CHAPTER TWO

ENGINE CYCLES

2-1 Thermodynamics Principles:

2-1-1 For Isentropic Process:

$$P.V = m.R.T \Leftrightarrow P.v = R.T \Leftrightarrow P = \rho.R.T \qquad \ldots (2\text{-}1)$$

$$dh = C_p.dT \;\&\; du = C_v.dT \qquad \ldots (2\text{-}2)$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \ldots (2\text{-}3)$$

$$w_{1-2} = \frac{P_1.v_1 - P_2.v_2}{\gamma - 1} = \frac{R.(T_2 - T_1)}{1-\gamma} \quad \text{(in closed systems)} \qquad \ldots (2\text{-}4)$$

2-1-2 For Polytropic Process:

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \qquad \ldots (2\text{-}5)$$

$$w_{1-2} = \frac{P_1.v_1 - P_2.v_2}{k - 1} = \frac{R.(T_2 - T_1)}{1-k} \qquad \ldots (2\text{-}6)$$

$$\gamma = C_p / C_v \cong k \quad \text{(Specific Heat Ratio) (Index)}$$

$$C = \sqrt{\gamma.R.T} \qquad \ldots (2\text{-}7)$$

Where: $P =$ Gas pressure in cylinder $[KPa]$

$V =$ Gas volume $[m^3]$

$v =$ Specific volume of gas $= 1/\rho$ $[m^3/kg]$

$R =$ Gas constant $[kJ/kg.K]$

$T =$ Temperature $[K]$

$m =$ Mass of gas in cylinder $[kg]$

$\rho =$ Density of gas $[kg/m^3]$

$h =$ Specific enthalpy $[kJ/kg]$

$u =$ Specific internal energy $[kJ/kg]$

$C_p \;\&\; C_v =$ Specific heat at constant pressure and constant volume

$C =$ Speed of sound

$w =$ Specific work $[kJ/kg]$

The Ideal Cycles

2-2 Otto Cycle:

In 1876 Otto build a four-stroke internal combustion engine that compressed the air and gas before ignition.

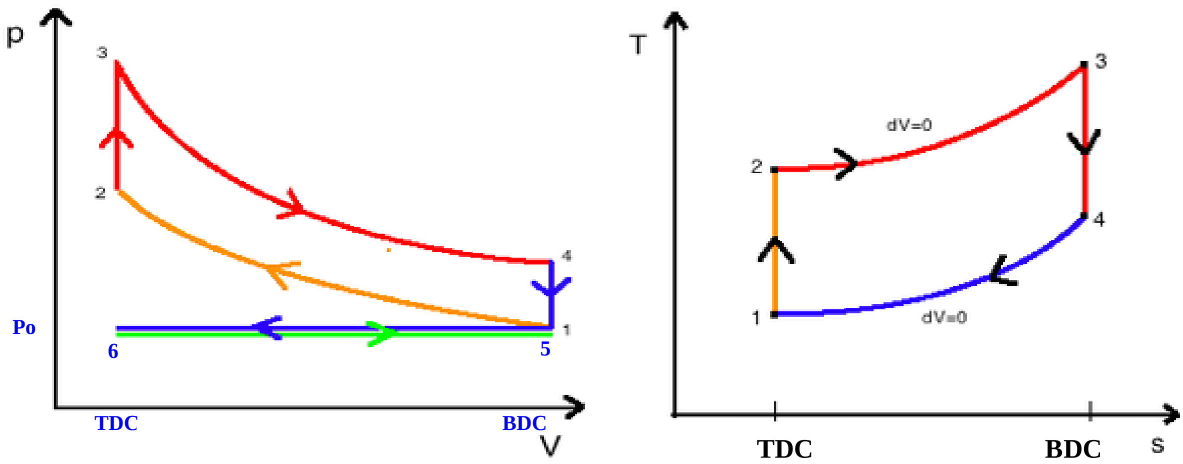


Figure (2-1): Otto Cycle.

Process (6-1): Constant pressure intake of air at (P_o). Intake valve open and exhaust valve is closed.

$$P_1 = P_6 = P_o$$

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

Process (1-2): Isentropic compression stroke. All valves are closed.

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

$$w_{1-2} = \frac{P_1.v_1 - P_2.v_2}{\gamma - 1} = \frac{m_m.(P_2.V_2 - P_1.V_1)}{\gamma - 1} = \frac{R.(T_1 - T_2)}{\gamma - 1}$$

$$w_{1-2} = (u_1 - u_2) = C_v.(T_1 - T_2)$$

m_m = mass of mixture = $m_a + m_f + m_{residual}$

Process (2-3): Constant-volume heat input (combustion). All valves are closed.

$$v_3 = v_2 = v_{TDC}$$
$$w_{2-3} = 0$$

$$Q_{2-3} = Q_A = m_f.H_L.\eta_C = m_m.C_v.(T_3 - T_2) = (m_a + m_f).C_v.(T_3 - T_2)$$
$$H_L.\eta_C = (AF + 1).C_v.(T_3 - T_2)$$

Where: m_m = maximum mass of mixture.

$$q_{2-3} = q_A = C_v.(T_3 - T_2) = (u_3 - u_2)$$
$$T_3 = T_{\max} \ \& \ P_3 = P_{\max}$$

Process (3-4): Isentropic power or expansion stroke. All valves are closed.

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

$$T_4 = T_3.\left(\frac{v_3}{v_4}\right)^{\gamma} = P_3.\left(\frac{v_3}{v_4}\right)^{\gamma-1} = T_3.\left(\frac{1}{r_C}\right)^{\gamma-1}$$

$$P_4 = P_3.\left(\frac{v_3}{v_4}\right)^{\gamma} = P_3.\left(\frac{v_3}{v_4}\right)^{\gamma} = P_3.\left(\frac{1}{r_C}\right)^{\gamma}$$

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

$$w_{3-4} = (u_3 - u_4) = C_v.(T_3 - T_4)$$

Process (4-5): Constant-volume heat rejection (exhaust blow down). Exhaust valve open and intake valve is closed.

$$v_5 = v_4 = v_1 = v_{BDC}$$
$$w_{4-5} = 0$$

$$Q_{4-5} = Q_R = m_m.C_v.(T_5 - T_4) = m_m.C_v.(T_1 - T_4)$$
$$q_{4-5} = q_R = C_v.(T_5 - T_4) = C_v.(T_1 - T_4) = (u_5 - u_4)$$

Process (5-6): Constant-pressure exhaust stroke at (P_o). Exhaust valve open and intake valve is closed.

$$P_5 = P_6 = P_o$$

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

$$\eta_{th})_{otto} = \frac{w_a}{q_A} = 1 - \frac{q_{out}(q_{reject})}{q_A} = 1 - \left[C_v (T_4 - T_1) \Big/ C_v (T_3 - T_2) \right] = 1 - \left[(T_4 - T_1) \Big/ (T_3 - T_2) \right]$$

$$T_2 / T_1 = \left(v_1 / v_2 \right)^{\gamma - 1} = \left(v_4 / v_3 \right)^{\gamma - 1} = T_3 / T_4$$

$$\therefore T_4 / T_1 = T_3 / T_2$$

$$\boxed{\eta_{th})_{otto} = 1 - \frac{T_1}{T_2}}$$

Using the above equations:

$$\boxed{\eta_{th})_{otto} = 1 - \left(\frac{1}{r_C} \right)^{\gamma - 1}} \qquad \text{… (2-8)}$$

<u>Example (1)</u>: A four-cylinder, 2.5 liter, and SI automobile engine operates at WOT on four-stroke air standard Otto cycle at 3000 RPM. The engine has a compression ratio of 8.6:1, a mechanical efficiency of 86% and stroke-to-stroke ratio $S/B = 1.025$. Fuel is isooctane with AF = 15, a heating value of 44300 (kJ/kg), and combustion efficiency ($\eta_c = 100\%$). At the start of compression stroke, condition in the cylinder combustion chamber are 100 kPa and $60^\circ C$. It can be assumed that there is a 4% exhaust residual left over the previous cycle. Do a complete thermodynamics analysis of this engine. Let $C_v = 0.821 (kJ/kg.K)$

<u>Solution</u>:

For one cylinder:

Displacement volume:

$$V_d = \frac{2.5(liter)}{4} = 0.625(L) = 0.000625(m^3)$$

Clearance volume: by using the equation below:

$$r_C = V_1 / V_2 = V_T / V_C = (V_d + V_C) / V_C = 8.6 = (0.000625 + V_C) / V_C$$

$$\therefore V_C = 0.0000822(m^3) = 0.0822(L) = 82.2(cm^3)$$

Bore and stroke:

$$V_d = \frac{\pi.B^2}{4}.S = \frac{\pi.B^2}{4}.(1.025.B) = 0.000625(m^3)$$

$$Then:$$
$$B = 0.0919(m) = 9.19(cm)$$
$$S = 1.025.B = 0.0942(m) = 9.42(cm)$$

State (1):

$$T_1 = 60^\circ C = 333K \ \& \ P_1 = 100KPa$$
$$V_1 = V_T = V_d + V_c = 0.000625 + 0.0000822 = 0.000707(m^3)$$

Mass of gas in cylinder can be calculated at state (1). The mass within the cylinder will be then the same for the entire cycle.

$$m_m = \frac{P_1.V_1}{R.T_1} = \frac{100(KPa)*0.000707(m^3)}{0.287\left(\frac{kJ}{kg.K}\right)*333(K)} = 0.00074(kg)$$

State (2): The compression stroke (1-2) is isentropic.

$$P_2 = P_1.(r_c)^k = 100.(8.6)^{1.35} = 1826(kPa)$$
$$T_2 = T_1.(r_c)^{k-1} = 333.(8.6)^{1.35-1} = 707(K) = 434(^\circ C)$$
$$V_2 = {(m.R.T_2)}\big/{P_2} = {(0.00074*0.287*707)}\big/{1826} = 0.0000822(m^3) = V_c$$

This is the *clearance volume of one cylinder*, which agrees with the above. Another way of getting this value is by using the following equation:

$$V_2 = {V_1}\big/{r_c} = {0.000707}\big/{8.6} = 0.0000822(m^3) = V_c$$

The *mass of gas mixture* (m_m) in the cylinder is made up of air (m_a), fuel (m_f), and exhaust residual (m_{exh}):

$$m_a = \left(\frac{15}{16}\right)*(0.96)*(0.00074) = 0.000666(kg)$$
$$m_f = \left(\frac{1}{16}\right)*(0.96)*(0.00074) = 0.000044(kg)$$
$$m_{exh} = (0.04)*(0.00074) = 0.000030(kg)$$
$$Total :\Rightarrow m_m = m_a + m_f + m_{exh} = 0.000666 + 0.000044 + 0.00003 = 0.00074(kg)$$

State (3):

For the *heat added during one cycle*:

$$Q_{in} = Q_{2-3} = Q_A = m_f . H_L . \eta_C = m_m . C_v . (T_3 - T_2)$$
$$0.000044 * 44300 = 0.00074 * 0.821 * (T_3 - 707)$$

Solving this for (T_3): $\Rightarrow \begin{array}{l} T_3 = 3915(K) = 3642(^\circ C) = T_{\max} \\ V_3 = V_2 = 0.0000822(m^3) \end{array}$

For constant volume: $\Rightarrow P_3 = P_2 . \left(T_3 / T_2 \right) = 1826 * \left(3915 / 707 \right) = 10111(kPa) = P_{\max}$

State (4):

Power stroke (3-4) is isentropic, and then we can find temperature and pressure:

$$T_4 = T_3 . \left(1 / r_C \right)^{\gamma - 1} = 3915 * \left(1 / 8.6 \right)^{1.35-1} = 1844K = 1571^\circ C$$
$$P_4 = P_3 . \left(1 / r_C \right)^{\gamma} = 10111 * \left(1 / 8.6 \right)^{1.35} = 554(kPa)$$
$$V_4 = (m.R.T_4) / P_4 = (0.00074 * 0.287 * 1844) / 554 = 0.000707(m^3) = V_1$$

This agrees with the value of (V_1) found earlier.

Work produced in isentropic power stroke for one cylinder during one cycle:

$$W_{3-4} = \frac{m.R.(T_4 - T_3)}{1-\gamma} = \frac{0.00074 * 0.287 * (1844 - 3915)}{1 - 1.35} = 1.257(kJ)$$

Work absorbed during isentropic compression stroke for one cylinder during one cycle:

$$W_{1-2} = \frac{m.R.(T_2 - T_1)}{1-\gamma} = \frac{0.00074 * 0.287 * (707 - 333)}{1 - 1.35} = -0.227(kJ)$$

Net indicated work for one cylinder during one cycle is:

$$W_{net} = W_{1-2} + W_{3-4} = (+1.257) + (-0.227) = +1.03(kJ)$$

To find *heat added for one cylinder during one cycle*:

$$Q_{in} = Q_A = m_f * Q_{HV} * \eta_C = 0.000044 * 44300 * 1.00 = 1.949(kJ)$$

Indicated thermal efficiency:

$$\eta_{th}\big)_{otto} = 1 - \left(\frac{T_1}{T_2}\right) = 1 - \left(\frac{1}{r_C}\right)^{\gamma-1} = 1 - \left(\frac{333}{707}\right) = 1 - \left(\frac{1}{8.6}\right)^{1.35-1} = 0.529 = 52.9\%$$

Or it could be found:

$$\eta_{th}\big)_{otto} = \frac{W_{net}}{Q_{in}} = \frac{1.03}{1.949} = 0.529 = 52.9\%$$

Indicated mean effective pressure:

$$imep = \frac{W_{net}}{V_1 - V_2} = \frac{W_i}{V_1 - V_2} = \frac{1.03}{0.000707 - 0.0000822} = 1649(kPa)$$

Indicated power at 3000 RPM is obtained by using the equation below:

$$\dot{W}_i = \frac{W.N}{n} = \left[\frac{1.03\left(\frac{kJ}{cylinder - cycle}\right) * \frac{3000}{60}\left(\frac{rev}{\sec}\right)}{2\left(\frac{rev}{cycle}\right)}\right] * 4(cylinder) = 103(kW) = 138(hp)$$

Mean piston speed:

$$\overline{U_P} = 2.S.N = 2\left(\frac{strokes}{rev}\right) * 0.0942\left(\frac{m}{stroke}\right) * \frac{3000}{60}\left(\frac{rev}{\sec}\right) = 9.42\left(m/\sec\right)$$

Brake work for one cylinder during one cycle:

$$W_b = \eta_m * W_i = 0.86 * 1.03 = 0.886(kJ)$$

Brake power at 3000 RPM:

$$\dot{W}_b = \eta_m * \dot{W}_i = 0.86 * 103 = 88.6(kJ)$$

Torque is calculated from the equation:

$$T = \frac{\dot{W}_b}{2.\pi.N} = \frac{88.6\left(\frac{kJ}{\sec}\right)}{2 * \pi\left(\frac{radians}{rev}\right) * \frac{3000}{60}\left(\frac{rev}{\sec}\right)} = 0.282(kN.m) = 282(N.m)$$

Friction power lost:

$$\dot{W}_{lost} = \dot{W}_i - \dot{W}_b = 103 - 88.6 = 14.4(kW) = 19.3(hp)$$

Brakes mean effective pressure:

$$bmep = \eta_m * imep = 0.86 * 1649 = 1418(kPa)$$

Brake specific power:

$$BSP = \frac{\dot{W}_b}{A_P} = \frac{88.6(kW)}{\left[\frac{\pi}{4} * (9.19cm)^2\right] * 4(cylinder)} = 0.334\left(kW/cm^2\right)$$

Output per displacement:

$$OPD = \frac{\dot{W}_b}{V_d} = \frac{88.6(kW)}{2.5(L)} = 35.4\left(kW/L\right)$$

Brake specific fuel consumption:

$$bsfc = \frac{\dot{m}_f}{\dot{W}_b} = \frac{0.000044\left(\frac{kg}{cylinder-cycle}\right) * 50\left(\frac{rev}{\sec}\right) * 0.5\left(\frac{cycle}{rev}\right) * 4(cylinder)}{88.6(kW)}$$

$$bsfc = 0.00005\left(\frac{kg}{kW.\sec}\right) = 180\left(\frac{gm}{kW.hr}\right)$$

Volumetric efficiency:

$$\eta_V = \frac{m_a}{\rho_a * V_d} = \frac{0.000666(kg)}{1.181\left(\frac{kg}{m^3}\right) * 0.000625(m^3)} = 0.902 = 90.2\%$$

2-3 Diesel Cycle:

In 1890/1892 Rudolf Diesel and Akroyd Stuart planed and produced a new type of engine which was burning coal dust as fuel.

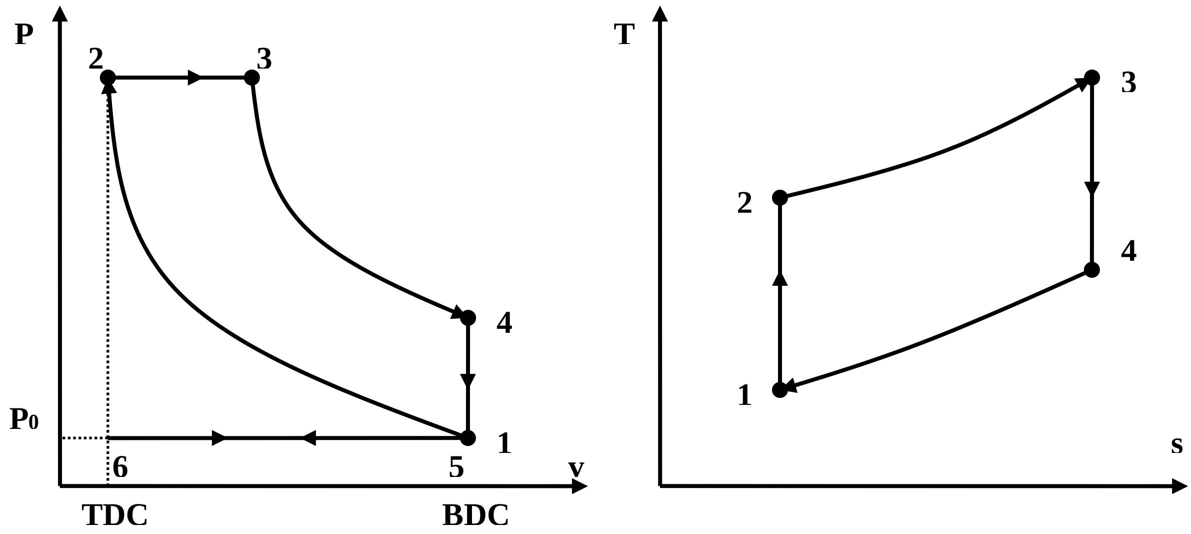


Figure (2-3): Diesel Cycle.

Air-standard:

Process (6-1): Constant pressure intake of air at (P_o). Intake valve open and exhaust valve is closed.

$$P_1 = P_6 = P_o$$

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

Process (1-2): Isentropic compression stroke. All valves are closed.

$$V_2 = V_{TDC} \ \& \ q_{1-2} = 0$$

$$w_{1-2} = \frac{P_2.v_2 - P_2.v_2}{1-\gamma} = \frac{R.(T_2 - T_1)}{1-\gamma}$$

$$w_{1-2} = (u_1 - u_2) = C_v.(T_1 - T_2)$$

Process (2-3): Constant-pressure heat input (combustion). All valves are closed.

$$P_3 = P_2$$

$$Q_{2-3} = Q_A = m_f.H_L.\eta_C = m_m.C_p.(T_3 - T_2) = (m_a + m_f).C_p.(T_3 - T_2)$$
$$H_L.\eta_C = (AF + 1).C_p.(T_3 - T_2)$$
$$q_{2-3} = q_A = C_p(T_3 - T_2) = .(h_3 - h_2)$$

$$w_{2-3} = q_{2-3} - .(u_3 - u_2) = P_2..(v_3 - v_2)$$
$$T_3 = T_{\max}$$

Process (3-4): Isentropic power or expansion stroke. All valves are closed.

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

$$w_{3-4} = \frac{P_4.v_4 - P_3.v_3}{1-\gamma} = \frac{R.(T_4 - T_3)}{1-\gamma}$$
$$w_{3-4} = (u_3 - u_4) = C_v.(T_3 - T_4)$$

Process (4-5): Constant-volume heat rejection (exhaust blow down). Exhaust valve open and intake valve is closed.

$$v_5 = v_4 = v_1 = v_{BDC}$$
$$w_{4-5} = 0$$

$$Q_{4-5} = Q_R = m_m.C_v.(T_5 - T_4) = m_m.C_v.(T_1 - T_4)$$
$$q_{4-5} = q_R = C_v.(T_5 - T_4) = C_v.(T_1 - T_4) = (u_5 - u_4)$$

Process (5-6): Constant-pressure exhaust stroke at (P_o). Exhaust valve open and intake valve is closed.

$$P_5 = P_6 = P_o$$

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

$$\eta_{th})_{Diesel} = \frac{w_a}{q_A} = 1 - \frac{q_{out}}{q_A} = 1 - \left[C_v (T_4 - T_1) \Big/ C_p (T_3 - T_2) \right] = 1 - \left[(T_4 - T_1) \Big/ \gamma.(T_3 - T_2) \right]$$

$$\eta_{th})_{Diesel} = 1 - \frac{1}{\gamma} \left[\frac{\left(T_4 / T_1 - 1 \right)}{\left(T_3 / T_2 - 1 \right)} * \frac{T_1}{T_2} \right]$$

$$T_1 / T_2 = \left(v_2 / v_1 \right)^{\gamma - 1} \ \& \left(v_3 / v_4 \right)^{\gamma - 1} = T_4 / T_3$$

$$v_3 / v_2 = T_3 / T_2 \ \& \ v_4 = v_1$$

$$\frac{T_4}{T_1} = \frac{T_3 \left(\frac{v_3}{v_4} \right)^{\gamma - 1}}{T_2 \left(\frac{v_2}{v_1} \right)^{\gamma - 1}}$$

With rearrangement, this can be shown to equal:

$$\boxed{\begin{aligned} \eta_{th})_{Diesel} &= 1 - \frac{1}{\gamma} \left[\frac{\left(v_3 / v_2 \right)^{\gamma} - 1}{\left(v_3 / v_2 \right) - 1} \right] * \left(v_2 / v_1 \right)^{\gamma - 1} \\ \therefore \eta_{th})_{Diesel} &= 1 - \left(\frac{1}{r_C} \right)^{\gamma - 1} * \left[\frac{\beta^{\gamma} - 1}{\gamma (\beta - 1)} \right] \end{aligned}} \quad \ldots (2\text{-}9)$$

Where: $\beta = \text{Cutoff ratio} = \frac{v_3}{v_2} = \frac{T_3}{T_2}$

Cutoff ratio is defined as the change in volume that occurs during combustion.

Note: The Diesel cycle efficiency dependent on $(r_c \ \& \ \beta)$. The cutoff ratio lies in the range $(1 < \beta < r_c)$ and is thus always greater than unity. Consequently the expression in square brackets is always greater than unity and the Diesel cycle efficiency is less than Otto cycle efficiency for the same (r_c). In the actual engines; the (r_c) of Diesel engine is usually greater than for petrol engine, so the former is usually more efficient.

2-4 Dual Cycle: it is also called Limited-Pressure or Mixed Cycle.

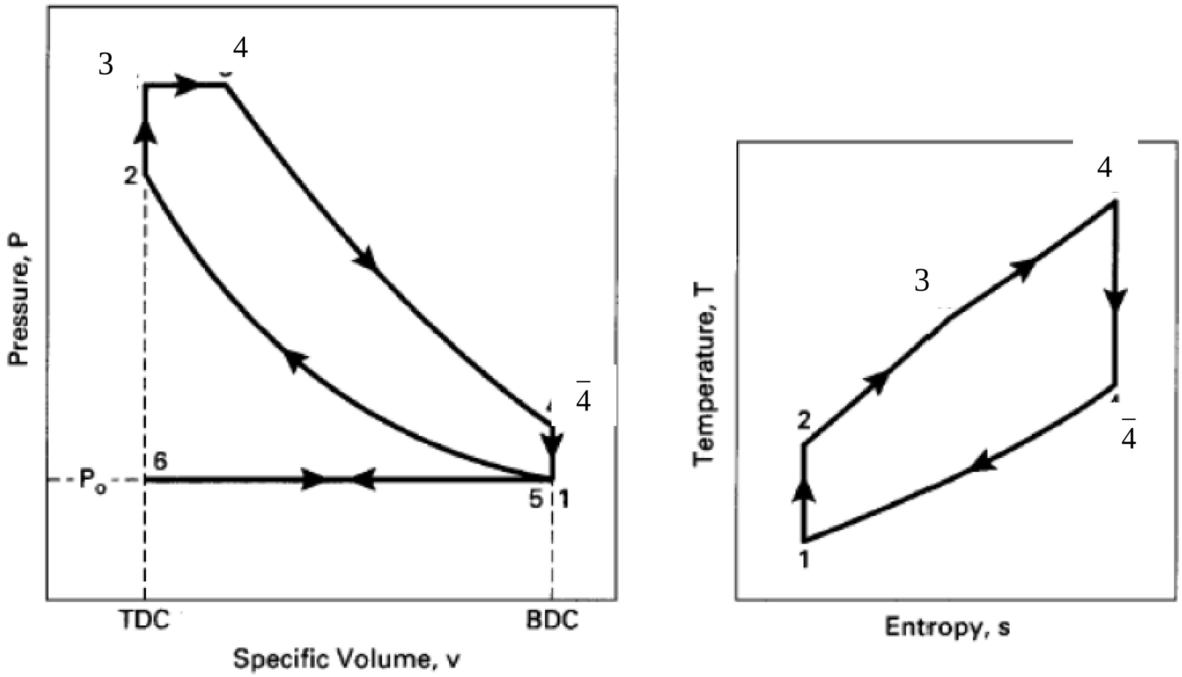


Figure (2-4): Dual Cycle.

$$\eta_{th} = 1 - \frac{Q_R}{Q_A}$$

$$Q_A = C_v.(T_3 - T_2) + C_p.(T_4 - T_3)$$

$$Q_R = C_v.(T_{\bar{4}} - T_1)$$

$$\therefore \eta_{th})_{Dual} = 1 - \frac{T_{\bar{4}} - T_1}{(T_3 - T_2) + \gamma(T_4 - T_3)}$$

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

$$T_3 = T_2.\left(\frac{P_3}{P_2}\right) = T_1.\alpha.(r_C)^{\gamma-1}$$

Where: $\alpha = \left(\frac{P_3}{P_2}\right)$ (pressure ratio or degree of pressure increasing during combustion.

$\frac{v_4}{v_{\bar{4}}} = \frac{\beta}{r_C}$, and hence $T_{\bar{4}} = T_1.\alpha.\beta^{\gamma}$

$$\boxed{\therefore \eta_{th})_{Dual} = 1 - \left(\frac{1}{r_C}\right)^{\gamma-1}.\left[\frac{\alpha\beta^{\gamma} - 1}{(\alpha - 1) + \gamma\alpha(\beta - 1)}\right]}$$

1) If $(\beta = 1)$, the above equation becomes the thermal efficiency of Otto cycle.
2) If $(\alpha = 1)$, the above equation becomes the Diesel cycle efficiency.

$$\text{Now: } \begin{cases} \eta_{th})_{Otto} = f(r_C \,\&\, \gamma) \\ \eta_{th})_{Diesel} = f(r_C, \gamma \,\&\, \beta) \\ \eta_{th})_{Dual} = f(r_C, \gamma, \beta \,\&\, \alpha) \end{cases}$$

<u>Example (2)</u>: A small truck has a four-cylinder, 4 liter CI engine that operates on the air-standard Dual cycle (Fig. 2-4) using light diesel fuel at an air-fuel ratio of 18. The compression ratio of the engine is 16:1 and the cylinder bore diameter is 10 cm. At the start of the compression stroke, conditions in the cylinders are 60°C and 100 KPa with a 2% exhaust residual. It can be assumed that half of the heat input from combustion is added at constant volume and half at constant pressure. If heating value of the diesel fuel is 42500 (kJ/kg), Calculate:

1. Temperature and pressure at each state of the cycle
2. Indicated thermal efficiency
3. Exhaust temperature
4. Engine volumetric efficiency

<u>Solution</u>:

1) For one cylinder:

$$V_d = {}^{4L}\!/_{4} = 1(L) = 0.001\left(m^3\right) = 1000\left(cm^3\right)$$

$$r_C = {}^{V_{BDC}}\!/_{V_{TDC}} = {}^{(V_d + V_c)}\!/_{V_c}$$

$$16 = {}^{(1000+V_c)}\!/_{V_c} \Rightarrow V_c = 66.7\left(cm^3\right) = 0.0667(L) = 0.0000667\left(m^3\right)$$

$$V_d = \frac{\pi . B^2}{4} . S$$

$$0.001\left(m^3\right) = \frac{\pi * [0.1(m)]^2}{4} * S \Rightarrow S = 0.127(m) = 12.7(cm)$$

State (1): $T_1 = 60^\circ C = 333(K) \,\&\, P_1 = 100(kPa)$

$$V_1 = V_{BDC} = V_d + V_c = 0.001 + 0.0000667 = 0.0010667\left(m^3\right)$$

Mass of gas in one cylinder at start of compression:

$$m_m = \frac{P_1 * V_1}{R * T_1} = \frac{100(kPa) * 0.0010667\left(m^3\right)}{0.287\left(\frac{kJ}{kg.K}\right) * 333(K)} = 0.00112(kg)$$

Mass of fuel injected per cylinder per cycle:

$$m_f = 0.00112(kg) * \left(\frac{1}{19}\right) * 0.98 = 0.0000578(kg)$$

State (2):

$$T_2 = T_1.(r_C)^{\gamma-1} = 333(K)*(16)^{0.35} = 879(K) = 606^{\circ}C$$

$$P_2 = P_1.(r_C)^{\gamma} = 100(kPa)*(16)^{1.35} = 4222(kPa)$$

$$V_2 = {m*R*T_2}\Big/{P_2} = \left[0.00112(kg)*0.287\left(\frac{kJ}{kg.K}\right)*879(K)\right]\Big/4222(kPa)$$

$$V_2 = 0.000067(m^3) = V_d$$

Or (V_2) can be found by using the following equation:

$$V_2 = {V_1}\Big/{r_C} = 0.0010667\Big/16 = 0.0000667(m^3)$$

State (3):

$$Q_A = m_f * Q_{HV} = 0.0000578(kg)*42500\left(\frac{kJ}{kg}\right) = 2.46(kJ)$$

If half of (Q_A) occurs at constant volume, then using the following equation gives us:

$$Q_{2-3} = m_m * C_v * (T_3 - T_2)$$

$$\frac{2.46}{2}(kJ) = 0.00112(kg)*0.821\left(\frac{kJ}{kg.K}\right)*(T_3 - 879K)$$

$$\therefore T_3 = 2217(K) = 1944^{\circ}C$$

$$V_3 = V_2 = 0.0000667(m^3)$$

$$P_3 = \frac{m*R*T_3}{V_3} = \frac{0.00112(kg)*0.287\left(\frac{kJ}{kg.K}\right)*2217(K)}{0.0000667(m^3)} = 10650(kPa) = P_{\max}$$

$or:$

$$P_3 = P_2 * \left(\frac{T_3}{T_2}\right) = 4222(kPa)*\left(\frac{2217}{879}\right) = 10650(kPa)$$

State (4):

$$P_4 = P_3 = 10650(kPa) = P_{\max}$$

$$Q_{3-4} = m_m * C_p * (T_4 - T_3)$$

$$1.23(kJ) = 0.00112(kg)*1.108\left(\frac{kJ}{kg.K}\right)*(T_4 - 2217K)$$

$$\therefore T_4 = 3208(K) = 2935^{\circ}C = T_{\max}$$

$$V_4 = \frac{m*R*T_4}{P_{4=3=\max}} = \frac{0.00112*0.287*3208}{10650} = 0.000097(m^3)$$

State ($\bar{4}$):

$$V_{\bar{4}} = V_1 = 0.0010667\left(m^3\right)$$

$$T_{\bar{4}} = T_4 . \left(V_4 \big/ V_{\bar{4}} \right)^{\gamma-1} = 3208 * \left(0.000097 \big/ 0.0010667 \right)^{0.35} = 1386\left(K\right) = 1113^{\circ} C$$

$$P_{\bar{4}} = P_4 . \left(V_4 \big/ V_{\bar{4}} \right)^{\gamma} = 10650 * \left(0.000097 \big/ 0.0010667 \right)^{1.35} = 418\left(kPa\right)$$

Work out for process (3-4) for one cylinder for one cycle:

$$W_{3-4} = P_{\max} .\left(V_4 - V_3\right) = 10650 * \left(0.000097 - 0.0000667\right) = 0.323\left(kJ\right)$$

Work out for process ($4-\bar{4}$):

$$W_{4-\bar{4}} = \frac{m * R * .\left(T_{\bar{4}} - T_4\right)}{1-\gamma} = \frac{0.00112 * 0.287 * \left(1386 - 3208\right)}{1 - 1.35} = 1.673\left(kJ\right)$$

Work in for process (1-2):

$$W_{1-2} = \frac{m * R * .\left(T_2 - T_1\right)}{1-\gamma} = \frac{0.00112 * 0.287 * \left(897 - 333\right)}{1 - 1.35} = -0.501\left(kJ\right)$$

$$W_{net} = W_{3-4} + W_{4-\bar{4}} + W_{1-2} = \left(+0.323\right) + \left(+1.673\right) + \left(-0.501\right)$$
$$\therefore W_{net} = +1.495\left(kJ\right)$$

2) Indicated thermal efficiency:

$$\left.\eta_{th}\right)_{Dual} = \frac{\left|W_{net}\right|}{\left|Q_A\right|} = \frac{1.495}{2.46} = 0.607 = 60.7\%$$

Pressure ratio:

$$\alpha = P_3 \big/ P_2 = 10650 \big/ 4222 = 2.52$$

Cutoff ratio:

$$\beta = V_4 \big/ V_3 = 0.000097 \big/ 0.0000667 = 1.45$$

Also, thermal efficiency can be found from the following equation

$$\left.\eta_{th}\right)_{Dual} = 1 - \left(\frac{1}{r_C}\right)^{\gamma-1} . \left[\frac{\left(\alpha\beta^{\gamma} - 1\right)}{\gamma\alpha\left(\beta - 1\right) + \left(\alpha - 1\right)}\right] = 1 - \left(\frac{1}{16}\right)^{0.35} . \left[\frac{\left(2.52 * 1.45^{1.35}\right) - 1}{1.35 * 2.52 * \left(1.45 - 1\right) + 2.52 - 1}\right] = 0.607$$

1) Assuming exhaust pressure is the same as intake pressure, then exhaust temperature is found from:

$$T_{exh} = T_{\bar{4}} * \left(\frac{P_{exh}}{P_{\bar{4}}}\right)^{\frac{\gamma-1}{\gamma}} = 1386 * \left(\frac{100}{418}\right)^{\frac{0.35}{1.35}} = 957\left(K\right) = 684^{\circ} C$$

4) Mass of air entering one cylinder during intake:

$m_a = 0.00112 * 0.98 = 0.0011(kg)$

Then, volumetric efficiency is found by using the equation below:

$$\eta_v = \frac{m_a}{\rho_a * V_d} = \frac{0.0011(kg)}{1.181\left(\frac{kg}{m^3}\right) * 0.001(m^3)} = 0.931 = 93.1\%$$

2-5 Comparison of Otto, Diesel and Dual Cycles

2-5-1 *For the same inlet conditions and the same compression ratio*:

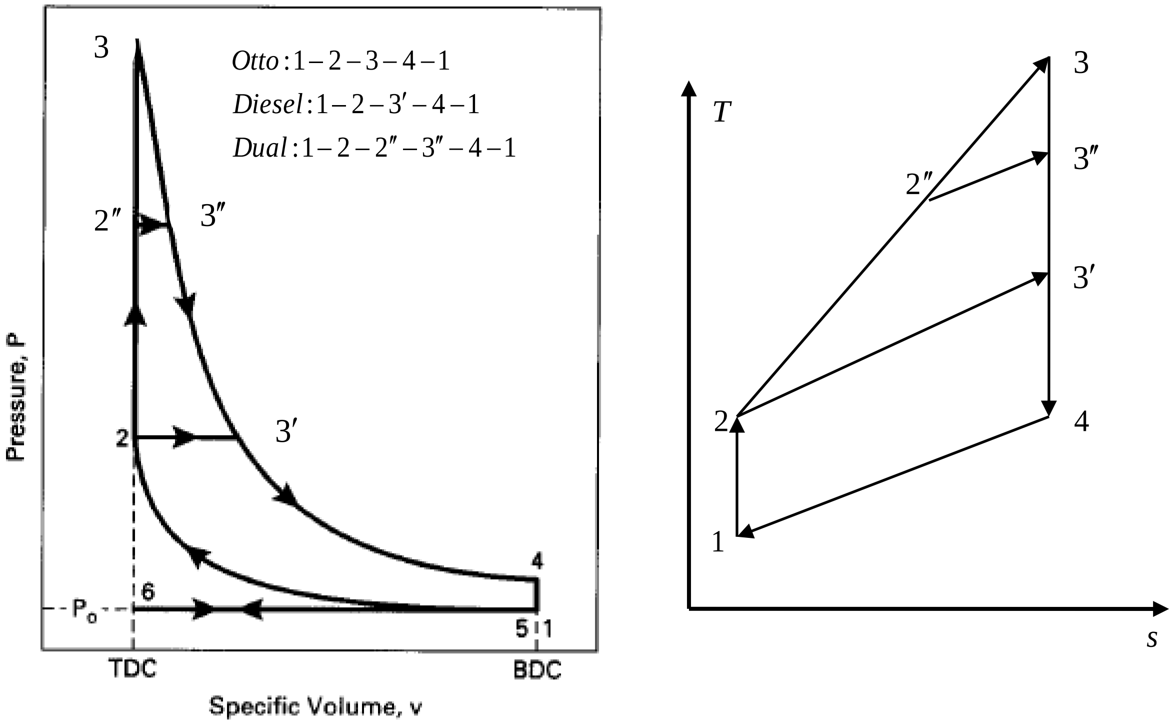


Figure (2-5): Comparison of Otto, Diesel and Dual Cycles at same inlet conditions & compression ratio.

Heat input for Otto is the area under $(2-3)$

Heat input for Diesel is the area under $(2-3')$

Heat input for Dual is the area under $(2-2''-3'')$.

Heat rejection for Otto = Heat rejection for Diesel = Heat rejection for Dual Cycles = Area under $(1-4)$.

$Q_A)_{Otto} > Q_A)_{Diesel}$ also: $Q_A)_{Otto} > Q_A)_{Diual}$

$\therefore W_{Otto} > W_{Dual} > W_{Diesel}$

$\& \eta_{th})_{Otto} > \eta_{th})_{Dual} > \eta_{th})_{Diesel}$

2-5-2 ***For the same inlet conditions and same maximum temperature***:

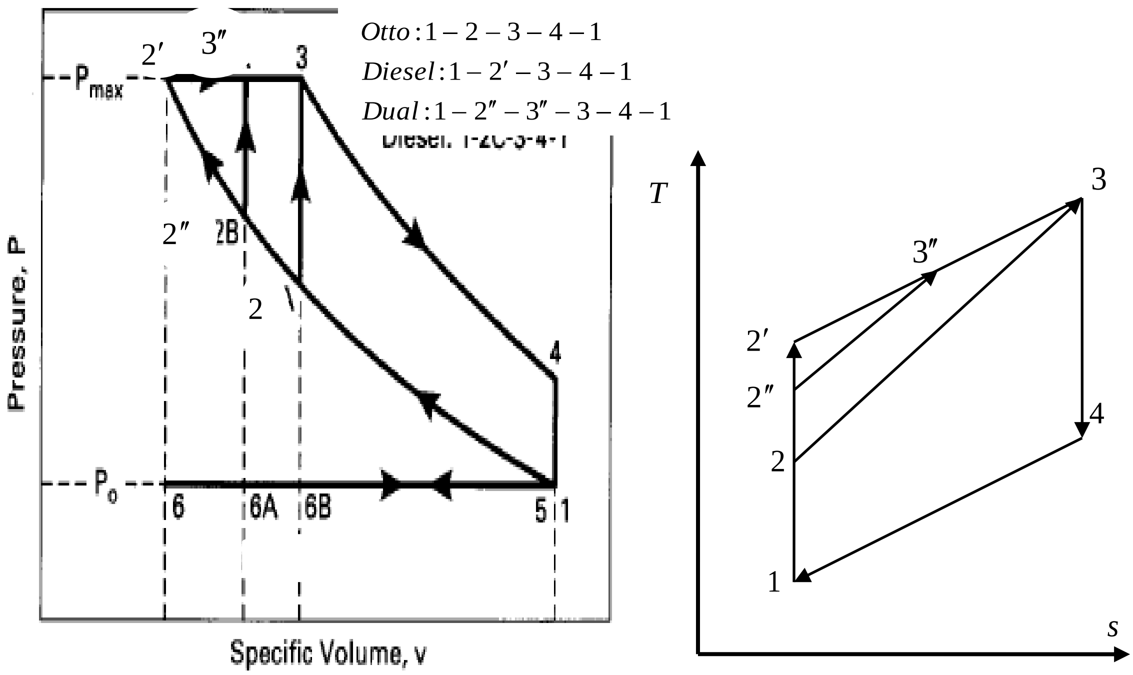


Figure (2-6): Comparison of Otto, Diesel and Dual Cycles at same inlet conditions & maximum temperatures.

Heat input (Q_A) for Otto is the area under $(2-3)$

Heat input (Q_A) for Diesel is the area under $(2'-3)$

Heat input (Q_A) for Dual is the area under $(2''-3''-3)$.

Heat rejection (Q_R) for Otto = Heat rejection (Q_R) for Diesel = Heat rejection (Q_R) for Dual Cycles = Area under $(1-4)$.

$$\left.Q_A\right)_{Diesel} > \left.Q_A\right)_{Dual} > \left.Q_A\right)_{Otto}$$

$$\therefore W_{Diesel} > W_{Dual} > W_{Otto} >$$

$$\& \left.\eta_{th}\right)_{Diesel} > \left.\eta_{th}\right)_{Dual} > \left.\eta_{th}\right)_{Otto}$$

2-5-3 ***For the same heat input and same maximum pressure***:

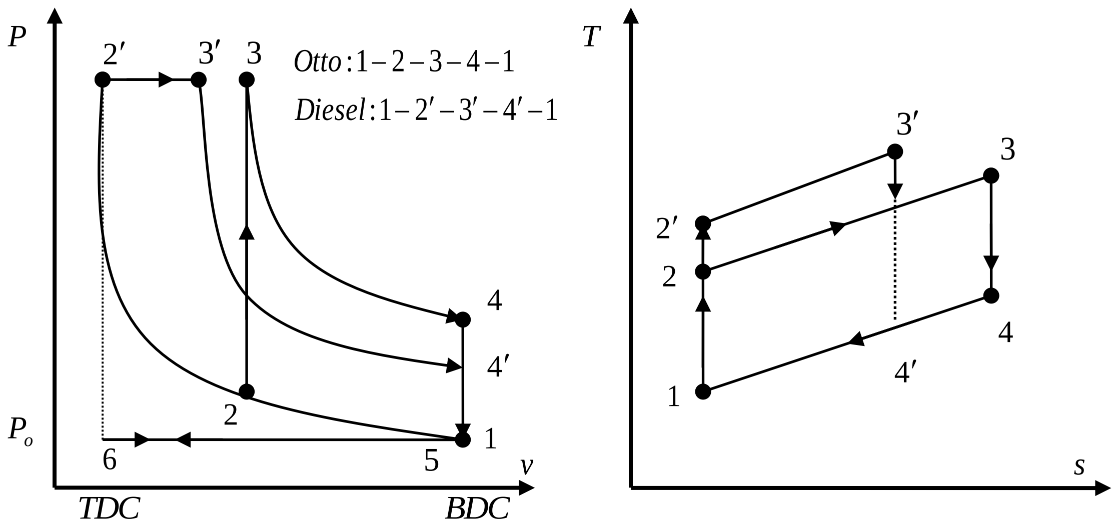


Figure (2-7): Comparison of Otto, Diesel Cycles at same heat input & maximum pressure.

Heat input (Q_A) for Otto is the area under $(2-3)$

Heat input (Q_A) for Diesel is the area under $(2'-3')$

But: $\left.Q_A\right)_{Otto} = \left.Q_A\right)_{Diesel}$

Heat rejection (Q_R) for Otto = area under $(1-4)$

Heat rejection (Q_R) for Diesel = area under $(1-4')$

$$\left.Q_A\right)_{Diesel} < \left.Q_A\right)_{Otto}$$

$$W_{Diesel} > W_{Otto}$$
$$\& \left.\eta_{th}\right)_{Diesel} > \left.\eta_{th}\right)_{Otto}$$

2-5-4 ***For the same maximum pressure and same maximum temperature***:

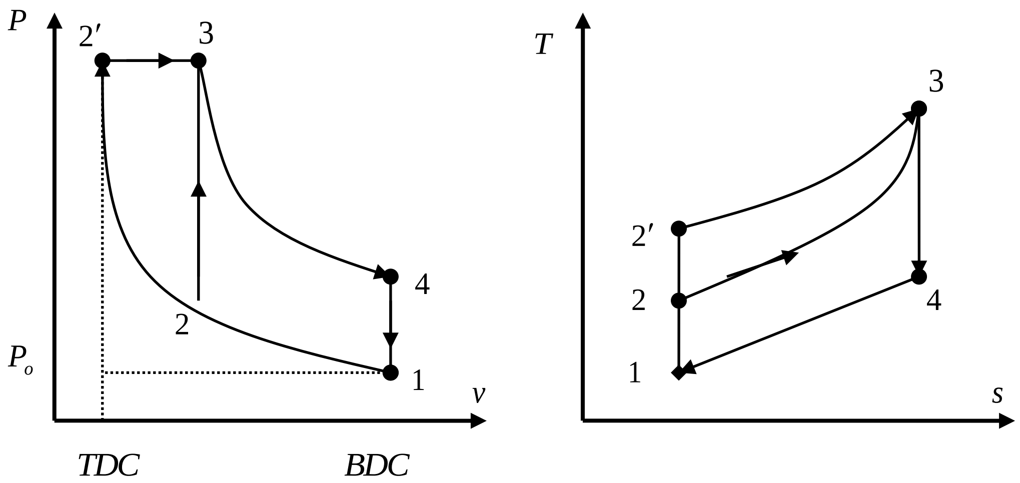


Figure (2-8): Comparison of Otto, Diesel Cycles at same maximum temperature & maximum pressure.

$$Otto: 1-2-3-4-1$$
$$Diesel: 1-2'-3-4-1$$
$$\left.Q_A\right)_{Otto}: \underrightarrow{under}(2-3)$$
$$\left.Q_A\right)_{Diesel}: \underrightarrow{under}(2'-3)$$
$$\left.Q_R\right)_{Otto}: \underrightarrow{under}(4-1)$$
$$\left.Q_R\right)_{Diesel} \underrightarrow{under}(4-1)$$
$$\left.Q_A\right)_{Ottol} < \left.Q_A\right)_{Diesel}$$
$$\left.Q_R\right)_{Ottol} = \left.Q_R\right)_{Diesel}$$
$$\therefore W_{Diesel} > W_{Otto} \;\&\; \left.\eta_{th}\right)_{Diesel} > \left.\eta_{th}\right)_{Otto}$$

2-5-5 ***For the same heat input and same compression ratio***:

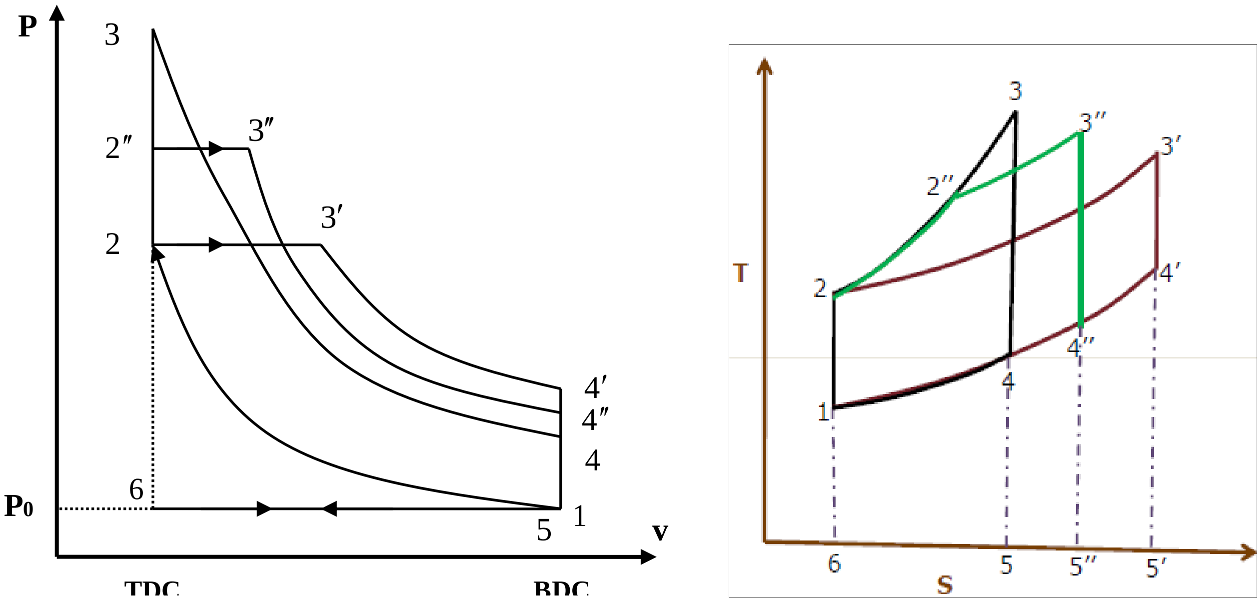


Figure (2-9): Comparison of Otto, Diesel and Dual Cycles at same heat input & same compression ratio.

$$\begin{aligned}
&Otto: 1-2-3-4-1\\
&Diesel: 1-2-3'-4'-1\\
&Dual: 1-2-2'-3''-4''-1\\
&Q_A)_{Ottol} = Q_A)_{Diesel} = Q_A)_{Dual}\\
&Q_R)_{Ottol}\, \underrightarrow{under} (1-4)\\
&Q_R)_{Diesel}\, \underrightarrow{under} (1-4')\\
&Q_R)_{Dual}\, \underrightarrow{under} (1-4'')\\
&Q_A)_{Ottol} < Q_R)_{Dual} < Q_A)_{Diesel}\\
&\therefore W_{Otto} > W_{Dual} > W_{Diesel}\\
&\&\\
&\eta_{th})_{Otto} > \eta_{th})_{Dual} > \eta_{th})_{Diesel}
\end{aligned}$$

2-6 Atkinson Cycle (also called: Over-expanded Cycle):

In Otto and Diesel cycles, when the exhaust valve is opened near the end of expansion stroke, pressure in the cylinder is still on the order of three to five atmospheres. A potential for doing additional work during the power stroke is there lost when the exhaust valve is opened and pressure is reduced to atmospheric. If the exhaust valve is not opened until the gas in the cylinder is allowed to expand to atmospheric pressure, a greater amount of work would be obtained in the expansion stroke with an increase in engine thermal efficiency. Atkinson cycle is illustrated in figure (2-10) below.

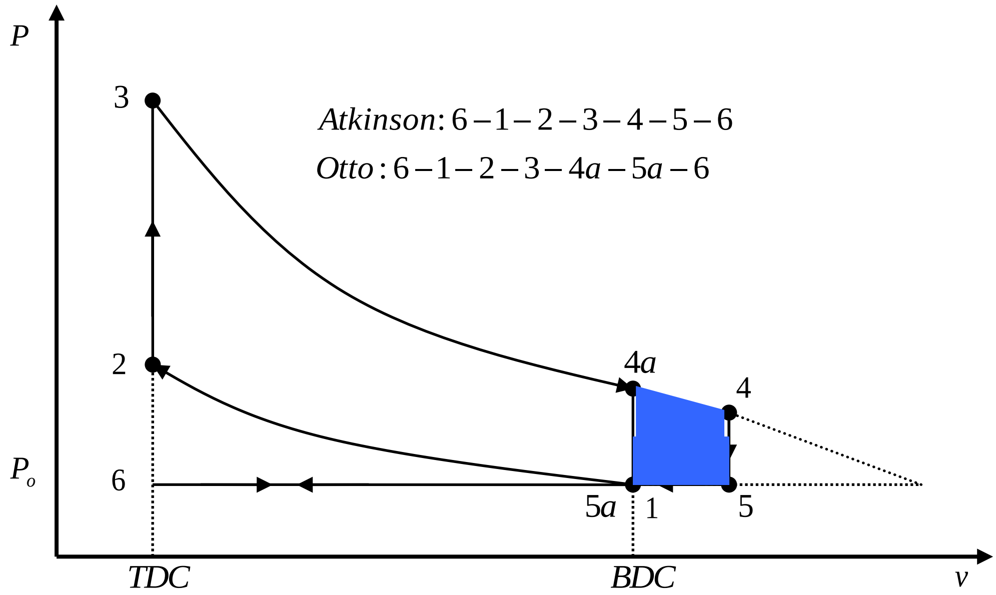


Figure (2-10): Atkinson Cycle.

This cycle starting in 1885, a number of crank and valve mechanisms were tried to achieve this cycle, which has a longer expansion stroke than compression stroke. No large number of these engines has ever been marked, indicating the failure of this development.

Compression ratio: $r_c = {v_1}/{v_2}$

Expansion ratio: $r_e = {v_4}/{v_2}$

$$\boxed{\eta_{th})_{Atkinston} = 1 - \gamma\left(\frac{r_e - r_c}{r_e^{\gamma} - r_c^{\gamma}}\right)}$$

2-7 Miller Cycle:

This cycle is named after R. H. Miller (1890-1967), it is a modern modification of the Atkinson cycle and has an expansion ratio greater than the compression ratio. Miller cycle is shown in figure (2-11).

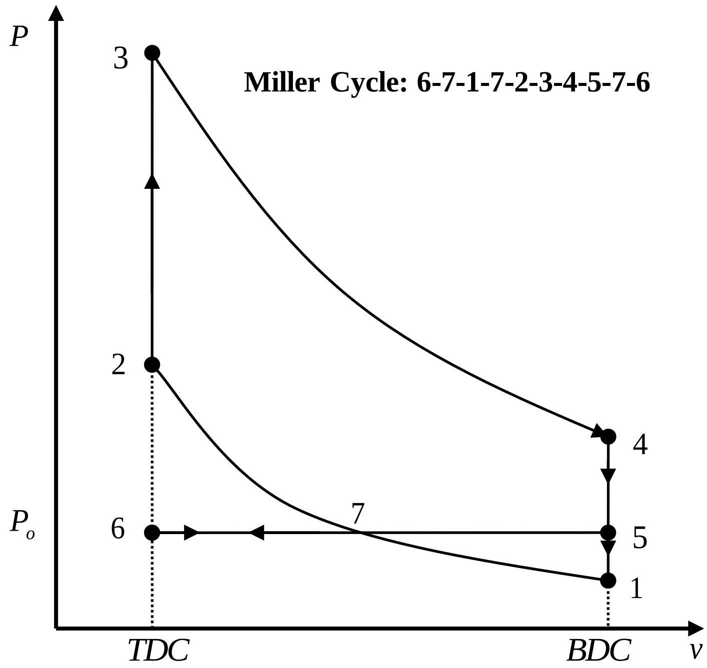


Figure (2-11): Miller Cycle.

Air intake in a Miller cycle is un-throttled. The amount of air ingested into each cylinder is thus controlled by closing the intake valve at the proper time, long before BDC (point-7). As the piston then continues towards BDC during the last part of the intake stroke, cylinder pressure is reduced along process (7-1). When the piston reaches BDC and starts back toward TDC, cylinder pressure is again increased during process (1-7). The work produced in the first part of the intake process (6-7) is canceled by part of the exhaust stroke (7-6), process (7-1) is canceled by process (1-7), and the net indicated work is the area within loop (7-2-3-4-5-7). There is essentially no pump work.

The shorter compression stroke which absorbs work combined with the longer expansion stroke which produces work, results in a greater net indicated work per cycle.

Compression ratio: $r_c = {}^{v_7}\!/_{v_2}$

Expansion ratio: $r_e = {}^{v_4}\!/_{v_2} = {}^{v_4}\!/_{v_3}$

$$\boxed{\eta_{th})_{Miller} = \frac{W}{Q_A}}$$

Automobiles with Miller cycle engines were first marked in the later half of the 1990s. A typical value of the compression ratio is about $(r_c = 8:1)$with an expansion ratio about$(r_e = 10:1)$.

2-7 Joule (Brayton) Cycle:

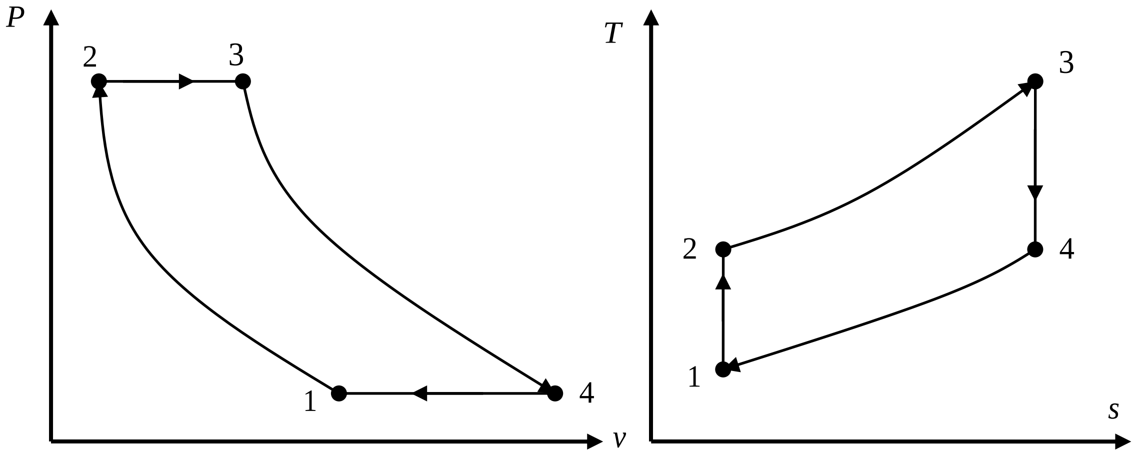


Figure (2-12): Brayton Cycle.

Compression ratio: $r_c = {}^{v_7}\!/\!{}_{v_2}$

Pressure ratio: $r_p = {}^{P_2}\!/\!{}_{P_1}$

$$\boxed{\eta_{th})_{Brayton} = 1 - \frac{Q_R}{Q_A} = 1 - \frac{C_p.(T_4 - T_1)}{C_p.(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}}$$

$$T_4 = {}^{T_3}\!/\!{}_{(r_c)^{\gamma-1}} \;\&\; T_1 = {}^{T_2}\!/\!{}_{(r_c)^{\gamma-1}}$$

$$\eta_{th} = 1 - \frac{\left[{}^{T_3}\!/\!{}_{(r_c)^{\gamma-1}} - {}^{T_2}\!/\!{}_{(r_c)^{\gamma-1}}\right]}{T_3 - T_2} = 1 - \frac{1}{{r_c}^{\gamma-1}}$$

$also;$

$$\eta_{th} = 1 - \frac{1}{{r_p}^{\gamma-1/\gamma}}$$

The Actual Cycle

The actual cycle experienced by the internal combustion engine is not in the true sense thermodynamic cycle. An ideal air-standard thermodynamic cycle occurs on a closed system of constant composition. This is not what actually happens in an IC engine and for this reason air-standard gives a best only approximations to actual conditions and output. For actual cycle the efficiency is lower than the ideal efficiency and this duo to the following losses:

1) ***Losses duo to variation of Specific Heats with Temperature***:

All gases except monatomic gases show an increase in specific heat at high temperatures.

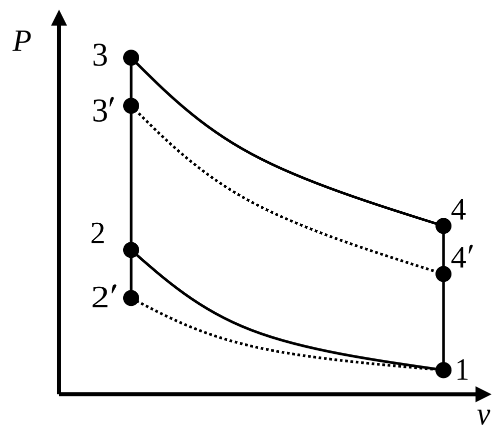
$$\left.\begin{aligned} C_p &= a + k_1.T_1 + k_2.T_2 + \ldots \\ C_v &= b + k_1.T_1 + k_2.T_2 + \ldots \end{aligned}\right\} \text{ for temper range from } (300-1500)K$$

But (γ) degrease with increasing temperature, because the difference between $(C_p \,\&\, C_v)$ constant.

$$\gamma = C_p / C_v$$

$$\because R = C_p - C_v \Rightarrow C_p = R + C_v$$

$$\therefore \gamma = \frac{R + C_v}{C_v} = \frac{R}{C_v} + 1$$



2) ***Dissociation Losses***:

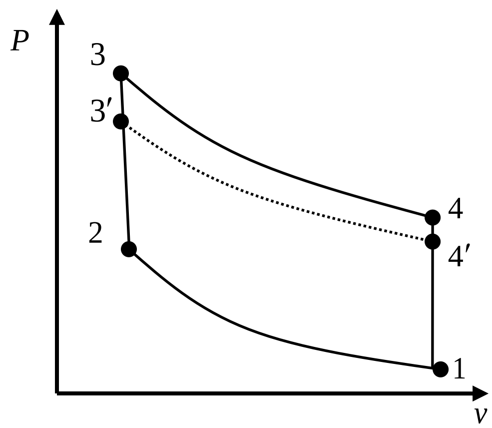
Dissociation increases with increasing temperature, also considerable amount of heat is absorbed. This heat will be liberated when the element recombine as the temperature falls.

$$C + O_2 \Leftrightarrow CO_2$$

$$2CO_2 + Heat \Leftrightarrow 2CO + O_2$$

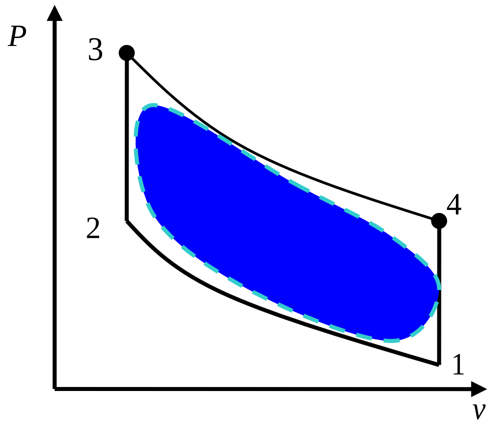
This effect appears at temperature:

$(1000 - 1500)^\circ C$



3) ***Time Losses***:

Burning in theoretical cycle is assumed to be completed instantaneously, while burning in actual cycle is completed in a definite interval of time during which there will be a change in volume (the burning combustion is completed during 40° of crank turn, i.e. 10° before BDC and 30° after TDC) and thus the maximum pressure dose not occurs sometime after TDC.



4) ***Losses due incomplete combustion***:

Some fuel dose not burns or burn partially and thus CO and CO_2 or even fuel will appears in the exhaust gases. This is due hydrogenous mixture, i.e. excess air in one part of the cylinder and excess fuel in another part. Therefore, energy release in actual engine is about (90-93) % of fuel energy input.

5) ***Direct Heat Losses***:

During the combustion process and subsequent expansion (power) stroke, heat will be loosed according to:

a. From hot gases to cylinder wall to water or cooling fins.

b. Heat into piston or cylinder wall carried away by engine oil.

*There for C_P and C_v are not constant.

6) ***Exhaust Blow-down Losses***:

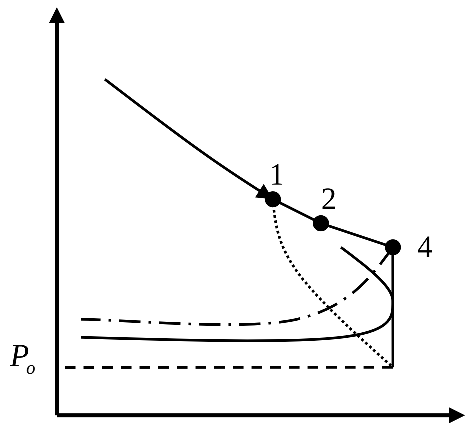
The cylinder pressure at the end of expansion stroke is about (3-5) bars know:

a. If the exhaust valve is opened at BDC (point 4), the piston has to do work against high cylinder pressure.

b. If the exhaust valve is opened too early (point 2), part of the expansion stroke is lost.

c. The best compromise is to open exhaust valve $\left(40^{\circ} - 60^{\circ}\right)$ before BDC, thus reducing the cylinder pressure to half way to atmospheric before the exhaust stroke begins:

- ❖ Exhaust valve open at point (4) too late.
- ❖ Exhaust valve open at point (1) too early.
- ❖ The best is to open at point (2).



7) ***Pumping Losses***:

These losses are due to pumping gas from atmospheric (low inlet pressure) to higher exhaust pressure. Pumping losses increase as a part throttle because throttling reduces the suction pressure also increases with speed.

8) ***Rubbing Friction Losses***:

These losses are duo to:

- **a.** Friction between piston and cylinder walls.
- **b.** Friction in various bearings.
- **c.** Friction in auxiliary equipment as pumps (oil, water, fuel … etc), fans generators … etc.

These losses increased with increasing engine speed.