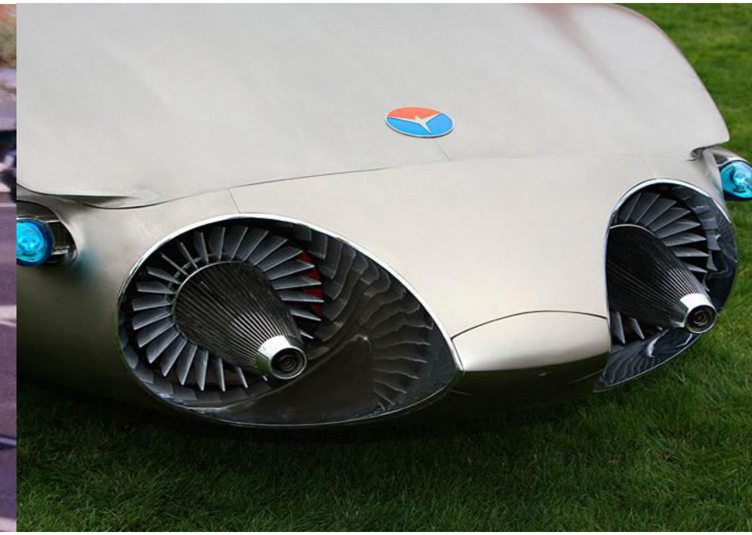
Combined gas-vapour cycles

Introduction to gas cycles

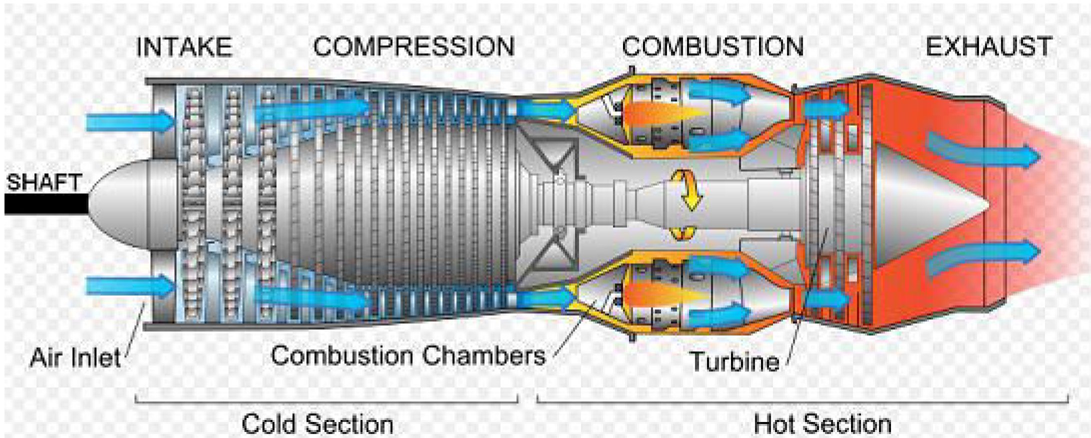
- In gas cycles the **working fluid** does **not undergo** any **phase change** ⇒ all the heat transfer processes are sensible processes, and hence are non-isothermal
- Among the gas cycles, the **Brayton cycle** is **most widely used** in many applications including for large scale power generation
- Since Brayton cycle employs a gas turbine for generating power, Brayton cycle is also called as a "gas turbine" cycle
- All **commercial aircraft** systems are **based on** the **gas turbine** cycle
- Gas turbines are **also used** in various industries for **driving** mechanical and electrical equipment such as **compressors, pumps** etc
- Due to their **high power-to-weight ratios**, gas turbines were also used in some of the **racing cars** and there are **efforts to use them in railways also**!

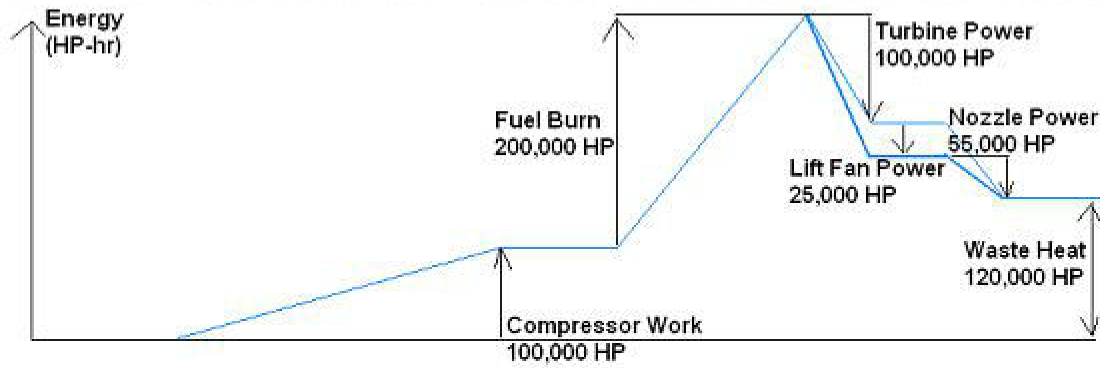
Chrysler gas turbine car

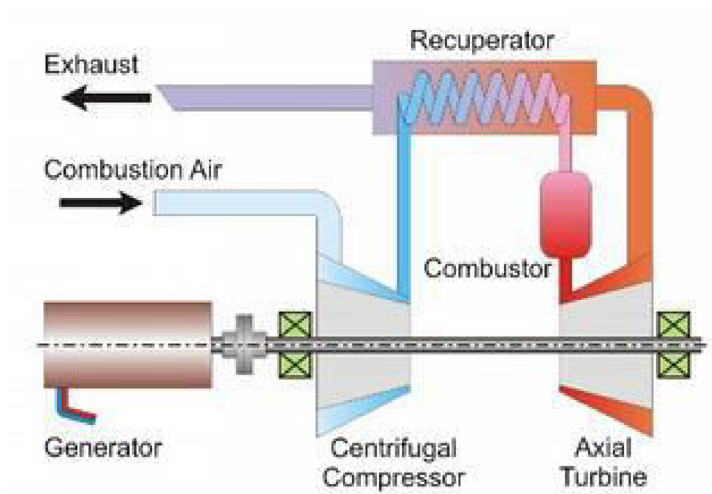


Close view of a gas turbine

Bombardier’s experimental JetTrain, 2002 (Canada)


INTAKE
COMPRESSION
COMBUSTION
EXHAUST
SHAFT
Air Inlet
Combustion Chambers
Turbine
Cold Section
Hot Section


Energy
(HP-hr)
Fuel Burn
200,000 HP
Turbine Power
100,000 HP
Nozzle Power
55,000 HP
Lift Fan Power
25,000 HP
Waste Heat
120,000 HP
Compressor Work
100,000 HP


Recuperator
Exhaust
Combustion Air
Combustor
Generator
Centrifugal
Compressor
Axial
Turbine

Introduction to gas turbines (contd.)

- **Compared to steam power plants**, **gas turbine** based power plants **offer** several **advantages**:

1. **For the same output**, they are **smaller in size** and **lighter in weight**
2. **For the same output**, they also **cost less**
3. They take **less time to procure**, **install** and **commission**
4. They are **quick-starting** and **run smoothly**
5. They **can use a wide variety** of **liquid or gaseous fuels**, e.g. **natural gas**, **fuel oil**, **syngas**, **naphtha**, **crude oil etc.**
6. **Environmentally**, they can provide **better emissions** with fewer restrictions

Introduction to gas turbines (contd.)

- However, **gas turbines** do suffer from some **major disadvantages**:
- For the **same maximum and minimum temperatures**, their **efficiency is much lower** compared to a vapour cycle
- ⇒ They are **not preferred** for **continuous**, stand-alone power generation applications
- They are **not compatible** with **solid fuels** such as **coal**

  ⇒ However, using **gasification**, **solid fuels** can be **converted** into **gaseous fuels** (e.g. **syngas**) and used in gas turbine plants, e.g. *integrated gasification combined cycle* (*IGCC*)

Introduction to gas turbines (contd.)

- Considering **the low initial cost**, but **low efficiency of gas turbine** as compared to **high initial cost** and **high efficiency** of **steam power plants**, it would be **advantageous to develop systems**, wherein:
  - Steam power plant would be **operating continuously** at base loads with high load factor and high efficiency
  - while the **gas turbine plant** would be put **into operation**, only during peak loads
- **Alternately**, since due to **improved material and manufacturing techniques**, it is possible to **operate gas turbines** at very high temperatures (as high as 1600°C) with high efficiency, they can be used as:
  - **topping cycles** in **steam power plants**, thus improving the overall plant efficiency tremendously!

Introduction to gas turbines (contd.)

- **Depending upon** the arrangement for **heat supply** and **heat rejection**, gas turbine cycles can be classified into:
  1. Direct open gas turbine cycle
  2. Direct closed gas turbine cycle
  3. Indirect open gas turbine cycle
  4. Indirect closed gas turbine cycle
- Depending upon how they are **coupled to the load**, they can also be classified into:
  1. Single shaft model, or
  2. Two shaft model

Introduction to gas turbines (contd.)

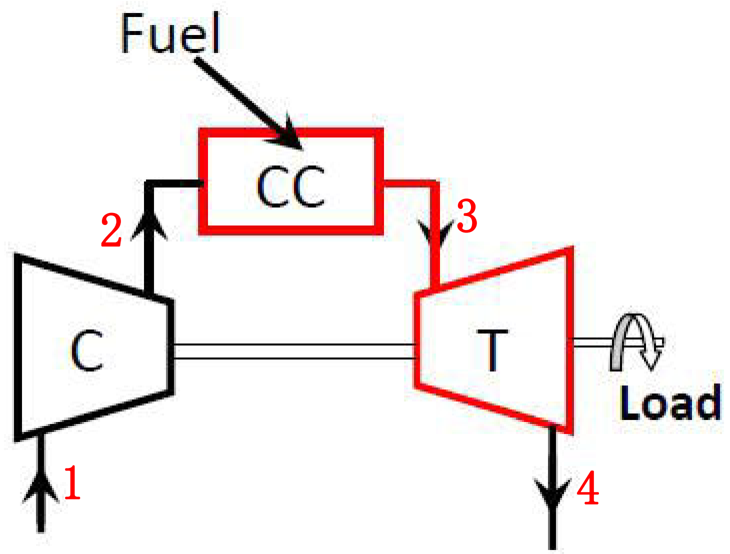
1. Gas enters the compressor (C) at point 1
2. Gas is compressed to point 2
3. Compressed gas enters the combustion chamber (CC) or reactor, as the case may be at point 2
4. Compressed gas is heated to point 3
5. Hot compressed gas at point 3 enters the turbine and expands to a lower pressure at point 4
6. Exhaust gas from turbine at point 4 is expelled into the atmosphere

Note:

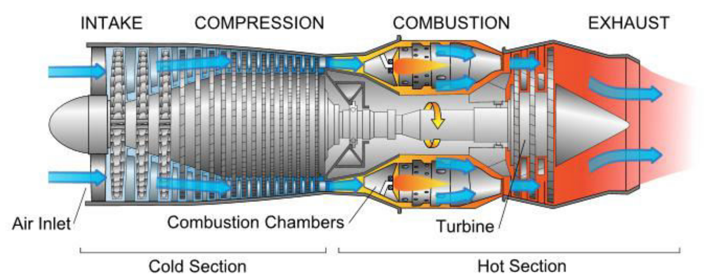
a) Since this is an open cycle, only air can be used as the working fluid

b) The pressure at point 1 and point 4 have to be atmospheric

c) Used in air crafts

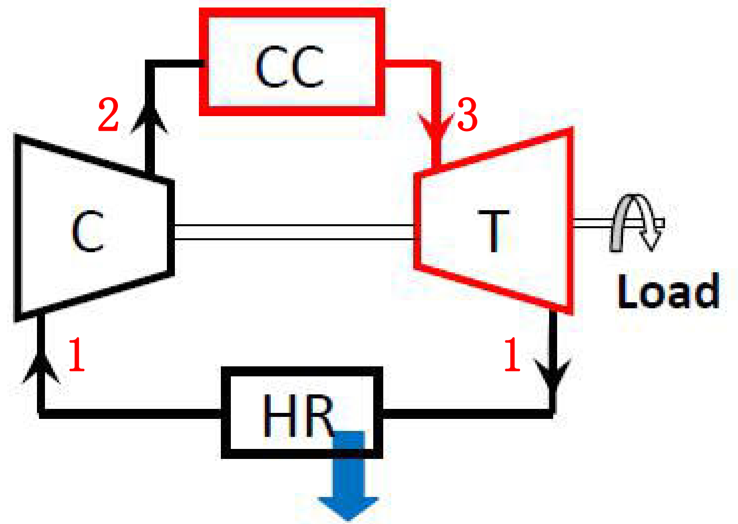


Direct, open gas turbine cycle



Introduction to gas turbines (contd.)

1. Gas enters the compressor (C) at point 1
2. Gas is compressed to point 2
3. Compressed gas enters the combustion chamber (CC) or reactor, as the case may be at point 2
4. Compressed gas is heated to point 3
5. Hot compressed gas at point 3 enters the turbine and expands to a lower pressure at point 4
6. Exhaust gas from turbine at point 4 is cooled in the heat exchanger (HR) to initial condition 1



Direct, closed gas turbine cycle
(Ideal Brayton cycle)

Note:

a) This is a theoretical cycle, since in practice, mass balance cannot be maintained with continuous addition of fuel, unless heat is directly added by some other way.

b) Since this is closed cycle, any gas can be used as the working fluid

c) The pressure at point 1 (and point 4) can be

Introduction to gas turbines (contd.)

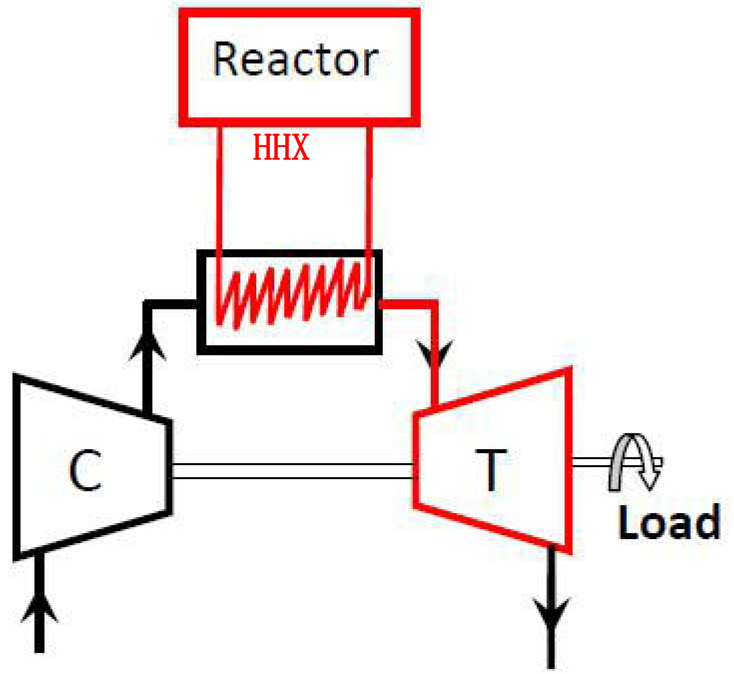
1. Gas enters the compressor (C) at point 1
2. Gas is compressed to point 2
3. Compressed gas enters the high temperature heat exchanger (HHX) at point 2
4. Compressed gas is heated to point 3
5. Hot compressed gas at point 3 enters the turbine and expands to a lower pressure at point 4
6. Exhaust gas from turbine at point 4 is expelled into the atmosphere

Note:

a) Used in applications that prevent direct heating of air, **e.g. in nuclear power stations**

b) Since this is an open cycle, **only air** can be used as the **working fluid**

c) The pressure at point 1 (and point 4) is same as **atmospheric pressure**



Indirect, open gas turbine cycle

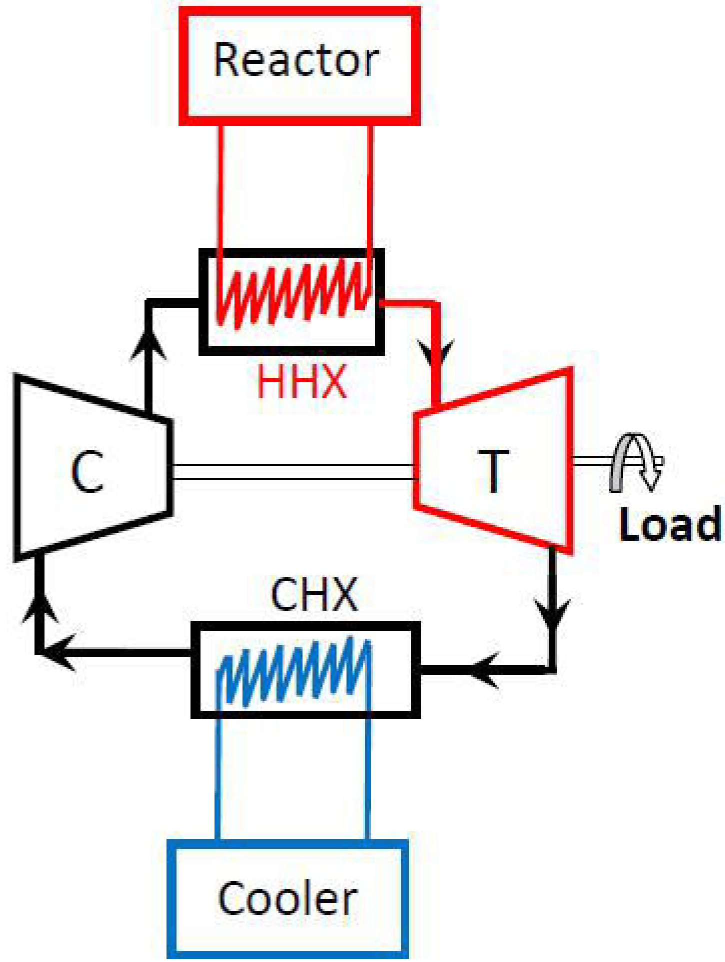
Introduction to gas turbines (contd.)

1. Gas enters the compressor (C) at point 1
2. Gas is compressed to point 2
3. Compressed gas enters the high temperature heat exchanger (HHX) at point 2
4. Compressed gas is heated to point 3
5. Hot compressed gas at point 3 enters the turbine and expands to a lower pressure at point 4
6. Exhaust gas from turbine at point 4 is cooled in the low temperature heat exchanger CHX

Note:

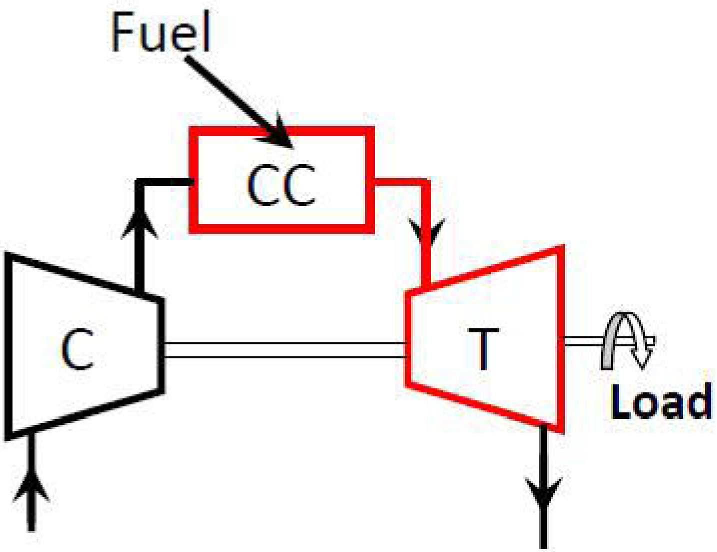
a) Used in applications that prevent direct heating of air, **e.g. in nuclear power stations**

b) The pressure at point 1 (and point 4) can be higher than the **atmospheric pressure**

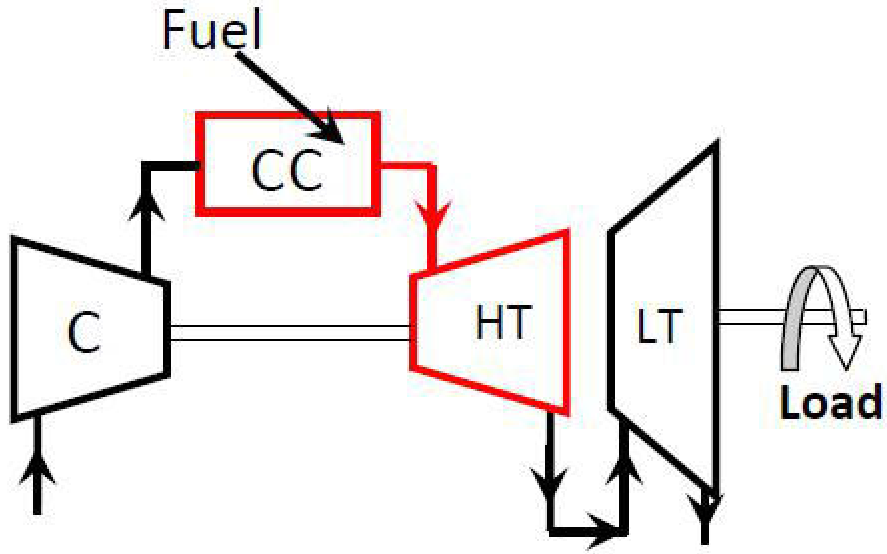


Indirect, closed gas turbine cycle

Introduction to gas turbines (contd.)



Single shaft, open gas turbine cycle



Two shaft, open gas turbine cycle

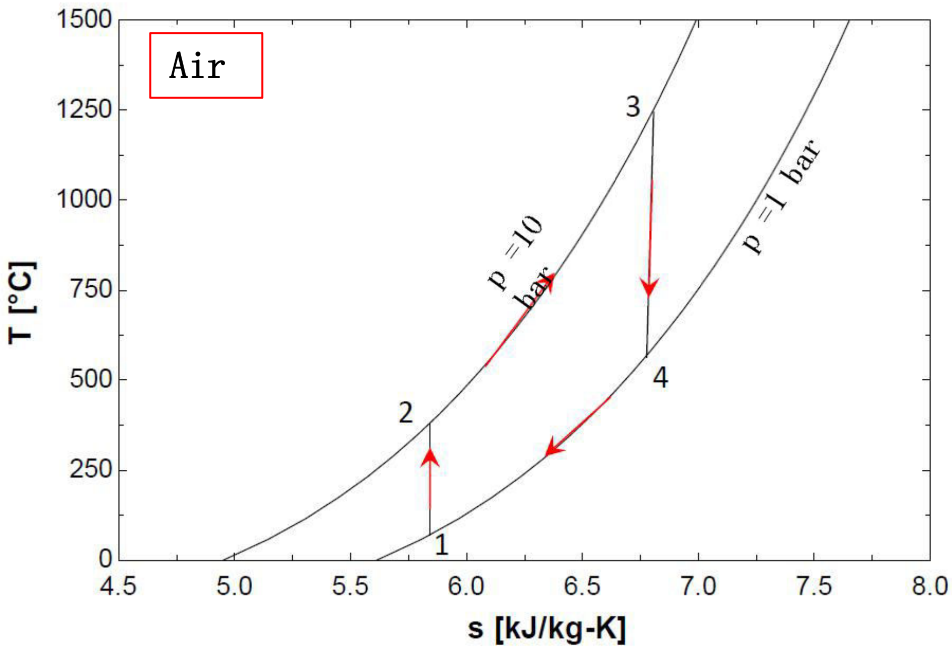
In single shaft systems, the rotational speed of gas turbine and the external load are same, as they are mounted on the same shaft

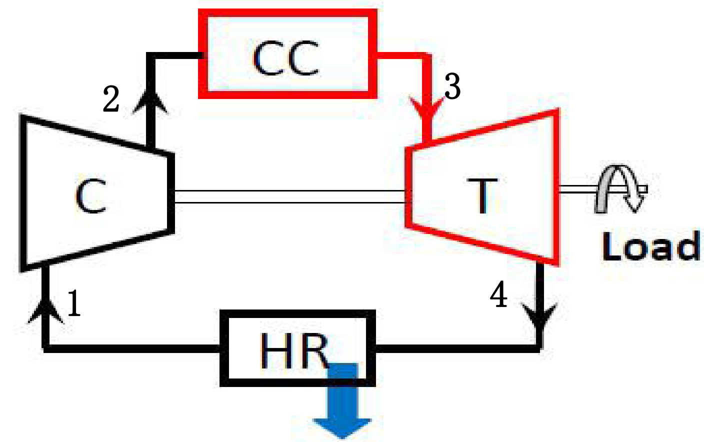
In a two shaft system, the **speed** at which the **gas turbine** and **load** operate can be **different**. This is done by splitting the turbine into two parts - **HT** and **LT**

The **high pressure turbine (HT)** called as gas generator is connected to the compressor and drives the compressor

The low pressure turbine (LT) connected to the load can operate at

Analysis of an Ideal Brayton cycle



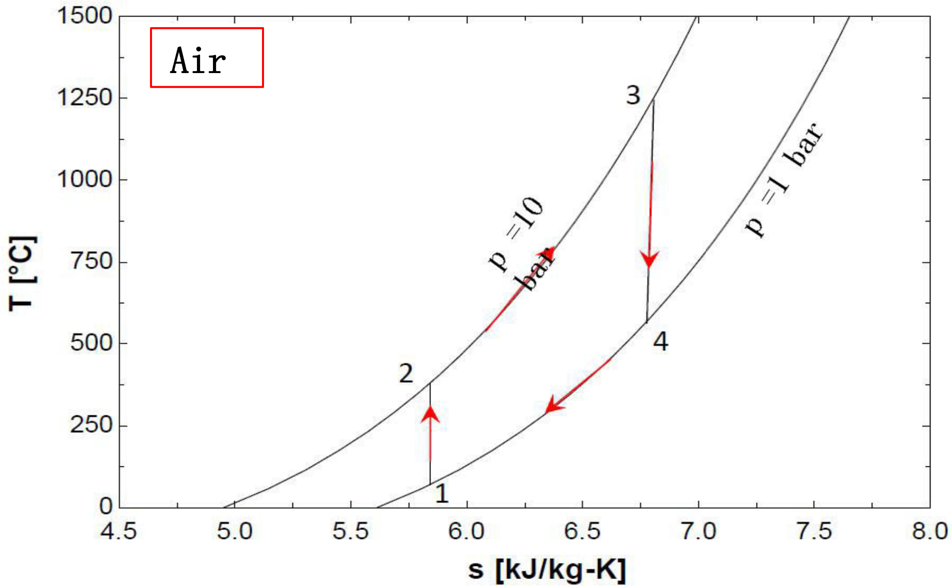


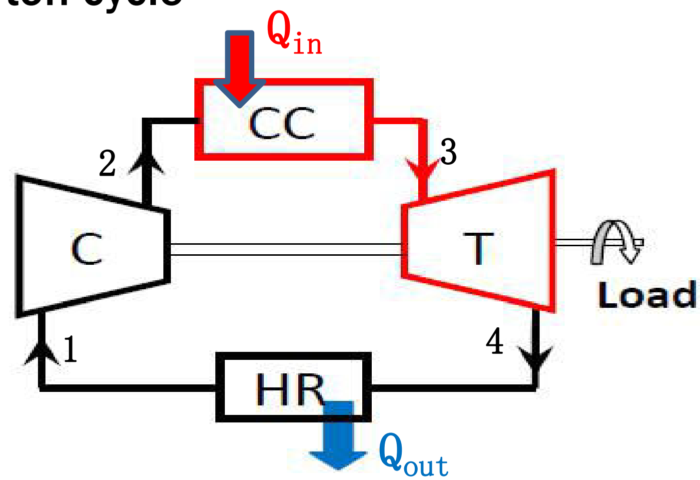
Ideal Brayton cycle

Assumptions: Cold Air Standard Cycle Analysis

1. The **working fluid** is a **pure and ideal fluid** that circulates through the **closed system** without undergoing any change in its **composition**
2. The working fluid **receives heat** from an **external source** (in CC) and **rejects heat** to an external sink (in HR)
3. All the **internal processes** are **reversible**
4. The **specific heat** of the working fluid **(c_p) is constant**
5. The system operates at **steady state**
6. **Kinetic and potential changes** across the components are **negligible**

Analysis of an Ideal Brayton cycle





Ideal Brayton cycle

<u>Applying steady flow energy balance across each component:</u>

1. **Compressor** (Process 1-2: reversible and adiabatic compression)

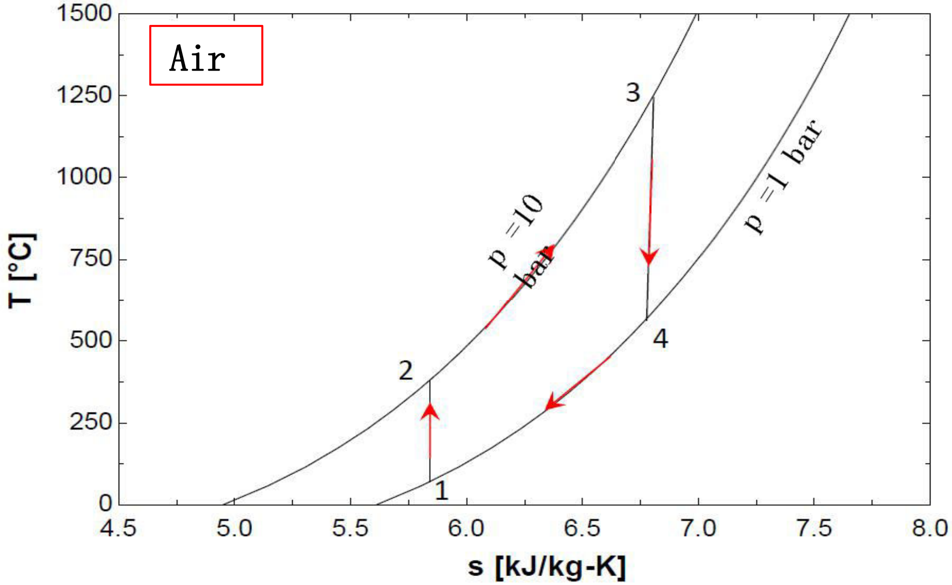
$$\dot{W}_c = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

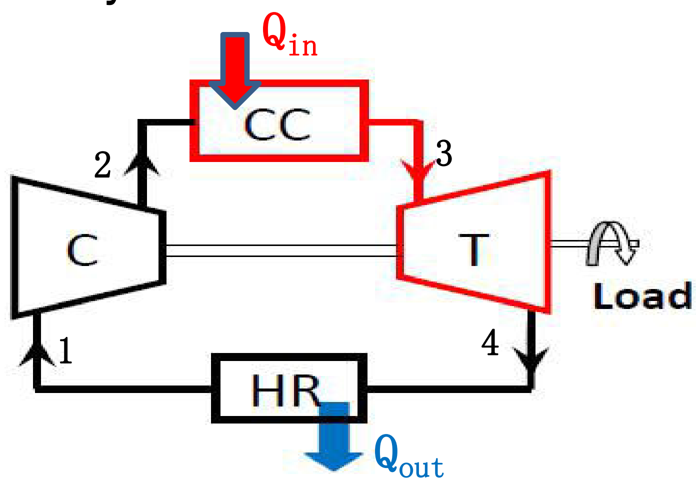
$$s_2 = s_1$$

$$p_1(v_1)^\gamma = p_2(v_2)^\gamma \Rightarrow \left(\frac{p_2}{p_1}\right) = r_{pC} = \left(\frac{v_1}{v_2}\right)^\gamma = \left(\frac{T_2}{T_1}\right)^{\gamma/(\gamma-1)}$$

$$\dot{W}_c = \dot{m}c_p(T_2 - T_1) = \dot{m}c_p T_2\left(1 - \frac{1}{r_{pC}^{(\gamma-1)/\gamma}}\right)$$

Analysis of an Ideal Brayton cycle





Ideal Brayton cycle

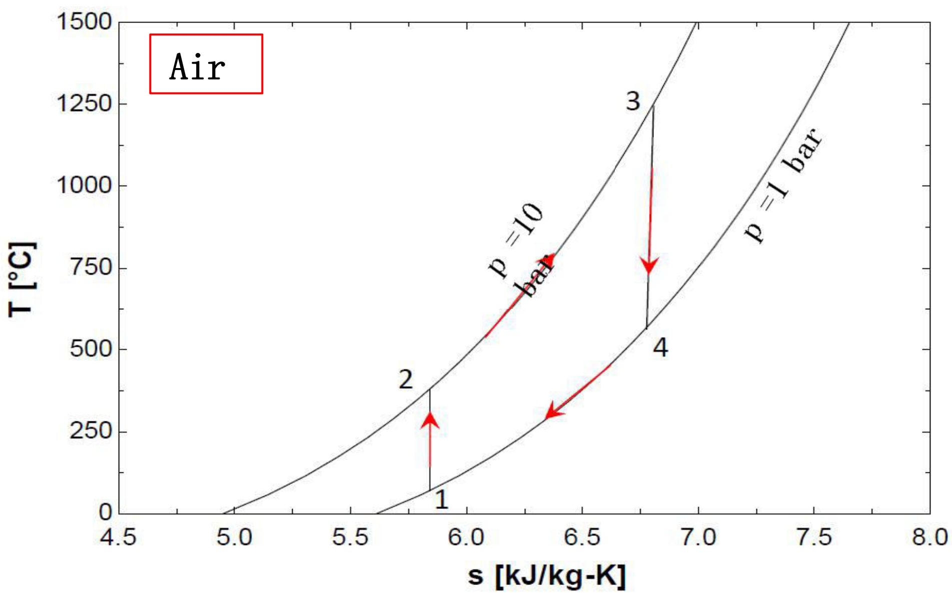
2. High temperature heat exchanger, CC (Process 2-3: Isobaric heat addition):

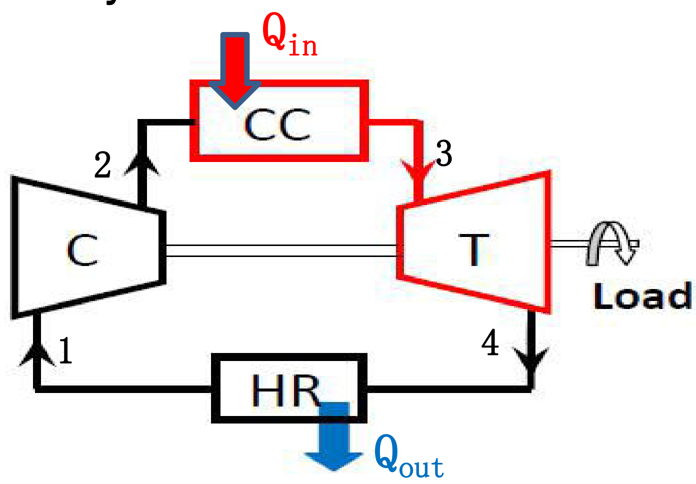
$$Q_{in} = \dot{m}(h_3 - h_2) = \dot{m}c_p(T_3 - T_2)$$

3. Low temperature heat exchanger (Process 4-1: Isobaric heat rejection):

$$Q_{out} = \dot{m}(h_4 - h_1) = \dot{m}c_p(T_4 - T_1)$$

Analysis of an Ideal Brayton cycle





Ideal Brayton cycle

4. **Turbine** (Process 3-4: reversible and adiabatic expansion):

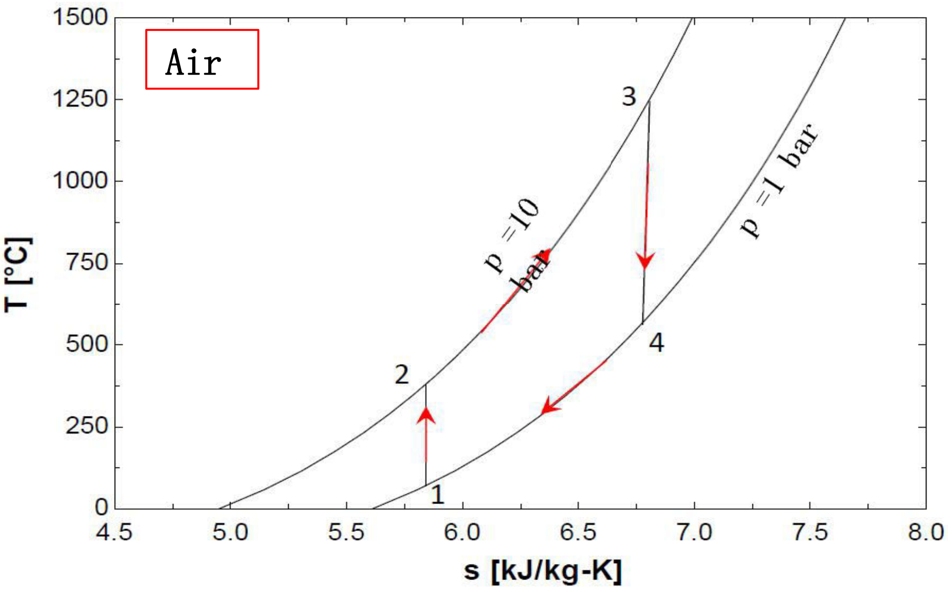
$$\dot{W}_T = \dot{m}(h_3 - h_4) = \dot{m}c_p(T_3 - T_4)$$

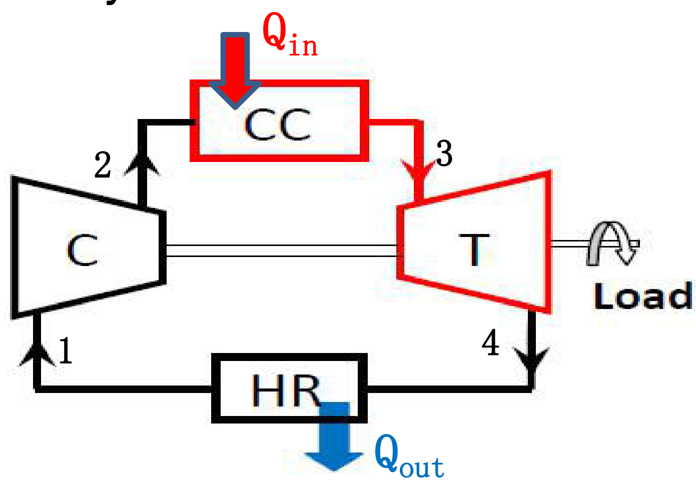
$$s_3 = s_4$$

$$p_3(v_3)^\gamma = p_4(v_4)^\gamma \Rightarrow \left(\frac{p_3}{p_4}\right) = r_{pT} = \left(\frac{v_4}{v_3}\right)^\gamma = \left(\frac{T_3}{T_4}\right)^{\gamma/(\gamma-1)}$$

$$\dot{W}_T = \dot{m}c_p(T_3 - T_4) = \dot{m}c_pT_3\left(1 - \frac{1}{r_{pT}^{(\gamma-1)/\gamma}}\right)$$

Analysis of an Ideal Brayton cycle





Ideal Brayton cycle

From overall energy balance:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C = Q_{in} - \ Q_{out} = \dot{m}c_p(T_3 - T_2) - \dot{m}c_p(T_4 - T_1)$$

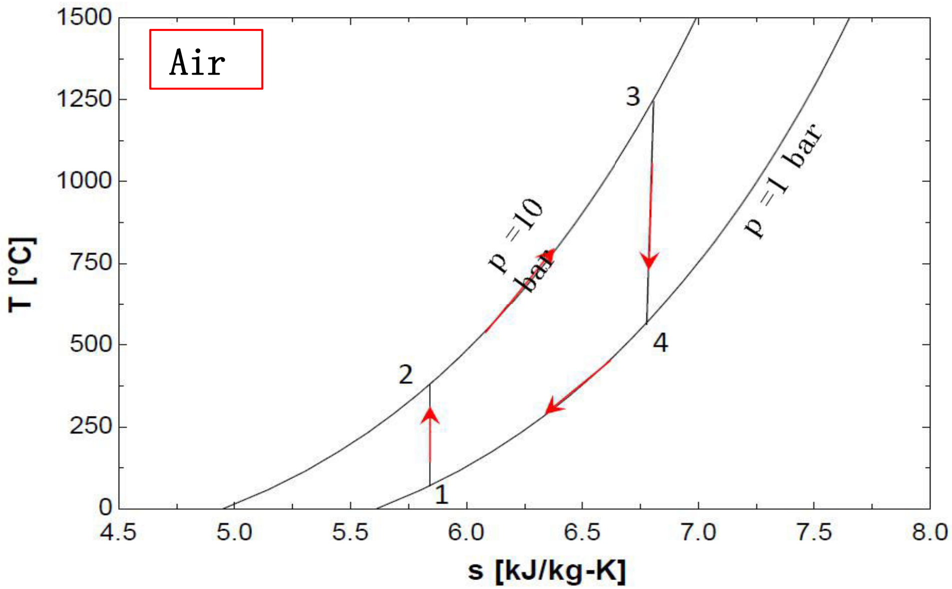
Since there is no pressure drop during heat addition (2-3) and heat rejection

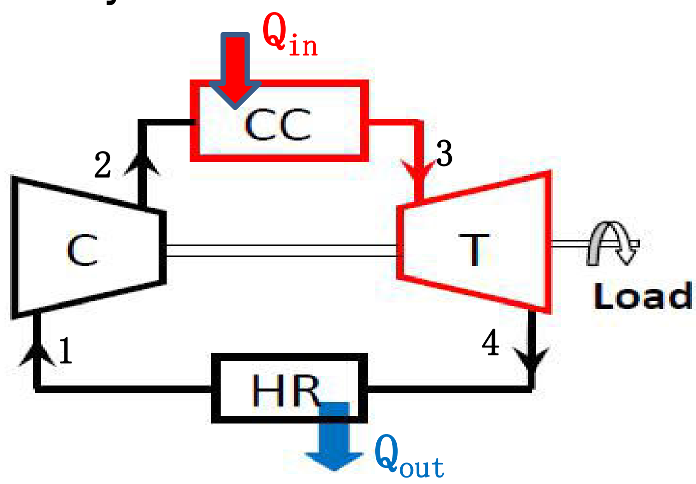
$$r_{pC} = \left(\frac{p_2}{p_1}\right) = \left(\frac{p_3}{p_4}\right) \Rightarrow r_{pC} = r_{pT} = r_p \quad and \quad \left(\frac{T_2}{T_1}\right) = \left(\frac{T_3}{T_4}\right)$$

Therefore, the net power output is given by:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_{C\,out} = \dot{m}c_p(T_3 - T_2)\left(1 - \frac{1}{r_p^{\,(\gamma-1)/\gamma}}\right)$$

Analysis of an Ideal Brayton cycle





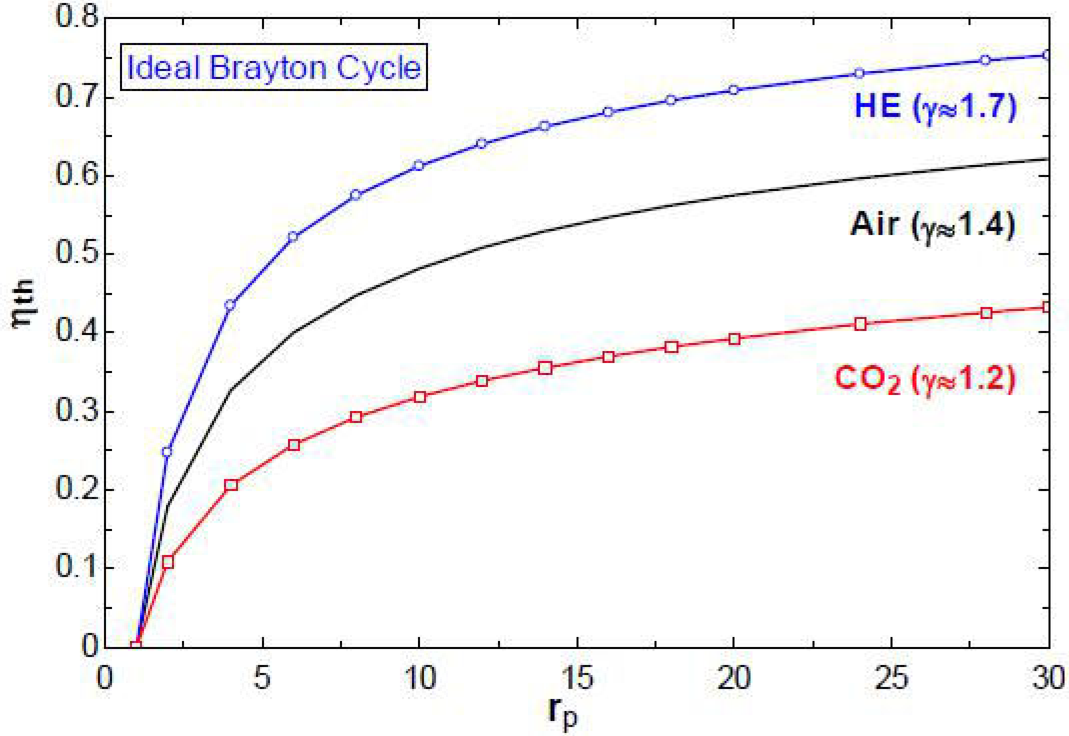
Ideal Brayton cycle

The thermal efficiency η_{th} is given by:

$$\eta_{th} = \frac{\dot{W}_{net}}{Q_{in}} = \left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}} \right) = f(r_p, \gamma)$$

From the above equations it is clear that **for a given working fluid** (**fixed γ**) the **thermal efficiency** of a simple, **ideal Brayton cycle** is **independent** of the minimum and maximum temperatures **(T_1 and T_3)** and **depends only** on the **pressure ratio r_p**, and **increases continuously with r_p**.

Variation of thermal efficiency of a simple Brayton cycle with pressure ratio



$$\eta_{th} = \frac{\dot{W}_{net}}{Q_{in}} = \left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right) = f(r_p, \gamma)$$

Analysis of an Ideal Brayton cycle

However, it can be shown that the **net specific power output** (kJ/kg) **depends on** T_1 and T_3 as well as γ and r_p.

$$\frac{\dot{W}_{net}}{\dot{m}} = c_p(T_3 - T_2)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right) = c_p\left(T_3 - T_1\left(r_p^{(\gamma-1)/\gamma}\right)\right)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right)$$

The above equation shows that:

•For a **given minimum and maximum temperatures** (T_1 and T_3) the **net specific work output increases as**:

1. c_p increases, and/or,
2. γ increases and/or,

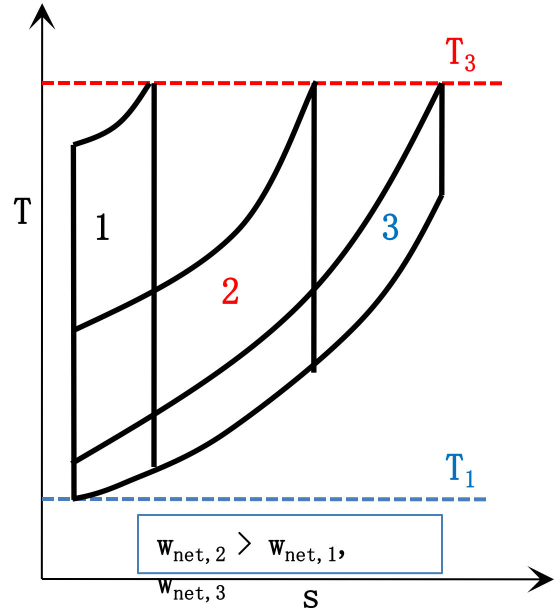
•For **a given gas (fixed values of c_p and γ) and fixed pressure ratio r_p**, the **net specific work output increases** as:

1. maximum temperature T_3 **increases** and/or
2. minimum temperature T_1 **decreases**

Analysis of an Ideal Brayton cycle

$$\frac{\dot{W}_{net}}{\dot{m}} = c_p(T_3 - T_2)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right) = c_p\left(T_3 - T_1\left(r_p^{(\gamma-1)/\gamma}\right)\right)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right)$$

Higher the net specific work output, more compact will be the system as the mass flow rate of the gas for the same net power output will be lower

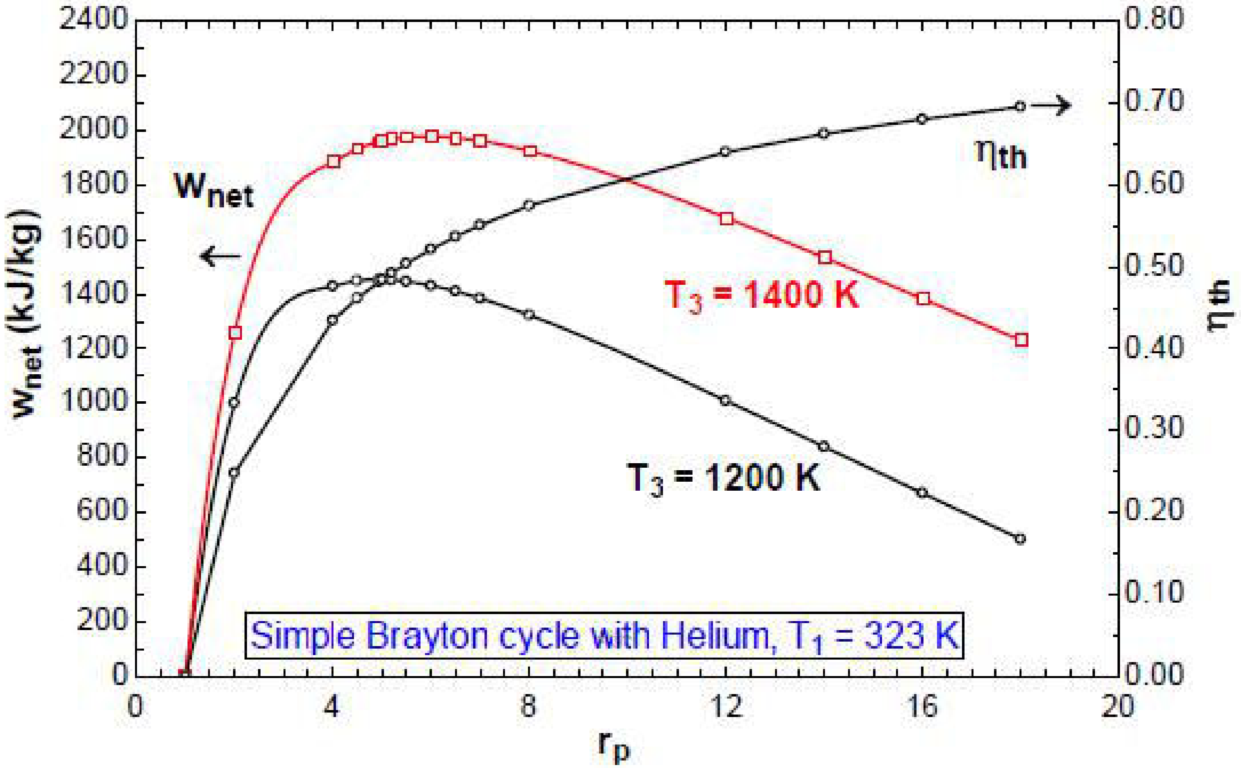


The **maximum temperature *T_3* is limited by the metallurgical considerations**, while the **minimum temperature is limited by the available heat sink**

From the expression for net specific power output it can be shown that for a given gas the **net specific power output reaches a maximum when**:

$$T_{2,opt} = (T_1 . T_3)^{1/2} \Rightarrow r_{p,opt} = \left(\frac{T_{2,opt}}{T_1}\right)^{\left(\gamma/(\gamma-1)\right)} = \left(\frac{T_3}{T_1}\right)^{\left(\gamma/2(\gamma-1)\right)}$$

Analysis of an Ideal Brayton cycle

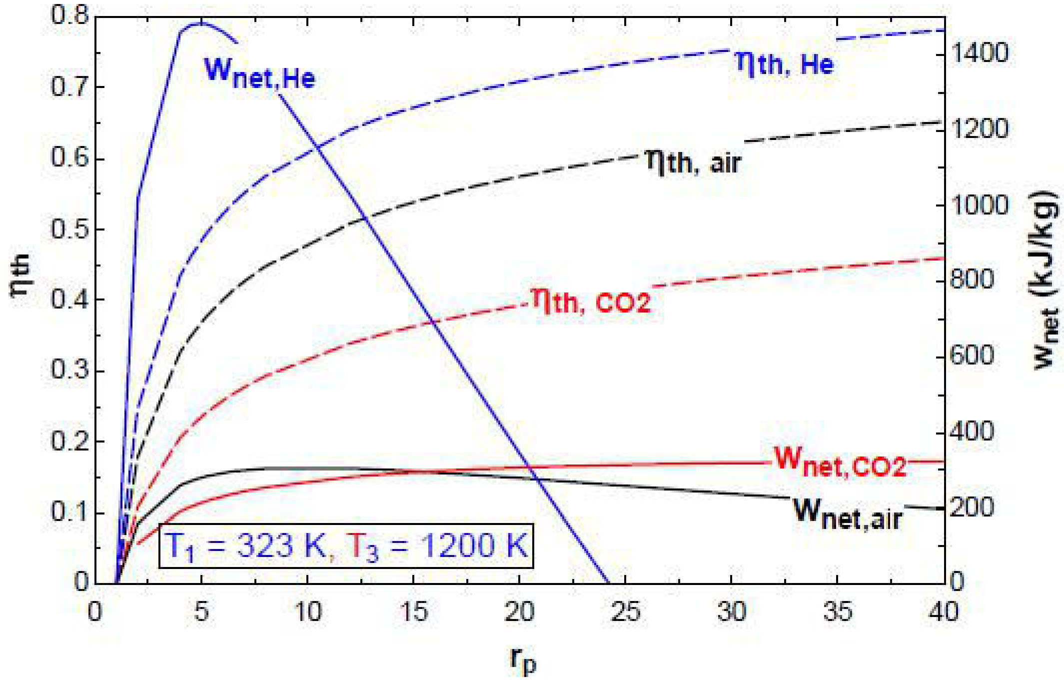


$$\eta_{th} = \frac{\dot{W}_{net}}{Q_{in}} = \left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right) = f(r_p, \gamma)$$

$$\frac{\dot{W}_{net}}{\dot{m}} = c_p(T_3 - T_2)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right) = c_p\left(T_3 - T_1\left(r_p^{(\gamma-1)/\gamma}\right)\right)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right)$$

$$T_{2,opt} = (T_1 . T_3)^{1/2} \Rightarrow r_{p,opt} = \left(\frac{T_{2,opt}}{T_1}\right)^{\left(\gamma/(\gamma-1)\right)} = \left(\frac{T_3}{T_1}\right)^{\left(\gamma/2(\gamma-1)\right)}$$

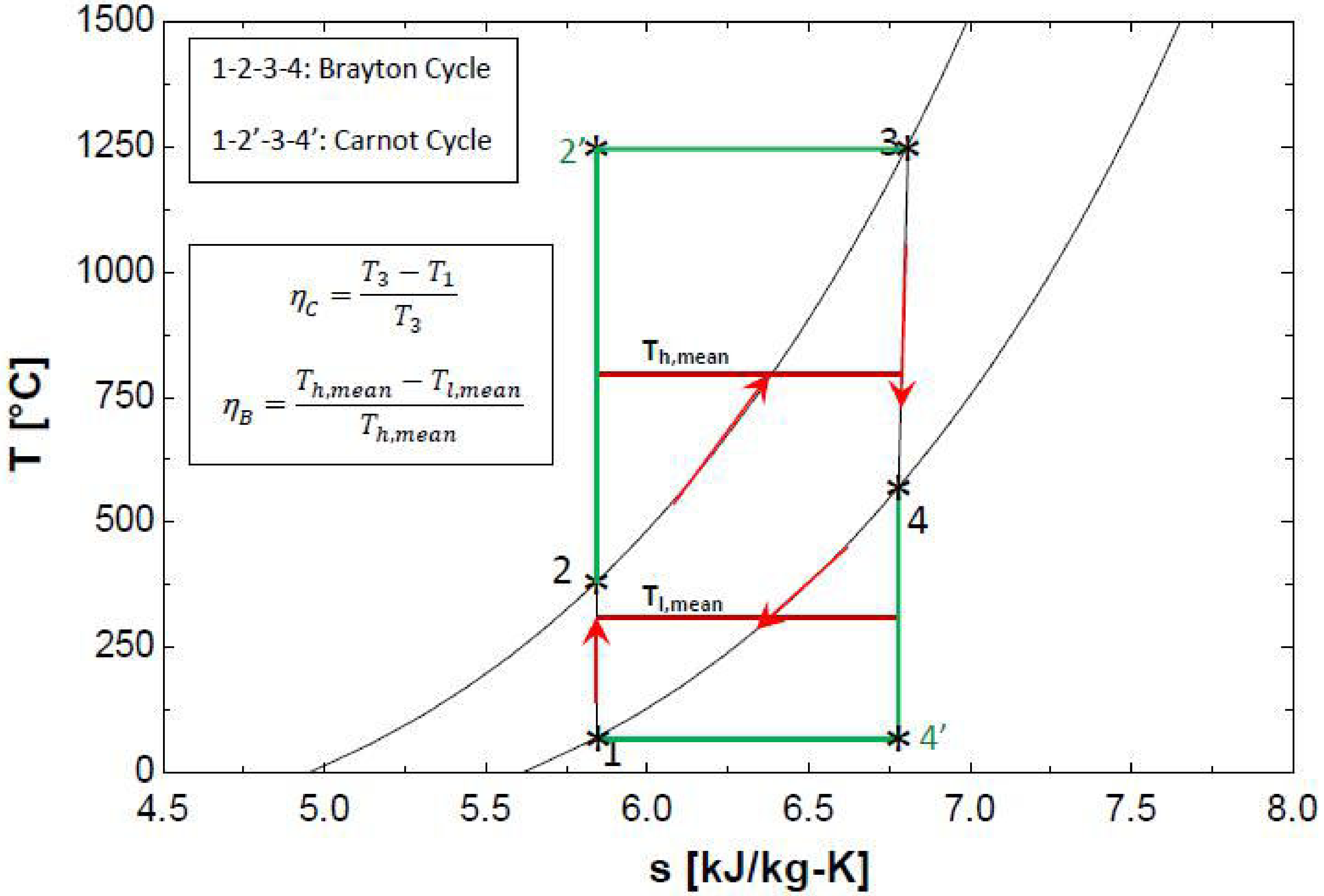
Effect of working fluid and pressure ratio on w_{net} and η_{th}



The above figure shows that:

1. Compared to other gases, Helium offers **very high specific power output** as well as **thermal efficiency** due to **high values of c_p and γ**
2. The **pressure ratio** at which the **net specific power output reaches a maximum** is **much lower for hel**ium compared to other gases

Comparison between Brayton & Carnot Cycles



Actual Brayton cycles

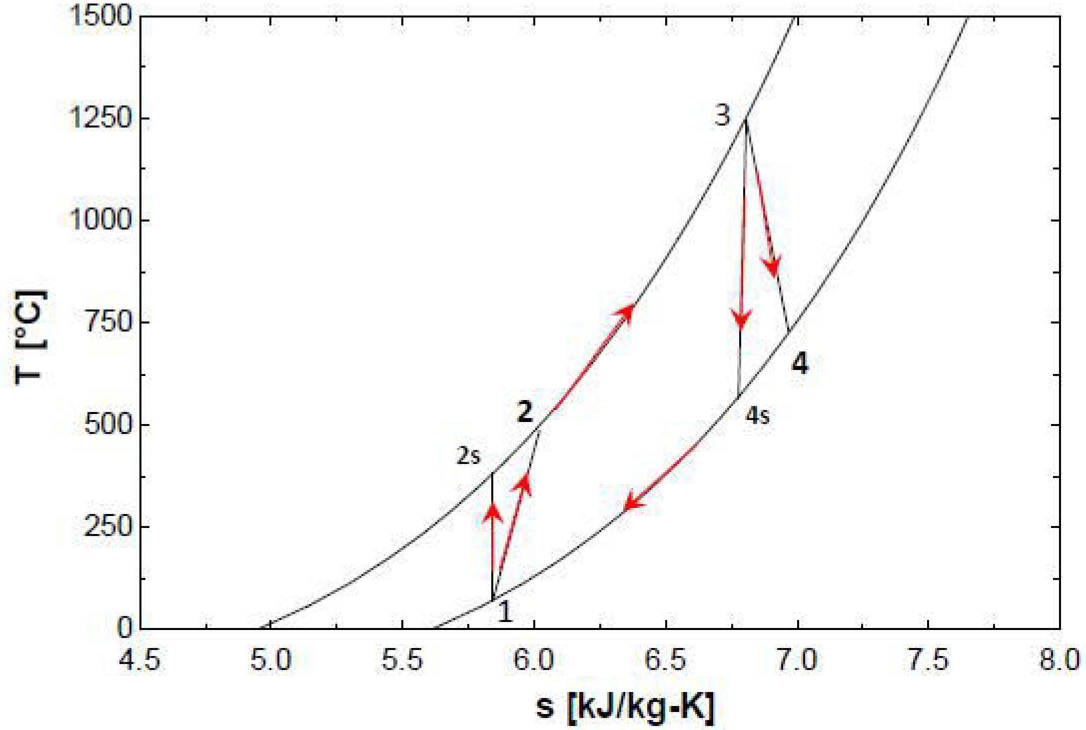
- In actual systems based on Brayton cycle:

1. **Compression and expansion** processes are **non-isentropic**
2. **Heat addition and heat rejection** are **non-isobaric**
3. **Mechanical losses** in bearings etc. **reduce** the useful net **power output**
4. **Properties** of the working fluid **vary** along the cycle due to variation in **gas composition** and **operating conditions**

As a result of the above, the **performance character**istics of the actual Bratyon cycles **differ** from that of an ideal cycle

Actual Brayton cycles

Non-isentropic compression and expansion:



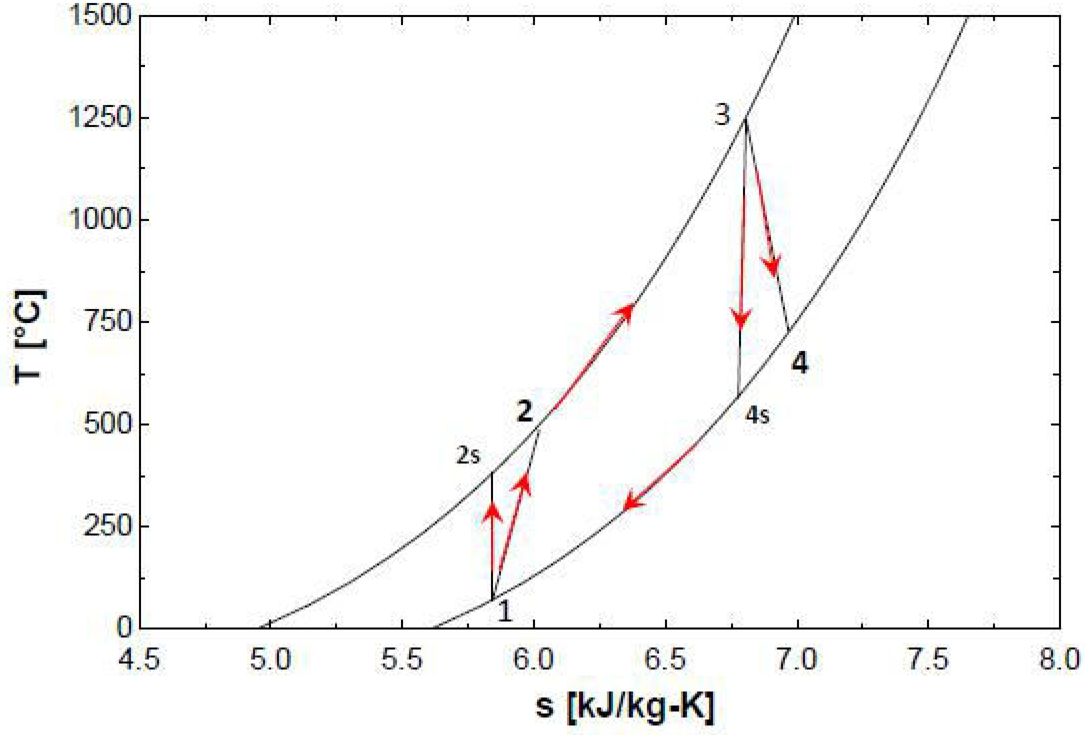
The compressor power input is given by:

$$\dot{W}_c = \dot{m}(h_2 - h_1) = \frac{\dot{m}(h_{2s} - h_1)}{\eta_C} = \frac{\dot{m}.c_p(T_{2s} - T_1)}{\eta_C}$$

Actual Brayton cycles

Non-isentropic compression and

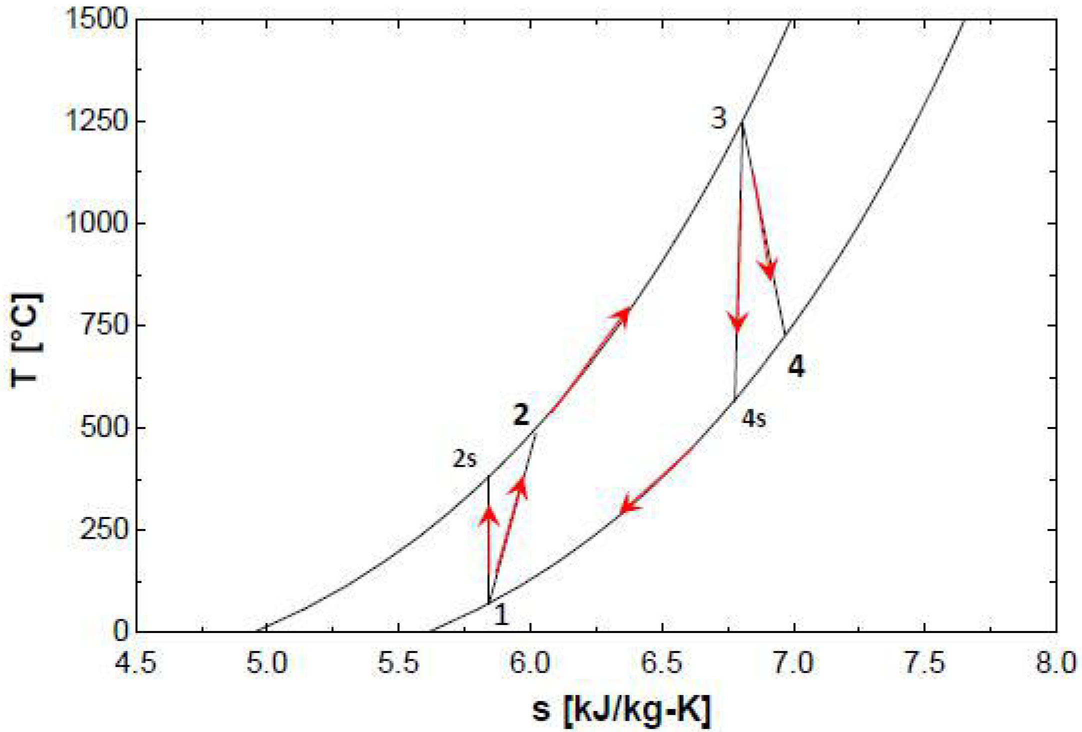
expansion:



The turbine power output is given by:

$$\dot{W}_T = \dot{m}(h_3 - h_4) = \dot{m}.\,\eta_T(h_3 - h_{4s}) = \dot{m}.\,\eta_T.\,c_p(T_3 - T_{4s})$$

Non-isentropic compression and



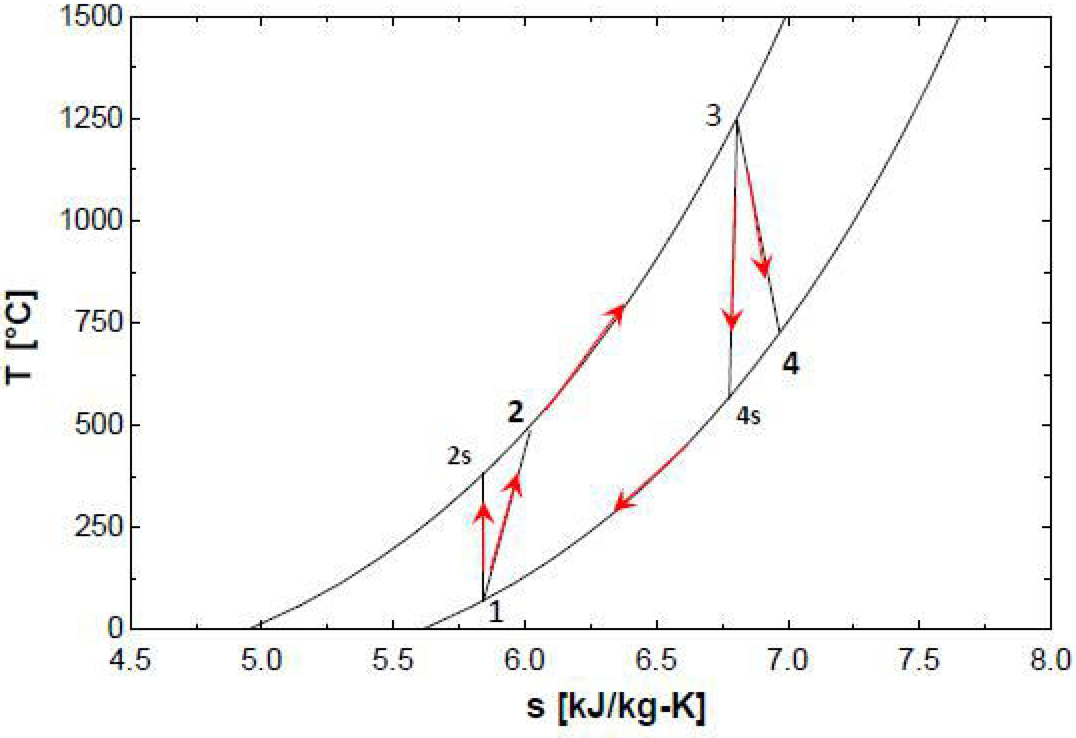
The net power output is given by:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_C = \dot{m}c_p.\,\eta_T(T_3 - T_{4s}) - \frac{\dot{m}c_p(T_{2s} - T_1)}{\eta_c}$$

In terms of the maximum and minimum temperatures and pressure ratios, the net power output is given by:

$$\dot{W}_{net} = \dot{m}c_p.T_1\left[\left(\eta_T\frac{T_3}{T_1} - \frac{r_p^{(\gamma-1)/\gamma}}{\eta_c}\right)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right)\right]$$

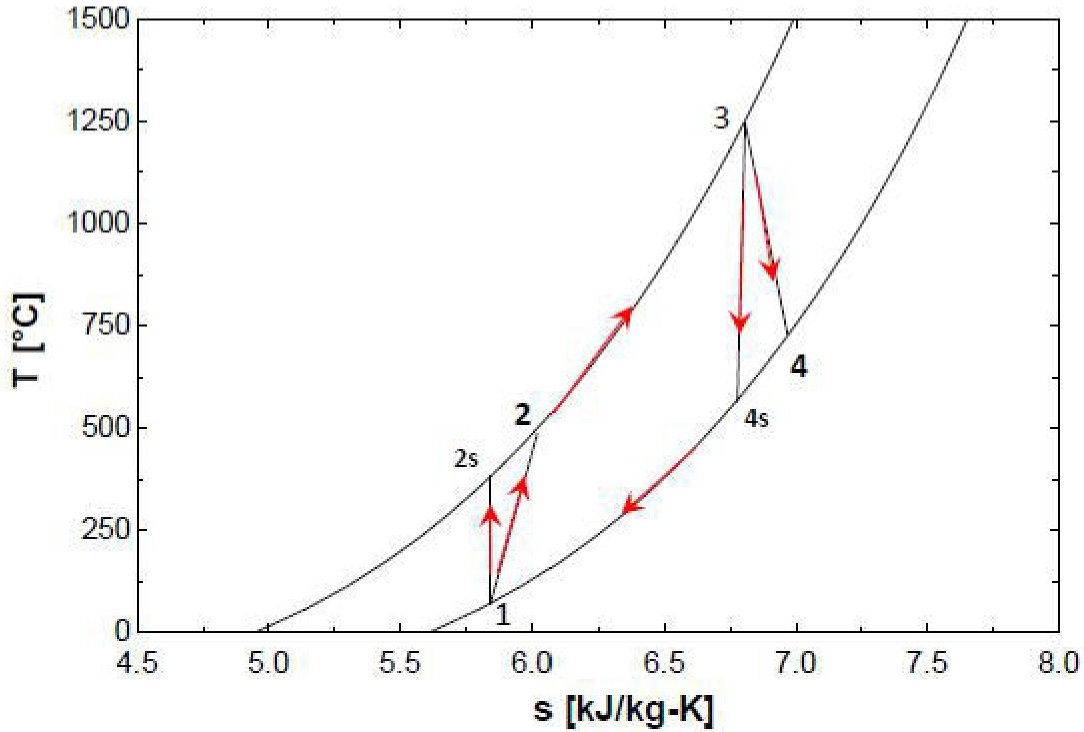
Non-isentropic compression and



The heat input is given by:

$$Q_{in} = \dot{m}c_p\left[(T_3 - T_1) - \left(T_1 \frac{r_p^{(\gamma-1)/\gamma} - 1}{\eta_c}\right)\right]$$

Non-isentropic compression and



The thermal efficiency is given by:

$$\eta_{th,act} = \frac{\dot{W}_{net}}{Q_{in}} = T_1 \frac{\left[\left(\eta_T \frac{T_3}{T_1} - \frac{r_p{}^{(\gamma-1)/\gamma}}{\eta_c}\right)\left(1 - \frac{1}{r_p{}^{(\gamma-1)/\gamma}}\right)\right]}{\left[(T_3 - T_1) - \left(T_1 \frac{r_p{}^{(\gamma-1)/\gamma} - 1}{\eta_c}\right)\right]}$$

Non-isentropic compression and

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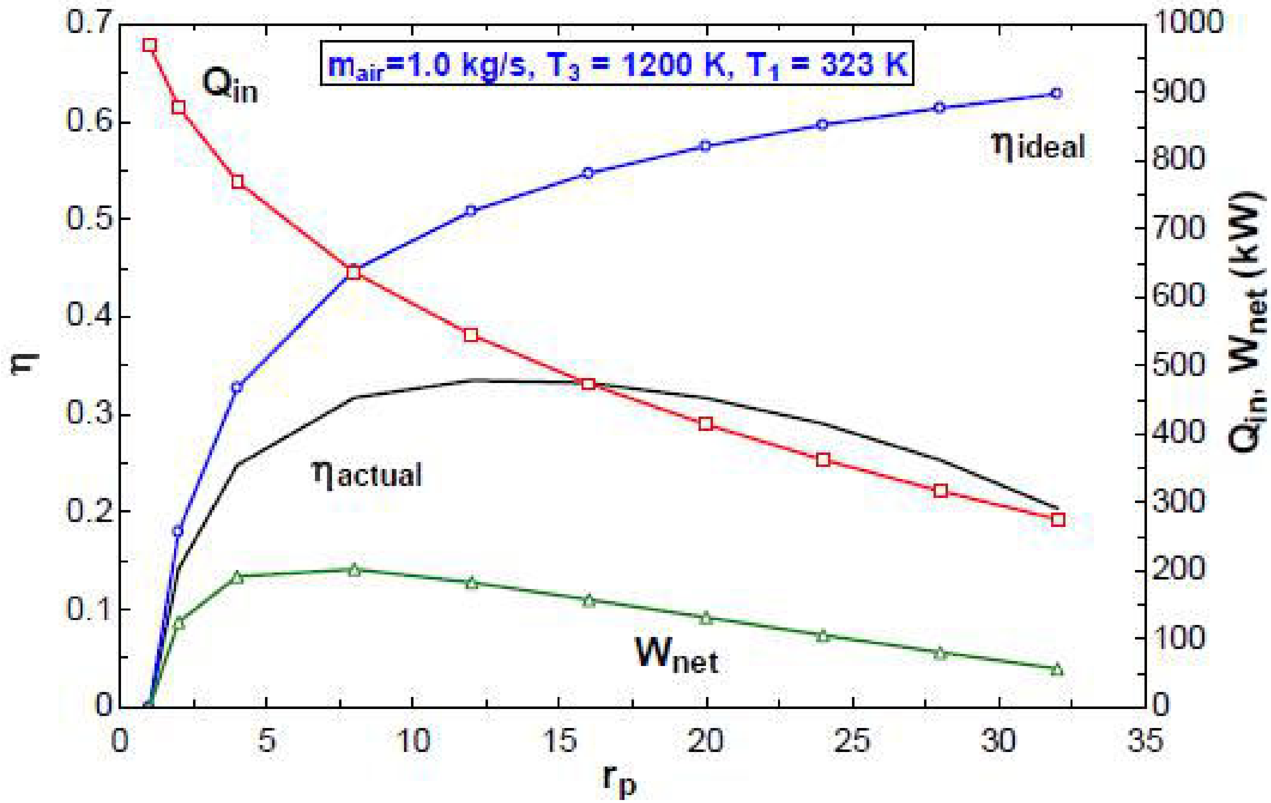


$$\eta_{th,act} = \frac{\dot{W}_{net}}{Q_{in}} = T_1 \frac{\left[\left(\eta_T \frac{T_3}{T_1} - \frac{r_p^{(\gamma-1)/\gamma}}{\eta_c}\right)\left(1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}\right)\right]}{\left[(T_3 - T_1) - \left(T_1 \frac{r_p^{(\gamma-1)/\gamma}-1}{\eta_c}\right)\right]}$$

Evaluation of an actual Brayton cycle

- Given:

a) Mass flow rate of air = 1 kg/s

b) Max. temperature of heat addition = 1200 K

c) Min. temperature of heat rejection = 323 K

d) Isentropic efficiency of turbine = 90 %

e) Isentropic efficiency of compressor = 87 %

f) Pressure ratio = 12

- Find:

a) Temperature at the exit of compressor and turbine **(706.9 K, 651 K)**

b) Turbine power output and compressor power input **(606.7 kW and 424.2 kW)**

c) Thermal efficiency of the cycle **(33.49 %)**

d) Total entropy generation **(0.668 kW/K)**

e) Lost work **(215.8 kW)**

Evaluation of an actual Brayton cycle

Component	Entropy generation (kW/K)	% of total
Compressor	0.0809	12.1
HT HX	0.1307	19.6
Turbine	0.1087	16.3
LT HX	0.3476	52.0
Total	0.6679	100

The above results show that:

a) **Maximum entropy generation** is in low temperature heat exchanger (LT HX) followed by the high temperature heat exchanger (HT HX)

This is **due to the large temperature difference** over which heat transfer takes place in the heat exchangers

b) To improve efficiency entropy generation in HXs should be minimized

Modified Brayton cycle

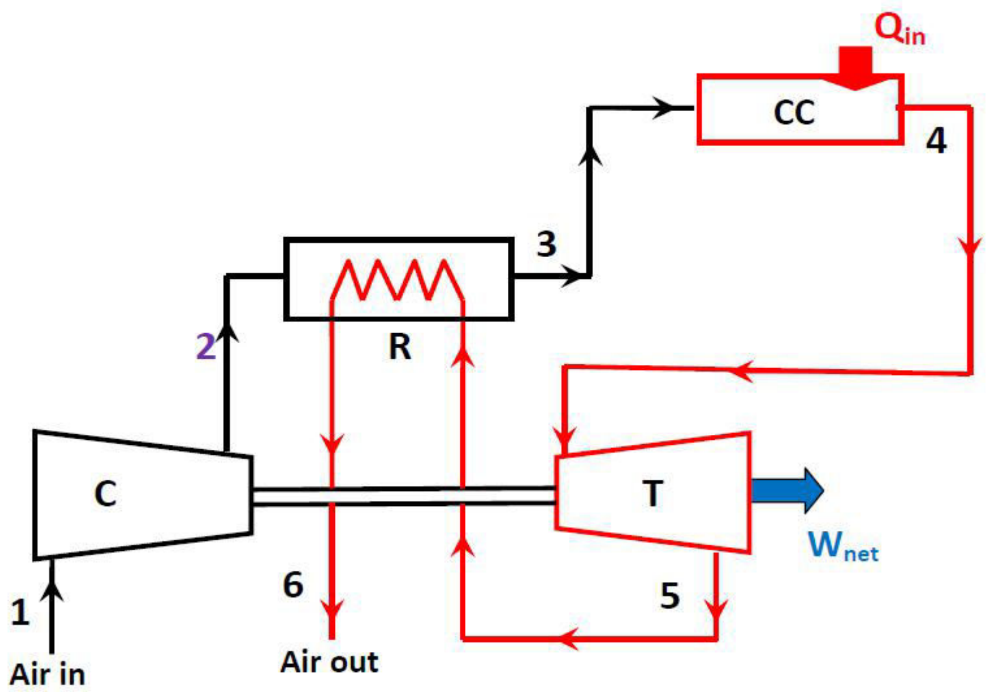
- Performance of Brayton cycle can be improved significantly by:

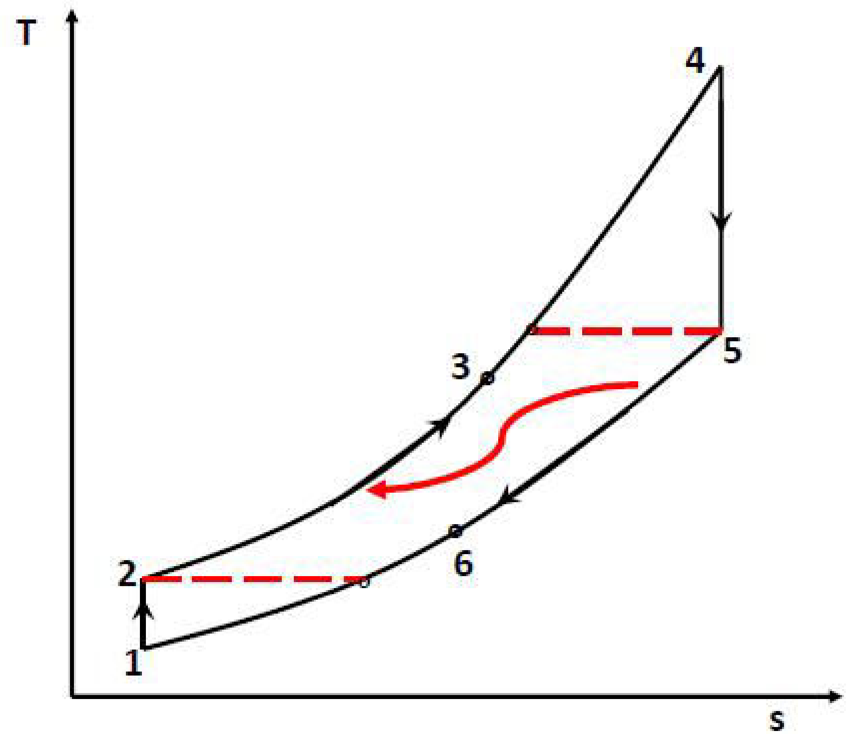
1. **Regeneration**
2. **Intercooling** between compression processes
3. **Reheating** between expansion processes
4. Water injection after compression

Out of the above,

- **Regeneration** is useful for **low to medium pressure ratios**
- **Intercooling** and **reheating** are useful for **high pressure ratios**
- **Water injection** (after compression and before regeneration) improves power output but has a marginal effect on

Regeneration





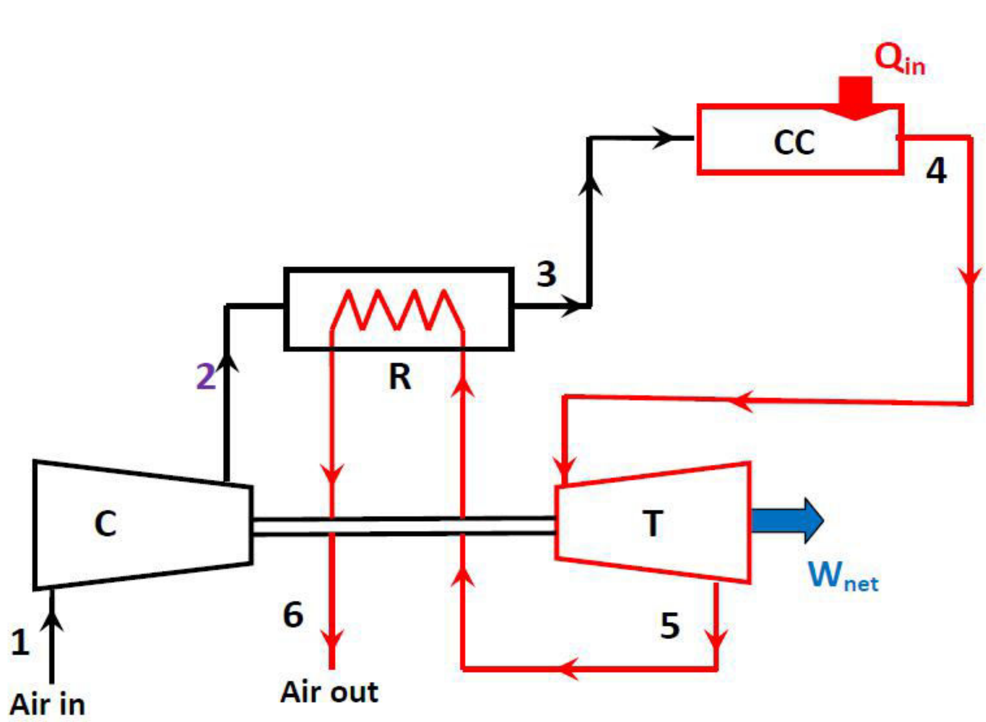
$$Q_{in} = \dot{m}c_p(T_4 - T_3)$$

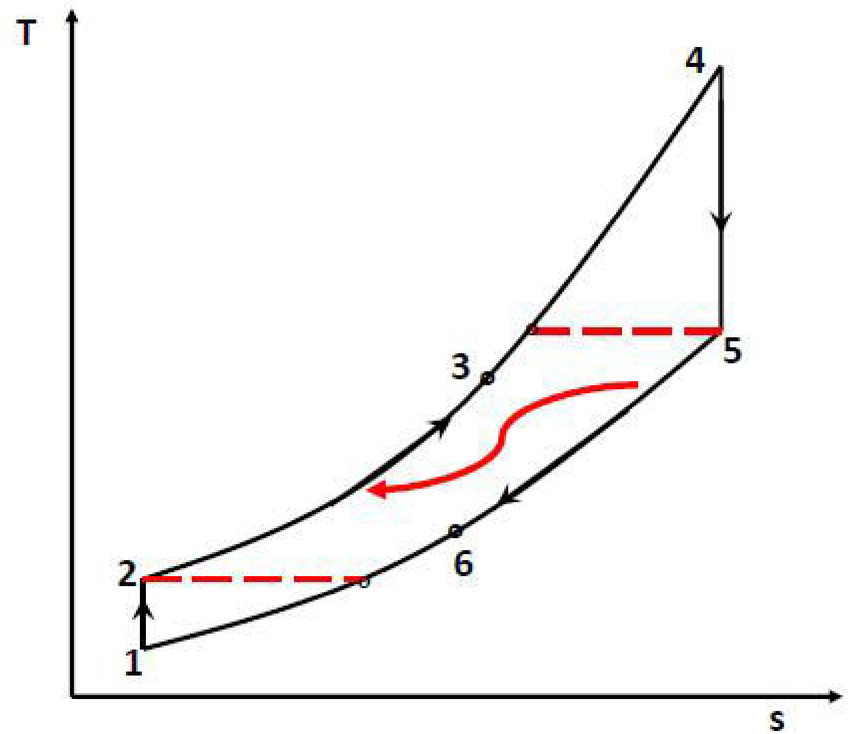
$$Q_{out} = \dot{m}c_p(T_6 - T_1)$$

$$W_{net} = \dot{m}c_p[(T_4 - T_5) - (T_2 - T_1)]$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{[(T_4 - T_5) - (T_2 - T_1)]}{(T_4 - T_3)}$$

Regeneration





Heat transfer rate in regenerator (R):

$$Q_R = \dot{m}c_p(T_5 - T_6) = \dot{m}c_p(T_3 - T_2)$$

Effectiveness of regenerator (R):

$$\varepsilon_R = \frac{(T_5 - T_6)}{(T_5 - T_2)} = \frac{(T_3 - T_2)}{(T_5 - T_2)}$$

Evaluation of Brayton cycle with regeneration

- Given:

a) Mass flow rate of air = 1 kg/s

b) Max. temperature of heat addition = 1200 K

c) Min. temperature of heat rejection = 323 K

d) Isentropic efficiency of turbine = 90 %

e) Isentropic efficiency of compressor = 87 %

f) Pressure ratio = 6

g) Effectiveness of regenerator = 0.90

- Find:

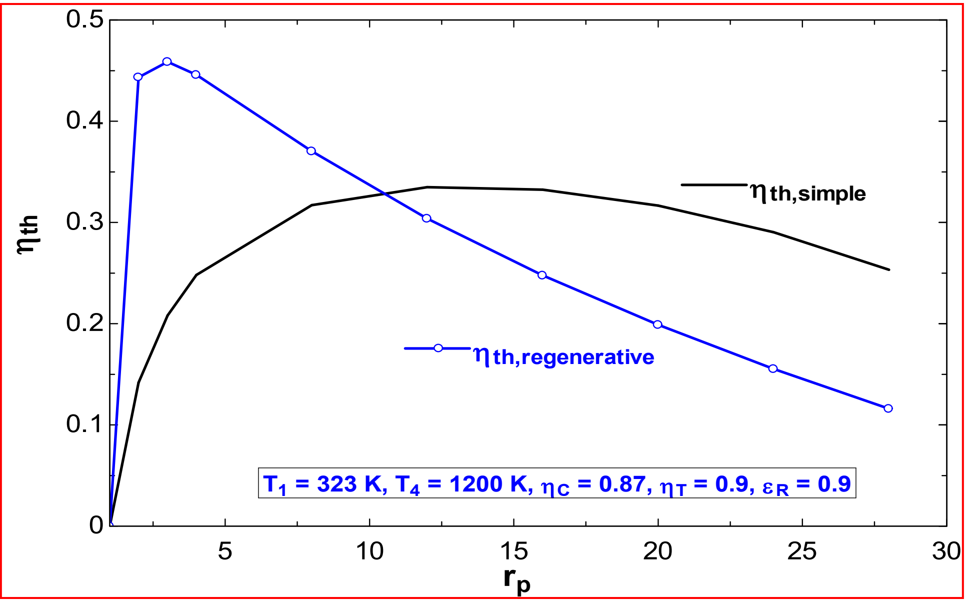
a) Temperature at the exit of compressor and turbine: (571.2 K, 767.3 K)

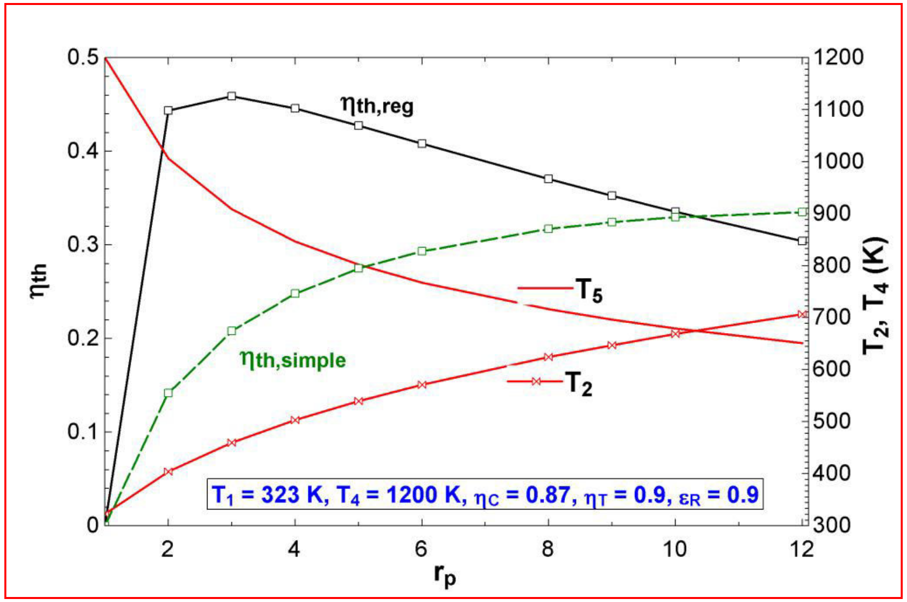
b) Turbine power output and compressor power input: (478.2 kW, 274.3 kW)

c) Thermal efficiency of the cycle: (40.8 %)

d) Heat transfer rate in regenerator: (195 kW)

Comparison between simple and regenerative Brayton cycles



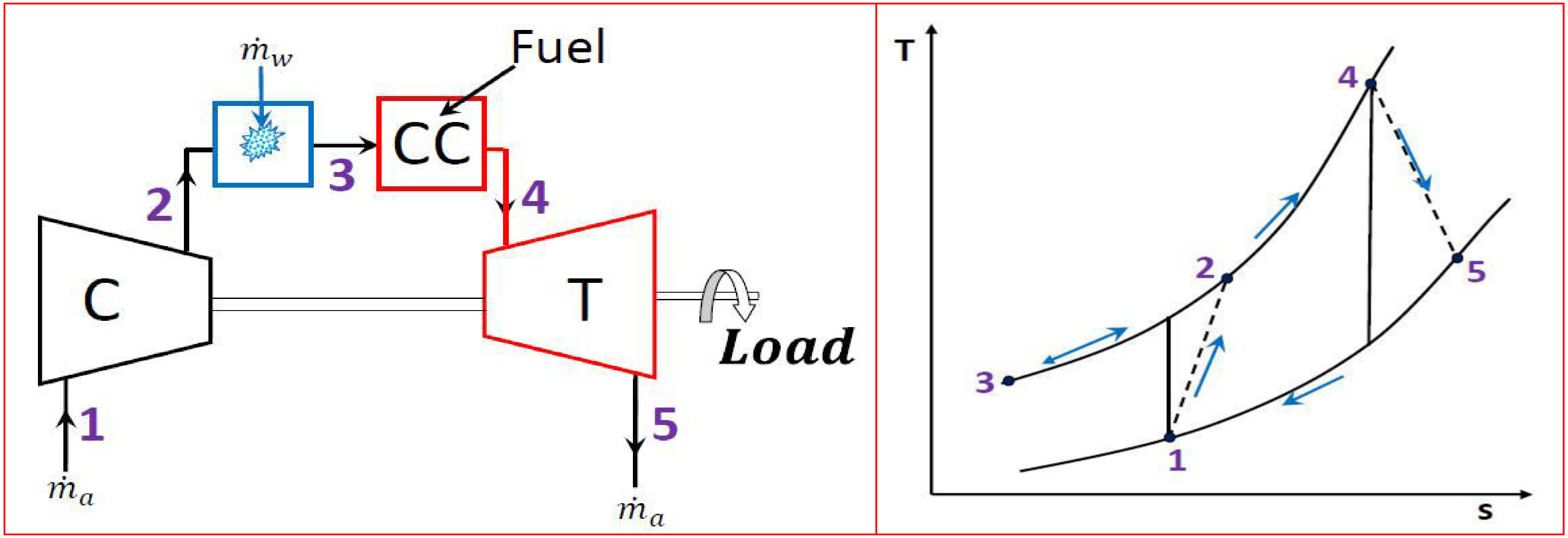


Results show that for a given maximum heat addition and minimum heat rejection temperatures and **non-ideal compressor and turbine**,

a) Regeneration is not possible beyond a certain pressure ratio ($T_2 > T_5$)

b) The **efficiency** of the **regenerative Brayton cycle** reaches a **maximum** at a **particular pressure ratio**, which is much less than that of a simple cycle

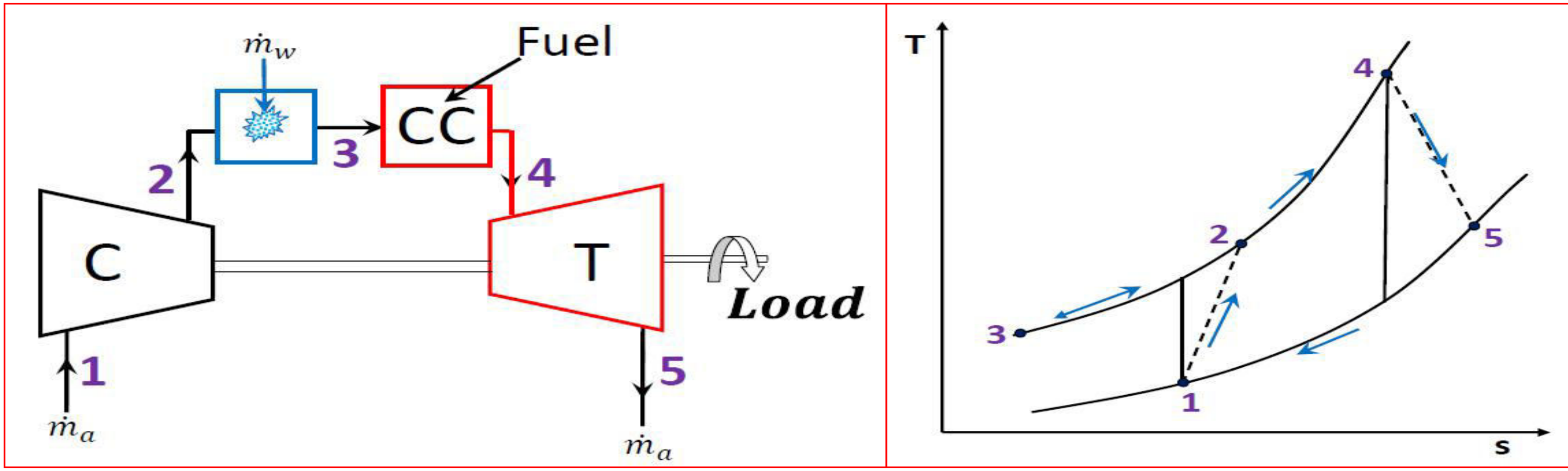
System with water injection



In a simple Brayton cycle with water injection,

1. **Liquid water** is injected into the **air stream after compression**
2. The **injection rat**e should be such that all the liquid water evaporates in the injector and the **moist air** that leaves the injector **(3) does not contain any liquid water**
3. **Due to evaporation of water** (assumed to be adiabatic), **the temperature of moist air decreases** ($T_3 < T_2$)and its **specific volume increases** due to presence of water vapour
4. The **heat input** required in the combustion chamber increases. However, **the net work output also increases** (**main reason for injecting water!**)
5. The thermal efficiency may increase or decrease depending upon the operating conditions

Analysis of a simple Brayton cycle with water injection



Combustion chamber: $Q_{in} = \dot{m}_a(h_4 - h_3)$

Turbine: $W_T = \dot{m}_a(h_4 - h_5)$

Rate of heat rejection: $Q_{out} = \dot{m}_a(h_5 - h_1)$

Compressor: $W_C = \dot{m}_a(h_2 - h_1)$

Energy balance for water injector: $\dot{m}_a h_1 + \dot{m}_w h_w = \dot{m}_a h_3$

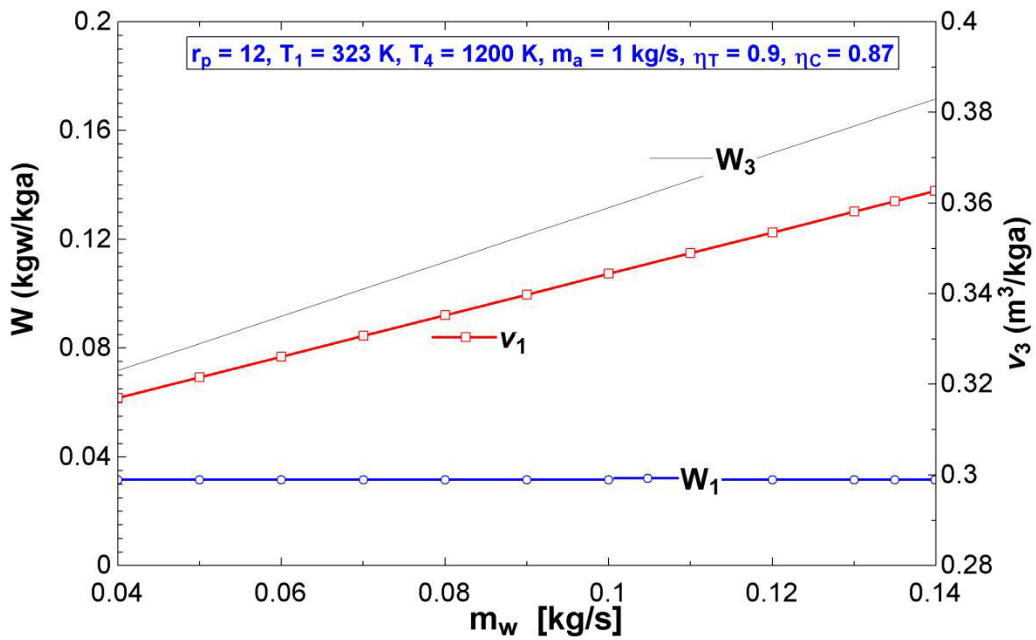
Thermal efficiency: $\eta_{th} = \left(\frac{W_T - W_C}{Q_{in}}\right)$

humidity ratio: $W = 0.622 \frac{p_v}{(p_t - p_v)}$

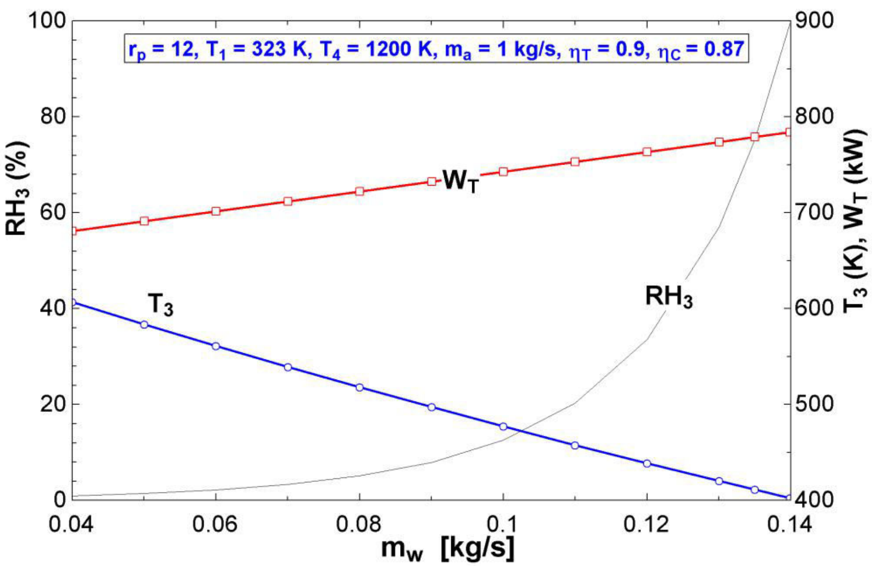
specific volume: $v = \frac{R_a T}{(p_t - p_v)}$

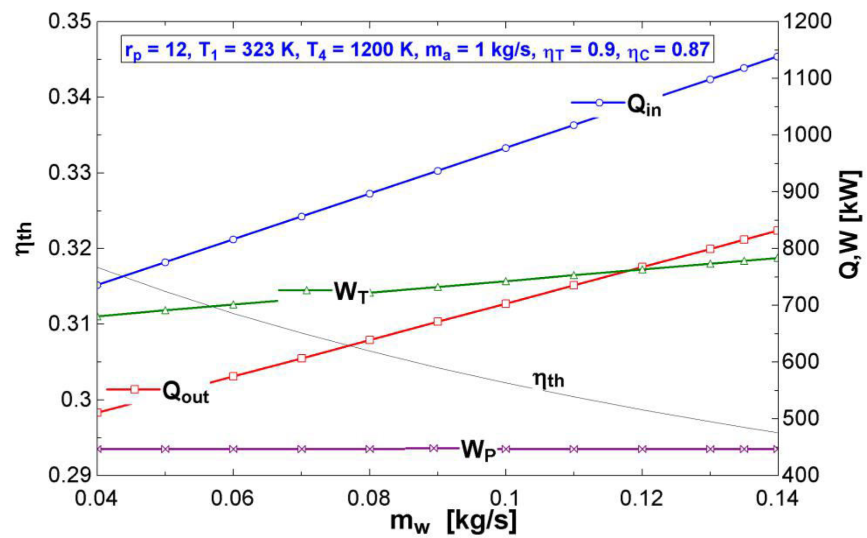
enthalpy of moist air: $h = c_{pa} t + W\left(h_{fg} + c_{pv} t\right)$

Typical results

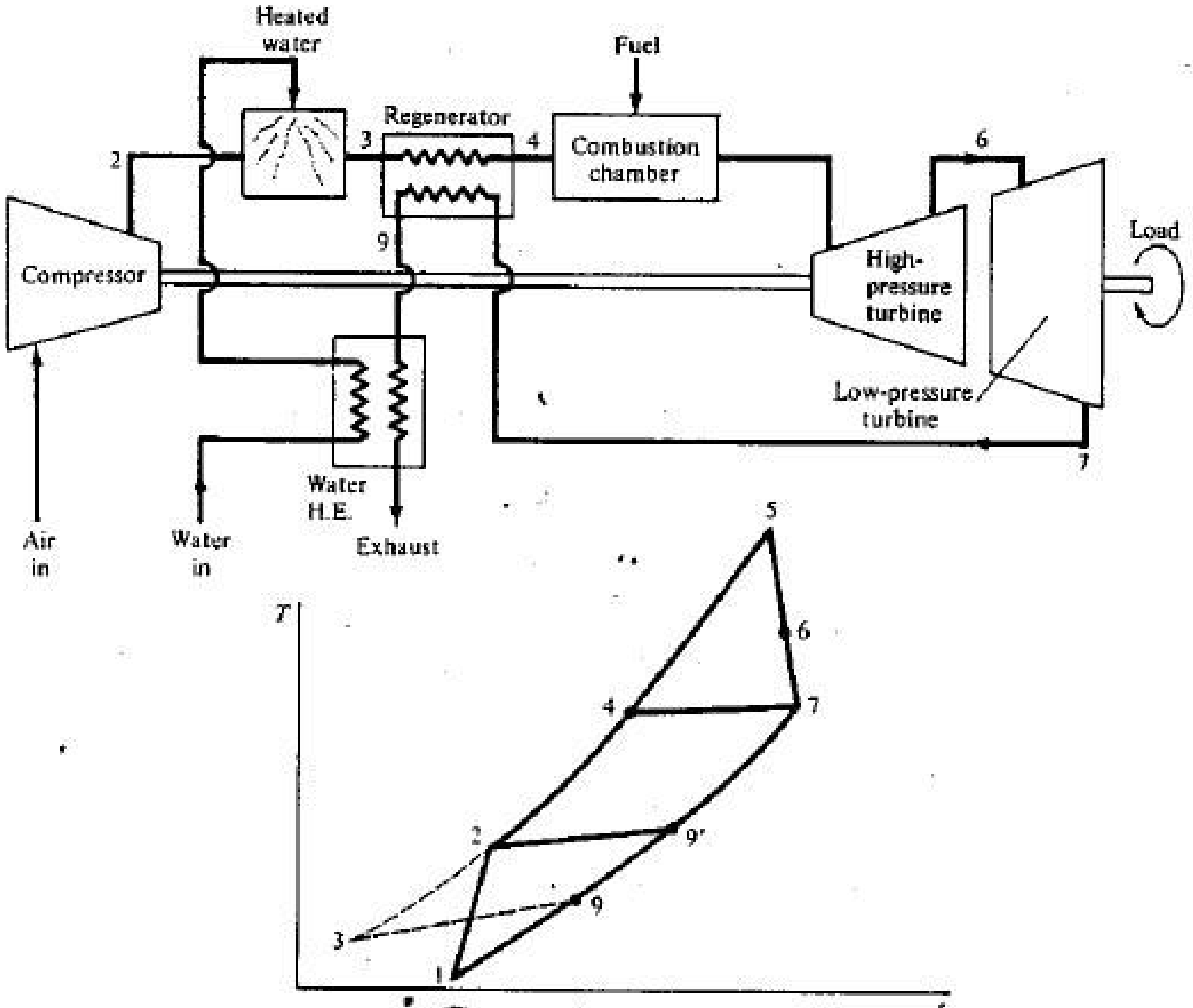


C_p (dry air) = 1.105 kJ/kga.K
C_p (water vapour) = 1.88 kJ/kgw.K
C_p (water) = 4.18 kJ/kgw.K
γ of air = 1.4





Open Brayton cycle with regeneration and water injection



Schematic and T-s diagram of a regenerative cycle with water injection

Open Brayton cycle with regeneration and water injection

- Given:

a) Mass flow rate of air = 1 kg/s

b) Max. temperature of heat addition = 1200 K

c) Min. temperature of heat rejection (compressor inlet) = 323 K

d) Isentropic efficiency of turbine = 90 %

e) Isentropic efficiency of compressor = 87 %

f) Pressure ratio = 6 (Pressure at compressor inlet = 1 atm.)

g) Effectiveness of regenerator = 0.90

h) Rate of water injection = 0.10 kg/s

i) Enthalpy of water = 125 kJ/kgw

j) Relative humidity of air at compressor inlet = 40%

C_p (dry air) = 1.105 kJ/kga.K
C_p (water vapour) = 1.88 kJ/kgw.K
C_p (water) = 4.18 kJ/kgw.K

γ of air = 1.4

- Find:

a) Temperatures at all the state points

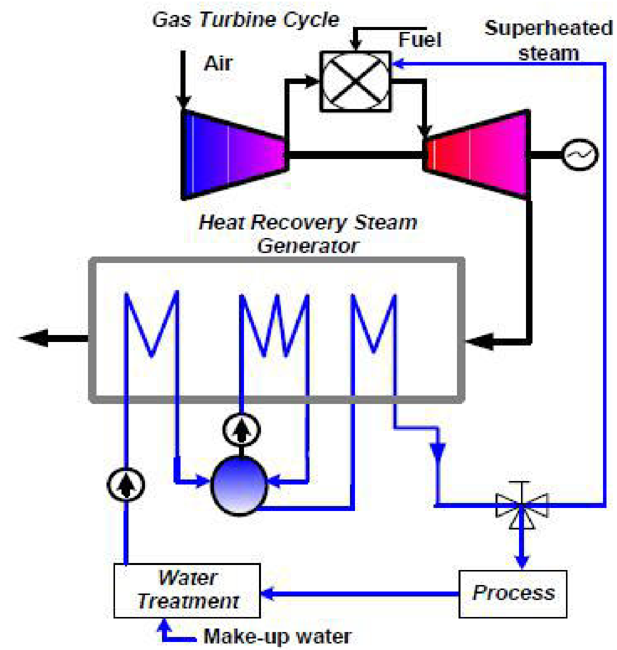
b) Relative humidity at the exit of water injector

c) Turbine power output and compressor power input (in kW) (

d) Thermal efficiency of the cycle (48.9 %)

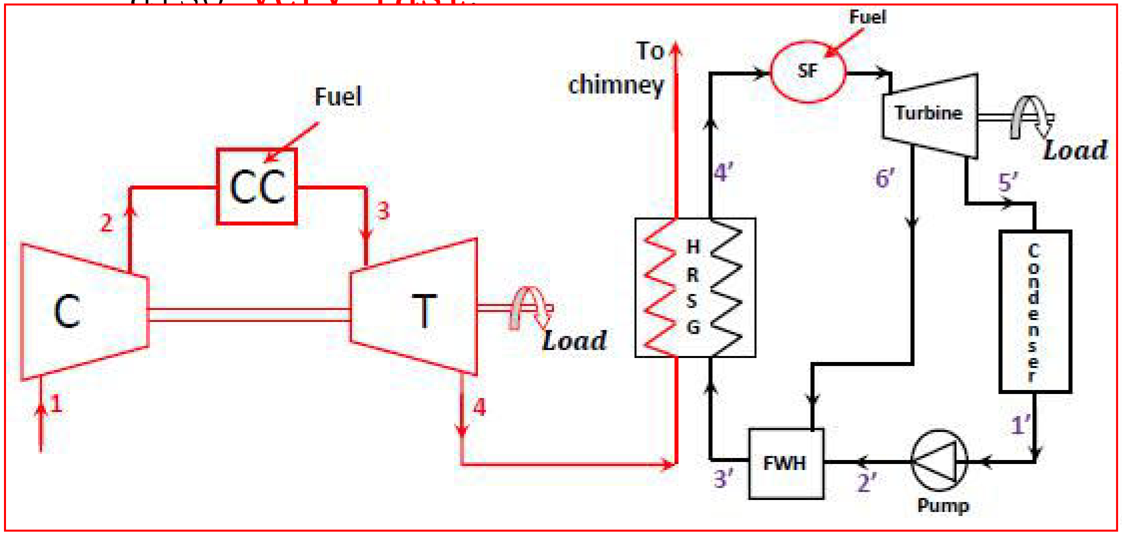
Other modifications to gas turbine cycles

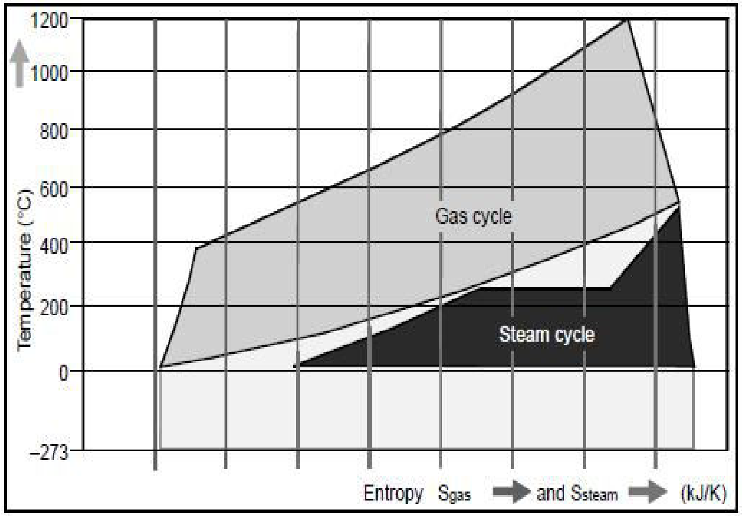
- Evaporative cooling:
  - The **air at the inlet to the compressor** is cooled by making it pass through an **evaporative cooler**
  - Since the **compressor and turbine** in gas turbine plants are typically, constant volume flow components, the lower temperature air at the inlet to the compressor **increases the mass flow rate** and **hence the power output**
- Steam Injected Gas Turbine (STIG):
  - **Injection of steam** into the **compressed air increases** both the **power output** and **efficiency**
  - **Mainly used in cogeneration plants,** wherein the steam injection rate is increased when the requirement for



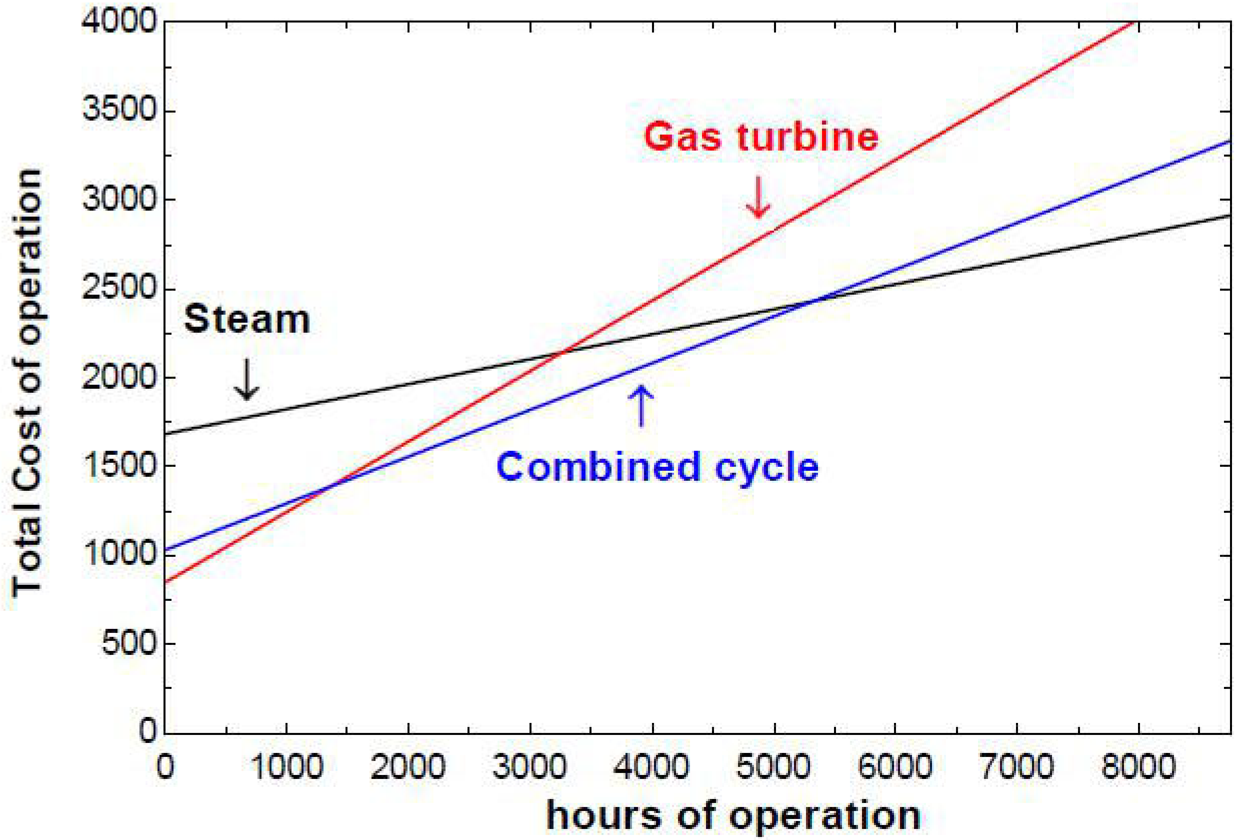
Combined gas-vapour cycles

- The **large amount of energy** available at the **exit of the turbine** in a gas turbine power plant can be **used as heat input** for a **steam power plant**
- Such a system which combines a gas turbine cycle with a steam power plant cycle is called as a **combined cycle power plants**
- **Combined cycle power plants** offer **very high overall efficiency** of the order of **50% or more**, in addition to other **environmental benefits**
- These plants are **simpler** compared to steam power plants due to the **absence of coal handling units, scrubbers** etc. Their **start-up** is also **very fast.**



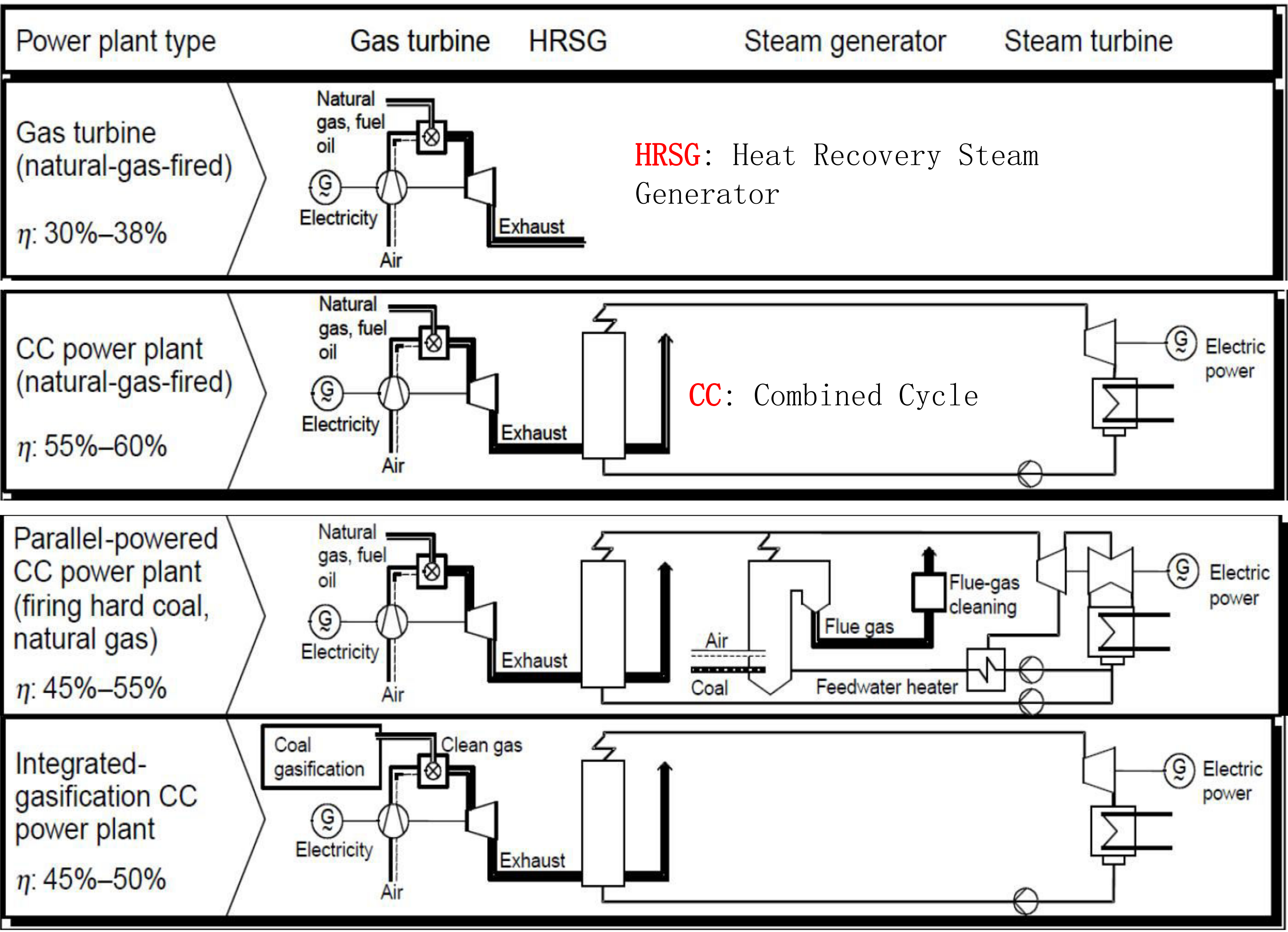


Comparison between steam, gas turbine and combined cycles

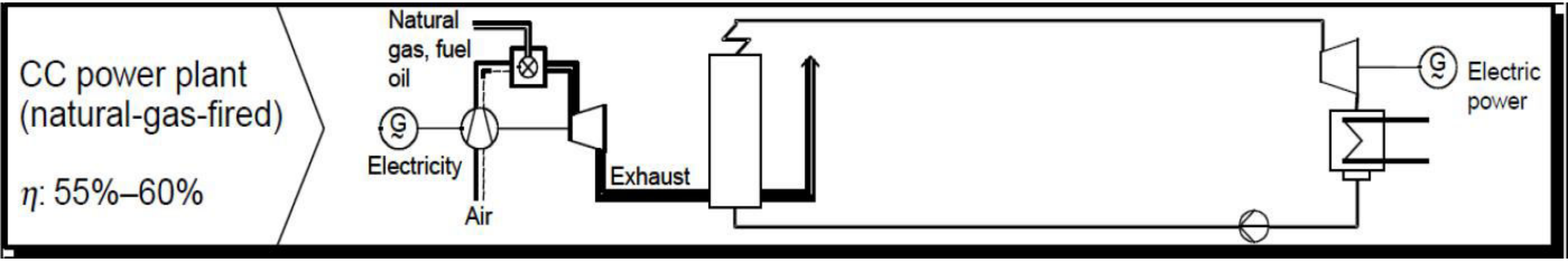


Studies show that from **total cost of operation point of view**:

1. **Gas turbine** plants are **good** for **peak load** operations, while
2. **Steam turbine** plants are good for **base load operation**
3. **Combined power plants** are good a **compromise** between gas turbine and steam power plants

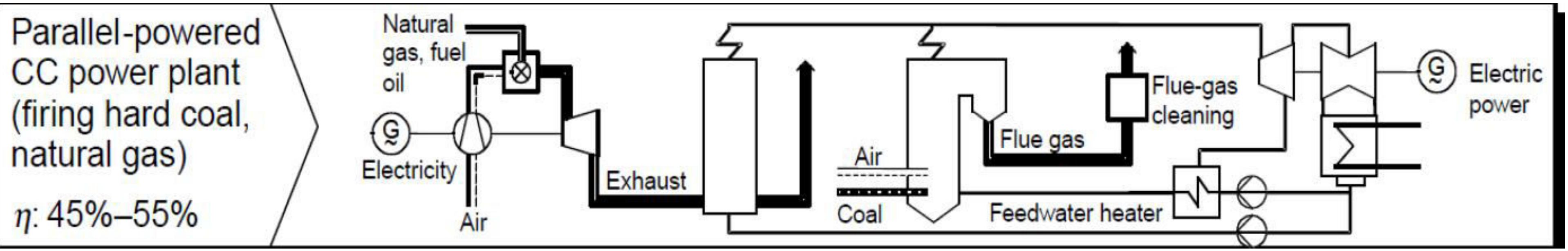


Combined cycle power plant concepts (Alex Lezuo, Siemens, Taylor & Francis, 2007)



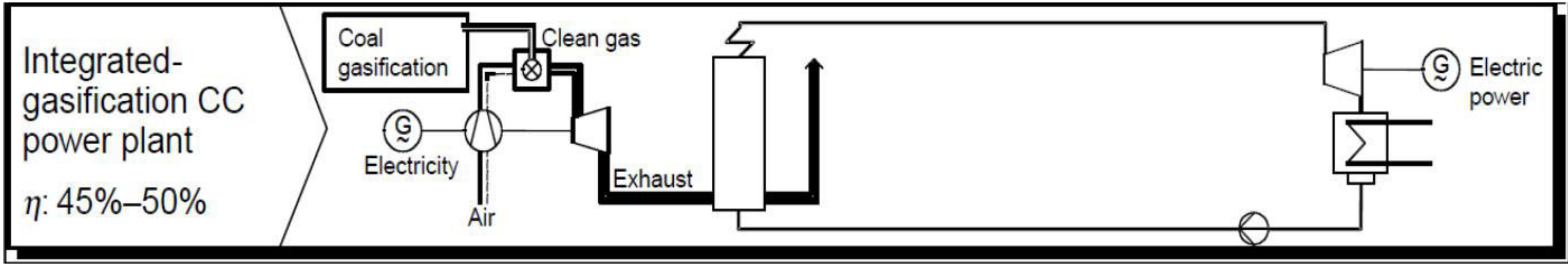
Natural gas fired Combined Cycle (CC) power plant:

1. Highest possible efficiency (+)
2. Simplest and lowest specific investment cost (+)
3. Only natural gas can be used with high efficiency (–)
4. Most commonly used arrangement



Parallel powered CC power plant:

1. Simple system for improving the heat rate of existing coal fired power plants with minimum investment and minimum lead time (+)
2. Offers excellent part-load performance (+)
3. In stead of generating steam, the gas turbine exhaust gases can also be used for heating the feedwater, thus eliminating the bleed stem from steam turbine, thereby improving the output of the steam turbine (+)
4. Higher initial cost (–)



Integrated Gasification Combined Cycle (IGCC) plant:

1. Can be used with fuels other natural gas, e.g. coal (+)
2. Permits use of lower cost fuels such as coal in an environment friendly manner (+)
3. Very complex system and suitability depends upon relative costs of coal and NG (–)

Characteristics of different types of Combined Cycle (CC) plants

1. **Natural gas fired CC power plant:**
   1. Highest possible efficiency (+)
   2. Simplest and lowest specific investment cost (+)
   3. Only natural gas can be used with high efficiency (–)
   4. Most commonly used arrangement
2. **Parallel powered CC power plant:**
   1. Simple system for improving the heat rate of existing coal fired power plants with minimum investment and minimum lead time (+)
   2. Offers excellent part-load performance (+)
   3. In stead of generating steam, the gas turbine exhaust gases can also be used for heating the feedwater, thus eliminating the bleed stem from steam turbine, thereby improving the output of the steam turbine (+)
3. **Integrated Gasification Combined Cycle (IGCC) plant:**
   1. Can be used with fuels other natural gas, e.g. coal (+)
   2. Permits use of lower cost fuels such as coal in an environment friendly manner (+)
   3. Very complex system and suitability depends upon relative costs of coal and NG (–)

Cogeneration

- **Cogeneration** refers to the **simultaneous generation** of electricity and heat or steam (or hot water)
- It has long been **used in industries and by municipalities** that **need both electricity and** steam (say for house heating in winter)
- Cogeneration is **beneficial only** if it **results in saving of primary energy** when **compared** to **separate pro**duction of electricity and steam by two different systems
- The cogeneration plant efficiency is given by:

$$\eta_{co} = \frac{P_E + Q_u}{Q_{in}}$$

- Where:
- η_{co} = cogeneration plant efficiency
- P_E = Electrical power generated (MWh)
- Q_u = Useful heat supplied from the plant (MWh)
- $Q_{in,co}$ = Heat added to the cogeneration plant through the fuel (MWh)

Cogeneration (contd.)

If **electricity and steam are generated individually**, then the amount of **heat to be added per unit total energy** (electrical + heat) output is given by:

$$q_{in,ind} = \frac{P_E}{\eta_E} + \frac{Q_u}{\eta_Q}$$

η_E = Efficiency of electrical power generation of a stand-alone power plant

η_Q = Efficiency of heat generation of the heat/steam generator

If E = Fraction of electrical energy of the total energy output, i.e, $E = \frac{P_E}{P_E + Q_u}$

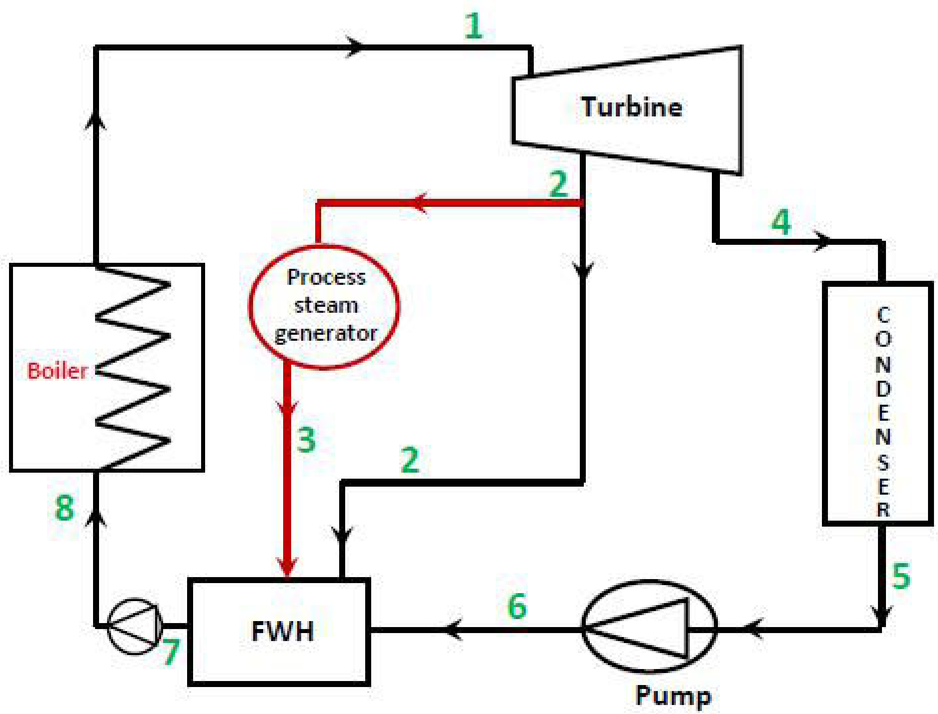
Then, the combined efficiency for individual plants η_{ind} for electricity and heat is given by:

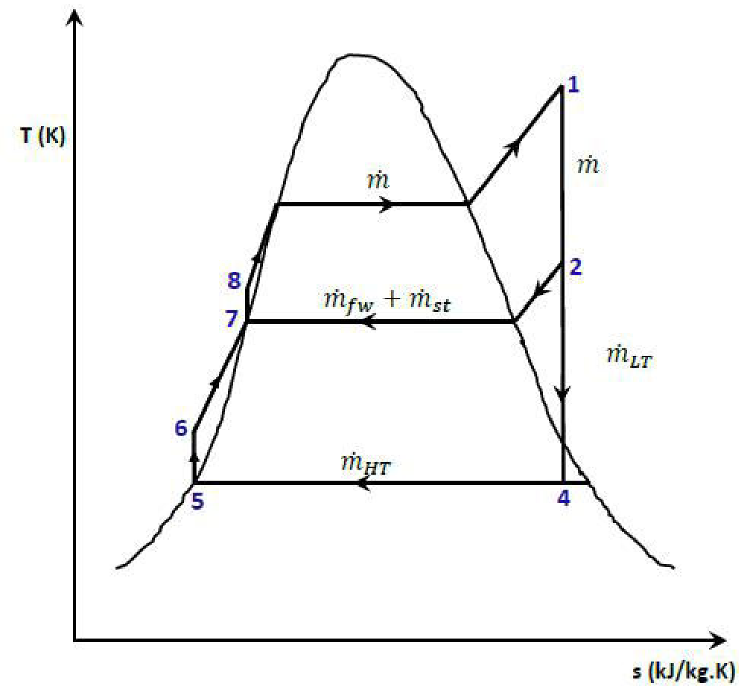
$$\eta_{ind} = \frac{1}{\left(E/\eta_E\right) + \left[\left((1-E)/\eta_Q\right)\right]}$$

Hence, cogeneration is beneficial if:

$$\eta_{co} > \eta_{ind}$$

Worked out example: Cogeneration plant with extraction-condensation turbine





Given: Net power output: **200 MW** Steam for process steam generator: **75 kg/s**

Boiler inlet conditions: **165 bar and 400°C** Boiler efficiency: **90 %**

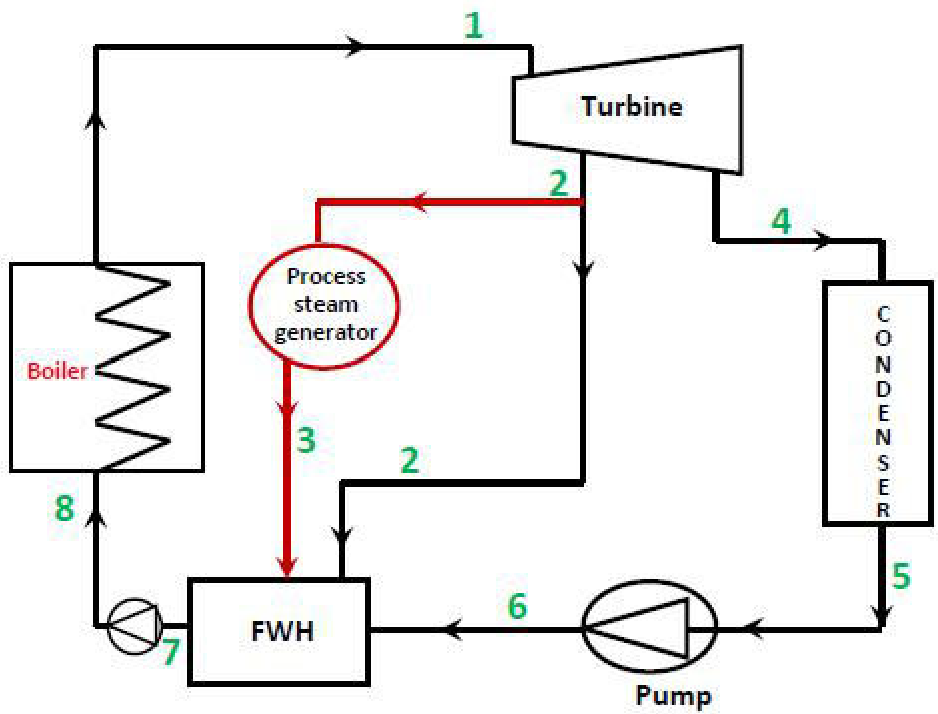
Condenser pressure: **0.07 bar** Calorific value of fuel: **20000 kJ/kg**

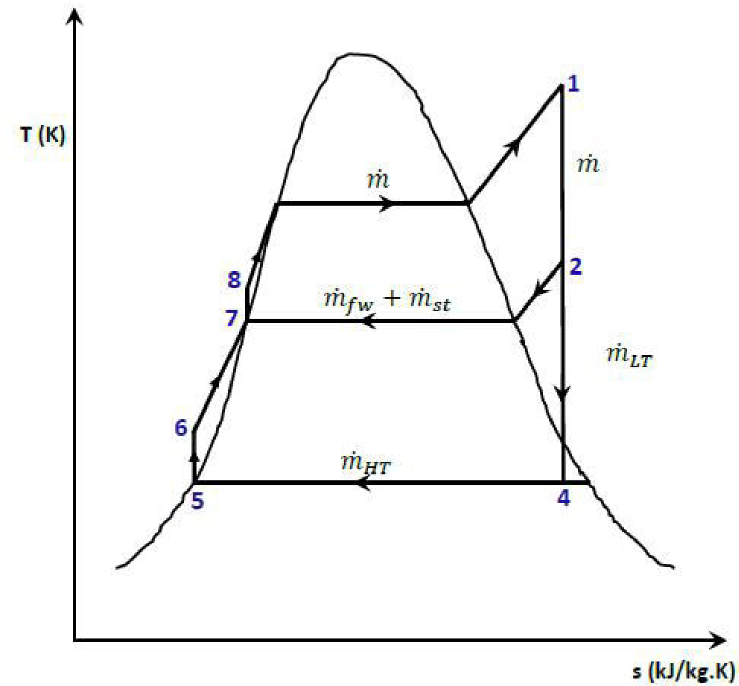
Assume **saturated conditions** at the **exit of condenser**, **process steam generator** and **feed water heater (FWH)**

Find the **fuel consumption rate (kg/day)** for the cogeneration system and compare this with system that uses individual power plant and process heat generator.

Ans.: a) 28.22 kg/s; b) 31.45 kg/s

Worked out example: Cogeneration plant with extraction-condensation turbine

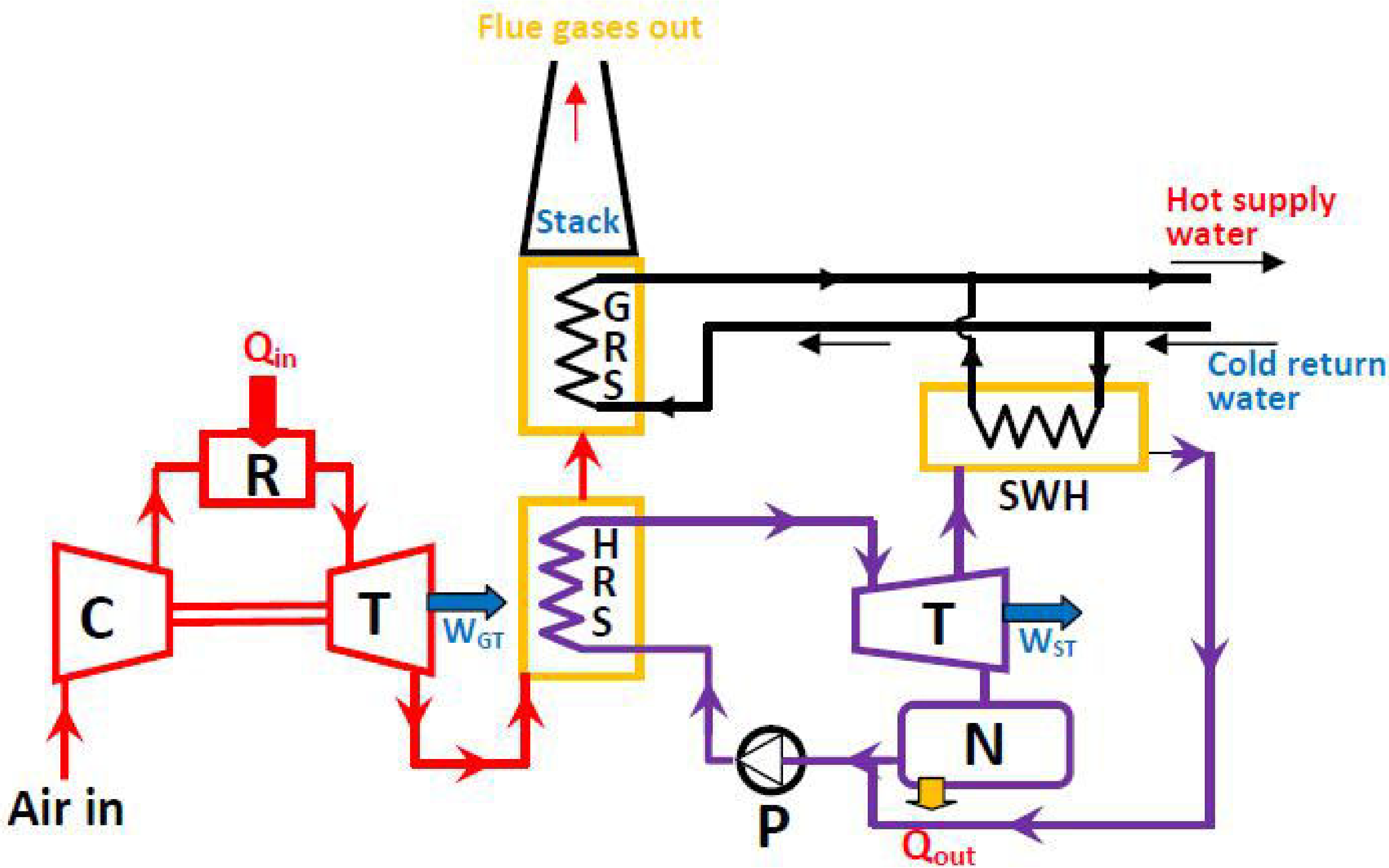




Property data:

Pressure, bar	Temperature, °C	Quality	Specific enthalpy, kJ/kg	Specific entropy, kJ/kg.K
0.07	39.01	0	163.4	0.559
0.07	39.01	1	2572	8.274
3	133.6	0	561.6	1.672
3	133.6	1	2725	6.992
165	465	Superheated	3180	6.136

Practical example of a large combined cycle CHP Plant
(Operating in The Hague, Netherlands)



C: Compressor; **N:** Condenser; **P:** Pump; **R:** Combustor; **T:** Turbine
GRS: Gas water Heater; **HRS:** Heat Recovery Steam Generator; **SWH:** Steam water heater

Assignment on the large combined cycle CHP Plant

(Data from the plant operating in The Hague, Netherlands

- Given Data:
- Steam turbine:
  - Power output (net): 25 MW;
  - Steam supply to turbine: 30 bar, 450°C,
  - pressure of bleed steam to SWH: 2 bar,
  - mass flow rate of bleed steam to SWH: 17 kg/s,
  - Isentropic eff.: Turbine: 80%, Condenser sat. temp: 25°C,
  - no subcooling in condenser
  - Electric generators: 95 % efficiency
  - District heating: Supply temp: 115°C, return temperature: 75°C
- Gas Turbines (2 in number):
  - Power output (net): 25 MW each;
  - Pressure ratio: 12;
  - Maximum cycle temperature: 1013°C
  - Exhaust gas temperature: 83°C;
  - Isentropic eff.: Turbine: 85%, Compressor: 83%, Combustion Eff.: 0.98
  - Cp of gas = 1.11 kJ/kg.K, γ = 1.333

Assignment on the large combined cycle CHP Plant

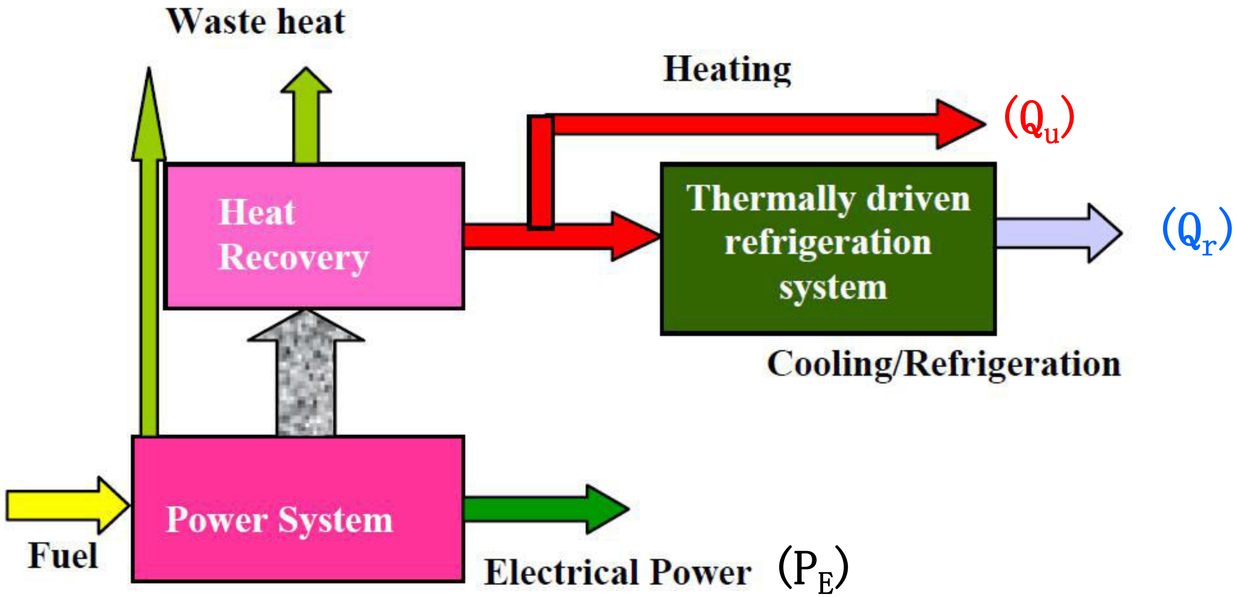
(Data from the plant operating in The Hague, Netherlands

- **To find:**

1. Overall efficiency of the system: (ans. **68.9%)**
2. Mass flow rate of water for district water heating: (ans. **331.5 kg/s**)
3. Air flow through each gas turbine: (ans. **131 kg/s**)
4. Steam flow rate (total): (ans. **35.39 kg/s**)
5. Heating output (water heating): (ans. **55.69 MW (total)**)

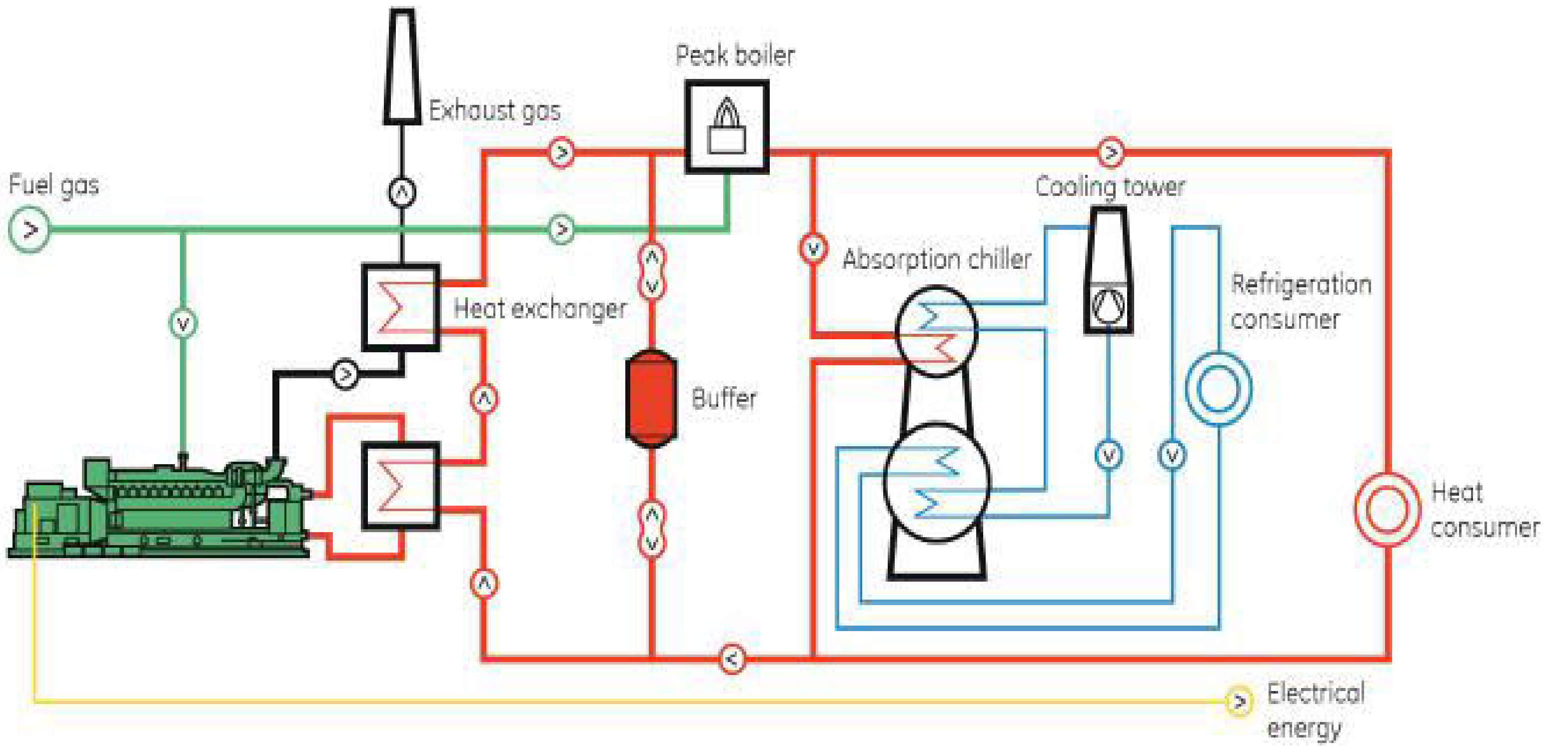
Tri-generation

- Tri-generation is the production of electricity, heat and cooling in a single power plant (**steam or gas turbine**)
- Steam from the boiler or gas from a gas turbine is used for:
  - Production of electricity in the steam turbine-generator (P_E)
  - Production of heat or process steam/hot water for heating purposes (Q_u)
  - Production of refrigeration/air conditioning using an absorption chiller (Q_r)
- The ratio of electricity (P_E), heat (Q_u) and refrigeration (Q_r) can vary depending upon the requirements



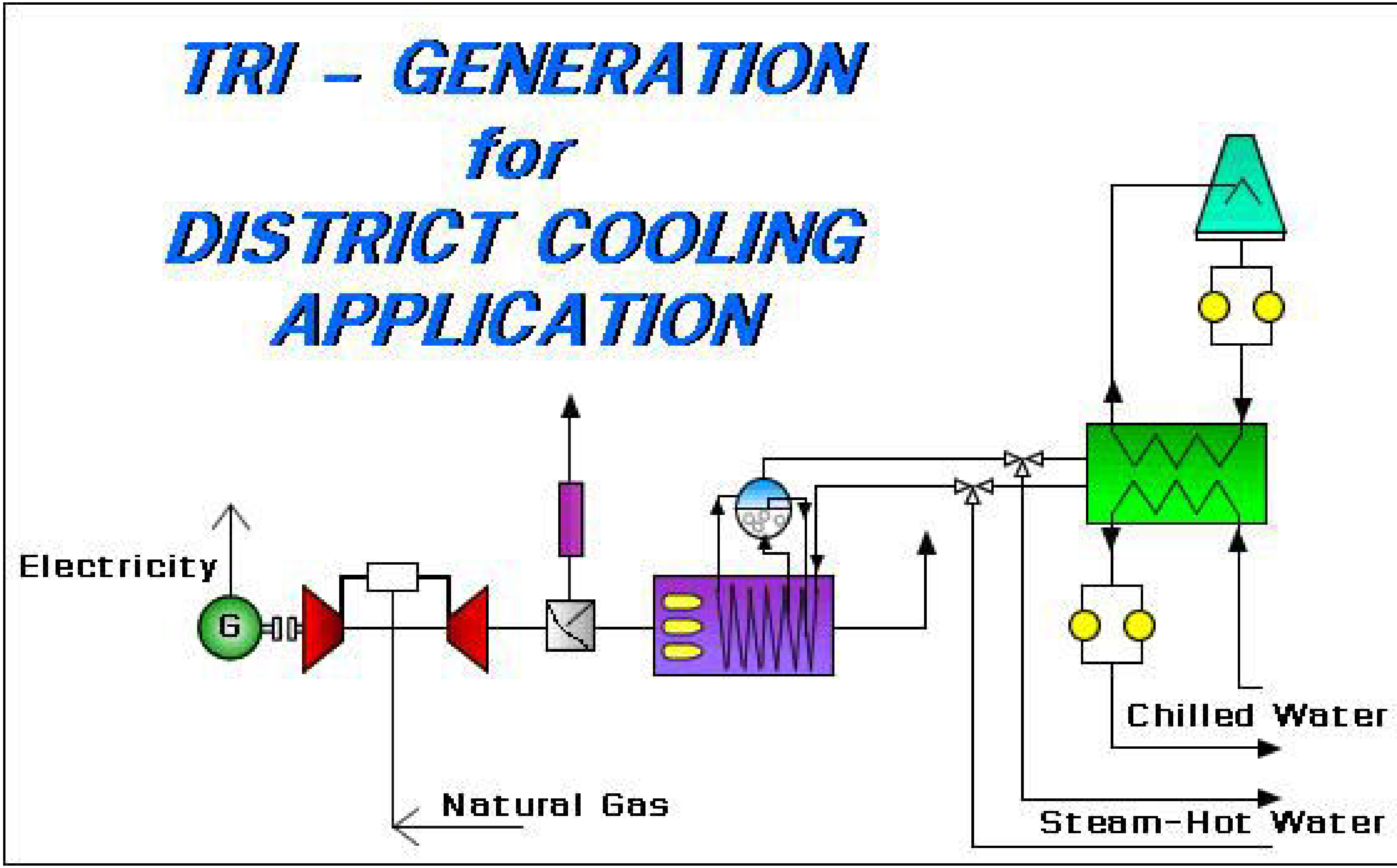
Concept of tri-generation

Tri-generation



A typical, gas turbine based tri-generation plant (Dusan Medved, 2011)

Tri-generation



End of Module-1