Thermodynamic analysis of IC Engine

Air-Standard Cycle

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Introduction

- The Three Thermodynamic Analysis of IC Engines are
  1. **Ideal Gas Cycle (Air Standard Cycle)**
     - Idealized processes
     - Idealize working Fluid
  2. **Fuel-Air Cycle**
     - Idealized Processes
     - Accurate Working Fluid Model
  3. **Actual Engine Cycle**
     - Accurate Models of Processes
     - Accurate Working Fluid Model

Introduction

- The operating cycle of an IC engine can be broken down into a sequence of separate processes
  - Intake, Compression, Combustion, Expansion and Exhaust.

- Actual IC Engine does not operate on ideal thermodynamic cycle that are operated on open cycle.

- The accurate analysis of IC engine processes is very complicated, to understand it well, it is advantageous to analyze the performance of an Idealized closed cycle

Air Standard Cycles

Air-Standard cycle differs from the actual by the following

1. The gas mixture in the cylinder is treated as air for the entire cycle, and property values of air are used in the analysis.

2. The real open cycle is changed into a closed cycle by assuming that the gases being exhausted are fed back into the intake system.

3. The combustion process is replaced with a heat addition term Q_{in} of equal energy value

Air standard cycles

4. The open exhaust process, which carries a large amount of enthalpy is denoted by Q_{out} of the equal energy value
5. Actual engine processes are approximated with ideal processes
    a. The almost-constant-pressure intake and exhaust strokes are assumed to be constant pressure.
    b. Compression strokes and expansion strokes are approximated by isentropic processes

Air Standard Cycles Assumption

c. The combustion process is idealized by a constant-volume process (SI cycle), a constant-pressure process (CI cycle), or a combination of both (CI Dual cycle).

d. Exhaust blow down is approximated by a constant-volume process.

e. All processes are considered reversible.

Charles Law

- For an ideal gas, volume is directly proportional Kelvin's temperature, if moles of gas and pressure are constant.

- $V \propto T$ If n and P are constant

- Decreased temperature ⟹ Decreased volume
- Increased temperature ⟹ Increased volume

Gay Lussac's Law

- The pressure of an ideal gas is directly proportional to the Kelvin temperature of the gas if the volume and moles of gas are constant

- $P \propto T$ if n and V are constant

- Decreased temperature ⟹ Decreased pressure
- Increased temperature ⟹ Increased pressure

Ideal Gas Law

- Combining Boyle's and Charles' laws allows for developing a single equation:

$$Pv = RT$$

  - Where the constant of proportionality R is called the ***gas constant***.

- The gas constant R is different for each gas and is determined from:

$$R = \frac{R_U}{M} \qquad [kJ/kg.K]$$

  - Ru is the ***Universal gas constant*** and M is the ***molar mass (molecular weight)***

Polytropic Process

- During Expansion and compression process, pressure and volume are often related by:

$$PV^n = C \quad or \quad P = CV^{-n}$$

  - Where C is a constant, and *n* is a value between 1.0 and 1.4.

- Assuming the combustion chamber is perfectly insulated (adiabatic), $n=k=1.4$. For an isothermal process, n=1.0.

Isentropic process

- For an ideal gas *k* is constant.

$$PV^k = C \qquad \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k$$

- Using the equation of state for an ideal gas,

$$PV = mRT$$

- $$TV^{k-1} = C \qquad TP^{\frac{1-k}{k}} = C$$

Isentropic work

- Work done during isentropic Process becomes:

$$W = \int_1^2 PdV = \int_1^2 CV^{-n}dV = C\frac{V_2^{-n+1} - V_1^{-n+1}}{-n+1} = \frac{P_2V_2 - P_1V_1}{1-n}$$

- For an ideal gas this equation can be written as:

$$W = \frac{mR(T_2 - T_1)}{1-k}$$

$$P = \rho RT$$

$$dh = c_P dT$$

$$du = c_v dT$$

Specific heat constants

- When analyzing what occurs within engines during the *operating cycle* and *exhaust flow*, the following air property values are used:

$$C_P = 1.108 \quad [kJ/kg-K]$$

$$C_V = 0.821 \quad [kJ/kg-K]$$

$$k = C_P / C_V = 1.108/0.821 = 1.35$$

$$R = C_P - C_V = 0.287 \quad [kJ/kg-K]$$

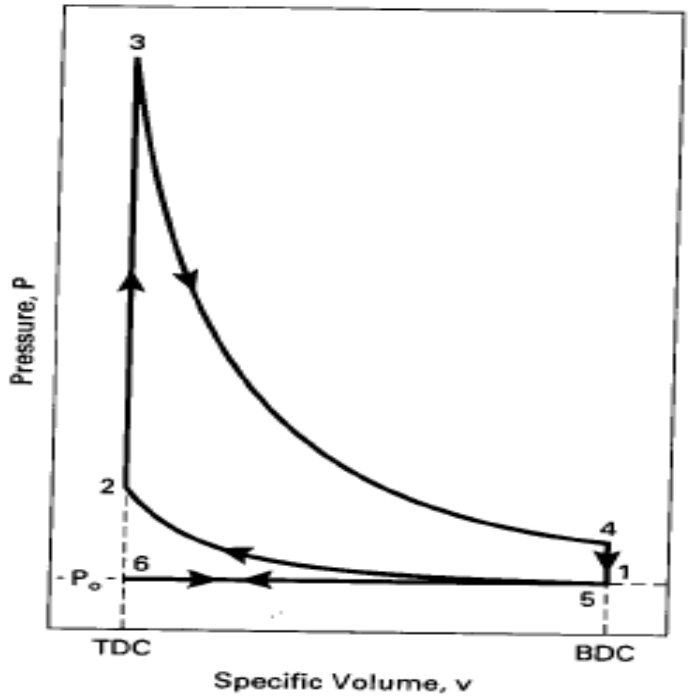
Air properties

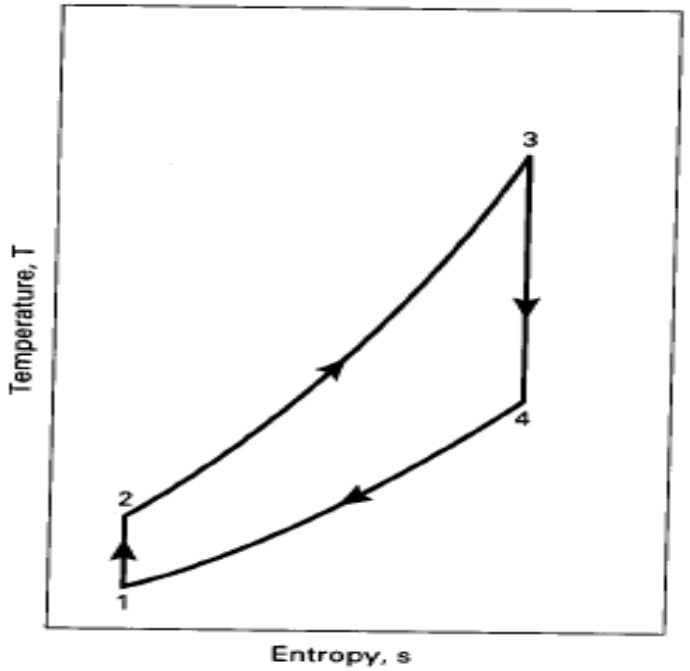
- For processes such as *inlet flow in superchargers, turbochargers, and carburetors*, and *air flow through the engine radiator,* the following air property values are used:

$$C_P = 1.005 \quad [kJ/kg-K]$$
$$C_V = 0.718 \quad [kJ/kg-K]$$
$$k = C_P/C_V = 1.005/0.718 = 1.4$$
$$R = C_P - C_V = 0.287 \quad [kJ/kg-K]$$

Otto Cycle

- Point 6 to 1 - Intake
- Point 1 to 2 - Isentropic Compression
- Point 2 to 3 - Constant-Volume Heat Input
- Point 3 to 4 – Isentropic Expansion
- Point 4 to 5 – Blow Down
- Point 5 to 6 - Exhaust





Intake Stroke

- Starts with the piston at TDC
- Constant pressure process at the inlet pressure of one atmosphere.
- In real engine process 6-1 will be slightly less than atmospheric due to pressure losses in the inlet air flow.
- The temperature of the air during the inlet stroke is increased as the air passes through the hot intake manifold.

Compression Stroke

- It is an isentropic compression from BDC to TDC (process 1-2)
- In real engine, the beginning of the stroke is affected by the intake valve not being fully closed until slightly after BDC.
- The end of compression is affected by the firing of the spark plug before TDC.
- In addition to increase in pressure there is also increase in temperature due to compressive heating

Combustion Process

- It is a constant-volume heat input process 2-3 at TDC.
- In real engines combustion starts slightly bTDC, reaches its maximum speed near TDC, is terminated a little aTDC.
- Peak cycle pressure and temperature is reached at point 3 due to energy added to the air within the cylinder.

Power (Expansion) Stroke

- High pressure on the piston face forces the piston back towards BDC and produces the work and power output of the engine.
- The power stroke of the real engine cycle is approximated with an isentropic process in the Otto cycle.
- The beginning of the power stroke is affected by the last part of the combustion process.
- The end of the power stroke is affected by the exhaust valve being opened before BDC.
- Values of both the temperature and pressure within the cylinder decrease as volume increases from TDC to BDC.

Exhaust Blowdown

- Exhaust valve is opened near the end of the power stroke
- A large amount of exhaust gas is expelled from the cylinder, reducing the pressure to that of the exhaust manifold
- The exhaust valve is opened bBDC to allow for the finite time of blowdown to occur.
- The Otto cycle replaces the exhaust blowdown open-system process of the real cycle with ***a constant–volume pressure reduction***, closed-system process 4-5.
- Enthalpy loss during this process is replaced with heat rejection in the engine analysis.

Exhaust Stroke

- Occurs as the piston travels from BDC to TDC.

- Process 5-6 is the exhaust stroke that occurs at a constant pressure of one atmosphere due to the open exhaust valve.

Thermodynamic Analysis of Air-Standard Otto Cycle

- ***Process 6-1***-Constant-pressure intake of air at Po.
  - Intake valve open and exhaust valve closed:

$$P_1 = P_6 = P_o$$

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

Process 1-2- Isentropic Compression Stroke

- All valves closed

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

$$P_2 = P_1\left(v_1/v_2\right)^{k} = P_1\left(V_1/V_2\right)^{k} = P_1\left(r_c\right)^{k}$$

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

$$w_{1-2} = \frac{\left(P_2v_2 - P_1v_1\right)}{\left(1-k\right)} = \frac{R\left(T_2 - T_1\right)}{\left(1-k\right)}$$

$$= \left(u_1 - u_2\right) = c_v\left(T_1 - T_2\right)$$

Process 2-3- Constant – volume heat input (combustion).

- All valves closed

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

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

$$Q_{2-3} = Q_{in} = m_f Q_{HV} \eta_C = m_m c_v (T_3 - T_2)$$

$$= (m_a + m_f) c_v (T_3 - T_2)$$

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

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

$$T_3 = T_{\max}$$

$$P_3 = P_{\max}$$

Process 3-4: Isentropic power or expansion stroke.

- All valves closed:

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

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

$$P_4 = P_3\left(v_3/v_4\right)^{k-1} = P_3\left(V_3/V_4\right)^{k-1} = P_3\left(1/r_C\right)^{k-1}$$

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

$$= \left(u_3 - u_4\right) = c_v\left(T_3 - T_4\right)$$

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 Po

- Exhaust valve open and intake valve closed:

$$P_5 = P_6 = P_o$$
$$w_{5-6} = P_o\left(v_6 - v_5\right) = P_o\left(v_6 - v_1\right)$$

Thermal efficiency of Otto Cycle

$$
\begin{aligned}
(\eta_t)_{OTTO} &= \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|} \\
&= 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)} \\
&= 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}
\end{aligned}
$$

Thermal efficiency of Otto Cycle

- For isentropic compression and expansion strokes and recognizing that $v_1 = v_4$ and $v_2 = v_3$

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

- Rearranging the temperature terms gives:

$$\left(T_4/T_1\right)=\left(T_3/T_2\right)$$

- Rearranging the above equation gives:

$$\left(\eta_t\right)_{OTTO}=1-\left(\frac{T_1}{T_2}\right)\left\{\frac{\left(\frac{T_4}{T_1}-1\right)}{\left(\frac{T_3}{T_2}-1\right)}\right\}$$

Thermal efficiency of Otto Cycle

- Thus efficiency becomes:

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

- Combining this with:

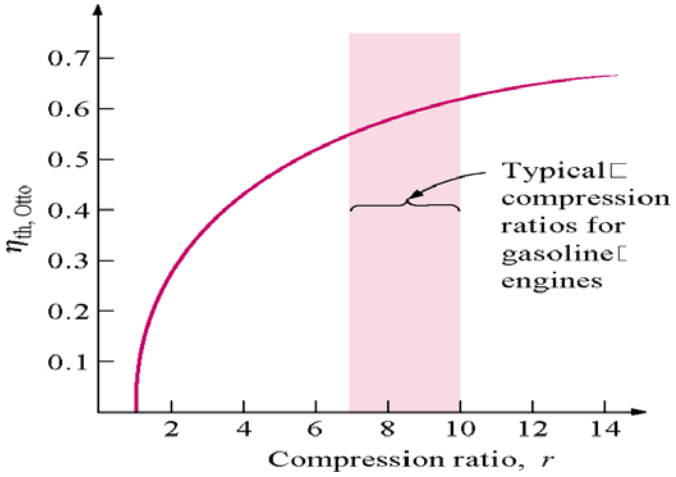
$$T_2 = T_1\left(v_1/v_2\right)^{k-1} = T_1\left(V_1/V_2\right)^{k-1} = T_1\left(r_c\right)^{k-1}$$

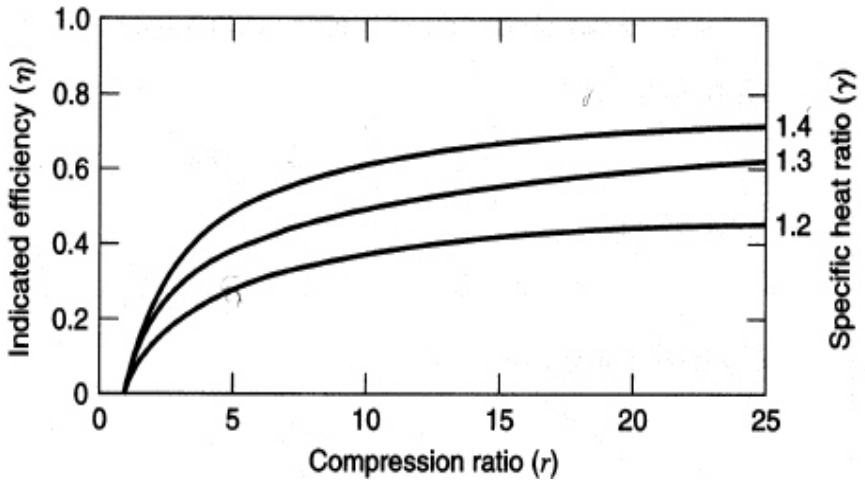
$$\left(\eta_t\right)_{OTTO} = 1 - \left(1/\left(v_1/v_2\right)^{k-1}\right)$$

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

OTTO CYCLE

- Spark ignition engine compression ratio limited by
  - T_3 (Auto ignition)
  - P_3 (material strength),
- For $r_c = 8$ the efficiency is 56%
- Cylinder temperatures vary between 300K and 2000K so $1.2 < k < 1.4$, $k = 1.3$ most representative





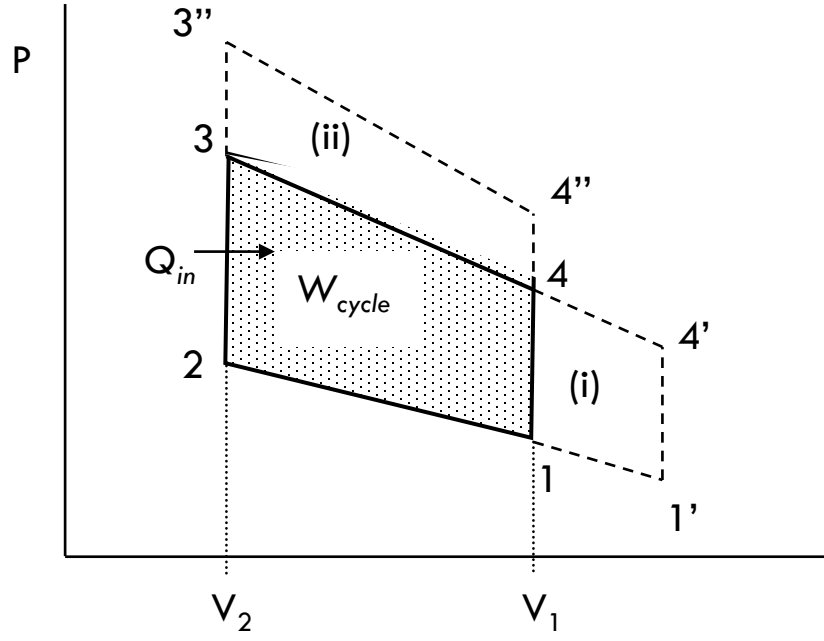
OTTO CYCLE

Factors Affecting Work per Cycle

- The indicted cycle work of an engine can be increased by either:
  - i) Increasing the *r* (1'→2)
  - ii) Increase Q*in* (2→3")

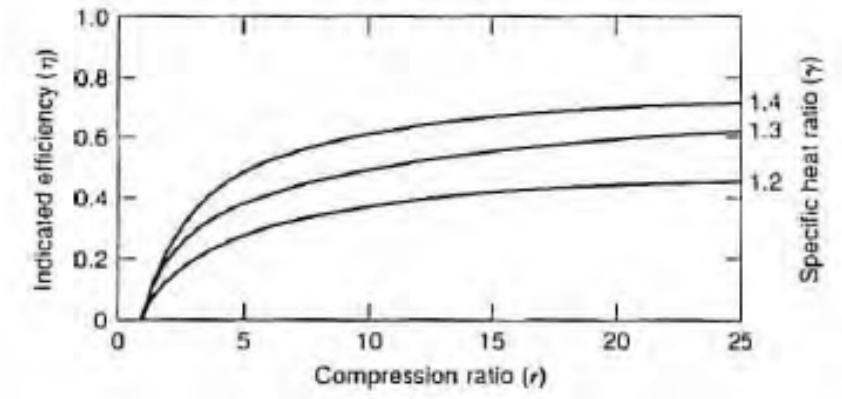
$$mep = \frac{W}{V_1 - V_2} = \frac{Q_{in}}{V_1}\left(\frac{r_c}{r_c - 1}\right)\eta_{th}$$

$$\frac{mep}{P_1} = \frac{W}{P_1(V_1 - V_2)} = \frac{Q_{in}}{P_1 V_1}\left(\frac{r_c}{r_c - 1}\right)\eta_{th}$$

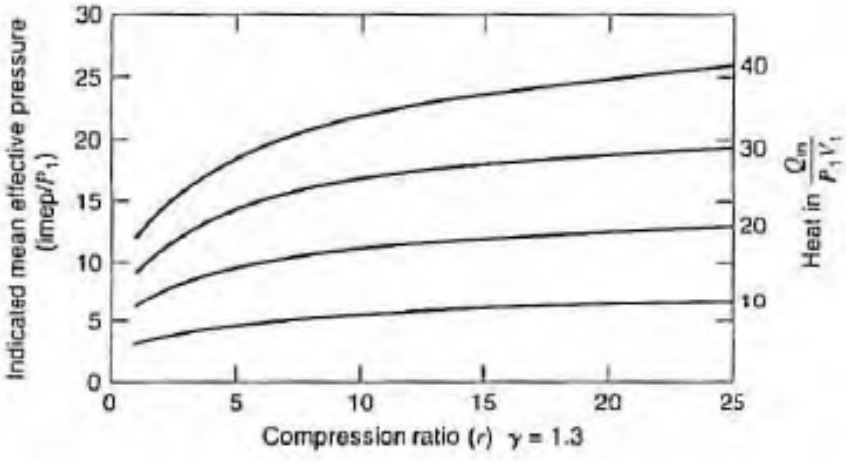


OTTO CYCLE

- The thermal efficiency of the Otto cycle depends only on compression ratio and specific heat ratio



- As expected the indicated mean effective pressure (imep) depends on heat addition and compression ratio



OTTO CYCLE

$$mep = \eta_{th} \frac{Q_{in} \dfrac{p_1}{mRT_1}}{1 - \dfrac{1}{r_c}} \qquad V_1 = m\frac{RT_1}{P_1}$$

- Non-dimensional mep with p_1 we get

$$\frac{mep}{p_1} = \eta_{th}\left[\frac{1}{1-\frac{1}{r_c}}\right]\left[\frac{q}{RT_1}\right] \qquad R = c_v(\gamma - 1) \qquad \frac{mep}{p_1} = \eta_{th}\frac{q}{c_v T_1}\frac{1}{\left[1-\frac{1}{r_c}\right][\gamma - 1]}$$

Mep/p_1 is a function of heat added, initial temperature, compression ratio properties of air, namely, cv and γ

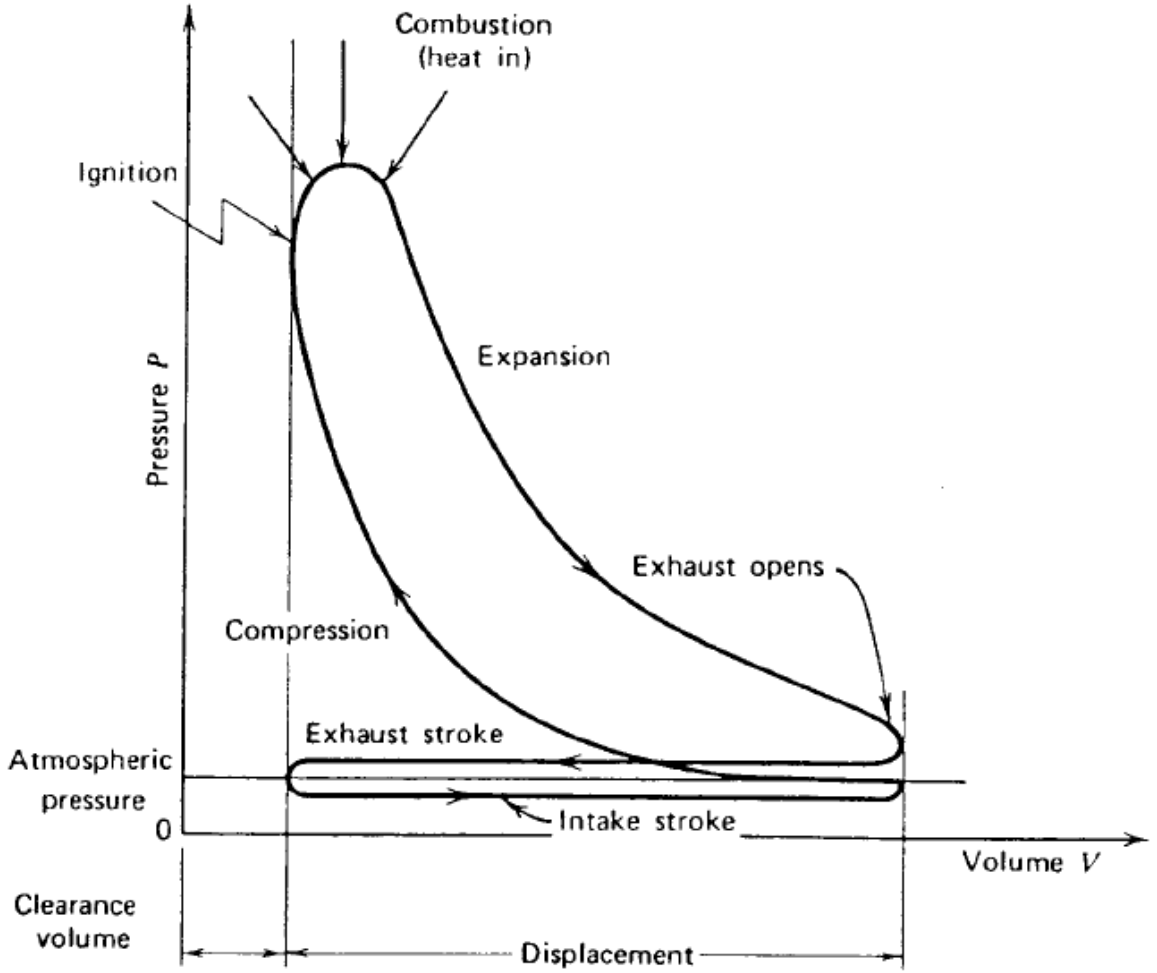
Determine the relation for mep/p3

$$\frac{mep}{p_3} = \frac{mep}{p_1}\frac{1}{r_c^{\gamma}}\frac{1}{\dfrac{q}{c_v T_1 r_c^{\gamma-1}} + 1}$$

Closing Thoughts on the Otto Cycle

- The Otto-cycle efficiency serves as an upper limit to the efficiency of SI engines.
- In practice this efficiency is never achieved.
- This theoretical analysis is flawed in that it ignores friction and heat transfer.

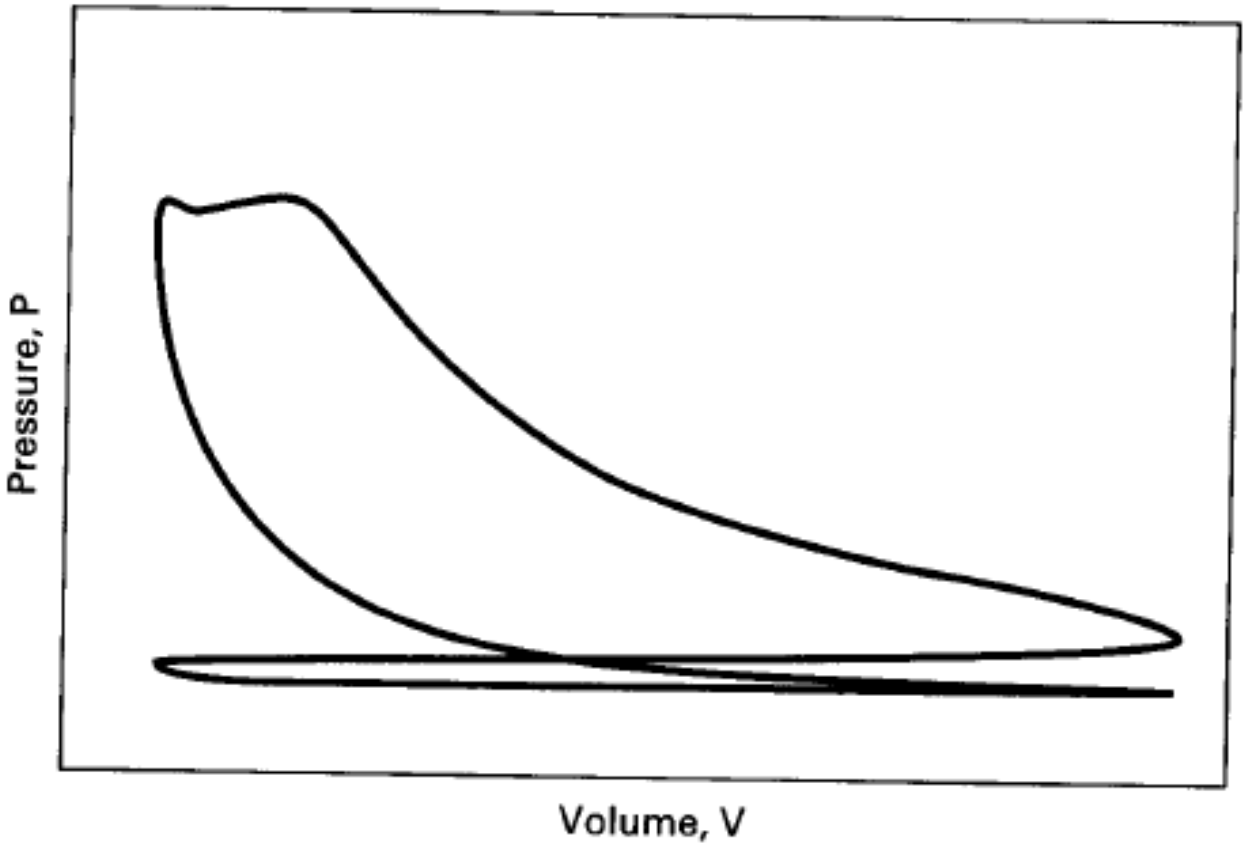
P-V Diagram of a SI Engine



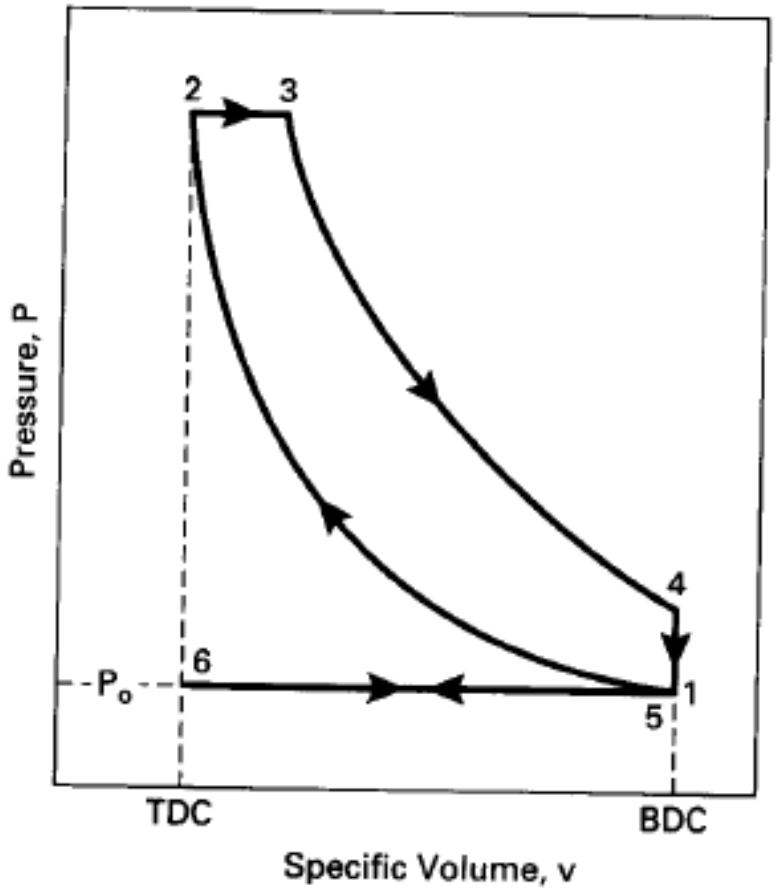
Diesel Cycle

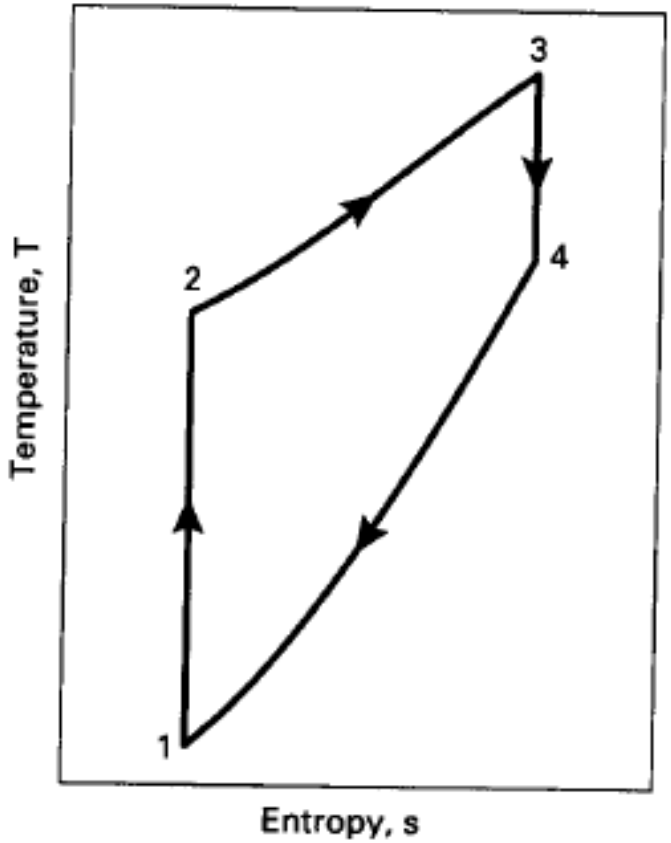
- Fuel is injected into the combustion chamber very late in the compression stroke
- Due to ignition delay and the finite time required to inject the fuel, combustion lasted into the expansion stroke
- This keeps the pressure at peak levels well past TDC.
- Combustion process is approximated as ***constant-pressure heat input*** in an air standard cycle

Indicator diagram of CI engine



Air–Standard diesel cycle





Thermodynamic Analysis of Air-Standard Diesel Cycle

- ***Process 6-1***-Constant-pressure intake of air at Po.
- Intake valve open and exhaust valve closed:

$$P_1 = P_6 = P_o$$

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

Process 1-2- Isentropic Compression Stroke

- All valves closed:

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

$$P_2 = P_1\left(v_1/v_2\right)^{k} = P_1\left(V_1/V_2\right)^{k} = P_1\left(r_c\right)^{k}$$

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

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

$$= \left(u_1 - u_2\right) = C_V\left(T_1 - T_2\right)$$

Process 2-3- Constant – Pressure heat input (combustion).

- All valves closed:

$$
\begin{aligned}
Q_{2-3} &= Q_{in} = m_f Q_{HV} \eta_C = m_m c_p \left(T_3 - T_2\right) \\
&= \left(m_a + m_f\right) c_p \left(T_3 - T_2\right) \\
Q_{HV} \eta_C &= \left(AF + 1\right) c_p \left(T_3 - T_2\right) \\
q_{2-3} &= q_{in} = c_p \left(T_3 - T_2\right) = \left(h_3 - h_2\right) \\
w_{2-3} &= q_{2-3} = \left(u_3 - u_2\right) = P_2 \left(v_3 - v_2\right) \\
T_3 &= T_{\max}
\end{aligned}
$$

Cut of ratio

Cut of Ratio (β): The change in volume that occurs during combustion, given as a ratio.

From ideal gas equation $P_2V_2/T_2 = P_3V_3/T_3$ ⇒

$T_3/T_2 = V_3/V_2$, $P_3 = P_2$

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

Process 3-4: Isentropic power or expansion stroke.

- All valves closed:

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

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

$$P_4 = P_3\left(v_3/v_4\right)^{k} = P_3\left(V_3/V_4\right)^{k}$$

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

$$= \left(u_3 - u_4\right) = c_v\left(T_3 - T_4\right)$$

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:

$$P_5 = P_6 = P_o$$
$$w_{5-6} = P_o\left(v_6 - v_5\right) = P_o\left(v_6 - v_1\right)$$

Thermal efficiency of diesel Cycle

$$\eta_{th,Diesel} = \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|}$$

$$= 1 - \frac{c_v(T_4 - T_1)}{c_p(T_3 - T_2)}$$

$$= 1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)}$$

$$= 1 - \frac{T_1(T_4/T_1 - 1)}{T_2 k(T_3/T_2 - 1)}$$

From combined gas law $\frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} \rightarrow \frac{T_3}{T_2} = \frac{V_3}{V_2} = \beta$

$$T_3 = T_2\beta$$

Thermal efficiency of diesel Cycle

From Isentropic compression

$$T_2 = T_1 (r_c)^{k-1}$$

Substituting isentropic expression into expression obtained from combined law we arrive at

$$T_3 = T_1 (r_c)^{k-1} \beta$$

From Isentropic Expansion

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} \rightarrow T_4 = T_3 \left(\frac{V_3}{V_2}\frac{V_2}{V_4}\right)^{k-1} \rightarrow T_4 = T_3 \left(\beta \frac{1}{r_c}\right)^{k-1}$$

$$T_4 = T_1 (r_c)^{k-1} \beta \left(\beta \frac{1}{r_c}\right)^{k-1}$$

$$T_4 = T_1 (\beta)^k$$

Thermal efficiency of diesel Cycle

- With rearrangement, this can be shown to equal:

$$(\eta_t)_{Diesel} = 1 - \left(\frac{1}{r_c}\right)^{k-1}\left[\frac{\beta^k - 1}{k(\beta - 1)}\right]$$

- Where:

$$r_c = Compression\ ratio$$
$$k = C_P / C_V$$
$$\beta = cutoff\ \ ratio$$

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

Note: CI needs higher r_c compared to SI to ignite fuel

Thermal Efficiency - Diesel cycle

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

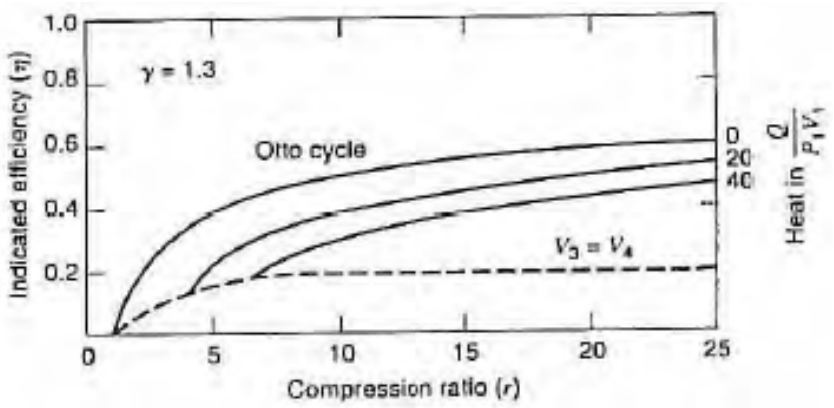
$$\beta = 1 - \frac{k-1}{k}\left(\frac{Q_{in}}{P_1 V_1}\right)\frac{1}{r_c^{\,k-1}}$$

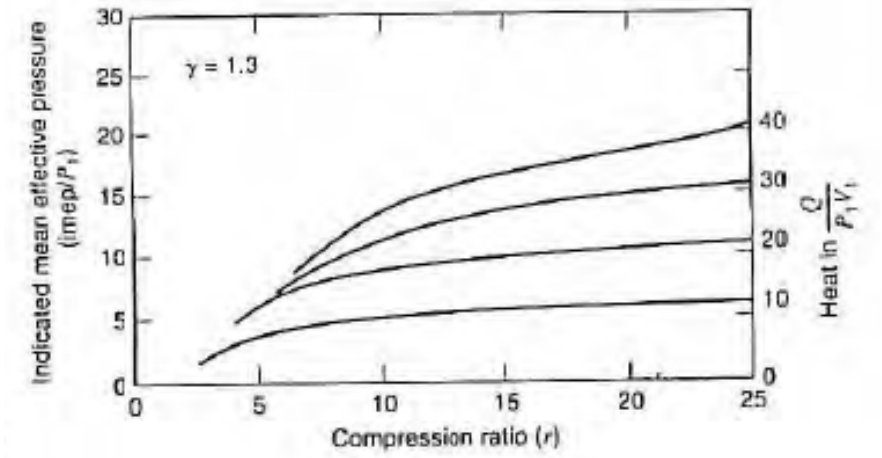
Substituting the cutoff ratio equation into the thermal efficiency equation for diesel cycle, it is possible to plot some curves

Thermal Efficiency - Diesel cycle

- Higher efficiency is obtained by adding less heat per cycle, Q_{in}

- In diesel engine the fuel cutoff ratio depends on the output maximum for maximum output

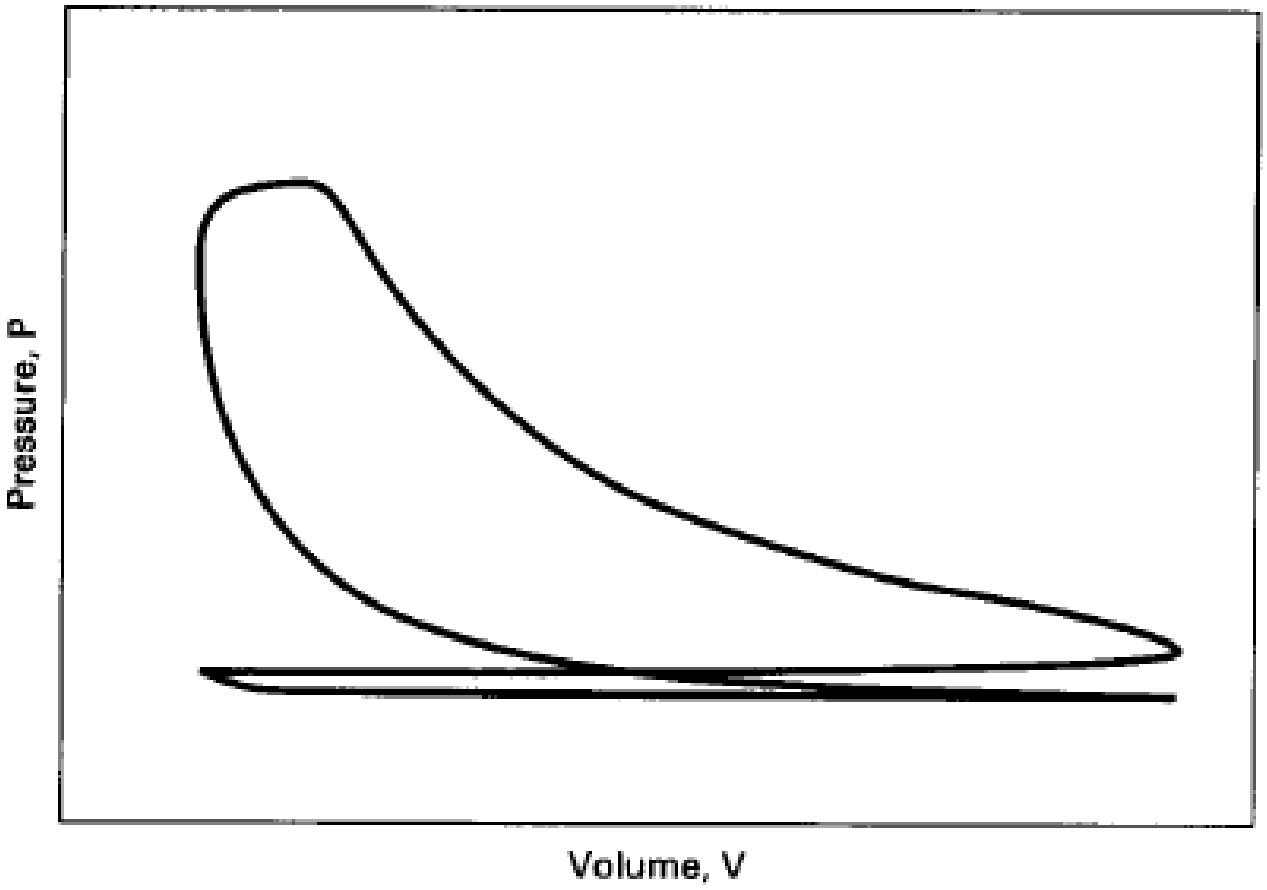




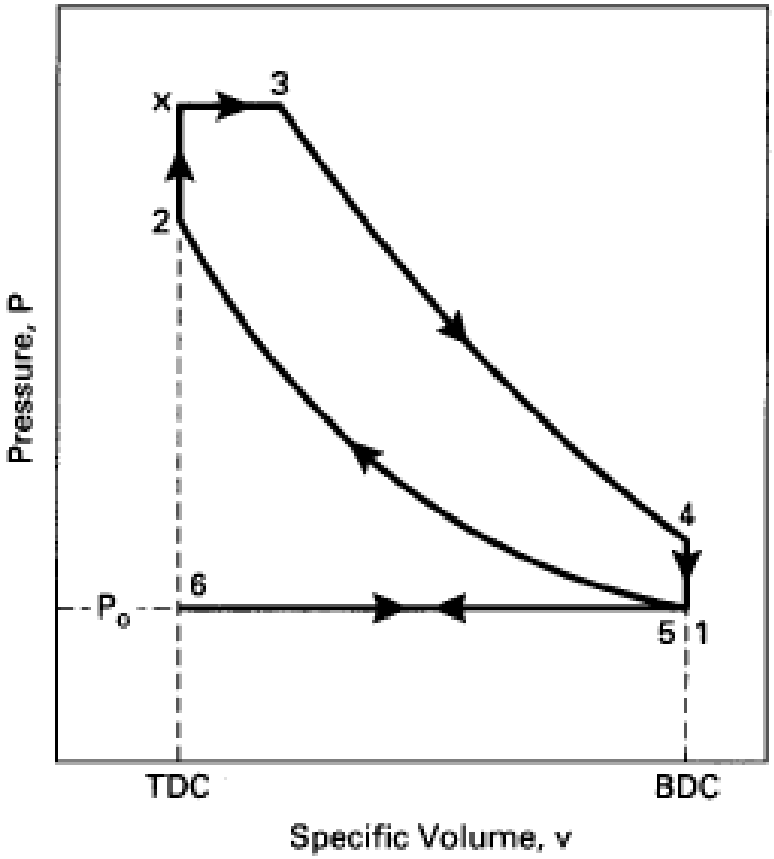
DUAL CYCLE

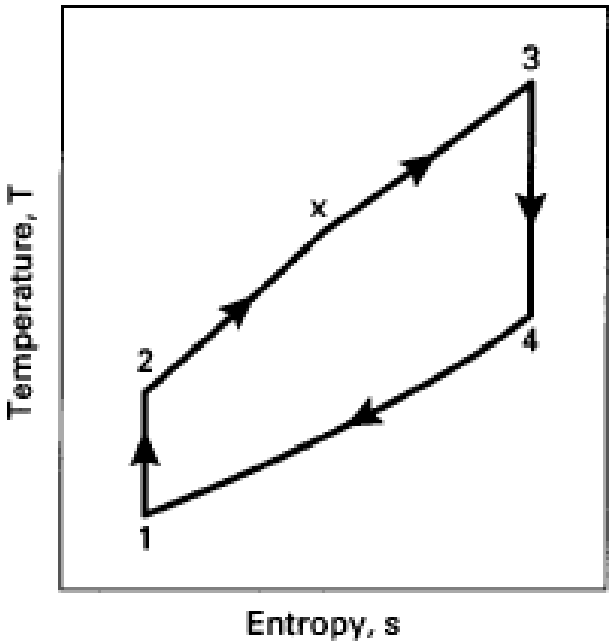
- Fuel is injected earlier in the compression stroke, around 20° bTDC
- Some of the combustion occurs almost at constant volume at TDC similar to Otto Cycle
- The fuel is being injected at TDC, and combustion of this fuel keeps the pressure high into the expansion stroke.
- The heat input process of combustion is approximated by a dual process of ***constant volume*** followed by ***constant pressure***

Indicator diagram of Dual cycle engine



Air Standard Dual cycle





Thermodynamic Analysis of Air-Standard Dual Cycle

- The analysis of an air-standard Dual cycle is the same as that of the Diesel cycle except for the heat input process (combustion) 2-x-3

- *Process 2-x: Constant-volume heat input (first part of combustion)*
- All valves closed:

$$V_x = V_2 = V_{TDC}$$

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

$$Q_{2-x} = m_m c_v (T_x - T_2) = (m_a + m_f) c_v (T_x - T_2)$$

$$q_{2-x} = c_v (T_x - T_2) = (u_x - u_2)$$

$$P_x = P_{\max} = P_2 (T_X / T_2)$$

Thermodynamic Analysis of Air-Standard Dual Cycle

- Pressure ratio is defined as the rise in pressure during combustion, given as a ratio:

$$\alpha = P_X / P_2 = P_3 / P_2 = T_X / T_2 = \left(1/r_c\right)^K \left(P_3 / P_1\right)$$

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

- *Process x-3- Constant-Pressure heat input (Second part of combustion)*
- All valves are closed:

$$P_3 = P_X = P_{\max}$$
$$w_{2-x} = 0$$
$$Q_{x-3} = m_m c_p \left(T_3 - T_x\right) = \left(m_a + m_f\right) c_p \left(T_3 - T_x\right)$$
$$q_{x-3} = c_p \left(T_3 - T_x\right) = \left(h_3 - h_x\right)$$
$$w_{x-3} = q_{x-3} - \left(u_3 - u_x\right) = P_x \left(v_3 - v_x\right) = P_3 \left(v_3 - v_x\right)$$
$$T_3 = T_{\max}$$

Thermal efficiency of Dual Cycle:

$$(\eta_t)_{DUAL} = \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|}$$

$$= 1 - \frac{c_v(T_4 - T_1)}{\left[c_v(T_x - T_2) + c_p(T_3 - T_x)\right]}$$

$$= 1 - \frac{(T_4 - T_1)}{\left[(T_x - T_2) + k(T_3 - T_x)\right]}$$

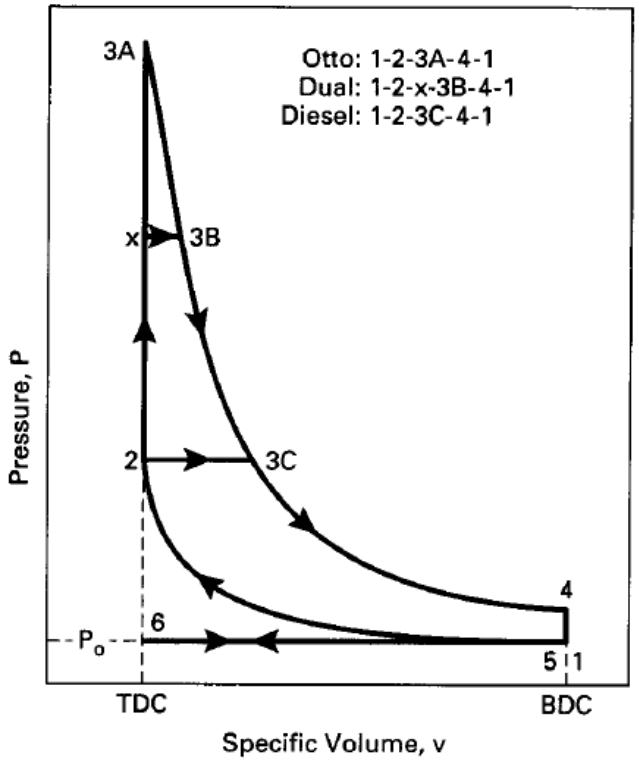
- This can rearranged to give:

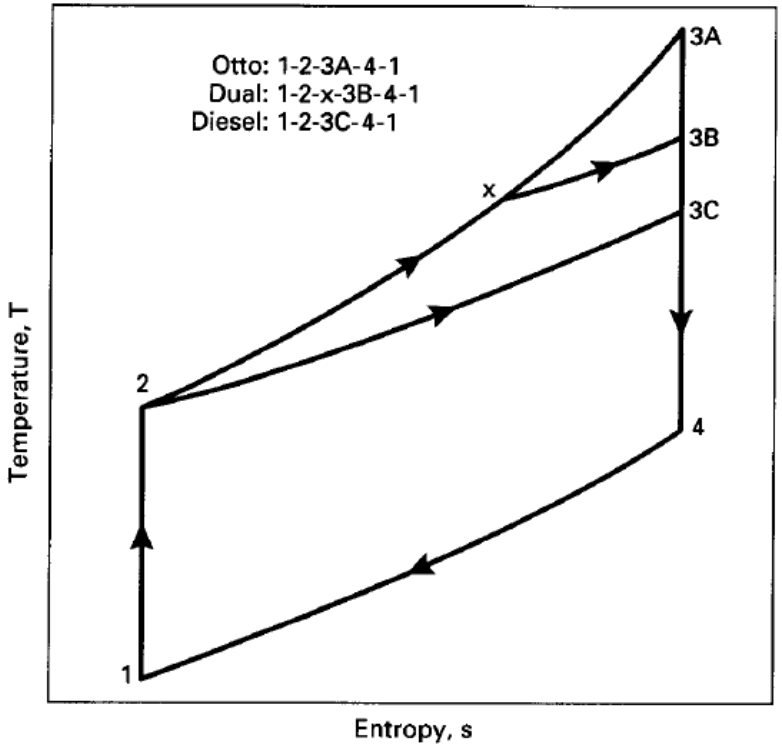
$$(\eta_t)_{DUAL} = 1 - \left(\frac{1}{r_c}\right)^{k-1}\left[\frac{\{\alpha\beta^k - 1\}}{\{k\alpha(\beta - 1) + \alpha - 1\}}\right]$$

Comparison of OTTO,DIESEL, and DUAL Cycles

- For the same inlet conditions, the same compression ratios and same heat removal:

$$(\eta_t)_{OTTO} > (\eta_t)_{DUAL} > (\eta_t)_{DIESEL}$$

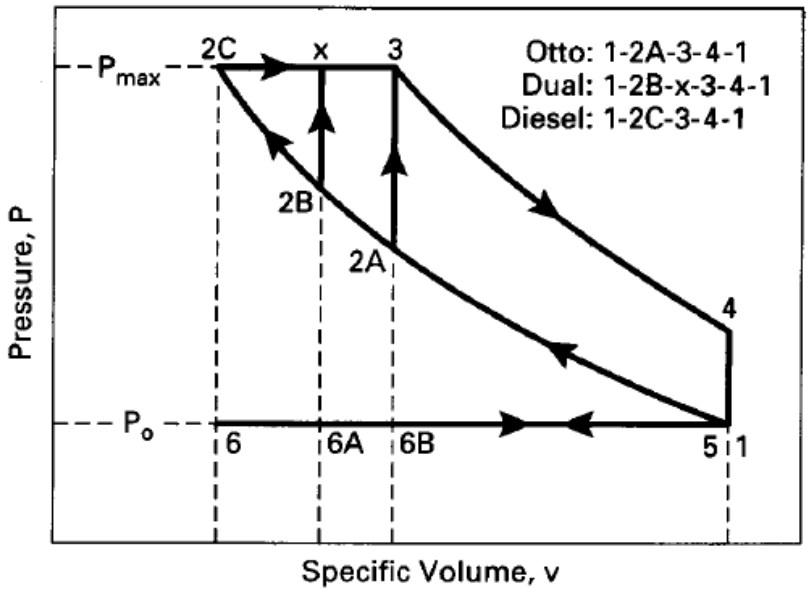


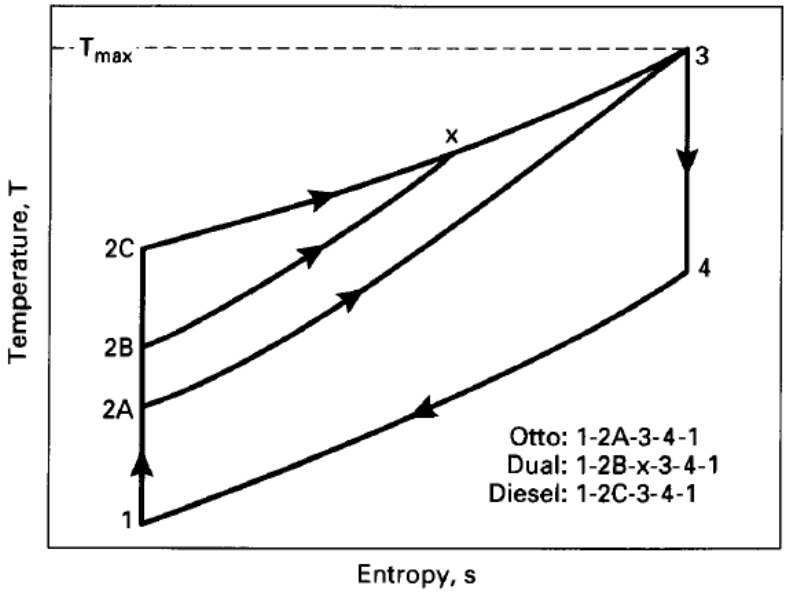


Comparison of OTTO,DIESEL, and DUAL Cycles

- For the same inlet conditions, the same peak pressure and same heat removal :

$$(\eta_t)_{DIESEL} > (\eta_t)_{DUAL} > (\eta_t)_{OTTO}$$

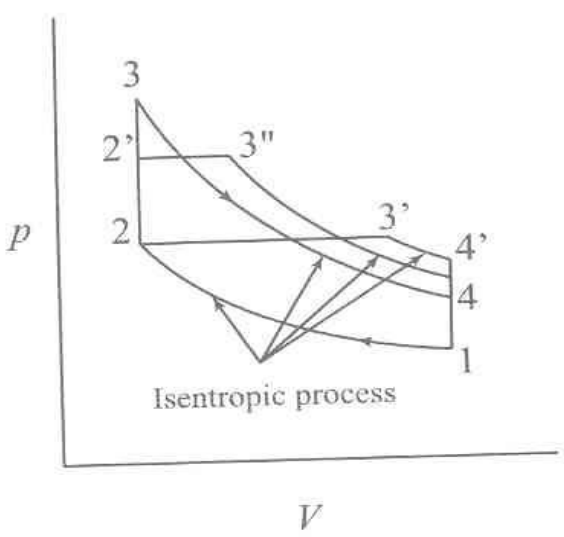


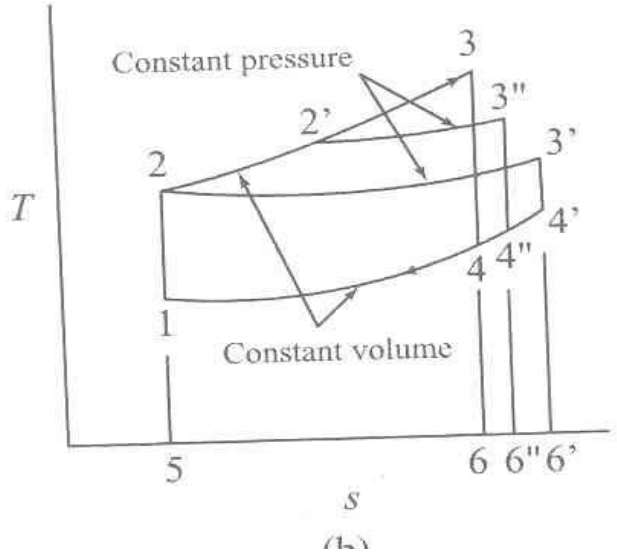


Comparison of OTTO,DIESEL, and DUAL Cycles

- For the same inlet conditions, the same compression ratios and same heat addition:

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



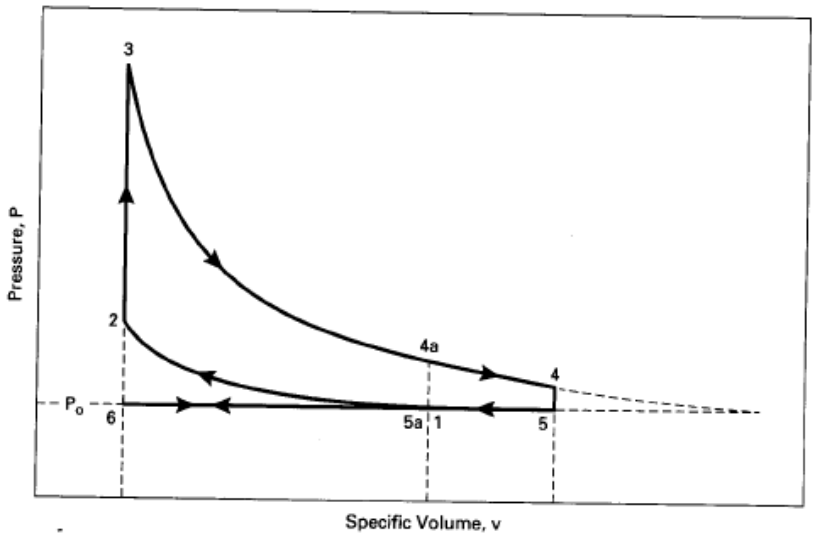


Atkinson Cycle

- Pressure in Otto and Diesel cycle is in order of three to five atmospheres when the exhaust valve opens

- If the exhaust valve is not opened until the gas in the cylinders allowed to expand down to atmospheric pressure a greater amount of work can be obtained in the expansion stroke

Air Standard Atkinson Cycle



- Such an air standard cycle is called an Atkinson Cycle or Over expanded Cycle
- Mechanical Linkage of some kind is needed to achieve this cycle

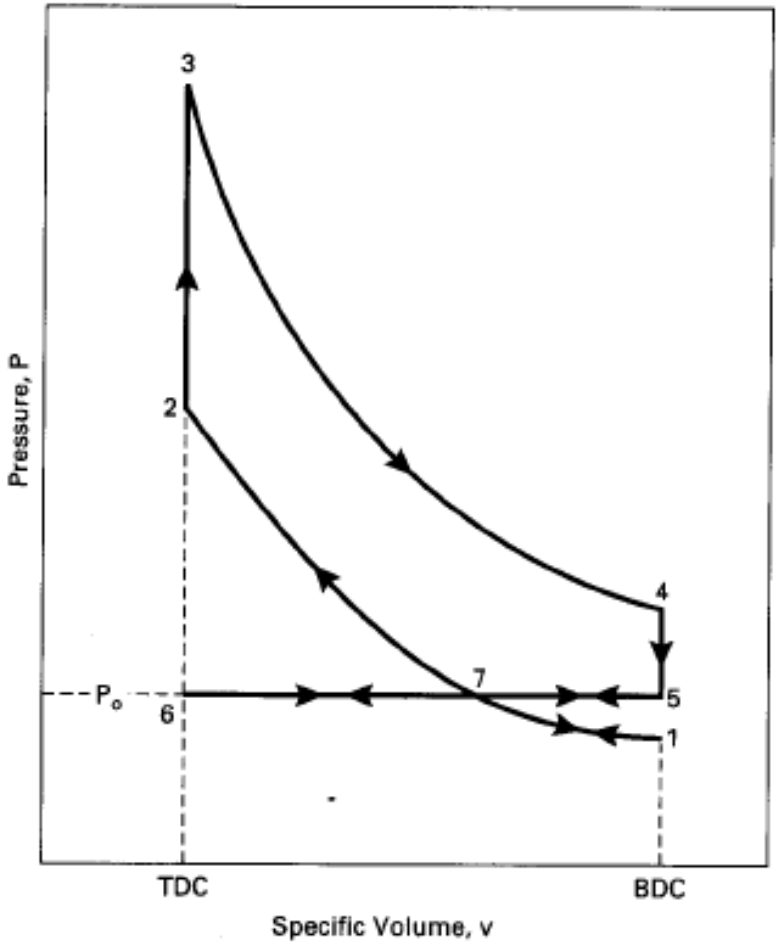
Miller Cycle

- Introduced by Ralph Miller in the 1940s
- It is a modified Otto Cycle that improves fuel efficiency by 10%-20%.
- It relies on a supercharger/turbocharger, and takes advantage of the supercharger's greater efficiency at low compression levels.
- This is applicable for the late closing cycle
- It uses unique time valve timing

Miller Cycle

- Air intake in a Miller cycle is unthrottled
- Closing the intake valve at the proper time long before BDC controls the amount of air ingested into each cylinder
- Has an expansion ratio greater than the compression ratio.

Air standard Miller Cycle



- Cylinder pressure is reduced along process 7-1
- When the piston reaches BDC and starts back towards TDC cylinder pressure increases during process 1-7
- Air-standard Miller cycle for unthrottled naturally aspirated four stroke cycle SI engine.

Air standard Miller Cycle

- If the engine has early intake valve closing, the cycle will be 6-7-1-7-2-3-4-5-7-6
- If the engine has late intake valve closing, the cycle will be 6-7- 5-7-2-3-4-5-7-6
- 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.

Air standard Miller Cycle

- The net indicated work is then the area within loop 7-2-3-4-5-7 so that there is essentially no pump work resulting in higher thermal efficiency

- The compression ratio is:

$$r_c = \frac{V_7}{V_2}$$

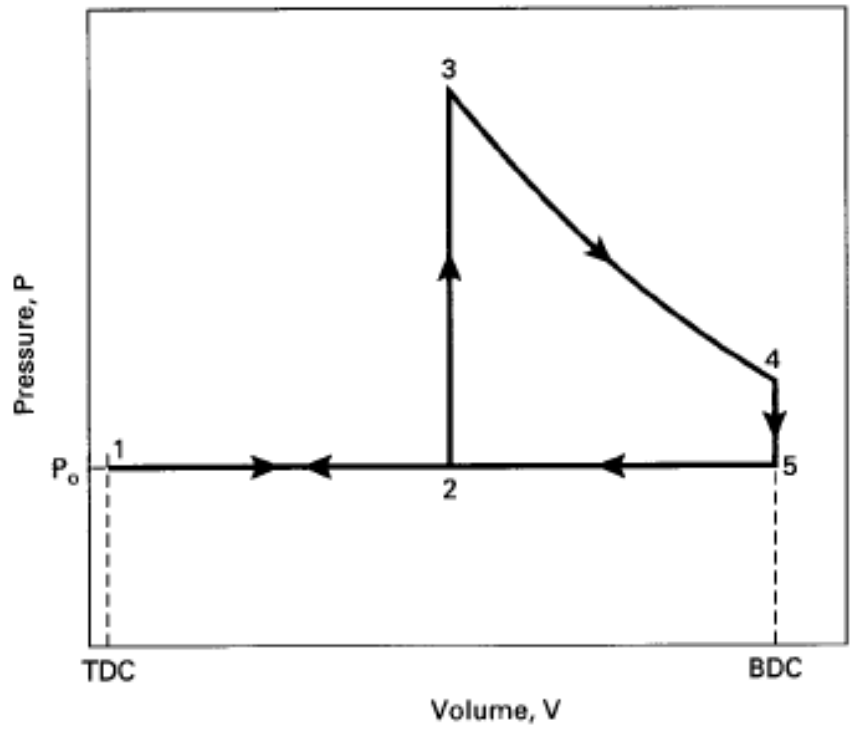
- The larger expansion ratio is:

$$r_e = \frac{V_4}{V_2} = \frac{V_4}{V_3}$$

Lenoir Cycle

- The Lenoir cycle is approximated by the air-standard cycle with two strokes
- The first half of the first stroke was intake with air-fuel entering the cylinder at atmospheric pressure (process 1-2)
- Halfway through the first stroke the intake valve was closed and the air-fuel mixture was ignited without any compression

Air standard approximation for Lenoir Engine Cycle



There was essentially no clearance volume !

Air standard approximation for Lenoir Engine Cycle

- Combustion raised the temperature and pressure in the cylinder almost at constant volume (process 2-3)
- The second half of the first stroke then became the power or expansion process 3-4.
- Near BDC, the exhaust valve opened and blowdown occurred (4-5).
- This is followed by the exhaust stroke 5-1, completing the two stroke cycle.

Thermodynamic Analysis of Air-Standard Lenoir Cycle

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

$$P_2 = P_1 = P_0$$

$$v_3 = v_2$$

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

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

Process 3-4: Isentropic power or expansion stroke.

- All valves closed:

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

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1}$$

$$P4 = P_3\left(\frac{v_3}{v_4}\right)^{k}$$

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

$$= c_v\left(T_3 - T_4\right) = \left(u_3 - u_4\right)$$

Process 4-5: Constant-volume heat rejection (exhaust blowdown)

- Exhaust valve open and intake valve closed

$$v_5 = v_4 = v_{BDC}$$
$$w_{4-5} = 0$$
$$q_{4-5} = q_{out} = c_v\left(T_5 - T_4\right) = \left(u_5 - u_4\right)$$

Process 4-5: Constant-volume heat rejection (exhaust blowdown)

$$P_5 = P_2 = P_1 = P_0$$
$$w_{5-2} = P_0\left(v_2 - v_5\right)$$
$$q_{5-2} = q_{out} = c_p\left(T_2 - T_5\right) = \left(h_2 - h_5\right)$$

Thermal Efficiency of Lenoir Cycle:

$$(\eta_t)_{LENOIR} = \frac{|w_{net}|}{|q_{in}|} = 1 - \frac{|q_{out}|}{|q_{in}|}$$

$$= 1 - \frac{\left[c_v(T_4 - T_5) + c_p(T_5 - T_2)\right]}{c_v(T_3 - T_2)}$$

$$= 1 - \frac{\left[(T_4 - T_5) + k(T_5 - T_2)\right]}{(T_3 - T_2)}$$