Chapter 9 GAS POWER 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 (Fig. 9–9).
4. The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.

9–4 . AN OVERVIEW OF RECIPROCATING ENGINES

The basic components of a reciprocating engine are shown in Fig. 9–10. 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.**

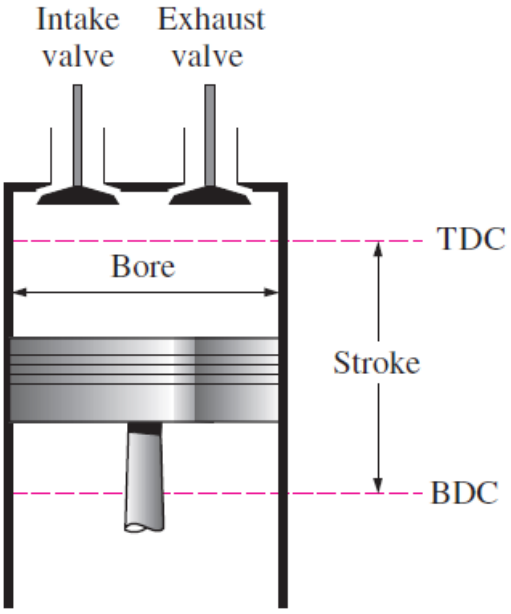


FIGURE 9–10

Nomenclature for reciprocating engines.

9–5 ▪ OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

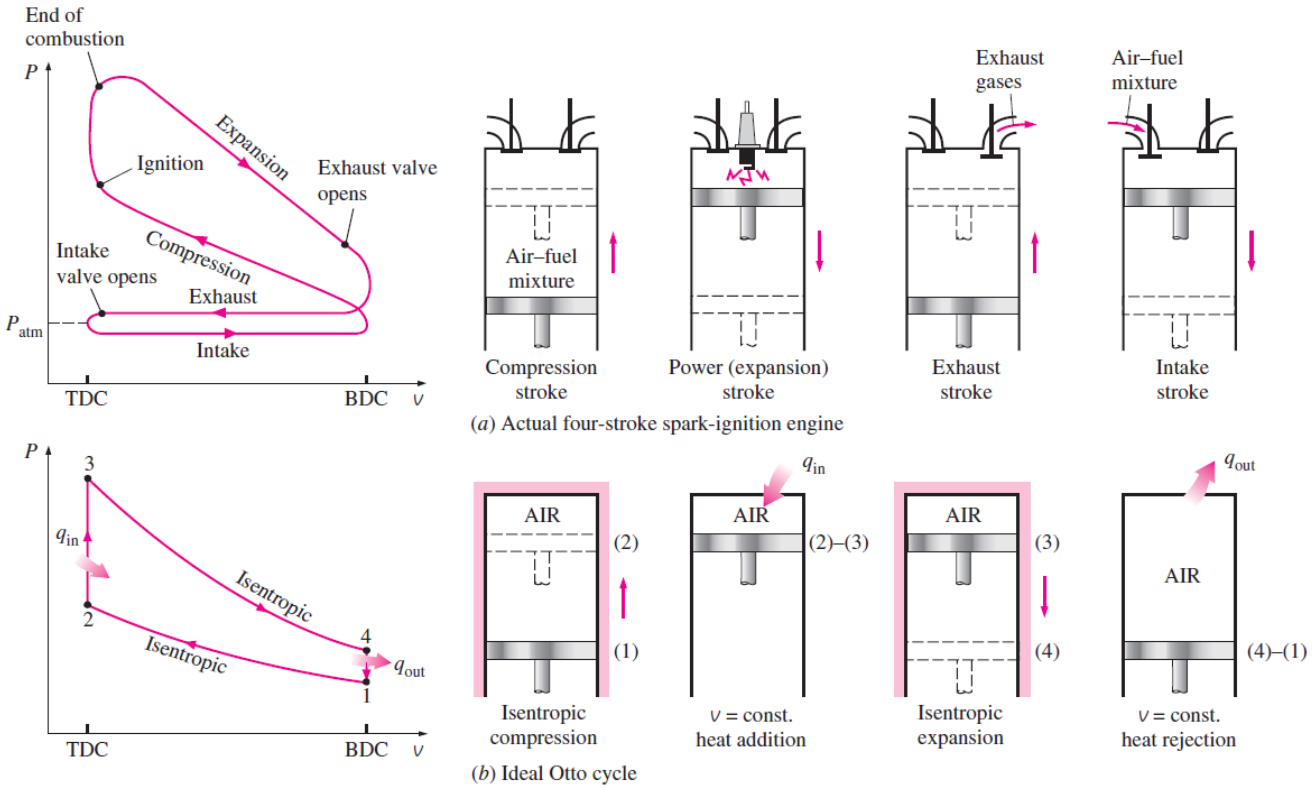


FIGURE 9–13

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

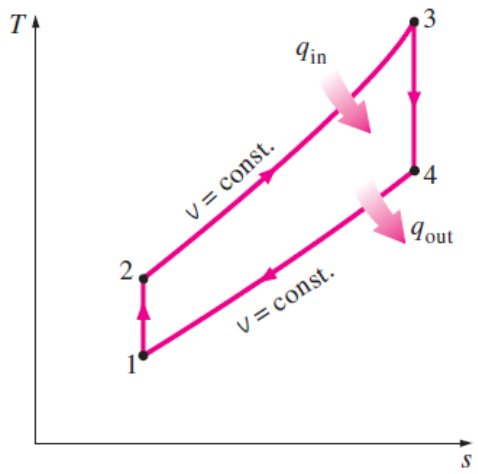


FIGURE 9–16

T-s diagram of the ideal Otto cycle.

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}} \tag{9–8}$$

$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2} \tag{9–9}$$

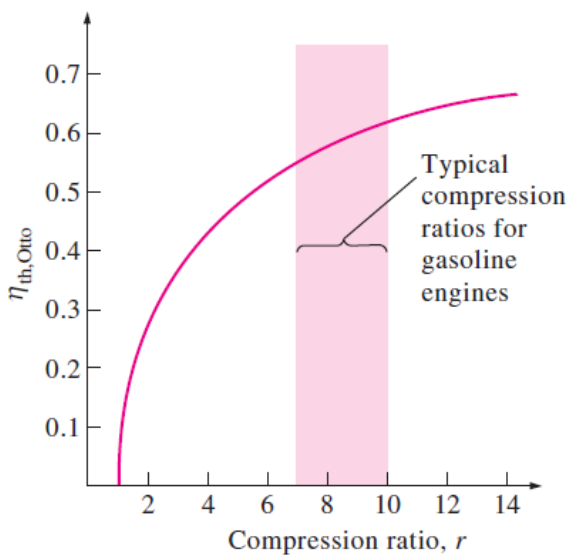


FIGURE 9–17

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

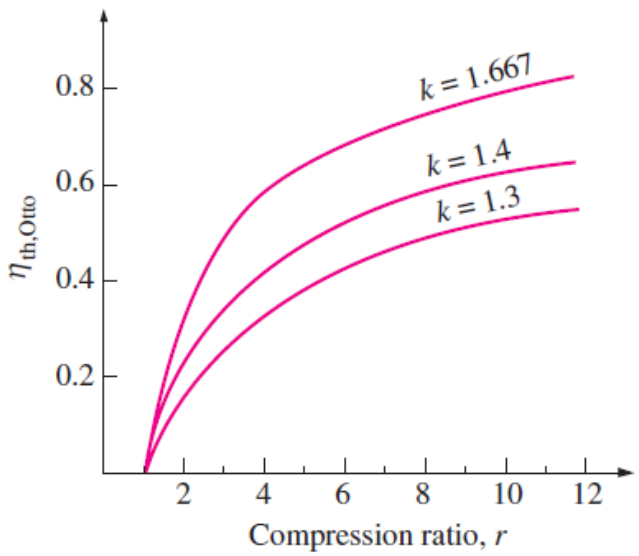


FIGURE 9–18

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

9–6 ■ DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

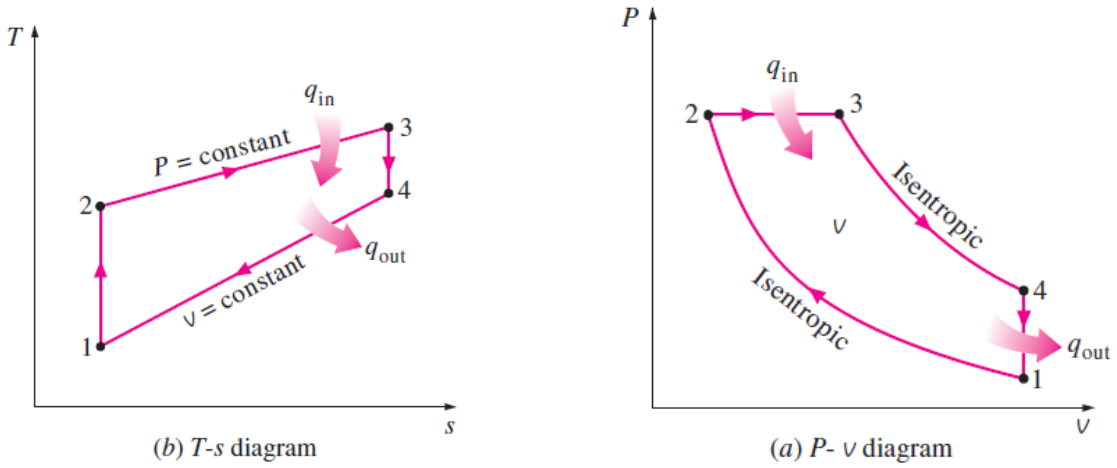


FIGURE 9–21

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

$$\eta_{\text{th,Diesel}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right] \tag{9–12}$$

$$\eta_{\text{th,Otto}} > \eta_{\text{th,Diesel}} \tag{9–13}$$

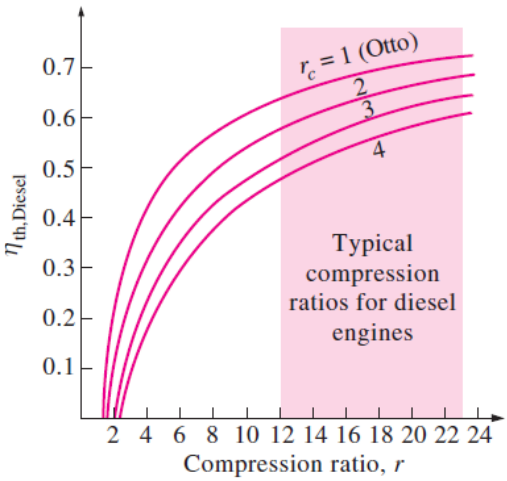


FIGURE 9–22

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

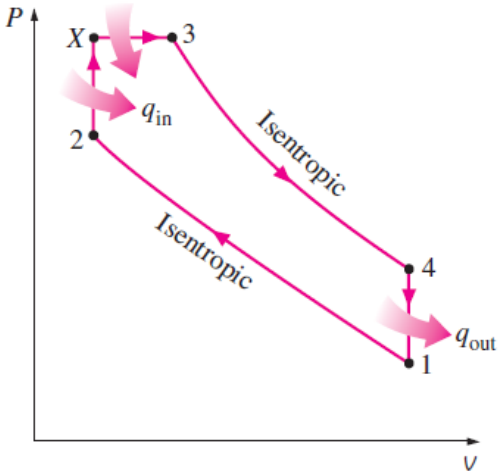


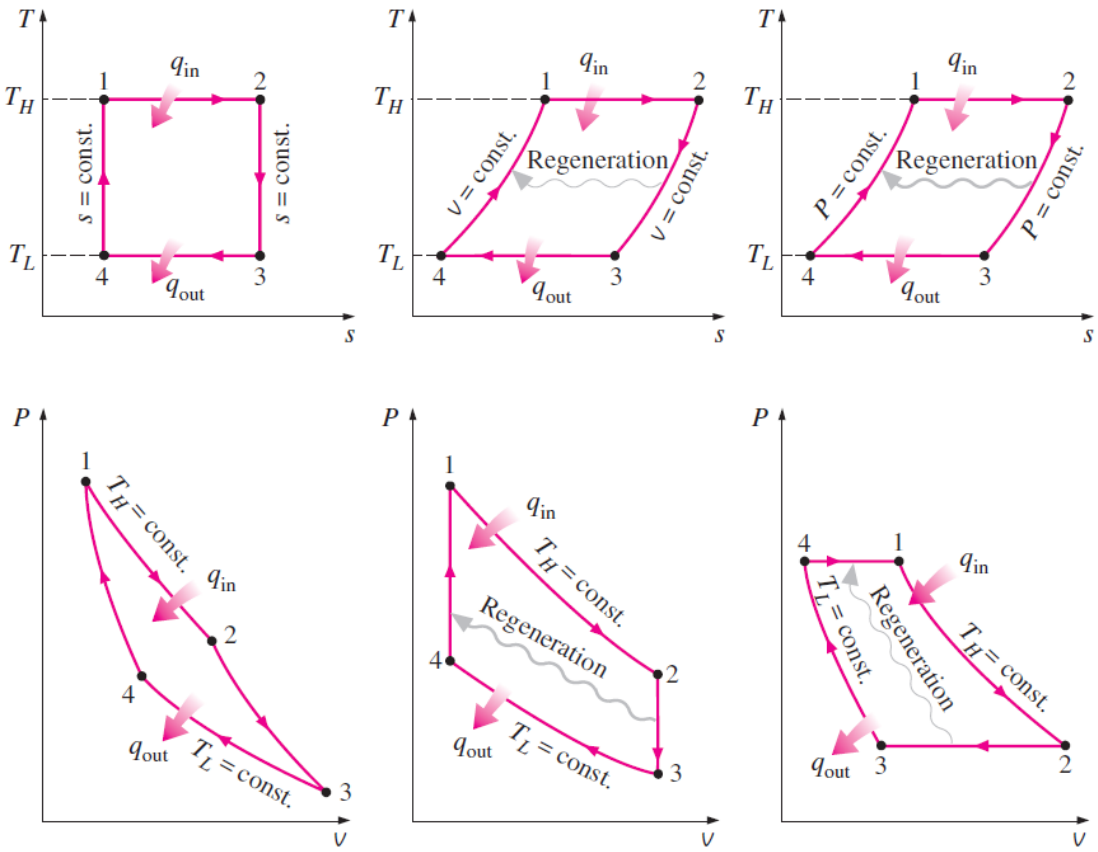
FIGURE 9–23

P-v diagram of an ideal dual cycle.

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. 9–23. 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.

9–7 .STIRLING AND ERICSSON CYCLES

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*.



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

Initially, the left chamber houses the entire working fluid (a gas), which is at a high temperature and pressure. During process 1-2, heat is transferred to the gas at T_H from a source at T_H. As the gas expands isothermally, the left piston moves outward, doing work, and the gas pressure drops. During process 2-3, both pistons are moved to the right at the same rate (to keep the volume constant) until the entire gas is forced into the right chamber. As the gas passes through the regenerator, heat is transferred to the regenerator and the gas temperature drops from T_H to T_L. For this heat transfer process to be reversible, the temperature difference between the gas and the regenerator should not exceed a differential amount dT at any point. Thus, the temperature of the regenerator will be T_H at

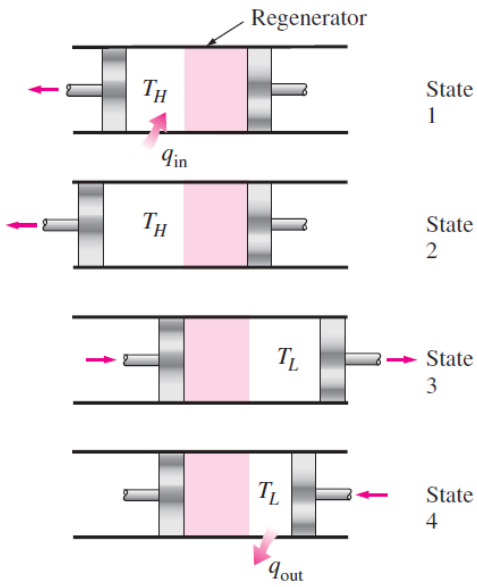


FIGURE 9–27
The execution of the Stirling cycle.

the left end and T_L at the right end of the regenerator when state 3 is reached. During process 3-4, the right piston is moved inward, compressing the gas. Heat is transferred from the gas to a sink at temperature T_L so that the gas temperature remains constant at T_L while the pressure rises. Finally, during process 4-1, both pistons are moved to the left at the same rate (to keep the volume constant), forcing the entire gas into the left chamber. The gas temperature rises from T_L to T_H as it passes through the regenerator and picks up the thermal energy stored there during process 2-3. This completes the cycle.

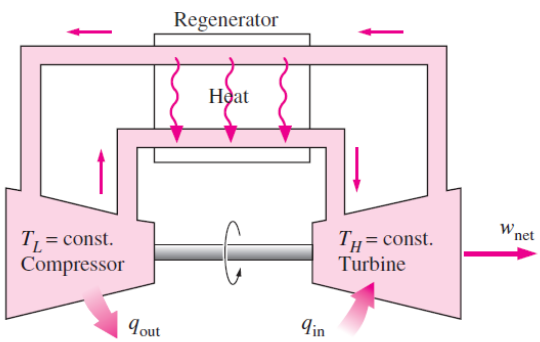


FIGURE 9–28
A steady-flow Ericsson engine.

Both the Stirling and Ericsson cycles are totally reversible.

$$\eta_{th,Stirling} = \eta_{th,Ericsson} = \eta_{th,Carnot} = 1 - \frac{T_L}{T_H} \qquad (9\text{–}14)$$

Both the Stirling and the Ericsson engines are *external combustion* engines.

External combustion offers several advantages.

First, a variety of fuels can be used as a source of thermal energy.

Second, there is more time for combustion, and thus the combustion process is more complete, which means less air pollution and more energy extraction from the fuel.

Third, these engines operate on closed cycles, and thus a working fluid that has the most desirable characteristics (stable, chemically inert, high thermal conductivity) can be

utilized as the working fluid. Hydrogen and helium are two gases commonly employed in these engines.

Regeneration can increase efficiency.

9–8 ▪BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

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

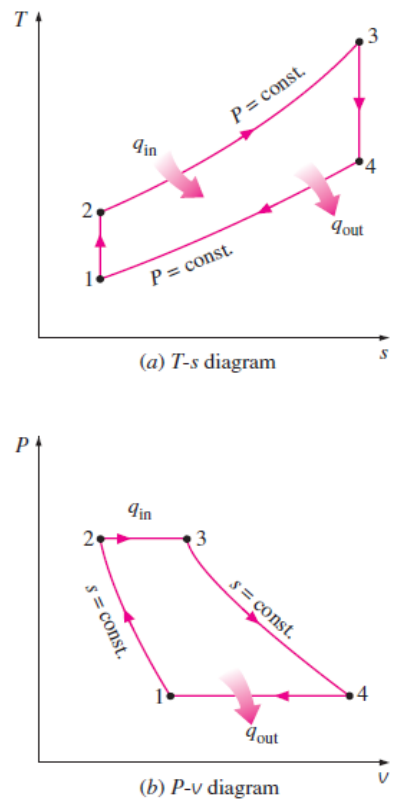


FIGURE 9–31

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

$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} \qquad (9\text{–}17)$$

$$r_p = \frac{P_2}{P_1} \qquad (9\text{–}18)$$

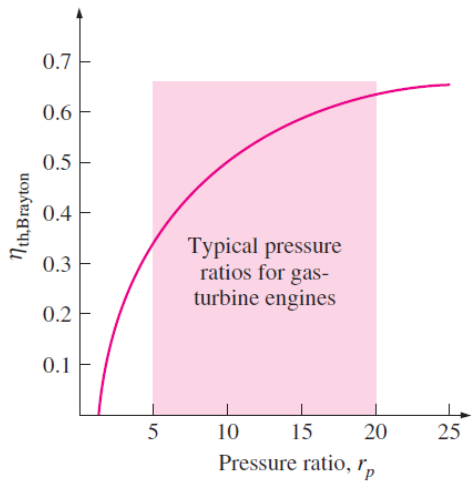


FIGURE 9–32

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

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.

The air in gas turbines performs two important functions: 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 safe limits.

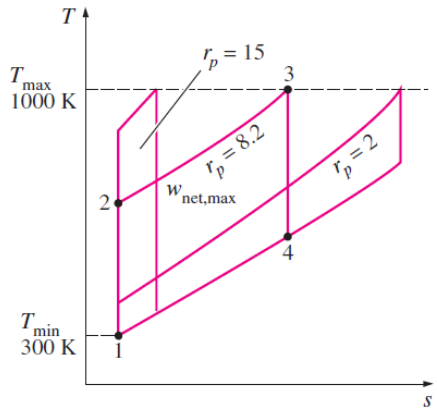


FIGURE 9–33

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 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,** is very high (Fig. 9–34). Usually more than one-half of the turbine work output is used to drive the compressor.

The situation is even worse when the isentropic efficiencies of the compressor and the turbine are low. This is quite in contrast to steam power plants, where the back work ratio is only a few percent. This is not surprising, however, since a liquid is compressed in steam power plants instead of a gas, and the steady-flow work is proportional to the specific volume of the working fluid.

Development of Gas Turbines

The efforts to improve the cycle efficiency concentrated in three areas:

1. **Increasing the turbine inlet (or firing) temperatures**.
2. **Increasing the efficiencies of turbomachinery components.**
3. **Adding modifications to the basic cycle**.

Deviation of Actual Gas-Turbine Cycles from Idealized Ones

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

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

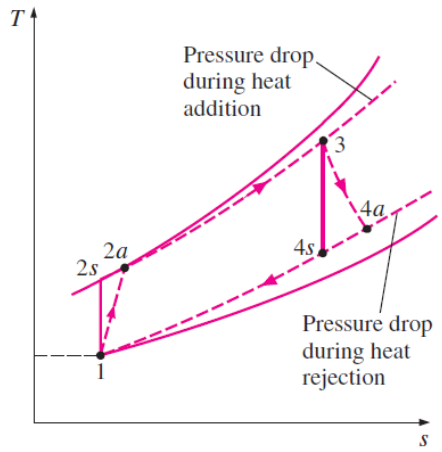


FIGURE 9–36

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

9–9 ▪THE BRAYTON CYCLE WITH REGENERATION

In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the high-pressure air leaving the compressor can be heated by transferring heat to it from the hot exhaust gases in a counter-flow heat exchanger, which is also known as a ***regenerator*** or a ***recuperator***.

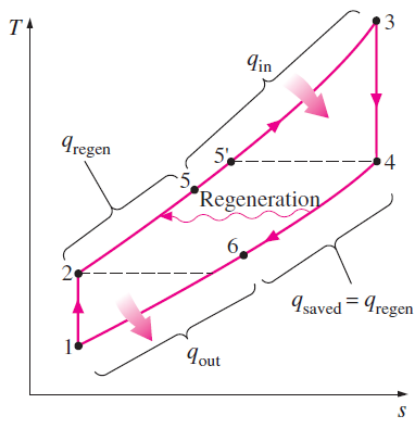


FIGURE 9–39

T-s diagram of a Brayton cycle with regeneration.

$$q_{regen,act} = h_5 - h_2 \qquad \textbf{(9–21)}$$

$$q_{\text{regen,max}} = h_{5'} - h_2 = h_4 - h_2 \qquad \textbf{(9–22)}$$

The extent to which a regenerator approaches an ideal regenerator is called the **effectiveness** ϵ and is defined as

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

When the cold-air-standard assumptions are utilized, it reduces to

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

Under the cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle with regeneration is

$$\eta_{\text{th,regen}} = 1 - \left(\frac{T_1}{T_3}\right)(r_p)^{(k-1)/k} \qquad \textbf{(9–25)}$$

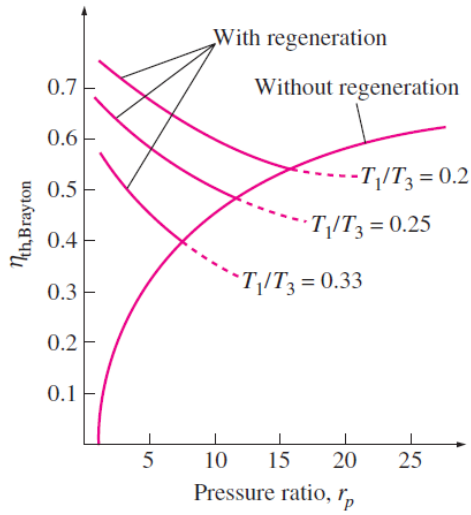


FIGURE 9–40

Thermal efficiency of the ideal Brayton cycle with and without regeneration.

9–10 ▪ THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

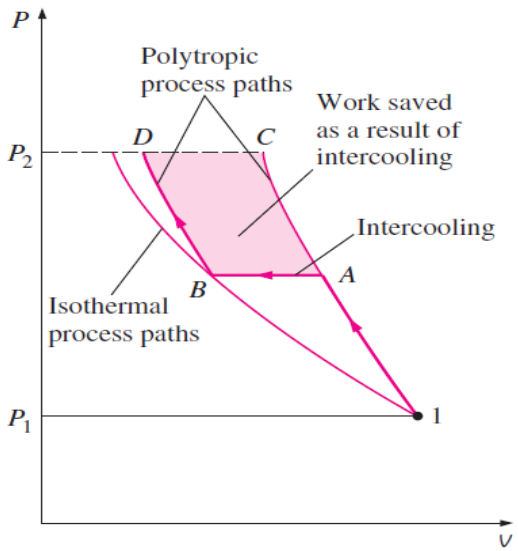


FIGURE 9–42

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

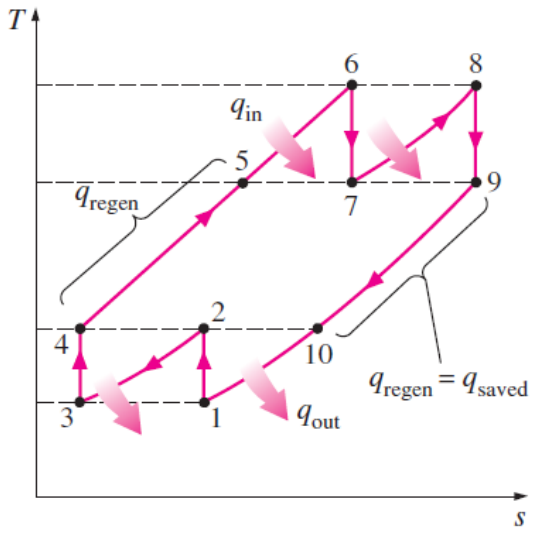


FIGURE 9–44

T-s diagram of an ideal gas-turbine cycle with intercooling, reheating, and regeneration.

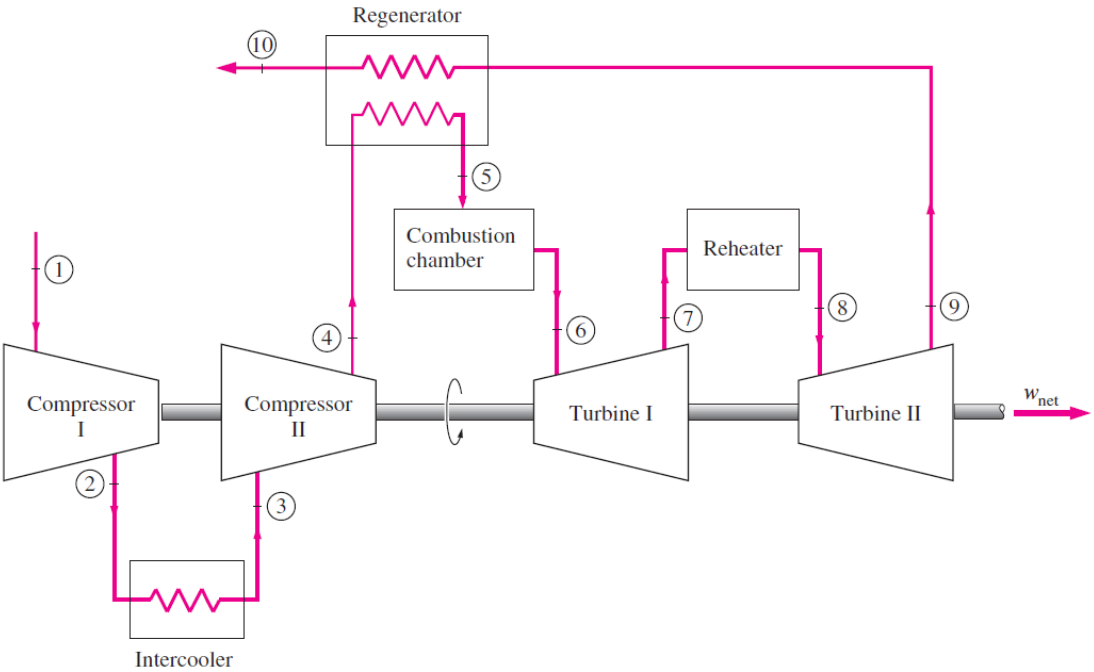


FIGURE 9–43

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

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9} \tag{9–26}$$

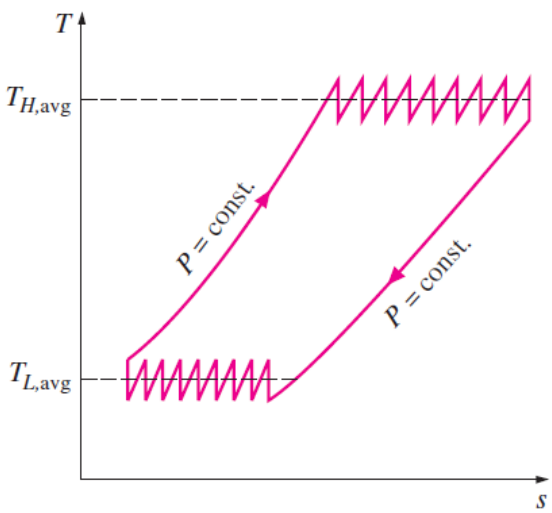


FIGURE 9–45

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

9–11 ▪IDEAL JET-PROPULSION CYCLES

Gas-turbine engines are widely used to power aircraft because they are light and compact and have a high power-to-weight ratio. Aircraft gas turbines operate on an open cycle called a **jet-propulsion cycle.** The ideal jet-propulsion cycle differs from the simple ideal Brayton cycle in that the gases are not expanded to the ambient pressure in the turbine. Instead, they are expanded to a pressure such that the power produced by the turbine is just sufficient to drive the compressor and the auxiliary equipment, such as a small generator and hydraulic pumps. That is, the net work output of a jet-propulsion cycle is zero. The gases that exit the turbine at a relatively high pressure are subsequently accelerated in a nozzle to provide the thrust to propel the aircraft (Fig. 9–47). Also, aircraft gas turbines operate at higher pressure ratios (typically between 10 and 25), and the fluid passes through a diffuser first, where it is decelerated and its pressure is increased before it enters the compressor.

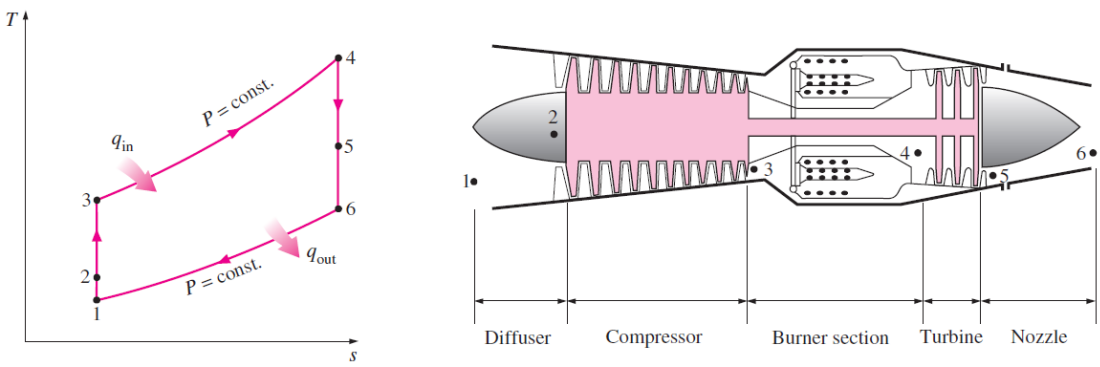


FIGURE 9–48
Basic components of a turbojet engine and the *T-s* diagram for the ideal turbojet cycle.

The pressures at the inlet and the exit of a turbojet engine are identical (the ambient pressure); thus, the net thrust developed by the engine is

$$F = (\dot{m}V)_{\text{exit}} - (\dot{m}V)_{\text{inlet}} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}}) \qquad \text{(N)} \qquad \textbf{(9–27)}$$

The power developed from the thrust of the engine is called the **propulsive power** $\dot{W}_P$

$$\dot{W}_P = FV_{\text{aircraft}} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}} \qquad (\text{kW}) \qquad \textbf{(9–28)}$$

The desired output in a turbojet engine is the *power produced* to propel the aircraft $\dot{W}_P$, and the required input is the *heating value of the fuel* $\dot{Q}_{\text{in}}$. The ratio of these two quantities is called the **propulsive efficiency** and is given by

$$\eta_P = \frac{\text{Propulsive power}}{\text{Energy input rate}} = \frac{\dot{W}_P}{\dot{Q}_{\text{in}}} \qquad \textbf{(9–29)}$$

9–12 ▪SECOND-LAW ANALYSIS OF GAS POWER CYCLES

$$X_{\text{dest}} = T_0 S_{\text{gen}} = T_0(\Delta S_{\text{sys}} - S_{\text{in}} + S_{\text{out}})$$
$$= T_0\left[(S_2 - S_1)_{\text{sys}} - \frac{Q_{\text{in}}}{T_{b,\text{in}}} + \frac{Q_{\text{out}}}{T_{b,\text{out}}}\right] \qquad (\text{kJ}) \qquad \textbf{(9–30)}$$

$$\dot{X}_{\text{dest}} = T_0\dot{S}_{\text{gen}} = T_0(\dot{S}_{\text{out}} - \dot{S}_{\text{in}}) = T_0\left(\sum_{\text{out}} \dot{m}s - \sum_{\text{in}} \dot{m}s - \frac{\dot{Q}_{\text{in}}}{T_{b,\text{in}}} + \frac{\dot{Q}_{\text{out}}}{T_{b,\text{out}}}\right) \qquad (\text{kW})$$

(9–31)

$$X_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{b,\text{in}}} + \frac{q_{\text{out}}}{T_{b,\text{out}}}\right) \qquad (\text{kJ/kg}) \qquad \textbf{(9–32)}$$

$$x_{\text{dest}} = T_0\left(\sum \frac{q_{\text{out}}}{T_{b,\text{out}}} - \sum \frac{q_{\text{in}}}{T_{b,\text{in}}}\right) \qquad (\text{kJ/kg}) \qquad \textbf{(9–33)}$$

$$x_{\text{dest}} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right) \qquad (\text{kJ/kg}) \qquad \textbf{(9–34)}$$

The exergies of a closed system ϕ and a fluid stream ψ at any state can be determined from

$$\phi = (u - u_0) - T_0(s - s_0) + P_0(v - v_0) + \frac{V^2}{2} + gz \qquad (\text{kJ/kg}) \quad \textbf{(9–35)}$$

$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \qquad (\text{kJ/kg}) \quad \textbf{(9–36)}$$

Where subscript “0” denotes the state of the surroundings.