Chapter 5: Air-Standard Power Cycles

5.1 Air-standard assumptions

- In gas power cycles, the working fluid remains a gas throughout the entire cycle.
  - Spark-ignition engines, diesel engines, and conventional gas turbines are familiar examples of devices that operate on gas cycles.

  In all these engines, energy is provided by burning a fuel within the system boundaries.
- That is, they are ***internal combustion engines***.
- Because of this combustion process, the composition of the working fluid changes from air and fuel to combustion products during the course of the cycle.
- However, considering that **air is predominantly nitrogen** that undergoes hardly any chemical reactions in the combustion chamber, **the working fluid closely resembles air at all times**.
- Even though internal combustion engines operate on a mechanical cycle (the piston returns to its starting position at the end of each revolution), the working fluid does not undergo a complete thermodynamic cycle. It is thrown out of the engine at some point in the cycle (as exhaust gases) instead of being returned to the initial state. Working on an open cycle is the characteristic of all internal combustion engines.

- The actual gas power cycles are rather complex. 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. 5.1).
4. The exhaust process is replaced by a **heat-rejection process** that restores the working fluid to its initial state.

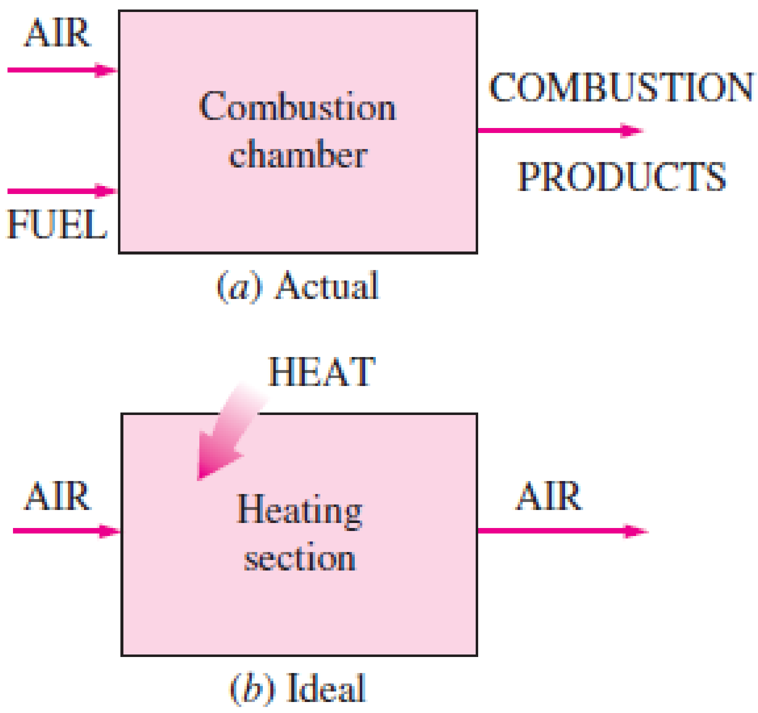


FIGURE 5.1: The combustion process is replaced by a heat-addition process in ideal cycles.

- Another assumption that is often utilized to simplify the analysis even more is that **air has constant specific heats** whose values are determined at ***room temperature*** (25°C, or 77°F).
- When this assumption is utilized, the air-standard assumptions are called the **cold-air-standard assumptions.**
- A cycle for which the air-standard assumptions are applicable is frequently referred to as an **air-standard cycle.**
- The air-standard assumptions previously stated provide considerable simplification in the analysis without significantly deviating from the actual cycles.
- This simplified model enables us to study qualitatively the influence of major parameters on the performance of the **actual engines**.

5.2 An overview of reciprocating engines

- The reciprocating engine (basically a piston–cylinder device)
- It is the powerhouse of the vast majority of automobiles, trucks, light aircraft, ships, and electric power generators, as well as many other devices.
- The basic components of a reciprocating engine are shown in Fig. 5.2.
  - 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
  - The **bottom dead center (BDC)—the** position of the piston when it forms the largest volume in the cylinder

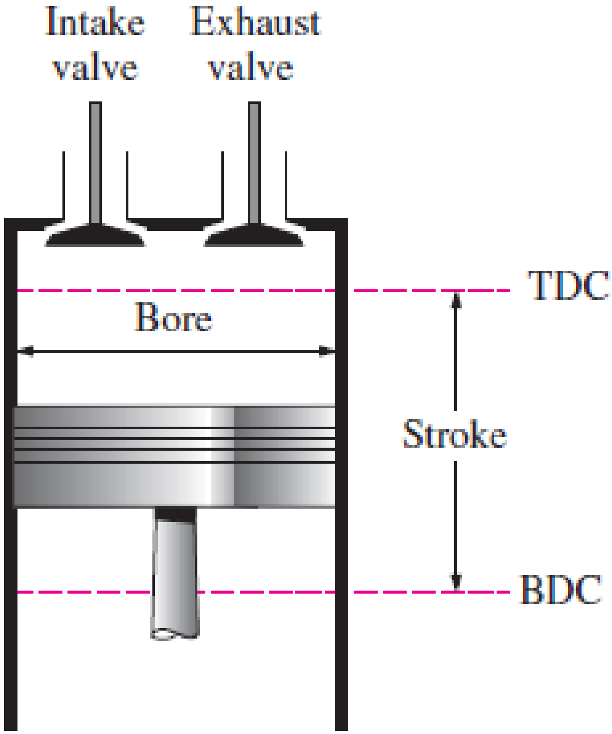


FIGURE 5.2: Nomenclature for reciprocating engines.

- 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,** and 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** (Fig. 5.3).
- 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:

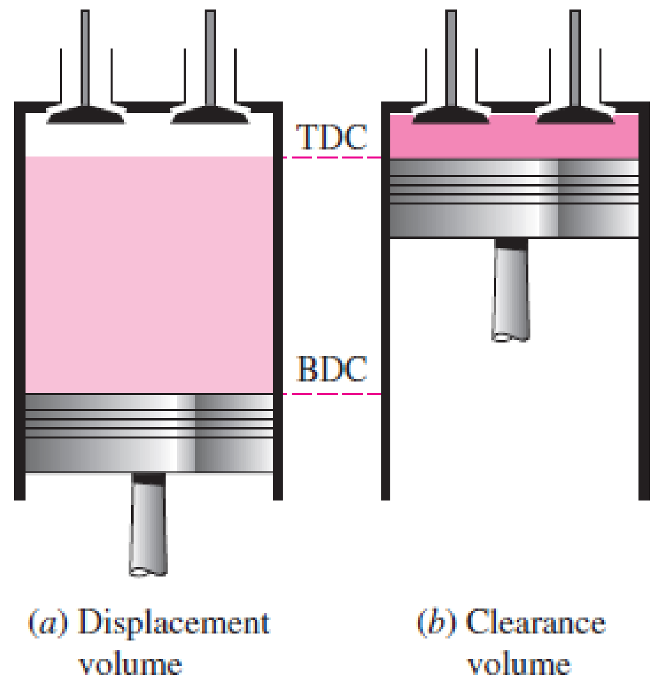


FIGURE 5.3: Displacement and clearance volumes of a reciprocating engine.

$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}} \qquad \text{..........................5.2}$$

Notice that the compression ratio is a *volume ratio* and should not be confused with the pressure ratio.

- Another term frequently used in conjunction with reciprocating engines is the **mean effective pressure** (MEP).
  - It 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 (Fig. 5.4). That is,

W_{net} = MEP * Piston area * Stroke = MEP * Displacement volume

or

$$MEP = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}} \quad (kPa) \qquad \text{..........................5.3}$$

- **The MEP** 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.

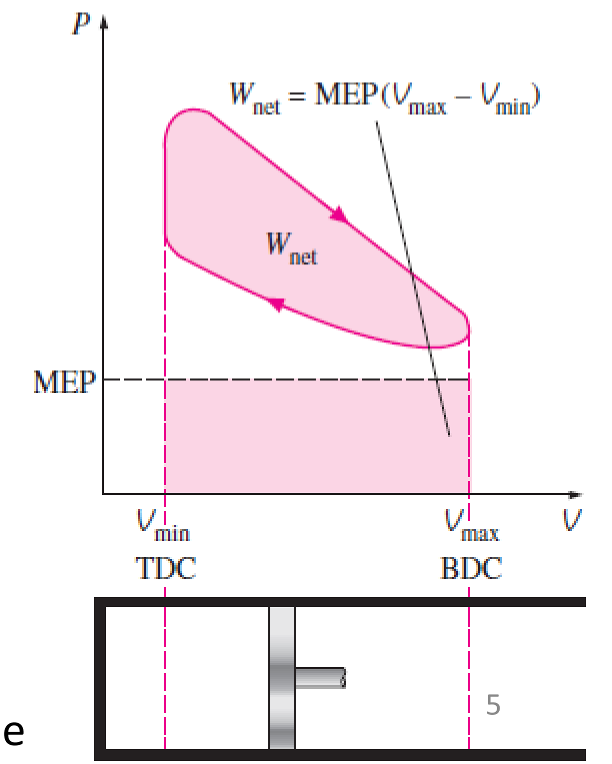


FIGURE 5.4: The net work output of a cycle is equivalent to the product of the mean effective pressure & the displacement volume

- 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.
- In **CI engines**, the air–fuel mixture is self-ignited as a result of compressing the mixture above its **self-ignition temperature**.
- In the next two sections, we will discuss the
  - ***Otto*** *which is the ideal cycle for SI engines,* and
  - ***Diesel cycles*** which is the ideal cycles for CI engines.

5.3 OTTO CYCLE: The ideal cycle for spark-ignition engines

- The Otto cycle is the ideal cycle for spark-ignition reciprocating 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. A schematic of each stroke as well as a *P-v* diagram for an actual four-stroke spark-ignition engine is given in Fig. 5.5(*a*).

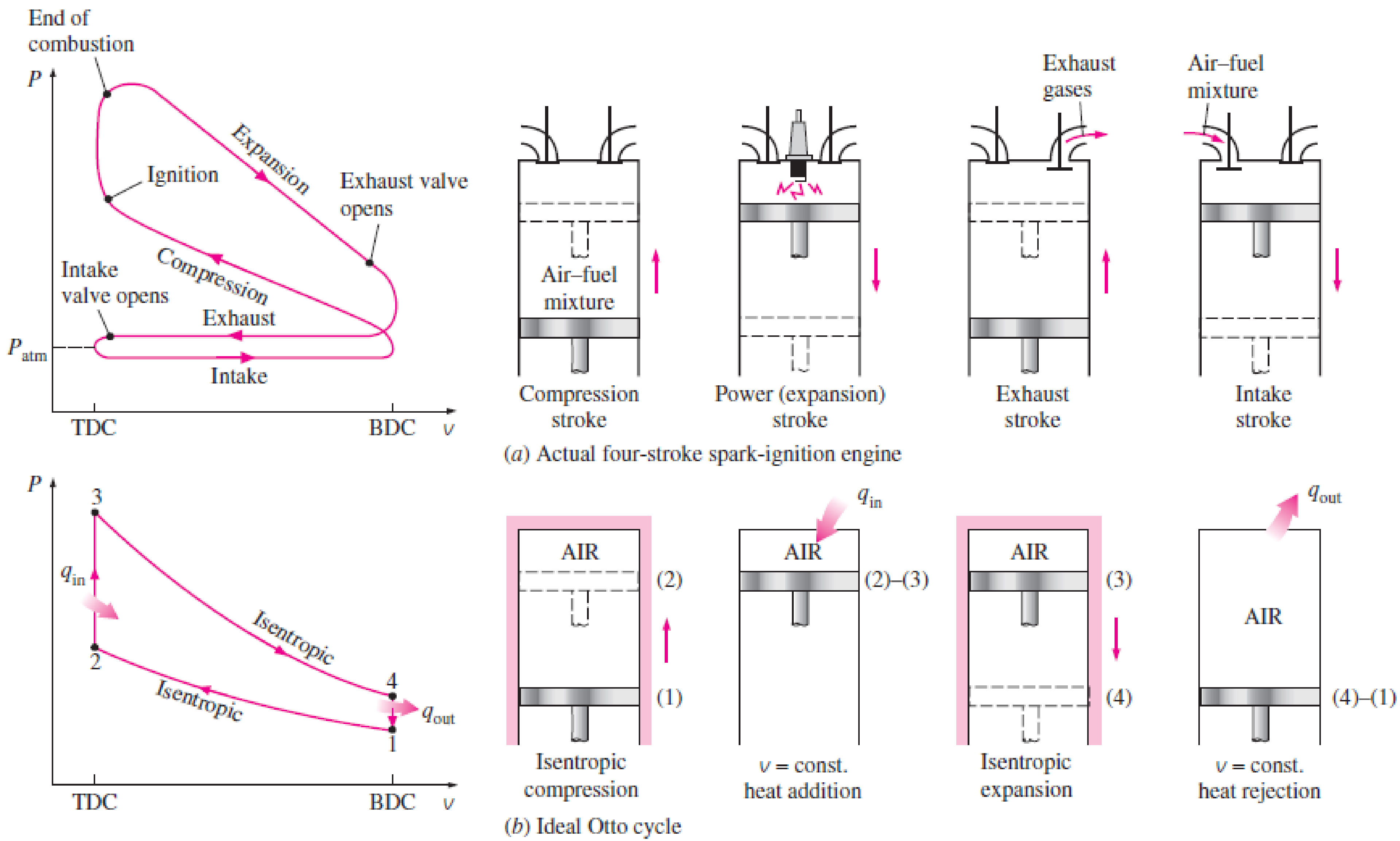


FIGURE 5.5: Actual and ideal cycles in spark-ignition engines and their *P-v* diagrams.

- 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.* At 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*). Notice that the pressure in the cylinder is slightly above the atmospheric value during the exhaust stroke and slightly below during the intake stroke.
- The ideal **Otto cycle** consists of f**our** internally reversible processes:

    1-2 Isentropic compression

    2-3 Constant-volume heat addition

    3-4 Isentropic expansion

    4-1 Constant-volume heat rejection

The *T-s* diagram of the Otto cycle is given in Fig. 5.6.

- 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:

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u \qquad (kJ/kg) \quad ....5.4$$

- 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:

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2) \quad ....5.5a$$

and

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \quad .....5.5b$$

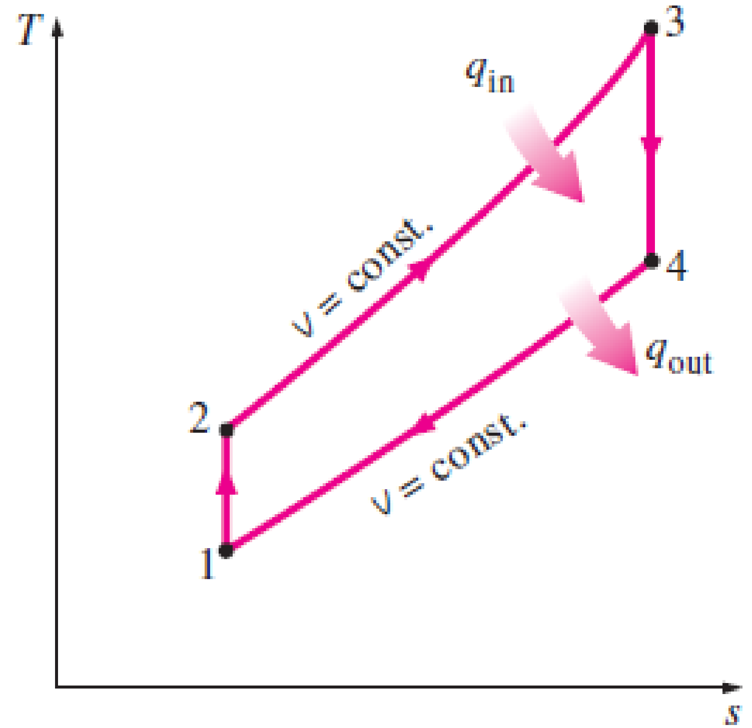


FIGURE 5.6: *T-s* diagram of the ideal Otto cycle.

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

$$\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)} \quad ....5.6$$

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

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{v_3}{v_4}\right)^{k-1} = \frac{T_4}{T_3} \quad \text{..........5.7}$$

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

$$\eta_{\text{th,Otto}} = 1 - \frac{1}{r^{k-1}} \quad \text{..........5.8}$$

where

$$r = \frac{V_{\max}}{V_{\min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2} \quad \text{..........5.9}$$

r - is the **compression ratio** and

k is the **specific heat ratio, i.e.** k **=** c_p/c_v.

- Equation 5.8 shows that under the cold-air-standard assumptions, the thermal efficiency of an ideal Otto cycle depends on the compression ratio of the engine and the specific heat ratio of the working fluid.
- The thermal efficiency of the ideal Otto cycle increases with both the compression ratio and the specific heat ratio.

- For a given compression ratio, the thermal efficiency of an actual spark-ignition engine is less than that of an ideal Otto cycle because of the **irreversibilities,** such as friction, and other factors such as incomplete combustion.
- The increase in thermal efficiency with the compression ratio is not as pronounced at high compression ratios.
- Also, when high compression ratios are used, the temperature of the air–fuel mixture rises above the auto ignition 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 end gas
- This premature ignition of the fuel, called **auto ignition, produces** an audible noise, which is called **engine knock.**
- **Auto ignition in** spark-ignition engines cannot be tolerated because it hurts performance and can cause engine damage.
- The requirement that auto ignition not be allowed places an upper limit on the compression ratios that can be used in spark ignition internal combustion engines
- Generally the thermal efficiencies of **actual spark-ignition engines** range from about **25 to 30 percent**.

example

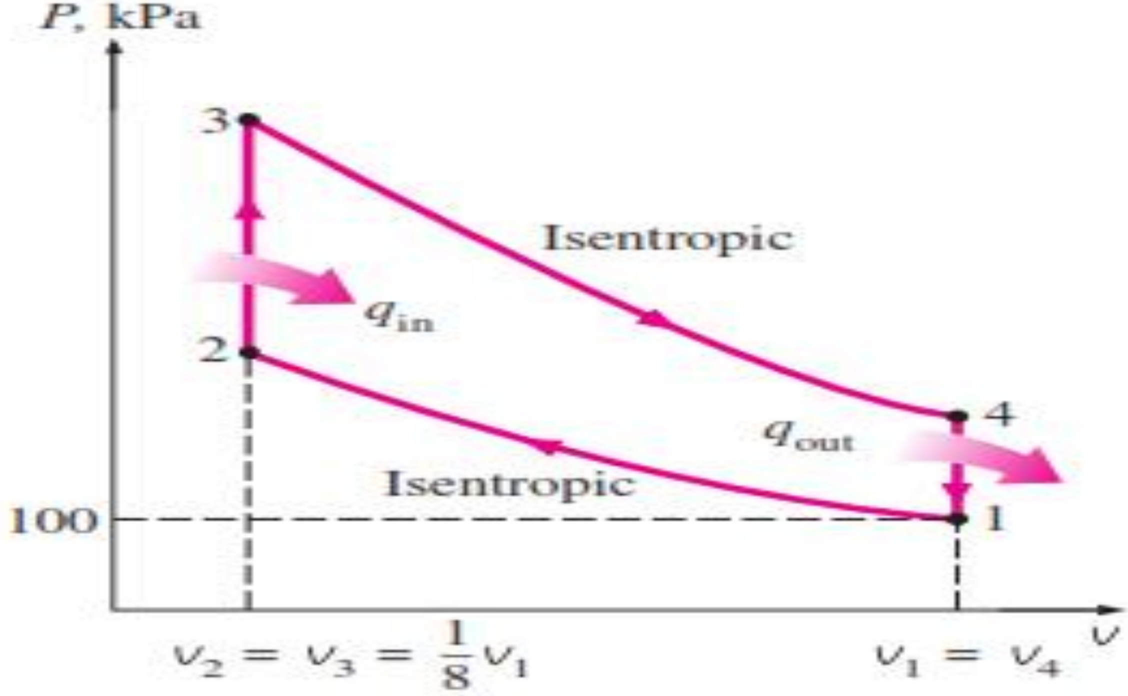
- An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 100kPa and 17°C, and 800 kJ/kg of heat is transferred to air during the constant-volume heat-addition process. Accounting for the variation of specific heats of air with temperature, determine

(*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.


P, kPa
3
2
1
4
Isentropic
Isentropic
q_{in}
q_{out}
100
$v_2 = v_3 = \frac{1}{8} v_1$
$v_1 = v_4$
v

$$T_1 = 290 \text{ K} \rightarrow u_1 = 206.91 \text{ kJ/kg}$$
$$v_{r1} = 676.1$$

Process 1-2 (isentropic compression of an ideal gas):

$$\frac{v_{r2}}{v_{r1}} = \frac{v_2}{v_1} = \frac{1}{r} \rightarrow v_{r2} = \frac{v_{r1}}{r} = \frac{676.1}{8} = 84.51 \rightarrow T_2 = 652.4 \text{ K}$$
$$u_2 = 475.11 \text{ kJ/kg}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \rightarrow P_2 = P_1\left(\frac{T_2}{T_1}\right)\left(\frac{v_1}{v_2}\right)$$
$$= (100 \text{ kPa})\left(\frac{652.4 \text{ K}}{290 \text{ K}}\right)(8) = 1799.7 \text{ kPa}$$

Process 2-3 (constant-volume heat addition):

$$q_{in} = u_3 - u_2$$
$$800 \text{ kJ/kg} = u_3 - 475.11 \text{ kJ/kg}$$
$$u_3 = 1275.11 \text{ kJ/kg} \rightarrow T_3 = \mathbf{1575.1 \text{ K}}$$
$$v_{r3} = 6.108$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \rightarrow P_3 = P_2\left(\frac{T_3}{T_2}\right)\left(\frac{v_2}{v_3}\right)$$

$$= (1.7997 \text{ MPa})\left(\frac{1575.1 \text{ K}}{652.4 \text{ K}}\right)(1) = \mathbf{4.345 \text{ MPa}}$$

(*b*) The net work output for the cycle is determined either by finding the boundary (*P dV*) work involved in each process by integration and adding them or by finding the net heat transfer that is equivalent to the net work done during the cycle. We take the latter approach. However, first we need to find the internal energy of the air at state 4:

Process 3-4 (isentropic expansion of an ideal gas):

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3} = r \rightarrow v_{r4} = r v_{r3} = (8)(6.108) = 48.864 \rightarrow T_4 = 795.6 \text{ K}$$

$$u_4 = 588.74 \text{ kJ/kg}$$

Process 4-1 (constant-volume heat rejection):

$$-q_{\text{out}} = u_1 - u_4 \rightarrow q_{\text{out}} = u_4 - u_1$$

$$q_{\text{out}} = 588.74 - 206.91 = 381.83 \text{ kJ/kg}$$

Thus,

$$w_{\text{net}} = q_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 800 - 381.83 = \mathbf{418.17 \text{ kJ/kg}}$$

(*c*) The thermal efficiency of the cycle is determined from its definition:

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{418.17 \text{ kJ/kg}}{800 \text{ kJ/kg}} = \mathbf{0.523 \text{ or } 52.3\%}$$

Under the cold-air-standard assumptions (constant specific heat values at room temperature), the thermal efficiency would be (Eq. 9–8)

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}} = 1 - r^{1-k} = 1 - (8)^{1-1.4} = 0.565 \text{ or } 56.5\%$$

which is considerably different from the value obtained above. Therefore, care should be exercised in utilizing the cold-air-standard assumptions.

(*d*) The mean effective pressure is determined from its definition, Eq. 9–4:

$$\text{MEP} = \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1 - v_1/r} = \frac{w_{net}}{v_1(1 - 1/r)}$$

where

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})}{100 \text{ kPa}} = 0.832 \text{ m}^3/\text{kg}$$

Thus,

$$\text{MEP} = \frac{418.17 \text{ kJ/kg}}{(0.832 \text{ m}^3/\text{kg})(1 - \frac{1}{8})}\left(\frac{1 \text{ kPa}\cdot\text{m}^3}{1 \text{ kJ}}\right) = \mathbf{574 \text{ kPa}}$$

5–4 ■ Diesel cycle: the ideal cycle for compression-ignition engines

- The Diesel cycle is the ideal cycle for CI reciprocating engines. The CI engine, is very similar to the SI engine, differing mainly in the **method of initiating combustion**.
- In spark-ignition engines (also known as *gasoline engines*), the air–fuel mixture is compressed to a temperature that is below the autoignition temperature of the fuel, and the combustion process is initiated by firing a spark plug. In CI engines (also known as *diesel engines*), the air is compressed to a temperature that is above the autoignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines (Fig. 5.7).

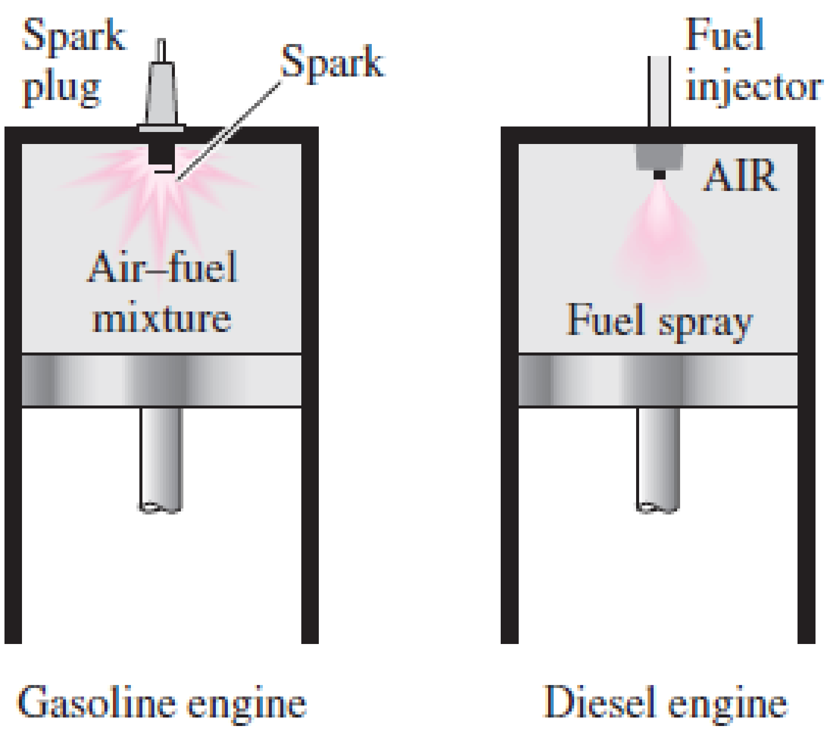


FIGURE 5.7: In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

- 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.
- In diesel engines, 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. Not having to deal with the problem of autoignition has another benefit: many of the stringent requirements placed on the gasoline can now be removed, and fuels that are less refined (thus less expensive) can be used in diesel engines.
- 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. In fact, this is the only process where the Otto and the Diesel cycles differ. The remaining three processes are the same for both ideal cycles. That is, process 1-2 is isentropic compression, 3-4 is isentropic expansion, and 4-1 is constant-volume heat rejection. The similarity between the two cycles is also apparent from the *P*-*v* and *T*-*s* diagrams of the Diesel cycle, shown in Fig. 5.8.

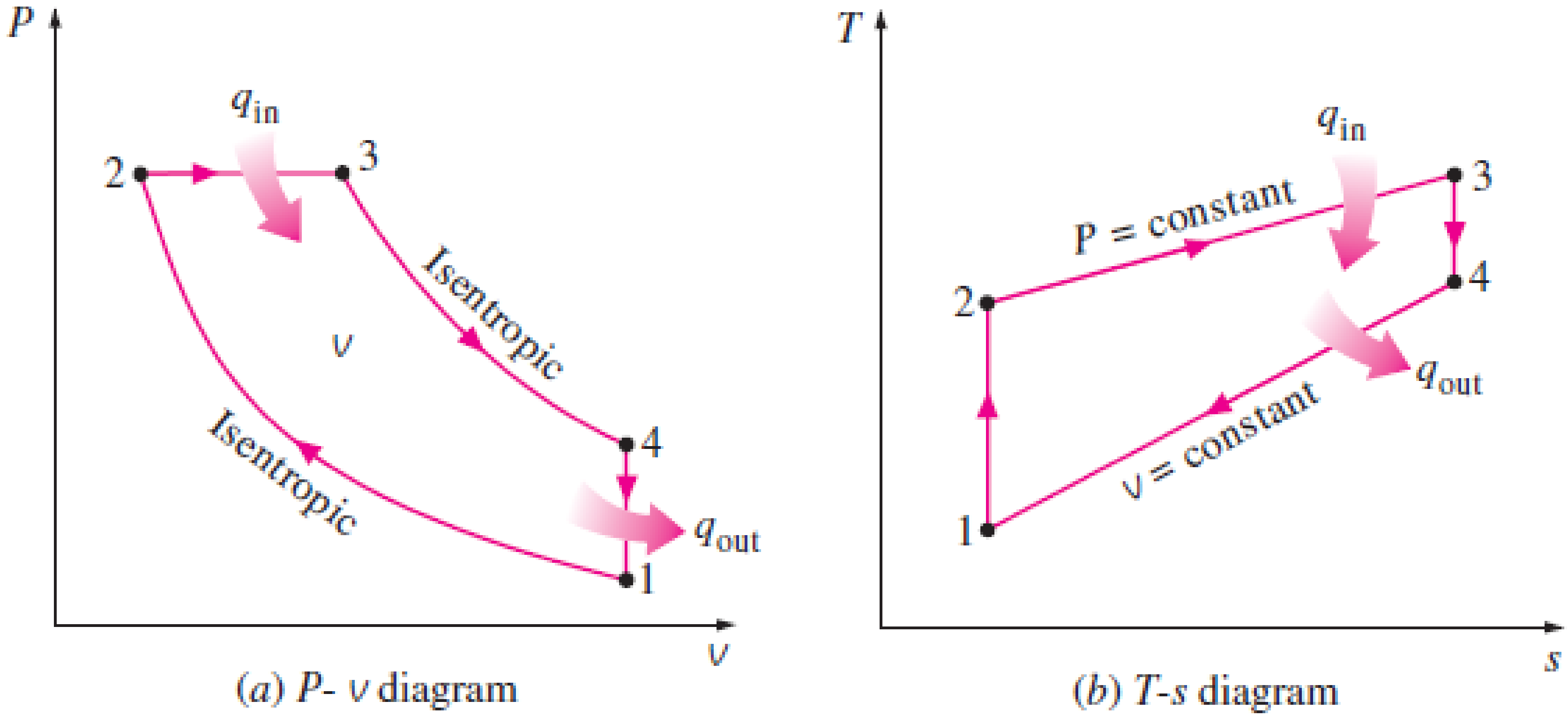


FIGURE 5.8: *T-s* and *P-v* diagrams for the ideal Diesel cycle.

- 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

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

$$= h_3 - h_2 = c_p(T_3 - T_2) \quad \text{..........5.10a}$$

and

$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \quad \text{..........5.10b}$$

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

$$\eta_{\text{th,Diesel}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)} \quad \text{..........5.11}$$

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

$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2} \quad \text{..........5.12}$$

- 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

$$\eta_{\text{th,Diesel}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right] \quad \text{..........5.13}$$

- Looking at Eq. 5.13 carefully, one would notice that under the cold-air-standard assumptions, 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. Therefore, when both cycles operate on the same compression ratio.

$$\eta_{th,Otto} > \eta_{th,Diesel} \quad \text{...........5.14}$$

- Also, as the cutoff ratio decreases, the efficiency of the Diesel cycle increases (Fig. 5.9). For the limiting case of r_c = 1, the quantity in the brackets becomes unity, and the efficiencies of the Otto and Diesel cycles become identical.

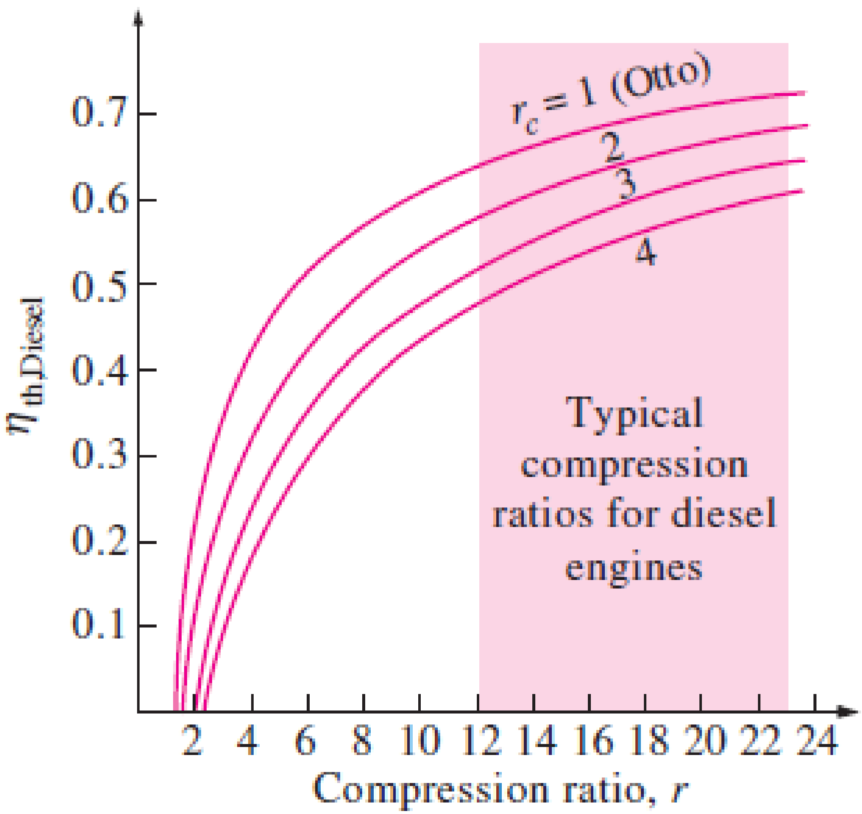


FIGURE 5.9: Thermal efficiency of the ideal Diesel cycle as a function of compression and cutoff ratios (k = 1.4).

- Approximating the combustion process in internal combustion engines as a constant volume or a constant-pressure heat-addition process is overly simplistic and not quite realistic. Probably a better (but slightly more complex) approach would be to 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.

- Approximating the combustion process in internal combustion engines as a constant volume or a constant-pressure heat-addition process is overly simplistic and not quite realistic. Probably a better (but slightly more complex) approach would be to 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.
  - The ideal cycle based on this concept is called the **dual cycle,** and a *P-v* diagram for it is given in Fig. 5.10. The relative amounts of heat transferred during each process can be adjusted to approximate the actual cycle more closely. Note that both the Otto and the Diesel cycles can be obtained as special cases of the dual cycle.

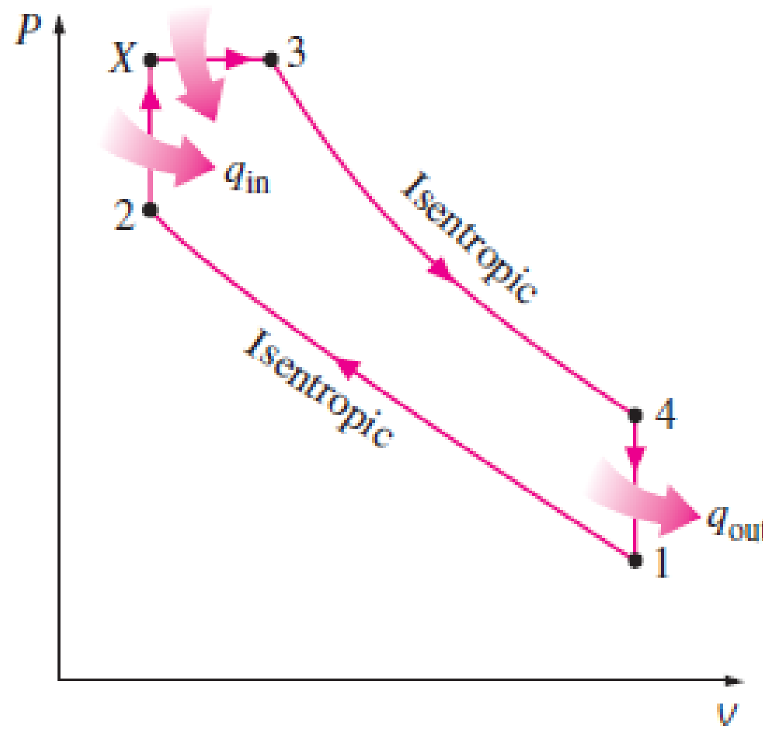


FIGURE 5.10: *P-v* diagram of an ideal dual cycle.

Example

- An ideal Diesel cycle with air as the working fluid has a compression ratio of 18 and a cutoff ratio of 2. At the beginning of the compression process, the working fluid is at 14.7psia, 80°F, & 117in^3. Utilizing the cold-air-standard assumptions, determine

  (*a*) the temperature and pressure of air at the end of each process,

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

  (*c*) the mean effective pressure.

5.5 ■ Stirling and ericsson cycles

- The ideal Otto and Diesel cycles discussed in the preceding sections are composed entirely of internally reversible processes and thus are internally reversible cycles. These cycles are not totally reversible, however, since they involve heat transfer through a finite temperature difference during the nopnisothermal heat-addition and heat-rejection processes, which are irreversible. Therefore, the thermal efficiency of an Otto or Diesel engine will be less than that of a Carnot engine operating between the same temperature limits.
- Consider a heat engine operating between a heat source at T_H and a heat sink at T_L. For the heat-engine cycle to be totally reversible, the temperature difference between the working fluid and the heat source (or sink) should never exceed a differential amount dT during any heat-transfer process. That is, both the heat-addition and heat-rejection processes during the cycle must take place isothermally, one at a temperature of T_H and the other at a temperature of T_L. This is precisely what happens in a Carnot cycle.
- There are two other cycles that involve an isothermal heat-addition process at T_H and an isothermal heat-rejection process at T_L: 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 (Fig. 5.11).

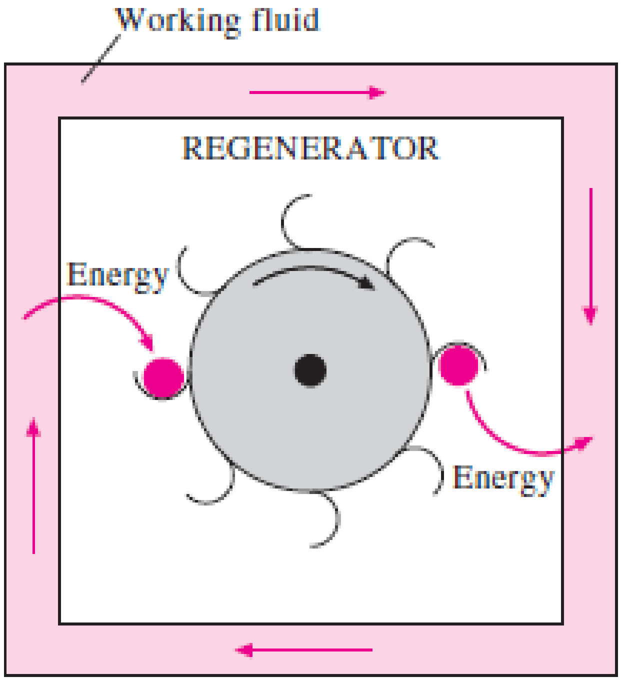


FIGURE 5.11: A regenerator is a device that borrows energy from the working fluid during one part of the cycle and pays it back (without interest) during another part.

- Figure 5.12(*b*) shows the *T*-*s* and *P*-*v* diagrams of the **Stirling cycle,** which is made up of four totally reversible processes:

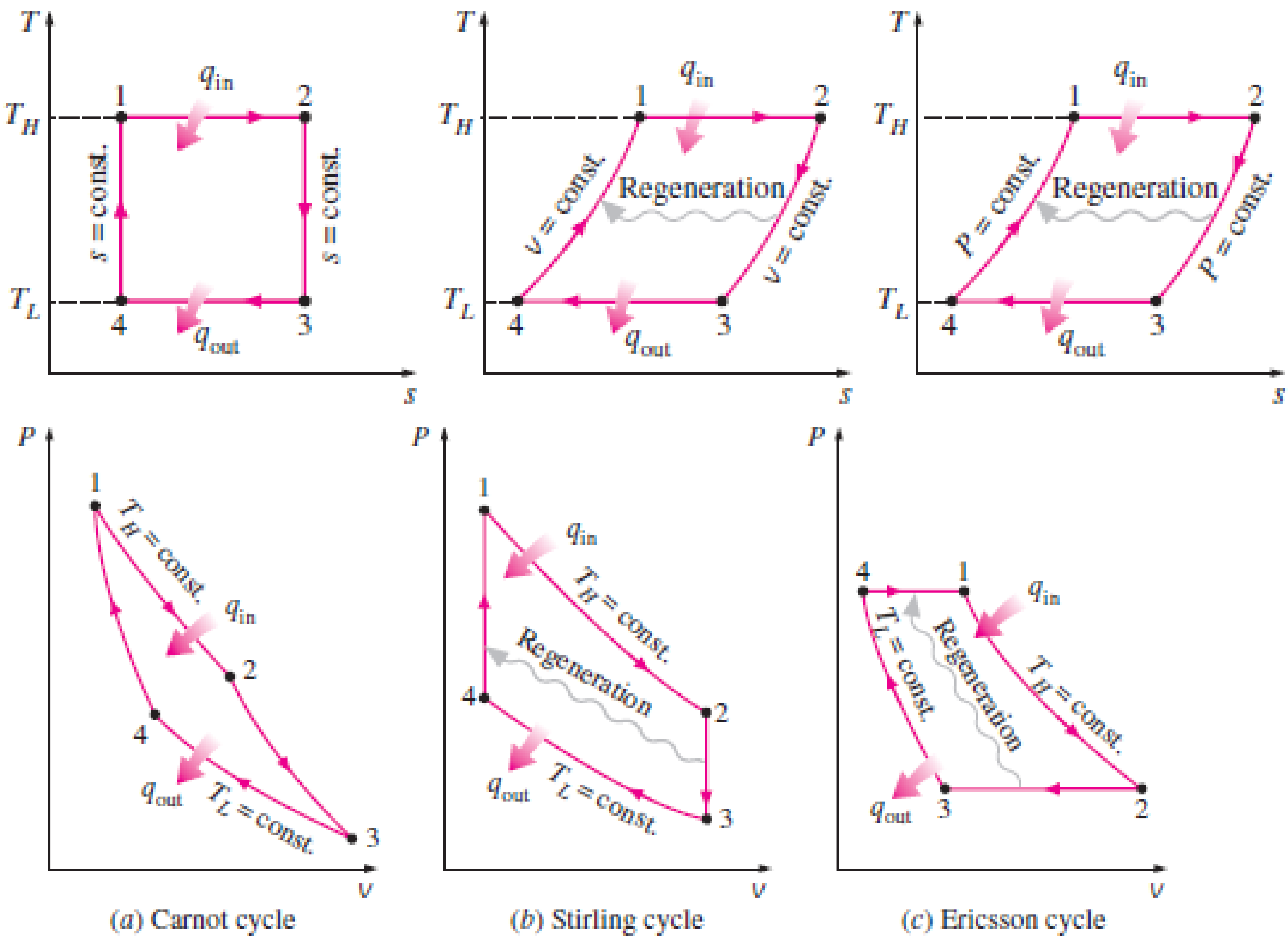


FIGURE 5.12: *T-s* and *P-v* diagrams of Carnot,Stirling, and Ericsson cycles.

The processes are:

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)

- To spare the reader the complexities, the execution of the Stirling cycle in a closed system is explained with the help of the hypothetical engine shown in Fig. 5.13.

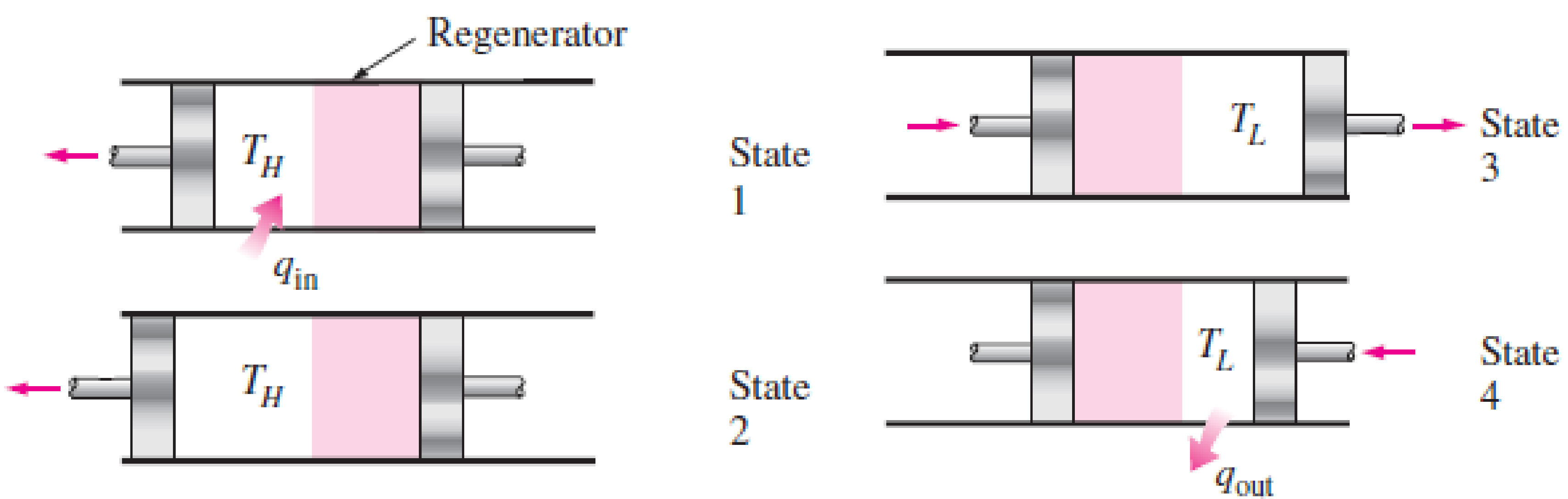


FIGURE 5.13: The execution of the Stirling cycle.

- This system consists of a cylinder with two pistons on each side and a regenerator in the middle.

- The regenerator can be a wire or a ceramic mesh or any kind of porous plug with a high thermal mass (mass times specific heat). It is used for the temporary storage of thermal energy. The mass of the working fluid contained within the regenerator at any instant is considered negligible.

5.6: Brayton cycle: The ideal cycle for gas-turbine engines

- Gas turbines usually operate on an ***open cycle***, as shown in Fig. 5.14.

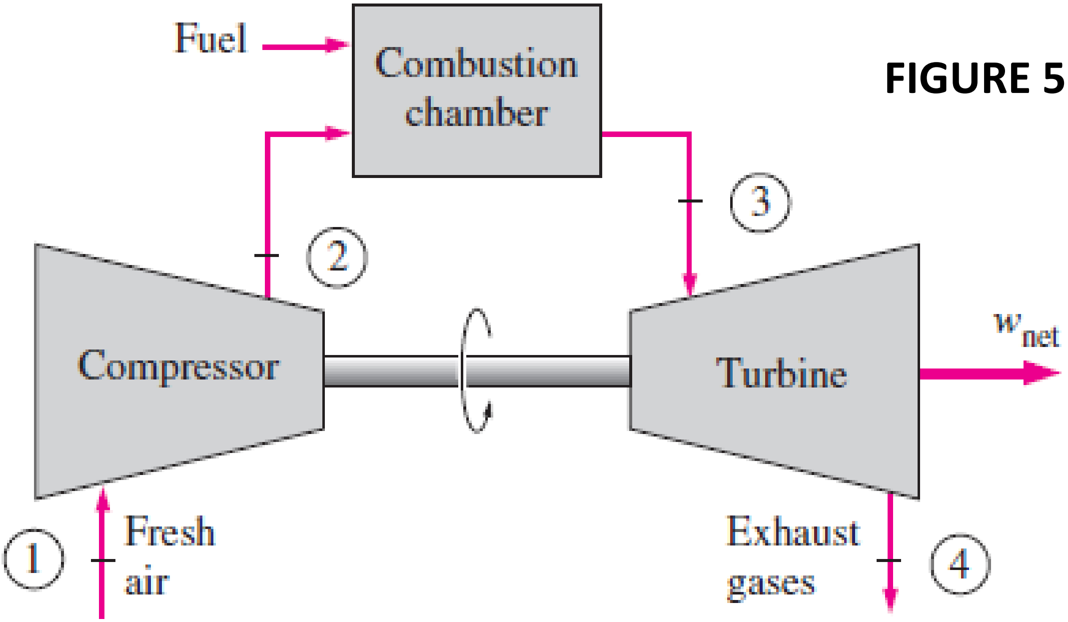


FIGURE 5.14: An open-cycle gas-turbine engine.

- 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 open gas-turbine cycle described above can be modeled as **a *closed cycle***, as shown in Fig. 5.15, by utilizing the air-standard assumptions.

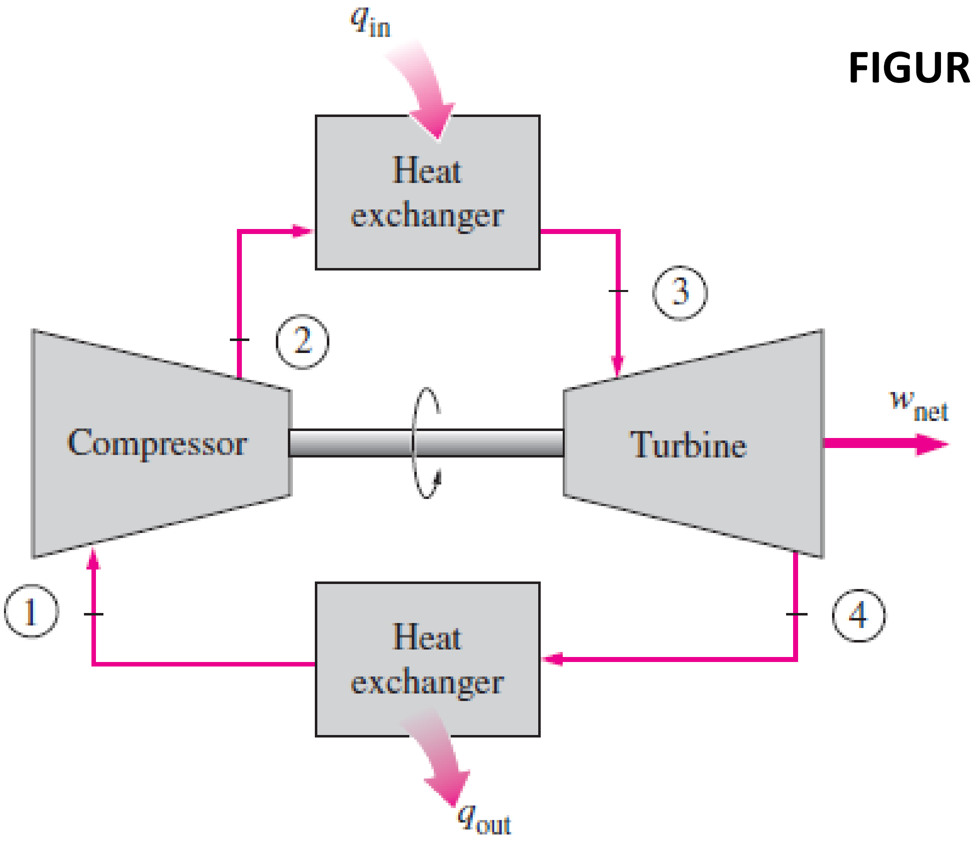


FIGURE 5.15: A closed-cycle gas-turbine engine.

- Here 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, and
  - ✓ the exhaust process is replaced by a constant pressure heat-rejection process to the ambient air.

- 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)

  4-1 Constant-pressure heat rejection

- The *T-s* and *P-v* diagrams of an ideal Brayton cycle are shown in Fig. 5.16.

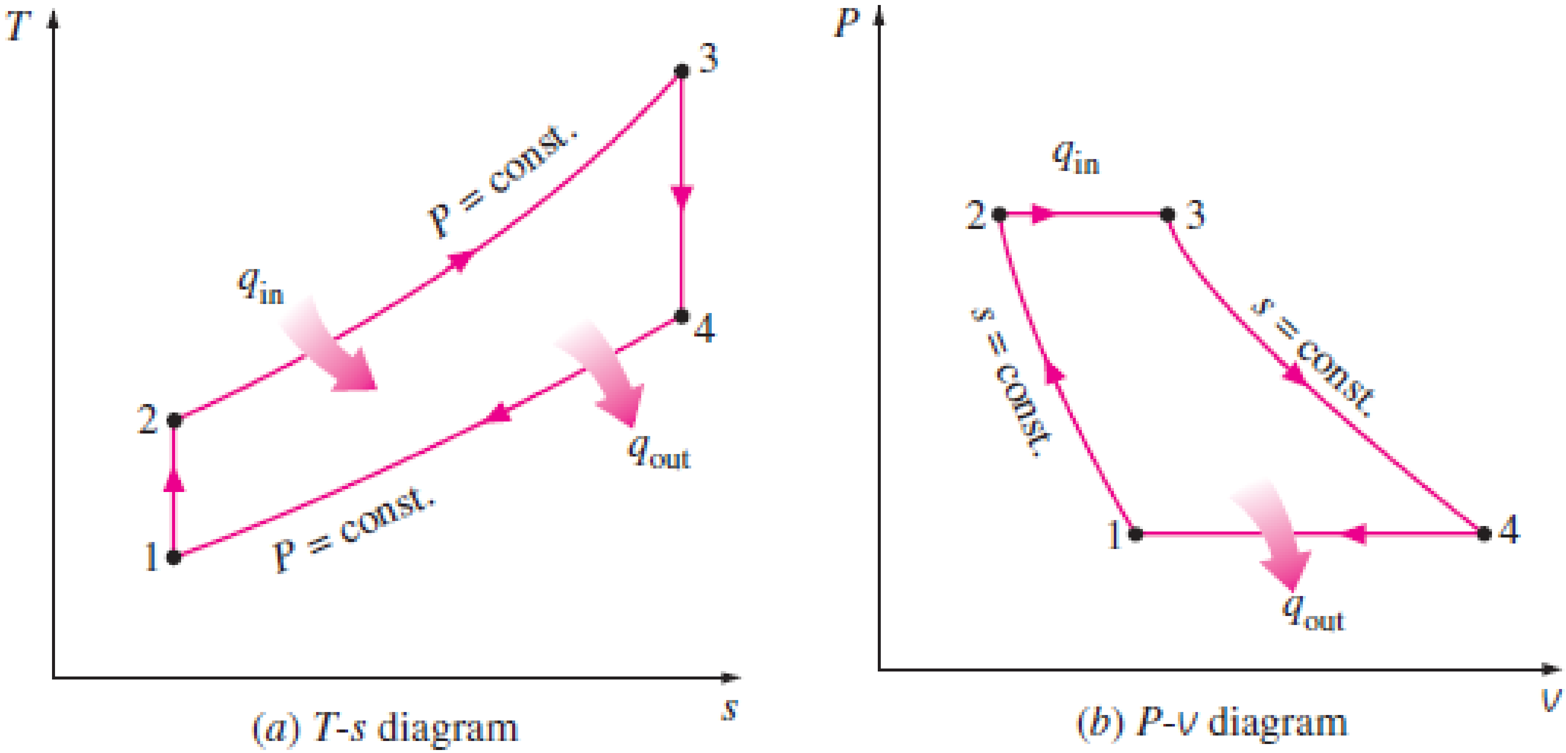


FIGURE 5.16: *T-s* and *P-v* diagrams for the ideal Brayton cycle.

- Notice that all four processes of the Brayton cycle are executed in steadyflow devices; thus, they should be analyzed as steady-flow processes. When the changes in kinetic and potential energies are neglected, the energy balance for a steady-flow process can be expressed, on a unit–mass basis, as

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

Therefore, heat transfers to and from the working fluid are

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

and

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

- Then the thermal efficiency of the ideal Brayton cycle under the cold-airstandard assumptions becomes

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

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

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

Substituting these equations into the thermal efficiency relation and simplifying give

$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} \quad \text{..........f}$$

where

$$r_p = \frac{P_2}{P_1} \quad \text{..........g}$$

r_p- is the **pressure ratio** and *k* is the specific heat ratio.

- Equation f shows that under the cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle depends on the pressure ratio of the gas turbine and the specific heat ratio of the working fluid.
- The thermal efficiency increases with both of these parameters, which is also the case for actual gas turbines.
- The two major application areas of gas-turbine engines are *aircraft propulsion* and *electric power generation.*

- In gas-turbine power plants, the ratio of the compressor work to the turbine work, called the **back work ratio.** Usually
- more than one-half of the turbine work output is used to drive the compressor

EXAMPLE 1. **The Simple Ideal Brayton Cycle**

- A gas-turbine power plant operating on an ideal Brayton cycle has a pressure ratio of 8. The gas temperature is 300 K at the compressor inlet and 1300K at the turbine inlet. Utilizing the air-standard assumptions, determine

  (*a*) the gas temperature at the exits of the compressor and the turbine,

  (*b*) the back work ratio, and

  (*c*) the thermal efficiency.

Deviation of Actual Gas-Turbine Cycles from Idealized Ones

- The actual gas-turbine cycle differs from the ideal Brayton cycle on several accounts.
  - For one thing, some pressure drop during the heat-addition and heat-rejection processes is inevitable.
  - More importantly, the actual work input to the compressor is more, and the actual work output from the turbine is less because of irreversibilities.
- The deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accurately accounted for by utilizing the **isentropic efficiencies of the turbine and compressor** as:

$$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1} \quad \text{........h} \qquad \text{and} \qquad \eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \quad \text{......i}$$

- where states 2_a and 4_a are the actual exit states of the compressor and the turbine, respectively, and 2_s and 4_s are the corresponding states for the isentropic case, as illustrated in Fig. 5.17.

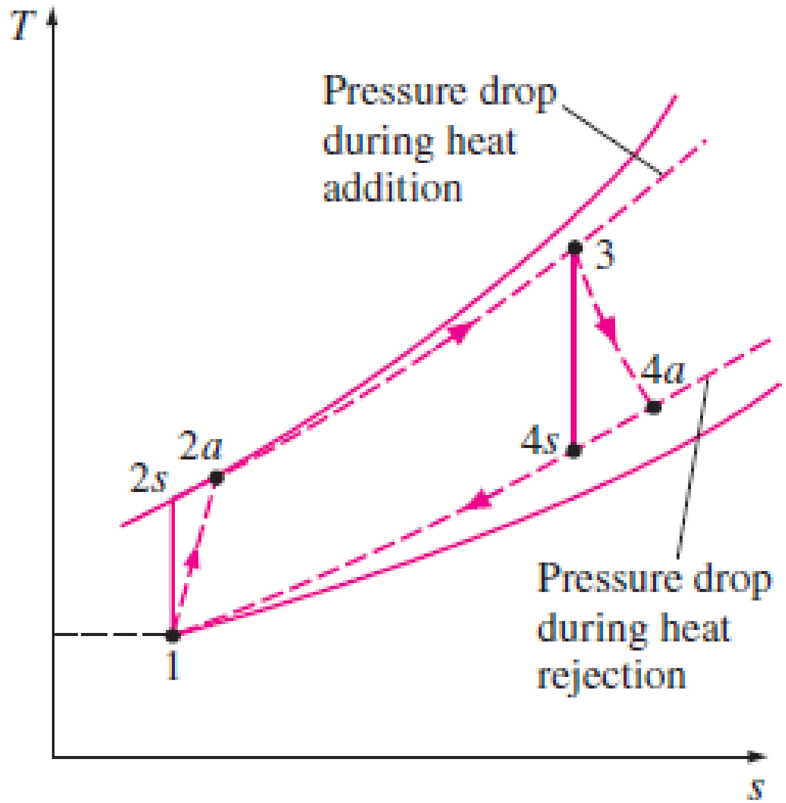


FIGURE 5.17: The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

- The effect of the turbine and compressor efficiencies on the thermal efficiency of the gas-turbine engines is illustrated below with an example.

- Assuming a compressor efficiency of 80 percent and a turbine efficiency of 85 percent, determine (*a*) the back work ratio, (*b*) the thermal efficiency, and (*c*) the turbine exit temperature of the gas-turbine cycle discussed in Example 1 above.