III. Engine Cycles

III.A – Air-Standard Cycles

III.B – Air-Fuel Cycles

III.C – Actual IC Cycles

Internal Combustion Engines

University of Al-Basrah
College of Engineering
Mechanical Engineering Department
3rd Stage

Perfect Gas: Thermodynamic Relations

$$Pv = RT$$

$$PV = mRT$$

$$P = \rho RT$$

$$dh = c_P dT$$

$$du = c_v dT$$

$c_p = $ 1.108 kJ/kg-K = 0.265 BTU/lbm-°R

$c_v = $ 0.821 kJ/kg-K = 0.196 BTU/lbm-°R

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

$R = c_P - c_v = $ 0.287 kJ/kg-K

$=$ 0.069 BTU/lbm-°R = 53.33 ft-lbf/lbm-°R

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

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

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

$w_{1-2} = (P_2v_2 - P_1v_1)/(1-k)$ isentropic work in closed system

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

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

The Air-Cycle Approximations

- Due to the importance of thermal efficiency, it is necessary to know how it is affected by engine design, adjustment and operating conditions, by making certain simplified assumptions called air-cycle approximations.
- The accurate analysis of IC engine processes is very complicated. In order to understand them it is advantageous to analyze the performance of an idealized closed cycle that closely approximates the real cycle.
- One such approach is the air-standard cycle.

Simplifications to the Real Cycle

1) Fixed amount of air (ideal gas) for working fluid.
2) Combustion process is not considered.
3) Intake and exhaust processes are not considered.
4) Engine friction and heat losses are not considered.

Air-Standard Cycles Assumptions

- Air-standard cycles are based on the following assumptions:

1. The working substance is assumed to be a perfect gas and follows the relation PV = mRT or P = ρ RT .
2. The working substance is taken as the air of constant specific heats. (C_p, C_v and M are taken at standard atmospheric conditions:

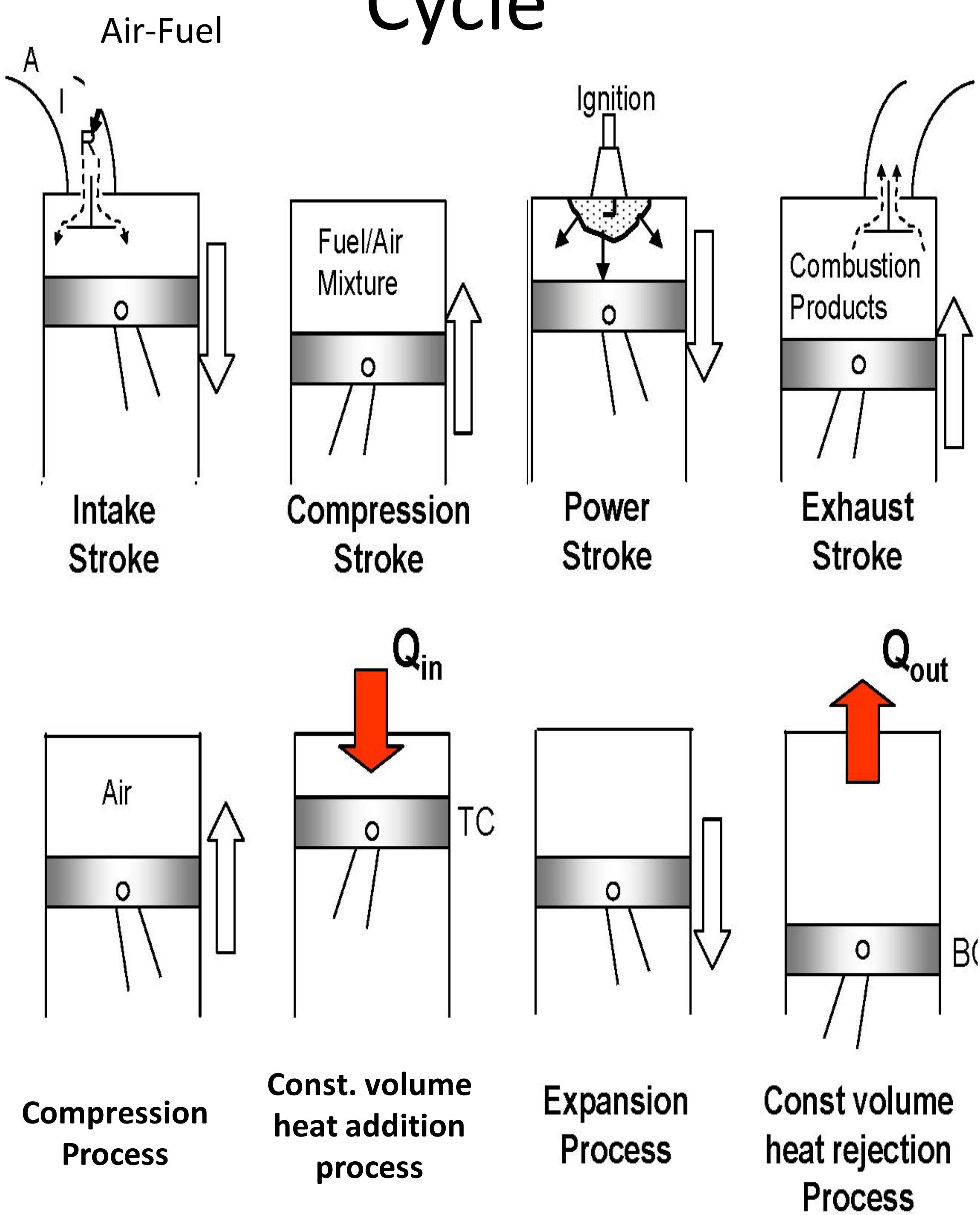
C_p = 1.005 kJ/kgK, C_v = 0.717 kJ/kgK,   M = 29 kg/kmol

and   k = 1.4

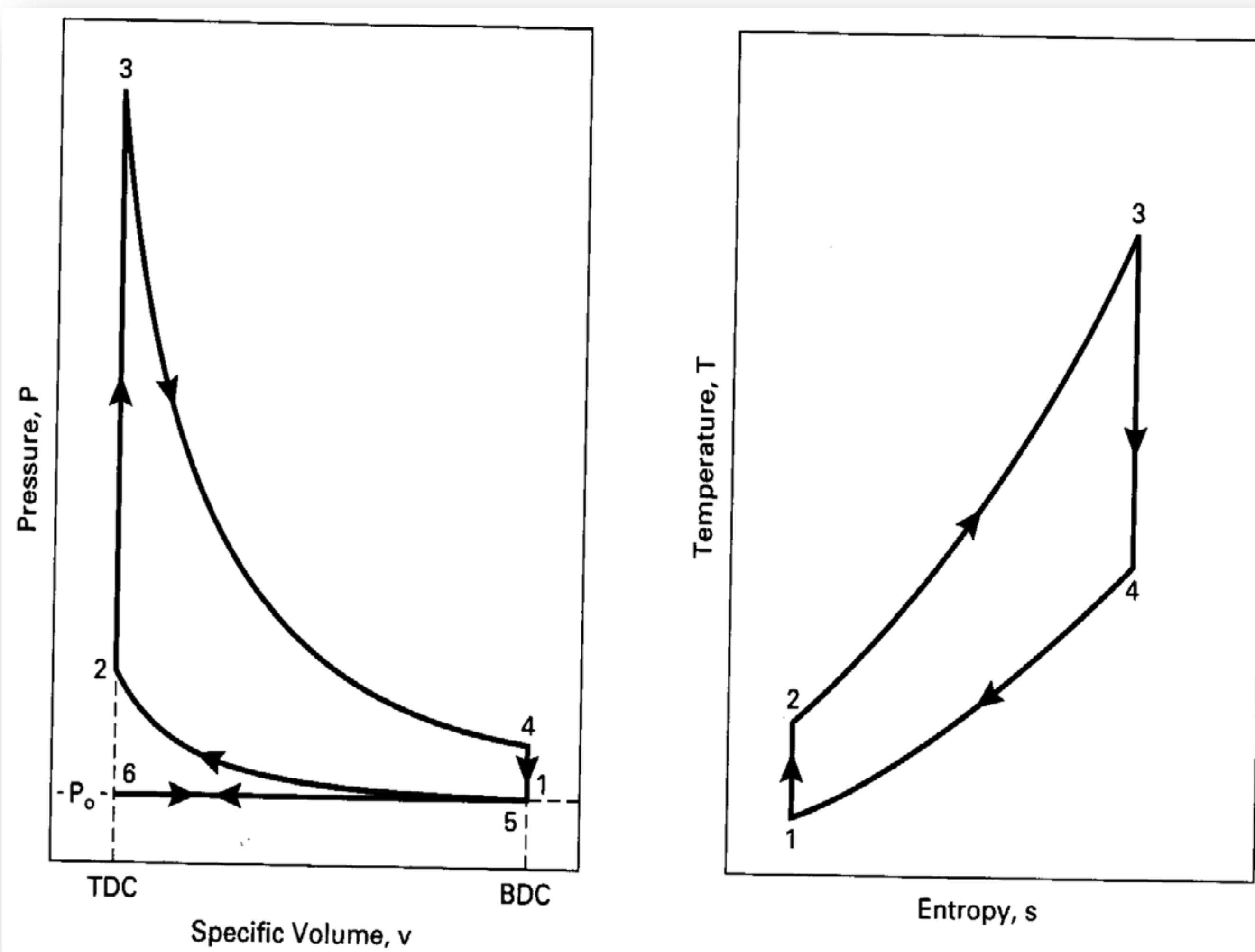
Air-Standard Cycles Assumptions

3. There is no change in the mass of the working substance.
4. All the processes that constitute the cycle are reversible
5. Heat assumed to be added from a constant high temperature source and not from chemical reactions during the cycle.
6. Some heat is assumed to be rejected to a constant low temperature sink during the cycle.
7. No heat losses from the system to the surroundings.

SI Engine Cycle vs. Thermodynamic Ideal Otto Cycle



Thermodynamic Analysis of Air-Standard Otto-Cycle



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

$$P_1 = P_6 = P_o$$

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

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

$$T_2 = T_1(v_1/v_2)^{k-1} = T_1(V_1/V_2)^{k-1} = T_1(r_c)^{k-1} \quad \text{.....(A)}$$

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

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

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1-k) = R(T_2 - T_1)/(1-k)$$
$$= (u_1 - u_2) = c_v(T_1 - T_2)$$

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

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

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

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

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

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

$$T_3 = T_{\max}$$

$$P_3 = P_{\max}$$

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

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

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

$$P_4 = P_3(v_3/v_4)^k = P_3(V_3/V_4)^k = P_3(1/r_c)^k$$

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

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

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

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

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

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

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

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

$$P_5 = P_6 = P_o$$

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

Thermal efficiency of Otto cycle:

$$(\eta_t)_{\text{OTTO}} = |w_{\text{net}}|/|q_{\text{in}}| = 1 - (|q_{\text{out}}|/|q_{\text{in}}|) \quad \text{............(B)}$$

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

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

Only cycle temperatures need to be known to determine thermal efficiency. This can be simplified further by applying ideal gas relationships for the isentropic compression and expansion strokes and recognizing that $v_1 = v_4$ and $v_2 = v_3$:

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

Rearranging the temperature terms gives:

$$T_4/T_1 = T_3/T_2 \qquad \text{............(C)}$$

Equation **(B)** can be rearranged to:

$$(\eta_t)_{\text{OTTO}} = 1 - (T_1/T_2)\{[(T_4/T_1) - 1]/[(T_3/T_2) - 1]\}$$

Using Eq. **(C)**) gives:

$$(\eta_t)_{\text{OTTO}} = 1 - (T_1/T_2)$$

Combining this with Eq. **(A)**

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

With $v_1/v_2 = r_c$, the compression ratio:

$$(\eta_t)_{\text{OTTO}} = 1 - (1/r_c)^{k-1}$$

Indicated Mean Effective Pressure (Pi)

- Work output = (W_{out})

W_{out} = Pi . Vs = η_t . Q_{in} = η_t . Q_{23}

- Let Qs = heat supplied per unit volume, Qs = ($-\Delta h_r$) ρ_s
- Where ρ_s (kg/m^3) is density at point of heat supplied, ($-\Delta h_r$) is enthalpy of reaction (J/kg) ,Thus :

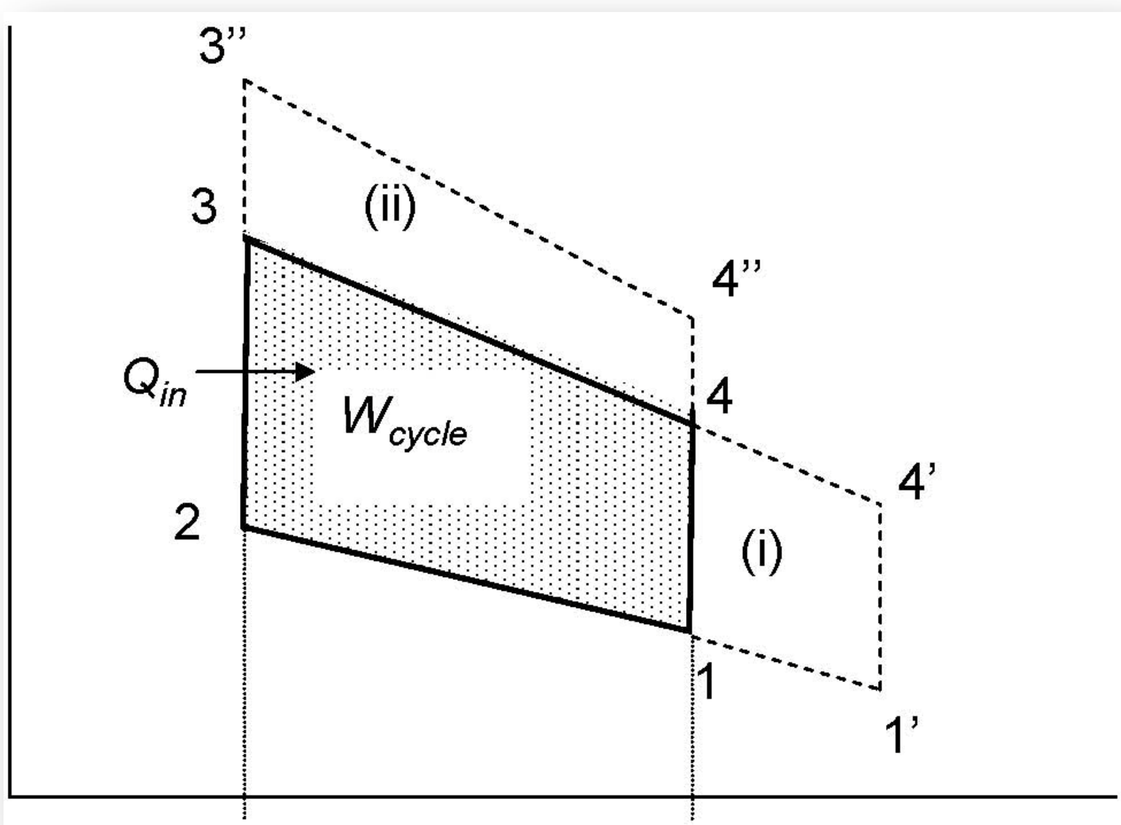
$$P_i = \frac{Q_{23}}{V_1 - V_2}\left(1 - \frac{1}{r_c^{\,k-1}}\right) = Q_s\left(1 - \frac{1}{r_c^{\,k-1}}\right) = (-\Delta h_r)\rho_s\left(1 - \frac{1}{r_c^{\,k-1}}\right)$$

- Note:

1) Pi depends on Qs
2) Pi is independent of T_3 and T_1
3) = η_t is independent of Qs, T_3 and T_1, and depends on r_c and k only.

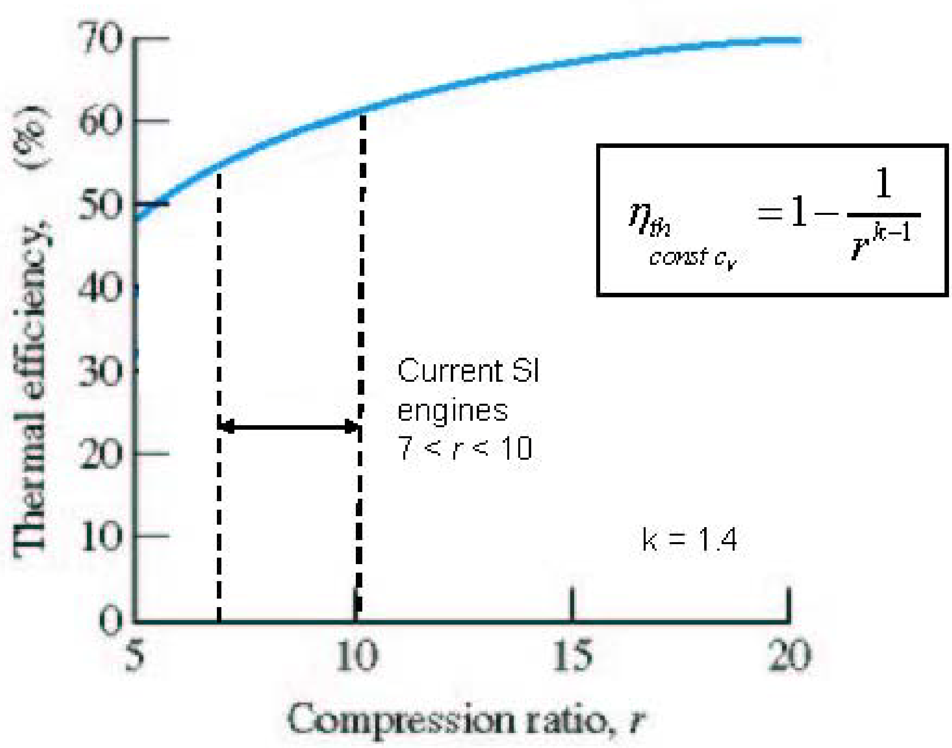
Cycle Work

- The net work output per cycle *W_{cycle} can be increased by either:*
- *i) Increasing the compression ratio, or*
- *ii) Increase Q_{in} (increase the engine bore).*



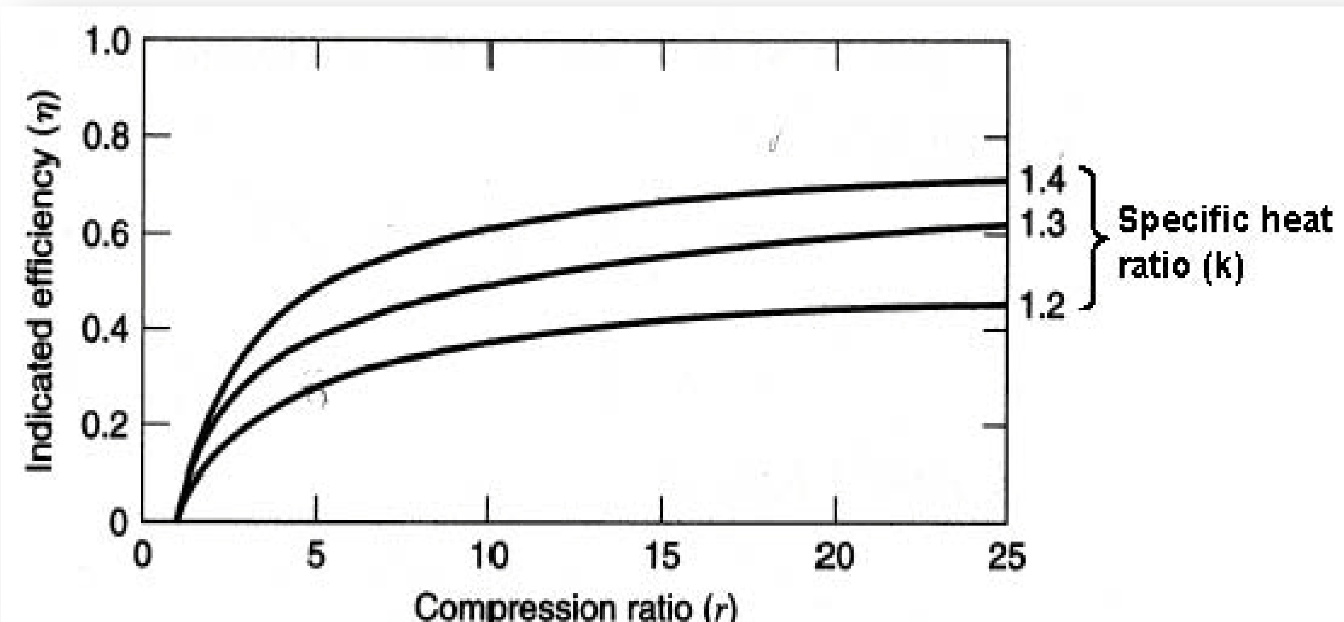
Effect of Compression Ratio on Thermal Efficiency

- Spark ignition engine compression ratio limited due to "knock"
- For r = 8 the efficiency is 56% which is about twice the actual value



Effect of Specific Heat Ratio (k) on Thermal Efficiency

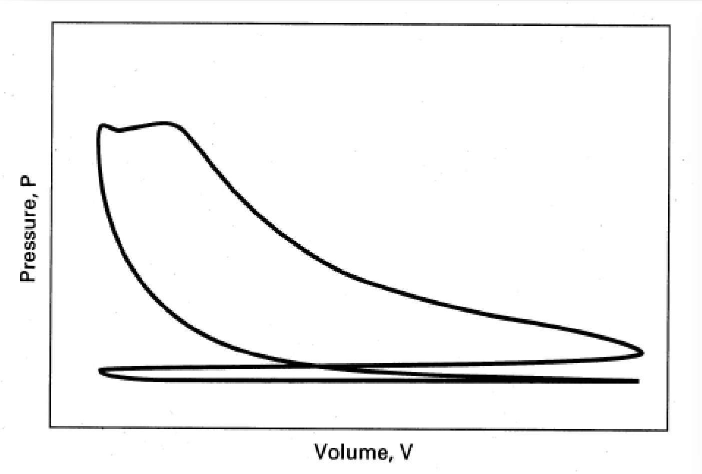
- $\eta_t = 1 - (1 / r_c^{\,k-1})$

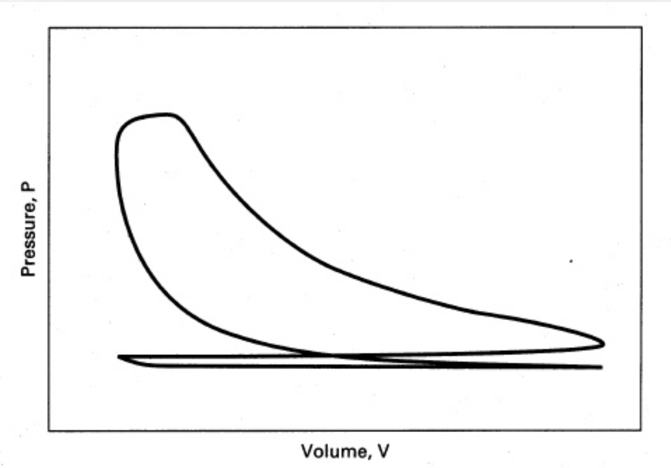


- Cylinder temperatures vary between 20K and 2000K so 1.2 < k < 1.4
- k = 1.3 most representative

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 (about 20^o bTDC).**

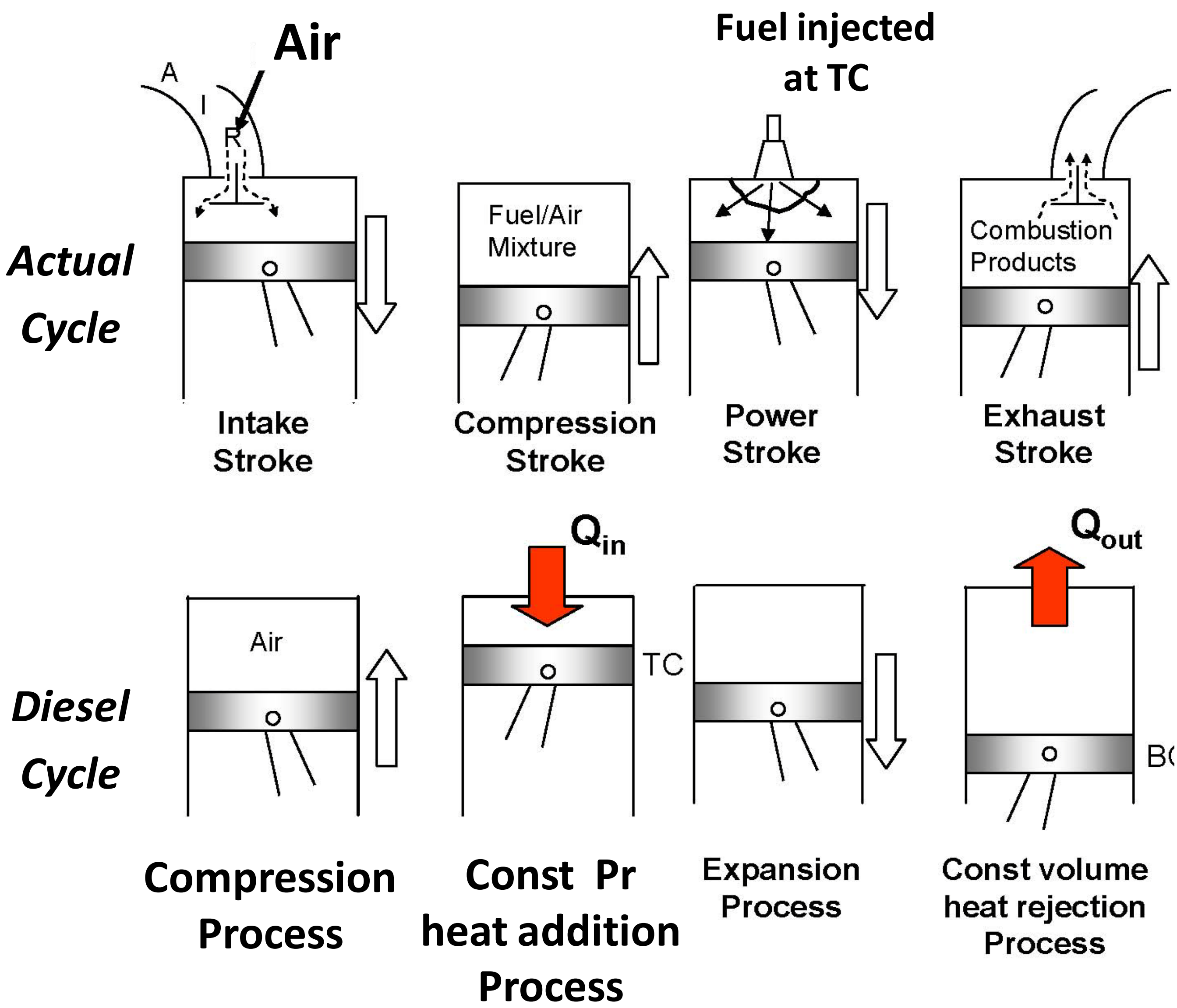




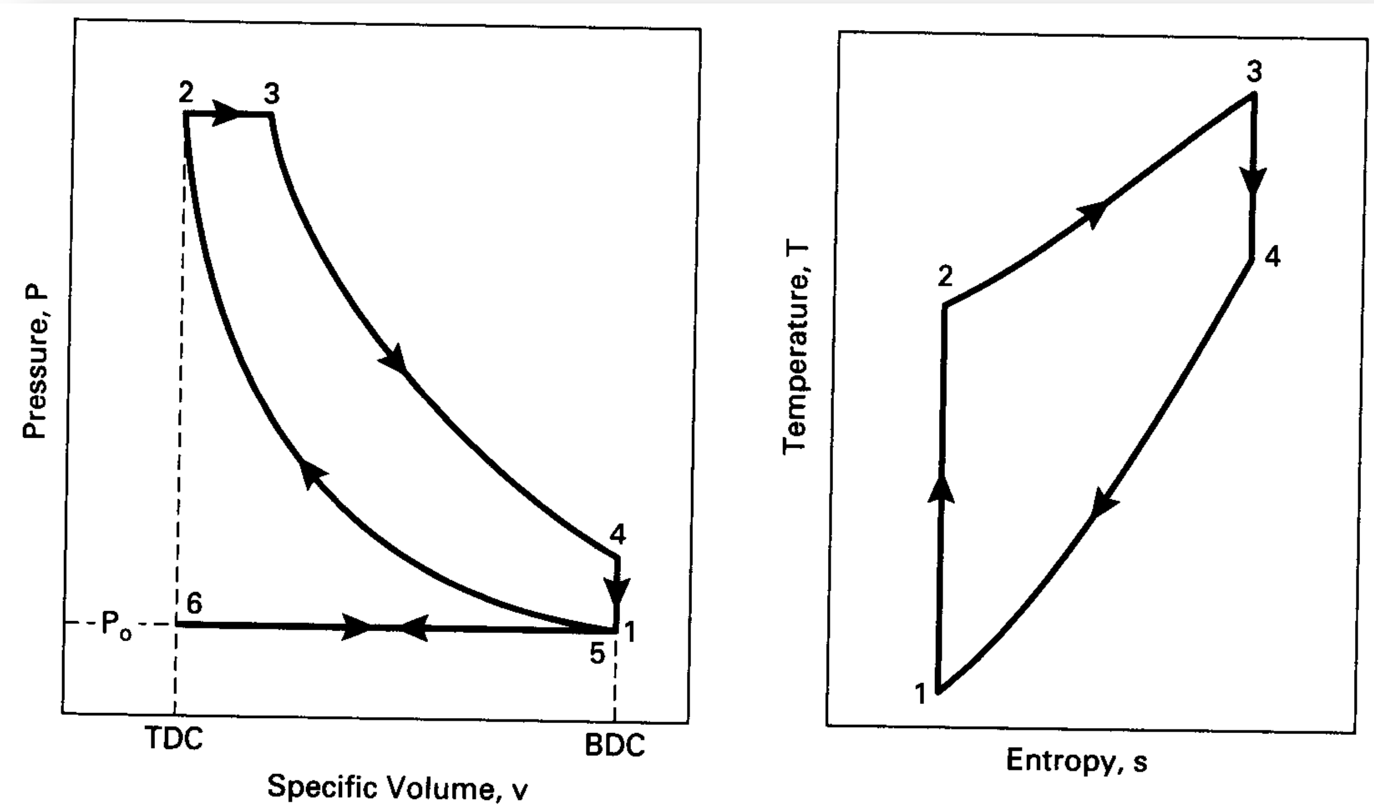
❑ The combustion process in the early CI engines is best approximated by a (P=C) heat addition process **Diesel Cycle**

❑ The combustion process in the modern CI engines is best approximated by a combination of (v=C) and (P=C) **Dual Cycle Fuel**

Early CI Engine Cycle and the Thermodynamic Diesel Cycle



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:

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

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

$$V_2 = V_{\text{TDC}}$$

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

$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1 - k) = R(T_2 - T_1)/(1 - k)$$
$$= (u_1 - u_2) = c_v(T_1 - T_2)$$

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 (β) is defined as the change in volume that occurs during combustion:

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

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

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

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

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

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

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

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

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

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

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

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

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:

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

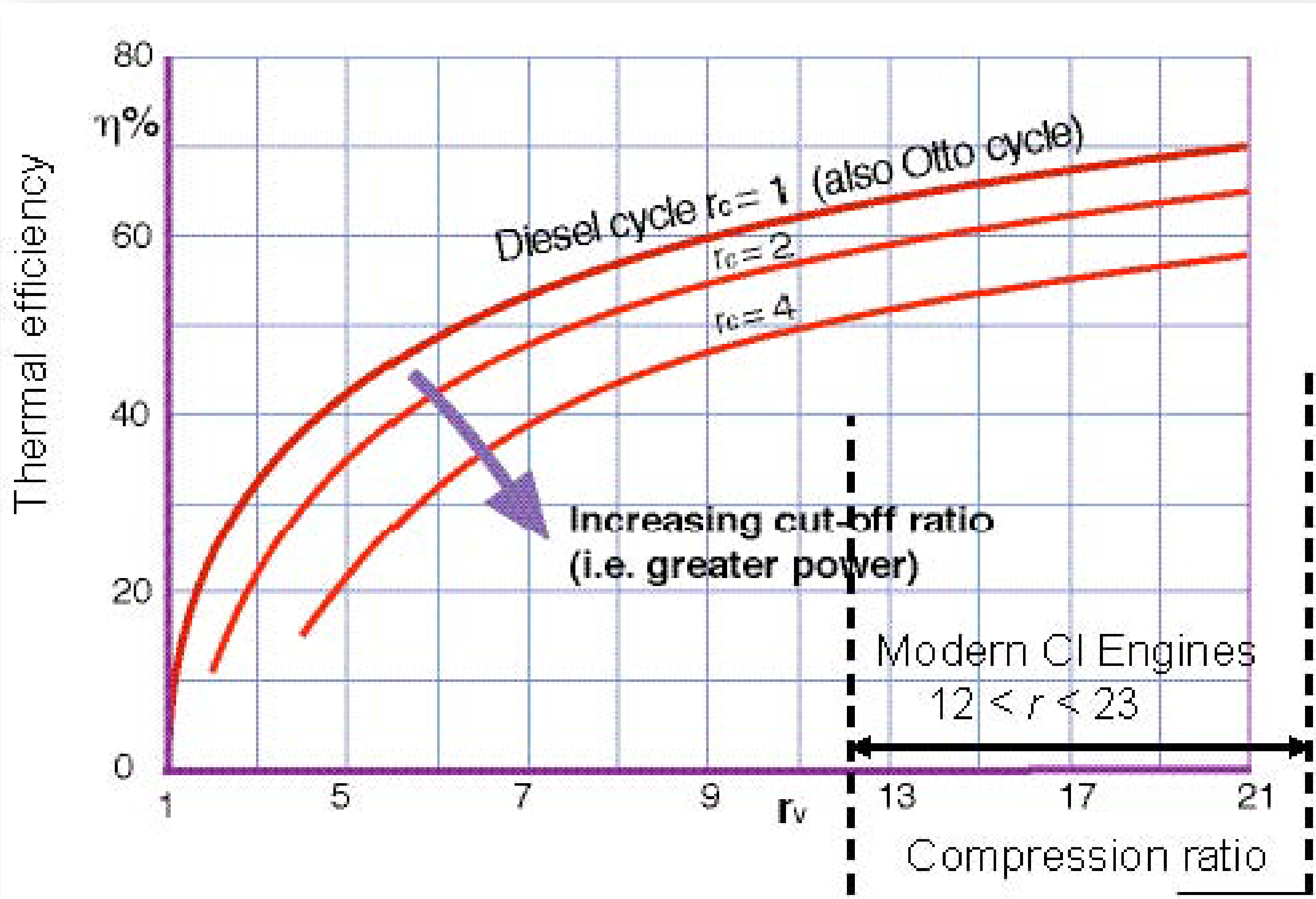
With rearrangement, this can be shown to equal:

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

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

- The value of the term in brackets > 1 .
- Comparing $\eta_{t,DIESEL}$ with ($\eta_{t,OTTO} = 1 - (1 / r_c)^{k-1}$,for a given $\mathbf{r_c}$ $\eta_{t,OTTO} > \eta_{t,DIESEL}$. Const.-volume combustion at TDC is more efficient than const.-pressure combustion.
- CI engines operate with much higher (CR) than SI engines (12 to 24 versus 8 to 11) and thus have higher thermal efficiencies.

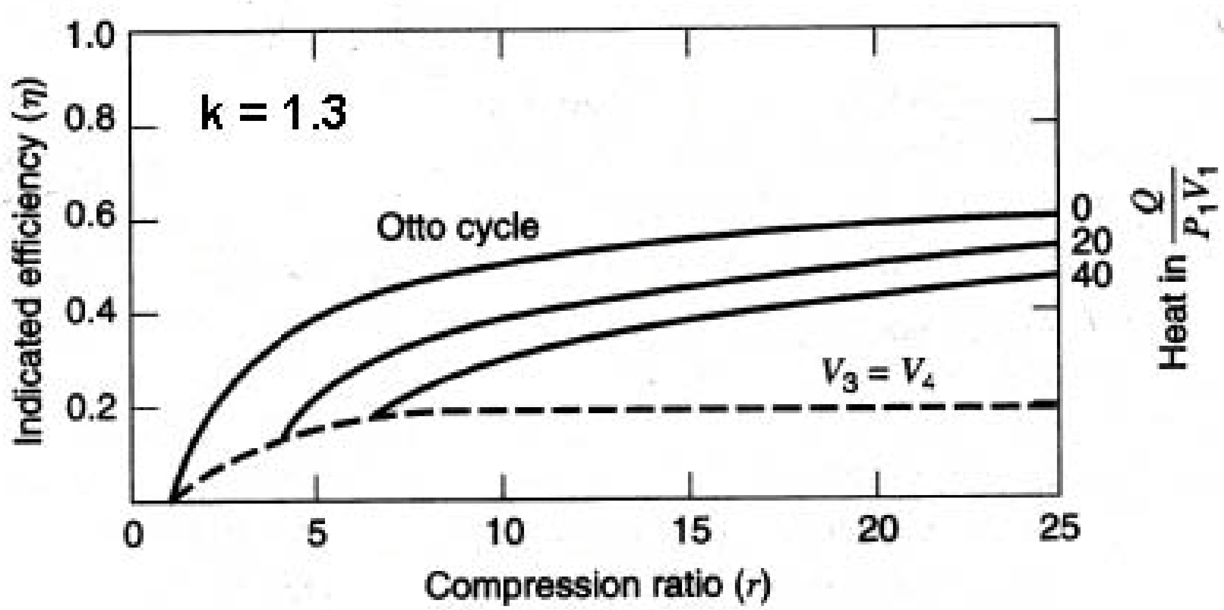
Effects of Compression Ratio and Cut-off Ratio on Thermal Efficiency

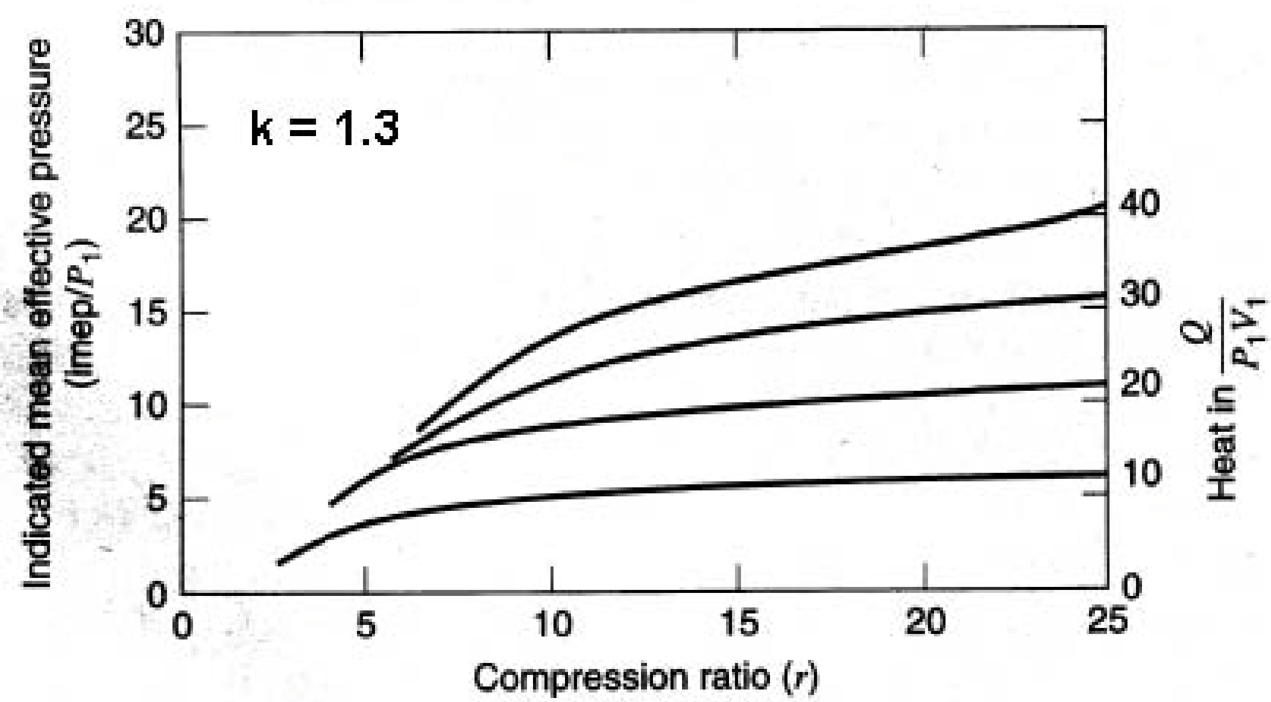


- The cut-off ratio is not a natural choice for the independent variable. A more suitable parameter is the heat input, the two are related by:

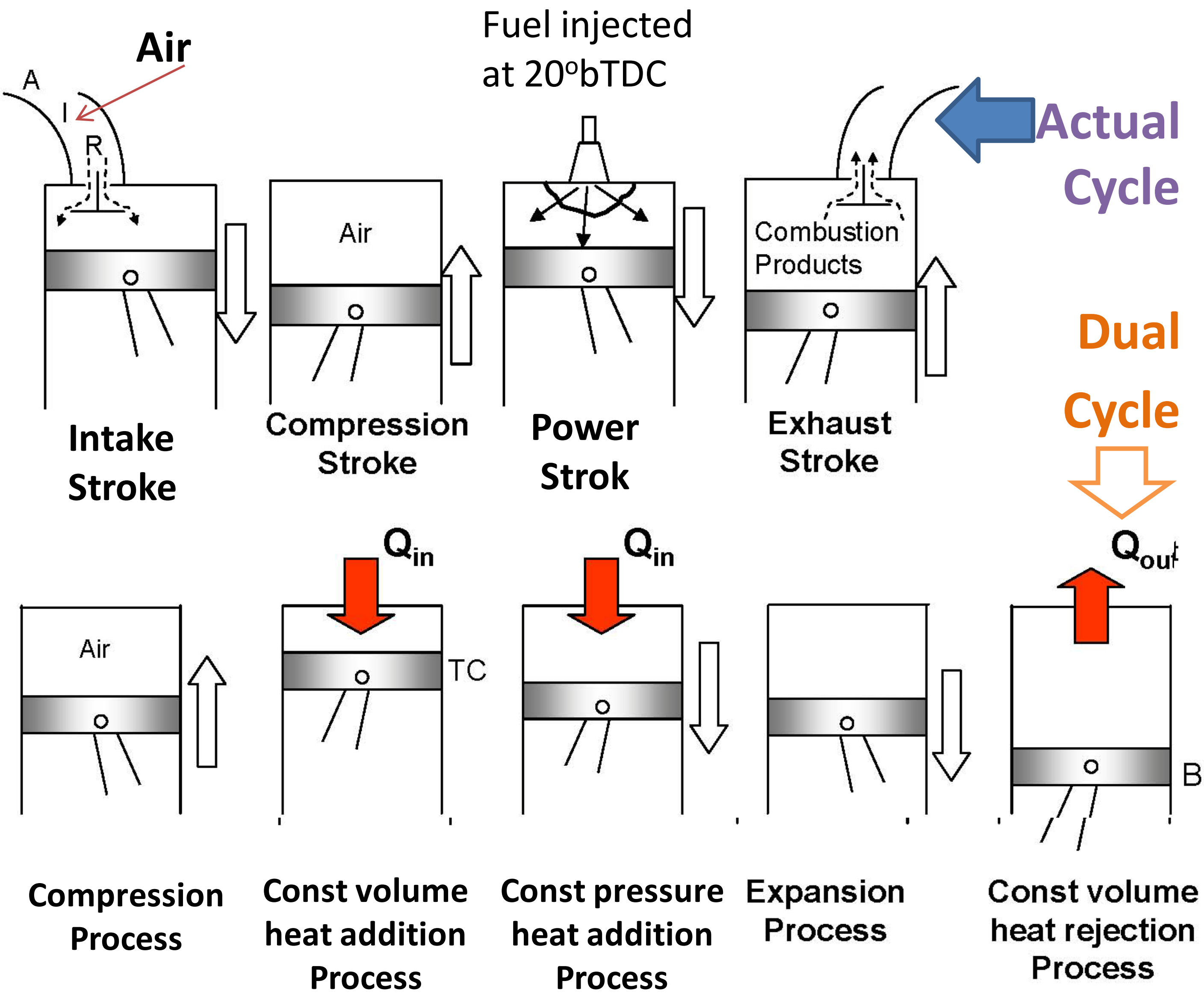
$$\beta = 1 - \frac{k-1}{k}\left(\frac{Q_{in}}{P_1 V_1}\right)\frac{1}{r_c^{\,k-1}} \Rightarrow \ as\ Q_{in} \to 0,\ \beta \to 0, and\ \eta \to \eta_{OTTO}$$

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

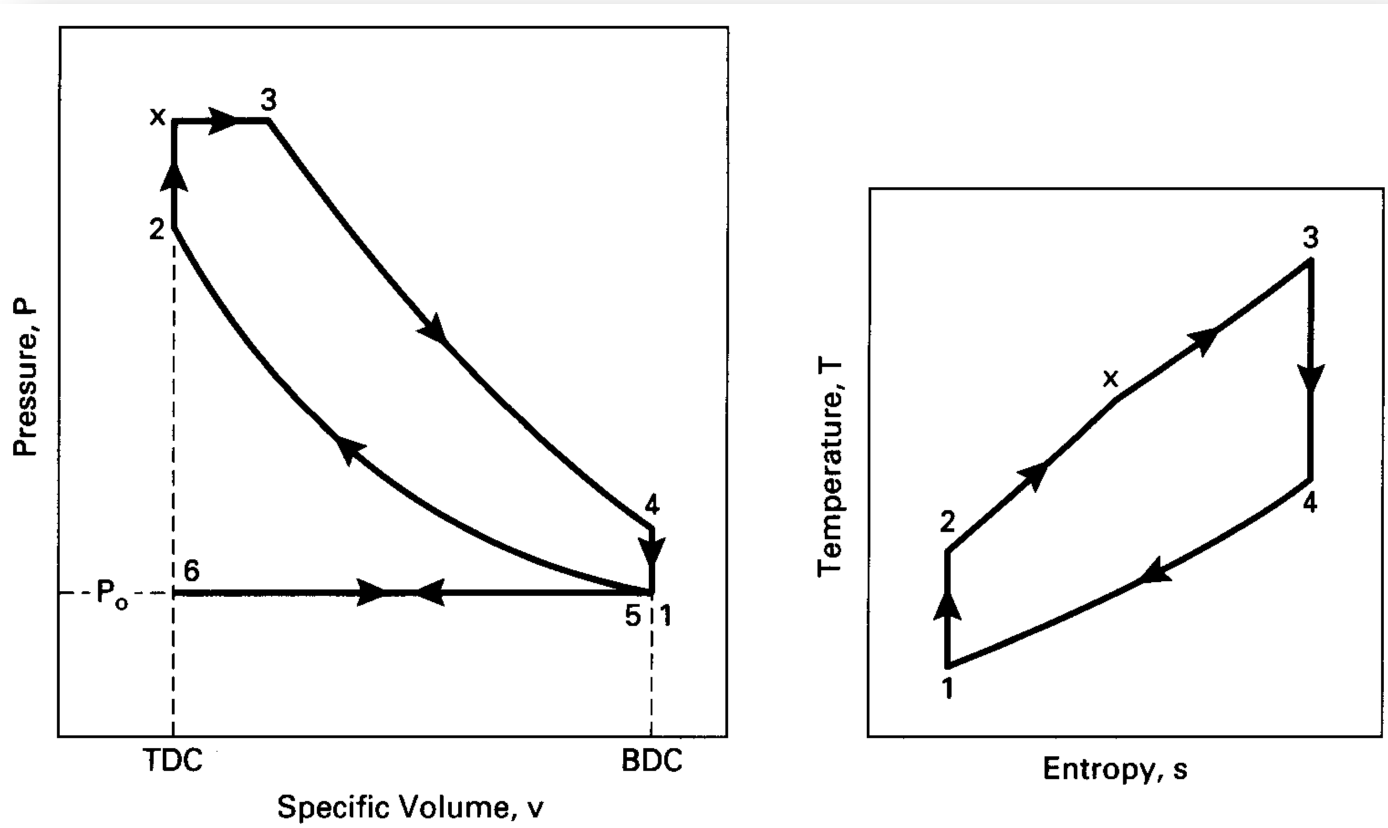




Modern CI Engine Cycle and the Thermodynamic Dual Cycle



Thermodynamic Analysis of Air-Standard Dual-Cycle



Process x-3—constant-pressure heat input (second part of combustion). All valves closed:

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

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

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

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

$$T_3 = T_{max}$$

Process x-3—constant-pressure heat input (second part of combustion). All valves closed:

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

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

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

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

$$T_3 = T_{\max}$$

Cutoff ratio:

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

Heat in:

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

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

Thermal efficiency of Dual cycle:

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

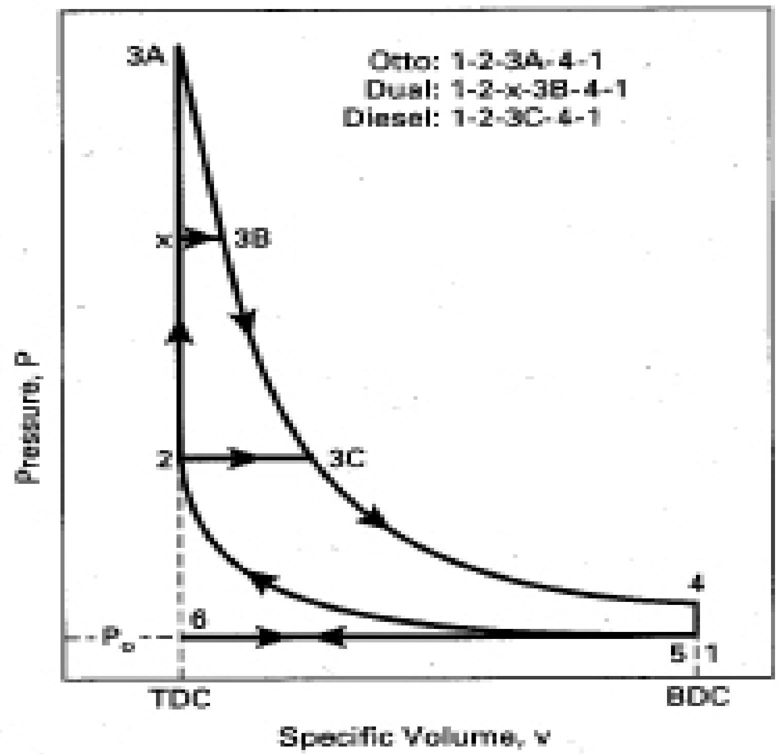
This can be rearranged to give:

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

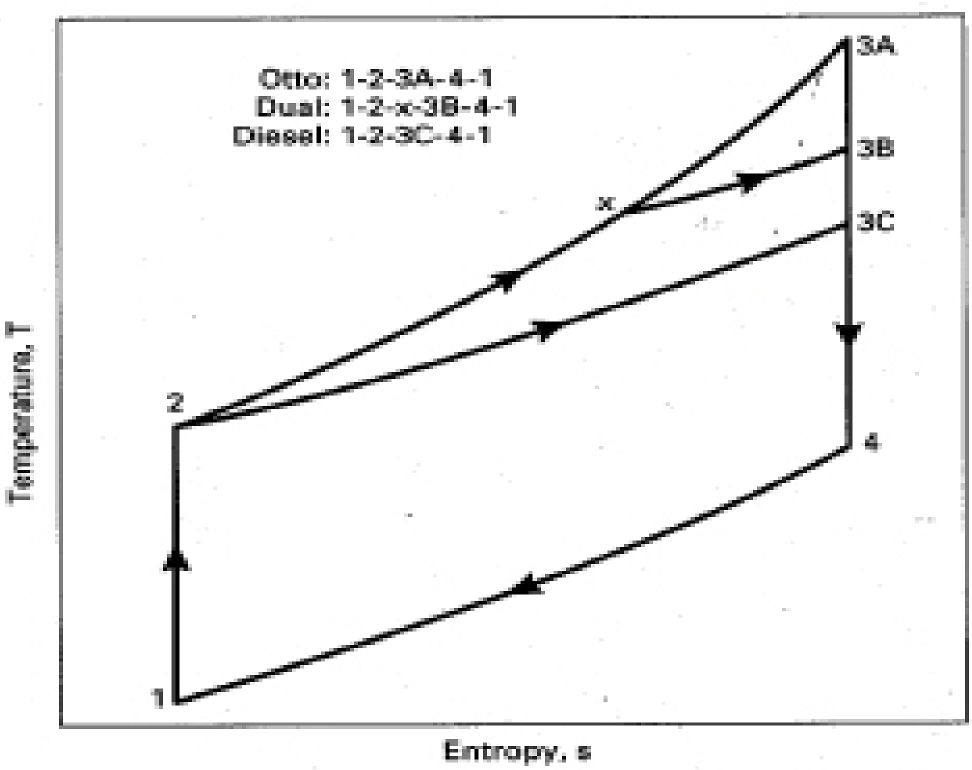
where: r_c = compression ratio
$k = c_p/c_v$
α = pressure ratio
β = cutoff ratio

Comparison of Otto, Diesel and Dual Cycles

For the same inlet conditions P1, V1 and the same compression ratio P2/P1:

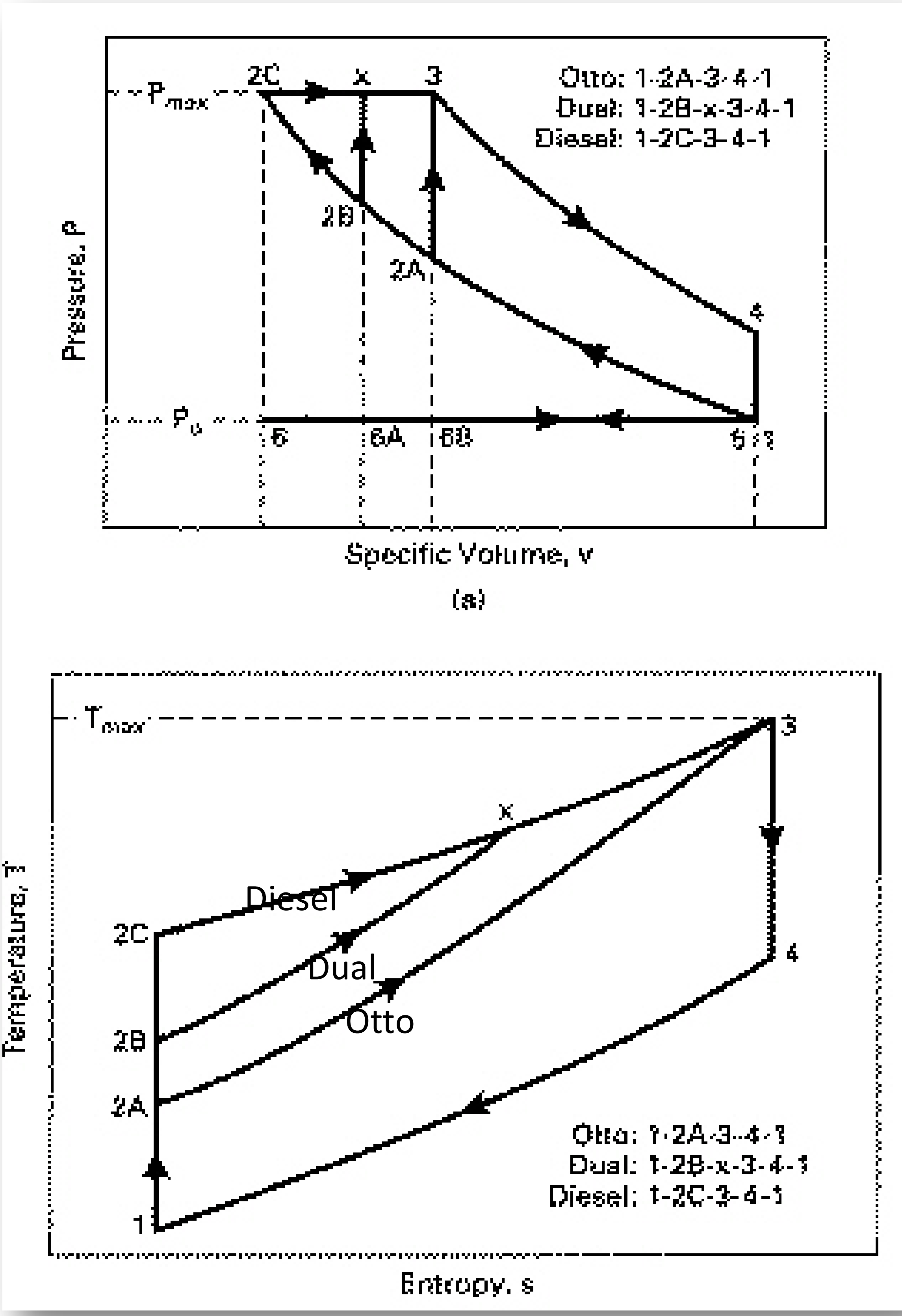


(a)



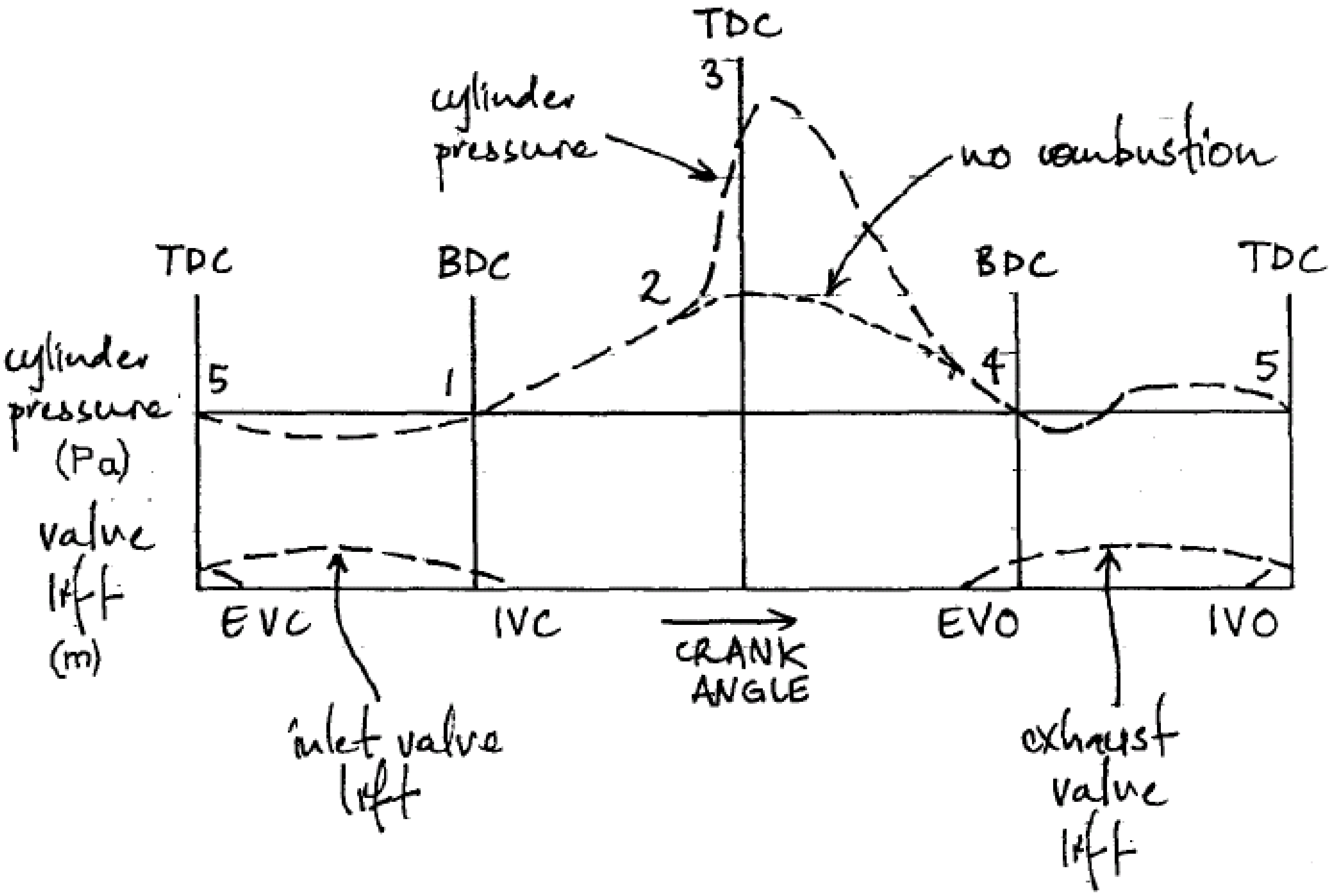
$$\eta_t = 1 - Q_{out}/Q_{in}$$

For the same inlet conditions P1, V1 and the same peak pressure P3:



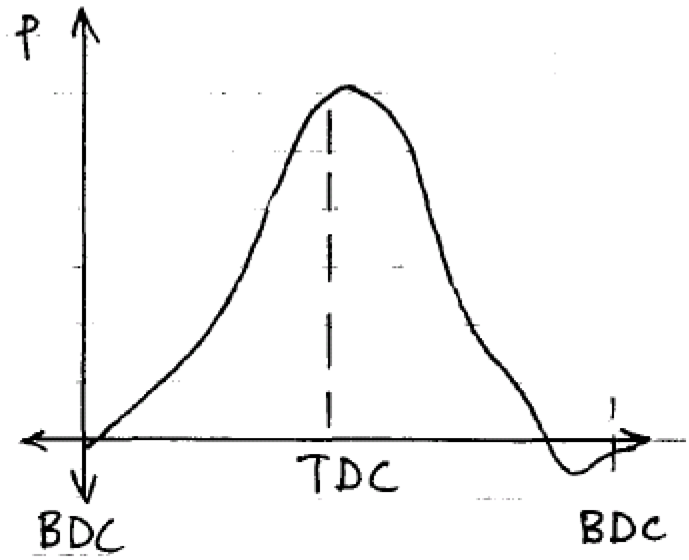
$$\eta_t = 1 - Q_{out}/Q_{in}$$

4 - Stroke SI Engine

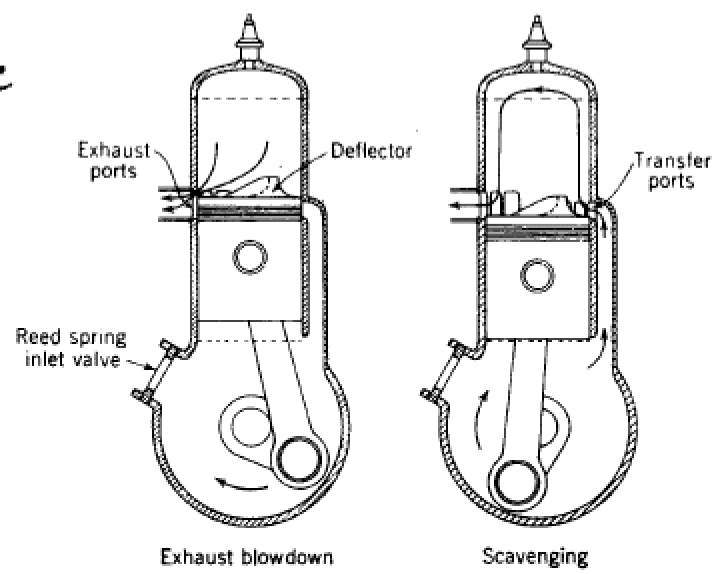


indicator diagram (pressure vs. crank angle) and valve timing diagram for a typical 4 stroke SI engine

2 - Stroke SI Engine

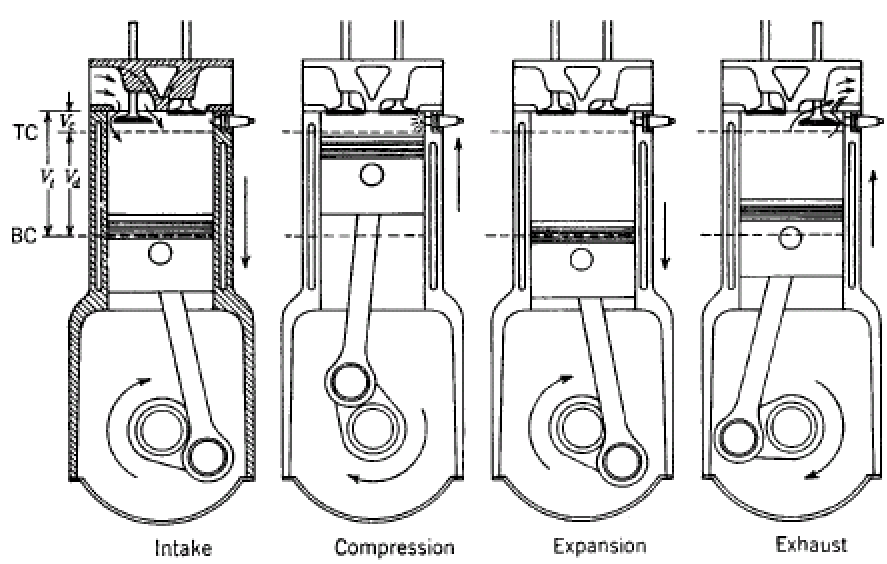


Two - stroke indicator diagram in terms of crank angle



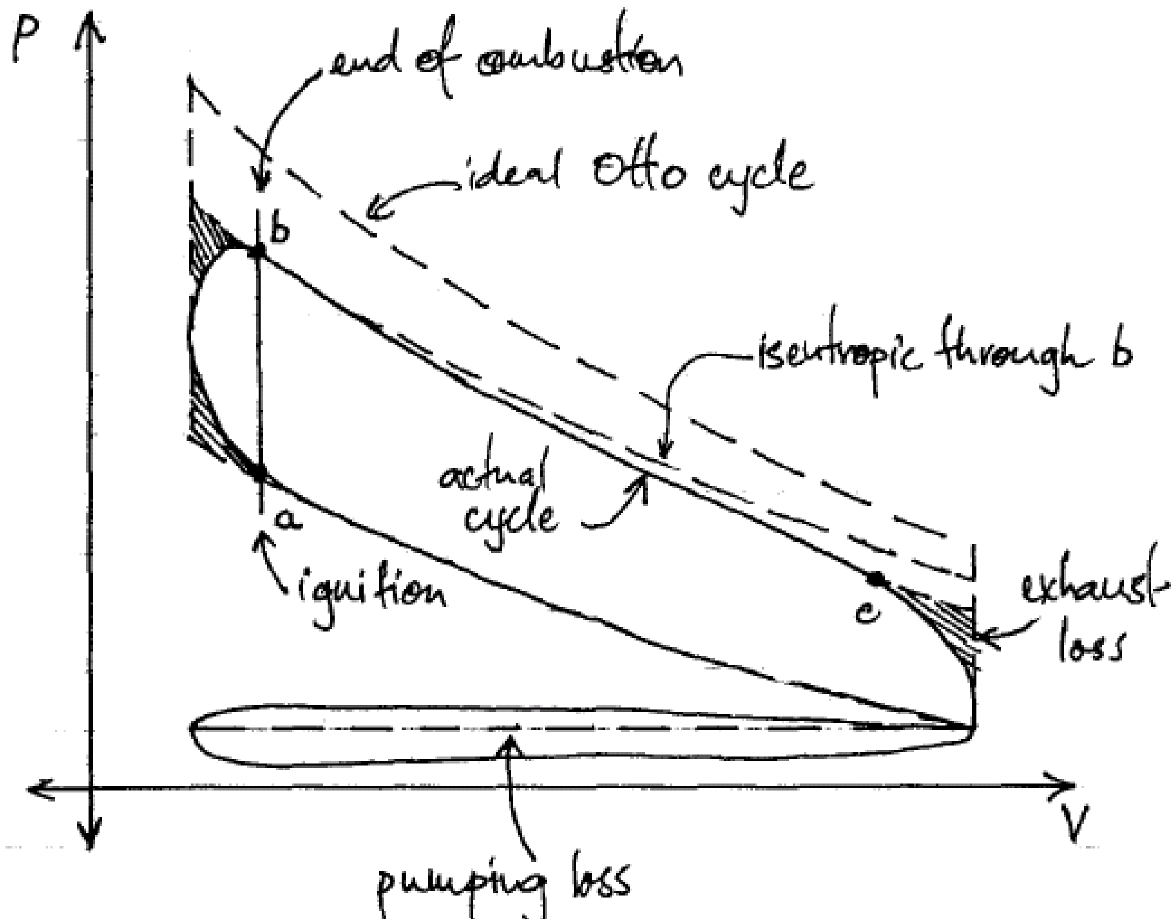
✓ ***H.W:***

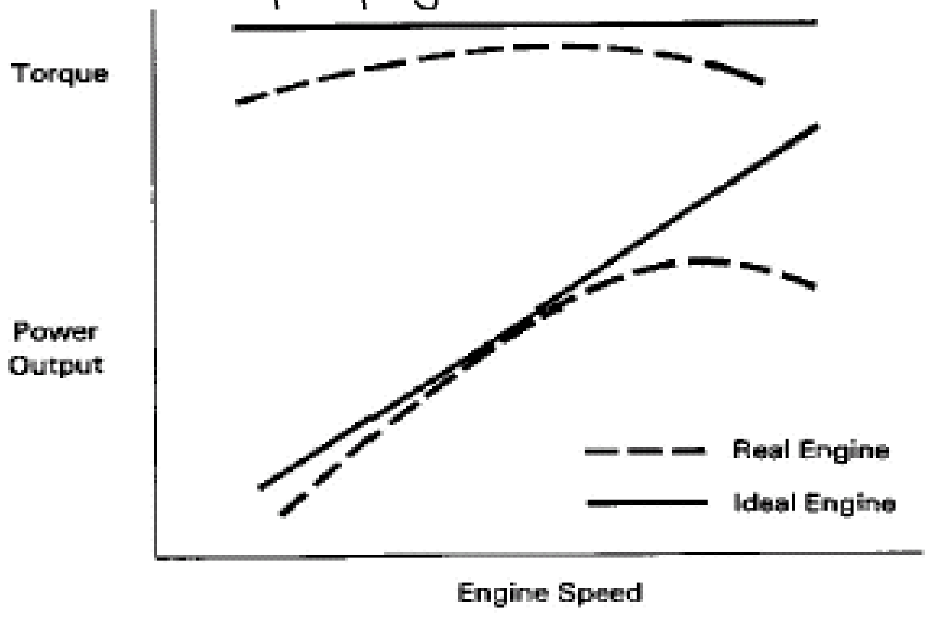
Compare between 4-stroke & 2-stroke engine cycles



Actual Otto cycles

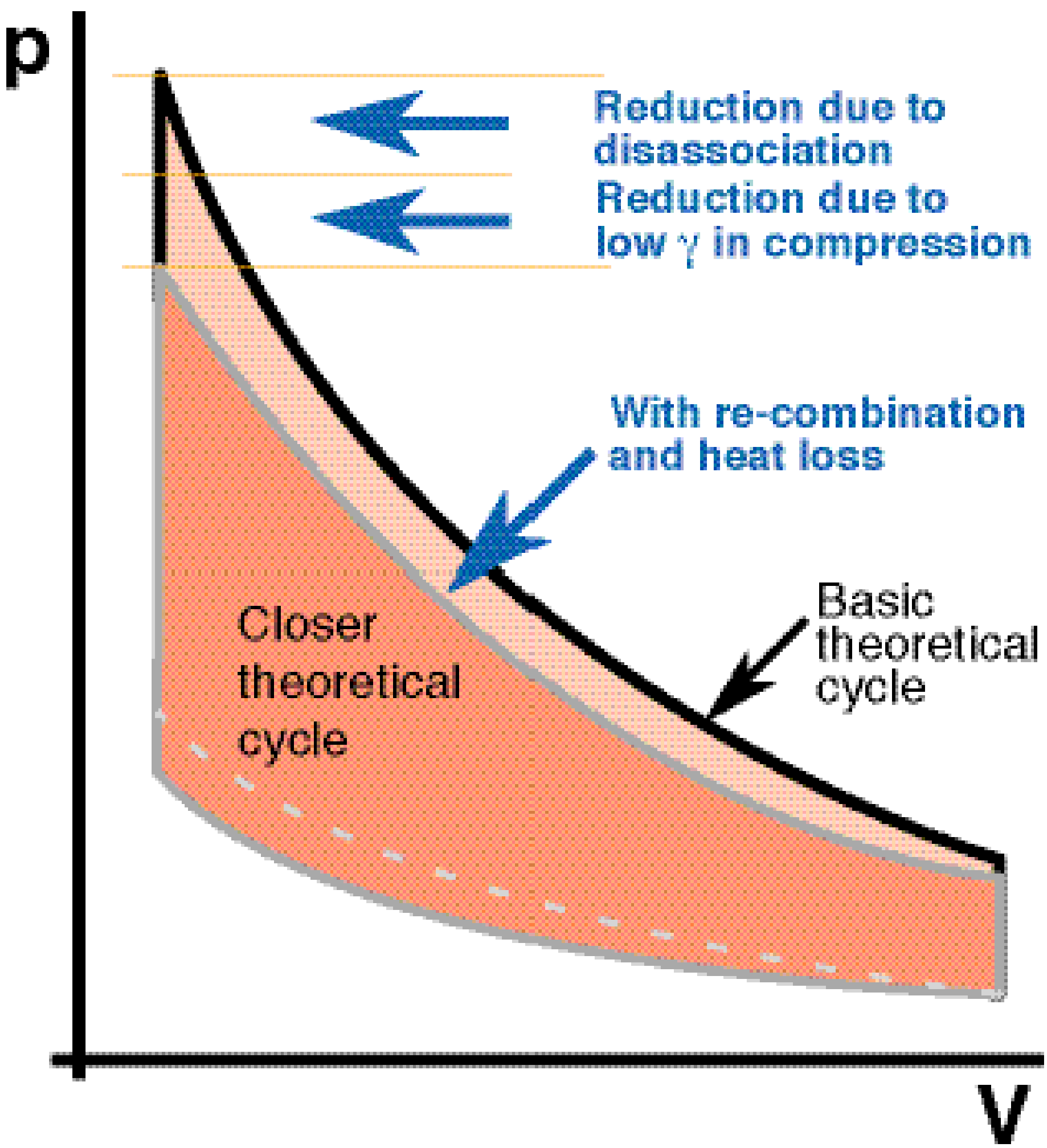
- For a SI engine using the Otto- cycle, a typical indicator diagram may look like:





Actual Otto cycles

For a SI engine, comparing real engine and ideal air standard Otto cycle, a typical indicator diagram may look like:



The Differences Between the Actual and the Ideal Cycles:

1. **compression:** processes are similar, with only a small heat loss.
2. **combustion:** actual combustion is not instantaneous and therefore cannot be at a constant volume. A finite combustion rate gives a lower maximum pressure.
3. **expansion:** heat transfer is significant. The cylinder pressure also falls from ‘c’ due to exhaust blowdown.
4. **pumping loss:** negative work out because the exhaust needs to be expelled and the fresh mixture induced
5. **other:** gas leakage past pistons

combustion duration depends on:
type of fuel chamber design and shape number and position of ignition sites engine speed
variations in specific heats with temperature (particularly of products) and species dissociation reduces lower pressure and lower temperature, thus lower heat release during combustion.

typical performance:

	actual	ideal
IMEP (kPa)	1430	1780
η_{TH}	29	36

as a rough guess, about 30% of difference in efficiency due to from timing, 60% due to heat losses and 10% due to exhaust loss

Actual Diesel cycle

***** most of the above also applies to diesel engines.*****

- The combustion duration is determined by:

1. **physical** delay: the time taken to atomize, vaporize and mix the fuel with the air
2. **chemical delay:** the time taken for pre-flame reactions to initiate a fuel combustion action. Therefore**,** to minimize the delay we require:

1. **good atomization**
2. **volatile fuel**
3. **good self-ignition quality (cetane no.)**
4. **good spray penetration**
5. **high air temperature**

Fuels and ignition

- An important difference between the diesel and petrol (Otto) engines is their respective use of 'heterogeneous' and 'homogeneous' combustion.

1. Homogeneous combustion (petrol) :

a) the fuel and air are premixed as a gaseous mixture. Combustion proceeds from initiation at one (or more) points, with ignition is by one or a number of sparks

b) conditions up to spark ignition must be such that the fuel does not ignite.

c) fuels are either a volatile liquid or gas: petrol, LPG, alcohol, natural gas.

d) fuels must be resistant to auto-ignition

2. Heterogeneous combustion (Diesel)

a) fuel is injected as a finely atomized spray of liquid droplets. The fuel burns when it is evaporated and mixed with air.

b) ignition is spontaneous, due to the high mixture temperature. The compression ratio must therefore be high.

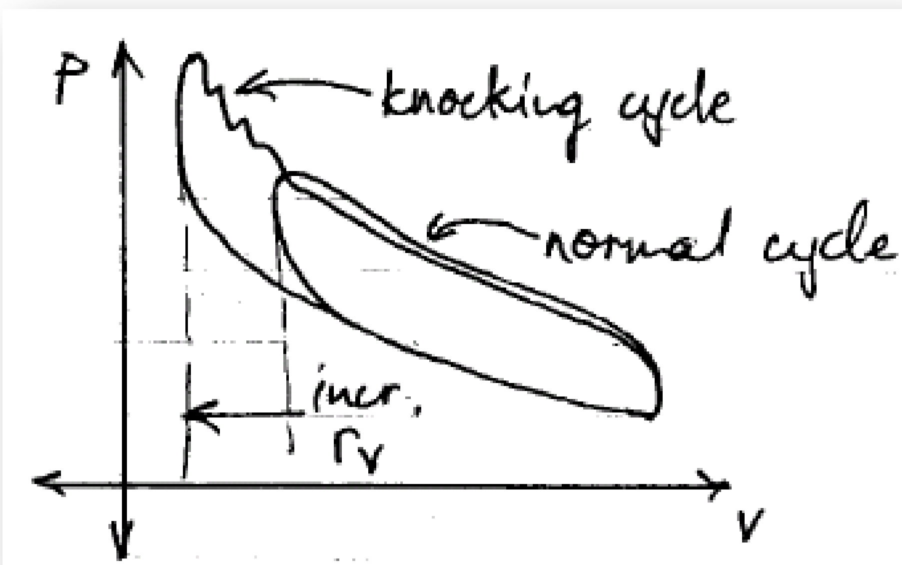
c) the engine compresses the air only, and the fuel is injected late into the compression stroke.

d) fuel must readily auto-ignite.

Limitations on the Compression Ratio

- ***spark ignition engines***

knock: as the CR increases, the peak pr. and temp. increase. Eventually the self-ignition temp. is reached and non-spark initiated combustion occurs. This gives uncontrolled combustion and the related pressure waves are called knock, and can result in poor performance and engine damage.



to prevent knock, usually either retard ignition, improve combustion chamber designs or increase the fuel octane number.

Limitations on the Compression Ratio

- ***Diesel engines***

The trade-off in CR is between:

1. improved efficiency.
2. increased heat losses with higher *CR* .
3. *increased engine strength and friction Heat losses and engine strength are considerations which can be controlled by improved design*