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Subject: TEGD (ME-601)

Unit: V

Topic:Nozzles & Condensers

## 1.1 INTRODUCTION

A nozzle is a flow passage of varying cross-sectional area in which the velocity of fluid increases and pressure drops in the direction of flow. Thus, in nozzle the fluid enters the variable cross section area duct with small velocity and high pressure and leaves it with high velocity and small pressure. During flow through nozzle the enthalpy drops and heat drop in expansion is spent in increasing the velocity of fluid. Similar to nozzle a duct with variable cross-section area will be called diffuser if the fluid gets decelerated, causing a rise in pressure along the direction of flow. Nozzles are generally used in turbines, jet engines, rockets, injectors, ejectors etc.

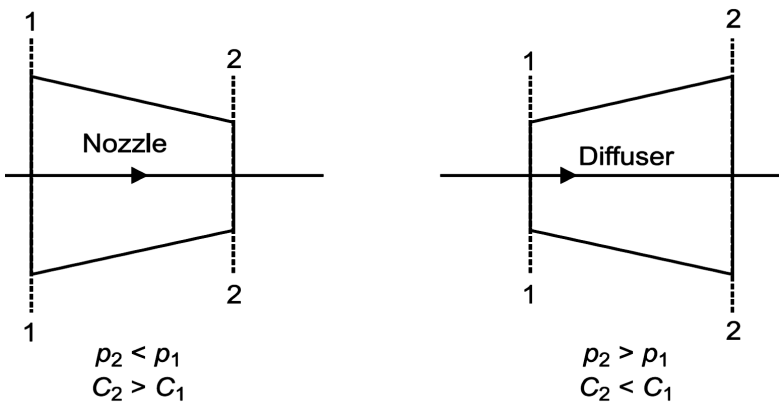



***Fig. 13.1*** *General arrangement in nozzle and diffuser*

Here in this chapter the one-dimensional analysis of nozzle has been considered.

Different types of nozzles, thermodynamic analysis and different phenomenon in nozzles are discussed ahead.

Momentum transfer across the control volume may be accounted for as,

$$[Time\ rate\ of\ momentum\ transfer\ across\ the\ control\ volume] = m'C$$

Newton's law states that resultant force $F$ acting on the control volume equals the difference between the rates of momentum leaving and entering the control volume accompanying mass flow.

Momentum equation says;

$$F = m_2' C_2 - m_1' C_1$$

Since at steady state, $m_2' = m_1'$ i.e. continuity equation being satisfied

$$F = m' (C_2 - C_1)$$

The resultant force $F$ includes forces due to pressure acting at inlet and exit, forces acting on the portion of the boundary through which there is no mass flow, and force due to gravity.

*Nozzles*

## 1.2 ONE DIMENSIONAL STEADY FLOW IN NOZZLES

Here one dimensional steady flow analysis in nozzle is carried out assuming the change in cross-sectional area and axis to be gradual and thermodynamic properties being uniform across planes normal to axis of duct. In general real flow through nozzle is not truly one-dimensional but this assumption offers fairly correct analysis. Flow through nozzle occurs at comparatively high velocities in many engineering applications and so exhibits changes in fluid density. Therefore, it is necessary to first look at the compressible flow preliminaries.

**COMPRESSIBLE FLOW PRELIMINARIES:** Let as consider compressible flow and obtain momentum equation for one dimensional steady flow.

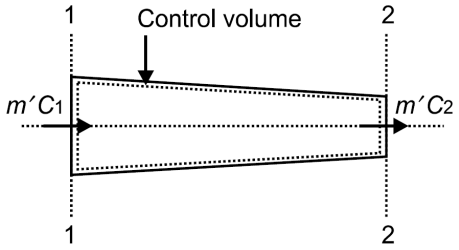
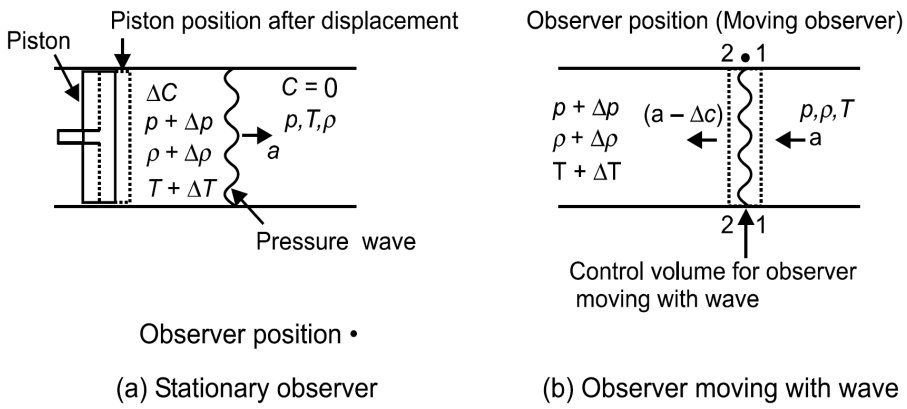



***Fig. 1.2***

The one dimensional steady flow through a duct is shown above. For control volume shown the principle of conservation of mass, energy and Newton's law of motion may be applied.

By Newton's law of motion, $F = m \times a$ where $F$ is the resultant force acting on system of mass '$m$' and acceleration '$a$'.

**PRESSURE WAVES AND SOUND WAVES:** Let us consider a cylindrical duct having piston on one end for generating the pressure wave. Figure 1.3 shows the arrangement for producing a pressure wave moving to right with velocity '$a$'. Sound wave is also a small pressure disturbance that propagates through a gas, liquid or solid with velocity '$a$' that depends on the properties of medium.




(a) Stationary observer (b) Observer moving with wave

***Fig. 1.3*** *Propagation of pressure wave (sound wave)*

Figure 1.3 shows how the generation of pressure wave causes displacement of fluid thereby causing rise in pressure, density and temperature by $\Delta p$, $\Delta\rho$ and $\Delta T$ in respect to the region on the right of wave (undisturbed region). In the undisturbed region say pressure, density, temperature and fluid velocity be $p, \rho, T$ and $C = 0$ respectively. Due to piston movement fluid velocity increases by $\Delta C$ and other properties also change by elemental values as shown. For analysing there are two approaches available as shown in Figs. 1.3 (a) and (b). One approach considers observer to be stationary and gas moving and second approach considers observer to be moving along with wave i.e. relative velocity of observer with respect to wave is zero.

Respective values of fluid velocity, wave propagation velocity, pressure, density and temperature are labelled on figure. For an observer at rest relative to wave (observer moving with wave) it seems as if the fluid is moving towards the stationary wave from right with velocity $a$, pressure $p$, density $\rho$ and temperature $T$ and moves away on left with velocity '$a - \Delta C$', pressure '$p + \Delta p$', density '$\rho + \Delta\rho$' and temperature '$T + \Delta T$'.

From conservation of mass, applying continuity equation upon control volume we get

$$m_1' = m_2' = m'$$

$$\rho.A.a = (\rho + \Delta\rho).A(a - \Delta C)$$

where $A$ is constant cross section area of duct.

$$\rho.A.a = (\rho.A.a) - (\rho.A.\Delta C) + (\Delta\rho.A.a) - \Delta\rho.A.\Delta C$$

Upon neglecting higher order terms and rearranging we get,

$$(\Delta\rho.a) - (\rho.\Delta C) = 0$$

or,

$$\Delta C = \frac{\Delta\rho.a}{\rho}$$

Applying momentum equation to the control volume;

$$(p.A) - \{(p + \Delta p).A\} = \{m'(a - \Delta C)\} - (m'.a)$$

$$-\Delta p.A = m'.(-\Delta C)$$

for mass flow rate $m'$ we can write, $m' = \rho.A.a$

so,

$$\Delta p.A = \rho.A.a.\Delta C$$

or,

$$\Delta C = \frac{\Delta p}{\rho.A}$$

Equating two values obtained for '$\Delta C$' we get

$$\frac{\Delta\rho.a}{\rho} = \frac{\Delta p}{\rho.A}$$

$$a = \sqrt{\frac{\Delta p}{\Delta\rho}}$$

Thus, velocity of wave propagation comes out as the square root of the ratio of change in pressure and change in density.

In case of sound waves the magnitude of changes in pressure, density and temperature are infinitesimal and so these may also be called as infinitesimal pressure wave. It is also seen that thermodynamic process occurring across an infinitesimal pressure wave may be considered nearly isentropic.

Therefore the velocity of sound can be given as square root of derivative of pressure with respect to density across the wave under isentropic conditions.

$$a = \sqrt{\left(\frac{\partial p}{\partial\rho}\right)_{s=constt}}$$

In terms of specific volume values; $d\rho = \frac{-dv}{v^2}$

$$a = \sqrt{-v^2\left(\frac{\partial p}{\partial v}\right)_{s=constt}}$$

Let us consider fluid to be a perfect gas following isentropic process given by $pv^k = constt$. Taking log of both sides and then partially differentiating we get,

$$\sqrt{\left(\frac{\partial p}{\partial v}\right)_s} = \frac{-k.p}{v}$$

Substituting in expression for sound velocity

$$a = \sqrt{k.pv}$$

For ideal gas,

$a = \sqrt{k\,RT}$ . In case of air, $a = \sqrt{\gamma RT}$

Using the velocity of sound and fluid velocity a non dimensional parameter called Mach number can be defined. Mach number is given by the ratio of velocity of fluid (object) to the velocity of sound. It is generally denoted by *M*.

$$M = \frac{C}{a}$$

Based upon Mach no. value flow can be classified as given below.

For

$M < 1$ flow is called subsonic flow.

$M = 1$ flow is called sonic flow.

$M > 1$ flow is called supersonic flow.

***Nozzle flow analysis:*** Let us consider one dimensional steady flow in nozzles. Let us take a varying cross-section area duct such that velocity increases and pressure decreases from inlet to exit.

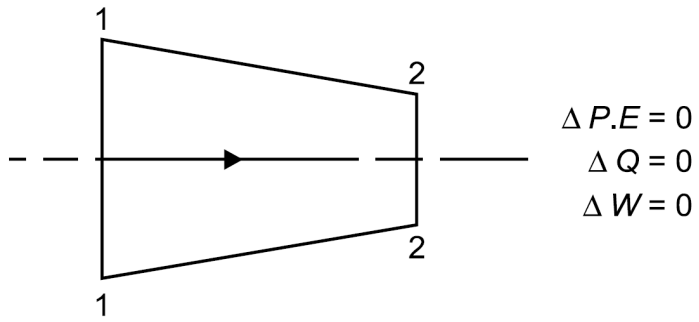



***Fig. 1.4***

From conservation of mass, upon applying continuity equation, it can be given that,

$$\rho . A . a = constant$$

Taking log of both the sides,

$$ln\, \rho + ln\, A + ln\, C = ln\, constant$$

Differentiating partially we get,

$$\frac{d\rho}{\rho} + \frac{dA}{A} + \frac{dC}{C} = 0$$

Let us now apply steady flow energy equation assuming no change in potential energy, no heat interaction and no work interaction during the flow across control volume.

S.F.E.E. yields, $dq = dh + dw + d(KE) + d(PE)$

Applying assumptions,

$$dh + d(KE) = 0$$

It can be rewritten for section 1 and 2 as, $KE = \frac{C^2}{2}$

or $$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

$$h_{o1} = h_{o2}$$

Stagnation enthalpy at section 1 = Stagnation enthalpy at section 2.

From differential form, $dh + d\left(\frac{C^2}{2}\right) = 0$

$$dh + 2.\frac{C.dC}{2} = 0$$

$$dh = -C.dC$$

From first and second law combined we know,

$$dh = Tds + vdp$$

Using the adiabatic flow considerations, *ds* = 0, so

$$dh = vdp = \frac{dp}{\rho}$$

Above shows that with increase or decrease in pressure along the direction of flow the specific enthalpy also change in same way.

From thermodynamic property relations pressure can be given as function of density and entropy i.e. $p = p(\rho, s)$.

or, $$dp = \left(\frac{\partial p}{\partial \rho}\right)_s . d\rho + \left(\frac{\partial p}{\partial s}\right)_\rho . ds$$

For isentropic flow considerations

$$dp = \left(\frac{\partial p}{\partial \rho}\right)_s . d\rho$$

We know from sound velocity $a = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s}$

so, $$dp = a^2 . d\rho$$

Combining two expressions for *dh* we get

$-C . dC = \frac{dp}{\rho}$. This shows that as pressure increases in direction of flow then velocity must decrease. Substituting from $dp$ as obtained above, it yields,

$$-C . dC = \frac{a^2 d\rho}{\rho}$$

Or $$-\left(\frac{dC}{C}\right) = \frac{a^2}{C^2}\left(\frac{d\rho}{\rho}\right) \quad \Rightarrow \quad \left(\frac{d\rho}{\rho}\right) = -\frac{C^2}{a^2}\left(\frac{dC}{C}\right)$$

Substituting above in the equation available from continuity equation,

$$\frac{d\rho}{\rho} + \frac{dA}{A} + \frac{dC}{C} = 0$$

Or $$\frac{dA}{A} = -\frac{dC}{C} - \frac{d\rho}{\rho}$$

$$\frac{dA}{A} = \frac{dC}{C}\left\{\left(\frac{C^2}{a^2}\right) - 1\right\}$$

As Mach no. $$M = \frac{C}{a}$$

$$\frac{dA}{A} = \frac{dC}{C}\{M^2 - 1\}$$

Using above relation the effect of area variation upon the flow can be seen in subsonic, sonic and supersonic flow regimes.

***Case 1***

*For subsonic flow i.e. M* < 1

***Nozzle:*** For positive velocity gradient i.e. velocity increases along the direction of flow as in case of nozzle $\frac{dC}{C}$ is +ve, it yields $\frac{dA}{A}$ as – ve. Negative area gradient means cross section area decreases along the direction of flow.

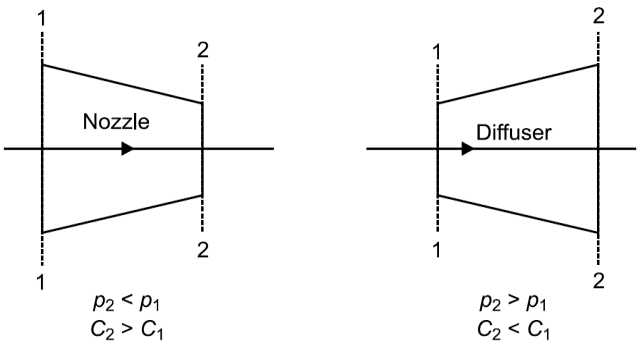



***Fig. 1.5***

***Diffuser:*** For negative velocity gradient i.e., $\frac{dC}{C}$ is $-ve$ the velocity decreases along the direction of flow as in case of diffuser, it yields $\frac{dA}{A}$ as +ve. Positive area gradient means duct has diverging cross section area along the direction of flow.

***Case 2: For supersonic flow i.e. M > 1***

*Nozzle:* For positive velocity gradient i.e. $\frac{dC}{C}$ being + ve, it yields $\frac{dA}{A}$ as + ve. It means that in supersonic flow the nozzle duct shall have diverging cross-sectional area along the direction of flow.

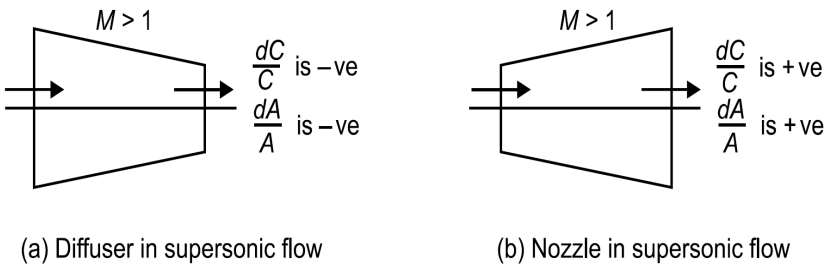



***Fig. 1.6***

***Diffuser:*** For negative velocity gradient i.e. $\frac{dC}{C}$ being $-ve$ it yields $\frac{dA}{A}$ as $-ve$. It means in supersonic flow the diffuser duct shall have converging cross-sectional area along the direction of flow.

From above discussion it can be concluded that

(i) Nozzle must be of convergent duct type in subsonic flow region and such nozzles are called subsonic nozzles or convergent nozzles.

(ii) Nozzle must be of divergent duct type in supersonic flow region and such nozzles are called supersonic nozzles or divergent nozzles.

(iii) For acceleration of fluid flow from subsonic to supersonic velocity the nozzle must be first of converging type till flow becomes sonic and subsequently nozzle should be of diverging type in supersonic flow. The portion of duct at which flow becomes sonic ($M$ = 1) and $dA$ is zero i.e. duct is constant cross-section area duct, is called throat. Thus in this type of flow from subsonic to supersonic the duct is of converging type followed by throat and a diverging duct. Such nozzles are also called convergent-divergent nozzles. Throat gives the minimum cross-section area in convergent-divergent nozzles.

Let us consider the expansion through a nozzle between sections 1 and 2. In nozzle the velocity of fluid is so high that there is no time available for heat exchange with the surroundings and the expansion may be considered adiabatic. Also the change in potential energy may be negligible if the elevation does not change from inlet to exit. Work done during flow is absent.

Application of steady flow energy equation yields,

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

Velocity at exit from nozzle:

$$C_2 = \sqrt{2(h_1 - h_2) + C_1^2}\text{, m/s}$$

For negligible velocity of fluid at inlet to nozzle, $C_1 = 0$

$C_2 = \sqrt{2(h_1 - h_2) +}$, m/s, where $h_1$ and $h_2$ are enthalpy in J/kg at sections 1 and 2 respectively.

Expansion of fluid on $p - v$ diagram is shown below.

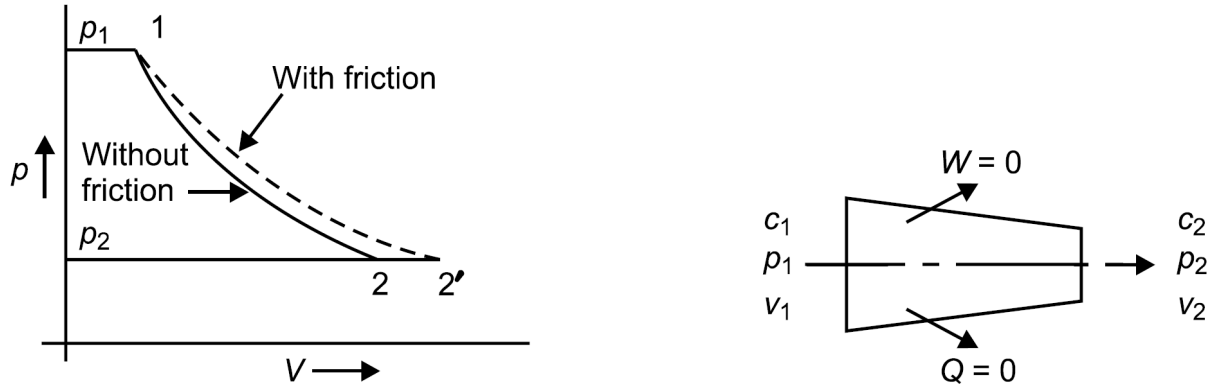



***Fig. 1.7*** P-V *diagram for flow through nozzle*

Expansion of gases on T-s diagram is as shown in Fig. 1.8.

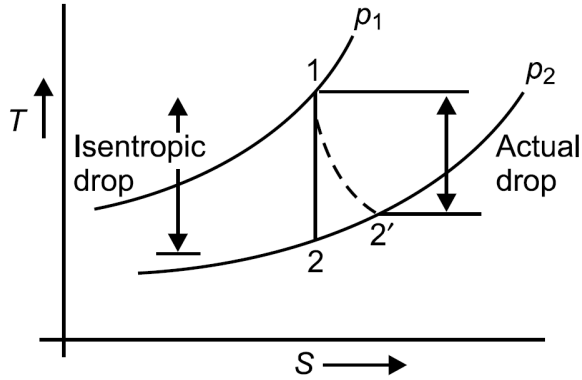



***Fig. 1.8*** T-s *diagram for flow through nozzle*

Expansion of steam on *T-s* and *h-s* diagram for superheated steam and wet steam is shown by 1–2 and 3–4 respectively under ideal conditions.

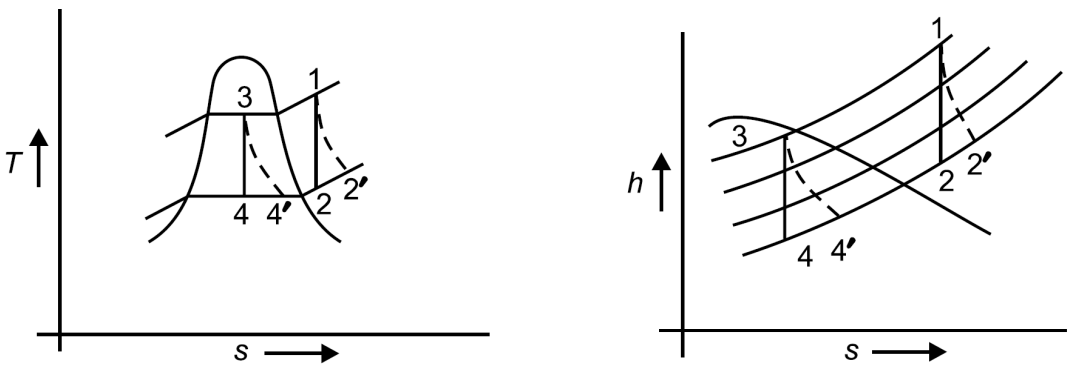



***Fig. 1.9*** *T-s and* h-s *representation for steam flow through nozzle*

In above representations the isentropic heat drop shown by 1–2 and 3–4 is also called 'Rankine heat drop'.

Mass flow through a nozzle can be obtained from continuity equation between sections 1 and 2.

$$m' = \frac{A_1\, C_1}{v_1} = \frac{A_2\, C_2}{v_2}$$

Mass flow per unit area; $\frac{m'}{A_2} = \frac{C_2}{v_2}$

From different from of S.F.E.E.

$$dq = dh + dw + d(K.E.) + d(P.E.)$$

or, $$dh + d(K.E.) = 0$$

$$du + pdv + vdp + d(K.E.) = 0$$

also as $$dq = du + pdv = 0,$$

so $$d(K.E.) = -\, vdp$$

or $$\frac{C_2^2 - C_1^2}{2} = -\int_{p_1}^{p_2} v.dp$$

For the expansion through a nozzle being governed by process $pv^n = constt$,

$$C_2^2 - C_1^2 = 2\left(\frac{n}{n-1}\right)p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right)$$

or,

Velocity at exit from nozzle $C_2 = \sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right) + C_1^2}$

For negligible inlet velocity, say $C_1 = 0$

Velocity at exit from nozzle

$$C_2 = \sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right)}$$

If the working fluid is perfect gas then $n = \gamma$ and for air $\gamma = 1.4$. However, if working fluid is steam a good approximation for $n$ can be obtained from some polytropic considerations. For steam being dry saturated initially and process of expansion occurring in wet region the index $n$ can be approximated as 1.135. For steam being initially superheated and expanded in superheated region the index $n$ can be approximated as 1.3.

Looking at mathematical expression for exit velocity it could be concluded that maximum exit velocity is possible only when fluid is expanded upto zero pressure. The maximum velocity is,

$$C_{max} = \sqrt{2\left(\frac{n}{n-1}\right)p_1v_1}$$

Mass flow rate,

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

Mass flow rate per unit area,

$$\frac{m'}{A_2} = \frac{\sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right)}}{v_2}$$

From expansion's governing equation, $p_1v_1^n = p_2v_2^n$

or, $$v_2 = \left(\frac{p_1}{p_2}\right)^{1/n}.v_1$$

$$\frac{m'}{A_2} = \frac{\sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right)}}{\left(\frac{p_1}{p_2}\right)^{1/n}v_1}$$

This expression for mass flow rate through nozzle depends upon inlet and exit pressures, initial specific volume and index of expansion. It has been seen earlier that the mass flow per unit area is maximum at throat and nozzle should be designed for maximum discharge per unit area. Thus there will be some value of throat pressure ($p_2$) which offers maximum discharge per unit area. Mathematically this pressure value can be obtained by differentiating expression for mass flow per unit area and equating it to zero. This pressure at throat for maximum discharge per unit area is also called 'critical pressure' and pressure ratio with inlet pressure is called 'critical pressure ratio'.

Let pressure ratio $\frac{p_2}{p_1} = r$, then mass flow per unit area can be re-written as;

$$\frac{m'}{A_2} = \left\{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left(r^{2/n} - r^{\frac{n+1}{n}}\right)\right\}^{1/2}$$

$$\frac{d\frac{m'}{A_2}}{dr} = \frac{d}{dr}\left\{2\left(\frac{n}{n-1}\right)\frac{p_1}{v_1}\left(r^{2/n} - r^{\frac{n+1}{n}}\right)\right\}^{1/2}$$

Here $p_1$, $v_1$ are inlet conditions and remain constant. Also $n$ being index of expansion remains constant so differentiating and putting equal to zero.

$$\frac{2}{n}.r^{\frac{2-n}{n}} - \left(\frac{n+1}{n}\right).r^{1/n} = 0$$

Or

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

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

or, Critical pressure ratio,

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

Let critical pressure at throat be given by $p_c$ or $p_t$ then,

$$\frac{p_c}{p_1} = \left(\frac{n+1}{2}\right)^{\frac{1-n}{n}} \quad \Rightarrow \quad \frac{p_t}{p_1} = \left(\frac{n+1}{2}\right)^{\frac{1-n}{n}}$$

Here subscript '$c$' and't' refers to critical and throat respectively.

While designing a nozzle the critical pressure ratio at throat is equal to the one obtained above.

Critical pressure ratio value depends only upon expansion index and so shall have constant value. Value of adiabatic expansion index and critical pressure ratio are tabulated ahead;

**Table 1.1:** Adiabatic expansion index and critical pressure ratio for selected fluids

| Fluid | Adiabatic expansion index, *n* | Critical pressure ratio $\frac{p_c}{p_1} = \left[\frac{2}{n+1}\right]^{\frac{n}{n-1}}$ |
|---|---|---|
| Wet steam | 1.135 ($n = 1.035 + 0.1x$, where $x$ is dryness fraction of wet steam) | 0.577 |
| Superheated steam | 1.3 | 0.545 |
| Air | 1.4 | 0.528 |

The maximum discharge per unit area can be obtained by substituting critical pressure ratio in expression for mass flow per unit area at throat section.

$$\frac{m'}{A_t} = \sqrt{\left[2.\left(\frac{n}{n-1}\right).\frac{p_1}{v_1}\left\{\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right\}\right]}$$

$$\frac{m'}{A_t} = \left[\left(\frac{2n}{n-1}\right).\frac{p_1}{v_1}.\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left\{\frac{n+1}{2} - 1\right\}\right]^{1/2}$$

$$\frac{m'}{A_t} = \left[n.\frac{p_1}{v_1}.\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]^{1/2}$$

Maximum discharge per unit area = $\sqrt{\left[n.\frac{p_1}{v_1}.\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$

For this maximum discharge per unit area at throat the velocity at throat can be obtained for critical pressure ratio. This velocity may also be termed as 'critical velocity'.

$$C_2 = \sqrt{2\left(\frac{n}{n-1}\right)(p_1v_1 - p_2v_2)}$$

At throat $$C_t = \sqrt{2\left(\frac{n}{n-1}\right)p_tv_t\left(\frac{p_1v_1}{p_tv_t} - 1\right)}$$

$$C_t = \sqrt{2\left(\frac{n}{n-1}\right)p_tv_t\left\{\left(\frac{p_t}{p_1}\right)^{\frac{1-n}{n}} - 1\right\}}$$

Substituting critical pressure ratio $\left(\frac{p_t}{p_1}\right)$

$$C_t = \sqrt{2\left(\frac{n}{n-1}\right)p_tv_t\left\{\left(\frac{n+1}{2}\right) - 1\right\}}$$

$C_t = \sqrt{n\,p_tv_t}$ Hence, critical velocity = $\sqrt{n\,p_tv_t}$

For perfect gas; $C_t = \sqrt{n\,RT_t}$

For $n = \gamma$, $C_t = \sqrt{\gamma.R.T_t}$ = $a$ = Velocity of sound.

Thus it can be concluded that for maximum discharge per unit area at throat the fluid velocity (critical velocity) equals to the sonic velocity. At the throat section mach no. $M$ = 1 for critical pressure ratio.

***For perfect gas:***

All the above equations obtained for the flow through nozzle can also be obtained for perfect gas by substituting $n = \gamma$ and $pv = RT$

Velocity at exit from nozzles

$$C_2 = \sqrt{2\left(\frac{\gamma}{\gamma-1}\right)(p_1v_1 - p_2v_2)}$$

$$C_2 = \sqrt{2\left(\frac{\gamma}{\gamma-1}\right)\mathrm{R}(T_1 - T_2)}$$

$$C_2 = \sqrt{2\,\mathrm{C_p}(T_1 - T_2)} \qquad \text{Since} \quad \mathrm{C_p} = \frac{\gamma \mathrm{R}}{\gamma-1}$$

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

Critical velocity at throat, $C_t = \sqrt{\gamma.R.T_t}$

Mass flow rate per unit area,

$$\frac{m'}{A_2} = \left[2\left(\frac{\gamma}{\gamma-1}\right)\frac{p_1}{v_1}\left\{\left(\frac{p_2}{p_1}\right)^{2/\gamma} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right\}\right]^{1/2}$$

Maximum discharge per unit area at throat for critical conditions,

$$\frac{m'}{A_t} = \left[\gamma.\frac{p_1}{v_1}.\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}\right]^{1/2}$$

Critical pressure ratio,

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

1.3 CHOKED FLOW

Let us consider a converging nozzle as shown in Fig. 1.10 with arrangement for varying back pressure. A valve is provided at exit of nozzle for regulating the back pressure at section 2-2. Let us denote back pressure by $p_b$. Expansion occurs in nozzle from pressure $p_1$ to $p_b$.

Initially when back pressure $p_b$ is equal to $p_1$ there shall be no flow through the nozzle but as back pressure $p_b$ is reduced the mass flow through nozzle increases. With the reduction in back pressure a situation comes when pressure ratio equals to critical pressure ratio (back pressure attains critical pressure value) then mass flow through nozzle is found

maximum. Further reduction in back pressure beyond critical pressure value does not affect the mass flow i.e. mass flow rate does not increase beyond its' limiting value at critical pressure ratio. Thus under these situations flow is said to be choked flow or critical flow.

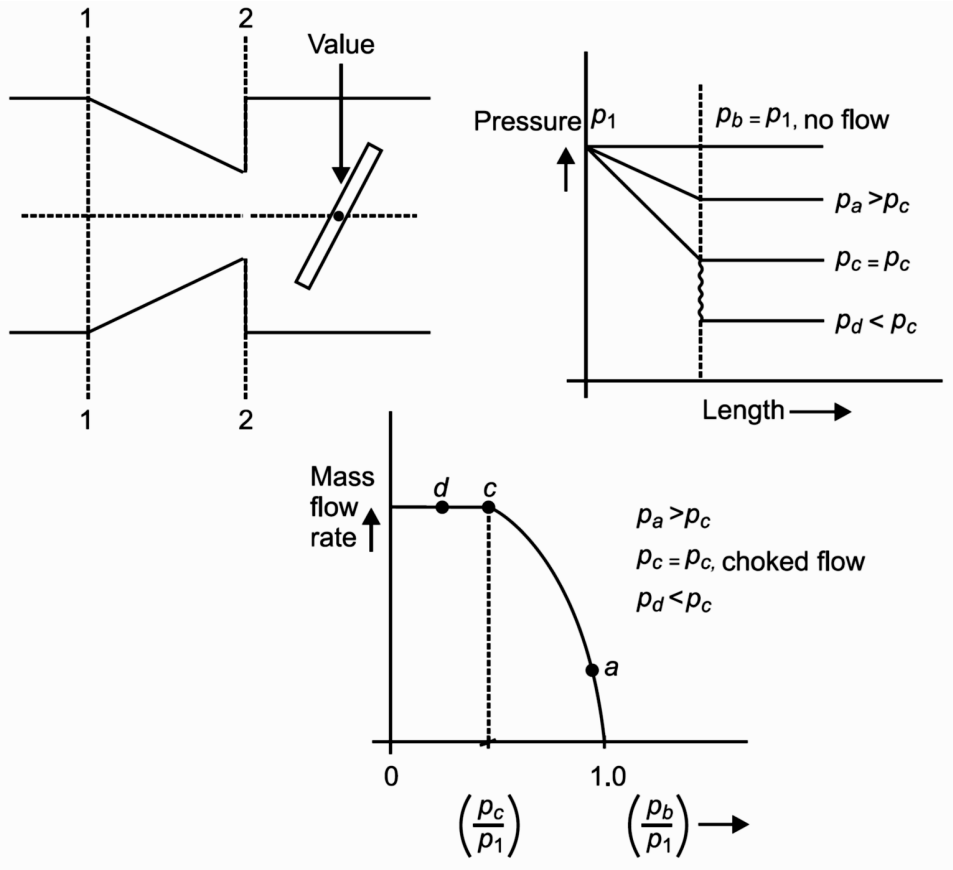



***Fig. 1.10*** *Flow through a convergent nozzle*

A nozzle operating with maximum mass flow rate condition is called choked flow nozzle. At the critical pressure ratio the velocity at exit is equal to the velocity of sound. If the back pressure is reduced below critical pressure then too the mass flow remains at maximum value and exit pressure remains as critical pressure and the fluid leaving nozzle at critical pressure expands violently down to the reduced back pressure value. Graphical representation of mass flow rate with pressure ratio and variation of pressure along length of nozzle explain the above phenomenon. State *a* refers to the state having back pressure more than critical pressure, state *c* refers to the state having back pressure equal to critical pressure and state *d* refers to state having back pressure less than critical pressure.

In case of convergent-divergent nozzle also the maximum mass flow through such nozzle shall be obtained when pressure ratio at throat section equals critical pressure ratio and velocity at throat equals sonic velocity. The cross-sectional area of throat decides the mass flow through nozzle for given inlet conditions.

## 1.4 OFF DESIGN OPERATION OF NOZZLE

Design operation of nozzle refers to the nozzle operating with pressure ratio equal to critical pressure ratio and maximum discharge rate per unit area then nozzle is said to be operating under design conditions. If the nozzle does not operate under design conditions then it is called off design operation of nozzle. Depending upon the back pressure value in reference to design value of pressure at exit of nozzle, the nozzle can be classified as under-expanding, over-expanding nozzles.

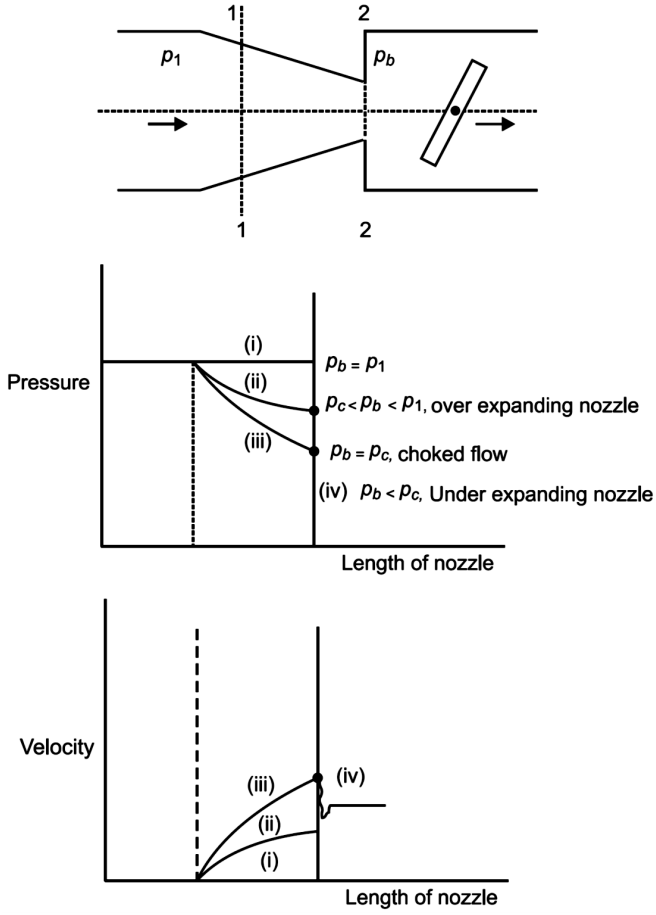



***Fig. 1.11*** *Off design operation of converging nozzle*

Nozzle is called under-expanding if the back pressure of nozzle is below the designed value of pressure at exit of nozzle. Nozzle is called over expanding if the back pressure of a nozzle is above designed value of pressure at exit of nozzle. Detail discussion about the off design operation of nozzle is given ahead for convergent and convergent-divergent nozzle.

***Convergent nozzle:*** Let us look at convergent nozzle having arrangement for varying back pressure. Fluid enters the nozzle at state 1, say pressure $p_1$. Variation of back pressure using value at exit of nozzle shows the pressure and velocity variation as shown in Fig. 1.11. Following significant operating states are shown here.

(i) When back pressure $p_b = p_1$, there is no flow.

(ii) When back pressure is such that back pressure is more than critical pressure i.e. $pc < p_b < p_1$, there is flow through nozzle. Here $p_c$ is critical pressure at exit. This operating state of nozzle having back pressure higher than critical pressure is called over expanding nozzle. In this over expanding nozzle the mass flow rate through nozzle is less than designed value.

(iii) When back pressure is such that back pressure is equal to critical pressure i.e. $p_b = p_c$. In this situation the mass flow through nozzle is maximum and nozzle is said to be choked.

(iv) When back pressure is further lowered such that back pressure is less than critical pressure i.e. $p_b < p_c$, the nozzle is said to be under expanding nozzle. In under-expanding nozzle there is no change in specific volume, velocity and mass flow rate through exit as that at choked flow state of nozzle. Since back pressure at exit is less than critical pressure while fluid leaves nozzle at critical pressure so fluid expands violently and irreversibly upto backpressure outside the nozzle.

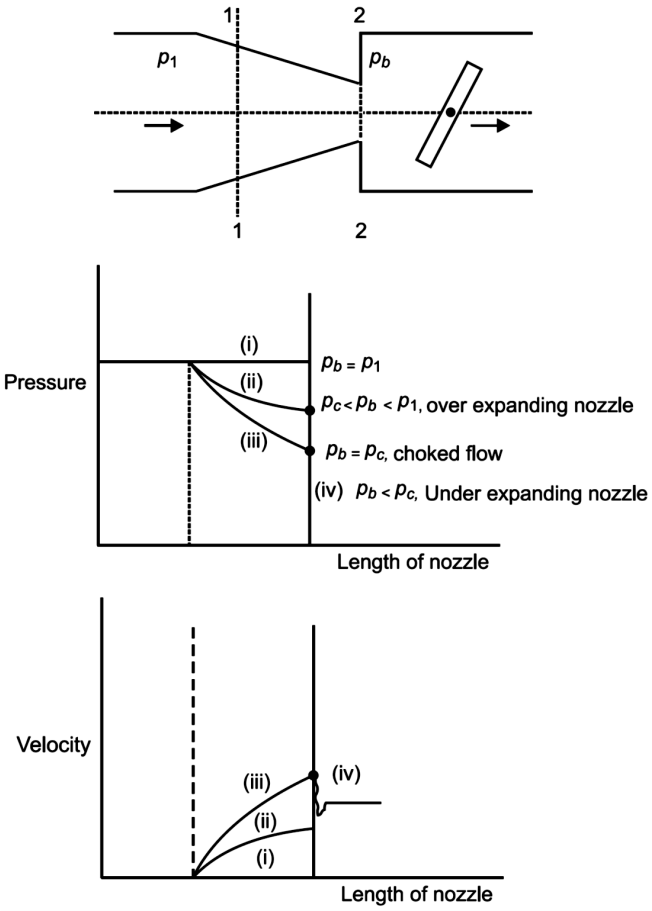



***Fig. 1.12*** *Off design operation of convergent-divergent nozzle.*

***Convergent-divergent nozzle:*** Converging-diverging nozzles are generally used for accelerating flow up to supersonic velocity. Arrangement for varying back pressure is as shown in Fig. 1.12.

Different operating regime of nozzle is obtained by varying back pressure using valve at exit. Let us start with back pressure equal to inlet pressure. Following operating states are shown here;

(i) When back pressure $p_b$ is equal to inlet pressure $p_1$ then there is no flow as shown by state $e$ on pressure-nozzle length plot.

(ii) When back pressure is reduced and is slightly below $p_1$, then there is some flow through nozzle shown by state $f$. The maximum velocity and minimum pressure occurs at throat section. With further reduction in back pressure from $e$ to $h$, the flow rate increases. Flow remains subsonic for back pressure between $e$ to $h$. In subsonic region the diverging portion of nozzle acts as diffuser, thereby increasing pressure and decreasing velocity in the direction of flow. In this regime the convergent-divergent nozzle is also used as venture-meter for flow rate measurement. Nozzle is said to be over expanding nozzle.

(iii) With further reduction in back pressure the throat pressure decreases and the throat velocity increases. Back pressure at which throat velocity becomes sonic and throat pressure equals to critical pressure $p_c$ is shown by state $h$. Here maximum velocity occurs at throat so the diverging portion of nozzle still acts as diffuser. Mass flow rate through nozzle has become maximum as the sonic flow conditions are obtained at throat. Thus it can be seen that flow through converging duct with subsonic velocity at inlet can never result in the velocity higher than sonic velocity and pressure less than critical pressure. This is choked flow state of nozzle.

(iv) Further lowering of back pressure less than critical pressure causes no effect on the flow in converging portion of nozzle and the pressure at throat remains equal to critical pressure and velocity at throat remains sonic. Also the flow rate through nozzle does not change. However, the nature of flow in diverging section of the duct changes as the breakage of flow occurs.

As pressure is reduced to $i$ and $j$ the fluid passing through the throat continues to expand and accelerate in diverging portion of nozzle. Flow velocity beyond throat is supersonic in diverging portion of nozzle. At the section downstream of throat there occurs

discontinuity in the flow due to abrupt irreversible increase in pressure accompanied by deceleration from supersonic to subsonic velocity. This discontinuity in flow is called shock and generally plane of discontinuity is normal to direction of flow so it may also be called normal shock. Flow through shock is of irreversible and steady adiabatic type. Beyond shock the fluid undergoes further isentropic deceleration as diverging section acts as a subsonic diffuser.

With further reduction in back pressure $p_b$ the shock moves downstream till it approaches nozzle exit plane and $p_b$ approaches the pressure given by state $k$. For the back pressure equal to pressure given by point $k$ i.e. $p_k$ the normal shock reaches at exit end of nozzle. Here flow within nozzle is isentropic, subsonic in converging portion, sonic at throat and supersonic in diverging portion. Due to shock the flow leaving nozzle becomes subsonic. These are all over expanding states of nozzle.

When back pressure is further lowered and back pressure becomes equal to exit plane pressure as at state $l$ i.e. $p_b = p_l$ , the isentropic expansion occurs throughout nozzle and no shock is found during flow. Fluid leaving nozzle is supersonic.

(v) Further lowering of back pressure below $p_l$ flow remains same as for pressure upto point $j$, but the back pressure being less than design pressure causes breaking of flow at downstream of nozzle exit. An abrupt expansion of irreversible type occurs at nozzle exit.

Irrespective of reduced back pressure the pressure at nozzle exit does not go below design pressure $p_l$ and mass flow rate and exit velocity also do not change. This operating state of nozzle is also called under expanding nozzle.

## 1.5 EFFECT OF FRICTION ON NOZZLE

In spite of the inside surface of nozzle being smooth the frictional losses always prevail due to friction between fluid and nozzle surface and friction within fluid itself. Due to friction prevailing during fluid flow through nozzle the expansion process through nozzle becomes irreversible. Expansion process since occurs at quite fast rate and time available is very less for heat transfer to take place so it can be approximated as adiabatic.

Frictions prevailing during flow through nozzle causes heat drop by about 10–15% and reduces the exit velocity. For the flowing fluid to be gas the T-S diagram representation is as follows:

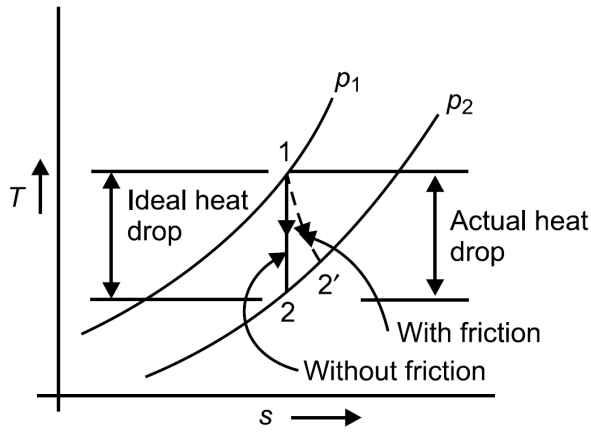



***Fig. 1.13*** *T-s representation for expansion through nozzle*

Non ideal operation of nozzle causes reduction in enthalpy drop. This inefficiency in nozzle can be accounted for by nozzle efficiency. Nozzle efficiency is defined as ratio of actual heat drop to ideal heat drop. Nozzle efficiency,

$$\eta_{Nozzle} = \frac{Actual\ heat\ dropIdeal\ heat\ drop}{Ideal\ Heat\ Drop}$$

$$\eta_{Nozzle} = \frac{h_1 - h_{2'}}{h_1 - h_2}$$

$$\eta_{Nozzle} = \frac{T_1 - T_{2'}}{h_1 - T_2}$$

In case of working fluid being steam the friction causes heating of steam flowing through nozzle thereby increasing dryness fraction. The volume of steam at exit also increases due to increase in dryness fraction. *T–s* and *h-s* representation of steam expanding through nozzle is given in Fig. 1.14.

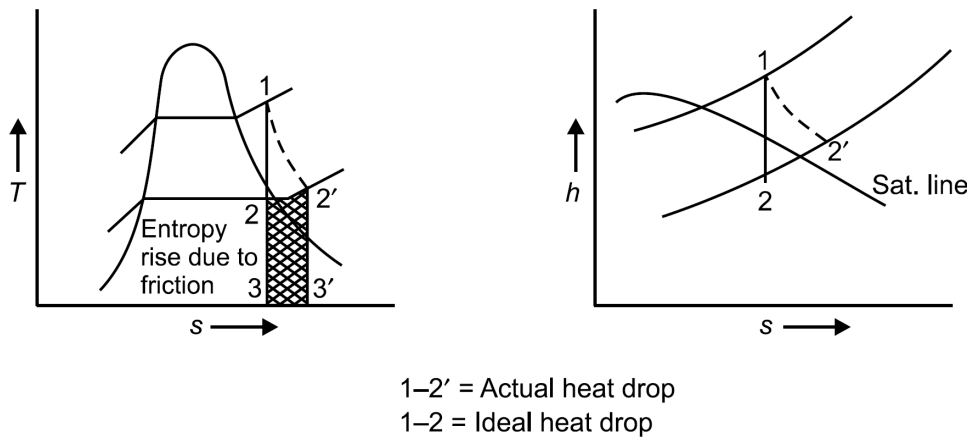



***Fig. 1.14*** T-s *and* h-s *representation for steam expanding through nozzle*

Due to friction the velocity at exit from nozzle gets modified by nozzle efficiency as given below.

Velocity at exit, $C_2 = \sqrt{2\,(h_1 - h_2) + C_1^2}$ , for no friction

In case of nozzle with friction the enthalpy drop, $(h_1 - h_{2'})$ gives velocity at exit as,

$$C_{2'} = \sqrt{2\,(h_1 - h_{2'}) + C_1^2}$$

or, $$(h_1 - h_2) = \frac{C_2^2 - C_1^2}{2}$$

$$(h_1 - h_{2'}) = \frac{C_{2'}^2 - C_1^2}{2}$$

Substituting in nozzle efficiency,

$$\eta_{Nozzle} = \frac{C_{2'}^2 - C_1^2}{C_2^2 - C_1^2}$$

For negligible inlet velocity i.e., $C_1 \approx 0$

Nozzle efficiency, $$\eta_{Nozzle} = \frac{C_{2'}^2}{C_2^2}$$

Thus it could be seen that friction loss will be high with higher velocity of fluid. Generally frictional losses are found to be more in the downstream after throat in convergent-divergent nozzle because of simple fact that velocity in converging section upto throat is smaller as compared to after throat. Expansion upto throat may be considered isentropic due to small frictional losses. Apart from velocity considerations the significantly high frictional loss in diverging portion of nozzle compared to converging portion can be attributed to the contact surface area. Length of converging section upto throat is quite small compared to length of diverging portion after throat as it has subsonic acceleration which can be completed in short length. Diverging section of nozzle is designed comparatively longer than converging section so as to avoid flow separation due to adverse duct geometry (diverging type). Turbulence losses are also significant in diverging portion compared to converging portion. Due to the different factors discussed above the frictional losses are found to be more in diverging portion compared to converging portion.

Normally angle of divergence in divergent portion is kept between $10^0$ and $25^0$ so as to avoid flow separation. But small divergence angle causes increase in length of diverging portion therefore increasing frictional losses. Thus a compromise should be striked in selecting angle of divergence as very small angle is desirable from flow separation point of view but undesirable due to long length and larger frictional losses point of view. Length of diverging portion of nozzle can be empirically obtained as below

$$L = \sqrt{15 \cdot A_t}$$

where $A_t$ is cross-sectional area at throat.

While designing the nozzle parameters due care should be taken for smoothness of nozzle profile, surface finish for minimum friction and ease of manufacturing etc. Thus finally, it can be concluded that nozzle efficiency depends upon nozzle material, size and shape of nozzle, angle of divergence, nature of fluid flowing and its properties etc.

**Coefficient of velocity:** The 'coefficient of velocity' or the 'velocity coefficient' can be given by the ratio of actual velocity at exit and the isentropic velocity at exit. Thus it measures the amount of deviation from ideal flow conditions. Mathematically,

$$Coefficient\ of\ velocity\ = \frac{C_{actual\ at\ exit}}{C_{isentropic\ at\ exit}}$$

Coefficient of discharge: The 'coefficient of discharge' or 'discharge coefficient' is given by the ratio of actual discharge and the discharge during isentropic flow through nozzle. Mathematically,

$$Coefficient\ of\ discharge\ = \frac{m_{actual}}{m_{isentropic}}$$

Here *m* refers to discharge rate.

### 1.6 SUPERSATURATION PHENOMENON IN STEAM NOZZLES

The phenomenon of supersaturation in steam nozzles is also called as supersaturated flow or metastable flow in steam nozzle. When superheated steam flows through a nozzle and expands upto the back pressure such that exit state of steam lies in wet region, then during expansion steam vapours expand isentropically and slowly get condensed up to exit state. During such expansion steam also passes across saturated steam line or saturation line having unity dryness fraction. Thus it is obvious that expansion of steam is accompanied by simultaneous state change from superheated state to wet state. At every point along expansion line there exist a mixture of vapour and liquid in equilibrium. An expansion process starting at 1 goes up to state 2 in thermal equilibrium as shown on *T-S* and *h-s* diagram.

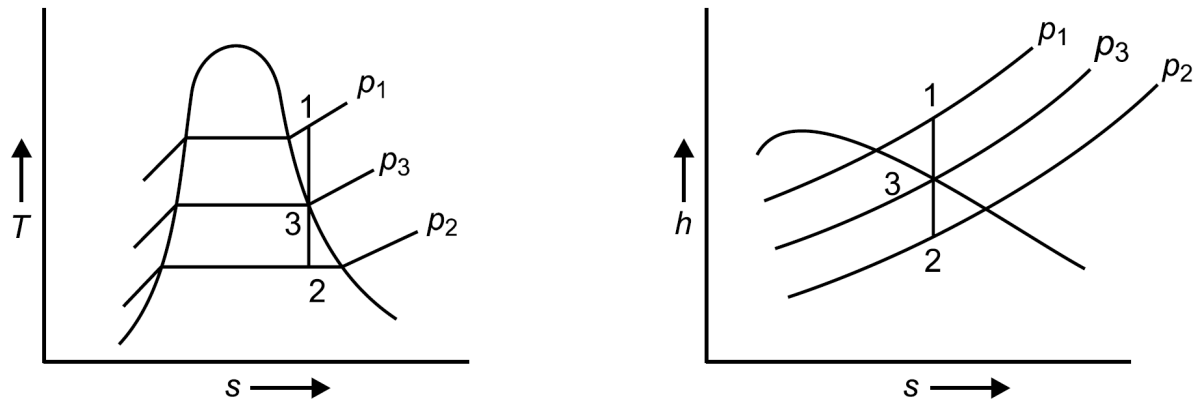



***Fig. 1.15*** *Expansion of steam on* T-s *and* h-s *diagram under equilibrium*

Superheated steam undergoes continuous change in state and becomes dry saturated steam at state 3 and subsequently wet steam leaving steam turbine at state 2. Sometimes expansion of steam occurs in metastable equilibrium or in equilibrium in which change of steam state could not maintain its pace with expanding steam. This phenomenon in which change of steam state could not occur simultaneously with expanding steam in nozzle is called phenomenon of supersaturation and flow is called supersaturated flow or metastable flow.

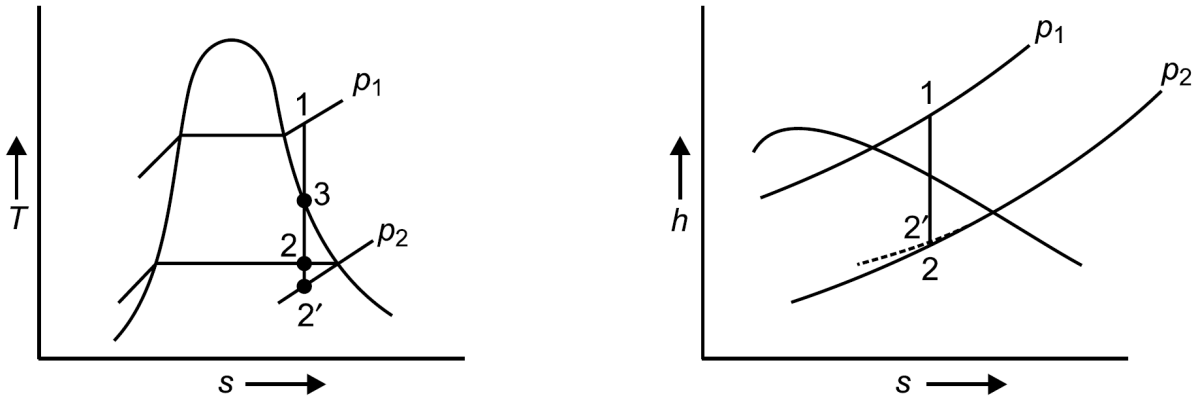



***Fig. 1.16*** *Metastable flow through steam nozzle*

In supersaturated flow the condensation of steam lags behind the expansion and so steam does not condense at the saturation temperature corresponding to the pressure. It may be understood as the shift in state 2 from 2 to 2' i.e. condensation gets extended up to 2'. The dry saturated steam state which should be attained at state 3 cannot be realized at 3, but below 3 on vertical expansion line 1–2'. This delayed phase transformation of steam causing supersaturation phenomenon may be attributed to the following.

(i) Steam flow through nozzle may be so fast that sufficient time is not available for heat transfer to take place and so the phase change lags behind the expansion. Generally time available may be of the order of $10^{-2}$ second for steam to flow through nozzle along with its condensation.

(ii) Also the condensation of steam may have inherent requirement of nuclei of condensation which act as initiators for condensation. These nuclei of condensation may be provided by foreign particles, solid boundary etc. In the absence of nuclei of condensation the phase change of steam may get delayed and lags behind.

These could be the factors responsible for supersaturation. Phenomenon of supersaturation or metastable equilibrium continues up to generally 94–95% dryness fraction. Beyond this the condensation of steam occurs suddenly at very fast rate irreversibly and the

expansion process attains stable equilibrium. The locci of points up to which metastable equilibrium is observed is called Wilson line.

Law of expansion for supersaturated flow is considered as $pv^{1.3} = constant$. Phenomenon of supersaturation causes increase in discharge by 2–5% because of increase in density at throat and also the heat drop gets slightly reduced thereby causing reduced velocity at exit.

Supersaturation causes slight increase in dryness fraction and entropy.

Figure 1.17 shows the supersaturated flow and Wilson line.

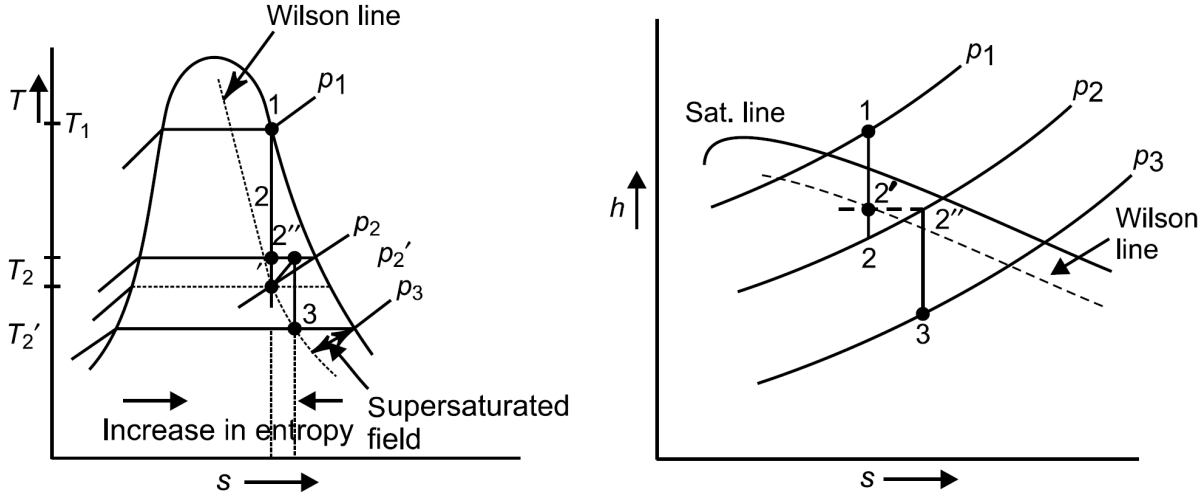



***Fig. 1.17*** *Supersaturated flow on* T-s *and* h-s *diagram.*

Region between saturation line and Wilson line is called supersaturated field. Here 1–2' line shows isentropic process. In the absence of supersaturated flow expansion occurs as 1–2 while with metastable flow it gets extended up to 1–2' as shown on *T–s* diagram. Meta stable equilibrium gets settled and stable equilibrium is attained as shown by 2'–2" and then normal expansion in stable equilibrium continues from 2" to 3.

Thus it is obvious that in supersaturated flow the expansion occurs as if there is no saturated steam line and state 2' lies on the extended constant pressure line.

The temperature at 2' is less than saturation pressure corresponding to $p_2$ due to excess kinetic energy of steam at the cost of sensible heat.

Metastable flow is characterized by a parameter called "degree of supersaturation" and "degree of undercooling". 'Degree of supersaturation' refers to the ratio of saturation pressures corresponding to temperatures of states in stable equilibrium and metastable equilibrium (i.e. saturation pressures corresponding to 2 and 2' states). Degree of super saturation has value more than unity. 'Degree of undercooling' refers to the difference of two

temperatures i.e. saturation temperature at state in stable equilibrium and temperature of the state in unstable equilibrium.

## 1.7 STEAM INJECTOR

Steam injector refers to the device for injecting water into boiler using steam which may be available from boiler or exhaust steam from engine.

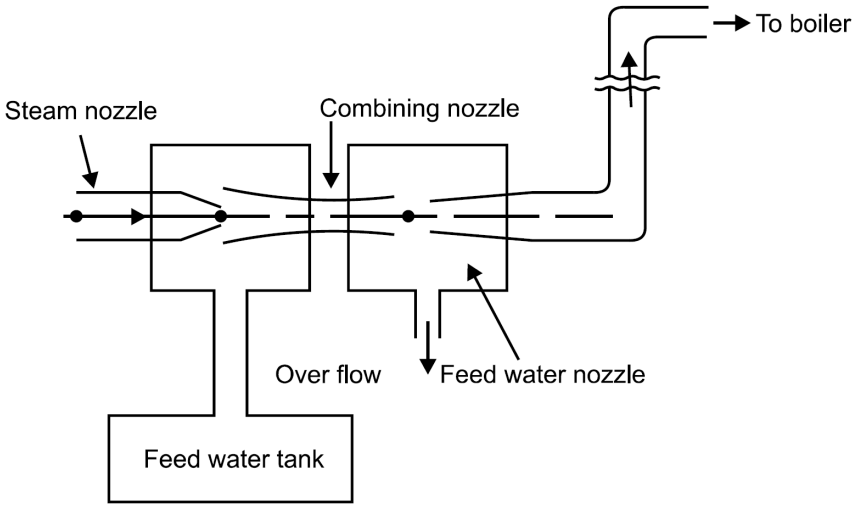



***Fig. 1.18*** *Steam injector*

Schematic for steam injector is shown in Fig. 1.18. Here high pressure steam enters a converging steam nozzle and leaves with high velocity and low pressure. Water gets entrained from feed tank and is carried by steam through combining nozzle. In due course steam gets condensed and the resulting mixture enters the divergent feed water nozzle where kinetic energy of water gets transformed into pressure head. Pressure head available in feed water nozzle is sufficiently above boiler pressure so that water can be fed to the boiler. Surplus water, if any gets discharged from over flow. The steam injector works on its own. It may be noted that the potential energy removed from live steam is many times more than the potential energy returned.

$$Potential\ energy\ removed\ from\ live\ steam\ =\ Boiler\ steam\ pressure \times Volume\ of\ steam.$$

$$Potential\ energy\ returned\ =\ Boiler\ pressure\ \times Volume\ of\ condensate\ and\ boiler\ feed$$

Difference in the potential energy exists due to large decrease in volume as steam condenses and this difference is only used for pumping water.

## 5A.1 INTRODUCTION

Condenser is one of the essential components of steam power plants as it facilitates condensation of steam at given conditions with minimum expenditure of energy and minimum loss of heat and finally gives condensate which can be recirculated by feed pump to boiler for steam generation. Condenser generally operates at pressure less than atmospheric pressure. In the steam power plant the use of condenser permits expansion in steam turbine even up to less than atmospheric pressure and subsequently condensing steam to yield condensate for recirculation thus improving plant efficiency and output. Expansion in steam turbine/engine cannot be extended to pressures less than atmospheric in the absence of condenser.

"Condenser can be defined as device used for condensation of steam at constant pressure; generally pressure is less than atmospheric pressure". Condenser is thus a closed vessel which is generally maintained at vacuum and cold fluid is circulated for picking heat from steam to cause its condensation. Use of Condenser offers advantages such as hotter feed water for being sent to boiler', 'removal of air and non condensable dissolved gases from feed water', 'recovery of condensate reduces treated water requirement', 'expansion up to sub atmospheric conditions and capital cost is reduced by recycling of feed water' etc. Increase in expansion work due to use of condenser is shown in Fig. 5A.1 on *p-V* diagram.

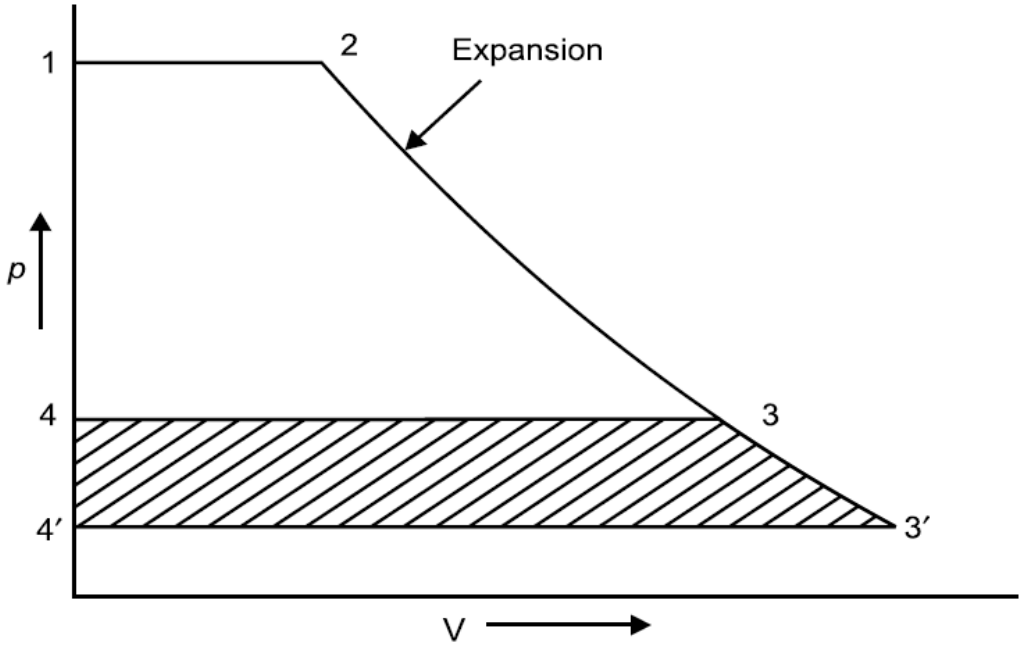



***Fig. 5A.1*** p-V *diagram showing how condenser increases work output in steam engine*

Steam power plant employing condenser and the condensing plant are shown in Fig. 5A.2.

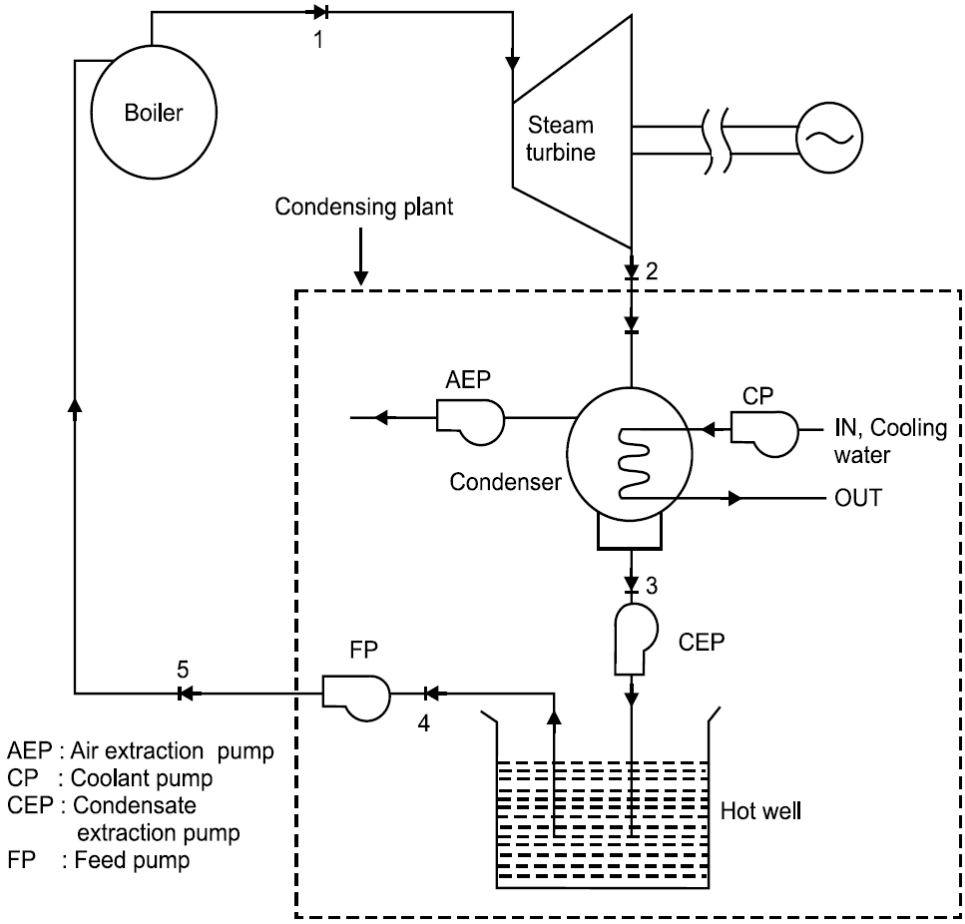



***Fig. 5A.2*** *Schematic for steam power plant having condensing plant*

Discharge from steam turbine passes into condenser where it is condensed using cooling water being circulated employing coolant pump. Condensate being at pressure less than atmospheric pressure is to be sucked out using condensate extraction pump. Condensate is extracted and sent to hot well from where it is pumped to boiler using feed pump. Dissolved gases and air etc. if any are extracted out from condenser using air extraction pump. This air or vapour may be present because of air leaking into vacuum system and air coming with steam. Cooling water for supply to condenser is taken either from some river or from cooling tower. Cooling water requirement may be up to 100 kg water per kg of steam or even more depending upon the type of condenser and its capacity. Cooling tower cools the hot cooling water leaving condenser to get cooled by evaporation of water and heat exchange with air.

Water evaporated or lost in cooling tower is compensated by the makeup treated water available from feed water treatment plant.

## 5A.2 CLASSIFICATION OF CONDENSER

Condenser can be broadly classified on the basis of type of heat exchange i.e. direct or indirect contact condensers.

(i) Direct contact type or mixing type or Jet condenser

(ii) Indirect contact type or Non-mixing type or Surface condenser

(iii) Evaporative condenser

Jet condensers have direct contact between steam and cooling fluid thereby causing contamination of condensate. Surface condensers have indirect heat exchange through metal interface and the two fluids do not come in direct contact to each other. An evaporative condenser use evaporation of water for heat extraction and is well suited for dry weather so that evaporation is not difficult. Due to direct contact of two fluids the circulating water requirement is much less in jet condenser as compared to other types of condensers. Space requirement and size of condenser etc. are also less with jet condensers.

Surface condenser is advantageous over direct contact type condensers because any type of cooling fluid can be used in it and also there is no scope of contamination etc. Different types of condensers are discussed ahead.

**(i) Jet condenser:** In jet condenser the steam to be condensed and cooling water are intimately mixed by breaking up of water in the form of spray and allowing small sized water particles to fall down through the body of steam. The water may also be discharged out through suitably shaped nozzles into body of steam. Thus it is desired to atomize water into small sized particles so that increased surface area is available for heat exchange between hot and cold fluid. Number of arrangements for flow of steam and water are available such as; counter flow type having steam entering from bottom and flowing upwards while water enters from top and falls downwards with air pump connected on top where air is colder etc. Jet condenser may be further classified based on relative movement of two fluids, and based on arrangement used for removal of condensate.

Based on relative moment of two fluids jet condenser can be,

(a) Counter flow jet condenser

(b) Parallel flow jet condenser

Based on arrangement for removal of condensate jet condenser can be,

(a) Low level jet condenser

(b) High level jet condenser

(c) Ejector condenser

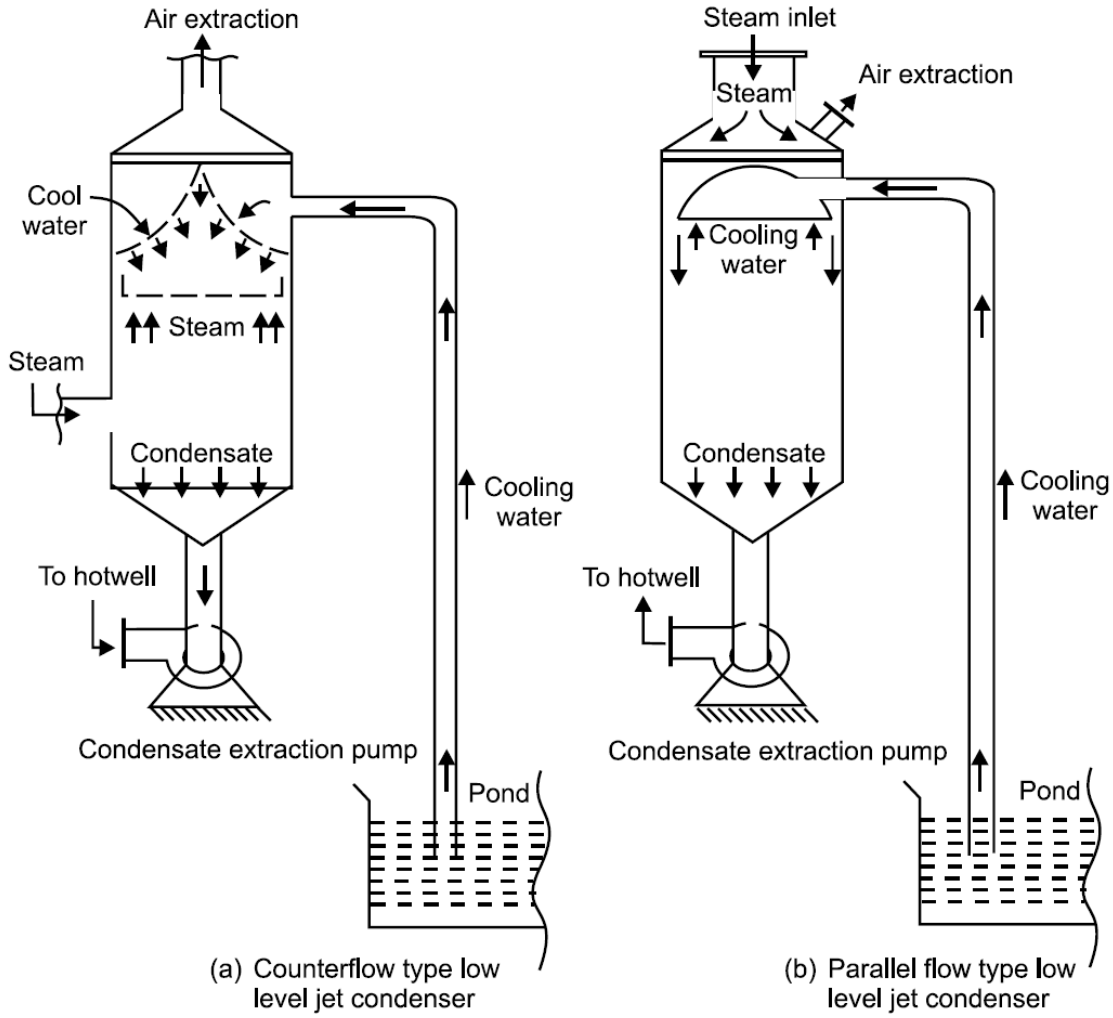



***Fig. 5A.3*** *Schematic of low level jet condenser*

**(a) Low level jet condenser:** Low level jet condenser is the one which is placed at low level such that vacuum inside condenser draws cooling water into condenser from river/pond/cooling tower.

Difference between atmospheric pressure (at which cooling water is available) and condenser pressure causes flow of cooling water from cooling water reservoir to condenser i.e. $(p_{atm} - p_{cond})$. Flow of steam and cooling water could be parallel flow or counter flow type. Counter flow type and parallel flow type low level jet condensers are shown in Fig. 5A.3. There is provision for extraction of air and dissolved gases from top of condenser by using air extraction pump. Condensate extraction pump is used for taking out condensate from condenser and sending it to hot well.

Cooling water supplied to jet condenser has generally a large percentage of dissolved air which gets liberated due to atomization of water, vacuum and heating

of water and is extracted out. Low level jet condenser suffers from inherent drawback that in the event of failure of condensate extraction pump condenser shall be flooded with cooling water.

**(b) High level jet condenser:** High level jet condenser is the one which is placed at a height more than that of water and water is to be injected into condenser using a pump and the condensate will flow out of condenser because of gravity. Here no condensate extraction pump is required; instead pump is required for pumping water up to condenser inlet. High level jet condenser is also called as 'barometric condenser'. High level jet condenser is placed at suitable height depending upon efficient drainage and capacity of sump (hot well) into which tail pipe of condenser discharges out. Mathematically, it could be said that jet condenser placed above hotwell by 10.36 m shall be high level jet condenser or barometric condenser. High level jet condenser may also be of counter flow type or parallel flow type depending upon the direction of flow of steam and cooling water. Figure 5A.4 shows counter flow high level jet condenser.

High level jet condenser do not pose problem of flooding of condenser in the event of failure of pump as it is in case of low level jet condensers. But high level jet condensers are costlier than low level jet condenser. Also there is loss of vacuum between turbine and condenser in case of high level jet condenser.

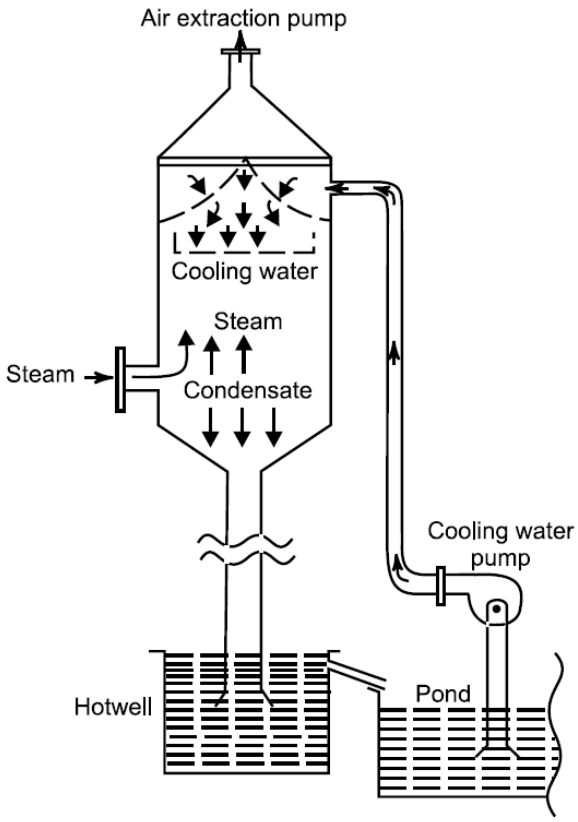



***Fig. 5A.4*** *Schematic of high level condenser*

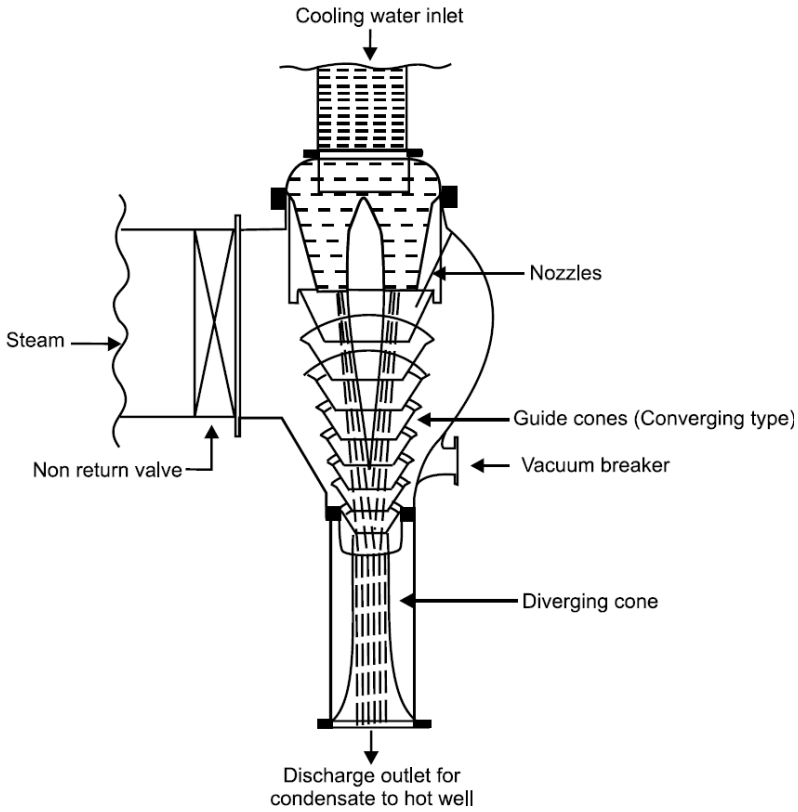



***Fig. 5A.5*** *Ejector condenser*

**Ejector condenser:** Ejector condenser has water jet discharging through the series of guide cones which guide steam on to the surface of water jet. Discharge of water through these convergent nozzles causes partial vacuum due to conversion of potential energy into kinetic energy. Subsequently water jet enters the diffuser nozzle where kinetic energy is converted into the pressure head and water is discharged against the vacuum pull. Ejector condensers are well suited for moderate vacuum only.

Steam is injected in condenser with non return valve in between and is condensed by the mixing with cooling water. Condensation of steam further increases vacuum.

Ejector condenser does not require air pump because of air entraining effect of water jet itself. Here condensing jet has number of nozzles arranged concentrically and have their axis inclined at such an angle that water jet assumes the form of inverted cone. Around the water jet the guide cones are arranged with increasing area from bottom to top. Water will be colder in upper part of condensing cone as compared to lower down. The temperature difference between steam and water at top will be greater than at lower end and so the condensation is greatest at top and gradually diminishes to zero at bottom.

In case of failure of cooling water supply water may be sucked from hot well to go into steam pipe, but this is prevented by non-return valve in steam supply line.

**(ii) Non mixing type or surface condensers:** Surface condensers are the most common types of condenser and offer great advantage in terms of no contamination of feed water. In these condensers the steam to be condensed and cooling fluid (water) do not come in contact with one another, instead the heat transfer occurs between two fluids through surface in between. Generally, cooling water flows through the pipes/tubes and steam surrounds them. These condensers are preferred in the locations where large quantity of poor quality cooling fluid (impure water) is available and condensate is to be re-circulated. Surface condensers can be classified based on number of passes of condenser i.e. single pass or multipass. Number of times the cooling water crosses any transverse section is called a pass.
Surface condensers may be of ‘down flow type’ or ‘central flow type’ depending on the type of flow of condensate and tube arrangement. Typical surface condenser

having two passes, down flow type and central flow type arrangement are shown in Fig. 5A.6.

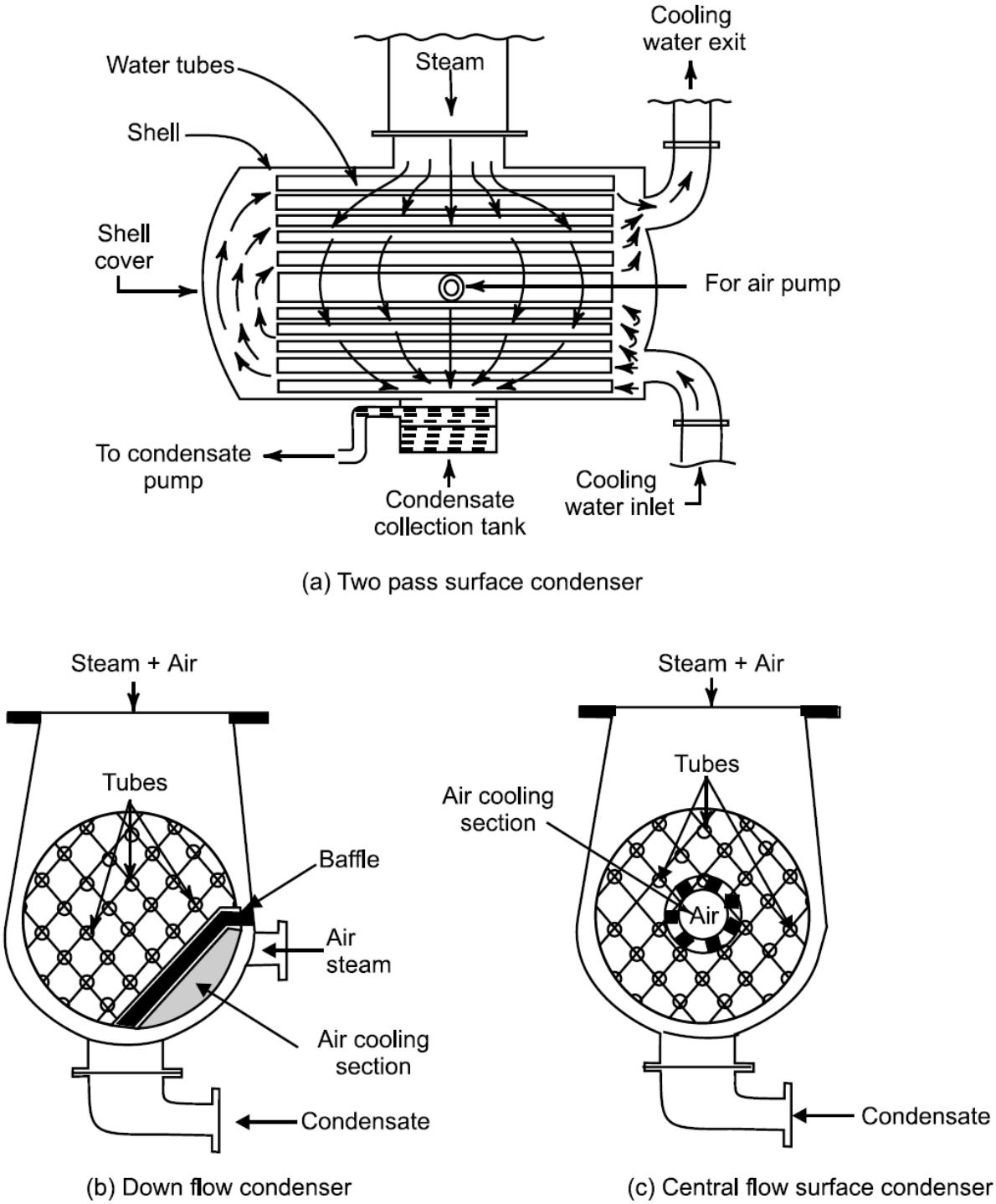



***Fig. 5A.6*** *Surface condenser*

Two pass surface condensers has cooling water entering from one end and coming out after twice traversing through the tubes (generally, brass) containing water and surrounded by steam to be condensed. Condensate gets collected at bottom and is subsequently sucked by condensate extraction pump. Steam is admitted from the top. Cooling water may be picked directly from river/pond/cooling

tower. For extraction of air the provision is made for air pump. Thus, this type of condenser has three pumps i.e. one for circulating cooling water, second for condensate extraction and third for air extraction.

In surface condenser the space occupied by tubes in shell is about 10% of shell volume. Steam is not passed through the tubes because at this steam pressure the specific volume of steam is large requiring large number of tubes.
Down flow condenser has steam and air entering from top and flowing downwards across the bundle of tubes having cooling water flowing through them. Air is extracted from bottom and before being handled by air pump it is flown through air cooler so as to reduce the temperature of air. Low temperature of air enhances the air handling capacity of pump. With the flow of steam down and simultaneous heat exchange the condensate is taken out by condensate extraction pump.

Central flow condenser has air cooling section in the centre of condenser. Steam enters from top and passes over the tube banks of similar type as in case of down flow condenser. As air is being sucked from centre so the flow of steam is radially inwards towards the centre. During this flow steam passes over tubes. Condensate is collected from bottom. In this type of condenser there is better contact between steam and tubes because of radial flow of steam in whole of condenser, thus arrangement is better as compared to down flow condenser.

In different designs of condenser it is always attempted to have maximum heat transfer between two fluids. Also air extraction should be done effectively.
Thus designer of condenser should keep following things in consideration for making a better design surface condenser.

(i) Steam should be uniformly distributed over cooling water tubes. i.e. cooling surface.

(ii) Distribution of steam should be such that there is minimum pressure loss.

(iii) Number of tubes should be minimum. Water must be flown inside tubes and steam should surround them.

(iv) Tubes should be cleaned from inside and outside both. Although on external surface the steam surrounding tubes prevents deposition. For internal cleaning of tubes mechanical or chemical means of cleaning be used at frequent intervals.

(v) Leakage of air into condenser (due to vacuum) should be prevented as it reduces the work output. Also this reduces the heat transfer rate. Even if there is leakage of air, arrangement should be made for quick and effective removal of air with minimum work input.

(vi) Air should be cooled to maximum extent inside condenser before being thrown out as this shall cause condensation if possible within condenser and thus reduce loss of condensate. Also the cool air shall enhance air handling capacity of pump.

(vii) Rate of circulation of cooling water should be such that the range of temperature variation in cooling water lies near the optimum temperature range. Generally, the cooling water temperature rise is limited to 10°C for having maximum heat exchange between two fluids.

(viii) Material of tubes is generally taken as brass. Tube material should be such as to offer maximum heat transfer rate i.e. high thermal conductivity. Generally, surface condensers are bulky and require large space.

(ix) Cost of surface condenser should be kept low. Capital cost, running and maintenance cost should be maintained as low as possible. Generally, these costs are high in case of surface condenser as compared to other types of condensers.

**(iii) Evaporative condenser:** Evaporative condensers are generally used where the availability of water is very poor. Figure 5A.7 shows the schematic of such type of condenser where water falls from top through the nozzles over the condenser coil. Water picks up heat from the steam flowing through condenser coil and gets warmed up. This water is recirculated by circulation pump. Air flow inside condenser is maintained by using exhaust fan. This flow of air across condenser coil may be natural or forced to enhance the cooling rate.

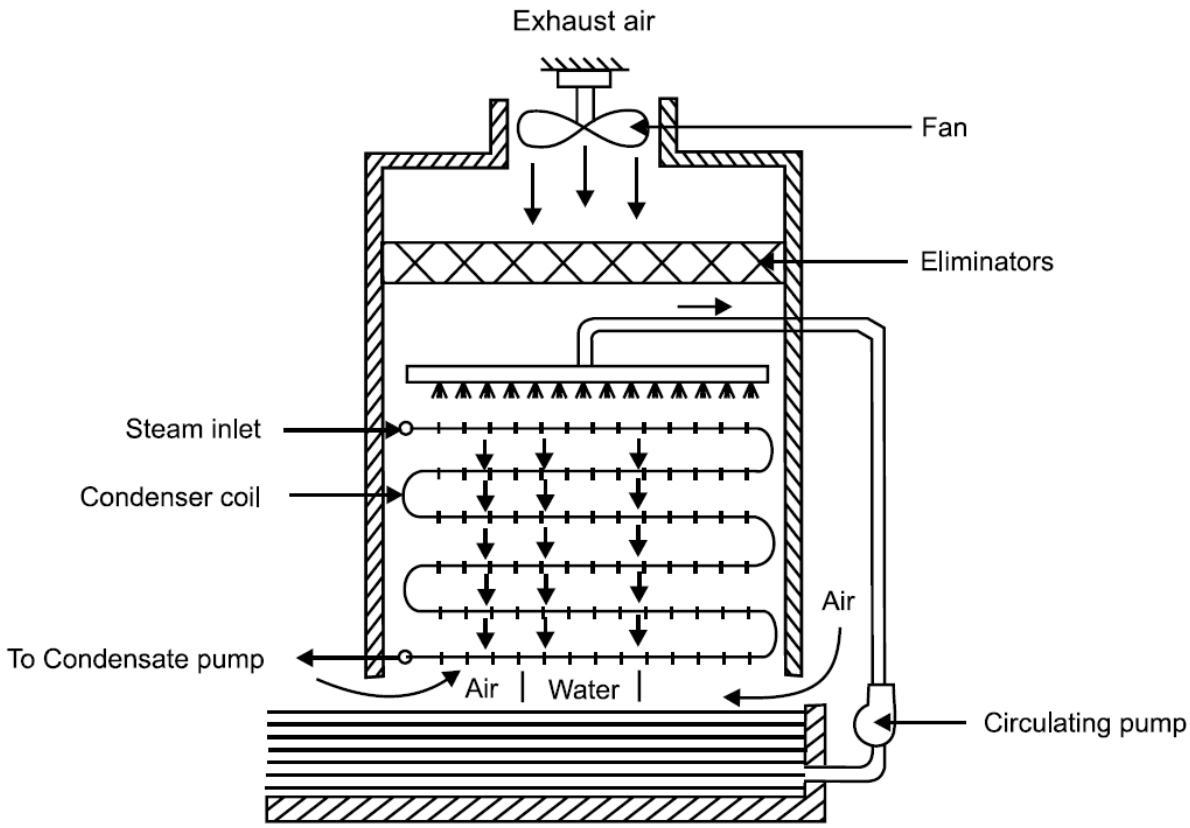



***Fig. 5A.7*** *Evaporative condenser*

Water gets evaporated and evaporated vapours are taken by air leaving condenser. Heat required for evaporation is extracted finally from the steam flowing inside tubes and thus causing its phase transformation. For preventing the exit of water vapours with air going out the separator/eliminator is put on the top before the final exit by which water vapour are recovered up to certain extent. Evaporative condensers are named so because the technique of evaporation is used for realizing the cooling. Amount of water to be sprinkled on condenser tubes should be just sufficient to maintain tube surface in thoroughly wet state. In case of air being humid the vapourising capacity of wet air gets reduced compared to dry air and so the performance of evaporative condenser deteriorates when humidity in atmosphere is high.

Evaporative condenser is advantageous over the surface condenser as the vacuum maintained in evaporative condenser is not very high and the water requirement is small. These condensers are generally used in small capacity power plants where shortage of water supply is there.

## 5A.3 AIR LEAKAGE

Generally, inside the condenser pressure less than atmospheric pressure is maintained, thereby increasing the chances of air leakage into condenser. Leakage of air occurs due to leaking joints, packings, glands etc. along with air in dissolved form coming with feed water. This leakage of air accounts up to 0.005% and 0.5% of steam condensed in case of jet type condenser and surface condensers respectively. Thus, leakage of air is practically always there in the condensers. Air leakage causes the reduction in work done per kg of steam as it increases the back pressure. Also the quantity of water required for condensation of steam is increased due to lowering of partial pressure of steam due to pressure of air. At low pressure the latent heat of steam to be released is more than at higher pressure. Air (having lower conductivity) when present between water and steam hampers the heat exchange and also takes away a portion of heat. Because of this reason also the more quantity of cooling water is required. Hence, leakage of air reduces the condenser efficiency and auxiliary devices such as reciprocating pump, rotary pump, steam ejector or air pumps etc. are required. Also the presence of air increases corrosive action as the corrosion depends largely upon the oxygen content.

Since air leakage in condenser is quite damaging to the performance of condenser so air leakage should be detected and subsequent extraction of air being done. Air leakage is detected by isolating condenser from the rest of plant after the steady states are attained in it i.e. pressure and temperature become steady. After isolation of condenser from plant by stopping the steam and cooling fluid pump, if the vacuum gauge and thermometer readings change then it shows that there is air leakage. For identifying location of leak points, the soap bubble test is carried out in which the bubble formation occurs at leak point if soapy water is put on that after filling condenser with air. Thus, for soap bubble test condenser needs to be emptied and filled with high pressure air. This test has drawback that condenser is to be made non functional and running of plant suffers till the test is performed.

For locating the sources of air leakage during the operational state of condenser peppermint oil test may be used. In this peppermint oil is applied at suspected sources of leak point and in case of leakage at the joint peppermint oil fumes enter the condenser and will come out with the air. Odour of peppermint could be felt in the air leaving condenser. Leak joints can also be detected by passing candle flame over the probable joint. In case of leakage the flame gets distorted.

## 5A.4 CONDENSER PERFORMANCE MEASUREMENT

The vacuum inside the condenser can be quantified by looking at barometer reading which gives atmospheric pressure and vacuum gauge reading and taking their difference to get absolute pressure inside condenser.

Thus, **Absolute pressure (in cm) in condenser**

**= (Barometric head in cm of Hg) – (Vacuum pressure in cm of Hg)**

Generally, this barometric head depends upon the atmospheric conditions and so the absolute pressure also keeps on changing depending upon it. In order to take care for these variations a pressure head called as corrected vacuum in condenser is being defined. This corrected vacuum pressure is defined in reference to 76 cm of mercury which is the standard barometric head as below,

**Corrected vacuum pressure (in cm of Hg) = 76 – Absolute pressure in condenser (in cm of Hg).**

Therefore, the corrected vacuum pressure is used in cases where barometric head differs from 76 cm of mercury.

By the **Dalton's law of partial pressures**, the absolute pressure inside condenser is the sum of partial pressures of steam and air inside it. The partial pressure of steam shall be equal to the saturation pressure corresponding to entering steam temperature. This partial pressure of steam could be seen from steam table. Mathematically, absolute pressure in condenser (*pc*), as per Dalton's law;

$$p_c = p_a + p_s$$

Where $p_a$ is partial pressure of air and $p_s$ is partial pressure of steam.

Theoretically the vacuum in condenser can be given as, $p_{v,\,th} = p_b - p_s$ where $p_{v,\,th}$ is theoretical vacuum is condenser and $p_b$ is barometric pressure. It could be understood that the leakage of air into condenser shall disturb the vacuum inside the condenser and actually due to this air leaked into condenser the condenser pressure is always greater than the theoretical condenser pressure.

In the absence of air leakage and with air leakage there is loss in performance of condenser and so we need to quantify this effect. 'Vacuum efficiency' of condenser is such parameter which is defined by the ratio of actual vacuum to theoretical

vacuum inside condenser. Actually vacuum in condenser in the presence of air leakage can be given by;

$$p_{v,act} = p_b - (p_a + p_s)$$

$$Thus, vacuum\ efficiency\ of\ condenser = \frac{\text{Actual vacuum in condenser } p_{v,act}}{\text{Theoretical vacuum in condenser}(\mathrm{p}_{v,\mathrm{th}})}$$

Here, partial pressure of air ($p_a$) shall be zero in the absence of air leakage and vacuum efficiency shall be 100%. If we look at mathematical expression of efficiency, it is obvious that vacuum efficiency increases with reduction in partial pressure of air.

Also the vacuum efficiency shall increase with decrease in barometric pressure for constant exit steam pressure and condenser actual pressure. In case of less cooling water the condenser pressure increases and reduces the vacuum efficiency of condenser for other pressures remaining same. Designer always wishes to have condenser with highest vacuum efficiency i.e. close to 100%.

'Condenser efficiency' is another condenser performance parameter. It is given by the ratio of actual rise in cooling water temperature to the maximum possible temperature rise. Condenser facilitates heat exchange between two fluids and under ideal conditions the steam should only reject latent heat to cooling water so as to yield condensate at saturated liquid condition. Thus, there should be no undercooling of condensate in ideal condenser. Therefore, ideal condenser may be defined as condenser in which steam rejects only latent heat to cooling fluid and condensate is available without any undercooling. Ideal condenser requires minimum quantity of cooling water and shows maximum gain in cooling water temperature so as to condense the steam. Mathematically,

$$Condenser\ efficiency = \frac{Actual\ rise\ in\ cooling\ water\ temperature}{Maximum\ possible\ temperature\ rise}$$

Here, the maximum possible temperature rise

= {(Saturated temperature corresponding to condenser pressure) – (Cooling water inlet temperature)}

## 5A.5 COOLING TOWER

Cooling tower is similar to evaporative condenser where water used for cooling is being cooled effectively. Water used for cooling becomes hotter after extracting heat from condenser steam and needs to be cooled down if it is to be

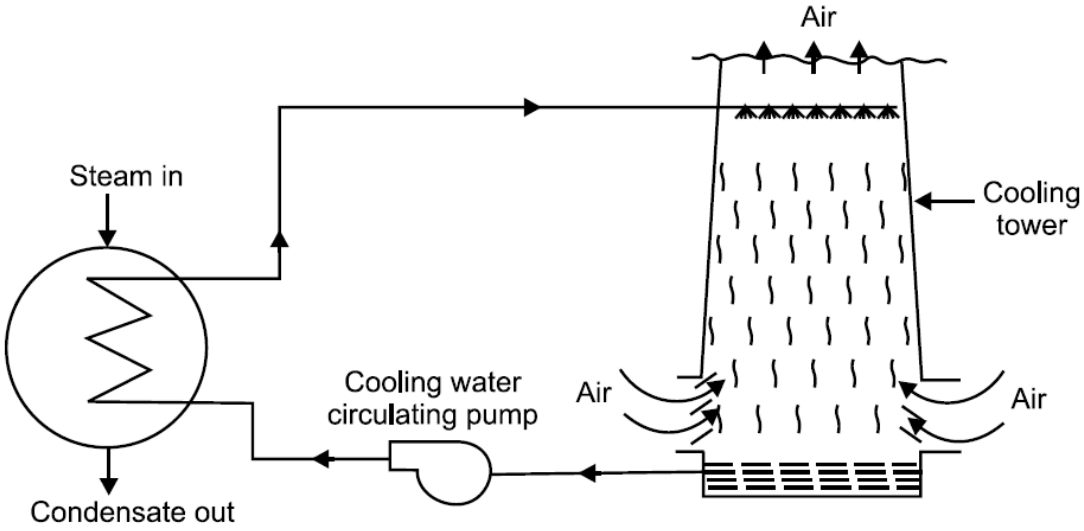
recycled. Cooling towers are preferably used where the water supply is limited and cooling water has to be recirculated without being thrown out.

Cooling tower is such an arrangement made of wood or metal structure having baffles inside to facilitate better heat exchange between hot water falling down and atmospheric air blowing across it.

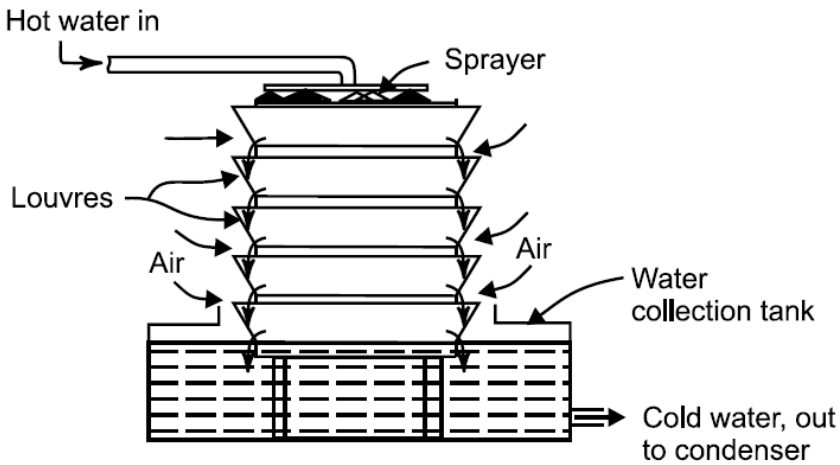
Generally, hot water is admitted from top and is broken into small size (atomized) while falling down. Air enters tower at bottom and flows upward either due to natural draught or forced draught as the case may be. Air picks up heat by intimate contact with hot water particles and leaves cooling tower from exit passage at top. Cooled water falls down and is collected in a tank at bottom of cooling tower. The heat transfer from hot water to air occurs due to evaporative cooling of water and convective heating of air both. The effectiveness of cooling tower diminishes in humid weather conditions due to reduced capacity of air. Dry air shall offer better cooling effectiveness as compared to moist air. During cooling there occurs some loss of water as it is carried away by air. This water loss may be from 1 to 4% due to evaporation and drift losses.

Typical values for a 500 MW steam power plant indicate that this plant has exhaust of steam at the rate of 3.6 kg/kWh at full load, requiring 18 x $10^7$ kg per hour of cooling water and to cool this cooling water air requirement is about 30 x $10^6$ kg per hour in cooling tower.

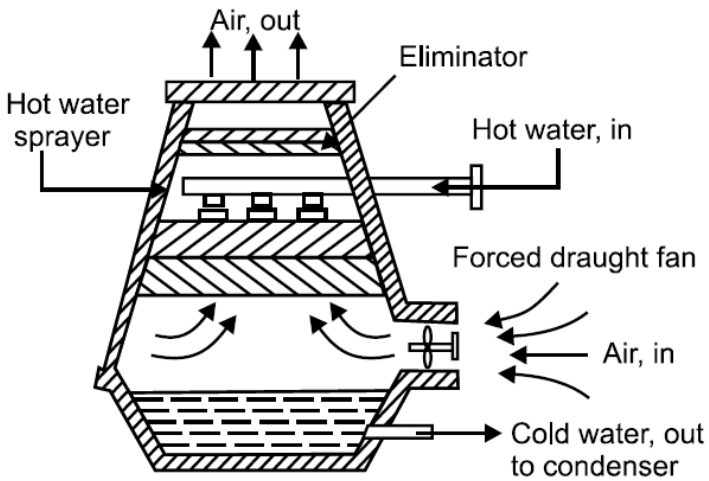
Figure 5A.8 shows schematic of different cooling towers. The performance of cooling tower depends largely upon the duration of contact between water particle and air, surface area of contact between water particle and air, humidity of air and relative velocity of air and water flow etc.




(a) Schematic of cooling tower




(b) Atmospheric cooling tower




(c) Mechanical draught (forced) cooling tower

***Fig. 5A.8*** *Cooling tower*

**1.** ***In a condenser the vacuum of 71 cm of Hg is maintained with barometer reading of 76 cm in Hg. Temperature in condenser is 35°C while hot well is at temperature of 30°C. The cooling water is circulated at the rate of 800 kg/min and condensate is available at 25 kg/min. The temperature of cooling water at inlet and outlet are 15°C and 25°C. Determine the mass of air in kg/m3 of condenser volume, dryness fraction of steam entering condenser and vacuum efficiency. Take mercury density as 0.0135951 kg/cm3, g = 9.81$m/s^2$.***

**Solution:**

Absolute pressure in the condenser

$$p_t = (76 - 71) \times 10^{-2} \times 0.0135951 \times 10^6 \times 9.81 = 6668.396\ N/m^2 = 6.67\ kPa$$

Partial pressure of steam in condenser

=Saturation pressure of steam corresponding to 35°C

(From steam table)

$$p_s = 5.62\ kPa$$

Partial pressure of air, $p_a = p_t - p_s = 6.67 - 5.63 = 1.04\ kPa$

Mass of air per $m^3$ of condenser volume can be obtained from gas equation,

$$m_a = \frac{p_a V}{RT} = \frac{1.04 \times 1}{(273+35) \times 0.287} = 0.012\ \text{kg/m}^3$$

Let the enthalpy of steam entering condenser be $h_s$ so by heat balance,

$$m_w \times C_{p_w} \times (T_{w_o} - T_{w_i}) = m_s \times (h_s - C_{p_w} \times T_c)$$

$$m_w = 800\ kg/min,\ m_s = 25\ kg/min,\ T_{w_o} = 25°C,\ T_{w_i} = 15°C,\ T_c = 30°C$$

$$800 \times 4.18\ (25 - 15) = 25\ (h_s - 4.18 \times 30)$$

$$h_s = 1463\ kJ/kg$$

Let dryness fraction of steam entering be *x*.

$$h_s = 1463 = h_{f\ at\ 35°C} + x\,.\,h_{fg\ at\ 35°C}$$

$$1463 = 146.68 + x \times 2418.6$$

$x = 0.5442$

$$Vacuum\ efficiency = \frac{(76 - 5)\ 0.0135951 \times 10^4 \times 9.81}{(76 \times 0\ 0135951 \times 10^4 \times 9.81) - 5.63\ 10^3}$$

= 0.9891 or 98.91%

| |
|---|
| Mass of air in kg/m$^3$ of condenser volume=0.012 kg/m$^3$, |
| Dryness fraction of steam entering = 0.5442, |
| Vacuum efficiency = 98.91% ***Ans.*** |

**2.** ***A condenser has vacuum of 70 cm of Hg when barometer reading is 76 cm. Condenser has temperature of 30°C. Air leaks into condenser at the rate of 1 kg air per 2500 kg steam. Calculate (i) the capacity of air pump per kg of steam for removal of air from steam entering condenser, and (ii) the mass of water vapour accompanying this air.***

**Solution:**

Absolute pressure in condenser = (76 – 70) cm of Hg

$$= 6 \times 0.0135951 \times 10^4 \times 9.81 = 8002.67\ Pa$$

$$p_t = 8.003\ kPa$$

Partial pressure of steam, $p_s$= Saturation pressure corresponding to 30°C from steam table

$$p_s = 4.246\ kPa$$

Partial pressure of air, $p_a$ = Total pressure in condenser – Partial pressure of steam

$$p_a = p_t - p_s = 3.757\ kPa$$

Mass of air accompanying per kg steam due to leakage $= \frac{1}{2500}$ kg

= 0.0004 kg

Using gas equation, $p_a . V = mRT$

Volume of air per kg of steam $= \frac{mRT}{p_a}$

$$= \frac{0\ 0004 \times 0.287 \times (273+30)}{3.757}$$

$$= 9.26 \times 10^{-3}\ m^3/kg$$

Capacity of air pump = $9.26 \times 10^{-3}\ m^3/kg$ steam ***Ans.***

Volume of water vapour accompanying air shall be equal to the volume of air.

So volume of water vapour accompanying air = $9.26 \times 10^{-3}\ m^3/kg$

Specific volume of dry steam at condenser temperature of 30°C

$$= v_{g\ at\ 30°C} = 32.89 \times 10^{-3}\ m^3/kg$$

Mass of water vapour accompanying air $= \frac{9.26 \times 10^{-3}}{32.89}$

$$= 2.82 \times 10^{-4}\ kg/kg\ of\ steam$$

Mass of water vapour accompanying air = $2.82 \times 10^{-4}\ kg/kg\ of\ steam$ ***Ans.***

**3.** ***During the trial on a condenser it is seen to have vacuum of 67 cm of Hg while barometer reading is 75 cm of Hg. The mean condenser temperature is 40°C and temperature of hot well is 35°C. Circulating water flows at 1000 kg/min for giving condensate at the rate of 50 kg/min. Temperature of cooling water at inlet and exit are 10°C and 25°C. Determine, (i) the vacuum corrected to standard barometer reading of 76 cm. (ii) the vacuum efficiency of condenser, (iii) the undercooling of condensate, (iv) the condenser efficiency, (v) the state of steam entering condenser, (vi) the mass of air per m3 of condenser volume, and (vii) the mass of air per kg of uncondensed steam.***

**Solution:**

Vacuum corrected to standard barometer reading of 76 cm

= 76 – (75 – 67) = 68 cm Hg

Corrected vacuum = 68 cm Hg ***Ans.***

Absolute pressure in condenser = 75 – 67 = 8 cm of Hg

or $(8 \times 0.0135951 \times 10^4 \times 9.81 = 10.67\ kPa)$

or, $p_t = 10.67\ kPa$

Partial pressure of steam, $p_s$ = Saturation pressure at 40°C from steam table

$p_s = 7.384\ kPa$

Partial pressure of air, $p_a = p_t - p_s = 3.286\ kPa$

Vacuum efficiency $= \frac{Actual\ vacuum}{Theoretical\ vacuum}$

$= \frac{(75-8)\ 0.0135951\times10^4\times 9\ 81}{(75\times0.0145951\times10^4\times9.81)-7.384\times10^3}$

= 0.9645 or 96.45%

Vacuum efficiency = 96.45% ***Ans.***

Undercooling of condensate = 40 – 35 = 5°C

**Undercooling = 5°C** ***Ans.***

Condenser efficiency $= \frac{(T_{w\ out}-T_{w\ in})}{(T_{w\ sat\ corresponding\ to\ p_t} - T_{w\ in})}$

Saturation temperature corresponding to absolute pressure in condenser, $40.01\ kPa$.

$T_{sat\ corresponding\ to\ p_t} = 46.9°C$, from steam table.

$T_{w\ out}$ = 25°C, $T_{w\ in}$ = 10°C

Condenser efficiency $= \frac{25-10}{46.9-10}$ = 0.4065 or 40.65%

Let us consider enthalpy of steam entering be $h$ kJ/kg and dryness fraction be $x$. By applying heat balance,

$$m_s \times (h - c_{pw}\,.T_c) = m_w\,.c_{pw}.(T_{w_o} - T_{w_i})$$

$m_s$= 50 kg/min, $m_w$ = 1000 kg/min

$50 \times (h - 4.18 \times 40) = 1000 \times 4.18\,(25 \times 10)$

$$h = 1421.2\frac{kJ}{kg} = h_{f\,at\ 40°C} + x\,.h_{fg\ at\ 40°C}$$

$$1421.2 = 167.57 + x \times 2406.7$$

$$x = 0.5209$$

| Dryness fraction of steam entering = 0.5209 ***Ans.*** |
|---|

Mass of air per m$^3$ of condenser volume $m = \frac{p_a.V}{RT_c} = \frac{3.286\times1}{0.287\times(273+40)} = 0.0366$ kg/m$^3$

Volume of per kg of uncondensed steam = $v_{g\ at\ 40°C} = 19.52\ m^3/kg$

Mass of air in one kg of uncondensed steam = Mass of air in 19.52 m$^3$ volume

$$= \frac{3.286\times19.52}{0.287\times(273+40)}$$

= 0.714 kg

| Mass of air/m3 of condenser volume = 0.0366 kg/m3 ***Ans.*** |
|---|
| Mass of air in per kg of uncondensed steam = 0.714 kg |

**4.** ***In a surface condenser operating with steam turbine the vacuum near inlet of air pump is 69 cm of Hg when barometer reading is 76 cm of Hg. Temperature at inlet of vacuum pump is 30°C. Air leakage occurs at the rate of 60 kg/hr. Determine,***
***(i) The capacity of air pump in m3/hr and mass of vapour extracted with air in kg/hr.***
***(ii) The dimension of reciprocating air pump cylinder if it runs at 240 rpm and L/D ratio is 1.5.***

***Solution:***

Absolute pressure at inlet to air pump $=(76-69) = 7\ cm\ Hg$, or

$$= 7 \times 0.0135951 \times 10^4 \times 9.81$$

$$p_t = 9.336\ kPa$$

$$Partial\ pressure\ of\ vapour\ at\ 30°C = Saturation\ pressure\ at\ 30°C$$

$$p_s = 4.246\ kPa$$

Partial pressure of air, $p_a = p_t - p_s = 5.09\ kPa$

Volume of 60 kg air at pressure of 89.112 kPa

$$V = \frac{60 \times 0.287 \times (273+30)}{5.09} = 1025.1\ m^3/hr$$

| Capacity of air pump in $m^3$/hr = 1025.1 $m^3$/hr ***Ans.*** |
|---|

$$Volume\ handled, m^3/hr = \frac{\pi}{4} \times D2 \times L \times N \times 60 = 1025.1$$

$$1025.1 = \frac{\pi}{4} \times 1.5\ D^3 \times 240 \times 60$$

$D$ = 0.3924 m or 39.24 cm

$L$ = 58.86 cm

Mass of water vapour going with air $= \frac{V}{V_{gat\ 30°\ C}} = \frac{1025.1}{32.89} = 31.17\ \frac{kg}{hr}$

| Bore = 39.24 cm, Stroke = 58.86 cm |
|---|
| Mass of water vapour extracted with air = 31.17 kg/hr ***Ans.*** |

**5.** ***A steam condenser is supplied with 1000 kg/min steam in 0.9 dry state. The pressure at suction of air extraction pump on condenser is 70 cm of Hg and barometer reads 77 cm of Hg. Temperature in suction pipe is 30°C and air leaks at the rate of 5 _ 10–4 kg per kg of steam. Cooling water temperature gets increased by 15°C. Determine the mass handled by dry air extractor and cooling water circulation rate in kg/min.***

**Solution:**

Absolute pressure in condenser = $(77-70) = 7\ cm\ of\ Hg\ or\ (7 \times 0.0135951 \times 10^4 \times 9.81)$

$$p_t = 9.34\ kP_a$$

Partial pressure of steam, $p_s = Saturation\ pressure\ corresponding\ to\ 30°C = 4.246\ kPa$

Partial pressure of air, $p_a = p_t - p_s = 5.094\ kPa$

Rate of air extraction per minute = $5 \times 10^{-4} \times 1000 = 0.5\ kg/min$

Volume of air extracted per minute = $\frac{mRT}{p_a} = \frac{0.5 \times 0.287 \times (273+30)}{5.094}$

= 8.54 m3/min

Specific volume corresponding to 30°C, $v_g = 32.89\ m^3/kg$

Volume of air extracted = Volume of mixture sucked per minute = 8.54 $m^3$/min

Mass of steam extracted in mixture handled per minute = $\frac{8.54}{32.89} = 0.2596\ kg/min$

Therefore, mass handled by air extraction pump = 0.5 + 0.2596 = 0.7596 kg/min

| Mass handled by air pump = 0.7596 kg/min ***Ans.*** |
|---|

Enthalpy of steam entering condenser, $h = h_{f\,at\,30°C} + 0.9 \times h_{fg\ at\ 30°C}$

$= 125.79 + (0.9 \times 2430.5)$

= 2313.24 kJ/kg

Mass flow rate of circulating water can be obtained by energy balance on condenser,

$$m_w \times C_{pw} \times (\Delta T_w) = m_s \times (h - C_{pw} \times T_c)$$

$$m_w \times 4.18 \times 15 = 1000 \times (2313.24 - 4.18 \times 303)$$

$$m_w = 16693.78\ \text{kg/min}$$

| Water circulation rate = 16693.78 kg/min ***Ans.*** |
|---|

**6.** ***In a surface condenser vacuum of 70 cm Hg is maintained when the barometric pressure is 76 cm Hg. Steam enters 0.85 dry into condenser at the rate of 300 kg/min. Temperature of condensate is 30°C and the rise in circulating water temperature is 20°C. For sending water through condenser and piping a pressure head of 5 m is required. For surface condenser determine,***

***(i) The flow surface area required when water flows at 50 m/min,***

***(ii) The cooling surface area required when heat transfer rate is*** $\mathbf{15 \times 10^5 \frac{kJ}{m^2}.hr\ °C}$.

***(iii) The total head required to be developed by pump***

**Solution:**

Absolute pressure in condenser = $76 - 70 = 6\ cm\ of\ Hg,$

Or $= (6 \times 0.0135951 \times 10^4 \times 9.81)\ Pa$

$$p_t = 8.002\ kPa$$

Partial pressure of steam, $p_s = Saturation\ pressure\ corresponding\ to\ 30°C = 4.246\ kPa$

Applying heat balance on condenser,

$$m_w \times 4.18 \times 20 = (h_{fat\ 30°C} + 0.85 \times h_{fg\ at\ 30°C} - 4.18 \times 30) \times 300$$

$$m_w = 7415\ kg/min$$

Volume flow of water = $\frac{7415}{1000} m^3/min = 7.415\ m^3/min$

Flow surface area requirement = $\frac{Velocity\ flowrateof\ water}{Velocityof\ water\ flow} = \frac{7.415}{50}$

$= 0.1483\ m^2$

Flow surface area required = 0.1483 m2 ***Ans.***

Cooling surface area required $= \frac{Total\ heat\ given\ by\ steam}{Heat\ transfer\ rate}$

$$= \frac{300 \times (h_{fat\ 30°C} + 0.85 \times h_{fg\ at\ 30°C} - 4.18 \times 30) \times 60}{15 \times 10^5}$$

$= 0.413 \times 60$

$= 24.79\ m^2$

Cooling surface area required = 24.79 m$^2$ ***Ans.***

Velocity head present $= \frac{1}{2} \times \left(\frac{50}{60}\right)^2 \times \frac{1}{9.81} = 0.0354$ m

Total head required = Pressure head + Velocity head

= 5 + 0.0354 = 5.0354 m

Head required = 5.0354 m ***Ans.***

**7.** ***A jet condenser has steam entering at 350 kg/min when vacuum of 680 mm is maintained in it and the barometer reads 760 mm. Air mass going into condenser is 0.05% of steam mass entering. Water at 20°C enters the condenser to condense the steam such that temperature of condensate is 30°C. Volume of water required is 0.02 m³ per kg steam. The volume of air dissolved in the water injected may be considered as 5% of volume of water at atmospheric pressure. Determine the volume handling capacity of air pump for removing air and condensate when pump has volumetric efficiency of 90%.***

**Solution:**

Absolute pressure in condenser = (76 – 68) = 8 cm of Hg

or, $= (8 \times 0.0135951 \times 9.81 \times 10^4 \times 10^{-3})$

$$p_t = 10.67\ kPa$$

Partial pressure of steam,

$p_s$ = Saturation pressure corresponding to 30°C

$p_s$= 4.246 kPa

Partial pressure of air, $p_a = p_t - p_s = 10.67 - 4.246 = 6.424\ kPa$

Volume of cooling water required per minute = $350 \times 0.02 = 7m^3/min$

Mass of air going into condenser with steam per minute = $\frac{350\times 0.05}{100} = 0.175$ kg/min

Volume of air entering per minute with cooling water = $\frac{7\times5}{100}$

$= 0.35\ m^3/min$

Mass of air with cooling water, using $pV = mRT$,

$$m = \frac{pV}{RT}$$

Here $p$ = atmospheric pressure = 101.3 kPa, $V$ = 0.35 $m^3$, $R$ = 0.287 kJ/kg K

$T$ = 273 + 20

= 293 K

Mass of air with cooling water $= \frac{101.3\times0.35}{0.287\times293}$

$= 0.422\ kg/min$

Thus, total mass of air inside condenser per minute

= (Mass of air with steam + Mass of air with cooling water)

= 0.175 + 0.422 = 0.597 kg/min

Volume of air corresponding to 0.597 kg/min = $\frac{0.597\times0.287\times(273+30)}{6.424}$

$= 8.08\ m^3/min$

Volume of steam condensed (condensate) $= 350 \times v_{f\ at\ 30°C}$

$= 350 \times 0.001004 = 0.3514\ m^3/min$

Total capacity of air pump (wet air pump) = Volume of condensate/min + Volume of air/min + Volume of cooling water/min.

$= 0.3514 + 8.08 + 7$

$= 15.4314\ m^3/min$

Actual capacity of air pump $= \frac{15.4314}{0.9}$

$= 17.15\ m^3/min$

Capacity of air pump = 17.15 m3/min ***Ans.***

**8.** ***A barometric jet condenser has steam entering at the rate of 20 kg/min and 12 kg cooling water per kg of steam is supplied into it for condensation. After condensation the cooling water and condensate leave at 40°C. Cooling water enters the condenser at 20°C. Vacuum of 650 mm Hg is maintained inside condenser while barometer read 760 mm Hg. An air pump is put for extracting air from condensate at the rate of 2m3 of wet mixture per minute. At the suction side of air pump a vacuum of 660 mm Hg and temperature of 35°C is observed.***

***Determine,***

***(i) The dryness fraction of steam entering condenser,***

***(ii) The mass of air entering condenser per minute,***

***(iii) The effective pressure head at tail of barometric condenser.***

*Solution:*

Absolute pressure in condenser, $p_t = (760 - 650) \times 10^{-1} = 11\ cm\ Hg$

or, $p_t = 11 \times 10^4 \times 0.0135951 \times 9.81 \times 10^{-3}, kPa$

$p_t = 14.67\ kPa$

Partial pressure of steam, $p_s = Saturation\ pressure\ corresponding\ to\ 40°C$

= 7.384 kPa

Partial pressure of air, $p_a = p_t - p_s = 7.286\ kPa$

Cooling water required $= 12 \times 20 = 240\ kg/min$

Let the dryness fraction of steam entering condenser be *x* and enthalpy *h*.

Heat balance on condenser yields;

$$m_s \times h + m_w \times c_{pw} \times T_{wi} = (m_s + m_w)\, c_{pw} \_ T_{wo}$$

$$20 \times h + (240 \times 4.18 \times 20) = (20 + 240) \times 4.18 \times 40$$

$$h = 1170.4\ kJ/kg = h_{f\ at\ 40°C} + x \times h_{fg\ at\ 40°C}$$

$$1170.4 = 167.57 + (x \times 2406.7)$$

$$x = 0.4167$$

Dryness fraction of steam entering = 0.4167 ***Ans.***

Absolute pressure at suction of air pump = $(760 - 660) \times 10^{-1} = 10\ cm\ Hg$

Or $(10 \times 0.0135951 \times 10^4 \times 9.81 \times 10^{-3} = 13.34\ kPa)$

Partial pressure of steam at suction of air pump = Saturation pressure at 35°C.

$$p'_s = 5.628\ kPa$$

Partial pressure of air, $p'_a = 13.34 - 5.628 = 7.712\ kPa$

Now, at suction of pump volume of air will be equal to the volume of mixture.

Volume of mixture = Volume of air = 2 $m^3$

Mass of air entering $= \frac{p'_a V}{R T'_a}$

$$= \frac{7.712 \times 2}{0.287 \times (273 + 35)}$$

= 0.1745 kg/min

Effective pressure head tail of barometric condenser = Head corresponding to (Barometric pressure – Absolute pressure in condenser)

$$= \frac{(101.3 - 14.67) \times 10^3}{9.81 \times 0.0135951 \times 10^6}$$

= **0.649 m**

| |
|---|
| Mass of air entering = 0.1745 kg/min ***Ans.*** |
| Head = 0.649 m |

**9.** ***A steam condenser has steam entering at 35°C and condensate being removed at 34°C. Condenser has two pumps one for extracting air and other for extraction of condensate. Air is removed at temperature of 33°C. The air leaks into condenser at the rate of 3 kg/hr. consider the pressure inside condenser to remain uniform and neglect change in pressure due to air at steam inlet. Determine the volume of air handled by air pump in kg/hr and also determine the volume to be handled if a combined air and condensate pump is being used.***

**Solution:**

Partial pressure of steam at 35°C = Saturation pressure corresponding to 35°C.

$$p_s = 5.628 \text{ kPa}$$

If the pressure of air at inlet is neglected then the total pressure in condenser,

$$p_t = p_s = 5.628\ kPa.$$

At the suction of air pump, partial pressure of steam= Saturation pressure corresponding to 33°C

$$p_s' = 5.075 \text{ kPa}$$

Partial pressure of air, $p_a' = p_t - p_s' = 5.628 - 5.075 = 0.553$ kPa

Volume of air handled by air pump $= \frac{mRT_a'}{p_a'} = \frac{3\times0.287\times(273+33)}{0.553}$

$= 467.43\ m^3/hr$

In case the air and condensate mixture is to be handled by same pump then,

Partial pressure of steam = Saturation pressure corresponding to 34°C

$$p_{s''} = 5.352\ kPa$$

Partial pressure of air, $p_{a''} = p_t - p_{s''} = 5.628 - 5.352 = 0.276\ kPa$

Volume of air = Volume of mixture handled by pump $= \frac{3\times0.287\times(273+34)}{0.276}$

$= 957.71\ m^3/hr$

| |
|---|
| Volume of air handled = 467.43 m3/hr ***Ans.*** |
| Volume of mixture handled = 957.71 m3/hr |

**10.** ***In a surface condenser the vacuum at inlet is seen to be 72 cm Hg and at outlet it is 73 cm Hg.The barometer reading is 76 cm and the dryness fraction of steam at inlet is 0.92. Cooling water entering the condenser is at 20°C. Considering no air in the condenser and the temperature rise in cooling water to be maximum determine.***

***(i) the minimum amount of undercooling.***

***(ii) the amount of cooling water required per kg of steam.***

**Solution:**

Inlet pressure in condenser = (76 – 72) cm Hg = 4 cm Hg.

$= 4 \times 10^4 \times 0.0135951 \times 9.81 \times 10^{-3}$

= 5.335 kPa

Outlet pressure in condenser = (76 – 73) cm Hg = 3 cm Hg

$= 3 \times 10^4 \times 0.0135951 \times 9.81 \times 10^{-3}$

= 4.001 kPa

Since there is no air in condenser so the 5.335 kPa and 4.001 kPa will be the pressure of steam. Saturation temperature corresponding to above pressures gives temperature at inlet and outlet respectively.

Saturation temperature at inlet = 33.87°C, (from steam table at 5.335 kPa)

Saturation temperature at outlet = 28.96°C, (from steam table at 4.001 kPa)

Thus, steam will leave at maximum temperature of 28.96°C

The minimum amount of undercooling = 33.87 – 28.96 = 4.91°C

For maximum temperature rise of cooling water the temperature of cooling water outlet will be equal to the temperature of steam at inlet of 33.87°C.

Therefore, the maximum rise in cooling water temperature

$$= 33.87 - 20 = 13.87°C$$

Enthalpy of steam entering, $h = h_{f\ at\ 33.87°C} + 0.92 \times h_{fg\ at\ 33.87°C}$

$$h = 141.97 + (0.92 \times 2421.33) = 2369.59\ kJ/kg$$

Let mass of cooling water required be *m* kg per kg steam.

Heat balance on condenser yields,

$$m \times 4.18 \times 13.87 = 1 \times (2369.59 - 4.18 \times 28.96)$$

*m* = 38.78 kg water per kg of steam

| |
|---|
| Undercooling = 4.91°C ***Ans.*** |
| Cooling water requirement = 38.78 kg/kg steam |