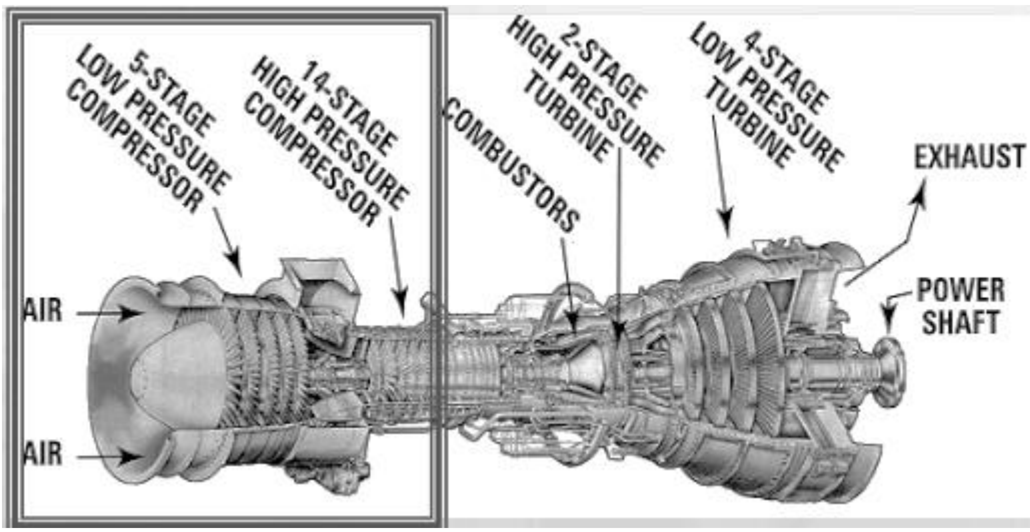
Gas turbine power plant:

Gas turbine engines derive their power from burning fuel in a combustion chamber and using the fast flowing combustion gases to drive a turbine in much the same way as the high pressure steam drives the steam turbine. A simple gas turbine is comprised of three main sections a compressor, a combustor and a power turbine. The gas turbine operates on the principle of the Brayton cycle, where compressed air is mixed with fuel, and burned under constant pressure conditions. The resulting gas is allowed to expand through a turbine to perform work. Gas turbines are widely used in Aircraft propulsion system, Electric power generation, Marine vehicle propulsion and Combined-cycle power plant (with steam power plant).

Main Components of Gas Turbine Power Plant



1. Compressor

The compressor sucks in air form the atmosphere and compresses it to pressures in the range of 15 to 20 bars. The compressor consists of a number of rows of blades mounted on a shaft. The shaft is connected and rotates along with the main gas turbine.

2. Combustor

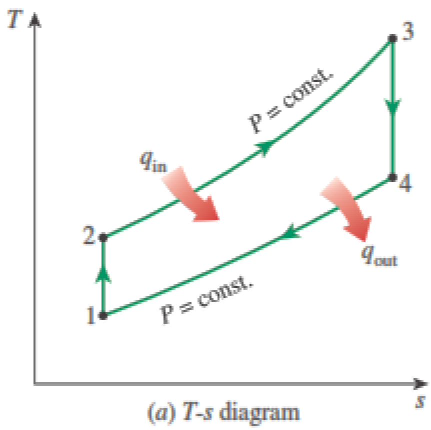
This is an annular chamber where the fuel burns and is similar to the furnace in a boiler. The hot gases in the range of 1400 to 1500 ^{O}C leave the chamber with high energy levels. The chamber and the subsequent sections are made of special alloys and designs that can withstand this high temperature

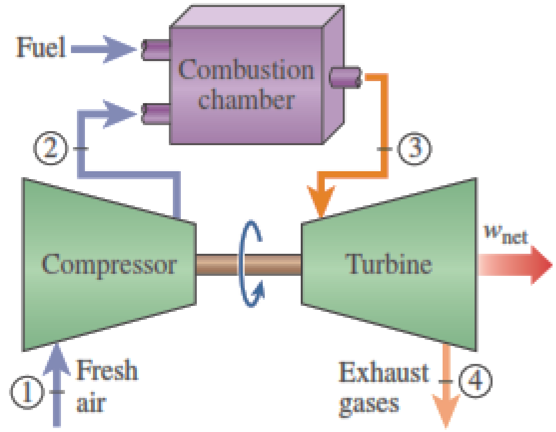
3. Turbine

The turbine does the main work of energy conversion. The turbine portion also consists of rows of blades fixed to the shaft. The kinetic energy of the hot gases impacting on the blades rotates the blades and the shaft. The gas temperature leaving the Turbine is in the range of 500 to 550 ^{O}C. The gas turbine shaft connects to the generator to produce electric power.

Gas Turbine Cycle (Brayton cycle):

The gas turbine operates on the principle of the Brayton cycle which is as follow:





- 1-2: Isentropic compression, fresh air at ambient conditions is drawn into the compressor; its temperature and pressure are raised.
- 2-3: Constant-pressure heat addition, the high-pressure air enters the combustion chamber, the fuel is burned at constant pressure (Heat is supplied)
- 3-4: Isentropic expansion, the high temperature and pressure gases enter the turbine and expand to the atmospheric pressure while producing power.
- 4-1: Constant-pressure heat rejection, the exhaust gases leaving the turbine are thrown out (not re-circulated), causing the cycle to be classified as an open cycle

Thermal Efficiency

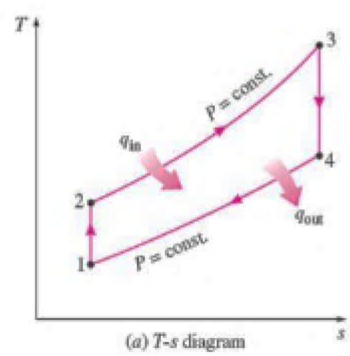
The **energy balance** for a steady-flow process can be expressed, on a unit–mass basis, as

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

The **heat transfers** to and from the working fluid are:

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

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



The **thermal efficiency** of the ideal Brayton cycle,

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

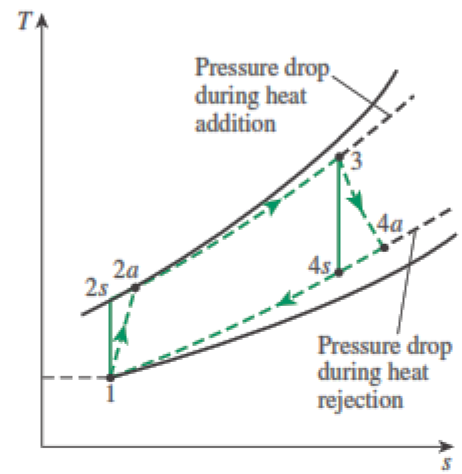
Practical Gas Turbine Cycle and Isentropic Efficiencies:

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. 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

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

and

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



Where states 2a and 4a are the actual exit states of the compressor and the turbine, respectively, and 2s and 4s are the corresponding states for the isentropic case, as illustrated in Figure above.

The Brayton Cycle with Intercooling, Reheating, and Regeneration

The net work of a gas-turbine cycle is the difference between the turbine work output and the compressor work input, and it can be increased by either decreasing the compressor work or increasing the turbine work, or both.

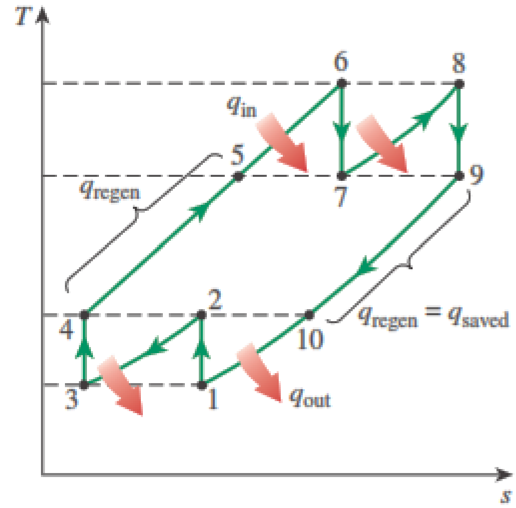
One way to reduce compressor work is to reduce the irreversibilities such as friction and turbulence. Other way of reducing the compressor work is to keep the specific volume of the gas as small as possible during the compression process. This is done by maintaining the temperature of the gas as low as possible during compression since the specific volume of a gas is proportional to temperature. Therefore, reducing the work input to a compressor requires that the gas be cooled as it is compressed. This is done by using multistage compression with intercooling. As the number of stages is increased, the compression process becomes nearly isothermal at the compressor inlet temperature, and the compression work decreases.

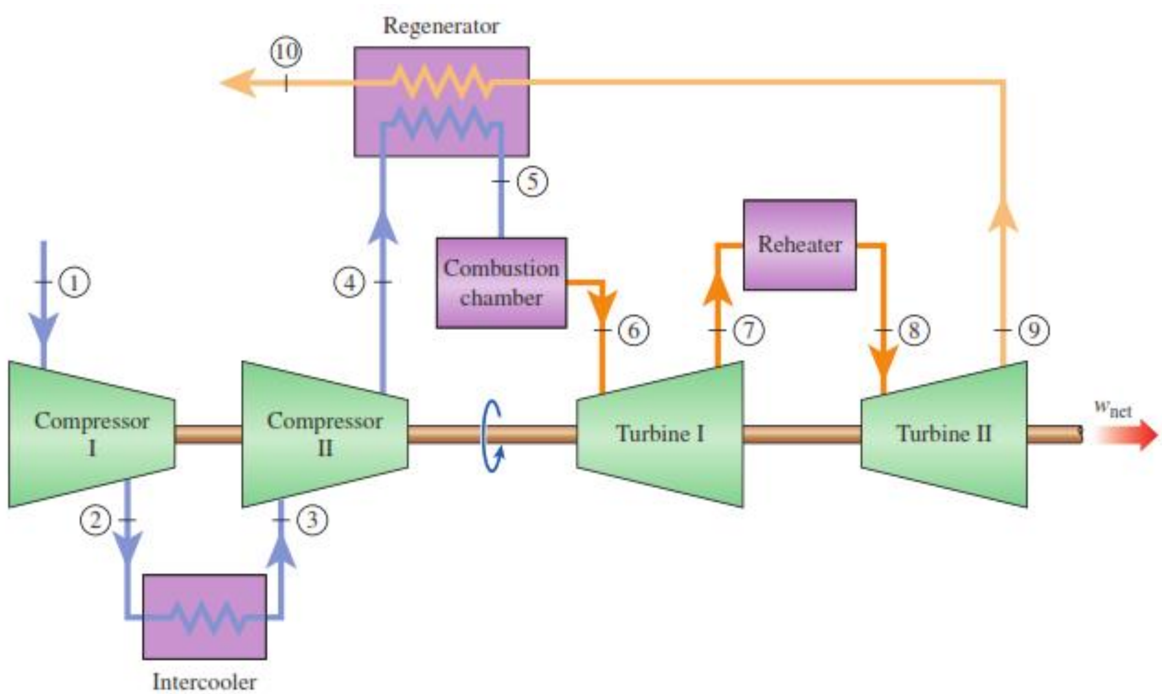
Likewise, the work output of a turbine operating between two pressure levels can be increased by expanding the gas in stages and reheating it in between—that is, utilizing multistage expansion with reheating. This is accomplished without raising the maximum temperature in the cycle. As the number of stages is increased, the expansion process becomes nearly isothermal. 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.

Combustion in gas turbines typically occurs at four times the amount of air needed for complete combustion to avoid excessive temperatures. Therefore, the exhaust gases are rich in oxygen, and reheating can be accomplished by simply spraying additional fuel into the exhaust gases between two expansion states.

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. Also, the gases leaving the compressor can be heated to a higher temperature before they enter the combustion chamber because of the higher temperature of the turbine exhaust.

A schematic of the physical arrangement and the T-s diagram of an ideal two-stage gas-turbine cycle with intercooling, reheating, and regeneration are shown in Figure. The gas enters the first stage of the compressor at state 1, is compressed isentropically to an intermediate pressure P_2 , is cooled at constant pressure to state 3 ($T_3 = T_1$), and is compressed in the second stage isentropically to the final pressure P_4 . At state 4 the gas enters the regenerator, where it is heated to T_5 at constant pressure. In an ideal regenerator, the gas leaves the regenerator at the temperature of the turbine exhaust, that is, $T_5 = T_9$. The primary heat addition (or combustion) process takes place between states 5 and 6. The gas enters the first stage of the turbine at state 6 and expands isentropically to state 7, where it enters the reheater. It is reheated at constant pressure to state 8 ($T_8 = T_6$), where it enters the second stage of the turbine. The gas exits the turbine at state 9 and enters the regenerator, where it is cooled to state 10 at constant pressure. The cycle is completed by cooling the gas to the initial state (or purging the exhaust gases).





In the analysis of the actual gas-turbine cycles, the irreversibilities that are present within the compressor, the turbine, and the regenerator as well as the pressure drops in the heat exchangers should be taken into consideration.

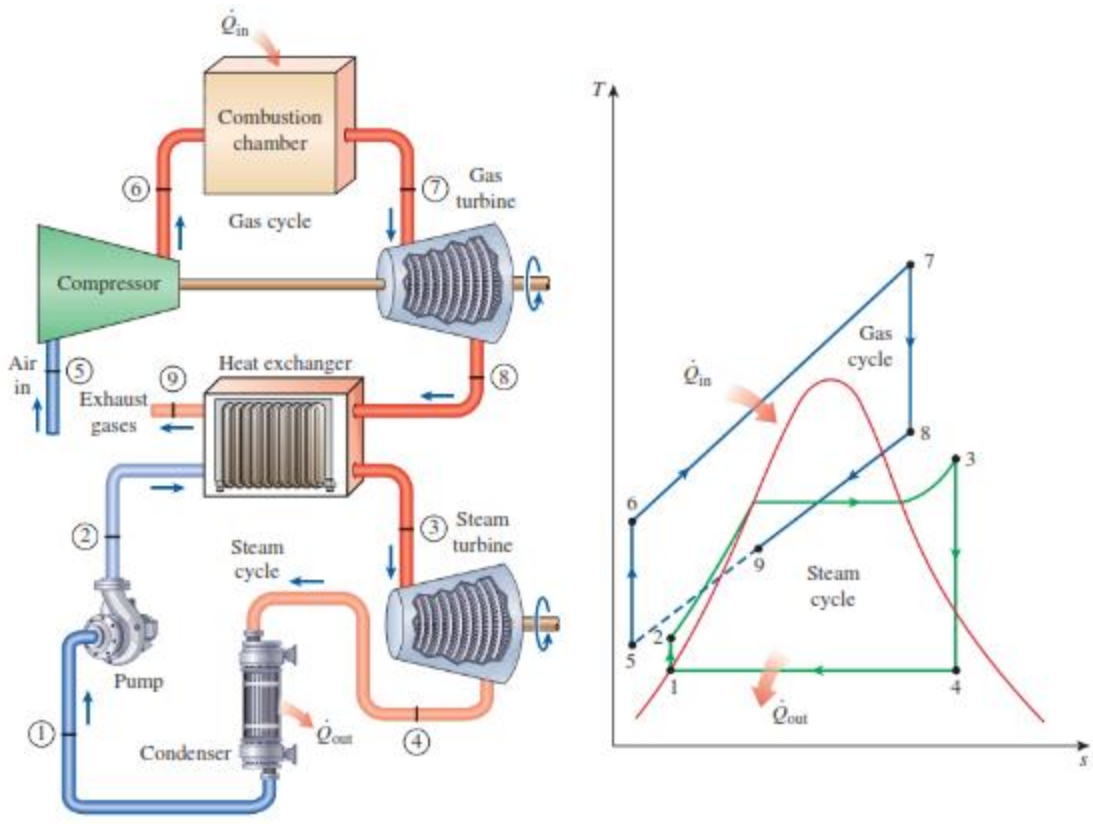
Combine Cycle Power Plants:

The continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants. A more popular modification involves a gas power cycle topping a vapor power cycle, which is called the combined gas–vapor cycle, or just the combined cycle. The combined cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam turbine (Rankine) cycle, which has a higher thermal efficiency than either of the cycles executed individually.

Gas-turbine cycles typically operate at considerably higher temperatures than steam cycles. The maximum fluid temperature at the turbine inlet is about 620 ^{O}C for modern steam power plants, but over 1425 ^{O}C for gas-turbine power plants. It is over 1500 ^{O}C at the burner exit of turbojet engines. The use of higher temperatures in gas turbines is made possible by developments in cooling the turbine blades and coating the blades with high-temperature-resistant materials such as ceramics.

Because of the higher average temperature at which heat is supplied, gas turbine cycles have a greater potential for higher thermal efficiencies. However, the gas-turbine cycles have one inherent disadvantage: The gas leaves the gas turbine at very high temperatures (usually above 500 ^{O}C), which erases any potential gains in the thermal efficiency. The situation can be improved somewhat by using regeneration, but the improvement is limited

It makes engineering sense to take advantage of the very desirable characteristics of the gas-turbine cycle at high temperatures and to use the high temperature exhaust gases as the energy source for the bottoming cycle such as a steam power cycle. The result is a combined gas–steam cycle, as shown in Figure. In this cycle, energy is recovered from the exhaust gases by transferring it to the steam in a heat exchanger that serves as the boiler. In general, more than one gas turbine is needed to supply sufficient heat to the steam. Also, the steam cycle may involve regeneration as well as reheating. Energy for the reheating process can be supplied by burning some additional fuel in the oxygen-rich exhaust gases.



Developments in gas-turbine technology have made the combined gas– steam cycle economically very attractive. The combined cycle increases the efficiency without increasing the initial cost greatly. Consequently, many new power plants operate on combined cycles, and many more existing steam- or gas-turbine plants are being converted to combined-cycle power plants. Thermal efficiencies well over 40 percent are reported as a result of conversion.

A 1090-MW Tohoku combined plant that was put in commercial operation in 1985 in Niigata, Japan, is reported to operate at a thermal efficiency of 44 percent. This plant has two 191-MW steam turbines and six 118-MW gas turbines. Hot combustion gases enter the gas turbines at 1154 ^{O}C, and steam enters the steam turbines at 500 ^{O}C. Steam is cooled in the condenser by cooling water at an average temperature of 15 ^{O}C. The compressors have a pressure ratio of 14, and the mass flow rate of air through the compressors is 443 kg/s.

A 1350-MW combined-cycle power plant built in Ambarli, Turkey, in 1988 by Siemens of Germany is the first commercially operating thermal plant in the world to attain an efficiency level as high as 52.5 percent at design operating conditions. This plant has six 150-MW gas turbines and three 173-MW steam turbines. Some recent combined-cycle power plants have achieved efficiencies above 60 percent.