Chapter 7

SOME VAPOR POWER CYCLES

Vapour Power Cycle

- The devices or systems used to produce a net power output are often called engines, and the thermodynamic cycles they operate on are called power cycles.
- Thermodynamic cycles can also be categorized as gas cycles and vapour cycles, depending on the phase of the working fluid.
- In gas cycles, the working fluid remains in the gaseous phase throughout the entire cycle, whereas in vapour cycles the working fluid exists in the vapor phase during one part of the cycle and in the liquid phase during another part.

AIR STANDARD EFFICIENCY

- The efficiency of engine using air as the working medium is known as Air Standard Efficiency
- The actual gas power cycles are rather complex. To reduce the analysis to a manageable level, we utilize the following approximations, commonly known as the **air-standard assumptions**

AIR STANDARD ASSUMPTIONS

- The working fluid is air
- In the cycle, all the processes are reversible
- Mass of working fluid remains constant through entire cycle
- The working fluid is homogenous throughout the cycle and no chemical reaction takes place
- The air behaves as an ideal gas and its specific heat is constant at all temperatures
- The cycle is considered closed with the same ‘air’ always remaining in the cylinder to repeat the cycle

CARNOT CYCLE

- This cycle is a hypothetical cycle having highest possible efficiency
- Consists of four simple operations namely:

1. Isothermal expansion
2. Adiabatic expansion
3. Isothermal compression
4. Adiabatic compression

ASSUMPTIONS MADE IN CARNOT CYCLE

- The piston moving in the cylinder does not produce any friction during motion
- The cylinder head is arranged such a way that it can be perfect heat conductor or insulator
- The walls of cylinder and piston are considered as perfect insulators of heat
- Heat transfer does not affect the temperature of source or sink
- Compression and expansion processes are reversible
- Working medium is a perfect gas and has constant specific heat

Is Carnot Cycle Practical?

The Carnot cycle is **NOT** a suitable model for actual power cycles because of several **impracticalities** associated with it:

Process 1-2

Limiting the heat transfer processes to **two-phase systems** severely limits the maximum temperature that can be used in the cycle (374°C for water).

Process 2-3

The turbine cannot handle steam with a high **moisture content** because of the impingement of liquid droplets on the turbine blades causing **erosion** and **wear.**

Process 4-1

It is not practical to design a compressor that handles two phases.

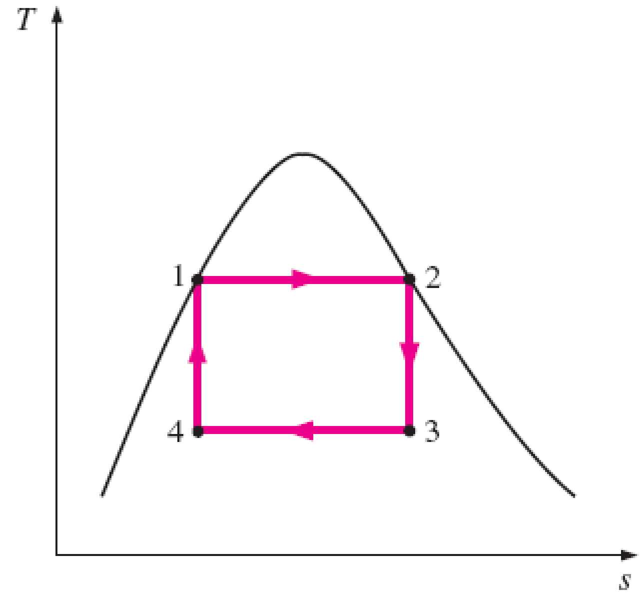


FIGURE 10–1

T-s diagram of two Carnot vapor cycles.

Carnot Vapor Cycle

Carnot cycle is the **most efficient** power cycle operating between two specified temperature limits (Fig. 10-1).

We can adopt the Carnot cycle first as a prospective **ideal cycle** for vapor power plants.

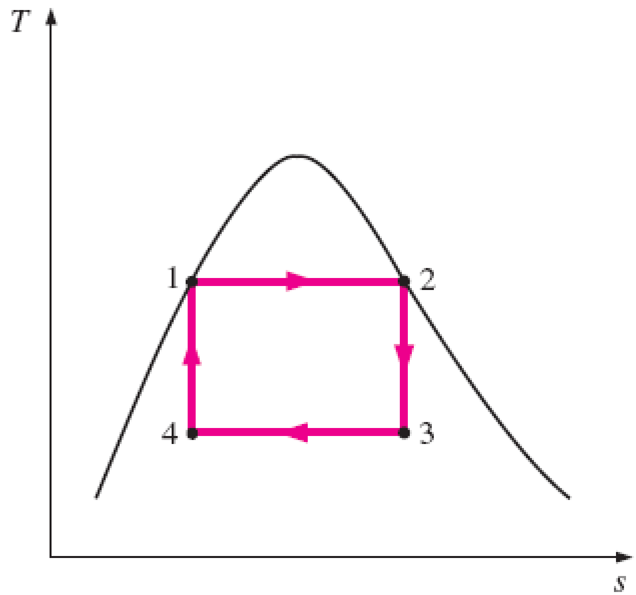


FIGURE 10–1

T-s diagram of two Carnot vapor cycles.

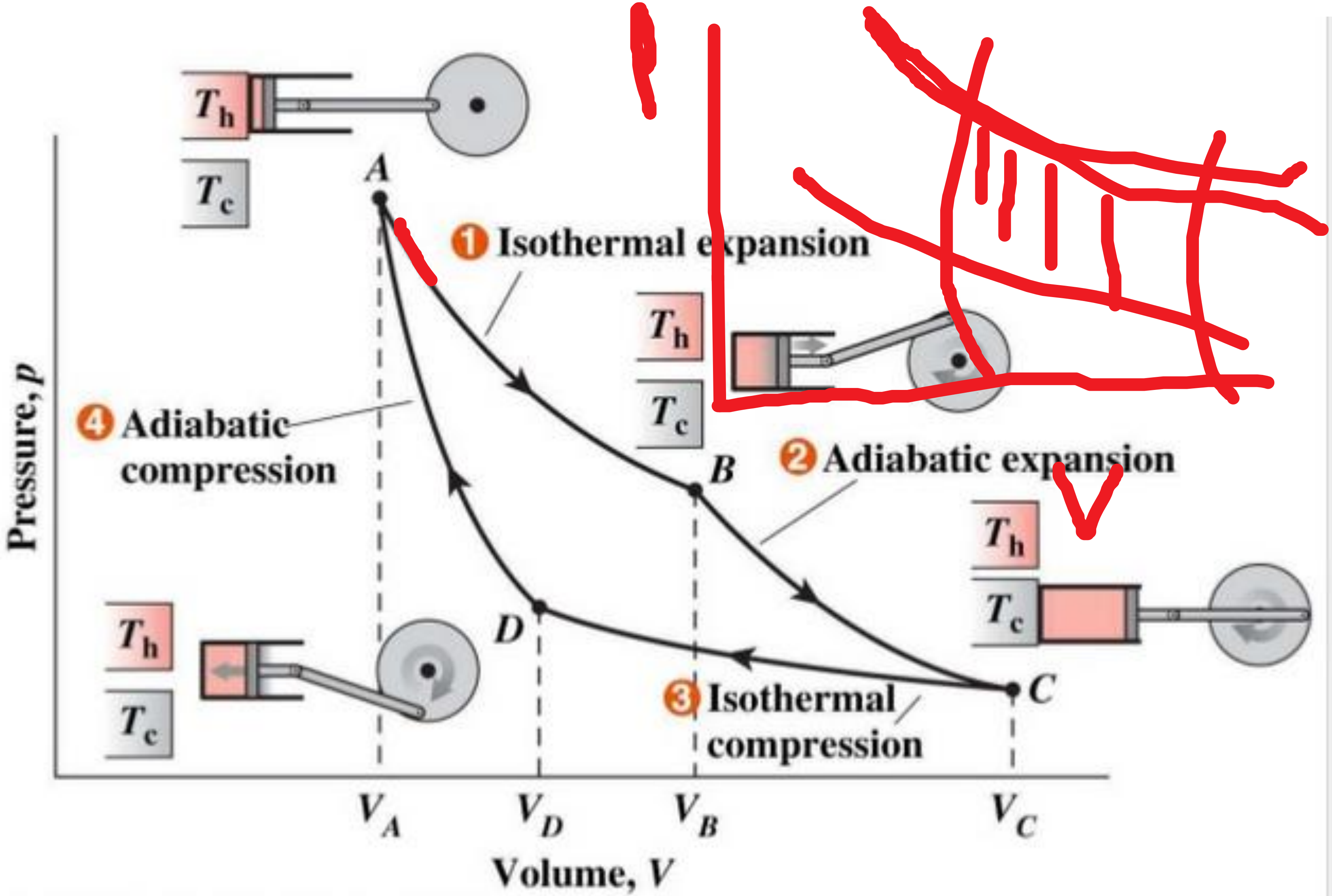
Sequence of Processes:

1-2 Reversible and isothermal heating (in a boiler);

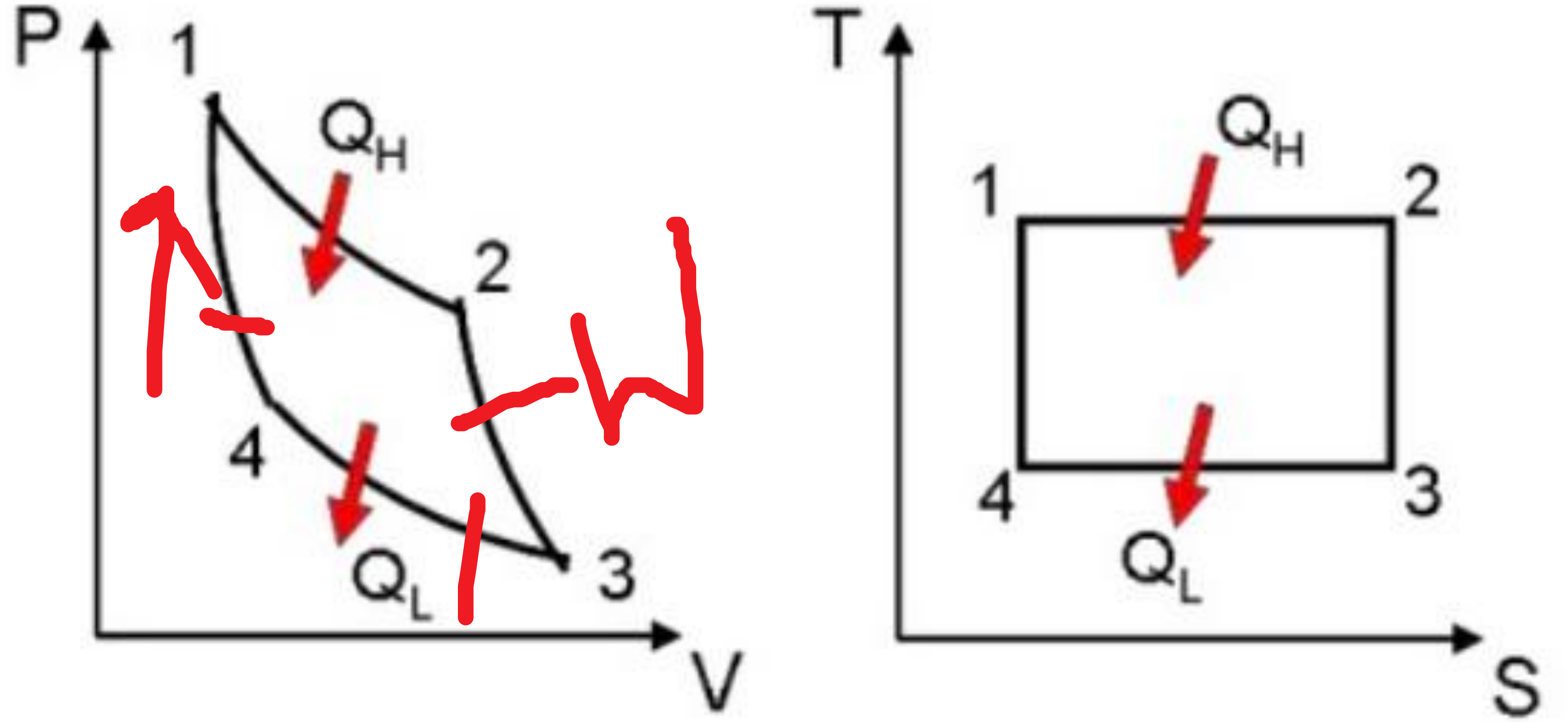
2-3 Isentropic expansion (in a turbine);

3-4 Reversible and isothermal condensation (in a condenser); and

4-2 Isentropic compression (in a compressor).



T_h
T_c
A
1 Isothermal expansion
B
2 Adiabatic expansion
C
3 Isothermal compression
D
4 Adiabatic compression
Pressure, p
V_A
V_D
V_B
V_C
Volume, V



$$\eta_{th} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{T_L}{T_H}$$

13.3. THE CARNOT CYCLE

This cycle has the *highest possible efficiency* and consists of four simple operations namely,

(*a*) Isothermal expansion

(*b*) Adiabatic expansion

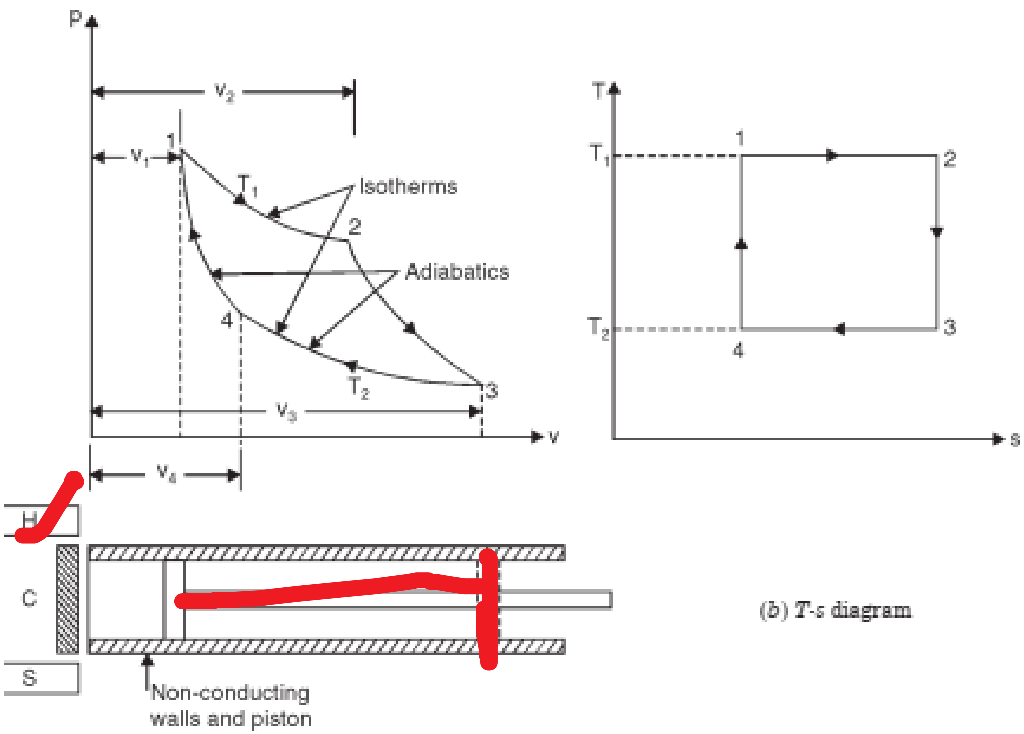
(*c*) Isothermal compression

(*d*) Adiabatic compression.

The condition of the Carnot cycle may be imagined to occur in the following way :

One kg of a air is enclosed in the cylinder which (except at the end) is made of perfect non-conducting material. A source of heat '*H*' is supposed to provide unlimited quantity of heat, non-conducting cover '*C*' and a sump '*S*' which is of infinite capacity so that its temperature remains unchanged irrespective of the fact how much heat is supplied to it. The temperature of source *H* is T_1 and the same is of the working substance. The working substance while rejecting heat to sump '*S*' has the temperature. T_2 *i.e.*, the same as that of sump *S*.

Following are the *four stages* of the Carnot cycle. Refer Fig. 13.1 (*a*).



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

(*a*) Four stages of the carnot cycle

Stage (1). Line 1-2 [Fig. 13.1 (*a*)] represents the isothermal expansion which takes place at temperature T_1 when source of heat *H* is applied to the end of cylinder. Heat supplied in this case is given by $RT_1 \log_e r$ and where r is the ratio of expansion.

Stage (2). Line 2-3 represents the application of non-conducting cover to the end of the cylinder. This is followed by the adiabatic expansion and the temperature falls from T_1 to T_2.

Stage (3). Line 3-4 represents the isothermal compression which takes place when sump '*S*' is applied to the end of cylinder. Heat is rejected during this operation whose value is given by $RT_2 \log_e r$ where r is the ratio of compression.

Stage (4). Line 4-1 represents repeated application of non-conducting cover and adiabatic compression due to which temperature increases from T_2 to T_1.

It may be noted that ratio of expansion during isotherm 1-2 and ratio of compression during isotherm 3-4 must be equal to get a closed cycle.

Fig. 13.1 (*b*) represents the Carnot cycle on *T-s* coordinates.

PV = RT

Now according to law of conservation of energy,

Heat supplied = Work done + Heat rejected

Work done = Heat supplied − Heat rejected

$$= RT_1 \, . \log_e r - RT_2 \log_e r$$

$Q_1 - Q_2 = W$

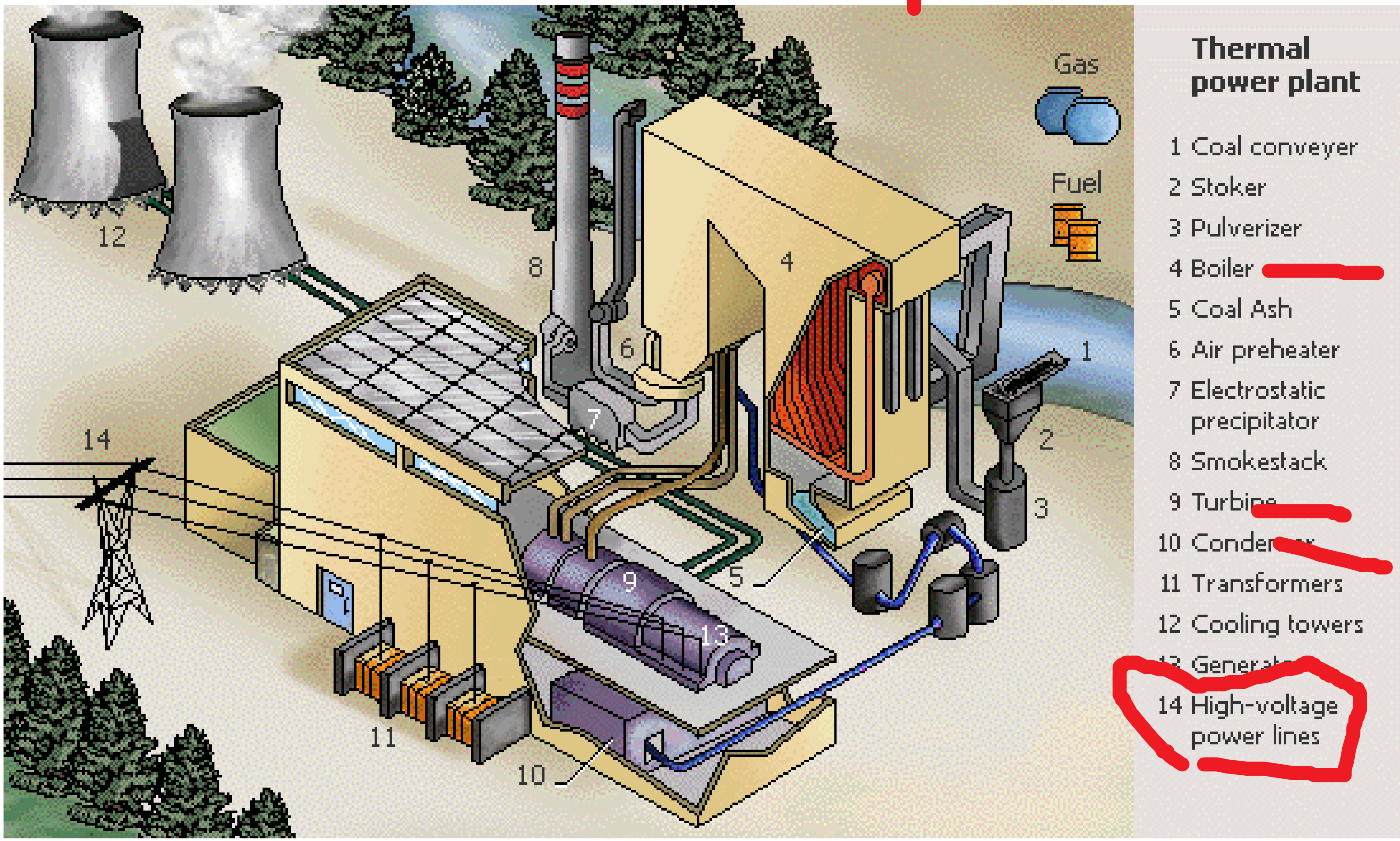
$$\text{Efficiency of cycle} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{R \log_e r \,(T_1 - T_2)}{RT_1 \, . \log_e r}$$

$$= \frac{T_1 - T_2}{T_1} \qquad \text{...(13.2)}$$

$1 - T_2/T_1$

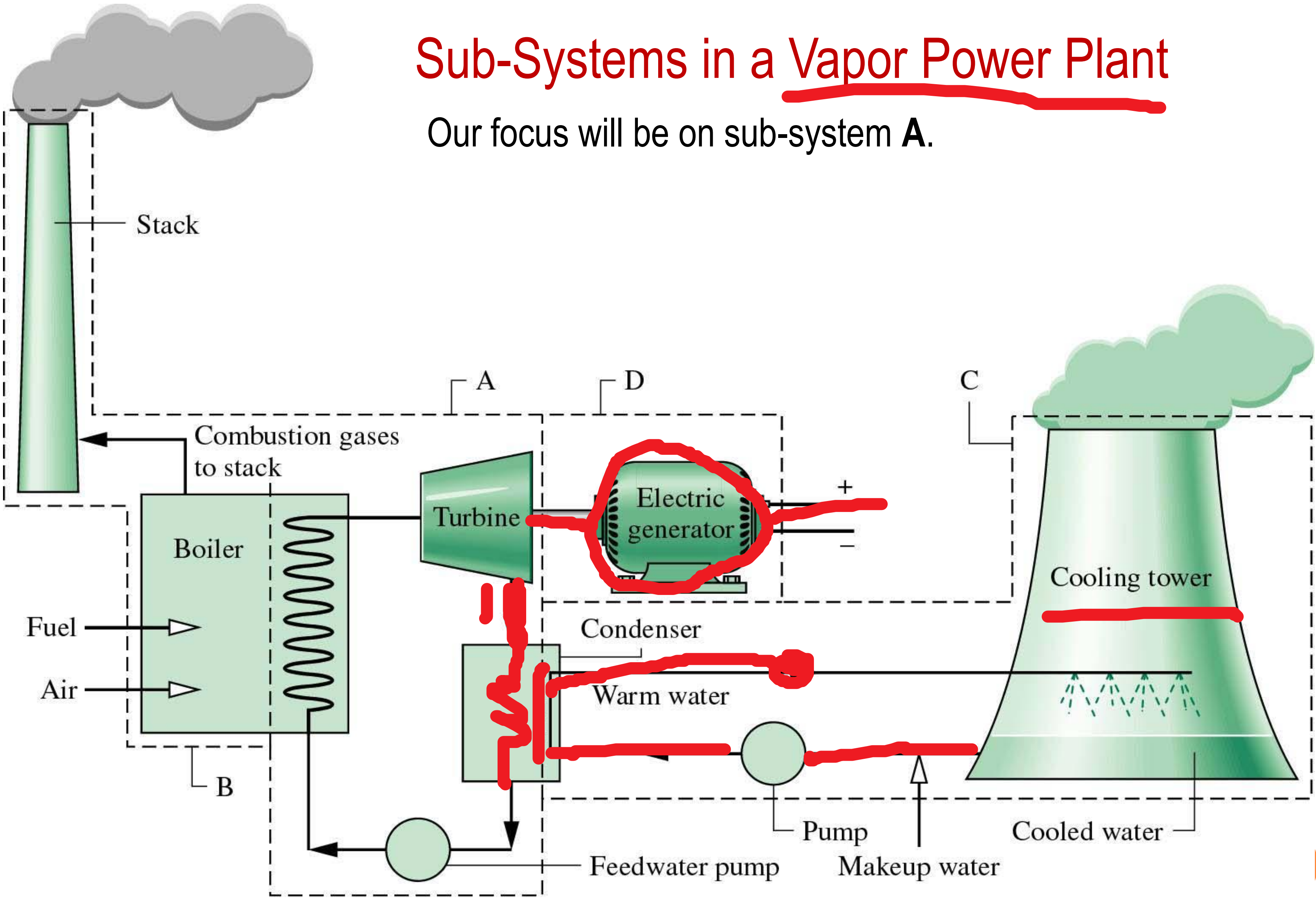
From this equation, it is quite obvious that if temperature T_2 decreases efficiency increases

Thermal Power Plant



Sub-Systems in a Vapor Power Plant

Our focus will be on sub-system **A**.

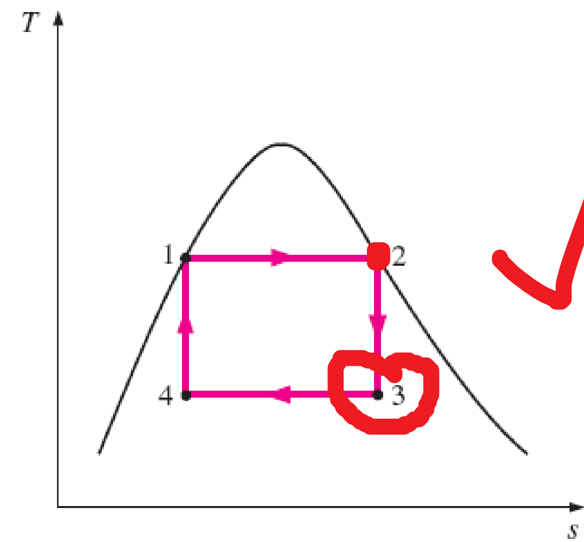


The Rankine Cycle

Many of the impracticalities associated with the Carnot cycle can be eliminated by: (a) **superheating** the steam in the boiler, and (b) condensing the steam **completely** in the condenser.

The modified Carnot cycle is called the **Rankine cycle**, which is the ideal and practical cycle for vapor power plants (Figure 10-2).

This ideal cycle does not involve any internal irreversibilities.



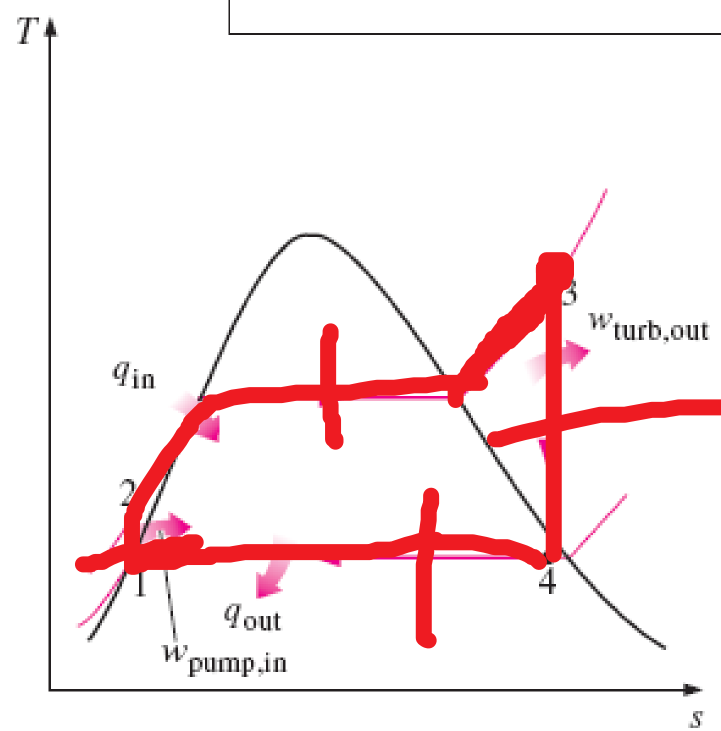


FIGURE 10–2

The simple ideal Rankine cycle.

Sequence of Processes

The ideal Rankine cycle consists of **four** processes:

1-2 Isentropic compression in a water pump;

2-3 Constant pressure heat addition in a boiler;

3-4 Isentropic expansion in a turbine;

4-1 Constant pressure heat rejection in a condenser.

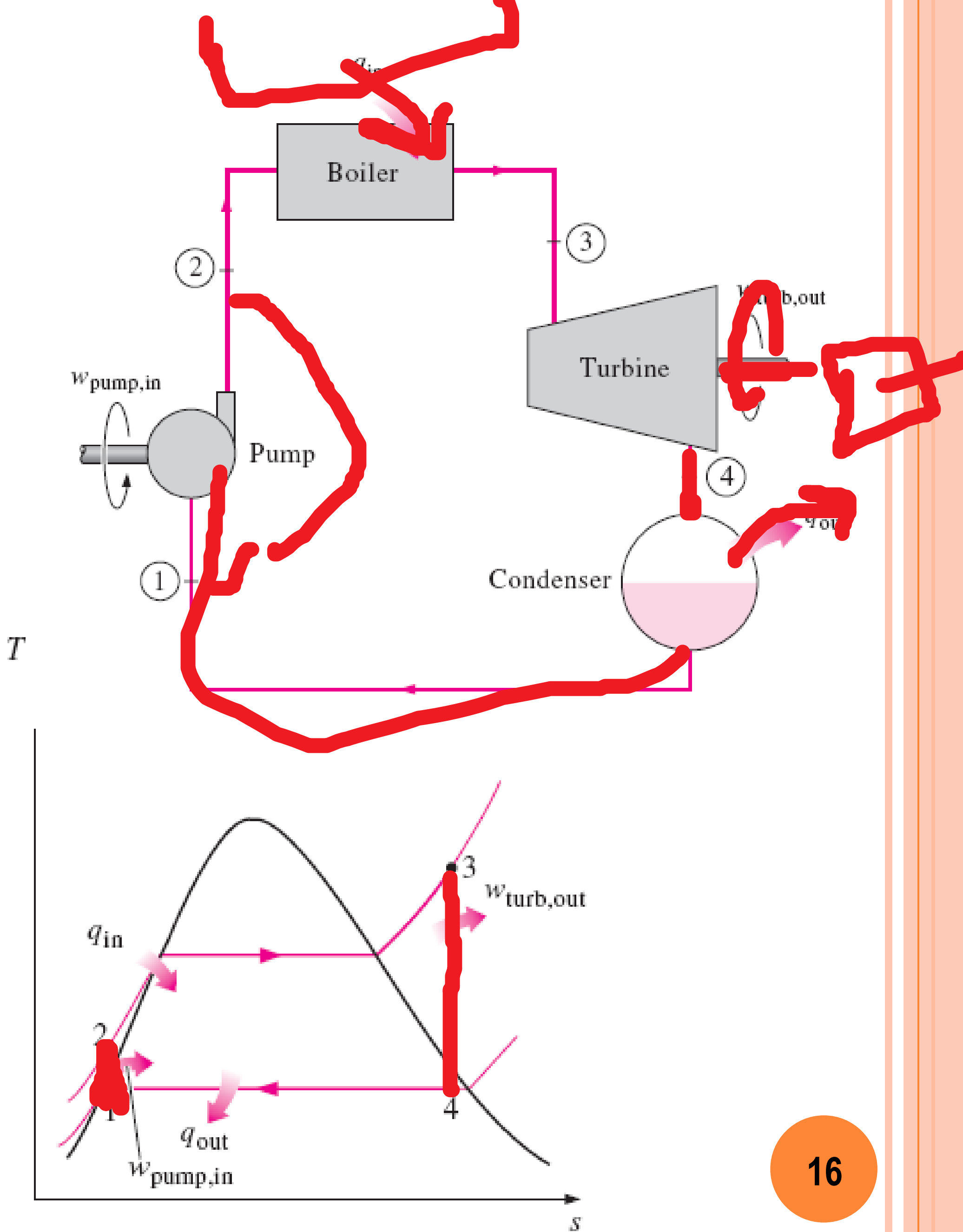


FIGURE 10–2

The simple ideal Rankine cycle.

SEE TUTORIAL CH. 10, SEC. 1 ON T

Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser, as shown schematically on a *T-s* diagram in Fig. 10–2. The cycle that results is the **Rankine cycle,** which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:

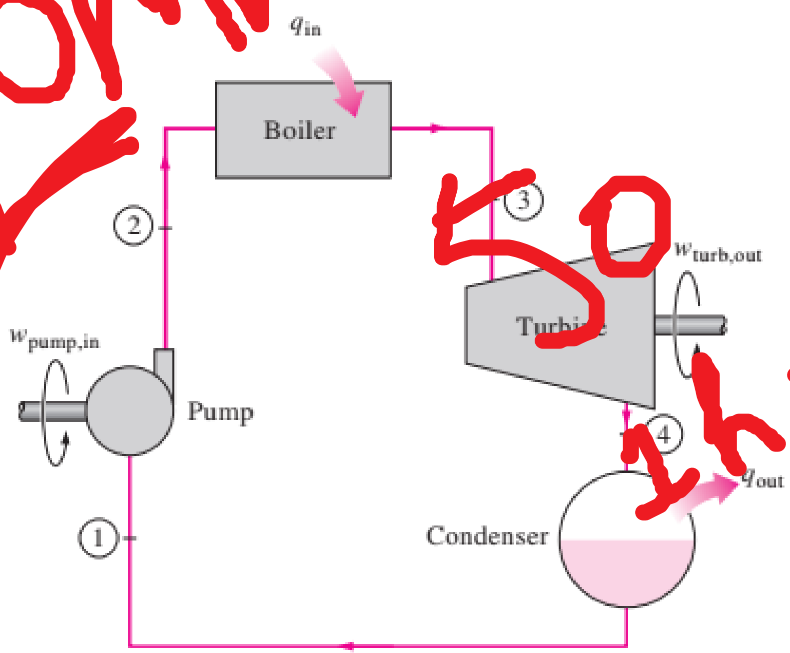
1-2 Isentropic compression in a pump
2-3 Constant pressure heat addition in a boiler
3-4 Isentropic expansion in a turbine
4-1 Constant pressure heat rejection in a condenser

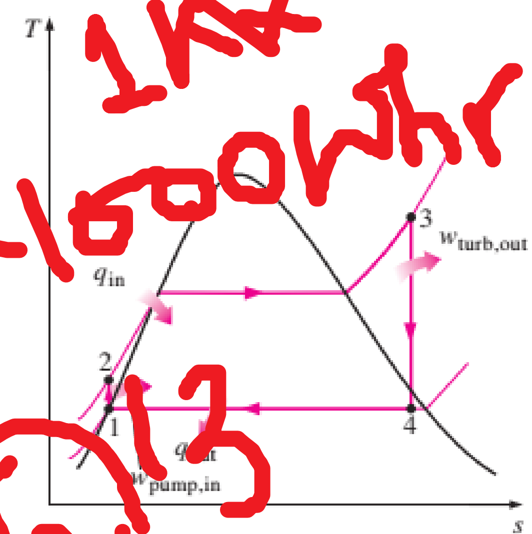
50

40 → 42MW

40MW

50





1kW

1h × 1000whr

$\eta = 1 - \frac{T_2}{T_1}$

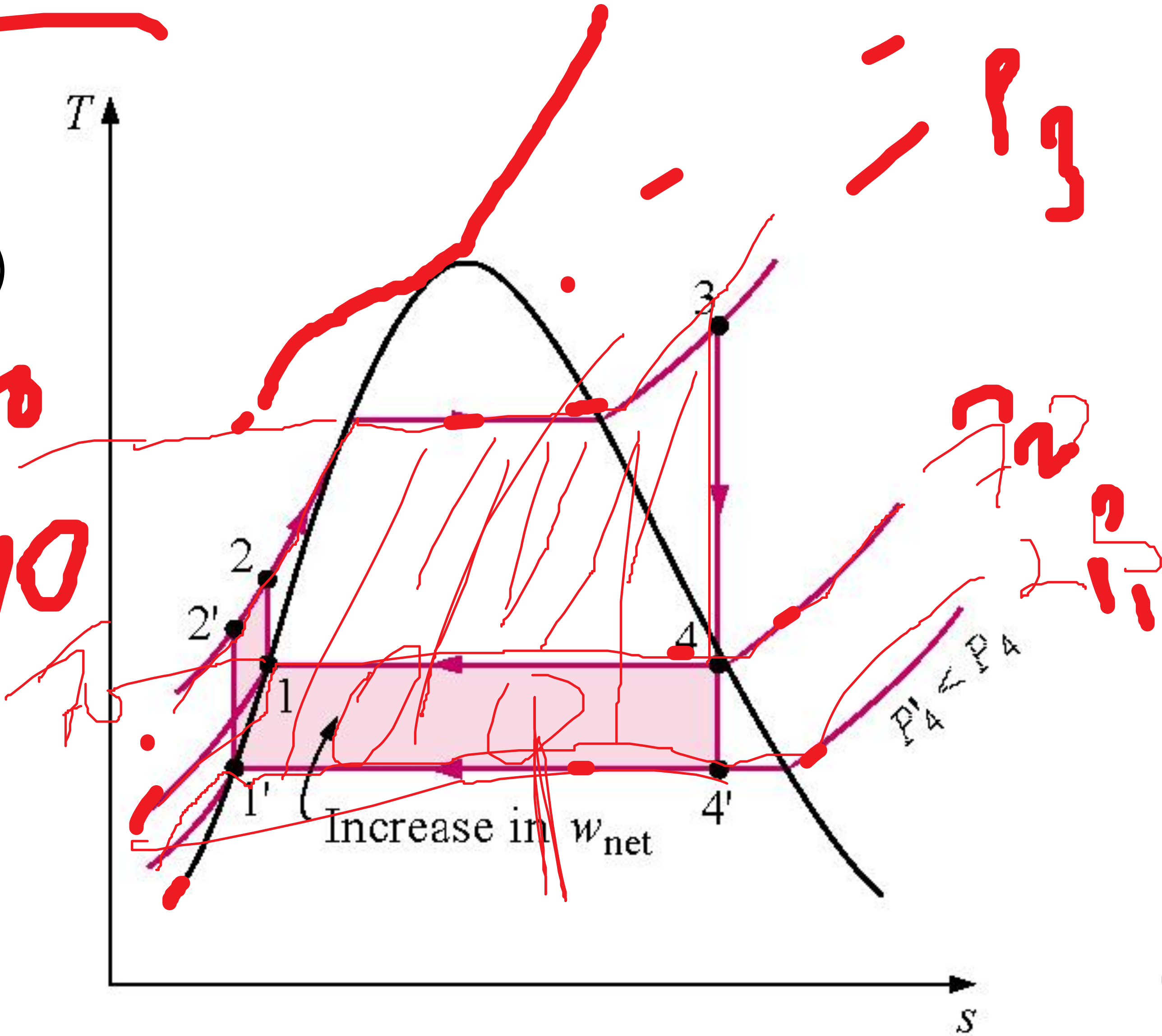
10–4 ▪ HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

Steam power plants are responsible for the production of most electric power in the world, and even small increases in thermal efficiency can mean large savings from the fuel requirements. Therefore, every effort is made to improve the efficiency of the cycle on which steam power plants operate.

The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same: *Increase the average temperature at which heat is transferred to the working fluid in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser.* That is, the average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection. Next we discuss three ways of accomplishing this for the simple ideal Rankine cycle.

Effect of Lowering Condenser Pressure on the Ideal Rankine Cycle

- (Fig. 9-6)



Lowering the Condenser Pressure (*Lowers* $T_{low,avg}$)

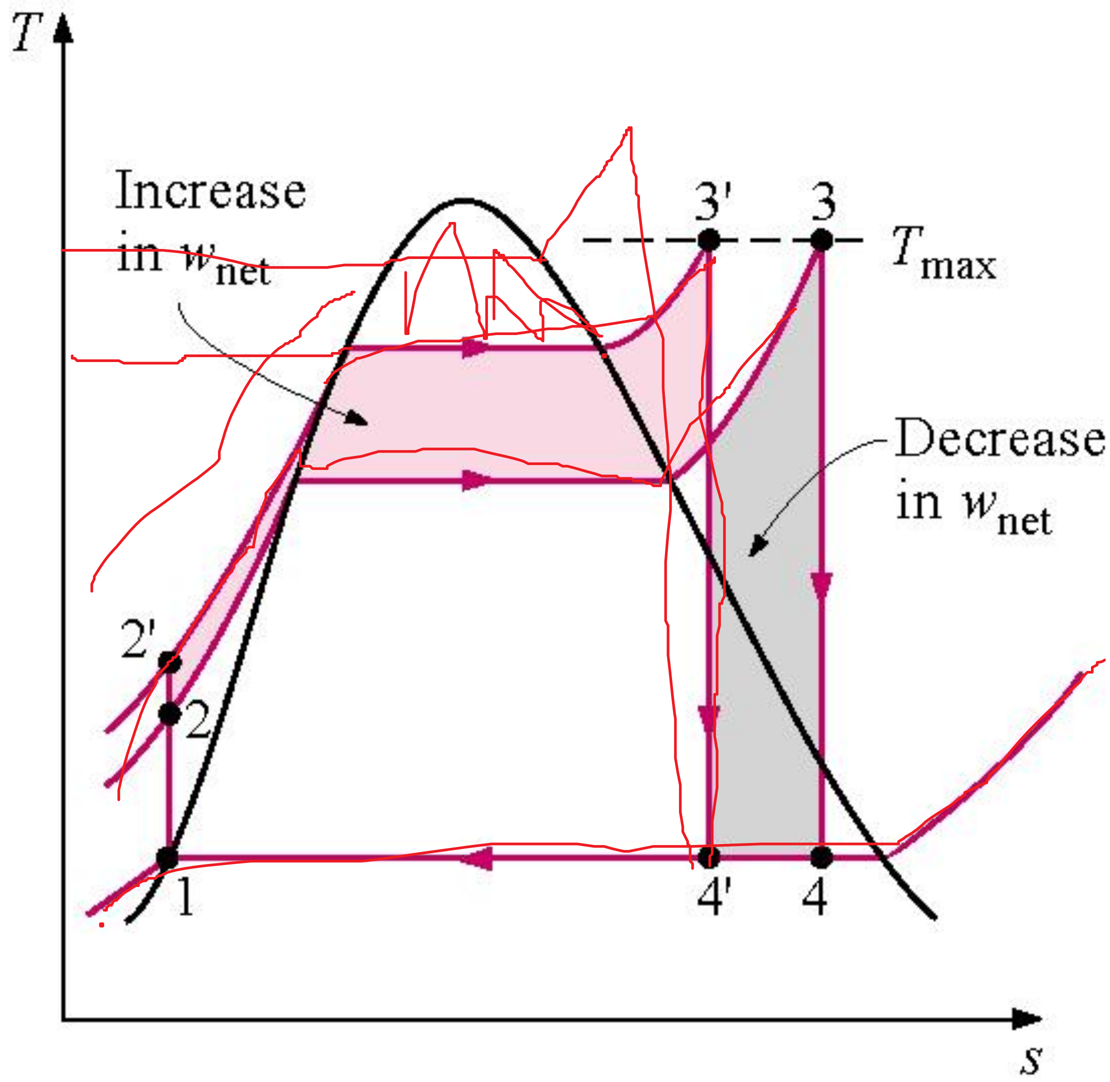
Steam exists as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. Therefore, lowering the operating pressure of the condenser automatically lowers the temperature of the steam, and thus the temperature at which heat is rejected.

The effect of lowering the condenser pressure on the Rankine cycle efficiency is illustrated on a *T-s* diagram in Fig. 10–6. For comparison purposes, the turbine inlet state is maintained the same. The colored area on this diagram represents the increase in net work output as a result of lowering the condenser pressure from P_4 to P_4'. The heat input requirements also increase (represented by the area under curve 2′-2), but this increase is very small. Thus the overall effect of lowering the condenser pressure is an increase in the thermal efficiency of the cycle.

Superheating the Steam to High Temperatures ($Increases\ T_{high,avg}$)

The average temperature at which heat is transferred to steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of vapor power cycles is illustrated on a *T-s* diagram in Fig. 10–7. The colored area on this diagram represents the increase in the net work. The total area under the process curve 3-3′ represents the increase in the heat input. Thus both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency, however, since the average temperature at which heat is added increases.

Effect of Increasing Boiler Pressure on the Ideal Rankine cycle



Increasing the Boiler Pressure (*Increases* $T_{high,avg}$)

Another way of increasing the average temperature during the heat-addition process is to increase the operating pressure of the boiler, which automatically raises the temperature at which boiling takes place. This, in turn, raises the average temperature at which heat is transferred to the steam and thus raises the thermal efficiency of the cycle.

The effect of increasing the boiler pressure on the performance of vapor power cycles is illustrated on a *T-s* diagram in Fig. 10–8. Notice that for a fixed turbine inlet temperature, the cycle shifts to the left and the moisture content of steam at the turbine exit increases. This undesirable side effect can be corrected, however, by reheating the steam, as discussed in the next section.

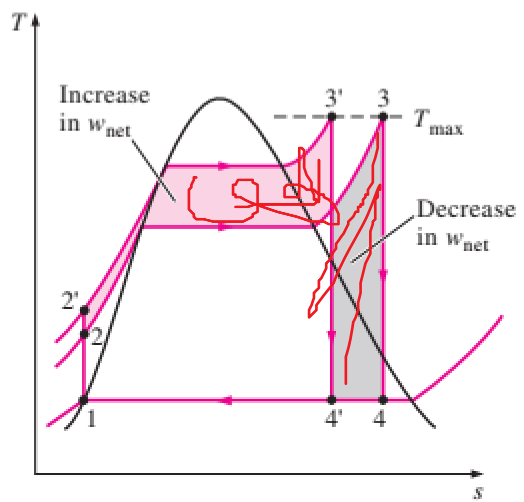


FIGURE 10–8

The effect of increasing the boiler pressure on the ideal Rankine cycle.

Energy Interactions in Each Device

SFEE

Pump: The work needed to operate the water pump,

DP

$$w_{\text{pump,in}} = h_2 - h_1$$

$$w_{\text{pump,in}} = v(P_2 - P_1)$$

where,

$$h_1 = h_{f\,@\,P_1} \quad \text{and} \quad v \cong v_1 = v_{f\,@\,P_1}$$

Boiler: The amount of heat supplied in the steam boiler,

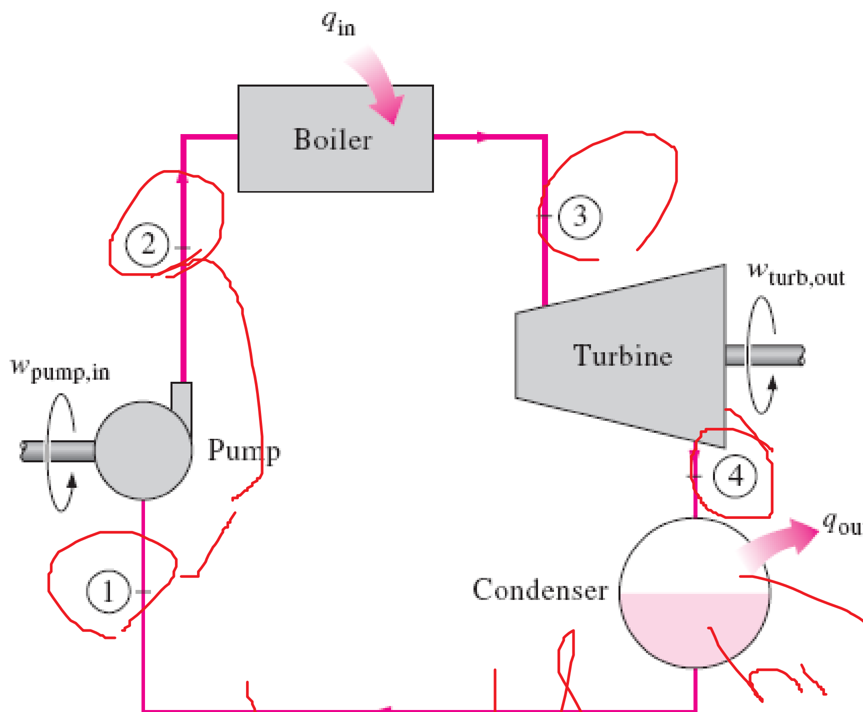
$$q_{\text{in}} = h_3 - h_2$$

Turbine: The amount of work produced by the turbine,

$$w_{\text{turb,out}} = h_3 - h_4$$

Condenser: The amount of heat rejected to cooling medium in the condenser,

$$q_{\text{out}} = h_4 - h_1$$



Performance of Ideal Rankine Cycle

Thermal Efficiency

The thermal efficiency of the Rankine cycle is determined from,

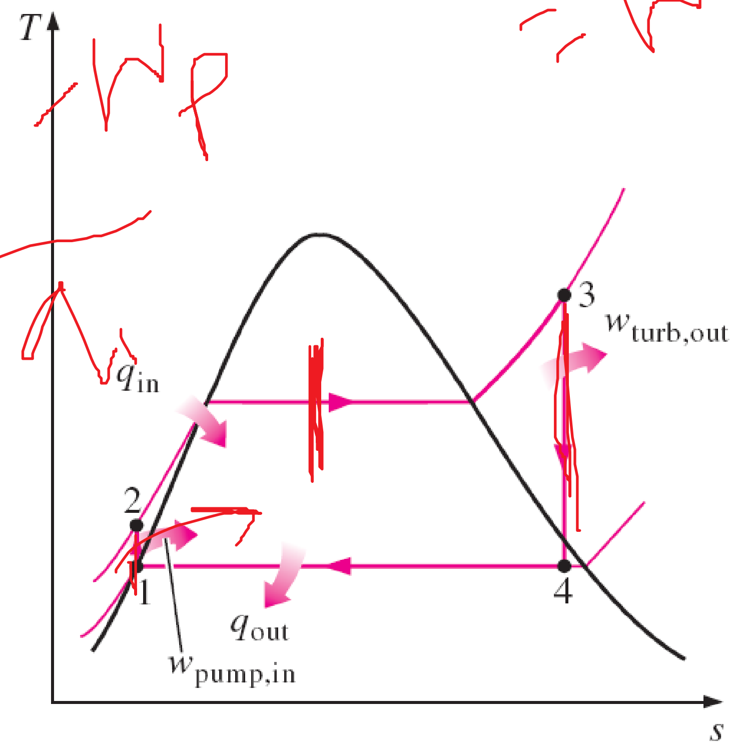
$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

where the **net work** output,

$$w_{net} = q_{in} - q_{out} = w_{turb,out} - w_{pump,in}$$

Wnet

Note: +ve quantities only!



Thermal efficiency of Rankine cycle can also be interpreted as the ratio of the area enclosed by the cycle on a *T-s* diagram to the area under the heat-addition process.

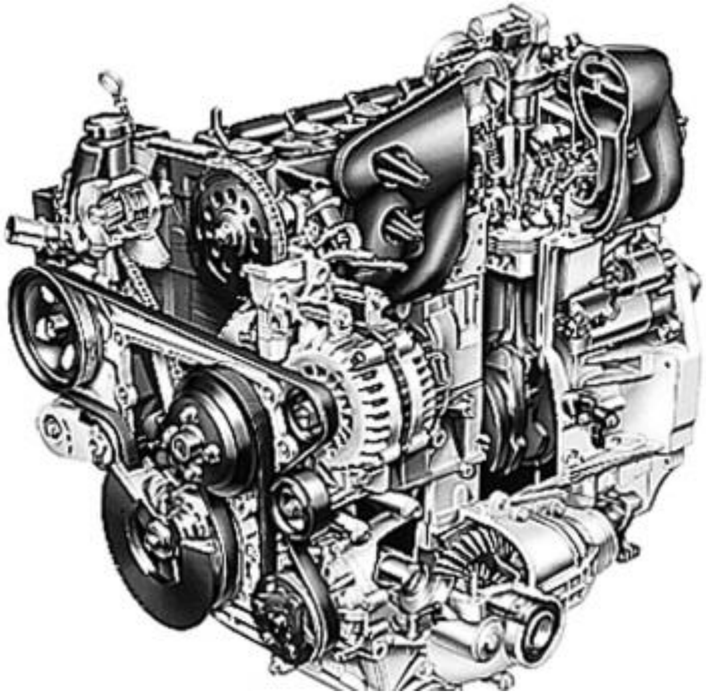


FIGURE 9–4

An automotive engine with the combustion chamber exposed.

9–1 ▪ BASIC CONSIDERATIONS IN THE ANALYSIS OF POWER CYCLES

Most power-producing devices operate on cycles, and the study of power cycles is an exciting and important part of thermodynamics. The cycles encountered in actual devices are difficult to analyze because of the presence of complicating effects, such as friction, and the absence of sufficient time for establishment of the equilibrium conditions during the cycle. To make an analytical study of a cycle feasible, we have to keep the complexities at a manageable level and utilize some idealizations (Fig. 9–1). When the actual cycle is stripped of all the internal irreversibilities and complexities, we end up with a cycle that resembles the actual cycle closely but is made up totally of internally reversible processes. Such a cycle is called an **ideal cycle** (Fig. 9–2).

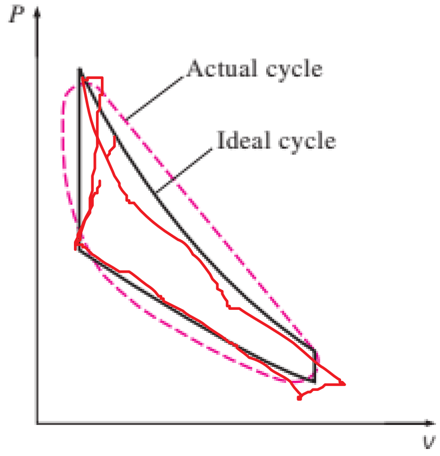


FIGURE 9–2

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

for a more in-depth study.

Heat engines are designed for the purpose of converting thermal energy to work, and their performance is expressed in terms of the **thermal efficiency** η_{th}, which is the ratio of the net work produced by the engine to the total heat input:

$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}} \tag{9–1}$$

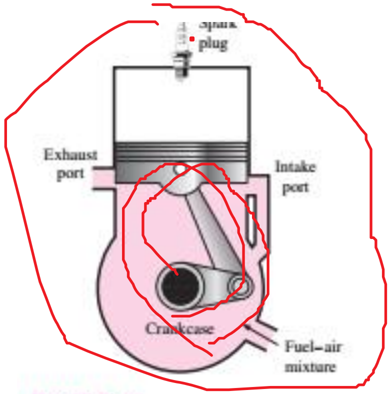


FIGURE 9–14

Schematic of a two-stroke reciprocating engine.

FIGURE 9–15

Two-stroke engines are commonly used in motorcycles and lawn mowers.

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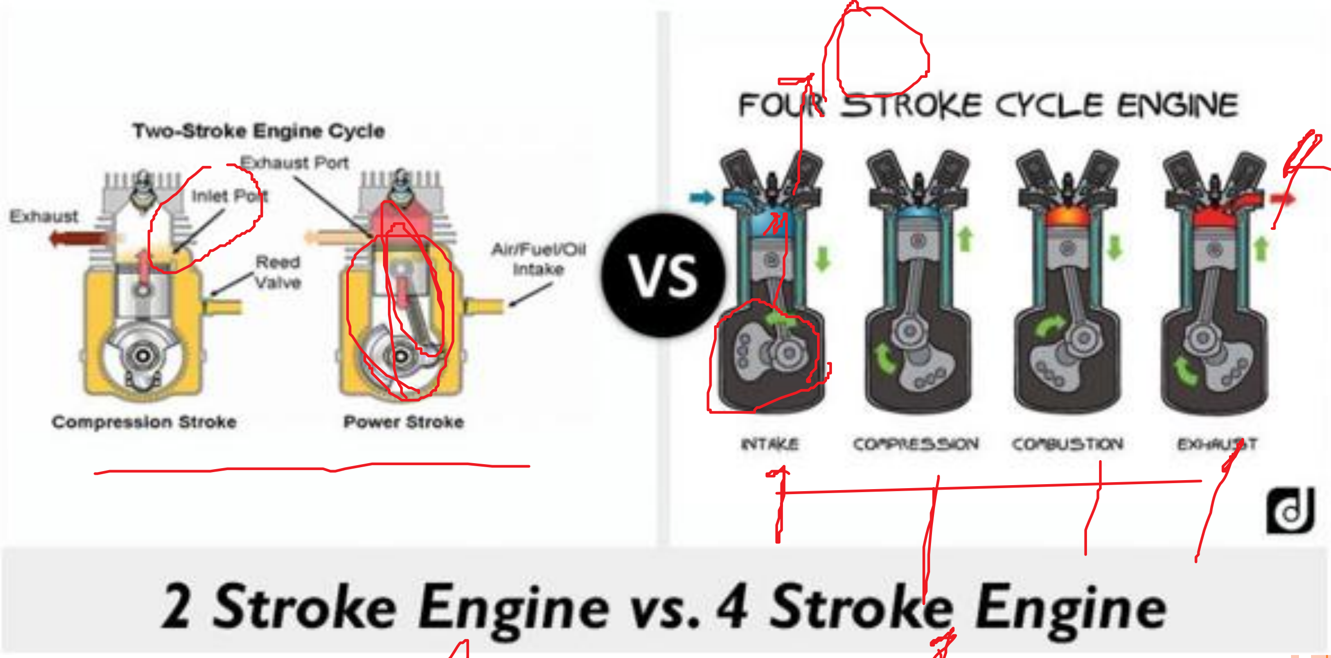

DIFERRENECE BETWEEN 2 AND 4 STROKE ENGINE (PART 1)

S. No.	Two Stroke Engine	Four Stroke Engine
1.	It has one revolution of crankshaft within one power stroke.	It has two revolution of crankshaft between one power strokes.
2.	It can generate high torque compare to 4 strokes engine.	It generates less torque due to 2 revolution of crankshaft between one power strokes.
3.	It used port to inlet and outlet of fuel.	It used valve to inlet and outlet.
4.	2 stroke engines require lighter flywheel compare to other engines because it generates more balanced force due to one revolution for one power stroke.	It requires heavy flywheel because it generates unbalance force due to two revolutions for one power stroke.
5.	The charge is partially burn and mix with the burn gases during inlet. It is due to port mechanism.	In four stroke engine charge is fully burn and does not mix with burn charge in ideal condition.
6.	Easy lubrication due to lubrication oil mix with the fuel.	Comparatively complicated lubrication.
7.	More lubricating oil requires because some oil burns with fuel.	Comparatively less lubricating oil requires.
8.	These engines give less thermal efficiency.	These engines give more thermal efficiency.
9.	It has high power to weight ratio compare to others.	4 stroke engines have less power to weight ratio.
10.	It creates more noise.	It is less noisy.
11.	Two stroke engines are less efficient and generate more smoke.	Four stroke engines are more efficient and generate less smoke.

12.	These engines are comparatively cheaper.	These engines are expansive due to valve and lubrication mechanism.
13.	These engines are easy to manufacture.	These engines are comparatively hard to manufacture.
14.	These engines are generally lighter.	These engines are comparatively heavier than 2 strokes due to heavy flywheel and valve mechanism.
15.	These are mostly used in ships, scooters etc.	These engines mostly used in car, truck, and other automobiles.
16.	Due to poor lubrication more wear and tear occurs	Less wear and tear occurs.

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

The Otto cycle is the ideal cycle for spark-ignition reciprocating engines. It is named after Nikolaus A. Otto, who built a successful four-stroke engine in 1876 in Germany using the cycle proposed by Frenchman Beau de Rochas in 1862. In most spark-ignition engines, the piston executes four complete strokes (two mechanical cycles) within the cylinder, and the crankshaft completes two revolutions for each thermodynamic cycle. These engines are called **four-stroke** internal combustion engines. A schematic of each stroke as well as a P-v diagram for an actual four-stroke spark-ignition engine is given in Fig. 9–13(a).

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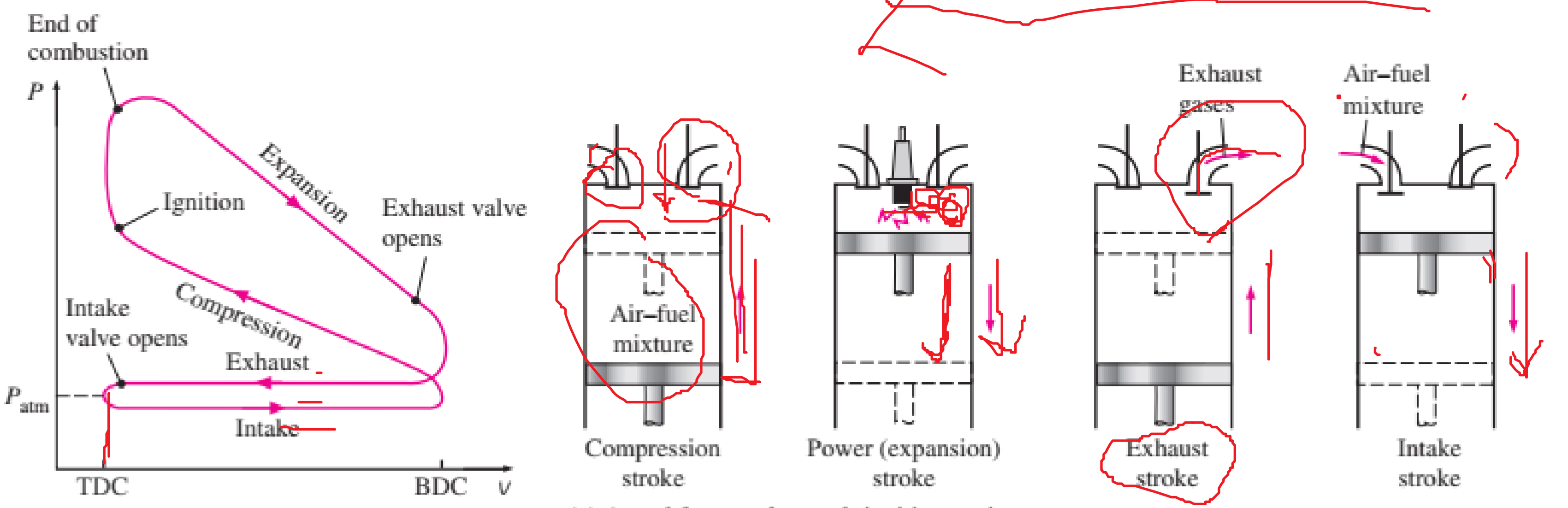
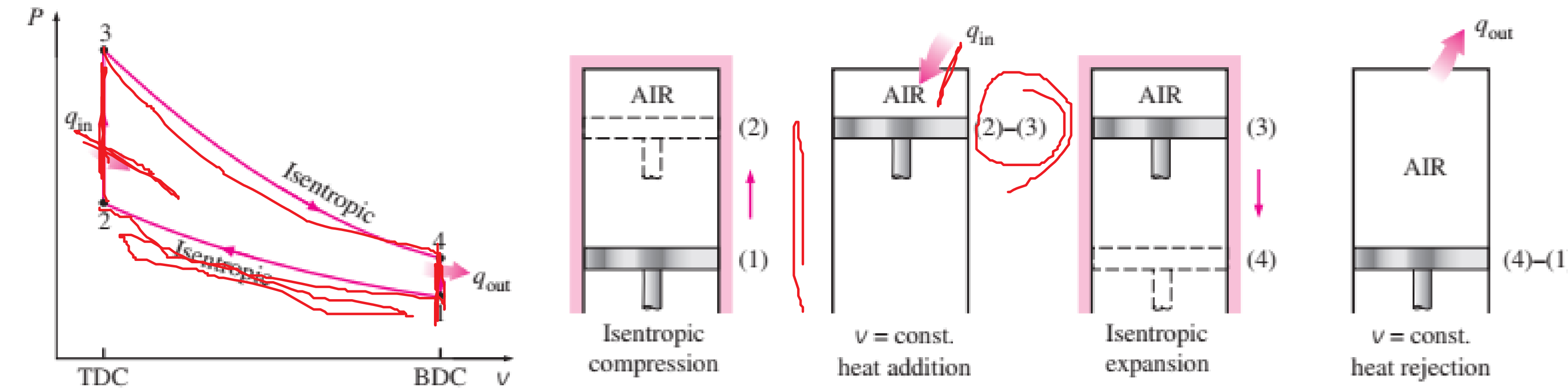


(*a*) Actual four-stroke spark-ignition engine



(*b*) Ideal Otto cycle

Major car companies have research programs underway on two-stroke engines which are expected to make a comeback in the future.

The thermodynamic analysis of the actual four-stroke or two-stroke cycles described is not a simple task. However, the analysis can be simplified significantly if the air-standard assumptions are utilized. The resulting cycle, which closely resembles the actual operating conditions, is the ideal **Otto cycle.** It consists of four internally reversible processes:

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

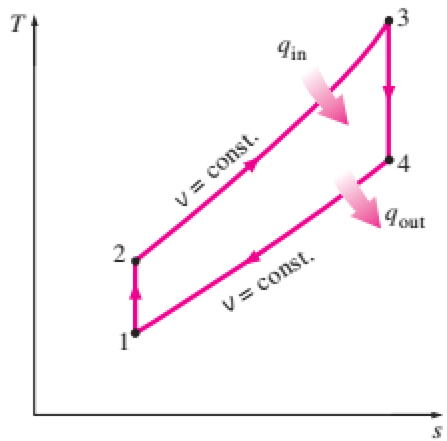


FIGURE 9–16

T-s diagram of the ideal Otto cycle.

The execution of the Otto cycle in a piston–cylinder device together with a P-v diagram is illustrated in Fig. 9–13*b*. The T-s diagram of the Otto cycle is given in Fig. 9–16.

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

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

No work is involved during the two heat transfer processes since both take place at constant volume. Therefore, heat transfer to and from the working fluid can be expressed as

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2) \tag{9–6a}$$

and

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \tag{9–6b}$$

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

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

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

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

Substituting these equations into the thermal efficiency relation and simplifying give

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

where

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

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

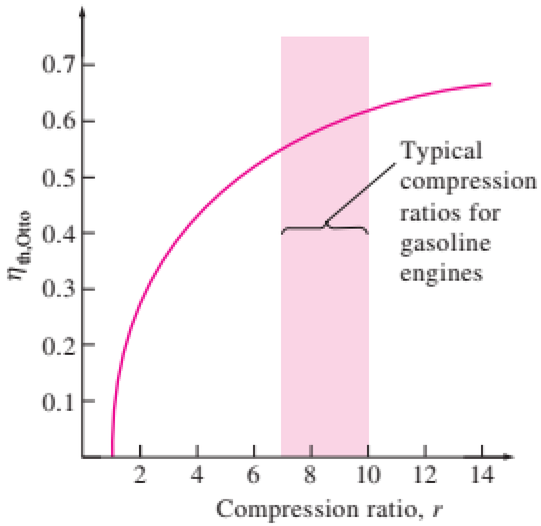


FIGURE 9–17

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

The thermal efficiency of the ideal Otto cycle increases with both the compression ratio and the specific heat ratio.
This is also true for actual spark-ignition internal combustion engines.

A plot of thermal efficiency versus the compression ratio is given in Fig. 9–17 for k 1.4, which is the specific heat ratio value
of air at room temperature.

For a given compression ratio, the thermal efficiency of an actual spark-ignition engine is less than that of an ideal Otto cycle because of the irreversibilities, such as friction, and other factors such as incomplete combustion

We can observe from Fig. 9–17 that the thermal efficiency curve is rather steep at low compression ratios but flattens out starting with a compression ratio value of about 8. Therefore, the increase in thermal efficiency with the compression ratio is not as pronounced at high compression ratios. Also, when high compression ratios are used, the temperature of the air–fuel mixture rises above the autoignition temperature of the fuel (the temperature at which the fuel ignites without the help of a spark) 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. This premature ignition of the fuel, called **autoignition,** produces an audible noise, which is called **engine knock.** Autoignition in spark-ignition engines cannot be tolerated because it hurts performance and can cause engine damage. The requirement that autoignition not be allowed places an upper limit on the compression ratios that can be used in spark-ignition internal combustion engines.

Improvement of the thermal efficiency of gasoline engines by utilizing higher compression ratios (up to about 12) without facing the autoignition problem has been made possible by using gasoline blends that have good antiknock characteristics, such as gasoline mixed with tetraethyl lead. Tetraethyl lead had been added to gasoline since the 1920s because it is an inexpensive method of raising the *octane rating*, which is a measure of the engine knock resistance of a fuel. Leaded gasoline, however, has a very undesirable side effect: it forms compounds during the combustion process that are hazardous to health and pollute the environment. In an effort to combat air pollution, the government adopted a policy in the mid-1970s that resulted in the eventual phase-out of leaded gasoline. Unable to use lead, the refiners developed other techniques to improve the antiknock characteristics of gasoline. Most cars made since 1975 have been designed to use unleaded gasoline, and the compression ratios had to be lowered to avoid engine knock. The ready availability of high octane fuels made it possible to raise the compression ratios again in recent years. Also, owing to the improvements in other areas (reduction in overall automobile weight, improved aerodynamic design, etc.), today's cars have better fuel economy and consequently get more miles per gallon of fuel. This is an example of how engineering decisions involve compromises, and efficiency is only one of the considerations in final design.

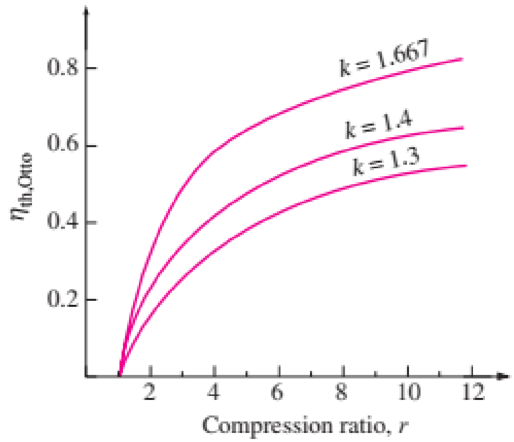


FIGURE 9–18

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

The second parameter affecting the thermal efficiency of an ideal Otto cycle is the specific heat ratio k. For a given compression ratio, an ideal Otto cycle using a monatomic gas (such as argon or helium, $k = 1.667$) as the working fluid will have the highest thermal efficiency. The specific heat ratio k, and thus the thermal efficiency of the ideal Otto cycle, decreases as the molecules of the working fluid get larger (Fig. 9–18). At room temperature it is 1.4 for air, 1.3 for carbon dioxide, and 1.2 for ethane. The working

fluid in actual engines contains larger molecules such as carbon dioxide, and the specific heat ratio decreases with temperature, which is one of the reasons that the actual cycles have lower thermal efficiencies than the ideal Otto cycle. The thermal efficiencies of actual spark-ignition engines range from about 25 to 30 percent.

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

The Diesel cycle is the ideal cycle for CI reciprocating engines. The CI engine, first proposed by Rudolph Diesel in the 1890s, is very similar to the SI engine discussed in the last section, differing mainly in the method of initiating combustion. In spark-ignition engines (also known as *gasoline engines*), the air–fuel mixture is compressed to a temperature that is below the autoignition temperature of the fuel, and the combustion process is initiated by firing a spark plug. In CI engines (also known as *diesel engines*), the air is compressed to a temperature that is above the autoignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines (Fig. 9–20).

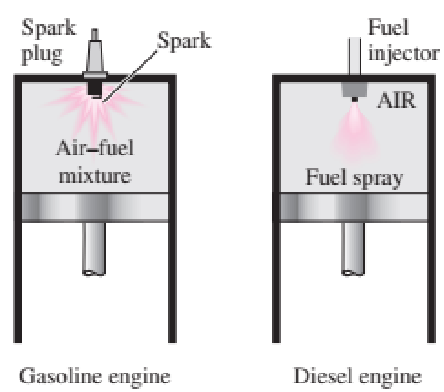
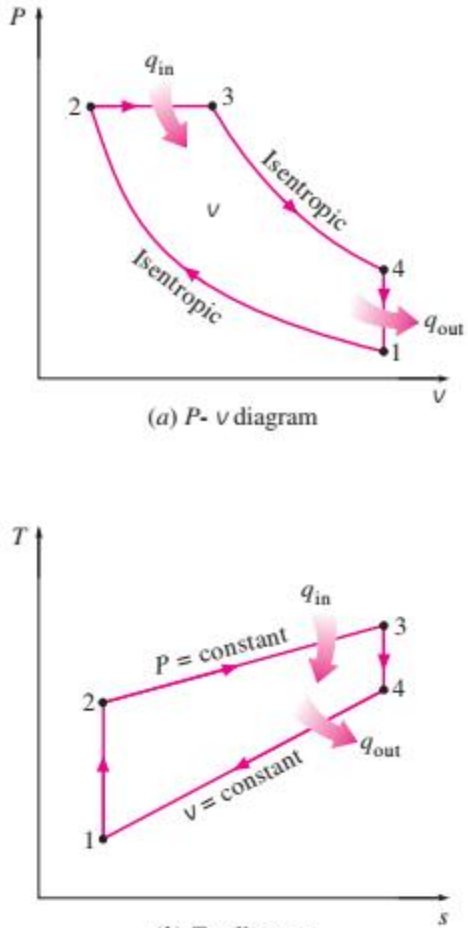


FIGURE 9–20

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

In gasoline engines, a mixture of air and fuel is compressed during the compression stroke, and the compression ratios are limited by the onset of autoignition or engine knock. In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition. Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 and 24. Not having to deal with the problem of autoignition has another benefit: many of the stringent requirements placed on the gasoline can now be removed, and fuels that are less refined (thus less expensive) can be used in diesel engines.



(*a*) *P*-v diagram

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

FIGURE 9–21

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

The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke. Therefore, the combustion process in these engines takes place over a longer interval. Because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat-addition process. In fact, this is the only process where the Otto and the Diesel cycles differ. The remaining three processes are the same for both ideal cycles. That is, process 1-2 is isentropic compression, 3-4 is isentropic expansion, and 4-1 is constant-volume heat rejection. The similarity between the two cycles is also apparent from the *P*-v and *T*-*s* diagrams of the Diesel cycle, shown in Fig. 9–21.

Noting that 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

$$q_{in} - w_{b,out} = u_3 - u_2 \rightarrow q_{in} = P_2(v_3 - v_2) + (u_3 - u_2)$$
$$= h_3 - h_2 = c_p(T_3 - T_2) \qquad \textbf{(9–10}a\textbf{)}$$

and

$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1) \qquad \textbf{(9–10}b\textbf{)}$$

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

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

We now define a new quantity, the **cutoff ratio** r_c, as the ratio of the cylinder volumes after and before the combustion process:

$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2} \tag{9–11}$$

Utilizing this definition and the isentropic ideal-gas relations for processes 1-2 and 3-4, we see that the thermal efficiency relation reduces to

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

where r is the compression ratio defined by Eq. 9–9. Looking at Eq. 9–12 carefully, one would notice that under the cold-air-standard assumptions, the efficiency of a Diesel cycle differs from the efficiency of an Otto cycle by the quantity in the brackets. This quantity is always greater than 1. Therefore,

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

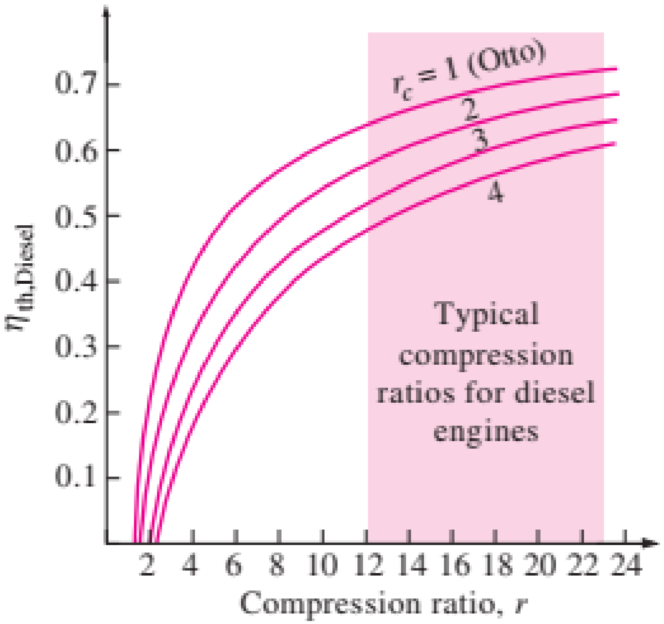


FIGURE 9–22

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

when both cycles operate on the same compression ratio. Also, as the cutoff ratio decreases, the efficiency of the Diesel cycle increases (Fig. 9–22). For the limiting case of $r_c = 1$, the quantity in the brackets becomes unity (can you prove it?), and the efficiencies of the Otto and Diesel cycles become identical. Remember, though, that diesel engines operate at much higher compression ratios and thus are usually more efficient than the spark-ignition (gasoline) engines. The diesel engines also burn the fuel more completely since they usually operate at lower revolutions per minute and the air–fuel mass ratio is much higher than spark-ignition engines. Thermal efficiencies of large diesel engines range from about 35 to 40 percent.

The higher efficiency and lower fuel costs of diesel engines make them attractive in applications requiring relatively large amounts of power, such as in locomotive engines, emergency power generation units, large ships, and heavy trucks. As an example of how large a diesel engine can be, a 12-cylinder diesel engine built in 1964 by the Fiat Corporation of Italy had a normal power output of 25,200 hp (18.8 MW) at 122 rpm, a cylinder bore of 90 cm, and a stroke of 91 cm.

Comparison of Otto and Diesel cycle is given below:

	Otto Cycle	Diesel Cycle
1	Heat is addition in system at constant volume.	Heat is addition in system at constant pressure.
2	Compression ration varies from 6 to 10.	Compression ration varies from 14 to 22.
3	High Efficiency at same compression ratio.	Low efficiency at same compression ratio.
4	Petrol engine work on Otto cycle.	Diesel engine is work on diesel cycle.
5	Spark plug required.	Fuel injector required.
6	Air Fuel mixture inserted from inlet valve.	Only Air is inserted from inlet valve.
7	After generating spark, combustion take place.	Combustion take place due to hot air and fuel receive from fuel injector.
8	In compression stroke, air fuel is compressed.	In compression stroke, only air is compressed.
9	Fuel is inserted during the suction stroke.	Fuel is inserted at the end of compression stroke.
10	No need of high pressure and temperature because spark starts the combustion.	High pressure and temperature is required to start combustion when fuel is injected.
11	Low compression ratio.	High compression ratio.

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

The Brayton cycle was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery. Gas turbines usually operate on an *open cycle,* as shown in Fig. 9–29. 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. 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.

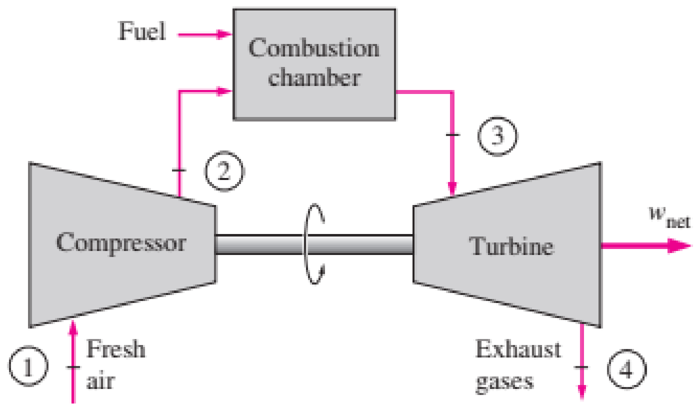


FIGURE 9–29

An open-cycle gas-turbine engine.

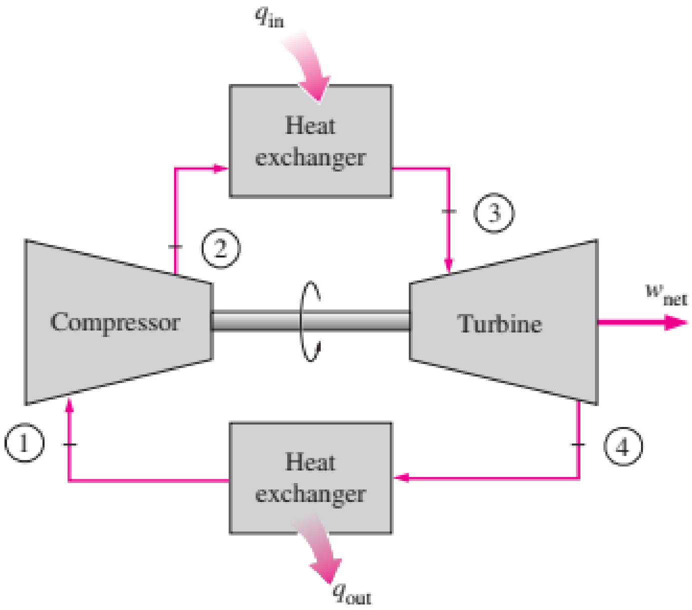
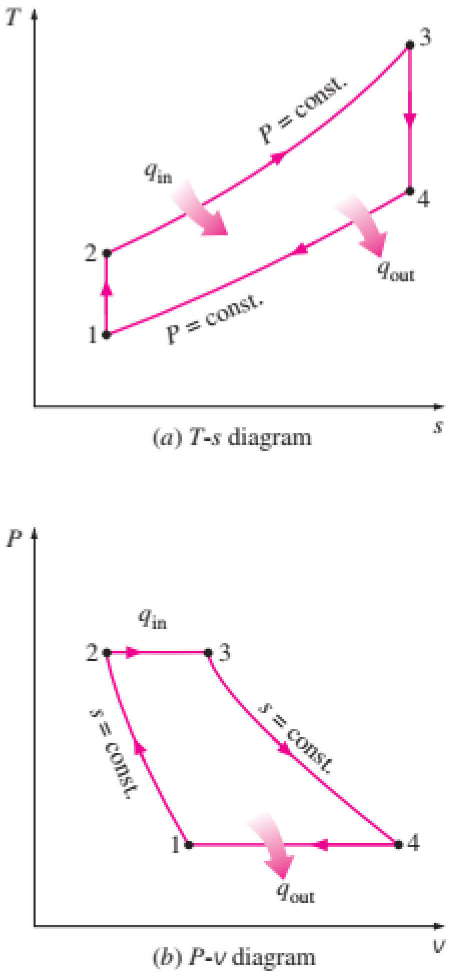


FIGURE 9–30

A closed-cycle gas-turbine engine.

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



The T-s and P-v diagrams of an ideal Brayton cycle are shown in Fig. 9–31. Notice that all four processes of the Brayton cycle are executed in steady-flow devices; thus, they should be analyzed as steady-flow processes. When the changes in kinetic and potential energies are neglected, the energy balance for a steady-flow process can be expressed, on a unit–mass basis, as

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

Therefore, heat transfers to and from the working fluid are

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

and

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

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

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

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

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

Substituting these equations into the thermal efficiency relation and simplifying give

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