

Open Gas-Turbine Cycle

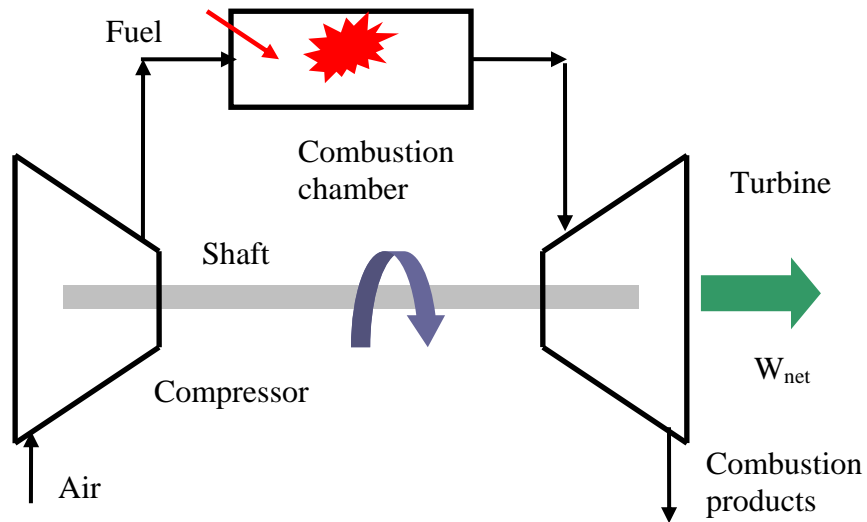


Fig.1: Schematic for an open gas-turbine cycle.

Working Principal

Fresh air enters the compressor at ambient temperature where its pressure and temperature are increased.

The high pressure air enters the combustion chamber where the fuel is burned at constant pressure.

The high temperature (and pressure) gas enters the turbine where it expands to ambient pressure and produces work.

Features:

- Gas-turbine is used in aircraft propulsion and electric power generation.
- High thermal efficiencies up to 44%.
- Suitable for combined cycles (with steam power plant)
- High power to weight ratio, high reliability, long life
- Fast start up time, about 2 min, compared to 4 hr for steam-propulsion systems
- High back work ratio (ratio of compressor work to the turbine work), up to 50%, compared to few percent in steam power plants.

Brayton Cycle

Brayton cycle is the ideal cycle for gas-turbine engines in which the working fluid undergoes a closed loop. That is the combustion and exhaust processes are modeled by constant-pressure heat addition and rejection, respectively.

The Brayton ideal cycle is made up of four internally reversible processes:

| | |
|-----|---|
| 1-2 | isentropic compression (in compressor) |
| 2-3 | const. pressure heat-addition (in combustion chamber) |
| 3-4 | isentropic expansion (in turbine) |
| 4-1 | const. pressure heat rejection (exhaust) |

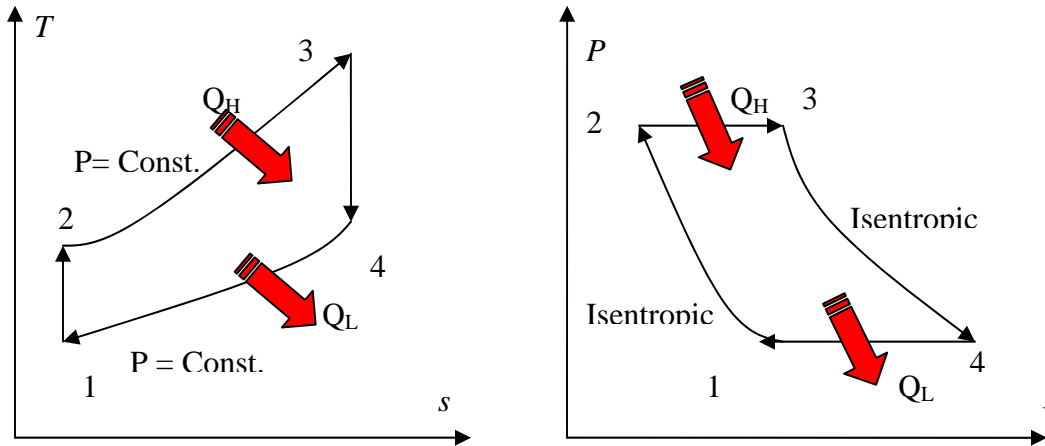


Fig. 2: T-s and P-v diagrams for ideal Brayton cycle.

Thermal efficiency for the Brayton cycle is:

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

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

thus

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

$$r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4}$$

where r_p is called the pressure ratio and $k = c_p/c_v$ is the specific heat ratio.

Maximum Pressure Ratio

Given that the maximum and minimum temperature can be prescribed for the Brayton cycle, a change in the pressure ratio can result in a change in the work output from the cycle.

The maximum temperature in the cycle T_3 is limited by metallurgical conditions because the turbine blades cannot sustain temperatures above 1300 K. Higher temperatures (up to 1600 K can be obtained with ceramic turbine blades). The minimum temperature is set by the air temperature at the inlet to the engine.

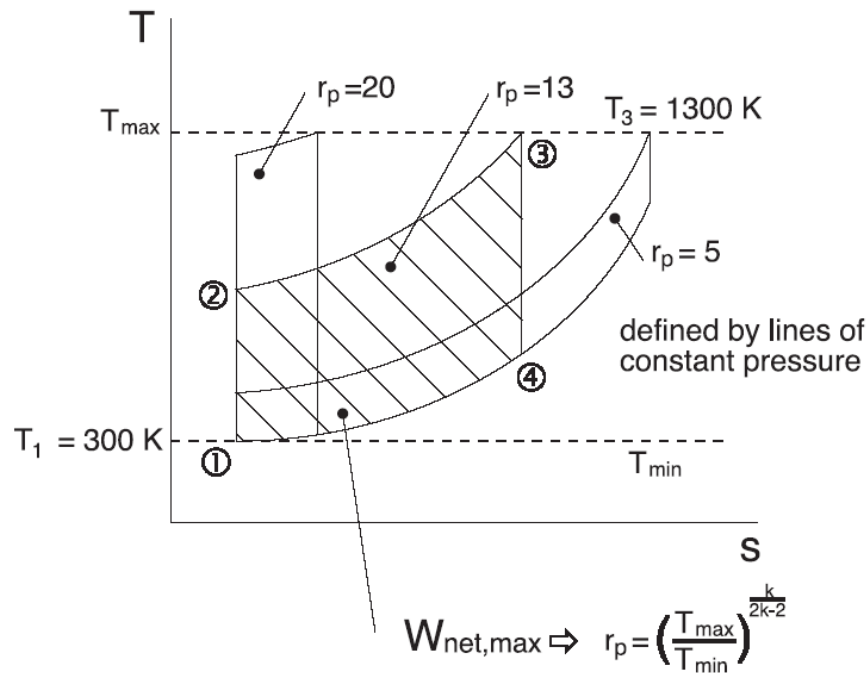


Fig. 3: An optimum pressure ratio exists for a given max., min. temperature ratio that maximized the net work output.

Actual Brayton Cycle

Irreversibilities exist in actual cycle. Most important differences are deviations of actual compressor and turbine from idealized isentropic compression/expansion, and pressure drop in combustion chamber.

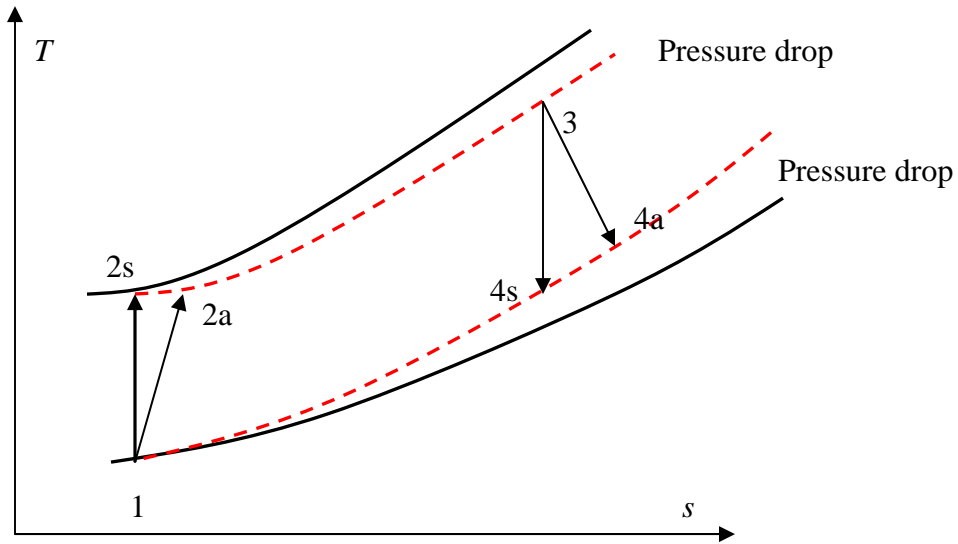


Fig. 4: Actual Brayton cycle.

$$\eta_c = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1} \quad \eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

The Brayton Cycle with Regeneration

The high pressure air leaving the compressor can be heated by transferring heat from exhaust gases in a counter-flow heat exchanger which is called a regenerator.

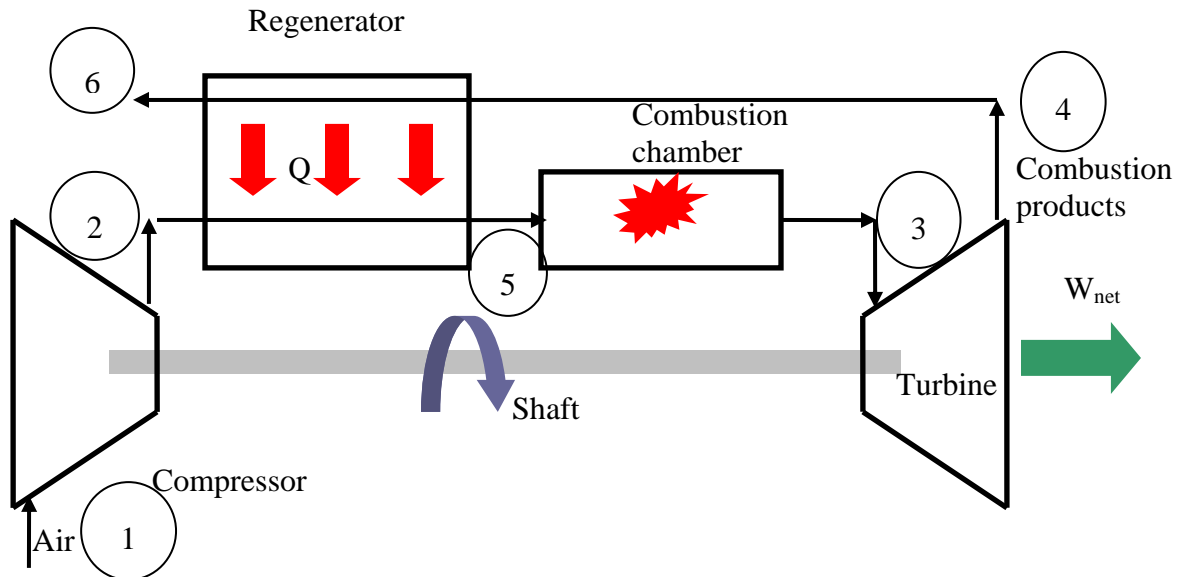


Fig. 5: Schematic for a Brayton cycle with regenerator.

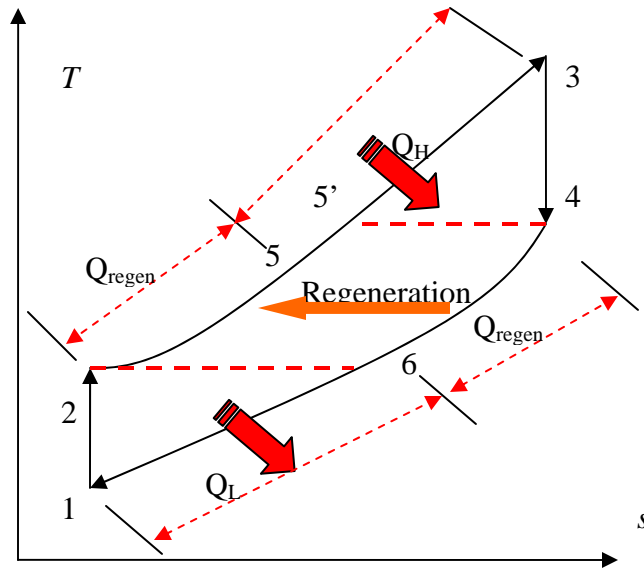


Fig. 6: T-s diagram for a Brayton cycle with regeneration.

In the ideal case, the air exits the regenerator at the inlet temperature of the exhaust gases, T_4 . One can write:

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

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

The effectiveness of regenerator is defined as:

$$\varepsilon = \frac{q_{regen, act}}{q_{regen, max}} = \frac{h_5 - h_2}{h_4 - h_2} = \frac{T_5 - T_2}{T_4 - T_2} \text{ (assuming cold - air - standard conditions)}$$

Thermal efficiency of an ideal Brayton cycle with regenerator can be found from:

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

The Brayton Cycle with Intercooling, Reheating, and Regeneration

The net work output of the cycle can be increased by reducing the work input to the compressor and/or by increasing the work output from turbine (or both).

Using multi-stage compression with intercooling reduces the work input the compressor. As the number of stages is increased, the compression process becomes nearly isothermal at the compressor inlet temperature, and the compression work decreases.

Likewise utilizing multistage expansion with reheat (in a multi-turbine arrangement) will increase the work produced by turbines.

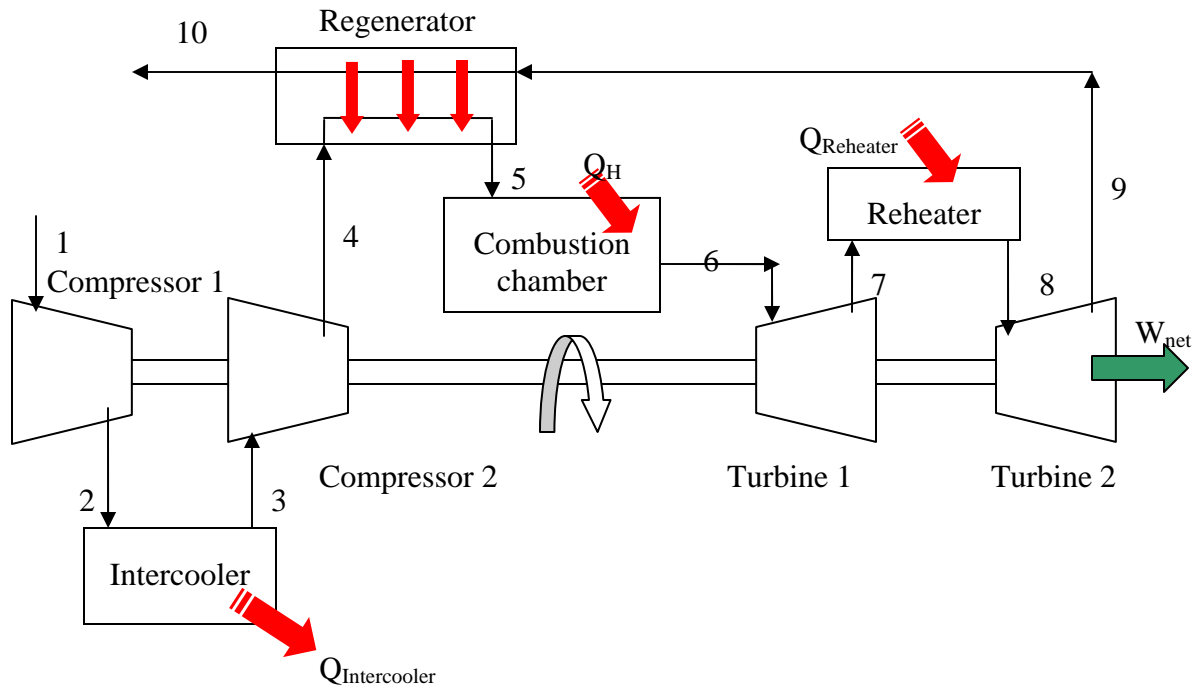


Fig. 7: A gas-turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration.

When intercooling and reheating are used, regeneration becomes more attractive since a greater potential for regeneration exists.

The back work ratio of a gas-turbine improves as a result of intercooling and reheating. However; intercooling and reheating *decreases thermal efficiency* unless they are accompanied with regeneration.

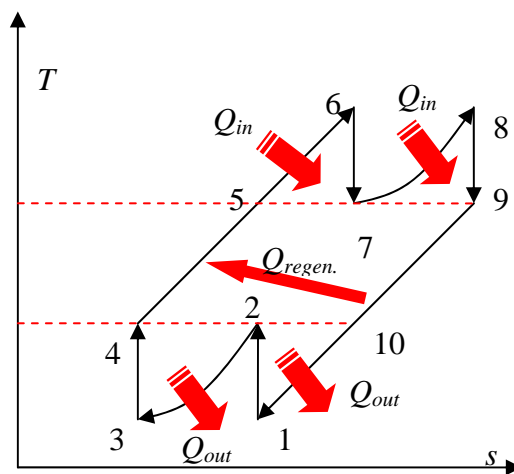


Fig. 8: T-s diagram for an ideal gas-turbine cycle with intercooling, reheating, and regeneration.

As shown in Fig. 8:

$$T_1 = T_3, T_2 = T_4$$

In an ideal regenerator, $T_5 = T_9$. In practice (actual regenerator), $T_5 < T_9$.

$$T_8 = T_6, T_7 = T_9$$

The net work input to a two-stage compressor is minimized when equal pressure ratios are maintained across each stage. That is:

$$\frac{P_2}{P_1} = \frac{P_4}{P_3}$$

This procedure also maximizes the turbine work output.

$$\frac{P_6}{P_7} = \frac{P_8}{P_9}$$