STEAM NOZZLES

TECHNICAL TERMS:

1. Wet steam: The steam, which contains some water particles in superposition.

2. Dry steam / dry saturated steam:

When whole mass of steam is converted into steam then it is called as dry steam.

3. Super heated steam: When the dry steam is further heated at constant pressure, the temperature increases the above saturation temperature. The steam has obtained is called super heated steam.

4. Degree of super heat: The difference between the temperature of saturated steam and saturated temperature is called degree of superheat.

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

6. Coefficient of nozzle: It is the ratio of actual enthalpy drop to isentropic enthalpy drop.

7. Critical pressure ratio: There is only one value of ratio (P2/P1) which produces maximum discharge from the nozzle . then the ratio is called critical pressure ratio.

8. Degree of reaction: It is defined as the ratio of isentropic heat drop in the moving blade to isentropic heat drop in the entire stages of the reaction turbine.

9. Compounding: It is the method of absorbing the jet velocity in stages when the steam flows over moving blades. (i)Velocity compounding (ii) Pressure compounding and (iii) Velocity-pressure compounding

10. Enthalpy: It is the combination of the internal energy and the flow energy.

11. Entropy: It is the function of quantity of heat with respective to the temperature.

12. Convergent nozzle: The crossectional area of the duct decreases from inlet to the outlet side then it is called as convergent nozzle.

13. Divergent nozzle: The crossectional area of the duct increases from inlet to the outlet then it is called as divergent nozzle.

Flow of steam through nozzles:

The flow of steam through nozzles may be regarded as adiabatic expansion. - The steam has a very high velocity at the end of the expansion, and the enthalpy decreases as expansion takes place. - Friction exists between the steam and the sides of the nozzle; heat is produced as the result of the resistance to the flow. - The phenomenon of super saturation occurs in the flow of steam through nozzles. This is due to the time lag in the condensation of the steam during the expansion.

Continuity and steady flow energy equations

Through a certain section of the nozzle: m.v = A.C m is the mass flow rate, v is the specific volume, A is the cross-sectional area and C is the velocity. For steady flow of steam through a certain apparatus, principle of conservation of energy states:

$$h_1 + C_1^2/2 + gz_1 + q = h_2 + C_2^2/2 + gz_2 + w$$

For nozzles, changes in potential energies are negligible, w = 0 and q ≅ 0

$$h_1 + C_1^2/2 = h_2 + C_2^2/2$$

Types of Nozzles:

1. Convergent Nozzle
2. Divergent Nozzle
3. Convergent-Divergent Nozzle

Convergent Nozzle:

A typical convergent nozzle is shown in fig. in a convergent nozzle, the cross sectional area decreases continuously from its entrance to exit. It is used in a case where the back pressure is equal to or greater than the critical pressure ratio.

Divergent Nozzle:

The cross sectional area of divergent nozzle increases continuously from its entrance to exit. It is used in a case, where the back pressure is less than the critical pressure ratio.

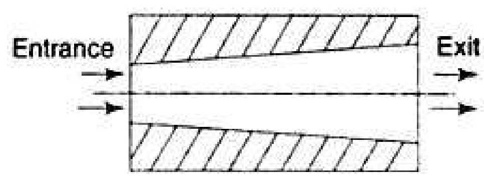


Fig. 3.1. Divergent Nozzle

Convergent-Divergent Nozzle:

In this case, the cross sectional area first decreases from its entrance to throat, and then increases from throat to exit.it is widely used in many type of steam turbines.

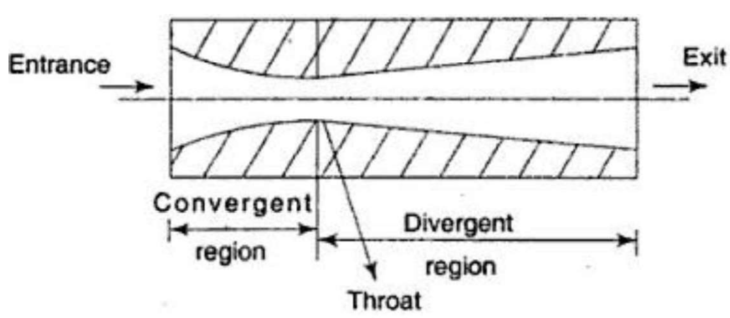
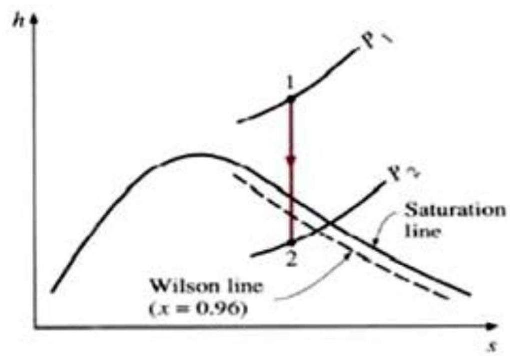


Fig. 3.2. Convergent-Divergent Nozzle

Supersaturated flow or Meta stable flow in Nozzles: As steam expands in the nozzle, its pressure and temperature drop, and it is expected that the steam start condensing when it strikes the saturation line. But this is not always the case. Owing to the high velocities, the residence time of the steam in the nozzle is small, and there may not sufficient time for the necessary heat transfer and the formation of liquid droplets. Consequently, the condensation of steam is delayed for a little while. This phenomenon is known as super saturation, and the steam that exists in the wet region without containing any liquid is known as supersaturated steam.

The locus of points where condensation will take place regardless of the initial temperature and pressure at the nozzle entrance is called the Wilson line. The Wilson line lies between 4 and 5 percent moisture curves in the saturation region on the h-s diagram for steam, and is often approximated by the 4 percent moisture line. The super saturation phenomenon is shown on the h-s chart below:



The *h-s* diagram for the isentropic expansion of steam in a nozzle.

Critical Pressure Ratio: The critical pressure ratio is the pressure ratio which will accelerate the flow to a velocity equal to the local velocity of sound in the fluid.

Critical flow nozzles are also called **sonic chokes**. By establishing a shock wave the sonic choke establish a fixed flow rate unaffected by the differential pressure, any fluctuations or changes in downstream pressure. A sonic choke may provide a simple way to regulate a gas flow.

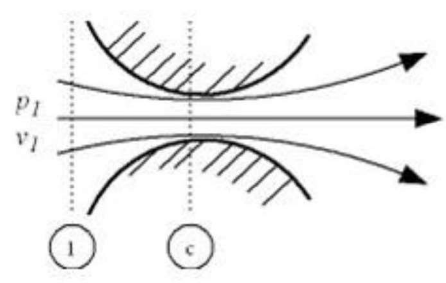


Fig. 3.3 ritical flow nozzles

The ratio between the critical pressure and the initial pressure for a nozzle can expressed as

$$P_c / p_1 = (2 / (n + 1))^{n/(n-1)}$$

Where, p_c = critical pressure (Pa)

p_1 = inlet pressure (Pa)

n = index of isentropic expansion or compression or polytrophic constant

For a perfect gas undergoing an adiabatic process the index – n – is the ratio of specific heats k = cp / cv. There is no unique value for – n. Values for some common gases are

- Steam where most of the process occurs in the wet region: n = 1.135
- Steam super-heated: n = 1.30
- Air: n = 1.4
- Methane: n = 1.31
- Helium: n = 1.667

Effect of Friction on Nozzles:

1) Entropy is increased.
2) Available energy is decreased.
3) Velocity of flow at throat is decreased.
4) Volume of flowing steam is decreased.
5) Throat area necessary to discharge a given mass of steam is increased.

Most of the friction occurs in the diverging part of a convergent-divergent nozzle as the length of the converging part is very small. The effect of friction is to reduce the available enthalpy drop by about 10 to 15%. The velocity of steam will be then

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

Where, k is the co-efficient which allows for friction loss. It is also known as nozzle efficiency.

Velocity of Steam at Nozzle Exit:

$$V_2^2 = 2000(H_1 - H_2) + V_1^2 \quad \therefore \quad V_2 = \sqrt{2000(H_1 - H_2) + V_1^2}$$

As the velocity of steam entering the nozzle is very small, V_1 can be neglected.

$$\therefore \qquad V_2 = \sqrt{2000(H_1 - H_2)} = 44.72\sqrt{(H_1 - H_2)} \text{ m/s}$$

If frictional losses are taken into account then

$$V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n} \text{ m/s}$$

Mass of steam discharged through nozzle:

$$m = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

Condition for maximum discharge through nozzle: The nozzle is always designed for maximum discharge

$$\frac{m}{A} = \sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

The mass flow per unit area will be maximum at the throat because the throat area is minimum.

It is seen from the above equation that the discharge through a nozzle is a function of $\frac{P_2}{P_1}$ only, as the expansion index is fixed according to the steam supplied to the nozzle.

Therefore, $\frac{m}{A}$ is maximum when

$$\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right] \text{ is minimum}$$

Values for maximum discharge:

$$m = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

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

Putting the value of $\frac{P_2}{P_1}$ in the above equation

$$m_{max} = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}-\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

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$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}}-1\right]}$$

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$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\frac{n+1}{2}-1\right]}$$

$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left(\frac{n-1}{2}\right)}$$

$$m_{max} = A\sqrt{1000n\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}$$

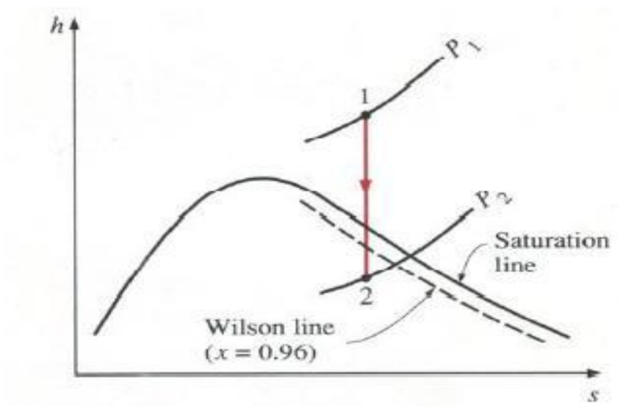
Where P_1 is the initial pressure of the steam in kpa and v_1 is the specific volume of the steam in m3/kg at the initial pressure.

STEAM NOZZLE PROBLEMS

1. Describe about supersaturated flow or metastable flow in a nozzleand state effect of super saturation

As steam expands in the nozzle, its pressure and temperature drop, and it is expected that the Steam start condensing when it strikes the saturation line. But this is not always the case. Owing to the high velocities, the residence time of the steam in the nozzle is small, and there may not Sufficient time for the necessary heat transfer and the formation of liquid droplets. Consequently, the condensation of steam is delayed for a little while. This phenomenon is known as Super saturation, and the steam that exists in the wet region without containing any liquid is Known as supersaturated steam.

The locus of points where condensation will take place regardless of the initial temperature and Pressure at the nozzle entrance is called the Wilson line. The Wilson line lies between 4 and 5 percent moisture curves in the saturation region on the h-s diagram for steam, and is often Approximated by the 4 percent moisture line. The super saturation phenomenon is shown on the h-s chart below:



The h-s diagram for the isentropic expansion of steam in a nozzle.

Effects of Supersaturation:

The following are the effects of supersaturation in a nozzle.

(a) The temperature at which the supersaturation occurs will be less than the saturation temperature corresponding to that pressure. Therefore, the density of supersaturated steam will be more than that of equilibrium condition which gives the increase in the mass of steam discharged.

(b) Supersaturation increases the specific volume and entropy of the steam.

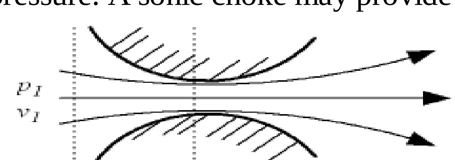
(c) Supersaturation reduces the heat drop. Thus the exit velocity of steam is reduced.

(d) Supersaturation increases the dryness fraction of the steam.

Critical Pressure Ratio:

The critical pressure ratio is the pressure ratio which will accelerate the flow to a velocity equal to the local velocity of sound in the fluid.

Critical flow nozzles are also called **sonic chokes**. By establishing a shock wave the sonic choke establish a fixed flow rate unaffected by the differential pressure, any fluctuations or changes in downstream pressure. A sonic choke may provide a simple way to regulate a gas flow.



The ratio between the critical pressure and the initial pressure for a nozzle can expressed as

$p_c / p_1 = (2 / (n + 1))^{n / (n - 1)}$

where

p_c = critical pressure (Pa)

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n = index of isentropic expansion or compression - or polytropic constant

For a perfect gas undergoing an adiabatic process the index - n - is the ratio of specific heats - k = c_p / c_v. There is no unique value for - n. Values for some common gases are

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2. Describe the Effect of Friction on Nozzles.

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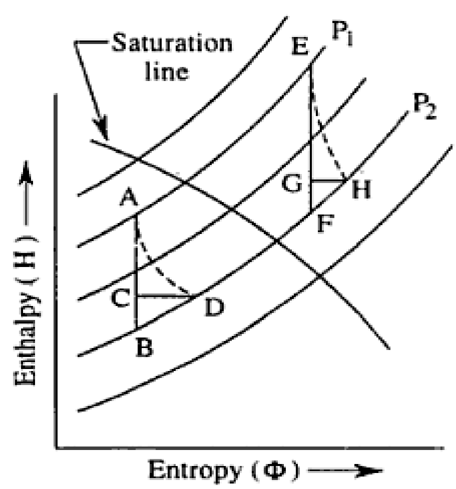
Most of the friction occurs in the diverging part of a convergent-divergent nozzle as the length of the converging part is very small. The effect of friction is to reduce the available enthalpy drop by about 10 to 15 per cent. The velocity of steam will be then $V_2 = 44.72\sqrt{K(H_1 - H_2)}$ where K is the coefficient which allows for friction loss. It is also known as nozzle efficiency (η_n)

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



Velocity of steam at nozzle exit:

$$V_2^2 = 2000(H_1 - H_2) + V_1^2 \quad \therefore \quad V_2 = \sqrt{2000(H_1 - H_2) + V_1^2}$$

As the velocity of steam entering the nozzle is very small, V_1 can be neglected.

$$\therefore \qquad V_2 = \sqrt{2000(H_1 - H_2)} = 44.72\sqrt{(H_1 - H_2)} \text{ m/s}$$

If frictional losses are taken into account then

$$V_2 = 44.72\sqrt{(H_1 - H_2)\eta_n} \text{ m/s}$$

3. Derive the expression for maximum discharge through a nozzle.

Mass of steam discharged through a nozzle:

$$m = A\sqrt{2000\frac{n}{n-1} \times \frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

Condition for maximum discharge through nozzle:

The nozzle is always designed for maximum discharge

$$\frac{m}{A} = \sqrt{2000\frac{n}{n-1} \times \frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

The mass flow per unit area will be maximum at the throat because the throat area is minimum.

It is seen from the above equation that the discharge through a nozzle is a function of $\frac{P_2}{P_1}$ only, as the expansion index is fixed according to the steam supplied to the nozzle.

Therefore, $\frac{m}{A}$ is maximum when

$$\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right] \text{ is minimum}$$

Values for maximum discharge:

$$m = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

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

Putting the value of $\frac{P_2}{P_1}$ in the above equation

$$m_{\max} = A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}-\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

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$$= A\sqrt{2000\frac{n}{n-1}\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left[\left(\frac{2}{n+1}\right)^{\frac{1-n}{n-1}}-1\right]}$$

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$$m_{\max} = A\sqrt{1000n\times\frac{P_1}{v_1}\times\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}$$

W3here P_1 is the initial pressure of the steam in kpa and v_1 is the specific volume of the stea

4. ***Dry saturated steam at a pressure of 15 bar enters in a nozzle and is discharged at a pressure of 1.5 bar. Find the final velocity of the steam. When the initial velocity of the steam is negligible.***

Given :

p_1 = 15 bar;

p_2 = 1.5 bar

Solution.

Final velocity of the steam

From steam tables, corresponding to a pressure of 15 bar, we find that enthalpy of dry saturated steam,

h_1 = 2789.9 kJ/Kg

and corresponding to a pressure of 1.5 bar, enthalpy of dry saturated steam,

h_2 = 2693.4 kJ/Kg

$\therefore$ Heat drop, $h_d = h_1 - h_2 = 2789.9 – 2693.4 = 96.5$ kJ/Kg

We know that final velocity of the steam,

$V_2 = 44.72\sqrt{h_d} = 44.72\sqrt{96.5} = 439.3 m/s$ Ans.

Percentage reduction in the final velocity

We know that heat drop lost in friction

= 10% = 01.

$\therefore$ Nozzle coefficient or nozzle efficiency

K = 1 – 01 = 0.9

We know that final velocity of the steam,

$V_2 = 44.72\sqrt{Kh_d} = 44.72\sqrt{0.9 \times 96.5} = 416.8 m/s$

$\therefore$ Percentage reduction in final velocity

$$= \frac{439.3 - 416.8}{439.3} = 0.051 \; or \; 5.1\%$$

5. 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, and 2. Initial velocity of the steam is 135 m/s

Given :

$p_1 = 10$ bar ;

$p_2 = 0.1$ bar

Solution .

Dryness fraction of the steam at exit

Let x_2 = Dryness fraction of the steam at exit.

From steam tables, corresponding to a pressure of 10 bar, we find that entropy of dry saturated steam,

$s_1 = s_{g_1} = 6.583 kJ/kgK$ s

and corresponding to a pressure of 0.1 bar,

we find that

$s_{f2} = 0.649 kJ/kgK$, And $s_{fg2} = 7.502 kJ/kgK$

Since the expansion of steam is isentropic,

Therefore

Entropy of steam at inlet (s_1) = Entropy of steam at exit (s_2)

$6.583 = s_{fg2} + x_2 s_{fg_2} = 0.649 + x_2 + 7.502$

X2 = 0.791 Ans.

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

From steam tables, corresponding to a pressure of 10 bar, we find that enthalpy or total heat of dry saturated steam,

$h_1 = h_{g1} = 2776.2$ kJ/kg

and Corresponding to a pressure of 0.1 bar,

h_{f2} = 191.8 kJ/kg, and h_{fg2} = 2392.9 kJ/kg

∴ Enthalpy or total heat of steam of exit,

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

$$= 191.8 + 0.791 \times 2392.2$$

$$= 2084.6 \text{ kJ/kg}$$

and heat drop, $h_d = h_1 - h_2 = 2776.2 - 2084.6$

$$= 691.6 \text{ kJ/kg}$$

We know that velocity of steam leaving the nozzle,

$$V_2 = 44.72\sqrt{h_d} = 44.72\sqrt{691.6} = 1176 \text{ m/s} \quad \text{Ans.}$$

Velocity of steam leaving the nozzle when initial velocity,

$$\boldsymbol{V_1 = 135 \text{ m/s}}$$

We know that velocity of steam leaving the nozzle,

$$V_2 = \sqrt{V_1^2 + 2000\,h_d} = \sqrt{(135)^2 + 2000 \times 691.69} = 1184\, m/s \quad \textbf{Ans.}$$

6. ***Dry saturated steam enters a nozzle at a pressure of 10 bar and with an initial velocity of 90 m/s. The outlet pressure is 6 bar and the outlet velocity is 435 m/s. The hear loss from the nozzle id 9 kJ/kg of stream flow.***

Calculate the dryness fraction and the area at the exit, if the area at the inlet is 1256 mm²

Given:

p_1 =10 bar; V_1 = 90 m/s;

p_3= 6 bar; V_3= 435 m/s;

Loses = 9 kJ/Kg;

A_1=1256 mm^2 = 1256 x 10^{-6} m^2

Solution.

Dryness fraction of steam

Let x_3 = Dryness fraction of steam at the exit.

From steam tables,

corresponding to a pressure of 10 bar, we find that enthalpy of dry saturated steam,

h_1 = 2776.2 kJ/kg; and $v_{g1} = 0.1943 m^3/kg$ and corresponding to a pressure of 6 bar, we find that

$$h_{\beta} = 670.4 kJ/kg;\ h_{fg3} = 2085 kK/kg;\ v_{g3} = 0.3155 m^3/kg$$

We know that for a steady flow through the nozzle,

$$h_1 + \frac{1}{1000}\left(\frac{V_1^2}{2}\right) = h_3 + \frac{1}{1000}\left(\frac{V_3^2}{2}\right) + Losses$$

$$\therefore \quad h_3 = h_1 + \frac{2}{1000}(V_1^2 - V3^2) - Losses$$

$$= 2776.2 + \frac{1}{2000}[(90)^2 - (435)^2] - 9$$

=2776.2-99.6

= 2676.6 kJ/kg

We also know that enthalpy of wet stream (h_3),

$$2676.6 = h_{\beta} + x_3 h_{fg3} = 670.4 + x_3 \times 2085$$

$\therefore$ $x_3 = 0.962$ Ans.

Area at exit

Let A_3 Area at exit in m^2

We know that $\frac{A_1 V_1}{x_1 v_{g1}} = \frac{A_3 V_{31}}{x_{31} v_{g3}}$ or $\frac{1256 x 10^{-6} \times 90}{1 \times 0.1943} = \frac{A_3 \times 435}{0.962 \times 0.3155}$

…(For dry saturated steam, x_1 =1)

$A_3 = 406 \ x\ 10^{-6} m^2 = 406\ mm^2$ Ans.

7. ***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 flow is isentropic, and the corresponding expansion index is 1.135; find the ratio of cross – sectional area at exit and throat for maximum discharge.***

Solution. Given : p_1= 8 bar; p_3=1.5 bar; n=1.135

Let A_2 =Cross-sectional area at throat,

A_3 = Cross-sectional area at exit, and

m = Mass of steam discharged per second.

We know that for dry saturated steam (or when n= 1.135), critical pressure ratio,

$$\frac{p_2}{p_1} = 0.577$$

$\therefore$ p_2=0.577p1=0.577x 8 = 4.616 bar

Now complete the Mollier diagram for the expansion of steam through the nozzle, as shown in Fig.21.8.

From Mollier diagram, we find that

h_1 = 2775kJ/kg; h_2 = 2650kJ/kg; h_3 = 2465 kJ/kg; x_2 = 0.965; and x_3 = 0.902

From steam tables, we also find that the specific volume of steam at throat corresponding to 4.616 bar,

$$\upsilon_{g^2} = 0.405 m^3 / kg$$

and specific volume of steam at exit corresponding to 1.5 bar,

$$\upsilon_{g^3} = 1.159 m^3 / kg$$

Heat drop between entrance and throat,

$$h_{a2} = h_1 - h_2 = 2775 - 2650 = 125 kJ / kg$$

$\therefore$ Velocity of steam at throat,

$$V_2 = 44.72\sqrt{h_{d2}} = 44.72\sqrt{125} = 500 m/s$$

and $$m = \frac{A_2 V_2}{x_2 \upsilon_{g2}}$$

or $$A_2 = \frac{m x_2 \upsilon_{g2}}{V_2} = \frac{m \times 0.965 \times 0.405}{500} = 0.000786 m \qquad \ldots(i)$$

Heat drop between entrance and exit,

$$h_{d3} = h_1 - h_3 = 2775 - 2465 = 310 kJ/kg$$

$\therefore$ Velocity of steam at throat,

$$V_3 = 44.72\sqrt{h_{d3}} = 44.72\sqrt{310} = 787.4 m/s$$

and $$m = \frac{A_2 V_2}{x_2 \upsilon_{g2}}$$

or $$A_3 = \frac{m x_3 \upsilon_{g3}}{V_3} = \frac{m \times 0.902 \times 1.159}{787.4} = 0.001.33m \qquad \ldots(ii)$$

$\therefore$ Ratio of cross-sectional area at exit and throat,

$$\frac{A_3}{A_2} = \frac{0.00133m}{0.000786m} = 1.7 \text{ Ans}$$

8. Derive the expression for the critical pressure ratio in a steam nozzle.

There is only one value of the ratio (called critical pressure ratio) $\frac{p_2}{p_1}$ which will produce the maximum discharge. This can be obtained by differentiating '*m*' with respect to $\left(\frac{p_2}{p_1}\right)$ and equating it to zero. Other quantities except the ratio $\frac{p_2}{p_1}$ are constant.

$$\frac{d}{d[p_2/p_1]}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right] = 0$$

$$\frac{2}{n}\left[\frac{p_2}{p_1}\right]^{\frac{2}{n}-1} - \frac{n+1}{n}\left[\frac{p_2}{p_1}\right]^{\frac{n+1}{n}-1} = 0$$

$$\frac{2}{n} \times \left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1} = \frac{n+1}{n}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

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

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

$$\frac{\left(\frac{p_2}{p_1}\right)^{2-n}}{\left(\frac{p_2}{p_1}\right)} = \left(\frac{n+1}{2}\right)^n$$

$$\left(\frac{p_2}{p_1}\right)^{2-n-1} = \left(\frac{n+1}{2}\right)^n$$

$$\left(\frac{p_2}{p_1}\right)^{1-n} = \left(\frac{n+1}{2}\right)^n \implies \frac{p_2}{p_1} = \left(\frac{n+1}{2}\right)^{n\times\frac{1}{1-n}}$$

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

Critical pressure ratio of steam nozzle,

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

9. A convergent nozzle required to discharge 2kg of steam per second. The nozzle is supplied with steam at 7 bar and 180^{0}C and discharge takes place againsta back pressure of 1 bar. The expansion up to throat is isentropic and the frictional resistance between the throat and exit is equivalent to 63 kJ/kg of steam.Take approach velocity of 75 m/s and throat pressure 4 bar, estimate

i) suitable areas for the throat and exit,and

ii) Overall efficiency of the nozzle based on enthalpy drop between the actual inlet pressure and temperature and the exit pressure.

Final velocity of the steam

From steam tables, corresponding to a pressure of 15 bar, we find that enthalpy of dry saturated steam,

h_1 = 2789.9 kJ/Kg

and corresponding to a pressure of 1.5 bar, enthalpy of dry saturated steam,

h_2 = 2693.4 kJ/Kg

$\therefore$ Heat drop, $h_d = h_1 - h_2$=2789.9 – 2693.4 = 96.5 kJ/Kg

We know that final velocity of the steam,

$$V_2 = 44.72\ \sqrt{h_d} = 44.72\sqrt{96.5} = 439.3 m/s$$

10.Define critical pressure ratio of a nozzle and discuss why attainment of sonic velocity determines the maximum discharge through steam nozzle.

Critical Pressure Ratio:

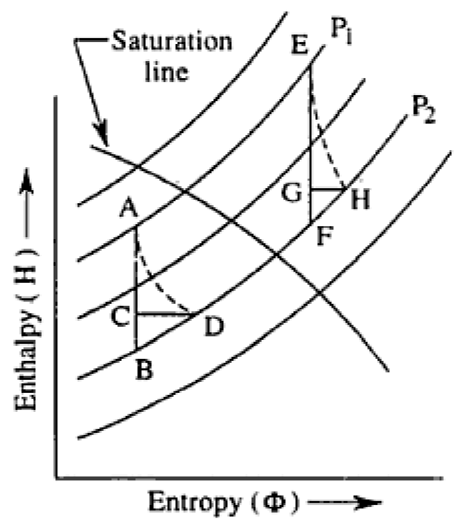
The critical pressure ratio is the pressure ratio which will accelerate the flow to a velocity equal to the local velocity of sound in the fluid.

Critical flow nozzles are also called **sonic chokes**. By establishing a shock wave the sonic choke establish a fixed flow rate unaffected by the differential pressure, any fluctuations or changes in downstream pressure. A sonic choke may provide a simple way to regulate a gas flow.

11. Explain the metastable expansion of steam in a nozzle with help of h-s diagram.

Most of the friction occurs in the diverging part of a convergent-divergent nozzle as the length of the converging part is very small. The effect of friction is to reduce the available enthalpy drop by about 10 to 15 per cent. The velocity of steam will be then $V_2 = 44.72\sqrt{K(H_1 - H_2)}$ where K is the coefficient which allows for friction loss. It is also known as nozzle efficiency (η_n)

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



$$\theta = 29^{o}$$

(b) During force on the wheel

We know that driving force on the wheel,

$$F_x = m(V_w + V_{w1}) = 0.5\ (1130 + 190) = 660\ N$$

(c) Arial thrust on the wheel

We know that axial thrust on the wheel,

$$F_\gamma = m(V_f - V_{f1}) = 0.5(410 - 310) = 50N$$

(d) Power development by the turbine

We know that power development by the turbine,

$$P = m(V_w + V_{w1})\ V_b$$
$$= 0.5(1130 + 190)\ 375 = 247\ 500\ W$$
$$= 247.5\ kW$$

12. 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 flow is isentropic and if the corresponding expansion index is 1.133, find the ratio of cross sectional area at exit and throat for maximum discharge.

Heat drop between entrance and exit,

$$h_{d3} = h_1 - h_3 = 2775 - 2465 = 310 kJ/kg$$

∴ Velocity of steam at throat,

$$V_3 = 44.72\sqrt{h_{d3}} = 44.72\sqrt{310} = 787.4 m/s$$

and $$m = \frac{A_2V_2}{x_2\upsilon_{g2}}$$

or $$A_3 = \frac{mx_3\upsilon_{g3}}{V_3} = \frac{m\times 0.902\times 1.159}{787.4} = 0.001.33m$$

∴ Ratio of cross-sectional area at exit and throat,

$$\frac{A_3}{A_2} = \frac{0.00133m}{0.000786m} = 1.7$$

13. Dry saturated steam at a pressure of 11 bar enters a convergent divergent nozzle and leaves at at pressure of 2 bar. If the flow is adiabatic and frictionless, determine

i) The exit velocity of steam and

ii) Ratio of cross section of exit and that at throat.

From steam tables, corresponding to a pressure of 10 bar, we find that enthalpy or total heat of dry saturated steam,

$$h_1 = h_{g1} = 2776.2 \text{ kJ/kg}$$

and

corresponding to a pressure of 0.1 bar,

$$h_{f2} = 191.8 \text{ kJ/kg, and } h_{fg2} = 2392.9 \text{ kJ/kg}$$

∴ Enthalpy or total heat of steam of exit,

$$h_2 = h_{f2} + x_2h_{fg2}$$

$$= 191.8 + 0.791 \times 2392.2$$

$$= 2084.6 \text{ kJ/kg}$$

and heat drop, $h_d = h_1 - h_2 = 2776.2 - 2084.6$

$= 691.6$ kJ/kg

MODULE I

STEAM TURBINE

STEAMTURBNES

The steam turbine is a prim-mover in which the potential energy of steam istransferred Into kinetic energy and later in its turn transferred into the mechanical energyofrotation of The turbine shaft.

Based on action of steam the steam turbines may be classifiedas

(i) Impulse turbine
(ii) Reaction turbine
(iii) Impulse and reaction turbine

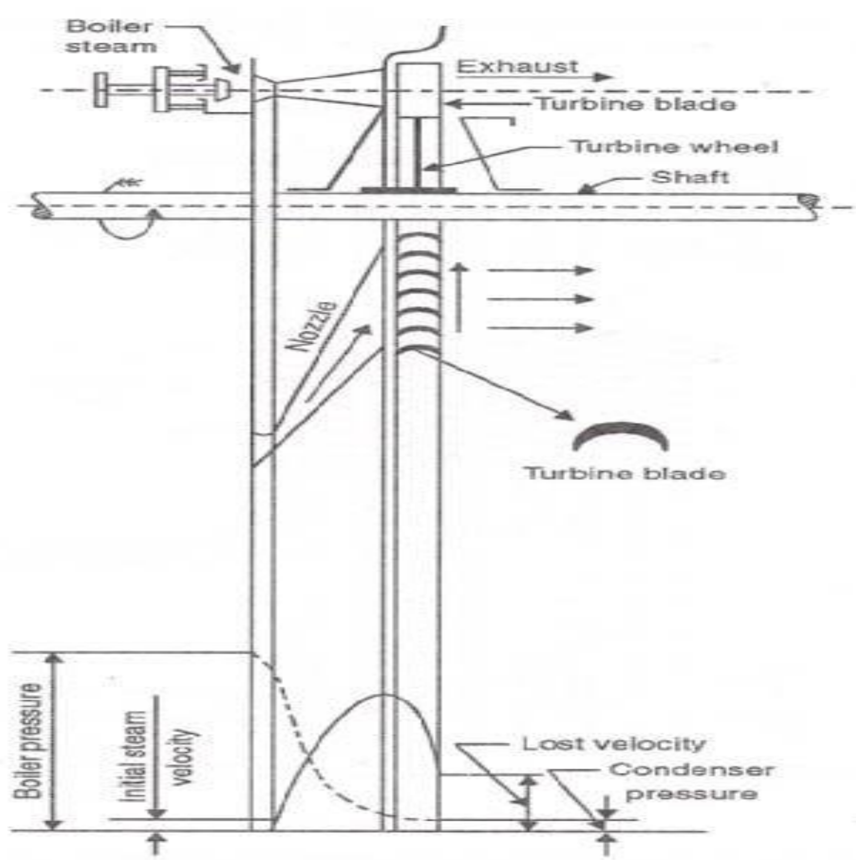
According to the direction of steam flow

(i) Axial flow turbine
(ii) Radial flow turbine

According to the number of stages

(i) Single stage turbine
(ii) Multi stage turbine

Simple Impulse turbine

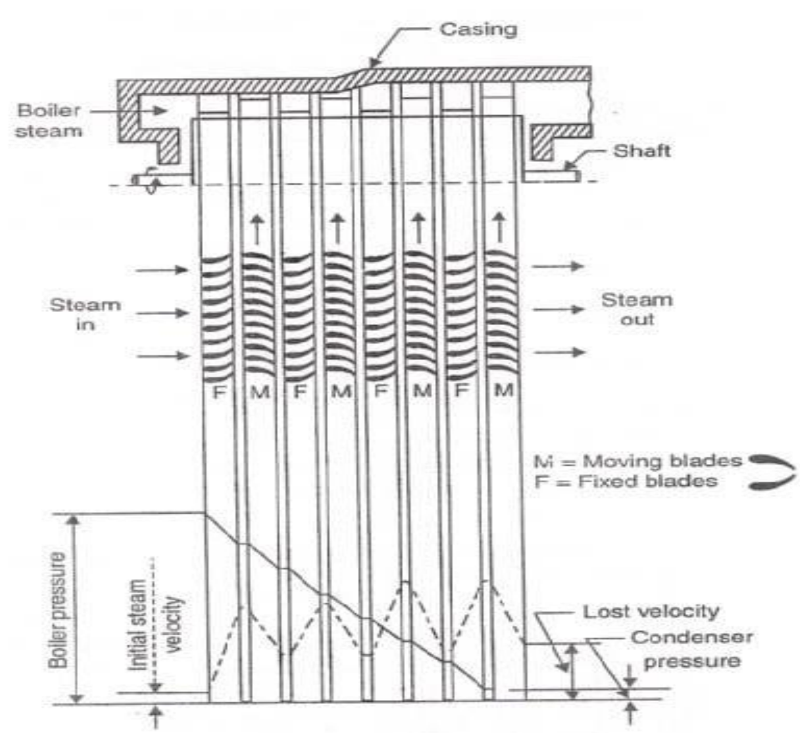


An impulse turbine runs by the impulse of steam jet. In this turbine the steam isfirst made to flow through a nozzle. Then the steam jet impinges on the turbine blades. The steam jet after Impinging on the rotor blades glides over the concave surface of the blades and finally Leaves the turbine.

A De-Laval turbine is the simple impulse turbine and is commonly used with fixed nozzles and a rotor with a ring of blades inside a casing. The surface of the blades are

Generallyverysmoothtominimizethefrictionallosses.Thebladesaregenerallymadeof Special steel alloys. Steam supplied to an impulse turbine expands completely in the nozzle. As the steam flows through the nozzle its pressure falls from steam chest pressure to Condenser pressure. Due to this relatively higher ratio of expansion of steam in the nozzlesthe Steam leaves the nozzle with a very high velocity. It can be observed that the velocity of the Steam leaving the moving blades is comparatively higher .The loss of energy due to this Higher exit velocity is called “Carry over loss” or “leaving loss”.This loss may amount to 3 to % of the nozzle velocity. The moving blades of impulse turbine are ‘constant flow area profile type blades'. Therefore The pressure remains constant during the flow of steam through the moving bladesof impulse Turbine.

Reaction turbine



Inthistypeofturbine,thereisagradualpressuredropandtakesplacecontinuously over the fixed and moving blades. The function of the fixed blades is that they alter the direction of the steam as well as allow it expand to a larger velocity. As the steam passes over the moving blades its kinetic energy is absorbed by them. Instead of a set of nozzles, steam is admitted for whole of the circumference and therefore there is all round admission. In passing through the first row of fixed blades, the steam undergoes a small drop in pressure and its velocity is increased. It then enters the first row of moving blades and it suffers a change in direction and therefore momentum. This gives impulse to the blades. But the moving blades are of aerofoil type and hence there is also a pressure drop in the moving blades.

The reaction turbines which are used these days are really impulse-reaction turbines. Pure reaction turbines are not in general use. The expansion of steam and heatdropoccurbothinfixedandmovingblades.Thevelocityofsteaminthistypeofturbinesiscomparatively low,themaximumbeingaboutequaltobladevelocity.This type of turbine very successful lin practice. Itis also called "Parson's Reaction Turbine".

Difference between Impulse and Reaction turbines

S. No.	Particulars	Impulse turbine	Reaction turbine
1.	*Pressure drop*	Only in nozzles and not in moving blades.	In fixed blades (nozzles) as well as in moving blades.
2.	*Area of blade channels*	Constant.	Varying (converging type).
3.	*Blades*	Profile type.	Aerofoil type.
4.	*Admission of steam*	Not all round or complete.	All round or complete.
5.	*Nozzles / fixed blades*	Diaphram contains the nozzle.	Fixed blades similar to moving blades attached to the casing serve as nozzles and guide the steam.
6.	*Power*	Not much power can be developed.	Much power can be developed.
7.	*Space*	Requires less space for same power.	Requires more space for same power.
8.	*Efficiency*	Low.	High.
9.	*Suitability*	Suitable for small power requirements.	Suitable for medium and higher power requirements.
10.	*Blade manufacture*	Not difficult.	Difficult.

Methods of reducing rotor speed

Incaseofsimpleimpulseturbine,thesteamisexpandedfromtheboilerpressure to the condenser pressure in one stage only. Hence the speed of the rotorbecomes very high for practical purposes. In order to make the rotor speed practicablecompounding of steam turbine is done. Compounding is the method of reducing rotorspeed by adding stages to a simple impulse turbine without affecting the turbine workoutput.Therotor speedcanbe reduced by thefollowing methods.

(i) Velocity compounding

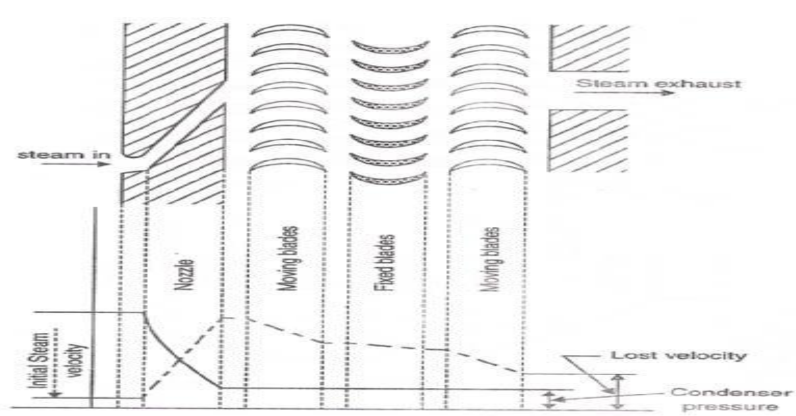
(ii) Pressure compounding

(iii) Pressure-Velocity compounding

(iv) Reaction turbine

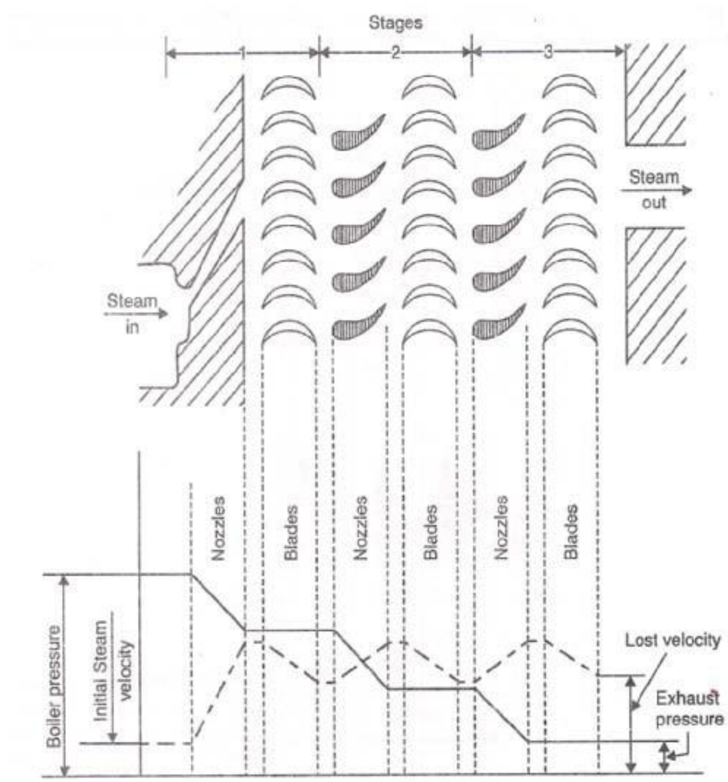
Velocity Compounding

Steamisexpandedthroughastationarynozzlefromtheboilerorinletpressureto condenser pressure. So, the pressure in the nozzle drops, the kinetic energy of thesteamincreasesduetoincreaseinvelocity.Aportionofthisavailableenergyisabsorbed by a row of moving blades. The steam (whose velocity has decreased while moving over the moving blades) then flows through the second row of blades which are fixed. The function of these fixed blades is to redirect the steam flow without altering its velocity to the following next row moving blades where again work is done on them and steam leaves the turbine with a low velocity. Fig shows a cut away section of such stage and changes in pressure and velocity as the steam passes through the nozzle, fixed and moving blades. Though this method has the advantage that the initial cost is low due to lesser number of stages yet its efficiency is low.



Pressure Compounding

Fig shows rings of fixed blades incorporated between the rings of moving blades. The steam at boiler pressure enters the first set of nozzles and expands partially. The kinetic energy of the steam thus obtained is absorbed by the moving blades. The steam then expands partially in the second set of nozzles where its pressure again falls and thevelocityincreases;thekineticenergysoobtainedisabsorbedbythesecondringofmoving blades (stage-2). This is repeated in stage-3 and steam finally leaves the turbine at low velocity and pressure. The number of stages depends on the number of rows of nozzles through which the steam must pass.
This method of compounding is used in Rateau and Zoelly turbine. This is most efficient turbine since the speed ratio remains constant but it is expensive owing to a large number of stages



Pressure-Velocity Compounding

Thismethodofcompoundingisthecombinationofpressureandvelocitycompounding. The total drop in steam pressure is divided into stages and the velocity obtained in each stage is also compounded. The rings of nozzles are fixed at the beginning of each stage and pressure remains constant during each stage. The changes in pressure and velocity are shown. This method of compounding is used in Curtis and Moore turbine.