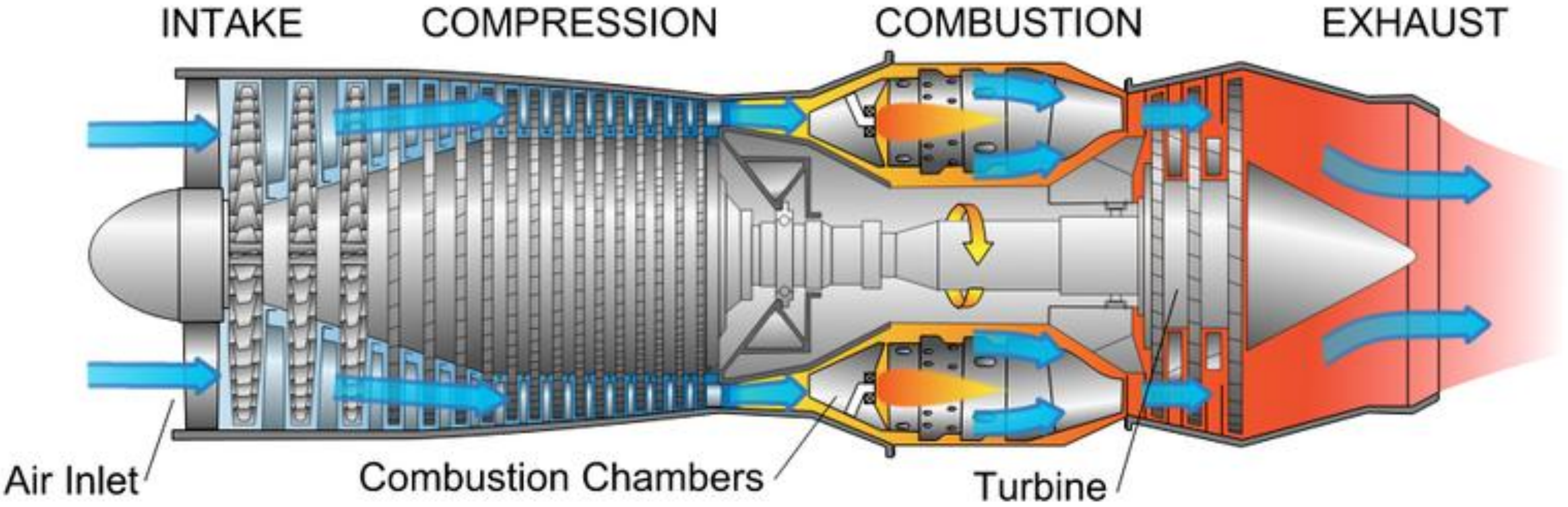
ENG 3165 LECTURE 9

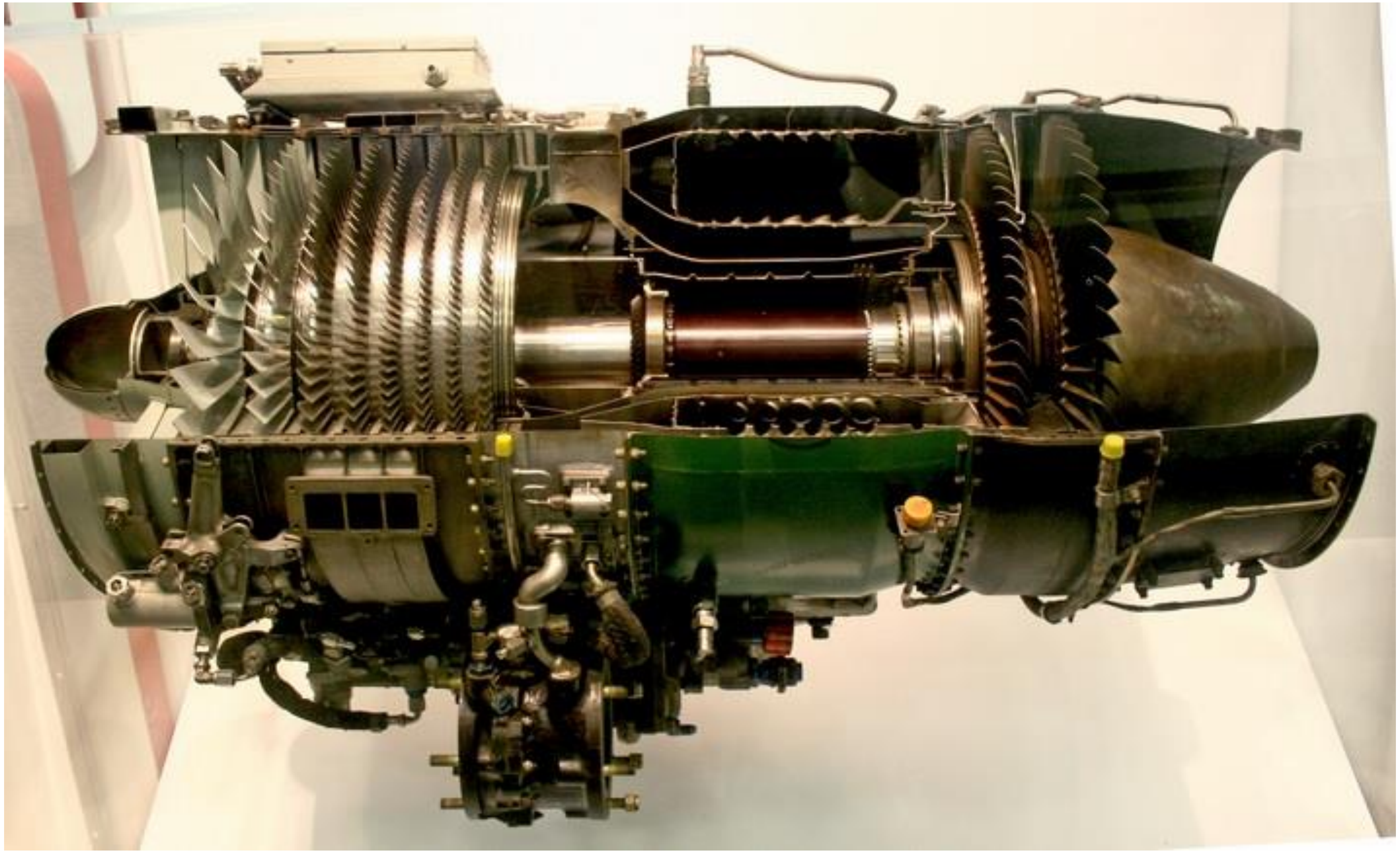
THERMODYNAMICS COMPONENT

Gas Power Cycles Continued…..

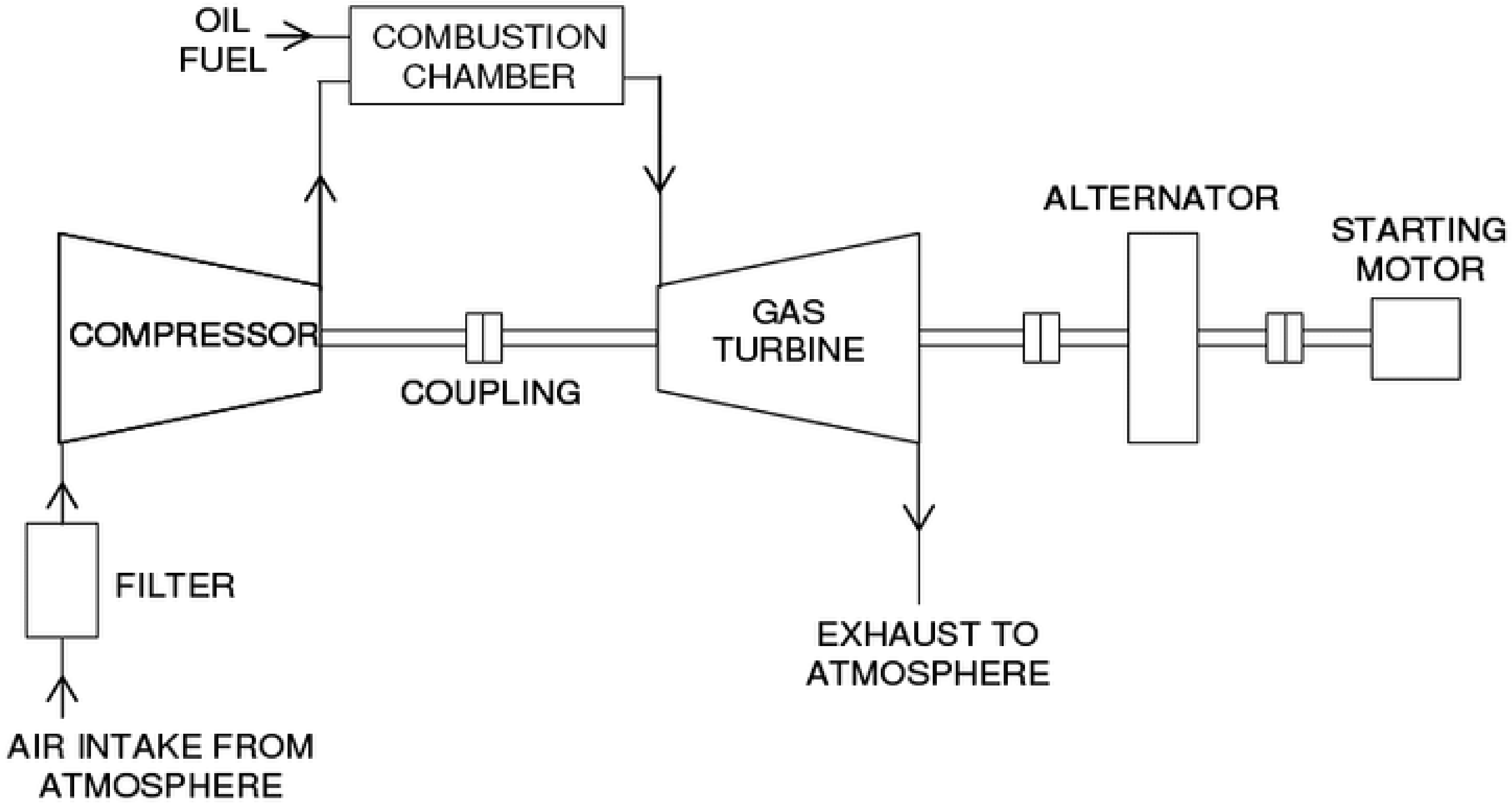
BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES



BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES



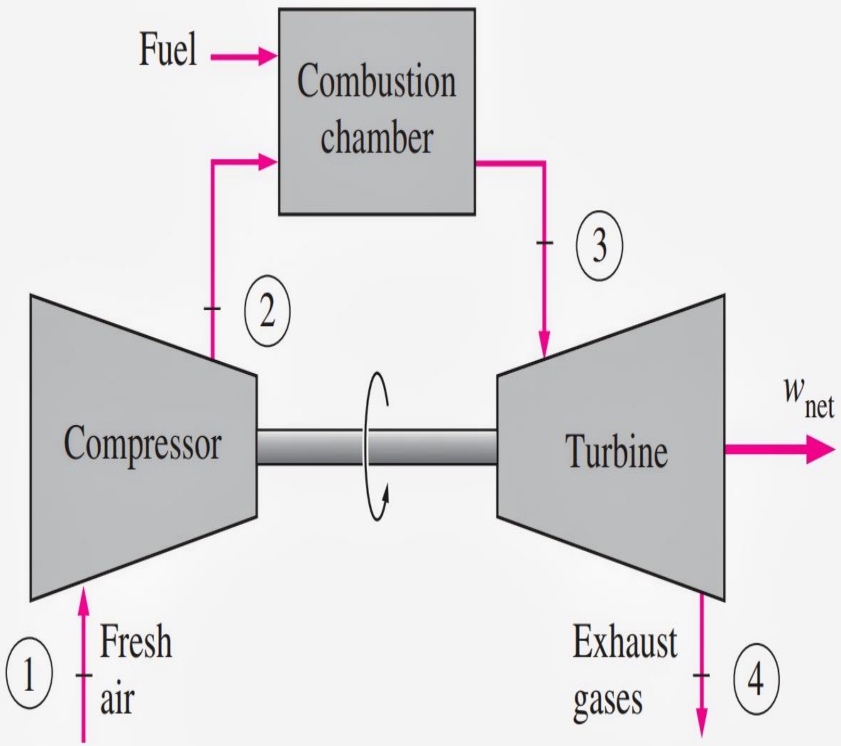
BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES



BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

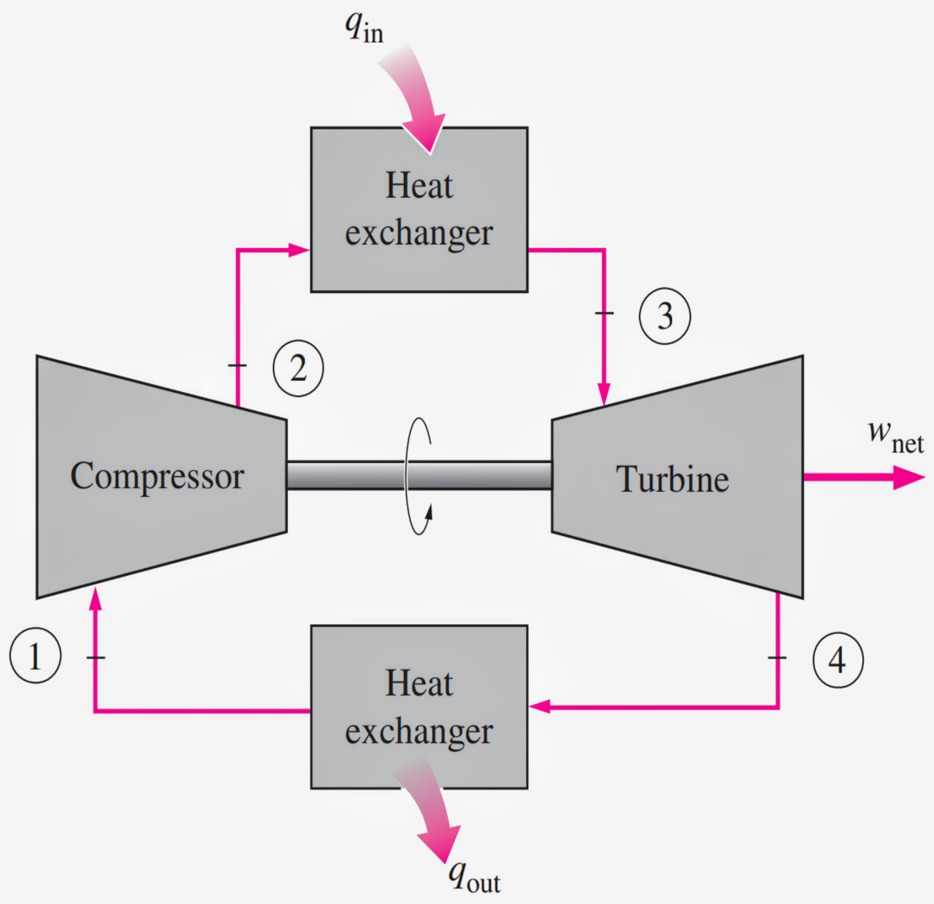
- The Brayton cycle was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery.
- Gas turbines usually operate on an open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

1. Fresh air at ambient conditions is drawn into the **compressor**, where its temperature and pressure are raised.
2. The high pressure air proceeds into the **combustion chamber**, where the fuel is burned at constant pressure.
3. The resulting high-temperature gases then enter the **turbine**, where they expand to the atmospheric pressure while producing power.
4. 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 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.
- The ideal cycle that the working fluid undergoes in this closed loop is the **Brayton cycle**



BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

- The Brayton Cycle which is made up of four internally reversible processes:

  1-2 Isentropic compression (in a compressor)

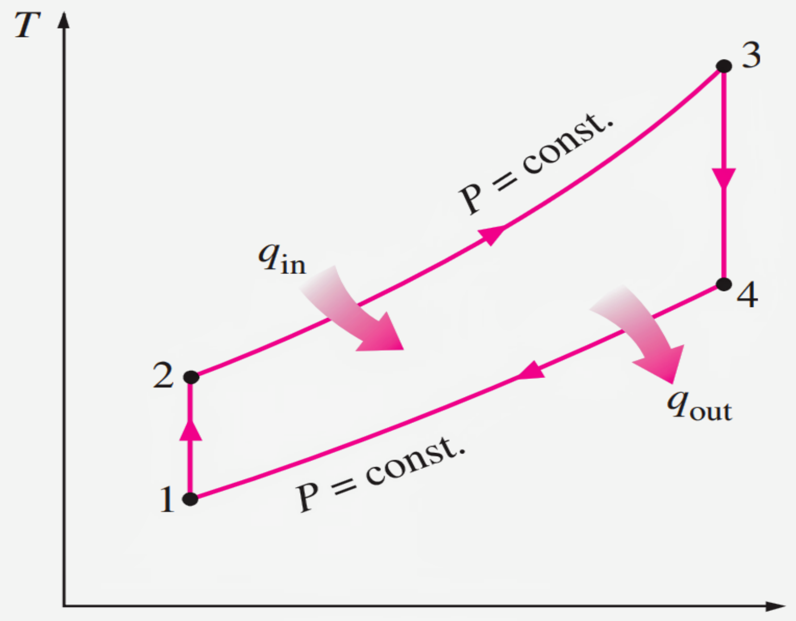
  2-3 Constant-pressure heat addition

  3-4 Isentropic expansion (in a turbine)

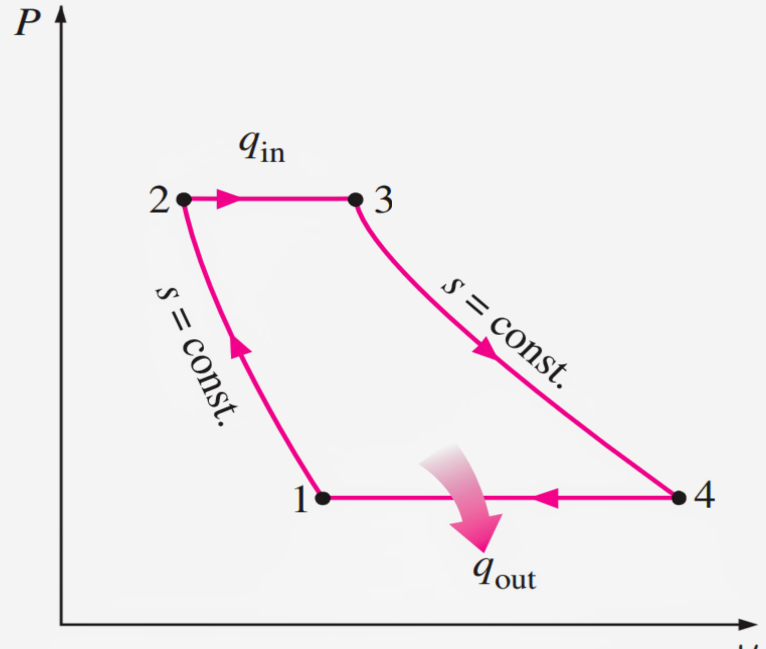
  4-1 Constant-pressure heat rejection

- 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



(a) T-s diagram



(b) P-v diagram

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

- 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

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_{exit} - h_{inlet} \quad \text{(kJ/kg}\ldots\ldots\ldots\ldots\text{(1)}$$

- Note that since the compressor is coupled to the turbine then,

Net work output =Turbine output – Compressor work

Turbine output = $\dot{m}c_p(T_3 - T_4)$

Compressor work = $\dot{m}c_p(T_2 - T_1)$

- Therefore, heat transfers to and from the working fluid are

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

$$q_{out} = h_4 - h_1 = c_p(T_4 - T_1) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3$$

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

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

$$\eta_{th,Brayton} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)} \ldots\ldots\ldots\ldots\ldots(4)$$

- Processes 1-2 and 3-4 are isentropic, and P2 = P3 and P4 = P1. Thus,

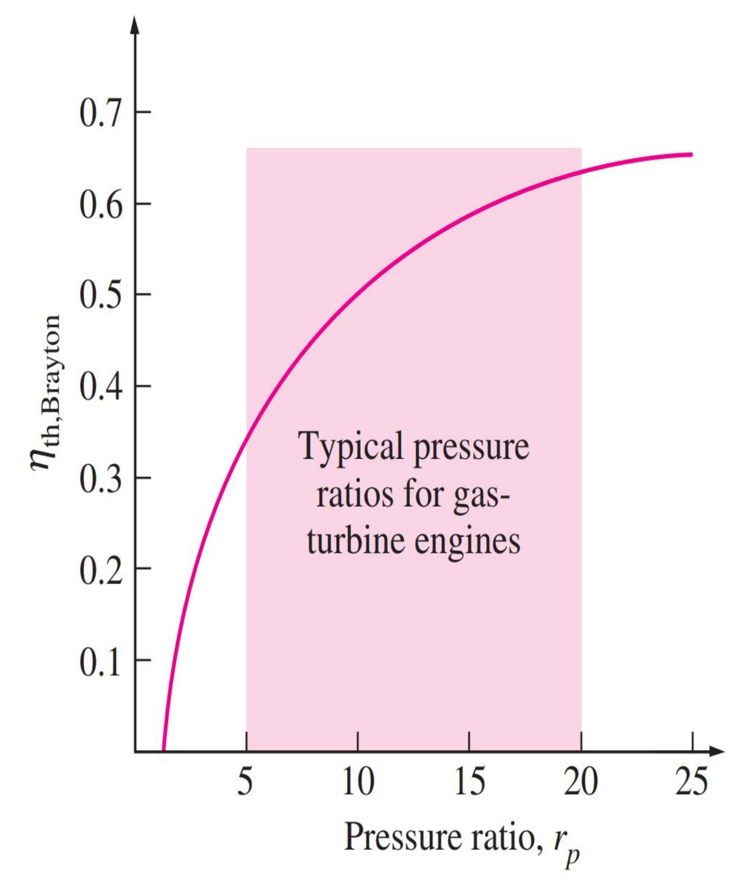
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k} = \frac{T_3}{T_4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

- Substituting these equations into the thermal efficiency relation and simplifying gives

$$\boldsymbol{\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}}} \ldots\ldots\ldots\ldots(6)$$

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

- Where $r_p = \frac{P_2}{P_1}$ is the pressure ratio and k is the specific heat ratio.
- Equation 6 shows that under the cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle depends on the pressure ratio of the gas turbine and the specific heat ratio of the working fluid. The thermal efficiency increases with both of these parameters, which is also the case for actual gas turbines.
- A plot of thermal efficiency versus the pressure ratio is shown for k = 1.4, which is the specific-heat-ratio value of air at room temperature.



BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

- The highest temperature in the cycle occurs at the end of the combustion process (state 3), and it 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.
- For a fixed turbine inlet temperature T3, the net work output per cycle increases with the pressure ratio, reaches a maximum, and then starts to decrease. Therefore, there should be a compromise between the pressure ratio (thus the thermal efficiency) and the net work output.
- With less work output per cycle, a larger mass flow rate (thus a larger system) is needed to maintain the same power output, which may not be economical.
- In most common designs, the pressure ratio of gas turbines ranges from about 11 to 16

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

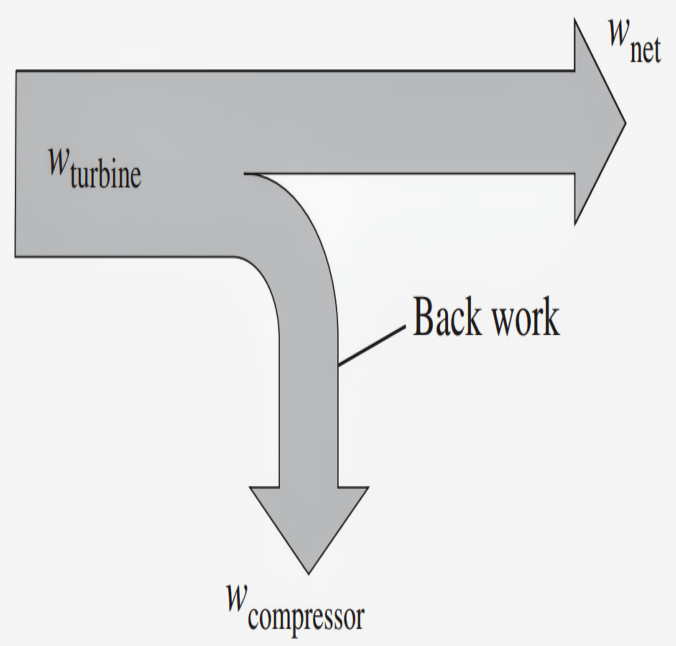
- The air in gas turbines performs two important functions: It supplies the necessary oxidant for the combustion of the fuel, and it serves as a coolant to keep the temperature of various components within safe limits.
- The second function is accomplished by drawing in more air than is needed for the complete combustion of the fuel. In gas turbines, an air–fuel mass 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.
- Also, the mass flow rate through the turbine is greater than that through the compressor, the difference being equal to the mass flow rate of the fuel. Thus, assuming a constant mass flow rate throughout the cycle yields conservative results for open-loop gas-turbine engines.

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

- The two major application areas of gas-turbine engines are aircraft propulsion and electric power generation. When it is used for aircraft propulsion, the gas turbine produces just enough power to drive the compressor and a small generator to power the auxiliary equipment. The high-velocity exhaust gases are responsible for producing the necessary thrust to propel the aircraft.
- Gas turbines are also used as stationary power plants to generate electricity as stand-alone units or in conjunction with steam power plants on the high-temperature side. In these plants, the exhaust gases of the gas turbine serve as the heat source for the steam.
- The gas-turbine cycle can also be executed as a closed cycle for use in nuclear power plants. This time the working fluid is not limited to air, and a gas with more desirable characteristics (such as helium) can be used.

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

- 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. The situation is even worse when the isentropic efficiencies of the compressor and the turbine are low.
- This is quite in contrast to steam power plants, where the back work ratio is only a few percent. This is not surprising, however, since a liquid is compressed in steam power plants instead of a gas, and the steady-flow work is proportional to the specific volume of the working fluid.
- A power plant with a high back work ratio requires a larger turbine to provide the additional power requirements of the compressor.
- Therefore, the turbines used in gas-turbine power plants are larger than those used in steam power plants of the same net power output.



Example 1 - The Simple Ideal Brayton Cycle

A gas turbine operating on a simple constant pressure cycle has a pressure compression ratio of 8:1. The turbine has a thermal efficiency of 60% of the ideal. The fuel used has a calorific value of 43 MJ/kg. If $\gamma = 1{\cdot}4$, determine:

(a) the actual thermal efficiency of the turbine;
(b) the specific fuel consumption of the turbine in kg/kW h.

(a) Ideal thermal efficiency $= 1 - \dfrac{1}{r_v^{(\gamma-1)}}$ where $r_v = \dfrac{V_1}{V_2}$

But the pressure ratio only is given $= \dfrac{P_2}{P_1} = \underline{8}$

Now,

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{(\gamma-1)/\gamma} = \left(\frac{V_2}{V_1}\right)^{(\gamma-1)}$$

$$\therefore \left(\frac{1}{r_v}\right)^{(\gamma-1)} = \left(\frac{V_2}{V_1}\right)^{(-1)} = \left(\frac{P_1}{P_2}\right)^{(\gamma-1)/\gamma}$$

Example 1 - The Simple Ideal Brayton Cycle

∴ Ideal thermal efficiency

$$= 1 - \left(\frac{P_1}{P_2}\right)^{(\gamma-1)/\gamma} = 1 - \frac{1}{8^{(1\cdot4-1)/1\cdot4}} = 1 - \frac{1}{8^{0\cdot4/1\cdot4}}$$

$$= 1 - \frac{1}{8^{1/3\cdot5}} = 1 - \frac{1}{1\cdot81} = 1 - 0\cdot552$$

$$= \underline{0\cdot448}$$

$$= 0\cdot448 \times 100$$

$$= \underline{44\cdot8\,\%}$$

∴ Actual thermal efficiency $= 44\cdot8 \times 0\cdot6 = \underline{26\cdot88\,\%}$

(b) Energy to work/kg of fuel

$$= 43\,000 \times 0\cdot268\,8 = \underline{115\,58 \text{ kJ}}$$

Energy equivalent of 1 kW h

$$= \underline{3\,600 \text{ kJ}}$$

∴ Specific fuel consumption $= \frac{3\,600}{11\,558} = \underline{0\cdot311 \text{ kg/kW h}}$

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 heatrejection 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

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

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

- 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 Fig. 9–36. The effect of the turbine and compressor efficiencies on the thermal efficiency of the gas-turbine engines is illustrated below with an example.

Example 1 - An Actual Gas-Turbine Cycle

In a continuous combustion, constant pressure, gas turbine, air is taken into a rotary compressor at a pressure of 100 kN/m^2 and temperature 18°C. It is compressed through a pressure ratio of 8:1 with an isentropic efficiency of 85%. From the compressor, the compressed air is passed to a combustion chamber where its temperature is raised to 1000°C. From the combustion chamber, the high temperature air is passed to a gas turbine in which it is expanded down to 100 kN/m^2 with an isentropic efficiency of 88%. From the turbine, the air is passed to exhaust. If the air used is 4·5 kg/s and neglecting the mass of fuel as small, determine,

(a) the net power output of the turbine plant if the turbine is coupled to the compressor,
(b) the thermal efficiency of the plant,
(c) the work ratio.

Take $c_p = 1{\cdot}006$ kJ/kg K, $\gamma = 1{\cdot}4$.

(a) For the compressor,

$$T_2' = T_1\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} = 291 \times 8^{(1\cdot4-1)/1\cdot4} = 291 \times 8^{0\cdot4/1\cdot4}$$

$$= 291 \times 8^{1/3\cdot5} = 291 \times 1{\cdot}811 = \underline{527\text{ K}}$$

$$\text{Isentropic } \eta_{comp} = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad T_2 - T_1 = \frac{T_2' - T_1}{\text{Isentropic } \eta_{comp}} = \frac{527-291}{0{\cdot}85} = \frac{236}{0{\cdot}85} = \underline{278\text{ K}}$$

$$\therefore \quad T_2 = 291 + 278 = \underline{569\text{ K}}$$

$$t_2 = 569 - 273 = \underline{296°\text{C}}$$

For the turbine

$$T_4' = T_3\left(\frac{P_4}{P_3}\right)^{(\gamma-1)/\gamma} = \frac{1273}{8^{(1\cdot4-1)/1\cdot4}} = \frac{1273}{1\cdot811} = \underline{703\ \text{K}}$$

$$\text{Isentropic } \eta_{\text{tub}} = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$\therefore \quad T_3 - T_4 = (T_3 - T_4') \times \text{Isentropic } \eta_{\text{turb}}$$

$$= (1273 - 703) \times 0\cdot88 = 570 \times 0\cdot88$$

$$= \underline{502\ \text{K}}$$

$$\therefore \quad T_4 = 1273 - 502 = \underline{771\ \text{K}}$$

$$t_4 = 771 - 273 = \underline{498°\text{C}}$$

$$\text{Net power output} = \dot{m}c_p\{(T_3 - T_4) - (T_2 - T_1)\}$$

$$= 4\cdot5 \times 1\cdot006\{502 - 278\}$$

$$= 4\cdot5 \times 1\cdot006 \times 224$$

$$= \underline{1014\ \text{kW}}$$

(b) Thermal $\eta = \dfrac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$

$$= \frac{502 - 278}{1\,273 - 569} = \frac{224}{704} = \underline{0{\cdot}32}$$

$$= 0{\cdot}32 \times 100$$

$$= \underline{32\,\%}$$

(c) Work ratio $= \dfrac{\text{Net cycle work}}{\text{Positive cycle work}}$

$$= \frac{1\,014}{\dot{m}c_p(T_3 - T_4)}$$

$$= \frac{1\,014}{4{\cdot}5 \times 1{\cdot}006 \times 502}$$

$$= \frac{1\,014}{2\,273}$$

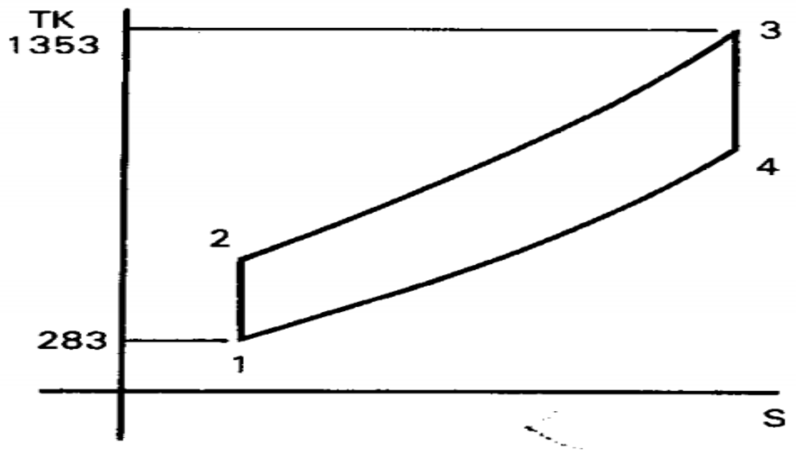
Example 2 - An Actual Gas-Turbine Cycle

A gas turbine plant has the temperature limits of 1080°C and 10°C. Compression in the compressor and expansion in the turbine are isentropic.

Take, $\gamma = 1{\cdot}41$ and $c_p = 1{\cdot}007$ kJ/kg K

Determine,

(a) the pressure ratio which will give the maximum net work output,
(b) the maximum net specific work output,



(c) the thermal efficiency at maximum work output,
(d) the work ratio at maximum work output,
(e) the Carnot efficiency within the cycle temperature limits.

(a) $T_3 = 1080 + 273 = \underline{1353 \text{ K}}$

$T_1 = 10 + 273 = \underline{283 \text{ K}}$

Maximum pressure ratio $= r_{pmax} = \left(\frac{T_3}{T_1}\right)^{\gamma/(\gamma-1)}$ (see equation (16))

$$= \left(\frac{1353}{283}\right)^{1\cdot41/(1\cdot41-1)}$$

$$= 4{\cdot}78^{1\cdot41/0\cdot41}$$

$$= 4{\cdot}78^{3\cdot44}$$

$$= \underline{217{\cdot}4}$$

For maximum net work output,

Pressure ratio $= r_p = \sqrt{r_{pmax}}$ (see equation 22)

$$= \sqrt{217{\cdot}4}$$

$$= 14{\cdot}74$$

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

$\therefore \quad T_2 = T_1 r_p^{(\gamma-1)/\gamma} = 283 \times 14{\cdot}74^{(1{\cdot}41-1)/1{\cdot}41}$

$= 283 \times 14{\cdot}74^{0.41/1{\cdot}41} = 283 \times 14{\cdot}74^{1/3{\cdot}44}$

$= 283 \times 2{\cdot}19$

$= \underline{620\ \text{K}}$

$= T_4$ (see equation (23))

Maximum net specific work output

$= c_p\{(T_3 - T_4) - (T_2 - T_1)\}$

$= 1{\cdot}007\{(1\,353 - 620) - (620 - 283)\}$

$= 1{\cdot}007\{733 - 337\}$

$= 1{\cdot}007 \times 396$

$= \underline{399\ \text{kJ}}$

(c) Thermal $\eta = \dfrac{399}{c_p(T_3 - T_2)}$

$$= \frac{399}{1{\cdot}007(1\,353 - 620)}$$

$$= \frac{399}{1{\cdot}007 \times 733}$$

$$= \frac{399}{738}$$

$$= \underline{0{\cdot}54}$$

$$= 0{\cdot}54 \times 100$$

$$= \underline{54\,\%}$$

(d) Work ratio = Thermal η = $\underline{0{\cdot}54}$ (see equation (26))

(e) Carnot $\eta = \dfrac{T_3 - T_1}{T_3}$

$$= \frac{1\,353 - 283}{1\,353}$$

$$= \frac{1\,070}{1\,353}$$

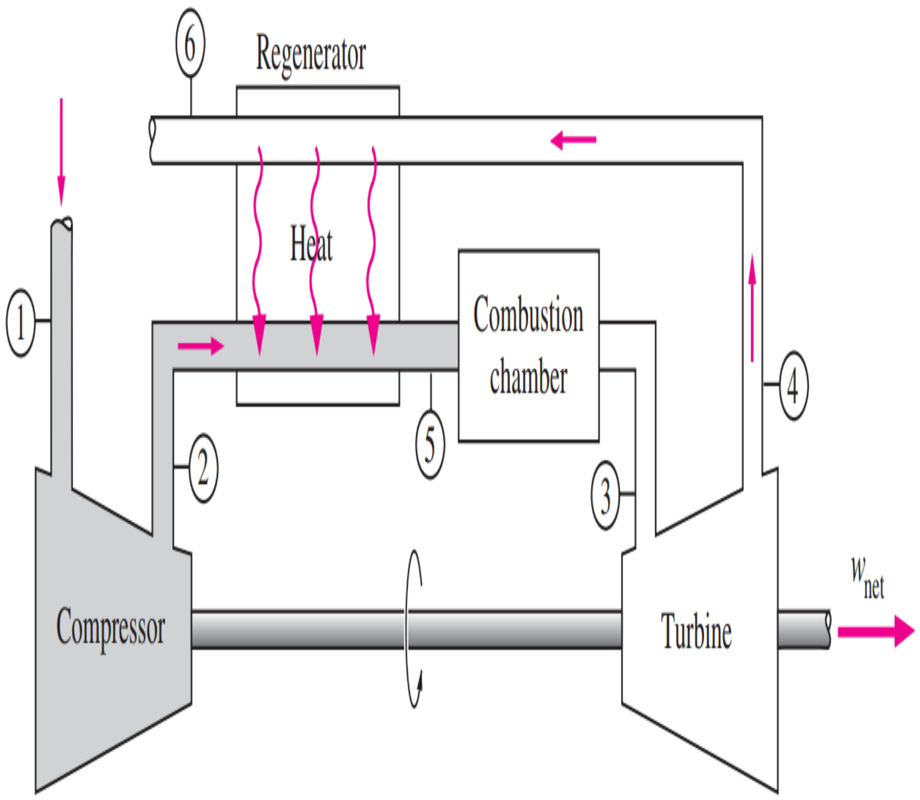
$$= \underline{0{\cdot}79}$$

$$= 0{\cdot}79 \times 100$$

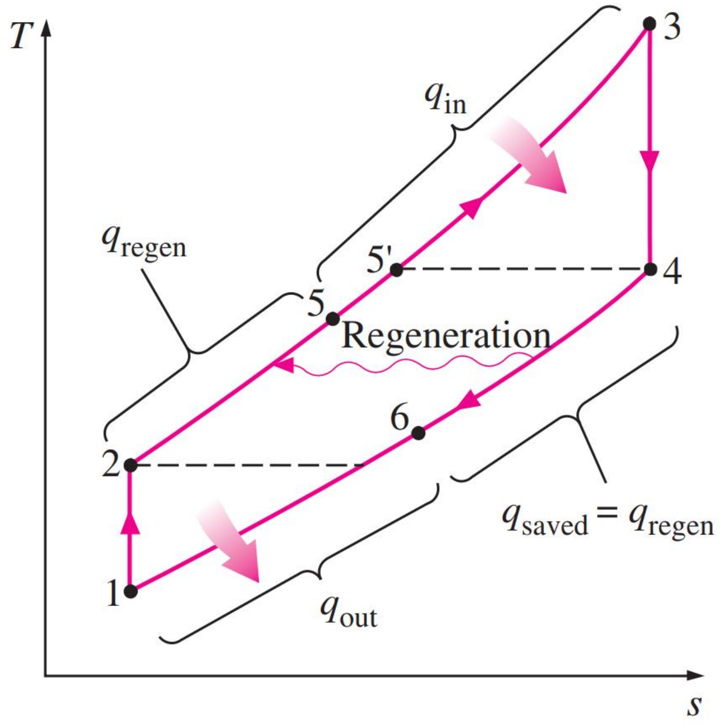
$$= \underline{79\,\%}$$

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
- 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 BRAYTON CYCLE WITH REGENERATION

- The highest temperature occurring within the regenerator is T4, the temperature of the exhaust gases leaving the turbine and entering the regenerator. Under no conditions can the air be preheated in the regenerator to a temperature above this value. Air normally leaves the regenerator at a lower temperature, T5. In the limiting (ideal) case, the air exits the regenerator at the inlet temperature of the exhaust gases T4. Assuming the regenerator to be well insulated and any changes in kinetic and potential energies to be negligible, the actual and maximum heat transfers from the exhaust gases to the air can be expressed as:

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

And

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

THE BRAYTON CYCLE WITH REGENERATION

- The extent to which a regenerator approaches an ideal regenerator is called the **effectiveness** and is defined as

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

- When the cold-air-standard assumptions are utilized, it reduces to

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

- A regenerator with a higher effectiveness obviously saves a greater amount of fuel since it preheats the air to a higher temperature prior to combustion.
- The effectiveness of most regenerators used in practice is below 0.85.

THE BRAYTON CYCLE WITH REGENERATION

- Under the cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle with regeneration is

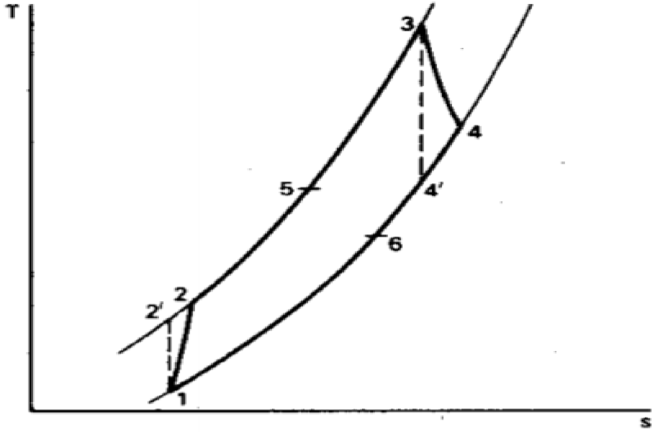
$$\eta_{th,regen} = 1 - \left(\frac{T_1}{T_3}\right) r_p^{(k-1)/k}$$

- 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

An open circuit, continuous combustion, constant pressure, gas turbine takes in air at a pressure of 101 kN/m^2 and at a temperature of 15°C. The air is compressed in a rotary compressor through a pressure ratio of 6:1. The air then passes at constant pressure through a heat exchanger, the effectiveness of which is 65%. From the heat exchanger the air passes at constant pressure through a combustion chamber in which its temperature is raised to 870°C. From the combustion chamber the air passes through a gas turbine in which it is expanded to a pressure of 101 kN/m^2 and it then passes through the heat exchanger to exhaust. The isentropic efficiency of the compressor is 85% while that of the turbine is 80%. Neglect the mass fuel and take the air mass flow rate as 4 kg/s.

Take $\gamma = 1{\cdot}4$ and $c_p = 1{\cdot}005$ kJ/kg K.
Determine:

(a) the nett power output of the plant;
(b) the exhaust temperature from the heat exchanger;
(c) the thermal efficiency of the plant;
(d) the thermal efficiency of the plant if there was no heat exchanger;
(e) the work ratio.



Example 3 - Actual Gas-Turbine Cycle with Regeneration

(a) For the compressor

$$T_2' = T_1\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} = 288 \times 6^{(1\cdot4-1)/1\cdot4} = 288 \times 6^{0\cdot4/1\cdot4}$$

$$= 288 \times 6^{1/3\cdot5}$$

$$= 288 \times 1\cdot67 = \underline{481\ \mathrm{K}}$$

$$0\cdot85 = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad T_2 - T_1 = \frac{T_2' - T_1}{0{\cdot}85} = \frac{481 - 288}{0{\cdot}85} = \frac{193}{0{\cdot}85} = \underline{227\ \text{K}}$$

$$\therefore \quad T_2 = 227 + 288 = \underline{515\ \text{K}}$$

$$t_2 = 515 - 273 = \underline{242°C}$$

For the turbine

$$T_4' = T_3\left(\frac{P_4}{P_3}\right)^{(\gamma-1)/\gamma} = 1\,143 \times \left(\frac{1}{6}\right)^{(1{\cdot}4-1)/1{\cdot}4} = \frac{1\,143}{1{\cdot}67} = \underline{684\ \text{K}}$$

$$0{\cdot}80 = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$\therefore \quad T_3 - T_4 = 0.80(T_3 - T_4') = 0{\cdot}80(1\,143 - 684) = 0{\cdot}80 \times 459$$

$$= \underline{367\ \text{K}}$$

$$\therefore \quad T_4 = 1\,143 - 367 = \underline{776\ \text{K}}$$

$$t_4 = 776 - 273 = \underline{503°C}$$

$$\text{Net power output} = \dot{m}c_p\{(T_3 - T_4) - (T_2 - T_1)\}$$

$$= 4 \times 1{\cdot}005 \times \{367 - 227\}$$

$$= 4 \times 1{\cdot}005 \times 140$$

$$= \underline{563\ \text{kW}}$$

(b) Maximum temperature drop available for heat transfer $= (T_4 - T_2)$

Actual temperature drop $= 0{\cdot}65(T_4 - T_2)$

$$= 0.65(776 - 515) = 0{\cdot}65 \times 261$$

$$= \underline{170\ \text{K}}$$

$\therefore$ Exhaust temperature from heat exchanger

$$= T_6 T_4 - 170 = 776 - 170 = \underline{606\ \text{K}}$$

$$t_6 = 606 - 273 = \underline{333°\text{C}}$$

(c) Thermal $\eta = \dfrac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_5)}$

and $T_5 = T_2 + 170 = 515 + 170 = \underline{685\ \text{K}}$

$\therefore$ Thermal $\eta = \dfrac{367 - 227}{1\,143 - 685} = \dfrac{140}{458}$

$$= \underline{0{\cdot}306}$$

$$= 0{\cdot}306 \times 100$$

$$= \underline{30{\cdot}6\,\%}$$

(d) With no heat exchanger,

$$\text{Thermal } \eta = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$$

$$= \frac{367 - 227}{1\,143 - 515} = \frac{140}{628}$$

$$= \underline{0{\cdot}223}$$

$$= 0{\cdot}223 \times 100$$

$$= \underline{22{\cdot}3\,\%}$$

(c) Work ratio $= \dfrac{\text{Net cycle work}}{\text{Positive cycle work}}$

$$= \frac{563}{\dot{m}c_p(T_3 - T_4)}$$

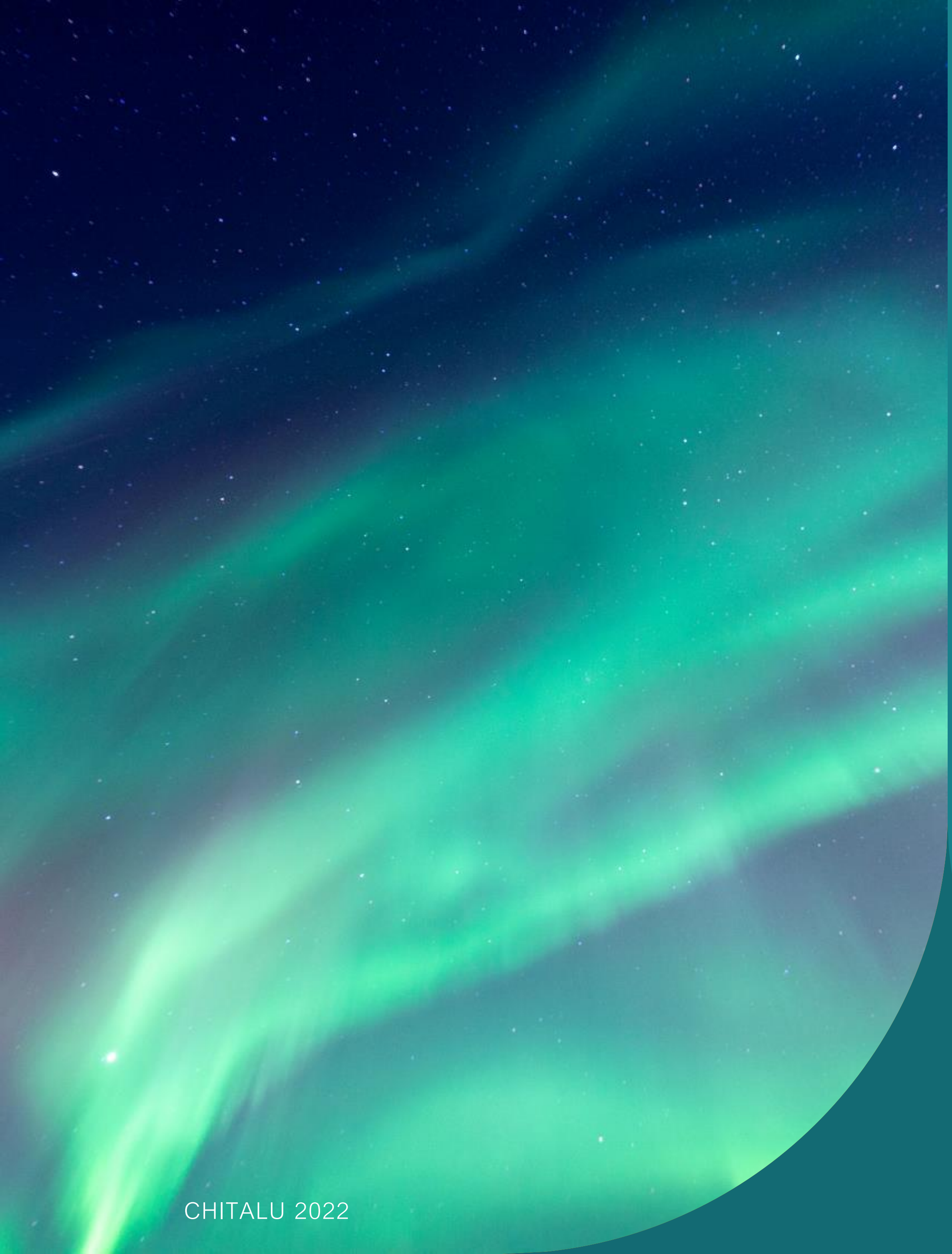
$$= \frac{563}{4 \times 1{\cdot}005 \times 367}$$

$$= \frac{563}{1\,475}$$

$$= \underline{0{\cdot}38}$$

“History, like thermodynamics, won’t let you out”

Ira Haron



Thank You

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