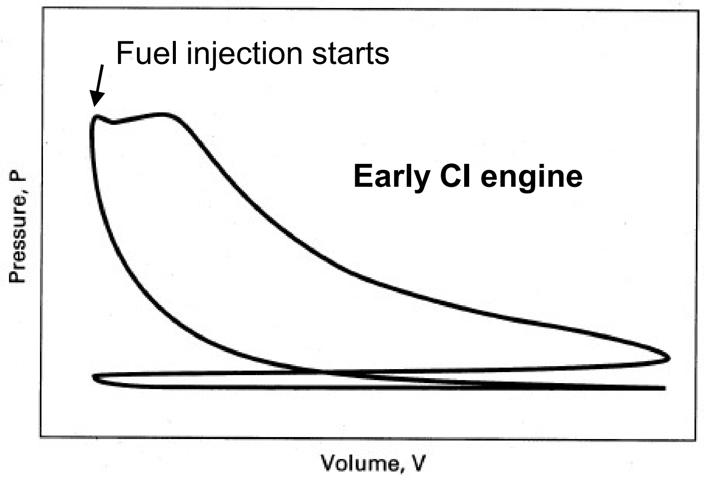
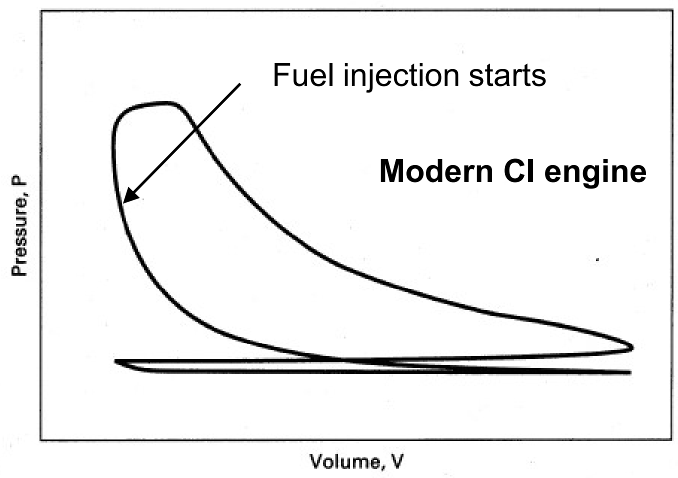
Thermodynamic Cycles for CI engines

- In early CI engines the fuel was injected when the piston reached TDC and thus combustion lasted well into the expansion stroke.

- In modern engines the fuel is injected before TDC (about 20^o)

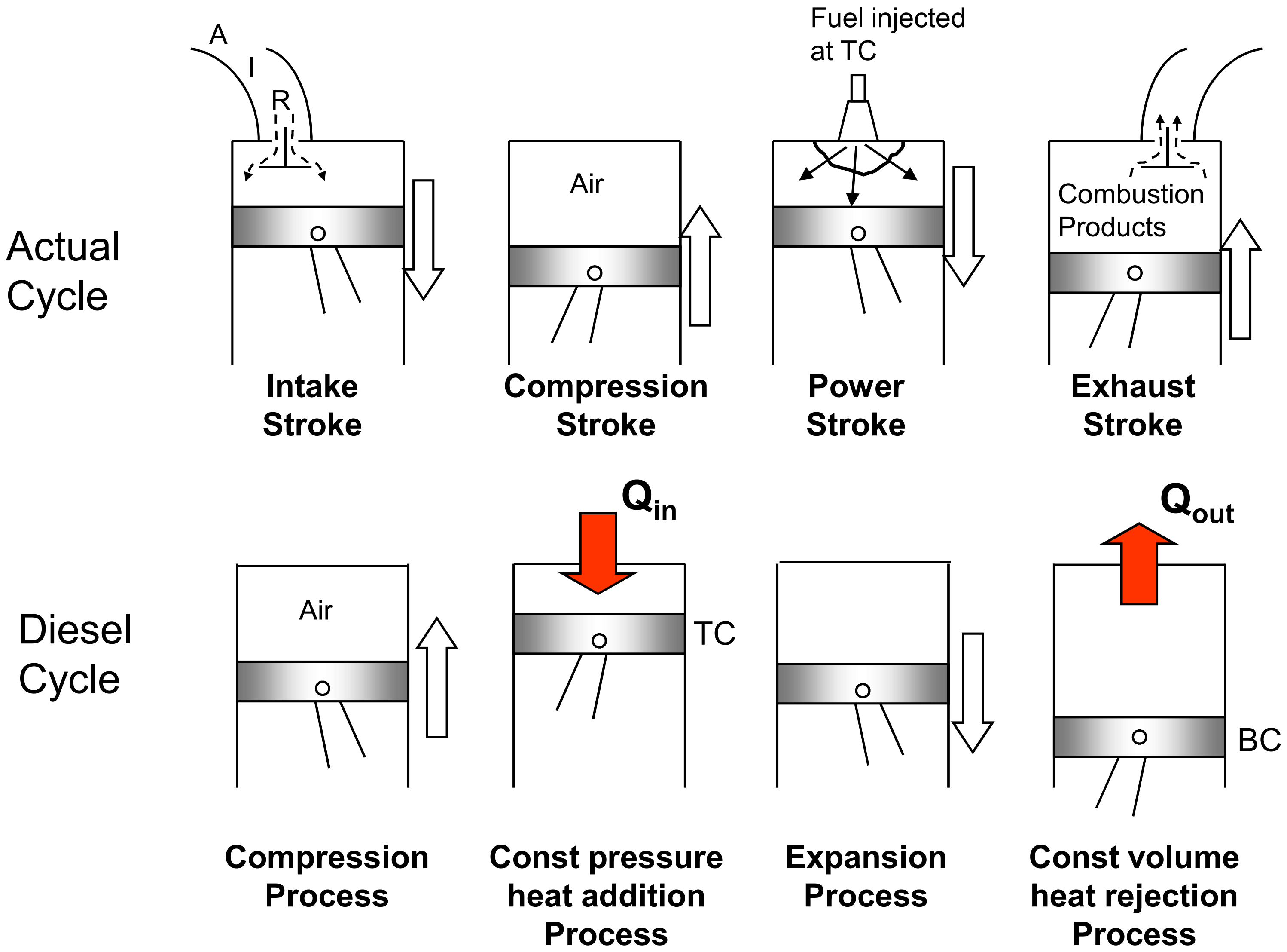




- The combustion process in the early CI engines is best approximated by a constant pressure heat addition process → **Diesel Cycle**

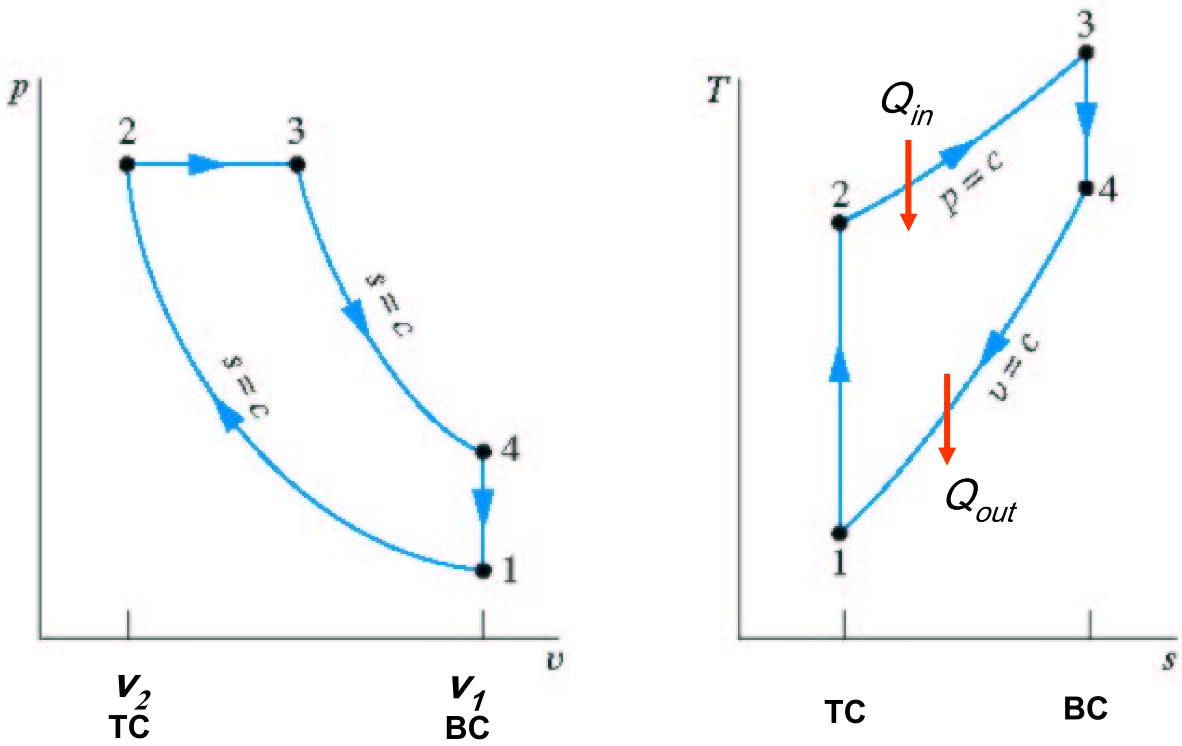
- The combustion process in the modern CI engines is best approximated by a combination of constant volume & constant pressure → **Dual Cycle**

Early CI Engine Cycle vs Diesel 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



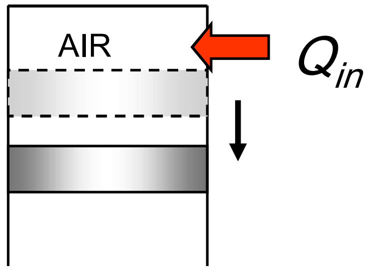
Cut-off ratio:

$$r_c = \frac{v_3}{v_2}$$

First Law Analysis of Diesel Cycle

Equations for processes 1→2, 4→1 are the same as those presented for the Otto cycle

2→3 <u>Constant Pressure Heat Addition</u>



$$(u_3 - u_2) = (+\frac{Q_{in}}{m}) - \frac{P_2(V_3 - V_2)}{m}$$

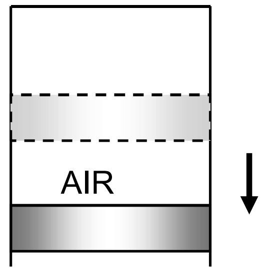
$$\frac{Q_{in}}{m} = (u_3 + P_3 v_3) - (u_2 + P_2 v_2)$$

$$\boxed{\frac{Q_{in}}{m} = (h_3 - h_2)}$$

$$P = \frac{RT_2}{v_2} = \frac{RT_3}{v_3} \rightarrow \boxed{\frac{T_3}{T_2} = \frac{v_3}{v_2} = r_c}$$

3 → 4 Isentropic Expansion

$$(u_4 - u_3) = \frac{\not{Q}}{m} - \left(+\frac{W_{out}}{m}\right)$$



$$\boxed{\frac{W_{out}}{m} = (u_3 - u_4)}$$

Note that $v_4 = v_1$, so: $\frac{v_4}{v_3} = \frac{v_4}{v_2} \cdot \frac{v_2}{v_3} = \frac{v_1}{v_2} \cdot \frac{v_2}{v_3} = \frac{r}{r_c}$ → $\boxed{\frac{v_4}{v_3} = \frac{r}{r_c}}$

$$\frac{P_4 v_4}{T_4} = \frac{P_3 v_3}{T_3} \rightarrow \boxed{\frac{P_4}{P_3} = \frac{T_4}{T_3} \cdot \frac{r_c}{r}}$$

Thermal Efficiency

$$\eta_{\substack{Diesel \\ cycle}} = 1 - \frac{Q_{out}/m}{Q_{in}/m} = 1 - \frac{u_4 - u_1}{h_3 - h_2}$$

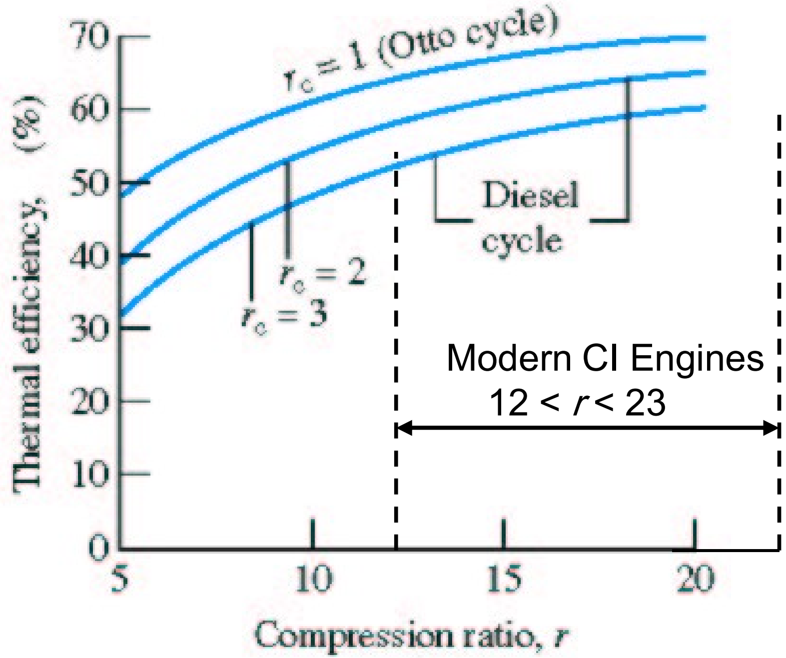
For cold air-standard the above reduces to:

$$\eta_{\substack{Diesel \\ const\ c_V}} = 1 - \frac{1}{r^{k-1}}\left[\frac{1}{k} \cdot \frac{\left(r_c^k - 1\right)}{\left(r_c - 1\right)}\right] \quad \text{recall,} \quad \eta_{Otto} = 1 - \frac{1}{r^{k-1}}$$

Note the term in the square bracket is always larger than one so for the same compression ratio, *r*, the Diesel cycle has a *lower* thermal efficiency than the Otto cycle

When r_c ($=v_3/v_2$)$\rightarrow$1 the Diesel cycle efficiency approaches the efficiency of the Otto cycle

Thermal Efficiency

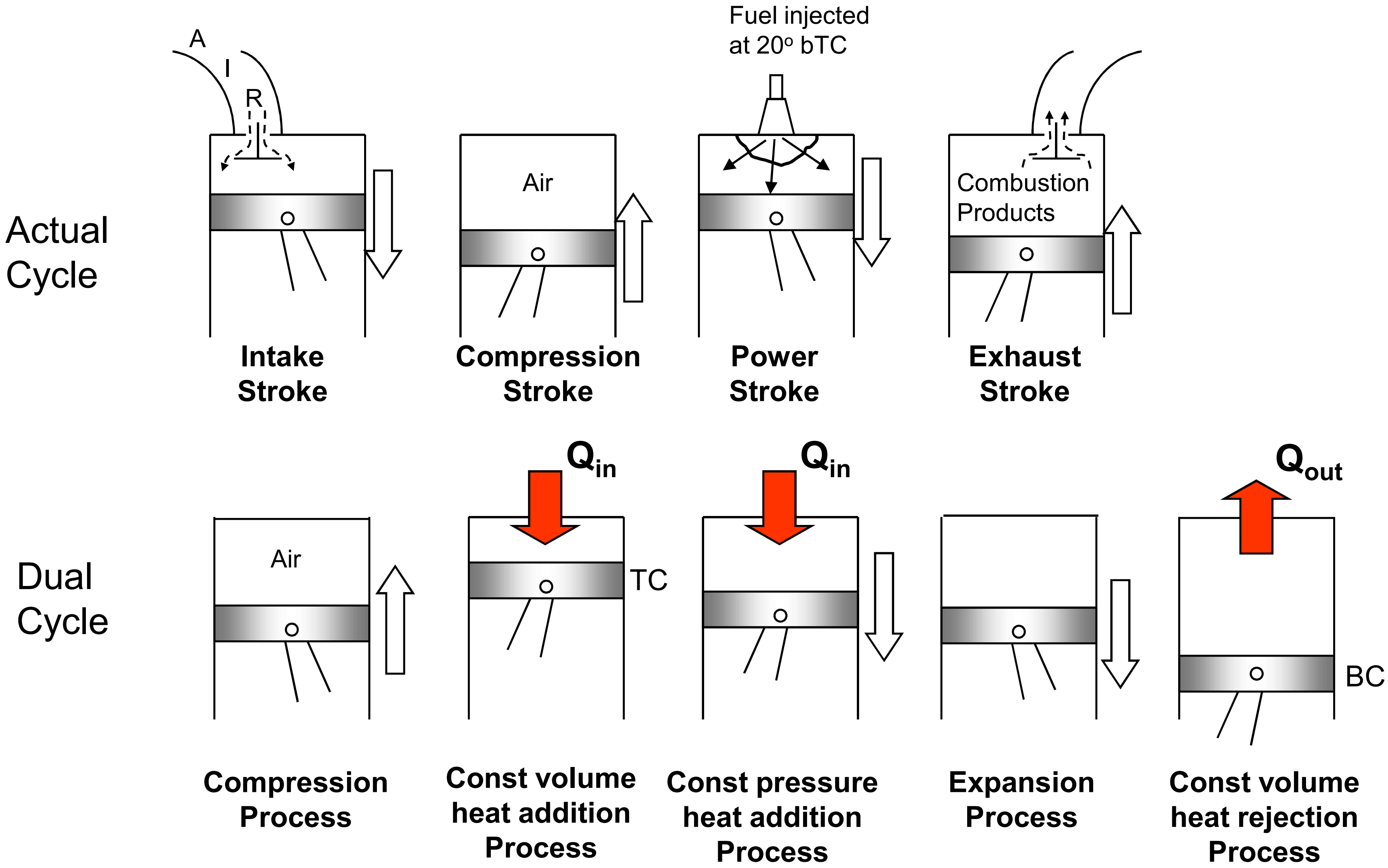


The cut-off ratio is not a natural choice for the independent variable
A more suitable parameter is the heat input, the two are related by:

$$r_c = 1 - \frac{k-1}{k}\left(\frac{Q_{in}}{P_1 V_1}\right)\frac{1}{r^{k-1}}$$

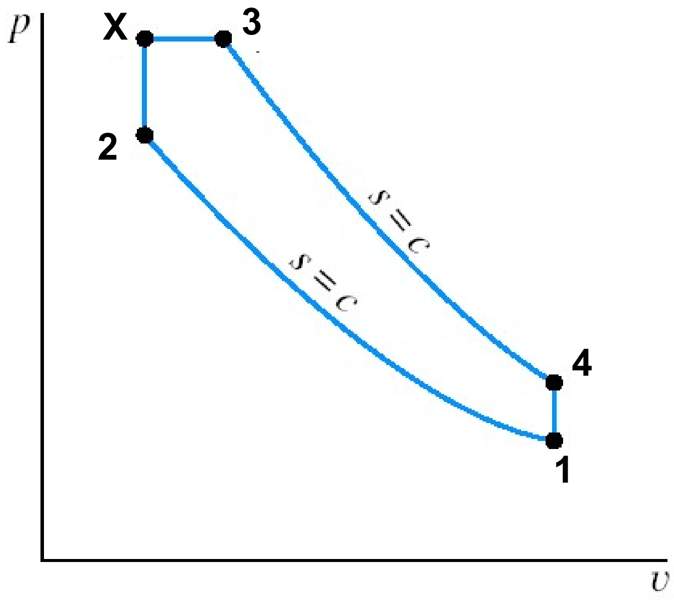
as $Q_{in} \rightarrow 0$, $r_c \rightarrow 1$ and $\eta \rightarrow \eta_{Otto}$

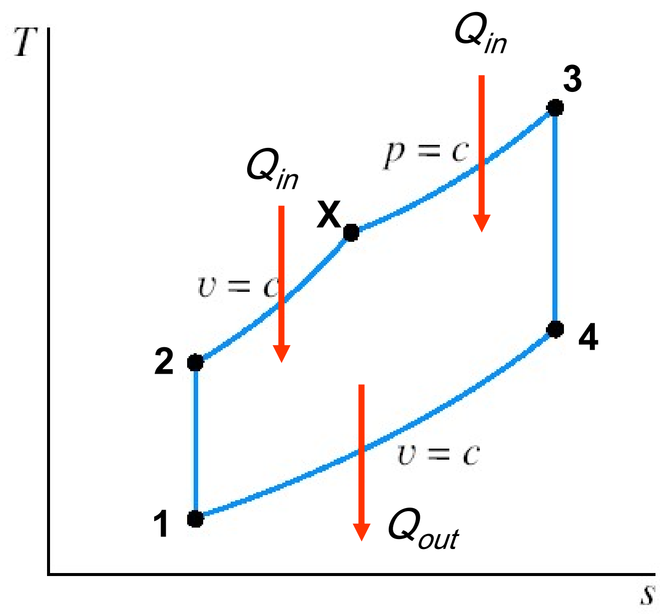
Modern CI Engine Cycle vs Dual Cycle



Dual Cycle

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





Thermal Efficiency

$$\eta_{\substack{Dual\\cycle}} = 1 - \frac{Q_{out}/m}{Q_{in}/m} \qquad \eta_{\substack{Dual\\cycle}} = 1 - \frac{u_4 - u_1}{(u_x - u_2) + (h_3 - h_x)}$$

For cold air-standard the above reduces to:

$$\eta_{\substack{Dual\\const\, c_v}} = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\alpha r_c^{\gamma} - 1}{(\alpha - 1) + \alpha\gamma(r_c - 1)}\right]$$

$$\text{where } r_c = v_3/v_x \text{ and } \alpha = p_3/p_2$$

Note, the Otto cycle ($r_c = 1$) and the Diesel cycle ($\alpha = 1$) are special cases:

$$\eta_{Otto} = 1 - \frac{1}{r^{\gamma-1}} \qquad \eta_{Diesel} = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{1}{k} \cdot \frac{(r_c^{\gamma} - 1)}{(r_c - 1)}\right]$$

Comparison between Otto, Diesel and Dual cycles

The use of the Dual cycle requires information about either:

i) the fractions of constant volume and constant pressure heat addition
(common assumption is to *equally* split the heat addition), or

ii) maximum pressure p_3.

iii) Transformation of r_c and α into more natural variables yields

$$r_c = 1 - \frac{\gamma - 1}{\alpha k}\left[\left(\frac{Q_{in}}{p_1 V_1}\right)\frac{1}{r^{\gamma-1}} - \frac{\alpha - 1}{\gamma - 1}\right] \qquad \alpha = \frac{1}{r^{\gamma}}\frac{p_3}{p_1}$$

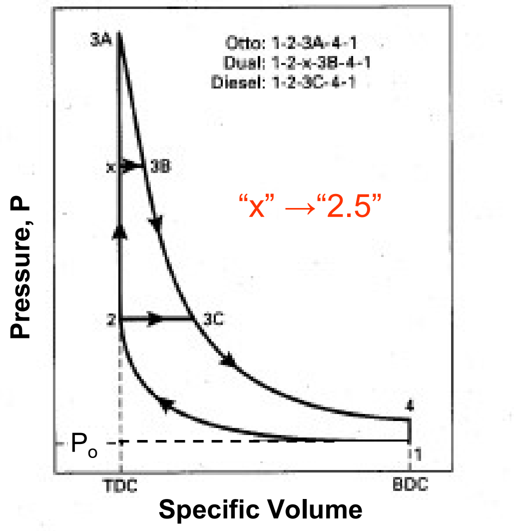
For the same inlet conditions p_1, V_1 and the same compression ratio:

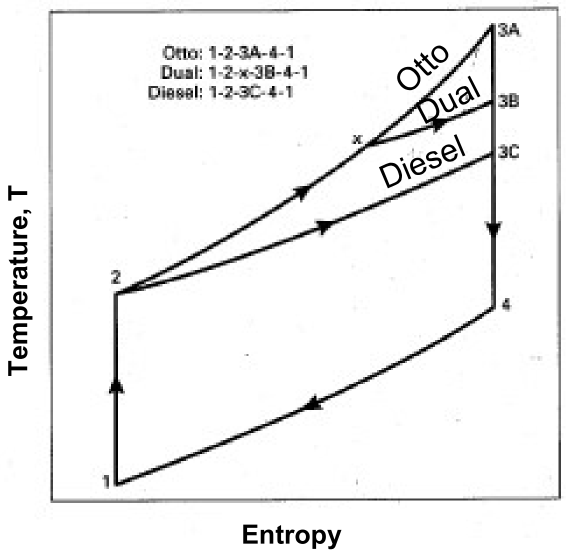
$$\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$$

For the same inlet conditions p_1, v_1 and the same peak pressure p_3 (actual design limitation in engines):

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

For the *same compression ratio* p_2/p_1:

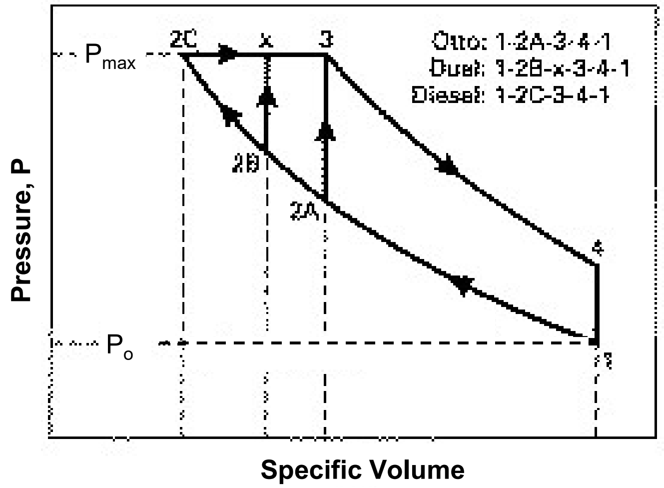


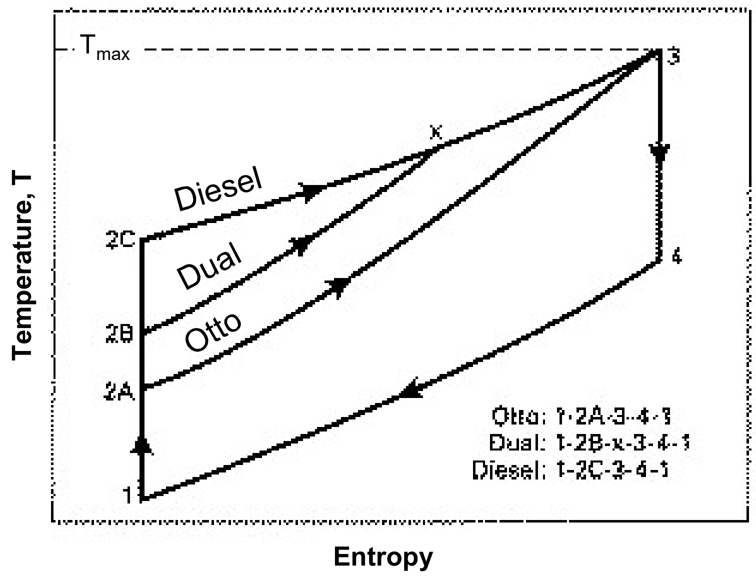


$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}}$$

$$= 1 - \frac{\int_4^1 Tds}{\int_2^3 Tds}$$

For the *same peak pressure* p_3:





Type of Fuel Vs Combustion Strategy

- Highly volatile with High self Ignition Temperature: Spark Ignition. Ignition after thorough mixing of air and fuel.
- Less Volatile with low self Ignition Temperature: Compression Ignition , Almost simultaneous mixing & Ignition.