Engine Thermodynamic Cycles

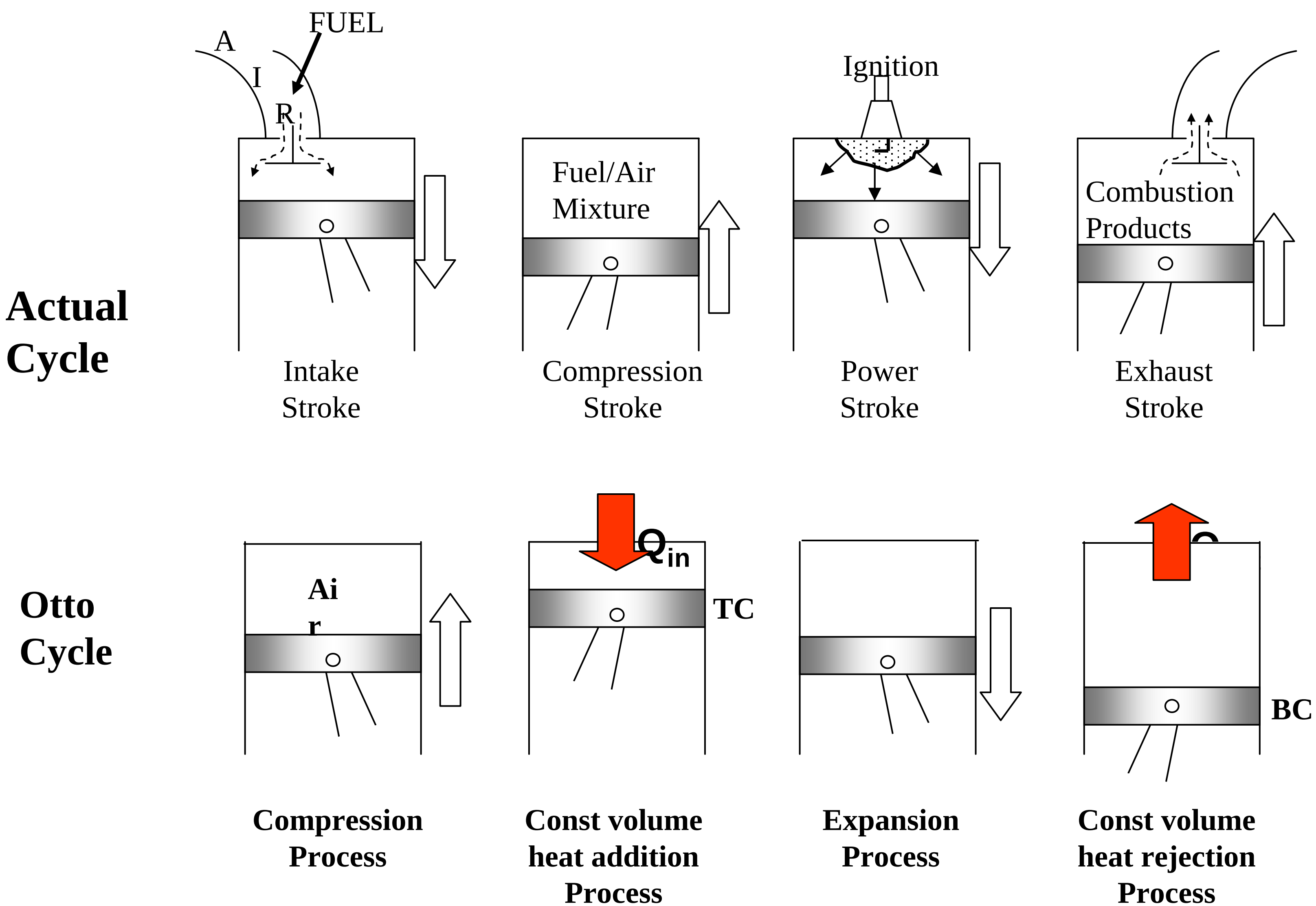
Thermodynamic Cycles

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

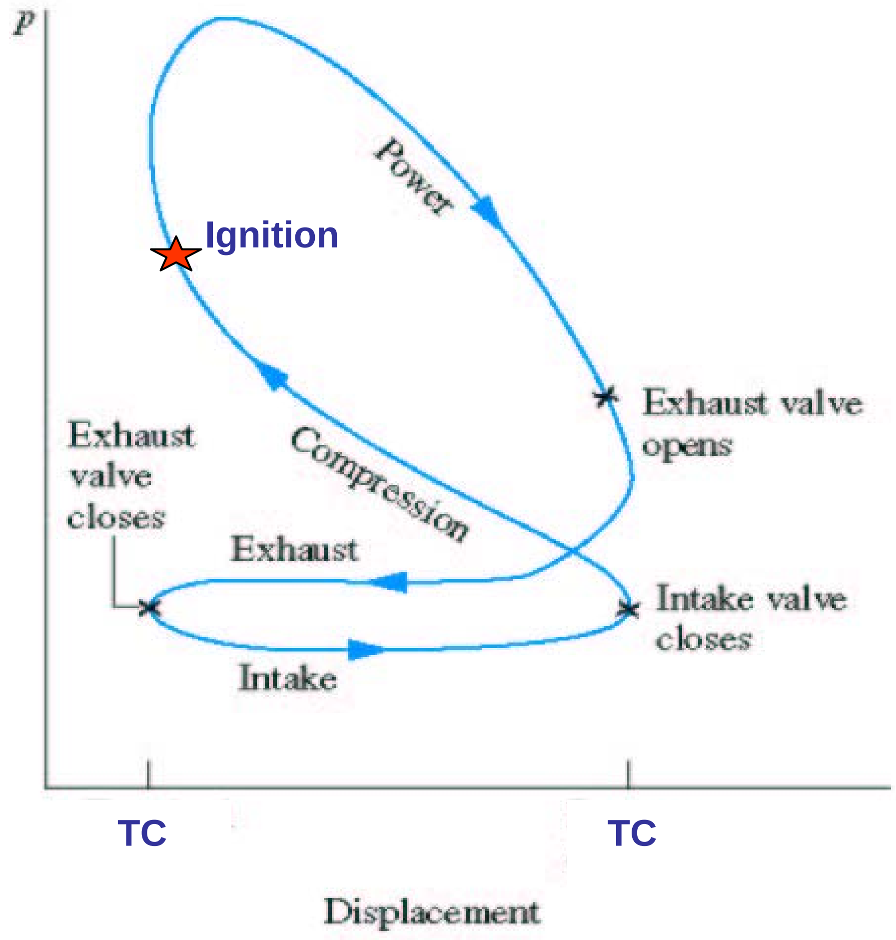
- Simplifications compared to the real cycle are:
    1) Fixed amount of air for working mixture
    2) Ideal gas assumption
    3) Combustion process itself not considered
    4) Intake and exhaust processes not considered - assumed instantaneous
    5) Engine friction and heat losses not considered

- The two types of reciprocating engine cycles:
    1) Spark ignition – Otto cycle
    2) Compression ignition – Diesel cycle

SI Engine: Thermodynamic Otto Cycle



Actual SI Engine cycle



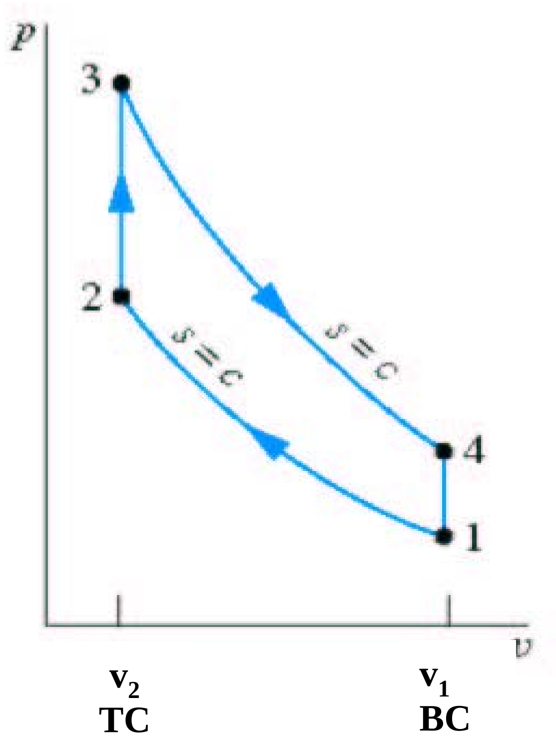
Air-Standard Otto cycle

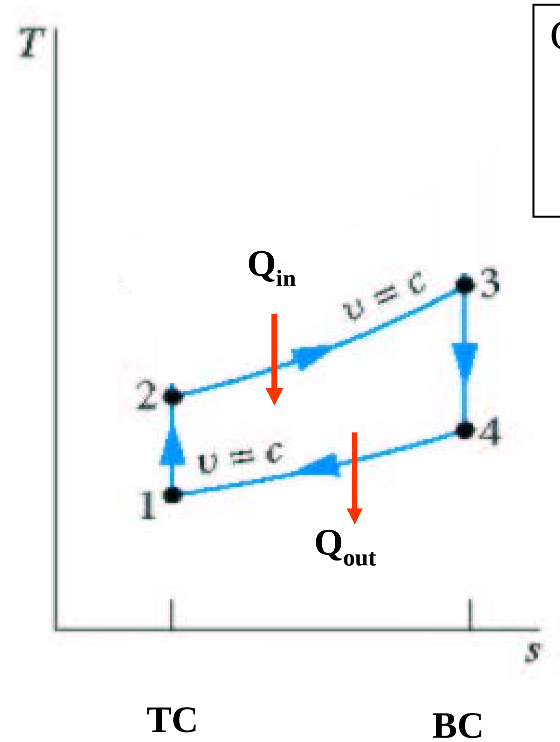
$1 \Rightarrow 2$: Isentropic compression

$2 \Rightarrow 3$: Constant volume heat addition

$3 \Rightarrow 4$: Isentropic expansion

$4 \Rightarrow 1$: Constant volume heat rejection



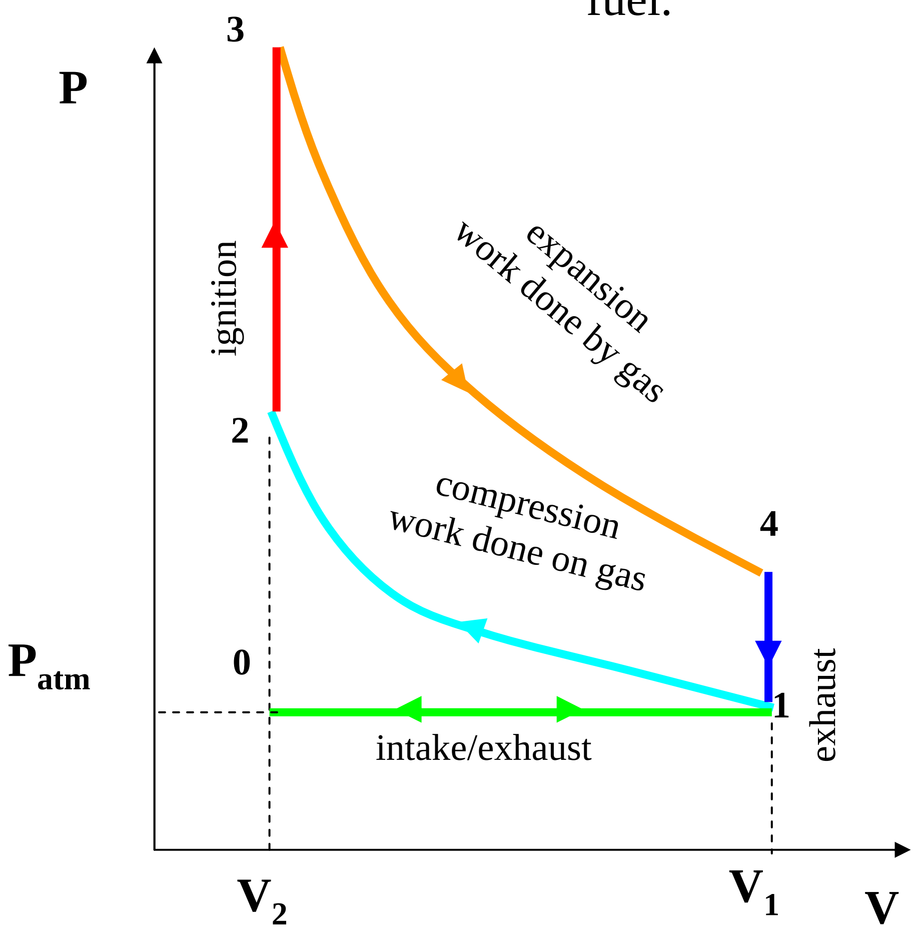


Compression ratio:

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

Internal Combustion Engines (Otto cycle)

Otto cycle. Working substance – a mixture of air and vaporized gasoline. No hot reservoir – thermal energy is produced by burning fuel.



$0 \Rightarrow 1$: intake (fuel+air is injected into the cylinder by the retreating piston)

$1 \Rightarrow 2$: isentropic compression

$3 \Rightarrow 3$: isochoric heating

$3 \Rightarrow 4$: isentropic expansion

$4 \Rightarrow 1 \Rightarrow 0$: exhaust

Internal Combustion Engines (Otto cycle)

The efficiency:

$$\eta = 1 - \left(\frac{V_1}{V_2}\right)^{\gamma - 1} = 1 - \frac{T_2}{T_1}$$

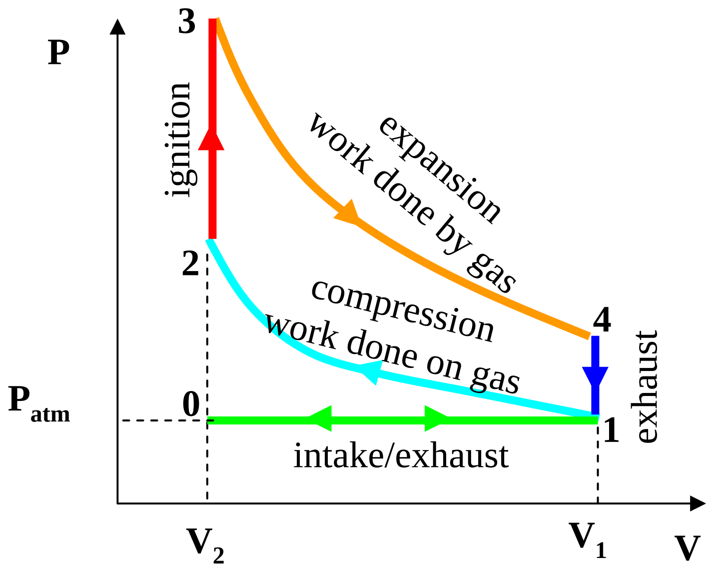
V_2 – maximum cylinder volume

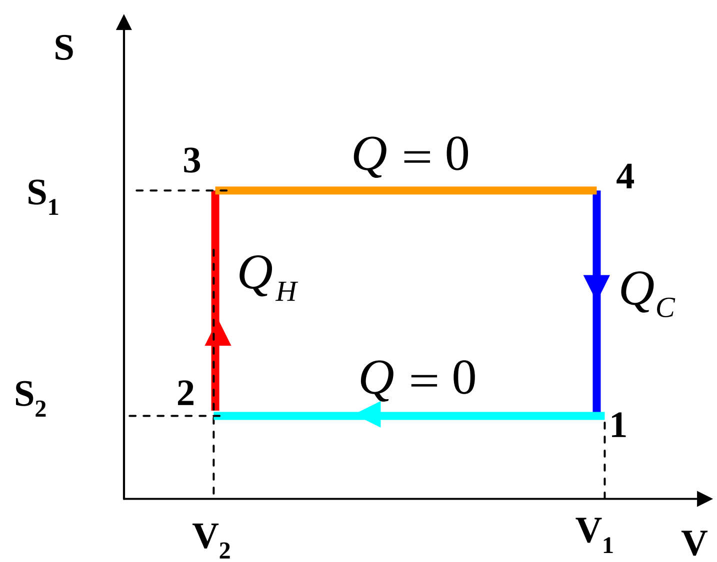
V_2 – minimum cylinder volume

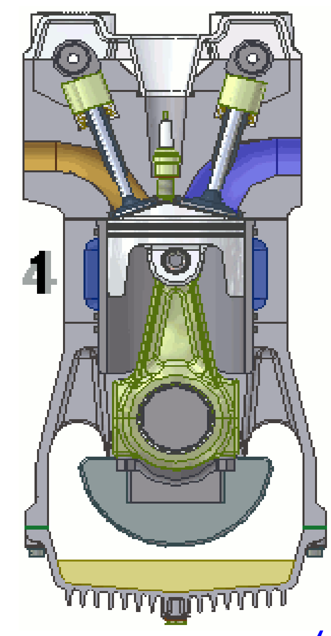
$\frac{V_2}{V_1}$ – compression ratio

$\gamma = c_P / c_V$ – adiabatic constant

Typical numbers $\mathbf{V_1/V_2}$ ~8 , γ ~ 7/5 $\rightarrow$ η = 0.56, (reality: $\eta \approx 0.2 - 0.3$) - *smaller* than the second law limit $\mathbf{1\text{-}}\boldsymbol{T_3/T_1}$)



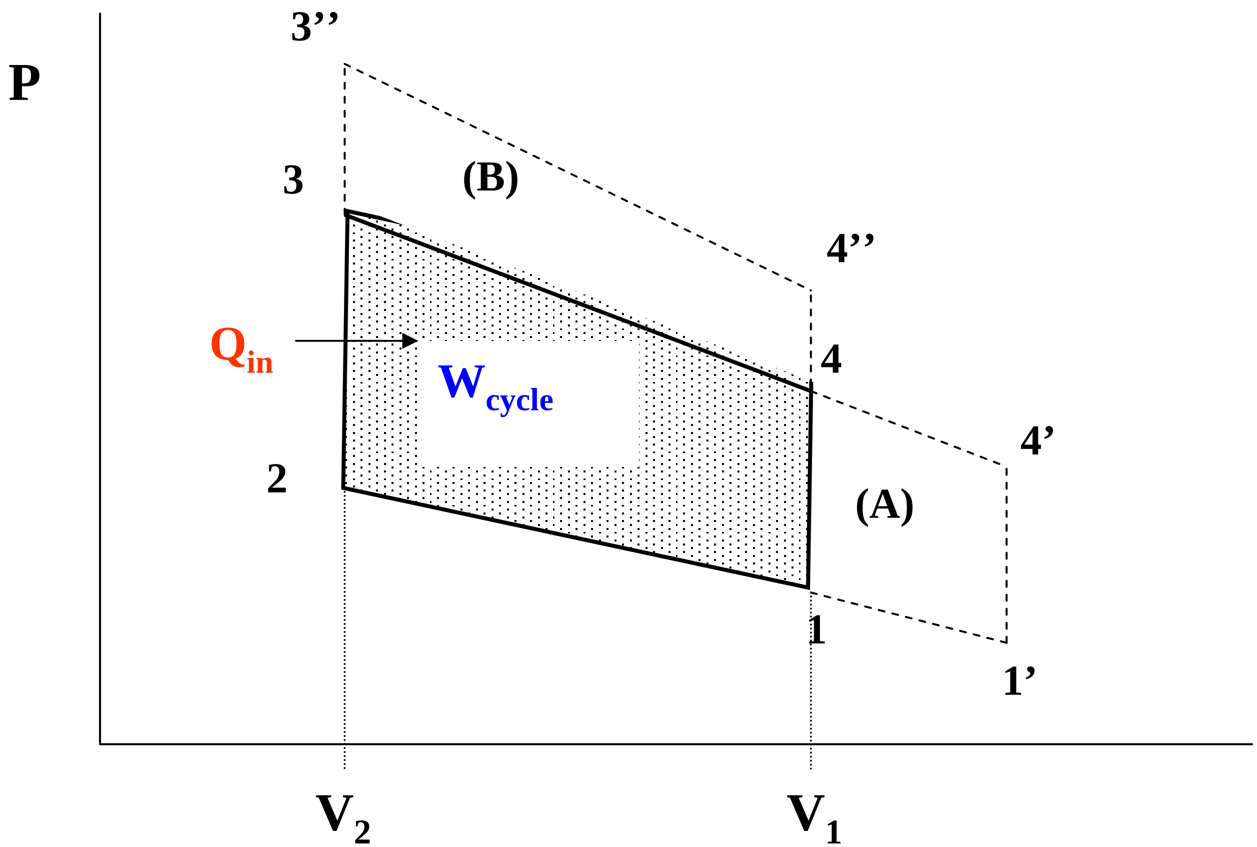




The net work output per cycle W_{cycle} can be increased by either:

A) Increasing the compression ratio.

B) Increasing Q_{in} (increase the engine bore).



For a cold air-standard analysis the specific heats are assumed to be constant evaluated at ambient temperature values ($\gamma = c_p/c_v = 1.4$).

For the two isentropic processes in the cycle, assuming ideal gas with constant specific heat using $PV^{\gamma} = \text{const.} \quad PV = RT$:

$$1 \Rightarrow 2: \qquad \frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = r_c^{\gamma-1} \qquad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

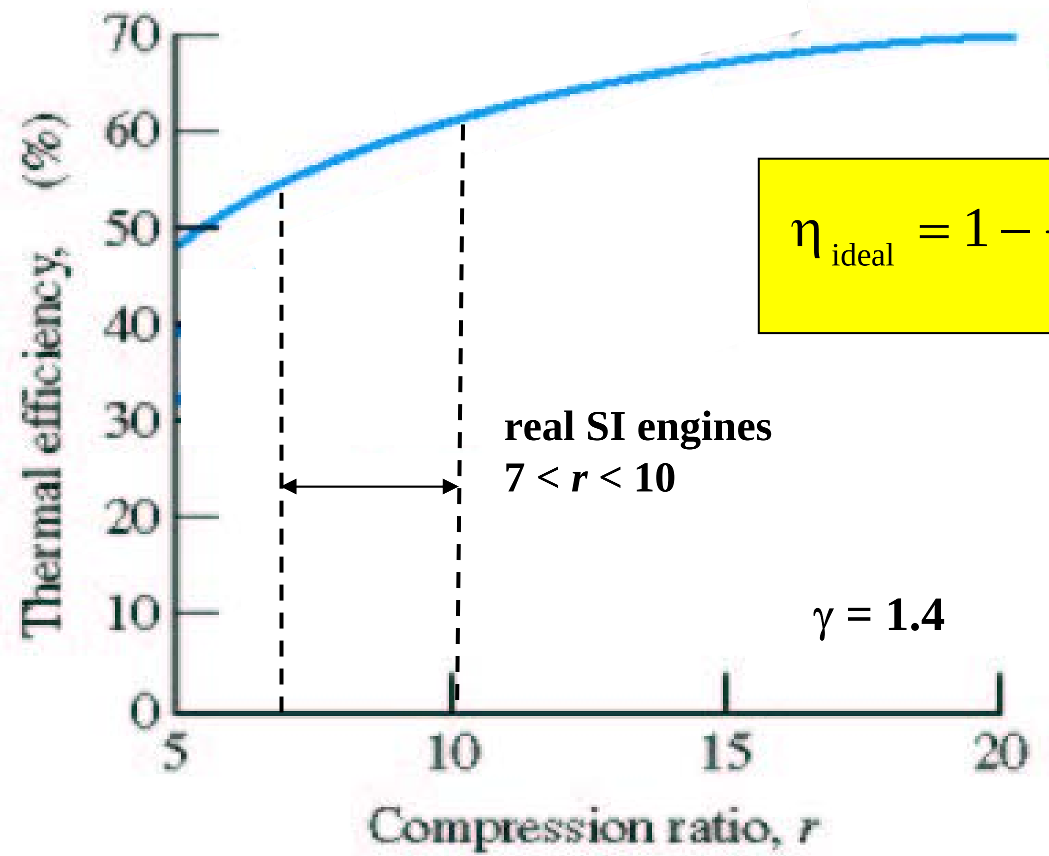
$$3 \Rightarrow 4: \qquad \frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{\gamma-1} \qquad \frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\eta = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{r^{\gamma-1}}$$

Effect of Compression Ratio on Thermal Efficiency

For SI engines compression ratio limited due to "knock"

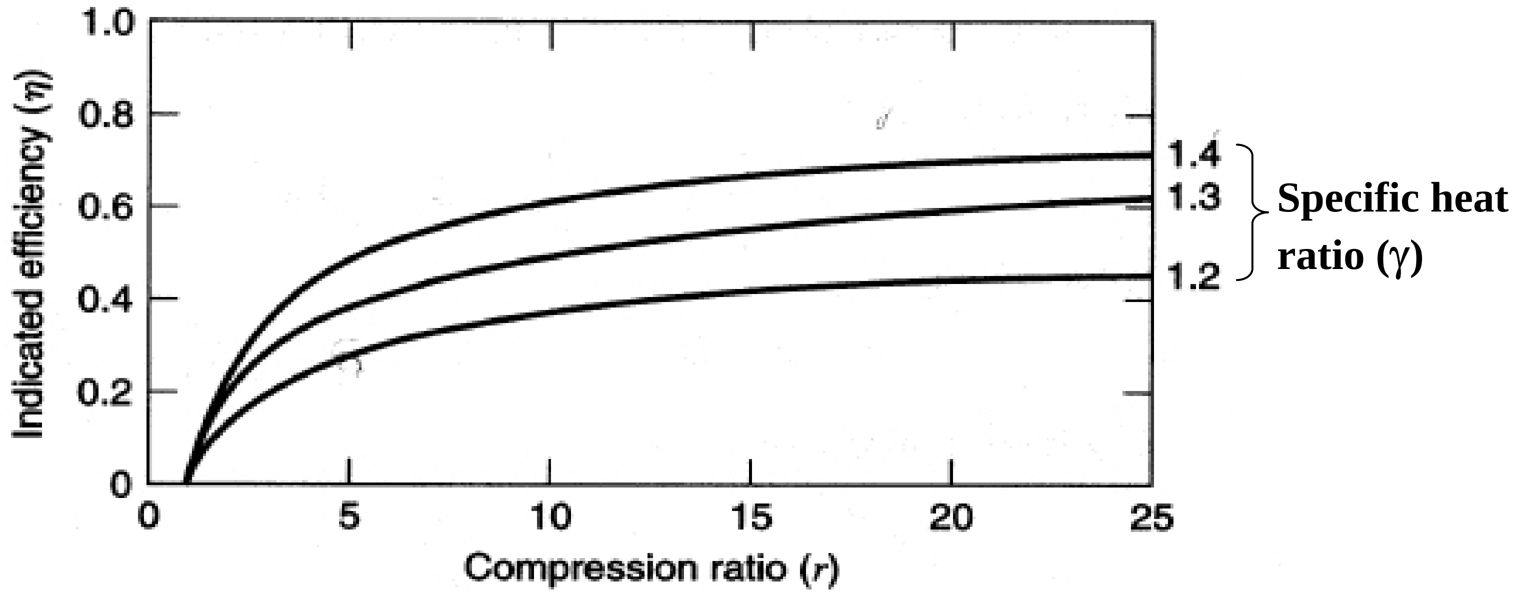
For r_c = 8 the efficiency is 56%   about twice of the actual efficiency value



$$\eta_{ideal} = 1 - \frac{1}{r^{\gamma-1}}$$

Effect of Specific Heat Ratio on Thermal Efficiency

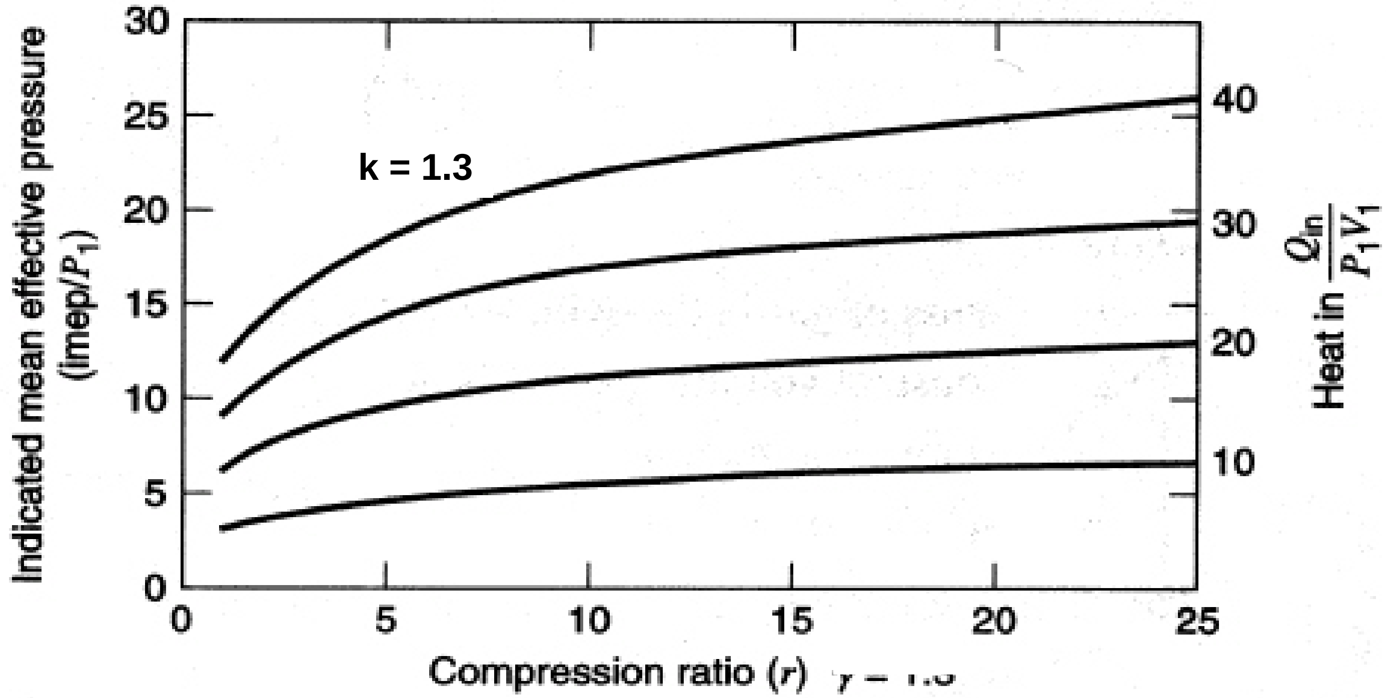
$$\eta_{ideal} = 1 - \frac{1}{r^{\gamma - 1}}$$



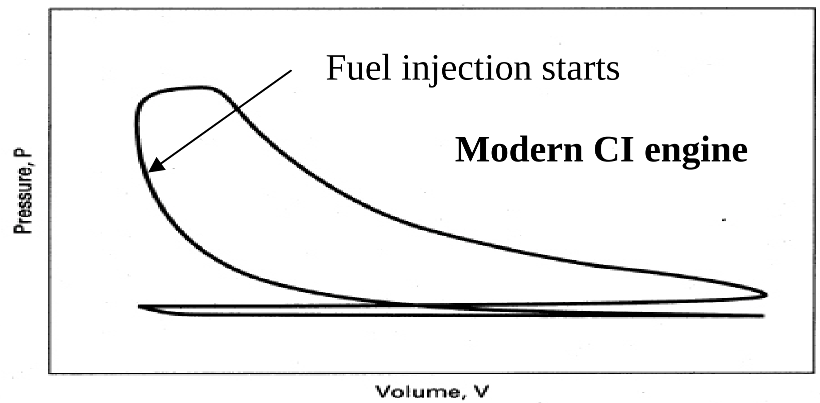
For temperatures between 20K and 2000K, $\gamma = 1.3$ is most representative

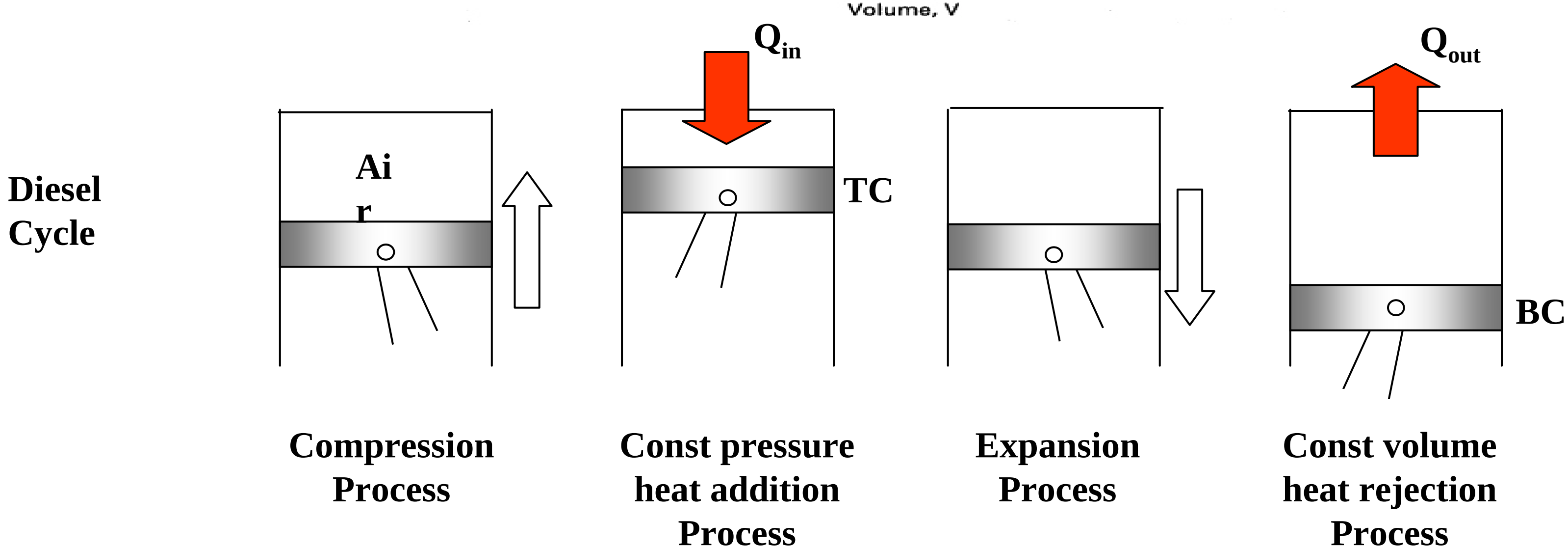
Effect of Compression Ratio on Thermal Efficiency and Indicated Mean Effective Pressure (IMEP)

$$\frac{IMEP}{P_1} = \frac{Q_{in}}{P_1 V_1}\left(\frac{\gamma}{\gamma - 1}\right)\eta_{ideal}$$



Thermodynamic Cycles for CI engines

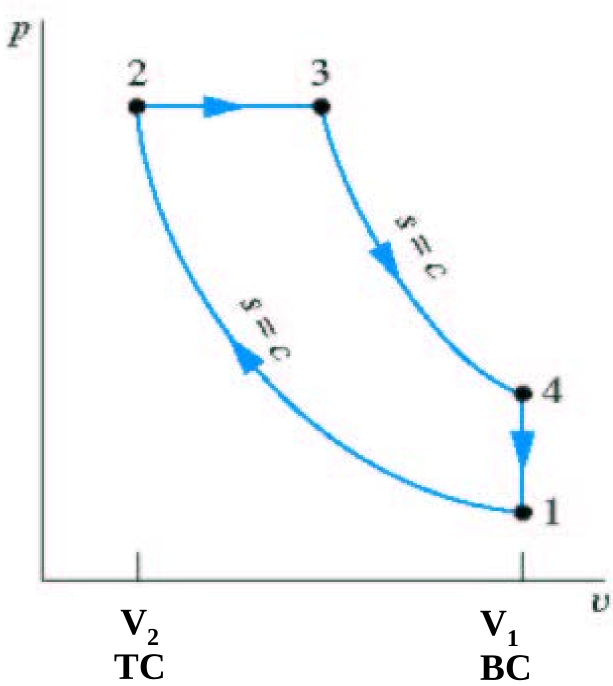


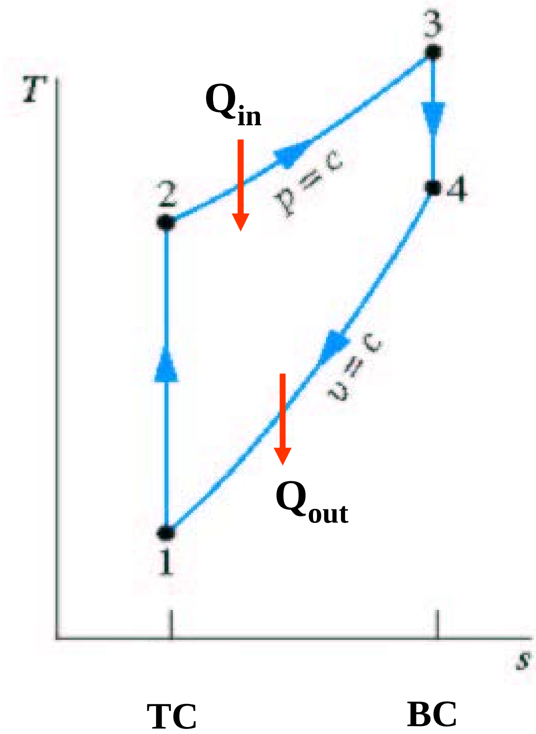


In modern CI engines the fuel is injected before TC (about 20^o). Therefore, the combustion process in the modern CI engines is best approximated by a combination of constant volume and constant pressure ⇒ **Dual Cycle**

Air-Standard Diesel cycle

1⇒ 2: Isentropic compression
2 ⇒ 3: Constant pressure heat addition
3 ⇒ 4: Isentropic expansion
4 ⇒ 1: Constant volume heat rejection





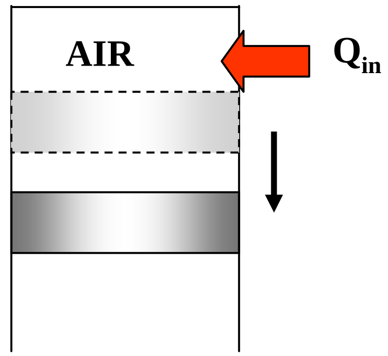
Cut-off ratio:

$$r_c = \frac{V_3}{V_2}$$

Thermodynamic Analysis of Diesel Cycle

Equations for processes 1⇒2, 4⇒1 are the same as for the Otto cycle

2 ⇒ 3: Constant Pressure Heat Addition



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

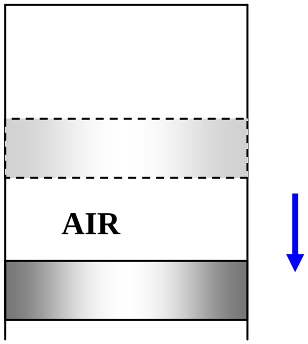
$$\frac{Q_{in}}{m} = (u_3 + P_3 v_3) - (u_2 + P_2 v_2)$$

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

$$P = \frac{RT_2}{V_2} = \frac{RT_3}{V_3} \Rightarrow \frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$$

$3 \Rightarrow 4$: Isentropic Expansion

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



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

$$\frac{V_4}{V_3} = \frac{V_4}{V_2} \cdot \frac{V_2}{V_3} = \frac{V_1}{V_2} \cdot \frac{V_2}{V_3} = \frac{r}{r_c}$$

$$V_4 = V_1 \Rightarrow \boxed{\frac{V_{r_4}}{V_{r_3}} = \frac{V_4}{V_3} = \frac{r}{r_c}}$$

$$\frac{P_4 V_4}{T_4} = \frac{P_3 V_3}{T_3} \Rightarrow \boxed{\frac{P_4}{P_3} = \frac{T_4}{T_3} \cdot \frac{r}{r_c}}$$

Thermal Efficiency

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

For cold air-standard analysis:

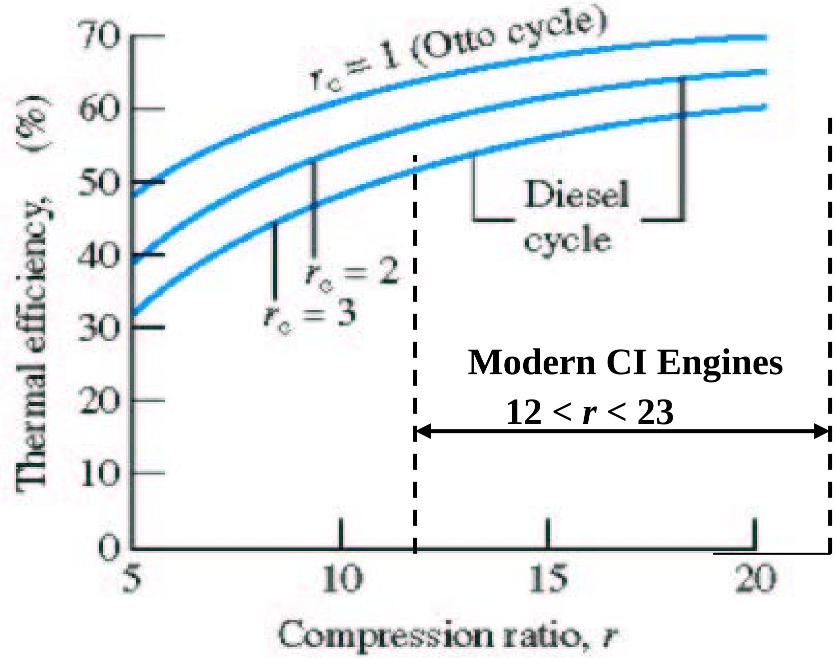
$$\eta_{\substack{\text{Diesel}\\\text{const } c_V}} = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{1}{\gamma}\cdot\frac{\left(r_c^{\gamma} - 1\right)}{\left(r_c - 1\right)}\right]$$

Compared to

$$\eta_{\substack{\text{Otto}\\\text{Cycle}}} = 1 - \frac{1}{r^{\gamma-1}}$$

- ✓ Note that for the same compression ratio, *r*, the Diesel cycle has a *lower* thermal efficiency than the Otto cycle, since the term in the square bracket is always larger than one.
- ✓ The Diesel cycle efficiency approaches the efficiency of the Otto cycle for $r_c = V_3/V_2 \Rightarrow 1$.

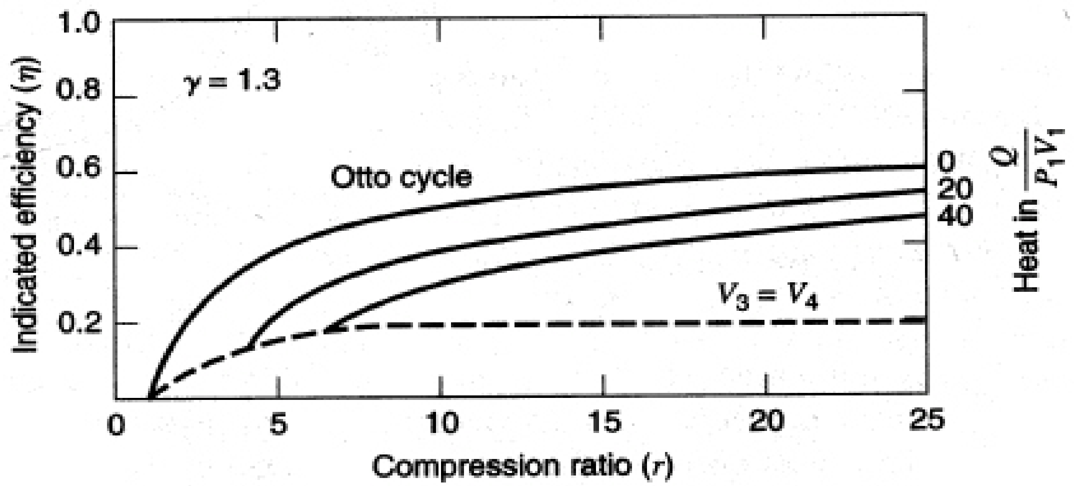
Thermal Efficiency of Diesel Engines

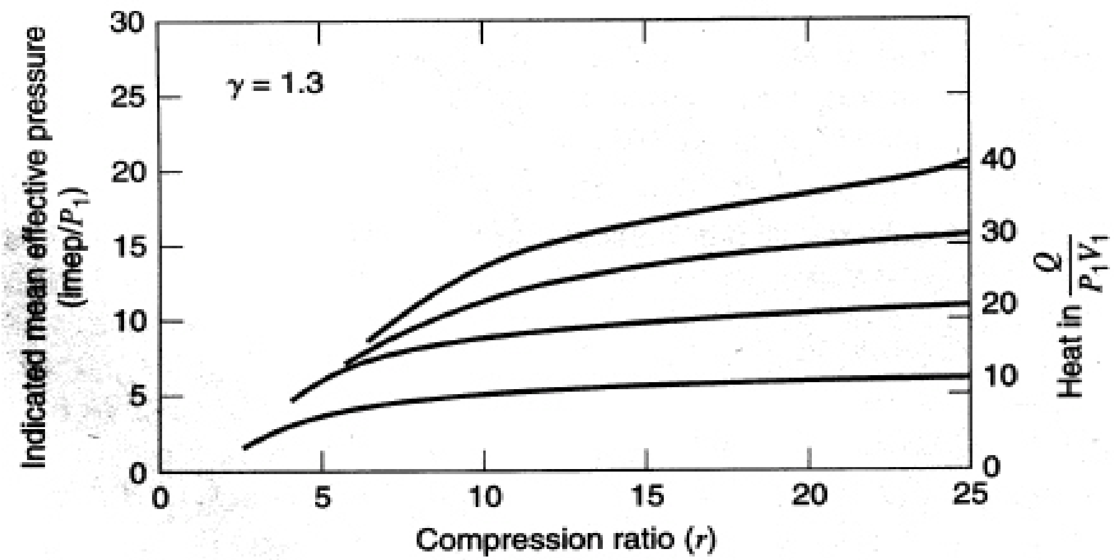


The cut-off ratio is not a natural choice for the independent variable - a more suitable parameter is the heat input. The two are related by:

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

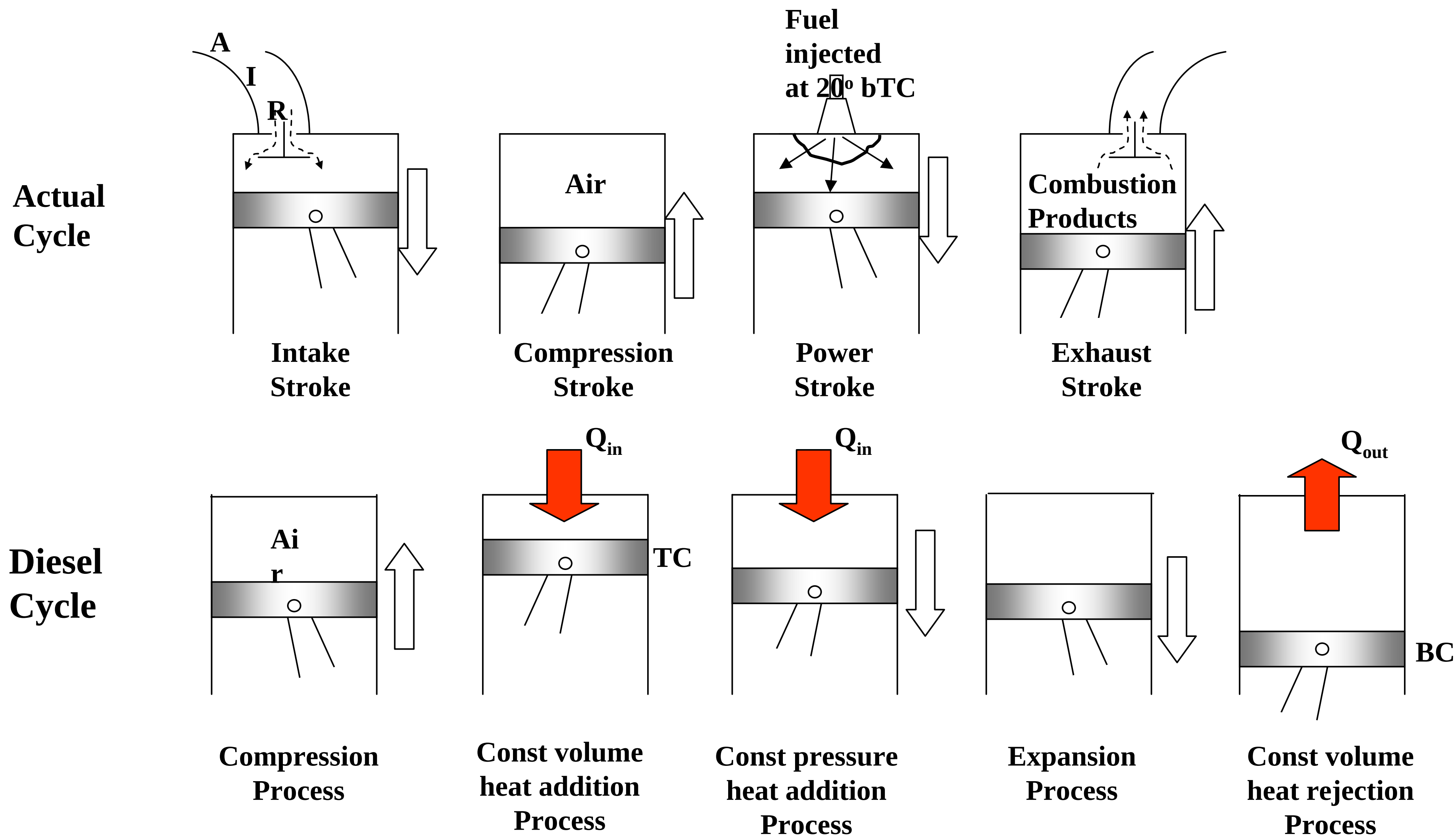
as $Q_{in} \Rightarrow 0$, $r_c \Rightarrow 1$ and $h \Rightarrow h_{Otto}$





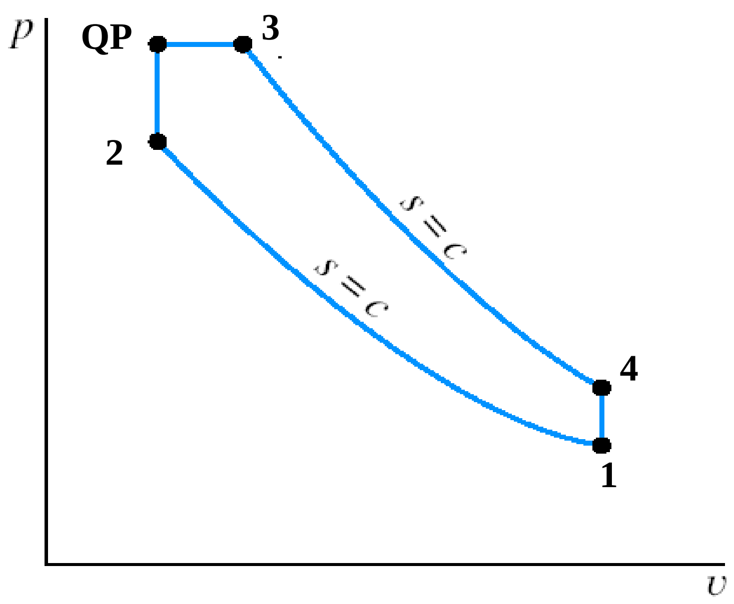
Higher efficiency is obtained by adding less heat per cycle, Q_{in}, need to run engine at higher speed to get the same power.

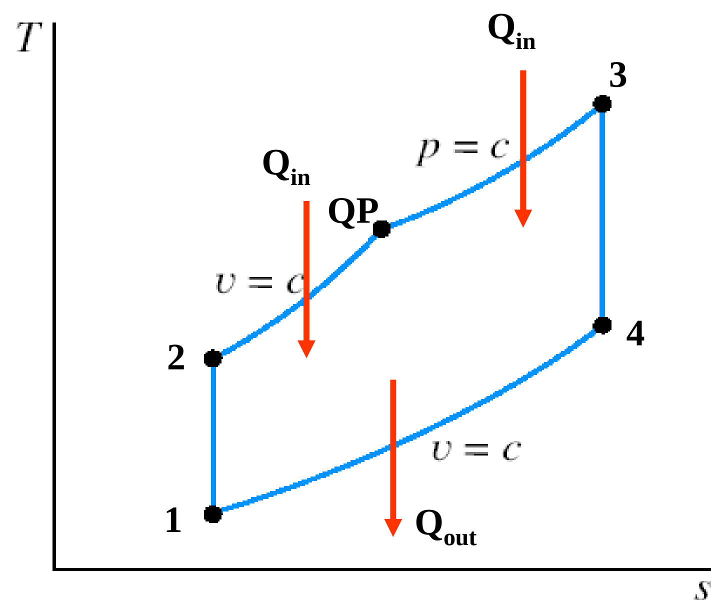
CI Engine Cycle and the Thermodynamic Dual Cycle



Dual Cycle

1	⇒ 2	Isentropic compression
2	⇒ QP	Constant volume heat addition
QP	⇒ 3	Constant pressure heat addition
3	⇒ 4	Isentropic expansion
4	⇒ 1	Constant volume heat rejection





Dual Cycle - Thermal Efficiency

$$\eta_{\substack{\text{Dual}\\ \text{cycle}}} = 1 - \frac{Q_{out}/m}{Q_{in}/m} = 1 - \frac{u_4 - u_1}{(u_X - u_2) + (h_3 - h_{QP})}$$

For cold air-standard the above reduces to:

$$\eta_{\substack{\text{Diesel}\\ \text{const } c_V}} = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\alpha r_c^{\gamma} - 1}{(\alpha - 1) + \alpha\gamma\left(r_c - 1\right)}\right]$$

where $r_c = v_3/v_{QP}$ and $a = P_3/P_2$

The Otto cycle ($r_c = 1$) and the Diesel cycle (**α=1**) are special cases:

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{\gamma-1}}$$

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

The use of the Dual cycle requires information about either the fractions of constant volume and constant pressure heat addition (common assumption is to equally split the heat addition), or the maximum pressure P_3.

More natural variables

$$r_c = 1 - \frac{\gamma - 1}{\alpha\gamma}\left[\left(\frac{Q_{in}}{P_1 V_1}\right)\frac{1}{r^{\gamma-1}} - \frac{\alpha - 1}{\gamma - 1}\right] \qquad \alpha = \frac{1}{r^{\gamma}}\frac{P_3}{P_1}$$

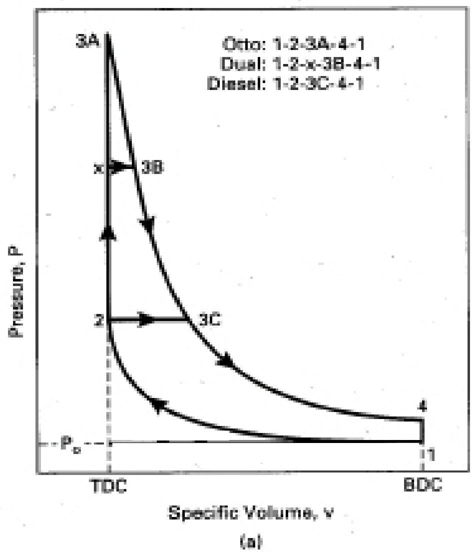
For the same inlet conditions (P_1, V_1) and the same compression ratio:

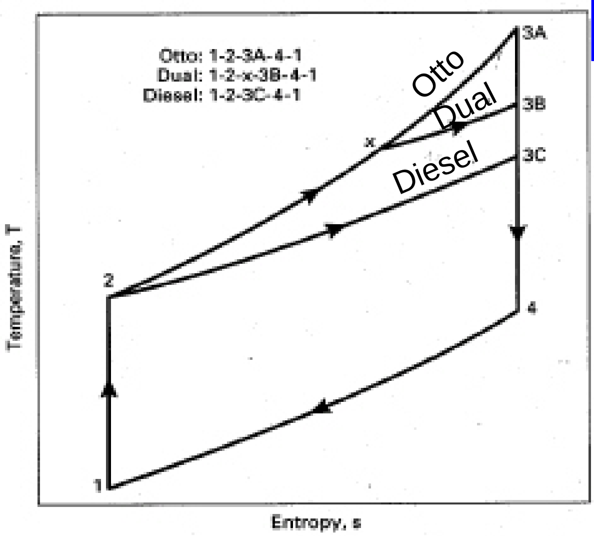
$$\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$$

For the same inlet conditions (P_1, V_1) and the same peak pressure (design limitation in engines):

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

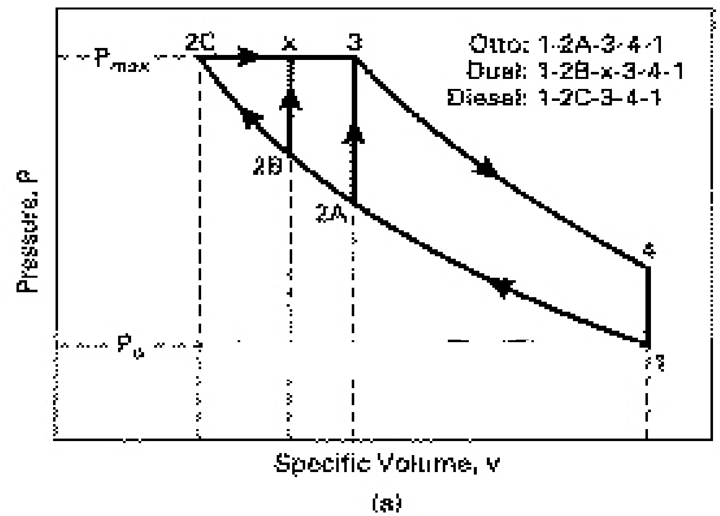
The same inlet conditions (P_1,V_1)
The same compression ratio P_2/P_1:

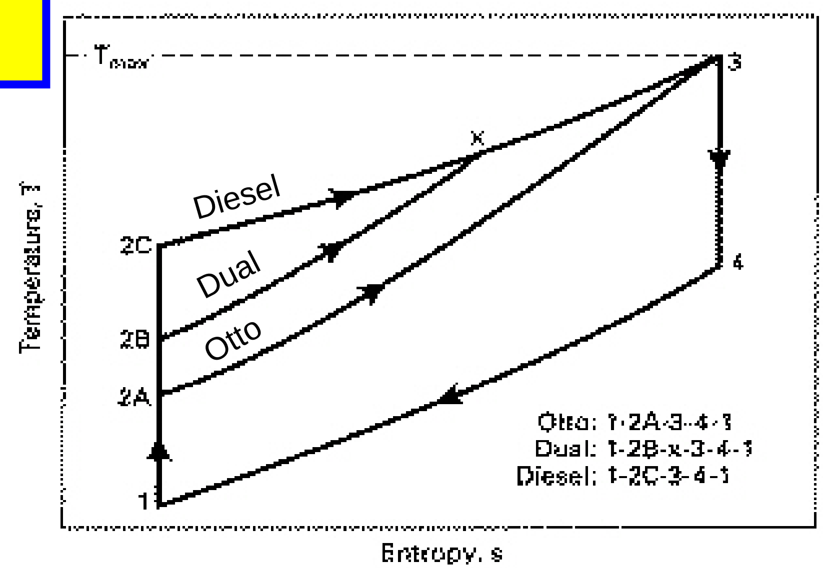




$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{\int_4^1 Tds}{\int_2^3 Tds}$$

For the same inlet conditions (P_1,V_1)
The same peak pressure P_3:





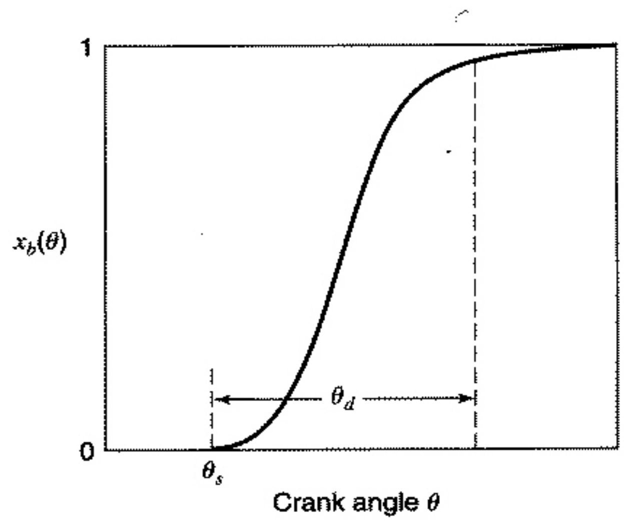
Finite Heat Release Model

In the Otto cycle it is assumed that the heat is release instantaneously.
A finite heat release model specifies heat release as a function of crank angle.
This model can be used determine the effect of spark timing or heat transfer on engine work and efficiency.

The cumulative heat release or “burn fraction” for SI engines is given by:

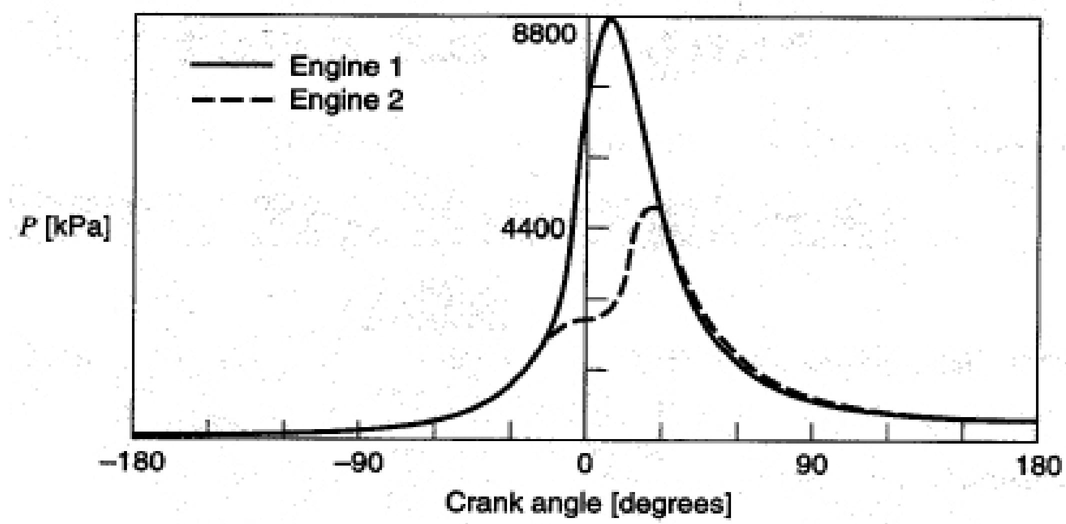
Finite Heat Release

It is assumed that in the Otto cycle the heat is release instantaneously. A finite heat release model specifies heat release as a function of crank angle.
A typical heat release curve consists of an initial spark ignition phase, followed by a rapid burning phase and ends with burning completion phase



The curve asymptotically approaches 1 so the end of combustion is defined by an arbitrary limit, such as 90% or 99% complete combustion where x_b = 0.90 or 0.99 corresponding values for efficiency factor α is 2.3 and 4.6

Finite Heat Release Model - Results



Start of heat release:
Engine 1 - 20º TC
Engine 2 - TC

Duration 40º