88 Pg

5 Oct, 17

Power-Plant

Gate
- Gas turbine
- Rankine cycle
- Reci- compressor
- centr- compressor
- Axial flow comp.

+

IES
- Actual Gas turbine
- Binary vap cycle
- Impulse Turbine
- Reaction
- Boiler & its component
- condenser
- cooling tower.

Gas Turbine

Engine

* It is a mechanical device which converts one form of energy into another useful form.

IC Engine

* In IC Engine the products of combustion transfer their heat directly in the engine or fuel is the working fluid or combustion & expansion takes place at same location.

EC Engine

* In this type the products of combustion are transfer their heat to the other working fluid which is utilised for producing some output or combustion & expansion takes place at different location.

$$\eta = 1 - \frac{T_L}{T_H}$$

T_L — To (sink)

if $T_h \uparrow - T_L =$ constant $\rightarrow \eta \uparrow$

if $T_h =$ const $\rightarrow T_L \downarrow \rightarrow \eta \uparrow$

*** Max Blade efficiency, $\eta_{B_{max}} = \cos^2\alpha$

α — ** Nozzle angle

$\eta \uparrow \Rightarrow$ In most of the cases bcoz of $\uparrow$ in mean temp. of heat addition

Thermodynamic

↓

Energy Interaction

W $\neq$ Q → HMT ($\Delta T \neq 0$)

W: +ve, −ve

+ve: KP → Engine → IC, EC

−ve: CS → RAC

- IC → CE → HE → R.E
- PP → CE → HE → R.E → EE

> Advantage of Gas turbine over IC engine

* High speeds can be developed due to rotary engine
* Easy Balancing.
* Compact i.e weight to power ratio is less
* Simple mechanism.

> Disadvantages

* As the compresor is handling the gaseous phase of the working fluid which is not negligible in comparission to the turbine work. Therefore the net work o/p is having lower value. & it results a decrease in efficiency.

$$\downarrow\eta = \frac{W_{net}\downarrow}{Q_s}$$

$$\downarrow W_{net} = W_T - \uparrow W_C$$

$$\int v_g dp \cdot \int v_g dp$$

* High heat resistant material are required in the gas turbine as these are subjected to higher temp continuously.

* High speed reduction gears are required as the value of centrifugal forces are high.

$$F_c = m r \omega^2$$

$$F_c = m r \left(\frac{2\pi N}{60}\right)^2$$

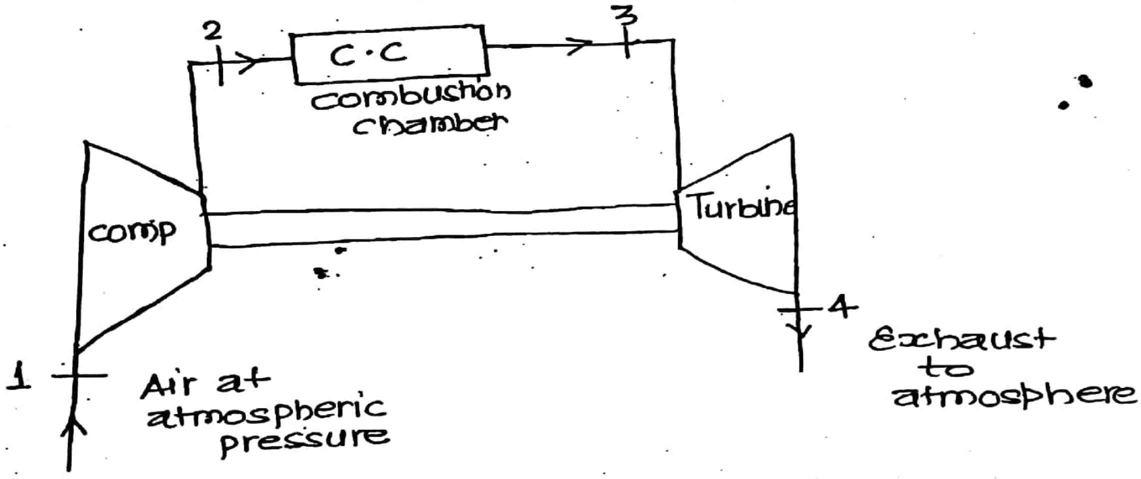
$$F_c \propto N$$

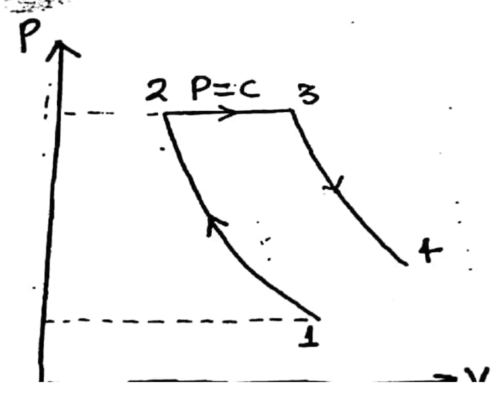
> Types of Gas Turbine

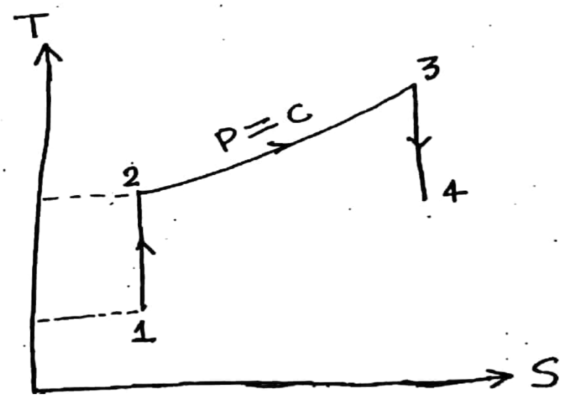
1. open cycled gas Turbine

* In the case of open cycle gas turbine in every cycle fresh charge is introduced.

* process 1-2 → Isentropic compression
  2-3 → const. pressure heat addition
  3-4 → isentropic Expansion.

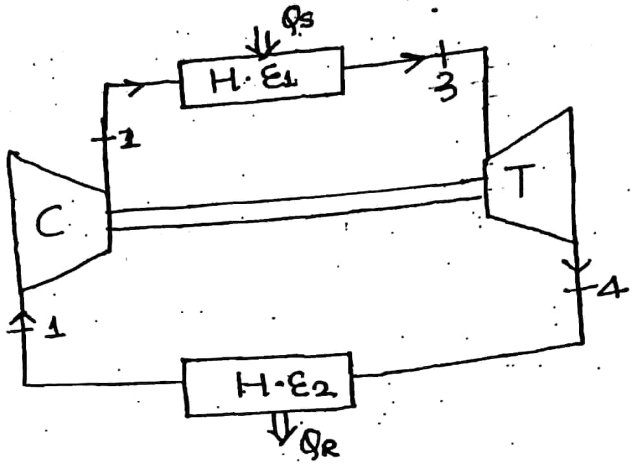


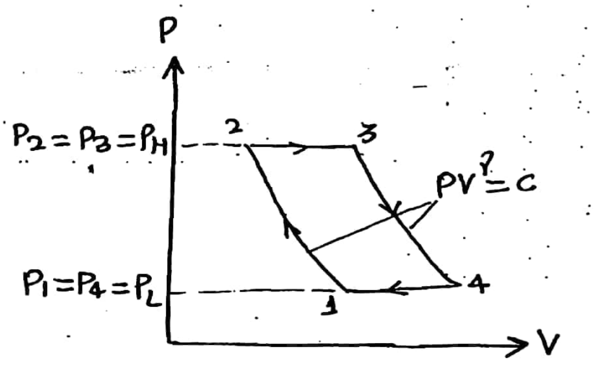


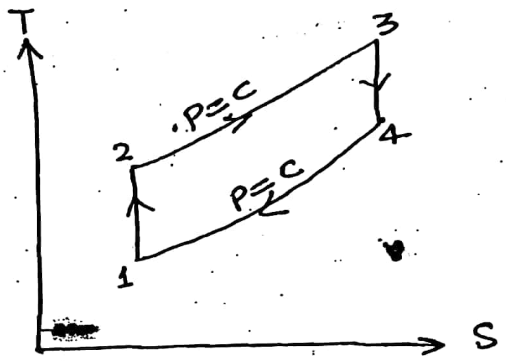


2. Closed cycle gas turbine

* In this cycle same working fluid is recirculated.







let $r_p = \frac{P_H}{P_L}$

→ Advantage of close cycle gas turbine cycle over open cycle.

* Wide variety of fuel can be used because the product of combustion does not enter the turbine.
* Better working fluids can be used like Argon & Helium having the value of index 1.67.
* It can operate even below atmospheric pressure.
* Having more efficiency b/w the same Max & Min temp limits.

$$\uparrow \eta_{otto} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} \downarrow$$

$$\uparrow \eta_{Bray} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} \downarrow$$

Scanned by CamScanner

→ Disadvantage of close cycle gas turbine

* A coolant is required is to cool down the turbine exhaust before entering the compressor where as in open cycle gas turbine atmosphere is acting a sink

* Absolute leak proofing is very difficult to achieve.

* System is complicated, costly, & Bulky. theirfore these are not prefered in air-craft application.

– Back work ratio (r_{bw})

$$r_{bw} = \frac{-ve\ work}{+ve\ work}$$

Rankine cycle

* $r_{bw} = \frac{W_P}{W_T}$ → $\int v_f dp$

* 1% to 2%

* $r_w = 98\%$ to 99%

Brayton cycle

* $r_{bw} = \frac{W_c}{W_T}$ $\int v_g dp$

* 40 to 60%

$r_w = 60\%$ to 40%

–

– work ratio (r_w)

$$r_w = \frac{Net\ work}{+ve\ work}$$

$$= \frac{W_T - W_c}{W_T} = 1 - \frac{W_c}{W_T}$$

∴

$$\boxed{r_w = 1 - r_{bw}}$$

– Efficiency of Brayton cycle

$$\eta = \frac{O/P}{I/P}$$

$$\boxed{\eta = \frac{W_{net}}{Q_s}}$$

$$\eta = \frac{Q_{net}}{Q_s}$$

$$Q_{net} = \cancelto{0}{Q_{1-2}} + Q_{2-3} + \cancelto{0}{Q_{3-4}} + \cancel{Q_{4-1}}$$

$$= Q_{2-3} + Q_{4-1}$$

$$= Q_S - Q_R$$

$$\therefore \eta = \frac{Q_S - Q_R}{Q_S}$$

$$\boxed{\eta = 1 - \frac{Q_R}{Q_S}}$$

Air, $PV = mRT$

$h = C_pT$

$$\eta = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

$$\eta = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} \quad -① $$

Process (1-2) :- $PV^{\gamma} = c$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{x} = \left(\frac{P_H}{P_L}\right)^{x} = (r_p)^{x} \quad -② \qquad \because x = \frac{\gamma - 1}{\gamma}$$

Process (3-4) : $PV^{\gamma} = c$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{x} = \left(\frac{P_2}{P_1}\right)^{x} = (r_p)^{x} \quad -③$$

compare eq ② & 3.

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

*** $$\boxed{T_2 T_4 = T_1 T_3} \quad -4$$

use eqn ④ & 1

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

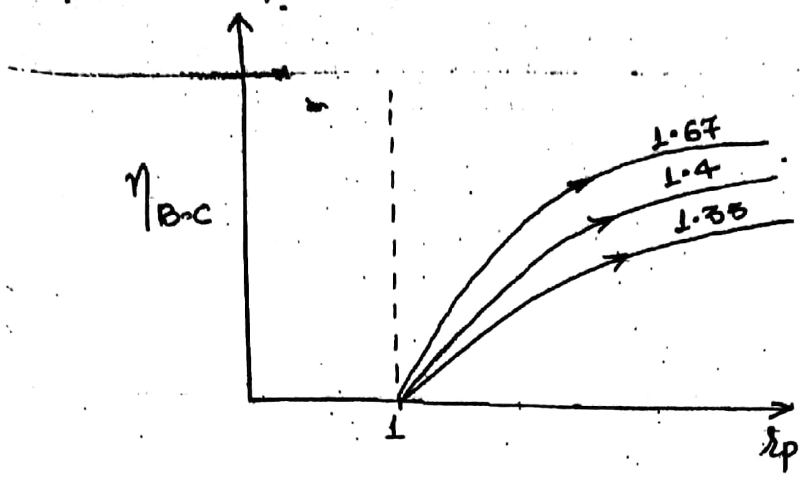
$$\therefore \eta = 1 - \frac{T_1}{T_2}$$

$$\eta = 1 - \frac{1}{T_2/T_1}$$

*** $$\boxed{\eta = 1 - \frac{1}{(r_p)^{\frac{\gamma - 1}{\gamma}}}}$$

* The efficiency of Brayton cycle is a function of pressure ratio & Index.

$$\uparrow \eta = 1 - \left[\frac{1}{r_p^{\frac{\gamma - 1}{\gamma}} \rightarrow 1 - \left(\frac{1}{\gamma\uparrow}\right)\downarrow}\right]\downarrow$$



$\gamma \uparrow \rightarrow \eta \uparrow$

$r_p \uparrow \rightarrow \eta \uparrow$

Note above

* The expression of efficiency of Brayton cycle is applicable when both compression & expansion are isentropic.

> Assumptions of Air-standard cycle:

* Air is the working fluid treated as ideal gas.
* The value of cp, cv, & γ are assumed as constant w.r.t temp.
* Both compression & expansion are assumed to be isentropic
* should be a close system analysis.
* All process are internally reversible.
* Neglecting K.E & P.E changes
* Neglecting pressure loss.

> Actually

* As T $\uparrow$ — Cp $\uparrow$, Cv $\uparrow$, $\gamma \downarrow$

$C_p - C_v = R \rightarrow C_p/C_v$

$3 - 2 = 1 \rightarrow 1.5$

$4 - 3 = 1 \rightarrow 1.33$

* $ds = S_{gen} + \frac{dQ}{T}$ (rever, actual)
* Isentropic, $\eta_c \neq 100\%$, $\eta_T \neq 100\%$
* $\Delta p \neq 0$
* $h_1 + \cancel{KE_1} + \cancel{PE_1} + Q = h_2 + \cancel{KE_2} + \cancel{PE_2} + W$ (insul)

> Effect of pressure ratio on efficiency & net work output.

Their is a upper limit of 1700 K is imposed due to the metalurgical limit condition of the turbine blades. with the increase in pressure ratio the efficiency increases but for the same power o/p the mass flow rate increases.

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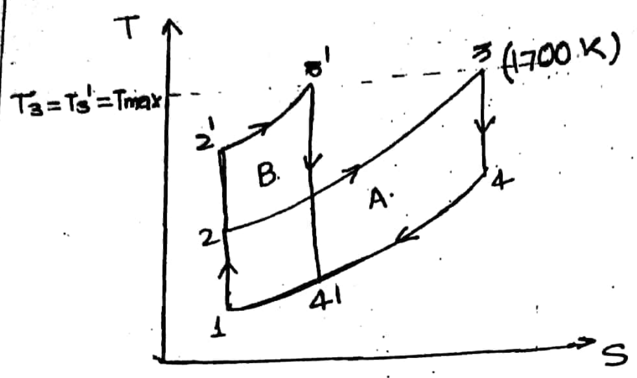


A → 1 − 2 − 3 − 4 − 1 → $(r_p)_A = \frac{P_2}{P_1}$

B → 1 − 2' − 3' − 4' − 1 → $(r_p)_B = \frac{P_2'}{P_1}$

$P_2' > P_2$

$\frac{P_2'}{P_1} > \frac{P_2}{P_1}$

$(r_p)_B > (r_p)_A$

$\eta_B > \eta_A$

Power output $= \uparrow \dot{m} \times \downarrow W_{Net}$

$(W_{net})_A > (W_{net})_B$

$(\dot{m})_A < (\dot{m})_B$

> optimum pressure ratio for maximum work output

$W_{net} = W_T - W_C$

$= (h_3 - h_4) - (h_2 - h_1)$

$= h_3 - h_4 - h_2 + h_1$

$W_{net} = C_p(T_3 - T_4 - T_2 + T_1) \quad \text{--- (1)}$

$T_1 = T_{min}$, $T_3 = T_{max}$,

$T_2 T_4 = T_1 T_3$

$T_2 = \frac{T_1 T_3}{T_4} \quad \text{--- (2)}$

Use (2) in (1)

$W_{net} = C_p \left[T_3 - \frac{T_1 T_3}{T_4} - T_4 + T_1 \right]$

$\frac{dw}{dT_4} = 0 \qquad T_4^{-1} = -1 \, T_4^{-2} \rightarrow \frac{-1}{T_4^2}$

$+ \frac{T_1 T_3}{T_4^2} - 1 = 0$

$\frac{T_1 T_3}{T_4^2} = 1$

$T_4^2 = T_1 T_3$

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

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

Use in eqn 1.

$$W_{net} = C_p\left[T_3 - T_4 - T_2 + T_1\right]$$
$$= C_p\left[T_3 - \sqrt{T_1T_3} - \sqrt{T_1T_3} + T_1\right]$$
$$= C_p\left[T_3 + T_1 - 2\sqrt{T_1T_3}\right]$$

$$W_{max} = C_p\left[(\sqrt{T_3})^2 + (\sqrt{T_1})^2 - 2\sqrt{T_1T_3}\right]$$

$$W_{max} = C_p\left[\sqrt{T_3} - \sqrt{T_1}\right]^2$$

***
$$\boxed{W_{max} = C_p\left[\sqrt{T_H} - \sqrt{T_L}\right]^2}$$

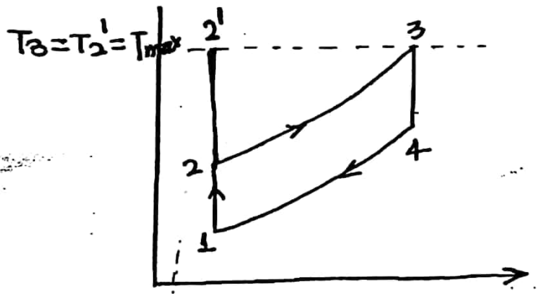
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

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

$$(r_p)_{opt} = \left(\frac{\sqrt{T_1 \cdot T_3}}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\sqrt{\frac{T_3}{T_1}}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

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

> Prove that $(r_p)_{max} = (r_p)^2_{opt}$



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

$$(r_p)_{max} = \frac{P_2'}{P_1} = \left(\frac{T_2'}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{\gamma-1}}$$

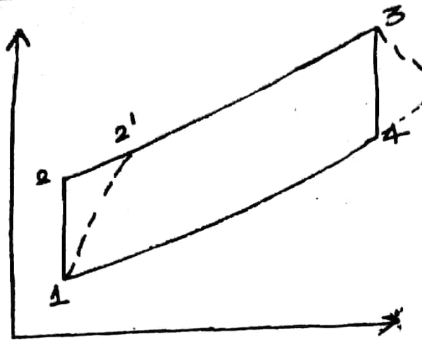
$$= \left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{\gamma-1}}$$

***
$$\boxed{(r_p)_{max} = (r_p)^2_{opt}}$$

> Isentropic efficiency of compressor & Turbine.

Isentropic efficiency of compressor is defined as the ratio of ideal temp rise to that of actual temp rise.

Isentropic efficiency of ~~compr~~ Turbine is defined as the ratio of actual temp drop to that of ideal temp drop.

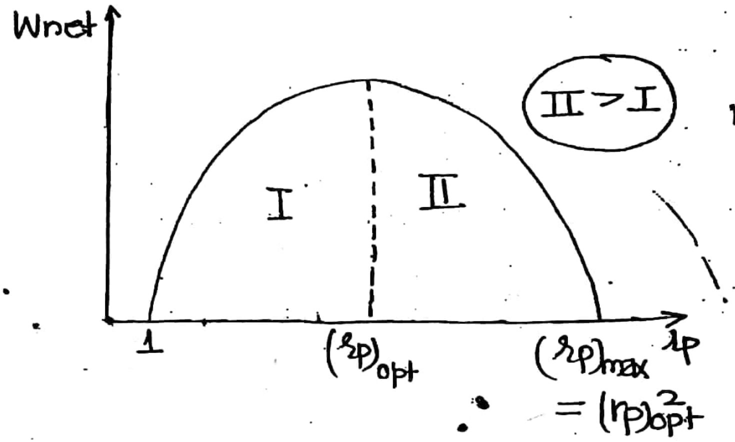


$\eta_{Isentropic}$ → Rev adiabatic
└→ Irrev Adiabatic

$ds = S_{gen} + \frac{dQ}{T}$

$$(\eta_{is})_C = \frac{T_2 - T_1}{T_{2'} - T_1}$$

$$(\eta_{is})_T = \frac{T_3 - T_{4'}}{T_3 - T_4}$$



II > I

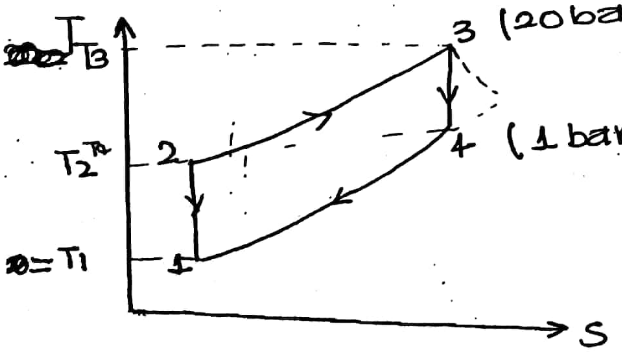
$$\eta = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}}$$

$\Rightarrow r_p = 1 \rightarrow \eta = 0 = W_{net}$

$(r_p) \rightarrow (r_p)_{max}$

$W_{net} = 0$

Pb In a gas turbine hot combustion product having a value of C_p & C_v is 0.98 & 0.7538 kJ/kg K resp. enters the turbine at a pressure of 20 bar, 1500 K & then exit the turbine at a pressure of 1 bar. Assuming the value of Index 1.3 & the Isentropic efficiency of the turbine is 94%. Then find the actual work done o/p of the turbine.



$$\eta_{is} = \frac{T_3 - T_{4'}}{T_3 - T_4}$$

$$0.94 = \frac{1500K - T_{4'}}{1500 - T_4}$$

$$\frac{1500}{T_4} = \frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$= \left(\frac{20}{1}\right)^{\frac{1.3-1}{1.3}}$$

$T_4 = \cancel{750.6}$
$= 751$ K

$$0.94 = \frac{1500 - T_{4'}}{1500 - 751}$$

$T_{4'} = 796$ K

$W_T = h_3 - h_{4'}$
$= C_p(T_3 - T_{4'})$
$= 0.98(1500 - T_{4'})$
$= 689.5 \frac{kJ}{kg}$

Pb An ideal Brayton cycle having a net work o/p of 150 kJ/kg & the work ratio is 0.4. If the isentropic efficiency of both comp & turbine are 84%. Then find out the actual net work o/p.

For ideal cycle.

$$\eta = 0.84 = \frac{T_2 - T_1}{T_2' - T_1} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$\gamma = 1.4$

$W_{o/p} = 150$ kJ/kg

$$r_w = 0.4 = 1 - \frac{W_C}{W_T}$$

$$= \frac{W_T - W_C}{W_T} = \frac{W_{net}}{W_T}$$

$$0.4 = \frac{150}{W_T}$$

$$W_T = \frac{\overset{75}{\cancel{150}} \times \cancel{10.5}}{\cancel{4}\,2} = 375 \text{ kJ}$$

$r_w = 1 - r_{bw} = 0.4$

$= r_{bw} = 0.6 = \frac{W_C}{W_T}$

$W_C = 0.6 W_T$

$W_T - W_C = 150$

$0.4 W_T = 150$

$W_T = 375$ kJ/kg

$W_C = 225$ kJ/kg

or,

$$\text{Actual } W_C = \frac{\text{Ideal } W_C}{\eta_T}$$

$$\text{Actual } W_T = \text{Ideal } W_T \times \eta_T$$

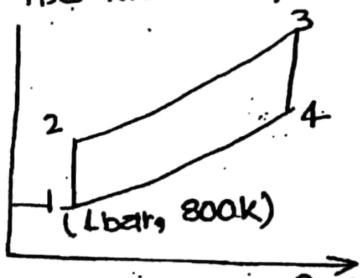
$$(W_{net}) = (W_T)_A - (W_C)_A$$

$$= W_T \cdot \eta_T - \frac{W_C}{\eta_C}$$

$$= 0.85 \cdot W_T - \frac{W_C}{0.85}$$

$$W_{net} = 54 \frac{\text{kJ}}{\text{kg}}$$

Pb Air enters the compressor at a pressure of 1 bar & temp of 27°C operating on Brayton cycle. Having a pressure ratio of 6 if the turbine work is 2.5 times compressor Wc then find out the max temp in the cycle & efficiency of the cycle.



$$(r_p) = 6 = \left(\frac{P_h}{P_L}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\gamma-1/\gamma} =$$

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

$$\eta = 1 - \frac{1}{(6)^{\frac{.4}{1.4}}}$$

$= 40\%$.

$W_T = 2.5 W_C$

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

$$T_3\left(1 - \frac{T_4}{T_3}\right) = 2.5 T_1\left(\frac{T_2}{T_1} - 1\right)$$

$$T_3\left(1 - \frac{1}{(r_p)^x}\right) = 2.5 T_1 (r_p^x - 1)$$

$$\frac{T_3 \cancel{(r_p^x - 1)}}{r_p^x} = 2.5 T_1 \cancel{(r_p^x - 1)}$$

$$T_3 = 2.5 T_1 (r_p)^{\gamma-1/\gamma} = 1251 \text{ K}$$

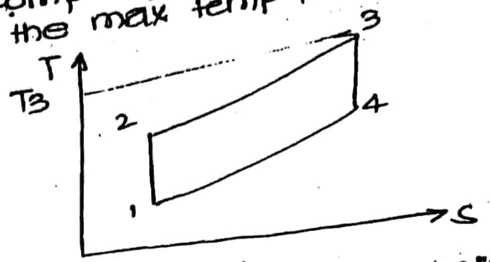
Pb Air enters the comp of a gas turbine having a pressure ratio of 5. Find the max temp in the sec

$P_1 = 1$ bar

$T_1 = 300K$

$r_p = 5$

$r_{bw} = 0.4$



$$\frac{W_C}{W_T} = 0.4$$

$$\frac{T_2 - T_1}{T_3 - T_4} = 0.4$$

$$\eta = 1 - \frac{1}{(6)^{\frac{.4}{1.4}}} = 98\%$$

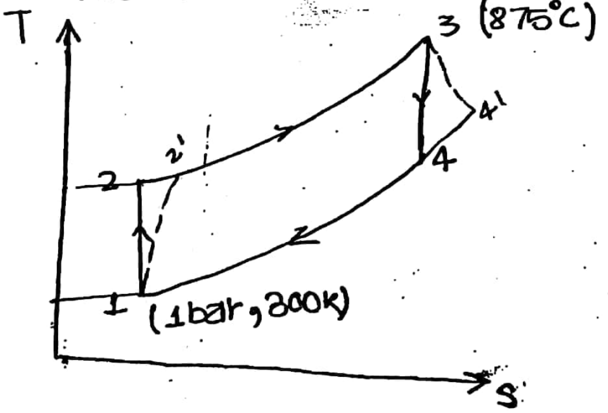
$$\frac{T_2 - T_1}{T_3 - T_4} = \frac{T_1\left(\frac{T_2}{T_1} - 1\right)}{T_3\left(1 - \frac{T_4}{T_3}\right)} = \frac{T_1\left(r_p^x - 1\right)}{T_3\left(1 - \frac{1}{r_p^x}\right)}$$

$$= \frac{T_1\cancel{(r_p^x - 1)}}{\frac{T_3\cancel{(r_p^x - 1)}}{r_p^x}} = 0.4$$

$$0.4 = \frac{(r_p)^x}{T_3}$$

$$T_3 = 1187 K$$

Pb Find the required air fuel ratio in a gas turbine having 80% efficiency of compressor & 85% of Turbine. The max temp in the cycle is 875°C having specific heat of air is 1 kJ/kg K and the value of γ is 1.4 and air enters the compressor at a pressure of 1 bar & temp of 27°C having pressure ratio of 4 & the calorific value of fuel is 42 MJ/kg & their is the loss of 10% of the calorific value in the combustion chamber.



$$\eta_c = 80\% = \frac{\text{Ideal } W_c}{\text{Actual } W_c}$$

$$\eta_T = 85\% = \frac{\text{Actual}}{\text{Ideal}}$$

$$r_p = 4 = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{T_2}{T_1}\right)$$

$$4 = \left(\frac{P_2}{P_1}\right)^{0.285}$$

$$\boxed{128 = P_2}$$

$$\left(\frac{T_2}{T_1}\right) = 4^{\frac{0.4}{1.4}}$$

$$T_2 = 445.79 K$$

$$\eta_c = \frac{T_2 - T_1}{T_2' - T_1}$$

0.80

$$T_2' = 482.24 K$$

$$\boxed{E_3 = E_2' + (Q_s)_{Actual}} \qquad h_3 - h_2' = (Q_s)_{actual} = m_f \times C_v \times \eta_{cc}'$$

$$\frac{(m_a + m_f)\, C_{pa}\, T_3}{m_f} = \frac{m_a C_{pa} T_2'}{m_f} + \frac{m_F\, CV \times \eta_{cc}}{m_f}$$

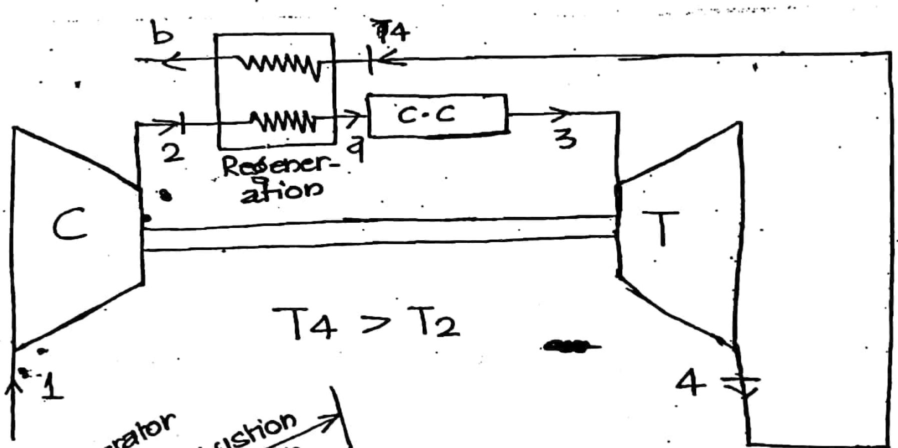
$$\left(\frac{A}{F} + 1\right) 1 \times 1148 = \frac{A}{F} \times 1 \times 482.24 + 42000 \times 0.90$$

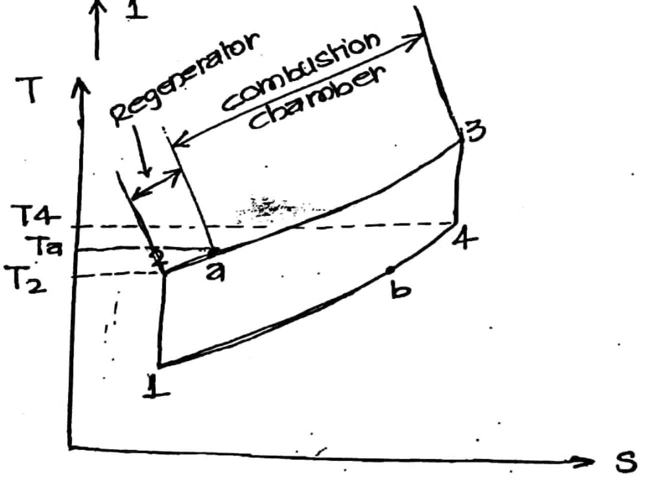
$$\frac{A}{F} = 56.7$$

> Method of improving the performance of Gas Turbine.

Case 1: Regeneration in Gas Turbine

* It is a process in which turbine exhaust are utilised to increased the temp of compressed air before entering the combustion chamber.





(VI) $Q_R \downarrow$ T_{MA} – mean temp of Heat add

(VII) $T_{MA} \uparrow$ T_{MR} – mean temp of Heat reje.

(VIII) $T_{MR} \downarrow$

(XI) SFC $\downarrow$ – specific fuel consumption

1–2–3–4–1	1–2–a–3–4–b–1
(i) $W_C = h_2 - h_1$ const	$W_C = h_2 - h_1$ const
(ii) $W_T = h_3 - h_4$ const	$W_T = h_3 - h_4$ const
(iii) $W_{net} = W_T - W_C$ const	$W_{net} = W_T - W_C$ const
(iv) $Q_S = h_3 - h_2$	$\downarrow Q_S = h_3 - h_a$
(v) $\eta = \frac{W_{net}}{Q_S}$	$\uparrow \eta = \frac{W_{net}}{Q_S \downarrow}$

- Effectiveness of Regenerator / Thermal ... Regeneration.

* It is defined as the ratio of actual temp rise in the regenerator to the max possible temp rise in the regenerator.

$$\boxed{\epsilon = \frac{T_a - T_2}{T_4 - T_2}}$$

* Generally the value of effectiveness is varries from 70% to 80%.

Note

* In the case of Ideal regenerative cycle

$$\epsilon = 100\% = 1 = \frac{T_a - T_2}{T_4 - T_2}$$

$$T_a - T_2 = T_4 - T_2$$

$$\boxed{T_a = T_4}$$

$$\boxed{T_b = T_2}$$

- Effeciency of Ideal regenerative cycle

$$\eta = \frac{o/p}{i/p} = \frac{W_{net}}{Q_s} = \frac{Q_s - Q_R}{Q_s} = 1 - \frac{Q_R}{Q_s}$$

$$\eta_{\text{simple Brayton cycle}} = 1 - \frac{h_4 - h_1}{h_3 - T_2} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Air, $PV = mRT$

$h = C_pT$

$$\eta_{\text{regenerative cycle}} = 1 - \frac{T_b - T_1}{T_3 - T_a}$$

$$\eta_{\text{ideal Reg. cycle}} = 1 - \frac{T_2 - T_1}{T_3 - T_4} \qquad \because T_b = T_2, \; T_a = T_4$$

$$\eta_{\text{ideal reg}} = 1 - \frac{T_1\left(\frac{T_2}{T_1} - 1\right)}{T_3\left(1 - \frac{T_4}{T_3}\right)}$$

$$\eta_{\text{ideal reg}} = 1 - \frac{T_1}{T_3}\left(\frac{r_p^{x} - 1}{1 - \frac{1}{(r_p)^{x}}}\right)$$

**

$$\boxed{\eta_{\text{ideal reg}} = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{\gamma}}}$$

$$x = \frac{\gamma - 1}{\gamma}$$

Observation

* Beyond a certain value of pressure ratio the use of regenerator becomes ineffective as the compressor outlet temp exceeds the turbine outlet temp.

$\eta_{ideal\ Reg}$

$T_1 \rightarrow \downarrow$

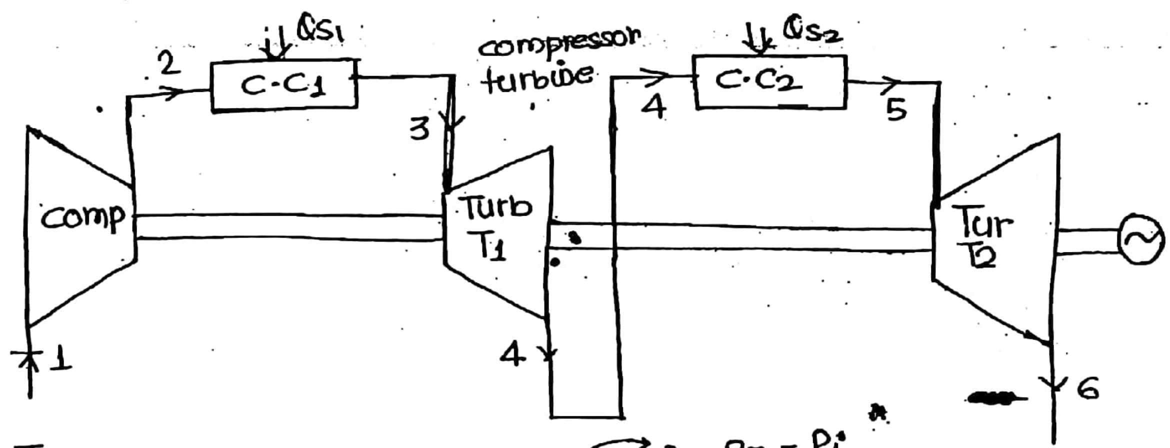
$T_3 \rightarrow \uparrow$

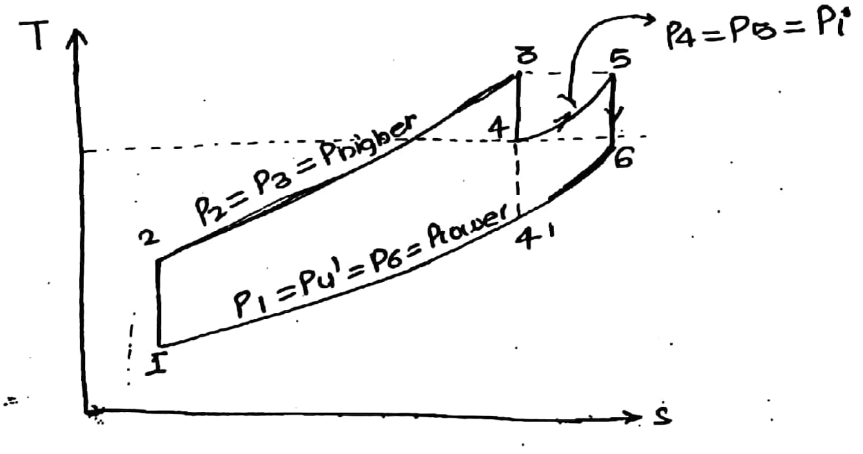
$r_p \rightarrow \downarrow$

$\gamma \rightarrow \downarrow$

Case II : Reheating in Gas turbine

* The main purpose of reheating is to improve the net work output by increasing the turbine work.





1-2-3-4'-1

* $W_C = h_2 - h_1$
* $W_T = (h_3 - h_{4'})$
* $W_{net} = W_T - W_C$
* $Q_S = h_3 - h_2$

R·H - 1-2-3-4-5-6-1

$W_C = h_2 - h_1$

$W_T \uparrow = (h_3 - h_4) + (h_5 - h_6)$

$\boxed{\uparrow W_{net}} = \uparrow W_T - W_C$

$Q_S = (h_3 - h_2) + (h_5 - h_4)$

* $Q_R \uparrow$
* $T_{MA} \uparrow$
* $T_{MR} \uparrow$

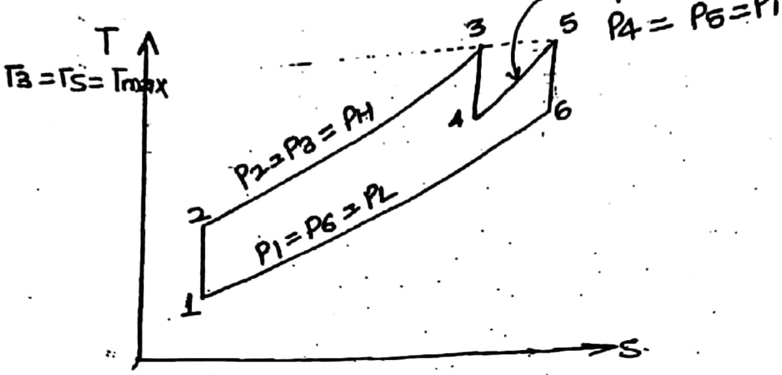
* $\boxed{\eta \downarrow}$

* $T_6 > T_{4'}$

Note

* With the use of reheating the scope of regeneration increases because the turbine outlet temp increases. ($T_6 > T_4'$) Therefore reheating is generally coupled with regeneration.

- condition for max work output with perfect reheating



Perfect reheating $\Rightarrow T_3 = T_5$

$\uparrow W_{net} = \uparrow W_T - W_C$

$W_T = W_{T_1} + W_{T_2}$

$W_T = (h_3 - h_4) + (h_5 - h_6)$

$W_T = C_p(T_3 - T_4) + C_p(T_5 - T_6)$

$W_T = C_p T_3\left(1 - \frac{T_4}{T_3}\right) + C_p T_5\left(1 - \frac{T_6}{T_5}\right)$ — A.

$$\left.\begin{aligned} \frac{T_4}{T_3} &= \left(\frac{P_4}{P_3}\right)^x = \left(\frac{P_i}{P_H}\right)^x \\ \frac{T_6}{T_5} &= \left(\frac{P_6}{P_5}\right)^x = \left(\frac{P_L}{P_i}\right)^x \end{aligned}\right] B. \qquad \boxed{x = \frac{\gamma - 1}{\gamma}}$$

Use B in A.

$$W_T = C_p T_3\left[1 - \left(\frac{P_i}{P_H}\right)^x\right] + C_p T_5\left[1 - \left(\frac{P_L}{P_i}\right)^x\right]$$

$$W_T = C_p T_3\left[2 - \left(\frac{P_i}{P_H}\right)^x - \left(\frac{P_L}{P_i}\right)^x\right] \quad — C$$

$\frac{dW_T}{dP_i} = 0$

$$\frac{-x P_i^{x-1}}{P_H^x} + \frac{x P_L^{x}}{P_i^{x+1}} = 0$$

$$\frac{\cancel{x} P_i^{x-1}}{P_H^x} = \frac{\cancel{x} P_L^x}{P_i^{x+1}}$$

$$P_i^{2x} = (P_H \cdot P_L)^x$$

***
$$\boxed{P_i = \sqrt{P_H \cdot P_L}}$$

- observation

$$W_{T_1} = h_3 - h_4 = C_p(T_3 - T_4) = C_pT_3\left(1 - \frac{T_4}{T_3}\right) = C_pT_3\left(1 - \left(\frac{P_4}{P_3}\right)^x\right)$$

$$= C_pT_3\left[1 - \left(\frac{P_i}{P_H}\right)^x\right]$$

$$= C_pT_3\left[1 - \left(\frac{\sqrt{P_H P_L}}{P_H}\right)^x\right]$$

$$= C_pT_3\left[1 - \left(\sqrt{\frac{P_L}{P_H}}\right)^x\right]$$

$$= C_pT_3\left[1 - \left(\frac{P_L}{P_H}\right)^{x/2}\right]$$

$$W_{T_2} = h_5 - h_6 = C_p(T_5 - T_6) = C_p T_5\left(1 - \frac{T_6}{T_5}\right) = C_pT_5\left[1 - \left(\frac{P_6}{P_5}\right)^x\right]$$

$$= C_pT_5\left[1 - \left(\frac{P_L}{P_i}\right)^x\right]$$

$$= C_pT_5\left[1 - \left(\frac{P_L}{\sqrt{P_H \cdot P_L}}\right)^x\right]$$

$$= C_pT_5\left[1 - \left(\frac{P_L}{P_H}\right)^{x/2}\right] \quad - B$$

Perfect reheating 3 – 5

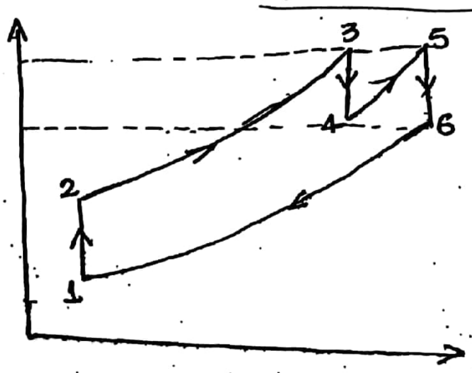
$$T_3 = T_5$$

compare equation A & B)

$\boxed{W_{T_1} = W_{T_2}}$ ⟶ In perfect reheating.

Note :

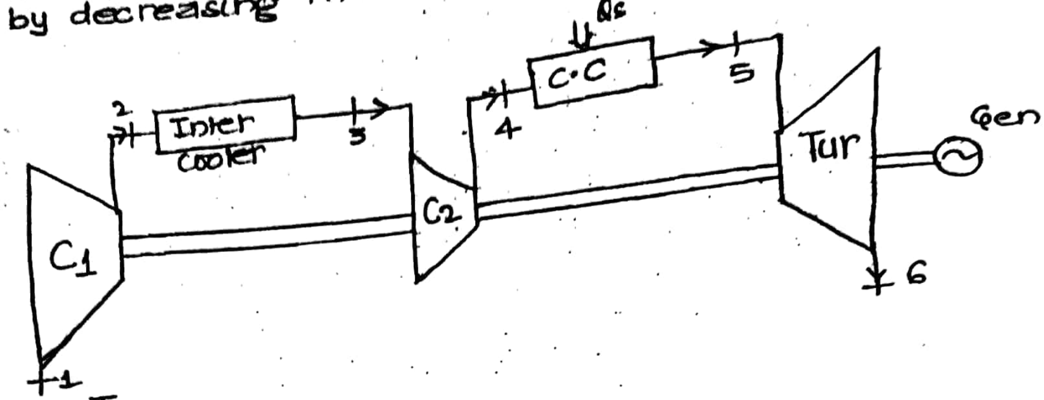
* condition for maximum work o/p in perfect reheating

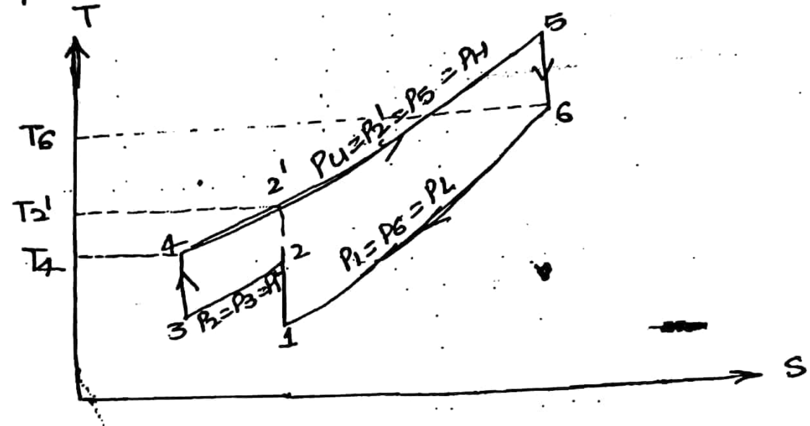


$$\boxed{\begin{array}{l} T_3 = T_5 \\ P_i = \sqrt{P_L \cdot P_H} \\ W_{T_1} = W_{T_2} \\ T_4 = T_6 \end{array}}$$

case III : Intercooling in Gas Turbine

* It is the another way of improving a net work o/p by decreasing the compressor work.





$1-2^1-5-6-1$

$W_T = h_5 - h_6$

$W_C = h_2^1 - h_1$

$W_{net} = W_T - W_C$

$Q_s = h_5 - h_2^1$

$1-2-3-4-5-6-1$

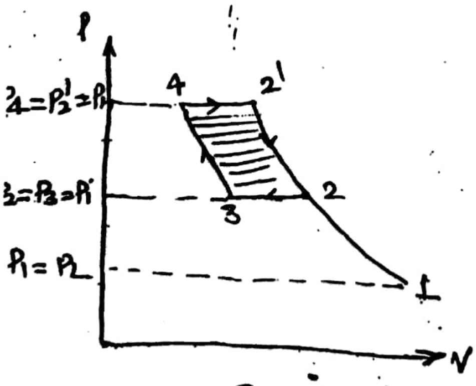
$W_T = h_5 - h_6$ const

$\downarrow W_C = (h_2 - h_1) + (h_4 - h_3)$

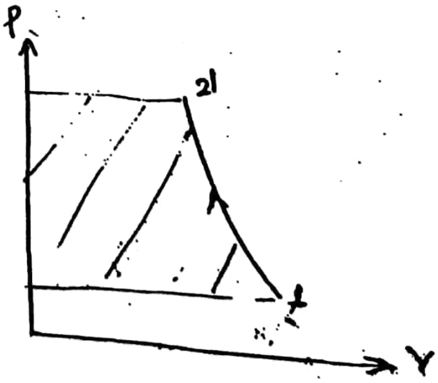
$\uparrow W_{net} = W_T - W_C \downarrow$

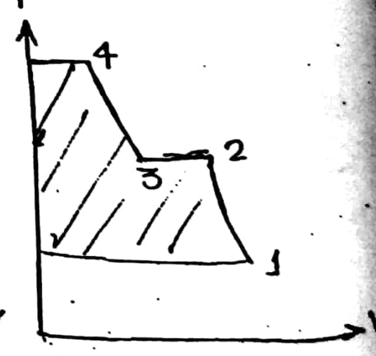
$\uparrow Q_s = h_5 - h_4$

- $W_C \downarrow$
- W_T = const
- $\boxed{W_{net} \uparrow}$
- $Q_s \uparrow$
- $Q_R \uparrow$
- $T_{MA} \downarrow$
- $P_m \downarrow$
- $\boxed{\eta \downarrow}$
- $T_4 < T_2^1$



Serving in work due to intercooling

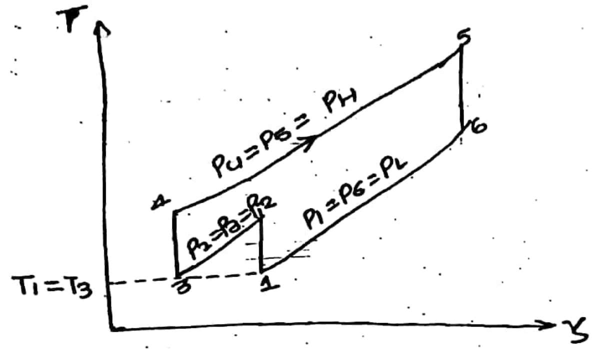




6 oct, 17

Note
* with the use of intercooling the scope of regeneration is increases as the compressor outlet temp decreases ($T_4 < T_2'$)
Theirfore intercooling is generally coupled with regenerator

> condition for minimum work input with perfect intercooling



$$\uparrow W_{net} = W_T - W_C \downarrow$$

$$W_C = W_{C1} + W_{C2}$$
$$= (h_2-h_1)+(h_4-h_3)$$
$$= C_p(T_2-T_1) + C_p(T_4-T_3)$$
$$= C_pT_1\left(\frac{T_2}{T_1}-1\right) + C_pT_3\left(\frac{T_4}{T_3}-1\right)$$
$$= C_pT_1\left[\left(\frac{P_2}{P_1}\right)^x-1\right] + C_pT_3\left[\left(\frac{P_4}{P_3}\right)^x-1\right]$$
$$= C_pT_1\left[\left(\frac{P_i}{P_L}\right)^x-1\right] + C_pT_3\left[\left(\frac{P_H}{P_i}\right)^x-1\right]$$

For perfect inter cooling

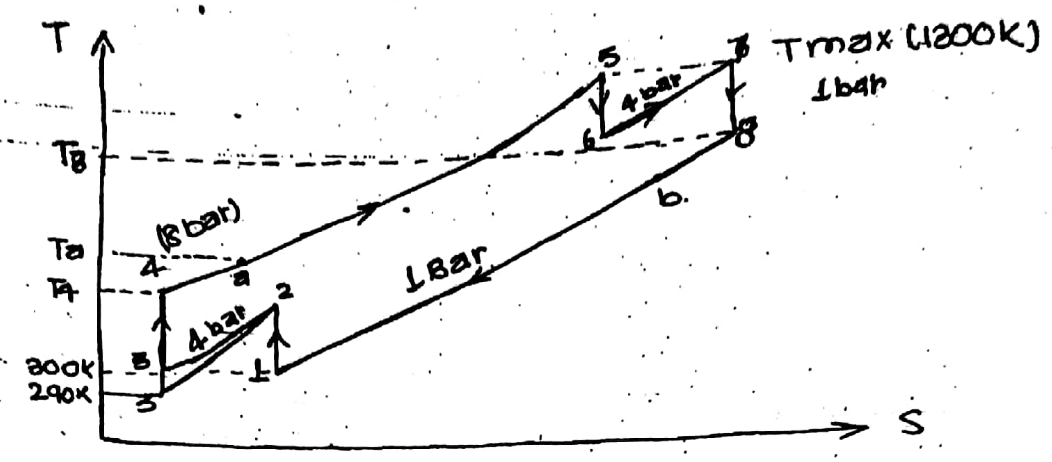
$$W = C_pT_1\left[\left(\frac{P_H}{P_i}\right)^x + \left(\frac{P_i}{P_L}\right)^x - 2\right]$$

$$\frac{dw}{dt} = 0 \qquad *** \boxed{P_i = \sqrt{P_H \cdot P_L}}$$

Note
* condition for maximum work output with perpect intercooling

$$\boxed{\begin{array}{c} T_1 = T_3 \\ P_i = \sqrt{P_H \cdot P_L} \\ W_{C1} = W_{C2} \\ T_2 = T_4 \end{array}}$$

ESE 2008 20 Marks
A regenerative, reheat cycle has air entering at 1 bar & 300K into compresor having intercooling b/w the 2 stage of compression. Air leaving first stage of compressor is cooled upt 290 K & 4 bar press. in inter cooler & subsiquently comp to 8 bar. compressed air leaving 2 compressor passed through regenerator having a efficetive ness of 80%. subsequently combustion produces 1300 K at the inlet of turbine having expansion upto 4 bar & then reheated upto 1300 K before being expanded upto 1 bar exhaust from turbine passes through regenerator consider compres & expansion to be Isentropic & calculate over all efficiency.



$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{x}$$

$$\frac{T_2}{300} = \left(\frac{4}{1}\right)^{\frac{0.4}{1.4}} \quad \therefore T_2 = 445\,K$$

$$\frac{T_4}{290} = \left(\frac{8}{4}\right)^{\frac{0.4}{1.4}} \quad \therefore T_4 = 353\,K$$

$$\frac{1800}{T_6} = \left(\frac{8}{6}\right)^{\frac{0.4}{1.4}} \quad \therefore T_6 = 1066\,K$$

$$\frac{1300}{T_8} = \left(\frac{4}{1}\right)^{\frac{0.4}{1.4}} \quad \therefore T_8 = 874\,K$$

$$e = \frac{T_a - T_4}{T_8 - T_4}$$

$$T_a = 270\,K$$

$$\eta = \frac{W_T + W_C}{Q_S}$$

$$= \frac{[(T_5 - T_6) + (T_7 - T_8)] - [(T_2 - T_1) + (T_4 - T_3)]}{(T_5 - T_a) + (T_7 - T_6)}$$

$$= 58.9\,\%$$

Rankine Cycle
(steam vapour cycle)

→ Reason for using water as a working fluid
* It is chemically stable & harmless substance.
* Available in large quantity.
* cheaper & economic.

→ specific steam consumption.
* It is the amount of steam consumed to produce one KW power output

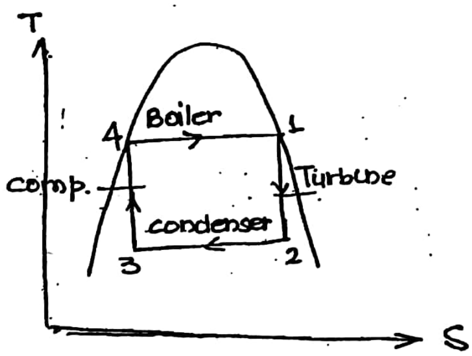
$$\boxed{SSC = \frac{3600}{W_{net}} \quad \frac{kg}{KWHr}}$$

→ Heat Rate
* It is the "rate" of heat input required to produce one "KW" power" output.

$$HR = \frac{\dot{m} \times Q_s}{\dot{m} \times W_{net}} = \frac{Q_s}{W_{net}} = \frac{1}{\frac{W_{net}}{Q_s}} = \frac{1}{\eta}$$

$$\boxed{HR = \frac{1}{\eta}}$$

→ Carnot vapour cycle
* Process 1-2 → Isentropic Expansion
  2-3 → Isothermal heat rejection
  3-4 → Isentropic compresion
  4-1 → Isothermal heat addition

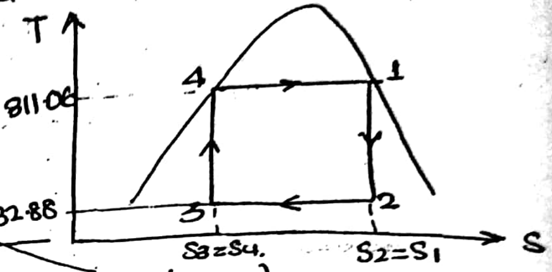


* Drawbacks
• The state of the working fluid at the entry of turbine is vapour & at the exit of turbine is wet mixture which increases the chances of erosion of turbine blades.

- Design and control of partial condenser is not possi[ble]
- It is very difficult to design a compressor handling a mixture of liq & vapour.
- The compression work is having higher value for the carnot cycle which reduces the net work output of the cycle.

Pb: Carnot cycle operates between a source temp of 311.06°c & 10 MPa where as the sink temp is 32.88°c & 5 kpa then determine.

(i) work ratio
(ii) Specific steam consumption
(iii) efficiency of the cycle.



Pr	t°c	hf	hg	sf	sg
		kJ/kg			kg/kg-K
10MPa	311.06	1407.56	2724.7	3.3596	5.6141
5 kPa	32.88	137.88	2561.42	0.4764	8.3951

$$\eta = 1 - \frac{T_L}{T_H}$$

$$= 1 - \frac{32.88+27[3]}{311.06+27[3]}$$

$$= 47\%$$

~~h3 = h4 + x hfg~~

$h_B = h_f$

$h_D = h_g$

$h_C = \cancel{h_f + x h_{fg}}$

h_E

$h_1 = h_{g1} = 2724.7$ kJ/kg

$h_4 = h_{f4} = 1407.56$ kJ/kg

$h_2 = h_f + x_2(h_{fg}) = 137.88 + x_2(2561.42 - 137.88)$

$= 1710$ kJ/kg

$h_3 = h_f + x_3(h_{fg}) = 137.88 + x_3(2561.42 - 137.88)$

$= 1020$ kJ/kg

$S_1 = S_2$

$s_{g1} = s_{f2} + x_2(s_{g2} - s_{f2})$

$5.6141 = 0.4764 + x_2(8.3951 - 0.4764)$

$x_2 = 0.648.$

$S_3 = S_4$

$s_{f3} + x_3(s_{g3} - s_{f3}) = s_{f4}$

$0.4764 + x_3(8.3951 - 0.4764) = 3.3596$

$x_3 = 0.364$

- $\underline{SSC} = \frac{3600}{W_T - W_C}$ ← put kJ

$= 5.74$ kg/kwhr

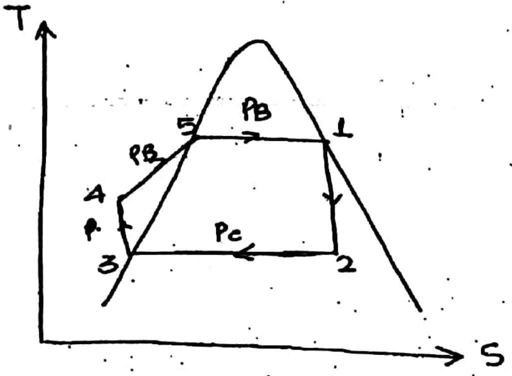
$W_R = 1 - \text{bw}$

$= 1 - \frac{W_C}{W_T}$

$= \underline{0.6182}$

Simple Rankine Cycle.

* Process 1-2 → Isentropic exp.
  2-3 → const pressure HR
  3-4 → pump work
  4-5-1 → const pressure heat addition

*


Isentropic eff. of Pump = ~~Actual work / Ideal work~~

$= \dfrac{\text{Ideal work}}{\text{Actual work}}$

$\text{Actual work} = \dfrac{v_{f_3}(P_4 - P_3)}{\eta_{ise}}$

1-2 → Turbine → $W_T = h_1 - h_2$

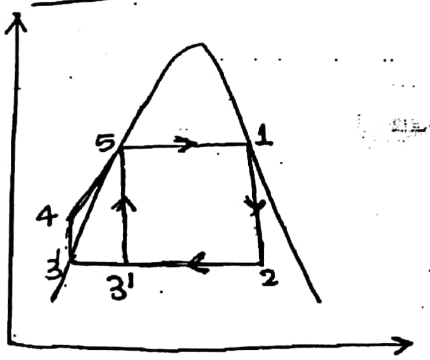
2-3 → condenser → $Q_C = h_2 - h_3$

3-4 → pump → $W_P = h_4 - h_3 = vdp = V_{f\,entry}(P_{high} - P_{low})$

4-5-1 → Boiler → $Q_B = h_1 - h_4$ $\quad = V_{f3}(P_B - P_C) = V_{f3}(P_4 - P_3)$

— Reason for lower efficiency of Rankine cycle w.r.t carnot cycle

The net work output of the rankine cycle is more than the carnot cycle but it does not mean that its efficiency is higher than the carnot because the mean temp of heat addition for carnot cycle is higher value.



carnot cycle → 1-2-3'-5-1

Rankine cycle → 1-2-3-4-5-1.

$$\eta = \frac{O/P}{I/P} = \frac{W_{net}}{Q_S} = \frac{Q_{net}}{Q_S} = \frac{Q_S - Q_R}{Q_S} = 1 - \frac{Q_R}{Q_S}$$

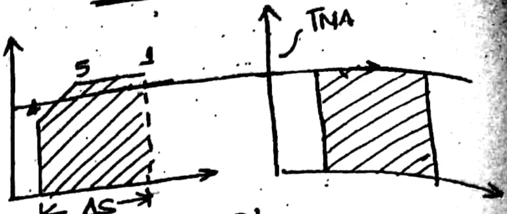
$$\eta = 1 - \frac{T_L \Delta S}{Q_S} \longrightarrow (i)$$

$$\eta = 1 - \frac{T_L \Delta S}{T_{MA} \cdot \Delta S} = \uparrow \eta = 1 - \frac{T_L}{T_{MA}\uparrow}\downarrow$$

Imp

Q:- $\boxed{\begin{gathered}(T_{MA})_{CC} > (T_{MA})_{RC} \\ \eta_{CC} > \eta_{RC} \\ (W_{net})_{RC} > (W_{net})_{CC}\end{gathered}}$

Note

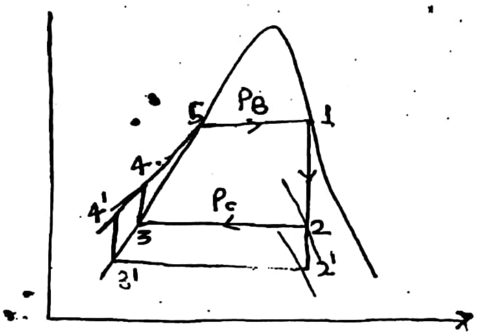


Mean temp of heat addition =
$Q_s = T_{MA} \cdot \Delta s$

→ Methods of Improving the ~~efficiency~~ performance of Rankine cycle.

* The efficiency of the cycle is increased by increasing the mean temp. of heat addition or decreasing the mean temp of heat rejection.

1. case I : Decrease in condenser pressure.



- $P_{C1} \rightarrow 1-2-3-4-5-1$
- $P_{C2} \rightarrow 1-2'-3'-4'-5-1$

- $W_T \uparrow$
- $W_P \uparrow = \int v_f\, dp$
- $W_{net} \uparrow = \uparrow W_T - W_P \uparrow$
- $Q_s \uparrow$
- Q_R can't say
- $T_{MA} \downarrow$
- $T_{MR} \downarrow$
- $\eta \uparrow$

x (Dryness Fract^n) $= \dfrac{m_v}{m_v + m_L}$

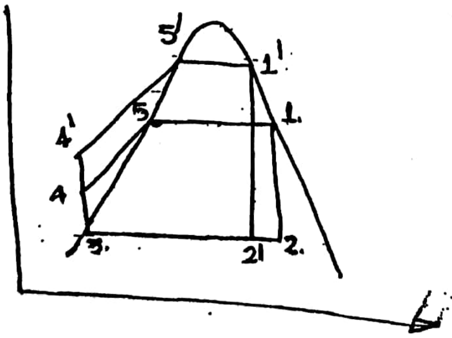
$x_2' < x_2$

$\downarrow x \rightarrow \downarrow$ vapour $\rightarrow \uparrow$ Liquid $\rightarrow \uparrow$ Erosion of Turbine Blade.

(-ve way to increase eff)

2. case II : Increase in Boilure pressure.

*



$P_{B1} \rightarrow 1-2-3-4-5-1$
$P_{B2} \rightarrow 1'-2'-3-4'-5'-1'$

Scanned by CamScanner

- $W_T \uparrow$
- $W_P \uparrow$
- $W_{net} \uparrow \rightarrow W_T - W_P \uparrow$
- Q_S - can't say
- $Q_R \downarrow$
- $T_{MA} \uparrow$
- $\eta \uparrow$
- x (Dryness fraction) $= \frac{m_V}{m_V + m_L}$

$x_1 < x_2$

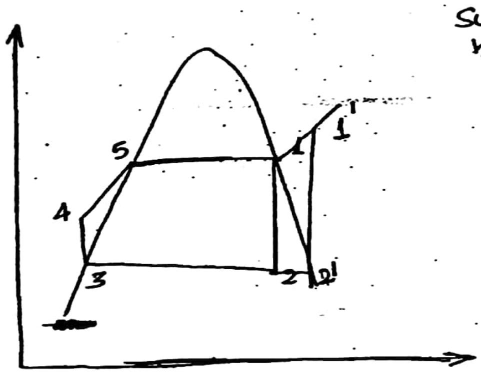
- $x \downarrow \rightarrow$ Vapour $\downarrow \rightarrow$ Liquid $\uparrow \rightarrow \uparrow$ Erosion of Turbine Blade
- $T_{MR} \rightarrow$ const

3. case III : Super heating

* It is a process of increasing the temp at constant pressure above saturated vapour.

Super → 1-2-3-4-5-1

heating → 1'-2'-3-4-5-1'



- $W_T \uparrow$
- $W_P \rightarrow$ const
- $W_{net} \uparrow = W_T - W_P$
- $Q_S \uparrow$
- $Q_R \uparrow$
- $T_{MA} \uparrow$
- $\eta \uparrow$

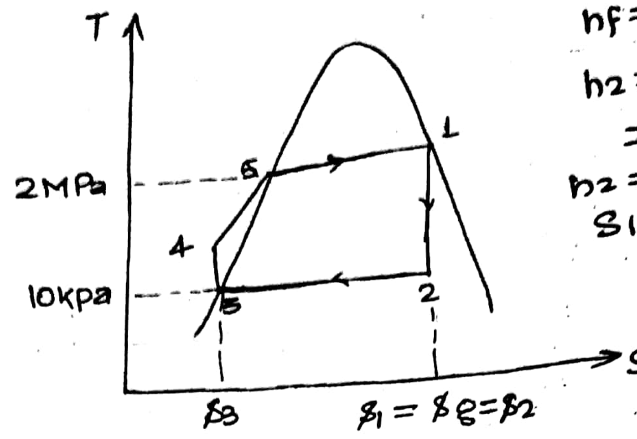
x (Dryness fraction) $= \frac{m_V}{m_V + m_L}$

- $x \uparrow \rightarrow \uparrow$ vapour $\rightarrow \downarrow$ liquid $\rightarrow \downarrow$ Erosion of Turbine Blade

- $x_2' > x_2$

Pb Water is the working fluid in the rankine cycle. Sat. vapour enters the turbine at 2 MPa & condensor pressure is 10 kpa. Pump work is negligible then determine efficiency of the cycle.

Pr	$t\,°C$	V_f (m^3/kg)	V_g (m^3/kg)	h_f (kJ/kg)	h_g (kJ/kg)	S_f (kJ/kg)	S_g (kJ/kg)
2 MPa	212.42	0.00177	0.09963	908.77	2799.5	2.4473	6.3468
10 KPa	45.81	0.00101	14.6735	191.81	2584.63	0.6492	8.15



$h_g = h_1 = 2799.5$ kJ/kg

$h_f = h_5 = 908.77$ kJ/kg

$h_2 = h_{f2} + x(h_g - h_f)$

$= 191.81 + x_2(2584.63 - 191.81)$

$h_2 = 2009$ kJ/kg

$s_1 = s_2 = s_g$

$s_{g1} = s_{f2} + x_2(s_{g2} - s_{f2})$

$6.8468 = 0.6492 + x_2(8.15 - 0.6492)$

$\boxed{x_2 = 0.7595}$

$\eta = \dfrac{W_T - \cancel{W_P}^{\,0}}{Q_S}$ Neglecting pump work

$h_4 - h_3 = 0$

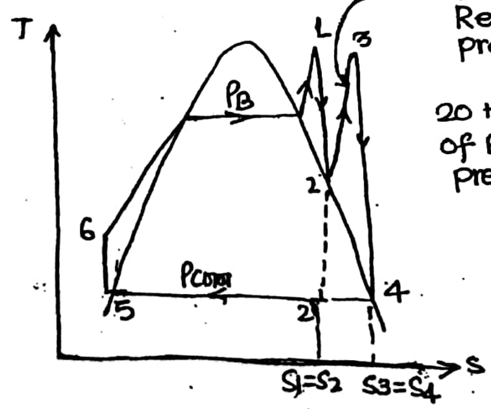
$h_4 = h_3 = 191.81$

$= \dfrac{h_1 - h_2}{h_1 - h_4} = 30.3\%$

case IV : Reheating.

* The main purpose of re-heating is to improve the quality of the working fluid at the exit of turbine which results a decrease in erosion of turbine blades.

* It is use to improve the net work output therefore specific steam consumption reduces which reduce the size of the plant



optimum Reheater pressure
↓
20 to 25% of Boiler pressure

- $W_T \uparrow$
- W_P - const
- $W_{net}\uparrow = \uparrow W_T - W_P$
- $\downarrow SSC = \dfrac{3600}{W_{net}\uparrow}$
- $Q_s \uparrow$
- $QR \uparrow$
- $T_{MA} \uparrow\downarrow$
- $\eta \uparrow\downarrow$
- $x = \dfrac{m_v}{m_v + m_L}$

$x_4 > x_2'$

- $\uparrow x \rightarrow \uparrow$ vapour $\rightarrow \downarrow$ liquid $\rightarrow \downarrow$ erosion of turbine blades.

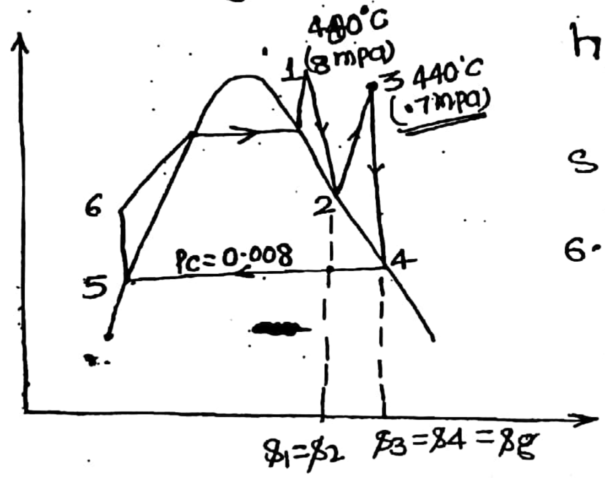
Pb. steam is the working fluid in the cycle, the superheated & reheated steams enters the first stage of turbine in super heated state at 8 mpa, 480°c having h = 3348 KJ/kg, s = 6.6586 KJ/kgK then expanded to a state of 0.7MPa at which $SF = 1.9922$, $Sfg = 6.708$ KJ/kgK, $hf = 697.22$, $hfg = 2066.3$ KJ/kg. It is then reheated to 440 °c super heated state before entereing second stage turbine having h = 3353.3 kJ/kg, s = 7.7571 KJ/kg K where it expands to a pressure of .008 MPa (condenser) having $hf = 173.88$, $Sf = 0.5926$, $hfg = 2403$, $Sg = 8.2287$. Assuming that the specific volume of saturated liquid at .008 mpa is 1.0085×10^{-3} m³/kg. cycle generates 100mw power output then determine.

(i) efficiency.

(ii) Mass flow rate of the steam in KJ/s.

(iii) rate of heat transfer from the condensing steam as it passes through condenser in mw.

Soln :-



$h_2 = (h_f + x h_{fg})$

$=$

$h_1 = 3348$ KJ/kg
$h_3 = 3353.3$
$h_5 = 173.88$

$S_1 = S_2$
$= Sf_2 + x_2 Sfg_2$

$6.8586 = 1.9922 + x_2 \times 6.708$

$x_2 = 0.69$

$h_2 = hf_2 + x_2(hg_2 - hf_2)$
$= 697.22 + 0.69\,(2066.3 - 697.22)$

$h_2 = 1649$ KJ/kg

$S_3 = S_4$

$S_3 = Sf_4 + x_4 Sfg_4$

$7.7571 = 0.5926 + x_4 \times 8.2287$

$x_4 = 0.987$

$h_4 = hf_4 + x_4 hfg$
$= 173.88 + 0.987 \times 2403$
$= 2428$ KJ/kg

$h_6 = h_5 + v_{f5}(P_6 - P_5)$
$= 173.88 + 1.0085 \times 10^{-3}(8 - 0.008) \times 10^3$
$= 182$ KJ/kg

$\eta = \dfrac{W_T - W_C}{Q_S}$ excludes that pump work

$W_T = (h_1 - h_2) + (h_3 - h_4)$

$W_P = (h_6 - h_5) + (h_3 - h_2)$

$= 53.5\%$

(ii) $P = \dot{m}\,(W_T - W_P)$

100×10^3 KJ/sec $= \dot{m}\,[(h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5)]$

$\dot{m} = 38.10$ kg/sec

(iii) $Q_C = \dfrac{(h_4 - h_5) \times \dot{m}}{1000}$

$= 85.8$ MW.

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* Principle stirling gas cycle

case V :- Regeneration

* In the ideal regeneration cycle the efficiency is increased by exchanging the heat internally b/w the expanding steam in the turbine and the feed water entering the boiler.

* Ideal regenerative cycle is not practically possible because the heat exchanger can not installed inside the turbine as their more chances of erosion of turbine blades (last stage

* In the practical power plant high temp steam is extracted from the intermediate stages of the turbine which is utilised to increase the temp of feed water before entering the boiler

-> Types

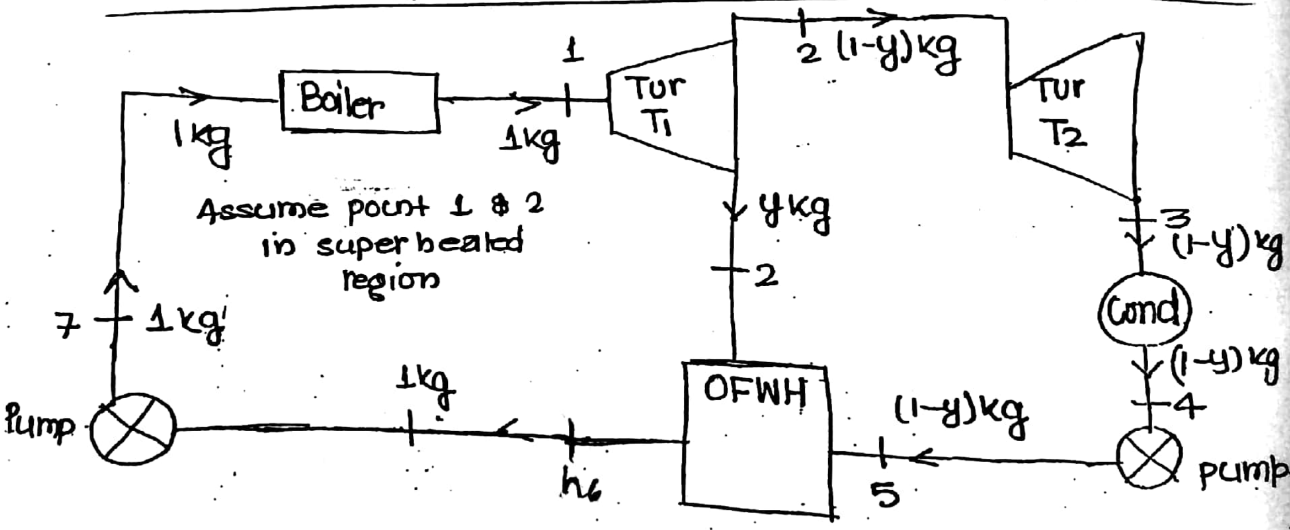
(i) open feed water heater

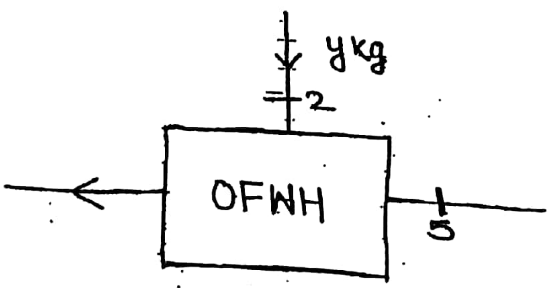
* It is a mixing chamber where the high temp steam is extracted from turbine is utilised to increase the temp of feed water. The pressure of working fluid should be same to minimise the loss of energy.

* Therefore more no of pumps are required.

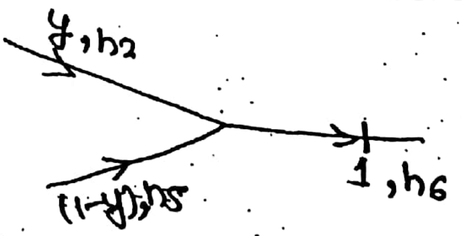
eg :- Deaerator

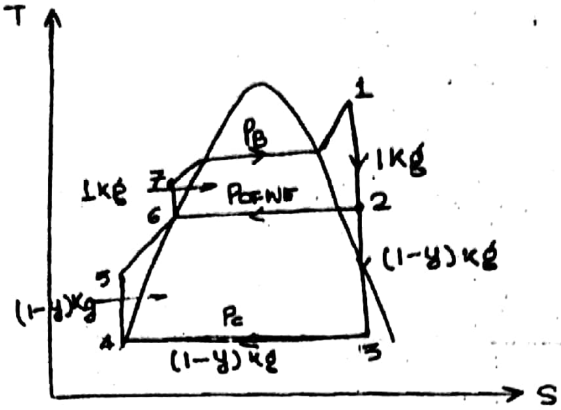
* The purpose of deaerator to remove the air mixed with the feed water before entering the boiler.





$$\boxed{y h_2 + (1-y) h_5 = h_6}$$





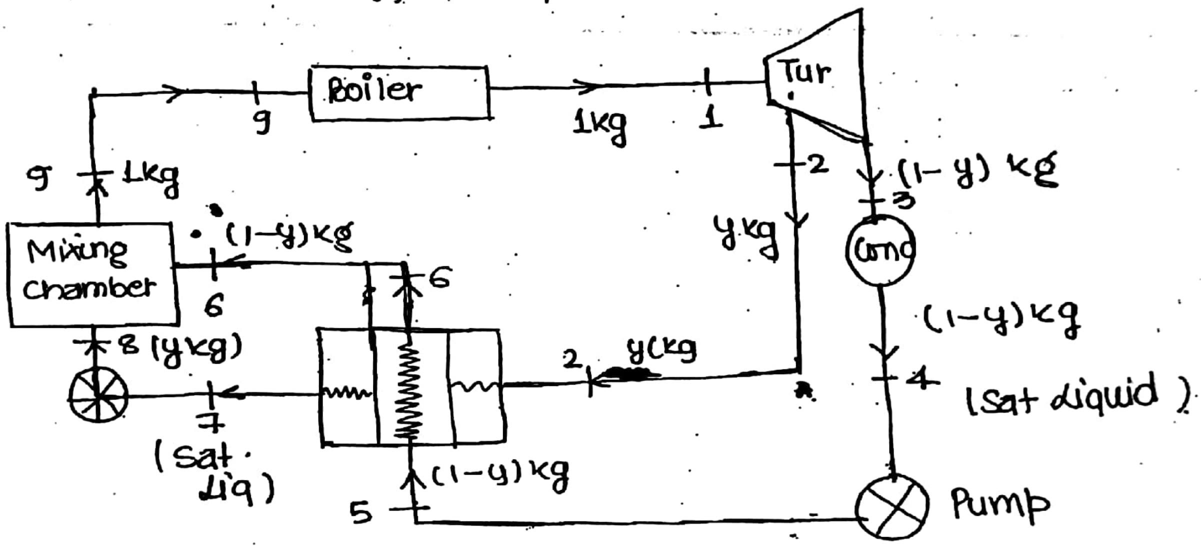
$$W_T = (h_1 - h_2) + (1-y)(h_2 - h_3)$$

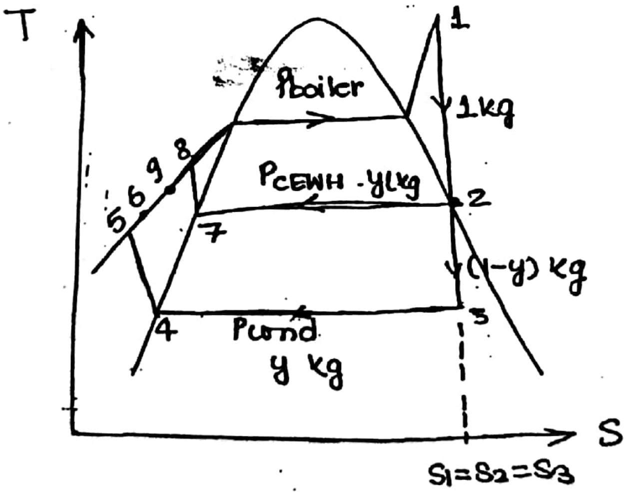
$$W_P = (1-y)(h_5 - h_4) + 1(h_7 - h_6)$$

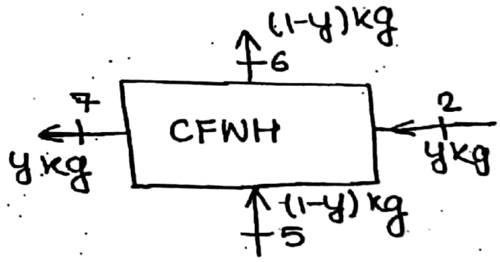
$$Q_S = h_1 - h_7$$

(ii) Closed feed water heater

* In this type the high temp steam rejects its heat energy to the feed water without mixing so these can be kept at different pressure.





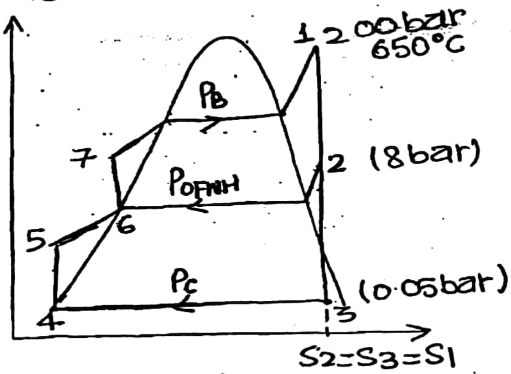


$$\boxed{y h_2 + (1-y) h_5 = (1-y) h_6 + y h_7}$$

Effects of Regeneration

* cycle ~~effieiency~~ increases at heat rate decreases
* For the same mass flow rate turbine work decreases
* condenser size descreases as the mass flow rate decreases
* It provides convinent method of removing the air from the condenser.
* For the same power o/p steam flow rate increases therefore it requires large size boiler.

Pb A regenerative Rankine cycle having steam entry at 200 bar, 650°c & leaving at ·05 bar considered the feed water heater to be <u>open type</u> & calculate the efficiency of the plant when the feed water heater is operating at 8 bar



at 0.005 bar

$h_f = 137.89$ } kJ/kg

$h_{fg} = 2423.7$

$s_f = 0.4764$ } kJ/kgK

$s_{fg} = 7.9187$

$v_f = 0.001005\ m^3/kg$

at 8 bar

$h_f = 721.11$ } kg/kg

$h_{fg} = 2048$

$s_f = 2.0462$ } kJ/kgK

$s_{fg} = 4.6166$

$v_f = 0.00115\ m^3/kg$

$h_1 = h_g = 3675.3\text{kg}$. — $h_1 = 3675.3$, $s_1 = 6.6582$

$h_2 = h_{fg} \therefore = h_{fg} = h_g - h_f$

$h_g = h_{fg} + h_f$

$h_g = 2048 + 721.11$

$= 2769.11\ kJ/kg$.

$S_1 = S_2$

$S_1 = S_{f2} + x_2 S_{fg2}$

$6.6582 = 2.0462 + x_2(4.616)$

$x = 0.998 \simeq 1$

$h_3 = h_{f3} + x_3 h_{fg3}$

$= 137.89 + 0.78 \times 2423.7$

$= 2029\ \frac{kJ}{kg}$

$S_1 = S_3$

$s_1 = s_{g3} + x_3(s_{fg})$

$6.6582 = 0.4764 + x_3(7.9187)$

$x_3 = 0.78$.

$h_4 = h_{f4} = 137.89$

$h_6 = h_{f6} = 721.11$

$h_5 = h_4 + v_{f4}(P_5 - P_4)$

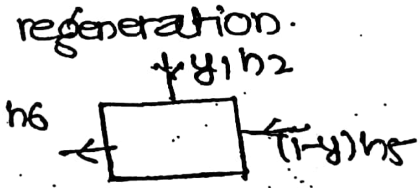
$= 137.89 + 0.001005\ (8 - 0.05) \times \underline{100}$

$= 138.67\ kJ/kg$

$h_7 = h_6 + v_{f6}(P_7 - P_6) \times 100$

$= 721.11 + 0.00115\ (200 - 8) \times 100$

For regeneration.



$y h_2 + (1-y) h_5 = 1 \times h_6$

$y = 0.221$

$$W_T = W_{T1} + W_{T2}$$
$$= 1 \times (h_1 - h_2) + (h_2 - h_3)(1-y)$$
$$W_P = (1-y)(h_5 - h_4) + 1 \times (h_7 - h_6)$$
$$Q_s = h_1 - h_7$$
$$\eta = \frac{W_T - W_P}{Q_s} = 49.8\%$$

Compressor

- Compressor

* Compressor is used to increase the pressure of the gases or vapour

- classification of

1. on the basic of principle of operation
   a. positive displacement compressor

* In this type of compressor the pressure of the gas is increased by decreasing the volume in a confined cylinder i.e rise in pressure is obtained by displacing the working fluid to a lower volume

eg :- Reciprocating compressor

b Rotary or Dynamic

* In this a high kinetic energy is imparted to the working fluid by the means of rotating element called as rotor & this high K.E is converted into pressure energy in the diffusor
* eg :- centrifugal comp, Axial flow compressor

2. On the basic of pressure developed
   a. low pressure - upto 10 bar
   Midium pressure - 10 to 80 bar
   high pressure - above 80 bar

3. On the basic of volume handling capacity
   a. small handling capacity - upto 9 m^3/min
   medium : 9 to 3000 m^3/min
   large more than 3000 m^3/min