SKMM 2423 Applied Thermodynamics

Chapter 2 Gas turbine

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The Use of Gas Turbine

Gas turbine plants are widely used in the following engineering fields:

1. Aircraft propulsion system
2. Electric power generation
3. Marine vehicle propulsion
4. Combined-cycle power plant (with steam power plant)

Note: The processes taking place in an actual gas turbine plant are complicated. To carry out thermodynamics study on the system, we will develop a simplified model of the system.
Gas Turbine Application

GE
Rolls Royce (RR)
Pratt and Whitney
Gas Turbine Application
Gas Turbine Application
GAS TURBINE CYCLES
Main Components of Gas Turbine Cycle

1. Compressor
   - The compressor sucks in air from 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.
Main Components of Gas Turbine Cycle

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 °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 °C.
Types of Gas Turbine Cycles

There are two types of gas turbine cycle; Brayton/Joule cycle and Atkinson cycle

➔ Brayton Cycle

- Heat added and rejected is at constant pressure

➔ Atkinson Cycle

- Heat added at constant volume, therefore needs valve to control gas flow to ascertain constant volume
- Heat rejected at constant pressure
- Not popular for gas turbine study
- Out of the SKMM2423 scope
Brayton Cycle: Ideal Cycle for Gas Turbine Cycle

- Gas turbines usually operate on an **open cycle**.
- 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 high-temperature gases then enter the turbine where they expand to atmospheric pressure while producing power output.
- Some of the output power is used to drive the compressor.
- The exhaust gases leaving the turbine are thrown out (not re-circulated), causing the cycle to be classified as an **open cycle**.

An open-cycle gas-turbine engine.
Brayton Cycle - Closed Cycle Model

- The open gas-turbine cycle can be modelled as a closed cycle, using the **air-standard** assumptions.

- 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.

- 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 the closed loop is the *Brayton cycle*. It is made up of four internally reversible processes:

- **1-2** Isentropic compression;
- **2-3** Constant-pressure heat addition;
- **3-4** Isentropic expansion;
- **4-1** Constant-pressure heat rejection.

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

**Note**: All four processes of the Brayton cycle are executed in steady-flow devices thus, they should be analyzed as **steady-flow processes**.
~ Review ~

➔ Steady Flow Energy Equation

\[ \dot{Q} - \dot{W} = \sum_{out} \dot{m}(h + ke + pe) - \sum_{in} \dot{m}(h + ke + pe) \]

➔ Relationship of P, v, T between two states under polytropic process for ideal gases

\[ \frac{T_2}{T_1} = \left( \frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = \left( \frac{v_1}{v_2} \right)^{n-1} \]

➔ For an isentropic process

\[ n = k \]

➔ Specific Heat Ratio

\[ k = \frac{C_p}{C_v} \quad \quad C_p - C_v = R \]
Brayton / Closed Cycle

Cycle analysis – Energy & Thermal efficiency

By using steady flow energy equation, this closed ideal cycle can be analysed as follows.

\[ \begin{align*}
q_{in} &= q_{23} = h_3 - h_2 = c_p (T_3 - T_2) \\
q_{out} &= q_{41} = h_4 - h_1 = c_p (T_4 - T_1) \\
w_{tur} &= w_{34} = h_3 - h_4 = c_p (T_3 - T_4) \\
w_{com} &= w_{12} = h_2 - h_1 = c_p (T_2 - T_1)
\end{align*} \]
Brayton / Closed Cycle

Cycle analysis – Energy & Thermal efficiency

First law of thermodynamic states, net heat received by any cyclic device is the same with the net work produced.

\[ w_{net} = q_{in} - q_{out} = w_{tur} - w_{comp} \]

\[ \eta_{th} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{\phi_p (T_4 - T_1)}{\phi_p (T_3 - T_2)} \]

\[ \eta_{th} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)} \]

**Note:** Only valid for ideal closed cycle.
But, for isentropic processes,

\[ \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} \quad \text{and} \quad \frac{T_3}{T_4} = \left( \frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}} \]

So,

\[ T_2 = T_1 \cdot r_p^{\frac{\gamma-1}{\gamma}} \quad \text{and} \quad T_3 = T_4 \cdot r_p^{\frac{\gamma-1}{\gamma}} \]

Where,

\( r_p = \) pressure ratio

and

\( \gamma = \frac{c_p}{c_v} = \) specific heat ratio (air = 1.4)
Therefore,
\[
\frac{T_2}{T_1} = \frac{T_3}{T_4} = r_p \gamma
\]

So,
\[
\frac{T_4 - T_1}{T_3 - T_2} = \frac{1}{r_p \gamma}
\]
or
\[
(T_3 - T_2) = (T_4 - T_1) r_p \gamma
\]

Re-express thermal efficiency,
\[
\eta_{th} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)} = 1 - \frac{1}{r_p \gamma}
\]
Brayton / Closed Cycle

Cycle analysis – Work ratio, $w_r$

$$w_r = \frac{w_{net}}{w_{turbine}} = \frac{w_{34} - w_{12}}{w_{34}} = \frac{c_p (T_3 - T_4) - c_p (T_2 - T_1)}{c_p (T_3 - T_4)}$$

$$w_r = 1 - \frac{(T_2 - T_1)}{(T_3 - T_4)}$$

or, using $T \& r_p$ relation

$$T_2 = T_1 \cdot r_p^{\frac{\gamma - 1}{\gamma}}$$
$$T_4 = \frac{T_3}{r_p^{\frac{1}{\gamma}}}$$

$$w_r = 1 - \frac{T_1}{T_3} \cdot r_p^{\frac{\gamma - 1}{\gamma}}$$

(a) $T$-$s$ diagram
Modifications of Brayton Gas Turbine Cycles

Brayton/Joule Cycle (Closed & Ideal)

Open cycle

Basic

Two stage expansion

Two stage compression with intercooler

Two stage compression + Two stage expansion + Reheating

Two stage compression + Two stage expansion + Reheating + Heat exchanger
Modifications of Brayton Gas Turbine Cycles

Brayton/Joule Cycle (Closed & Ideal)

Open cycle

Basic

Two stage expansion

Two stage compression with intercooler

Two stage compression + Two stage expansion + Reheating + Heat exchanger
Basic Gas Turbine – Open Cycle

Process Description:

1-2 Compression process – Atmospheric air at pressure P1 and temperature T1 is induced and compressed adiabatically to higher pressure P2 and Temperature T2. The process is not reversible, thus not isentropic.

2-3 Combustion process – Fuel is injected into the air stream. Mixture of air and fuel is burned at constant pressure inside a combustion chamber, thus producing hot gases.
Basic Gas Turbine – Open Cycle

Process Description:

3-4 Expansion process – Hot combustion gases expands through the turbine. The process is assumed adiabatic but not reversible, thus not isentropic. Mechanical work produced by the turbine. Part of this work used to drive compressor.

*The air exiting the turbine is exhausted to atmosphere. New fresh air is induced into the compressor and the processes are repeated.
Energy analysis

*using steady flow energy equation

Neglecting the change in the kinetic and potential energy of the working fluid, we have, work to drive the compressor:

\[ w_C = c_{pa} \cdot (T_2 - T_1) \]

Heat added to the compressed air

\[ q_{CC} = c_{pg} \cdot (T_3 - T_2) \]

Work produced by the turbine

\[ w_T = c_{pg} \cdot (T_3 - T_4) \]

**Note:**
The turbine is connected to the compressor via common shaft. Thus, part of the work produced by the turbine is used to drive the compressor.
Properties of working fluid

The air and combustion gases are assumed to have the following properties

For air: $c_{pa} = 1.005 \text{ kJ/kg.K}$, $\gamma_a = 1.4$

For gas: $c_{pg} = 1.110 \text{ kJ/kg.K}$, $\gamma_g = 1.33$
Isentropic efficiency

The compression and expansion processes are not isentropic. The isentropic efficiency of the compressor and the turbine is defined as follows:

**Compressor:**

\[ \eta_{is,C} = \frac{(T_{2s} - T_1)}{(T_2' - T_1)} \]

**Turbine:**

\[ \eta_{is,T} = \frac{(T_3 - T_4')}{(T_3 - T_{4s})} \]
Isentropic process

The isentropic process between path 1-2 is

\[ T_{2s} = T_1 \left( \frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = T_1 r_p^{\frac{\gamma-1}{\gamma}} \]

And the isentropic process between path 3-4 is

\[ T_{4s} = T_3 \left( \frac{p_4}{p_3} \right)^{\frac{\gamma-1}{\gamma}} = T_3 \left( \frac{1}{r_p} \right)^{\frac{\gamma-1}{\gamma}} \]

where the values of \( \gamma \) for air and gas are different.
Basic Gas Turbine – Open Cycle

Cycle performance

The performance of the basic gas turbine cycle is measured by the following criteria

1) Thermal efficiency

Thermal efficiency of the basic gas turbine cycle is

\[ \eta_{th} = \frac{W_{net}}{q_{in}} = \frac{W_T - W_C}{q_{CC}} \]

\[ \eta_{th} = \frac{c_{pg} (T_3 - T_{4'}) - c_{pa} (T_{2'} - T_1)}{c_{pg} (T_3 - T_{2'})} \]
2) **Work ratio**

The work ratio of the plant is

\[ w_r = \frac{W_{net}}{W_T} = \frac{w_T - w_C}{w_T} = 1 - \frac{w_C}{w_T} \]

\[ w_r = 1 - \frac{c_{pa} (T_2' - T_1)}{c_{pg} (T_3 - T_4')} \]

**Assumptions**

The following assumptions are made on the simplified model of the gas turbine plant:

1. The mass of fuel injected into the air is ignored since the air-fuel ratio is usually large.

   \[ m_g = m_a + m_f \]

2. The mass flow rate of the working fluid is considered constant.
Example 9.1

A gas turbine unit has a pressure ratio of 10/1 and a maximum cycle temperature of 700°C. The isentropic efficiencies of the compressor and turbine are 82% and 85% respectively. Calculate the power output of an electric generator geared to the turbine when the air enters the compressor at 15°C at the rate of 15 kg/s. Take \( c_p = 1.005 \) kJ/kgK and \( \gamma = 1.4 \) for the compression process, and take \( c_p = 1.11 \) kJ/kgK and \( \gamma = 1.333 \) for the expansion process.

\( (1098 \text{ kW}) \)
Modifications of Brayton Gas Turbine Cycles

Brayton/Joule Cycle (Closed & Ideal)

Open cycle

- Basic
- Two stage expansion
- Two stage compression with intercooler
- Two stage compression + Two stage expansion + Reheating + Heat exchanger
Two Stage Expansion Gas Turbine

- It is more practical to expand the hot gases in two turbine stages.
- The high-pressure, HP turbine is solely used to drive compressor.
- The low-pressure, LP turbine produces net power output.
Two Stage Expansion Gas Turbine

System diagram:
Two Stage Expansion Gas Turbine

System diagram:

T-s diagram:
Two Stage Expansion Gas Turbine

Energy Analysis
Neglecting changes in the kinetic and potential energy of the working fluid

Compressor work
The high-pressure turbine is dedicatedly used to drive the compressor, therefore

\[
W_{\text{com}} = W_{\text{HPT}}
\]
Two Stage Expansion Gas Turbine

Net work output

The low-pressure turbine produces the net work output for the plant, thus,

\[ w_{net} = w_{TLP} \]

\[ w_{net} = c_{pg}(T_4 - T_5) \]
Two Stage Expansion Gas Turbine

Example 9.3 (Eastop)

A gas turbine unit takes in air at 17°C and 1.01 bar with an overall pressure ratio 8:1 and a maximum cycle temperature of 650°C. The compressor is driven by high-pressure turbine and low-pressure turbine drives a separate power shaft. The isentropic efficiencies of the compressor, HP turbine and LP turbine are 0.8, 0.85 and 0.83 respectively. The mechanical efficiency of both shaft is 100%. Calculate the:

i) Pressure and temperature at the inlet of LP turbine
ii) Net power output for each kg/s mass flow rate
iii) Work ratio of the plant
iv) Thermal efficiency of the cycle

Take $c_p=1.005$ kJ/kgK and $\gamma=1.4$ for the compression process, and take $c_p=1.15$ kJ/kgK and $\gamma=1.333$ for the expansion process.
Modifications of Brayton Gas Turbine Cycles

Brayton/Joule Cycle (Closed & Ideal)

Open cycle

Basic
Two stage expansion
Two stage compression with intercooler
Two stage compression + Two stage expansion + Reheating
Two stage compression + Two stage expansion + Reheating + Heat exchanger
Two Stage Compression with Intercooler

- In real gas turbine plant, the compression process is carried out in more than one stage.
- An intercooler is used between each stage
- Reasons:
  1. The work required to drive compressor can be reduced
  2. Since the work output of LP turbine is unchanged, the work ratio is increased
Two Stage Compression with Intercooler

System diagram:

T-s diagram:
Two Stage Compression with Intercooler

Analysis of the cycle

i) The work required to drive compressor with intercooler

\[ w_c = c_{pa}(T_2 - T_1) + c_{pa}(T_4 - T_3) \]

ii) The work required to drive compressor without intercooler

\[ w_c = c_{pa}(T_2 - T_1) + c_{pa}(T_A - T_2) \]

Since the pressure lines on the T-s diagram diverge to the right, hence,

\[ c_{pa}(T_4 - T_3) < c_{pa}(T_A - T_2) \]

Therefore, the compressor work input with intercooler is **LESS** than the work input without intercooler.
Two Stage Compression with Intercooler

Minimum compressor work

The work input with intercooler will be MINIMUM when:

i) The pressure ratio in each stage is equal

\[
\frac{P_2}{P_1} = \frac{P_4}{P_3}
\]

ii) The intercooling process is complete, i.e. the temperature of the air entering between the stages high-pressure and low-pressure is the same.

\[T_3 = T_1\]
Two Stage Compression with Intercooler

Advantage

Two-stage compression with intercooling between the stages reduces the work required to drive the compressor

\[ w_r = \frac{w_T - w_C}{w_T} \]

Since \( w_C \) is reduced while \( w_T \) remains unchanged, the work ratio of the plant increased.
Two Stage Compression with Intercooler

Disadvantage

The heat added to the air during combustion with 2-stage compression and intercooling is

\[ q_{cc} = c_{pg}(T_5 - T_4) \]

and that without 2-stage & intercooling

\[ q_{cc} = c_{pg}(T_5 - T_A) \]

From T-s diagram,

\[ (T_5 - T_4) > (T_5 - T_A) \]

Therefore, the heat added to the air increases with 2-stage compression with intercooling. Hence, thermal efficiency of the cycle decreases.
Example 3

A gas turbine unit has a pressure ratio of 10/1 and a maximum temperature of 700°C. The compression process is carried out in 2 stages and intercooling is used between the stages. The isentropic efficiencies of the compressors and the turbine are 0.82 and 0.85 respectively. The air enters the compressor at 15°C at rate of 15 kg/s. Calculate,

i) Power output of an electric generator geared to the turbine

ii) Work ratio of the plant

iii) Thermal efficiency of the cycle

Assume the conditions for minimum compressor works. Take $c_p = 1.005$ kJ/kgK and $\gamma=1.4$ for the compression process, and take $c_p = 1.11$ kJ/kgK and $\gamma=1.333$ for the expansion process.
A 5000 kW gas turbine generating set operates with two compressor stages with intercooling between stages; the overall pressure ratio is 9/1. A HP turbine is used to drive the compressors, and a LP turbine drives the generator. The temperature of the gases at entry to the HP turbine is 650°C and the gases are reheated to 650°C after expansion in the first turbine. The exhaust gases leaving the LP turbine are passed through a heat exchanger to heat the air leaving the HP stage compressor. The compressors have equal pressure ratios and intercooling is complete between stages. The air inlet temperature to the unit is 15°C. The isentropic efficiency of each compressor stage is 0.8 and the isentropic efficiency of each turbine stage is 0.85; the heat exchanger thermal ratio is 0.75. A mechanical efficiency of 98% can be assumed for both the power shaft and the compressor turbine shaft. Neglecting all pressure losses and changes in kinetic energy, calculate:

(i) the cycle efficiency;
(ii) the work ratio;
(iii) the mass flow rate.

For air take $c_p = 1.005 \text{ kJ/kg K}$ and $\gamma = 1.4$, and for the gases in the combustion chamber and in the turbines and heat exchanger take $c_p = 1.15 \text{ kJ/kg K}$ and $\gamma = 1.333$. Neglect the mass of fuel.
Modifications of Brayton Gas Turbine Cycles

Brayton/Joule Cycle (Closed & Ideal)

Open cycle

Basic

Two stage expansion

Two stage compression with intercooler

Two stage compression + Two stage expansion + Reheating

Two stage compression + Two stage expansion + Reheating + Heat exchanger
Two Stage Expansion and Reheating

With 2-stage expansion, the gas exiting high-pressure turbine can be reheated in a second combustion chamber, before expanding in the low-pressure turbine.
Two Stage Expansion and Reheating

Analysis of the cycle

Neglecting any mechanical losses, the work produced by low-pressure turbine WITH reheating,

\[ w_{LPT} = c_{pg}(T_5 - T_6) \]

And the work produces WITHOUT reheating,

\[ w_{LPT} = c_{pg}(T_4 - T_A) \]

The constant pressure lines diverge to the right on the T-s diagram, hence,

\[ (T_5 - T_6) > (T_4 - T_A) \]

Thus, reheating between the turbine stage increases the work output of the low-pressure (LP) turbine.

Hence, reheating between the turbine stages increases the work output of the low-pressure (LP) turbine.

\[ (T_5 - T_6) > (T_4 - T_A) \]
Two Stage Expansion and Reheating

Advantage

The work ratio of the plant is given by,

$$w_r = \frac{\sum w_T - w_c}{\sum w_T}$$

The total turbine work increases with reheating between the turbine stages. Therefore, the work ratio of the plant increases.
Two Stage Expansion and Reheating

Disadvantage

The heat added to the working fluid in the combustion chambers

\[
\sum q_{cc} = c_{pg} (T_3 - T_2) + c_{pg} (T_5 - T_4)
\]

“\(c_{pg} (T_5 - T_4)\)” is the additional heat added to the working fluid in the second combustion chamber.

Clearly, reheating between the turbine stages increases the amount of heat supplied in the combustion chambers.

This cause the thermal efficiency of the plant to decrease.
Example 4

A gas turbine unit is used in a marine ship consists of a compressor, two combustion chambers and two stages of turbines. The compressor is driven by the high-pressure (HP) turbine and the low-pressure (LP) turbine drives the ship rotor. The overall pressure ratio is 10:1. The atmosphere air enters the compressor at 1.01 bar and 30°C with the rate of 20 kg/s. The gas enters the HP turbine at 900°C and enters the LP turbine at 700°C. The mechanical efficiency for both shafts is 90%. The isentropic efficiency for both turbines is 85%. The power received by the rotor is 2590 kW.

a) Show the plant components arrangement and all the processes on a T-s diagram,

b) Calculate the intermediate pressure between the HP turbine and LP turbine,

c) Calculate the compressor power;

d) Calculate the isentropic efficiency of the compressor;

e) Thermal efficiency of the cycle.
Two Stage Expansion, Two Stage Compression and Reheating

T-s diagram:
Example 5

A gas turbine power plant consists of 2-stages compressor and 2-stages turbine. The high-pressure and low-pressure compressor are driven by the high-pressure (HP) turbine and the low-pressure (LP) turbine drives a separate power shaft. The low-pressure compressor takes in air at 100 kPa and 27°C with the mass flow rate of 5.80 kg/s. The compressors have equal pressure ratios and inter-cooling is complete between stages. The temperature of the gases at entry to the HP turbine is 1327°C and the gases are reheated to 1127°C at 300 kPa after expansion in the first turbine. The overall pressure ratio is 10. The isentropic efficiency of each compressor stage and each turbine stage is 80%. Calculate:

a) net power output (kW),

b) work ratio of the plant,

c) thermal efficiency of the cycle.

Sketch the schematic diagram of the plant and all the processes on a T-s diagram.
EXAMPLE 9–8 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 9–5.
Modifications of Brayton Gas Turbine Cycles

Brayton/Joule Cycle (Closed & Ideal)

Open cycle

Basic
Two stage expansion
Two stage compression with intercooler
Two stage compression + Two stage expansion + Reheating + Heat exchanger
Gas Turbine with Heat Exchanger

- The gases exiting low-pressure turbine is still has a high temperature.
- The heat energy contained in the exhaust gases can be utilized for improving the thermal efficiency of the cycle.
- One scheme is by using a heat exchanger unit to preheat the air leaving the compressor, before it enters the combustion chamber.
Gas Turbine with Heat Exchanger

Ideal vs. Actual temperature

➢ In an **ideal** cycle, temperatures

\[ T_2 = T_6 \text{ and } T_3 = T_5 \]

➢ In an **actual** cycle, temperatures

\[ T_2 < T_6 \text{ and } T_3 < T_5 \]

The finite temperature difference is required, between the compressed air and the exhaust gases, for the heat transfer process to take place.
Gas Turbine with Heat Exchanger

Advantage of heat exchanger

When the compressed air is preheated before entering the combustion chamber, the amount of heat required to raise the temperature of the working fluid from $T_2$ to $T_4$ is reduced.

Heat only required to raise the temperature of the air from $T_3$ to $T_4$. This leads to save in fuel consumption and hence the operating cost of the plant.

Assuming that net work output of the plant remains unchanged, the thermal efficiency of the cycle increases by using heat exchanger.
Gas Turbine with Heat Exchanger

Heat energy balance

Assuming the heat exchanger is well insulated (no heat loss), we have

Heat transferred from the exhaust gases = Heat transferred to the compressed air

\[ m_g c_{pg}(T_5 - T_6) = m_a c_{pa}(T_3 - T_2) \]
Gas Turbine with Heat Exchanger

Heat exchanger performance

- The performance of the heat exchanger is measured by its **thermal ratio**, defined as

  \[
  \eta_{he} = \frac{\text{Temperature rise of the air}}{\text{Maximum available temperature difference}}
  \]

  \[
  \eta_{he} = \frac{(T_3 - T_2)}{(T_5 - T_2)}
  \]

- Heat supply without HE,

  \[
  q_{cc} = c_p g (T_4 - T_2)
  \]

- Heat supply with HE,

  \[
  q_{cc} = c_p g (T_4 - T_3)
  \]
Gas Turbine with Heat Exchanger

Conditions for using heat exchanger

To use heat exchanger, there must be a sufficiently large temperature difference between the exhaust gases and the compressed air.

The use of heat exchanger will not be feasible when the temperature of the exhaust gases is lower than the temperature of the compressed air leaving the compressor.
Gas Turbine with Heat Exchanger

Example 6

You are required to carry out an energy audit on a gas turbine plant. In this plant the high pressure turbine (HPT) drives a compressor and the low pressure turbine (LPT) drives an electric generator (EG). Atmospheric air, after being compressed, enters a first combustion chamber (CC1) and the hot gas is then expanded in the HPT. The gas is reheated in a second combustion chamber (CC2) before it is expanded in the LPT. Based on the true measurements made, followings are the data recorded by your Technical Assistant:

- output power of generator, 1080 kW;
- mass flow-rate of air, 6.85 kg/s;
- compressor pressure ratio, 6.3 and HPT pressure ratio, 2.9;
- compressor inlet and outlet air temperatures, 20°C and 252°C;
- HPT inlet and outlet gas temperatures, 701°C and 487°C;
- LPT inlet and outlet gas temperatures, 631°C and 486°C.
Sketch all processes on a T-s diagram and determine the following values, for you to prepare the audit report:

i. compressor isentropic efficiency;
ii. HPT isentropic efficiency;
iii. HPT/compressor mechanical efficiency;
iv. LPT isentropic efficiency;
v. LPT/EG mechanical efficiency;
vi. plant thermal efficiency, and

vii. in the same report, you are proposing that a heat exchanger with a thermal ratio of 0.78, be installed to preheat the compressed air by the exhaust gas. To support your proposal what are:

(a) the percentage saving on heat supply by CC1, and
(b) the new value of the plant thermal efficiency.

Please use the following assumptions: for air, \( c_p = 1.005 \text{ kJ/kg} \) and \( \gamma = 1.4 \) for gas, \( c_p = 1.15 \text{ kJ/kg} \) and \( \gamma = 1.333 \).