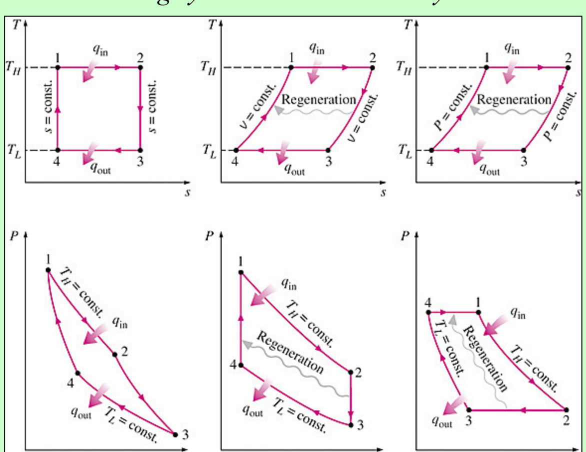
Chapter 8:

Gas Power Cycles

1

STIRLING AND ERICSSON CYCLES

- There are two other cycles that involve an isothermal heat-addition process at T_H and an isothermal heat-rejection process at T_L: the *Stirling cycle* and the *Ericsson cycle.*



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

- They differ from the Carnot cycle in that the two isentropic processes are replaced by:
  - two constant-volume regeneration processes in the Stirling cycle and
  - two constant-pressure regeneration processes in the Ericsson cycle.

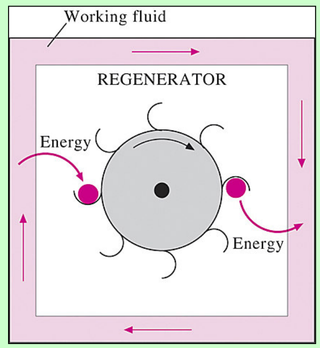
2

STIRLING AND ERICSSON CYCLES

- Both cycles utilize **regeneration,** a process during which heat is transferred to a thermal energy storage device (called a *regenerator*) during one part of the cycle and is transferred back to the working fluid during another part of the cycle.

- The regenerator can be a wire or a ceramic mesh or any kind of porous plug with a high thermal mass (mass times specific heat).

- Stirling and Ericsson cycles are difficult to achieve in practice because they involve heat transfer through a differential temperature difference in all components including the regenerator.
- Neither is practical.



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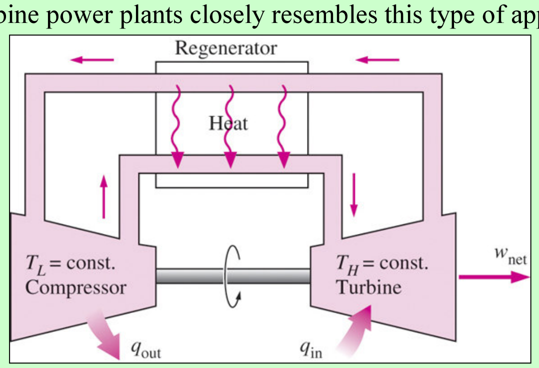
STIRLING AND ERICSSON CYCLES

- Despite the physical limitations and impracticalities associated with them, both the Stirling and Ericsson cycles give a strong message to design engineers: *Regeneration can increase efficiency.*
- It is no coincidence that modern gas-turbine and steam power plants make extensive use of regeneration.
- In fact, the Brayton cycle with regeneration, which is utilized in large gas-turbine power plants closely resembles this type of applications.



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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.

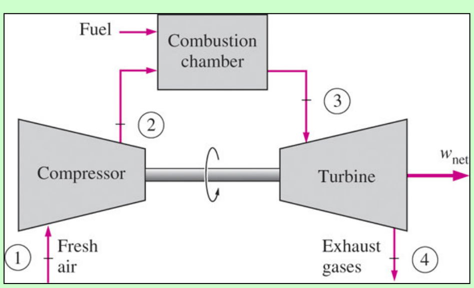
Gas turbines usually operate on an *open cycle*.

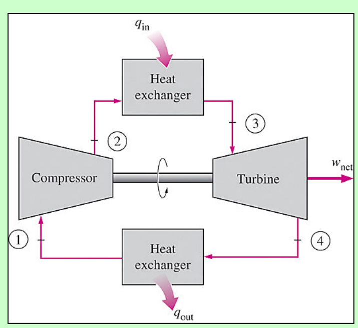
Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised.
The high pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure.
The resulting high-temperature gases then enter the turbine, where they expand to the 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.

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

The open gas-turbine cycle described above can be modeled as a *closed cycle*, by utilizing the air-standard assumptions.

Here the compression and expansion processes remain the same, but the combustion process is replaced by a constant-pressure heat-addition process from an external source, and the exhaust process is replaced by a constant pressure heat-rejection process to the ambient air.





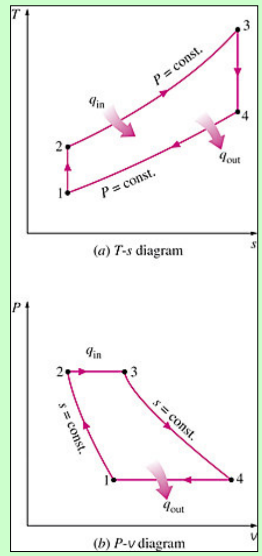
BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

The ideal cycle that the working fluid undergoes in this closed loop is the **Brayton cycle,** which is made up of four internally reversible processes:

1-2 Isentropic compression (in a compressor)
2-3 Constant-pressure heat addition
3-4 Isentropic expansion (in a turbine)
4-1 Constant-pressure heat rejection

The *T-s* and *P-v* diagrams of an ideal Brayton cycle are shown.

Notice that all four processes of the Brayton cycle are executed in steady flow devices; thus, they should be analyzed as steady-flow processes.



BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

Thermal efficiency of the Brayton cycle

$$\eta_{th,\, Brayton} = \frac{W_{net}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

Now to find Q_{in} and Q_{out}

Apply the conservation of energy to process 2-3 for P = constant (no work), steady-flow, and neglect changes in kinetic and potential energies. For constant specific heats

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2)$$
$$\dot{Q}_{in} = \dot{m}C_p(T_3 - T_2)$$

The conservation of energy for process 4-1 yields for constant specific heats

$$\dot{Q}_{out} = \dot{m}(h_4 - h_1)$$
$$\dot{Q}_{out} = \dot{m}C_p(T_4 - T_1)$$

The thermal efficiency becomes

$$\eta_{th,\, Brayton} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}} = 1 - \frac{C_p(T_4 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

Recall processes 1-2 and 3-4 are isentropic, so

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

Since $P_3 = P_2$ and $P_4 = P_1$, we see that

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \text{or} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

The Brayton cycle efficiency becomes

$$\eta_{th,\,Brayton} = 1 - \frac{T_1}{T_2}$$

Since process 1-2 is isentropic,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = r_p^{(k-1)/k}$$

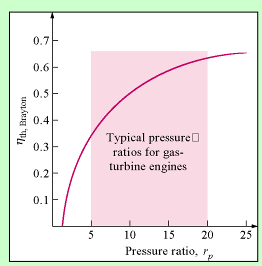
$$\frac{T_1}{T_2} = \frac{1}{r_p^{(k-1)/k}}$$

where the pressure ratio is $r_p = P_2/P_1$ and

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

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

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

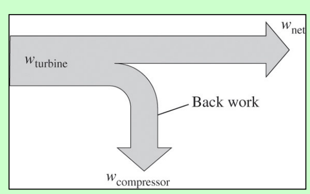


The two major application areas of gas-turbine engines are:

aircraft propulsion and

electric power generation.

In gas-turbine power plants, the ratio of the compressor work to the turbine work, called the **back work ratio** r_{bw}, is very high.



$$r_{bw} = w_{comp} / w_{turb}$$

Usually more than one-half of the gas turbine work output is used to drive the compressor.

This is quite in contrast to steam power plants, where the back work ratio is only a few percent.

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

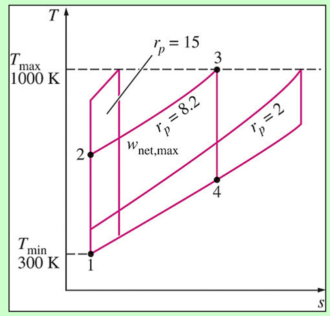
What happens to η_{th}, w_{in} /w_{out}, and w_{net} as the pressure ratio r_p is increased?

Consider the T-s diagram for the cycle and note that the area enclosed by the cycle is the net heat added to the cycle.

By the first law applied to the cycle, the net heat added to the cycle is equal to the net work done by the cycle.

Thus, the area enclosed by the cycle on the T-s diagram also represents the net work done by the cycle.

$$r_{p,\,\text{max work}} = \left(\frac{T_3}{T_1}\right)^{k/[2(k-1)]}$$

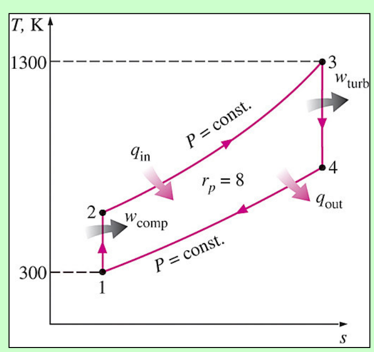


For fixed values of T_{min} and T_{max}, the net work of the Brayton cycle first increases with the pressure ratio, then reaches a maximum at $W_{net,max}$ and finally decreases.

EXAMPLE 8–5

The Simple Ideal Brayton Cycle

- A gas-turbine power plant operating on an ideal Brayton cycle has a pressure ratio of 8. The gas temperature is 300 K at the compressor inlet and 1300 K at the turbine inlet. 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.



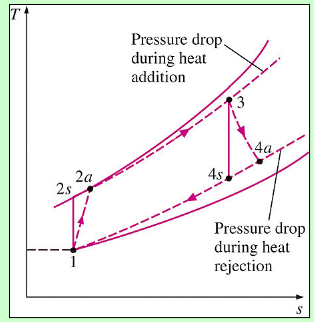
Deviation of Actual Gas-Turbine Cycles from Idealized Ones

The actual gas-turbine cycle differs from the ideal Brayton cycle on several accounts.

For one thing, some pressure drop during the heat-addition and heat rejection processes is inevitable (predictable).

More importantly, the actual work input to the compressor is more, and the actual work output from the turbine is less because of irreversibilities.

The deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accurately accounted for by utilizing the isentropic efficiencies of the turbine and compressor as:



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

$$\eta_T = \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$

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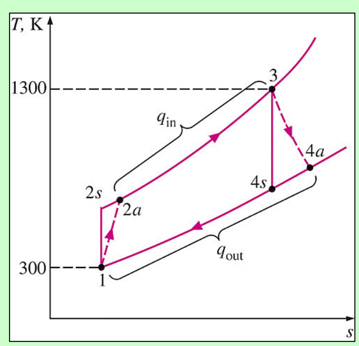
EXAMPLE 8–6

An Actual Gas-Turbine Cycle

- Assuming a compressor efficiency of 80 percent and a turbine efficiency of 85 percent, determine
- (*a*) the back work ratio,
- (*b*) the thermal efficiency, and
- (*c*) the turbine exit temperature of the gas-turbine cycle discussed in Example 8–5.



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Development of Gas Turbines

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

1. Increasing the turbine inlet (or firing) temperatures.

The turbine inlet temperatures have increased steadily from about 540°C (1000°F) in the 1940s to 1425°C (2600°F) and even higher today.

2. Increasing the efficiencies of turbo-machinery components.

the advent (arrival) of computers and advanced techniques for computer-aided design made it possible to design these components aerodynamically with minimal losses.

3. Adding modifications to the basic cycle.

The simple-cycle efficiencies of early gas turbines were practically doubled by incorporating inter-cooling, regeneration (or recuperation), and reheating, discussed in the next two sections.

These improvements, of course, come at the expense of increased initial and operation costs, and they can be justified by the decrease in fuel costs.

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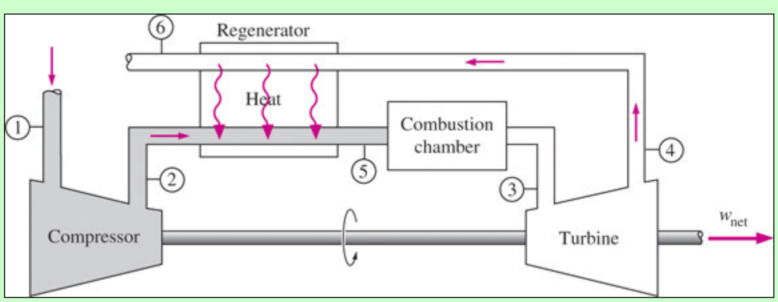
THE BRAYTON CYCLE WITH REGENERATION

In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor.

Therefore, the high-pressure air leaving the compressor can be heated by transferring heat to it from the hot exhaust gases in a counter-flow heat exchanger, which is also known as a *regenerator* or a *recuperator.*



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THE BRAYTON CYCLE WITH REGENERATION

The thermal efficiency of the Brayton cycle increases as a result of regeneration since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber.

This, in turn, decreases the heat input (thus fuel) requirements for the same net work output.

Note, however, that the use of a regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature.

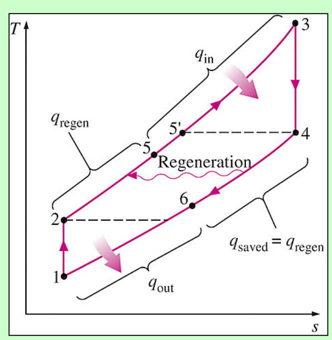
Otherwise, heat will flow in the reverse direction (*to* the exhaust gases), decreasing the efficiency. This situation is encountered in gas-turbine engines operating at very high pressure ratios.

THE BRAYTON CYCLE WITH REGENERATION

The highest temperature occurring within the regenerator is T_4.
Air normally leaves the regenerator at a lower temperature, T_5.
In the limiting (ideal) case, the air exits the regenerator at the inlet temperature of the exhaust gases $T_{5'}=T_4$.
The actual and maximum heat transfers from the exhaust gases to the air can be expressed as:

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

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



We define the regenerator effectiveness ε_{regen} as the ratio of the heat transferred to the compressor gases in the regenerator to the maximum possible heat transfer to the compressor gases.

$$\varepsilon_{regen} = \frac{q_{regen,act}}{q_{regen,\max}} = \frac{h_5 - h_2}{h_4 - h_2} \cong \frac{T_5 - T_2}{T_4 - T_2}$$

THE BRAYTON CYCLE WITH REGENERATION

Using the closed cycle analysis and treating the heat addition and heat rejection as steady-flow processes, the regenerative cycle thermal efficiency is

$$\eta_{th,\,Brayton\ with\ regen} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{h_6 - h_1}{h_3 - h_5}$$

Notice that the heat transfer occurring within the regenerator is **not included** in the efficiency calculation because this energy is not heat transferred across the cycle boundary.

Assuming an ideal regenerator $\varepsilon_{regen} = 1$ and constant specific heats, the thermal efficiency becomes:

$$\eta_{th,\,regen} = 1 - \frac{T_1}{T_3}\left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$= 1 - \frac{T_1}{T_3}(r_p)^{(k-1)/k}$$

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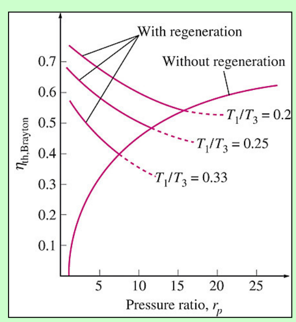
THE BRAYTON CYCLE WITH REGENERATION

Therefore, the thermal efficiency of an ideal Brayton cycle with regeneration depends on the ratio of the minimum to maximum temperatures as well as the pressure ratio.

The thermal efficiency is plotted in Fig. 9–40 for various pressure ratios and minimum-to-maximum temperature ratios.

This figure shows that regeneration is most effective at lower pressure ratios and low minimum-to-maximum temperature ratios.



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THE BRAYTON CYCLE WITH REGENERATION

When does the efficiency of the air-standard Brayton cycle equal the efficiency of the air-standard regenerative Brayton cycle? If we set $\eta_{th,Brayton} = \eta_{th,regen}$ then

$$\eta_{th,\,Brayton} = \eta_{th,\,regen}$$

$$1 - \frac{1}{(r_p)^{(k-1)/k}} = 1 - \frac{T_1}{T_3}(r_p)^{(k-1)/k}$$

$$r_p = \left(\frac{T_3}{T_1}\right)^{k/[2(k-1)]}$$

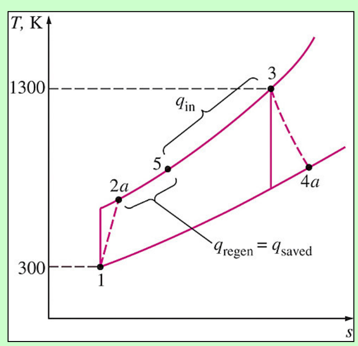
Recall that this is the pressure ratio that maximizes the net work for the simple Brayton cycle and makes $T_4 = T_2$.

What happens if the regenerative Brayton cycle operates at a pressure ratio larger than this value?

EXAMPLE 8–7

Actual Gas-Turbine Cycle With Regeneration

- Determine the thermal efficiency of the gas-turbine described in Example 8–6 if a regenerator having an effectiveness of 80 percent is installed.

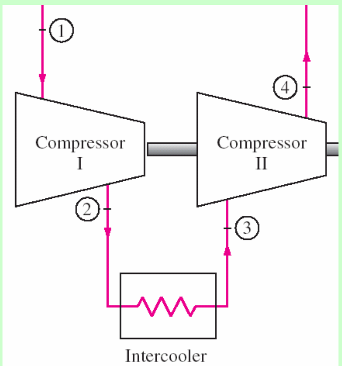


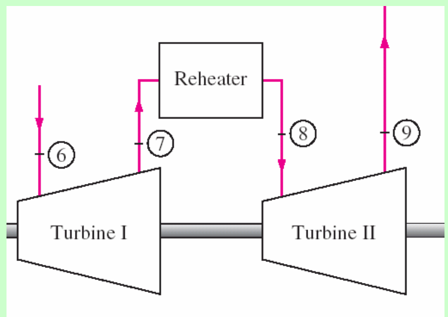
THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

$$w_{net} = w_{turb} - w_{comp}$$

Other Ways to Improve Brayton Cycle Performance

Inter-cooling and reheating are two important ways to improve the performance of the Brayton cycle with regeneration.





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THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

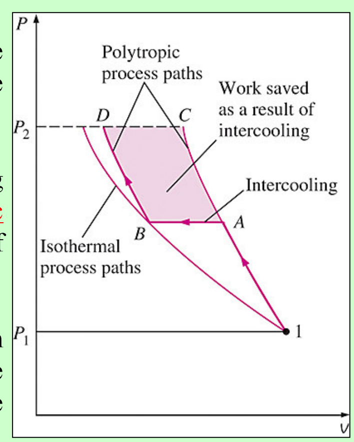
Intercooling

When using multistage compression, cooling the working fluid between the stages will reduce the amount of compressor work required.

The compressor work is reduced because cooling the working fluid reduces the average specific volume of the fluid and thus reduces the amount of work on the fluid to achieve the given pressure rise.

Recall that the intermediate pressure at which intercooling should take place to minimize the compressor work, can be found by following the approach shown in Chapter 6.

$$\frac{P_2}{P_B} = \frac{P_A}{P_1}$$



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THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

Reheating

Likewise, the work output of a turbine operating between two pressure levels can be increased by expanding the gas in stages and reheating it in between. That is, utilizing *multistage expansion with reheating.*

This is accomplished without raising the maximum temperature in the cycle.

The foregoing argument is based on a simple principle:
The steady-flow compression or expansion work is proportional to the specific volume of the fluid. Therefore, the specific volume of the working fluid should be as low as possible during a compression process and as high as possible during an expansion process.

This is precisely what intercooling and reheating accomplish.

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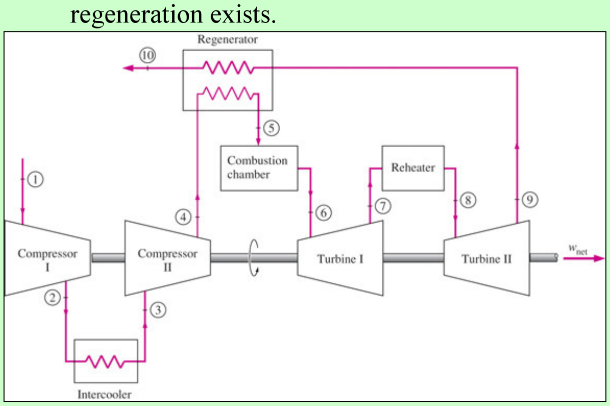
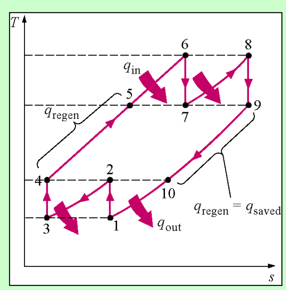
THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

The working fluid leaves the compressor at a lower temperature, and the turbine at a higher temperature, when intercooling and reheating are utilized.
This makes regeneration more attractive since a greater potential for regeneration exists.





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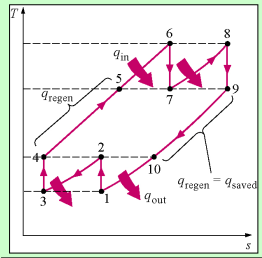
THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

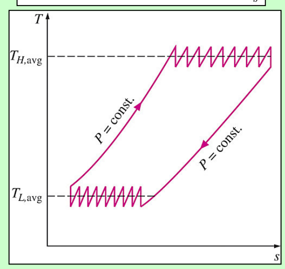
It was shown previously, that the work input to a two-stage compressor is minimized when equal pressure ratios are maintained across each stage.
It can be shown that this procedure also maximizes the turbine work output. Thus, for best performance we have

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



As the number of compression stages is increased, the compression process becomes nearly isothermal at the compressor inlet temperature, and the compression work decreases.

Similarly, as the number of expansion stages is increased, the expansion process becomes nearly isothermal.



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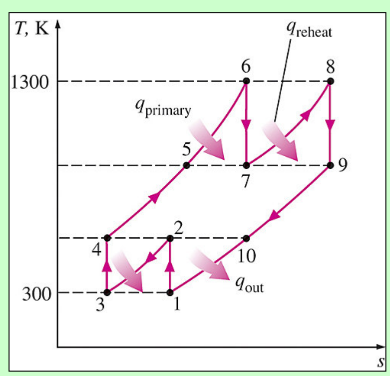
EXAMPLE 8–8

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.
- Compare the results with those obtained in Example 8–5.



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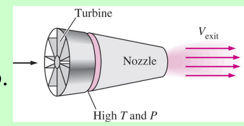
IDEAL JET-PROPULSION CYCLES

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 totally expanded to the ambient pressure in the turbine.

Instead, they are expanded to a pressure such that the power produced by the turbine is just sufficient to drive the compressor and the auxiliary equipment, such as a small generator and hydraulic pumps.



That is, the net work output of a jet propulsion cycle is zero.

The gases that exit the turbine at a relatively high pressure are subsequently accelerated in a nozzle to provide the thrust to propel the aircraft.

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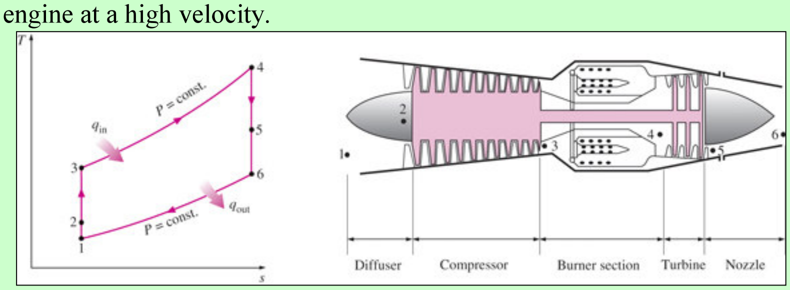
IDEAL JET-PROPULSION CYCLES

A schematic of a turbojet engine and the *T-s* diagram of the ideal turbojet cycle are shown.
The pressure of air rises slightly as it is decelerated in the diffuser.
Air is compressed by the compressor.
It is mixed with fuel in the combustion chamber, where the mixture is burned at constant pressure.
The high-pressure and high-temperature combustion gases partially expand in the turbine, producing enough power to drive the compressor and other equipment.
Finally, the gases expand in a nozzle to the ambient pressure and leave the engine at a high velocity.



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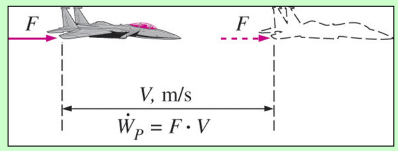
IDEAL JET-PROPULSION CYCLES

The **thrust** developed in a turbojet engine is the unbalanced force that is caused by the difference in the momentum of the low-velocity air entering the engine and the high-velocity exhaust gases leaving the engine, and it is determined from Newton's second law.

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

The power developed from the thrust of the engine is called the **propulsive power** W_P, which is the *propulsive force* (*thrust*) times the *distance* this force acts on the aircraft per unit time, that is, the thrust times the aircraft velocity



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

The ratio of the *power produced* to propel the aircraft W_P and the required input which is the *heating value of the fuel* Q_{in} is called the **propulsive efficiency** and is given by:

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

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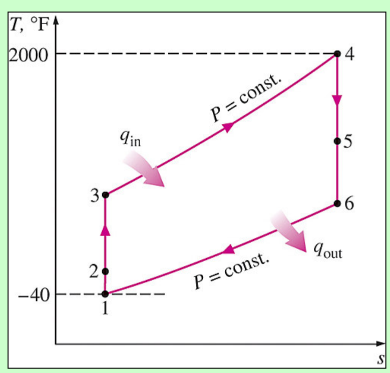
EXAMPLE 8–9
The Ideal Jet-Propulsion Cycle

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



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