MODULE -5 STEAM TURBINE AND GAS TURBINE

- Steam turbine-Impulse and Reaction turbine-Performance

- Gas turbine-Open and Closed cycle gas turbine, Reheating, Regeneration and Intercooling.

STEAM TURBINE

A **steam turbine** is a prime mover in which potential energy (steam) is converted into kinetic energy and then to Mechanical energy.

Potential Energy

↓

Kinetic energy

↓

Mechanical Energy

STEAM TURBINE

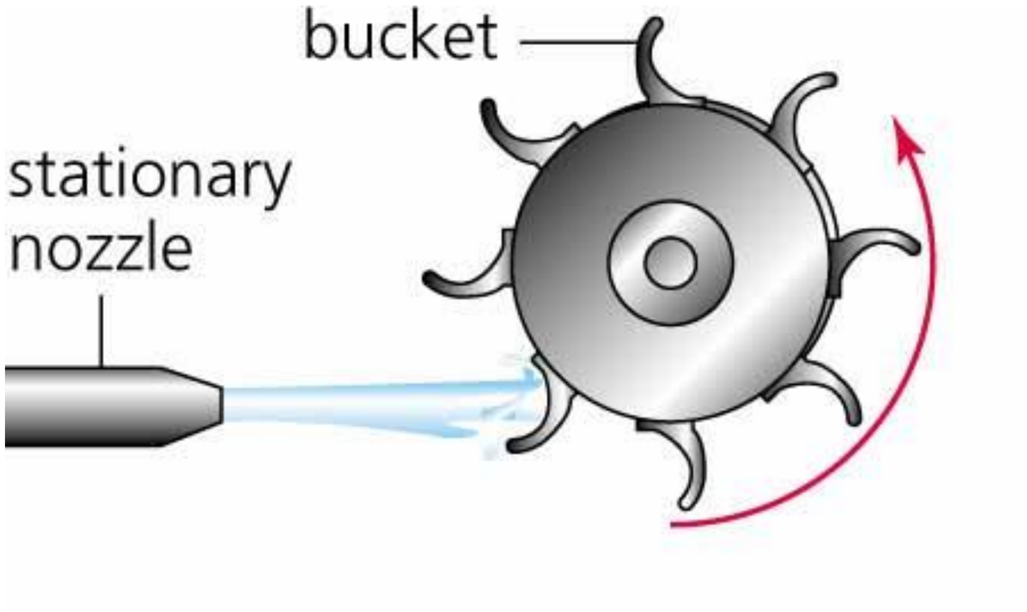
Common Types of Turbines

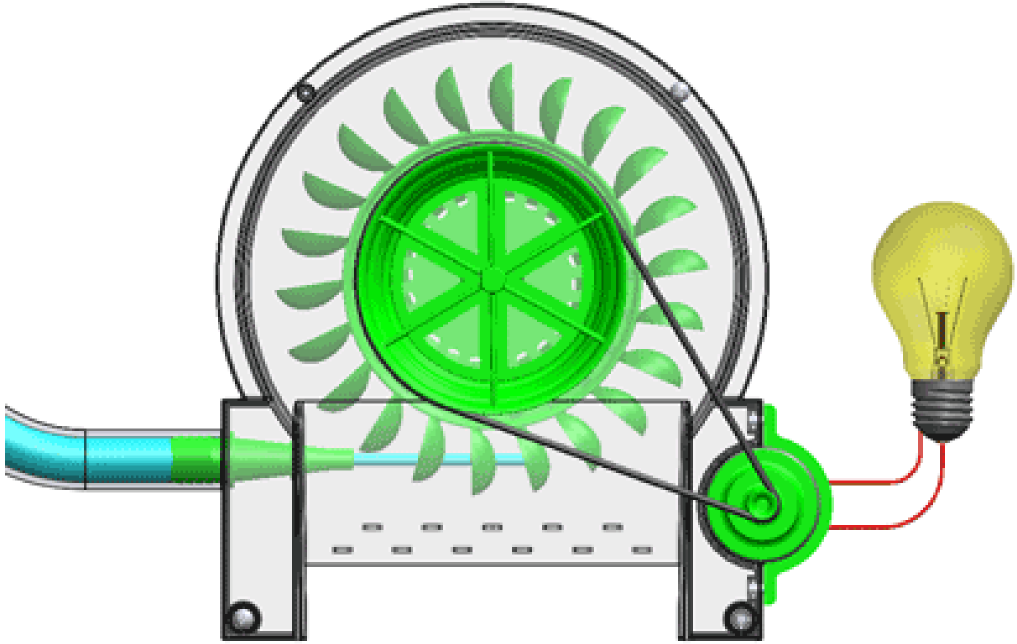
1. Impulse Turbine and 2. Reaction Turbine.

The main difference between these two turbines lies in the way of expanding the steam.

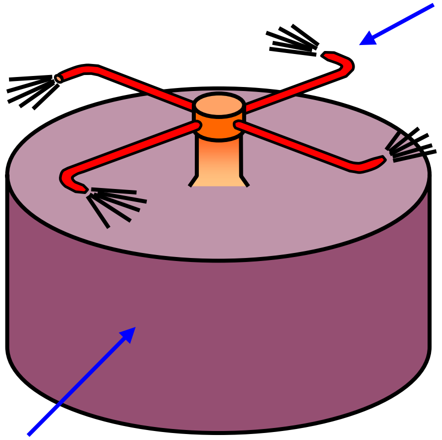
- In the **impulse turbine**, the steam expands in the nozzles and it's pressure does not alter as it moves over the blades.
- In the **reaction turbine** the steam expanded continuously as it passes over the blades and thus there is gradually fall in the pressure during expansion below the atmospheric pressure.

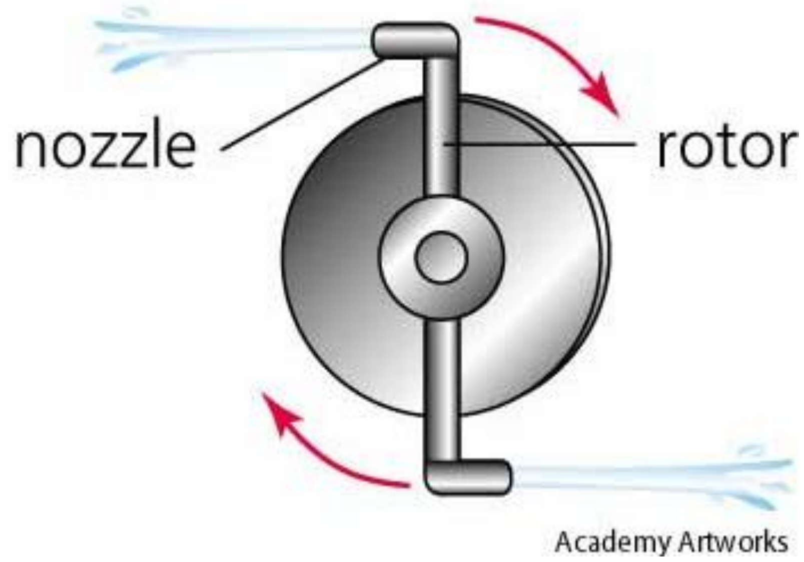
STEAM TURBINE-Impulse type





STEAM TURBINE-Reaction Type





IMPULSE Vs REACTION tURNINE

Impulse turbine	*Reaction turbine*
➢ The steam completely expands in the nozzle and its pressure remains constant during its flow through the blade passages	➢ The steam expands partially in the nozzle and further expansion takes place in the rotor blades
➢ The relative velocity of steam passing over the blade remains constant in the absence of friction	➢ The relative velocity of steam passing over the blade increases as the steam expands while passing over the blade
➢ Blades are symmetrical	➢ Blades are asymmetrical
➢ The pressure on both ends of the moving blade is same	➢ The pressure on both ends of the moving blade is different
➢ For the same power developed, as pressure drop is more, the number of stages required are less	➢ For the same power developed, as pressure drop is small, the number of stages required are more
➢ The blade efficiency curve is less flat	➢ The blade efficiency curve is more flat
➢ The steam velocity is very high and therefore the speed of turbine is high.	➢ The steam velocity is not very high and therefore the speed of turbine is low.

Compounding of Steam Turbine

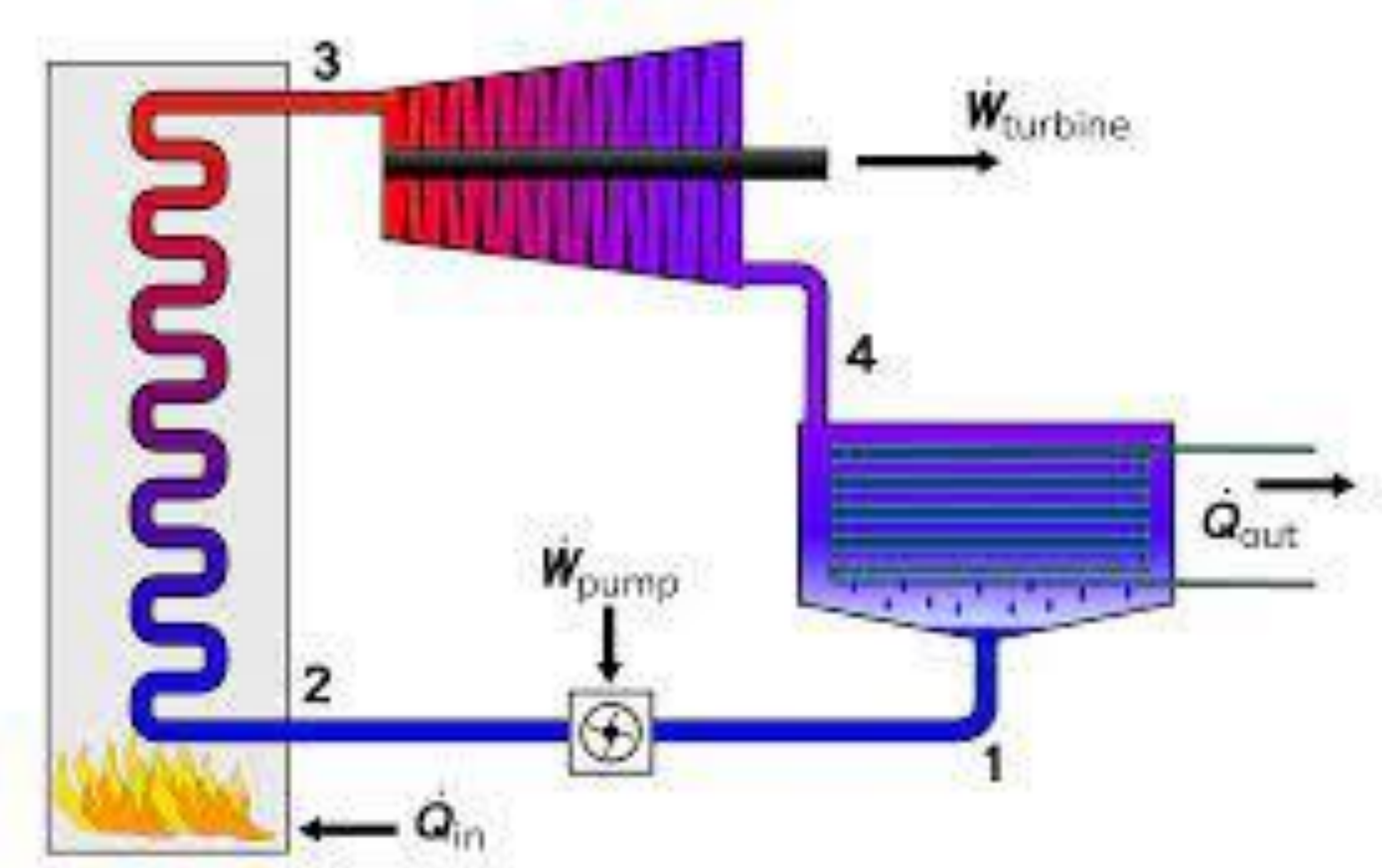
- Compounding of a steam turbine refers to the method of using multiple stages to extract energy from the expanding steam, increasing efficiency and power output.

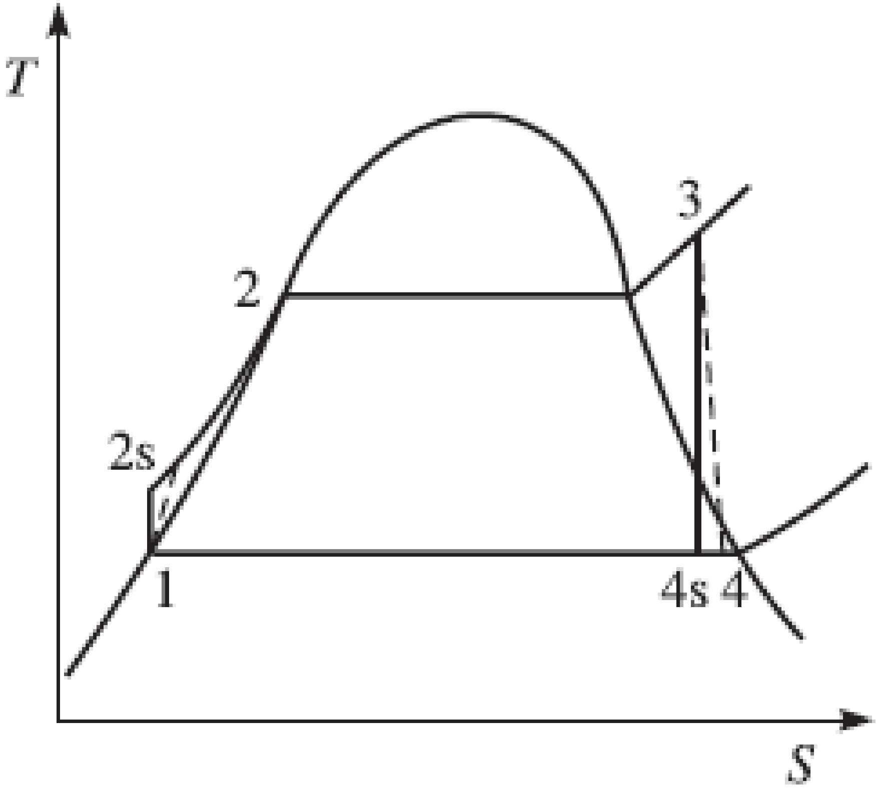
Types of Compounding:

1. Pressure Compounding: Divides the pressure drop across multiple stages.
2. Velocity Compounding: Uses multiple rows of blades to extract energy from the steam velocity.
3. Pressure-Velocity Compounding: Combines both pressure and velocity compounding.

- Stages in a Compounded Steam Turbine:1. High-Pressure (HP) Stage: Expands high-pressure steam.2. Intermediate-Pressure (IP) Stage: Expands steam from the HP stage.3. Low-Pressure (LP) Stage: Expands steam from the IP stage.4. Last Stage: Extracts remaining energy from the steam.
- Advantages: 1. Increased efficiency 2. Higher power output 3. Reduced steam consumption4. Improved part-load performance

STEAM TURBINE





Numerical on Rankine cycle

A steam power plant operating on the simple ideal Rankine cycle. The steam enters the turbine at 3 Mpa & 350°C and is condensed in the condenser at a pressure of 75 kPa. Determine the thermal efficiency of this cycle.

Solution:

In Ideal Rankine cycle,

(i) Pump and turbine are isentropic

(ii) No pressure drop in boiler and condenser

(iii) Steam leaves the condenser and enters the boiler as saturated liquid at the condenser pressure.

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{q_{in} - q_{out}}{q_{in}} = \frac{w_{turbine} - w_{pump}}{q_{in}}$$

Numerical on Rankine cycle

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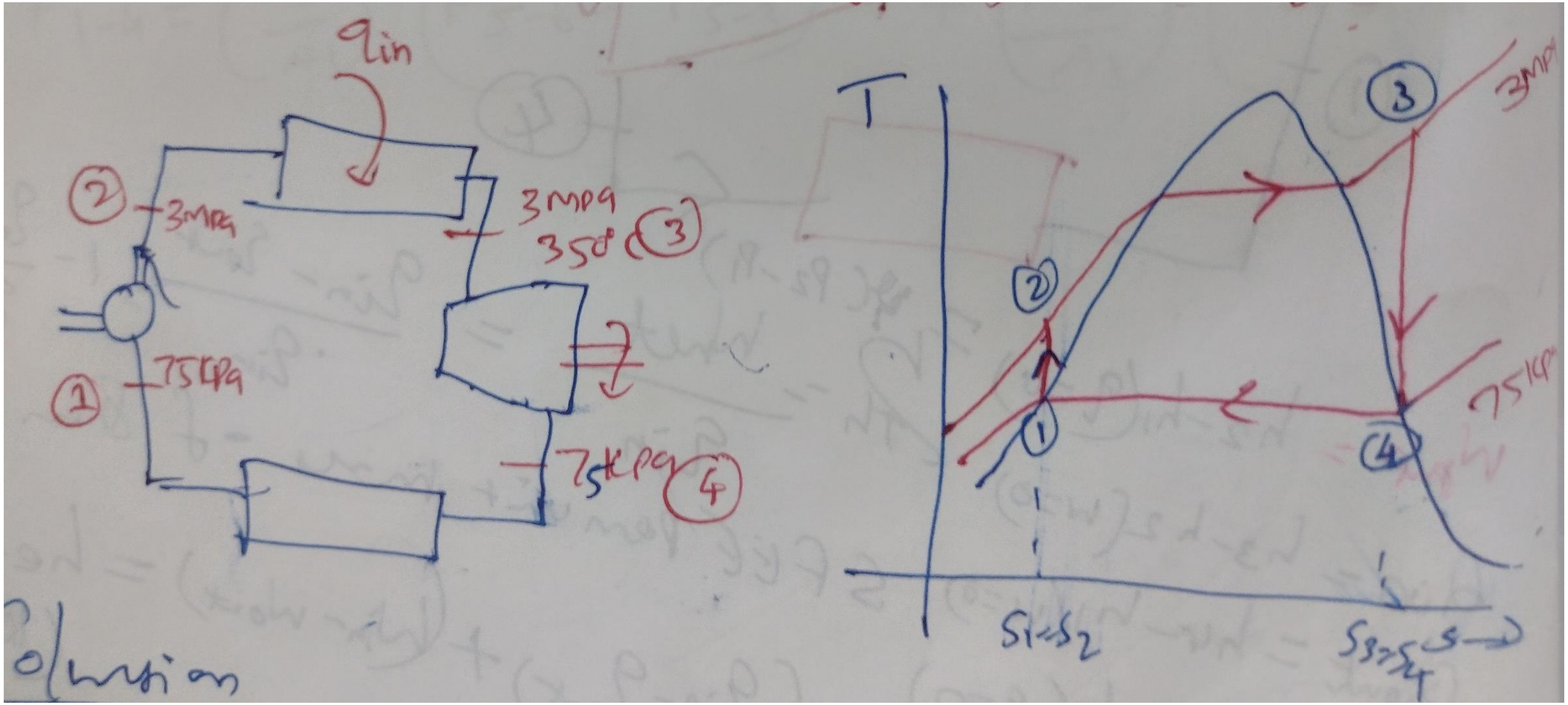


Numerical on Rankine cycle

State-1 (Saturated Liquid)

For, P_1= 75 kPa, from steam Table,

$$h_1 = h_f = 384.39\ kJ/kg$$

$$v_1 = v_f = 0.001037\ m^3/kg$$

Numerical on Rankine cycle

State-2 (Subcooled)

For, $P_2 = 3\ Mpa$

$$W_{pump} = v_1 (P_2 - P_{1)} = h_2 - h_1$$

$$W_{pump} = 0.001037 \times (3000\text{-}75) = 3.03\ kJ/kg$$

$$h_2 = h_1 + 3.03 = 387.42\ kJ/kg$$

Numerical on Rankine cycle

State-3 (Super heated steam)

P_3= 3 Mpa and T_3=350°C

(From Steam Table page.22)

h_3=3117.5 kJ/kg

s_3=6.747 kJ/kg-K

Note: $s_3 = s_4$

Numerical on Rankine cycle

State-4 (Saturated Mixture)

For, P_4= 75 kPa

(From saturated pressure Table, page. 9)

$$h_4 = h_f + x_4 h_{fg}$$

$$s_4 = s_f + x_4 s_{fg} \Rightarrow$$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = 0.8857$$

$$h_4 = 2402.6 \, kJ / kg$$

Numerical on Rankine cycle

$$Cycle - efficiency$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}}$$

$$q_{in} = h_3 - h_2 = 2727.9\, kJ / kg$$

$$q_{out} = h_4 - h_1 = 2018.2\, kJ / kg$$

$$\eta_{th} = 26\%$$

Numerical on Rankine cycle

$$Carnot - efficiency$$

$$\eta_{carnot} = 1 - \left(\frac{T_{\min}}{T_{\max}} \right)$$

$$= 1 - \left(\frac{91.78 + 273}{350 + 273} \right)$$

$$= 0.414 = 41.4\%$$

Back Work Ratio

Note:

- This difference between the two-efficiencies is due to large temperature difference between the steam and the combustion gases during the heat addition process.

- Back Work Ratio:

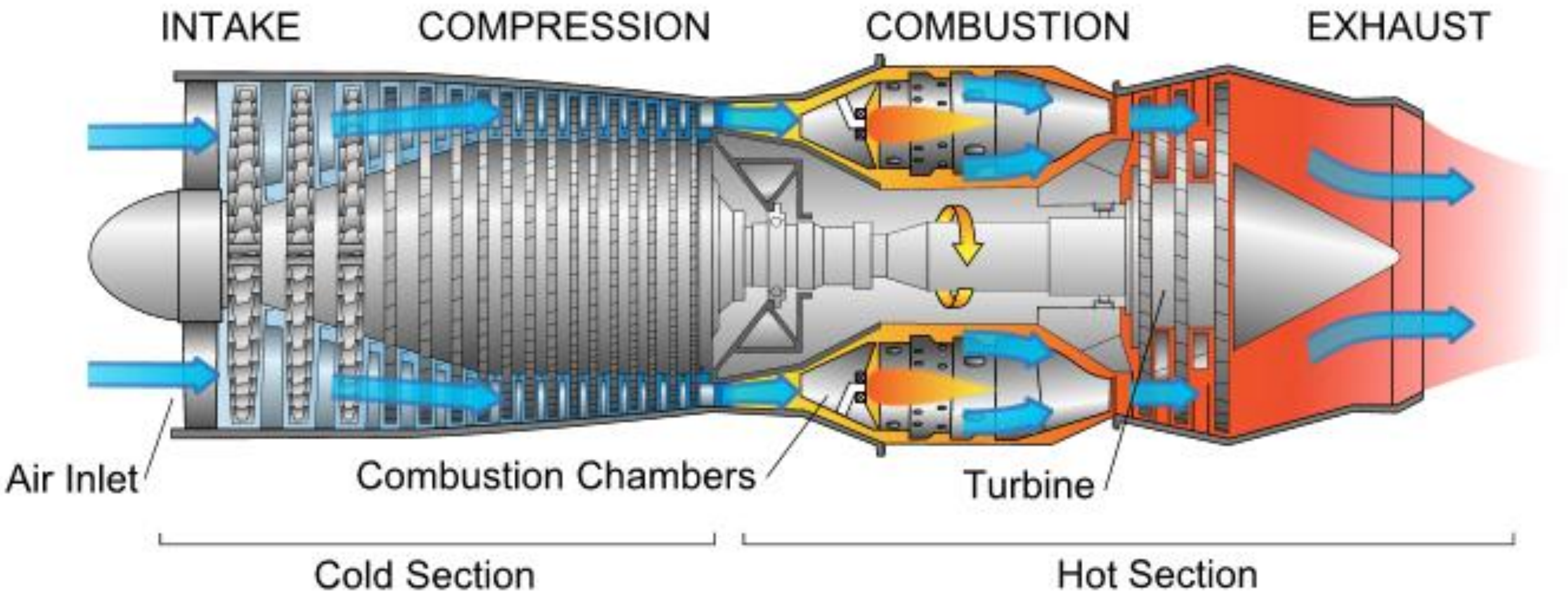
$$r = \frac{W_{input}}{W_{output}} = \frac{W_{pump}}{W_{Turbine}} = \frac{h_2 - h_1}{h_3 - h_4} = \frac{3.03}{721.7} = 0.004 = 0.4\%$$

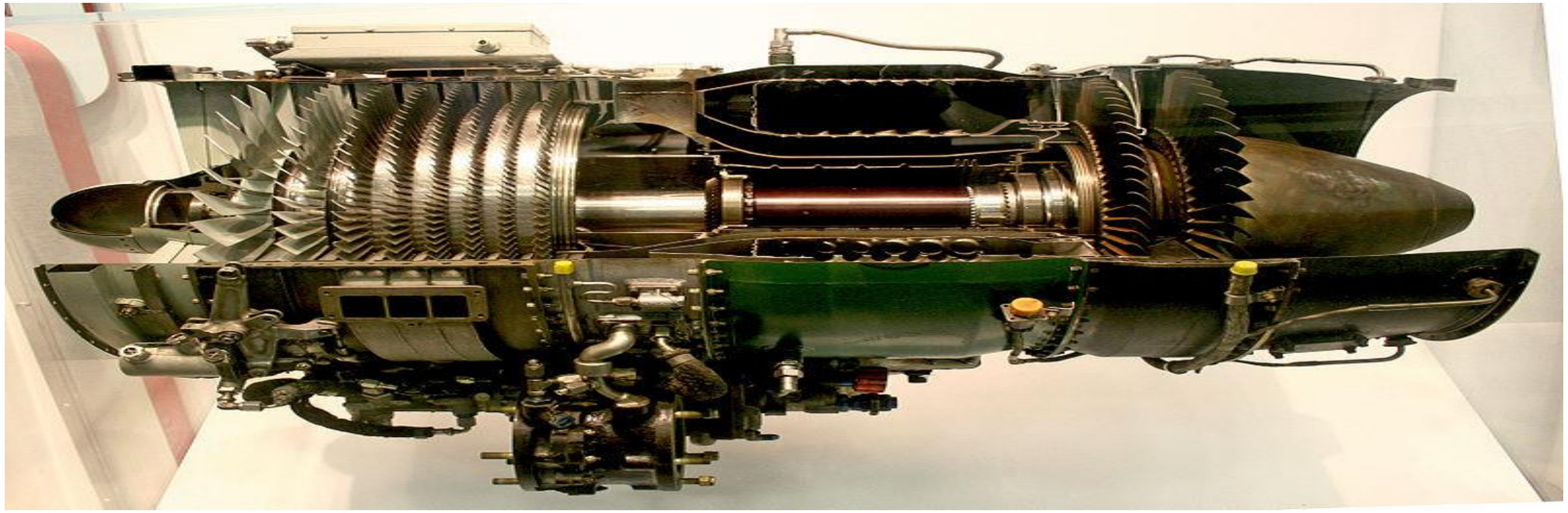
- Generally gas power cycles have very high back work ratio (40%-80%)

ADVANTAGES OF STEAM TURBINES OVER GAS TURBINES

- The load control in steam turbine is easy, simply by throttle governing or by cut-off governing.
- In a gas turbine, the air fuel ratio becomes too high, 100 to 150 at part loads. This causes problems to sustain the flame.
- A steam turbine works on Rankine Cycle. In this cycle, most of the heat is supplied at constant temperature in the form of latent heat of evaporation. Also, the heat is rejected in the condenser isothermally. Hence, **the cycle is more efficient**, and its efficiency is close to that of Carnot Cycle efficiency.
- On the other hand, the gas turbine works on Brayton cycle whose efficiency is much less than that of Carnot cycle working between the same maximum and minimum limits of temperatures.
- The efficiency of steam turbine, at part load, is not very much reduced. In gas turbines, the maximum cycle temperature decreases considerably at part load; therefore, its part load efficiency is very low.
- The blade material for steam turbine is not costly. For gas turbine, the blade material is very costly as it is required to sustain considerably high temperatures.

Components of GAS TURBINE





Brayton Cycle: Actual Gas-Turbine Engines

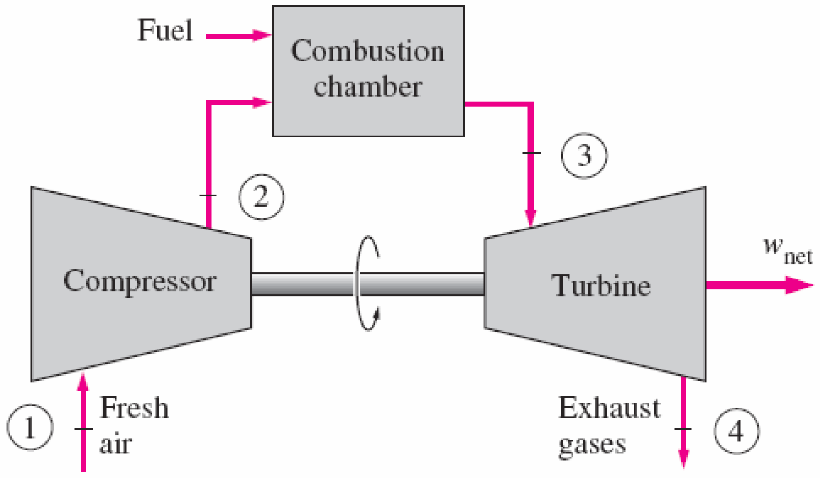
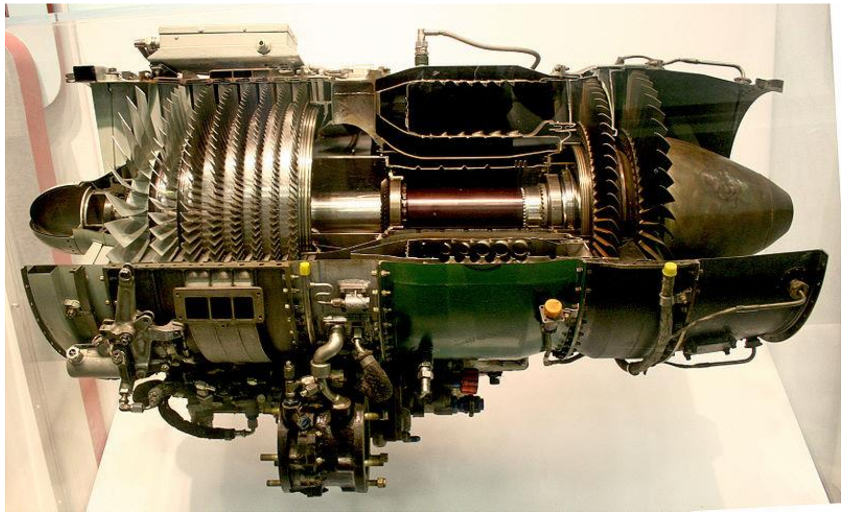


FIGURE 9–29

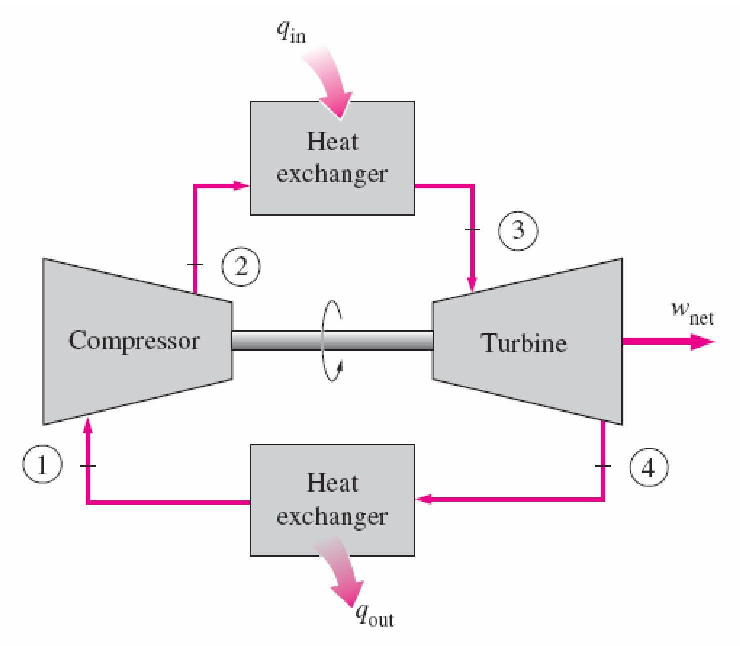
An open-cycle gas-turbine engine.



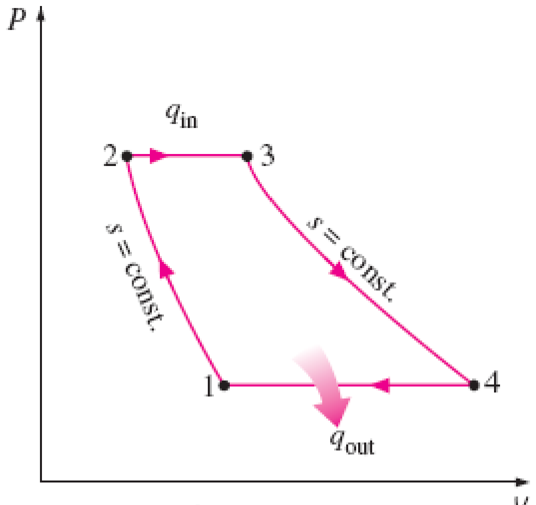
- Gas turbines usually operate on an open cycle
- Air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. The high pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure.

- 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**.

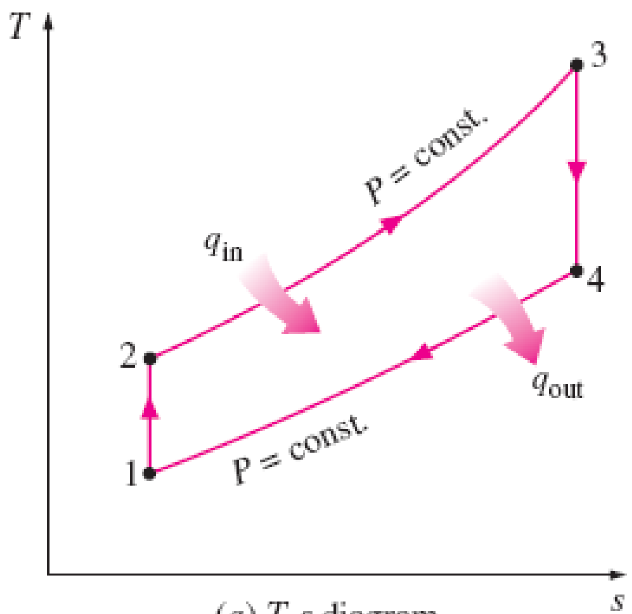
Closed Cycle- IDEAL GAS TURBINE



- The open gas-turbine cycle can be modelled as a closed cycle, using the **air-standard** assumptions **(Ideal cycle)** .
- 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** s to the ambient air.



(b) P-v diagram



(a) T-s diagram

The Ideal-Brayton Cycle

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:

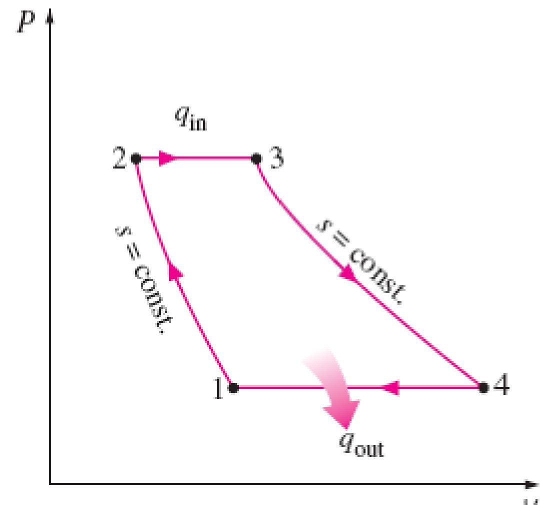
1-2 Isentropic compression;

2-3 Constant-pressure heat addition;

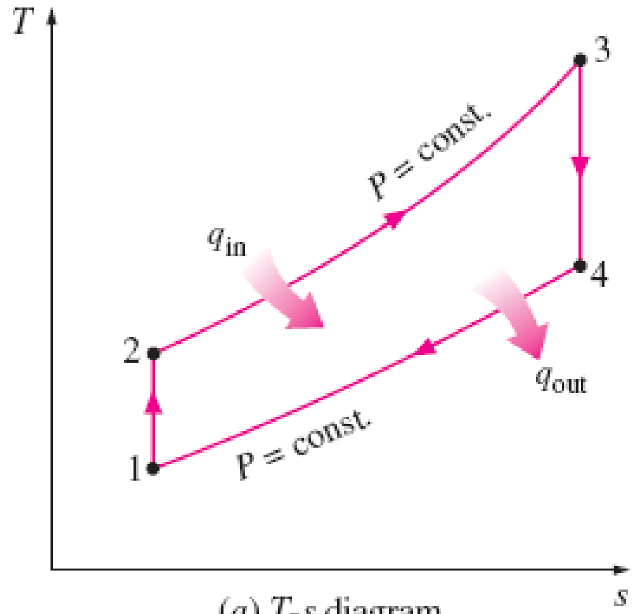
3-4 Isentropic expansion;

4-1 Constant-pressure heat rejection.

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



(b) P-v diagram



(a) T-s diagram

Thermal Efficiency

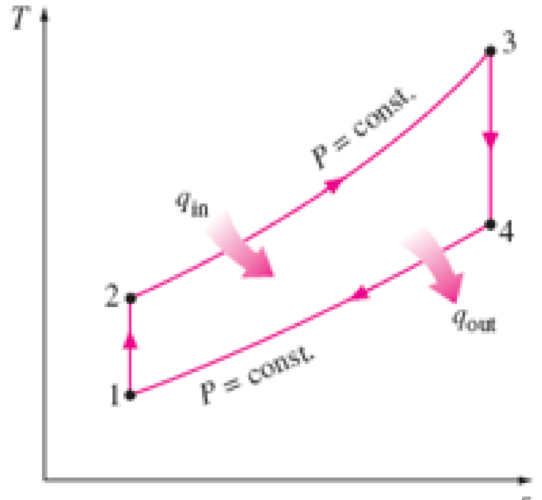
The **energy balance** for a steady-flow process can be expressed, on a unit–mass basis, as

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

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

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

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



(a) T-s diagram

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

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

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

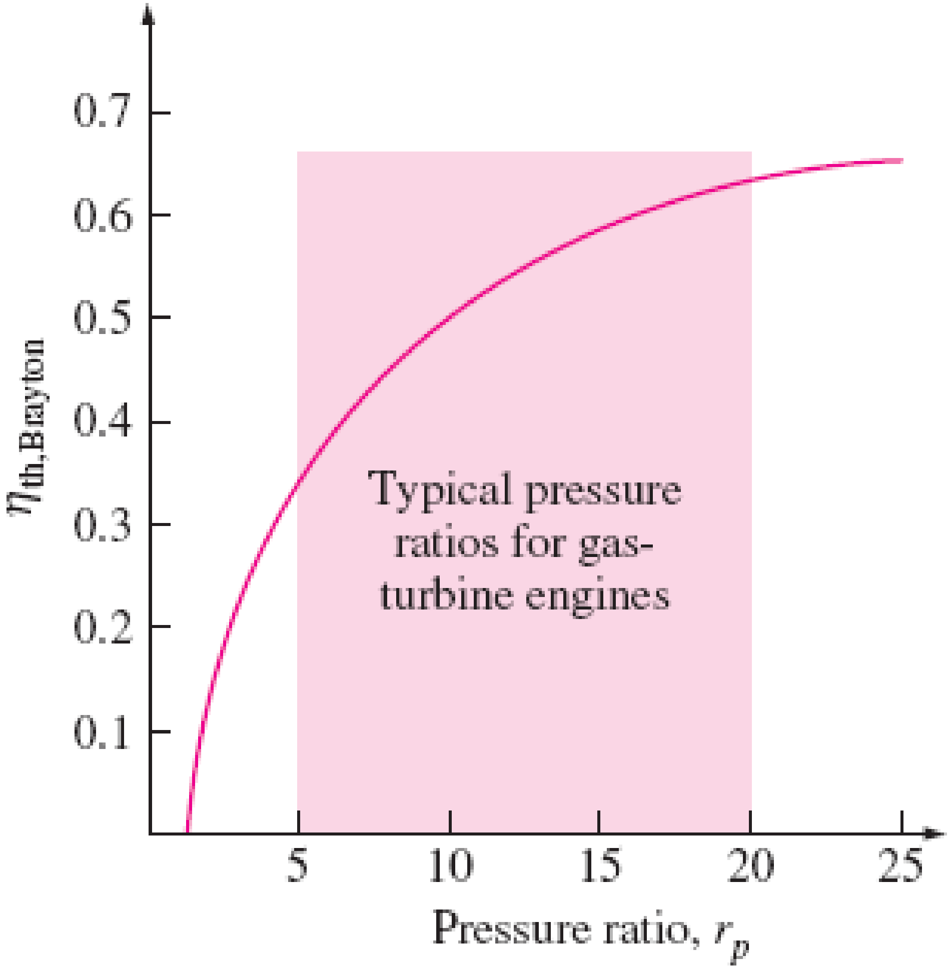
$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}}$$ Constant specific heats

where $r_p = \frac{P_2}{P_1}$ is the **pressure ratio**.

GAS TURBINE

Parameters Affecting Thermal Efficiency

- The thermal efficiency of an ideal Brayton cycle depends on the pressure ratio, r_p of the gas turbine and the specific heat ratio, *k* 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 in Fig, for the case of k =1.4.



Improvements of Gas Turbine's Performance

1. **Increasing the turbine inlet (or firing) temperatures.**
   The turbine inlet temperatures have increased steadily from about 540°C (1000°F) in the 1940s to 1425°C (2600°F) and even higher today.

2. **Increasing the efficiencies of turbo-machinery components (turbines, compressors).**
   The advent of computers and advanced techniques for computer-aided design made it possible to design these components aerodynamically with minimal losses.

3. **Adding modifications to the basic cycle (intercooling, regeneration, and reheating).**
   The simple-cycle efficiencies of early gas turbines were practically doubled by incorporating intercooling, regeneration (or recuperation), and reheating.

ACTUAL GAS TURBINE

- Some **pressure drop** occurs during the heat-addition and heat rejection processes.
- The actual work input to the compressor is more, and the actual work output from the turbine is less, because of **irreversibilities**.

Deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accounted for by utilizing **isentropic efficiencies** of the turbine and compressor.

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

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

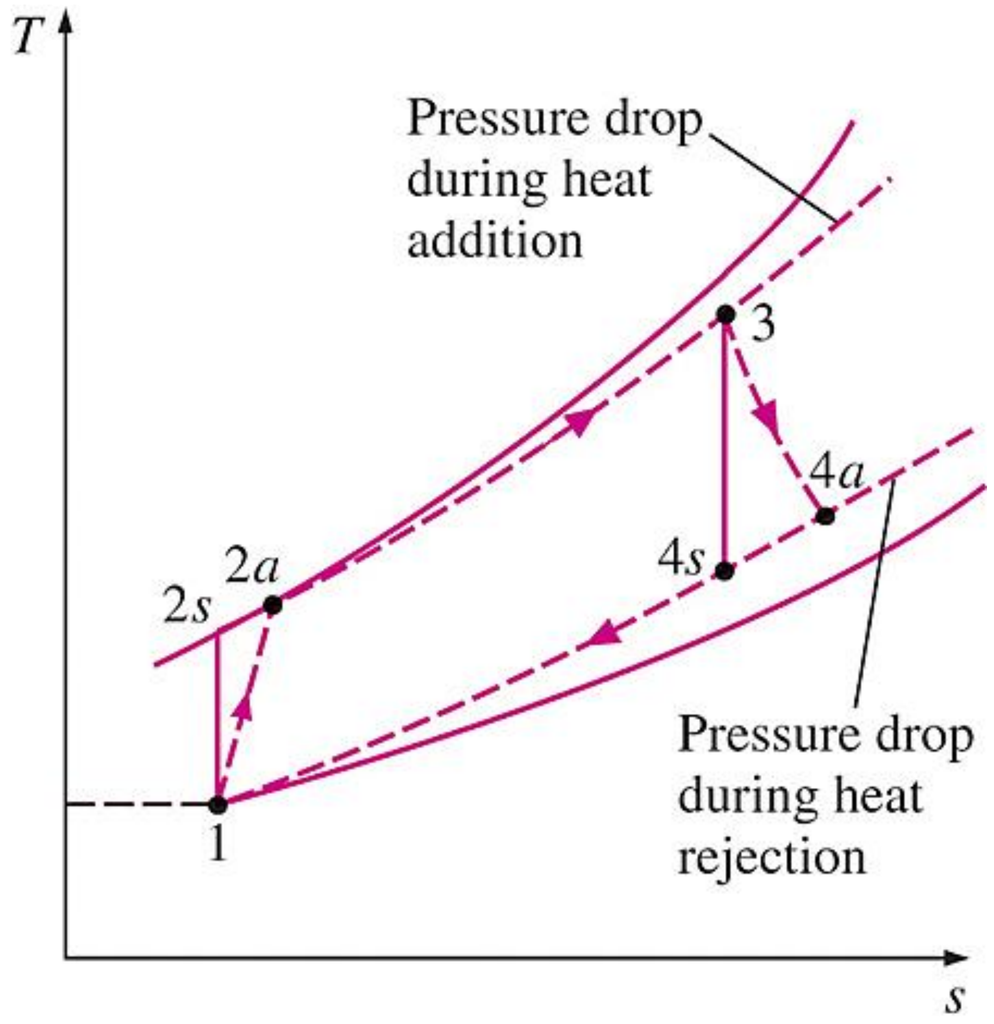


FIGURE 9–36

The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

Brayton Cycle With Regeneration

- Temperature of the exhaust gas leaving the turbine is **higher** than the temperature of the air leaving the compressor.
- The air leaving the compressor can be heated by the hot exhaust gases in a **counter-flow** heat exchanger (a *regenerator* or *recuperator*) – a process called **regeneration** (Fig. 9-38 & Fig. 9-39).
- The thermal efficiency of the Brayton cycle **increases** due to regeneration since less fuel is used for the same work output.

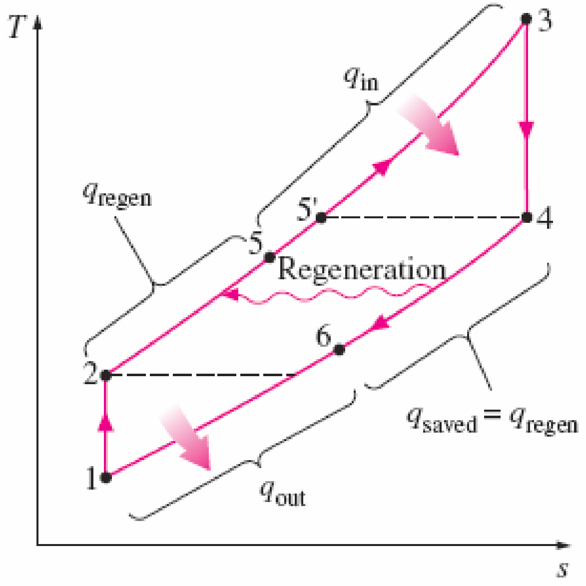


FIGURE 9–39

T-s diagram of a Brayton cycle with regeneration.

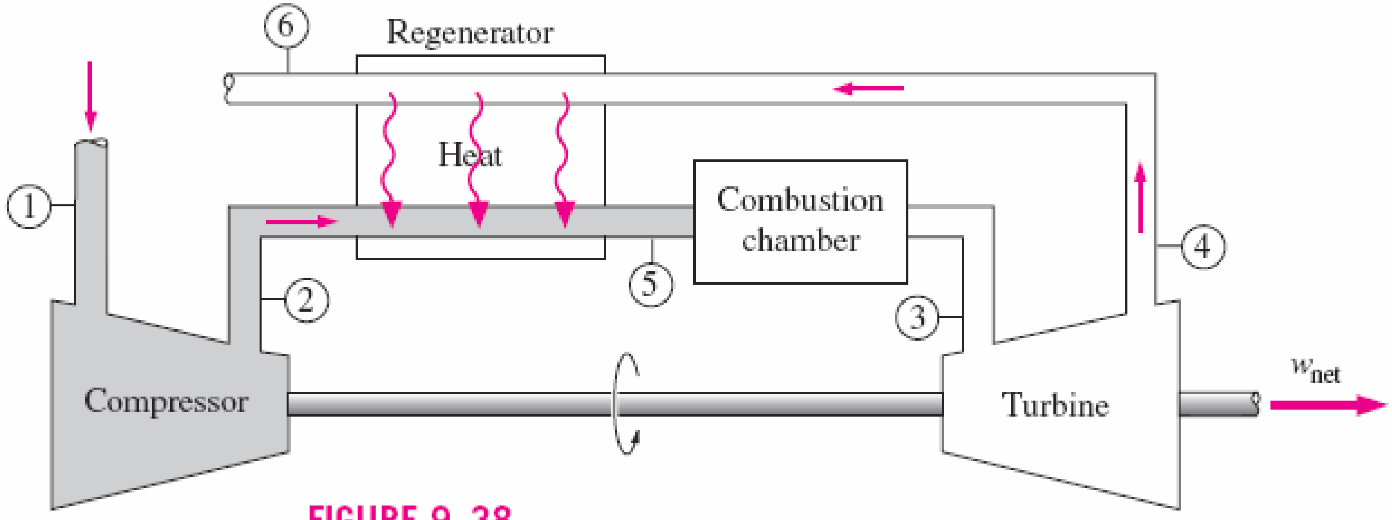


FIGURE 9–38

A gas-turbine engine with regenerator.

Note:
The use of a regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature.

Effectiveness of the Regenerator

Assuming the regenerator is well insulated and changes in kinetic and potential energies are negligible, the **actual** and **maximum heat transfers** from the exhaust gases to the air can be expressed as

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

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

$$effectiveness\ of\ regenerator\ (Heat - exchanger)$$

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

Under Air standard assumptions

$$\varepsilon = \frac{T_5 - T_2}{T_4 - T_2} \qquad \eta_{th} = 1 - \left(\frac{T_1}{T_3}\right)\left(r_p\right)^{\frac{\gamma-1}{\gamma}}$$

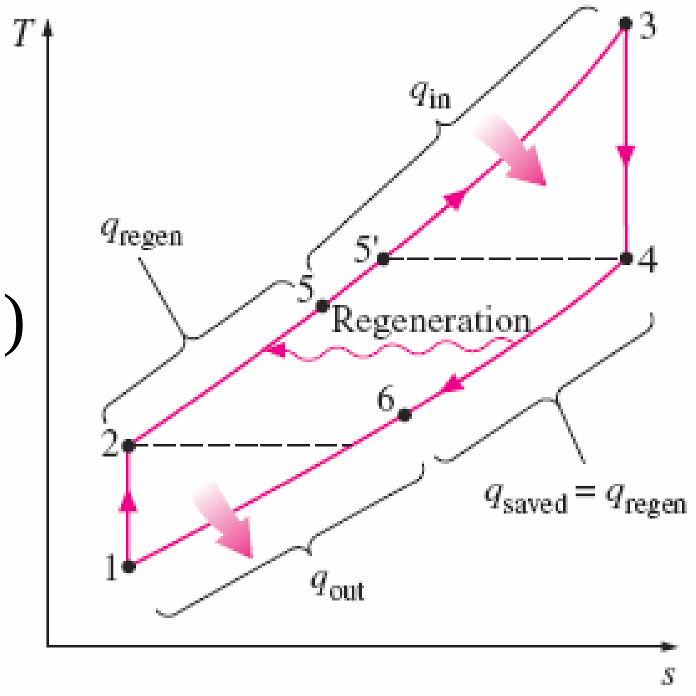


FIGURE 9–39

T-s diagram of a Brayton cycle with regeneration.

Factors Affecting Thermal Efficiency

Thermal efficiency of Brayton cycle with regeneration depends on:

a) ratio of the minimum to maximum temperatures,

b) the pressure ratio.

Conclusion:

Regeneration is most effective at **lower** pressure ratios and **small** minimum-to-maximum temperature ratios.

$$\eta_{th} = 1 - \left(\frac{T_1}{T_3}\right)\left(r_p\right)^{\frac{\gamma-1}{\gamma}}$$

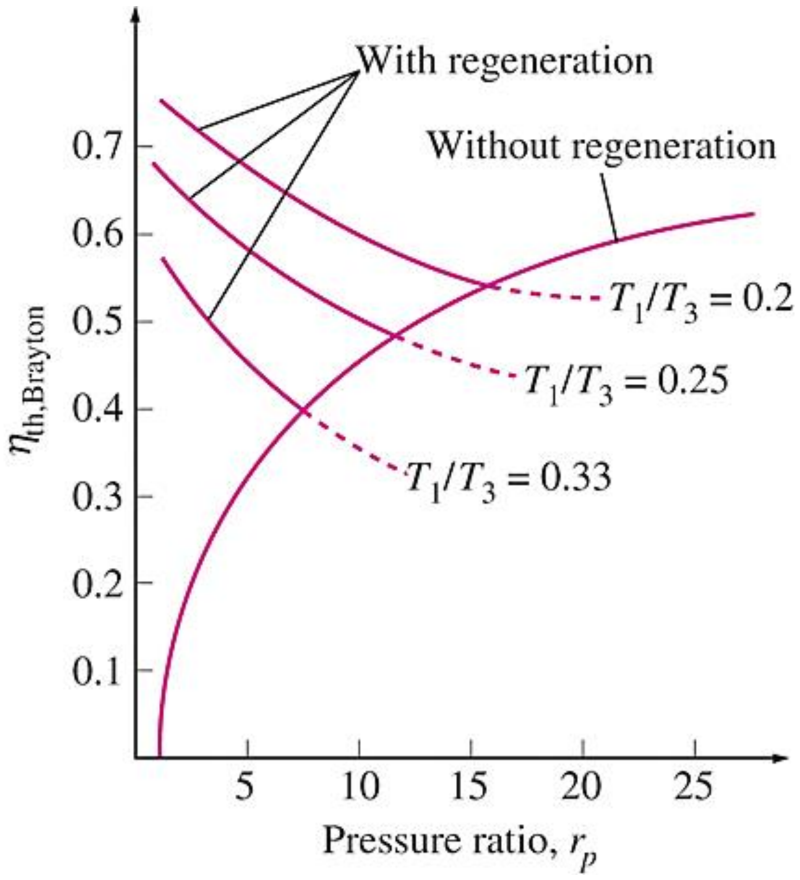


FIGURE 9–40

Thermal efficiency of the ideal Brayton cycle with and without regeneration.

Brayton Cycle With Intercooling, Reheating, & Regeneration

The **net work output** of a gas-turbine cycle can be increased by either:

a) decreasing the compressor work, or
b) increasing the turbine work, or
c) both.

The compressor work input can be decreased by carrying out the compression process in stages and cooling the gas in between (Fig. 9-42), using **multistage compression** with **intercooling**.

The work output of a turbine can be increased by expanding the gas in stages and reheating it in between, utilizing a **multistage expansion** with **reheating.**

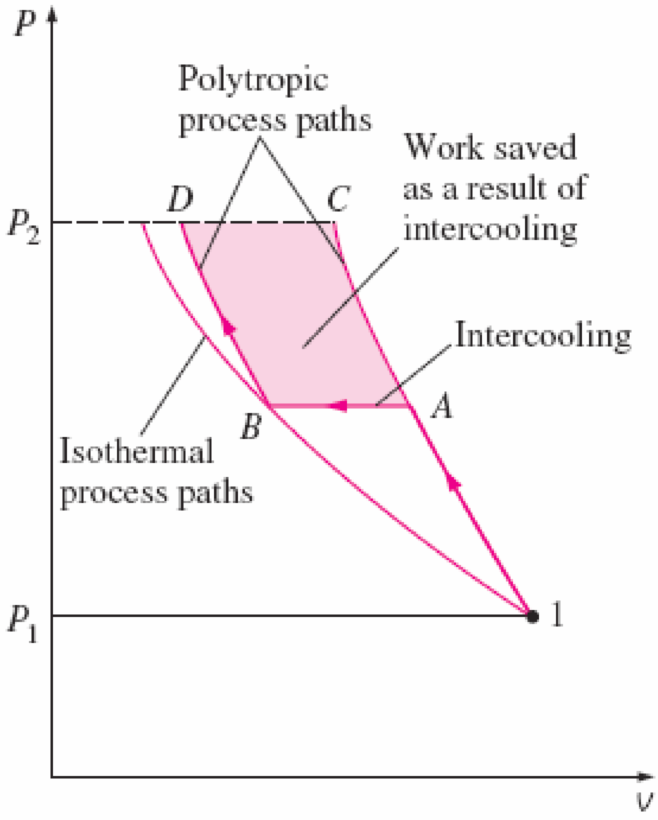
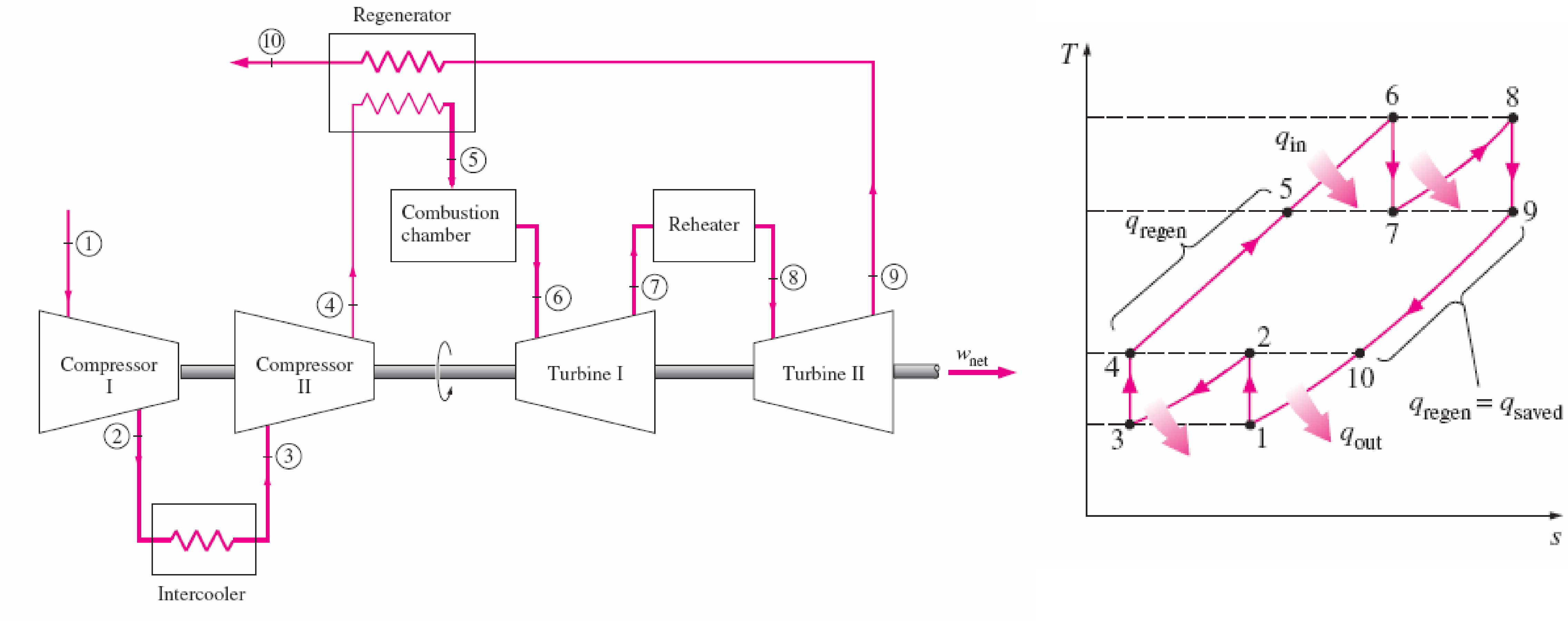


FIGURE 9–42

Comparison of work inputs to a single-stage compressor (1*AC*) and a two-stage compressor with intercooling (1*ABD*).

Brayton Cycle With Intercooling, Reheating, & Regeneration



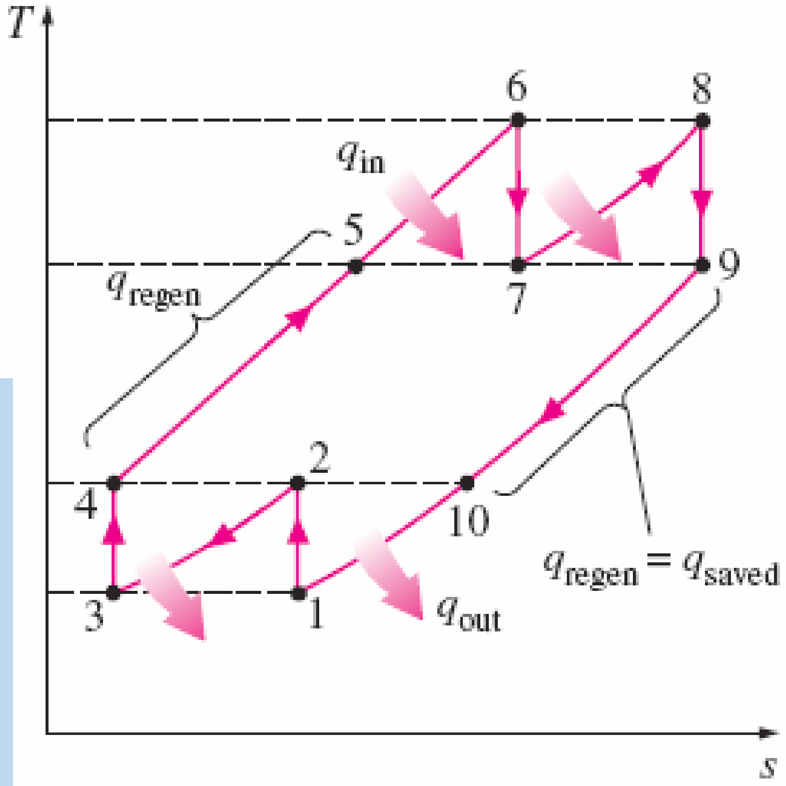
Physical arrangement of an **ideal two-stage** gas-turbine cycle with intercooling, reheating, and regeneration

Brayton Cycle With Intercooling, Reheating, & Regeneration

Conditions for Best Performance:

The work input to a two-stage compressor is **minimized** when equal pressure ratios are maintained across each stage. This procedure also maximizes the turbine work output. Thus, for best performance we have,

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9}$$



- Intercooling and reheating always **decreases** thermal efficiency unless are accompanied by **regeneration**.
- Therefore, in gas turbine power plants, intercooling and reheating are always **used in conjunction** with regeneration.

OPEN CYCLE GAS TURBINE Vs CLOSED CYCLE GAS TURBINE

Open cycle:

1. **Warm-up time.** Once the turbine is brought up to the rated speed by the starting motor and the fuel is ignited.

2. **Low weight and size.** The weight in kg per kW developed is less.
3. Open cycle plants ***occupy comparatively little space***.
4. Open-cycle gas turbine power plant, except those having an intercooler, ***does not require cooling water***.
5. The ***part load efficiency of the open cycle plant decreases*** rapidly as the considerable percentage of power developed by the turbine is used to drive the compressor.
6. The open-cycle gas turbine plant has high air rate compared to the other cycles, therefore, it results in increased loss of heat in the exhaust gases.

OPEN CYCLE GAS TURBINE Vs CLOSED CYCLE GAS TURBINE

Closed cycle:

1. The machine can be **smaller and cheaper** than the machine used to develop the same power using open cycle plant.

2. ***The closed cycle avoids erosion of the turbine blades*** due to the contaminated gases and fouling of compressor blades due to dust. Therefore, it is practically free from deterioration of efficiency in service.
3. The need for filtration of the incoming air which is a severe problem in open cycle plant is completely eliminated.
4. The ***maintenance cost is low*** and reliability is high due to longer useful life.
5. The system is dependent on external means as considerable quantity of cooling water is required in the pre-cooler.

6. The response to the load variations is poor compared to the open-cycle plant.

NUMERICAL ON GAS TURBINE

1. A gas turbine is supplied with gas at 5 bar and 1000 K and expands it adiabatically to 1 bar. The mean specific heat at constant pressure and constant volume are 1.0425 kJ/kg-K and 0.7662 kJ/kg-K respectively.

(i) Draw the *p-v* and *T-s* diagram to represent the simple gas turbine system.

(ii) Calculate the power developed (in kW/kg of gas per second) and

(iii) Exhaust gas temperature

Given Data:

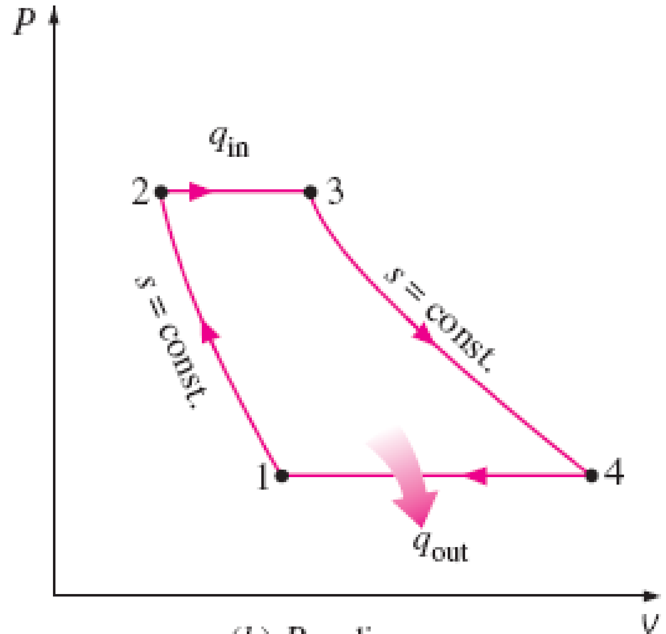
$$p_4 = 1 bar$$

$$p_3 = 5 bar$$

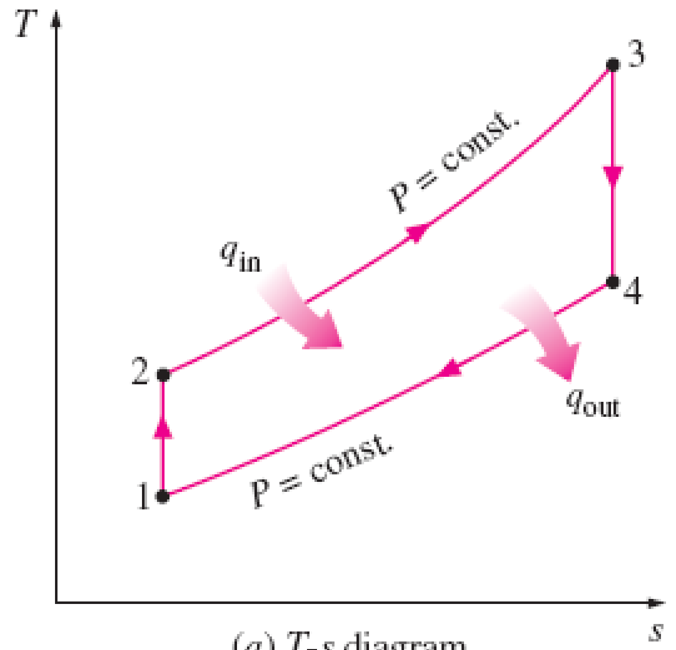
$$T_3 = 1000\,K$$

$$c_p = 1.0425\,kJ/kg-K$$

$$c_v = 0.7662\,kJ/kg-K$$



(*b*) *P-v* diagram



(*a*) *T-s* diagram

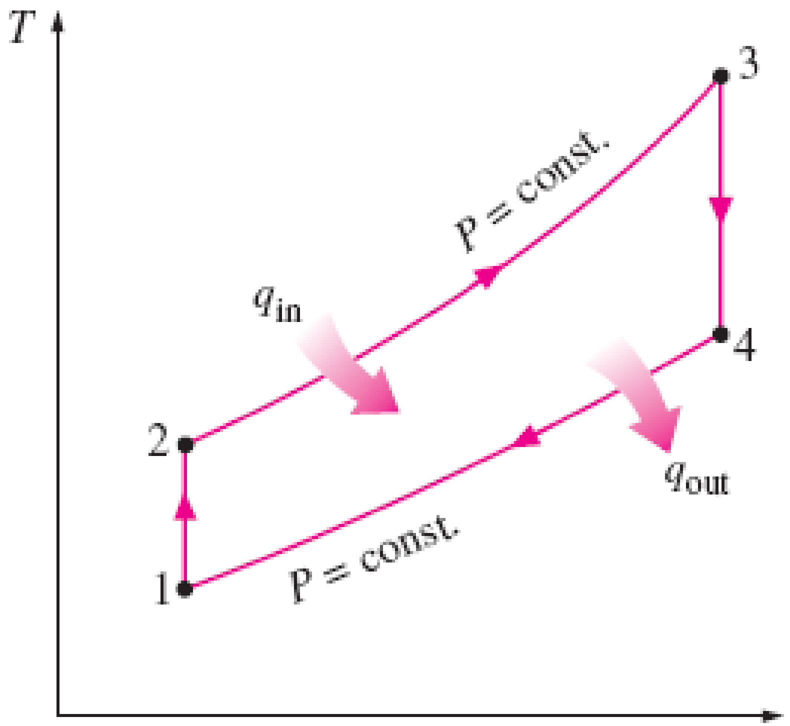
NUMERICAL ON GAS TURBINE

$$Formula:$$

$$(i)\ Power\ developed = c_p\,(T_3 - T_4)$$

$$(ii)\ \frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$where,\ \gamma = \frac{c_p}{c_v}$$



(a) T-s diagram

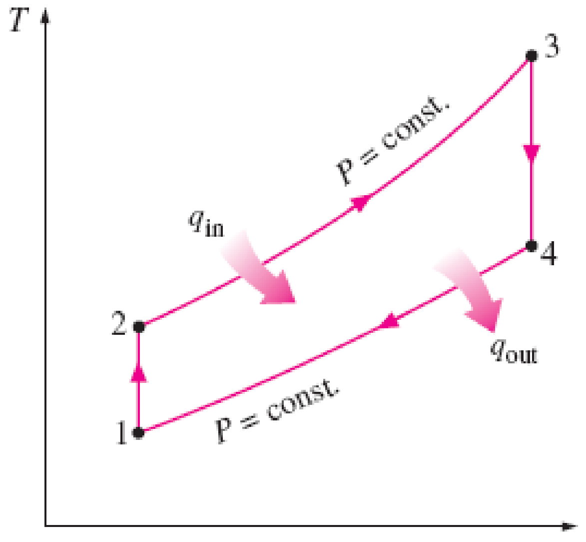
NUMERICAL ON GAS TURBINE

$Solution:$

$$(i)\ Power = c_p\,(T_3 - T_4) = 361.7\,kW/kg$$

$$(ii)\ T_4 = T_3\left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = 1000 \times 0.653 = 653\,K$$

$$where,\ \gamma = \frac{c_p}{c_v} = 1.36$$



(a) T-s diagram

NUMERICAL ON GAS TURBINE

2. Air enters the compressors of a gas turbine plant operating on Brayton cycle at 1 bar, 27°C. The pressure ratio in the cycle is 6. if W_T=2.5 W_C, where W_T and W_C are the turbine and compressor work respectively. Calculate the maximum temperature and the cycle efficiency.

Given Data:

$$p_1 = 1 bar$$

$$\frac{p_2}{p_1} = 6 bar$$

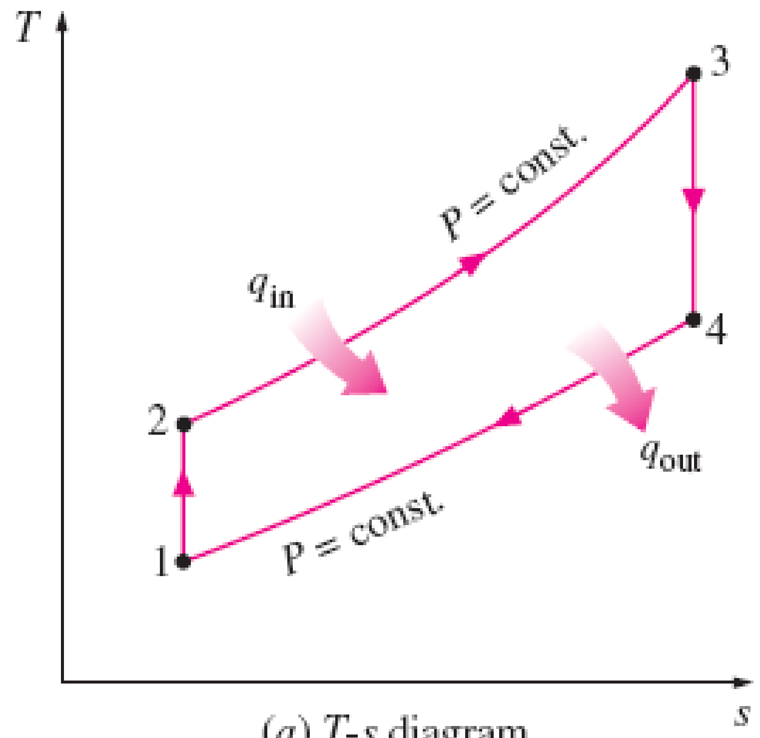
$$T_1 = 27°C = 300K$$

$$W_T = 2.5 W_C$$

$To\,Find:$

$(i) T_3$

$(ii) \eta_{cycle}$



(a) T-s diagram

NUMERICAL ON GAS TURBINE

Solution:

$1. T_3 ...?$

$$W_T = 2.5 \times W_C$$

$$(T_3 - T_4) = 2.5 \times (T_2 - T_1)$$

$$T_3 - \frac{T_3}{1.668} = 2.5 \times (500.4 - 300)$$

$$T_3 = 1251 K$$

where,

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

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4} \right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}}$$

NUMERICAL ON GAS TURBINE

$$Efficiency\ of\ cycle:$$

$$\eta_{cycle} = 1 - \frac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}}} \quad (\because \gamma = 1.4)$$

$$\eta_{cycle} = 40\%$$

NUMERICAL ON GAS TURBINE

3. Calculate the required air-fuel ratio in a gas turbine, whose turbine and compressor efficiencies are 85% and 80%, respectively. Maximum cycle temperature is 875°C. The working fluid can be taken as air (c_p=*1.0 kJ/kg-K*, γ=*1.4*) which enters the compressor at 1bar and 27 °C. the pressure ratio is 4. The calorific value of fuel is 42,000 kJ/kg. There is a loss of 10% calorific value in the combustion chamber.

$Given\,Data:$

$\eta_T = 85\%$

$\eta_c = 80\%$

$T_3 = 875°C + 273 = 1148K$

$T_1 = 27°C = 300K$

$c_p = 1.0\,kJ/kg-K$

$\gamma = 1.4$

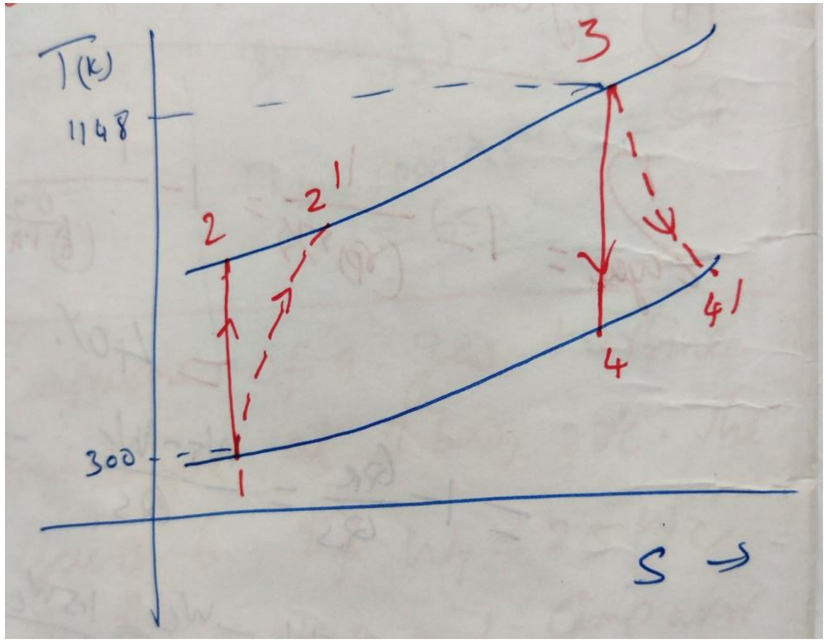
$p_1 = 1bar$

$p_2 = 4\,bar$

$C.V. = 42000\,kJ/kg.K$

$\eta_{cc} = 90\%$

$To\,Find: \dfrac{m_a}{m_f}$



NUMERICAL ON GAS TURBINE

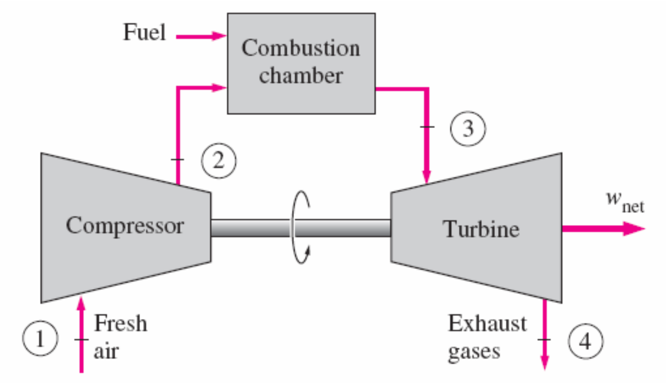
Energy Balance on Combustion Chamber

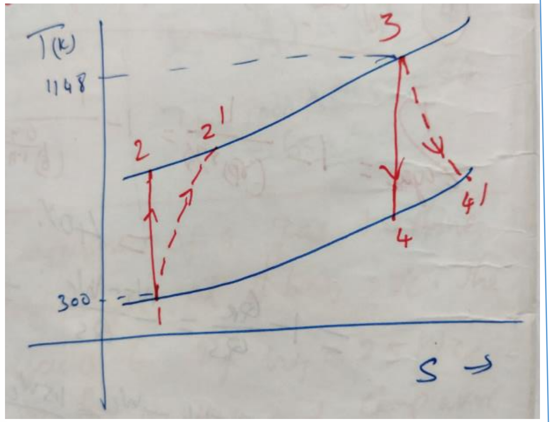
Heat utilized out of fuel supply = Heat Leaving the Combustion Chamber

$$\eta_{CC} \times m_f \times C.V. = \left(m_a + m_f\right) \times c_p \left(T_3 - T_2^1\right)$$

$$\div m_f \Rightarrow \quad \eta_{CC} \times C.V. = \left(\frac{m_a}{m_f} + 1\right) \times c_p \left(T_3 - T_2^1\right)$$

$$\frac{m_a}{m_f} = \ ?$$





$$To\ Find : T_2^1$$

$$\eta_{comp} = \frac{w_{isen}}{w_{act}} = \frac{T_2 - T_1}{T_2^1 - T_1}$$

$$where, T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2^1 = \ ?$$

NUMERICAL ON GAS TURBINE

$$To\,Find : T_2^1$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = 300 \times 1.486 = 445.8\,K$$

$$\eta_{comp} = \frac{w_{isen}}{w_{act}} = \frac{T_2 - T_1}{T_2^1 - T_1} \Rightarrow 0.8 = \frac{(445.8 - 300)}{(T_2^1 - 300)}$$

$$T_2^1 = 482.2\,K$$

NUMERICAL ON GAS TURBINE

Heat supplied by fuel = Heat carried away by Exhaust gas

$$0.9 \times C.V. = \left(\frac{m_a}{m_f} + 1 \right) \times c_p \left(T_3 - T_2^1 \right)$$

$$0.9 \times 42000 \times 10^3 = \left(\frac{m_a}{m_f} + 1 \right) \times 1 \times 10^3 (1148 - 482.2)$$

$$\frac{m_a}{m_f} = 56$$

NUMERICAL ON GAS TURBINE

4. In a closed cycle gas turbine there is two-stage compressor and a two-stage turbine. The pressure and temperature at the inlet of the first stage compressor are 1.5 bar and 20°C. The maximum cycle temperature and pressure are limited to 750°C and 6 bar. A perfect intercooler is used between the two compressor. Gases are heated in the reheater to 750°C before entering into the L.P. turbine.

Assuming the compressor and turbine efficiencies as 0.82, calculate

(i) The efficiency of the cycle without regenerator

(ii) The efficiency of the cycle with regenerator, whose effectiveness is 0.7.

(iii) The mass of the fluid circulated if the power developed by the plant is 350 kW.

The working fluid is air, $\gamma=1.4$ *and* $c_p=1.005$ *kJ/kg-K.*

NUMERICAL ON GAS TURBINE

4. In a closed cycle gas turbine there is two-stage compressor and a two-stage turbine. The pressure and temperature at the inlet of the first stage compressor are 1.5 bar and 20°C. The maximum cycle temperature and pressure are limited to 750°C and 6 bar. A perfect intercooler is used between the two compressor. Gases are heated in the reheater to 750°C before entering into the L.P. turbine.

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(iii) The mass of the fluid circulated if the power developed by the plant is 350 kW

The working fluid is air, $\gamma=1.4$ *and* $c_p=1.005\ kJ/kg\text{-}K$

$Given\ Data:$

$T_1 = 20°C = 293K$

$P_s = P_1 = 1.5\,bar$

$T_5 = T_7 = 750°C = 1023K$

$P_d = P_4 = P_5 = 6\,bar$

$\eta_T = \eta_c = 0.82$

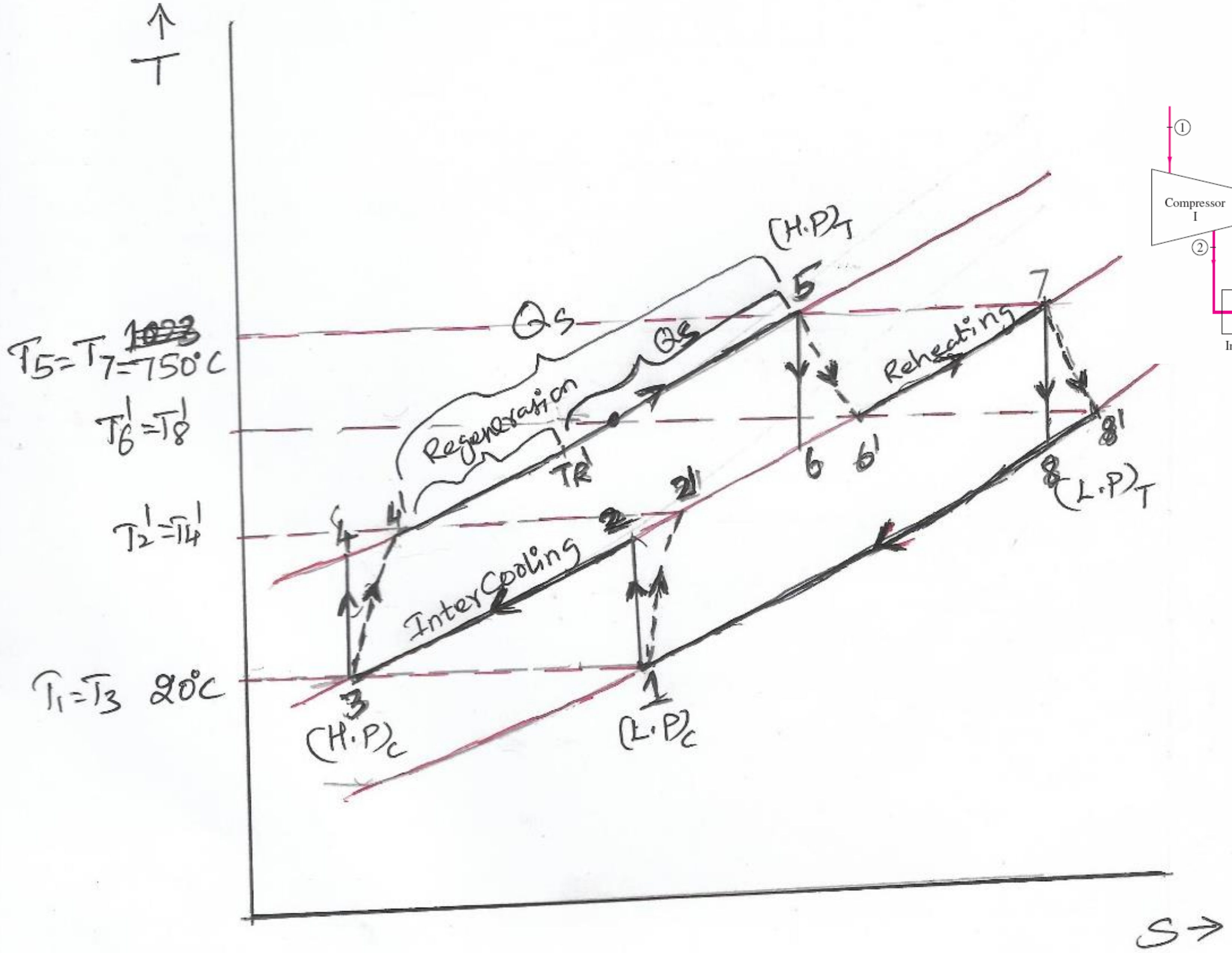
$\varepsilon = 0.7$

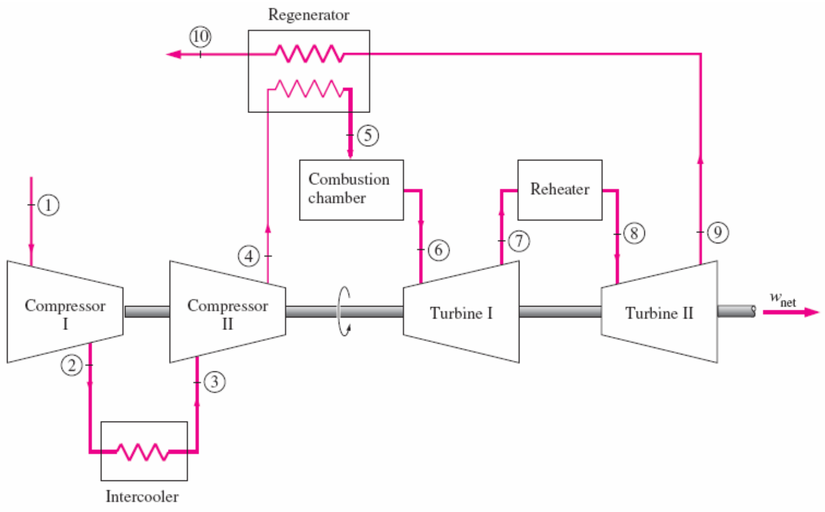
$P = 350kW$

$For\ air,$

$c_p = 1.005\,kJ/kg - K$

$\gamma = 1.4$


T
S
$T_5 = T_7 = \cancel{1023}$ 750°C
$T_6' = T_8'$
$T_2' = T_4'$
$T_1 = T_3$ 20°C
(H.P)T
Qs
Qs
Regeneration
Reheating
InterCooling
TR'
5
6
6'
7
8
8'
(L.P)T
4
4'
2
2'
3
(H.P)C
1
(L.P)C


Regenerator
10
5
Combustion chamber
Reheater
1
4
6
7
8
9
Compressor I
Compressor II
Turbine I
Turbine II
w_{net}
2
3
Intercooler

NUMERICAL ON GAS TURBINE

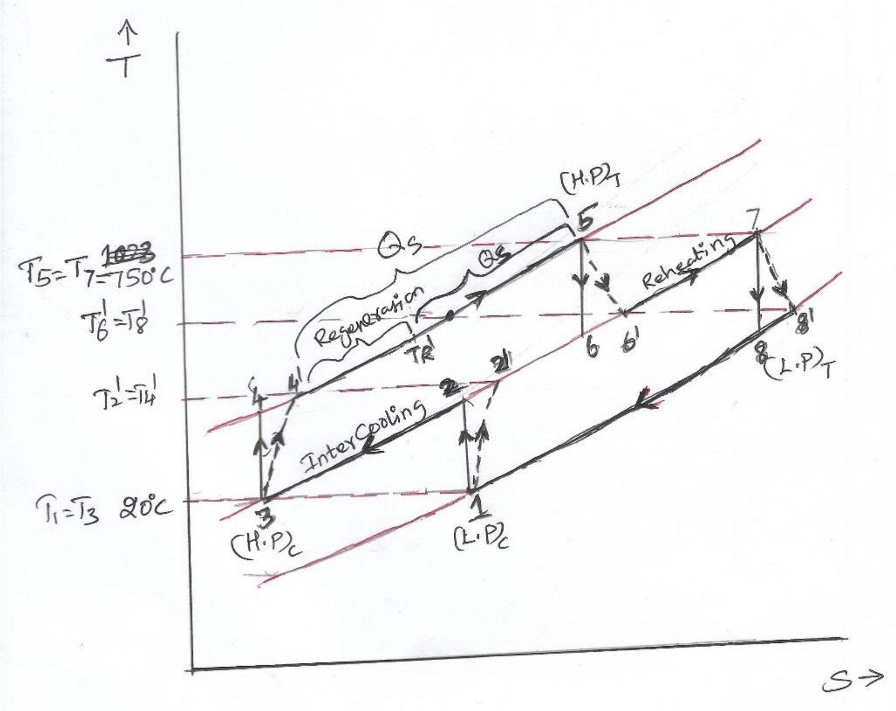
Formula,

$$1.\ \eta = \frac{W_{net}}{Q_s}$$

$$(i)\ W_{net} = \left(W_{T-HP} + W_{T-LP}\right) - \left(W_{C-HP} + W_{C-LP}\right)$$

$$= 2\left[W_{T-LP} - W_{C-LP}\right]$$

$$= 2 \times c_p\left[\left(T_5 - T_6^1\right) - \left(T_2^1 - T_1\right)\right]$$



(ii) Heat supplied per kg of air without regenerator:

$$Q_s = c_p\left[\left(T_5 - T_4^1\right) + \left(T_7 - T_6^1\right)\right]$$

(iii) Heat supplied per kg of air with regenerator:

$$Q_s = c_p\left[\left(T_5 - T_R^1\right) + \left(T_7 - T_6^1\right)\right]$$

2. Mass of fluid

$$P = W_{net} \times \dot{m}$$

NUMERICAL ON GAS TURBINE

$$Unknowns: (T_2^1 = T_4^1), T_6^1, T_R^1$$

$$To\ find: T_2^1$$

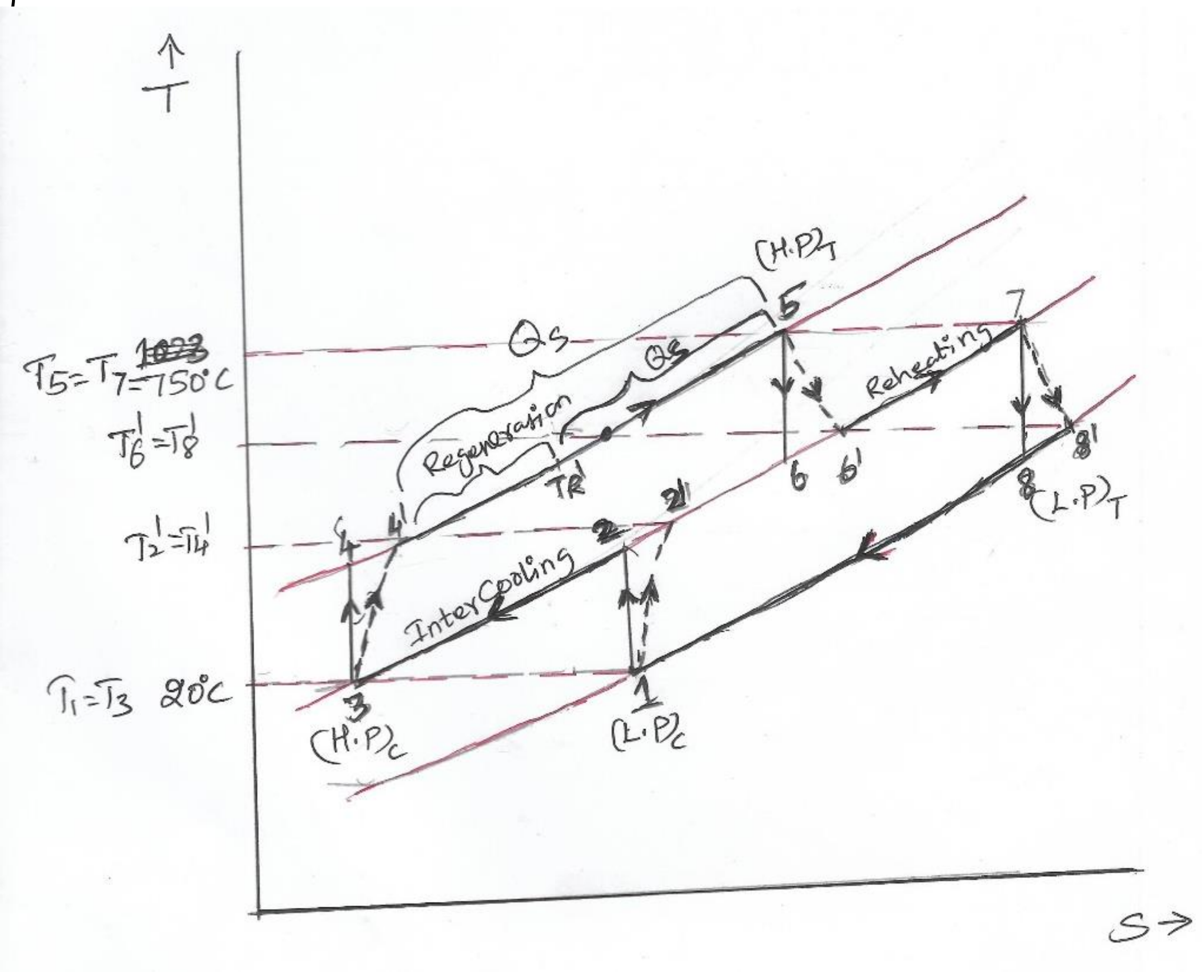
$$\eta_{comp} = \frac{W_{isentropic}}{W_{Actual}} = \frac{c_p\left(T_2 - T_1\right)}{c_p\left(T_2^1 - T_1\right)}$$

$$T_2^1 = \ ?$$

where,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad and \quad P_2 = \sqrt{P_s P_d}$$

$$T_2 = \ ?$$



NUMERICAL ON GAS TURBINE

$$To\ find : T_6^1$$

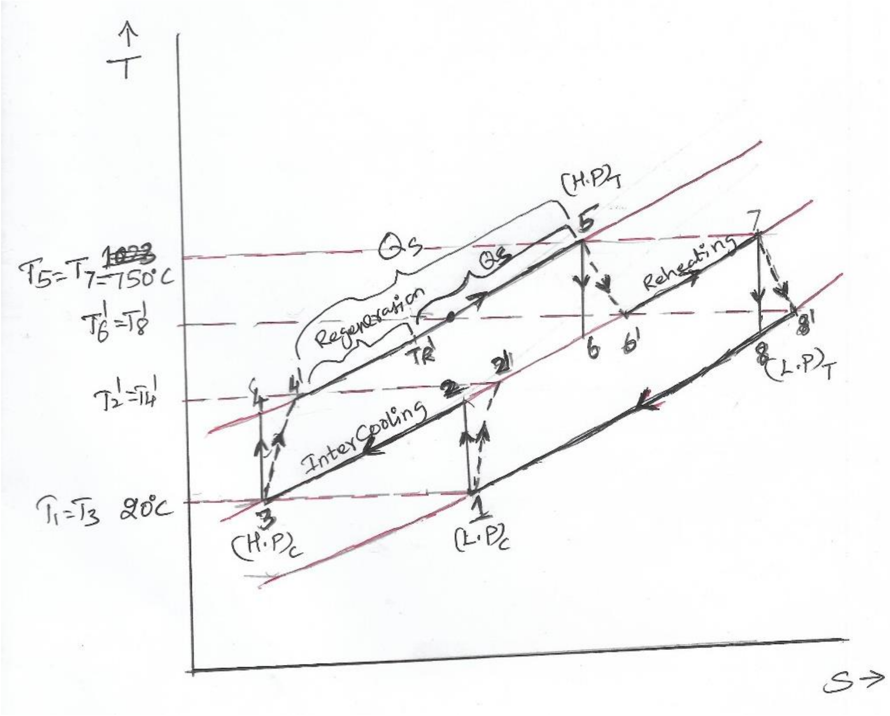
$$\eta_{turbine} = \frac{W_{Actual}}{W_{isentropic}} = \frac{c_p\left(T_5 - T_6^1\right)}{c_p\left(T_5 - T_6\right)}$$

$$T_6^1 = \ ?$$

where,

$$\frac{T_5}{T_6} = \left(\frac{P_5}{P_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_d}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_6 = \quad ?$$



NUMERICAL ON GAS TURBINE

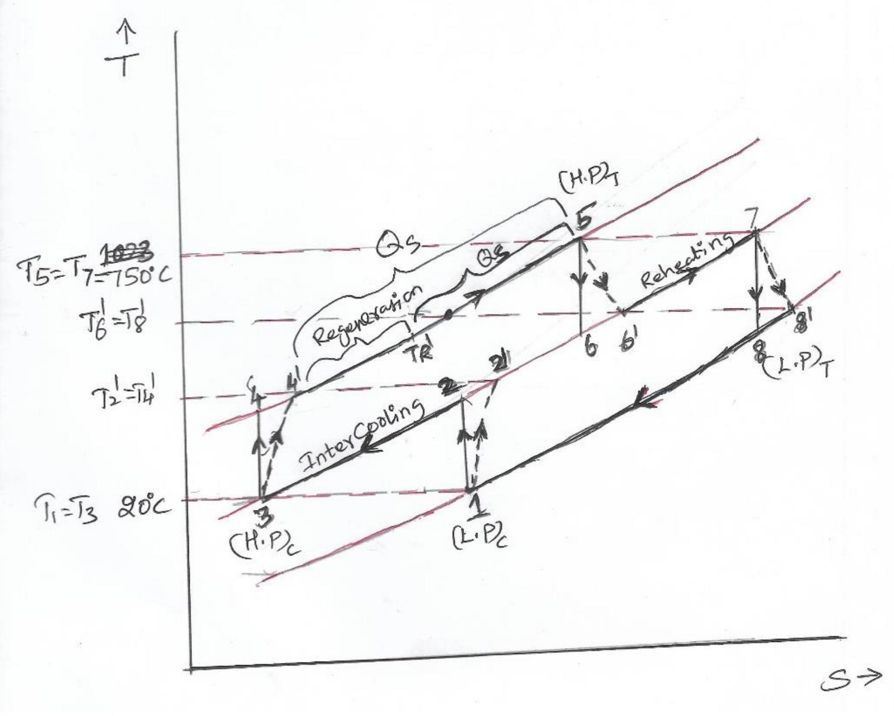
$$To\ find : T_R^1\ (Temp. of\ air\ comes\ out\ of\ regenerator)$$

$$\varepsilon = \frac{\left(T_R^1 - T_4^1\right)}{\left(T_8^1 - T_4^1\right)}$$

$$T_R^1 = \ ?$$

where,

$$T_6^1 = T_8^1$$



NUMERICAL ON GAS TURBINE

Answers:

$$T_2 = 357\,K$$

$$T_2^1 = 371\,K = T_4^1$$

$$T_6 = 839\,K$$

$$T_6^1 = 872\,K = T_8^1$$

$$T_R^1 = 722\,K$$

$$(i) W_{net} = 146.73\,kJ/kg\ of\ air$$

$$(ii) Q_s\,(without\ regenerator) = 807\,kJ/kg\ of\ air$$

$$(iii) Q_s\,(with\ regenerator) = 454.3\,kJ/kg\ of\ air$$

$$1.\ \eta\,(without\ regenerator) = \frac{W_{net}}{Q_s} = 18.2\%$$

$$2.\ \eta\,(with\ regenerator) = \frac{W_{net}}{Q_s} = 32.3\%$$

$$3.\ Mass\ of\ Fluid\ circulated:$$

$$\dot{m} = \frac{P}{W_{net}} = \frac{350\,(kW = kJ/s)}{146.73(kJ/kg)} = 2.38\,kg/s$$

NUMERICAL ON GAS TURBINE

5. A stationary gas-turbine power plant operates on an ideal Brayton cycle with air as the working fluid. The air enters the compressor at 95 kPa and 290 K and the turbine at 760 kPa and 1100 K. Heat is transferred to air at a rate of 35,000 kJ/s. Determine the power delivered by this plant assuming constant specific heats at room temperature.

NUMERICAL ON GAS TURBINE

5. A stationary gas-turbine power plant operates on an ideal Brayton cycle with air as the working fluid. The air enters the compressor at 95 kPa and 290 K and the turbine at 760 kPa and 1100 K. Heat is transferred to air at a rate of 35,000 kJ/s. Determine the power delivered by this plant assuming constant specific heats at room temperature.

Given Data :

$$T_1 = 290 K$$

$$P_s = P_1 = 95\, kPa$$

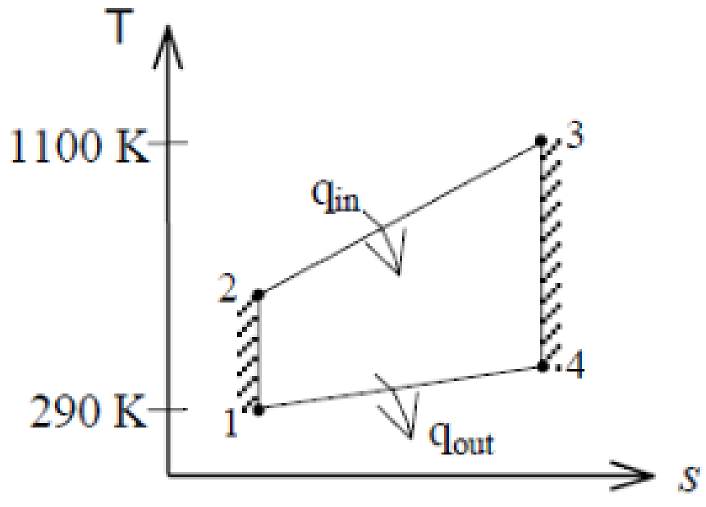
$$P_3 = 760\, kPa$$

$$T_3 = 1100 K$$

$$Q_{in} = 35000\, kJ / s$$

Formula,

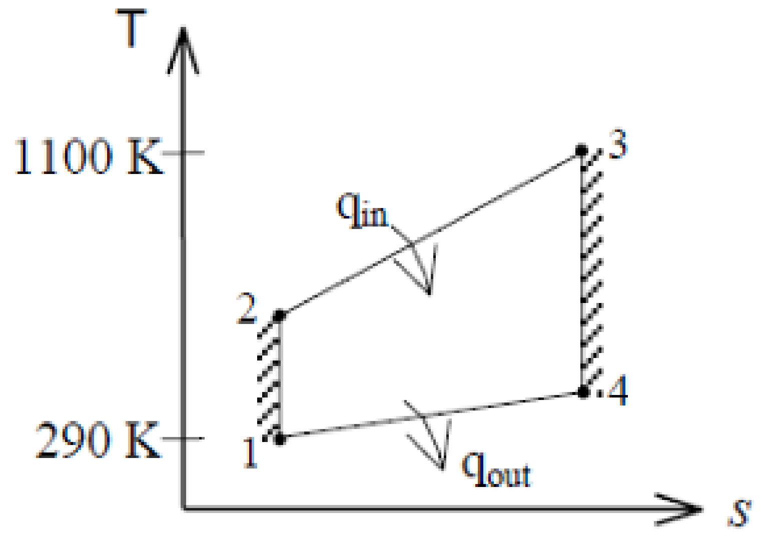
$$\eta = \frac{W_{net}}{Q_s} = \frac{Q_s - Q_{out}}{Q_s} = 1 - \frac{Q_{out}}{Q_s}$$



NUMERICAL ON GAS TURBINE

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kineti and potential energy changes are negligible. **4** Air is an ideal gas.

Analysis (*a*) Assuming constant specific heats,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (290\text{ K})(8)^{0.4/1.4} = 525.3\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1100\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 607.2\text{ K}$$

$$\eta_{\text{th}} = 1-\frac{q_{\text{out}}}{q_{\text{in}}} = 1-\frac{c_p(T_4-T_1)}{c_p(T_3-T_2)} = 1-\frac{T_4-T_1}{T_3-T_2} = 1-\frac{607.2-290}{1100-525.3} = 0.448$$

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_{\text{in}} = (0.448)(35{,}000\text{ kW}) = \mathbf{15{,}680\text{ kW}}$$

NUMERICAL ON GAS TURBINE

4. In a closed cycle gas turbine there is two-stage compressor and a two-stage turbine. The pressure and temperature at the inlet of the first stage compressor are 1 bar and 30°C. The maximum cycle temperature and pressure are limited to A°C and 'B' bar. A perfect intercooler is used between the two compressor. Gases are heated in the reheater to A°C before entering into the L.P. turbine.

Assuming the compressor and turbine efficiencies as 0.85, calculate

(i) The efficiency of the cycle without regenerator

(ii) The efficiency of the cycle with regenerator, (effectiveness of regenerator is 0.75)

(iii) The mass of the fluid circulated if the power developed by the plant is 330 kW

The working fluid is air, $\gamma = 1.4$ *and* $c_p = 1.005$ *kJ/kg-K*