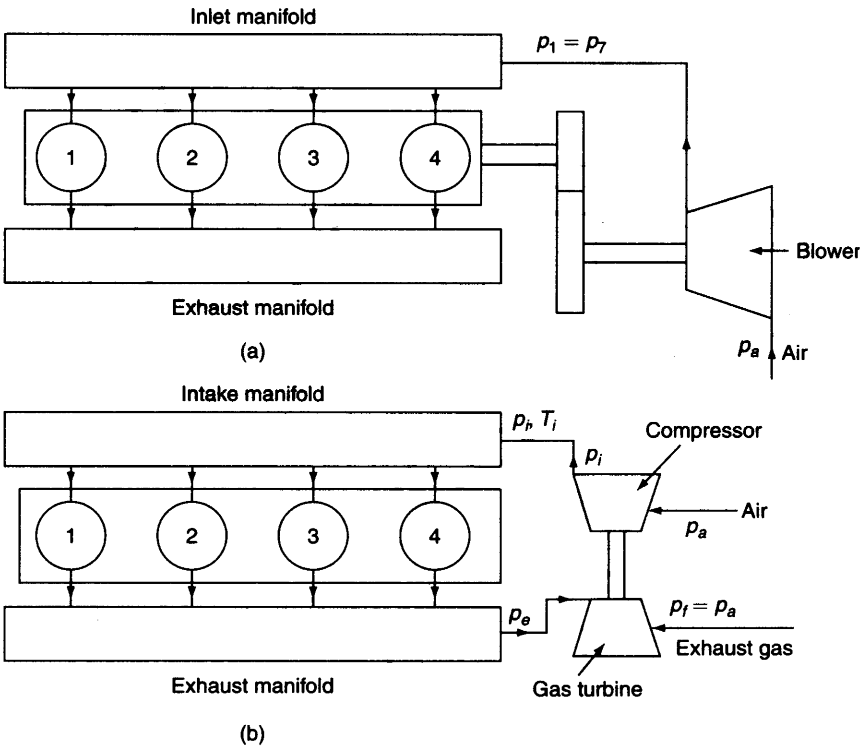



Fig. 20.43 *Diagrams of a four-stroke four-cylinder CI engine with (a) Mechanical supercharging and (b) turbocharging*

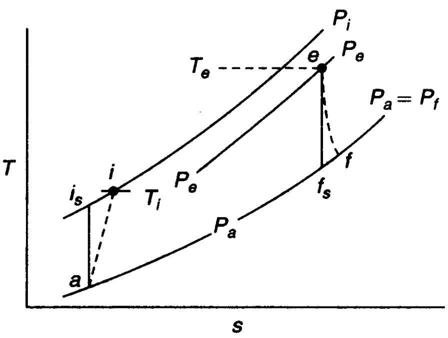



Fig. 20.44 *T-s diagram for a turbocharger of Fig.* 20.43

The turbine power output is,

$$\dot{W}_T = \dot{m}_g c p_e T_e \left[1-(p_a/p_e)^{(\gamma_e-1)\gamma_e}\right]\times\eta_T$$

where $\eta_c$ and $\eta_T$ ar e the isentropic efficiencies of the compressor and turbine respectively. Now,

$$\dot{m}_e = \text{rate of flow of exhaust gas from the engine}$$

$$= \dot{m}_a + \dot{m}_f$$

$$\dot{m}_e/\dot{m}_a = 1+(\dot{m}_f/\dot{m}_a) = 1+F/A$$

where *F/A* is the fuel air ratio.

Also, $W_c = W_T \times \eta_M$

where, $\eta_M$ is the mechanical efficiency.

$$\left[(p_i/p_a)^{(\gamma_a-1)/\gamma_a}-1\right] = \left[1-(p_a/p_e)^{(\gamma_e-1)/\gamma_e}\right]\left[c_{p_e}/c_{p_a}\right]\left[T_e/T_a\right]$$

$$[1+F/A]\times\eta_0$$

where, $\eta_0 = \eta_M \times \eta_T \times \eta_c =$ Overall efficiency of the supercharger.

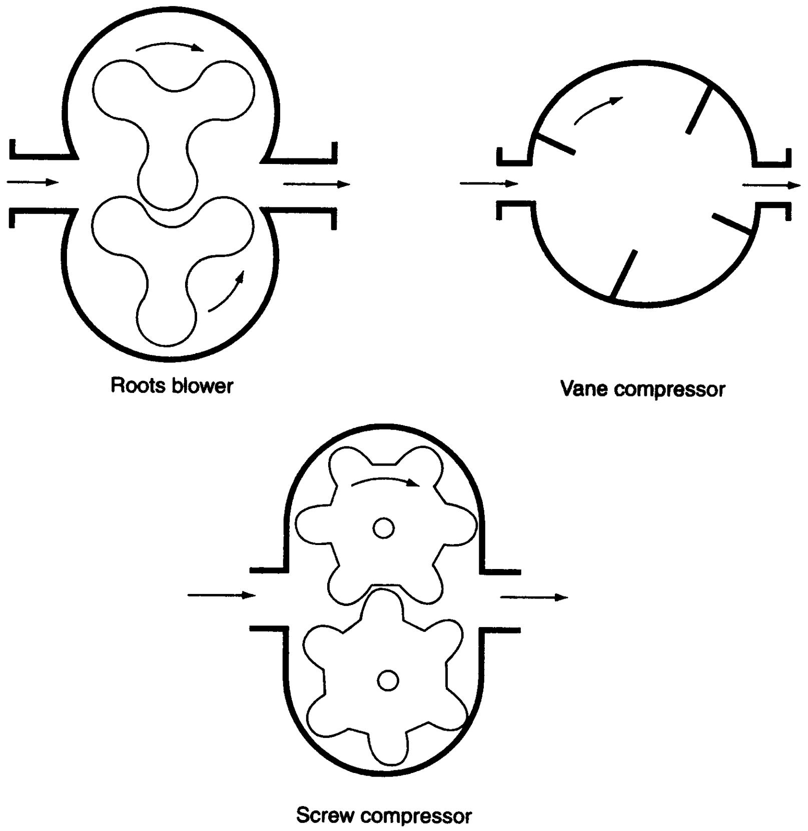



Fig. 20.45 *Types of positive displacement compressor*

Compressors for turbocharging can be divided into the following two classes.

(*i*) *Positive displacement* type like roots blower, vane compressor and screw compressor (Fig 20.45),

(*ii*) *Non-positive displacement* type like axial and radial compressors (Fig. 20.46). Turbochargers are superior to superchargers, since the former use the exhaust gas energy during blowdown and the latter consume a part of the engine output.

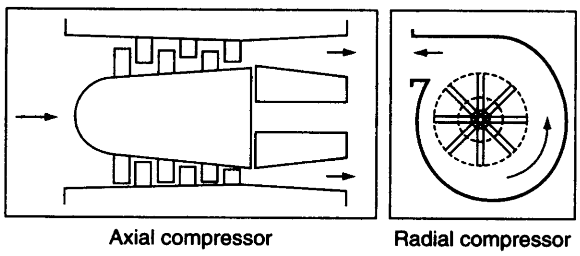
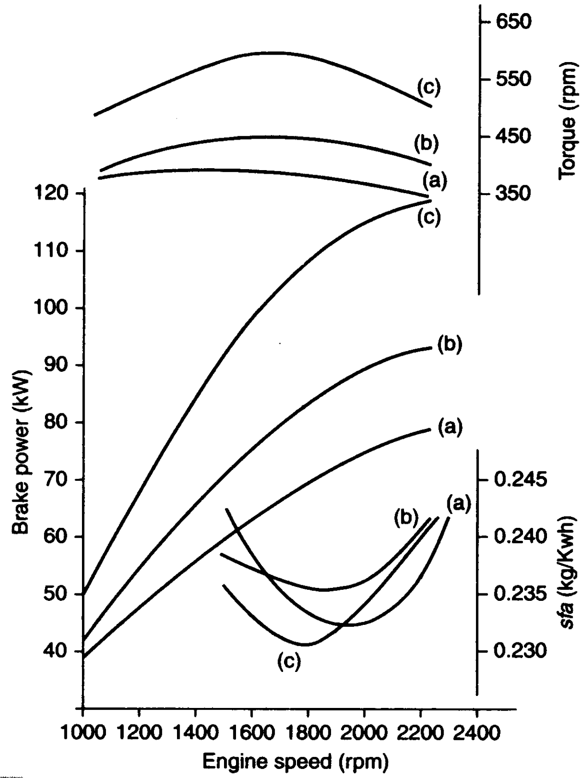



Fig. 20.46 *Types of dynamic or non-positive displacement compressor*

Figure 20.47 shows the continuous running performance characteristics of a diesel engine in three modes, i.e. (a) normally aspirated (b) turbocharged and (c) turbocharged with intercooling for a specific engine as given by Eastop and McConkey. It is seen that intercooling of air has a very good influence on the engine performance. Typical performances for a CI engine at full load and part load are shown in Fig. 20.48.




*Continuous running performance characteristics of a CI engine in three modes: (a) Normally aspirated, (b) Turbocharged and (c) Turbocharged with intercooling*

## Solved Examples

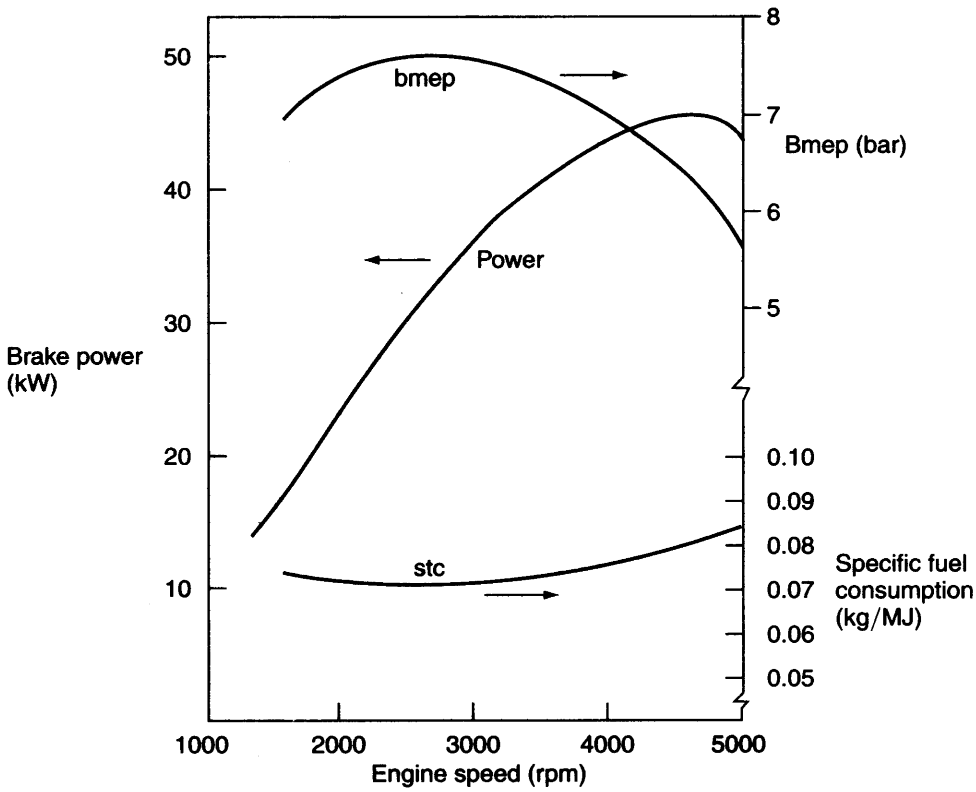
### Example 20.1

*A four cylinder four-stroke gasoline engine has a 65-mm diameter and 95-mm stroke. On test, it developed a torque of 64 Nm when running at 3000 rpm. If the clearance volume in each cylinder is 63 cm³, the brake efficiency ratio based on air standard efficiency is 0.5 and calorific value of gasoline is 42 MJ/kg, determine the fuel consumption in kg/h and the bmep.*

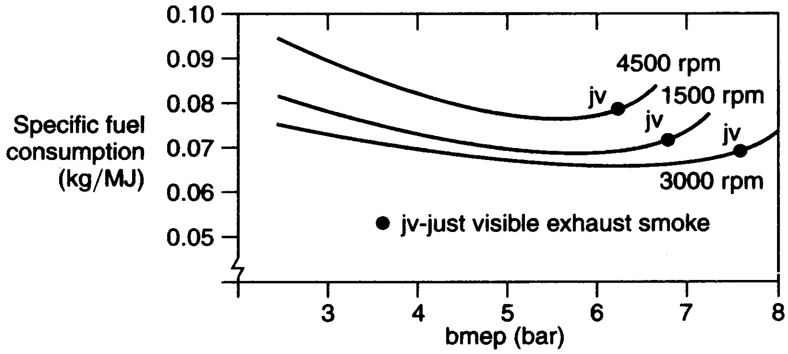
*Solution* Swept volume per cylinder, $V_s$

$$= \frac{\pi}{4} d^2 L = \frac{\pi}{4}(6.5)^2 \times 9.5 = 315.24 \text{ cm}^3$$

Compression ratio, $$r_k = \frac{V_s + V_c}{V_c} = \frac{315.24 + 63}{63} = 6$$




(a)




(b)

**Fig. 20.48** *Typical performance for a CI engine at (a) Full load and (b) Part load at different speeds*

$$\text{Air standard efficiency, } \eta_{as} = 1 - \frac{1}{r_k^{\gamma-1}} = 1 - \frac{1}{6^{0.4}} = 0.5125$$

$$\text{Relative efficiency} = \frac{\text{Brake thermal efficiency}}{\text{Air standard efficiency}} = 0.5$$

$$\text{Brake thermal efficiency} = 0.5 \times 0.5125 = 0.2562$$

$$\text{BP} = \frac{2\pi TN}{60} = \frac{2\pi \times 3000 \times 64}{60 \times 1000} = 20.1 \text{ kW}$$

$$\eta_{br.th} = \frac{\text{BP}}{m_f \times cv} = \frac{20.1}{\dot{m}_f \times 42000} = 0.2562$$

$$\therefore \quad m_f = \frac{20.1 \times 3600}{0.2562 \times 42000} = 6.726 \text{ kg/h} \qquad \textit{Ans.}$$

$$\text{BP} = \frac{\text{BMEP}\frac{\pi}{4}D^2 L N}{60 \times 2} = \frac{20.1}{4}$$

$$\text{BMEP} = \frac{5.025 \times 2 \times 60 \times 1000 \times 4}{0.095 \times \pi \times (0.065)^2 \times 3000}$$

$$= 627887 \cdot 4 \frac{\text{N}}{\text{m}^2} = 627 \cdot 8874 \text{ kN/m}^2 \qquad \textit{Ans.}$$

**Example 20.2**

*A four-cylinder two-stroke cycle petrol engine develops 30 kW at 2500 rpm. The mean effective pressure of each is 800 kN/m² and mechanical efficiency is 80%. Calculate the diameter and stroke of each cylinder if the stroke-to-bore ratio is 1.5. Also calculate the brake specific fuel consumption of the engine, if brake thermal efficiency is 28%. The heating value of petrol is 44 MJ/kg.*

*Solution* No. of cylinders = 4, B.P = 30 kW

$$N = 2500 \text{ RPM}, \; P_m = 800 \text{ kN/m}^2, \; \eta_{mech} = 80$$

$$L/d = 1 \cdot 5, \; \eta_{br \cdot th.} = 28\%, \; \text{C.V} = 44 \text{ MJ/kg.}$$

$$\text{I} \cdot \text{P} = \frac{\text{B.P.}}{\eta_{mech}} = \frac{30}{0 \cdot 8} = 37.5 \text{kW}$$

$$\text{I} \cdot \text{P} = \frac{p_m \text{LAN} n}{60}$$

$$37 \cdot 5 = \frac{800 \times 10^3 \times 1 \cdot 5d \times \frac{\pi}{4} d^2 \times 2500 \times 4}{60 \times 1000}$$

$$d^3 = 2 \cdot 3873 \times 10^{-4} \text{m}^3$$

$$\therefore \quad d = 0.062\,\text{m} = 6.2\,\text{cm}, L = 1.5 \times 6.2 = 9.3\,\text{cm}$$

$$\eta_{br.th} = \frac{BP}{\dot{m}_f \times cv}$$

$$0.28 = \frac{B.P.}{\dot{m}_f \times 44{,}000}$$

$$\text{bsfc} = \frac{\dot{m}_f}{BP} = \frac{1 \times 3600}{0.28 \times 44{,}000} = 0.2922\,\text{kg/kWh} \qquad \textit{Ans.}$$

### *Example* 20.3

*The following particulars refer to a 2-stroke diesel engine: bore = 10cm, stroke = 15cm, piston speed = 300 m/min, torque developed = 58 Nm, mechanical efficiency = 80%, indicated thermal efficiency = 40%, calorific value of fuel = 44 MJ/kg. Determine (a) indicated power, (b) indicated mean effective pressure, and (c) fuel consumption per kWh on brake power output.*

*Solution* Given: $d = 10$ cm $= 0.1$m, $L = 15$ cm $= 0.15$ m.

Piston Speed $= 2LN = 300$ m/min, $T = 58$ Nm.

Type of engine = 2-Stroke diesel engine.

$\text{C.V} = 44\ \text{MJ/kg},\ \eta_{mech} = 0{\cdot}80,\ \eta_{i.th.} = 0.40$

$$\text{Now, } 2LN = 300, \ \therefore N = \frac{300}{2 \times 0.15} = 1000\ \text{rpm}.$$

$$\eta_{mech} = BP/IP.\ \text{Also, B.P.} = \frac{2\pi TN}{60} = \frac{2\pi \times 58 \times 1000}{60 \times 1000} = 6.073\,\text{kW}$$

$$\text{I.P.} = \frac{6.073}{0{\cdot}8} = 7{\cdot}592\,\text{kW} \qquad \textit{Ans.}\text{(a)}$$

$$\text{I.P.} = \frac{p_m L \frac{\pi}{4} d^2 Nn}{60}$$

$$p_m = i_{mep} = \frac{60 \times 7.592 \times 4 \times 1000}{0.15 \times \frac{\pi}{4}(0.1)^2 \times 1000 \times 1}$$

$$= 386666.67\ \text{N/m}^2 = 386.67\ \text{kN/m}^2 \qquad \textit{Ans.}\text{(a)}$$

$$\eta_{i.th} = \frac{IP}{\dot{m}_f \times C.V.} = \frac{B.P.}{\dot{m}_f \times C.V. \times \eta_{mech}} = \frac{6.053 \times 3600}{\dot{m}_f \times 44000 \times 0.8}$$

$$\therefore \quad \dot{m}_f = \frac{7.5921 \times 3600}{0{\cdot}4 \times 44000} = 1.5529\ \text{kg/h}.$$

$$\therefore \quad \text{bsfc} = 1.5529/6.073 = 0.2557\,\text{kg/kWh} \qquad \textit{Ans.}$$

**Example 20.4**

*Following are the observations made for a 20 minute trial of a 2-stroke diesel engine:*

*Net brake load = 680N, mep = 3 bar, N = 360 rpm, Fuel consumption = 1.56 kg, Cooling water = 160 kg, Water inlet temperature = 57°C, Air used/kg fuel = 30 kg, Room temperature = 27°C, Exhaust gas temperature = 310°C, cylinder dimensions = 210-mm bore × 290-mm stroke, Brake diameter = 1m, Calorific value of fuel = 44 MJ/kg, Steam formed per kg fuel in the exhaust = 1.3 kg, specific heat of steam in the exhaust = 2.093 kJ/kg k. Specific heat of dry exhaust gases =1.01 kJ/kg k.*

*Calculate the indicated power and the brake power, and make an energy balance of the engine*

*Solution* For a two-stroke engine,

$$i_p = \frac{p_m \text{LANn}}{60} = \frac{3\times100\times0\cdot29\times\frac{\pi}{4}(0\cdot21)^2\times360\times1}{60}$$

$$= 18\cdot08\,\text{kW} \qquad \textit{Ans.}$$

$$b_p = \frac{2\pi TN}{60} = \frac{2\pi\times\left(680\times\frac{1}{2}\right)\times360}{60} = 12.818 \text{ kW}$$

$$\eta_M = bp/ip = 0.7089 \text{ or } 70.89\%$$

Heat supplied during the trial = 1.56×44,000 = 68,640 kJ.

Energy equivalent of *ip* in trial period = 18.08 kJ/s × (20 × 60) s = 21,696 kJ

Energy carried away by cooling water = 160 × 4.187 × (57 – 32) = 16,748 kJ

Total mass of exhaust gas = 1.56 × 30 + 1.56 = 48.36 kg

Mass of steam formed = 1.3 × 1.56 = 2.028 kg

Mass of dry exhaust gas = 48.36 – 2.028 = 46.332 kg

Energy carried away by dry exhaust gases

$$= 46.332 \times 1.01 \times (310 - 27) = 13{,}243 \text{ kJ}$$

Energy carried away by steam

$$= 2.028\,[4.187\,(100 - 27) + 2257.9 + 2.093(310 - 100)] = 6090 \text{ kJ}$$

Total energy carried away by exhaust gases = 19.333 kJ

| *Energy Balance Sheet* | | |
|---|---|---|
| *Energy released by combustion of fuel* | 68,640 kJ | |
| 1. Energy equivalent of ip | 21,696 kJ | 31.61% |
| 2. Energy carried away by cooling water | 16,748 kJ | 24.40% |
| 3. Energy carried away by exhaust gases | 19,333 kJ | 28.17% |
| 4. [illegible] energy loss (by difference) | 10,863 kJ | 15.82% |
| | 68,640 kJ | 100.00% |

**Example 20.5**

*The following particulars refer to a 2-Stroke oil engine: bore = 20 cm, stroke = 30 cm, speed = 350 rpm, i.m.e.p. = 275 kN/m², net brake load = 610 N, diameter of brake drum = 1 m, oil consumption = 4.25 kg/h, calorific value of fuel = 44 MJ/kg. Determine (a) IP, (b) B.P., (c) $\eta_{mech}$ (d) indicated thermal efficiency, and (e) brake thermal efficiency.*

*Solution*

$$\text{IP} = \frac{\text{imep LAN}}{60 \times 1000}\text{kW} = \frac{275 \times 10^3 \frac{\text{KN}}{\text{m}^2} \times 0 \cdot 3 \text{ m} \times \frac{\pi}{4}(0 \cdot 2^2 \text{ m}^2 \times 350 \text{ rpn}}{60 \times 1000}$$

$$= 15 \cdot 1189 \text{ kW} \qquad \textit{Ans.}(a)$$

$$\text{BP} = \frac{2\pi \text{N}(W - S)R}{60 \times 1000}\text{kW} = \frac{2\pi \times 350 \times 610 \times 0 \cdot 5}{60 \times 1000}$$

$$= 11.1788 \text{ kW} \qquad \textit{Ans.}(b)$$

$$\eta_{mech} = \frac{\text{BP}}{\text{IP}} = 0.7393 \text{ or } 73.93\% \qquad \textit{Ans.}(c)$$

$$\eta_{i.th.} = \frac{IP \times 3600}{\dot{m}_f \times CV} = \frac{15 \cdot 1189 \times 3600}{4.25 \times 44000} = 0.291 \text{ or } 29.1\% \qquad \textit{Ans.}(d)$$

$$\eta_{br.th.} = 0.291 \times 0.7393 = 0.2151 \text{ or } 21.51\% \qquad \textit{Ans.}(e)$$

**Example 20.6**

*A 4-stroke 6-cylinder gas engine with a stroke volume of 1.75 litres develops 26.25 kW at 506 rpm. The m.e.p. is 600 kN/m². Find the average number of times each cylinder misfired in one minute.*

*Solution* Given : $V_s = 1.75 \times 10^{-3} \text{m}^3$, $\text{IP} = 26.25$ kW, N = 506 rpm, imep = 600 kN/m², $n = 6$

$$\text{IP/cylinder} = \frac{\text{imep} \times \text{LA}\frac{N}{2}}{60}$$

$$\frac{26.25}{6} = \frac{600 \times 10^3 \times 1.75 \times 10^{-3} \times n}{60,000}$$

where $n$ = number of explosions

$$n = \frac{26.25}{6 \times 1 \cdot 75 \times 10^{-3}} = 250$$

As it is a 4-stroke, the number of explosions per min. should be 506/2 =253.

∴ no. of misfiring per min per cylinder = 253 – 250 = 3 *Ans.*

**Example 20.7**

*An engine is used on a job requiring 110 kW brake power. The mechanical efficiency of the engine is 80% and the engine used 50 kg/h of fuel under the conditions of operation. A design improvement is made which reduces the engine friction by 5 kW. Assuming that the indicated thermal efficiency remains the same, how many kg of fuel per hour will be saved?*

*Solution* BP = 110 kW, $\eta_{mech} = 80\%$

$\therefore$ IP = 110/0.8 = 137.5 kW

Frictional power = IP – BP = 137.5 – 110 = 27.5 kW

New frictional power = 27.5 – 5 = 22.5 kW

New IP = 110 + 22.5 = 132.5 kW

As the indicated thermal efficiency remains the same in both cases,

$$\frac{\text{New IP}}{\dot{m}_{f_1} \times \text{C.V.}} = \frac{\text{IP}}{\dot{m}_f \times \text{CV}}, \quad \text{where } m_{f_1} = \text{new fuel consumption per hour.}$$

$$\therefore \quad \frac{132.5}{\dot{m}_{f_1}} = -\frac{137.5}{50}$$

$$\therefore \quad \dot{m}_{f_1} = \frac{132.5 \times 50}{137.5} = 48.18 \text{ kg/h.}$$

$\therefore$ Saving in fuel = 50 – 48.18 = 1.82 kg/h *Ans.*

### *Example* 20.8

*During a trail of a four-cylinder four-stroke petrol engine coupled to a hydraulic dynamometer at constant speed, the following readings were obtained:*

*B.P. with all cylinders operating* = 14.7 *kW*
*B.P. with cylinder no.* 1 *cut out* = 10.14 *kW*
*B.P. with cylinder no.* 2 *cut out* =10.3 *kW*
*B.P. with cylinder no.* 3 *cut out* = 10.36 *kW*
*B.P. with cylinder no.* 4 *cut out* = 10.21 *kW*
*Fuel consumption* = 5.5 *kg/h*
*Calorific value of fuel* = 42 *MJ/kg*
*Diameter of cylinder* = 8 *cm*
*Stroke of piston* = 10 *cm*
*Clearance volume* = 0.1 *litre*

*Calculate (a) mechanical efficiency and (b) relative efficiency on indicated power basis.*

*Solution* $IP_1$ = 14.7 – 10.14 = 4.56

$IP_2$ = 14.7 – 10.3 = 4.4

$IP_3$ = 14.7 – 10.36 = 4.34

$IP_4$ = 14.7 – 10.21 = 4.49

Total IP = 4.56 + 4.4 + 4.34 + 4.49 = 17. 79 kW

$$\text{Mechanical efficiency} = \frac{\text{BP}}{\text{IP}} = \frac{14.7}{17.79} = 0.8263 \quad \text{or} \quad 82.63\% \qquad \textit{Ans. (a)}$$

$$V_s = \frac{\pi}{4}d^2 l = \frac{\pi}{4}(0.08)^2 \times (0.1) = 5.0265 \times 10^{-4}\,\text{m}^3$$

$V_c = 0.1\text{litre} = 0.1 \times 10^{-3} = 10^{-4}\text{m}^2$

Compression ratio, $r_k = \dfrac{V_s + V_c}{V_c} = \dfrac{5.0265\times10^{-4}+10^{-4}}{10^{-4}} = 6.0265$

Air standard efficiency, $\eta_{\text{A.S.E.}} = 1-\dfrac{1}{r_k^{\gamma-1}} = 1-\dfrac{1}{(6.0265)^{0.4}} = 0.5215$

Indicated thermal efficiency $= \dfrac{\text{IP}\times3600}{\dot{m}_f\times\text{C.V.}} = \dfrac{17.79\times3600}{5.5\times42000} = 0.2772$

$\therefore$ Relative efficiency $= \dfrac{0.2772}{0.5215} = 0.5316$ or $53.16\%$ *Ans.* (*b*)

### *Example* 20.9

*A four-cylinder four-stroke petrol engine of* 82-*mm bore,* 130-*mm stroke develops* 28.35 *kW brake power while running at* 1500 *rpm and using a* 20% *rich mixture. If the volume of the air into the cylinder when measured at* 15.5°*C and* 760 *mm of mercury is* 70% *of the swept volume, the theoretical air fuel ratio is* 14.8, *the heating value of petrol is* 44 *MJ/kg and the mechanical efficiency of the engine is* 90%. *Find* (*a*) *indicated thermal efficiency, and* (*b*) *brake mean effective pressure.*

*Solution* Given: $n = 4$, 4-stroke, $d = 0.082$ mm, L = 0.13 m, BP = 28.35 kW, $N = 1500$ rpm, Mixture = 20% rich.

Volume of air at 15.5°C and 760 mm Hg = 70% of swept volume

Theoretical A/F ratio = 14.8, CV = 44000 kJ/kg

$$\eta_{\text{mech}} = 0.9,\quad R = 0.287 \text{ kJ/kgK.}$$

$$\text{IP} = \frac{\text{BP}}{\eta_{\text{mech}}} = \frac{28.35}{0.9} = 31.5\text{kW}$$

760 mm Hg = 1.01325 kN/m$^2$

Volume of air sucked into the cylinder in m$^3$/min = 70% of swept volume × 4

$$= 0.7\times\frac{\pi}{4}(0.082)^2\times0.13\times\frac{1500}{2}\times4 = 1.4417\,\text{m}^3/\text{min}$$

Now, $P\dot{V} = \dot{m}RT$

$$\dot{m} = \frac{1.01325\times1.4417}{0.287(15.5+273)} = 1.764\,\text{kg/min}$$

Theoretically, 14.8 kg of air are required for 1 kg fuel, but the mixture is 20% rich,

Actual fuel consumption $= \dfrac{1.764}{14.8}\times1.2 = 0.143\,\text{kg/min}$

Indicated thermal efficiency $= \dfrac{31.5\times3600}{0.143\times44000\times60} = 0.30$ or $30\%$ *Ans.*

$$\text{Brake power} = b\text{mep} \times \frac{\frac{\pi}{4} d^2 l \times \frac{N}{2} \times n}{60}$$

$$28.35 = \frac{b\text{mep} \times \frac{\pi}{4}(0.082)^2 \times 0.13 \times \frac{1500}{2} \times 4}{60} = b\text{mep} \times 0.0343$$

$$\therefore \quad b\text{mep} = 825.53\,\text{kN/m}^2$$

*Ans.* (b)

**Example 20.10**

*In a simple carburetor, the throat diameter is 25 mm and the main jet diameter is 1.2 mm. The coefficients of discharge for the venturi and the fuel jet are 0.85 and 0.65 respectively. The gasoline surface is 6 mm below the throat. Estimate the minimum velocity of air required to start the flow of fuel. The ambient conditions are pressure 1 bar, $T_1 = 300$ K and $\rho_f = 760$ kg/m³.*

*Solution* The lip is provided in the carburetor to avoid the fuel wastage. The fuel will start flowing when the depression at the throat overcomes the effect of the lip. Thus,

$$\Delta p = h\rho_f g = 760 \times 6 \times 10^{-3} \times 9.81$$

$$= 44.75\ \text{N/m}^2$$

$$p_2 = p_1 - \Delta p = 1.0 - 44.75 \times 10^{-5}\ \text{bar}$$

Since the depression is very small, we can consider air as an incompressible fluid, and we have

$$p_1 = p_2 + \rho\frac{v^2}{2}$$

$$\therefore \quad \rho\frac{v^2}{2} = p_1 - p_2 = \Delta p = 44.75 \times 10^{-5}\ \text{bar} = 44.75\ \text{N/m}^2$$

$$V = \left(\frac{2 \times 44.75}{1.16}\right)^{\frac{1}{2}}, \text{ since } \rho_{\text{air}} = \frac{p}{RT} = \frac{10^5}{287 \times 300} = 1.168\ \text{kg/m}^3$$

$$= 8.75\ \text{m/s}$$

*Ans.*

**Example 20.11**

*A six-cylinder four-stroke S.I. engine develops 40 kW. During a Morse test conducted on the engine at 2000 rpm, the power output with each cylinder made inoperative turn by turn was 32.2, 32.0, 32.5, 32.4, 32.1 and 32.3 kW respectively. Estimate the mechanical efficiency, bmep, air standard efficiency when the bore = 100 mm, stroke = 125 mm, clearance volume = $1.23 \times 10^{-4} m^4$, the brake thermal efficiency when fuel consumption is 9 kg/h and the heating value is 40 MJ/kg, and the relative efficiency.*

*Solution* Morse test is conducted to determine the IP of multi-cylinder engine. The engine is made to run at constant speed and after steady-state conditions are reached, one cylinder, at one time, is made inoperative (by short circuiting the spark plug or by disconnecting the injector). Under these conditions, all other cylinders 'motor' this cut-out cylinder. The output is measured by keeping the speed constant at its original value. The difference in the output gives the indicated power of the cut-out cylinder. Thus, the IP produced by the engine is

$$IP = (40 - 32.2) + (40 - 32) + (40 - 32.5) + (40 - 32.4) + (40 - 32.1) + (40 - 32.3) = 46.5 \text{ kW}$$

$$\therefore \quad \eta_{mech} = 40/46.5 = 0.86 \quad \text{or} \quad 86\%$$ *Ans.*

$$bmep = \frac{BP \times 2 \times 60}{LAN} = \frac{40 \times 2 \times 60}{0.125 \times \frac{\pi}{4}(0.1)^2 \times 2000} = 305.54 \text{ kPa}$$

$$= 3.055 \text{ bar}$$ *Ans.*

Stroke volume, $V_s = \frac{\pi}{4} \times (0.1)^2 \times 0.125 = 9.819 \times 10^{-4} \text{ m}^2$

Compression ratio, $r_k = \frac{V_c + V_s}{V_c} = 1 + \frac{9.819 \times 10^{-4}}{1.23 \times 10^{-4}} = 8.99$

$\therefore$ air-standard efficience (ASE) $= 1 - \frac{1}{r_k^{\gamma-1}} = 0.5835$ or $58.35\%$ *Ans.*

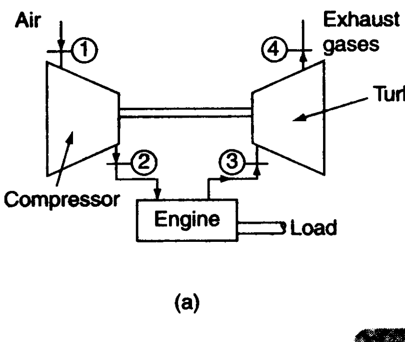
Brake thermal efficiency $= \frac{40 \times 3600}{9 \times 40,000} = 0.4$ or $40\%$ *Ans.*

Relative efficiency $= \frac{0.4}{0.5835} = 0.686$ or $68.6\%$ *Ans.*

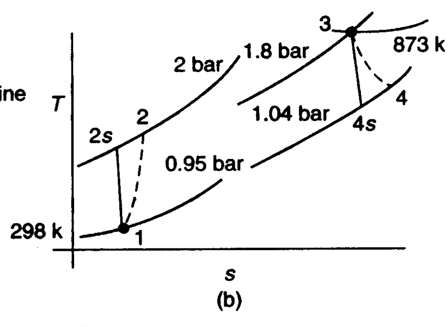
### Example 20.12

*Air is sucked into a diesel engine fitted with a turbocharger at a pressure of* 0.95 *bar and a temperature of* 25°C. *The delivery pressure is* 2 *bar. The air-fuel ratio is* 18:1. *The temperature of the gases leaving the engine is* 600°C. *and the pressure in* 1.8 *bar. The gases come out of the turbine at* 1.04 *bar. The efficiency of the compresor is* 0.75 *and that of turbine is* 0.85. *Calculate the power lost in the supercharger expressed as a percentage of the power developed by the turbine. For air, take* $c_p$=1.005 *and for gases* $c_p$=1.15 *kJ/kgk,* $\gamma = 1.4$.

*Solution* The compressor in driven by the turbine (turbocharging) and both of them are mounted on the same shaft (Fig. Ex. 20.12). The energy available in the exhaust gases from the engine is converted into useful work by the turbine which drives the compressor. The T-s diagram is shown in the figure. For isentropic compression,




(a)




(b)

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{2s} = 298\left(\frac{2}{0.95}\right)^{0.286} = 368.7\,\text{K}$$

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} = 0.75 = \frac{368.7 - 298}{T_2 - 298}$$

$$\therefore \qquad T_2 = 392.28\ \text{K}$$

Power required by the compressor, $W_c = mc_p\,(T_2 - T_1)$

$$= 1 \times 1.005\,(392.28 - 298) = 94.75\ \text{kJ/kg}$$

For isentropic expansion in the turbine

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$\frac{T_3}{T_{4s}} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.8}{1.04}\right)^{0.286} = 1.17$$

$$T_{4s} = \frac{873}{1.17} = 746.23\,\text{K}$$

$$T_4 = T_3 - (T_3 - T_{4s})\eta_T = (873 - 746.23) \times 0.85 = 765.2\ \text{K}$$

$$\frac{A}{F} = \frac{18}{1},\ m_g = m_a\left(1 + \frac{1}{18}\right) = \frac{19}{18}m_a.$$

Power developed by the turbine, $W_T = \frac{19}{18} \times 1 \times 1.15(873 - 765.2) = 130.86\,\text{kJ/kg air}$

Power lost as a percentage of the power produced by the turbine $= \dfrac{130.86 - 94.75}{130.86}$

$= 0.276$ or $27.6\%$ *Ans.*

### *Example* 20.13

*A four-stroke 6-cylinder diesel engine develops 250 kW at 2000 rpm. The bsfc is 0.20 kg/kWh. At the beginning of injection, pressure is 35 bar and the maximum cylinder pressure is 55 bar. The injection is expected to be 180 bar and the maximum pressure at the injector is set to be about 520 bar. Assuming $c_d = 0.78$, specific gravity of fuel oil = 0.85, atmospheric pressure 1 bar, estimate the total arifice area per injector if the injection takes place over 18° crank angle.*

*Solution*

$$\text{BP per cylinder} = \frac{250}{6} = 41.67\,\text{kW}$$

$$\dot{m}_f = 41.67 \times 0.2 = 8.33\ \text{kg/h} = 0.139\ \text{kg/min}$$

$$\text{Fuel injected per cycle } = 0.139 \times \frac{2}{N} = \frac{0.139}{1000} = 0.139 \times 10^{-3}\ \text{kg}$$

$$\text{Time for injection } = \frac{\theta}{360 \times \frac{N}{60}} = \frac{18}{360 \times \frac{2000}{60}} = 1.5 \times 10^{-3}\,\text{s}$$

Pressure difference at the beginning = 180 − 35 = 145 bar

Pressure difference at the end = 520 − 55 = 465 bar

$$\text{Average pressure difference } = \frac{465 + 145}{2} = 305\,\text{bar}$$

$$\text{Velocity of injection } = c_d \sqrt{\frac{2\Delta^{\ 2}}{\rho_f}} = 0.78\sqrt{\frac{2 \times 305 \times 10^5}{850}}$$

$$= 208.96\ \text{m/s}$$

$$\text{Volume of fuel injected per cycle } = \frac{0.139 \times 10^{-3}}{850}$$

$$\therefore \qquad \text{area of orifice} = \frac{\text{Volume of fuel injected per cylcle}}{\text{Injection velocity} \times \text{Injection timing}}$$

$$= \frac{0.139 \times 10^{-3}}{850 \times 208.96 \times 1.5 \times 10^{-3}} = 0.522\ \text{mm}^2 \qquad \textit{Ans.}$$

**Example 20.14**

*The compression curve on the indicator digram from a gas engine follows the law $pv^{1.3}$ = constant. At two points on the curve at ¼ stroke and ¾ stroke, the pressures are 140 kN/m² and 360 kN/m² respectively. Determine the compression ratio of the engine. Calculate the thermal efficiency and the gas consumption per kWh on indicated power basis if the relative efficiency is 0.4 and gas has the calorific value of 18840 KJ/m³.*

*Solution* Let the point 1 is at $\frac{1}{4}$ th stroke and point 2 is at $\frac{3}{4}$ th stroke of the compression curve AB (Fig. Ex. 20.14).

Let $V_s$ = stroke volume

$V_C$ = clearance volume

$$\therefore \qquad V_1 = V_C + V_S - \frac{V_S}{4} = V_C + 0.75 V_S$$

$$\text{Volume at point 2,} \quad V_2 = V_C + V_S - \frac{3}{4}V_S = V_C + 0.25 V_S$$

$$\frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{1.3}} = \left(\frac{360}{140}\right)^{\frac{1}{1.3}} = 2.067 = \frac{V_C + 0.75 V_S}{V_C + 0.25 V_S}$$

$$\therefore \qquad \frac{V_S}{V_C} = 4.5735$$

$$\text{Compression ratio, } \gamma_k = \frac{V_S + V_C}{V_C} = \frac{V_S}{V_C} + 1 = 5.5735 \quad \textit{Ans.}$$

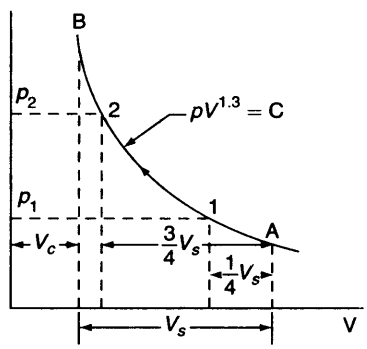



Fig. Ex. 20.14

$$\eta_{ASE} = 1 - \frac{1}{\gamma_k^{\gamma-1}} = 1 - \frac{1}{(5.5735)^{0.4}} = 0.4971$$

Relative efficiency $= \frac{\eta_{i.th.}}{\eta_{ASE}} = 0.4$

$\therefore$ $\eta_{i.th.} = 0.4 \times 0.4971 = 0.19884$ or $19.884\%$ *Ans.*

$\therefore$ $0.19884 = \frac{3600}{18840 \times \dot{V}_f}$

$\therefore$ $\dot{V}_f = 0.961 \text{ m}^3/\text{kWh}$ *Ans.*

**Example 20.15**

*During an engine trial on a 6-cylinder 4-stroke diesel engine, bore of 180 mm, stroke of 200 mm, the following observations are recorded. BP = 245 kW, speed = 1500 rpm, mep = 8 bar, fuel consumption 70 kg/h, HV of fuel = 42 MJ/kg, hydrogen content of fuel = 12% by mass; air consumption 26 kg/min, mass of cooling water 82 kg/min, cooling water temperature rise 44°C, cooling oil circulated through the engine = 50 kg/min, temperature rise of cooling oil 24°C, specific heat of cooling oil = 2.1 kJ/kgK, room temperature 30°C, exhaust gas temperature 400°C, $c_p$ of dry exhaust gas, 1.045 kJ/kgK, partial pressure of steam in exhaust gas is 0.035 bar, estimate the mechanical efficiency and draw up an energy balance sheet.*

*Solution*

Heat energy supplied by fuel $= \frac{70 \times 42,000}{3600} = 816.67 \text{ kW}$

Indicated power $= \frac{p_m LAN}{2 \times 60} = \frac{8 \times 10^5 \times 0.2 \times \frac{\pi}{4}(0.18)^2 \times 1500 \times 6}{2 \times 60 \times 1000} = 305.4 \text{ kW}$

$\eta_{mech} = \frac{BP}{IP} = \frac{245}{305.4} = 0.802$ or $80.2\%$ *Ans.*

Heat carried away by cooling water $= \frac{82}{60} \times 4.187 \times 44 = 251.78 \text{ kW}$

Heat carried away by cooling oil $= \frac{50}{60} \times 2.1 \times 24 = 42 \text{ kW}$

Mass of exhaust gas $= \dot{m}_f + \dot{m}_a$

$= \frac{70}{60} + 26 = 27.167 \text{ kg/min}$

Mass of vapour in the exhaust gases $= 0.12 \times 9 \times \frac{70}{60} = 1.26 \text{ kg/min}$

($2H_2 + O_2 = 2H_2O$

4 kg 32 36 kg

1 kg 8 9 Kg)

Mass of dry exhaust gas = (27.167 − 1.26)/60 = 0.4318 kg/s

From steam table, the enthalpy of steam at the exhaust condition (0.035 bar) = 3060 kJ/kg

Heat carried away by steam $= 0.21 \times 3060 = 64.28$ kW

Heat carried away by dry exhaust gases

$$= 0.4318 \times 1.045 \times (400 - 30) = 166.96 \text{ kW}$$

| *Energy Balance* | | |
|---|---|---|
| | Input | Output |
| Heat supplied by fuel | 816.67 kW | - |
| Useful work (BP) | - | [illegible] kW |
| Heat carried by cooling water | - | 251.78 kW |
| Heat carried away by steam | - | 64.28 kW |
| Heat carried away by cooing oil | - | 42.0 kW |
| Heat carried away by dry exhaust gas | - | 166.96 kW |
| Heat transferred to surroundings | - | 46.65 kW |
| | 816.67 kW | 816.67 kW |

**Example 20.16**

*A 4-stroke 4-cylinder S.I. engine running at* 3000 *rpm produces a torque of* 66.5 *Nm. It has a bore of* 60 *mm and a stroke of* 100 *mm. The clearance volume in each cylinder is* 60 *cc and the relative efficiency with respect to brake efficiency is* 0.5. *If the calorific value of fuel is* 42 *MJ/kg, determine the fuel consumption in kg/h and the bmep.*

*Solution* Swept volume, $V_s = \frac{\pi}{4} \times (0.60)^2 \times 0.1 = 2.83 \times 10^{-4} \text{ m}^3$

Compression ratio, $$\gamma_k = \frac{2.83 \times 10^{-4} + 0.6 \times 10^{-4}}{0.6 \times 10^{-4}} = 5.71$$

$$\therefore \quad \eta_{\text{air std}} = 1 - \frac{1}{5.71^{1.4-1}} = 0.50$$

$$\eta_{br} = 0.50 \times \eta_{rel} = 0.50 \times 0.50 = 0.25$$

$$\text{BP} = \frac{2\pi \text{TN}}{60}$$

$$= \frac{2\pi \times 66.5 \times 3000}{60} \times 10^{-3} = 20.89 \text{ kW}$$

$$\eta_{br} = \frac{\text{bp}}{\dot{m}_f \times \text{C.V.}} = \frac{20.89}{\dot{m}_f \times 42 \times 10^3} = 0.25$$

$$\therefore \quad \text{Fuel consumption, } \dot{m}_f = \frac{20.89}{42 \times 10^3 \times 0.25} \times 3600 = 7.16 \text{ kg/h}$$ *Ans.*

$$bp = \frac{\text{bmep} \times \frac{\pi}{4} D^2 L \times \frac{N}{2}}{60}$$

$$20.89 = \frac{b\text{mep} \times 2.83 \times 10^{-4} \times \frac{3000}{2}}{60} \times 10^{-3}$$ *Ans.*

$$b\text{mep} = 7.38 \times 10^5 \text{ Pa} = 7.38 \text{ bar}$$

## Review Questions

20.1 What are the differences between internal and external combustion engines?

20.2 What are the advantages and applications of IC engines?

20.3 Describe how IC engines are classified.

20.4 Give with a neat sketch an overview of a reciprocating IC engine.

20.5 Explain the differences between S.I. and C.I. engines.

20.6 Describe with a neat sketch the main components of an S.I. engine.

20.7 Explain the working principle of a 4-stroke S.I. engine with the help of *p-v* and valve timing diagrams.

20.8 Explain with sketches the operation of 4-stroke C.I. engine.

20.9 How does the operation of a 2-stroke S.I. engine differ from that of a 4-stroke S.I. engine?

20.10 Explain the valve-timing diagram of a 2-stroke S.I. engine.

20.11 Define the terms: (a) brake power, (b) indicated power, (c) mechanical efficiency, (d) friction power, (e) imep, (f) bmep, (g) isfc, and (h) bsfc.

20.12 What do you mean by volumetric efficiency? On what factors does it depend?

20.13 Show that the brake power of an IC engine depends on the density of air used.

20.14 How are these measured: (a) bp, (b) ip, (c) bmep, (d) imep and (e) bsfc?

20.15 Why are air-standard cycles conceived? Show that the efficiency of the Otto cycle depends on the compression ratio only.

20.16 Explain the effects of compression ratio and specific heat ratio on Otto cycle efficiency.

20.17 Derive the expressions for net work output and m.e.p. of the Otto cycle in terms of $r_k$, $r_p$, and $\gamma$.

20.18 Explain the air-standard cycle of a C.I. engine. Show that the cycle efficiency depends on $r_k$ and $r_c$.

20.19 Describe the types of fuels used for I.C. Engines.

20.20 What are the important qualities of S.I. engine fuels?

20.21 Explain the octane number rating of S.I. engine fuels. Why is unleaded petrol used in automobiles?

20.22 Explain the main qualities of C.I. engine fuels. What do you mean by cetane number rating of diesel oil?

20.23 Describe the combustion mechanism in an S.I. engine. What is the relationship between pressure and crank angle for different rates of combustion?

20.24 What is abnormal combustion? Explain the phenomena of preigntion and detonation. What is auto-ignition?

20.25 Explain with a sketch the ignition system of an S.I. engine. How does the spark plug operate?

20.26 What is the function of a carburetor? Describe the operation of a simple float carburetor with a neat sketch. What are idling jet and compensating jet? When is the accelerating pump used?

20.27 Describe the fuel injection systems in C.I. engines. Explain the operation of a fuel pump. What are the types of fuel injectors?

20.28 Explain the combustion mechanism in C.I. engines. What is ignition delay? What are the factors which influence ignition delay?

20.29 Explain the C.I. engine combustion chambers.

20.30 Describe the cooling system of I.C. engines with neat sketches.

20.31 What are the various components to be lubricated in an engine? Explain how it is accomplished.

20.32 Explain the various mechanisms of lubrication in an I.C. engine.

20.33 What are the common methods of starting an engine?

20.34 What are the distinctive features in combustion in a C.I. engine as compared to an S.I. engine?

20.35 Explain the performance characteristics of an I.C. engine.

20.36 Describe the Morse test in measuring the I.P. of a multi-cylinder engine.

20.37 What do you mean by volumetric efficiency? What are the variables which affect the volumetric efficiency of an engine?

20.38 Give an energy balance of an IC engine.

20.39 What is the objective of supercharging? Why is it more beneficial in an IC engine compared to a SI engine?

20.40 Explain the main features of supercharging with the help of a $p-V$ diagram. What do you mean by mechanical supercharging and turbocharging? What is the effect of intercooling in turbocharging?

20.41 What are the different types of compressors used for supercharging? Why are turbochargers superior to superchargers?

20.42 Explain the running performance characteristics of a diesel engine operating in various modes.

## Problems

20.1 Calculate the air flow rate, fuel flow rate, bmep and imep for a 4-stroke engine developing 300 kW at 2500 rpm when the stroke volume is 2000 cc, thermal efficiency is 0.30 and air/fuel ratio is 15.
*Ans.* 0.075 kg/s, 0.9 kg/s, 7.2 MPa, 9 MPa.

20.2 A petrol engine having a compression ratio of 8, has a brake thermal efficiency which is 40% of air standard efficiency. The calorific value of fuel is 44 MJ/kg. Calculate the fuel consumption in kg/h, if the engine delivers 7.5 kW. *Ans.* 2.7165 kg/h

20.3 A four-cylinder engine running at 1200 rpm gave 18.6 kW brake power. The average torque when one cylinder was cut off was 105 Nm. Determine the indicated thermal efficiency if the heating value of fuel is 42000 kJ/kg and the engine uses 0.34 kg of petrol per kW brake power.
*Ans.* 29.3%

20.4 A 4-stroke engine uses natural gas and has a stroke of 400 mm, bore of 350 mm. It runs at 1000 rpm and the air-fuel ratio 5:1 by volume and the volumetric efficiency of 0.80. Estimate the bmep when the brake thermal efficiency is 0.28 and the HV of the gas is 8 MJ/m$^3$ at standard conditions.
*Ans.* 298.6 kPa.

20.5 In a simple carburetor, the gasoline in the float chamber stands 5-mm below the nozzle opening. The engine consumes 6.1 kg of fuel per hour. The fuel jet diameter is 1.25mm and the discharge coefficient for fuel orifice is 0.64. Estimate the air velocity at the throat, the throat diameter if the air-fuel ratio is 16:1. $C_d$ for air is 0.87 and the ambient conditions are: $p_1$ = 1 bar, $T_1$ = 300 K, $\rho_f$ = 775 kg/m$^3$. *Ans.* $d_2$ = 2.19 cm

20.6 A single cylinder 4-stroke diesel engine running at 2000 rpm uses 2.8 kg of fuel (sp. gravity 0.88) per hour. It has an injector with a single orifice nozzle and the injection period is 22° crank angle. If the injection pressure is 150 bar and the average pressure inside the cylinder is 30 bar, estimate the fuel orifice diameter. Take $C_d$ for the nozzle = 0.88. *Ans.* 0.503 mm

20.7 A Diesel engine has a compression ratio of 14:1 and fuel is cut off at 0.08 of stroke. Calculate the mass of fuel used per kWh if the calorific value is 42 MJ/kg and the relative efficiency 0.54.
*Ans.* 0.269 kg/kWh.

20.8 A 4-cylinder 4-stroke single acting S.I. engine gave the following results:
Petrol consumption =5 kg/h
Speed =750 rpm
Air-fuel ratio = 11:1
Temperature at the end of compression=375°C
Pressure at the end of compression = 1200 kN/m$^2$
Relative efficiency = 0.50
Diameter of cylinder = 8.5 cm
Stroke of piston = 12 cm
Calorific value of fuel = 42 MJ/kg
R for mixture=0.285 kJ/kgK.
Estimate specific fuel consumption, (isfc)
*Ans.* 0.308 kg/kWh

20.9 Find the diameter of the cylinder of a single acting diesel engine, working on four-stroke cycle, which is required to give 38 kW indicated power at 200 rpm from the following data: compression ratio 14:1, fuel cut-off 5% of the stroke, index of compression curve 1.4; index of expansion curve 1.3, pressure at the beginning of compression 100 kPa, ratio of stroke to bore 1.5:1.

*Ans.* 33.76 cm

20.10 A four-stroke diesel engine of 2000 cc produces 12.1 kW per $m^3$ of free air inducted per minute. The speed is 3000 rpm and the volumetric efficiency is 0.84. When the ambient conditions are: $p_1 = 1$ bar, $T_1 = 300$ K, a supercharger is introduced in the system which raises the pressure to 1.8 bar with an isentropic efficiency of 0.8. Estimate the increase in the output of the engine.

*Ans.* 55.72 kW

20.11 A single cylinder 4- stroke gas engine has a bore of 180 mm, stroke of 330 mm and is governed to run at 750 rpm. At full load, indicator cards are taken which give a working loop mep of 7.8 bar and a pumping loop mep 0.41 bar. The dead cycle mep is 0.68 far. The number of firing strokes were 125 per min at no load conditions. Estimate the brake power and the mechanical efficiency.

*Ans.* 28.25 kW, 72.8%

20.12 A 4-stroke 3000 cc 6 cylinder engine has a maximum torque of 240 Nm at 2000 rpm. The specific fuel consumption is 0.09 kg/MJ and the air-flow rate is 4.08 $m^3$/min. The ambient conditions are $p_1$=1bar, $T_1$=300 K. Estimate the volumetric efficiency, air-fuel ratio and the thermal efficiency if the heating value of fuel is 44 MJ/kg.

*Ans.* 0.907, 11.64, 0.2525

20.13 Estimate the cylinder dimensions of 4-stroke 3 cylinder engine producing 25 kW at 3500 rpm, when the air-fuel ratio is 15, the heating value of fuel is 40 MJ/kg, the volumetric efficiency is 0.85, brake thermal efficiency is 0.30.

*Ans.* $d$ =74.8 mm, $L$ = 82.32 mm

20.14 Find the air velocity and the corresponding venturi throat diameter of a carburetor when the fuel jet cross-sectional area is 1.95 $mm^2$ and the fuel flow rate is 4.98 kg/h. The intake conditions are $p$ = 1 bar, $T$ = 300 K, Air fuel ratio = 14.5:1, $C_d$ for fuel jet = 0.6, $C_d$ for venturi = 0.84, $\rho_f$ =780 kg/$m^3$.

*Ans.* 25.6 m/s, 32 mm.

20.15 A 6-cylinder 4-stroke SI engine having a bore of 90 mm and a stroke of 100 mm has a compression ratio of 7. The relative efficiency is 0.55 when the indicated specific fuel consumption is 300 g/kWh. Calculate (i) the calorific value of the fuel, (ii) the fuel consumption in kg/h, when the imep is 8.5 bar and speed is 2500 rpm.

*Ans.* (i) 40.4 MJ/kg, (ii) 20.28 kg/h.

20.16 A 4-cylinder gasoline engine having a bore of 75 mm and a stroke of 90 mm, and running at 3000 rpm develops a brake power of 20.9 kW. A Morse Test was conducted on this engine, and the brake power (kW) obtained when each cylinder was made inoperative by short circuiting the spark plug are 14.9, 14.3, 14.8 and 14.5 respectively. The test was conducted at constant speed Estimate the indicated power, mechanical efficiency and the bmep, when all the cylinders are firing.

*Ans.* 25.1 kW, 83.3%, 5.25kW

20.17 The following observations were made during a trial of a single-cylinder four stroke cycle gas engine having a bore of 180 mm and stroke 240 mm:

Duration of trial=30 min
Total number of revolutions = 9000
Total number of explosions = 4450
Mean effective pressure = 5 bar
Net load on the brake wheel = 40 kg
Effective diameter of the brake wheel = 1m
Total gas used at NTP = 2.4 $m^3$
Calorific value of gas at NTP = 19 MJ/$m^3$
Total air used = 36 $m^3$
Pressure of air = 720 mm Hg
Temperature of air = 17°C
Density of air at WTP = 1.29 kg/$m^3$
Temperature of exhaust gas = 350°C
Room temperature = 17°C
Specific heat of exhaust gas = 1 kJ/kgk.
Cooling water circulated = 80 kg
Temperature rise of cooling water = 30°C.

Draw up an energy balance of the engine and estimate the mechanical and indicated efficiencies of the engine. *Ans.* 81.6%, 20.8%

20.18 A six-cylinder 4-stroke gasoline engine having a bore of 80 mm and a stroke of 100 mm for each cylinder consumes fuel at the rate of 20 kg/h and develops a torge of 150 Nm at the speed of 4000 rpm. The clearance volume in each cylinder is 70 cc. Estimate (i) the brake power, (ii) the brake mean effective pressure, (iii) the brake efficiency if the calorific value of fuel is 43 MJ/kg, and (iv) the relative efficiency based on brake power.

*Ans.* (i) 62.8 kW, (ii) 6.25 bar, (iii) 26.3%, (iv) 46.3%

CHAPTER 21

# Gas Turbines and Propulsion Systems

The economics of power generation by gas turbines is now quite attractive due to low capital cost and its high reliability and flexibility in operation. Another outstanding feature is its capability of quick starting and using a wide variety of fuels from natural gas to residual oil or powdered coal. Its consumption of lubricating oil is quite low because of the absence of rubbing and reciprocating parts, and the balancing does not pose a major problem. Due to better materials being made available and with the use of adequate blade cooling, the inlet gas temperature to the gas turbine (GT) blades can now exceed 1200°C, as a result of which the overall efficiency of a GT plant can be about 35%, almost the same as that of a conventional steam power plant.

Because of its low weight per unit power, gas turbine is exclusively used to drive aviation system of all kinds of aircraft. It is also being increasingly used in land vehicles like buses and trucks and also to drive locomotives and marine ships. In oil and gas industries, the gas turbine is widely employed to drive auxiliaries like compressors blowers and pumps.

## 21.1 CLOSED CYCLE AND OPEN CYCLE PLANTS

The essential components of a gas turbine (GT) power plant are the compressor, combustion chamber and the turbine. The air standard cycle of a GT plant is the Brayton cycle.

A GT plant can either be open or closed. Figure 21.1 shows the arrangement of an open-cycle plant which is more common. The compressor takes in ambient air and raises its pressure. The temperature of air is increased when it flows through a combustion chamber where a fossil fuel is burned or a heat exchanger (nuclear fuel being used as a source of energy) is present. The high-pressure, high-temperature working fluid, mostly a gas, enters a turbine where it expands to a low-pressure (equal to or a little above the atmospheric pressure) fluid. In an open unit, the gas is released from the turbine to the surroundings and in a closed unit, the working fluid is cooled in a cooler after the exhaust from the turbine and is returned to the compressor (Fig. 21.2). A significant part of the power developed by the turbine is utilized to drive the compressor and any other auxiliaries, and the remainder is available as useful work.

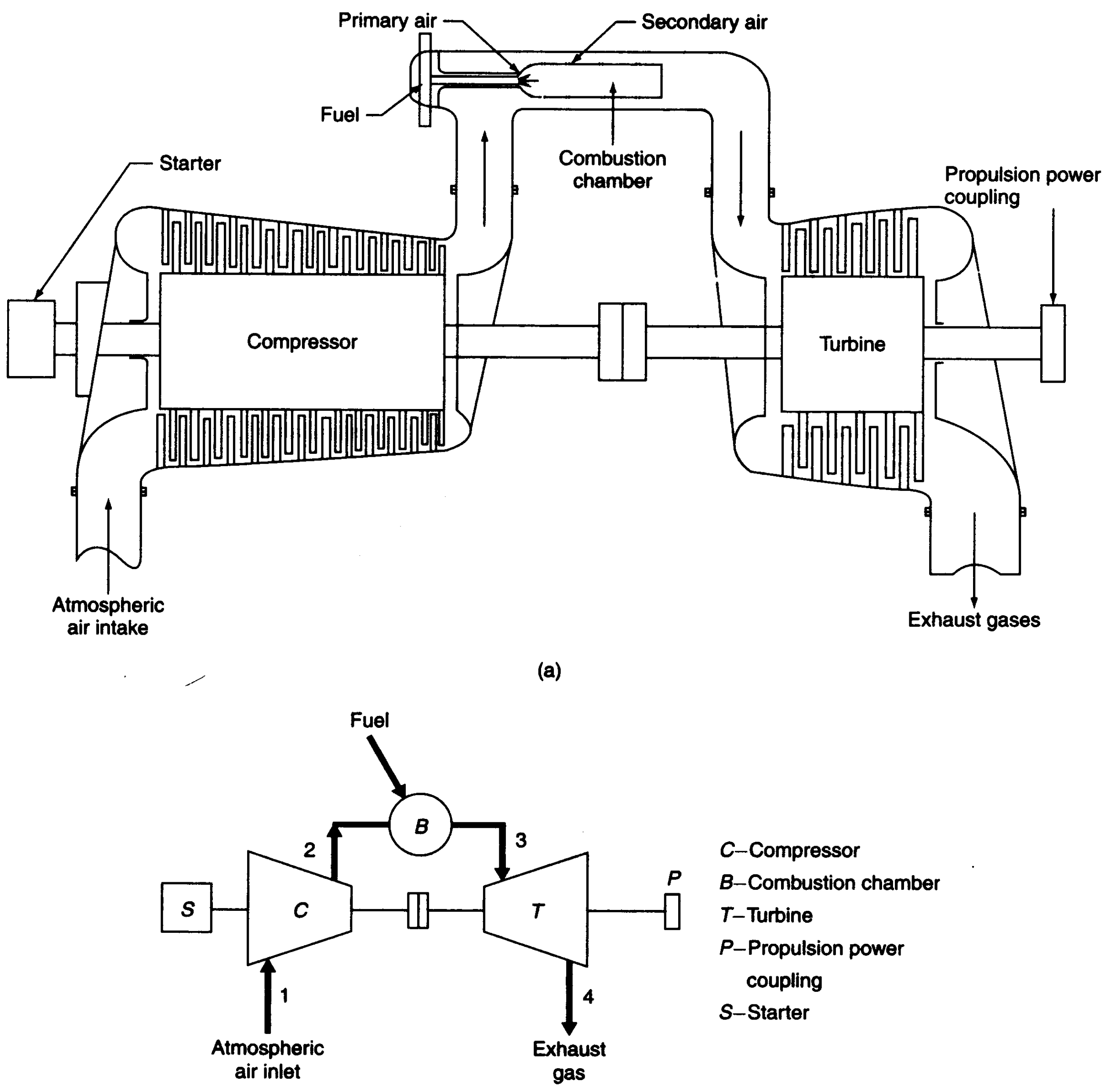
## 21.2 ADVANTAGES OF A GT PLANT

The advantages of a GT plant for power generation are being enumerated below:

**(1)** ***Warm-up Time*** Once the turbine is brought up to the rated speed by the starting motor and the fuel is ignited, the GT will accelerate from cold start to full load without warm-up time.

**(2)** ***Low Weight and Size*** The weight of the plant per kW output is low, which is a favourable feature in all vehicles (land, air and sea). In utilities also, the foundation of the plant is lighter.

**(3)** ***Fuel Flexibility*** Any hydrocarbon fuel from high octane gasoline to heavy diesel oil and pulverized coal can be used effectively.

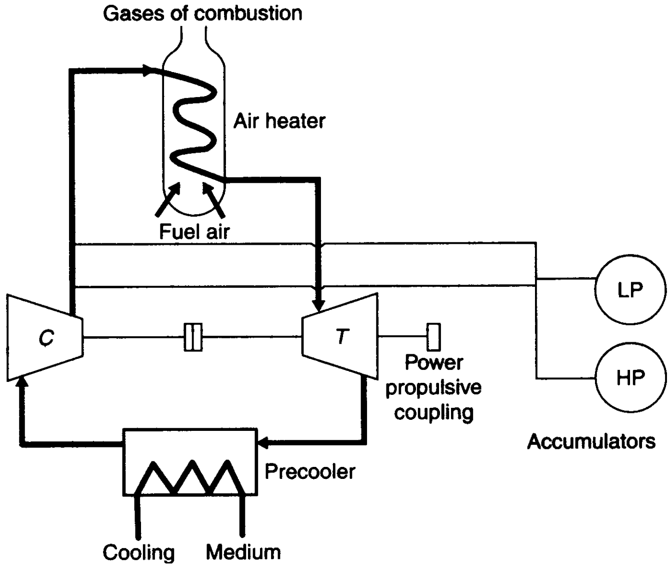



[redacted] *(a) Component parts of a simple open cycle (constant-pressure combustion) gas turbine, (b) Simple open cycle gas turbine*

**(4)** ***Floor Space*** Because of its smaller size, the floor space required for its installation is less.

**(5)** ***Start-up and Shut-down*** A GT plant can be started up as well as shut down quickly, like a diesel engine. Thus it is eminently suitable to meet the peak load demand of a certain region.

**(6)** ***High Efficiency*** Suitable blade cooling permits the use of high GT inlet temperature (as high as 1300°C) yielding a high thermal efficiency (on the order of 37%).




Simple closed-cycle GT plant

**(7)** ***Combined Cycle Mode*** A GT plant can be used in conjunction with a bottoming steam plant in the combined cycle mode to yield an overall fuel-to-electricity efficiency of 55%.

**(8)** ***Cooling Water*** The requirement of cooling water is not much. Water availability is not a restriction for installing a GT plant.

**(9)** ***Lubricating Oil*** The absence of rubbing and reciprocating parts does not warrant high consumption of lubricating oil and there is no problem of balancing as in a reciprocating engine.

**(10)** ***Ash Disposal*** In a thermal power station ash disposal from the site often poses a serious problem. This is not so in a GT power plant.

**(11)** ***Transmission Loss*** It can be located at the load centre itself. Therefore, the transmission loss is minimal in such a plant.

**(12)** ***Cost of Installation*** The installation cost is much less compared to a thermal plant. Only a foundation is required. The whole plant comes from the factory to the site, almost fully assembled.

**(13)** ***Scope of Cogeneration*** A GT exhaust can be used to produce process heat for various uses.

**(14)** ***Low Capital Cost*** GT plants are available in standard sizes. The capital cost per kW is considerably less than a thermal plant.

## 21.3 DISADVANTAGES OF A GT PLANT

These are enlisted below:

1. Part load efficiency is low
2. Highly sensitive to component efficiency like $\eta_c$ and $\eta_T$