Chapter Three

Engine Cycles

Introduction

1- The cycle experienced in the cylinder of an internal combustion engine is very complex. The cycle in SI and diesel engine were discussed in detail in the previous chapter.

2- Instead ideal cycles were assumed to make the process more manageable.

3- Ideal cycle resemble true cycle but it is made of a lot of assumption.

Air-Standard Cycles

Ideal Air-standard cycle : Differences between Air-standard cycle and actual engine cycle

1. Gas mixture in the cylinder is treated as air (an ideal gas with constant specific heats) for the entire cycle, and property values of air are used in the analysis.

- the first half of cycle, air with only up to about 7% fuel vapor
- the second half of cycle, mostly CO2 , H2O and N2 doesn't great large errors

2. Real open cycle is changed into a closed cycle by assuming that the gases being exhausted are fed back into the intake system.

- both intake and exhaust gases are air, composition doesn't change

3. Combustion process is replaced with a heat addition process Q in of equal energy value ***(air alone cannot combust)***

4. Open exhaust process, which carries a large amount of enthalpy out of the system, is replaced with a closed system heat rejection process ***Q*** out of equal energy value

5. Actual engine processes are approximated with ideal process (a) almost-constant-pressure intake and exhaust strokes are assumed to be constant pressure

(b) compression and expansion strokes are approximated by isentropic processes (reversible and adiabatic) – surface frictions and gas motion friction and heat transfer (ignored)

(c) the combustion process is idealized by a constant-volume process(SI cycle) and a constant-pressure process(CI cycle), or a combination of both (CI dual cycle)

(d) exhaust blow down is approximated by a constant-volume process

(e) all processes are considered reversible

$$Pv = RT$$

$$PV = mRT$$

$$P = \rho RT$$

$$dh = c_P dT$$

$$du = c_v dT$$

$$Pv^k = \text{constant} \qquad \text{isentropic process}$$

$$Tv^{k-1} = \text{constant} \qquad \text{isentropic process}$$

$$TP^{(1-k)/k} = \text{constant} \qquad \text{isentropic process}$$

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1 - k) \qquad \text{isentropic work in closed system}$$

$$= R(T_2 - T_1)/(1 - k)$$

$$c = \sqrt{kRT} \qquad \text{speed of sound}$$

where:
P = gas pressure in cylinder
V = volume in cylinder
v = specific volume of gas
R = gas constant of air
T = temperature
m = mass of gas in cylinder
ρ = density
h = specific enthalpy
u = specific internal energy
c_p, c_v = specific heats
$k = c_p/c_v$
w = specific work
c = speed of sound

$$c_p = 1.108 \text{ kJ/kg-K} = 0.265 \text{ BTU/lbm-°R}$$

$$c_v = 0.821 \text{ kJ/kg-K} = 0.196 \text{ BTU/lbm-°R}$$

$$k = c_P/c_v = 1.108/0.821 = 1.35$$

$$R = c_P - c_v = 0.287 \text{ kJ/kg-K}$$

$$= 0.069 \text{ BTU/lbm-°R} = 53.33 \text{ ft-lbf/lbm-°R}$$

AF = air–fuel ratio

$\dot{m}$ = mass flow rate

q = heat transfer per unit mass for one cycle

$\dot{q}$ = heat transfer rate per unit mass

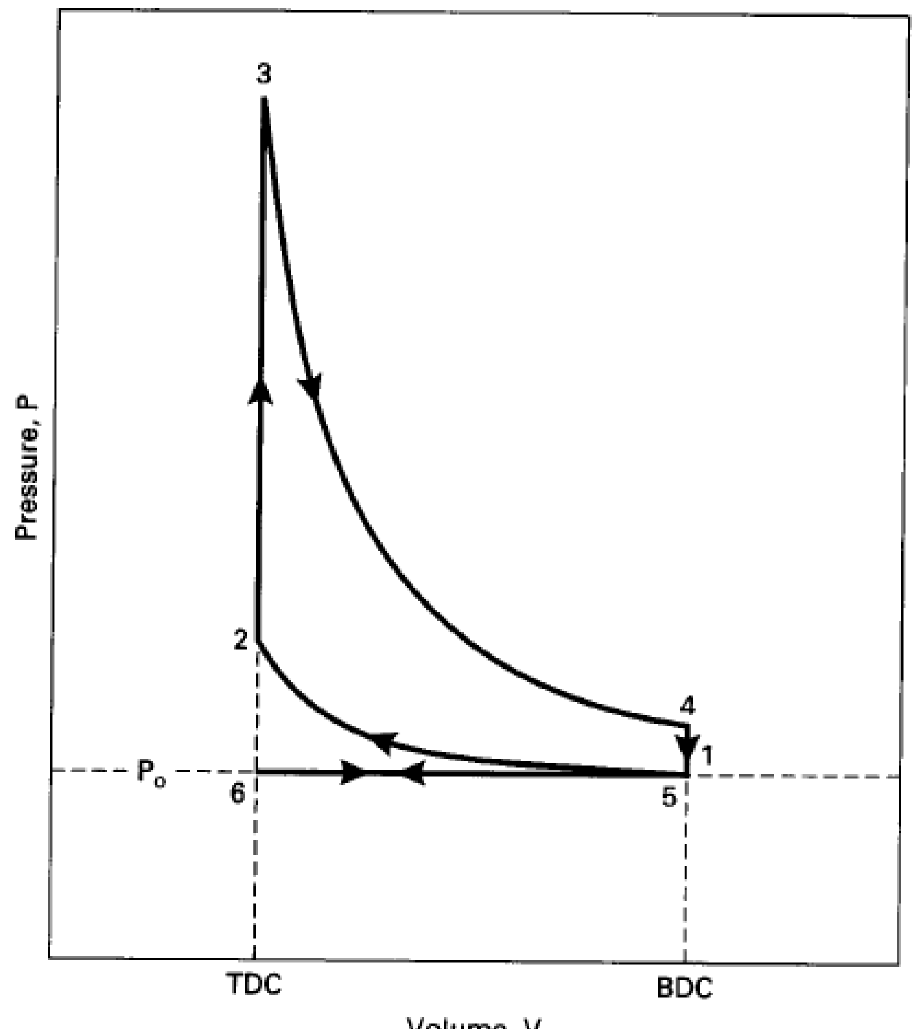
Q = heat transfer for one cycle

$\dot{Q}$ = heat transfer rate

Q_{HV} = heating value of fuel

Otto cycle

The cycle of a four-stroke, SI, naturally aspirated engine at WOT. This is the cycle of most automobile engines and other four-stroke SI engines.

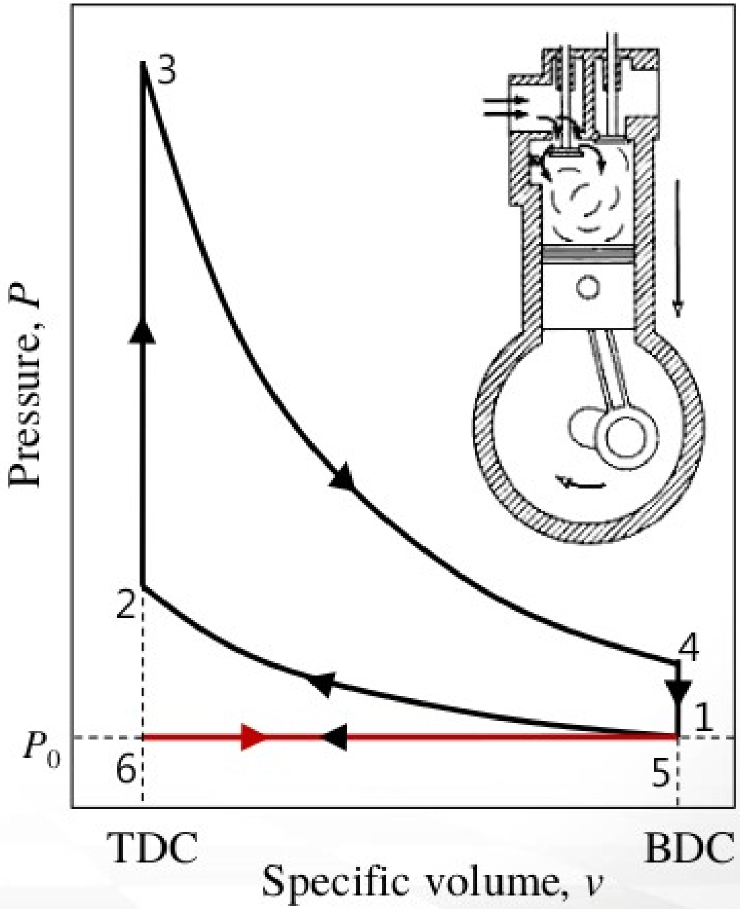


- Process 6-1: Constant pressure

  Intake valve open and exhaust valve closed

$$P_1 = P_6 = P_0$$

$$w_{6-1} = P_0\left(v_1 - v_6\right)$$



Real cycles
◆ Actual pressure > P_0 ◆ Temperature rise bet' n 25-35° C

- Process 1-2: Isentropic

  All valves closed

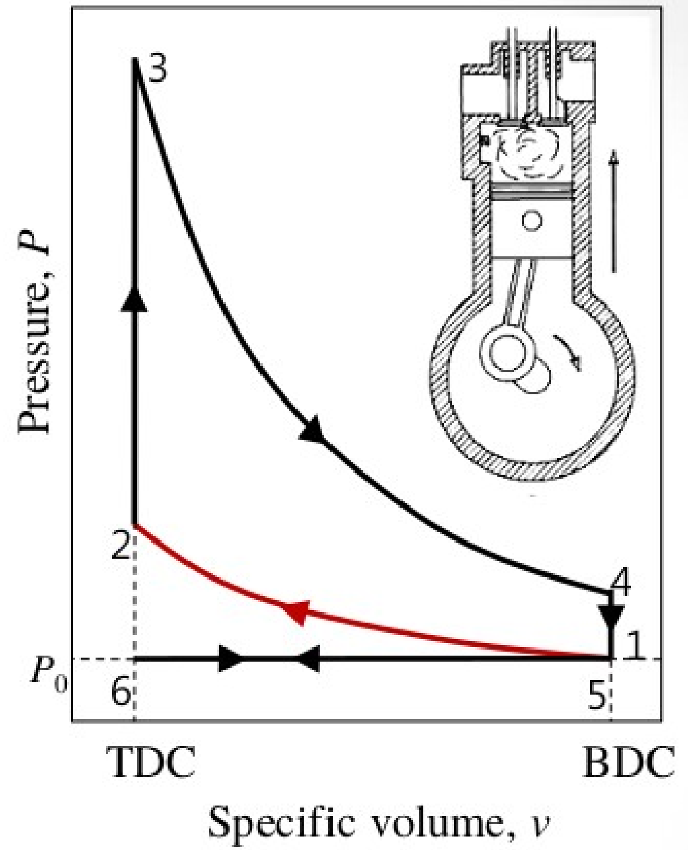
$$T_2 = T_1\left(v_1 / v_2\right)^{k-1} = T_1\left(r_c\right)^{k-1}$$

$$P_2 = P_1\left(v_1 / v_2\right)^{k} = P_1\left(r_c\right)^{k}$$

$$q_{1-2} = 0$$

$$w_{1-2} = \frac{P_2 v_2 - P_1 v_1}{1-k} = \frac{R\left(T_2 - T_1\right)}{1-k}$$

$$= u_1 - u_2 = c_v\left(T_1 - T_2\right)$$



Real cycles
- Beginning: Slight opening of intake vale aBDC - End: Firing of spark plug bTDC

Process 2-3: Constant-volume heat input

All valves closed

$$v_3 = v_2 = v_{\mathrm{TDC}}$$

$$w_{2-3} = 0$$

$$Q_{2-3} = Q_{\mathrm{in}} = m_f Q_{HV} \eta_c$$

$$= m_m c_v (T_3 - T_2)$$

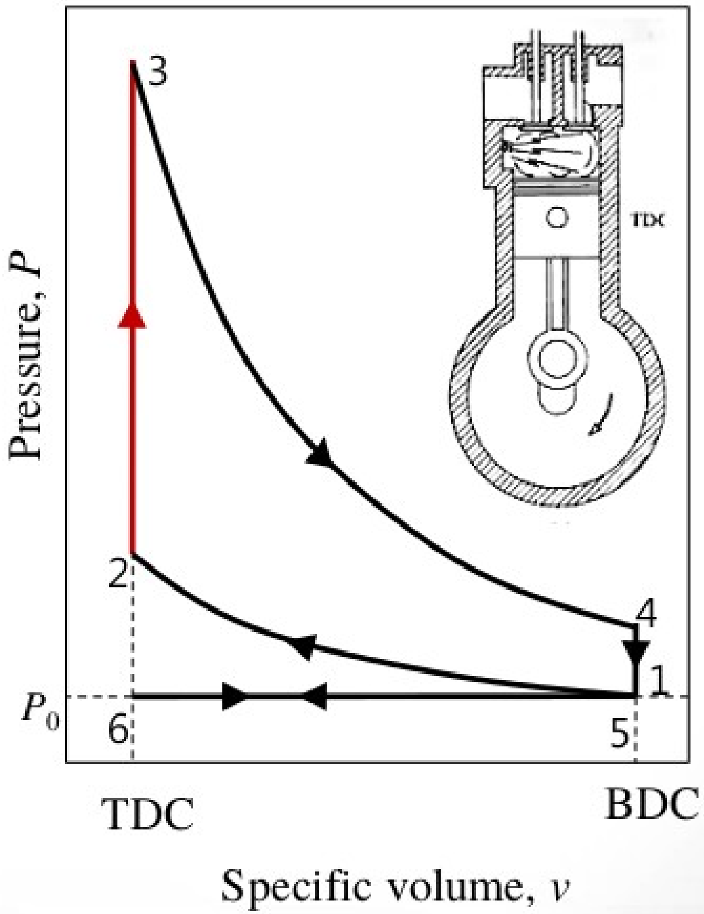
$$= (m_a + m_f) c_v (T_3 - T_2)$$

$$Q_{HV} \eta_c = (\mathrm{AF} + 1) c_v (T_3 - T_2)$$

$$q_{2-3} = q_{\mathrm{in}} = c_v (T_3 - T_2) = u_3 - u_2$$

$$T_3 = T_{\max}$$

$$P_3 = P_{\max}$$



- Process 3-4: Isentropic

  All valves closed

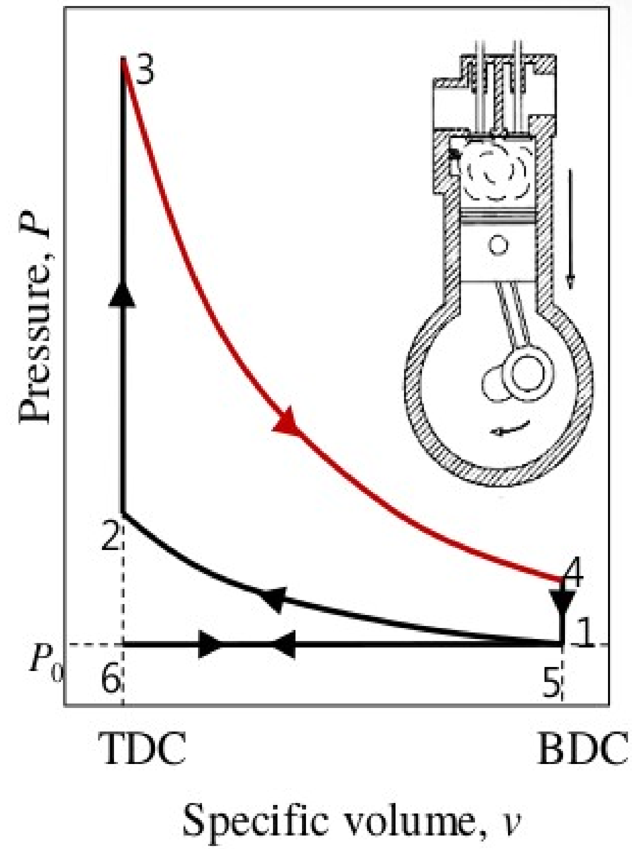
$$q_{3-4} = 0$$

$$T_4 = T_3 \left(v_3 / v_4 \right)^{k-1} = T_3 \left(1 / r_c \right)^{k-1}$$

$$P_4 = P_3 \left(v_3 / v_4 \right)^{k} = P_3 \left(1 / r_c \right)^{k}$$

$$w_{3-4} = \frac{P_4 v_4 - P_3 v_3}{1-k} = \frac{R\left(T_4 - T_3\right)}{1-k}$$

$$= u_3 - u_4 = c_v \left(T_3 - T_4 \right)$$



Real cycles
Beginning: Combustion
End: Exhaust valve opening bBDC

- Process 4-5: Constant-volume heat rejection

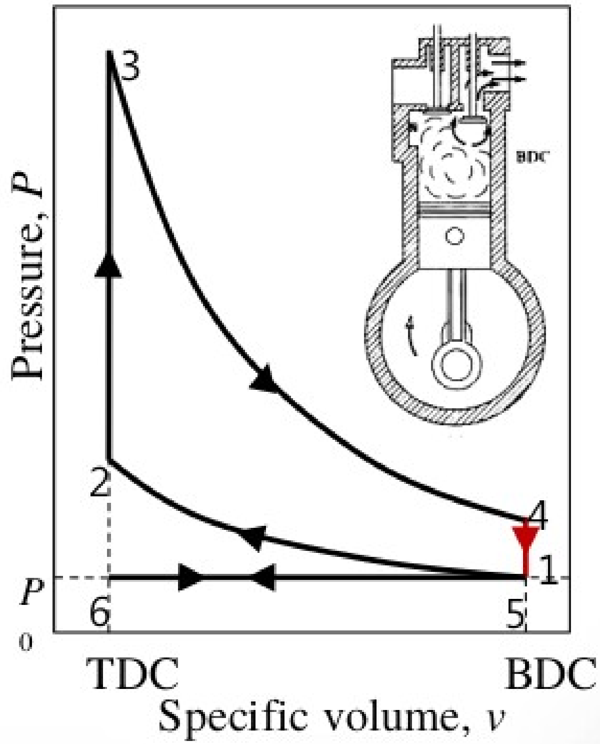
  Exhaust valve open and intake valve closed

$$v_5 = v_4 = v_1 = v_{\mathrm{BDC}}$$

$$w_{4-5} = 0$$

$$Q_{4-5} = Q_{\mathrm{out}} = m_m c_v (T_5 - T_4) = m_m c_v (T_1 - T_4)$$

$$q_{4-5} = q_{\mathrm{out}} = c_v (T_5 - T_4) = u_5 - u_4 = c_v (T_1 - T_4)$$



- Enthalpy carried away, limiting thermal efficiency

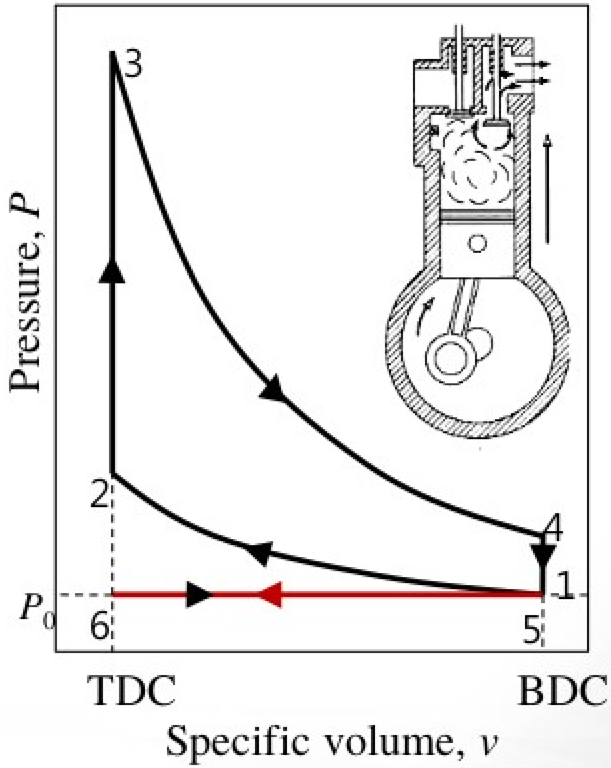
- Process 5-6: Constant pressure

  Exhaust valve open and intake valve closed

$$P_6 = P_5 = P_0$$

$$w_{5-6} = P_0\left(v_6 - v_5\right)$$
$$= P_0\left(v_6 - v_1\right)$$

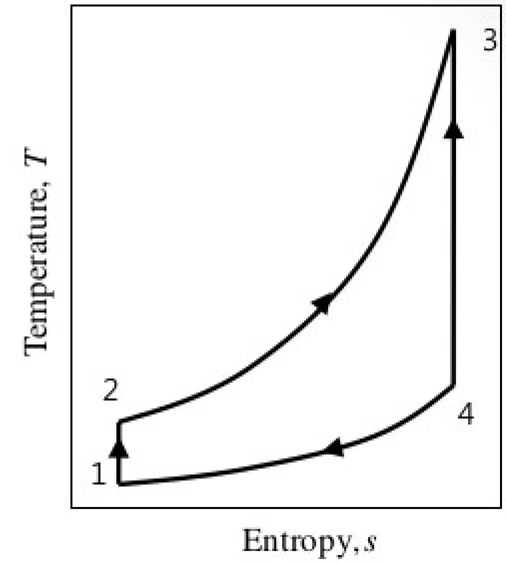
Real cycles
Actual pressure < P_0



- Processes 6-1, 5-6 sometimes left off since these cancel each other

- Thermal efficiency: Depending only on temperature

$$(\eta_t)_{\text{OTTO}} = |w_{net}|/|q_{in}| = 1 - |q_{out}|/|q_{in}|$$
$$= 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

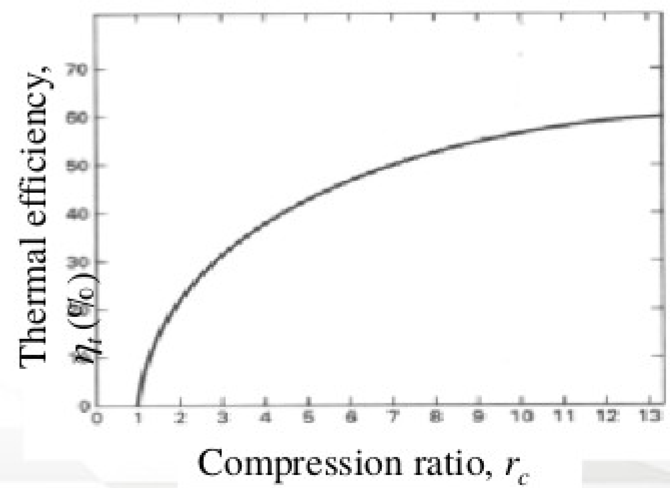


- Using relations for processes 1-2, 3-4

$$T_2/T_1 = (v_1/v_2)^{k-1} = (v_4/v_3)^{k-1} = T_3/T_4$$
$$T_4/T_1 = T_3/T_2$$

- Therefore,

$$(\eta_t)_{\text{OTTO}} = 1 - T_1/T_2$$
$$= 1 - \left[1/(v_1/v_2)^{k-1}\right]$$
$$= 1 - (1/r_c)^{k-1}$$



Diesel cycle

Early CI engines injected fuel into the combustion chamber very late in the compression stroke, resulting in the indicator diagram shown in Figure below. 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 Figure below. The rest of the cycle is similar to the air-standard Otto cycle. The diesel cycle is sometimes called a ***Constant· Pressure cycle.***

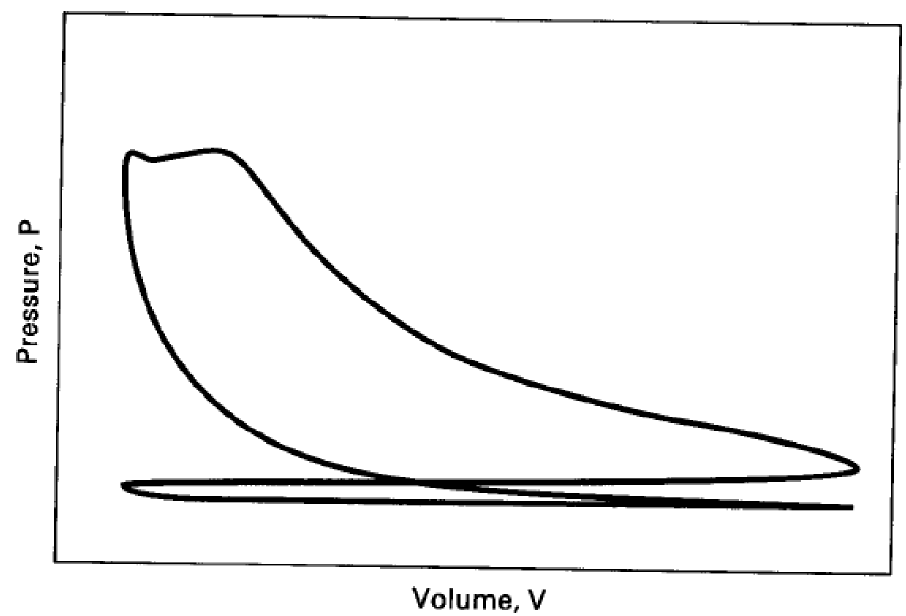


Figure 3-7 Indicator diagram of a historic CI engine operating on an early four-stroke cycle.

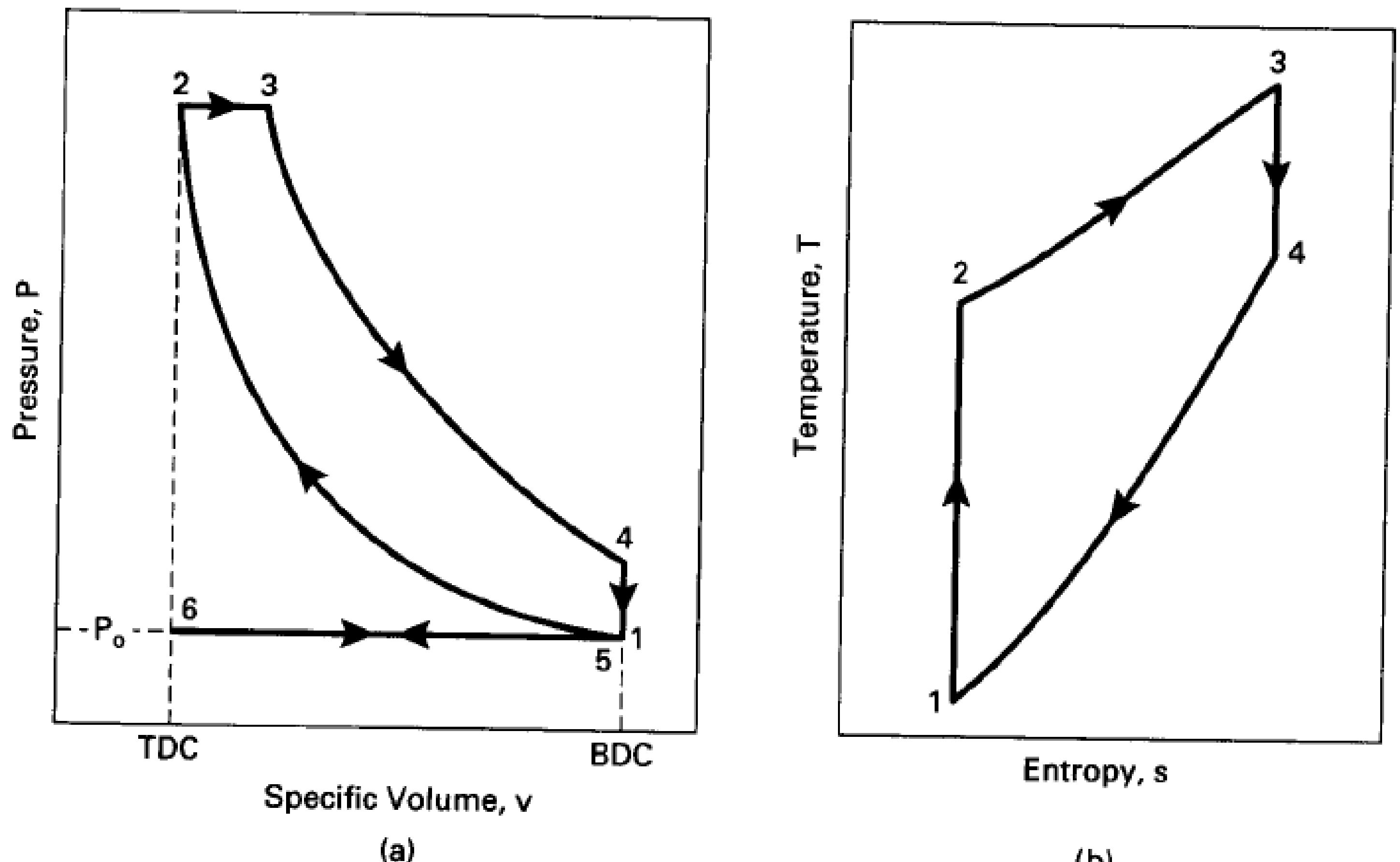


Figure 3-8 Air-standard diesel cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an early CI engine on **(a)** pressure-specific volume coordinates, and **(b)** temperature-entropy coordinates.

Thermodynamic Analysis of Air-Standard Diesel Cycle

Process 6-1—constant-pressure intake of air at P_o.
Intake valve open and exhaust valve closed:

$$w_{6-1} = P_o(v_1 - v_6)$$

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

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

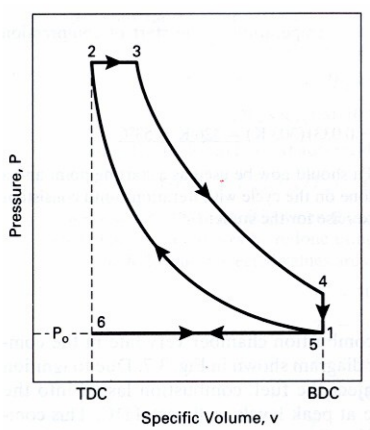
$$P_2 = P_1(v_1/v_2)^k = P_1(V_1/V_2)^k = P_1(r_c)^k$$

$$V_2 = V_{TDC}$$

$$q_{1-2} = 0$$

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

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



(a)

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

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

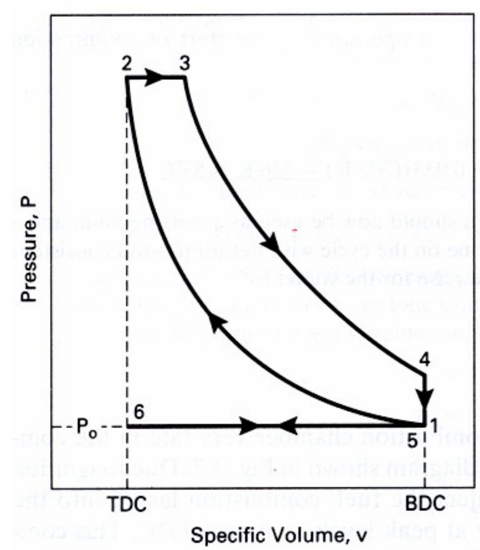
$$Q_{HV}\,\eta_c = (AF + 1)c_p(T_3 - T_2)$$

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

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

$$T_3 = T_{max}$$

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



(a)

$$\beta = V_3/V_2 = v_3/v_2 = T_3/T_2$$

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

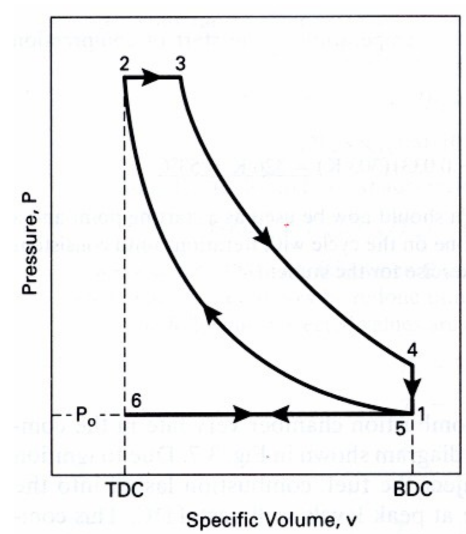
$$q_{3-4} = 0$$

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

$$P_4 = P_3(v_3/v_4)^k = P_3(V_3/V_4)^k$$

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

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

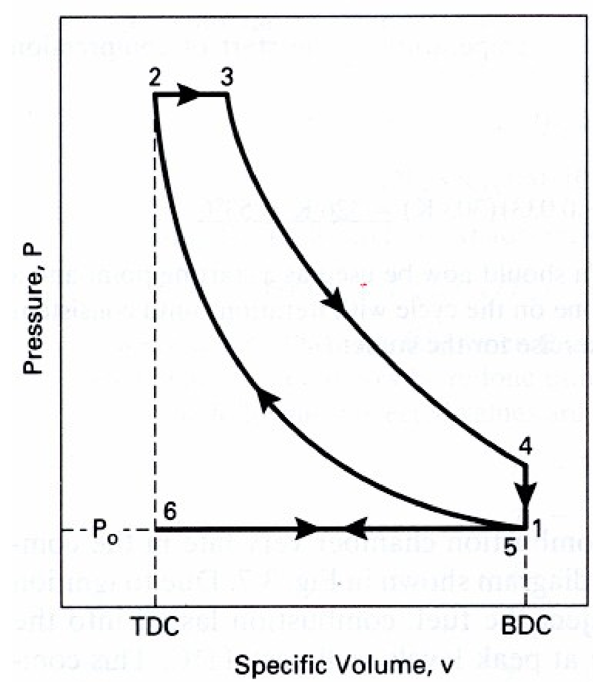


(a)

Process 4-5—constant-volume heat rejection (exhaust blowdown). Exhaust valve open and intake valve closed:

$$v_5 = v_4 = v_1 = v_{BDC}$$

$$w_{4-5} = 0$$



(a)

$$Q_{4-5} = Q_{out} = m_m c_v(T_5 - T_4) = m_m c_v(T_1 - T_4)$$

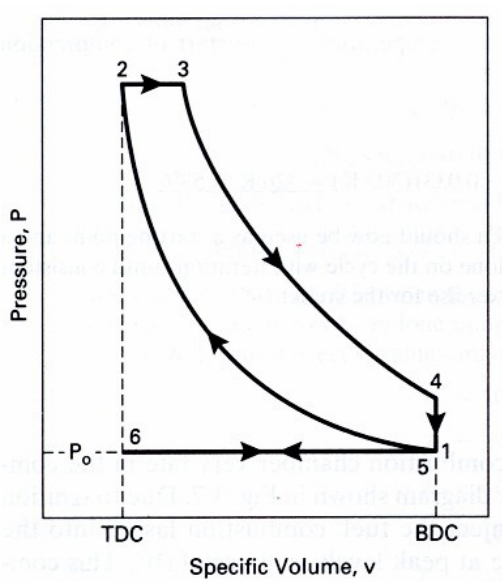
$$q_{4-5} = q_{out} = c_v(T_5 - T_4) = (u_5 - u_4) = c_v(T_1 - T_4)$$

Process 5-6—constant-pressure exhaust stroke at P_o.
Exhaust valve open and intake valve closed:

$$w_{5-6} = P_o(v_6 - v_5) = P_o(v_6 - v_1)$$

Thermal efficiency of diesel cycle:

$$\begin{aligned}(\eta_t)_{\text{DIESEL}} &= |w_{net}|/|q_{in}| = 1 - (|q_{out}|/|q_{in}|) \\ &= 1 - [c_v(T_4 - T_1)/c_p(T_3 - T_2)] \\ &= 1 - (T_4 - T_1)/[k(T_3 - T_2)]\end{aligned}$$



(a)

With rearrangement, this can be shown to equal:

$$(\eta_t)_{\text{DIESEL}} = 1 - (1/r_c)^{k-1}[(\beta^k - 1)/\{k(\beta - 1)\}]$$

where: r_c = compression ratio
$k = c_p/c_v$
β = cutoff ratio

Dual cycle

If equations of the efficiency are compared, it can be seen that to have the best of both worlds, an engine ideally would be compression ignition but would operate on the Otto cycle.

Compression ignition would operate on the more efficient higher compression ratios, while constant-volume combustion of the Otto cycle would give higher efficiency for a given compression ratio. The modern high-speed CI engine accomplishes this in part by a simple operating change from early diesel engines.

Instead of injecting the fuel late in the compression stroke near TDC, as was done in early engines, modern CI engines start to inject the fuel much earlier in the cycle, somewhere around 20° bTDC. The first fuel then ignites late in the compression stroke, and some of the combustion occurs almost at constant volume at TDC, much like the Otto cycle.

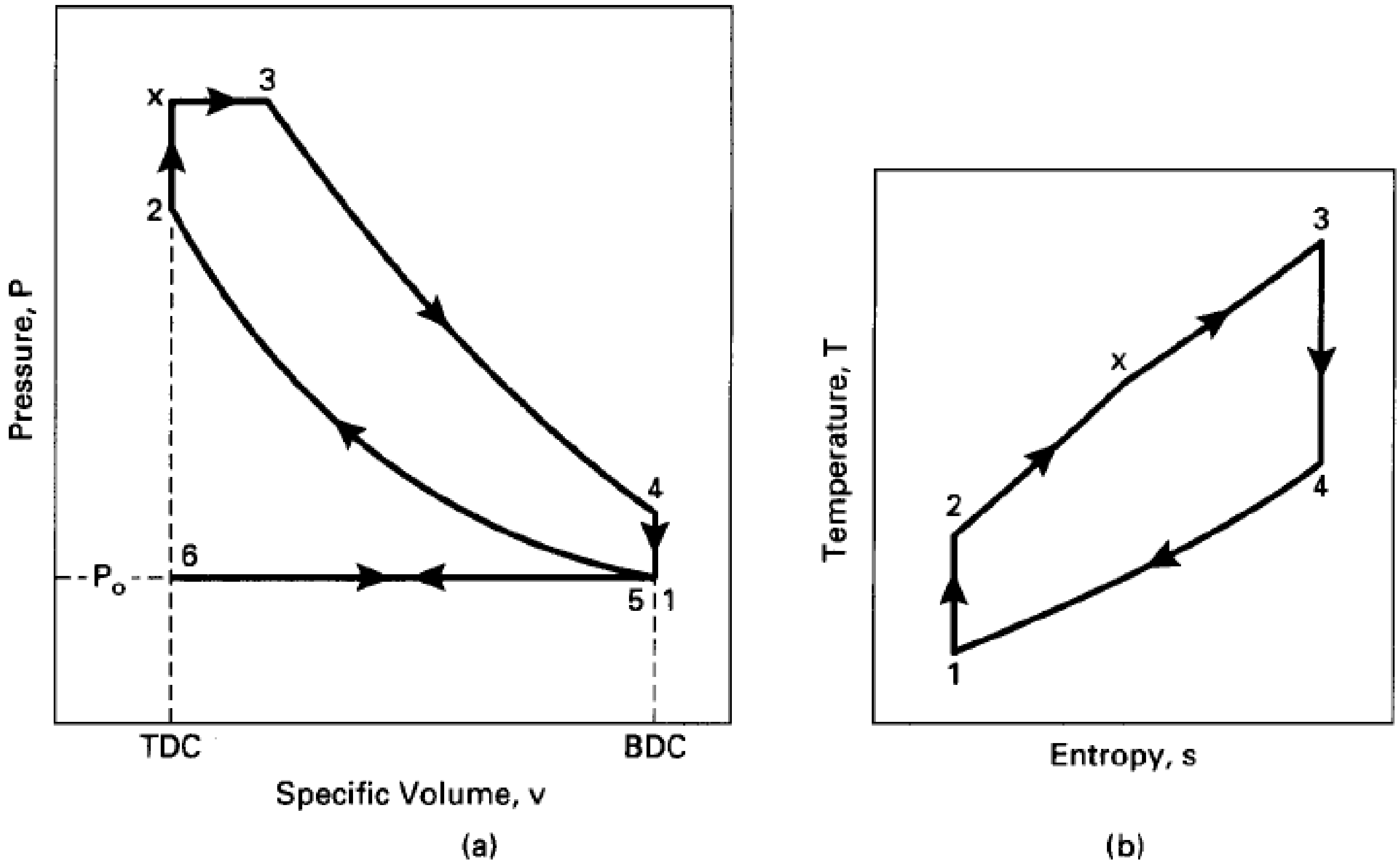


Figure 3-10 Air-standard Dual cycle, 6-1-2-x-3-4-5-6, which approximates the four-stroke cycle of a modern CI engine on **(a)** pressure-specific volume coordinates, and **(b)** temperature-entropy coordinates.

Thermodynamic Analysis of Air-Standard Dual Cycle

The analysis of an air-standard Dual cycle is the same as that of the Diesel cycle except for the heat input process (combustion) *2-x-3*.

Process 2-*x*—constant-volume heat input (first part of combustion).
All valves closed:

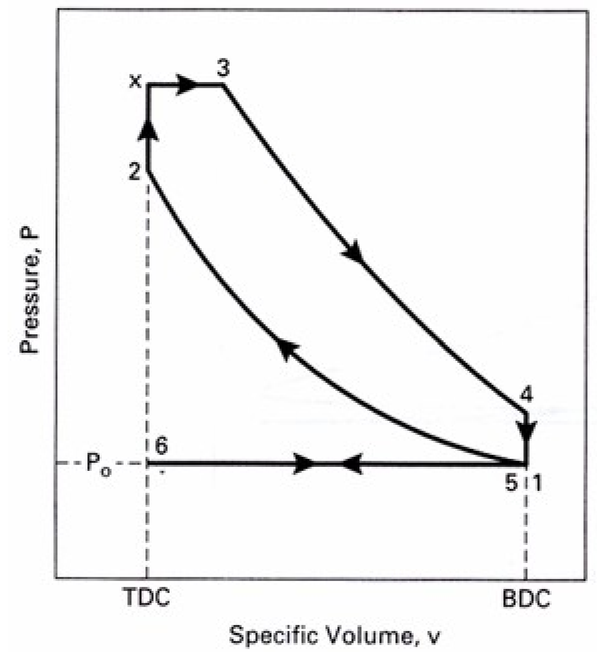
$$V_x = V_2 = V_{\mathrm{TDC}}$$

$$w_{2-x} = 0$$

$$Q_{2-x} = m_m c_v(T_x - T_2) = (m_a + m_f)c_v(T_x - T_2)$$

$$q_{2-x} = c_v(T_x - T_2) = (u_x - u_2)$$

$$P_x = P_{\max} = P_2(T_x/T_2)$$



(a)

Pressure ratio is defined as the rise in pressure during combustion, given as a ratio:

$$\alpha = P_x/P_2 = P_3/P_2 = T_x/T_2 = (1/r_c)^k(P_3/P_1)$$

Process x-3—constant-pressure heat input (second part of combustion)

All valves closed:

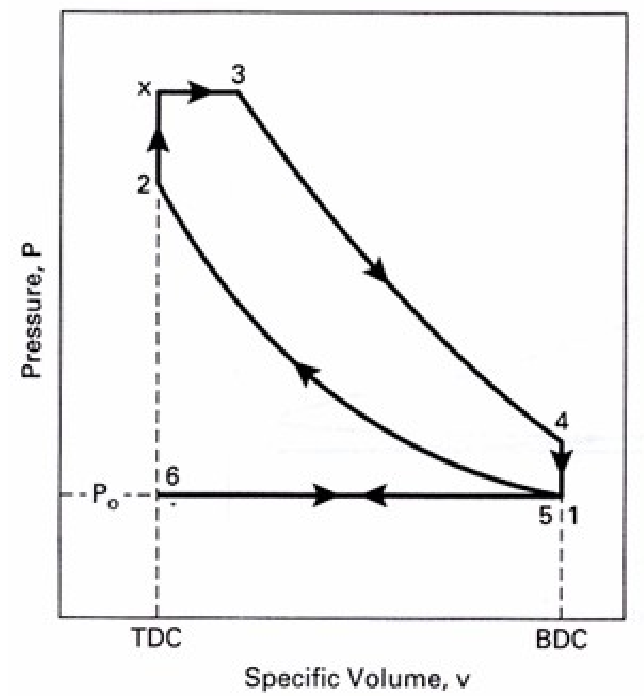
$$P_3 = P_x = P_{max}$$

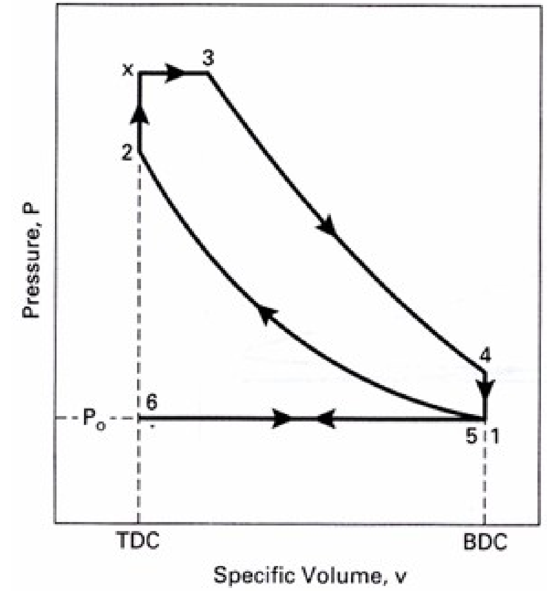
$$Q_{x-3} = m_m c_p(T_3 - T_x) = (m_a + m_f)c_p(T_3 - T_x)$$

$$q_{x-3} = c_p(T_3 - T_x) = (h_3 - h_x)$$

$$w_{x-3} = q_{x-3} - (u_3 - u_x) = P_x(v_3 - v_x) = P_3(v_3 - v_x)$$

$$T_3 = T_{max}$$





(a)

Cutoff ratio:

$$\beta = v_3/v_x = v_3/v_2 = V_3/V_2 = T_3/T_x$$

Heat in:

$$Q_{in} = Q_{2-x} + Q_{x-3} = m_f Q_{HV} \eta_c$$

$$q_{in} = q_{2-x} + q_{x-3} = (u_x - u_2) + (h_3 - h_x)$$

Thermal efficiency of Dual cycle:

$$(\eta_t)_{DUAL} = |w_{net}|/|q_{in}| = 1 - (|q_{out}|/|q_{in}|)$$

$$= 1 - c_v(T_4 - T_1)/[c_v(T_x - T_2) + c_p(T_3 - T_x)]$$

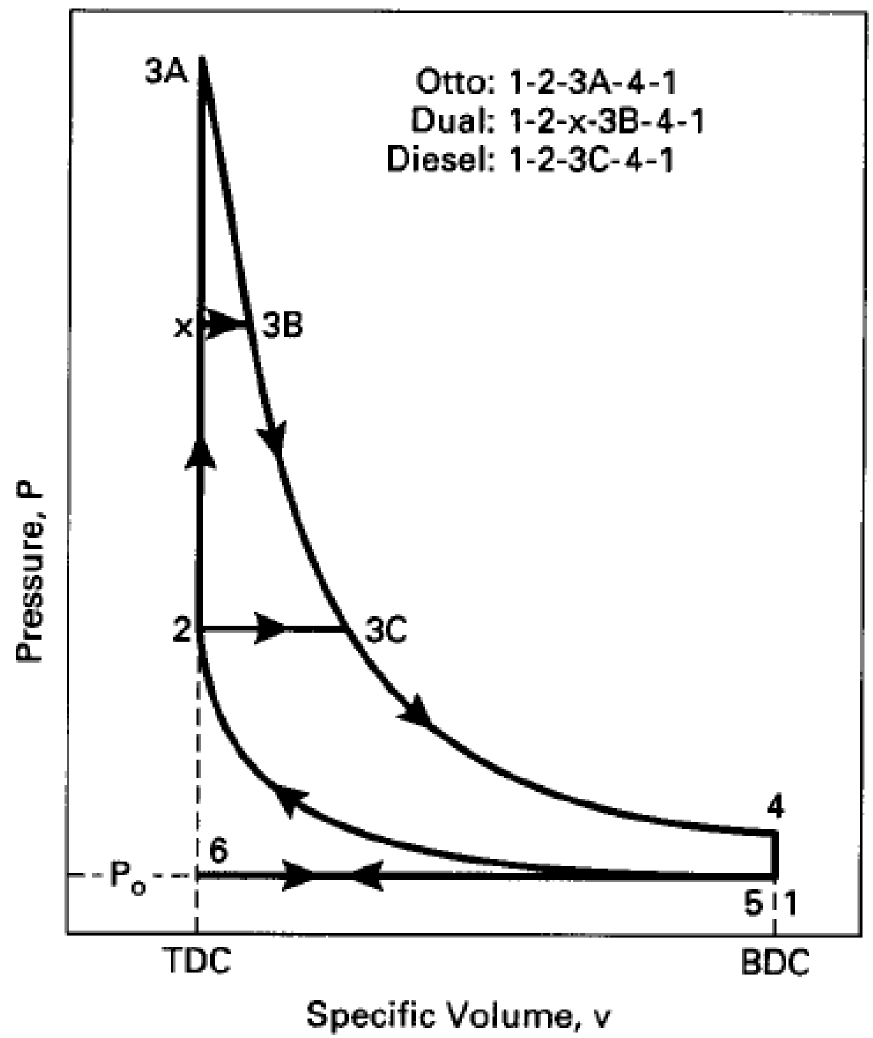
$$= 1 - (T_4 - T_1)/[(T_x - T_2) + k(T_3 - T_x)]$$

This can be rearranged to give:

$$(\eta_t)_{\mathrm{DUAL}} = 1 - (1/r_c)^{k-1}[\{\alpha\beta^k - 1\}/\{k\alpha(\beta - 1) + \alpha - 1\}]$$

where: r_c = compression ratio
$k = c_p/c_v$
α = pressure ratio
β = cutoff ratio

Comparison of Otto, Diesel and Dual cycles



Notes