Lecture 8

DIESEL CYCLE

INTERNAL COMBUSTION ENGINES

By

Lecturer

Dr.Hyder H. Abd

Automobile Department
Engineering Technical College / Najaf
Al-furat Al-awsat Technical University

3.1.2 DIESEL CYCLE

Early CI engines injected fuel into the combustion chamber very late in the compression stroke, resulting in the indicator diagram shown in Fig. 3-7. Due to ignition delay and the finite time required to inject the fuel, combustion lasted into the expansion stroke. This kept the pressure at peak levels well past TDC. This combustion process is best approximated as a constant-pressure heat input in an air-standard cycle, resulting in the **Diesel cycle** shown in Fig. 3-8. The rest of the cycle is similar to the air-standard Otto cycle. The diesel cycle is sometimes called a **Constant· Pressure cycle**.

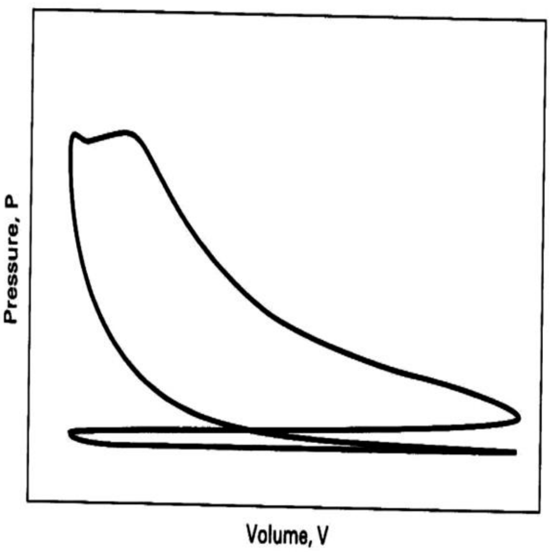

Figure 3-7 Indicator diagram of a historic CI engine operating on an early four-stroke cycle.

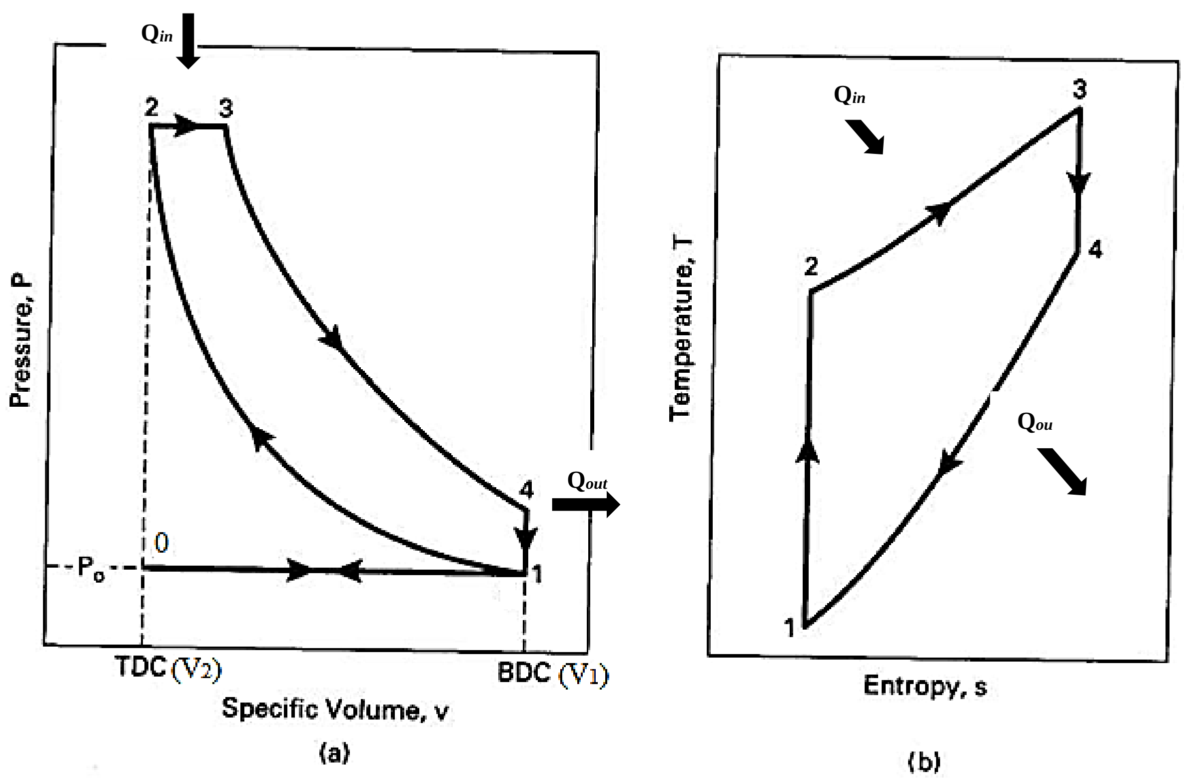

Figure 3-8 Air-standard diesel cycle, 0-1-2-3-4-1-0, which approximates the four-stroke cycle of an early CI engine on **(a)** pressure-specific volume coordinates, and **(b)** temperature-entropy coordinates.

Thermodynamic Analysis of Air-Standard Diesel Cycle

Process 0-1—constant-pressure intake of air at P_o.
Intake valve open and exhaust valve closed:

$$w_{0\text{-}1} = P_o(v_1 - v_o) \tag{3-51}$$

Process 1-2—isentropic compression stroke.
All valves closed:

$$T_2 = T_1(v_1/v_2)^{k-1} = T_1(V_1/V_2)^{k-1} = T_1(r_c)^{k-1} \tag{3-52}$$

$$P_2 = P_1(v_1/v_2)^k = P_1(V_1/V_2)^k = P_1(r_c)^k \tag{3-53}$$

$$V_2 = V_{TDC} \tag{3-54}$$

$$q_{1-2} = 0 \tag{3-55}$$

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1-k) = R(T_2 - T_1)/(1-k) \tag{3-56}$$

$$= (u_1 - u_2) = c_v(T_1 - T_2)$$

Process 2-3—constant-pressure heat input (combustion).
All valves closed:

$$Q_{2-3} = Q_{in} = m_f Q_{HV}\eta_c = m_m c_p(T_3 - T_2) = (m_a + m_f)c_p(T_3 - T_2) \tag{3-57}$$

$$Q_{HV}\eta_c = (AF + 1)c_p(T_3 - T_2) \tag{3-58}$$

$$q_{2-3} = q_{in} = c_p(T_3 - T_2) = (h_3 - h_2) \tag{3-59}$$

$$w_{2-3} = q_{2-3} - (u_3 - u_2) = P_2(v_3 - v_2) \tag{3-60}$$

$$T_3 = T_{max} \tag{3-61}$$

Cutoff ratio is defined as the change in volume that occurs during combustion, given as a ratio:

$$\beta = V_3/V_2 = v_3/v_2 = T_3/T_2 \tag{3-62}$$

Process 3-4—isentropic power or expansion stroke.
All valves closed:

$$q_{3-4} = 0 \tag{3-63}$$

$$T_4 = T_3(v_3/v_4)^{k-1} = T_3(V_3/V_4)^{k-1} \tag{3-64}$$

$$P_4 = P_3(v_3/v_4)^k = P_3(V_3/V_4)^k \tag{3-65}$$

$$w_{3-4} = (P_4 v_4 - P_3 v_3)/(1-k) = R(T_4 - T_3)/(1-k) \tag{3-66}$$

$$= (u_3 - u_4) = c_v(T_3 - T_4)$$

Process 4-1—constant-volume heat rejection (exhaust blowdown).
Exhaust valve open and intake valve closed:

$$v_4 = v_1 = v_{BDC} \tag{3-67}$$

$$w_{4-1} = 0 \tag{3-68}$$

$$Q_{4-1} = Q_{out} = m_m c_v(T_4 - T_1) = m_m c_v(T_4 - T_1) \tag{3-69}$$

$$q_{4-1} = q_{out} = c_v(T_4 - T_1) = (u_4 - u_1) \tag{3-70}$$

Process 0-1—constant-pressure exhaust stroke at P_o.
Exhaust valve open and intake valve closed:

$$w_{1-0} = P_o(v_0 - v_1) = P_o(v_0 - v_1) \tag{3-71}$$

The thermal efficiency of Diesel cycle:

$$(\eta_t)_{DIESEL} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \tag{3.72}$$

$$= 1 - \frac{c_v(T_4 - T_1)}{c_P(T_3 - T_2)}$$

$$= 1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)}$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} \rightarrow T_2 = T_1(r_c)^{k-1} \rightarrow T_1 = \frac{T_2}{(r_c)^{k-1}} \tag{3.73}$$

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1} = \left(\frac{v_2\beta}{v_1}\right)^{k-1} \rightarrow \left(\frac{\beta}{r_c}\right)^{k-1} \quad T_4 = T_3\left(\frac{\beta}{r_c}\right)^{k-1} \tag{3.74}$$

$$\mathrm{k} = \frac{c_P}{c_v}\ , \ \ \beta = \frac{T_3}{T_2} \rightarrow \boldsymbol{T_3 = T_2\,\beta} \quad *$$

$$(\eta_t)_{DIESEL} = 1 - \frac{\frac{1}{(r_c)^{k-1}}\ (T_3\ \beta^{k-1} - T_2)}{k(T_3 - T_2)} \qquad (3.75)$$

By substitution eq.(*) in eq. (3.75) we get:

$$(\eta_t)_{DIESEL} = 1 - \frac{1}{(r_c)^{k-1}}\left[\frac{T_2 \quad (\beta^k - 1)}{T_2\ k \quad (\beta - 1)}\right] \qquad (3.76)$$

$$(\eta_t)_{DIESEL} = 1 - \frac{1}{(r_c)^{k-1}}\left[\frac{(\beta^k - 1)}{k \quad (\beta - 1)}\right] \qquad (3.77)$$

If representative numbers are introduced into Eq. (3-77), it is found that the value of the term in brackets is greater than one. When this equation is compared with Eq. (3-31), it can be seen that for a given compression ratio the thermal efficiency of the Otto cycle would be greater than the thermal efficiency of the Diesel cycle. Constant-volume combustion at TDC is more efficient than constant-pressure combustion. However, it must be remembered that CI engines operate with much higher compression ratios than SI engines (12 to 24 versus 8 to 11) and thus have higher thermal efficiencies.