POWER PLANTS

REFERENCES:

1-Power Plant System Design by KamW.Li and A.PaulPriddy

2- Thermal Engineering by R.K.Rajput

3- Power Plant Technology by El-Wakil M.M.

4- Power Generation Handbook by Philip Kiameh

5- Thermodynamic Fundamentals by Eistop

Types of power plants:

1. Oil fired power plant.
2. Gas fired power plant.
3. Nuclear power plant.
4. Hydro power plant.
5. Solar power plant.
6. Wind turbine power plant.
7. Coal fired power plant.
8. Biomass power plant.
9. Tidal power plant.

Thermodynamic principles:

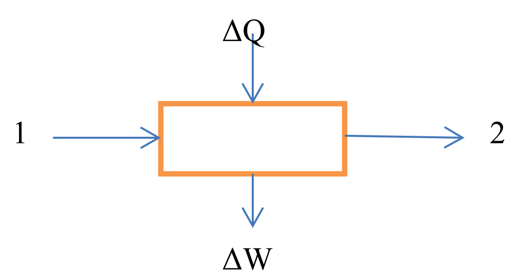
1st law of thermodynamics:

$$PE_1 + KE_1 + IE_1 + FE_1 + \Delta Q = PE_2 + KE_2 + IE_2 + FE_2 + \Delta W_{SF} \quad \text{--------1}$$

PE: Potential Energy (Z)

KE: Kinetic Energy ($V^2/2g$)

IE: Internal Energy (U)



FE: Flow Energy (PV)

ΔQ: Net Heat Added ($Q_2 - Q_1$)

ΔW: Net steady flow mechanical work done by the system ($W_1 - W_2$)

$$H = U + PV \qquad \text{-----------------2}$$

$$h = u + Pv \qquad \text{------------------3}$$

$$\Delta Q = \Delta U - \Delta W \qquad \text{------------------4}$$

The 2[nd] law of thermodynamic:

$$\sum \Delta m_i s_i - \sum \Delta m_e s_e + \sum \frac{\Delta Q_{C.V.}}{T} + \sum \Delta\sigma = \Delta(ms)_{c.v.} \qquad \text{------------5}$$

$\sum \frac{\Delta Q_{C.V.}}{T}$: Entropy increase by heat transfer

$\sum \Delta\sigma$: Entropy increase due to internal irreversibility (such as friction) $= \frac{LW}{T}$

$\sum \Delta m_i s_i$: Entropy associated with the mass flow entering the C.V.

$\sum \Delta m_e s_e$:Entropy associated with the mass flow leaving the C.V.

$\Delta(ms)_{c.v.}$: Entropy change in the C.V.

For SSSF, $\Delta(ms)_{c.v.} = 0$, Eq. 5 becomes:

$$\sum \Delta m_i s_i - \sum \Delta m_e s_e + \sum \frac{\Delta Q_{C.V.}}{T} + \sum \Delta\sigma = 0 \qquad \text{-------------6}$$

For closed system, no mass convection, Eq.5 becomes:

$$\frac{\Delta Q_{sys.}}{T} + \sum \Delta\sigma = \Delta(ms)_{sys.} \qquad \text{---------------7}$$

For reversible process; the internally generated entropy becomes zero:

$$\therefore \Delta(ms)_{sys.} = \frac{\Delta Q_{sys.}}{T} \qquad \text{----------------8}$$

Turbine process:

Adiabatic, no change in kinetic potential energy

$$\Delta m_i h_i - \Delta m_e h_e = \Delta W_{c.v.} \qquad \text{--------------------------9}$$

Or: $w_t = \dfrac{\Delta W_{c.v.}}{\Delta m_{.i}} = h_i - h_e$ -----------------------10

$\eta_t = \dfrac{h_i - h_e}{h_i - h_{es}}$ ---------------------11

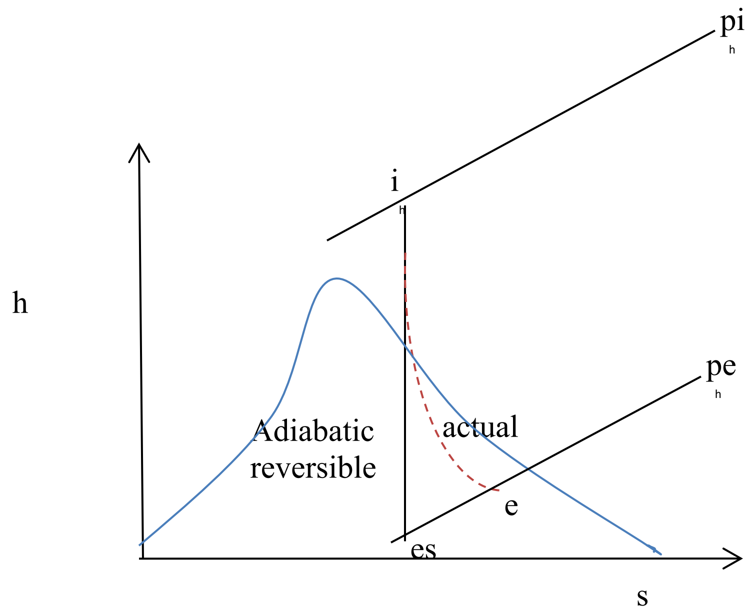
For gas (or air):

$w_t = C_p(T_i - T_e)$ ----------------------12

$\eta_t = \dfrac{T_i - T_e}{T_i - T_{es}}$ ----------------------13

$\dfrac{T_{es}}{T_i} = \left(\dfrac{P_e}{P_i}\right)^{\frac{k-1}{k}}$ -----------------------14

$k = \dfrac{C_p}{C_v}$, k=1.4 for air.



Irreversible expansion process in steam turbine

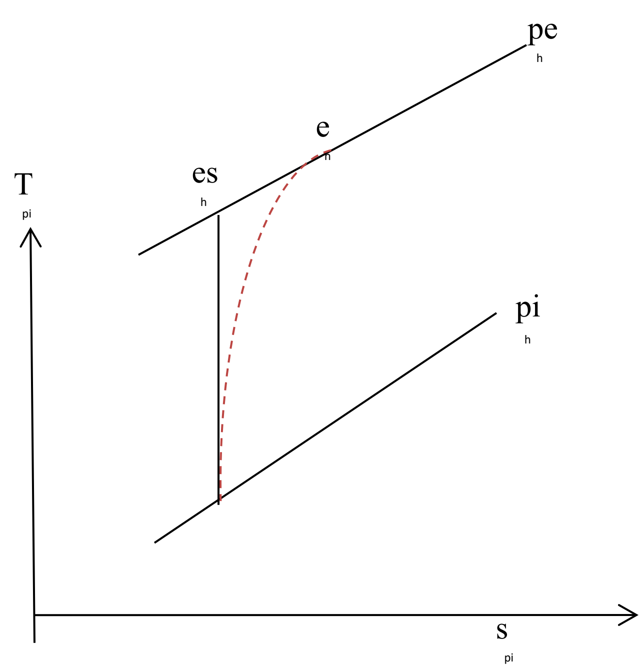
Compressor or pump process:

For compressor:

$\eta_c = \dfrac{T_i - T_{es}}{T_i - T_e}$ ----------------15

$w_c = C_p(T_i - T_e)$ ----------------16

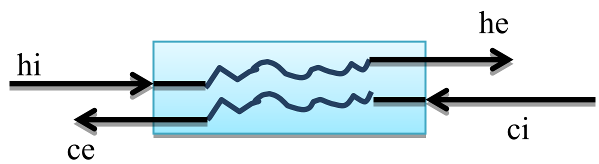
Irreversible compression process



For pump:

$w_p = -\int_i^e v\, dp = -\, v_i(p_e - p_i)$ --------------------17

Heat exchanger:



$\Delta m_h h_{hi} + \Delta m_c h_{ci} = \Delta m_h h_{he} + \Delta m_c h_{ce}$ or

$\Delta m_h(h_{hi} - h_{he}) = \Delta m_c(h_{ce} - h_{ci})$ ---------------18

For ideal gas:

$h_i - h_e = \int_i^e C_p dT$ ------------19

$\Delta Q = \Delta m_h(h_{hi} - h_{he}) = \Delta m_c(h_{ce} - h_{ci}) = \Delta m_c C_{pc}(T_{ce} - T_{ci})$ --------20

For boiler:

$\Delta Q_f = \frac{\Delta Q_s}{\eta_b} = \frac{\Delta m_s}{\eta_b}(h_{se} - h_{si})$ ----------------------21

ΔQ_f: heat released by fuel

ΔQ_s: heat gained by steam

Throttling process:

$h_i = h_e$ -------------------------22

For ideal gas $T_i = T_e$ (isothermal process)

Power system Economics:

Production cost "C"

C= A kW + B kWh

A: The constant costs for building and ground ….. etc. . per kW in ID.

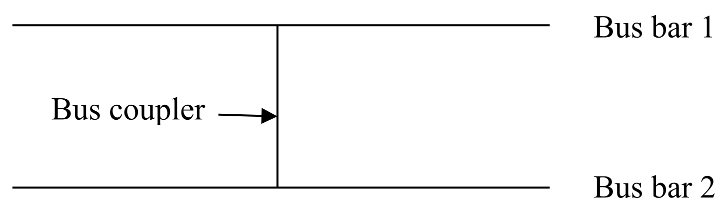
B: Operation cost per kWh included the fuel , water treatment etc.

Connected load : the total summation of connected load for all instruments connect to the system .

Firm power : it's the power that assumed to be continuously existed at all time ever at emergency cases.

Cold reserve : It's the generation reserve capacity that is standby but not in work or operation .

Spinning reserve : It's the generation reserve capacity for bus coupler which is standby for loading.



Demand factor : the ratio of maximum load to the summation of connected loads . It is less than one unit.

Demand factor $=\frac{actual\ max.load}{connected\ load}$

Load factor : the ratio of average load to max load through limited time (day, month, year)

Load factor $= \frac{average\ load\ through\ limited\ time\ (d,m,y)}{\max load\ through\ this\ time\ (peak\ load)}$

Average load : the ratio of (kWh) units (actual) through limited time (day, month, year) to the hours at this time .

Daily average load $= \frac{No.of\ units\ at\ this\ day\ kWh}{24\ h\ at\ this\ day}$

Diversity factor : the ratio of singular max. loads of consumers to max. loads when they connected as all to the station. It is larger than one unit.

Coincide factor : the inverse of diversity factor.

Plant capacity factor : the ratio of actual produced power to max. power which may be produced by power plant according to the capacity of it (installed).

Capacity factor $= \frac{actual\ energy\ produced}{max.energy\ produced\ according\ to\ the\ design} = \frac{average\ demand}{installed\ capacity}$

Plant use factor :the ratio of actual energy produced to the plant capacity multiplied by No. of operation hours.

$$plant\ use\ factor = \frac{actual\ energy\ produced}{plant\ capacity\ \times\ hours\ of\ operation}$$

Example 1:

In Hydro Power plant the head of water is **30m,** the average flow rate of falling water **7m³/s.** Find the load power that must be feed it when the turbine efficiency is **90%** and generation efficiency is **95%.**

Solution:

H=30m Q=7m³/s η_t = 90% η_g=95%

Power load output = power of falling water * η =(ρ*g*H*Q* η)/1000

= (9.81*30*1000*7*0.9*0.95)/1000

=1761.39 kwh

Example 2:

The peak load of power plant is **20MW** and the load factor is **60%** and the plant capacity factor is **48%** and the plant use factor is **80%**. Find:

1. daily output power .
2. plant reserve capacity.
3. max. power output at day of continuously operated.
4. max. power output at day of it operated with max load.

1)

Load factor $= \frac{average\ load}{peak\ load}$

$0.6 = \frac{av.load}{20\ MW}$

Average load = 12 MW

Average load $= \frac{KWh}{hours}$

12 MW $= \frac{KWh}{24\ hour}$

Energy produced (kWh) = 288000 kWh = 288 MWh

2) $$plant\ capacity\ factor = \frac{average\ load}{installed\ capacity}$$

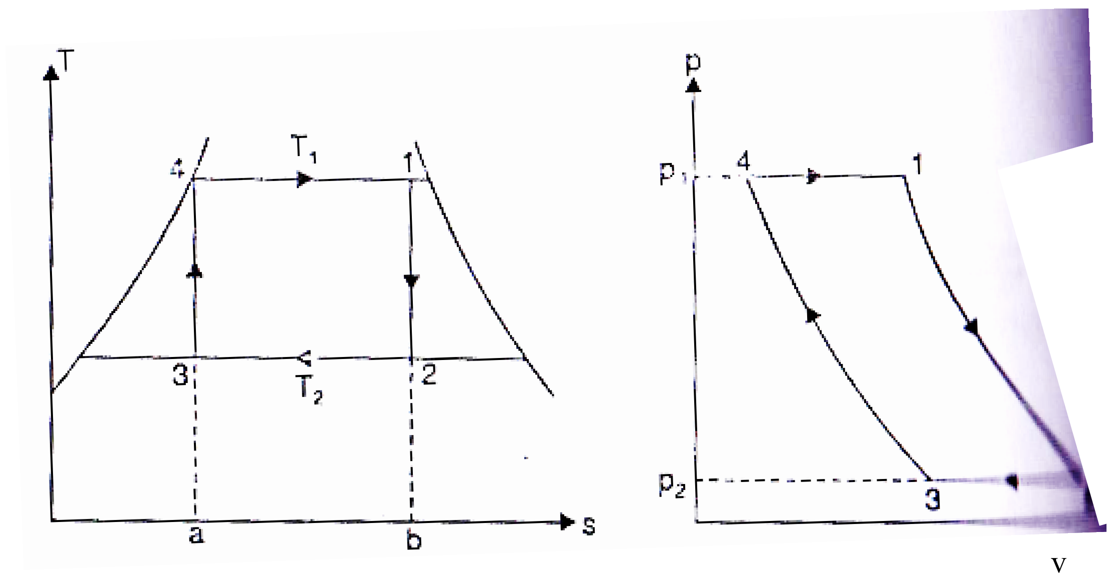
$$installed\ capacity = \frac{12\ MW}{0.48} = 25\ MW$$

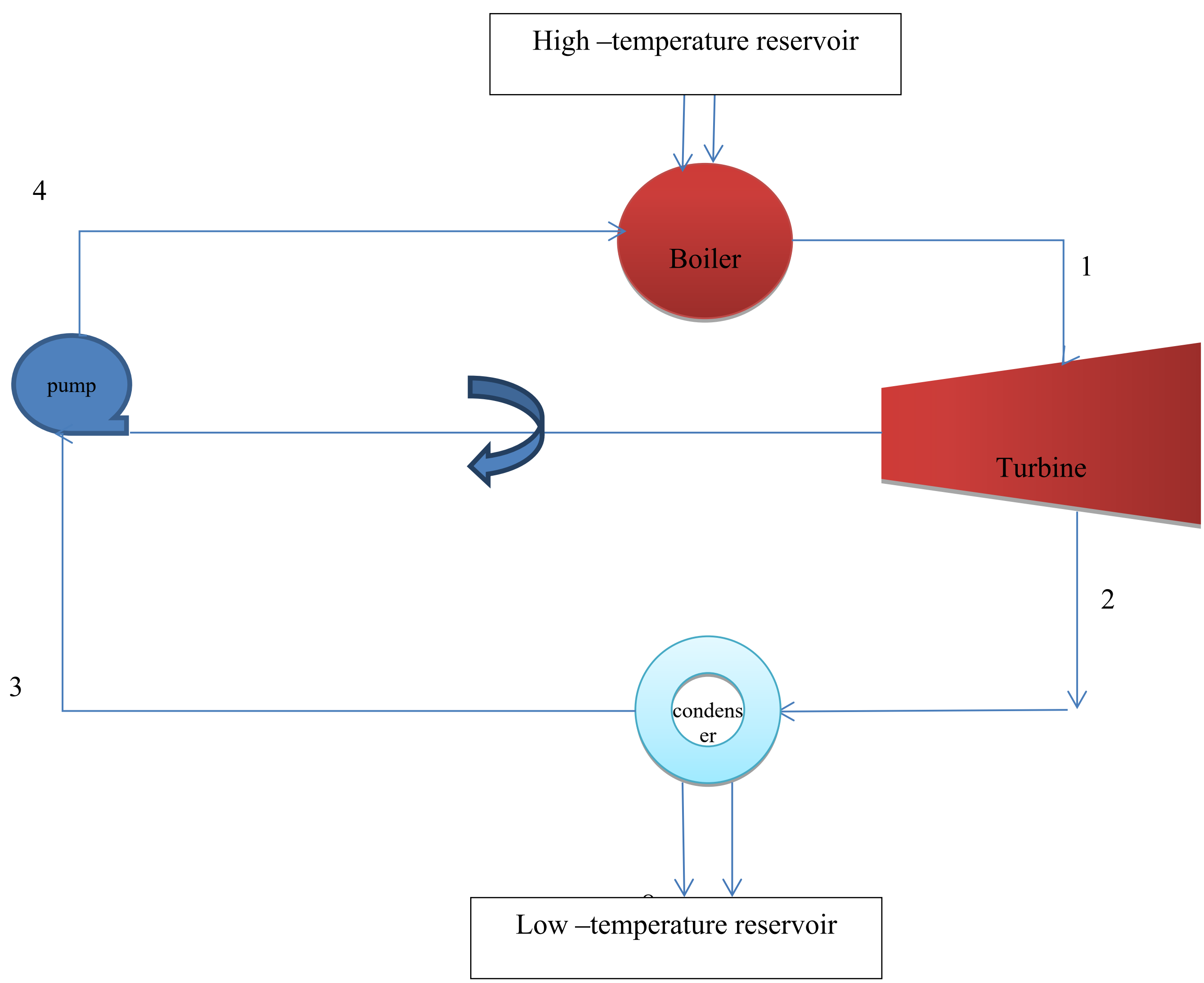
Reserve capacity = installed capacity – peak load = 25-20= 5MW

3) Max. power = installed capacity *24h = 25*24=600MWh

4) Plant capacity =$\frac{energy\ produed}{used\ factor} = \frac{288000}{0.8} = 360000\ kWh$

Carnot cycle:





1- The process 4-1 reversible and isothermal

Heat transferred from the high temperature reservoir.

2- The process 1-2 reversible and adiabatic

The working substance expands and has its temperature decreased to that of the low temperature reservoir.

3- The process 2-3 reversible and isothermal

Heat is transferred to the low temperature reservoir.

4- The process 3-4 reversible and adiabatic

The working substance is compressed and has its temperature increased back to that of the high temperature reservoir.

Heat supplied at constant temperature $T_1\,(4-1) = area\ (4-1-a-b-4) =$

$$= T_1\,(s_1 - s_4) = T_1\,(s_2 - s_3)$$

Heat rejected at constant temperature $T_2\,(2-3) = area\ (2-3-b-a-2)$ $=$

$$= T_2(s_2 - s_3)$$

Net work done =heat supplied – heat rejected

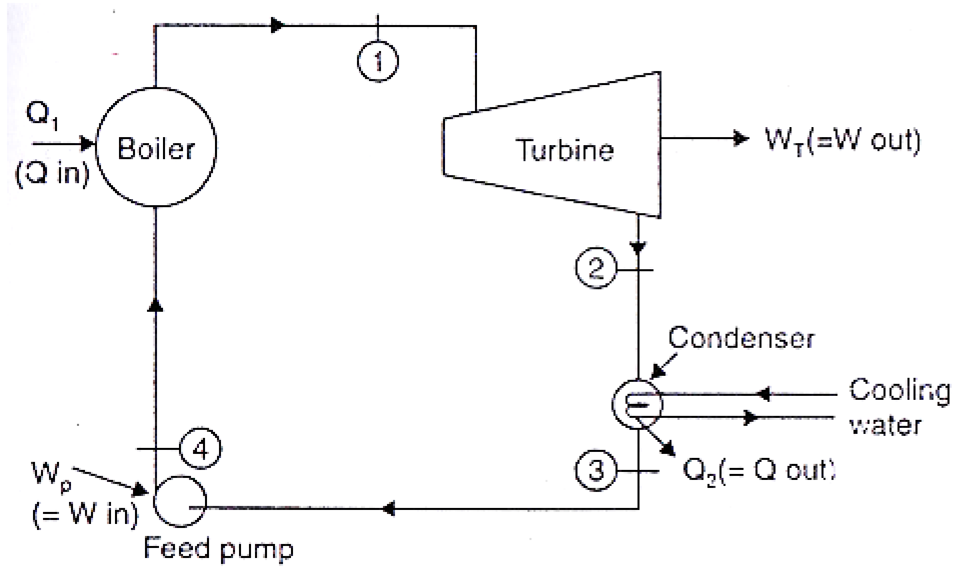
$$=T_1\,(s_2 - s_3) - T_2(s_2 - s_3)$$

$$=(T_1 - T_2)\,(s_2 - s_3)$$

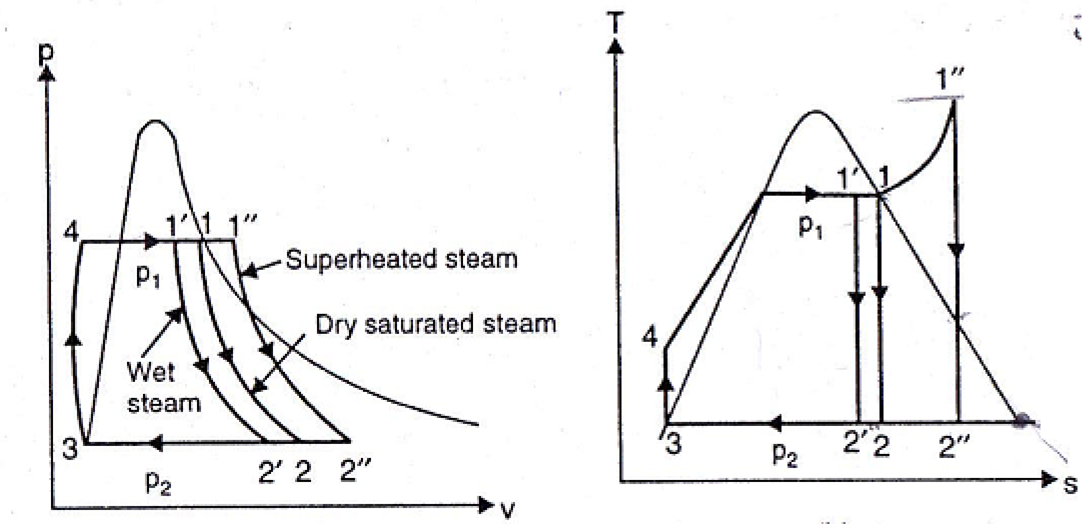
Carnot cycle efficiency $\eta_{cy} = \frac{work\ done}{heat\ supplied} = \frac{(T_1 - T_2)\,(s_2 - s_3)}{T_1\,(s_2 - s_3)} = \frac{T_1 - T_2}{T_1}$

Rankine Cycle:

1- The process 4-1 constant pressure, transfer of heat in the boiler.
2- The process 1-2 reversible adiabatic, expansion in the turbine (or steam engine).
3- The process 2-3 constant pressure, transfer of heat in the condenser.
4- The process 3-4 reversible adiabatic, pumping process in the feed pump.



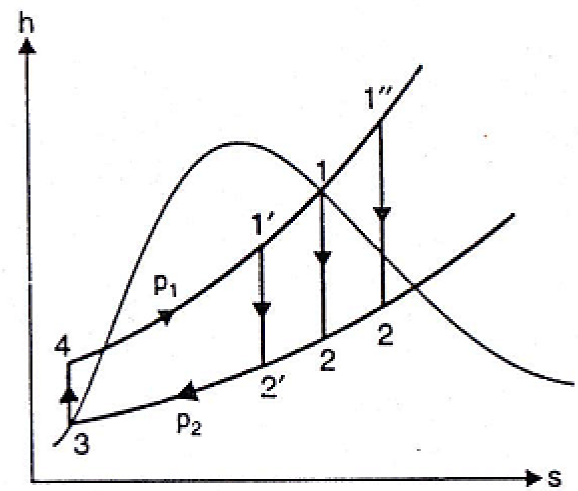
1
Q1
(Q in)
Boiler
Turbine
WT(=W out)
2
Condenser
Cooling
water
4
Wp
(= W in)
3
Q2(= Q out)
Feed pump



p
4
1′ 1 1″
Superheated steam
p1
Dry saturated steam
Wet
steam
3
2′ 2 2″
p2
v
T
1″
1′ 1
p1
4
3
p2
2′ 2
2″
s

(a)

(b)



h
1″
1
1′
p1
4
2
2
2′
3
p2
s

Applying S.F.E.E. (Steady Flow Energy Equation):

1- For boiler $q_1 = h_1 - h_4$

2- For turbine $W_t = h_1 - h_2$

3- For condenser $q_2 = h_2 - h_3$

4- For pump $W_p = h_4 - h_3$

$$\eta_{Rankine} = \frac{W_{net}}{q_1} = \frac{W_t - W_p}{q_1} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_3) - (h_4 - h_3)}$$

For feed pump (reversible adiabatic compression)

$T\,ds = dh - v\,dp$

$ds = 0 \quad , \quad dh = v\,dp$

$\Delta h = v\,\Delta p$ (since change in specific volume is negligible)

$h_4 - h_3 = v_3(p_4 - p_3)$

$(h_4 - h_3)$ small quantity in comparison with turbine work W_t

$$\eta_{Rankine} = \frac{(h_1 - h_2)}{(h_1 - h_3)}$$

For incompressible liquid (pump) (v=constant):

$(h_4 - h_3) = v_3(p_4 - p_3)$

$dQ = du + p\,dv$ -----------1

$h = u + pv$

$dh = du + d(pv)$

$dh = du + p\,dv + v\,dp$

$du = dh - p\,dv - v\,dp$ ------------2

Sub. Eq.1 in 2:

$dQ = dh - p\,dv - v\,dp + p\,dv = dh - v\,dp$

For isentropic process:

$0 = dh - v\,dp$

$dh = v\,dp$

$$\int_3^4 dh = \int_3^4 v\,dp$$

$$h_4 - h_3 = v\int_3^4 dp = v(p_4 - p_3) = pump\ work$$

$$\text{Efficiency Ratio} = \frac{cycle\ efficiency}{Rankine\ efficiency}$$

$isentropic\ efficiency = \frac{actual\ work}{isentropic\ work}$ for expansion process

$isentropic\ efficiency = \frac{isentropic\ work}{actual\ work}$ for compression process

$$turbine\ isentropic\ efficiency = \frac{W_{1\acute{2}}}{W_{12}} = \frac{h_1 - h_{\acute{2}}}{h_1 - h_2}$$

$$Work\ ratio = \frac{net\ work}{gross\ work}$$

Specific steam consumption (SSC) is the steam flow in kg/h required to develop 1 kW

$$W * SSC = 1 * 3600\ kJ/h$$

$$SSC = \frac{3600}{W_{net}} kg/kWh$$

Effect of operating conditions on Rankine cycle efficiency:

The Rankine cycle efficiency can be improved by:

1- Increasing the average temperature at which heat is supplied.

2- Decreasing the temperature at which heat is rejected.

This can be achieved by:

I. Increasing boiler pressure. It has been observed that by increasing the boiler pressure (other factors remaining the same) the cycle tends to rise and reaches a maximum value.

II. Superheating. All other factors remaining the same, if the steam is superheated before allowing it to expand the Rankine cycle efficiency may be increased. The use of superheated steam also ensures longer turbine blade life because of the absence of erosion from high velocity water particles that are suspended in wet vapor.

III. Reducing condenser pressure. The thermal efficiency of the cycle can be amply improved by reducing the condenser pressure (hence by reducing the temperature at which heat is rejected), especially in high vacuums. But the increase in efficiency is obtained at the increased cost of condensation apparatus.

Also it improved by:

1- Regenerative feed heating.

2- Reheating of steam.

3- Water extraction.

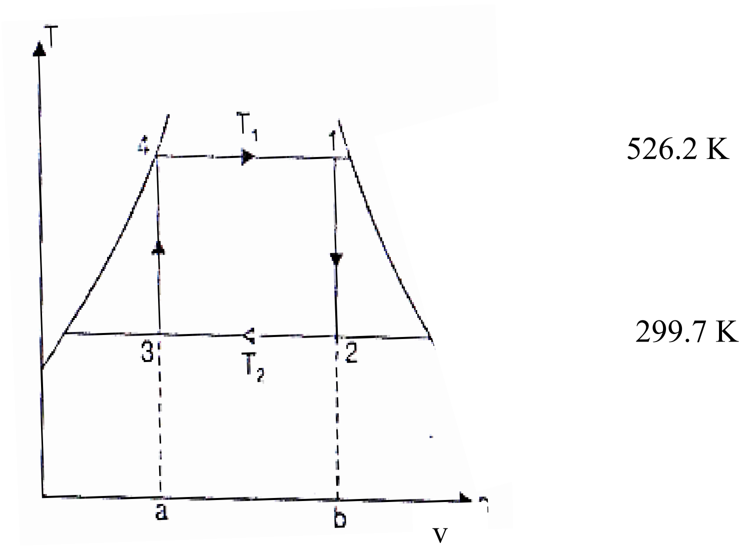
4- Using binary vapor.

Example 1:A steam power plant operates between a boiler pressure of 42 bar and a condenser pressure of 0.035 bar. Calculate for these limits the cycle efficiency, the work ratio, and the specific steam consumption:

a) For a Carnot cycle using wet steam.
b) For a Rankine cycle with dry saturated steam at entry to the turbine.
c) For the Rankine cycle of b when the expansion process has an isentropic efficiency of 80%.

Solution:

a)



A Carnot cycle shown:

T_1= saturated temperature at 42 bar= 253.2+273=526.2K

T_2= saturated temperature at 0.035 bar= 26.7+273=299.7K

$$\eta_{Carnot} = \frac{T_1 - T_2}{T_1} = \frac{526.2 - 299.7}{526.2} = 0.432 \quad or\ 43.2\%$$

Heat supplied=h_1-h_4=h_{fg} at 42bar=1698 kJ/kg

$\eta_{Carnot} = \frac{W}{Q} = 0.432$ or W=0.432*1698=734 kJ/kg

S_1=S_2, from tables h_1=2800kJ/kg, S_1=S_2= 6.049kJ/kg.K

S_2= 6.049= S_{f2}+ x_2 S_{fg2}= 0.391+x_2 *8.13

x_2= 0.696

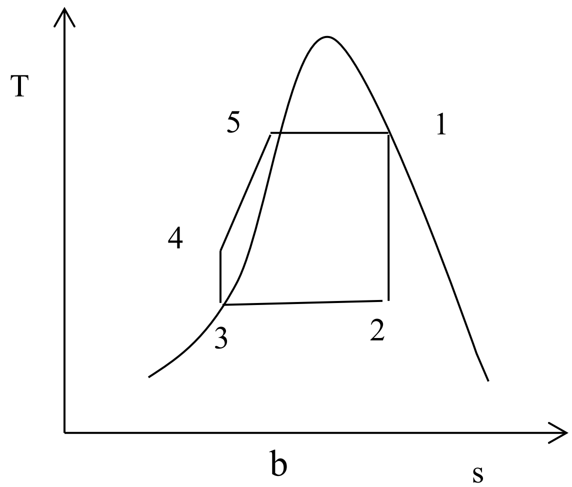
h_2= h_{f2}+ x_2 h_{fg2}= 112+0.696 *2438= 1808 kJ/kg

W_{12}=h_1-h_2= 2800 - 1808= 992kJ/kg

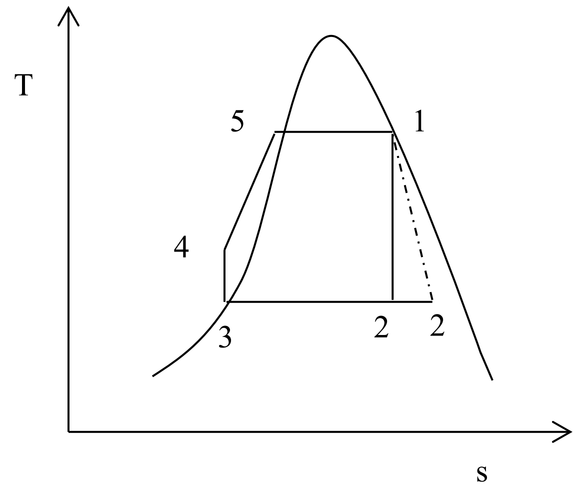
$$Work\ ratio = \frac{net\ work}{gross\ work} = \frac{734}{992} = 0.739$$

Specific steam consumption SSC= $\frac{3600}{W} = \frac{3600}{734} = 4.9 kg/kWh$

b)



b



c

The Rankine cycle:

From table and from a:

h_1= 2800kJ/kg, h_2=1808kJ/kg, h_3=h_f at0.035bar= 112kJ/kg

v=v_f at 0.035bar

pump work= $v_f\,(p_4 - p_3) = 0.001 * (42 - 0.035) * \frac{10^5}{10^3} = \frac{4.2kJ}{kg}$

$$W_{12} = h_1 - h_2 = 2800 - 1808 = \frac{992kJ}{kg}$$

$$\eta_{Rankine} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_3) - (h_4 - h_3)} = \frac{992 - 4.2}{(2800 - 112) - 4.2} = 0.368 = 36.8\%$$

$$work\ ratio = \frac{net\ work}{gross\ work} = \frac{992 - 4.2}{992} = 0.996$$

$$ssc = \frac{3600}{W} = \frac{3600}{992 - 4.2} = \frac{3.64kg}{kWh}$$

c) Irreversible expansion process

$$isentropic\ efficciency = \frac{h_1 - h_{2'}}{h_1 - h_2} = \frac{W_{12'}}{W_{12}}$$

$$0.8 = \frac{W_{12'}}{992} W_{12'} = 0.8 * 992 = 793.6kJ/kg$$

$$cycle\ efficiency = \frac{(h_1 - h_{\acute{2}}) - (h_4 - h_3)}{heat\ supplied} = \frac{793.6 - 4.2}{(2800 - 112) - 4.2} = 0.294 = 29.4\%$$

$$work\ ratio = \frac{W_{1\acute{2}} - pump\ work}{W_{1\acute{2}}} = \frac{793.6 - 4.2}{793.6} = 0.995$$

$$ssc = \frac{3600}{794.6 - 4.2} = 4.56 kg/kWh$$

4

3

Rankine cycle with superheat:

The average temperature at which heat is supplied in the boiler can be increased by superheating the steam. Usually the dry saturated steam from the boiler drum is passed through a second bank of smaller bore tubes within the boiler. This bank is situated such that it is heated by the hot gases from the furnace until the steam reaches the required temperature.

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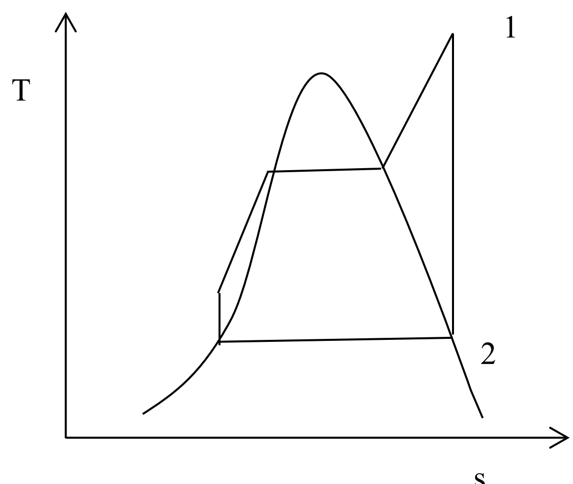

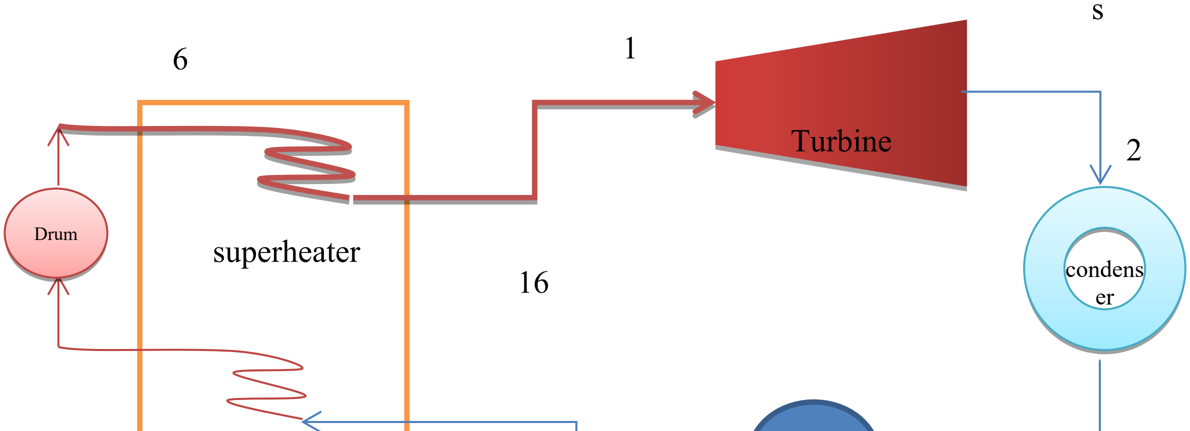

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4

5 3

Example 2: Compare the Rankine cycle performance of Example 1 with heat that obtained when the steam is superheated to 500 ^oC. Neglect the feed pump work.

Solution:from tables, at 42bar, h_1=3442.6kJ/kg, s_1=s_2=7.066kJ/kg. K

$s_2= s_{f2}+ x_2 \ s_{fg2}$

$$0.391 + x_2 * 8.13 = 7.066 \therefore \ x_2 = 0.821$$

$h_2= h_{f2}+ x_2 \ h_{fg2}$= 112+(0.821* 2438)= 2113 kJ/kg

From tables h_3=112 kJ/kg

$W_{12} = h_1 - h_2 = 3442.6 - 2113 = 1329.6 \ kJ/kg$

Neglect the feed pump term,

$Heat \ supplied = h_1 - h_3 = 3442.6 - 112 = 3330.6 \ kJ/kg$

$$Thermal \ efficiency = \frac{h_1 - h_2}{h_1 - h_3} = \frac{1329.6}{3330.6} = 0.399 = 39.9\%$$

$$ssc = \frac{3600}{W_{12}} = \frac{3600}{1329.6} = 2.71 \ kg/kWh$$

To calculate cooling load of water for condenser for both examples by the law:

$ssc * (h_2 - h_3)$

1- Dry saturated steam

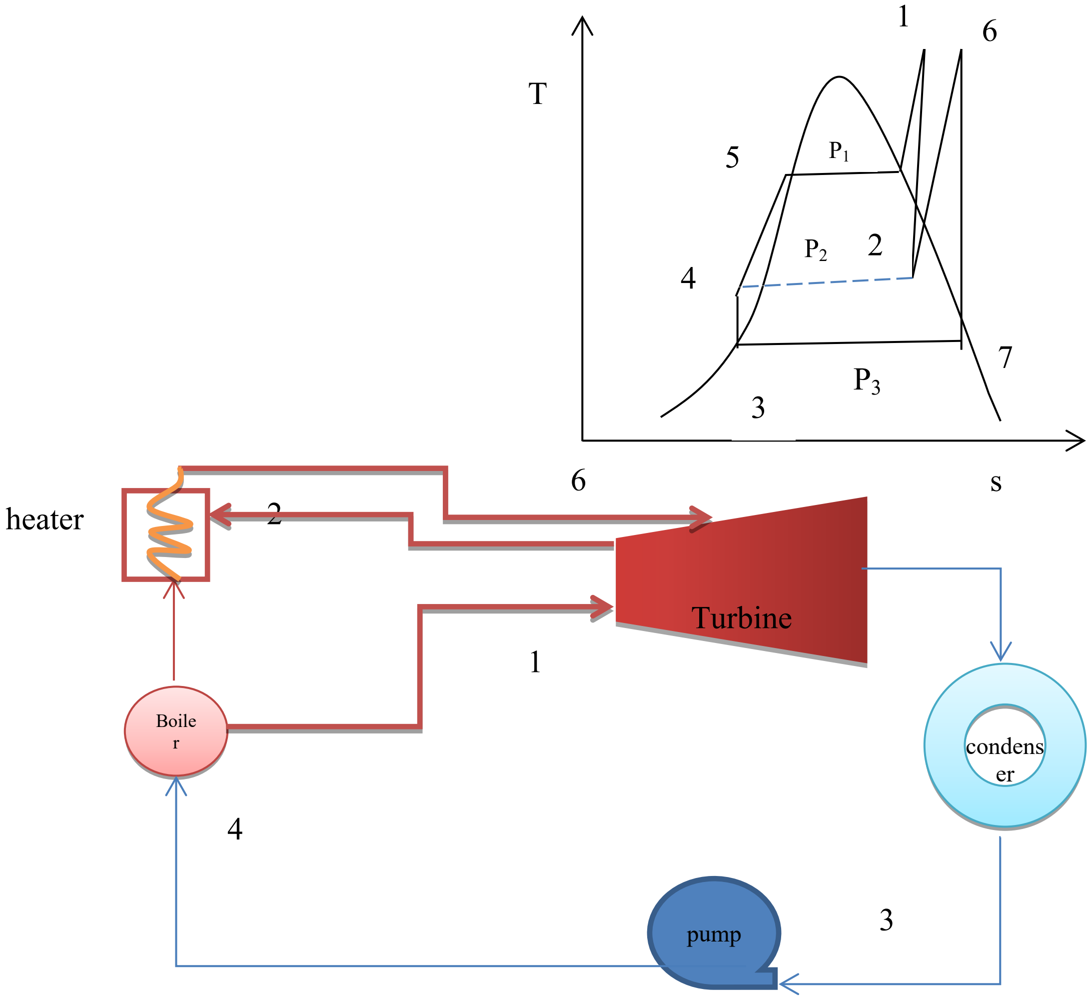
Condenser heat load= 3.64(1808-112)=6175 (kJ/h)/kW

2- with superheated steam

Condenser heat load= 2.71(2113-112)= 5420 (kJ/h)/kW

Reheat cycle:

It is desirable to increase the average temperature at which heat is supplied to the steam, and also to keep the steam as dry as possible in the lower pressure stages of the turbine. The wetness of exhaust should be no greater than 10%. High boiler pressures are required for high efficiency, but the expansion in one stage can result in exhaust steam which is wet. The 5 7 exhaust steam condition can be improved most effectively by reheating the steam, the expansion being carried out in two stages or more.



1-2 represents isentropic expansion in the high pressure turbine.

6-7 represents isentropic expansion in the low pressure turbine.

The steam is reheated at constant pressure in process 2-6.

The reheat can be carried out by returning the steam to the boiler and passing it through a special bank of tubes which are situated in the proximity of the superheat tubes, or in a separate reheater situated near the turbine. This encourages use of high pressure (100-250bar) or high temperature (500-600 °C) boilers. This improves the cycle efficiency by about 5% for 85/15 bar cycle.

$Heat\ supplied = Q_{451} + Q_{26}$

$Q_{451} = h_1 - h_3$ (Neglecting the feed pump work)

$Q_{26} = h_6 - h_2$

$Work\ output = W_{12} + W_{67}$

$W_{12} = h_1 - h_2$, $W_{67} = h_6 - h_7$

$$Cycle\ efficiency = \frac{W_{12} + W_{67}}{Q_{451} + Q_{26}} = \frac{(h_1 - h_2) + (h_6 - h_7)}{(h_1 - h_3) + (h_6 - h_2)}$$

Advantages of Reheating:

1. There is an increased output of the turbine.
2. Erosion and corrosion problems in the steam turbine are eliminated/ avoided.
3. There is an improvement in the thermal efficiency of the turbines.
4. Final dryness fraction of steam is improved.
5. There is an increase in the nozzle and blade efficiencies.

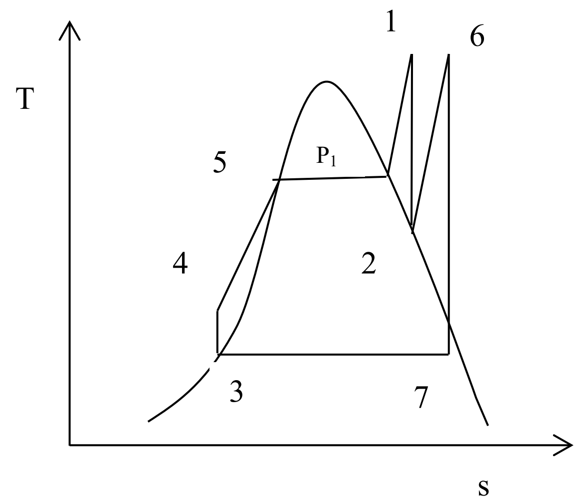
Disadvantages of Reheating:

1- Reheating requires more maintenance.

2- The increase in thermal efficiency is not appreciable in comparison to the expenditure incurred in reheating.

Example3: Calculate the new cycle efficiency and specific steam consumption if reheat is included in the plant of Ex.2. The steam conditions at inlet to the turbine are 42bar and 500°C and the condenser pressure is 0.035bar as before. Assume that the steam is just dry saturated on leaving the first turbine, and is reheated to its initial temperature. Neglect feed pump work.

Solution:



From tables:

h_1=3442.6kJ/kg, h_2=2713kJ/kg, h_6=3487kJ/kg (at 2.3bar, 500°C),

$s_6=s_7=$

$x_7=$

h_7=2535kJ/kg

from tables: h_3=112kJ/kg

$$Turbine\ work = (h_1 - h_2) + (h_6 - h_7)$$

$$= (3443 - 2713) + (3487 - 2535) = 1682\ \ kJ/kg$$

$$heat\ supplied = (h_1 - h_3) + (h_6 - h_2)$$

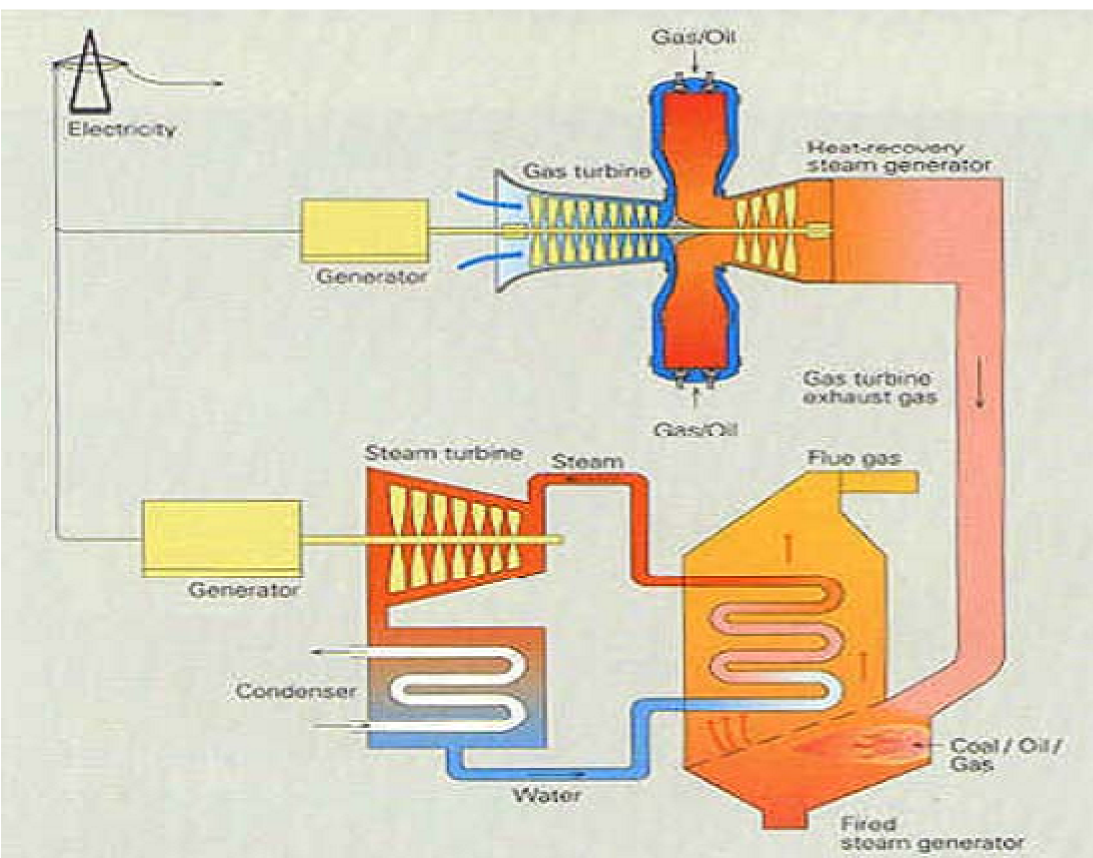
$$= (3443 - 112) + (3487 - 2713) = 4105\ \ kJ/kg$$

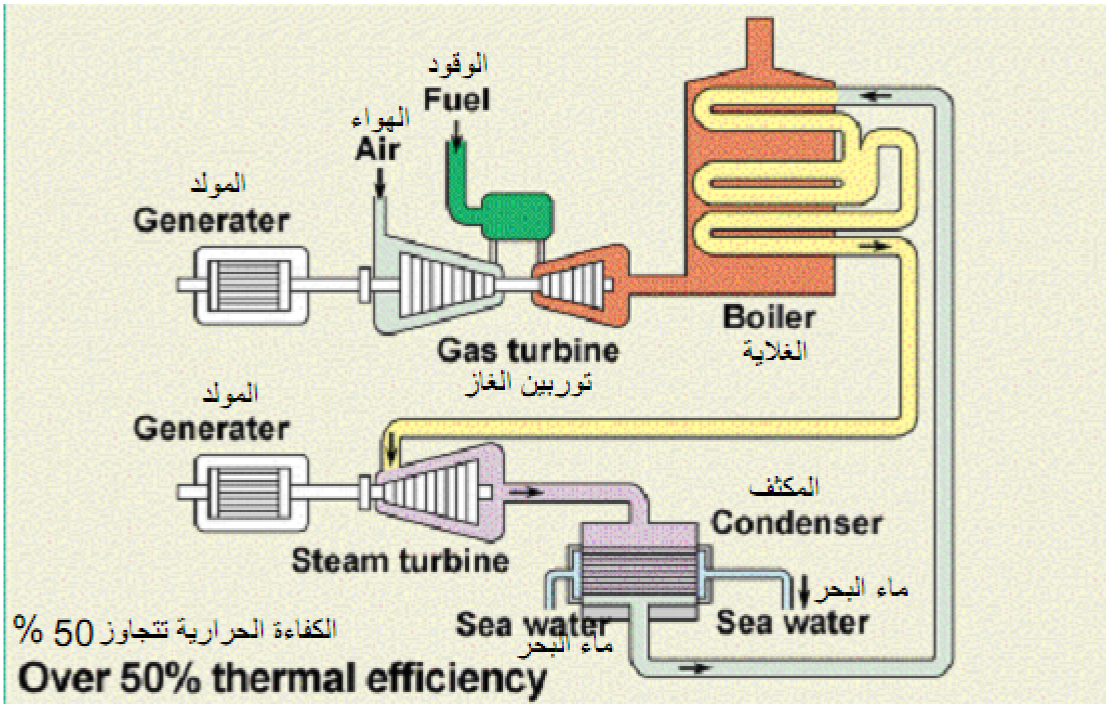
$$cycle\ efficiency = \frac{1682}{4105} = 0.41\ or\ 41\%$$

$$SSC = \frac{3600}{W} = \frac{3600}{1682} = 2.14\ kg/kWh$$

Home Work:

Steam at a pressure of 15bar and 250°C is expanded through a turbine at first to a pressure of 4bar. It is then reheated at constant pressure to the initial temperature of 250°C and is finally expanded to 0.1bar. Estimate the work done per kg of steam flowing through the turbine and amount of heat supplied during the process of reheat. Compare the work output when the expansion is directed from 15bar to 0.1bar without any reheat. Assume all expansion process to be isentropic.

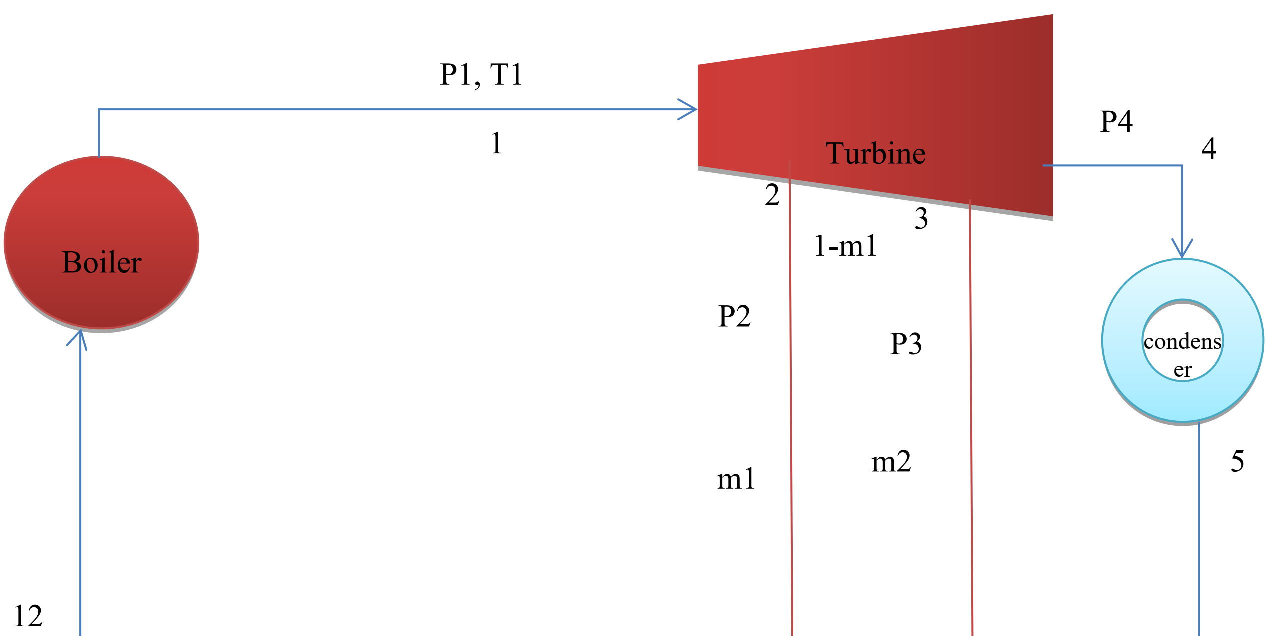


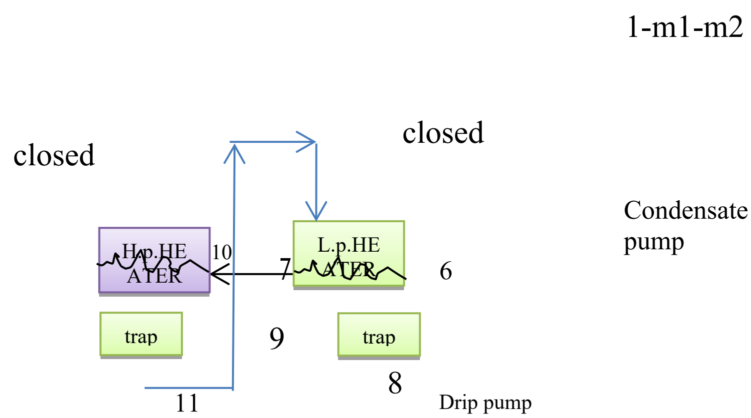


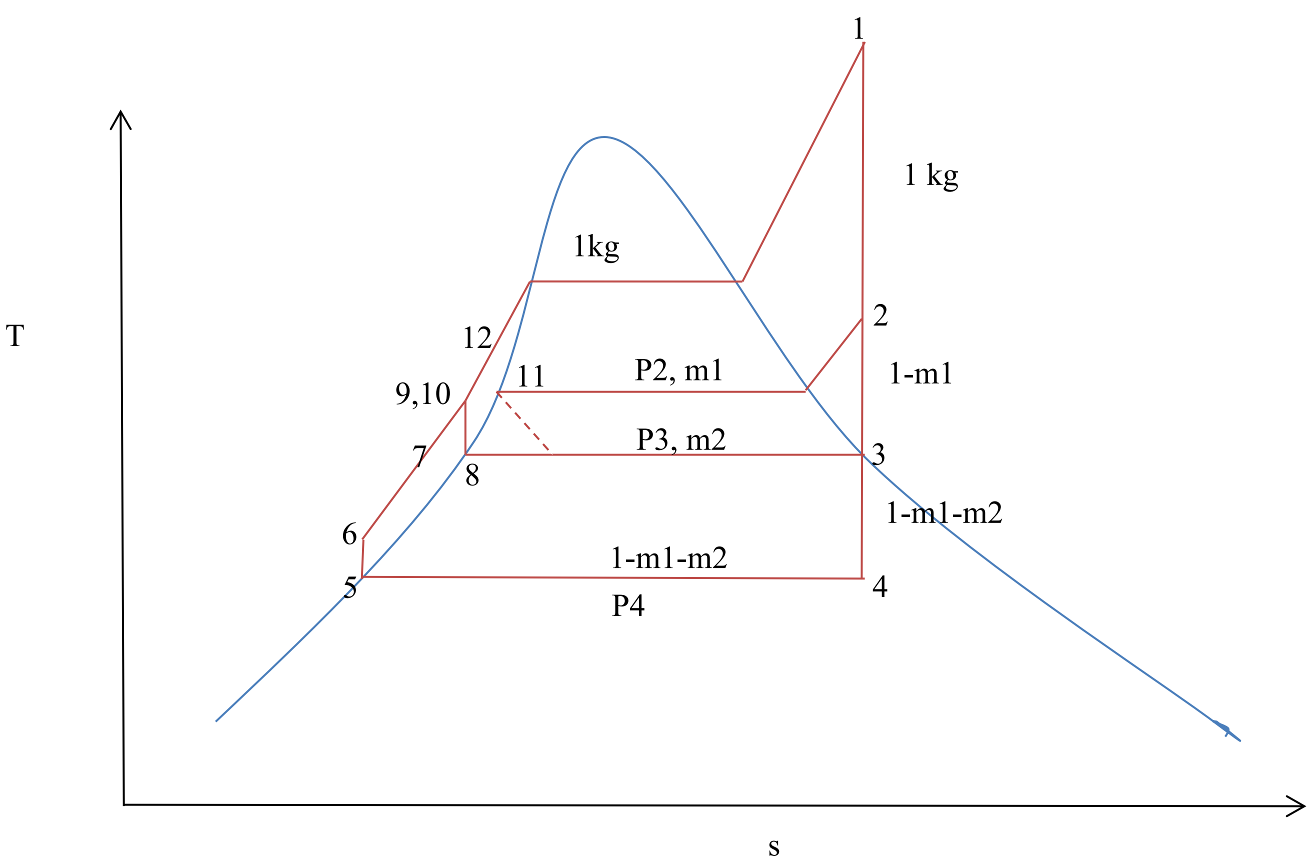
FEEDWATER HEATERS:

There are two types: open (or contact) and closed heaters. In open heater, the extracted steam is allowed to mix with feedwater and both leave the heater at a common temperature. In closed heater, the fluids are kept separate and are not allowed to mix together.

The condensate (saturated water at the steam extraction pressure), sometimes called the heater drip, then passes through a trap into the next lower pressure heater.



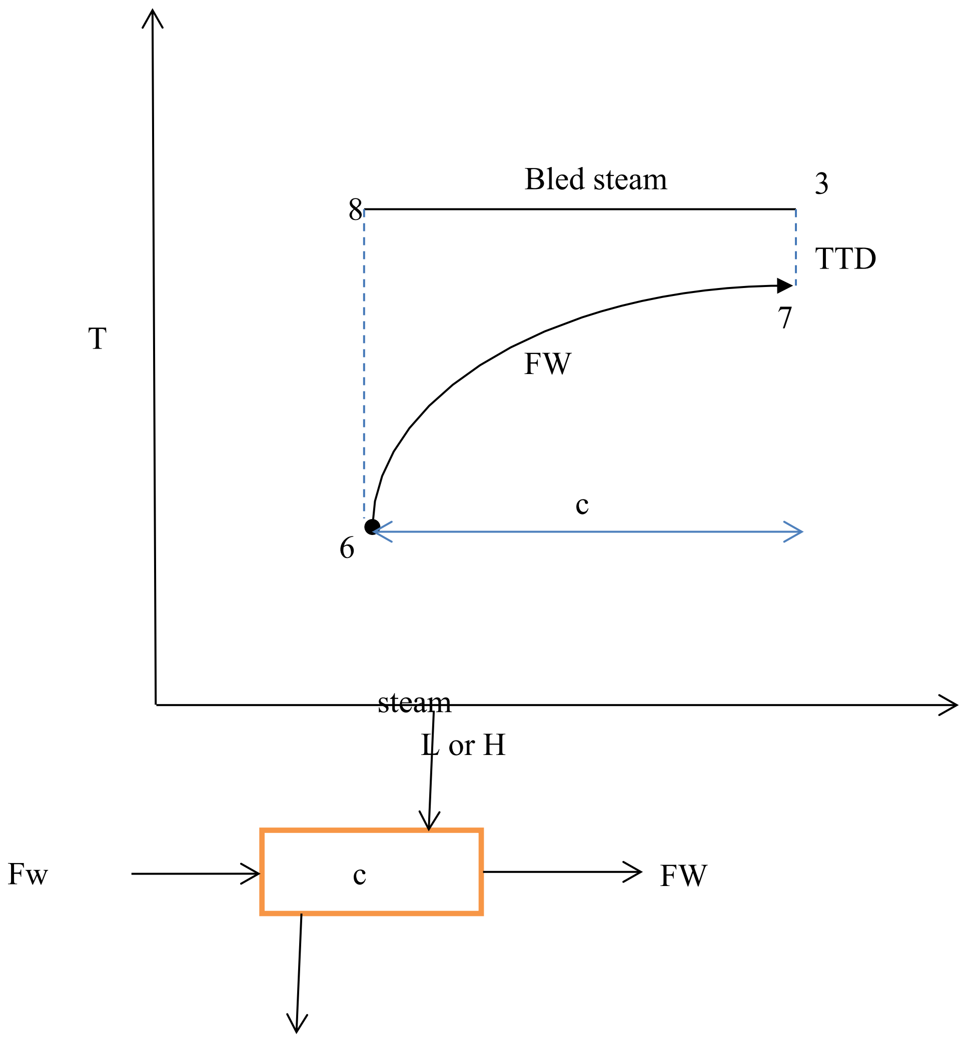

1-m1-m2
closed
closed
Condensate pump
H.p.HE ATER
10
7
L.p.HE ATER
6
trap
9
trap
8
11
Drip pump


1
1 kg
1kg
T
12
2
11
P2, m1
1-m1
9,10
P3, m2
3
7
8
1-m1-m2
6
1-m1-m2
5
4
P4
s

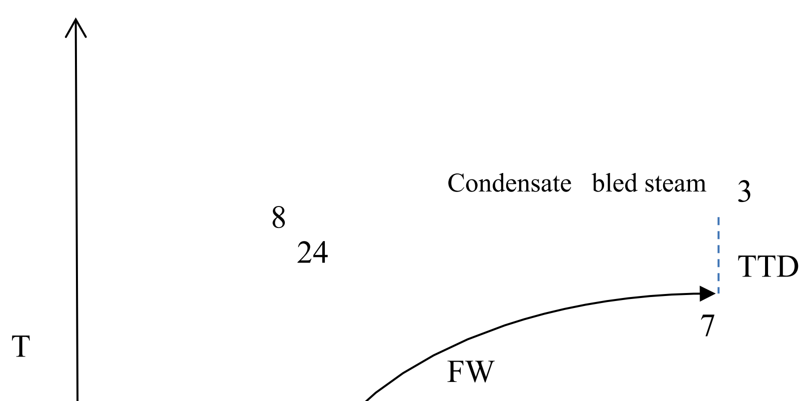
$$m1(h2 - h11) = 1(h12 - h10) = 1C_{pw}(T12 - T10)$$

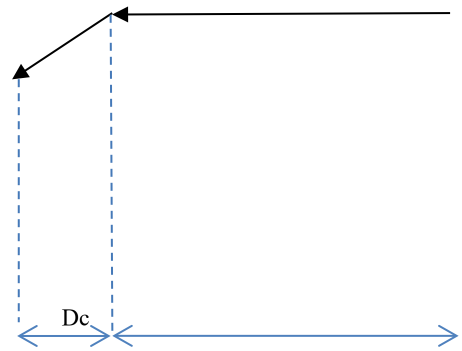
$$m2(h3 - h8) + m1(h11 - h8) = (1 - m1 - m2)(h7 - h6) = (1 - m1 - m2)C_{pw}(T7 - T6)$$

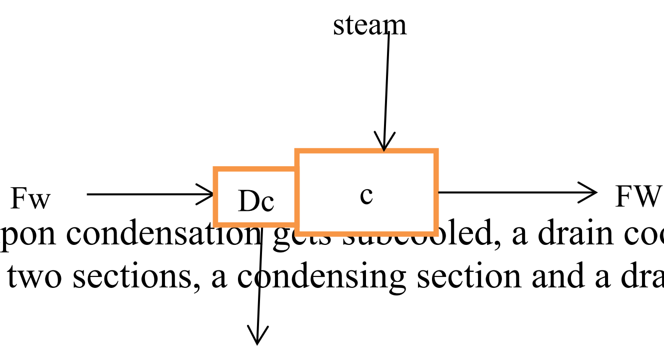
L,H: Low or High pressure turbine



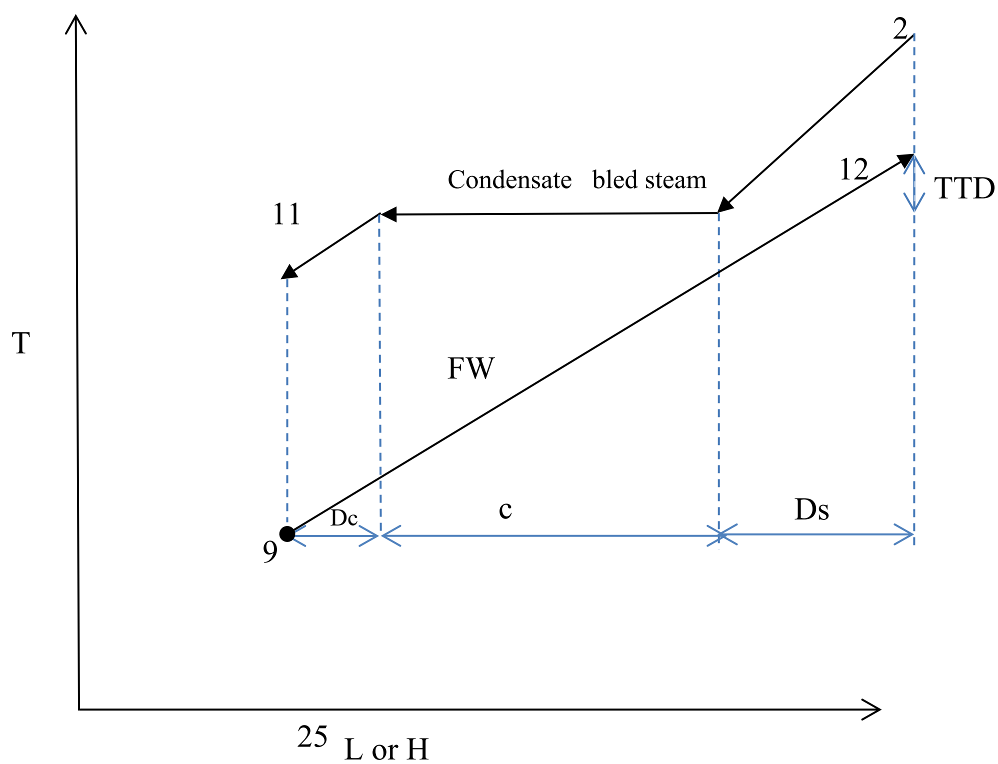
TTD: saturation temperature of bled steam (exit water temperature) "terminal temperature difference".
For low pressure heaters receiving wet steam. TTD is positive (about 3°C)

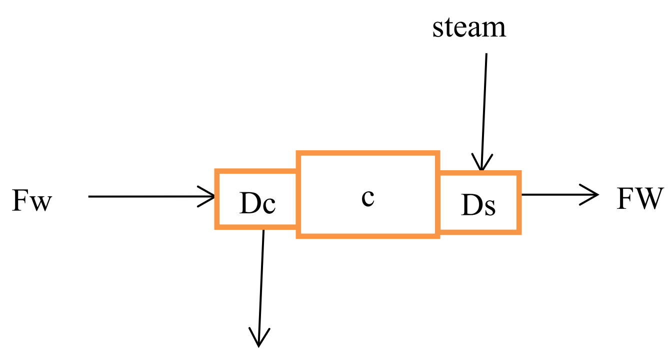






If the extracted steam upon condensation gets subcooled, a drain cooler may be used. The heater would then have two sections, a condensing section and a drain cooler section.





C:condenser, Dc: drain cooler, Ds: desuperheater

For the high pressure heater receiving superheated steam, bled from the turbine at state 2, the steam is first desuperheated, then condensed and finally subcooled to state 11, where as the feedwater gets heated from 9 to 12. It may be noted that the exit water temperature T12 is higher than the saturation temperature at p2, and the TTD is here negative.

The advantages of the open heater are simplicity, lower cost, and high heat transfer capacity. The disadvantages is the necessity of a pump at each heater to handle the large feedwater stream.

A closed heater requires only a single pump for the main feedwater stream (the drip pump is relatively small). Closed heater are favored in P.P. but at least one open heater is used for the purpose of feedwater deaeration (deaerator).

OPEN OR DIRECT-CONTACT FEEDWATER HEATERS

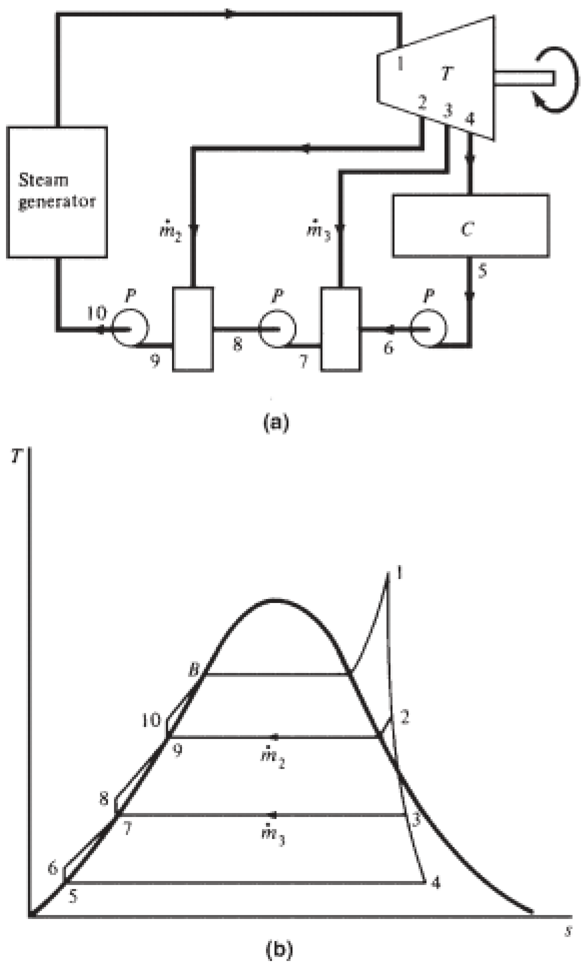
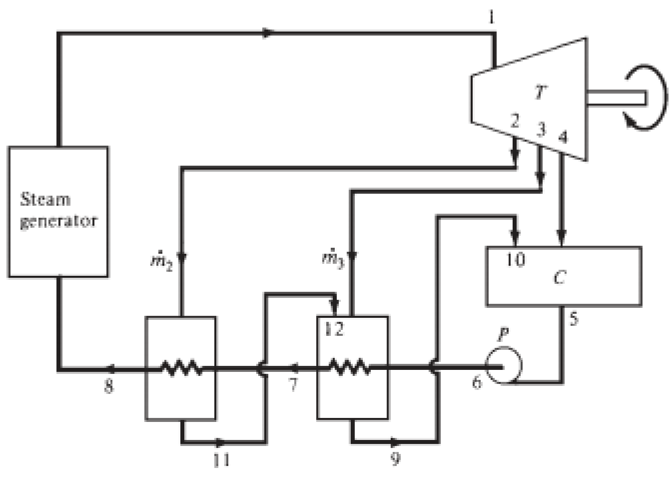


FIGURE 2.8 (*a*) Schematic flow and (*b*) *T-s* diagrams of a nonideal superheat Rankine cycle with two open-type feedwater heaters.

CLOSED-TYPE FEEDWATER HEATER WITH DRAINS CASCADED BACKWARD



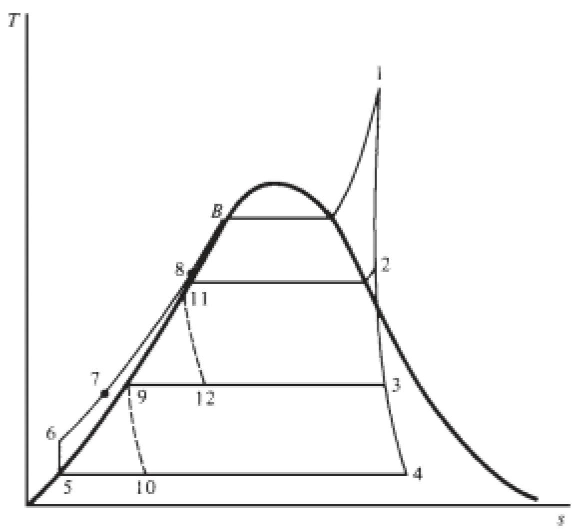
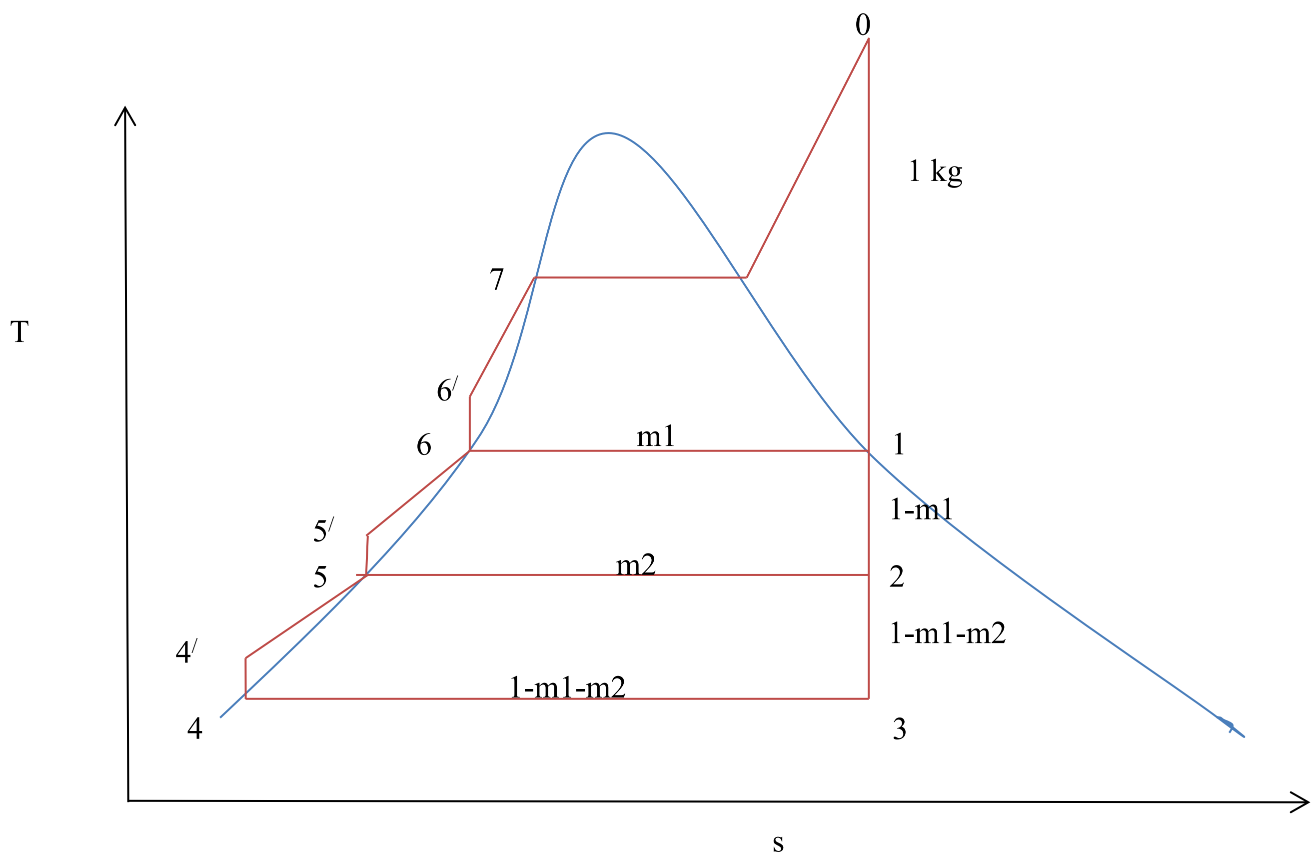
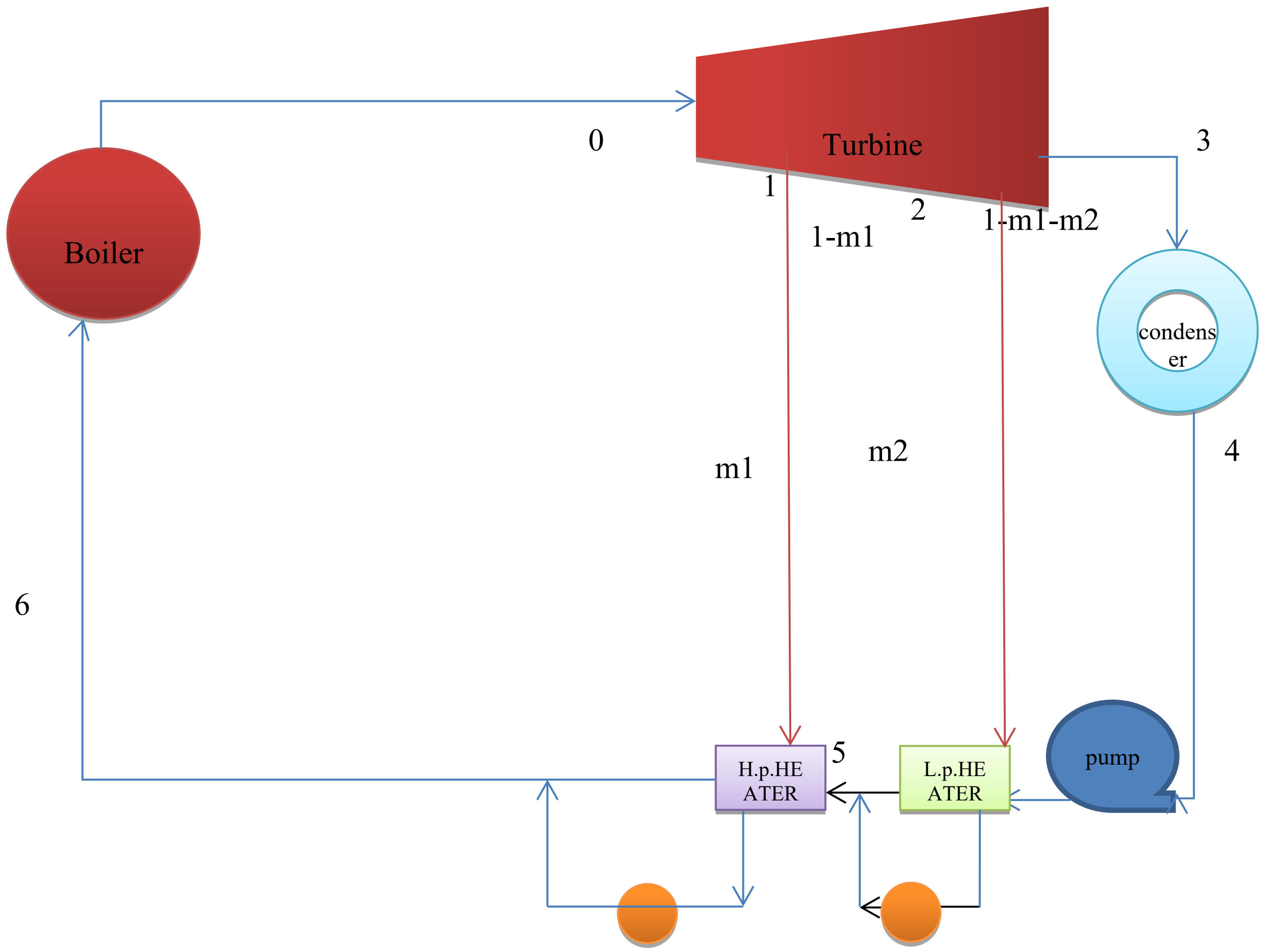


FIGURE 2.10 (*a*) Schematic flow and (*b*) *T-s* diagrams of a nonideal superheat Rankine cycle with two closed-type feedwater heaters with drains cascaded backward.

REGENERATIVE CYCLE:

The Rankine efficiency can be improved by bleeding off some of the steam at an intermediate pressure during the expansion, and mixing this steam with feedwater which has been pumped to the same pressure. The mixing process is carried out in a feed heater.





m1=kg of high pressure (H.p.) steam extracted per kg of steam flow

m1=kg of low pressure (L.p.) steam extracted per kg of steam flow

(1-m1-m2)=kg of steam entering condenser per kg of steam flow

Energy/ heat balance equation for high pressure heater:

$$m1(h1 - hf6) = (1 - m1)(hf6 - hf5)$$

$$m1[(h1 - hf6) + (hf6 - hf5)] = (hf6 - hf5)$$

$$m1 = \frac{hf6 - hf5}{h1 - hf5} \text{------------------------}(1)$$

Energy/ heat balance equation for low pressure heater:

$m2(h2 - hf5) = (1 - m1 - m2)(hf5 - hf4)$

$m2[(h2 - hf5) + (hf5 - hf4)] = (1 - m1)(hf5 - hf4)$

$m2 = \frac{(1 - m1)(hf5 - hf4)}{h2 - hf4}$ ----------(2)

The heat supplied externally in the cycle= $ho - hf6$ -----(3)

Isentropic work done = $m1(ho - h1) + m2(ho - h2) + (1 - m1 - m2)(ho - h3)$

$$\zeta_{thermal} = \frac{work\ done}{heat\ supplied} = \frac{m1(ho - h1) + m2(ho - h2) + (1 - m1 - m2)(ho - h3)}{ho - hf6}$$

Work done by turbine= $(ho - h1) + (1 - m1)(h1 - h2) + (1 - m1 - m2)(h2 - h3)$

Advantages of Regenerative Cycle:

1- The heat process in the boiler tends to become reversible.

2- The thermal stresses set up in the boiler are minimized. This is due to the fact that temperature ranges in the boiler are reduced.

3- The thermal efficiency is improved because the average temperature of heat addition to the cycle is increased.

4- Heat rate is reduced.

5- The blade height is less due to the reduced amount of steam passed through the low pressure stages.

6- Due to many extractions there is an improvement in the turbine drainage and it reduces erosion due to moisture.

7- A small size condenser is required.

Disadvantages:

1- The plant becomes more complicated.

2- Because of addition of heaters greater maintenance is required.

3- For given power a large capacity boiler is required.

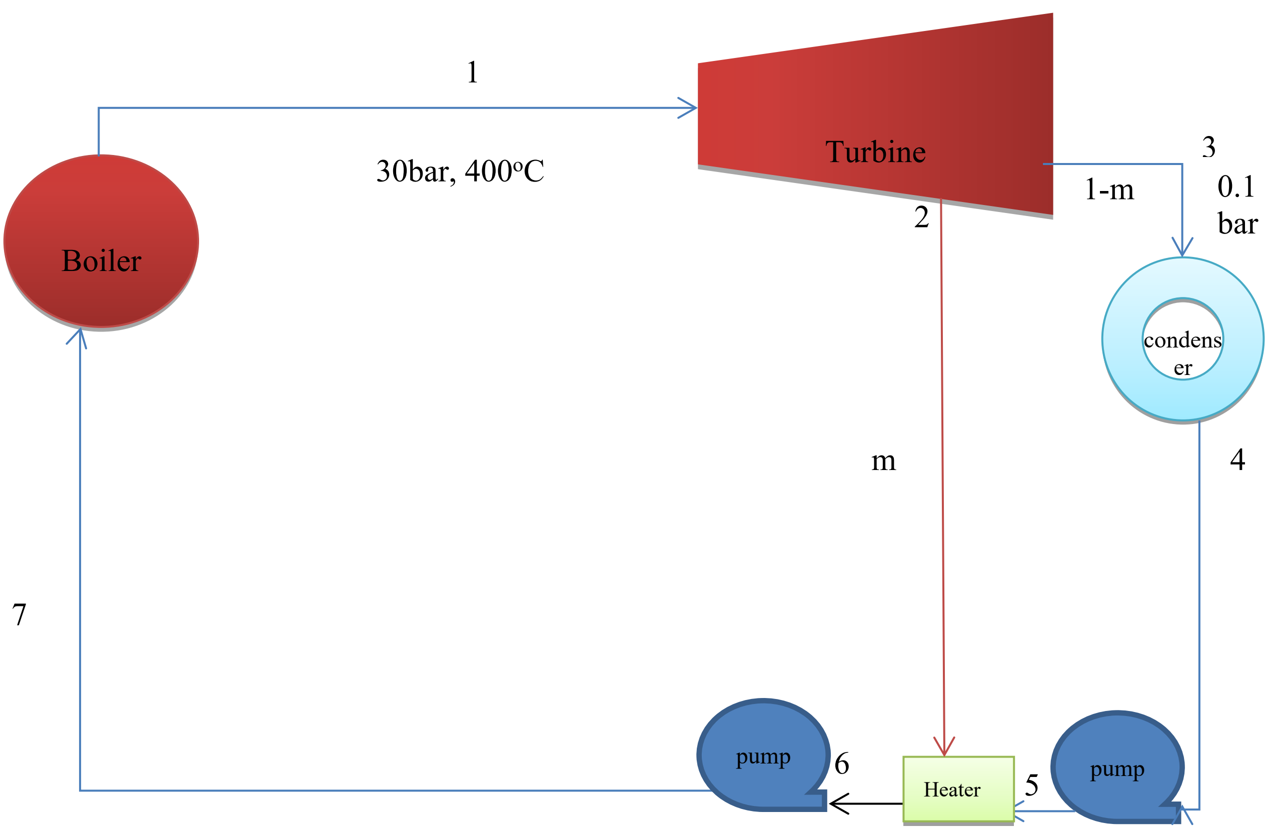
4- The heaters are costly and the gain in thermal efficiency is not much in comparison to the heavier costs.

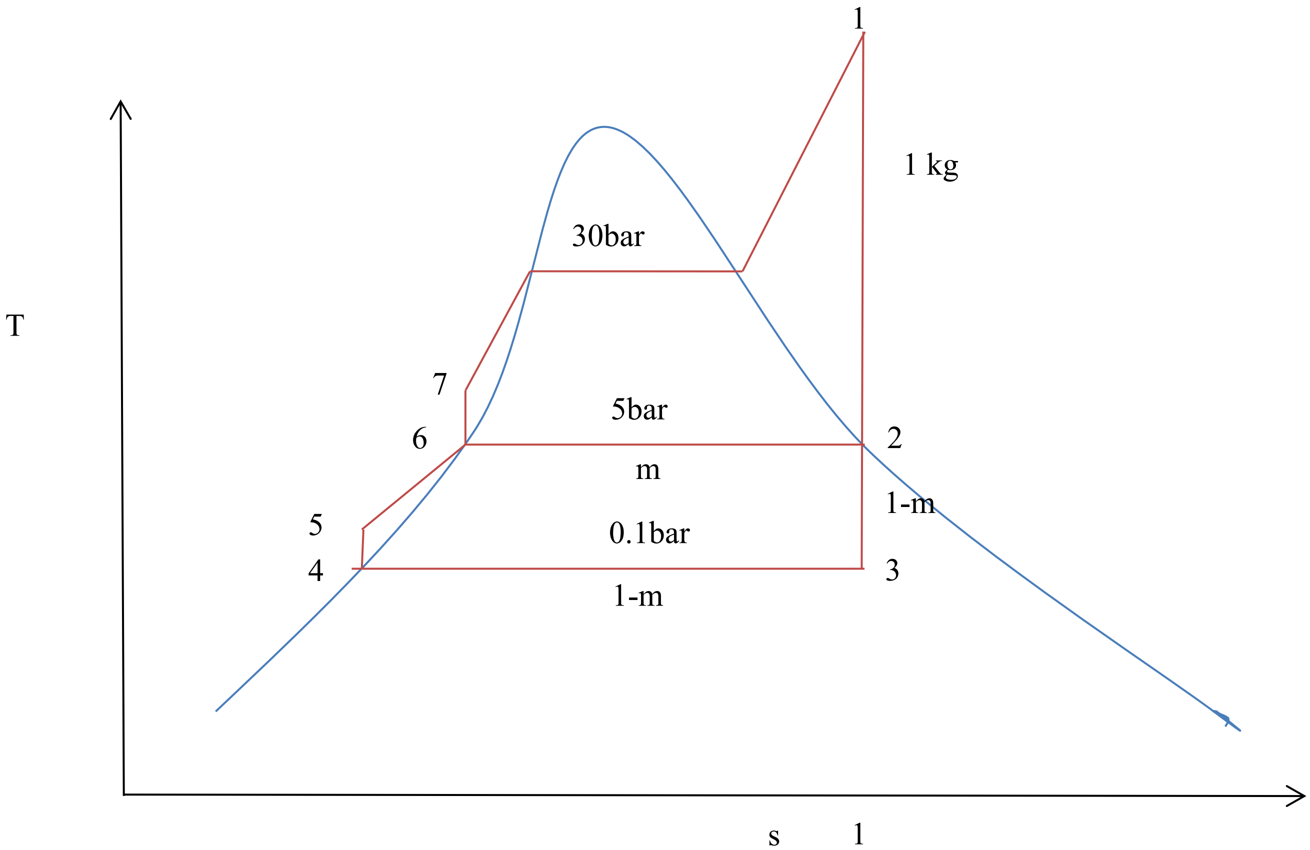
Ex.

In a single –heater regenerative cycle the steam enters the turbine at 30 bar, 400°C and the exhaust pressure is 0.1bar. The feedwater heater is a direct contact type which operates at 5bar. Find:

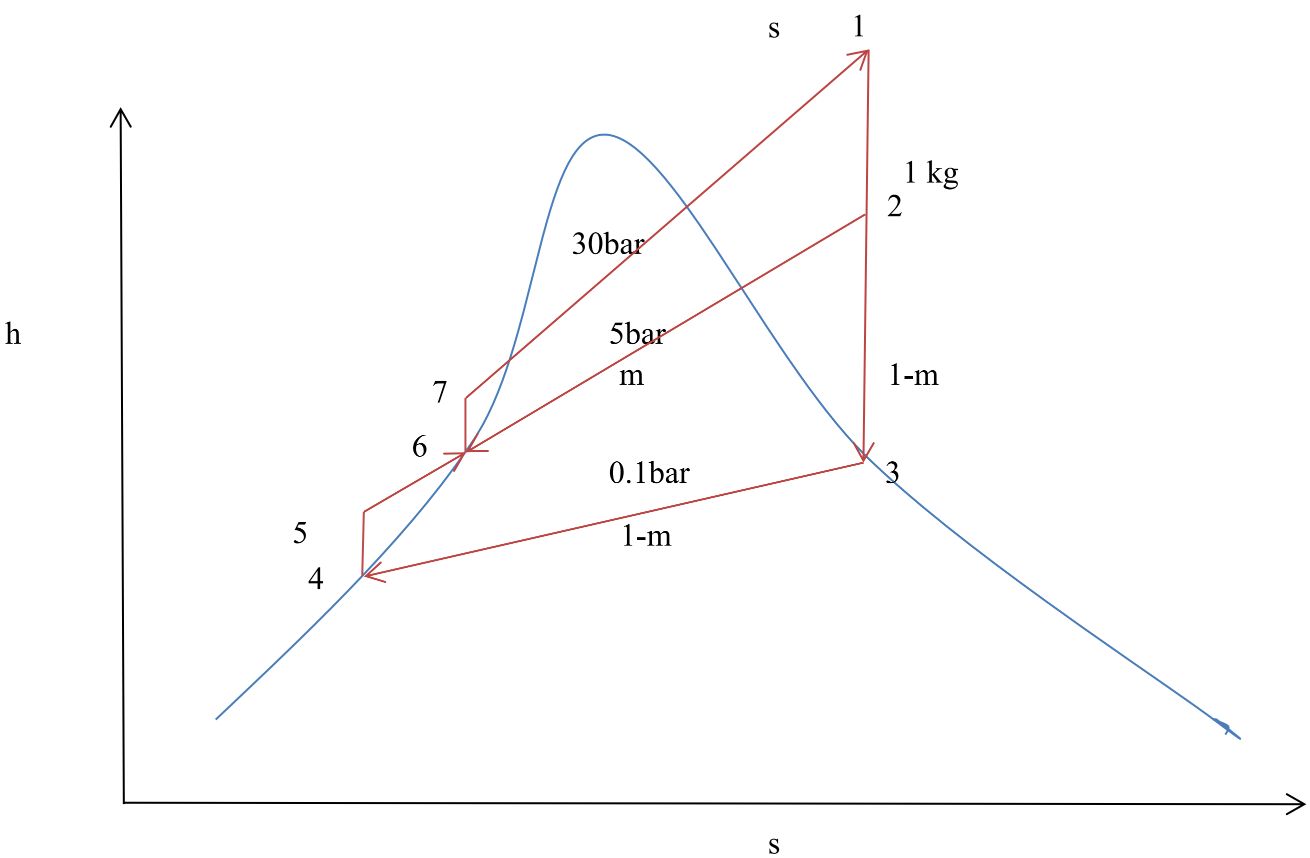
1- The efficiency and the steam rate of the cycle.

2- The increase in mean temperature of heat addition, efficiency and steam rate as compared to the Rankine cycle (without regeneration). Pump work may be neglected.





1
1 kg
30bar
T
7
5bar
6
2
m
1-m
5
0.1bar
4
3
1-m
s



1
1 kg
2
30bar
h
5bar
m
1-m
7
6
0.1bar
3
5
1-m
4
s

Example:

Steam at pressure of 20bar, 250°C enters a turbine and leaves it finally at a pressure of 0.05bar. Steam is bled off at pressure of 5, 1.5 and 0.3 bar. Assuming i- that the condensate is heated in each heater up to the saturated temperature of the steam in that heater, ii- that the drain water from each heater is cascaded through a trap into the next heater on the low pressure side of it, iii- that the combined drains from the heater operating at 0.3bar are cooled in a drain cooler to condenser temperature, calculate the following:

1- Mass of bled steam for each heater per kg of steam entering the turbine.
2- Thermal efficiency of the cycle.
3- Thermal efficiency of the Rankine cycle.
4- Theoretical gain due to regenerative feed heating.
5- Steam consumption in kg/kWh with or without regenerative feed heating.
6- Quantity of steam passing through the last stage nozzle of a 50000kW turbine with and without regenerative feed heating.

HOMEWORK:

A steam turbine plant developing 120 MW electrical output is equipped with reheating and regenerative feed heating arrangement consisting of two feed heaters – one surface type on H.P. side and other direct contact type on L.P. side. The steam conditions before the steam stop valve are 100bar and 530°C. A pressure drop of 5 bar takes place due to throttling in valves.

Steam exhausts from the H.P. turbine at 25bar. A small quantity of steam is bled off at 25bar for H.P. surface heater for feed heating and the remaining is reheated in a reheater to 550°C and the steam enters at 22bar in L.P. turbine for further expansion. Another small quantity of steam is bled off at a pressure 6bar for L.P. heater and the rest of steam expands up to the back pressure of 0.05bar. The drain from the H.P. heater is led to the L.P. heater and the combined feed from the L.P. heater is pumped to the high pressure feed heater and finally to the boiler with the help of boiler feed pump.

The component efficiencies are: turbine efficiency 85%, pump efficiency 90%, generator efficiency 96%, boiler efficiency 90%, and mechanical efficiency 95%. It may be assumed that the feed water is heated up to the saturation temperature at the prevailing pressure in feed heater. Work out the following:

1- Sketch the feed heating system and show the process on T-s and h-s diagram.
2- Amounts of steam bled off.
3- Overall thermal efficiency of turbo-alternator considering pump work.
4- Specific steam consumption in kg/kWh.

BOILERS:

Classification of boilers:

1- Horizontal, vertical or inclined.
2- Fire tube and water tube.
3- Externally fired and internally fired.
4- Forced circulation and natural circulation.
5- High pressure and low pressure.
6- Stationary and portable.
7- Single tube and multi-tube boilers.

	Particulars	Fire-tube boilers	Water-tube boiler
1	Position of water and hot gases	Hot gases inside the tubes and water outside the tubes.	Water inside the tubes and hot gases outside the tubes
2	Mode of firing	Generally internally fired	Externally fired
3	Operation pressure	Operating pressure limited to 16bar.	Can work under as high pressure as 100bar
4	Rate of steam production	Lower	Higher
5	Suitability	Not suitable for large power plants	suitable for large power plants
6	Risk on bursting	Involves lesser risk on explosion due to lower pressure	Involves more risk on bursting due to high pressure
7	Floor area	For a given power it occupies more floor area	For a given power it occupies less floor area
8	Construction	Difficult	Simple
9	Transportation	Simple	Difficult
10	Shell diameter	Large for same power	small for same power
12	Treatment of water	Not so necessary	More necessary
13	Accessibility of various parts	Various parts not so easily accessible for cleaning, repair and inspection	Various parts are more accessible
14	Requirement of skill	Require less skill for efficient and economic working	Require more skill and careful attention

Essentials of a good steam boiler:

1- The boiler should produce the maximum weight of steam of the required quantity at minimum expenses.

2- Steam production rate should be as per requirements.

3- It should be absolutely reliable.

4- It should occupy minimum space.

5- It should be light in weight.
6- It should be capable of quick starting.
7- There should be an easy access to the various parts of the boiler for repairs and inspection.
8- The boiler components should be transportable without difficulty.
9- The installation of the boiler should be simple.
10- The tubes of the boiler should not accumulate soot or water deposits and should be not sufficiently strong to allow for wear and corrosion.
11- The water and gas circuits should be such as to allow minimum fluid velocity (for low frictional losses).

Method of water circulation in high pressure boilers:

The circulation of water through the boiler may be natural circulation due to density difference or forced circulation. In all modern high pressure boiler plants, the water circulation is maintained with the help of pump which forces the water through the boiler plant. The use of natural circulation is limited to sub critical boilers due to its limitations.

Advantages of high pressure boilers:

1- Pumps are used to maintain forced circulation of water which increase evaporative capacity and lessen number of steam drums.
2- Efficient heat combustion by using small diameter tubes in large number and multiple circuits.
3- Pressurized combustion is used which increases rate of firing of fuel thus increasing the rate of heat release.
4- Due to compactness less floor space is required.

5- The tendency of scale formation is eliminated due to high velocity of water through the tubes.

6- The danger of overheating is reduced and thermal stress problem is simplified because all the parts are uniformly heated.

7- The differential expansion is reduced due to uniform temperature and this reduces the possibility of gas and air leakage.

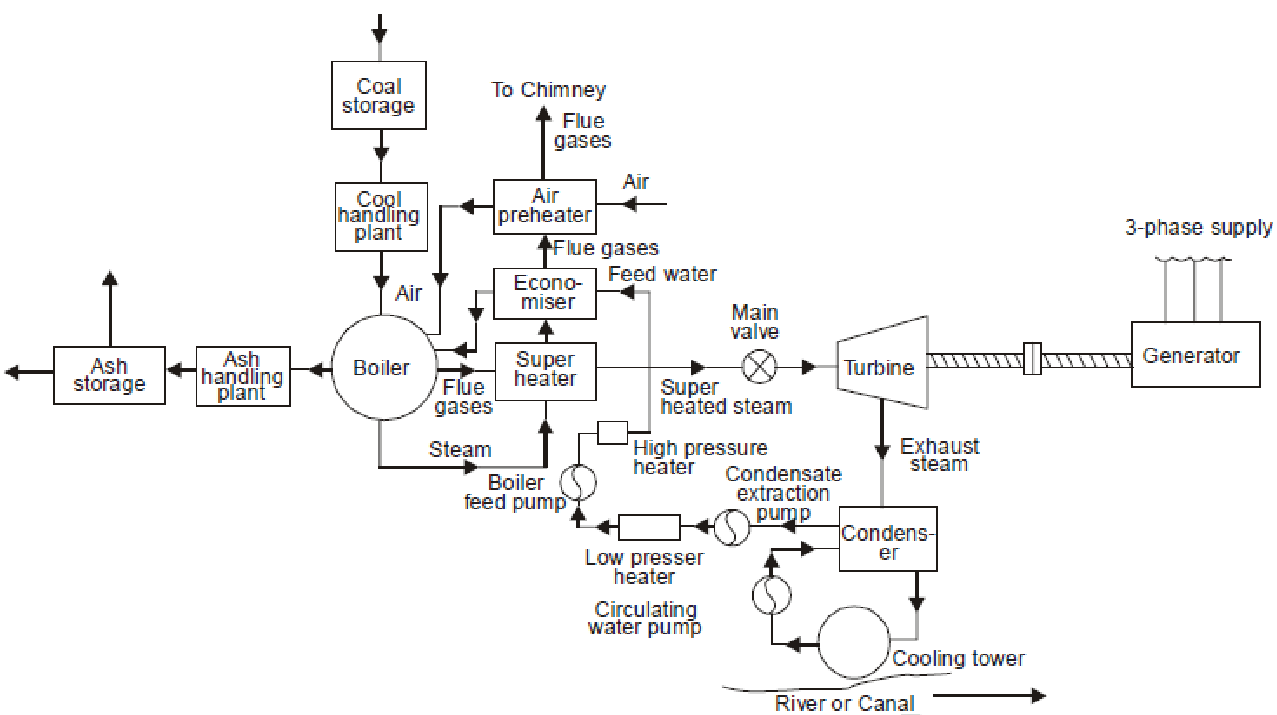
8- The components are arranged with great flexibility.

9- The steam can be raised quickly to meet the variable load requirements without the use of complicated control devices.

10- The efficiency of plant is increased up to 40 to 42% by using high pressure and high temperature steam.

11- A very rapid start from cold is possible if an external supply of power is available. Hence the boiler can be used for carrying peak loads or stand by purposes with hydraulic station.

12- Use of high pressure and high temperature steam is economical.

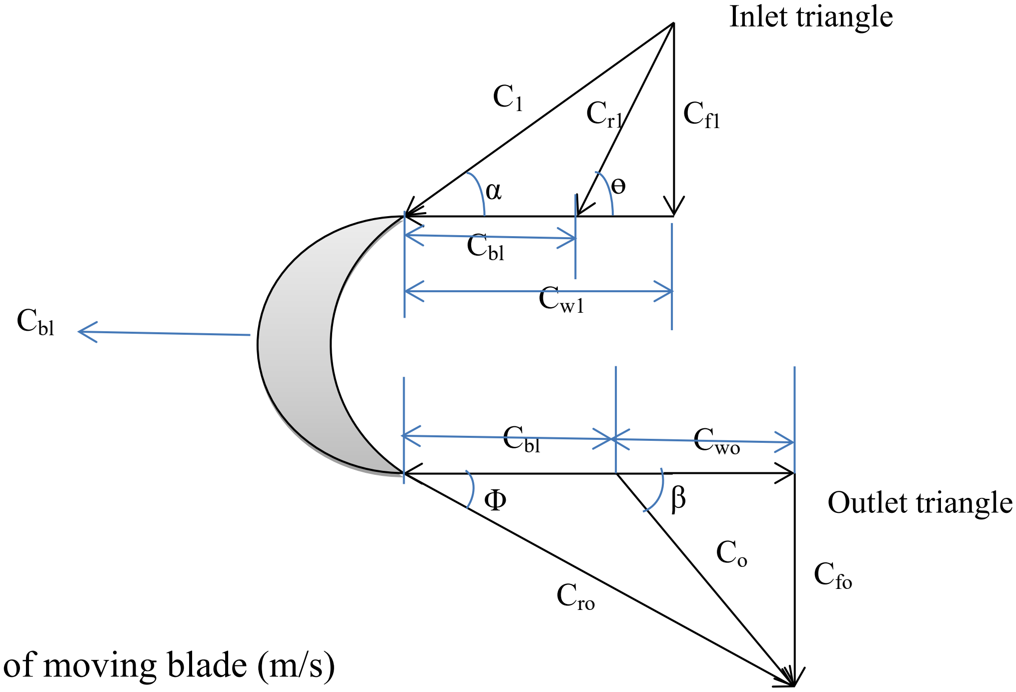


STEAM TURBINES:

There are several ways in which the steam turbines may be classified. The most important and common division being with respect to the action of the steam, as:

1- Impulse
2- Reaction
3- Combination of impulse and reaction.

1- Impulse turbine:



C_{bl}: linear velocity of moving blade (m/s)
C_1: absolute velocity of steam entering moving blade (m/s).
C_o : absolute velocity of steam leaving moving blade (m/s).
C_{w1} : velocity of whirl at the entrance of moving blade
= tangential component of C_1
C_{wo}:velocity of whirl at the exit of moving blade
= tangential component of C_o
C_{f1}: velocity of flow at the entrance of moving blade
= axial component of C_1
C_{fo}: velocity of flow at the exit of moving blade
= axial component of C_o
C_{r1}: relative velocity of steam at moving blade at entrance
C_{ro}: relative velocity of steam at moving blade at exit

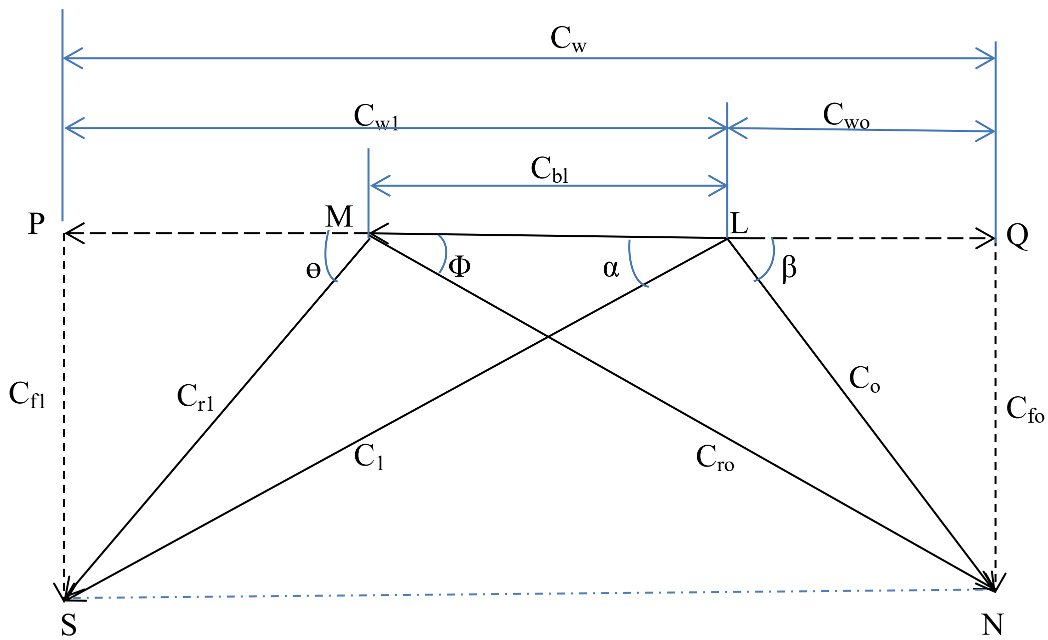
α: angle with the tangent of the wheel at which the steam with velocity C_1 enters (nozzle angle).

β: angle which the discharging steam makes with the tangent of the wheel at the exit of moving blade.

ϴ: entrance angle of moving blade.

Φ: exit angle of moving blade.

$C_{ro} \leq C_{r1}$ for impulse turbine blade



From Newton's second law of motion:

Force (tangential) on the wheel= mass of steam × acceleration

=mass of steam/sec× change of velocity

$$= \dot{m}_S (C_{w1} + C_{wo})$$

Work done on blades/sec = force × distance travelled/sec

$$= \dot{m}_S (C_{w1} + C_{wo}) \times C_{bl}$$

$$= \dot{m}_S C_w C_{bl}$$

Power per wheel $= \frac{\dot{m}_S C_w C_{bl}}{1000}$ kW

$$C_w = C_{w1} + C_{wo} = PQ = MP + MQ = C_{r1}\cos\theta + C_{ro}\cos\Phi$$

$$Blade\ or\ diagram\ efficiency = \frac{work\ done\ on\ the\ blade}{energy\ supplied\ to\ the\ blade}$$

$$= \frac{\dot{m}_S\,(C_{w1} + C_{wo}) \times C_{bl}}{\frac{\dot{m}_S C_1^2}{2}} = \frac{2C_{bl}(C_{w1} + C_{wo})}{C_1^2}$$

$$stage\ efficiency\ \eta_{stage} = \frac{Work\ done\ on\ blade\ per\ kg\ of\ steam}{Total\ energy\ supplied\ per\ kg\ of\ steam}$$

$$= \frac{C_{bl}(C_{w1} + C_{wo})}{(h_1 - h_2)}$$

h_1: the total heat before expansion through the nozzles
h_2: the total heat after expansion through the nozzles
(h_1-h_2): the heat drop through a stage of fixed blades ring and moving blades ring

$$\text{Nozzle efficiency} = \frac{C_1^2}{2(h_1 - h_2)}$$

$$\eta_{stage} = Blade\ efficiency \times Nozzle\ efficiency$$

$$= \frac{2C_{bl}(C_{w1} + C_{wo})}{C_1^2} \times \frac{C_1^2}{2(h_1 - h_2)} = \frac{C_{bl}(C_{w1} + C_{wo})}{(h_1 - h_2)}$$

The axial thrust on the wheel is due to difference between the velocities of flow at entrance and outlet.

Axial force on the wheel=Mass of steam × axial acceleration $=\dot{m}_S\,(C_{f1} - C_{fo})$

Energy converted to heat by blade friction=loss of kinetic energy during flow over blades

$$= \dot{m}_S \frac{(C_{r1}^2 - C_{ro}^2)}{2}$$

$$Blade\ velocity\ coefficient\ k = \frac{C_{ro}}{C_{r1}}$$

$$C_w = C_{r1}\cos\theta\left[1 + \frac{C_{ro}\cos\emptyset}{C_{r1}\cos\theta}\right]$$

$$= C_{r1}\cos\theta(1 + k\,z) \text{-------------(1)}$$

$$z = \frac{\cos\emptyset}{\cos\theta}$$

ɵ and Φ are nearly equal, then z=constant

$C_{r1}\cos ɵ = MP = LP - LM = C_1\cos\alpha - C_{bl}$

From eq.1 C_w=$(C_1\cos\alpha - C_{bl})(1+k\,z)$

From above $\eta_{bl} = \frac{2C_{bl}C_w}{{C_1}^2} = \frac{2C_{bl}(C_1\cos\alpha - C_{bl})(1+k\,z)}{{C_1}^2}$

=$2(\rho\cos\alpha - \rho^2)(1+k\,z)$

=$2\rho(\cos\alpha - \rho)(1+k\,z)$-----------------(2)

$\rho = \frac{C_{bl}}{C_1}$ blade speed ratio

If α, k and z may assumed to be constant.

$\frac{d\eta_{bl}}{d\rho}$=$2(\cos\alpha - 2\rho)(1+k\,z)$

For max.or min. η_{bl}, $\frac{d\eta_{bl}}{d\rho} = 0$

$\cos\alpha - 2\rho = 0$, $\rho = \frac{\cos\alpha}{2}$

$\frac{d^2_{\eta_{bl}}}{d\rho^2} = 2(-2)(1+k\,z) = -4(1+k\,z)$ negative value, hence the value of **ρ** is max.

$\rho_{opt} = \frac{\cos\alpha}{2}$-------------------(3) sub. In eq. (2)

η_{blmax}=$2\frac{\cos\alpha}{2}\left(\cos\alpha - \frac{\cos\alpha}{2}\right)(1+k\,z)$

=$\frac{\cos^2\alpha}{2}(1+k\,z)$

If symmetrical blades (ɵ = Φ) , no friction in fluid passage for the purpose of analysis.

z=1, and k=1

$\eta_{blmax} = \cos^2\alpha$

$w = (C_{w1} + C_{wo})C_{bl} = C_wC_{bl}$

=$(C_1\cos\alpha - C_{bl})(1+k\,z)C_{bl} = 2C_{bl}(C_1\cos\alpha - C_{bl})$

k=1, z=1, sub. cosα from eq. (3)

$\cos\alpha = 2\rho = 2\frac{C_{bl}}{C_1}$

$$w_{max} = 2C_{bl}(2C_{bl} - C_{bl}) = 2{C^2}_{bl}$$

REACTION TURBINES:

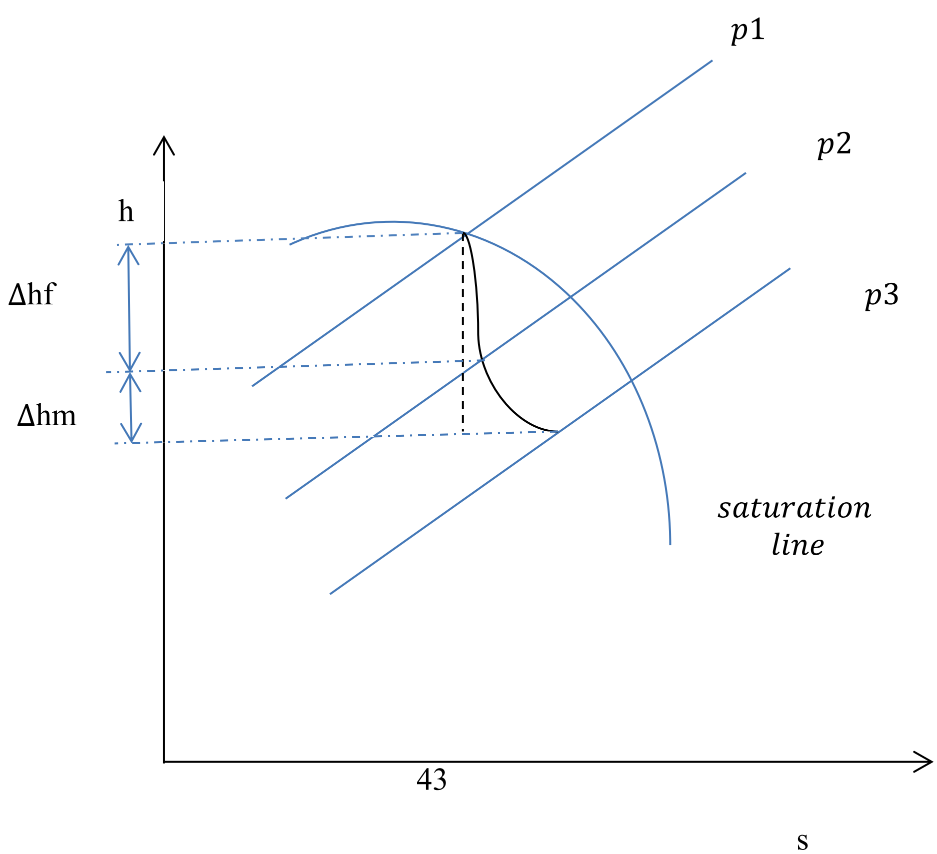
The steam continuously expands at it flows over the blades. The effect of continuous expansion of steam during the flow over the blade is to increase the relative velocity of steam.

$C_{ro} > C_{r1}$ for reaction turbine blade

$C_{ro} \leq C_{r1}$ for impulse turbine blade

The degree of reaction of reaction turbine stage is defined as the ratio of heat drop over moving blades to the total heat drop in the stage.

$$R_d = \frac{heat\ drop\ in\ moving\ blades}{heat\ drop\ in\ the\ stage} = \frac{\Delta h_m}{\Delta h_f + \Delta h_m}$$



$$\Delta h_m = \frac{{C_{ro}}^2 - {C_{r1}}^2}{2}$$

The total heat drop in the stage $\Delta h_f + \Delta h_m$=work done by the steam in the stage

$\Delta h_f + \Delta h_m = C_{bl}(C_{w1} + C_{wo})$

$R_d = \frac{{C_{ro}}^2 - {C_{r1}}^2}{2C_{bl}(C_{w1} + C_{wo})}$---------------(1)

$C_{ro} = C_{fo}\, csc\Phi, \quad C_{r1} = C_{f1}\, csc\theta$

From fig. of velocity diagram:

$$(C_{w1} + C_{wo}) = C_{f1} cot\theta + C_{fo}\, cot\Phi$$

$C_{f1} = C_{fo} = C_f$ (the velocity of flow remains constant through the blades)

Sub. C_{ro}, C_{r1} , $(C_{w1} + C_{wo})$ in equation 1

$$R_d = \frac{{C_f}^2(csc^2\Phi - csc^2\theta)}{2C_{bl}C_f(cot\theta + cot\Phi)} = \frac{C_f}{2C_{bl}}\left[\frac{(cot^2\Phi + 1) - (cot^2\theta + 1)}{cot\theta + cot\Phi}\right] = \frac{C_f}{2C_{bl}}\left[\frac{cot^2\Phi - cot^2\theta}{cot\theta + cot\Phi}\right] = \frac{C_f}{2C_{bl}}(cot\Phi - cot\theta)$$

The conditions for maximum efficiency done by the following assumptions:

1- The degree of reaction is 50%.
2- The moving and fixed blades are symmetrical.
3- The velocity of steam at exit from the preceding stage is same as velocity of steam at the entrance to the succeeding stage.

$$W = C_{bl}(C_{w1} + C_{wo}) = C_{bl}[C_1\, cos\alpha + (C_{ro}\, cos\Phi - C_{bl})]$$

$\Phi = \alpha$ and $C_{ro} = C_1$ (assumptions)

$W = C_{bl}[2C_1\, cos\alpha - C_{bl}]$

$$W = C_1{}^2\left[\frac{2C_{bl}C_1\, cos\alpha}{C_1{}^2} - \frac{C_{bl}{}^2}{C_1{}^2}\right] = C_1{}^2[2\rho\, cos\alpha - \rho^2]\text{-----------(2)}$$

$$\rho = \frac{C_{bl}}{C_1}$$

K.E. supplied to the fixed blade $= \frac{C_1{}^2}{2}$

K.E. supplied to the moving blade $= \frac{C_{ro}{}^2 - C_{r1}{}^2}{2}$

Total energy supplied to the stage $\Delta h = \frac{C_1{}^2}{2} + \frac{C_{ro}{}^2 - C_{r1}{}^2}{2}$

$C_{ro} = C_1$ for symmetrical triangles

$\Delta h = \frac{C_1{}^2}{2} + \frac{C_1{}^2 - C_{r1}{}^2}{2} = C_1{}^2 - \frac{C_{r1}{}^2}{2}$--------------------(3)

From the triangle LMS

$C_{r1}{}^2 = C_1{}^2 + C_{bl}{}^2 - 2C_1C_{bl}cos\alpha$ sub. In eq. 3

$$\Delta h = C_1{}^2 - \left(\frac{C_1{}^2 + C_{bl}{}^2 - 2C_1C_{bl}cos\alpha}{2}\right)$$

$$= \frac{C_1{}^2 - C_{bl}{}^2 + 2C_1C_{bl}cos\alpha}{2}$$

$$= \frac{C_1{}^2}{2}\left[1 + \frac{2C_{bl}}{C_1}cos\alpha - \left(\frac{C_{bl}}{C_1}\right)^2\right] = \frac{C_1{}^2}{2}[1 + 2\rho\, cos\alpha - \rho^2]$$

$$\eta_{bl} = \frac{W}{\Delta h} = \frac{C_1{}^2[2\rho\, cos\alpha - \rho^2]}{\frac{C_1{}^2}{2}[1 + 2\rho\, cos\alpha - \rho^2]} = \frac{2(2\rho\, cos\alpha - \rho^2)}{(1 + 2\rho\, cos\alpha - \rho^2)} = \frac{2(1 + 2\rho\, cos\alpha - \rho^2) - 2}{(1 + 2\rho\, cos\alpha - \rho^2)} = 2 - \frac{2}{1 + 2\rho\, cos\alpha - \rho^2}\text{--------------------------(4)}$$

η_{bl}becomes maximum when the value of $(1 + 2\rho\ cos\alpha - \rho^2)$ becomes maximum.

$$\frac{d}{d\rho}(1 + 2\rho\ cos\alpha - \rho^2) = 0$$

$$2cos\alpha - 2\rho = 0$$

$\rho = cos\alpha$---------------------(5) sub. In eq. 4

$$\eta_{bl_{max}} = 2 - \frac{2}{1 + 2\ cos^2\alpha - cos^2\alpha} = 2\left[1 - \frac{1}{1 + cos^2\alpha}\right] = \frac{2cos^2\alpha}{1 + cos^2\alpha}$$

Condenser:

A condenser where the exhaust steam from the turbine is condensed operates at a pressure lower than atmosphere. There are two objects of using a condenser in a steam plant:

1- To reduce the turbine exhaust pressure so as to increase the specific output of the turbine. If the circulating cooling water temperature in a condenser is low enough, it creates a low back pressure (vacuum) for the turbine. This pressure is equal to the saturation pressure corresponding to the condensing steam temperature, which, in turn, is a function of the cooling water temperature. It is known that the enthalpy drop or turbine work per unit pressure drop is much greater at the low pressure end than at the high pressure end of a turbine. A condensation by lowering the back pressure increases the plant efficiency and reduces the steam flow for a given output.

2- To recover high quality feedwater in the form of condensate and feed it back to the steam generator without any further treatment.

There are two types of condensers:

1- Jet condenser.

2- Surface condenser.

$$condenser\ efficiency = \frac{Rise\ in\ temperature\ of\ cooling\ water}{\begin{pmatrix} Temperature\ corresponding \\ to\ vacuum\ in\ the\ condenser \end{pmatrix} - \begin{pmatrix} inlet\ temperature \\ of\ cooling\ water \end{pmatrix}}$$

$$= \frac{T_{w2} - T_{w1}}{T_s - T_{w1}}$$

$$Heat\ lost\ by\ steam = m_s[x\ h_{fg2} + C_{pw}(T_s - T_c)]$$

Heat gained by water = $m_w C_{pw}(T_{w2} - T_{w1})$

$$m_s[x\ h_{fg2} + C_{pw}(T_s - T_c)] = m_w C_{pw}(T_{w2} - T_{w1})$$

$$m_w = \frac{m_s[x\ h_{fg2} + C_{pw}(T_s - T_c)]}{C_{pw}(T_{w2} - T_{w1})}$$

m_s: mass of steam condensed (kg/h)

m_w: mass of cooling water (kg/h)

T_s: saturation temperature of steam °C

T_c: temperature of the condensate leaving the condenser

T_{w1}: temperature of the cooling water at inlet °C

T_{w2}: temperature of the cooling water at outlet °C

C_{pw}: specific heat of water at constant pressure

h_{fg}: latent heat of 1 kg of steam entering the condenser

x: dryness fraction

Type equation here.

COOLING TOWERS:

In power plants, the hot water from condenser is cooled in cooling tower, so that it can be reused in condenser for condensation of steam. In a cooling tower water is made to trickle down drop by drop so that it comes in contact with the air moving in the opposite direction. As a result of this some water is evaporated and is taken away with air. In evaporation, the heat is taken away from the bulk of water, which is thus cooled. Factors affecting cooling of water in a cooling tower are:

1- Temperature of air.
2- Humidity of air.
3- Temperature of hot water.
4- Size and height of tower.
5- Velocity of air entering tower.
6- Accessibility of air to all parts of tower.
7- Degree of uniformity in descending water.
8- Arrangement of plates in tower.

Cooling towers may be classified according to the material of which these are made:

1- Timber towers: rarely used due to following disadvantages:

a- Due to exposure to sun, wind, water, etc.; timber rots easily.
b- Short life.
c- High maintenance type.
d- The design generally does not facilitate proper circulation of air.
e- Limited cooling capacity.

2- Concrete towers: possess the following advantages:
   a- Large capacity sometimes of the order of $5*10^3$ m^3/h.
   b- Improved draught and air circulation.
   c- Increased stability under air pressure.
   d- Low maintenance.
3- Steel duct type: are rarely used in case of modern power plants owing to their small capacity.

The cooling towers require a draught of air for condensation of water sprayed. The draught may be created by a chimney or the available natural air velocity (natural draught) or by fans (mechanical draught). The mechanical draught may be forced or induced depending on the placement of fans.

WATER CIRCULATION:

Water circulates within the tubes and partially becomes steam as it receives heat from the products of combustion. When water circulation within the boiler takes place due to its own density difference, it is called the natural-circulation boiler. In this type of boiler, water from the boiler drum first flows downward to the bottom of the heated evaporative tubes through several pipes (downcomers). Then, the water reverses its flow direction and returns to the drum as it receives the heat from the furnace. Since, the evaporative tubes (risers) contain a mixture of steam and water, the average density in the riser is always lower than that in the downcomer. This density difference gives rise to a driving force that will overcome all friction in the water-steam circuit.

Natural circulation of water is a simple and efficient technique and is frequently employed in boiler designs.

As the boiler pressure becomes higher and higher, the difference in density of the fluid between the downcomers and the risers will becomes less and less. At a certain boiler pressure, the driving force, which is proportional to the density difference, is not sufficiently large to balance the frictional resistance. This will employ pumps to force the water through the evaporative tubes. The boiler using circulation pumps is called the force circulation boiler.

In force circulation water-tube boiler, the circulation pumps take the water from the drum and supply it to the headers at the bottom of the boiler. From the headers water moves upward as it receives heat from the products of combustion. Because sufficient driving force is available, smaller diameter tubes can be used in the force circulation boiler. Furthermore, it is possible to apply an orifice to each tube so that more uniform flow and tube temperature can be achieved. These advantages frequently offset the cost of circulation pumps and their pumping power.

In forced circulation there is frequently no boiler drum. Water flows through the evaporation section without any circulation. This arrangement is frequently employed when the steam pressure in the boiler is supercritical.

The economizer is a heat exchanger used to increase feedwater temperature. The evaporation section, which usually surrounds the boiler furnace, is to produce saturated steam and supply it to the superheater.

GAS TURBINE POWER PLANT

The gas turbine obtains its power by utilizing the energy of burnt gases and air, which is at high temperature and pressure by expanding through the several ring of fixed and moving blades.

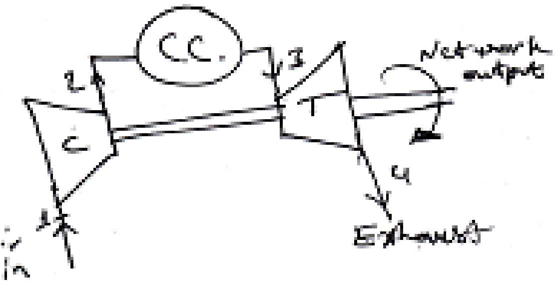
A simple gas turbine cycle consists of:

1- Compressor.
2- Combustion chamber.
3- Turbine.

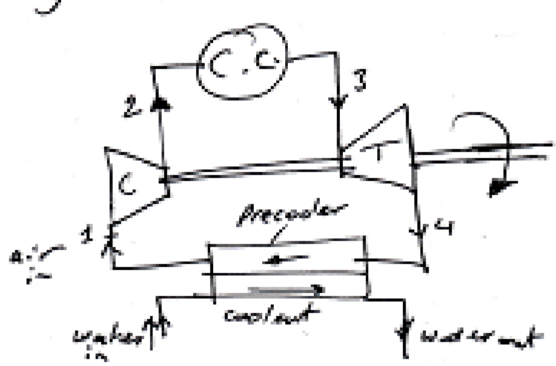
Gas turbines have been constructed to work on the oil, natural gas, coal gas, producer gas, and pulverized coal.

The gas turbine power plants which are used in electric power industry are classified into two groups as per the cycle of operation:

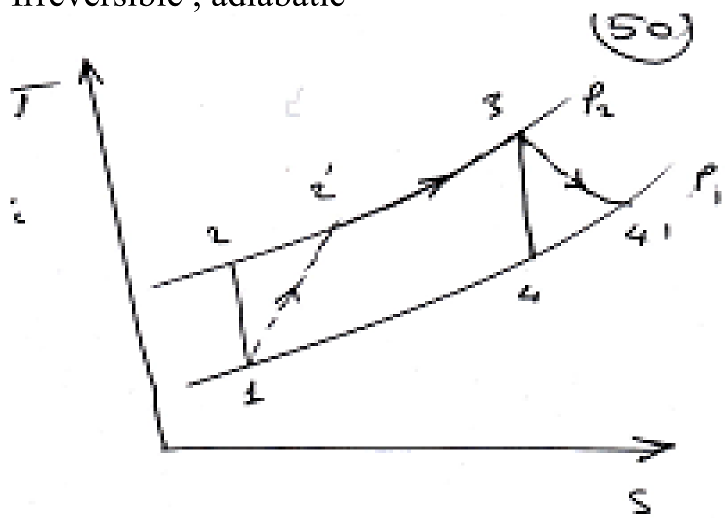
1- Open cycle gas turbine power plant.
2- Closed cycle gas turbine power plant.



② Closed cycle G.T.P.P.



Compression process:
Irreversible , approximately adiabatic
Expansion process : turbine
Irreversible , adiabatic



1 – 2′ : irreversible, adiabatic
2′ – 3 : constant pres. , heat- supply.
3 – 4′ : irreversible , adiabatic.
1 – 2 : ideal isentropic process.
Compressor : Work input = Cp (T2′ – T1)
Combustion chamber : heat supplied = Cp (T3 – T2′)
Turbine : work output = Cp (T3 – T4′)
Network output = Cp (T3 – T4′) – Cp (T2′ – T1)

$$\text{Thermal eff.} = \frac{Net\ work\ output}{heat\ supplied} = \frac{Cp\ (T3 - T4') - Cp\ (T2' - T1)}{Cp\ (T3 - T2')}$$

Neglecting changing in k.E .

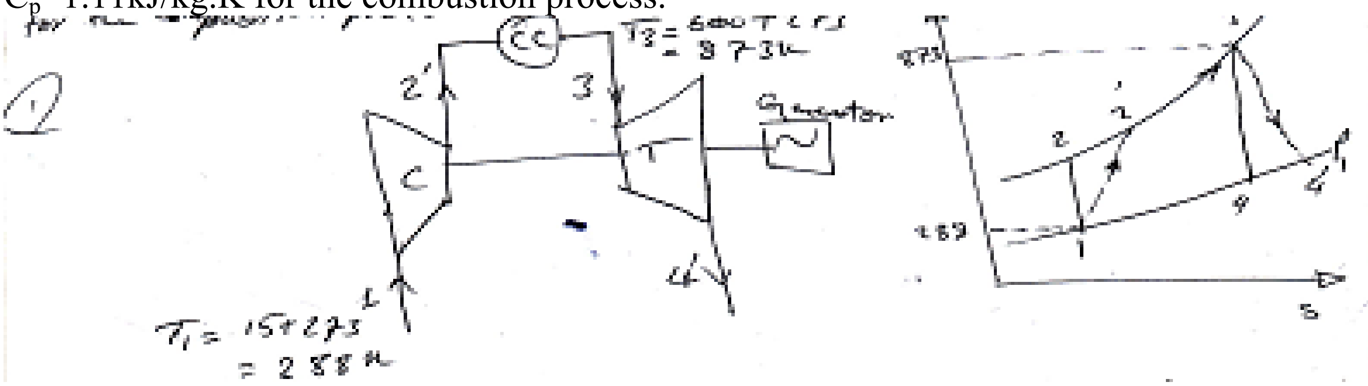
$$\text{Compressor isentropic eff.} = \frac{Cp\ (T2 - T1)}{Cp\ (T2' - T1)} = \frac{(T2 - T1)}{(T2' - T1)}$$

Turbine isentropic eff. $= \frac{Cp\,(T3 - T4')}{Cp\,(T3 - T4)}$

$= \frac{(T3 - T4')}{(T3 - T4)}$

Example1:

A gas turbine unit has a pressure ratio of 6/1 and a maximum cycle temperature of 600°C. The isentropic efficiencies of the compressor and turbine are 0.82 and 0.85 respectively. 1- Calculate the power output in kilowatts of an electric generator geared to the turbine when the air enters the compressor at 15°C at the rate of 15kg/s. Take C_p=1.005kJ/kg.K and γ=1.4 for the compression process, and take C_p=1.11kJ/kg.K and γ=1.333 for the expansion process. 2- Calculate the thermal efficiency and the work ratio of the plant assuming that C_p=1.11kJ/kg.K for the combustion process.



T1 = 15+273= 288 k

$\frac{T2}{T1} = \left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}}$

$T_2 = 288(6)^{\frac{0.4}{1.4}} = 481$ K

$\mu_c = \frac{T_2 - T_1}{T_{2'} - T_1}$

$= \frac{481 - 288}{T_{2'} - 288} = 0.82$ »»» $T_{2'} = 523.5$ K

$\frac{T3}{T4} = \left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}}$

$T_4 = \frac{873}{6^{\frac{0.333}{1.333}}} = \frac{873}{1.562} = 558$ K

$\mu_T = \frac{T3 - T_{4'}}{T_3 - T4}$ ==» T4′= 605K

$0.85 = \frac{873 - T4'}{873 - 558}$

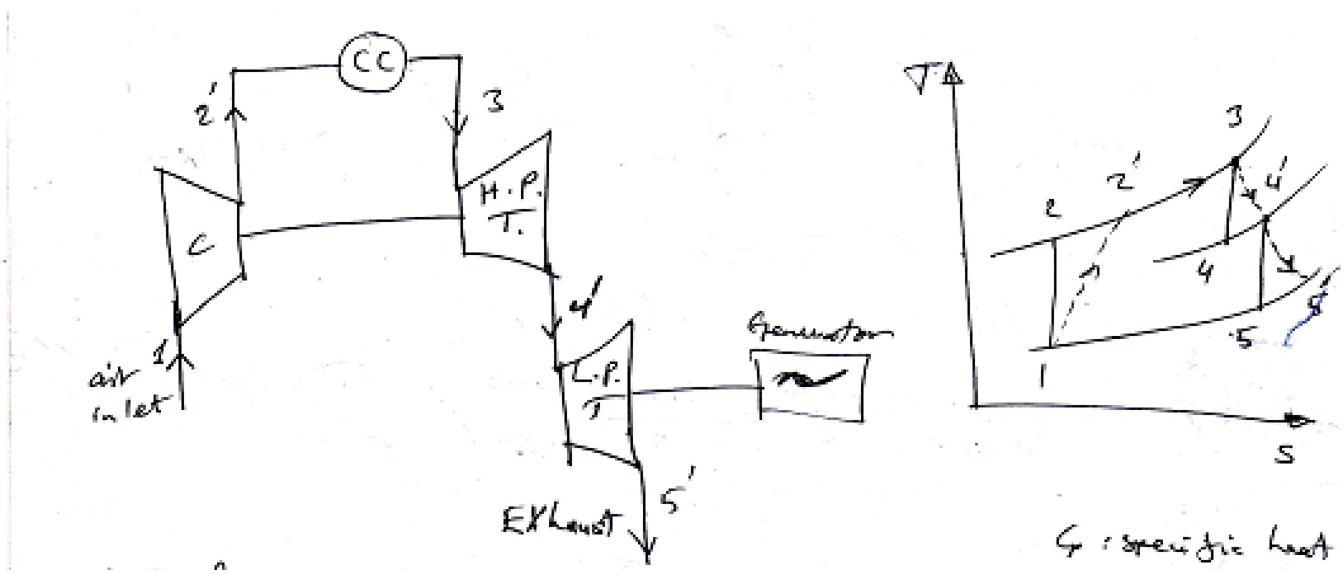
Compressor work input = C_P(T2′ - T1) = 1.005* 235.5 = 236.2 KJ/Kg
Turbine work output = C_P(T3 – T4′) = 1.11* 268 = 297.5 KJ/Kg

Network output = 297.5 – 236.2 = 61.3 kJ/kg
Power kilowatts = 61.3 * 15 =290 kW
2) heat supplied = Cp (T3 – T2′)= 1.11(873 – 523.5) = 338 kJ/kg

$$\mu_{th} = \frac{net\ work\ output}{heat\ supplied}$$

$$= 0.158 \quad = \frac{61.3}{388}$$

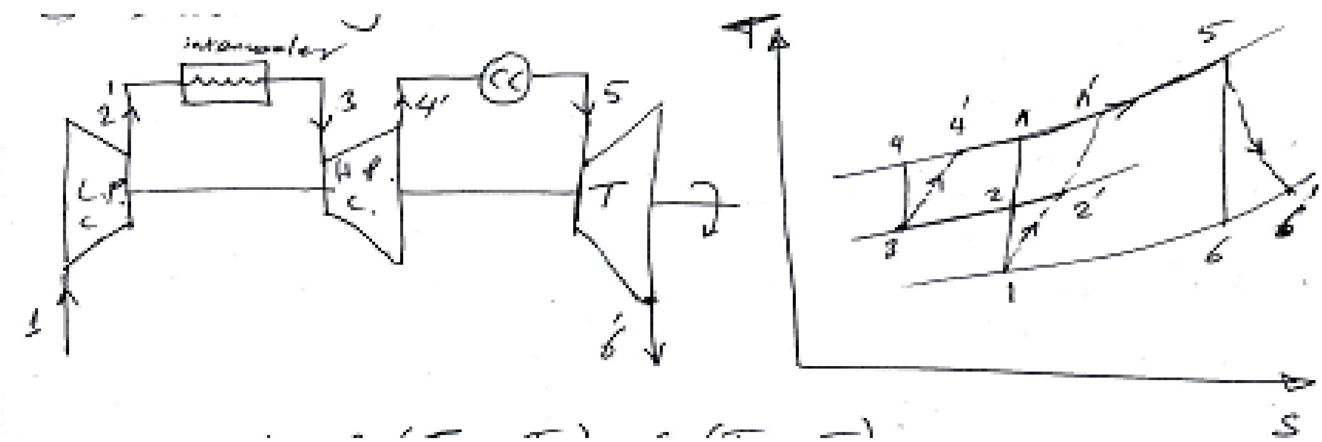
$$\text{Work ratio} = \frac{net\ work\ output}{gross\ work\ output} = \frac{61.3}{297.5} = 0.206$$



Work from H.P turbine = work input to compressor
Cpg (T3 –T4′) = Cpa(T2′ - T1)
The work net output = Cpg (T4′ - T5′)

Modification of the basic cycle

1 – Intercooling



Work input = $Cp\,(T2' - T1) + Cp\,(T4' - T3)$

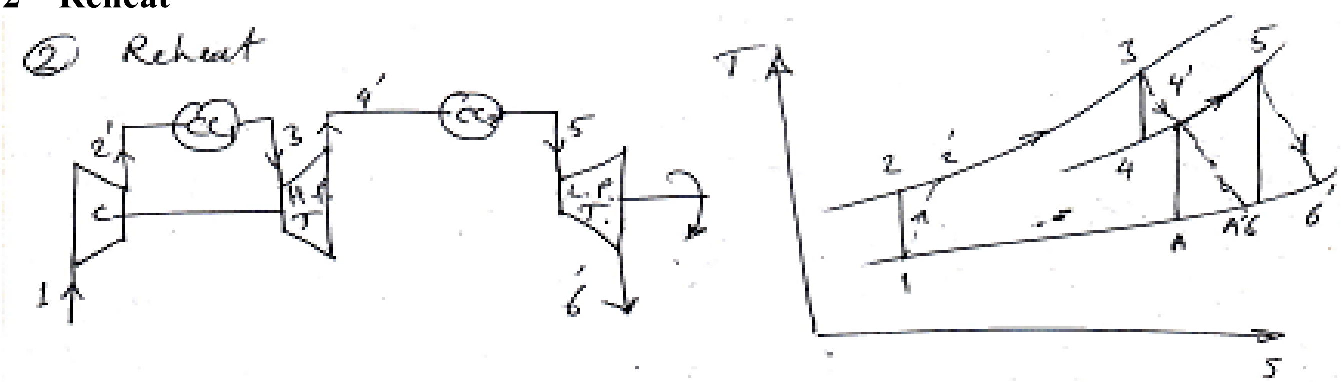
Work input (no intercooling)= $Cp(T_{A'} - T_1) = Cp(T_{2'} - T_1) + Cp(T_{A'} - T_{2'})$

Heat supplied = $Cp\,(T_5 - T_{4'})$

Heat supplied (no intercooler)= $Cp\,(T_5 - T_{A'})$

Work ratio = $\frac{net\ work\ output}{gross\ work\ output} = \frac{work\ of\ expansion - work\ of\ compression}{work\ of\ expansion}$

2 – Reheat

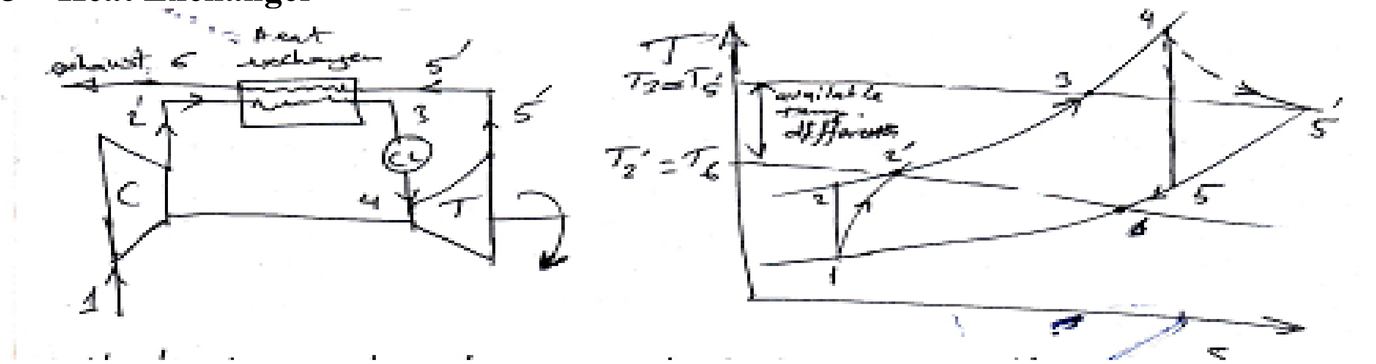


Work output of H.P.T =work input compression

$Cp(T2' - T1) = Cpg\,(T3 - T4')$

Network output = Cpg (T5-T6′) from low pressure turbine
Network output (no reheat)=Cpg($T_{4'}$ – TA′)
Heat supplied= C_{pg}(T3 – T2′) + C_{pg}(T5-T4′)

3 – Heat Exchanger



Heat given up by the gasses = Heat taken up by the air

$$m_a\, C_{pa}(T_3 - T_{2'}) = m_g C_{pg}\,(T_{5'} - T_6)$$

Heat exchanger eff.= $\dfrac{heat\ received\ by\ the\ air}{\max possible\ heat\ which\ could\ be\ transfered\ from\ the\ gasses\ on\ the\ heat\ exchanger} = \dfrac{m_a\, C_{pa}\,(T_3 - T_{2'})}{m_g C_{pg}(T_{5'} - T_{2'})}$

Thermal ratio = $\dfrac{temp.rise\ of\ the\ air}{\max temp.difference\ available} = \dfrac{T_3 - T_{2'}}{T_{5'} - T_{2'}}$

Heat supplied by the fuel(without H.E) =Cpg(T4 – T2′)
Heat supplied by the fuel(with H.E)= Cpg (T4 – T3)

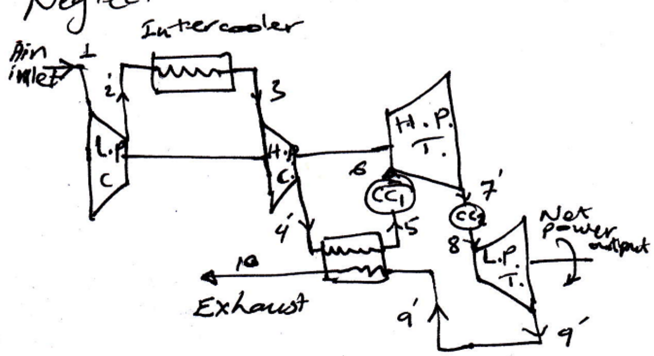
Example 2

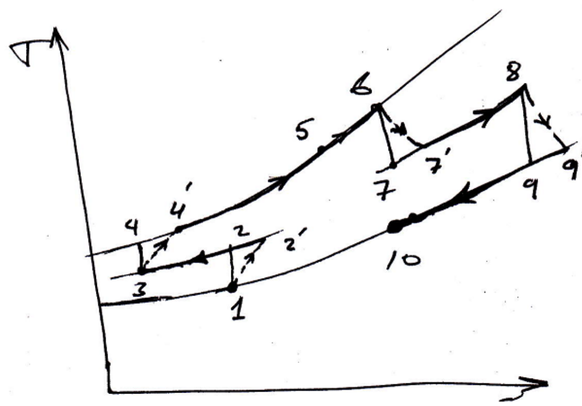
A 5000 kW gas turbine generating set operates with two compressor stages with intercooling between stages, the overall pressure ratio is 9/1. A high pressure turbine is used to drive the compressors, and a low pressure turbine drives the generator. The temperature of the gases at entry to the high pressure turbine is 650°C and the gases are reheated to 650°C after expansion in the first turbine. The exhaust gases leaving the low pressure turbine are passed through a heat exchanger to heat the air leaving the high pressure stage compressor. The compressors have equal pressure ratios and intercooling is complete between stages. The air inlet temperature to the unit is 15°C. The isentropic efficiency of each compressor stage is

0.8, and the isentropic efficiency of each turbine stage is 0.85; the heat exchanger thermal ratio is 0.75. A mechanical efficiency of 98% can be assumed for both the power shaft and the compressor turbine shaft. Neglecting all pressure losses and changes in kinetic energy, calculate the thermal efficiency and work ratio of the plant, and the mass flow in kg/s. For air take c_p is 1.005kJ/kg.K and γ=1.4, and for the gases in the combustion chamber and in the turbines and heat exchanger take c_p is 1.15kJ/kg.K γ=1.333. Neglect the mass of fuel.

Solution:

Neglect the mass of fuel.





Since the pressure ratio and isentropic eff. Of each compressor is the same , then the work input required for each compressor is the same since both compressor have the same air inlet

$T_1 = T_3$ $\quad\quad\quad\quad$ $T_{2'} = T_{4'}$

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

$$\frac{P_4}{P_1} = \frac{P_4}{P_2} * \frac{P_2}{P_1} = 9$$

$$\frac{P_2}{P_1} = \sqrt{9} = 3$$

$$T_2 = 288 * 3^{\frac{0.4}{1.4}} = 394K$$

$$\mu_{L.P.C} = \frac{T_2 - T_1}{T_{2'} - T_1} = 0.8 = \frac{394 - 288}{T_{2'} - 288} \quad\quad »» T_{2'} = 420.5 \text{ K}$$

Work input per compressor stage = $C_{Pa}\,(T_{2'} - T_1) = 1.005 * 132.5 = 133.1\frac{kj}{kg}$

The H.P.T is required to drive both compressor and to overcome mech. Friction ,

work output of H.P.T. = $\frac{2 * 133.1}{0.98} = 272\frac{KJ}{KG}$

$C_{pg}(T_6 - T_{7'}) = 272$ »» $T_{7'} = 686.5$ K

$$\mu_{H.P.T} = \frac{T_6 - T_{7'}}{T_6 - T_7} = 0.85 = \frac{923 - 686.5}{923 - T_7} \quad »» T_7 = 645K$$

$$\frac{P_6}{P_7} = \left(\frac{T_6}{T_7}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{923}{645}\right)^{\frac{1.333}{0.333}} = 4.19$$

$$\frac{P_8}{P_9} = \frac{9}{4.19} = 2.147$$

$$\frac{T_8}{T_9} = \left(\frac{P_8}{P_9}\right)^{\frac{\gamma-1}{\gamma}} = 2.147^{(\frac{0.333}{1.333}} = 1.211 \quad »» T_9 = 762.6K$$

$$\mu_{L.P.T} = \frac{T_8 - T_{9'}}{T_8 - T_9} = 0.85 = \frac{923 - T_{9'}}{923 - 762.6} \quad \ggg T_{9'} = 286.7K$$

NET WORK OUTPUT = $C_{pg}\,(T_8 - T_{9'}) * 0.98$

= 1.15 *136.3 *0.98= 153.7 kj/kg

Thermal ratio of heat exchanger $= \frac{T_5 - T_{4'}}{T_{9'} - T_{4'}} = 0.75 = \frac{T_5 - 420.5}{786.7 - 420.5}$ $\ggg T_5 = 695.2k$

Heat supplied = $C_{pg}\,(T_6 - T_5) + C_{pg}\,(T_8 - T_{7'})$

= 1.15 (923 -695.2)+ 1.15(923-686.5)= 534KJ/Kg

$$\mu_{th} = \frac{w}{Q} = \frac{153.7}{534} = 0.288 \; or \; 28.8\ \%$$

$$gross\ work\ of\ the\ plant = work\ of\ H.P.turbine + work\ of\ L.P.turbine$$

$$272 + \frac{153.7}{0.98} = 429\frac{kj}{kg}$$

Work ratio = $\frac{net\ work\ output}{gross\ work\ output}$

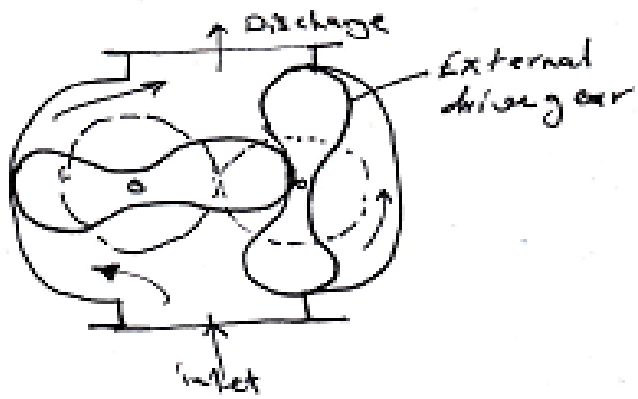
$= \frac{153.7}{429} = 0.358$

Power = m*W

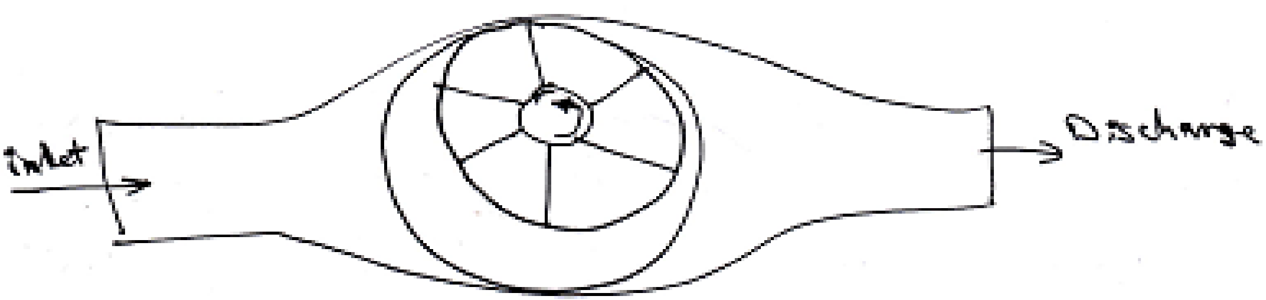
5000= m*153.7 $\lll m = 32.6\frac{kg}{s}$

Rate of flow of air = 32.6 kg/s

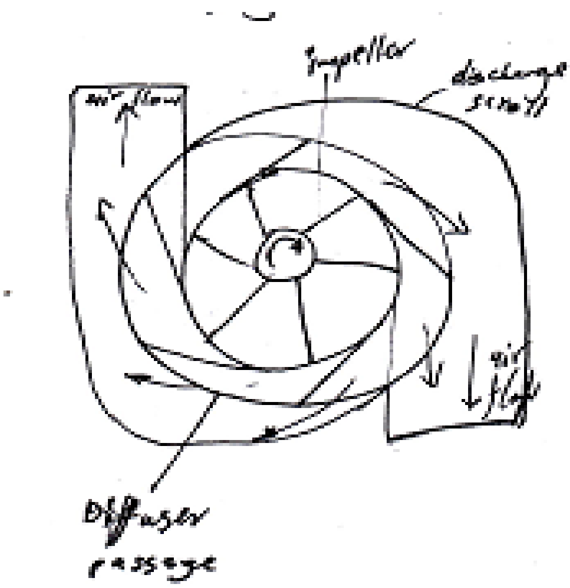
Rotary Compressor
1- Displacement compressor
a)roots blower.

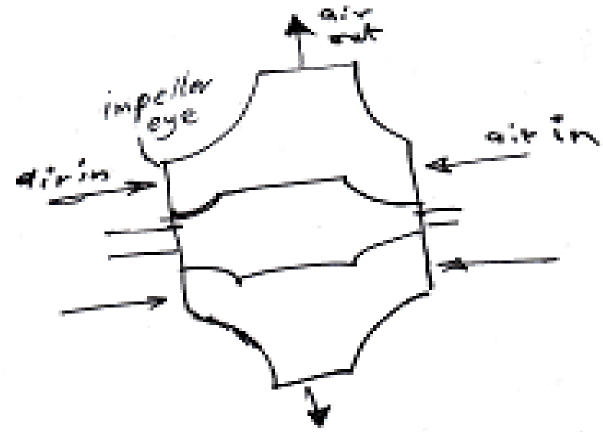


b) Vane blower

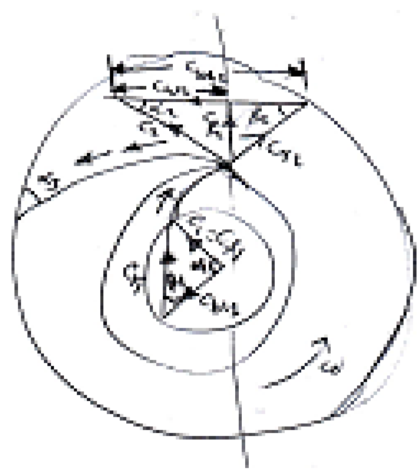


2) Steady flow compressors
a) centrifugal compressors





A centrifugal compressor consists of an impeller with a series of curved radial vanes . Air is drawn in near the hub, called the impeller eye. And is whirled at high speed by the vanes on the impeller rotate at high rotational speed.



C_{b11}:$Mean\ blade\ velocity\ at\ entrance$
C_{b12}:$Mean\ blade\ velocity\ at\ exit$
C_1 :$absolute\ velocity\ at\ inlet\ to\ the\ rotor$
C_2 :$absolute\ velocity\ at\ outlet\ to\ the\ rotor$
C_{r1} :$relative\ velocity\ of\ air\ at\ entry\ of\ rotor$
C_{r2} :$relative\ velocity\ of\ air\ at\ exit\ of\ rotor$
C_{w1} :$velocity\ of\ whirl\ at\ inlet$
C_{w2} :$velocity\ of\ whirl\ at\ outlet$
C_{f1} :$velocity\ of flow\ at\ inlet$

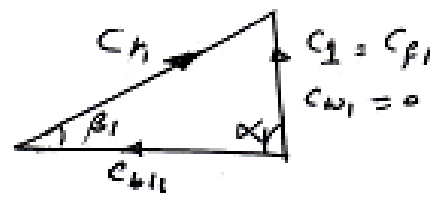
C_{f2} :$velocity\ of\ flow\ at\ outlet$

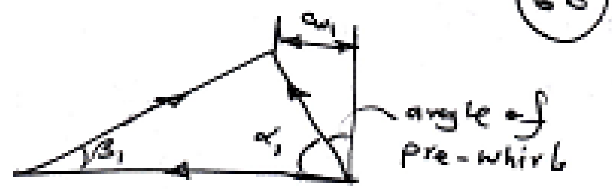
$\alpha 1$:$exit\ angle\ from\ the\ guide\ vane\ or\ inlet\ angle\ of\ the\ guide\ vane.$

β_1 :$Inlet\ angle\ to\ the\ rotor\ or\ impeller$

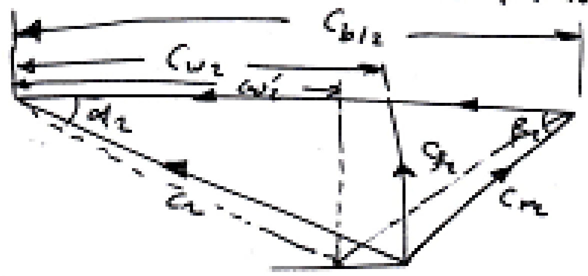
β_2 :$outlet\ angle\ to\ the\ rotor\ or\ impeller$

$\alpha_2 = inlet\ angle\ of\ diffuser$





the inlet velocity to the impeller eye is inclined at an angle by using fixed guide vanes is known as pre-whirl.



In practice the inertia of air trapped between the impeller blades causes the actual whirl velocity at exit (C_{w2}'), to be less than (C_{w2}), this phenomenon is known as slip.

$$\text{Slip factor} = \frac{C_{w2}'}{C_{w2}} = \frac{C_{w2}'}{C_{bl2} - C_{f2}\cot\beta_2}$$

$$\text{power input} = \dot{m}\left(C_{bl2}\,C_{w2}' - C_{bl1}\,C_{w1}\right)$$

$$\text{tangential force } F_t = \dot{m}\left(C_{w2} - C_{w1}\right)$$

$$\text{work done} = F_t \times C_{bl2} = \dot{m}\left(C_{w2} - C_{w1}\right) C_{bl2}$$

$C_{w1} = 0$ as air enters radially

61

Ex. A centrifugal compressor has a pressure ratio of 4/1 with an isentropic efficiency of 80% when running at 15000 rpm and inducing air at 20°C. Guide vanes at inlet give the air a pre-whirl of 25° to the axial direction at all radii and the mean diameter of the eye is 250 mm; the absolute air velocity at inlet is 150 m/s. At exit the blades are radially inclined and the impeller tip diameter is 590 mm. Calculate the slip factor of the compressor.

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

Sol.

Temp. after isentropic compression $= (20+273) \times 4^{0.286} = 435.4\,K$

Isentropic temp. rise $= \overset{T_2}{435.4} - \overset{T_1}{293} = 142.4\,K$

Actual temp. rise $= \frac{142.4}{0.8} = 178\,K$

power input per unit mass flow rate $= C_p \times$ actual temp. rise (ΔT)

$= 1.005 \times 178 = 178.9$ kJ/kg

$C_1 = 150$ m/s

angle of prewhirl $= 25°$ $\qquad C_{bl_1} = \omega r$

$$C_{bl_1} = \frac{15000 \times \pi \times 250}{60 \times 10^3} = 196.4 \text{ m/s}$$

$C_{w_1} = C_1 \sin 25 = 150 \sin 25 = 63.4$ m/s

$$C_{bl_2} = \frac{15000 \times \pi \times 590}{60 \times 10^3} = 463.4 \text{ m/s} = C_{w_2}$$, since the blades are radial

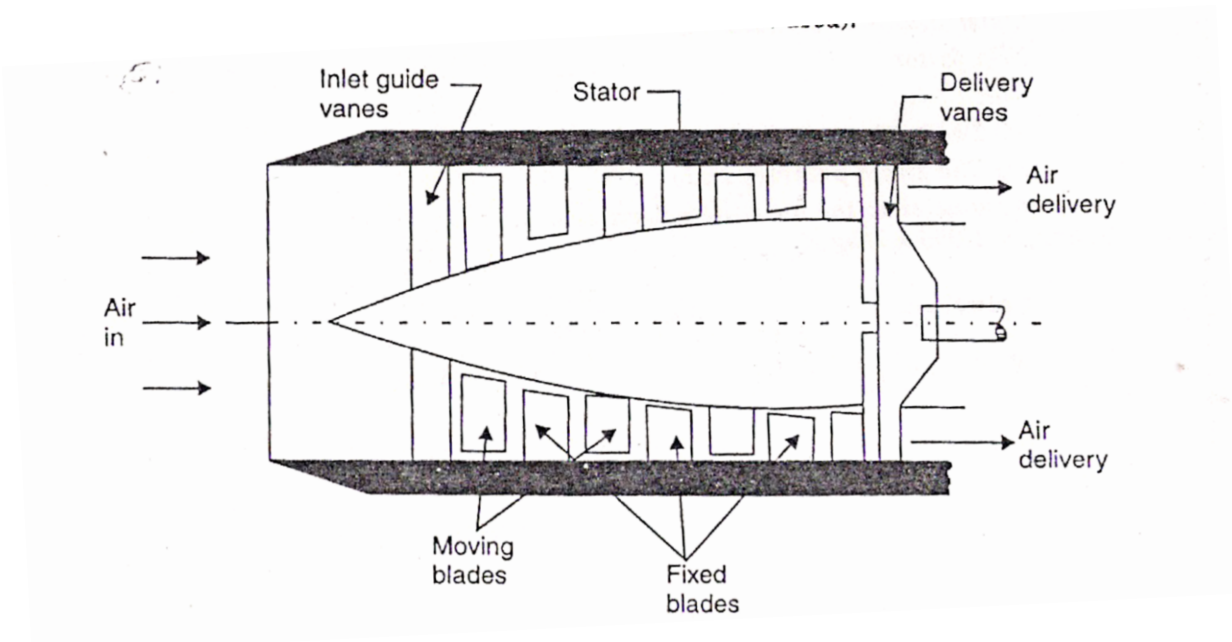
power input per unit mass flow rate $= C_{bl_2} C'_{w_2} - C_{bl_1} C_{w_1} = 178.9$ kJ

$463.4\, C'_{w_2} - 196.4 \times 63.4 = 178.9 \times 10^3$

$C'_{w_2} = 412.9$ m/s

$$\text{slip factor} = \frac{C'_{w_2}}{C_{w_2}} = \frac{412.9}{463.4} \simeq 0.89$$

Axial compressor



An axial flow compression stage consists of a row of moving blades arranged round the circumference of a rotor, and a row of fixed blades arranged round the circumference of a stator. The air flows axially through the moving and fixed blades in turn; stationary guide vanes are provided at entry to the first row of moving blades. The work input to the rotor shaft is transferred by the moving blades to the air, thus accelerating it. The blades are arranged so that the spaces between blades form diffuser passage, and hence the velocity of the air relative to the blades is decreased as the air passes through them, and there is a rise in pressure. The air is then further diffused in the stator blades, which are also arranged to form diffuser passages. In the fixed stator blades the air is turned through an angle so that its direction is such that it can be allowed to pass to a second row of moving rotor blades. It is usual to have a relatively large number of stages and to maintain a constant work input per stage (from 5 to 14 stage have been used).

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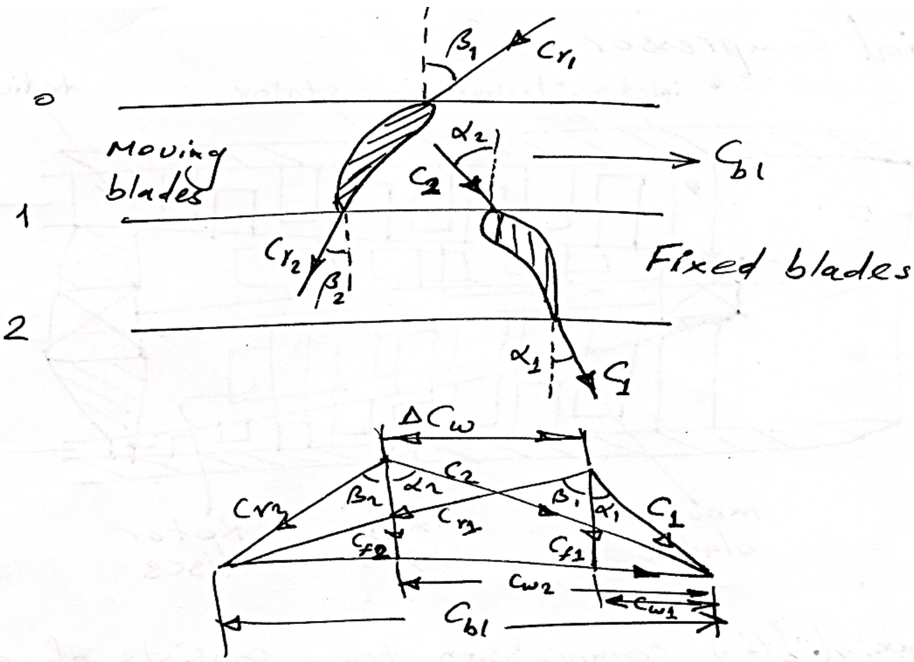


power input $= \dot{m}\, C_{b1}\, \Delta C_w$

ΔC_w : increase in the velocity of whirl

$\Delta C_w = C_{r_1} \sin\beta_1 - C_{r_2} \sin\beta_2$

$$\text{Degree of reaction} = \frac{\text{Enthalpy rise in rotor}}{\text{Enthalpy rise in the stage}}$$

$$= \frac{h_1 - h_0}{h_2 - h_0}$$

$$= \frac{C_{r_1}^2 - C_{r_2}^2}{(C_{r_1}^2 - C_{r_2}^2) + (C_2^2 - C_1^2)}$$

$$\text{Work done factor } (Y) = \frac{\text{Actual power input}}{\dot{m}\, C_{b1}\, \Delta C_w}$$ about 0.85

$$\text{Isentropic temp. rise for any stage} = \eta_s \times \frac{C_{b1}\, \Delta C_w}{C_p} \times (Y)$$

$$\text{Actual temp. at exit} = T_1 + y \left(\frac{C_{b1}\, \Delta C_w}{C_p}\right) Y$$

y : number of stages

actual temp. at inlet $= T_1 + (y-1)\dfrac{C_{b1}\,\Delta C_w}{C_p}\, y$

<u>EX.</u> In an axial flow air compressor producing a pressure ratio of 6/1 with air entering at 20°C the mean velocity of the rotor blades is 200 m/s and the inlet and exit angles of both the moving and the fixed blades are 45° and 15° respectively. The degree of reaction is 50%, the work done factor is 0.86 throughout, there are twelve stages, and the axial velocity may be taken as const. through the compressor. Calculate the isentropic efficiency of the compressor.

<u>Sol.</u>

Draw velocity diagram

$\Delta C_w = 115\ m/s$

Specific power input per stage $= C_{b1}\,\Delta C_w \times$ work done factor

$= 200 \times 115 \times 0.86 = 19780\ J = 19.78\ kJ$

Compressor specific power input $= 12 \times 19.78 = 237.4\ kJ$

For isentropic compression:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = 293 \times 6^{\left(\frac{1.4-1}{1.4}\right)} = 489.1\ K$$

Isentropic specific power input $= 1.005\,(489.1-293) = 197.1\ kJ$

Compression isentropic efficiency $= \dfrac{197.1}{237.4} = 0.83$ or 83%

	Particulars	Axial compressor	Centrifugal compressor
1	Type of flow	Axial	Radial
2	Pressure ratio per stage	Low (1.2:1)	High (5:1)
3	Efficiency	About 88%	About 82%
4	Frontal area	Small	Large
5	Range of operation	Narrow between surging and shocking limit	Wide (flat head capacity curve)
6	Starting torque	High	Low
7	Construction	Complex and costly	Simple and cheap
8	Application	Jet engine, gas turbine power plants	Low pressure refrigeration, big central air conditioning plants, supercharging, gas pumping

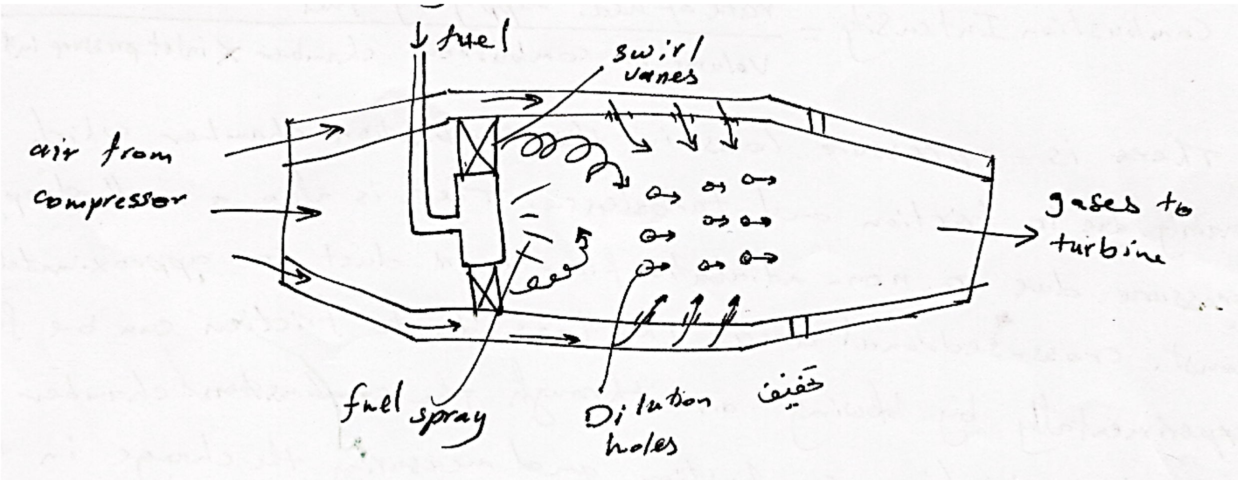
Combustion Chamber

In the closed cycle gas turbine unit heat is transferred to the air in a heat exchanger, but in the open cycle unit the fuel must be sprayed into the air continuously, and combustion is a continuous process unlike the cyclic combustion of the I.C. engine.

There are two main combustion systems for open cycles; one in which the air leaving the compressor is split into several streams and each stream is supplied to a separate cylindrical “can” type combustion chamber; and the other in which the air flows from the compressor through an annular combustion chamber. The annular type would appear to be more suitable for a unit using an axial flow compressor, but it is difficult to obtain good fuel / air distribution and research and development work on this type is harder than with the simpler can type. The annular type can be modified by having a series of interconnected cans placed in a ring; this is known as the annular type.

In industrial plants where space is not important, the combustion may be arranged to take place in one or two large cylindrical combustion chambers with ducting to convey the hot gases to the turbine; this system gives better control over the combustion process.

In all types of combustion chamber, combustion is initiated by electrical ignition, and once the fuel starts burning, a flame is stabilized in the chamber. In the can type it is usual to have interconnecting pipes between cans, to stabilize the pressure and to allow combustion to be initiated by a spark in one chamber on starting up.



Some of the air from the compressor is introduced directly to the fuel burner; this is called primary air, and presents about 25% of the total air flow. The remaining air enters the annulus round the flame tube, thus cooling the upper portion of the flame tube, and then enters the combustion zone through dilution holes. The primary air forms a comparatively rich mixture and the temperature is high in this zone. The air entering the dilution holes completes the combustion and helps to stabilize the flame in the high temperature region of the chamber.

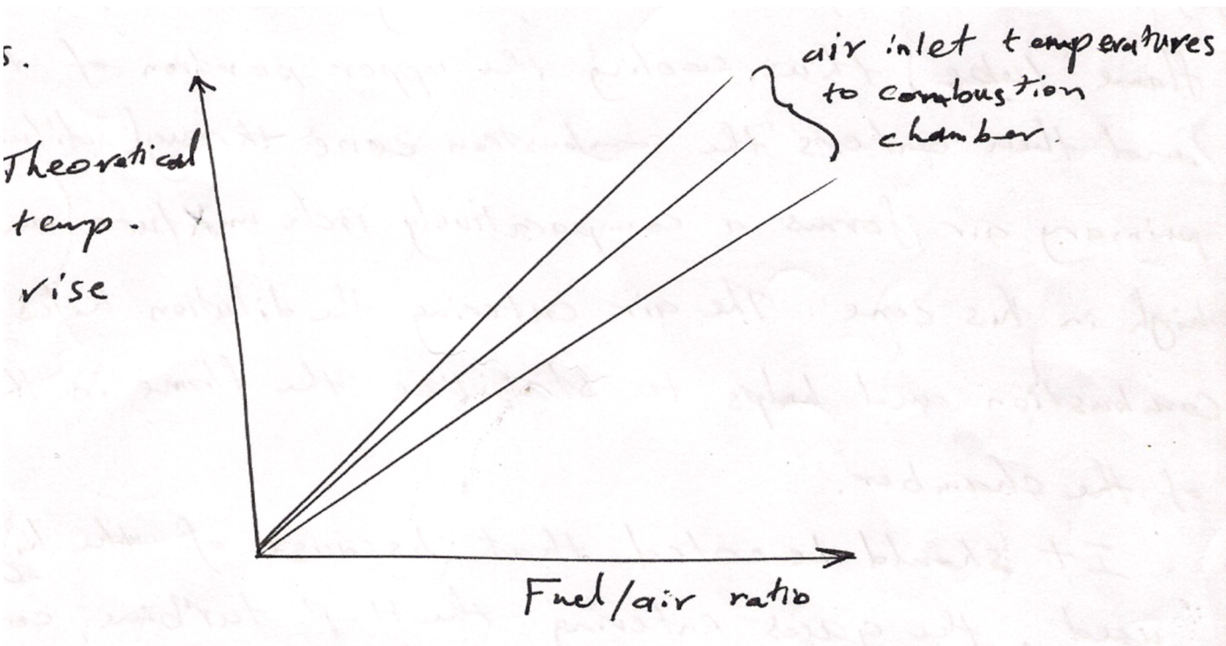
It should be noted that because of the high air /fuel ratios used, the gases entering the H.P. turbine contain a high percentage of oxygen, and therefore if reheating is performed between turbine stages, the additional fuel can be burned satisfactorily in the exhaust gas from the H.P.turbine.

$$combustion\ efficiency = \frac{actual\ temperature\ rise}{theoretical\ temperature\ rise}$$

The theoretical temperature rise is a function of the calorific value of the fuel used, the fuel/air ratio, and the initial temperature of the air.

$$combustion\ intensity = \frac{rate\ of\ heat\ supplied\ by\ fuel}{volume\ of\ combustion\ chamber\ \times inlet\ pressure\ in\ Atm.}$$

There is a pressure loss in the combustion chamber which is mainly due to friction and turbulence. There is also a small drop in pressure due to non-adiabatic flow in a duct to approximately constant cross sectional area. The loss due to friction can be found experimentally by blowing air through the combustion chamber without initiating combustion and measuring the change in total pressure. This friction loss in pressure is therefore called the cold loss. The loss due to the heating process alone is called the fundamental loss.



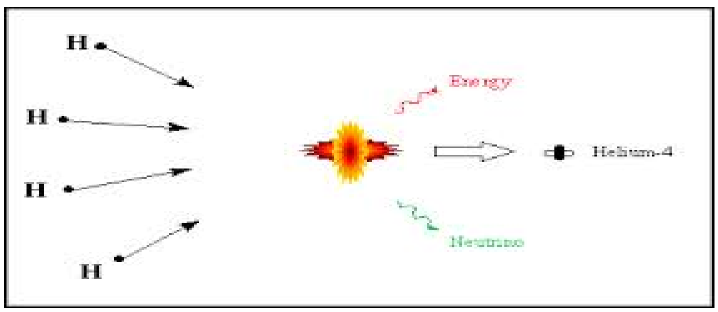
Nuclear Power Plants:

The energy released in nuclear reaction is very large in comparison to the chemical reaction. The heat evolved by one kg of uranium is equivalent to the combustion of 1,130,350 quintals of coal or 30×10^6 litres of diesel oil.

Nuclear reactions are of three types namely fusion, fission and radioactivity.

1- Fusion:

Energy is produced in the sun and stars by continuous fusion reactions. In this fusion process four nuclei of hydrogen fuse in a series of reactions and culminates in one nucleus of helium and two positrons :



$4\,{}_1H^1 \rightarrow {}_2He^4 + 2\,{}_1e^0$

This results in a decrease in mass of about 0.0276 amu (atomic mass unit) and thus release of 25.7 MeV. Many problems have to be solved before artificially made fusion reactor becomes a reality.

2- Fission

Fission is a practical preposition and can be caused by the neutron, which being electrically neutral, can strike and fission the positively charged nucleus at high, moderate or low speeds without being repulsed and sustains chain reaction because two or three neotrons are usually released for each one absorbed in fusion. These keep the reactions going.

$\Delta E = \Delta m\, C^2$ Einstein's Law

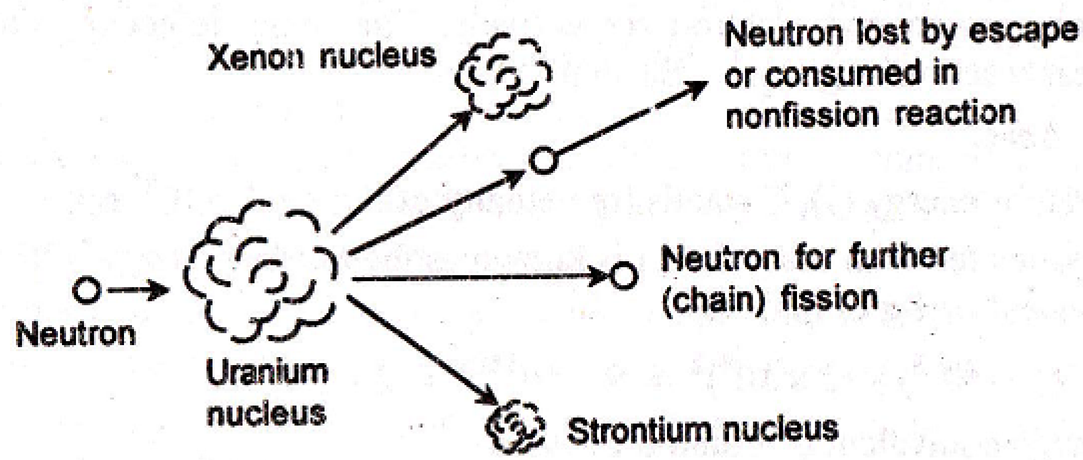
C: velocity of light = 3×10^8m/s

1 eV= 1.6021×10^{-9}J= 4.44×10^{-26}kWh,

therefore energy equivalent of 1 amu of mass is:

$\Delta E = (1.66 \times 10^{-27}) \times (3 \times 10^{8})^{2} = 14.94 \times 10^{-11}\text{J} = 9.3 \times 10^{8}$eV= 931MeV

$${}_{92}U^{235} + {}_{0}n^{1} \rightarrow {}_{54}Xe^{140} + {}_{38}Sr^{94} + 2{}_{0}n^{1}$$



This reaction yields 196 MeV.

$${}_{92}U^{235} + {}_{0}n^{1} \rightarrow {}_{56}Ba^{137} + {}_{36}Kr^{97} + 2{}_{0}n^{1}$$

Taking mass balance:

235.0439+1.00867→136.9061+96.9212+2×1.00867

Or 236.0526→235.8446

$$\Delta m = 235.8446 - 236.0526 = -0.208 \text{ amu}$$

$$\Delta E = 931 \times (-0.208) = -193.6\ MeV$$

The total energy produced per fission reaction is about 200 MeV. The complete fission of 1g of U^{235} nuclei produces:

$$Energy\ released = \frac{Avogadro's\ No.}{\mathrm{U}^{235} isotope\ mass} \times 200\ MeV = \frac{0.60225 \times 10^{24}}{235.0439} \times 200$$

$= 0.513\times10^{24}$ MeV $= 2.276\times10^{4}$ kWh = 0.948 MW-day

3- Radioactivity:

Radioactivity is one of the important aspect of nuclear science. It provides an important source of energy for small power devices and a source of radiation for use in research industry, medicine and a wide variety of applications as well as an environment concern.

Principal components of a nuclear reactor:

The core of the reactor is its heart, the place where the nuclear fuel is placed and where the nuclear reaction takes place. The fuel is most frequentlyformed into pellets roughly 2 cm in diameter and 1 cm long. These pelletsare loaded into a fuel rod, a hollow tube of a special corrosion-resistantmetal; this is frequently a zirconium alloy. Each fuel rod is 3–4m long anda single reactor core may contain close to 50,000 such rods. Fuel rods must be replaced once the fissile uranium-235 they contain has been used up. This is a lengthy process which can take as much as 3 weeks to complete.In between the fuel rods there are control rods, made of boron, whichare used to control the nuclear reaction. These rods can be moved in andout of the core. The core will also contain a moderator to slow the neutrons
released by the fission of uranium atoms. In some cases the moderator isalso the coolant used to carry heat away from the core.

The outside of the core may be surrounded by a material which acts asa reflector to return some of the neutrons escaping from the core. Thishelps maintain a uniform power density within the core. There may alsobe a similar reflecting material in the centre of the core.

The coolant collects heat within the core and transfers to an externalheat exchanger where it can be exploited to raise steam to drive a steamturbine. The coolant may be water (light water), deuterium (heavy water),a gas such as helium or a metal such as sodium. The core and its ancillaryequipment is normally called the 'nuclear island' of a nuclear power plantwhile the boiler, steam turbine and generator are called the 'conventionalisland'. The coolant system will link the nuclear and conventional islands.

A nuclear power plant will contain a host of systems to ensure that theplant remains safe and can never release radioactive material into the environment. The most important of these is the containment. This is a heavy concrete and steel jacket which completely surrounds the nuclear reactor. Inthe event of a core failure it should be able to completely isolate the core fromthe surroundings and remained sealed, whatever happens within the core.

Components summary:

1- Fuel core ($_{92}U^{235}$, $_{92}U^{233}$, $_{94}Pu^{239}$)

2- Moderator (to slow down the neotron (1MeV or 13200km/s) to (0.25eV or 2200m/s), H_2, D_2(Deuterium), N_2, O_2, C, Be(Beryllium)).

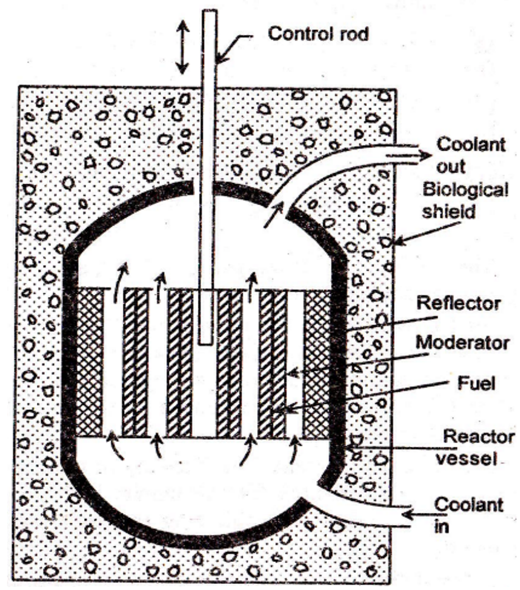
3- Reflector (moderator materials or H_2O, D_2O).

4- Coolant (the water, heavy water, gas (He, CO_2), a metal in liquid from Na and organic liquids). The good coolant should: not absorbs the neotrons, non-corrosive, have high boiling point (if liquid) and low melting point (if solid), non-oxidizing and non-toxic, high density, low viscosity, high conductivety and high specific heat.

5- Control rods (contain neotron absorber such as boron, cadmium or indium).

6- Biological shield (prevent damage of human body due to radiation).

7- Reactor vessel (encloses the reactor core, reflector and shield. Withstands the pressure at 200 bar or above).



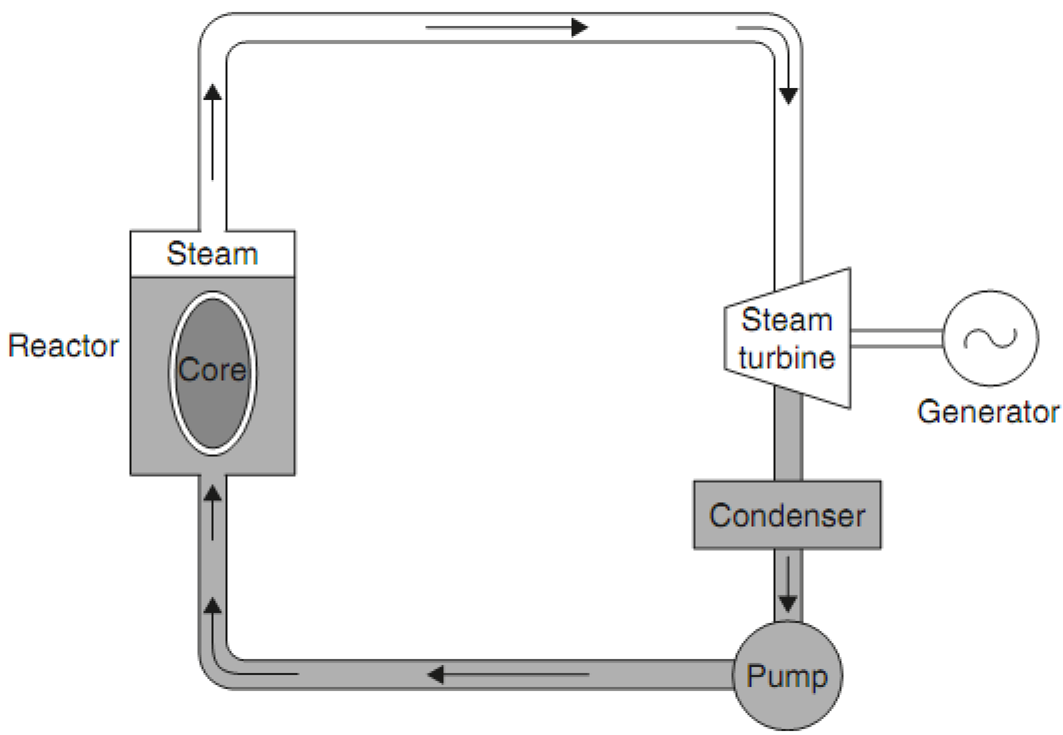
Principal parts of a nuclear reactor

Types of nuclear reactors:

i. Pressurized Water Reactor (PWR)
ii. Boiling Water Reactor (BWR)
iii. Canadian – Deuterium- Uranium Reactor (CANDU)
iv. Steam Generator Heavy Water Reactor (SGHWR)
v. High Temperature Gas Cooled Reactor (HTGR)
vi. Advanced Gas Cooled Reactor (AGR)
vii. Gas Cooled Reactor (GCR)

Boiling Water Reactor

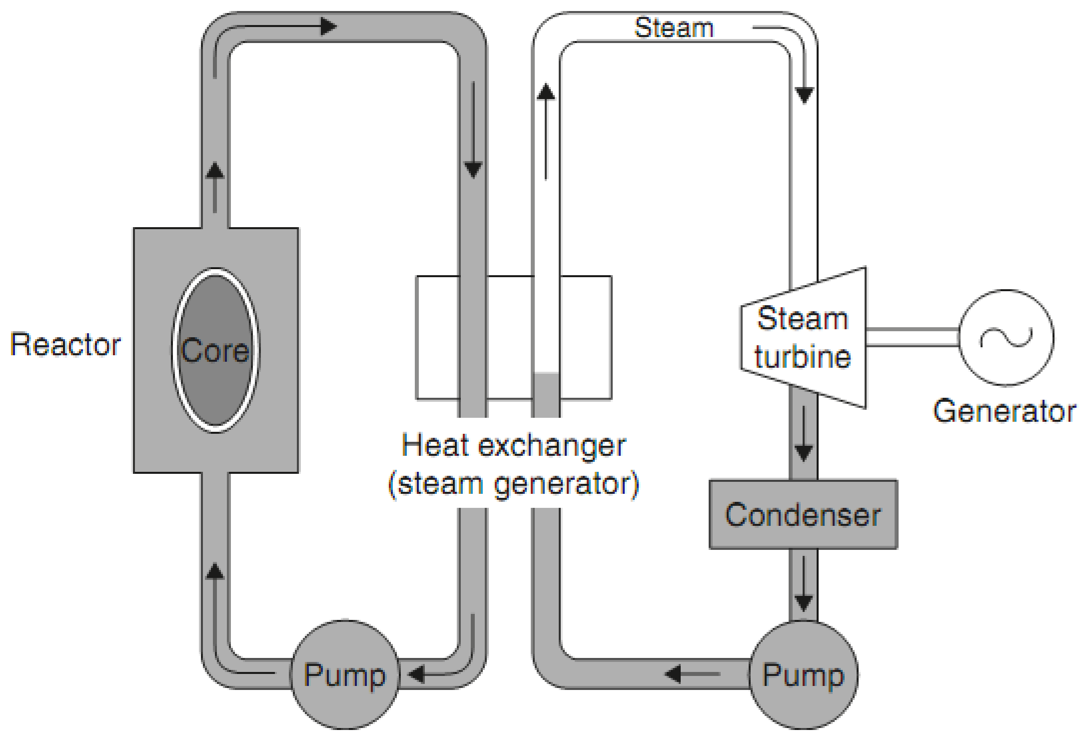
The boiling water reactor (BWR) uses ordinary water (light water) as bothits coolant and its moderator. In the BWR the water in the reactor core ispermitted to boil, and the steam generated is used directly to drive a steamturbine. This steam is then condensed and recycled back to the reactor core.This arrangement represents probably the simplest possible for anuclear reactor because no additional steam generators are required.However the internal systems within a BWR are complex. Steam pressureand temperature are low compared to a modern coal-fired power plantand the steam turbine is generally very large. BWRs have capacities of upto 1400MW and an efficiency of around 33%.The BWR uses enriched uranium as its fuel. This fuel is placed into thereactor in the form of uranium oxide pellets in zirconium alloy tubes.Refuelling a BWR involves removing the top of the reactor. The core itselfis kept under water, the water shielding operators from radioactivity.In common with all reactors, the fuel rods removed from a BWR reactorcore are extremely radioactive and continue to produce energy for someyears. They are normally kept in a carefully controlled storage pool at theplant before, in principle at least, being shipped for either reprocessing orfinal storage.



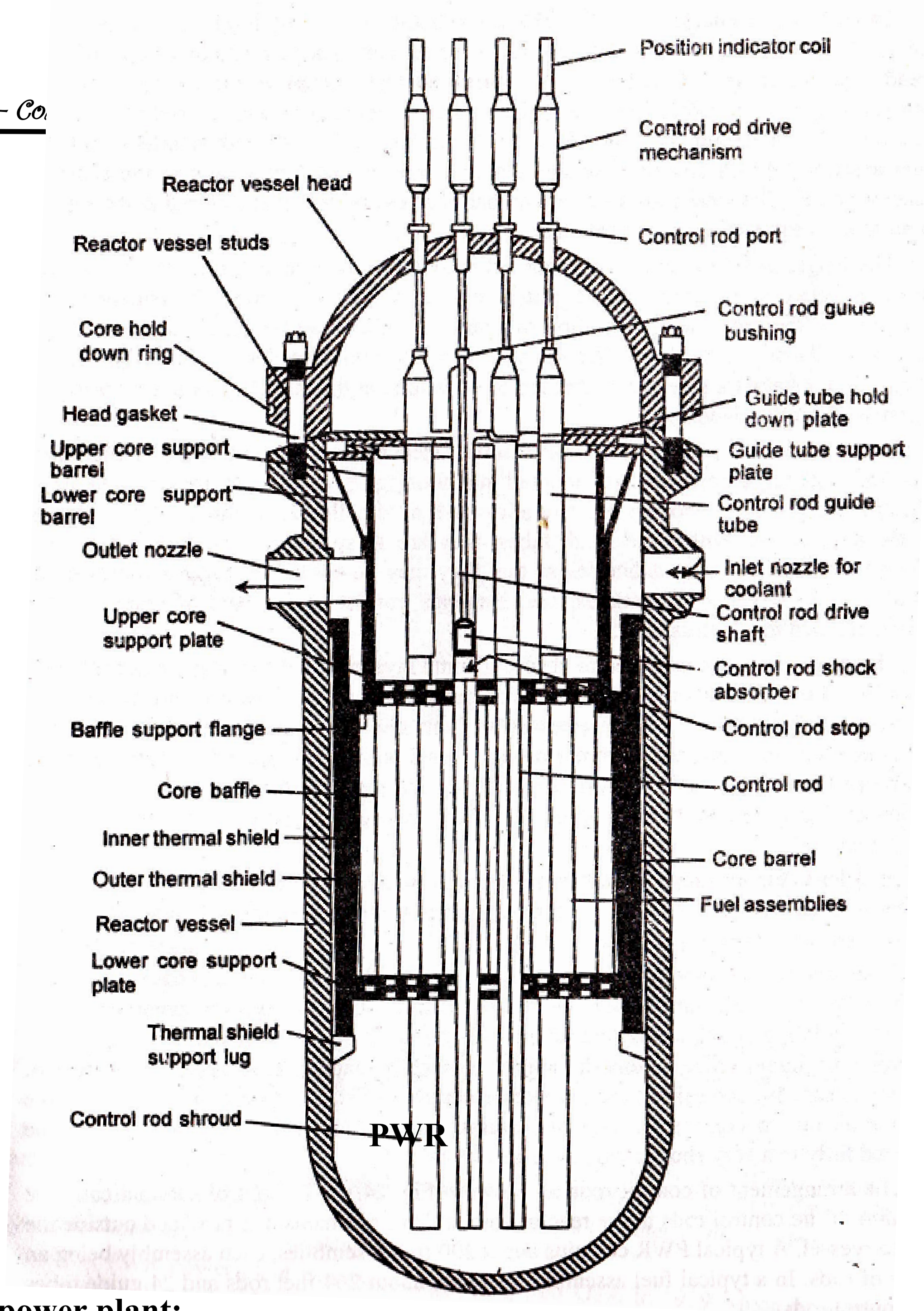
BWR

Pressurised Water Reactor

The pressurised water reactor (PWR) also uses ordinary or light water asboth coolant and moderator. However in the pressurised water system thecooling water is kept under pressure so that it cannot boil.The PWR differs in another respect from the BWR; the primary coolantdoes not drive the steam turbine. Instead heat from the primary watercooling system is captured in a heat exchanger and transferred to water ina secondary system. It is the water in this second system which is allowedto boil and generate steam to drive the turbine.The use of a second water cycle introduces energy losses which makethe PWR less efficient at converting the energy from the nuclear reactioninto electricity. However the arrangement has other advantages regardingfuel utilization and power density, making it competitive with the BWR.The PWR uses enriched uranium fuel with a slightly higher enrichmentlevel than in a BWR. This is responsible for a higher power density withinthe reactor core. As with the BWR, the fuel is introduced into the core in theform of uranium oxide pellets. A typical PWR has a generating capacity of1000MW. The efficiency is around 33%.



PWR



Hydro-electric power plant:

The hydraulic turbine is a prime mover in which the potential energy of water is converted into mechanical work of rotary shaft. According to the hydraulic machine relationship the input power to the turbine is calculated as follows:

$$P_{ti} = \gamma . Q . H$$

$$\eta = \frac{P_t}{P_{ti}}$$

P_{ti}: input power to the turbine (W)

P_t: output power from the turbine (W)

γ : specific weight of water (N/m^3)

H: net head of water (m)

Q: flow rate of water (m^3/s)

$$E = 9.81\ Q\ H\ \eta\ t$$

E: the electricity generated in kWh

t: the operating time in hours (8760 h/year)

Advantages of hydro p.p.:

1. No fuel requirement.
2. Low running cost.
3. No problem of disposal of ash.
4. Pollution free electricity generation.
5. Easily switched on and off in a short period.
6. Simple in concept, self-contained and reliable in operation.
7. Greater life expectancy.
8. Act as ideal spinning reserve.
9. Higher plant efficiency.
10. Ancillary benefits.
11. Less skilled workers.
12. Quick response to the change of load.

Disadvantages:

1) High capital cost.
2) Power dependent on quantity of water available.
3) Site selection dependent on water availability.
4) Long erection time.

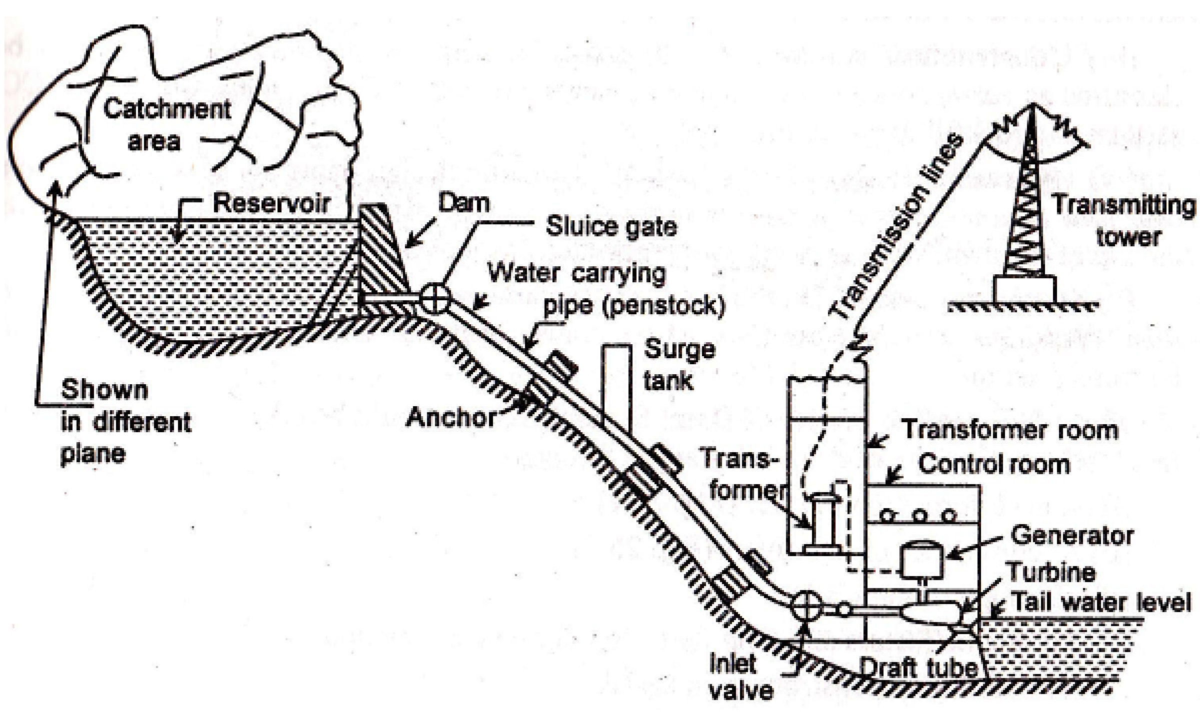
5) Disturbed ecology of the area.

Site selection for a hydro power plant:

1- Availability of water.
2- Water storage capacity.
3- Available water head.
4- Accessibility of site.
5- Distance from the load centre.
6- Type of the land on site.
7- Environmental aspects.
8- Sociological aspects.

Main elements of a hydro power plant:

1- Reservoir, 2- Dam, 3- Sluice Gate or Valve, 4- Penstock or Conduit,5- Surge Tanks, 6- Power House



Hydro p.p.

Wind Power Plants:

Characteristics of Wind Energy:

1- Wind power systems do not pollute the atmosphere.
2- Fuel provision and transport are not required in wind power systems.
3- Wind energy is a renewable source of energy.
4- Wind energy when produced on small scale is cheaper, but competitives with conventional power generating systems when produced on a large scale.

Wind energy problems:

1. It is fluctuating in nature.
2. Due to its irregularity it needs storage devices.
3. Wind power generating systems produce ample noise.

Types of Wind Mills:

1) Multiple blade type.
2) Savonius type.
3) Darrieus type.

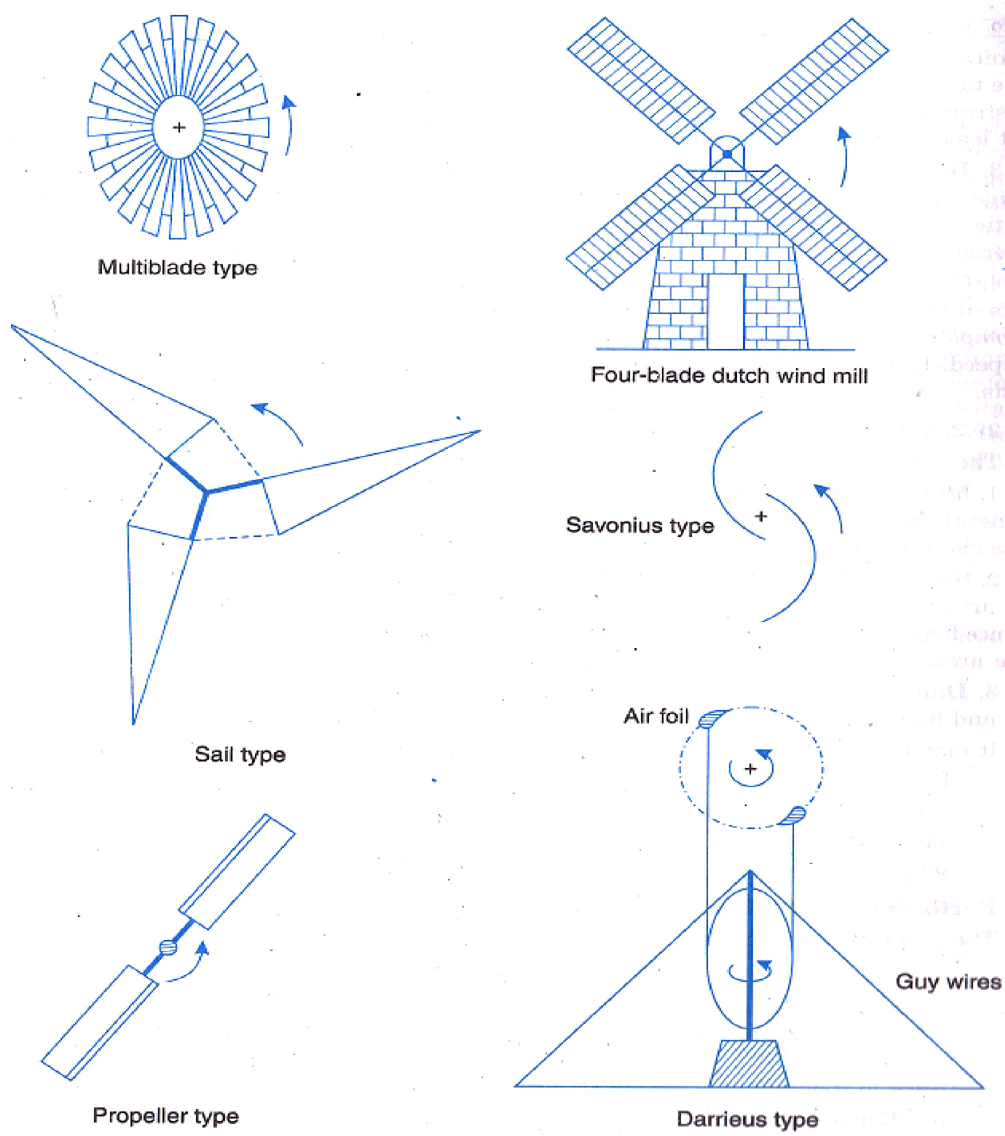
$$wind\ mill\ coefficient\ of\ performance = \frac{power\ delivered\ by\ the\ rotor}{maximum\ power\ available\ in\ the\ wind}$$

$$K_p = \frac{P}{P_{max}} = \frac{P}{\frac{1}{2}\rho A {U_w}^3}$$

ρ:density of air

A: swept area

U_w : velocity of wind

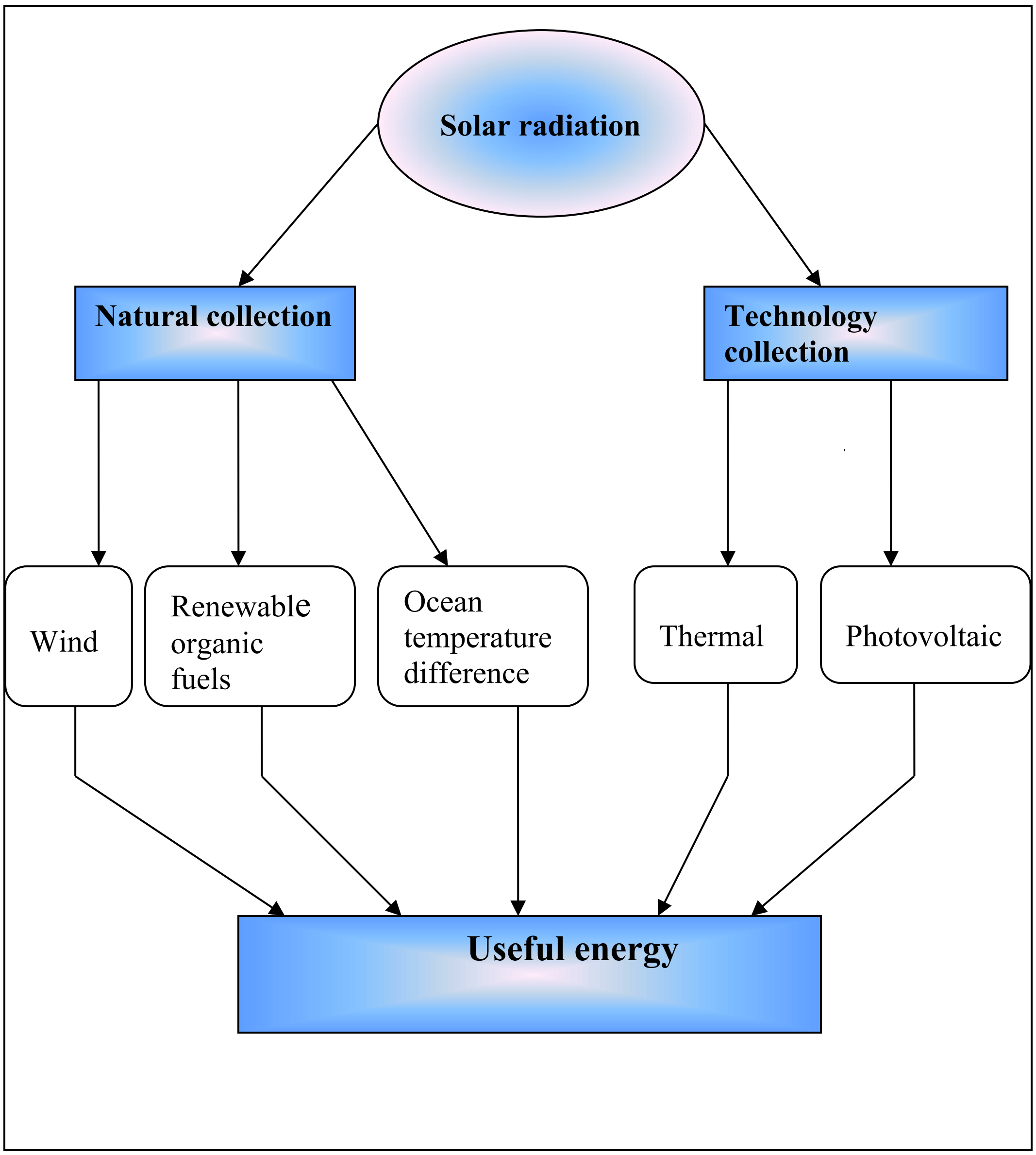


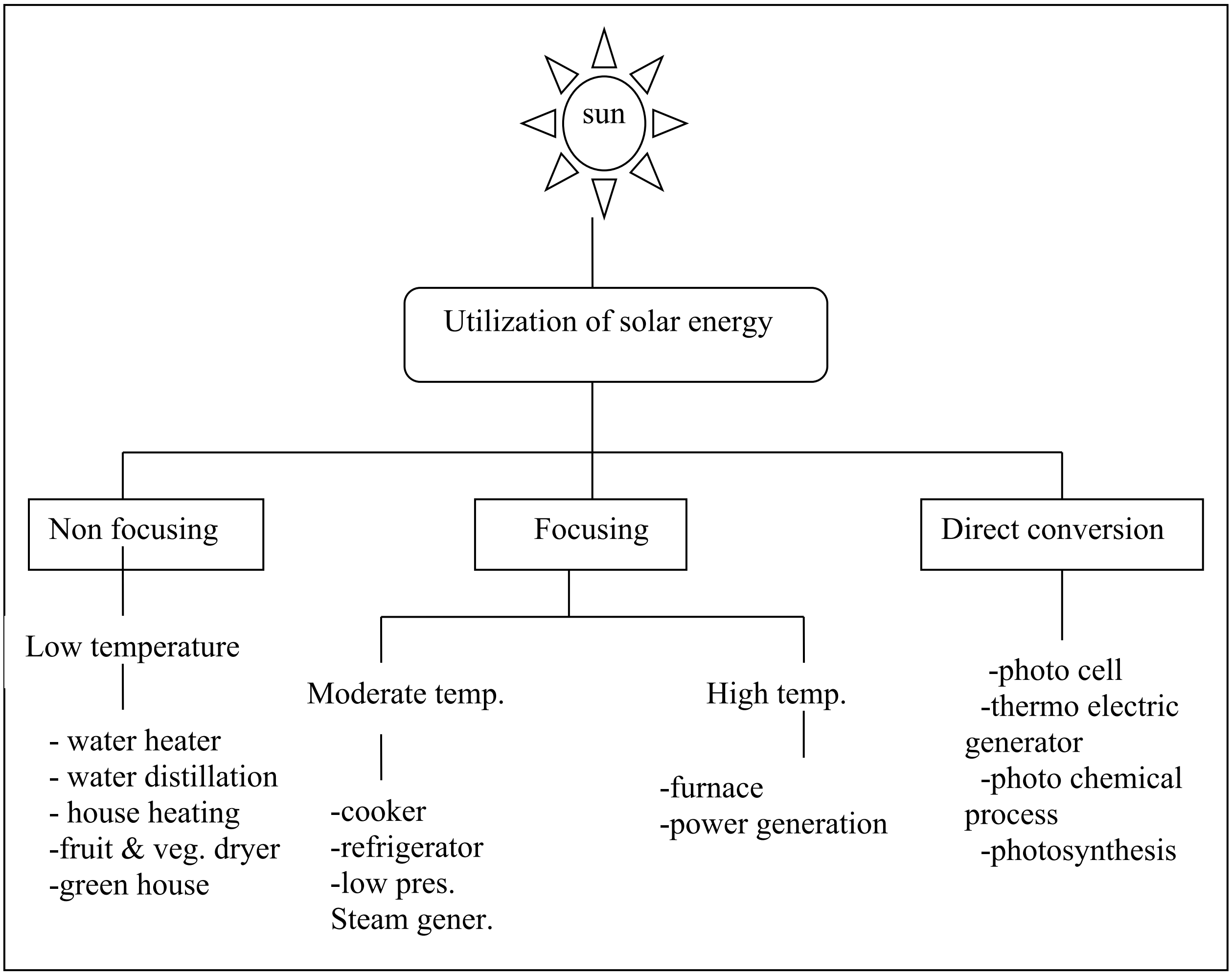
Problems in Operating Large Wind Power Generators:

1) Location of site.
2) Variation in wind velocity.
3) Need of a storage system.
4) Strong supporting structure.
5) Occupation of large areas of land.

Solar power plants:

1- Parabolic trough
2- Parabolic dish
3- Solar tower
4- Solar pond
5- Central reciever





$$\alpha + \rho + \tau = 1$$

$$\alpha P + \acute{\alpha}\acute{P} = h_c(T - T_a) = \varepsilon\sigma T^4$$

α: absorption coefficient of direct arrays

$\acute{\alpha}$: absorption coefficient of diffused radiations

ρ: reflection coefficient

τ: transmission coefficient

P: intensity of sun radiation

$\acute{P}$:intensity of the diffused radiations

σ: Stephan- Boltzman's constant = $5.67*10^{-8}$

ε: emission coefficient

T: body temperature

T_a:atmospheric temperature

h_c : convection heat transfer coefficient