# POWER PLANT ENGG

550°C

Steam Power Plant

Rankine Vapour Power cycle

Working fluid = Steam

9000 : 10000rpm

1300 K

Gas Turbine Power Plant

Joule or Brayton cycle

25000rpm

2500°C

I/C Engine Power Plant

Otto, Diesel, Dual

Working fluid = GAS

Hydroelectric Power Plant

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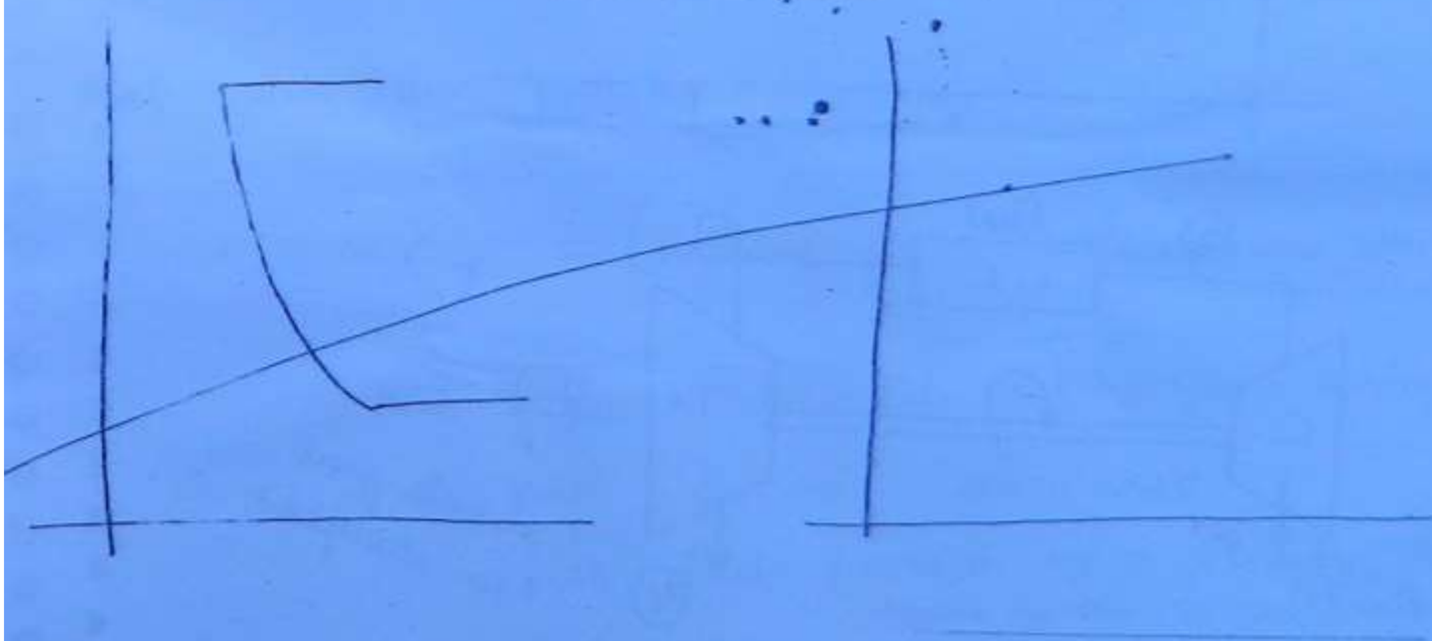
# GAS POWER CYCLE

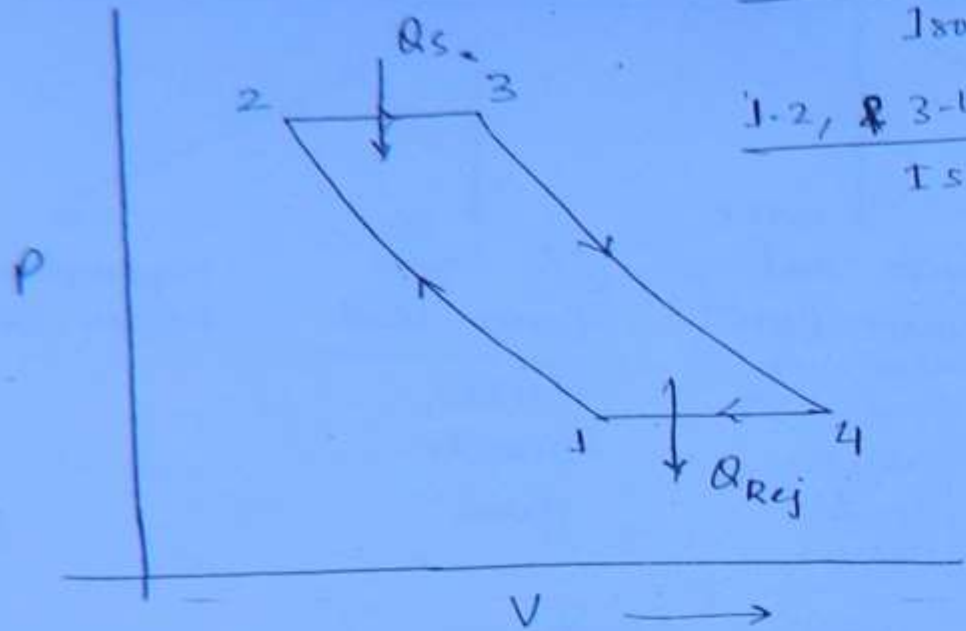
## BRAYTON CYCLE or IDEAL JOULE CYCLE

(Turbo jets)



1-4 & 2-3,
Isobaric Process

1-2, & 3-4,
Isentropic Process
(Reversible adiabatic)

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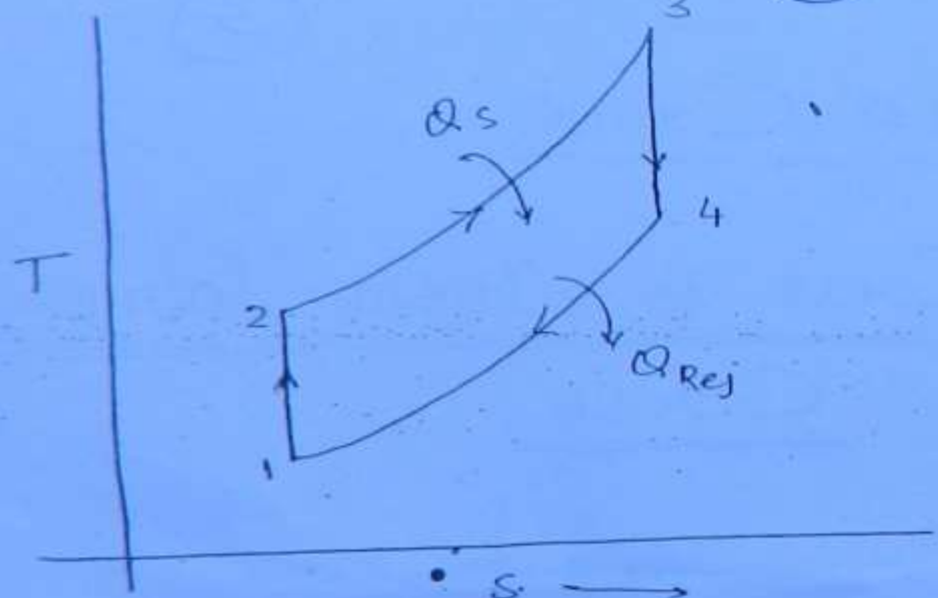


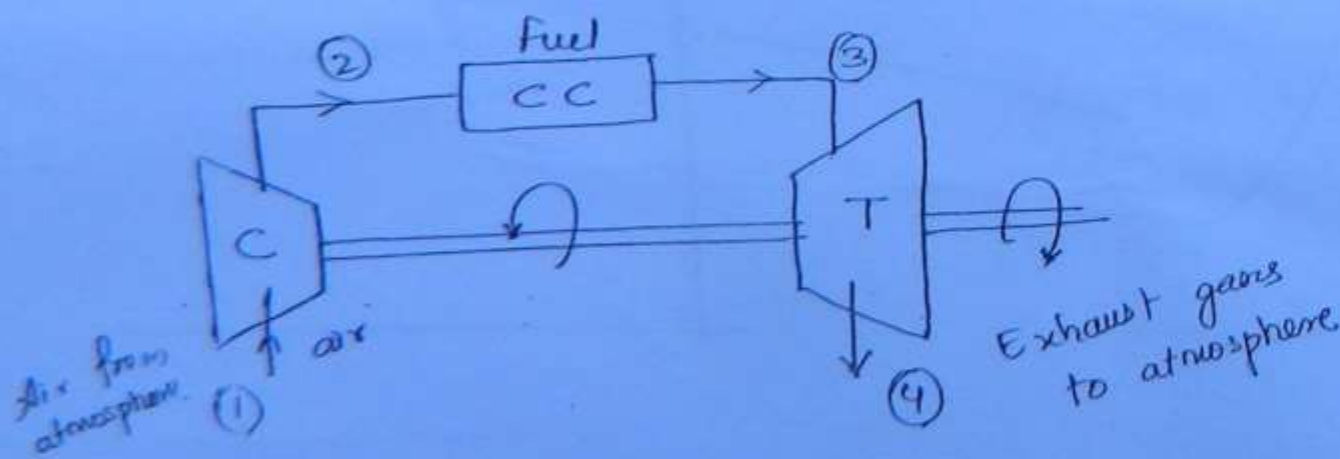

$\therefore$ from (1) & (2)

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \Rightarrow \boxed{\frac{T_4}{T_1} = \frac{T_3}{T_2}}$$

$$\eta_{th} = 1 - \frac{T_1\cancel{\left(\frac{T_4}{T_1} - 1\right)}}{T_2\cancel{\left(\frac{T_3}{T_2} - 1\right)}}$$ (as above)

$$\boxed{\eta_{th} = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}}$$ (5)

Note:- As the pressure ratio of the Brayton Cycle, increases, the Ideal Air Standard Efficiency of the brayton cycle increases

Net work done per kg —

$$\boxed{\text{Net W.D}/kg = (\text{Turbine work}) - (\text{compressor Work})}$$

$$\text{Turbine Work} = \Delta h = (h_3 - h_4)\ KJ/kg$$

$\left[ \because TdS = dh - vdP, \text{ if } ds = 0 \quad dh = vdP \right.$

$\therefore \int dh = \int vdP \; \{= \text{W.D by a steady flo open system in rev. process}$

Compressor Work $= \Delta h$

For Ideal gas, $\Delta h = C_p \cdot \Delta T$

$\therefore$ (TW) Turbine work $= C_{p_g}(T_3 - T_4)$ KJ/kg.

(CW) Compressor Work $= C_{p_a}(T_2 - T_1)$ KJ/kg

$$\text{Net W.D/kg} = TW - CW$$

$$\boxed{\text{Net W.D./kg} = C_{p_g}(T_3 - T_4) - C_{p_a}(T_2 - T_1)}$$

$C_{p_g} > C_{p_a} = 1.005$ KJ/kg K

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WORK RATIO (WR)

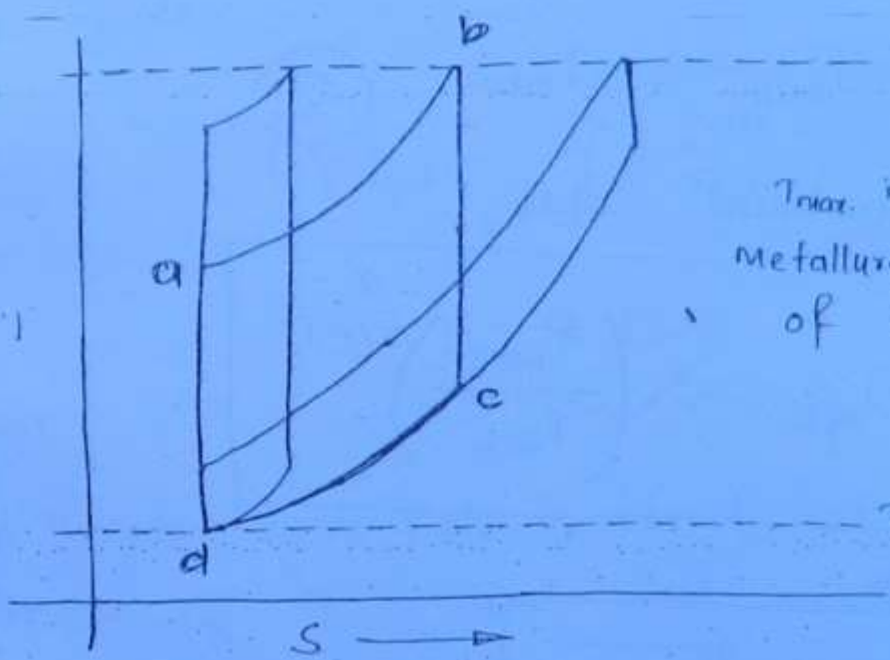
It is defined as net work done per kg and the turbine work done per kg

$$\boxed{WR = \frac{T.W. - C.W.}{T.W.}}$$

$$\boxed{WR = \frac{C_{p_g}(T_3 - T_4) - C_{p_a}(T_2 - T_1)}{C_{p_g}(T_3 - T_4)}}$$

Work ratio also depends on pressure ratio ($r_p$) of the cycle. As the pressure ratio increases work ratio also increase, reaches a maximum and then decreases

## Effect of ~~Work~~ Pressure Ratio on net work done

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$T_{max}$ is based on Max. Metallurgical tolerable limit of Gas Turbines blades

$T_{min}$ = atm. temp

For a given maximum and minimum tempreatures of the cycle, there is a certain pressure ratio known as Optimum Pressure Ratio at which specific work output of cycle or its work ratio is maximum. Similarly, for a given $T_{max}$. and $T_{min}$, the thermal efficiency of the Ideal joule cycle, shall be maximum, when,

$$\boxed{r_p = (r_p)_{max}}$$

$$\boxed{r_p = (r_p)_{max} = \left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{\gamma-1}}}$$

But for a given $T_{max}$ & $T_{min}$ 8

W.R. shall be maximum, when,

$$\boxed{r_p = (r_p)_{opt} = \left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{2(\gamma-1)}}}$$

$$\boxed{(r_p)_{opt} = \sqrt{(r_p)_{max}}}$$

Note: Value of '$\gamma$' —

for Helium — $\gamma = 1.67$

$\gamma_{diatomic} = 1.4$

$\gamma_{monoatomic} = 1.3$

if. $T_{max} = 1100$, $T_{min} = 300$, $\gamma = 1.4$

$$\therefore \; (r_p)_{max.} = 9.439$$

$$\text{and } (r_p)_{opt} = 9.71$$

Gate 2000

Q.) In an ideal air standard, Gas Turbine cycle, the minimum and maximum temperatures are respectively 310 K and 1100 K. Cal. optimal pressure ratio of the cycle.

Solⁿ

$$(r_p)_{opt} = \sqrt{\left(\frac{T_{max}}{T_{min}}\right)^{\frac{r}{r-1}}} \quad (r = 1.4)$$

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$$= 9.17$$

Gate 2001

Q.) A Brayton cycle (air standard) has a pressure ratio of 4 and inlet condition of 1 std. atm pressure and 27 °C. Find the air flow rate required for 100 kW power output if the max. temp. in cycle is 1000 °C. Assume $r = 1.4$, $C_p = 1$ kJ/kg K

Solⁿ

$$\frac{P_2}{P_1} = \frac{P_3}{P_4} = 4 \qquad r_p = 4$$

$$\therefore \; T_2 = T_1 \cdot (r_p)^{\frac{r-1}{r}} =$$

Sol$^n$

$T_{max}\ (T_3) = 1000^\circ C = 1273\ K$

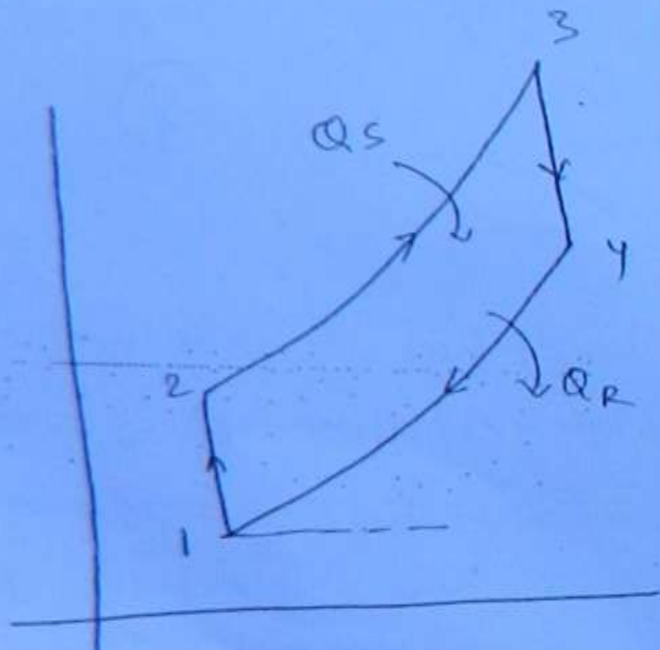
$T_{min}\ (T_1) = 27^\circ C = 300\ K$

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

$$= 300\ (4)^{\frac{1.4-1}{4}}$$

$$= 445.7\ K$$

and $T_4 = 856.67\ K$




TW = 416.3

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Turbine Work $= C_{p_a} (T_3 - T_4)$

$= 416.33\ kJ/kg$

Compressor Work $= C_{p_g} (T_2 - T_1)$

$= 145.7\ kJ/kg$

$\therefore$ Net WD/kg = TW − CW

= 270.6 kJ/kg

$\therefore$ Mass flow rate

$\therefore$ Power Output = $\dot{m}$ × Net. W.D./kg

$\therefore \dot{m} = \frac{100}{270.6}$

$\boxed{\dot{m} = 0.369 \text{ kg/sec.}}$

(11)

GATE 1999

Q.) An isentropic air turbine is used to supplied 0.1 kg/sec. of air at 0.1 MN/m² and at 285 K to a cabin. The pressure at inlet to the turbine is 0.4 MN/m². Determine the temperature at turbine inlet and the power developed.

Soln

$P_1 = P_4 = 400$ Pa. ,

$P_2 = P_3 = 100$ Pa , $T_3 = 285$ K

$\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4} \quad \therefore T_4 = 423.5$ K

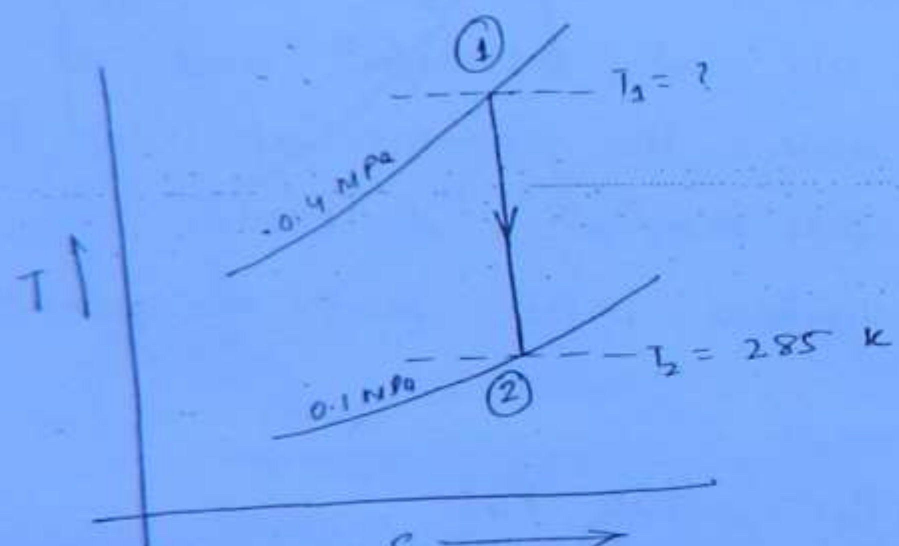
$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1}$$

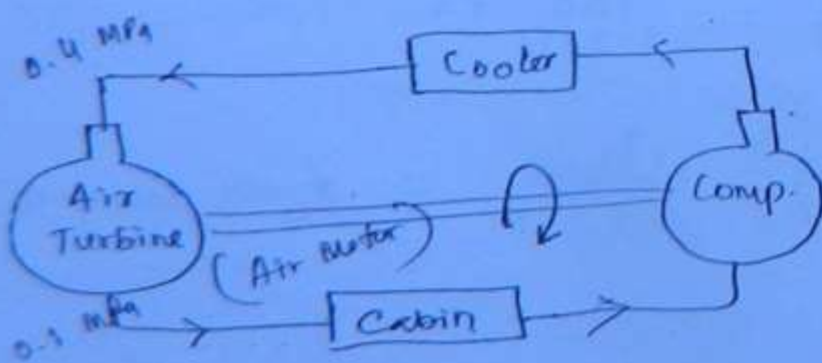
The above given turbine is a part of air craft refrigeration system, running on Bel- Colemmon Cycle (Reversed brayton)

Now, $r_p = 0.25$

(12)

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$



$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \qquad (T_2 = 285\ K)$$

$$T_1 = T_2 \left(\frac{400}{100}\right)^{\frac{1.4-1}{1.4}}$$

$$= 423.7\ K$$

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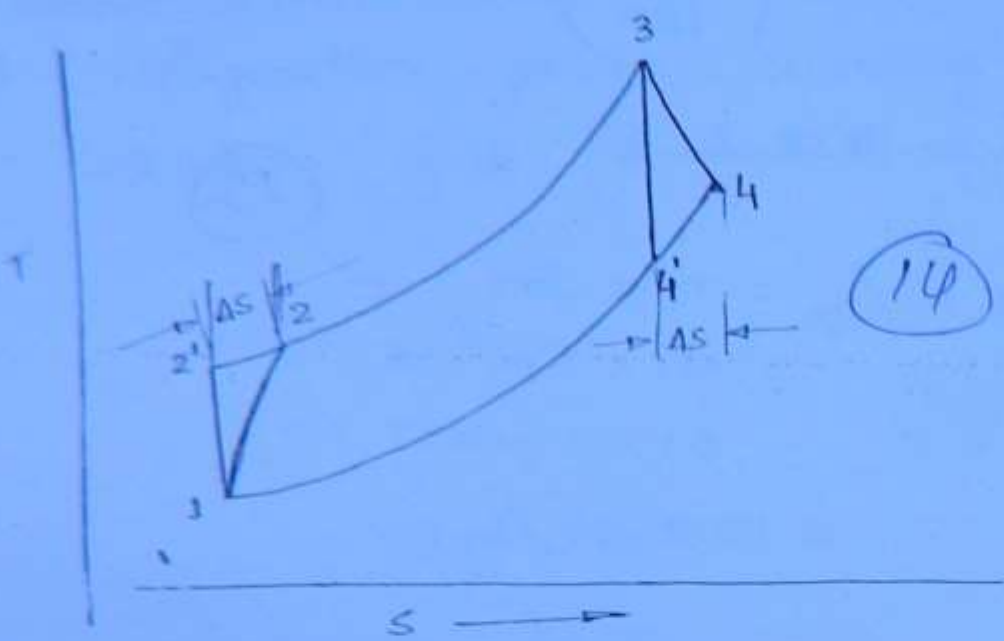
$$\therefore \text{Net WD/sec} = \dot{m} \times \Delta h$$

$$= \dot{m} \times C_p \left(T_{\cancel{2}} - T_1\right)$$

$$= 0.1 \times 1 \left(423.7 - 285\right)$$

$$= 13.85\ kJ/kg\ sec.$$

# Gas Turbine Cycle With Machine Efficiency




$1-2' \Rightarrow$ Reversible adiabatic or Isentropic Compression.

$1-2 \Rightarrow$ Actual compression with friction.

Isoentropic efficieny of the compressor is defined as the ratio between isoentropic work input to the compressor and actual work input

$$\eta_{(iso)\,compressor} = \frac{\text{Isentropic Work}}{\text{Actual work}}$$

$$\boxed{\eta_{\substack{ISO \\ (compressor)}} = \frac{\cancel{C_{p_a}}(T_2' - T_1)}{\cancel{C_{p_a}}(T_2 - T_1)}} \quad (\simeq 80-85\%)$$

Isentropic efficiency of the turbine is defined as the ratio between actual work output of the turbine with friction and the isentropic work output of the turbine without friction.

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$$\eta_{\substack{(ISO) \\ Turbine}} = \frac{\text{Actual Work}}{\text{Isentropic work output}} = \frac{\cancel{C_{p_g}}(T_3 - T_4)}{\cancel{C_{p_g}}(T_3 - T_4')}$$

$$\boxed{\eta_{\substack{(ISO) \\ turbine}} = \frac{T_3 - T_4}{T_3 - T_4'}} \quad (\cong 80-85\%)$$

Note:

(Steam Engine running on modified Rankine cycle)

Oldest Engine

Gate 1998

Q) Find the required air fuel ratio in a gas turbine whos turbine and compressor efficieny are 85% and 80%. Max. cycle temp. is 875°. The working fluid can be taken as air. $C_p = 1$ kJ/kg K, $\gamma = 1.4$, which enter the compressor at 1 atm and 27°C. The pressure ratio is 4. The fuel used has calorific value of 42000 kJ/kg. There is a loss of 10% of calorific value in combustion chamber.

Sol^n

$$T_3 = 875°, \quad \eta_{comp} = 80\% \quad \eta_{turb} = 85\%$$

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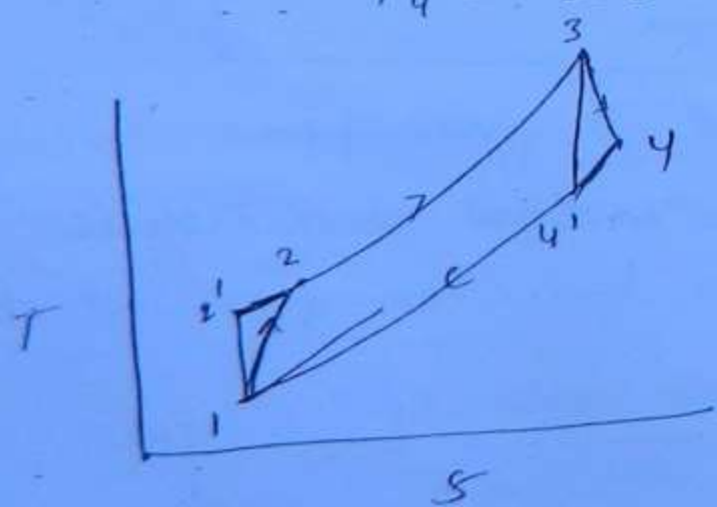
$$\frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} \qquad T_3 = 1148$$

$$T_4' = \cancel{588.8}\ 772.5 \text{ K}$$



$T_1 = 300 \text{ K}$

$$\therefore \frac{T_2'}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore T_2' = 445.79 \text{ K}.$$

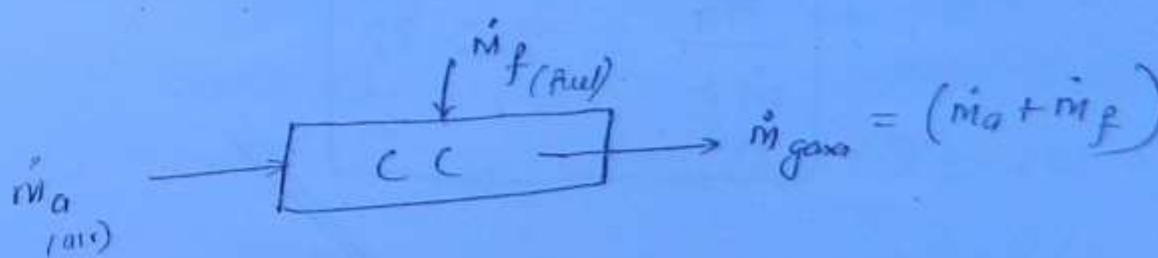
$$\eta_{comp.} = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\boxed{T_2 = 482.25 \text{ K}}$$

$$\eta_{turb} = \frac{T_3 - T_4}{T_3 - T_4'}$$ 17

$$\boxed{T_4 = 828.83 \text{ K}}$$

for combustion chamber (CC)



Energy balance

Heat liberated due to combustion of fuel

= Rate of change of Enthalpy of gases in CC.

$$\dot{m}_f \times L.C.V = (\dot{M}_a + \dot{m}_f)\, c_{pg}\,(T_3 - T_2)$$

$$L.C.V = \left(\frac{\dot{m}_a}{\dot{m}_f} + 1\right) c_{pg}\,(T_3 - T_2)$$

∴ Efficiency of combustion is 90% (due to 10% loss)

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$$\Rightarrow \quad 0.9 \times LCV = \left(\frac{\dot{m}_a}{m_f} + 1\right) c_{pg}\,(T_3 - T_2)$$

$$\Rightarrow \quad 0.9 \times 42000 = \left(\frac{\dot{m}_a}{\dot{m}_f} + 1\right) 1\,(1148 - 482.25)$$

$$\therefore \boxed{\frac{\dot{m}_a}{\dot{m}_f} = 55.77}$$

Q) Air enters the compressor of gas turbine plant operating on brayton cycle at 1 bar, 27°C. The pressure ratio in cycle is 6 if $w_T = 2.5 w_c$ where, $w_T$ & $w_c$ are turbine & compressor work respectively. Cal$^c$ the max. temperature and the cycle efficiency.

Sol$^n$

$$W_T = 2.5 W_C$$

$P_1 = 1$ bar
$= 100$ Pa

$T_1 = 300$ K

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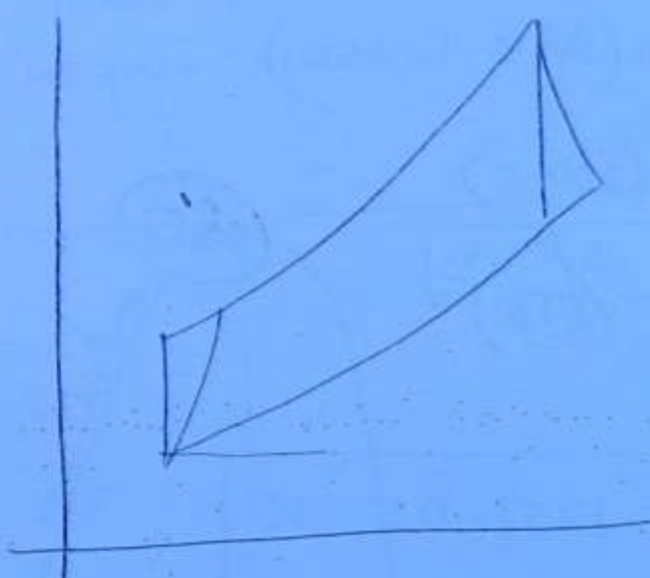

$$\frac{T_2}{T_1} = 6^{\frac{0.4}{1.4}} \Rightarrow T_2 = 500.55 \text{ K}$$

$$W_C = C_p (T_2 - T_1) = 200.55 \text{ kJ}$$

$$W_T = 501.38 \text{ kJ/kg}$$

$$r_p = \cancel{\sqrt{}} \left(\frac{T_{max}}{T_{min}}\right)^{\frac{r}{r-1}}$$

$$\eta = 1 - \frac{T_1}{T_2} \text{ or } 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}$$

$$\therefore \eta = 40\%$$

$$W_T = 2.5\, W_C$$

$$\cancel{C_p}(T_3 - T_4) = 2.5\cancel{C_p}(T_2 - T_1)$$

$$T_3\left(1 - \frac{T_4}{T_3}\right) = 2.5\,(500.5 - 300)$$

$$T_3 = \frac{2.5\,(200.5)}{1 - 1/(r_p)^{\frac{\gamma-1}{\gamma}}}$$

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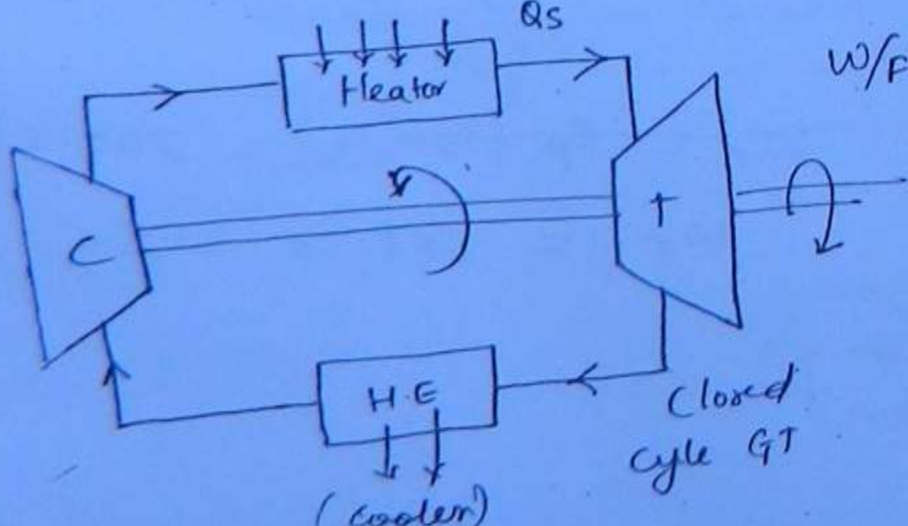
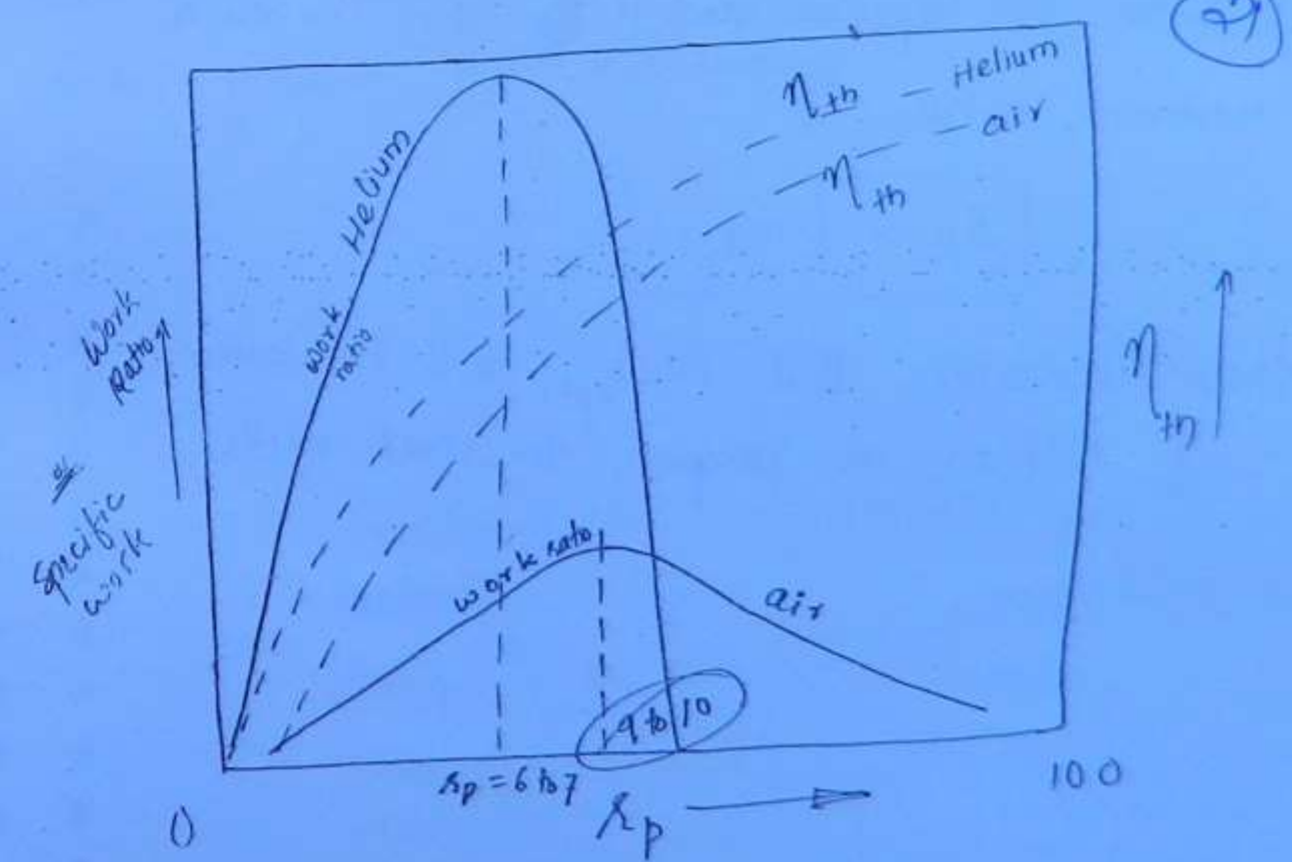
$$\boxed{T_3 = 1251.05 \text{ K}}$$




$W/F$ = Helium
$\gamma = 1.66$

In closed cycle gas turbine power plant, the chemical composition of working fluid will not change because heat is supplied to it ~~from~~ in the heater, after having combustion of the fuel outside the heater (which means fuel) Also that, working fluid may be circulating in the system with higher pressure.

Since helium has better thermodynamic properties than air i.e. higher value 'γ', it can give higher thermal efficiencies.




(For given $T_{max}$ & $T_{min}$.)
(1100 K & 300 K)

For a given $T_{min.}$ & $T_{max.}$ in the cycle, as the pressure ratio $r_p$ increases, the thermal efficiency of the cycle increases either case of Helium or air, reaches a maximum, when,

$$\boxed{r_p = (r_p)_{max.}}$$

At any $r_p$ (Pressure Ratio),

$$\boxed{\eta_{He} > \eta_{air}}$$

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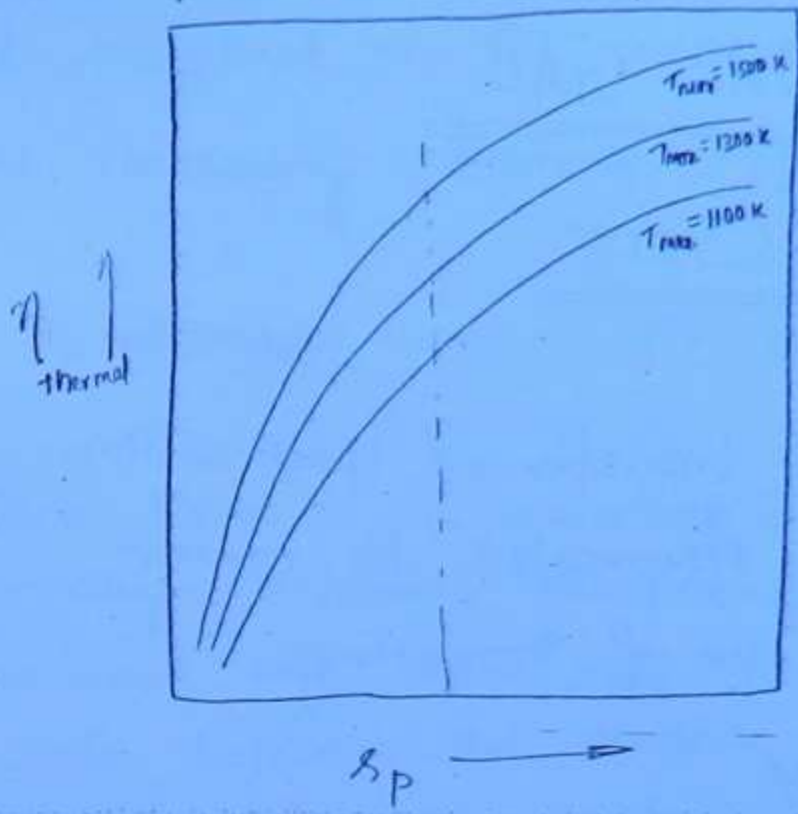
For work ratio, as $r_p$ increases, it also increases, reaches maximum, when,

$$\boxed{r_p = (r_p)_{opt}}$$

and then decreases. But $(r_p)_{opt.}$ will be lesser in of helium as compare to that with air

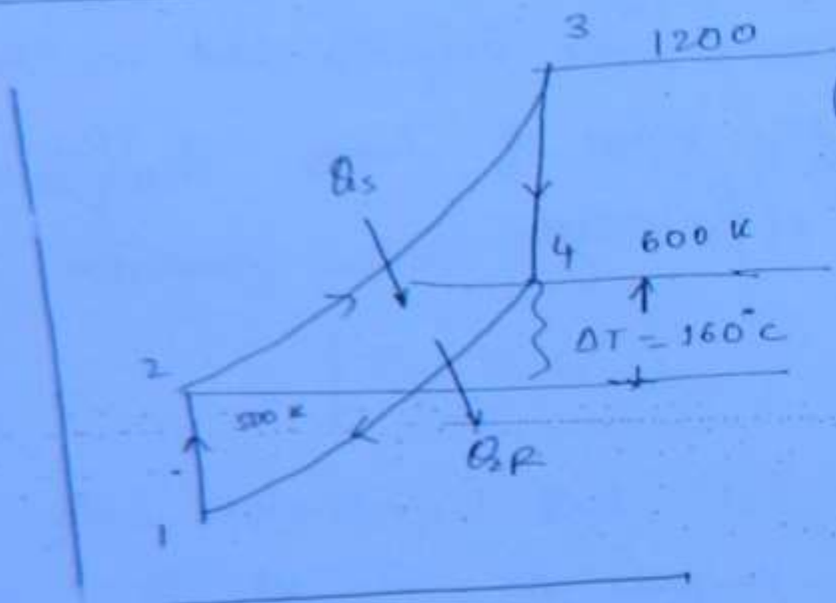



(23)

At any given pressure ratio improvided in the the cycle, higher the value of $T_{max}$ in the cycle, greater the efficiency.

# REGENERATION IN GAS TURBINE CYCLE

Regeneration means transfer of Heat within the cycle from Higher Tempreature to Lower tempreature for the sake of increasing Thermal Efficiency.




(24)

$r_p = 4$

$T_1 = 300$ K

$T_3 = 1200$ K

$T_2 = T_1 (4)^{\frac{\gamma-1}{\gamma}}$

$= \cancel{500.55}$ K 445.79 K

$T_4 = \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}} = \cancel{740.25 K}\ \cancel{659.27 K}$ 807 K

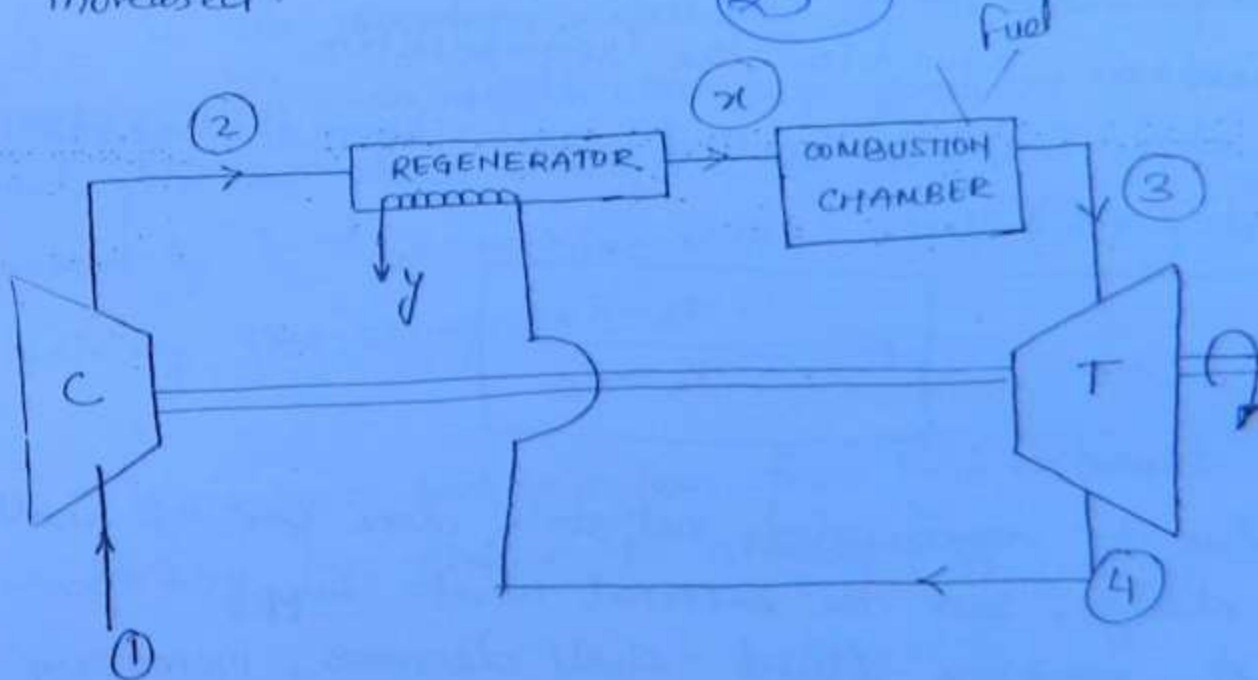
The Hot exhaust gases leaving the turbine are utilized for transferring the heat to the compressed air in a counter flow heat exchanger called **Regenerator**, there by reducing the external heat supplied in the combustion chamber, hence increasing thermal efficiency.

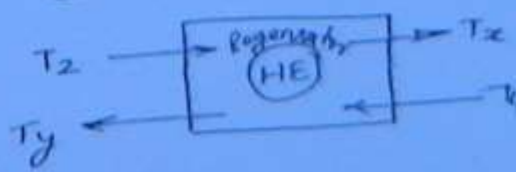
Note:

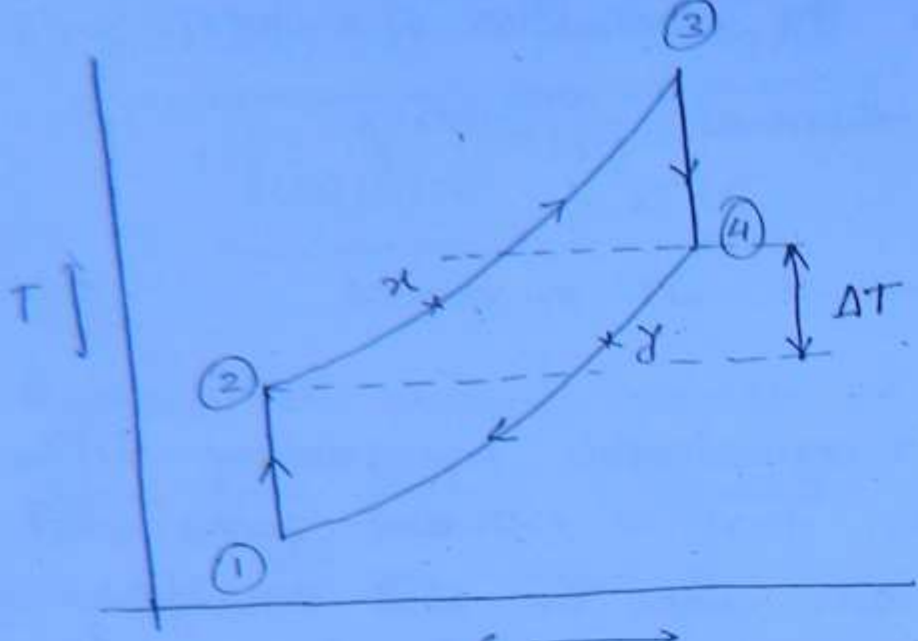
If $r_p$ decreases $\longrightarrow$ $\Delta T$ increases

Since there is a considerable temperature difference existing between turbine exhaust gases and compressed air, there could be heat transfer possible between the turbine exhaust and the compressed air in a heat exchanger called <u>REGENERATOR</u>, thereby external heat supplied can be reduced & thermal efficiency can be increased.








(26)

$T_x$ is the temperature of compressed air

The effectiveness of the regenerator is defined as the ratio between actual temperature rise of compressed air and the maximum possible temperature rise of air in the regenerator.

It is also called as Thermal Aspect Ratio ($\epsilon$).

$$\boxed{\epsilon = \frac{T_x - T_2}{T_4 - T_2}} \cong 0.7$$

Due to regeneration, net work done per kg shall not change, but the external heat supplied per kg of working fluid shall decrease, increasing the thermal efficiency.

~~The compressor~~

* 1-2, is the isentropic compression of air from atmospheric pressure to combuction chamber pressure increasing the pressure and tempreature of air in the rotary compresso: (centrifugal or axial)

* 2-3, is the isobaric heat addition to the air in the combustion chamber, due to the heat liberated by Burning of fuel which is sprayed directly into the air, rising the tempreature of the gases.

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* 3-4, is the isentropic expansion of the gases in the turbine from combustion chamber pressure to the exhaust pressure doing work on the turbine blade. Approximately $1/3^{rd}$ of turbine work shall be used for driving the compressor.

* 4-1, is isobaric heat rejection from hot exhaust gases leaving the turbine to atmosphere

Pressure Ratio $\boxed{(r_p) = \frac{P_2}{P_1} = \frac{P_3}{P_4}}$

$$\boxed{\eta_{(th)} = 1 - \frac{Q_R}{Q_S}}$$

Ideal Brayton

$Q_{Rej} = C_p (T_4 - T_1)$ KJ/kg

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$Q_s = C_p (T_3 - T_2)$ KJ/kg

$$\boxed{\eta_{th} = 1 - \frac{T_4 - T_1}{T_3 - T_2}}$$

Now, 1-2 is, Isentropic Process,

$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1} = \boxed{(r_p)^{\frac{\gamma-1}{\gamma}}} \quad \text{—— } (1)$$

3-4 is, Isentropic process,

$$\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4} = \boxed{(r_p)^{\gamma-1/\gamma}} \quad \text{—— } (2)$$

Actual thermal efficiency of regenerative cycle is,

$$\eta_{(th)\,Reg.} = \frac{\text{Net. Work Output}/kg}{Q_s\ /kg} = \frac{W_T - W_C}{Q_s}$$

$$\therefore \eta_{th} = \frac{C_{pg}(T_3 - T_4) - C_{pa}(T_2 - T_1)}{\cancel{C_{pg}}(T_3 - T_x)}$$

$\because$ Mostly $C_{pg} \cong C_{pa}$

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$$\therefore \boxed{\eta_{th} = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_x)}}$$

Hence, Regeneration in GT cycle results in,

1) No change is W.D/kg

2) [illegible] heat supply /kg get reduced.

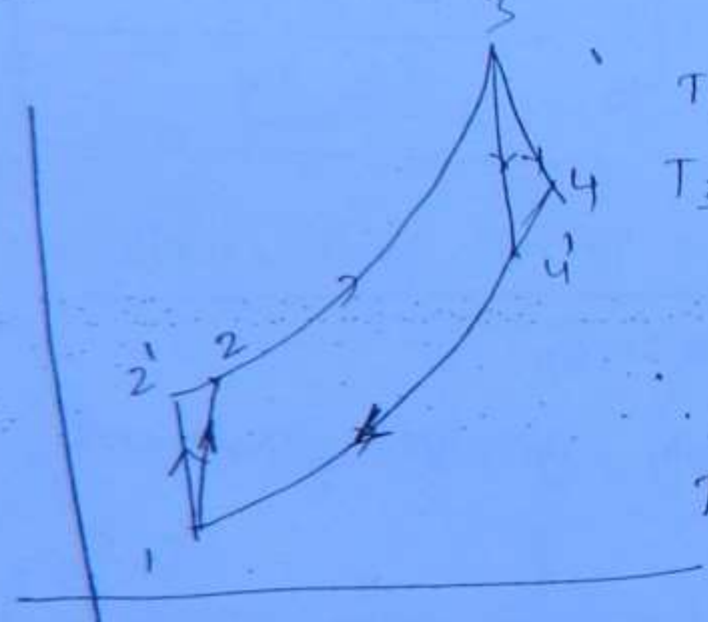
3) $\eta_{(th)\,cycle}$ increases

4) Specific fuel consumption gets ~~not~~ reduced
(Fuel Consumption required to produce 1kW Power)

Q) A Gas Turbine Cycle operates with air having constt. specific heats. The inlet air is at 1 bar and 20° C. The max. cycle temp. is 1200 K. The compressor & turbine have same pressure ratio and have internal efficiency of 0.85. A regenerator with 85% effectiveness is consider. Cal^c all temp. around the cycle, $\eta_{th}$, heat added to produce 10MW power, cycle efficieny if no regenerator is used. Assume $r_p = 5$

(30)

Sol^n




$T_1 = 293$ K

$T_3 = 1200$ K

$T_2' = 464.06$ K

$T_4' = 757.66$ K

$$\eta_{cycle} = 1 - \frac{T_1}{T_2} \quad \therefore \quad 0.85 - 1 = -\frac{T_1}{T_2}$$

$$T_2 = \frac{T_1}{1-0.85}$$

or

$$\therefore \cancel{T_x = 713.62\ K}$$

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

$$\therefore \cancel{T_2} = 0.85(T_2 - 293) = 464.06 - 293$$

$$\boxed{T_2 = 494.2}\ K$$

$$\cancel{\eta_T = \frac{T_3 - T_4'}{T_3 - T_4}}$$

(3P)

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_4'} \quad \therefore \boxed{T_4 = 824}\ K$$

$$\epsilon = \frac{T_x - T_2}{T_4 - T_2} \quad \therefore \boxed{T_x = 774.53\ K}$$

$$\therefore \eta_{th} = \frac{(T_3 - T_4) - (T_2 - T_1)}{T_3 - T_x}$$

$$= 0.41 = 41\%$$

Without regeneration,

$$\eta_{th} = \frac{(T_3 - T_4) - (T_2 - T_1)}{T_3 - T_2}$$

$$= 24.7\ \%$$

$$Q_s = \frac{\text{Power Supp.}}{\eta_{th}}$$

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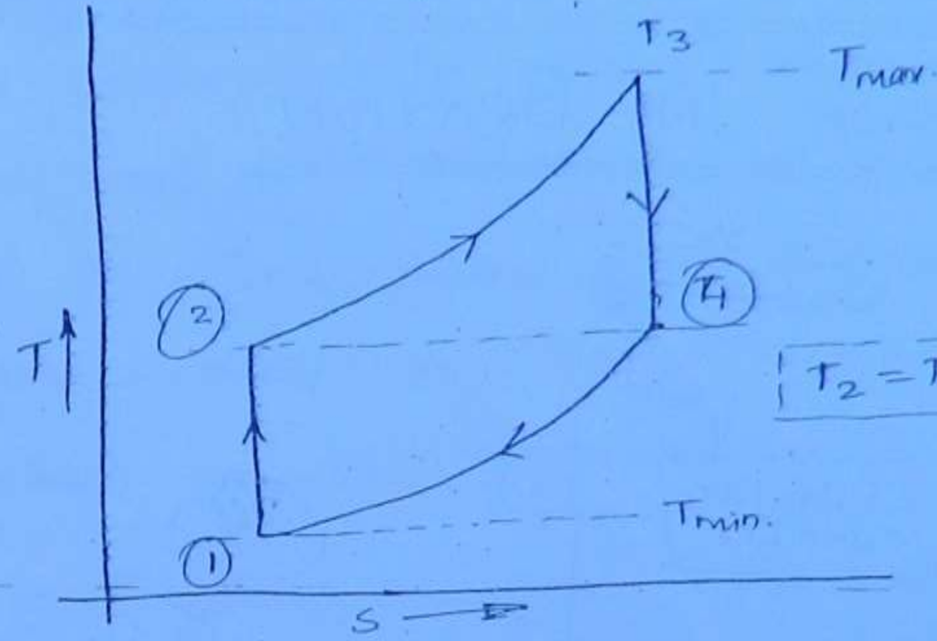
$$Q_s = \frac{10 \times 1000}{0.41} = 24.39\ \text{MW}$$

Note:

The effect of regeneration in increasing the efficieny is more pronounced at low air pressure ratio's, since there could greater "$\Delta T$" between turbine exhaust gases and compressed air.

There is one particular pressure ratio called Critical Pressure ratio where, there is no chance for regeneration because at these pressure ratio, compressed air temperature shall be equal to the turbine exhaust temperature.

For a given $T_{min}$ and $T_{max}$ —




$$\boxed{T_2 = T_4}$$

Here,

$$\frac{T_2}{T_1} = (r_p)_{critical}^{\frac{\gamma-1}{\gamma}} \quad \text{and} \quad \frac{T_3}{T_4} = (r_p)_{critical}^{\frac{\gamma-1}{\gamma}}$$

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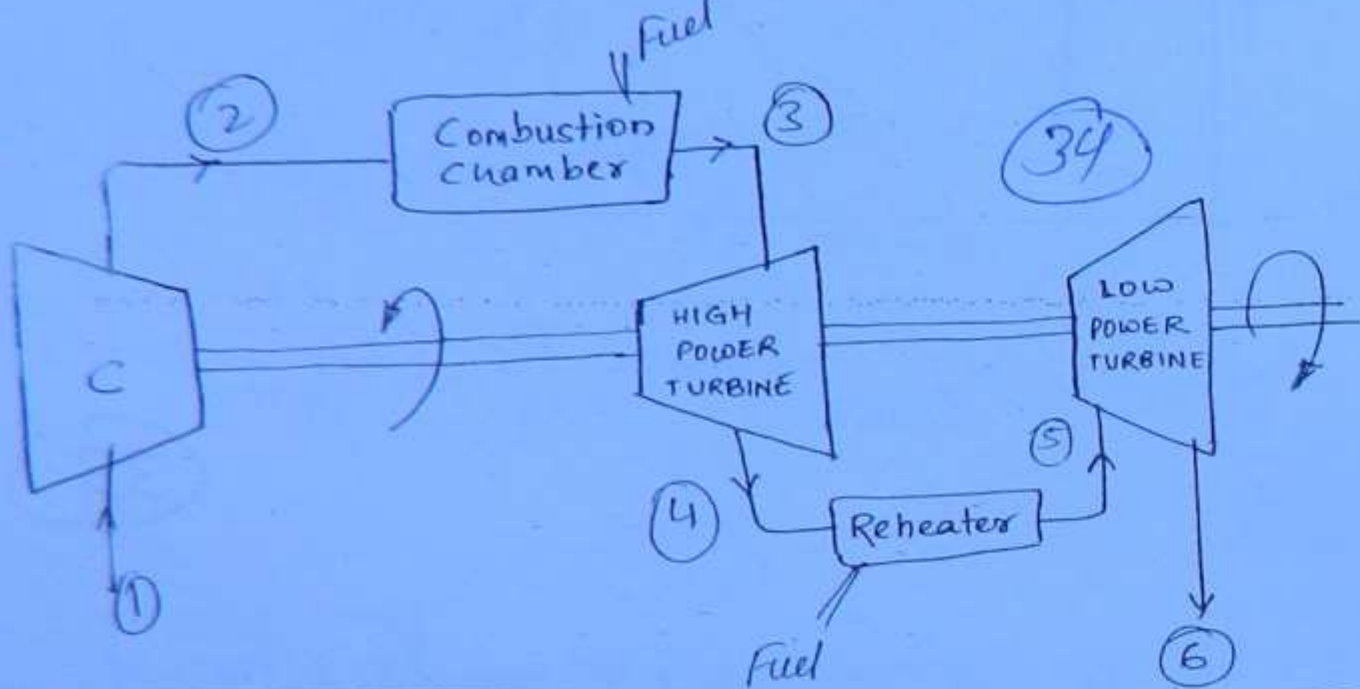
$$\boxed{\frac{T_2}{T_1} = \frac{T_3}{T_4}}$$

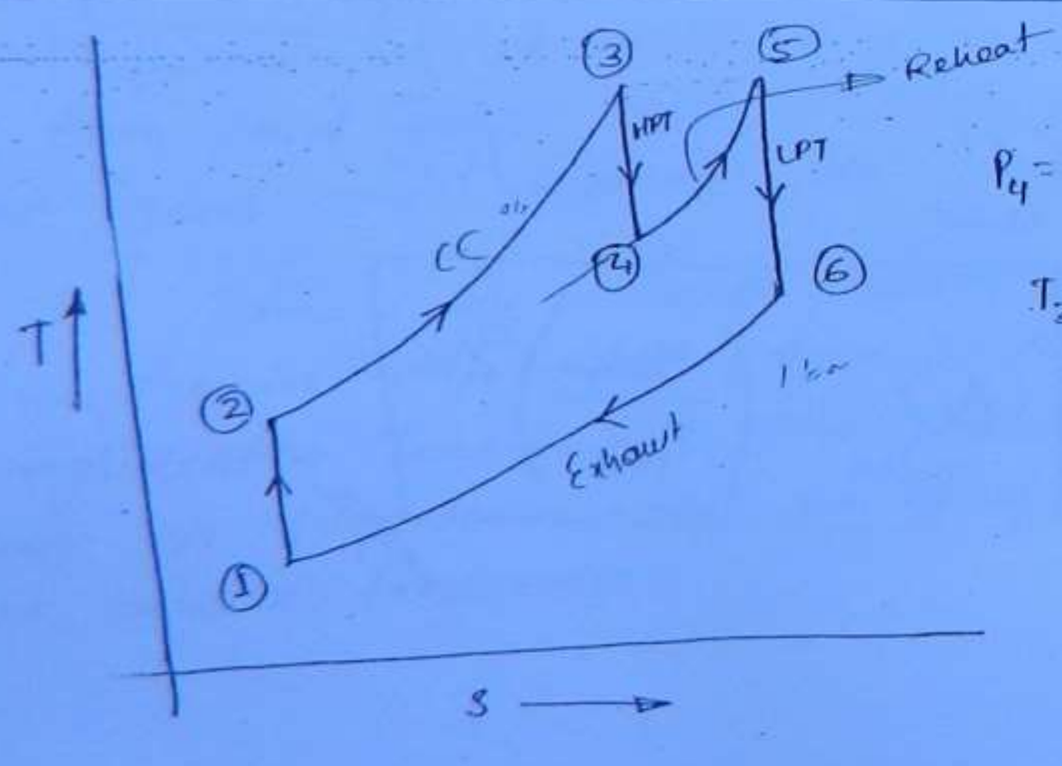
Put $T_2 = T_4 \Rightarrow T_2^2 = T_1 T_3 \Rightarrow \boxed{T_2 = \sqrt{T_1 T_3}}$

$$(r_p)_{Critical}^{\frac{\gamma-1}{\gamma}} = \frac{\sqrt{T_1 . T_3}}{T_1} = \sqrt{\frac{T_3}{T_1}}$$

$$\boxed{(r_p)_{Critical} = \left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{2(\gamma-1)}}}$$

# PREHEATING WITH MULTISTAGE EXPANSION IN BRAYTON CYCLE




$P_4 = P_5$ = Reheater Pressure

$T_3 = T_5$

Gases generated in the combustion chamber, expand in the high pressure turbine from combustion chamber to some intermediate reheat temperature, where it is again reheated ~~to some~~ at constant pressure back to the same initial peak tempreature. Then the gases expand in the low pressure turbine from reheat pressure to exhaust pressure.

For getting maximum total net work output from the cycle, when reheating is provided, the reheat pressure must be geometric mean of suction and delivery pressures i.e. pressure ratio across each stage of turbine, must be same

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$$\therefore \frac{P_3}{P_4} = \frac{P_5}{P_6} \quad \text{and} \quad P_4 = P_5$$

$$\therefore P_5 = \boxed{P_4 = \sqrt{P_3 P_6}}$$

Effects of Reheating :—

(1) Net workoutput per kg of cycle increases.

(2) For a given mass flow rate of air, power output of the cycle increases.

(3) For a given power output, overall size of the plant becomes smaller. $\left(\because P = \underset{\uparrow \text{decrease}}{\dot{m}_a} \times \underset{\uparrow \text{increase}}{W.D/sec}\right)$

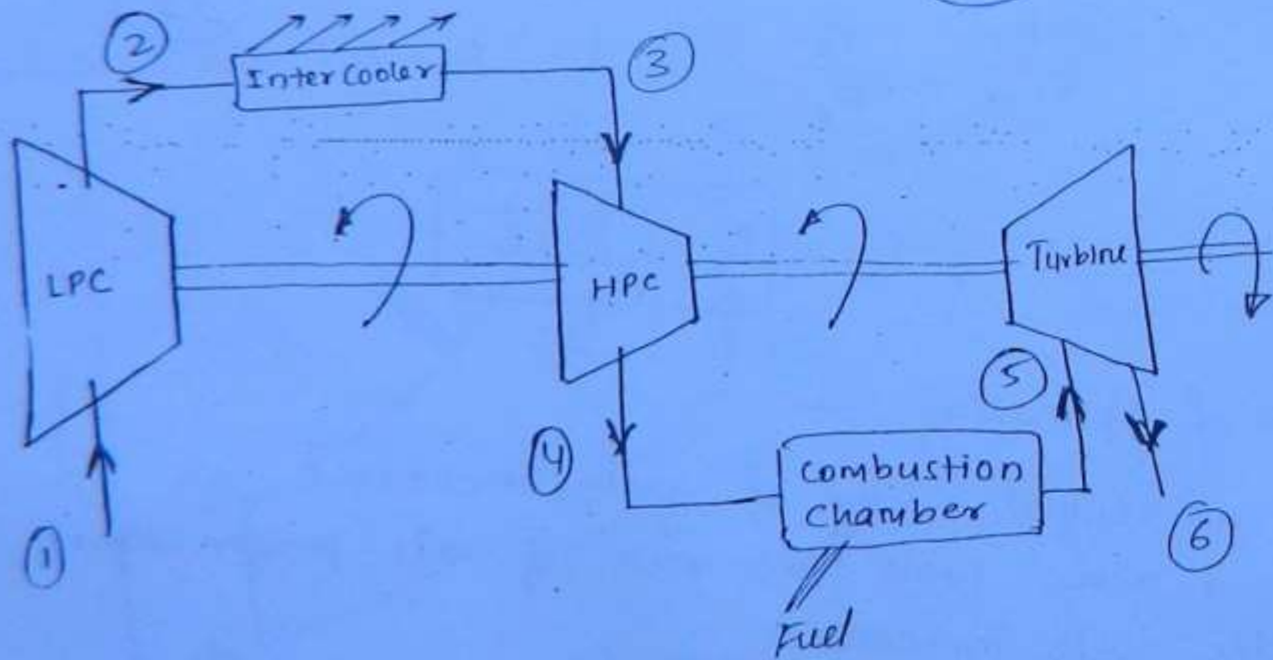
(4) Thermal efficiency marginally increases.

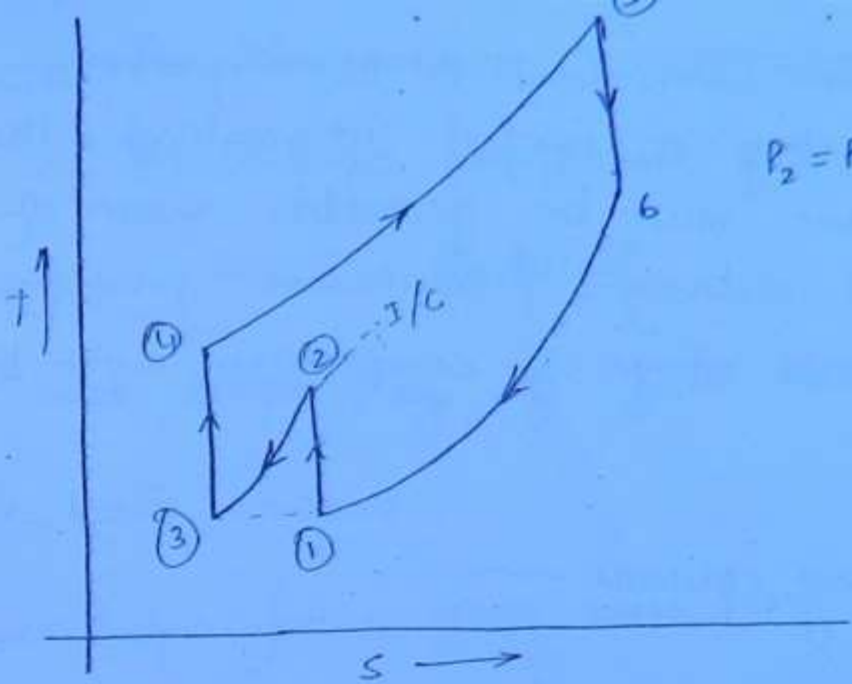
(5) Work Ratio increases.

# INTERCOOLING WITH MULTISTAGE COMPRESSION IN BRAYTON CYCLE

For a given mass of air and for a given pressure ratio, work required in the multistage compression with intercooling shall always be lesser than work required for single stage compression.








$P_2 = P_3 =$ Intercooler Pressure

Air entering from the atmosphere into the compressor is compressed in the low pressure compressor from atmospheric pressure to some intermediate pressure called Intercooler pressure, where it is cooled at constant pressure (②−③) in the intercooler back to the initial tempreature prior to L.P. compression, Then it is again compressd ~~the~~ in the H.P. compressor, from intercooler pressure to combustion chamber pressure. 37

Imp More the no. of stages of compression with intercooling, lesser the work input required. If infinite no. of stages of compression are provided, then it ideal most compression, which requires least work input ie. isothermal compression is obtained.

For total minimum work input to the compressor, with ideal intercooling or perfect intercooling, the intercooler pressure must be geometric mean of the suction and delivery pressures i.e. pressure ratio across each stage of compression must be same.

Perfect Intercooling means —

(1) $P_2 = P_3$ (No pressure drop in intercooler)

(2) $T_3 = T_1$ (Temp. of air after intercooling = Temp of air prior to L.P. stage compression)

(3) Then for total minimum work input,

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad (38)$$

$$\Rightarrow P_2 = \boxed{P_3 = P_{I/c} = \sqrt{P_1 P_4}}$$

All, 3 pressures $(P_1, P_3, P_4)$ are in geometric progression

$\boxed{HPC} < \boxed{LPC}$ (∵ there is const $\dot{m}$)

$\therefore \rho Q = C$

$\therefore v_3 < v_1$

$Q_3 < Q_1$

EFFECTS OF MULTISTAGE COMPRESSION

Effects of Multistage Compressor with Intercooler —

(1) Compressor $W_{input}$ Reduced.

(2) Net work output / kg of cycle increases

(3) Work Ratio increases.

(4) Power Output for a given mass flow rate increases.

(5) $\eta_{th}$ decreases!

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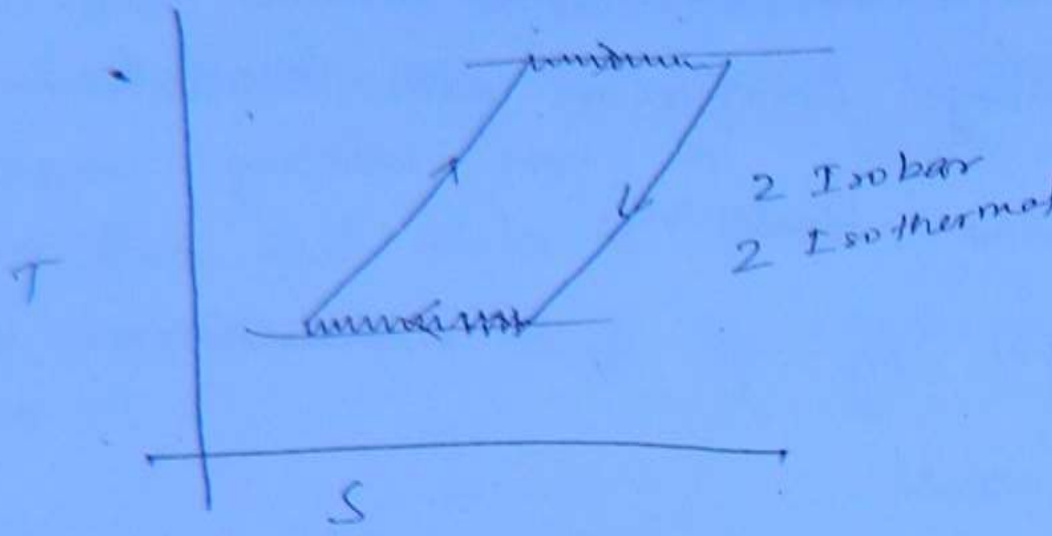
The decrease in $\eta_{th}$ because of multistage compression with intercooling, can be compensated by coupling the regenerator into these system, because, there is a greater scope for regeneration in these cycle because $\Delta T$ between turbine exhaust and compressed air is higher here.

Q.) A gas turbine cycle with infinite no. of stages of multistage compression with intercooling and infinite no. of stages of multistage expansion with reheating, shall result in what cycle —

P.T.O




"ERICSSON CYCLE"

NOTE

$$\boxed{\eta_{stirling} = \eta_{carnot} = \eta_{ericsson}}$$

if regeneration is provided, then only

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