ENGINEERING THERMODYNAMICS -II

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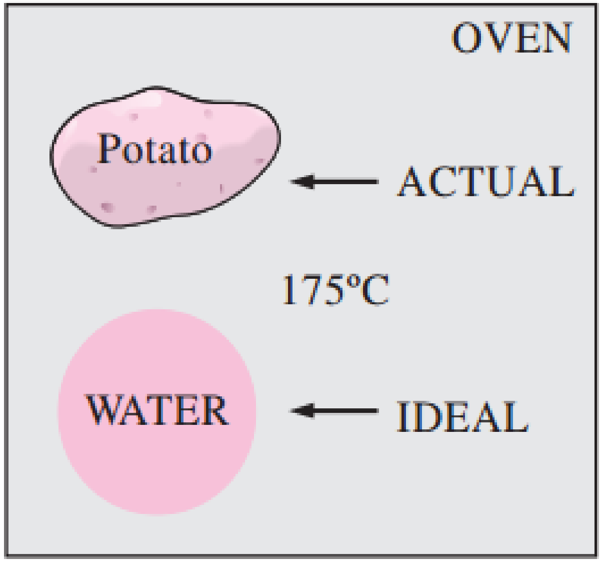

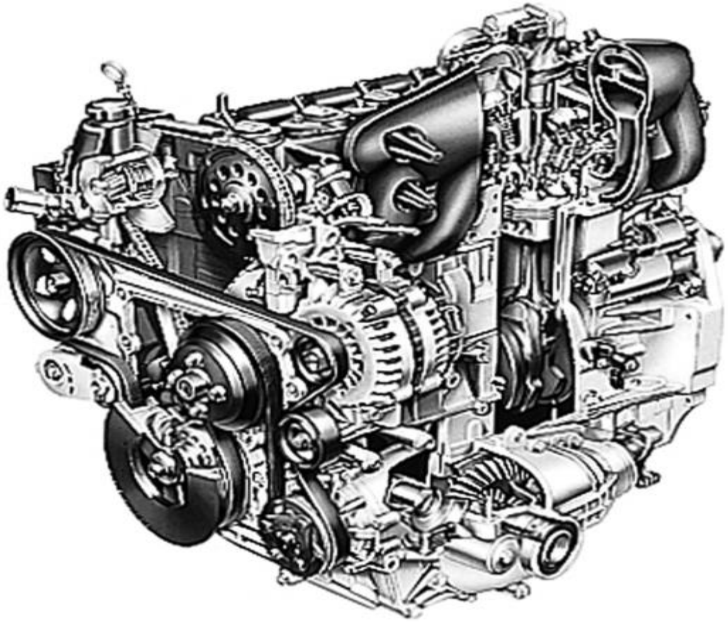

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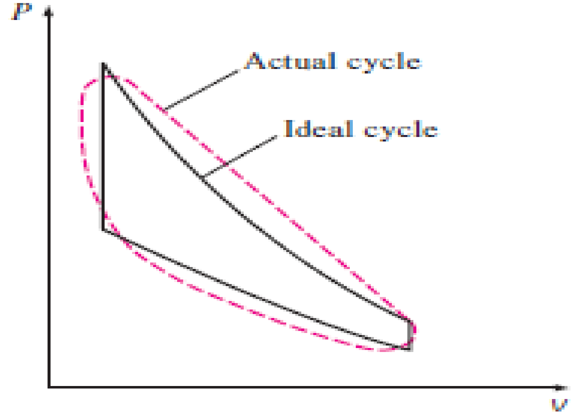
3. Air standard cycles

Objectives

- ✓ Evaluate the performance of gas power cycles for which the working fluid remains a gas throughout the entire cycle.
- ✓ Develop simplifying assumptions applicable to gas power cycles.
- ✓ Review the operation of reciprocating engines.
- ✓ Analyze both closed and open gas power cycles.
- ✓ Solve problems based on the Otto, Diesel, Stirling, and Ericsson cycles.
- ✓ Solve problems based on the Brayton cycle; the Brayton cycle with regeneration; and the Brayton cycle with intercooling, reheating, and regeneration.
- ✓ Analyze jet-propulsion cycles.
- ✓ Identify simplifying assumptions for second-law analysis of gas power cycles.
- ✓ Perform second-law analysis of gas power cycles.

BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

- ❖ Gas power cycles will involves study those heat engine in which the working fluid remains in gaseous state through out the cycle.
- ❖ **Ideal cycle:** A cycle that resembles the actual cycle closely but is made up totally of internally reversible processes.



- ❖ Heat engines are designed for the purpose of converting thermal energy to work, and their performance is expressed in terms of the **thermal efficiency** η_{th}, which is the ratio of the net work produced by the engine to the total heat input:

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

- The idealizations and simplifications commonly employed in the analysis of power cycles can be summarized as follows:

1. The cycle does not involve any *friction*.

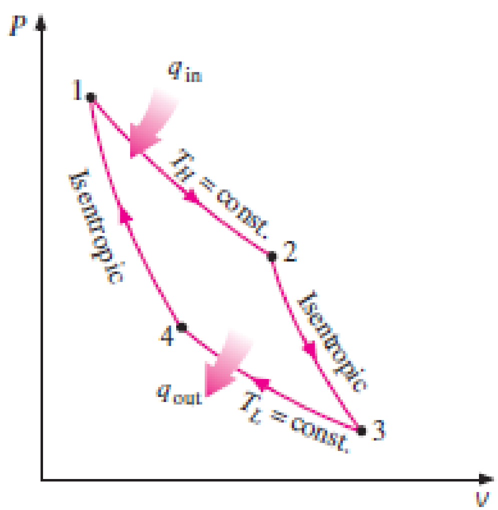
   Therefore, *the working fluid* does not experience any pressure drop as it flows in pipes or devices such as heat exchangers.
2. All expansion and compression processes take place in a *quasiequilibrium* manner.
3. The pipes connecting the various components of a system are well insulated, and *heat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

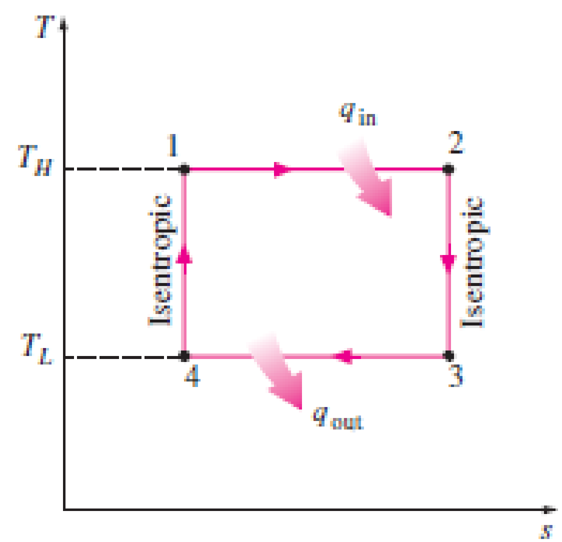
The Carnot Cycle and Its Value in Engineering

- Carnot yields maximum efficiency

  $\eta_{Carnot} = 1 - TL/TH$

- Executed in closed system or open steady-flow device
- Efficiency increases with increasing/decreasing high/low temperature
- Reversible isothermal heat transfer not practical





AIR-STANDARD ASSUMPTIONS

To reduce the analysis to a manageable level, we utilize the following approximations, commonly known as the **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 (Fig. below).
4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.
5. cold-air-standard assumption apply when the working fluid is air and has constant specific heats evaluate at *room temperature (25°C, or 77°F).*

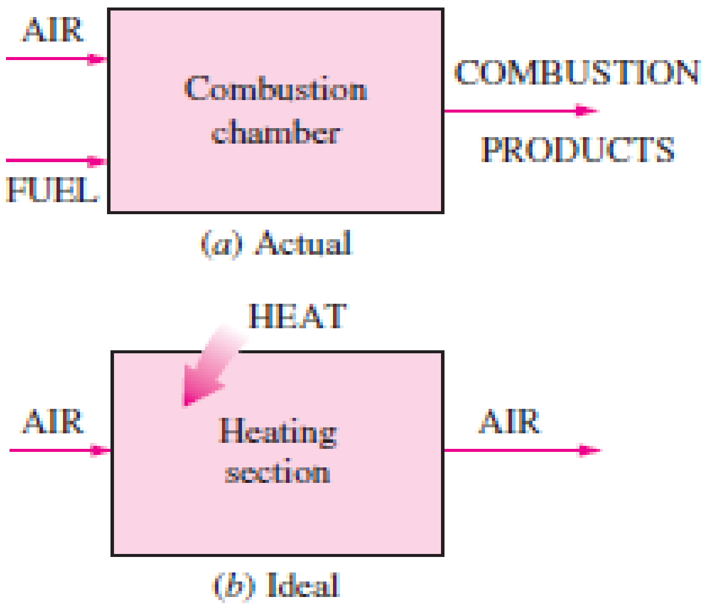


Fig: The combustion process is replaced by a heat-addition process in ideal cycles.

TERMINOLOGY FOR RECIPROCATING ENGINES

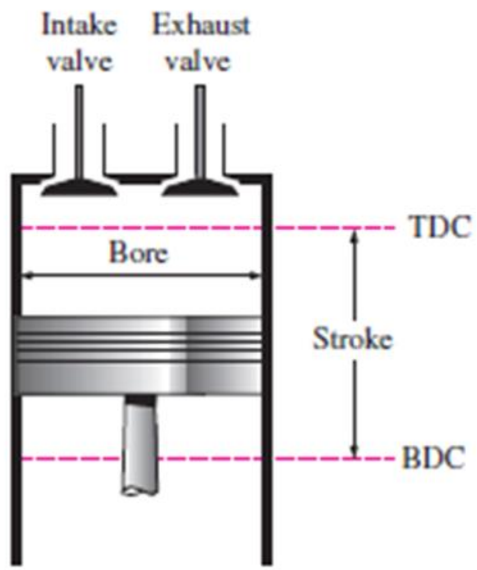
The following is terminology we need to understand for reciprocating engines (basically a piston–cylinder device)

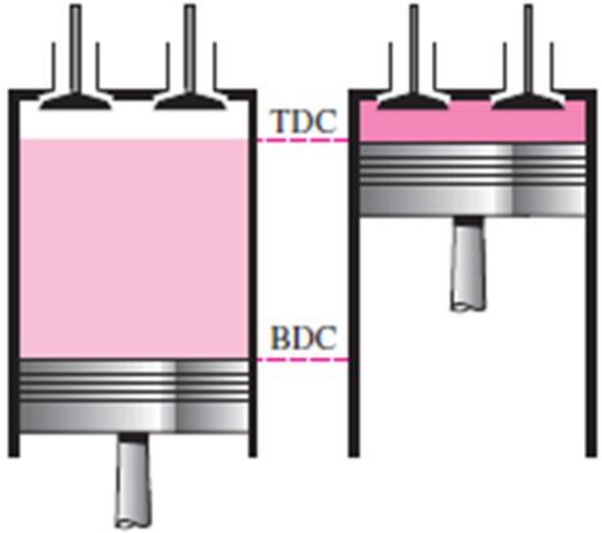
- The piston reciprocates in the cylinder between two fixed positions called the **top dead center (TDC)—**the position of the piston when it forms the smallest volume in the cylinder—and the **bottom dead center (BDC)—**the position of the piston when it forms the largest volume in the cylinder.
- The distance between the TDC and the BDC is the largest distance that the piston can travel in one direction, and it is called the **stroke of the engine.**
- The diameter of the piston is called the **bore.**
- The air or air–fuel mixture is drawn into the cylinder through the **intake valve.**
- the combustion products are expelled from the cylinder through the **exhaust valve.**
- The minimum volume formed in the cylinder when the piston is at TDC is called the **clearance volume.**

TERMINOLOGY FOR RECIPROCATING ENGINES

- The volume displaced by the piston as it moves between TDC and BDC is called **the displacement volume.**
- The ratio of the maximum volume formed in the cylinder to the minimum (clearance) volume is **called the compression ratio r of the engine:**

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





(*a*) Displacement volume

(*b*) Clearance volume

TERMINOLOGY FOR RECIPROCATING ENGINES

- **mean effective pressure (MEP)** is a fictitious pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle.
- *Wnet = MEP X Piston area Stroke = MEP X Displacement volume*

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

▪The mean effective pressure can be used as a parameter to compare the performances of reciprocating engines of equal size.

▪The engine with a larger value of MEP delivers more net work per cycle and thus performs better.

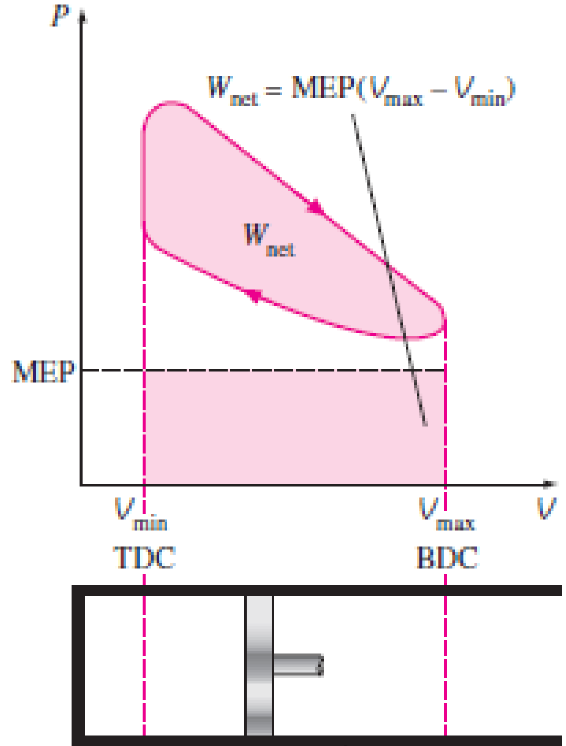


Fig: The net work output of a cycle is equivalent to the product of the mean effective pressure and the displacement volume.

TERMINOLOGY FOR RECIPROCATING ENGINES

Reciprocating engines are classified as spark-ignition (SI) engines or compression-ignition (CI) engines, depending on how the combustion process in the cylinder is initiated.

- In SI engines, the combustion of the air–fuel mixture is initiated by a spark plug.
- ➢ *Otto a cycles which is the ideal cycles for the SI reciprocating* engine.
- In CI engines, the air–fuel mixture is self-ignited as a result of compressing the mixture above its self ignition temperature.
- ➢ *Diesel cycles, which are the ideal cycles for the CI reciprocating* engine.

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

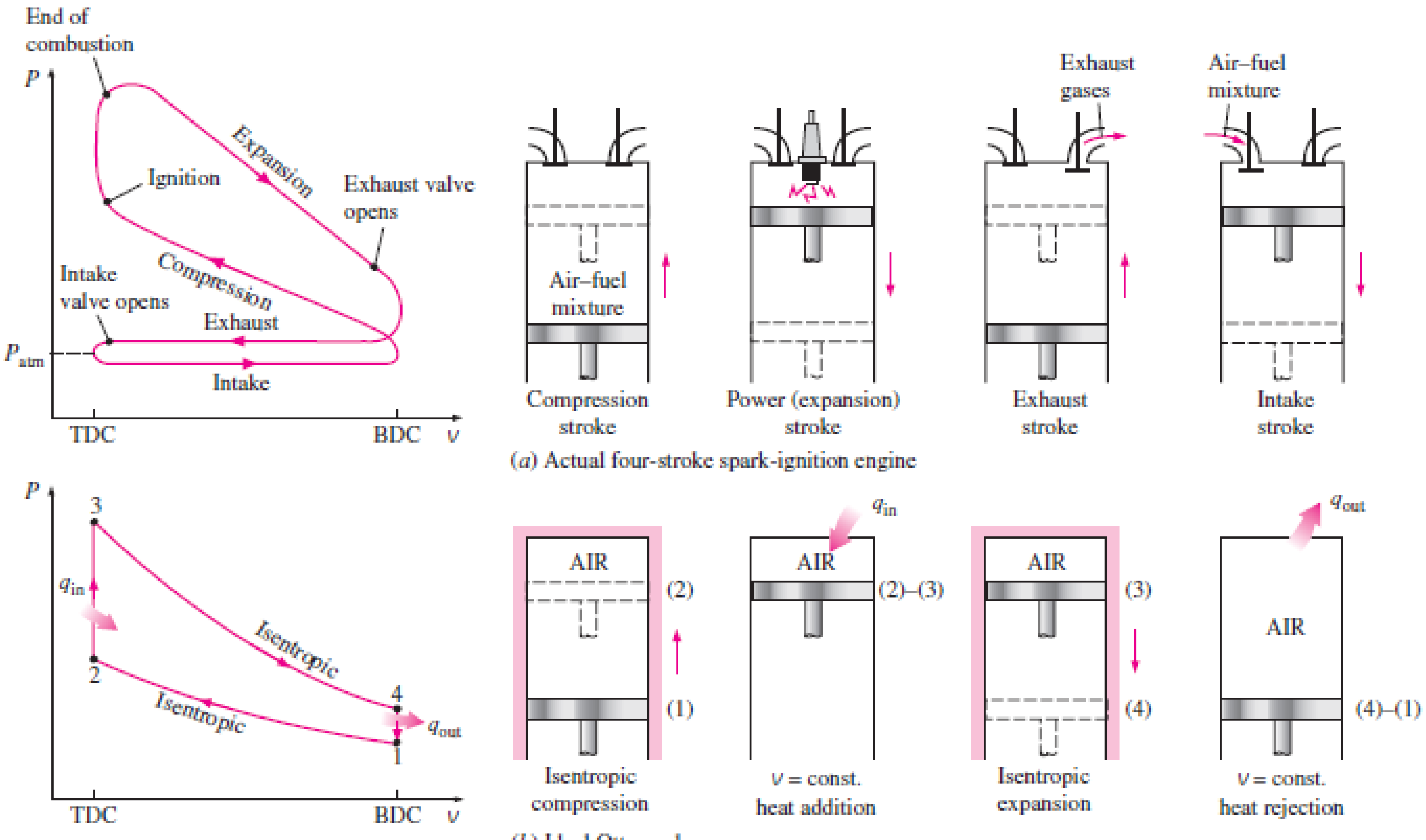
- In most spark-ignition engines, the piston executes four complete strokes (two mechanical cycles) within the cylinder, and the crankshaft completes two revolutions for each thermodynamic cycle.
- These engines are called **four-stroke internal combustion engines.**

These process:

- ✓ **Intake stroke**
- ✓ **Compression stroke**
- ✓ **Expansion stroke**
- ✓ **Exhaust stroke**

- Often the ignition and combustion process begins before the completion the compression stroke.
- The number of crank angle degrees before the piston reaches TDC on the number one piston at which spark occurs is called the engine timing.

OTTO CYCLE



(a) Actual four-stroke spark-ignition engine

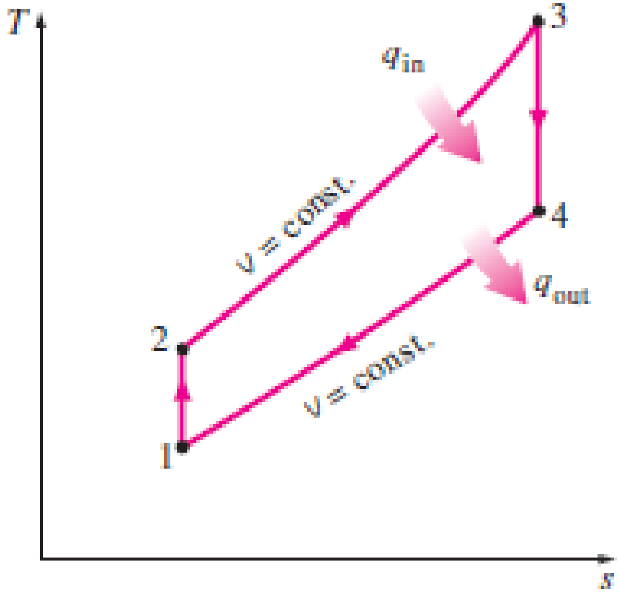
(b) Ideal Otto cycle

OTTO CYCLE

Air-Standard Otto cycle is the ideal cycle that approximation the spark ignition engine.

It consists of four internally reversible processes:

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



First Law Analysis of Otto Cycle

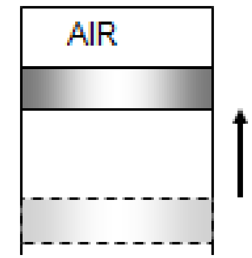
1→2 Isentropic Compression

$$(u_2 - u_1) = \cancel{\frac{Q}{m}} - \left(-\frac{W_{in}}{m}\right)$$

$$\frac{W_{in}}{m} = (u_2 - u_1) = c_v (T_2 - T_1)$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1} = r^{k-1}$$

$$\frac{P_2}{P_1} = \frac{T_2}{T_1} \cdot \frac{v_1}{v_2}$$



Where:
Compression ratio:

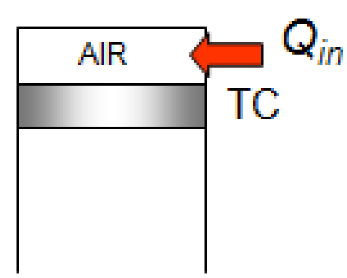
$$r = \frac{v_1}{v_2} = \frac{v_4}{v_3}$$

2→3 Constant Volume Heat Addition

$$(u_3 - u_2) = \left(+\frac{Q_{in}}{m}\right) - \cancel{\frac{W}{m}}$$

$$\frac{Q_{in}}{m} = (u_3 - u_2) = c_v (T_3 - T_2)$$

$$\frac{P_3}{P_2} = \frac{T_3}{T_2}$$



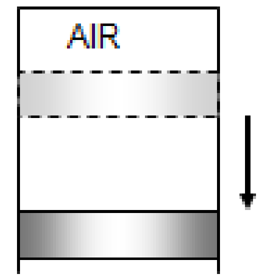
First Law Analysis of Otto Cycle

- 3 → 4 Isentropic Expansion

$$(u_4 - u_3) = \frac{\cancel{Q}}{m} - (+\frac{W_{out}}{m})$$

$$\frac{W_{out}}{m} = (u_3 - u_4) = c_v(T_3 - T_4)$$

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

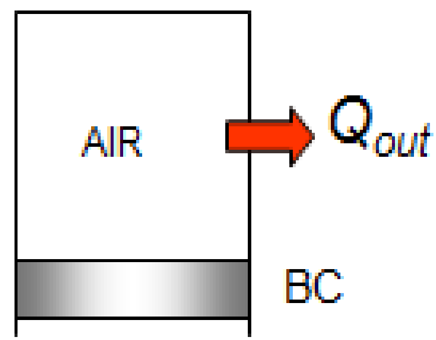


$$\frac{P_4}{P_3} = \frac{T_4}{T_3} \cdot \frac{v_3}{v_4}$$

- 4 → 1 Constant Volume Heat Removal

$$(u_1 - u_4) = (-\frac{Q_{out}}{m}) - \frac{\cancel{W}}{m}$$

$$\frac{Q_{out}}{m} = (u_4 - u_1) = c_v(T_4 - T_1)$$



$$\frac{P_4}{T_4} = \frac{P_1}{T_1}$$

First Law Analysis of Otto Cycle

Net cycle work:

$$W_{cycle} = W_{out} - W_{in} = m(u_3 - u_4) - m(u_2 - u_1)$$

Cycle thermal efficiency:

$$\eta_{th} = \frac{W_{cycle}}{Q_{in}} = \frac{(u_3 - u_4) - (u_2 - u_1)}{(u_3 - u_2)} = \frac{(u_3 - u_2) - (u_4 - u_1)}{u_3 - u_2} = 1 - \frac{u_4 - u_1}{u_3 - u_2}$$

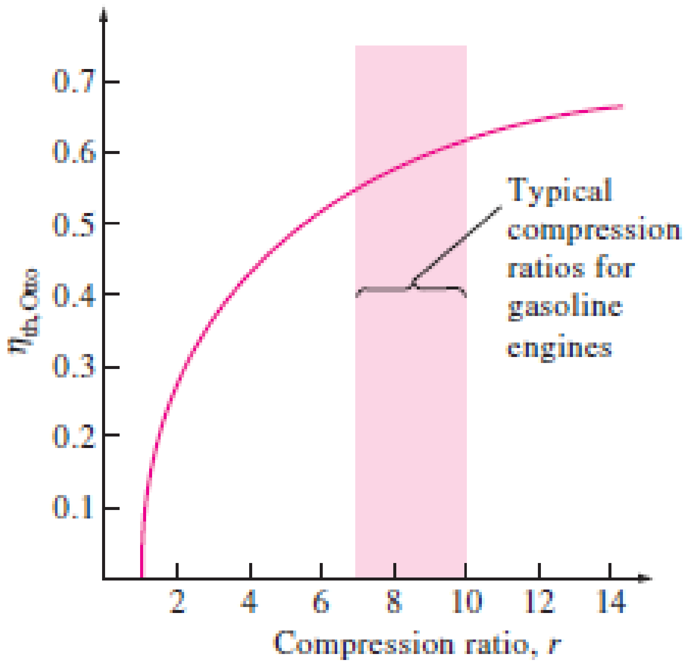
$$= 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{r^{k-1}}$$

Indicated mean effective pressure is:

$$imep = \frac{W_{cycle}}{V_1 - V_2} \rightarrow \frac{imep}{P_1} = \frac{Q_{in}}{P_1 V_1}\left(\frac{r}{r-1}\right)\eta_{th} = \frac{1}{k-1}\left(\frac{Q_{in}/m}{u_1}\right)\left(\frac{r}{r-1}\right)\eta_{th}$$

OTTO CYCLE

❖ Effect of Compression Ratio on Thermal Efficiency



$$\eta_{th \atop const\ c_V} = 1 - \frac{1}{r^{k-1}}$$

k = 1.4

- Spark ignition engine compression ratio limited by T_3 (auto ignition) and P_3 (material strength), both $\sim r^k$
- For r = 8 the efficiency is 56% which is twice the actual indicated value

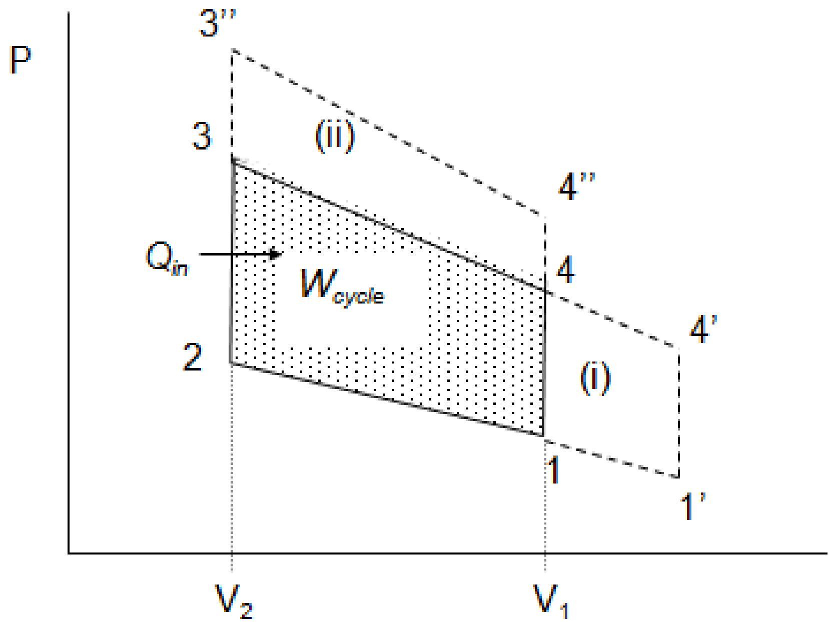
OTTO CYCLE

❖ Factors Affecting Work per Cycle

The net cycle work of an engine can be increased by either:

i) Increasing the r (1'→2)

ii) Increase Q_{in} (2→3")



$$imep = \frac{W_{cycle}}{V_1 - V_2} = \frac{Q_{in}}{V_1}\left(\frac{r}{r-1}\right)\eta_{th}$$

Example:1 An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 100 kPa and 17°C, and 800 kJ/kg of heat is transferred to air during the constant-volume heat-addition process. Accounting for the variation of specific heats of air with temperature, determine (*a) the maximum temperature and pressure that occur during the cycle*, (*b) the net work output*, *(c) the thermal efficiency*, *and (d) the mean effective* pressure for the cycle.

Assignment: 1

An Otto cycle having a compression of 9:1 uses air as the working fluid . Initially P_1 = 95KPa, T_1 = 17^{o}c, and V_1 = 3.8liters . During the heat addition , 7.5KJ of heat are added. Determine all T's, P's, η_{th}, the back work ration, and the effective pressure.

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

IN CI engines (also known as *diesel engines):*

- the air is compressed to a temperature that is above the auto ignition temperature of the fuel.
- combustion starts on contact as the fuel is injected into this hot air.
- Spark plug is replaced by injector (fine droplets)

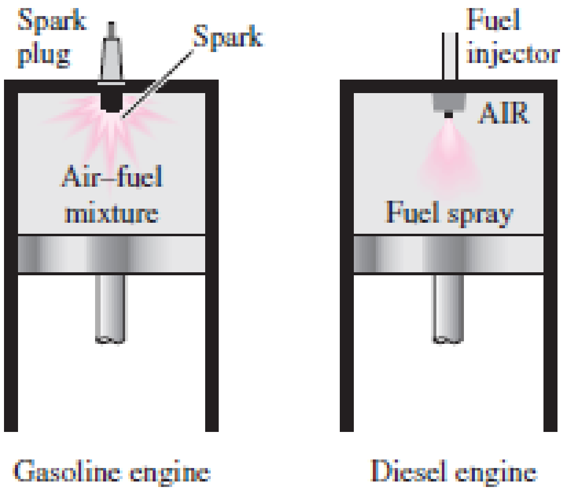
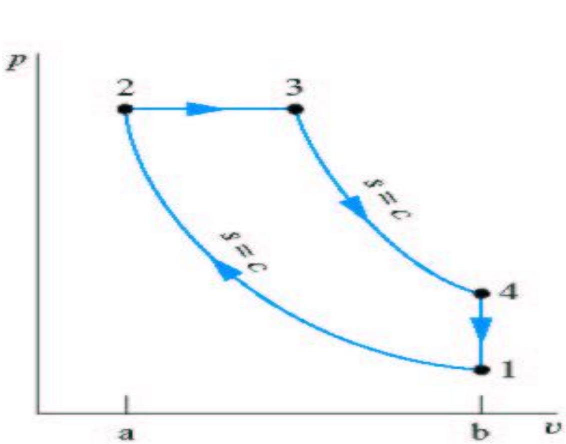


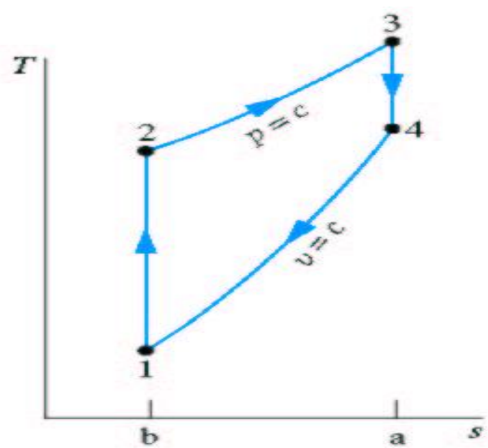
Fig: In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

DIESEL CYCLE

Air-Standard diesel cycle is the ideal cycle that approximation the spark ignition engine.

- ❖ Process 1→ 2 Isentropic compression
- ❖ Process 2 → 3 Constant pressure heat addition
- ❖ Process 3 → 4 Isentropic expansion
- ❖ Process 4 → 1 Constant volume heat rejection





Cut-off ratio r_c:

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c$$

cutoff ratio *rc,* as the ratio of the cylinder volumes after and before the combustion process:

Apply first law closed system for Diesel cycle (piston–cylinder device)

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

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

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

DIESEL CYCLE

cutoff ratio *rc*, *is the ratio of the cylinder* volumes after and before the combustion process

- Is a measure of the duration of the heat addition at constant pressure.
- Since the fuel is injected directly into the cylinder, the cutoff can be related to the number of degrees that crank rotated during the fuel injection into the cylinder.

❖For cold air-standard the above reduces to:

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

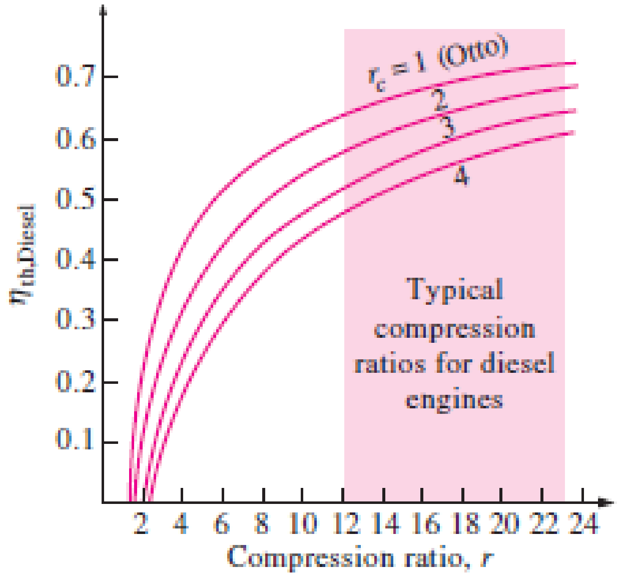
Note the term in the square bracket is always larger than one so for the same compression ratio, *r*, the Diesel cycle has a *lower* thermal efficiency than the Otto cycle

Note: CI needs higher *r* compared to SI to ignite fuel

DIESEL CYCLE

- When r_c (= v_3/v_2)→1 the Diesel cycle efficiency approaches the efficiency of the Otto cycle.
- When $r_c > 1$ for fixed r, $\eta_{th,Diesel} < \eta_{th,Otto}$. But, since $r_{Diesel} > r_{Otto}$,

$\eta_{th,Diesel} > \eta_{th,Otto}$



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

Example:2

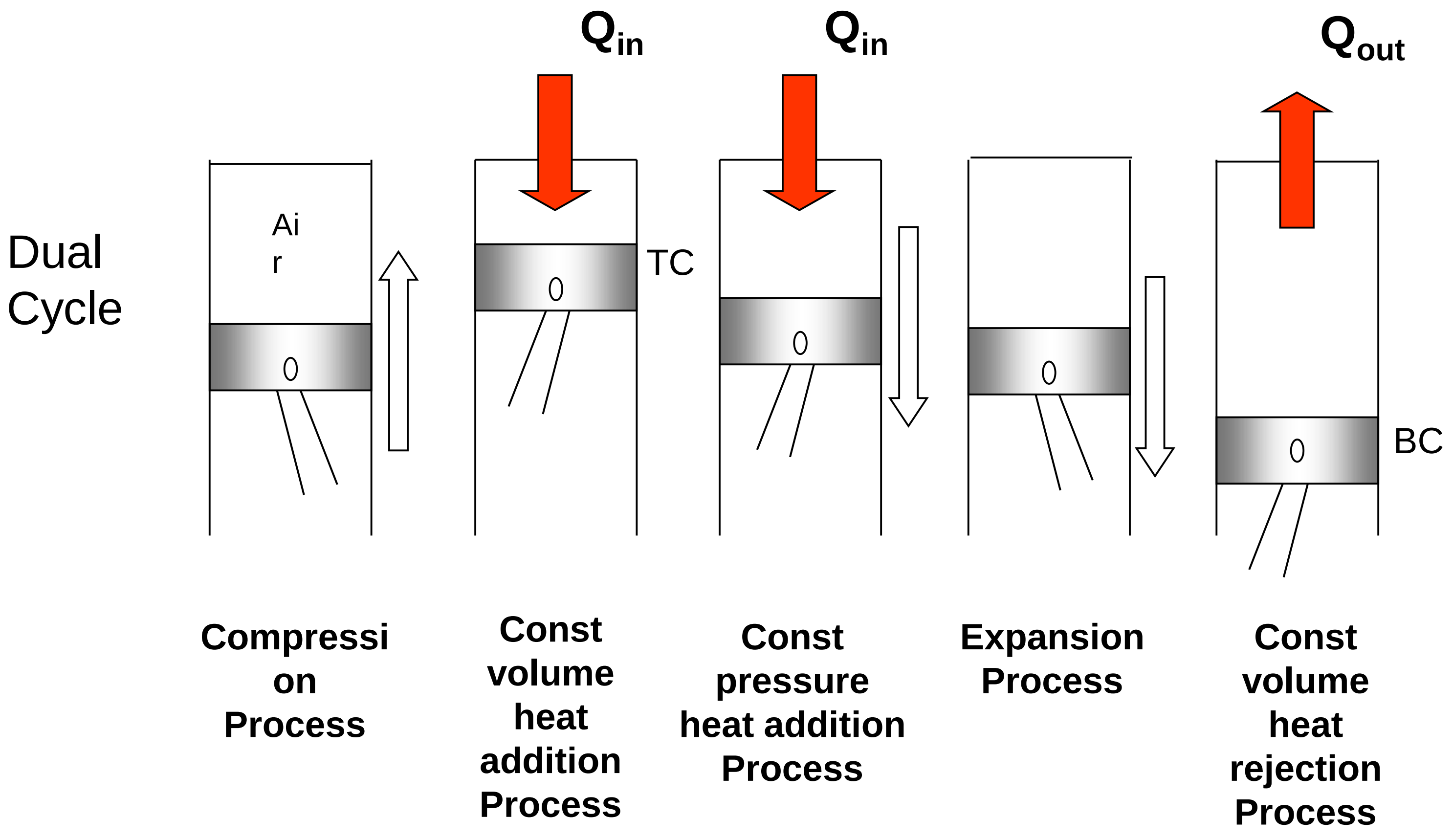
An air-standard Diesel cycle has a compression ratio of 16 and a cutoff ratio of 2. At the beginning of the compression process, air is at 95kPa and 27°C. Accounting for the variation of specific heats with temperature, determine (*a) the temperature after the heat-addition process*, *(b) the* thermal efficiency, and (*c) the mean effective pressure.*

Assignment :2

An ideal diesel engine has a compression ratio of 20 and uses air as the working fluid. The state of air at the beginning of the compression process is 95kPa and 20°C. If the maximum temperature in the cycle is not to exceed 2200K, determine (*a) the thermal efficiency and (b) the mean* effective pressure. Assume constant specific heats for air at room temperature.

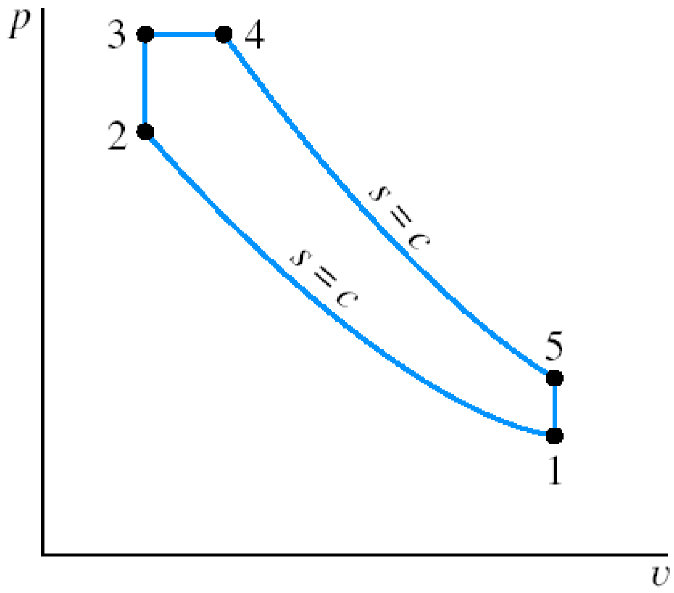
Dual cycle

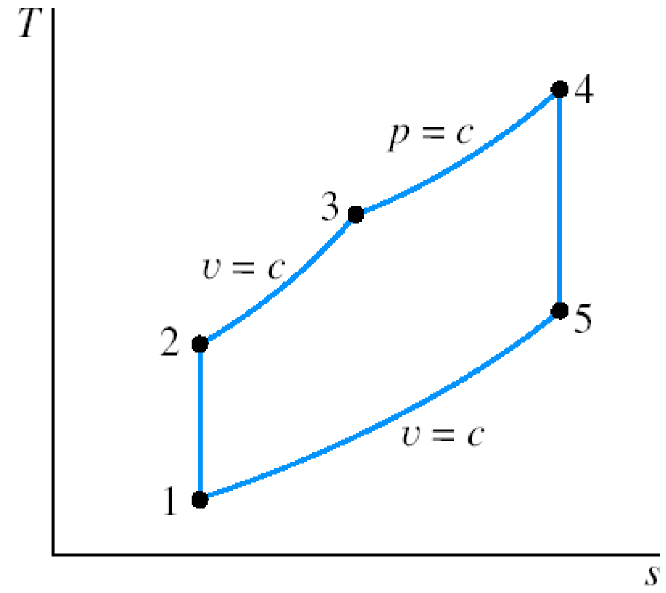
➢Is model the combustion process in both gasoline and diesel engines as a combination of two heat-transfer processes, one at constant volume and the other at constant pressure.



Dual cycle

- Process 1 → 2 Isentropic compression
- Process 2 → 2.5 Constant volume heat addition
- Process 2.5 → 3 Constant pressure heat addition
- Process 3 → 4 Isentropic expansion
- Process 4 → 1 Constant volume heat rejection





$$\frac{Q_{in}}{m} = (u_3 - u_2) + (h4 - h_3) = c_v(T_3 - T_2) + c_p(T_3 - T_4)$$

Dual cycle

- **Thermal Efficiency**

$$\eta_{\substack{Dual \\ cycle}} = 1 - \frac{Q_{out}/m}{Q_{in}/m} = 1 - \frac{u_4 - u_1}{(u_{2.5} - u_2) + (h_3 - h_{2.5})}$$

$$\eta_{\substack{Dual \\ const\, c_V}} = 1 - \frac{1}{r^{k-1}}\left[\frac{\alpha r_c^k - 1}{(\alpha - 1) + \alpha k (r_c - 1)}\right]$$

$$\text{where } r_c = {}^{v_3}\!/\!_{v_{2.5}} \text{ and } \alpha = {}^{P_3}\!/\!_{P_2}$$

Note, the Otto cycle (r_c=1) and the Diesel cycle (α=1) are special cases:

$$\eta_{Otto} = 1 - \frac{1}{r^{k-1}} \qquad \eta_{\substack{Diesel \\ const\, c_V}} = 1 - \frac{1}{r^{k-1}}\left[\frac{1}{k} \cdot \frac{\left(r_c^k - 1\right)}{\left(r_c - 1\right)}\right]$$

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

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

➢For the same inlet conditions P_1, V_1 and the same peak pressure P_3 (actual design limitation in engines):

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

Assignment :3

The compression ratio of an ideal dual cycle is 14. Air is at 100kPa and 300 K at the beginning of the compression process and at 2200 K at the end of the heat-addition process. Heat transfer to air takes place partly at constant volume and partly at constant pressure, and it amounts to 1520.4 kJ/kg. Assuming variable specific heats for air, determine (*a) the fraction of heat transferred at constant* volume and (*b) the thermal efficiency of the cycle.*

Stirling and Ericsson cycles

There are two other cycles that involve an isothermal heat-addition process at *TH and an isothermal heat-rejection process at TL: the Stirling cycle and* the *Ericsson cycle.*

- ➢ *They differ from the Carnot cycle in that the two isentropic* processes are replaced:
- ❖ by two constant-volume regeneration processes in the Stirling cycle, and
- ❖ by two constant-pressure regeneration processes in the Ericsson cycle.
- Both cycles utilize **regeneration,** a process during which heat is transferred to a thermal energy storage device (called a *regenerator)* during one part of the cycle and is transferred back to the working fluid during another part of the cycle
- ❖ Figure below (*b) shows the T-s and P-v diagrams of the* ***Stirling cycle,*** which is made up of four totally reversible processes:

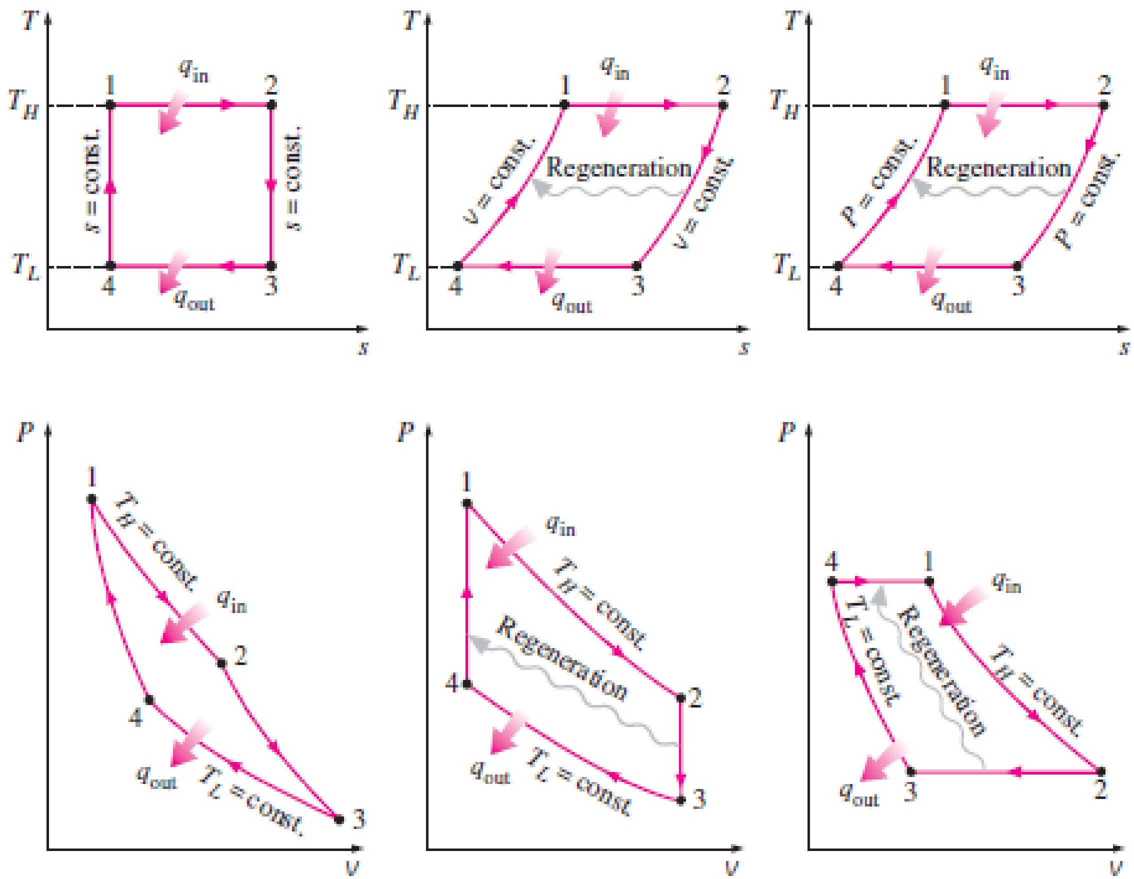
1-2 *T= constant expansion (heat addition from the external source)*

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)The execution of the Stirling cycle requires rather

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



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

Assignment .4

a. Consider an ideal Ericsson cycle with air as the working fluid executed in a steady-flow system. Air is at 27°C and 120 kPa at the beginning of the isothermal compression process, during which 150 kJ/kg of heat is rejected. Heat transfer to air occurs at 1200 K. Determine (*a) the maximum* pressure in the cycle, (*b) the net work output per unit mass of* air, and (*c) the thermal efficiency of the cycle*

b. An ideal Stirling engine using helium as the working fluid operates between temperature limits of 300 and 2000 K and pressure limits of 150 kPa and 3 MPa. Assuming the mass of the helium used in the cycle is 0.12 kg, determine (*a) the* thermal efficiency of the cycle, (*b) the amount of heat transfer* in the regenerator, and (*c) the work output per cycle.*

Brayton Cycle: Ideal Cycle for Gas-Turbine Engines

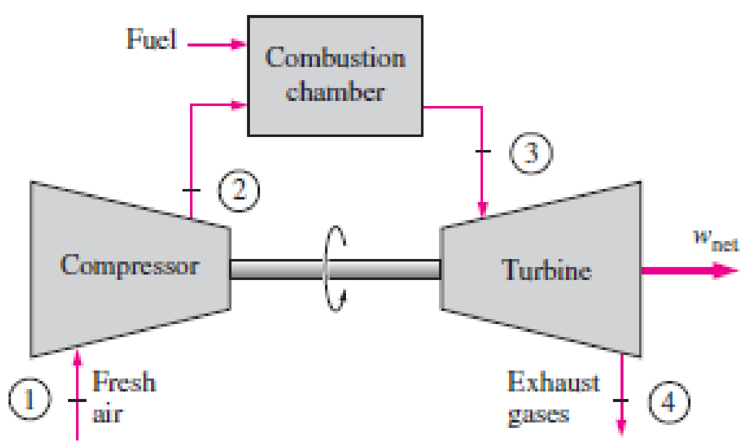
Gas turbines usually operate on an open cycle (fig below)

Working Principal:

- ✓ Fresh air enters the compressor at ambient temperature where its pressure and temperature are increased.
- ✓ The high pressure air enters the combustion chamber where the fuel is burned at constant pressure.
- ✓ The high temperature (and pressure) gas enters the turbine where it expands to ambient pressure and produces work.

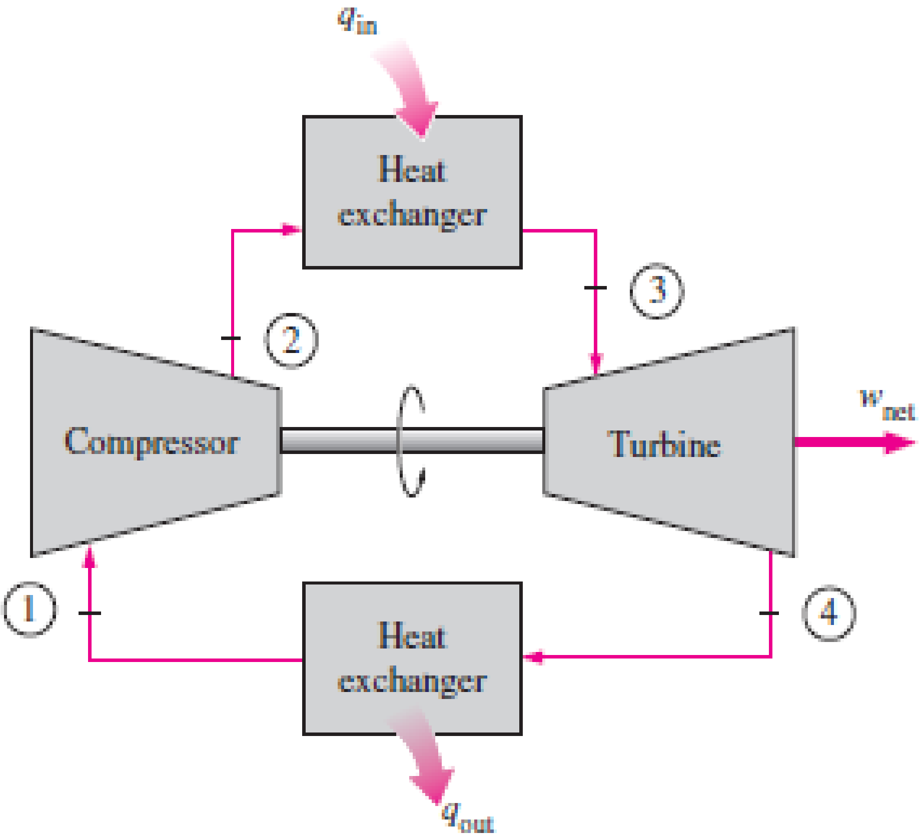
Some of the output power is used to drive the compressor.

- The exhaust gases leaving the turbine are thrown out (not re-circulated), causing the cycle to be classified as an **open cycle**.



Brayton Cycle: Closed Cycle Model

- The open gas-turbine cycle can be modelled as a closed cycle, using the **air-standard** assumptions (Fig. below).
- The compression and expansion processes remain the same, but the combustion process is replaced by a **constant-pressure heat addition** process from an external source.
- The exhaust process is replaced by a **constant-pressure heat rejection** process to the ambient air.

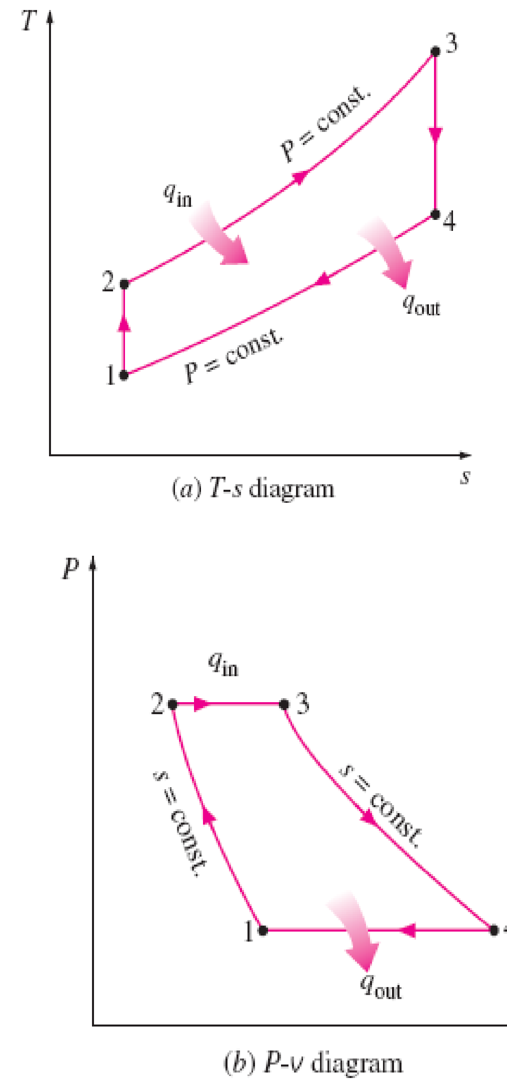


Brayton Cycle :processes

The ideal cycle that the working fluid undergoes in the closed loop is the **Brayton cycle.** It is made up of four internally reversible processes:

- **1-2** Isentropic compression;
- **2-3** Constant-pressure heat addition;
- **3-4** Isentropic expansion;
- **4-1** Constant-pressure heat rejection.

The *T-s* and *P-v* diagrams of an ideal Brayton cycle are shown in below.

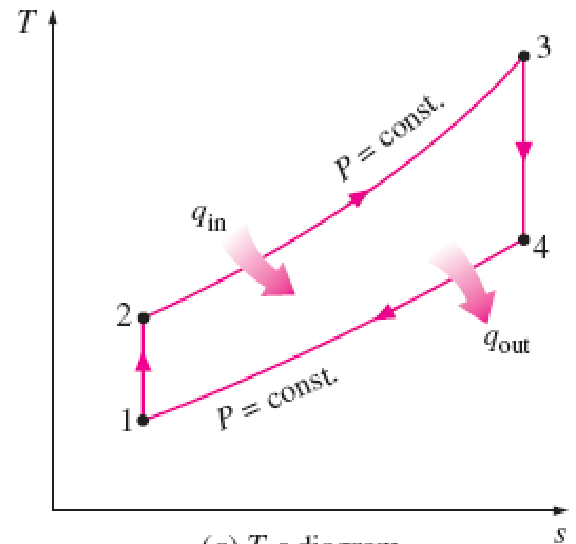


Note: All four processes of the Brayton cycle are executed in steady-flow devices thus, they should be analyzed as **steady-flow processes**.

Brayton Cycle : Thermal Efficiency

The **energy balance** for a steady-flow process can be expressed, on a unit–mass basis, as

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



(a) T-s diagram

The **heat transfers** to and from the working fluid are:

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2)$$

$$q_{\text{out}} = h_4 - h_1 = c_p(T_4 - T_1)$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k} = \frac{T_3}{T_4}$$

The **thermal efficiency** of the ideal Brayton cycle,

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

$$\eta_{\text{th,Brayton}} = 1 - \frac{1}{r_p^{(k-1)/k}}$$ Constant specific heats

where $r_p = \dfrac{P_2}{P_1}$ is the **pressure ratio**.

Example: 3

A simple **Brayton cycle** using air as the working fluid has a pressure ratio of 8. The minimum and maximum temperatures in the cycle are 310 K and 1160 K, respectively. Assuming an isentropic efficiency of 75 percent for the compressor and 82 percent for the turbine, determine:

a) the **air temperature** at the turbine exit,

b) the **net work output**, and

c) the **thermal efficiency**.

Assignment: 5

A stationary gas-turbine power plant operates on a simple ideal Brayton cycle with air as the working fluid. The air enters the compressor at 95kPa and 290 K and the turbine at 760kPa and 1100 K. Heat is transferred to air at a rate of 35,000 kJ/s. Determine the power delivered by this plant (*a) assuming constant specific heats at room temperature* and (*b) accounting for the variation of specific heats* with temperature.

Brayton Cycle: Parameters Affecting Thermal Efficiency

❖The thermal efficiency of an ideal Brayton cycle depends on the pressure ratio, r_p of the gas turbine and the specific heat ratio, k of the working fluid.

❖The thermal efficiency increases with both of these parameters, which is also the case for actual gas turbines.

❖A plot of thermal efficiency versus the pressure ratio is shown in Fig. below, for the case of k =1.4.

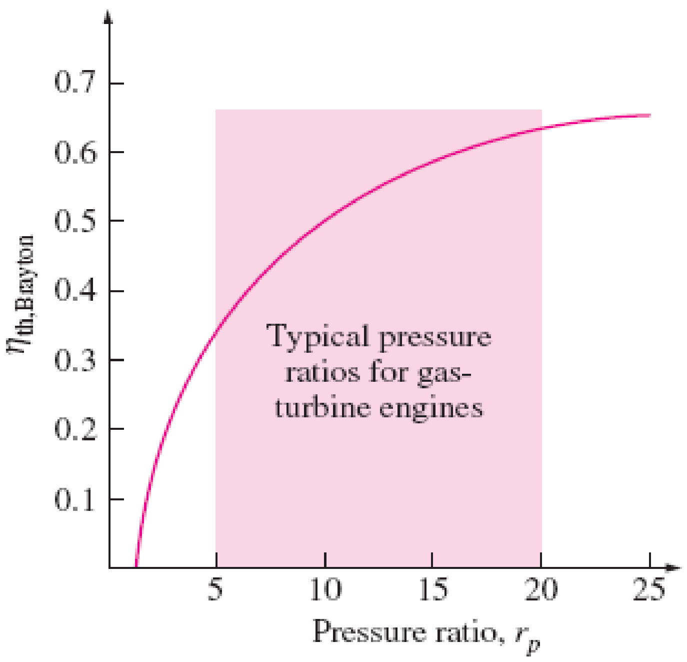
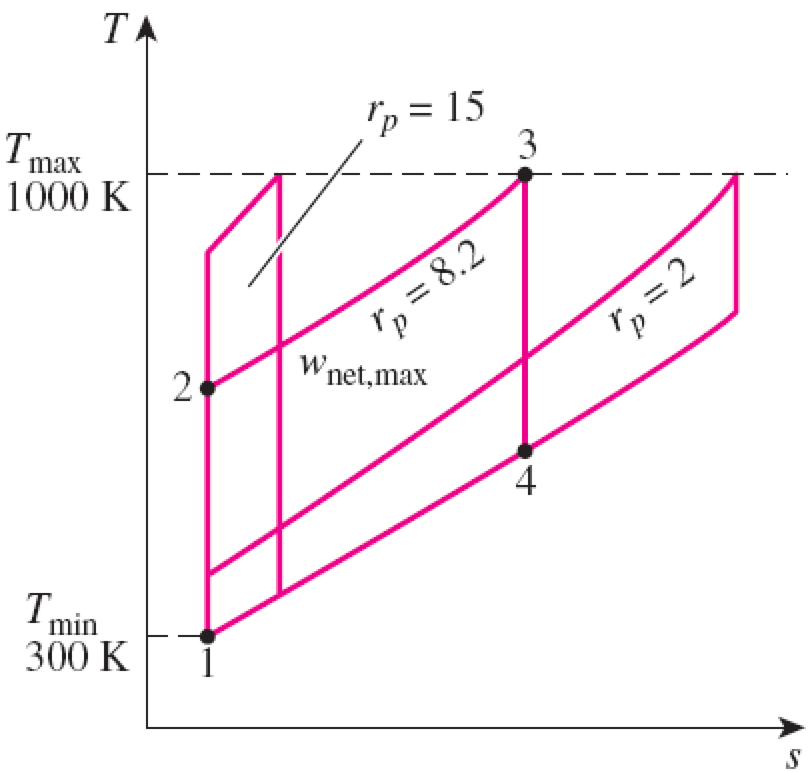


Fig: Thermal efficiency of the ideal Brayton cycle as a function of the pressure ratio.

The two major application areas of gas-turbine engines are *aircraft propulsion* and *electric power generation*.

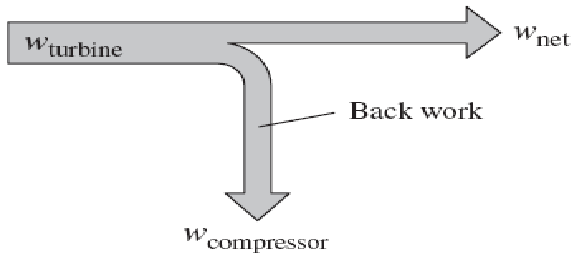


For fixed values of T_{min} and T_{max}, the net work of the Brayton cycle first increases with the pressure ratio, then reaches a maximum at $r_p = (T_{max}/T_{min})^{k/[2(k-1)]}$, and finally decreases.

❖The highest temperature in the cycle is limited by the maximum temperature that the turbine blades can withstand. This also limits the pressure ratios that can be used in the cycle.

❖The air in gas turbines supplies the necessary oxidant for the combustion of the fuel, and it serves as a coolant to keep the temperature of various components within safe limits. An air–fuel ratio of 50 or above is not uncommon.

Back Work Ratio



➢The fraction of the turbine work used to drive the compressor is called the back work ratio.

➢BWR is defined as the ratio of compressor work to the turbine work.

➢The BWR in gas turbine power plant is very high, normally one-half of turbine work output is used to drive the compressor.

$$r_{bw} = \frac{w_{compressor}}{w_{turbine}}$$

❖work ratio is the fraction of the turbine work that becomes the net work.

$$r_w = w_{net}/w_{turbine}$$

Brayton Cycle: Actual Gas-Turbine Cycles

- Some **pressure drop** occurs during the heat-addition and heat rejection processes.
- The actual work input to the compressor is more, and the actual work output from the turbine is less, because of **irreversibilities**.

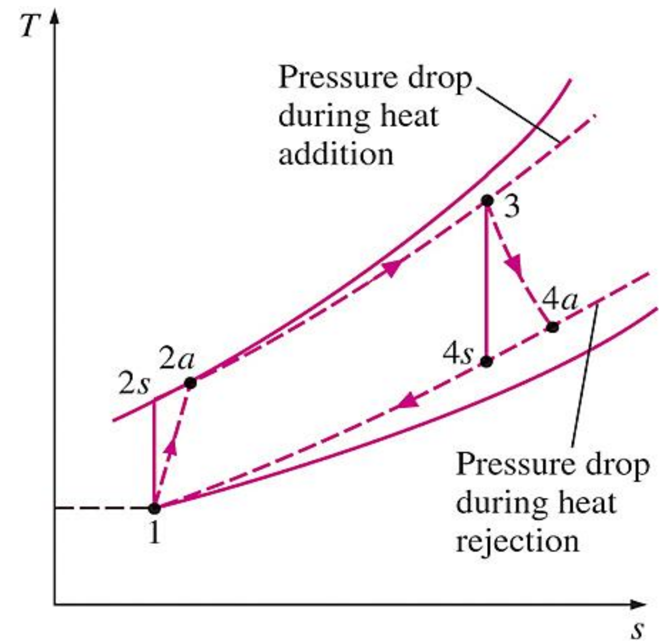


Fig: The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

- Deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accounted for by utilizing **isentropic efficiencies** of the turbine and compressor.

Turbine: $$\eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

Compressor: $$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

Brayton Cycle: Improvements of Gas Turbine's Performance

The early gas turbines (1940s to 1959s) found only limited use despite their versatility and their ability to burn a variety of fuels, because its thermal efficiency was only about 17%. Efforts to improve the cycle efficiency are concentrated in three areas:

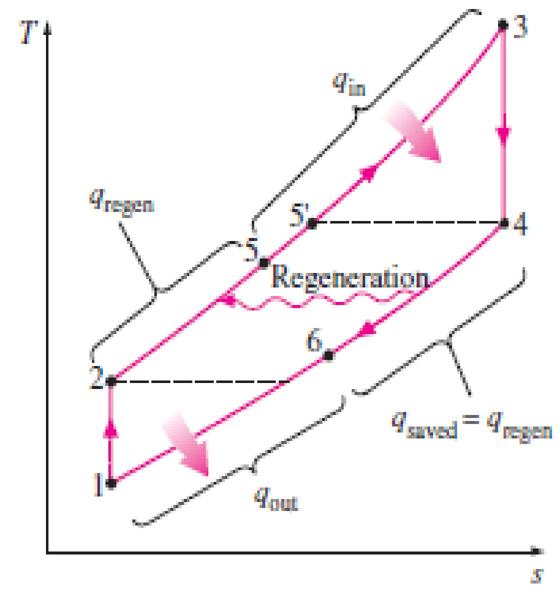
1. **Increasing the turbine inlet (or firing) temperatures.**
   - The turbine inlet temperatures have increased steadily from about 540°C (1000°F) in the 1940s to 1425°C (2600°F) and even higher today.
2. **Increasing the efficiencies of turbo-machinery components (turbines, compressors).**
   - The advent of computers and advanced techniques for computer-aided design made it possible to design these components aerodynamically with minimal losses.
3. **Adding modifications to the basic cycle (intercooling, regeneration or recuperation, and reheating).**
   - The simple-cycle efficiencies of early gas turbines were practically doubled by incorporating intercooling, regeneration (or recuperation), and reheating.

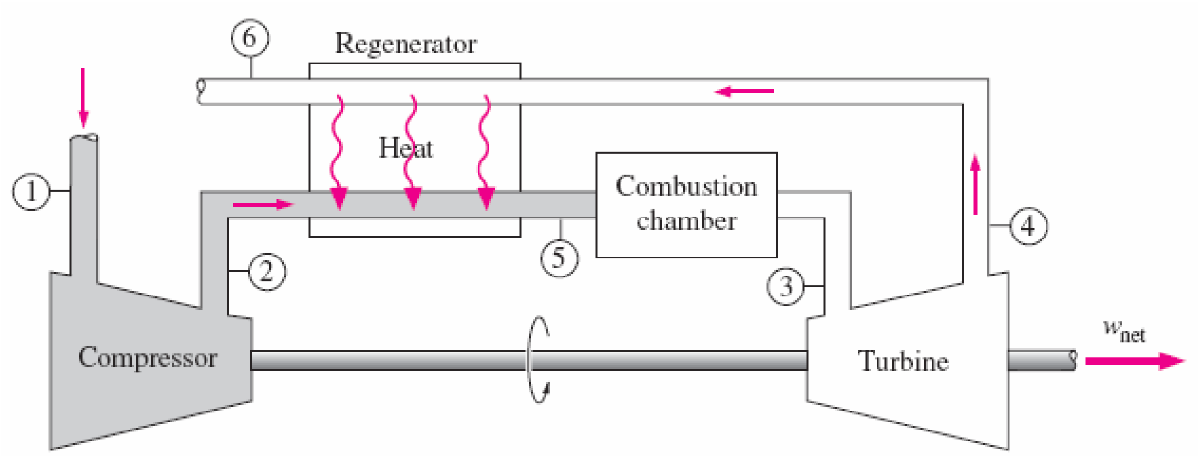
Brayton Cycle With Regeneration

➢Temperature of the exhaust gas leaving the turbine is **higher** than the temperature of the air leaving the compressor.

➢The air leaving the compressor can be heated by the hot exhaust gases in a **counter-flow** heat exchanger (a *regenerator* or *recuperator*) – a process called **regeneration**

➢The thermal efficiency of the Brayton cycle **increases** due to regeneration since less fuel is used for the same work output.





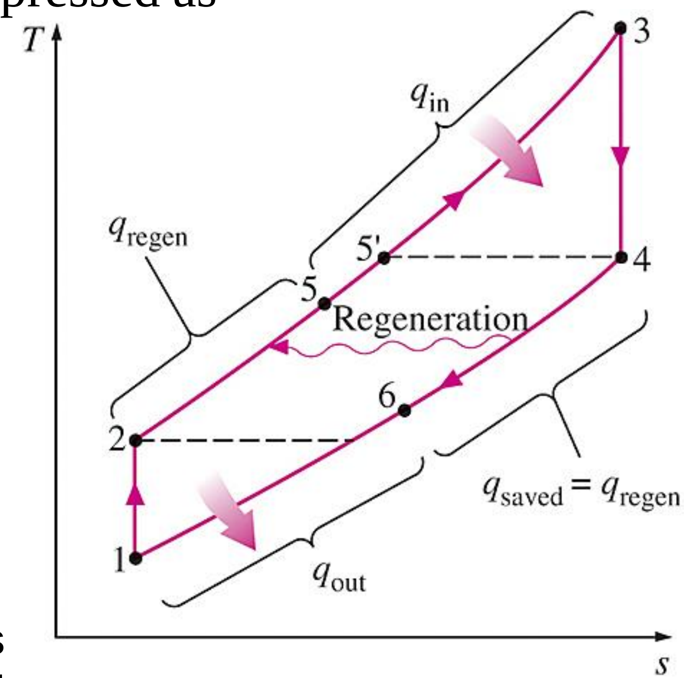
Note:
The use of a regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature.

Effectiveness of the Regenerator

Assuming the regenerator is well insulated and changes in kinetic and potential energies are negligible, the **actual** and **maximum heat transfers** from the exhaust gases to the air can be expressed as

$$q_{\text{regen,act}} = h_5 - h_2$$

$$q_{\text{regen,max}} = h_{5'} - h_2 = h_4 - h_2$$



Effectiveness of the regenerator,

$$\epsilon = \frac{q_{\text{regen,act}}}{q_{\text{regen,max}}} = \frac{h_5 - h_2}{h_4 - h_2}$$

Effectiveness under **cold-air standard** assumptions,

$$\epsilon \cong \frac{T_5 - T_2}{T_4 - T_2}$$

If written in terms of temperatures only, it is also called the thermal ratio

Thermal efficiency under **cold-air standard** assumptions,

$$\eta_{\text{th,regen}} = 1 - \left(\frac{T_1}{T_3}\right)(r_p)^{(k-1)/k}$$

Example: 4

A Brayton cycle with **regeneration** using air as the working fluid has a pressure ratio of 7. The minimum and maximum temperatures in the cycle are 310 and 1150 K respectively.

Assuming an isentropic efficiency of 75 percent for the compressor and 82 percent for the turbine and an **effectiveness** of 65 percent for the regenerator, determine:

(a) the **air temperature** at the turbine exit,

(b) the **net work output**, and

(c) the **thermal efficiency**.

Assignment: 6

The **7FA gas turbine** manufactured by General Electric is reported to have an efficiency of 35.9 percent in the simple-cycle mode and to produce 159 MW of net power. The pressure ratio is 14.7 and the turbine inlet temperature is 1288°C. The mass flow rate through the turbine is 1,536,000 kg/h.

Taking the ambient conditions to be 20°C and 100 kPa, determine:

a) the isentropic efficiency of the turbine and the compressor,

b) the thermal efficiency of this gas turbine if a **regenerator** with an **effectiveness** of 80 percent is added.

Assume **constant** specific heats at 300 K.

Assignment 7

A gas turbine plant with reheating is fitted with an exhaust heat exchanger. Compression is done in a single stage with a pressure ratio of 8, while expansion is done in two turbine stages. The high pressure turbine drives the compressor while the low pressure turbine supplies the net work of the plant. Inlet temperatures for the turbines are the same at 1073 K and the inlet temperature of the compressor is 303 K. The main combustion chamber (not including the reheater) supplies heat at a rate of 380 kJ/kg of working fluid.

Sketch the cycle on a T-s diagram, determine the temperature at each point, and calculate;

a) Thermal ratio of the heat exchanger
b) Thermal efficiency of the plant
c) The ratio of the fuel flow rate to the working fluid flow rate, provided that the calorific value of the fuel be 43000 kJ/kg fuel.

Given :	c_p = 1.005 kJ/kg.K and γ = 1.4	for air
	c_p = 1.15 kJ/kg.K and γ = 1.333	for gas

Brayton Cycle with Intercooling, Reheating & Regeneration

❖The **net work output** of a gas-turbine cycle can be increased by either:

a) decreasing the compressor work, or
b) increasing the turbine work, or
c) both.

❖The compressor work input can be decreased by carrying out the compression process in stages and cooling the gas in between (Fig. below), using **multistage compression** with **intercooling**.

❖The work output of a turbine can be increased by expanding the gas in stages and reheating it in between, utilizing a **multistage expansion** with **reheating.**

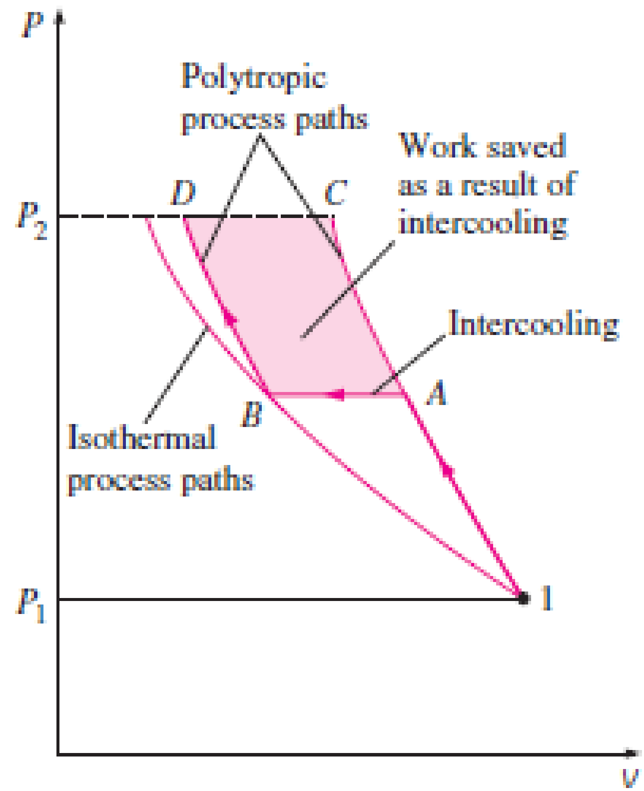


Fig: Comparison of work inputs to a single-stage compressor (1*AC) and a* two-stage compressor with intercooling (1*ABD).*

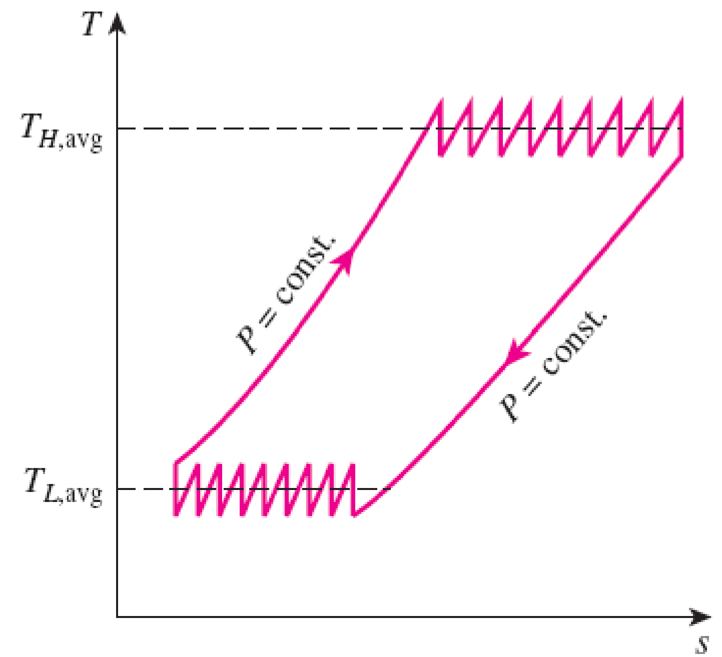
Brayton Cycle with Intercooling, Reheating & Regeneration

Intercooling and reheating always **decreases** thermal efficiency unless accompanied by **regeneration**. **Why?**
Therefore, in gas turbine power plants, intercooling and reheating are always **used in conjunction** with regeneration.

•**Reheating :** Adding fuel into hot gases and we have extra energy for next turbine.

• **Intercooling :** Cooling the flow before entering into next compressor stage.

• **Regeneration :** Using exhaust energy to heat up the air before entering the combustion chamber.



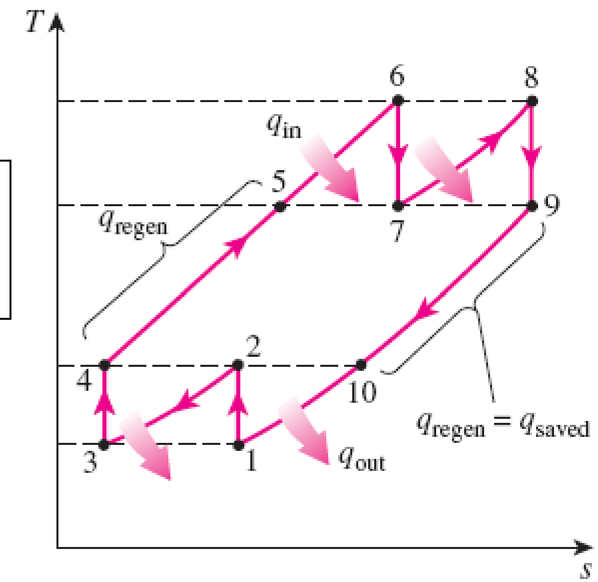
As the number of compression and expansion stages increases, the gas-turbine cycle with intercooling, reheating, and regeneration approaches the Ericsson cycle.

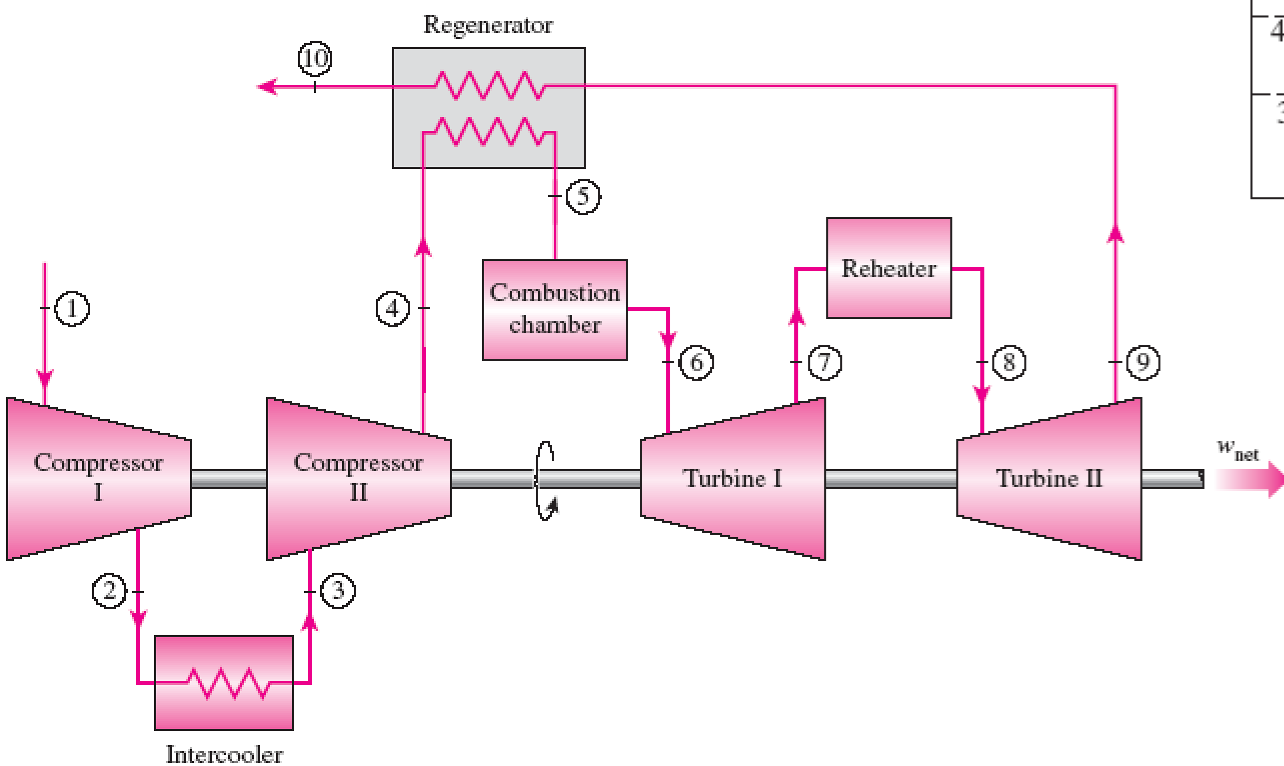
THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

For minimizing work input to compressor and maximizing work output from turbine:

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9}$$

Tmax limited by materials,
Tmin limited by environment





A gas-turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration and its *T-s* diagram.

Conditions for Best Performance

The work input to a two-stage compressor is **minimized** when

a) equal pressure ratios are maintained across each stage.

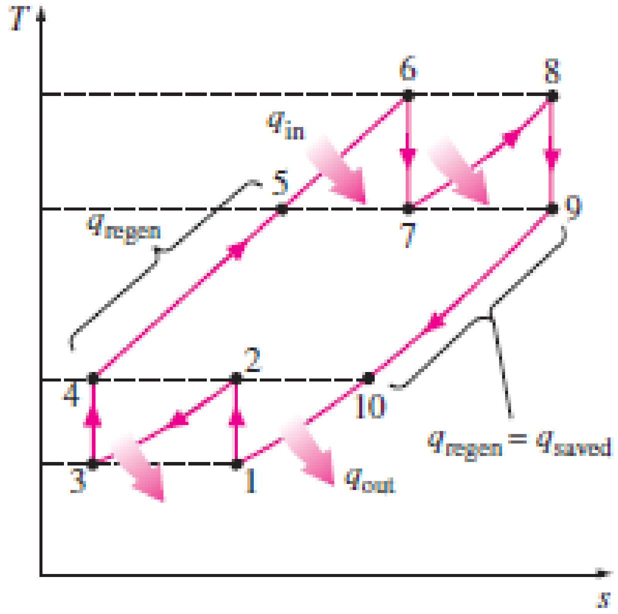
b) Complete intercooling is performed

This procedure also maximizes the turbine work output.

Thus, for best performance we have,

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9}$$

$$T_3 = T_1$$



Example:5

Consider an ideal gas-turbine cycle with **two stages** of compression and two stages of expansion. The pressure ratio across each stage of the compressor and turbine is 3. The air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K. Determine:

a) the **back work ratio**, and
b) the **thermal efficiency** of the cycle

assuming:

I)no regenerator is used, and

II)a regenerator with **75 percent** effectiveness is used.

Assignment. 8

Consider a **regenerative** gas-turbine power plant with two stages of compression and two stages of expansion. The overall pressure ratio of the cycle is 9. The air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K.

Determine the **minimum mass flow rate** of air needed to develop net power output of 110 MW.

Thank You for your attention!