INTERNAL COMBUSTION ENGINES

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Lecture # 4 (Engine Cycles)

AIR-STANDARD CYCLES

Real cycle is approximated with an ideal **air-standard cycle** which 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. This is a good approximation during the first half of the cycle, when most of the gas in the cylinder is air with only up to about 7% fuel vapor. Even in the second half of the cycle, when the gas composition is mostly CO2, H20, and N2, using air properties does not create large errors 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. This works with ideal air standard cycles, as both intake gases and exhaust gases are air. Closing the cycle simplifies the analysis.

3- The combustion process is replaced with a heat addition term Q_{in} of equal energy value. Air alone cannot combust.

4- The open exhaust process, which carries a large amount of enthalpy out of the system, is replaced with a closed system heat rejection process Q_{out} *of equal energy value*

5- Actual engine processes are approximated with ideal processes

AIR-STANDARD CYCLES

In air-standard cycles, air is considered an ideal gas such that the following ideal gas relationships can be used:

$$Pv = RT \qquad \text{(a)}$$
$$PV = mRT \qquad \text{(b)}$$
$$P = \rho RT \qquad \text{(c)}$$
$$dh = c_p\, dT \qquad \text{(d)}$$

$$du = c_v\, dT \qquad \text{(e)}$$
$$Pv^k = \text{constant} \quad \text{isentropic process} \qquad \text{(f)}$$
$$Tv^{k-1} = \text{constant} \quad \text{isentropic process} \qquad \text{(g)}$$
$$TP^{(1-k)/k} = \text{constant} \quad \text{isentropic process} \qquad \text{(h)}$$
$$w_{1-2} = (P_2 v_2 - P_1 v_1)/(1 - k) \quad \text{isentropic work in closed system}$$
$$= R(T_2 - T_1)/(1 - k) \qquad \text{(i)}$$
$$c = \sqrt{kRT} \quad \text{speed of sound} \qquad \text{(j)} \qquad (1)$$

P = gas pressure in cylinder
V = volume in cylinder
v = specific volume of gas
R = gas constant of air
T = temperature
m = mass of gas in cylinder
ρ = density
h = specific enthalpy
u = specific internal energy
c_p, c_v = specific heats
$k = c_p/c_v$
w = specific work
c = speed of sound

AIR-STANDARD CYCLES

In addition to these, the following variables are used in this chapter for cycle analysis:

$$
\begin{aligned}
\mathrm{AF} &= \text{air–fuel ratio} \\
\dot{m} &= \text{mass flow rate} \\
q &= \text{heat transfer per unit mass for one cycle} \\
\dot{q} &= \text{heat transfer rate per unit mass} \\
Q &= \text{heat transfer for one cycle} \\
\dot{Q} &= \text{heat transfer rate} \\
Q_{\mathrm{HV}} &= \text{heating value of fuel} \\
r_c &= \text{compression ratio} \\
W &= \text{work for one cycle} \\
\dot{W} &= \text{power} \\
\eta_c &= \text{combustion efficiency}
\end{aligned}
$$

Subscripts used include the following:

$$
\begin{aligned}
a &= \text{air} \\
f &= \text{fuel} \\
\mathrm{ex} &= \text{exhaust} \\
m &= \text{mixture of all gases}
\end{aligned}
$$

AIR-STANDARD CYCLES

When analyzing what occurs within engines during the operating cycle and exhaust flow, this text uses the following air property values:

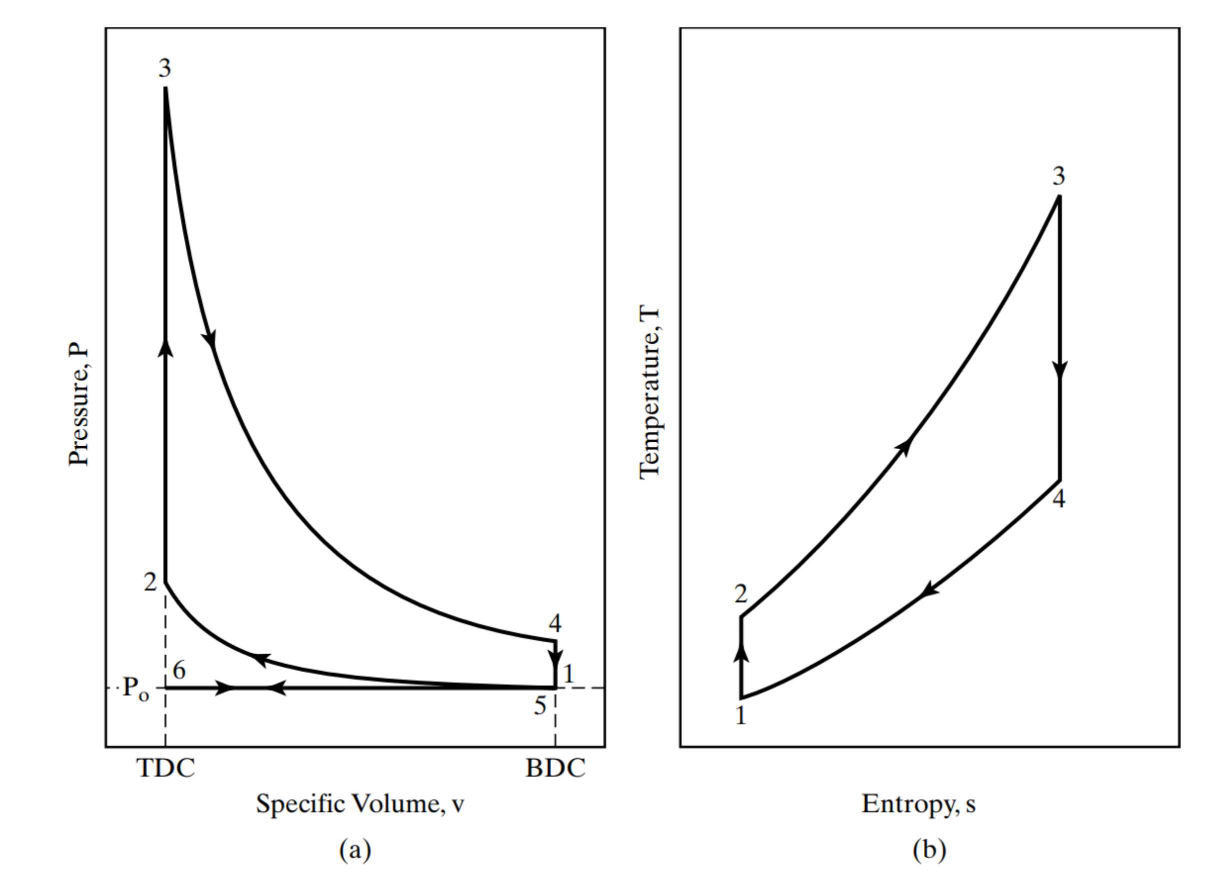
$$
\begin{aligned}
c_p &= 1.108 \text{ kJ/kg-K} = 0.265 \text{ BTU/lbm-°R} \\
c_v &= 0.821 \text{ kJ/kg-K} = 0.196 \text{ BTU/lbm-°R} \\
k &= c_p/c_v = 1.108/0.821 = 1.35 \\
R &= c_p - c_v = 0.287 \text{ kJ/kg-K} \\
&= 0.069 \text{ BTU/lbm-°R} = 53.33 \text{ ft-lbf/lbm-°R}
\end{aligned}
$$

Air flow before it enters an engine is usually closer to standard temperature, and for these conditions a value of $k = 1.4$ is correct. This would include processes such as inlet flow in superchargers, turbochargers, and carburetors, and air flow through the engine radiator. For these conditions, the following air property values are used:

$$
\begin{aligned}
c_p &= 1.005 \text{ kJ/kg-K} = 0.240 \text{ BTU/lbm-°R} \\
c_v &= 0.718 \text{ kJ/kg-K} = 0.172 \text{ BTU/lbm-°R} \\
k &= c_p/c_v = 1.005/0.718 = 1.40 \\
R &= c_p - c_v = 0.287 \text{ kJ/kg-K}
\end{aligned}
$$

OTTO CYCLE

- The Otto cycle is the air-standard model of most four-stroke SI engines of the last 140 years, including many of today's automobile engines.



It is not uncommon to find the Otto cycle shown with processes 6–1 and 5–6 left off the figure. The reasoning to justify this is that these two processes cancel each other thermodynamically and are not needed in analyzing the cycle.

OTTO CYCLE

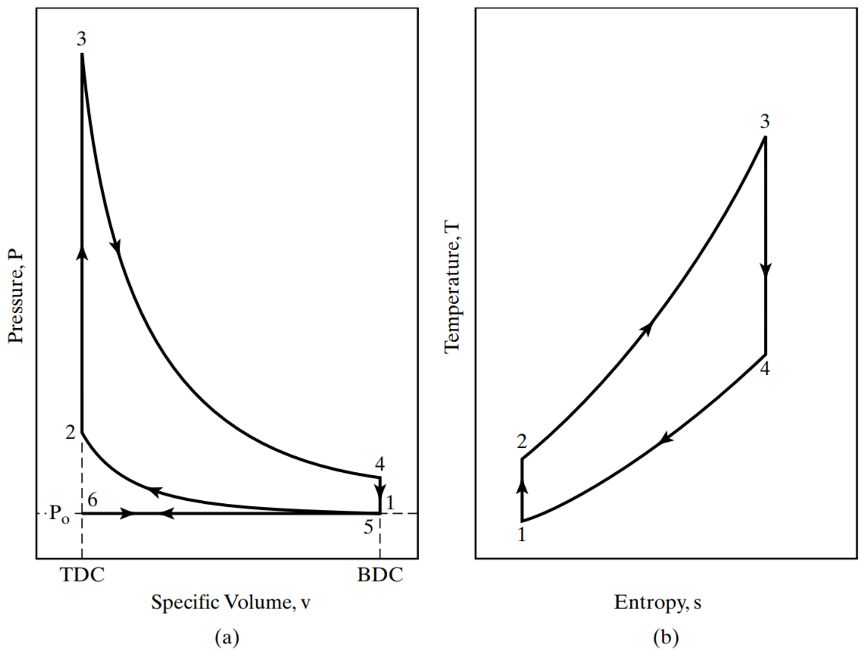
Thermodynamic Analysis of Air-Standard Otto Cycle at WOT

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

$$P_1 = P_6 = P_o \tag{2}$$
$$w_{6-1} = P_o(v_1 - v_6) \tag{3}$$

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

$$T_2 = T_1(v_1/v_2)^{k-1} = T_1(V_1/V_2)^{k-1} = T_1(r_c)^{k-1} \tag{4}$$
$$P_2 = P_1(v_1/v_2)^k = P_1(V_1/V_2)^k = P_1(r_c)^k \tag{5}$$
$$q_{1-2} = 0 \tag{6}$$



$$\begin{aligned} w_{1-2} &= (P_2v_2 - P_1v_1)/(1-k) = R(T_2 - T_1)/(1-k) \\ &= (u_1 - u_2) = c_v(T_1 - T_2) \end{aligned} \tag{7}$$

OTTO CYCLE

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

$$v_3 = v_2 = v_{\mathrm{TDC}} \tag{8}$$

$$w_{2-3} = 0 \tag{9}$$

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

$$= (m_a + m_f) c_v (T_3 - T_2) \tag{10}$$

$$Q_{\mathrm{HV}} \eta_c = (\mathrm{AF} + 1) c_v (T_3 - T_2) \tag{11}$$

$$q_{2-3} = q_{\mathrm{in}} = c_v (T_3 - T_2) = (u_3 - u_2) \tag{12}$$

$$T_3 = T_{\max} \tag{13}$$

$$P_3 = P_{\max} \tag{14}$$

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

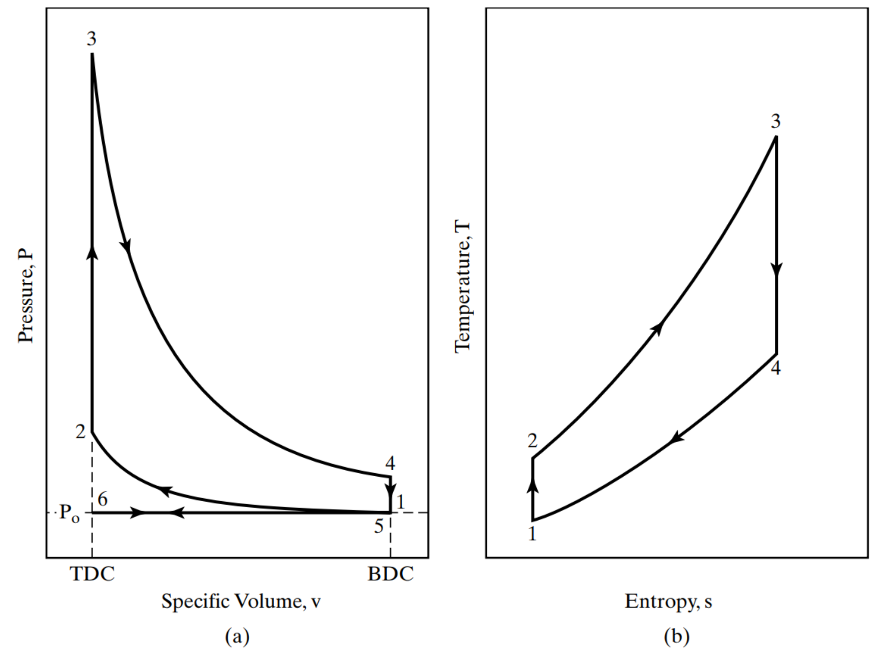
$$q_{3-4} = 0 \tag{15}$$

$$T_4 = T_3 (v_3/v_4)^{k-1} = T_3 (V_3/V_4)^{k-1} = T_3 (1/r_c)^{k-1} \tag{16}$$

$$P_4 = P_3 (v_3/v_4)^k = P_3 (V_3/V_4)^k = P_3 (1/r_c)^k \tag{17}$$

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

$$= (u_3 - u_4) = c_v (T_3 - T_4) \tag{18}$$



OTTO CYCLE

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

$$v_5 = v_4 = v_1 = v_{\mathrm{BDC}} \tag{19}$$

$$w_{4-5} = 0 \tag{20}$$

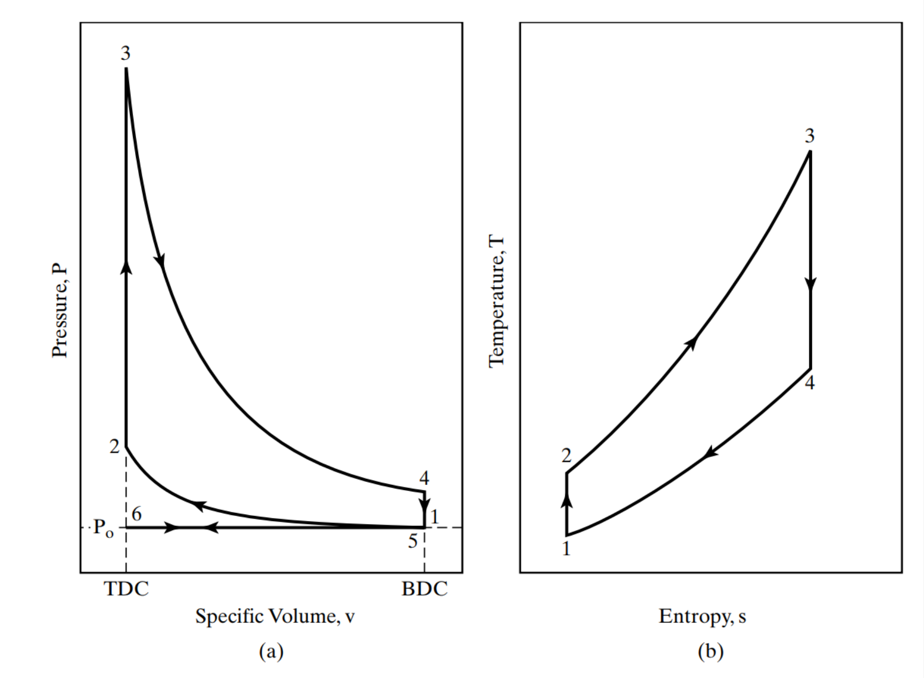
$$Q_{4-5} = Q_{\mathrm{out}} = m_m c_v (T_5 - T_4) = m_m c_v (T_1 - T_4) \tag{21}$$

$$q_{4-5} = q_{\mathrm{out}} = c_v (T_5 - T_4) = (u_5 - u_4) = c_v (T_1 - T_4) \tag{22}$$

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

$$P_5 = P_6 = P_o \tag{23}$$

$$w_{5-6} = P_o (v_6 - v_5) = P_o (v_6 - v_1) \tag{24}$$



OTTO CYCLE

Thermal efficiency of Otto cycle:

$$
\begin{aligned}
(\eta_t)_{\mathrm{OTTO}} &= |w_{\mathrm{net}}|/|q_{\mathrm{in}}| = 1 - (|q_{\mathrm{out}}|/|q_{\mathrm{in}}|) \\
&= 1 - [c_v(T_4 - T_1)/c_v(T_3 - T_2)] \\
&= 1 - [(T_4 - T_1)/(T_3 - T_2)]
\end{aligned}
\tag{25}
$$

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 $v_2 = v_3$:

$$(T_2/T_1) = (v_1/v_2)^{k-1} = (v_4/v_3)^{k-1} = (T_3/T_4) \tag{26}$$

Rearranging the temperature terms gives

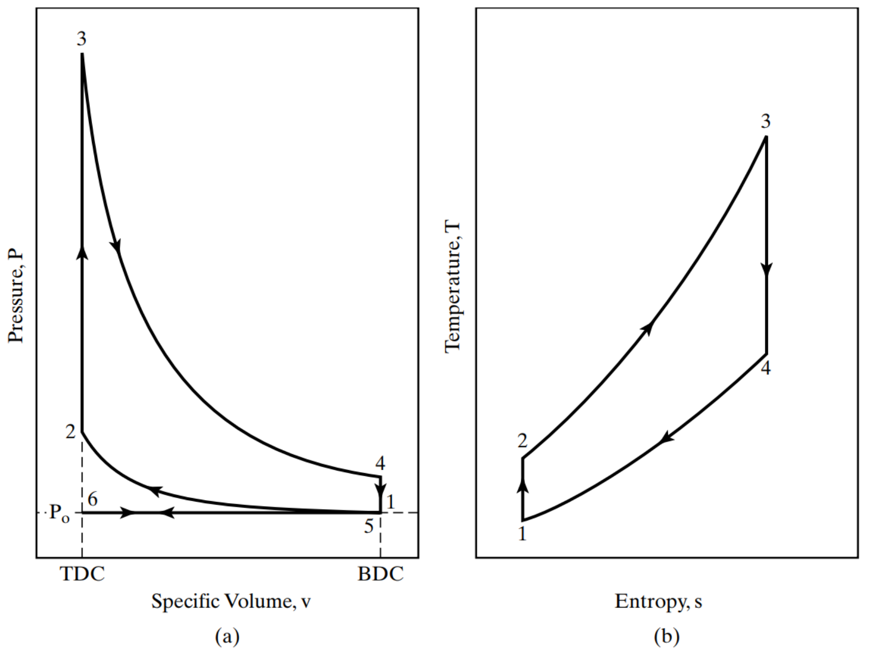
$$T_4/T_1 = T_3/T_2 \tag{27}$$

Equation (25) can be rearranged to

$$(\eta_t)_{\mathrm{OTTO}} = 1 - (T_1/T_2)\{[(T_4/T_1) - 1]/[(T_3/T_2) - 1]\} \tag{28}$$

Using Eq. (27) gives

$$(\eta_t)_{\mathrm{OTTO}} = 1 - (T_1/T_2) \tag{29}$$



OTTO CYCLE

Equation (25) can be rearranged to

$$(\eta_t)_{\text{OTTO}} = 1 - (T_1/T_2)\{[(T_4/T_1) - 1]/[(T_3/T_2) - 1]\} \tag{28}$$

Using Eq. (27) gives

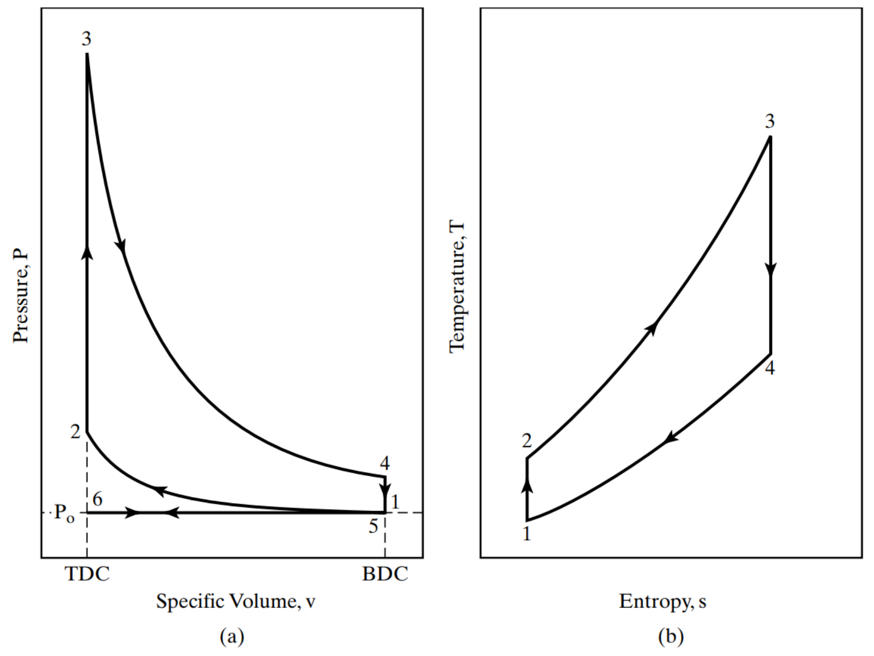
$$(\eta_t)_{\text{OTTO}} = 1 - (T_1/T_2) \tag{29}$$

Combining this with Eq. (4)

$$(\eta_t)_{\text{OTTO}} = 1 - [1/(v_1/v_2)^{k-1}] \tag{30}$$

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

$$(\eta_t)_{\text{OTTO}} = 1 - (1/r_c)^{k-1} \tag{31}$$



Only the compression ratio is needed to determine the thermal efficiency of the Otto cycle at WOT

OTTO CYCLE

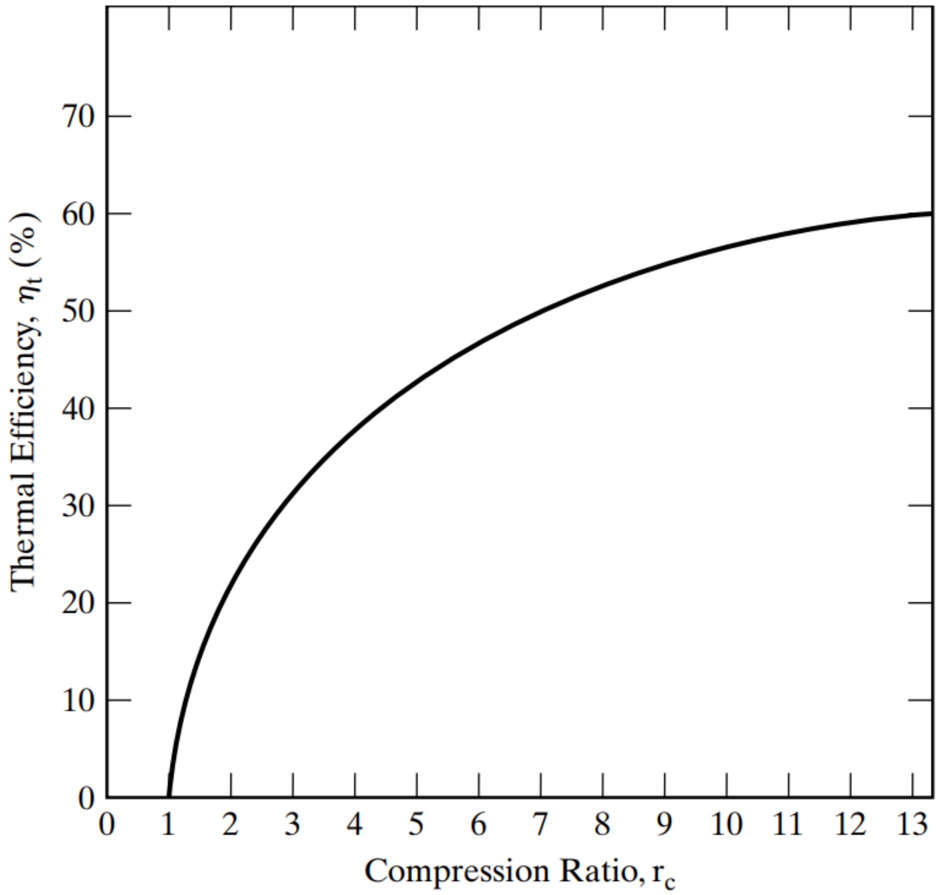


FIGURE 3

Indicated thermal efficiency as a function of compression ratio for SI engines operating at WOT on air-standard Otto cycle ($k = 1.35$).

REAL AIR-FUEL ENGINE CYCLES

Major differences between the standard cycles and Real cycles include:

1- Real engines operate on an open cycle with changing composition. Not only does the inlet gas composition differ from what exits, but often the mass flow rate is not the same.

2- Air-standard analysis treats the fluid flow through the entire engine as **air** and approximates air as an ideal gas. In a real engine inlet flow may be all air, or it may be air mixed with up to 7% fuel, either gaseous or as liquid droplets, or both.

3- There are **heat losses** during the cycle of a real engine which are neglected in air-standard analysis.

4- Combustion requires a short but finite time to occur, and heat addition is not instantaneous at TDC, as approximated in an Otto cycle.

5- The blowdown process requires a finite real time and a finite cycle time and does not occur at constant volume as in air-standard analysis.

REAL AIR-FUEL ENGINE CYCLES

Major differences between the standard cycles and Real cycles include:

7- Engine **valves require a finite time** to actuate. Ideally, valves would open and close instantaneously, but this is not possible when using a camshaft.

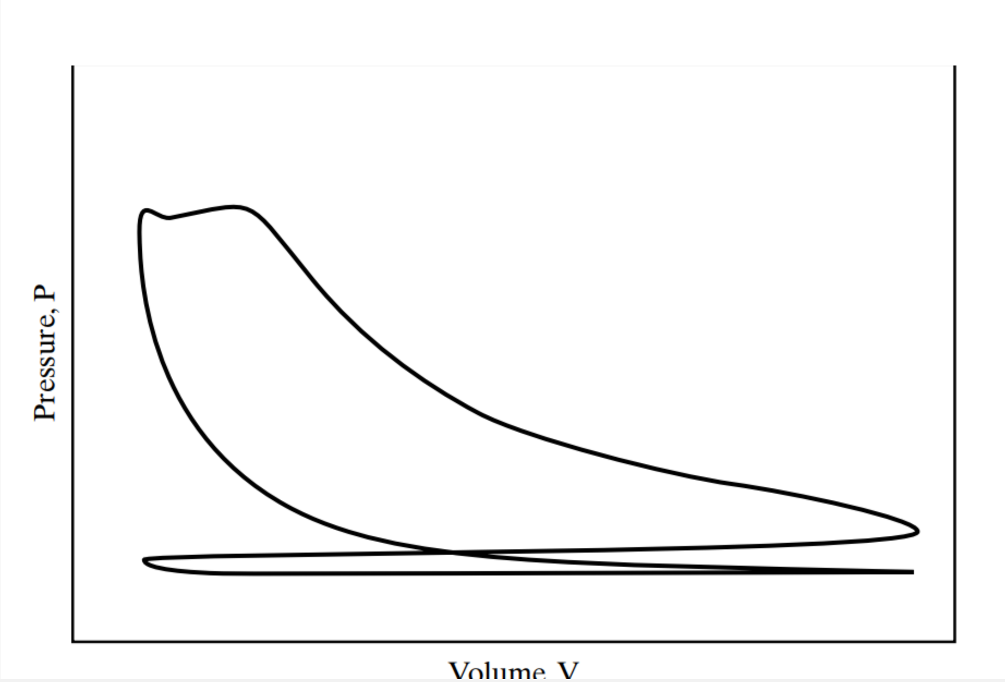
Indicated thermal efficiency of a real four-stroke SI engine is always somewhat less than what air-standard Otto cycle analysis predicts. This is caused by the heat losses,. friction, ignition timing, valve timing, finite time of combustion and blowdown, and deviation from ideal gas behavior of the real engine.

the indicated thermal efficiency of an actual SI four-stroke cycle engine can be approximated by;

$$(\eta_t)_{\text{actual}} \approx 0.85(\eta_t)_{\text{OTTO}}$$

DIESEL CYCLE

1. Early CI engines injected fuel into the combustion chamber very late in the compression stroke, resulting in the indicator diagram shown in Fig.
2. Due to ignition delay and the finite time required to inject the fuel, combustion lasted into the expansion stroke. This kept the pressure at peak levels well past TDC.
3. This combustion process is best approximated as a constant-pressure heat input in an air-standard cycle, resulting
   in the **Diesel cycle**.
4. The rest of the cycle is similar to the air-standard Otto cycle.
5. The Diesel cycle is sometimes called a **Constant-Pressure cycle**.



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:

$$w_{6-1} = P_o(v_1 - v_6) \quad (51)$$

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

$$T_2 = T_1(v_1/v_2)^{k-1} = T_1(V_1/V_2)^{k-1} = T_1(r_c)^{k-1} \quad (52)$$
$$P_2 = P_1(v_1/v_2)^k = P_1(V_1/V_2)^k = P_1(r_c)^k \quad (53)$$
$$V_2 = V_{TDC} \quad (54)$$
$$q_{1-2} = 0 \quad (55)$$
$$w_{1-2} = (P_2v_2 - P_1v_1)/(1-k) = R(T_2 - T_1)/(1-k)$$
$$= (u_1 - u_2) = c_v(T_1 - T_2) \quad (56)$$

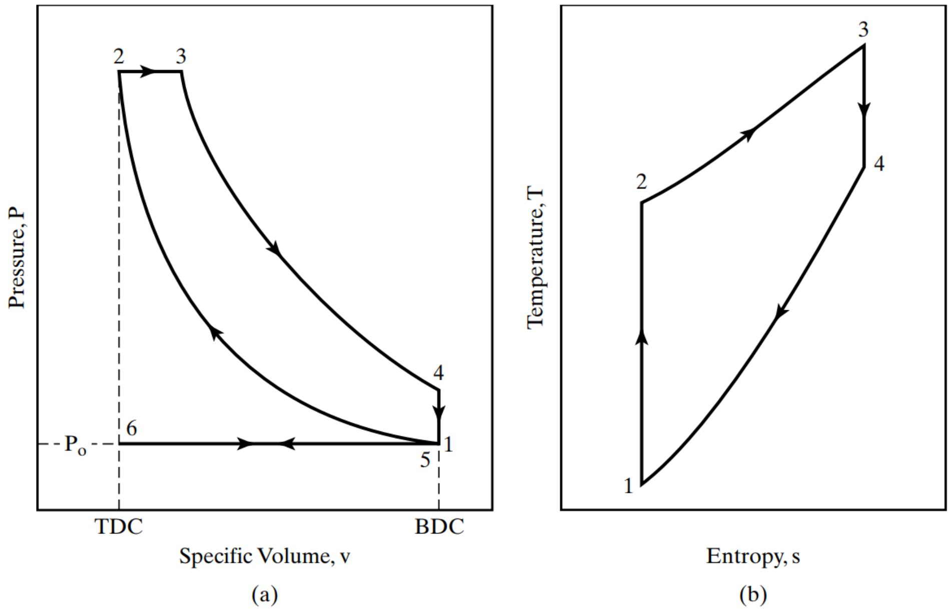


FIGURE 8

Air-standard Diesel cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an early CI engine on **(a)** pressure-specific volume coordinates, and **(b)** temperature-entropy coordinates.

DIESEL CYCLE

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

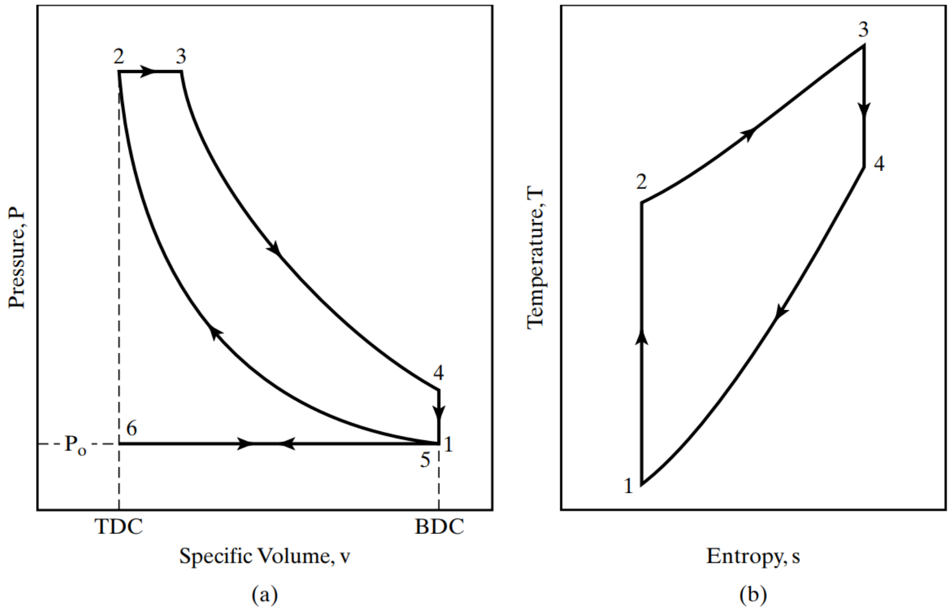
$$Q_{2-3} = Q_{in} = m_f Q_{HV} \eta_c = m_m c_p(T_3 - T_2) = (m_a + m_f)c_p(T_3 - T_2) \quad (57)$$
$$Q_{HV}\eta_c = (AF + 1)c_p(T_3 - T_2) \quad (58)$$
$$q_{2-3} = q_{in} = c_p(T_3 - T_2) = (h_3 - h_2) \quad (59)$$
$$w_{2-3} = q_{2-3} - (u_3 - u_2) = P_2(v_3 - v_2) \quad (60)$$
$$T_3 = T_{max} \quad (61)$$

Cutoff ratio is defined as the change in volume that occurs during combustion, given as a ratio:

$$\beta = V_3/V_2 = v_3/v_2 = T_3/T_2 \quad (62)$$

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

$$q_{3-4} = 0 \quad (63)$$
$$T_4 = T_3(v_3/v_4)^{k-1} = T_3(V_3/V_4)^{k-1} \quad (64)$$
$$P_4 = P_3(v_3/v_4)^k = P_3(V_3/V_4)^k \quad (65)$$
$$w_{3-4} = (P_4v_4 - P_3v_3)/(1 - k) = R(T_4 - T_3)/(1 - k)$$
$$= (u_3 - u_4) = c_v(T_3 - T_4) \quad (66)$$



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DIESEL CYCLE

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

$$v_5 = v_4 = v_1 = v_{\mathrm{BDC}} \tag{67}$$

$$w_{4-5} = 0 \tag{68}$$

$$Q_{4-5} = Q_{\mathrm{out}} = m_m c_v(T_5 - T_4) = m_m c_v(T_1 - T_4) \tag{69}$$

$$q_{4-5} = q_{\mathrm{out}} = c_v(T_5 - T_4) = (u_5 - u_4) = c_v(T_1 - T_4) \tag{70}$$

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

$$w_{5-6} = P_o(v_6 - v_5) = P_o(v_6 - v_1) \tag{71}$$

Thermal efficiency of Diesel cycle

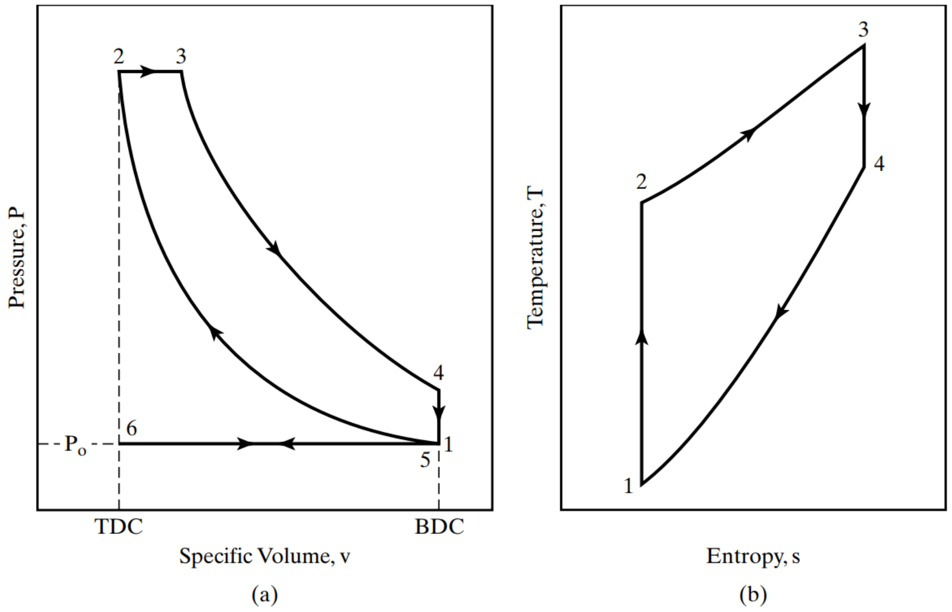
$$\begin{aligned}(\eta_t)_{\mathrm{DIESEL}} &= |w_{\mathrm{net}}|/|q_{\mathrm{in}}| = 1 - (|q_{\mathrm{out}}|/|q_{\mathrm{in}}|) \\ &= 1 - [c_v(T_4 - T_1)/c_p(T_3 - T_2)] \\ &= 1 - (T_4 - T_1)/[k(T_3 - T_2)]\end{aligned} \tag{72}$$

With rearrangement, this can be shown to equal

$$(\eta_t)_{\mathrm{DIESEL}} = 1 - (1/r_c)^{k-1}[(\beta^k - 1)/\{k(\beta - 1)\}] \tag{73}$$

where

$$\begin{aligned}r_c &= \text{compression ratio} \\ k &= c_p/c_v \\ \beta &= \text{cutoff ratio}\end{aligned}$$



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DIESEL CYCLE

$$(\eta_t)_{\text{DIESEL}} = 1 - (1/r_c)^{k-1}[(\beta^k - 1)/\{k(\beta - 1)\}] \qquad (73)$$

where

$$r_c = \text{compression ratio}$$
$$k = c_p/c_v$$
$$\beta = \text{cutoff ratio}$$

If representative numbers are introduced into Eq. (73), it is found that the value of the term in brackets is greater than one. When this equation is compared with Eq. (31), it can be seen that for a given compression ratio the thermal efficiency of the Otto cycle would be greater than the thermal efficiency of the Diesel cycle. Constant-volume combustion at TDC is more efficient than constant-pressure combustion. However, it must be remembered that CI engines operate with much higher compression ratios than SI engines (12 to 24 versus 8 to 11) and thus have higher thermal efficiencies.

DIESEL CYCLE

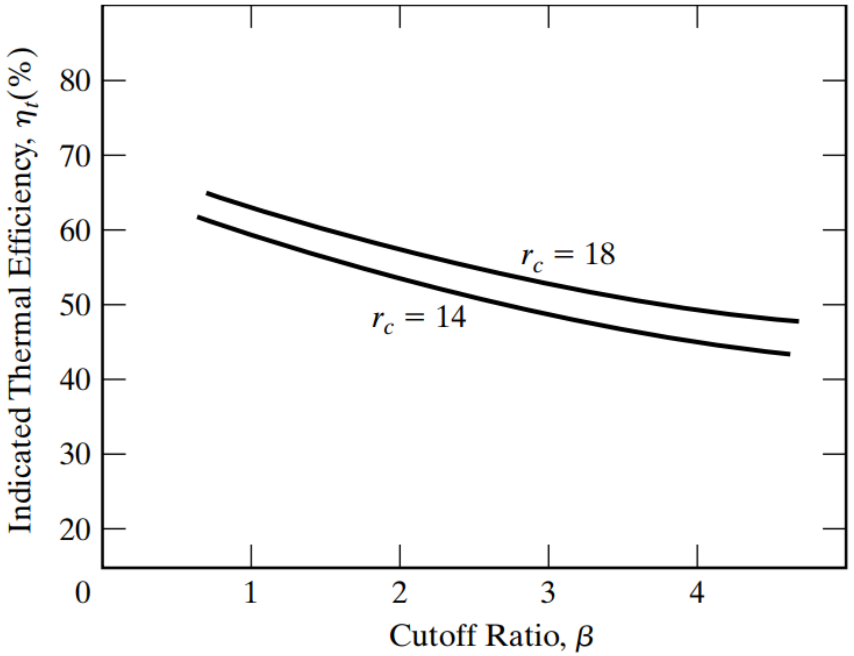

FIGURE 9

Indicated thermal efficiency as a function of cutoff ratio for air-standard Diesel cycle (k = 1.35).

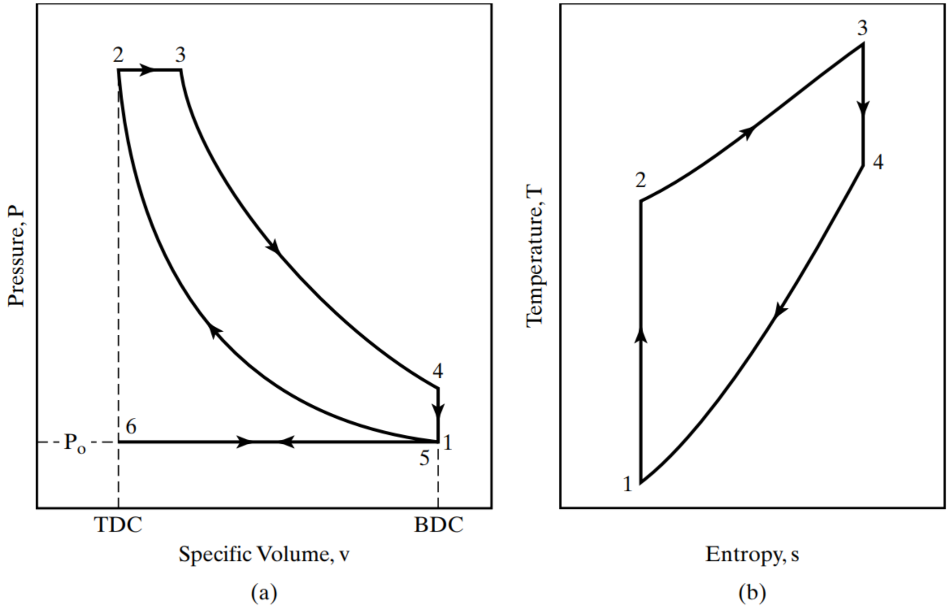

FIGURE 8

Air-standard Diesel cycle, 6-1-2-3-4-5-6, which approximates the four-stroke cycle of an early CI engine on **(a)** pressure-specific volume coordinates, and **(b)** temperature-entropy coordinates.

Cutoff ratio is defined as the change in volume that occurs during combustion, given as a ratio:

$$\beta = V_3/V_2 = v_3/v_2 = T_3/T_2 \tag{62}$$

END OF THE LECTURE