ME 430 Internal Combustion Engines

# Engineering Fundamentals of Internal Combustion Engines

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# ME 430 Internal Combustion Engines

## Lecture Notes for the Undergraduate Course

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# Chapter 3
# Air-Standart Ideal Engine Cycles Otto/Diesel/Dual

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# BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

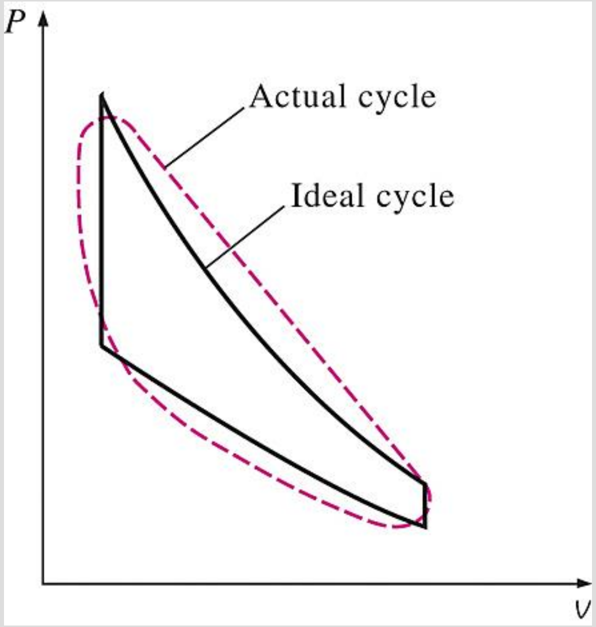
Thermal efficiency of heat engines

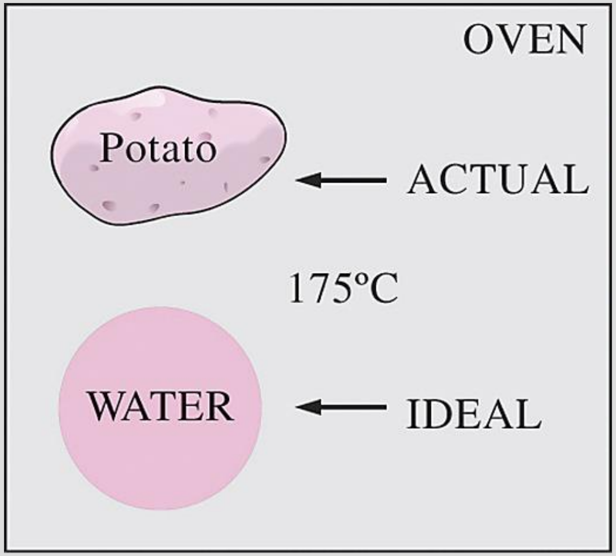
$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}}$$

*Most power-producing devices operate on cycles.*

*Ideal cycle: A cycle that **resembles the actual cycle** closely but is made up **totally of internally reversible** processes is called an Ideal Cycle.*

*Reversible cycles such as Carnot cycle **have the highest thermal efficiency** of all heat engines operating between the same temperature levels. Unlike ideal cycles, they are unsuitable as a realistic model.*




*Modeling is a powerful engineering tool that provides great insight and simplicity at the expense of some loss in accuracy.*

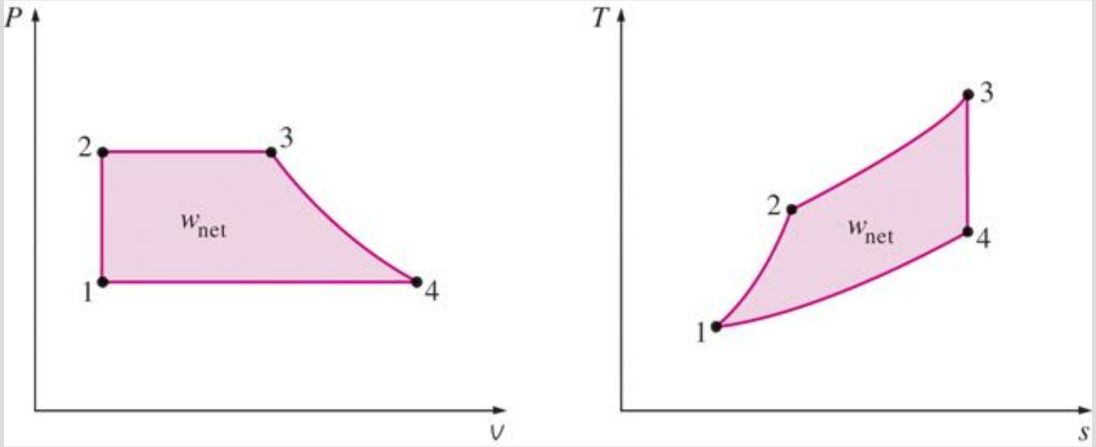
*The analysis of many complex processes can be reduced to a manageable level by utilizing some idealizations*

On a *T*-*s* diagram, the ratio of the area enclosed by the cyclic curve to the area under the heat-addition process curve represents the thermal efficiency of the cycle. Any modification that increases the ratio of these two areas will also increase the thermal efficiency of the cycle.


Care should be exercised in the interpretation of the results from ideal cycles

**The idealizations and simplifications in the analysis of power cycles:**

1. *The cycle does not involve any friction. Therefore, the working fluid does not experience any pressure drop as it flows in pipes or devices such as heat exchangers.*
2. *All expansion and compression processes take place in a quasi-equilibrium manner.*
3. *The pipes connecting the various components of a system are well insulated, and heat transfer through them is negligible.*

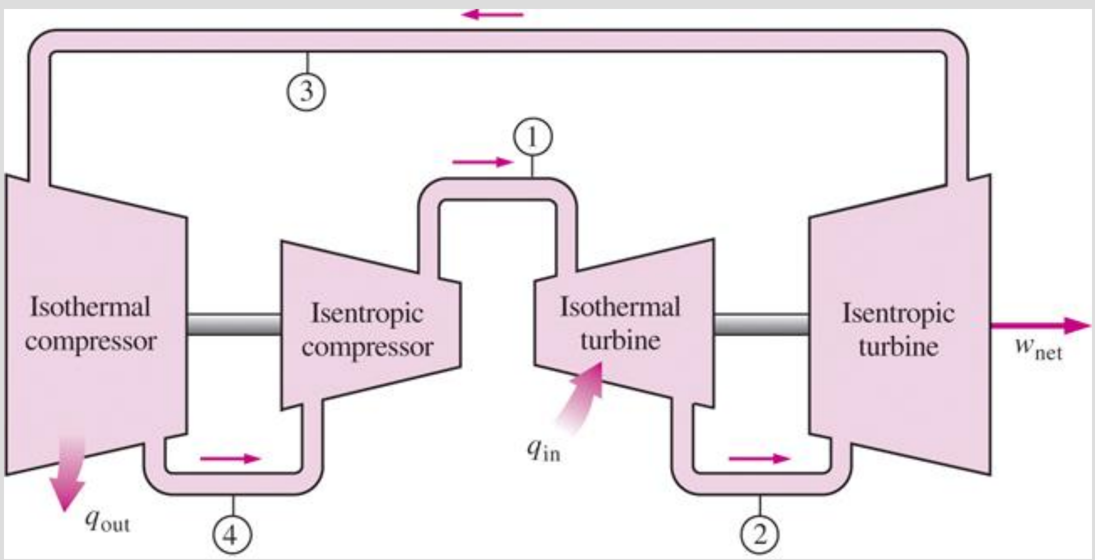



On both *P*-*v* and *T*-*s* diagrams, the area enclosed by the process curve represents the net work of the cycle

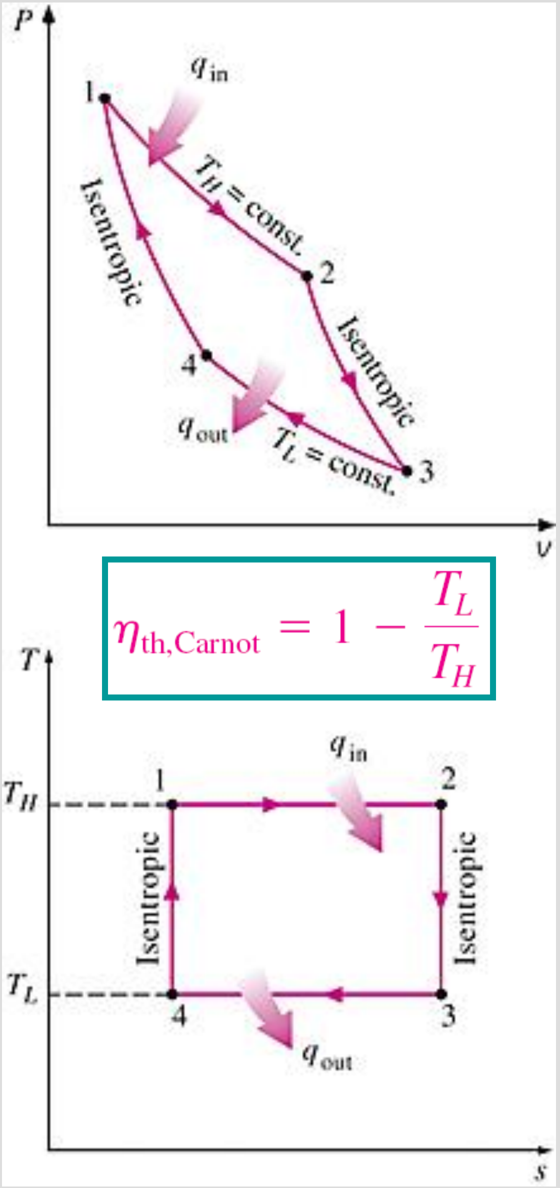
# THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING

The Carnot cycle is composed of four totally reversible processes: isothermal heat addition, isentropic expansion, isothermal heat rejection, and isentropic compression.

***For both ideal and actual cycles:*** *Thermal efficiency increases with an increase in the average temperature at which heat is supplied to the system or with a decrease in the average temperature at which heat is rejected from the system.*



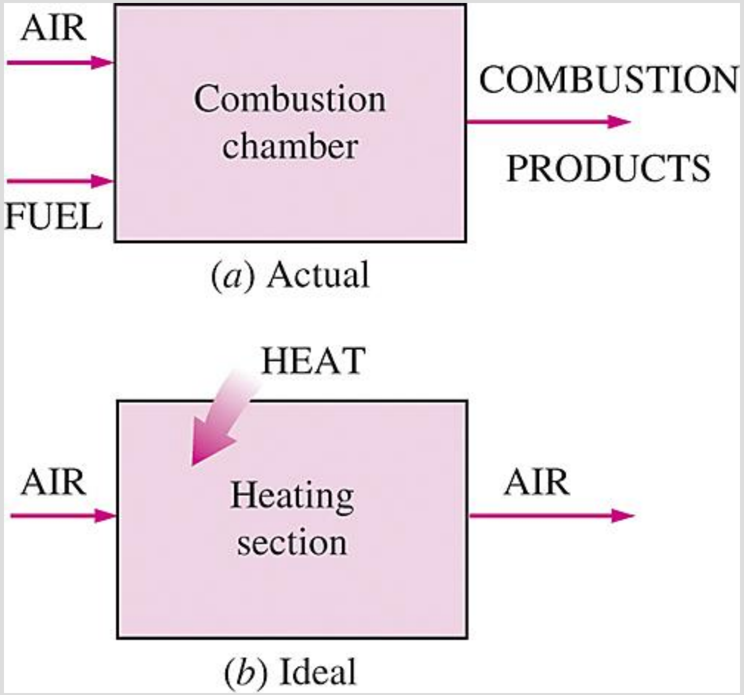

A steady-flow Carnot engine




$$\eta_{th,Carnot} = 1 - \frac{T_L}{T_H}$$

*P-v* and *T-s* diagrams of a Carnot cycle

# AIR-STANDARD ASSUMPTIONS




The combustion process is replaced by a heat-addition process in ideal cycles.

**Air-standard assumptions**:

1. *The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.*
2. *All the processes that make up the cycle are internally reversible.*
3. *The combustion process is replaced by a heat-addition process from an external source.*
4. *The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.*

**Cold-air-standard assumptions**: When the working fluid is considered to be air with constant specific heats at room temperature (25°C).

**Air-standard cycle:** A cycle for which the air-standard assumptions are applicable.

# AN OVERVIEW OF RECIPROCATING ENGINES

Compression ratio

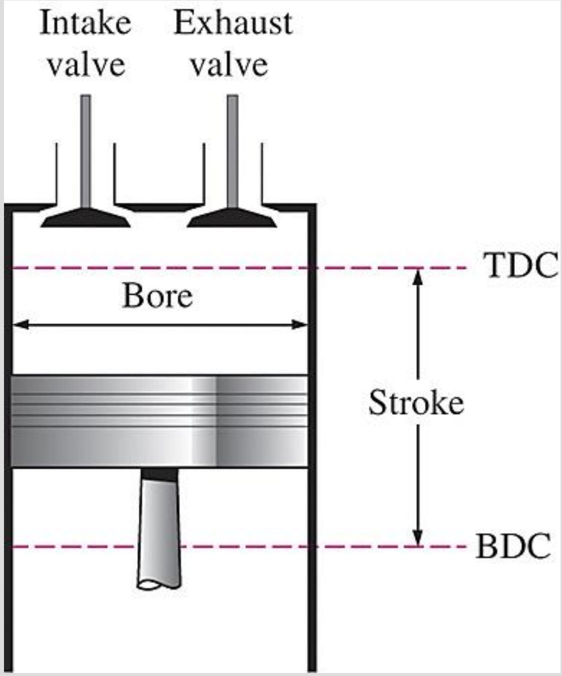
$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}}$$

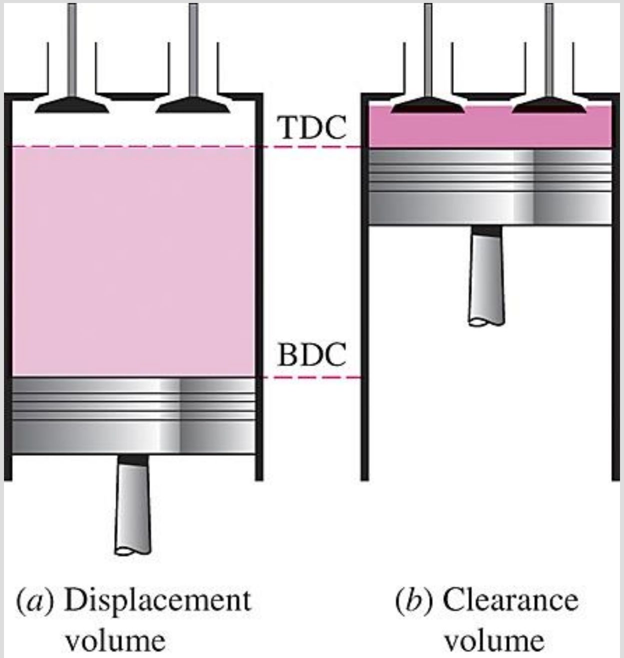
$$W_{net} = \text{MEP} \times \text{Piston area} \times \text{Stroke} = \text{MEP} \times \text{Displacement volume}$$

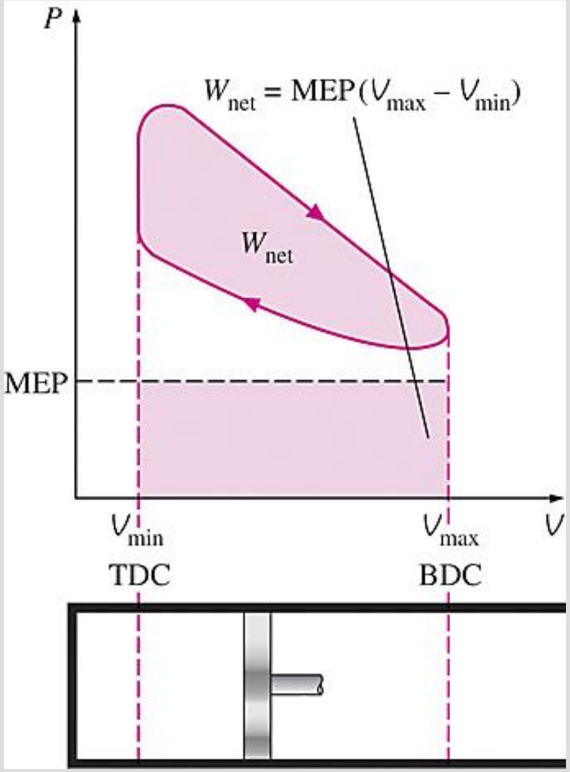
Mean effective pressure

$$\text{MEP} = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}} \quad \text{(kPa)}$$

- **Spark-ignition (SI) engines**
- **Compression-ignition (CI) engines**

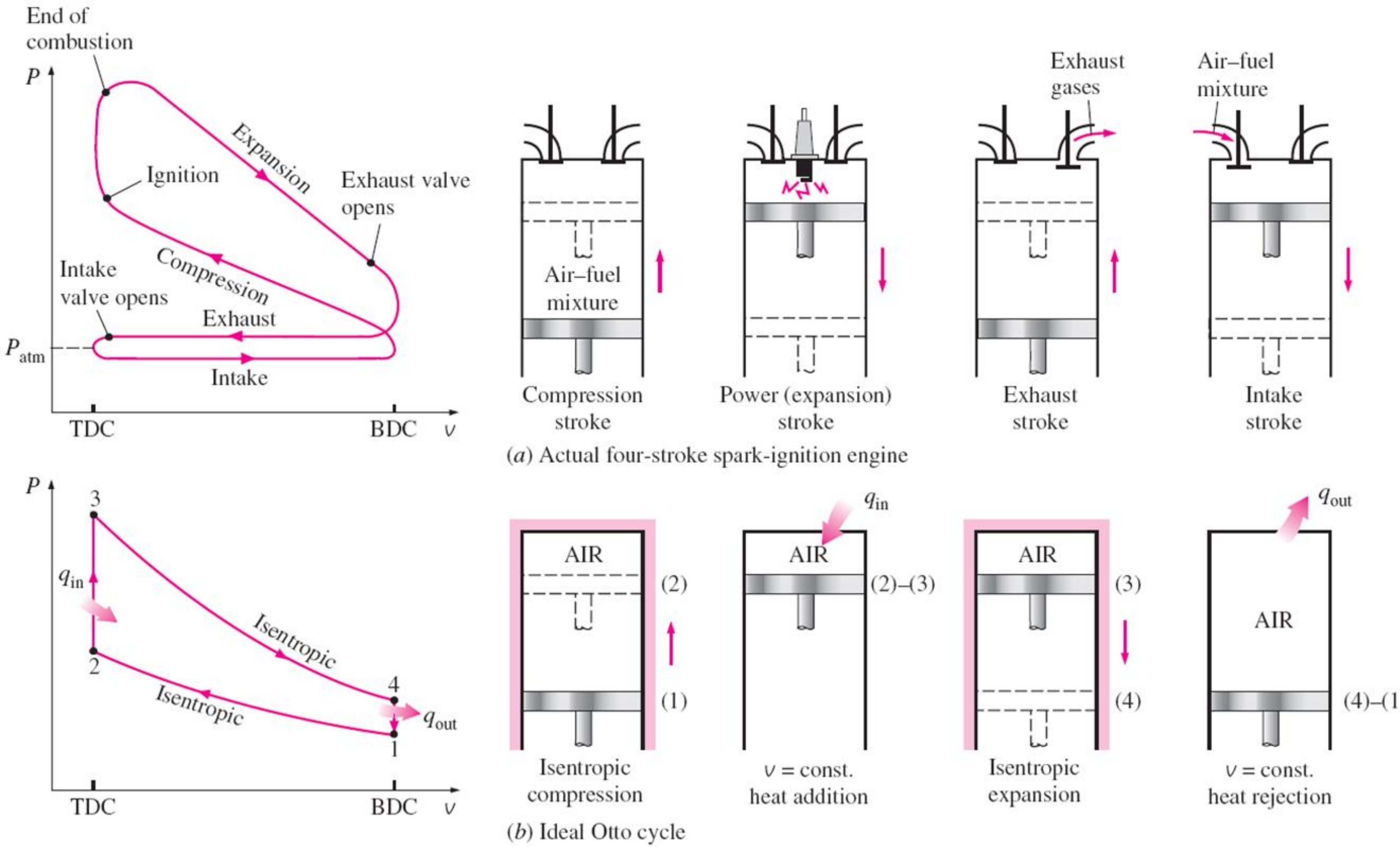



Nomenclature for reciprocating engines.

# OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES




*Actual and ideal cycles in spark-ignition engines and their P-v diagrams.*

# OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

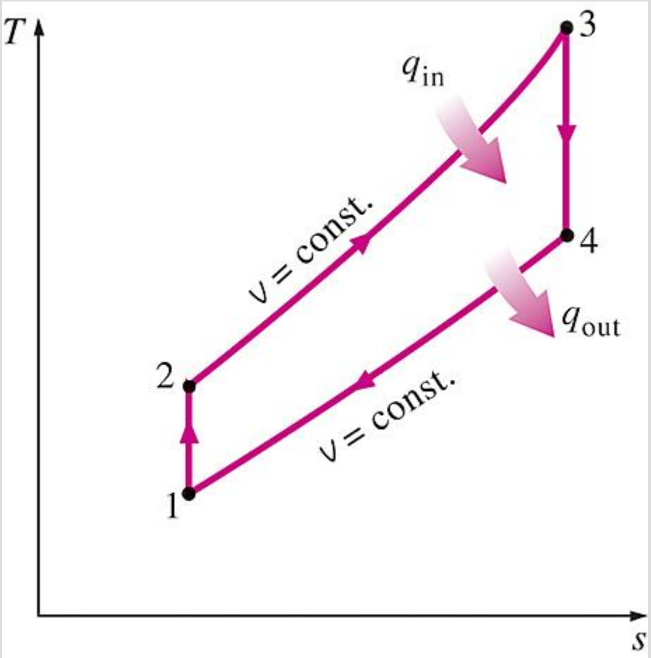
**Four-stroke cycle**

1 cycle = 4 stroke = 2 revolution

**Two-stroke cycle**

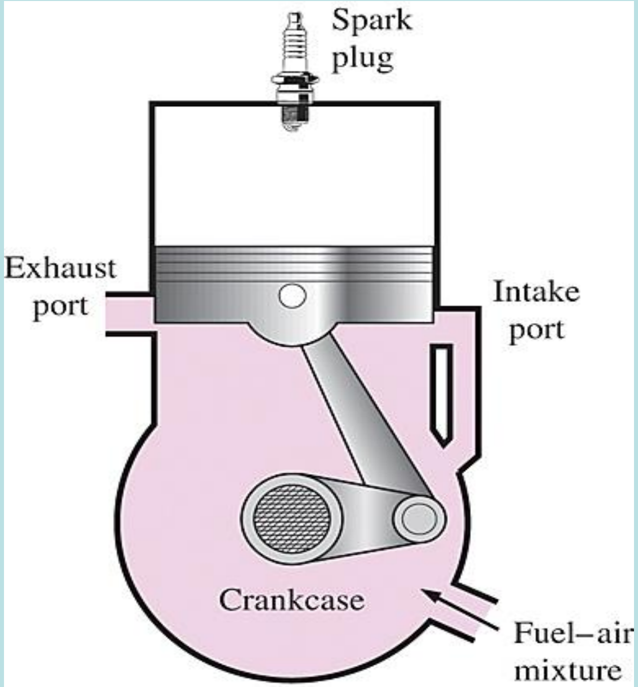
1 cycle = 2 stroke = 1 revolution

| | |
|---|---|
| 1-2 | Isentropic compression |
| 2-3 | Constant-volume heat addition |
| 3-4 | Isentropic expansion |
| 4-1 | Constant-volume heat rejection |



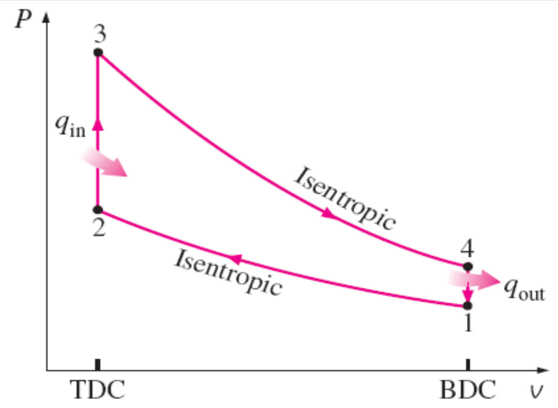

*T-s* diagram of the ideal Otto cycle.

The two-stroke engines are generally less efficient than their four-stroke counterparts but they are relatively simple and inexpensive, and they have **high power-to-weight** and **power-to-volume ratios.**




Schematic of a two-stroke reciprocating engine.

# OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES




$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{exit} - h_{inlet}$$

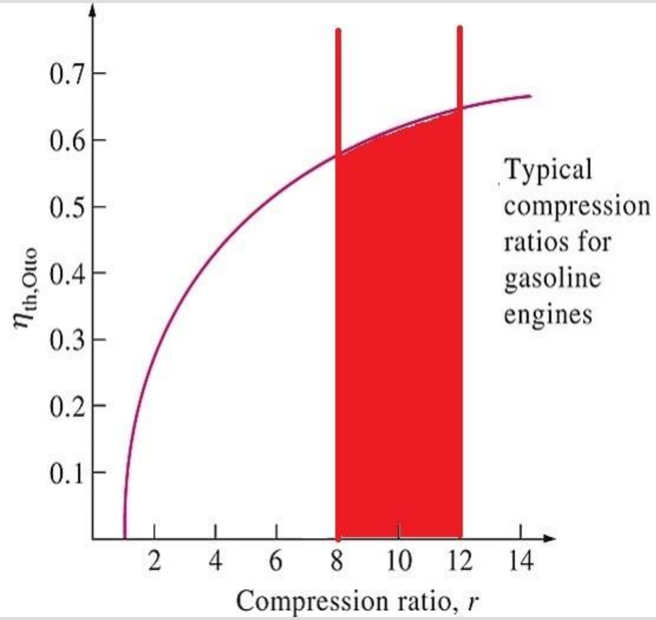
$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

$$\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

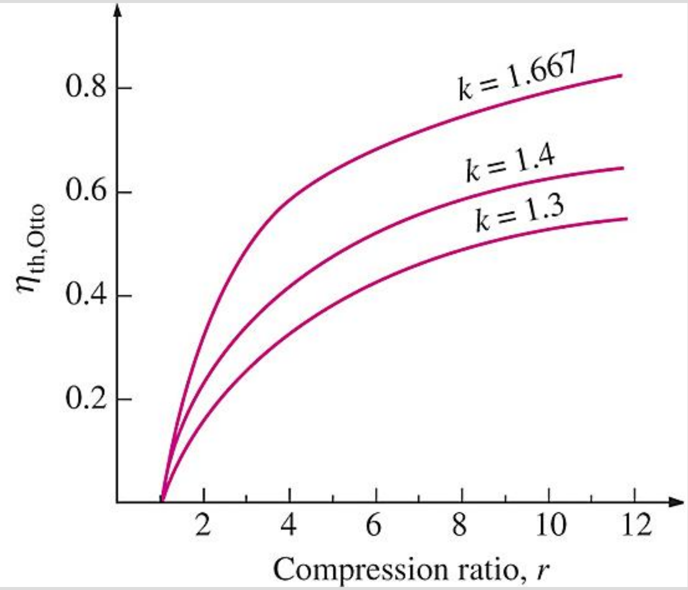
$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{v_3}{v_4}\right)^{k-1} = \frac{T_4}{T_3}$$

$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2}$$




$$\boxed{\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}}$$

In SI engines, the compression ratio is limited **by autoignition or engine knock**.




The thermal efficiency of the Otto cycle increases with the specific heat ratio *k* of the working fluid.

# OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

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

$$
\begin{aligned}
Pv &= RT \\
PV &= mRT \\
P &= \rho RT \\
dh &= c_p\, dT
\end{aligned}
$$

$$
\begin{aligned}
du &= c_v\, dT \\
Pv^k &= \text{constant} \quad \text{isentropic process} \\
Tv^{k-1} &= \text{constant} \quad \text{isentropic process} \\
TP^{(1-k)/k} &= \text{constant} \quad \text{isentropic process} \\
w_{1-2} &= (P_2 v_2 - P_1 v_1)/(1-k) \quad \text{isentropic work in closed system} \\
&= R(T_2 - T_1)/(1-k) \\
c &= \sqrt{kRT} \quad \text{speed of sound}
\end{aligned}
$$

where

$P$ = 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
$\rho$ = 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

# OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

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:

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

## OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

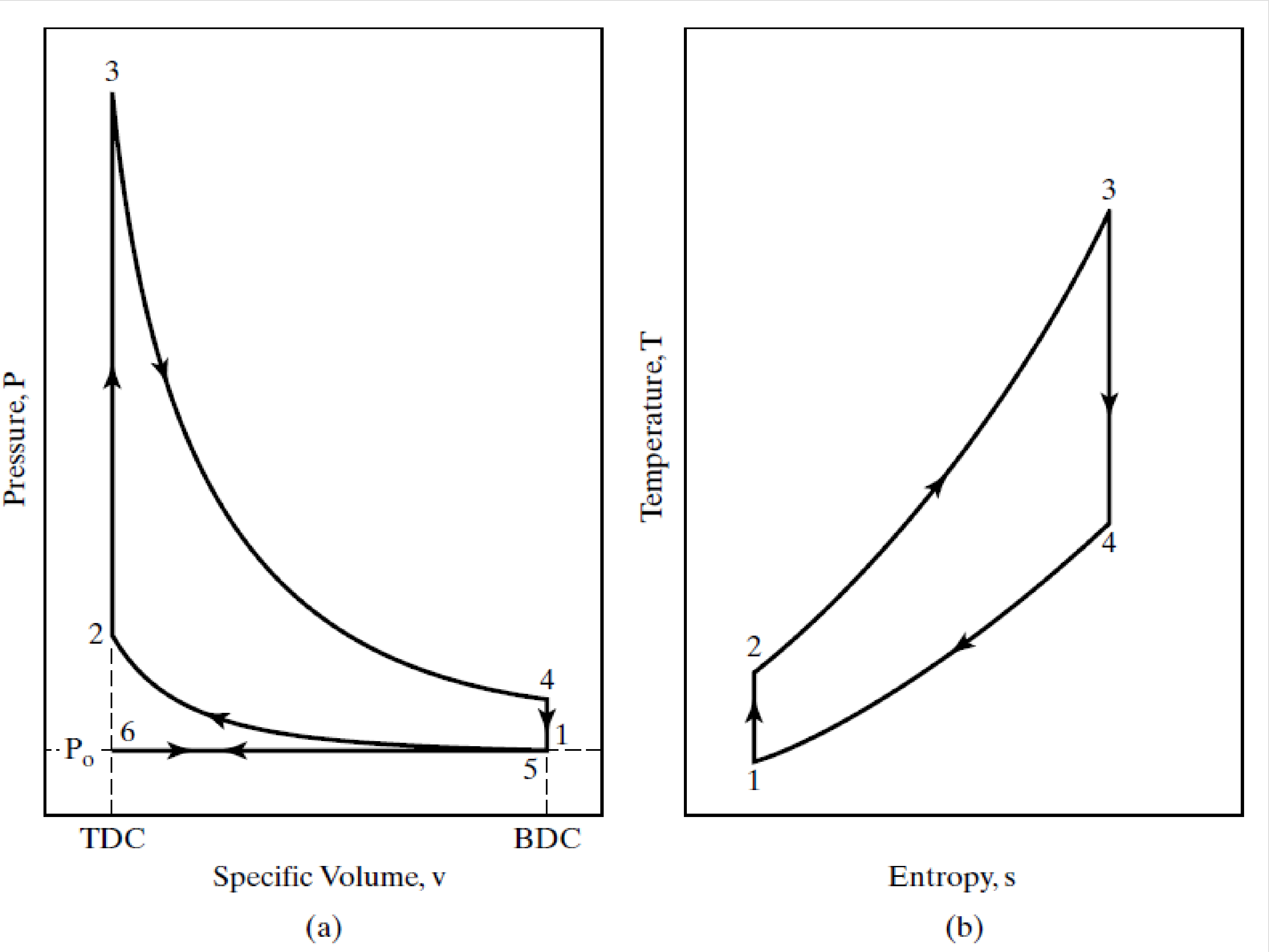
When analyzing what occurs within engines during the **operating cycle and exhaust flow**, this text uses the following air property values:

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

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. For these conditions, the following air property values are used:

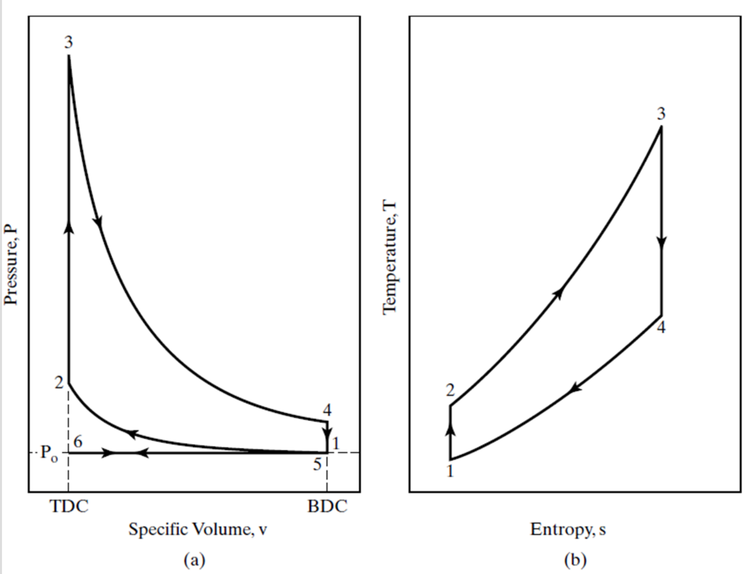
$$c_p = 1.005 \text{ kJ/kg-K} = 0.240 \text{ BTU/lbm-°R}$$
$$c_v = 0.718 \text{ kJ/kg-K} = 0.172 \text{ BTU/lbm-°R}$$
$$k = c_p/c_v = 1.005/0.718 = 1.40$$
$$R = c_p - c_v = 0.287 \text{ kJ/kg-K}$$

# OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES




It is common 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 <u>**cancel each other thermodynamically**</u> and are not needed in analyzing the cycle.

# Thermodynamic Analysis of Air-Standard Otto Cycle at WOT




**Process 6-1—constant-pressure intake of air**

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:

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

$$w_{1-2} = (P_2v_2 - P_1v_1)/(1-k) = R(T_2 - T_1)/(1-k)$$
$$= (u_1 - u_2) = c_v(T_1 - T_2)$$

**Process 2-3—constant-volume heat input (combustion)**

All valves closed:

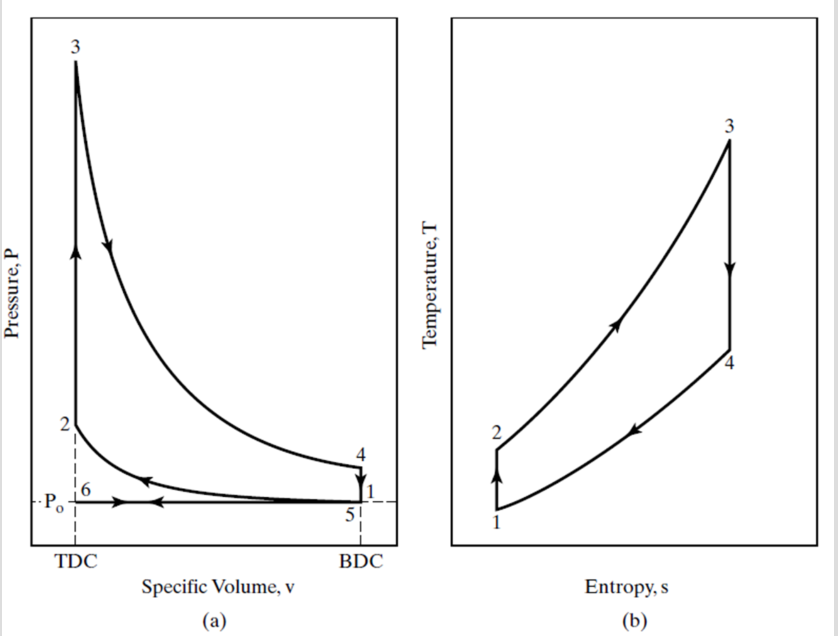
$$v_3 = v_2 = v_{\mathrm{TDC}}$$
$$w_{2-3} = 0$$
$$Q_{2-3} = Q_{\mathrm{in}} = m_f Q_{\mathrm{HV}} \eta_c = m_m c_v (T_3 - T_2)$$
$$= (m_a + m_f) c_v (T_3 - T_2)$$
$$Q_{\mathrm{HV}} \eta_c = (\mathrm{AF} + 1) c_v (T_3 - T_2)$$
$$q_{2-3} = q_{\mathrm{in}} = c_v(T_3 - T_2) = (u_3 - u_2)$$
$$T_3 = T_{\mathrm{max}}$$
$$P_3 = P_{\mathrm{max}}$$

# Thermodynamic Analysis of Air-Standard Otto Cycle at WOT

**Process 3-4—isentropic power or expansion stroke**

<u>All valves closed:</u>

$$
\begin{aligned}
q_{3-4} &= 0 \\
T_4 &= T_3(v_3/v_4)^{k-1} = T_3(V_3/V_4)^{k-1} = T_3(1/r_c)^{k-1} \\
P_4 &= P_3(v_3/v_4)^{k} = P_3(V_3/V_4)^{k} = P_3(1/r_c)^{k} \\
w_{3-4} &= (P_4v_4 - P_3v_3)/(1-k) = R(T_4 - T_3)/(1-k) \\
&= (u_3 - u_4) = c_v(T_3 - T_4)
\end{aligned}
$$




**Process 4-5—constant-volume heat rejection (exhaust blowdown)**

<u>Exhaust valve open and intake valve closed:</u>

$$
\begin{aligned}
v_5 &= v_4 = v_1 = v_{\mathrm{BDC}} \\
w_{4-5} &= 0 \\
Q_{4-5} &= Q_{\mathrm{out}} = m_m c_v(T_5 - T_4) = m_m c_v(T_1 - T_4) \\
q_{4-5} &= q_{\mathrm{out}} = c_v(T_5 - T_4) = (u_5 - u_4) = c_v(T_1 - T_4)
\end{aligned}
$$

**Process 5-6—constant-pressure exhaust stroke**

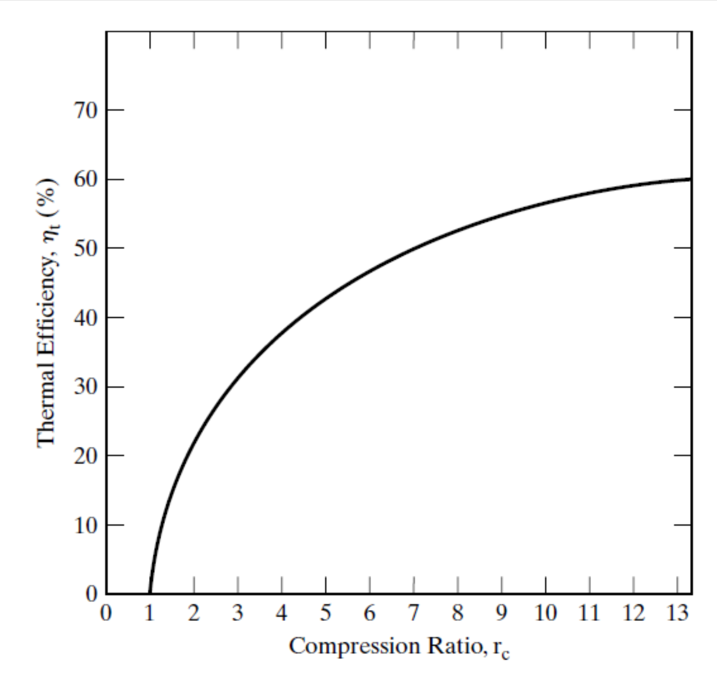
<u>Exhaust valve open and intake valve closed:</u>

$$
\begin{aligned}
P_5 &= P_6 = P_o \\
w_{5-6} &= P_o(v_6 - v_5) = P_o(v_6 - v_1)
\end{aligned}
$$

# Thermodynamic Analysis of Air-Standard Otto Cycle at WOT

## Thermal efficiency of Otto cycle:

$$(\eta_t)_{\text{OTTO}} = |w_{\text{net}}|/|q_{\text{in}}| = 1 - (|q_{\text{out}}|/|q_{\text{in}}|)$$
$$= 1 - [c_v(T_4 - T_1)/c_v(T_3 - T_2)]$$
$$= 1 - [(T_4 - T_1)/(T_3 - T_2)]$$




Isentropic compression and expansion strokes recognizing

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

Rearranging the temperature terms gives

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

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

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

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

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

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

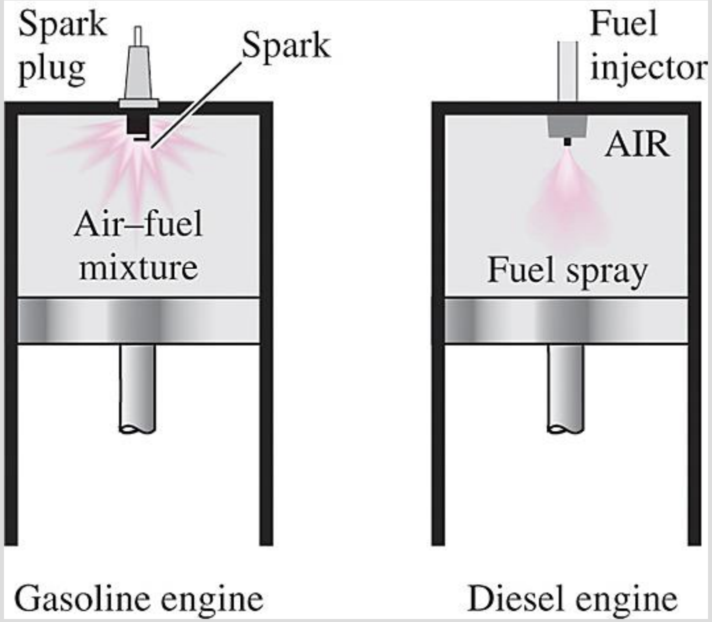
## Example Problem 1

A four-cylinder, 2.5-liter, SI automobile engine operates at WOT on a four-stroke air-standard Otto cycle at 3000 RPM. The engine has a compression ratio of 8.6:1, a mechanical efficiency of 86%, and a stroke-to-bore ratio $S/B = 1.025$. Fuel is isooctane with AF = 15, a heating value of 44,300 kJ/kg, and combustion efficiency $\eta_c = 100\%$. At the start of the compression stroke, conditions in the cylinder combustion chamber are 100 kPa and 60°C. It can be assumed that there is a 4% exhaust residual left over from the previous cycle.

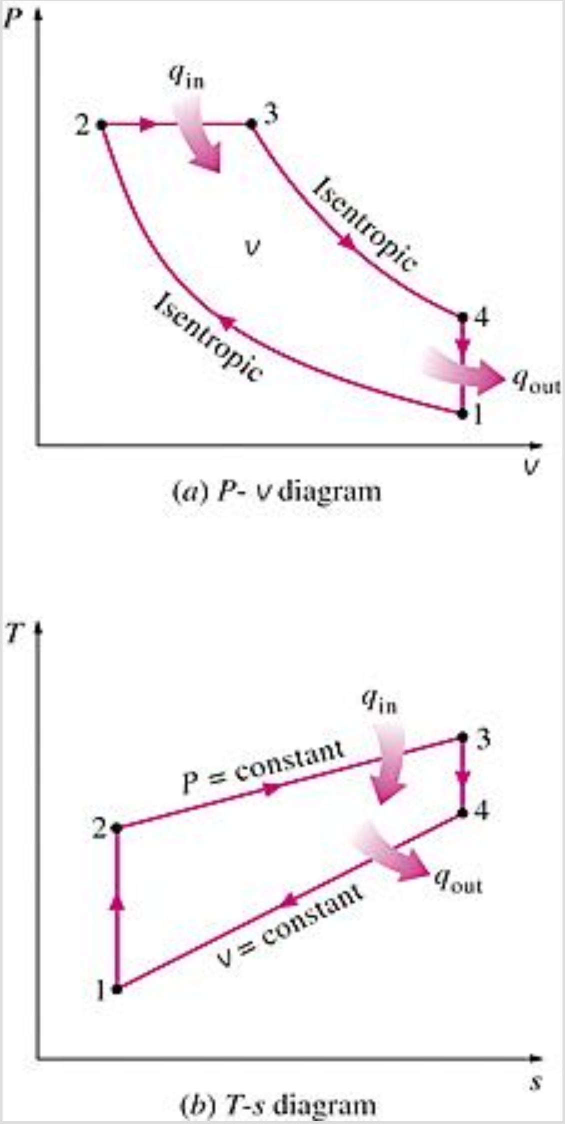
Do a complete thermodynamic analysis of this engine.

# DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

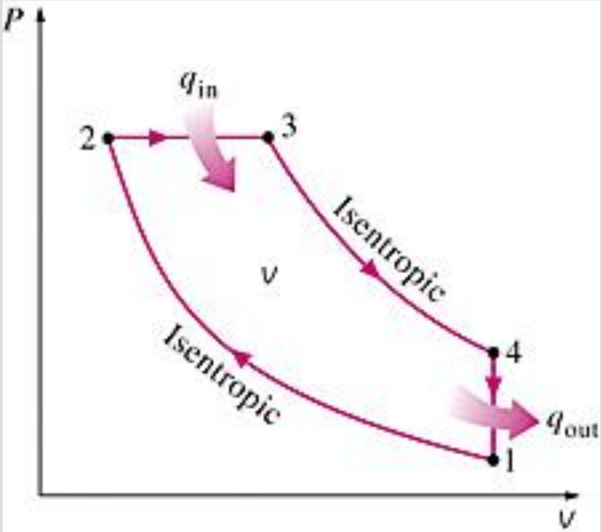
In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition (engine knock). Therefore, diesel engines can be designed to operate at much higher compression ratios than SI engines, typically between 12 and 24.



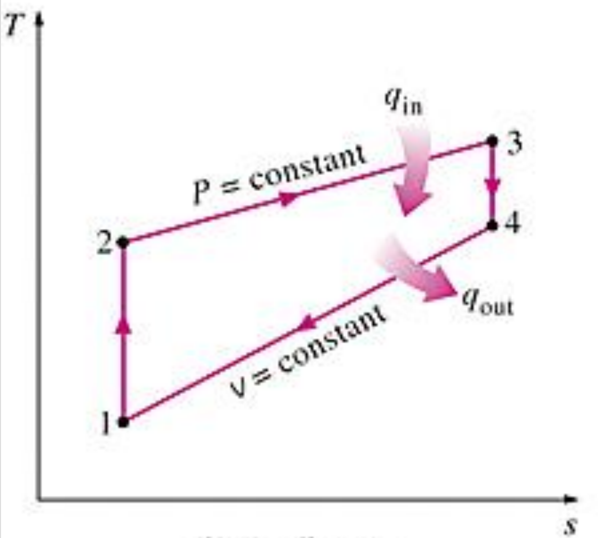

- **1-2** isentropic compression
- **2-3** constant-pressure heat addition
- **3-4** isentropic expansion
- **4-1** constant-volume heat rejection.




In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.



(*a*) *P*- *v* diagram



(*b*) *T*-*s* diagram

$$q_{in} - w_{b,out} = u_3 - u_2 \rightarrow q_{in} = P_2(v_3 - v_2) + (u_3 - u_2)$$
$$= h_3 - h_2 = c_p(T_3 - T_2)$$

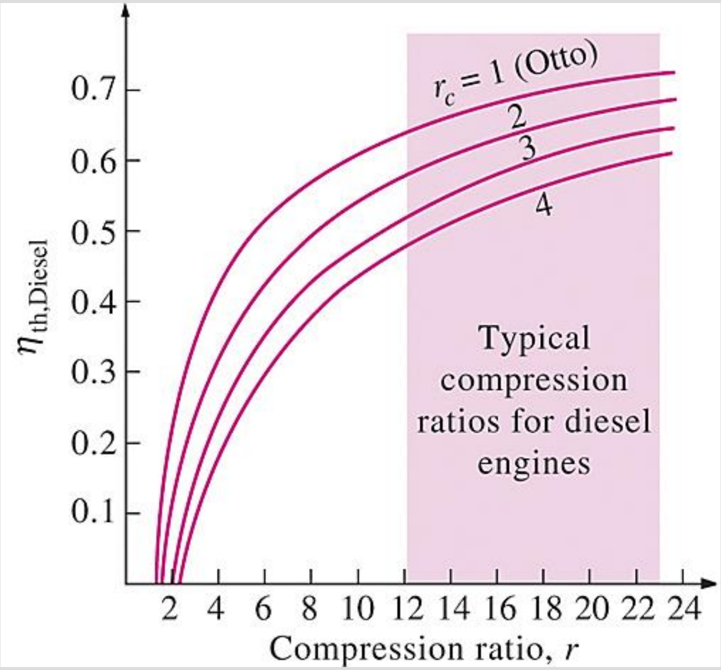
$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

$$\eta_{th,Diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)}$$

$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2}$$ Cutoff ratio

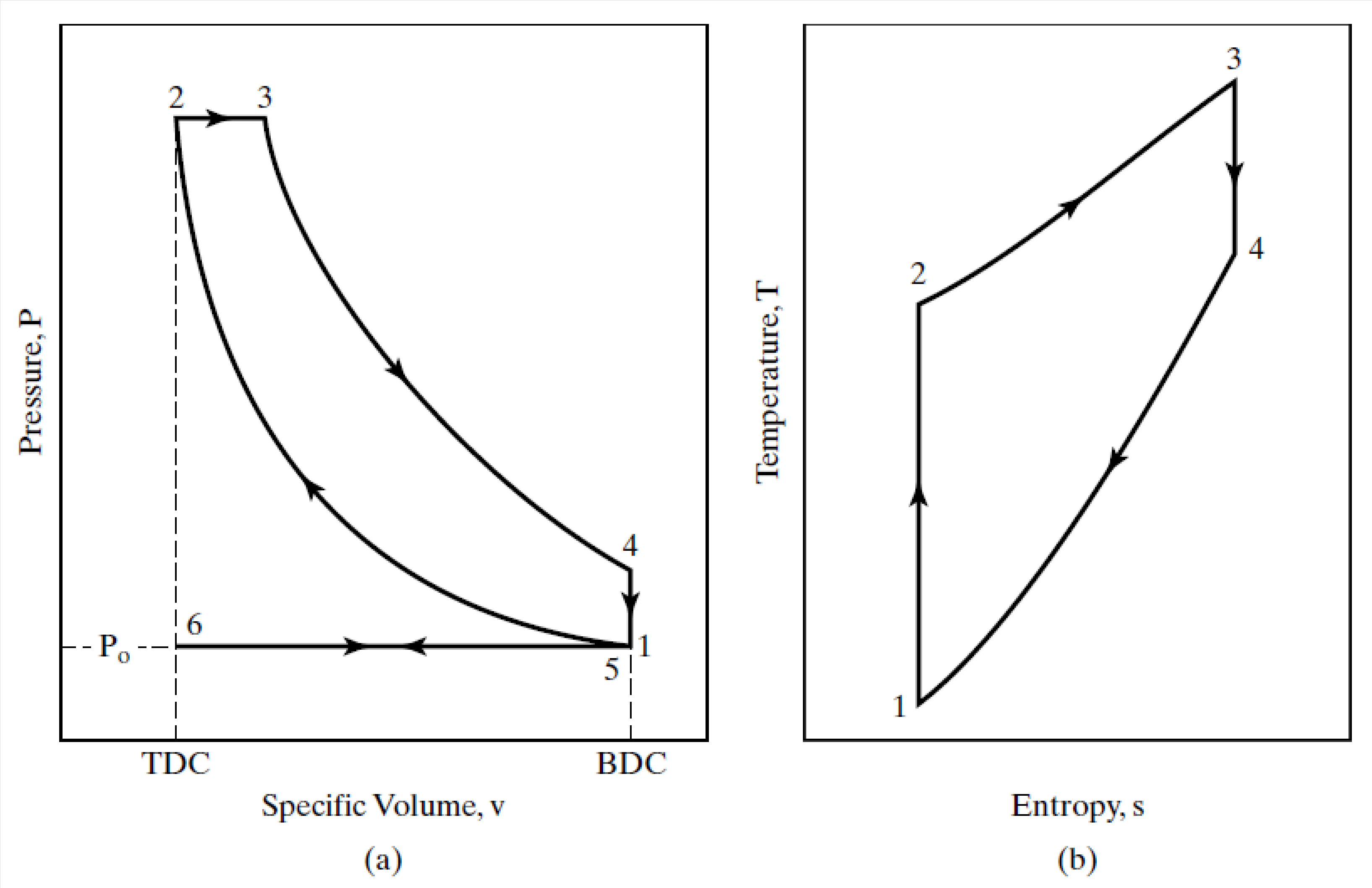
$$\boxed{\eta_{th,Diesel} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right]}$$

$\boxed{\eta_{th,Otto} > \eta_{th,Diesel}}$ for the same compression ratio




Thermal efficiency of the ideal Diesel cycle as a function of compression and cutoff ratios (*k*=1.4)

# DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES




The pressure at peak levels well past TDC. This combustion process is best approximated as a constant-pressure heat input in an air-standard cycle, resulting in the **Diesel cycle**

# Thermodynamic Analysis of Air-Standard Diesel Cycle

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

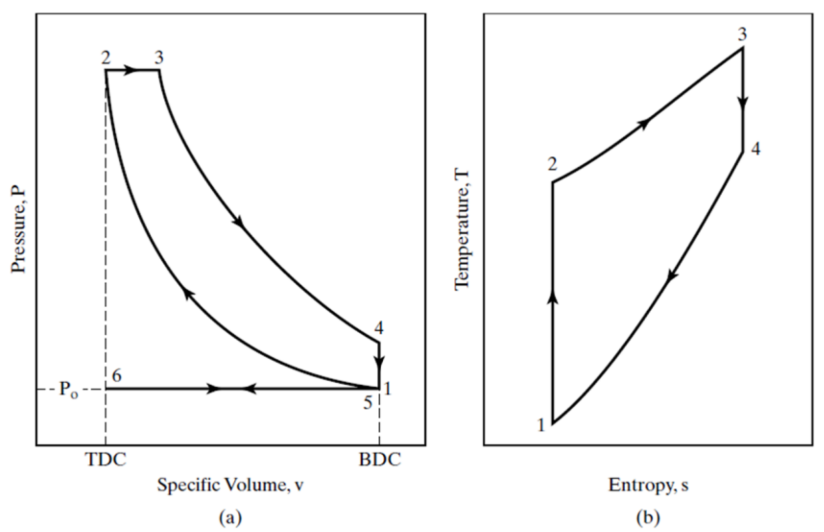
$$w_{6-1} = P_o(v_1 - v_6)$$

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

$$\begin{aligned} T_2 &= T_1(v_1/v_2)^{k-1} = T_1(V_1/V_2)^{k-1} = T_1(r_c)^{k-1} \\ P_2 &= P_1(v_1/v_2)^k = P_1(V_1/V_2)^k = P_1(r_c)^k \\ V_2 &= V_{\mathrm{TDC}} \\ q_{1-2} &= 0 \\ w_{1-2} &= (P_2v_2 - P_1v_1)/(1-k) = R(T_2 - T_1)/(1-k) \\ &= (u_1 - u_2) = c_v(T_1 - T_2) \end{aligned}$$

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

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




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

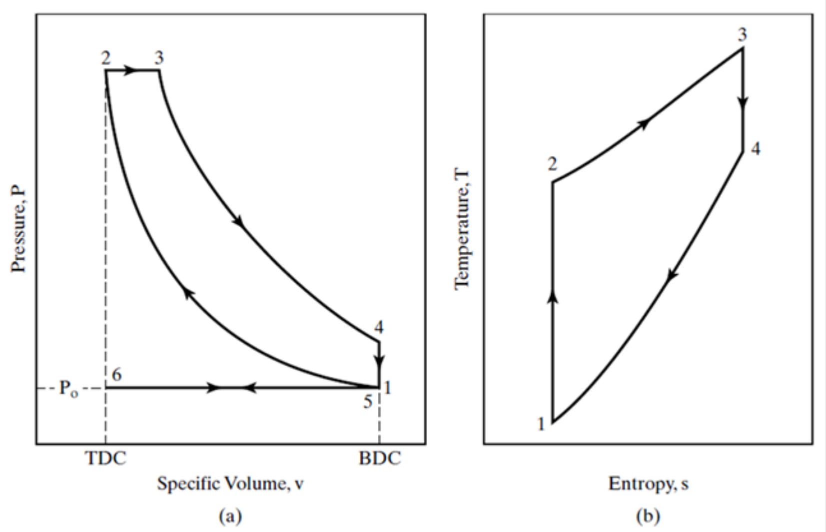
$$\beta = V_3/V_2 = v_3/v_2 = T_3/T_2$$

# Thermodynamic Analysis of Air-Standard Diesel Cycle

**Process 3-4—isentropic power or expansion stroke**

<u>All valves closed:</u>

$$
\begin{aligned}
q_{3-4} &= 0 \\
T_4 &= T_3(v_3/v_4)^{k-1} = T_3(V_3/V_4)^{k-1} \\
P_4 &= P_3(v_3/v_4)^{k} = P_3(V_3/V_4)^{k} \\
w_{3-4} &= (P_4 v_4 - P_3 v_3)/(1-k) = R(T_4 - T_3)/(1-k) \\
&= (u_3 - u_4) = c_v(T_3 - T_4)
\end{aligned}
$$




**Process 4-5—constant-volume heat rejection (exhaust blowdown)**

<u>Exhaust valve open and intake valve closed:</u>

$$
\begin{aligned}
v_5 &= v_4 = v_1 = v_{\mathrm{BDC}} \\
w_{4-5} &= 0 \\
Q_{4-5} &= Q_{\mathrm{out}} = m_m c_v(T_5 - T_4) = m_m c_v(T_1 - T_4) \\
q_{4-5} &= q_{\mathrm{out}} = c_v(T_5 - T_4) = (u_5 - u_4) = c_v(T_1 - T_4)
\end{aligned}
$$

**Process 5-6—constant-pressure exhaust stroke**

<u>Exhaust valve open and intake valve closed:</u>

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

# Thermodynamic Analysis of Air-Standard Diesel Cycle

## Thermal efficiency of Diesel cycle:

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

With rearrangement, this can be shown to equal

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

It is found that the value of the term in brackets is greater than one.

When this equation is compared with Otto thermal efficiency eqn. can be seen that <u>for a given compression ratio</u> ***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, CI engines operate with much higher compression ratios than SI engines (12 to 24 versus 8 to 12) and thus have higher thermal efficiencies.**

Where;

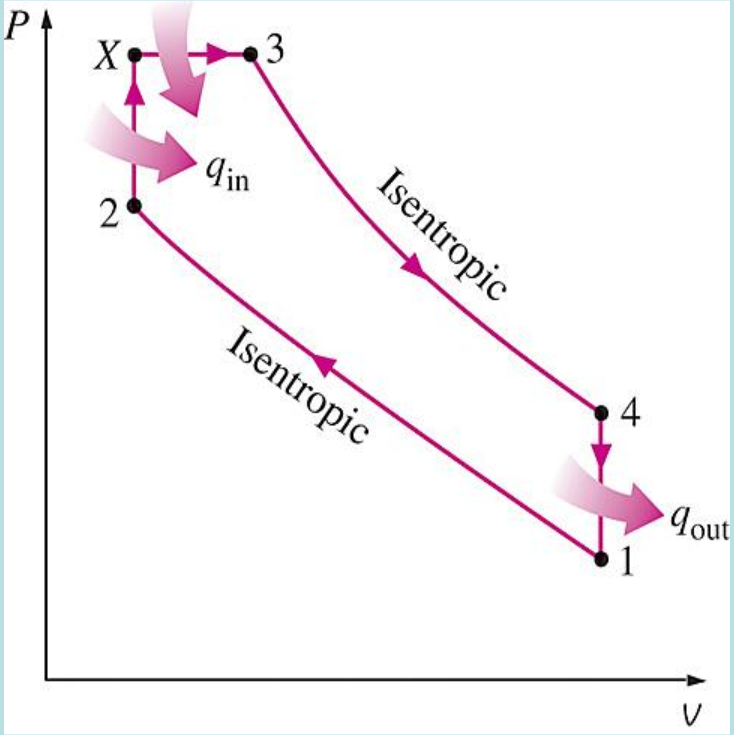
$$
\begin{aligned}
r_c &= \text{compression ratio} \\
k &= c_p/c_v \\
\beta &= \text{cutoff ratio}
\end{aligned}
$$

Recall that the Otto thermal efficiency;

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

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

**Dual cycle:** A more realistic ideal cycle model for modern, high-speed compression ignition engine.




*P-v* diagram of an ideal dual cycle.

## QUESTIONS

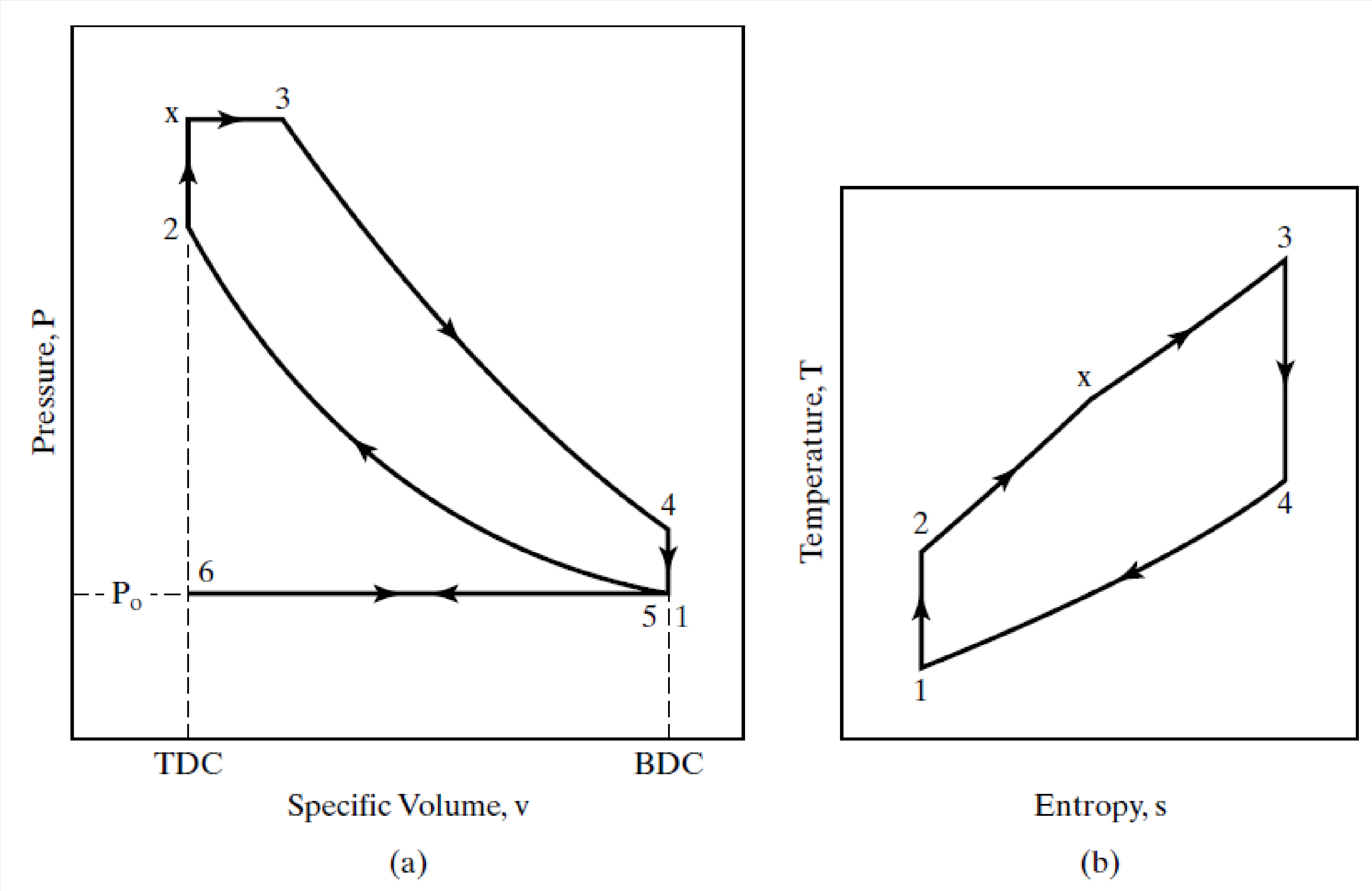
Diesel engines operate at higher air-fuel ratios than gasoline engines. Why?

Despite higher power to weight ratios, two-stroke engines are not used in automobiles. Why?

The stationary diesel engines are among the most efficient power producing devices (about 50%). Why?

What is a turbocharger? Why are they mostly used in diesel engines compared to gasoline engines.

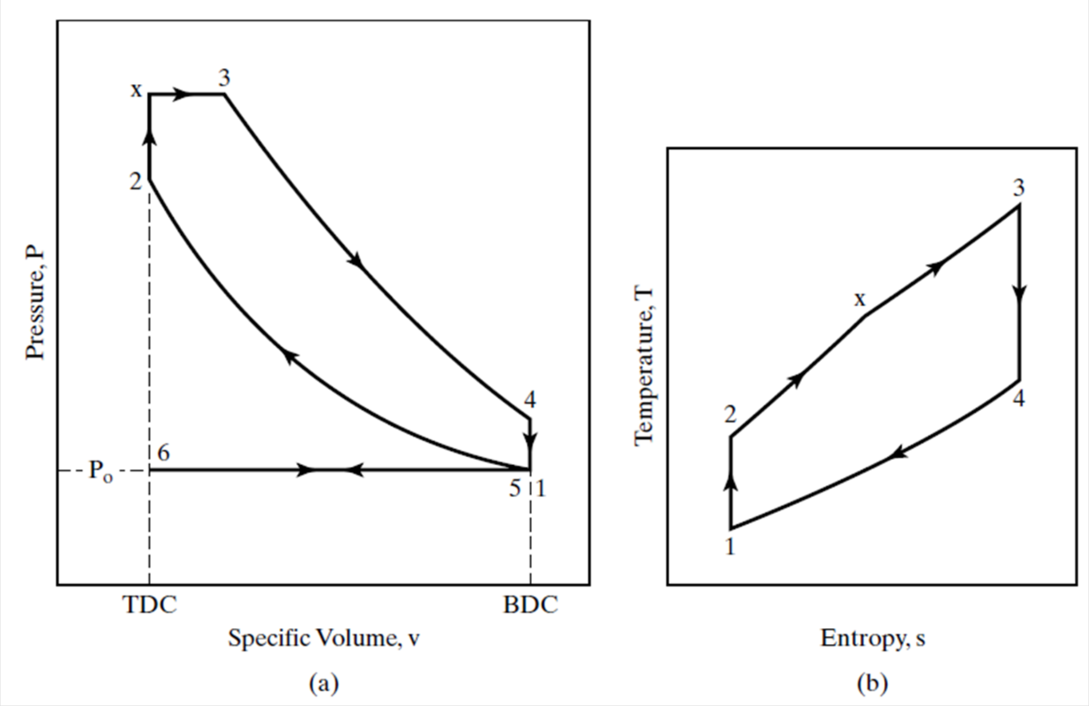
# DUAL CYCLE: THE IDEAL CYCLE FOR MODERN COMPRESSION-IGNITION ENGINES




The air-standard cycle used to analyze this modern CI engine cycle is called a **Dual cycle** or sometimes a **Limited Pressure cycle.** It is a Dual cycle because the <u>heat input process</u> of combustion can best be approximated by a Dual process of <u>constant volume followed by constant pressure</u>. It can also be considered a modified Otto cycle with a limited upper pressure.

# 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:

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

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

# 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.

$$
\begin{aligned}
Q_{2-x} &= m_m c_v(T_x - T_2) = (m_a + m_f)c_v(T_x - T_2) \\
q_{2-x} &= c_v(T_x - T_2) = (u_x - u_2) \\
P_x &= P_{\max} = P_2(T_x/T_2)
\end{aligned}
$$

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

$$
\alpha = P_x/P_2 = P_3/P_2 = T_x/T_2 = (1/r_c)^k(P_3/P_1)
$$

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

$$
\begin{aligned}
P_3 &= P_x = P_{\max} \\
Q_{x-3} &= m_m c_p(T_3 - T_x) = (m_a + m_f)c_p(T_3 - T_x) \\
q_{x-3} &= c_p(T_3 - T_x) = (h_3 - h_x) \\
w_{x-3} &= q_{x-3} - (u_3 - u_x) = P_x(v_3 - v_x) = P_3(v_3 - v_x) \\
T_3 &= T_{\max}
\end{aligned}
$$

# 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.

**Cutoff ratio:**

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

**Heat in:**

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

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

# Thermodynamic Analysis of Air-Standard Dual Cycle

## 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
$k = c_p/c_v$
$\alpha$ = pressure ratio
$\beta$ = cutoff ratio

Recall that the Diesel thermal efficiency;

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

Recall that the Otto thermal efficiency;

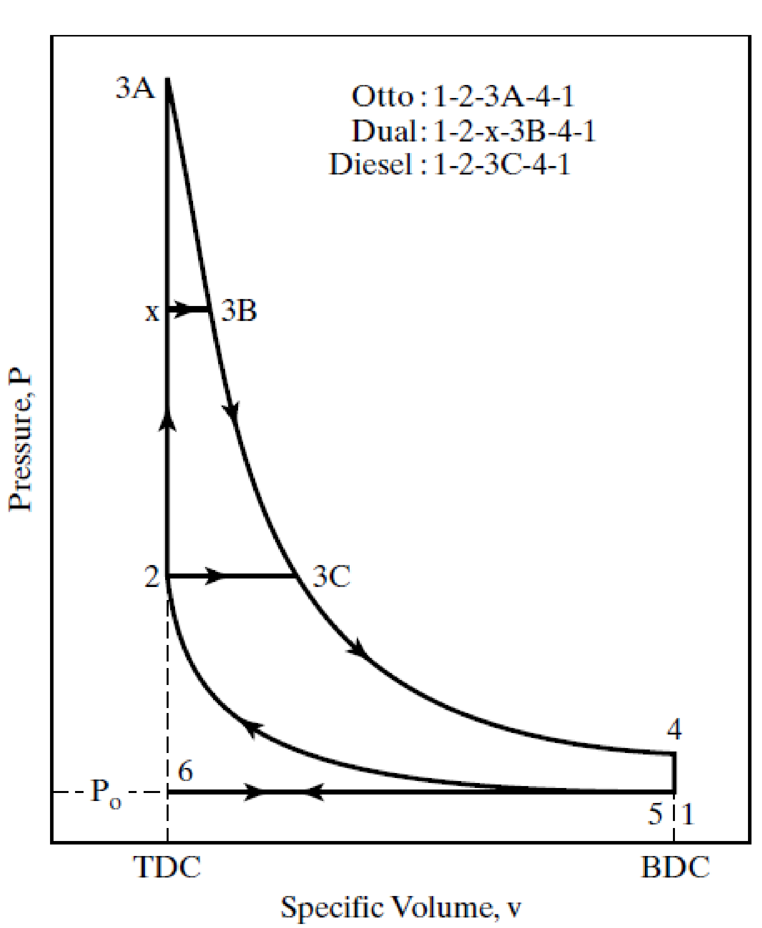
With $v_1/v_2 = r_c$, the compression ratio is

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

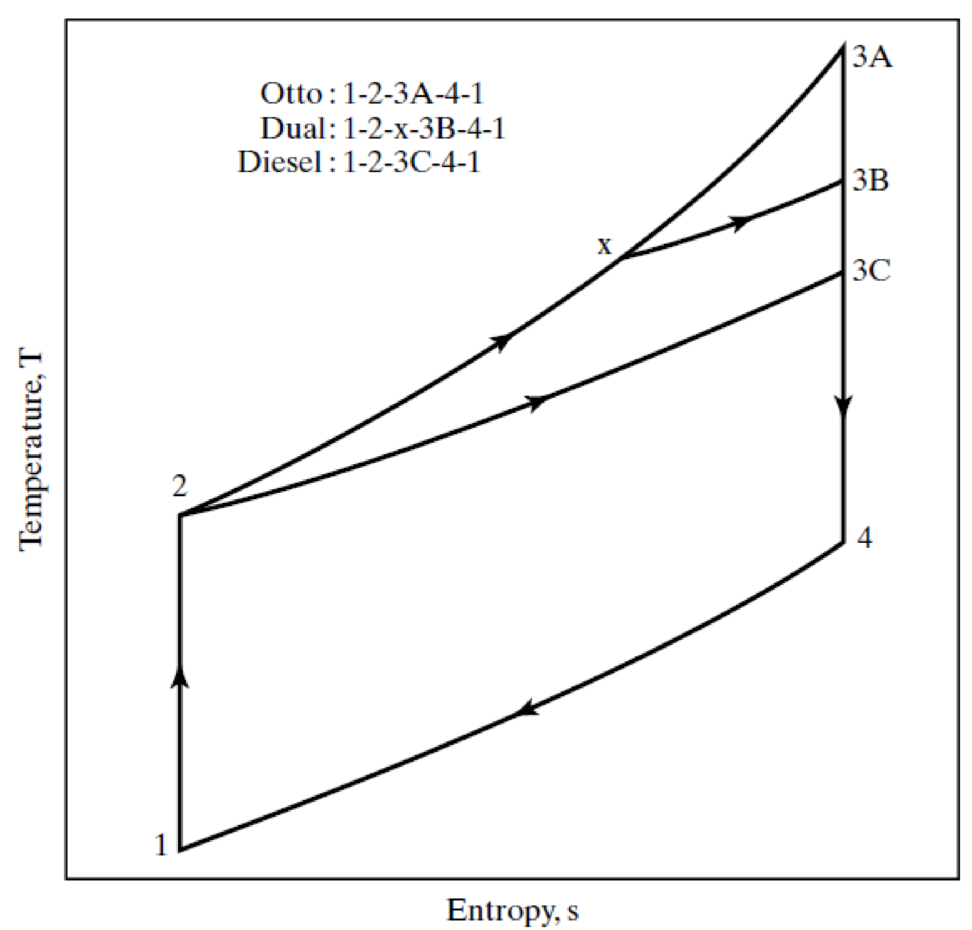
$$(\eta_t)_{\text{actual}} \approx 0.85(\eta_t)_{\text{DIESEL}}$$
$$(\eta_t)_{\text{actual}} \approx 0.85(\eta_t)_{\text{DUAL}}$$

# COMPARISON OF OTTO, DIESEL, AND DUAL CYCLES

First compares Otto, Diesel, and Dual cycles with ***the same inlet conditions and the same compression ratios***



(a)

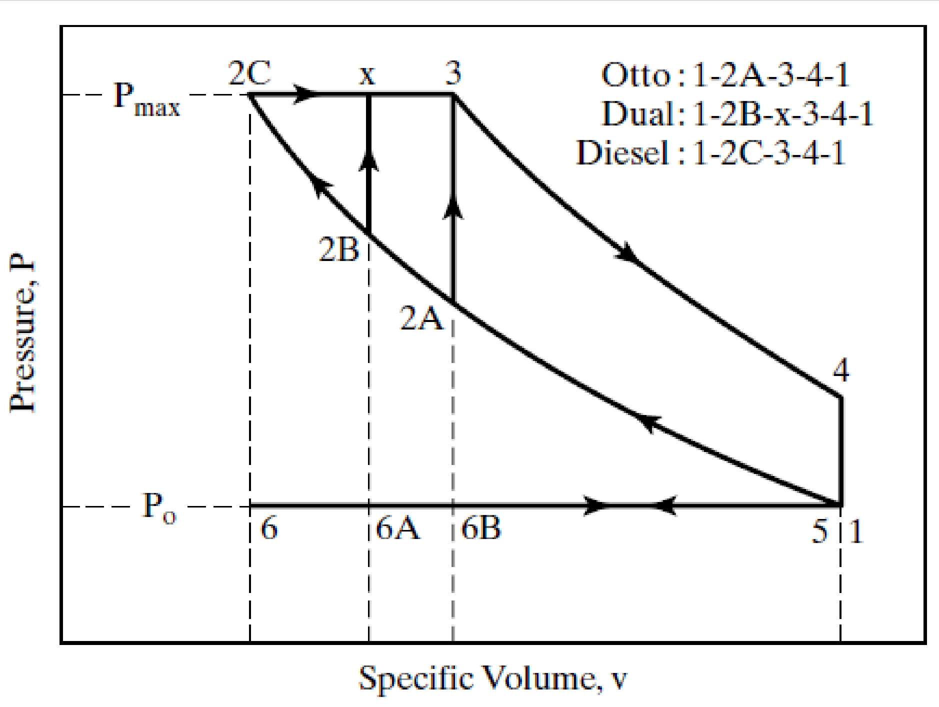


(b)

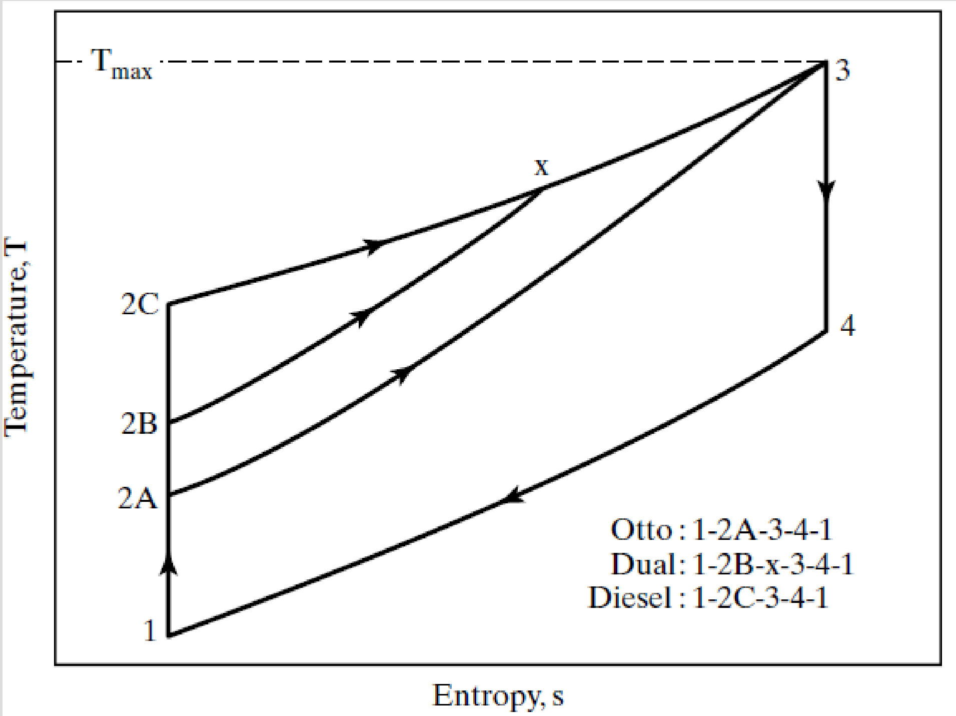
$$(\eta_t)_{\mathrm{OTTO}} > (\eta_t)_{\mathrm{DUAL}} > (\eta_t)_{\mathrm{DIESEL}}$$

# COMPARISON OF OTTO, DIESEL, AND DUAL CYCLES

A more realistic way to compare these three cycles would be to ***have the same peak pressure***—an actual design limitation in engines.



(a)



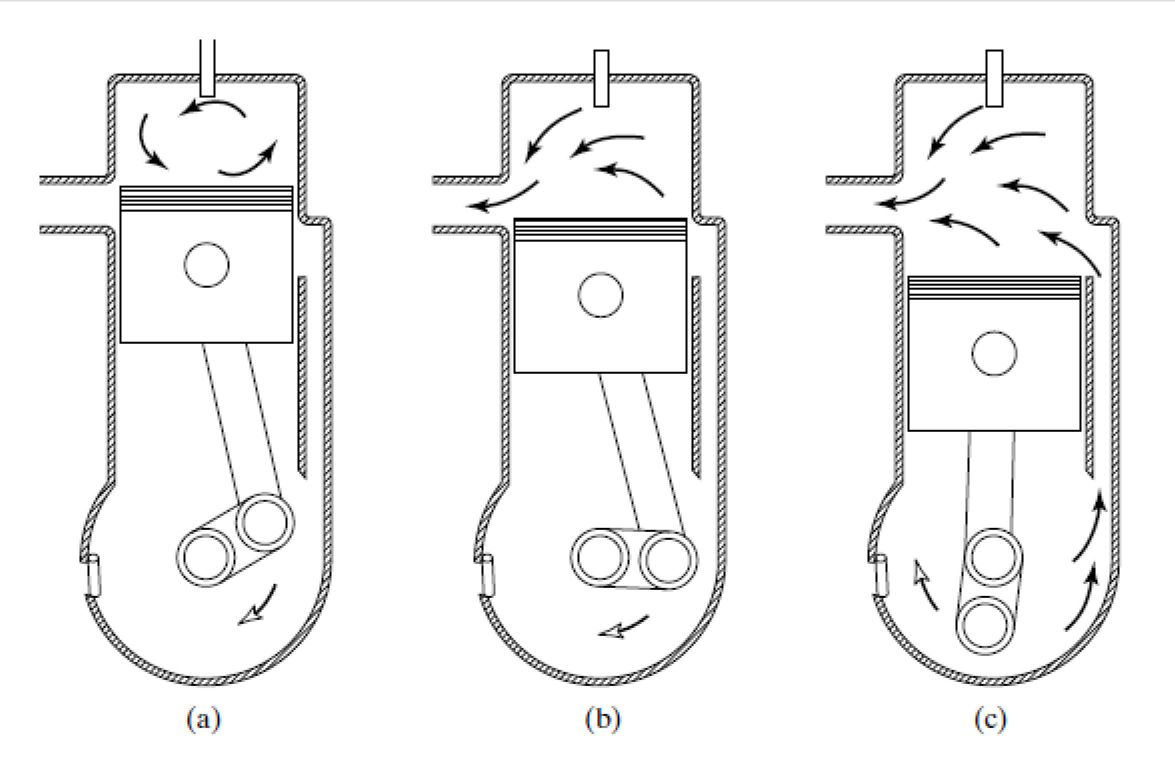
(b)

Comparing the ideas, It would suggest that the most efficient engine would have ***combustion as close as possible to constant volume but would be compression ignition and operate at the higher compression ratios that are required***.

**This is an area where more research and development is needed.**

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

## BASIC CYCLES OF TWO STROKE SI ENGINES



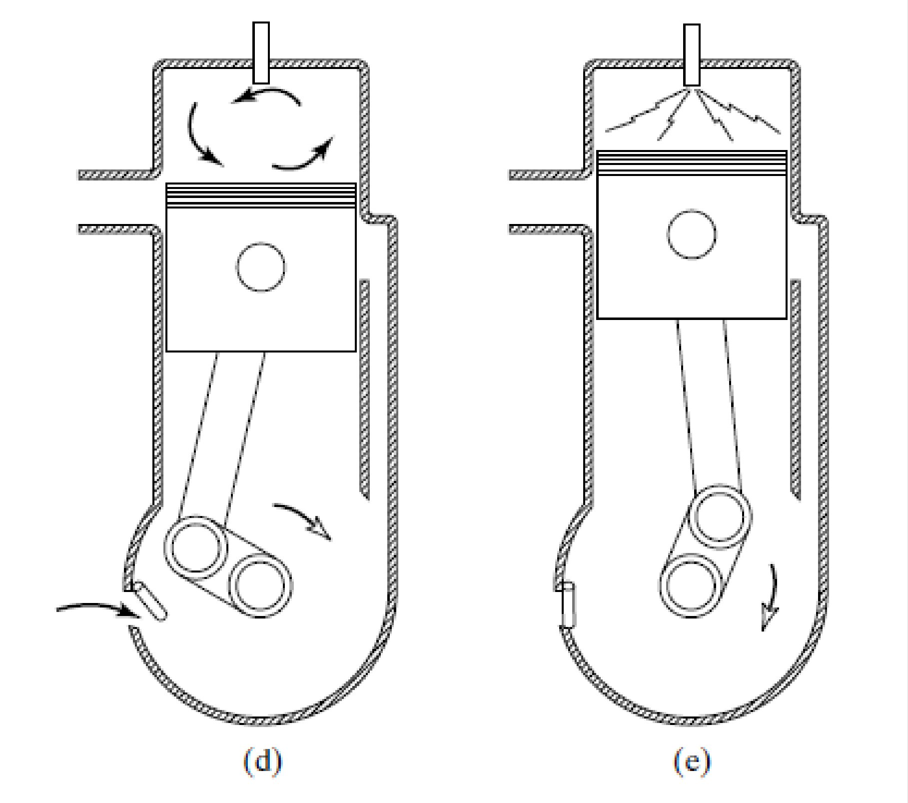

*Two-stroke SI engine operating cycle with crankcase compression.*

**Four-Stroke Cycle.** A four-stroke cycle has **four piston movements over two engine revolutions** for each cycle. **Two-Stroke Cycle.** A two-stroke cycle has **two piston movements over one revolution** for each cycle.

**(a)** Power or expansion stroke. High cylinder pressure pushes piston from TDC towards BDC with all ports closed. Air in crankcase is compressed by downward motion of piston.

**(b)** Exhaust blowdown when exhaust port opens near end of power stroke.
**(c)** Cylinder scavenging when transfer port opens and air-fuel is forced into cylinder under pressure. Intake mixture pushes some of the remaining exhaust out the open exhaust port. Scavenging lasts until piston passes BDC and closes transfer and exhaust ports.

# BASIC CYCLES OF TWO STROKE SI ENGINES




*Two-stroke SI engine operating cycle with crankcase compression.*

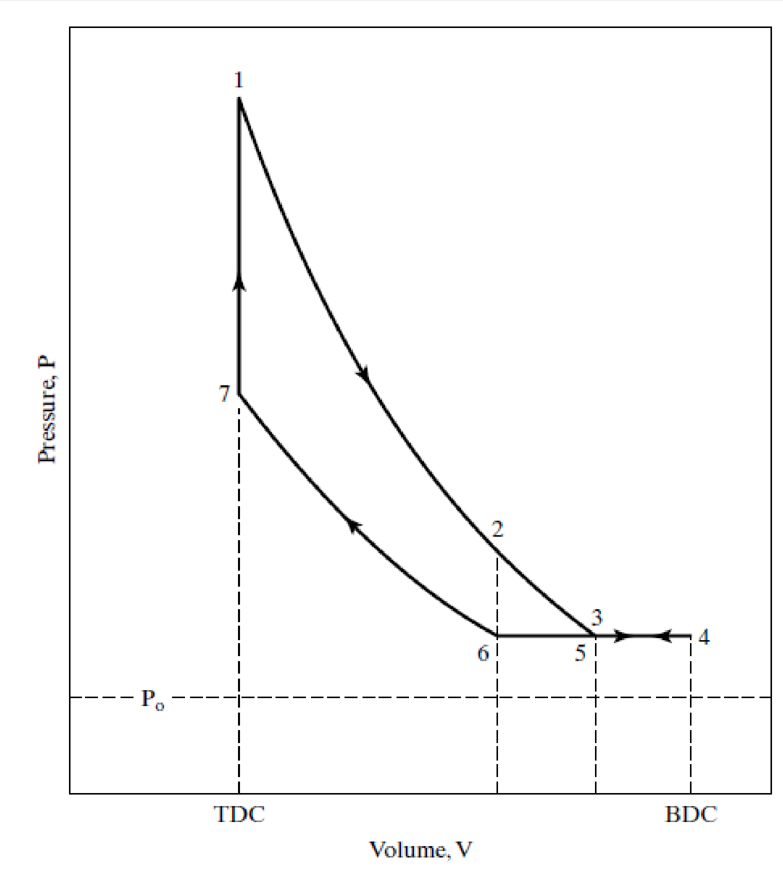
**(d)** Compression stroke. Piston moves from BDC to TDC with all ports closed. Intake air fills crankcase. Spark ignition occurs near end of compression stroke.

**(e)** Combustion at almost constant volume near TDC.

## Two-Stroke SI Engine Cycle

*Intake and Scavenging* when blowdown is nearly complete, at about 50° bBDC, the intake slot on the side of the cylinder is uncovered and intake air–fuel enters under pressure. Fuel is added to the air with either a carburetor or fuel injection. This incoming mixture pushes much of the remaining exhaust gases out the open exhaust valve and fills the cylinder with a combustible air–fuel mixture, a process called **scavenging**. The piston passes BDC and very quickly covers the transfer port and then the exhaust port (or the exhaust valve closes). The higher pressure at which the air enters the cylinder is established in one of two ways. Large two-stroke cycle engines generally have a supercharger, while small engines will intake the air through the crankcase. On these engines the crankcase is designed to serve as a compressor in addition to serving its normal function.

# AIR STANDART APPROXIMATION FOR TWO STROKE SI ENGINE




Process 1-2—isentropic power or expansion stroke.
All ports (or valves) closed:

$$T_2 = T_1(V_1/V_2)^{k-1}$$
$$P_2 = P_1(V_1/V_2)^k$$
$$q_{1-2} = 0$$
$$w_{1-2} = (P_2v_2 - P_1v_1)/(1-k) = R(T_2 - T_1)/(1-k)$$

Process 2-3—exhaust blowdown.
Exhaust port open and intake port closed.

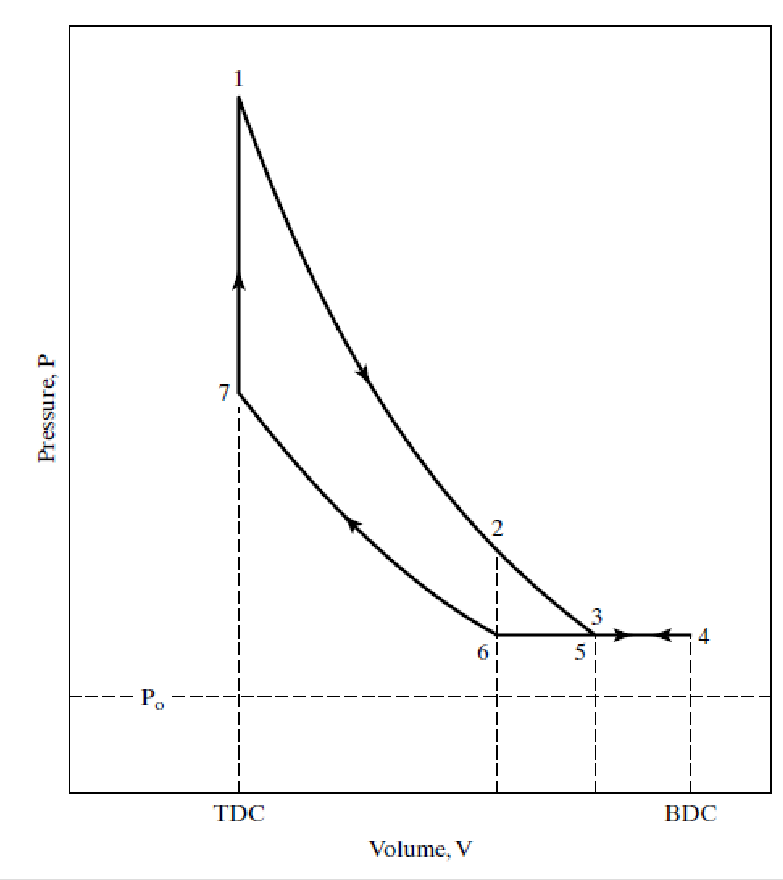
Process 3-4-5—intake, and exhaust scavenging.
Exhaust port open and transfer port open.

Intake air entering at an absolute pressure on the order of 140–180 kPa fills and scavenges the cylinder. ***Scavenging*** is a process in which the air pushes out most of the remaining exhaust residual from the previous cycle through the open exhaust port into the exhaust system, which is at about one atmosphere pressure. The piston uncovers the intake port at point 3, reaches BDC at point 4, reverses direction, and again closes the intake port at point 5. In some engines fuel is mixed with the incoming air. In other engines the fuel is injected later, after the exhaust port is closed.

Exhaust scavenging continues until the exhaust port is closed at point 6.

Process 5-6—exhaust scavenging.
Exhaust port open and transfer port closed.

# AIR STANDART APPROXIMATION FOR TWO STROKE SI ENGINE




Process 6-7—isentropic compression.
<u>All ports closed:</u>

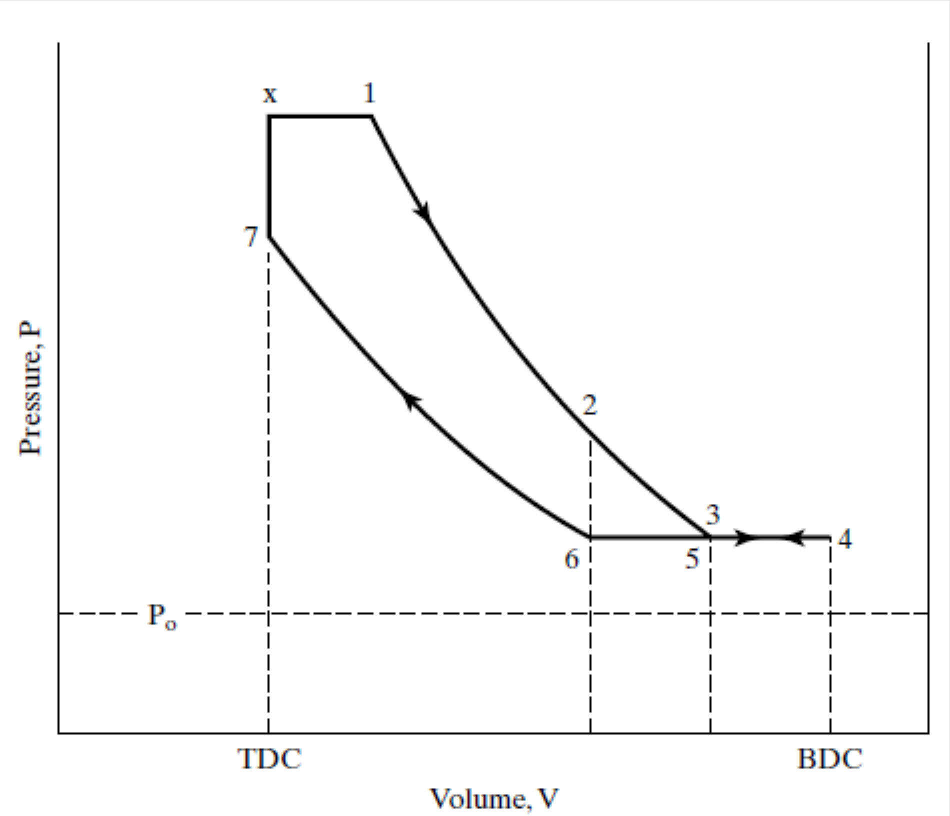
$$T_7 = T_6(V_6/V_7)^{k-1}$$
$$P_7 = P_6(V_6/V_7)^k$$
$$q_{6-7} = 0$$
$$w_{6-7} = (P_7 v_7 - P_6 v_6)/(1-k) = R(T_7 - T_6)/(1-k)$$

In some engines, fuel is added very early in the compression process. The spark plug is fired near the end of process 6-7.

Process 7-1—constant-volume heat input (combustion).
<u>All ports closed</u>:

$$V_7 = V_1 = V_{\mathrm{TDC}}$$
$$W_{7-1} = 0$$
$$Q_{7-1} = Q_{\mathrm{in}} = m_f Q_{\mathrm{HV}} \eta_c = m_m c_v (T_1 - T_7)$$
$$T_1 = T_{\max}$$
$$P_1 = P_{\max} = P_7(T_1/T_7)$$

# AIR STANDART APPROXIMATION FOR TWO STROKE CI ENGINE




Process 7-*x*—constant-volume heat input (first part of combustion).

All ports closed:

$$V_7 = V_x = V_{\mathrm{TDC}}$$
$$W_{7-x} = 0$$
$$Q_{7-x} = m_m c_v (T_x - T_7)$$
$$P_x = P_{\max} = P_7 (T_x / T_7)$$

Process *x*-1—constant-pressure heat input (second part of combustion).

All ports closed:

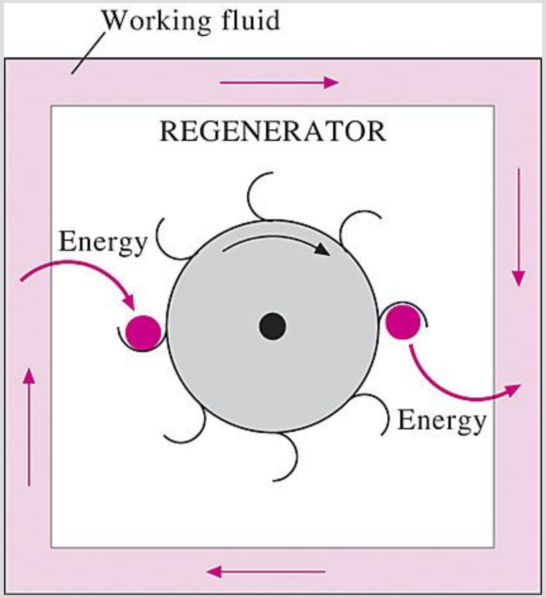
$$P_1 = P_x = P_{\max}$$
$$W_{x-1} = P_1 (V_1 - V_x)$$
$$Q_{x-1} = m_m c_p (T_1 - T_x)$$
$$T_1 = T_{\max}$$

Many compression ignition engines—**especially large ones**—operate on two-stroke cycles. These cycles can be approximated by the air-standard cycle shown in Fig. This cycle is the same as the two-stroke SI cycle **except for the fuel input and combustion process**. Instead of adding fuel to the intake air or early in the compression process, fuel is added with injectors late in the compression process, the same as with four-stroke cycle CI engines. Heat input or combustion can be approximated by a two-step (dual) process.

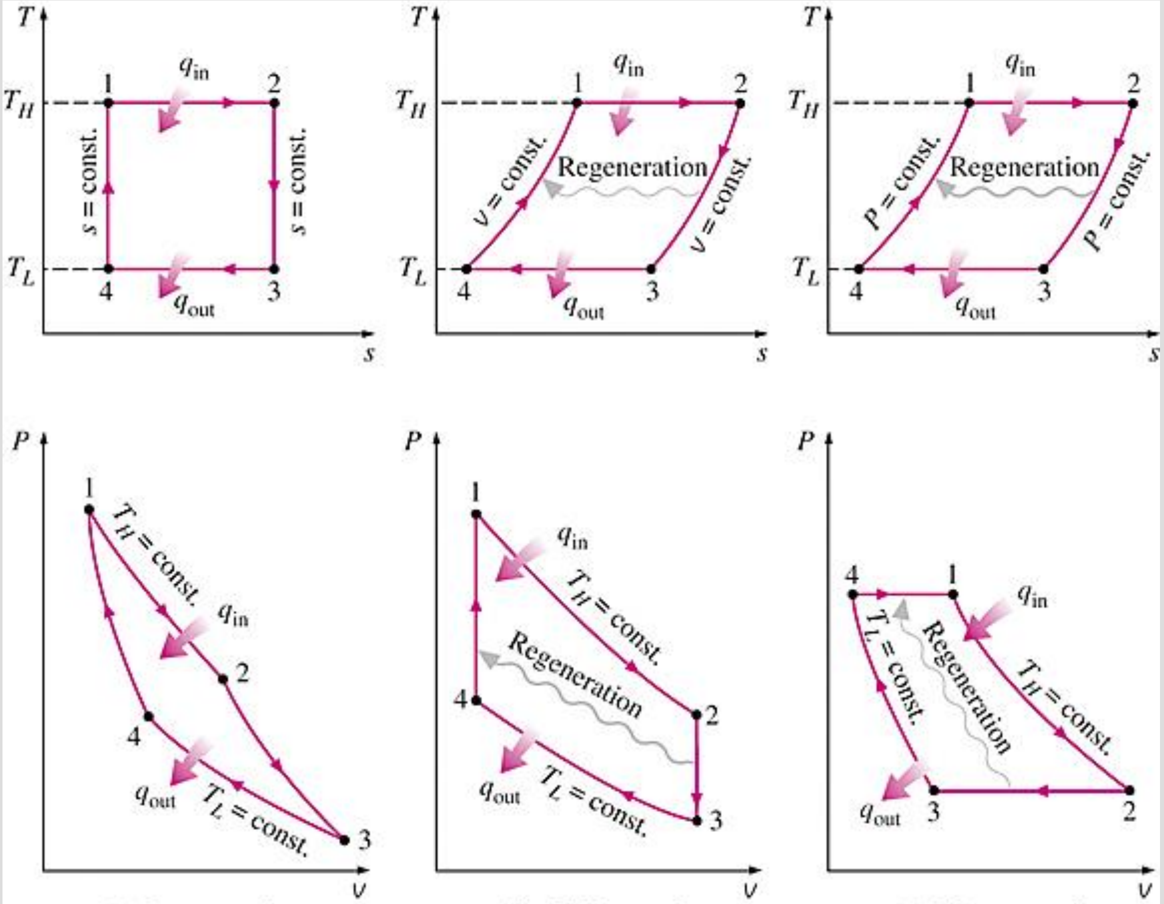
# STIRLING AND ERICSSON CYCLES

**Stirling cycle**

- **1-2** *T = constant* expansion (heat addition from the external source)
- **2-3** *v = constant* regeneration (internal heat transfer from the working fluid to the regenerator)
- **3-4** *T = constant* compression (heat rejection to the external sink)
- **4-1** *v = constant* regeneration (internal heat transfer from the regenerator back to the working fluid)




A regenerator is a device that borrows energy from the working fluid during one part of the cycle and pays it back (without interest) during another part.



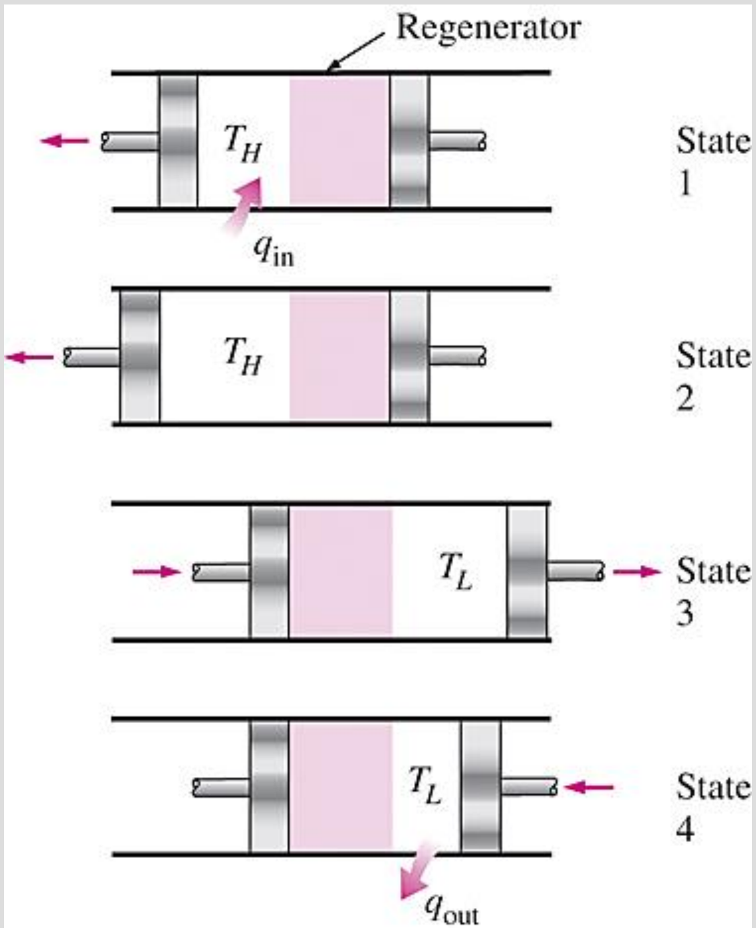

(a) Carnot cycle (b) Stirling cycle (c) Ericsson cycle

The Stirling and Ericsson cycles give a message: *Regeneration can increase efficiency.*

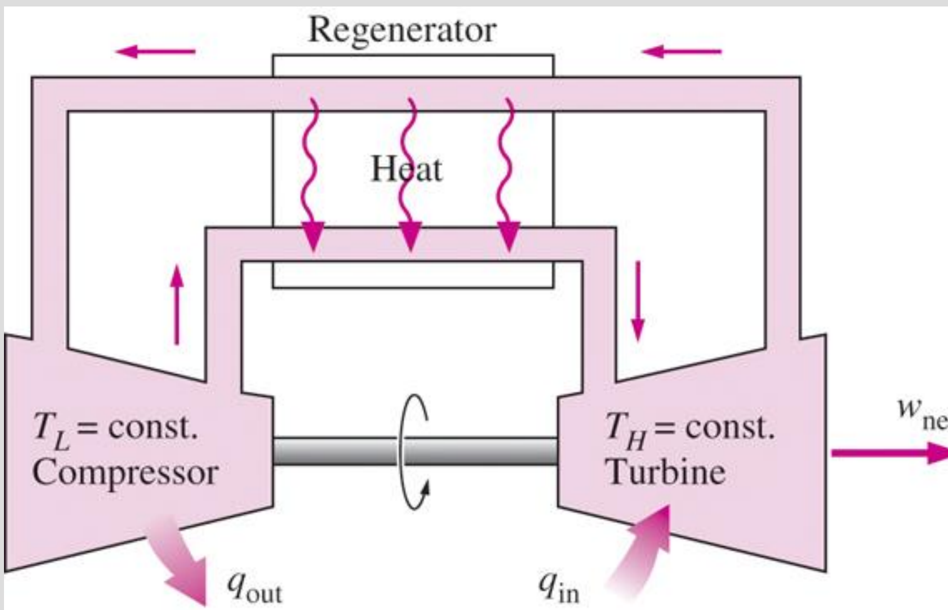
Both the Stirling and Ericsson cycles are totally reversible, as is the Carnot cycle, and thus:

$$\eta_{\text{th,Stirling}} = \eta_{\text{th,Ericsson}} = \eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H}$$

The Ericsson cycle is very much like the Stirling cycle, except that the two constant-volume processes are replaced by two constant-pressure processes.




The execution of the Stirling cycle.




A steady-flow Ericsson engine.

# SECOND-LAW ANALYSIS OF GAS POWER CYCLES

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} \geq 0$$

$$X_{\text{destroyed}} \begin{cases} > 0 & \text{Irreversible process} \\ = 0 & \text{Reversible process} \\ < 0 & \text{Impossible process} \end{cases}$$

Exergy destruction for a closed system

$$\dot{X}_{\text{dest}} = T_0 \dot{S}_{\text{gen}} = T_0(\dot{S}_{\text{out}} - \dot{S}_{\text{in}}) = T_0\left(\sum_{\text{out}} \dot{m}s - \sum_{\text{in}} \dot{m}s - \frac{\dot{Q}_{\text{in}}}{T_{b,\text{in}}} + \frac{\dot{Q}_{\text{out}}}{T_{b,\text{out}}}\right) \quad (\text{kW})$$

For a steady-flow system

$$X_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{b,\text{in}}} + \frac{q_{\text{out}}}{T_{b,\text{out}}}\right) \quad (\text{kJ/kg})$$

Steady-flow, one-inlet, one-exit

$$x_{\text{dest}} = T_0\left(\sum \frac{q_{\text{out}}}{T_{b,\text{out}}} - \sum \frac{q_{\text{in}}}{T_{b,\text{in}}}\right) \quad (\text{kJ/kg})$$

Exergy destruction of a cycle

$$x_{\text{dest}} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right) \quad (\text{kJ/kg})$$

For a cycle with heat transfer only with a source and a sink

$$\phi = (u - u_0) - T_0(s - s_0) + P_0(v - v_0) + \frac{V^2}{2} + gz$$

Closed system exergy

$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

Stream exergy

A second-law analysis of these cycles reveals where the largest irreversibilities occur and where to start improvements.

**Exp 1:** The compression ratio of an engine running on the ideal air standard dual cycle is 15. The maximum pressure in the cycle is 6500 kN/$m^2$, and the maximum temperature is 1950 K. At the start of compression, the temperature and pressure are 27 °C and 100 kN/$m^2$.

a) Find the thermal efficiency of the cycle.
b) Find the mean effective pressure of the cycle.

**Exp 2:** In an Otto cycle with a compression ratio r=8, the pressure and temperature at point 1 are 100 kPa and 300K, respectively. Since $M_a$=1 kg, $Q_i$=3000 kJ, find the net work and efficiency for the cycle.

# Summary

- Basic considerations in the analysis of power cycles
- The Carnot cycle and its value in engineering
- Air-standard sssumptions
- An overview of reciprocating engines
- Otto cycle: The ideal cycle for spark-ignition engines
- Diesel cycle: The ideal cycle for compression-ignition engines
- Stirling and Ericsson cycles
- Second-law analysis of gas power cycles