Thermodynamics: An Engineering Approach
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CHAPTER 9
GAS POWER CYCLES

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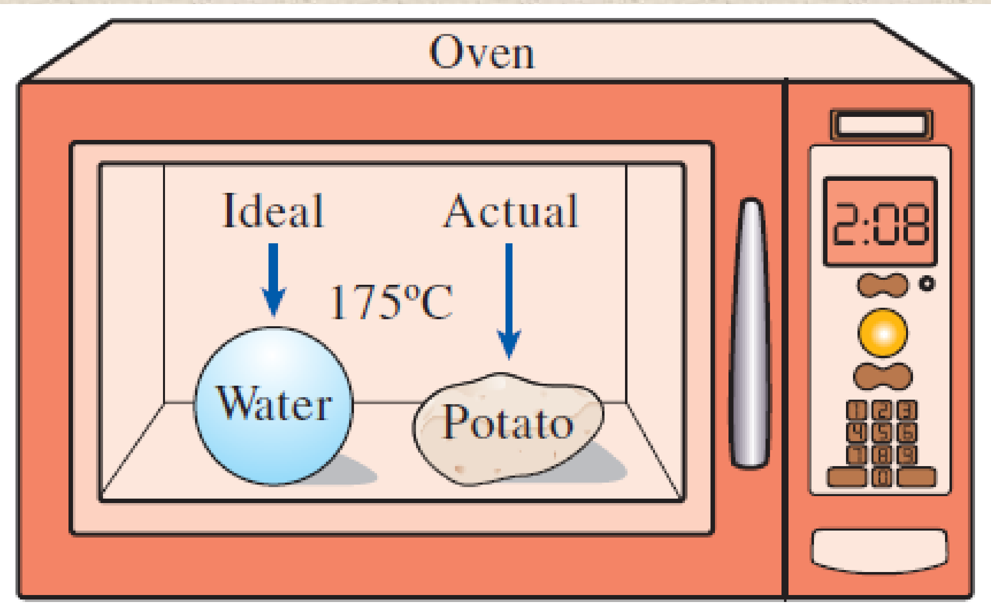
Objectives

- 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
- 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
- Identify simplifying assumptions for second-law analysis of gas power cycles
- Perform second-law analysis of gas power cycles

BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

- Most power-producing devices operate on cycles
- The cycles encountered in *actual devices* are difficult to analyze because of the presence of complicating effects, such as *friction*, and the *absence of sufficient time for establishment of the equilibrium*
- ***Ideal cycle:*** When the actual cycles are stripped of all the internal irreversibilities and complexities, we end up with a cycle that *resembles the actual cycle* closely but is made up entirely of ***internally reversible processes***. Such a cycle is called an ***ideal cycle***
- A simple idealized model enables engineers to study the effects of the major parameters that dominate the cycle without getting bogged down in the details



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

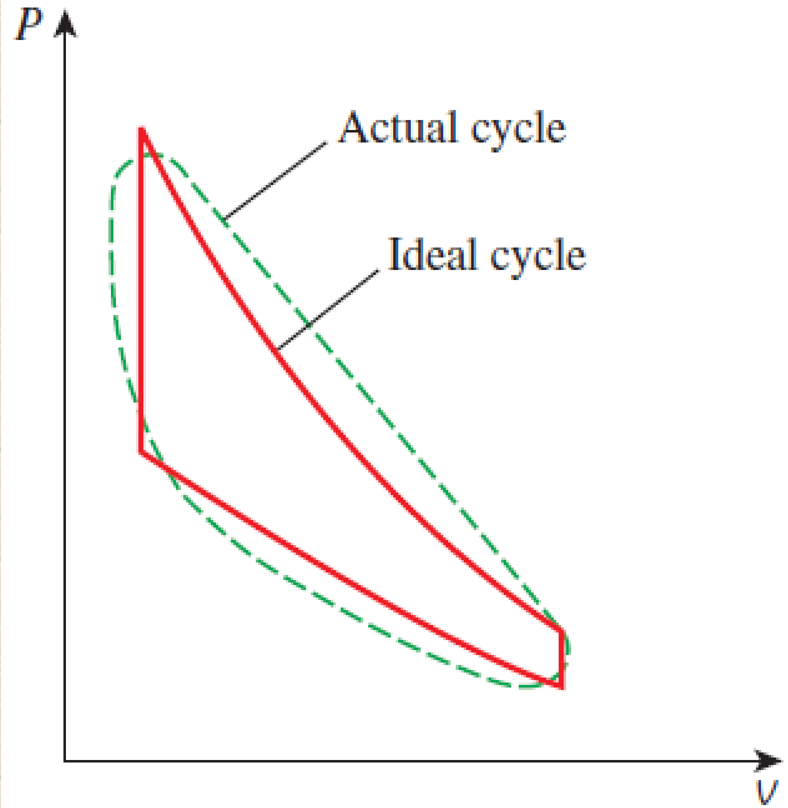


FIGURE 9–2
The analysis of many complex processes can be reduced to a manageable level by utilizing some idealizations.

BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

- *Heat engines* are designed for the purpose of *converting thermal energy to work*, and their performance is expressed in terms of the ***thermal efficiency***:

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

Thermal efficiency of heat engines:

- Heat engines that operate on a totally **reversible cycles** such as **Carnot cycle** have the highest thermal efficiency of all heat engines operating between the same temperature levels

- The **ideal cycles** are *internally reversible*, but, unlike the Carnot cycle, they are *not necessarily externally reversible*
- That is, ideal cycles *may involve irreversibilities external to the system* such as heat transfer through a finite temperature difference

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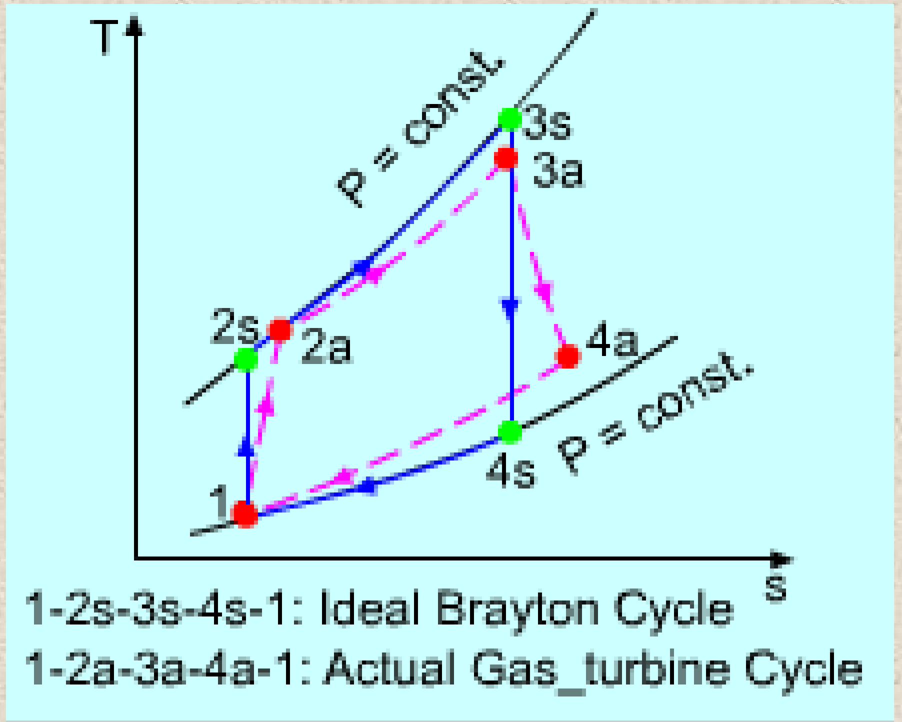


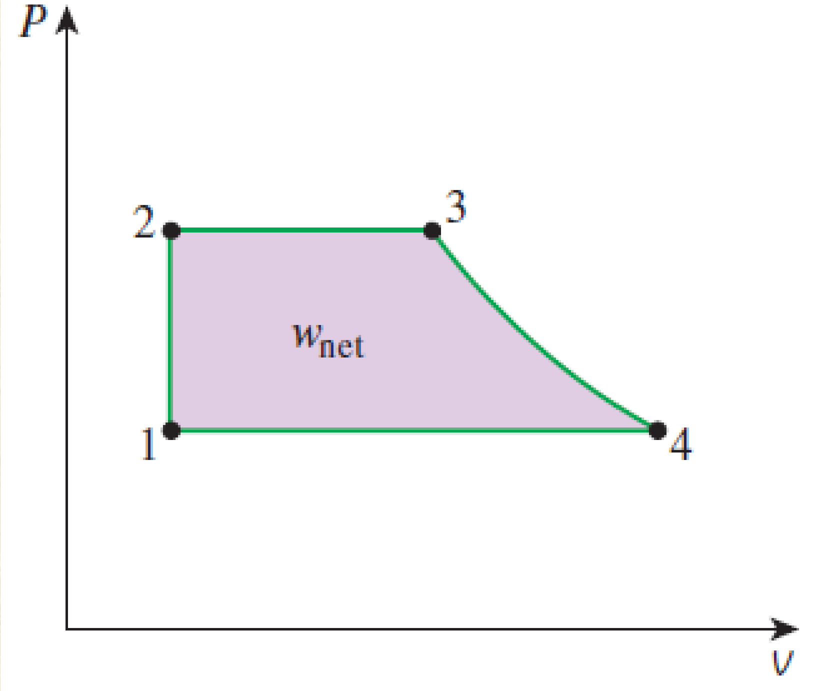
- Therefore, the thermal efficiency of an ideal cycle, in general, is less than that of a totally reversible cycle operating between the same temperature limits
- However, it is still considerably higher than the thermal efficiency of an actual cycle because of the idealizations utilized

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

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.

- *Neglecting the changes in kinetic and potential energies of the working fluid* is another commonly utilized simplification in the analysis of power cycles



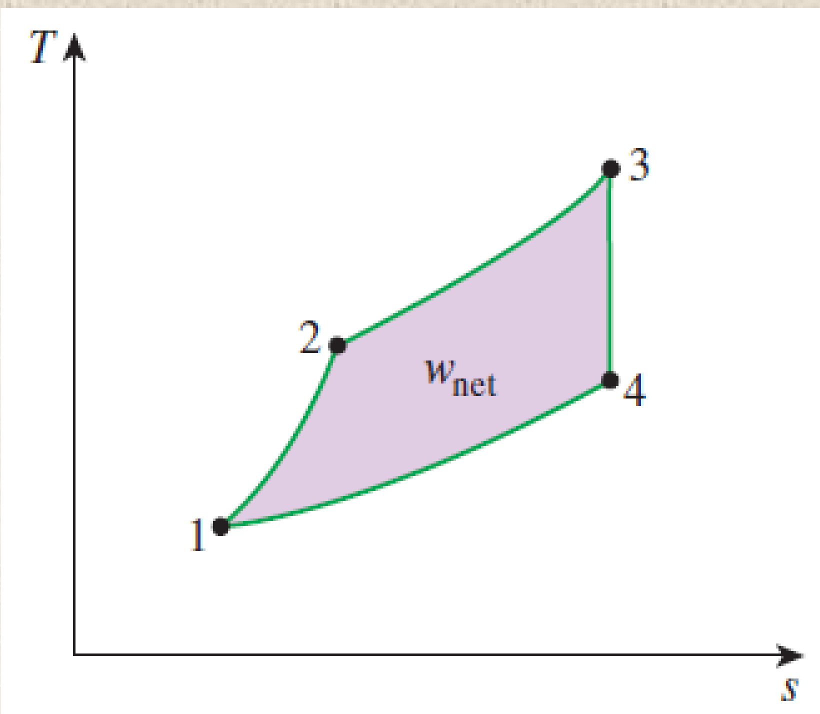


FIGURE 9–4

On both *P*-*v* and *T*-*s* diagrams, the area enclosed by the process curve represents the net work of the cycle. which is also equivalent to the net heat transfer for that cycle

On a *T*-*s* diagram, a *heat-addition* process proceeds in the direction of *increasing entropy*, since all the processes are *internally reversible*. Similarly a heat-rejection process proceeds in the direction of decreasing entropy.

The *area under the process curve* on the T-s diagram represents the *heat transfer* for that process

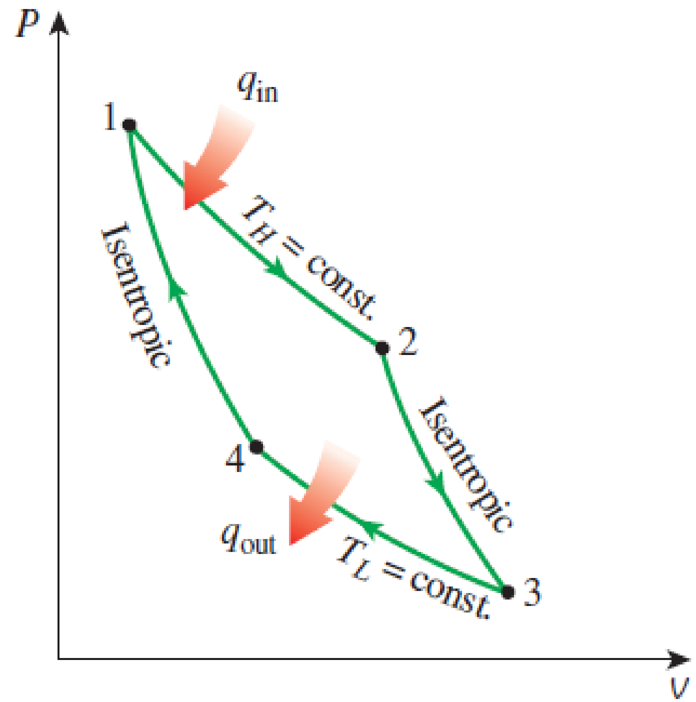
On a T-s diagram, the ratio of the *area enclosed by the cyclic curve* to the *area under the heat-addition process curve* represents the *thermal efficiency* of the cycle.

THE CARNOT CYCLE

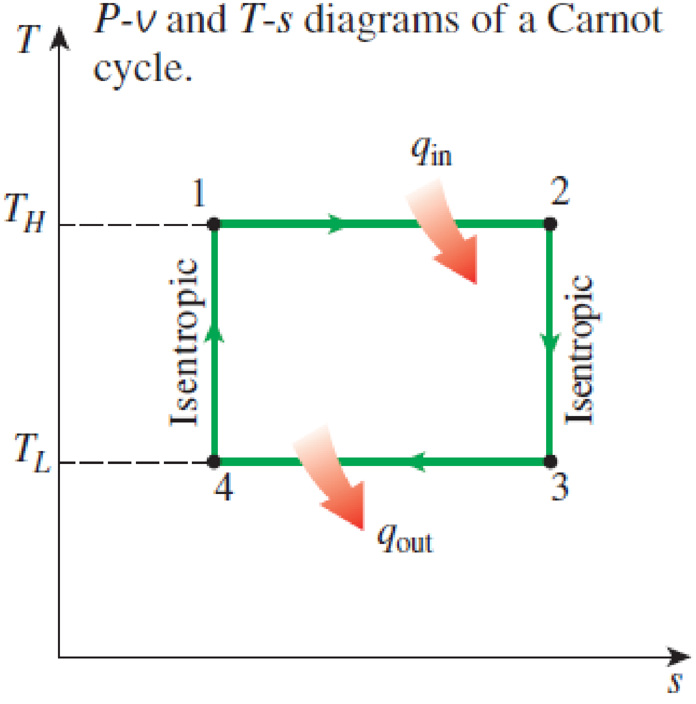
- The Carnot cycle is composed of four totally reversible processes: isothermal heat addition, isentropic expansion, isothermal heat rejection, and isentropic compression

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

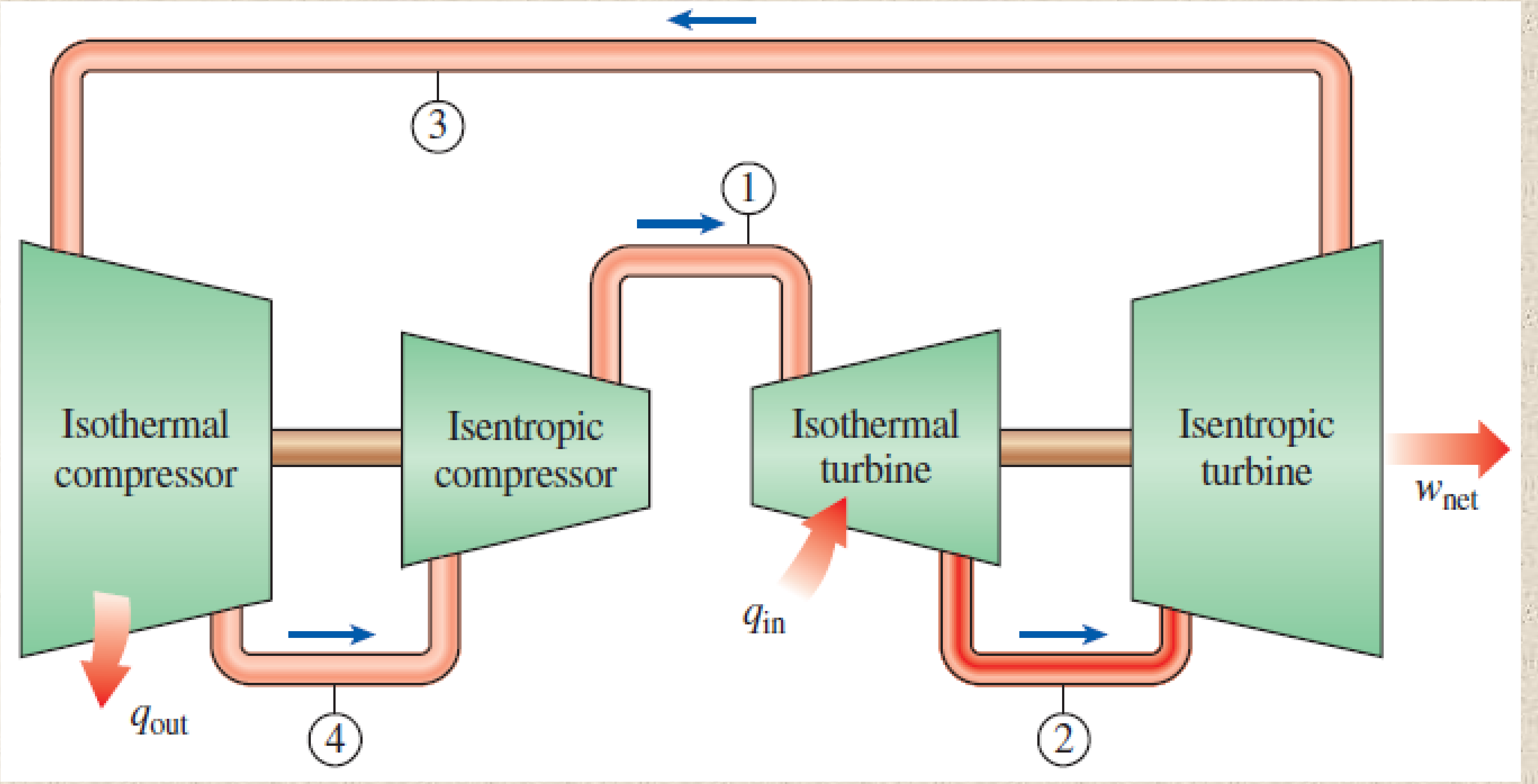
- Reversible isothermal heat transfer is very difficult to achieve in reality because it would require very large heat exchangers and it would take very long time.
- Therefore, *it is not practical to build an engine that would operate on a cycle that closely approximates the Carnot cycle*



P-*v* and *T*-*s* diagrams of a Carnot cycle.



A steady-flow Carnot engine



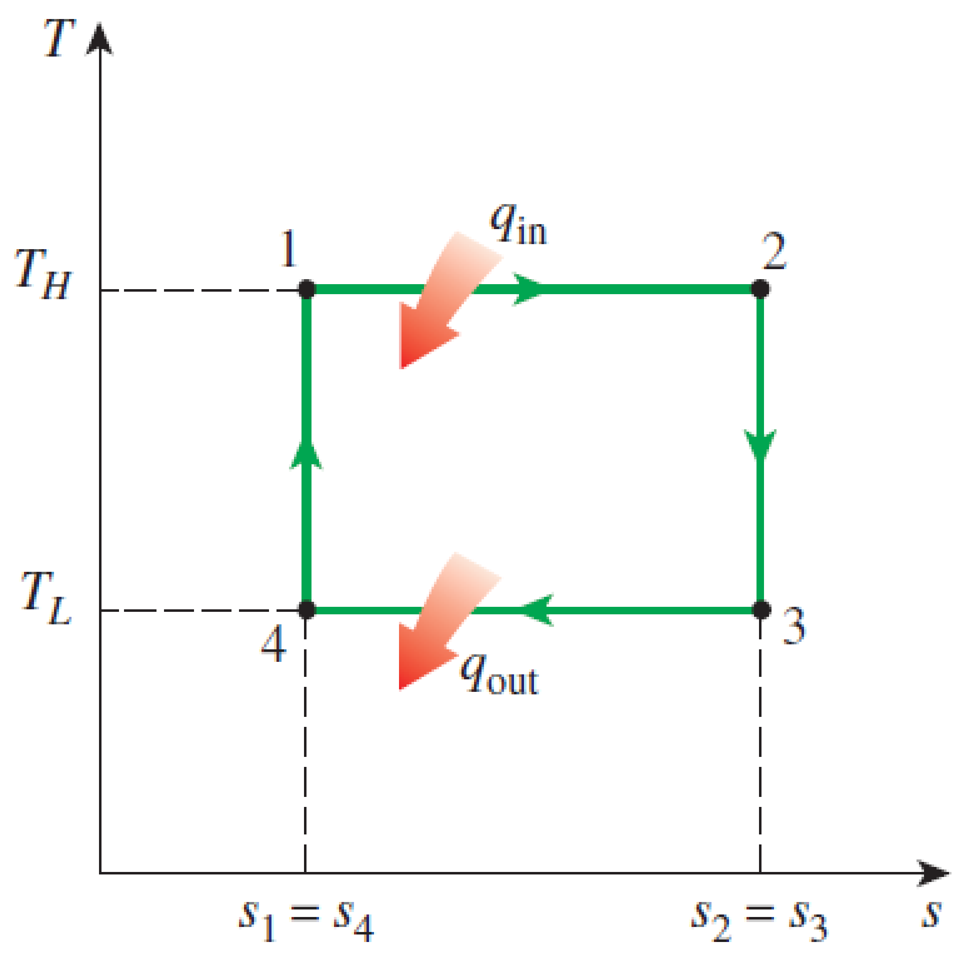
- The Carnot cycle can be executed in a closed system (a piston-cylinder device) or a steady-flow system (utilizing two turbines and two compressors), and either a gas or a vapor can be utilized as the working fluid

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 $s_4 = s_1$



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

- The thermal efficiency of a Carnot cycle is independent of the type of the working fluid used (an ideal gas, steam, etc…) or whether the cycle is executed in a closed or steady flow system

AIR-STANDARD ASSUMPTIONS

- *In gas power cycles, the working fluid remains a gas throughout the engine cycle*. Spark-ignition engines, diesel engines, and conventional gas turbines are familiar examples of devices that operate on gas cycles
- In all these engines, *energy is provided by burning a fuel* within the system boundaries. Because of this combustion process, the *composition of the working fluid changes* from air and fuel to combustion products during the course of the cycle
- However, considering that *air is predominantly nitrogen* that hardly undergoes a any chemical reactions, *the working fluid closely resembles air at all times*.

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

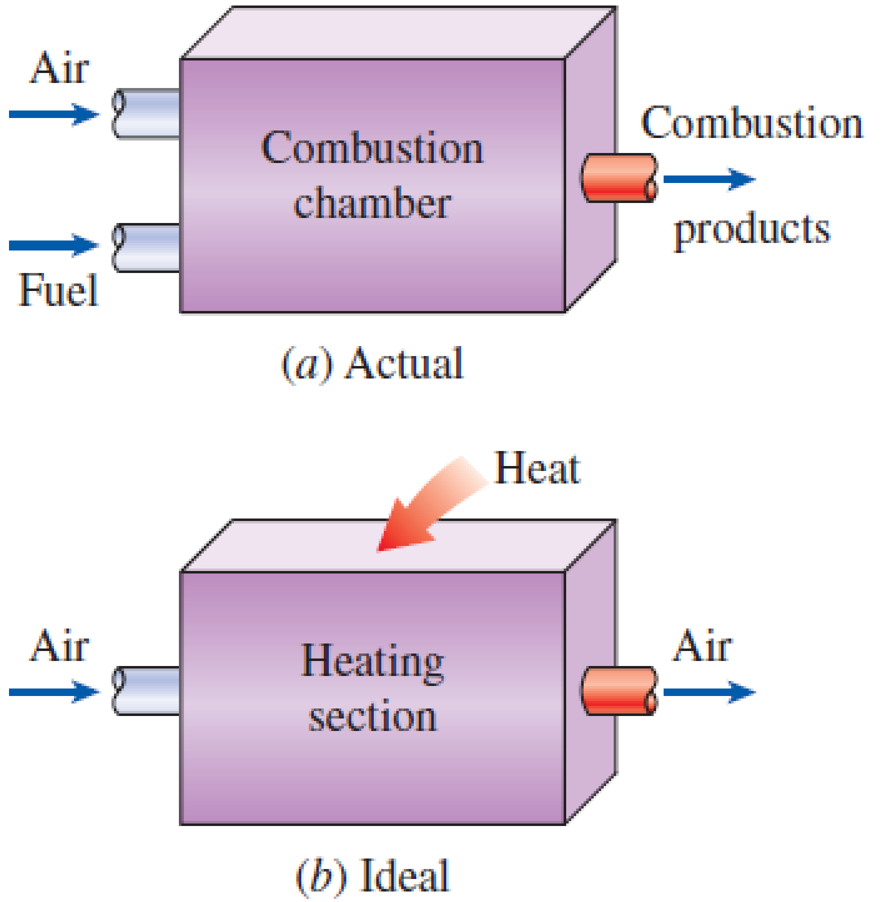


FIGURE 9–8

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

AN OVERVIEW OF RECIPROCATING ENGINES

- **Top dead center (TDC)** – the position of the piston when it forms the smallest volume in the cylinder
- **Bottom dead center (BDC)** – the position of the piston when it forms the largest volume in the cylinder
- The distance between the TDC and BDC is called the **stroke** of the engine
- The diameter of the piston is called the **bore**

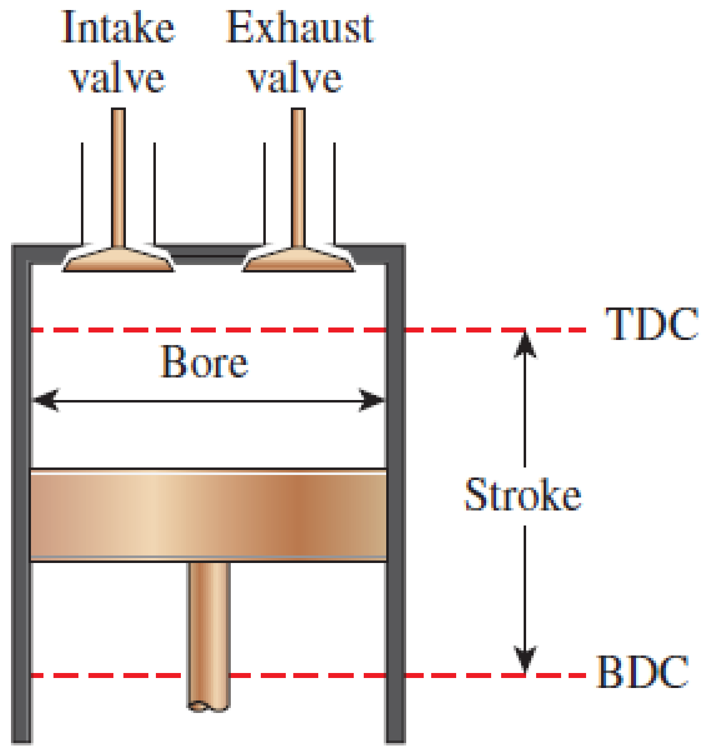


FIGURE 9–9
Nomenclature for reciprocating engines.

- 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**

AN OVERVIEW OF RECIPROCATING ENGINES

- The minimum volume formed in the cylinder when the piston is at TDC is called the **clearance volume**
- The volume displaced by the piston as it moves between TDC and BDC is called the **displacement volume**
- The ratio of maximum volume formed in the cylinder to the minimum (clearance) volume is called the **compression ratio *r*** of the engine

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

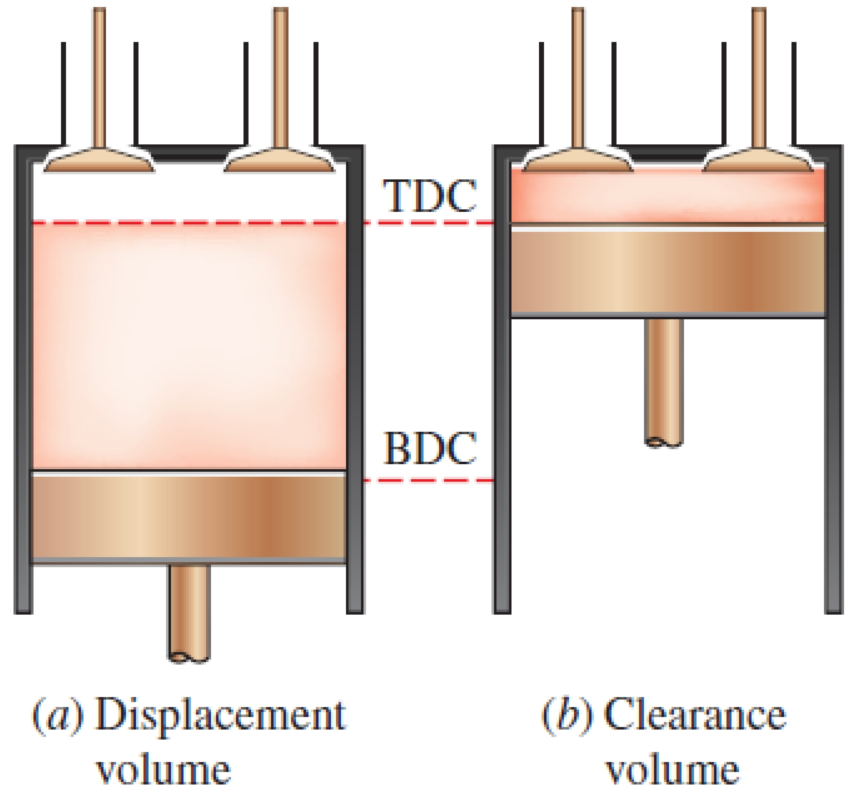
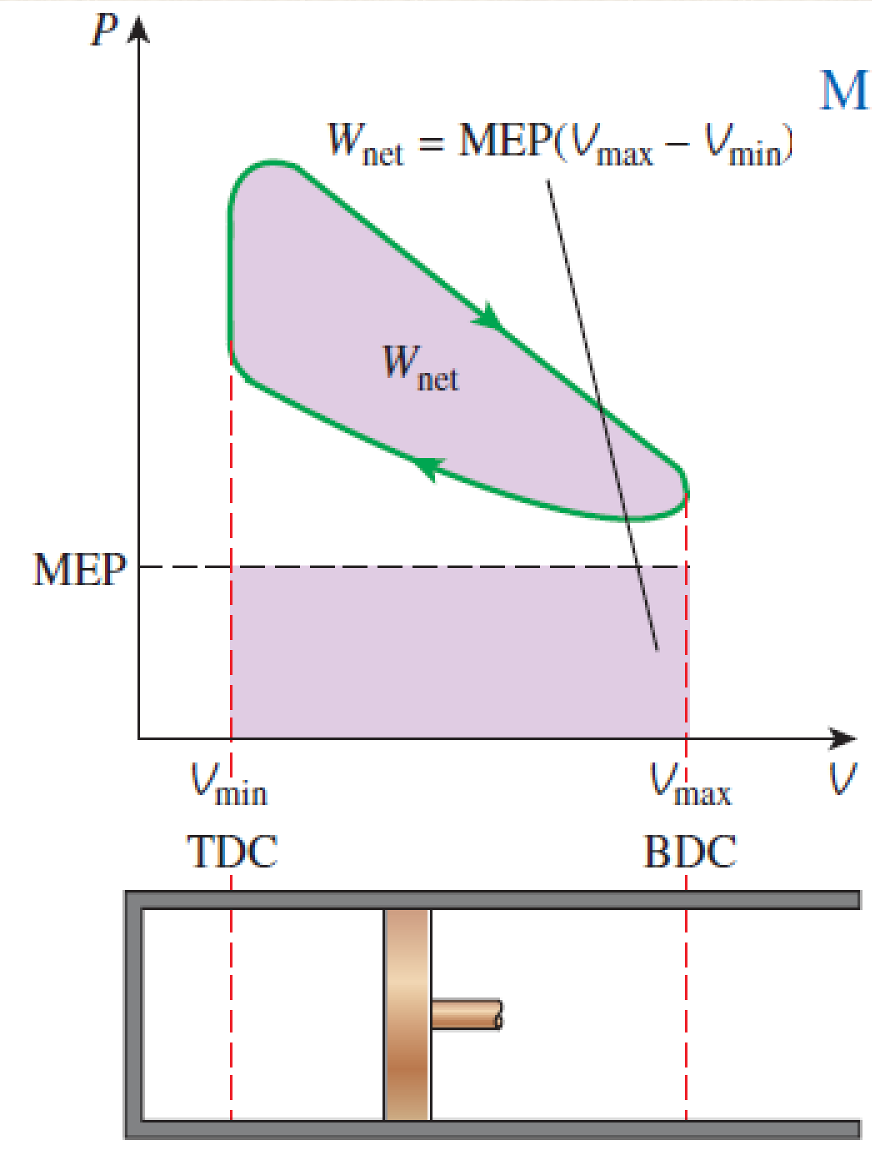


FIGURE 9–10
Displacement and clearance volumes of a reciprocating engine.

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

Mean effective pressure

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



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

Mean effective pressure (MEP) is a fictitious pressure that, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle

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

OTTO CYCLE: THE IDEAL CYCLE FOR SPARK-IGNITION ENGINES

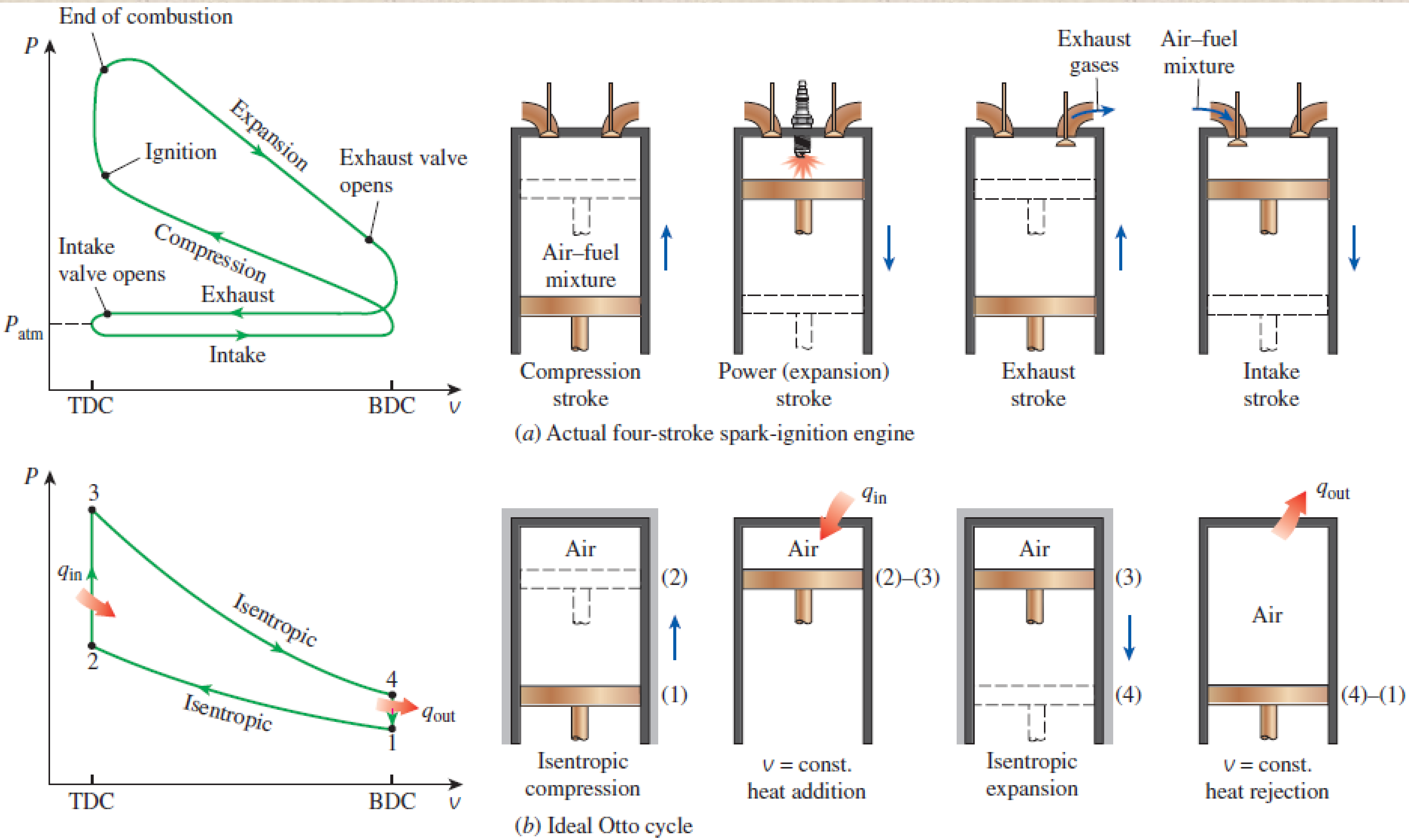


FIGURE 9–12

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

Four-stroke cycle

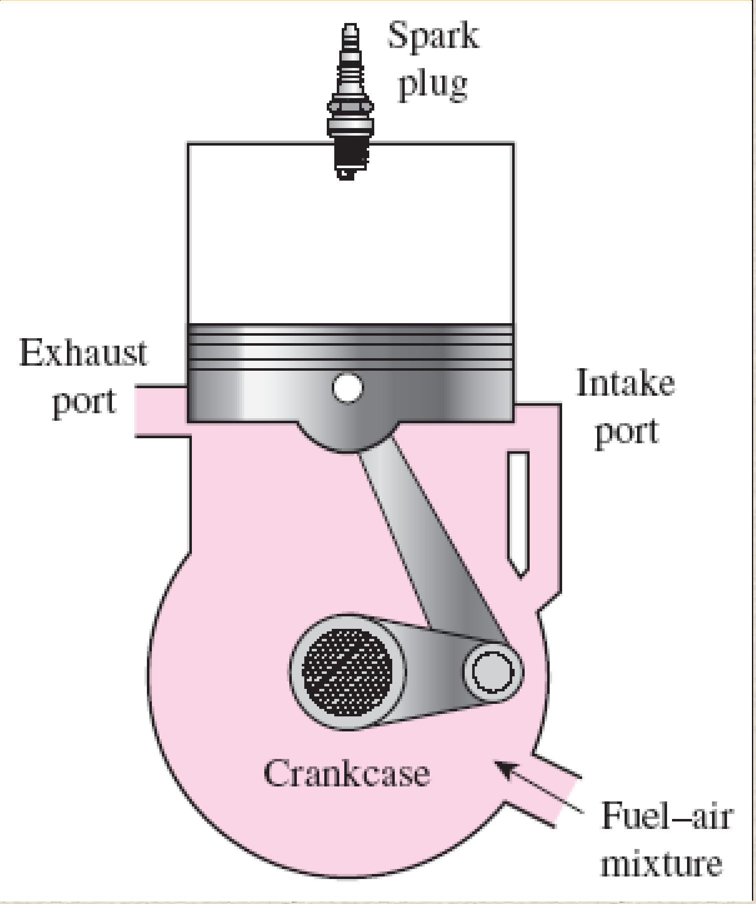
1 cycle = 4 stroke = 2 revolution

Two-stroke cycle

1 cycle = 2 stroke = 1 revolution

- In two stroke engines, during later part of the **power stroke**, the piston uncovers first the exhaust port, allowing the exhaust gases to be partially expelled, and then the intake port, allowing the fresh air-fuel mixture to rush in and drive most of the remaining exhaust gases out of the cylinder
- This mixture is then compressed as the piston moves upward during the **compression stroke** and is subsequently ignited by the spark plug

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.



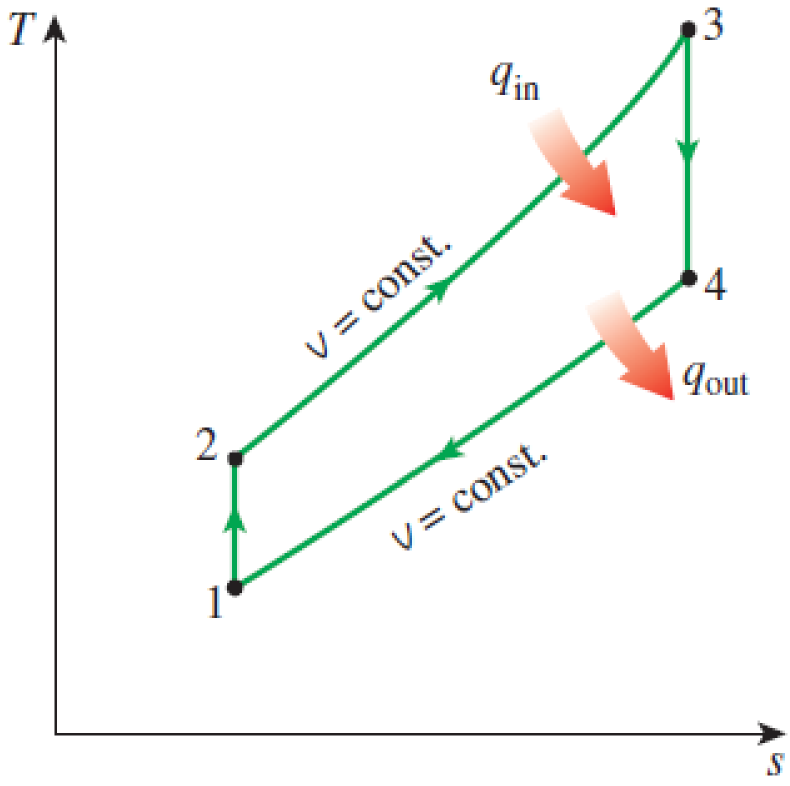
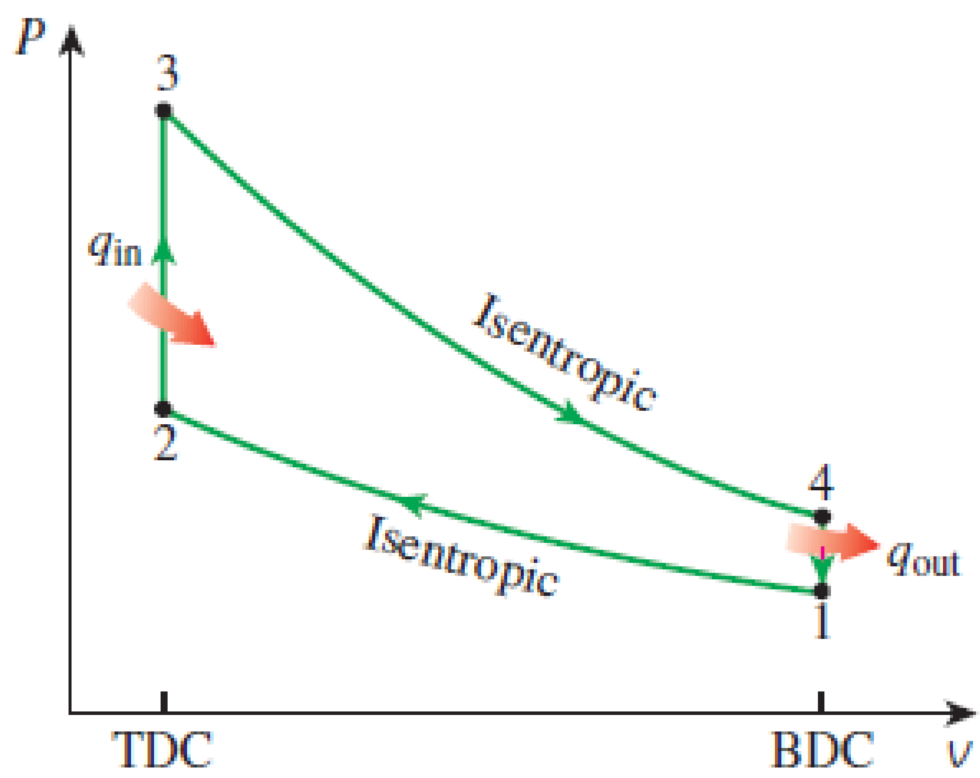


FIGURE 9–15
T-s diagram of the ideal Otto cycle.



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

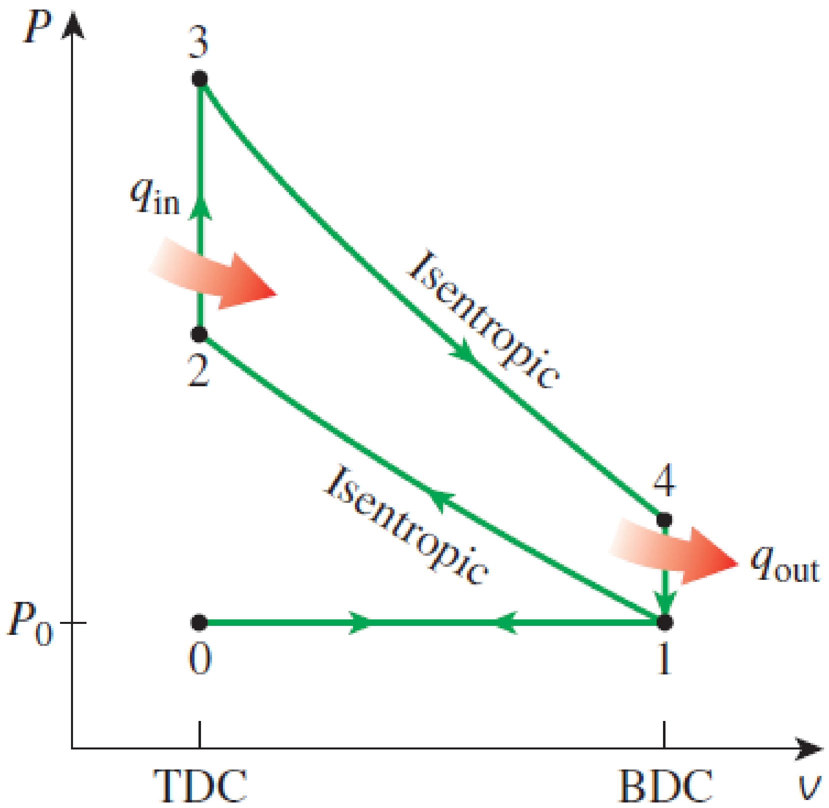


FIGURE 9–16

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

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

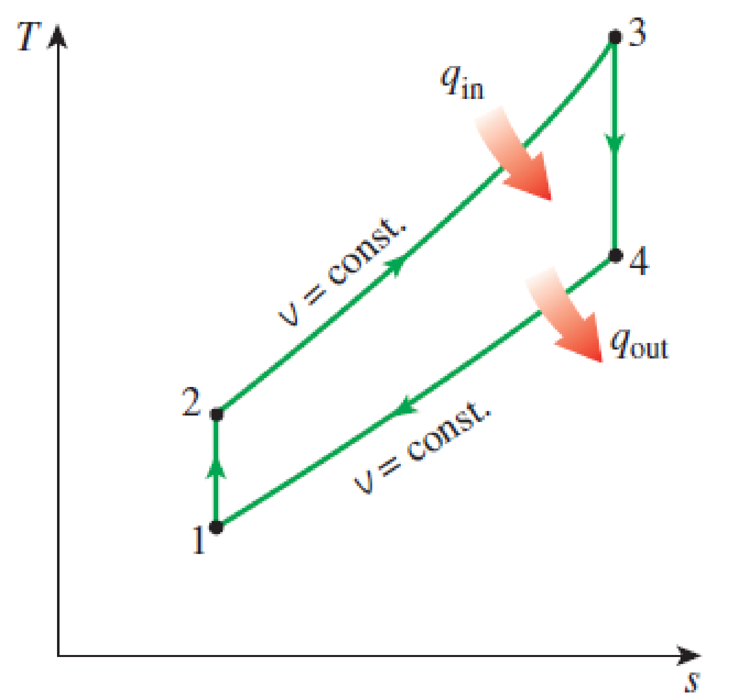
The intake valve is closed at state 1 and air is compressed isentropically to state 2. Heat is transferred at constant volume (process 2-3); it is expanded isentropically to state 4; and heat is rejected at constant volume (process 4-1).

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

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

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

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.



The Otto cycle is executed in a closed system, and disregarding the changes in kinetic and potential energies, the energy balance for any of the processes is expressed, on a unit-mass basis:

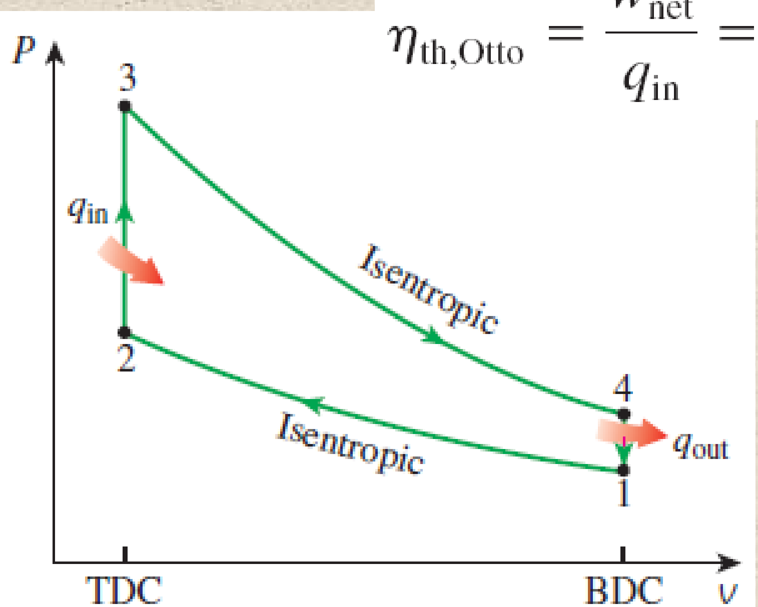
$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u$$

Process (2-3) and (4-1) are constant volume:
(for constant specific heats)

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

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

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



Process (1-2) and (3-4) are isentropic:
(for constant specific heats)

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

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

Compression ratio (r):

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

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

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

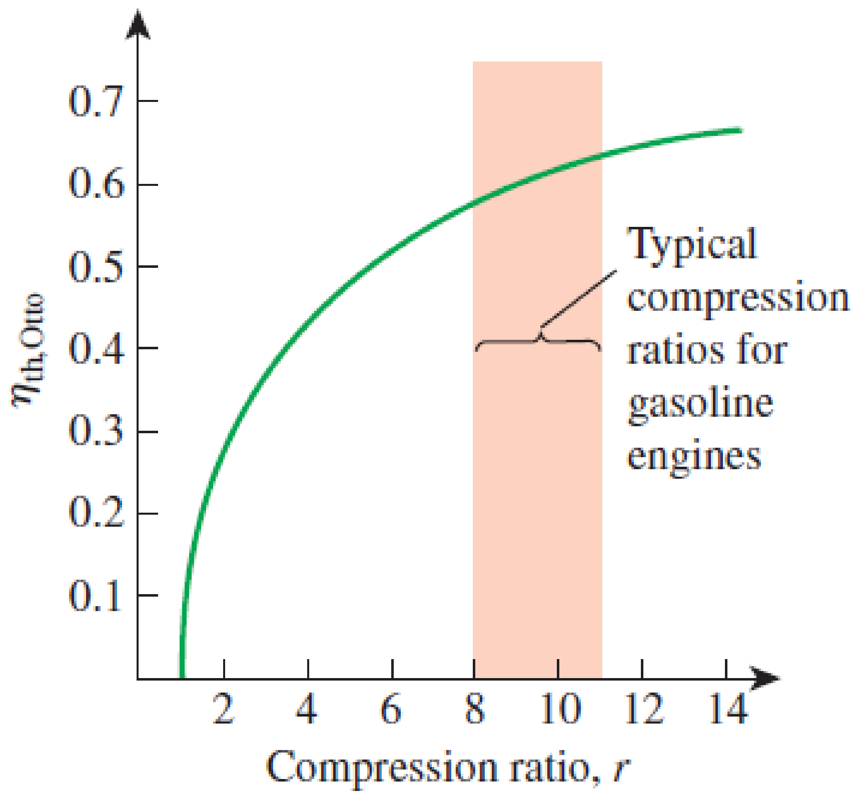


FIGURE 9–17

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

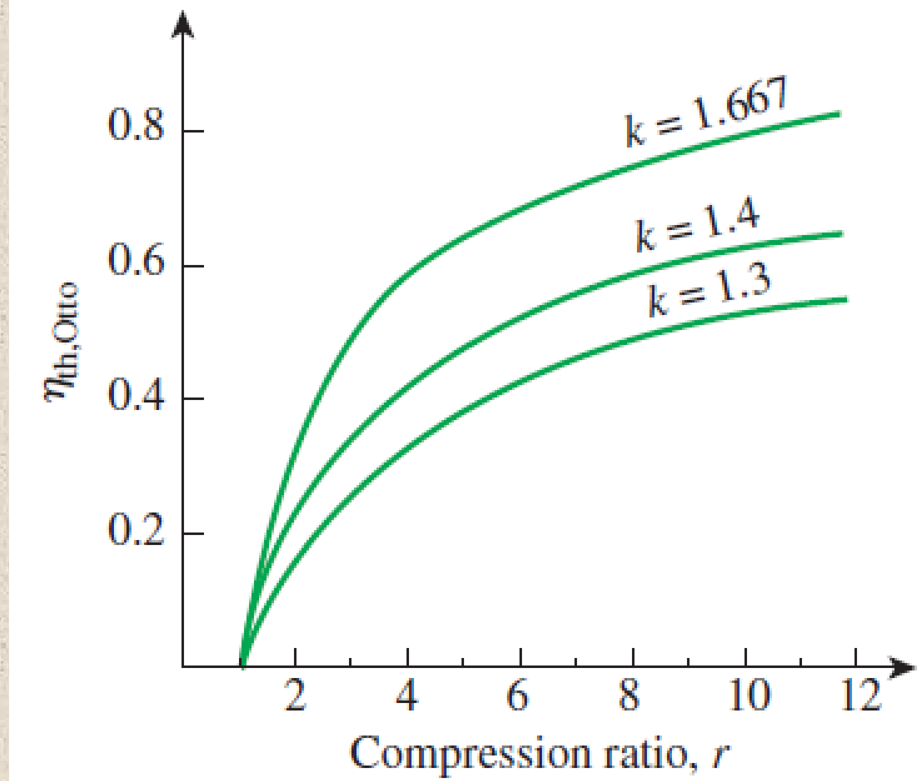


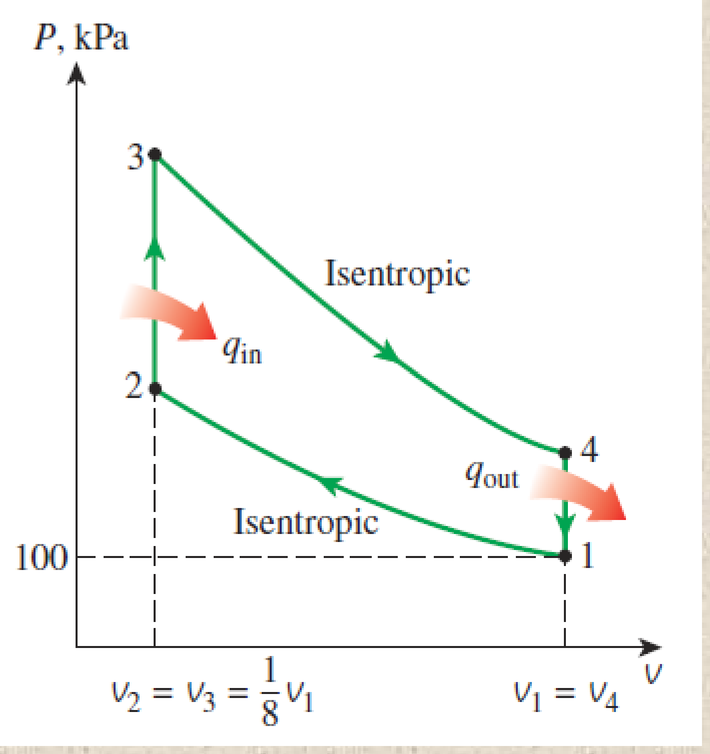
FIGURE 9–18

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

Thermal efficiencies of actual spark-ignition engines range from about 25 to 30 percent

EXAMPLE: 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



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

Table A–17

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

Considering variation of specific heats of air with temperature:

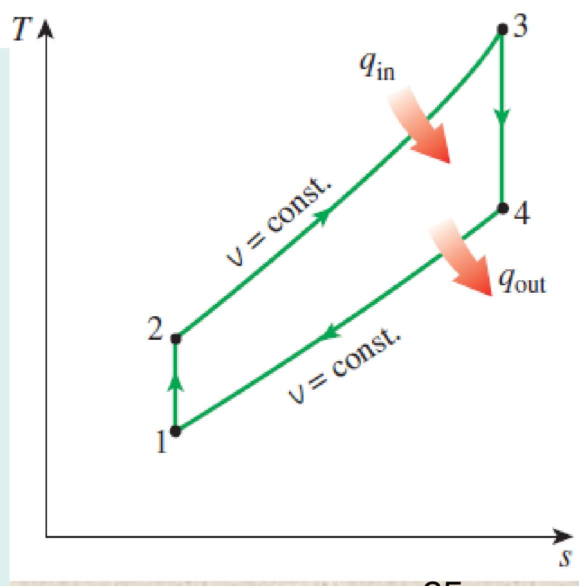
$$\frac{v_{r2}}{v_{r1}} = \frac{v_2}{v_1} = \frac{1}{r} \rightarrow v_{r2} = \frac{v_{r1}}{r} = \frac{676.1}{8} = 84.51 \rightarrow T_2 = 652.4 \text{ K}$$

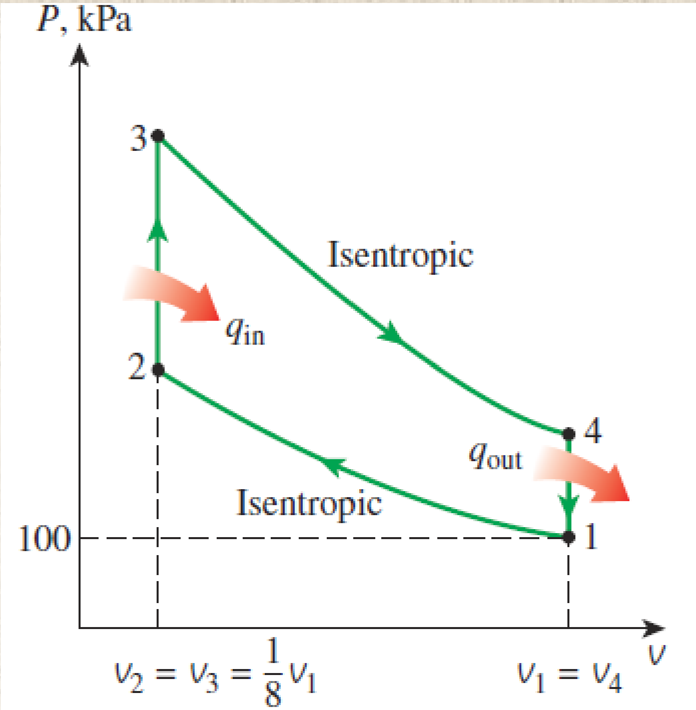
$$u_2 = 475.11 \text{ kJ/kg}$$

Use ideal gas law:

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \rightarrow P_2 = P_1\left(\frac{T_2}{T_1}\right)\left(\frac{v_1}{v_2}\right)$$

$$= (100 \text{ kPa})\left(\frac{652.4 \text{ K}}{290 \text{ K}}\right)(8) = 1799.7 \text{ kPa}$$





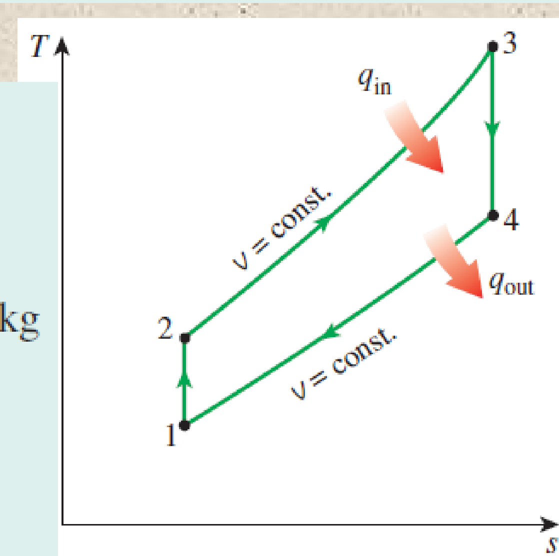
Process 2-3 (constant-volume heat addition):

Use first law:

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

Use ideal gas law:

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \rightarrow P_3 = P_2\left(\frac{T_3}{T_2}\right)\left(\frac{v_2}{v_3}\right)$$
$$= (1.7997 \text{ MPa})\left(\frac{1575.1 \text{ K}}{652.4 \text{ K}}\right)(1) = \mathbf{4.345\ MPa}$$



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

Considering variation of specific heats of air with temperature:

$$\frac{v_{r4}}{v_{r3}} = \frac{v_4}{v_3} = r \rightarrow v_{r4} = r v_{r3} = (8)(6.108) = 48.864 \rightarrow T_4 = 795.6 \text{ K}$$
$$u_4 = 588.74 \text{ kJ/kg}$$

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

Use first law: $-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1$

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

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

Thus,

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

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

- In diesel engines, *only air is compressed* to a temperature that is above the autoignition temperature of the fuel, *and combustion starts as the fuel is injected into this hot air*
- Therefore, diesel engines can be designed to *operate at much higher compression ratios* than SI engines, typically between 12 and 24
- *Combustion process* in diesel engines takes place over a longer interval, hence, it is approximated as a *constant-pressure heat addition* process in the ideal cycle

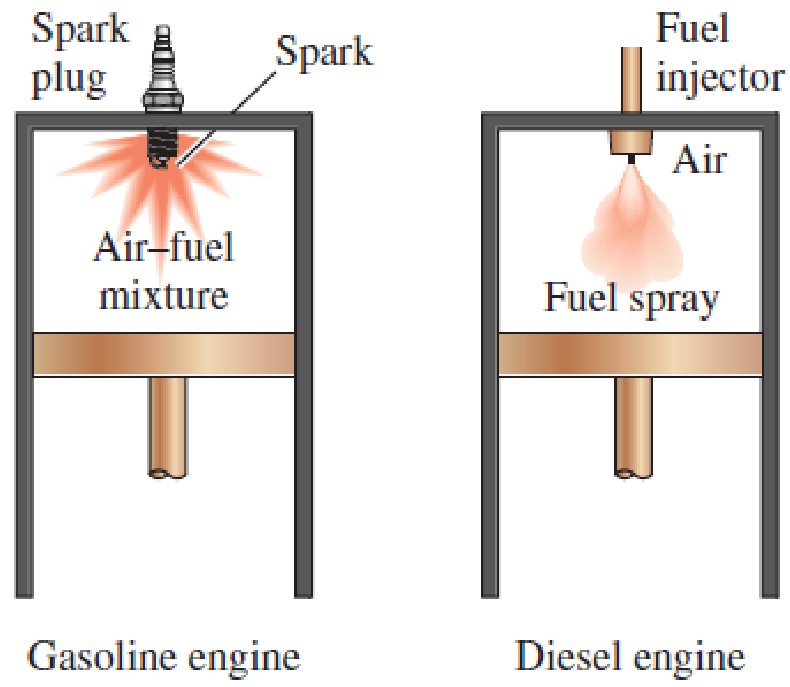


FIGURE 9–20
In diesel engines, the spark plug is replaced by a fuel injector, and only air is compressed during the compression process.

DIESEL CYCLE: THE IDEAL CYCLE FOR COMPRESSION-IGNITION ENGINES

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

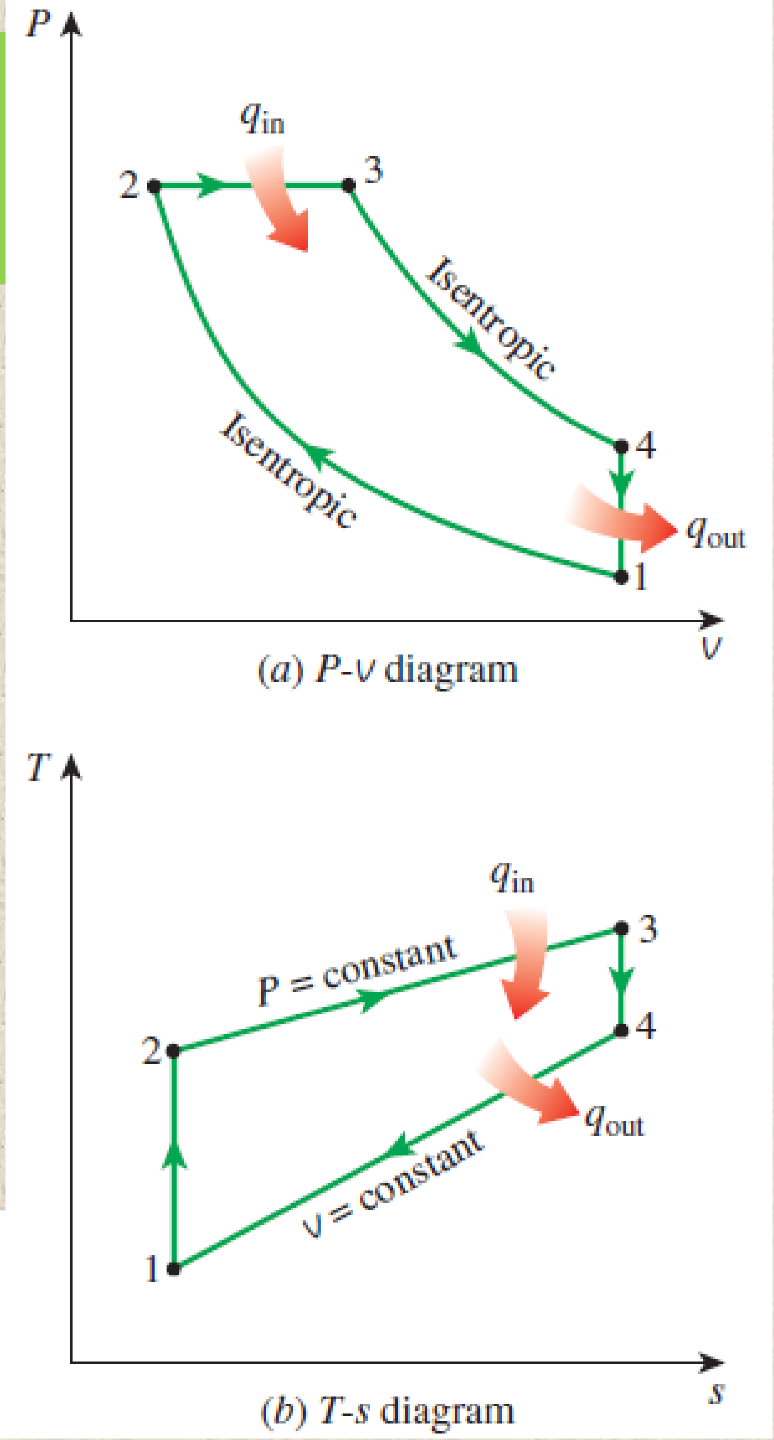
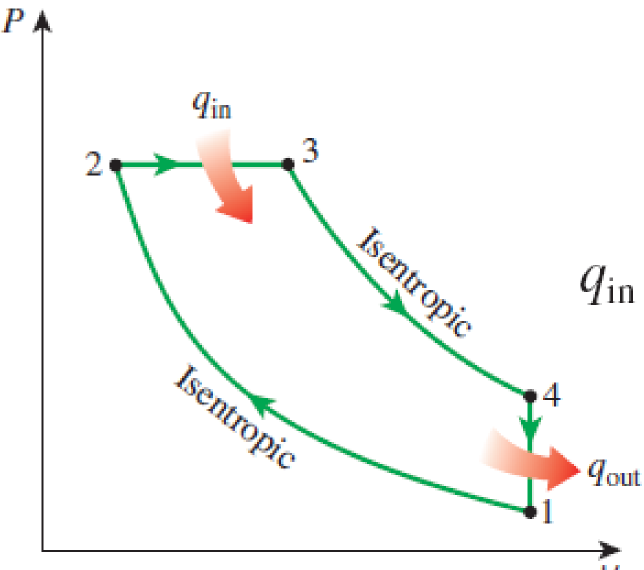


FIGURE 9–21
T-*s* and *P*-*v* diagrams for the ideal Diesel cycle.

The Diesel cycle is executed in a piston-cylinder device, which forms a closed system, the amount of heat transferred to the working fluid at constant pressure and rejected from it at constant volume can be expressed as (for constant specific heats):

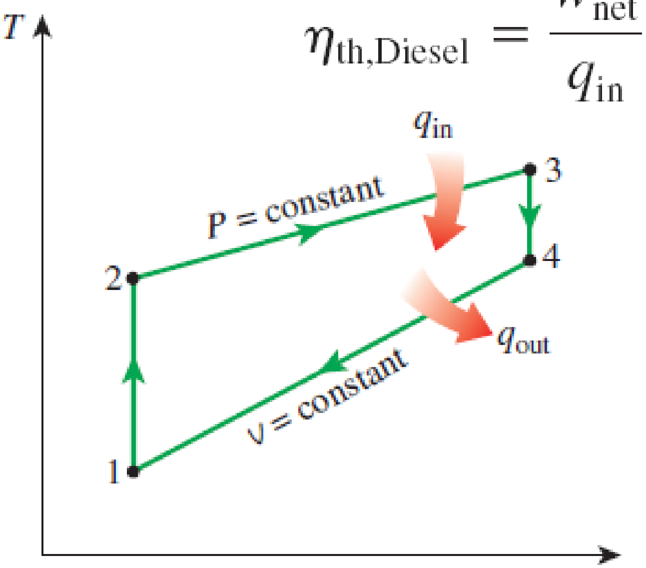


(*a*) *P*-*v* diagram

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

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

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



(*b*) *T*-*s* diagram

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

Define cutoff ratio r_c: as the ratio of the cylinder volumes after and before the combustion process

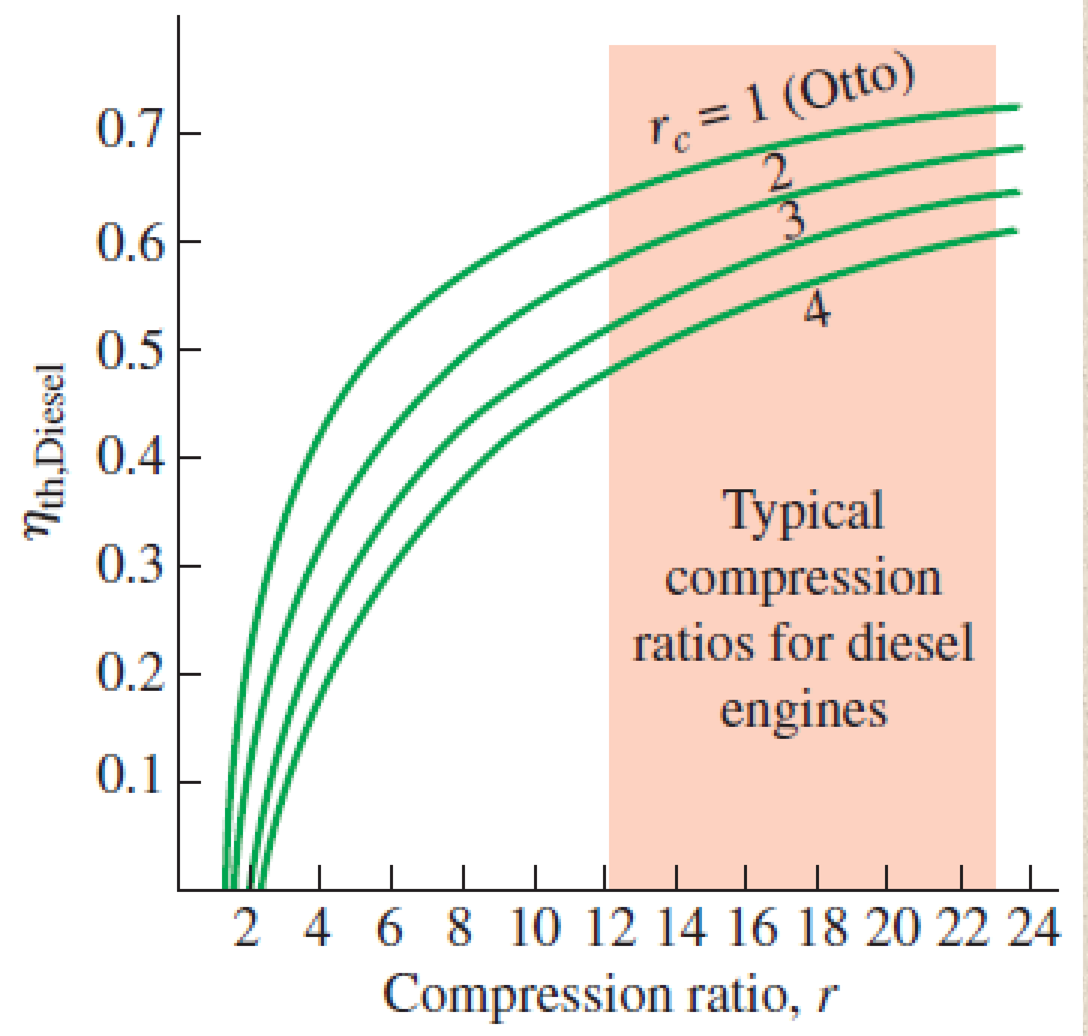
Processes (1-2) and (3-4) are isentropic:
(for constant specific heats)

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

Compression ratio: $r = v_1/v_2 = V_1/V_2$

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

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



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

For the same compression ratio ®:

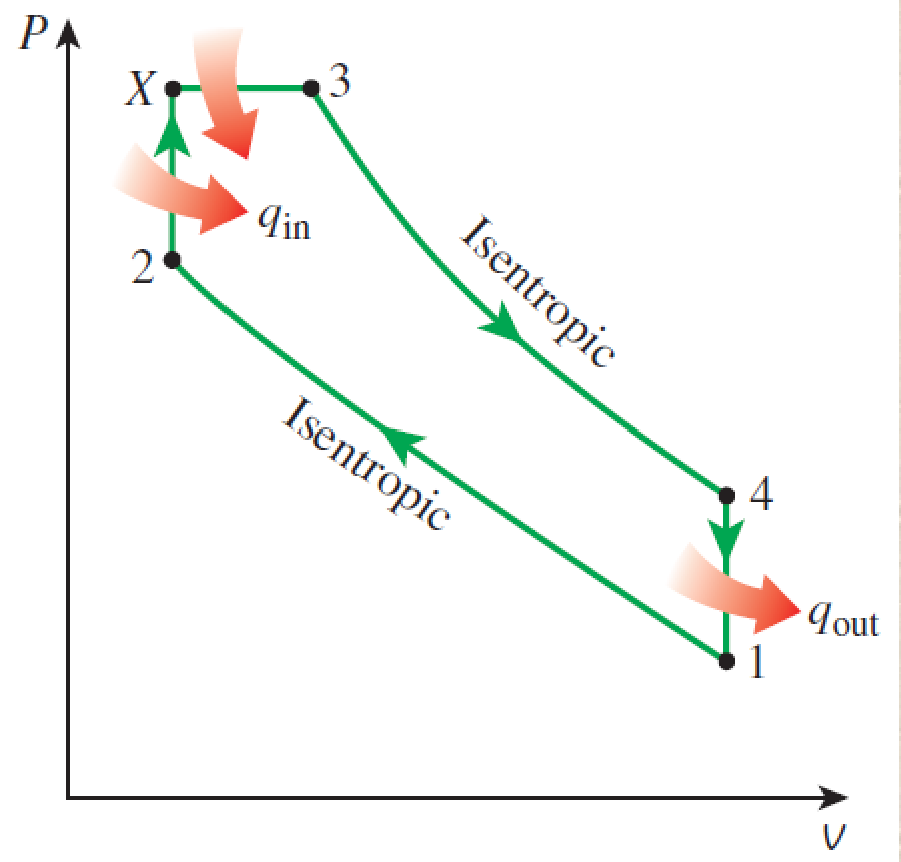
$$\eta_{\text{th,Otto}} > \eta_{\text{th,Diesel}}$$

However, diesel engines operate at much higher compression ratios and thus are usually more efficient than the spark-ignition engines

Thermal efficiencies of large diesel engines range from about 35 to 40 percent

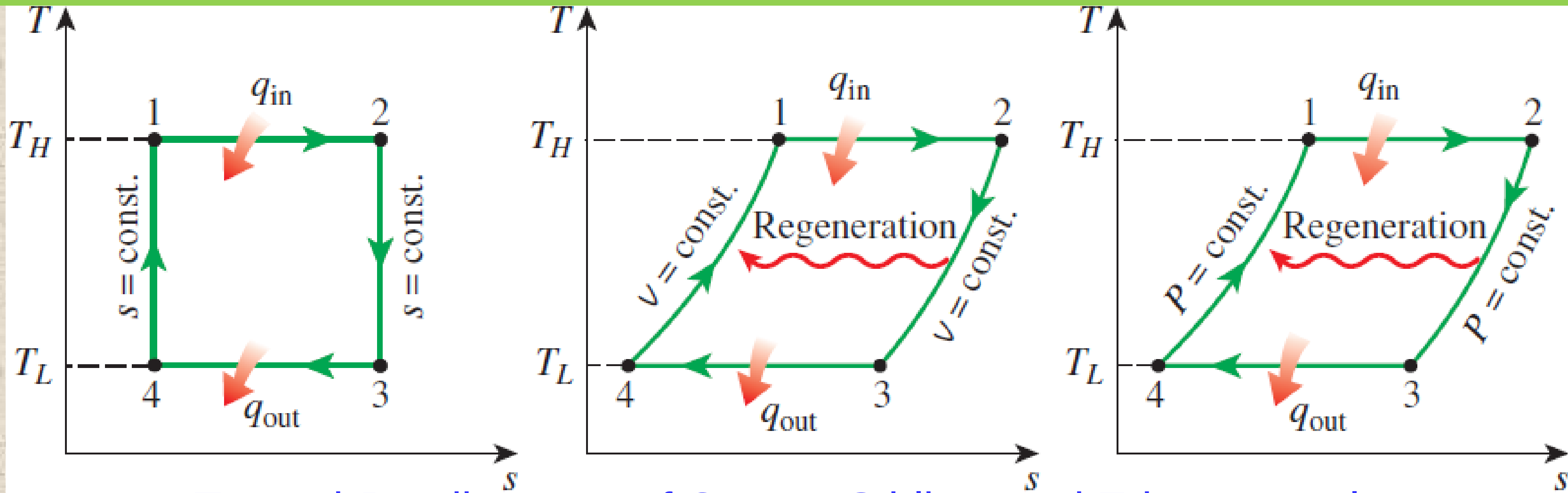
Dual cycle:

- In modern high-speed CI 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*

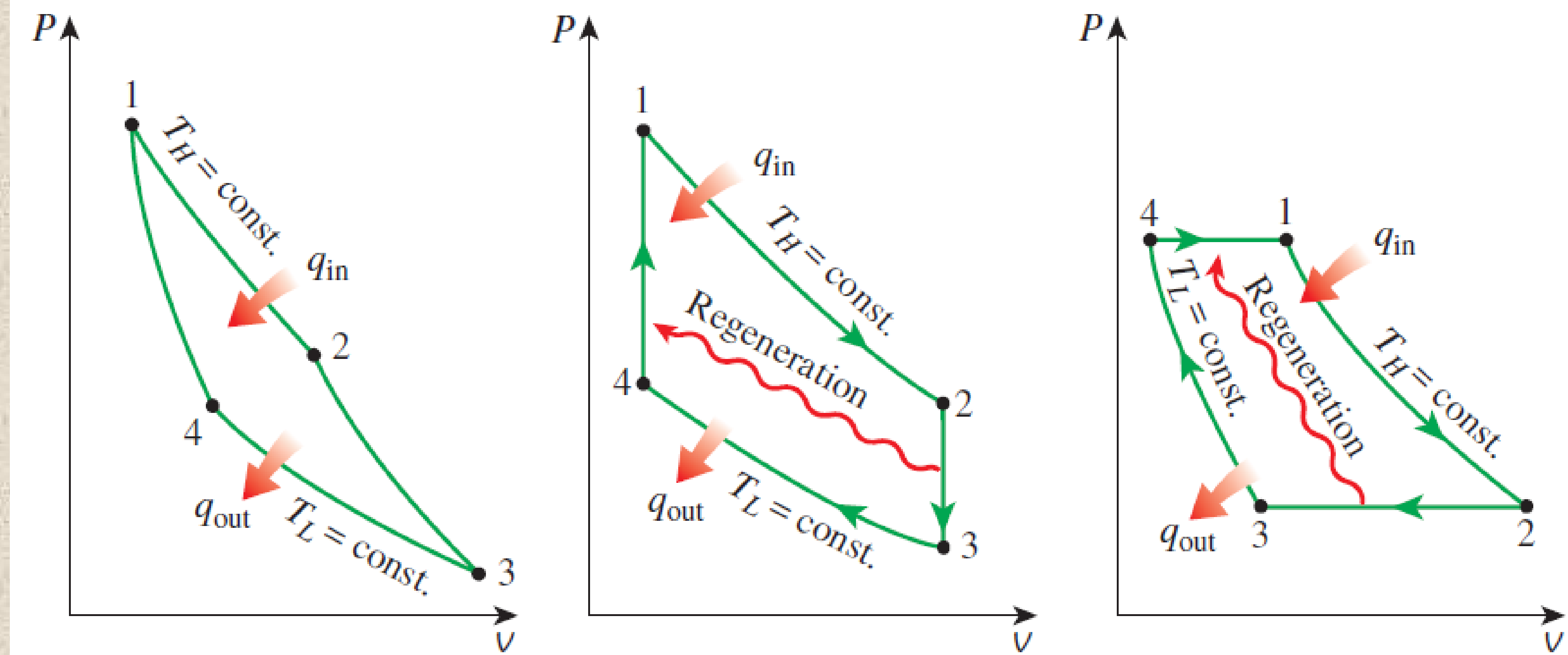


- *Fuel injection continues until the piston reaches the top dead center*, and combustion of the fuel keeps the pressure high well into the expansion stroke
- Thus, the *entire combustion process can better be modeled as the combination of constant-volume and constant-pressure processes*

STIRLING AND ERICSSON CYCLES – Totally Reversible



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



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

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)

Ericson cycle

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

2-3 p = *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 p = *constant* regeneration (internal heat transfer from the regenerator back to the working fluid)

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

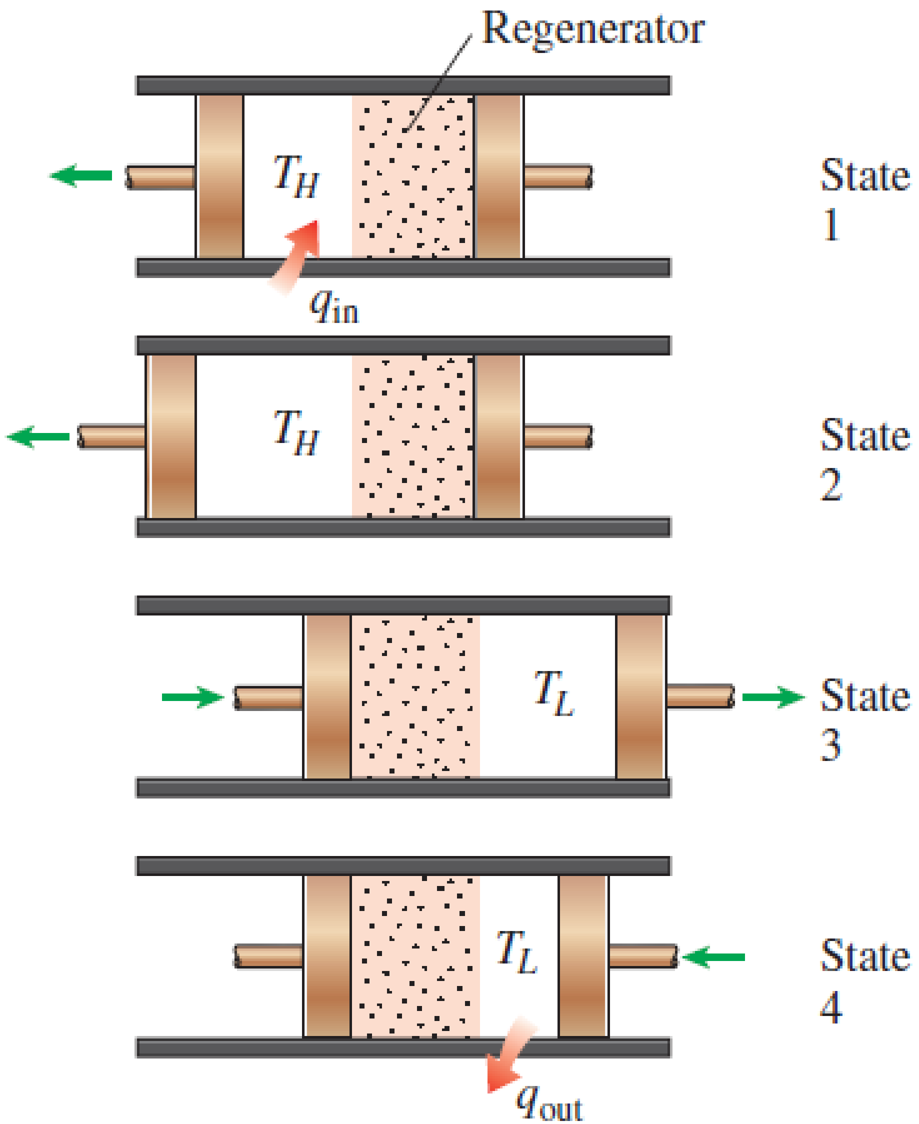


FIGURE 9–27
The execution of the Stirling cycle.

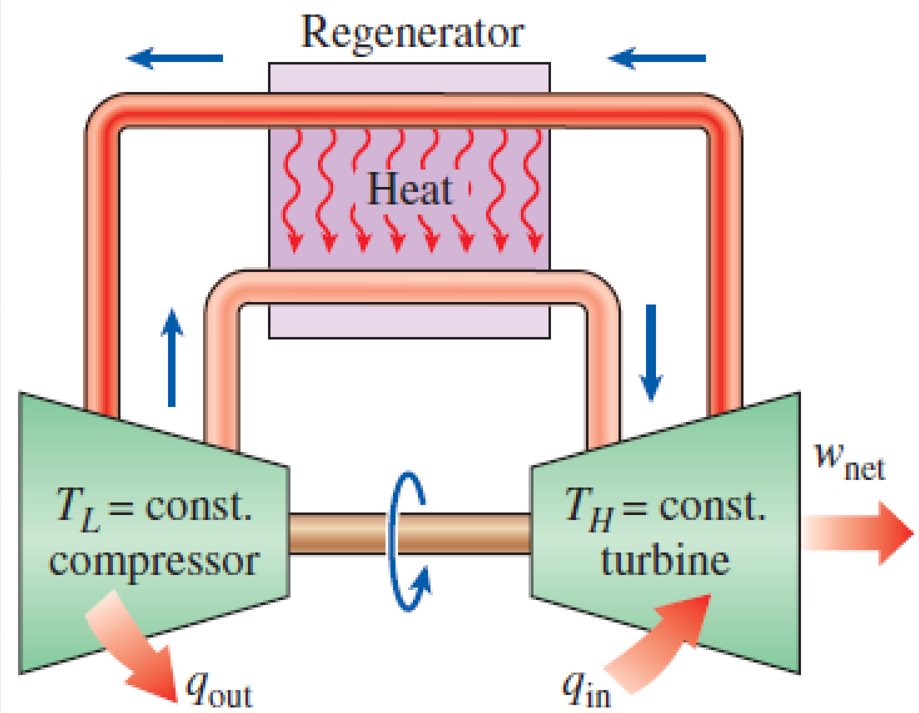


FIGURE 9–28
A steady-flow Ericsson engine.

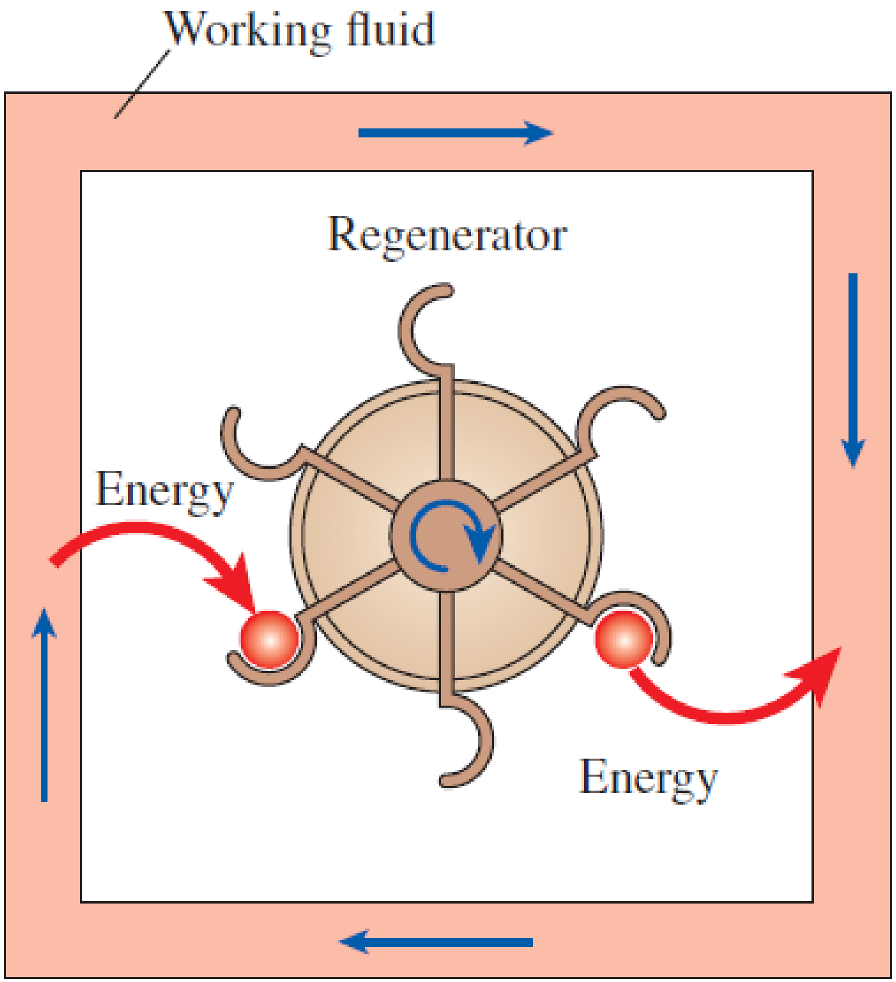


FIGURE 9–25

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.

Both Ericsson and Stirling cycles utilize **regeneration**, a process during which heat is transferred to the 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

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

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

- Stirling and Ericsson cycles 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

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

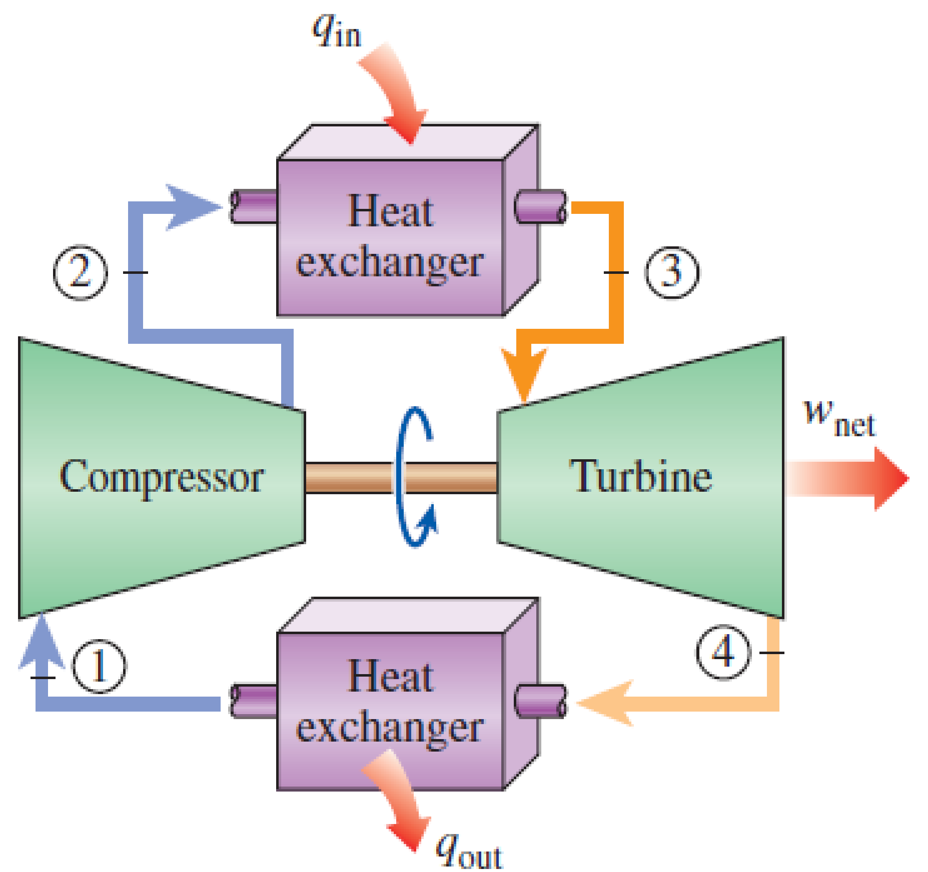


FIGURE 9–30
A closed-cycle gas-turbine engine.

Gas turbine usually operate on an open cycle:

- Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised
- The high-pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure.

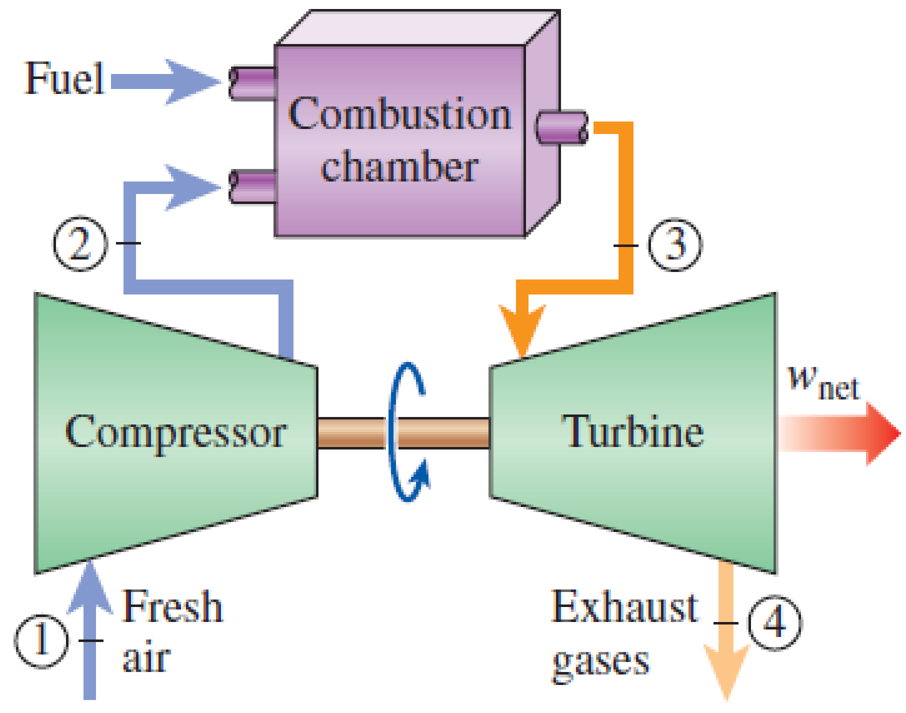
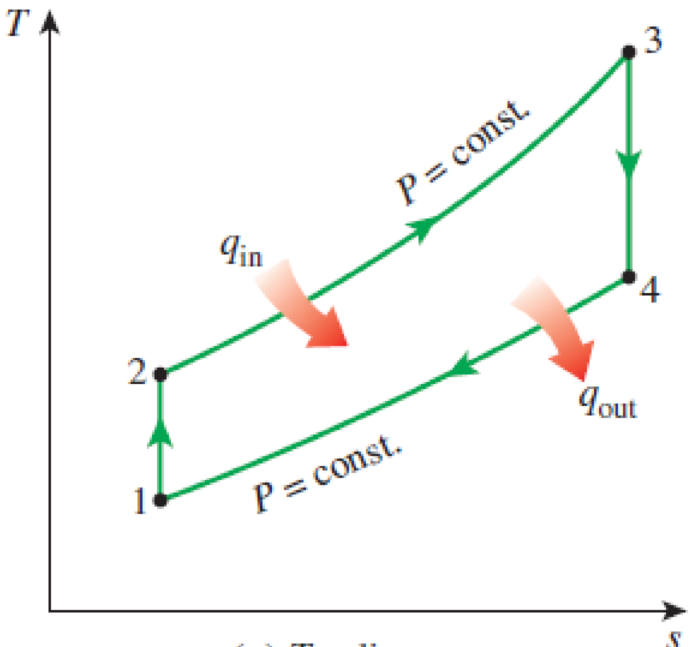


FIGURE 9–29
An open-cycle gas-turbine engine.

- The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power.
- The exhaust gases leaving the turbine are thrown out (not recirculated), causing the cycle to be classified as an open cycle



(*a*) *T*-*s* diagram

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

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

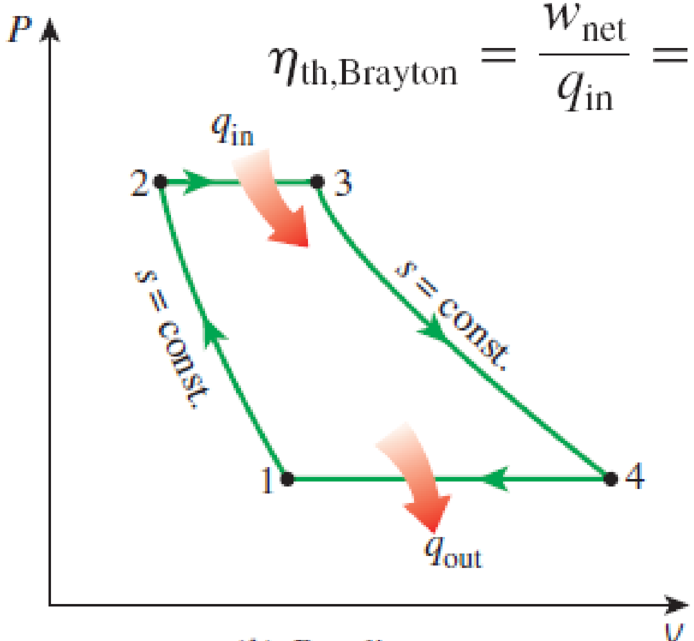
$$q_{out} = h_4 - h_1 = c_p(T_4 - T_1)$$

Constant specific heats:

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

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

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



(*b*) *P*-*v* diagram

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

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

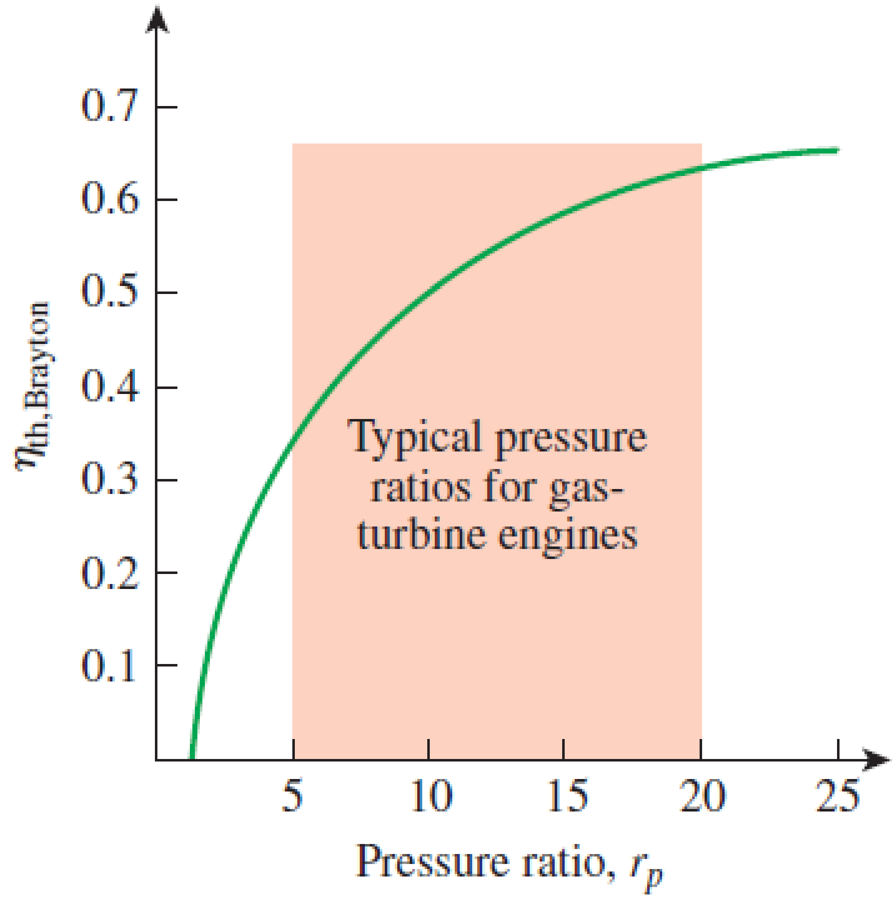


FIGURE 9–32

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

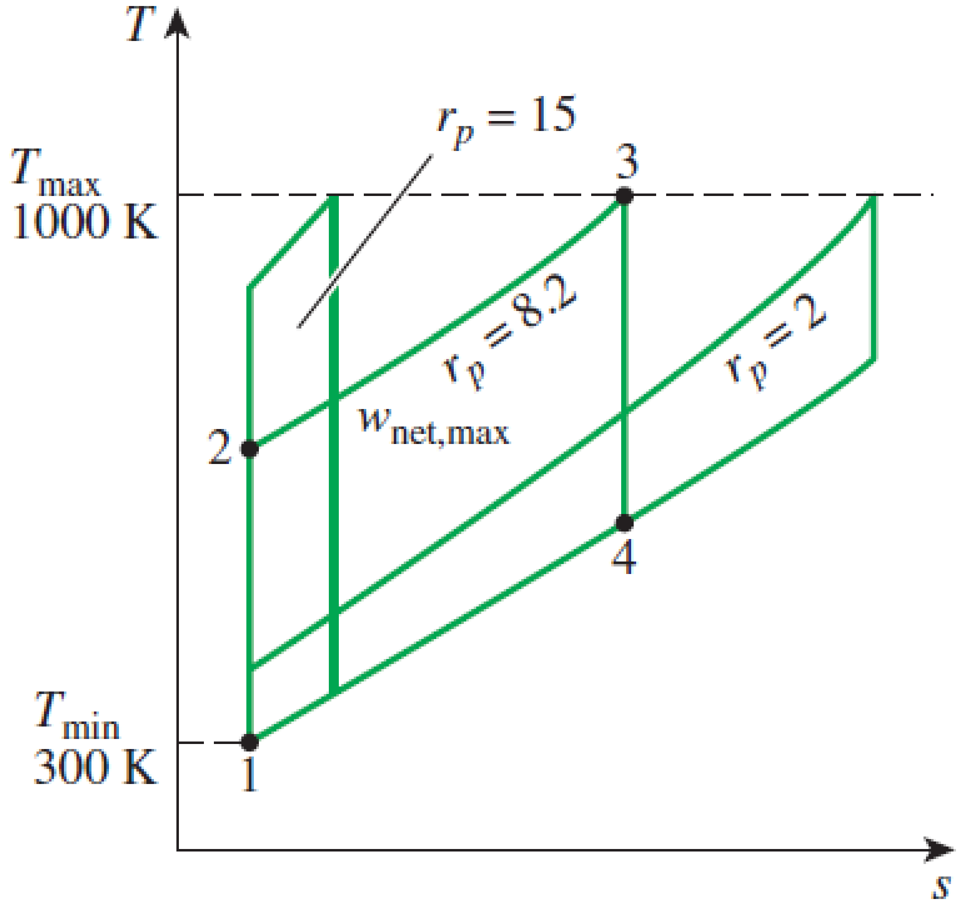


FIGURE 9–33

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.

GAS-TURBINE ENGINES

- The two major application areas of gas-turbine engines are *aircraft propulsion* and *electric power generation*
- 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. Therefore, in a cycle analysis, treating the combustion gases as air does not cause any appreciable error

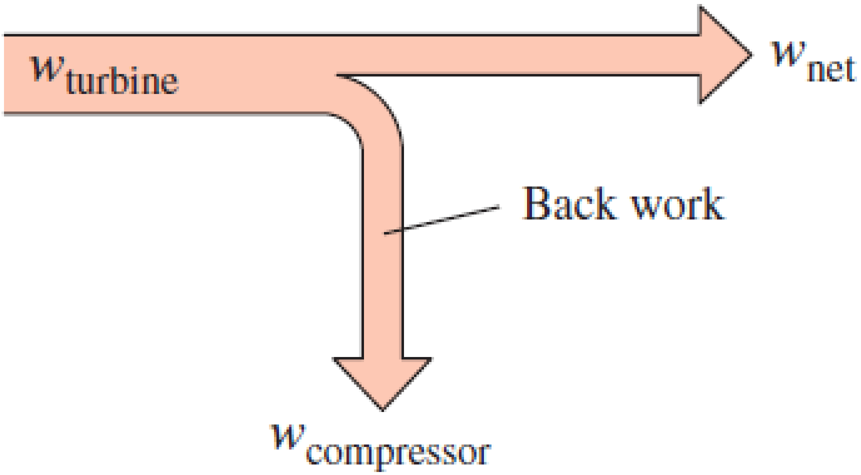
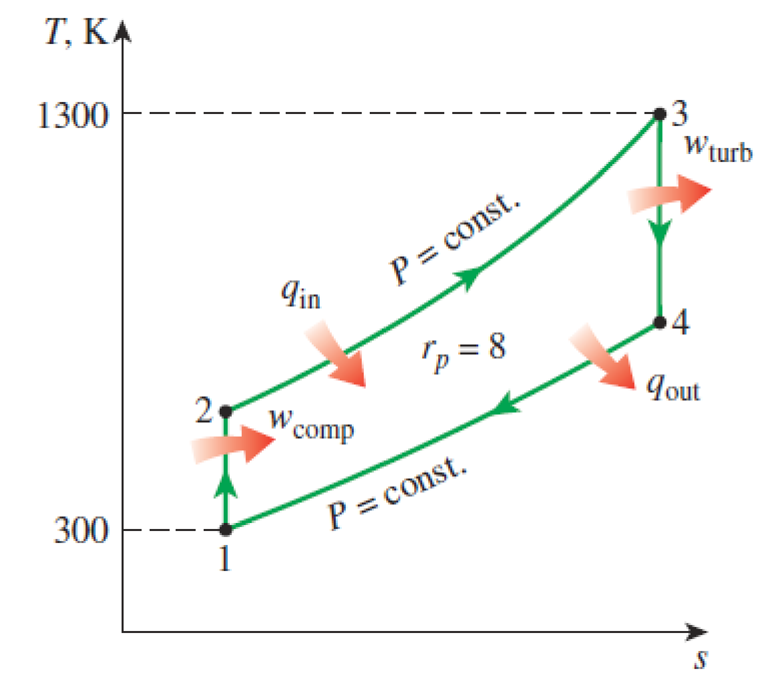


FIGURE 9–34
The fraction of the turbine work used to drive the compressor is called the back work ratio.

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

EXAMPLE: 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. Utilizing the air- standard assumptions, determine (*a*) the gas temperature at the exits of the compressor and the turbine, (*b*) the back work ratio, and (*c*) the thermal efficiency



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):

$$T_3 = 1300\text{ K} \rightarrow h_3 = 1395.97\text{ kJ/kg}$$

$$P_{r3} = 330.9$$

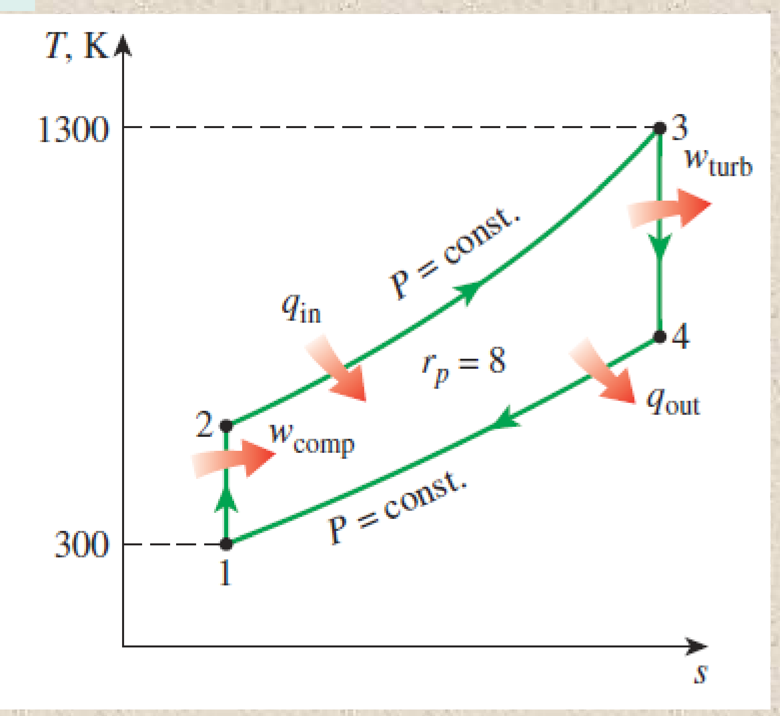
$$P_{r4} = \frac{P_4}{P_3} P_{r3} = \left(\frac{1}{8}\right)(330.9) = 41.36 \rightarrow T_4 = \mathbf{770\ K} \quad \text{(at turbine exit)}$$

$$h_4 = 789.37\text{ kJ/kg}$$

$$w_{\text{comp,in}} = h_2 - h_1 = 544.35 - 300.19 = 244.16 \text{ kJ/kg}$$

$$w_{\text{turb,out}} = h_3 - h_4 = 1395.97 - 789.37 = 606.60 \text{ kJ/kg}$$

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



$$q_{\text{in}} = h_3 - h_2 = 1395.97 - 544.35 = 851.62 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{out}} - w_{\text{in}} = 606.60 - 244.16 = 362.4 \text{ kJ/kg}$$

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

Development of Gas Turbines

1. *Increasing the turbine inlet (or firing) temperatures*: The turbine inlet temperatures have increased steadily from about 540°C in the 1940s to 1425°C and even higher today, with development of new materials and the innovative cooling techniques for the critical components
2. *Increasing the efficiencies of turbomachinery components* (turbines, compressors): Advanced techniques for computer-aided design made it possible to design these components aerodynamically with minimal losses
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

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

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

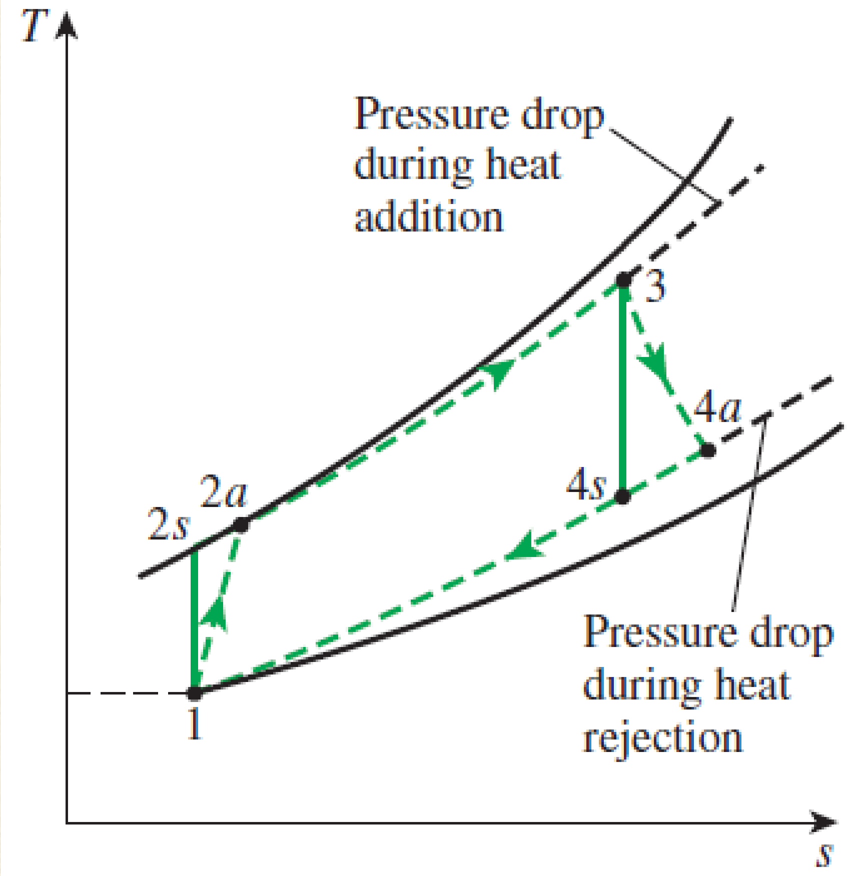


FIGURE 9–36
The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

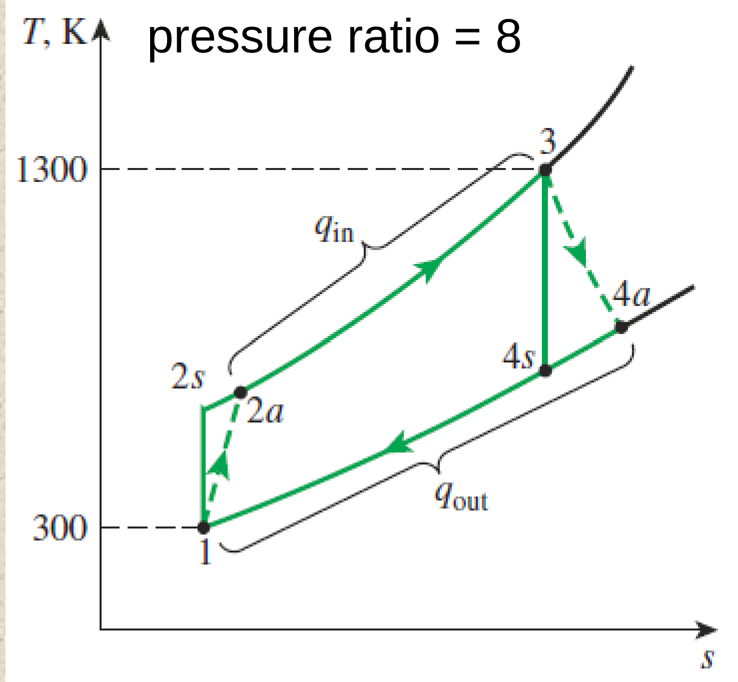
EXAMPLE: ACTUAL GAS TURBINE CYCLE

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 previously (simple ideal cycle)

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

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

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



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

$$q_{\text{in}} = h_3 - h_{2a} = 1395.97 - 605.39 = 790.58\ \text{kJ/kg}$$

$$w_{\text{net}} = w_{\text{out}} - w_{\text{in}} = 515.61 - 305.20 = 210.41\ \text{kJ/kg}$$

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

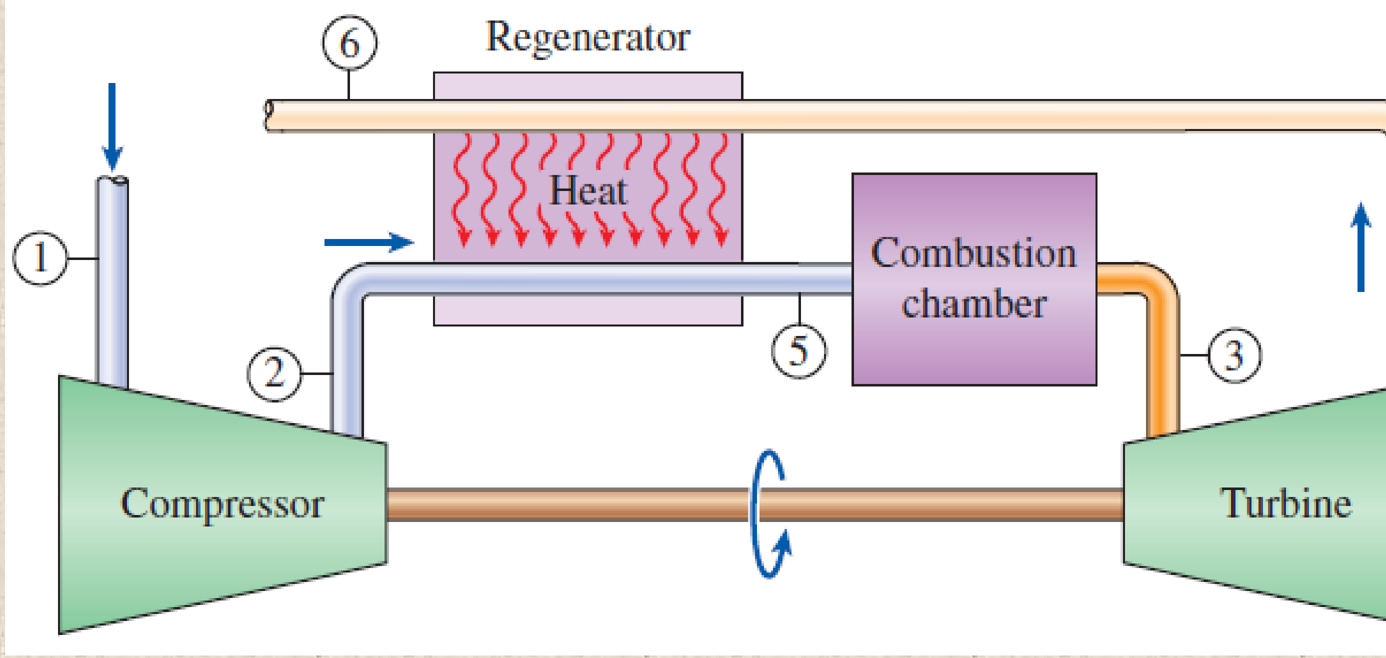
$$\begin{aligned} w_{\text{turb,out}} = h_3 - h_{4a} \rightarrow h_{4a} &= h_3 - w_{\text{turb,out}} \\ &= 1395.97 - 515.61 \\ &= 880.36\ \text{kJ/kg} \end{aligned}$$

Table A–17,

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

THE BRAYTON CYCLE WITH REGENERATION

- The air leaving the compressor can be heated by the hot exhaust gases leaving the turbine in a counter-flow heat exchanger (a *regenerator* or a *recuperator*)
- The thermal efficiency of the Brayton cycle increases since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering combustion chamber, thus reducing the fuel consumption



A gas-turbine engine with regenerator

THE BRAYTON CYCLE WITH REGENERATION

- Regeneration is only useful when the temperature of the exhaust gas leaving the turbine (State-4) is higher than the temperature of the air leaving the compressor (State-2)
- In the limiting (ideal case), the air exits the regenerator at the inlet temperature of the exhaust gases (T_4)

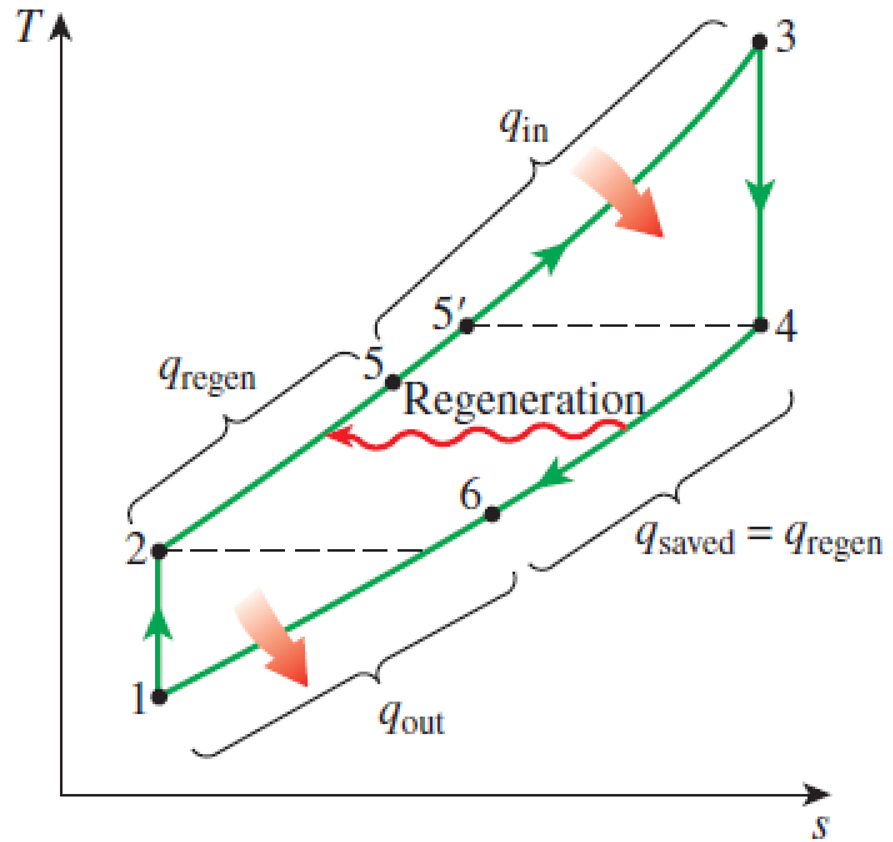
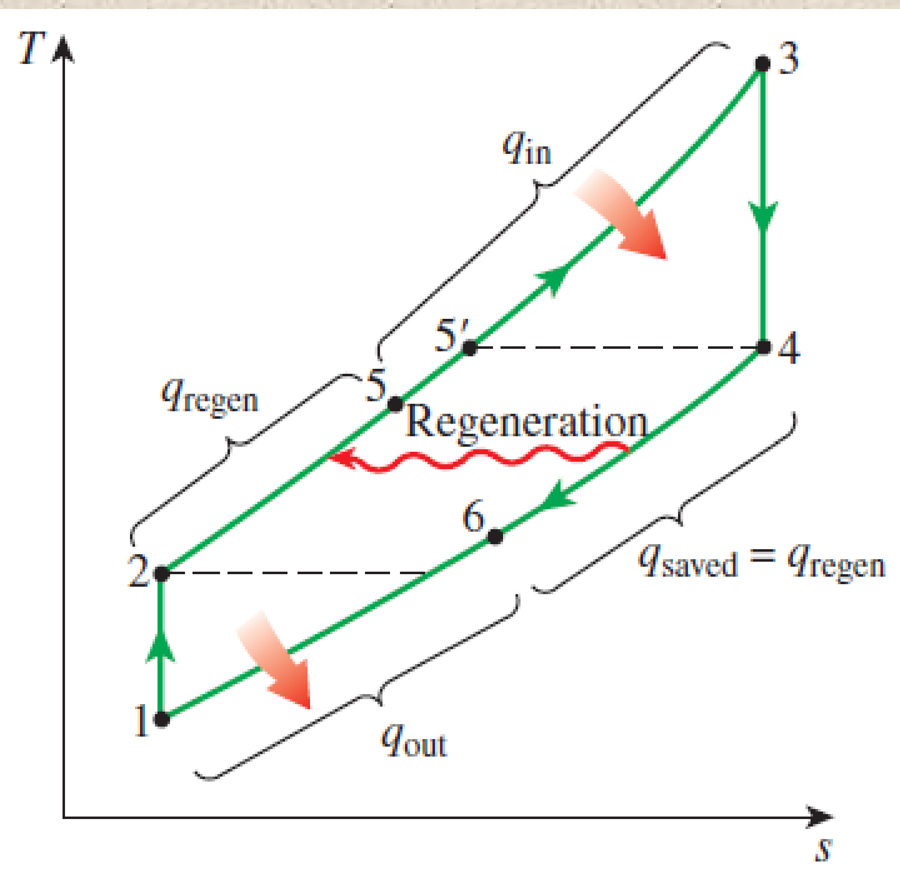


FIGURE 9–39
T-s diagram of a Brayton cycle with regeneration.



$$q_{regen,act} = h_5 - h_2$$

$$q_{regen,max} = h_{5'} - h_2 = h_4 - h_2$$

Effectiveness of regenerator

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

Effectiveness under cold-air standard assumptions

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

Generally: $\epsilon < 0.85$

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

- Under cold-air standard assumptions the thermal efficiency of an ideal Brayton cycle 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

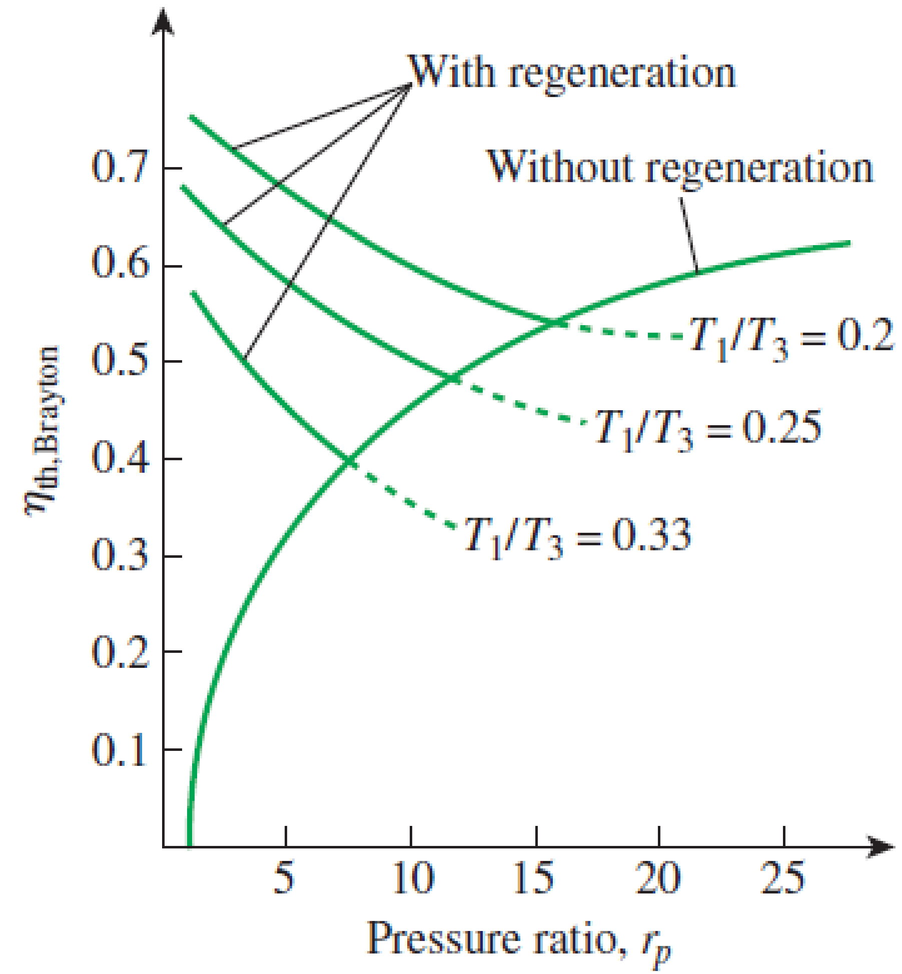
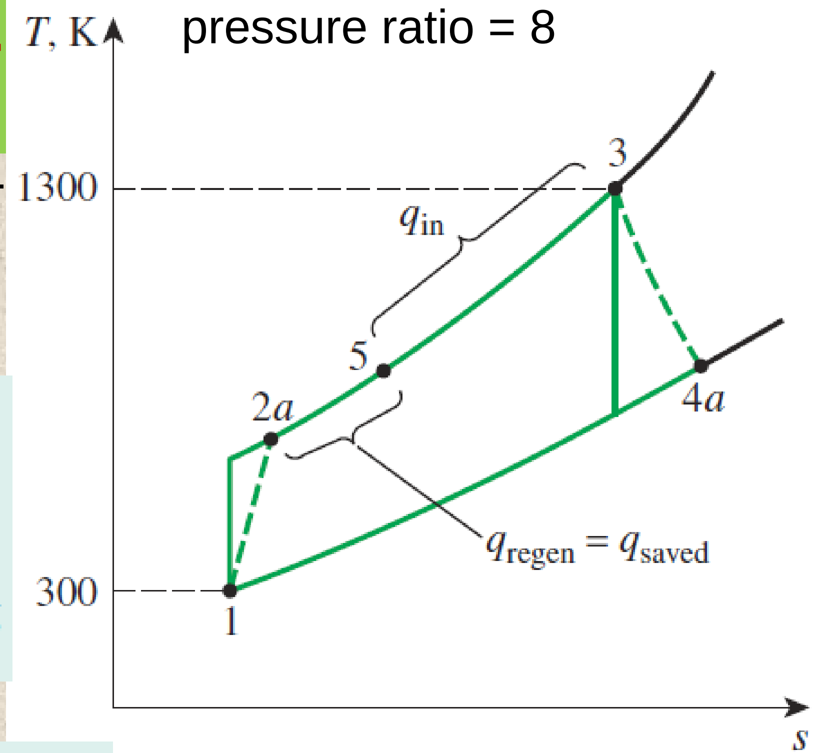


FIGURE 9–40

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

EXAMPLE: ACTUAL GAS TURBINE CYCLE WITH REGENERATION

Determine the thermal efficiency of the actual gas-turbine cycle discussed previously if a regenerator having an effectiveness of 80 percent is installed



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

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

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

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

THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

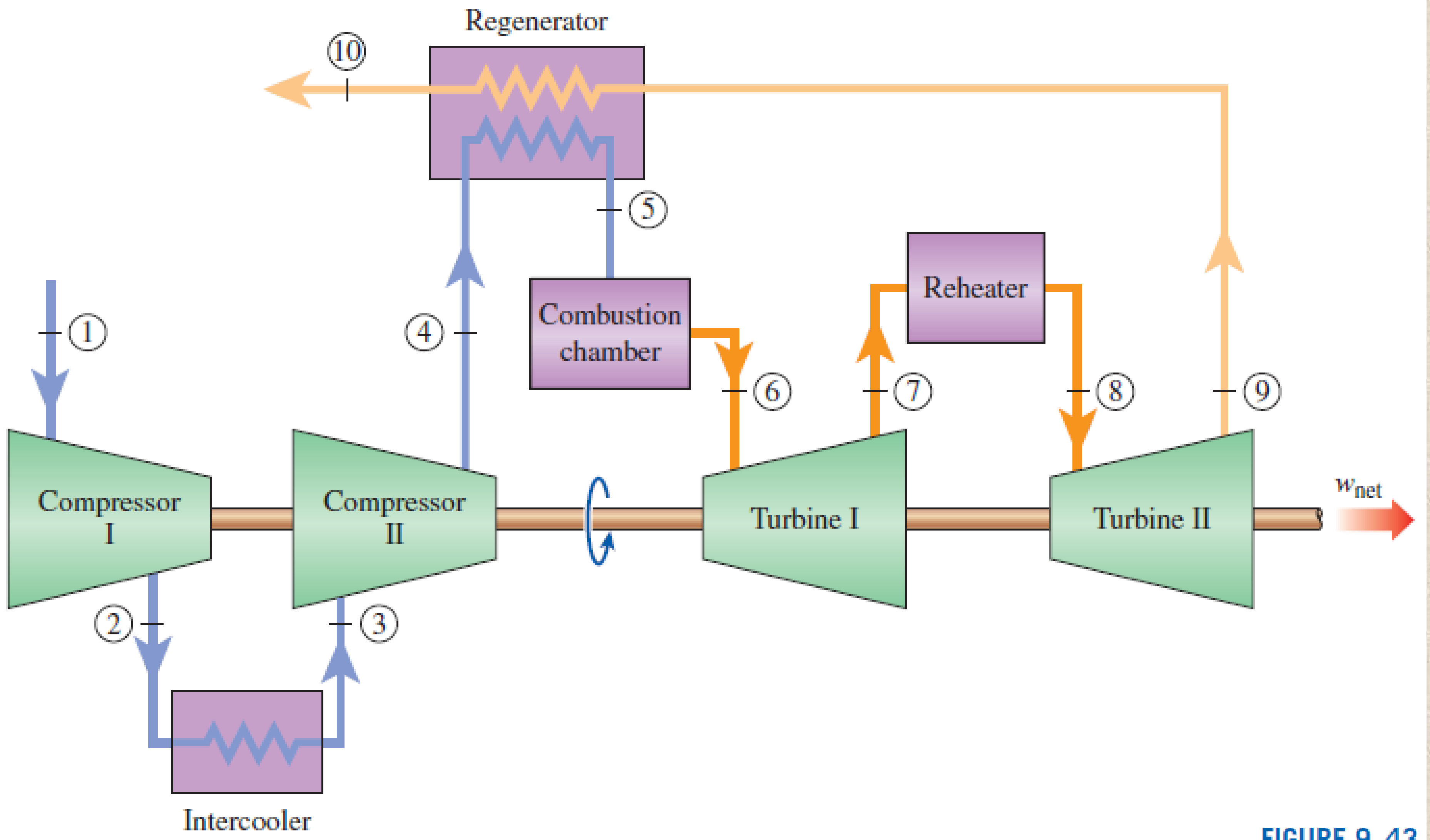


FIGURE 9–43

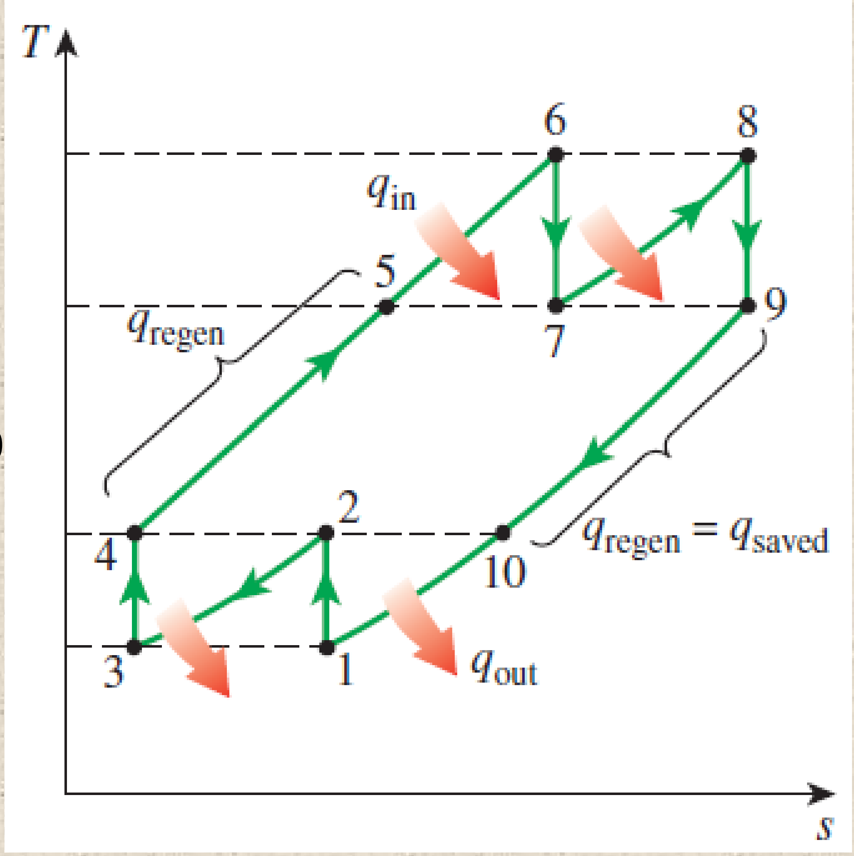
A gas-turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration.

THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

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

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

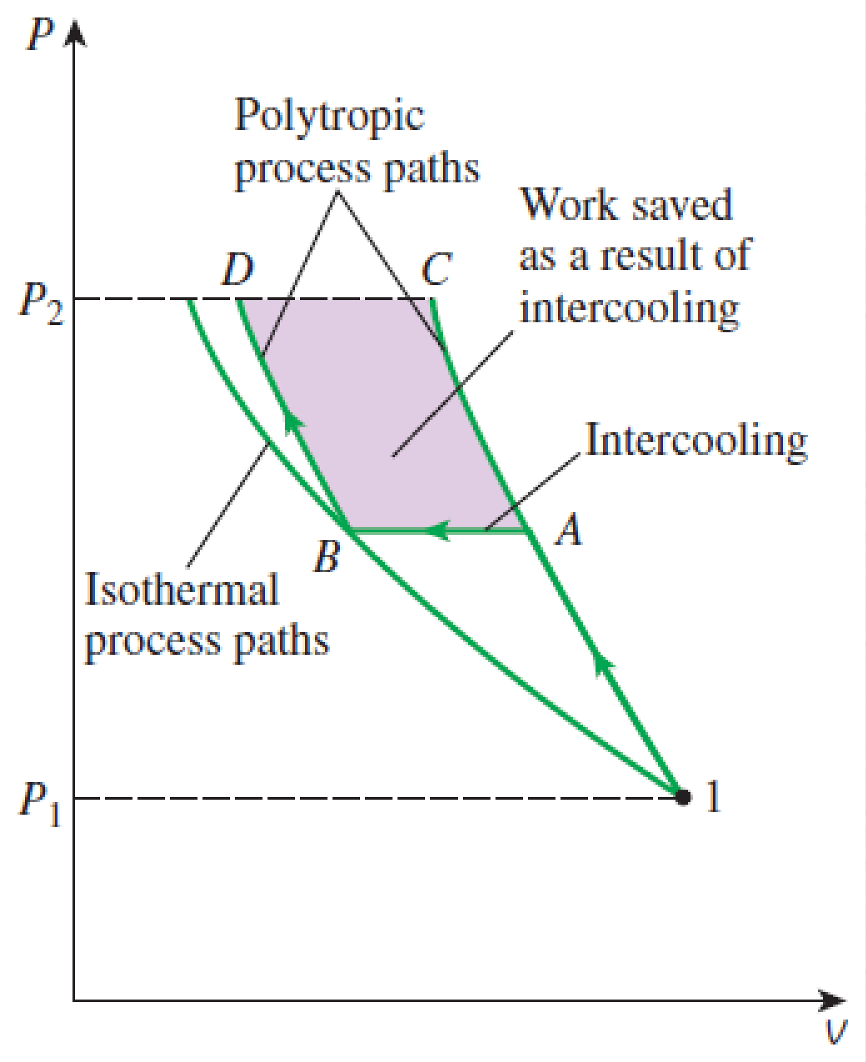
- *Reheating* can be accomplished by simply spraying additional fuel into the exhaust gases between the two expanding states
- *Intercooling and reheating always decreases the thermal efficiency unless they are accompanied by regeneration*. This is because intercooling decreases the average temperature at which heat is added and reheating increases the average temperature at which heat is rejected



T-s diagram of an ideal gas-turbine 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.



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

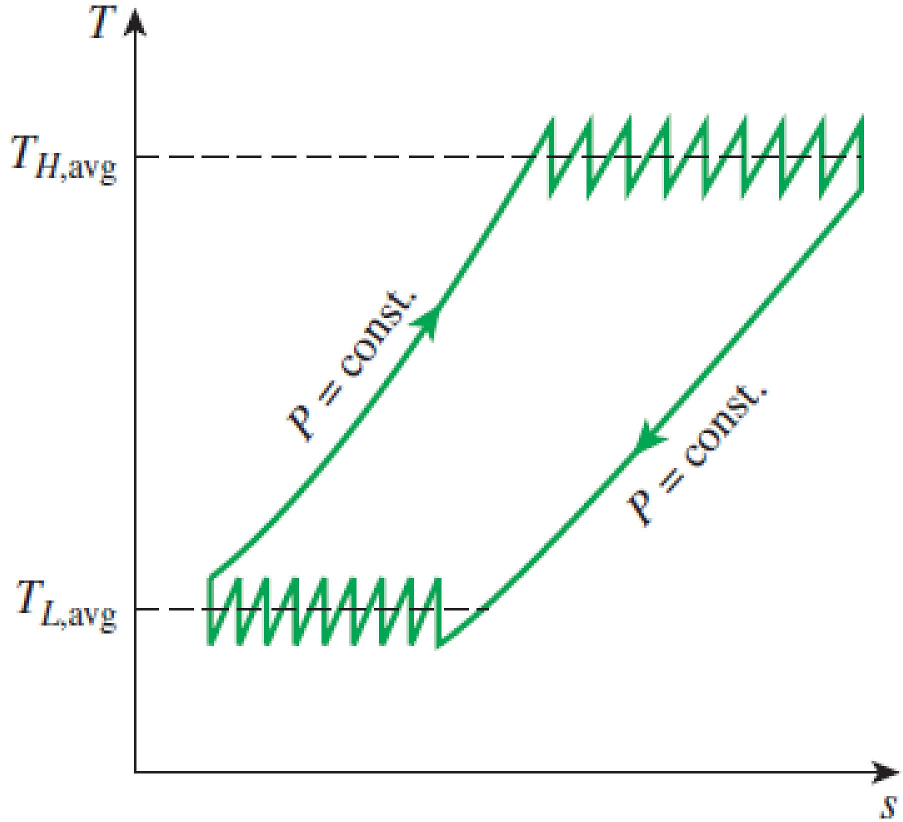


FIGURE 9–45

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

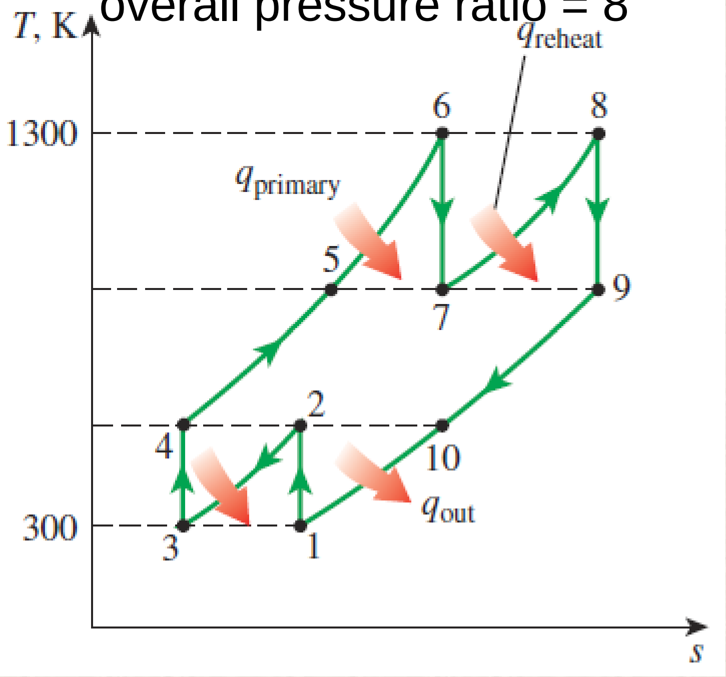
EXAMPLE: A GAS TURBINE WITH REHEATING AND INTERCOOLING

An ideal gas-turbine cycle with two stages of compression and two stages of expansion has an overall pressure ratio of 8. Air enters each stage of the compressor at 300 K and each stage of the turbine at 1300 K. Determine the back work ratio and the thermal efficiency of this gas-turbine cycle, assuming (*a*) no regenerators and (*b*) an ideal regenerator with 100 percent effectiveness.

For two-stage compression and expansion, the work input is minimized and the work output is maximized when both stages of the compressor and the turbine have the same pressure ratio. Thus,

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} = \sqrt{8} = 2.83 \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9} = \sqrt{8} = 2.83$$

overall pressure ratio = 8

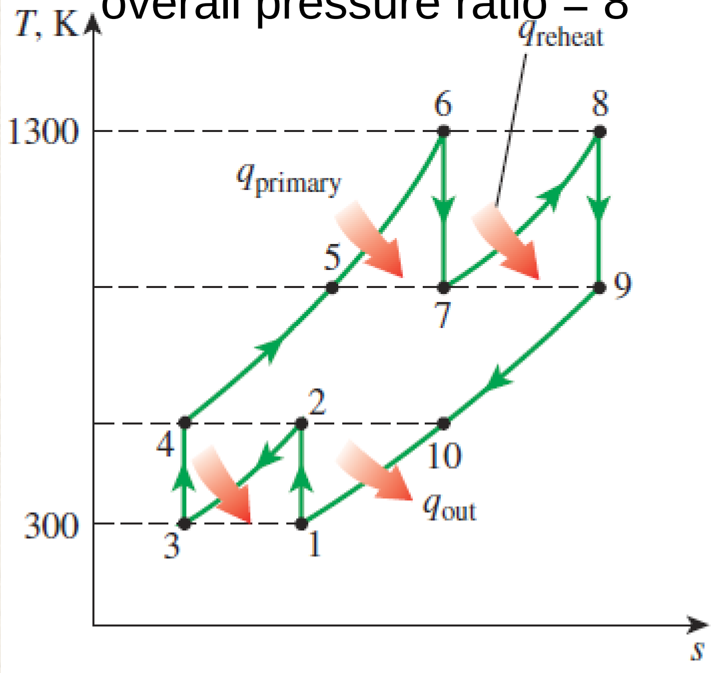


At inlets: $T_1 = T_3, \quad h_1 = h_3 \quad \text{and} \quad T_6 = T_8, \quad h_6 = h_8$

At exits: $T_2 = T_4, \quad h_2 = h_4 \quad \text{and} \quad T_7 = T_9, \quad h_7 = h_9$

$$T_1 = 300\text{ K} \rightarrow h_1 = 300.19\text{ kJ/kg}$$
$$P_{r1} = 1.386$$
$$P_{r2} = \frac{P_2}{P_1} P_{r1} = \sqrt{8}(1.386) = 3.92 \rightarrow T_2 = 403.3\text{ K}$$
$$h_2 = 404.31\text{ kJ/kg}$$
$$T_6 = 1300\text{ K} \rightarrow h_6 = 1395.97\text{ kJ/kg}$$
$$P_{r6} = 330.9$$
$$P_{r7} = \frac{P_7}{P_6} P_{r6} = \frac{1}{\sqrt{8}}(330.9) = 117.0 \rightarrow T_7 = 1006.4\text{ K}$$
$$h_7 = 1053.33\text{ kJ/kg}$$

overall pressure ratio = 8



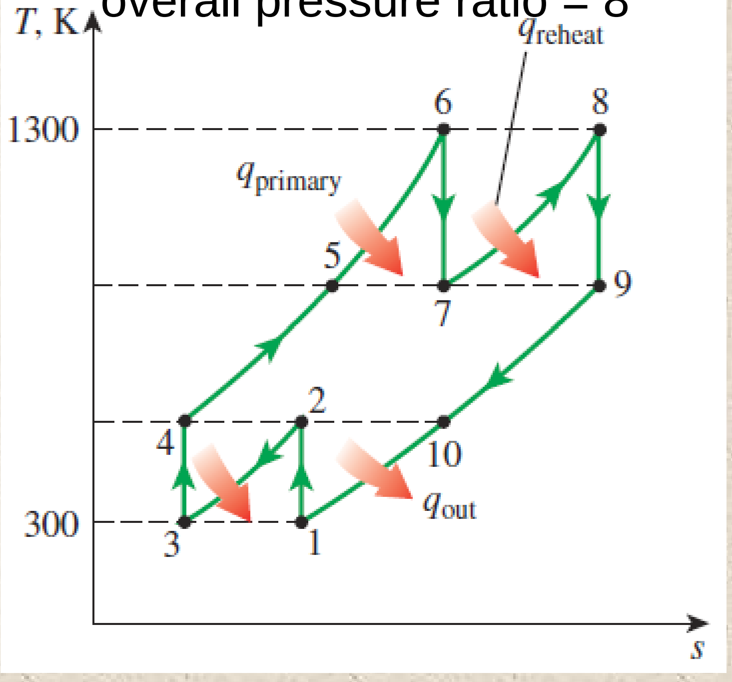
$$w_{\text{comp,in}} = 2(w_{\text{comp,in,I}}) = 2(h_2 - h_1) = 2(404.31 - 300.19) = 208.24\text{ kJ/kg}$$
$$w_{\text{turb,out}} = 2(w_{\text{turb,out,I}}) = 2(h_6 - h_7) = 2(1395.97 - 1053.33) = 685.28\text{ kJ/kg}$$
$$w_{\text{net}} = w_{\text{turb,out}} - w_{\text{comp,in}} = 685.28 - 208.24 = 477.04\text{ kJ/kg}$$
$$q_{\text{in}} = q_{\text{primary}} + q_{\text{reheat}} = (h_6 - h_4) + (h_8 - h_7)$$
$$= (1395.97 - 404.31) + (1395.97 - 1053.33) = 1334.30\text{ kJ/kg}$$

$$r_{\text{bw}} = \frac{w_{\text{comp,in}}}{w_{\text{turb,out}}} = \frac{208.24\text{ kJ/kg}}{685.28\text{ kJ/kg}} = \mathbf{0.304}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{477.04\text{ kJ/kg}}{1334.30\text{ kJ/kg}} = \mathbf{0.358}\text{ or }\mathbf{35.8\%}$$

without regenerator:

overall pressure ratio = 8



$$q_{\text{in}} = q_{\text{primary}} + q_{\text{reheat}} = (h_6 - h_5) + (h_8 - h_7)$$

$$= (1395.97 - 1053.33) + (1395.97 - 1053.33) = 685.28 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{477.04 \text{ kJ/kg}}{685.28 \text{ kJ/kg}} = \mathbf{0.696} \text{ or } \mathbf{69.6\%}$$

with ideal regenerator with 100 percent effectiveness

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***)

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

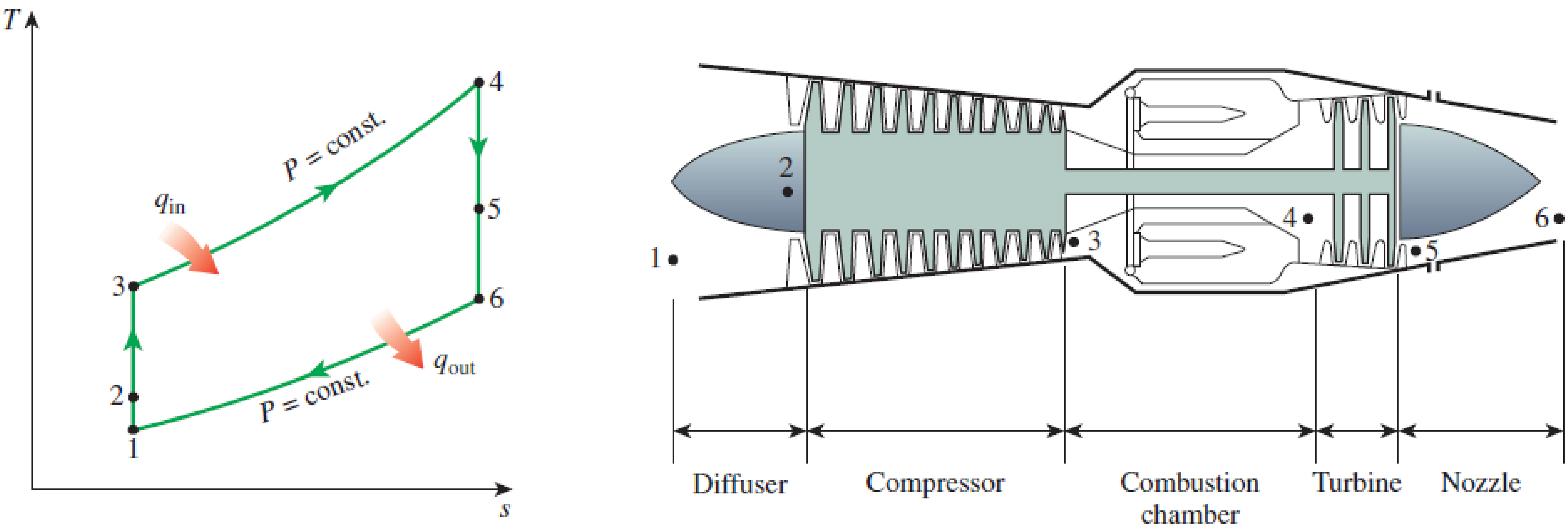
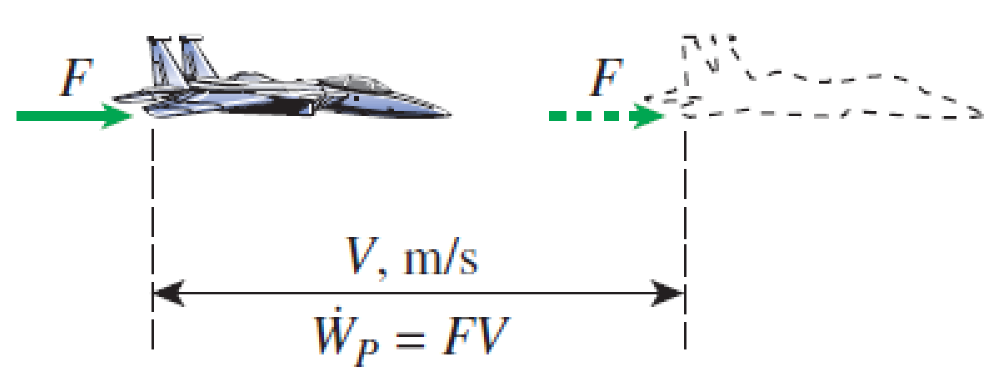


FIGURE 9–48
Basic components of a turbojet engine and the *T-s* diagram for the ideal turbojet cycle.

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



Using Newton's second law and considering that the pressures at the inlet and the exit are identical (ambient pressure):

Thrust (propulsive force)

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

Propulsive power

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

Propulsive efficiency

$$\eta_P = \frac{\text{Propulsive power}}{\text{Energy input rate}} = \frac{\dot{W}_P}{\dot{Q}_{\text{in}}}$$

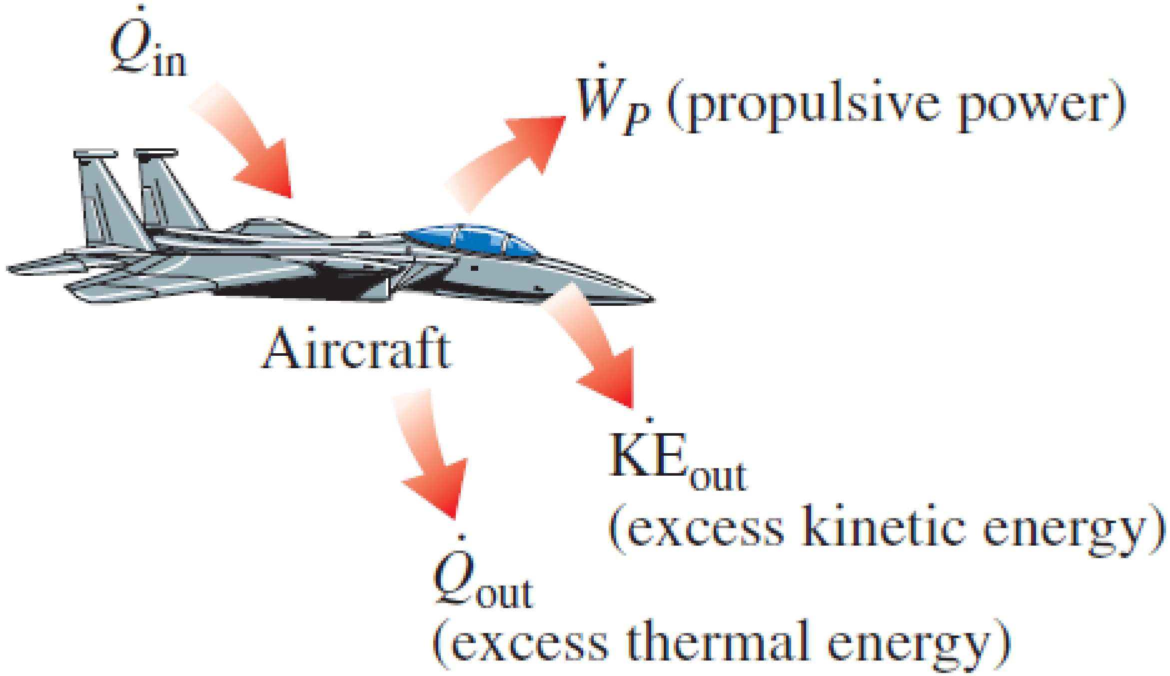


FIGURE 9–51

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 ***turboprop 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.

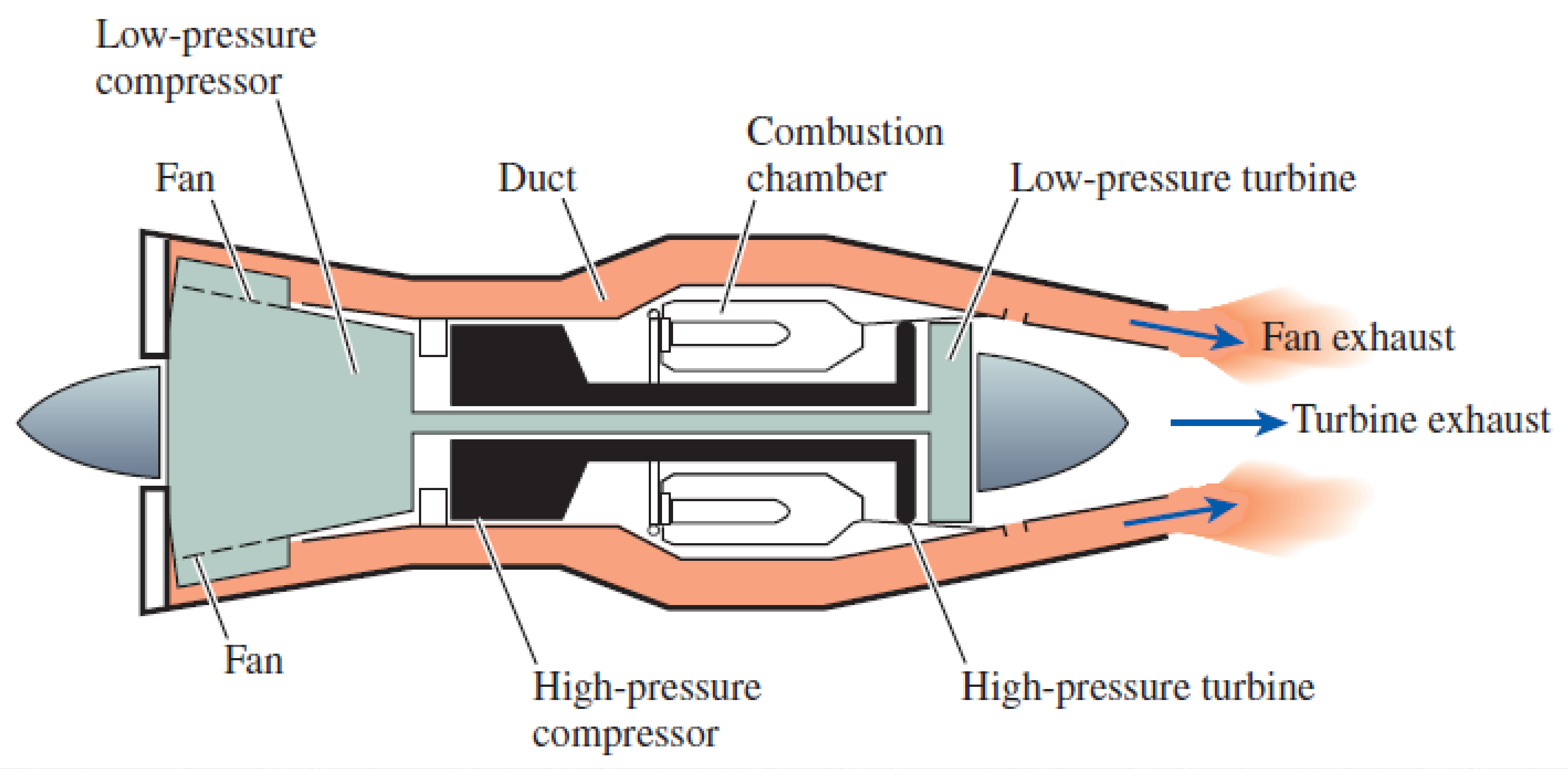


FIGURE 9–52
A turbofan engine.

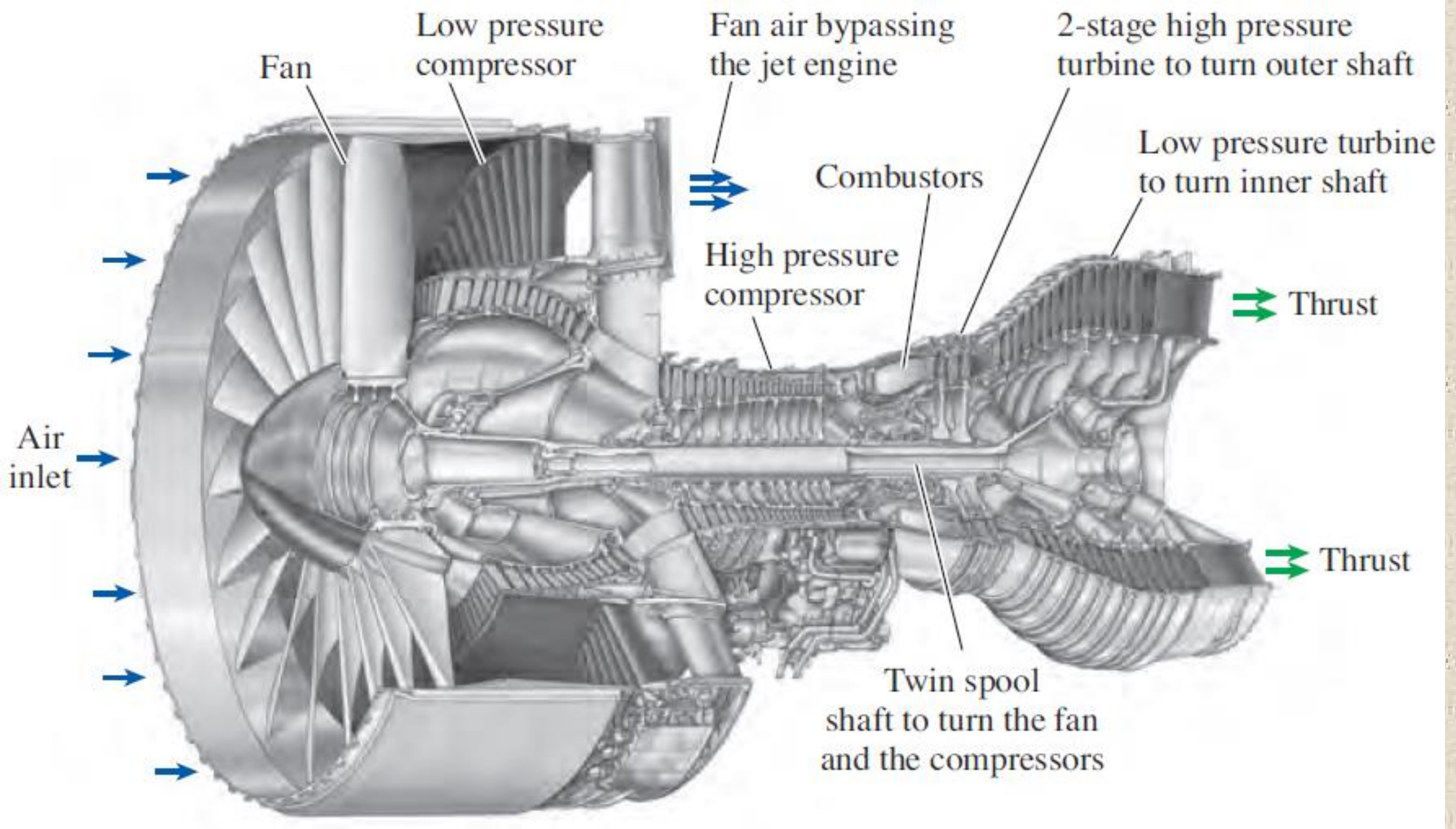


FIGURE 9–53

A modern jet engine used to power Boeing 777 aircraft. This is a Pratt & Whitney PW4084 turbofan capable of producing 84,000 pounds of thrust. It is 4.87 m (192 in) long, has a 2.84 m (112 in) diameter fan, and it weighs 6800 kg (15,000 lbm).

Various engine types:

Turbofan, Turboprop, Ramjet, Sacramjet, Rocket

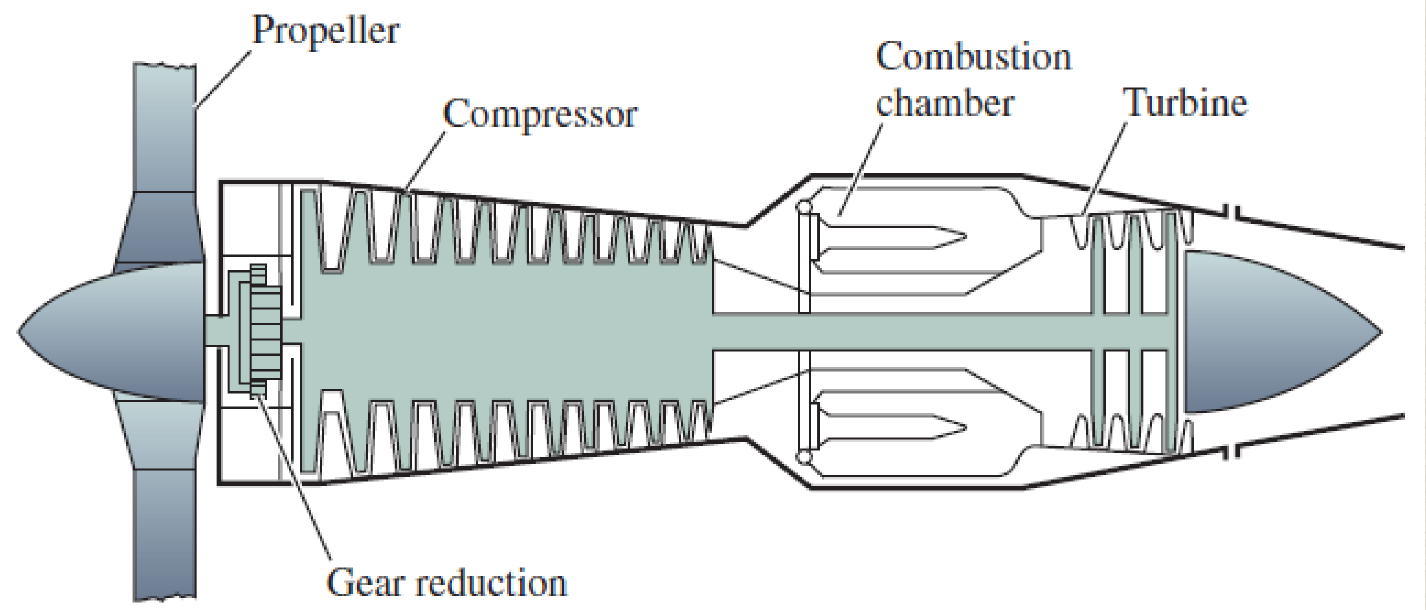


FIGURE 9–54

A turboprop engine.

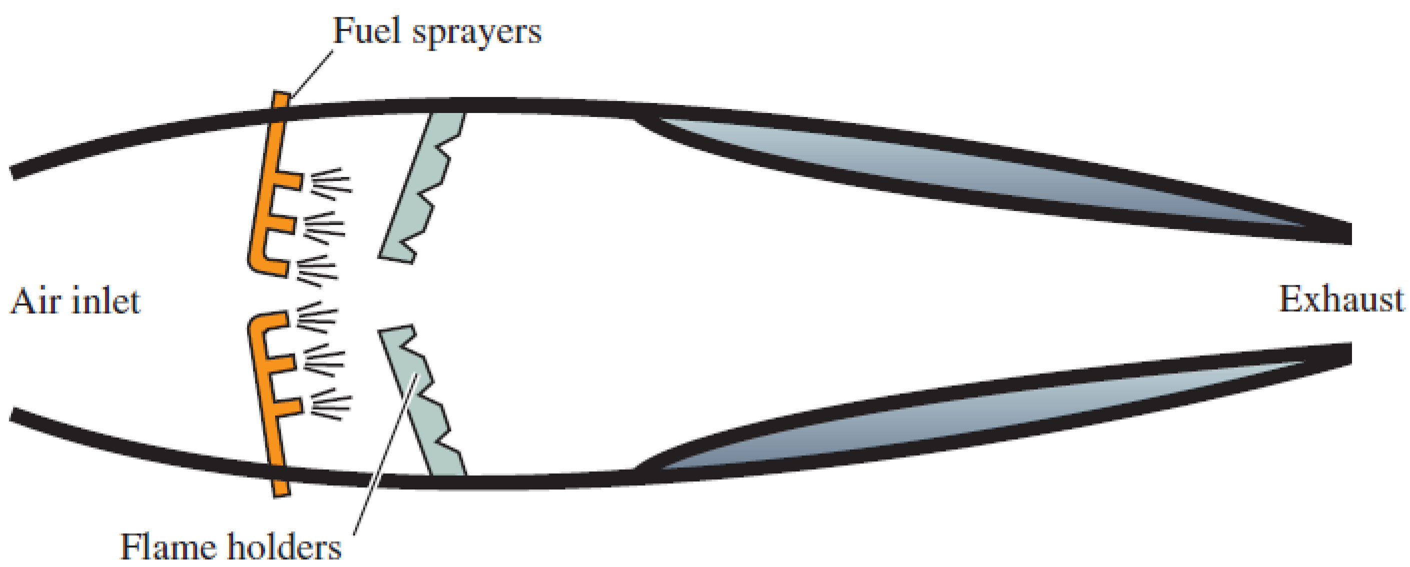


FIGURE 9–55

A ramjet engine.

SECOND-LAW ANALYSIS OF GAS POWER CYCLES

- The ideal Carnot, Ericsson, and Stirling cycles are ***totally reversible***; thus they do not involve any irreversibilities
- The ideal Otto, Diesel, and Brayton cycles, however, are only ***internally reversible***, and they may involve irreversibilities external to the system
- A second-law analysis of these cycles reveals where the largest irreversibilities occur and where to start improvements

EXERGY DESTRUCTION IN A SYSTEM

Exergy destruction for a closed system

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

Rate of exergy destruction for a steady-flow system

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

Steady-flow system, one-inlet, one-exit, unit-mass basis

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

Subscripts *i* and *e* denote the inlet and exit states, respectively

$T_{b,in}$ and $T_{b,out}$ are the temperatures of the **system boundary** where heat is transferred into and out of the system, respectively

EXERGY DESTRUCTION OF A CYCLE

- Exergy destruction of a ***cycle*** is the sum of the exergy destruction of the **processes** that compose that cycle
- Exergy destruction of a cycle can also be determined without tracing the individual processes by considering the entire cycle as a single process
- Entropy is a property and its value depends on the state only. For a cycle, reversible or actual, the initial and final states are identical; thus $s_e = s_i$
- The *exergy destruction of a cycle* depends on the *magnitude of the heat transfer* with the *high- and low-temperature reservoirs* involved and on *their temperatures*

Exergy destruction (unit-mass basis) for a **cycle** that involves heat transfer only with a source at T_H and a sink at T_L:

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

Exergy of a closed system ϕ:

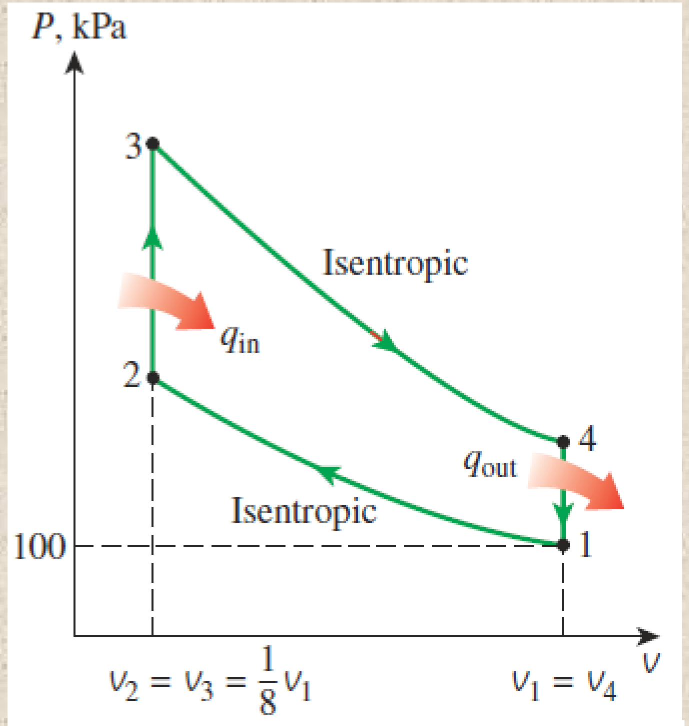
$$\phi = (u - u_0) - T_0(s - s_0) + P_0(v - v_0) + \frac{V^2}{2} + gz$$

Exergy of a fluid stream ψ:

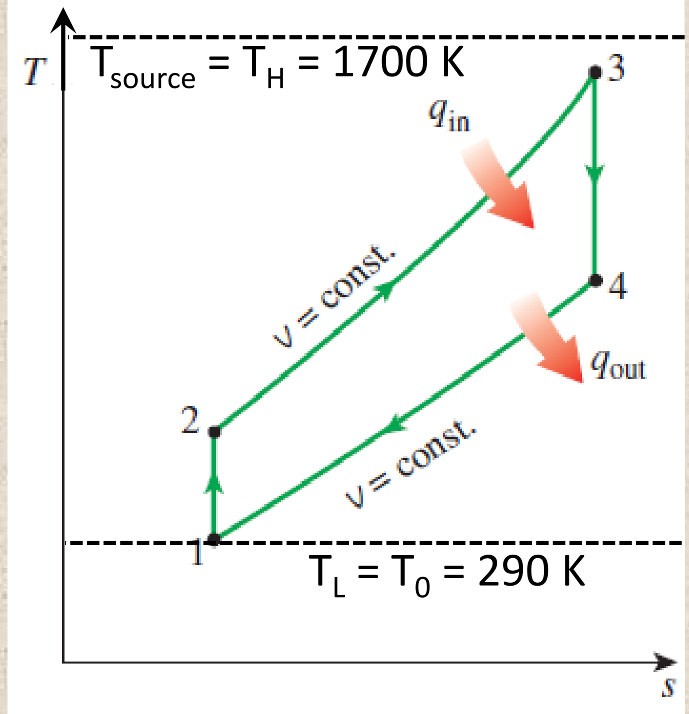
$$\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$$

EXAMPLE: SECOND LAW ANALYSIS OF OTTO CYCLE

Consider an engine operating on the ideal Otto cycle with a compression ratio of 8. At the beginning of the compression process, air is at 100 kPa and 17°C. During the constant-volume heat-addition process, 800 kJ/kg of heat is transferred to air from a source at 1700 K and waste heat is rejected to the surroundings at 290 K. Accounting for the variation of specific heats of air with temperature, determine (*a*) the exergy destruction associated with each of the four processes and the cycle and (*b*) the second-law efficiency of this cycle.



$r = 8$	$P_2 = 1.7997$ MPa
$T_0 = 290$ K	$P_3 = 4.345$ MPa
$T_1 = 290$ K	$q_{in} = 800$ kJ/kg
$T_2 = 652.4$ K	$q_{out} = 381.83$ kJ/kg
$T_3 = 1575.1$ K	$w_{net} = 418.17$ kJ/kg



Processes 1–2 and 3–4 are isentropic ($s_1 = s_2$, $s_3 = s_4$) and therefore do not involve any internal or external irreversibilities; that is, $X_{dest,12} = 0$ and $X_{dest,34} = 0$

Processes 2–3 and 4–1 are constant-volume heat-addition and heat-rejection processes, respectively, and are **internally reversible**. However, the heat transfer between the working fluid and the source or the sink takes place through a finite temperature difference, rendering **both processes irreversible**.

Processes 2–3

$$s_3 - s_2 = s_3^\circ - s_2^\circ - R \ln \frac{P_3}{P_2} = 0.7540 \text{ kJ/kg·K}$$

$$q_{in} = 800 \text{ kJ/kg} \quad \text{and} \quad T_{source} = 1700 \text{ K}$$

$$x_{dest,23} = T_0\left[(s_3 - s_2)_{sys} - \frac{q_{in}}{T_{source}}\right] = \mathbf{82.2 \text{ kJ/kg}}$$

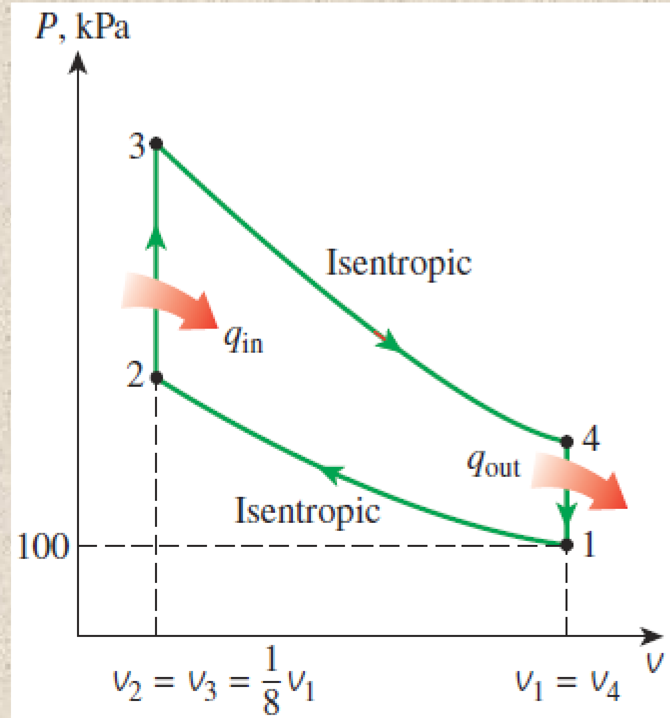
For process 4-1,

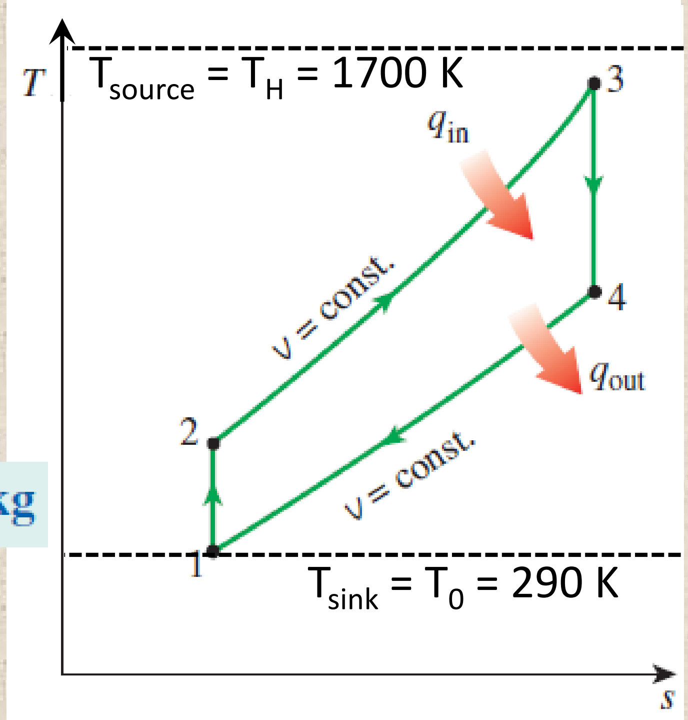
$$s_1 - s_4 = s_2 - s_3 = -0.7540 \text{ kJ/kg·K},$$

$$x_{dest,41} = T_0\left[(s_1 - s_4)_{sys} + \frac{q_{out}}{T_{sink}}\right] = \mathbf{163.2 \text{ kJ/kg}}$$

$$x_{dest,cycle} = x_{dest,12} + x_{dest,23} + x_{dest,34} + x_{dest,41} = \mathbf{245.4 \text{ kJ/kg}}$$

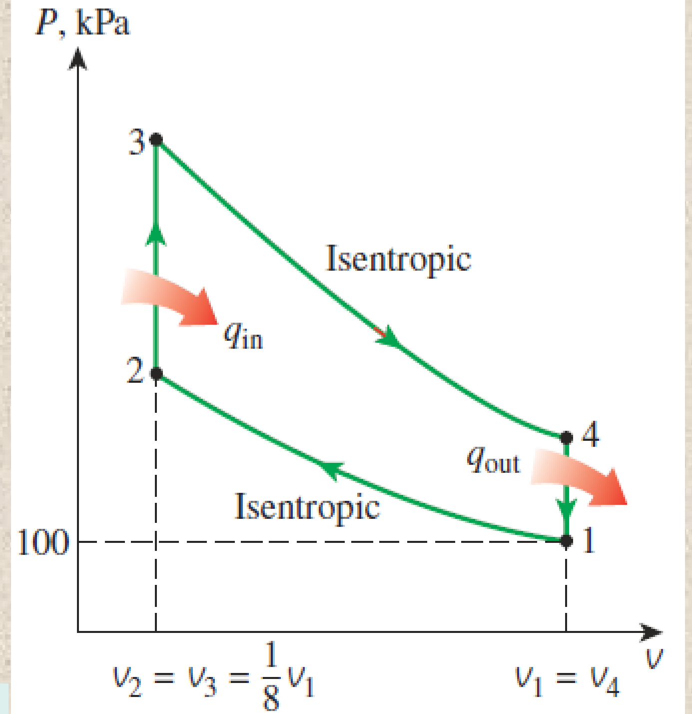
$$x_{dest,cycle} = T_0\left[\frac{q_{out}}{T_{sink}} - \frac{q_{in}}{T_{source}}\right] = 245.4 \text{ kJ/kg}$$

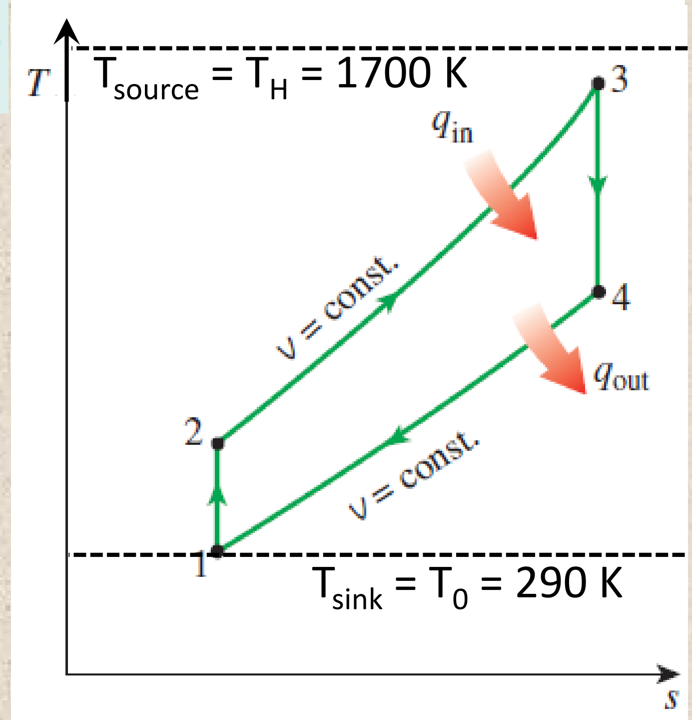




$$x_{\text{expended}} = x_{\text{heat, in}} = \left(1 - \frac{T_0}{T_H}\right) q_{\text{in}} = 663.5 \text{ kJ/kg}$$

$$\eta_{\text{II}} = 1 - \frac{x_{\text{destroyed}}}{x_{\text{expended}}} = 1 - \frac{245.4 \text{ kJ/kg}}{663.5 \text{ kJ/kg}} = 0.630 \text{ or } 63.0\%$$





Summary

- Basic considerations in the analysis of power cycles
- The Carnot cycle and its value in engineering
- Air-standard sssumptions
- 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