Steam nozzles

Flow of steam through nozzles, shapes of nozzles, effect of friction, Critical pressure ratio, Supersaturated flow, Impulse and Reaction Principles, compounding velocity diagram for simple and multi-stage turbines, speed regulations - Governors.

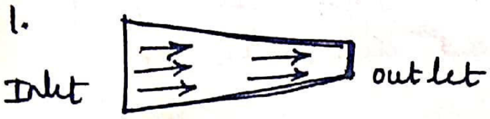
Nozzle

Nozzle is a duct of varying cross sectional area in which the velocity increases with corresponding drop in pressure.

Nozzles are used in steam turbines, gas turbines, jet engines, and rocket motors.

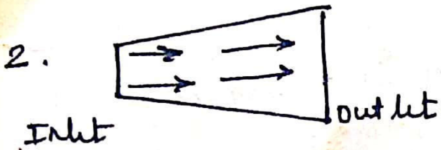
1. convergent nozzle
2. Divergent nozzle
3. Convergent - Divergent nozzle.

1.



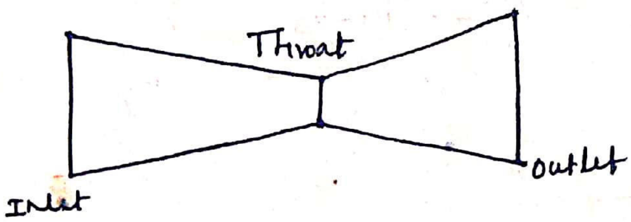
In convergent nozzle, the cross sectional area decreases from the inlet section to outlet section.

2.



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

3.



Steam flow Through nozzles.

(i) It is assumed as adiabatic flow.

(ii) since no heat is supplied (or) rejected by the steam during flow through a nozzle.

(iii) And there is no work done during the process

$Q = 0$ and $W = 0$.

(i) Velocity of steam.

$$V_2 = \sqrt{2000(h_1 - h_2)}$$

$$V_2 = 44.72\sqrt{(H_1 - H_2)}$$

mass of steam discharged through the nozzle

(ii) ~~Condition for maximum discharge~~

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

(iii) Condition for maximum discharge

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

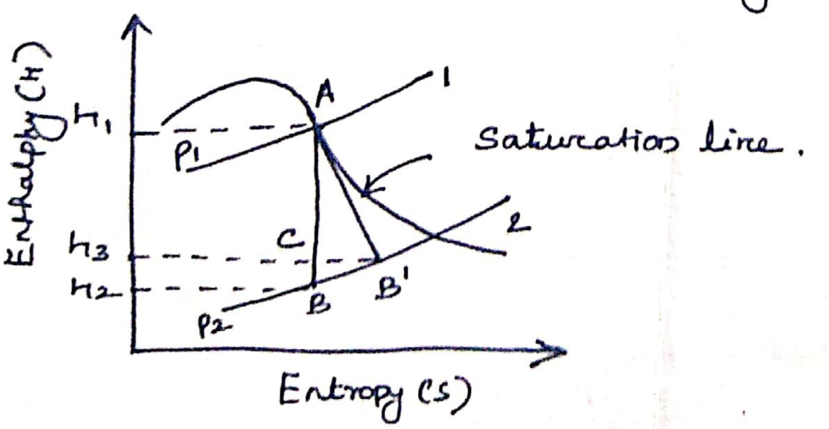
(iv) Nozzle efficiency. (or) Effect of friction in a nozzle.

When the steam flows through a nozzle the final velocity of steam for a given pressure drop is reduced due to the following reasons.

1. Due to the friction between the nozzle surface and steam.
2. Due to the internal fluid friction in the steam.
3. Due to shock losses.

These losses occur between the throat and exit in convergent-divergent nozzle.

$$\eta = \frac{\text{Actual enthalphy drop}}{\text{Isentropic drop enthalphy}} = \frac{AC}{AB} = \frac{h_1 - h_3}{H_1 - h_2}$$



1. Point A represents the initial condition of steam.
2. where the saturation line meet the intial pressure (P_1) line.
3. If the friction is neglected the expansion of steam from entry to throat is represented by the vertical line AB.
4. The enthalphy drop ($h_1 - h_2$) iseentropic enthalphy drop.
5. Due to friction in the nozzle the actual enthalphy drop in the steam will be less than ($h_1 - h_2$). The enthalphy drop is shown as AC instead of AB.
6. Actual enthalphy drop ($h_1 - h_3$)

Critical pressure ratio

Value of the pressure ratio $= \frac{P_2}{P_1}$ which produces max discharge from the nozzle. This ratio is called critical pressure ratio.

Where $P_1 \Rightarrow$ Inlet pressure.

$P_2 \Rightarrow$ Throat pressure.

(i) For saturated steam $n = 1.135$

Critical pressure ratio

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

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

$$\boxed{C_{pr} \quad \frac{P_2}{P_1} = 0.577}$$

(ii) For superheated steam $n = 1.3$

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

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

$$\boxed{C_{pr} \quad \frac{P_2}{P_1} = 0.546}$$

(iii) For gases $n = 1.4$

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

$$\boxed{\frac{P_2}{P_1} = 0.5282}$$

Problem no: 1

Dry saturated steam at 6.5 bar with negligible velocity expands isentropically in a convergent nozzle to 1.4 bar and dryness fraction 0.956. Determine the velocity of steam leaving the nozzle. If 13% heat is lost in friction, find the % reduction in the final velocity.

Given data

$P_1 = 6.5$ bar.

$P_2 = 1.4$ bar

$x_2 = 0.956$

Heat loss $= 13\%$

Solution

From steam table at 6.5 bar

$h_1 = h_g = 2758.8$ kJ/kg

At 1.4 bar

$h_f = 458.4$ kJ/kg $h_{fg} = 2231.9$ kJ/kg

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

$= 458.4 + 0.956 (2231.9)$

$h_2 = 2592.1$ kJ/kg.

$V_2 = \sqrt{2000 \times (h_1 - h_2)}$

$= \sqrt{2000 \times (2758.8 - 2592.1)}$

$= 577.39$ m/sec

Heat drop $13\% = 0.13$

Nozzle $\eta = 1 - 0.13$

$= 0.87$

Velocity of steam by considering the nozzle η

$V_2 = \sqrt{2000 \times (h_1 - h_2)\eta}$

$= \sqrt{2000 \times (2758.8 - 2592.1) \times 0.87}$

% Reduction in final velocity $= \dfrac{577.39 - 538.55}{577.39} \times 100$

$= 6.726\%$

Problem no: 2

Steam is expanded in a set of nozzles from 10 bar and 200°C to 5 bar. What type of nozzle is it? Neglecting the initial velocity find min area of the nozzle required to allow a flow of 3 kg/s under the given conditions. Assume that expansion of steam to be isentropic.

Sol:

Given:

Pressure at the entry to the steam nozzles

$P_1 = 10$ bar

$T_1 = 200°C$

Steam at exit pressure

$P_2 = 5$ bar

We know that

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

$$= \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{0.3}}$$

$$= \left(\frac{2}{2.3}\right)^{4.333} = 0.5457$$

$P_2 = P_1 \times 0.5457$

$= 10 \times 0.5457$

$P_2 = 5.5$ bar

Since throat pressure (P_2) is greater than the exit pressure the nozzle used is Convergent divergent nozzle. The minimum area will be at throat, where the pressure is 5.5 bar.

Pressure at 10 bar $h_1 =$

S_1

$h_1 = S_1$

$S_1 = S_2$

$S_1 = S_{f_2} + x_2 S_{fg_2}$

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

$h_2 = h_{f_2} + x_2 (h_{g_2} - h_{f_2})$

Specific volume of 5.5 bar

$v = 0.345\ m^3/kg.$

Velocity of Throat $V_2 = 44.72\sqrt{h_1 - h_2}$

$= 44.72\sqrt{}$

Throat area $A_2 = \frac{mv}{\cancel{a}_2 V_2}$

$= \frac{3 \times 0.345}{}$

$\boxed{A = 0.0021\ m^2}$

Problems no; 3

In a steam nozzle, the steam expands from 4 bar to 1 bar initial velocity is 60 m/s and the initial temperature 200°c. Determine the exit velocity if the nozzle efficiency is 92%. ①

pands

raction

f 13%.

final

<u>Solution</u>

Steam pressure at entry to the nozzle $P_1 = 4$ bar 200°c.

Steam Pressure at exit from the nozzle $P_2 = 1$ bar

Intial velocity of steam $C_1 = 60$ m/sec

Nozzle efficiency $\eta_{nozzle} = 92\%$

Exit velocity :?

<u>using steam table</u>

At $P_1 = 4$ bar 200°c

$h_1 = 2860.5$ kJ/kg. $S_1 = 7.171$ kJ/kg.

At $P_2 = 1$ bar

$hf_2 = 417.5$ kJ/kg $hfg_2 = 2257.9$ kJ/kg.

$Sf_2 = 1.3027$ kJ/kgK $Sfg_2 = 6.0571$ kJ/kg k.

Entropy remains Constant.

$$S_1 = S_2$$

$$S_1 = Sf_2 + x_2\, Sfg_2$$

$$7.171 = Sf_2 + x_2\, Sfg_2$$

$$7.171 = 1.3027 + x_2 (6.0571)$$

$$x_2 = \frac{7.171 - 1.3027}{6.0571}$$

$$x_2 = 0.969$$

$$h_2 = Hf_2 + x_2\, hfg_2$$

$$= 417.5 + 0.969 (2257.9)$$

$$\boxed{h_2 = 2605.4 \text{ kJ/kg}}$$

∴ Enthalphy drop $= h_1 - h_2 = 2860.5 - 2605.4$

$$\boxed{= 255.1 \text{ kJ/kg}}$$

Actual enthalpy drop $\cancel{h_1 - h_2}$

$$= \cancel{2860.5 - 2605.4}$$

$$= 25$$

$$= \eta_{nozzle} \times (h_1 - h_2)$$

$$= 0.92 \times (255.1)$$

$$\boxed{\text{Actual enthalpy drop} = 234.69\ kJ/kg.}$$

$$\text{Enthalpy drop Actual} = \frac{V_2^2 - V_1^2}{2}$$

$$\cancel{V_2^2} = \frac{V_2^2 - 60^2}{2} = 234.69$$

$$V_2^2 - 60^2 = 234.69 \times 2$$

$$V_2 = \sqrt{60^2 + 2 \times 234.69 \times 1000}$$

$$\boxed{V_2 = 687.7\ m/sec}$$

Home work no: 1

Dry saturated steam enters a steam nozzle at a pressure of 15 bar and is discharged at a pressure of 2.0 bar. If the dryness fraction of discharge steam is 0.96. What will be the final velocity of steam?

Problems no: 1

①

Dry saturated steam at 6.5 bar with negligible velocity expands isentropically in a convergent nozzle to 1.4 bar and dryness fraction 0.956. Determine the velocity of steam leaving the nozzle. If 13% heat is lost in friction, find the percentage reduction in the final velocity.

Given data

$P_1 = 6.5$ bar.

$P_2 = 1.4$ bar

$x_2 = 0.956$

Heat loss $= 13\%$

From steam table at 6.5 bar.

$h_1 = h_g = 2758.8$ kJ/kg.

At 1.4 bar.

$h_{f_2} = 458.4$ kJ/kg.

$h_{fg_2} = 2231.9$ kJ/kg.

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

$= 458.4 + 0.956 \times 2231.9$

$\boxed{h_2 = 2592.1 \text{ kJ/kg}}$

$V_2 = \sqrt{2000(h_1 - h_2)}$

$= \sqrt{2000(2758.8 - 2592.1)} = 577.39$ m/sec.

Heat drop is 13% = 0.13

Nozzle efficiency $\eta = 1 - 0.13 = 0.87$

velocity of steam by considering the nozzle efficiency

$V_2 = \sqrt{2000(h_1 - h_2)\eta}$

$= \sqrt{2000(2758.8 - 2592.1) \times 0.87}$

$\boxed{V_2 = 538.55 \text{ m/sec}}$

% Reduction in final velocity $= \dfrac{577.39 - 538.55}{577.39} \times 100$

$\boxed{= 6.726\%}$

Problem no: 2

Dry saturated steam at 2.8 bar is expanded through a convergent nozzle to 1.7 bar. The exit area is 3 cm^2. Calculate the exit velocity and mass flow rate for (i) Isentropic expansion and (ii) Super saturated flow.

Given data:

$P_1 = 2.8$ bar.

$P_2 = 1.7$ bar.

$A_2 = 3 cm^2 = 3 \times 10^{-4} m^2$

Solution

From steam table

At 2.8 bar

$h_1 = 2721.5$ kJ/kg

$s_1 = 7.014$ kJ/kg K.

$v_1 = 0.64600$ m^3/kg

At 1.7 bar.

$h_f = 483.2$ kJ/kg

$h_{fg} = 2215.6$ kJ/kg

$s_f = 1.475$ kJ/kg K.

$s_{fg} = 5.706$ kJ/kg K

$v_f = 0.001056$ m^3/kg

$v_g = 1.0309$ m^3/kg

For Isentropic flow

$S_1 = S_2 = 7.014$ kJ/kg K.

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

$7.014 = 1.475 + x_2 \times 5.706$

$x_2 = 0.97$

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

$= 483.2 + 0.97 \times 2215.6$

$\boxed{h_2 = 2634.15 \text{ kJ/kg}}$

$v_2 = x_2 \, v_{g_2}$

$= 0.97 \times 1.0309$

$v_2 =$ ... m^3/kg

Velocity of steam at exit

$$V_2 = \sqrt{2000(h_1 - h_2)}$$

$$= \sqrt{2000\,(2721.5 - 2631.15)}$$

$$\boxed{V_2 = 418 \text{ m/s}}$$

Mass flow rate of steam

$$m = \frac{A_2 V_2}{v_2} = \frac{3 \times 10^4 \times 418}{1.00}$$

$$\boxed{m = 0.1257 \text{ kg/sec}}$$

For Super Saturated flow

$$V_2 = \sqrt{\frac{2n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= \sqrt{\frac{2 \times 1.3}{1.3-1} \times 2.8 \times 10^5 \times 0.6460 \left[1 - \left(\frac{1.7}{2.8}\right)^{\frac{1.3-1}{1.3}}\right]}$$

$$V_2 = 413 \text{ m/sec}$$

Specific volume $v_2 = v_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}$

$$= 0.6460 \times \left(\frac{2.8}{1.7}\right)^{\frac{1}{1.3}}$$

$$= 0.94827 \text{ m}^3/\text{kg}.$$

Mass flow rate

$$m = \frac{A_2 V_2}{v_2} = \frac{3 \times 10^{-4} \times 413}{0.94827}$$

$$\boxed{m = 0.1306 \text{ kg/sec}}$$

Problem no: 3.

Dry air at a pressure of 12 bar and 573 K is expanded isentropically through a nozzle at a pressure of 2 bar. Determine the maximum mass flow rate through the nozzle of 0.00015 m^2 area.

Given data:

$P_1 = 12$ bar $= 1200$ kpa

$T_1 = 573$ K

$P_2 = 2$ bar $= 200$ kpa.

$A = 0.00015 m^2$

$\frac{12 \times 10^5}{1000} = $ kpa

To find

max mass flow rate

Formula

(i) Mass flow rate through nozzle

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

(ii) For iscentropic process

$$P_1 V_1^{\gamma} = P_2 V_2^{\gamma}$$

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

(iii) From ideal gas equation

$$P_1 V_1 = RT_1$$

$$V_1 = \frac{RT_1}{P_1}$$

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

$$C_2 = \sqrt{2000 \times C_p (T_1 - T_2)}$$

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

Solution

From ideal gas equation

$$P_1 V_1 = RT_1$$

$$V_1 = \frac{RT_1}{P_1}$$

$$V_1 = \frac{0.287 \times 573}{1200} = 0.137 m^3/kg.$$

For isentropic Process

$$P_1 V_1^{\gamma} = P_2 V_2^{\gamma}$$

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

$$= \left(\frac{12}{2}\right)^{\frac{1}{1.4}} \times 0.137$$

$$v_2 = 0.493 \ m^3/kg.$$

Relation between pressure and Temperature for isentropic process

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

$$T_2 = \left(\frac{2}{12}\right)^{\frac{1.4-1}{1.4}} \times 573$$

$$\boxed{T_2 = 343.42 \ K}$$

$$C_2 = \sqrt{2000 \times C_p (T_1 - T_2)}$$

$$= \sqrt{2000 \times 1.005 (573 - 343.42)}$$

$$C_2 = 679.305 \ m/sec.$$

Mass flow rate through nozzle

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

$$m = \frac{0.00015 \times 679.305}{0.493}$$

$$\boxed{m = 0.207 \ kg/s}$$

Problem no: 4

Dry Saturated steam at a pressure of 8 bar enters a convergent divergent nozzle and leaves it at a pressure of 1.5 bar. If the steam flow process is isentropic and if the corresponding expansion index is 1.135, find the ratio of cross sectional area at exit and throat for maximum discharge.

Given data

$P_1 = 8$ bar

$P_2 = 1.5$ bar

$n = 1.135$

To find

ratio of Cross Sectional area.

Formula

$$\frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2}$$

$$\boxed{\frac{A_2}{A_1} = \frac{v_2 V_1}{V_2 v_1}}$$

$$V_2 = \sqrt{2000(h_1 - h_2)}$$

h_1 - Pressure at 8 bar.

$h_2 = h_{f2} + x_2 h_{fg2}$ at 1.5 bar.

$S_2 = S_{f2} + x_2 \, S_{fg2}$ $\qquad S_1 = S_2$

$v_2 = x_2 \, V_{g2}$

$$\cancel{V_1 = \sqrt{2000(h_1 - h_{11})}}$$

$\cancel{h_{11} = h_{f11} + x_1 \, h_{fg11}}$

$\cancel{x_{11} =}$

$S_{11} = S_{f11} + x_{11} \times S_{fg11}$

(4)

Problem no: 5

Steam at 20 bar and 250°c enters a group of Convergent-divergent nozzles. The back up pressure of nozzle is 0.07 bar. Neglect the losses in convergent part. Assume a loss 10% of enthalpy drop available in the devergent part. Find the number of nozzles required to discharge 13.6 kg/s. The throat area of each nozzle is 3.97 cm². Also Determine area of exit of each nozzle.
Assume Critical pr ratio 0.546

Given data

$P_1 = 20$ bar $\qquad P_r = 0.546$

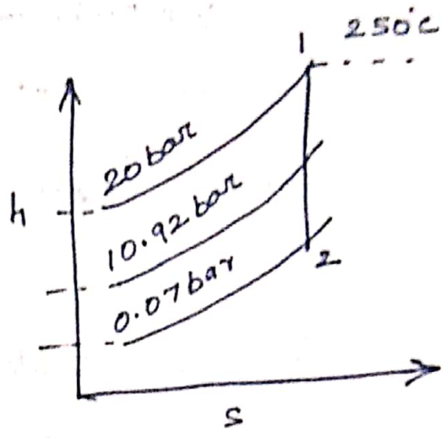
$T_1 = 250°c$

$P_2 = 0.07$ bar

$m = 13.6$ kg/sec

$\eta = 90\%$.

Throat area = $3.97\ cm^2$.



To find

(i) no of nozzles

(ii) ~~Throat~~ area of the exit.

Formula

(I) Exit area of nozzle $A_2 = \dfrac{m \times v_2}{V_2}$ $\qquad V_2 = \sqrt{2000(h_1 - h_2)\eta}$

(ii) Mass flow rate of steam/nozzle = $m = \dfrac{A_t \times V_t}{v_t}$

(iii) No of nozzle required = $\dfrac{\text{Total mass flow rate}}{m}$

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

Solution

Properties of steam (from mollier diagram) at 20 bar and 250°C

$h_1 = 2900$ kJ/kg

The process is isentropic. from $h_1 = 2900$ kJ/kg draw a vertical line in the Mollier diagram upto 0.07 bar pressure line.

$h_2 = 2030$ kJ/kg

$v_2 = 15.9313\ m^3/kg$

$$\frac{P_t}{P_1} = 0.546$$

$\therefore$ Throat Pressure $P_t = P_1 \times 0.546$

$= 20 \times 0.546$

$= 10.92$ bar

The expansion is iscentropic from $h_f = 2900$ kJ/kg. draw a vertical line in the Mollier diagram upto 10.92 bar Pressure line. Now note the following values at that point

$v_t = 0.177213 \ m^3/kg$

$h_t = 2780 \ kJ/kg$

$$V_t = \sqrt{2000(h_1 - h_t)} = \sqrt{2000(2900 - 2780)}$$

$V_t = 489.9$ m/sec

Velocity of steam at exit

$$V_2 = \sqrt{2000(h_t - h_2) \times \eta} = \sqrt{2000(2780 - 2030) \times 0.9}$$

$= 1161.895$ m/s.

(i) Mass flow rate of steam / nozzle

$$m = \frac{A_t \times V_t}{v_t} = \frac{3.97 \times 10^{-4} \times 489.9}{0.177213}$$

$m = 1.0975$ kg/s

(ii) No of nozzle required $= \dfrac{\text{Total mass flowrate}}{m}$

$$= \frac{13.6}{1.0975} = 12.39 \simeq 12$$

iv) Exit area of nozzle $\quad A_2 = \dfrac{m \times v_2}{V_2}$

$$= \frac{1.0975 \times 15.9313}{1161.895}$$

$= 150.48 \times 10^{-4} \ m^2$

Steam Turbines

The Steam Turbine is a prime mover in which the potential energy of the steam is transformed into kinetic energy, and latter in its turn is transformed into the mechanical energy of rotation of the turbine shaft.

Turbines are classified into

(a) impulse.

(b) Reaction

(c) Combination of impulse and Reaction.

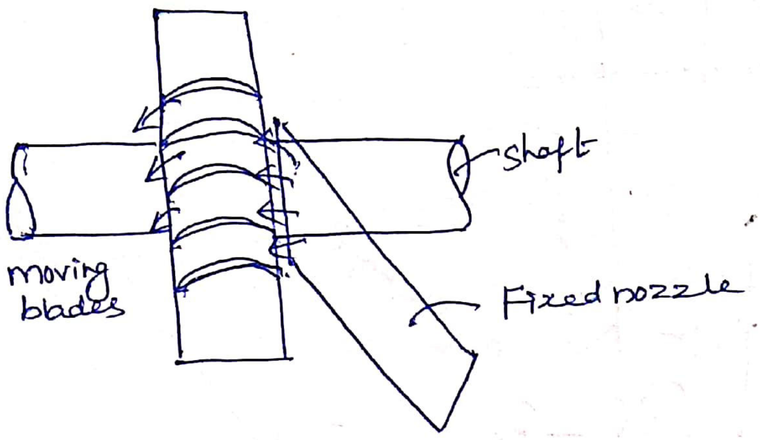
Classification of Steam Turbines

1. On the basis of method of steam expansion.
   - (a) Impulse turbine
   - (b) Reaction turbine
   - (c) Combination of impulse and reaction turbine.
2. On the basis of number of stages.
   - (a) Single stage turbine
   - (b) Multi stage turbine
3. On the basis of steam flow directions.
   - (a) Axial turbine
   - (b) Radial turbine
   - (c) Tangential turbine
   - (d) Mixed flow turbine.
4. On the basis of pressure of steam
   - (a) High pressure turbine
   - (b) Low pressure turbine
   - (c) Medium pressure turbine.

Steam Turbine.

1. A steam turbine is rotary machine which is designed to Convert the energy of high pressure and high temp steam into mechanical power. The operation of steam turbine wholly depends upon the dynamic action of the steam.

2. In this the steam is first expanded in a set of nozzles (or) passages upto exit pressure where in the pressure energy of steam is converted into kinetic energy.



There are mainly two types.

Impulse Turbine

Reaction Turbine.

Classification

1. According to principle of action of steam
   (a) Impulse (b) Reaction.

2. According to direction of steam flow
   (a) Axial (b) Radial (c) Tangential.

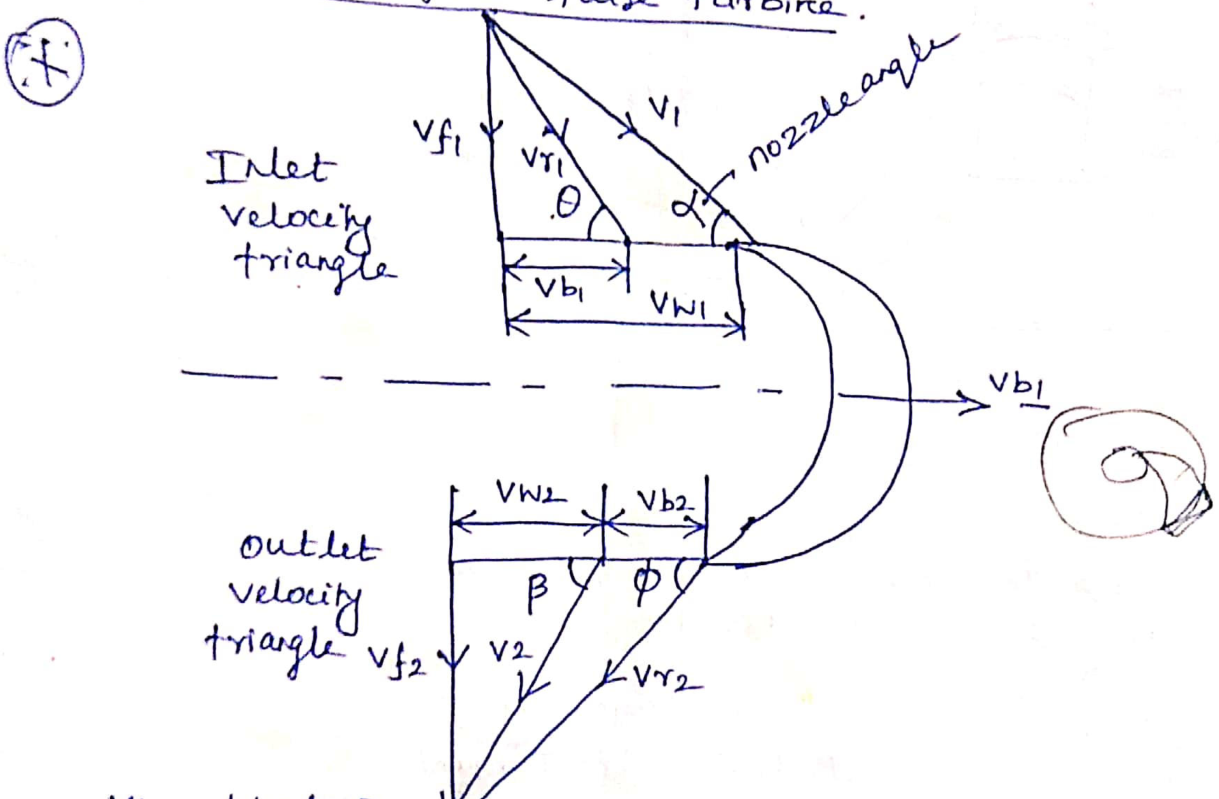
3. According to number of pressure stages.
   (a) single stage (b) multistage.

4. According to ~~method~~ basis of ~~governing~~ pressure
   (a) High pressure (b) Low pressure (c) Medium

Impulse Turbine

1. It Consist of nozzles and moving blades.
2. Pressure drop occurs only in nozzles not in moving blades.
3. Steam ~~passes over~~ strikes the blade with kinetic energy.
4. It has constant blade channels area
5. Due to more pressure drop per blade, number of stage required is less
6. power developed is less
7. velocity of turbine is more.

Velocity diagram for impulse turbine.



V_b - blade speed

V_1 - Absolute velocity of steam entering the moving blade.

V_{f1} - velocity of flow at entrance of moving blade.

V_{r1} - Relative velocity of jet entrance of moving blade.

V_{w1} - Velocity of whirl at entrance of moving blade.

α - Angle with the tangent of the wheel at which the steam with velocity V_1 enters.

θ – Entrance angle of moving blade.
ϕ – Exit angle of moving blade.
β – Angle which the discharging steam

$V_2, V_{f2}, V_{r2}, V_{w2}$ - Corresponding values of exit moving blade.

blade Co-efficient = $\frac{V_{r2}}{V_{r1}}$

Formulas

i) Tangential force on the wheel = mass of steam x acceleration

= mass of steam x change of Velocity

$$\boxed{F_x = m \times (V_{w1} + V_{w2})}$$

(ii) Work done on blades/s = Force x Distance traveled.

$= m \times (V_{w1} + V_{w2}) \times V_b$

iii) Power developed/wheel = $P = m \times (V_{w1} + V_{w2}) V_b$

iv) blade efficiency $\eta_b = \frac{\text{Work done on the blade}}{\text{Energy supplied}}$

$$\eta_b = \frac{m (V_{w1} + V_{w2}) V_b}{\frac{m V_1^2}{2}}$$

$\frac{m \times 1}{2V^2}$

$$\eta_b = \frac{2 m V_b (V_{w1} + V_{w2})}{V_1^2}$$

(v) Stage efficiency $\eta_{stage} = \frac{V_b (V_{w1} + V_{w2})}{h_1 - h_2}$ Total energy supplied.

(vi) Axial Thrust $F_y = m (V_{f1} - V_{f2})$

Problem no: 1

~~Steam at 10.5 bar and 0.95 dryness is~~ ~~expanded through~~
Convergent divergent

The following data refer to a single stage Impulse turbine. Isentropic nozzle entropy drop = 200 kJ/kg. Nozzle efficiency = 90%. Nozzle angle = 25°. Ratio of blade speed to whirl component of steam speed = 0.5. Blade coefficient = 0.9. The velocity of steam entering the nozzle 30 m/sec. Find (i) The blade angle at the inlet and outlet if the steam enters the blade without shock and leaves the blade in the axial direction (ii) Blade efficiency (iii) power developed and (iv) Axial thrust if the steam flow rate is 10 kg/s.

Given data

$H_e = 200 \text{ kJ/kg}$ (~~Chitke~~).

$\eta_N = 90\%$

nozzle angle: α

V_b = blade speed $\quad \frac{V_b}{V_W} = 0.5$.

V_W = Whirl speed

~~V_{r2}~~ blade co-eff $\frac{V_{r2}}{V_{r1}} = 0.9$

V_1 = Velocity of steam entering the nozzle $\}$ $V_1 = 30 \text{ m/sec}$.

$V_2 = V_{f2}$

$V_{W2} = 0$

$\beta = 90°$

Actual enthalpy drop = $h_i - h_e$

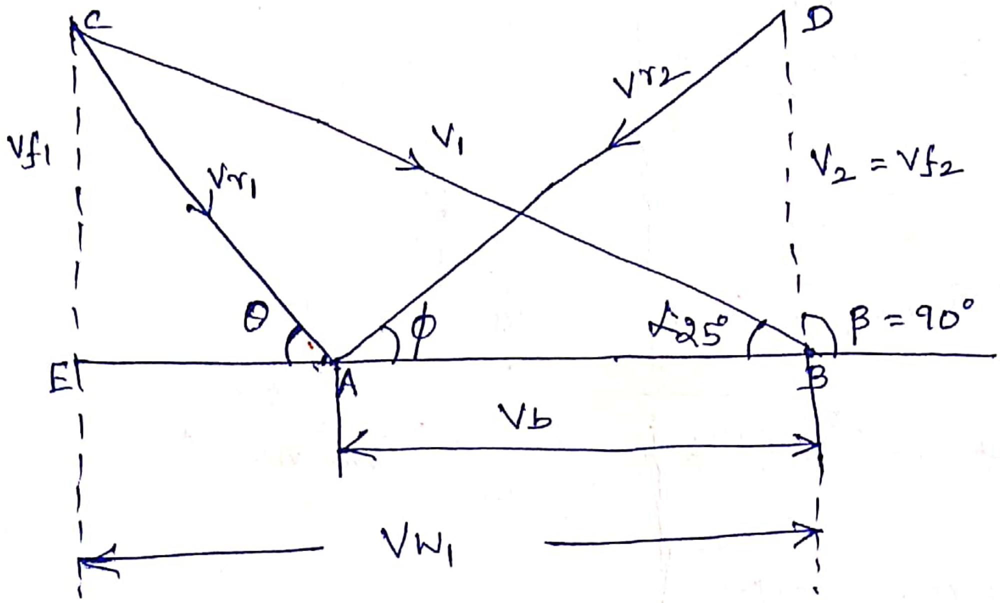
Formulas

(i) Power developed $P = m\,(V_{w_1} + V_{w_2}) \times V_b$

(ii) Blade efficiency $\eta_b = \dfrac{\cancel{m}\,(V_{w_1} + V_{w_2}) \times V_b}{\cancel{m} \times \frac{1}{2} V^2}$

(iii) Axial Thrust $F_y = m\,(V_{f_1} - V_{f_2})$

Velocity triangle



Actual enthalpy drop $h_i - h_e = (h_i - h_e) \times \eta_v$

$= 200 \times 0.9$

$= 180\ kJ/kg$.

$= 180 \times 10^3\ J/kg$

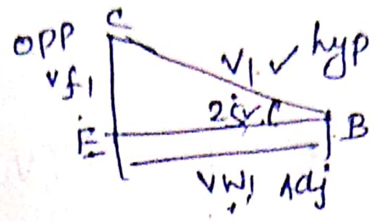
Exit velocity nozzle $V_e = \sqrt{2(h_i - h_e) + V_1^2}$

$= \sqrt{2 \times 180 \times 10^3 + 30^2}$

$V_e = 600.75\ m/s$

$V_e = V_1 = 600.75\ m/sec$

From Δ BCE



$\cos\theta = \frac{Adj}{hyp}$

$\cos\theta = \frac{V_{w1}}{V_1}$

$\boxed{V_{w1} = \cos\theta \cdot V_1}$

$V_{w1} = V_1 \cos 25^\circ$

$= 600.75 \cos 25^\circ = 544.46$

$V_{f1} = V_1 \sin 25^\circ$

$= 600.75 \sin 25^\circ = 253.89$

$\sin\theta = \frac{opp}{hyp}$

$\sin\theta = \frac{V_{f1}}{V_1}$

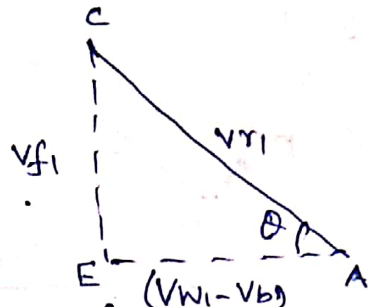
$\boxed{V_{f1} = \sin\theta \cdot V_1}$

$\frac{V_b}{V_{w1}} = 0.5$

$V_b = 0.5\, V_{w1}$

$= \underline{\underline{272.23}}$ m/sec.

From Δ ACE



$V_{r1} = \sqrt{V_{f1}^2 + (V_{w1} - V_b)^2}$

$= \sqrt{(253.89)^2 + (544.46 - 272.23)^2}$

$\boxed{V_{r1} = 372.25 \text{ m/sec.}}$

$\tan\theta = \frac{V_{f1}}{V_{w1} - V_b} = \frac{253.89}{544.46 - 272.23}$

$\boxed{\theta = 43^\circ}$

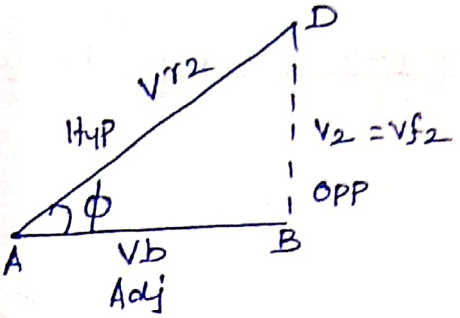
But

$V_{r2} = 0.9\, V_{r1}$

$= 0.9 \times 372.25 = 335.03$ m/sec

$\boxed{V_{r2} = 335.03 \text{ m/sec}}$

From Δ ADB $\cos\phi$



$$\cos\phi = \frac{Adj}{Hyp}$$

$$= \frac{AB}{AD} = \frac{V_b}{V_{r2}}$$

$$= \frac{272.23}{372.25}$$

$$\boxed{\phi = 35^\circ 39'}$$

$$V_2 = \sqrt{V_r^2 - V_b^2}$$

$$= \sqrt{335.03^2 - 272.03^2}$$

$$\boxed{V_{f2} = V_2 = 195.28 \text{ m/sec}}$$

Power developed $P = m(V_{w1} + V_{w2}) \times V_b$

m assume 1 $\quad = 1(544.46 + 0) \times 272.23$

$$P = 148.21 \text{ KW.}$$

$$\text{Blade } \eta = \frac{\cancel{m}(V_{w1} + V_{w2}) V_b}{\cancel{m} \times \frac{1}{2} V_1^2}$$

$$= \frac{(544.46 + 0) \, 272.23}{\frac{1}{2} \times (600.75)^2}$$

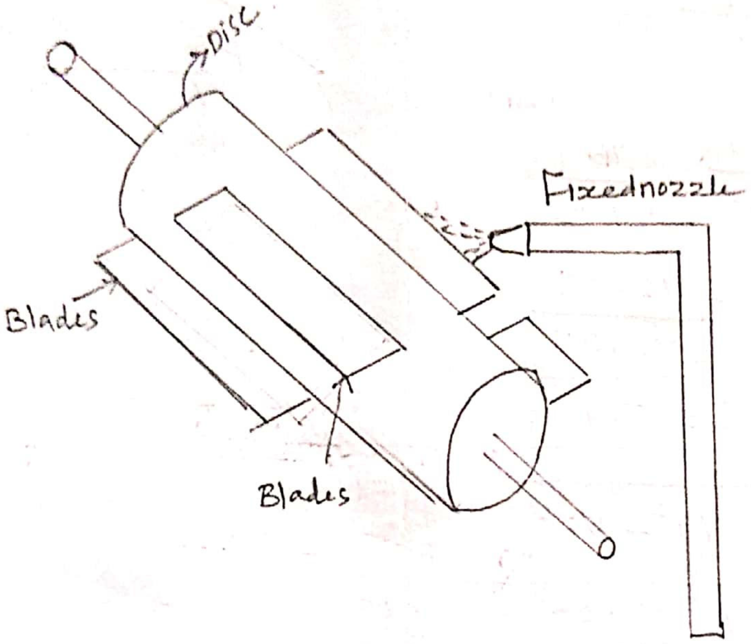
$$= \frac{148218.3}{180.45 \times 10^3} = 82\%$$

=

Axial Thrust $\quad F_y = m(V_{f1} - V_{f2})$

$$= 1(253.89 - 195.28)$$

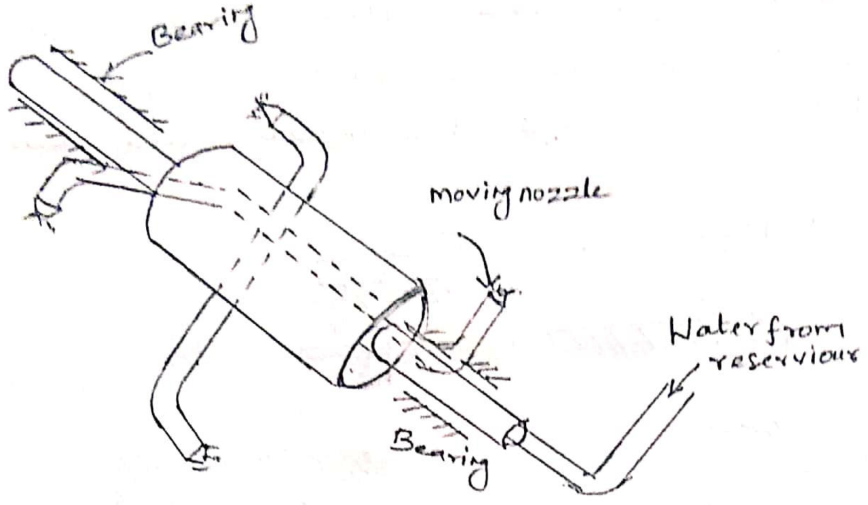
Principle of Impulse Turbine



In Impulse Turbine, the steam at high pressure and temperature with low Velocity expands through nozzles where the pressure reduces and the Velocity increases. The high velocity jet of steam which is obtained from the nozzle impinges on blade fixed on a rotor. The blade change the direction of steam flow without changing its pressure. It causes the change in momentum and thus the force is developed which drives the turbine rotor. Here the nozzle are stationary and fitted in a casing.

Ex:- De-laval, Curtis and Rateau turbines.

Principle of Reaction Turbine.



In reaction turbine, the steam expands both in fixed and moving blades continuously as the steam passes over them. As it expands, there is some increase in velocity thereby resulting the reaction force. The pressure drop occurs gradually and continuously over both moving and fixed blades.

Ex: Parson's turbine.

Working of Simple Impulse Turbine

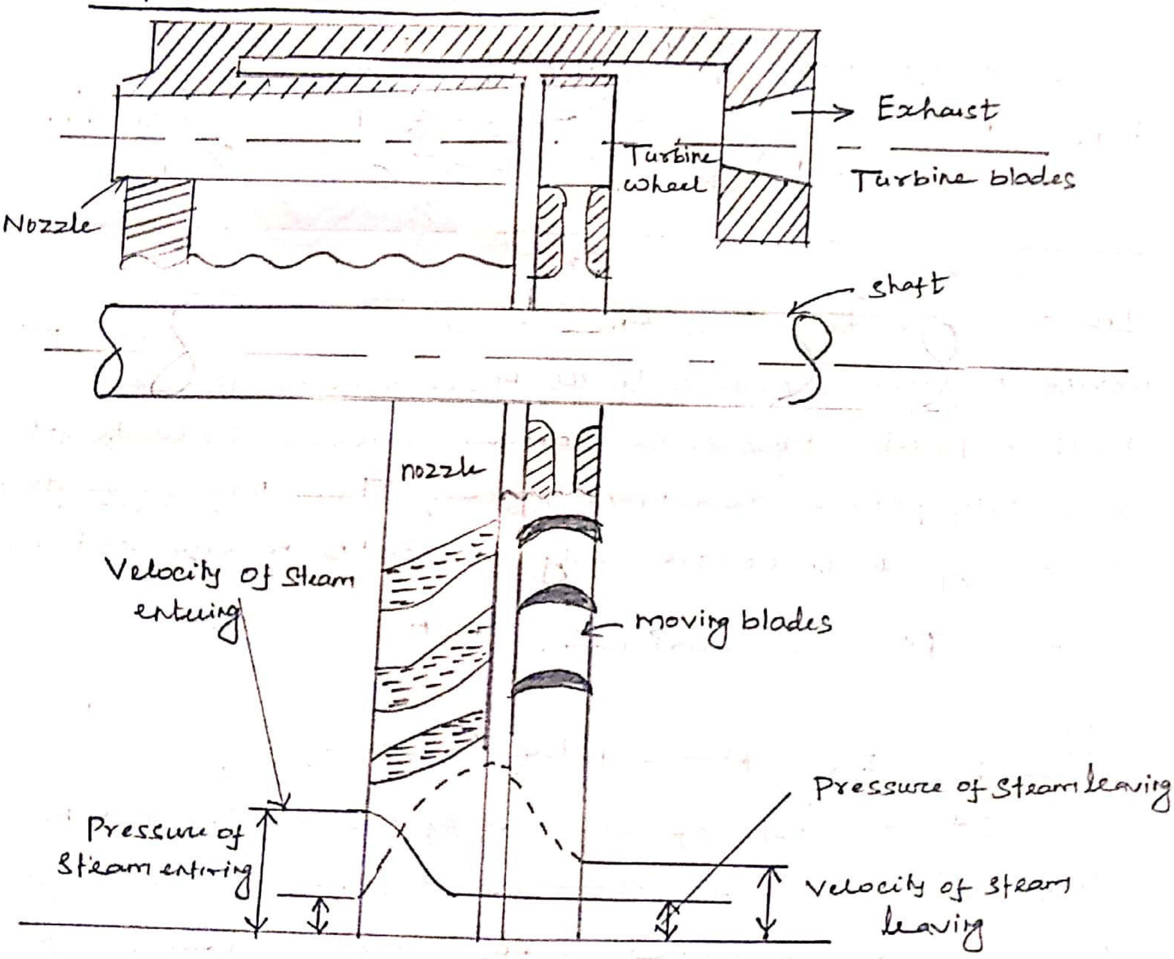
It consist of one set of nozzle followed by one set of moving blade. A rotor is mounted on a shaft. The moving blades are attached to the rotor. The steam from the boiler at high pressure and low velocity enters the nozzle which is fitted in the casing. The steam expands in the nozzle where the pressure drop to P_1 and the velocity increases to V_1.

This high velocity steam jet impinges over the blade mounted on the rotor attached to the shaft. It causes the rotation of the turbine shaft and thus the useful work is obtained.

Pressure of steam when it moves over the blades remains constant, but the velocity decreases.

The middle portion shows the development of nozzle and bladding. The bottom portion shows the variation of velocity and the pressure of the steam during which it passess through the nozzles and blades.

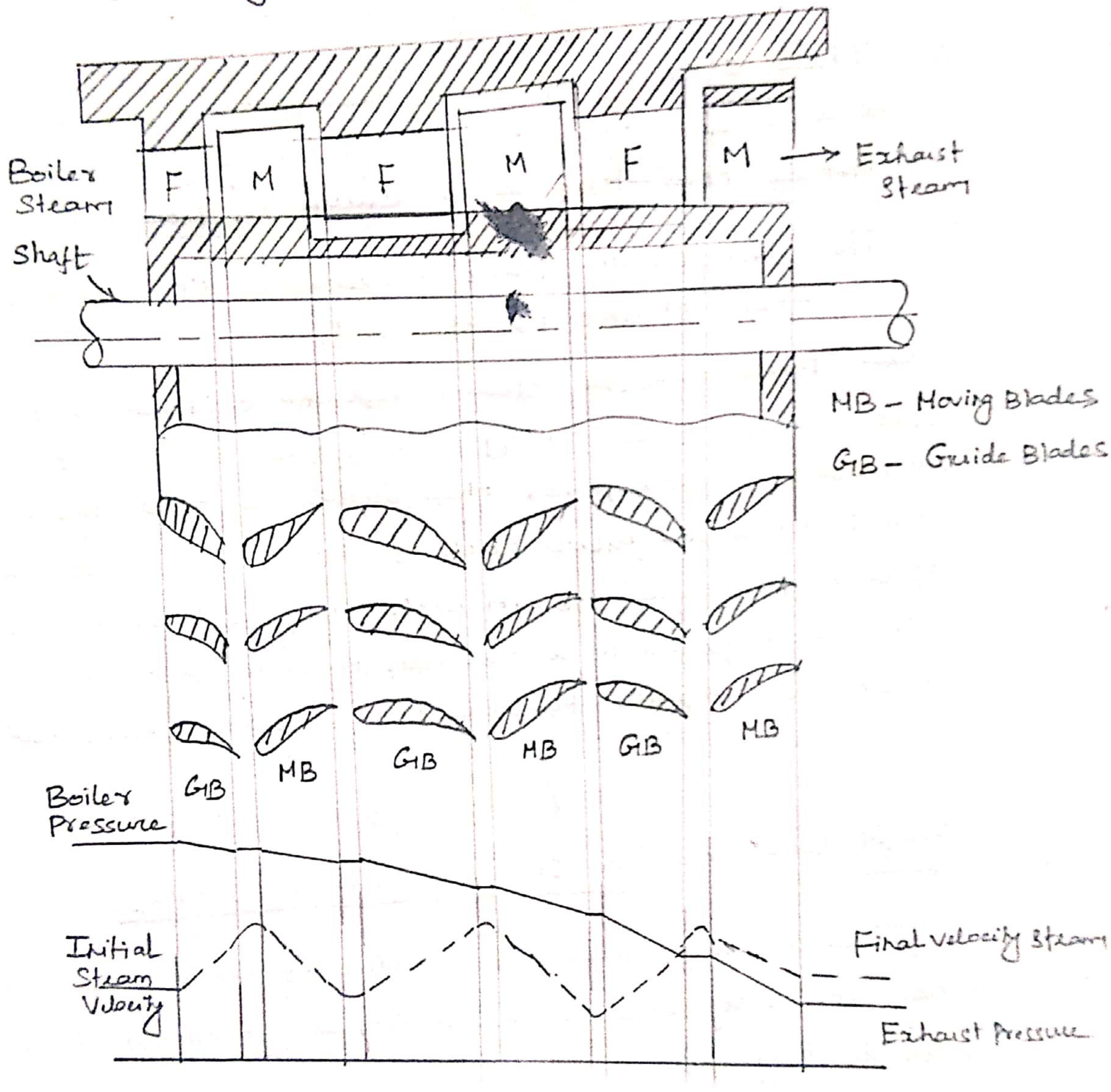
Simple Impulse Turbine



Simple impulse Turbine.

Working of Reaction Turbine

In reaction turbine there is no sudden pressure drop. There is a gradual pressure drop and takes place continuously over the fixed and moving blades. A number of wheels are fixed to the rotating shaft. Fixed guide ways (GB) are provided in between a pair of rotating wheels as shown in figure.



Reaction Turbine

The function of fixed blades (F) is that they guide the steam as well as well allow it to expand at high velocity. It is similar to nozzles as in the case of an impulse turbine.

The moving blades (M) serve the following functions.

1. It converts kinetic energy of a steam into useful mechanical energy.

2. The steam expands while following over the moving blades and thus it gives, reaction to moving blades, Hence the turbine is called reaction turbine.

3. The velocity of steam decreases as the kinetic energy of the steam is absorbed.

The pressure of steam reduces continuously as it follows over the moving blades, the velocity of steam increases. Therefore, the diameter of a reaction turbine must increase after each group of blade rings. Because of less pressure drop in each stage, the number of stages required in a reaction turbine is much greater than an impulse turbine of the same power output.

Velocity Diagram for Impulse Turbine (simple)

The velocity of steam relative to blades, the work done on blades etc can easily be found out the velocity diagrams.

Let c_1 = Absolute velocity of steam entering the moving blade.

c_{f1} = Velocity of flow at entrance of moving blade

c_{r1} = Relative velocity of jet at entrance of moving blade; It is the vertical difference between c_b and c_1

C_b = Linear velocity of moving blade

C_{W1} = Velocity of cohril at the entrance of moving blade

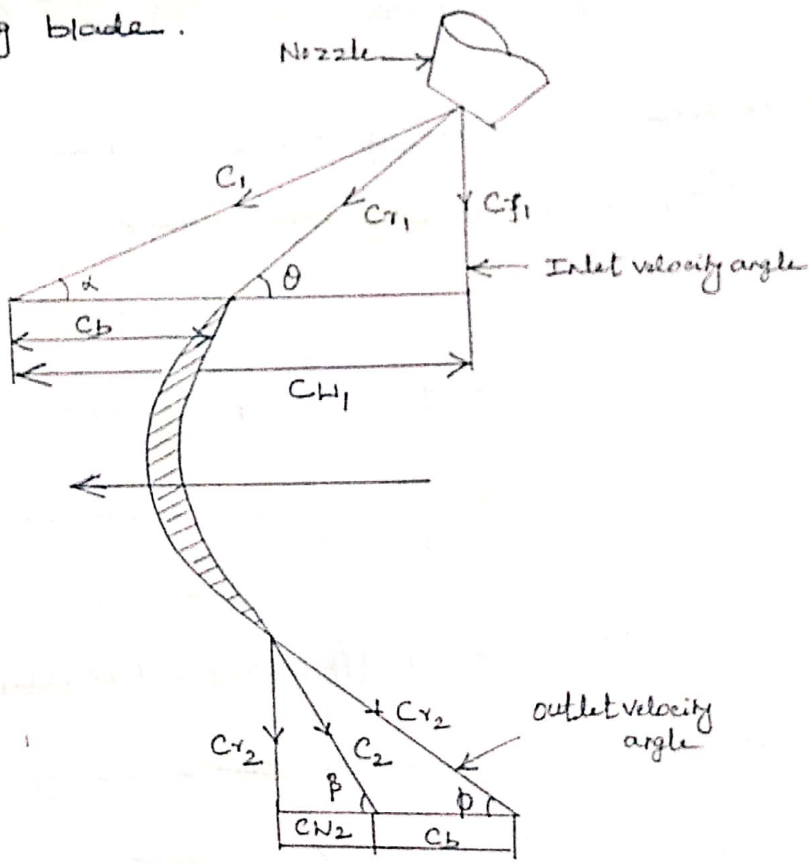
α = Angle with the tangent of the wheel of which the steam with velocity C_1 enters.

θ = Entrance angle of moving blade.

ϕ = Exit angle of moving blade

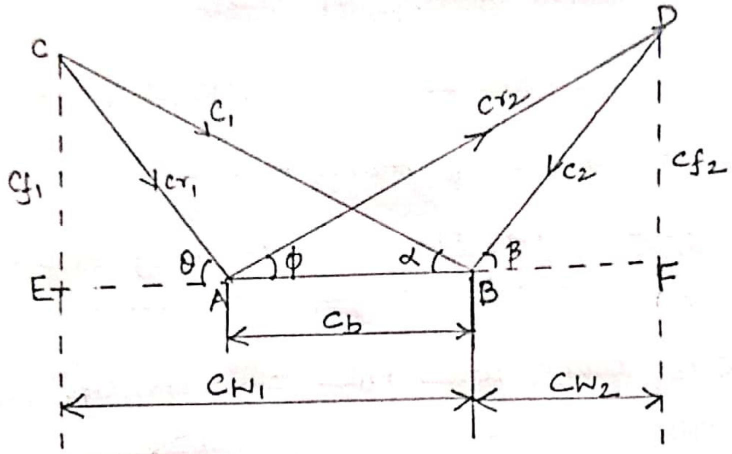
β = Angle which the discharging steam makes with the tangent of the wheel at the exit of moving blade.

C_2, C_{f2}, C_{r2}, C_{W2} are the corresponding values at exit of the moving blade.



Velocity diagram for simple Impulse turbine

Combined velocity diagram.



Work done on blades of Impulse Turbine

Tangential force on the wheel = mass of steam x Acceleration

= mass of steam/s x change of Velocity

Driving force $F_x = m \times (C_{W_1} + C_{W_2})$

$\therefore$ Work done on blade/s = Force x Distance traveled

$= m \times (C_{W_1} + C_{W_2}) \times C_b$

$\therefore$ Power developed/wheel $P = m \times (C_{W_1} + C_{W_2}) \times C_b$

Available energy of the steam entering the blade

$$= \frac{mC_1^2}{2}$$

Blade efficiency $\eta_b = \dfrac{\text{Work done on the blade}}{\text{Energy supplied to the blade}}$

$$= \frac{m(C_{W_1} + C_{W_2})C_b}{\frac{mC_1^2}{2}} = \frac{2C_b(C_{W_1} + C_{W_2})}{C_1^2}$$

Stage efficiency $\eta_{stage} = \dfrac{\text{Workdone on the blade}}{\text{Total energy supplied / stage}}$

$$= \frac{C_b (C_{w_1} + C_{w_2})}{h_1 - h_2}$$

η_{stage} = Blade efficiency × Nozzle efficiency

Axial force on wheel = Mass of steam × Acceleration in axial direction

= Mass of steam/s × change in axial Velocities

Axial Thrust $F_y = m(C_{f_1} - C_{f_2})$

Effect of friction on velocity diagram:

Friction factor $k = \dfrac{C_{r_2}}{C_{r_1}}$

$$AC \neq AD$$
$$C_{r_2} \neq C_{r_1}$$

Heat due to blade friction = Loss of kinetic energy during flow over blades

$$= \frac{m(C_{r_1}^2 - C_{r_2}^2)}{2}$$

Velocity Diagram for reaction Turbine

Tangential force $F_x = m(C_{w_1} + C_{w_2}) C_b$

Workdone per kg of steam $W = m C_b (C_{w_1} + C_{w_2})$

Power Produced by the turbine $P = m(C_{w_1} + C_{w_2}) C_b$

Axial Thrust on the wheel $F_y = m(C_{f_1} - C_{f_2})$

Problem no: 2

The velocity of steam leaving the nozzle of an impulse turbine is 1000 m/s and the nozzle angle is 20°. The blade velocity is 350 m/sec and the blade velocity of co-efficient is 0.85. Assuming no losses due to shock at inlet calculate for a mass flow of 1.5 kg/sec and symmetrical blading (a) blade inlet angle (b) driving force on the wheel (c) axial thrust on the wheel (d) Power developed by the turbine

Given data.

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

$\alpha = 20°$

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

$K = 0.85$

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

For symmetrical blading $\theta = \phi$

Solution

From Δ EBC $\quad C_{w1} = C_1 \cos 20° = 1000 \cos 20° = 939.69 \text{ m/sec}$

$C_{f1} = C_1 \sin 20° = 1000 \sin 20° = 342.02 \text{ m/sec}$

$$\tan\theta = \frac{C_{f1}}{C_{w1} - C_b} = \frac{342.02}{939.68 - 350}$$

$$\boxed{\theta = 30°7'}$$

From Δ EAC

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

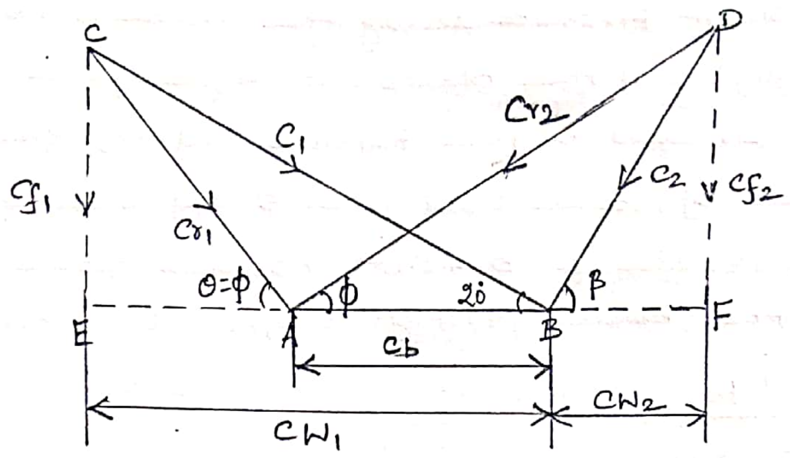
$$= \sqrt{(342.02)^2 + (939.69 - 350)^2}$$

$$= 681.7 \text{ m/sec}$$

But $\quad K = \dfrac{C_{r2}}{C_{r1}} = 0.85$

$$\therefore C_{r2} = 0.85 C_{r1}$$

$$= 0.85 \times 681.7 = 579.45 \text{ m/sec}$$



For symmetrical blading

$$\theta = \phi = 30^\circ 7'$$

From Δ DAF $\quad C_{f2} = C_{r2} \sin\phi = 579.44 \sin 30^\circ 7'$

$$= 290.79 \text{ m/sec.}$$

Illy

$$C_b + C_{w2} = C_{r2}\cos\phi$$

$$350 + C_{w2} = 579.45 \cos 30^\circ 7'$$

$$\therefore C_{w2} = 151.76 \text{ m/sec.}$$

$\therefore$ From Δ BDF $\quad C_2 = \sqrt{C_{f2}^2 + C_{w2}^2} = \sqrt{(290.79)^2 + (151.76)^2}$

$$= 328 \text{ m/sec}$$

Driving force $\quad F_x = m(C_{w1} + C_{w2})$

$$= 1.5\,(939.69 + 151.76)$$

$$= 1637.18 \text{ N}$$

Axial Thrust $\quad F_y = m(C_{f1} - C_{f2})$

$$= 1.5\,(342.02 - 290.79)$$

$$= 76.85 \text{ N}$$

Power developed $\quad P = m\,C_b\,(C_{w1} + C_{w2})$

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

$$= 1637.18 \times 350 = 573.01 \text{ kW}$$

$$\boxed{P = 573.01 \text{ kW}}$$

Problem no: 3.

Saturated steam at 8 bar is supplied to a single stage steam turbine through a convergent divergent steam nozzle. The nozzle angle is 20° and the mean blade speed is 450m/sec. The steam pressure leaving the nozzle is 1 bar. Find the (a) best angle if the blades are equiangular, (b) maximum power developed by the turbine and (c) stage efficiency if the number of nozzles used are 5 and area at the throat of each nozzle is 0.4 cm². Assume nozzle efficiency of 95% and the blade friction co-efficient of 0.85.

Given data.

$P_1 = 8$ bar

$\alpha = 20°$

$C_b = 450$ m/sec

$P_e = 1$ bar

$\theta = \phi$

$n = 5$

Area at throat $A = 0.4\,cm^2 = 4\times10^{-5}\,m^2$

$\eta_N = 95\%$

$\frac{C_{r2}}{C_{r1}} = 0.85$

Sol

From Saturated steam table, Corresponding to $P_1 = 8$ bar, the values of enthalpy and entropy are read.

$h_1 = 2767.4$ kJ/kg $\quad S_1 = 6.66$ kJ/kg K

$\because h_1 = h_g$; $S_1 = S_g$

Corresponding to 1 bar, the values of Parameters

$h_{fe} = 417.5$ kJ/kg $\quad h_{fge} = 2257.9$ kJ/kg

$S_{fe} = 1.303$ kJ/kg K $\quad S_{fge} = 6.057$ kJ/kg K

Since the expansion between inlet and exit of the nozzle is isentropic

$S_i = S_e = 6.66\ kJ/kg\,K$

$S_e = S_{fe} + x_e\, S_{fge}$

$6.66 = 1.303 + x_e \times 6.057$

$x_e = 0.88$

$H_e = h_{fe} + x_e\, h_{ffe}$

$= 417.5 + 0.88 \times 2257.9$

$h_e = 2404.45\ kJ/kg$

Exit velocity of steam from the nozzle

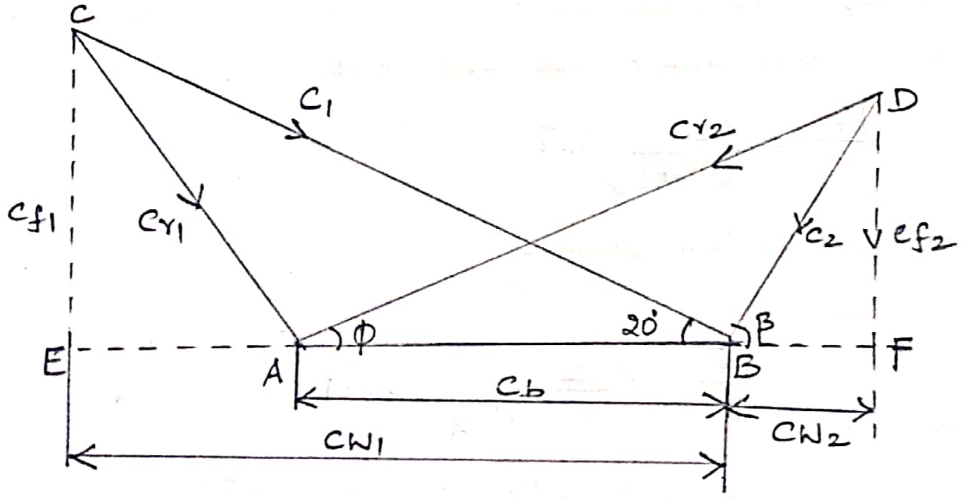
$C_e = \sqrt{2000\,(h_i - h_e) \times \eta}$

$= \sqrt{2000 \times (2767.4 - 2404.45) \times 0.95}$

$C_e = 830.43\ m/sec$

Exit velocity of nozzle is same as the inlet velocity of steam entering the turbine.

$C_1 = C_e = 830.43\ m/sec.$



From Δ EBC $\quad C_{W1} = C_1 \cos\alpha = 830.43 \cos 20°$

$= 780.35\ m/sec$

$C_{f1} = C_1 \sin\alpha = 830.43 \sin 20°$

$= 284.02\ m/sec$

From Δ EAC $\quad \tan\theta = \dfrac{C_{f1}}{C_{W1} - C_b} = \dfrac{284.02}{780.35 - 450}$

$\theta = 40° 41'$

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

$$= \sqrt{(284.02)^2 + (780.35 - 450)^2}$$

$$= 435.66 \text{ m/sec}$$

But $\frac{C_{r2}}{C_{r1}} = 0.85$

$\therefore \quad C_{r2} = 0.85 \times 435.66 = 370.31 \text{ m/sec}$

From Δ AFD

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

$$= 370.31 \sin 40^\circ 11' = 241.56 \text{ m/sec}$$

$$C_b + C_{w2} = C_{r2} \cos\phi$$

$$= 370.31 \cos 40^\circ 11'$$

$$= 282.88 \text{ m/sec}.$$

$\therefore \quad C_{w2} = 282.88 - 450 = -167.12 \text{ m/sec}.$

Both velocity and specific volume at the throat can be found to calculate mass flow rate of steam.

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

For saturated steam

$$n = 1.135$$

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

$$P_t = 0.577 \times 8 = 4.62 \text{ bar}.$$

Corresponding to $P_t = 4.62$ bar from steam table

$h_{f1} = 627.38$ kJ/kg $\qquad h_{fg1} = 2116.7$ kJ/kg

$s_{f1} = 1.8306$ kJ/kg K $\qquad s_{fg1} = 5.015$ kJ/kg K

$v_{f1} = 0.00109$ m^3/kg $\qquad v_{g1} = 0.40364$ m^3/kg

Since the expansion between inlet and throat of the nozzle is Isentropic.

$s_1 = s_t = 6.66\ \text{kJ/kg K}$

$s_1 = s_{ft} + x_t s_{fgt}$

$6.66 = 1.8366 + x_t \times 5.015$

$x_t = 0.96$

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

$= 627.24 + 0.96 \times 2116.7$

$h_t = 2659.41\ \text{kJ/kg}$

Velocity of steam at throat of the nozzle

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

$$= \sqrt{2000(2767.4 - 2659.41) \times 0.95}$$

$$= 452.97\ \text{m/sec.}$$

Specific volume of steam at throat of the nozzle

$$V_t = x_t v_{gt}$$

$$= 0.96 \times 0.40264$$

$$= 0.388\ \text{m}^3/\text{kg}$$

∴ Mass flow rate of steam through the nozzle

$$m = \frac{A_t C_t}{V_t} = \frac{4 \times 10^{-5} \times 452.97}{0.288}$$

$$= 0.047\ \text{kg/sec.}$$

Total Mass of Steam $m = 0.047 \times$ Number of nozzles

$$= 0.047 \times 5$$

$$= 0.235\ \text{kg/sec.}$$

Maximum power developed in the turbine $\Big\}$ $P = m(C_{w1} + C_{w2}) \times C_b$

$$P = 0.235(780.35 - 167.12) \times 450$$

$$= 64.85\ \text{kW}$$

Blade efficiency $\eta_b = \dfrac{2\,(C_{w1} + C_{w2})\,C_b}{C_1^2}$

$= \dfrac{2 \times (780.35 + 167.12) \times 450}{(870.43)^2}$

$= 0.8003 = 80.03\%$

Stage efficiency $\eta_{stage} = \eta_b \times \eta_N$

$= 80.03 \times 0.95$

$= 76.03\%$

— x —

Problem no: 4 (Reaction Turbine)

300 kg/min of steam (2 bar, 0.08 dry) flows through a given stage of reaction turbine. The exit angles of fixed blades as well as moving blades are 20° and 3.68 kW of power is developed. If the rotor speed is 360 rpm and tip leakage is 5 percent, calculate the mean drum diameter and the blade height. The axial flow velocity is 0.8 times the blade velocity. (May 12)

Given data:

$m = 300$ kg/min $= 5$ kg/sec

$P = 2$ bar

$x = 0.08$

$\alpha = \phi = 20°$

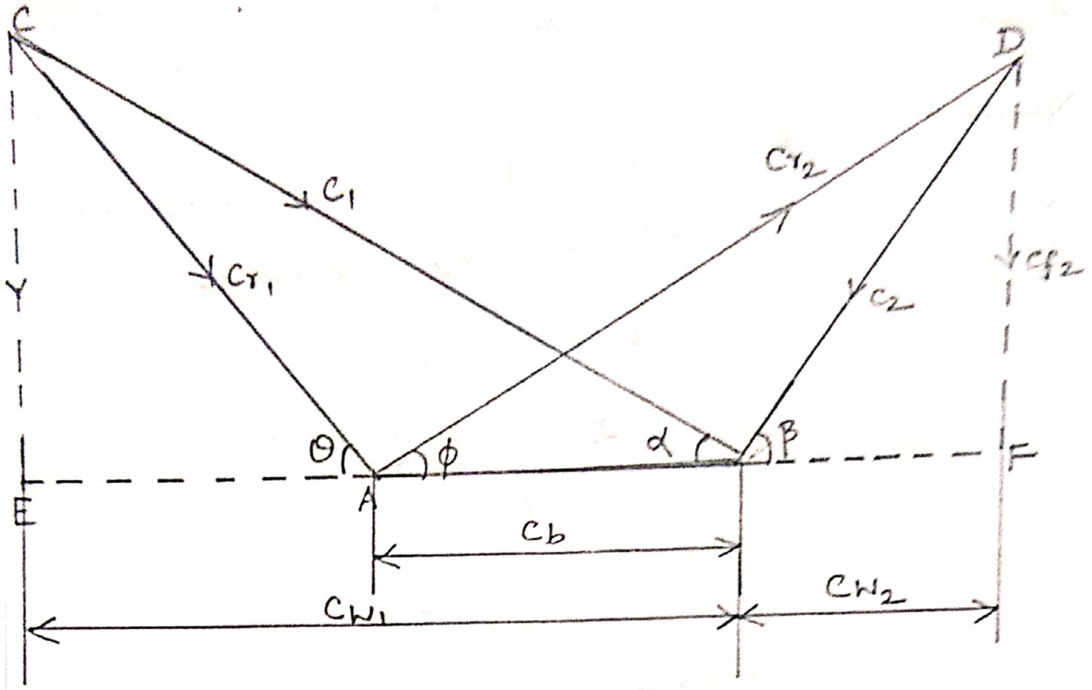
$P = 3.68$ kW $= 3680$ W

$N = 360$ rpm

Tip leakage $= 5\%$

$\dfrac{C_{f1}}{C_b} = \dfrac{C_{f2}}{C_b} = 0.8$

From the velocity diagram by considering ΔEBC



Velocity diagram

$\|^{ly}$ from ΔDAF

$$\tan\phi = \frac{C_{f2}}{AF} = \frac{C_{f2}}{C_b + C_{W_2}}$$

$$C_b + C_{W_2} = \frac{C_{f2}}{\tan\phi} = \frac{C_{f2}}{\tan 20^\circ} = 2.75 C_{f2}$$

$\therefore C_{f2} = C_{f1}$

$[C_{f1} = 0.34 C_1]$

$$C_b + C_{W_2} = 2.75 C_{f1}$$

$$0.43 C_1 + C_{W_2} = 2.75 \times 0.34 C_1$$

$$0.43 C_1 + C_{W_2} = 0.935 C_1$$

$$C_{W_2} = 0.505 C_1$$

Power developed $P = m\,(C_{W_1} + C_{W_2})\,C_b$

$$3680 = 5\,(0.94 C_1 + 0.505 C_1)\,0.43 C_1$$

$$C_1 = 34.42 \text{ m/sec}$$

$$C_b = 0.43 C_1 = 0.43 \times 34.42 = 14.8 \text{ m/sec}$$

$$C_{f1} = C_{f2} = 0.8\, C_b$$

$$= 0.8 \times 14.8 = 11.84 \text{ m/sec}$$

$$C_b = \frac{\pi D_1 N}{60}$$

$$14.8 = \frac{\pi \times D_m \times 360}{60}$$

Mean drum diameter $D_m = 0.79m$.

Actual mass of steam used by considering tip leakage

$$M = 5-(5 \times 0.05) = 4.75 m/sec.$$

From steam table at $P_1 = 2$ bar

$$V_g = V_s = 0.8854 m^3/kg$$

Mass of steam flow /sec

$$M = \frac{\pi D_m h C_{f2}}{x V_s}$$

$$4.75 = \frac{\pi \times 0.79 \times h \times 11.84}{0.8 \times 0.8854}$$

Height of the blade $h = 0.1145 m = 114.5mm$.

Problem no: 5

At a particular stage of reaction turbine, the mean blade speed is 66 m/sec and the steam pressure is 3.5 bar with a temperature of 175°c. The identical fixed and moving blades have inlet angles of 30° and outlet angle of 20°. Determine the

(i) blade height if it is 1/10 of the blade ring diameter for a flow rate of 13.5 kg/sec

(ii) Power developed by pair

(iii) Spefic enthalphy drop if the stage efficiency is 85%.

Given data

Mean

St

Given data

Mean blade speed $C_b = 60 m/sec$

Steam Pressure $= 3.5 bar$

Temperature $= 175°c$

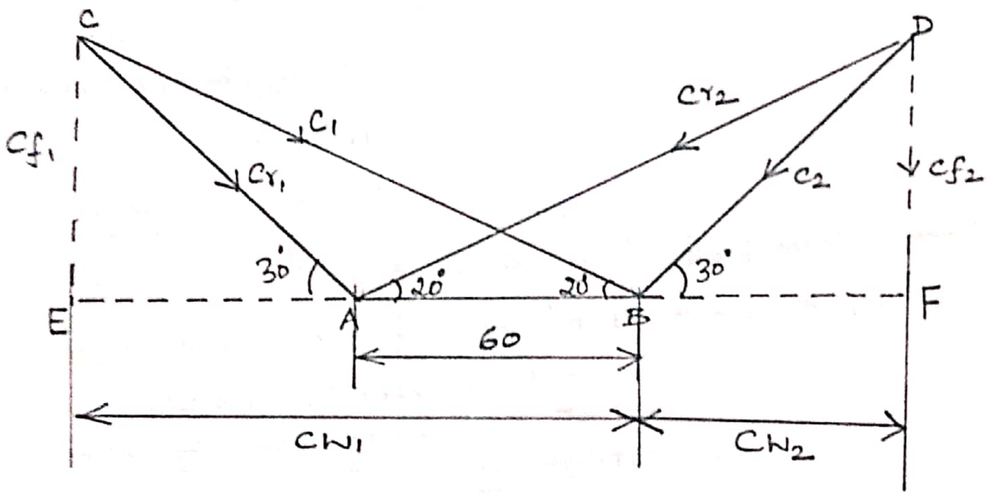
For identical fixed and moving blade $\theta = \beta = 30°$

$\alpha = \phi = 20°$

$m = 13.5 kg/sec$

$h = 1/10 d.$

Solution



According to the sine rule

Δ ABC $\quad \frac{C_1}{\sin 150°} = \frac{C_{r1}}{\sin 20°} = \frac{C_b}{\sin 10°}$

$$C_{r1} = \frac{C_b}{\sin 10°} \times \sin 20°$$

$$= \frac{60}{0.1736} \times 0.3420 = 118.18 m/sec$$

$$C_{f1} = C_{r_1} \sin 30° = 118.18 \times \sin 30°$$

$$= 59.09 m/sec$$

$EA = C_{r1} \cos 30^\circ$

$= 118.18 \times \cos 30^\circ = 102.35$ m/sec

For identical blading $EA = BF = 102.35$ m/sec

$(C_{w1} + C_{w2}) = EA + AB + BF$

$= 102.35 + 60 + 102.35$

$= 264.7$ m/sec.

Pressure of 3.5 bar and 175°C from steam table

$V_s = V_{sup} = 0.73\ m^3/kg$

Mass of steam flow (m) $m = \frac{\pi(d+h)h c_{f1}}{V_s}$

$13.5 = \frac{\pi(10h+h)h \times 59.09}{0.73}$

$13.5 = 2797.27h^2$

$h = 0.06945$ m

$\boxed{h = 69.45m}$

Power developed

By a pair of fixed and moving blade rings

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

$= 13.5 \times 264.7 \times 60 = 214.4$ kW

Stage efficiency

$\eta_{stage} = \frac{m C_b (C_{w1} + C_{w2})}{m(h_1 - h_2)}$

$0.85 = \frac{214.41}{13.5 \times (h_1 - h_2)}$

$h_1 - h_2 = 18.69$ kJ/kg

$\therefore$ Specific enthalpy drop $= 18.69$ kJ/kg.

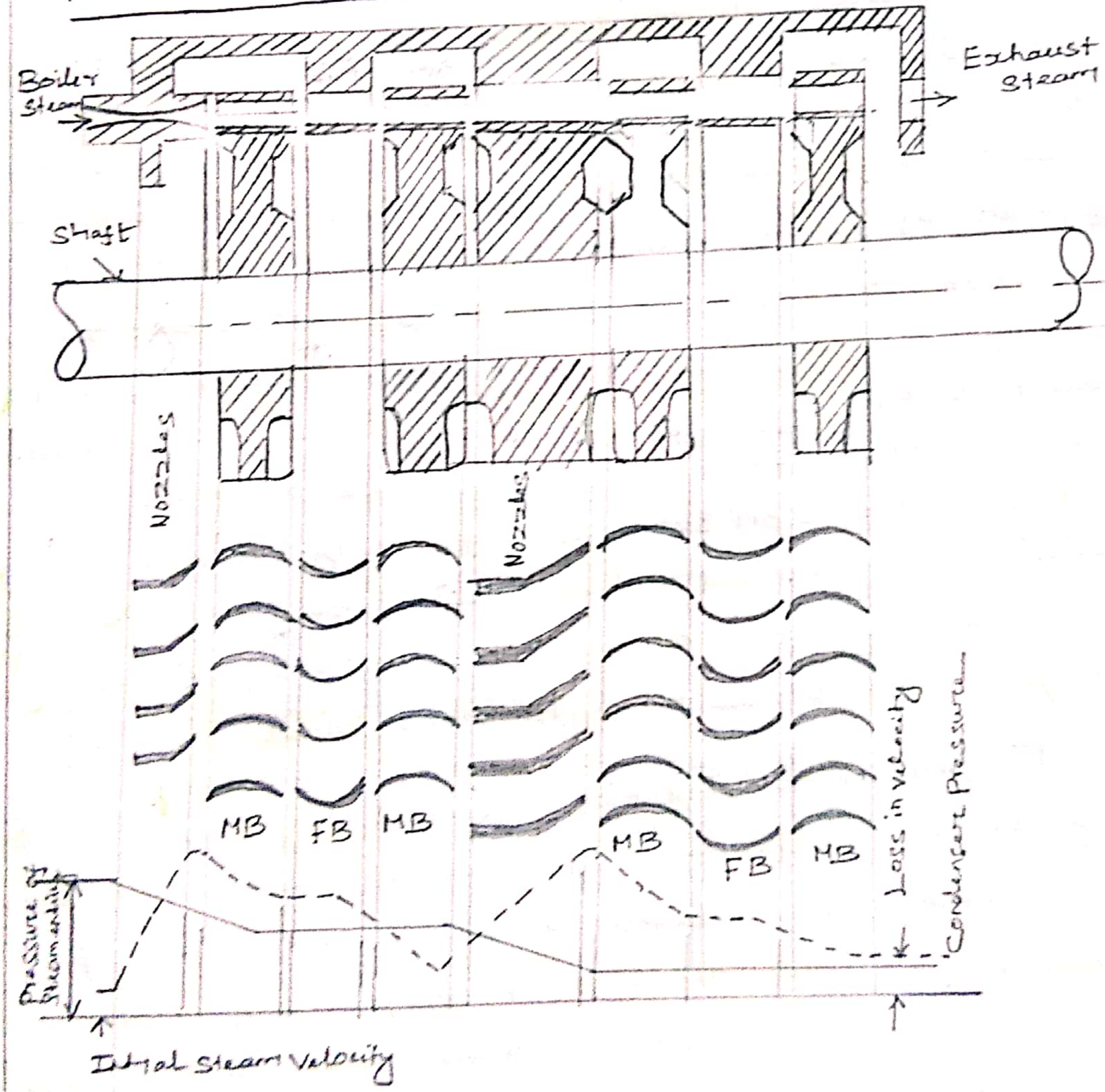
Multistage Turbines:

Compounding is a method of absorbing the jet velocity in more than one stage when the steam flows over moving blades.

The different methods of compounding are given below.

1. Velocity compounding
2. Pressure compounding
3. Pressure - velocity compounding.

Pressure velocity compounding



This method is a combination of pressure and Velocity compounding. The total pressure drop is carried out in two stages and the velocity obtained in each stage is also compounded. Steam pressure from the boiler to the condenser is dropped in stages through convergent divergent nozzles. Velocity compounding is done by using a guide blade rings in between every two moving blade rings.

High pressure steam expands through the first ring nozzles, it does work on the first row of moving blades. (M.B) and then enters guide blades (FB). Though the guide blades, the steam comes out with a change in direction of flow.

Then the steam flows through the second row of moving blades where it does work. The remaining reduction of pressure up to condenser pressure takes place in the second set of nozzles and the process of doing work is on two sets of moving blade and a guide blade is continued. Thus, the total pressure drop is obtained in stages through nozzle sets and the velocity changes takes place through moving blades.

Turbine employing this method may be to combine many of the advantages of both pressure and velocity compounding. By allowing a large pressure drop in each stage, less number of stages are used and hence a shorter turbine will be obtained for a given pressure drop.

Losses in Steam Turbines:

1. Losses in regulating Valves.
2. Losses due to Steam friction.
3. Losses due to mechanical friction.
4. Losses due to leakage
5. Residual velocity losses
6. Carryover losses.
7. Losses due to Wetness of steam
8. Losses due to radiation.

Stage efficiency

Stage efficiency is defined as the ratio of actual heat drop to the isentropic heat drop. It is given by

$$\eta_s = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

$$\eta_s = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

Reheat factor

$$RF = \frac{\text{cumulative heat drop}}{\text{Isentropic (or) Rankine heat drop}}$$

$$RF = \frac{(h_1 - h_{2s}) + (h_2 - h_{3s}) + (h_3 - h_{4s}) + (h_4 - h_{5s}) + (h_5 - h_{6s})}{h_1 - h_{7s}}$$

$$RF = \frac{\Delta H_c}{\Delta H_{isen}}$$

$$\eta_{overall} = \eta_{stage} \times RF$$

Problem no: 6.

The following particulars refer to a two-row velocity compounded impulse wheel which forms the first stage of a combination turbine.

Stage velocity at nozzle outlet = 630m/sec

Blade Velocity = 125m/sec

Nozzle angle = 16°

outlet angle, first row of moving blades = 18°

outlet angle, fixed guide blades = 22°

outlet angle, second row of moving blades = 36°

Steam flow rate = 2.6 kg/sec.

The ratio of the relative velocity at outlet to that at inlet is 0.85 for all blades. Determine (a) velocity of whirl (b) tangential thrust on the blades, (c) axial thrust on the blades, (d) power developed (e) blading efficiency.

Given data:

$c_1 = 630$ m/sec

$C_b = 125$ m/sec

$\alpha_1 = 16^\circ$

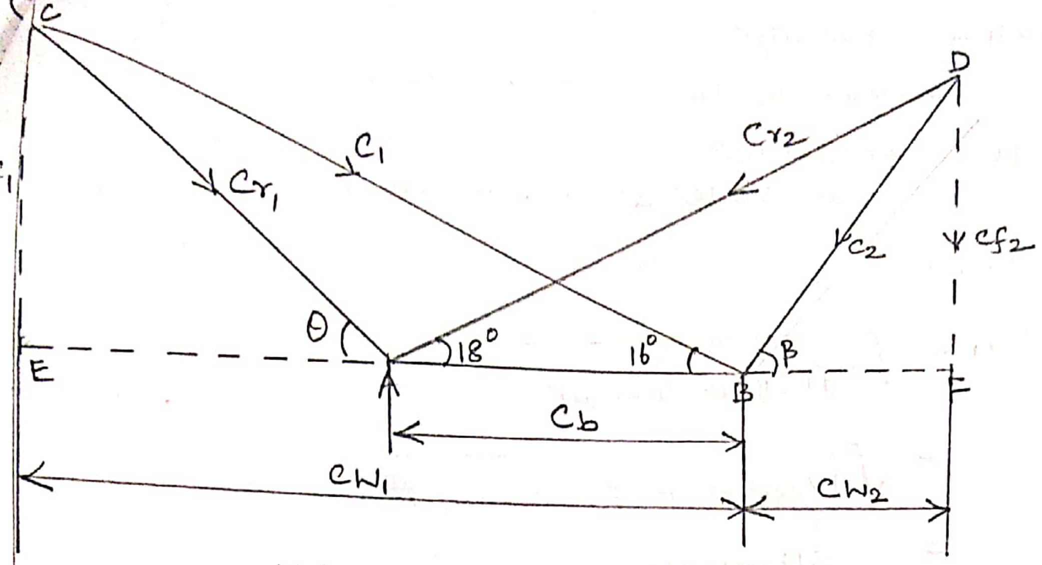
$\phi_1 = 18^\circ$

$\alpha_2 = 22^\circ$

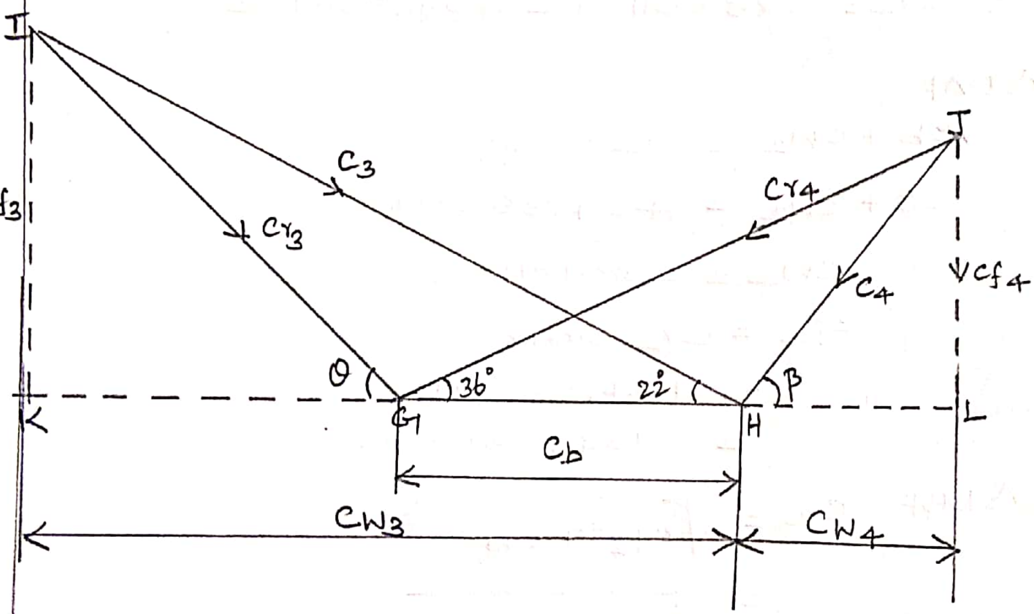
$\phi_2 = 36^\circ$

$m = 2.6$ kg/sec

$$\frac{c_{r2}}{c_{r1}} = \frac{c_3}{c_2} = \frac{c_{r4}}{c_{r3}} = 0.85$$



Velocity diagram for the first stage.



Velocity diagram for the second stage

From Δ EBC

$$C_{w1} = C_1 \cos 16^\circ$$

$$= 630 \cos 16^\circ = 605.6 \text{ m/sec}$$

$$C_{f1} = C_1 \sin 16^\circ$$

$$= 630 \sin 16^\circ = 173.65 \text{ m/sec}$$

From Δ ACE

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

$$= \sqrt{173.65^2 + (605.6 - 125)^2}$$

$$= 511.01 \text{ m/sec}.$$

But $\dfrac{C_{r2}}{C_{r1}} = 0.85$

$\therefore$ $C_{r2} = 0.85 \times 511.01 = 434.36 \text{ m/sec}$

From Δ DAF

$$C_b + C_{w2} = C_{r2} \cos 18^\circ$$

$$125 + C_{w2} = 434.36 \cos 18^\circ$$

$$C_{w2} = 288.1 \text{ m/sec}$$

$$C_{f2} = C_{r2} \sin 18^\circ$$

$$= 434.36 \sin 18^\circ$$

$$= 134.23 \text{ m/sec}$$

From Δ DBF $\quad C_2 = \sqrt{C_{f2}^2 + C_N^2}$

$$= \sqrt{134.23^2 + 288.1^2}$$

$$= 317.84 \text{ m/sec}$$

But $\dfrac{C_3}{C_2} = 0.85$

$$C_3 = 0.85 \times 317.84 = 270.16 \text{ m/sec}$$

From Δ HIK

$$C_{w3} = C_3 \cos 22^\circ$$

$$= 270.16 \cos 22^\circ$$

$$= 250.49 \text{ m/sec}$$

$$C_{f3} = C_3 \sin 22^\circ = 270.16 \sin 22^\circ$$

$$= 101.2 \text{ m/sec}.$$

$$\therefore C_{r3} = \sqrt{C_{f3}^2 + (C_{w3} - C_b)^2}$$

$$= \sqrt{(101.2)^2 + (250.49 - 125)^2}$$

$$= 161.21 \text{ m/sec}.$$

$$\frac{C_{r4}}{C_{r3}} = 0.85$$

$$\therefore C_{r4} = 0.85 \times 161.21 = 137.03 \text{ m/sec}$$

From ΔGJL

$$C_b + C_{w4} = C_{r4} \cos 36^\circ$$

$$125 + C_{w4} = 137.03 \cos 36^\circ$$

$$\therefore C_{w4} = 137.03 \cos 36^\circ - 125$$

$$= -14.14 \text{ m/sec}$$

$$C_{f4} = C_{r4} \sin 36^\circ$$

$$= 137.03 \sin 36^\circ$$

$$= 80.54 \text{ m/sec}.$$

Total whirl Velocity $C_w = \left[C_{w1} + C_{w2}\right]_{Stage 1} + \left[C_{w3} + C_{w4}\right]_{Stage 2}$

$$= 605.6 + 288.1 + 250.49 - 14.14$$

$$= 1130.05 \text{ m/sec}$$

Tangential thrust $F_x = m\left[(C_{w1} + C_{w2})_{Stage 1} + (C_{w3} + C_{w4})_{Stage 2}\right]$

$$= 2.6 \times 1130.05$$

$$= 2938.13 \text{ N or } 2.94 \text{ kN}$$

Axial Thrust $F_y = m\left[(C_{f1} - C_{f2})_{Stage 1} + (C_{f3} - C_{f4})_{Stage 2}\right]$

$$= 2.6\left[(173.65 - 134.23) + (101.2 - 80.54)\right]$$

$$= 156.21 \text{ N}$$

power developed $P = m\left[(C_{w1}+C_{w2})_{stage1} + (C_{w3}+C_{w4})_{stage2}\right]U$

$= 2938.13 \times 125$

$= 367266.25\ W$

$= 367.27\ kW$

Total energy supplied

$= (\text{Energy supplied})_{stage1} + (\text{Energy supplied})_{stage2}$

$= m\left[\frac{1}{2}C_1^2 + \frac{1}{2}C_3^2\right]$

$= 2.6\left[\frac{1}{2}630^2 + \frac{1}{2}270.16^2\right]$

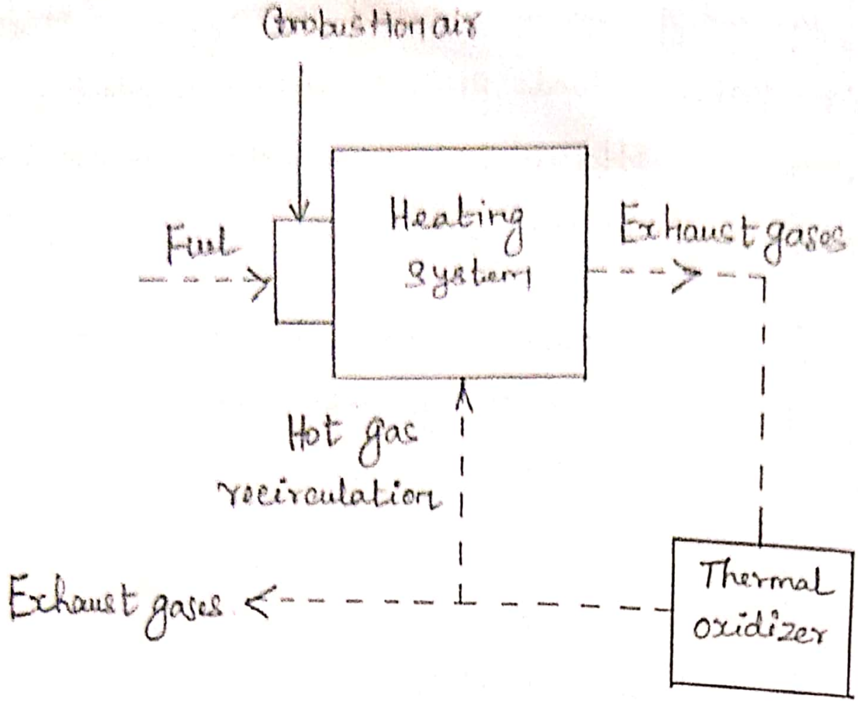
$= 610.85\ kW$

∴ Blading efficiency

$\eta_b = \frac{\text{power developed}}{\text{Energy supplied}}$

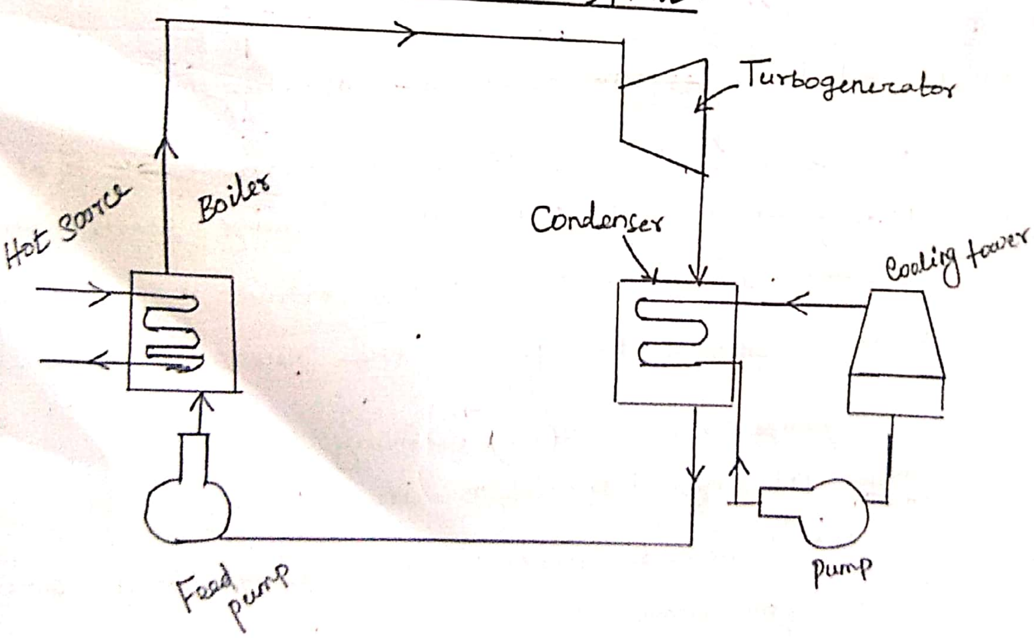
$= \frac{367.27}{610.85}$

$= 0.6012 = 60.12\%$



Regenerative Thermal oxidizers (RTO) stack is great opportunity for the manufacturers to capture waste heat. The waste streams to be destroyed by thermal oxidation are maintained for a set period of time at an elevated temperature in an oxygen-rich environment to achieve the desired destruction efficiency.

<u>Organic Rankine cycle (ORC) plant.</u>

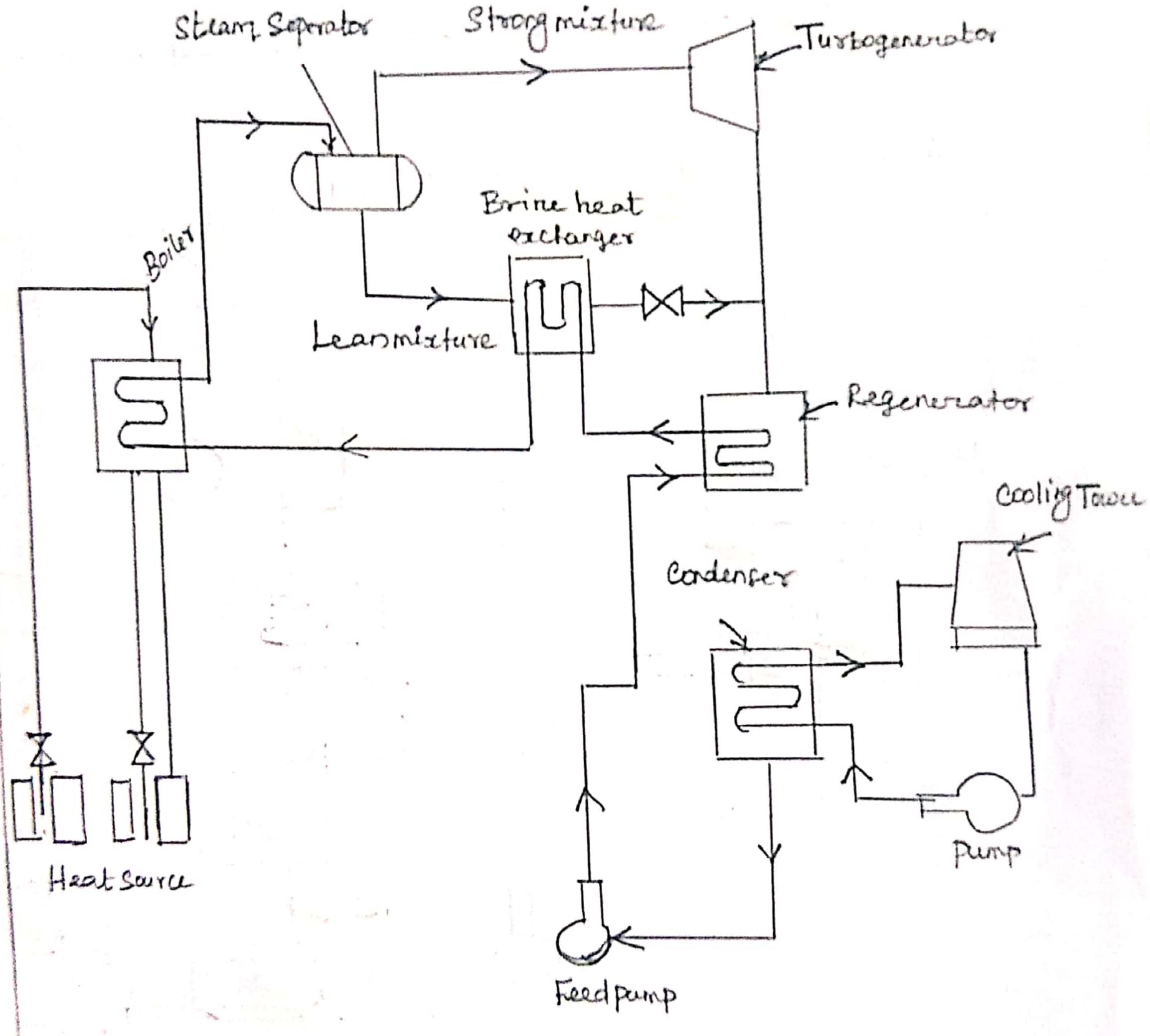


other working fluids with better efficiencies at lower heat source temperatures are used in ORC heat engines.

ORC plants use an organic working fluid that has low boiling point, high vapour pressure, high molecular mass and high mass flow compared to water.

The working mediums are varieties of organic liquid such as Freon, butane, propane, ammonia and new environmentally-friendly refrigerants.

Ammonia - water system



Kalina cycle

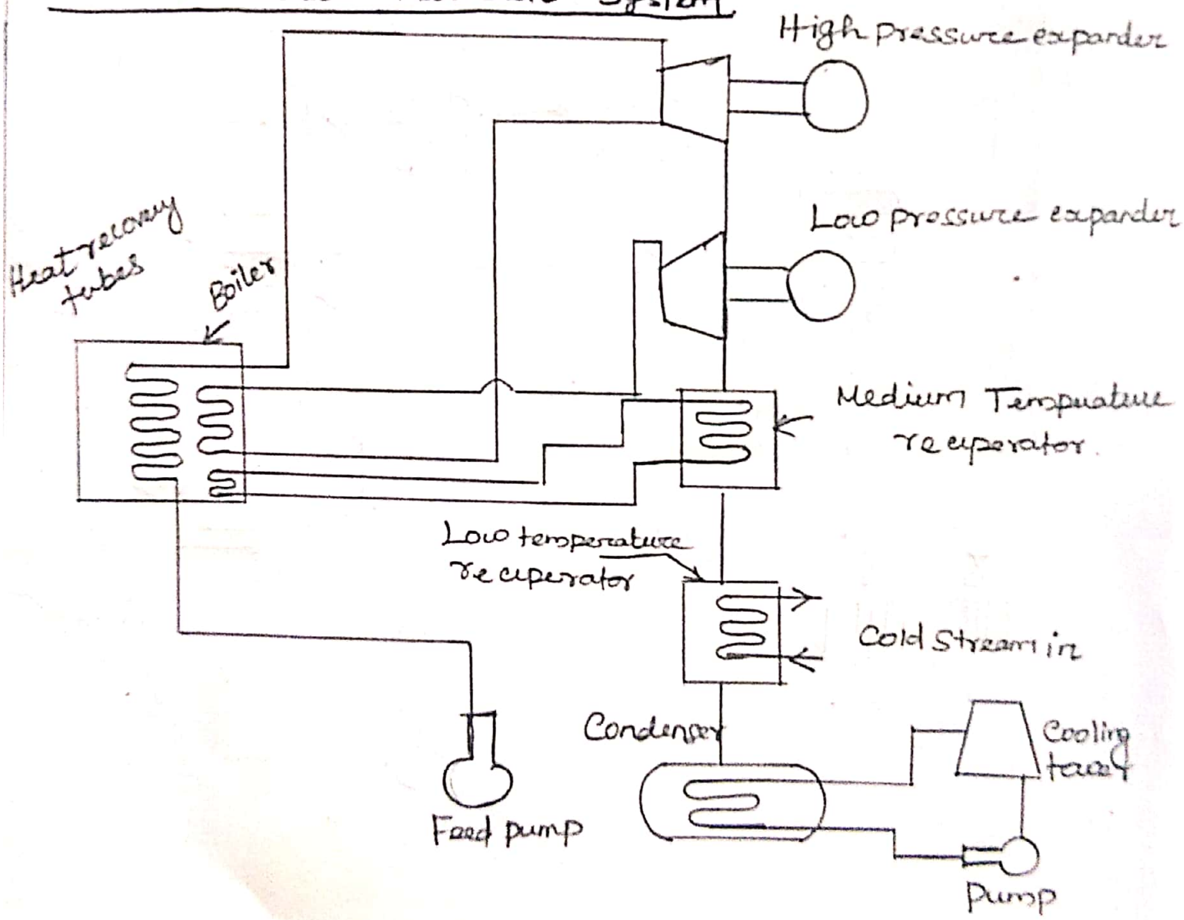
Kalina cycle is another type of Rankine cycle using a mixture of water and ammonia as the working fluid which provides a more efficient energy extraction from the heat source.

The working medium is ammonia - water vapour. It's operating temperature ranging from 200°F to 1000°F, and it is 15% to 25% more efficient than ORC at the same temperature level.

Applications:

Geo-thermal and other non-heavy industrial areas.

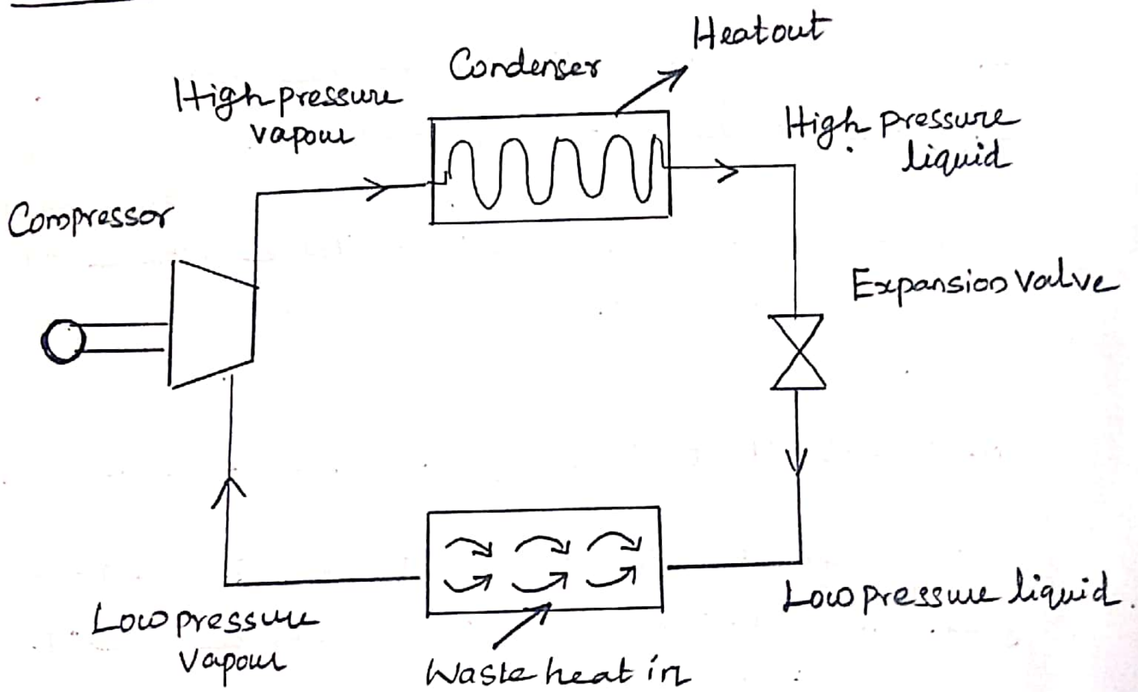
Ammonia - water "Neo-Gen" system



In this system, the working medium is ammonia — water vapour. Neo-Gen steam cycle provides 15% to 20% more efficient than ORC cycle with much configuration compared to kalina system.

Operating temperature range is 250°F to 800°F. It needs relatively moderate cost per kW capacity and estimated power cost in the range of 0.07 to 0.10 per kWh for 250 kW plant. This system requires less number of heat exchangers as compared to kalina system.

Heat pump:



Heat pump is a device which is used to suppy heat to hot system. In otherwords, it is used to maintain the temperature of the body higher than the surroundings. Ex: Room heater.

The waste heat is transferred from a hot fluid to a fluid at a low temperature. Heat must flow spontaneously downhill. It always flow from high temperature system to low temperature system. when energy is repeatedly transferred or transferred it becomes less and less available for the use. The heat pump was developed as a space heating system where low temperature energy from the ambient air.

The heat is extracted from the heat source to boil the circulating substance.

1. The compressor compresses the circulating substance. During compression, both pressure and temperature increase. The low temperature vapour is compressed by a compressor which requires external work. The work done on the vapour raises its pressure and temperature levels where its energy becomes available for the use.

2. The heat is delivered to the condenser.

3. The pressure of the circulating substance (working fluid) is reduced back to the evaporator condition in the throttling valve where the cycle is repeated.

Residual Heat Recovery Equipment and Systems.

1. High to ultra-high temperature (650° to 870°c)
2. Low to medium temperature (120° to 650°c)
3. Low temperature (below 120°c)