Air Standard Cycles

Gas Power Cycles

- Most power-producing devices operate on cycles and can be studies using Thermodynamics

- The cycles encountered in actual devices are difficult to analyze because of the presence of complicating effects, such as friction, and the absence of sufficient time for establishment of the equilibrium conditions during the cycle.

- When the actual cycle is stripped of all the internal irreversibilities and complexities, we end up with a cycle that resembles the actual cycle closely but is made up totally of internally reversible processes. Such a cycle is called an **ideal cycle**

- A simple idealized model enables engineers to study the effects of the major parameters that dominate the cycle without getting bogged down in the details

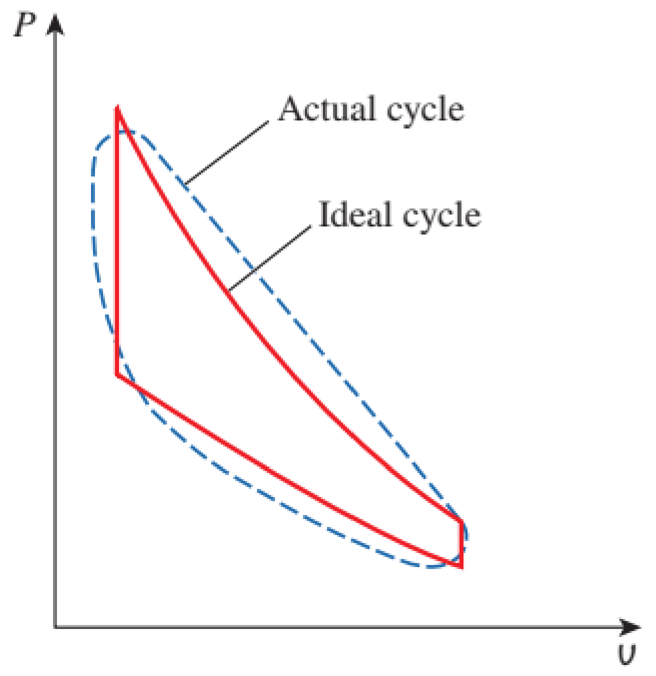


FIGURE 10–2
The analysis of many complex processes can be reduced to a manageable level by utilizing some idealizations.

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 quasi-equilibrium manner.
3. The pipes connecting the various components of a system are well insulated, and heat transfer through them is negligible.
4. Neglecting the changes in kinetic and potential energies of the working fluid (The only devices where the changes in kinetic energy are significant are the nozzles and diffusers)

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:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}} \tag{10–1}$$

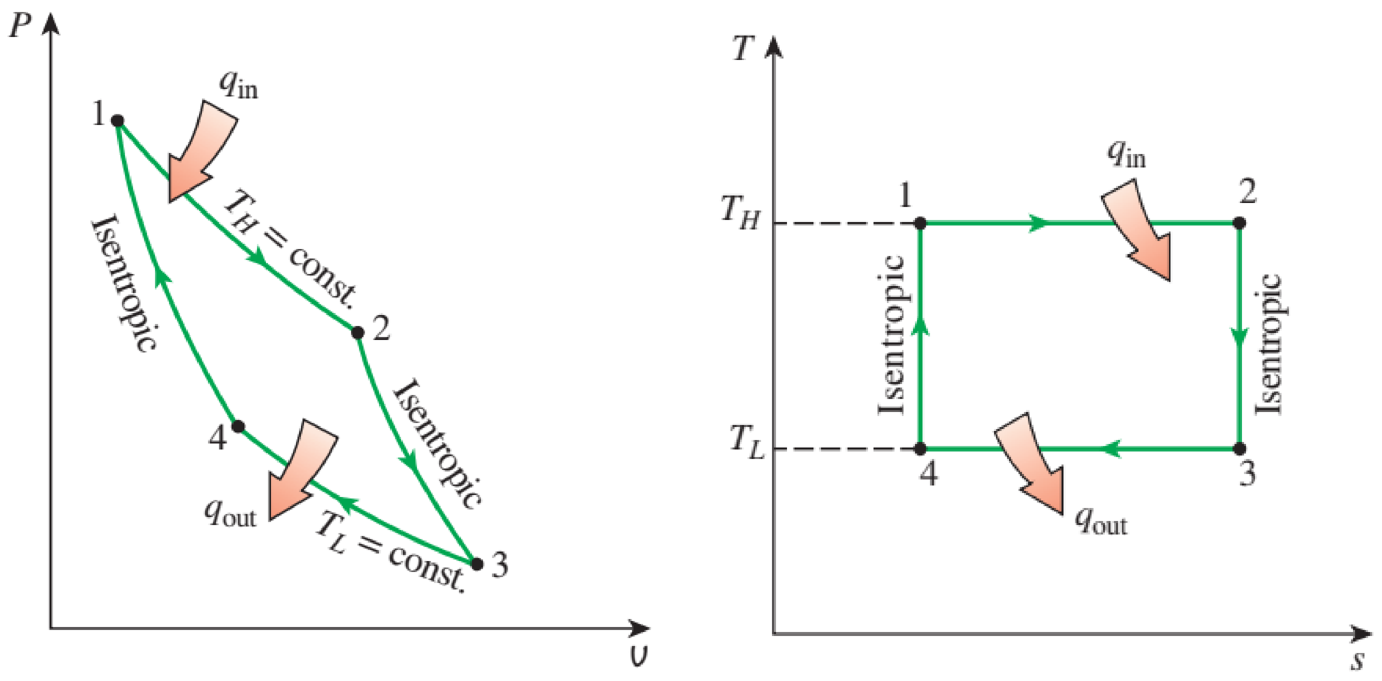
- Recall that heat engines that operate on a totally reversible cycle, such as the Carnot cycle, have the highest thermal efficiency of all heat engines operating between the same temperature levels. That is, nobody can develop a cycle more efficient than the *Carnot cycle.*

- Most cycles encountered in practice differ significantly from the Carnot cycle, which makes it unsuitable as a realistic model. Each ideal cycle discussed in this chapter is related to a specific work producing device and is an *idealized* version of the actual cycle

THE CARNOT CYCLE

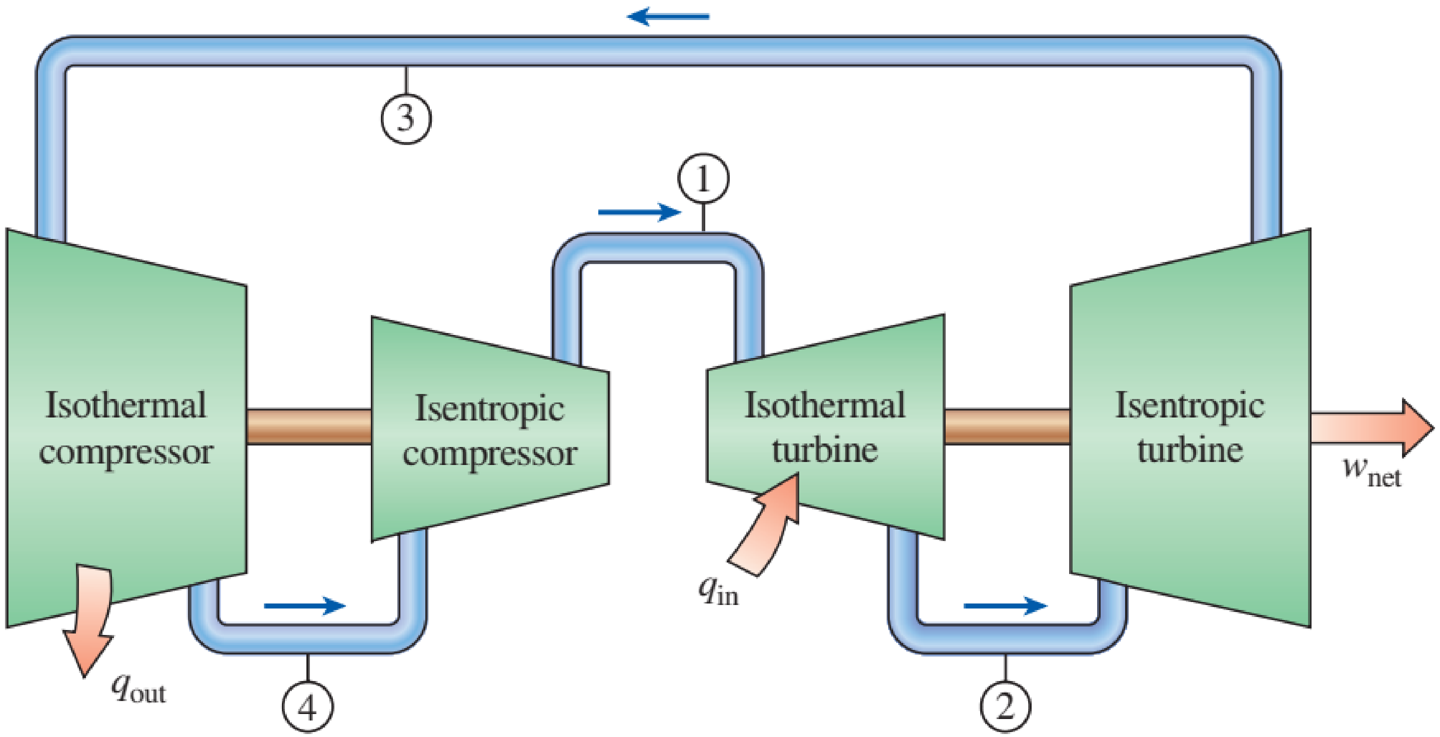
The Carnot cycle is composed of four totally reversible processes:

1-2 isothermal heat addition
2-3 isentropic expansion
3-4 isothermal heat rejection
4-1 isentropic compression.



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

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



- Reversible isothermal heat transfer is very difficult to achieve in reality because it would require very large heat exchangers and it would take a very long time (a power cycle in a typical engine is completed in a fraction of a second).
- Therefore, it is not practical to build an engine that would operate on a cycle that closely approximates the Carnot cycle.

AIR-STANDARD ASSUMPTIONS

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.

4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state

5. air has constant specific heats whose values are determined at *room temperature*

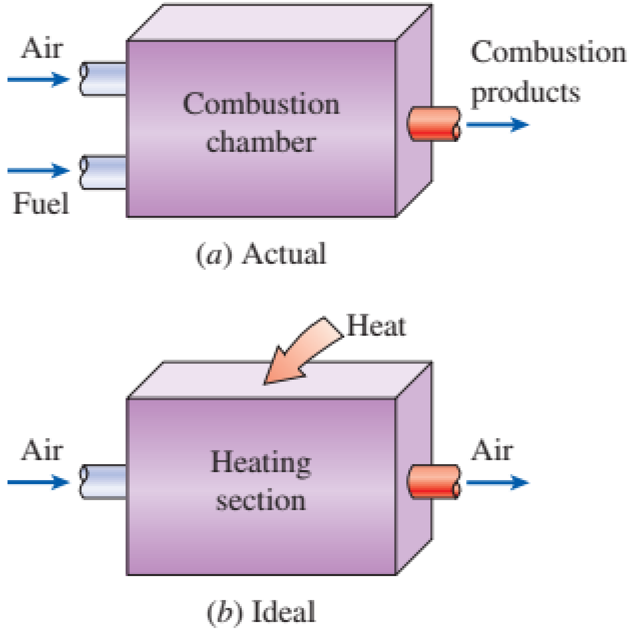


FIGURE 10–8
The combustion process is replaced by a heat-addition process in ideal cycles.

RECIPROCATING ENGINES – An overview

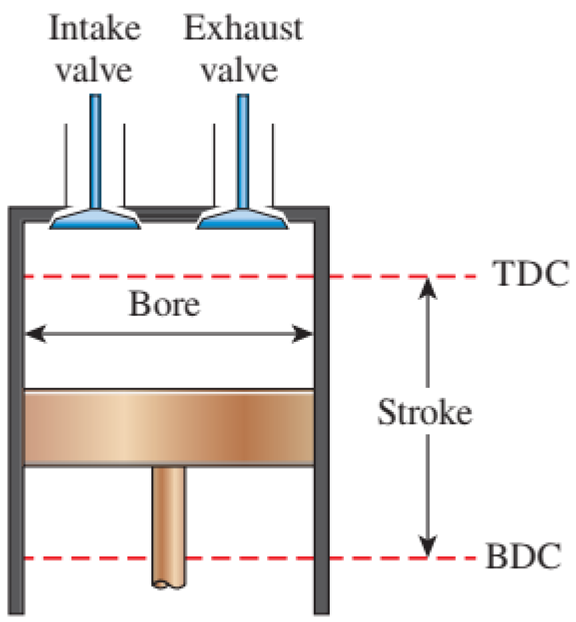


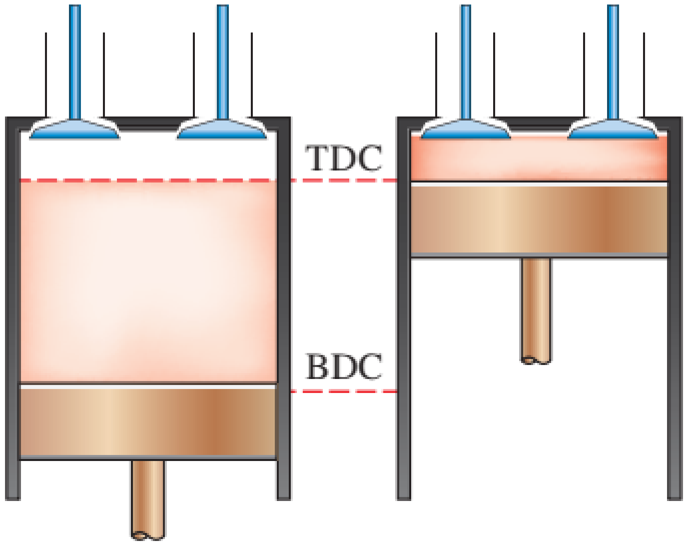
FIGURE 10–10
Nomenclature for reciprocating engines.

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**, and the combustion products are expelled from the cylinder through the **exhaust valve**.



(*a*) Displacement volume

(*b*) Clearance volume

The minimum volume formed in the cylinder when the piston is at TDC is called the **clearance volume** .
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:

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

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

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

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

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

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.

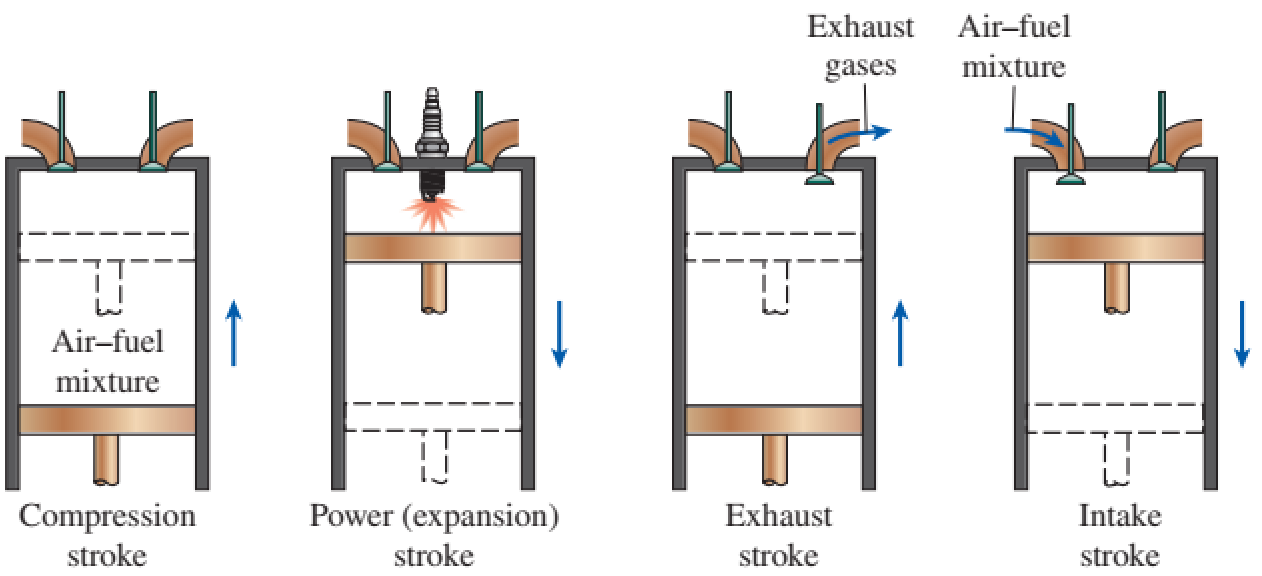
In SI engines, the combustion of the air–fuel mixture is initiated by a spark plug.

In CI engines, the air–fuel mixture is self-ignited as a result of compressing the air above the self-ignition temperature of the fuel and injecting fuel into the compressed air.

Otto cycle and *Diesel cycle* are the ideal cycles for the SI and CI reciprocating engines

OTTO CYCLE

- Ideal cycle for Spark Ignition Engines
- named after Nikolaus A. Otto, who built a successful four-stroke engine in 1876 in Germany
- 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*) Actual four-stroke spark-ignition engine

In **two-stroke engines**, all four functions are executed in just two strokes: the power stroke and the compression stroke.

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*. Toward the end of expansion stroke, the exhaust valve opens and the combustion gases that are above the atmospheric pressure rush out of the cylinder through the open exhaust valve. This process is called **exhaust blowdown**, and most combustion gases leave the cylinder by the time the piston reaches BDC. The cylinder is still filled by the exhaust gases at a lower pressure at BDC. 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.

In **two-stroke engines**, all four functions are executed in just two strokes: the power stroke and the compression stroke.

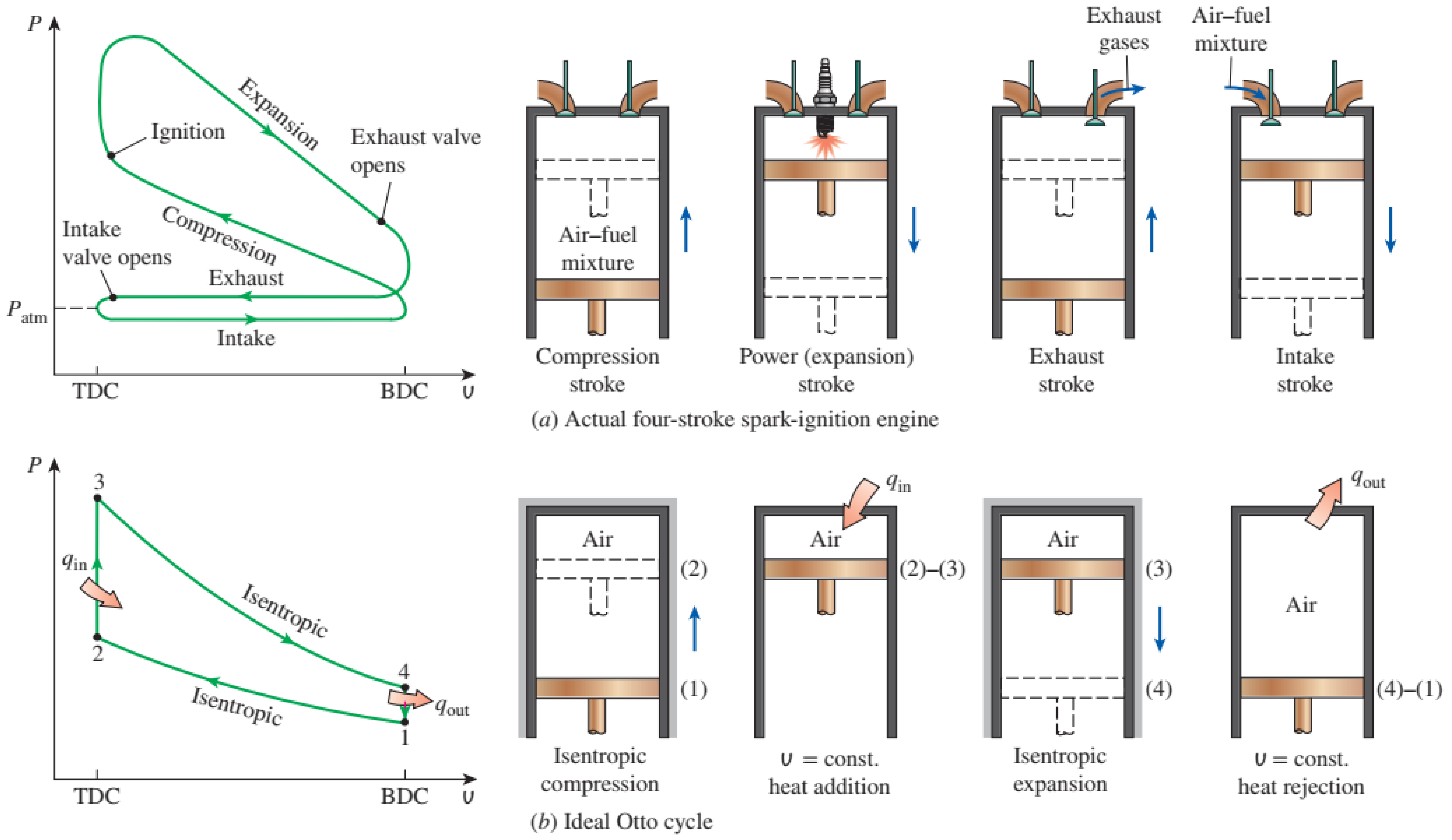
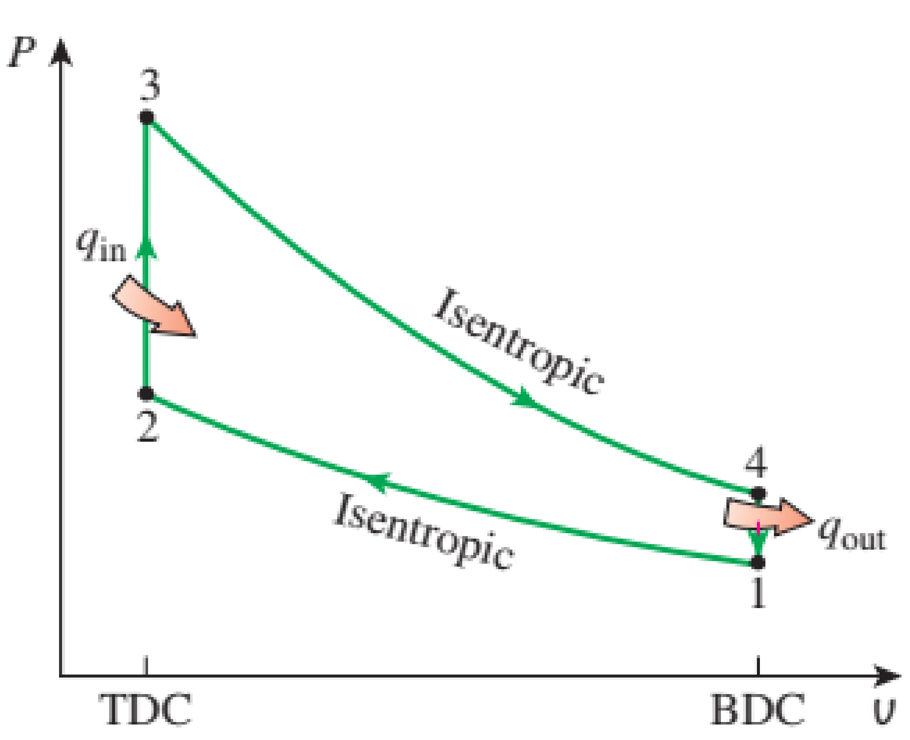


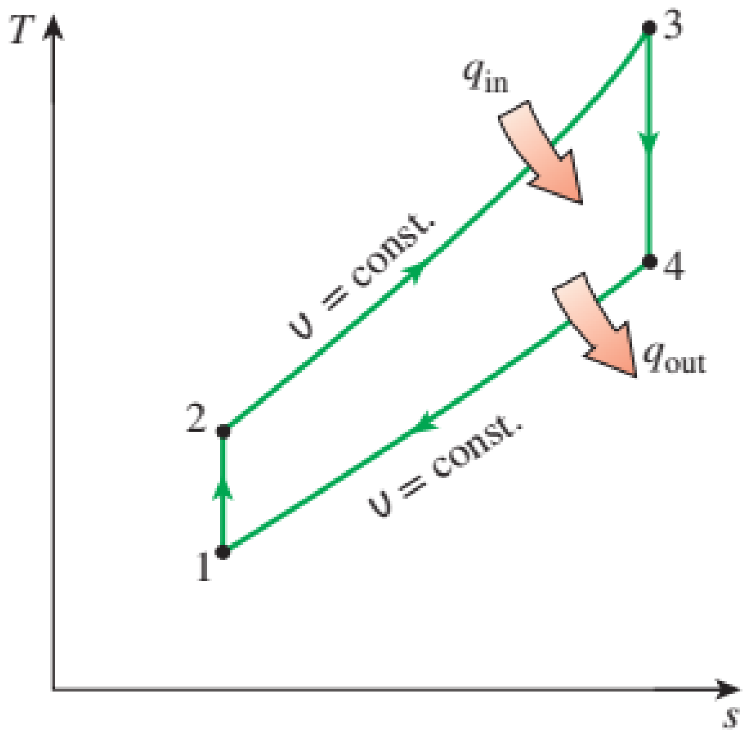
FIGURE 10–13

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

Otto Cycle 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





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

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u$$

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

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

and

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

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

$$\eta_{\text{th,Otto}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}}$$

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

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

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

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

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

Substituting these equations into the thermal efficiency relation and simplifying give

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

where,

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

r is the compression ratio and *k* is the specific heat ratio c_p/c_v.

- 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.
- It increases with an increase in both Compression ratio and specific heat ratio

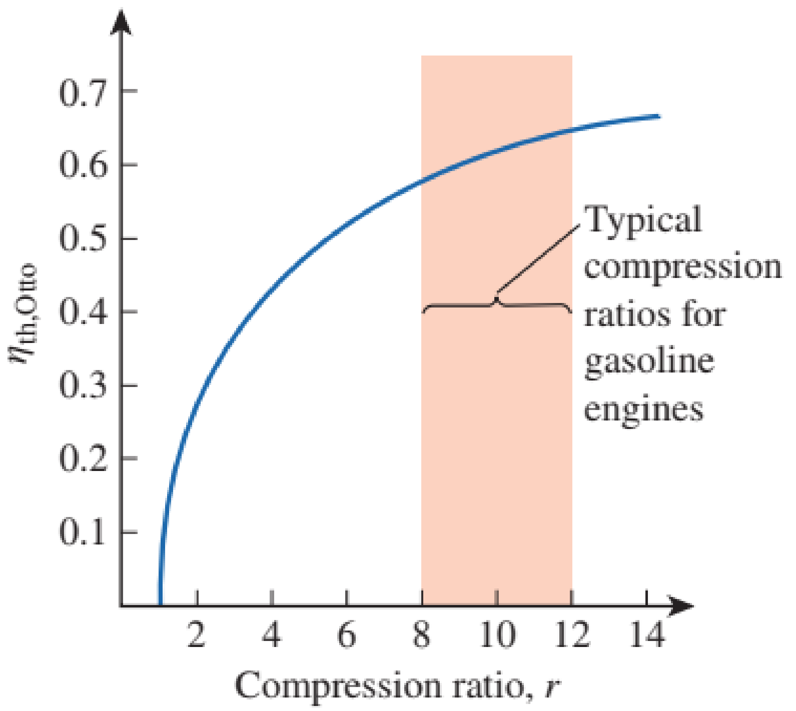


FIGURE 10–18
Thermal efficiency of the ideal Otto cycle as a function of compression ratio ($k = 1.4$).

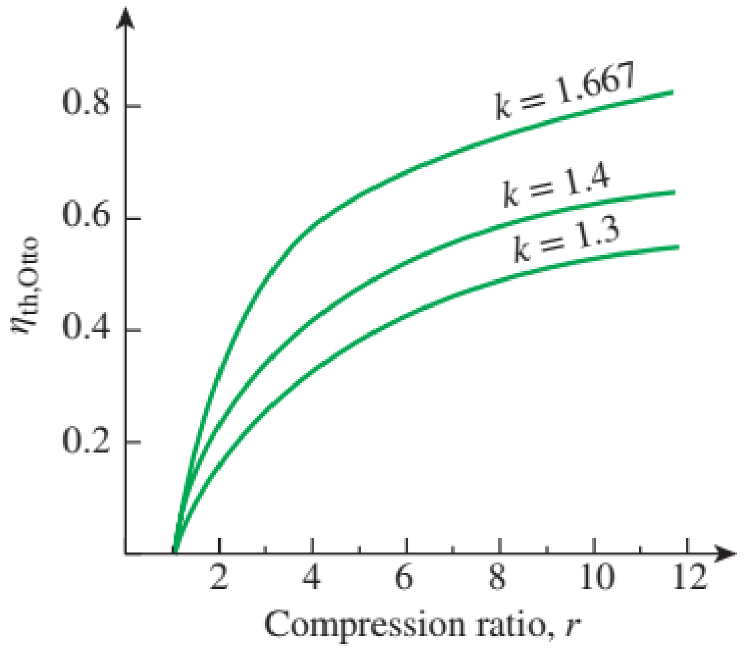


FIGURE 10–19
The thermal efficiency of the Otto cycle increases with the specific heat ratio k of the working fluid.

DIESEL CYCLE

- Diesel cycle is the ideal cycle for CI reciprocating engines
- first proposed by Rudolph Diesel in the 1890s

- 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.
- 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. the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat-addition process.

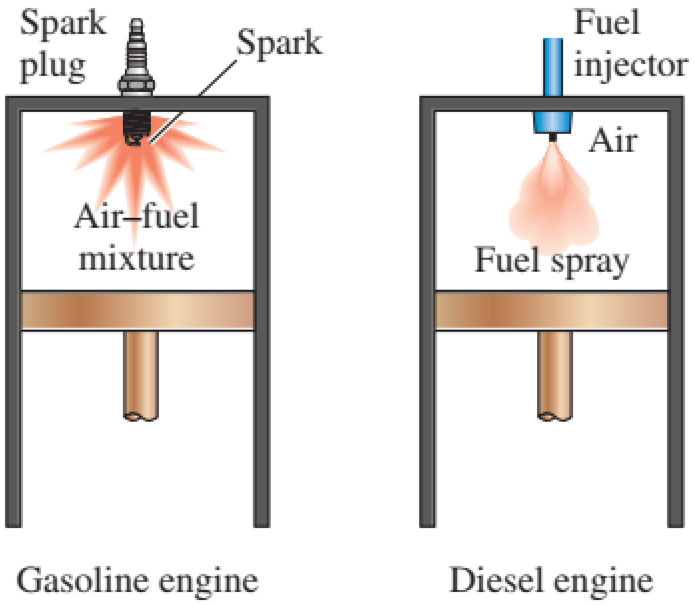
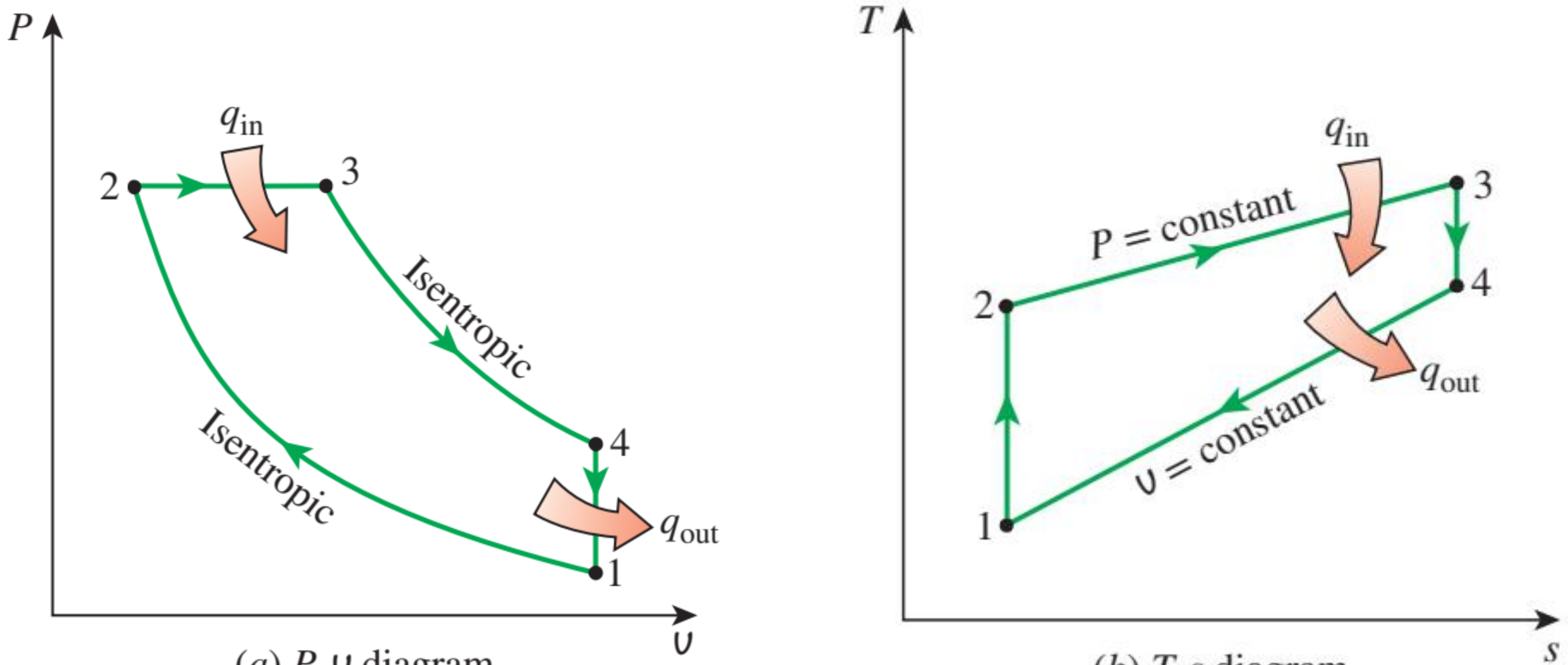


FIGURE 10–21
In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

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



(a) P-υ diagram

(b) T-s diagram

the amount of heat transferred to the working fluid at constant pressure

$$q_{in} - w_{b,out} = u_3 - u_2$$

$$q_{in} = P_2(v_3 - v_2) + (u_3 - u_2)$$

$$= h_3 - h_2$$

$$= c_p(T_3 - T_2)$$

the amount of heat rejected from it at constant volume can be expressed as

$$-q_{out} = u_1 - u_4$$

$$q_{out} = u_4 - u_1$$

$$= c_v(T_4 - T_1)$$

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

$$\eta_{\text{th,Diesel}} = \frac{w_{\text{net}}}{q_{\text{in}}}$$

$$= 1 - \frac{q_{\text{out}}}{q_{\text{in}}}$$

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

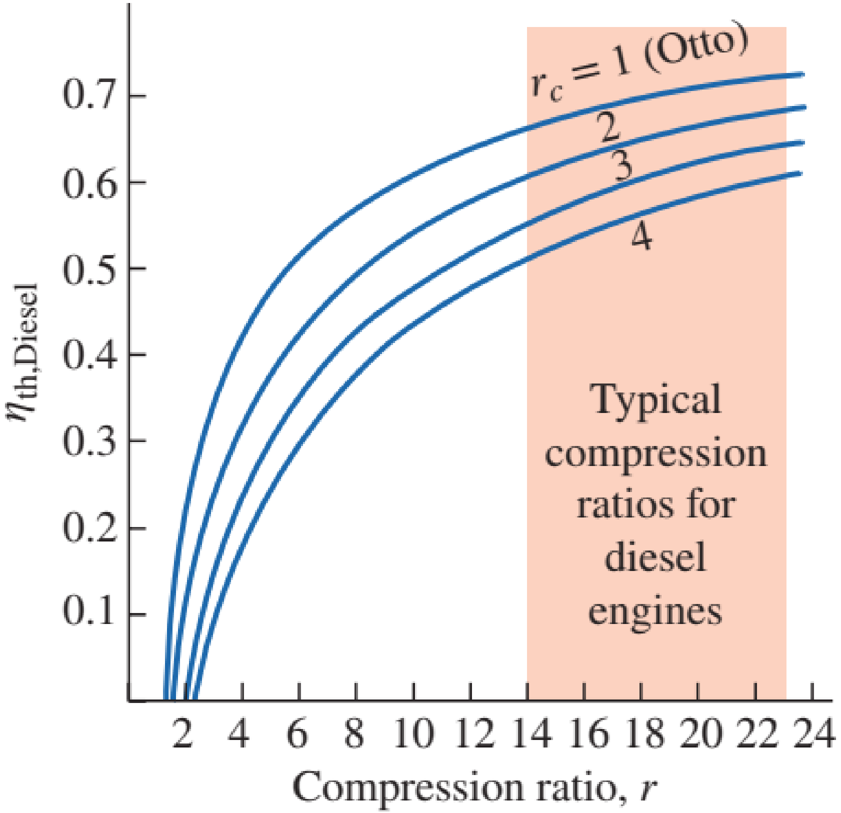
$$= 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)}$$

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

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



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

Dual Cycle

- In modern high-speed compression ignition engines, fuel is injected into the combustion chamber much sooner than in the early diesel engines.
- Fuel starts to ignite late in the compression stroke, and consequently part of the combustion occurs almost at constant volume.
- Fuel injection continues until the piston reaches the top dead center, and combustion of the fuel keeps the pressure high well into the expansion stroke.
- Thus, the entire combustion process can be better modeled as the combination of constant-volume and constant-pressure processes.
- The ideal cycle based on this concept is called the **dual cycle**,

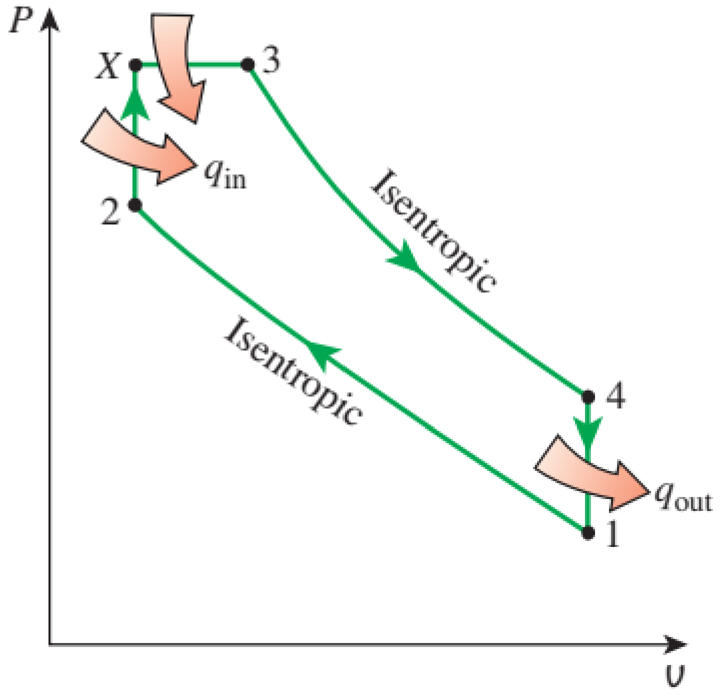
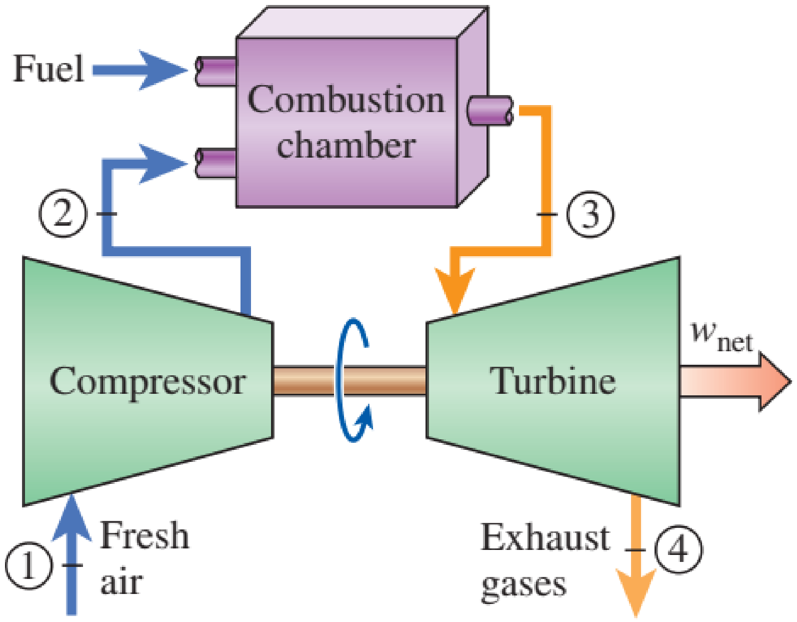


FIGURE 10–24
P-υ diagram of an ideal dual cycle.

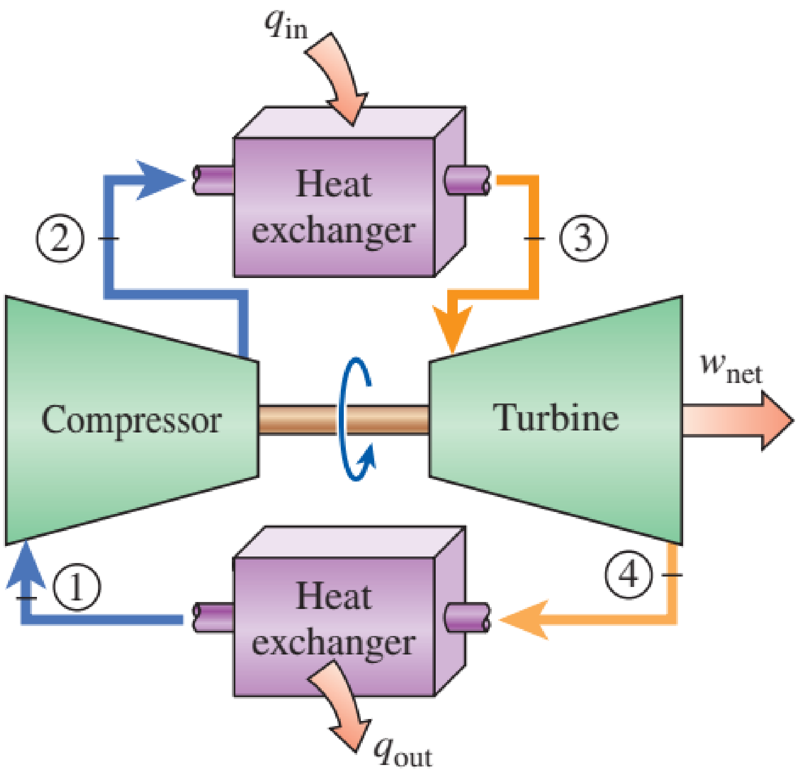
BRAYTON CYCLE

IDEAL CYCLE FOR GAS-TURBINE ENGINES

Actual cycle (Open cycle)

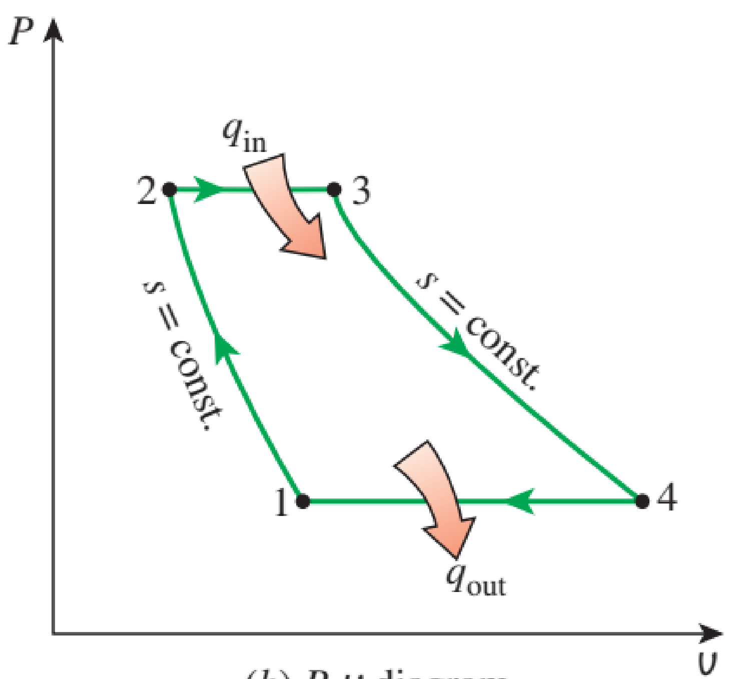


Ideal cycle (Closed cycle)

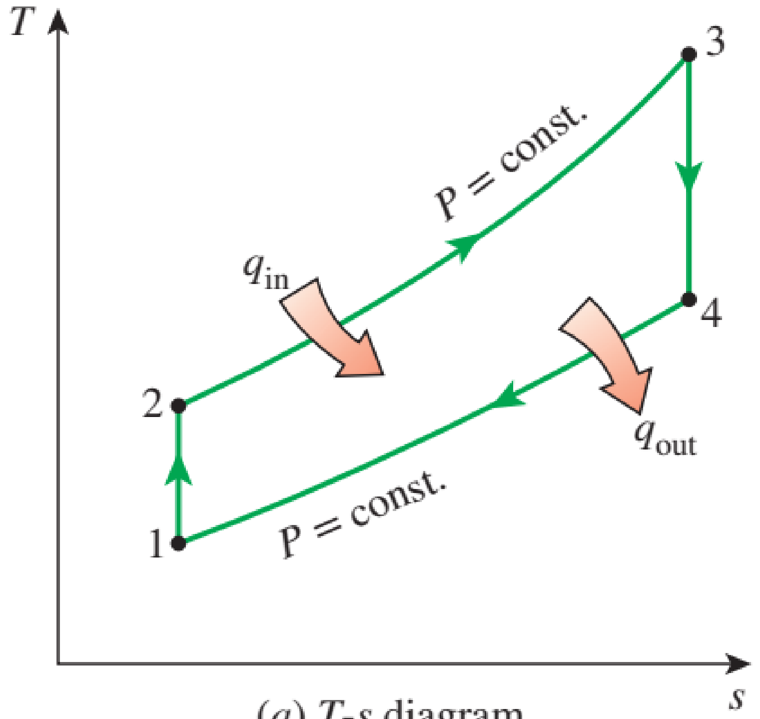


The ideal cycle that the working fluid undergoes in this closed loop is the **Brayton cycle**, which is made up of four internally reversible processes:

1-2 Isentropic compression (in a compressor)
2-3 Constant-pressure heat addition
3-4 Isentropic expansion (in a turbine)
4-1 Constant-pressure heat rejection



(b) P-υ diagram



(a) T-s diagram

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

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

Therefore, heat added to the working fluid is

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

Heat removed from the working fluid is

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

Then the thermal efficiency of the ideal Brayton cycle

$$\eta_{th,Brayton} = \frac{w_{net}}{q_{in}}$$

$$= 1 - \frac{q_{out}}{q_{in}} :$$

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

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

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

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

$$= \left(\frac{P_3}{P_4}\right)^{(k-1)/k}$$

$$= \frac{T_3}{T_4}$$

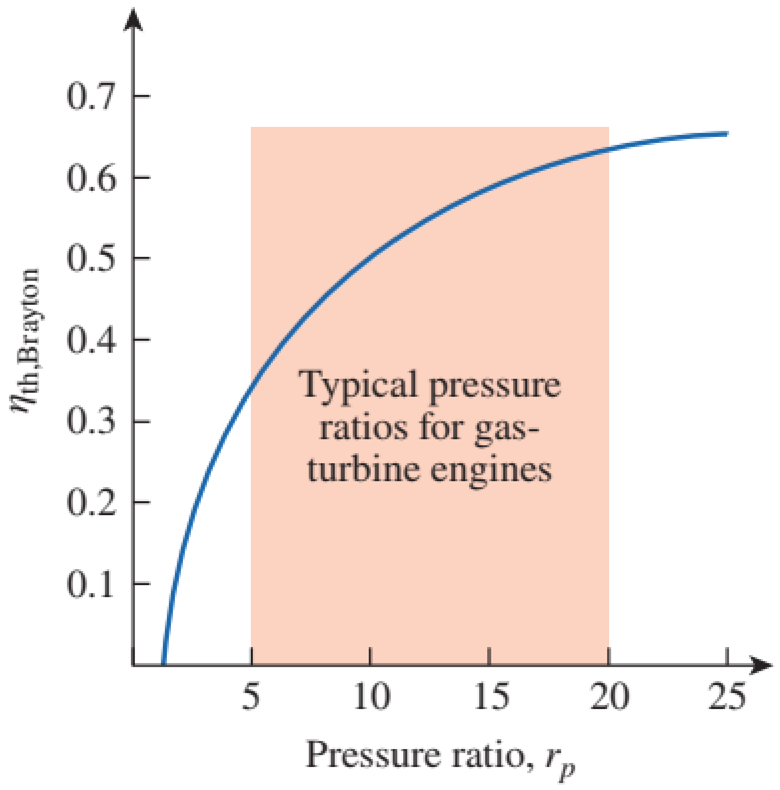
Substituting these equations into the thermal efficiency relation and simplifying give

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

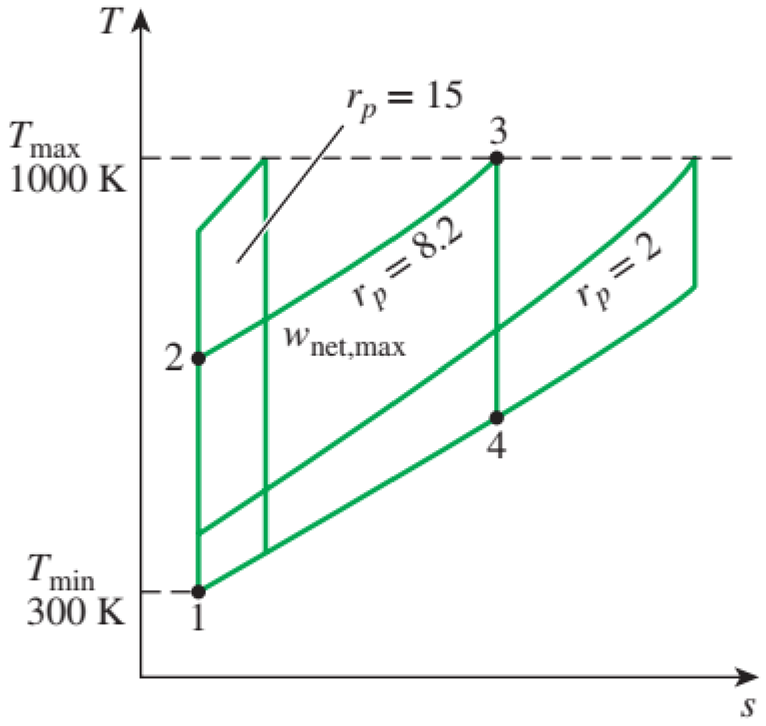
Where, $r_p = \dfrac{P_2}{P_1}$ is the **pressure ratio** and *k* is the specific heat ratio.

- The thermal efficiency of an ideal Brayton cycle increases with both pressure ratio of the gas turbine and the specific heat ratio of the working fluid.

- 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.



For a fixed turbine inlet temperature T_3, the net work output per cycle increases with the pressure ratio, reaches a maximum, and then starts to decrease, as shown the figure. Therefore, there should be a compromise between the pressure ratio (thus the thermal efficiency) and the net work output

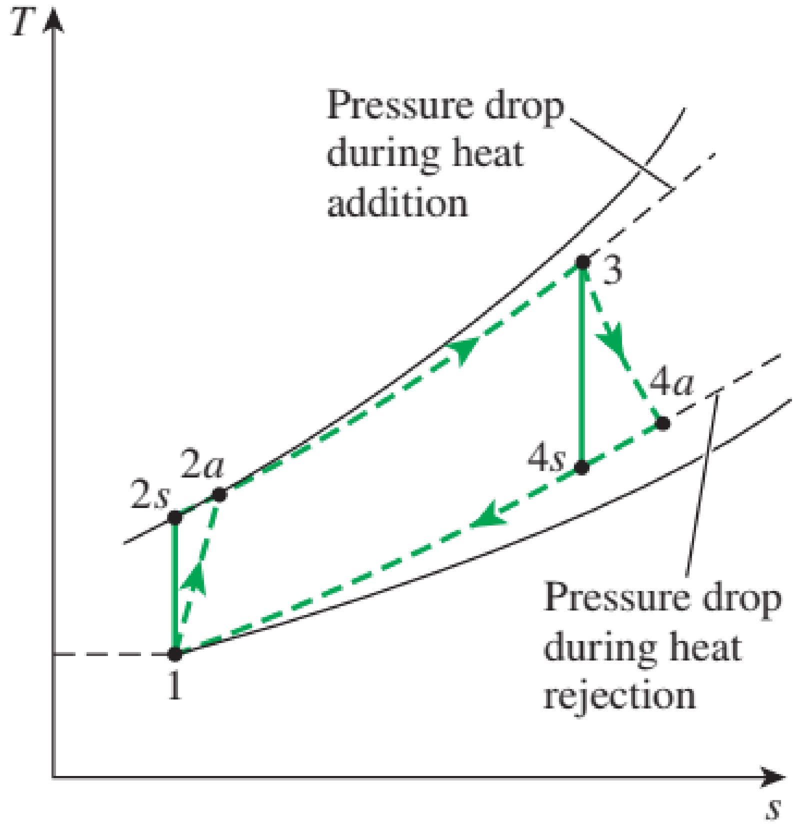


Isentropic efficiency of Turbine and Compressor

- The actual gas-turbine cycle differs from the ideal Brayton cycle on several accounts.
- 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

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

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



Effect of Reheat, Regeneration and Intercooling

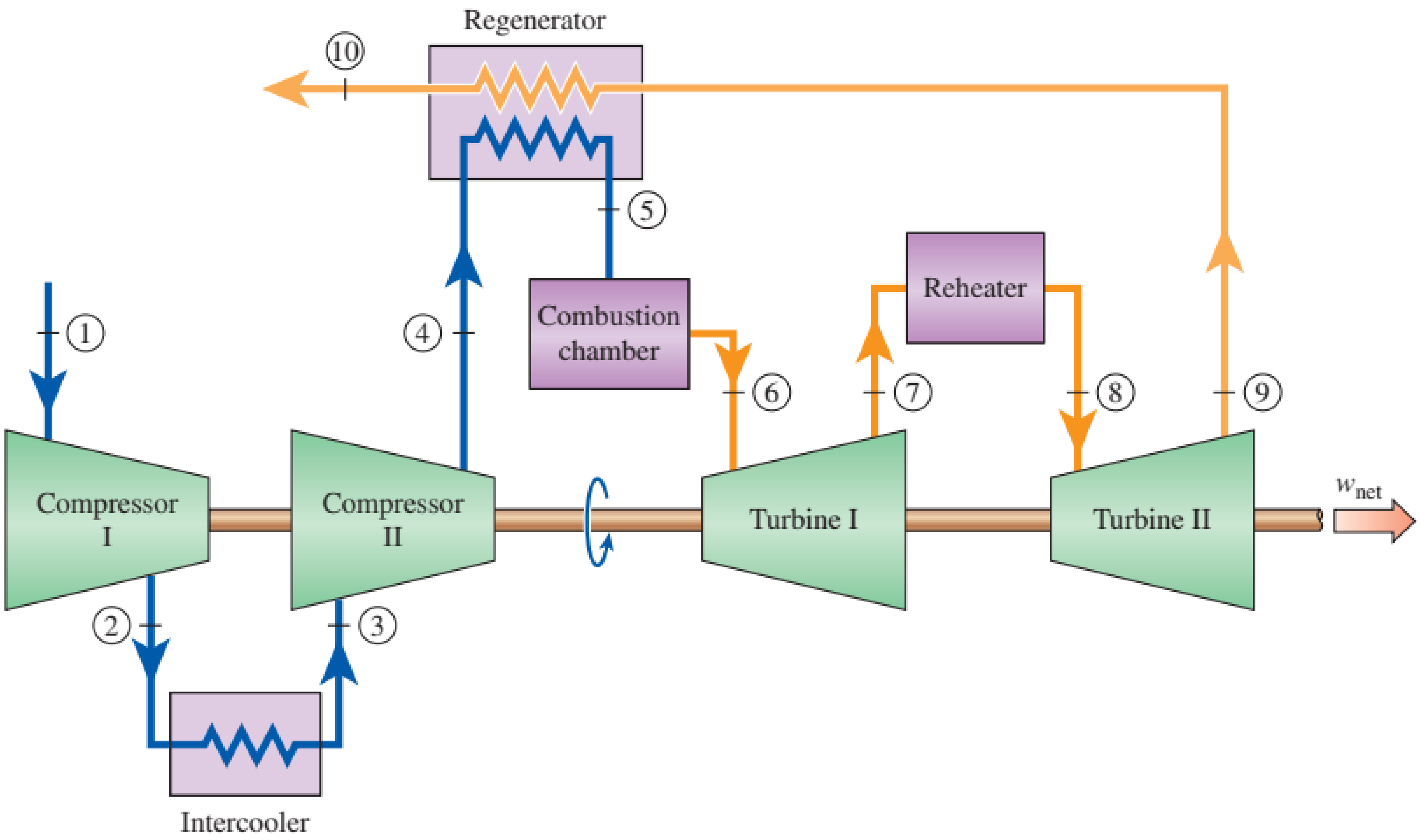


FIGURE 10–44
A gas-turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration.