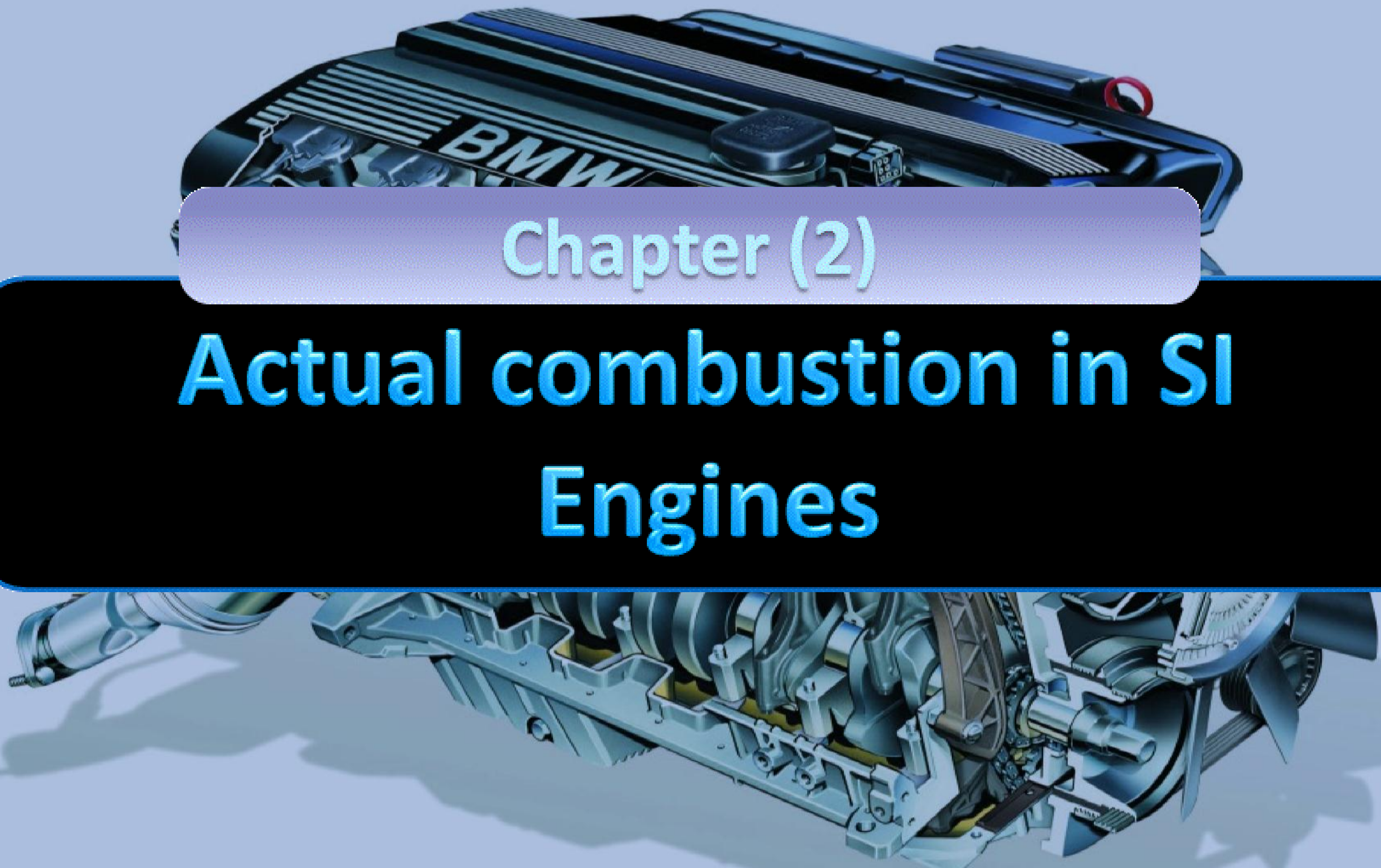
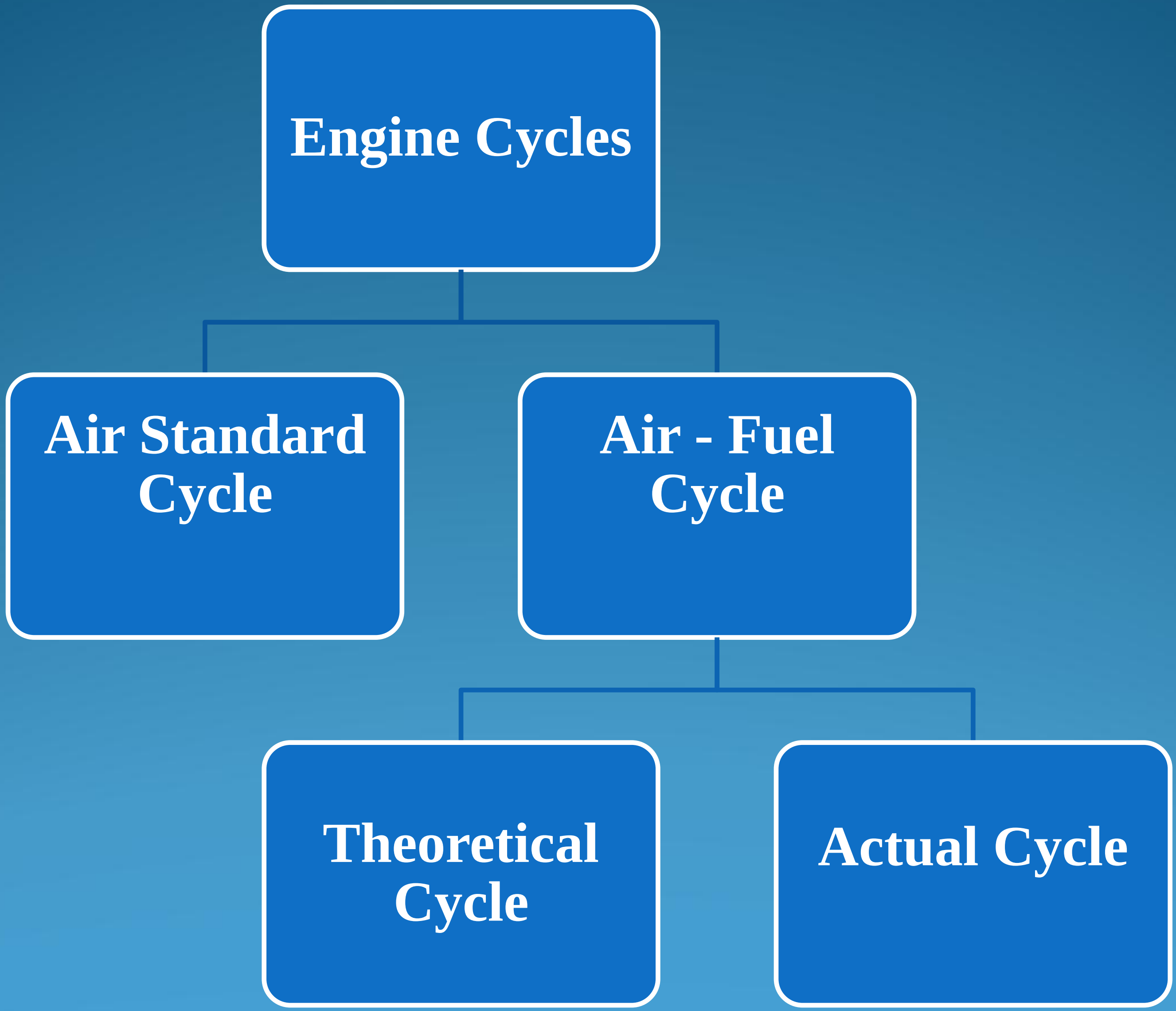


Chapter (2)

Actual combustion in SI Engines

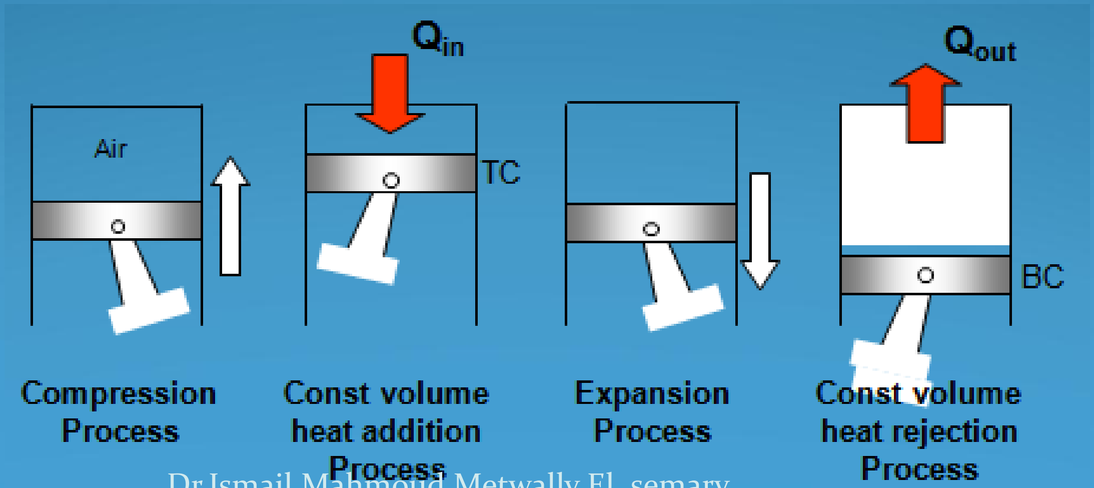
Thermodynamic cycles

Thermodynamic cycles



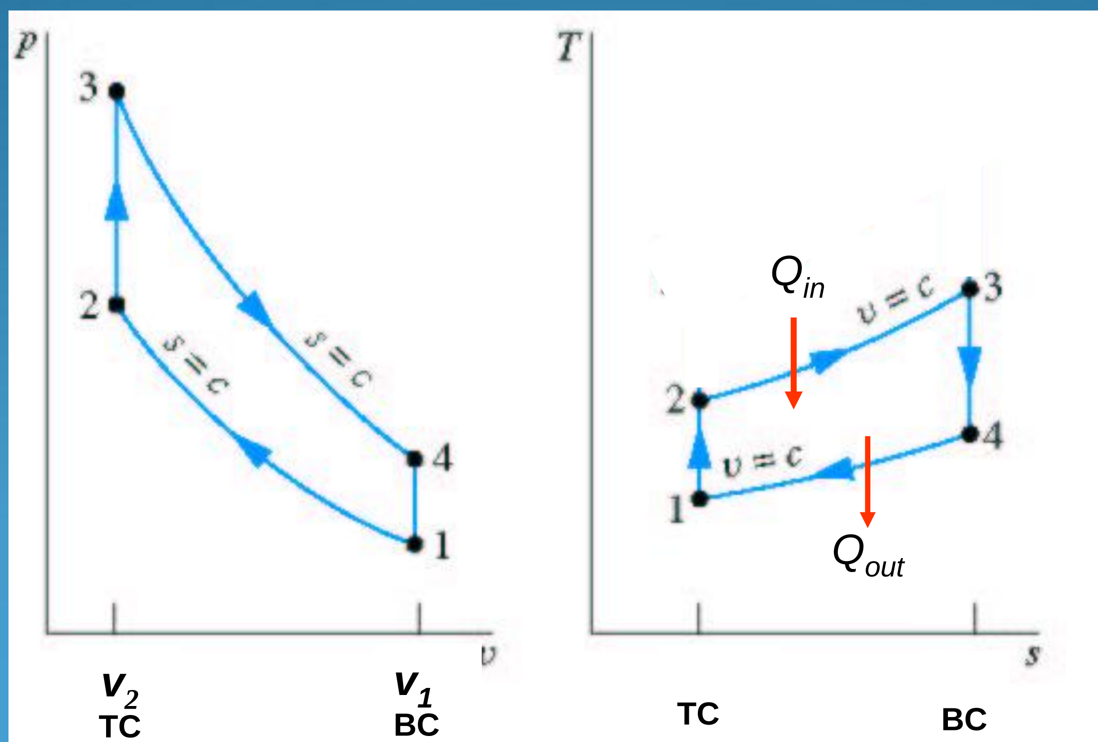
- **Air-standard analysis** is used to perform elementary analyses of IC engine cycles.

- **Simplifications to the real cycle include:**
  1) Fixed amount of air (ideal gas) for working fluid
  2) Combustion process not considered
  3) Intake and exhaust processes not considered
  4) Engine friction and heat losses not considered
  5) Specific heats independent of temperature



Air-Standard Otto cycle

Process 1→ 2 Isentropic compression
Process 2 → 3 Constant volume heat addition
Process 3 → 4 Isentropic expansion
Process 4 → 1 Constant volume heat rejection



Compression ratio:

$$r = \frac{v_1}{v_2} = \frac{v_4}{v_3}$$

Thermodynamic Analysis of Air-Standard Otto Cycle:

Process 1-2 **Isentropic Compression.**

All valves closed.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1(r_c)^{k-1} \quad \text{and} \quad P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1(r_c)^{k}$$

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

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

Process 2-3 **Constant Volume Heat Input.**

All valves closed.

$$V_3 = v_2 = v_{TDC} \quad \text{so that} \quad \frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$T_3 = T_{Max.} \quad \text{and} \quad P_3 = P_{Max.}$$

$$q_{2-3} = q_{in} = Cv(T_3 - T_2)$$

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

Process 3-4 Isentropic Expansion

All valves closed.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r_c}\right)^{k-1} \quad \text{and} \quad P_4 = P_3\left(\frac{v_3}{v_4}\right)^{k} = P_3\left(\frac{1}{r_c}\right)^{k}$$

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

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

Process 4-5 Constant Volume Heat Rejection

Exhaust valve open and intake valve closed.

$$v_5 = v_4 = v_1 = v_{BDC} \quad \text{so that} \quad \frac{T_5}{T_4} = \frac{P_5}{P_4}$$

$$q_{4-5} = q_{out} = Cv(T_5 - T_4) = Cv(T_1 - T_4)$$

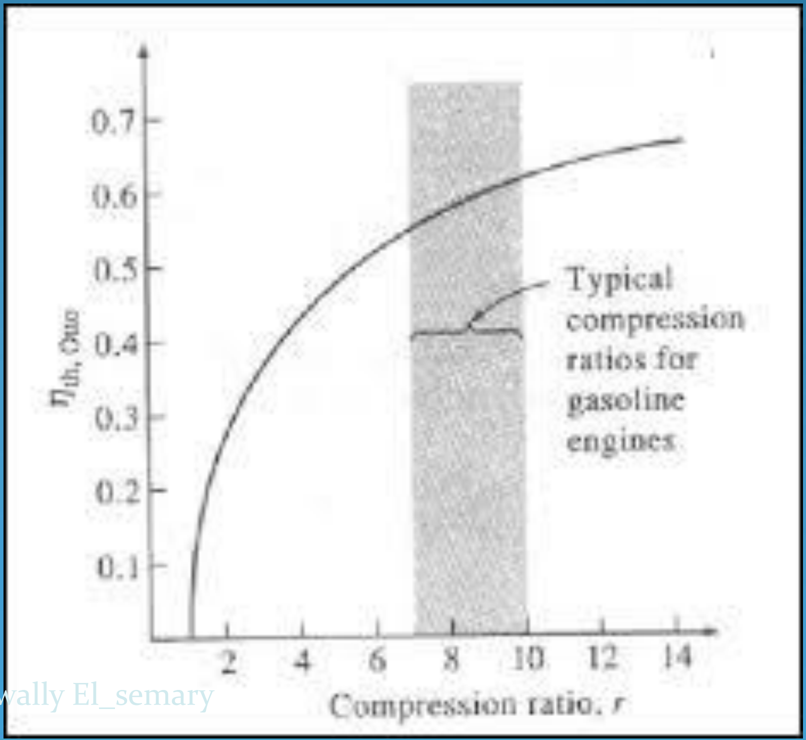
$$w_{4-5} = 0$$

Thermal efficiency of Otto Cycle:

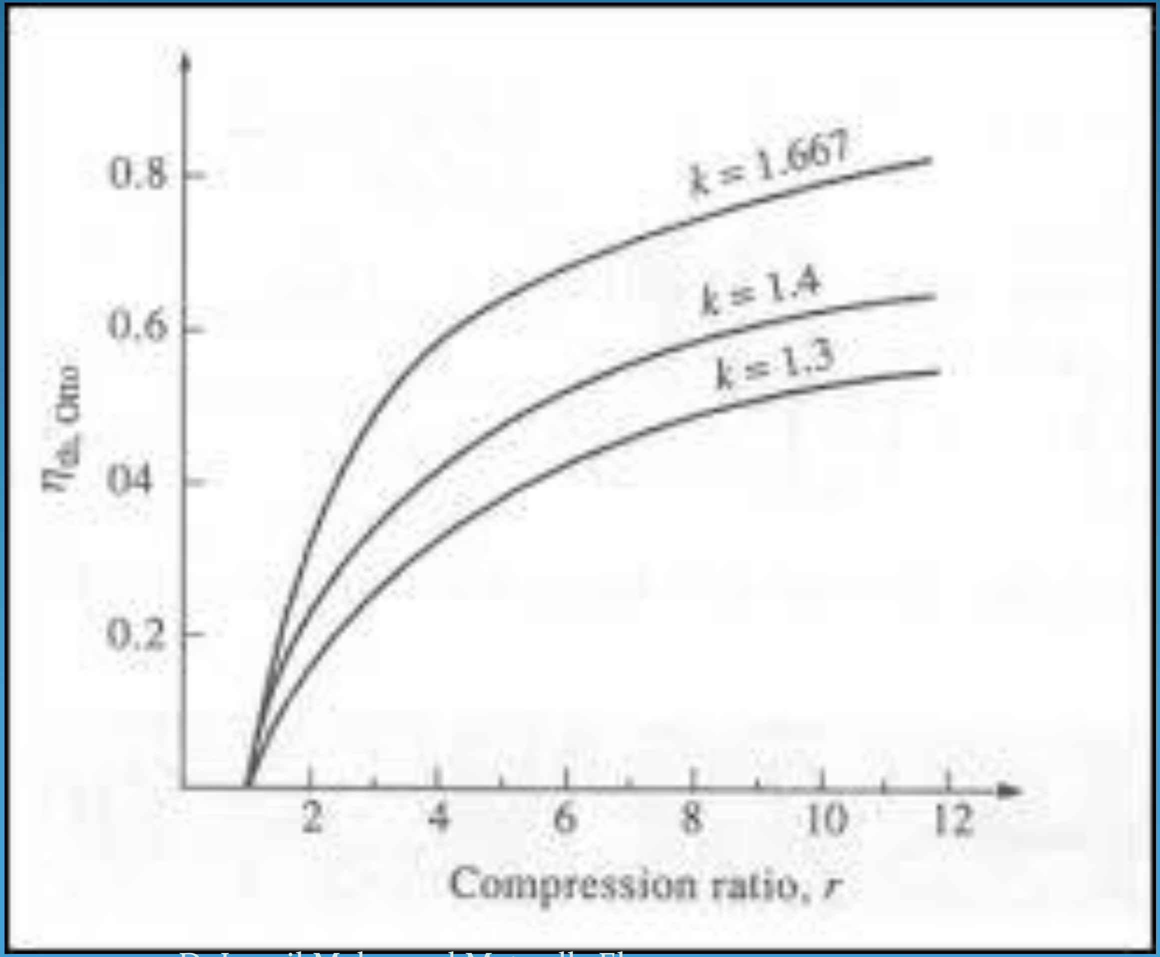
$$\eta_{(Otto)} = \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|} = 1 - \frac{Cv(T_4 - T_1)}{Cv(T_3 - T_2)} = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)}$$

$$\therefore \eta_{(Otto)} = 1 - \left(\frac{T_1}{T_2}\right) = 1 - \left(\frac{1}{r_c}\right)^{k-1}$$

The effect of compression ratio on thermal efficiency of Otto Cycle:

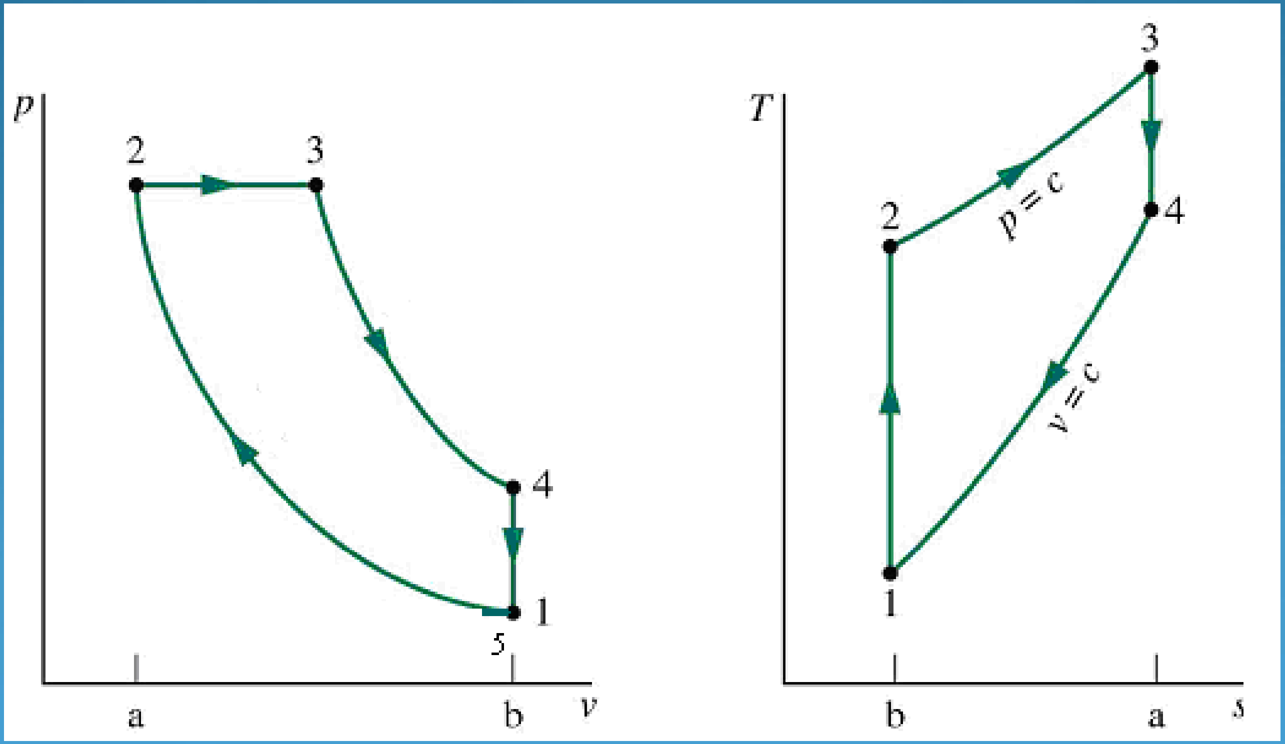


The effect of variation of specific heat constant γ = Cp/Cv on thermal efficiency of Otto Cycle:



Air-Standard Diesel cycle

Process 1 → 2 Isentropic compression
Process 2 → 3 Constant pressure heat addition
Process 3 → 4 Isentropic expansion
Process 4 → 1 Constant volume heat rejection



Cut off ratio:

$$\left(r_{cut\ off}\right)=\left(\frac{v_3}{v_2}\right)=\left(\frac{T_3}{T_2}\right)$$

Thermodynamic Analysis of Air-Standard Diesel Cycle:

Process 1-2 **Isentropic Compression Stroke.**

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1(r_c)^{k-1} \quad \text{and} \quad P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1(r_c)^{k}$$

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

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

Process 2-3 **Constant Pressure Heat Input. (Combustion)**

$$P_3 = P_2 \quad \text{and} \quad T_3 = T_2\left(\frac{v_3}{v_2}\right) = T_2\left(r_{cut\,off}\right)$$

$$q_{2-3} = q_{in} = Cp(T_3 - T_2)$$

$$w_{2-3} = P_3(v_3 - v_2)$$

Process 3-4 **Isentropic Power or Expansion Stroke.**

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{r_{cut\,off}}{r_c}\right)^{k-1} \quad \text{and} \quad P_4 = P_3\left(\frac{v_3}{v_4}\right)^{k} = P_3\left(\frac{r_{cut\,off}}{r_c}\right)^{k}$$

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

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

Process 4-5 **Constant Volume Heat Rejection. (Exhaust stroke)**

$$v_5 = v_4 = v_1 = v_{BDC} \quad \text{so that} \quad \frac{T_5}{T_4} = \frac{P_5}{P_4}$$

$$q_{4-5} = q_{out} = Cv(T_5 - T_4) = Cv(T_1 - T_4)$$

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

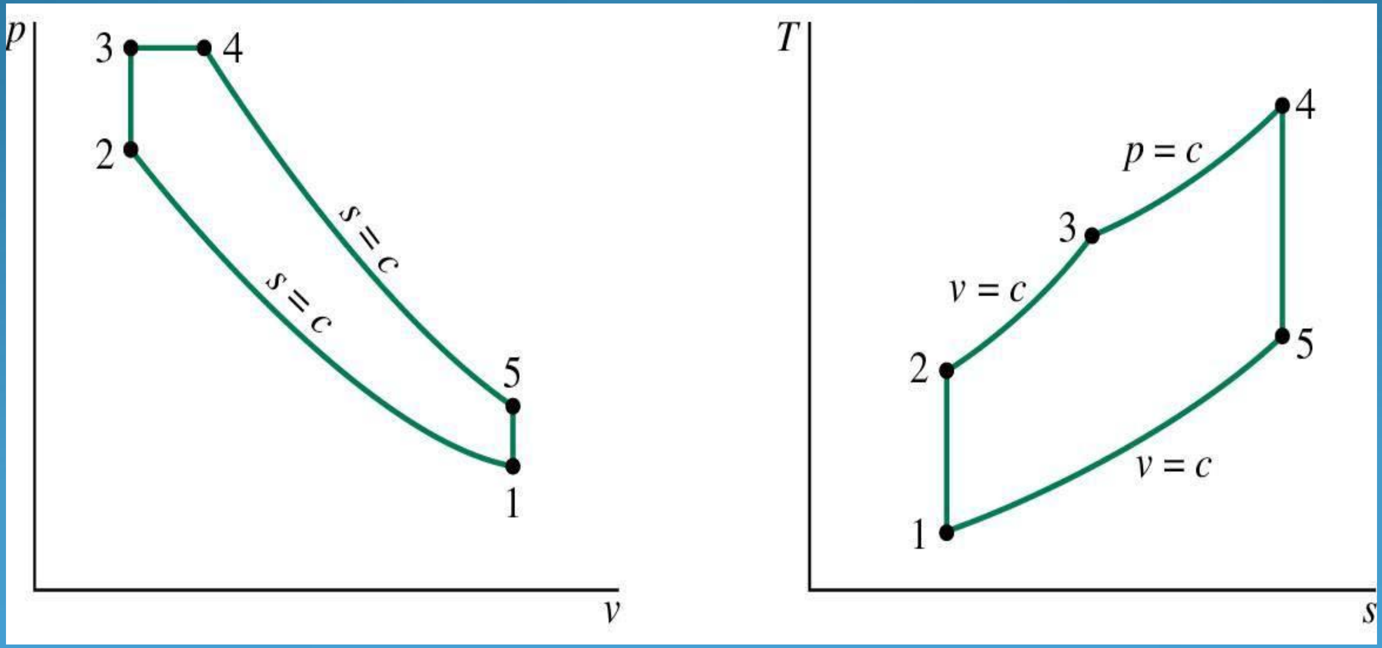
Thermal efficiency of Diesel Cycle:

$$\eta_{(Diesel)} = \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|} = 1 - \frac{Cv(T_4 - T_1)}{Cp(T_3 - T_2)}$$

$$\therefore \eta_{(Diesel)} = 1 - \left(\frac{1}{r_c}\right)^{k-1} \times \left[\frac{r_{\text{cutt off}}^{\text{k}} - 1}{k(r_{\text{cut off}} - 1)}\right]$$

Air-Standard Dual cycle

Process 1 → 2 Isentropic compression
Process 2 → 3 Constant volume heat addition
Process 3 → 4 Constant pressure heat addition
Process 4 → 5 Isentropic expansion
Process 5 → 1 Constant volume heat rejection



Thermodynamic Analysis of Air-Standard Dual Cycle:

Process 1-2 **Isentropic Compression Stroke.**

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1\left(r_c\right)^{k-1} \quad \text{and} \quad P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1\left(r_c\right)^{k}$$

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

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

Process 2-3 **Constant Volume Heat Input. (Combustion)**

$$V_3 = v_2 = v_{TDC} \quad \text{so that} \quad \frac{T_3}{T_2} = \frac{P_3}{P_2}$$

$$q_{2-3} = q_{in} = Cv\left(T_3 - T_2\right)$$

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

Process 3-4 **Constant Pressure Heat Input. (Combustion)**

$$P_4 = P_3 \quad \text{and} \quad T_4 = T_3\left(\frac{v_4}{v_3}\right) = T_3\left(r_{cut\,off}\right)$$

$$q_{3-4} = q_{in} = Cp\left(T_4 - T_3\right)$$

$$w_{3-4} = P_4\left(v_4 - v_3\right)$$

Process 4-5 **Isentropic Power or Expansion Stroke.**

$$T_5 = T_4\left(\frac{v_4}{v_5}\right)^{k-1} = T_3\left(\frac{r_{cut\,off}}{r_c}\right)^{k-1} \quad \text{and} \quad P_5 = P_4\left(\frac{v_4}{v_5}\right)^{k} = P_3\left(\frac{r_{cut\,off}}{r_c}\right)^{k}$$

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

$$w_{4-5} = \frac{P_5 v_5 - P_4 v_4}{1-k} = \frac{R(T_5 - T_4)}{1-k} = Cv\left(T_4 - T_5\right)$$

Process 5-6 **Constant Volume Heat Rejection. (Exhaust stroke)**

$$v_6 = v_5 = v_1 = v_{BDC} \qquad \text{so that} \qquad \frac{T_6}{T_5} = \frac{P_6}{P_5}$$

$$q_{5-6} = q_{out} = Cv(T_6 - T_5) = Cv(T_1 - T_5)$$

$$w_{5-6} = 0$$

Thermal efficiency of Dual cycle:

$$\eta_{(Dual)} = \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|} = 1 - \frac{Cv(T_5 - T_1)}{Cv(T_3 - T_2) + Cp(T_4 - T_3)}$$

$$\therefore \eta_{(Dual)} = 1 - \left(\frac{1}{r_c}\right)^{k-1} \times \left[\frac{r_p \times r_{cutt\,off}^{k} - 1}{k \times r_p (r_{cut\,off} - 1) + r_p - 1}\right]$$

Differences between Otto, Diesel and Dual Cycles

The comparison of Otto, Diesel and Dual cycles can be made on the basis of compression ratio, maximum pressure, maximum temperature, heat input, work output etc.

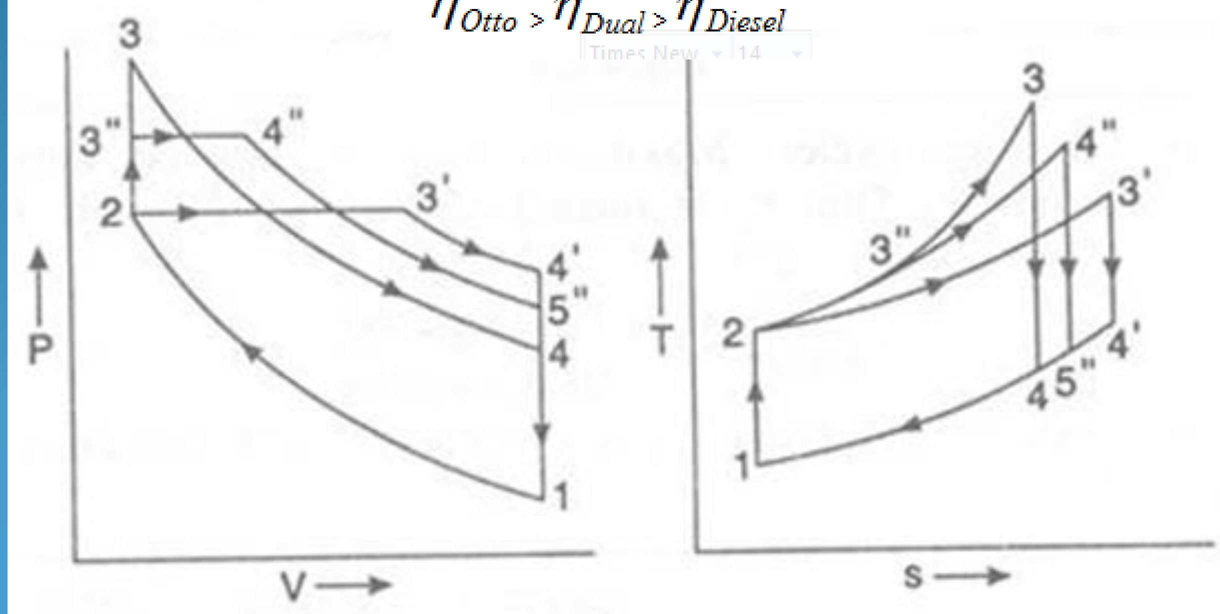
(A) For same compression ratio and same heat input:

When compression ratio is constant process (1-2) remains the same for all the three cycles. And same heat is transferred in all three cycles.

Q_L (heat rejected) from Otto cycle < Q_L from Dual cycle < Q_L from Diesel cycle

So that:

$$\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$$



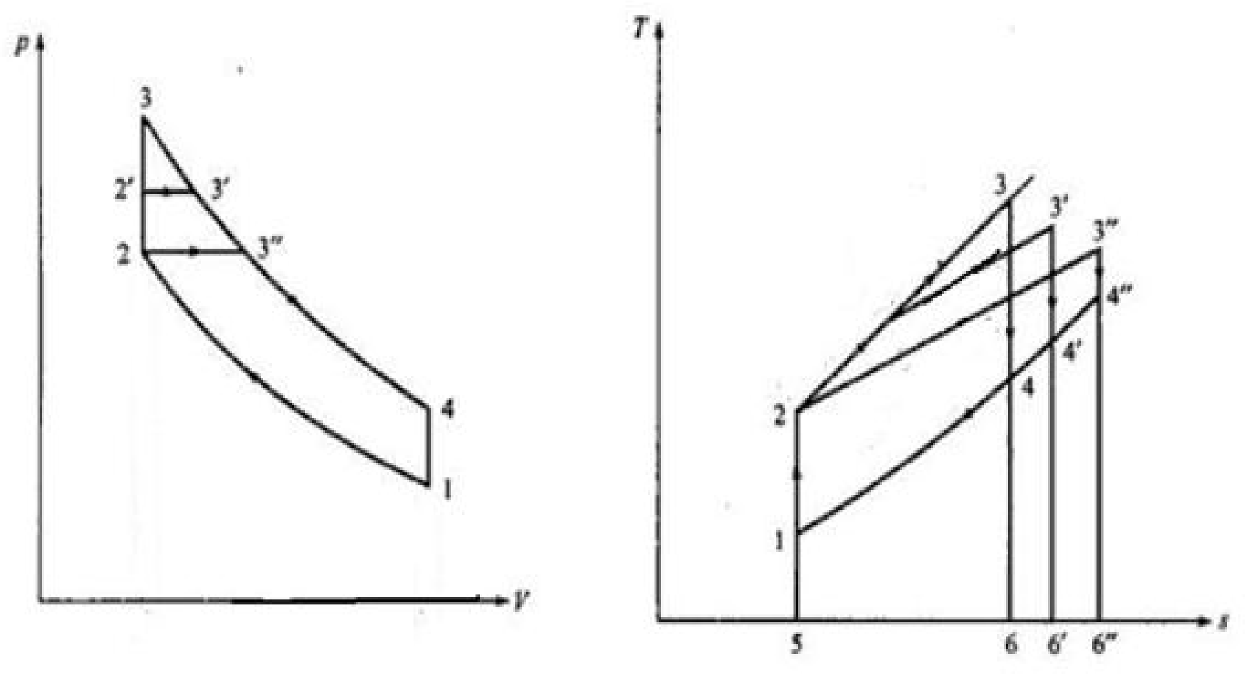
B: For the same compression ratio and heat rejected.

When compression ratio is constant process (1-2) remains the same for all the three cycles. And same heat reject in all three cycles.

Q_H (heat added) from Otto cycle > Q_H from Dual cycle > Q_H from Diesel cycle

So that:

$$\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$$

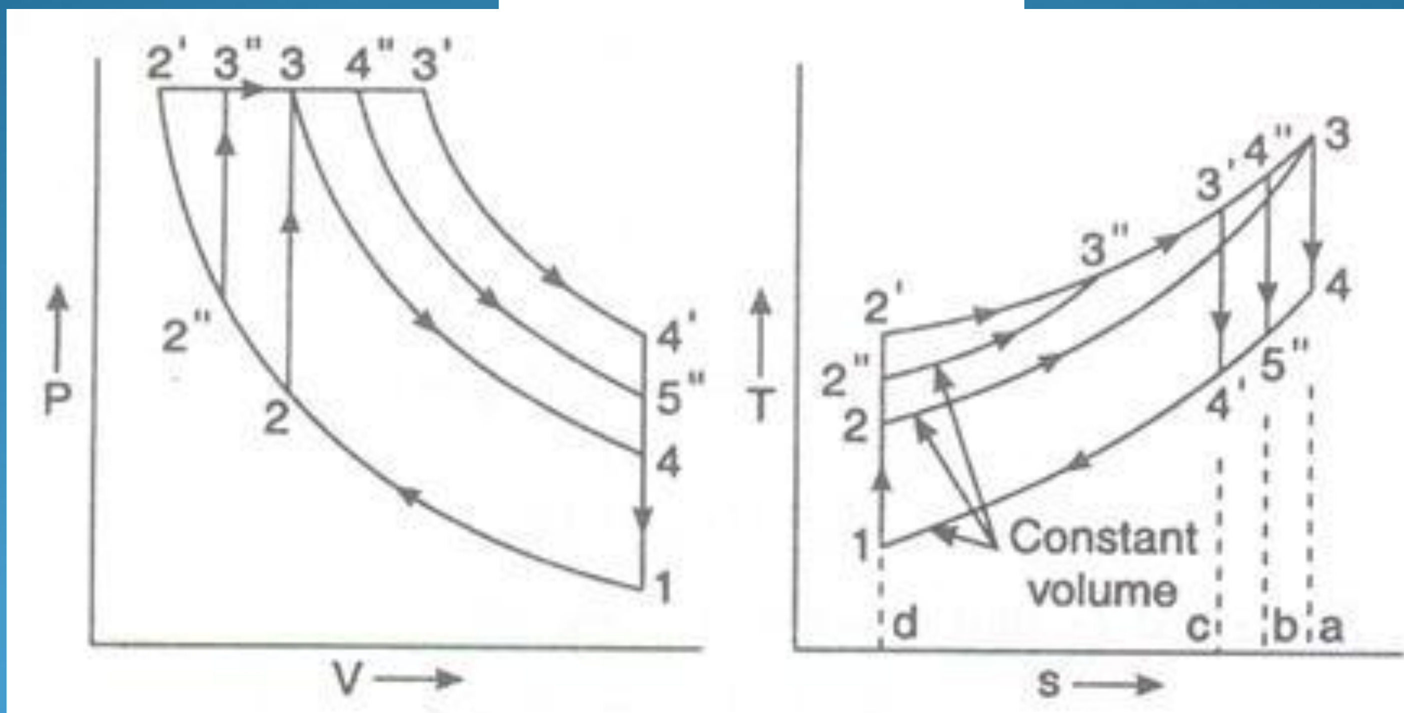


(C) For same maximum pressure and same heat input:

For the same maximum pressure 3.3’. and 3” must be on same pressure line and for the same heat input the area should be equal.

Q_L (heat rejected) from Diesel cycle < Q_L from Dual cycle < Q_L from Otto cycle

$$\eta_{Otto} < \eta_{dual} < \eta_{diesel}$$

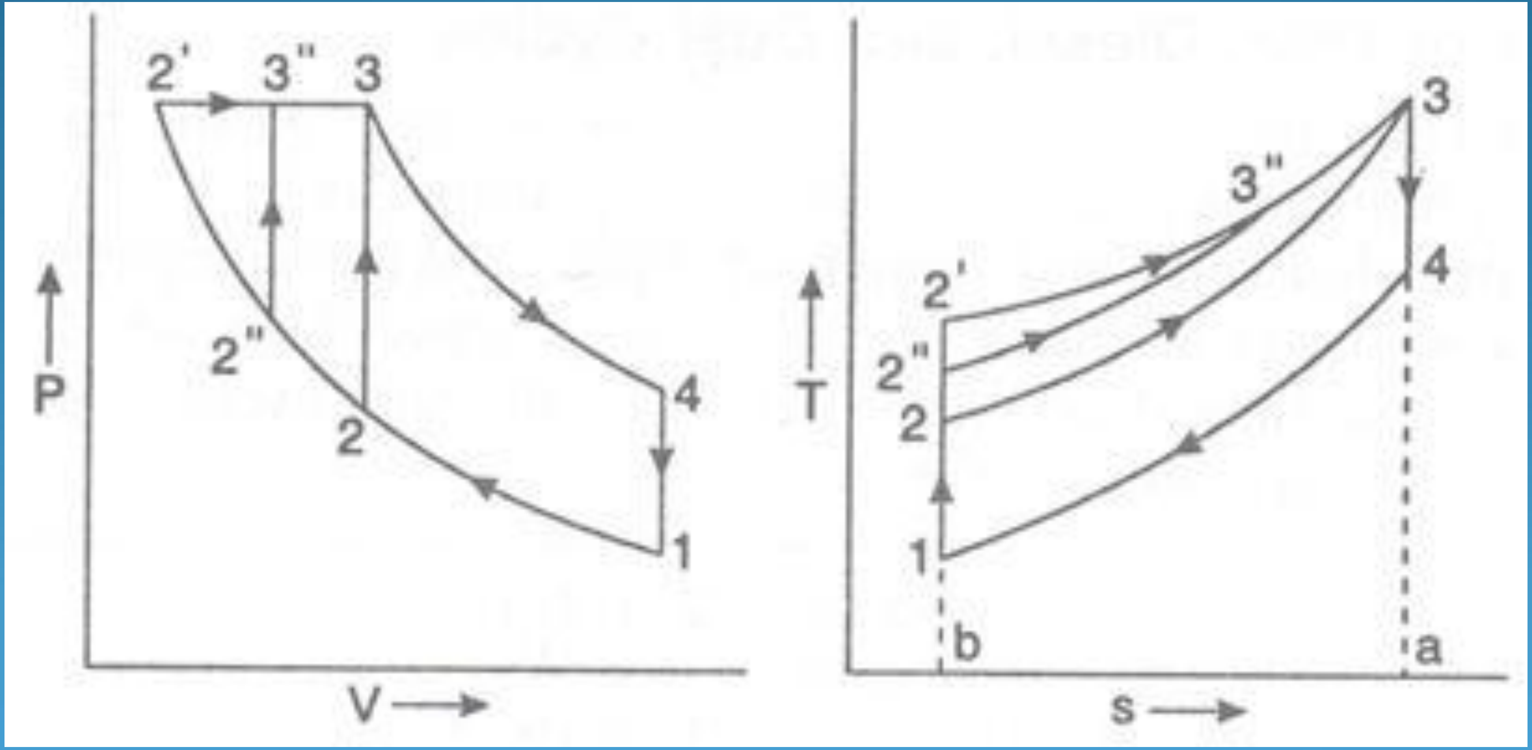


(D) For same pressure and temperature:

It is clear from the figure that the heat rejected by all three cycles, Otto, Diesel and Dual cycle remains the same .

Q_H (heat added) from Diesel cycle > Q_H from Dual cycle > Q_H from Otto cycle

$$\eta_{Otto} < \eta_{dual} < \eta_{diesel}$$



Thermodynamic cycles

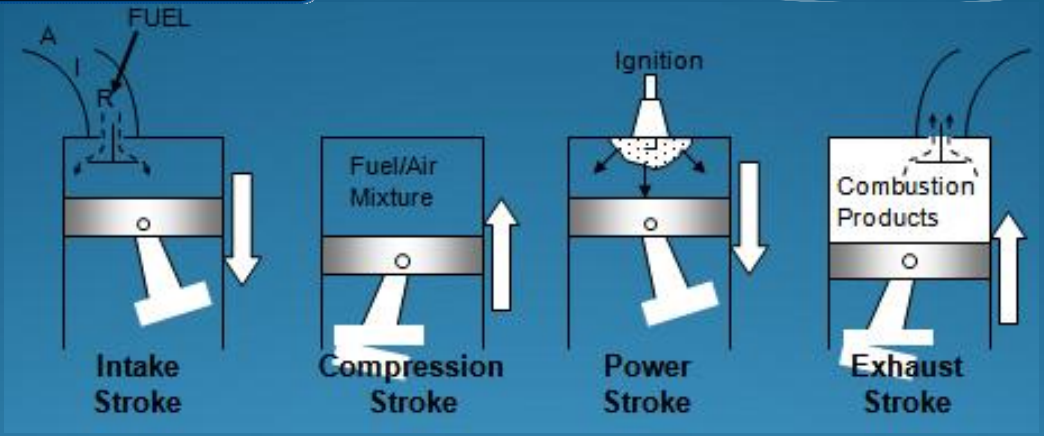
- **Air-fuel cycle**

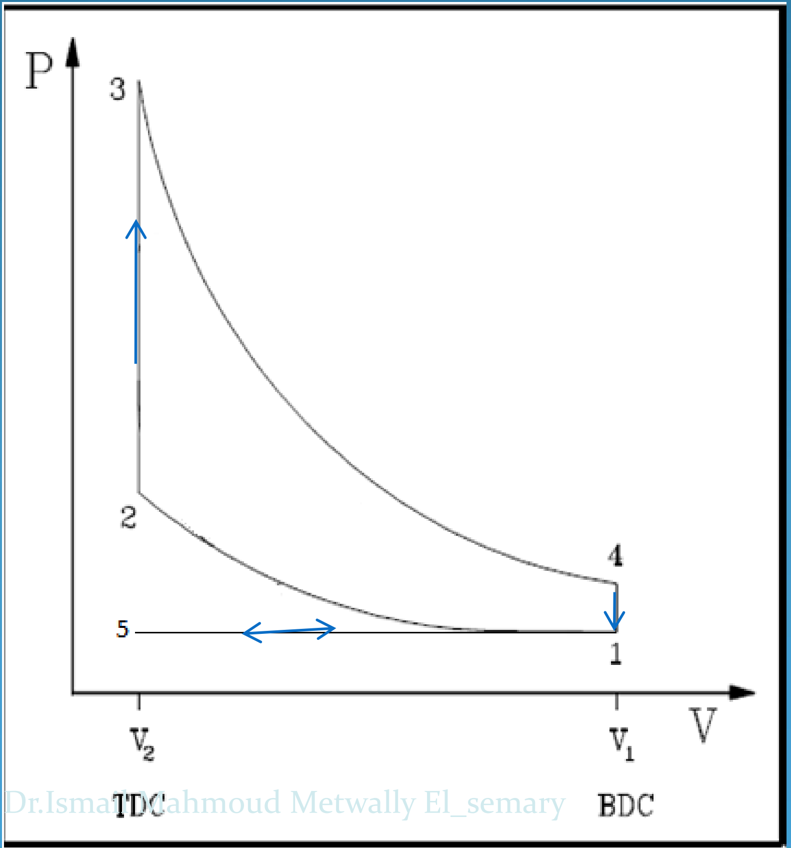
❑ Is the theoretical cycle based on the actual properties of the cylinder contents.

- **Simplifications to the Theoretical Air-Fuel Cycle approximation.**

1- The actual composition of the cylinder contents.
2- The variation in the specific heat of the gases in the cylinder.
3- The dissociation effect.
4- The variation in the number of moles present in the cylinder as the pressure and temperature change.
5- No chemical changes in either fuel or air prior to combustion.
6- Combustion takes place instantaneously at top dead center.
7- All processes are adiabatic.
8- The fuel is mixed well with air.

Thermodynamic cycles





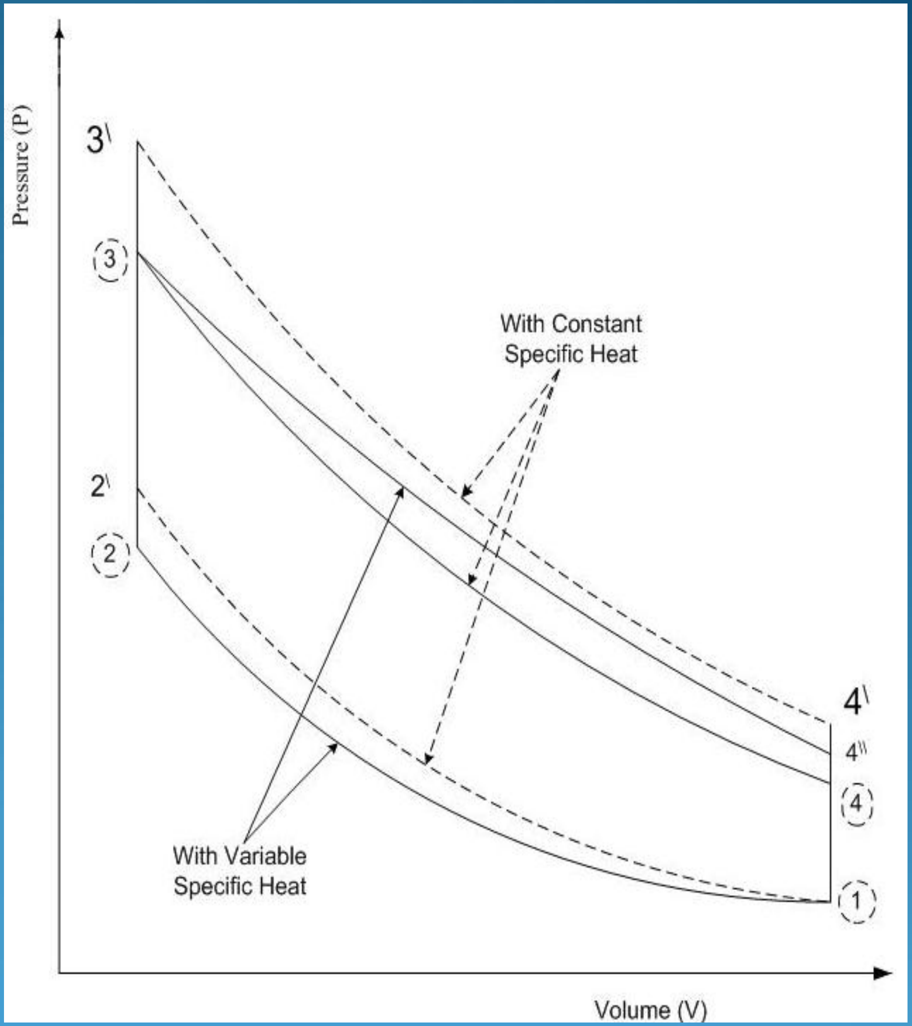
Variation of specific heats:

- All gases, except mono-atomic gases, show an increase in specific heat as temperature increase. The specific heat may be written in the form:

$$C = a+bT+cT^2$$

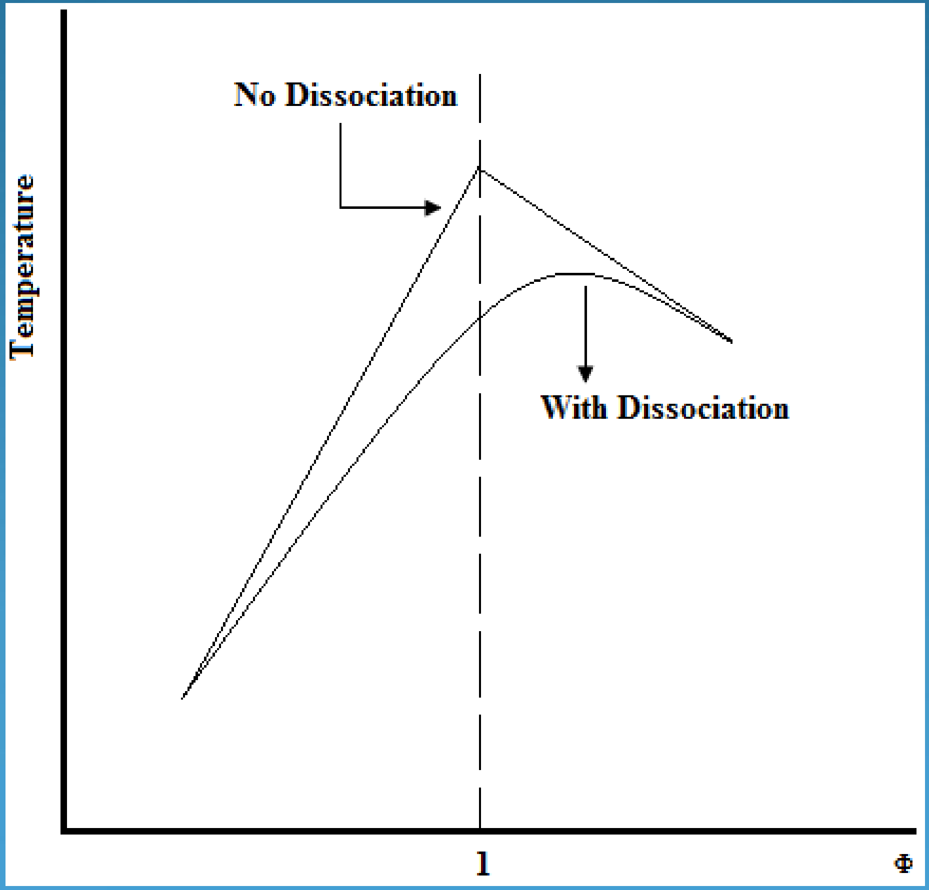
where T is the absolute temperature and a, b and c are constants for any specific gas.

C_p and C_v increase with temperature but $\gamma = \dfrac{c_p}{c_v}$ *decrease as the temperature increase.*



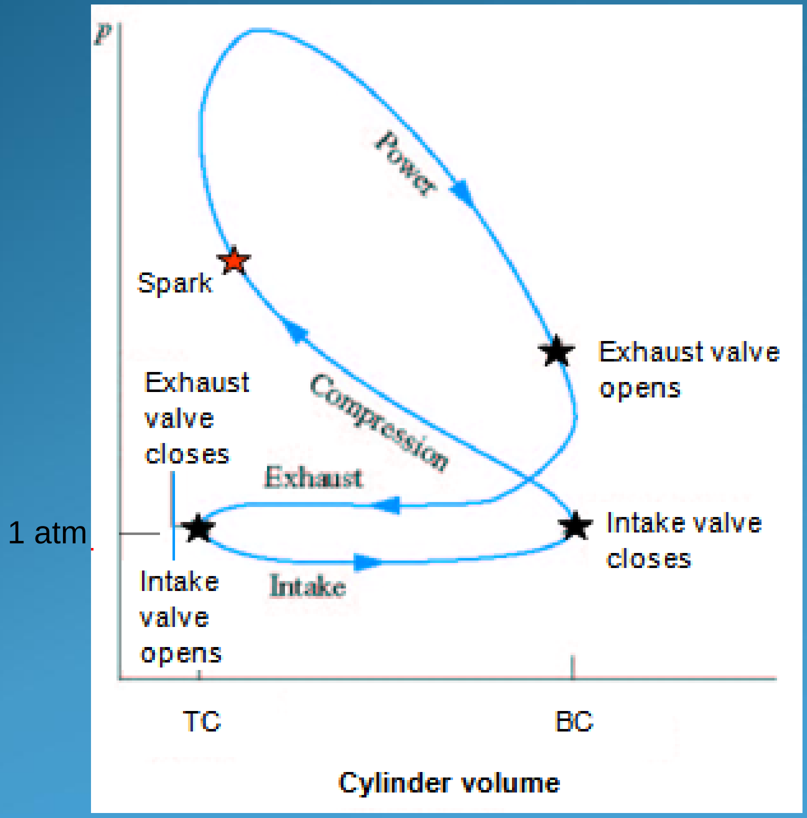
Dissociation effect:

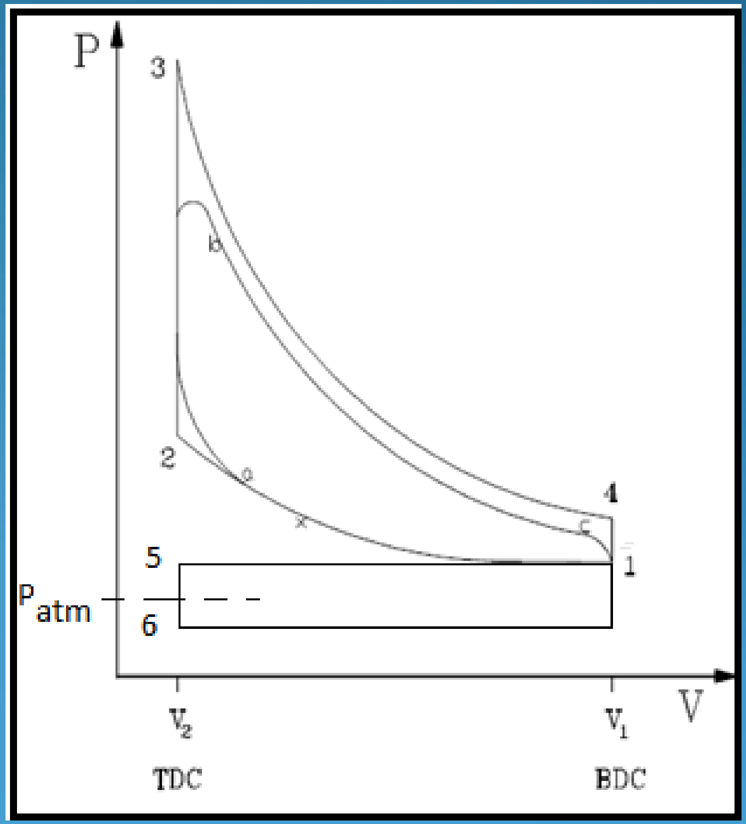
The effect of dissociation is a suppression of a part of the heat release during combustion and the liberation of it as expansion proceeds



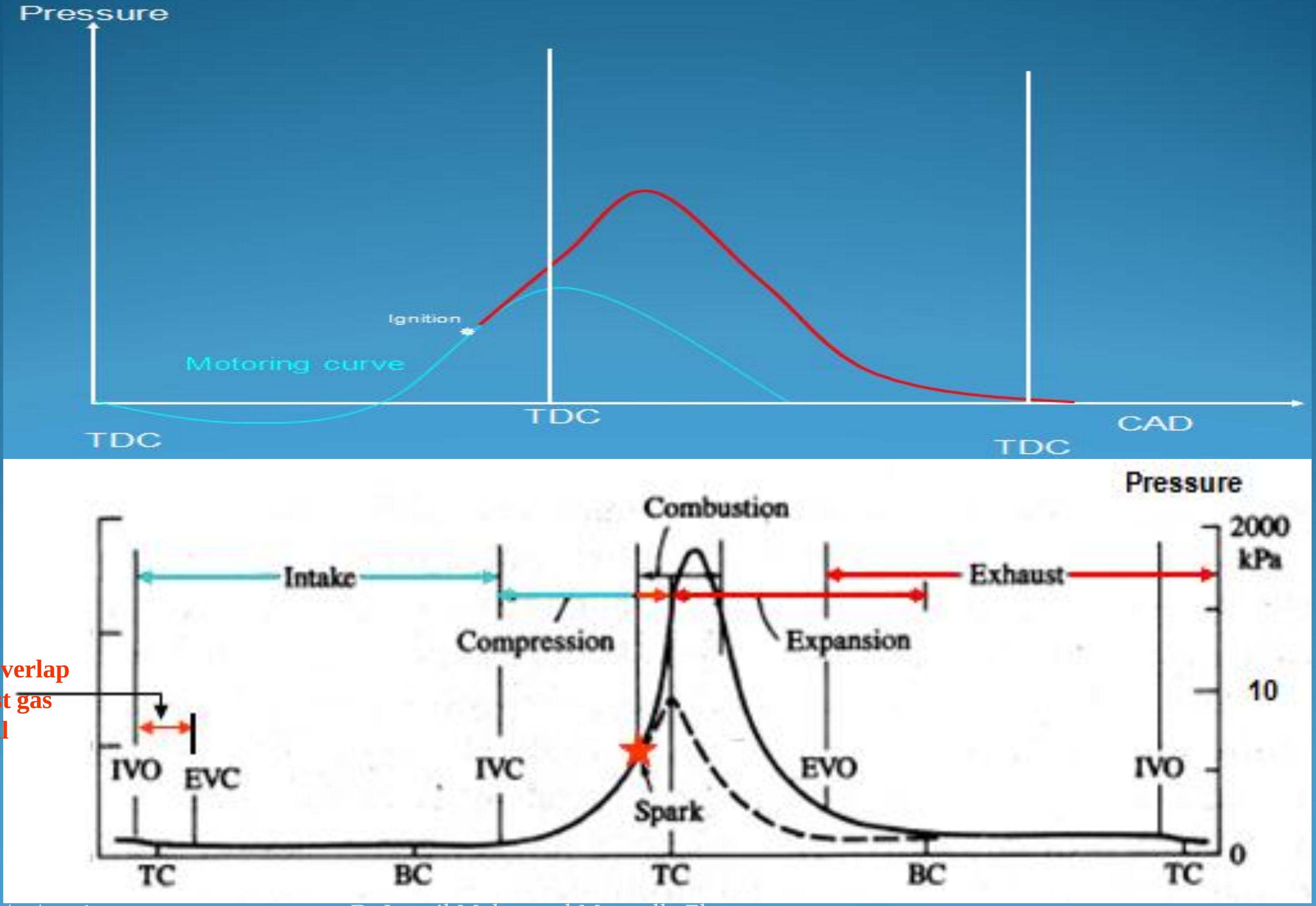
The actual cycle (Otto Cycle)

The actual cycle experienced by internal combustion engines is an open cycle with changing composition, actual cycle efficiency is much lower than the air standard efficiency due to various losses occurring in the actual engine.





P- θ Diagram



Valve overlap
Exhaust gas residual

Differences between Ideal and Actual Cycles

Differences between Ideal and Actual Cycles

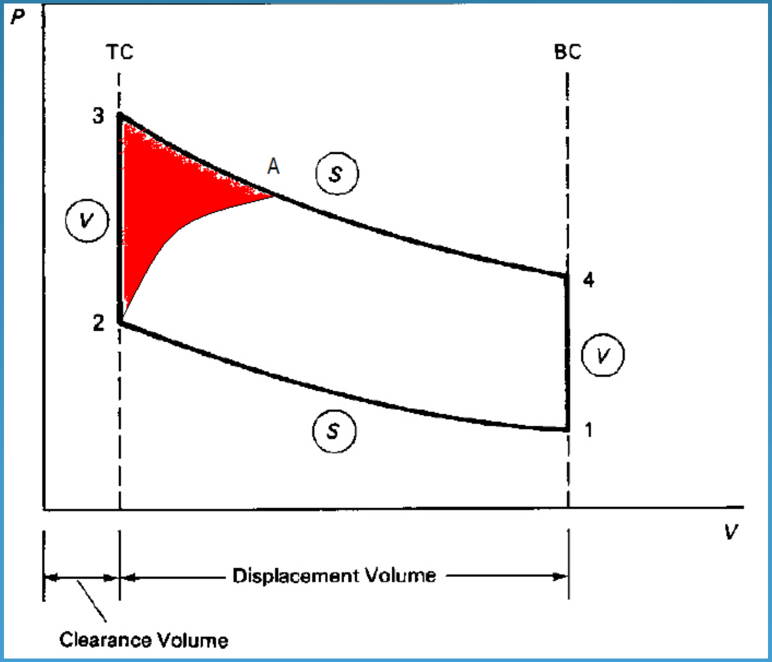
1. Leakage :-

A small amount of combustion pressure can leak past piston rings, Valves or Gasket.

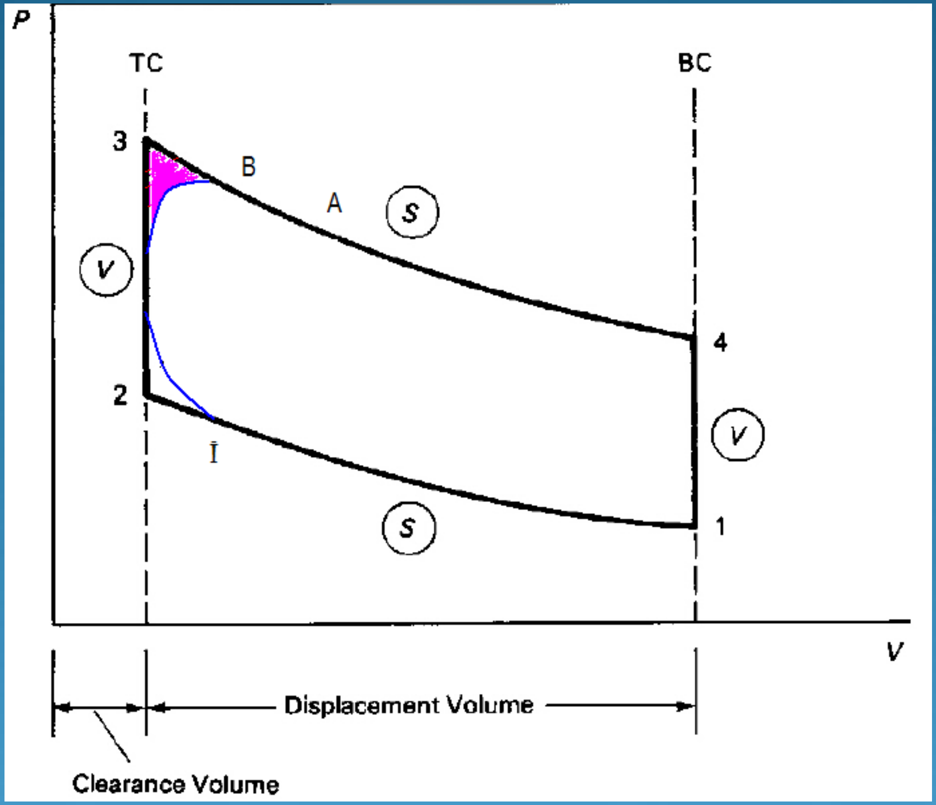
To find this problem make a **Compression Test** by measure the pressure inside the combustion chamber in the compression stroke and compared this with the standard value if P_{test} less than $P_{standard}$ there is leakage.

2. Time Losses:-

The loss of work due to piston movement during combustion process. Where, combustion does not occur "instantaneously", as a result, there is some piston motion during combustion, so combustion does not occur at constant volume.



Differences between Ideal and Actual Cycles



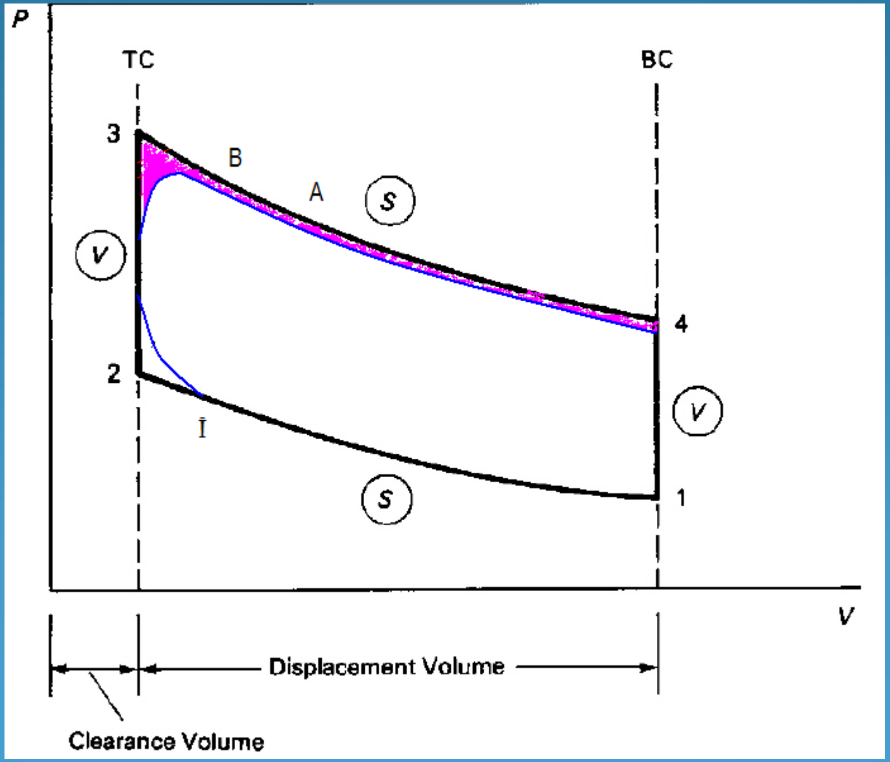
Ignition point: The point at which ignition starts

Spark Advance: "Number of CAD BTD at which Ignition Starts"

Differences between Ideal and Actual Cycles

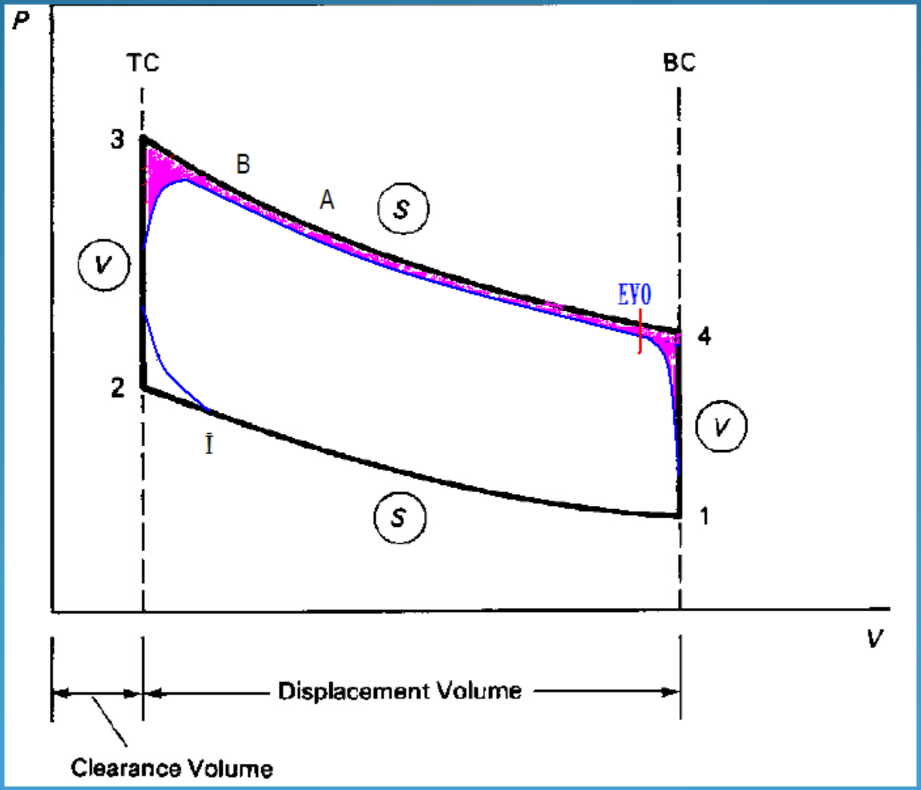
3. Heat Loss:-

The loss of work due to heat transfer during compression, combustion and expansion strokes



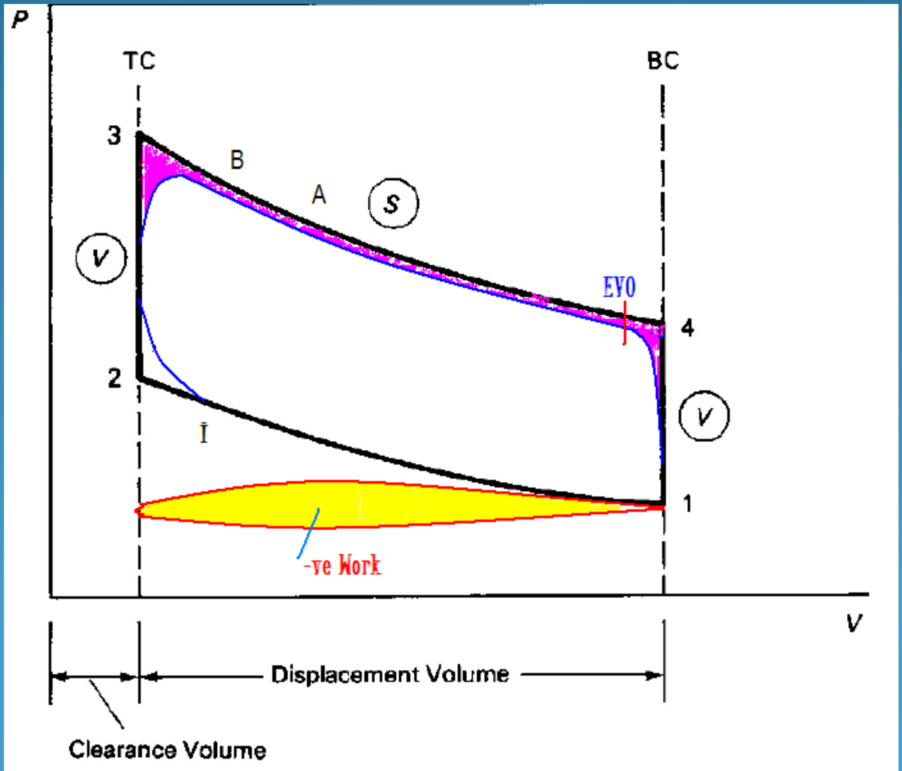
4. Exhaust loss:-

The loss of work due to opening the exhaust valve before B.D.C. by the hot exhaust gasses



5. Pumping work:-

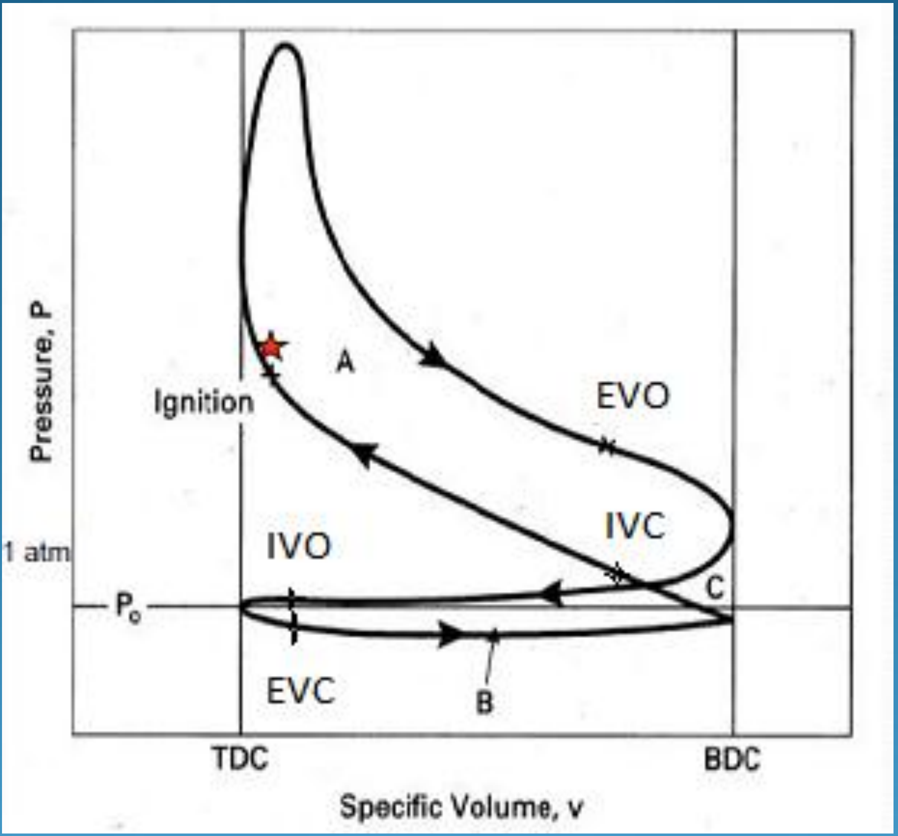
The work required to pump the charge inside the cylinder and exhaust gases outside the cylinder.



6. Progressive Burning :-

Combustion usually starts at a single point, and proceeds with a moving "flame front". Combustion time varies with fuel composition (which affects flame speed), combustion chamber size and shape, location and number of ignition sources, engine speed, and other engine operating conditions.

Differences between Ideal and Actual Cycles



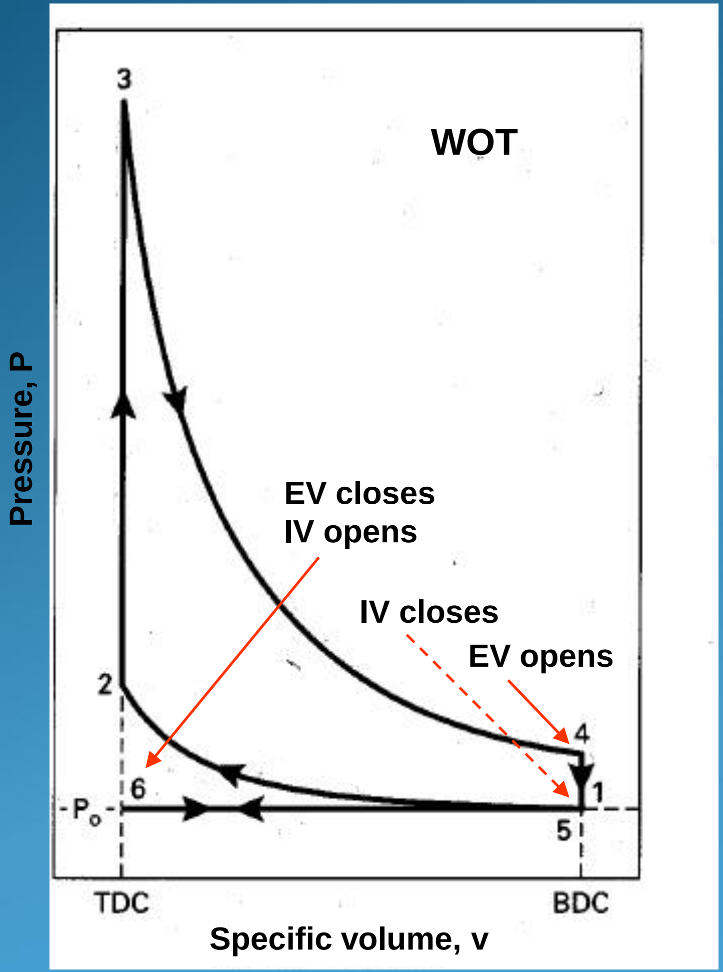
Actual cycle efficiency < Air standard cycle efficiency

$$\eta_r = \frac{\eta_{act}}{\eta_{airst}}$$

Intake and Exhaust Strokes

Ideal Intake and Exhaust Strokes

(Neglecting the effect of the throttle)

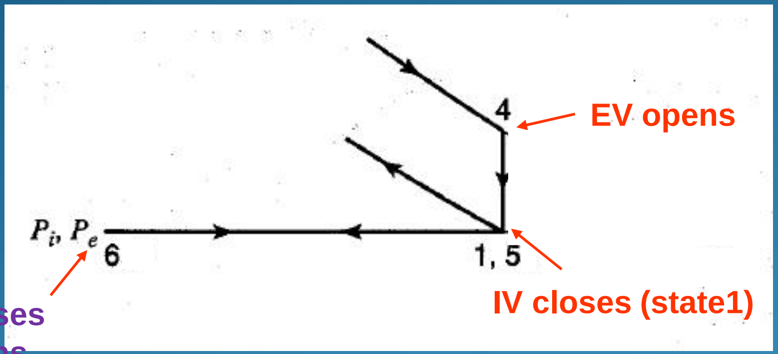


The intake & exhaust processes are often shown as a constant pressure processes, e.g., 5→6 and 6→1
For WOT states 1 and 5 are the same.

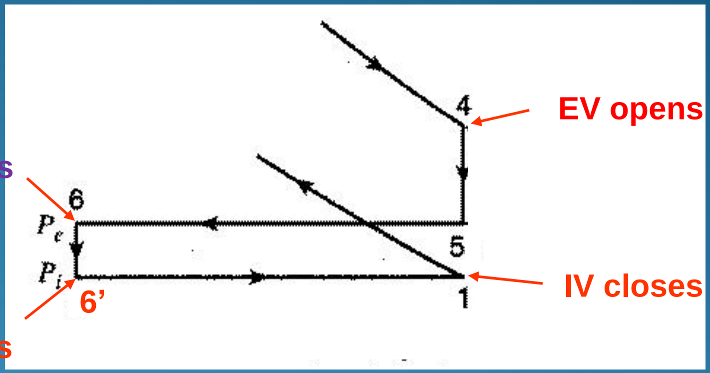
Accordingly for Ideal process:
Valves operate instantaneously, intake and exhaust process are adiabatic and constant pressure.

Ideal Intake and Exhaust Strokes

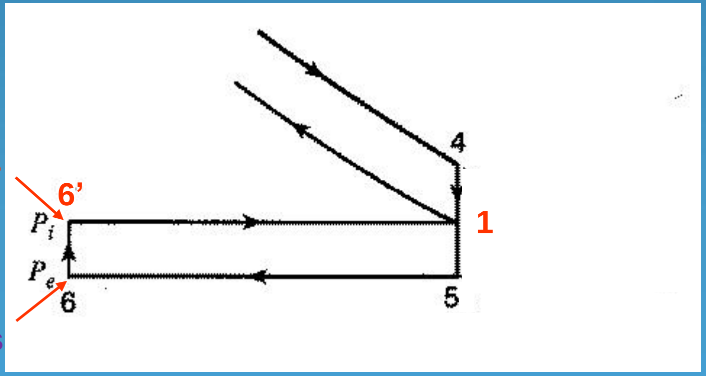
(Considering the effect of the throttle)



Unthrottled: $P_i = P_e = 1$ atm



Throttled: $P_i < P_e$

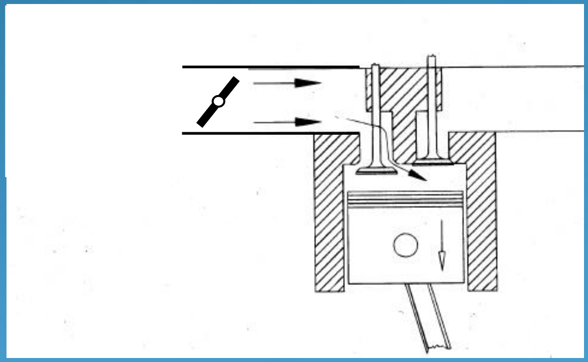


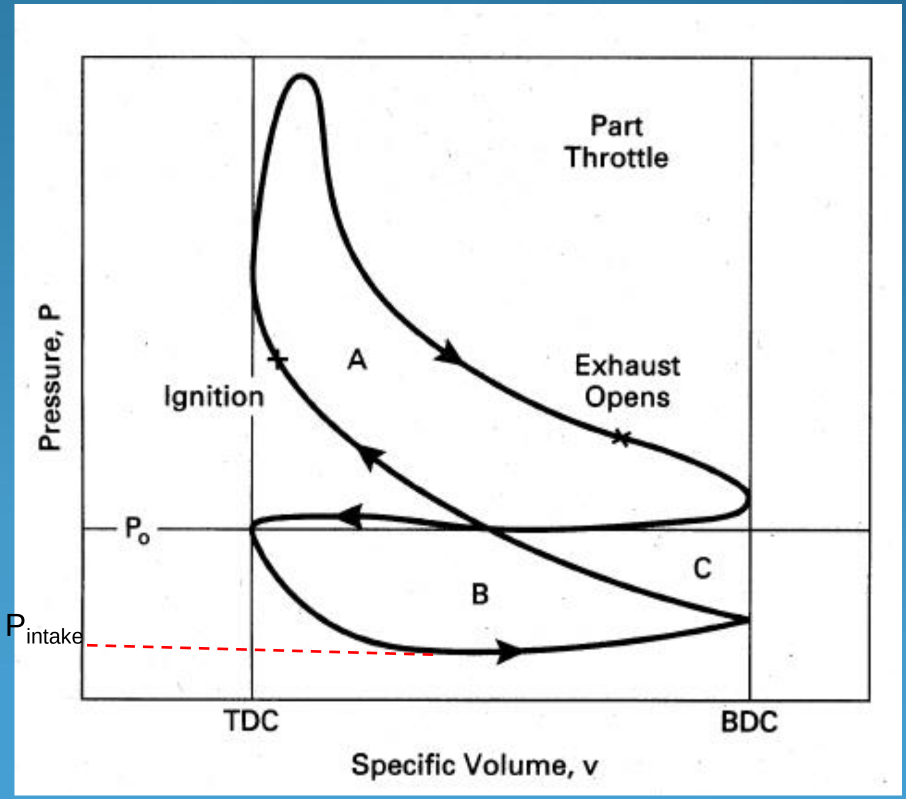
Supercharged: $P_i > P_e$

Actual Intake and Exhaust Strokes

Throttled stroke

The pressure at the intake port is significantly lower than atmospheric pressure



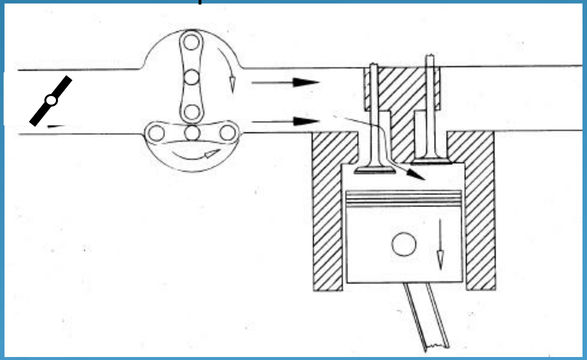


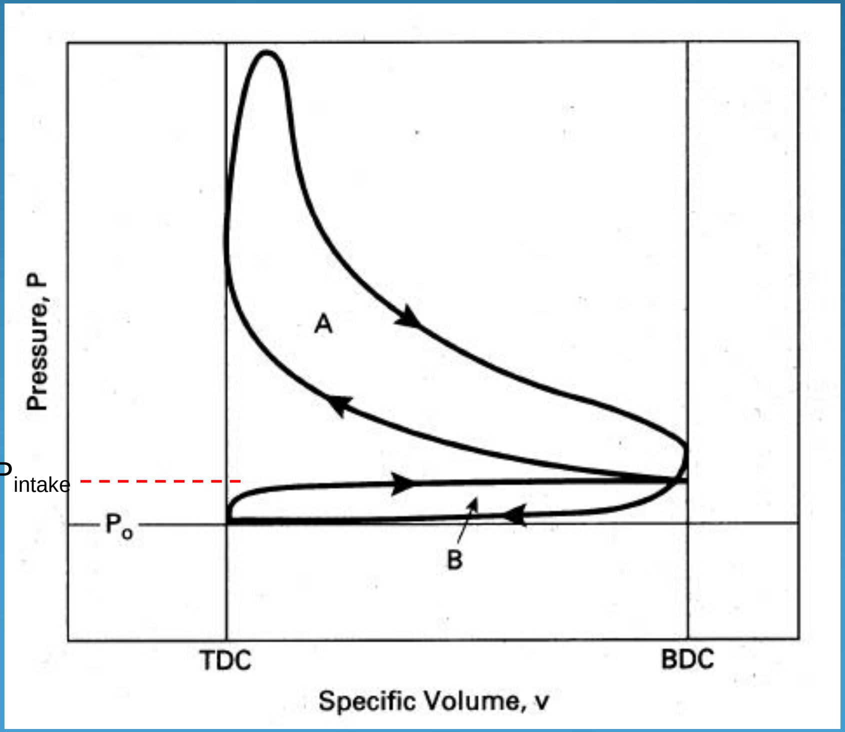
Actual Intake and Exhaust Strokes

Supercharging

Engines with superchargers or turbochargers have intake pressures greater than the exhaust pressure, yielding a positive pump work

Compressor

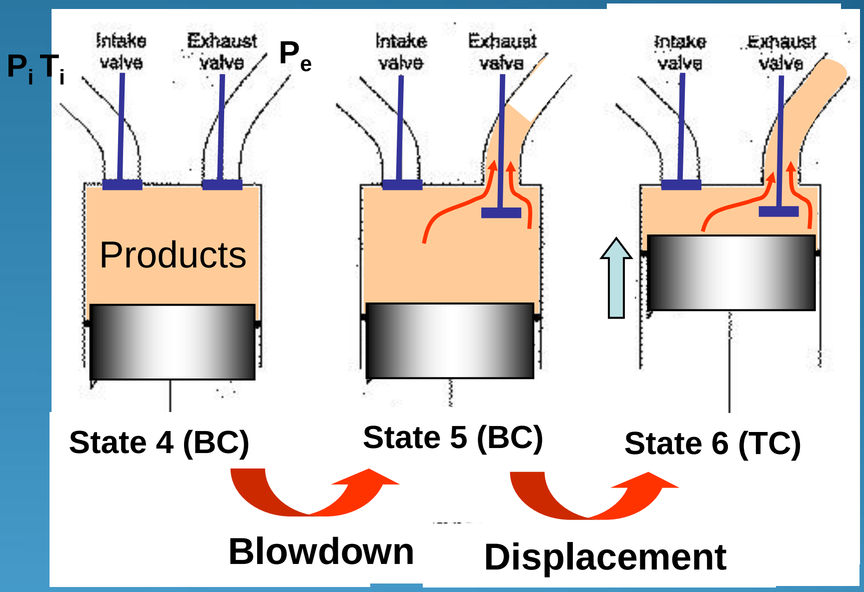




Actual Exhaust Stroke

The actual exhaust process consists of two phases:
i) Blowdown ii) Displacement

Blowdown – At the end of the power stroke when the exhaust valve opens the cylinder pressure is much higher than the exhaust manifold pressure which is typically at 1 atm ($P_4 > P_e$), so the cylinder gas flows out through the exhaust valve and the pressure drops to P_e.

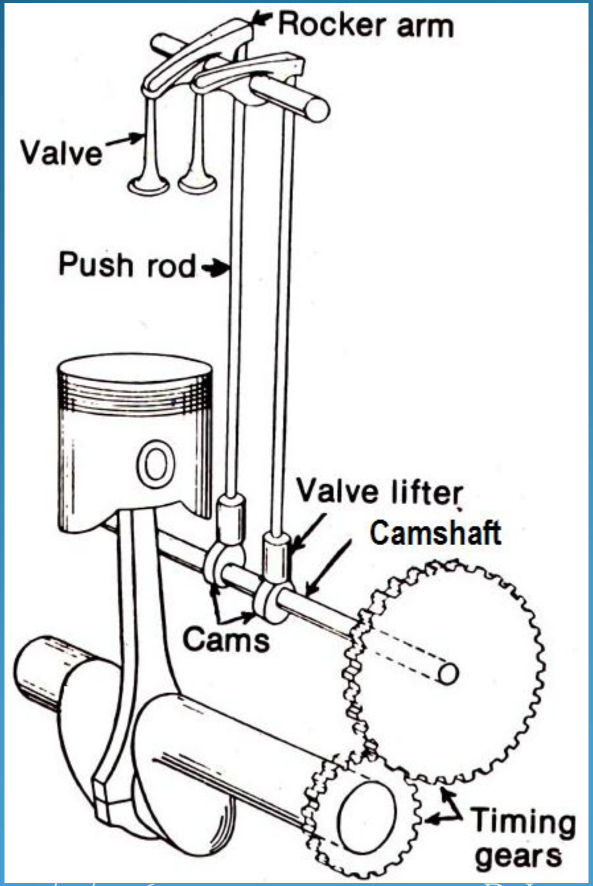


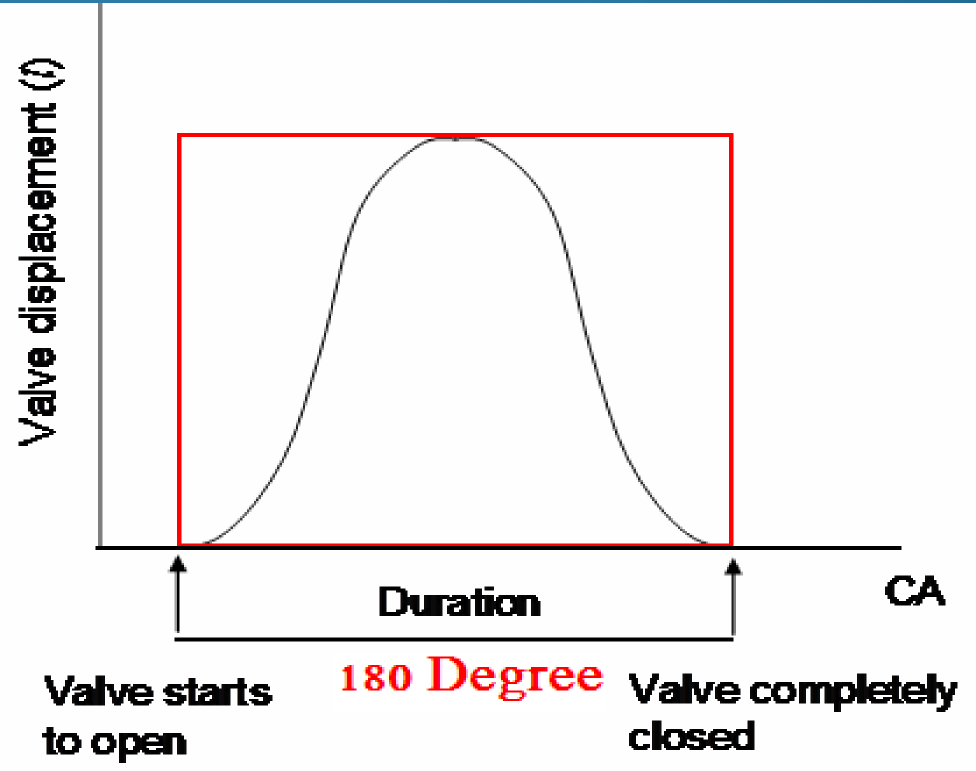
Displacement – Remaining gas is pushed out of the cylinder by the piston moving to TDC.

Valve Timing

Valve Opening and Closing

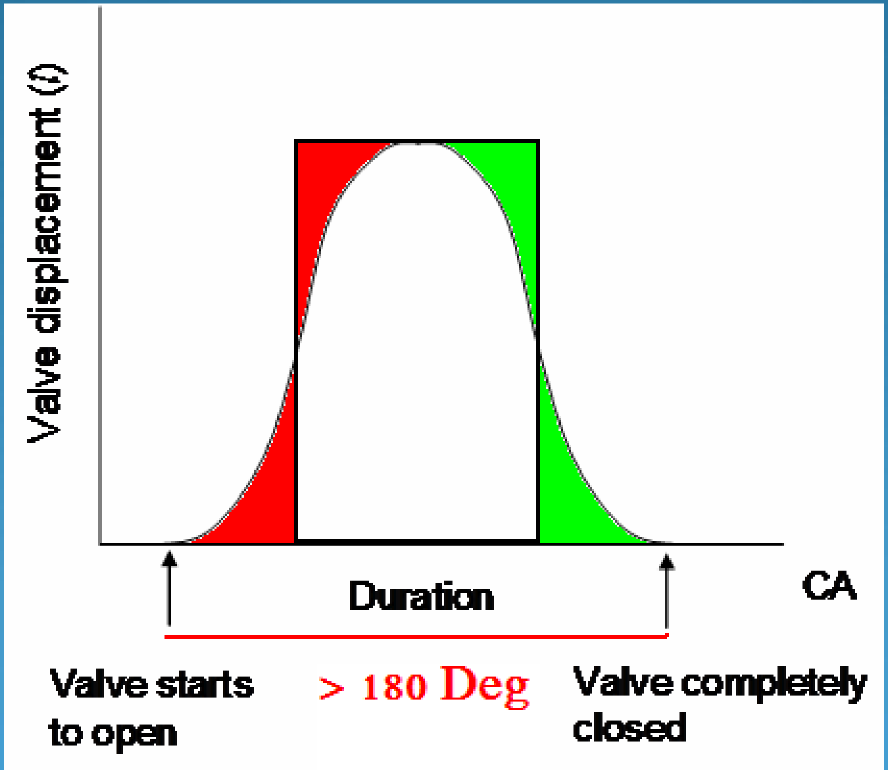
In thermo cycles it is assumed the valves open and close instantaneously
In reality a cam is used to progressively open and close the valves, the lobes are contoured so that the valve land gently on the seat.





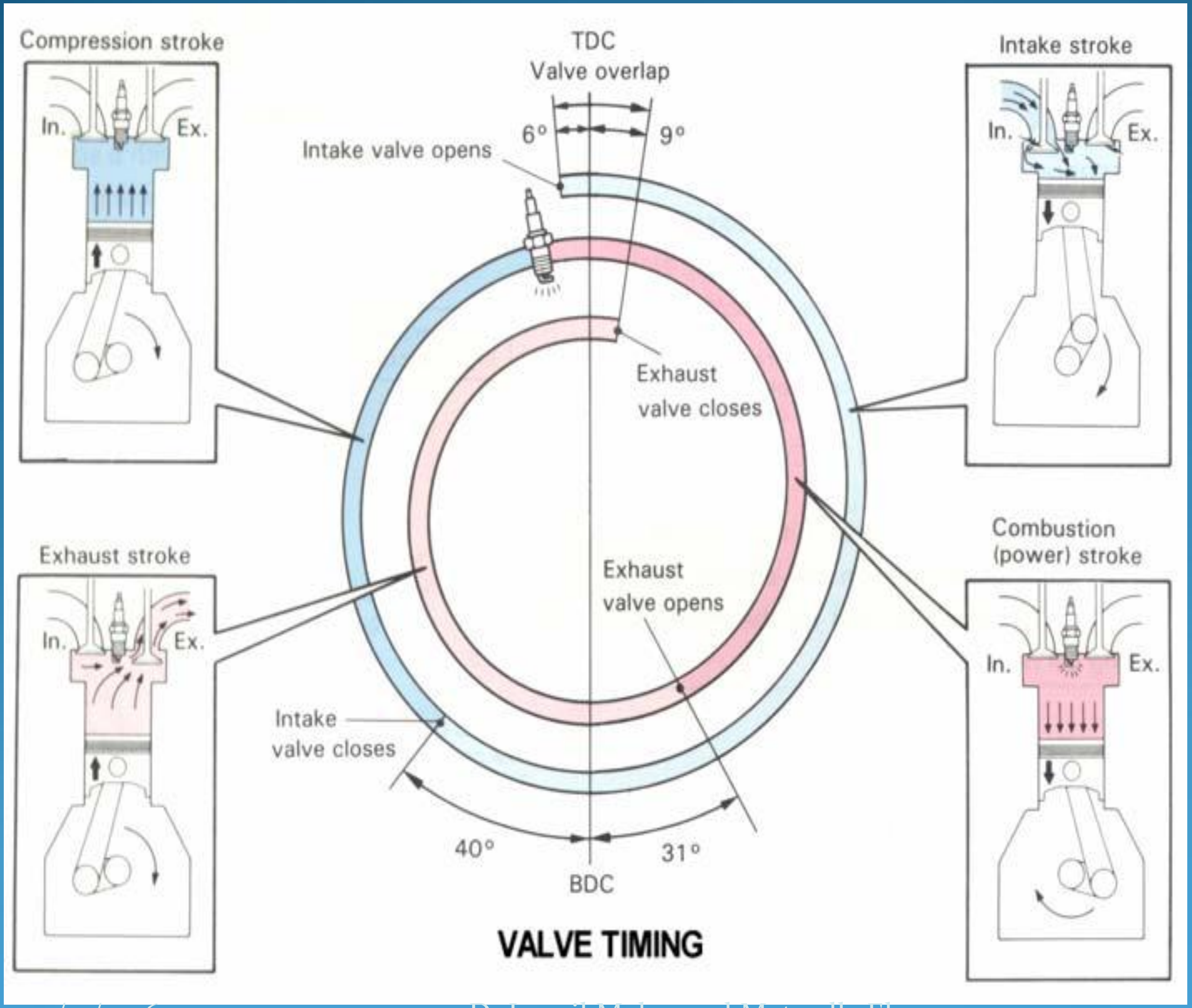
Valve Opening and Closing

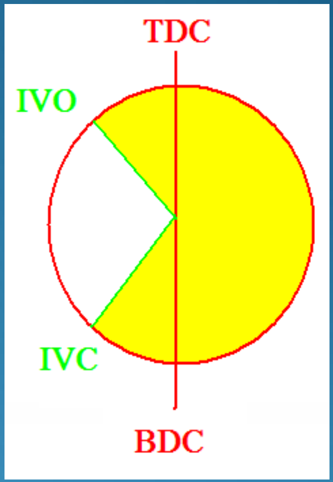
In real engines in order to ensure that the valve is ***fully*** open during a stroke, for high volumetric efficiency, the valves are open for longer than 180o.

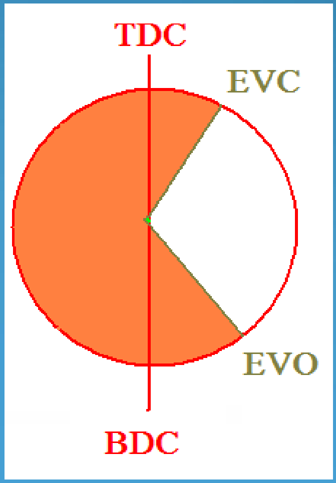


Valve Timing

The exhaust valve opens before BDC and closes after TDC and the intake valve opens before TDC and closes after BDC.

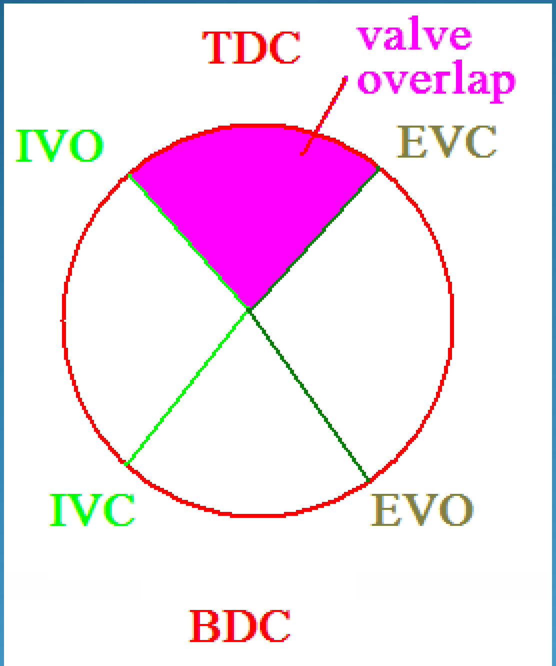


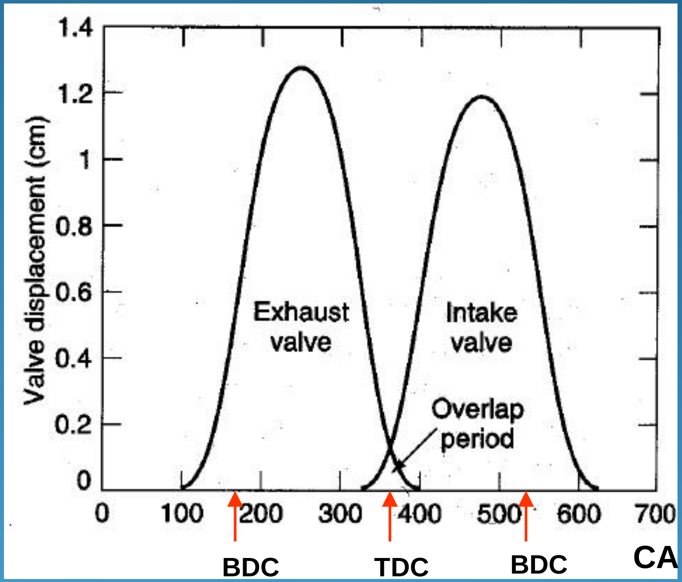




Valve Timing

At TDC there is a period of time called **valve overlap** where both the intake and exhaust valves are open.





Definitions:-

Valve timing:-

No. of crank angles during which the valve is open.

Valve overlap:-

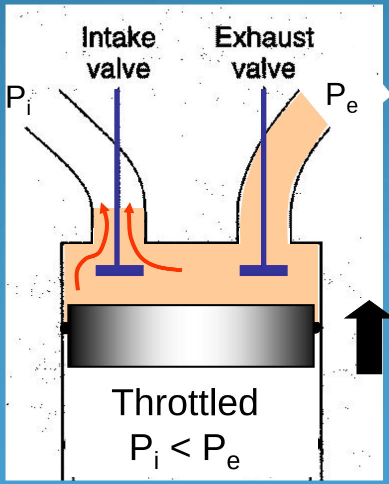
No. of crank angles during which the two valves are open at the same time

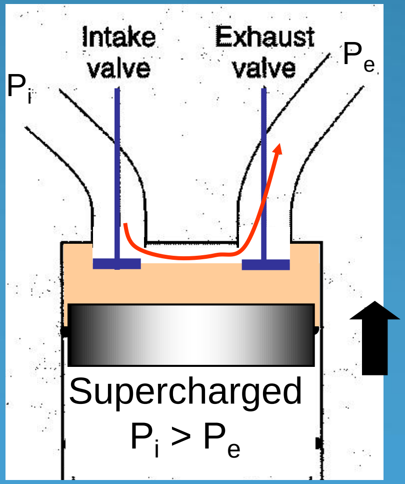
Valve overlap

When the intake valve opens BTDC the cylinder pressure is at roughly P_e

Part throttle ($P_i < P_e$): residual gas flows into the intake port. During intake stroke the residual gas is first returned to the cylinder then fresh gas is introduced. Residual gas reduces part load performance.

Supercharged ($P_i > P_e$): fresh gas can flow out the exhaust valve

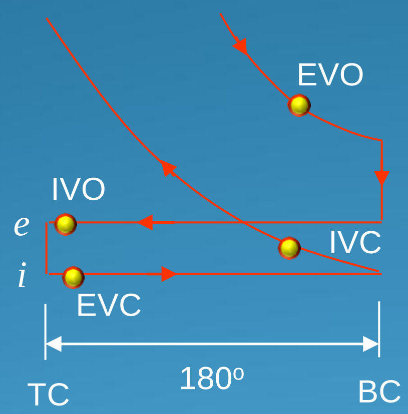




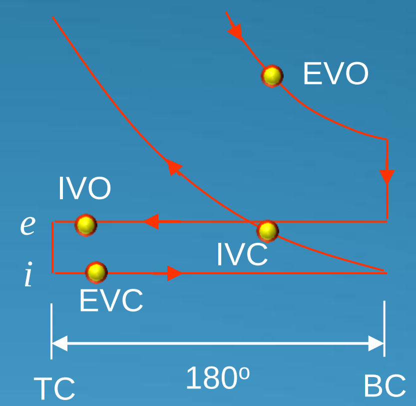
Valve Timing

		Open	Close	Duration
Intake	Conventional	5° before tdc	45° after bdc	230°
	High performance	30° before tdc	75° after bdc	285°
Exhaust	Conventional	45° before bdc	10° after tdc	235°
	High performance	70° before bdc	35° after tdc	285°

Conventional



Performance



@1000 rpm intake duration: 230º = 38.4 ms
@2500 rpm 230º = 15.4 ms
@5000 rpm 230º = 7.7 ms, 285º = 9.5 ms

Valve Timing

		Open	Close	Duration
Intake	Conventional	5° before tdc	45° after bdc	230°
	High performance	30° before tdc	75° after bdc	285°
Exhaust	Conventional	45° before bdc	10° after tdc	235°
	High performance	70° before bdc	35° after tdc	285°

Overlap

15°

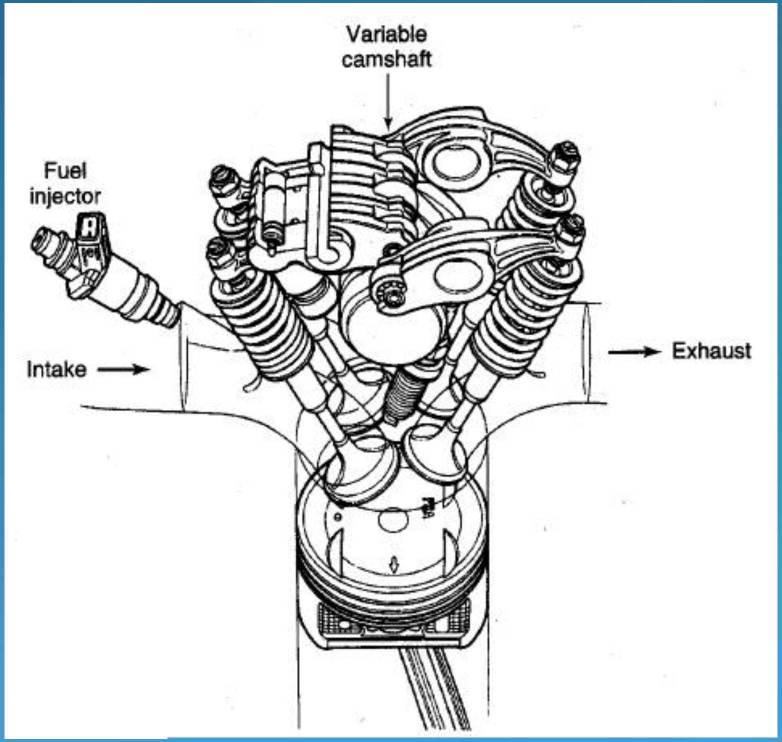
65°

At high engine speeds less time available for fresh gas intake so need more crank angles to get high volumetric efficiency → large valve overlap

At low engine speed and part throttle valve overlap is minimized by reducing the number of CA the intake valve stays open.

Valve Timing

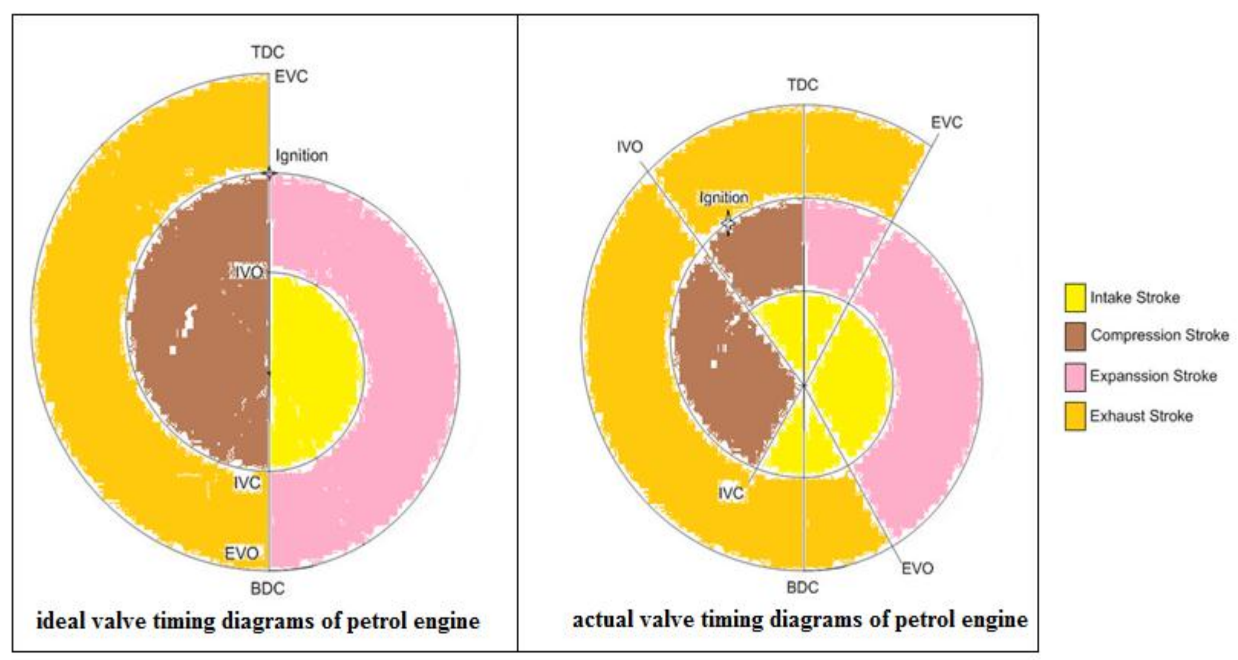
Honda Variable valve Timing and lift Electronic Control (VTEC)



Solenoid Activated Valves

Valve Timing

➢**The difference between ideal and actual valve timing diagrams of petrol engine**



For ideal valve time:

The intake valve open at TDC and closed at BDC.

The exhaust valve open at BDC and closed at TDC.

For actual valve time:

The intake valve open (10^o) before TDC and closed (50^o:70^o)after BDC.

The exhaust valve open (40^o:50^o) before BDC and closed (15^o:30^o) after TDC.

This difference to overcome the mechanical motion of the valve.