Thermodynamics: An Engineering Approach
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CHAPTER 9
GAS POWER CYCLES

Lecture slides by
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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.
- Solve problems based on the Otto and Diesel cycles.
- Solve problems based on the Brayton cycle.

BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

Thermal efficiency of heat engines:

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

Most power-producing devices operate on cycles.

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

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

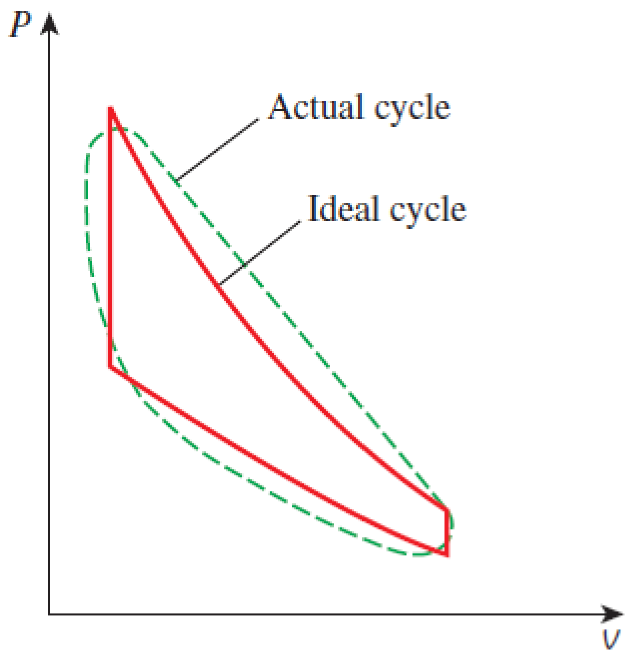


FIGURE 9–2
The analysis of many complex processes can be reduced to a manageable level by utilizing some idealizations.

The ideal cycles are *internally reversible*, but, unlike the Carnot cycle, they are not necessarily externally reversible.

Therefore, the thermal efficiency of an ideal cycle, in general, is less than that of a totally reversible cycle operating between the same temperature limits.

However, it is still considerably higher than the thermal efficiency of an actual cycle because of the idealizations utilized.

The idealizations and simplifications in the analysis of power cycles:

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

AIR-STANDARD ASSUMPTIONS

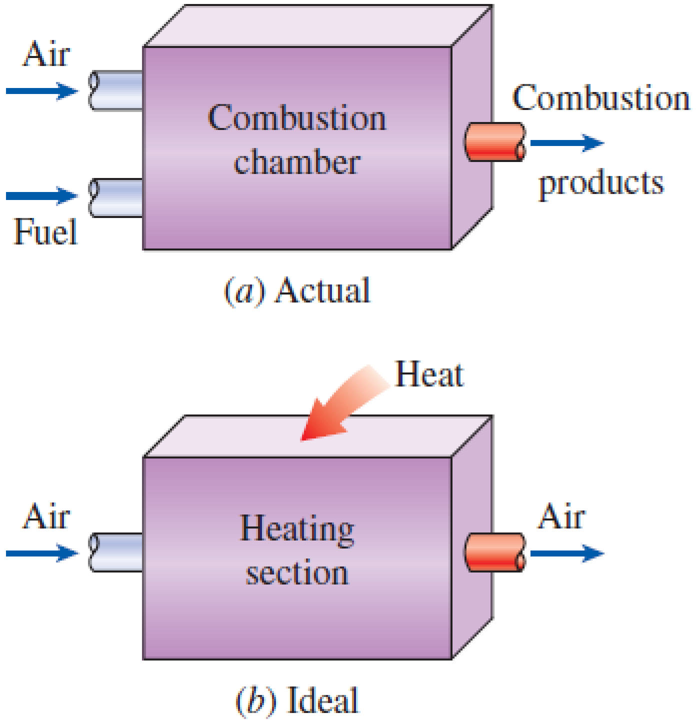


FIGURE 9–8
The combustion process is replaced by a heat-addition process in ideal cycles.

Air-standard assumptions:

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

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

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

AN OVERVIEW OF RECIPROCATING ENGINES

Compression ratio

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

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

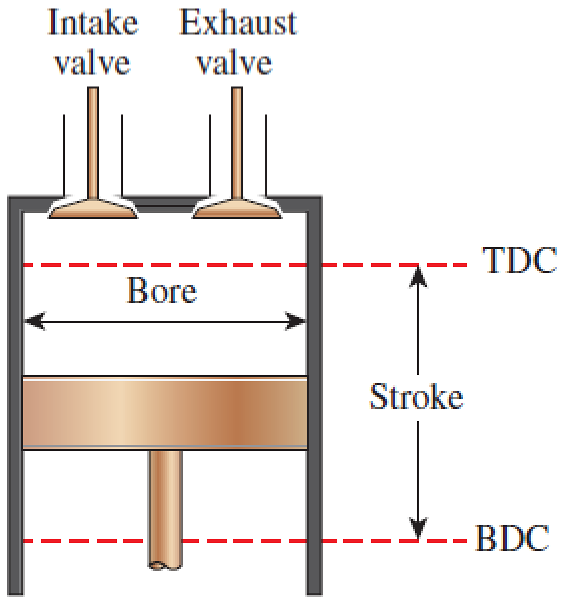


FIGURE 9–9
Nomenclature for reciprocating engines.

TDC: Top Dead Center

BDC: Bottom Dead Center

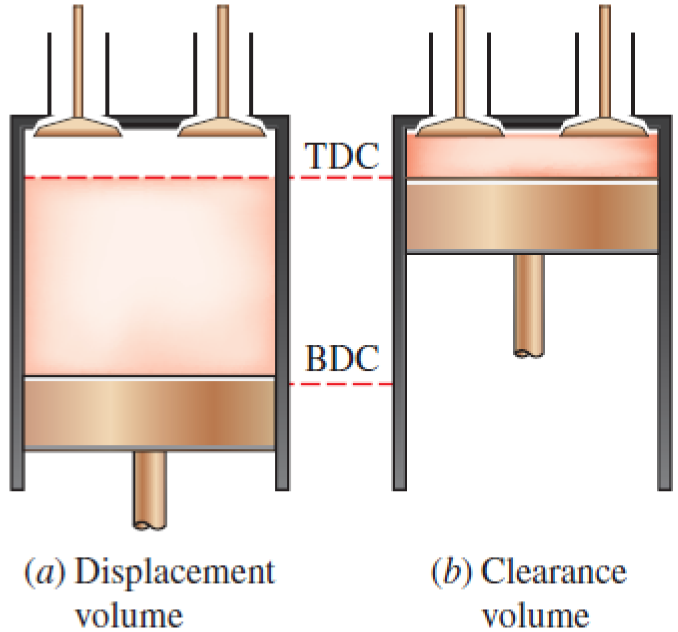


FIGURE 9–10
Displacement and clearance volumes of a reciprocating engine.

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

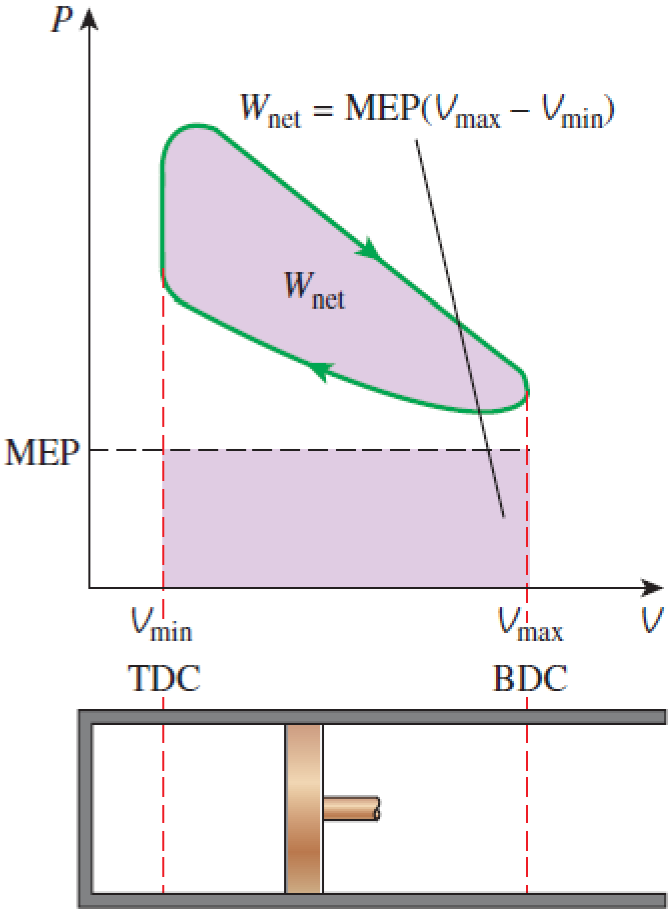


FIGURE 9–11
The net work output of a cycle is equivalent to the product of the mean effective pressure and the displacement volume.

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

Mean effective pressure

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.

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

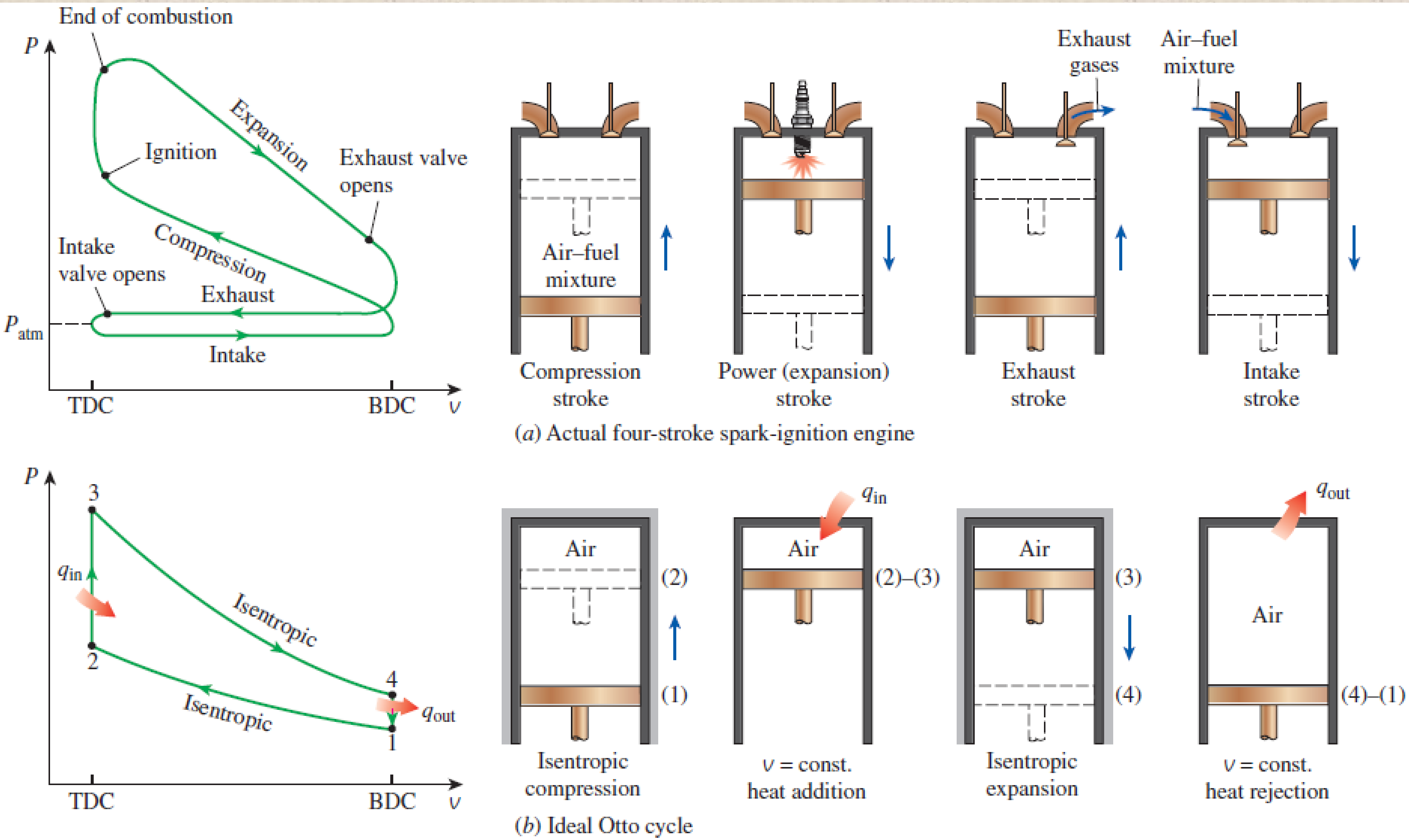


FIGURE 9–12

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

Four-stroke cycle

1 cycle = 4 stroke = 2 revolution

Two-stroke cycle

1 cycle = 2 stroke = 1 revolution

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

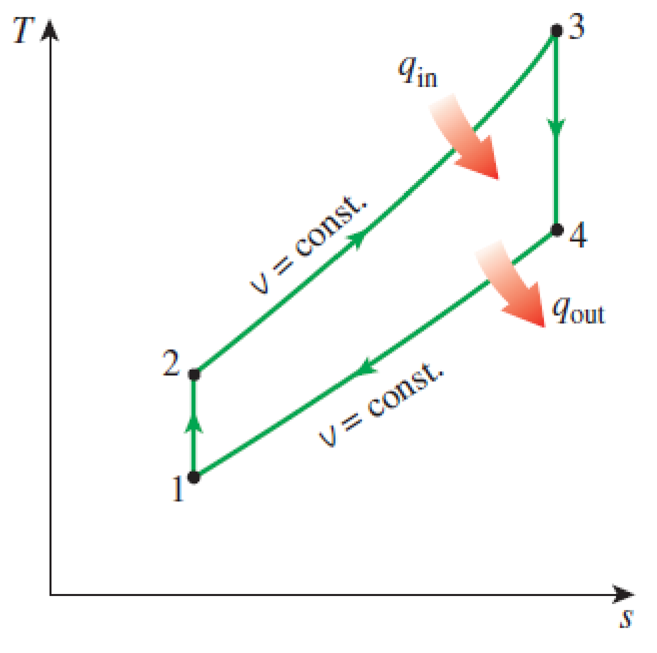
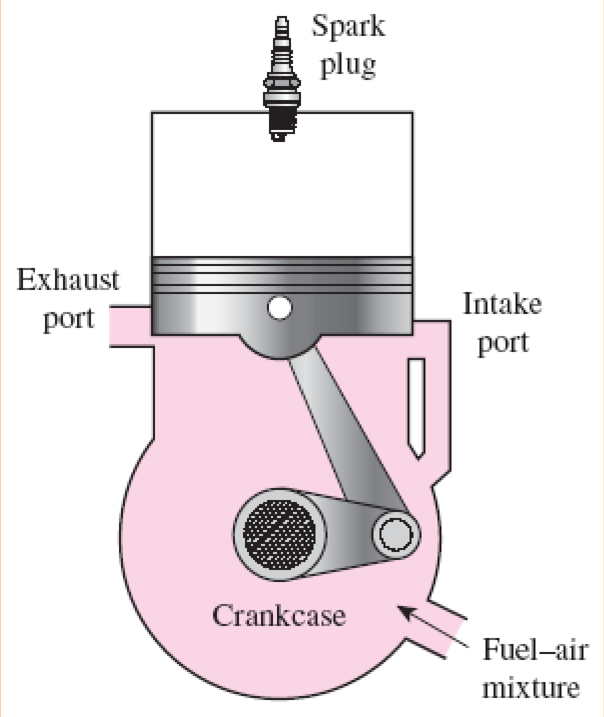


FIGURE 9–15
T-s diagram of the ideal Otto cycle.

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



Schematic of a two-stroke reciprocating engine.

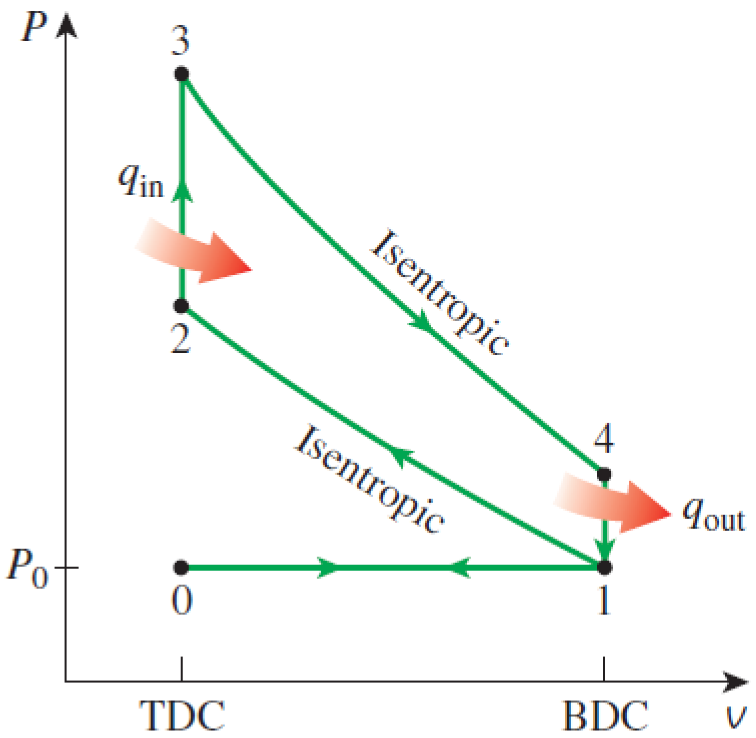


FIGURE 9–16
P-v diagram of the ideal Otto cycle that includes intake and exhaust strokes.

$$W_{out,0-1} = P_0(v_1 - v_0)$$

$$W_{in,1-0} = P_0(v_1 - v_0)$$

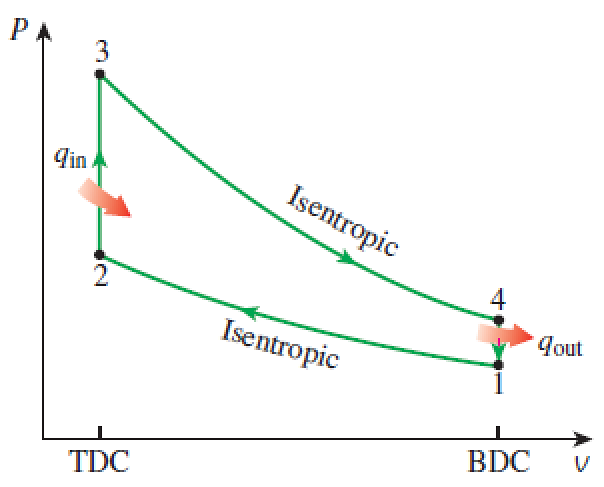
Air enters the cylinder through the open intake valve at atmospheric pressure P_0 during process 0-1 as the piston moves from TDC to BDC.

The intake valve is closed at state 1 and air is compressed isentropically to state 2. Heat is transferred at constant volume (process 2-3); it is expanded isentropically to state 4; and heat is rejected at constant volume (process 4-1).

Air is expelled through the open exhaust valve (process 1-0).

Work interactions during intake and exhaust cancel each other, and thus inclusion of the intake and exhaust processes has no effect on the net work output from the cycle.

However, when calculating power output from the cycle during an ideal Otto cycle analysis, we must consider the fact that the ideal Otto cycle has four strokes just like actual four-stroke spark-ignition engine.



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

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

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

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

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

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

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

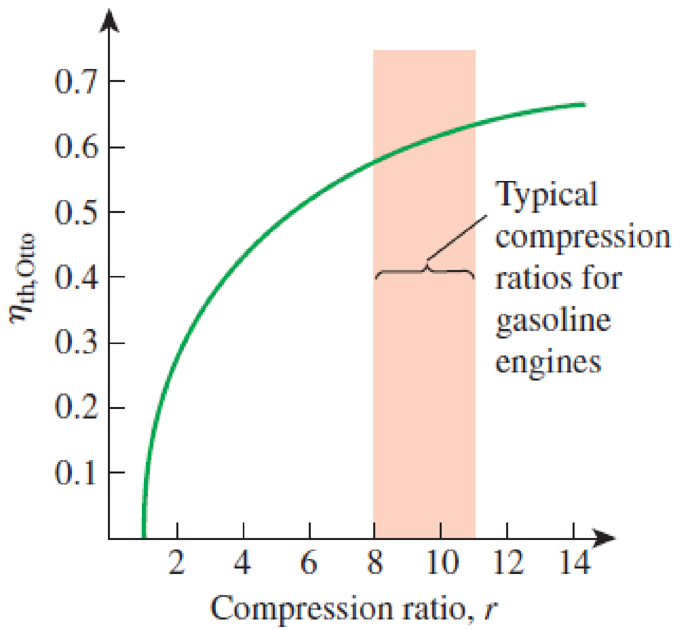


FIGURE 9–17

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

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

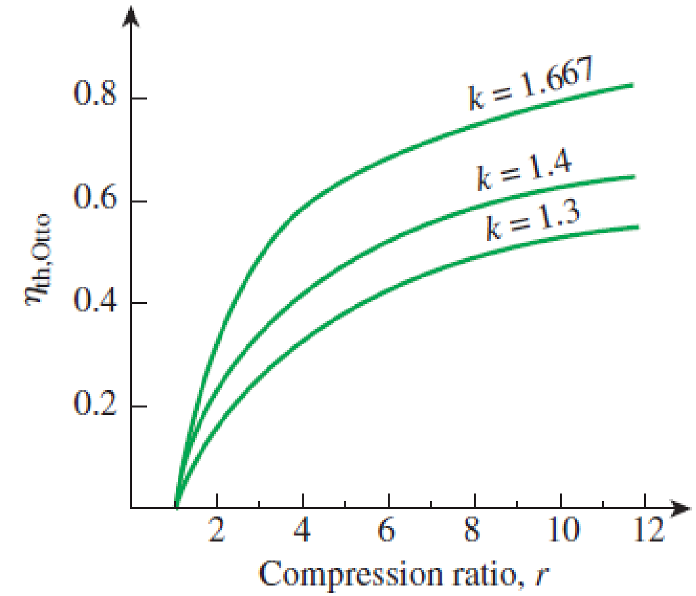


FIGURE 9–18

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

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

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

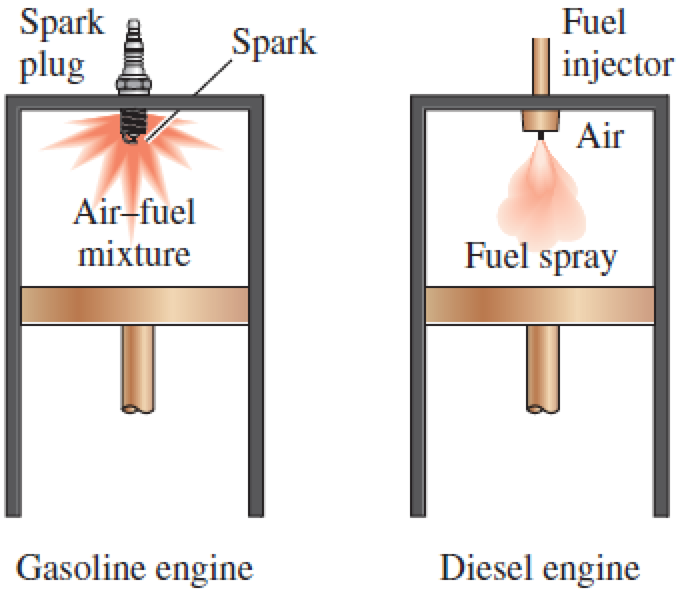


FIGURE 9–20
In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

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

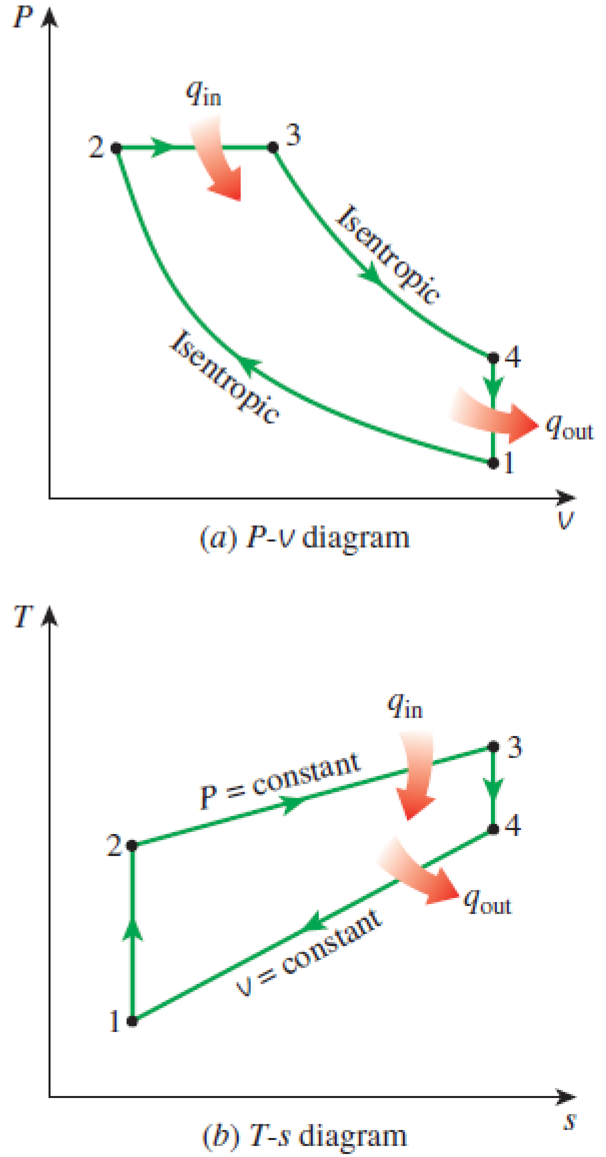


FIGURE 9–21
T-s and *P-v* diagrams for the ideal Diesel cycle.

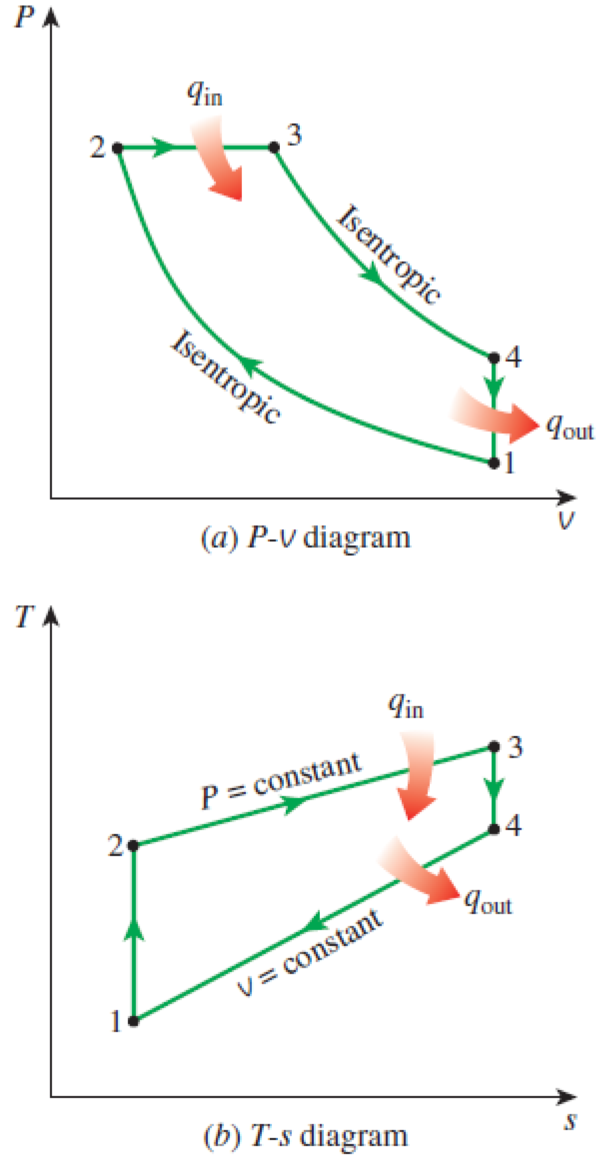


FIGURE 9–21
T-s and P-v diagrams for the ideal Diesel cycle.

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

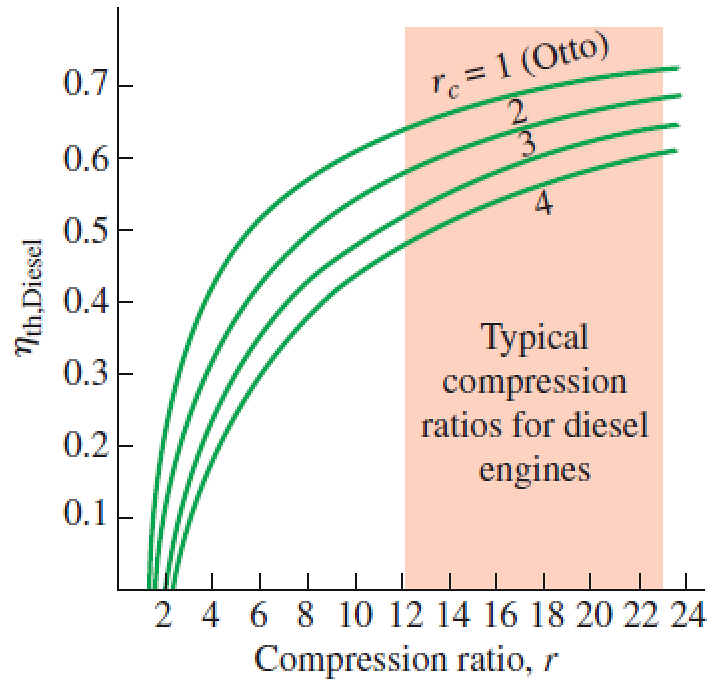
$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

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

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

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

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



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

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

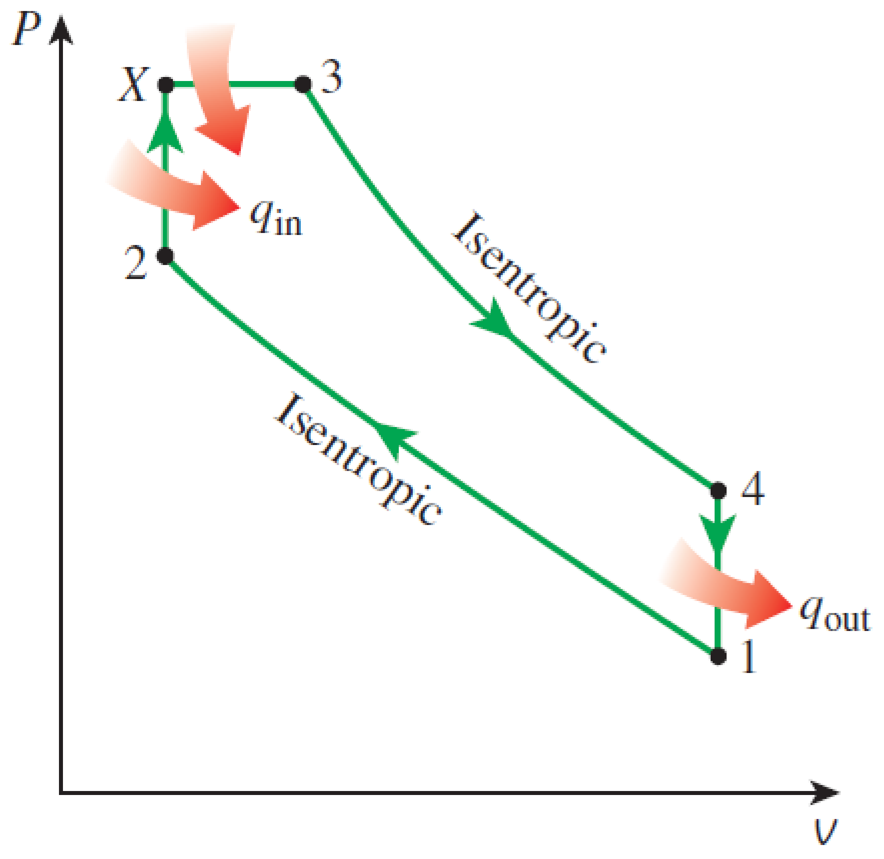


FIGURE 9–23

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

In modern high-speed compression ignition engines, fuel is injected into the combustion chamber much sooner compared to 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 better be modeled as the combination of constant-volume and constant-pressure processes.

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

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.

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

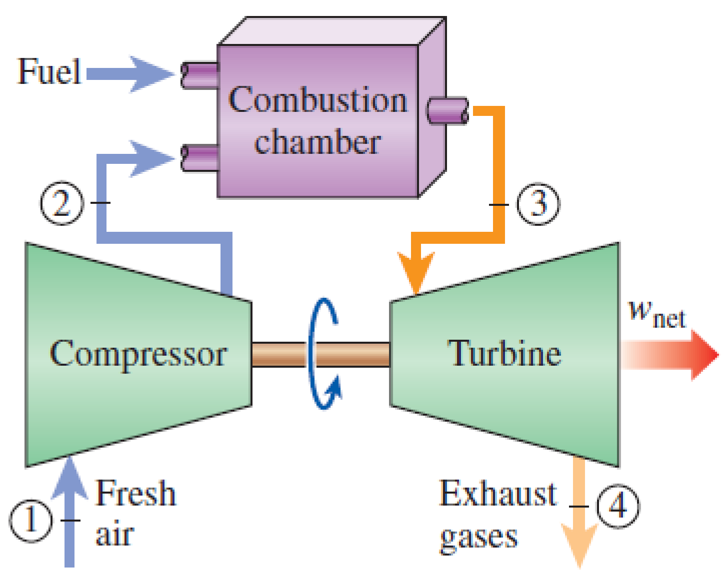


FIGURE 9–29
An open-cycle gas-turbine engine.

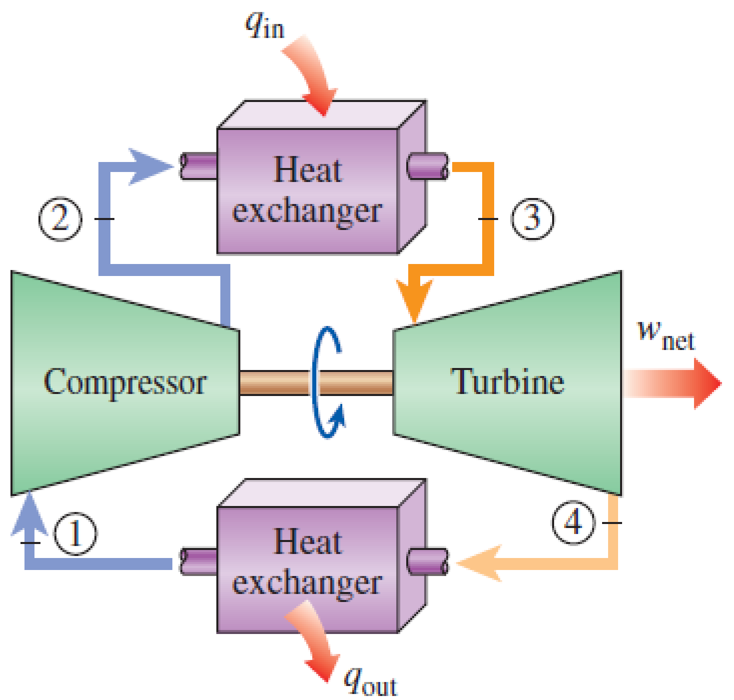


FIGURE 9–30
A closed-cycle gas-turbine engine.

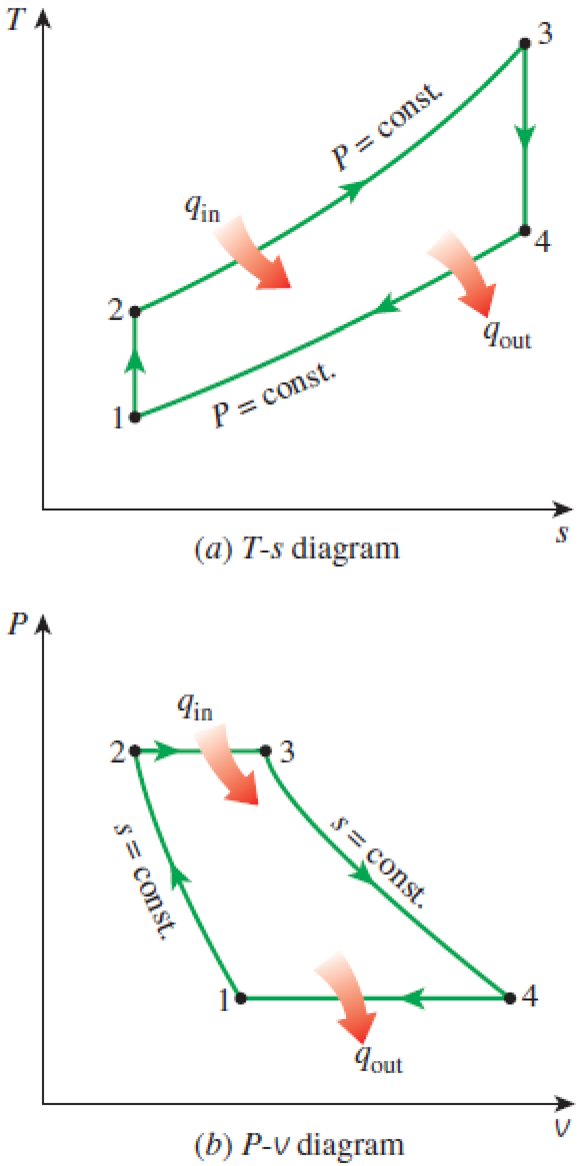


FIGURE 9–31
T-s and P-v diagrams for the ideal Brayton cycle.

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

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

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

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

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

$$r_p = \frac{P_2}{P_1}$$ Pressure ratio

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

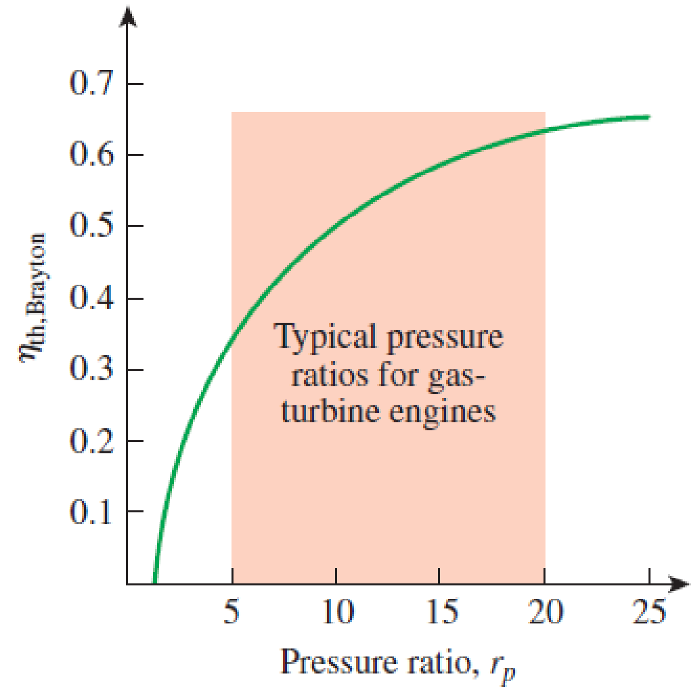


FIGURE 9–32
Thermal efficiency of the ideal Brayton cycle as a function of the pressure ratio.

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

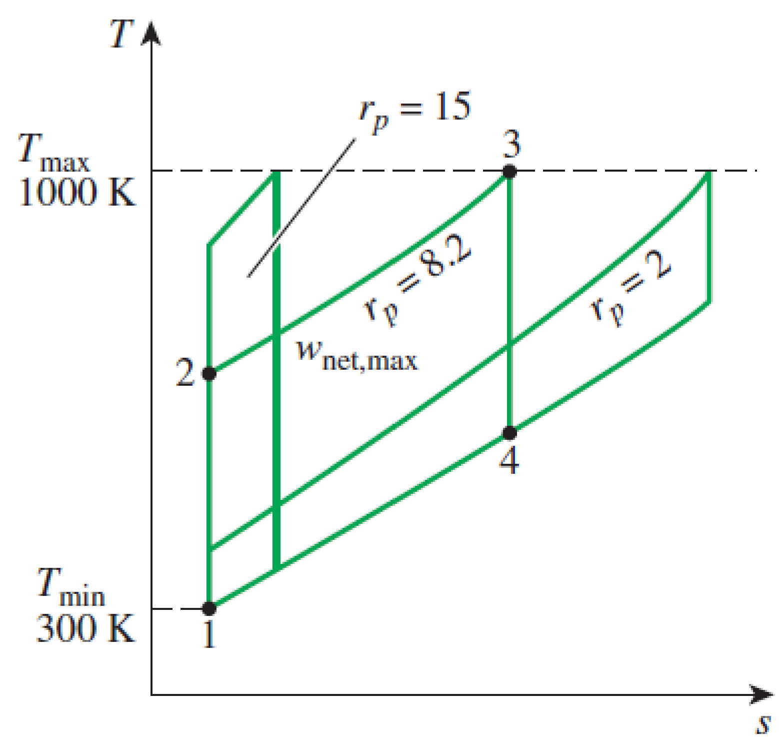


FIGURE 9–33
For fixed values of T_{min} and T_{max}, the net work of the Brayton cycle first increases with the pressure ratio, then reaches a maximum at $r_p = (T_{max}/T_{min})^{k/[2(k-1)]}$, and finally decreases.

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.

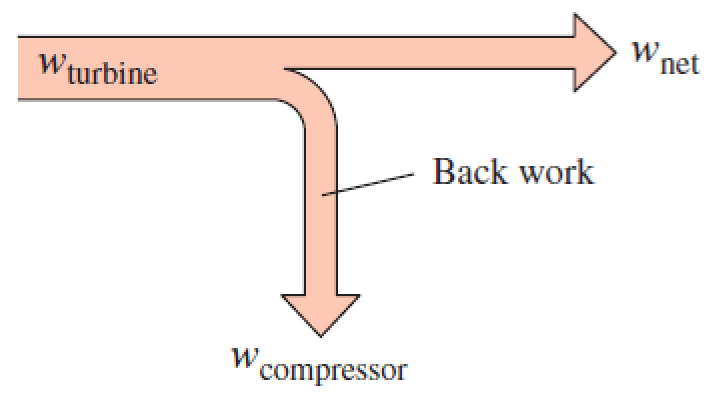


FIGURE 9–34
The fraction of the turbine work used to drive the compressor is called the back work ratio.

Development of Gas Turbines

1. Increasing the turbine inlet (or firing) temperatures
2. Increasing the efficiencies of turbomachinery components (turbines, compressors):
3. Adding modifications to the basic cycle (intercooling, regeneration or recuperation, and reheating).

Deviation of Actual Gas-Turbine Cycles from Idealized Ones

Reasons: Irreversibilities in turbine and compressors, pressure drops, heat losses

Isentropic efficiencies of the compressor and turbine

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

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

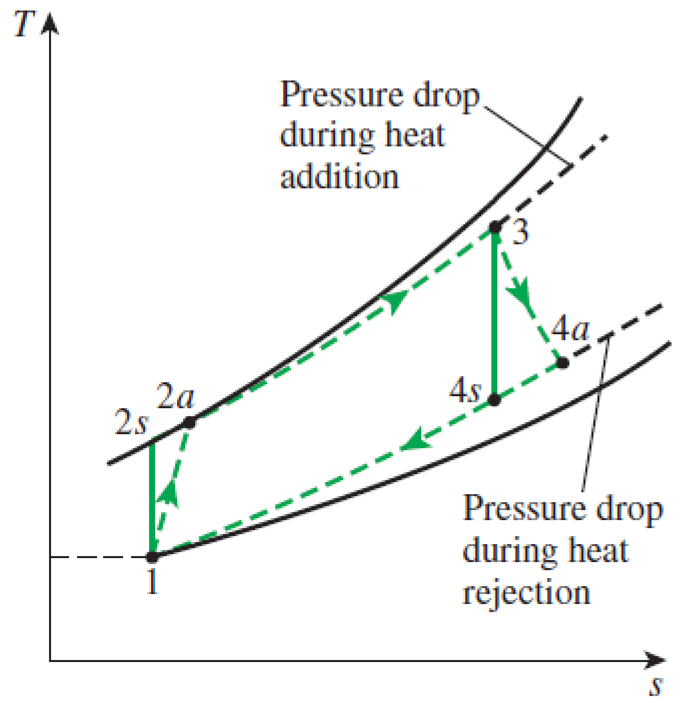


FIGURE 9–36
The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.