CHAPTER
9

Gas Power Cycles

1. Basic Consideration
2. The Carnot Cycle and Its Value in Engineering
3. Air-Standard Assumptions
4. An Overview of Reciprocating Engines
5. Otto Cycle
6. Diesel Cycle
7. Stirling and Ericsson Cycles
8. Brayton Cycle
9. Brayton Cycle with Regeneration
10. Brayton Cycle with Intercooling, Reheating
11. Ideal Jet-Propulsion Cycles
12. Second-Law Analysis of Gas Power Cycles

2022 Fall semester

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.
- Analyze both closed and open gas power cycles.
- Solve problems based on the Otto, Diesel, Stirling, and Ericsson cycles.
- Solve problems based on the Brayton cycle; the Brayton cycle with regeneration; and the Brayton cycle with intercooling, reheating, and regeneration.
- Analyze jet-propulsion cycles.
- Perform second-law analysis of gas power cycles.

9-1. BASIC CONSIDERATIONS

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 is called an**.**

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.

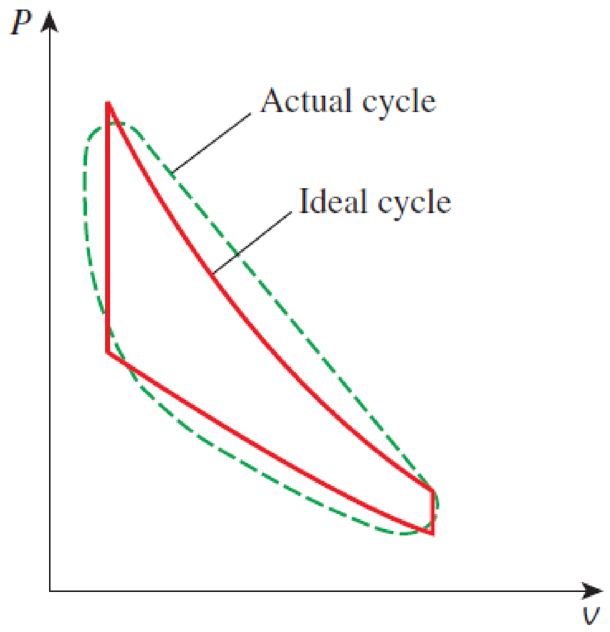


FIGURE 9–2
The analysis of many complex processes can be reduced to a manageable level by utilizing some idealizations.

Thermal efficiency of heat engines

$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}} \qquad (9\text{-}1)$$

9-1. BASIC CONSIDERATIONS

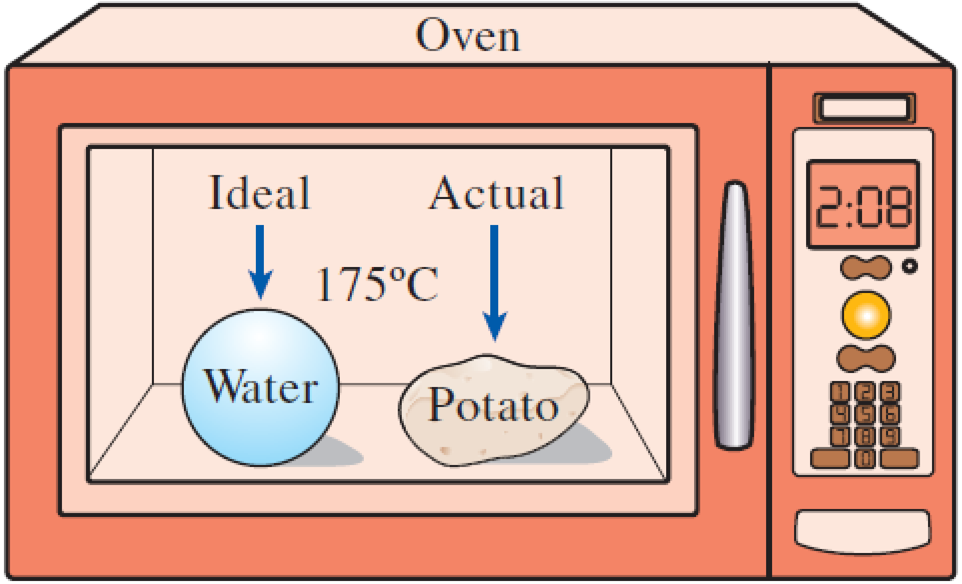


FIGURE 9–1

Modeling is a powerful engineering tool that provides great insight and simplicity at the expense of some loss in accuracy.

9-1. BASIC CONSIDERATIONS

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

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.

FIGURE 9–3
An automotive engine with the combustion chamber exposed.

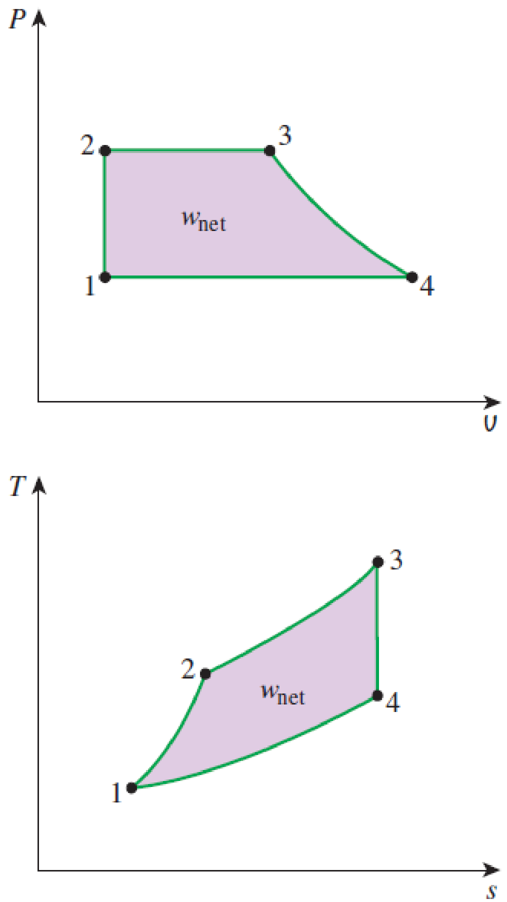


FIGURE 9–4
On both *P*-υ and *T*-*s* diagrams, the area enclosed by the process curve represents the net work of the cycle.

The idealizations and simplifications in the analysis of power cycles:

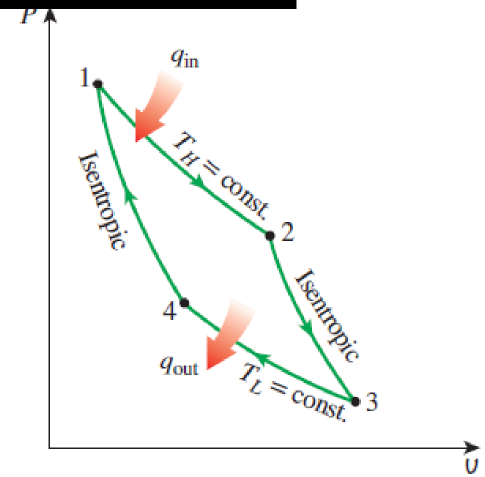
1. The cycle does not involve any *friction.* Therefore, the working fluid does not experience any pressure drop as it flows in pipes or devices such as heat exchangers.

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.

On a *T*-*s* diagram, the ratio of the area enclosed by the cyclic curve to the area under the heat-addition process curve represents the thermal efficiency of the cycle. Any modification that increases the ratio of these two areas will also increase the thermal efficiency of the cycle.

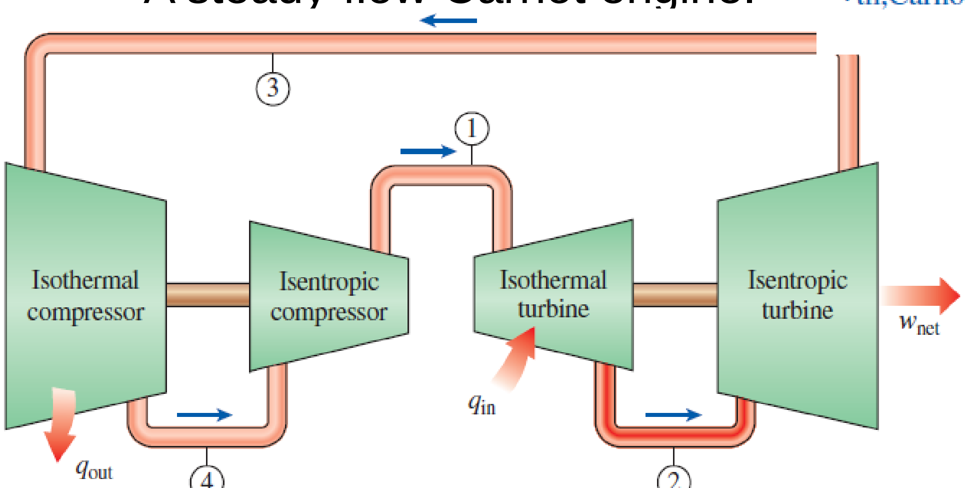
9-2. THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING



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

A steady-flow Carnot engine.

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



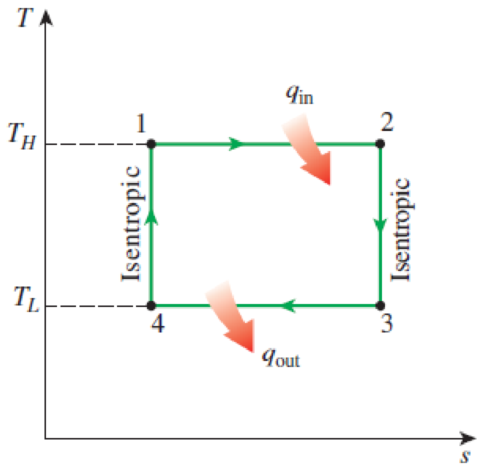


FIGURE 9–5
P-υ and *T*-*s* diagrams of a Carnot cycle.

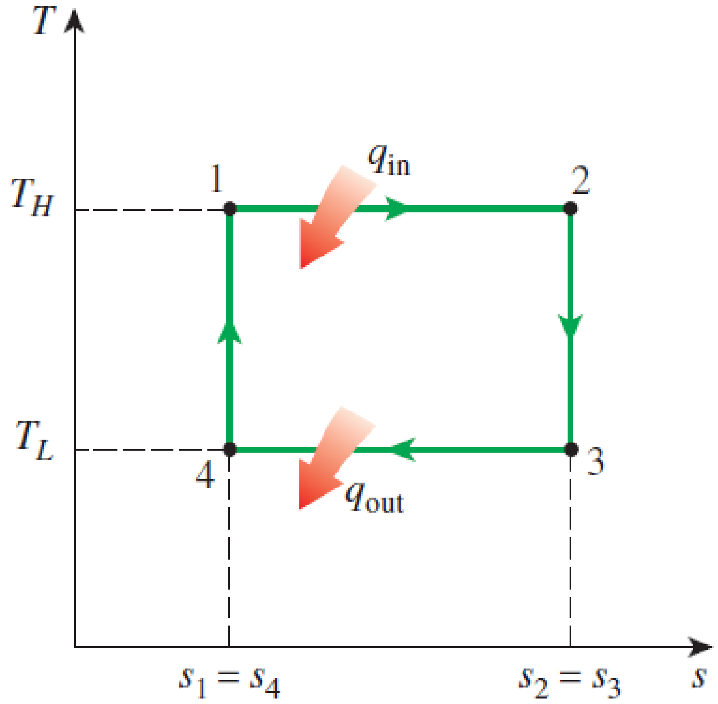
9-2. THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING

$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} \qquad (9\text{-}2)$$

For both ideal and actual cycles:

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.

9-2. THE CARNOT CYCLE AND ITS VALUE IN ENGINEERING



Derivation of the Efficiency of the Carnot Cycle

$$q_{in} = T_H(s_2 - s_1)$$

$$q_{out} = T_L(s_3 - s_4) = T_L(s_2 - s_1)$$

$s_2 = s_3$ and $s_4 = s_1$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_L(s_2 - s_1)}{T_H(s_2 - s_1)} = 1 - \frac{T_L}{T_H}$$

9-3. AIR-STANDARD ASSUMPTIONS

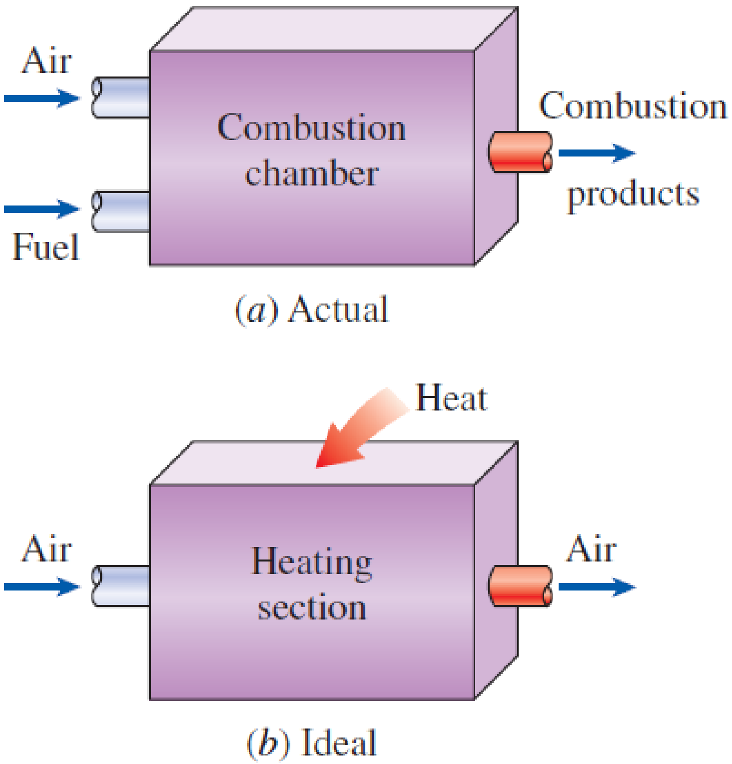


FIGURE 9–8
The combustion process is replaced by a heat-addition process in ideal cycles.

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

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).

9-4. OVERVIEW OF RECIPROCATING ENGINES

Compression ratio

$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}} \qquad (9\text{-}3)$$

Spark-ignition (SI) engines

Compression-ignition (CI) engines

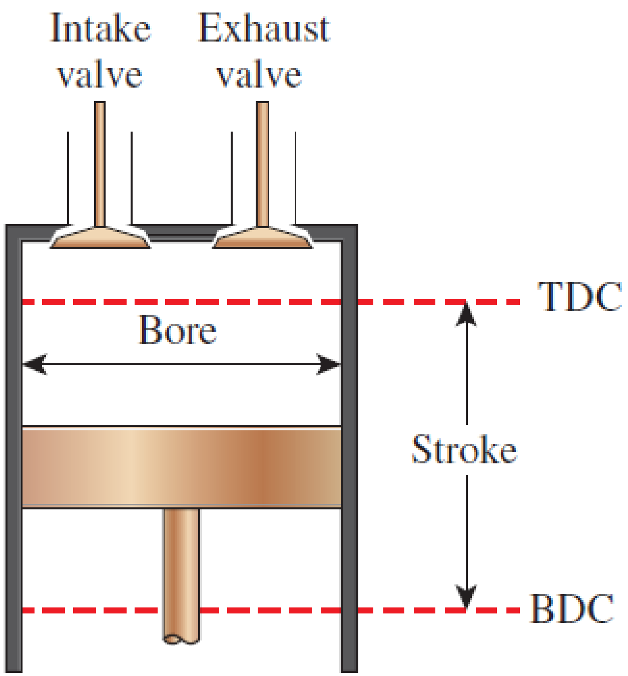


FIGURE 9–9

Nomenclature for reciprocating engines.

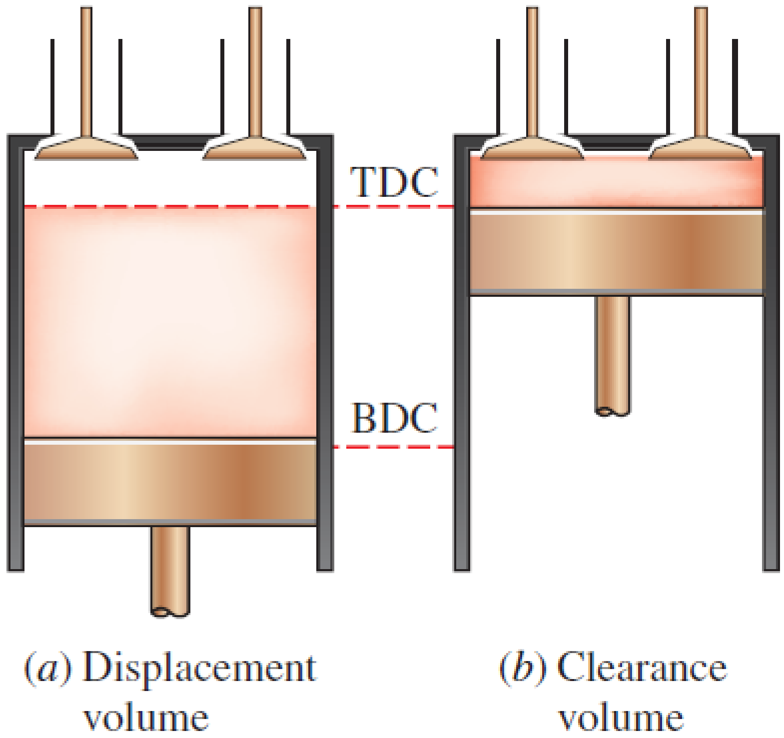


FIGURE 9–10

Displacement and clearance volumes of a reciprocating engine.

9-4. OVERVIEW OF RECIPROCATING ENGINES

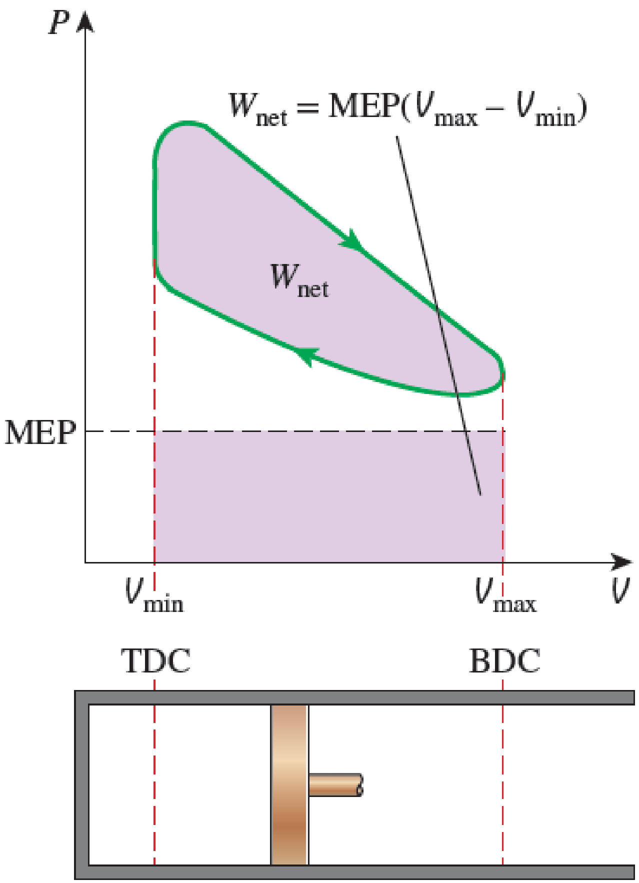


FIGURE 9–12
The net work output of a cycle is equivalent to the product of the mean effective pressure and the displacement volume.

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

Mean effective pressure

$$MEP = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}} \quad (kPa) \qquad (9\text{-}4)$$

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.

9-5. OTTO CYCLE: IDEAL CYCLE FOR SI ENGINES

2ea Isentropic process
2ea Isometric process

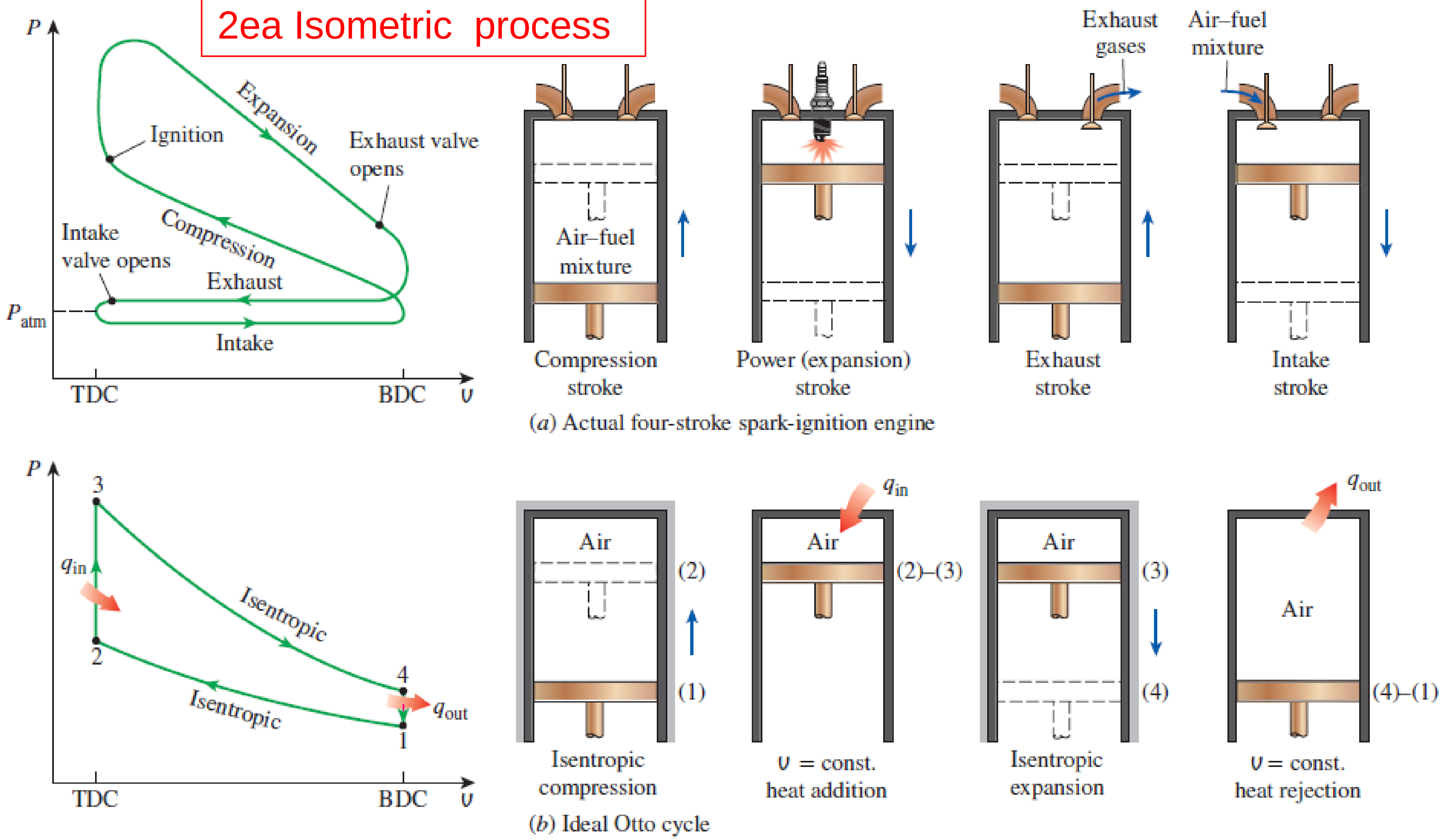


FIGURE 9–13
Actual and ideal cycles in spark-ignition engines and their *P*-υ diagrams.

Four-stroke cycle
1 cycle = 4 stroke = 2 revolution
Two-stroke cycle
1 cycle = 2 stroke = 1 revolution

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.

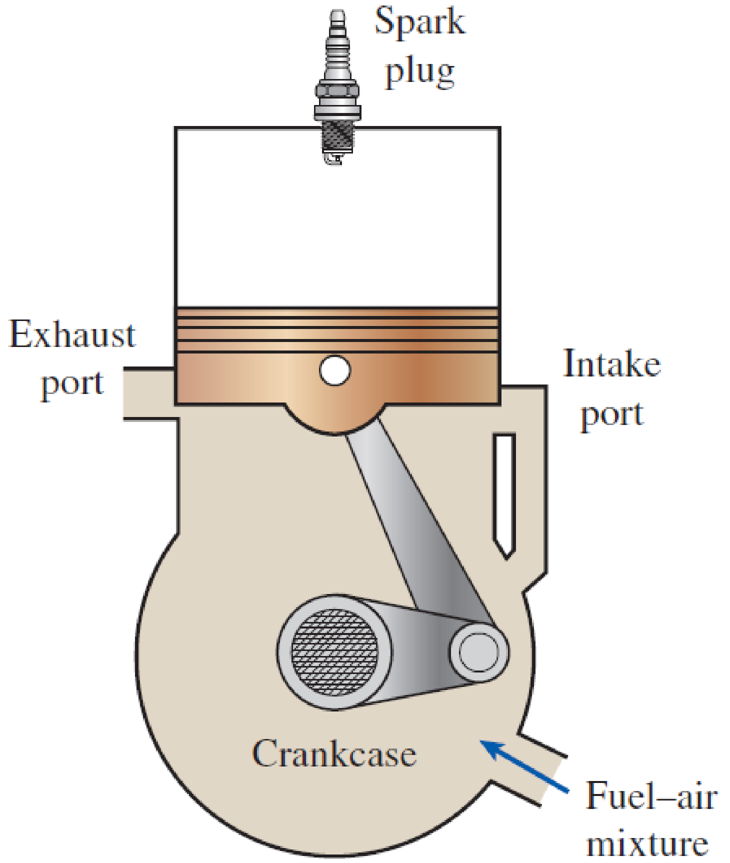


FIGURE 9–13
Schematic of a two-stroke reciprocating engine.

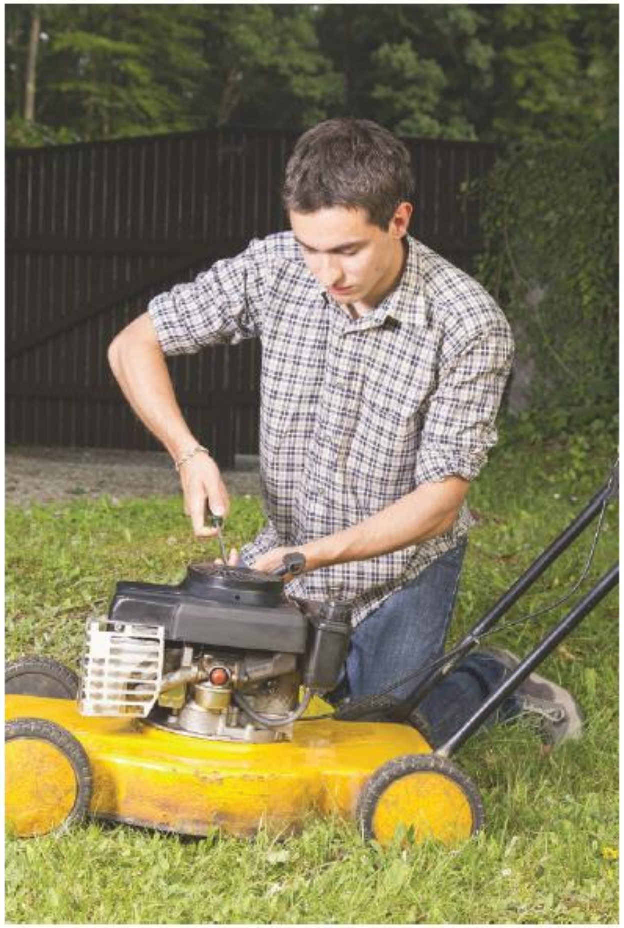
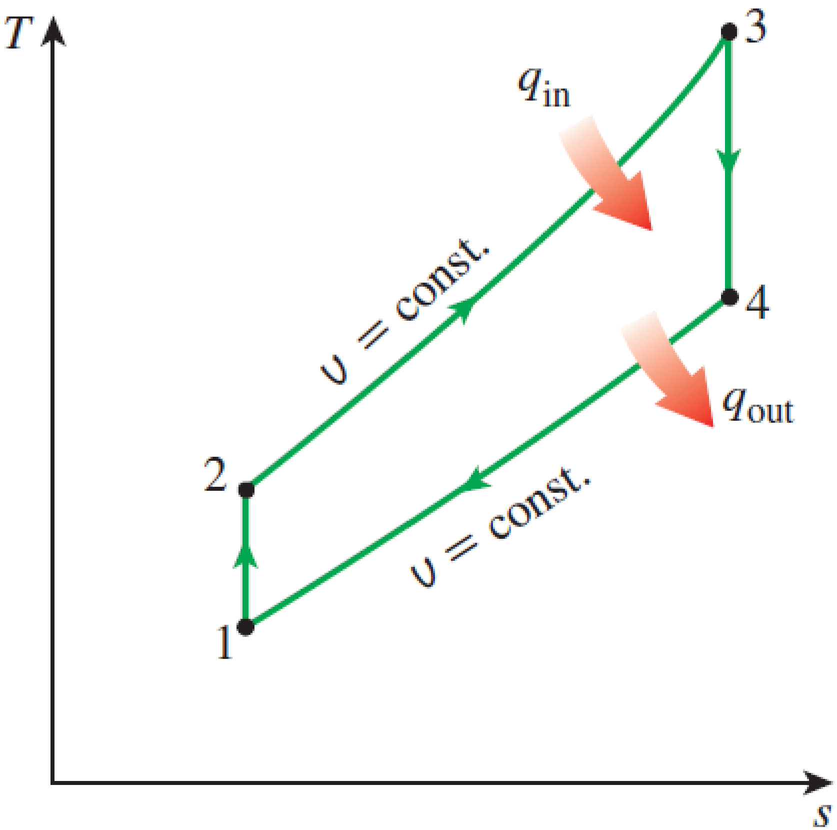


FIGURE 9–15
Two-stroke engines are commonly used in motorcycles and lawn mowers.

2ea Isentropic process
2ea Isometric process



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.

2ea Isentropic process
2ea Isometric process

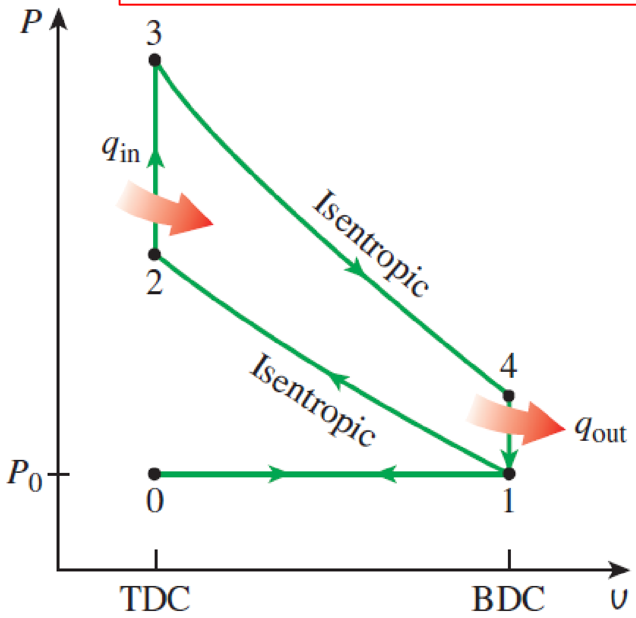


FIGURE 9–17
P-υ diagram of the ideal Otto cycle that includes intake and exhaust strokes.

$$w_{out,0\text{-}1} = P_0(\upsilon_1 - \upsilon_0)$$

$$w_{in,1\text{-}0} = P_0(\upsilon_1 - \upsilon_0)$$

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

Process 1-2: The intake valve is closed at state 1 and air is compressed isentropically to state 2. Piston moves from BDC to TDC.

Process 2-3: Heat is transferred at constant volume.

Process 3-4: Air is expanded isentropically.

Process 4-1: Heat is rejected at constant volume.

Process 1-0: Air is expelled through the open exhaust valve.

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.

Isometric process

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u \quad (kJ/kg) \qquad (9\text{-}5)$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2) \qquad q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \qquad (9\text{-}6)$$

Closed system — Closed system

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

From isentorpic process

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{v_3}{v_4}\right)^{k-1} = \frac{T_4}{T_3} \qquad (9\text{-}7)$$

From definition of compression ratio

$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2} = \frac{v_1}{v_2} \qquad (9\text{-}9)$$

$$\therefore \quad \eta_{th,Otto} = 1 - \frac{1}{r^{k-1}} \qquad (9\text{-}8)$$

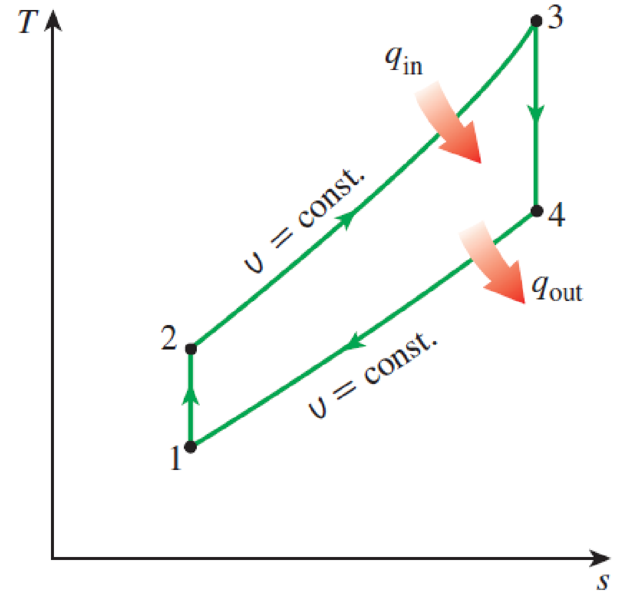


FIGURE 9–16
T-s diagram of the ideal Otto cycle.

9-5. OTTO CYCLE: IDEAL CYCLE FOR SI ENGINES

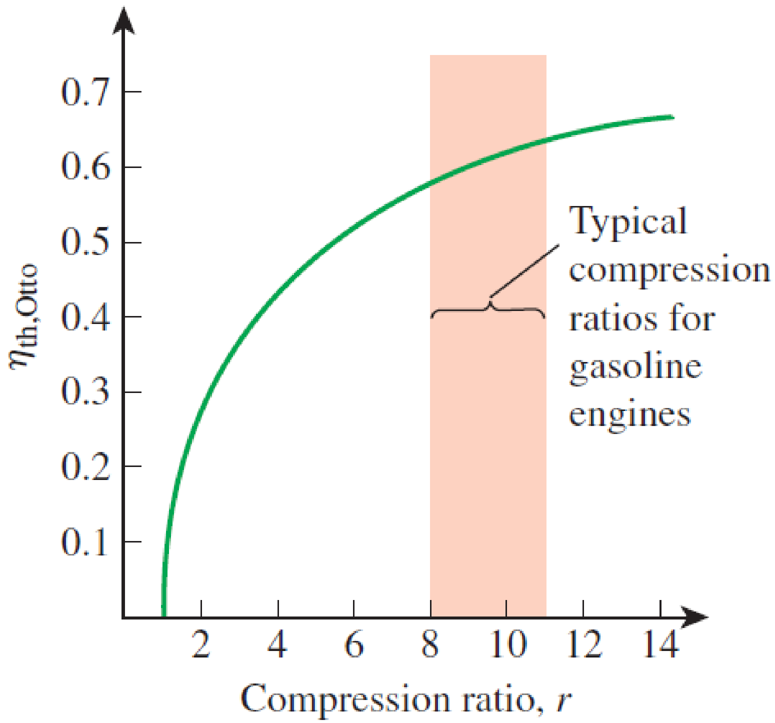


FIGURE 9–17
Thermal efficiency of the ideal Otto cycle as a function of compression ratio ($k = 1.4$).

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

Engine knock: The temperature of the air–fuel mixture rises above the autoignition temperature of the fuel 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.

Engine knock hurts performance and can cause engine damage

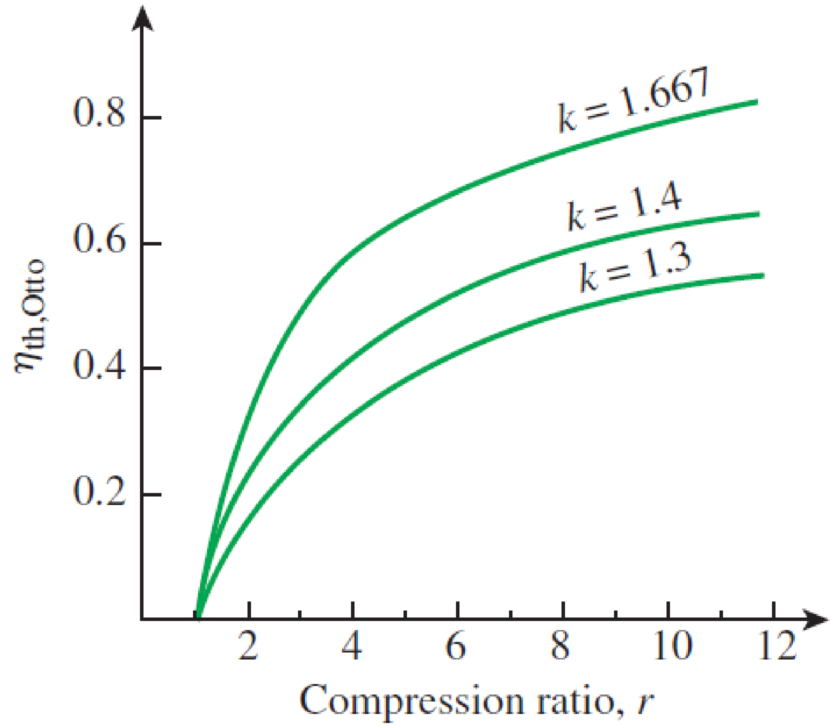


FIGURE 9–18

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

Example 9-3, 비열이 변화는 경우

EXAMPLE 9–3 The Ideal Otto Cycle

An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 100 kPa 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.

(*e*) Also, determine the power output from the cycle, in kW, for an engine speed of 4000 rpm (rev/min). Assume that this cycle is operated on an engine that has four cylinders with a total displacement volume of 1.6 L.

SOLUTION An ideal Otto cycle is considered. The maximum temperature and pressure, the net work output, the thermal efficiency, the mean effective pressure, and the power output for a given engine speed are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** The variation of specific heats with temperature is to be accounted for.

Analysis The *P*-*v* diagram of the ideal Otto cycle described is shown in Fig. 9–20. We note that the air contained in the cylinder forms a closed system.

(*a*) The maximum temperature and pressure in an Otto cycle occur at the end of the constant-volume heat-addition process (state 3). But first we need to determine the temperature and pressure of air at the end of the isentropic compression process (state 2), using data from Table A–17:

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

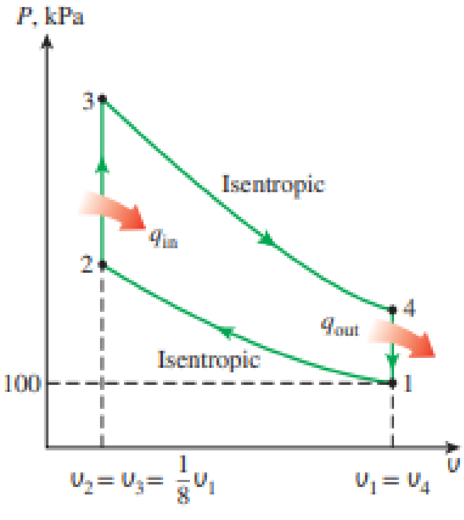


FIGURE 9–20
P-*v* diagram for the Otto cycle discussed in Example 9–3.

(*a*) The maximum temperature and pressure in an Otto cycle occur at the end of the constant-volume heat-addition process (state 3). But first we need to determine the temperature and pressure of air at the end of the isentropic compression process (state 2), using data from Table A–17:

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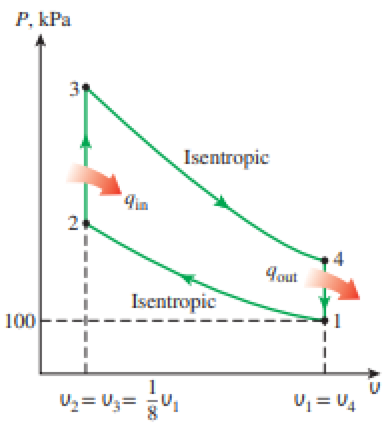


FIGURE 9–20
P-υ diagram for the Otto cycle discussed in Example 9–3.

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

$$\frac{\upsilon_{r2}}{\upsilon_{r1}} = \frac{\upsilon_2}{\upsilon_1} = \frac{1}{r} \rightarrow \upsilon_{r2} = \frac{\upsilon_{r1}}{r} = \frac{676.1}{8} = 84.51 \rightarrow T_2 = 652.4\text{ K}$$
$$u_2 = 475.11\text{ kJ/kg}$$

$$\frac{P_2\upsilon_2}{T_2} = \frac{P_1\upsilon_1}{T_1} \rightarrow P_2 = P_1\left(\frac{T_2}{T_1}\right)\left(\frac{\upsilon_1}{\upsilon_2}\right)$$
$$= (100\text{ kPa})\left(\frac{652.4\text{ K}}{290\text{ K}}\right)(8) = 1799.7\text{ kPa}$$

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

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

$$\frac{P_3\upsilon_3}{T_3} = \frac{P_2\upsilon_2}{T_2} \rightarrow P_3 = P_2\left(\frac{T_3}{T_2}\right)\left(\frac{\upsilon_2}{\upsilon_3}\right)$$
$$= (1.7997\text{ MPa})\left(\frac{1575.1\text{ K}}{652.4\text{ K}}\right)(1) = \mathbf{4.345\text{ MPa}}$$

(*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):

$$\frac{\upsilon_{r4}}{\upsilon_{r3}} = \frac{\upsilon_4}{\upsilon_3} = r \rightarrow \upsilon_{r4} = r\upsilon_{r3} = (8)(6.108) = 48.864 \rightarrow T_4 = 795.6\text{ K}$$
$$u_4 = 588.74\text{ kJ/kg}$$

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

$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1$$
$$q_{out} = 588.74 - 206.91 = 381.83\text{ kJ/kg}$$

Thus,

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

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

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

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

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

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:

$$\text{MEP} = \frac{w_{net}}{\upsilon_1 - \upsilon_2} = \frac{w_{net}}{\upsilon_1 - \upsilon_1/r} = \frac{w_{net}}{\upsilon_1(1 - 1/r)}$$

where

$$\upsilon_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa·m}^3\text{/kg·K})(290\text{ K})}{100\text{ kPa}} = 0.8323\text{ m}^3\text{/kg}$$

Thus,

$$\text{MEP} = \frac{418.17\text{ kJ/kg}}{(0.8323\text{ m}^3\text{/kg})(1 - \frac{1}{8})}\left(\frac{1\text{ kPa·m}^3}{1\text{ kJ}}\right) = \mathbf{574\text{ kPa}}$$

(*e*) The total air mass taken by all four cylinders when they are charged is

$$m = \frac{V_d}{\upsilon_1} = \frac{0.0016\text{ m}^3}{0.8323\text{ m}^3\text{/kg}} = 0.001922\text{ kg}$$

The net work produced by the cycle is

$$W_{net} = mw_{net} = (0.001922\text{ kg})(418.17\text{ kJ/kg}) = 0.8037\text{ kJ}$$

That is, the net work produced per thermodynamic cycle is 0.8037 kJ/cycle. Noting that there are two revolutions per thermodynamic cycle ($n_{rev} = 2$ rev/cycle) in a four-stroke engine (or in the ideal Otto cycle including intake and exhaust strokes), the power produced by the engine is determined from

$$\dot{W}_{net} = \frac{W_{net}\dot{n}}{n_{rev}} = \frac{(0.8037\text{ kJ/cycle})(4000\text{ rev/min})}{2\text{ rev/cycle}}\left(\frac{1\text{ min}}{60\text{ s}}\right) = \mathbf{26.8\text{ kW}}$$

Discussion If we analyzed a two-stroke engine operating on an ideal Otto cycle with the same values, the power output would be calculated as

$$\dot{W}_{net} = \frac{W_{net}\dot{n}}{n_{rev}} = \frac{(0.8037\text{ kJ/cycle})(4000\text{ rev/min})}{1\text{ rev/cycle}}\left(\frac{1\text{ min}}{60\text{ s}}\right) = 53.6\text{ kW}$$

Note that there is one revolution in one thermodynamic cycle in two-stroke engines.

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

2ea Isentropic process
1ea Isometric process
1ea Isobaric process

In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition.

Diesel engines can be designed to operate at much higher compression ratios than SI engines, typically between 12 and 24.

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.

The spark plug is replaced by a fuel injector in diesel engines.

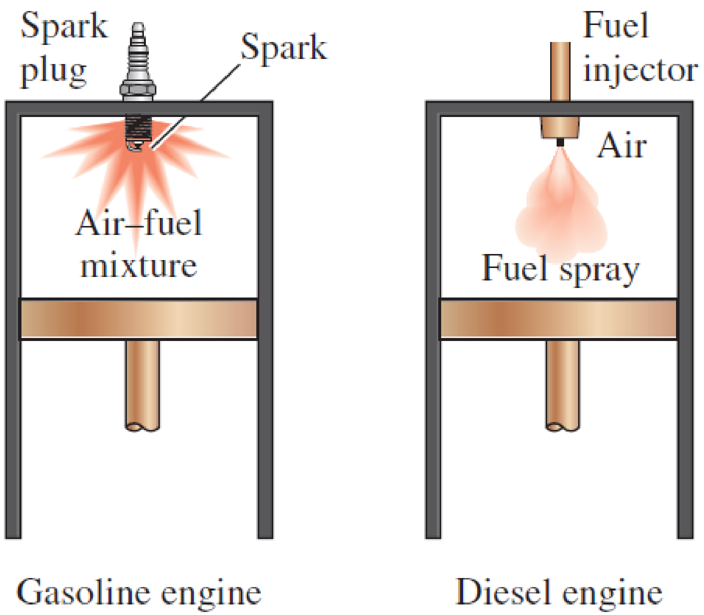
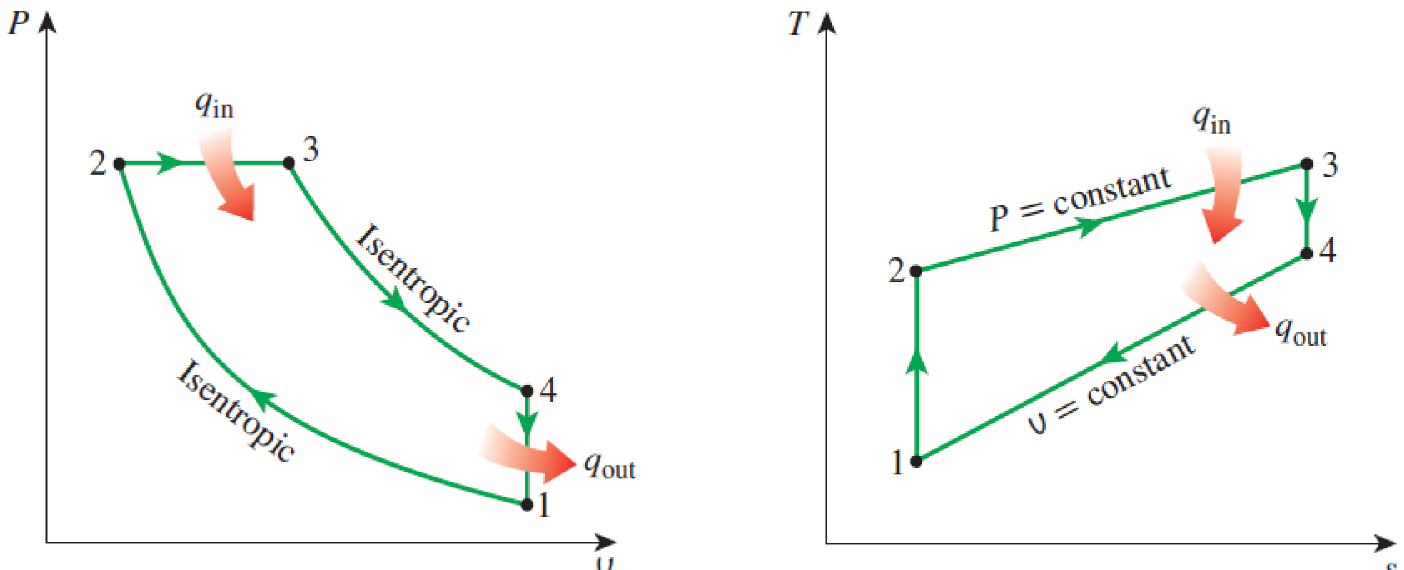


FIGURE 9–20
In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

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

2ea Isentropic process
1ea Isometric process
1ea Isobaric process

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



(*a*) *P*-υ diagram (*b*) *T*-*s* diagram

FIGURE 9–22

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

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

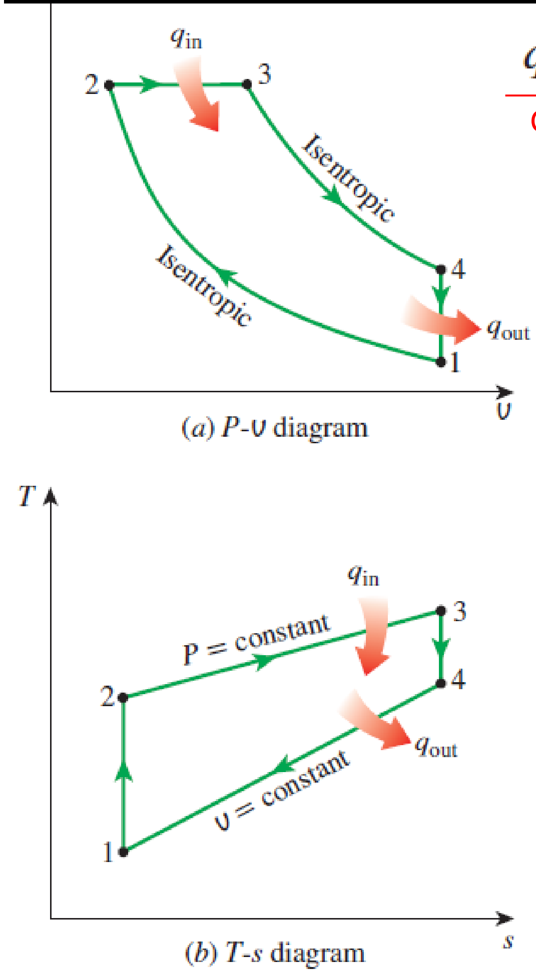


FIGURE 9–22
T-s and P-υ diagrams for the ideal Diesel cycle.

$$q_{in} - w_{b,out} = u_3 - u_2 \rightarrow q_{in} = P_2(\upsilon_3 - \upsilon_2) + (u_3 - u_2)$$

Closed system for cont. press.

$$= h_3 - h_2 = c_p(T_3 - T_2) \quad (9\text{-}10a)$$

$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_\upsilon(T_4 - T_1) \quad (9\text{-}10b)$$

Close system for cont. volume

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

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

$$r_c = \frac{V_3}{V_2} = \frac{\upsilon_3}{\upsilon_2} \quad \text{Cutoff ratio} \quad (9\text{-}11)$$

From 2→3 정압 과정, 3→4 등엔트로피 과정과 차단비 정의에 따라서 정리하면,

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

$$\eta_{th,Otto} > \eta_{th,Diesel}$$ for the same compression ratio (9-13)

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

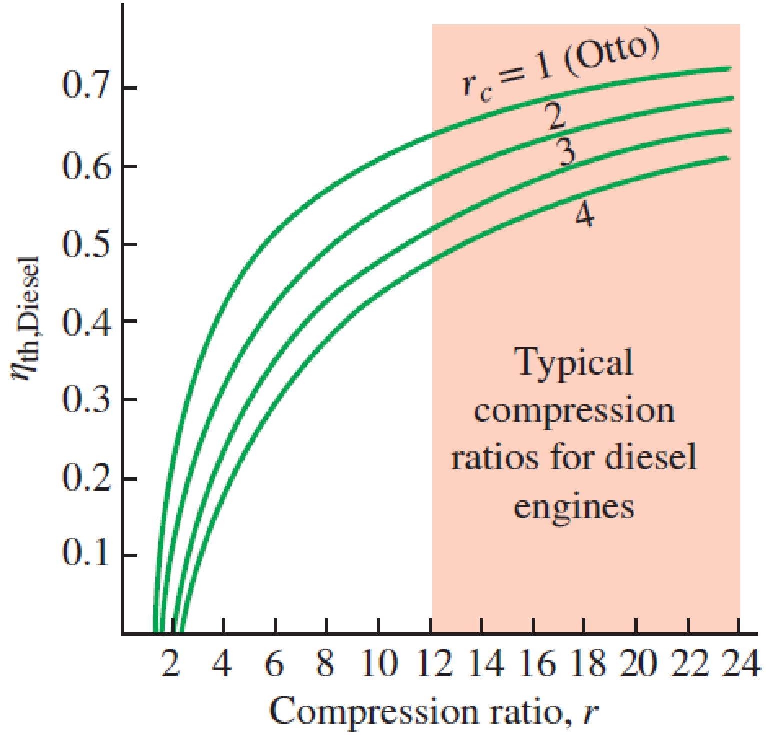


FIGURE 9–23

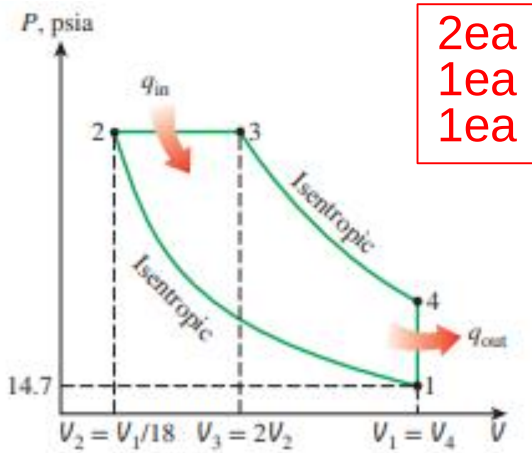
Thermal efficiency of the ideal Diesel cycle as a function of compression and cutoff ratios ($k = 1.4$).

9-6. DIESEL CYCLE: CI ENGINES

Example 9-4, 비열이 일정한 경우

EXAMPLE 9–4 The Ideal Diesel Cycle

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 14.7 psia, 80°F, and 117 in^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.



2ea Isentropic process
1ea Isometric process
1ea Isobaric process

FIGURE 9–25
P-V diagram for the ideal Diesel cycle discussed in Example 9–4.

Analysis The *P-V* diagram of the ideal Diesel cycle described is shown in Fig. 9–25. We note that the air contained in the cylinder forms a closed system.

(*a*) The temperature and pressure values at the end of each process can be determined by utilizing the ideal-gas isentropic relations for processes 1-2 and 3-4. But first we determine the volumes at the end of each process from the definitions of the compression ratio and the cutoff ratio:

$$V_2 = \frac{V_1}{r} = \frac{117\ \text{in}^3}{18} = 6.5\ \text{in}^3$$
$$V_3 = r_c V_2 = (2)(6.5\ \text{in}^3) = 13\ \text{in}^3$$
$$V_4 = V_1 = 117\ \text{in}^3$$

Process 1-2 (isentropic compression of an ideal gas, constant specific heats):

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (540\ \text{R})(18)^{1.4-1} = \mathbf{1716\ R}$$
$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{k} = (14.7\ \text{psia})(18)^{1.4} = \mathbf{841\ psia}$$

Process 2-3 (constant-pressure heat addition to an ideal gas):

$$P_3 = P_2 = \mathbf{841\ psia}$$
$$\frac{P_2V_2}{T_2} = \frac{P_3V_3}{T_3} \rightarrow T_3 = T_2\left(\frac{V_3}{V_2}\right) = (1716\ \text{R})(2) = \mathbf{3432\ R}$$

Process 3-4 (isentropic expansion of an ideal gas, constant specific heats):

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = (3432\ \text{R})\left(\frac{13\ \text{in}^3}{117\ \text{in}^3}\right)^{1.4-1} = \mathbf{1425\ R}$$
$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = (841\ \text{psia})\left(\frac{13\ \text{in}^3}{117\ \text{in}^3}\right)^{1.4} = \mathbf{38.8\ psia}$$

(*b*) The net work for a cycle is equivalent to the net heat transfer. But first we find the mass of air:

$$m = \frac{P_1V_1}{RT_1} = \frac{(14.7\ \text{psia})(117\ \text{in}^3)}{(0.3704\ \text{psia·ft}^3/\text{lbm·R})(540\ \text{R})}\left(\frac{1\ \text{ft}^3}{1728\ \text{in}^3}\right) = 0.00498\ \text{lbm}$$

Process 2-3 is a constant-pressure heat-addition process for which the boundary work and Δu terms can be combined into Δh. Thus,

$$Q_{in} = m(h_3 - h_2) = mc_p(T_3 - T_2)$$
$$= (0.00498\ \text{lbm})(0.240\ \text{Btu/lbm·R})[(3432 - 1716)\ \text{R}]$$
$$= 2.051\ \text{Btu}$$

Process 4-1 is a constant-volume heat-rejection process (it involves no work interactions), and the amount of heat rejected is

$$Q_{out} = m(u_4 - u_1) = mc_v(T_4 - T_1)$$
$$= (0.00498\ \text{lbm})(0.171\ \text{Btu/lbm·R})[(1425 - 540)\ \text{R}]$$
$$= 0.754\ \text{Btu}$$

Thus,

$$W_{net} = Q_{in} - Q_{out} = 2.051 - 0.754 = \mathbf{1.297\ Btu}$$

Then the thermal efficiency becomes

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{1.297\ \text{Btu}}{2.051\ \text{Btu}} = 0.632 \quad \text{or} \quad \mathbf{63.2\%}$$

The thermal efficiency of this Diesel cycle under the cold-air-standard assumptions could also be determined from Eq. 9–12.

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

$$\text{MEP} = \frac{W_{net}}{V_{max} - V_{min}} = \frac{W_{net}}{V_1 - V_2} = \frac{1.297\ \text{Btu}}{(117 - 6.5)\ \text{in}^3}\left(\frac{778.17\ \text{lbf·ft}}{1\ \text{Btu}}\right)\left(\frac{12\ \text{in}}{1\ \text{ft}}\right)$$
$$= \mathbf{110\ psia}$$

Discussion Note that a constant pressure of 110 psia during the power stroke would produce the same net work output as the entire Diesel cycle.

H₂ Combustion SCNU Comb. Lab.

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

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

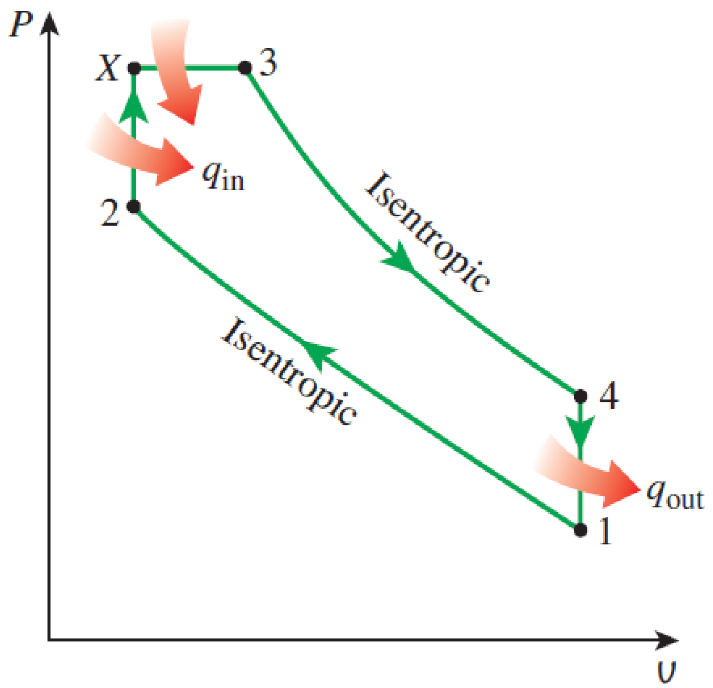


FIGURE 9–24

P-υ 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.

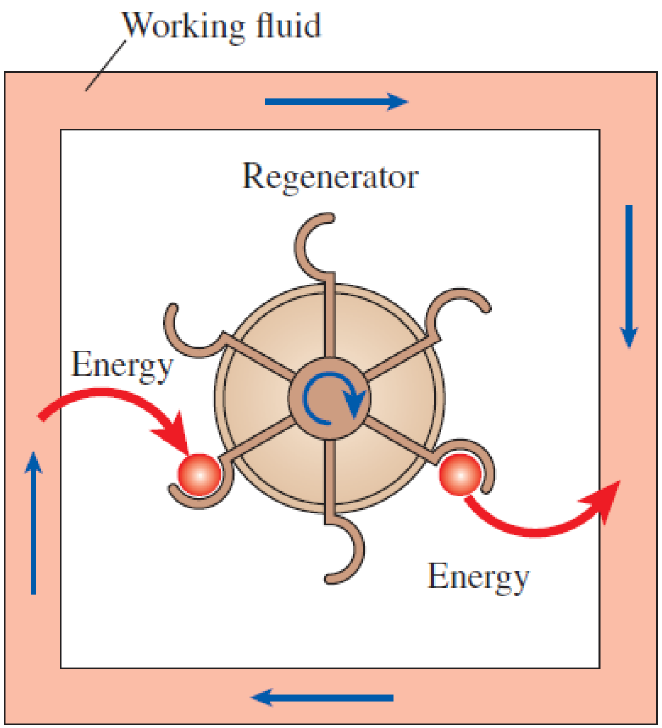


FIGURE 9–25
A regenerator is a device that borrows energy from the working fluid during one part of the cycle and pays it back (without interest) during another part.

Stirling cycle

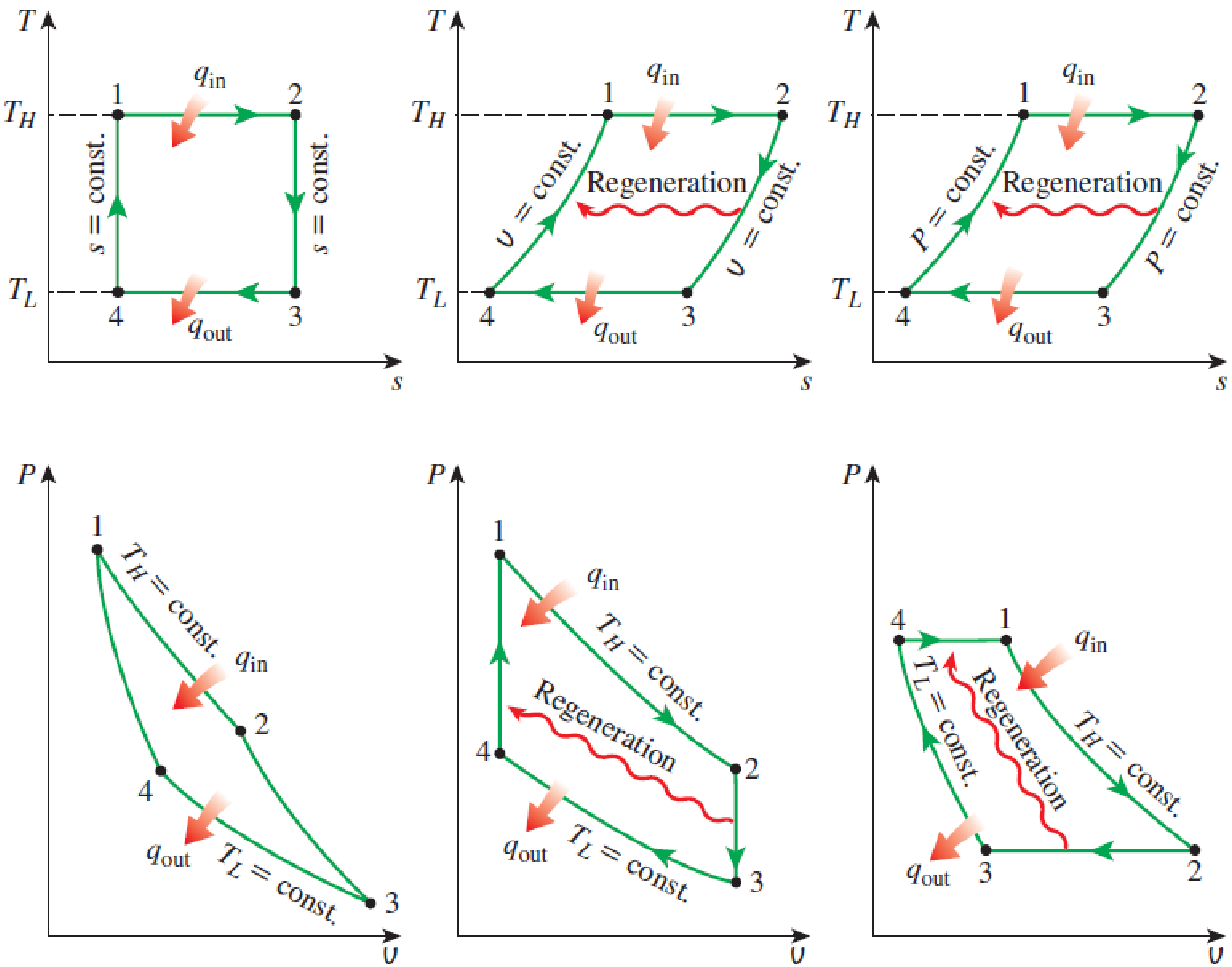
1-2 *T = constant* expansion (heat addition from the external source)

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)

9-7. STIRLING AND ERICSSON CYCLES



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

FIGURE 9–27

T-s and *P-υ* diagrams of Carnot, Stirling, and Ericsson cycles.

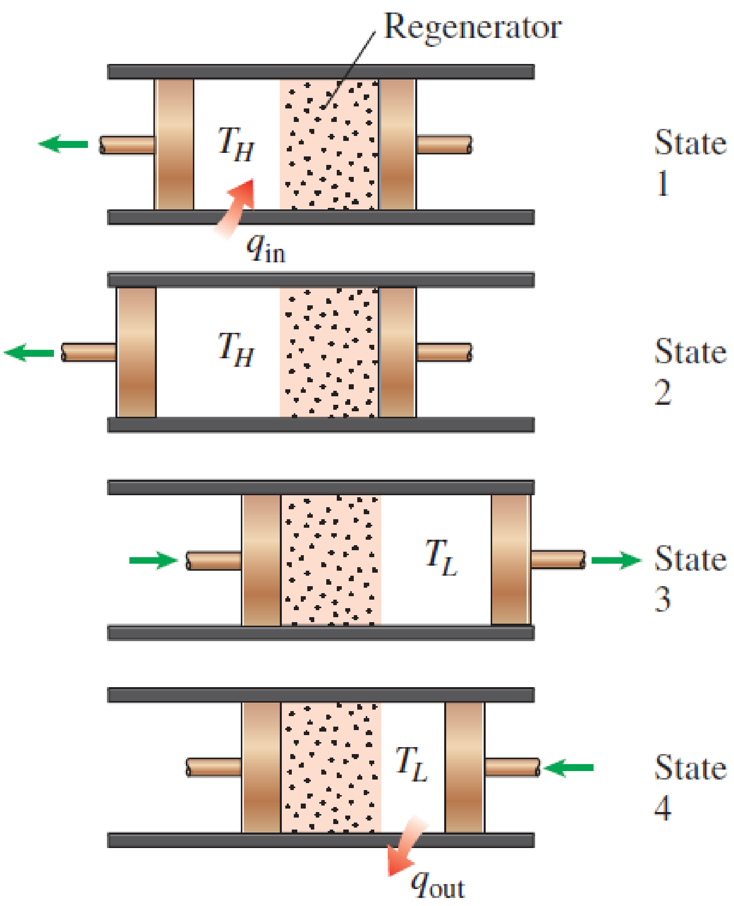


FIGURE 9–27
The execution of the Stirling cycle.

Both the Stirling and Ericsson cycles are totally reversible, as is the Carnot cycle, and thus:

$$\eta_{th,Stirling} = \eta_{th,Ericsson} = \eta_{th,Carnot} = 1 - \frac{T_L}{T_H}$$

The Stirling and Ericsson cycles give a message: *Regeneration can increase efficiency.*

9-7. STIRLING AND ERICSSON CYCLES

The Ericsson cycle is very much like the Stirling cycle, except that the two constant-volume processes are replaced by two constant-pressure processes.

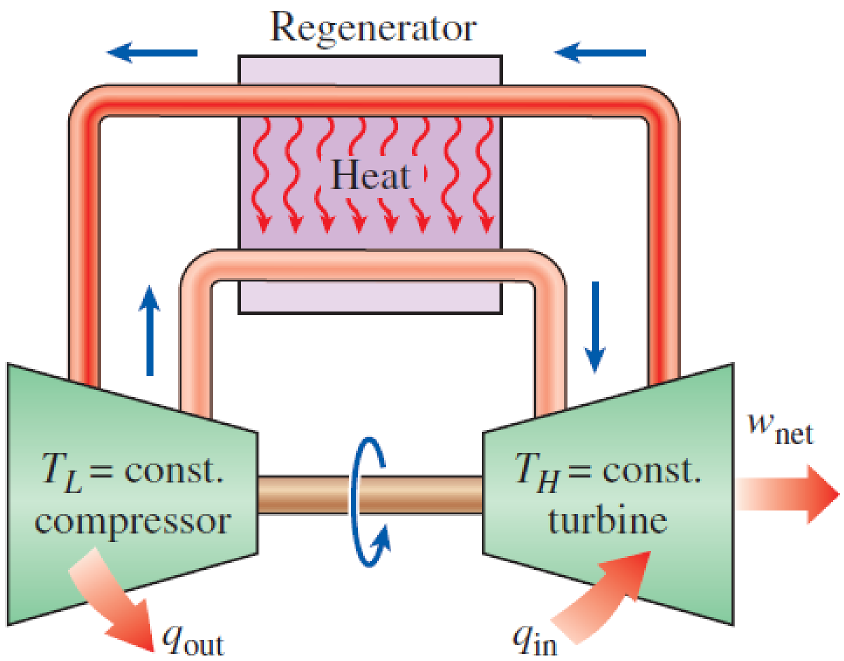


FIGURE 9–28

A steady-flow Ericsson engine.

9-8. 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

2ea Isentropic process
2ea Isobaric process

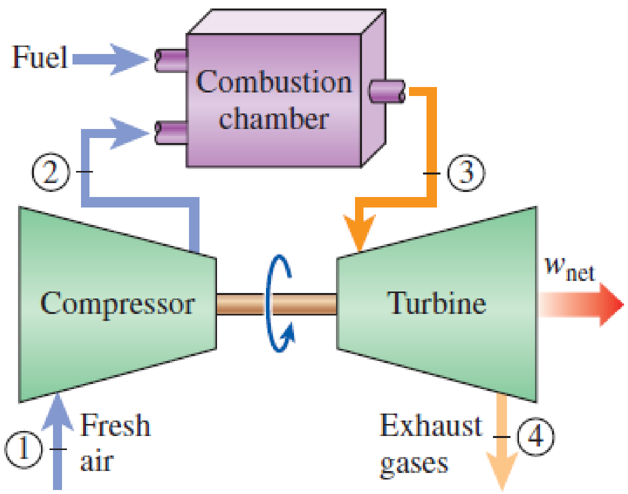


FIGURE 9–29
An open-cycle gas-turbine engine.

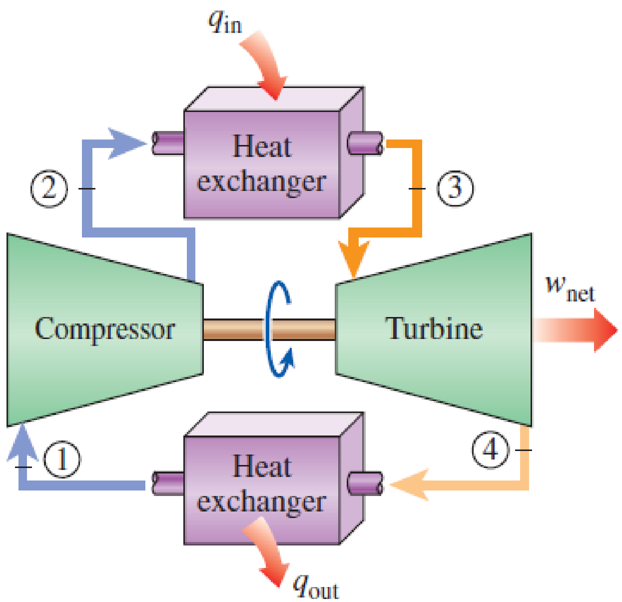


FIGURE 9–30
A closed-cycle gas-turbine engine.

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

2ea Isentropic process
2ea Isobaric process

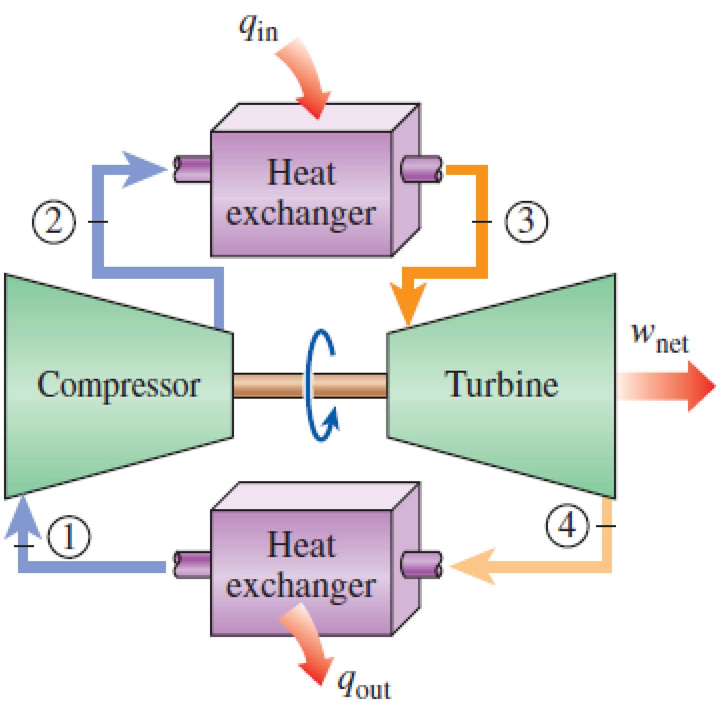


FIGURE 9–30

A closed-cycle gas-turbine engine.

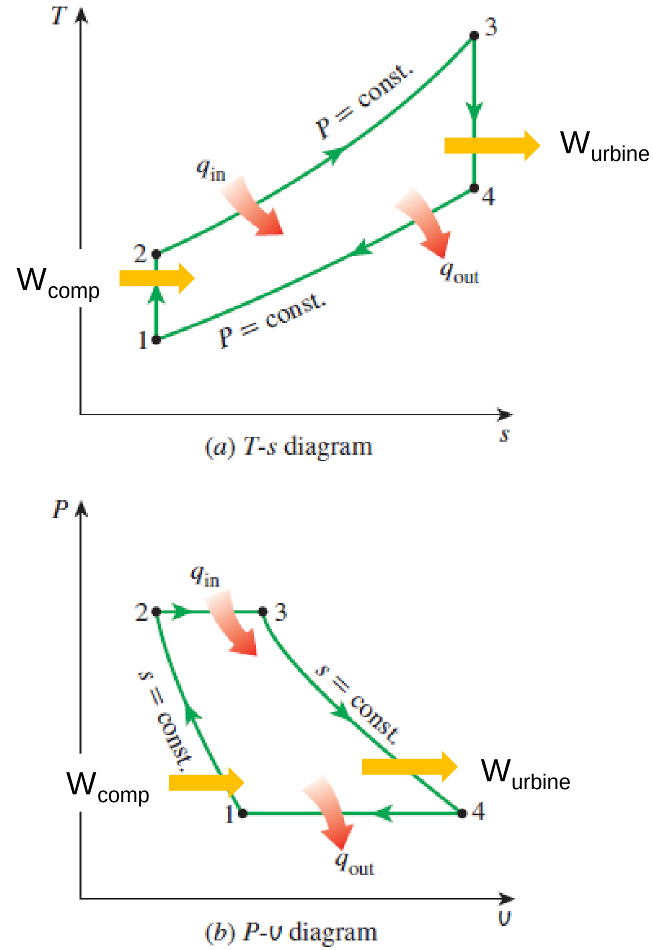


FIGURE 9–32

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

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

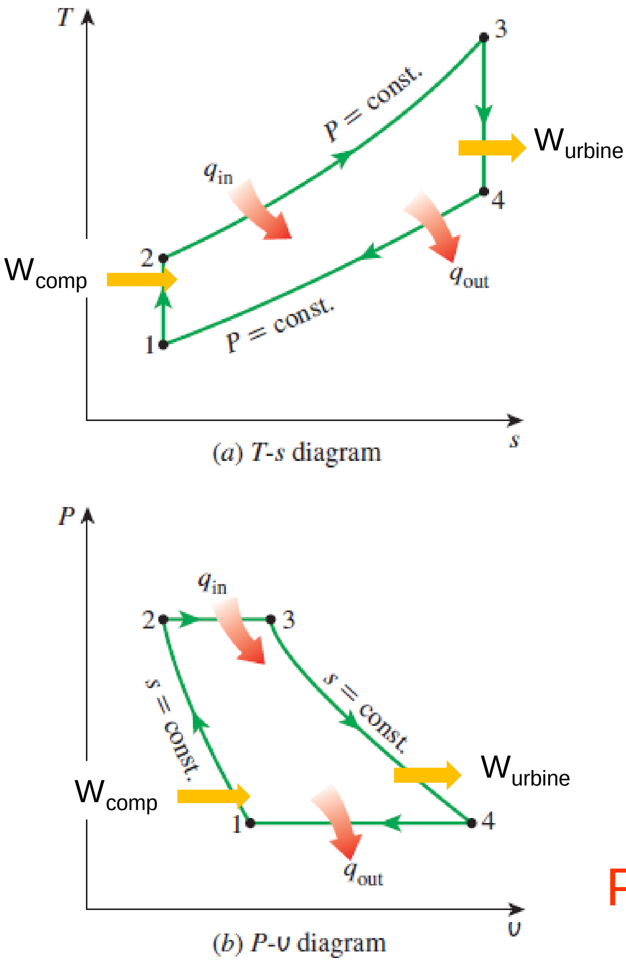


FIGURE 9–32
T-s and *P-υ* diagrams for the ideal Brayton cycle.

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{exit} - h_{inlet} \qquad \text{(9-15)}$$

Energy balancing eq'n for open system

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) \qquad \text{(9-16a)}$$

$$q_{out} = h_4 - h_1 = c_p(T_4 - T_1) \qquad \text{(9-16b)}$$

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

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

Pressure ratio $r_p = \frac{P_2}{P_1}$ (9-18)

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

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

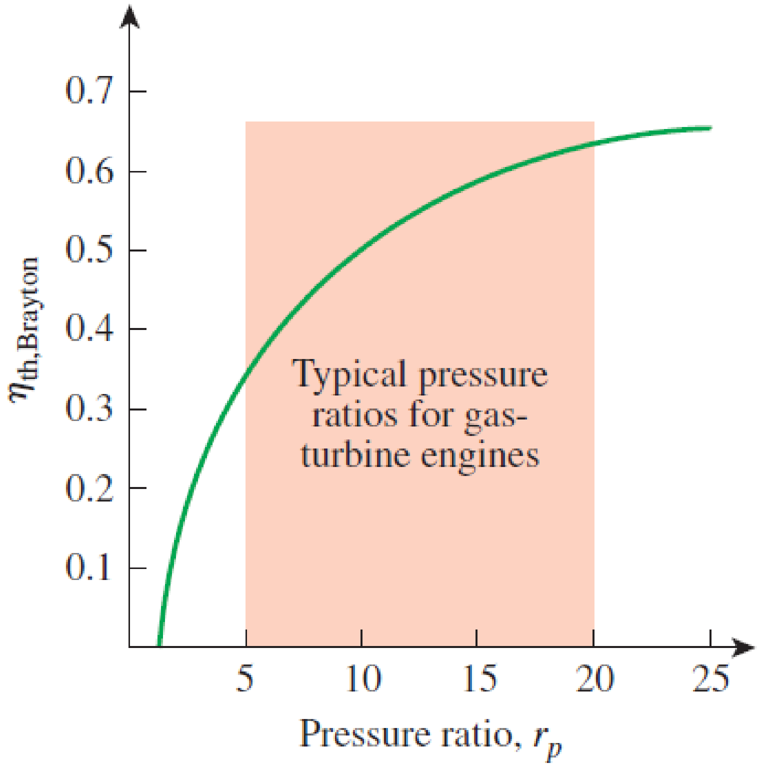


FIGURE 9–32
Thermal efficiency of the ideal Brayton cycle as a function of the pressure ratio.

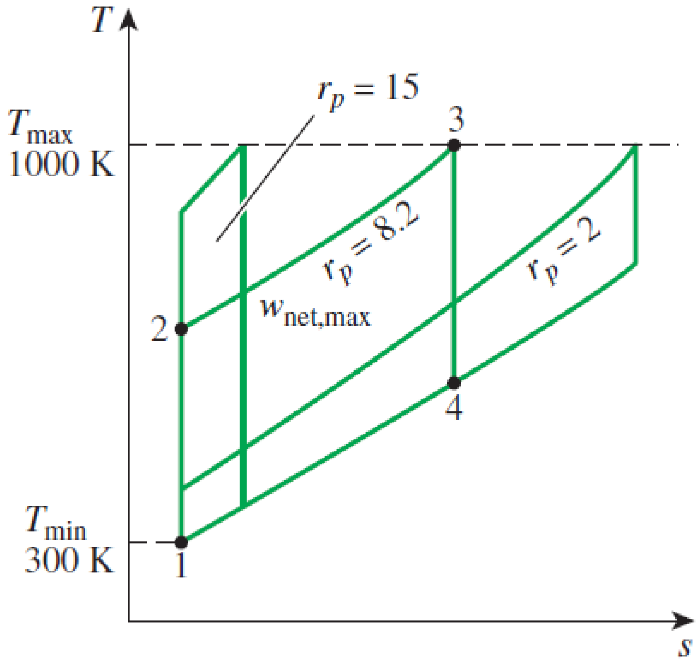


FIGURE 9–33
For fixed values of T_{min} and T_{max}, the net work of the Brayton cycle first increases with the pressure ratio, then reaches a maximum at $r_p = (T_{max}/T_{min})^{k/[2(k-1)]}$, and finally decreases.

Example 9-6, 이상적인 브레이튼 사이클, 비열이 변하는 경우

EXAMPLE 9–6 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 1300 K at the turbine inlet. Using 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.

SOLUTION A power plant operating on the ideal Brayton cycle is considered. The compressor and turbine exit temperatures, back work ratio, and thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** The variation of specific heats with temperature is to be considered.

Analysis The *T-s* diagram of the ideal Brayton cycle described is shown in Fig. 9–36. We note that the components involved in the Brayton cycle are steady-flow devices.

(*a*) The air temperatures at the compressor and turbine exits are determined from isentropic relations:

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

$$T_1 = 300\text{ K} \rightarrow h_1 = 300.19\text{ kJ/kg}$$
$$P_{r1} = 1.386$$
$$P_{r2} = \frac{P_2}{P_1}P_{r1} = (8)(1.386) = 11.09 \rightarrow T_2 = \mathbf{540\ K} \quad \text{(at compressor exit)}$$
$$h_2 = 544.35\text{ kJ/kg}$$

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

$$T_3 = 1300\text{ K} \rightarrow h_3 = 1395.97\text{ kJ/kg}$$
$$P_{r3} = 330.9$$
$$P_{r4} = \frac{P_4}{P_3}P_{r3} = \left(\frac{1}{8}\right)(330.9) = 41.36 \rightarrow T_4 = \mathbf{770\ K} \quad \text{(at turbine exit)}$$
$$h_4 = 789.37\text{ kJ/kg}$$

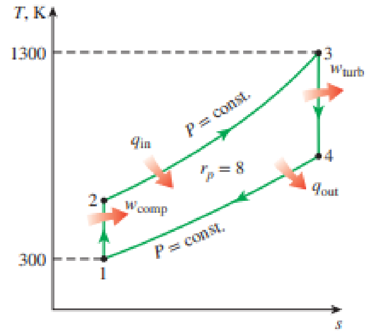


FIGURE 9–36
T-s diagram for the Brayton cycle discussed in Example 9–6.

(*b*) To find the back work ratio, we need to find the work input to the compressor and the work output of the turbine:

$$w_{comp,in} = h_2 - h_1 = 544.35 - 300.19 = 244.16\text{ kJ/kg}$$
$$w_{turb,out} = h_3 - h_4 = 1395.97 - 789.37 = 606.60\text{ kJ/kg}$$

Thus,

$$r_{bw} = \frac{w_{comp,in}}{w_{turb,out}} = \frac{244.16\text{ kJ/kg}}{606.60\text{ kJ/kg}} = \mathbf{0.403}$$

That is, 40.3 percent of the turbine work output is used just to drive the compressor.

(*c*) The thermal efficiency of the cycle is the ratio of the net power output to the total heat input:

$$q_{in} = h_3 - h_2 = 1395.97 - 544.35 = 851.62\text{ kJ/kg}$$
$$w_{net} = w_{out} - w_{in} = 606.60 - 244.16 = 362.4\text{ kJ/kg}$$

Thus,

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{362.4\text{ kJ/kg}}{851.62\text{ kJ/kg}} = 0.426 \quad \text{or} \quad \mathbf{42.6\%}$$

The thermal efficiency could also be determined from

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

where

$$q_{out} = h_4 - h_1 = 789.37 - 300.19 = 489.2\text{ kJ/kg}$$

Discussion Under the cold-air-standard assumptions (constant specific heat values at room temperature), the thermal efficiency would be, from Eq. 9–17,

$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} = 1 - \frac{1}{8^{(1.4-1)/1.4}} = 0.448 \quad \text{or} \quad 44.8\%$$

which is sufficiently close to the value obtained by accounting for the variation of specific heats with temperature.

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

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

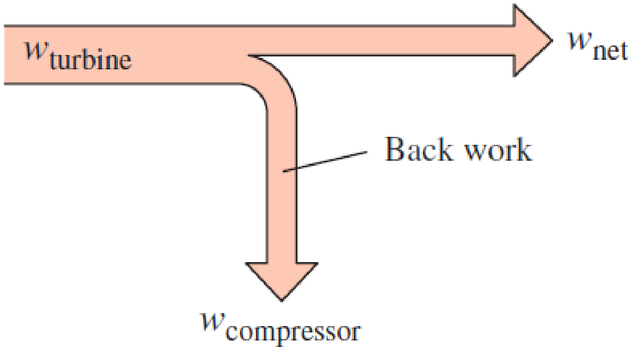


FIGURE 9–34
The fraction of the turbine work used to drive the compressor is called the back work ratio.

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.

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

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

$$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1} \qquad (9\text{-}19)$$

$$\eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \qquad (9\text{-}20)$$

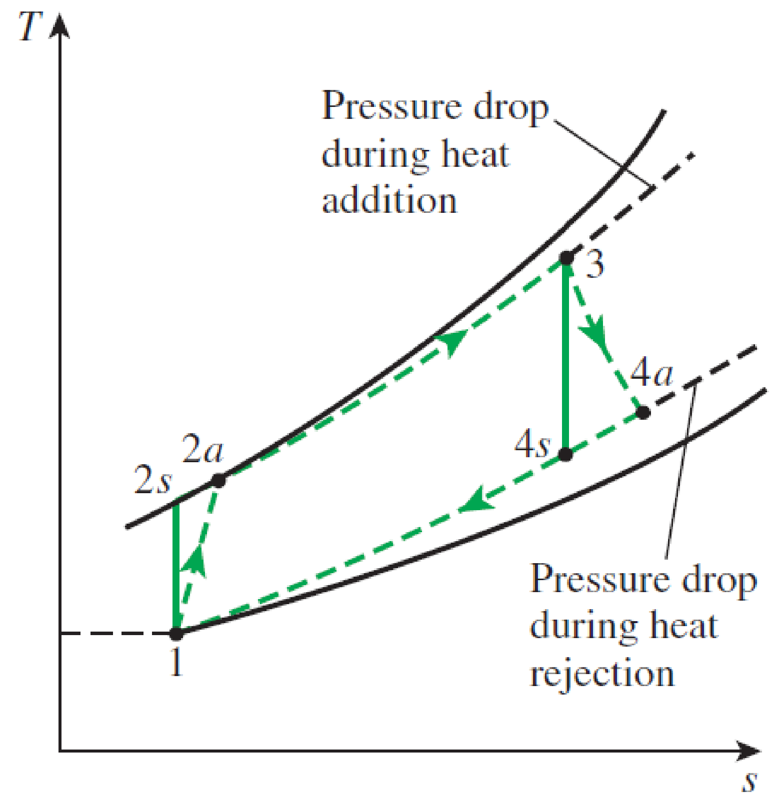


FIGURE 9–36
The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

Example 9-7, 실제 가스터빈 사이클, 비열이 변하는 경우

EXAMPLE 9–7 **An Actual Gas-Turbine Cycle**

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 9–6.

Analysis (*a*) The *T-s* diagram of the cycle is shown in Fig. 9–38. The actual compressor work and turbine work are determined by using the definitions of compressor and turbine efficiencies, Eqs. 9–19 and 9–20:

Compressor:
$$w_{comp,in} = \frac{w_s}{\eta_C} = \frac{244.16 \text{ kJ/kg}}{0.80} = 305.20 \text{ kJ/kg}$$

Turbine:
$$w_{turb,out} = \eta_T w_s = (0.85)(606.60 \text{ kJ/kg}) = 515.61 \text{ kJ/kg}$$

Thus,

$$r_{bw} = \frac{w_{comp,in}}{w_{turb,out}} = \frac{305.20 \text{ kJ/kg}}{515.61 \text{ kJ/kg}} = \mathbf{0.592}$$

That is, the compressor is now consuming 59.2 percent of the work produced by the turbine (up from 40.3 percent). This increase is due to the irreversibilities that occur within the compressor and the turbine.

(*b*) In this case, air leaves the compressor at a higher temperature and enthalpy, which are determined to be

$$w_{comp,in} = h_{2a} - h_1 \rightarrow h_{2a} = h_1 + w_{comp,in}$$
$$= 300.19 + 305.20$$
$$= 605.39 \text{ kJ/kg} \quad (\text{and } T_{2a} = 598 \text{ K})$$

Thus,

$$q_{in} = h_3 - h_{2a} = 1395.97 - 605.39 = 790.58 \text{ kJ/kg}$$
$$w_{net} = w_{out} - w_{in} = 515.61 - 305.20 = 210.41 \text{ kJ/kg}$$

and

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{210.41 \text{ kJ/kg}}{790.58 \text{ kJ/kg}} = 0.266 \quad \text{or} \quad 26.6\%$$

c.f 42.6%

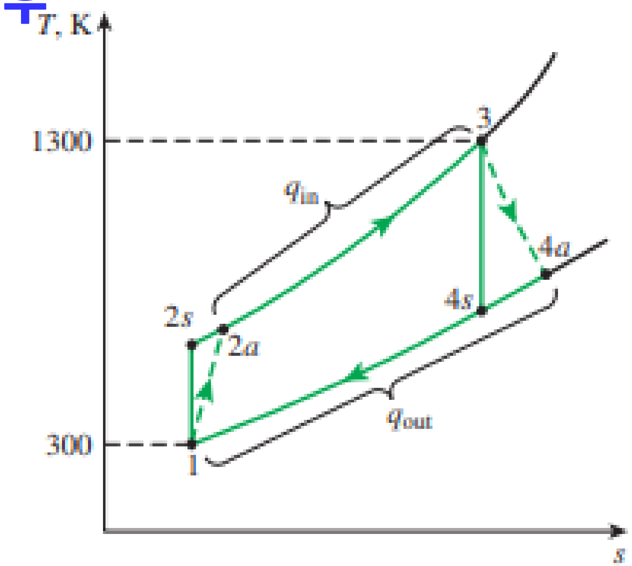


FIGURE 9–38

T-s diagram of the gas-turbine cycle discussed in Example 9–7.

(*c*) The air temperature at the turbine exit is determined from an energy balance on the turbine:

$$w_{turb,out} = h_3 - h_{4a} \rightarrow h_{4a} = h_3 - w_{turb,out}$$
$$= 1395.97 - 515.61$$
$$= 880.36 \text{ kJ/kg}$$

Then, from Table A–17,

$$T_{4a} = \mathbf{853 \text{ K}}$$

Discussion The temperature at turbine exit is considerably higher than that at the compressor exit (T_{2a} = 598 K), which suggests the use of regeneration to reduce fuel cost.

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.

The high-pressure air leaving the compressor can be heated by the hot exhaust gases in a counter-flow heat exchanger (a *regenerator* or a *recuperator*).

The thermal efficiency of the Brayton cycle increases as a result of regeneration since less fuel is used for the same work output.

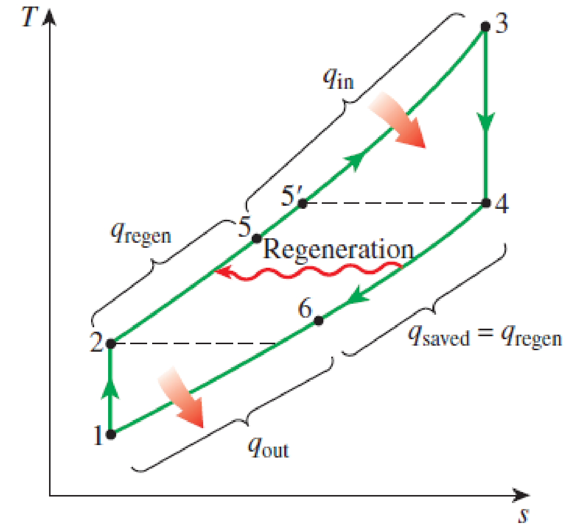
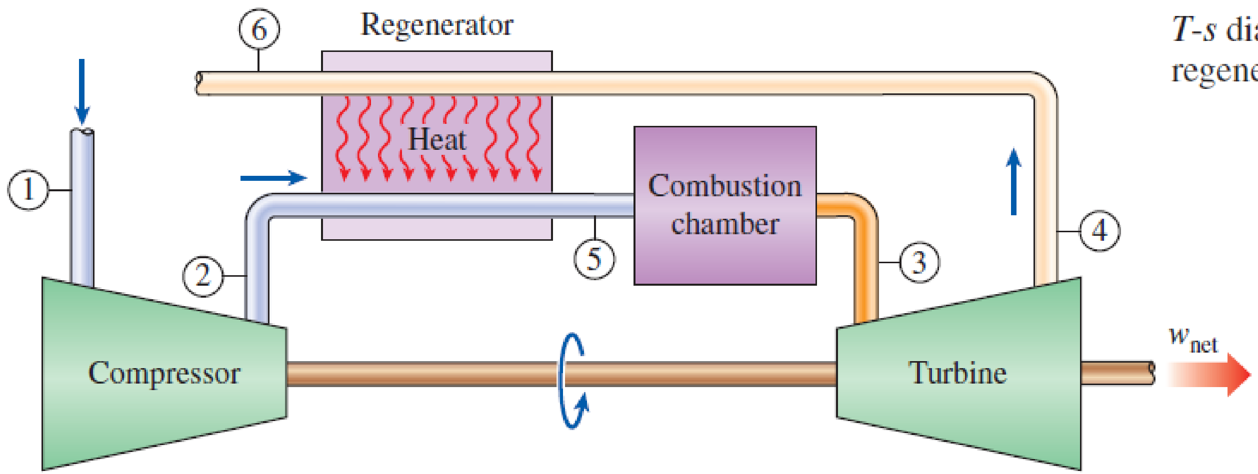
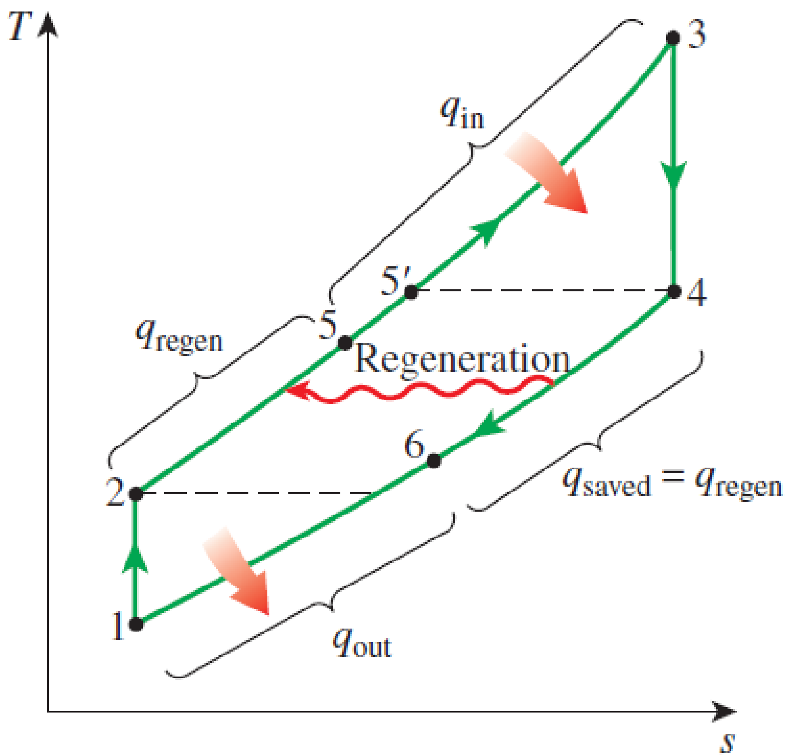


FIGURE 9–39
T-s diagram of a Brayton cycle with regeneration.



A gas-turbine engine with regenerator.

9-9. THE BRAYTON CYCLE WITH REGENERATION



$$q_{regen,act} = h_5 - h_2 \quad (9\text{-}21)$$

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

Effectiveness of regenerator

$$\epsilon = \frac{q_{regen,act}}{q_{regen,max}} = \frac{h_5 - h_2}{h_4 - h_2} \quad (9\text{-}23)$$

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

Effectiveness under cold-air standard assumptions

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

Efficiency under cold-air standard assumptions

The thermal efficiency depends on the ratio of the minimum to maximum temperatures as well as the pressure ratio.

Regeneration is most effective at lower pressure ratios and low minimum-to-maximum temperature ratios.

9-9. THE BRAYTON CYCLE WITH REGENERATION

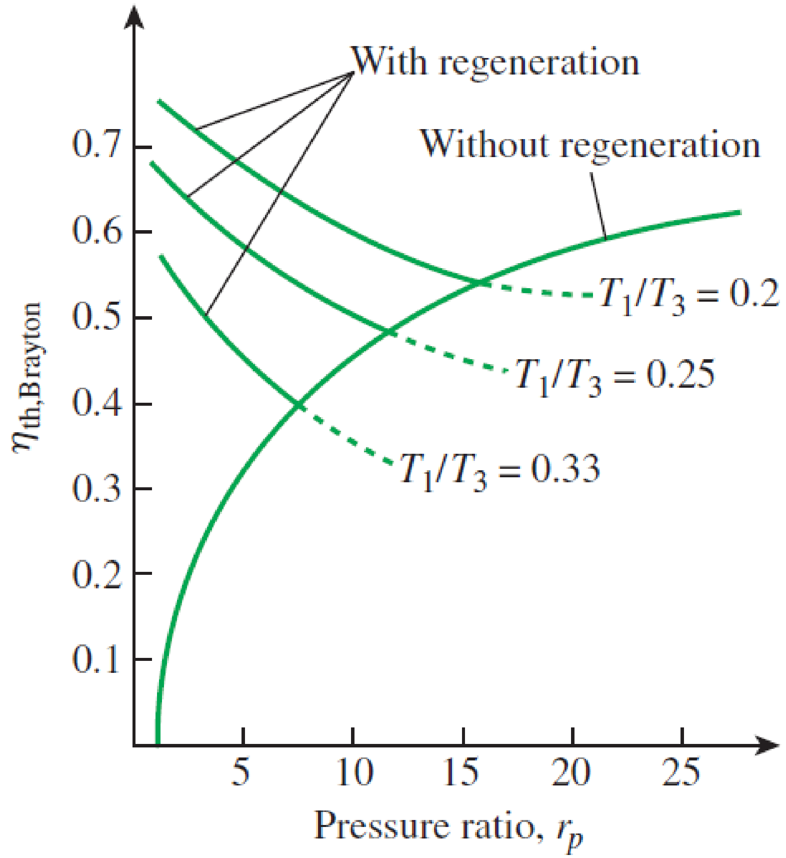


FIGURE 9–40

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

Example 9-8, 실제 재생 가스터빈 사이클, 비열이 변하는 경우

EXAMPLE 9–8 Actual Gas-Turbine Cycle with Regeneration

Determine the thermal efficiency of the gas turbine described in Example 9–7 if a regenerator having an effectiveness of 80 percent is installed.

SOLUTION The gas turbine discussed in Example 9–7 is equipped with a regenerator. For a specified effectiveness, the thermal efficiency is to be determined.

Analysis The *T-s* diagram of the cycle is shown in Fig. 9–42. We first determine the enthalpy of the air at the exit of the regenerator, using the definition of effectiveness:

$$\epsilon = \frac{h_5 - h_{2a}}{h_{4a} - h_{2a}}$$

$$0.80 = \frac{(h_5 - 605.39)\ \text{kJ/kg}}{(880.36 - 605.39)\ \text{kJ/kg}} \rightarrow h_5 = 825.37\ \text{kJ/kg}$$

Thus,

$$q_{in} = h_3 - h_5 = (1395.97 - 825.37)\ \text{kJ/kg} = 570.60\ \text{kJ/kg}$$

This represents a savings of 220.0 kJ/kg from the heat input requirements. The addition of a regenerator (assumed to be frictionless) does not affect the net work output. Thus,

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{210.41\ \text{kJ/kg}}{570.60\ \text{kJ/kg}} = 0.369 \quad \text{or} \quad 36.9\%$$

c.f 26.6%

Discussion Note that the thermal efficiency of the gas turbine has gone up from 26.6 to 36.9 percent as a result of installing a regenerator that helps to recover some of the thermal energy of the exhaust gases.

FIGURE 9–41
Thermal efficiency of the ideal Brayton cycle with and without regeneration.

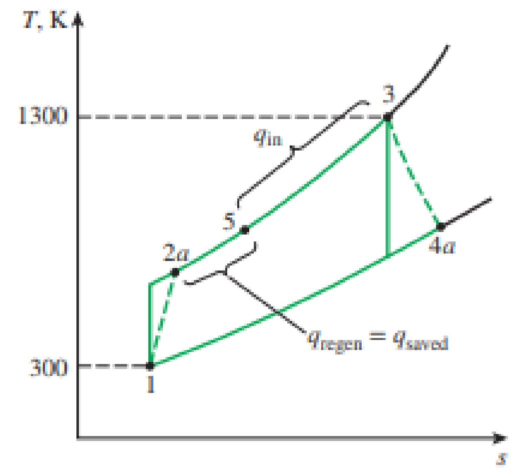
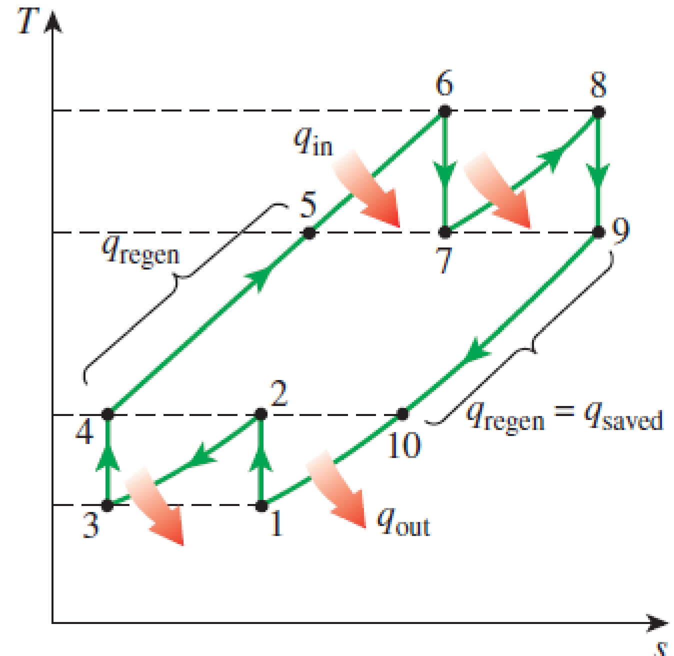


FIGURE 9–42
T-s diagram of the regenerative Brayton cycle described in Example 9–8.

9-10. THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

For minimizing work input to compressor and maximizing work output from turbine:

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9} \qquad (9\text{-}26)$$



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

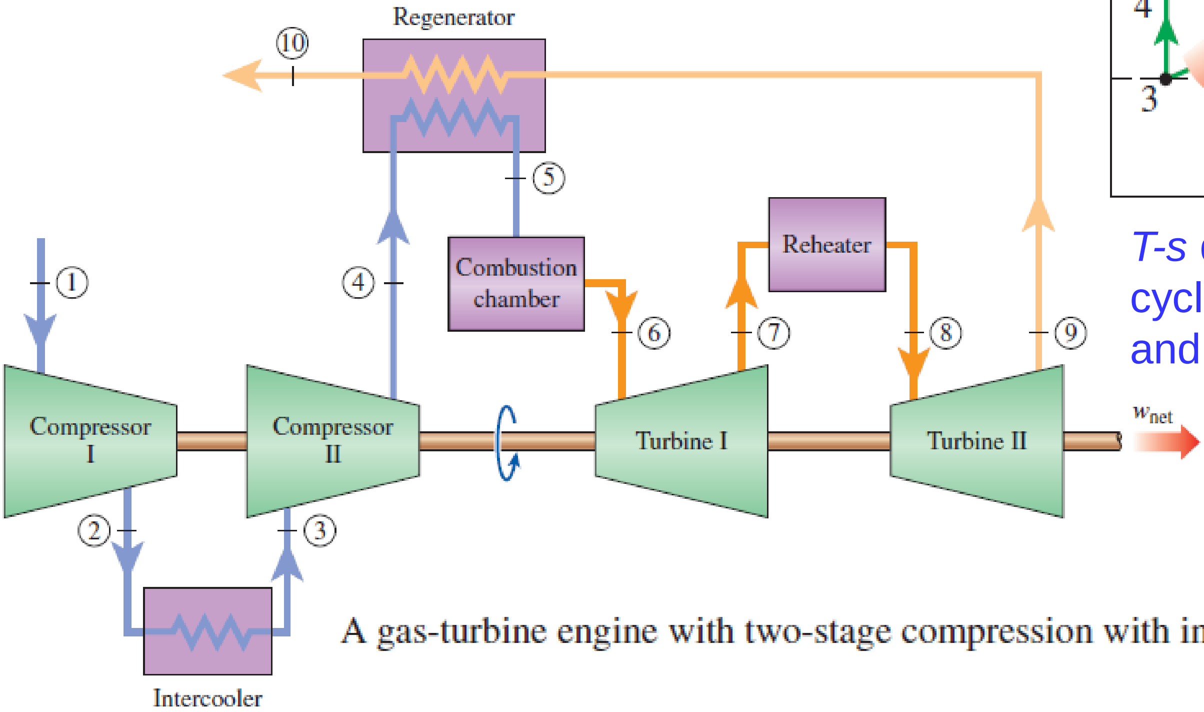


FIGURE 9–43

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

9-10. THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

Multistage compression with intercooling: The work required to compress a gas between two specified pressures can be decreased by carrying out the compression process in stages and cooling the gas in between. This keeps the specific volume as low as possible.

Multistage expansion with reheating keeps the specific volume of the working fluid as high as possible during an expansion process, thus maximizing work output.

Intercooling and reheating always decreases the thermal efficiency unless they are accompanied by regeneration.

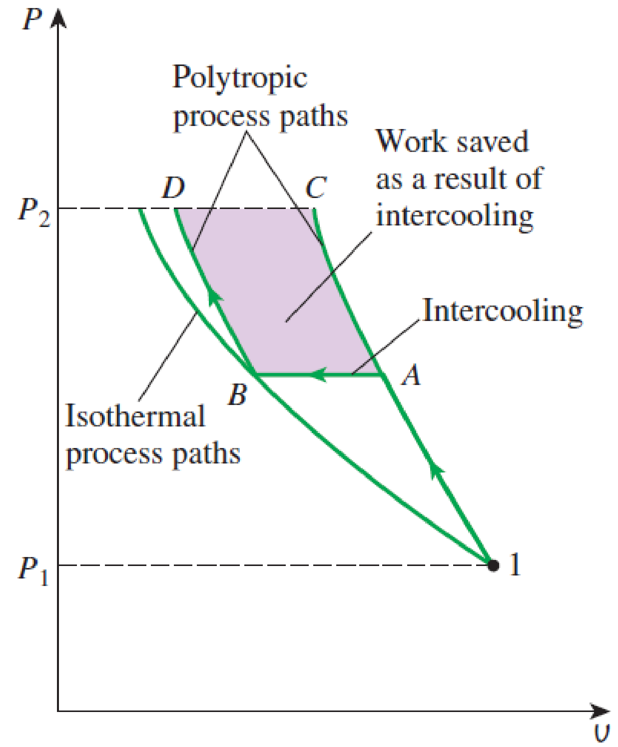


FIGURE 9–43
Comparison of work inputs to a single-stage compressor (1*AC*) and a two-stage compressor with intercooling (1*ABD*).

9-10. THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

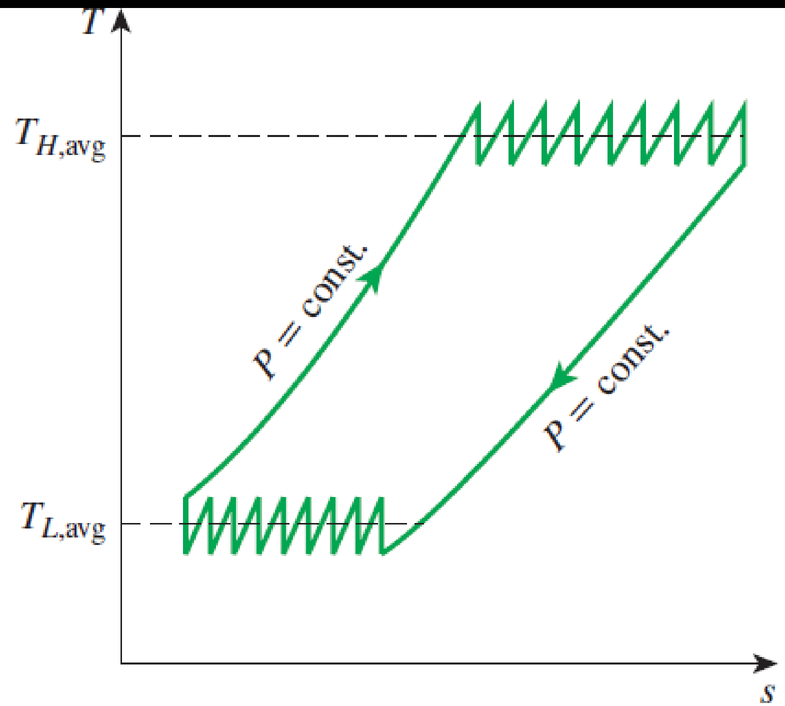


FIGURE 9–46

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.

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.

Aircraft are propelled by accelerating a fluid in the opposite direction to motion. This is accomplished by either slightly accelerating a large mass of fluid (***propeller-driven engine***) or greatly accelerating a small mass of fluid (***jet*** or ***turbojet engine***) or both (***turboprop engine***).

9-11. IDEAL JET-PROPULSION CYCLES

FIGURE 9–47
In jet engines, the high-temperature and high-pressure gases leaving the turbine are accelerated in a nozzle to provide thrust.

4ea Isentropic process
2ea Isobaric process

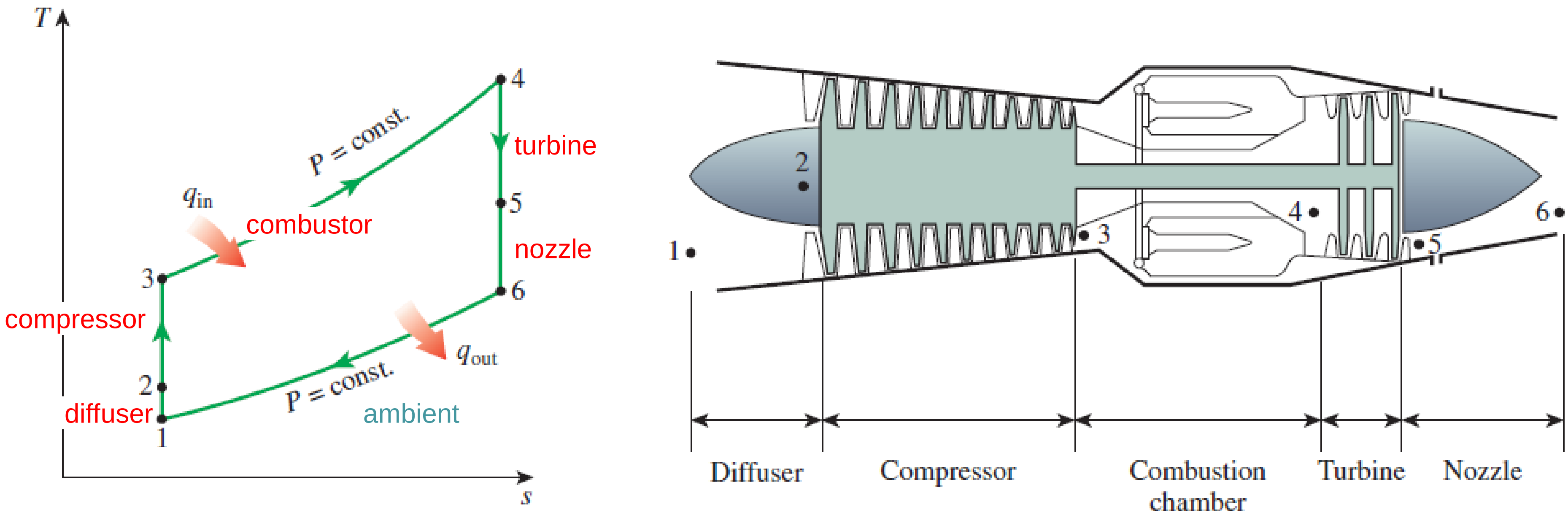


FIGURE 9–48
Basic components of a turbojet engine and the *T*-*s* diagram for the ideal turbojet cycle.

9-11. IDEAL JET-PROPULSION CYCLES

Thrust (propulsive force)

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

Propulsive power

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

Propulsive efficiency

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

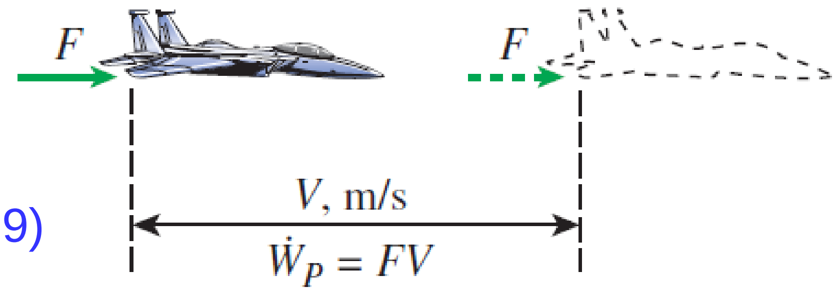


FIGURE 9–49
Propulsive power is the thrust acting on the aircraft through a distance per unit time.

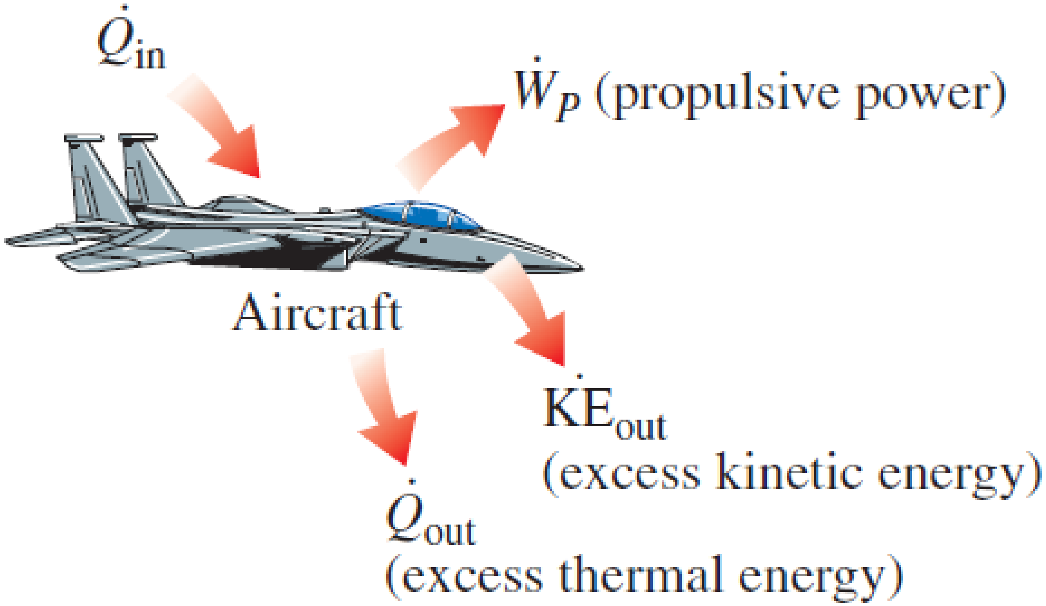


FIGURE 9–51

Energy supplied to an aircraft (from the burning of a fuel) manifests itself in various forms.

Modifications to Turbojet Engines

The first airplanes built were all propeller-driven, with propellers powered by engines essentially identical to automobile engines.

Both propeller-driven engines and jet-propulsion-driven engines have their own strengths and limitations, and several attempts have been made to combine the desirable characteristics of both in one engine.

Two such modifications are the ***propjet engine*** and the ***turbofan engine***.

The most widely used engine in aircraft propulsion is the **turbofan** (or *fanjet*) engine wherein a large fan driven by the turbine forces a considerable amount of air through a duct (cowl) surrounding the engine.

9-11. IDEAL JET-PROPULSION CYCLES

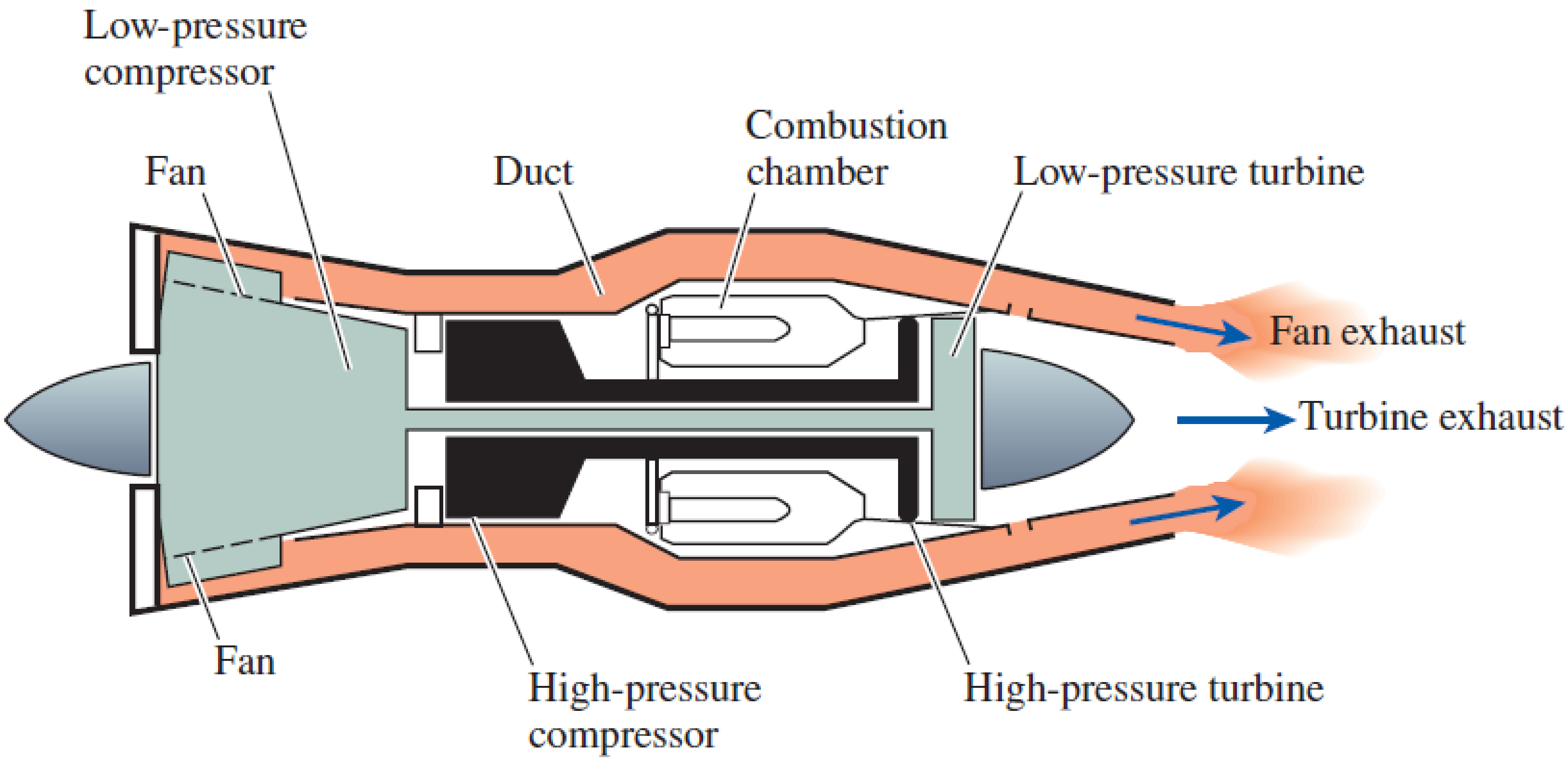


FIGURE 9–53

A turbofan engine.

9-11. IDEAL JET-PROPULSION CYCLES

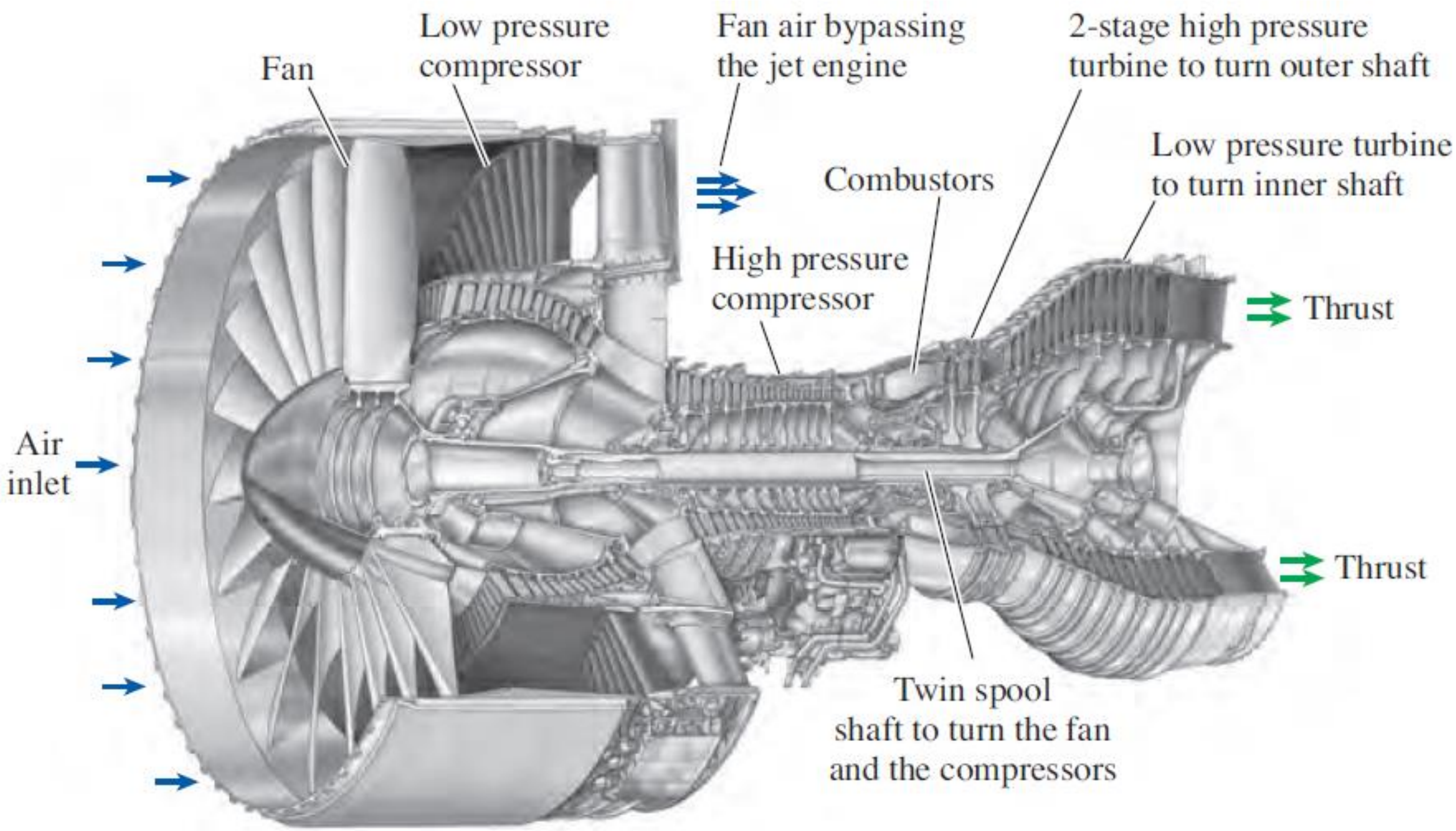


FIGURE 9–54

A modern jet engine used to power Boeing 777 aircraft. This is a Pratt & Whitney PW4084 turbofan capable of producing 375 kN of thrust. It is 4.87 m long, has a 2.84 m diameter fan, and it weighs 6800 kg.

9-11. IDEAL JET-PROPULSION CYCLES

Various engine types:

Turbofan, Propjet, Ramjet, Sacramjet, Rocket

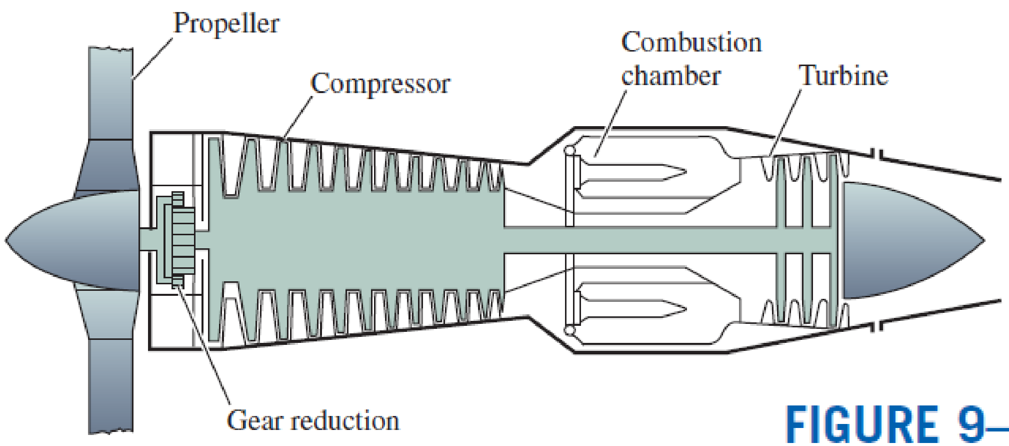


FIGURE 9–54

A turboprop engine.

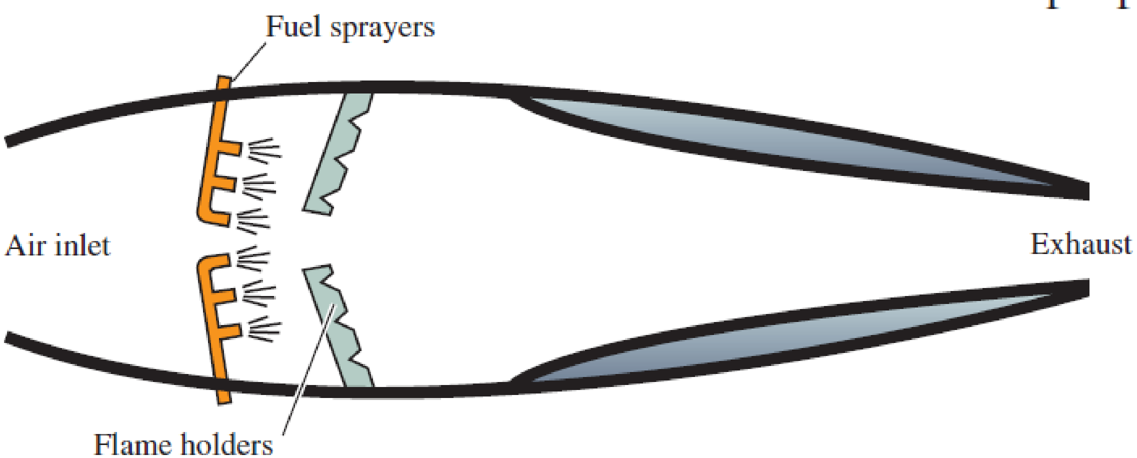


FIGURE 9–55

A ramjet engine.

Example 9-10, 이상적인 제트추진 사이클, 비열이 일정한 경우

EXAMPLE 9–10 **The Ideal Jet-Propulsion Cycle**

A turbojet aircraft flies with a velocity of 850 ft/s at an altitude where the air is at 5 psia and −40°F. The compressor has a pressure ratio of 10, and the temperature of the gases at the turbine inlet is 2000°F. Air enters the compressor at a rate of 100 lbm/s. Utilizing the cold-air-standard assumptions, determine (*a*) the temperature and pressure of the gases at the turbine exit, (*b*) the velocity of the gases at the nozzle exit, and (*c*) the propulsive efficiency of the cycle.

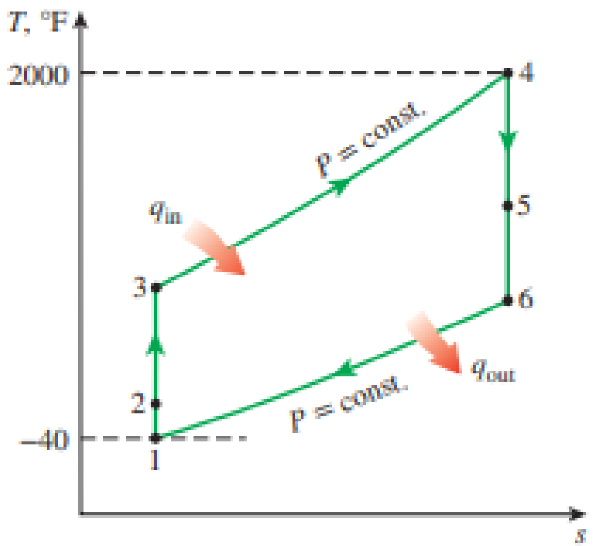


FIGURE 9–51
T-s diagram for the turbojet cycle described in Example 9–10.

SOLUTION The operating conditions of a turbojet aircraft are specified. The temperature and pressure at the turbine exit, the velocity of gases at the nozzle exit, and the propulsive efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The cold-air-standard assumptions are applicable, and thus air can be assumed to have constant specific heats at room temperature (c_p = 0.240 Btu/lbm·°F and k = 1.4). **3** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit. **4** The turbine work output is equal to the compressor work input.

Analysis The *T-s* diagram of the ideal jet propulsion cycle described is shown in Fig. 9–51. We note that the components involved in the jetpropulsion cycle are steady-flow devices.

(*a*) Before we can determine the temperature and pressure at the turbine exit, we need to find the temperatures and pressures at other states:

Process 1–2 (isentropic compression of an ideal gas in a diffuser): For convenience, we can assume that the aircraft is stationary and the air is moving toward the aircraft at a velocity of V_1 = 850 ft/s. Ideally, the air exits the diffuser with a negligible velocity ($V_2 \cong 0$):

$$h_2 + \frac{V_2^2}{2}^{\nearrow 0} = h_1 + \frac{V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - \frac{V_1^2}{2}$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p}$$

$$= 420\text{ R} + \frac{(850\text{ ft/s})^2}{2(0.240\text{ Btu/lbm·R})}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)$$

$$= 480\text{ R}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (5\text{ psia})\left(\frac{480\text{ R}}{420\text{ R}}\right)^{1.4/(1.4-1)} = 8.0\text{ psia}$$

Example 9–10, 이상적인 제트추진 사이클, 비열이 일정한

Process 2–3 (isentropic compression of an ideal gas in a compressor):

$$P_3 = (r_p)(P_2) = (10)(8.0\text{ psia}) = 80\text{ psia } (= P_4)$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (480\text{ R})(10)^{(1.4-1)/1.4} = 927\text{ R}$$

Process 4–5 (isentropic expansion of an ideal gas in a turbine): Neglecting the kinetic energy changes across the compressor and the turbine and assuming the turbine work to be equal to the compressor work, we find the temperature and pressure at the turbine exit to be

$$w_{\text{comp,in}} = w_{\text{turb,out}}$$

$$h_3 - h_2 = h_4 - h_5$$

$$c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

$$T_5 = T_4 - T_3 + T_2 = 2460 - 927 + 480 = \mathbf{2013\text{ R}}$$

$$P_5 = P_4\left(\frac{T_5}{T_4}\right)^{k/(k-1)} = (80\text{ psia})\left(\frac{2013\text{ R}}{2460\text{ R}}\right)^{1.4/(1.4-1)} = \mathbf{39.7\text{ psia}}$$

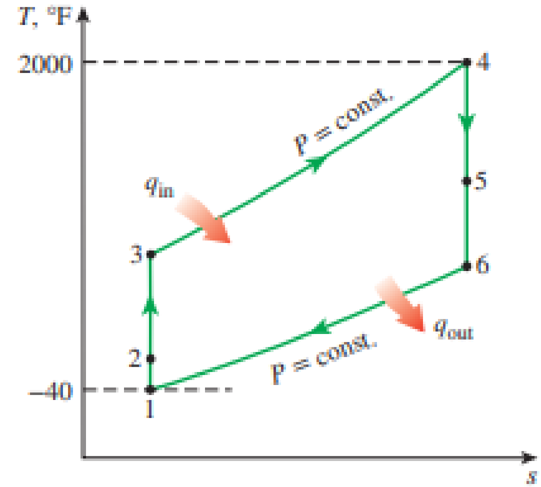


FIGURE 9–51
T-s diagram for the turbojet cycle described in Example 9–10.

(*b*) To find the air velocity at the nozzle exit, we need to first determine the nozzle exit temperature and then apply the steady-flow energy equation.

Process 5-6 (isentropic expansion of an ideal gas in a nozzle):

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (2013\text{ R})\left(\frac{5\text{ psia}}{39.7\text{ psia}}\right)^{(1.4-1)/1.4} = 1114\text{ R}$$

$$h_6 + \frac{V_6^2}{2} = h_5 + \frac{V_5^2}{2}^{\nearrow 0}$$

$$0 = c_p(T_6 - T_5) + \frac{V_6^2}{2}$$

$$V_6 = \sqrt{2c_p(T_5 - T_6)}$$

$$= \sqrt{2(0.240\text{ Btu/lbm·R})[(2013 - 1114)\text{R}]\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)}$$

$$= \mathbf{3288\text{ ft/s}}$$

(*c*) The propulsive efficiency of a turbojet engine is the ratio of the propulsive power developed $\dot{W}_P$ to the total heat transfer rate to the working fluid:

$$\dot{W}_P = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$

$$= (100\text{ lbm/s})[(3288 - 850)\text{ft/s}](850\text{ ft/s})\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)$$

$$= 8276\text{ Btu/s} \quad (\text{or } 11{,}707\text{ hp})$$

$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3)$$

$$= (100\text{ lbm/s})(0.240\text{ Btu/lbm·R})[(2460 - 927)\text{R}]$$

$$= 36{,}794\text{ Btu/s}$$

$$\eta_P = \frac{\dot{W}_P}{\dot{Q}_{\text{in}}} = \frac{8276\text{ Btu/s}}{36{,}794\text{ Btu/s}} = 0.255 \text{ or } \mathbf{22.5\%}$$

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

Exergy destruction for a closed system (9-30)

$$\dot{X}_{dest} = T_0\dot{S}_{gen} = T_0(\dot{S}_{out} - \dot{S}_{in}) = T_0\left(\sum_{out}\dot{m}s - \sum_{in}\dot{m}s - \frac{\dot{Q}_{in}}{T_{b,in}} + \frac{\dot{Q}_{out}}{T_{b,out}}\right) \quad \text{(kW)}$$ (9-31)

For a steady-flow system

$$x_{dest} = T_0 s_{gen} = T_0\left(s_e - s_i - \frac{q_{in}}{T_{b,in}} + \frac{q_{out}}{T_{b,out}}\right) \quad \text{(kJ/kg)}$$

Steady-flow, one-inlet, one-exit (9-32)

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

Exergy destruction of a cycle (9-33)

9-12. SECOND-LAW ANALYSIS OF GAS POWER CYCLES

$$x_{dest} = T_0\left(\frac{q_{out}}{T_L} - \frac{q_{in}}{T_H}\right) \quad \text{(kJ/kg)}$$

For a cycle with heat transfer only with a source and a sink

$$\phi = (u - u_0) - T_0(s - s_0) + P_0(\upsilon - \upsilon_0) + \frac{V^2}{2} + gz \quad \text{(kJ/kg)}$$

(9-35)

Closed system exergy

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

Stream exergy

(9-36)

A second-law analysis of these cycles reveals where the largest irreversibilities occur and where to start improvements.

Summary

- Basic considerations in the analysis of power cycles
- The Carnot cycle and its value in engineering
- Air-standard assumptions
- An overview of reciprocating engines
- Otto cycle: The ideal cycle for spark-ignition engines
- Diesel cycle: The ideal cycle for compression-ignition engines
- Stirling and Ericsson cycles
- Brayton cycle: The ideal cycle for gas-turbine engines
- The Brayton cycle with regeneration
- The Brayton cycle with intercooling, reheating, and regeneration
- Ideal jet-propulsion cycles
- Second-law analysis of gas power cycles