Engineering Thermodynamics

Module -4.4

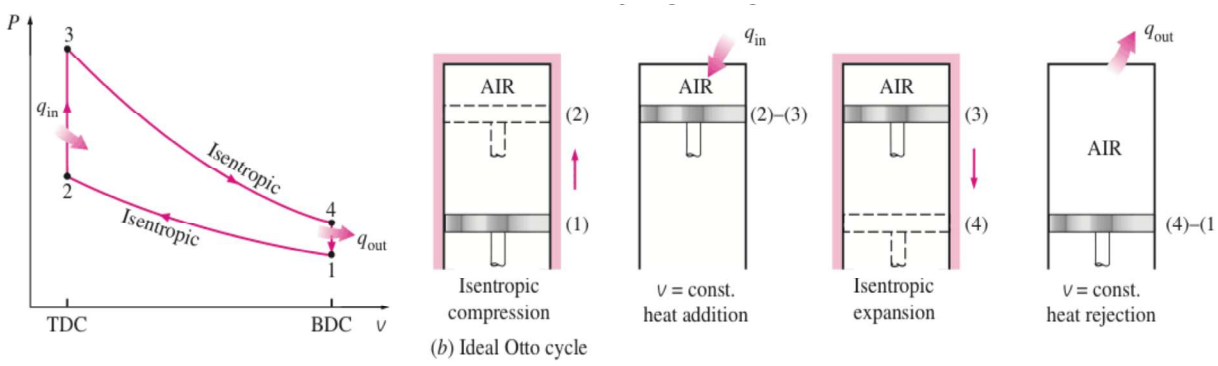
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9.5 OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES



(b) Ideal Otto cycle

FIGURE 9–13

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

- Working
  - Initially, both the intake and the exhaust valves are closed, and the piston is at its lowest position (BDC)
  - During the compression stroke, the piston moves upward, compressing the air–fuel mixture
  - Shortly before the piston reaches its highest position (TDC), the spark plug fires and the mixture ignites, increasing the pressure and temperature of the system
  - The high-pressure gases force the piston down, which in turn forces the crankshaft to rotate, producing a useful work output during the expansion or power stroke
  - the end of this stroke, the piston is at its lowest position (the completion of the first mechanical cycle), and the cylinder is filled with combustion products
  - Now the piston moves upward one more time, purging the exhaust gases through the exhaust valve (the exhaust stroke), and down a second time, drawing in fresh air–fuel mixture through the intake valve (the intake stroke).
  - the analysis can be simplified significantly if the air-standard assumptions are utilized
  - The resulting cycle, which closely resembles the actual operating conditions, is the ideal Otto cycle.
  - It consists of four internally reversible processes

1-2 – isentropic compression
2-3 – Constant volume heat addition
3-4 – Isentropic expansion
4-1 → Constant volume heat rejection.

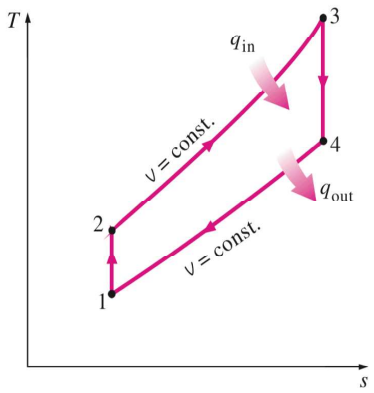


FIGURE 9–16

T-s diagram of the ideal Otto cycle.

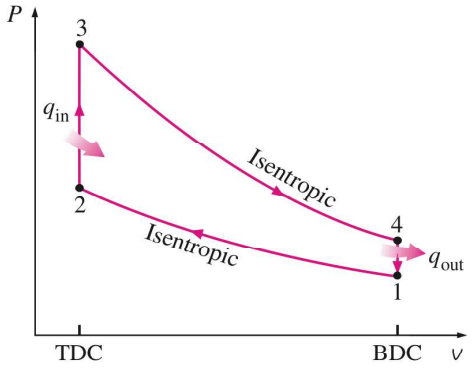


FIGURE 9–13

Actual and ideal cycles in spark-ignition en

- The Otto cycle is executed in a closed system, and disregarding the changes in kinetic and potential energies, the energy balance for any of the processes is expressed, on a unit-mass basis, as

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u \quad (kJ/kg) \qquad (9.5)$$

- No work is involved during the two heat transfer processes since both take place at constant volume
- Therefore, heat transfer to and from the working fluid can be expressed as

$$q_{in} = u_3 - u_2 = c_v (T_3 - T_2) \qquad (9.6a)$$

and

$$q_{out} = u_4 - u_1 = c_v (T_4 - T_1) \qquad (9.6b)$$

- Then the thermal efficiency of the ideal Otto cycle under the cold air standard assumptions becomes

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

Process 1-2 & 3-4 are isentropic. and $v_2 = v_3$ and $v_4 = v_1$. Thus

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

- Substituting these equations into the thermal efficiency relation and simplifying give

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

where,

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

where, r is the compression ratio and k is the specific heat ratio c_p/c_v

- thermal efficiency of an ideal Otto cycle depends on the **compression ratio** of the engine and the **specific heat ratio** of the working fluid
  - thermal efficiency of the ideal Otto cycle increases with both the compression ratio and the specific heat ratio
  - Fig. 9–17 shows that the thermal efficiency curve is rather steep at low compression ratios but flattens out starting with a compression ratio value of about 8
  - the increase in thermal efficiency with the compression ratio is not as pronounced at high compression ratios

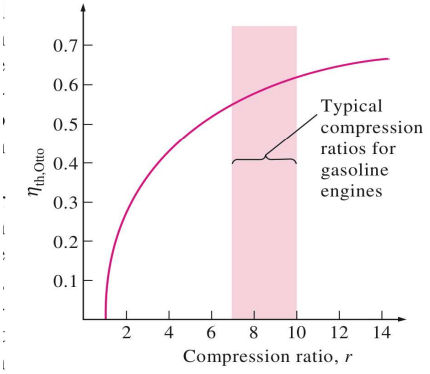


FIGURE 9–17
Thermal efficiency of the ideal Otto cycle as a function of compression ratio ($k = 1.4$).

- Autoignition
  - Also, when high compression ratios are used, the temperature of the air–fuel mixture rises above the autoignition temperature of the fuel (the temperature at which the fuel ignites without the help of a spark) during the combustion process, causing an early and rapid burn of the fuel at some point or points ahead of the flame front, followed by almost instantaneous inflammation of the

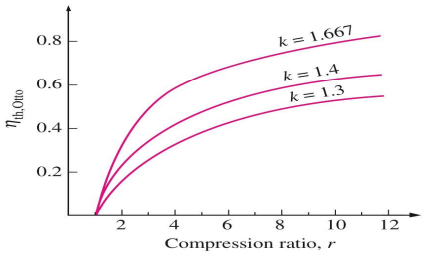


FIGURE 9–18
The thermal efficiency of the Otto cycle increases with the specific heat ratio k of the working fluid.

end gas.
    - This premature ignition of the fuel, called autoignition, produces an audible noise, which is called engine knock
    - Autoignition in spark-ignition engines cannot be tolerated because it hurts performance and can cause engine damage
- The second parameter affecting the thermal efficiency of an ideal Otto cycle is the specific heat ratio k.
    - For a given compression ratio, an ideal Otto cycle using a monatomic gas (such as argon or helium, k = 1.667) as the working fluid will have the highest thermal efficiency
    - The specific heat ratio k, and thus the thermal efficiency of the ideal Otto cycle, decreases as the molecules of the working fluid get larger (Fig. 9–18). At
    - At room temperature it is 1.4 for air, 1.3 for carbon dioxide, and 1.2 for ethane.
- The working
  fluid in actual engines contains larger molecules such as carbon dioxide, and the specific heat ratio decreases with temperature, which is one of the reasons that the actual cycles have lower thermal efficiencies than the ideal Otto cycle

DIESEL CYCLE: THE IDEAL CYCLE

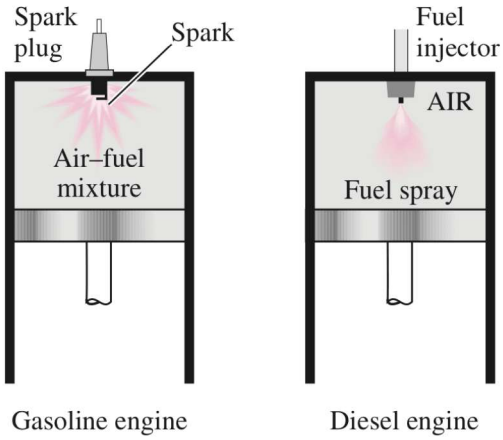


FIGURE 9–20

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

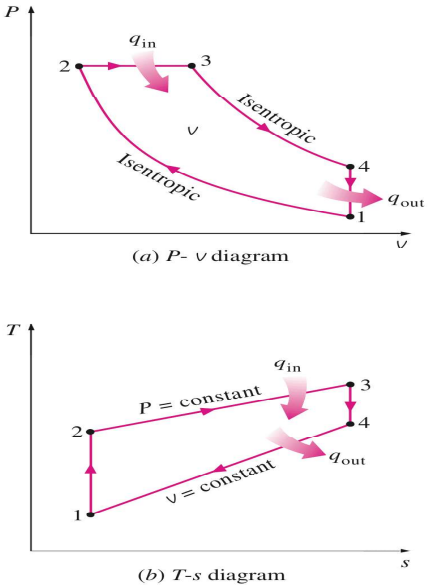


FIGURE 9–21

T-s and *P-v* diagrams for the ideal Diesel cycle.

- Workinng of a CI engines (also known as diesel engines),
  - the air is compressed to a temperature that is above the autoignition temperature of the fuel
  - combustion starts on contact as the fuel is injected into this hot air.
  - spark plug and carburetor are replaced by a fuel injector in diesel engines
  - In gasoline engines, a mixture of air and fuel is compressed during the compression stroke, and the compression ratios are limited by the onset of autoignition or engine knock
  - only air is compressed during the compression stroke, eliminating the possibility of autoignition.
  - Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 and 24.
  - The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke.
  - Therefore, the combustion process in these engines takes place over a longer interval

    - Because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat-addition process
    - Noting that the Diesel cycle is executed in a piston–cylinder device, which forms a closed system
- the amount of heat transferred to the working fluid at constant pressure and rejected from it at constant volume can be expressed as

$$q_{in} - w_{b,out} = u_3 - u_2 \Rightarrow q_{in} = P_2(v_3 - v_2) + (u_3 - u_2)$$
$$= h_3 - h_2 = c_p(T_3 - T_2) \quad (9.10a)$$

and

$$-q_{out} = u_1 - u_4 \Rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \quad (9.10b)$$

Then the thermal efficiency of ideal diesel cycle under cold air standard assumptions becomes :-

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

- We now define a new quantity, the cutoff ratio r_c,—
- —as the ratio of the cylinder volumes after and before the combustion process:

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

- Utilizing this definition and the isentropic ideal-gas relations for processes 1-2 and 3-4, we see that the thermal efficiency relation reduces to

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

- the efficiency of a Diesel cycle differs from the efficiency of an Otto cycle by the quantity in the brackets
- This quantity is always greater than 1.

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

- when both cycles operate on the same compression ratio
- Also, as the cutoff ratio decreases, the efficiency of the Diesel cycle increases (Fig. 9)
- For the limiting case of $r_c = 1$, the quantity in the brackets becomes unity (can you prove it?), and the efficiencies of the Otto and Diesel cycles become identical

Engineering Thermodynamics

Module -4.5

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BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

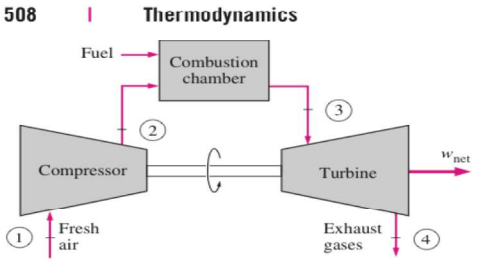


FIGURE 9–29
An open-cycle gas-turbine engine.

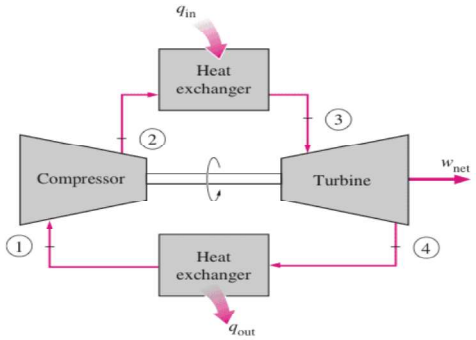


FIGURE 9–30
A closed-cycle gas-turbine engine.

- Gas turbines usually operate on an open cycle, as shown in Fig. 9–29
- Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised
- The high-pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure.
- The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power.
- The exhaust gases leaving the turbine are thrown out (not recirculated), causing the cycle to be classified as an open cycle
- The ideal cycle that the working fluid undergoes in this closed loop is the Brayton cycle, which is made up of four internally reversible processes:
- 1-2 Isentropic compression (in a compressor)
- 2-3 Constant-pressure heat addition
- 3-4 Isentropic expansion (in a turbine)

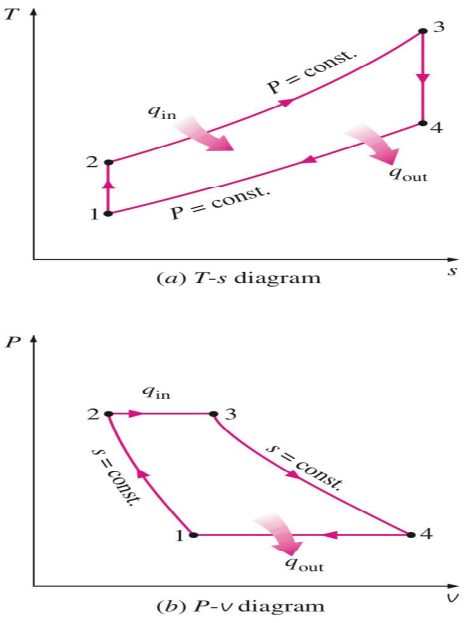


FIGURE 9–31
T-s and *P-v* diagrams for the ideal Brayton cycle.

- 4-1 Constant-pressure heat rejection

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{exit} - h_{inlet} \qquad -(9.15)$$

Therefore, heat transfer to and from the working fluid are

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) \qquad -(9.16a)$$

and

$$q_{out} = h_4 - h_1 = c_p(T_4 - T_1) \qquad (9.16b)$$

Then the thermal efficiency of ideal Brayton cycle under cold air standard assumption becomes :-

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

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

a

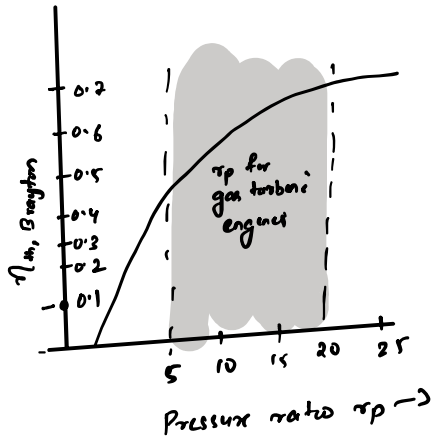
Process 1-2, 3-4 are isentropic, and $P_2 = P_3$ and $P_4 = P_1$. Thus,

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

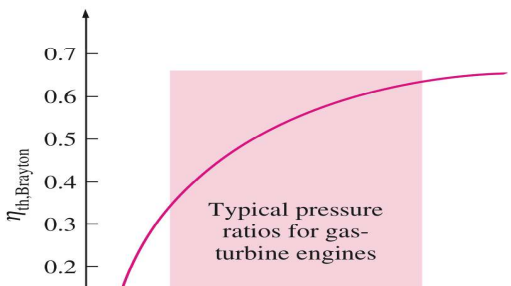
Substituting these eqns. into the thermal efficiency relation and simplifying give :-

$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{k-1/k}} \qquad (9.17)$$

where $r_p = P_2/P_1$



- For a fixed turbine inlet temperature T3, the net work output per cycle increases



with the pressure ratio, reaches a maximum, and then starts to decrease, as shown in Fig. 9–33. Therefore, there should be a compromise between the pressure ratio (thus the thermal efficiency) and the net work output.

- The highest temperature in the cycle occurs at the end of the combustion process (state 3), and it 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 performs two important functions: It
- It supplies the necessary oxidant for the combustion of the fuel,
- and it serves as a coolant to keep the temperature of various components within
- gas turbines, an air–fuel mass ratio of 50 or above is not uncommon
- Therefore, in a cycle analysis, treating the combustion gases as air does not cause any appreciable error

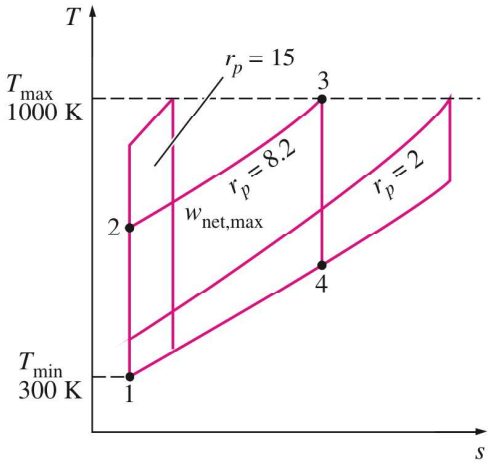


FIGURE 9–33

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.