NM INSTITUTE OF ENGINEERING & TECHNOLOGY
Sijua, Patrapada, Bhubaneswar-751019,Odisha

DEPARTMENT OF MECHANICAL ENGINEERING

LECTURE NOTES ON

THERMAL ENGINEERING-II

Er. SANGRAM BEHERA
ASSIATANT PROFESSOR

27/06/2019

THERMAL ENGINEERING - II

Unit-1 STEAM NOZZLE:

Introduction to steam nozzle:

→ Steam nozzle:

* Nozzle is a device with varying cross-sectional area to increase the velocity and corresponding to the pressure drop.

* The main function of a nozzle is to produce high velocity jet of stream.

* Nozzles are generally used in steam turbines, gas turbines, rocket engines, flow measurement devices and many other industrial applications.

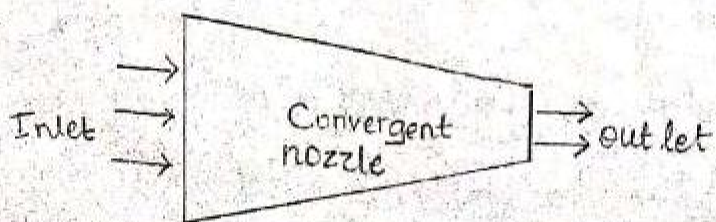
→ Types & Shapes of nozzles:

* The foll. 3 types of nozzles are generally used in industrial application,

(a) Convergent nozzle.
(b) Divergent nozzle.
(c) Convergent - Divergent nozzle.

• Convergent nozzle:

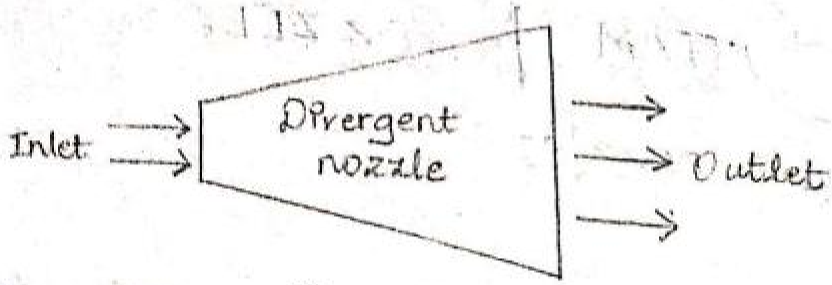
* In convergent nozzle, the cross-sectional area decreases from inlet section to outlet section.



* Mainly used in steam engines, etc.

• **Divergent nozzle:**

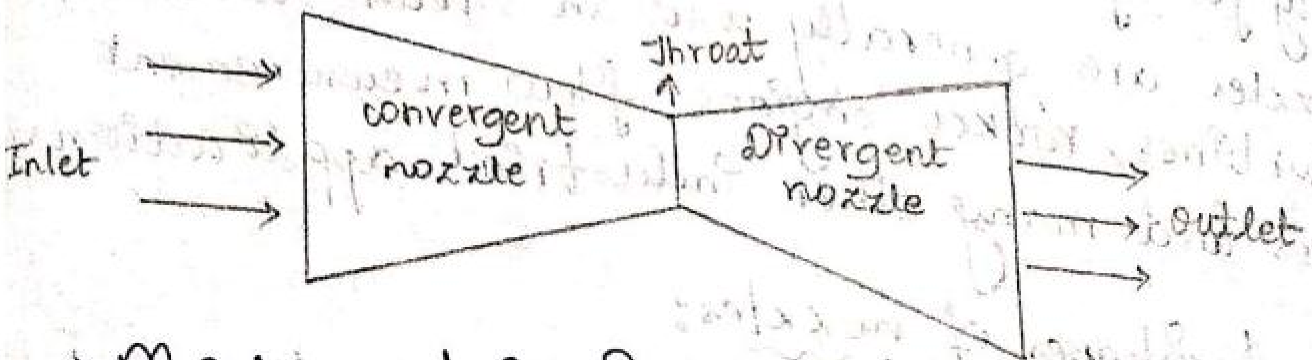
* In divergent nozzle, the cross-sectional area increases from inlet section to outlet section.



* Mainly used in Rocket engines, etc.

• **Convergent - Divergent nozzle:**

* In convergent - divergent nozzle, first the cross-sectional area decreases from inlet section to throat and increases from throat to outlet section.



* Mainly used in Rocket engines, etc.

* Mac number determines the purpose of usage of nozzle at that work.

$$m = c/a$$

generally, $m = 2, 2.5$ used in aeroplanes, jets but $m > 3$ used for supersonic, ultrasonic jets.

Important topics:
* Velocity of nozzle (C_2) exit.
* Mass flow rate.
* Max. condition for discharge.
* Critical pressure ratio. (P_2/P_1)
* Efficiency of nozzle.

vary them

- Steam flow through a nozzle:
* The flow of steam in a nozzle has assumed as adiabatic process.
* There is no heat supplied (or) rejected during the flow of steam, there is no work transfer during the flow.
* So, $Q=0$
$W=0$

- Velocity of nozzle: (C_2)
* Steam enters the nozzle at high pressure & low velocity. It leaves from nozzle at high-velocity and low pressure.
* The exit velocity of nozzle are calculated by foll. method,

Consider the flow of steam at unit mass,

let, $C_1 \rightarrow$ velocity of steam at inlet of nozzle (m/s)

$C_2 \rightarrow$ velocity of steam at outlet of nozzle (m/s)

$h_1 \rightarrow$ Enthalpy at inlet of nozzle (KJ/kg)

$h_2 \rightarrow$ Enthalpy at outlet of nozzle (KJ/kg)

By Steady flow energy equation,

$$mgz_1 + \frac{1}{2}mC_1^2 + U_1 + P_1V_1 + Q = mgz_2 + \frac{1}{2}mC_2^2 + U_2 + P_2V_2 + W$$

by applying adiabatic condition,

$(m=1)$ $(Q=0)$ $(W=0)$ $(z_1 = z_2)$

$$\frac{1}{2}C_1^2 + h_1 = \frac{1}{2}C_2^2 + h_2 \qquad \because (h = u + pv)$$

So, $$h_1 - h_2 = \frac{C_2^2 - C_1^2}{2}$$

$$C_2^2 - C_1^2 = 2(h_1 - h_2)$$

compare to exit velocity, inlet velocity is very less,
so neglect the inlet velocity (C_1),

$$C_2^2 = 2(h_1 - h_2)$$

$$\therefore \quad C_2 = \sqrt{2(h_1 - h_2)}$$

Enthalpy unit is kJ/kg,

So, $C_2 = \sqrt{2000(h_1 - h_2)}$ → use when to find $(h_1 - h_2)$

$$C_2 = 44.72\sqrt{(h_1 - h_2)} \text{ m/s}$$

$\because h_1 - h_2 =$ enthalpy drop.

for convergent - divergent nozzle,

$C_t = \sqrt{2000(h_1 - h_t)}$ is used

to find velocity of throat.

28/06/2019

- **Mass flow rate:**

* The isentropic process is approximately represented by an equation,

$$[PV^n = C]$$

* Let, $P_1 \rightarrow$ initial pressure of steam at inlet.
  $C_1 \rightarrow$ initial velocity of steam at inlet.
  $v_1 \rightarrow$ specific volume of steam at inlet.
  $P_2 \rightarrow$ final pressure of steam at exit.
  $C_2 \rightarrow$ final velocity of steam at exit.
  $v_2 \rightarrow$ specific volume of steam at exit.

* Take, $n = 1.135$ for saturated steam, take $n = 1.3$ for superheated steam.

* The steam passed through the nozzle, its pressure is dropped. So, the enthalpy is also reduced.

* This reduction in enthalpy must be equal to the increase in kinetic energy. Hence, the work done by steam is equal to the drop of enthalpy.

wkt,
$$m = \frac{\text{vol. of flow of steam per second}}{\text{Specific volume of steam}} = \frac{V}{v}$$

$$m = \frac{A \times C_2}{v_2} \qquad \because A \rightarrow \text{area.}$$

$$m = \frac{A \times C_2}{v_2}.$$

Gain in K.E = work done by isentropic process given, $PV^n = C$

work done for isentropic process,

$$W = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

Gain in K.E = work done by isentropic process

$$\frac{C_2^2}{2} - \frac{C_1^2}{2} = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

as the inlet velocity (C_1) of steam is very less when compared to exit velocity (C_2) of steam, so, neglect initial velocity (C_1).

So, the above equation becomes,

$$\frac{C_2^2}{2} = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

$$\left[\frac{C_2^2}{2} = \frac{n}{n-1} P_1V_1 \left(1 - \frac{P_2V_2}{P_1V_1}\right)\right] \rightarrow ①$$

for isentropic process,

$$PV^n = C$$

So, $P_1V_1^n = P_2V_2^n$

we get,

$$\left[\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{1/n}\right] \rightarrow ②$$

(Specific volume ration)

Subs. ② in ①,

$$\frac{C_2^2}{2} = \frac{n}{n-1} P_1V_1 \left(1 - \frac{P_2}{P_1}\left(\frac{P_1}{P_2}\right)^{1/n}\right)$$

$$\frac{C_2^2}{2} = \frac{n}{n-1} P_1 V_1 \left(1 - \frac{P_2}{P_1}\left(\frac{P_2}{P_1}\right)^{-1/n}\right)$$

$$\frac{C_2^2}{2} = \frac{n}{n-1} P_1 V_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{1-1/n}\right)$$

$$C_2^2 = \frac{2n}{n-1} P_1 V_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)$$

$$\left[C_2 = \sqrt{\frac{2n}{n-1} P_1 V_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}\right] \rightarrow ③ \quad //$$

Subs. ③ in mass flow rate equation,

$$m = \frac{A \times C_2}{v_2}$$

$$\left[m = \frac{A}{v_2}\left(\sqrt{\frac{2n}{n-1} P_1 V_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}\right)\right] \rightarrow ④$$

* The mass of steam discharged through nozzle per second is obtained above.

$$m = \frac{\text{volume of steam flow through nozzle per second}}{\text{Specific volume at exit}}$$

from equation ②,

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{1/n} \rightarrow V_2 = V_1 \left(P_1/P_2\right)^{1/n}$$

$$m = \frac{A}{v_2}\left(\sqrt{\frac{2n}{n-1} P_1 V_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}\right)$$

$$m = \frac{A}{v_1 (P_1/P_2)^{1/n}}\left[\sqrt{\frac{2n}{n-1} P_1 v_1 \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}\right]$$

$$m = \frac{A}{v_1}\left(P_1/P_2\right)^{-1/n}\left[\sqrt{\frac{2n}{n-1} P_1 v_1 \left(1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}\right]$$

$$m = \frac{A}{v_1}\left(\frac{P_2}{P_1}\right)^{1/n}\left[\sqrt{\frac{2n}{n-1} P_1 v_1 \left(1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}\right]$$

$$m = \frac{A}{v_1}\left[\sqrt{\frac{2n}{n-1} P_1 v_1 \left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}\left(1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right)}\right]$$

$$m = \frac{A}{v_1}\left[\sqrt{\frac{2n}{n-1} P_1 v_1 \left(\left(\frac{P_2}{P_1}\right)^{2/n} - \left(\frac{P_2}{P_1}\right)^{\frac{2}{n}+\frac{n-1}{n}}\right.}\right]$$

$$m = \frac{A}{v_1}\left[\sqrt{\frac{2n}{n-1} P_1 v_1 \left(\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right)}\right]$$

$$\cancel{m = \frac{A}{v_1}} \quad \boxed{m = A\left[\sqrt{\frac{2n}{n-1}\,\frac{P_1}{v_1}\left(\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right)}\right]} \quad (5)$$

* The above equation is known as mass flow rate of steam.

$\because P_2/P_1 \rightarrow$ (critical pressure ratio)

by differentiating above equation of m, we get : max. condition equation.

$m_{max} \Rightarrow$

Maximum discharge condition: (by $\uparrow P_2/P_1$)

* From mass flow rate equation, the mass of steam discharged through nozzle,

$$m = \frac{A}{v_1}\sqrt{\frac{2n}{n-1} P_1 v_1 \left(\left(P_2/P_1\right)^{2/n} - \left(P_2/P_1\right)^{\frac{n+1}{n}}\right)}$$

* There is only one value of ratio (called critical pressure ratio) 'P_2/P_1', which will produce the maximum discharge.
* It can be obtained by the mass flow rate equation by differentiating '$\dot{m}$' with respect to 'P_2/P_1' and equating to zero.
* Other quantities except 'P_2/P_1' is constant.

$$\frac{d}{d(P_2/P_1)}\left[\left(P_2/P_1\right)^{2/n} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right] = 0$$

$$\frac{2}{n}\left[P_2/P_1\right]^{\frac{2}{n}-1} - \frac{n+1}{n}\left[P_2/P_1\right]^{\frac{n+1}{n}-1} = 0$$

$$2/n\left(P_2/P_1\right)^{\frac{2-n}{n}} = \frac{n+1}{n}\left(P_2/P_1\right)^{1/n} \rightarrow \left(\frac{P_2}{P_1}\right)^{\frac{2-n}{n}} = \frac{n+1}{2}\left(\frac{P_2}{P_1}\right)^{1/n}$$

$$\frac{2}{n}\left(P_2/P_1\right)^{2-n} = \left[\frac{n+1}{n}\left(P_2/P_1\right)^{1/n}\right]^{n} \quad \left(\frac{P_2}{P_1}\right)^{2-n} = \left(\frac{n+1}{2}\right)^{n}\left(\frac{P_2}{P_1}\right)$$

$$\frac{\left(P_2/P_1\right)^{2-n}}{\left(P_2/P_1\right)} = \left(\frac{n+1}{2}\right)^{n}$$

$$\left(P_2/P_1\right)^{1-n} = \left(\frac{n+1}{2}\right)^{n}$$

$$\left(P_2/P_1\right) = \left(\frac{n+1}{2}\right)^{\frac{n}{1-n}}$$

$$\left[\left(P_2/P_1\right) = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right] \rightarrow (6)$$

Subs. (6) in (5),

$$\dot{m}_{max.} = A\sqrt{\frac{2n}{n-1}\left(\frac{P_1}{v_1}\right)\left[\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}\times\frac{2}{n}} - \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}\times\frac{n+1}{n}}\right]}$$

$$\left[\dot{m}_{max.} = A\sqrt{\frac{2n}{n-1}\left(\frac{P_1}{v_1}\right)\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}\right] \rightarrow (7)$$

the above equation is known as, "Maximum discharge conditions for nozzle".

*Now, applying $n = 1.135$ for saturated conditions & $n = 1.3$ for superheated conditions for above equation,

(i) $n = 1.135$

$$\dot{m}_{max.} = A\sqrt{\frac{2n}{n-1}\left(\frac{P_1}{V_1}\right)\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

$$\dot{m}_{max.} = A\sqrt{\frac{2(1.135)}{1.135-1}\left(\frac{P_1}{v_1}\right)\left[\left(\frac{2}{1.135+1}\right)^{\frac{2}{1.135-1}} - \left(\frac{2}{1.135+1}\right)^{\frac{1.135+1}{1.135-1}}\right]}$$

$$\dot{m}_{max.} = A\sqrt{16.8148\left(\frac{P_1}{v_1}\right)\left[0.3799 - 0.3559\right]}$$

$$\dot{m}_{max.} = A\sqrt{16.8148\left(\frac{P_1}{v_1}\right)(0.024)}$$

$$\dot{m}_{max.} = A\sqrt{0.40355\,(P_1/v_1)}$$

$$\boxed{\dot{m}_{max.} = A\,(0.63525)\sqrt{P_1/v_1}}$$

(ii) $n = 1.3$

$$\dot{m}_{max.} = A\sqrt{\frac{2(1.3)}{1.3-1}\left(\frac{P_1}{v_1}\right)\left[\left(\frac{2}{1.3+1}\right)^{\frac{2}{1.3-1}} - \left(\frac{2}{1.3+1}\right)^{\frac{1.3+1}{1.3-1}}\right]}$$

$$\dot{m}_{max.} = A\sqrt{8.6666\left(\frac{P_1}{v_1}\right)\left[0.3938 - 0.3424\right]}$$

$$\dot{m}_{max.} = A\sqrt{0.4454\,(P_1/v_1)}$$

$$\boxed{\dot{m}_{max.} = A\,(0.6673)\sqrt{P_1/v_1}}$$

1/07/2019 Efficiency of nozzle (or) Effect of friction in nozzle:

* When, the steam flows through the nozzle, the final velocity of steam 'C_2' is reduced due to the foll. reason,

(i.) Due to friction b/w nozzle surface and steam.

(ii.) Due to internal fluid friction of steam.

(iii.) Due to shock losses.

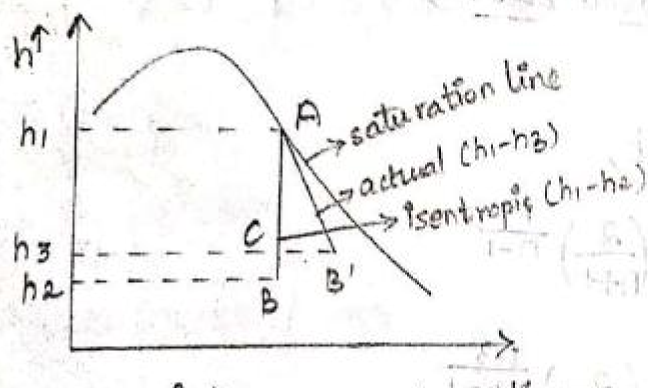
• Efficiency of nozzle:

* It is defined as, "The ratio b/w actual enthalpy drop to isentropic enthalpy drop".

$$\eta_n = \frac{\text{actual enthalpy drop}}{\text{Isentropic enthalpy drop}} \%$$

Enthalpy drop ↑ , Work output ↑ , pressure ↓, velocity ↑

Enthalpy drop ↓ , work output ↓, pressure ↑, velocity ↓



h-s diagram

from above h-s diagram,

$$\left[\eta_n = \frac{AB'}{AB} = \frac{h_1 - h_3}{h_1 - h_2}\right] \%$$

- **Critical pressure ratio:**

* There is only one ratio 'P_2/P_1', which produce max. discharge is called "critical pressure ratio".

* So, $P_1 \rightarrow$ initial pressure of steam at inlet.

  $P_2 \rightarrow$ final/throat pressure of steam at exit.

(i.) For saturated steam,

$n = 1.135$

we know that,

$$P_2/P_1 = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$ $\rightarrow$ refer max. mass flow rate (discharge) equation.

$$\frac{P_2}{P_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}}$$

$$\boxed{P_2/P_1 = 0.5774}$$ for saturation condition.

(ii.) For superheated steam,

$n = 1.3$

we know that,

$$P_2/P_1 = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$$P_2/P_1 = \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}}$$

$$\boxed{P_2/P_1 = 0.5457}$$ for superheated condition

(iii.) If $n = 1.4$ (adiabatic condition)

$$P_2/P_1 = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$$P_2/P_1 = \left(\frac{2}{1.4+1}\right)^{\frac{1.4}{1.4-1}}$$

$[P_2/P_1 = 0.5282]$ for adiabatic condition.

2/3/19

Saturated steam (or) Dry saturated steam:

* The whole steam without water suspension is called "Saturated (or) dry saturated steam".

Super heated steam:

* In saturated steam, if heat is added further then the temperature will increase, which is known as super heating and the steam obtained is called "Super heated steam".

Dryness fraction:

* It is the ratio of mass of dry steam to the total mass of steam.
* Denoted by "x".

$$x = \frac{m_g}{m_f + m_g}$$

∵ $m_g \to$ mass of dry steam
$m_f \to$ mass of wet steam

for dry steam, $m_f = 0$

So, $x = 1$ for dry steam.

Important formulas:

① Exit velocity of steam,

$$\left[C_2 = \sqrt{2000(h_1 - h_2)}\right] \, (m/s)$$

where, $h_1 \to$ Enthalpy at inlet (KJ/kg).

$h_2 \to$ Enthalpy at exit (KJ/kg)

② Mass flow rate,

$$m = \frac{A \times C_2}{V_2}$$

where, $A \to$ area of nozzle

$C_2 \to$ exit velocity

$V_2 \to$ specific volume

③ **Critical pressure ratio:**

$$P_2/P_1 = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

for saturated steam $(n = 1.135)$

$$\left(P_2/P_1 = 0.5774\right)$$

for superheated steam $(n = 1.3)$

$$\left(P_2/P_1 = 0.5457\right)$$

for $(n = 1.4)$ adiabatic steam

$$\left(P_2/P_1 = 0.5282\right)$$

④ **Enthalpy of steam:**

$$h_{wet} = h_f + x h_{fg}$$

$$h_{dry} = h_f + h_{fg} \qquad \text{as } (x = 1)$$

⑤ **Entropy of steam:**

$$S_{wet} = S_f + x\, S_{fg}$$

$$S_{dry} = S_f + (1) S_{fg} \qquad \text{as } (x = 1)$$

⑥ **Specific volume of steam:**

$$V_{wet} = x \times V_g$$

$$V_{dry} = V_g \qquad \text{as } (x = 1)$$

Problem:

1. Dry saturated steam at 3.5 bar is supplied to a convergent-divergent nozzle, whose throat area is 4.4 cm². The exit pressure is 1.1 bar. Determine the max. possible discharge through nozzle per minute, area of nozzle at exit when flow is maximum. Assume the flow has frictionless adiabatic.

Sol: given,

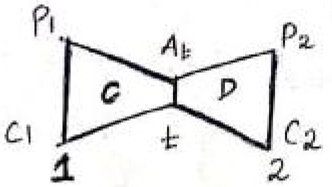
$P_1 = 3.5$ bar

$P_2 = 1.1$ bar

$A_t = 4.4\,cm^2$

$n = 1.135$

$x = 1$



find max. possible discharge at throat, becoz it is convergent-divergent nozzle.

Area at exit is A_2,

$$m_{max.} = \frac{A_t \times C_t}{V_t}$$

$$A_2 = \frac{m_{max.} \times V_2}{C_2}$$

for adiabatic process
$(S_1 = S_t = S_2)$

$$C_t = \sqrt{2000(h_1 - h_t)}$$

$$V_t = x_t \times V_{fg_t}$$

$$C_2 = \sqrt{2000(h_1 - h_2)}$$

$$V_2 = x_2 \times V_{fg_2}$$

from steam table,

P_1, 3.5 bar

$h_{f_1} = 584.3$ KJ/Kg

$h_{fg_1} = 2147.3$ KJ/Kg

$S_{f_1} = 1.727$ KJ/Kg K

$S_{fg_1} = 5.212$ KJ/Kg K

$V_{f_1} = 0.001079$ m³/kg

~~$V_{fg_1} = V_{f_1} + V_{g_1}$~~ $V_{g_1} = 0.52397$ m³/kg

P_2, 1.1 bar

$h_{f_2} = 428.8$ KJ/Kg

$h_{fg_2} = 2250.8$ KJ/Kg

$S_{f_2} = 1.333$ KJ/Kg K

$S_{fg_2} = 5.995$ KJ/Kg K

$V_{f_2} = 0.001046$ m³/kg

~~$V_{fg_2} = V_{f_2} + V_{g_2} =$~~

$V_{g_2} = 1.5492$ m³/kg

at inlet,

①

$h_1 = h_{f1} + h_{fg1}$

$h_1 = 584.3 + 2147.3$

$\boxed{h_1 = 2731.6}$ KJ/Kg

$S_1 = S_{f1} + S_{fg1}$

$S_1 = 1.727 + 5.212$

$\boxed{S_1 = 6.939}$ KJ/KgK

$\frac{P_t}{P_1} = 0.5774$

$P_t = 3.5 \times 0.5774$

$\boxed{P_t = 2.0209 \text{ bar}}$

P_t,

$h_{ft} = 504.7$ KJ/Kg

$h_{fgt} = 2201.6$ KJ/Kg

$S_{ft} = 1.530$ KJ/KgK

$S_{fgt} = 5.597$ KJ/Kg K

$V_{ft} = 0.001061$ m^3/kg

$V_{gt} = 0.88540$ m^3/kg

① - t,

$h_t = h_{ft} + x_t h_{fgt}$

$S_t = S_{ft} + x_t S_{fgt}$

as, $S_1 = S_2 = S_t$

$6.939 = 1.53 + x_t (5.597)$

$\boxed{x_t = 0.966} = 0.97$

$h_t = 504.7 + 0.97 (2201.6)$

$\boxed{h_t = 2640.252}$ KJ/Kg

$t - ②$

$h_2 = h_{f2} + x_2 h_{fg2}$

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

$6.939 = 1.333 + x_2(5.995)$

$(x_2 = 0.93)$

$h_2 = 428.8 + 0.93(2250.8)$

$[h_2 = 2522.044]$ KJ/Kg

$\dot{m}_{max} = \frac{A_t \times C_t}{V_t}$

$C_t = \sqrt{2000(h_1 - h_t)}$

$C_t = \sqrt{2000(2731.6 - 2640.252)}$

$C_t = \sqrt{182696}$

$[C_t = 427.42 \text{ m/s}]$

$V_t = x_t \times V_{gt}$

$V_t = 0.97 \times 0.8854$

$[V_t = 0.858838]$ m³/kg

$\dot{m}_{max} = \frac{4.4 \times 10^{-4} \times 427.42}{0.858838}$

$[\dot{m}_{max} = 0.2189]$ Kg/s $\rightarrow$ Kg/min $\Rightarrow \dot{m}_{max} = 3.645 \times 10^{-3}$ kg/min

$A_2 = \frac{\dot{m}_{max} \times v_2}{C_2}$

$A_2 \Rightarrow C_2 = \sqrt{2000(h_1 - h_2)} = \sqrt{2000(2731.6 - 2522.044)}$

$C_2 = \sqrt{419112}$

$[C_2 = 647.38 \text{ m/s}]$

$v_2 = x_2(V_{g2})$

$v_2 = 0.93 \times 1.5492$

$[V_2 = 1.440756]$ m³/kg

$A_2 = \frac{0.2189 \times 1.440756}{647.38}$

$[A_2 = 4.87 \times 10^{-4} \text{ m}^2]$

3/7/19

(2) Steam at 10·5 bar and 0·95 dryness is expanded through a convergent-divergent nozzle. The pressure of steam leaving the nozzle is 0·85 bar. Find the velocity of steam at throat for max. discharge and area at exit, also steam discharge if throat area is 1·2 cm^2. Assume the flow has isentropic and there are no friction losses. Take $n = 1·135$.

Sol: given,

$P_1 = 10·5$ bar

$x_1 = 0·95$

$P_2 = 0·85$ bar

$C_t = ?$

$A_2 = ?$

$A_t = 1·2\ cm^2$

$n = 1·135$

P_1,

$V_{f_1} = 0·001130\ m^3/kg$

$V_{g_1} = 0·18548\ m^3/kg$

$h_{f_1} = 772\ KJ/kg$

$h_{fg_1} = 2006\ KJ/kg$

$h_{g_1} = 2778\ KJ/kg$

$S_{f_1} = 2·159\ KJ/KgK$

$S_{fg_1} = 4·407\ KJ/KgK$

$S_{g_1} = 6·566\ KJ/KgK$

P_2,

$V_{f_2} = 0·001040\ m^3/kg$

$V_{g_2} = 1·9721\ m^3/kg$

$h_{f_2} = 398·6\ KJ/kg$

$h_{fg_2} = 2269·8\ KJ/Kg$

$h_{g_2} = 2668·4\ KJ/Kg$

$S_{f_2} = 1·252\ KJ/KgK$

$S_{fg_2} = 6·163\ KJ/KgK$

$S_{g_2} = 7·415\ KJ/KgK$

$h_1 = h_{f_1} + x_1 h_{fg_1}$

$S_1 = S_{f_1} + x_1 S_{fg_1}$

$h_1 = 772 + (0·95)(2006)$

$$\boxed{h_1 = 2677·7\ KJ/Kg}$$

$S_1 = 2·159 + (0·95)(4·407)$

$$\boxed{S_1 = 6·34565\ KJ/KgK}$$

as it is isentropic,

$(S_1 = S_2 = S_t)$

$h_2 = h_{f_2} + x_2 h_{fg_2}$

$S_2 = S_{f_2} + x_2 S_{fg_2}$

$6·34565 = 1·252 + x_2(6·163)$

$(x_2 = 0·82)$

$h_2 = 398·6 + (0·82)(2269·8)$

$$\boxed{h_2 = 2259·836}$$

$$P_t/P_1 = \left(\frac{2}{n+1}\right)^{n/n-1}$$

$$P_t = 10.5\left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}}$$

$$P_t = 10.5 \times 0.5774$$

$$[P_t = 6.06 \text{ bar}]$$

$v_{ft} = 0.001101 \text{ m}^3/\text{kg}$

$v_{gt} = 0.31546 \text{ m}^3/\text{kg}$

$h_{ft} = 670.4 \text{ kJ/kg}$

$h_{fgt} = 2085.1 \text{ kJ/kg}$

$h_{gt} = 2755.5 \text{ kJ/kg}$

$s_{ft} = 1.931 \text{ kJ/kgK}$

$s_{fgt} = 4.827 \text{ kJ/kgK}$

$s_{gt} = 6.758 \text{ kJ/kgK}$

$$h_t = h_{ft} + x_t h_{fgt}$$

$$s_t = s_{ft} + x_t s_{fgt}$$

$$6.34565 = 1.931 + x_t(4.827)$$

$$[x_t = 0.91]$$

$$h_t = 670.4 + (0.91)(2085.1)$$

$$[h_t = 2568.892 \text{ kJ/kg}]$$

Process ①-t,

$$V_t = x_t \times v_{gt}$$

$$V_t = 0.92 \times 0.31546$$

$$[V_t = 0.29022 \text{ m}^3/\text{kg}]$$

Process t-②,

$$V_2 = x_2 \times v_{g2}$$

$$V_2 = 0.82 \times 1.9721$$

$$[V_2 = 1.617122 \text{ m}^3/\text{kg}]$$

$$C_t = \sqrt{2000(h_1-h_t)} = \sqrt{2000(2677.7-2567.841)}$$

$$C_t = \sqrt{219718}$$

$$\boxed{C_t = 468.74\ m/s}$$

$$C_t = \sqrt{2000(h_1-h_t)}$$

$$C_t = \sqrt{2000(2677.7-2588.692)}$$

$$C_t = \sqrt{178016}$$

$$\boxed{C_t = 421.91\ m/s}$$

$$m_t = \frac{A_t \times C_t}{v_t}$$

$$m_t = \frac{1.2\times10^{-4}\times 421.91}{0.29022}$$

$$\boxed{m_t = 0.1744\ kg/s}$$

$$C_2 = \sqrt{2000(h_1-h_2)} = \sqrt{2000(2677.7-2259.836)}$$

$$C_2 = \sqrt{835728}$$

$$\boxed{C_2 = 914.18\ m/s}$$

According to mass balance,

$$m_t = m_2$$

$$m_2 = \frac{A_2 \times C_2}{V_2} \Rightarrow A_2 = \frac{m_2 \times V_2}{C_2} = \frac{0.1744\times1.617122}{914.18}$$

$$\boxed{A_2 = 3.085\times10^{-4}\ m^2}$$

③ Dry saturated steam at 2.8bar is expanded through a convergent nozzle to 1.7bar. The exit area is 3cm². Calculate exit velocity and mass flow rate for,

(i.) Isentropic expansion.

(ii.) Supersaturated flow.

Sol: given,

$P_1 = 2.8\,bar$

$P_2 = 1.7\,bar$

$A_2 = 3\,cm^2$

$C_2 = ?\quad m = ?$

$x_1 = 1$

(i.) $C_2 = ?$ Isentropic

$m = ?$ expansion

P_1,

$V_{f1} = 0.001071\ m^3/kg$

$V_{g1} = 0.64600\ m^3/kg$

$h_{f1} = 551.4\ KJ/kg$

$h_{fg1} = 2170.1\ KJ/kg$

$h_{g1} = 2721.5\ KJ/kg$

$s_{f1} = 1.647\ KJ/kgK$

$s_{fg1} = 5.367\ KJ/kgK$

$s_{g1} = 7.014\ KJ/kgK$

P_2,

$V_{f2} = 0.001056\ m^3/kg$

$V_{g2} = 1.0309\ m^3/kg$

$h_{f2} = 483.2\ KJ/kg$

$h_{fg2} = 2215.8\ KJ/kg$

$h_{g2} = 2699.0\ KJ/kg$

$s_{f2} = 1.475\ KJ/kgK$

$s_{fg2} = 5.706\ KJ/kg\ K$

$s_{g2} = 7.181\ KJ/kg\ K$

$h_1 = h_{f1} + h_{fg1}$

$h_1 = 551.4 + 2170.1$

$\boxed{h_1 = 2721.5\ KJ/kg}$

$S_1 = S_{f1} + S_{fg1}$

$S_1 = 1.647 + 5.367$

$\boxed{S_1 = 7.014\ KJ/Kg\ K}$

$V_1 = V_{g1}$

$V_1 = 0.64600\ m^3/kg$

$C_2 = \sqrt{2000(h_1 - h_2)}$

$h_2 = h_{f2} + x_2 h_{fg2}$

$S_2 = S_{f2} + x_2 S_{fg2}$

$V_2 = x_2 \times V_{g2}$

$V_2 = 0.97 \times 1.0309$

$\boxed{V_2 = 0.99\ m^3/kg}$

For isentropic process,

$(S_1 = S_2)$

$7.014 = 1.475 + x_2(5.706)$

$\boxed{x_2 = 0.97}$

$h_2 = 483.2 + (0.97)(2215.8)$

$\boxed{h_2 = 2632.526\ KJ/kg}$

$C_2 = \sqrt{2000(2721.5 - 2632.526)}$

$C_2 = \sqrt{177948}$

$\boxed{C_2 = 421.83\ m/s}$ → ans

$m = \frac{A_2 \times C_2}{V_2} = \frac{3 \times 10^{-4} \times 421.83}{0.99}$

$\boxed{m = 0.1278\ kg/s}$ → ans

(ii) Super saturated flow,

$n = 1.3$

here, $C_2 = \sqrt{\frac{2n}{n-1} \times P_1 V_1 \times \left(1 - (P_2/P_1)^{\frac{n-1}{n}}\right)}$

2.8 bar = 2.8×10^5 N/m²

$$C_2 = \sqrt{\frac{2(1.3)}{1.3-1} \times \frac{2.8}{(100000)^{-1}} \times 0.64600 \times \left(1 - \left(\frac{1.7}{2.8}\right)^{\frac{1.3-1}{1.3}}\right)}$$

$$C_2 = \sqrt{\frac{2.6}{0.3} \times 2.8\times10^5 \times 0.64600 \times (1-0.89)}$$

$$C_2 = \sqrt{172438.9333}$$

$$\boxed{C_2 = 415.25 \text{ m/s}} \rightarrow \text{ans}$$

$$m = \frac{A_2 \times C_2}{V_2} = \frac{3\times10^{-4} \times 415.25}{V_2}$$

$$m = \frac{0.124575}{V_2}$$

V_2,

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{1/n}$$

$$V_2 = 0.64600 \times \left(\frac{2.8}{1.7}\right)^{1/1.3}$$

$$\boxed{V_2 = 0.948 \text{ m}^3/\text{kg}}$$

$$m = \frac{0.124575}{0.94826}$$

$$\boxed{m = 0.1313 \text{ kg/s}} \rightarrow \text{ans}$$

5/7/19

Problems using Mollier diagram:

① Steam approaches a nozzle with velocity of 200 m/s, pressure of 4 bar and dryness fraction of 0.98. If the isentropic expansion in nozzle proceeds till the pressure of exit is 1 bar, determine the change in enthalpy and dryness fraction of steam using mollier diagram. Also calculate the exit velocity of steam from nozzle and area of exit of nozzle for flow of 0.8 kg/s.

Sol: given,

$C_1 = 200 m/s$

$P_1 = 4 bar$

$x_1 = 0.98$

$P_2 = 1 bar$

$m = 0.8 kg/s$

to find,

$C_2 = ?$

$x_2 = ?$

$a_2 = ?$

$h_1 - h_2 = \Delta h = ?$

* from mollier diagram, the corresp. pressure and dryness-fraction (x_1); $P_1 = 4 bar$; $x_1 = 0.98$; the enthalpy has taken, $[h_1 = 2690 kJ/kg]$.

* Since, the expansion is isentropic, from $h_1 = 2690 kJ/kg$ a vertical line is drawn in mollier diagram upto exit pressure '$P_2 = 1 bar$' line to obtain point '2'.

* Now, the foll. values are noted at point '2'.

$[h_2 = 2470 kJ/kg]$

$[x_2 = 0.905]$

$[V_2 = 1.5 m^3/kg]$

we get,

$\Delta h = h_1 - h_2 = 2690 - 2470$

$[\Delta h = 220 kJ/kg] \rightarrow ans$

$C_2 = \sqrt{2000(h_1 - h_2)} = \sqrt{2000(220)}$

$[C_2 = 663.32 m/s] \rightarrow ans$

a_2, $m = \frac{a_2 \times C_2}{V_2}$

$$a_2 = \frac{m \times V_2}{C_2} = \frac{0.8 \times 1.5}{663.32}$$

$$\left[a_2 = 1.8 \times 10^{-3}\, m^2\right] \rightarrow \text{ans}$$

② In a steam nozzle, the steam expands from 4bar to 1bar. The initial velocity is 60m/s and initial temp. is 200°C. Determine C_2, if nozzle efficiency is 92% and dryness fraction at exit.

Sol: given,

$P_1 = 4$ bar
$P_2 = 1$ bar
$C_1 = 60$ m/s
$T_1 = 200°C$

to get,

$C_2 = ?$
$x_2 = ?$

here,
$C_2 = \sqrt{2000(h_1 - h_2) \times \text{eff}}$

from mollier diagram,
by using $P_1 = 4$bar and $T_1 = 200°C$ we get

$h_1 = 2880$ KJ/Kg
$h_2 = 2610$ KJ/Kg
$x_2 = 0.97$

$$C_2 = \sqrt{2000(h_1 - h_2) \times 0.92} = \sqrt{2000(2880 - 2610)\, 0.92}$$

$$\left[C_2 = 704.84 \text{ m/s}\right] \rightarrow \text{ans}$$

$$\left[x_2 = 0.97\right] \rightarrow \text{ans}$$

③ Dry saturated steam at a pressure of 11 bar enters a convergent-divergent nozzle and leaves at a pressure of 2bar. If the flow is adiabatic and frictionless, determine the exit velocity of steam and ratio of cross-section of exit and throat.

sol: given, to get,

$(x_1 = 1)$ $C_2 = ?$

$P_1 = 11$ bar $a_2/a_t = ?$

$P_2 = 2$ bar

$(s_1 = s_2 = s_t)$

as it is convergent - divergent nozzle,

$$m_t = m_2$$

$$\frac{a_t \times C_t}{v_t} = \frac{a_2 \times C_2}{v_2}$$

$$\left[\frac{a_2}{a_t} = \frac{v_2}{v_t} \times \frac{C_t}{C_2}\right]$$

from chart, [Saturation line based]

$h_1 = 2780$ KJ/kg

$h_2 = 2480$ KJ/kg

$h_t = 2680$ KJ/kg.

$V_t = 0.28$

$V_2 = 0.8$

we know,

$$P_t/P_1 = 0.5774$$

$$P_t = 11 \times 0.5774$$

$$[P_t = 6.35 \text{ bar}]$$

$$C_2 = \sqrt{2000(h_1 - h_2)} = \sqrt{2000(2780 - 2480)}$$

$$\left[C_2 = 774.59 \text{ m/s}\right] \rightarrow \text{ans}$$

$$C_t = \sqrt{2000(h_1 - h_t)}$$

$$C_t = \sqrt{2000(2780 - 2680)}$$

$$[C_t = 447.21 \text{ m/s}]$$

$$\frac{a_2}{a_t} = \frac{0.8 \times C_t}{0.28 \times 774.59}$$

$$a_2/a_t = \frac{357.768}{216.8852}$$

$$\left[a_2/a_t = 1.64\right] \rightarrow \text{ans}$$

9/7/19

① Steam enters a group of convergent-divergent nozzle at 21 bar and 270°C. The discharge pressure of nozzle is 0.07 bar. The expansion is equilibrium throughout and the loss of friction in convergent portion of nozzle is negligible. But the loss by friction in divergent section of nozzle is equivalent to 10% of enthalpy drop available in that section. Calculate the throat and exit area to discharge 14 kg/s of steam.

Sol: given,

$P_1 = 21 \text{ bar}$

$T_1 = 270°C$

$P_2 = 0.07 \text{ bar}$

$m = 14 \text{ kg/s}$

$\text{loss} = 10\% \rightarrow \eta = 90\%$

to find,

$a_t = ?$

$a_2 = ?$

by mollier diagram,

at inlet condition:

$P_1 = 21 \text{ bar}, \; T_1 = 270°C$

we get, $\boxed{h_1 = 2990 \text{ kJ/kg}}$

throat section,

$P_t/P_1 = 0.5457$

$P_t = 0.5457 \times 21$

$\boxed{P_t = 11.45 \text{ bar}}$

by comparing $T_1 = 270°C$ in mollier diagram, we assume steam to be superheated steam.

$(n = 1.3)$

So, for $P_t = 11.45$ bar we get

$h_t = 2820 \text{ kJ/kg}$

$v_t = 0.18 \text{ m}^3/\text{kg}$

at exit,

$P_2 = 0.07$ bar

from mollier diagram,

$h_2 = 2060$ KJ/kg

$V_2 = 18\ m^3/kg$

So,
$$m_t = \frac{A_t \times C_t}{V_t}$$

$$A_t = \frac{m_t \times V_t}{C_t}$$

$$C_t = \sqrt{2000(h_1 - h_t)} = \sqrt{2000(2990 - 2820)}$$

$$(C_t = 583.9\ m/s)$$

$$A_t = \frac{14 \times 0.18}{583.9}$$

$$[A_t = 43.80\ cm^2] \rightarrow ans.$$

$$m_2 = \frac{a_2 \times C_2}{V_2}$$

$$a_2 = \frac{m_2 \times V_2}{C_2}$$

$$C_2 = \sqrt{2000(h_1 - h_t) + \eta(h_1 - h_2)}$$

$$C_2 = \sqrt{2000(2990 - 2820) + 0.9(2990 - 2060)}$$

$$[C_2 = 1306.90\ m/s]$$

$$A_2 = \frac{14 \times 0.18}{1306.90}$$

$$[A_2 = 1928\ cm^2] \rightarrow ans$$

12/7/19

Unit-2 BOILER:

Introduction:

* Boiler is a closed vessel in which the steam is generated From water by applying heat.
* The heated or vaporized fluid exit the boiler for using various process and heating applications.
* A "boiler" or "steam generator" is used where a source of steam is needed.
* Boiler are mainly used in steam engines such as locomotive, marine engines.
* Industry installation and power station (power plants), steam powered road vehicles, large hotels (cooking), bio gas plants, sugar mills.

List of boilers: (mostly fire tube type)

① Cochran boiler. ⊗
② Lancashire boiler. ⊗
③ Locomotive boiler.
④ Cornish boiler.
⑤ Wabcock boiler (or) wilcox boiler. ⊗

High pressure boilers: (mostly water tube type)

① Lamont boiler. ⊗
② Benson boiler. → seminar
③ Loeffler boiler.
④ Volex boiler.

Super critical boiler:

① Drum type boiler.
② Once-through boiler.

Fluidized Bed boilers:

① Bubbling fluidized bed boiler.

② Circulating fluidized bed boiler.

Classification of boiler:

* Steam boiler are classified on basis of boiler pressure, fuel (solid/liquid), boiler material, boiler tube type (fire/water), circulation, method of combustion (inside/outside of boiler), type of furnace construction, use (application), mobility (ship), ASME (American society of mech. Engi.) codes and heat source.

(i) According to flow of water and hot gases:

a) Fire tube boiler. (inside pipe gas is passed) Eg: Cochran boiler

b) Water tube boiler. (" " water " ") Eg: Lancashire boiler

(ii) According to axis of shell:

a) Vertical boiler. Eg: High pressure boilers.

b) Horizontal boiler. Eg: 1-4 low & medium pressure boiler.

c) Inclined boiler. Eg: Wabcock boiler

(iii) According to location/position of furnace:

a) Internally fired boiler.

b) Internally plus externally fired boiler.

(iv) According to the method of water circulation:

a) Natural circulation boiler. Eg: Remaining high press. boiler.

b) Forced circulation boiler.

(v) According to application/mobility:

a) Stationery boiler.

b) Mobile boiler.

(vi) According to steam pressure:

a) Low pressure boilers. (3.5 - 10 bar)

b) Medium pressure boilers. (10 - 25 bar)

c) High pressure boilers. (25 - 250 bar), 500°C

d) Supercritical boilers. (125 - 300 bar), 660°C

(vii) According to no. of tubes used:

a) Single tube boiler.

b) Multi tube boiler.

(viii) According to draft used:

a) Natural draft boiler.

b) Artificial draft boiler.

(ix) According to types of fuels:

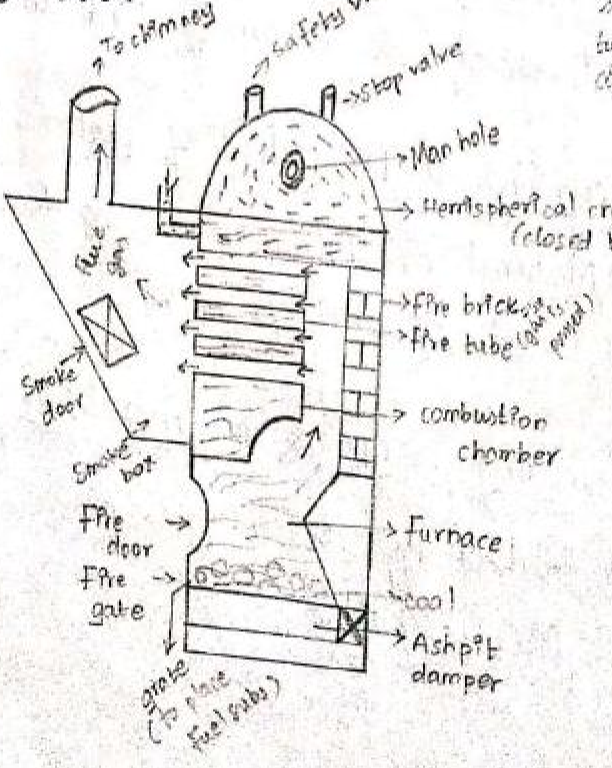
a) Solid fuel boiler.

b) Liquid / Gas fuel boiler.

c) Electrical or Nuclear power boiler.

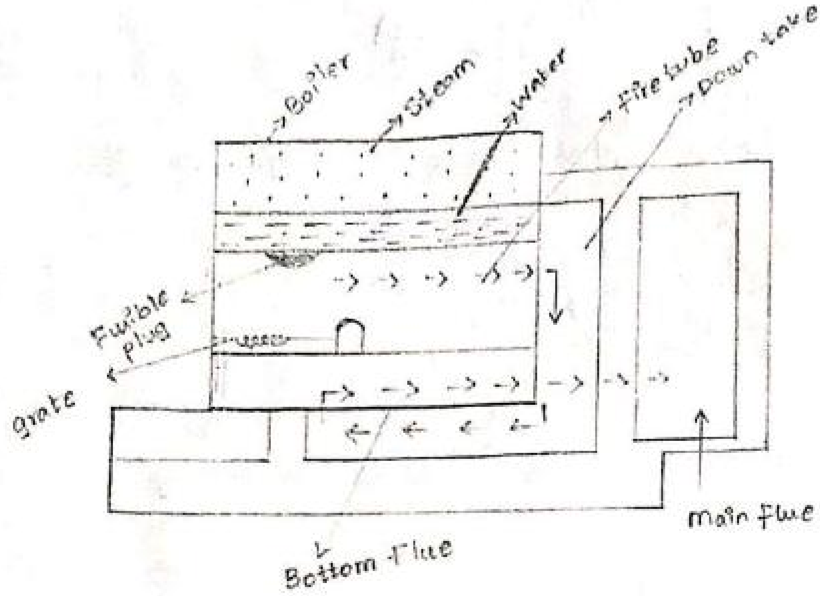
Cochran boiler:

[vertical axis, solid fuel, fire tube boiler, Natural circulation, lowpres boiler.]

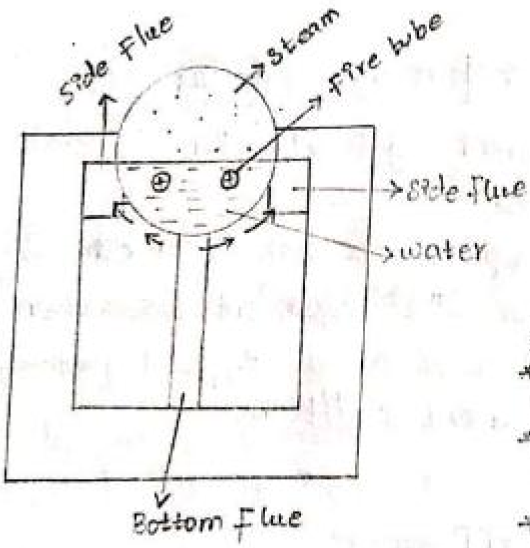


16/7/19

Lancashire boilers:

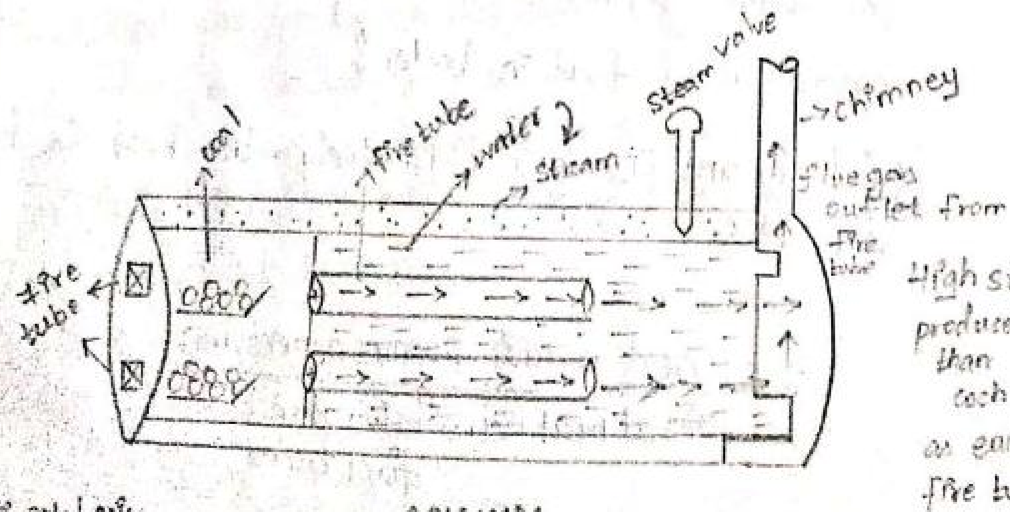


SIDE VIEW



FRONT VIEW

Adv:
* More steam can be produced.
* More efficient than cochran boiler
* More space to burn more fuel, more flue gas circulation.

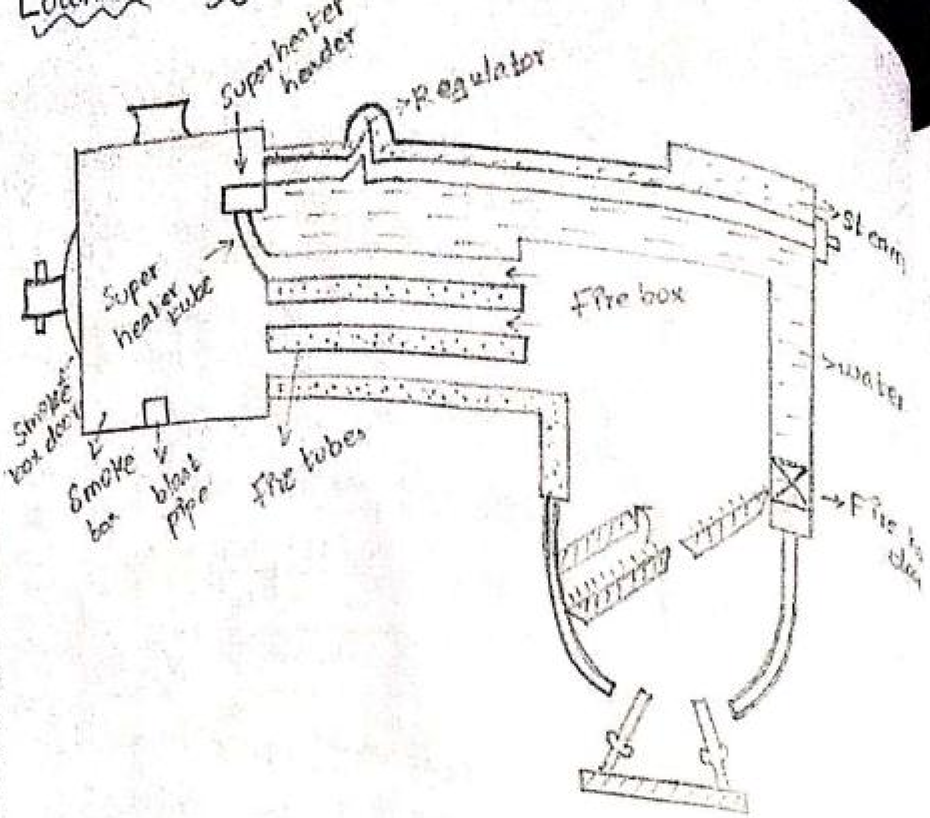


High steam produced than cochran as each fire tube has separate coal charge

[horizontal axis, S.F, F.t.b, N.C, L.P.B]

ORIGINAL BOILER CONSTRUCTION

Locomotive boiler:



16/7/19 Boiler performance calculations:

* Different types of boilers generate steam at different conditions.

* for comparing one boiler with other is very difficult. So, a parameter called "Equivalent evaporation" is used for comparing capacity of boilers at different pressure, temperature, different final steam conditions.

Boiler efficiency:

* Boiler efficiency is "the ratio of heat actually used for steam generation and total heat available due to combustion of fuel in boiler".

* It quantifies how effectively the heat is being used in boilers.

$$\eta_b = \frac{\text{heat used in steam generation}}{\text{Total heat available due to fuel burns}}$$

$$\left[\eta_b = \frac{m_w(h-h_w)}{m_f \times CV} = \frac{m_a(h-h_w)}{C.V}\right]\% \quad \because \left[m_a = \frac{m_w}{m_f}\right]$$

where,

m_f = mass of fuel/hour (kg/hr)

m_w = mass of feed water/hour (kg/hr)

m_a = actual evaporation in kg/kg of fuel

$h-h_w$ = enthalpy level at final steam and feed water in KJ/kg

Actual evaporation:

* It is "the ratio of mass of feed water, to the mass of fuel used in boiler".
* It is denoted by "m_a".

$$\therefore m_a = \frac{\text{mass of feed water}}{\text{mass of fuel}} = \frac{m_w}{m_F}$$

Boiler tests and trials:

(i) To estimate the amount of steam generating at full load conditions.

(ii) To calculate the thermal efficiency of boiler.

Heat losses:

* The following heat losses reduce efficiency in boiler,

(a) Heat loss due to Flue gas.

(b) Heat loss due to unburned coal.

(c) Heat loss due to incomplete combustion.

(d) Heat loss due to moisture present in fuel.

(e) Heat loss due to excess air in combustion chamber.

(f) Unaccounted heat losses.

19/7/19

- **Loss due to flue gas: (Q_g)**

$$\left[Q_g = m_g \cdot C_{pg}\,(T_g - T_a)\right] \text{ KJ}$$

where $m_g \Rightarrow$ mass of gases in combustion chamber.

$C_{pg} \rightarrow$ Specific heat of flue gas.

$T_g \rightarrow$ Temp. of flue gas.

$T_a \rightarrow$ ambient air temp.

- **Loss due to unburnt coal: (Q_u)**

$$\left[Q_u = m_u \times CV_u\right] \text{ KJ}$$

m_u = mass of unburnt coal.

CV_u = calorific value.

- **Loss due to moisture present in fuel: (Q_m)**

$$\left[Q_m = m_m \times (h_{sm} - h_a)\right] \text{ KJ}$$

m_m = mass of moisture in the fuel.

h_{sm} = Enthalpy of superheated steam at corresponding atm. press. and T_g.

h_a = Enthalpy of feed water at T_a (ambient).

- **Loss due to incomplete combustion: (Q_c)**

(i) 1kg of 'C' gives 33822 KJ of heat when it completely burnt,

$$C + O_2 \Rightarrow CO_2 \quad \text{(complete combustion)}$$

(ii) If amount of air is insufficient,

1kg of 'C' gives 10130 KJ of heat due to incomplete combustion,

$$C + O \Rightarrow CO \quad \text{[incomplete combustion]}$$

$\therefore$ heat loss due to incomplete combustion $\Big\} = 33822 - 10130$ KJ

22/7/19

- Loss due to excess air in combustion chamber: (Q_{air})

$$\left[Q_{air} = m_{air} \times C_{p\,air} (T_g - T_a)\right] \text{ KJ}$$

where,
$m_{air} \rightarrow$ mass of excess air.
$C_{p\,air} \rightarrow$ Specific capacity of air.
$T_g \rightarrow$ flue gas temp.
$T_a \rightarrow$ ambient temp.

- Unaccounted losses: ($Q_{acc.}$)

$$\left[Q_{acc.} = Q_s - \left(Q_b + Q_g + Q_u + Q_m + Q_c + Q_{air}\right)\right] \text{ KJ}$$

where,

Q_b = heat utilized to generate steam = $m_a (h - h_w)$

$Q_g = m_g \times C_{pg} (T_g - T_a)$

$Q_u = m_u \times CV_u$ $\qquad CV_u \rightarrow$ calorific value.

$Q_m = m_m \times (h_{sm} - h_a)$

$Q_c = 33822 - 10130$

$Q_{air} = m_{air} \times C_{p\,air} (T_g - T_a)$

17/9 Heat balance sheet: (1 Kg of fuel)

Heat supplied:	KJ:	%:	Heat utilized:	KJ:	%:
heat supplied by the fuel $= m_f \times CV$	Q_s	100	(a) By (To) steam generation	Q_b	ans
			(b) Loss due to flue gas	Q_g	ans
			(c) Loss due to unburnt coal	Q_u	ans
			(d) Loss due to moisture in fuel	Q_m	ans
always $m_f = 1$ kg to coal.			(e) Loss due to incomplete combustion	Q_c	ans
			(f) Loss due to excess air in combustion chamber	Q_{air}	ans
			(g) Loss due to unaccounted losses	$Q_{acc.}$	ans

$Q_b = m_a (h - h_w)$ KJ

$\% Q_b = \frac{Q_b}{Q_s} \times 100$

$Q_g = m_g \times C_{pg} (T_g - T_a)$ KJ

$\% Q_g = \frac{Q_g}{Q_s} \times 100$

$Q_u = m_u \times CV$ KJ

$Q_m = m_m \times (h_{sm} - h_a)$ KJ

$Q_c = 33822 - 10130$ KJ

$Q_{air} = m_{air} \times C_{p\,air} (T_g - T_a)$ KJ

$Q_{acc} = Q_s - [Q_b + Q_g + Q_u + Q_m + Q_c + Q_{air}]$

Boiler inspection:

* Inspection of boilers comprises such activities which are necessary for getting the correct knowledge regarding physical condition of all parts of boiler accesories.

* The inspection of furnaces, chimney and interior parts of boiler are related to inspection.

* The following areas should be checked carefully during inspection of boiler,

  (i) Checking the tube for corrosion and pittings.
  (ii.) Checking boiler heads, shells (walls) and welded portions.

  (iii.) Checking the feed pipe, whether it is clean.
  (iv.) Checking all safety walls, pressure gauges, Feed water valves and other Flow valves.
  (v) Checking the scale built up, internally blow tubes.

* The various inspection processes are carried out during boiler inspection,

  (a) Inspection of registration.
  (b) Hammer test.
  (c) Hydraulic test.
  (d) Steam test.
  (e) Inspection under emergency condition of steam.
  (f) Annual inspection.
  (g) Internal inspection. (higher offical)
  (h) Casual inspection.
  (i) Accident inspection.

/19 Important formulas:

① Actual evaporation,

$$\left(m_a = m_w / m_f \right) \text{ Kg/kg of fuel}$$

$m_w \rightarrow$ feed water mass
$m_f \rightarrow$ fuel mass

$h \rightarrow$ final steam enthalpy
$h_w \rightarrow$ enthalpy of feed water

② Equivalent evaporation,

$$\left(m_{equi.} = \frac{m_a (h - h_w)}{2256.9} \right)$$

2256.9 → heat supplied to generate steam at 100°C from (const. value) feed water

③ Efficiency of boiler,

$$\left(\eta_b = \frac{m_a (h - h_w)}{C.V} \right) \%$$

④ Heat supplied,

$$(Q_s = m_f \times CV)$$

⑤ Heat utilized to generate steam,

$$(Q_b = m_a (h - h_w))$$

Heat losses:

① $Q_g = m_g \times C_{pg} (T_g - T_a)$

② $Q_u = m_u \times CV$

③ $Q_m = m_m \times (h_{sm} - h_a)$

④ $Q_{air} = m_{air} \times C_{p\,air} (T_g - T_a)$

⑤ $Q_{acc.} = Q_s - [Q_b + Q_g + Q_u + Q_m + Q_c + Q_{air}]$

Problem:

① A boiler working at a pressure of 14bar evaporates 8.6kg of water per kg of coal fired from feed water entering at 39°C. The steam at the boiler stop valve is 0.92 dry. Determine the equi. evaporation from and at 100°C. Also determine the thermal efficiency of boiler if the calorific value of coal is 30,200KJ/kg

Sol: given,

$P_{steam} = 14bar \quad x = 0.92$

$m_a = 8.6 kg/kg\ of\ fuel \quad T_w = 39°C = 312K \quad CV = 30,200 KJ/kg$

$\{ m_a = m_w / m_f \quad T_2 = 100°C = 373K \}$ no use

$$M_{eq.} = \frac{m_a(h - h_w)}{2256.9} \; ?$$

$P_{steam} \rightarrow$ steam pressure
$x \rightarrow$ dryness fraction
$T_w \rightarrow$ feed water temp.
$m_a \rightarrow$ actual evaporation
$CV \rightarrow$ calorific value of fuel (coal)

$$\eta_b = \frac{m_a(h - h_w)}{CV} \,\% \; ?$$

from steam table, (pressure)

For, $P = 14$ bar

we get,

$h_f = 830.1$ KJ/kg

$h_{fg} = 1957.7$ KJ/kg

$$h = h_f + x h_{fg}$$

$$h = 830.1 + (0.92 \times 1957.7)$$

$$\boxed{h = 2631.184 \text{ KJ/kg}}$$

From, $T_w = 39°C$ using temperature table at $39°C$

$h_{f_w} = 163.3$ KJ/kg $\rightarrow$ $h_f = h_w = 163.3$ KJ/kg

$\cancel{h_{fg_w} = 2409.3 \text{ KJ/kg}}$

$\cancel{h_w = 163.3 + (0.92)(2409.3)}$

$\cancel{\boxed{h_w = 2379.856 \text{ KJ/kg}}}$

So,

$$M_{eq.} = \frac{m_a(h - h_w)}{2256.9} = \frac{8.6(2631.184 - h_w)}{2256.9}$$

$$M_{eq.} = \frac{8.6(2631.184 - 163.3)}{2256.9}$$

$$\boxed{M_{eq} = 9.403 \text{ KJ/kg}} \text{ ans}$$

$$\eta_b = \frac{m_a(h - h_w)}{CV} = \frac{8.6(2631.184 - 163.3)}{30{,}200} \times 100$$

$$\boxed{\eta_b = 70.27\,\%} \text{ ans}$$

25/7/19

② The foll. observation were made in boiler trial, coal used = 250 kg; calorific value = 29,800 KJ/kg; water evaporated = 2000 kg; steam pressure = 11.5 bar; dryness fraction = 0.95; feed H_2O temp. = 34°C. calculate the equi. evaporation from and at 100°C per kg of coal and efficiency of boiler.

Sol:

$P_{steam} = 11.5$ bar

$CV = 29{,}800$ KJ/kg

$m_f = 250$ kg

$x = 0.95$

$m_w = 2000$ kg

$T_w = 34°C$

$m_{equi.} = ?$ at 100°C

$\eta_b = ?$

$\boxed{h = h_f + x\, h_{fg}}$

W

T_f Sup

h h_w

$T_w = h_f = h_w$

wkt,

$$m_a = m_w / m_f = 2000/250$$

$$(m_a = 8 \text{ kg/kg of fuel})$$

for $P_{steam} = 11.5$ bar, from steam table (pressure)

$h_f = 789.9$ KJ/kg

$h_{fg} = 1991.4$ KJ/kg

$$h = h_f + x\, h_{fg}$$

$$h = 789.9 + (0.95 \times 1991.4)$$

$$[h = 2681.73 \text{ KJ/kg}]$$

for $T_w = 34°C$, from steam table (temp),

$$[h_f = h_w = 142.4 \text{ KJ/Kg}]$$

So,

$$m_{equi} = \frac{m_a (h - h_w)}{2256.9} = \frac{8 (2681.73 - 142.4)}{2256.9}$$

$$[m_{equi.} = 9.001 \text{ KJ/Kg}] \text{ ans}$$

$$\eta_b = \frac{m_a(h-h_w)}{CV} \cancel{\left(\frac{8(2681.73-142.4)}{2256.9}\right)\left(29{,}800\right)}$$

$$\eta_b = \left(\frac{8(2681.73-142.4)}{29{,}800} \times 100\right)$$

$$\left[\eta_b = 68.16\%\right] \text{ans}$$

③ A coal fired boiler consumes 400 kg of coal per hr. The boiler evaporates 3,200 kg of water at 45°C into superheated steam at a pressure of 12 bar and 275°C. If calorific value of fuel is 32,760 KJ/kg of coal. Calculate,

(i) Equi. evaporation from and at 100°C.

(ii) Thermal efficiency of boiler.

Take spf. heat of super heated steam as 2.01 KJ/kg K.

Sol: given,

$m_f = 400\,kg$

$m_w = 3{,}200\,kg$

$T_w = 45°C$

$P_s = 12\,bar$

$T_s = 275°C$

$CV = 32{,}760\,KJ/kg$

$m_{equi} = ?$ at 100°c

$\eta_b = ?$

$C_p = 2.1\,kJ/kg$

$m_a = m_w/m_f = 3{,}200/400$

$[m_a = 8\,kg/kg \text{ of fuel}]$

for $T_w = 45°c$, from steam table (temp.),

$[h_f = h_w = 188.4\,KJ/kg]$

For superheated steam,

enthalpy, $h = h_{sup} = (h_g + C_p(T_{sup.} - T_{sat.}))$

for water,

$$T_w \Rightarrow h_w = (h_f)$$

for steam,

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

So,

for superheated steam,

$P = 12\,\text{bar}$, $T = 275\,°C = 548\,K$

from steam table (pressure) for $P_1 = 12$ bar,

$[h_g = 2782.7\ \text{KJ/kg}]$ $T_{sat} = 188\,°C$

$$h = h_{sup.} = (2782.7 + 2.1[\underset{(275°C)}{548} - \underset{(188°C)}{461}])$$

$$[h = 2965.4\ \text{KJ/kg}]$$

$$m_{equi.} = \frac{m_a(h - h_w)}{2256.9} = \frac{8(2965.4 - 188.4)}{2256.9}$$

$[M_{equi} = 9.84\ \text{KJ/kg}]$ ans

$$\eta_b = \frac{m_a(h - h_w)}{CV} \times 100$$

$$\eta_b = 8(2965.4 - 188.4)/32760 \times 100$$

$\left[\eta_b = 67.81\%\right]$ ans.

So, $\left[m_{equi.} = 2965.4 \text{ kJ/kg}\right]$

$\left[\eta_b = 67.81\%\right]$

hint;

for , $h_{sup.} = \left[h = h_g + C_P\left(T_{sup.} - T_{sat}\right)\right]$

h_g and $T_{sat.}$ are taken from steam table (pressu

31/7/19

Problems on Heat balance sheet:

① The foll. data were obtained in a boiler trial,

(i) Feed H_2O supplied per hour = 690 kg at 28°c.

(ii) Steam produced = 0.97 dry at 8 bar.

(iii) Coal fired per hour = 91 kg of C.V 27,255 KJ/kg

(iv) Ash and unburnt coal are collected at bottom of grate = 7.5 kg/hr of CV 3,700 KJ/kg

(v) Mass of flue gases per kg of coal burnt = 17.. kg (m_g)

(vi) Temp. of flue gas = 325°C. (T_g)

(vii) Room temp. = 17°C. (T_a)

(viii) Spf. heat of flue gas = 1.005 KJ/kg K

Estimate the boiler efficiency and draw heat balance sheet.

Sol: given,

$m_w = 690\,kg$

$T_w = 28°C$

$x = 0.97$

$P_{steam} = 8\,bar$

$m_f = 91\,kg$

$CV_f = 27,255\,kJ/kg$

$m_u = 7.5\,kg/hr$ ~~= 7.5/3600 kg/s~~

mass, $u = \frac{m_u}{m_f} = \frac{7.5}{91} = 0.08$

So, mass $u = 0.08\,kg/kg$

$CV_u = 3,700\,kJ/kg$

$m_g = 17.4\,kg$

$T_g = 325°C$

$T_a = 17°C$

$C_{pg} = 1.005\,kJ/kg\,k$

$\eta_b = ?$ Heat balance sheet cal. => ?

$$\eta_b = \frac{m_a(h - h_w)}{CV_f}$$

$m_a = m_w / m_f = 690/91$

$[m_a = 7.58 \text{ kg}]$

$h \Rightarrow$ From steam table (pressure), $P_{steam} = 8 \text{ bar}$

we get,

$h_F = 720.9 \text{ KJ/Kg}$

$h_{fg} = 2046.5 \text{ KJ/kg}$

$h = h_F + x\, h_{fg}$

$h = 720.9 + (0.97 \times 2046.5)$

$[h = 2706.005 \text{ KJ/Kg}]$

$h_w \Rightarrow$ from steam table (temp.), $T_w = 28°C$

we get,

$[h_F = h_w = 117.3 \text{ KJ/Kg}]$

$$\eta_b = \frac{7.58\,(2706.005 - 117.3)}{27255} \times 100\ \%$$

$[\eta_b = 71.99\%]$ ans.

Heat balance sheet,

Heat supplied	KJ	%	Heat utilized	KJ	%
$Q_s = m_f \times CV_f$	27,255	100	(a) Q_b (x)	19622.3839	71.99%
			(b) Q_g ✓	5385.996	19.76%
			(c) Q_u ✓	296	1.08%
			(d) Q_m ×		
			(e) Q_c ×		
			(f) Q_{air} ×		
			(g) $Q_{acc.}$ ✓	1950.6201	7.1%
				27,255	99.93% ≃ 100%

no data given

here taken $m_f = 1$ kg, $CV = 27255$

$Q_b = m_a(h - h_w)$ KJ

$Q_b = 7.58(2706.005 - 117.3)$ KJ

$[Q_b = (19622.3839 \text{ KJ})]$

$$\% Q_b = \frac{Q_b}{Q_s} \times 100 = \frac{19622.3839}{27255} \times 100 \%$$

$$\left[\% Q_b = 71.99 \%\right]$$

$$Q_g = m_g \times C_{pg} \, (T_g - T_a) \quad KJ$$

$$Q_g = 17.4 \times 1.005 \left((325+273) - (17+273)\right) \, KJ$$

$$\left[Q_g = 5385.996 \, KJ\right]$$

$$\% Q_g = Q_g / Q_s \times 100 = \frac{5385.996}{27255} \times 100$$

$$\left[\% Q_g = 19.76 \%\right]$$

$$M_u = \frac{m_u \; 7.5}{m_f \; 91} \quad 1kg$$

$$m_u = 1$$

$$\cancel{Q_u = M_u \times CV_u \quad KJ}$$

$$\cancel{Q_u = \frac{7.5}{3600} \times 3700 \quad KJ}$$

$$\cancel{\left[Q_u = 7.70 \, KJ\right]}$$

$$\cancel{\% Q_u = Q_u / Q_s \times 100 = \frac{7.70}{27255} \times 100 \%}$$

$$\cancel{\left[\% Q_u = 2.8 \%\right]}$$

$$Q_u \Rightarrow \quad m_{req.\,u} = \frac{m_u}{m_f} = \frac{7.5}{91}$$

$$m_{req.\,u} = 0.08$$

$$\left[Q_u = 0.08 \times 3700 = 296 \, KJ\right]$$

$$\% Q_u = Q_u / Q_s \times 100 = \frac{296}{27255} \times 100$$

$$\left[\% Q_u = 1.08 \%\right]$$

$$Q_{acc.} = Q_s - \left[Q_b + Q_g + Q_u\right]$$

$$Q_{acc.} = 27255 - (19622.3839 + 5385.996 + 296)$$

$$\left[Q_{acc.} = 1950.6201 \, KJ\right]$$

$$\% Q_{acc.} = \frac{Q_{acc.}}{Q_s} \times 100 = \frac{1950.6201}{27255} \times 100$$

$$\left[\% \, Q_{acc} = 7 \cdot 15 \, \% \right]$$

6/8/19 Unit 3 STEAM TURBINES:

1869

* Steam turbine is a device, used to convert kinetic energy into mechanical energy. In turbine, the enthalpy of steam first converted into kinetic energy in nozzle section.

* The high velocity of steam flow on curved blades and its direction of flow is changed it causes a change of momentum, a force is developed in turbine shaft.

* In large size power plant (turbine) is used to drive electric generator. The small size power is used to drive fans, pumps, compressor, etc.

Classification of steam turbines:

① On basis of steam expansion – (i) Impulse turbine.
(ii) Reaction turbine.
(iii) Combination of impulse and turbines

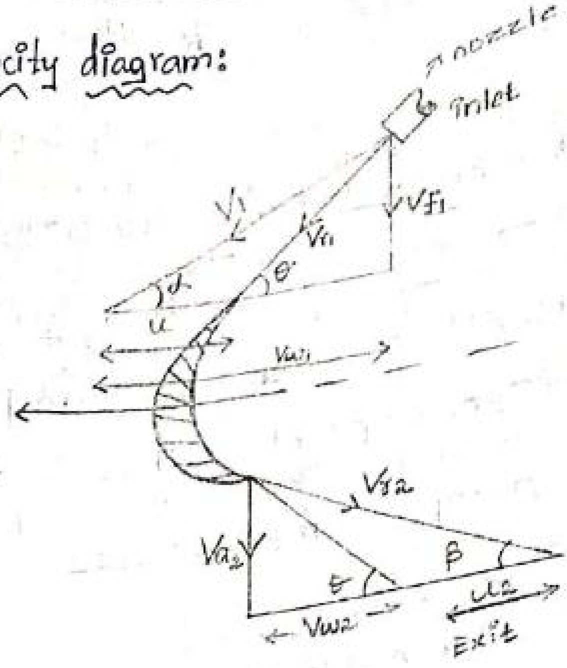
② On basis of no. of stage of expansion – (i) Single stage turbine.
(ii) Multi-stage turbine.

③ On basis of direction of flow – (i) Axial flow turbine
(ii) Radial flow turbine
(iii) Tangential flow turbine.
(iv) Mixed flow turbine

④ On basis of steam pressure – (i) Low pressure turbine.
(ii) Medium pressure turbine.

20/8/19

Velocity diagram:



let,

$V_1 \rightarrow$ absolute velocity of steam entering the moving blade

$V_{f1} \rightarrow$ velocity of flow at entrance of moving blade.

$V_{r1} \rightarrow$ relative velocity of jet at entrance of moving blade.

$u \rightarrow$ Linear velocity of moving blade.

$V_{w1} \rightarrow$ velocity of whirl at entrance of moving blade.

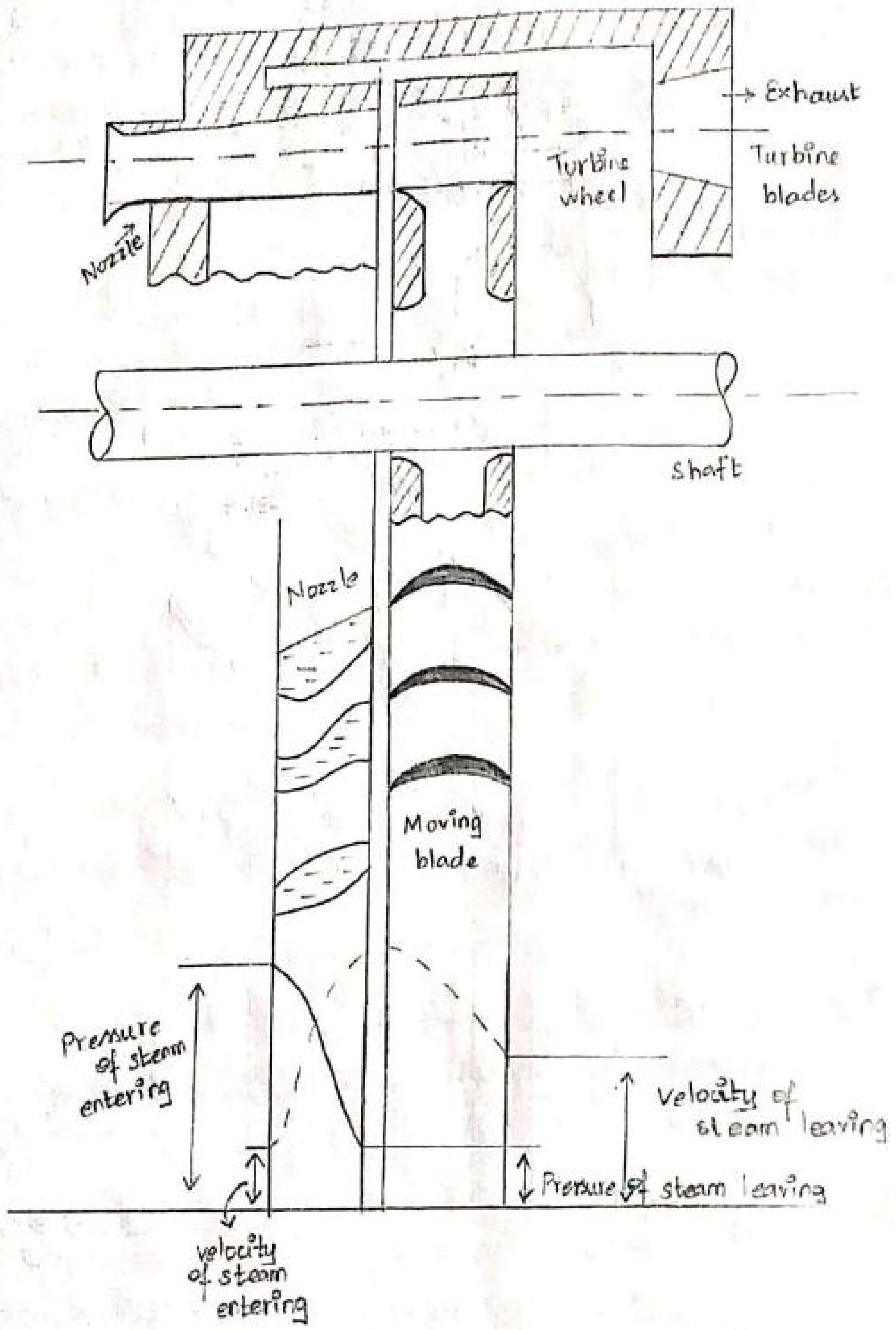
α - angle with tangent of wheel at which the steam with velocity V_1 enters. Also called as "nozzle angle".

θ - entrance angle of moving blade.

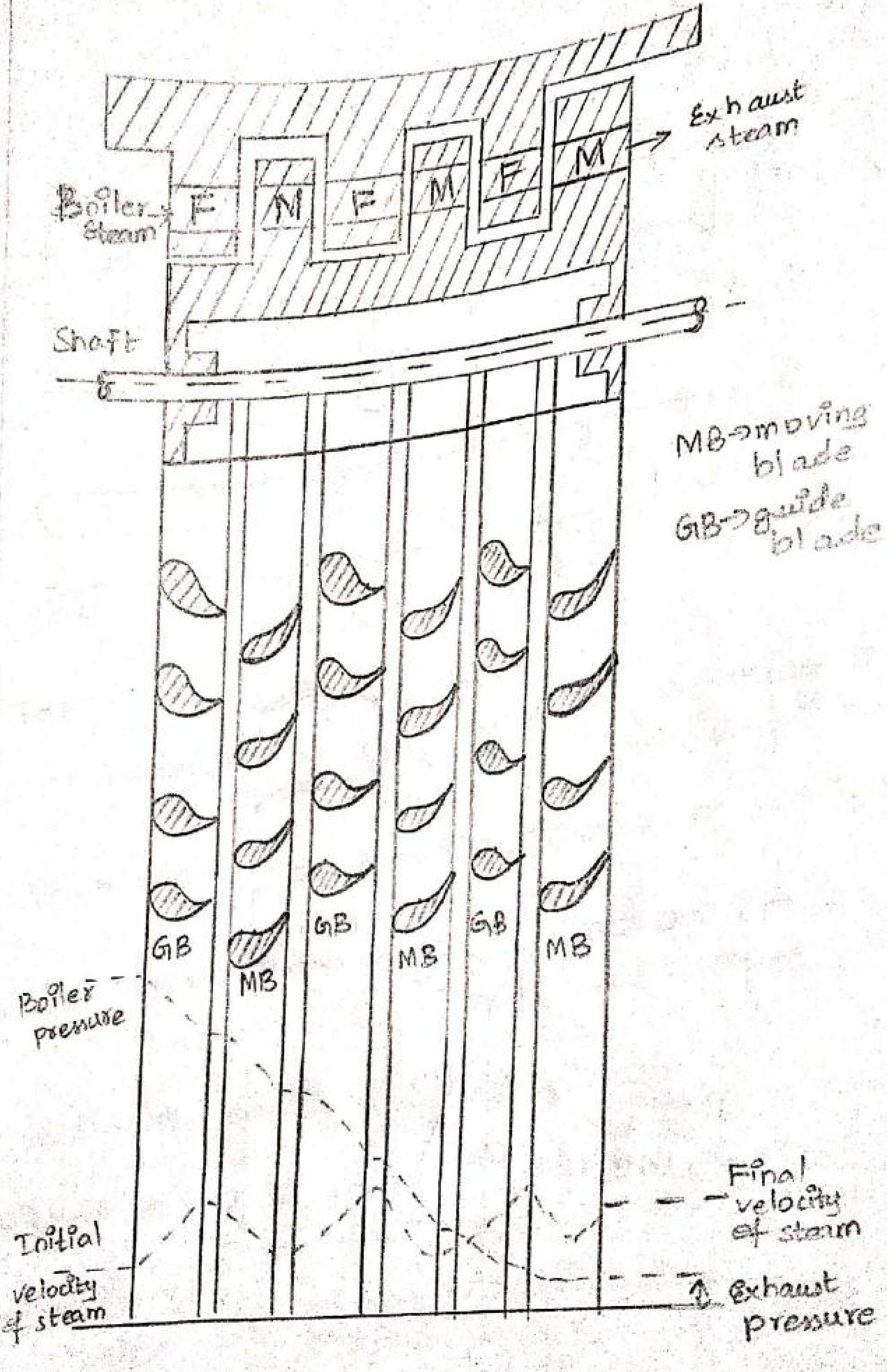
ϕ - exit angle of moving blade.

β - angle which the discharging steam ma with tangent of the wheel at exit of moving blade.

Simple impulse turbine:



Reaction turbine:



Velocity diagram:

* With its use, we determine the rate of momentum of steam across the moving blades in order to find the forces on blade.
* The power developed can be determined from knowledge of flow rate and blade speed. It is also equally important that the steam should enter and leave the blade without any stock.
* Hence, suitable inlet angle of moving blade should be evaluated.
* This is made possible with the help of velocity-diagram at inlet and outlet of moving blade.

Velocity diagram for impulse turbine:

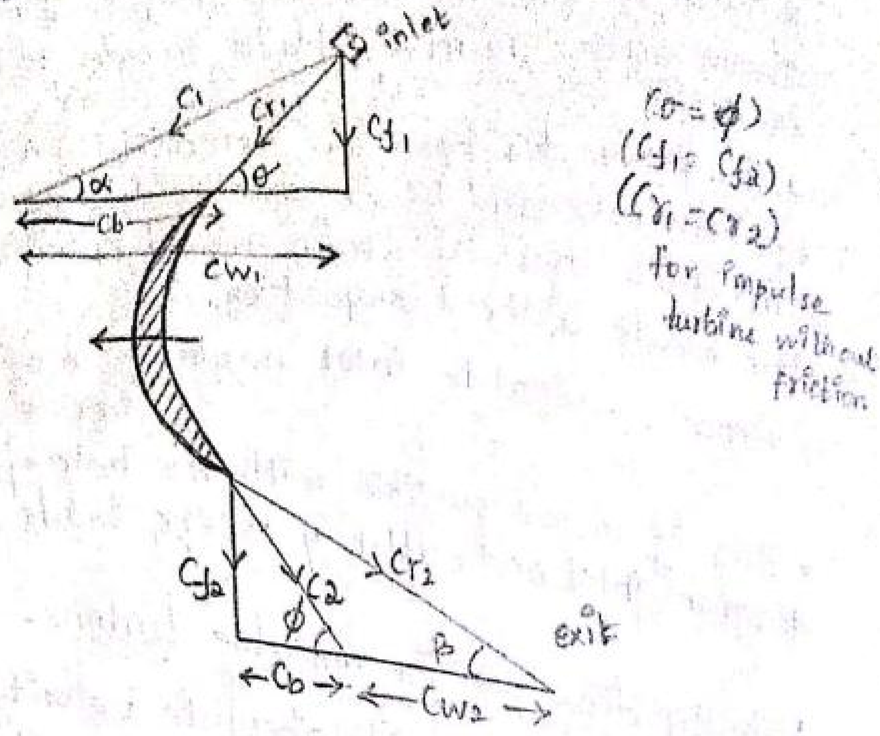
→ The steam of jet with absolute velocity (V_1) incident on moving blade at angle (α). The tangential of jet (V_{w1}) perform work on blade which is called 'velocity whirl (C_{f1})'.

→ The axial component of velocity (V_{f1}) of jet (V_1) doesn't produce work on blade but it causes steam to flow through turbine. This component is known as 'Velocity of Flow'.

→ As the blade moves with a tangential velocity (V_1), the entering stream of jet has a relative velocity (V_{r1}) which makes angle (θ).

→ The velocity of flow makes an axial thrust on a rotor, the steam then slide over blades without any shock and discharge at a relative velocity (V_{r2}) at an angle (ϕ) with tangent of blade's the absolute to tangent at the wheel.

Velocity diagram for impulse turbine.



$C_1 \rightarrow$ absolute velocity at entrance $\longrightarrow C_2 \rightarrow$ exit

$C_{f1} \rightarrow$ flow velocity at entrance $\longrightarrow C_{f2} \rightarrow$ exit

$C_{r1} \rightarrow$ relative velocity at entrance $\longrightarrow C_{r2} \rightarrow$ exit

$C_b \rightarrow$ Blade velocity

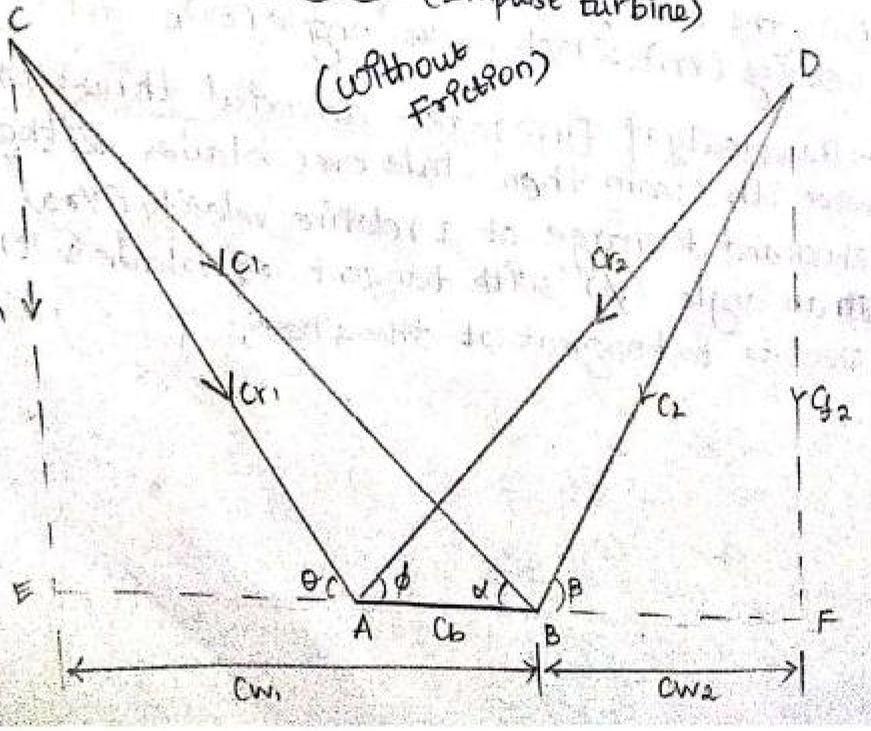
$C_{w1} \rightarrow$ whril velocity at entrance

$\alpha \rightarrow$ angle made between C_1 and C_{w_1} $\rightarrow$ nozzle angle/inlet steam angle (α)

$\theta \rightarrow$ Entrance angle for moving blade (θ) with C_b & C_{r1}

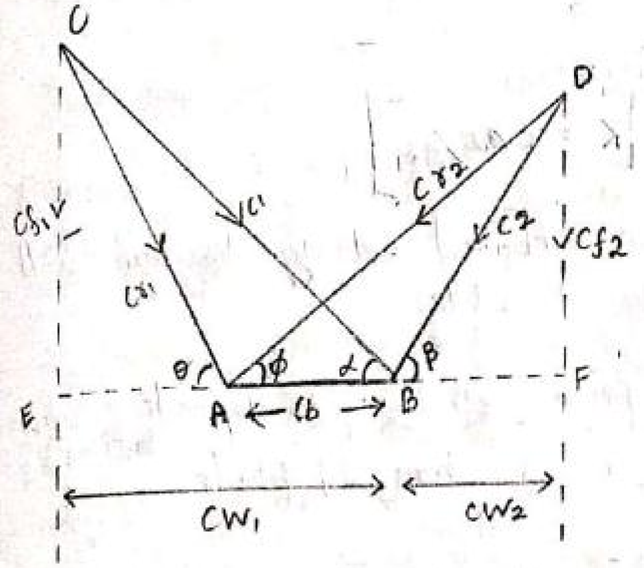
$\phi, \beta \rightarrow$ Exit angle for moving blade (ϕ) $\rightarrow$ Exit angle of steam (β)

Combined velocity diagram: (Impulse turbine)



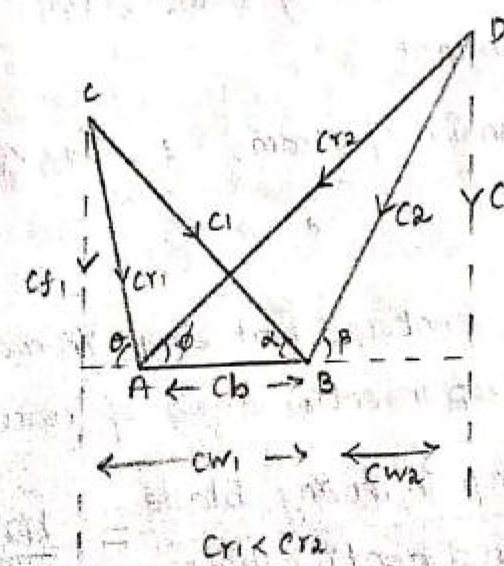
* For solving problems on turbines, it is a common practise to combine two velocity diagrams on a common basis representing blade velocity (C_b), when there is no friction.
* Hence, the outlet velocity triangle is turned through 180°.
* Without friction, the foll. conditions are made
  (i.) $C_{r1} = C_{r2}$
  (ii.) $C_{f1} = C_{f2}$
  (iii.) $\theta = \phi$

26/8/19 **Effect of friction on velocity diagrams: (Impulse turbine)**
(with friction)



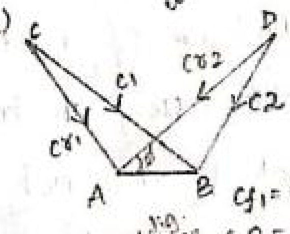
$C_{r1} > C_{r2}$ $(C_{r1} \neq C_{r2})$

Velocity diagram for Reaction turbine:



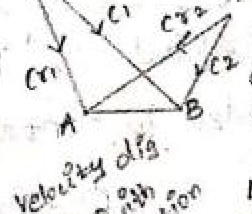
$C_{r1} < C_{r2}$
$\phi \neq \theta$

Velocity dig. Impulse turbine without friction

(a)


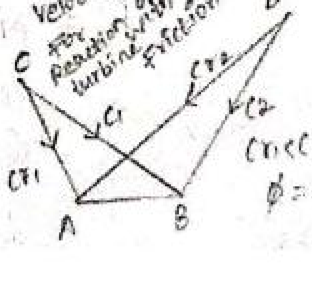
$C_{f1} = C_{f2}$
$\theta = \phi$
$C_{r1} = C_{r2}$

Velocity dig. for impulse turbine with friction

(b)


$C_{f1} \neq C_{f2}$
$C_{r1} > C_{r2}$

Velocity dig. for Reaction turbine with friction

(c)


$C_{r1} < C_{r2}$
$\phi \neq \theta$

30/8/19

Effect of friction on velocity diagram:

* As stated earlier, if friction is neglected "$C_{r1} = C_{r2}$", but in actual practise there is always some frictional resistance to flow of steam over plates.

* The effect of this friction is to reduce the relative velocity of steam as it passes over the plates.
* The loss in relative velocity is generally taken as 10 - 15 %

* The ratio of C_{r2} to C_{r1} is known as frictional factor or blade velocity coefficient.

So, Frictional factor,

$$\left[K = C_{r2}/C_{r1}\right]$$

* Effect of friction on combined velocity diagram will reduce relative velocity at outlet.

* The value 'k' varies from 0.5 to 0.75 depending upon shape of blade.

Velocity diagram for reaction turbine:

* In case of reaction turbines, since the steam expands continuously in both fixed and moving blades, the relative velocity does not remain constant.
* It increase due to the expansion of steam, i.e, $C_{r2} > C_{r1}$

Degree of reaction (R):

→ Defined as, "The ratio of isentropic heat drop in moving blades to isentropic heat drop in entire stage of reaction turbine."

$$R = \frac{\text{enthalpy drop in moving blade}}{\text{enthalpy drop in entire stage of reaction turbine}} = \frac{h_2 - h_3}{h_1 - h_3}$$

For impulse turbine → Blade speed, $\rho = \frac{C_b}{C_1} = 50\%$

For reaction turbine → Degree of reaction, $R = \frac{h_2 - h_3}{h_1 - h_3}$

Important formulas:

① Impulse turbines:

(I)(a) Driving (or) tangential force:

$$F_x = m\,(C_{w1} + C_{w2})$$

C_w → whirl velocity

1, 2 → inlet, exit

(b) Workdone: (blade)

$$W_b = C_b\,(C_{w1} + C_{w2})\,m$$

C_b → blade velocity

(c) Power developed:

$$P = m \times C_b\,(C_{w1} + C_{w2})$$

(d) Axial thrust:

$$F_y = m\,(C_{f1} - C_{f2})$$

(e) Efficiency of blade:

$$\eta_b = \frac{\text{workdone on blade}}{\text{energy supplied to blade}}$$

$$\eta_b = \frac{2\,C_b\,[C_{w1} + C_{w2}]}{C_1^2}$$

if enthalpy drop given then, stage

$$\eta_{b'} = \frac{\text{workdone on blade}}{\text{total energy supplied per stage}}$$

$$\eta_{b'} = \frac{C_b\,[C_{w1} + C_{w2}]}{h_1 - h_2}$$

(f) Stage efficiency:

$$\eta_{stage} = \eta_b \times \eta_{nozzle}$$

Effect of friction on velocity diagram:

(i) Friction factor,

$$k = C_{r2}/C_{r1}$$

Value of 'k' varies from (0.75 to 0.85)

(ii) Speed ratio,

$$\rho = C_b/C_1$$

$C_1 \to$ absolute velocity at inlet

4/9/19 Problems on impulse turbine:

① In a D-laval turbine (single stage impulse turbine) steam enters the wheel through a nozzle with velocity of 350 m/s and at an angle of 20° to direction of motion of blade. The blade speed of 250 m/s and exit angle of moving blade is 35°. Find the ~~inlet~~ exit angle of moving blade, exit velocity and workdone per kg of steam.

Sol: given,

$C_1 = 350$ m/s [absolute inlet nozzle velocity]

$\alpha = 20°$ (steam inlet angle)

$C_b = 250$ m/s [absolute blade speed]

$\phi = 35°$ [exit angle of blade]

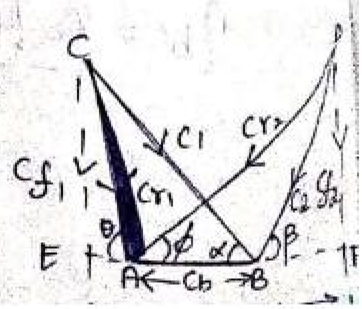
($C_{r1} = C_{r2}$ as no friction given)

to find:

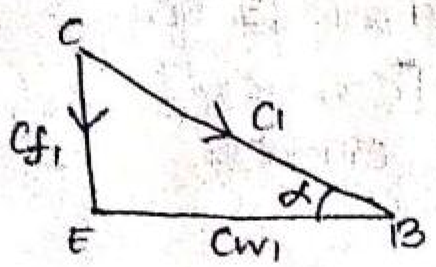
(i) Exit angle of moving blade (θ)

(ii) Exit velocity (C_2)

(iii) Workdone (W_D)



take ΔBCE,

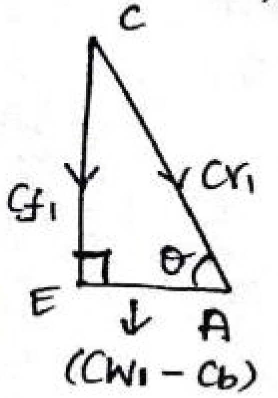


$\sin\alpha = Cf_1 / C_1$ $\qquad$ $\cos\alpha = \frac{Cw_1}{C_1}$

$\sin 20^\circ = Cf_1 / 350$ $\qquad$ $\cos 20^\circ = \frac{Cw_1}{350}$

$[Cf_1 = 119.70\ m/s]$ // $\qquad$ $[Cw_1 = 328.89\ m/s]$ //

take ΔACE,



$\tan\theta = Cf_1 / (Cw_1 - C_b)$

$\tan\theta = \frac{119.7}{328.89 - C_b} \Rightarrow \tan\theta = \frac{119.7}{328.89 - 250}$

$\tan\theta = 1.517$

$[\theta = 56.61^\circ]$ ans //.

$Cr_1^2 = Cf_1^2 + (Cw_1 - C_b)^2$

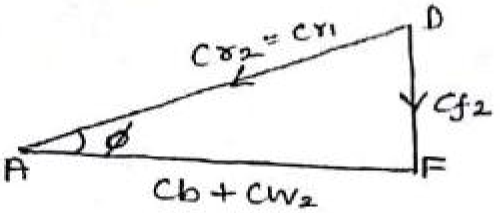
$Cr_1^2 = 119.7^2 + (328.89 - 250)^2$

$[Cr_1 = 143.35\ m/s]$ //

take ΔADF,

wkt, for impulse turbine without friction,

$$[C_{r1} = C_{r2}]$$



$$\sin\phi = C_{f2}/C_{r2}$$

$$\sin 35^\circ = C_{f2}/143.35$$

$$[C_{f2} = 82.22\,m/s]$$

$$\tan\phi = C_{f2}/C_b + C_{w2}$$

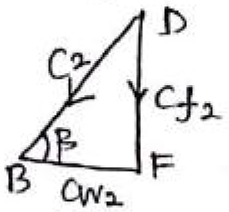
$$\tan 35^\circ = 82.22/250 + C_{w2}$$

$$250 + C_{w2} = 82.22/\tan 35^\circ$$

$$[C_{w2} = -132.57\ m/s]$$

take triangle

ΔBDF,



$$C_2^2 = C_{f2}^2 + C_{w2}^2$$

$$C_2 = 155.99\,m/s$$

$$\tan\beta = C_{f2}/C_{w2} = 82.22/-132.57$$

$$\tan\beta = -0.6202$$

$$[\beta = -31.80^\circ]$$

Work done,

$$W_b = C_b(C_{w1} + C_{w2}) = 250\,(328.89 - 132.57)$$

$$[W_b = 49{,}080\ J]$$

So, we get

$$[\theta = 56.61^\circ]\ [C_2 = 155.99\,m/s]$$

$$[W_b = 49{,}080\ J]$$

Q) The velocity of steam leaving the nozzle of an impulse turbine is 1000 m/s and nozzle angle is 20°. The blade velocity is 350 m/s and blade velocity of coeff. is 0.85. Assuming no losses due to shock at inlet, calculate for mass flow of 1.5 kg/s and symmetrical blading,

(a) Blade inlet angle.

(b) Driving force on wheel.

(c) Axial thrust on wheel.

(d) Power developed by turbine.

Sol. given, $K = 0.85 = C_{r2}/C_{r1}$

$C_1 = 1000 \text{ m/s}$

$\alpha = 20°$

$C_b = 350 \text{ m/s}$

$m = 1.5 \text{ kg/s}$

to find,

$\theta = ?$

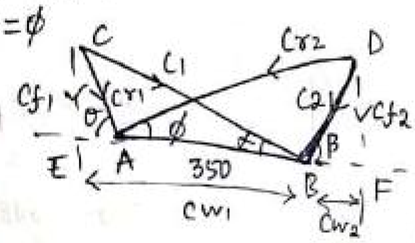
$F_x = ?$ $m(C_{w1} + C_{w2})$

$P = ?$ $m \times C_b (C_{w1} + C_{w2})$

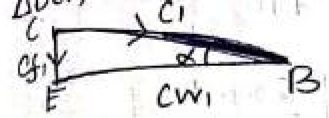
$F_y = ?$ $m(C_{f1} - C_{f2})$

for symmetrical Blade,

$\theta = \phi$



ΔBCE,



$\sin\alpha = C_{f1}/C_1$

$\sin 20° = C_{f1}/1000$

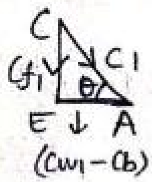
$[C_{f1} = 342.02 \text{ m/s}]$

$\cos\alpha = \dfrac{C_{w1}}{C_1}$

$\cos 20° = \dfrac{C_{w1}}{1000}$

$[C_{w1} = 939.69 \text{ m/s}]$

ΔACE,



$\tan\theta = \dfrac{C_{f1}}{C_{w1} - C_b}$

$\theta = \tan^{-1}\left[\dfrac{342.02}{939.69 - 350}\right]$

$[\theta = 30°]$

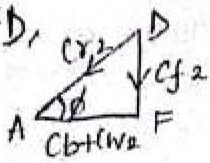
$C_{r1} = \sqrt{(C_{w1} - C_b)^2 + C_{f1}^2} = \sqrt{589.69^2 - 342.02^2}$

$[C_{r1} = 681.69 \text{ m/s}]$

wkt, $K = C_{r2}/C_{r1} \Rightarrow 0.85 = \frac{C_{r2}}{681.69}$

$$\boxed{C_{r2} = 579.43\, m/s}$$

Δ AFD,



$\sin\phi = C_{f2}/C_{r2}$

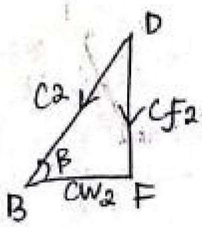
$\sin 30^\circ = C_{f2}/579.43$

$$\boxed{C_{f2} = 289.715\, m/s}$$

$\cos\phi = \frac{C_b + C_{w2}}{C_{r2}} \Rightarrow \cos 30^\circ = \frac{350 + C_{w2}}{579.43}$

$$\boxed{C_{w2} = 151.80\, m/s}$$

ΔDBF,



$C_2 = \sqrt{C_{f2}^2 + C_{w2}^2}$

$C_2 = \sqrt{289.715^2 + 151.80^2}$

$$\boxed{C_2 = 327.07\, m/s}$$

$\tan\beta = C_{f2}/C_{w2} \Rightarrow \beta = \tan^{-1}\left[\frac{289.715}{151.80}\right]$

$$\boxed{\beta = 62.34^\circ}$$

$F_x = m(C_{w1} + C_{w2}) = 1.5\,(939.69 + 151.80)$

$$\boxed{F_x = 1637.23\,N}$$

$F_y = m[C_{f1} - C_{f2}] = 1.5\,(342.02 - 289.715)$

$$\boxed{F_y = 78.45\,N}$$

$P = m \times C_b\,(C_{w1} + C_{w2}) = 1.5 \times 350\,(939.69 + 151.80)$

$$\boxed{P = 573032.25\,W}$$

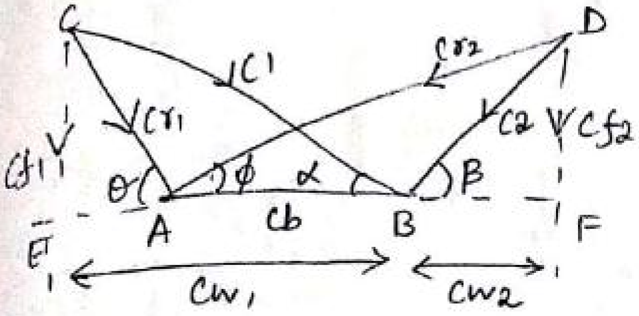
Steam enters the blade row of an impulse turbine with a velocity of 500 m/s at an angle of 30° to plane of rotation of blade. The mean blade speed is 280 m/s. The blade angle on the exit side is 35°. The blade friction coeff. is 12%. Determine,
(i) Blade angle at inlet.
(ii) Work done per kg of steam.
(iii) Diagram efficiency.
(iv) Axial thrust.

Sol: given,

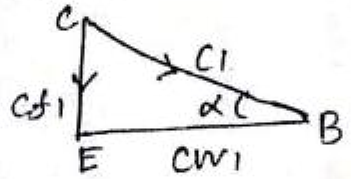
$C_1 = 500\,m/s$
$\alpha = 30^\circ$
$C_b = 280\,m/s$
$\phi = 35^\circ$
$K = 0.12$

to find,

$\theta = ?$
$W = ?$
$F_x = ?$
$\eta_b = ?$



Δ BEC,



$\sin\alpha = C_{f1}/C_1$

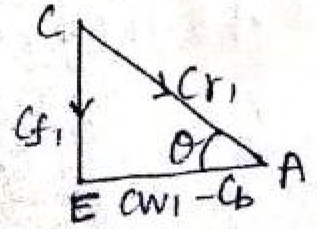
$\sin 30^\circ = C_{f1}/500$

$$\boxed{C_{f1} = 250\,m/s}$$

$\cos\alpha = \dfrac{C_{w1}}{C_1} \Rightarrow \cos 30^\circ = \dfrac{C_{w1}}{500}$

$$\boxed{C_{w1} = 433.01\,m/s}$$

Δ AEC,



$\tan\theta = \dfrac{C_{f1}}{C_{w1} - C_b}$

$\theta = \tan^{-1}\left(\dfrac{250}{433.01 - 280}\right)$

$$\boxed{\theta = 58.53^\circ}$$

$\sin\theta = C_{f1}/C_{r1}$

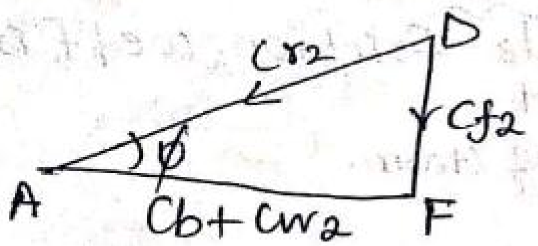
$\Rightarrow \sin 58.53^\circ = \dfrac{250}{C_{r1}}$

$$\boxed{C_{r1} = 293.11\,m/s}$$

WKt, $K = C_{r2}/C_{r1} \Rightarrow 0.12 = C_{r2}/293.11$

$$\boxed{C_{r2} = 35.17 \text{ m/s}}$$

ΔADF,



$\sin\phi = C_{f2}/C_{r2}$

$\sin 35^\circ = C_{f2}/35.17$

$$\boxed{C_{f2} = 20.17 \text{ m/s}}$$

$$\cos\phi = \frac{C_b + C_{w2}}{C_{r2}} \Rightarrow \cos 35^\circ = \frac{280 + C_{w2}}{35.17}$$

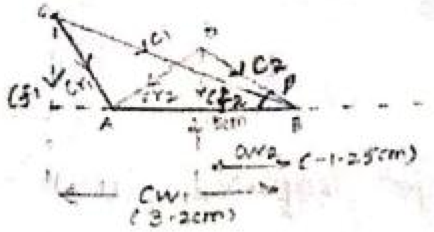
$C_{w2} =$

Problem ① graphical method.

① given,
$C_1 = 350\ m/s$
$\alpha = 20^\circ$
$C_b = 250\ m/s$
$\phi = 35^\circ$

take scale $10\ m/s = 1\ mm$
$250\ m/s = 25\ mm = 2.5\ cm$
$350\ m/s = 35\ mm = 3.5\ cm$

to find,
$\theta = ?$
$C_2 = ?$
$w = ?$



here no friction,
so, $C_{r1} = C_{r2}$
$C_{r1} = 1.4\ cm = 140\ m/s$
$C_2 = 1.5\ cm = 150\ m/s$
$C_{f2} = 0.8\ cm = 80\ m/s$
$\beta = -31^\circ$
$C_{w2} = -1.25\ cm = (-125)\ m/s$
$\theta = 58^\circ$
$C_{w1} = 3.2\ cm = 320\ m/s$

② given, $k = 0.85$
$C_1 = 1000\ m/s$
$\alpha = 20^\circ$
$C_b = 350\ m/s$
$m = 1.5\ kg/s$

take scale,
$10\ m/s = 1\ mm$
$1000\ m/s = 100\ mm = 10\ cm$
$350\ m/s = 35\ mm = 3.5\ cm$

$C_{w1} = 9.4\ cm$

$\theta = 30^\circ$
$C_{r1} = 6.8\ cm = 680\ m/s$
$C_{w1} = 9.4\ cm = 940\ m/s$
$C_{f1} = 3.4\ cm = 340\ m/s$
$k = C_{r2}/C_{r1}$
$0.85 = \frac{C_{r2}}{680}$
$C_{r2} = 578$

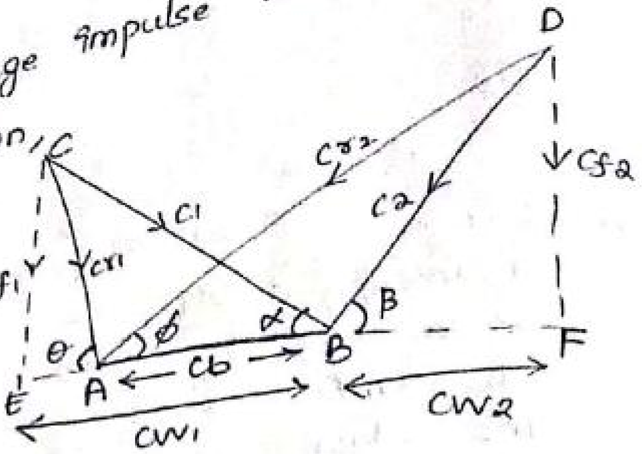
16/9/19 Impulse turbine.

① In De-Laval turbine, steam comes from nozzle with a velocity of 1,200 m/s and nozzle angle is 20°. The mean blade velocity is 400 m/s and inlet & outlet angles of blade are equal. The mass of steam flowing through the turbine per hour is 1000 kg. Calculate the following,

(i) Blade angles. (θ, ϕ)
(ii) Relative velocity of steam entering the blade.
(iii) Tangential force on blades.
(iv) Power developed.
(v) Blade efficiency.

Sol: given, single stage impulse turbine.

this is without friction,



$C_{r1} = C_{r2}$

$\theta = \phi$

given,

$C_1 = 1200\,m/s$

$\alpha = 20°$

$C_b = 400\,m/s$

$\theta = \phi$

$m = 1000\,kg/hr = 0.27\,kg/s$

to find,

$\theta = ?$ $\phi = ?$

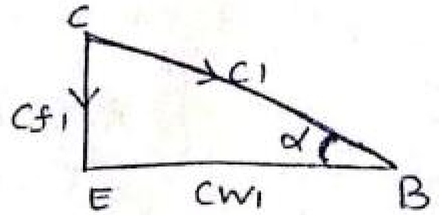
$C_{r1} = ?$

$F_x = ?$

$P = ?$

$\eta_{blade} = ?$

from ΔCEB,



$\sin\alpha = C_{f1}/C_1$

$\sin 20° = C_{f1}/1200$

$\boxed{C_{f1} = 410.424172\,m/s}$

~~as $C_{f1} = C_{f2}$~~

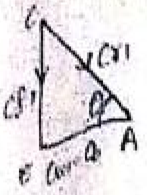
~~$C_{f2} = 410.424172\,m/s$~~

$\cos\alpha = \dfrac{C_{W1}}{C_1}$

$\cos 20° = \dfrac{C_{W1}}{1200}$

$\boxed{C_{W1} = 1127.63145\,m/s}$

from ΔCEA,



$$\left(\cos\theta = \frac{C_{w1} - C_b}{C_{r1}}\right) \left(\sin\theta = \frac{C_{f1}}{C_{r1}}\right) \left(\tan\theta = \frac{C_{f1}}{C_{w1} - C_b}\right)$$

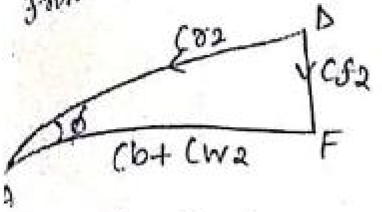
$$\tan\theta = \frac{410.424172}{1127.631145 - 400} \Rightarrow \theta = \tan^{-1}(0.5640)$$

$$[\theta = 29.42°]$$

$$\sin 29.42° = \frac{410.424172}{C_{r1}}$$

$$[C_{r1} = C_{r2} = 835.5407691\,m/s] \text{ ans}$$

from ΔADF,



as $(\theta = \phi = 29.42°)$ ans

$$\sin\phi = C_{f2}/C_{r2}$$

$$\sin 29.42° = C_{f2}/835.5407691$$

$$\tan\phi = C_{f2}/C_b + C_{w2}$$

$$[C_{f2} = 410.424172\,m/s]$$

$$\tan 29.42° = \frac{410.424172}{400 + C_{w2}}$$

$$400 + C_{w2} = 727.7914371$$

$$[C_{w2} = 327.7914371\,m/s]$$

tangential force,

$$F_x = m\,(C_{w1} + C_{w2})$$

$$F_x = 0.27\,(1127.631145 + 327.7914371)$$

$$[F_x = 392.9640\,N] \text{ ans}$$

Power developed,

$$P = m \times C_b\,(C_{w1} + C_{w2})$$

$$P = 0.27 \times 400\,(1127.631145 + 327.7914371)$$

$$[P = 157185.6389\,W] \text{ ans}$$

Efficiency of blade,

$$\eta_b = \frac{2C_b(C_{w1} + C_{w2})}{C_1^2} = \frac{2 \times 400 \times (1455.422582)}{1200^2}$$

$$[\eta_b = 80.85\,\%] \text{ ans}$$

so, we get

$(\theta = 29.42^\circ)$

$(\phi = 29.42^\circ)$

$(C_{r1} = 835.5407691 \, m/s)$

$(F_x = 392.9640 \, N)$

$(P = 157185.6389 \, W)$

$(\eta_{blade} = 80.85\%)$

18/9/19

Problem: Impulse turbine.

① The foll. data refer to a single stage impulse turbine. Isentropic nozzle enthalpy drop equal to 200 KJ/kg, nozzle angle is 20°, ratio of blade speed to whirl component of steam speed is 0.5, nozzle efficiency is 85%, blade velocity coefficient is 0.9, velocity of steam entering the nozzle is 35 m/s. Find the,

(i) Blade angle at inlet and outlet, if steam enters the blades without shock and leaves the blade in axial direction.

(ii) Blade efficiency.

(iii) Power developed.

(iv) Axial thrust, if steam flow rate is 5 Kg/s.

Sol: given,

$\Delta h = 200 KJ/kg \; (h_1 - h_2)$

$\eta_{nozzle} = 85\%$

$\alpha = 20^\circ$ (nozzle)

$C_b/(C_{w1}+C_{w2}) = 0.5$

$K = C_{r2}/C_{r1} = 0.9$

$C_1 = 35 m/s$ (nozzle inlet velocity)

nozzle exit velocity,

$C_2 = \sqrt{2000(h_1 - h_2) + C_1^2} \; m/s$ = Inlet velocity of turbine to find,

$\theta = ?$ $\phi = ?$

$\eta_b = ?$ $P = ?$

$F_y = ?$

<u>Note</u> In unit 1 (nozzle), if enthalpy drop given in ques, it is actual enthalpy drop

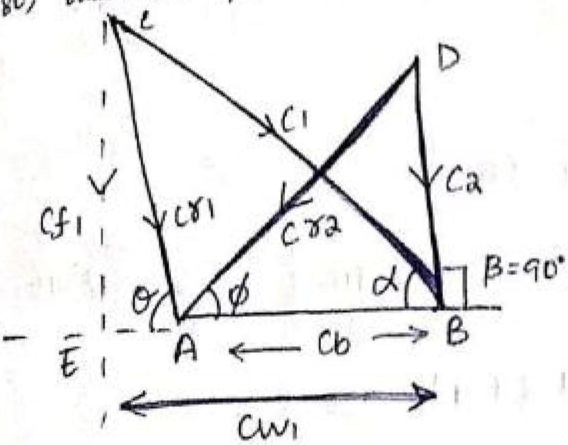
$C_2 = \sqrt{2000(h_1 - h_2)}$

But, if we find h_1, h_2 by using steam table, i.e, $h_1 = h_{f1} + x_1 h_{fg1}$, it is isentropic enthalpy drop. $(h_1 - h_2)$

here, $C_2 = \underbrace{\sqrt{2000(h_1 - h_2) + C_1^2}}_{\text{actual enthalpy drop}}$

$$\eta_{nozzle} = \frac{\text{actual enthalpy drop}}{\text{isentropic enthalpy drop} \to \text{given}} \%$$

So, subs. and find 'C_2' which is inlet angle of turbine.



$(C_2 = C_{f2})$

$(C_{w2} = 0)$

$$0.85 = \frac{\text{act. enth. drop}}{200}$$

(act. enth. drop = 170 KJ/Kg)

$C_2 = \sqrt{2000(170) + 35^2}$

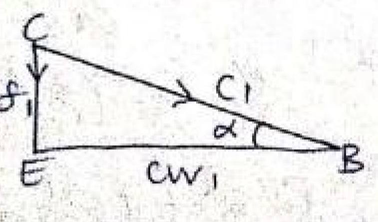
$\left[C_2 = 584.1446 \text{ m/s}\right] =$ 'C_1' (turbine)

$$C_b / (C_{w1} + C_{w2}) = 0.5$$

$$C_b / (C_{w1} + 0) = 0.5$$

$$\left[C_b / C_{w1} = 0.5\right]$$

from ΔCEB,

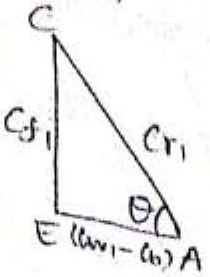


$\sin\alpha = C_{f1}/C_1$

$\sin 20° = C_{f1} / 584.1446$

$\left[C_{f1} = 199.7892 \text{ m/s}\right]$

$\Delta CEA,$



from ΔCEB_1

$$\cos\alpha = \frac{Cw_1}{C_1}$$

$$\cos 20^\circ = \frac{Cw_1}{584.1446}$$

$$\boxed{Cw_1 = 548.9163 \ m/s}$$

$(Cb = 0.5\ (w_1)$

$(Cr_1 = Cr_2/0.9)$

$$\sin\theta = \frac{Cf_1}{Cr_1}$$

$$\cos\theta = \frac{Cw_1 - Cb}{Cr_1}$$

$$\cos\theta = \frac{548.9163 - 0.5(548.9163)}{Cr_1}$$

$$\tan\theta = Cf_1 / Cw_1 - Cb$$

$$\tan\theta = 199.7892 / (548.9163 - 0.5(548.9163))$$

$$\theta = \tan^{-1}(0.7279)$$

$$\boxed{\theta = 36.05^\circ}$$

$$\sin 36.05^\circ = 199.7892 / Cr_1$$

$$\boxed{Cr_1 = 339.4940 \ m/s}$$

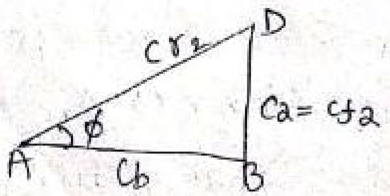
$$Cr_2 = 0.9 \times 339.4940$$

$$\boxed{Cr_2 = 305.5446 \ m/s}$$

$$Cb = 0.5 \times Cw_1 = 0.5 \times 548.9163$$

$$\boxed{Cb = 274.45815 \ m/s}$$

from ΔABD,



$$\cos\phi = \frac{Cb}{Cr_2}$$

$$\phi = \cos^{-1}\left(\frac{274.4581}{305.5446}\right)$$

$$\boxed{\phi = 26.06^\circ}$$

$\sin\phi = C_2 / C_{r2} \Rightarrow \sin 26.06° = C_2 / 305.5446$

$$\boxed{C_2 = 134.2294\ m/s}$$

Power developed,

$P = m C_b (C_{w1} + C_{w2})$

$P = 5 \times 274.4581 (548.9163)$

$P = 753272.6238$ W

$\boxed{P = 753.27 \text{ KW}}$

Efficiency of blade,

$\eta_b = \frac{2 C_b (C_{w1} + C_{w2})}{C_1^2} \times 100$

$\eta_b = \frac{2 \times 274.4581 (548.9163 + 0)}{584.1446^2}$

$\boxed{\eta_b = 88.30\%}$

Axial thrust,

$F_y = m (C_{f1} - C_{f2})$ $(C_2 = C_{f2})$

$F_y = 5 (199.7892 - 134.2294)$

$\boxed{F_y = 327.799 N}$

So, we get

$\boxed{\theta = 36.05^\circ}$ $\boxed{F_y = 327.799 N}$

$\boxed{\phi = 26.06^\circ}$

$\boxed{\eta_b = 88.30\%}$

$\boxed{P = 753.72 \text{ KW}}$

20/1/19 Problem: (Reaction turbine)

(1) The angles at inlet and discharge of blading of 50% of reaction turbine are 20°, 35° respec. The speed of rotating is 1600rpm and at a particular stage, the mean ring shaft dia. 0.72m and steam condition is 1.8bar and 0.98 dryness.

Estimate: (a) The required height of blading to pass 4.2 kg/s of steam.

(b) Power developed by ring when 50% of RT.

note: (two angle given reaction turbine)

(i) $\alpha = \phi$ (inlet)

(ii) $\theta = \beta$ (exit)

(iii) $C_{f1} = C_{f2} = C_f$

(iv) $C_{r1} = C_2$

(v) $C_{r2} = C_1$

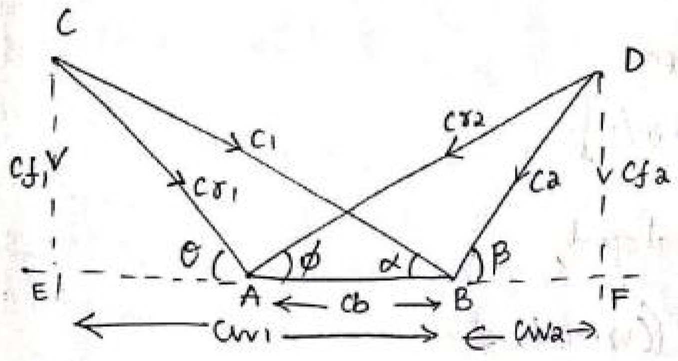
Sol:
given, $N = 1600 rpm$ $P = 1.8 bar$ $x = 0.98$ $\alpha = \phi = 20°$

$\theta = \beta = 35°$ $(d = 0.72 m)$ $m = 4.2 kg/s$ of steam

to find,

(i) $P = m\, C_b\, (C_{w1} + C_{w2})$ kW

(ii) $h = ?$ $\left[m = \frac{Q}{x V_g} = \frac{\pi d h C_f}{x V_g} \right]$ → 1.8bar pressure in steam table



For 50% Reaction turbine,

$\alpha = \phi$ $C_{f1} = C_{f2} = C_f$

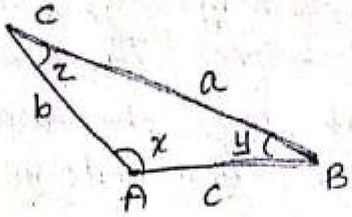
$\theta = \beta$ $C_{r2} = C_1$; $C_{r1} = C_2$

First,

$C_b = \pi d N / 60 = \pi \times 0.72 \times 1600 / 60$

$\boxed{C_b = 60.3185 m/s}$

Sine rule, take ΔABC



$$\frac{a}{\sin x} = \frac{b}{\sin y} = \frac{c}{\sin z}$$

$x = 180° - \theta = 145°$

So, $x + y + z = 180°$

$145° + 20° + z = 180°$

$z = 15°$

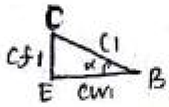
$$\frac{C_1}{\sin x} = \frac{C_{r1}}{\sin 20°} = \frac{C_b}{\sin z}$$

$$\frac{C_1}{\sin 145°} = \frac{C_{r1}}{\sin 80°} = \frac{60.3185}{\sin 15°}$$

$$\boxed{C_{r1} = 79.70 \text{ m/s}} = C_2$$

$$\boxed{C_1 = 133.67 \text{ m/s}} = C_{r2}$$

From ΔBEC,

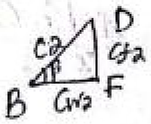


$$\cos\alpha = \frac{C_{w1}}{C_1}$$

$$\cos 20° = \frac{C_{w1}}{133.67}$$

$$\boxed{C_{w1} = 125.60 \text{ m/s}}$$

From ΔBDF,



$$\cos\beta = C_{w2}/C_2$$

$$\cos 35° = C_{w2}/79.70$$

$$\boxed{C_{w2} = 65.28 \text{ m/s}}$$

So, power developed,

$$P = m \times C_b \times (C_{w1} + C_{w2})$$

$$P = 4.2 \times 60.3185\,(125.6 + 65.28)$$

$$P = 48357.10018 \text{ W}$$

$$\boxed{P = 48.3571 \text{ kW}}$$

height,

$$m = \frac{\pi d h \times C_f}{x \times V_g}$$

From steam table,

for $P = 1.8$ bar

$[V_g = 0.97713 \ m^3/kg]$

$$4.2 = \frac{\pi (0.72)(h)(45.7232)}{0.98 \times 0.97713}$$

$[h = 0.0388 m]$

So,

$\left[\begin{array}{l} P = 45.35771 \ KW \\ h = 0.0388 \ m \end{array}\right]$

ΔBDF,

$$C_2^2 = C_{w2}^2 + C_{f2}^2$$

$$79.7^2 = 65.28^2 + C_{f2}^2$$

$$C_{f2}^2 = 2090.6116$$

$[C_{f2} = C_{f1} = C_f = 45.7232 \ m/s]$

9/10/19

Cogeneration & Residual Heat recovery.

Co-generation: → Electricity + heat required place.

→ Types of co-generation plant

(i) Topping cycle cogeneration plant;

(a) Gas turbine topping cogeneration.
(b) Steam turbine topping cogeneration.

(ii) Bottoming cycle cogeneration plant;

(a) Factors influencing cogeneration system.
(b) Application of cogeneration system.
(c) Advantages of " "
(d) Disadvantages of " "
(e) Cycle analysis.

Residual waste heat recovery:

Three 'R's of residual heat,

(a) Reduction. (b) Recycling. (c) Recovery.

Utilization of Residual heat:

→ In process recycling.
→ In-plant recovery.
→ Electricity generation.

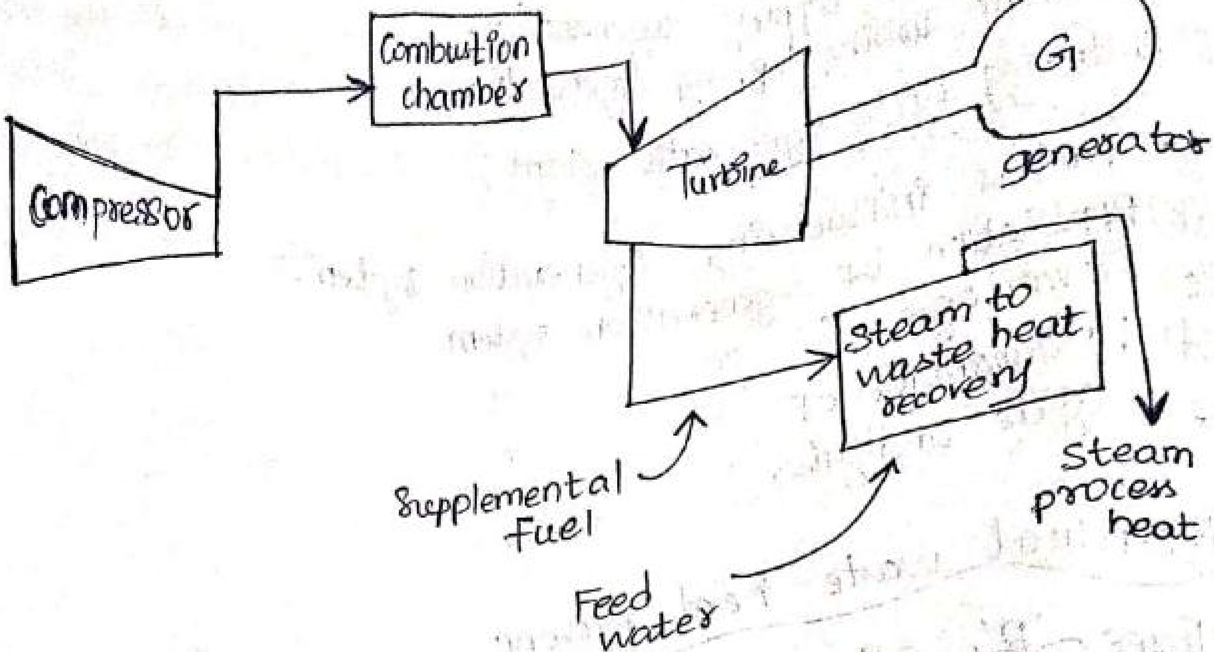
Residual heat & power generation:

→ Conventional steam power plants using Rankine cycle.
→ Organic rankine cycle.
→ Ammonia - water system.
→ Thermoelectric power generation System.

Residual heat Recovery equipments and systems:

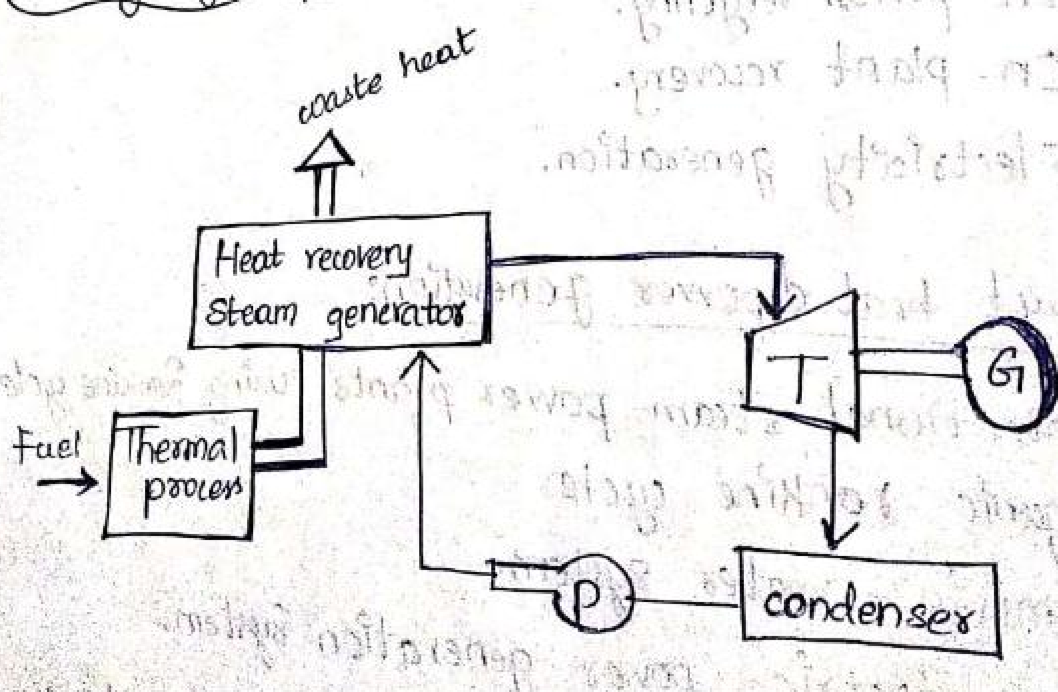
→ Recuperators (heat exchanger). → Heat wheels.
→ Regenerators.
→ Heat pipes. → Economizers.

- Topping cycle power plants:



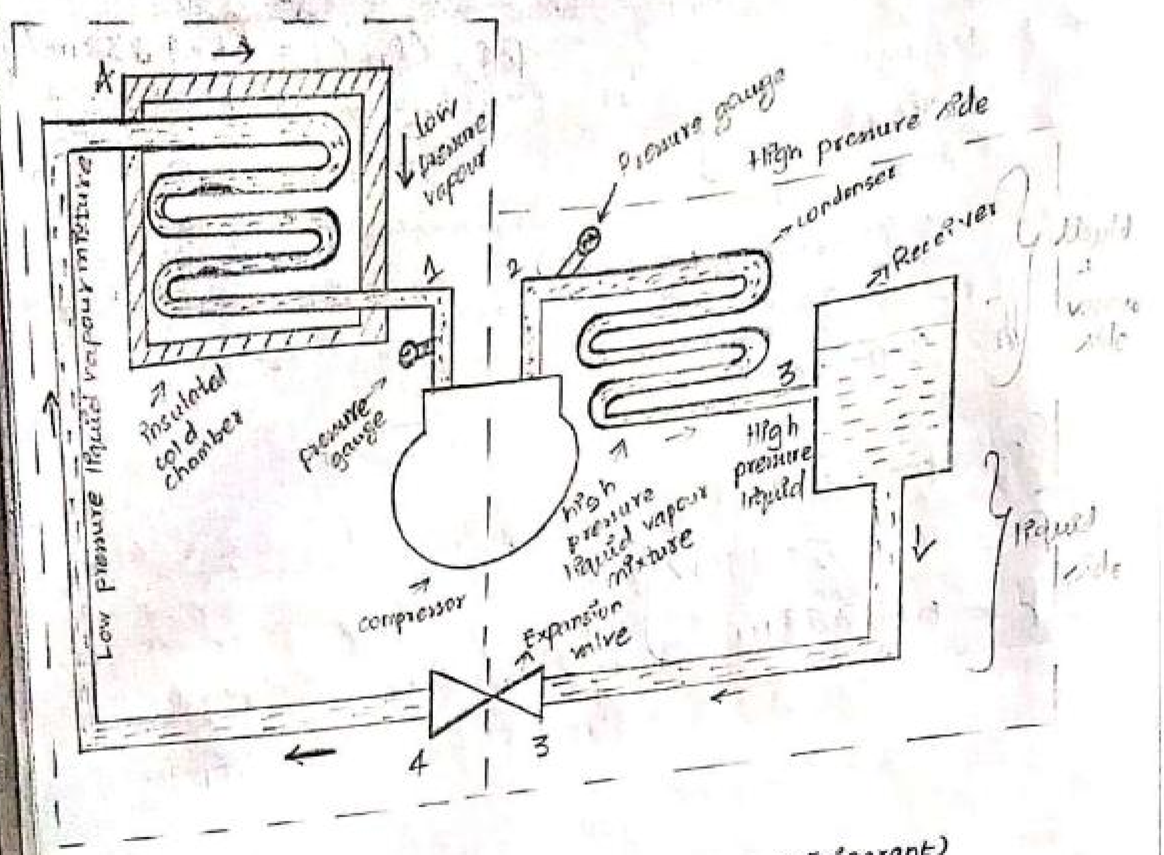
: Gas turbine topping cycle:

- Bottoming cycle plant:



24/9/19 Unit-5. REFRIGERATION AND AIR CONDITIONING:

Vapour Compression refrigeration system:



Main parts:

① Compressor. (low pressure refrigerant vapour to high pressure refrigerant vapour)

② Condenser. (high pressure refrigerant vapour to high pressure liquid refrigerant)

③ Expansion valve. (Expansion of high pressure liquid refrigerant to low press. liq. vapour mixture of refrigerant)

④ Evaporator. (sending out hot air by evaporation from refrigerator)

* (2m) Refrigeration can be defined as, "a process of removing heat from a substance or space and maintaining its temp. below surrounding temperature."

* Refrigerant - The refrigeration system vary according to the purpose and type of refrigerant used. Some of commonly used refrigerant are,

(a) Liquid Ammonia (NH_3)

(b) Carbon dioxide (CO_2)

(c) Sulphur dioxide (SO_2

(d) "Feron" which has types of R-11, R-12, R-21, R-134a, R-22, R-50, R-502, R-134

Types of Refrigeration system:

(i) Vapour compression refrigeration system. (X)

(ii) Vapour absorption refrigeration system.

Main components of Refrigeration system:

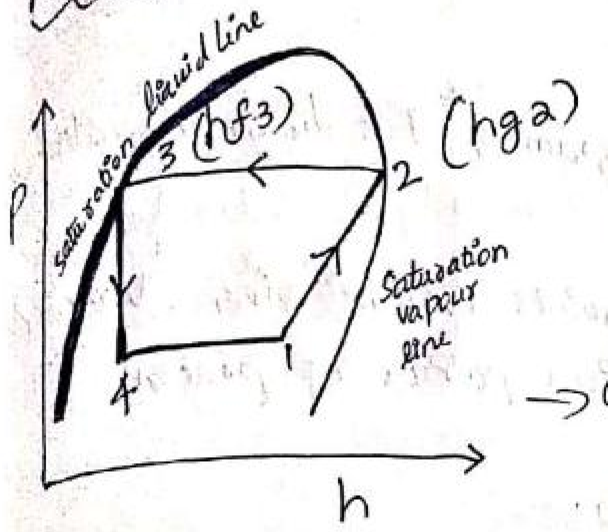
① Compressor.

② Condenser.

③ Expansion valve.

④ Evaporator.

p-h diagrams



inlet outlet

1-2 compression.

2-3 Condensation.

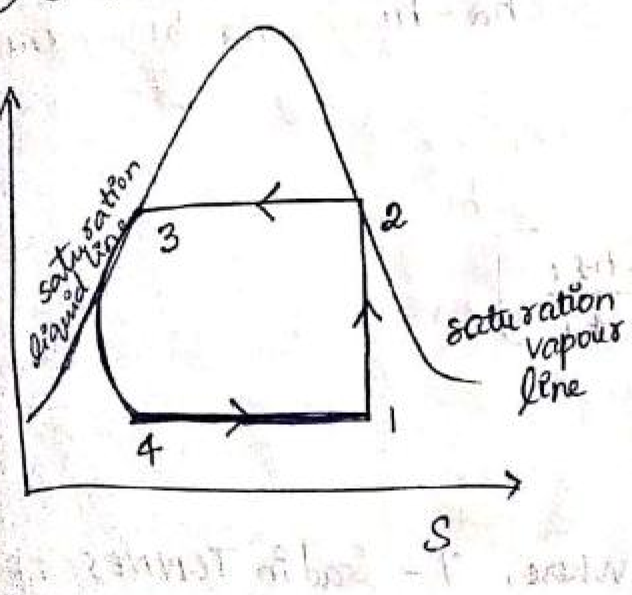
3-4 Expansion

4-1 Evaporation.

→ $(P_4 = P_1)$ $(P_2 = P_3)$

$(h_3 = h_4)$

T-S diagram:



→ $(S_1 = S_2)$

$(T_1 = T_4)$

$(T_2 = T_3)$

Performance calculations:

* The performance of refrigeration system is generally measured by a factor "COP - coefficient of performance".

* The refrigeration load in system is given in terms of TONNES of refrigeration.

* A TONNE of refrigeration is defined as, "the quantity of heat required to be removed from 1 TONNE of water (1000kg) at 0°C to convert into ice in 24 hours."

* 1 TONNE of refrigeration consumes 3.5KW in 24 hrs.

Important formulas:

→ With reference to 'P-h' diagram & 'T-s' diagram, calculations are followed.

→ Usually, the pressure at various points will be given. We need to calculate enthalpy at various points of process.

(i) Coefficient of performance (COP):

$$\boxed{COP = \frac{\text{Refrigerant effect}}{\text{compressor work}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}}$$

↓
for cooling

$$\boxed{COP = \frac{h_2 - h_3}{h_2 - h_1} = \frac{h_2 - h_{f3}}{h_2 - h_1}}$$

↓
for heating

(ii) Mass of refrigerant:

$$m = \frac{3.5T}{h_1 - h_{f3}}$$

where, T - load in TONNES of refrigeration

h_1 - Enthalpy at compressor inlet.

h_3 - Enthalpy at condenser exit.

(iii) Indicated power:

$$P = m\,(h_2 - h_1)$$

(iv) Quantity of cooling air (or) water required by condenser:

$$[m_w \times C_{pw} \times \Delta T = m\,(h_2 - h_3)]$$

where,

m_w = mass of water (or) air

C_{pw} = spf. heat of water (or) air

ΔT = change in temp. of water (or) air

(v) The dimensions of single acting compressor:

$$(\pi/4 \times D^2 \times L)\,\eta_v \times (N/60) = m \times v_{s1}$$

where,

$D \rightarrow$ diameter of cylinder.

$L \rightarrow$ stroke length.

$\eta_v \rightarrow$ volumetric efficiency.

$N \rightarrow$ speed of compressor.

$v_{s1} \rightarrow$ spf. volume of saturated refrigerant at point 1

$$\text{Capacity} = ((h_1 - h_4) \times m) \div 210 \quad \text{in tonne}$$

$$\text{Power} = \text{heat rejected} / \text{Actual COP}$$

$$1 \text{ tonne} = 210 \text{ kJ/min}$$

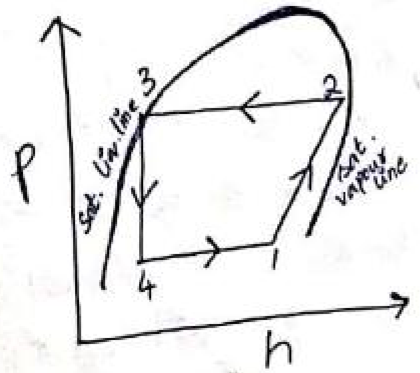
Problem:

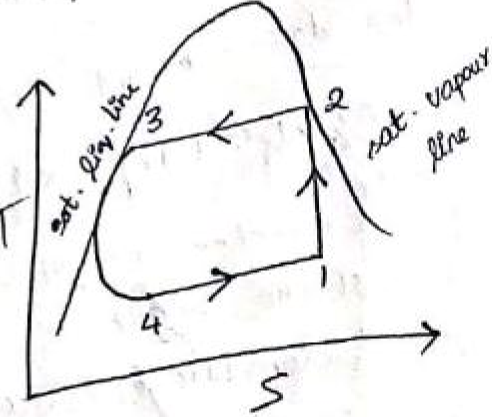
① 4kg per minute of ammonia (NH_3) is circulated in an ammonia vapour compression refrigeration system. Pressure in the evaporator is 2bar. Dryness fraction of ammonia at entry and exit of evaporation is 0.19 and 0.85 respec. Workdone during the compression is 150 KJ/kg of ammonia. Calculate COP of system, volume of vapour entering the system per minute. Latent heat of ammonia at 2bar is 1325 kJ/kg and specific volume is $0.58 m^3/kg$.

Sol:

T-s diagram

P-h diagram





1-2 compression
2-3 condensation
3-4 Expansion
4-1 Evaporation

given, $m = 4\ kg/min$
$P_1 = P_4 = 2bar$
$x_4 = 0.19$ $\quad x_1 = 0.85$

compressor work,
$h_2 - h_1 = 150 KJ/kg$
$h_{fg} = 1325 KJ/kg$

$$COP = \frac{\text{refrigerant effect}}{\text{compressor work}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$h_1 = x_1 \times h_{fg} = 0.85 \times 1325 = 1126.25$ KJ/kg
$h_4 = x_4 \times h_{fg} = 0.19 \times 1325 = 251.75$ KJ/kg

so, $$COP = \frac{1126.25 - 251.75}{150}$$

$$\boxed{COP = 5.83}$$

Specific volume $= 0.58 m^3/kg$

$m = \frac{V}{v} \Rightarrow 4 = V/0.58 \Rightarrow V = 2.32\ m^3/min$

So,

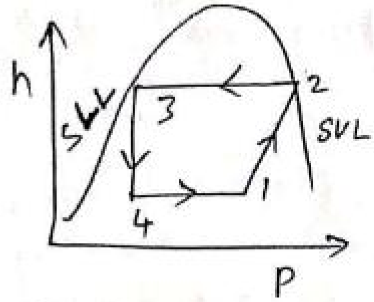
$$\boxed{\begin{array}{l} COP = 5.83 \\ V = 2.32\ m^3/min \end{array}}$$

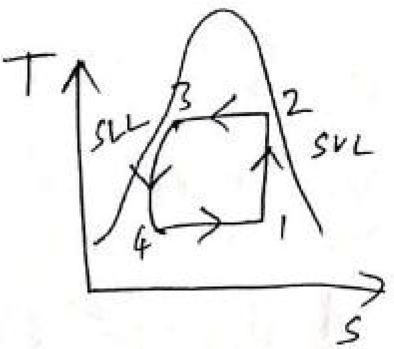
---

② A vapour compression system works b/w pressure limits of 58 bar and 25 bar. At end of compression, fluid is just dried and there is no sub cooling. Determine, (i) COP (ii) capacity of system if refrigerant flow at a rate of 5 KJ/min.
Properties of refrigerant is given below,

Pressure (bar)	Saturation temp (K)	Enthalpy (KJ/kg) liquid	Enthalpy (KJ/kg) vapour	Entropy liquid	Entropy vapour	
58	295	152 $h_{f2} = h_{f3} = h_3$	293.3 (h_{g2})	0.554 $S_{f2} = S_{f3} = S_3$	1.0332 S_{g2}	(2, 3) point
25	261	56.32 h_{f1}	322.6 h_{g1}	0.226 S_{f1}	1.2464 S_{g1}	(4, 1) point

Sol:





given from table,

$h_{f3} = h_4 = 152\ KJ/kg$ $\qquad S_{f3} = S_4 = 0.554\ KJ/kg\,K$

$h_{f1} = 56.32\ KJ/kg$ $\qquad S_{f1} = 0.226\ KJ/kg\,K$

$h_{g2} = 293.3\ KJ/kg$ $\qquad S_{g2} = 1.0332\ KJ/kg\,K$

$h_{g1} = 322.6\ KJ/kg$ $\qquad S_{g1} = 1.2464\ KJ/kg\,K$

$P_2 = P_3 = 58\ bar$ $\qquad S_1 = S_2 = S_{g2}$

$P_1 = P_4 = 25\ bar$ $\qquad COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_{g2} - h_1}$

$COP = \frac{h_1 - 152}{293.3 - h_1} \rightarrow \boxed{h_1 = h_{f1} + x_1 \times h_{fg1}}$

From $s_1 = s_2 = s_{g2}$

$1.0332 = 0.226 + x_1 (s_{fg1})$

$s_{fg1} = S_{g1} - S_{f1} = 1.0204$

$(x_1 = 0.79)$

$h_1 = 56.32 + 0.79 (h_{fg1})$

$h_{fg1} = h_{g1} - h_{f1} = 266.28$

$(h_1 = 266.6812\ KJ/kg)$

$$COP = \frac{266.68 - 152}{293.3 - 266.68} = 4.3$$

$\boxed{COP = 4.3}$ ans.

capacity $= m(h_1 - h_4) = 5(266.68 - 152)$

$\boxed{\text{Capacity} = 573.4 \text{ KJ/min}}$ ans

1 tonne of refrigeration = 210 KJ/min.

So, capacity $= \frac{573.4}{210} = 2.73$

$\boxed{\text{Capacity} = 2.73 \text{ tonne}}$

27/9/19

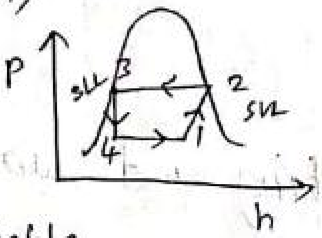
③ An ammonia (NH_3) refrigerator produces 30 tonnes of ice from and at 0°C in a day of 24 hrs. The temp range in compressor is from 25°C to −15°C. The vapour is dry saturated at end of compression. Assume COP is 60% theoretical. Calculate power req'd to drive compressor. Assume latent heat of ice is 335 KJ/kg, for properties of NH_3 refer table below,

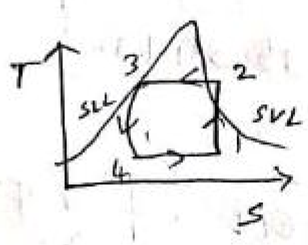
temp °C	hf (L) (KJ/kg)	hg (V) (KJ/kg)	sf (L) (KJ/kgK)	sg (V) (KJ/kgK)
25	298.9 $h_{f3}=h_3$	1465.8 h_{g2}	1.124 $s_{f3}=s_3$	5.039 s_{g2}
−15	112.34 h_{f1}	1426.5 h_{g1}	0.4572 s_{f1}	5.549 s_{g1}

(3, 2) points

(4, 1) points

Sol: given,





From table,

$h_{f3} = h_3 = 298.9\ KJ/kg = h_{f2}$ $\quad s_{g2} = 50.39\ KJ/kgK$

$h_{g2} = 1465.8\ KJ/kg$ $\quad s_{f3} = 1.124\ KJ/kgK = s_{f2}$

$h_{f1} = 112.34\ KJ/kg$ $\quad s_{g1} = 5.549\ KJ/kgK$

$h_{g1} = 1426.5\ KJ/kg$ $\quad s_{f1} = 0.4572\ KJ/kgK$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1} \qquad (h_2 = h_{g2})$$

$$\left[COP = \frac{h_1 - 298.9}{1465.8 - h_1}\right]$$

$s_{fg} = s_g - s_f$

$s_{fg1} = s_{g1} - s_{f1}$

$s_{fg1} = 5.0918$

as, $s_1 = s_2 = s_{g2}$

$$5.039 = 0.4572 + x_1 (5.0918)$$

$$(x_1 = 0.89)$$

$h_{fg1} = h_{g1} - h_{f1}$

$h_{fg1} = 1426.5 - 112.34$

$h_{fg1} = 1314.16$

$$h_1 = h_{f1} + x_1 h_{fg1}$$

$$h_1 = 112.34 + (0.89 \times 1314.16)$$

$$[h_1 = 1281.9424\ KJ/kg]$$

$$COP = \frac{1281.9424 - 298.9}{1465.8 - 1281.9424}$$

$$\boxed{COP = 5.34}$$

$$\boxed{\text{Actual } COP = 5.34 \times 0.6 = 3.2}$$

$$Q_{actual} = \frac{\overset{(m)}{30000} \times \overset{(LHF)}{335}}{24 \times 3600} = 116.31 \text{ KJ/kg}$$

Power,

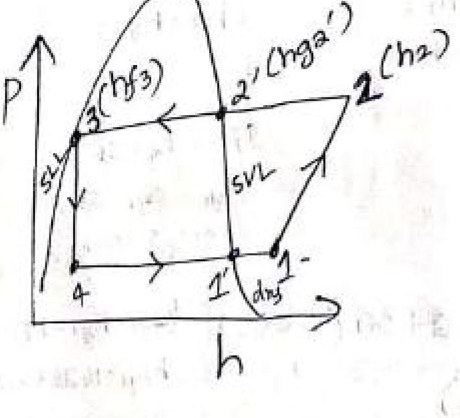
$$P = \frac{Q_{actual}}{\text{Actual } COP} = \frac{116.31}{3.2} = 36.34$$

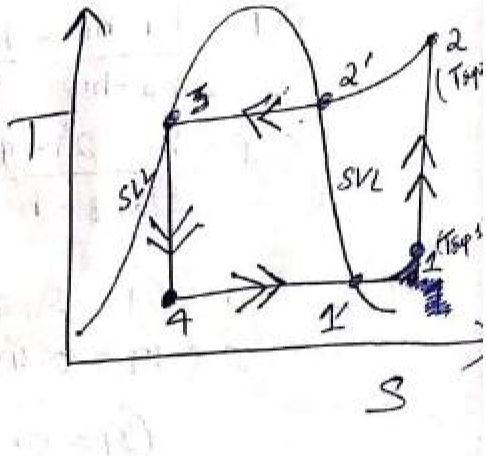
$$\boxed{P = 36.34 \text{ KW}}$$

30/9/19 Effect of superheating and sub-cooling of refrigerant:

① Super heating: $\Rightarrow P_3 = P_2' = P_2$, $P_4 = P_1' = P_1$, $h_3 = h_4$

$\uparrow S_3 = S_4$, $S_1 = S_2$, $T_4 = T_1'$, $T_2' = T_3$





$$COP = \frac{h_1 - h_4}{h_2 - h_1}$$

$h_{sup} = h_g + C_p (T_{sup} - T_{sat})$ (Superheated)

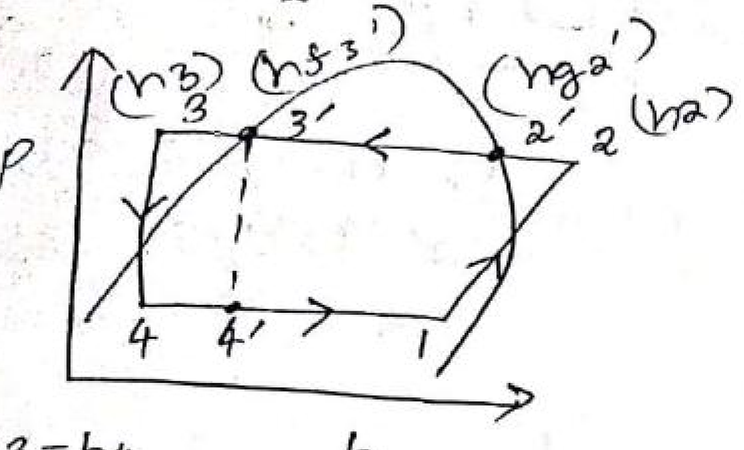
$h_g = h_f + h_{fg}$ (dry)

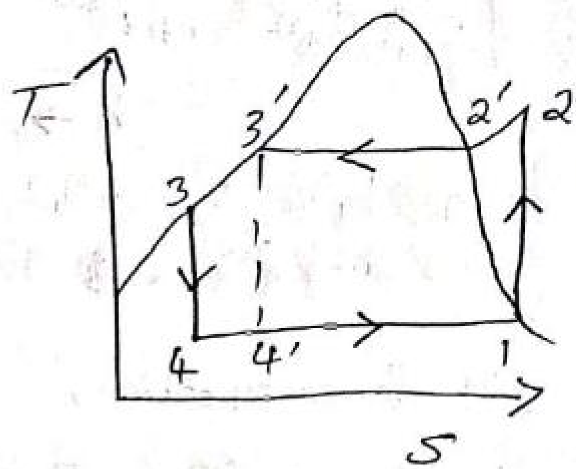
$h_{wet} = h_f + x h_{fg}$ (wet

Effect of superheating:

(i) Refrigerant effect is increased due to superheating.
(ii) Specific volume is increased by superheating.
(iii) Refrigeration work is increased.

② Subcooling:





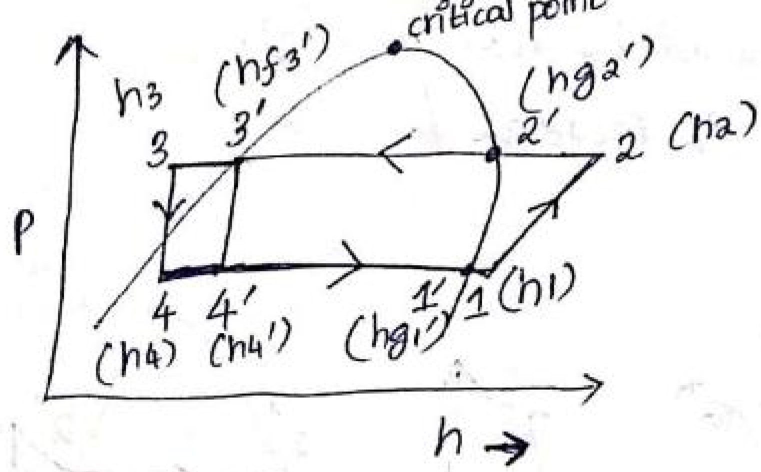
$h3 = h4$
$h3' = h4'$
$3' - 3$ is subcooling

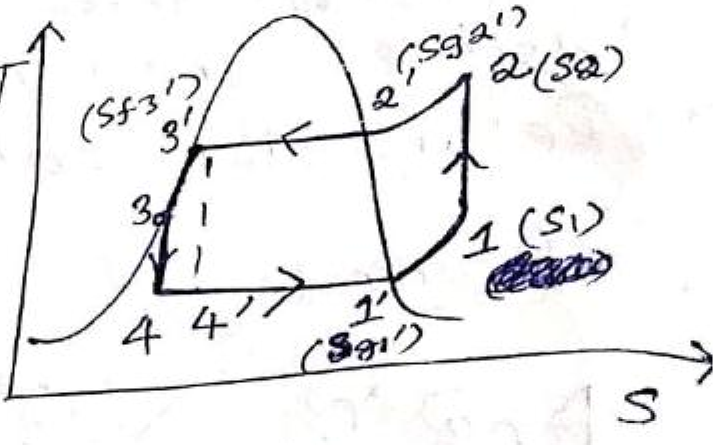
$$\left(COP = \frac{h_1 - h_4}{h_2 - h_1} \right)$$

Problem 5·8, 5·9, 5·10 (with superheating, subcooling)

1/10/19

Combined superheating and subcooling:





$(h_3 = h_4)(P_1 = P_1' = P_4' = P_4)$

$(h_3' = h_4')(P_2 = P_2' = P_3 = P_3')$

$(T_1' = T_4' = T_4)(S_1 = S_2)$

$(T_2' = T_3')(S_3 = S_4)$

$(S_3' = S_4')$

wkt for superheating,

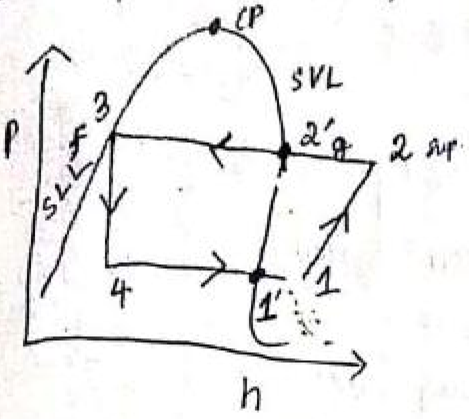
$$h_{sup} = h_g + C_p (T_{sup} - T_{sat})$$

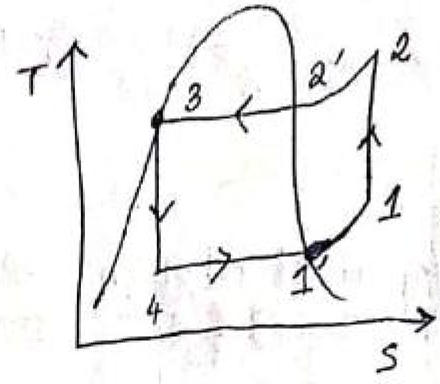
Problem:

① A vapour compression refrigeration plant works b/w pressure limits of 5.3 bar and 2.1 bar. The vapour is superheated at end of compression. Its temp. being 37°C. The vapour is superheated by 5°C before entering the compressor. If the spf. heat of superheated vapour is 0.63 kJ/kg K. Find COP of plant. Use data given below.

Pressure (bar)	Saturation temp. (°C)	Liquid heat (KJ/kg)	Latent heat (KJ/kg)
5.3	15.5	56.15 (h_{f2})	144.9 (h_{fg2})
2.1	-14.0	25.12 (h_{f1})	158.7 (h_{fg1})

Sol: Superheated condition,





~~$P_2 = 5.3$ bar $P_1 = 2.1$ bar~~

$P_3 = P_2' = 5.3\,bar$

$P_4 = P_1' = P_1 = 2.1\,bar$

$T_2 = 37°C$

$T_3 = T_2' = 15.4\,°C$

$h_{f2} = 56.15\ KJ/kg = h_{f3} = h_3 = h_4$

$h_{f1} = 25.12\ KJ/kg$

$h_{fg2} = 144.9\ KJ/kg$

$h_{fg1} = 158.7\ KJ/kg$

$C_p = 0.63\ KJ/KgK$

before compression, refrigerant is superheated by 5°C,

$h_1 = h_{g1} + C_p \times \underset{(5°C)}{\text{degree of heat}}$

where,

$h_{g1} = h_{f1} + h_{fg1} \Rightarrow h_{g1} = 25.12 + 158.7$

$h_{g1} = 183.82\ KJ/Kg$

$h_1 = 183.82 + 0.63 \times (5)$

$\boxed{h_1 = 186.97\ KJ/Kg}$

End of compression,

$h_2 = h_{g2} + C_p\,(T_{sup} - T_{sat})$

where, $h_{g2} = h_{f2} + h_{fg2} \Rightarrow h_{g2} = 56.15 + 144.9 = 201.05\ KJ/Kg$

Solve by (°C) not (K)

$h_2 = 201.05 + 0.63((37+273) - (15.5+273))$

$h_2 = 201.05 + 0.63(310 - 288.5)$

$\boxed{h_2 = 214.595\ KJ/Kg}$

$(h_3 = h_{f3} = h_4)$

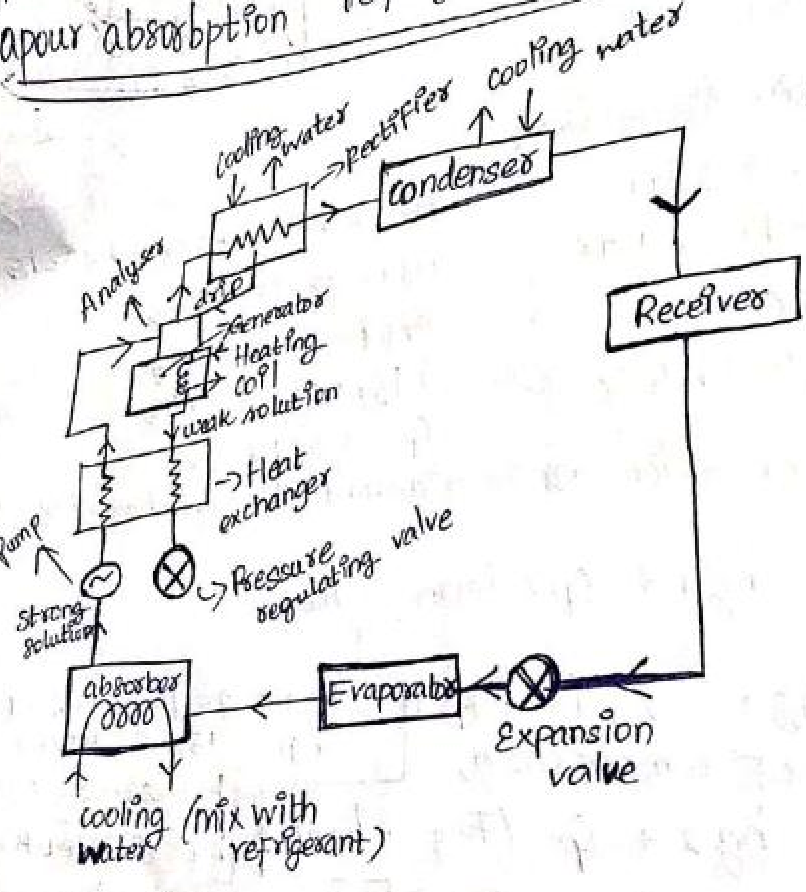
$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{186.97 - h_{f3}}{214.595 - 186.97} = \frac{186.97 - 56.15}{27.625}$$

$$\boxed{COP = 4.73}$$

So, we get $COP = 4.73$.

1/10/19

Vapour absorbption refrigeration system:



Psychrometry Formulas:

~~① Psychrometry properties:~~

① spf. humidity/humidity ratio/moisture content:

(i) $\omega = \frac{m_v}{m_a}$ (ii) $\omega = 0.622 \frac{P_v}{P_o - P_v}$

② Saturation ratio:

$$\mu = \frac{w}{w_s}$$

$$\mu = \frac{P_v}{P_s}\left(\frac{P_b - P_s}{P_b - P_v}\right)$$

$P_v \rightarrow$ partial pressure of water vapour.

$P_s \rightarrow$ partial pressure of saturated air.

$P_b \rightarrow$ Barometric pressure.

③ Relative velocity:

$$\phi = \frac{m_v}{m_s}$$

④ Total enthalpy:

$$h = (C_p \times t_d) + (w \times h_g)$$

$C_p \rightarrow$ spf. heat at const. pressure = 1.005 KJ/kgK

$t_d \rightarrow$ dry bulb temp.

$w \rightarrow$ Spf humidity.

$h_g \rightarrow$ spf. enthalpy of air to DBT.

⑤ Dalton's law of partial pressure:

$$[P_b = P_a + P_v]$$

where,

$P_b \rightarrow$ barometric pressure.

$P_a \rightarrow$ partial pressure of air.

$P_v \rightarrow$ Partial pressure of water vapour.

$$P_v = P_{sw} - \frac{(P_b - P_{sw}) \times (t_d - t_w)}{1527.4 - 1.3 t_w}$$

P_{sw} = Saturation pressure corresponding to WBT

$t_d \Rightarrow$ DBT

$t_w \rightarrow$ WBT