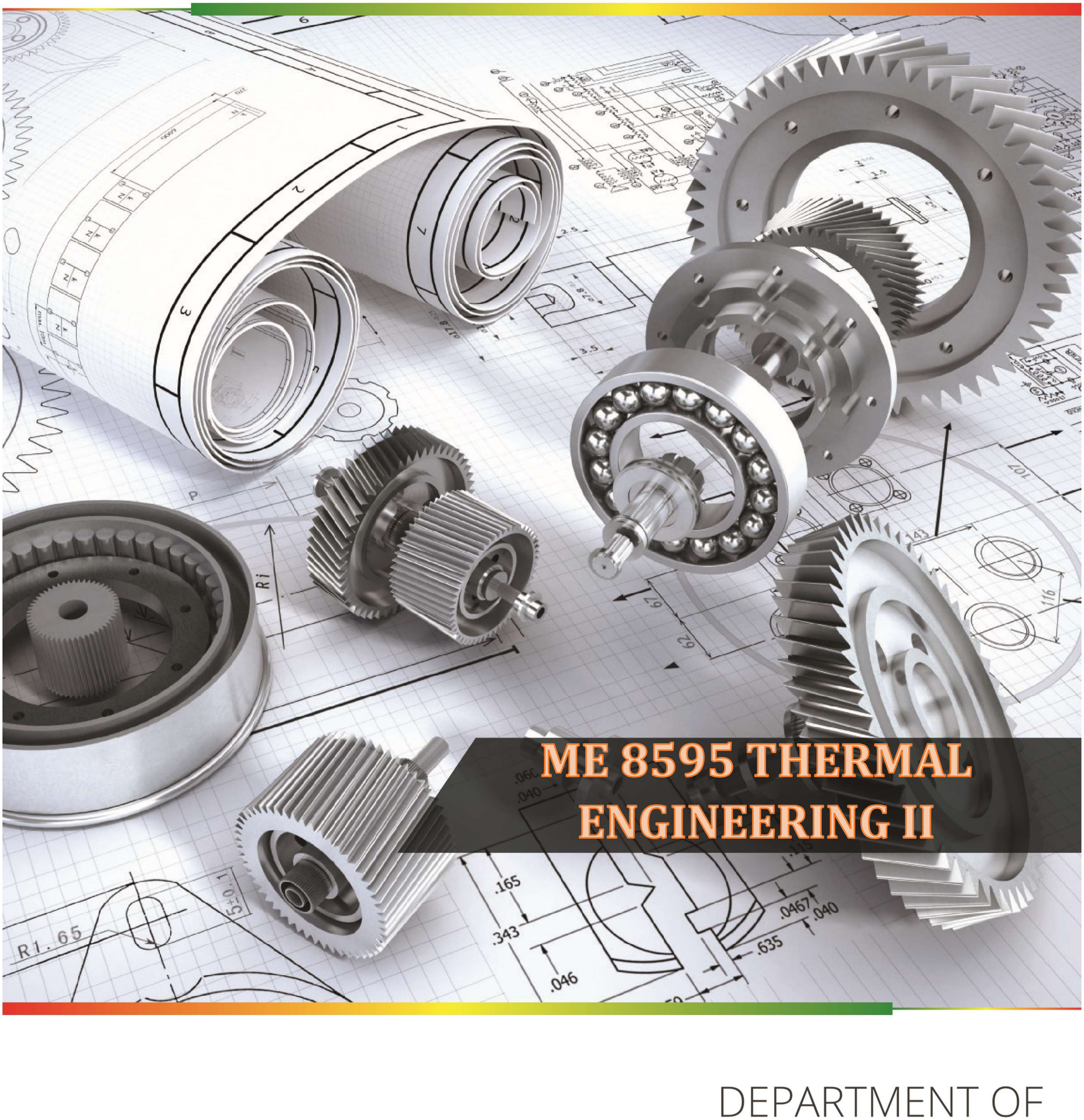
ROHINI
COLLEGE OF ENGINEERING & TECHNOLOGY
Approved by AICTE and Affiliated to Anna University, (An ISO Certified Institution)



ME 8595 THERMAL ENGINEERING II

DEPARTMENT OF
MECHANICAL ENGINEERING

ROHINI COLLEGE OF ENGINEERING AND TECHNOLOGY

DEPARTMENT OF MECHANICAL ENGINEERING

ME8595 - THERMAL ENGINEERING – II

YEAR / SEM: III / V **REGULATION: 2017**

LECTURE NOTES

(Both typed and written notes)

Prepared by

MANOJ.J.K

Assistant Professor

ME8595 - THERMAL ENGINEERING - II

UNIT I STEAM NOZZLE

- Types and Shapes of nozzles
- Flow of steam through nozzles
- Critical pressure ratio,
- Variation of mass flow rate with pressure ratio
- Effect of friction
- Metastable flow

UNIT II BOILERS

- Types and comparison
- Mountings and Accessories
- Fuels-Solid, Liquid and Gas
- Performance calculations
- Boiler trial

UNIT III STEAM TURBINES

- Types of steam turbine
- Impulse and reaction principles
- Velocity diagrams
- Work done and efficiency
- Optimal operating conditions
- Multi-staging,
- Compounding
- Governing

UNIT IV COGENERATION AND RESIDUAL HEAT RECOVERY

- Cogeneration Principles
- Cycle Analysis
- Applications
- Source and utilisation of residual heat
- Heat pipes, Heat pumps
- Recuperative and Regenerative heat exchangers
- Economic Aspects

UNIT V REFRIGERATION AND AIR – CONDITIONING

- Vapour compression refrigeration cycle
- Effect of Superheat and Sub-cooling
- Performance calculations
- Working principle of air cycle
- Vapour absorption system
- Thermoelectric refrigeration
- Air conditioning systems
- Concept of RSHF, GSHF and ESHF
- Cooling load calculations
- Cooling towers - concept and types.

TEXT BOOKS

1. Kothandaraman, C.P., Domkundwar .S and Domkundwar A.V.,**"A course in Thermal Engineering"**, Dhanpat Rai & Sons, 2016.
2. Mahesh. M. Rathore, **"Thermal Engineering"**, 1st Edition, Tata Mc Graw Hill Publications, 2010.

REFERENCES

1. Arora .C.P., "Refrigeration and Air Conditioning", Tata Mc Graw Hill, 2008
2. Ballaney. P.L ." Thermal Engineering", Khanna publishers, 24th Edition 2012
3. Charles H Butler : Cogeneration" McGraw Hill, 1984.
4. Donald Q. Kern, " Process Heat Transfer", Tata Mc Graw Hill, 2001.
5. Sydney Reiter "Industrial and Commercial Heat Recovery Systems" Van Nostrand Reinhols,1985

CO No.	Course Outcomes	Highest Cognitive Level
17C301.1	**Explain the types of nozzles and apply the concept for different velocity flow through a nozzle**	**K4**
17C301.2	**Explain the functioning and features of different types of boiler and auxiliaries and calculate performance parameters**	**K2**
17C301.3	**Explain the flow in steam turbines, plot velocity diagram and calculate the work done & efficiency**	**K4**
17C301.4	**Summarize the concept of cogeneration, working features of heat pump and heat exchangers**	**K2**
17C301.5	**Explain the types of Refrigeration & Air-conditioning and its performance**	**K4**

UNIT I

STEAM NOZZLE

- Types and Shapes of nozzles
- Flow of steam through nozzles
- Critical pressure ratio,
- Variation of mass flow rate with pressure ratio
- Effect of friction
- Metastable flow

MANOJ.J.K

AP/MECH
ROHINI CET

✓flow of steam through

✓Critical Pressure ratio

✓Variation of mass flow ratio with Press

✓Effect of friction

✓metastable flow

Nozzle:

Nozzle is a device which increases th
Kinatic energy or velocity and reduces
Pressure or Pressure drops

Steam Nozzle:

SteamNozzle is a device which increases th
Kinatic energy of the steam and reduce
Pressure of the steam

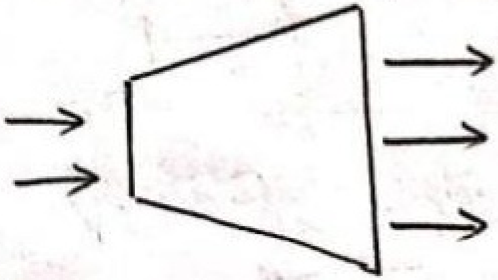
Types and shapes of Nozzle.

1 - convergent Nozzle

Divergent Nozzle

In Divergent Nozzle the Cross se

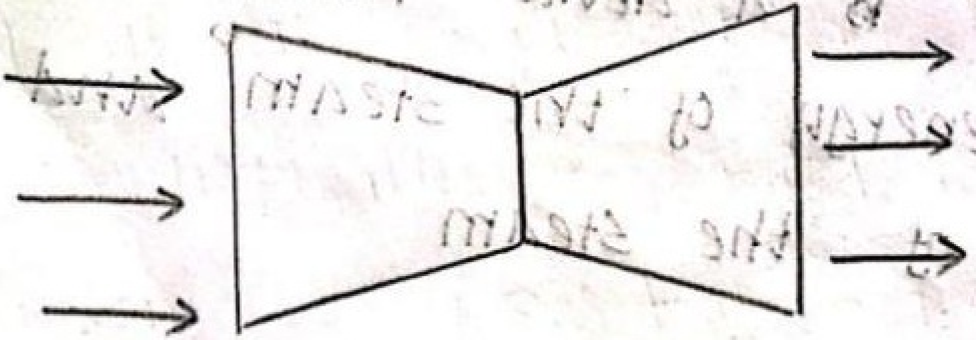
Area Increases from Inlet to Exit.



Convergent Divergent Nozzle : [CD Nozzle]

In convergent Divergent Nozzle th

Crosssectional area first decreases f
Inlet to throat and the Increases f
throat to Exit



Flow of steam through Nozzle

The flow of steam through a

The steam entering the nozzle at low elocity and high pressure and leaves at igh velocity and low pressure

nsider the unit mass of steam ($m = 1$ kg/s)

t C_1 = Velocity of steam at inlet

$C_2 \rightarrow$ Velocity of steam at Exit

$h_1 \rightarrow$ enthalpy of steam at inlet

$h_2 \rightarrow$ enthalpy of steam at Exit

r steady flow Process,

or steady flow energy equation (SFEE)

$$z_1 + \frac{C_1^2}{2} + h_1 + Q = gz_2 + \frac{C_2^2}{2} + h_2 + W$$

nditions of Nozzle,

$Q = 0$

$W = 0$

$z_1 = z_2$

pply all the above Conditions in SFEE

$$C^2 \quad h = C_2^2 + h_2$$

$$C_2 = \sqrt{2(h_1 - h_2) + C_1^2}$$

$C_1 \cancel{<<} C_2$ so C_1 is negligible ($C_1 << C_2$)

$C_2 =$

for unit conversion,

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

WKT, $h = C_p T$

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

Problem 1.1

Dry air at a pressure of 10 bar a

is isentropically expended to a nozzle

a pressure of 1.2 bar determine

max discharge through the nozzle

$A = 130 mm^2$

To find:

mass flow, m

Solution,

The characteristic gas eqn

$$PV = mRT$$

$$P_1 V_1 = mRT_1$$

For unit mass

$m = 1$ Kg/s

For Air

$R = 0.286$ KJ/Kg K

$C_p = 1.005$ KJ/Kg K

$\gamma = 1.4$

From gas equation

$$1000 \times v_1 = 1 \times 0.287 \times 623$$

$$\boxed{v_1 = 0.178 \ m^3/Kg}$$

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

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

$$V_2 = 0.178\left(\frac{1000}{120}\right)^{1/1.4}$$

$$V_2 = 0.809$$

$$\boxed{V_2 = 0.81\ m^3/Kg}$$

For isentropic process,

PT relation,

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

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

$$T_2 = 623\left(\frac{120}{1000}\right)^{\frac{1.4-1}{1.4}}$$

$m = \rho A C$

$m = \rho_2 A_2 C_2$ $\qquad \rho = m/V$

$m = \dfrac{A_2 C_2}{v_2}$ $\qquad v = V/m$

$\qquad \rho = 1/v$

$$= \frac{130 \times 10^{-6} \times 754.2}{0.81}$$

$$\boxed{m = 0.121 \text{ kg/s}}$$

Mass of steam discharge through the

For polytropic process,

$$Pv^n = C$$

$n_1 = 1.35$ for saturated steam

$n = 1.3$ for super saturated steam

let P_1 = Initial pressure of steam

also reduces

This reduction in enthalpy must be to the increase in kinetic engergy, the workdone by the steam is equal the enthalpy drop

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

Gain in kinetic energy = Workdone during Isentropic pr

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

$C_1 << C_2$, C_1 is neglated

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

$$\boxed{\frac{C_2}{2} = \frac{n}{n-1} P_1V_1 \left(1 - \frac{P_2V_2}{P_1V_1}\right)} \text{ — } ①$$

$$\frac{V_2}{V_1} = \left(P_1/P_2\right)^{1/n} \quad \text{——} 2$$

Sub equ ② in ①

$$① \Rightarrow \frac{C_2^2}{2} = \frac{n}{n-1} P_1 V_1 \left[1 - \frac{P_2}{P_1}\left(\frac{V_2}{V_2}\right)\right]$$

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

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

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

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

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

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

$m = \frac{AC}{V}$

Sub eqn ③ in above eqn:

$$m = \frac{A_2}{V_2}\sqrt{\frac{2n}{n-1}P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]} \quad —5$$

From equation 2

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

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

Sub the Value of V_2 in m

$$m = \frac{A_2}{V_1\left(P_1/P_2\right)^{1/n}}\sqrt{\frac{2n}{n-1}P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= \frac{A_2}{V_1}\left(\frac{P_2}{P_1}\right)^{1/n}\sqrt{\frac{2n}{n-1}P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

$$m = A_2 \sqrt{\frac{2n}{n-1} \frac{P_1 v_1}{v_1^2} \left[\left(\frac{P_2}{P_1}\right)^{2/n} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right.}$$

Condition for Maximum discharge

$$m = A_2 \sqrt{\frac{2n}{n-1} \frac{P_1}{v_1} \left(P_2/P_1\right)^{2/n} - \left(P_2/P_1\right)^{\frac{n+1}{n}} \Big]}$$

There is only one value of the ratio (crit Pressure ratio) $\left(\frac{P_2}{P_1}\right)$, which will produce a maximum discharge

This can be obtained by differentiatin mass flow m w.r.t P_2/P_1 and equat it to zero

other quatities except the ratio P_2/P constant

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

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

$$\left(\;/P_1\right) = \left(\frac{\;}{2}\right)\left(\;/P_1\right)$$

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

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

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

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

$$\boxed{P_2/P_1 = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}} \text{ --- ⑦}$$

$$\text{ⓑ} \Rightarrow m = A_2 \sqrt{\frac{2n}{n-1}\left(\frac{P_1}{V_1}\right)\left[\left(P_2/P_1\right)^{2/n} - \left(P_2/P_1\right.\right.}$$

in $\Rightarrow$

$\xrightarrow{} m = 0.6356 \times A\sqrt{P_1/v_1}$

Substitute $n = 1.135$ for saturated steam

and $n = 1.3$ for super heated steam.

$\searrow m = 0.6670 A \times \sqrt{P_1/v_1}$

Case (i)

Put $n = 1.135$

$$m_{max} = A\sqrt{\underset{16.81}{\frac{2\times1.135}{1.135-1}}\left(\frac{P_1}{v_1}\right)\left[\overset{0.3799}{\left(\frac{2}{1.135+1}\right)^{\frac{2}{1.13}}} - \left(\frac{2}{(1.135+1)}\right)^{\frac{1.135+1}{1.135-1}}\right.}$$

2.2062

$$= A\sqrt{5.984\left(\frac{P_1}{v_1}\right) - 0.3559}$$

$$= A\sqrt{16.81\left(\frac{P_1}{v_1}\right)\left[0.3799 - 0.3\right.}$$

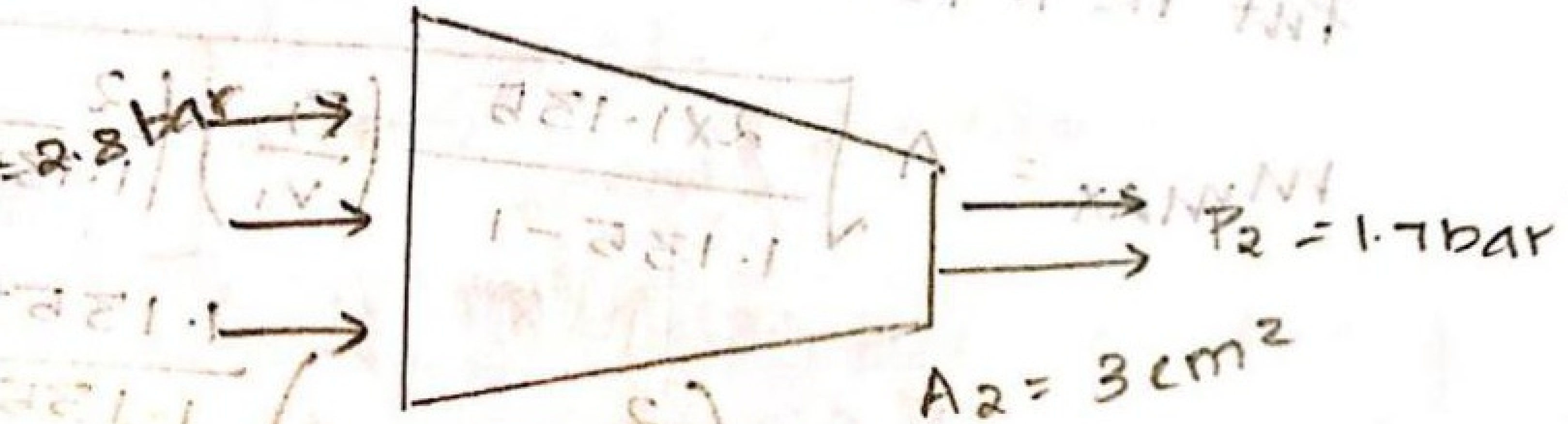
$$= A\sqrt{1681 \times 0.024\,(P_1/v_1)}$$

$= 0.6356$

case (ii) put $n = 1.3$

$$m_{max} = A\sqrt{\frac{2\times1.3}{1.3-1}\left(\frac{P_1}{v_1}\right)\left[\frac{2}{1.3+1}\right]^{\frac{2}{1.3-1}} - \left(\frac{2}{1.3+1}\right)^{\frac{1.3+}{1.3-}}}$$

e exit velocity · and mass flow
te for
(i) Isentropic expansion
(ii) super heated flow



ven :

$P_1 = 2.8$ bar $= 280\ kN/m^2$

$P_2 = 1.7$ bar $= 170\ kN/m^2$

$A_2 = 3\,cm^2 = 3\times10^{-4}\,m^2$

find.

C_2 & m for Isentropic expansion

for exit condition

from steam tables for $P_2 = 1.7$ bar

$v_{f2} = 0.001056$ m^3/Kg

$v_{g2} = 1.0304$ m^3/Kg

$h_{f2} = 483.2$ KJ/Kg

$h_{fg2} = 2215.8$ KJ/Kg

$h_{g2} = 2699.0$ KJ/Kg

$s_{f2} = 1.475$ KJ/KgK

$s_{fg2} = 5.706$ KJ/KgK

$s_{g2} = 7.181$ KJ/KgK

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

For expansion the entropy remains const

$s_1 = s_2$

$s_2 = 7.014$ KJ/Kg·K

$h_2 = 2632.5$ kJ/kg

To find Exit Velocity

$$c_2 = \sqrt{2000(h_1 - h_2)}$$

$$= \sqrt{2000(2721.5 - 2632.5)}$$

$$\boxed{c_2 = 421.8 \text{ m/s}}$$

To find the mass flow rate:-

$$m = \rho_2 A_2 c_2$$

$$m = \frac{A_2 c_2}{v_2}$$

$$v_2 = x_2 v_{g_2}$$

$$v_2 = 0.97 \times 1.0309$$

$$v_2 = 1 \text{ m}^3/\text{kg}$$

$C_2 = \sqrt{\frac{2n}{n-1}\left(P_1V_1\right)\left[1-\left(\frac{P_2}{P_1}\right)\right.}$

for super heated steam = $n = 1.3$

$$C_2 = \sqrt{\frac{2\times1.3}{1.3-1}\left(\underset{2.8\times10^5}{280\times0.646}\right)\left[1-\left(\frac{\overset{170}{\cancel{280}}}{\underset{280}{\cancel{110}}}\right)\right.}$$

$$\boxed{C_2 = 412.92 \text{ m/s}}$$

To find:

From polytropic Pv relation:

$$Pv^n = c$$

$$P_1V_1^n = P_2V_2^n$$

$$P_1/P_2 = \left(V_2/V_1\right)^n$$

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

$$V_2 = V_1\left(P_1/P_2\right)^{1/n} \qquad 1/1.3$$

$$\boxed{m = 0.131 \text{ Kg/s}}$$

Answer:

) Isentropic expantion $c_2 = 421.8$ m/s

$m = 0.126$ Kg/s

, Super heated flow $c_2 = 415.09$ m/s

$m = 0.13$ Kg/s

itical Temperature:

ere is only one value of ratio P_2/P_1 which roduces maximum discharge from the ozzle this ratio is called critical Pressur atio

where P_1 = inlet Pressure

P_2 = Exit Pressure

$$= \left[\frac{2}{(1.135+1)}\right]^{\overline{1.135-1}}$$

$$P_2/P_1 = 0.577$$

Case(ii) for super heated steam $\boxed{n = 1.3}$

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

$$P_2/P_1 = 0.545$$

Case (iii) for gas $n = 1.4$

$$P_2/P_1 = \left[\frac{2}{[1.4+1]}\right]\left[{}^{1.4}/_{1.4-1}\right]$$

steam flow process is

correspond expansion index is 1.135

the ratio of crosssectional Area at

and throat for max discharge

Given Data:

$P_1 = 8$ bar

$P_2 = 1.5$ bar

$n = 1.135$

To find:

A_2 / A_t

Solution:

Critical Pressure ratio

$P \quad [\;]^{\frac{n}{}}$

$P_t = 4.6$ bar

At inlet Condition (dry

for steam tables for $P_1 = 8$ ba

$h_1 = 2767.4$

$s_1 = s_g = 6.660$

$v_1 = s_g = 0.24026$

At throat condition

from steam tables for throat Pressur

$P_t = 4.6$ bar

$v_{f_t} = 0.001090$ m³/kg.

$v_{g_t} = 0.40526$ m³/kg

$h_{f_t} = 626.7$ KgJ/kg

$h_{fg_t} = 2117.2$ kJ/kg

$h_{g_t} = 2743.9$ kJ/kg

$hf_2 = 467.1$ kJ/kg

$fg_2 = 2226.3$ kJ/kg

$hg_2 = 2693.4$ kJ/kg

$f_2 = 1.433$ kJ/kgK

$fg_2 = 5.790$ kJ/kgK

$g_2 = 7.223$ kJ/kgK.

onsider Section 1-t.

$st = sft + x_t\, sfgt$

or isentropic expansion

$s_1 = st = s_2$

$st = s_1 = 6.66$ kJ/kgK

$6.66 = 1.829 + x_t (5.018)$.

$= \sqrt{2000(267.4 - 2654.8}$

$c_t = 463.89$ m/s

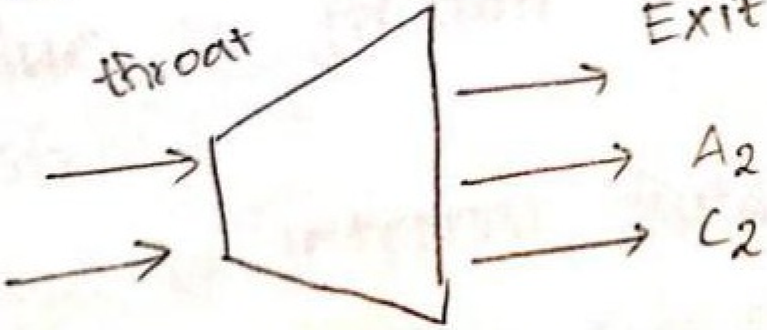
e

$V_t = x_t \times V_{gt}$

$= 0.96 \times 0.40526$

$\boxed{V_t = .384 \text{ m}^3/\text{kg}}$

onsider section t−2



N KT

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

or isentropic expansion $s_1 = s_t = s_2$

6.66 KJ/Kg K

$h_2 = 2470.77$ KJ/Kg

K.T

$h_t = 2654.21$ KJ/Kg

$_2 = \sqrt{2000(h_t - h_2)}$

$= \sqrt{2000(2654.21 - 2470.77)}$

$c_2 = 613.9$ m/s

$_2 = x_2 V_{f_2}$

$= 0.9 \times 1.159$

$V_2 = 1.04$ m³/kg

Mass flow rate of steam at throat } = { mass of flow rat of steam at exi

$A_2/A_t = 2.02$

Effect of friction in a Nozzle

When a steam flows through a Nozz the final Velocity of steam for a d given pressure drop is reduce due to following reason

- → due to friction between nozzle surface ar steam
- → due to internal fluid friction in steam
- → due to shock losses

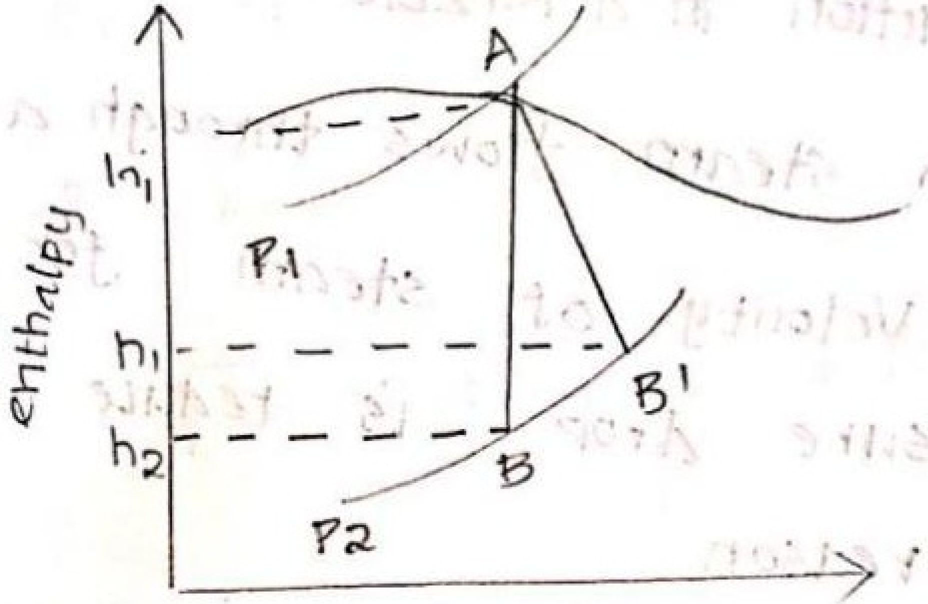
Most of the friction losses occur between th and Exit in a convergent Divergent N

The effect frictional losses are

- → The expansion is no more isentropic and enthalpy drop is reduced

lower exit Velocity

this friction reheating



Entropy (s) →

h-s diagram or Moller diagram

The enthalpy drop $h_1 - h_2$ is known as isentropic enthalpy drop. The actual enthalpy drop $(h_1 - h_3')$

Nozzle efficiency or co-efficient of Nozzle

It is defined as the ratio of

supplied with

and discharge takes place against

back pressure of 1 bar. The expansion upto throat is isentropic and the frictional resistance between the throat and exit is equalent to 63 KJ/kg. Take approach velocity of 75 m/s and throat Pressur 4 bar. estimate

1). Suitable area for the throat to Exit

2) over all efficiency of the nozzle

3) based on the enthalpy b/w the actual inlet pressure and Tempessure and the Exit pressure

Given:

CD Nozzle

$m = 2 kg/s$

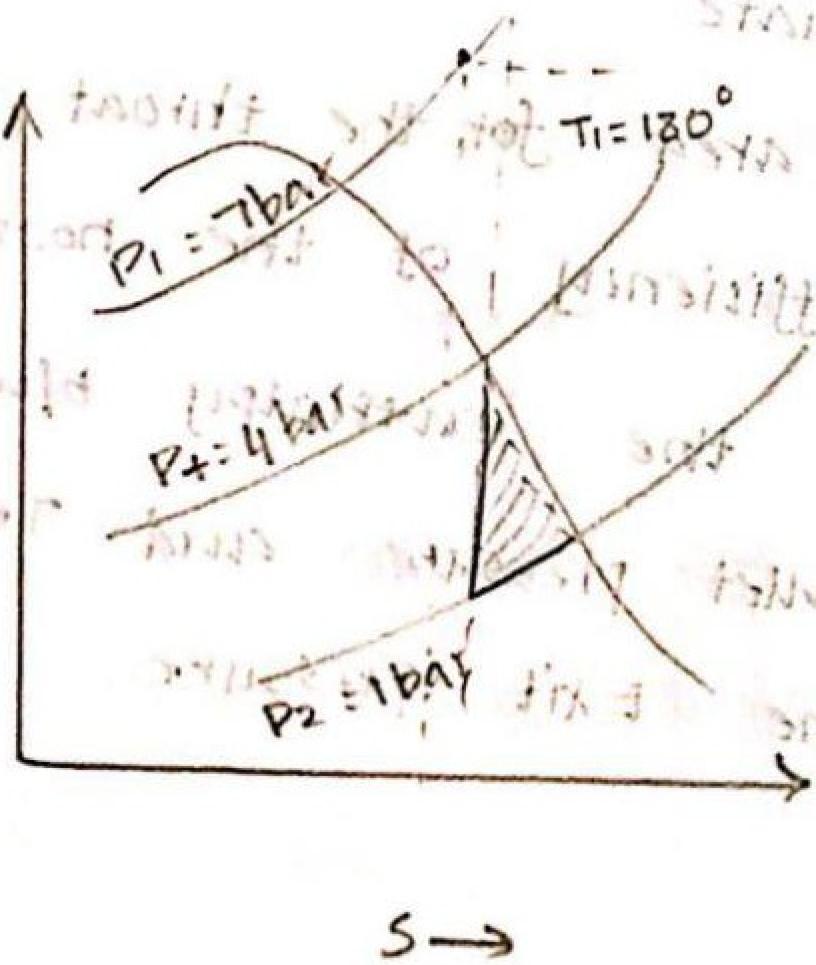
i) A_t / A_2

ii) η_{nozzle}

Solution :

At inlet Condition :

from Mollier chart for P_1 7bar * $T_1 = 1$



$h_1 = 2745$ KJ/Kg

$S_1 = 6.78$ KJ/KgK

$V_1 = 0.2880$ m^3/

At throat Condition

from steam tables $P_2 = 4$bar

At Exit Condition:

From steam stables for $P_2 = 1 bar$

$V_{f2} = 0.001043 \ m^3/Kg$

$V_{g2} = 1.6938 \ m^3/Kg$

$hf_2 = 417.5 \ KJ/Kg$ $\quad sf_2 = 1.303 \ KJ/Kg K$

$hfg_2 = 2257.9 \ KJ/Kg$ $\quad sfg_2 = 6.057 \ KJ/K$

$hgt = 2675.4 \ KJ/Kg$ $\quad sg_2 = 7.360 \ KJ/Kg$

Consider 1-t

$S_t = Sf_t + X_t \cdot Sfg_t$

$6.78 = 1.776 + X_t \ 5.118$

$$\boxed{X_t = 0.97}$$

At throat the steam is WET

The condition of steam at throught is

$h = hf_t + xhfg_t$

$$C_t = \sqrt{2000(h_1 - h_t) + C_1^2}$$

$$= \sqrt{2000(2795 - 2673.6 + 75^2}$$

$$C_t = 498.42 \text{ m/s}$$

At Exit Condition:

$$s_2 = s_{f_2} + x_2 (s_{fg_2})$$

$s_2 = s_t = s_1$ (isentropic expansion

$$6.78 = 1.303 + x_2 (6.057)$$

$$\boxed{x_2 = 0.90}$$

The condition of the steam at Exit

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

$$h_2 = 417.5 + 0.9 \times 2257.9$$

$= 2512.61$

$$\eta_{nozzle} = \frac{\text{Actual enthalpy drop}}{\text{Isentropic enthalpy drop}}$$

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

$$\eta_{nozzle} = \frac{2795 - 2512.61}{2795 - 2449.61}$$

$$= 81.7$$

To find At

$$m = \frac{A_t C_t}{V_t}$$

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

$$2512.6 = 417.5 + x_2'(2257.9)$$

$$x_2' = 0.92$$

$$v_2 = x_2' \, v_{g_2}$$

$$= 0.92 \times 1.6938$$

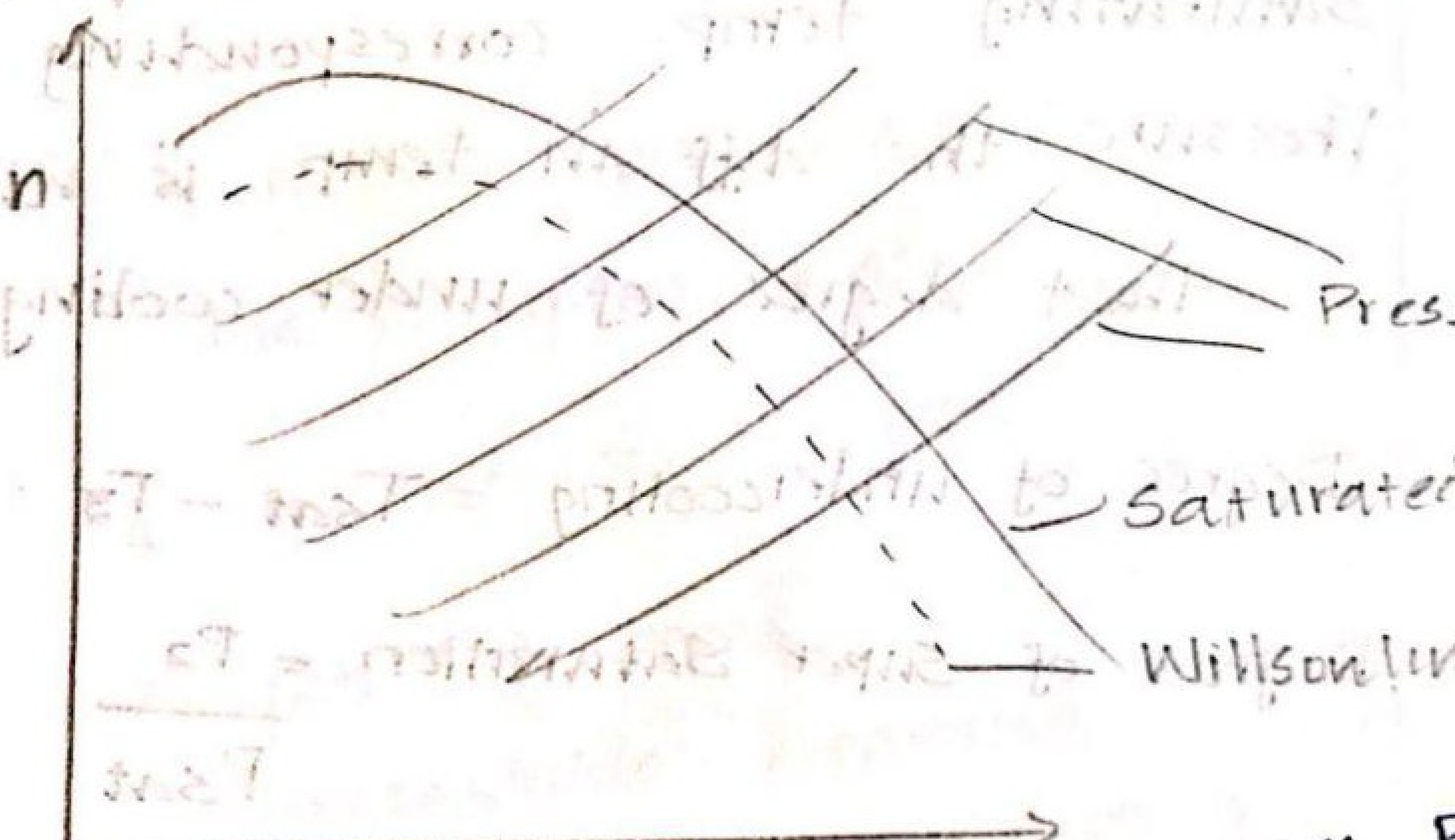
$$v_2 = 155 . \, m^3/kg$$

$$m = \frac{A_2 C_2}{v_2}$$

$$2 = \frac{A_2 \times 755.26}{1.55}$$

$$\boxed{A_2 = 0.00413 \, m^2}$$

the condensation
the expected rate. so the equal equ
b/w liquid and vapour phase is dela
and the steam continues to expan
a dry stage. The steam in such
set of condition is set to be supe
Saturated or Metastable



The Vapour between → Pressures P
is said to be super saturated
and this type of flow in the noz
is known as supersaturated flow

The flow is also called as super cooled flow because at a Pressure b/w P_2 and P_3 the tem of the vapour is always lower than saturating temp. corresponding to Pressure the diff in temp is know that degree of under cooling

Degree of undercooling $= T_{sat} - T_2$

Degree of super saturation $= \frac{P_2}{P_{sat}}$

cross sectional flow area with respe
to aspect ratio 3:1 · the expansion
may be consider as metastable
and friction is neglected

2) degree of undercooling and super saturation.

iven: (3) loss in avaialable heat drop due to irreversability

4) Increase in entropy

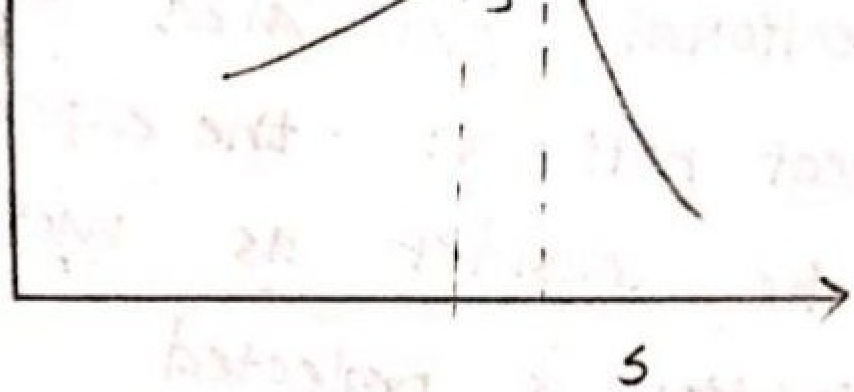
5) ratio of mass flow rate with metastable expansion to that if exp is in thermal equlibirium.

ven:

Super heated steam

$m = 5.2\ kg/s$

$10^5\ N/m^2$



Solution :

At inlet condition :

from steam tables $P_1 = 30$ bar, $T_1 = 350^\circ C$

$h_1 = 3115$ KJ/kg.

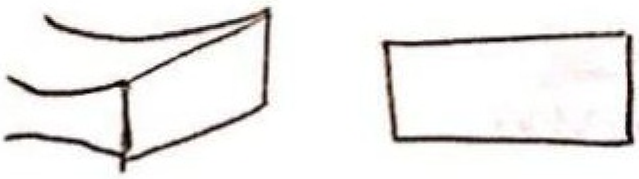
$s_1 = 6.7$ KJ/kgK

$v_1 = 0.09\ m^3/kg$.

(i) Dimension of the Nozzle

Rectangular cross section

Aspect ratio 3 : 1



for super heated steam $n = 1.3$

$$\frac{c_2}{2} = \frac{n}{n-1} P_1V_1\left[1 - \left(\frac{\quad}{P_1}\right)\right]$$

$$gz_1 + \frac{c_1^2}{2} + h_1 + Q = gz_2 + \frac{c_2^2}{2} + h_2 + QW$$

$$h_1 - h_2 = \frac{c_2^2}{2} - \frac{c_1^2}{2}$$

$$c_2 = \sqrt{2000\,(h_1 - h_2)}$$

$$c_2 = \boxed{432.7\ m/s}$$

$$P_1V_1^n = P_2V_2^n$$

$$V_2/V_1 = (P_1/P_2)^{1/n}$$

$$V_2 = V_1\,(P_1/P_2)^{1/n}$$

$$= 0.09\,(30/4)^{1/1.3}$$

$$V_2 = 0.424\ m^3/kg$$

$A_2 = l \times b$

$= 3x \times x$

$A_2 = 3x^2$

Total Nozzle = 6

$A_2 = 6 \times 3x^2$

$A_2 = 18x^2$

$0.00236 = 18x^2$

$$\boxed{x^2 = 0.0114\,m}$$

length $l = 3x = 3 \times 0.0114$

$l = 0.0343\,m$

length of the Nozzle = 0.0345 m

width of the Nozzle = 0.0114 m

) Degree of under cooling and Super saturati

Degree of undercooling = $T_{sat} - T_2$

$n-1$

$T_2 = 391.3\,K = 118.3$

from steam table saturation temp at 4

$T_{sat} = 143.6\,°c$

Degree of undercooling $= T_{sat} - T_2$

$= 143.6 - 118.3$

$= \overline{25.27}\,°c$

Degree of super saturation $= \dfrac{P_2}{P_{sat}}$

from steam table for saturation Pressure

$T_2 = 118.3\,°c$

$P_{sat} = 1.863$ bar

Degree of super saturation $= \dfrac{4}{1.863}$

$= 2.14$

Degree of undercooling = ~~118.3 °c~~