ANNAMACHARYA UNIVERSITY

DEPARTMENT OF MECHANICAL ENGINEERING

LECTURE NOTES

THERMAL ENERGY SYSTEMS

[24AMEC44T]

Prepared by
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ANNAMACHARYA UNIVERSITY

EXCELLENCE IN EDUCATION; SERVICE TO SOCIETY

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DEPARTMENT OF MECHANICAL ENGINEERING

Title of the Course:	**Thermal Energy Systems**
Category:	PCC
Semester:	IV Semester
Course Code:	24AMEC44T
Branch/es:	Mechanical Engineering

Lecture Hours	Tutorial Hours	Practice Hours	Credits
3	0	0	3

Course Objectives:

1. To familiarize about the Rankine cycle used for steam power plant & steam boilers.
2. To impart knowledge on the working of nozzles and condensers used in steam power plants.
3. To impart knowledge on the working of steam turbines.
4. To understand the principle and operation of various Refrigeration methods.
5. To acquire knowledge on different Psychometric Processes & Air conditioning systems.

Course Outcomes:

At the end of the course, the student will be able to

1. Calculate the efficiency of Rankine cycle and summarize the working of different boilers used in steam power plant.
2. Analyze the flow through Nozzles and condensers used in steam power plant.
3. Solve the performance parameters of Steam Turbines.
4. Explain various refrigeration methods for specific uses.
5. Analyze various heat load concepts using RSHF and GSHF using psychrometric processes for different air conditioning systems.

Unit 1 Introduction to steam power plant 10

Rankine cycle - Schematic layout, Thermodynamic Analysis, Concept of Mean Temperature of Heat addition, Methods to improve cycle performance – Regeneration – reheating.

Boilers: Classification based on Working principles - Fire tube and water tube boilers – High pressure Boilers.

Unit 2 Steam Nozzles & Condensers 10

Steam Nozzles: Function of nozzle – applications - types, Flow through nozzles, thermodynamic analysis – assumptions -velocity of nozzle at exit- Ideal and actual expansion in nozzle, velocity coefficient, condition for maximum discharge, critical pressure ratio.

Condensers: Requirements of steam condensing plant, rare fraction – Classification of condensers – working principle of different types – vacuum efficiency and condenser efficiency – air leakage, sources and its effects.

Unit 3 Steam Turbines 10

Classification of steam turbines -impulse turbine and reaction turbine -compounding of turbines - velocity diagrams for impulse and reaction turbines, efficiency, degree of reaction - governing of turbines.

Unit 4 Refrigeration 12

Introduction: Refrigeration, Necessity and applications – Unit of refrigeration and C.O.P. – Different refrigeration methods.

Air Refrigeration: Bell Coleman Cycle-Simple air-cooling system- Refrigeration needs of Air craft.
Vapour Compression Refrigeration: Basic cycle - working principle and essential components of the plant – COP – Expander vs. Throttling, effect of sub cooling and super heating – numerical Problems.
Vapour Absorption Refrigeration System: Description and working of NH_3- water system, Li Br – water (Two shell) System, Principle of operation of three Fluid absorption systems, properties of common refrigerants.

Unit 5 Air Conditioning **12**

Psychometric Properties & Processes – Need for Ventilation, Consideration of Infiltrated air – Heat Load concepts: RSHF, GSHF- Problems. Requirements of human comfort and concept of Effective Temperature- Comfort chart –Comfort Air Conditioning-Summer, winter & year-round air-conditioning systems (elementary treatment).

Prescribed Textbooks:

1. Thermal Engineering, R.K. Rajput, S.Chand & Co., 10th edition 2020, Laxmi publications, ISBN-9788131808047
2. Refrigeration and Air Conditioning by C P Arora, 4th Edition 2021. ISBN-13 : 978-9390385843

Reference Books:

1. Thermodynamics: An Engineering Approach, Cengel .Y.A and Boles M.A, 5/e, McGraw-Hill, 9th Ed 2019, ISBN-13: 978-9339221652
2. A Textbook of Thermal Engineering by R. S Khurmi & JS Gupta, S.Chand, 16th Ed.2020, ISBN 9788121925730
3. A Course In Refrigeration And Air-Conditioning By Domkundwar, Arora, Domkundwar, 2022. ISBN-13 : 978-1111644475
4. A text book of Refrigeration and Air Conditioning by R.K Rajput , S K Kataria & sons, 3rd Edition 2015. ISBN-13 : 978-93-5014-255-4

Web resources:

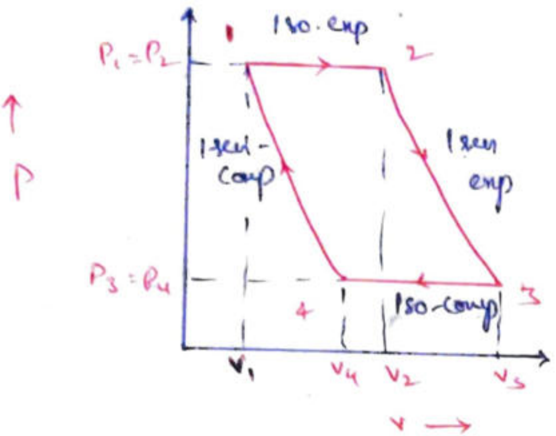
1. https://archive.nptel.ac.in/courses/112/107/112107291/
2. https://youtu.be/DuLFDzQVTU4
3. https://youtu.be/h1Yt4ibYXfA
4. https://archive.nptel.ac.in/courses/112/107/112107216/
5. https://archive.nptel.ac.in/courses/112/105/112105128/

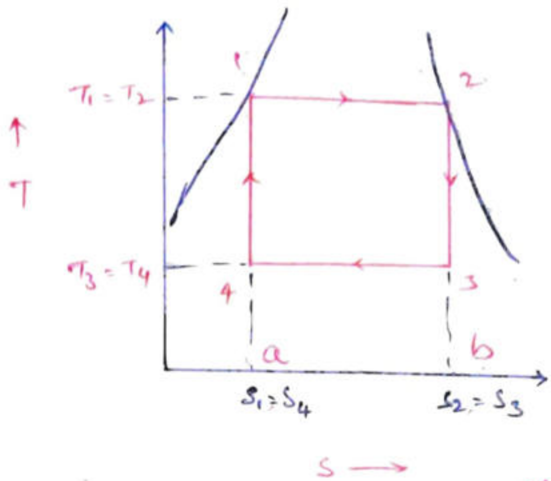
CO-PO Mapping:

Course Outcomes	Engineering Knowledge	Problem Analysis	Design/Development of solutions	Conduct investigations of complex problems	Modern tool usage	The engineer and society	Ethics	Individual and team work	Communication	Project management and finance	Life-long learning	**PSO1**	**PSO2**
24AMEC44T.1	3	3	2	3	-	-	-	-	-	-	1	2	1
24AMEC44T.2	3	2	2	2	-	-	-	-	-	-	1	2	1
24AMEC44T.3	3	3	2	3	-	-	-	-	-	-	1	2	1
24AMEC44T.4	3	2	2	1	-	-	-	-	-	-	2	2	2
24AMEC44T.5	3	3	2	3	-	-	-	-	-	-	2	2	3

Carnot cycle :- [Steam is the working substance]

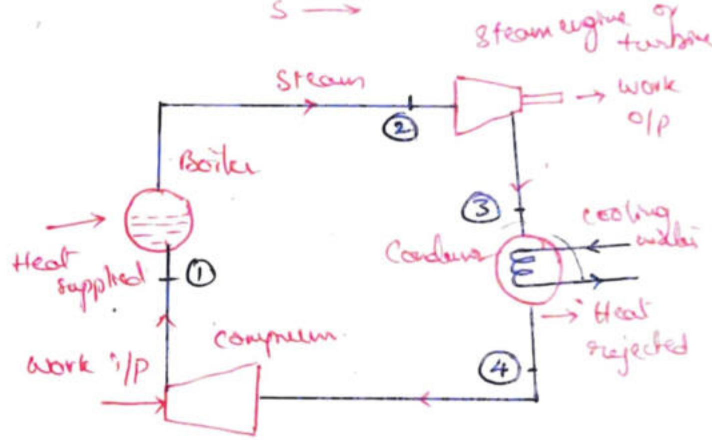
It consists of two const. pressure operations (1-2) & (3-4). and two frictionless adiabatics (2-3) & (4-1).





Carnot engine

Consider 1 kg of saturated water at pressure P_1 & abs. temp T_1 as represented by point ①.



process (1-2) :-

The saturated water at point ① is isothermally converted to dry saturated steam in a boiler & the heat is absorbed at a const. temp T_1 & pressure P_1. The dry state of steam is represented by point ②. It means $T_1 = T_2$ & $P_1 = P_2$ from P-V & T-S diagrams.

We know that heat absorbed by the saturated water during its conversion into dry steam is its latent heat of evaporation (ie., $h_{fg_1} = h_{fg_2}$) corresponding to a pressure P_1 (or) P_2 ($\because P_1 = P_2$).

Area (1-2-b-a) → heat absorbed to some scale, during the isothermal process.

∴ Heat absorbed during isothermal process (area (1-2-b-a))

$$q_{(1-2)} = \text{change in entropy} \times \text{Abs. temp} \quad [\because T_1 = T_2]$$
$$= (S_2 - S_1)T_1 = (S_2 - S_1)T_2$$

process (2-3) :-

The dry steam at point ② now expands isentropically in a steam engine or turbine. The pressure & temp falls from P_2 to P_3. & T_2 to T_3. Since no heat is supplied or rejected during this process, therefore there is no change of entropy. The isentropic exp. is represented by curve (2-3)

process (3-4) :-

The wet steam at point ③ is now isothermally condensed in a condensor & the heat is rejected at a const. temp T_3 & pressure P_3. It means that the temp T_4 & press. P_4 is equal to temp T_3 & pressure P_3 resp.

we know that area (3-4-b-a) represents heat rejected to some scale during the isothermal process.

∴ Heat rej during Isothermal Comp [area (3-4-a-b)]

$$q_{(3-4)} = (S_2 - S_1)T_3 = (S_2 - S_1)T_4$$

Process (4-1) :-

The wet steam at point ④ is finally compressed isentropically in a compressor, till it returns back to its original state (point ①). The pressure & temp rises from P_4 to P_1 & T_4 to T_1 resp. The The isentropic comp. represents by the curve (4-1). Since no heat is absorbed or rejected during this process, therefore entropy remains constant.

Workdone during cycle = Heat absorbed - heat rejected

$$= (S_2 - S_1)T_1 - (S_2 - S_1)T_3$$

$$= (S_2 - S_1)(T_1 - T_3)$$

Efficiency of Carnot Cycle,

$$\eta = \frac{WD}{\text{Heat absorbed}} = \frac{(S_2 - S_1)(T_1 - T_3)}{(S_2 - S_1)T_1} = \frac{T_1 - T_3}{T_1} = 1 - \frac{T_3}{T_1}$$

Note :-

→ Heat absorbed is at highest temp & rejected at lowest temp, the carnot cycle would give maximum possible efficiency.

→ It is impossible to make a steam engine working on Carnot cycle. The simple reason for the same is that isothermal expansion (1-2) will have to be carried out extremely slow to ensure that the steam is always at temp T_1. Similarly, the isothermal comp (3-4) will have to be carried out extremely slow. But isentropic expansion (2-3) & isentropic comp. (4-1) should be carried out as quickly as possible in order to approach ideal isentropic conditions. we know that sudden changes in the speed of an engine are not possible in actual practice.

→ It is impossible to completely eliminate friction b/w the various moving parts of the engine & also heat losses due to conduction, radiation etc.

→ Such an imaginary engine is used as an ultimate standard of comparison of all steam engines.

Limitations of Carnot cycle :-

→ It is difficult to compress a wet vapour isentropically to the saturated state as required by the process (4-1).

→ It is difficult to control the quality of the condensate coming out of condenser so that state ④ is exactly obtained.

→ The cycle is still more difficult to operate in practice with super heated steam due to the necessity of supplying the superheat at constant temp instead of const. pressure (as it is customary).

RANKINE CYCLE :-

Rankine cycle is an ideal cycle for comparing the performance of steam plants. It is the modified form of Carnot cycle, in which the condensation process (2-3) is continued until the steam

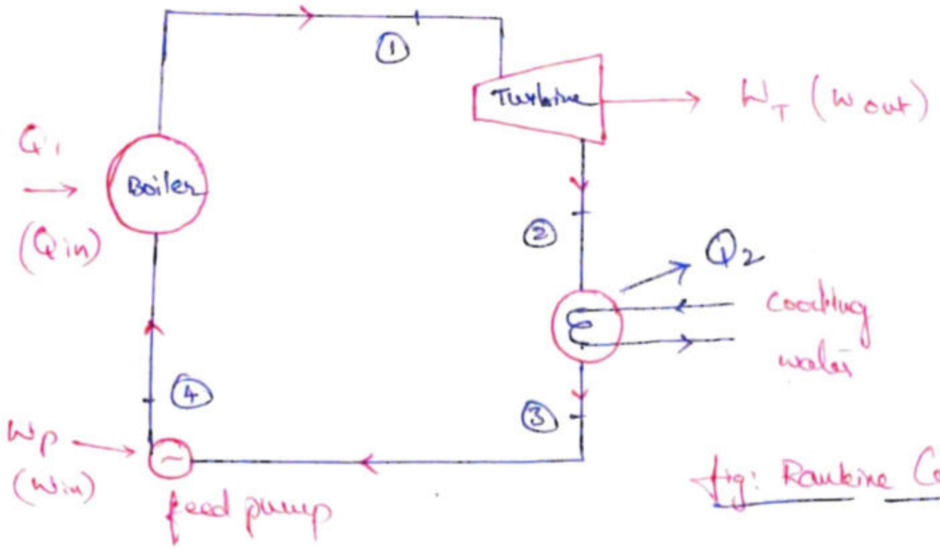
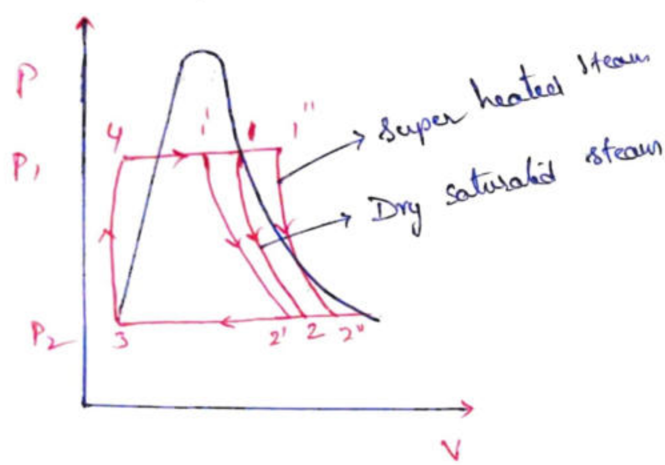
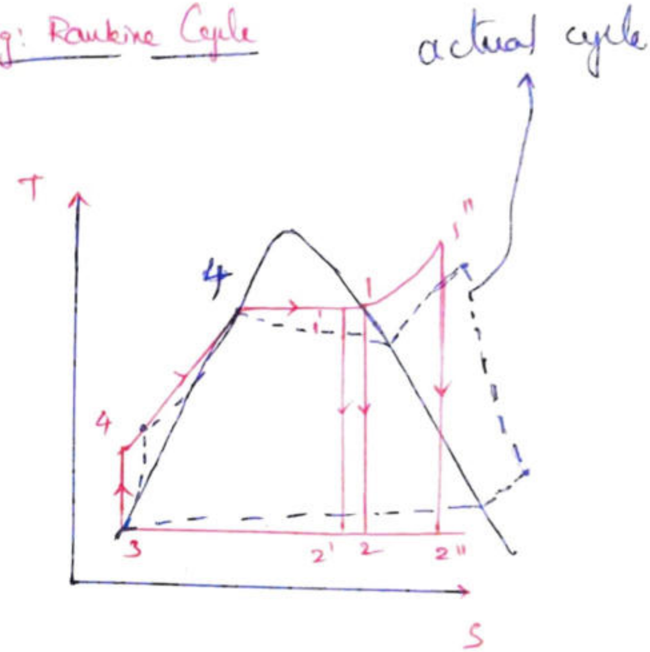
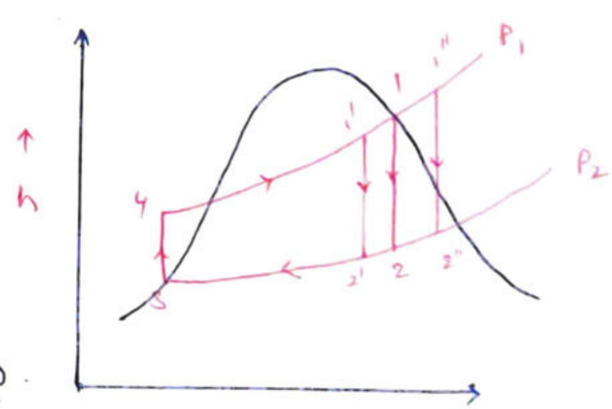


fig: Rankine Cycle







process (1-2) : Reversible adiabatic expansion in turbine

process (2-3) :- Const. pressure heat transfer in Condenser

process (3-4) :- Reversible adiabatic pumping process in feed pump.

process (4-1) :- Const. pressure heat transfer in boiler.

The above fig. shows the Rankine cycle on p-v, T-s & h-s diagrams (when saturated steam enters the turbine, the steam can be wet or superheated also.)

Consider 1 kg of fluid,

Applying SFEE to boiler, turbine, condenser & pump

(i) For boiler (as control vol)

$$h_{f4} + Q_1 = h_1 \Rightarrow Q_1 = h_1 - h_{f4}$$

(ii) For turbine.

$$h_1 = W_T + h_2 \Rightarrow W_T = h_1 - h_2$$

(iii) For condenser,

$$h_2 = Q_2 + h_{f3} \Rightarrow Q_2 = h_2 - h_{f3}$$

$Tds = dh - vdp$
$ds = 0$
$dh = v\,dp$
$\Delta h = v\,\Delta P$
$h_{f4} - h_{f3} = v_3(P_1 - P_3)$

(iv) For the feed pump. $h_{f3} + W_P = h_{f4} \Rightarrow W_P = h_{f4} - h_{f3}$

$\therefore$ Efficiency of Rankine cycle is given by

$$\eta_{Rankine} = \frac{W_{net}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{(h_1 - h_2) - (h_{f4} - h_{f3})}{(h_1 - h_{f4})}$$

The feed pump terms $(h_{f4} - h_{f3})$ being a small quantity in comparison with turbine work W_T, is usually neglected especially when boiler pressures are low.

$$\therefore \eta_{Rankine} = \frac{h_1 - h_2}{h_1 - h_{f4}}$$

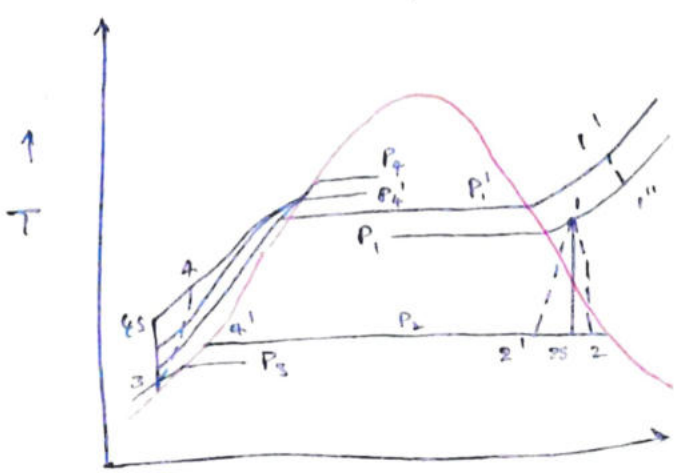
Actual vapour cyclic processes:-

The actual process or cycle differ from those of an ideal cycle. The thermal efficiency of the cycle is

$$\eta_{th} = \frac{W_{net}}{Q_1}$$

where the work & heat quantities are the measured values for the actual cycle which are different from corresponding quantities of the ideal cycle.

Actual cycle



Piping losses:-

Pressure drop due to friction & heat loss to the surroundings are the most important piping losses. States 1' & 1 represent the states of the steam leaving boiler & entering the turbine, 1'-1" represents the frictional losses and 1"-1 shows the const. pressure heat loss to the surroundings. Both the pressure drop & heat transfer reduce the availability of steam entering to the turbine.

A similar loss is the pressure drop in boiler & also in the pipe line from the pump to the boiler. Due to this pressure drop, the water entering the boiler must be pumped to a much higher pressure than the desired steam pressure leaving the boiler & this requires additional pump work.

Turbine losses:-

The losses in the turbine are those associated with frictional effects & heat loss to the surroundings.

The SFEE for turbine is $h_1 = h_2 + W_T + Q_{loss}$

$$W_T = h_1 - h_2 - Q_{loss}.$$

For the reversible adiabatic expansion, path will be 1-2s. For an ordinary real turbine the heat loss is small, & W_T is $h_1 - h_2$, with Q_2 equal to zero. Since actual turbine work is less than the reversible ideal work o/p, h_2 is greater than h_{2s}. However if there is heat loss to the surroundings, h_2 will decrease, accompanied by a decrease in entropy. If the heat loss is large, the end state of steam from the turbine may be 2'. It may so happen that the entropy increase due to frictional effects just balances the entropy decrease due to the heat loss, with result that the initial & final entropies of steam in the expansion process are equal, but the expansion is neither adiabatic nor reversible. Except for very small turbines, heat loss from turbines is generally negligible. The isentropic efficiency of the turbine is defined as follows:

$$\boxed{\eta_T = \frac{w_T}{h_1 - h_{2s}} = \frac{h_1 - h_2}{h_1 - h_{2s}}}$$

w_T = Actual turbine work

$(h_1 - h_{2s})$ = isentropic enthalpy drop in the turbine.

Pump losses:-

The losses in the pump are similar to those of the turbine, and are primarily due to the irreversibilities associated with fluid friction. Heat transfer is usually negligible. The pump η is defined as

$$\boxed{\eta_P = \frac{h_{4s} - h_3}{W_P} = \frac{h_{4s} - h_3}{h_4 - h_3}}$$

w_P = actual pump work.

Condenser losses:-

The losses in the condenser are usually small. These include the loss of pressure & cooling of condensate below the saturation temperature.

Performance Criteria for thermodynamic Cycles:

The following terms, in addition to the efficiency, are commonly used for the comparison of performance of TD vapour cycles.

(1) Efficiency ratio: It is also known as relative efficiency. It is defined as the ratio of thermal η (or actual cycle η) to Rankine η (or ideal cycle η).

$$\text{Efficiency ratio} = \frac{\eta_{thermal}}{\eta_{Ran}}$$

$$\eta_{thermal} = \frac{\text{Heat equivalent to one kilowatt hour (kWh)}}{\text{Total heat supplied to the steam per kWh}} = \frac{3600 \times P}{m_s(h_1 - h_{f_4})}$$

m_s = mass of steam supplied in kg/h

P - Power developed in kW.

(2) Work ratio :- It is defined as the ratio of net work o/p to the gross (engine or turbine) output.

$$\text{Work ratio} = \frac{\text{Net work o/p}}{\text{Gross o/p}} = \frac{\text{Turbine work} - \text{Compressor Work}}{\text{Turbine Work}}$$

Carnot cycle have high ideal η_{thermo} & it has low work ratio. In order to have better performance of the plant, both efficiency ratio & work ratio are important criteria. It is desirable to have the value of work ratio almost unity. The higher value of work ratio also means a smaller size of the plant.

(3) Specific steam Consumption :- (steam rate or Sp. rate of flow of steam)

It is defined as the mass of steam that must be supplied to a steam engine or turbine in order to develop a unit amount of work or power output. The amount of work or power o/p is usually expressed in Kwh. (1 kwh = 3600 KJ)

$$\text{Sp. steam Consumption} = \frac{1\,\text{Kwh}}{w} = \frac{3600}{w} = \frac{3600}{h_1 - h_2}\ \text{kg/kWh}$$

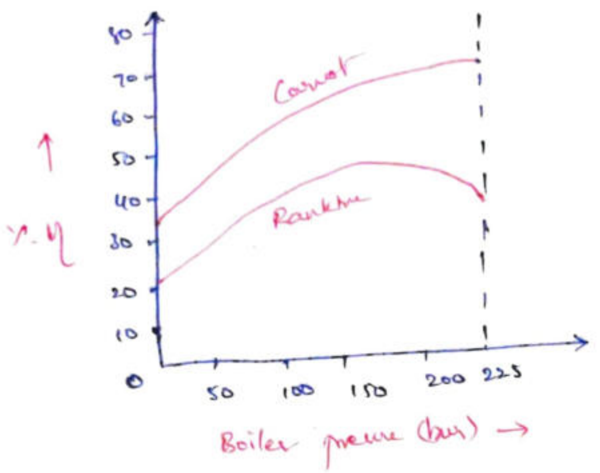
$$w = \text{net workdone } \cancel{\text{of power}} \text{ o/p} = (h_1 - h_2)\ \text{KJ/kg}$$

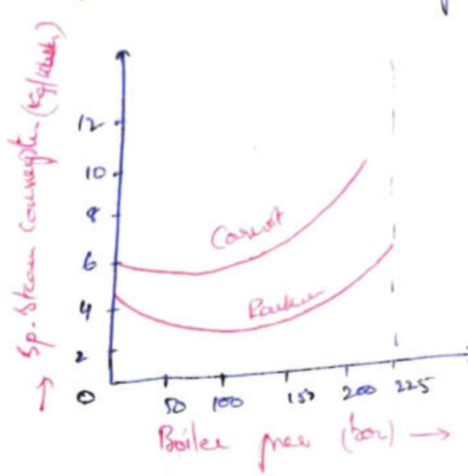
Comparison b/w Rankine & Carnot Cycle :-

⇒ B/w the same temp limits Rankine cycle provides a higher specific work o/p than Carnot cycle, Consequently Rankine cycle requires a smaller steam flow rate resulting in smaller size plant for a given power o/p. However, Rankine cycle calls for higher rates heat transfer in boiler & condenser.

⇒ Since in Rankine cycle only part of heat is supplied isothermally at const higher temp T_1, therefore its η is lower than that of Carnot cycle. The η of Rankine Cycle will approach that of Carnot cycle more nearly if the super heated temp rise is reduced.

⇒ The advantage of using pump to feed liquid to the boiler instead of compressing a wet vapour is obvious that the work for compression is very large compared to the pump.





1) In a carnot cycle, heat is supplied at 350°C & is rejected at 25°C. The working fluid is water, which while receiving heat, evaporates from liquid at 350°C to steam at 350°C. From the ST, the entropy change for this process is 1.438 KJ/kg K. If the cycle operates on stationary mass of 1 kg of water, find the heat supplied, work done & heat rejected per cycle. What is the pressure of water during heat reception?

Sol: Given: $T_1 = 350°C = 623\ K$

$T_3 = 25°C = 298\ K$

$(S_2 - S_1) = 1.438\ KJ/kg\ K$

Heat supp per cycle $= (S_2 - S_1)T_1$

$= 1.438 \times 623 = 895.87$ KJ/kg. → ①

work done per cycle $= (S_2 - S_1)(T_1 - T_3) = 1.438(623 - 298) = 467.35$ KJ/kg

Heat rejected / cycle $: (S_2 - S_1)T_3 = 1.438 \times 298 = 428.52$ KJ/kg

Pressure of water during heat reception is the formation pressure of steam corresponding to 350°C. From ST, corresponding to 350°C, the pressure is 165.35 bar.

$\boxed{\text{interpolation} \to \dfrac{y - y_1}{y_2 - y_1} = \dfrac{x_0 - x_1}{x_2 - x_1}}$

$x_1 = 360$, $y_1 =$
$x = 365$, $y_0 = ?$
$x_2 = 400$, $y_2 =$

(2) In a steam power cycle, the steam supply is at 15 bar and dry saturated. The condenser pressure is 0.4 bar. Calculate the Carnot & Rankine efficiencies of the cycle. Neglect pump work.

Sol: Steam supply pressure $P_1 = 15$ bar, $x_1 = 1$

Condenser pressure, $P_2 = 0.4$ bar

From ST,

At 15 bar, $t_s = 198.3^\circ c$, $h_g = 2789.9$ KJ/kg; $S_g = 6.4406$ KJ/kg k

At 0.4 bar; $t_s = 75.9^\circ c$; $h_f = 317.7$ KJ/kg, $h_{fg} = 2319.2$ KJ/kg.

$S_{f_2} = 1.0261$ KJ/kg k ; $S_{fg_2} = 6.6448$ KJ/kg K

$T_1 = 198.3 + 273 = 471.3k$; $T_2 = 75.9 + 273 = 348.9$ k

$$\eta_{carnot} = \frac{T_1 - T_2}{T_1} = \cancel{0.259} = \cancel{25.9}\% \quad 61.7$$

$$\eta_R = \frac{\text{Adiabatic or isentropic heat drop}}{\text{Heat supplied}} = \frac{h_1 - h_2}{h_1 - h_{f_4}}$$

where $h_2 = h_{f_2} + x_2 h_{fg_2} = 317.7 + x_2 \times 2319.2$

As the steam expands isentropically.

$$S_1 = S_2$$

$$6.4406 = 1.0261 + x_2 \times 6.6448$$

$$x_2 = 0.815$$

$$\therefore h_2 = 2207.8 \text{ KJ/kg}$$

$w_p = h_{f_4} - h_{f_3}$
$= v_f (P_1 - P_3) \times 100$
$= 0.001027(15 - 0.4) \times 100$
$w_p = 1.4994$
$h_{f_4} = 317.7 + 1.4994$

$$\therefore \eta_R = \frac{2789.9 - 2207.8}{2789.9 - \cancel{317.7}} = \cancel{23.54}\% \quad 29.9$$

844.6

(3) In a steam turbine steam at 20 bar, 360°c is expanded to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. Assume ideal processes, find per kg of steam the net work & cycle efficiency

Sol:

Boiler pressure $P_1 = 20$ bar $(360^\circ C)$

Condenser " $P_2 = 0.08$ bar

From ST,

At 20 bar, $h_1 = 3159.3$ KJ/kg ; $S_1 = 6.9917$ KJ/kg K

At 0.08 bar, $h_3 = h_{f(2)} = 173.88$ KJ/kg.

$S_3 = S_{f(2)} = 0.5926$ KJ/kg K

$h_{fg(2)} = 2403.1$ KJ/kg $\quad S_{g(2)} = 8.2287$ KJ/kg K

$v_{f(2)} = 0.001008$ m³/kg $\quad \therefore S_{fg(2)} = 7.6361$ KJ/kg K

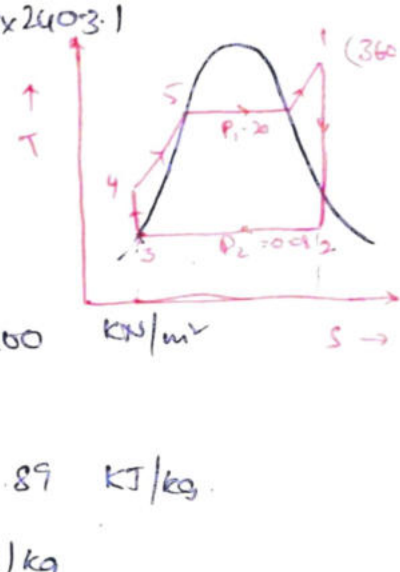
Now, $S_1 = S_2$

$6.9917 = S_{f(2)} + x_2 S_{fg(2)} = 0.5926 + x_2 \times 7.6361$

$x_2 = 0.838$

$h_2 = h_{f2} + x_2 h_{fg2} = 173.88 + 0.838 \times 2403.1$

$h_2 = 2187.68$ KJ/kg.



Net work $W_{net} = W_T - W_P$

$W_P = h_{f4} - h_{f3} \; (= h_{f3})$

$= v_{f2}(P_1 - P_2) = 0.001008 \times (20 - 0.08) \times 100$ KN/m²

$W_P = 2.008$ KJ/kg.

$h_{f4} = 2.008 + h_{f2} = 2.008 + 173.88 = 175.89$ KJ/kg.

$W_T = h_1 - h_2 = 3159.3 - 2187.68 = 971.62$ KJ/kg

$W_{net} = 971.62 - 2.008 = 969.61$ KJ/kg

Cycle efficiency η_{cycle}:

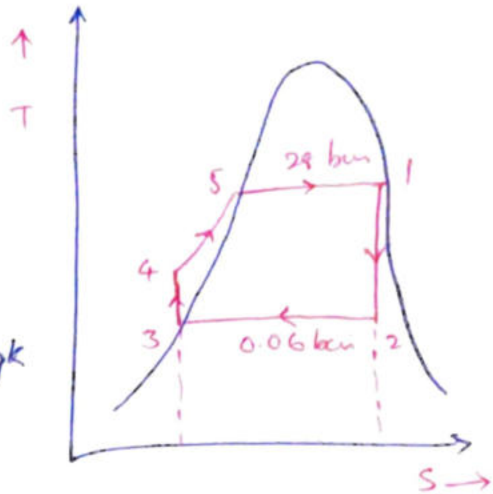
$Q_1 = h_1 - h_{f4} = 3159.3 - 175.89 = 2983.41$ KJ/kg

$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{269.61}{2983.41} = 32.5\ \%$

(4) A simple Rankine cycle works b/w pressures 28 bar & 0.06 bar the initial condition of steam being dry saturated. Calculate the

cycle efficiency, work ratio & specific steam consumption.

Sol: From ST,



At 28 bar. $h_1 = 2802$ KJ/kg.

$S_1 = 6.2104$ KJ/kg k

At 0.06 bar, $h_{f_2} = h_{f_3} = 151.5$ KJ/kg

$h_{fg_2} = 2415.9$ KJ/kg

$s_{f_2} = 0.521$ KJ/kgk, $s_{fg_2} = 7.809$ KJ/kgk

$v_f = 0.001$ m³/kg.

Considering turbine process (1-2) we have:

$$S_1 = S_2$$

$$6.2104 = s_{f_2} + x_2 \, s_{fg_2} = 0.521 + x_2(7.809)$$

$$x_2 = \frac{6.2104 - 0.521}{7.809} = 0.728$$

$$h_2 = h_{f_2} + x_2 \, h_{fg_2} = 151.5 + 0.728 \times 2415.9 = 1910.27 \text{ KJ/kg}$$

$$W_T = h_1 - h_2 = 2802 - 1910.27 = 891.73 \text{ KJ/kg}.$$

$$W_P = h_{f_4} - h_{f_3} = v_f(P_1 - P_2) = \frac{0.001(28 - 0.06) \times 10^5}{1000} = 2.79 \text{ KJ/kg}$$

$$[\because h_{f_4} = h_{f_3} + 2.79 = 151.5 + 2.79 = 154.29]$$

$$\therefore W_{net} = W_T - W_P = 891.73 - 2.79 = 888.94 \text{ KJ/kg}.$$

$$\text{cycle efficiency} = \frac{W_{net}}{Q_1} = \frac{888.94}{h_1 - h_{f_4}} = \frac{888.94}{2802 - 154.29} = \underline{33.57\%}$$

$$\text{work ratio} = \frac{W_{net}}{W_T} = \frac{888.94}{891.73} = \underline{0.997}$$

$$\text{Sp. steam Consumption} = \frac{3600}{W_{net}} = \frac{3600}{888.94} = \underline{4.049 \text{ kg/kwh}}.$$

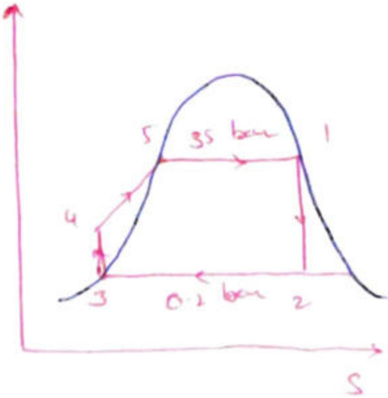
(5) In a Rankine cycle, the steam at inlet to the turbine is saturated at a pressure of 35 bar & exhaust pressure is 0.2 bar. Determine (i) pump work (ii) Turbine work (iii) $\eta_{Rankine}$ (iv) Condenser heat flow (v) Dryness at the end of expansion. Assume flow rate of 9.5 kg/s.

Sol:-

$P_1 = 35$ bar $\Rightarrow x = 1$

$P_2 = 0.2$ bar

$m = 9.5$ kg/s.



From ST,

At 35 bar, $h_1 = h_{g_1} = 2802$ KJ/kg

$s_{g_1} = 6.1228$ KJ/kg K

At 0.2 bar, $h_{f_3} = 251.5$ KJ/kg, $h_{fg} = 2358.4$ kJ/kg

$v_f = 0.00107$ m³/kg $s_f = 0.8321$ KJ/kg K ; $S_{fg} = 7.0773$ KJ/kg K

(i) pump work $= (P_4 - P_3) v_f = \frac{(35 - 0.2) \times 10^5}{1000} \times 0.001017 = 3.54$ KJ/kg.

Also pump work $= h_{f_4} - h_{f_3}$

$3.54 = h_{f_4} - 251.5 \Rightarrow h_{f_4} = 255.04$ KJ/kg

Now power required to drive the pump $= 9.5 \times 3.54$ KJ/s

$= 33.63$ KW.

(ii) Turbine work

$s_1 = s_2 = s_{f_2} + x_2 \times s_{fg_2}$

$6.1228 = 0.8321 + x_2 \times 7.0773 \Rightarrow x_2 = 0.747$

$h_2 = h_{f_2} + x_2 h_{fg_2} = 251.5 + 0.747 \times 2358.4 = 2013$ KJ/kg

Turbine work $= m(h_1 - h_2) = 9.5(2802 - 2013) = 7495.5$ KW.

It may be noted that pump work (33.63 KW) is very small as compared to the turbine work (7495.5 kW).

(iii) Rankine η. $\eta_R = \frac{h_1 - h_2}{h_1 - h_{f_4}} = \frac{2802 - 2013}{2802 - 255.5} = ~~30.93~~$ % $= 31$ %.

(iv) The condenser heat flow $= m(h_2 - h_{f_3}) = 9.5(2013 - 251.5)$

$= 16734.25$ KW

(v) The dryness at the end of expansion x_2

$x_2 = 0.747$ or 74.7 %.

(6) Calculate the fuel oil consumption required in a industrial steam plant to generate 5000 kW at the turbine shaft. The calorific value of fuel is 40000 KJ/kg & η_R is 50%. Assume appropriate values for isentropic turbine efficiency, boiler heat transfer efficiency & combustion efficiency.

Sol: Power to be generated $P = 5000$ KW

$$CV = 40000 \text{ KJ/kg}$$

$$\eta_R = 50\%$$

fuel consumption m_f :

Assume; $\eta_{turbine} = 90\%$; $\eta_{heat\ transfer} = 85\%$; $\eta_{combustn} = 98\%$

$$\eta_R = \frac{\text{shaft power} / \eta_t}{m_f \times CV \times \eta_{HT} \times \eta_{com}} \Rightarrow \frac{5000/0.9}{m_f \times 40000 \times 0.85 \times 0.98} = 0.5$$

$$\therefore m_f = 0.3335 \text{ kg/s or } 1200.6 \text{ kg/h}$$

(7) Steam at 20 bar, 360°C is expanded in a steam turbine to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. (a) Assuming ideal processes, find per kg of steam the net work and cycle efficiency. (b) If the turbine & pump have each 80% efficiency, find the percentage reduction in the network & cycle efficiency.

Sol: From ST, at 20 bar

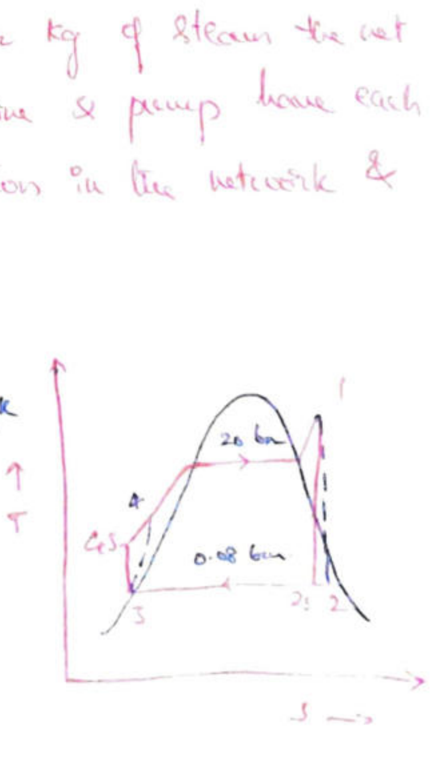
$h_1 = 3159.3$ KJ/kg $\quad s_1 = 6.9917$ KJ/kgK

At 0.08 bar

$h_3 = h_{f_2} = 173.88$ $\quad s_{g_2} = 8.2287$

$s_3 = s_{f_2} = 0.5926$ $\quad s_{fg_2} = 7.6361$

$h_{fg_2} = 2403.1$ $\quad v_{f_2} = 0.001008$ m³/kg



Now $s_1 = s_2 = 6.9917 = s_{f_2} + x_2 s_{fg_2}$

$$6.9917 = 0.5926 + x_2 (7.6361)$$

$$x_2 = 0.838$$

$\therefore h_{2s} = h_{f_2} + x_2 h_{fg_2} = 173.88 + 0.838 \times 2403.1 = 2187.68$ KJ/kg

(a) $W_p = h_{4s} - h_3 = v_{f_2} (P_1 - P_2) = 0.001008 (20 - 0.08) \times 100$

$$w_p = 2.008 \text{ KJ/kg.}$$

$\therefore w_p = h_{4s} - h_3 \Rightarrow 2.008 = h_{4s} - 173.88$

$$h_{4s} = 175.89 \text{ KJ/kg}$$

$W_T = h_1 - h_{2s} = 3159.3 - 2187.68 = 971.62$ KJ/kg

$W_{net} = W_T - W_p = 971.62 - 2.008 = 969.61$ KJ/kg.

$Q_1 = h_1 - h_{4s} = 3159.3 - 175.89 = 2983.41$ KJ/kg.

$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{969.61}{2983.41} = 0.325 = 32.5\%$$

(b) If $\eta_p = 80\%$ & $\eta_T = 80\%$.

$$w_p = \frac{2.008}{0.8} = 2.51 \text{ KJ/kg}$$

$W_T = 0.8 \times 971.62 = 777.3$ KJ/kg.

$W_{net} = W_T - W_p = 774.8$ KJ/kg.

$\therefore$ % reduction in work o/p $= \dfrac{969.61 - 774.8}{969.61} \times 100 = 20.1\%$

$h_{4s} = 173.88 + 2.51 = 176.39$ KJ/kg

$Q_1 = 3159.3 - 176.39 = 2982.91$ KJ/kg

$$\eta_{cycle} = \frac{774.8}{2982.91} = 0.2597 = 25.97\%$$

$w_p = h_{f4} - h_{f3}$

$h_{f4} = w_p + h_{f3}$

% reduction in cycle efficiency

$$= \frac{0.325 - 0.2597}{0.325} \times 100$$

$$= 20.1\% \quad \text{Ans}$$

Modified Rankine cycle:-

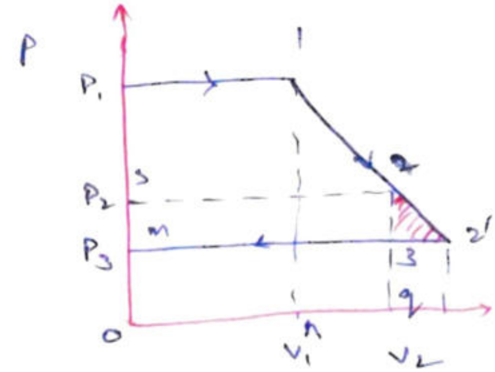
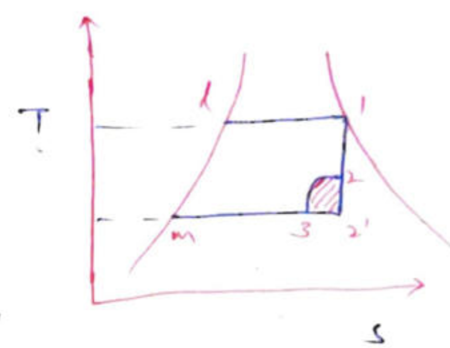


Fig. shows modified Rankine cycle (neglecting pump work)
It will be noted that p-v diagram is very narrow at the toe i.e., point 2' & the work obtained near toe is very small. In fact this work is too inadequate to overcome friction even. ∴ The adiabatic is terminated at 2, the pressure drop decreases suddenly whilst the volume remains constant. This operation is represented by line 2-3. By thus doing the stroke length is reduced, in other words the cylinder dimensions reduce but at the expense of small loss of work (area 2-3-2') which however, is negligibly small.



The WD during the modified Rankine cycle can be calculated by

let $P_1 V_1$, u_1 & h_1 correspond to initial condition of steam at '1'

Similarly P_2, V_2, u_2 & h_2

P_3, h_3

∴ WD during the cycle / kg of steam = area $l-1-2-3-m$

$= \text{area}(0-l-1-n) + \text{area}(1-2-q-n) - \text{area}(0-m-3-q)$

$= P_1V_1 + (u_1 - u_2) - P_2V_2$

Heat supplied $= h_1 - h_{f_3}$

The modified Rankine efficiency $= \dfrac{WD}{HS}$

$= \dfrac{P_1V_1 + (u_1 - u_2) - P_3V_3}{h_1 - h_{f3}}$

Alternative method for finding mod. R. η.

WD / kg of steam = area (l-1-2-3-m)

= area (l-1-2-S) + area (S-2-3-m)

$= (h_1 - h_2) + (P_2 - P_3) v_2$

Heat Supp = $h_1 - h_{f3}$

$$\eta_{mod\,R} = \frac{WD}{HS} = \frac{(h_1 - h_2) + (P_2 - P_3)v_2}{h_1 - h_{f3}}.$$

Note: Mod. Rankine cycle is used for reciprocating steam engine. becoz stroke length & hence cylinder size is reduced with the sacrifice of practically a quite negligible amount of work.

(8) Steam at a pressure of 15 bar & 300°C is delivered to the throttle of an engine. The steam expands to 2 bar when release occurs. The steam exhaust takes place at 1.1 bar. A performance test gave the result of the sp. steam consumption of 12.8 kg/kwh & a η_{mech} of 80%. Determine

(i) Ideal work of the mod. Rankine engine work. per kg

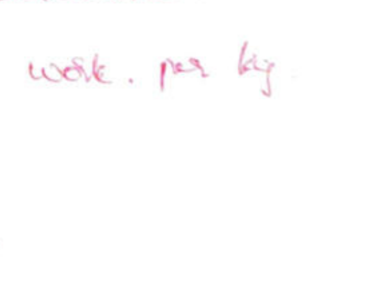
(ii) $\eta_{mod.R}$ (or) ideal thermal η.

(iii) The indicated & brake work per kg.

(iv) η_{bth}

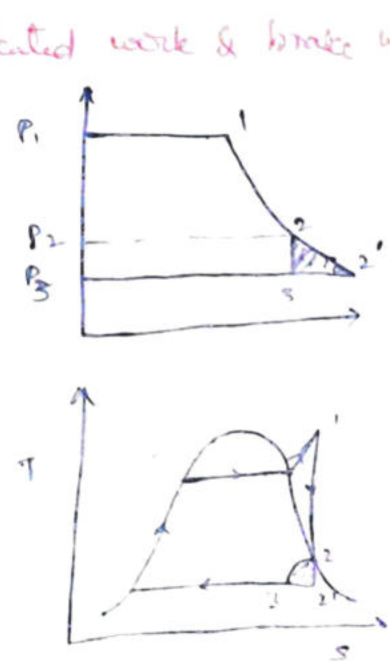
(v) The relative η on the basis of indicated work & brake work

Sol: From ST,



(1) At 15 bar, 300°C

$h_1 = 3037.6 \quad v_1 = 0.169, \; S_1 = 6.918$



(2) At 2 bar.

$t_{s2} = 120.2°C \quad h_{f2} = 504.7 \quad h_{fg2} = 2201.6$

$S_{f2} = 1.5301 \quad S_{fg2} = 5.5967 \quad v_{f2} = 0.00106$

$v_{fg2} = 0.885 \; m^3/kg$

(3) At 1.1 bar

$t_{s_3} = 102.3°C \quad h_{f_3} : 428.8 \quad h_{fg_3} = 2250.8$

$s_{f_3} = 1.333 \quad s_{fg_3} = 5.9947 \quad v_{f_3} : 0.001 \quad v_g : 1.549$

During isentropic expansion (1-2) we have,

$$S_1 = S_2$$

$$6.918 = s_{f_2} + x_2 s_{fg_2} = 1.5301 + x_2 \times 5.5967$$

$$\Rightarrow x_2 = 0.96$$

$$h_2 = h_{f_2} + x_2 h_{fg_2} = 504.7 + 0.96 \times 2201.6 = 2618.2 \text{ kJ/kg}$$

Then $v_2 = x_2 v_{g_2} + (1-x_2) v_{f_2}$

$$= 0.96 \times 0.885 + (1-0.96) 0.00106 = 0.849 \text{ m}^3/\text{kg}.$$

(i) Ideal work / kg : $W = (h_1 - h_2) + (P_2 - P_3) V_2$

$$= (3037.6 - 2618.2) + (2 - 1.1) 10^2 \times 0.849$$

$$= 495.8 \text{ kJ/kg}.$$

(ii) $\eta_R = \dfrac{WD}{HS} = \dfrac{495.8}{h_1 - h_{f_3}} = \dfrac{495.8}{3037.6 - 428.8} = 19\%.$

(iii) Indicated & Brake ~~power~~ work per kg.

Indicated work, $W_{ind} = \dfrac{IP}{m} = \dfrac{1 \times 3600}{12.8} = 281.25 \text{ kJ/kg}.$

Brake work $W_{bra} = \dfrac{BP}{m} = \dfrac{\eta_{mech} \times IP}{m} = \dfrac{0.8 \times 1 \times 3600}{12.8}$

$$= 225 \text{ kJ/kg}$$

(iv) Brake thermal η $\quad \eta_{bth} = \dfrac{W_{brak}}{h_1 - h_{f_3}} = \dfrac{225}{3037.6 - 428.8} = 8.6\%.$

(v) Relative efficiency on the basis of indicated work

$$= \frac{W_{ind}/h_1 - h_{f_3}}{W/h_1 - h_{f_3}} = \frac{W_{ind}}{W} = \frac{281.25}{495.8} = 56.7\%.$$

Relative η on the basis of brake work

$$= \frac{W_{brak}/h_1 - h_{f_3}}{W/h_1 - h_{f_3}} = \frac{W_{brake}}{W} = \frac{225}{495.8} = 45.38\%.$$

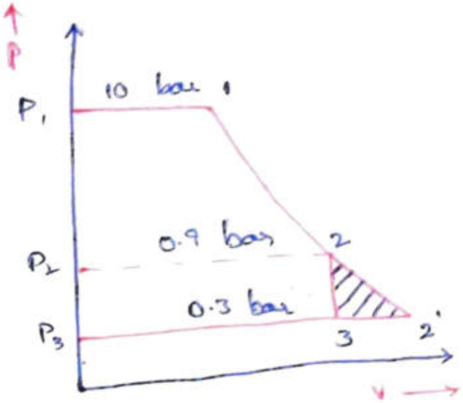
9 Superheated steam at a pressure of 10 bar & 400°C is supplied to a steam engine. Adiabatic expansion takes place to release point at 0.9 bar & it exhaust into a condenser at 0.3 bar. Neglecting clearance determine for a steam flow rate of 1.5 kg/s.

(i) Quality of steam at the end of expansion & the end of constant volume operation.

(ii) Power developed

(iii) Specific steam consumption

(iv) Modified Rankine cycle efficiency.

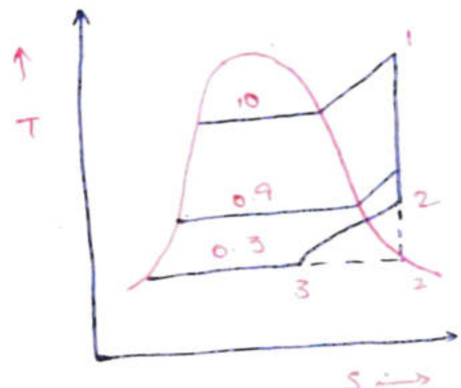


Sol:- From S.T,

① At 10 bar, 400°C

$h_1 = 3263.9$ $\quad v_1 = 0.307$

$s_1 = 7.465$



② At 0.9 bar,

$t_{s_2} = 96.7°C$ $\quad hg_2 = 2670.9$

$sg_2 = 7.3954$ $\quad vg_2 = 1.869$

③ At 0.3 bar,

$hf_3 = 289.3$ $\quad vg_3 = 5.229$

(i) Quality of steam at the end of expansion, T_{sup_2}:

For isentropic expansion 1-2, we have

$$s_1 = s_2$$
$$= sg_2 + c_p \log_e \frac{T_{sup_2}}{T_{s_2}}$$
$$7.465 = 7.3954 + 2.1 \log_e \frac{T_{sup_2}}{(96.7+273)}$$

$$T_{sup_2} = 382 \text{ K} = 109°C$$

$$h_2 = hg_2 + Cp_s (T_{sup_2} - T_{s_2}) = 2703.4 \text{ kJ/kg}.$$
$$= 2670.9 + 2.1(382 - 366.5)$$

(ii) Quality of steam at the end of constant volume operation x_3:

For calculating v_2 using the relation

$$\frac{v_{g2}}{T_{s2}} = \frac{v_2}{T_{sup2}} \quad (\text{Approximately})$$

$$\frac{1.869}{369.7} = \frac{v_2}{382} \Rightarrow v_2 = 1.931 \ m^3/kg.$$

$$v_2 = v_3 = x_3 \, v_{g3}$$

$$1.931 = x_3 \times 5.229 \Rightarrow x_3 = \frac{1.931}{5.229} = 0.37$$

(iii) Power developed, P:

$$\text{work done} = (h_1 - h_2) + (P_2 - P_3)v_2$$

$$= (3263.9 - 2703.4) + 0.75 - 0.3 \times 10^2 \times 1.931$$

$$= 647.4 \ kJ/kg.$$

$\therefore$ ~~stea~~ Power developed = Steam flow rate × work done

$$= 1 \times 647.4 = 647.4 \ kW$$

(iv) Specific steam consumption, SSC:

$$SSC = \frac{3600}{\text{Power}} = \frac{1 \times 3600}{647.4} = 5.56 \ kg/kWh$$

(v) Modified Rankine cycle efficiency, η_{MR}

$$\eta_{MR} = \frac{(h_1 - h_2) + (P_2 - P_3)v_2}{h_1 - h_{f3}} = \frac{647.4}{3263.9 - 289.3} = 21.7\%$$

Mean Temperature of Heat addition:-

In the Rankine cycle, heat is added reversibly at a constant pressure, but at infinite temperatures. If T_{m1} is the mean temp of heat addition, so that the area under 4s and 1 is equal to the area under 5 & 6, then heat added.

$Q_1 = h_1 - h_{4s} = T_{m_1}(s_1 - s_{4s})$

$\therefore T_{m_1}$ = Mean temp of heat addition.

$$T_{m_1} = \frac{h_1 - h_{4s}}{s_1 - s_{4s}}$$

Heat rejected $Q_2 = h_{2s} - h_3$

$= T_2(s_1 - s_{4s})$

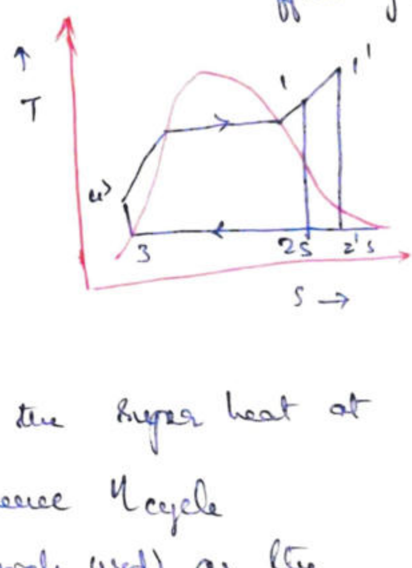
$$\therefore \eta_R = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2(s_1 - \cancel{s_{4s}})}{T_m(s_1 - \cancel{s_{4s}})} \Rightarrow \boxed{\eta_R = 1 - \frac{T_L}{T_{m_1}}}$$

where T_2 = temp of heat rejection.

(*) lowest possible or practicable temp of heat rejection is the temp of surroundings (T_0).

$$\therefore \eta_R = f(T_{m_1}) \text{ only.}$$

⇒ The higher the mean temp (T_{m_1}), the higher will be the cycle efficiency.



⇒ The effect of increasing ~~the increasing~~ the initial temp at const. pressure is shown in fig.

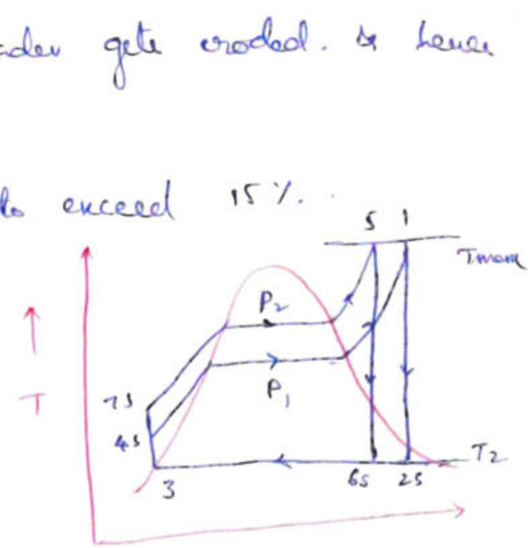
⇒ when the initial state changes from 1 to 1', T_{m_1} b/w 1 & 1' is higher than T_{m_1} b/w 4S & 1. So an increase in the super heat at constant pressure increases the T_{m_1} & hence η_{cycle}

⇒ when the max. temp is fixed (A/c materials used) as the operating system pressure at which heat is added in the boiler increases from P_1 to P_2, the T_{m_1} increases, since T_{m_1} b/w 7s & 5 is higher than that b/w 4S & 1.

⇒ But when the turbine inlet pressure increases from P_1 to P_2 the ideal expansion line shifts to the left & the moisture content at the turbine exhaust increases. (bcoz $x_{6s} < x_{2s}$).

⇒ If moisture content is large, turbine blades gets eroded. & hence life of blades decreases.

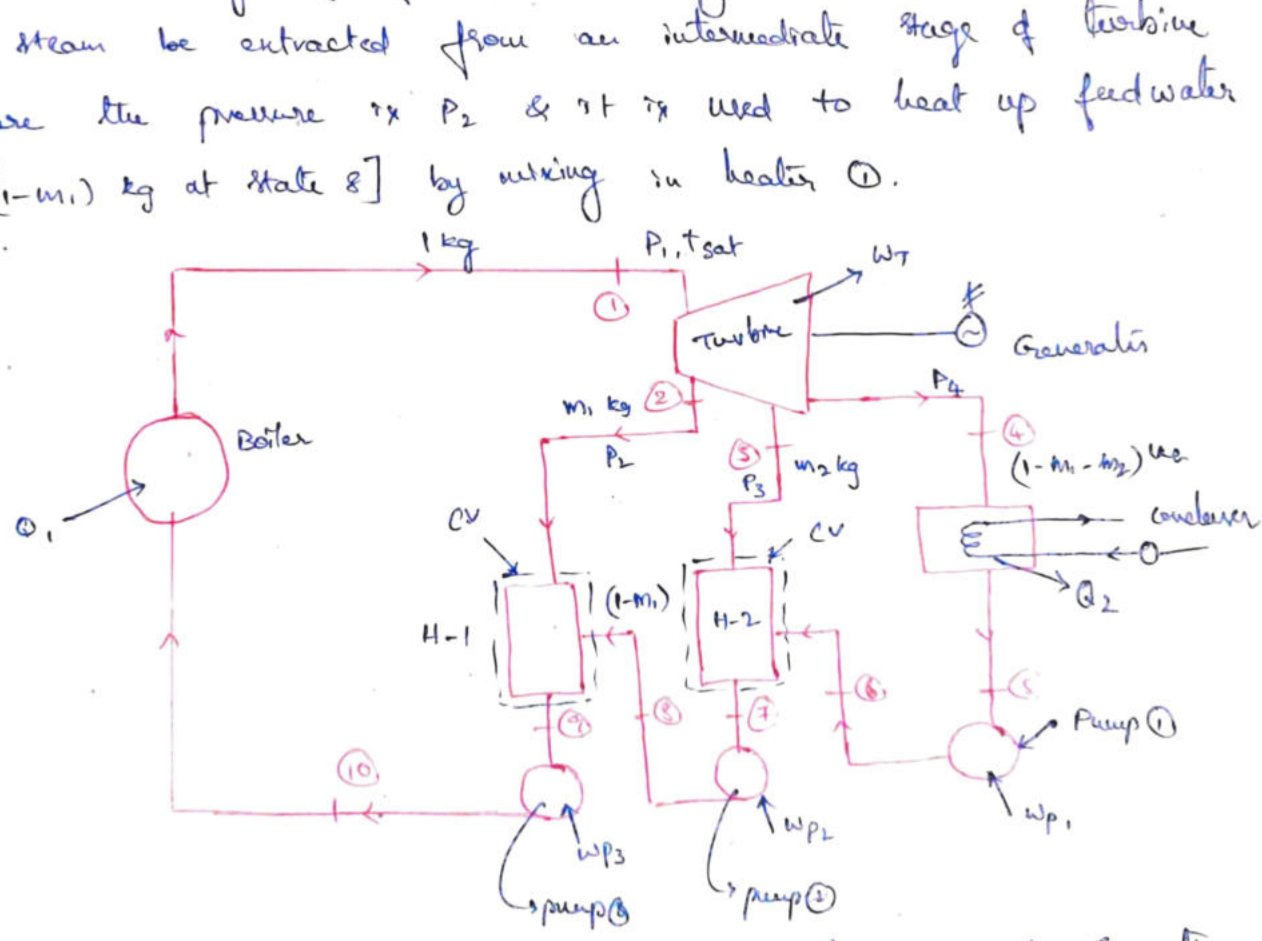
⇒ So moisture content is not allowed to exceed 15%.



REGENERATIVE CYCLE :-

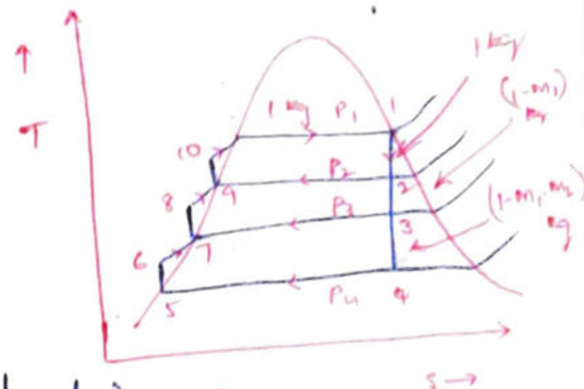
In a practical regeneration cycle, the feed water enters the boiler at a temp b/w 4 & 4' & is heated by steam extracted from intermediate stages of turbine.

For every kg of steam entering the turbine, let m_1 kg of steam be extracted from an intermediate stage of turbine where the pressure is P_2 & it is used to heat up feedwater [$(1-m_1)$ kg at state 8] by mixing in heater ①.



The remaining $(1-m_1)$ kg of steam then expands in the turbine from pressure P_2 (State ②) to pressure P_3 (State ③) when m_2 kg of steam is extracted for heating feed water in heater ②.

So $(1-m_1-m_2)$ kg of steam then expands in the remaining stages of turbine to pressure P_4, gets condensed into water in the condenser, & then pumped to heater (2), where it mixes with m_2 kg of steam extracted at press. P_3. Then $(1-m_1)$ kg of water is pumped to heater (1) where it mixes with m_1 kg of steam extracted at press. P_2. The resulting 1 kg of steam is then pumped to the boiler where heat from external source is supplied



Heat & work transfer.

$$W_T = 1(h_1-h_2)+(1-m_1)(h_2-h_3)+(1-m_1-m_2)(h_3-h_4) \rightarrow (1)$$

$$w_p = w_{p_1}+w_{p_2}+w_{p_3} \rightarrow (2)$$

$$= (1-m_1-m_2)(h_6-h_5)+(1-m_1)(h_8-h_7)+1(h_{10}-h_9) \rightarrow (3)$$

$$Q_1 = 1(h_1-h_{10}) \rightarrow (4)$$

$$Q_2 = (1-m_1-m_2)(h_4-h_5) \rightarrow (5)$$

$$\text{cycle } \eta = \frac{Q_1-Q_2}{Q_1} = \frac{w_T-w_p}{Q_1} \rightarrow (6)$$

$$\text{Steam rate} = \frac{3600}{w_T-w_p} \text{ kg/kwh} \rightarrow (7)$$

In the Rankine cycle operating at given pressures, P_1 & P_4, the heat addition would have been from state (6) to state (1). By using two stages of regenerative feed water heating, feed water enters the boiler at state (10) instead of (6), & heat addition is therefore from state (10) to (1). Therefore

$$(T_{m_1}) \text{ with regeneration} = \frac{h_1-h_{10}}{s_1-s_{10}} \rightarrow (8)$$

$$(T_{m_1}) \text{ without regeneration} = \frac{h_1-h_6}{s_1-s_6} \rightarrow (9)$$

$\therefore$ Since $(T_{m_1})_{\text{regn}} > (T_{m_1})_{\text{without regn.}}$

∴ efficiency of the regenerative cycle will be higher than that of Rankine cycle.

The energy balance for heater ① gives.

$$m_1 h_2 + (1-m_1) h_8 = 1\, h_9$$

$$m_1 = \frac{h_9 - h_8}{h_2 - h_8} \rightarrow (10)$$

$m_1 h_2 + h_8 - m_1 h_8 = h_9$

$m_1(h_2 - h_8) = h_9 - h_8$

$m_1 = \frac{h_9 - h_8}{h_2 - h_8}$

The energy balance for heater ② gives

$$m_2 h_3 + (1-m_1-m_2) h_6 = (1-m_1) h_7$$

$$m_2 = (1-m_1)\frac{h_7 - h_6}{h_3 - h_6} \rightarrow (11)$$

From eq's (10) & (11) m_1 & m_2 can be evaluated. It can also be written alternatively as

$$(1-m_1)(h_9 - h_8) = m_1(h_2 - h_9)$$

$$(1-m_1-m_2)(h_7 - h_6) = m_2(h_3 - h_7)$$

Energy gain of feed water = Energy given off by vapour in condensation

Heaters have been assumed to be adequately insulated, & there is no heat gain from, or heat loss to, the surroundings.

Advantages of Regenerative Cycle over Simple Rankine Cycle:-

⇒ The heating process in the boiler tends to become reversible.

⇒ The thermal stresses set up in the boiler are minimised. This is due to the fact that temp ranges in boiler are reduced.

⇒ The thermal η is improved becoz the avg temp of heat addition to the cycle is increased.

⇒ Heat rate is reduced.

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

⇒ Due to many extractions there is an improvement in the turbine drainage & it reduces erosion due to moisture

⇒ A small size condenser is required.

Disadvantages:

⇒ The plant becomes more complicated.

⇒ Becoz of addition of heaters greater maintenance is required.

⇒ For given power a large capacity boiler is required.

⇒ The heaters are costly & the gain in thermal efficiency is not much in comparison to the heavier costs.

Assumptions:-

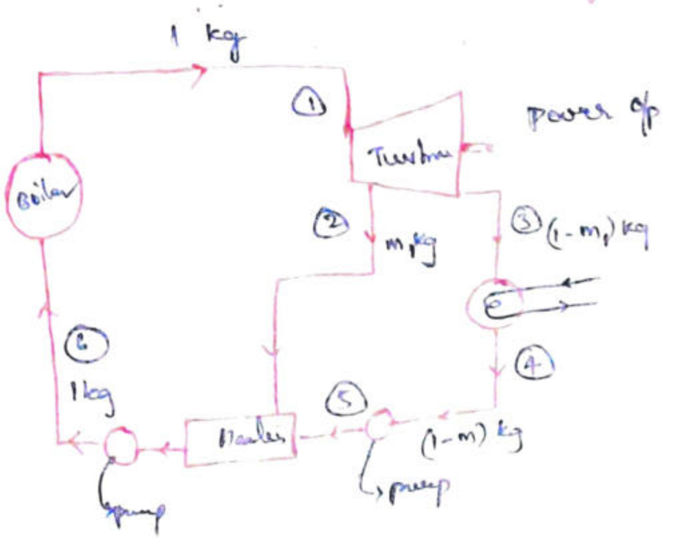
⇒ Each heater is ideal & bled steam just condenses.

⇒ The feed water is heated to saturation temp at the pressure of bled steam.

⇒ Unless otherwise stated the workdone by the pumps is considered negligible.

⇒ There is equal temp rise in all the heaters (usually 10°C to 15°C).

(10) A steam turbine is fed with steam having an enthalpy of 3100 KJ/kg. It moves out of the turbine with an enthalpy of 2100 KJ/kg. Feed heating is done at a pressure of 3.2 bar with steam enthalpy of 2500 KJ/kg. The condensate from the condenser with an enthalpy of 125 KJ/kg enters into the feed heater. The quantity of bled steam is 11200 kg/h. Find the power developed by the turbine. Assume that the water leaving the feed heater is saturated liquid at 3.2 bar & the heater is direct mixing type. Neglect pump work.



Sol:

$h_1 = 3100$ KJ/kg

$h_3 = 2100$ KJ/kg

$h_2 = 2500$ KJ/kg

$h_{f_5} = 125$ KJ/kg

$m = 11200$ kg/h

At 3.2 bar $h_{f_2} = 570.9$ KJ/kg

Energy balance for the feed heater is

$$m_1 h_2 + (1-m_1) h_{f_5} = 1 \times h_{f_2}$$

$$m_1 (2500) + (1-m_1) 125 = 1 \times 570.9$$

$x \to$ total mass
$y \to m_1 = 0.187$ kg/kg of steam
$z \to (1-m_1)$
$x = y + z \Rightarrow x = y + 11200$
$(x-y) = 11200$
$x - 0.187x = 11200$
$x(1-0.187) = 11200$
$x = \dfrac{11200}{0.813} = 13776$ kg

$$m_1 = 0.187 \text{ kg per kg of steam supplied to the turbine}$$

Steam supplied to the turbine per hour $= \dfrac{11200}{\cancel{0.226}\ \cancel{0.187}\ 0.813} = \cancel{49557.5}\ \cancel{59893}\ 13776$ kg/h

Net work developed per kg of steam $= 1(h_1 - h_2) + (1-m_1)(h_2 - h_3)$

$$= (3100 - 2500)1 + (1 - 0.187)(2500 - 2100)$$

$$= 925.2 \text{ KJ/kg.}$$

$\therefore$ Power developed by the turbine $= 925.2 \times \dfrac{13776\ \cancel{59873}}{3600}$

$$= \cancel{15392.5}\ 3540 \text{ KW}$$

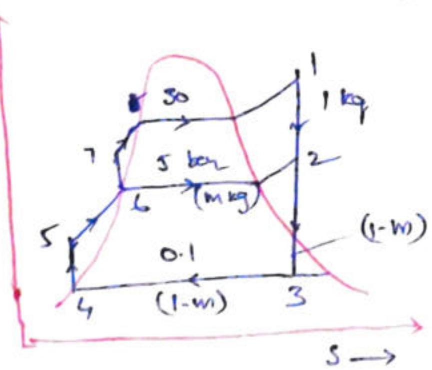
(11) In a single heater regenerative cycle the steam enters the turbine at 30 bar, 400°C & exhaust pressure is 0.1 bar. The feed water heater is a direct contact type which operates at 5 bar. Find:

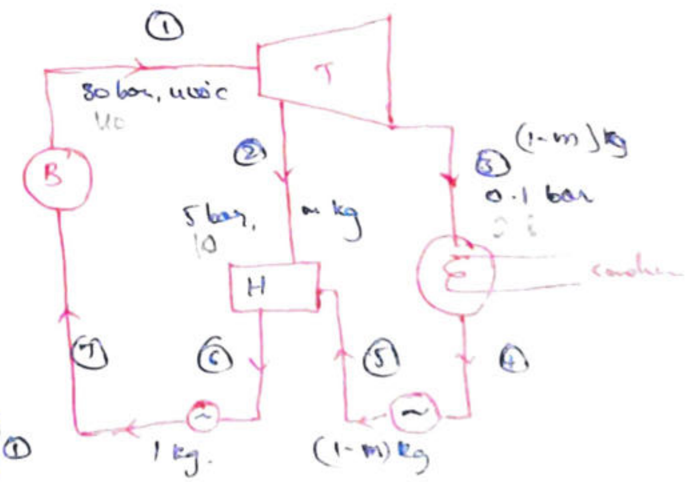
(i) The efficiency & the steam rate of the cycle.

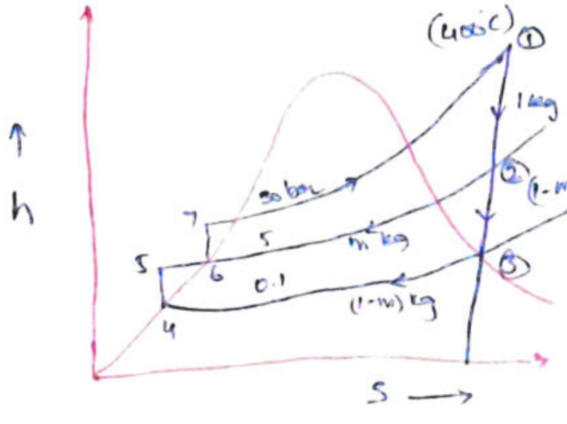
(ii) The increase in mean temp of heat addition, efficiency & steam rate as compared to the Rankine cycle (without regeneration).

Pump work may be neglected.

Sol:







From ST,

At ~~30~~ 40 bar, 400°C, $h_1 = 3230.9$, $s_1 = 6.921 = s_2 = s_3$

At ~~5~~ 10 bar, $s_f = 1.8604$, $s_{g_2} = 6.8192$, $h_{f_2} = 640.1$

Since $s_2 > s_g$ the state ② must be in the superheated region.

From the ST, for superheated steam

$t_2 = 172°C$; $h_2 = 2796$ KJ/kg ? ~~3277.1~~

At 0.1 bar; $s_{f_3} = 0.649$, $s_{fg_3} = 7.501$, $h_{f_3} = 191.8$, $h_{fg_3} = 2392.8$

$$s_2 = s_3$$

$$6.921 = s_{f_3} + x_3 s_{fg_3}$$

$$x_3 = 0.836$$

$$\therefore h_3 = h_{f_3} + x_3 h_{fg_3} = 2192.2 \text{ KJ/kg.}$$

Since pump work is neglected,

$$h_{f_4} = h_{f_3} = 191.8 \text{ KJ/kg} = h_{f_5}$$

(at 5 bar) $h_{f_6} = 640.1 = h_{f_7}$

Energy balance for heater given,

$$m(h_2) + (1-m)\,h_{f_5} = 1 \times h_{f_6}$$

$$m h_2 - m h_{f_5} + h_{f_5} = h_{f_6}$$

$$m = \frac{h_{f_6} - h_{f_5}}{h_2 - h_{f_5}} = \frac{640.1 - 191.8}{2796 - 191.8} = \cancel{0.172}\ 0.145 \text{ kg}$$

$\therefore$ Turbine work $W_T = 1(h_1 - h_2) + (1-m)(h_2 - h_3)$

$$= (3230.9 - 2796) + (1 - 0.172)(2796 - 2192.2)$$

$$= 434.9 + 499.9 = 934.8 \text{ KJ/kg.}$$

Heat supplied, $Q_1 = h_1 - h_{f_8}$ h_{f_7}(or) $= 3230.9 - 640.1 = 2590.8$ KJ/kg

(i) Efficiency of cycle, $\eta_{cycle} = \frac{W_T}{Q_1} = \frac{934.8}{2590.8} = 36.08\ \%$.

steam rate $= \frac{3600}{934.8} = 3.85$ kg/kwh

(ii) $T_{m_1} = \frac{h_1 - h_{f_7}}{s_1 - s_{f_7}} = \frac{3230.9 - 640.1}{6.921 - \cancel{0.649}\ 1.8604} = 511.9 \text{ K} = 238.9\ °C$

T_{m_1} (without regeneration) $= \dfrac{h_1 - h_{f_4}}{s_1 - s_4} = \dfrac{3230.9 - 191.8}{6.921 - \cancel{1.8604}\ 0.649} = 484.5\ K$
$= 211.5^\circ C$

Increase in T_{m_1} due to regeneration $= 238.9 - 211.5$
$= 27.4^\circ C$

W_T (without regeneration) $= h_1 - h_3 = 3230.9 - 2192.2$
$= 1038.7$ KJ/kg

Steam rate without regeneration $= \dfrac{3600}{1038.7} = 3.46$ kg/kWh

∴ Increase in steam rate due to regeneration $= 3.85 - 3.46$
$= 0.39$ kg/kWh

η_{cycle} (without reg) $= \dfrac{h_1 - h_3}{h_1 - h_{f_4}} = \dfrac{1038.7}{3230.9 - 191.8} = 34.18\%$

Increase in cycle η due to regeneration,
$= 36.08 - 34.18 = 1.9\%$

(12) Steam at a pressure of 20 bar & 250°C enters a turbine & leaves it finally at a pressure of 0.05 bar. Steam is bled off at a pressure of 5.0, 1.5 & 0.3 bar. Assuming (i) that the condensate is heated in each heater upto the saturation temp 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 0.3 bar are cooled in a drain cooler to a condenser temp, Calculate the following.

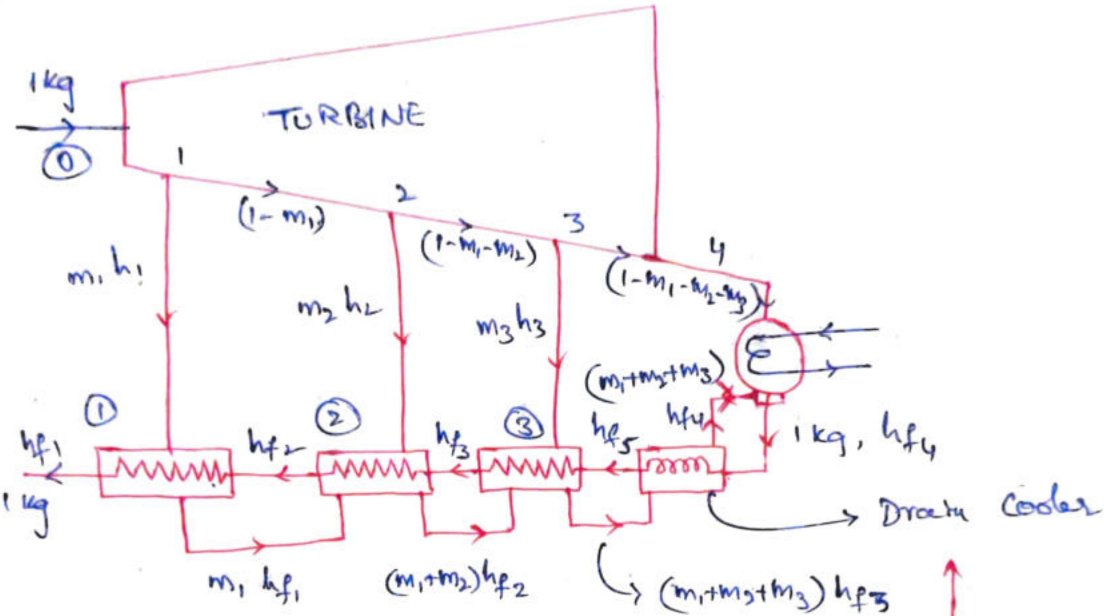
(i) Mass of bled steam for each heater/kg of steam entering the turbine
(ii) $\eta_{thermal}$ of cycle
(iii) $\eta_{thermal}$ of Rankine cycle

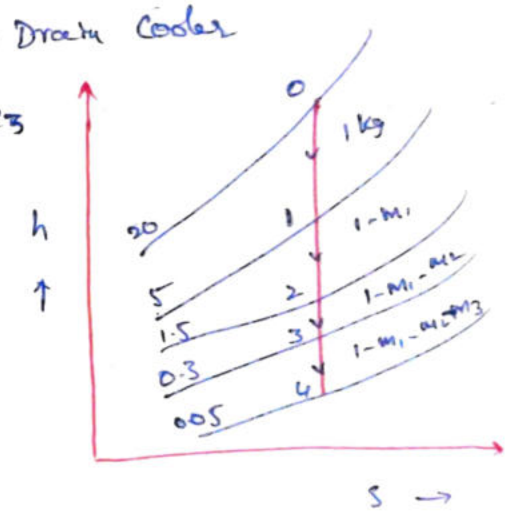
(iv) Theoretical gain due to regenerative feed heating.

(v) Steam consumption in kg/kwh with or without regenerative feed heating &

(vi) Qty of steam passing through the last stage nozzle of a 50000 kw turbine with & without regenerative feed heating.

Sol:





From Mollier chart,

$h_0 = 2905$, $h_1 = 2600$, $h_2 = 2430$

$h_3 = 2210$, $h_4 = 2000$

From ST,

At 5 bar, $h_{f1} = 640.1$

At 1.5 bar, $h_{f2} = 467.1$

At 0.3 bar $h_{f3} = 289.3$

At 0.05 bar, $h_{f4} = 137.8$

(i) Mass of bled steam for each heater/kg of steam.

Using heat balance eq,

At heater no: 1.

$$m_1 h_1 + h_{f2} = m_1 h_{f1} + h_{f1}$$

$$m_1 = \frac{h_{f1} - h_{f2}}{h_1 - h_{f1}} = \frac{640.1 - 467.1}{2600 - 640.1} = 0.088 \text{ KJ/kg of entering steam}$$

At heater 2.

$$m_2 h_2 + h_{f3} + m_1 h_{f1} = h_{f2} + (m_1+m_2) h_{f2}$$

$$\therefore m_2 = 0.0828 \text{ KJ/kg of entering steam}$$

At heater (3)

$$m_3 h_3 + h_{f5} + (m_1+m_2) h_{f2} = h_{f3} + (m_1+m_2+m_3) h_{f3} \rightarrow (1)$$

At drain cooler

$$(m_1+m_2+m_3) h_{f3} + h_{f4} = h_{f5} + (m_1+m_2+m_3) h_{f4}$$

$$h_{f5} = (m_1+m_2+m_3)(h_{f3} - h_{f4}) + h_{f4} \rightarrow \text{eq } (2)$$

Substitute eq (2) in (1)

$$\therefore m_3 h_3 + (m_1+m_2+m_3)(h_{f3} - h_{f4}) + h_{f4} + (m_1+m_2) h_{f2} = h_{f3} + (m_1+m_2+m_3) h_{f3}$$

$$m_3 = \frac{(h_{f3} - h_{f4}) - (m_1+m_2)(h_{f2} - h_{f4})}{h_3 - h_{f4}} = 0.046 \text{ KJ/kg}$$

workdone / kg (neglecting pump work)

$$= (h_0 - h_1) + (1-m_1)(h_1 - h_2) + (1-m_1-m_2)(h_2 - h_3) + (1-m_1-m_2-m_3)(h_3 - h_4)$$

$$= 806.93 \text{ KJ/kg.}$$

Heat supplied / kg $= h_0 - h_{f_1} = 2905 - 640.1 = 2264.9$ KJ/kg.

(ii) $\eta_{ther} = \frac{WD}{HS} = \frac{806.93}{2264.9} = 0.3563 = 35.63\%.$

(iii) $\eta_R = \frac{h_0 - h_4}{h_0 - h_{f4}} = \frac{2905 - 2000}{2905 - 137.8} = 32.7\%.$

(iv) Theoretical gain due to regenerative feed heating

$$= \frac{35.63 - 32.7}{35.63} = 8.22\%$$

(v) steam consumption with regenerative feed heating

$$= \frac{1 \times 3600}{WD/kg} = \frac{1 \times 3600}{806.93} = 4.46 \text{ kg/kwh}$$

Steam consumption without regenerative $= \frac{1 \times 3600}{WD/kg \text{ without regenerat.}}$

$$= \frac{1 \times 3600}{h_0 - h_4} = \frac{3600}{2905 - 2000} = 3.97 \text{ kg/kwh.}$$

(vi) Qty of steam passing through the last stage of a 50000 kw turbine with regenerative feed heating.

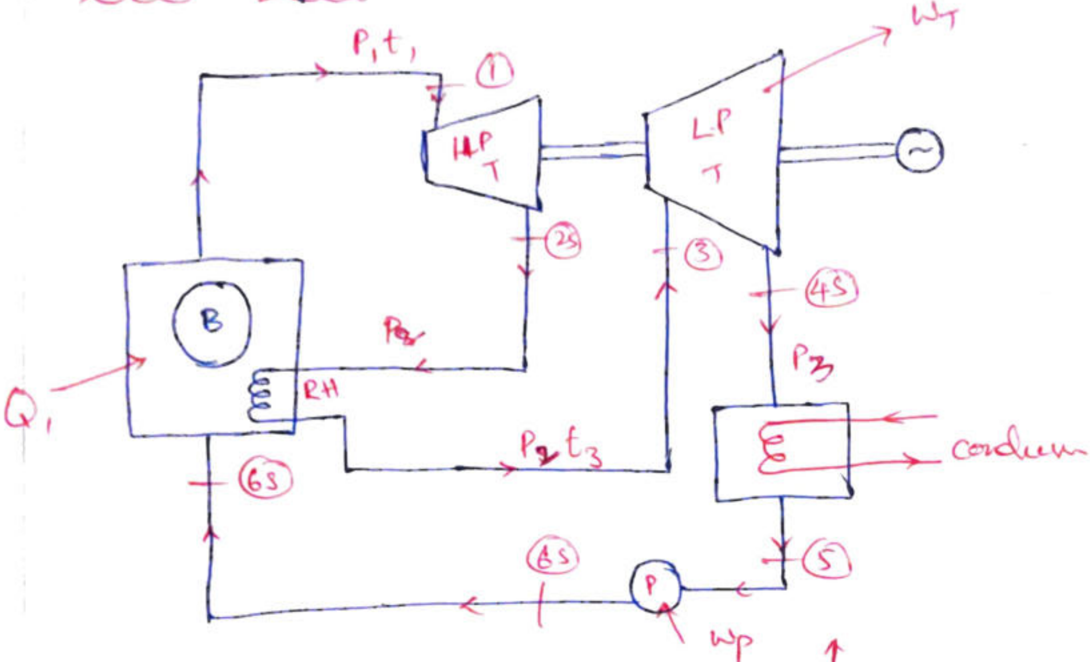
$= 4.46\,(1 - m_1 - m_2 - m_3) \times 50000$

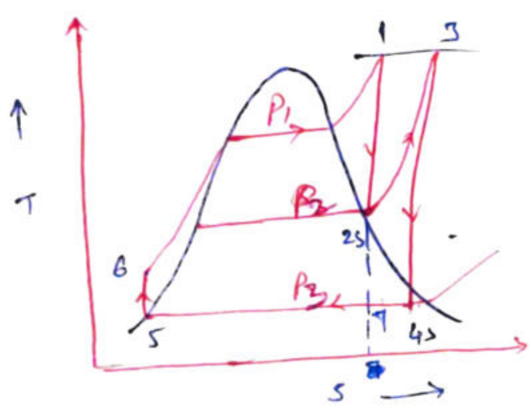
$= 174653.6$ kg/h.

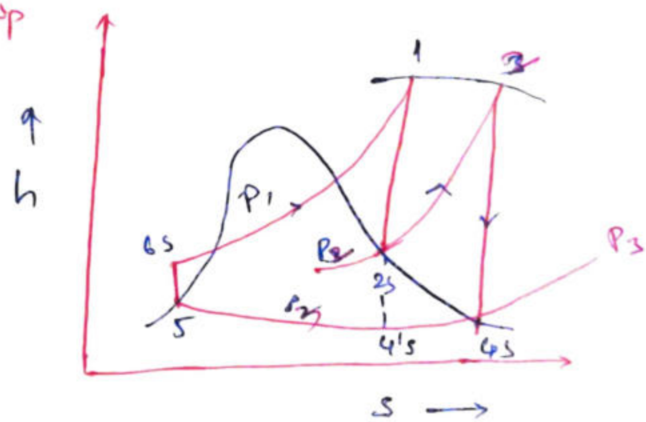
Same without regeneration arrangement

$= 3.97 \times 50000 = 198500$ kg/h

Reheat Cycle :-







→ primary object of superheating steam & supplying it to the prime movers is to avoid too much wetness at the end of expansion.

Advantages of superheated steam :-

1) Super heating reduces the initial condensation losses.

2) Use of superheated steam results in improving plant η by effecting a saving in cost of fuel

3) when a superheater is used in a boiler it helps in reducing the stack temp's by extracting heat from the flue gases before these are passed out of chimney.

Thermal efficiency with Reheating (neglecting pump work)

Heat supplied $= (h_1 - h_{f_5}) + (h_3 - h_2)$

Heat rejected $= (h_4 - h_{f_5})$

Work done by turbine $= (h_1 - h_2) + (h_3 - h_4)$

$$\eta_{th} = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f_5}) + (h_3 - h_2)}$$

If pump work $W_P = \frac{v_f(P_1 - P_b)}{1000}$

$$\eta_{th} = \frac{W_T - W_P}{Q_1}$$

Thermal η without reheating

is $\eta_{th} = \frac{h_1 - h_7}{h_1 - h_{f_5}}$. $[\because h_{f_5} = h_{f_7}]$.

Note:-

→ Reheating should be done at optimum pressure.

Advantages:-

1) There is an increased output of the turbine.
2) Erosion & corrosion problems in turbine are eliminated.
3) Improvement in η_{th}
4) Final dryness fraction of steam is improved.
5) There is an increase in the nozzle & blade efficiencies.

Disadvantages:

1) It requires more maintenance.
2) The increase in η_{th} is not applicable in comparison to the expenditure incurred in reheating.
3) Adding additional turbine increases the initial cost.

① Steam at a press. of 15 bar & 250°C is expanded through a turbine at first to a pressure of 4 bar. It is then reheated at constant pressure to the initial temp of 250°C & is finally expanded to 0.1 bar. Using Mollier chart, estimate WD/kg of steam flowing through the turbine & amount of heat supplied during the process of reheat. Compare the work out put when the expansion is direct from 15 bar to 0.1 bar without any reheat. Assume all expansion processes to be isentropic.

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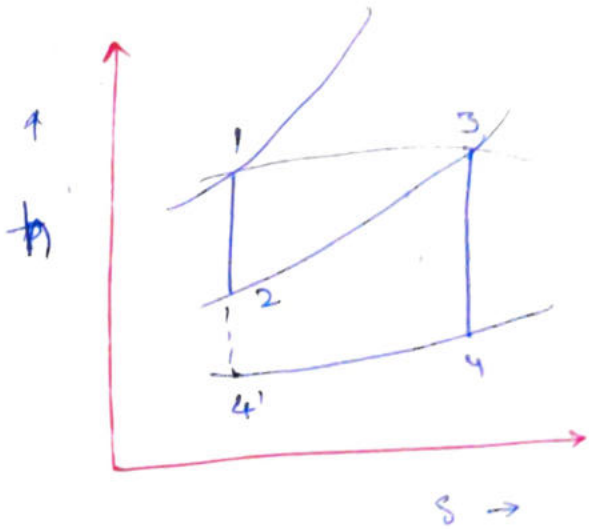
sol:- Given,

$P_1 = 15$ bar

$P_2 = 4$ bar

$P_4 = 0.1$ bar.



WD per kg of steam

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

From MD,

$h_1 = 2920$ $\quad h_2 = 2660$

$h_3 = 2960$ $\quad h_4 = 2335$

$\therefore WD = (2920 - 2660) + (2960 - 2335)$

$= 885$ KJ/kg.

Amount of heat supplied during reheating,

$h_{reheat} = h_3 - h_2 = 2960 - 2660 = 300$ KJ/kg

without reheating, $\quad$ From MD, $h_4' = 2125$

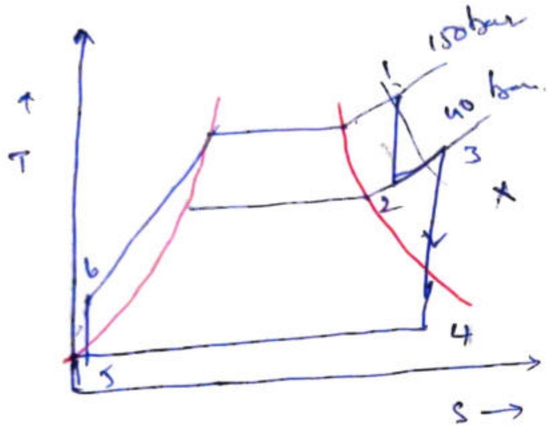
$W_1 = h_1 - h_4'$

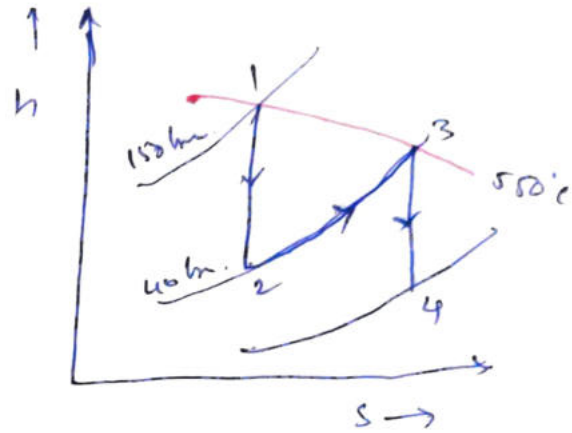
$= 2920 - 2125$

$W_1 = 795$ KJ/kg.

② A steam power plant operates on a theoretical reheat cycle. Steam at boiler at 150 bar, 550°C expands through the high pressure turbine. It is reheated at a constant pressure of 40 bar to 550°C & expands through the low pressure turbine to a condenser at 0.1 bar. Draw T-s & h-s diagram. Find (i) Quality of steam at turbine exhaust. (ii) cycle efficiency. (iii) steam rate in kg/kwh.

Sol:





From Mollier diagram (h-s diagram)

$h_1 = 3450$ kJ/kg $h_2 = 3050$ kJ/kg $h_3 = 3560$ kJ/kg

$h_4 = 2300$ kJ/kg

From ST, $h_{f4} = 191.8$ kJ/kg at ($p = 0.1$ bar)

(i) Quality of steam at turbine exhaust x_4,

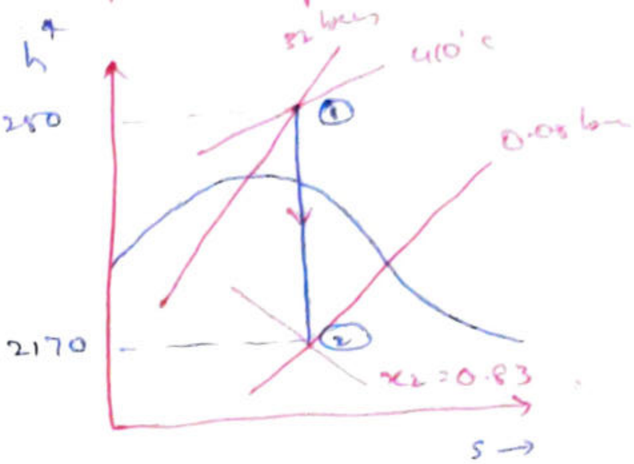
$x_4 = 0.88$ (From MD)

(ii) cycle η : $\eta_{cycle} = \dfrac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f4}) + (h_3 - h_2)} = 44.05\%$

(iii) Steam rate in kg/kwh.

$$\text{Steam rate} = \frac{3600}{(h_1 - h_2) + (h_3 - h_4)} = 2.17 \text{ kg/kwh.}$$

⑤ A turbine is supplied with steam at a pressure of ~~32~~ 40 32 bar and a temp of 410°C 450. The steam then expands isentropically to a pressure of 0.08 0.1 bar. Find the dryness fraction at the end of expansion & η_{therm} of cycle. If the steam is reheated at 5.5 bar (6) to a temp of 395°C (400°C) & then expanded isentropically to a pressure of 0.08 bar (0.1), what will be the dryness fraction & thermal η of the cycle



Sol: First case

From MD,

$h_1 = 3250$

$h_2 = 2170$

Heat drop (or workdone)

$= h_1 - h_2 = 3250 - 2170$

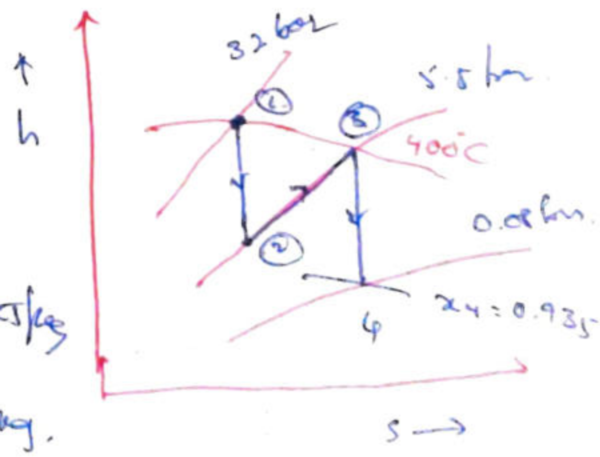
$= 1080$ kJ/kg

Heat supplied $= h_1 - h_{f_2} = 3250 - 173.9$ $\left[h_{f_2} = 173.9 \text{ at } 0.08 \text{ bar} \right]$

$= 3076.1$

$\eta_{thermal} = \frac{WD}{HS} = \frac{1080}{3076.1} = 35.1\%$

Exhaust steam condition, $x_2 = 0.83$ (from MD)



Second case:

From MD, $h_1 = 3250$

$h_3 = 3263$

$h_2 = 2807$

$h_4 = 2426$

workdone $= (h_1 - h_2) + (h_3 - h_4) = 1280$ kJ/kg

$HS = (h_1 - h_{f_4}) + (h_3 - h_2) = 3532$ kJ/kg.

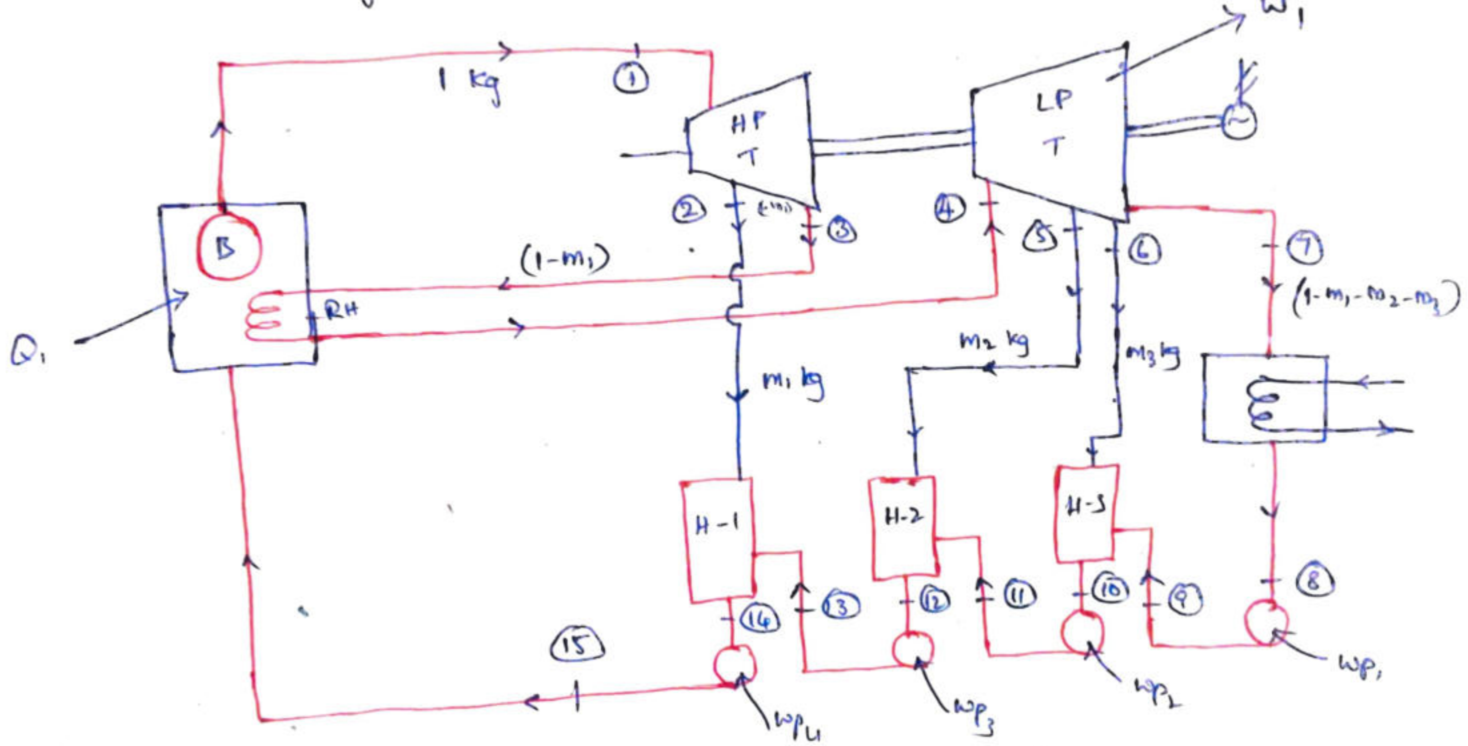
$\eta_{ther} = \frac{WD}{HS} = \frac{1280}{3532} = 36.2\%$

Condition of steam at the exhaust

$x_4 = 0.935$ (from MD)

Reheat - Regenerative cycle (combined cycle):

The reheating of steam is adopted when the vaporization pressure is high. The effect of reheat alone on the thermal η is very small. Regeneration or the heating up steam extracted from the turbine has a marked effect on cycle η. A modern steam power plant is equipped with both. The below fig shows plant with reheat & three stages of feed water heating.

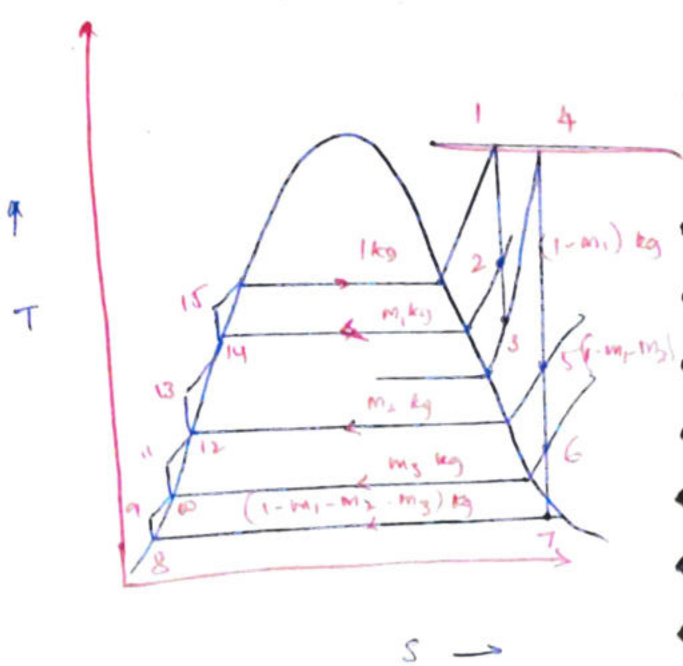


$$W_T = (h_1-h_2) + (1-m_1)(h_2-h_3) + (1-m_1)(h_4-h_5) + (1-m_1-m_2)(h_5-h_6) + (1-m_1-m_2-m_3)(h_6-h_7)$$

$$W_P = (1-m_1-m_2-m_3)(h_9-h_8) + (1-m_1-m_2)(h_{11}-h_{10}) + (1-m_1)(h_{13}-h_{12}) + 1(h_{15}-h_{14})$$

$$Q_1 = (h_1-h_{15}) + (1-m_1)(h_4-h_3)$$

$$Q_2 = (1-m_1-m_2-m_3)(h_7-h_8)$$



The energy balance of heaters 1, 2 & 3 gives.

$$m_1 h_2 + (1-m_1) h_{13} = 1 \times h_{14} \rightarrow (1)$$

$$m_2 h_5 + (1-m_1-m_2) h_{11} = (1-m_1) h_{12} \rightarrow (2)$$

$$m_3 h_6 + (1-m_1-m_2-m_3) h_9 = (1-m_1-m_2) h_{10} \rightarrow (3).$$

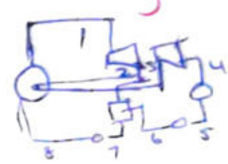
(1) A steam P.P equipped with regeneration as well as reheat arrangement is supplied with steam to the H.P turbine at 80 bar 470°C. For feed heating a part of steam is extracted at 7 bar & remainder of steam is reheated to 350°C in a reheater & then expanded in LP turbine down to 0.035 bar. Determine (i) Amount of steam bled-off for feed heating.

(ii) " " supplied to LP T

(iii) Q_s in boiler & reheater

(iv) η_{cycle}

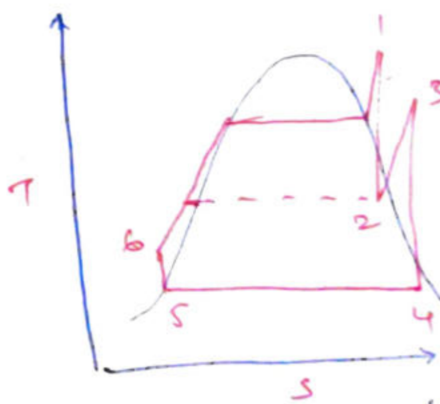
(v) Power, steam supplied by the boiler is 50 kg/s.



Sol: From h-s Chart & ST

$h_1 = 3315$
$h_2 = 2716$
$h_3 = 3165$
$h_4 = 2236$
} from MD

$h_{f6} = h_{f2} = 697.1$
$h_{f5} = h_{f4} = 101.9$
} From ST



(1) ~~Amt~~ Considering energy balance

$$m(h_2 - h_{f6}) = (1-m)(h_{f6} - h_{f5}) \Rightarrow m = 0.225 \text{ kg of steam supplied}$$

~~$m(h_2 - h_{f6})$~~ ∴ Hence amount of steam bled off is 22.5%

(2) Amt of steam supplied to LPT

$= 100 - 22.5 = 77.5\%$

(3) Q_s in boiler : $h_1 - h_{f6} = 3315 - 697.1 = 2617.9$ kJ/kg

in Reheater $Q_s = (1-m)(h_3 - h_2) = 347.97$ kJ/kg.

Total $Q_s = 2617.9 + 347.97 = 2965.87$ kJ/kg

(iv) η_{cycle}

$w_T = 1(h_1 - h_2) + (1-m_1)(h_3 - h_4)$ (neglect pump)

$= 1319$ KJ/kg

$\eta_{cycle} = \frac{W}{Q_s} = \frac{1319}{2965.87} = 44.47\%$.

(v) Power developed $= m_s \times W = 50 \times 1319 = 65950$ kW.

(2) In a steam power plant, the condition of steam at inlet to turbine is 20 bar & 300°C & the condenser pressure is 10 KPa. Two feed water heaters operate at 5 bar & 1 bar. By neglecting the pump work, determine

(Regenerative problem)

(a) Quality of steam at turbine exhaust

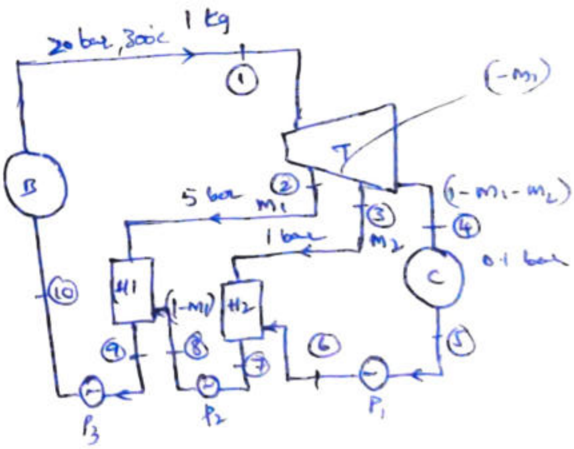
(b) masses of steam bled off at each pressure/kg of steam entering the turbine.

1 bar = 100 kP

(c) Net work done/kg of steam flow.

(d) η_{th} of the cycle &

(e) Specific steam consumption.



Sol:

$S_1 = S_2 = S_3 = S_4$

At 20 bar $S_1 = 6.77$ KJ/kg K

At 0.1 bar $S_{f4} = 0.649$

$S_{fg4} = 7.502$

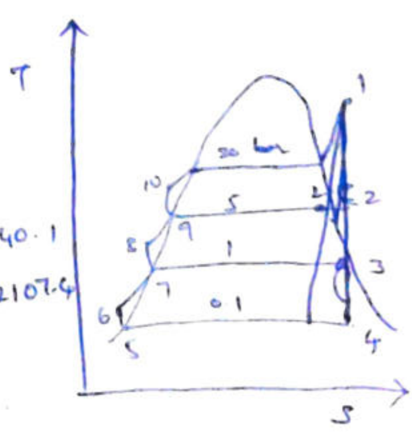
$S_1 = S_4$

$6.77 = S_{f4} + x_4 S_{fg4} = 0.649 + x_4(7.502)$

$x_4 = 0.815$

At 20 bar, $h_1 = 3025$

At 5 bar, $S_{g2} = 6.819$ $\therefore S_{g2} > S_2$, so point (2) will be inside the saturation line



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

$S_{f2} = 1.86$ | $h_{f2} = 640.1$

$S_{fg2} = 4.959$ | $h_{fg2} = 2107.4$

$S_1 = S_2$

$6.77 = 1.86 + x_2(4.959) \Rightarrow x_2 = 0.99$

$h_2 = 640.1 + 0.99(2107.4) = 2726.67$ KJ/kg

$S_1 = S_3$ $h_{f3} = 417.5$ $S_{f3} = 1.303$

At 1 bar, $h_{fg3} = 2257.9$ $S_{fg3} = 6.057$

$S_1 = S_3$ (20)

$6.77 = 1.303 + x_3 (6.087) \Rightarrow x_3 = 0.902$

$h_3 = 417.5 + 0.902 (2257.9) = 2455.46$

(0.1 bar) $h_4 = 191.8 + 0.815 (2392.9) = 2142.01$ $(x_4 = 0.815)$

$h_{f5} = h_{f6} = 191.8$

$h_{f7} = h_{f8} = 417.5$

$h_{f9} = h_{f10} = 640.1$

H① : Energy balance : $m_1 h_2 + (1-m_1) h_{f8} = 1 \times h_{f9}$

$m_1 (2726.67) + 417.5 - 417.5\, m_1 = 640.1 \Rightarrow m_1 = 0.0964$

H② : $m_2 h_3 + (1-m_1-m_2) h_{f6} = (1-m_1) h_{f7}$

$m_2 \times 2455.46 + (1-0.0964-m_2)\, 191.8 = (1-0.0964)\, 417.5$

Q) Consider a reheat regenerative power cycle with ... feed water heater. The steam enters the first turbine at 15 MPa 600°C & expands to 2 MPa. Then the steam is reheated to 600°C at the same pressure. The steam for feed water heater is extracted from the low pressure turbine at the pressure of 0.5 MPa & the remaining steam is further expanded to a condenser pressure of 10 kPa. Determine
(a) Fraction of steam extracted from turbine for FWH
(b) η_{th} (c) Mass flow rate of steam in kg/h if cycle produces 120 MW
The working fluid experiences no irreversibilities passes through turbine, pumps, heaters & condensers

Application of nozzles

→ Steam turbines, gas turbines, jet engines, flow measuring devices, fuel injection, carburetion system of IC engines, spray painting etc

→ Convergent nozzle is used when back pressure is equal to or greater than critical pressure

→ Divergent nozzle is used when the back pressure is less than critical pressure

→ Convergent - Divergent nozzle is used when the back pressure is less than the critical pressure. Widely used in steam & gas turbines

A PRESENTATION ON

BOILERS

❖ BOILER

Boiler, How it works .mp4

- Boiler is an apparatus which produce steam by supplying heat to the water.
- Thermal energy released by combustion of fuel is used to make steam at the desired temperature and pressure.

The steam produced is used for:

1) Generating power in steam engines or steam turbines.

2) Heating the residential and industrial buildings.

3)Performing certain processes in the sugar mills, chemical and textile industries.

Selection of boiler:

- The working pressure & quality of steam.
- Rate of steam generation.
- Floor area available.
- Accessibility for repair & inspection.
- Comparative initial cost.
- The fuel & water available.
- Operating & maintenance cost.
- Portable load factor.
- Erection facility.

CLASSIFICATION OF BOILERS :

The boiler may be classified as :

1. Horizontal, vertical or inclined
2. Fire tube & water tube boilers
3. Externally fired & internally fired boilers
4. Forced circulation & Natural circulation boilers
5. High pressure (>80 bar) & low pressure boilers
6. Stationary & portable boilers
7. Single tube & multi tube boilers

1. Horizontal , vertical or inclined

- If the axis of boiler is horizontal, the boiler is called horizontal.
- If the axis is vertical then it is called vertical.
- If the axis is inclined then it is called inclined.
- Advantage of horizontal boiler is that it can be repaired easily.
- Advantage of vertical boiler is that it occupies less floor area.

2. Fire tube & water tube boiler

- In the fire tube boiler the hot gases are inside the tubes & the water surrounds the tubes.

  ex. Cochran, locomotive etc.

- In the water tube boiler the water is inside the tube & the hot gases are surround them.

  ex. Stirling, Babcock etc..

3. Externally & internally fired

- The boiler is known as externally fired if the fire is outside the shell.

  ex. Babcock & Wilcox

- In case of internally fired boilers, the furnace is located inside the boiler shell.

  ex: Cochran, Lancashire etc.

4. Forced circulation & natural circulation

- In forced circulation type of boilers the circulation of water is done by pumps.

  ex. Velox, Lamont etc .

- In natural circulation type of boiler the circulation of water in boiler takes place due to density variation.

  ex. Lancashire, Babcock etc

5. High pressure & low pressure boilers

- The boilers which produce steam at pressure of 80 bar and above are called high pressure boiler.

  ex: Velox

- The boiler which produce steam at pressure below 80 bar are called low pressure boiler.

  ex. Cochran

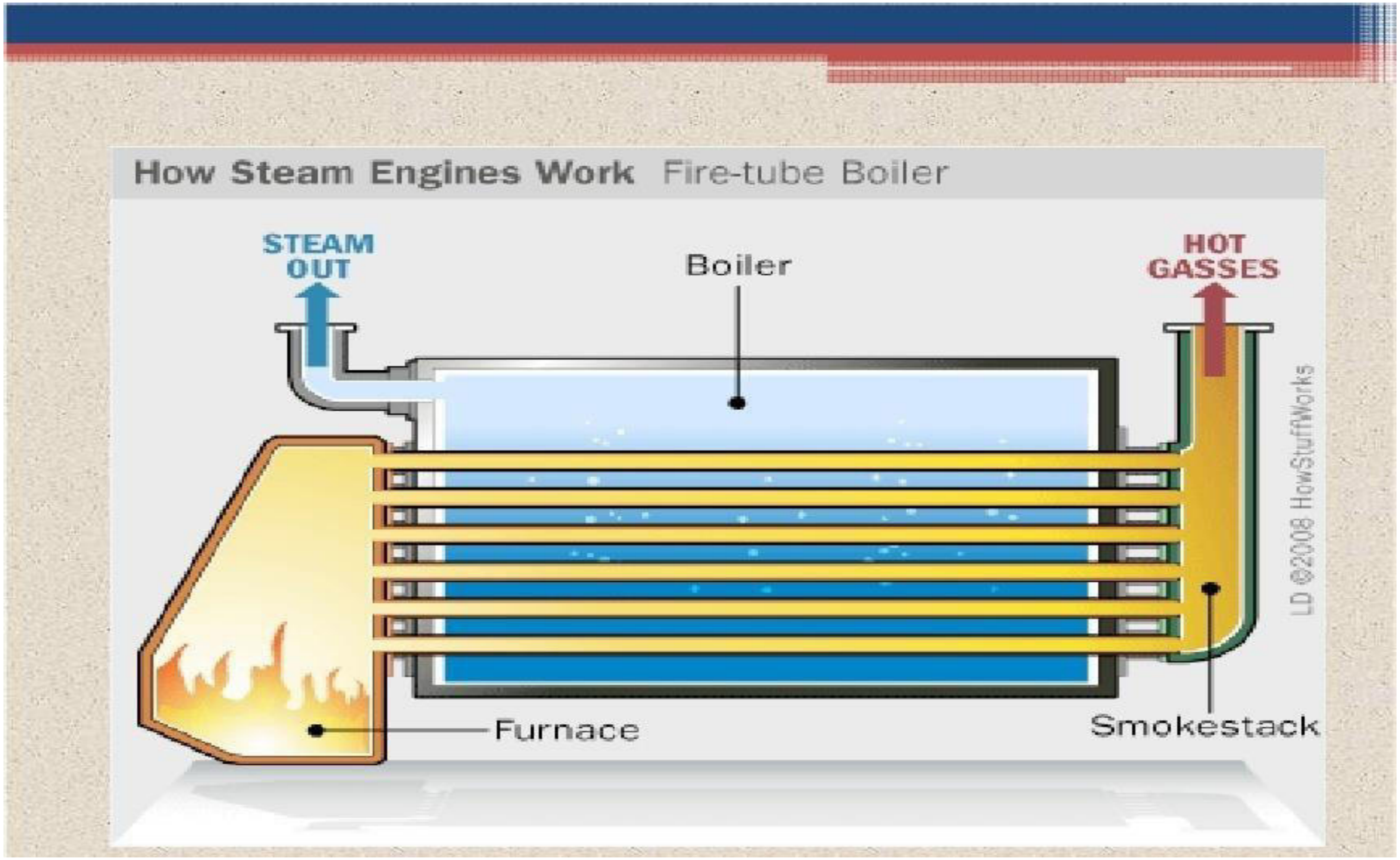
7. Single tube & multi-tube boilers

- The fire tube boilers are classified as single tube & multi tube boilers, depending upon the fire tube is one more than one.
- Ex: Cornish boiler

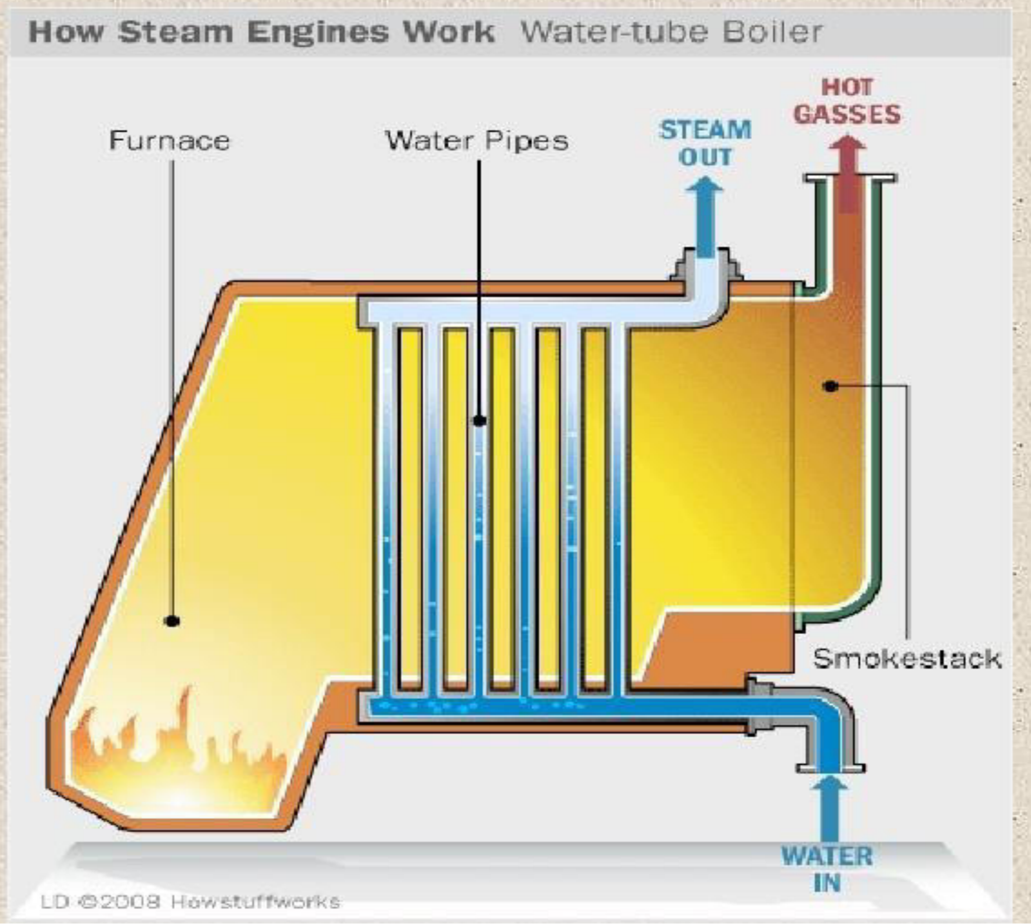
Basic classification

- Fire tube boilers
- Water tube boilers

FIRE TUBE BOILERS



WATER TUBE BOILERS



Comparison between Fire-tube & water-tube boilers:

S no.	Particulars	Fire tube boilers	Water tube boilers
1.	Mode of firing	Internally fired	Externally fired
2.	Rate of steam production	lower	Higher
3.	Construction	Difficult	Simple
4.	Transportation	Difficult	Simple
5.	Treatment of water	Not so necessary	More necessary
6.	Operating pressure	Limited to 16 bar	Under high pressure as 100 bar
7.	Floor area	More floor area	Less floor area
8.	Shell diameter	Large for same power	Small same power
9.	Explosion	Less	More
10.	Risk of bursting	lesser	More risk

TYPES OF FIRE TUBE BOILERS

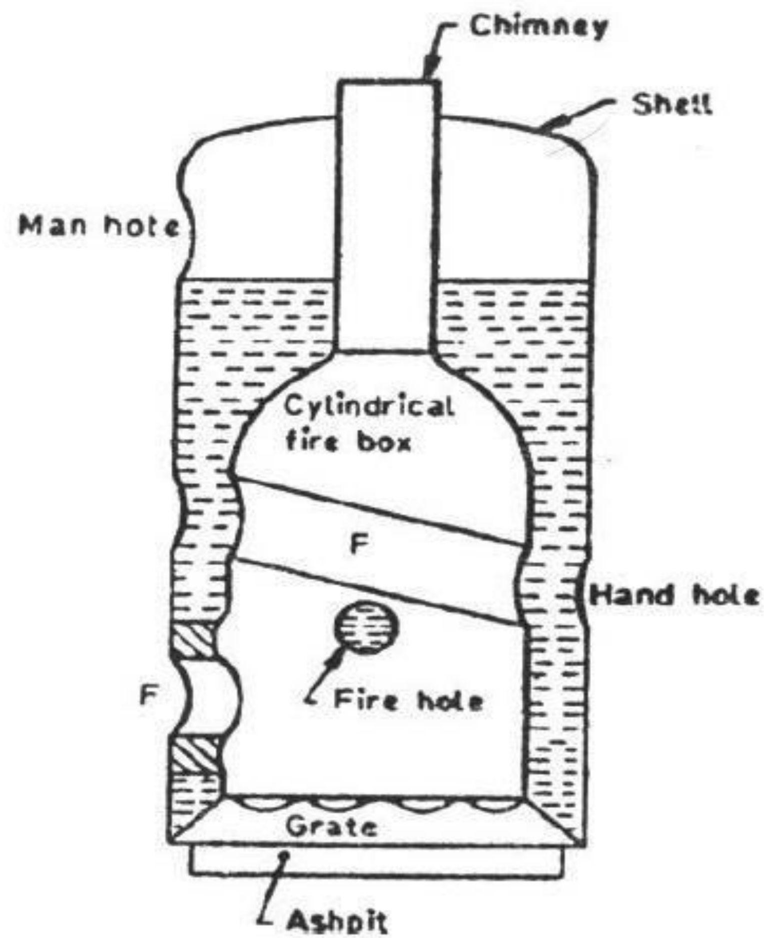
There are mainly five types of fire tube boilers :

1) Simple vertical boiler
2) Cochran boiler
3) Cornish boiler
4) Lancashire boiler
5) Locomotive boiler
6) Scotch boiler

1). SIMPLE VERTICAL BOILER

simple vertical boiler.mp4

- It consist of a cylindrical shell, the greater portion of which is full of water & remaining is the steam space.
- At the bottom of the fire box is grate on which fuel is burnt and the ash from it falls in the ash pit.
- A simple vertical boiler is self-contained & can be easily transported.



2) COCHRAN BOILER

- It is one of the best types of vertical multi-tubular boiler, and has a number of horizontal fire tubes.

- Cochran boiler consist of a cylindrical shell with a dome shaped top where the space is provided for steam.

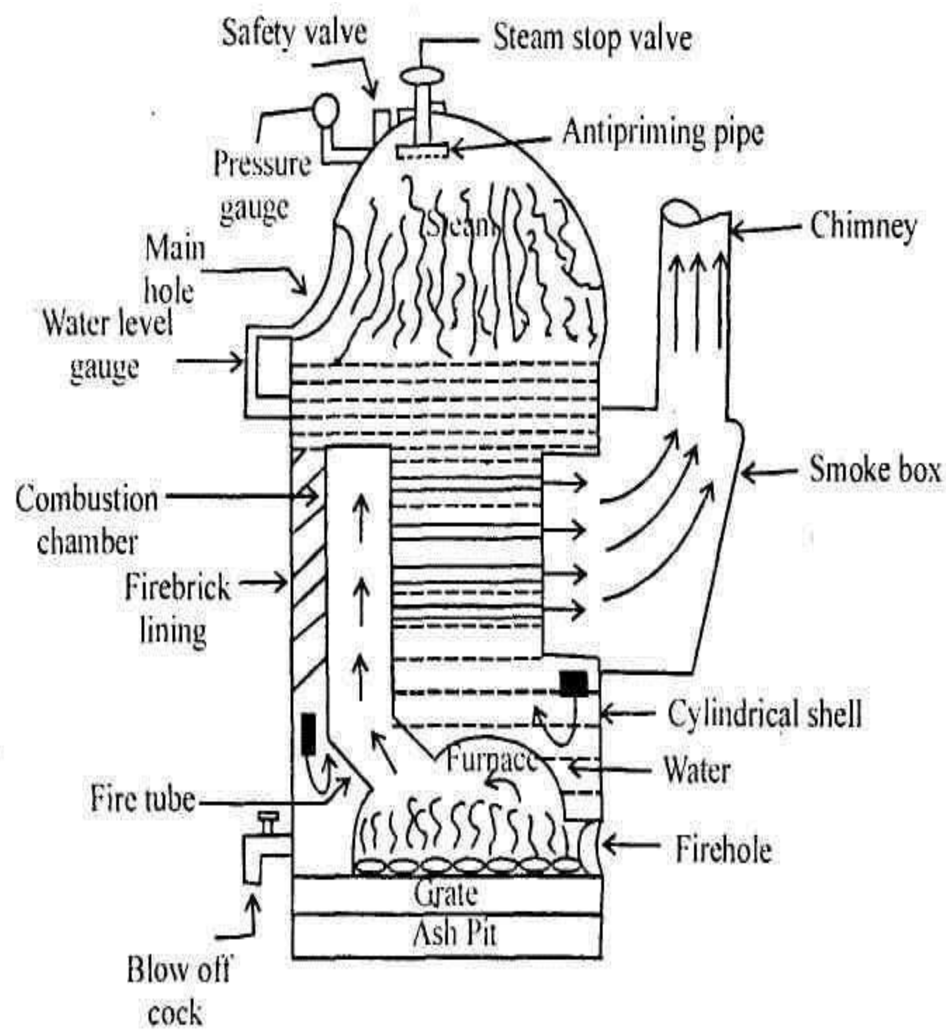
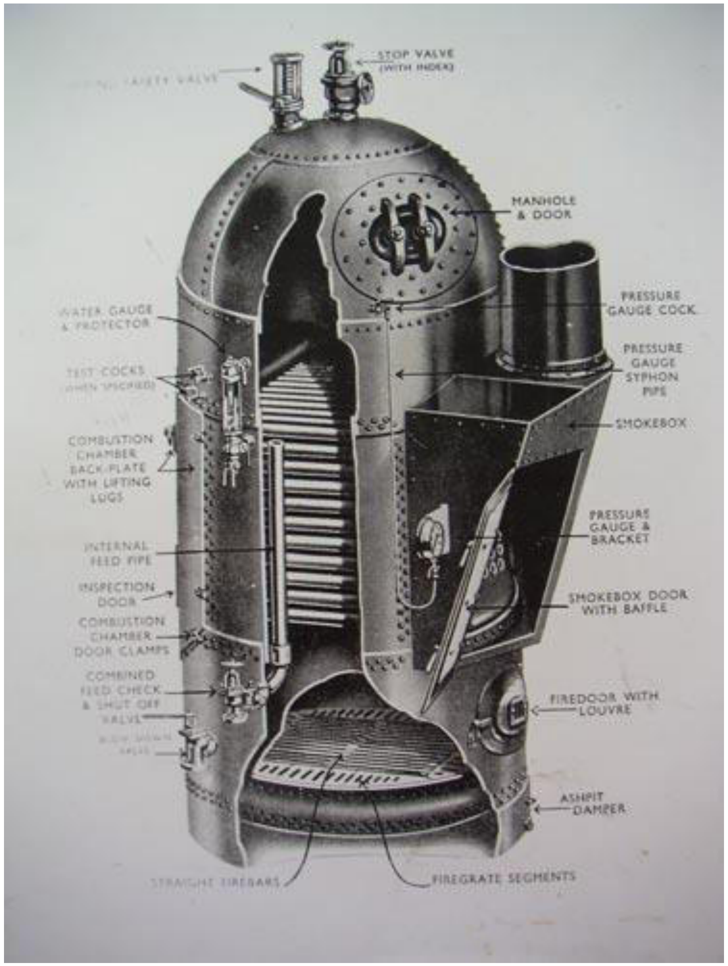


Fig. Cochran Boiler

COCHRAN BOILER.mp4

COCHRAN BOILER



- Working Pressure: 6.5 bar(max pressure=15 bar)
- Steam capacity =3500 Kg/hr (max capacity=4000Kg/hr)
- Heating surface =120 sq.mts
- Efficiency=70 to 75% (depending on the fuel used)

3 .CORNISH BOILER

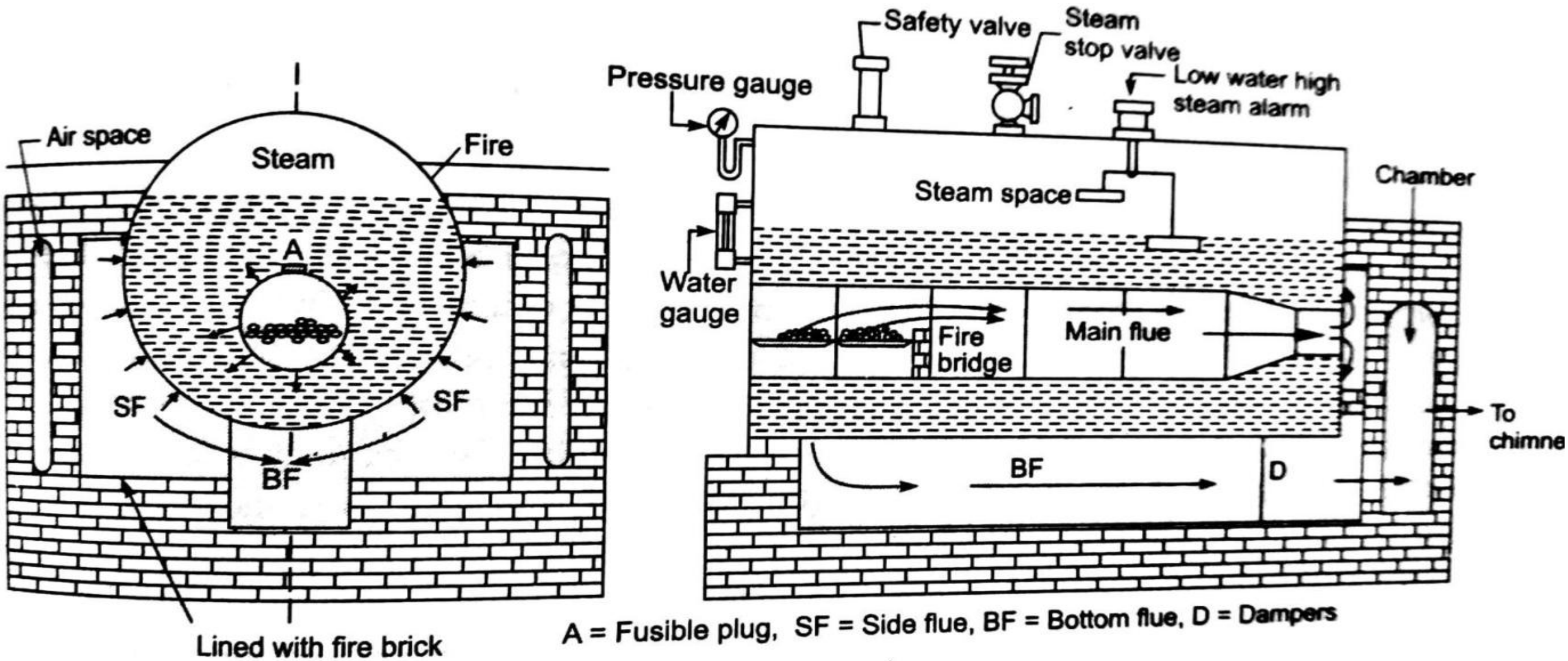


Fig. 17.5 *Cornish boiler*

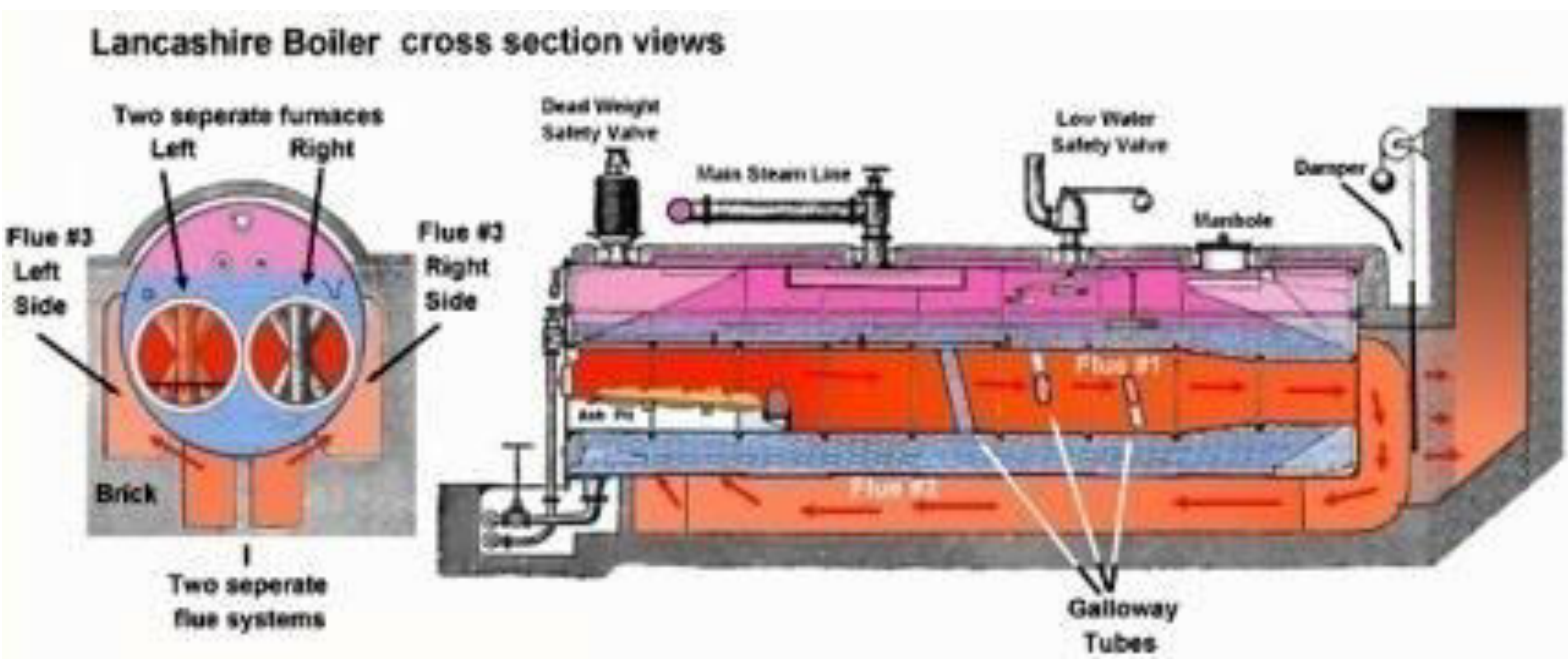
Differences with Lancashire Boiler

- **Small in size**
- **It has only one flue tube**
- **Dia of the shell =1.25m to 1.75m**
- **Length of the shell = 4 to 7 mts**
- **Maximum working pressure = 10.5 bar**
- **Steam Capacity = 6500 kg/hr**
- **Efficiency = 50 to 70 %.**

4) LANCASHIRE BOILER

Working & Construction of Lancashire Boiler - Magic Marks.mp4

- This boiler is reliable, has simplicity of design, ease of operation & less operating & maintenance costs.
- It is commonly used in sugar-mills & textile industries where along with the power system & steam for the process work is also needed.

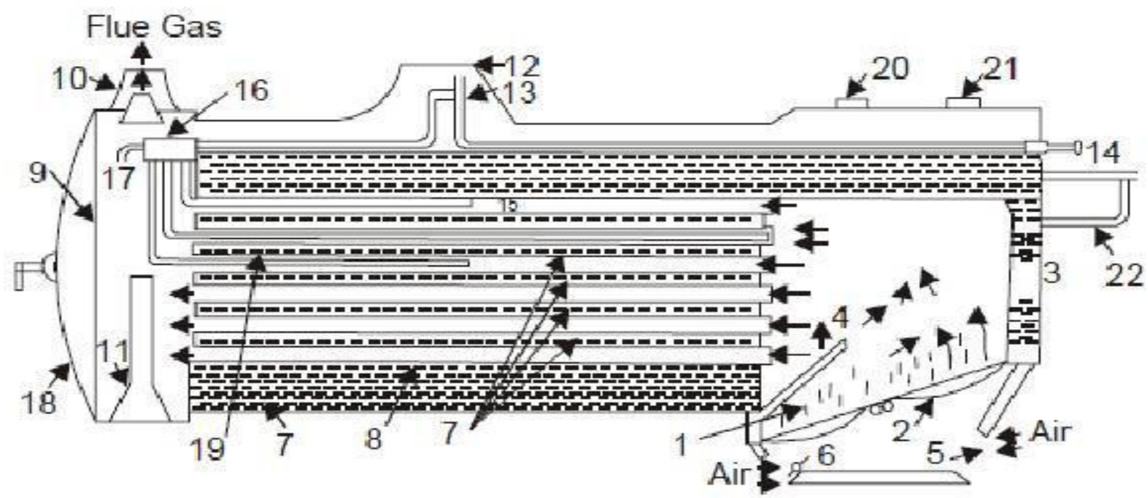


LANCASHIRE BOILER

- **Dia of the shell =2 to 3 mts**
- **Length of the shell = 7 to 9 mts**
- **Maximum working pressure = 16 bar**
- **Steam Capacity = 9000 kg/hr**
- **Efficiency = 50 to 70 %.**

5) LOCOMOTIVE BOILERS

ANUNIVERSE 22 - LOCOMOTIVE BOILER WORKING [FIRE TUBE BOILER].mp4



1. Fire box
2. Grate
3. Fire hole
4. Fire bride arch
5. Ash pit
6. Damper
7. Fine tubes
8. Barrel or shell
9. Smoke box
10. Chimney (short)
11. Exhaust steam pipe
12. Steam dome
13. Regulator
14. Lever
15. Superheater tubes
16. Superheater header
17. Superheater exist pipe
18. Smoke box door
19. Feed check valve
20. Safety valve
21. Whistle
22. Water gauge

About Locomotive boilers :

- Locomotive boiler is a horizontal fire tube type mobile boiler. The main requirement of this boiler is that it should produce steam at a very high rate. Therefore, this boiler requires a large amount of heating surface and large grate area to burn coal at a rapid rate. In order to provide the large heating surface area, a large number of fire tubes are setup and heat transfer rate is increased by creating strong draught by means of steam jet.

ADVANTAGES:

a. Large rate of steam generation per square meter of heating surface. To some extent this is due to the vibration caused by the motion.

b. It is very compact.

c. The pressure of the steam is limited to about 20 bar.

Locomotive boilers

- Barrel dia = 2.095 m
- Length of barallel = 5.206 m
- Size of tubes (Super heater) = 14 cm
- No of superheater tubes = 38
- Steam capacity = 9000 kg/hr
- working pressure = 14 bar
- Coal brunt per hour = 1600 kg
- Efficiency = 70%

SCOTCH BOILER

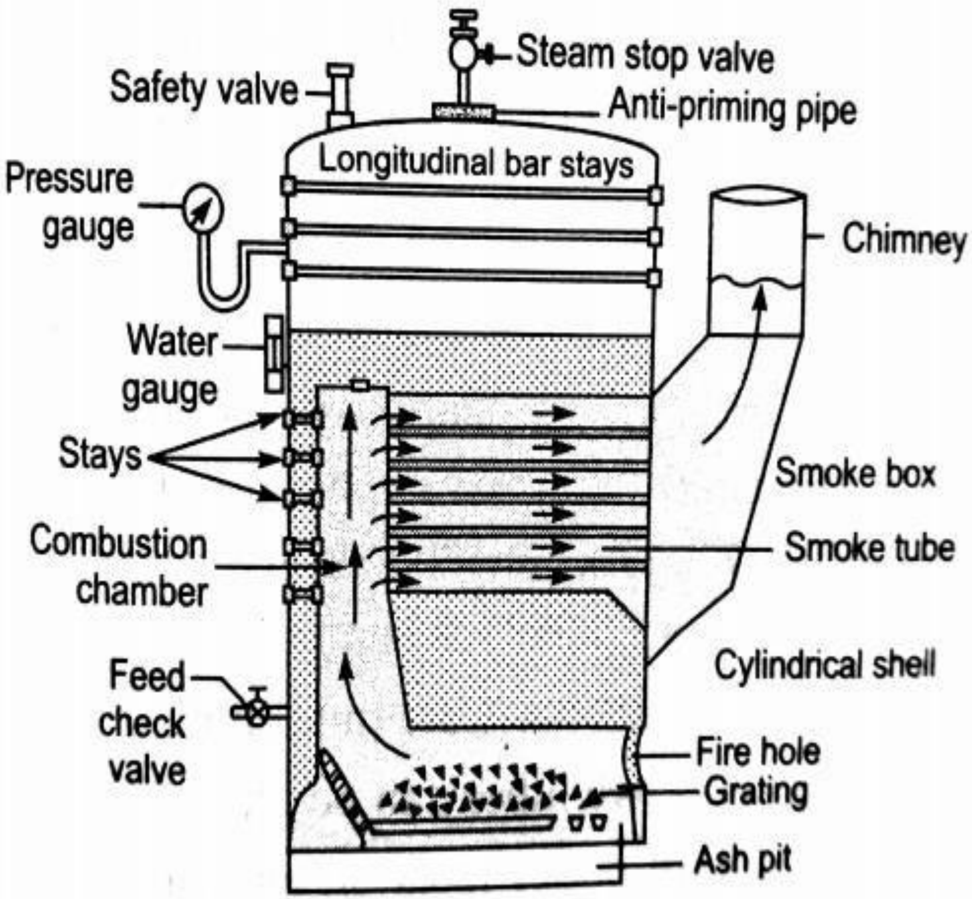


Fig. 17.7 *Scotch Marine boiler*

- Steam capacity about 1000 kg/hr
- Pressure developed around 17 bar
- Compact in size
- Occupies less space

WATER TUBE BOILERS ARE CLASSIFIED AS :

1). Horizontal Straight tube boiler

There are two types of Horizontal straight tube boiler:

a. Longitudinal drum

b. Cross drum

there are three types of cross drum boiler :

1. **Two drum**
2. **Three drum**
3. **Four drum**

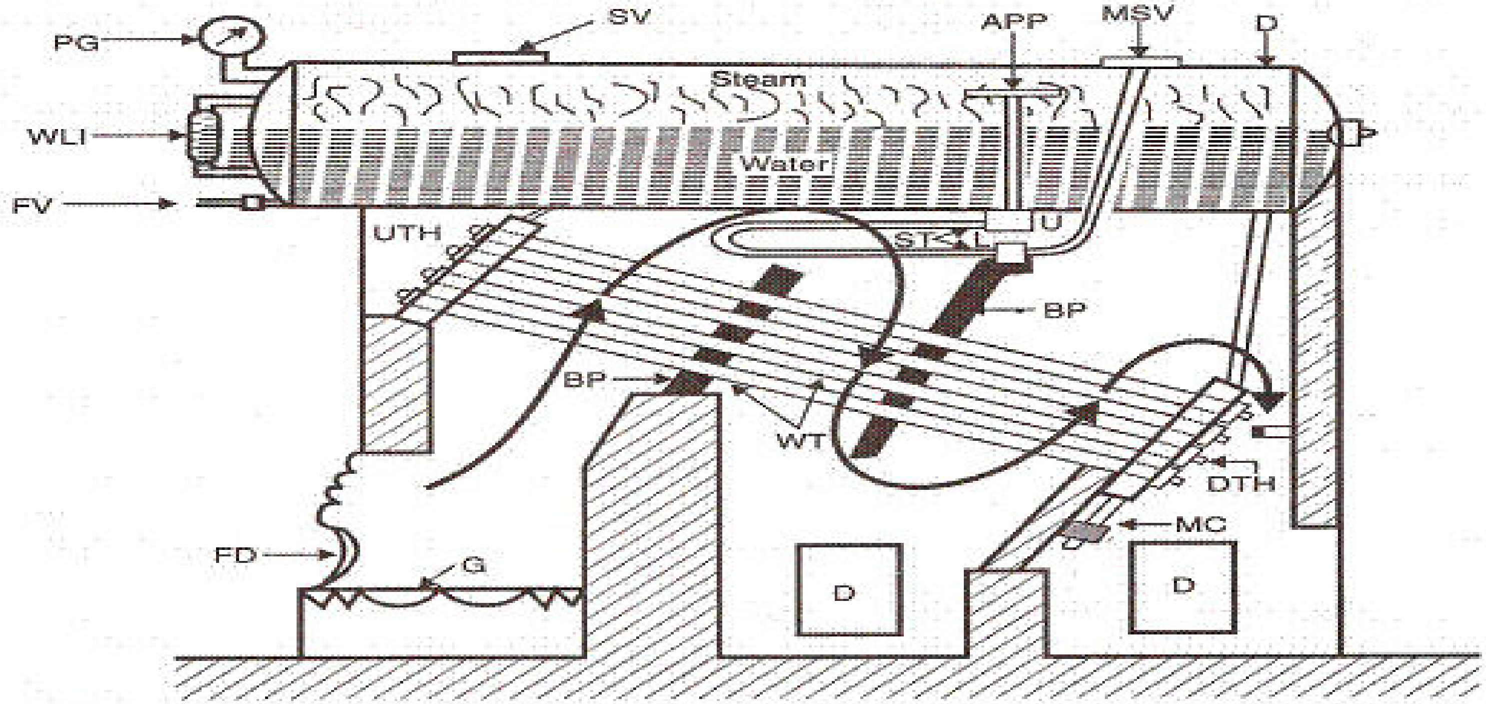
2). Bent tube boiler

There are one type of bent tube boiler :

a. Low head three drum

BABCOCK WILCOX BOILER

Construction & Working of Babcock & Wilcox Boiler - Magic Marks.mp4



D = Drum
DTH = Down take header
WT = Water tubes
BP = Baffle plates
D = Doors
G = Grate
FD = Fire door
MC = Mud collector
WLI = Water level indicator

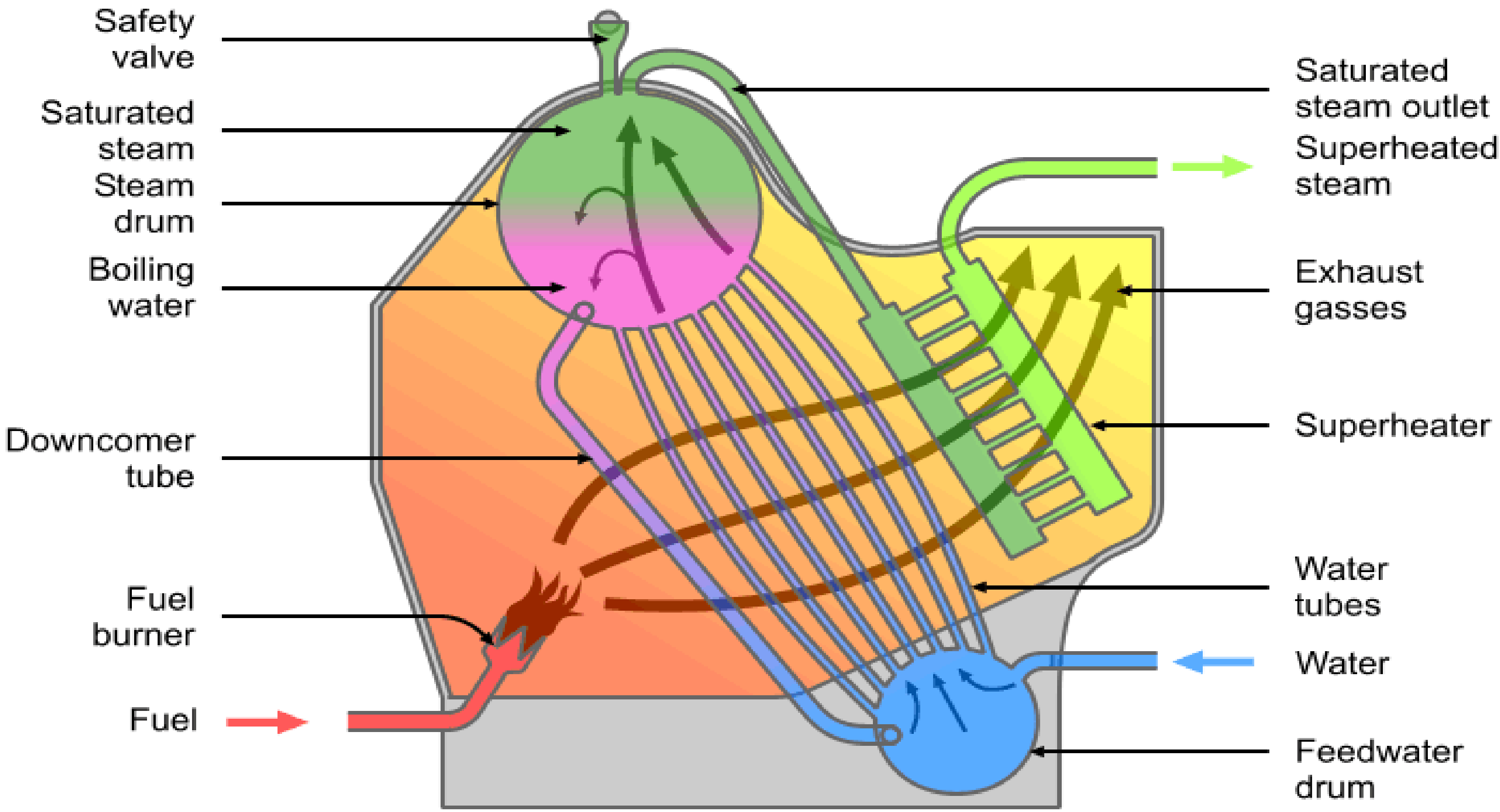
PG = Pressure gauge
ST = Superheater tubes
SV = Safety valve
MSV = Main stop valve
APP = Antipriming pipe
L = Lower junction box
U = Upper junction box
FV = Feed valve

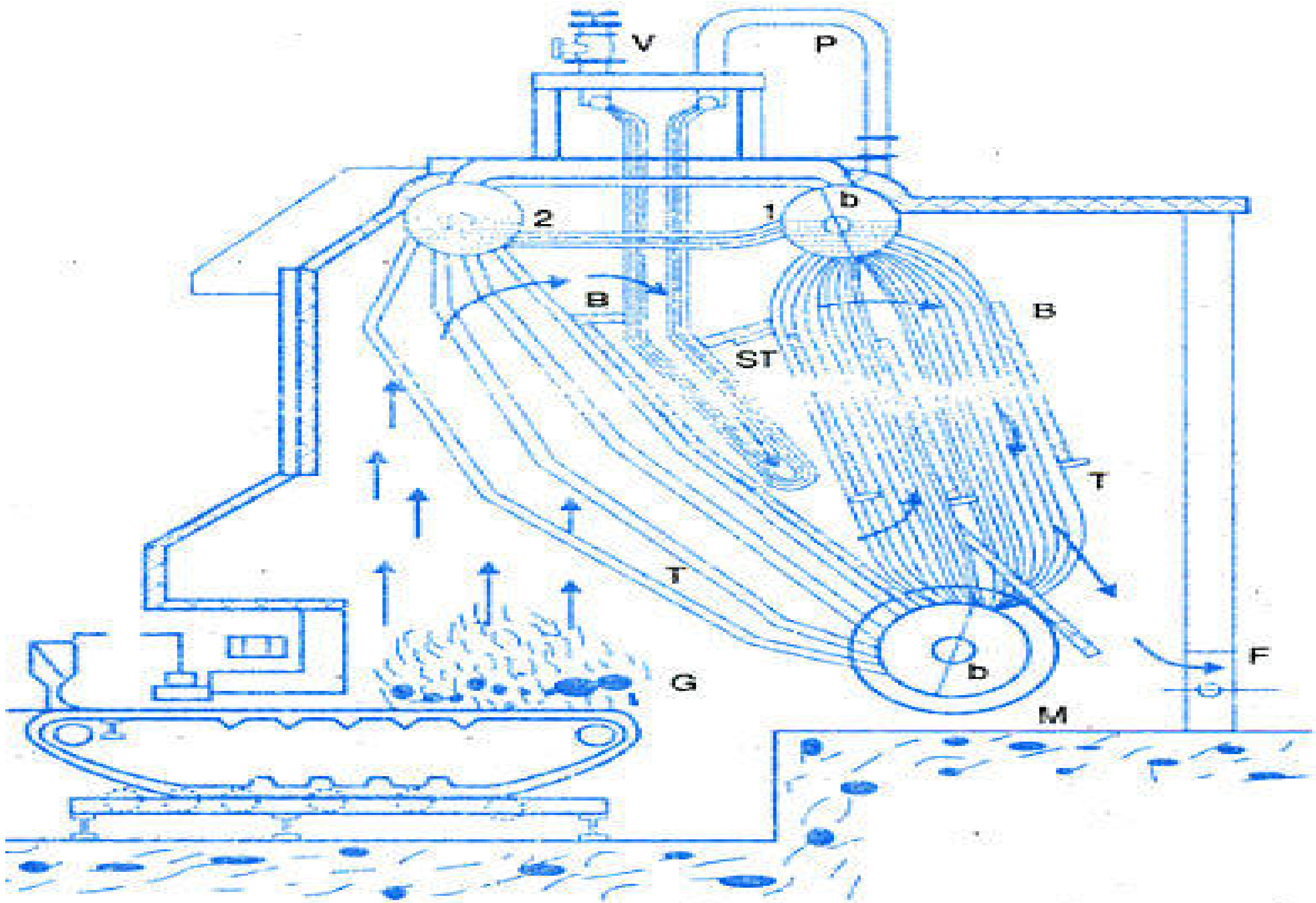
Features

1. The evaporative capacity of this boilers is high compared with other boilers (20,000 to 40,000 kg/hr). The operating pressure lies between 11.5 to 17.5 bar.
2. The draught loss is minimum compared with other boilers.
3. The defective tubes can be replaced easily.
4. The entire boiler rests over an iron structure, independent of brick work, so that the boiler may expand or contract freely. The brick walls which form the surroundings of the boiler are only to enclose the furnace and the hot gases.

Stirling Boiler

Construction and Working of Stirling Boiler - Magic Marks.mp4





V = Stop valve
P = Steam pipe
b = Water baffle
B = Baffle wall

ST = Superheater tubes
T = Water tubes
G = Grate
M = Mud drum


Thank You!

High Pressure Boilers

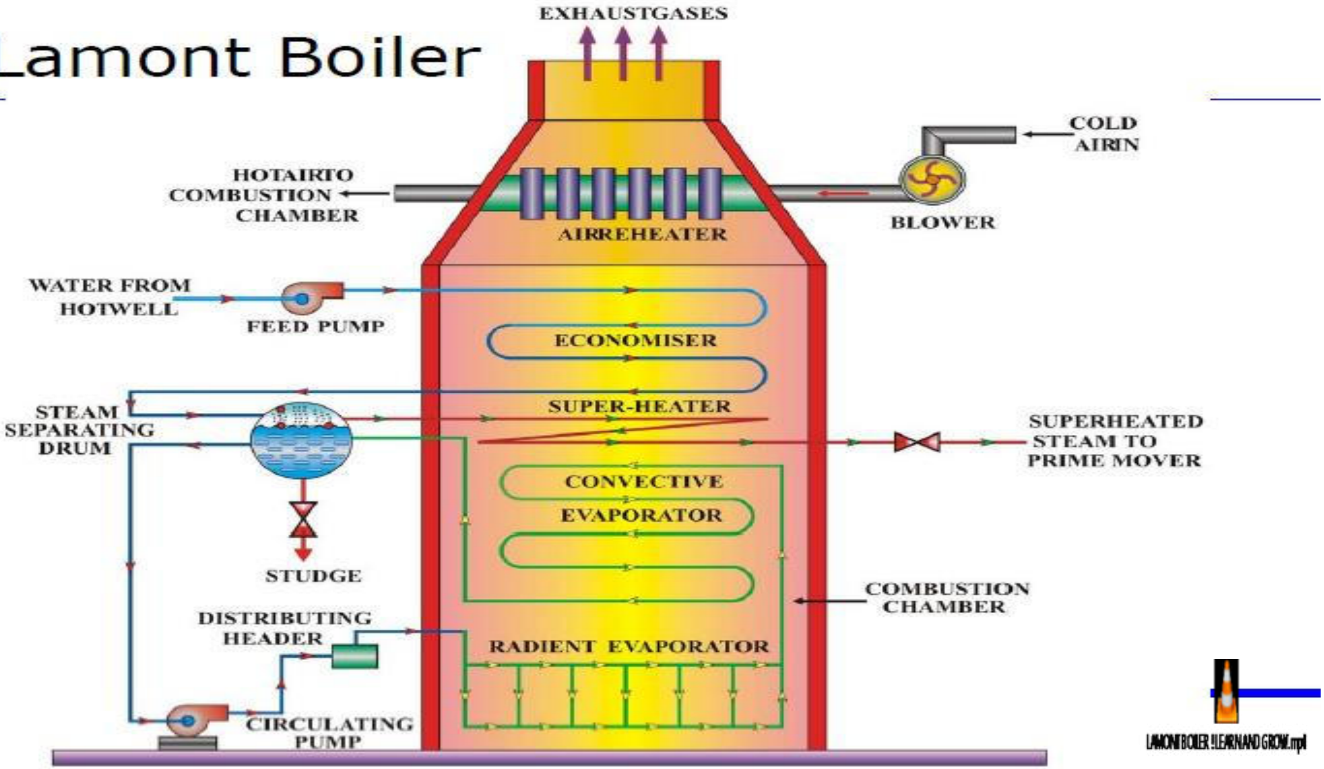
Advantages of High Pressure Boiler

- Due to use of forced circulation, there is freedom in the arrangement of furnace, tubes and boiler components.
- Because of the forced circulation usage, components can be arranged horizontally.
- All the parts are uniformly heated, therefore danger of overheating is reduced and thermal stress problem is simplified.
- The efficiency of plant is increased up to 40 to 42%.
- Rapid start from cold is possible, so it can be used as standby purpose.
- The differential expansion is reduced due to uniform temperature and this reduces the possibility of gas and air leakages.
- The tendency of scale formation is eliminated due to high velocity of water through the tubes.

High Pressure Boilers

- **Lamont Boiler**
- **Loeffler Boiler**
- **Benson Boiler**
- **Velox Boiler**

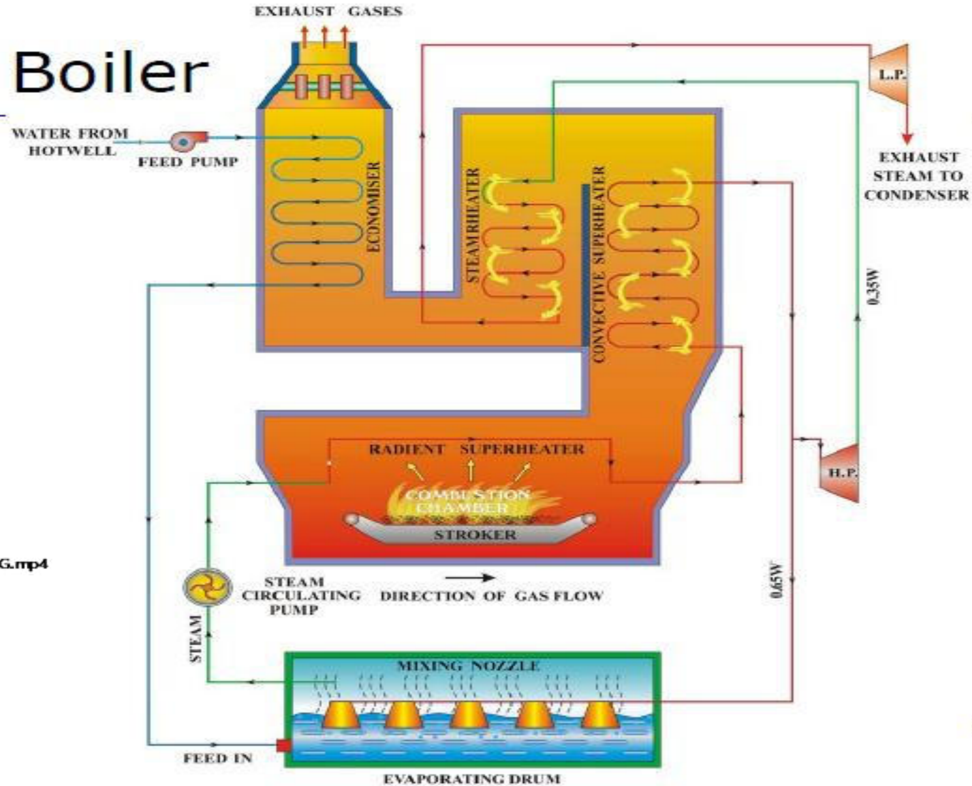
Lamont Boiler



Characteristic of La Mont Boiler

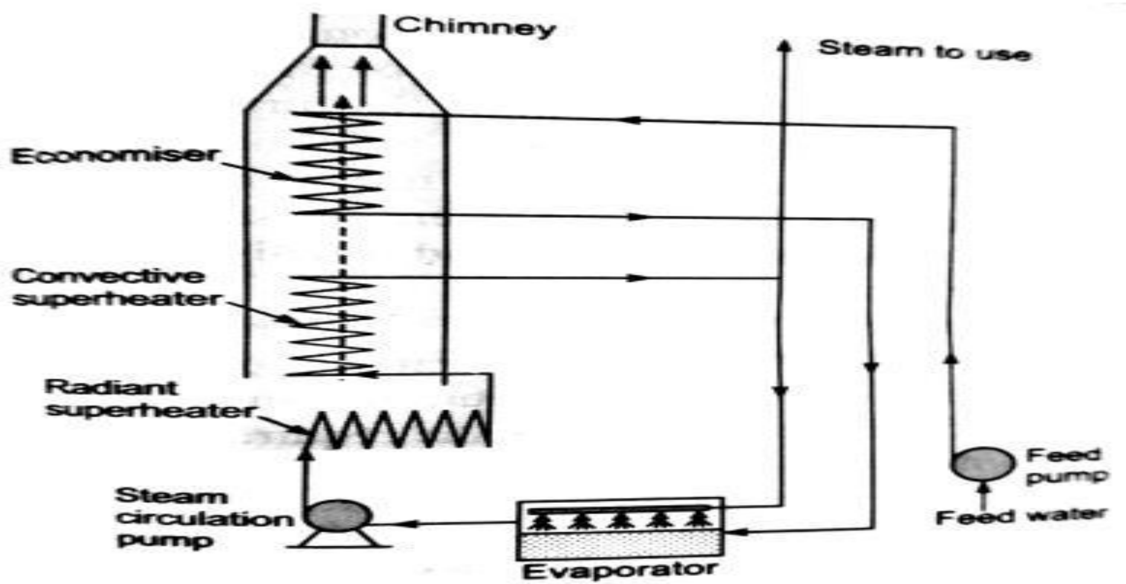
- A centrifugal pump circulates the water equal to 8 to 10 times the weight of steam evaporated.
- Because of the large circulation of water overheating is prevented.
- These boiler have been built to generate 45 to 50 tons of superheated steam at a pressure of 120 bar, and at a temperature of 500^0C
- Main disadvantage of this boiler is formation and attachment of bubbles on the inner surfaces of the heating tubes.

Loeffler Boiler



ANUNIVERSE 22 - LOEFFLER BOILER WORKING.mp4

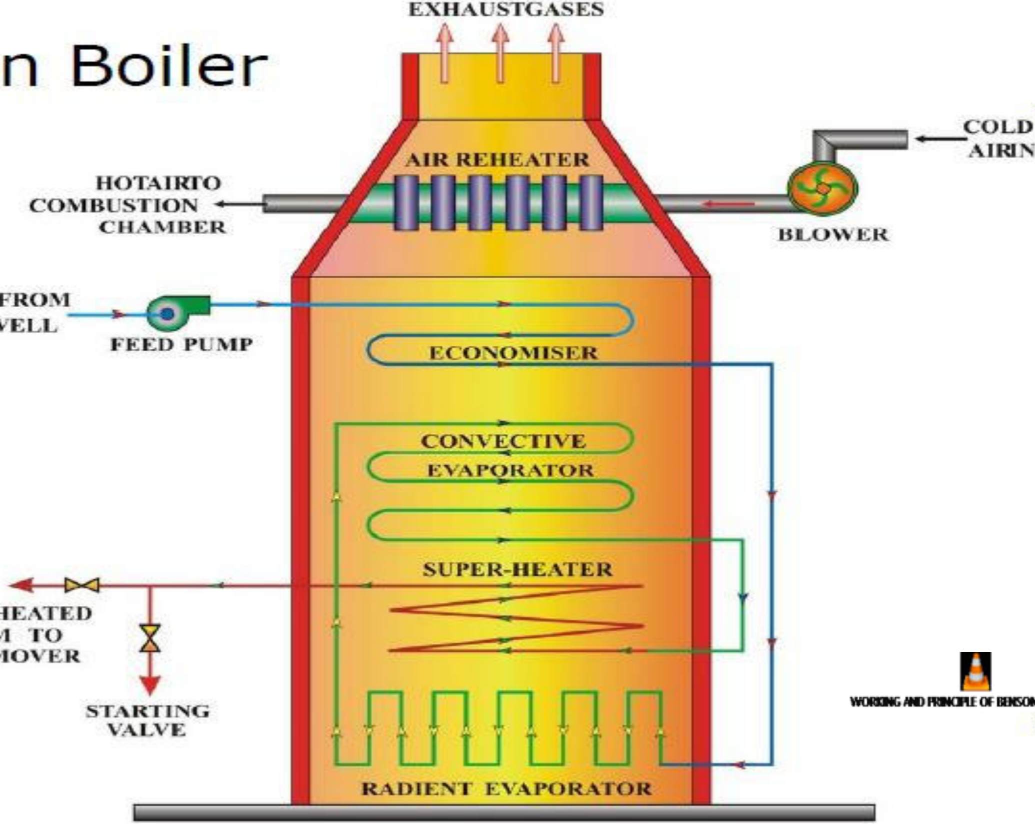
Loeffler Boiler



Characteristic Of Loeffler Boiler

- Problem of salt deposition in Benson boiler is eliminated in Loeffler boiler as most of the steam is generated outside from the feed water using part of the superheated steam coming out from the boiler.
- About 65% of steam from super-heater is used to evaporate the feed water and 35% of steam is supplied to H.P. turbine.
- Special design nozzle is used to distribute the superheated steam through the water into evaporator drum to avoid noise.
- This boiler can carry higher salt concentration than any other type and is more compact than indirectly heated boilers having natural circulation.
- Generating capacity is of 100 tones/hr and operating at 140 bar.

Benson Boiler



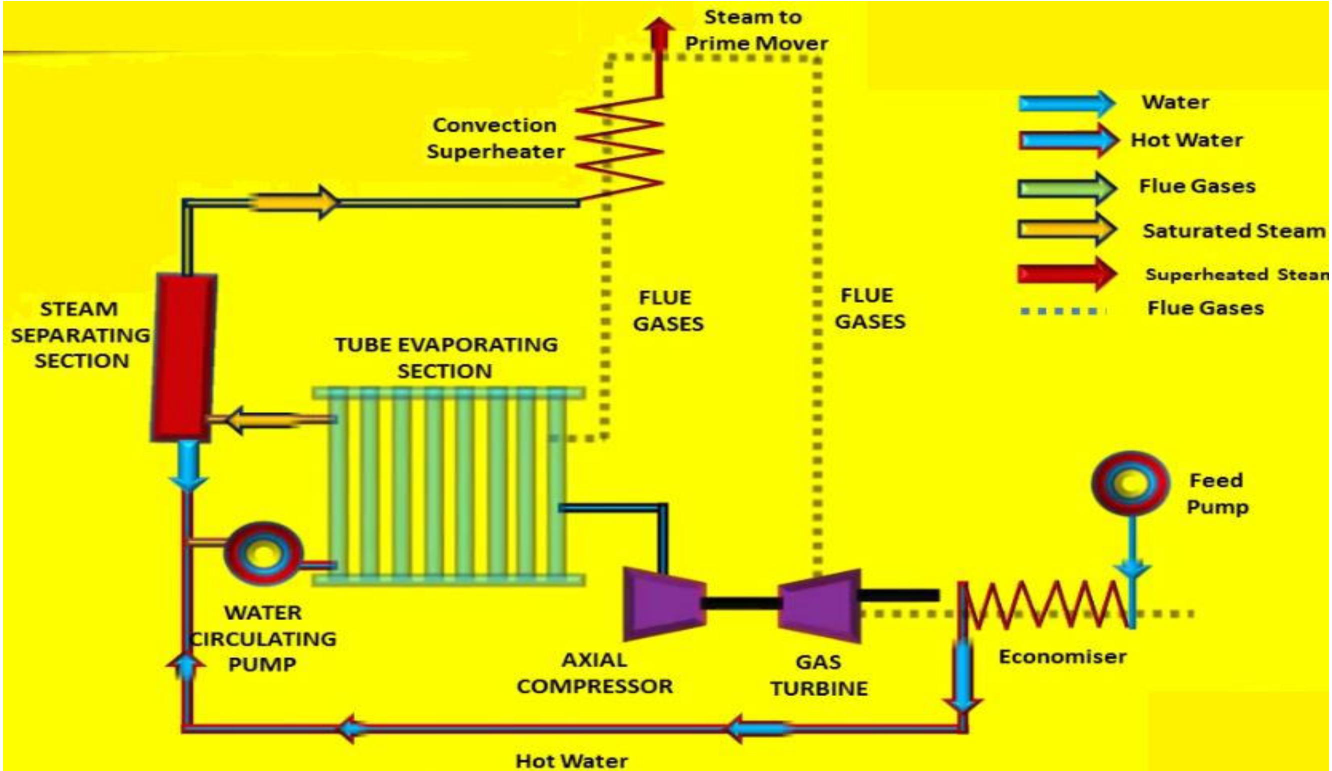
Benson Boiler

- The main difficulty experienced in the La Mont boiler is the formation and attachment of bubbles on the inner surfaces of the heating tubes.
- Benson in 1922 argued that if the boiler pressure was raised to critical pressure (225 atm.), the steam and water would have the same density and therefore the danger of bubble formation can be completely eliminated.
- Benson boiler having temperature as high as 650°C, pressure up to 500 bar and steam generation capacity of 150 tonnes/hr are in use.

Advantages and Disadvantages

- There is no drum so weight is reduced by 20% than other boilers.
- Pipe joints are welded, so erection of the boiler is easier and quicker as all parts are welded at sites.
- Transportation is easy as there is no drum and majority of parts are carried to sites without pre-assembly.
- Can be started very quickly.
- Insensitive to load fluctuation.
- Blow-down losses of Benson boiler are hardly 4% of natural circulation boilers of same capacity.
- Major problem is salt deposition when all remaining water is converted into steam in transformation zone. To avoid this, the boiler is flashed out after every 4000 working hours to remove salt.

VELOX BOILER



VELOX BOILER

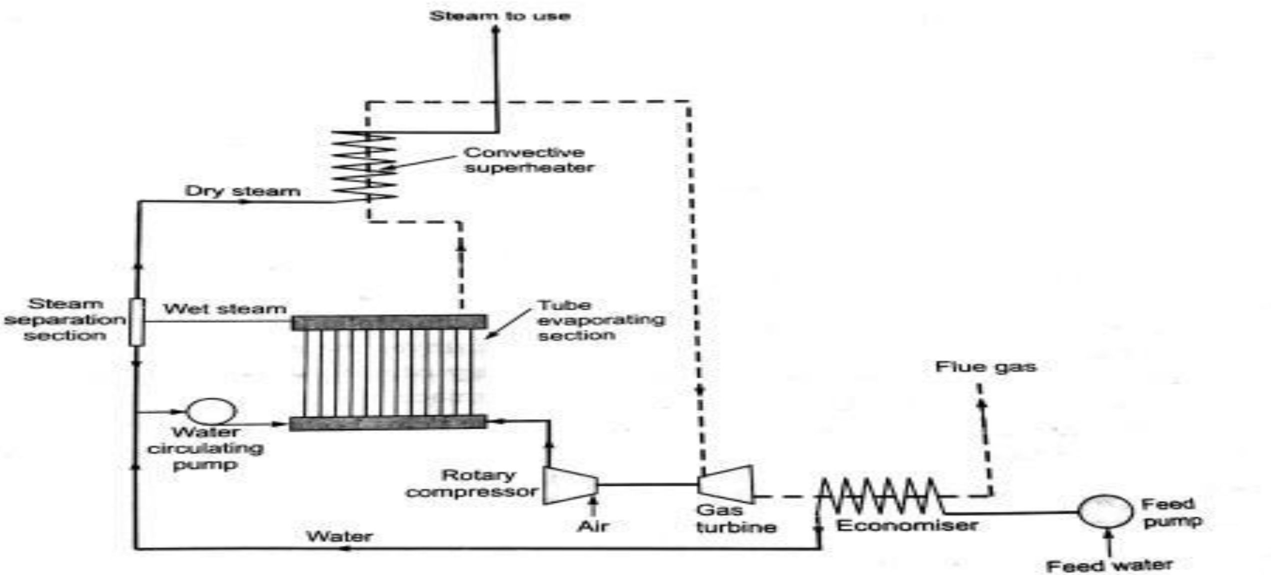


Fig. 17.15 *Velox boiler*

Characteristics of Velox Boiler

- When the gas velocity exceeds the sound-velocity, the heat is transferred from the gas at a much higher rate than rates achieved with sub-sonic flow.
- High heat release rates (40 MW/m3).
- Circulation ratio varies from 10 to 20%.
- Low excess air is required as the pressurised air is used and the problem of draught is simplified.
- It is very compact generating unit and has greater flexibility.
- It can be quickly started even though the separator has a storage capacity of about 10% of the maximum hourly output.

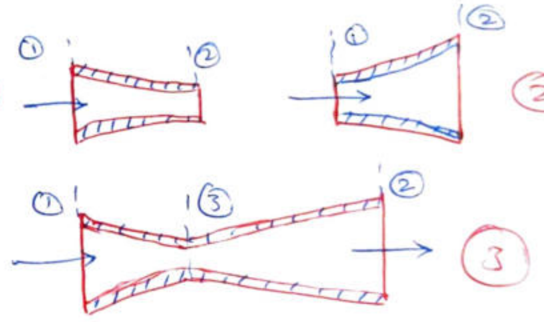
Thank You

Nozzle :-

A steam nozzle is a passage of varying cross section, which converts heat energy of steam into kinetic energy. During the first part of the nozzle, the steam increases its velocity. But in its later part, the steam gains more in volume than in velocity.

Since the mass of steam passing through any section of the nozzle remains constant, the variation of steam pressure in the nozzle depends upon the velocity, sp. vol & dryness fraction of steam. The main use of steam nozzle in steam turbines, is to produce a jet of steam with a high velocity. The smallest section of the nozzle is called throat.

Types of Steam Nozzles :-



1) Convergent nozzle
2) Divergent Nozzle
3) Convergent - divergent nozzle

Flow of Steam through Convergent - Divergent Nozzle :-

~~When the steam flows through a nozzle, some loss in its enthalpy or total heat takes place due to friction~~

The steam enters the nozzle with a high pressure, but with a negligible velocity. In the converging portion, there is a drop in the steam pressure with a rise in its velocity. There is also a drop in the enthalpy or total heat of the steam. This drop of heat is not utilised in doing some external work, but is converted into kinetic energy. In the divergent portion there is further drop of steam pressure with a further rise in its velocity

Again, there is a drop in the enthalpy or total heat of steam, which is converted into kinetic energy.

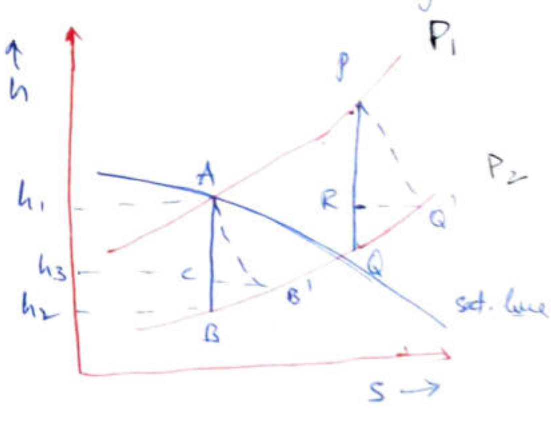
It will be interesting to know that the steam enters the nozzle with a high pressure & negligible velocity. But leaves the nozzle with a high velocity & small pressure.

⇒ The pressure at which the steam leaves the nozzle is known as back pressure. Moreover, no heat is supplied or rejected by the steam during flow through a nozzle.

⇒ Therefore it is considered as isentropic flow & the corresponding expansion is considered as an isentropic expansion.

Nozzle efficiency (or) Friction in a Nozzle :-

When the steam flows through a nozzle, some loss in its enthalpy or total heat takes place due to friction b/w the nozzle surface & the flowing steam. This can be understood by mollier chart as shown.



⇒ Locate point A for the initial condition of the steam. It is a point where the sat. line meets the initial pressure line. (P_1).

⇒ Now draw a vertical line through A to meet final pressure (P_2) line. This is done as the flow through the nozzle is isentropic. The heat drop ($h_1 - h_2$) is known as isentropic heat drop.

⇒ Due to friction in the nozzle the actual heat drop in the steam, will be less than ($h_1 - h_2$). Let this heat drop be shown as AC instead of AB.

⇒ As the expansion of steam ends at the pressure P_2, therefore final condition of steam is obtained by drawing a horizontal line through C to the meet the final pressure (P_2) line at B'.

⇒ Now the actual expansion of steam in the nozzle is expressed by the curve AB' (adiabatic expansion) instead of AB. (isentropic expansion). The actual heat drop $(h_1 - h_3)$ is known as useful heat drop.

"Coefficient of Nozzle" (OR) "Nozzle efficiency" is defined as the ratio of useful heat drop to the isentropic heat drop.

$$K = \frac{\text{Useful heat drop}}{\text{Isentropic heat drop}} = \frac{AC}{AB} = \frac{h_1 - h_3}{h_1 - h_2}$$

Note ① The dryness fraction of steam at B' is greater than that at B., i.e., effect of friction is to increase dryness of steam. Becoz the energy lost in friction is transferred into heat which tends to dry or superheat the steam.

② Similar effect produce when the steam is superheated at the entrance of nozzle. $\left[K = \frac{PR}{PQ}\right]$.

③ In general, if 15% of the heat drop is lost in friction, then η of nozzle is 85%.

Velocity of steam through a Nozzle :-

Consider a unit mass flow of steam through a Nozzle.

Let V_1 = vel. at entrance of nozzle in m/s.
V_2 = Vel. at any section
h_1 = enthalpy at entrance
h_2 = " " any section

Steady Flow process for in nozzle, for unit mass flow,

$$Q_{(1-2)} - W_{(1-2)} = (h_2 - h_1) + (KE_2 - KE_1) + (PE_2 - PE_1) = 0 \quad \begin{bmatrix} Q = 0 \\ W = 0 \\ PE = 0 \end{bmatrix}$$

$$(h_2 - h_1) + (KE_2 - KE_1) = 0$$

$$\frac{1}{1000}\left[\frac{V_2^2}{2} - \frac{V_1^2}{2}\right] = h_1 - h_2 \quad \text{(or)} \quad \because m_1 = 1\,kg$$

$$h_1 + \frac{1}{1000}\left(\frac{V_1^2}{2}\right) = h_2 + \frac{1}{1000}\left(\frac{V_2^2}{2}\right) + \text{losses}.$$

Neglecting losses,

$$(KJ)\ \frac{1}{1000}\left(\frac{V_2^2}{2} - \frac{V_1^2}{2}\right) = h_1 - h_2$$

$$V_2 = \sqrt{V_1^2 + 2000\,(h_1 - h_2)} = \sqrt{V_1^2 + 2000\,h_d}$$

$[\because h_1 - h_2 = h_d]$

$\because$ entrance velocity or velocity of approach (V_1) is negligible as compared to V_2, therefore from the eq.

$$V_2 = \sqrt{2000\,h_d} = 44.72\sqrt{h_d}$$

Note: In actual practice, there is always a certain amount of friction present b/w the steam & nozzle surfaces. This reduces the heat drop by 10 to 15 percent & thus the exit velocity of steam is also reduced correspondingly.

[k = nozzle η or coefficient

$$\therefore V_2 = 44.72\sqrt{k\,h_d}$$

① Dry saturated steam at 5 bar with negligible velocity expands isentropically in a convergent nozzle to 1 bar & dryness fraction 0.94. Determine the velocity of steam leaving the nozzle.

Sol: Given $P_1 = 5$ bar, $P_2 = 1$ bar $x_2 = 0.94$

From ST, At 5 bar

$$h_1 = h_{g_1} = 2747.5 \text{ KJ/kg}$$

At 1 bar, $h_{f_2} = 417.5$; $h_{fg_2} = 2257.9$ KJ/kg.

Enthalpy or total heat of final steam,

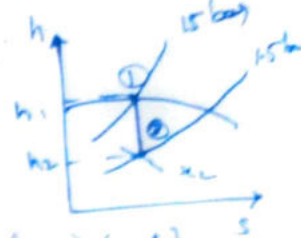
$$h_2 = h_{f_2} + x_2 h_{fg_2} = 417.5 + 0.94 \times 2257.9 = 2540 \text{ KJ/kg}$$

$$h_d = h_1 - h_2 = 2747.5 - 2540 = 207.5 \text{ KJ/kg}.$$

$$\therefore V_2 = 44.72\sqrt{h_d} = \underline{644 \text{ m/s}}.$$

(2) Dry saturated steam at a pressure of 15 bar enters in a nozzle & is discharged at a pressure of 1.5 bar. Find the final velocity of steam when V_1 is negligible. If 10% of heat drop is lost in friction find the % reduction in final velocity.

X



Sol: $P_1 = 15$ bar ; $P_2 = 1.5$ bar

From ST, At 15 bar , $h_1 = 2789.9$

At 1.5 bar , $h_2 = \cancel{2693.4}$ (~~superheated~~) (wet) ~~Dry saturated~~

Heat drop $hd = h_1 - h_2 = 2789.9 - 2693.4 = 96.5$ kJ/kg

$\therefore V_2 = 44.72\sqrt{hd} = \cancel{439.3}$ m/s. & 903.6 m/s.

Percentage reduction in final velocity.

we know that heat drop lost in friction $= 10\% = 0.1$

$\therefore$ Nozzle coefficient $k = 1 - 0.1 = 0.9$

$\therefore \cancel{V_2 = \sqrt{44.72 \times 0.9}}$ $\quad V_2 = 44.72 \times \sqrt{0.9 \times 96.5}$

$V_2 = 416.8$ m/s

$\therefore$ Percentage reduction in final velocity, $= \dfrac{4\cancel{39}.3 - \cancel{416.8}}{\cancel{439.3}}$

$= 0.051$

$= 5.1\%$ ✓

(3) Dry saturated steam at 10 bar is expanded isentropically in a nozzle to 0.1 bar. Using steam tables only, find the dryness fraction of the steam at exit. Also find the velocity of steam leaving the nozzle when (1) initial velocity is negligible & (2) initial velocity of the steam is 135 m/s

Sol: $P_1 = 10$ bar

$P_2 = 0.1$ bar

At $P_1 = 10$ bar $S_1 = S_{g_1} = 6.583$

$S_{f_2} = 0.649$; $S_{fg_2} = 7.502$

$\therefore S_1 = S_2$

$6.583 = S_{f_2} + x_2 S_{fg_2}$

$x_2 = 0.791$

① Velocity of steam leaving nozzle when initial velocity is negligible.

From ST,

(10) $h_1 = h_{g_1} = 2776.2$

(0.1) $h_{f_2} = 191.8$ $h_{fg_2} = 2392.9$

$h_2 = h_{f_2} + x_2 h_{fg_2} = 2084.6$

$h_d = h_1 - h_2 = 691.6$

$V_2 = 44.72\sqrt{h_d} = 1176$ m/s.

② Velocity of steam leaving nozzle when $V_1 = 135$ m/s.

$V_2 = \sqrt{V_1^2 + 2000 h_d} = 1184$ m/s.

④ Dry saturated steam at a pressure of 10 bar is expanded in a nozzle to a pressure of 0.7 bar. With the help of Mollier diagram find the velocity & dryness fraction of steam issuing from the nozzle, if the friction is neglected.

Also find the velocity of steam ~~leaving the nozzle~~ & dryness fraction of the steam if 15% of the heat drop is lost in friction.

Sol: $P_1 = 10$ bar, $P_2 = 0.7$ bar

velocity & dryness fraction, if friction is neglected.

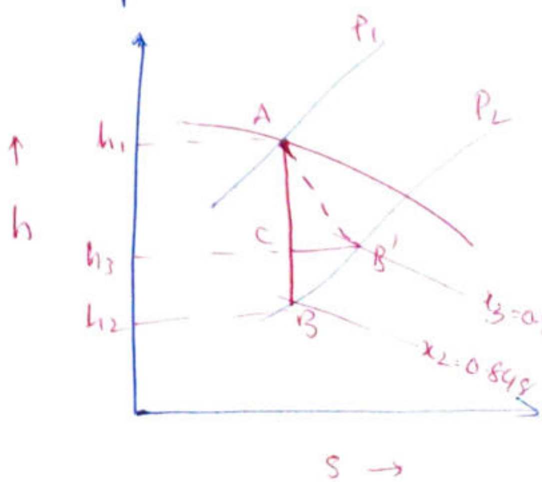
$h_1 = 2772$ KJ/kg

$h_2 = 2310$

$h_d = h_1 - h_2 = 462$

$V_2 = 44.72\sqrt{h_d} = 961$ m/s.

From MD, $x_2 = 0.848$



velocity & dryness fraction of steam, if 15% heat drop is lost in friction :

$\therefore\ K = 100 - 15 = 85\% = 0.85$

Heat drop due to friction $= 462 \times 0.15 = 69.3$ kJ/kg

we know that, $V_2 = 44.72\sqrt{K h_d}$

$= 44.72\sqrt{0.85 \times 462} = 886$ m/s.

Now,

1) locate point C on MD, on vertical line AB, such that BC = 69.3

2) Draw a horizontal line CB' to meet the final pressure line 0.7 bar at B'

then $x_3 = 0.878$. at B'

Mass of steam discharged through the nozzle :

Flow of steam through the nozzle is isentropic, which is approximately represented by general law :

$pv^n =$ constant.

Gain in Kinetic energy $= \frac{V_2^2}{2}$ (neglecting initial velocity)

Heat drop = workdone during Rankine Cycle.

$$= \frac{n}{n-1}(P_1 V_1 - P_2 V_2)$$

$\therefore$ Kinetic energy is equal to heat drop.

$$\therefore\ \frac{V_2^2}{2} = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

$$= \frac{n}{n-1} P_1V_1\left[1 - \frac{P_2V_2}{P_1V_1}\right] \longrightarrow (1)$$

we know, $P_1 V_1^n = P_2 V_2^n$

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

$$V_2 = V_1\left(\frac{P_2}{P_1}\right)^{-1/n} \longrightarrow (2)$$

Substitute the value in eq (1)

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

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

$$V_2 = \sqrt{2 \times \frac{n}{n-1} \times P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]} \longrightarrow (3)$$

Now,

The vol. of steam flowing /sec = Cross-sectional area of nozzle × velocity of steam $= AV_2$ m^3/s

vol. of 1 kg of steam i.e., sp. vol. of steam at press. P_2

$= v_2$ m^3/kg.

∴ Mass of steam discharged /sec

$$m = \frac{\text{vol. of steam flowing /sec}}{\text{vol. of 1kg of steam at } P_2} = \frac{AV_2}{v_2}$$

$$= \frac{A}{v_2}\sqrt{2 \times \frac{n}{n-1} P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]} \longrightarrow (4) \quad kg/s$$

$$v_2 = v_1 \left(\frac{P_2}{P_1}\right)^{-\frac{1}{n}}$$

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

Substitute eq (2) in (4)

$$\therefore \cancel{m = \frac{A}{v_1}\left(\frac{P_1}{P_2}\right)^{-\frac{1}{n}} \sqrt{\frac{2n}{n-1} \times P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}} \times$$

$$= \frac{A}{v_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \sqrt{\frac{2n}{n-1} \times P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= A\sqrt{\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} \times \frac{2n}{n-1} \times \frac{P_1 V_1}{v_1^2} \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

pressure should be in N/m^2

$$\boxed{m = A\sqrt{\frac{2n}{n-1} \times \frac{P_1}{V_1} \left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}} \longrightarrow (i) \quad kg/s$$

(5) Dry air at a temp of 27°C & pressure of 20 bar enters a nozzle and leaves at a pressure of 4 bar. Find the mass of air discharged, If the area of the nozzle is 200 mm^2.

For air $n = 1.4$

Sol:- $T_1 = 27^\circ C = 300\ K$

$P_1 = 20\ bar = 20\times10^5\ N/m^2$

$P_2 = 4\ bar = 4\times10^5\ N/m^2$

$A = 200\ mm^2 = 200\times10^{-6}\ m^2$

$R = 287\ J/kg\ K$

$P_1 v_1 = mRT_1$

$v_1 = \frac{mRT_1}{P_1} = \frac{1\times287\times300}{20\times10^5}$

$v_1 = 0.043\ m^3/kg$.

$n = 1.4$

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

$m = 0.02\ kg/s$.

Condition for Maximum discharge through a Nozzle (Critical Pressure Ratio)

A nozzle is normally, designed for maximum discharge by designing a certain throat pressure which produces this condition.

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

There is only one value of the ratio P_2/P_1, which produces maximum discharge from the Nozzle. This ratio P_2/P_1, is obtained by differentiating the right hand side of the equation. We see from this equation that except P_2/P_1, all other values are constant. Therefore, only that portion of the equation which contains P_2/P_1, is differentiated & equated to zero for maximum discharge.

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

$\frac{d}{dx}x^n = nx^{n-1}$

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

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

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

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

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

Note :-

1) The ratio P_2/P_1 is known as Critical Pressure ratio & the pressure P_2 at the throat is known as Critical pressure.

2) The maximum value of the discharge per second is obtained by substituting the value of P_2/P_1 in eq ①

Maximum discharge $m_{max} = A\sqrt{\frac{2n}{n-1} \times \frac{P_1}{V_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$

$$= A\sqrt{\frac{2n}{n-1} \times \frac{P_1}{V_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} \cdot \left[1 - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1} - \frac{2}{n-1}}\right]}$$

$$= A\sqrt{\frac{2n}{n-1} \times \frac{P_1}{V_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} \left[1 - \left(\frac{2}{n+1}\right)\right]}$$

$$= A\sqrt{\frac{2n}{n-1} \times \frac{P_1}{V_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} \left(\frac{n-1}{n+1}\right)}$$

$$\boxed{m_{max} = A\sqrt{\frac{2n}{n+1} \times \frac{P_1}{V_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}}}$$

③ The eq. derived above are true for gases also.

Values for Maximum discharge through a Nozzle :-

1) when the steam is initially dry satured.

For dry saturated $n = 1.135$

$$m_{max} = 0.637\, A\sqrt{\frac{P_1}{V_1}}$$

2) when the steam is initially superheated,

for superheated steam $n = 1.3$

$$m_{max} = 0.666\, A\sqrt{\frac{P_1}{V_1}}$$

For wet steam

$n = 1.11$

$$m_{max} = 0.63\, A\sqrt{\frac{P_1}{V_1}}$$

3) For gases, $n = 1.4$

$$m_{max} = 0.685\, A\sqrt{\frac{P_1}{V_1}}$$

⑥ steam at a pressure of 10 bar & 210°C is supplied to a convergent divergent nozzle with a throat area of 1500 mm². The exit is below critical pressure. Find the coefficient of discharge, if the flow is 7200 kg of steam per hour. (26)

Sol:- $P_1 = 10$ bar $= 10\times10^5$ N/m^2

$T_1 = 210°C = 483$ K

$A = 1500$ mm^2 $= 1500\times10^{-6}$ m^2

$m = 7200$ kg/h $= 2$ kg/s.

From ST, $P_1 = 10$ bar, $210°C$; $v_1 = 0.2113$ m^3/kg.

$n = 1.3$ (superheated)

$$m_{max} = A\sqrt{\frac{2n}{n+1}\times\frac{P_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}}$$

$$= 1500\times10^{-6}\sqrt{\frac{2\times1.3}{1.3+1}\times\frac{10\times10^5}{0.2113}\left(\frac{2}{1.3+1}\right)^{\frac{2}{1.3-1}}}\ \text{kg/s.}$$

$m_{max} = 2.17$ kg/s.

$$\text{Coefficient of discharge} = \frac{\text{Actual discharge}}{\text{Max. discharge}} = \frac{2}{2.17} = \underline{\underline{0.922}}$$

(or) $m_{max} = 0.666\,A\sqrt{\frac{P_1}{v_1}}$

Values for Critical Pressure Ratio:-

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

1) Dry saturated ; $n = 1.135 \Rightarrow \frac{P_2}{P_1} = 0.577 \Rightarrow P_2 = 0.577\,P_1$

2) Superheated ; $n = 1.3 \Rightarrow P_2 = 0.546\,P_1$

3) Steam is initially wet, $P_2 = 0.582\,P_1$; $n = 1.11$

4) For gases : $n = 1.4 \Rightarrow P_2 = 0.528\,P_1$

Velocity in terms of density,

$$V_2 = \sqrt{2\left(\frac{n}{n+1}\right)\frac{P_2}{\rho_2}\left(\frac{n+1}{2}\right)} \Rightarrow \boxed{V_2 = \sqrt{\frac{n P_2}{\rho_2}}}$$

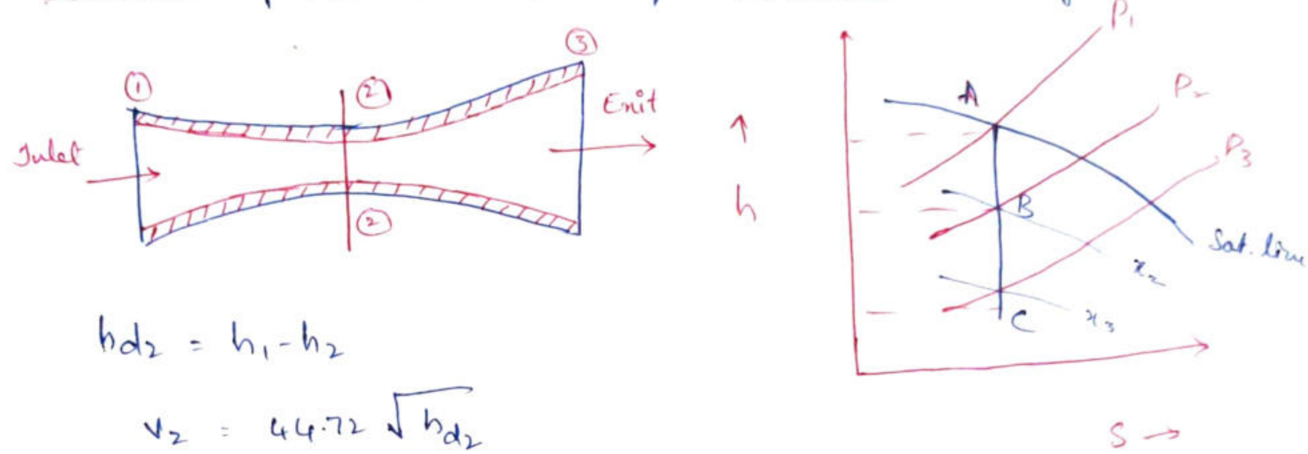
This is the value of velocity of sound in the medium at pressure P_2 & is known as sonic velocity.

1) The critical pressure gives the velocity of steam at the throat equal to the velocity of sound.

2) The flow in the convergent portion of the nozzle is sub-sonic & in the divergent portion it is supersonic.

3) To increase the velocity of steam above sonic velocity (supersonic) by expanding steam below the critical pressure, the divergent portion for the nozzle is necessary.

Diameters of Throat & exit for Maximum Discharge:



$$h_{d_2} = h_1 - h_2$$

$$V_2 = 44.72\sqrt{h_{d_2}}$$

$$m = \frac{\text{vol. of steam flowing at throat}}{\text{vol. of 1 kg of steam at press. } P_2} = \frac{A_2 V_2}{v_2} = \frac{A_2 V_2}{x_2 v_{g_2}}$$

Similarly, for exit conditions,

$$\boxed{m = \frac{A_3 V_3}{x_3 v_{g_3}} = \frac{A_2 V_2}{x_2 v_{g_2}}}$$

7) Steam enters a group of nozzles of a steam turbine at 12 bar & 220°C & leaves at 1.2 bar. The steam turbine develops 220 kW with a sp. steam consumption of 13.5 kg/kWh. If the diameter of nozzles at throat is 7 mm, calculate the no. of nozzles.

Sol) Given, $P_1 = 12$ bar, $T_1 = 220°C$; $d_2 = 7$ mm

$P_3 = 1.2$ bar, Power $= 220$ kW, $m_s = 13.5$ kg/kWh

we know that for superheated steam,

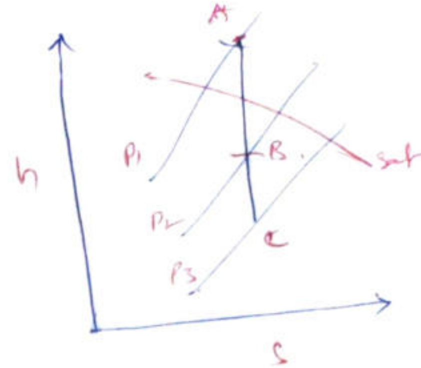
Pressure of steam at throat $P_2 = 0.546\, P_1$

$= 0.546 \times 12 = 6.552$ bar

From h-s diagram,

$h_1 = 2860$ kJ/kg at P_1

at P_2 ; $h_2 = 2750$ kJ/kg



& dryness fraction of steam at throat,

$x_2 = 0.992$

From ST, at $P_2 = 6.552$ bar

$v_{g2} = 0.29$ m^3/kg.

we know that heat drop from entrance to throat

$hd_2 = h_1 - h_2 = 2860 - 2750 = 110$ kJ/kg.

∴ velocity of steam at throat,

$V_2 = 44.72\sqrt{hd_2} = 44.72\sqrt{110} = 470$ m/s.

Area of nozzle at throat,

$A_2 = \frac{\pi}{4}(d_2)^2 = \frac{\pi}{4} \times 7^2 = 38.5$ mm^2 $= 38.5 \times 10^{-6}$ m^2

∴ Mass flow rate per nozzle,

$m = \frac{A_2 V_2}{v_2} = \frac{A_2 V_2}{x_2 v_{g2}} = \frac{38.5 \times 10^{-6} \times 470}{0.992 \times 0.29} = 0.063$ kg/s

$\left[P = \frac{\text{Total mass flow rate}}{\text{steam consumption}} \right]$

we know that total mass flow rate

$= 13.5 \times 220 = 2970$ kg/h $= 0.825$ kg/s.

∴ No. of nozzles $= \frac{\text{Total mass flow rate}}{\text{mass flow rate/nozzle}} = \frac{0.825}{0.063} = 13.1$ say $= 14$

(3) Dry saturated steam enters a nozzle at a pressure of 10 bar & with an initial velocity of 90 m/s. The outlet pressure is 6 bar & the outlet velocity is 435 m/s. The heat loss from the nozzle is 9 kJ/kg of steam flow. Calculate the dryness fraction & area at the exit if the area at inlet is 1256 mm^2.

Sol:- Given, $P_1 = 10$ bar   losses $= 9$ KJ/kg.

$v_1 = 90$ m/s $A_1 = 1256$ mm^2

$P_3 = 6$ bar $= 1256 \times 10^{-6}$ m^2

$V_3 = 435$ m/s

Dryness fraction of steam, x_3

From ST, at $P_1 = 10$ bar

$h_1 = 2776.2$ KJ/kg, $v_{g_1} = \cancel{0.1443}$ $\cancel{0.9143}$ 0.1943 m^3/kg.

at $P_3 = 6$ bar,

$h_{f_3} = 670.4$ KJ/kg, $h_{fg_3} = 2085$ KJ/kg, $v_{g_3} = 0.3155$ m^3/kg.

we know that, for a steady flow through the nozzle.

$$h_1 + \frac{v_1^2}{2000} = h_3 + \frac{V_3^2}{2000} + \text{losses}$$

$$h_3 = h_1 + \frac{v_1^2 - V_3^2}{2000} - \text{losses.}$$

$$= 2776.2 + \frac{(90)^2 - (435)^2}{2000} - 9$$

$$h_3 = 2676.6 \text{ KJ/kg.}$$

$\therefore$ Enthalpy of wet steam $h_3 = h_{f_3} + x_3 h_{fg_3}$

$$2676.6 = 670.4 + x_3 (2085)$$

$$\underline{x_3 = 0.962}$$

Area at exit, A_3

we know, $\frac{A_1 V_1}{x_1 v_{g_1}} = \frac{A_3 V_3}{x_3 v_{g_3}}$

$$\frac{1256 \times 10^{-6} \times 90}{1 \times \cancel{0.1443}\ 0.1943} = \frac{A_3 \times 435}{0.962 \times 0.3155}$$

$\therefore A_3 = \cancel{406 \times 10^{-6}}$ m^2 $\cancel{8.5 \times 10^{-5}}$ m^2

$= \cancel{406 \text{ mm}^2}$

$\underline{406 \text{ mm}^2}$

Q) A convergent - divergent nozzle is required to discharge 2 kg of steam per second. The nozzle is supplied with steam at 7 bar & 180° C & discharge takes place against a back pressure of 1 bar. The expansion upto throat is isentropic and the frictional resistance b/w throat & exit is equivalent to 63 KJ/kg of steam. Taking approach velocity of 75 m/s & throat pressure of 4 bar, estimate ① suitable areas at throat and exit ② Overall η of the nozzle based on the enthalpy drop b/w ~~actual~~ actual inlet pressure & temp & exit pressure.

Sol: Given, $m = 2$ kg/s   frictional resistance $= 63$ KJ/kg of steam

$P_1 = 7$ bar $v_1 = 75$ m/s

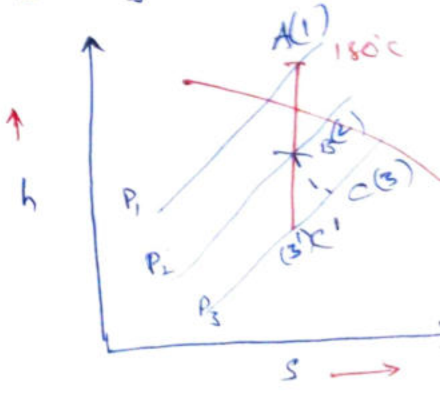
$T_1 = 180°C$

$P_3 = 1$ bar

$P_2 = 4$ bar

① Suitable areas for throat & exit, A_2 & A_3

From Mollier diagram,



$h_1 = 2810$ $x_2 = 0.97$

$h_2 = 2680$ $x_3 = 0.934$

$h_3' = 2470$

From ST, at $P_2 = 4$ bar

$v_{g_2} = 0.462$ m³/kg.

at $P_3 = 1$ bar, $v_{g_3} = 1.694$ m³/kg.

∴ Heat drop b/w entrance & throat

$h_{d_2} = h_1 - h_2 = 2810 - 2680 = 130$ KJ/kg

∴ velocity of steam at throat

$$v_2 = \sqrt{v_1^2 + 2000\, h_d} = \sqrt{(75)^2 + (2000 \times 130)}$$

$v_2 = 515$ m/s.

$$\therefore \quad m = \frac{A_2 V_2}{x_2 V_{g_2}} \Rightarrow 2 = \frac{A_2 \times 515}{0.97 \times 0.462}$$

$$\therefore A_2 = 1.74 \times 10^{-3} \ m^2 = 1740 \ mm^2$$

Since there is a frictional resistance of 63 kJ/kg of steam b/w the throat & exit, therefore

$$h_3 - h_3' = 63 \Rightarrow h_3 = 63 + h_3'$$

$$= 63 + 2470$$

$$h_3 = 2533 \ kJ/kg.$$

$\therefore$ heat drop b/w entrance & exit $= h_1 - h_3$

$$hd_3 = 2810 - 2533 = 277 \ kJ/kg.$$

$\therefore$ velocity at exit $V_3 = \sqrt{V_1^2 + 2000 \times hd_3}$

$$= \sqrt{(75)^2 + (2000 \times 277)}$$

$$V_3 = 748 \ m/s.$$

and $m = \dfrac{A_3 V_3}{x_3 V_{g_3}} \Rightarrow A_3 = \dfrac{m x_3 V_{g_3}}{V_3} = \dfrac{2 \times 0.934 \times 1.694}{748}$

$$A_3 = 4.23 \times 10^{-3} \ m^2 = 4230 \ mm^2$$

(2) Overall η of nozzle,

$$\eta = \frac{\text{Useful heat drop}}{\text{Isentropic heat drop}} = \frac{h_1 - h_3}{h_1 - h_3'} = \frac{2810 - 2533}{2810 - 2470}$$

$$\eta = 0.815 = 81.5 \%.$$

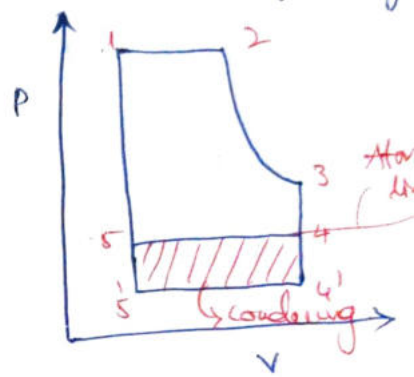
STEAM CONDENSERS

A steam condenser is a device in which steam condenses & heat released by steam is absorbed by water.

It serves the following purposes:

1) It maintains a very low back pressure on the exhaust side of piston of steam engine or turbine. Consequently steam expands to a greater extent which results in an increase in available heat energy for converting into mechanical work. The shaded area shows the increase in work obtained by fitting a condenser. η_{th} also increases.

2) It supplies to the boiler pure & hot feed water as the condensed steam which is discharged from the condenser & collected in a hot well, can be used as a feed water for the boiler.



Vacuum

Vacuum is sub atmospheric pressure. It is recorded as the pressure depression below atm. pressure. The condensation of steam in a closed vessel produces a partial vaccum by the reason of great reduction in the volume of the low pressure steam or vapour. The back pressure in the steam engine or steam turbine can be lowered from 1.013 to 0.2 bar abs. or even less.

Steam engines → 0.2 → 0.28 bar (back pressure)

steam turbines → about 0.025 bar abs.

Rare fraction: Most of the time, rarefraction refers to air or other gases becoming less dense. When rare fraction occurs, particles in steam becomes more spread out. The areas of lower density are called rarefractions.

Requirements of Steam Condensing plant:

It consists of

① Condenser (to condense the steam)

② Supply of cooling water (or injection)

③ wet air pump (To remove condensed steam, air & uncondensed water vapour & gases from condenser; separate pumps may be used to deal with air & condensate.)

④ Hot well

⑤ Arrangement for recooling the cooling water in case surface condenser is employed.

Advantages:

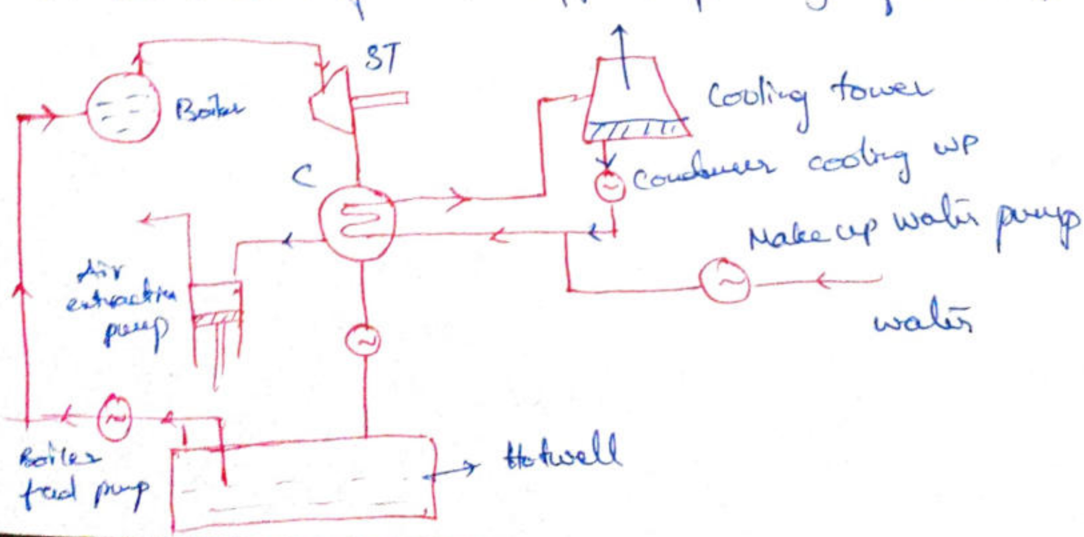
→ It increases expansion ratio of steam & thus increases η of the plant.

→ It reduces back pressure of steam, & thus more work can be obtained.

→ It reduces temp of the exhaust steam & thus more work can be obtained.

→ The reuse of condensate as feed water for boiler reduces the cost of power generation.

→ The temp of condensate is higher than that of fresh water.
∴ The amount of heat supplied per kg of steam is reduced.



Classification of Condensers

1) Jet condensers (or) mixing type condensers
2) Surface condensers (or) non-mixing type condensers.

Jet Condensers

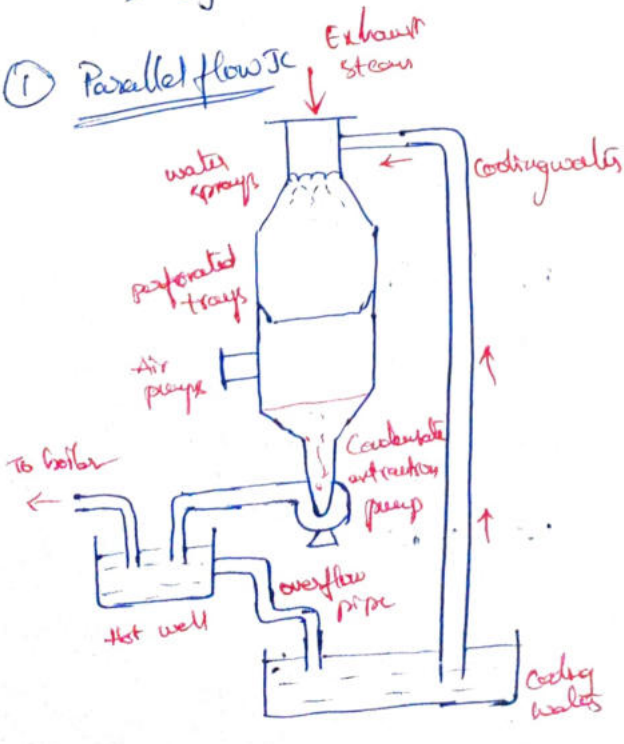
These are rarely used becoz there is some loss of condensate during the process of condensation & high power requirements for the pumps used. Moreover the condensate cannot be used as feed water to the boiler as it is not free from salt.

Jet condensers can be used at places where good quality of water is easily available in sufficient quantity.

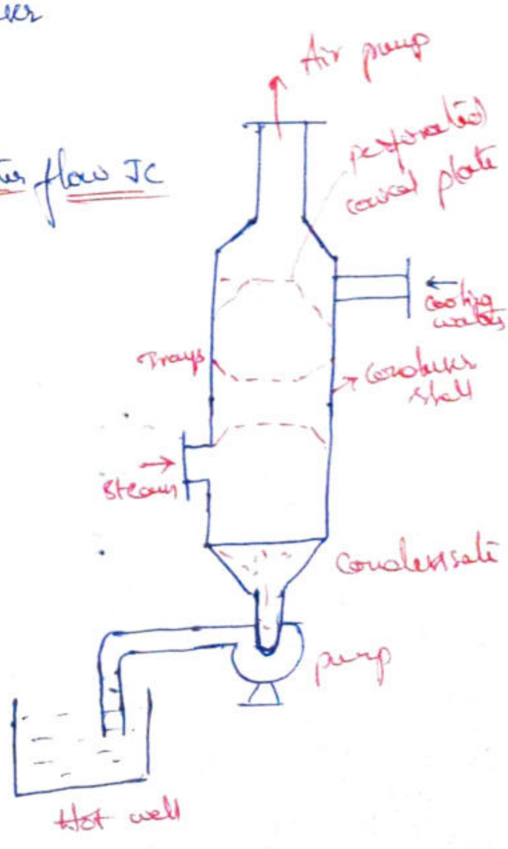
Types of Jet condensers:

1) Parallel flow jet condenser (low level)
2) counter flow (or) low level jet condenser
3) Barometric (or) high level jet condenser
4) Ejector condenser

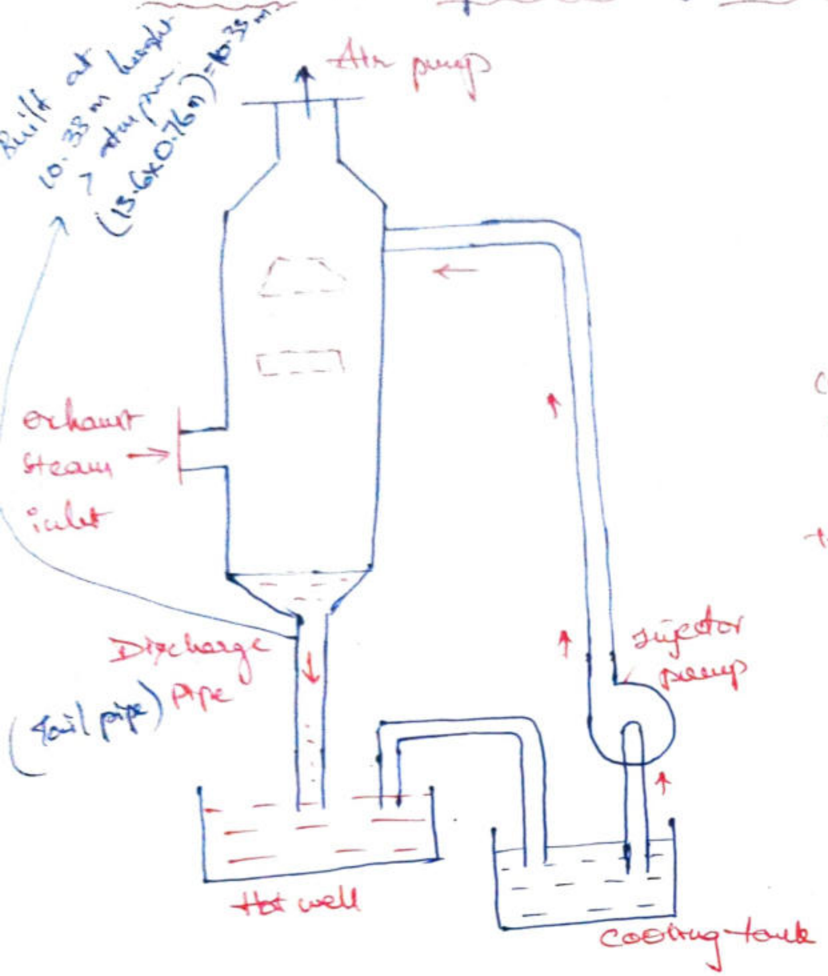
① Parallel flow JC



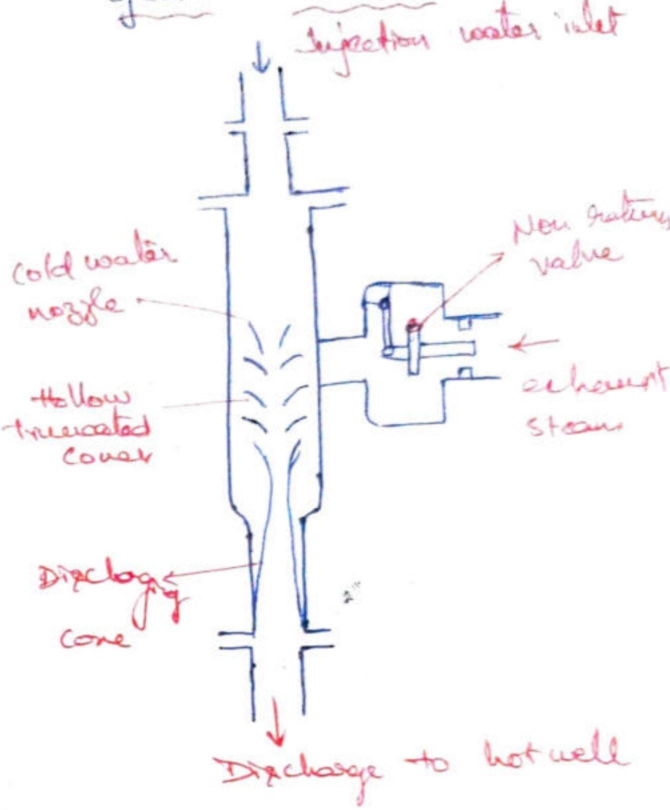
② Counter flow JC



Barometric (or) High level jet condenser > 10 m

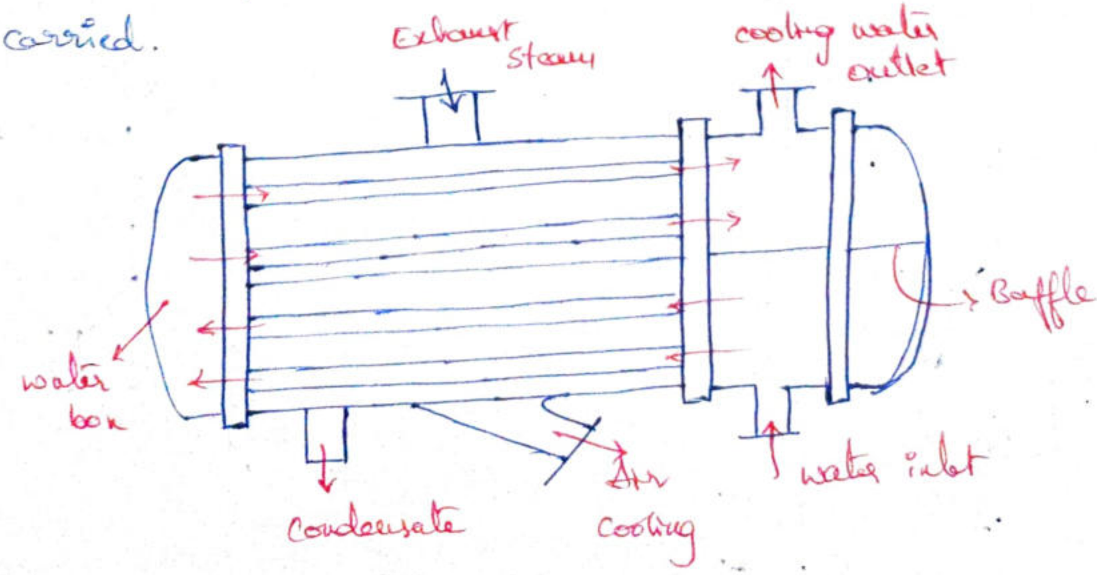


Ejector Condenser



Surface Condensers :-

A surface condenser has a great advantage over the jet condensers, as the condensate does not mix up with cooling water. whole condensate can be reused. This type of condensers is essential in ships where limited quantity of fresh water can be carried.



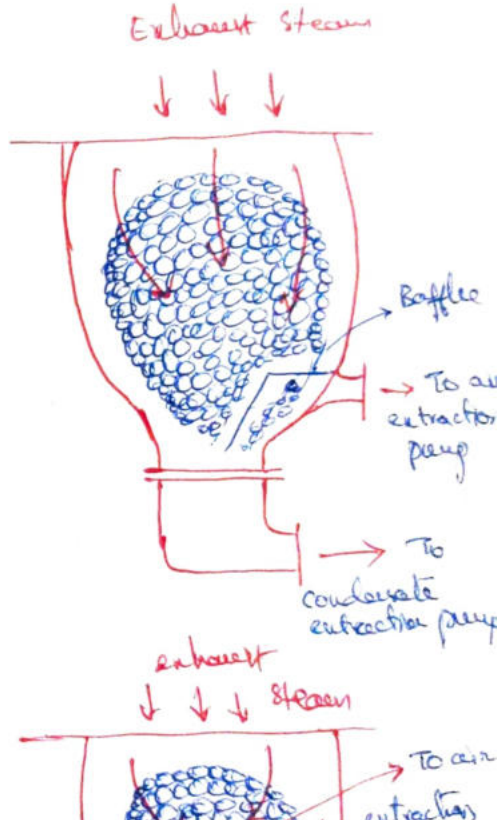
The water tubes pass horizontally through the main condensing space for the steam. The steam enters at the top & is forced to flow downwards over the ~~surface~~ tubes due to the suction of the extraction pump at bottom.

The cooling water flows in one direction through the lower half of the tubes & return in opposite direction through the upper half as shown in fig.

Types of Surface Condensers:-

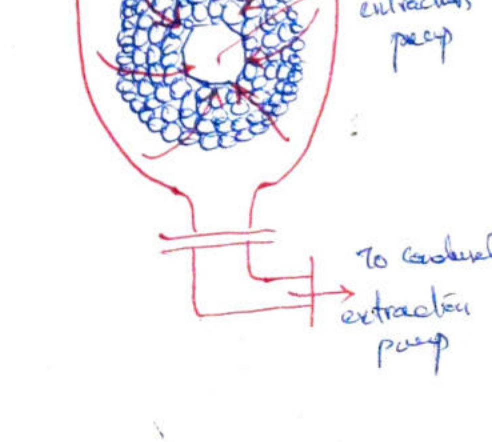
1) Down flow surface Condenser
2) Central " "
3) Regenerative " "
4) Evaporative condenser.

① Down flow Surface condenser.



As the steam flows perpendicular to the direction of flow of cooling water (inside the tubes), this is also called a cross surface condenser.

② Central flow Surface Condensers.



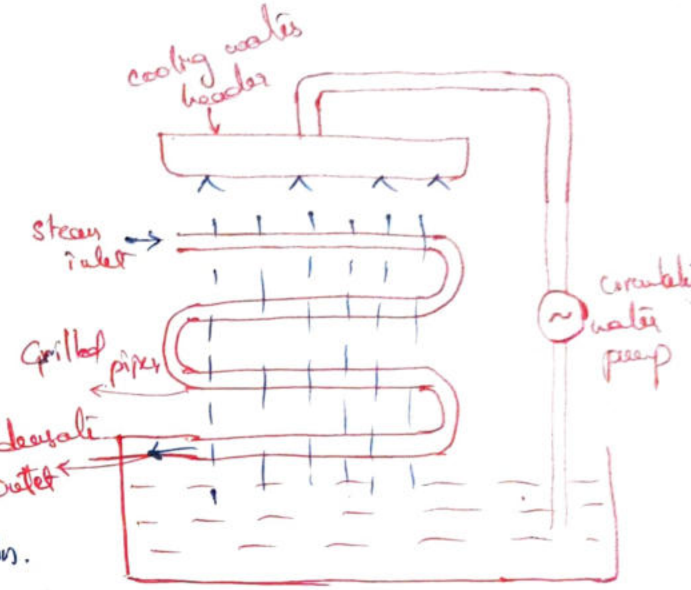
The central flow surface condenser is an improvement over the down flow type as the steam is directed radially inwards by a volute casing around the tube nest. It thus, gives an access to the whole periphery of the tubes.

Regenerative Surface Condenser

In this type, the condensate is heated by a regenerative method. The condensate after leaving the tubes is passed through the exhaust steam from the engine or turbine. It thus raises its temp for use as feed water for the boiler.

Evaporative Condenser

→ Steam enters at top
→ film of cold water is falling
→ Same time current of air circulates over the water film, causing rapid evaporation of water
→ Steam gets condensed.
→ Circulating water is reused again.



Comparison of Jet & Surface Condensers :-

Jet condenser	Surface Condenser
① cooling water & steam are mixed up	① Cooling water & Steam are not mix up.
② Less suitable for high capacity plants.	② More suitable for high capacity plants.
③ Condensate is wasted	③ condensate is reused.
④ It requires less qty of water	④ Requires more qty of circulating water
⑤ It is economical & simple	⑤ It is costly & complicated
⑥ Maintenance cost is low	⑥ Maintenance cost is high
⑦ More power is required for air pump	⑦ Less power is required for air pump
⑧ High power is required for water pumping.	⑧ Less power is required for water pumping.

Mixture of air & Steam (Dalton's law of partial Pressures)

"The pressure of mixture of air & steam is equal to the sum of the pressures, which each constituent would exert, if it occupied the same space by itself."

$\therefore$ Pressure in the condenser $\boxed{P_c = P_a + P_s}$

(or) P_c = Barometer reading - Vaccum reading

P_a = partial pres. of air

P_s = partial pres. of steam

Units :- 1 bar = 10^5 N/m²

1 mm of Hg = 0.00133 bar

760 mm of Hg = 1.013 bar

1 mm of Hg = 0.133 KPa

~~Vaccum eff~~

① The following observation were recorded during a condenser test. vaccum reading = 700 mm of Hg, Barometer reading = 760 mm of Hg. condensate temp = 34°C. Find ① Partial pressure of air ② Mass of air per m³ of condenser volume.

Sol. Given vaccum reading = 700 mm of Hg

Barometer " = 760 mm of Hg

$T = 34°C = 307\,K$; $V = 1\,m^3$

① Partial pressure of air

we know $P_c = P_a + P_s \Rightarrow P_a = P_c - P_s$

P_c = Barometer reading - vaccum reading

= 760 - 700 = 60 mm of Hg

= 60 × 0.00133 = 0.0798 bar.

From ST, at 34°C P_s = 0.0532 bar

$\therefore P_a = P_c - P_s = 0.0798 - 0.0532 = \underline{0.0266}$ bar

② Mass of air $\frac{m_a}{V} = \frac{P_a}{RT_a} = \frac{0.0266 \times 10^5 \times 1}{287 \times 307} = \underline{0.03\ kg/m^3}$

Vaccum efficiency :-

The ratio of actual vaccum to the ideal vaccum is known as vaccum efficiency.

$$\eta_v = \frac{\text{Actual vaccum}}{\text{Ideal vaccum}}$$

Actual vaccum = Barometric pressure - Actual pressure

Ideal vaccum = Barometric pressure - Ideal pressure.

(2) Calculate the vaccum efficiency if vaccum at steam inlet to condenser = 700 mm of Hg, Barometer reading : 760 mm of Hg, Hot well temp = 30°C

Sol:- Given Actual Vaccum or actual reading : 700 mm of Hg.

Bar. reading = 760 mm of Hg

$t = 30°C$

$P_C = 760 - 700 = 60$ mm of Hg.

From ST, at 30°C, Ideal pressure = 0.0424 bar

= 31.88 mm of Hg.

∴ Ideal vaccum = Bar. reading - Ideal press.

= 760 - 31.88 = 728.12 mm of Hg.

$$\therefore \eta_v = \frac{\text{Actual vaccum}}{\text{Ideal vaccum}} = \frac{700}{728.12} = 96.14\ \%$$

(3) In a surface condenser, the vacuum maintained is 700 mm of Hg. The barometer reads 754 mm of Hg. If the temp is 18°C, determine (1) Mass of air/sec (2) η_{vacuum}.

Sol:- Given Actual vaccum = 700 mm of Hg

bar. read = 754 "

$t = 18°C$

we know $P_c = \cancel{759}\ 754 - 700 = 54$ mm of Hg

From ST, at 18°C

$P_s = 0.0206$ bar $= 15.5$ mm of Hg.

Sp. vol of steam : $V_s = 65.09\ m^3/kg$.

(1) Mass of air/sec $M_a = \frac{P_a v}{RT}$

$P_a = P_c - P_s = 54 - 15.5 = 38.5$ mm of Hg.
$= 0.0512$ bar
$= 0.0512 \times 10^5\ N/m^2$

$\therefore M_a = \frac{0.0512 \times 10^5 \times 65.09}{287 \times 291} = 4\ kg/s.$

(2) vacuum efficiency.

$$\eta_v = \frac{\text{Actual vacuum}}{\text{Ideal vacuum.}}$$

Ideal vacuum = Bar. reading − Ideal press.
$= 754 - 15.5 = 738.5$ mm of Hg

$$\eta_v = \frac{700}{738.5} = 94.8\ \%.$$

(4) The air leakage into a surface condenser operating with a steam turbine is estimated as 84 kg/h. The vacuum near the inlet of air pump is 700 mm of Hg, when barometer reads 760 mm of Hg. The temp at inlet of vacuum pump is 20°C. Calculate (1) Minimum capacity of air pump in m^3/h (2) The dimension of reciprocating air pump to remove the air if it runs at 200 rpm. Take L/D ratio = 1.5 & volumetric efficiency = 100%. (3) The mass of vapour extracted per minute.

Sol:- Given, $m_a = 84$ kg/h
Vacuum = 700 mm of Hg.
Barometer = 760 " "
$T = 20°C = 293\ K$

(1) Minimum Capacity of air pump.

$$V_a = \frac{M_a RT}{P_a}$$

$P_a = P_c - P_s$

~~Pa = Barometer~~

P_c = Barometer − Vacuum

$= 760 - 700 = 60$ mm of Hg $= 0.0798$ bar

From ST, at $20^\circ C$

$P_s = 0.0234$ bar

$P_a = P_c - P_s = 0.0798 - 0.0234 = 0.0564$ bar

$= 5640 \; N/m^2$

$\therefore \; V_a = \dfrac{m_a R T}{P_a} = \dfrac{84 \times 287 \times 293}{5640} = \underline{\underline{1252.4 \; m^3/h}}$

(2) Dimensions of the reciprocating pump.

D = dia , $L = 1.5 D$, $\eta_v = 100\% = 1$; $N = 200$ rpm

we know, minimum capacity of air pump

$V_a = \dfrac{\pi}{4} d^2 l N$

$\dfrac{1252.4}{60} = \dfrac{\pi}{4} d^2 \times 1.5 d \times 200 \Rightarrow d = \underline{\underline{0.446}}$ m

$L = 1.5 D = \underline{\underline{0.669}}$ m.

(3) Mass of vapour extracted per minute

From ST, at $20^\circ C$ $\quad v_g = 57.84 \; m^3/kg$.

$\therefore$ Mass of vapour extracted /min $= \dfrac{V_a}{v_g} = \dfrac{1252.4}{60 \times 57.84}$

$m_v = \underline{\underline{0.361}}$ kg/min.

IMPULSE TURBINE

Advantages of steam turbines over steam Engines:-

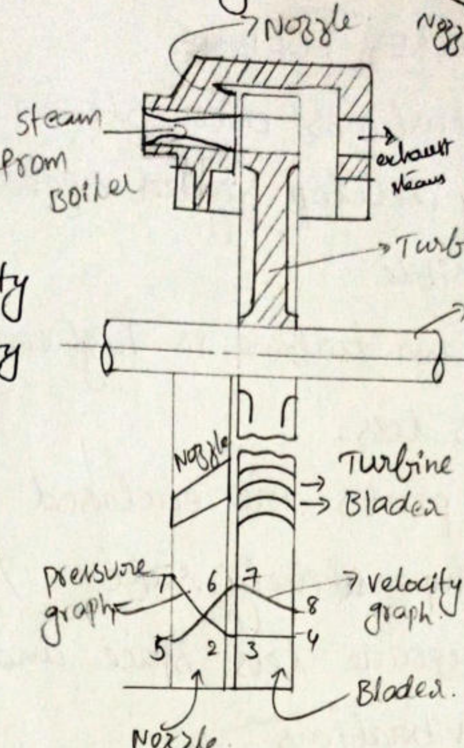
1. A steam turbine may develop higher speeds and a greater steam range is possible.
2. The efficiency of steam turbine is higher.
3. Steam consumption is less.
4. Since all the moving parts are enclosed in the casing, the steam turbine is comparatively safe.
5. A steam turbines require less space and lighter foundations, as there are little vibrations.
6. There is less frictional loss in steamturbines.
7. Applied torque is more uniform to the driven shaft.
8. Maintainance and repair cost is less.

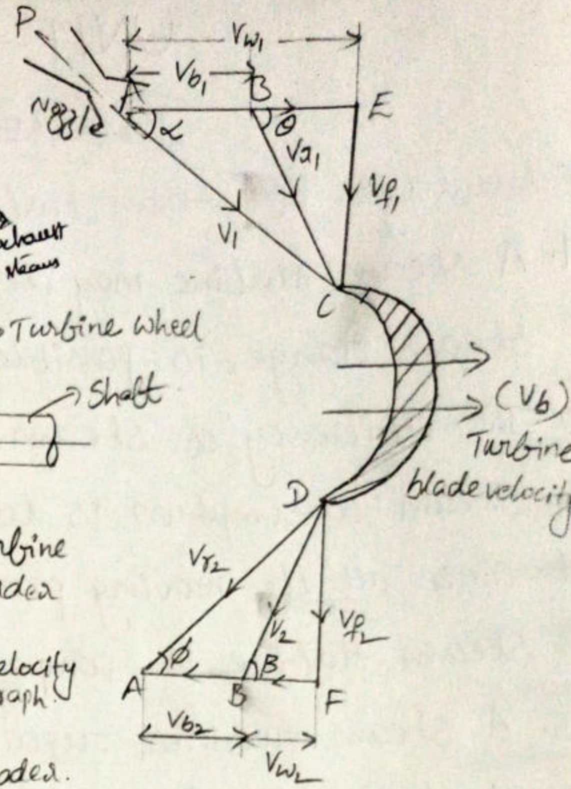
Classification of steam turbines:-

1. According to the mode of steam action.
   a) Impulse turbine.
   b) Reaction turbine.
2. According to the direction of steam flow.
   a) Tangential
   b) Axial
   c) Radial
3. According to the exhaust condition of the steam.
   a) condensing turbine.
   b) Non-condensing turbine.
4. According to the pressure of the steam.
   a) High pressure   b) Medium pressure
   c) Low pressure.

5) According to the Number of stages
a) Single stage
b) Multi stage.

Impulse Turbine :-





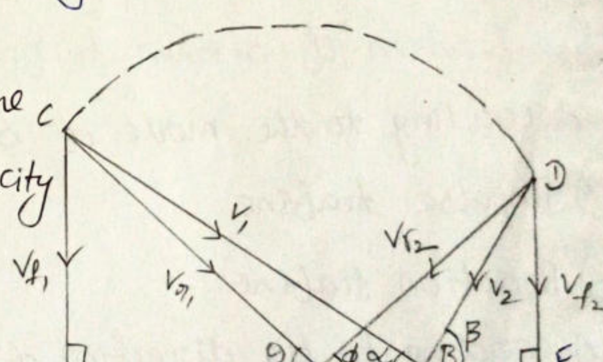
De-Laval Impulse Turbine

Let,
- V = Absolute velocity
- V_r = Relative velocity
- V_w = Whirl velocity.
- V_f = Flow velocity
- V_b = Blade velocity
- α, β = Jet angles
- θ, ϕ = Vane angles.

$$W.D/sec = m(V_{w_1} - (-V_{w_2}))V_b$$
$$= m(V_{w_1} + V_{w_2})V_b.$$

Combined Velocity triangle for moving Blades:-

Procedure:-



1. First of all draw a horizontal line and cutoff AB equal to the velocity of blade V_b to some suitable scale.
2. Now at B draw a line BC at an angle 'α' with AB. Cut off BC = V_1 to some scale.
3. Join AC ~~with~~ which represents relative velocity at Inlet (V_{r_1})
4. Now at A draw a line AD at an angle ϕ with AB.
5. Now with A as centre and radius equal to AC draw an arc meeting the line to A at D. such that AC = AD. (or) $V_{r_1} = V_{r_2}$
6. Join BD which represents velocity of jet at exit (i.e V_2)
7. From C and D draw ⊥'s meeting the line AB produce at E & F

8. EB and CE represents whirl velocity and flow velocity at Inlet (V_{f_1} and V_{w_1}) (2)

9. Similarly BF & DF represents whirl velocity and flow velocity at outlet (V_{w_2} & V_{f_2}).

Workdone on Blade = Force × distance / sec.

$= m/s \times (\text{Change in velocity}) \times V_b$.

$= m \times (V_{w_1} - (-V_{w_2})) \times V_b$.

→ V_{w_2} is −ve becoz it is opposite direction w.r.t blade motion.

$$W.D/sec = mV_b(V_{w_1} + V_{w_2})$$

$$\text{power} = \frac{mV_b(V_{w_1}+V_{w_2})}{1000} \text{ KW}.$$

Blade (or) Diagram Efficiency = $\frac{\text{W.D on Blade}}{\text{Energy supplied to the Blade}}$

$$= \frac{\cancel{m}V_b(V_{w_1}+V_{w_2})}{\frac{1}{2}\cancel{m}V_1^2}$$

$$= \frac{2V_b(V_{w_1}+V_{w_2})}{V_1^2}$$

If h_1 and h_2 be the total heats before and after expansion through the nozzles then $h_1 - h_2$ is the heat drop through a stage of fixed blade ring and moving blades rings.

Stage efficiency :- Stage efficiency $= \eta_{stage} = \frac{\text{W.D on blades per kg of steam}}{\text{Total energy supplied per kg of steam.}}$

$$\eta_{stage} = \frac{V_b(V_{w_1}+V_{w_2})/1000}{(h_1-h_2)}$$

Nozzle Efficiency $\eta_{nozz} = \frac{\text{Kinetic energy of steam at exit of nozzle}}{\text{Enthalpy drop in the nozzle.}}$

$$\eta_{nozz} = \frac{\frac{1}{2}\cancel{m}V_1^2}{\cancel{m}(h_1-h_2)} = \frac{V_1^2}{2(h_1-h_2) \times 1000}$$

Axial thrust (or) force :-

The axial thrust on the ~~whirl~~ wheel is due to difference b/w the velocity of flow at entrance and exit.

Axial thrust (or) force = Mass of steam × Axial acceleration.

$= m_s(V_{f_1} - V_{f_2})$

Energy converted to heat by Blade friction = Loss of kinetic energy during flow over Blades.

$= m_s(V_{r_1} - V_{r_2})$

Blade velocity coefficient:-

If friction is neglected $V_{r_1} = V_{r_2}$

But in actual practice the energy is lossed around 10 to 15 percent due to friction.

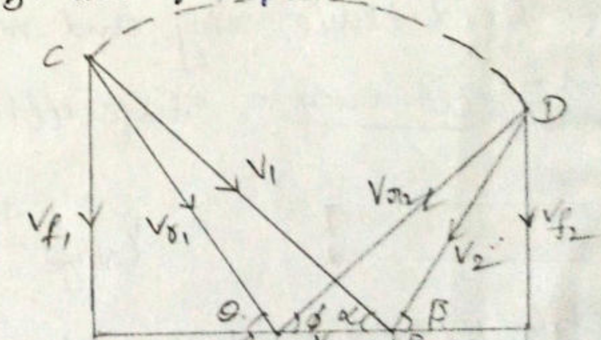
$\therefore V_{r_2} = K V_{r_1}$

where, K = Blade velocity coefficient.

1) In a De-laval turbine the steam enters the wheel through a nozzle with a velocity of 500 m/s. and at an angle of 20° to the direction of motion of the blade. The blade speed is 200m/s and exit angle of moving blade is 25°. Find the Inlet angle of moving blade., exit velocity of steam & its direction and workdone per kg of steam.

A) given that, $V_1 = 500$ m/s , $\alpha = 20°$, $V_b = 200$ m/s , $\phi = 25°$

From Inlet velocity Δ^{le}.



$$\cos\alpha = \frac{V_{w_1}}{V_1}$$

$$V_{w_1} = V_1 \cos\alpha.$$

$$\sin\alpha = \frac{V_{f_1}}{V_1}$$

$$V_1 \sin\alpha = V_{f_1}$$

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - V_b}$$

1) θ = ?
2) V_2 =
3) WD

$$V_{w_1} = V_1 \cos\alpha = 500 \times \cos(20) = 469.84 \text{ m/s}$$

$$V_{f_1} = V_1 \sin\alpha = 500 \times \sin(20) = 171.01 \text{ m/s}$$

$$\tan\theta = \frac{171.01}{469.84 - 200} = 0.633$$

$$\theta = \tan^{-1}(0.633) = 32.36°$$

(3)

$$\sin\theta = \frac{V_{f_1}}{V_{r_1}}$$

$$V_{f_1} = V_{r_1}\sin\theta.$$

$$V_{r_1} = \frac{V_{f_1}}{\sin\theta} = \frac{171.01}{\sin(32.3)} = 319.46 \text{ m/s}.$$

$$V_{r_1} = V_{r_2} = 319.46 \text{ m/s}.$$

$$\sin\phi = \frac{V_{f_2}}{V_{r_2}} \Rightarrow V_{f_2} = V_{r_2}\sin\phi$$

$$V_{f_2} = 319.46\sin(25) = 135.01 \text{ m/s}$$

$$\cos\phi = \frac{(V_{w_2}+V_b)}{V_{r_2}}$$

$$V_{w_2}+V_b = V_{r_2}\cos\phi$$

$$V_{w_2} = V_{r_2}\cos\phi - V_b$$

$$V_{w_2} = 319.46\cos(25) - 200 = 89.52$$

$$\tan\beta = \frac{V_{f_2}}{V_{w_2}} = \frac{135.01}{89.52} = 1.508$$

$$\beta = 56.45$$

$$\sin\beta = \frac{V_{f_2}}{V_2}$$

$$V_2 = \frac{V_{f_2}}{\sin\beta} = \frac{135.01}{\sin(56.45)} = 161.99 = 162 \text{ m/s}.$$

$$V_{w_1} - (-V_{w_2})$$

$$469.8 - (-89.52)$$

$$V_{w_1} + V_{w_2}$$

workdone $= \dot{m}(V_{w_1}+V_{w_2})V_b$

$$= (469.84 + 89.52)\,200 = 111872 \text{ J/s}.$$

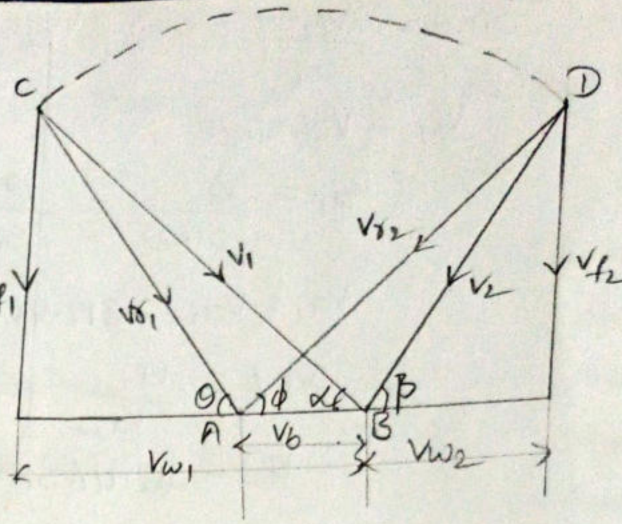
2) In a De-laval turbine, steam issued from the nozzle with ~~the~~ a velocity of 1200 m/s. the nozzle angle is 20° (α) the mean blade velocity is 400 m/s and the inlet & outlet angle of the blade are equal. The mass of steam flowing through the turbine per hr is 1000 kg. calculate i) Blade angles ii) Relative velocity of steam entering the blades. iii) tangential ~~flows~~ force on the blade iv) power developed v) Blade efficiency, take Blade velocity coefficient as 0.8 (k)

i) given that,

$V_1 = 1200\,m/s$, $\alpha = 20°$, $V_b = 400\,m/s$.

$\theta = \phi$. , $m = 1000\,kg/hr$, $K = 0.8$

$= 0.277\,kg/sec.$



$V_{r2} = K V_{r1}$

$\cos\alpha = \frac{V_{w1}}{V_1}$

$V_{w1} = V_1 \cos\alpha$ $\cancel{\Rightarrow V}$

$V_{w1} = 1200 \times \cos(20) = 1127.63$

$\sin\alpha = \frac{V_{f1}}{V_1}$

$V_{f1} = V_1 \sin\alpha$

$V_{f1} = 1200 \times \sin(20) = 410.42$

$\tan\theta = \frac{V_{f1}}{V_{w1} - V_b}$

$\tan\theta = \frac{410.42}{1127.63 - 400} = 0.564$

$\theta = 29.42$

$\therefore \theta = \phi = 29.42$

$\sin\theta = \frac{V_{f1}}{V_{r1}}$

$V_{r1} = \frac{V_{f1}}{\sin\theta} = \frac{410.42}{\sin(29.42)} = 835.39\ m/s$

$V_{r2} = K V_{r1}$

$V_{r1} = \frac{V_{r2}}{K}$ $\cancel{= \frac{835.39}{0.8}}$

$V_{r2} = K V_{r1}$

$V_{r2} = 0.8 \times 835.39 = 668.31$

tangential ~~flows~~ force on the blade $= m \times (V_{w1} + V_{w2}) = 362.8\ \frac{kg\,m}{s^2}$ N

$\cos\phi = \frac{(V_{w2} + V_b)}{V_{r2}}$

$V_{r2}\cos\phi = V_{w2} + V_b$

$V_{w2} = V_{r2}\cos\phi - V_b$

$V_{w2} = 668.31\cos(29.42) - 400$

$= 182.12$

$= m \times (V_{w_1} + V_{w_2}) \times V_b$

$= 0.277 \times (1127.63 + 182.12) \times 400$

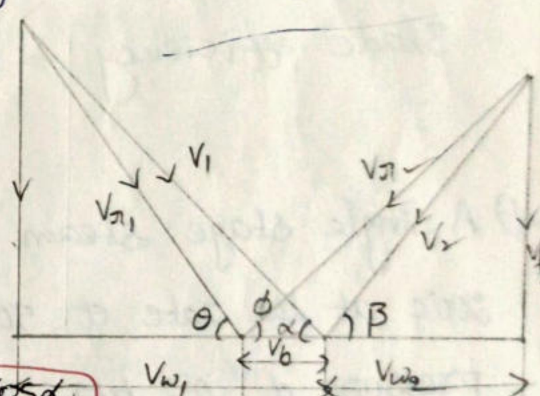
$= 362.8 \times 400 = 145120.3$

$$\text{power} = \frac{m V_b (V_{w_1} + V_{w_2})}{1000} = 145.12 \text{ kW}.$$

$$\text{Blade efficiency} = \frac{2V_b (V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 400 \times (1127.63 + 182.12)}{(1200)^2}$$

$$= 0.7276 = 72.7\%.$$

3) The velocity of steam exiting the nozzle of the impulse stage of turbine is 400 m/s (V_1). The blade operates close to the maximum blade efficiency. The nozzle angle is 20° (α) considering equiangular blades ($\theta = \phi$) and neglecting blade friction, calculate for a steam flow of 0.6 kg/sec, the ~~diagram~~ power and the diagram efficiency.



A) given that,

$V_1 = 400$ m/s , $\alpha = 20°$

$\theta = \phi$, $m = 0.6$ kg/sec

$V_{r_1} = V_{r_2}$

For maximum blade efficiency $\boxed{V_b = \frac{V_1 \cos\alpha}{2}}$

$$V_b = \frac{400 \times \cos(20)}{2} = 187.93 \text{ m/s}$$

$$\cos\alpha = \frac{V_{w_1}}{V_1}$$

$V_{w_1} = V_1 \cos\alpha = 400 \times \cos(20) = 375.87$ m/s.

$\sin\alpha = \frac{V_{f_1}}{V_1} \Rightarrow V_{f_1} = V_1 \sin\alpha = 400 \times \sin(20) = 136.8$ m/s.

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - V_b} = \frac{136.8}{375.8 - 187.9} = 0.728$$

$\theta = 36.05$

$\therefore \theta = \phi = 36.05$

$$\cos\phi = \frac{V_{w_2} + V_b}{V_{r_2}} \Rightarrow V_{w_2} + V_b = V_{r_2} \cos\phi$$

$V_{w_2} = V_{r_2}\cos\phi - V_b$

$V_{w_2} = V_{r_2}\cos\phi - V_b$.

$\sin\theta = \dfrac{V_{f_1}}{V_{r_1}}$

$V_{f_1} = V_{r_1}\sin\theta \Rightarrow V_{r_1} = \dfrac{V_{f_1}}{\sin\theta}$

$V_{r_1} = \dfrac{136.8}{\sin(36.05)} = 232.42$ m/s.

$V_{w_2} = V_{r_2}\cos\phi - V_b$.

$V_{w_2} = 232.42 \times \cos(36.05) - (187.93)$

$V_{w_2} = 0.013$

Blade power $= \dfrac{mV_b(V_{w_1}+V_{w_2})}{1000}$

$= \dfrac{0.6\times187.93(375.87+0.013)}{1000}$

$= 42.38$ kw. $= 42.42$ kw

Blade efficiency $= \dfrac{2V_b(V_{w_1}+V_{w_2})}{V_1^2} = \dfrac{2\times187.93\times(375.87+0.013)}{(400)^2}$

$= 88.29\%$.

4) A single stage steam turbine is supplied with steam at 5 bar, 200°c at the rate of 50 kg/min. It Expands into a condenser at pressure of 0.2 bar. The blade speed is 400 m/s. the Nozzles are inclined at an angle of 20° to the plane of the wheel. and outlet blade angle is 30°. Neglecting frictional losses. Determine the power developed, Bladefficiency and stage efficiency.

A) Given that, $P_1 = 5$ bar, 200°c

$m = 50$ kg/min $= 0.83$ kg/sec.

$P_2 = 0.2$ bar, $V_b = 400$ m/s.

$\alpha = 20°$, $\phi = 30°$.

$V_{r_1} = V_{r_2}$

$V_1 = 44.72\sqrt{h_1 - h_2}$

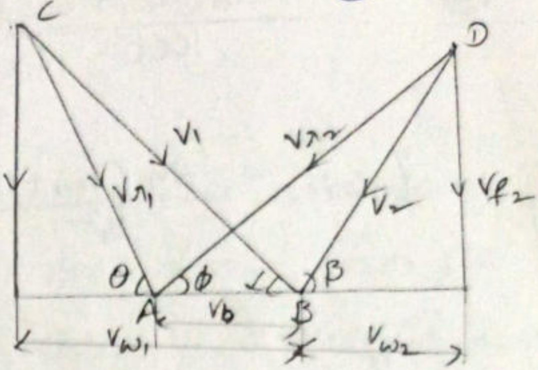
(5)

From steam tables,

At 5 bar, 200°C

$h_1 = 2855.1$ kJ/kg.

$S_1 = 7.0592$ kJ/kgK.



At 0.2 bar,

$h_{f2} = 251.5$, $S_{f2} = 0.8321$

$h_{fg2} = 2358.4$, $S_{fg2} = 7.0773$.

∴ It is Isentropic process.

$$S_1 = S_2$$

$$7.0592 = S_{f2} + x_2 S_{fg2}$$

$$7.0592 = 0.8321 + x_2(7.0773)$$

$$x_2 = 0.8798$$

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

$$h_2 = 251.5 + 0.879(2358.4)$$

$$h_2 = 2324.53$$

$$\boxed{V_1 = 44.72\sqrt{h_1 - h_2}}$$

$$= 44.72\sqrt{2855.1 - 2324.5}$$

$$V_1 = 1030.08 \text{ m/s}.$$

$$\cos\alpha = \frac{V_{w1}}{V_1}$$

$$V_{w1} = V_1 \cos\alpha = 1030.08\cos(20) = 967.96 \text{ m/s}.$$

$$\sin\alpha = \frac{V_{f1}}{V_1} \Rightarrow V_{f1} = V_1 \sin\alpha = 1030.08\sin(20) = 352.3 \text{ m/s}$$

$$\tan\theta = \frac{V_{f1}}{V_{w1} - V_b} = \frac{352.3}{967.96 - 400} = 0.6203$$

$$\theta = 31.81$$

$$V_{w2} = V_{r2}\cos\phi - V_b.$$

$$\sin\theta = \frac{V_{f1}}{V_{r1}} \Rightarrow V_{r1} = \frac{V_{f1}}{\sin\theta} = \frac{352.3}{\sin(31.81)} = 668.34 \text{ m/s}.$$

$$\therefore V_{r1} = V_{r2} = 668.34 \text{ m/s}.$$

$$V_{w2} = V_{r2}\cos\phi - V_b$$

$$V_{w2} = 668.34\cos(30°) - 400 = 178.79 \text{ m/s}.$$

$$\text{power} = \frac{m V_b (V_{w_1} + V_{w_2})}{1000} = \frac{0.83 \times 400 (967.96 + 178.79)}{1000}$$

$$= 380.72 \text{ kW.}$$

$$\eta_{Blade} = \frac{2 V_b (V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 400 (967.96 + 178.79)}{(1030.08)^2}$$

$$= 86.46\%.$$

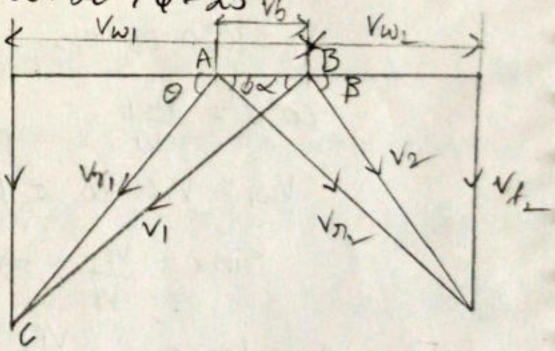
$$\eta_{stage} = \frac{V_b (V_{w_1} + V_{w_2})}{1000 (h_1 - h_2)} = \frac{400 (967.96 + 178.79)}{1000 (2855.1 - 2324.5)} = 0.8644$$

$$= 86.44\%.$$

5) The following data relates to a single stage impulse turbine
i) Steam velocity = 600 m/s.
ii) The Blade speed = 250 m/s, Nozzle angle (α) = 20°, blade outlet angle ϕ = 25°.
Neglecting friction. calculate work developed by the turbine with the steam flow rate of 20 kg/s, also calculate axial thrust on the bearings. By drawing the Diagram.

A) given that, $V_1 = 600$ m/s, $V_b = 250$ m/s, $\alpha = 20°$, $\phi = 25°$

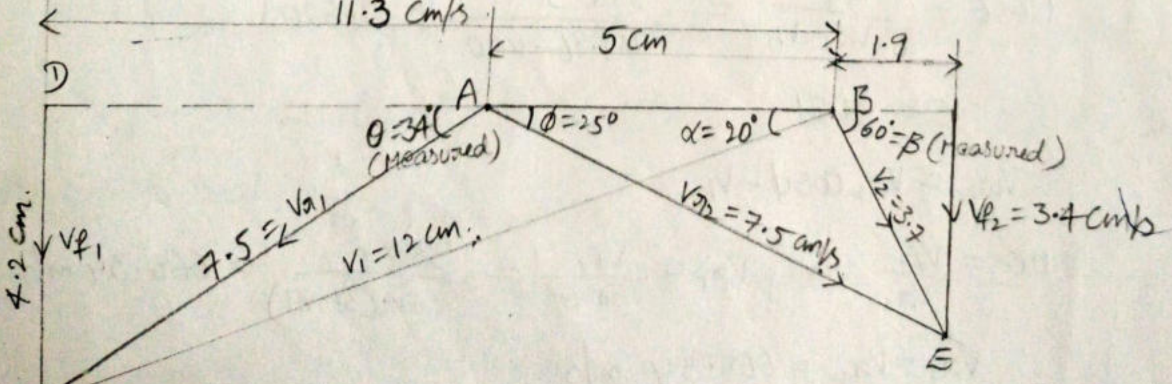
$V_{r_1} = V_{r_2}$, $m_s = 20$ kg/sec.



take scale, 50 m/s = 1 cm/sec.

$V_1 = 600$ m/s = 12 cm/sec

$V_b = 250$ m/s $= \frac{250}{50} = 5$ cm/sec.



$$\text{workdone} = m V_b (V_{w_1} + V_{w_2})$$

$$= \cancel{20 \times 250 (} \; 20 \times 250 (565 + \cancel{90}\, 95) = 3300000 \text{ ~~watt~~}$$

$$= 3300 \text{ kW.}$$

Axial thrust $= m_s(V_{f_1} - V_{f_2})$

$= 20 \times (210 - 170)$

$= 800N$

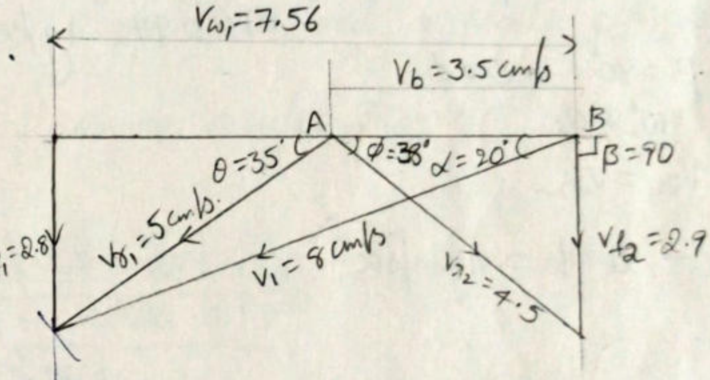
6) A single row Impulse turbine develops 132.4 kW at a Blade speed of 175 m/s using 2 kg of steam per sec. steam leaves the nozzle at 400 m/s velocity coefficient of Blade is 0.9 steam leaves the turbine axially. Determine the nozzle angle, Blade angle at entry and exit, assuming no shock.
$(\beta = 90^\circ; V_{w_2} = 0)$

A). given that, $P = 132.4$ kW. . Assume scale: 50 m/s = 1 cm/sec.

$V_b = 175$ m/s , $m = 2$ kg , $V_1 = 400$ m/s , $K = 0.9$
$= 3.5$ cm/s $\qquad = 8$ cm/s

$V_{r_2} = 0.9\, V_{r_1}$

$\beta = 90^\circ,\ V_{w_2} = 0.$



$$P = \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$$

$$132.4 = \frac{2 \times 175 \times (V_{w_1} + 0)}{1000}$$

$V_{w_1} = 378.28$ m/s

$= \frac{378.28}{50} = 7.56$ cm/s.

$V_{r_2} = 0.9 \times 250 = 0.9 \times V_{r_1}$ $\quad (V_{r_1} = 5$

$= 225$ m/s. $\qquad V_{r_1} = 5 \times 50 = 250)$.

$V_{r_2} = \frac{225}{50} = 4.5$ cm/s.

$\theta = 35^\circ,\ \phi = 38^\circ,\ \beta = 90^\circ,\ \alpha = 20^\circ.$

7) A simple impulse turbine has mean Blade speed of 200 m/s. the nozzles are inclined at 20° to the plane of rotation of the blades. The steam velocity from the nozzle is 600 m/s. The turbine uses 3500 kg/hr of the steam. the absolute velocity at exit is along the axis of turbine Determine i) The inlet & exit axis angles of Blade axial ($V_{w_2}=0, \beta = 90°$) ii) power out put of turbine iii) Diagram efficiency iv) Axial thrust (per kg steam per sec) assume inlet & outlet angles to be equal.

A) given that, $V_b = 200$ m/s.

$\alpha = 20°$.

$V_1 = 600$ m/s , $m = 3500$ kg/hr

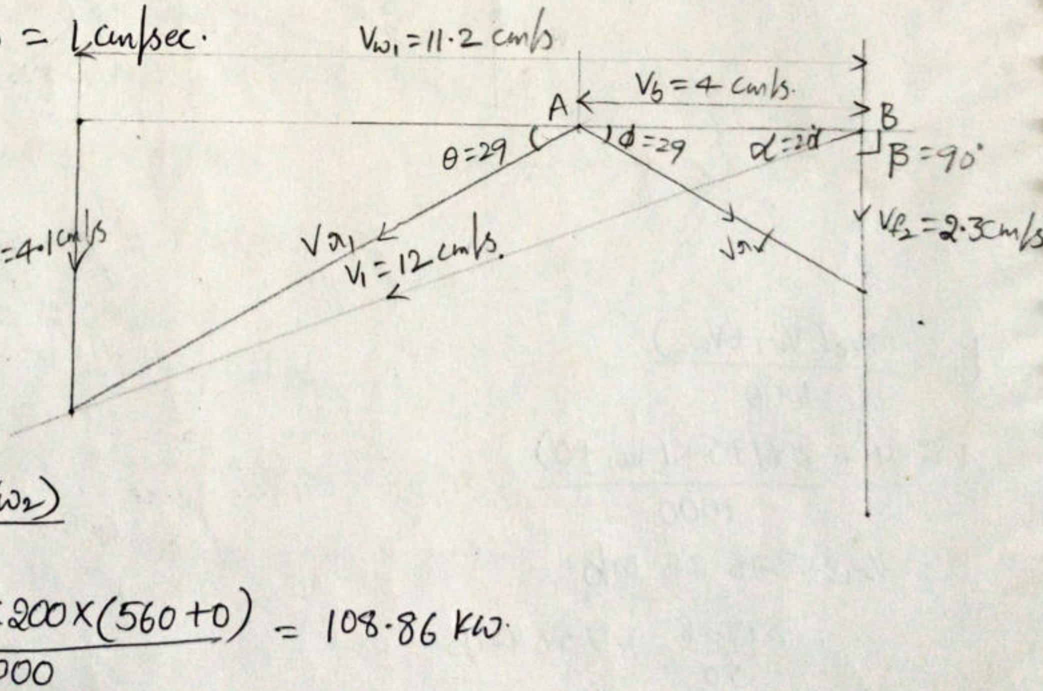
$m = 0.972$ kg/sec.

$\beta = 90°$, $\theta = \phi$

$V_{w_2} = 0$.

$V_{r_1} = V_{r_2}$

<u>Scale</u>:- 50 m/s = 1 cm/sec.



$$P = \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$$

$$P = \frac{0.972 \times 200 \times (560 + 0)}{1000} = 108.86 \text{ kW}.$$

Diagram efficiency $\eta_{di} = \frac{2 V_b (V_{w_1} + V_{w_2})}{V_1^2}$

$$= \frac{2 \times 200 \times (560 + 0)}{(600)^2}$$

$$= 62.22\%.$$

Axial thrust $= m_s(V_{f_1} - V_{f_2})$

$= 0.972(205-115)$

$= 87.48$ N.

8) In an Impulse turbine the mean dia. of Blade is ~~2.0~~ 1.05 met & speed is 3000 rpm. ratio of Blade speed to steam speed is 0.42 and ratio of outlet to inlet relative velocity $\left(\frac{V_{r_2}}{V_{r_1}}\right)$ is 0.84, The outlet angle of the Blade is to be made 8° less than the inlet angle. mass = 10 kg/s. Draw velocity diagram. Derive the following. Nozzle angle is 18°.

i) tangential thrust

ii) Axial thrust

iii) Resultant thrust iv) power developed v) Blade efficiency

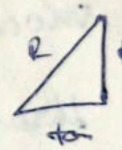
A) given that,

$d = 1.05$ met. , Assume scale 30 m/s $= 1$ cm/s.

$N = 3000$ rpm.

$V_b = \frac{\pi D N}{60} = \frac{3.14 \times 1.05 \times 3000}{60} = 164.85$ m/s.

$\frac{V_b}{V_1} = 0.42$, $\frac{V_{r_2}}{V_{r_1}} = 0.84$, $\phi = \theta - 3°$, $m = 10$ kg/s.

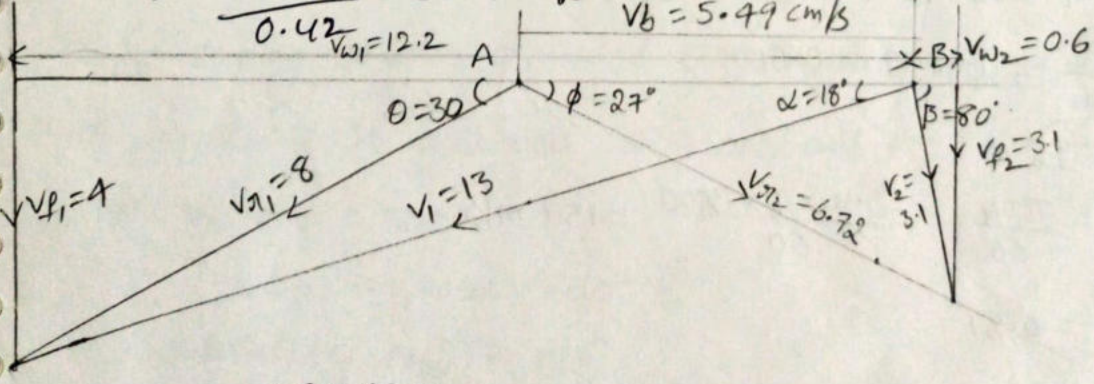


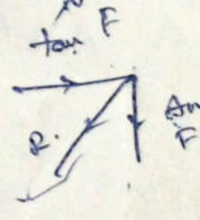
$\phi = 30-3$

$= 27°$

$V_1 = \frac{V_b}{0.42}$

$V_1 = \frac{164.85}{0.42} = 392.5$ m/s.





$V_{r_2} = 0.84 \times 240$

$V_{r_2} = 201.6$ m/s.

i) tangential thrust $= m_s(V_{w_1} + V_{w_2})$

$= 10(366+18) = 3840$. N

ii) Axial thrust $= m_s(V_{f_1} - V_{f_2})$

$= 10(120 - 93) = 270\,N$

iii) Resultant thrust $= \sqrt{(3840)^2 + (270)^2}$

$= 3849.48.$

iv) power $= \frac{m V_b (V_{w_1} + V_{w_2})}{1000} = \frac{10 \times 164.85 \times (366 + 18)}{1000}$

$= 633.024$

v) Blade efficiency $\eta = \frac{2V_b(V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 164.85 \times (366 + 18)}{(392.5)^2}$

$= 82.18\%.$

9) In a single stage Impulse turbine the mean diameter of the blade is 1met and rotational speed is 3000rpm. The steam is issued from the nozzle at 300 m/s & the nozzle angle is 20°. The blade angles are equiangular. The frictional loss in blade chanel is 19%. kinetic energy corresponding to the relative velocity at Inlet to the blade what is the power developed when axial thrust is 98 N

A) $d = 1$ met. , Assume scale:- 40 m/s = 1 cm/s.

$N = 3000$ rpm , $V_1 = 300$ m/s , $\alpha = 20°$, $\theta = \phi$

$K = \frac{V_{r_2}}{V_{r_1}} = 1 - 0.19 = 0.81$

Axial thrust = 98 N

$V_b = \frac{\pi D N}{60} = \frac{3.14 \times 1 \times 3000}{60} = 157$ m/s.

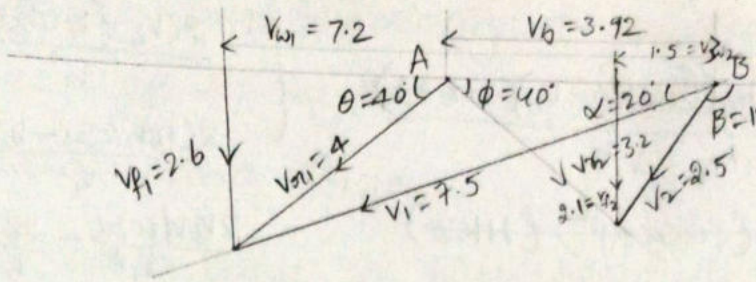
$\frac{V_{r_2}}{V_{r_1}} = 0.81$

$V_{r_2} = 0.81 \times V_{r_1} = 0.81 \times 160 = 129.6$

$P = \frac{m_s (V_{w_1} - V_{w_2}) V_b}{1000}$ $(\beta > 90°)$.

Axial thrust $= m_s(V_{f_1} - V_{f_2})$

(8)



$$98 = m_s(104 - 84)$$

$$m_s = 4.9 \text{ kg/s.}$$

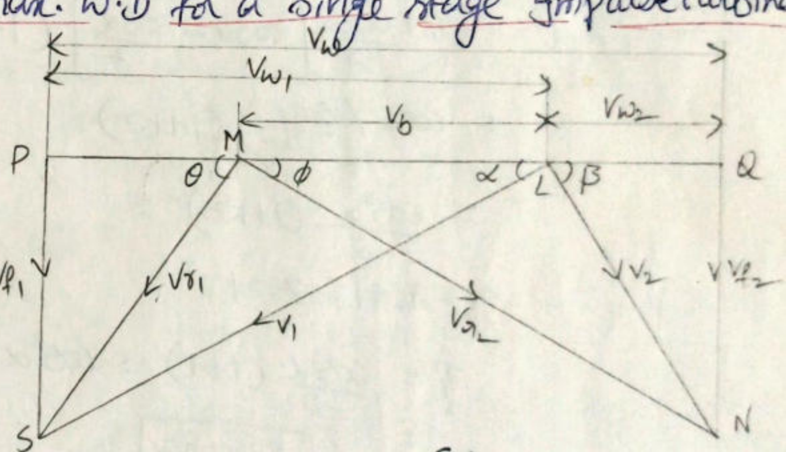
$$P = \frac{m_s(V_{w_1} - V_{w_2})V_b}{1000}$$

$$P = \frac{4.9(288 - 60) \times 157}{1000}$$

$$P = 175.4$$

Expression for optimum value of Blade speed ratio (For maximum efficiency) and max. W.D for a single stage impulse turbine

Blade speed ratio $= \frac{V_{bl}}{V_1} = e$



$$V_w = MP + MQ$$

$$= V_{r_1}\cos\theta + V_{r_2}\cos\phi$$

$$= V_{r_1}\cos\theta\left(1 + \frac{V_{r_2}\cos\phi}{V_{r_1}\cos\theta}\right]$$

$$V_w = V_{r_1}\cos\theta(1 + KZ) \longrightarrow (1)$$

$\left(\because \frac{V_{r_2}}{V_{r_1}} = K \quad \frac{\cos\phi}{\cos\theta} = Z\right]$

Generally angles θ and ϕ are nearly equal and hence Z is assumed as constant.

NOW, $V_{r_1}\cos\theta = V_{w_1} - V_b$

$$V_{r_1}\cos\theta = V_1\cos\alpha - V_b \longrightarrow (2)$$

substitute eq (2) in eq (1).

$$V_w = (V_1\cos\alpha - V_b)(1 + KZ) \longrightarrow (3).$$

$$\eta_{bla} = \frac{2V_b(V_{w_1} + V_{w_2})}{V_1^2}$$

$[\because V_w = V_{w_1} + V_{w_2})$

$$\eta_{bl} = \frac{2V_b V_w}{V_1^2} \longrightarrow (4)$$

Substitute in eq ③

$$\eta_{bl} = \frac{2V_b((V_1\cos\alpha - V_b)(1+kz))}{V_1^2}$$

$$= \cancel{2(\rho\cos\alpha - \rho^2)(1+kz)}$$

$$\eta_{bl} = 2\rho(\cos\alpha - \rho)(1+kz) \rightarrow ④$$

$$\left[\because \frac{2(V_b)(V_1\cos\alpha - V_b)}{V_1^2}\right.$$

$$\frac{2(V_bV_1\cos\alpha - V_b^2)}{V_1^2}$$

$$\frac{2V_bV_1\cos\alpha}{V_1^2} - \frac{2V_b^2}{V_1^2}$$

$$2\cdot\frac{V_b}{V_1}\cos\alpha - 2\left(\frac{V_b}{V_1}\right)^2$$

$$2\rho\cos\alpha - 2\rho^2$$

$$\left.2(\rho\cos\alpha - \rho^2)\right]$$

Blade efficiency dependence on only ρ.

Differentiate above eq. w.r.t 'ρ':

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

For max. (or) min. Blade efficiency the value is zero.

$$2(\cos\alpha - 2\rho)(1+kz) = 0.$$

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

$$\boxed{\rho = \frac{\cos\alpha}{2}} \rightarrow ⑤$$

$$\left[\because \frac{V_b}{V_1} = \frac{\cos\alpha}{2}\right.$$

$$\left.V_b = \frac{V_1\cos\alpha}{2}\right]$$

substitute eq ⑤ in ④

$$④ \Rightarrow \eta = \frac{\cancel{2}\times\cos\alpha}{\cancel{2}}\left[\cos\alpha - \frac{\cos\alpha}{2}\right](1+kz)$$

$$= \cos\alpha\left(\frac{\cos\alpha}{2}\right)(1+kz)$$

$$= \frac{\cos^2\alpha}{2}(1+kz)$$

If $k=1$, $z=1$

$$\eta = \frac{\cos^2\alpha}{\cancel{2}}(\cancel{1+1}) = \cos^2\alpha.$$

$$\boxed{\eta = \cos^2\alpha}$$

$$W_{max} = V_b(V_w)$$

$$V_w = (V_1\cos\alpha - V_b)(1+kz)$$

$$W_{max} = V_b(V_1\cos\alpha - V_b)(1+kz)$$

As $k=1$ & $z=1$

$$W_{max} = 2V_b(V_1\cos\alpha - V_b)$$

$$\left[V_b = \frac{V_1\cos\alpha}{2}\right.$$

$$\left.V_1\cos\alpha = 2V_b\right]$$

$$W_{max} = 2V_b(2V_b - V_b)$$

$$= 2V_b\cdot V_b$$

$$\boxed{W_{max} = 2V_b^2}$$

It is obivious that the blade velocity should be approximately

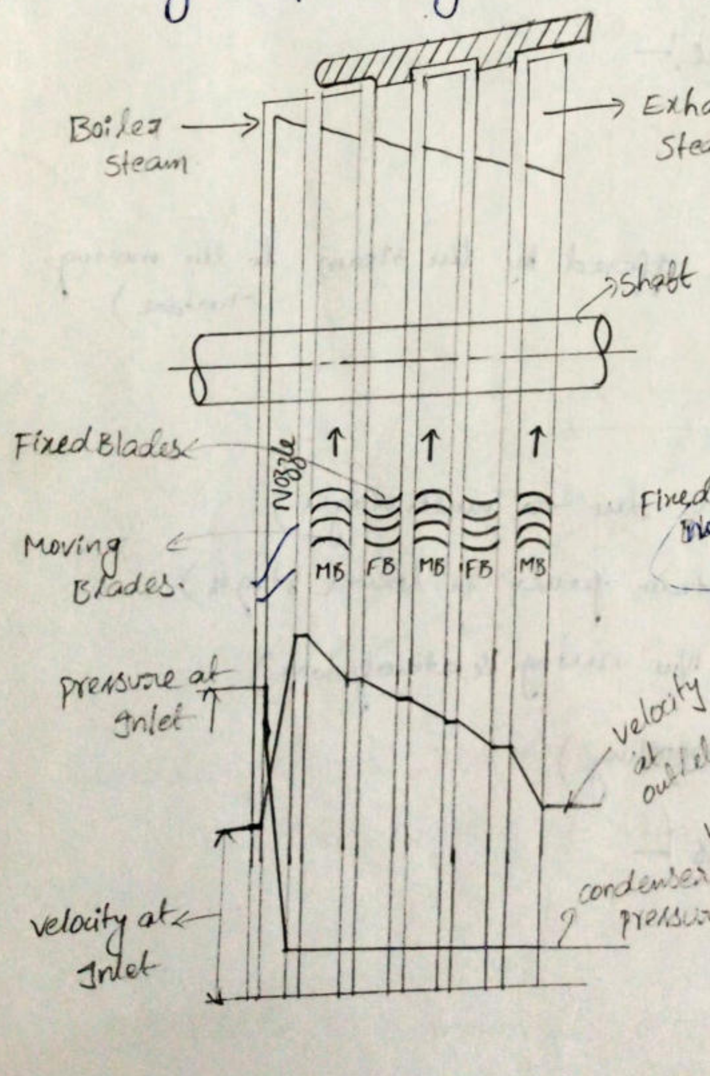
half of the absolute velocity of the steam jet coming out (9)
from the nozzle for the maximum work developed per kg of steam
(or) for maximum efficiency.

For other values of blade speed the absolute velocity at outlet will increase, consequently more energy will be carried away by the steam and hence efficiency will decrease.

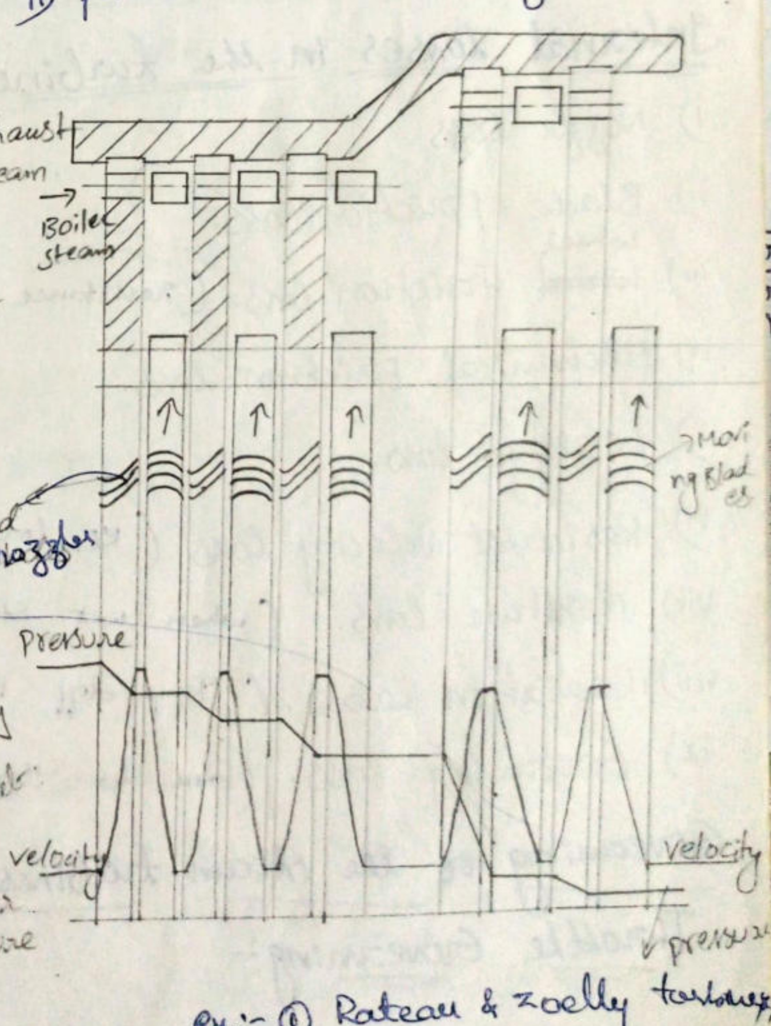
Methods of Reducing speed of the Rotor :-

1. Velocity compounding.
2. pressure compounding.
3. pressure velocity compounding.

i) Velocity compounding :-

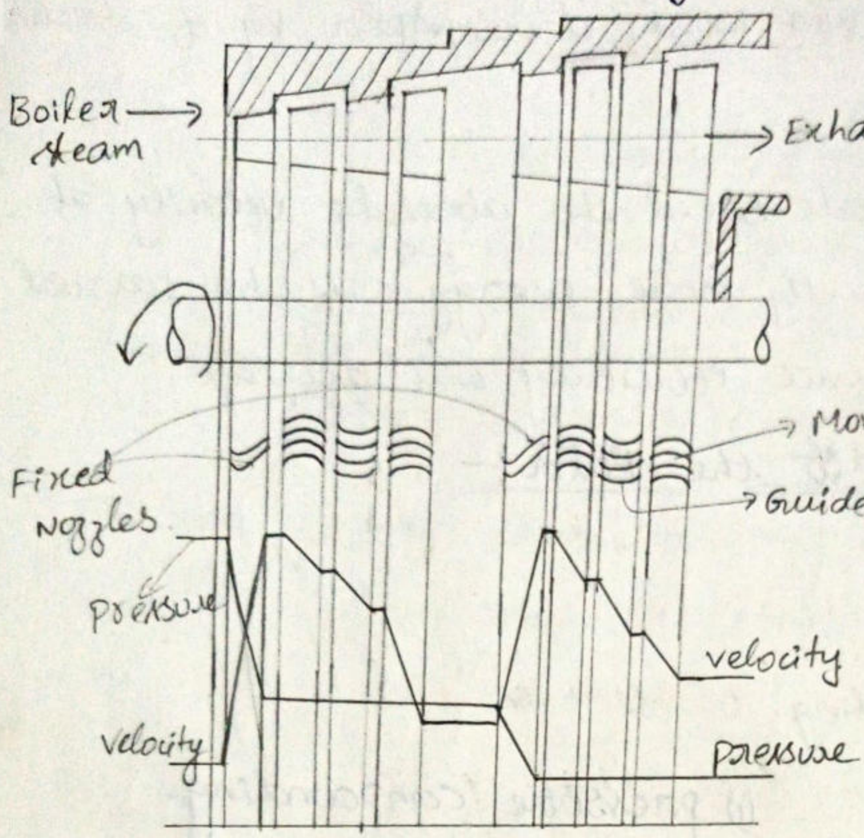


ii) pressure compounding.



eg:- ① Rateau & zoelly turbines.

Pressure - velocity compounding:-



ex: Curtis turbine.

Internal losses in the turbine:-

i) Nozzle loss.

ii) Blade Friction loss.

iii) ~~whirl~~ wheel friction loss. (resistance offered by the steam to the moving turbine)

iv) Mechanical Friction loss.

v) Leakage loss.

vi) Residual velocity loss. (KE loss due to multistages)

vii) Moisture loss. (when wet steam pass in lower stages)

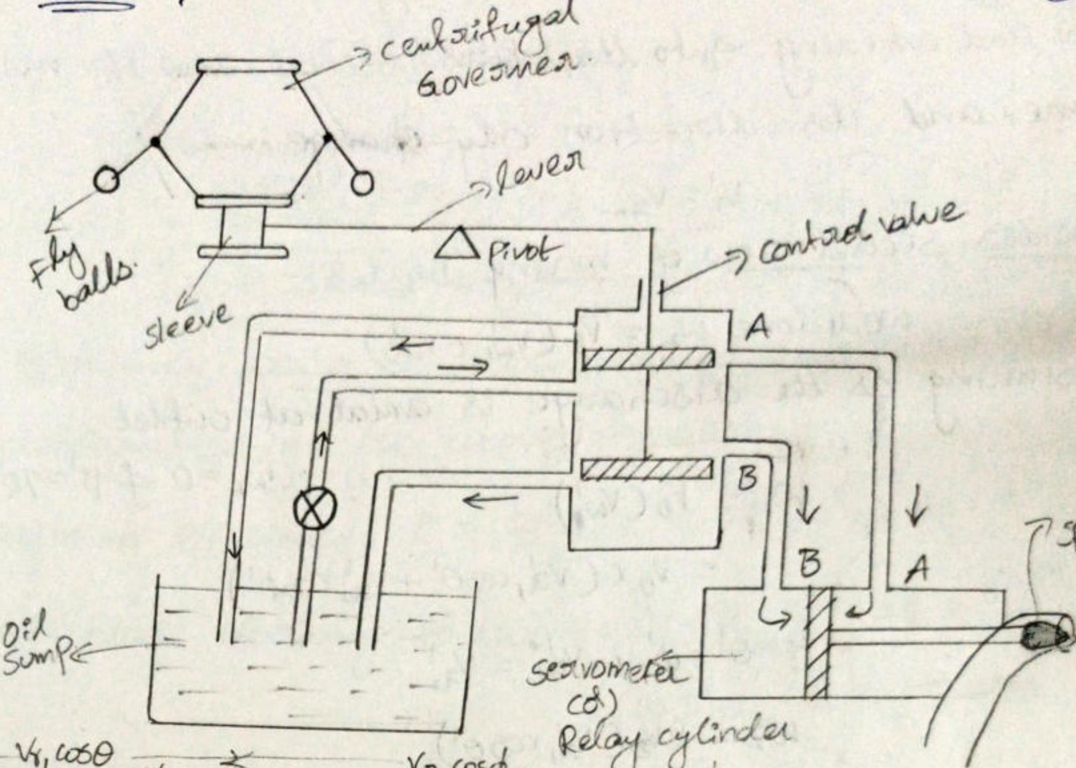
viii) Radiation losses. (Temp diff b/w casing & atmosphere)

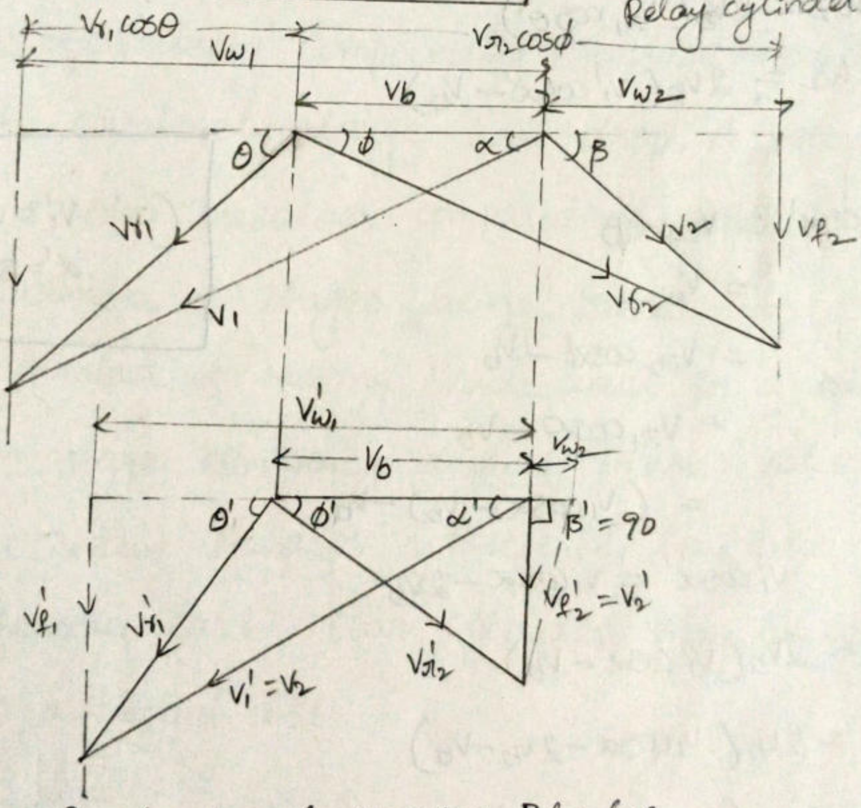
ix) Governing loss. (due to throttling)

Governing of the steam turbines:-

Throttle Governing:-

Throttle Governer :-





$V_{w_2} = 0$

$\beta = 90°$

$V'_1 = V_2$

Consider first row of moving Blades,

workdone per kg of steam $W_1 = V_b(V_{w_1} + V_{w_2})$

$$= V_b(V_w)$$

$$= V_b(V_{r_1}\cos\theta + V_{r_2}\cos\phi)$$

If blade is symetrical, $\theta = \phi$ & $V_{r_1} = V_{r_2}$

$$\therefore W_1 = V_b \times 2V_{r_1}\cos\theta$$

$$= V_b \times 2(V_{w_1} - V_b)$$

$$W_1 = 2V_b(V_1\cos\alpha - V_b) \longrightarrow ⓒ$$

The magnitude of absolute velocity of steam leaving the first row and entering into the ~~first~~ second row of moving blade is same. and its direction only change.

$$V_1' = V_2$$

<u>Consider second row of moving Blades</u>:-

workdone $W_2 = V_b(V_{w_1}' + V_{w_2}')$

Assuming as the discharge is axial at outlet,

$\therefore V_{w_2}' = 0$ & $\beta' = 90°$

$$W_2 = V_b(V_{w_1}')$$

$$= V_b \times (V_{r_1}' \cos\theta' + V_{r_2}' \cos\phi')$$

$$\theta' = \phi', \quad V_{r_1}' = V_{r_2}'$$

$$W_2 = 2V_b(V_{r_1}' \cos\theta')$$

$$W_2 = 2V_b(V_1' \cos\alpha' - V_b)$$

Now, $\alpha' = \beta$

$$V_1' \cos\alpha' = V_2 \cos\beta$$

$$= V_{w_2}$$

$$= V_{r_2} \cos\phi - V_b$$

$$= V_{r_1} \cos\theta - V_b$$

$$= (V_1 \cos\alpha - V_b) - V_b$$

$$V_1' \cos\alpha' = V_1 \cos\alpha - 2V_b$$

$(\because V_1' = V_2$
$\alpha' = \beta)$

$$W_2 = 2V_b(V_1' \cos\alpha' - V_b)$$

$$= 2V_b(V_1 \cos\alpha - 2V_b - V_b)$$

$$= 2V_b(V_1 \cos\alpha - 3V_b)$$

Total workdone $W = W_1 + W_2$

$$W = 2V_b(V_1 \cos\alpha - V_b) + 2V_b(V_1 \cos\alpha - 3V_b)$$

$$= 2V_b(2V_1 \cos\alpha - 4V_b)$$

$$\boxed{W = 4V_b(V_1 \cos\alpha - 2V_b)}$$

Blade efficiency for two stage Impulse turbine:- (11)

$$\eta_{bl} = \frac{W}{V_1^2/2} = \frac{4V_b(V_1\cos\alpha - 2V_b)}{\frac{V_1^2}{2}}$$

$$= 8\frac{V_b}{V_1}\left(\cos\alpha - 2\frac{V_b}{V_1}\right) \qquad \left[\because \rho = \frac{V_b}{V_1}\right]$$

$$\boxed{\eta_{bl} = 8\rho(\cos\alpha - 2\rho)}$$

The Blade efficiency would be maximum for two stage Impulse turbine. By differentiating the above equation.

$\therefore$ For maximum efficiency, $\rho = \frac{\cos\alpha}{4}$

maximum blade efficiency for two stage $\eta_{(bl)\,max} = 2\cos^2\alpha$.

$(W.D)_{max} = 8V_b^2$

Advantages of velocity compounded Impulse turbine:-

1. Owing to relatively large heat drop, A velocity compounded Impulse turbine requires comparatively small number of stages.
2. Due to number of stages being small its cost is less.
3. As the number of moving blade rows in a wheel increases the maximum stage efficiency and optimum value of 'ρ' decreases.
4. Since the steam temp. is sufficiently low in a two (or) three row wheel, therefore cast Iron cylinder may be used. this will cause saving in material cost.

Disadvantages:-

1. It has high steam consumption and low efficiency.
2. In a single row wheel the steam temp. is high so cast Iron cylinder can not be used. and therefore cast steel cylinder is used which is ~~castied~~ than cast Iron.

costlier

1) The following particulars relate to a two row velocity compounded - Impulse turbine. ~~steam~~ steam velocity at nozzle outlet (V_1) 650 m/s, mean blade velocity 125 m/s, nozzle outlet angle is 16°. outlet angle for the first ~~to~~ row of moving blades is 18° (ϕ) ~~outlet angles for the fixed guide vanes (β = α')~~ is ~~22°~~. outlet angle for the second row of moving blades (ϕ') is 36°. Steam flow is 2.5 kg/s. the ratio of relative velocity at outlet to that at inlet is 0.84 for all the blades. Determine the following.

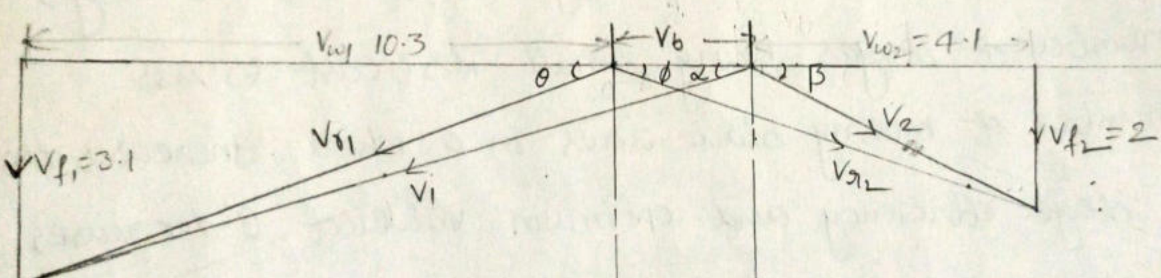
i) Axial thrust on the blades. ii) power developed iii) Efficiency of the wheel.

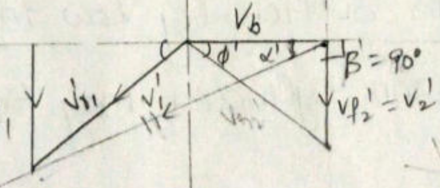
A) $V_1 = 650$ m/s, $V_b = 125$ m/s, $\alpha = 16°$, $\phi = 18°$ Scale :- 1 m/s = 50 cm/s

$\beta = \alpha' = 22°$, $\phi' = 36°$, $m = 2.5$ kg/s

$V_{r_2} = 0.84\, V_{r_1}$, $V_{r_2}' = 0.84\, V_{r_1}'$

$V_{r_2} = 8.9 \times 0.84$
$= 7.4$





$\left[V_1' = V_2 \quad \beta = \alpha' \right]$

~~$V_{r_2} = 0.84 \times 10.5$~~
~~$= 8.82$~~

$V_{a_2}' =$

i) Axial thrust $= m\left[(V_{f_1} - V_{f_2}) + (V_{f_1}' - V_{f_2}')\right] = 2.5\left((180-138) + (122-10\ldots\right.$

~~$= 2.5\left[(186 - 120) + 165\ (90 - 108)\right]$~~ (108-90) $= 142.5\,N$

ii) power developed $= \dfrac{m V_b (V_{w_1} + V_{w_2}) + (V_{w_1}' + V_{w_2}')}{1000} = \dfrac{\cancel{2.5 \times 125\,(618 + 246)}}{1000}$

$= \dfrac{m V_b (V_{w_1} + V_{w_2}) + (V_{w_1}' + V_{w_2}')}{1000} = \dfrac{2.5 \times 125 \times (924 + 324)}{1000}$ ~~= 270~~ $= 390$ kw.

iii) Efficiency of the wheel $= \frac{2V_b[(V_{w1}+V_{w2})+(V_{w1}'+V_{w2}')]}{V_1^2}$ (12)

$= \frac{2\times 125(924+324)}{(650)^2} = ~~51\%~~ \; 73.8\%$

2) The first stage of an impulse turbine is compounded for velocity and has two rings of moving blade & one ring of fixed blades the nozzle angle is 20° and leaving angles of blades are respectively as follows

i) First moving ~~20°~~ 25, (fixed 25°) not required

ii) Second moving 30°. Velocity of steam leaving the nozzle is 600 m/s and the steam velocity relative to the blade is reduced by 10% during the passage through each ring. Find the diagram efficiency and power developed ~~per~~ for a steam flow of 4 kg/s. Blade speed may be taken as 125 m/s.

A) $\phi = 20°$, $\alpha' = \beta = 25°$, $\phi' = 30°$, $m = 4$ kg/s, $V_b = 125$ m/s.

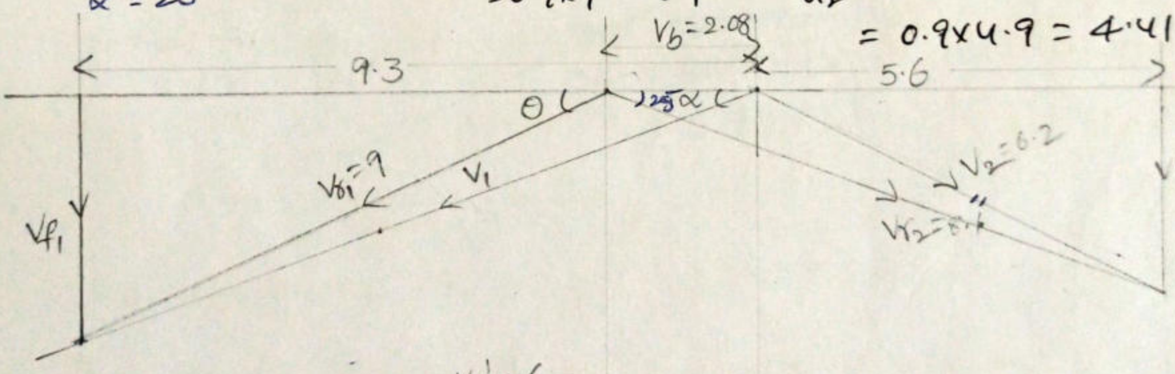
Scale :- 50 m/s = 1 cm/s.

$V_1 = 600$ m/s, $V_{r2} = 0.9 V_{r1}$, $V_{r2}' = 0.9 V_{r1}'$

$\alpha = 20$

$= 0.9\times 9 = 8.1$

$= 0.9\times 4.9 = 4.41$



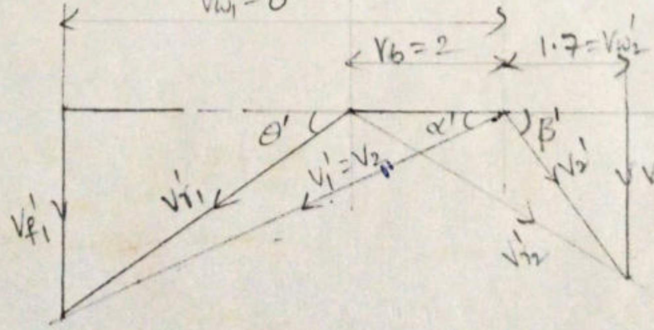


Diagram efficiency $= \frac{2V_b(V_{w1}+V_{w2})(V_{w1}'+V_{w2}')}{V_1^2} = \frac{2\times 125\times(558+336)(260+20)}{(600)^2}$

$= ~~62\%~~ = 78.47\%$ (values are wrong)

REACTION TURBINE

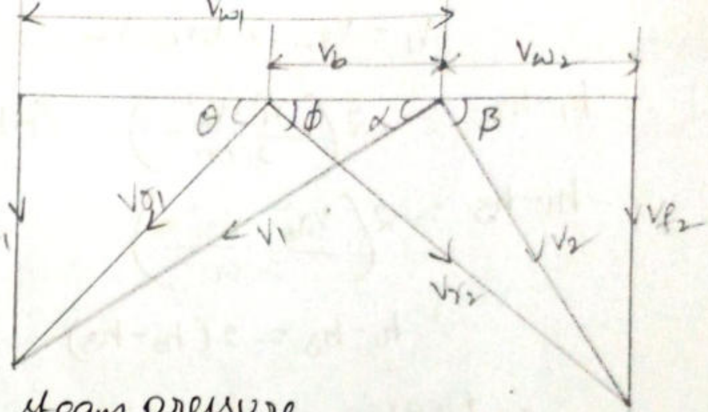
Differences b/w Impulse turbine and Reaction turbine.

Impulse	Reaction turbine.
1. pressure energy is converted into kinetic energy at the inlet of turbine.	1. partial pressure energy is converted into kinetic energy at the Inlet of turbine.
2. Steam flows through the nozzle and Impinges on the blade.	2. The steam flows first through the guide mechanism & then through the moving blades.
3. The steam may (or) may not admitted over the whole circumference.	3. The steam must be admitted over the whole circumference.
4. The steam pressure remains constant while moving through the blades.	4. The steam pressure reduces while moving through the blades
5. The relative velocity of steam remain constant (Assuming no friction)	5. Relative velocity increases while gliding over the moving blades (Assuming no friction)
6. The blades are symmetrical	6. Blades are unsymmetrical.
7. The Number of stages required are (or) less for the same power developed	7. The number of stages required are more for the same power developed.

Components :-

1. casing
2. Runner
3. Moving Blade
4. Draft tube

Velocity triangles:-



$V_{r_2} > V_{r_1}$

$\alpha = \phi$

$\beta = \theta$.

Consider a reaction turbine working and under action of steam pressure.

Let, m = Mass of steam flowing through the turbine.

$(V_{w_1}+V_{w_2})$ = Change in velocity of whirl in m/s.

Force F = mass/sec × change in velocity.

$$F = m(V_{w_1} - (-V_{w_2}))$$

$$= m(V_{w_1}+V_{w_2})$$

$$\text{workdone} = m(V_{w_1}+V_{w_2}) \times V_{bl}$$

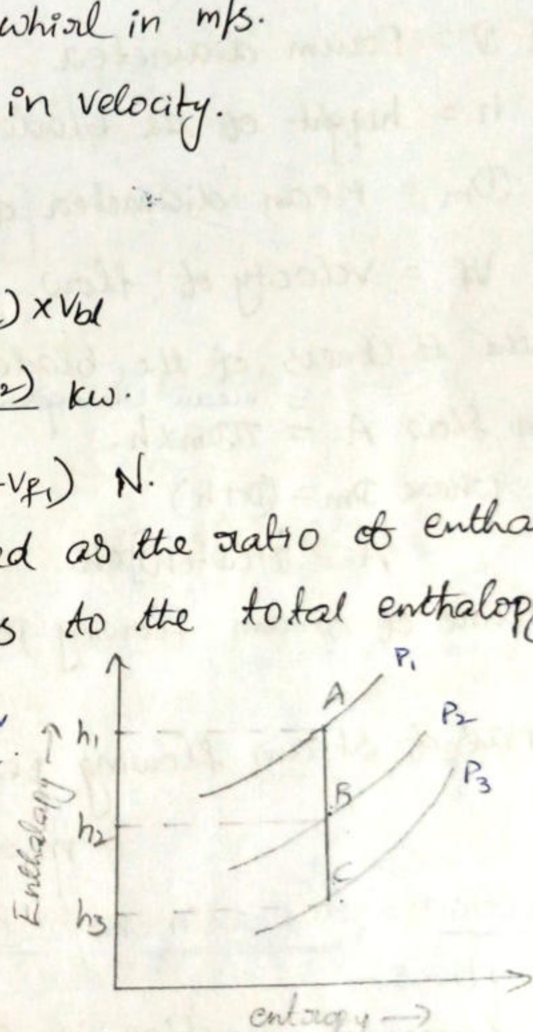
$$\text{power} = \frac{mV_b(V_{w_1}+V_{w_2})}{1000} \text{ kw.}$$

$$\text{Axial thrust force} = m(V_{f_2}-V_{f_1}) \text{ N.}$$

Degree of Reaction:- It is defined as the ratio of enthalopy (or) heat drop in the moving blades to the total enthalopy drop in moving and fixed blades.

= Isen. enthalpy change in rotor / " " " in stage.

$$\text{Degree of reaction} = \frac{h_2-h_3}{h_1-h_3}$$



∴ The Enthalapy drop in the fixed blades per kg of steam is given by,

$$(h_1-h_2) = \frac{V_1^2-V_2^2}{2000} \text{ kJ/kg}$$

and Enthalapy drop in the moving blades per kg of steam is given by,

$$(h_2-h_3) = \frac{V_{r_2}^2-V_{r_1}^2}{2000}$$

Total enthalapy drop in the stage $(h_1-h_3) = (h_1-h_2)+(h_2-h_3)$

$$= \left(\frac{V_1^2-V_2^2}{2000}\right) + \left(\frac{V_{r_2}^2-V_{r_1}^2}{2000}\right)$$

Now, in parson's reaction turbine (or) 50% Reaction turbine,

$V_1 = V_{r_2}$, $V_{r_1} = V_2$

$\therefore h_1 - h_3 = 2\left(\frac{V_1^2 - V_2^2}{2000}\right)$ (d)

$h_1 - h_3 = 2\left(\frac{V_{r_2}^2 - V_{r_1}^2}{2000}\right)$ $\left(\because h_2 - h_3 = \frac{V_{r_2}^2 - V_{r_1}^2}{2000}\right)$

$\therefore h_1 - h_3 = 2(h_2 - h_3)$

$\therefore$ Degree of reaction $= \frac{h_2 - h_3}{h_1 - h_3} = \frac{h_2 - h_3}{2(h_2 - h_3)} = \frac{1}{2} = 50\%$.

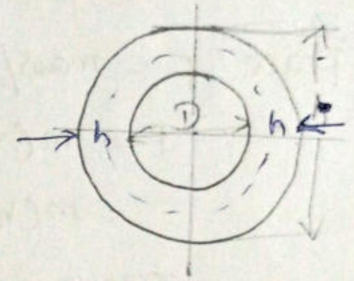
Mass of steam flowing over the blades of a reaction turbine:-

let D = Drum diameter

h = height of the blades.

D_m = Mean diameter of the blade

V_f = velocity of flow



If the thickness of the blade is neglected then total area of the steam flow $A = \pi D_m \times h$.

= mean circumference x blade height $= 2\pi r h$ (or) $(\pi(D+h))h$

$D_m = D + \frac{h}{2} + \frac{h}{2}$ πD

where $D_m = (D+h)$

$A = [\pi(D+h)] \times h$.

volume of steam flowing per sec = Area x velocity of flow.

$= [\pi(D+h)] \times h \times V_f$

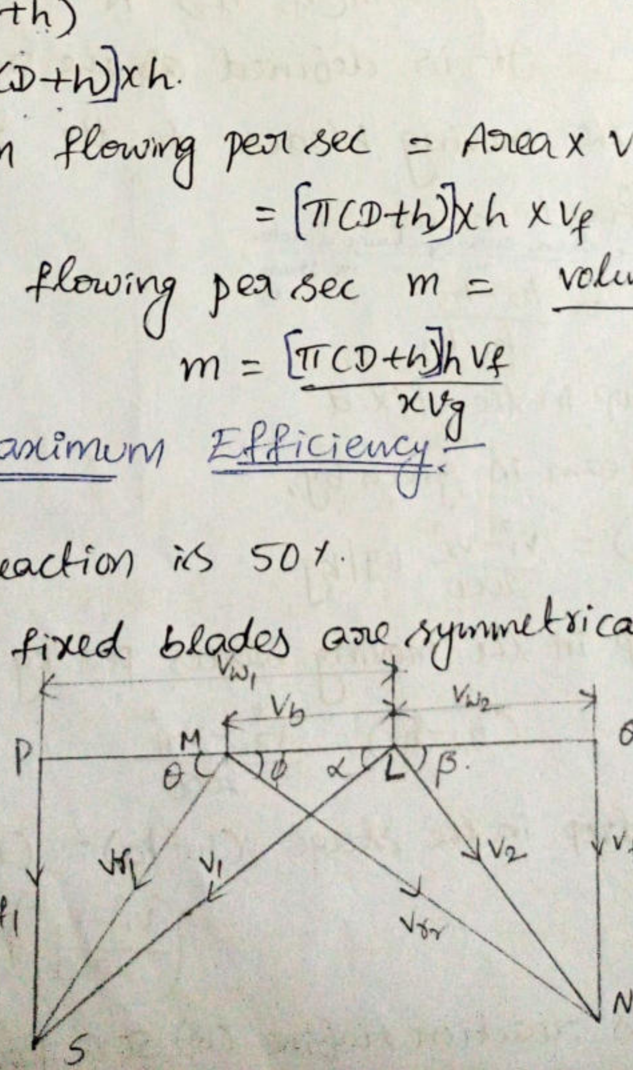
$\therefore$ Mass of steam flowing per sec $m = \frac{\text{volume of steam flowing}}{x \cdot v_g}$

$m = \frac{[\pi(D+h)] h V_f}{x V_g}$

Condition for Maximum Efficiency:-

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 the velocity of steam at the entrance to the succeding stage. (15)

$$W.D = V_b(V_{w_1}+V_{w_2})$$

$$= V_b\left(V_1\cos\alpha + (V_{r_2}\cos\phi - V_b)\right)$$

According to the assumptions, $\phi=\alpha$, $\theta=\beta$, $V_{r_2}=V_1$

$$W = V_b(2V_1\cos\alpha - V_b)$$

$$W = 2V_bV_1\cos\alpha - V_b^2$$

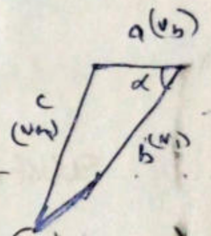
Multiply and divide the above equation with V_1^2

$$= V_1^2\left[\frac{2V_bV_1\cos\alpha}{V_1^2} - \frac{V_b^2}{V_1^2}\right]$$

$$\boxed{\therefore e = \frac{V_b}{V_1}}$$

$$W = V_1^2(2\rho\cos\alpha - e^2)$$

The kinetic energy supplied to the fixed blades $= \frac{V_1^2}{2}$



Kinetic energy for moving blades $= \frac{V_{r_2}^2 - V_{r_1}^2}{2}$

$\therefore$ Total energy supplied to one stage $\Delta h = \frac{V_1^2}{2} + \frac{V_{r_2}^2 - V_{r_1}^2}{2}$ $\quad (\because V_{r_2} = V_1)$

$$\Delta h = \frac{V_1^2}{2} + \frac{V_1^2 - V_{r_1}^2}{2}$$

$$\Delta h = V_1^2 - \frac{V_{r_1}^2}{2} \longrightarrow (1)$$

For obtuse triangle

$$\cos\alpha = \frac{a^2+b^2-c^2}{2ab}$$

$$\cos\alpha = \frac{V_b^2+V_1^2-V_{r_1}^2}{2V_bV_1}$$

$$V_{r_1}^2 = V_1^2+V_b^2-2V_bV_1\cos\alpha.$$

consider the $\triangle^{le}$ LMS,

$$V_{r_1}^2 = V_1^2+V_b^2-2V_1V_b\cos\alpha \longrightarrow (2)$$

substitute eq (2) in eq (1)

$$\Delta h = V_1^2 - \frac{V_1^2+V_b^2-2V_1V_b\cos\alpha}{2}$$

$$= \frac{2V_1^2-(V_1^2+V_b^2-2V_1V_b\cos\alpha)}{2} = \frac{V_1^2 - V_b^2+2V_1V_b\cos\alpha}{2}$$

$$= \frac{V_1^2}{2}\left[1-\frac{V_b^2}{V_1^2}+2\frac{V_b}{V_1}\cos\alpha\right]$$

$$= \frac{V_1^2}{2}\left[1-e^2+2e\cos\alpha\right] = \frac{V_1^2}{2}(1+2e\cos\alpha - e^2)$$

$\therefore$ The Blade efficiency of the reaction turbine is given by,

$$\eta_{bl} = \frac{\text{workdone}}{\text{total energy}} = \frac{W}{\Delta h}$$

$$= \frac{V_1^2(2e\cos\alpha - e^2)}{\frac{V_1^2}{2}(1+2e\cos\alpha - e^2)} = \frac{2(2e\cos\alpha - e^2)}{1+2e\cos\alpha - e^2}$$

$$= \frac{2(2e\cos\alpha - e^2+1-1)}{1+2e\cos\alpha - e^2}$$

$$= \frac{[2(2e\cos\alpha - e^2+1)]-2}{1+2e\cos\alpha - e^2}$$

$$\frac{2(1+2e\cos\alpha - e^2-1)}{1+2e\cos\alpha - e^2}$$

$$= \frac{2(1+2e\cos\alpha - e^2)}{1+2e\cos\alpha - e^2} - \frac{(1)2}{1+2e\cos\alpha - e^2}$$

$$= 2 - \frac{2}{1+2e\cos\alpha - e^2}$$

$$\boxed{\eta_{bl} = 2 - \frac{2}{1+2e\cos\alpha - e^2}} \rightarrow (3)$$

The Blade efficiency becomes maximum when the value of $(1+2e\cos\alpha - e^2)$ becomes maximum.

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

$$(2\cos\alpha - 2e) = 0$$

$$\cancel{2}e = \cancel{2}\cos\alpha$$

$$\boxed{e = \cos\alpha} \rightarrow (4)$$

Substitute eq (4) in (3)

$$(\eta_{bl})_{max} = 2 - \frac{2}{1+2\cos\alpha\cdot\cos\alpha - (\cos\alpha)^2}$$

$$= 2 - \frac{2}{1+2\cos^2\alpha - \cos^2\alpha} = 2 - \frac{2}{1+\cos^2\alpha}$$

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

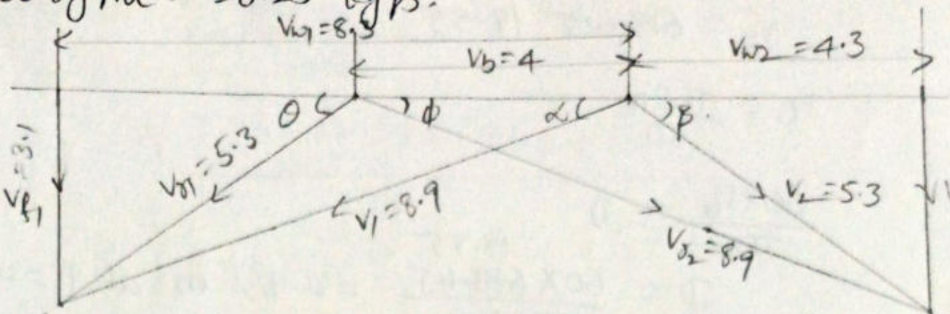
$$\boxed{(\eta_{bl})_{max} = \frac{2\cos^2\alpha}{1+\cos^2\alpha}}$$

1) In one stage of a reaction turbine both the fixed and moving blades have inlet and outlet blade tip angles of 35° & 20° respectively. The mean blade speed is 80 m/s and steam consumption is 22500 kg/hr. determine the power developed in the pair if the isentropic heat drop for the pair is 23.5 kJ/kg

A) $\theta = 35°$, $\phi = 20°$ Scale:- 20 m/s = 1 cm/s

$V_b = 80$ m/s $\theta = \beta$, $\phi = \alpha$

$m = 22500 \ kg/hr. = 6.25 \ kg/s.$



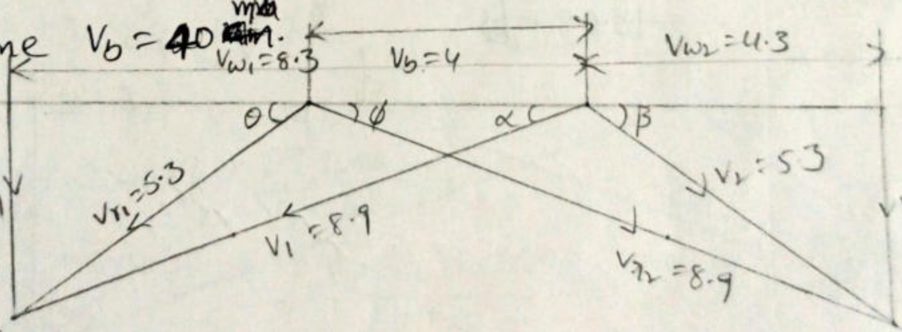
$$\text{power} = \frac{m V_b (V_{w_1} + V_{w_2})}{1000} = \frac{6.25 \times 80 \times (166 + 86)}{1000}$$

$$= 126 \ kw. \qquad (117.5 \ kw)$$

2) A ~~partial~~ parson's reaction turbine while running at 400 rpm consumes 30 tons of steam per hr. the steam at a certain stage is at 1.6 bar with dryness fraction of 0.9 and stage developes 10 kw. The axial velocity of flow is constant and is equal to 0.75 of the blade velocity. Find the mean diameter of the drum and volume of steam flowing per sec take blade tip angles at inlet and exit as 35° & 20° respectively.

A) $N = 400 \ rpm, \ x = 0.9, \ P = 10 kw. \quad , m = \frac{30 \times 1000}{3600} = 8.3 \ kg/s$

Assume $V_b = 40 \ m/s$.



$\theta = \beta = 35°, \ \phi = \alpha = 20°.$

$V_{w_1} + V_{w_2} = 12.6 \ cm = 126 \ mm$

$$\frac{V_{w_1} + V_{w_2}}{V_b} = \frac{126}{40} = 3.15$$

$$V_{w_1} + V_{w_2} = 3.15 V_b.$$

$$P = \frac{m V_b (V_{w_1} + V_{w_2})}{1000} \ kw$$

$$10 \times 10^3 = \frac{8.3 \times \cancel{40} \ V_b (3.15 V_b)}{1000} \Rightarrow V_b^2 = \frac{10000 \times \cancel{1000}}{26.145}$$

$V_b = \cancel{618.45}\ 19.55$

$V_b = \frac{\pi D N}{60}$

$\frac{60 \times V_b}{\pi N} = D$

$D = \frac{60 \times \cancel{618.45}^{19.55}}{\pi \times 400} = 0.93$ m at $P_1 = \cancel{0.6\text{ bar}}\ 1.6$ bar

$V_g = 1.0911\ m^3/kg$.

Volume of steam $= m x \times V_g$

$= 8.3 \times 0.9 \times 1.0911$

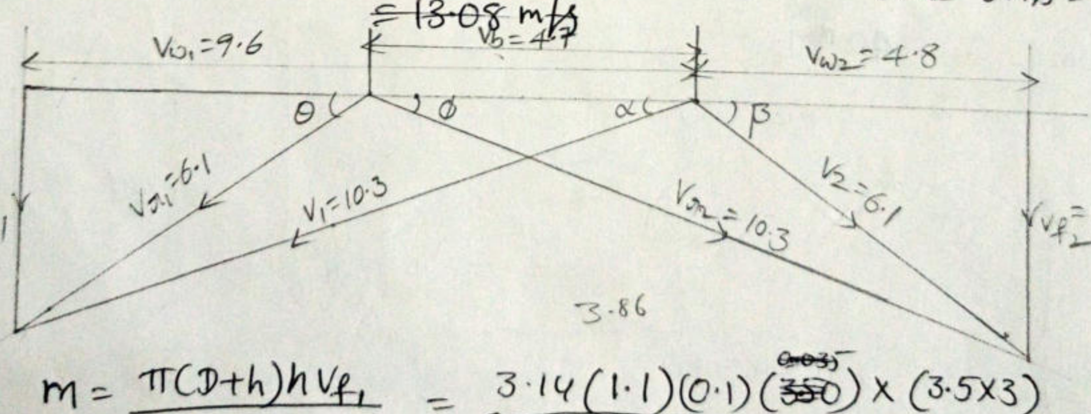
$= 8.15\ m^3/s$.

3) In a reaction turbine the bladetips are 35° & 20° in the direction of motion. the guide blades are of same shape as the moving blades. ~~At~~ a certain place in the turbine. The drum dia is 1m & blades are 10 cm height. ~~At~~ this place the steam has a pressure of 1.75 bar and dryness 0.935 If speed of turbine is 250rpm Find the power developed in the moving blades.

A) $V_b = \frac{\pi D N}{60}$, $\theta = \beta = 35°$, $\alpha = \phi = 20°$, $d = 1m$, $h = 0.1m$

$= \frac{3.14 \times \cancel{1000}^{1.1} \times 250}{60} = \cancel{13083.33}\ 14.39$ m/s. $d_m = d + h = 1.1$ m.

$\cancel{= 13.08\text{ m/s}}$ scale 3 m/s = 1 cm/s



$m = \frac{\pi(D+h) h V_{f_1}}{x V_g} = \frac{3.14(1.1)(0.1)(\cancel{350}^{\cancel{0.035}}) \times (3.5 \times 3)}{0.935 \times 1.00404}$

$= \frac{3.14 \times 1.1 \times 0.1 \times (10.5)}{0.935 \times 1.00404}$

$= 3.86$ kg/s

$V_f = 3.5$ cm/s $= 3.5 \times 3$, At $P = 1.75$ bar

$= 10.5$ m/s. $V_g = 1.00404$

$$power = \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$$

$$= \frac{3.86 \times 14.39 \times (28.8 + 14.4)}{1000}$$

$$P = 2.39 \text{ kW}.$$

4) A 50% reaction turbine running at 400 rpm has an exit angle of blades as 20° and the velocity of steam relative to the blades at exit is 1.35 times the mean blade speed. The steam flow rate is 8.33 kg/s. specific volume of steam is 1.381 m³/kg. calculate i) suitable height. assume ~~no.~~ rotor mean dia 12 times blade height. ii) workdone.

A) $\cancel{\theta = 20} = \beta$, $V_1 = 1.35 V_b$, $N = 400$ rpm. $(V_{r_2} = V_1)$

$V_1 = V_{r_2} = 1.35 V_b$, $D_m = 12h$, $\alpha = \phi = 20°$

$m = 8.33$ kg/s, $v_g = 1.381$ m³/kg, $h = D_m/12$

Assume $x = 1$

$V_{f_1} = V_1 \sin\alpha.$

$V_{f_1} = 1.35 V_b \times \sin 20.$

$= 0.4617 V_b.$

$= 0.4617 \times \frac{\pi(D_m \cancel{+h}) N}{60}$

$= 0.4617 \times \frac{\pi(D_m \cancel{+D/12}) \times 400}{60} = 0.4617 \times \frac{\pi(\overset{D_m}{12h}) \times 400}{60}.$

$$m = \frac{\pi(D_m \cancel{+h}) h \times V_{f_1}}{x \cdot v_g}.$$

$$8.33 = \frac{\pi(D_m \cancel{+D/12}) \frac{D_m}{12} \times \frac{0.4617 \times \pi \times (D_m \cancel{+D/12}) \times 400}{60}}{1 \times 1.381}$$

$$8.33 = \frac{\frac{D_m^2(\pi)}{12} \times \frac{0.4617 \times \pi \times D_m \times 400}{60}}{1.381}$$

$8.33 \times 1.381 = 0.2616\, D_m^2 \times 9.664\, D_m$, scale:- $10\ m/s = 1\ cm/s$.

$11.50 = 2.5283\, D_m^3$

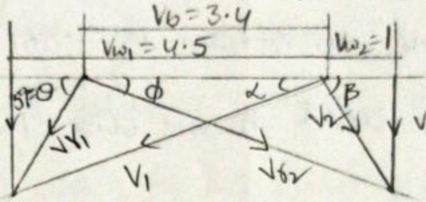
$D_m = 1.657 \Rightarrow D_m = 12h \Rightarrow h = \frac{D_m}{12} = 0.138\ m$

$V_b = \frac{\pi D_m N}{60} = \frac{3.14 \times 1.657 \times 400}{60}$

$= 34.68\ m/s.$

$V_1 = V_{r_2} = 1.35 V_b$

$= 1.35 \times 34.68 = 46.82\ m/s.$



workdone $= \cancel{2V_b V_1 \cos\alpha - V_b^2}$ $\frac{m V_b (V_{w_1} + V_{w_2})}{1000}$

$= \cancel{2 \times 34.6 \times 46.82 \cos(20) - (34.68)}$

$= \underline{15.88\ KW}.$

5) A Reaction turbine runs at 300 rpm and its steam consumption is 15400 kg/hr the pressure of steam at a certain ~~plate~~ stage is 1.9 bar and its dryness is 0.93 and power developed by the pair is 3 kw the discharge ~~plate~~ plane tip angle is 20° for both fixed & moving blades & axial velocity of flow 0.72 of the blade velocity. Find the drum dia. and blade height take the tip leakage & steam as 8%. but neglect the blade thickness.

A) $N = 300\ rpm$, $m = 15400\ kg/hr = 4.28\ kg/s$, $P = 1.9\ bar$, $x = 0.93$

$P = 3.5\ kw = 3.5 \times 10^3\ watt$, $\alpha = \phi = 20°$, $V_f = 0.72 V_b$.

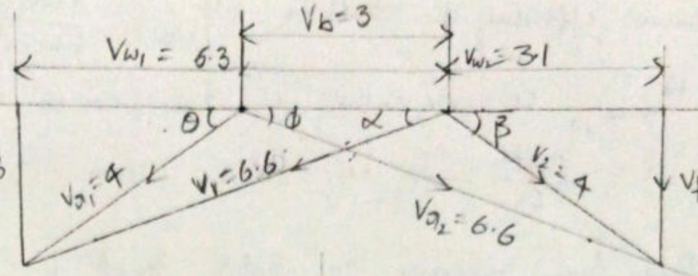
Since the tip leakage steam is 8%. therefore actual mass of steam flowing over the blades.

$m = 4.28 - (4.28 \times 0.08) = 3.94\ kg/s.$

$V_b = \frac{\pi d_m N}{60} = \frac{\pi d_m \times 300}{60} = 15.71 d_m\ m/s$

$V_{f_1} = 0.72 \times 15.71 d_m$
$= 11.3 d_m$ m/s.

scale: 5 m/s = 1 cm



$V_1 = \dfrac{V_{f_1}}{\sin\alpha}$

$= \dfrac{11.3 d_m}{\sin 20^\circ}$

$= 33.03.$

$V_{w_1} + V_{w_2} = ((9.4) \times 5)\ d_m$

$= 47 d_m$

$P = \dfrac{m V_b (V_{w_1} + V_{w_2})}{1000}$

$3.5 = \dfrac{3.94 \times 15.71 d_m (47 d_m)}{1000}$

$d_m^2 = \dfrac{3.5 \times 10^3}{2909.17} = 1.203$

$d_m = 1.09$ m.

$V_{f_1} = V_{f_2} = 11.3 d_m = 11.3 \times 1.09 = 12.39$

At $p = 1.9$ bar, $v_g = 0.929$ m³/kg.

$m = \dfrac{\pi d_m h V_{f_2}}{x v_g}$

$= \dfrac{3.14 \times 1.09 \times h \times 12.4}{0.93 \times 0.929} = 49.12 h$

$3.94 = 49.12 h \Rightarrow h = 0.08$
$= 80.2$

$d_m = d + h.$

$d = d_m - h$

$= 1.09 - \cancel{80}.08 = 1.01$ m

6) At a stage of reaction turbine the rotor diameter is 1.4 met and speed ratio is $0.7 \left(e = \frac{V_b}{V_1}\right)$ If the blade outlet angle is 20° & rotor speed 3000 rpm. Find the blade inlet angle & Blade efficiency. and also find % increase in blade efficiency and rotor speed If the turbine is designed to run the best theoretical speed.

Sol:

$D = 1.4$ m

$e = \dfrac{u}{V_1} = 0.7$

$\phi = 20 = \alpha$

$N = 3000$ rpm

$u = \dfrac{\pi D N}{60} = 220$ m/s.

$\therefore V_1 = \dfrac{220}{0.7} = 314.3$ m/s.

$e = 0.699 = \left(\dfrac{u}{V_1}\right)$

From diagram

i) $\theta = 55^\circ = \beta$

ii) Blade efficiency $\eta_{bl} = \dfrac{V_{r_2}^2 - V_{r_1}^2}{V_1^2} = \dfrac{(314.3)^2 - (130)^2}{(314.3)^2}$

$\eta_{bl} = 82.9\%.$

$$\boxed{\eta_{bl} = 2 - \dfrac{2}{1 + 2e\cos\alpha - e^2} = 90.5}$$

Maximum efficiency of turbine, $\eta_{max} = \frac{2\cos^2\alpha}{1+\cos^2\alpha} = 0.938$

For η_{max}, $u = V_1 \cos\alpha = 314.3 \cos 20^\circ = 295.3$ m/s.

$u = \frac{\pi D N_1}{60} \Rightarrow N_1 = 4044$ rpm.

$\therefore$ Percentage increase in rotor speed $= \frac{4044 - 3000}{\cancel{3000}\ 4044} \times 100 = \cancel{34.8}\ 25.8$ %

% increase in $\eta_{bl} = \frac{0.938 - 0.829}{0.938} \times 100 = 11.6\%$.

7) At a particular stage of reaction turbine the $V_b = 60$ m/s, steam is at a pressure of 3 bar with a temp of 200°C ~~with~~ if the fixed and moving blades at this stage have inlet angle 30° and exit angle 20°. Determine i) Blade height at this stage if the blade height is 1/10 of the mean blade ring dia. and steam flow is 10 kg/s. ii) power developed by pair iii) The heat drop required by the, pair if the steam expands with an efficiency of 85%.

A) $V_b = 60$ m/s, $P = 3$ bar, $T = 200^\circ C$

$\theta = \beta = 30^\circ$, $\phi = \alpha = 20^\circ$, $m = 10$ kg/s, $d/10 = h$.

$V_{w_1} + V_{w_2} = (8.5 + 5.1) = 13.6 \times 20 = 272$

$V_{f_2} = 3 \times 20 = 60$.

From steam tables, at $P = 3$ bar and $T = 200^\circ C$

$v_{sup} = 0.7164$ m³/kg

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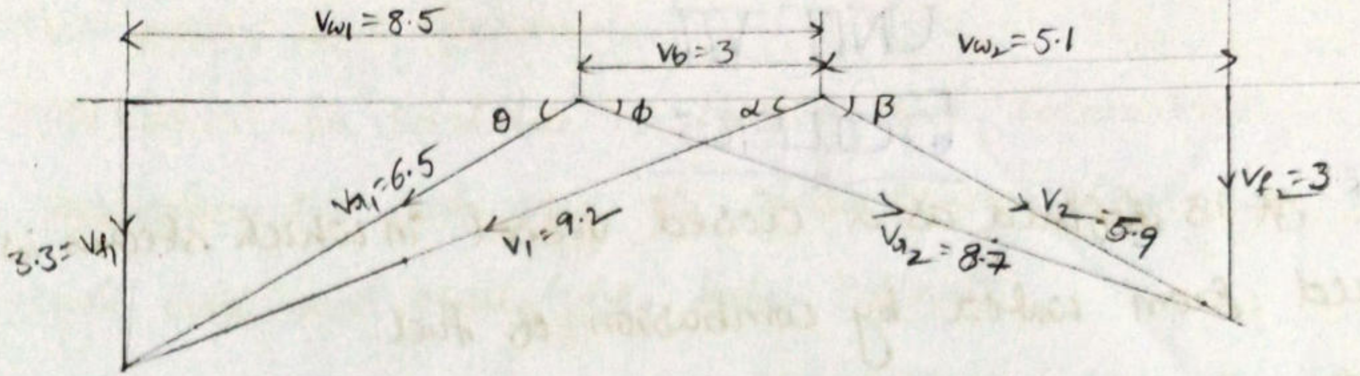


i) mass of steam flow $m = \dfrac{\pi(d+h)hV_{f_2}}{v_{sup}}$

$$10 = \frac{\pi(10h+h)h \times 60}{0.7164} = 2894h^2$$

$$h = 0.059\,m = 59\,mm.$$

ii) Power $P = \dfrac{mV_b(V_{w_1}+V_{w_2})}{1000}$

$$P = \frac{10 \times 60 \times (272)}{1000} = 163.2\ KW.$$

iii) Since the steam expands with an efficiency of 85%. therefore heat drop required by the pair.

$$\eta = \frac{Power}{\Delta h}$$

$$= \frac{163.2}{0.85} = 192\ kJ/s.$$

Introduction to Refrigeration

Refrigeration:
continuous extraction of heat from low temp. body & rejecting to high temp. body.

(or)

It is a process of controlling & maintaining the required temp. which is less than the atmospheric temperature.

Applications:

1) Human comfort.
2) For transportation of medicines, food materials from one place to another place.
3) Preserving the medicines.
4) For different industrial manufacturing processes.
5) Air crafts.
6) Automobile air conditioners
7) preserving the chemicals etc

Work: It is the displacement by some force.

Energy: The capacity to do work.

- stored energy (PE, KE, IE)
- transit energy (heat, work etc)

Heat:

- conduction
- convection
- Radiation

Sensible heat: Heat required to increase the temperature of the body upto boiling point. is called sensible heat.

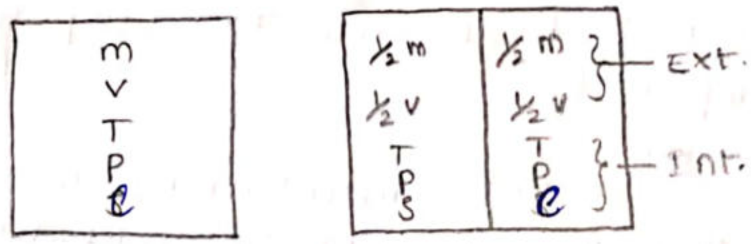
Latent heat: It is the heat required to change the phase.

Specific heat: Heat required to increase the temperature of 1°c per unit mass.

Property: Measurable quantity.

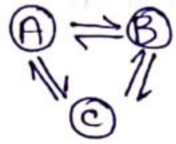
↳ Extensive property: Depends on mass

↳ Intensive property: Independent of mass.



Zeroeth law of thermodynamics:

If two bodies are in thermal equilibrium with third body then the two bodies are in thermal equilibrium with eachother



First law of thermodynamics:

Heat and work are mutually convertable

(or)

Energy can neither be created nor destroyed.

Second law of thermodynamics:

1) Kelvin - plank statement:

It is impossible to construct an engine working in a cyclic process whose sole purpose is to convert heat energy from a single thermal reservoir ~~into an~~ into an equivalent amount of work.

2) Classius statement:

It is impossible for a self acting machine working in a cyclic process to transfer heat from low temperature body to high temperature body without any aid of external agency.

Thermodynamic processes:

1) Constant pressure process:

$$\frac{V_1}{V_2} = \frac{T_1}{T_2}$$

2) constant volume process:

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}$$

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2) Evaporative Refrigeration:

Heat is observed when a the liquid evaporates. In this method dry atmospheric air is passed through the water soaked packings. cool air is blown into rooms (or) chambers to provide human comfort. Thermodynamically the evaporative cooling is an adiabatic exchange of heat between air and water. It is similar to a process called adiabatic saturation.

Applications:

1) Evaporation of moisture from the skin surface of human helps to keep him cool.

2) Desert bag which is used to keep drinking water cool. Always bag remains moisture because it is not water-proof. Under desert conditions usually both hot and dry, moisture on the surface of the bag evaporates rapidly. Then water inside the bag gets cool.

3) To make artificial snow.

4) Evaporative condensors.

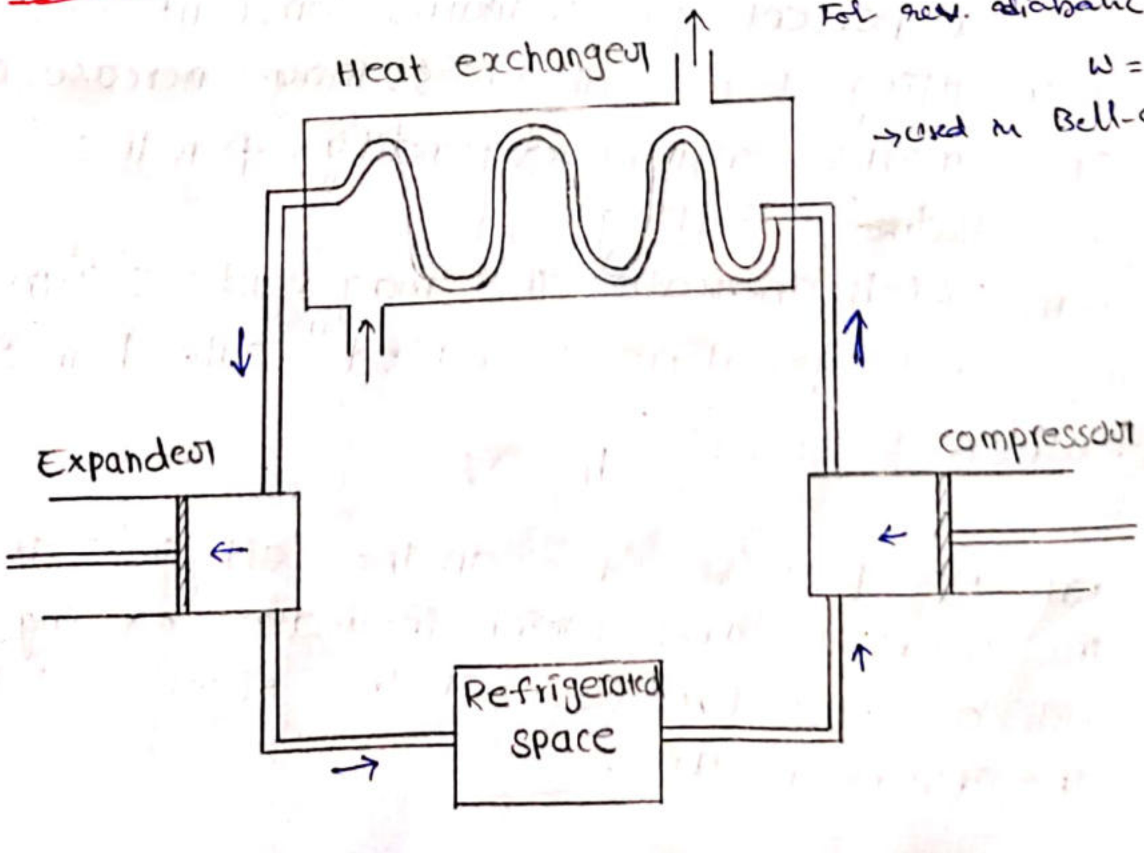
3) Air expansion Refrigeration:

I^{st} law $\rightarrow$ $Q = \Delta u + W$

For rev. adiabatic $Q = 0$

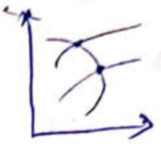
$W = -\Delta u$

$\rightarrow$ used in Bell-coleman ref. sys

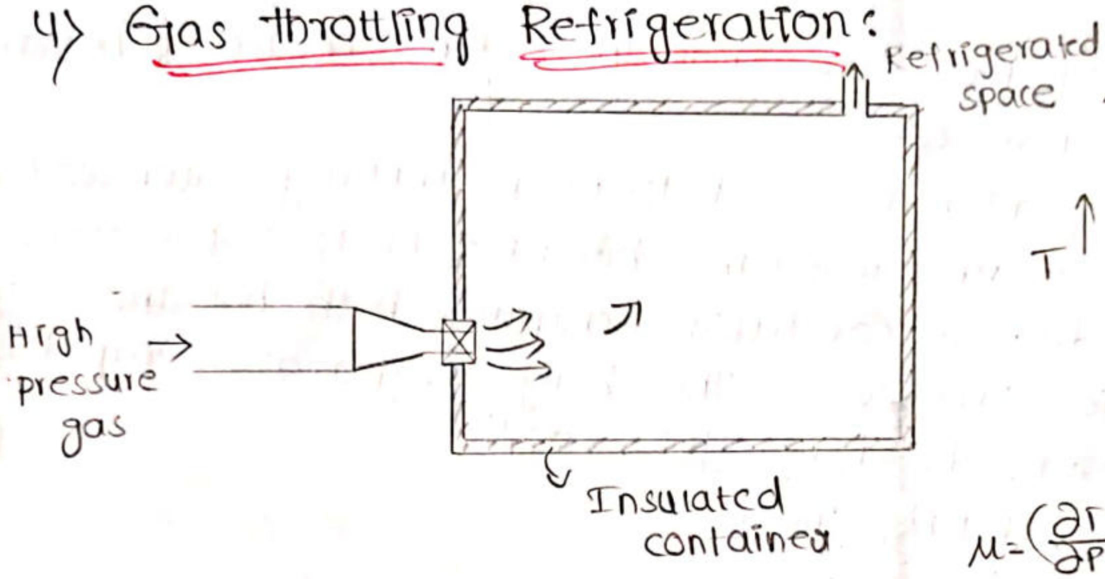


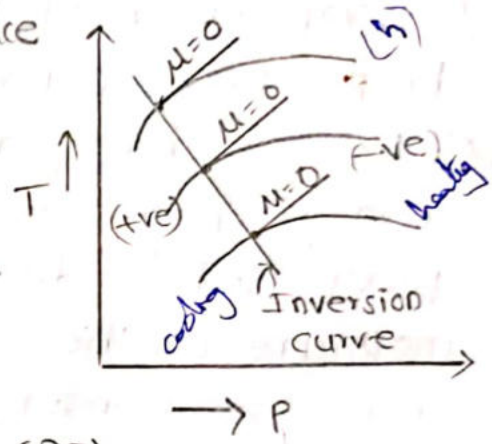
The temperature of the gas or air can be reduced by an adiabatic expansion of air. This method involves compression and expansion of the air.

For example if air is at 27°c & 5 bar pressure is allowed to expand down to 1 bar pressure adiabatically its temperature falls to −83.5°c by the relation $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$



4) Gas throttling Refrigeration:





$\mu = \left(\frac{\partial T}{\partial P}\right)_h$

Adiabatic throttling process is a constant enthalpy process. Enthalpy being a function of temperature, the temperature of a perfect gas remains constant before and after throttling. But real gases, may increase or decrease or remains constant depending upon the state of gas after ~~the~~ throttling.

The term which indicates the magnitude & sign of the change in temperature is called "Joule Thomson coefficient". where $\mu = \left(\frac{\partial T}{\partial P}\right)_h$

→ μ will be positive (+ve) on the left side of the inversion curve which indicates cooling.

→ μ will be negative (−ve) on the right side of the inversion curve.

3) Isothermal process:

$$P_1V_1 = P_2V_2, \quad T_1 = T_2$$

4) Adiabatic process:

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)\left(\frac{V_1}{V_2}\right)$$

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1}$$

5) Polytropic process:

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1}$$

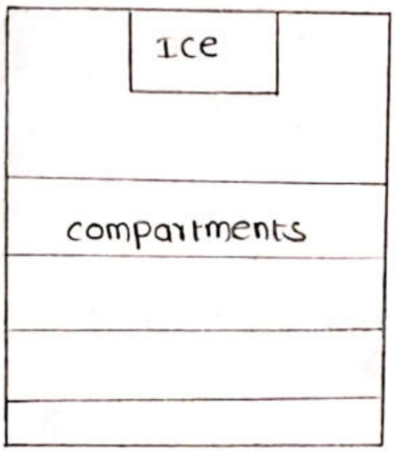
Methods of refrigeration:

1) Ice refrigeration
2) Evaporative refrigeration
3) Air expansion refrigeration
4) Gas throttling refrigeration
5) vapour refrigeration system
6) Steam jet refrigeration system.
7) Refrigeration by using liquid gas
8) Dry ice refrigeration

1) Ice refrigeration:

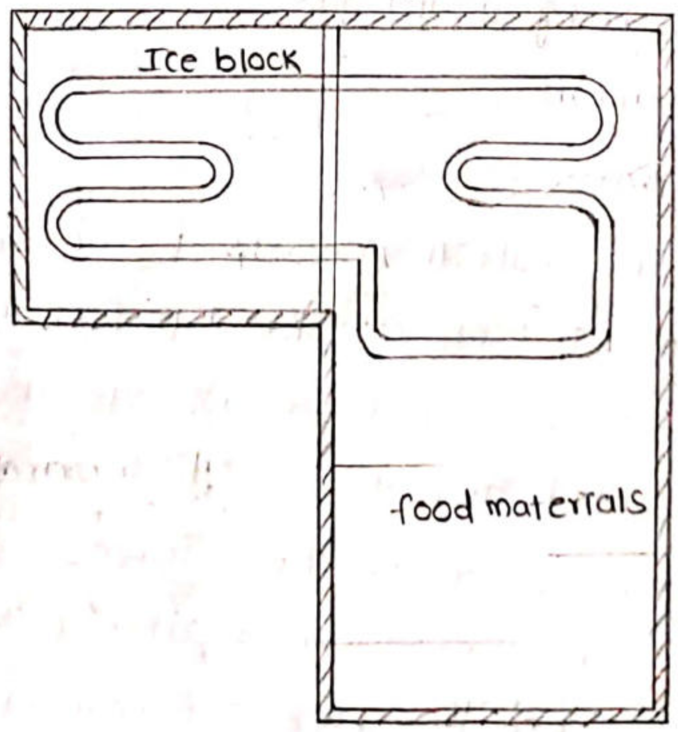
The ice refrigeration consists of an insulated cabin equipped with a tray or tank at the top for holding block of ice pieces. Shelves for food are located below the ice compartment. cold air flows downward from ice compartment & cools food in the shelves. below. Air returns from the bottom of the cabinet up, the sides and back up the cabinet which is warmer ~~close~~ flows over the ice and again ~~close~~ flows down over the shelves to be cooled. Temperature inside the refrigerator usually ranges from 5 to 10°c.

If it is necessary to decrease the temperature less than 0°c $\underbrace{\text{ice+salt}}_{\text{frigorific mixture}}$ are used.



Direct ice refrigeration

Indirect ice refrigeration



In this method the gas at high pressure is throttled down to a lower pressure through a porous plug or orifice. If the positive value of μ is maintained the temperature falls and gas is cooled. The temp. drop would be around 0.25° for each atmospheric drop in pressure.

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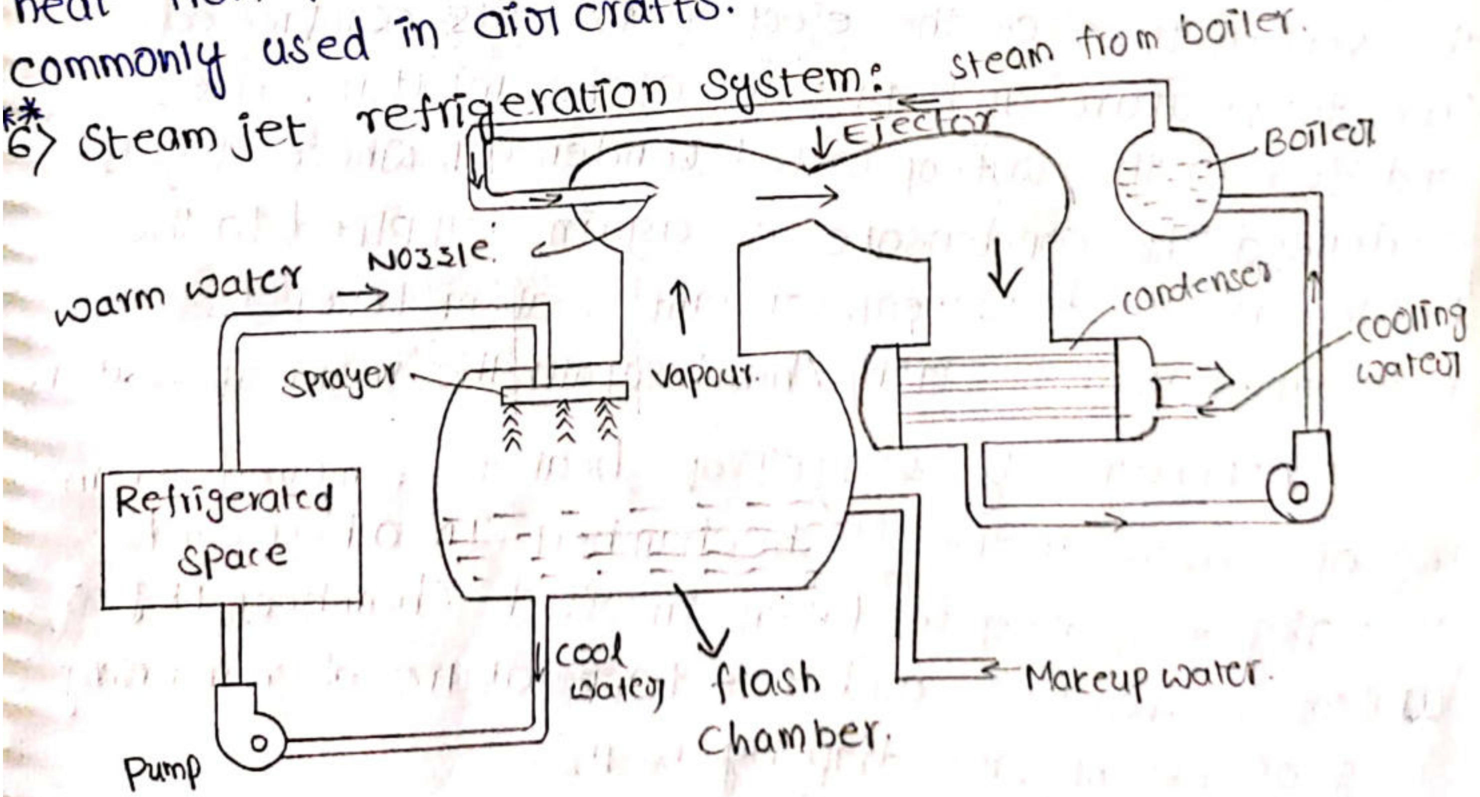
8) Dry ice refrigeration system:

Solid carbon dioxide is called dry ice. It can be pressed into different shapes and sizes of flakes (or) slices. It evoporates directly into vapour state without passing through liquid state. This phenomenon is called as sublimation. At standard atmospheric pressure dry ice sublimates as low as $-78^\circ C$. Therefore dry ice flakes are placed between food cartons during transportation over a long period. Dry ice observe heat from food cartons thus freezing the contents. It is commonly used in air crafts.

** 6) Steam jet refrigeration system:



Working principle:

Boiling point of water at which it starts evaporat~~ed~~ing
depends on the value of pressure, steam jet refrigeration
system use this principle. The flash chamber is a
large vessel & is heavily insulated to avoid the rise
in temperature of water due to high ambient temper-
ature. It is fitted with perforated pipes for spraying
water. The warm water coming out from the refrigera-
ted space is sprayed into the flash chamber. where
some of water converted to vapour after observing
the latent heat, there by cooling the rest of water.
The high pressure steam from the boiler is passed
through the nozzle the~~re~~ by increasing its velocity. This
high velocity steam in the ejector ~~is~~ to maintain ~~good~~ and
entrain the water vapour from the flash chamber
which would result in further formation of vapours.
The mixture of steam and water vapour passes through
the venturi tube of the ejector and gets compressed.
The temperature and pressure of the mixture rises
and fed to the water cooled condenser, where it gets
condensed. The condensate is again supplied to the
boiler as feed water. constant water level is
maintained in the flash chamber with the makeup water.

Approximately 2394 KJ of heat is removed from
1kg of water in the flash chamber. In other words
if 100kg of water is taken in flash chamber 1kg of
water evaporates and the temperature of remaining
99kg of water will drop by 5.75°c

$$Q = m\,c_p\,\Delta T$$

$$\Delta T = \frac{Q}{m\,c_p} = \frac{2394}{99(4.2)} = 5.75^\circ c$$

Evaporation of one more kg of remaining water (5) will become another 5.75°c colder. By continuing this process the water in the flash chamber get freezes.

7. Liquid gas refrigeration:

Liquid gas such as Nitrogen or carbon dioxide which is non-toxic is filled in an insulated cylinder. cylinder is connected to a pipe with intermittent perforations. operating a control valve, required quantity of liquid nitrogen is passed through this pipe. It is sprayed through perforations into the space to be refrigerated. Temperature produced is then nearly −20°c. This type of spray cooling is used in transportation of meat, fish, fruits etc. The liquid gas vapourises after absorbing heat & vapour is released ~~through~~ to the atmosphere.

Types.

1) cold plate cooling systems (plates (or) baffles are arranged)
2) spray cooling systems (spray headers are arranged)

5) vapour refrigeration system:

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In vapour refrigeration system instead of air vapours like ammonia (NH_3), carbon dioxide (CO_2), sulphur dioxide (SO_2) are used as working fluids. The heat carried away by the air from the refrigerator in air refrigeration system is in the form of sensible heat as there is no change of phase so that effectiveness of air refrigeration system is less.

In vapour refrigeration system the heat carried away by the vapours from the refrigerated space is in the form of latent heat of refrigerant so that the capacity of the refrigeration per unit kg of refrigerant is far superior than the air refrigeration system.

This is of two types

1) vapour compression refrigeration system (VCR)
2) vapour absorption refrigeration system (VAR)

Units of Refrigeration:

Unit of refrigeration is ton. The

A ton here ~~is~~ does not mean mass but it is the measure of rate of heat transfer.

The rate of removal of heat in cooling operation was expressed in terms of Killograms or tons of ice required per unit time usually hour (or) day.

A ton of refrigeration is defined as the quantity of heat required to be removed from one ton of ice within 24 hours when initial condition of water is at $0^\circ c$.

Latent heat of fusion of ice = 336 KJ/Kg

When 1000 kg of ice melts in 24 hours it produces a cooling effect at the rate of 233 KJ/min $\left(\frac{1000 \times 336}{24 \times 60}\right)$ (or)

If heat is removed from water at the rate of 233 KJ/min we get 1000kg of ice per day.

∴ The rate of cooling is designated as ton of refrigeration.

Approximate value taken is 210 KJ/min (or) 3.5 KJ/sec

∴ $\boxed{1\ T.R = 3.5\ KW}$

A machine which has a capacity of producing a cooling effect of 210 KJ/min is designated as One ton machine

Coefficient of performance (C.O.P):

It is defined as the ratio of net refrigeration effect to the work supplied.

$$C.O.P = \frac{\text{Net refrigeration effect}}{\text{work supplied}}$$

$$COP = \frac{N}{W}$$

C.O.P is always greater than 1.

$$\text{Relative } C.O.P = \frac{\text{actual COP}}{\text{theoritical COP}}$$

calculate heat removed with 1kg of water at 25°c is converted into ice at 0°c?

Sol:

Heat removed with 1kg of water at 25°c to ice at 0°c

Heat removed to convert water at 25°c to water at 0°c

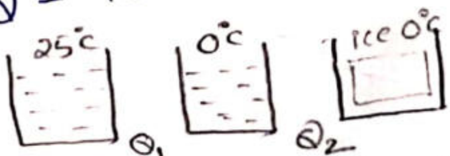
Q = Heat removed to convert water at 0°c to Ice at 0°c + Heat removed.

$= Q_1 + Q_2$

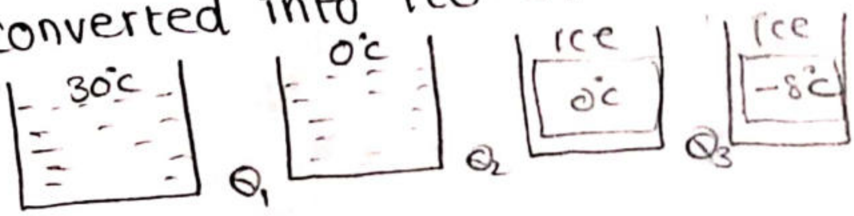
$Q_1 = mc_p\Delta T = 1 \times 4.2 \times (25-0) = 105$ KJ/kg.

$Q_2 = 336$ KJ/kg

$\therefore Q = 105 + 336 = 441$ KJ/kg.



calculate heat removed, with if 1kg of water at 30°c is converted into ice at −8°c.



$Q = Q_1 + Q_2 + Q_3$

$Q_1 = mc_p\Delta T = 1 \times 4.2 \times 30 = 126$ KJ/kg

$Q_2 = 336$ KJ/kg

$Q_3 = mC_p\Delta T$

$= 1 \times 2.1 \times (8)$

$= 16.8$ KJ

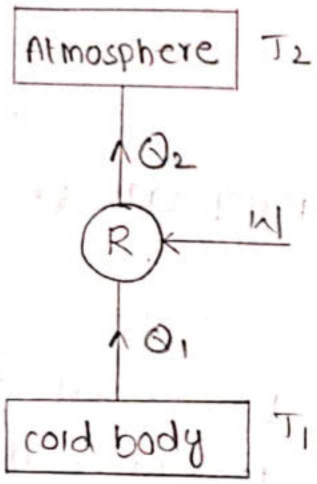
$Q = 478.8$ KJ

C_p of ice = 2.108 kJ/kgk

C_p of water = 4.187 "

C_p of steam = 1.996 "

Refrigerator



$$(COP)_R = \frac{O/P}{I/P}$$

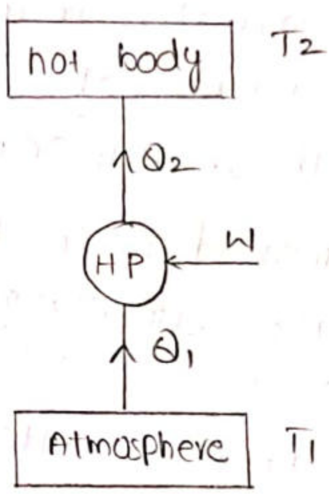
$$= \frac{Q_1}{W}$$

$$W = Q_2 - Q_1$$

$$(COP)_R = \frac{Q_1}{Q_2 - Q_1}$$

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

Heat pump

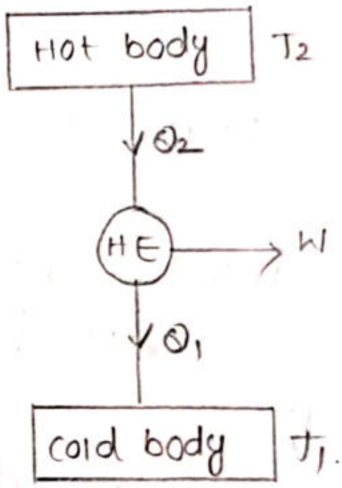


$$(COP)_{HP} = \frac{Q_2}{W}$$

$$= \frac{Q_2}{Q_2 - Q_1}$$

$$(COP)_{HP} = \frac{T_2}{T_2 - T_1}$$

Heat engine



$$\eta = \frac{W}{Q_2}$$

$$= \frac{Q_2 - Q_1}{Q_2}$$

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

COP of heat pump is also known as "~~energy~~ ~~permo~~ (7)
Performance ratio."

$$(COP)_{HP} = \frac{Q_2}{W} = \frac{Q_1+W}{W} = \frac{Q_1}{W}+1$$

$$(COP)_{HP} = (COP)_R + 1$$

$$\boxed{(COP)_{HP} - (COP)_R = 1}$$

from the above relation it is evident that C.O.P of refrigeration system may be less than (or) greater than 1 depending upon the type of the system, but C.O.P of heat pump is always greater than 1.

1) The refrigerator must work b/w the temp. difference of $-5°c$ & $20°c$. calculate the COP of the system.

Sol:

$T_1 = -5°C = 268\,K$

$T_2 = 20°C = 293\,K$

$$(C.O.P)_R = \frac{T_1}{T_2-T_1} = \cancel{\frac{+5}{20+5} = -0.2}$$

~~$(COP)_R = 27.2.8$~~

$$(C.O.P)_R = \frac{268}{293-268} = 10.72.$$

2) A machine operating on carnot cycle operates b/w 304K & 260K determine the following.

1) COP of refrigeration.

2) COP of heat pump.

3) efficiency of heat engine.

1) $T_1 = 260K$

$T_2 = 304K$

$$(COP)_R = \frac{260}{304-260} = 5.9$$

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2) $T_1 = 260K$

$T_2 = 304K$

$$(COP)_{HP} = \frac{\cancel{260}\ 304}{304-260}$$

$= 6.9$

3) $T_1 = 260K$, $T_2 = 304K$

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

$$= \frac{304-260}{304}$$

$\eta = 14.4\%$

3) The capacity of refrigeration is 150 TR & power required is 81.4KW calculate C.O.P.

sol: $Q_1 = 150\,TR$

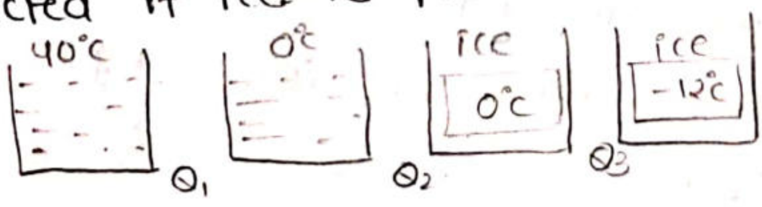
$W = 81.4KW$

$$(COP)_R = \frac{150 \times 3.5}{81.4}$$

$= 6.44$

4) Findout the heat extracted from the ice at 0°C from ~~fuel~~ water which is at 40°C, also find heat extracted if ice is produced at -12°C.

sol:



$Q_1 = 1 \times 4.2 \times 40 = 168\,KJ$

$Q_2 = 336\,KJ$

$Q_3 = 1 \times 2.1 \times 12 = 25.2\,KJ$

$Q = Q_1 + Q_2 + Q_3$

$= 529.2\,KJ$

5) 500 kg of fruits is supplied to a cold storage plant at 20°C. The cold storage is maintained at −5°C & the fruits gets cooled to the storage temperature in 10 hrs. The latent heat of freezing is 105 KJ/kg & specific heat of fruits is taken as 1.256 KJ/kgK. Find the refrigeration capacity of the plant.

Sol: $Q_1 = mC_p\Delta T$

$= 500 \times 1.256 \times (20+5)$

$= \cancel{15700\,KJ}$ $= \cancel{358588}$ KJ

Latent heat of freezing = 105 KJ/kg.

total latent heat of freezing = 500 × 105

= 52500 KJ

∴ Total heat extracted per 10 hrs = $\cancel{68200}$ KJ | = ($\cancel{15700 + 52500}$)

= $\cancel{411088}$ KJ

Total heat extracted per 1hr = $\frac{68200}{10}$ = $\cancel{6820}$ KJ/hr

$= \frac{411088}{10 \times 3600}$ = 1.894 KJ/sec

= $\cancel{11.419}$ KJ/sec. = 1.894 KW

= $\cancel{3.26}$ TR. = 0.54 TR

6) An ice plant produces 10 tons of ice per day at 0°C using water at 40°C if the plant C.O.P is 2.5. Find out power required to run the compressor.

Sol: Given

C.O.P = 2.5

$Q_1 = mC_p\Delta T$

$= 10 \times 10^3 \times 4.2 \times 40$ 313

$Q_1 = \cancel{168} \times 10^4$ KJ

$Q_2 = 336$ KJ/kg

$= 10 \times 10^3 \times 336$ KJ

$= 336 \times 10^4$ KJ

$Q = \cancel{504 \times 10^4}$ KJ $\cancel{16506 \times 10^3}$ KJ

$C.O.P = \frac{Q}{W}$ ⇒ $\frac{16506 \times 10^3}{2.5} = W$

= 6602400 KJ/day

= 1834 KJ/s

$W = \frac{\cancel{504 \times 10^4}}{2.5}$

$W = 2016 \times 10^3$ KJ/day

$\frac{2016 \times 10^3}{24 \times 60 \times 60} = \cancel{23.3}$ KW.

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7) A cold storage plant is required to store 20 tons of fish. The fish is supplied at a temperature of 30°c. The sp. heat of fish above the freezing point is 2.93 KJ/KgK. The sp. heat of fish below the freezing point is 1.26 KJ/KgK. The fish is stored in the cold storage which is maintained at −8°c. The freezing point of fish is −4°c. The latent heat of fish is 235 KJ/Kg. If the plant requires 75 KW to drive it, find i) the capacity of the plant ii) Time taken to achieve cooling & assume actual COP of the plant as 0.3 of carnot COP.

Sol:

mass = 20 tons

$T_1 = -8°c = 265K$

$T_2 = 30°C = 303K$

$T_3 = -4°c = 269K$

$P = 75 KW = 75 KJ/s$

$CP_{AF} = 2.93 \ KJ/KgK$

$CP_{BF} = 1.26 \ KJ/KgK$

Actual COP = 0.3 of carnot COP

h_{fg} fish = 235 KJ/Kg.

$$30°c \ (T_2) \xrightarrow{AF} -4°c \ (T_3) \xrightarrow{BF} -8°c \ (T_4)$$

$$\text{Carnot COP} = \frac{T_1}{T_2 - T_1} = \frac{265}{303-265} = 6.97$$

Actual COP = 0.3 × 6.97
= 2.09

$$C.OP = \frac{NRE}{W}$$

NRE = C.O.P × W = 2.09 × 75 = 156.9 KJ/S

$$NRE = \frac{156.9}{3.5} \ T.R = 44.83 T.R$$

NRE = 44.83 T.R.

Capacity of the plant = 44.83 T.R.

Heat extracted (or) removed from fish above freezing point (9)

$Q_1 = m C_{P_{AF}} (T_2 - T_3)$

$= 20000 \times 2.93 (303 - 269)$

$Q_1 = 1992400 \text{ KJ}$

Heat extracted from fish below freezing point

$Q_2 = m C_{P_{BF}} (T_3 - T_1)$

$= 20000 \times 1.26 (269 - 265)$

$= 1.008 \times 10^5 \text{ KJ}$

Total latent heat of the fish $Q_3 = m \times h_{fg}$

$= 20000 \times 235$

$= 47 \times 10^5 \text{ KJ}$

Total heat removed $Q = Q_1 + Q_2 + Q_3$

$= (19.92 + 1.008 + 47) 10^5$

$= 67.93 \times 10^5 \text{ KJ.}$

Time taken to achieve cooling $= \dfrac{\text{Total heat}}{\text{heat extracted/sec}}$

$= \dfrac{67.93 \times 10^5}{156.9}$

$= 43296.3 \text{ sec.}$

$t = 12$ hours.

Air Refrigeration System

→ Open Air Ref. System

→ closed (or) Dense air ref. system

open air refrigeration system:

In this cycle air is directly led to the space to be cooled, allowed to circulate through the cooler and then return to the compressor to start another cycle. Since the air is supplied to the refrigerator at atmospheric pressure therefore volume of air handled by the compressor & expander is large thus the size of the components should be large. Another disadvantage is that the moisture is regularly carried away by the air circulated through the cold space. This leads to the formation of frost at the end of expansion process & clog the line. Thus in an open system ~~the~~ a drier should be used.

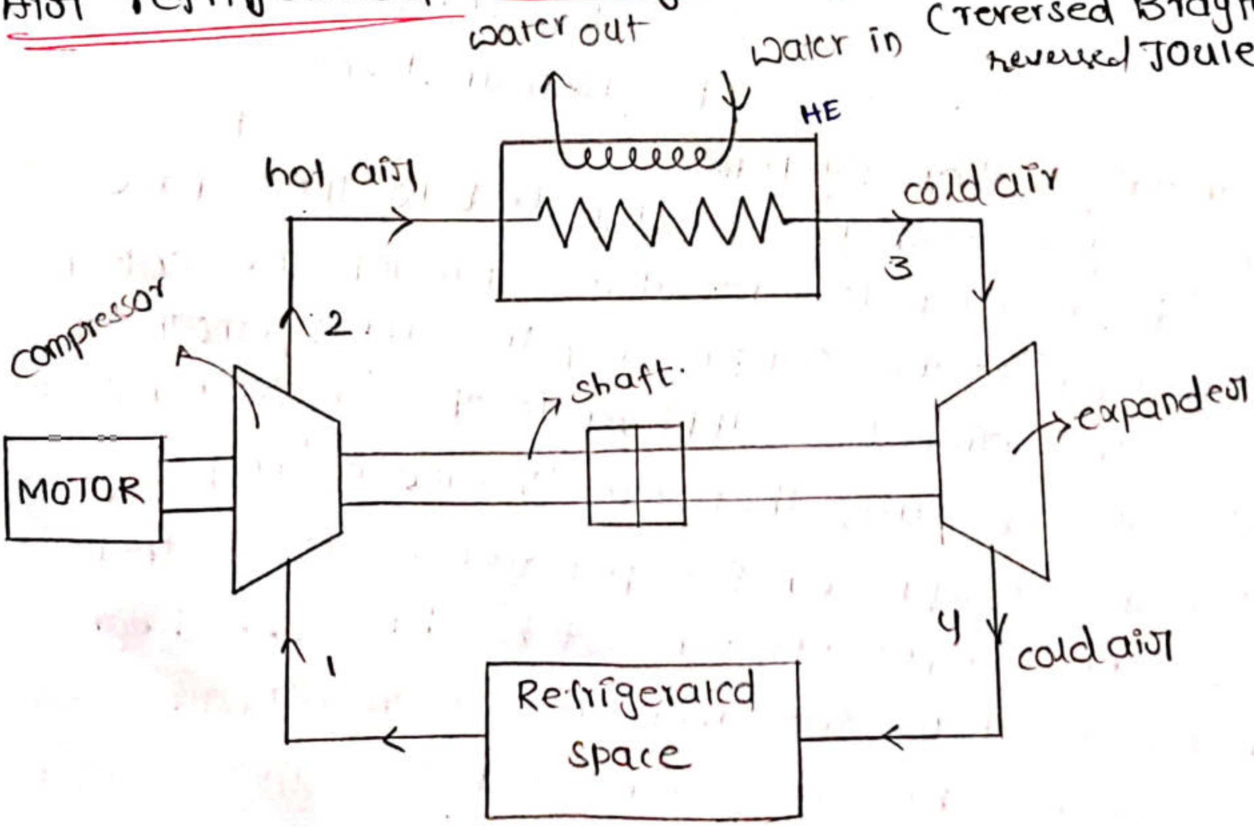
close con) dense air refrigeration system:

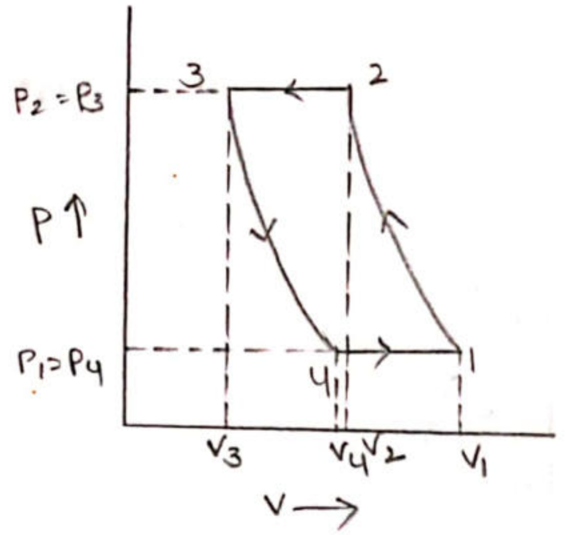
In this system the air is passed through the pipes & components in the system at all times. The air in this system is used for absorbing heat from other fluid (brine) & this cooled brine is circulated into the space which is to be cooled. The air in the closed system does not comes in contact directly with the refrigerated space.

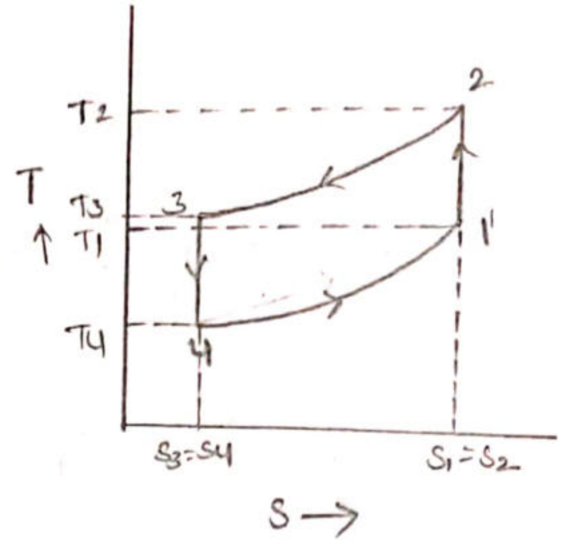
Thermodynamic advantages:

* Since it can work at a suction pressure ~~and~~ higher than that of atmospheric pressure, therefore the volume of air handled by compressor or expander is small as compared to open system.
* The operating pressure ratio can be reduced which results in higher COP.

Air refrigerator working on Bell-coleman cycle:
(reversed Brayton or reversed Joule cycle)







The cycle consists of following processes.

1) Isentropic compression process (No heat is transferred)

2) Constant pressure cooling process ($Q = mC_p(T_2 - T_3)$)

3) Isentropic expansion process (No heat is transferred $Q = 0$)

4) Constant pressure ~~expansion~~ heating process ($Q = mC_p(T_1 - T_4)$)

* This system mainly works on Bell-Coleman cycle.
* Air is used as a refrigerant in this system.
* Expansion cylinder will drop some power which is utilized to reduce the compressor power.
* Bell-coleman refrigeration system may work on open or closed system. (closed system is preferred)

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Wednesday

Heat rejected $= C_p(T_2 - T_3)$

Heat absorbed $= C_p(T_1 - T_4)$

work done = Heat rejected − heat absorbed

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

$$C.O.P = \frac{\text{Net ref. effect}}{\text{work done}}$$

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

$$C.O.P = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}$$

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

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

$$\therefore \frac{T_2}{T_1} = \frac{T_3}{T_4} \text{ (or) } \frac{T_2}{T_3} = \frac{T_1}{T_4}$$

$$COP = \frac{T_4\left(\frac{T_1}{T_4} - 1\right)}{T_3\left(\cancel{\frac{T_1}{T_4} - 1}\right) - T_4\left(\cancel{\frac{T_1}{T_4} - 1}\right)}$$

$$\boxed{COP = \frac{T_4}{T_3 - T_4}}$$

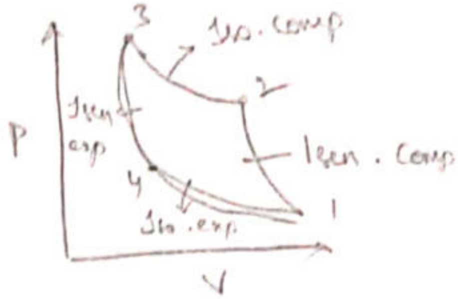
compression ratio $r_p = \frac{P_2}{P_1}$ (or) $\frac{P_3}{P_4}$

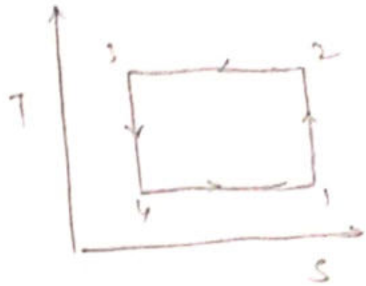
$$COP = \frac{1}{\frac{T_3}{T_4} - 1}$$

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

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

Reversed Carnot cycle





Bell-coleman is a modified cycle of rev Carnot cycle

Some times the compression & expansion process takes place according to the law of $\boxed{PV^n = \text{constant}}$

In isentropic process heat transfer during compression & expansion is 'zero' whereas in other

T.D processes there may be heat transfer from the surroundings. so that 11

work done = Heat rejected - heat absorbed

Work done by compressor during 1-2 process per kg of air

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

$$W_c = \frac{n}{n-1}(RT_2 - RT_1)$$

Work done by expander per kg of air,

$$W_e = \frac{n}{n-1}(P_3V_3 - P_4V_4)$$

$$= \frac{n}{n-1}R(T_3 - T_4)$$

Net work done $w = W_c - W_e$

$$= \frac{n}{n-1}R\left[(T_2 - T_1) - (T_3 - T_4)\right]$$

Heat absorbed during 4 to 1 process $= C_p(T_1 - T_4)$

$$COP = \frac{\text{Heat absorbed}}{\text{work done}}$$

$$= \frac{C_p(T_1 - T_4)}{\frac{n}{n-1}R\left[(T_2 - T_1) - (T_3 - T_4)\right]}$$

$$= \frac{C_p(T_1 - T_4)}{\frac{n}{n-1}C_v(\gamma - 1)\left[(T_2 - T_1) - (T_3 - T_4)\right]}$$

$R = C_p - C_v$

$\frac{C_p}{C_v} = \gamma$

$R = C_v(\gamma - 1)$

$C_v = \frac{R}{\gamma - 1}$

$$\boxed{COP = \frac{T_1 - T_4}{\frac{n}{n-1} \cdot \frac{\gamma - 1}{\gamma}\left[(T_2 - T_1) - (T_3 - T_4)\right]}}$$

Q) In an open type of refrigerating installation 1000kg of atmospheric air is circulated per hour. The air is drawn from the cold chamber at temperature 7°C and 1 bar & then compressed isentropically to 5bar. It is cooled at this pressure to 27°C and then ~~led~~ let to the expander where it expands isentropically down to atmospheric pressure and is discharged to the cold chamber. Find the following.

i) Heat extracted from the cold chamber per hour

ii) Heat rejected to the cooling water per hour

iii) cop of the system.

Take for air $\gamma = 1.4$ $C_p = 1$ KJ/kgK.

Sol:

$T_1 = 7°C = 280K$

$P_1 = 1$ bar

$P_2 = 5$ bar $= P_3$

$T_3 = 27°C = 300K$

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

$$\Rightarrow T_2 = \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} \times 280$$

$$\boxed{T_2 = 443.46K}$$

$P_4 = 1.01325$ bar $= P_1$

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

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

$$T_4 = \frac{300}{\left(\frac{5}{1}\right)^{\frac{0.4}{1.4}}} = 189.4K$$

$$\boxed{T_4 = 189.4K}$$

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i) Heat extracted $= mC_p(T_1 - T_4)$

$= 1(280 - 189.4) \times 1000$

$= \cancel{90.6\,KJ}\ 90600\ KJ/hr = 25.16\ KJ/s$

ii) Heat rejected $= mC_p(T_2 - T_3)$

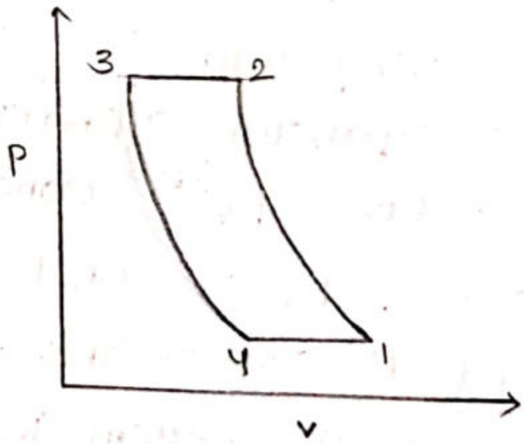
$= 1(443.46 - 300) \times 1000$

$= 143460\ KJ/hr$

iii) $COP = \dfrac{90600}{143460 - 90600} = 1.713$

9) A refrigerator working on Bell coleman cycle operates b/w pressure limits of 1.05 bar & 8.5 bar. Air is drawn from the cold chamber at 10°c. Air coming out of compressor is cooled to 30°c before entering into the expansion cylinder. Expansion & compression follows the law of $pv^{1.35}$ = constant. Determine theoritical cop of the system. Take $\gamma = 1.4$, $C_p = 1\ KJ/kgK$ & find out heat absorbed & work done.

Sol:



$P_1 = 1.05\ bar = P_4$

$P_2 = 8.5\ bar = P_3$

$T_1 = 10°c = 283$

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

$$T_2 = \left(\frac{8.5}{1.05}\right)^{\frac{1.35-1}{1.35}} \times 283$$

$$\boxed{T_2 = 486.69\,K}$$

$T_3 = 30°c = 303K$

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

$$\Rightarrow T_4 = \frac{T_3}{(P_3/P_4)^{\frac{n-1}{n}}}$$

$$T_4 = \frac{303}{\left(\frac{8.5}{1.05}\right)^{\frac{0.35}{1.35}}}$$

$$\boxed{T_4 = 176.18\,K}$$

Heat absorbed $= C_p(T_1 - T_4)$

$= 1(283 - 176.18)$

$= 106.82$ KJ

Work done $W = \frac{n}{n-1} R \left[(T_1 - T_2) - (T_3 - T_4)\right]$

$(T_2 - T_1)$

$= \frac{1.35}{0.35} \times 0.287 \left[(283 - 486.69) - (303 - 176.18)\right]$

$W = -85$ KJ (work consumed by the system)

$= 85$ KJ

$$C.O.P = \frac{106.8}{85} = 1.256$$

10) An air refrigeration system works b/w pressure limits of 1 bar & 4 bar. The temperature of air entering into the compressor is 15°c & entering into the expansion cylinder is 30°c. The expansion follows the law $PV^{1.25} = $ const. The compression follows the law of $PV^{1.35} = $ const. Take $C_p = 1$ KJ/kgK, $C_v = 0.7$ KJ/kgK. Find 1) COP of the system 2) If air circulation through the system is 25 kg/min, find the refrigerating capacity of the system.

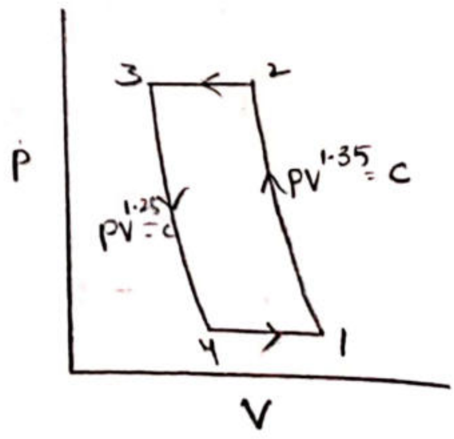
Sol:

$P_1 = 1$ bar $= P_4$

$P_2 = 4$ bar $= P_3$

$T_1 = 15°C = 288$ K

$T_3 = 30°C = 303$ K



Compression process:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{1.35-1}{1.35}}$$

$$T_2 = \left(\frac{4}{1}\right)^{\frac{0.35}{1.35}} \times 288$$

$$\boxed{T_2 = 412.5\ K}$$

$$W_c = \frac{n}{n-1} \cdot R(T_2 - T_1)$$

$$= \frac{1.35}{0.35} \times 0.287\ (412.5 - 288)$$

$W_c = 137.82$ KJ/kg 144

Expansion process:

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{1.25-1}{1.25}}$$

$$\frac{T_3}{T_4} = \left(\frac{4}{1}\right)^{\frac{0.25}{1.25}}$$

$$T_4 = \frac{303}{(4)^{\frac{0.25}{1.25}}} = 229.63\,K$$

$$\boxed{T_4 = 229.63\,K}$$

$$W_e = \frac{n}{n-1} R(T_3 - T_4)$$

$$= \frac{1.25}{1.25-1} \cdot 0.287(303 - 229.63)$$

$W_e = 105.28$ KJ/kg. 110.

$\therefore W = W_c - W_e = 137.82 - 105.28 = 32.54$ KJ/kg. $= 33.94$

Heat absorbed $= C_p(T_1 - T_4) = 1(288 - 229.63) = 58.37$ KJ/kg.

$$C.O.P = \frac{58.37}{32.54} = 1.793$$ 1.71

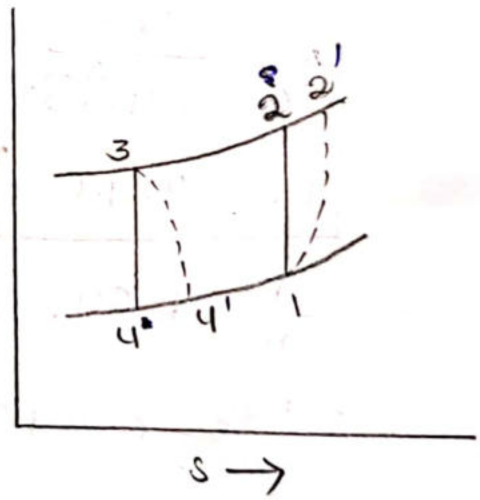
$$\left[C.O.P = \frac{NRE}{Q}\right]$$

$$NRE = \frac{58.37 \times 25}{60} = 24.32\,KW.$$

$$NRE = \frac{24.32}{3.5} = 6.94\,T.R.$$

11

In an air refrigerator working on Bell coleman cycle takes air into the compressor at 1 bar & −5°c it is compressed in the compressor to 5 bar & cooled to 25°c at the same pressure. It is further expanded to 1 bar & discharged to take the cooling load. The isentropic efficiency of the compressor is 85%, the isentropic efficiency of the expander is 90%. Find i) refrigerating capacity if air circulation is 40 kg/min. ii) KW capacity of the motor required to run the compressor. iii) COP of the system. Take $\gamma = 1.4$ $C_p = 1\,KJ/kgK$, $C_v = 0.7\,KJ/kgK$



Sol:

$P_1 = 1\,bar = P_4$

$T_1 = -5°C = 268K$

$P_2 = 5\,bar = P_3$

$T_3 = 25°C = 298K$

$$\eta_c = \frac{T_2^{s} - T_1}{T_2 - T_1} = 85\% \qquad \eta_c = \frac{T_2 - T_1}{T_2' - T_1}$$

$$\eta_e = \frac{T_3 - T_4}{T_3 - T_4^{s}} = 90\% \qquad \eta_e = \frac{T_3 - T_4'}{T_3 - T_4}$$

1−2'−3−4'−1 − Ideal cycle

1−2−3−4−1 − Actual cycle

$$\frac{T_2^{s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \eta_c = \frac{\text{Isentropic increase in temp}}{\text{Actual increase in temp}}$$

$$T_2^{s} = 268\left(\frac{5}{1}\right)^{\frac{0.4}{1.4}} \qquad \eta_e = \frac{\text{Actual decrease in temp}}{\text{Isentropic decrease in temp}}$$

$$\boxed{T_2^{s} = 424.46K} = T_2$$

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

$$T_4' = \frac{T_3}{\left(\frac{5}{1}\right)^{\frac{0.4}{1.4}}} = \frac{298}{(5)^{\frac{0.4}{1.4}}}$$

$$\boxed{T_4' = 188.15\,K} = T_4$$

W.K.T

$$0.85 = \frac{424.46 - 268}{T_2' - 268} \qquad \eta_c = \frac{T_2 - T_1}{T_2' - T_1}$$

$$\boxed{T_2' = 452.07\,K}$$

$$\cancel{0.85 = \frac{T_2 - 268}{424.46 - 268}}$$

$$\Rightarrow \boxed{\cancel{T_2 = 400.99\,K}}$$

$$\cancel{0.9 = \frac{298 - 188.15}{298 - T_4}}$$

$$\boxed{\cancel{T_4 = 175.94\,K}}$$

$$\eta_e = \frac{T_3 - T_4'}{T_3 - T_4} \Rightarrow 0.9 = \frac{298 - T_4'}{298 - 188.15}$$

$$T_4' = 199.18\ K$$

i)

$$\text{Heat absorbed} = C_p(T_1 - T_4')$$

$$= 1(268 - \cancel{175.94}\ 199.18) = 68.82\ KJ/kg$$

$$\cancel{= 92.06\ KJ/kg.}$$

$$NRE = \frac{\cancel{92.06}\ 68.82 \times 40}{60} = \cancel{61.37\,KW} = 45.88\ KW$$

$$NRE = 13.1\ TR$$

$$\cancel{NRE = 17.53\,TR.}$$

ii)

$$\cancel{W_{net} = \frac{n}{n-1} \cdot R\left((T_2 - T_1) - (T_3 - T_4)\right)}$$

$$\cancel{W = mC_p\left[(T_2 - T_3) - (T_1 - T_4)\right]}$$

$$\cancel{= 1\left[(400.99 - 298) - (268 - 175.94)\right]}$$

$$\cancel{= (62.01\ KJ/kg)}\ 10.931\ KJ/kg.$$

$$P = \frac{\cancel{62.01}\ 10.931 \times 40}{60} = \cancel{(41.34\,KW)}\ 7.287\ KW$$

(ii)

$$W_{net} = W_C - W_E$$

$$= m\,C_p(T_2' - T_1) - m\,C_p(T_3 - T_4')$$

$$= m\,C_p\left[(T_2' - T_1) - (T_3 - T_4')\right]$$

$$= \frac{40}{60} \times 1\left[(452 - 268) - (298 - 199.18)\right]$$

$$W_{net} = 56.78\ kW$$

(iii)

$$COP = \frac{NRE}{W_{net}}$$

$$= \frac{45.88}{56.78} = 0.807$$

A refrigerating machine using air as a working fluid and working on closed Bell coleman cycle operates under the following conditions.

Refrigerator temperature = 150K
cooler temperature = 300K

The air temperature at the entry of refrigerator is 40K less than the refrigerator temp. The pressure in the refrigerator = 1 bar. calculate i> Refrigerating effect ii> Net work done iii> COP of the system iv> cooler pressure. Assume expansion & compression are isentropic.

Sol: Given $T_1 = 150K$

$T_3 = 300K$

$T_4 = \cancel{40K}\ 150 - 40 = 110K$

$P_1 = 1 bar = P_4$

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

$$\frac{P_3}{1} = \left(\frac{300}{110}\right)^{\frac{1}{\frac{0.4}{1.4}}} \qquad \Rightarrow T_2 = 150 \times \left(\frac{33.5}{1}\right)^{\frac{0.4}{1.4}}$$

$\times\left[\ \boxed{\cancel{P_3 = 1.33 bar}}\right.$ $\boxed{P_3 = 33.5 bar = P_2}$ $\boxed{T_2 = 409.08K.}$

$\left.\cancel{P_2 = 1.33 bar}\ \right]\times$

i>

Heat absorbed $= c_p(T_1 - T_4)$

$= \cancel{2.4}(150 - 110) \times 1$

$= 40 KJ/kg.$

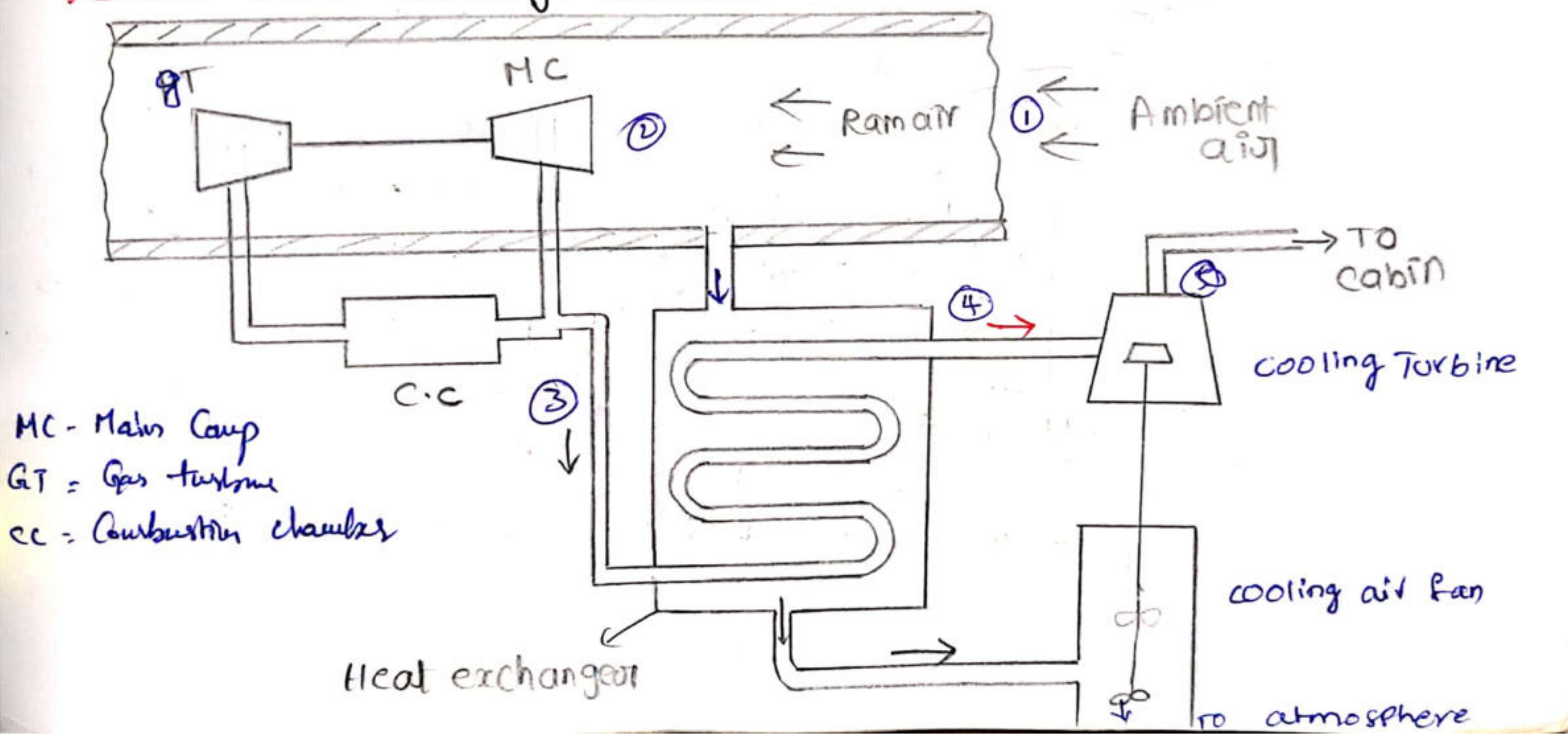
ii) $W = C_p(T_2 - T_3) - C_p(T_1 - T_4)$

$= 1[(409.08 - 300) - (150 - 110)]$

$= 69.08\ KJ/kg.$

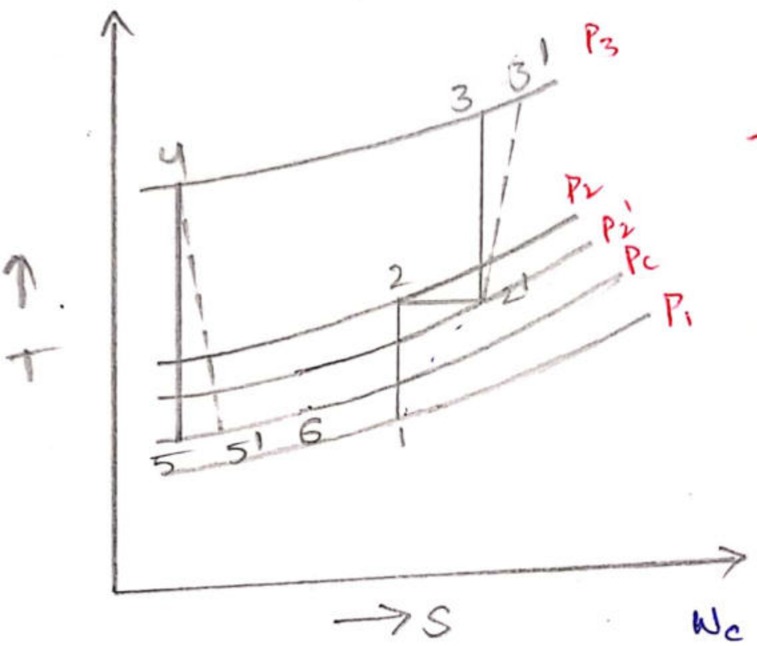
iii) $COP = \frac{H.A}{W} = \frac{40}{69.08} = 0.57$

Simple air cooling system:



T-s diagram:

at stagnation point KE is zero & it is converted to internal energy and it is added to static enthalpy.



$T_2 = T_2'$: Stagnation temp
P_1 = Ambient air press
P_c = Cabin pressure
P_2' = Actual ram press
P_2 = Ideal ram press.

$W_c = m_a C_p (T_3' - T_2')$

$Q_{R\,HE} = m_a C_p (T_3' - T_4)$

$W_{CT} = m_a C_p (T_4 - T_5')$

(Q) $NRE = m_a C_p (T_6 - T_5')$

$KE = \frac{V^2}{2000} = h_2 - h_1 = C_p (T_2 - T_1)$

$T_2 = T_1 + \frac{V^2}{2000 C_p}$

mass of air $m_a = \frac{210Q}{C_p (T_6 - T_5')}$ kg/min

power $P = \frac{m C_p (T_3' - T_2')}{60}$

$COP = \frac{210Q}{m C_p (T_3' - T_2')}$

18102114

The cockpit of a jet plane flying at a speed of 1200km/hr is to be cooled by a simple air cooling system. The cockpit is to be maintained at 25°C & pressure in the cockpit is 1 bar. The ambient air pressure & temp. are 0.85 bar & 30°C. The other data available is as follows.

cockpit cooling load = 10 T.R.

Main compressor pressure ratio = 4

Ram efficiency = 90%

Temperature of air leaving the H.E & entering the cooling turbine = 60°C

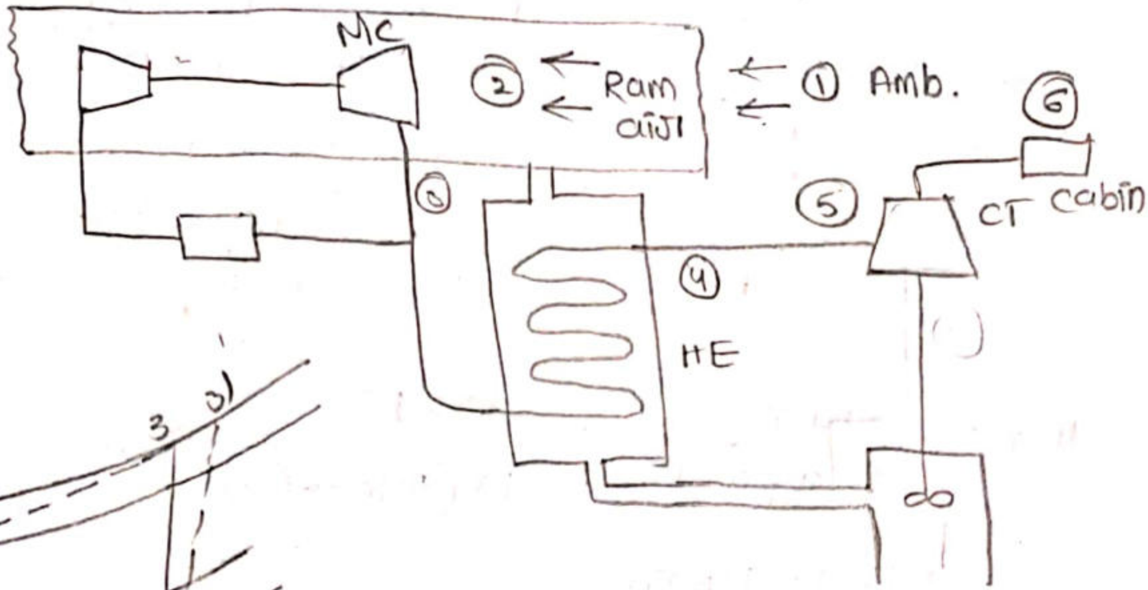
pressure drop in the heat exchanger - 0.5 bar

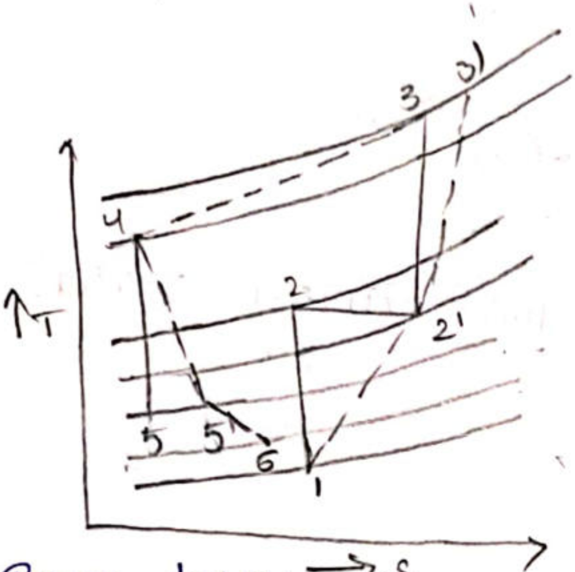
pressure loss b/w cooler turbine & cockpit = 0.2 bar

Assuming isentropic efficiency of main compressor and cooler turbine as 80%, Find the quantity of air passed through the cooling turbine and COP.

Take $\gamma = 1.4$, $C_p = 1$ KJ/kgK.

Sol:





Given data:

$V = 1200\,km/hr = 333.33\,m/s.$

$T_6 = 25°C = 298K$

$P_6 = 1\,bar$

$P_1 = 0.85\,bar$

$T_1 = 30°C = 303K$

$Q = 10\,T.R.$

$\frac{P_3}{P_2'} = 4$

$\eta_R = 0.9$

$T_4 = 60°C = 333K$

$P_4 = P_3 - 0.5$

$P_6 = P_5 - 0.2$

$P_5 = 1.2\,bar = P_5'$

$\eta_c = 0.8$

$\gamma = 1.4$

$C_p = 1\,KJ/kgK.$

$\eta_{CT} = 0.8$

$$T_2 = T_1 + \frac{V^2}{2000\,C_p}$$

$$= 303 + \frac{333.33^2}{2000 \times 1}$$

$$\boxed{T_2 = 358.55K = T_2'}$$

$$\left(\frac{P_2}{P_1}\right)^{\frac{0.4}{1.4}} = \frac{T_2}{T_1}$$

$$P_2 = \left(\frac{358.55}{303}\right)^{\frac{1.4}{0.4}} \times 0.85$$

$$\boxed{P_2 = 1.53\,bar}$$

$$\eta_R = \frac{P_2' - P_1}{P_2 - P_1} = 0.9 = \frac{P_2' - 0.85}{1.53 - 0.85}$$

$$\Rightarrow \boxed{P_2' = 1.462\,bar}$$

$P_3 = 4 \times P_2' = 4 \times 1.462$

$\boxed{P_3 = 5.848 \text{ bar}}$

$\frac{T_3}{T_2'} = \left(\frac{P_3}{P_2'}\right)^{\frac{\gamma-1}{\gamma}}$

$T_3 = 358.55\left(\frac{5.848}{1.462}\right)^{\frac{0.4}{1.4}}$

$\boxed{T_3 = 532.80 K}$

$P_4 = P_3 - 0.5$

$\boxed{P_4 = 5.348 \text{ bar}}$

$\eta_c = 0.8 = \frac{T_3 - T_2'}{T_3' - T_2'}$

$0.8 = \frac{532.8 - 358.55}{T_3' - 358.55}$

$\boxed{T_3' = 576.36 K}$

$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$

$T_5 = 333\left[\frac{1.2}{5.348}\right]^{\frac{0.4}{1.4}}$

$\boxed{T_5 = 217.27 K}$

$\cancel{\frac{T_5}{T_6} = P_5}$

$\eta_{CT} = 0.8 = \frac{T_4 - T_5'}{T_4 - T_5}$

$0.8 = \frac{333 - T_5'}{333 - 217.27}$

$\boxed{T_5' = 240.41 K}$

i) mass of air

$\dot{m}_a = \frac{210\,Q}{c_p(T_6 - T_5')}$

$= \frac{210 \times 10}{1 \times (298 - 240.41)}$

$\boxed{m_a = 36.46 \text{ kg/min}}$

$P = \frac{m c_p (T_3' - T_2')}{60}$

$= \frac{36.46 \times 1 \times (576.36 - 358.55)}{60}$

$\boxed{P = 132.35 \text{ KW}}$

ii) $COP = \frac{Q}{W}$

$= \frac{10 \times 3.5}{132.35}$

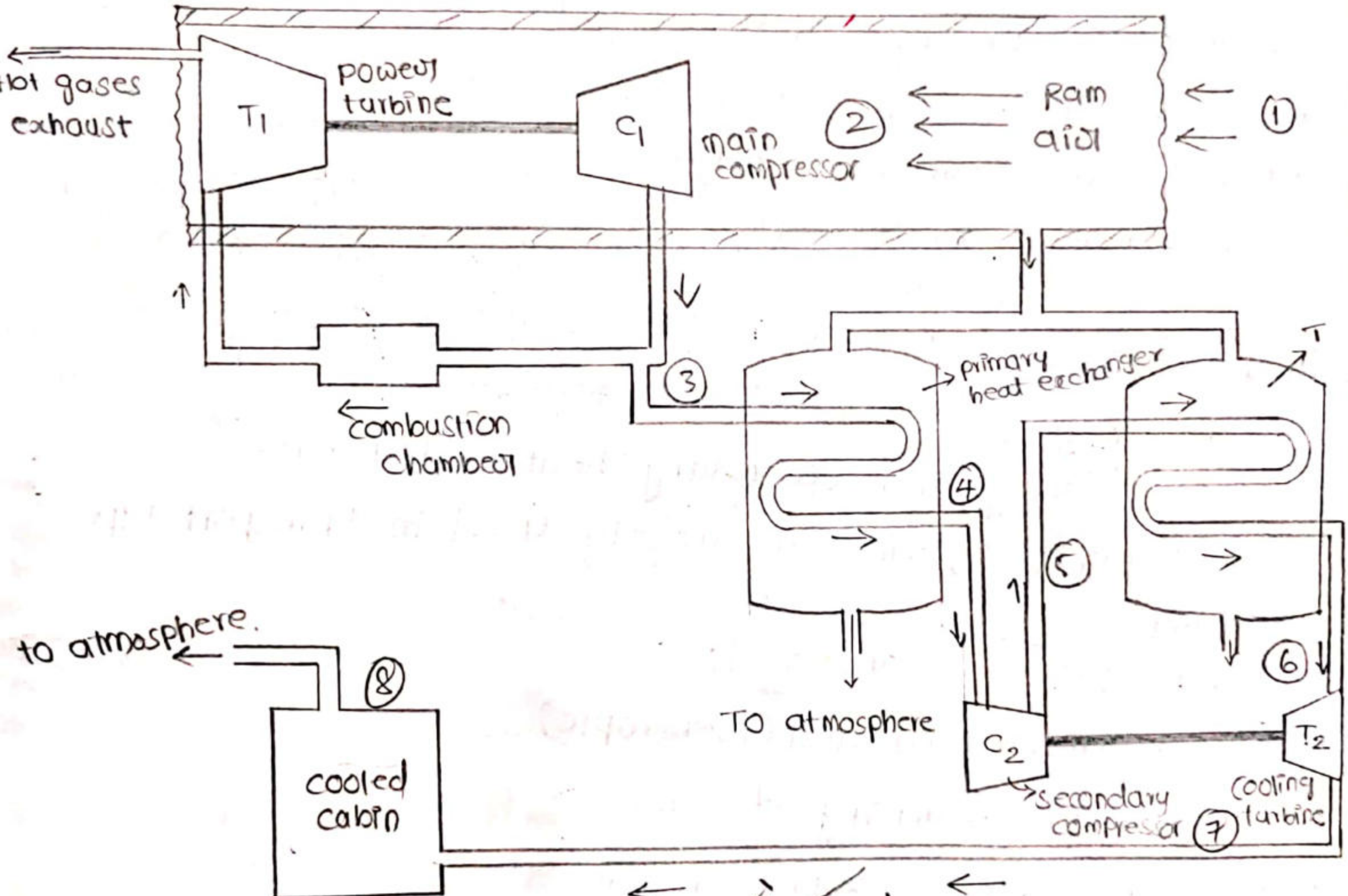
$\boxed{COP = 0.264}$

$P = 70$ KW

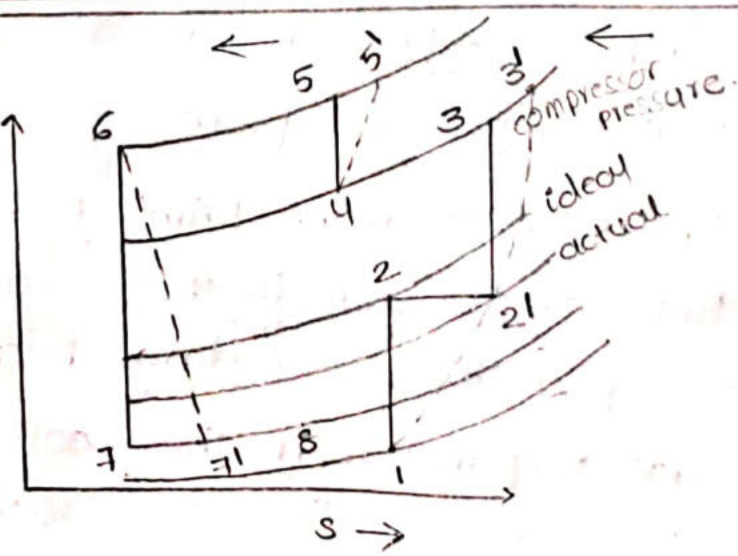
$COP = 0.5$

Scanned with CamScanner

Boot strap air cooling system:



T-S diagram:



Working:

The term boot strap as used in air cycle refrigeration system indicates a system in which the pressure of the working fluid is raised to a higher level than the main compressor by a separate compressor before expanding in the cooling turbine. The power required for the secondary compressor is taken from the cooling turbine.

This system consists of two heat exchangers (primary & secondary) and a cooling turbine. High pressure air from the main compressor is first cooled in the primary heat exchanger. Then the air is further compressed to a higher pressure by secondary compressor. It is then cooled in the secondary heat exchanger and further cooling action is completed by expanding the air through the cooling turbine. Ram air is used as heat sink in primary and secondary heat exchangers.

These type of systems are mostly used in transport type aircrafts.

Different processes involved:

1-2 → Ramming of air (isentropic)

1-2' → Actual ramming of air

2'-3 → Isentropic compression process

2'-3' → Actual compression process

3'-4 → Actual cooling in the primary heat exchanger

4-5' → Actual compression in the secondary compressor

5'-6 → cooling in the secondary heat exchanger

6-7' → Actual expansion in the cooling turbine

7'-8 → Heat extraction from the cabin.

* Assuming cooling load in the cabin is Q tons then the quantity of air circulated in the refrigerated system is

mass of air, $m_a = \dfrac{210\ Q}{C_p(T_8 - T_7')}$ kg/min

$= \dfrac{3.5\ Q}{C_p(T_8 - T_7')}$ kg/sec

where T_8 is the temperature of air leaving the cabin

* Power required for the system $P = m_a C_p (T_3' - T_2')$

Note:
Work required for the secondary compressor is delivered by the cooling turbine so no additional work is required to run the secondary compressor.

* cop of the system $= \dfrac{3.5 Q}{W}$

1) The Boot Strap cooling system of 10 T.R. capacity is used in an aeroplane. The ambient temperature and pressure of air are 20°C & 0.85 bar. The pressure of air increases from 0.85 bar to 1 bar due to ramming action of the air. The pressure of air discharged from the main compressor is 3 bar. The discharge pressure of air from the auxiliary compressor is 4 bar. The isentropic efficiency of each compressor is 80% and that of turbine efficiency is 85%. 50% of enthalpy of air discharged from the main compressor is removed in the first heat exchanger and 30% of enthalpy of air discharged from the auxiliary compressor is reduced in the secondary heat exchanger using rammed air. Assuming ramming action is isentropic the required

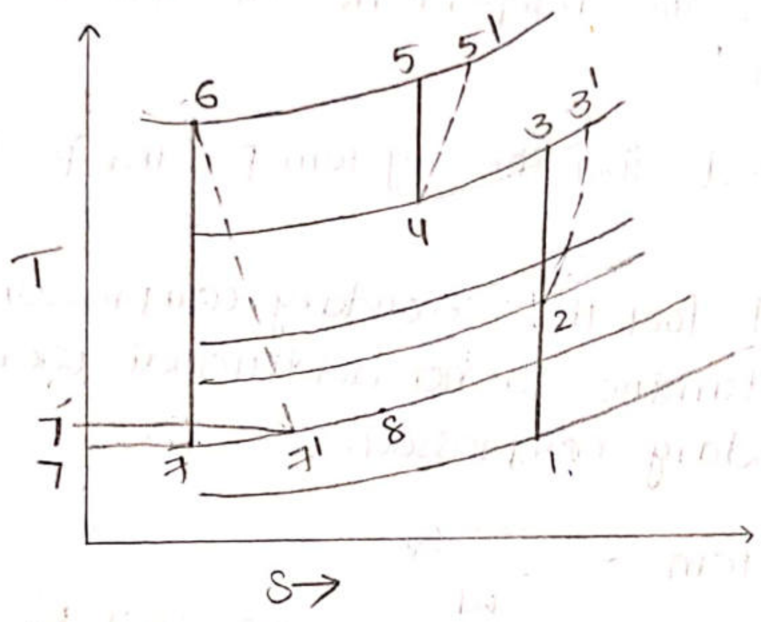
cabin pressure of 0.9 bar & temp. of air leaving the cabin not more than 20°c. Find.

i) power required to operate the system

ii) COP of the system and draw temperature, entropy diagram.

Take $\gamma = 1.4$ & $C_p = 1$ KJ/kgK.

Sol:
08/02/14



Given data:

$Q = 10$ TR

$T_1 = 20°C = 293K$

$P_1 = 0.85$ bar

$P_2 = 1$ bar

$P_3 = P_3' = P_4 = 3$ bar

$P_5 = P_5' = P_6 = 4$ bar

$\eta_{c_1} = \eta_{c_2} = 0.8$

$\eta_T = 0.85$

$P_7 = P_7' = P_8 = 0.9$ bar

$T_8 = 20°C = 293K$

$\gamma = 1.4$

$C_p = 1$

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

$$T_2 = 293\left(\frac{1}{0.85}\right)^{\frac{0.4}{1.4}}$$

$$\boxed{T_2 = 306.92\,K}$$

$$\eta_{c_1} = \frac{T_3 - T_2}{T_3' - T_2}$$

$$0.8 = \frac{420 - T_2}{T_3' - T_2}$$

$$0.8 = \frac{420 - 306.92}{T_3' - 306.92}$$

$$\boxed{T_3' = 448.27\,K.} = 175.27°C$$

$$\cancel{T_4 = 0.5 \times T_3' = 0.5 \times 448.27}$$

$$\boxed{T_4 = 224.135\,K}$$

$$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_5 = 224.135\left(\frac{4}{3}\right)^{\frac{0.4}{1.4}}$$

$$\boxed{T_5 = 243.3\,K}$$

$$\eta_{c_2} = \frac{T_5 - T_4}{T_5' - T_4}$$

$$0.8 = \frac{243.3 - 224.13}{T_5' - 224.13}$$

$$\boxed{T_5' = 248.09\,K}$$

$$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_3 = 306.92\left(\frac{3}{1}\right)^{\frac{0.4}{1.4}}$$

$$T_3 = 420\,K$$
$$= 147°C$$

$$T_4 = 0.5 \times T_3' = 0.5 \times 175.27$$
$$= 87.63°C$$

$$\boxed{T_4 = 360.6\,K.}$$

$$\left(\frac{T_5}{T_4}\right) = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_5 = 360.6\left(\frac{4}{3}\right)^{\frac{0.4}{1.4}}$$

$$\boxed{T_5 = 391.49\,K}$$

$$\eta_{c_2} = \frac{T_5 - T_4}{T_5' - T_4}$$

$$0.8 = \frac{391.49 - 360.6}{T_5' - 360.6}$$

$$\boxed{T_5' = 399.21\,K}$$
$$= 126.21°C$$

$$T_6 = 0.7 \times 126.21 = 88.34°C$$

$$\boxed{T_6 = 361.3\,K}$$

$$\frac{T_7}{T_6} = \left(\frac{P_7}{P_6}\right)^{\frac{0.4}{1.4}}$$

$$T_7 = 361.3 \times \left(\frac{0.9}{4}\right)^{\frac{0.4}{1.4}}$$

$$\boxed{T_7 = 235.95K}$$

$$\eta_T = \frac{T_7' - T_6}{T_7 - T_6} = 0.85$$

$$0.85 = \frac{T_7' - 361.3}{235.95 - 361.3}$$

$$\boxed{T_7' = 254.75K.}$$

$$Q = m C_p \Delta T$$

$$10 \times 3.5 \times 10^3 = m \times 1 \times 10^3 \times (T_8 - T_7')$$

$$10 \times 3.5 \times 10^3 = m \times 1 \times 10^3 \times (293 - 254.75)$$

$$\Rightarrow m = 0.915 \text{ kg/sec}$$

$$m = 54.9 \text{ kg/min}$$

$W =$ ~~NRE~~

power $P = m_a C_p (T_3' - T_2')$

$$= \frac{54.9}{60} \times 1 (448.27 - 306.92)$$

$$P = 129.33 \text{ kW}$$

$$COP = \frac{Q}{W} = \frac{10 \times 3.5}{129.33} = 0.27$$

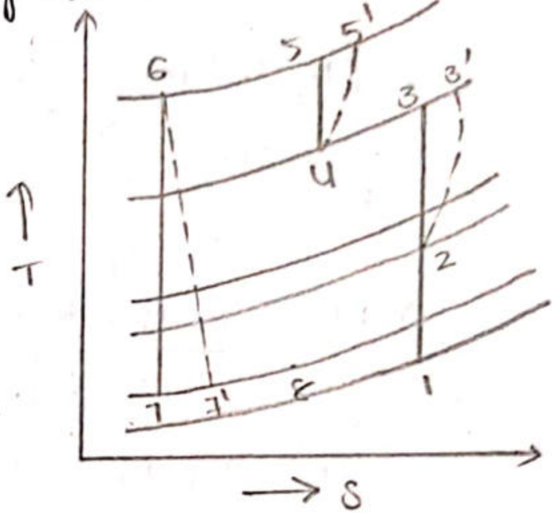
10/02/14

2) A Boot strap cooling system of 20 tons capacity is required for an aeroplane cabin. The temperature and pressure conditions of atmosphere are 20°C & 0.8 bar. The pressure of air is increased from 0.8 bar to 0.96 bar due to ramming action. The pressure of air leaving the main compressor & auxiliary compressor are 3.5 bar & 5.25 bar respectively. The isentropic efficiency of both the compressors is 85% & turbine efficiency is 80%, 60% of total

heat of air leaving the main compressor is removed in the first heat exchanger & 35% of total heat of air leaving the auxiliary compressor is removed from secondary heat exchanger. Assuming ramming as isentropic cabin pressure as 1.03 bar. Find (18)

i> kW power required for loading the cabin

ii> COP (temperature of air leaving the cabin should not exceed 27°C)



Sol: $Q = 20$ tons

$T_1 = 20°C$

~~$P_1 = 0.85 bar$~~

~~$P_2 = 0.85 bar$~~

~~$P_3 = 0.96 bar = P_3' = P_4$~~

$P_1 = 0.8$ bar

$P_2 = 0.96$ bar

$P_3 = 3.5$ bar $= P_3' = P_4$

$P_5 = P_5' = P_6 = 5.25$ bar

$\eta_c = 85\%$

$\eta_T = 80\%$

$P_8 = P_7 = P_7' = 1.03$ bar

$$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1} \Rightarrow T_2 = 20 \times \left[\frac{0.96}{0.8}\right]^{\frac{0.4}{1.4}} = \cancel{20.7°C} = 37°C = 310\ K$$

$$\boxed{T_2 = 37°C}$$

$$\frac{T_3}{T_2} = \left(\frac{P_3}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_3 = 37 \times \left[\frac{3.5}{0.96}\right]^{\frac{0.4}{1.4}}$$

$$\boxed{T_3 = \cancel{29.95°C}}\quad 177°C = 450\ K$$

$$\eta_c = \frac{T_3 - T_2}{T_3' - T_2} = \frac{177 - 37}{T_3' - 37} = 0.85 \Rightarrow T_3' = 202°C$$

$$0.85 = \frac{\cancel{29.95 - 20.7}}{\cancel{T_3' - 20.7}}$$

$$\Rightarrow \boxed{\cancel{T_3' = 31.58°C}}$$

$T_4 = 0.4 \times T_3' = 0.4 \times 31.58 = 12.632°C$

$\boxed{T_4 = 12.632°C}$ $+273 \Rightarrow T_4 =$

$$\frac{T_5}{T_4} = \left(\frac{P_5}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\Rightarrow T_5 = \left(\frac{5.25}{3.5}\right)^{\frac{0.4}{1.4}} \times 12.632$$

$\boxed{T_5 = 14.18°C}$

$$\eta_{c_2} = \frac{T_5 - T_4}{T_5' - T_4}$$

$$0.85 = \frac{14.18 - 12.632}{T_5' - 12.632}$$

$\boxed{T_5' = 14.45°C}$ $+273 =$

$T_6 = 0.65 \times 14.45 = 9.39°C$

$$\eta_T = \frac{T_7' - T_6}{T_7 - T_6} = 0.8$$

$$\frac{T_7}{T_6} = \left(\frac{P_7}{P_6}\right)^{\frac{0.4}{1.4}}$$

$$T_7 = 9.39 \times \left(\frac{1.03}{5.25}\right)^{\frac{0.4}{1.4}}$$

$\boxed{T_7 = 5.86°C}$

$$0.8 = \frac{T_7' - 9.39}{5.96 - 9.39}$$

$\boxed{T_7' = 6.646°C}$

$Q = mC_p \Delta T$

$\cancel{7}\cancel{2}0000 = m \times 1 \times 10^3 (27 - 6.646)$

$m = \cancel{0.98 \text{ kg/sec}}\ 3.439 \text{ kg/sec}$

$m = \cancel{58.9 \text{ kg/min}}\ 206.3 \text{ kg/min}$

Power $P = mcp(T_3' - T_2') = 0.98 \times 1 (31.58 - 20.7)$

$P = \cancel{10.66KW}.\ 2245.06 \text{ KW}$

$$COP = \frac{Q}{W} = \frac{20 \times 3.5}{2245.06} = \cancel{0.03}$$

0.435

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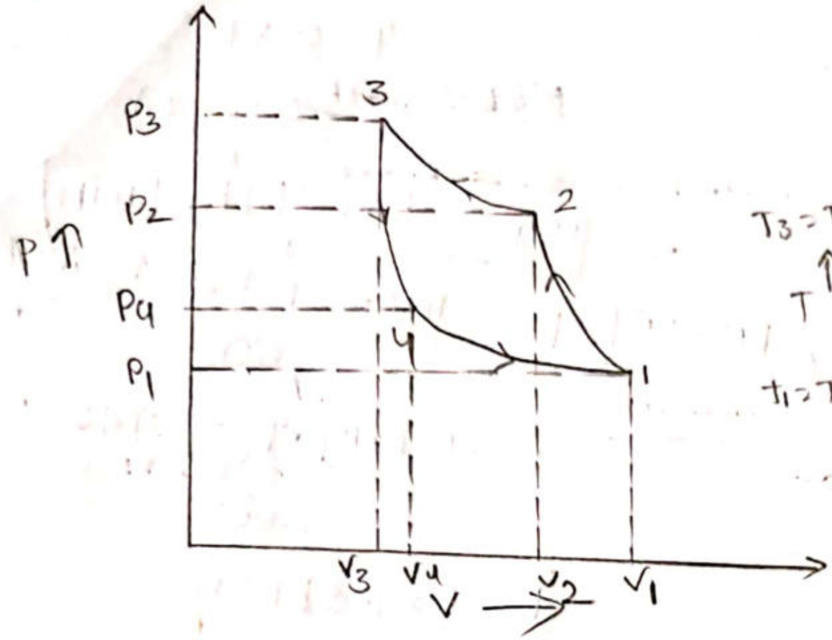
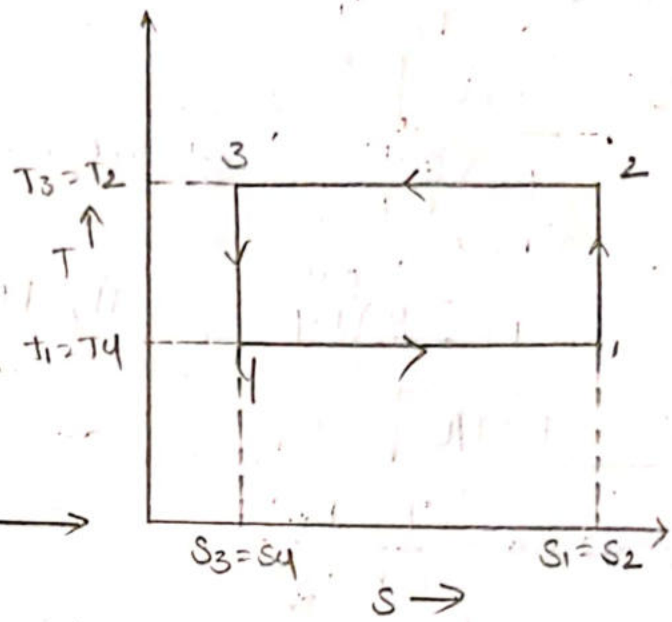
Air refrigerator working on reverse carnot cycle :





1. Isentropic compression
2. Isothermal compression
3. Isentropic expansion
4. Isothermal expansion

$Q_{absorbed} = T_1(S_2 - S_3)$ (or) $T_1(S_1 - S_4)$

$Q_{rejected} = T_2(S_2 - S_3)$

Work done $= Q_{ab} - Q_{rej} = T_1(S_2 - S_3) - T_2(S_2 - S_3)$

$= (S_2 - S_3)(T_1 - T_2)$

$COP = \frac{Q_{ab}}{W.D} = \cancel{(S_2 - S_3)(T_1 - T_2)}$

$= \frac{T_1\cancel{(S_2 - S_3)}}{\cancel{(S_2 - S_3)}(T_1 - T_2)}$

$$\boxed{COP = \frac{T_1}{T_1 - T_2}}$$

03/05/14

Unit-VII
Air conditioning Equipment

Air Cleaning & air filters:

Types of impurities in the air:

1) Dust: (coal, cement, lint) ~~fu~~ traffic industries waste
2) Fumes (These are from chemical reactions such as oxidation)
3) Smoke (Emitted from wood, coal, oil & tobacco)
4) Fogs (small particles suspended in the air)
5) Pollens (from trees, flowers & vegetables)
6) Bacteria (Minute living organisms)

Methods of air cleaning:

1) Air filtration (depends on nature, size & concentration of dust).
2) Air sterilization (killing of bacteria) through ultraviolet rays) propylene glycol can also be used.
3) Air ionization
4) Odour removal (or) supression (activated Carbon is used)

Types of air filters: (5 groups)

1) Dry filters
2) viscous filters
3) Wet filters
4) Electric filters
5) centrifugal dust collectors

Selection of Air filters.

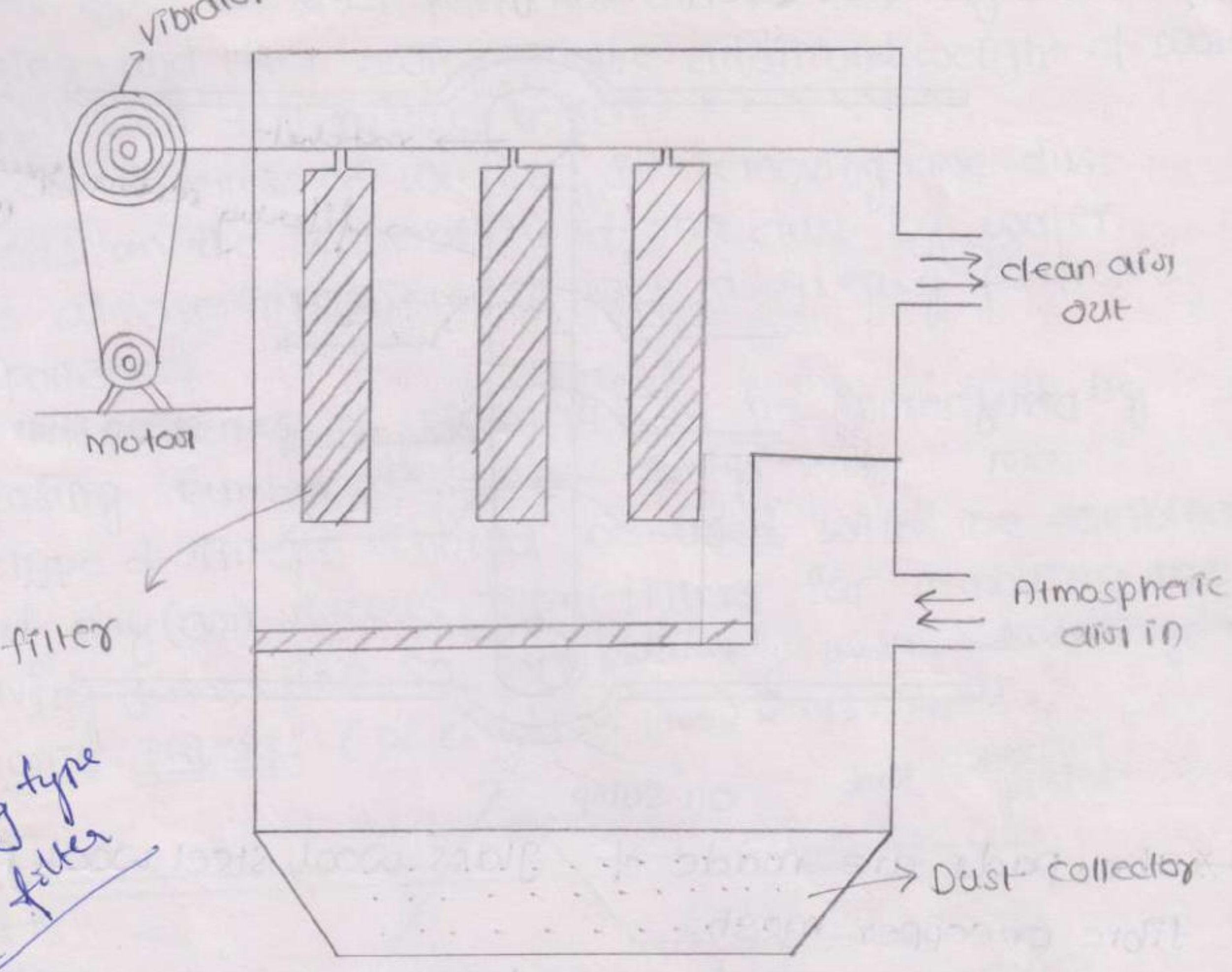
→ The degree of air cleanliness req.
→ Amount of air handled.
→ Type & amount of particulate in air to be filtered.
→ The method of disposal of the collected dust.

1) Dry filters: (Bag type)

Dry filters are of 2 types

1) cleanable filters
2) Throwaway filters.

* Dry filters are made up of cloth, coarse paper (or) wool etc.



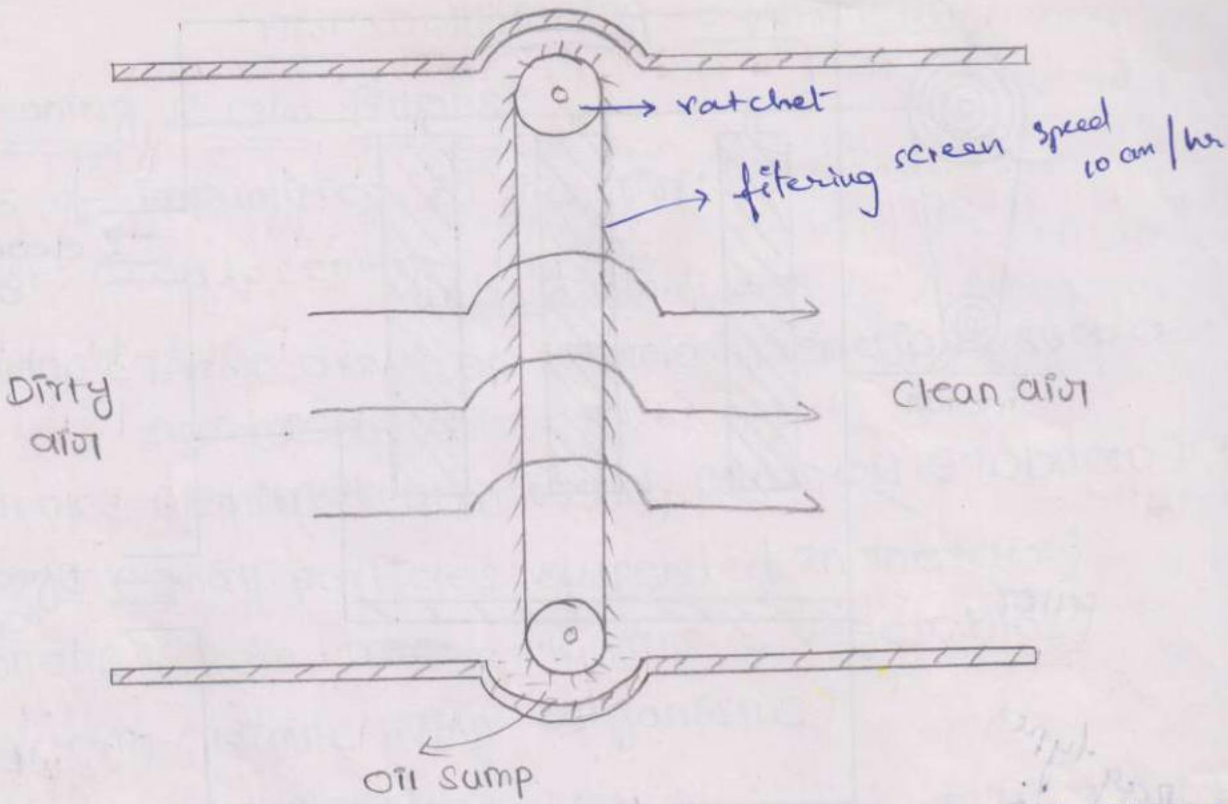
Bag type filter

* These type of filters are used where the velocity ranges are ~~2-3m/s~~ 2–15 m/min
* The cleaning of these filters is by shaking (or) rapping action of the filters.
* Dry filters are capable of collecting 99% of dust as small as 0.5 microns.
* Throw away filters are made of glass wool, plastic fibers, steel wool, animal hair (or) vegetable fibres.
* If the regsin powder is added to the dry filters filtering efficiency increases because it can attract small dia aerosols.
* These type of filters are applied for atmospheric dust, not for the industrial dust (or) process dust.

* 0.3 to 10 μ size dust can be removed.

→ Can use where dust concentration of 2-5 gms to $100m^3$

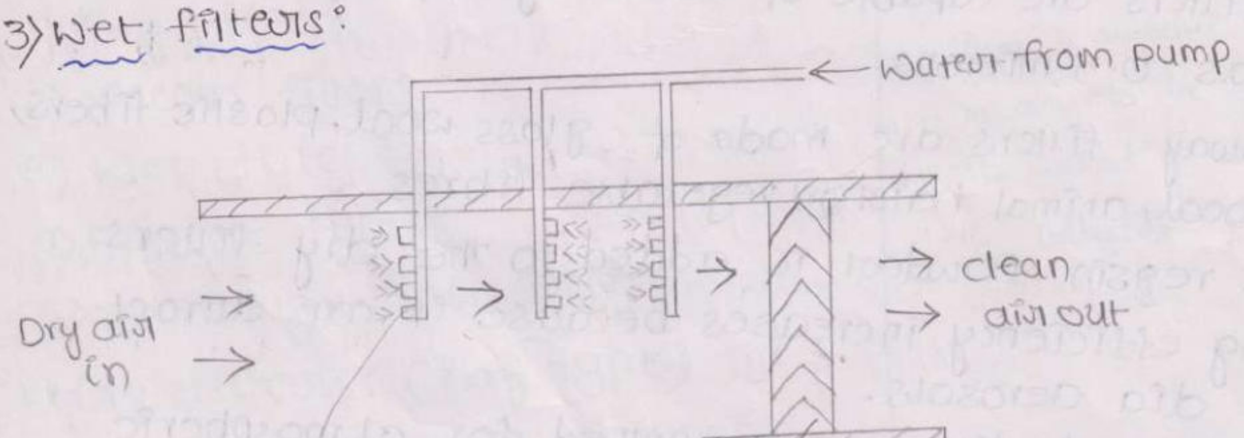
2> viscous filters: (self acting filters)



* The pads are made of glass wool, steel wool, plastic fibre or copper mesh.
* These pads are impregnated with viscosine which is oily substance. [viscosine properties :- constant viscosity, germicidal action & it should not evaporate in air]
* These filters can be washed in gasoline & they can be reused.

09/05/14

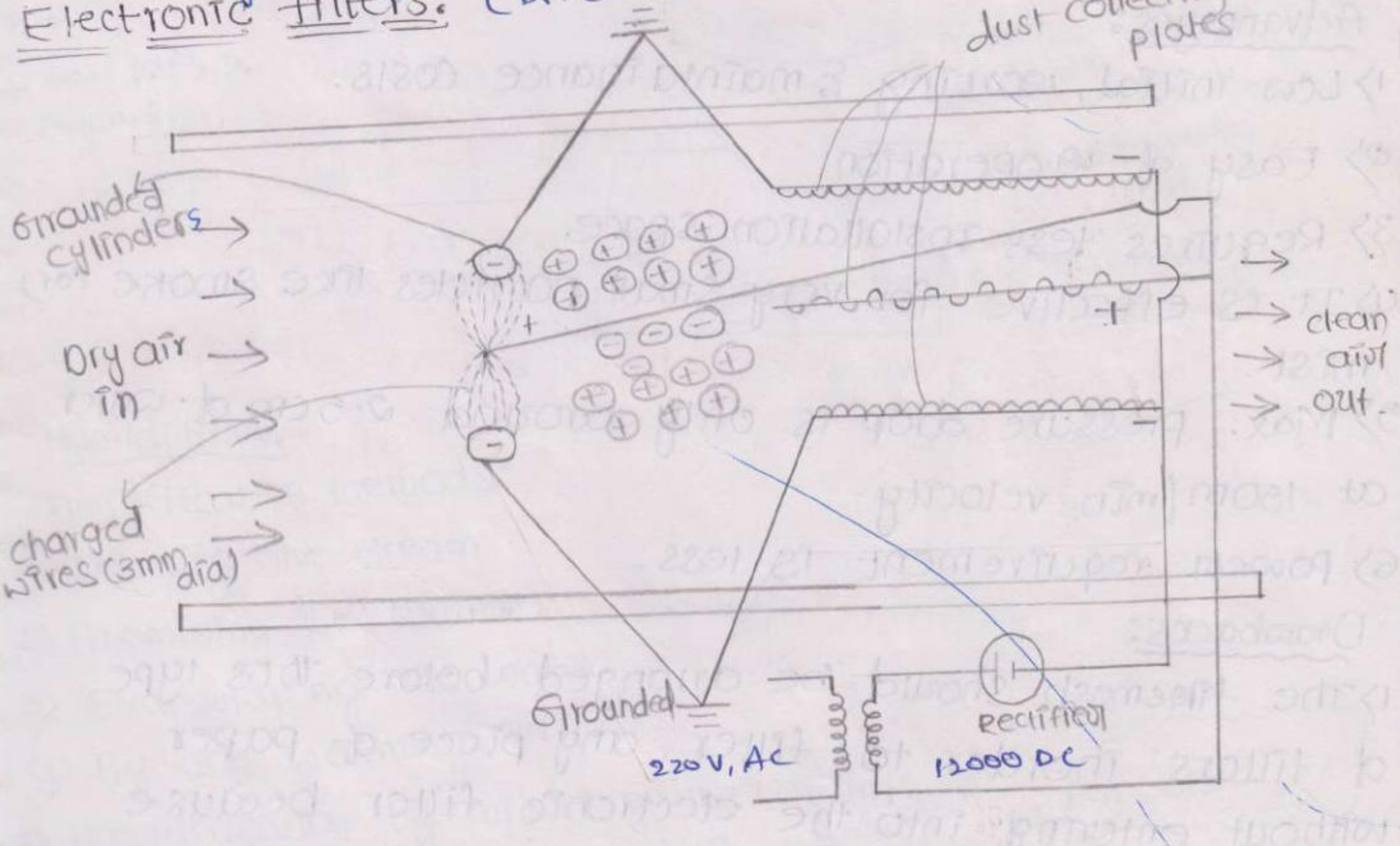
3> Wet filters:



Water spray type air washer

* In this type the dust particles are wetted by the water spraying and then owing to the additional weight of water the particles fall to the bottom.
* The effectiveness of washer in removing the dust depends on the wettability of the dust by water.
* It is almost impossible to wet a greasy particles like pollens.
* The effectiveness of washer can be increased by increasing number of ~~spaces~~. sprays
* This type of filters should be used with the combination of dry (or) viscous type filters for removing the dissolving gases like SO_2. (*) extensively used in industrial application

Electronic filters: (Dr. Cottell in 1906) & 1937 applied.



* Air is passed b/w oppositely charged conductors and it becomes ionized as the voltage between the conductors is sufficiently large (8000 V - 15000 V)
* As the air is passed both negative & positive ions are formed (20% -ve & 80% +ve)
* It is further passed to the collecting unit which

consists of set of metal plates ~~PI~~ spaced 15-20mm apart.

* Alternate plates are positively charged and earthed to attract negatively & positively charged ~~ions in the~~ dust particles.
* The voltage applied to the plates is approximately half of the voltage applied to the conductors.
* When the charged dust particles are passed b/w the plates, the electrostatic film exerts the force on a charged particles & drives them towards grounded plates.
* Grounded plates are cleaned periodically with hot water spray.

Advantages:

1) Low initial, working & maintainance costs.
2) Easy of operation.
3) Requires less installation space.
4) It is effective for very small particles like smoke (or) mist.
5) Max. pressure drop is only around 2.5cm of water at 150m/min velocity.
6) Power requirement is less.

Drawbacks:

1) The ~~fine~~ mesh should be arranged before this type of filters inorder to filter any piece of paper without entering into the electronic filter because there is a chance of catching fire in the filter by the sparks generated.
2) A ~~free~~ pre filter of other type is necessary before ~~control~~ electronic filter to reduce load on this filter.
3) The efficiency of the filter decreases as the quantity of air per unit time increases.

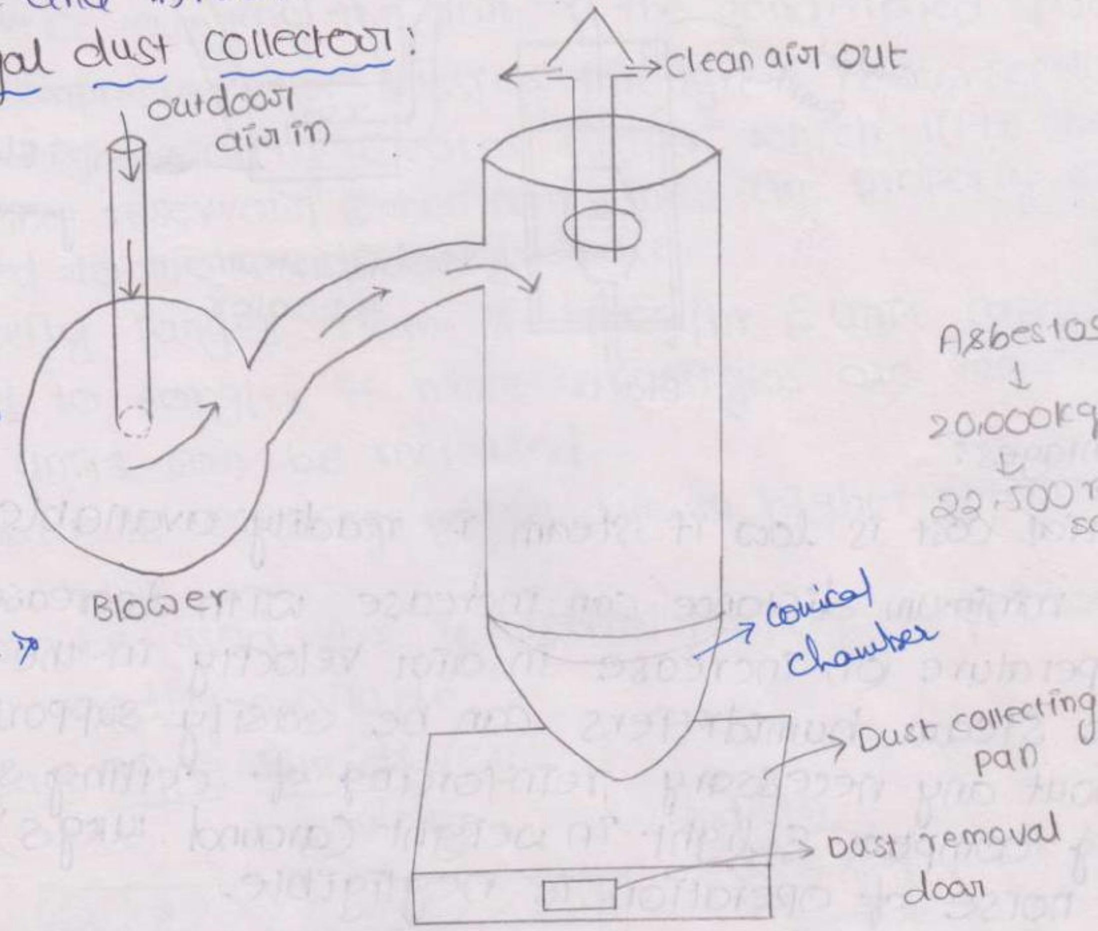
4) It forms small percentage of ozone when the air is passed through these type of filters. The formation of ozone is objectionable due to its irritating effect to the nose and throat.

Centrifugal dust collector:

10/05/14

Adv
→ large particles can be removed easily.

disadv
→ more power is required.



Asbestos
↓
20,000 kg
↓
22,500 rs/annum saving

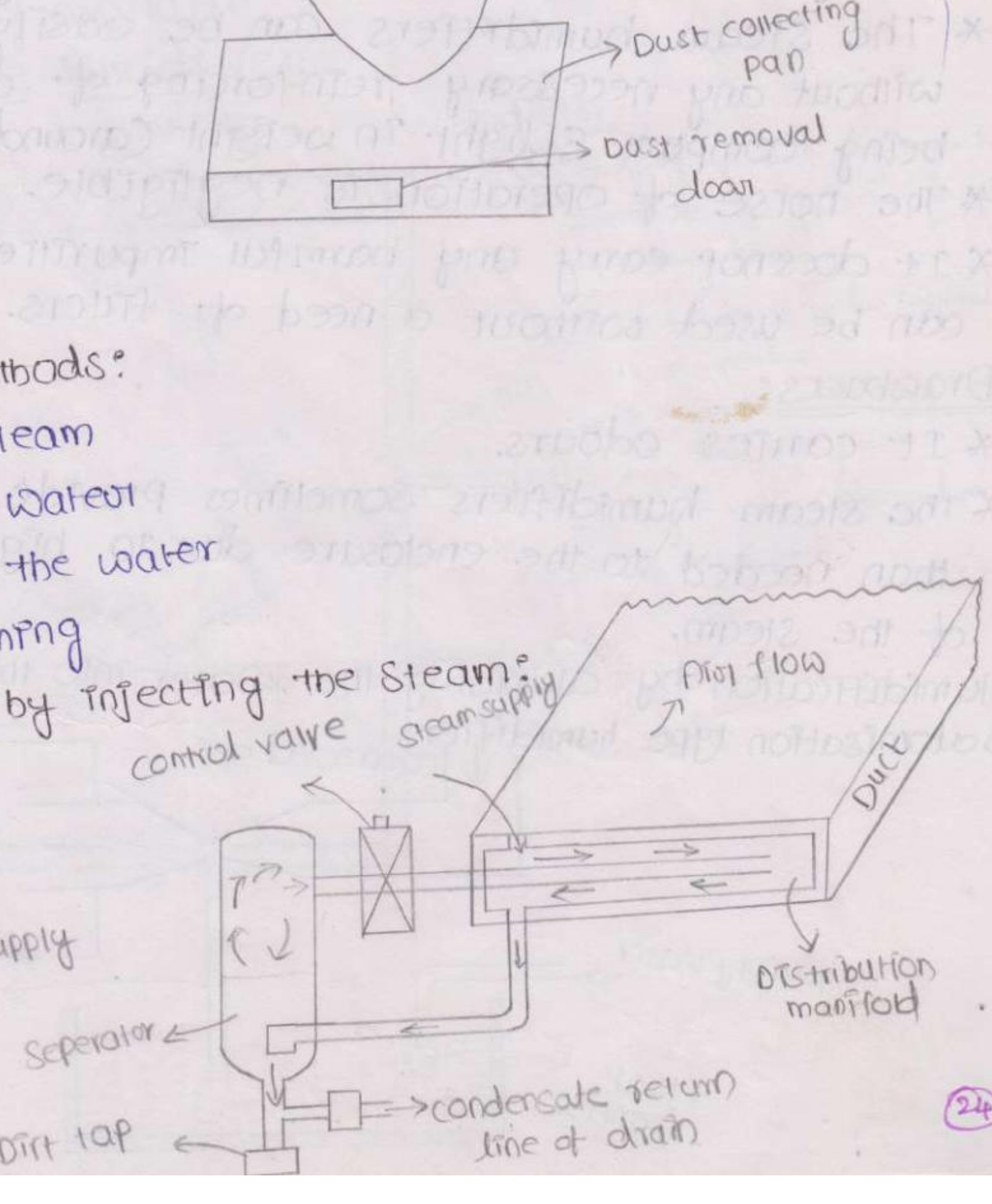
Humidifiers:

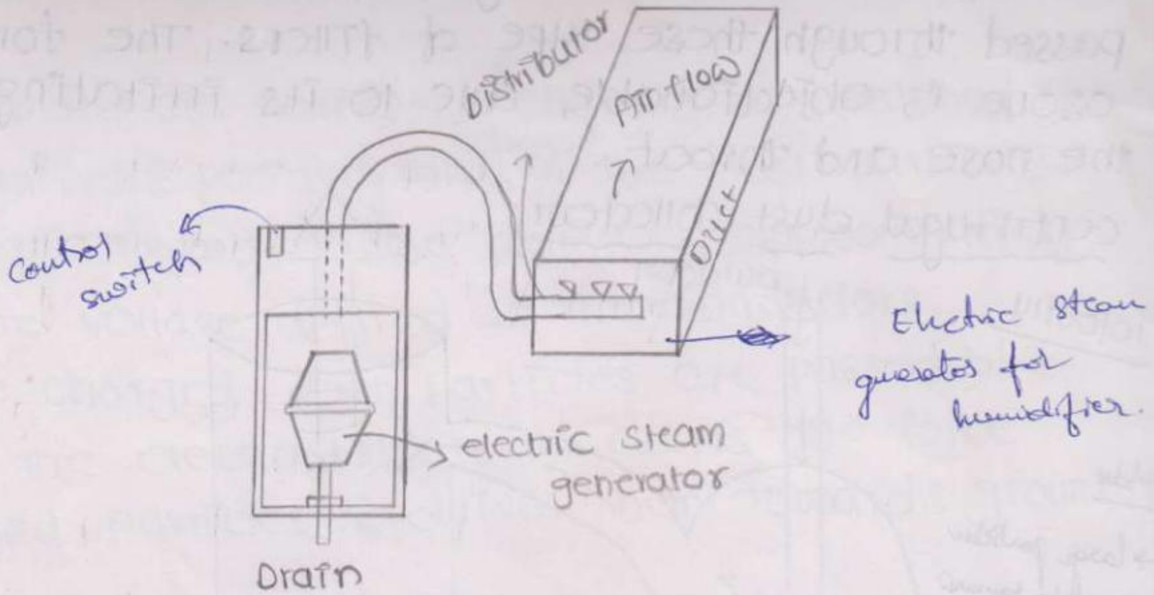
Humidification methods:

1) Injecting the steam
2) Atomising the water
3) Evaporating the water
4) By air washing

1) Humidification by injecting the steam:

Strainer & pressure reducer should be there before steam supply





Advantages:

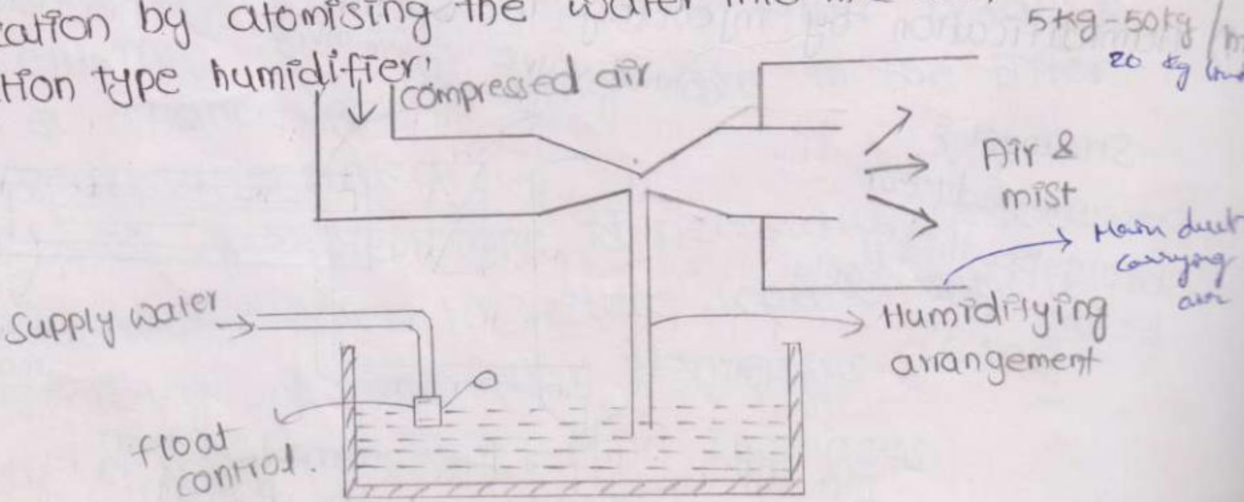
* Initial cost is low if steam is readily available.
* The minimum distance can increase with decrease in air temperature (or) increase in air velocity in the duct
* The steam humidifiers can be easily supported without any necessary reinforcing of ceiling spaces being compact & light in weight (around 14kgs)
* The noise of operation is negligible.
* It does not carry any harmful impurities and therefore can be used without a need of filters.

Drawbacks:

* It carries odours.
* The steam humidifiers sometimes provide more heat than needed to the enclosure due to high temperature of the steam.

2) Humidification by atomising the water into the air:

i) atomisation type humidifier: 5kg-50kg/hr 20 kg limited



1) An effective humidification can be achieved by using compressed air to draw water by aspiration from the supply tank & to blow it in the form of fine mist into the duct carrying the air to the conditioned space.

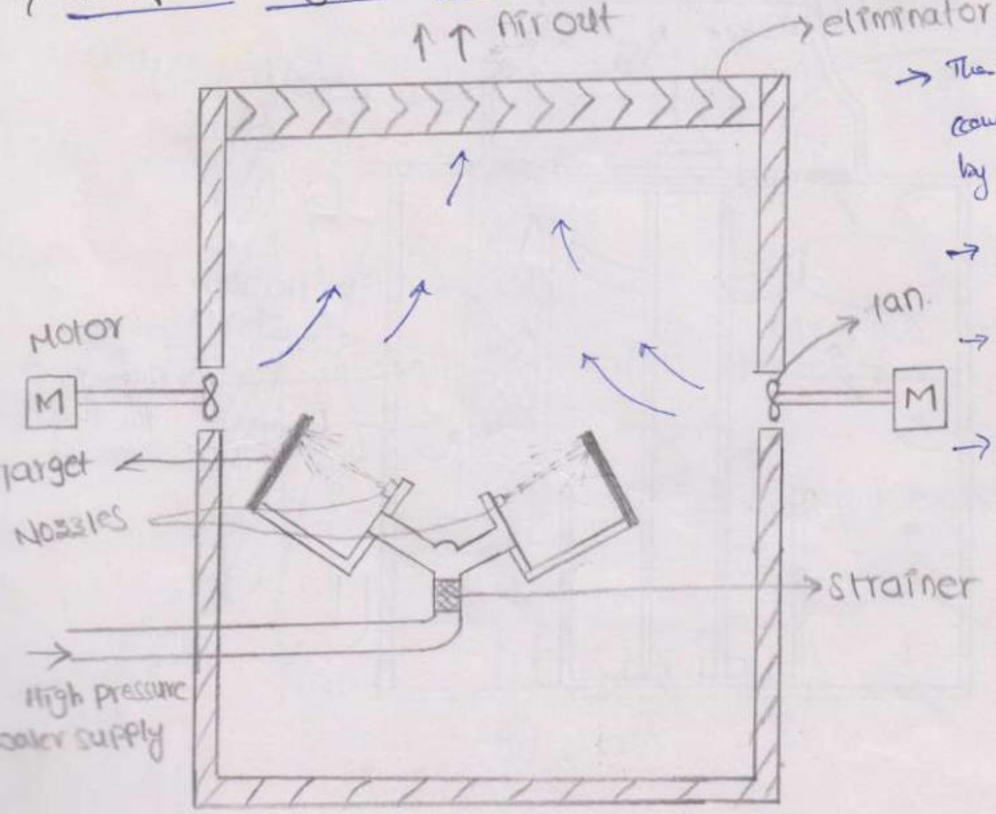
2) The compressed air passing through a hollow section of pipe at a high velocity creates suction which lifts the water from the reservoir & then air & mist are properly mixed & supplied to the conditioned space.

3) capacity ranges from 5-50kg/hr. & unit capacity is limited to 20kg/hr if more capacities are required then no. of. units can be increased.

4) Drawback is creating noise due to higher velocities of air.

5) It can be used for industrial purpose where compressed air is readily available.

ii) <u>Impact</u> <u>type</u> <u>humidifier</u>:



→ The spray formed by impact cannot be so fine as formed by atomization.

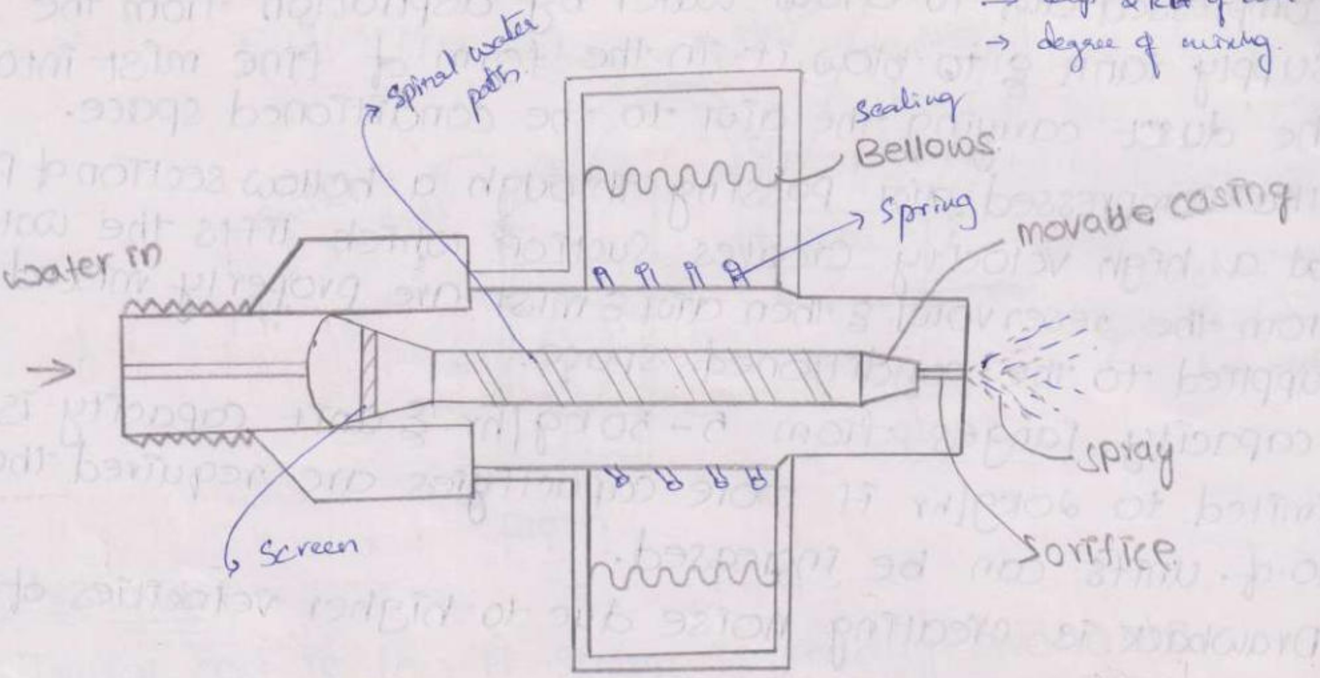
→ 10 to 20% of water supplied is evaporated.

→ It depends on velocity of water jet, temp & RH of air.

→ Target matl Carborundum (no rising voice)

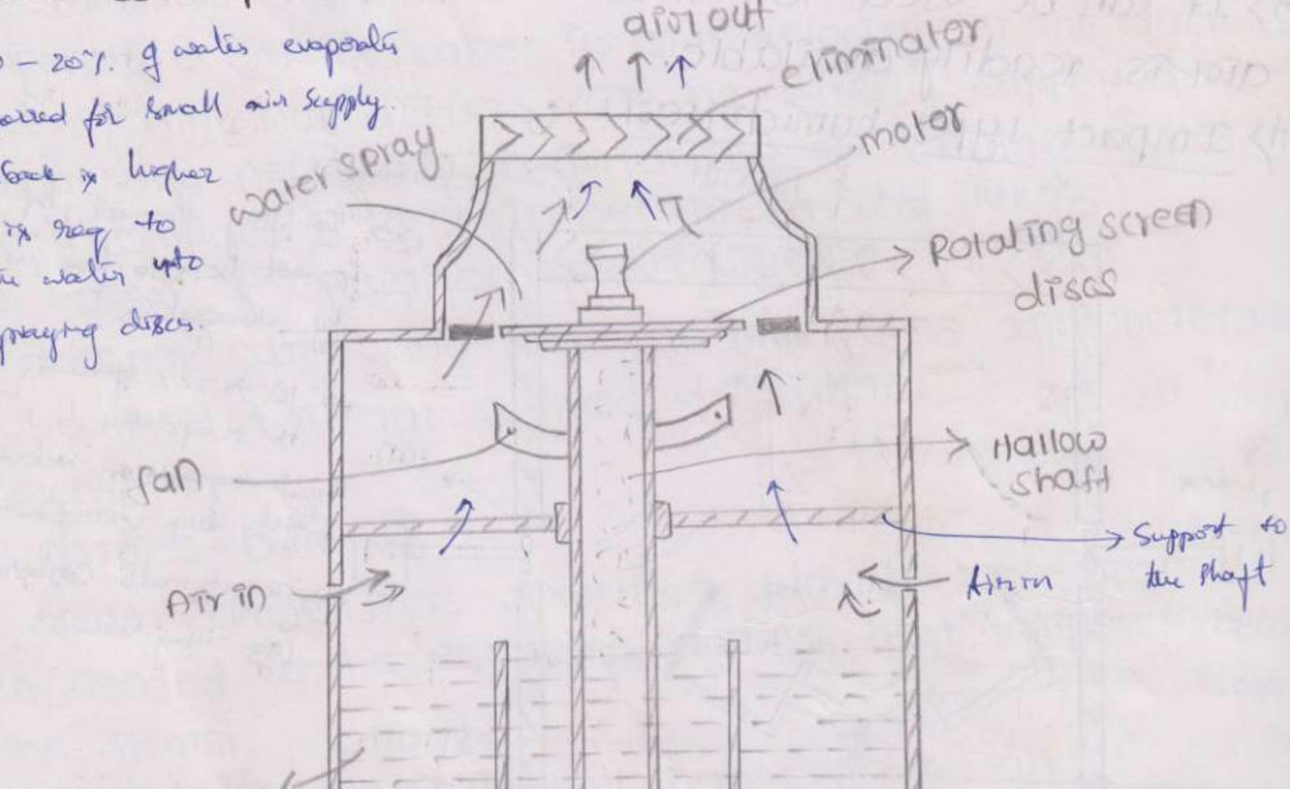
iii) Hydraulic seperation humidifier:

% of water evaporated depend on
→ velocity of jet
→ temp & RH of air
→ degree of mixing.



iv) Mechanical seperation humidifier:

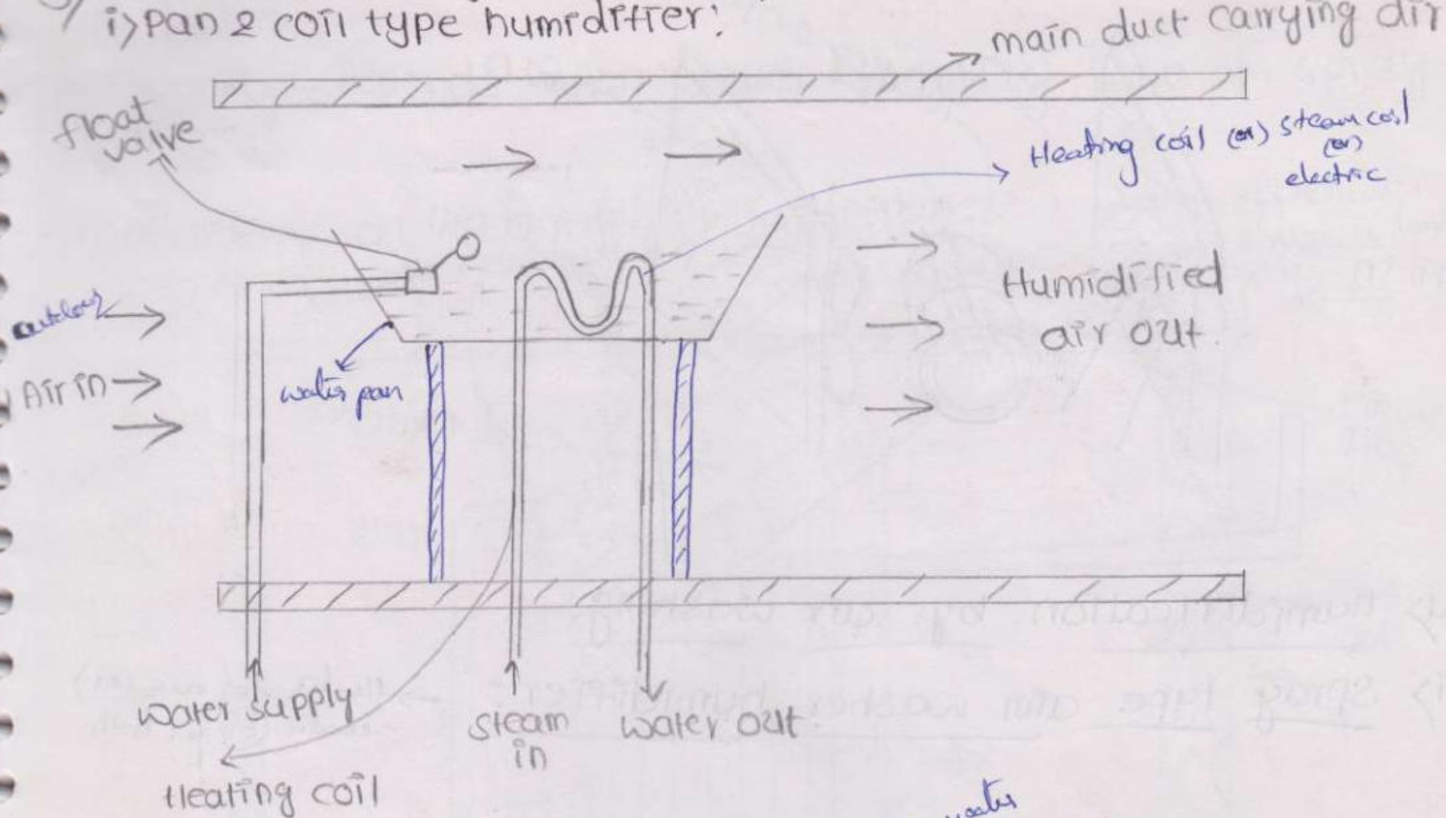
→ 10 – 20% of water evaporate
→ preferred for small air supply.
→ Drawback is higher speed is req to lift the water upto the spraying discs.



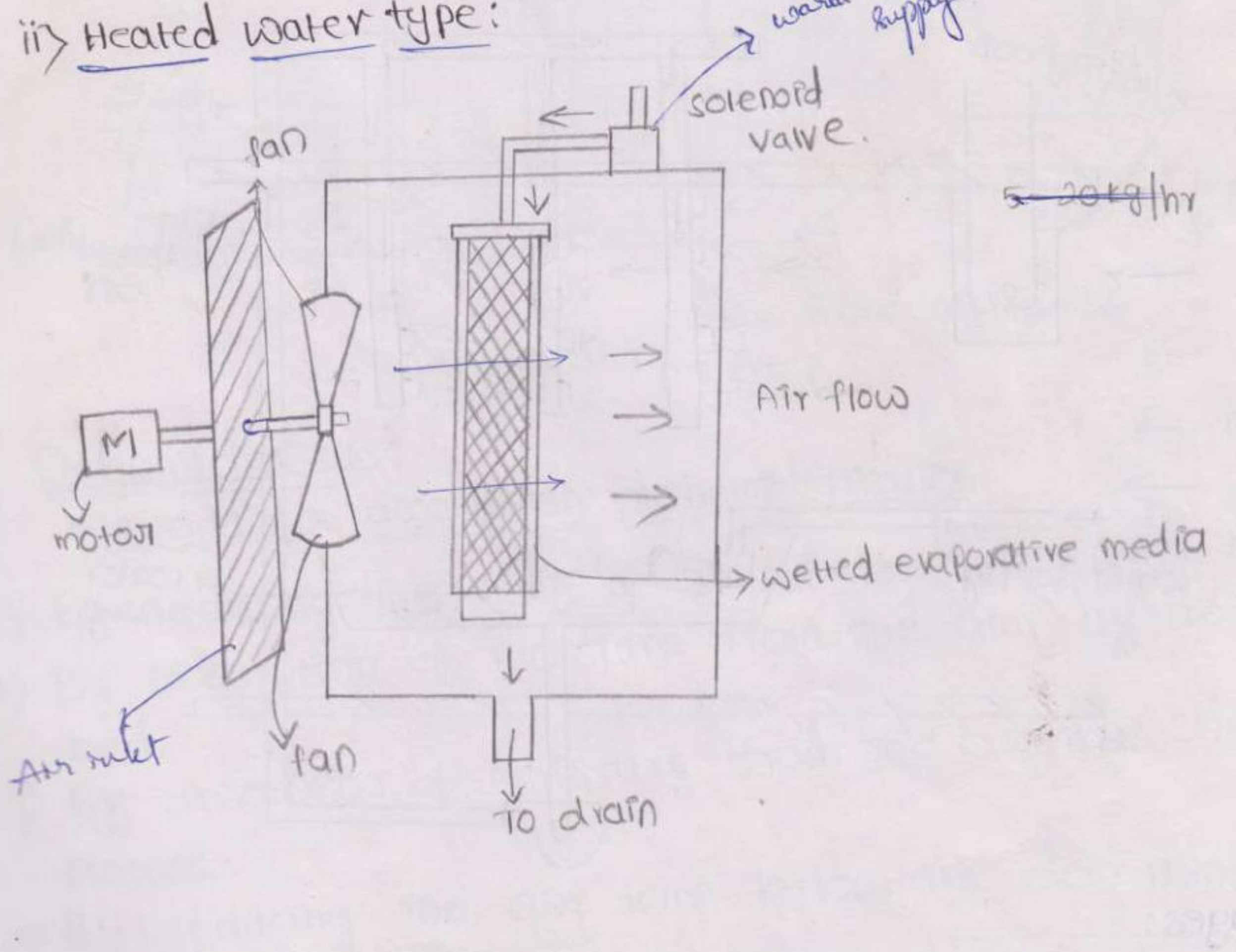
11/05/14

5> Humidification by evaporating the water into air:

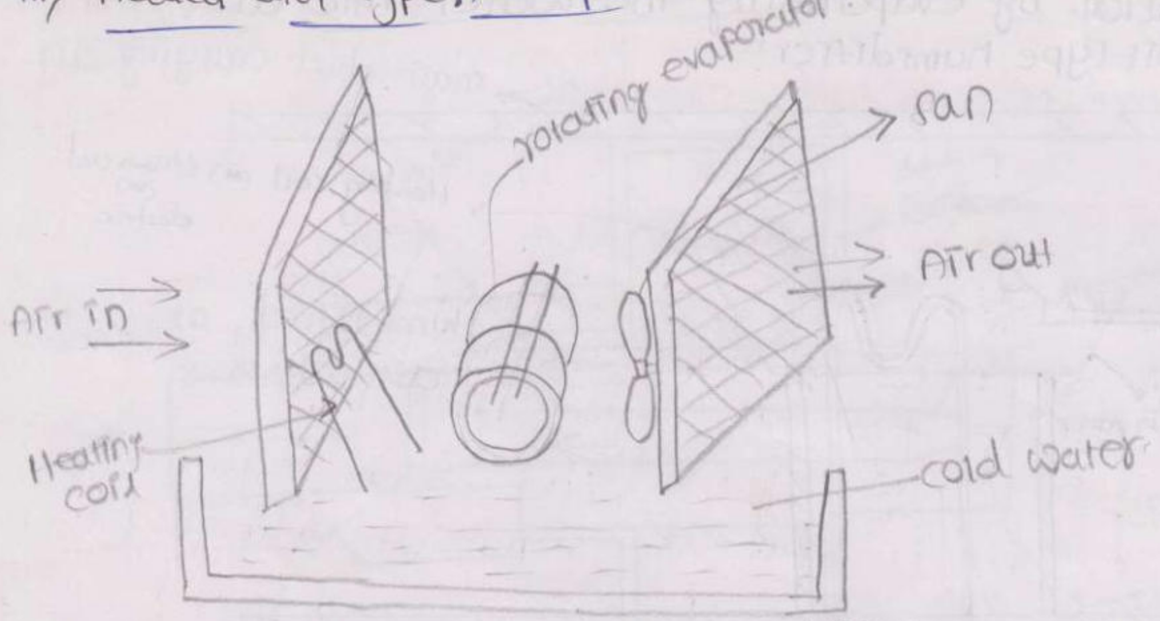
i> Pan & coil type humidifier:



ii> Heated water type:



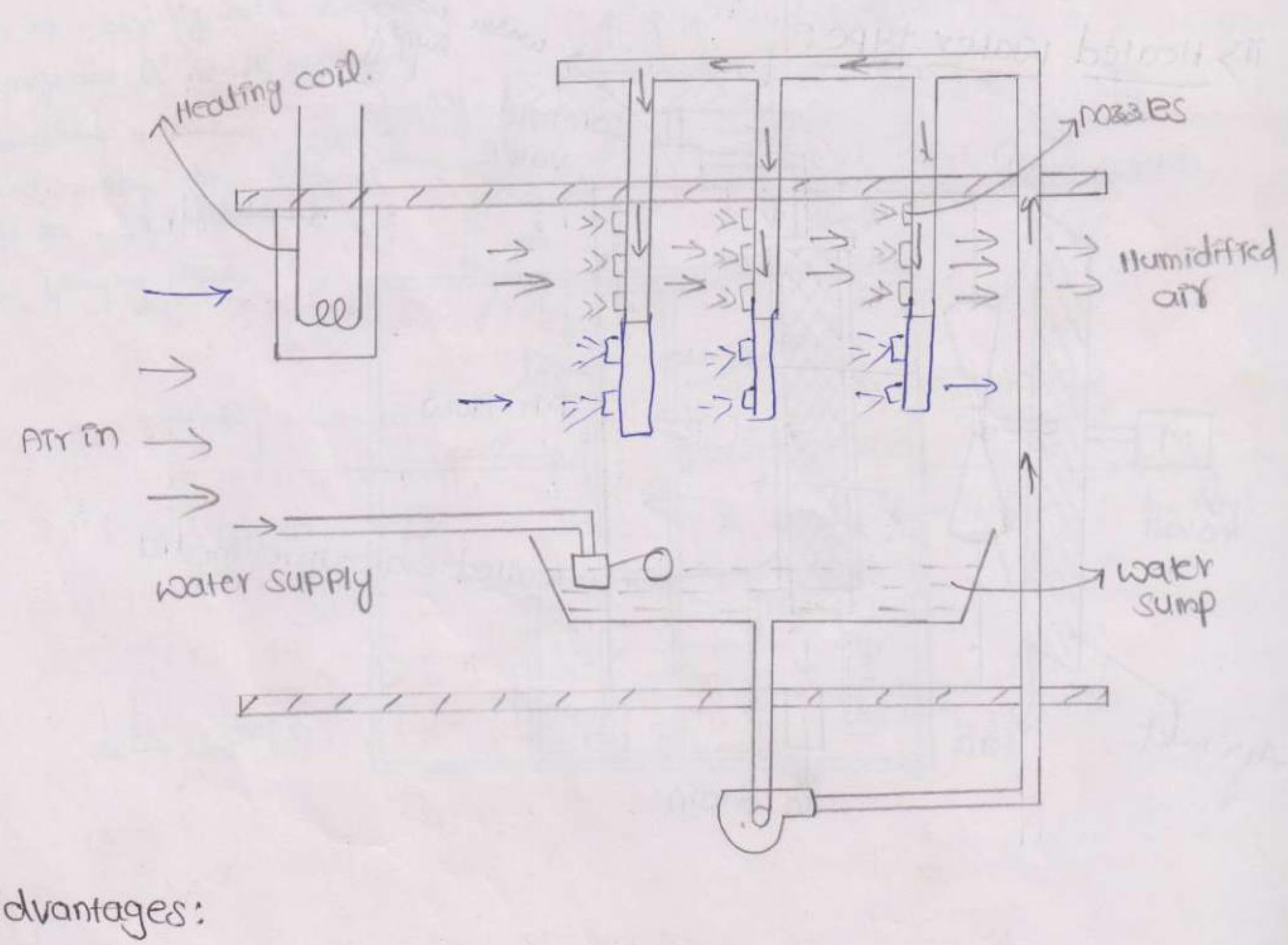
iii> Heated air type Humidifier



5-20 kg/hr

4> Humidification by air washing:

i> Spray type air washer humidifier: → Heats for air (or) water (or) for both.

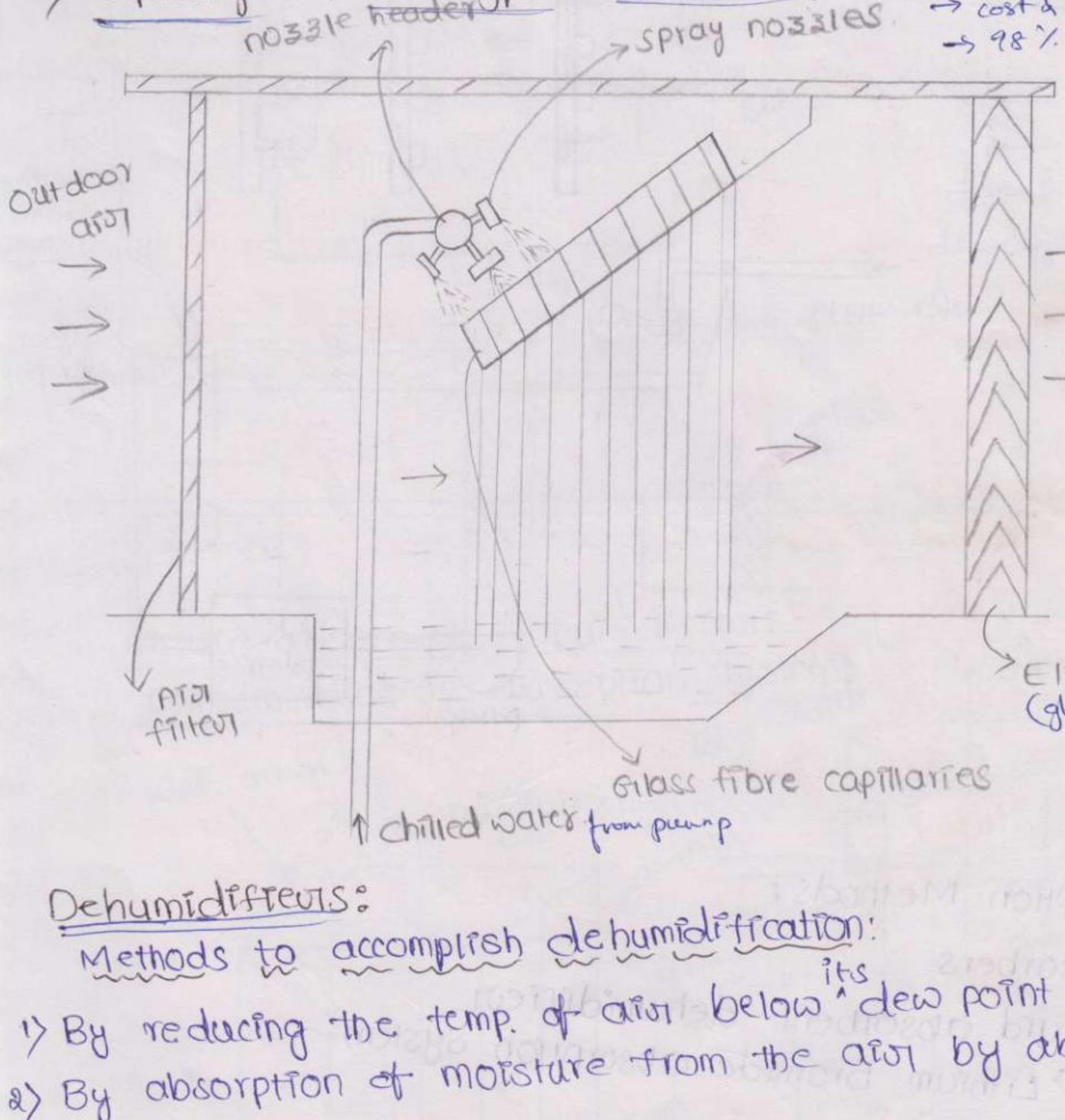


Advantages:

* Initial cost is less.
* Easy operation & maintainance.
* We can easily regulate the temperatures & relative humidity.

* poisonous gases will be filtered which are soluble in water.
* capacity depends upon water quantity & no. of spray nozzles.

ii) Capillary or cell type humidifier: → very efficient → cost & size is high → 98% η.



Dehumidifiers:

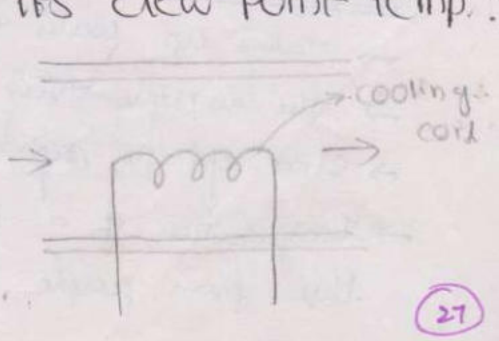
Methods to accomplish dehumidification:

1) By reducing the temp. of air below its dew point temp.
2) By absorption of moisture from the air by absorption bed.
3) By absorption of moisture from the air by chemical process.

1) By reducing the air temp. below its dew point temp.: (or) by refrigeration.

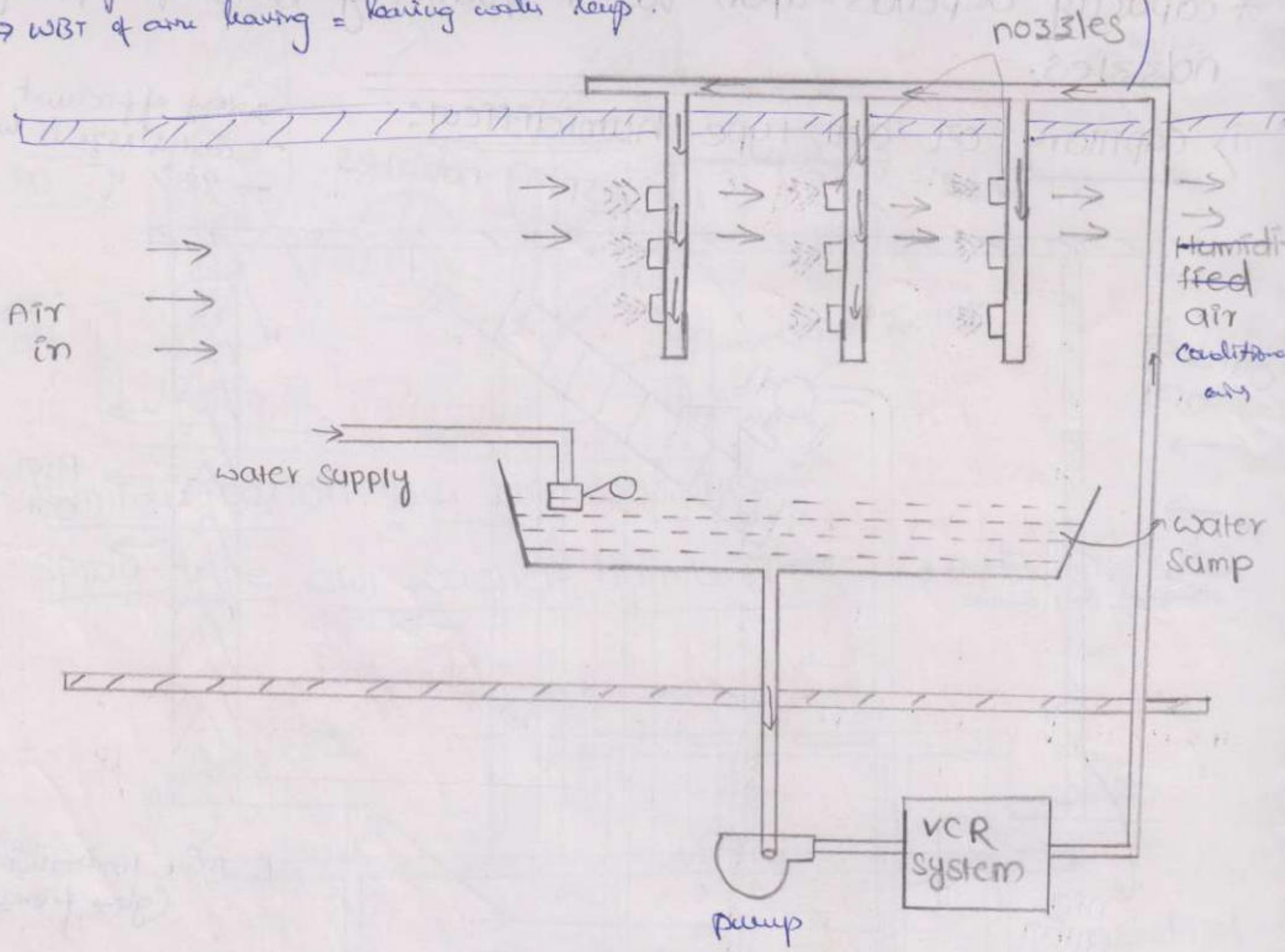
i) By using cooling coils:

(or) By passing water which is having DPT less than air.



ii) spray type dehumidifier:

→ temp of water is below DPT of air.
→ WBT of air leaving = leaving water temp.

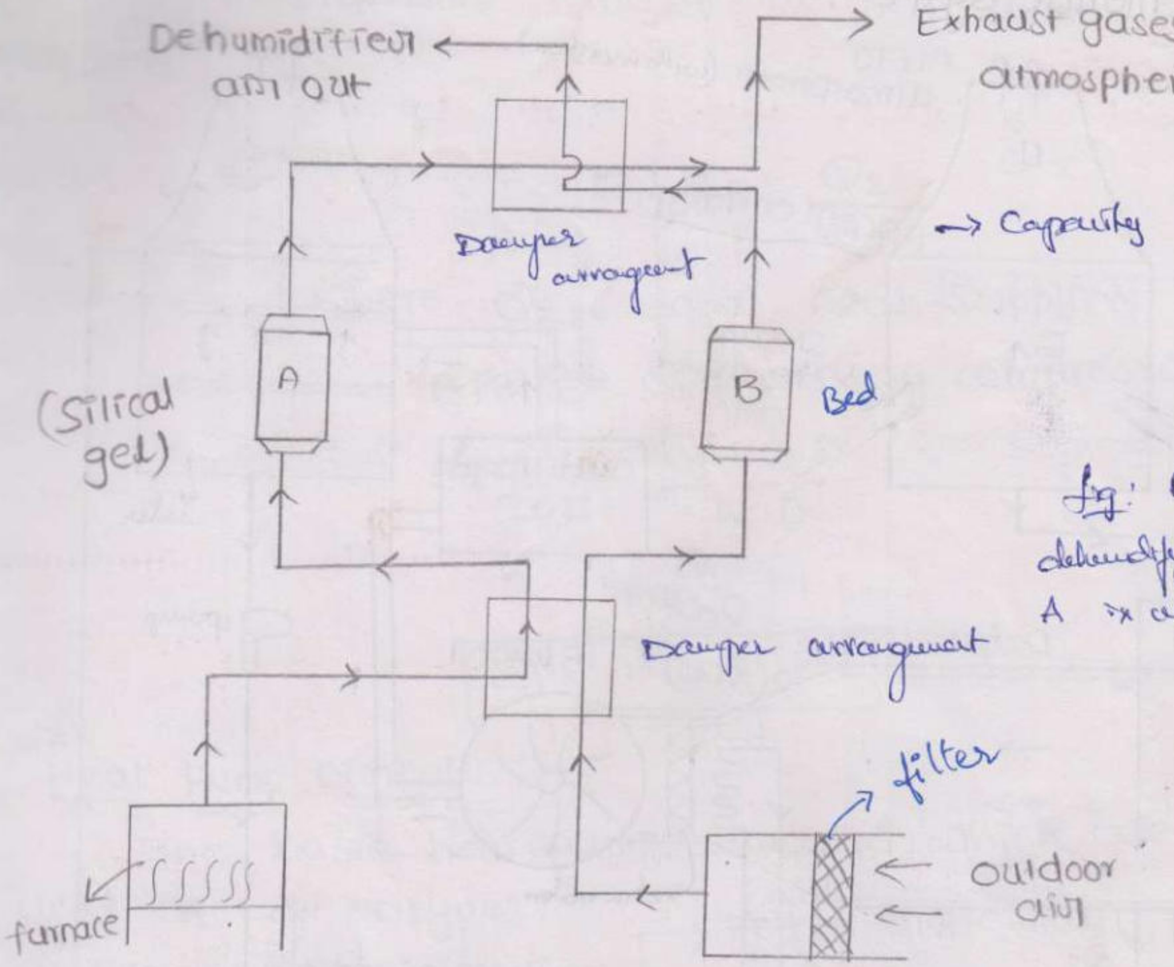


2) Adsorption Methods:

→ Adsorbers
→ Liquid absorbent dehumidifier
  → Lithium Bromide absorption system
  → calcium chloride system.

→ Adsorbers:

→ Takes up water vapour & hold without any chemical reactions.
→ The moisture thus adsorbed may be removed by reactivation process.
→ Silica gel is capable of adsorbing 40% of its own weight of water through its porous structure
→ Incorporation of a colouring agent like cobalt chloride which turns blue from purple, helps in knowing the stage of saturation of ~~dessicant~~ dissicant.

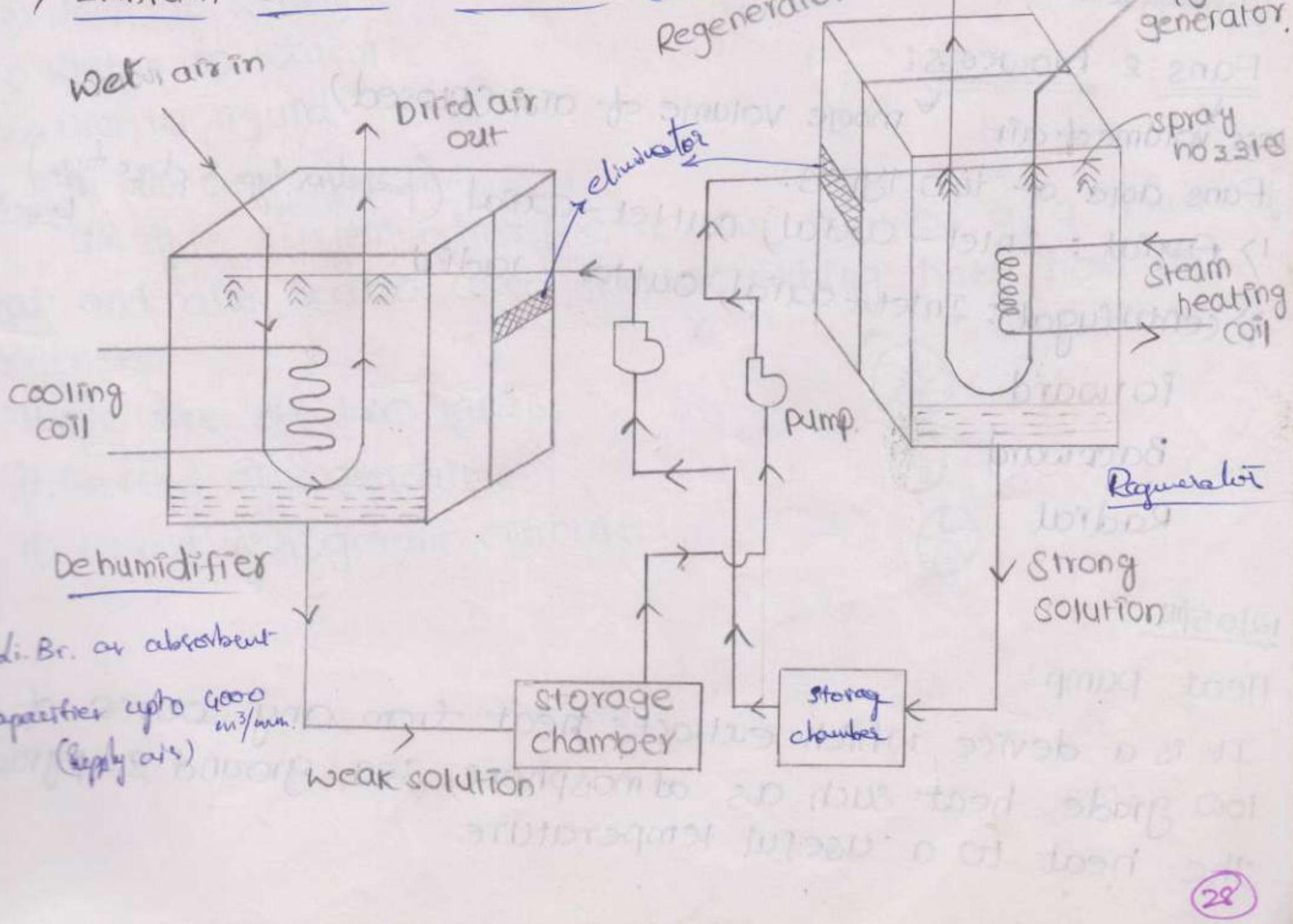


→ Capacity 1-3000 m³/min

fig: Bed B is used for dehumidifying air & bed A is used for reactivation

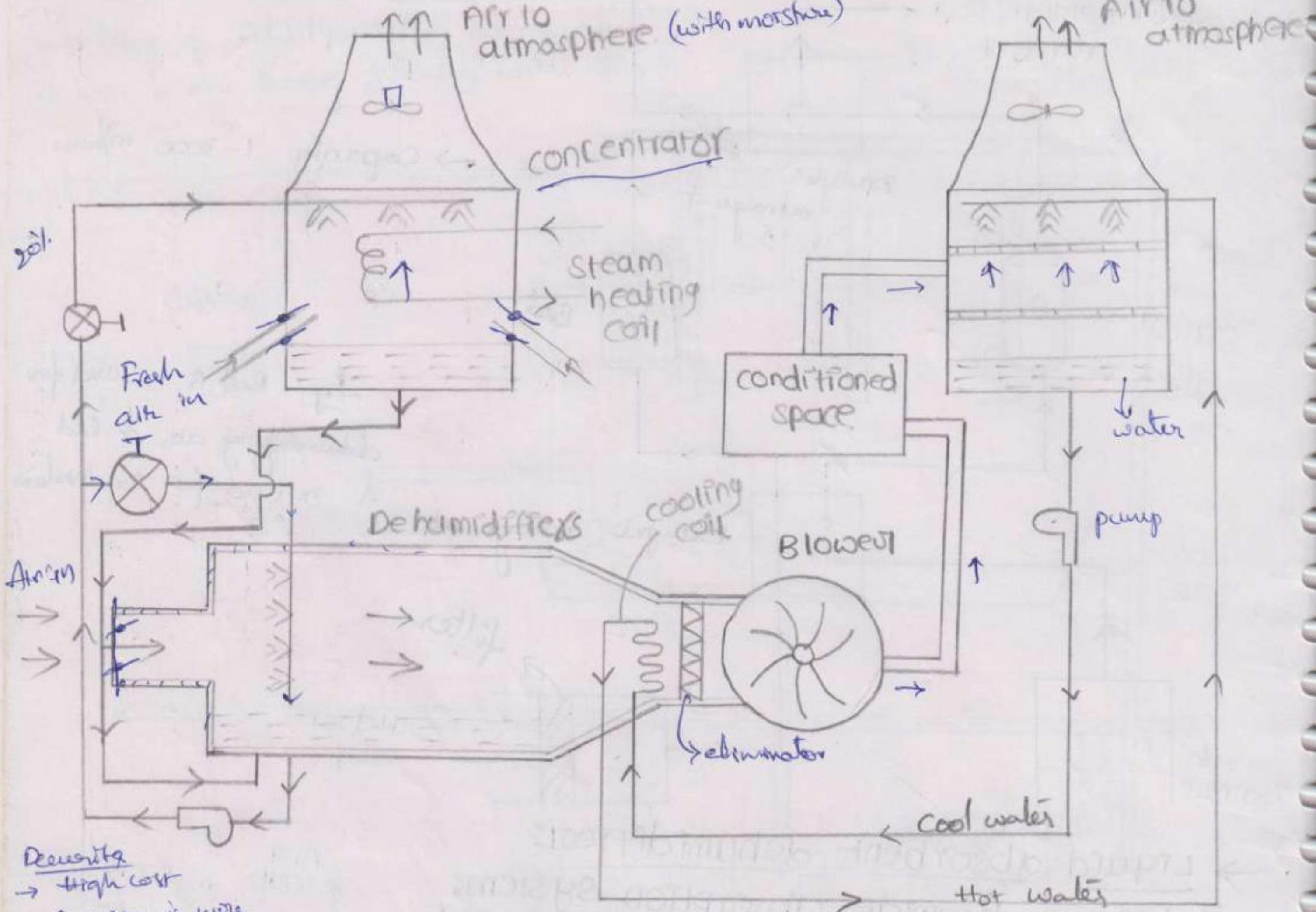
→ Liquid absorbent dehumidifieд:

i) Lithium Bromide absorption system:



→ Li.Br. as absorbent

-Capacities upto 4000 m³/min. (supply air) ii> Calcium Chloride System;



Demerits
→ High cost
→ Corrosion is more

Fans & blowers:
less volume of air — more volume of air (closed)

Fans are of two types.

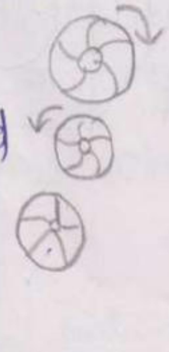
1> Axial : Inlet - axial, outlet - axial (propeller type & disc type) large vol

2> centrifugal : Inlet - axial, outlet - radial

forward

Backward

Radial



12/05/14

Heat pump:

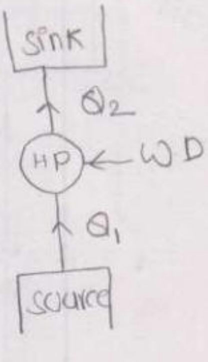
It is a device which extracts heat from any source of low grade heat such as atmosphere, sea, ground & upgrading the heat to a useful temperature.

Energy performance ratio of the heat pump = $\dfrac{\text{total heat supplied}}{\text{work done by the compressor}}$

$$EPR = \frac{Q_2}{W.D}$$

where Q_2 = total heat supplied

$\cancel{EPR}\, Q_2 = Q_1 + W.D$ by compressor



$$EPR = \frac{Q_1 + W.D}{W.D}$$

$$= \frac{Q_1}{W.D} + 1$$

$$\boxed{EPR = (COP)_R + 1}$$

***

Heat pump circuits:

The basic heat pump circuits which are commonly used are of 5 types.

1) Air to air circuit → ~~Forced~~ Fixed air circuit; → Fixed refrigerant circuit.
2) Water to air.
3) Air to water
4) Water to water
5) Air to liquid

1) Air to air circuit:

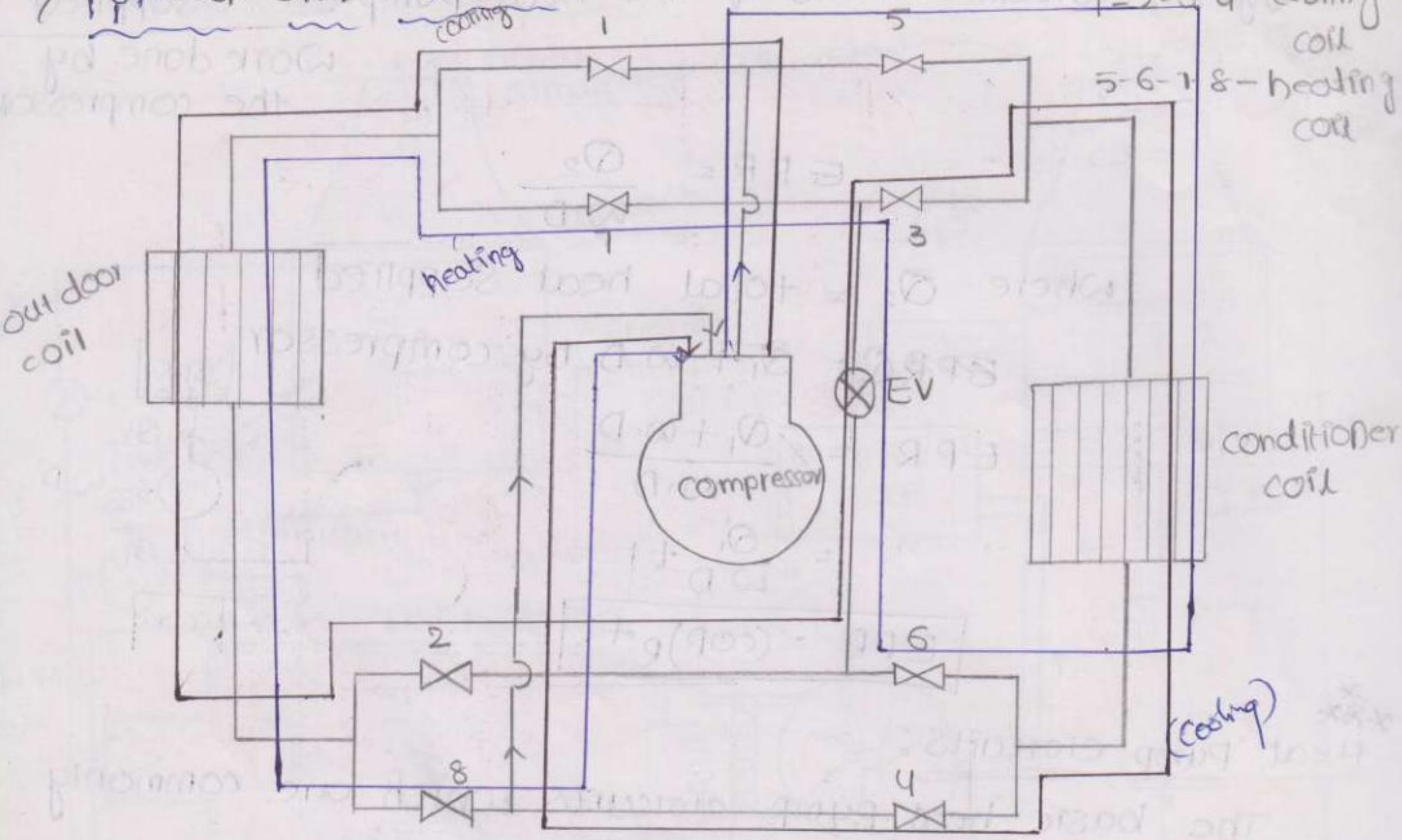
In this system atmospheric air is used as a source of heat and air is also used for absorbing heat from the condenser.

These are of two types.

i) ~~Forced~~ Fixed air circuit

ii) Fixed refrigerant circuit.

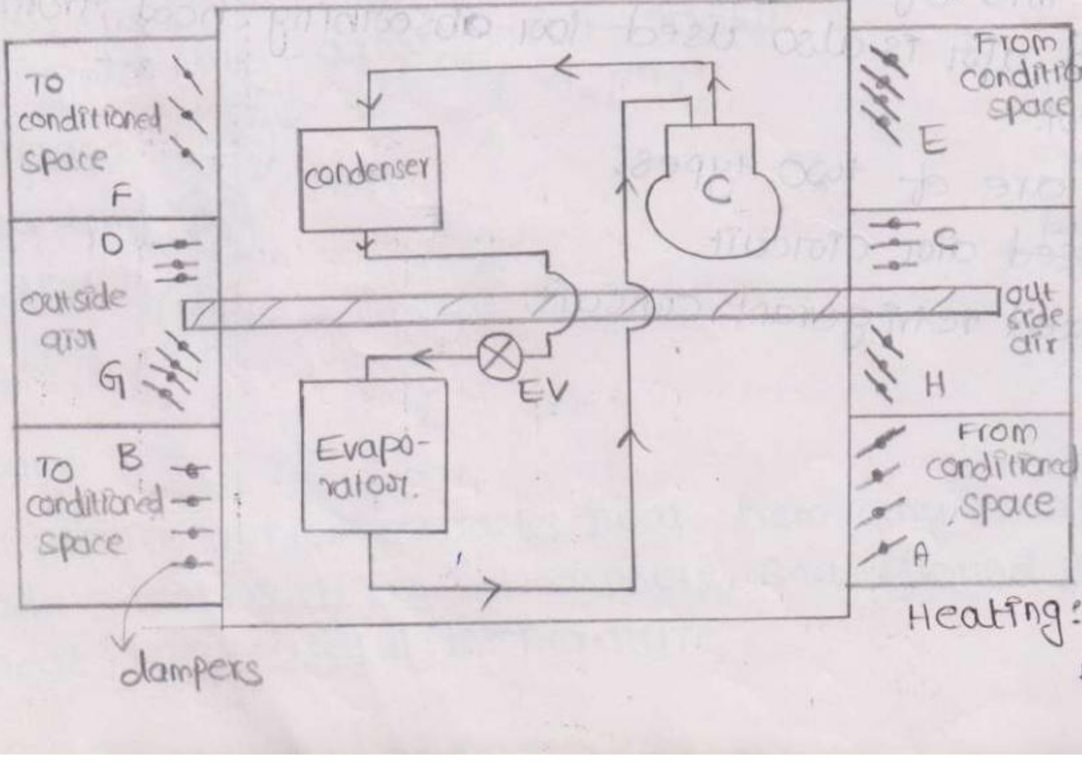
i) Forced air circuit:



* During heating cycle the heat is taken from the atmospheric air & it is pump to the room air and during cooling cycle the heat taken by the room air and it is pumped into the atmosphere.
* During heating cycle, the valves 5, 6, 7 & 8 remain open and 1, 2, 3 & 4 valves remains closed & vice versa for cooling.

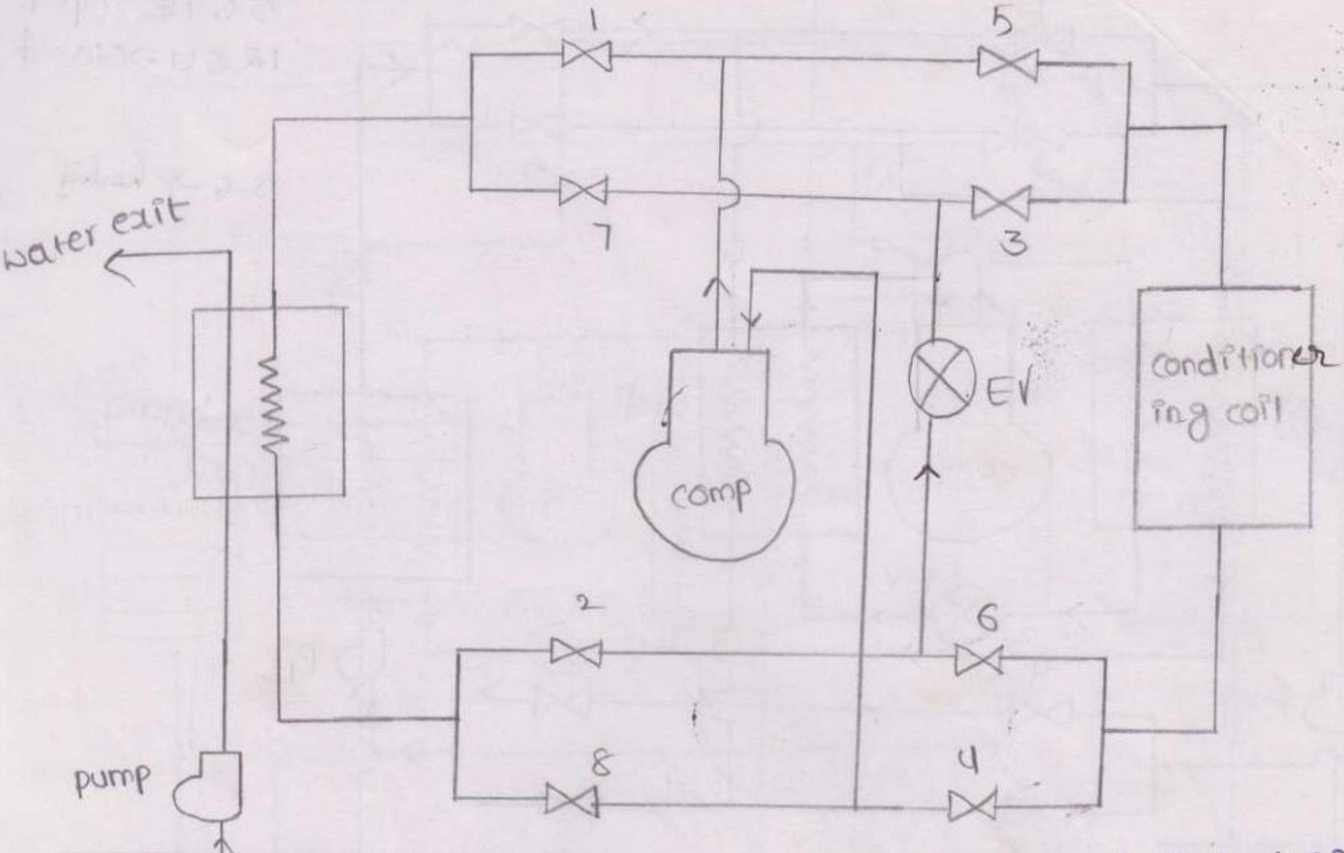
cooling: ABCD → open ✓
EFGH → ~~closed~~ open

ii) Fixed refrigerant circuit:



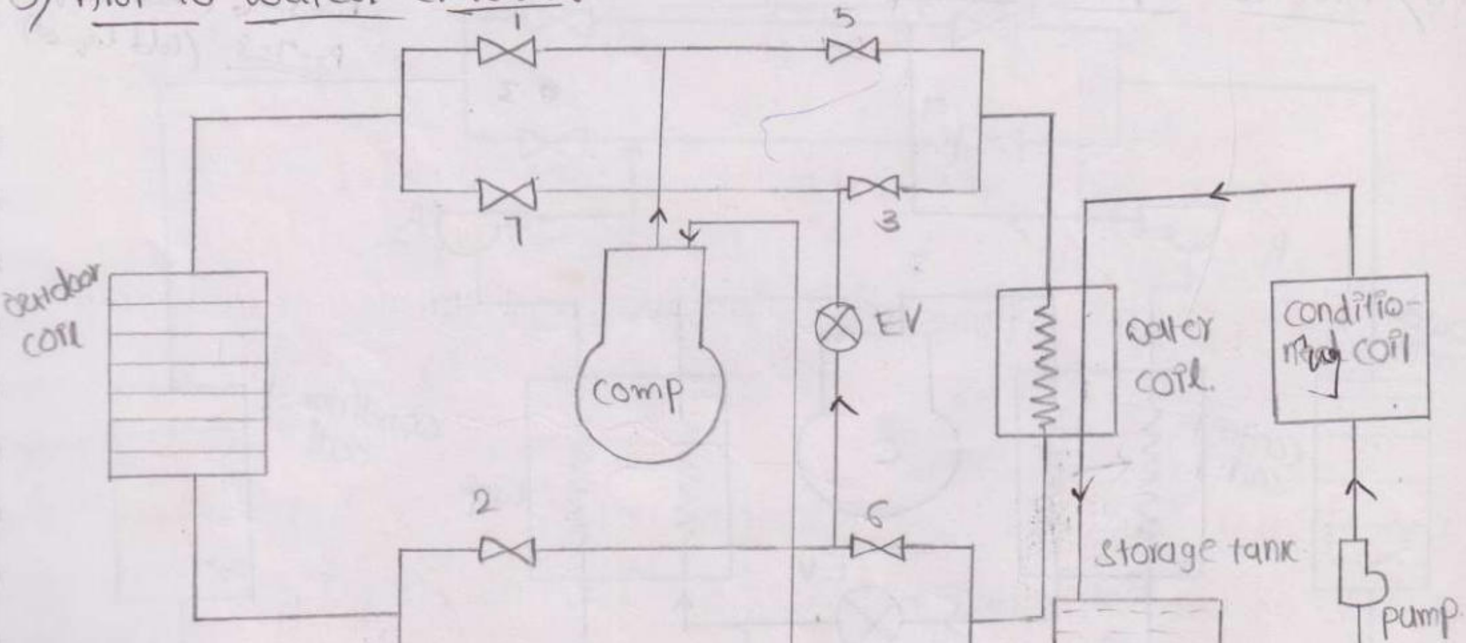
Heating: EFGH → open
ABCD → closed

2> Water to air circuit: & ~~Air to water~~ :-



cooling cycle 1-2-3-4 → open
5-6-7-8 → close

3> Air to water circuit:



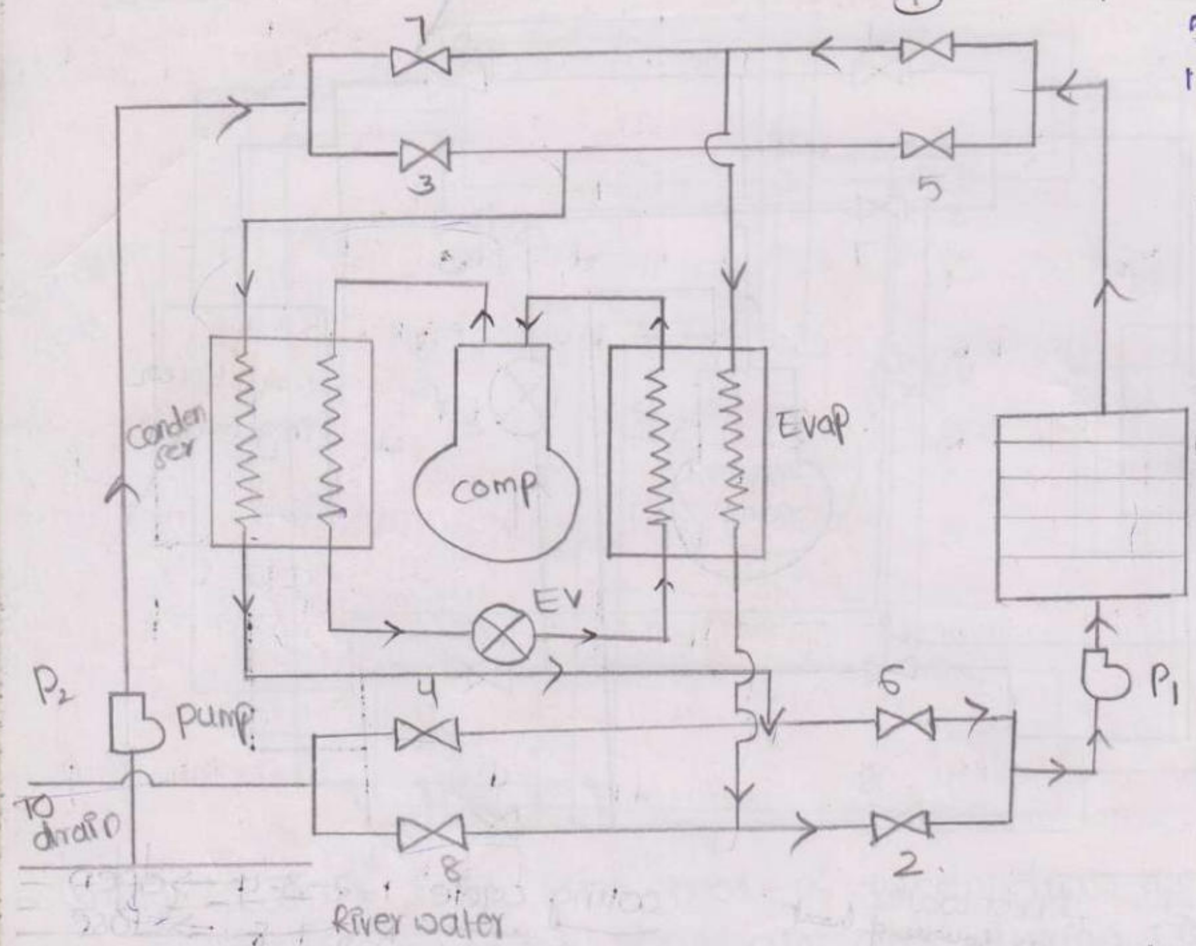
cooling cycle: 1-2-3-4 → open
5-6-7-8 → close

4> Water to water circuit:

water → source of heat fluid
→ medium of A/C
(1)

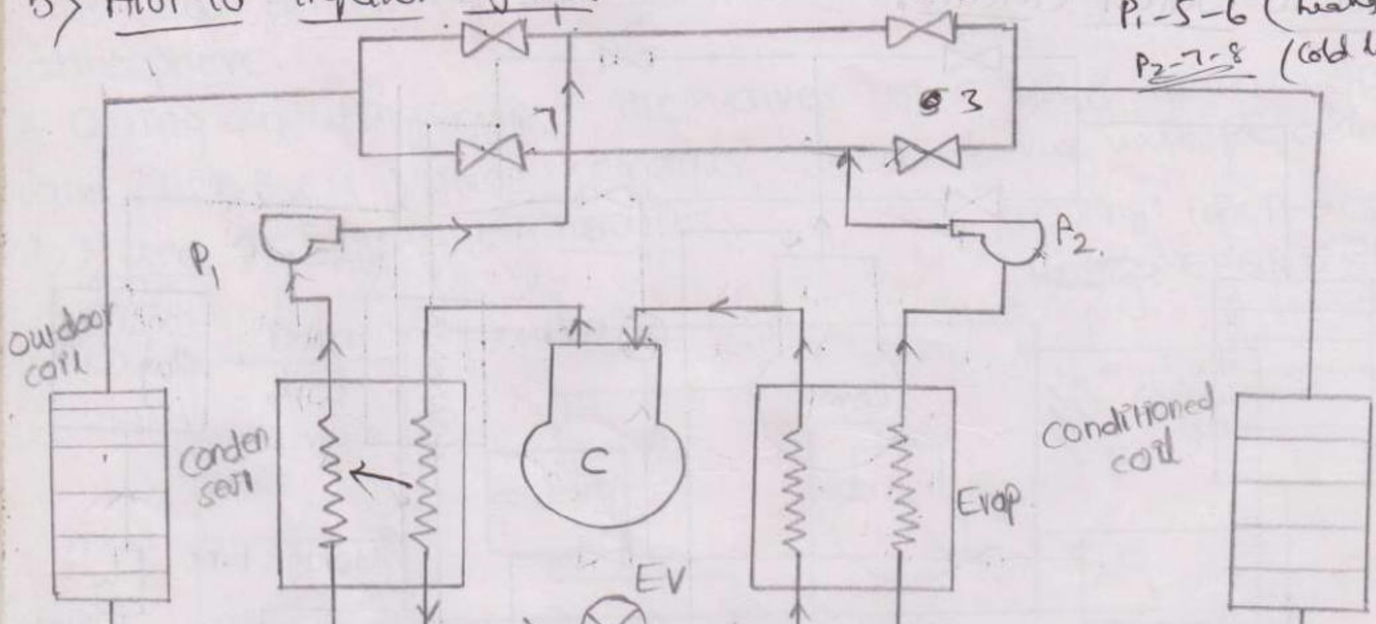
heating
5678 → open
1234 → closed

5-6 → heating



5> Air to liquid cycle:-

Heating → 5678 (open)
P_1-5-6 (heating liquid)
P_2-7-8 (cold liquid)



Air to liquid with auxilrary coil: