i.e., $$\text{Boiler efficiency} = \frac{m_a (h - h_{f_1})}{C} \quad ...(3.20)$$

where, m_a = Mass of water actually evaporated into steam per kg of fuel at the working pressure, and

C = Calorific value of the fuel in kJ/kg.

If the boiler, economiser, and superheater are considered as a single unit, then the *boiler efficiency is termed as overall efficiency of the boiler plant.*

The following are the *factors* on which the boiler efficiency depends :

1. Fixed factors 2. Variable factors.

1. **Fixed factors.** These are :

(*i*) ***Boiler design.*** It includes the arrangement and effectiveness of the heating surfaces, the shape and volume of the furnace, the arrangement of flues, the arrangement of steam and water circulation.

(*ii*) ***Heat recovery equipment.*** It includes the economiser, superheater, air preheater and feed water heater.

(*iii*) ***Built in losses.*** It includes the heat transfer properties of the settings and construction materials, flue gas and ash heat losses.

(*iv*) Rated rate of firing, the furnace volume and heating surface.

(*v*) Properties and characteristics of fuel burnt.

2. **Variable factors.** These are :

(*i*) Actual firing rate.

(*ii*) Fuel condition as it is fired.

(*iii*) The condition of heat absorbing surfaces.

(*iv*) Excess air fluctuations.

(*v*) Incomplete combustion and combustibles in the refuse.

(*vi*) Change in draught from the rated, due to atmospheric conditions.

(*vii*) Humidity and temperature of the combustion air.

3.17.5. Heat Losses in a Boiler Plant

The following heat losses occur in a boiler plant :

1. Heat lost to flue gases.
2. Heat lost due to incomplete combustion.
3. Heat lost due to unburnt fuel.
4. Convection and radiation losses.

1. **Heat lost to flue gases.** The flue gases contain dry products of combustion as well as the steam generated due to combustion of hydrogen in the fuel.

Heat lost through *dry flue gases*, Q_g :

$$Q_g = m_g c_{pg} (T_g - T_a) \quad ...(3.21)$$

where, m_g = Mass of gases formed per kg of coal,

c_{pg} = Specific heat of gases,

T_g = Temperature of the gases, and

T_a = Temperature of air entering the combustion chamber of boiler.

Heat carried away by the steam in flue gases, Q_s :

$$Q_s = m_{s_1} (h_{s_1} - h_{f_1}) \quad ...(3.22)$$

where, m_{s_1} = Mass of steam formed per kg of fuel due to the combustion of H_2 in the flue,

h_{f_1} = Enthalpy of water at boiler house temperature, and

h_{s_1} = Enthalpy of steam at the gas temperature and at a partial pressure of steam vapour in the gas.

This heat lost to flue gases can be reduced by passing the flue gases through the economiser and air preheater.

2. Heat lost due to incomplete combustion. The combustion is said to be incomplete if the carbon burns to CO instead of CO_2. One kg of carbon releases 10120 kJ of heat if it burns to CO, whereas, it can release 33800 kJ/kg if it burns to CO_2.

$\therefore$ Heat loss due to incomplete combustion of 1 kg of carbon = 33800 – 10120 = 23680 kJ.

If CO is present in the flue gases it indicates that combustion of fuel is incomplete. If the percentages of CO and CO_2 in flue gases by volume are known, then carbon burnt to CO instead of CO_2 per kg of fuel is given by

$$\text{Mass of carbon burnt to CO} = \frac{CO \times C}{CO_2 + CO} \qquad ...(3.23)$$

where CO and CO_2 are expressed as % by volume in flue gases and C as the fraction of carbon in one kg of fuel.

$\therefore$ Heat lost due to incomplete combustion of carbon per kg of fuel

$$= \frac{CO \times C}{CO_2 + CO} \times 23680 \text{ kJ/kg of fuel.}$$

This loss (due to incomplete combustion) can be reduced by supplying excess quantity of air and giving a turbulent motion to the air before it enters the furnace in order to help the mixing process.

3. Heat lost due to unburnt fuel. If m_{f_1} is the mass of unburnt fuel per kg of fuel used and C is the calorific value of the fuel, then heat lost due to unburnt fuel,

$$Q = m_{f_1} \times C \qquad ...(3.24)$$

In case of solid fuels this loss cannot be completely avoided.

4. Convection and radiation losses. As the hot surfaces of the boiler are exposed to the atmosphere, therefore, heat is lost to atmosphere by convection and radiation.

The loss of heat due to convection and radiation losses

= Heat released per kg of fuel – total of the heat losses given by eqns. (3.21), (3.22), (3.23) and (3.24).

These losses can be reduced by providing heat insulation on the boiler surface.

Table 3.1 is a resume of the more common causes of thermal losses associated with steam generators.

Table 3.1 Causes of Heat Loss in Steam Generators

1. Loss due to moisture in coal :

(*i*) Excessive wetting down of coal before firing.

(*ii*) High moisture absorption by coal in yard storage.

2. Loss due to moisture formed by combustion of hydrogen :

Irreducible for any specific fuel. This loss is larger for oil and gas fuels than for coal.

3. Loss due to heat carried away in dry chimney gas :

(*i*) High excess air as revealed in low CO_2 content of flue gas.

(*ii*) High flue gas temperature.

1. Dirty heating surfaces.

2. Poor water circulation. Scale on water side.
3. Dead gas pockets. Leaky or ineffective baffles.
4. Gas velocity too high.

4. Loss due to incomplete combustion :

(*i*) Insufficient air supply.
(*ii*) Fuel bed in poor condition.
(*iii*) Undercooling of furnace at low ratings.
(*iv*) Improper setting of boilers.

5. Loss due to combustible in ashpit :

(*i*) Grate or stoker not proportioned to the kind of fuel used.
(*ii*) Too high rate of combustion attempted.
(*iii*) Grates dumped or fuel bed sliced too frequently.
(*iv*) Furnace temperature is above fusion temperature of ash.

6. Loss from radiation and convection from boiler and setting :

(*i*) Boiler drums uninsulated.
(*ii*) Wall of setting too thin or of poor quality.
(*iii*) Furnace refractories in need of repair or renewal.

7. Loss due to moisture in the air :

(*i*) Moisture-laden air as from steam jet.
(*ii*) High excess air on days of high humidity. This loss is small and frequently included with several other small losses, usually unaccounted for, such as soot or cinder in the chimney gas, heat in ashes, etc.

Example 3.5. *The steam used by the turbine is 5.4 kg/kWh at a pressure of 50 bar and a temperature of 350°C. The efficiency of boiler is 82 percent with feed water at 150°C.*

(*i*) *How many kg of 28100 kJ coal are required/kWh ?*
(*ii*) *If the cost of coal/tonne is Rs. 500, what is fuel cost/kWh ?*

Solution. Mass of steam used, $m_s = 5.4$ kg/kWh
Pressure of steam, $p = 50$ bar
Temperature of steam, $t_{sup} = 350°C$
Boiler efficiency = 82%
Feed water temperature = 150°C
Calorific value of coal, $C = 28100$ kJ
Cost of coal/tonne = Rs. 500.

(*i*) Boiler efficiency is given by,

$$\eta_{\text{boiler}} = \frac{m_s (h - h_{f_1})}{m_f \times C} \qquad ...(i)$$

where m_f is the mass of fuel used, kg/kWh.

At 45 bar and 350°C. From steam tables,

$$h_{sup} = 3068.4 \text{ kJ/kg}$$

$$h_{f_1} \text{ (at 150°C)} = 1 \times 4.18 \times (150 - 0) = 627 \text{ kJ/kg}$$

Putting these values in eqn. (*i*), we get

$$0.82 = \frac{5.4 (3068.4 - 627)}{m_f \times 28100}$$

$$\therefore \qquad m_f = \frac{5.4\,(3068.4 - 627)}{0.82 \times 28100} = \mathbf{0.572\ kg/kWh.}\quad \textbf{(Ans.)}$$

(*ii*) The cost of fuel (coal)/kWh = m_f in tonnes/kWh × cost/tonne

$$= \frac{0.572}{1000} \times 500 \times 100 = \mathbf{28.6\ paise/kWh.}\quad \textbf{(Ans.)}$$

Example 3.6. *The following data refer to a boiler plant consisting of an economiser, a boiler and a superheater :*

Mass of water evaporated per hour = 5940 kg, mass of coal burnt per hour = 675 kg, L.C.V. of coal = 31600 kJ/kg, pressure of steam at boiler stop valve = 14 bar, temperature of feed water entering the economiser = 32°C, temperature of feed water leaving the economiser = 115°C, dryness fraction of steam leaving the boiler and entering superheater is 0.96, temperature of steam leaving the superheater = 260°C, specific heat of superheater steam = 2.3. Determine :

(*i*) *Percentage of heat in coal utilized in economiser, boiler and superheater.*

(*ii*) *Overall efficiency of boiler plant.*

Solution. Mass of water evaporated = 5940 kg/h

Mass of coal burnt = 675 kg/h

Lower calorific value of coal = 31600 kJ/kg

Pressure of steam at boiler stop valve, p_1 = 14 bar

Temperature of feed water entering the economiser, t_{e_1} = 32°C

Temperature of feed water leaving the economiser, t_{e_2} = 115°C

Dryness fraction of steam entering superheater = 0.96

Temperature of steam leaving the superheater, t_{sup} = 260°C

Specific heat of superheated steam, c_{ps} = 2.3 kJ/kg K

Heat utilised by 1 kg of feed water in *economiser* :

$$h_{f_1} = 1 \times 4.18 \times (t_{e_2} - t_{e_1}) = 1 \times 4.18 \times (115 - 32) = 346.9 \text{ kJ/kg}$$

Heat utilised in *boiler* per kg of feed water :

$$h_{\text{boiler}} = (h_f + xh_{fg}) - h_{f_1}$$

At 14 bar pressure : From steam tables,

$$t_s = 195°\text{C},\ h_f = 830.1 \text{ kJ/kg},\ h_{fg} = 1957.7 \text{ kJ/kg}$$

$$\therefore \qquad h_{\text{boiler}} = (830.1 + 0.96 \times 1957.7) - 346.9 = 2362.6 \text{ kJ/kg}$$

Heat utilised in *superheater* by 1 kg of feed water :

$$h_{\text{superheater}} = (1 - x)\,h_{fg} + c_{ps}\,(T_{sup} - T_s)$$
$$= (1 - 0.96) \times 1957.7 + 2.3\,(260 - 195) = 78.3 + 149.5 = 227.8 \text{ kJ/kg.}$$

Also, mass of water evaporated/hour/kg of coal burnt = $\frac{5940}{675}$ = 8.8 kg.

(*i*) **Percentage of heat utilised in economiser**

$$= \frac{346.9}{31600} \times 8.8 \times 100 = \mathbf{9.66\%.}\quad \textbf{(Ans.)}$$

Percentage of heat utilised in boiler

$$= \frac{2362.6}{31600} \times 8.8 \times 100 = \mathbf{65.7\%.}\quad \textbf{(Ans.)}$$

Percentage of heat utilised in superheater

$$= \frac{227.8}{31600} \times 8.8 \times 100 = \mathbf{6.34\%.} \quad \textbf{(Ans.)}$$

(*ii*) **Overall efficiency of boiler plant, $\eta_{overall}$** :

Total heat absorbed in kg of water

$$= h_{f_1} + h_{boiler} + h_{superheater}$$

$$= 346.9 + 2362.6 + 227.8 = 2937.3 \text{ kJ/kg}$$

Overall efficiency $\quad \eta_{overall} = \dfrac{8.8 \times 2937.3}{31600} = \mathbf{0.8179 \text{ or } 81.79\%.}$ **(Ans.)**

Example 3.7. *A boiler produces 200 kg of dry and saturated steam per hour at 10 bar and feed water is heated by an economiser to a temperature of 110°C. 225 kg of coal of a calorific value of 30100 kJ/kg are fired per hour. If 10% of coal remains unburnt, find the thermal efficiency of the boiler and boiler and grate combined.*

Solution. Rate of production of steam = 2000 kg/h

Quality of steam, $x = 1$

Steam pressure, $p = 10$ bar

Feed water temperature rise = 110°C

Rate of coal firing = 225 kg/h

Calorific value of coal = 30100 kJ/kg

Percentage of coal unburnt = 10%

From steam tables, corresponding to 10 bar, $h = h_g = 2776.2$ kJ/kg

Heat contained in 1 kg of feed water before entering the boiler,

$$h_{f_1} = 1 \times 4.18 \times (110 - 0) = 459.8 \text{ kJ}$$

Total heat given to produce 1 kg of steam in boiler

$$= h - h_{f_1} = 2776.2 - 459.8 = 2316.4 \text{ kJ/kg}$$

Mass of coal actually burnt $= 225 \times \dfrac{90}{100} = 202.5$ kg

Mass of steam produced per kg of coal (actually burnt), $m_a = \dfrac{2000}{202.5} = 9.87$ kg

Thermal efficiency of the boiler

$$= \frac{m_a (h - h_{f_1})}{C} = \frac{9.87\,(2776.2 - 459.8)}{30100}$$

$$= \mathbf{0.759 \text{ or } 75.9\%.} \quad \textbf{(Ans.)}$$

Thermal efficiency of boiler and grate combined

$$= \frac{\frac{2000}{225}(2776.2 - 459.8)}{30100} = \mathbf{0.684 \text{ or } 68.4\%.} \quad \textbf{(Ans.)}$$

Example 3.8. *A boiler generates 7.5 kg of steam per kg of coal burnt at a pressure of 11 bar, from feed water having a temperature of 70°C. The efficiency of boiler is 75% and factor of evaporation 1.15, specific heat of steam at constant pressure is 2.3 kJ/kg K. Calculate :*

(*i*) *Degree of superheat and temperature of steam generated.*

(*ii*) *Calorific value of coal in kJ/kg.*

(*iii*) *Equivalent evaporation in kg of steam per kg of coal.*

Solution. Steam generated per kg of coal, $m_a = 7.5$ kg

Steam pressure, $p = 11$ bar

Temperature of feed water $= 70°C$

Efficiency of boiler $= 75\%$

Factor of evaporation, $F_e = 1.15$

Specific heat of steam, $c_{ps} = 2.3$ kJ/kg K

At 11 bar : From steam tables,

$h_f = 781.1$ kJ/kg, $h_{fg} = 1998.5$ kJ/kg, $t_s = 184.1°C$ ($T_s = 273 + 184.1 = 457.1$ K)

Factor of evaporation, $$F_e = \frac{[\{h_f + h_{fg} + c_{ps}(T_{sup} - T_s)\} - h_{f_1}]}{2257}$$

$$1.15 = \frac{[781.1 + 1998.5 + 2.3(T_{sup} - 457.1) - 1 \times 4.18 \times (70 - 0)]}{2257}$$

$$= \frac{2779.6 + 2.3(T_{sup} - 457.1) - 292.6}{2257}$$

$$= \frac{2487 + 2.3(T_{sup} - 457.1)}{2257}$$

$$\therefore \quad (T_{sup} - 457.1) = \frac{(1.15 \times 2257 - 2487)}{2.3} = 47.2$$

i.e., $$T_{sup} = 504.3 \text{ K.}$$

(*i*) **Degree of superheat** $= (T_{sup} - T_s) = 504.3 - 457.1 =$ **47.2°C. (Ans.)**

(*ii*) **Calorific value of coal, C :**

Boiler efficiency $$= \frac{m_a(h - h_{f_1})}{C}$$

$$0.75 = \frac{m_a[\{h_f + h_{fg} + c_{ps}(T_{sup} - T_s)\} - h_{f_1}]}{C}$$

$$0.75 = \frac{7.5[\{781.1 + 1998.5 + 2.3(504.3 - 457.1)\} - 1 \times 4.18 \times (70 - 0)]}{C}$$

$$= \frac{7.5(2888.16 - 292.6)}{C} = \frac{19466.7}{C}$$

$$\therefore \quad C = \frac{19466.7}{0.75} = 25955 \text{ kJ/kg}$$

i.e., *Calorific value of coal* = **25955 kJ/kg. (Ans.)**

(*iii*) **Equivalent evaporation,**

$$m_e = \frac{m_a(h - h_{f_1})}{2257} = \frac{7.5(2888.16 - 292.6)}{2257}$$

$$= \mathbf{8.625 \text{ kg.} \ (Ans.)}$$

Example 3.9. *During the trial of water-tube boiler the following data were obtained :*

Steam pressure,	p = *13 bar*
Degree of superheat	= *77°C*
Temperature of feed water	= *85°C*
Water evaporated	= *3000 kg per hour*
Coal fired	= *410 kg per hour*
Ash	= *40 kg per hour*
Percentage of combustible in ash	= *9.6%*
Moisture in coal	= *4.5%*
Calorific value of dry coal/kg	= *30500 kJ/kg.*

Determine : (i) The efficiency of the boiler plant including superheater.

(ii) The efficiency of the boiler and furnace combined.

Take specific heat of superheated steam = 2.1 kJ/kg K.

Solution. ***At 13 bar :*** From steam tables,

$$t_s = 191.6°\text{C},\ h_f = 814.7 \text{ kJ/kg},\ h_{fg} = 1970.7 \text{ kJ/kg}$$

$$h = h_{sup} = h_f + h_{fg} + c_{ps}\,(T_{sup} - T_s)$$

$$= 814.7 + 1970.7 + 2.1 \times 77 = 2947.1 \text{ kJ/kg}$$

Also $\quad h_{f_1} = 1 \times 4.18 \times (85 - 0) = 355.3$ kJ/kg

∴ Total heat supplied to produce 1 kg of steam

$$= h_{sup} - h_{f_1} = 2947.1 - 355.3 = 2591.8 \text{ kJ/kg.}$$

(*i*) **Efficiency of boiler plant including superheater**

$$= \frac{m_a\,(h_{sup} - h_{f_1})}{C} \qquad \left[\begin{aligned}\text{Dry coal} &= 410 - 410 \times \frac{4.5}{100}\\ &= 391.55 \text{ kg}\end{aligned}\right]$$

$$= \frac{\dfrac{3000}{391.55}(2947.1 - 355.3)}{30500} = \mathbf{0.651 \text{ or } 65.1\%.} \quad \textbf{(Ans.)}$$

Combustible in ash per hour = $40 \times \dfrac{9.6}{100} = 3.84$ kg.

The combustible present in ash is practically carbon and its value may be taken as 33860 kJ/kg.

Heat actually supplied per hour

= Heat of dry coal – heat of combustible in the ash

= 391.55 × 30500 – 3.84 × 33860 = 11812253 kJ

Heat usefully utilised in the boiler per hour

$$= 3000(h_{sup} - h_{f_1}) = 3000\,(2947.1 - 355.3) = 777400 \text{ kJ.}$$

(*ii*) **Efficiency of the boiler and furnace combined**

$$= \frac{\text{Heat usefully utilised in boiler per hour}}{\text{Heat actually supplied per hour}}$$

$$= \frac{7775400}{11812253} = \mathbf{0.658 \text{ or } 65.8\%.} \quad \textbf{(Ans.)}$$

3.18. STEAM NOZZLES

3.18.1. Introduction

*A **steam nozzle** may be defined as a passage of varying cross-section, through which heat energy of steam is converted to kinetic energy.* Its major function is to produce steam jet with high velocity to drive steam turbines. A turbine nozzle performs *two functions* :

(*i*) It transforms a portion of energy of steam (obtained from steam generating unit) into kinetic energy.

(*ii*) In the impulse turbine it directs the steam jet of high velocity against blades, which are free to move in order to convert kinetic energy into shaft work. In reaction turbines the nozzles which are free to move, discharge high velocity steam. The reactive force of the steam against the nozzle produces motion and work is obtained.

The cross-section of a nozzle at first tapers to a smaller section (*to allow for changes which occur due to changes in velocity, specific volume and dryness fraction as the steam expands*) ; the smallest section being known as ***throat***, and then it diverges to a large diameter. The nozzle which converges to throat and diverges afterwards is known as ***convergent-divergent*** nozzle (Fig. 3.76). In convergent nozzle there is no divergence after the throat as shown in Fig. 3.77.

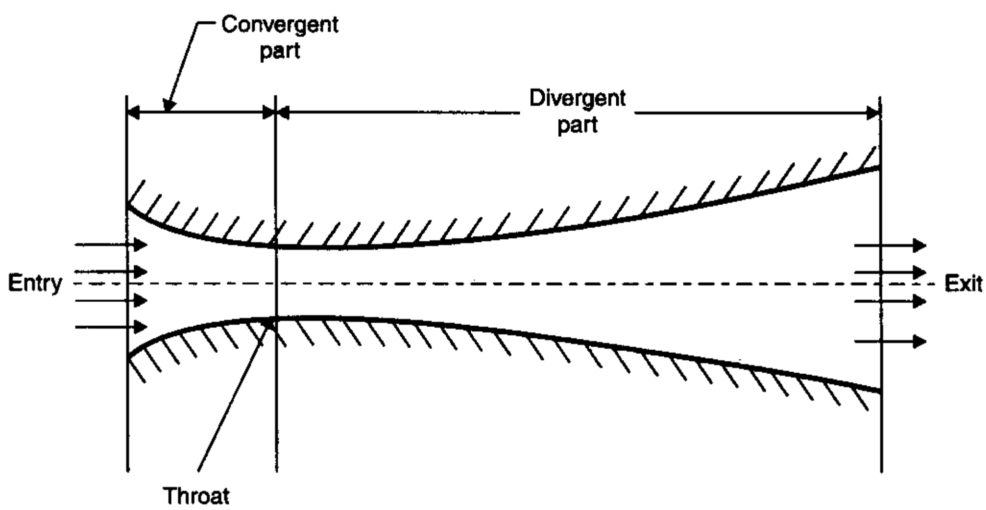

Fig. 3.76. Convergent-divergent nozzle.

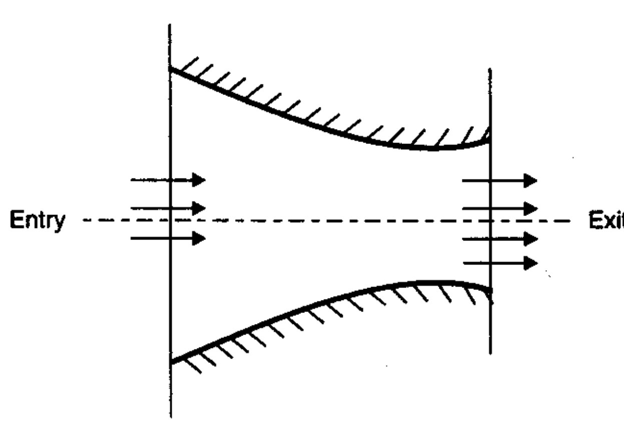

Fig. 3.77. Convergent nozzle.

In a convergent-divergent nozzle, because of the higher expansion ratio, addition of divergent portion produces steam at higher velocities as compared to a convergent nozzle.

3.18.2. Steam Flow Through Nozzles

The steam flow through the nozzle may be assumed as *adiabatic flow* since during the expansion of steam in nozzle neither any heat is supplied nor rejected, work, however, is performed by increasing the kinetic energy of the steam. As the steam passes through the nozzle it loses its pressure as well as the heat. *The work done is equal to the adiabatic heat drop which in turn is equal to Rankine area.*

3.18.2.1. Velocity of steam

Steam enters the nozzle with high pressure and low initial velocity (it is so small as compared to the final velocity that it is generally *neglected*) and leaves it with high velocity and low pressure. This is due to the reason that heat energy of steam is *converted* into kinetic energy as it (steam) passes through the nozzle. The final or outlet velocity of steam can be found as follows :

Let C = Velocity of steam at the section considered (m/sec),

h_1 = Enthalpy of steam entering the nozzle,

h_2 = Enthalpy of steam at section considered, and

h_d = Heat drop during expansion of steam is the nozzle = $(h_1 - h_2)$.

Consider 1 kg of steam and flow to be frictionless adiabatic.

Gain in kinetic energy = Adiabatic heat drop

$$\frac{C^2}{2} = h_d$$

$$\therefore \quad C = \sqrt{2 \times 1000\, h_d} \quad \text{where } h_d \text{ is in kJ}$$

$$= 44.72\sqrt{h_d} \quad \text{...(3.25)}$$

In practice, there is loss due to friction in the nozzle and its value varies from 10 to 15 per cent of total heat drop. Due to this, total heat drop is minimized. Let heat drop after deducting friction loss be kh_d.

$$\text{The velocity, } C = 44.72\sqrt{kh_d} \quad \text{...(3.26)}$$

3.18.3. Discharge through the Nozzle and Conditions for its Maximum Value

Let p_1 = Initial pressure of steam,

v_1 = Initial volume of 1 kg of steam at pressure p_1 (m^3),

p_2 = Steam pressure at the throat,

v_2 = Volume of 1 kg of steam at pressure p_2 (m^3),

A = Cross-sectional area of nozzle at throat (m^2), and

C = Velocity of steam (m/s).

The steam flowing through the nozzle follows approximately the equation given below :

$$pv^n = \text{Constant}$$

where, n = 1.135 for ***saturated steam***

and $\quad$ = 1.3 for ***superheated steam.***

[For **wet steam,** the value of n can be calculated by *Dr. Zenner's equation,*

$n = 1.035 + 0.1x$, where x is the initial dryness fraction of steam]

Work done per kg of steam during the cycle (Rankine area)

$$= \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

and, Gain in kinetic energy = Adiabatic heat drop
= Work done during Rakine cycle

or
$$\frac{C^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

$$= \frac{n}{n-1}\, p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right) \qquad ...(3.27)$$

Also
$$p_1v_1^{\,n} = p_2v_2^{\,n} \quad \text{or} \quad \frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{1/n} \qquad ...(3.28)$$

or
$$v_2 = v_1\left(\frac{p_1}{p_2}\right)^{1/n} \qquad ...(3.29)$$

Putting the value of v_2/v_1 from eqn. (3.28) in eqn. (3.27), we get

$$\frac{C^2}{2} = \frac{n}{n-1}\, p_1v_1\left[1 - \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{1/n}\right] = \frac{n}{n-1}\, p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}}\right]$$

$$= \frac{n}{n-1}\, p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$C^2 = 2\left(\frac{n}{n-1}\right) p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$C = \sqrt{2\left(\frac{n}{n-1}\right) p_1 v_1\left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\}} \qquad ...(3.30)$$

If m is the mass of steam discharged in kg/s.

Then
$$m = \frac{AC}{v_2} \qquad ...(3.31)$$

Substituting the value of v_2 from eqn. (3.29) in eqn. (3.31),

$$m = \frac{AC}{v_1\left(\frac{p_1}{p_2}\right)^{1/n}}$$

or
$$m = \frac{A}{v_1\left(\frac{p_1}{p_2}\right)^{1/n}}\sqrt{2\left(\frac{n}{n-1}\right) p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

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

$$= \frac{A}{v_1}\sqrt{\left[2\left(\frac{n}{n-1}\right)p_1 v_1\left\{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right\}\right]} \qquad ...(3.32)$$

It is obvious from above equation that there is only one value of the ratio (called *critical ratio pressure*) p_2/p_1 which will produce the *maximum discharge.* This can be obtained by differentiating 'm' with respect to (p_2/p_1) and equating it to zero.

As other quantities except the ratio p_2/p_1 are constant

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

or $$\frac{2}{n}\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1} - \left(\frac{n+1}{n}\right)\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}-1} = 0$$

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

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

or $$\left(\frac{p_2}{p_1}\right)^{2-n-1} = \left(\frac{n+1}{2}\right)^{n} \quad \text{or} \quad \frac{\mathbf{p_2}}{\mathbf{p_1}} = \left(\frac{\mathbf{2}}{\mathbf{n+1}}\right)^{\frac{\mathbf{n}}{\mathbf{n-1}}} \qquad ...(3.33)$$

Hence the discharge through the nozzle will be the maximum when critical pressure ratio, *i.e.*,

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

For saturated steam : $n = 1.135$

$$\frac{p_2}{p_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}} = 0.58$$

For superheated steam : $n = 1.3$

$$\frac{p_2}{p_1} = \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}} = \left(\frac{2}{2.3}\right)^{\frac{1.3}{0.3}} = 0.546$$

Substituting the value of $\frac{p_2}{p_1}$ from eqn. (3.33) into eqn. (3.32), we get the maximum discharge,

$$m_{max} = \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left[\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{2}{n}}\left\{\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}\right\}^{\frac{n+1}{n}}\right]}$$

$$= \frac{A}{v_1}\sqrt{2\left(\frac{n}{n-1}\right)p_1v_1\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}-\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

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

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

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

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

$$= A\sqrt{2\left(\frac{n}{n-1}\right)\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\left(\frac{n-1}{2}\right)}$$

i.e.,
$$m_{max} = A\sqrt{n\left(\frac{p_1}{v_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}} \quad \text{...[3.33 (a)]}$$

From the above equation it is evident that *the maximum mass flow depends only on the initial condition of the steam (p_1, v_1) and the throat area* and is *independent of the final pressure of steam i.e.* at the exit of the nozzle. *The addition of the divergent part of the nozzle after the throat does not affect the discharge of steam passing through the nozzle but it only accelerates the steam leaving the nozzle.*

It may be noted that the discharge through nozzle increases as the pressure at the throat of the nozzle (p_2) decreases, when the supply pressure p_1 is constant. But once the nozzle pressure p_2 reaches the critical value [given by equation (3.33)], the discharge reaches a maximum and after that the throat pressure and mass flow remains constant irrespective of the pressure at the exit.

The velocity of steam at the throat of the nozzle when the discharge is maximum is obtained by substituting the value of $\frac{p_2}{p_1}$ from eqn. (3.33) into eqn. (3.30).

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

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

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

i.e.,

$$C_{max} = \sqrt{2\left(\frac{n}{n+1}\right)p_1v_1} \qquad ...(3.34)$$

The above equation indicates that the *velocity is also dependent on the initial conditions of the steam.*

3.18.4. Nozzle Efficiency

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

(*i*) *The friction between the nozzle surface and steam ;*

(*ii*) *The internal friction of steam itself ; and*

(*iii*) *The shock losses.*

Most of these frictional losses occur between the throat and exit in convergent-divergent nozzle. These frictional losses entail the following *effects* :

(*i*) *The expansion is no more isentropic and enthalpy drop is reduced ;*

(*ii*) *The final dryness friction of steam is increased as the kinetic energy gets converted into heat due do friction and is absorbed by steam ;*

(*iii*) *The specific volume of steam is increased as the steam becomes more dry due to this frictional reheating.*

Fig. 3.78 represents on Mollier diagram the **effect of friction** on steam flow through a nozzle.

The point 1 represents the initial condition of steam which enters the nozzle in a dry saturated state. If the friction is neglected, the expansion of steam from entry to throat is represented by the vertical line 1-2 and that from the throat to the exit by 2-3. Now if the friction were taken into account the heat drop would have been somewhat less than 1-3. Let this heat drop be 1-3′. From 3′ draw a horizontal line which cuts the same pressure line on which point 3 lies, at the point 2′ which represents the final condition of steam. It may be noted that dryness fraction of steam is **more** at point 2′ than at point 3. Hence *the effect of friction is to improve the quality of steam.* The value of co-efficient '*k*' in the equation for the velocity of expanding steam is given by :

$$k = \frac{\text{Actual heat drop}}{\text{Isentropic heat drop}} = \frac{1-3'}{1-3} = \frac{h_1 - h_3'}{h_1 - h_3}.$$

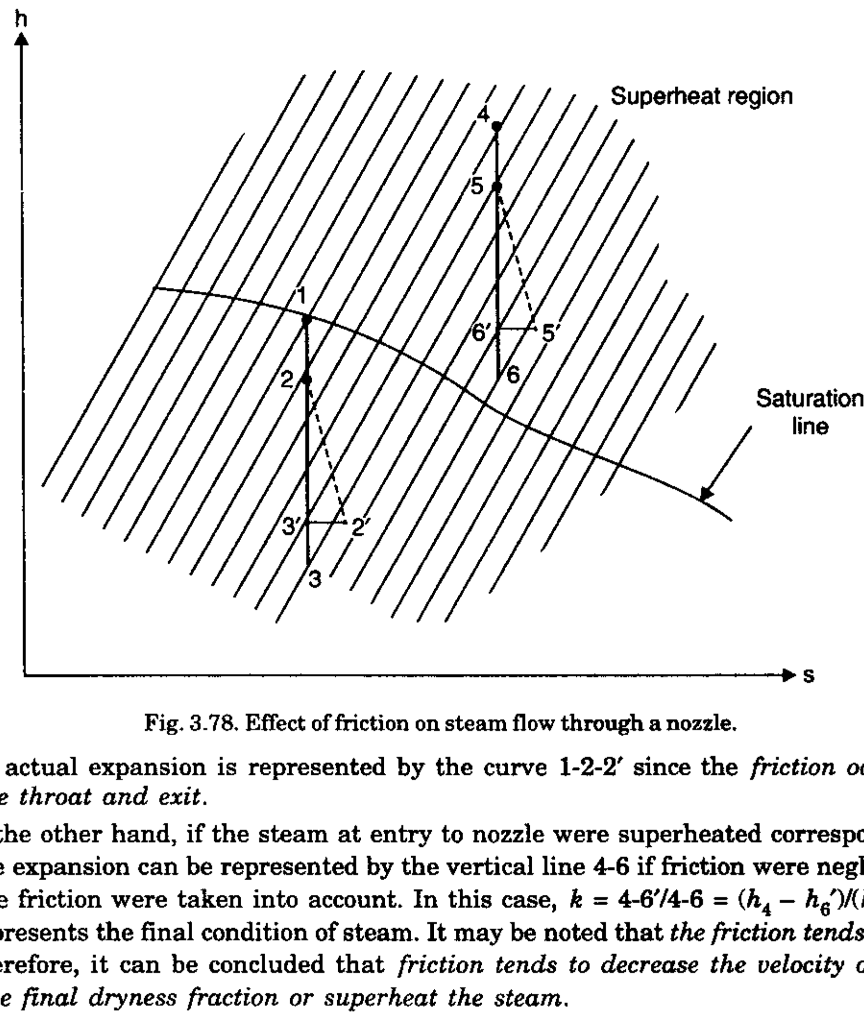

Fig. 3.78. Effect of friction on steam flow through a nozzle.

The actual expansion is represented by the curve 1-2-2′ since the *friction occurs mainly between the throat and exit.*

On the other hand, if the steam at entry to nozzle were superheated corresponding to the point 4, the expansion can be represented by the vertical line 4-6 if friction were neglected and by 4-5-5′ if the friction were taken into account. In this case, k = 4-6′/4-6 = $(h_4 - h_6')/(h_4 - h_6)$. The point 5′ represents the final condition of steam. It may be noted that *the friction tends to superheat steam*. Therefore, it can be concluded that *friction tends to decrease the velocity of steam and increase the final dryness fraction or superheat the steam.*

The *nozzle efficiency* is, therefore, defined *as the ratio of the actual enthalpy drop to the isentropy enthalpy drop between the same pressures,*

i.e., $$\text{Nozzle efficiency} = \frac{h_1 - h_3'}{h_1 - h_3} \text{ or } \frac{h_4 - h_6'}{h_4 - h_6} \text{ as the case may be} \qquad ...(3.35)$$

If the actual velocity at exit from the nozzle is C_2' and the velocity at exit when the flow is isentropic is C_3, then using the steady flow energy equation, in each case we have

$$h_1 + \frac{C_1^2}{2} = h_3 + \frac{C_3^2}{2} \quad \text{or} \quad h_1 - h_3 = \frac{C_3^2 - C_1^2}{2}$$

and $$h_1 + \frac{C_1^2}{2} = h_2' + \frac{C_2'^2}{2} \quad \text{or} \quad h_1 - h_2' = \frac{C_2'^2 - C_1^2}{2}$$

$$\therefore \text{ Nozzle efficiency} = \frac{C_2'^2 - C_1^2}{C_3^2 - C_1^2} \qquad ...(3.36)$$

When the inlet velocity, C_1, is negligibly small then

Nozzle efficiency $= \dfrac{C_2'^2}{C_3^2}$ ...(3.37)

Sometimes a *velocity co-efficient is defined as the ratio of the actual exit velocity to the exit velocity when the flow is isentropic between the same pressures,*

i.e., Velocity co-efficient $= \dfrac{C_2'}{C_3}$ ...(3.38)

It can be seen from eqns. (3.37) and (3.38) that the velocity co-efficient is the square root of the nozzle efficiency, when the inlet velocity is assumed to be negligible.

3.18.5. Supersaturated or Metastable Expansion of Steam in a Nozzle

When steam flows through a nozzle, it would normally be expected that the discharge of steam through the nozzle would be slightly less than the theoretical value. But it has been observed during experiments on flow of wet steam that the *discharge is slightly greater than that calculated by the formula.* This phenomenon is explained as follows : The converging part of the nozzle is so short and the steam velocity so high that the molecules of steam have insufficient time to collect and form droplets so that normal condensation does not take place. Such rapid expansion is said to be *metastable* and produces a *supersaturated state*. In this state of supersaturation the steam is undercooled to a temperature less than that corresponding to its pressure ; consequently the density of steam increases and hence the weight of discharge. Prof. Wilson through experiments showed that dry saturated steam, when suddenly expanded *in the absence of dust, does not condense until it density is about 8 times that of the saturated vapour of the same pressure.* This effect is discussed below :

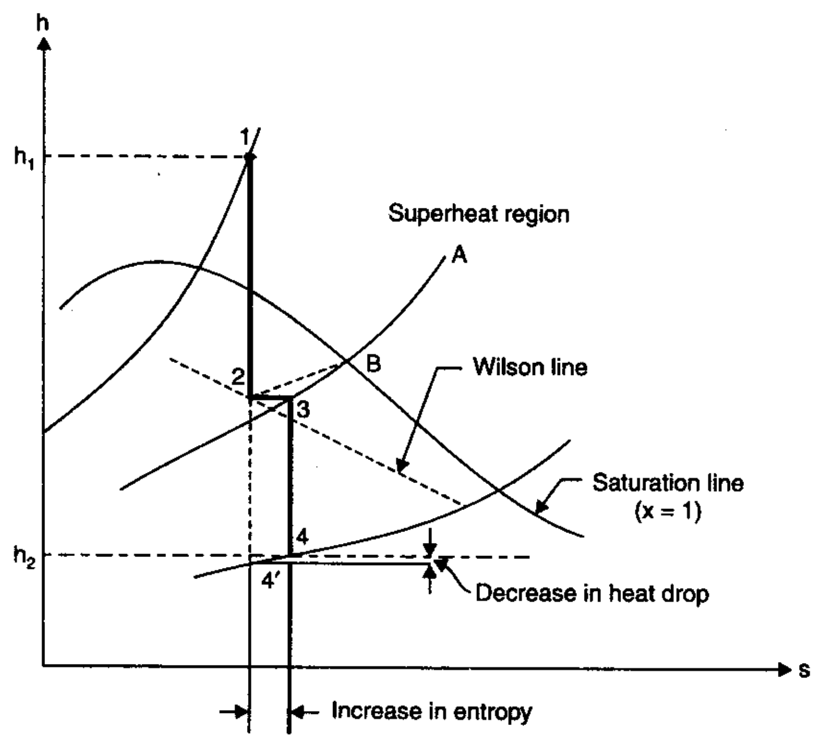

Fig. 3.79. Supersaturated flow of steam.

Refer Fig. 3.79. The point 1 represents initial state of the steam. The steam expands isentropically without any condensation to point 2, 2 being on the superheat constant pressure curve *AB* produced. At point 2 the limit of supersaturation is reached and steam reverts to its normal condition at 3 at the same enthalpy value as 2, and at the same pressure. The steam continues expanding isentropically to a lower pressure to point 4 instead of 4′ which would have been reached if thermal equilibrium had been maintained. Consequently, enthalpy drop is reduced and the condition of the final steam is improved. *The limiting condition of under-cooling at which condensation commences and is assumed to restore conditions of normal thermal equilibrium is called the "Wilson Line".*

It may be noted that when metastable conditions prevail the h-s chart/diagram should not be used and the expansion must be considered to follow the law $pv^{1.3} = C$, i.e., with the index of expansion for superheated steam. Thus,

$$\text{Enthalpy drop} \quad = \frac{n}{(n-1)} = p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

The relationship, $T_2/T_1 = (p_2/p_1)^{\frac{n-1}{n}}$ may be used to calculate *supercooled temperature*. The *'degree of undercooling' is then the difference between the saturation temperature and the supercooled temperature.*

Effects of supersaturation. In a nozzle in which supersaturation occurs the effects may be summarised as follows :

(*i*) There is an increase in the entropy and specific volume of steam.

(*ii*) The heat drop is reduced below that for thermal equilibrium as a consequence the exit velocity of steam is reduced.

(*iii*) Since the condensation does not take place during supersaturated expansion, so the temperature at which the supersaturation occurs will be *less* than the saturation temperature corresponding to the pressure. Therefore, the density of supersaturated steam will be more than that for the equilibrium conditions which gives the increase in the mass of steam discharged.

(*iv*) The dryness fraction of steam is improved.

The problems on supersaturated flow cannot be solved by Mollier chart unless Wilson line is drawn on it.

The velocity of steam at the end of expansion is found by using the relation,

$$C_2 = \sqrt{2 \times \left(\frac{n}{n-1}\right) p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

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

$$\text{Apparent temperature,} \quad T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

and

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

Example 3.10. *Dry saturated steam at a pressure of 11 bar enters a convergent-divergent nozzle and leaves at a pressure of 2 bar. If the flow is adiabatic and frictionless, determine :*

(i) The exit velocity of steam.

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

Assume the index of adiabatic expansion to be 1.135.

Solution. Refer Fig. 3.80.

$$p_1 = 11 \text{ bar} \; ; \; p_3 = 2 \text{ bar} \; ; \; p_2 = \text{Throat pressure} \; ; \; n = 1.135$$

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

$$= \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}} = 0.58$$

or
$$p_2 = 0.58 \times p_1 = 0.58 \times 11 = 6.38 \text{ bar.}$$

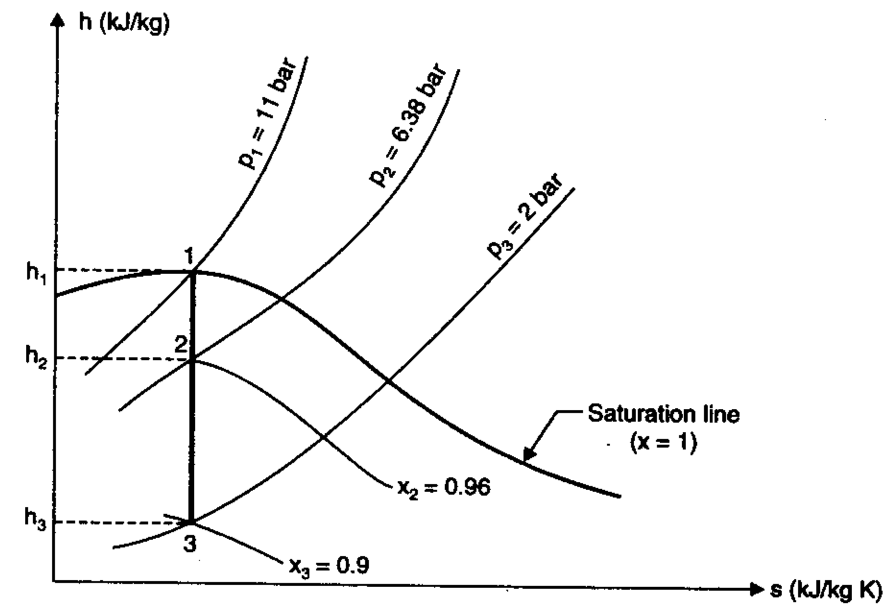

Fig. 3.80

- From Mollier chart (Fig. 3.80) point 1 is located on the dry saturation line corresponding to 11 bar pressure.
- From '1' vertical line 1-3 is drawn cutting the pressure line 2 bar.
- Point '2' corresponding to throat pressure 6.38 bar is located on the vertical line.

Adiabatic heat drop between inlet and throat,

$$h_d = h_1 - h_2$$
$$= 2780 - 2679 = 101 \text{ kJ/kg}$$
$$x_2 = 0.96$$
$$v_{g_2} = 0.297 \text{ m}^3/\text{kg}$$

Throat velocity,
$$C_2 = 44.72 \sqrt{h_d} = 44.72 \sqrt{101} = 449.4 \text{ m/s}$$

Also, $\dot{m} = \dfrac{A_2C_2}{v_2} = \dfrac{A_2C_2}{x_2 v_{g_2}}$ (where $\dot{m}$ = Mass flow rate in kg/s)

or Throat area, $A_2 = \dfrac{\dot{m}\, x_2\, v_{g_2}}{C_2} = \dfrac{\dot{m} \times 0.96 \times 0.297}{449.4} = 0.000634\ \dot{m}$

From Mollier chart, $x_3 = 0.9$

From steam tables, $v_{g_3} = 0.885$ m³/kg (at 2 bar)

h_d' = Adiabatic heat drop between inlet and exit

$= h_1 - h_3 = 2780 - 2480 = 300$ kJ/kg

Exit velocity, $C_3 = 44.72\ \sqrt{h_d'} = 44.72\ \sqrt{300}$ = **774.6 m/s. (Ans.)**

Exit area, $A_3 = \dfrac{\dot{m}\, x_3\, v_{g_3}}{C_3} = \dfrac{\dot{m} \times 0.9 \times 0.885}{774.6} = 0.001028\ \dot{m}$

∴ **Ratio** of $\dfrac{\text{exit area}}{\text{throat area}} = \dfrac{0.001028\, \dot{m}}{0.000638\, \dot{m}}$ = **1.62. (Ans.)**

Example 3.11. *The nozzles of a Delaval steam turbine are supplied with dry saturated steam at a pressure of 9 bar. The pressure at the outlet is 1 bar. The turbine has two nozzles with a throat diameter of 2.5 mm. Assuming nozzle efficiency as 90% and that of turbine rotor 35%, find the quality of steam used per hour and the power developed.*

Solution. $p_1 = 9$ bar, $p_3 = 1$ bar, p_2 = throat pressure, Number of nozzles = 2

We know that, $\dfrac{p_2}{p_1} = \left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}}$

Since the steam is dry and saturated, $n = 1.135$

∴ $\dfrac{p_2}{p_1} = \left(\dfrac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}} = 0.58$

or $p_2 = 0.58\, p_1 = 0.58 \times 9 = 5.22$ bar.

From Mollier chart :

$h_1 = 2770$ kJ/kg, $h_2 = 2670$ kJ/kg

$h_3 = 2400$ kJ/kg

$x_{2'} = 0.96$, $x_{3'} = 0.88$

Now, $h_d = h_1 - h_2 = 2770 - 2670 = 100$ kJ/kg

$h_d' = h_1 - h_3 = 2770 - 2400 = 370$ kJ/kg

From steam tables :

$v_{g_2} = 0.361$ m³/kg (at 5.22 bar)

$v_{g_3} = 1.694$ m³/kg (at 1.0 bar)

Velocity of steam at throat,

$C_2' = 44.72\ \sqrt{kh_d} = 44.72\ \sqrt{0.9 \times 100} = 424.2$ m/s

Exit velocity, $C_3' = 44.72\ \sqrt{kh_{d'}} = 44.72\ \sqrt{0.9 \times 370} = 816$ m/s

$D_2 = 2.5$ mm,

$$A_2 = \frac{\pi}{4} D_2^2 \times 2 = \frac{\pi}{4} \times \left(\frac{2.5}{1000}\right)^2 \times 2 = 9.82 \times 10^{-6} \text{ m}^2$$

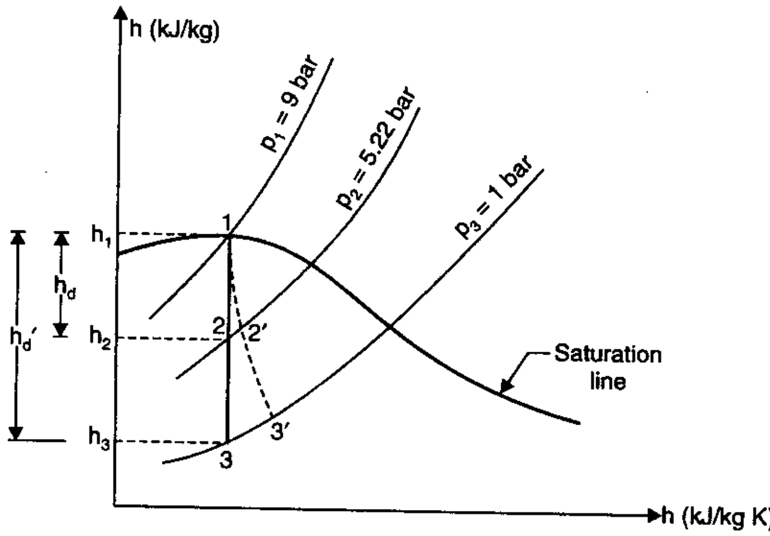

Fig. 3.81

Mass of steam used per sec.,

$$\dot{m} = \frac{A_2 C_2'}{x_2' v_{g_2}} = \frac{9.82 \times 10^{-6} \times 424.2}{0.96 \times 0.361} = 0.012 \text{ kg/s}$$

Energy supplied by the steam to the wheel per sec

$$= \frac{\dot{m} C_3'^2}{2} = \frac{0.012 \times 816^2}{2} = 3995 \text{ W} \simeq \mathbf{4 \text{ kW}}.$$

$\therefore$ Useful work $= \eta_{turbine} \times 4 = 0.35 \times 4 = 1.44$ kW.

i.e., *Power developed* = **1.44 kW. (Ans.)**

Example 3.12. *An impulse turbine having a set of 16 nozzles receives steam at 20 bar, 400°C. The pressure of steam at exit is 12 bar. If the total discharge is 260 kg/min and nozzle efficiency is 90%, find the cross-sectional area of the exit of each nozzle. If the steam has a velocity of 80 m/s at entry to the nozzles, find the percentage increase in discharge.*

Solution. Set of nozzles = 16, p_1 = 20 bar, 400°C

Total discharge = 260 kg/min, η_{nozzle} = 90%.

Since the steam supplied to the nozzle is superheated, the throat pressure is given by :

$$\frac{p_2}{p_1} = 0.546$$

$\therefore$ $p_2 = 20 \times p_1 = 20 \times 0.546$

$= 10.9$ bar

$$\left[\begin{array}{l} \text{For superheated steam : } n = 1.3 \\ \dfrac{p_2}{p_1} = \left(\dfrac{2}{n+1}\right)^{\frac{n}{n-1}} = \left(\dfrac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}} \\ = 0.546 \end{array}\right]$$

Since the pressure is *less* than the exit pressure of steam from nozzle, as such the nozzle is *convergent type.*

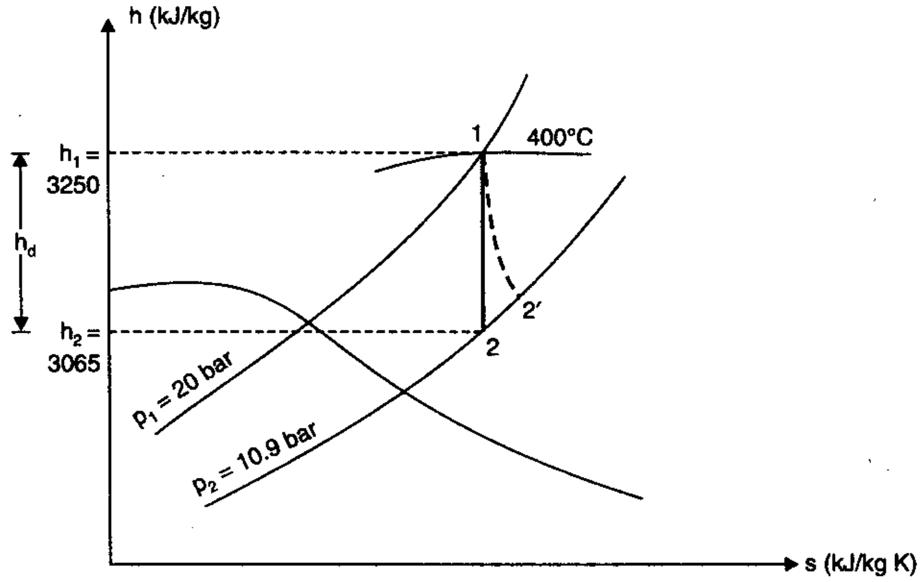

Fig. 3.82

From Mollier chart :

$$h_1 = 3250 \text{ kJ/kg}$$
$$h_2 = 3065 \text{ kJ/kg}$$
$$h_d = h_1 - h_2 = 3250 - 3065 = 185 \text{ kJ/kg}.$$

Velocity at exit neglecting initial velocity of steam,

$$C_2' = 44.72 \sqrt{kh_d} = 44.72 \sqrt{0.9 \times 185} = 577 \text{ m/s}$$

Specific volume at exit,

$$v_2' = 0.235 \text{ m}^3\text{/kg} \qquad \text{(From Mollier chart)}$$

Area at exit for one nozzle,

$$A_2 = \frac{m \times v_2}{C_2' \times \text{no. of nozzles}} = \frac{260 \times 0.235}{60 \times 577 \times 16} = \mathbf{1.1 \times 10^{-4} \ m^2}. \quad \textbf{(Ans.)}$$

Taking into account the initial velocity of steam as 80 m/s, the velocity of steam at exit, C_2' is calculated as follows :

$$\frac{C_2'^2 - C_1^2}{2} = kh_d$$

i.e.,
$$C_2'^2 = 2\,kh_d + C_1^2$$
$$= 2 \times 0.9 \times 185 \times 1000 + 80^2 = 339400$$

$\therefore$
$$C_2' = 582.6 \text{ m/s}$$

Percentage increase in velocity

$$= \frac{582.6 - 577}{577} \times 100 = \mathbf{0.97\%}.$$

This will result in 0.97% increase in discharge as specific volume will not be affected by velocity of approach.

Hence *percentage increase in discharge* = **0.97%. (Ans.)**

3.19. STEAM TURBINES

3.19.1. Introduction

*The **steam turbine** is a prime-mover in which the potential energy of the steam is transformed into kinetic energy, and latter in its turn is transformed into the mechanical energy of rotation of the turbine shaft.* The turbine shaft, directly or with the help of a reduction gearing, is connected with the driven mechanism. Depending on the type of the driven mechanism a steam turbine may be *utilised in most diverse fields of industry, for power generation* and *for transport.* Transformation of the potential energy of steam into the mechanical energy of rotation of the shaft is brought about by different means.

3.19.2. Classification of Steam Turbines

There are several ways in which the steam turbines may be classified. The most important and common division being with respect to the *action of the steam*, as :

(*a*) Impulse.

(*b*) Reaction.

(*c*) Combination of impulse and reaction.

Other classification are :

1. **According to the number of pressure stages :**

(*i*) *Single-stage turbines* with one or more velocity stages usually of small-power capacities ; these turbines are mostly used for driving centrifugal compressors, blowers and other similar machinery.

(*ii*) *Multi-stage impulse and reaction turbines* ; they are made in a wide range of power capacities varying from small to large.

2. **According to the direction of steam flow :**

(*i*) *Axial turbines* in which steam flows in a direction parallel to the axis of the turbine.

(*ii*) *Radial turbines* in which steam flows in a direction perpendicular to the axis of the turbine ; one or more low-pressure stages in such turbines are made axial.

3. **According to the number of cylinders :**

(*i*) Single-cylinder turbines.

(*ii*) Double-cylinder turbines.

(*iii*) Three-cylinder turbines.

(*iv*) Four-cylinder turbines.

Multi-cylinder turbines which have their rotors mounted on one and the same shaft and coupled to a single generator are known as *single shaft turbines* ; turbines with separate rotor shafts for each cylinder placed parallel to each other are known as ***multiaxial turbines.***

4. **According to the method of governing :**

(*i*) *Turbines with throttle governing* in which fresh steam enters through one or more (depending on the power developed) simultaneously operated throttle valves.

(*ii*) *Turbines with nozzle governing* in which fresh steam enters through two or more consecutively opening regulators.

(*iii*) *Turbines with by pass governing* in which steam turbines besides being fed to the first stage is also directly fed to one, two or even three intermediate stages of the turbine.

5. **According to heat drop process :**

(*i*) *Condensing turbines with generators* ; in these turbines steam at a pressure less than atmospheric is directed to a condenser ; besides, steam is also extracted from intermediate stages for feed water heating, the number of such extractions usually being from 2-3 to as much 8-9. The latent heat of exhaust steam during the process of condensation is completely lost in these turbines.

(*ii*) *Condensing turbines with one or two intermediate stage extractions* at specific pressures for industrial and heating purposes.

(*iii*) *Back pressure turbines*, the exhaust steam from which is utilised for industrial or heating purposes ; to this type of turbines can also be added (in a relative sense) turbines with deteriorated vacuum, the exhaust steam of which may be used for heating and process purposes.

(*iv*) *Topping turbines* ; these turbines are also of the back pressure type with the difference that the exhaust steam from these turbines is further utilised in medium and low-pressure condensing turbines. These turbines, in general, operate at high initial conditions of steam pressure and temperature, and are mostly used during extension of power station capacities, with a view to obtain better efficiencies.

(*v*) *Back pressure turbines with steam extraction from intermediate stages at specific pressure* ; turbines of this type are meant for supplying the consumer with steam of various pressures and temperature conditions.

(*vi*) *Low pressure turbines* in which the exhaust steam from reciprocating steam engines, power hammers, etc., is utilised for power generation purposes.

(*vii*) *Mixed pressure turbines* with two or three pressure stages, with supply of exhaust steam to its intermediate stages.

6. **According to steam conditions at inlet to turbine :**

(*i*) *Low pressure turbines*, using steam at a pressure of 1.2 to 2 ata.

(*ii*) *Medium pressure turbines*, using steam at pressure of up to 40 ata.

(*iii*) *High pressure turbines*, utilising steam at pressures above 40 ata.

(*iv*) *Turbines of very high pressures*, utilising steam at pressures of 170 ata and higher and temperatures of 550°C and higher.

(*v*) *Turbines of supercritical pressures*, using steam at pressures of 225 ata and above.

7. **According to their usage in industry :**

(*i*) *Stationary turbines with constant speed of rotation* primarily used for driving alternators.

(*ii*) *Stationary steam turbines* with variable speed meant for driving turbo-blowers, air circulators, pumps, etc.

(*iii*) *Non-stationary* turbines with variable speed ; turbines of this type are usually employed in steamers, ships and railway locomotives.

3.19.3. Advantages of Steam Turbine over the Steam Engines

The following are the principal *advantages* of steam turbine over steam engines :

1. The thermal efficiency of a steam turbine is much higher than that of a steam engine.
2. The power generation in a steam turbine is at a uniform rate, therefore necessity to use a flywheel (as in the case of steam engine) is not felt.
3. Much higher speeds and greater range of speed is possible than in case of a steam engine.
4. In large thermal stations where we need higher outputs, the steam turbines prove very suitable as these can be made in big sizes.
5. With the absence of reciprocating parts (as in steam engine) the balancing problem is minimised.
6. No internal lubrication is required as there are no rubbing parts in the steam turbine.
7. In a steam turbine there is no loss due to initial condensation of steam.
8. It can utilise high vacuum very advantageously.
9. Considerable overloads can be carried at the expense of slight reduction in overall efficiency.

3.19.4. Description of Common Types of Turbines

The common types of steam turbines are :

1. Simple Impulse turbine.
2. Reaction turbine.

The main difference between these turbines lies *in the way in which the steam is expanded while it moves through them. In the former type steam expands in the nozzles and its pressure does not alter as it moves over the blades while in the latter type the steam expands continuously as it passes over the blades and thus there is gradual fall in the pressure during expansion.*

1. Simple impulse turbine :

Fig. 3.83 shows a simple impulse turbine diagrammatically. The top portion of the figure exhibits a longitudinal section through the upper half of the turbine, the middle portion shows one set of nozzles which is followed by a ring of moving blades, while lower part of the diagram indicates approximately changes in pressure and velocity during the flow of steam through the turbine. This turbine is called *'simple'* impulse turbine since the expansion of the steam takes place in one set of the nozzles.

As the steam flows through the nozzle its pressure falls from steam chest pressure to condenser pressure (or atmospheric pressure if the turbine is non-condensing). Due to this relatively higher ratio of expansion of steam in the nozzles the steam leaves the nozzle with a very high velocity. Refer Fig. 3.83, it is evident that the velocity of the steam leaving the moving blades is a large portion of the maximum velocity of the steam when leaving the nozzle. The loss of energy due to this higher exit velocity is commonly the ***"carry over loss"*** or ***"leaving loss"***.

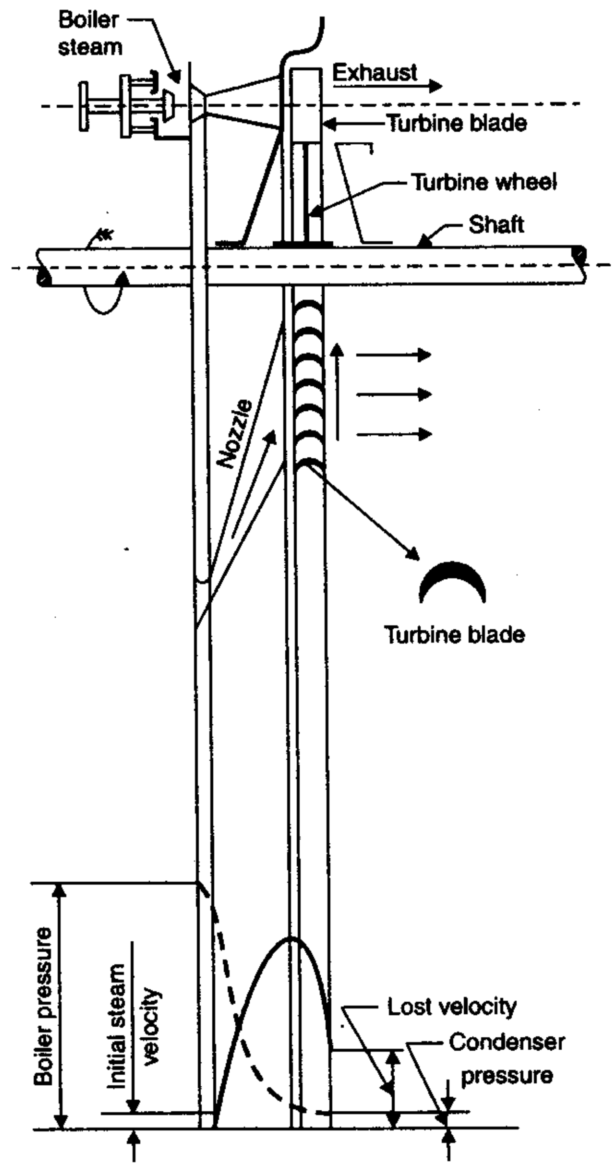

Fig. 3.83. Simple impulse turbine.

The principal example of this turbine is the well known ***"De laval turbine"*** and in this turbine the 'exit velocity' or 'leaving velocity' or 'lost velocity' may amount to 3.3 per cent of the nozzle outlet velocity. Also since all the kinetic energy is to be absorbed by one ring of the moving blades only, the velocity of wheel is too high (varying from 25,000 to 30,000 r.p.m.). This wheel or rotor speed however, can be reduced by different methods (discussed in the Art. 3.19.5.).

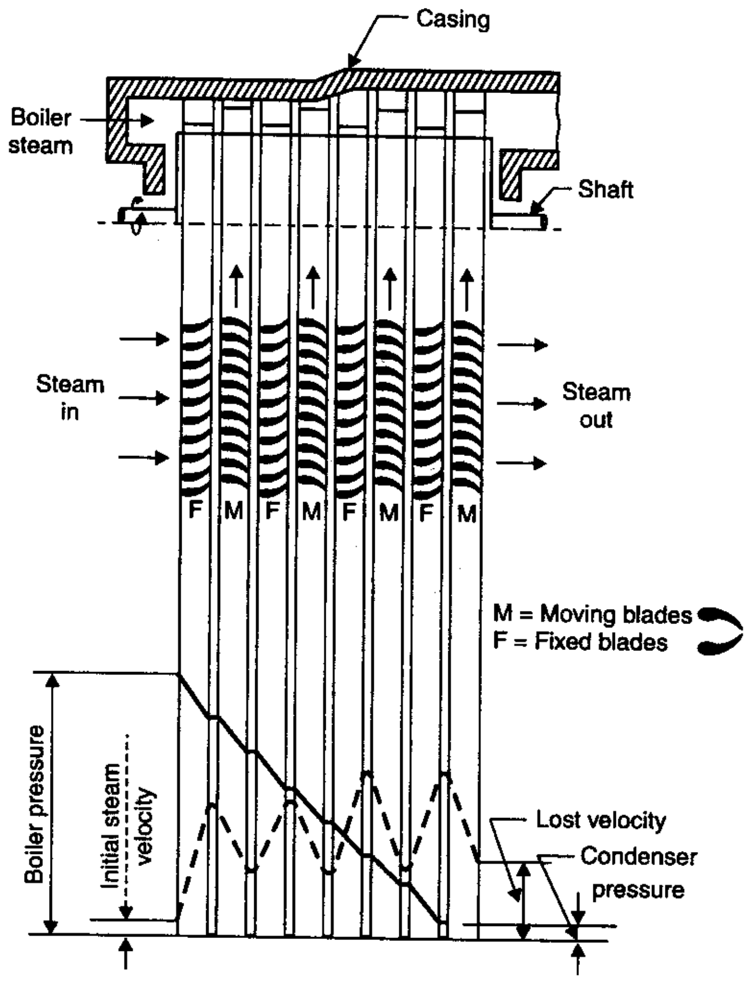

Fig. 3.84. Reaction turbine (three stage).

2. Reaction turbine :

In this type of turbine, *there is a gradual pressure drop and takes place continuously over the fixed and moving blades*. The function of the fixed blades is (the same as the nozzle) that they alter the direction of the steam as well as allow it to expand to a larger velocity. As the steam passes over the moving blades its kinetic energy (obtained due to fall in pressure) is absorbed by them. Fig. 3.84 shows a *multi-stage* reaction turbine. The changes in pressure and velocity are also shown therein.

As the volume of steam increases at lower pressures therefore, the diameter of the turbine must increase after each group of blade rings. It may be noted that in this turbine since the pressure drop per stage is small, therefore, the number of stages required is much higher than an impulse turbine of the same capacity.

3.19.5. Methods of Reducing wheel or Rotor Speed

As already discussed under the heading 'simple impulse turbine' that if the steam is expanded from the boiler pressure to condenser pressure in one stage the speed of the rotor becomes *tremendously high which crops up practical complicacies.* There are several methods of reducing this speed to lower value ; all these methods utilise a multiple system of rotor in series, keyed on a common shaft and the steam pressure of jet velocity is absorbed in stages as the steam flows over the blades. This is known as '**compounding**'. The different methods of compounding are :

1. Velocity compounding.
2. Pressure compounding.
3. Pressure velocity compounding.
4. Reaction turbine.

1. Velocity compounding :

Steam is expanded through a stationary nozzle from the boiler or inlet pressure to condenser pressure. So the pressure in the nozzle drops, the kinetic energy of the steam increases due to increase in velocity. A portion of this available energy is absorbed by a row of moving blades. The

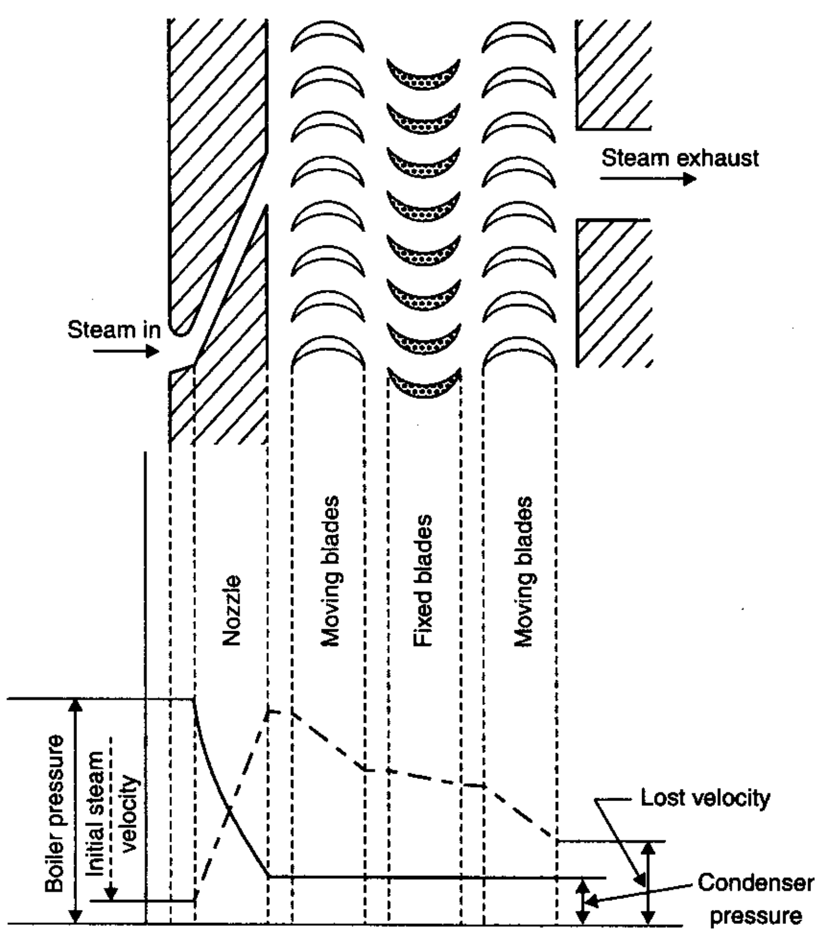

Fig. 3.85. Velocity compounding.

steam (whose velocity has decreased while moving over the moving blades) then flows through the second row of blades which are fixed. The function of these fixed blades is to redirect the steam flow without altering its velocity to the following next row moving blades where again work is done on them and steam leaves the turbine with a low velocity. Fig. 3.85 shows a cut away section of such a stage and changes in pressure and velocity as the steam passes through the nozzle, fixed and moving blades.

Though this method has the advantage that the initial cost is low due to lesser number of stages yet its efficiency is low.

2. Pressure compounding :

Fig. 3.86 shows rings of fixed nozzles incorporated between the rings of moving blades. The steam at boiler pressure enters the first set of nozzles and expands partially. The kinetic energy of the steam thus obtained is absorbed by the moving blades (stage 1). The steam then expands partially in the second set of nozzles where its pressure again falls and the velocity increases ; the

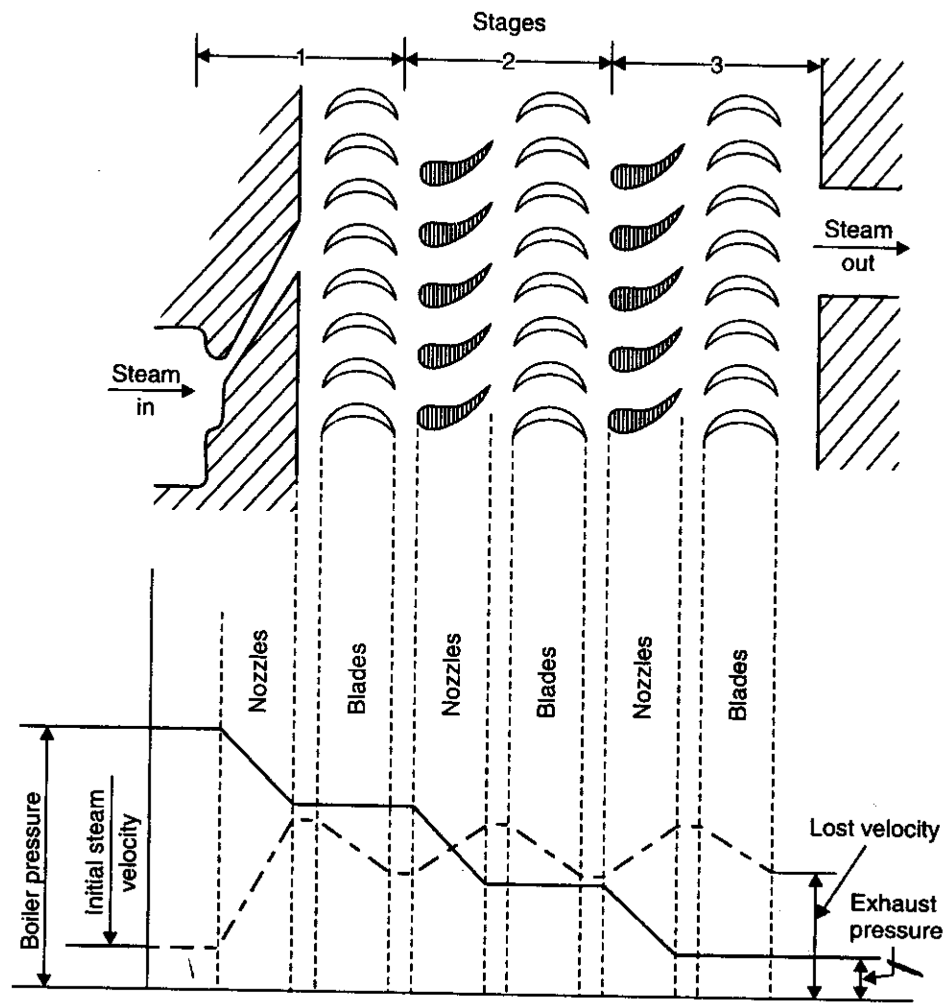

Fig. 3.86. Pressure compounding.

kinetic energy so obtained is absorbed by the second ring of moving blades (stage 2). This is repeated in stage 3 and steam finally leaves the turbine at low velocity and pressure. The number of stages (or pressure reductions) depends on the number of rows of nozzles through which the steam must pass.

This method of compounding is used in *Rateau and Zoelly turbine. This is most efficient turbine since the speed ratio remains constant but it is expensive owing to a large number of stages.*

3. Pressure velocity compounding :

This method of compounding is the combination of two previously discussed method. The total drop in steam pressure is divided into stages and the velocity obtained in each stage is also compounded. The rings of nozzles are fixed at the beginning of each stage and pressure remains constant during each stage. The changes in pressure and velocity are shown in Fig. 3.87.

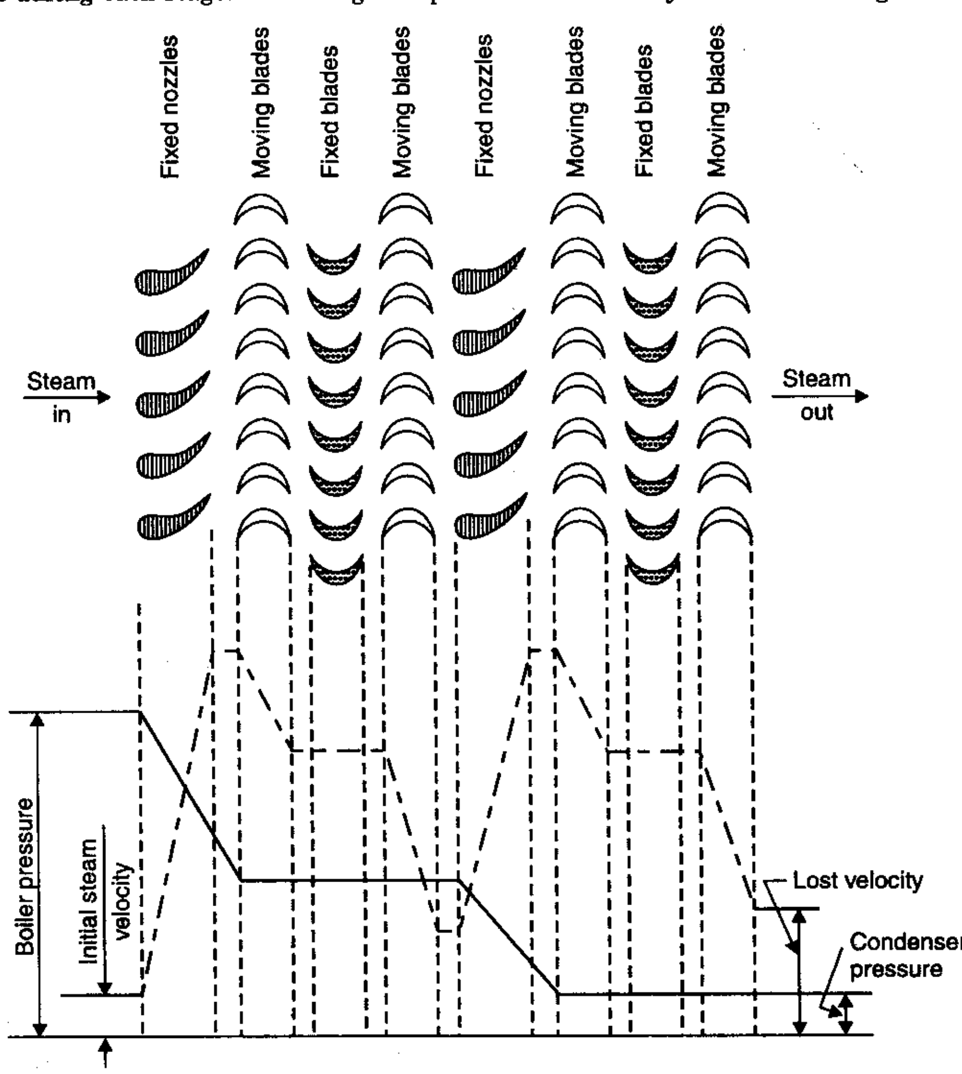

Fig. 3.87. Pressure velocity compounding.

This method of compounding is used in *Curits and Moore turbine.*

4. Reaction turbine

It has been discussed in Article 3.19.4.

3.19.6. Difference between Impulse and Reaction Turbines

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

3.19.7. Impulse Turbines

3.19.7.1. Velocity diagram for moving blade

Fig. 3.88 shows the velocity diagram of a *single stage impulse turbine.*

C_{bl} = Linear velocity of moving blade (m/s)

C_1 = Absolute velocity of steam entering moving blade (m/s)

C_0 = Absolute velocity of steam leaving moving blade (m/s)

C_{w_1} = Velocity of whirl at the entrance of moving blade
= tangential component of C_1

C_{w_0} = Velocity of whirl at exit of the moving blade
= tangential component of C_0

C_{f_1} = Velocity of flow at entrance of moving blade
= axial component of C_1

C_{f_0} = Velocity of flow at exit of the moving blade
= axial component of C_0

C_{r_1} = Relative velocity of steam to moving blade at entrance

C_{r_0} = Relative velocity of steam to moving blade at exit

α = Angle with the tangent of the wheel at which the steam with velocity C_1 enters. This is also called **nozzle angle**

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

θ = Entrance angle of moving blade

ϕ = Exit angle of moving blade.

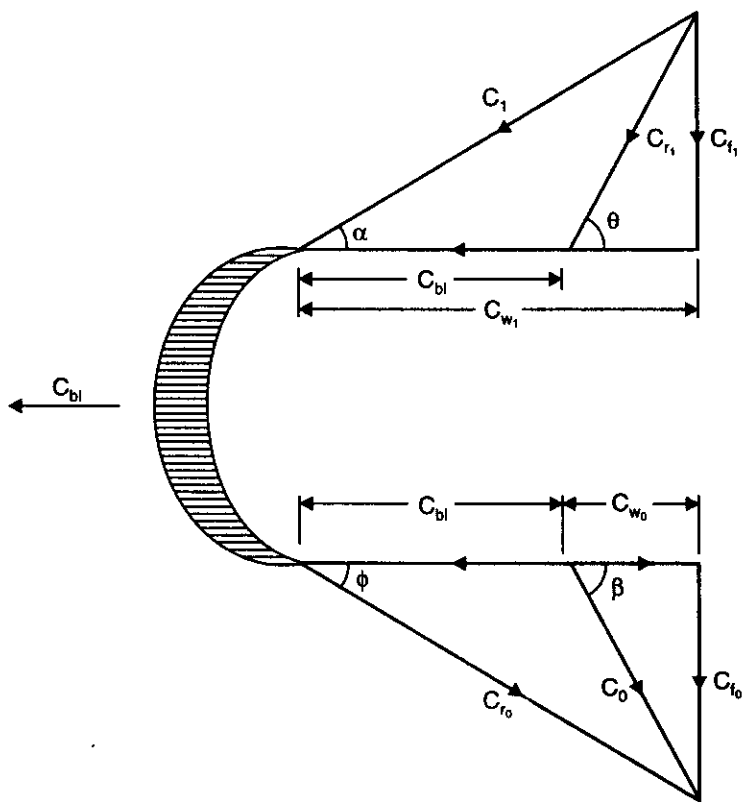

Fig. 3.88. Velocity diagram for moving blade.

The steam jet issuing from the nozzle at a velocity of C_1 impinges on the blade at an angle α. The tangential component of this jet (C_{w_1}) performs work on the blade, the axial component (C_{f_1}) however does no work but causes the steam to flow through the turbine. As the blades move with a tangential velocity of C_{bl}, the entering steam jet has a relative velocity C_{r_1} (with respect to blade) which makes an angle θ with the wheel tangent. The steam then glides over the blade without any shock and discharges at a relative velocity of C_{r_0} at an angle ϕ with the tangent of the blades. The relative velocity at the inlet (C_{r_1}) is the same as the relative velocity at the outlet (C_{r_0}) if *there is no frictional loss at the blade*. The absolute velocity (C_0) of leaving steam make an angle β to the tangent at the wheel.

To have convenience in solving the problems on turbines it is a common practice to combine the two vector velocity diagrams on a common base which represents the blade velocity (C_{bl}) as shown in Fig. 3.89. This diagram has been obtained by superimposing the inlet velocity diagram on the outlet diagram in order that the blade velocity lines C_{bl} coincide.

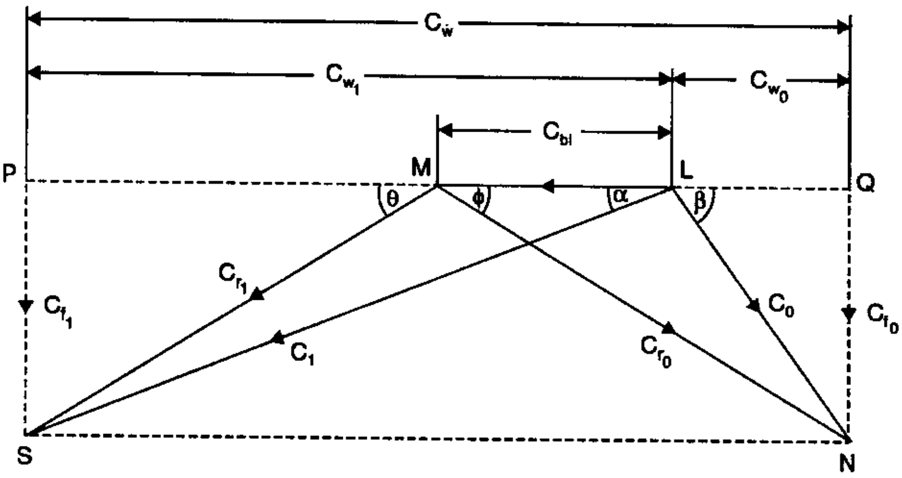

Fig. 3.89

3.19.7.2. Work done on the blade

The work done on the blade may be found out from the change of momentum of the steam jet during its flow over the blade. As earlier discussed, it is only the velocity of whirl which performs work on the blade since it acts in its (blade) direction of motion.

From Newton's second law of motion,

Force (tangential) on the wheel = Mass of steam × acceleration

= Mass of steam/sec. × change of velocity

$$= \dot{m}_s(C_{w_1} - C_{w_0}) \quad \text{...(3.39)}$$

The value of C_{w_0} is actually negative as the steam is discharged in the *opposite direction* to the blade motion, therefore, due consideration should be given to the fact that the *values of C_{w_1} and C_{w_0} are to be added while doing the solution of the problem.*

Work done on blades/sec. = Force × distance travelled/sec.

$$= \dot{m}_s(C_{w_1} + C_{w_0}) \times C_{bl}$$

Power per wheel

$$= \dot{m}_s(C_{w_1} + C_{w_0}) \times C_{bl}$$

$$= \frac{\dot{m}_s C_w C_{bl}}{1000} \text{ kW} \quad \text{...(3.40)}$$

$$(\because \; C_w = C_{w_1} + C_{w_0})$$

$$\textbf{Blade or diagram efficiency} = \frac{\text{Work done on the blade}}{\text{Energy supplied to the blade}}$$

$$= \frac{\dot{m}_s(C_{w_1} + C_{w_0}) \,.\, C_{bl}}{\dot{m}\, C_1^2}$$

$$= \frac{2\, C_{bl}(C_{w_1} + C_{w_0})}{C_1^2} \quad \text{...(3.41)}$$

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

$\therefore$ **Stage efficiency,** $\eta_{stage} = \dfrac{\text{Work done on blade per kg of steam}}{\text{Total energy supplied per kg of steam}}$

$$= \frac{C_{bl}(C_{w_1} + C_{w_0})}{(h_1 - h_2)} \quad ...(3.42)$$

Now, nozzle efficiency $= \dfrac{C_1^2}{2(h_1 - h_2)}$

Also η_{stage} = Blade efficiency × nozzle efficiency

$$= \frac{2C_{bl}(C_{w_1} + C_{w_0})}{C_1^2} \times \frac{C_1^2}{2(h_1 - h_2)} = \frac{C_{bl}(C_{w_1} + C_{w_0})}{(h_1 - h_2)}$$

The ***axial thrust*** on the wheel is due to *difference* between the velocities of flow at entrance and outlet.

Axial force on the wheel = Mass of steam × axial acceleration

$$= \dot{m}_s(C_{f_1} - C_{f_0}) \quad ...(3.43)$$

The *axial force on the wheel must be balanced or must be taken by a thrust bearing.*

Energy converted to heat by blade friction

= Loss of kinetic energy during flow over blades

$$= \dot{m}_s(C_{r_1}^{\,2} - C_{r_0}^{\,2}) \quad ...(3.44)$$

3.19.7.3. Blade velocity coefficient

In an impulse turbine, if friction is neglected the relative velocity will remain unaltered as it passes over blades. In practice the flow of steam over the blades is resisted by friction. The effect of the friction is to reduce the relative velocity of steam as it passes over the blades. In general, there is a loss of 10 to 15 per cent in the relative velocity. Owing to friction in the blades, C_{r_0} is less than C_{r_1} and we may write

$$C_{r_0} = K \,.\, C_{r_1} \quad ...(3.45)$$

where K is termed a ***blade velocity coefficient.***

3.19.7.4. Expression for optimum value of the ratio of blade speed to steam speed (for maximum efficiency) for a single stage impulse turbine

Refer Fig. 3.89.

$$C_w = PQ = MP + MQ = C_{r_1} \cos\theta + C_{r_0} \cos\phi$$

$$= C_{r_1} \cos\theta \left[1 + \frac{C_{r_0} \cos\phi}{C_{r_1} \cos\theta}\right]$$

$$= C_{r_1} \cos\theta\,(1 + K \,.\, Z) \text{ where } Z = \frac{\cos\phi}{\cos\theta} \quad ...(i)$$

Generally, *the angles* θ *and* ϕ *are nearly equal for impulse turbine* and hence it can safely be assumed that Z is a constant.

But, $C_{r_1} \cos\theta = MP = LP - LM = C_1 \cos\alpha - C_{bl}$

From equation (*i*), $C_w = (C_1 \cos\alpha - C_{bl})(1 + K \,.\, Z)$

We know that, Blade efficiency, $\eta_{bl} = \dfrac{2C_{bl} \,.\, C_w}{C_1^2}$ ...(*ii*)

$$\eta_{bl} = \frac{2C_{bl}(C_1 \cos\alpha - C_{bl})(1+KZ)}{C_1^2}$$
$$= 2(\rho \cos \alpha - \rho^2)(1 + KZ)$$
$$= 2\rho(\cos \alpha - \rho)(1 + KZ) \quad ...(iii)$$

where $\rho = \frac{C_{bl}}{C_1}$ is the ratio of *blade speed to steam speed* and is commonly called as ***"Blade speed ratio"***.

For particular impulse turbine α, K and Z may assumed to be constant and from equation (*iii*) it can be seen clearly that η_{bl} depends on the value of ρ only. Hence differentiating (*iii*),

$$\frac{d\eta_{bl}}{d\rho} = 2(\cos \alpha - 2\rho)(1 + KZ)$$

For a maximum or minimum value of η_{bl} this should be zero

$$\cos \alpha - 2\rho = 0, \quad \therefore \quad \rho = \frac{\cos\alpha}{2}$$

Now,
$$\frac{d^2\eta_{bl}}{d\rho^2} = 2(-2)(1 + KZ) = -4(1 + KZ)$$

which is a negative quantity and thus the value so obtained is the maximum.

Optimum value of ratio of blade speed to steam speed is

$$\rho_{opt} = \frac{\cos\alpha}{2} \quad ...(3.46)$$

Substituting this value of ρ in eqn. (*iii*), we get

$$(\eta_{bl})_{max} = 2 \times \frac{\cos\alpha}{2}\left(\cos\alpha - \frac{\cos\alpha}{2}\right)(1 + K \,.\, Z)$$
$$= \frac{\cos^2\alpha}{2}(1 + KZ) \quad ...(3.47)$$

It is sufficiently accurate to assume symmetrical blades ($\theta = \phi$) and no friction in fluid passage for the purpose of analysis.

$$\therefore \quad Z = 1 \text{ and } K = 1$$
$$\therefore \quad (\eta_{bl})_{max} = \cos^2 \alpha \quad ...(3.48)$$

The work done per kg of steam is given by

$$W = (C_{w_1} + C_{w_0})\, C_{bl}$$

Substituting the value of $C_{w_1} + C_{w_0}$ $(= C_w)$

$$W = (C_1 \cos \alpha - C_{bl})(1 + KZ)C_{bl} = 2C_{bl}(C_1 \cos \alpha - C_{bl}) \text{ when } K = 1 \text{ and } Z = 1$$

The maximum value of W can be obtained by substituting the value of cos α from equation (3.46),

$$\cos \alpha = 2\rho = 2\,\frac{C_{bl}}{C_1}$$
$$\therefore \quad W_{max} = 2C_{bl}(2C_{bl} - C_{bl}) = 2C_{bl}^2 \quad ...(3.49)$$

It is obvious from the equation (3.46) that the blade velocity should be *approximately* half of absolute velocity of steam jet coming out from the nozzle (fixed blade) for *the maximum work* *developed per kg of steam or for maximum efficiency. For the other values of blade speed the absolute velocity at outlet from the blade will increase, consequently, more energy will be carried away by the steam and efficiency will decrease.*

For equiangular blades with no friction losses, optimum value of $\frac{C_{bl}}{C_1}$ corresponds to the case, when the outlet absolute velocity is axial as shown in Fig. 3.90.

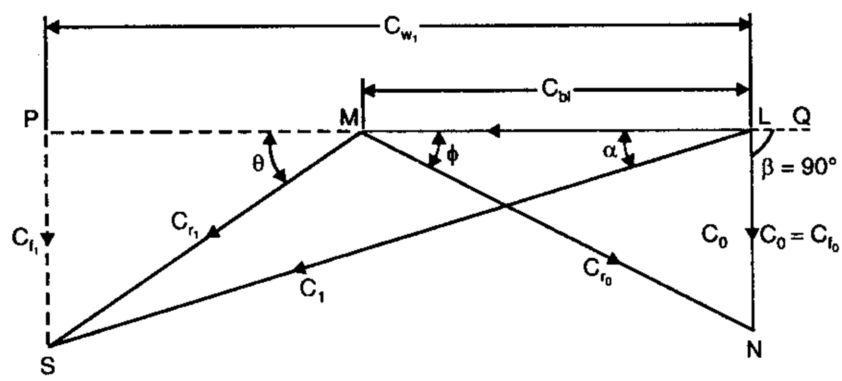

Fig. 3.90

Since the discharge is axial, $\beta = 90°$, $\therefore\ C_0 = C_{f_0}$ and $C_{w_0} = 0$.

The variations of η_{bl} or work developed per kg of steam with $\frac{C_{bl}}{C_1}$ is shown in Fig. 3.91. This figure shows that :

(*i*) When $\frac{C_{bl}}{C_1} = 0$, the work done becomes zero as the distance travelled by the blade (C_{bl}) is zero, even through the torque on the blade is maximum.

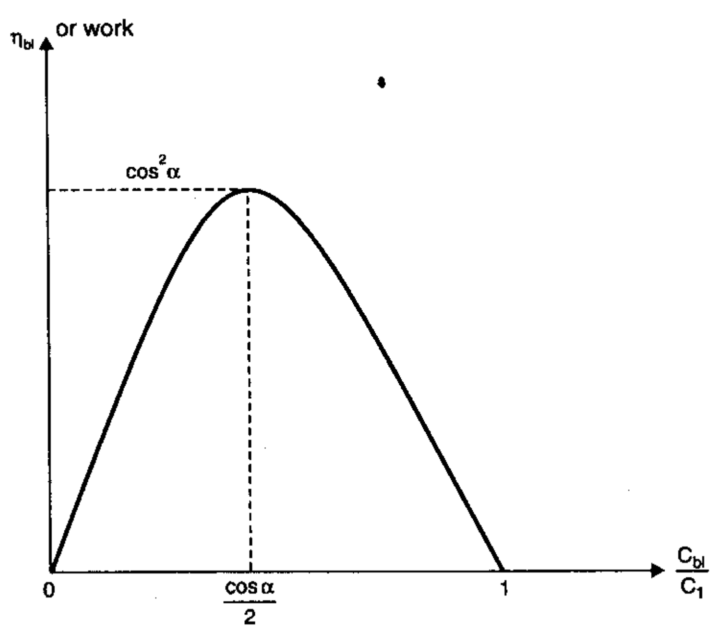

Fig. 3.91

(*ii*) The maximum efficiency is $\cos^2 \alpha$ and maximum work done per kg of steam is $2C^2_{bl}$ when $\frac{C_{bl}}{C_1} = \cos \alpha/2$.

(*iii*) When $\frac{C_{bl}}{C_1} = 1$, the work done is zero as the torque acting on the blade becomes zero even though the distance travelled by the blade is maximum.

When the high pressure steam is expanded from the boiler pressure to condenser pressure in a single stage of nozzle, the absolute velocity of steam becomes maximum and blade velocity also becomes tremendously high. In such a case, *a velocity compounded stage is used to give lower blade speed ratio* and *better utilization of the kinetic energy of the steam*. The arrangement of velocity compounding has already been dealt with.

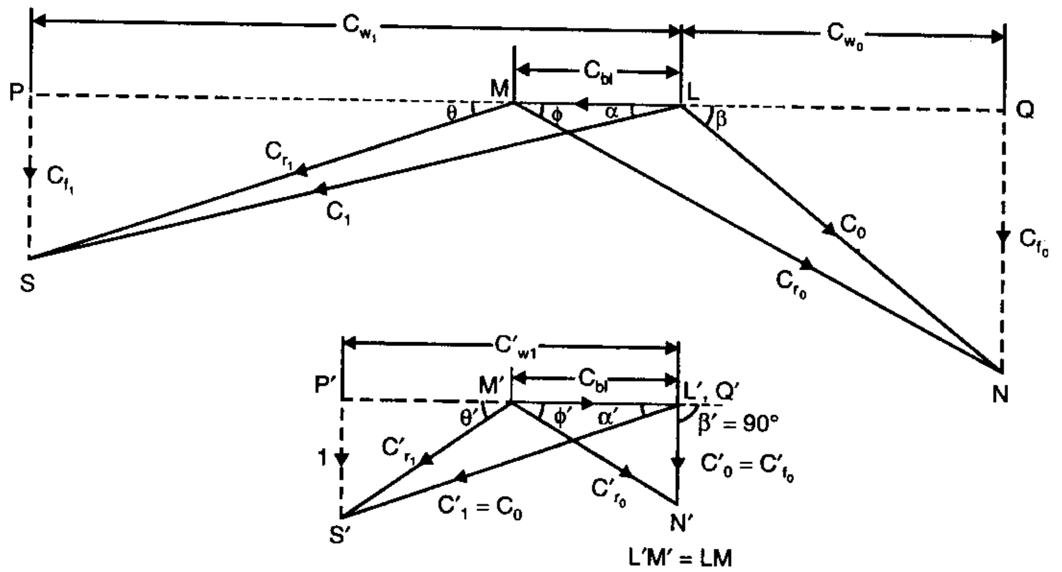

Fig. 3.92

Fig. 3.92 shows the velocity diagrams for the first and second row of moving blades of *velocity compounded unit*. The speed and angles are such that the final absolute velocity of the steam leaving the second row is *axial*. With this arrangement, *the K.E. carried by the steam is minimum*, therefore, *efficiency becomes maximum*.

The velocity of blades (C_{bl}) is same for both the rows since they are mounted on the same shaft.

Consider **first row of moving blades :**

Work done per kg of steam, $W_1 = C_{bl}\,(C_{w_1} + C_{w_0})$

$$= C_{bl}[C_{r_1} \cos \theta + C_{r_0} \cos \phi]$$

If there is *no friction loss and symmetrical blading* is used, then

$$C_{r_1} = C_{r_0} \quad \text{and} \quad \theta = \phi$$

$$\therefore \quad W_1 = C_{bl} \times 2C_{r_1} \cos \theta = 2C_{bl}(C_1 \cos \alpha - C_{bl}) \qquad \text{...(3.50)}$$

The magnitude of absolute velocity of steam leaving the first row and entering into the second row of moving blades is same and *its direction only is changed.*

$\therefore \qquad C_1' = C_0$

Consider **second row of moving blades :**

Work done per kg, $\quad W_2 = C_{bl} \cdot C'_{w_1}$ as $C'_{w_0} = 0$ because discharge is *axial* and $\beta' = 90°$

Alternately, $\quad W_2' = C_{bl}\,[C'_{r_1} \cos\theta' + C'_{r_0} \cos\phi']$

For symmetrical blades $\quad \theta' = \phi'$

and if there is *no friction loss*, then $C'_{r_1} = C'_{r_0}$

$$\therefore \qquad W_2 = 2C_{bl}\, C'_{r_1} \cos\theta'$$

$$= 2C_{bl}\,(C_1' \cos\alpha' - C_{bl}) \qquad \text{...(3.51)}$$

Now α' may be equal to β.

$$\therefore \qquad C_1' \cos\alpha' = C_0 \cos\beta = C_{r_0} \cos\phi - C_{bl}$$

$$= C_{r_1} \cos\theta - C_{bl} = (C_1 \cos\alpha - C_{bl}) - C_{bl}$$

$$= C_1 \cos\alpha - 2C_{bl}$$

Substituting the value of $C_1' \cos\alpha'$ in eqn. (3.51), we get

$$W_2 = 2C_{bl}\,[(C_1 \cos\alpha - 2C_{bl}) - C_{bl}]$$

$$= 2C_{bl}\,(C_1 \cos\alpha - 3C_{bl}) \qquad \text{...(3.52)}$$

Total work done per kg of steam passing through both stages is given by

$$W_t = W_1 + W_2$$

$$= 2C_{bl}\,[C_1 \cos\alpha - C_{bl}] + 2C_{bl}\,[C_1 \cos\alpha - 3C_{bl}]$$

$$= 2C_{bl}\,[2C_1 \cos\alpha - 4C_{bl}]$$

$$= 4C_{bl}\,(C_1 \cos\alpha - 2C_{bl}) \qquad \text{...(3.53)}$$

The blade efficiency for two stage impulse turbine is given by

$$\eta_{bl} = \frac{W_t}{\frac{C_1^2}{2}} = 4C_{bl}\,[C_1 \cos\alpha_1 - 2C_{bl}] \times \frac{2}{C_1^2}$$

$$= \frac{8C_{bl}}{C_1^2}\,(C_1 \cos\alpha - 2C_{bl}) = 8\,\frac{C_{bl}}{C_1}\left(\cos\alpha - 2\,.\frac{C_{bl}}{C_1}\right)$$

$$= 8\rho\,(\cos\alpha - 2\rho) \qquad \text{...(3.54)}$$

where ρ (velocity ratio) $= \dfrac{C_{bl}}{C_1}$.

The blade efficiency for two stage turbine will be maximum when $\dfrac{d\eta_{bl}}{d\rho} = 0$

$$\therefore \qquad \frac{d}{d\rho}\,[8\rho \cos\alpha - 16\rho^2] = 0$$

$$\therefore \qquad 8 \cos\alpha - 32\rho = 0$$

From which, $\rho = \dfrac{\cos\alpha}{4}$ ...(3.55)

Substituting this value in eqn. (3.54), we get

$$(\eta_{bl})_{\max} = 8\,.\,\frac{\cos\alpha}{4}\left[\cos\alpha - 2\,\frac{\cos\alpha}{4}\right] = \cos^2\alpha \qquad \text{...(3.56)}$$

The maximum work done per kg of steam is obtained by substituting the value of

$$\rho = \frac{C_{bl}}{C_1} = \frac{\cos\alpha}{4}$$

or

$$C_1 = \frac{4C_{bl}}{\cos\alpha} \text{ in the eqn. (3.53).}$$

$$\therefore \quad (W_b)_{max} = 4C_{bl}\left(\frac{4C_{bl}}{\cos\alpha}.\cos\alpha - 2C_{bl}\right)$$

$$= 8C_{bl}^2 \quad \text{...(3.57)}$$

The present analysis is done for *two stages* only. The similar procedure is adopted for analysing the problem with three or four stages.

In general, optimum blade speed ratio for maximum blade efficiency or maximum work done is given by

$$\rho = \frac{\cos\alpha}{2.n} \quad \text{...(3.58)}$$

and work done in the last row $= \frac{1}{2^n}$ of total work ...(3.59)

where n is the number of moving/rotating blade rows in series.

As the number of rows increases, the utility of last row decreases. In practice, *more than two rows are hardly preferred.*

3.19.7.5. Advantages of velocity compounded impulse turbine

1. Owing to relatively large heat drop, a velocity-compounded impulse turbine requires a comparatively small number of stages.

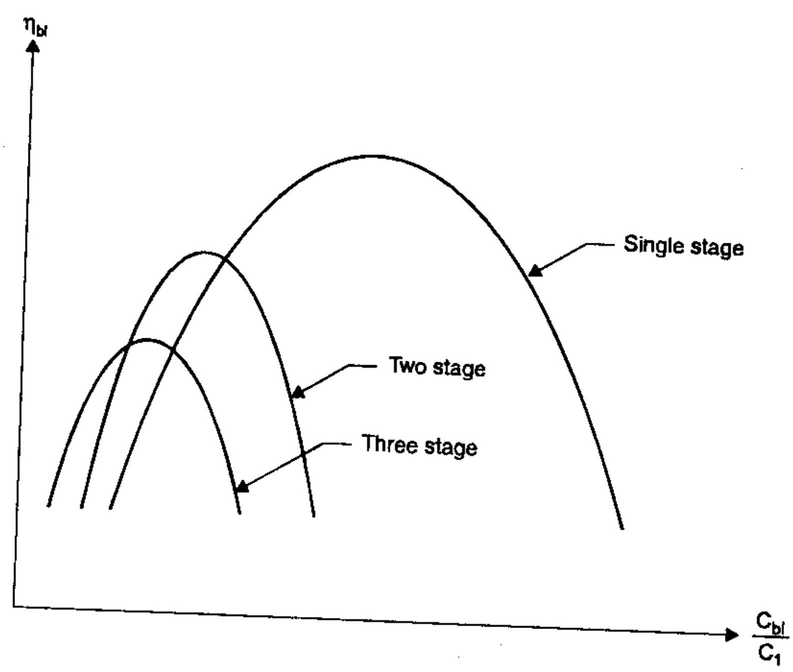

Fig. 3.93

2. Due to number of stages being small, its cost is less.
3. As the number of moving blades rows in a wheel increases, the maximum stage efficiency and optimum value of ρ decreases.
4. Since the steam temperature is sufficiently low in a two or three row wheel, therefore, cast iron cylinder may used. This will cause saving in material cost.

Disadvantages of velocity-compounded impulse turbine :

1. It has high steam consumption and low efficiency (Fig. 3.93).
2. In a single row wheel, the steam temperature is high so cast iron cylinder cannot be used due to phenomenon of growth ; cast steel cylinder is used which is costlier than cost iron.

3.19.8. Reaction Turbines

The reaction turbines which are used these days are really **impulse-reaction** turbines. Pure reaction turbines are not in general use. The expansion of steam and heat drop occur both in fixed and moving blades.

3.19.8.1. Velocity diagram for reaction turbine blade

Fig. 3.94 shows the velocity diagram for reaction turbine blade. In case of an impulse turbine blade the relative velocity of steam either remains constant as the steam glides over the blades or is reduced slightly due to friction. In reaction turbine blades, the steam continuously expands at it flows over the blades. *The effect of the continuous expansion of steam during the flow over the blade is to increase the relative velocity of steam.*

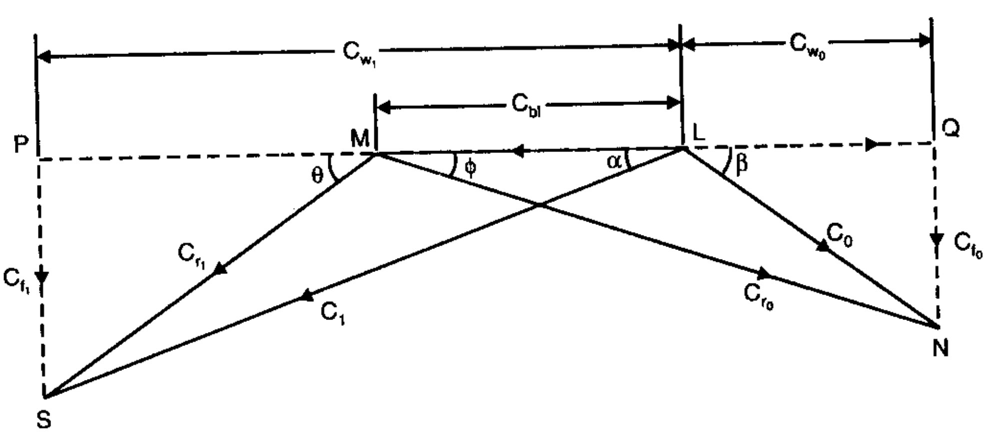

Fig. 3.94. Velocity diagram for reaction turbine blade.

∴ $C_{r_0} > C_{r_1}$ for *reaction turbine blade.*

$(C_{r_0} \leq C_{r_1}$ for *impulse turbine blade).*

3.19.8.2. Degree of reaction (R_d)

The *"degree of reaction"* of *reaction turbine stage is defined* as the *ratio of heat drop over moving blades to the total heat drop in the stage.*

Thus the *degree of reaction of reaction turbine* is given by,

$$R_d = \frac{\text{Heat drop in moving blades}}{\text{Heat drop in the stage}} = \frac{\Delta h_m}{\Delta h_f + \Delta h_m} \text{ as shown in Fig. 3.95.}$$

The heat drop in moving blades is equal to increase in relative velocity of steam passing through the blade.

$$\therefore \qquad \Delta h_m = \frac{C_{r_0}^{\ 2} - C_{r_1}^{\ 2}}{2}$$

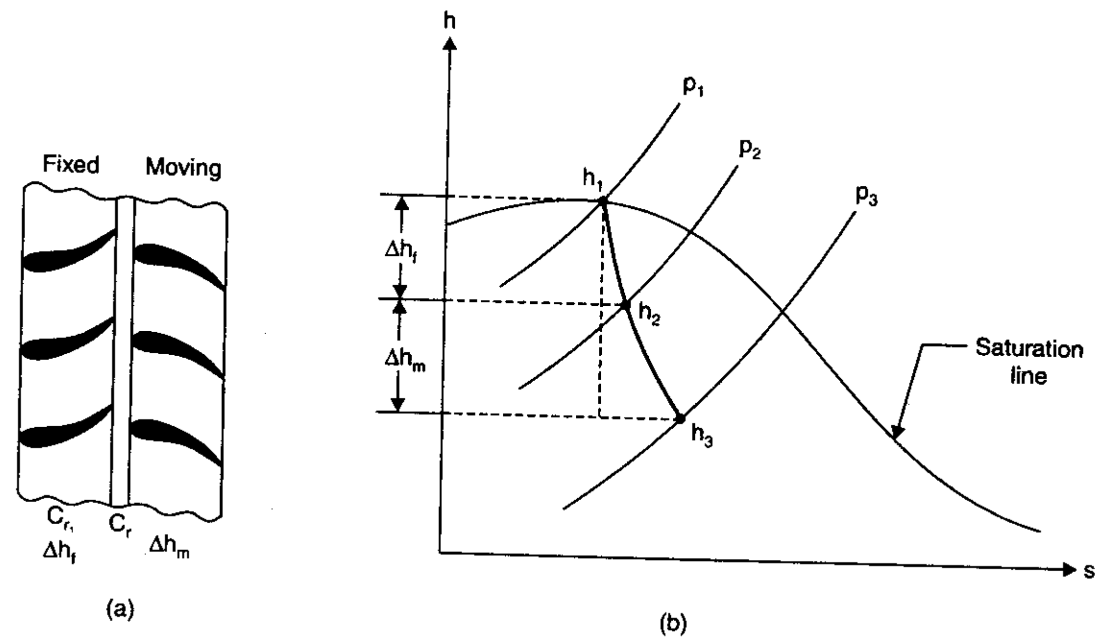

Fig. 3.95

The total heat drop in the stage $(\Delta h_f + \Delta h_m)$ is equal to the work done by the steam in the stage and it is given by

$$\Delta h_f + \Delta h_m = C_{bl}\ (C_{w_1} + C_{w_0})$$

$$\therefore \qquad R_d = \frac{C_{r_0}^{\ 2} - C_{r_1}^{\ 2}}{2C_{bl}(C_{w_1} + C_{w_0})} \qquad ...(3.60)$$

Referring to Fig. 3.94,

$$C_{r_0} = C_{f_0}\ \text{cosec}\ \phi \text{ and } C_{r_1} = C_{f_1}\ \text{cosec}\ \theta$$

and

$$(C_{w_1} + C_{w_0}) = C_{f_1}\ \cot\theta + C_{f_0}\ \cot\phi$$

The velocity of flow generally remains constant through the blades.

$$\therefore \qquad C_{f_1} = C_{f_0} = C_f\,.$$

Substituting the values of C_{r_1}, C_{r_0} and $(C_{w_1} + C_{w_0})$ in eqn. (3.60), we get

$$R_d = \frac{C_f^{\,2}\ (\text{cosec}^2\phi - \text{cosec}^2\theta)}{2\,C_{bl}\ C_f\ (\cot\theta + \cot\phi)} = \frac{C_f}{2C_{bl}}\left[\frac{(\cot^2\phi + 1) - (\cot^2\theta + 1)}{\cot\theta + \cot\phi}\right]$$

$$= \frac{C_f}{2C_{bl}}\left[\frac{\cot^2\phi - \cot^2\theta}{\cot\phi + \cot\theta}\right]$$

$$= \frac{C_f}{2C_{bl}} (\cot \phi - \cot \theta) \qquad ...(3.61)$$

If the turbine is designed for 50% reaction ($\Delta h_f = \Delta h_m$), then the equation (3.61) can be written as

$$\frac{1}{2} = \frac{C_f}{2C_{bl}} (\cot \phi - \cot \theta)$$

$$\therefore \quad C_{bl} = C_f (\cot \phi - \cot \theta) \qquad ...(3.62)$$

Also C_{bl} can be written as

$$C_{bl} = C_f (\cot \phi - \cot \beta) \qquad ...(3.63)$$

and

$$C_{bl} = C_f (\cot \alpha - \cot \theta) \qquad ...(3.64)$$

$C_{f_1} = C_{f_0} = C_f$ is assumed in writing the above equations.

Comparing the equations (3.62), (3.63), (3.64)

$$\theta = \beta \text{ and } \phi = \alpha$$

which *means that moving blade and fixed blade* must have the same shape if the degree of reaction is 50%. This condition gives symmetrical velocity diagrams. This type of turbine is known as **"Parson's reaction turbine"**. Velocity diagram for the blades of this turbine is given in Fig. 3.96.

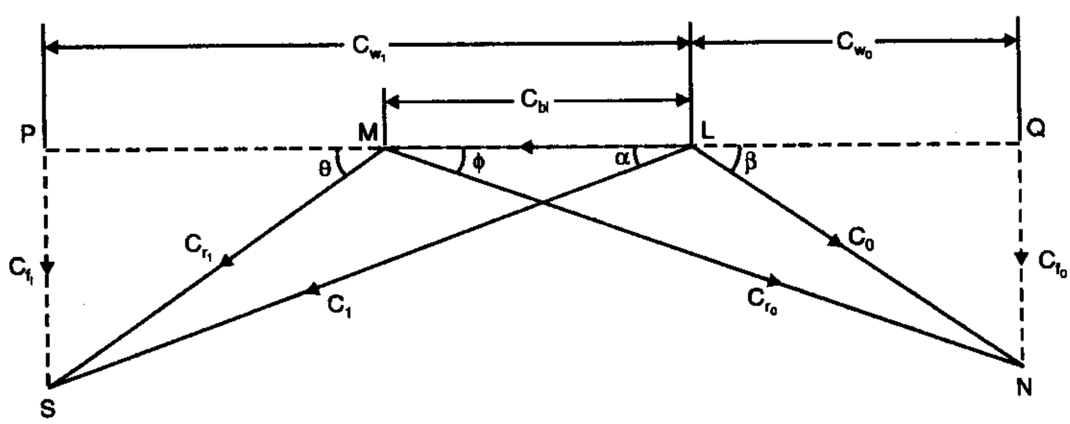

Fig. 3.96

3.19.8.3. Condition for maximum efficiency

The condition for *maximum efficiency* is derived by making the following *assumptions* :

(*i*) The degree of reaction is 50%.

(*ii*) The moving and fixed blades are symmetrical.

(*iii*) The velocity of steam at exit from the preceeding stage is same as velocity of steam at the entrance to the succeeding stage.

Refer Fig. 3.96 (velocity diagram for reaction blade).

Work done per kg of steam,

$$W = C_{bl} (C_{w_1} + C_{w_0}) = C_{bl}[C_1 \cos \alpha + (C_{r_0} \cos \phi - C_{bl})]$$

as $\phi = \alpha$ and $C_{r_0} = C_{r_1}$ as per the assumptions.

$$\therefore \quad W = C_{bl} [2C_1 \cos \alpha - C_{bl}]$$

or
$$W = C_1^2\left[\frac{2C_{bl}\,C_1\cos\alpha}{C_1^2} - \frac{C_{bl}^2}{C_1^2}\right]$$
$$= C_1^2\ [2\rho\ .\ \cos\alpha - \rho^2] \qquad ...(3.65)$$

where $\rho = \dfrac{C_{bl}}{C_1}$.

The K.E. supplied to the fixed blade $= \dfrac{C_1^2}{2g}$.

The K.E. supplied to the moving blade $= \dfrac{C_{r_0}^2 - C_{r_1}^2}{2}$.

$\therefore$ Total energy supplied to the stage,
$$\Delta h = \frac{C_1^2}{2} + \frac{C_{r_0}^2 - C_{r_1}^2}{2}$$

as $\quad C_{r_0} = C_1$ for symmetrical triangles.

$$\therefore \qquad \Delta h = \frac{C_1^2}{2} + \frac{C_1^2 - C_{r_1}^2}{2}$$
$$= C_1^2 - \frac{C_{r_1}^2}{2} \qquad ...(3.66)$$

Considering the ΔLMS (Fig. 3.94)
$$C_{r_1}^2 = C_1^2 + C_{bl}^2 - 2C_1\ .\ C_{bl}\ .\ \cos\alpha$$

Substituting this value of $C_{r_1}^2$ is equation (3.66).

Total energy supplied to the stage
$$\Delta h = C_1^2 - (C_1^2 + C_{bl}^2 - 2C_1\ .\ C_{bl}\ .\ \cos\alpha)/2$$
$$= (C_1^2 + 2C_1\,C_{bl}\cos\alpha - C_{bl}^2)/2$$
$$= \frac{C_1^2}{2}\left[1 + \frac{2C_{bl}}{C_1}\ .\ \cos\alpha - \left(\frac{C_{bl}}{C_1}\right)^2\right]$$
$$= \frac{C_1^2}{2}[1 + 2\rho\cos\alpha - \rho^2] \qquad ...(3.67)$$

The blade efficiency of the reaction turbine is given by,
$$\eta_{bl} = \frac{W}{\Delta h}$$

Substituting the value of W and Δh from equations (3.65) and (3.67),
$$\eta_{bl} = \frac{C_1^2\,[2\rho\cos\alpha - \rho^2]}{\dfrac{C_1^2}{2}(1 + 2\rho\cos\alpha - \rho^2)}$$
$$= \frac{2(2\rho\cos\alpha - \rho^2)}{(1 + 2\rho\cos\alpha - \rho^2)} = \frac{2\rho(2\cos\alpha - \rho)}{(1 + 2\rho\cos\alpha - \rho^2)} = \frac{2(1 + 2\rho\cos\alpha - \rho^2) - 2}{(1 + 2\rho\cos\alpha - \rho^2)}$$

$$= 2 - \frac{2}{1 + 2\rho \cos\alpha - \rho^2} \qquad ...(3.68)$$

The η_{bl} becomes maximum when the value of $(1 + 2\rho \cos \alpha - \rho^2)$ becomes maximum.

$\therefore$ The required equation is

$$\frac{d}{d\rho}(1 + 2\rho \cos \alpha - \rho^2) = 0$$

$$2 \cos \alpha - 2\rho = 0$$

$$\therefore \qquad \rho = \cos \alpha \qquad ...(3.69)$$

Substituting the value of ρ from eqn. (3.69) into the eqn. (3.68), the value of maximum efficiency is given by,

$$(\eta_{bl})_{max.} = 2 - \frac{2}{1 + 2\cos^2\alpha - \cos^2\alpha} = 2\left(1 - \frac{1}{1+\cos^2\alpha}\right) = \frac{2\cos^2\alpha}{1+\cos^2\alpha}$$

Hence $$(\eta_{bl})_{max.} = \frac{2\cos^2\alpha}{1+\cos^2\alpha} \qquad ...(3.70)$$

The variation of η_{bl} with blade speed ratio $\left(\frac{C_{bl}}{C_1}\right)$ for the reaction stage is shown in Fig. 3.97.

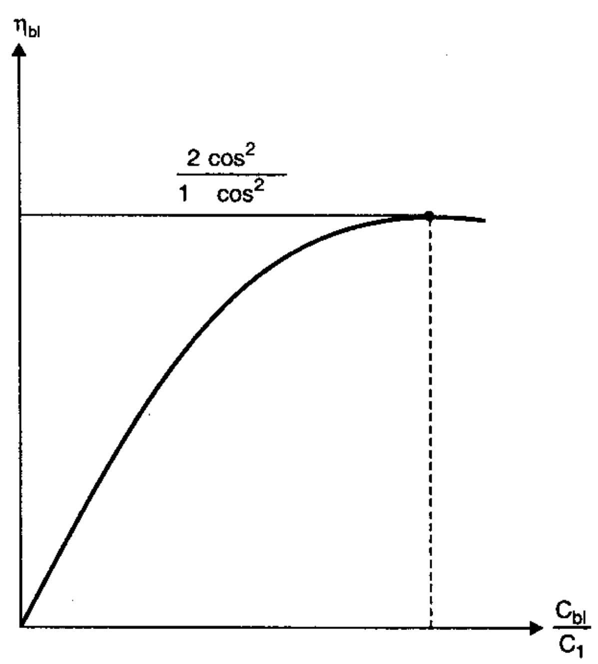

Fig. 3.97

3.19.9. Turbine Efficiencies

1. **Blade or diagram efficiency** (η_{bl}). *It is the ratio of work done on the blade per second to the energy entering the blade per second.*

2. **Stage efficiency** (η_{stage}). The stage efficiency covers all the losses in the nozzles, blades, diaphragms and discs that are associated with that stage.

$$\eta_{stage} = \frac{\text{Net work done on shaft per stage per kg of steam flowing}}{\text{Adiabatic heat drop per stage}}$$

$$= \frac{\text{Net work done on blades – Disc friction and windage}}{\text{Adiabatic heat drop per stage}}.$$

3. **Internal efficiency** ($\eta_{internal}$). This is equivalent to the stage efficiency when applied to the whole turbine, and is defined as :

$$= \frac{\text{Heat coverted into useful work}}{\text{Total adiabatic heat drop}}.$$

4. **Overall or turbine efficiency** ($\eta_{overall}$). This efficiency covers internal and external losses ; for example, bearings and steam friction, leakage, radiation etc.

$$\eta_{overall} = \frac{\text{Work delivered at the turbine coupling in heat units per kg of steam}}{\text{Total adiabatic heat drop}}.$$

5. **Net efficiency or efficiency ratio** (η_{net}). It is the ratio

$$\frac{\text{Brake thermal efficiency}}{\text{Thermal efficiency on the Rankine cycle}}$$

Also the actual thermal efficiency

$$= \frac{\text{Heat converted into useful work per kg of steam}}{\text{Total heat in steam at stop valve – Water heat in exhaust}}.$$

Again, Rankine efficiency

$$= \frac{\text{Adiabatic heat drop}}{\text{Total heat in steam at stop valve – Water heat in exhaust}}.$$

$$\eta_{net} = \frac{\text{Heat converted into useful work}}{\text{Total adiabatic heat drop}}.$$

Hence $\eta_{net} = \eta_{overall}.$

It is the overall or net efficiency that is meant when the efficiency of a turbine is spoken of without qualification.

3.19.10. Types of Power in Steam Turbine Practice

In steam turbine performance the following types of power are generally used :

1. **Adiabatic power (A.P.).** It is the power based on the total internal steam flow and adiabatic heat drop.

2. **Shaft power (S.P.).** It is the actual power transmitted by the turbine.

3. **Rim power (R.P.).** It is the power developed at the rim. It is also called *blade power.*

Power losses are usually expressed as follows :

(*i*) $(P)_D$ = Power lost in overcoming disc friction.

(*ii*) $(P)_{bw}$ = Power lost in blade windage losses.

Let us consider the case, of an impulse turbine. Let $\dot{m}_s$ be the total internal steam flow in kg/s.

Refer Fig. 3.98. The line (1-2) represents the adiabatic or isentropic expansion of steam in the nozzle from pressure p_1 to p_2. But the actual path of the stage point during expansion in nozzles is shown by (1-3) which takes into account the effect of 'nozzles losses'.

Then, $$\text{A.P.} = \dot{m}_s\,(h_1 - h_2)\ \text{kW} \quad \text{...(3.71)}$$

After expansion in the nozzle the steam enters the blades where the R.P. is developed. Due to blade friction the steam is somewhat reheated and this reheating is shown by (3-4) along the constant pressure p_2 line just for convenience. But in actual practice though the pressure at outlet of the blade is equal to that at the inlet, the pressure in the blade channels is not constant. However, with this simplification ;

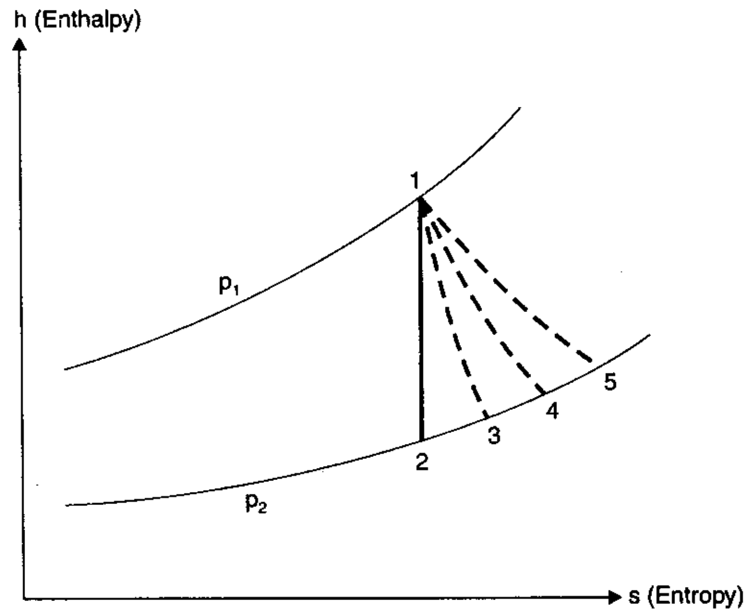

Fig. 3.98

$$\text{R.P.} = \dot{m}_s\,(h_1 - h_4)\ \text{kW} \quad \text{...[3.71 (a)]}$$

4-5 shows the further reheating due to friction and blade windage and these losses are given as

$$(P)_{w.f.} = \dot{m}_s\,(h_5 - h_4)\ \text{kW} \quad \text{...[3.71 (b)]}$$

Now points 1 and 5 are the initial and final stage points respectively for a single stage impulse turbine. It, therefore, follows that

$$\text{S.P.} = \dot{m}_s\,(h_1 - h_5)\ \text{kW.} \quad \text{...[3.71 (c)]}$$

3.19.11. Energy Losses in Steam Turbines

The increase in heat energy required for doing mechanical work in actual practice as compared to the theoretical value, in which the process of expansion takes place strictly according to the adiabatic process, is termed as energy loss in a steam turbine.

The losses which appear in an actual turbine may be divided into two following groups :

1. **Internal losses.** Losses directly connected with the steam conditions while in its flow through the turbine are called *internal losses*. They may be further classified as :

(*i*) Losses in regulating valves.

(*ii*) Losses in nozzles (guide blades).

(iii) Losses in moving blades :

(a) losses due to trailing edge wake ;

(b) impingement losses ;

(c) losses due to leakage of steam through the angular space ;

(d) frictional losses ;

(e) losses due to turning of the steam jet in the blades ;

(f) losses due to shrouding.

(iv) Leaving velocity losses (exit velocity).

(v) Losses due to friction of disc carrying the blades and windage losses.

(vi) Losses due to clearance between the rotor and guide blade discs.

(vii) Losses due to wetness of steam.

(viii) Losses in exhaust piping etc.

2. **External losses.** Losses which do not influence the steam conditions. They may be further classified as :

(i) Mechanical losses.

(ii) Losses due to leakage of steam from the labyrinth gland seals.

3.19.12. Steam Turbine Governing and Control

The objective of governing is to keep the turbine speed fairly constant irrespective of load.

The principal methods of steam turbine governing are as follows :

1. Throttle governing
2. Nozzle governing
3. By-pass governing
4. Combination of 1 and 2 and 1 and 3.

1. Throttle governing :

Throttle governing is the most widely used particularly on *small turbines,* because *its initial cost is less* and *the mechanism is simple.* The object of throttle governing is to *throttle the steam,* whenever, *there is a reduction of load* compared to economic or design load for maintaining speed and *vice versa.*

Fig. 3.99 (a) shows a simple throttle arrangement. To start the turbine for full load running valve A is opened. The operation of double beat valve B is carried out by an oil servomotor which is controlled by a centrifugal governor. As the steam turbine gains speed the valve B closes to throttle the steam and reduces the supply to the nozzle.

For a turbine governed by throttling the relationship between steam consumption and load is given by the well known *Willan's line* as shown in Fig. 3.99 (b). Several tests have shown that when a turbine is governed by throttling, the Willan's line is straight. It is expressed as :

$$m_s = KM + m_{s_0} \qquad ...(3.72)$$

where $\dot{m}_s$ = Steam consumption in kg/h at any load M,

m_{s_0} = Steam consumption in kg/h at no load,

m_{s_1} = Steam consumption in kg/h at full load,

M = Any other load in kW,

M_1 = full load in kW, and
K = constant.

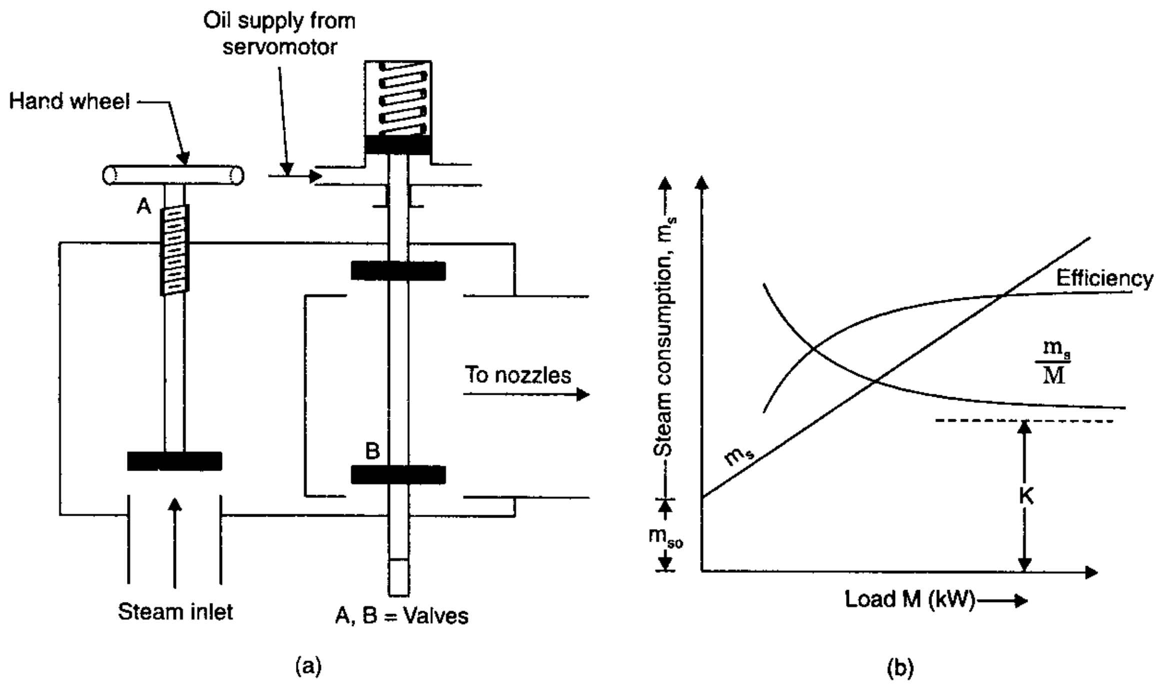

Fig. 3.99. Throttle governing.

m_{s_0} varies from about 0.1 to 0.14 times the full load consumption. The eqns. (3.72) can also be written as :

$$\frac{m_s}{M} = K + \frac{m_{s_0}}{M}$$

where $\frac{m_s}{M}$ is called the steam consumption per kWh.

2. **Nozzle governing**

The efficiency of a steam turbine is considerably reduced if throttle governing is carried out at low loads. An alternative, and more efficient form of governing is by means of nozzle control. Fig. 3.100 shows a diagrammatic arrangement of typical nozzle control governing. In this method of governing, the nozzles are grouped together 3 to 5 or more groups and supply of steam to each group is controlled by regulating valves. Under full load conditions the valves remain fully open.

When the load on the turbine becomes more or less than the design value, the supply of steam to a group of nozzles may be varied accordingly so as to restore the original speed.

Nozzle control can only be applied to the first stage of a turbine. It is suitable for *simple impulse turbine* and *larger units which have an impulse stage followed* by an *impulse-reaction turbine*. In pressure compounded impulse turbines, there will be some drop in pressure at entry to second stage when some of the first stage nozzles are cut out.

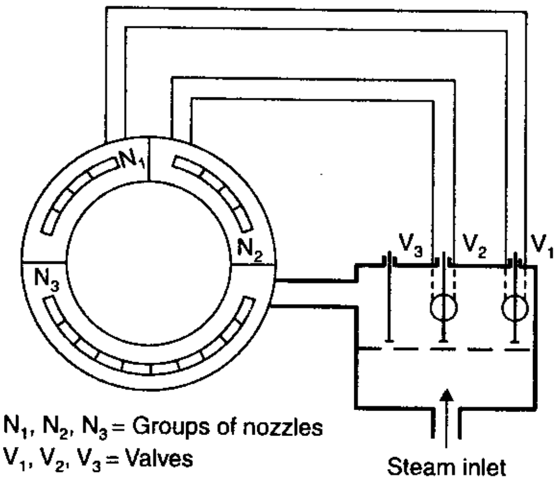

Fig. 3.100. Nozzle governing.

Comparison of Throttle and Nozzle Control Governing

S. No.	Particulars	Throttle control	Nozzle control
1.	*Throttling losses*	Severe	No throttling losses (Actually there are a little throttling losses in nozzles valves which are partially open).
2.	*Partial admission losses*	Low	High
3.	*Heat drop available*	Lesser	Larger
4.	*Use*	Used in impulse and reaction turbines both.	Used in impulse turbine and also in reaction turbine (if initial stage is impulse)
5.	*Suitability*	Small turbines	Medium and larger turbines.

3. **By-pass governing :**

The steam turbines which are designed to work at economic load it is desirable to have full admission of steam in the high pressure stages. At the maximum load, which is greater than the economic load, the additional steam required could not pass through the first stage since additional nozzles are not available. By-pass regulation allows for this in a turbine which is throttle governed, by means of a second by-pass valve in the first stage nozzle (Fig. 3.101). This valve opens when throttle valve has opened a definite amount. Steam is by-passed through the second valve to a lower stage in the turbine. When by-pass valve operates it is under the control of the turbine governor. The secondary and tertiary supplies of steam in the lower stages increase the work output in these stages, but there is a *loss in efficiency* and a curving of the Willian's line.

In reaction turbines, because of the pressure drop required in the moving blades, nozzles control governing is not possible, and *throttle governing plus by-pass governing, is used.*

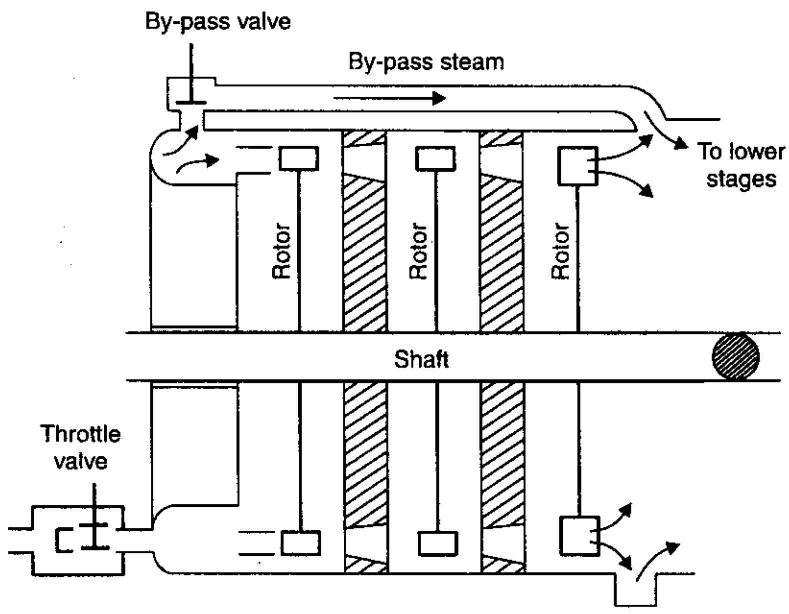

Fig. 3.101. By-pass governing.

3.19.13. Special Forms of Turbines :

In many industries such as chemical, sugar refining, paper making, textile etc. where combined use of power and heating and process work is required it is wasteful to generate steam for power and process purposes separately, because about 70 percent of heat supplied for power purposes will normally be carried away by the cooling water. On the other hand, if the engine or turbine is operated with a normal exhaust pressure then the temperature of the exhaust steam is too low to be of any use for heating process. It would be possible to generate the required power and still have available for process work a large quantity of heat in the exhaust steam, if suitable modification of the initial steam pressure and exhaust pressure is made. Thus in combined power and process plants following types of steam turbines are used : (1) Back pressure turbines, and (2) Steam extraction or pass-out turbines.

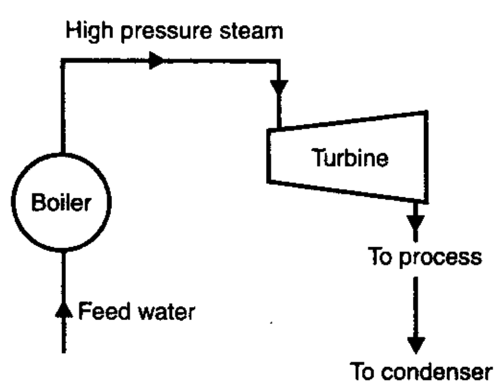

Fig. 3.102. Back pressure turbine.

1. ***Back pressure turbine :***

Refer Fig. 3.102. In this type of turbine steam at boiler pressure enters the turbine and is exhausted into a pipe. This pipe leads to process plant or other turbine. The back pressure turbine may be used in cases where the power generated (by expanding steam) from economical initial pressure down to the heating pressure is equal to, or greater than, the power requirements. The steam exhausted from the turbine is usually superheated and in most cases it is not suitable for process work due to the following *reasons* :

(*i*) It is impossible to control its temperature, and

(*ii*) Rate of the heat transfer from superheated steam to the heating surface is lower than that of saturated steam. Consequently a desuperheater is invariably used. To enhance the power capacity of the existing installation, a high pressure boiler and a back-pressure turbine are added to it. This added high pressure boiler supplies steam to the back pressure turbine which exhaust into the old low pressure turbine.

2. ***Extraction pass out turbine :***

Refer Fig. 3.103. It is found that in several cases the power available from a back pressure turbine (through which the whole of the steam flows) is appreciably less than that required in the factory and this may be due to the following reasons :

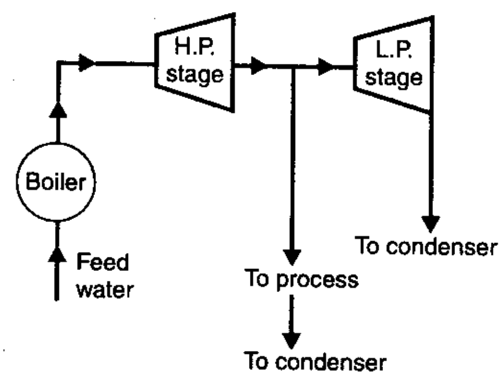

Fig. 3.103. Extraction pass out turbine.

(*i*) Small heating or process requirements ;

(*ii*) A relatively high exhaust pressure ; and

(*iii*) A combination of the both.

In such a case it would be possible to install a back-pressure turbine to provide the heating steam and a condensing tubing to generate extra power, but it is possible, and useful, to combine functions of both machines in a single turbine. Such a machine is called **extraction or pass out turbine** and here at some point intermediate between inlet and exhaust some steam is extracted or passed out for process or heating purposes. In this type of turbine a sensitive governor is used which controls the admission of steam to the high pressure section so that regardless of power or process requirements, constant speed is maintained.

Exhaust or low pressure turbine :

If an uninterrupted supply of low pressure steam is available (such as from reciprocating steam engines exhaust) it is possible to improve the efficiency of the whole plant by fitting an exhaust or low pressure turbine. The use of exhaust turbine is chiefly made where there are several reciprocating steam engines which work intermittently ; and are non-condensing (*e.g.*

rolling mill and colliery engines). The exhaust steam from these engines is expanded in an exhaust turbine and then condensed. In this turbine some form of heat accumulator is needed to collect the more or less irregular supply of low pressure steam from the non-condensing steam engines and deliver it to the turbine at the rate required. In some cases when the supply of low pressure steam falls below the demand, live steam from the boiler, with its pressure and temperature reduced ; is used to make up the deficiency.

The necessary drop in pressure may be obtained by the use of a reducing valve, or for large flows, more economically by expansion through another turbine. The high pressure and low pressure turbines are sometimes combined on a common spindle and because of two supply pressures this combined unit is known as '**mixed pressure turbine**'. (Fig. 3.104).

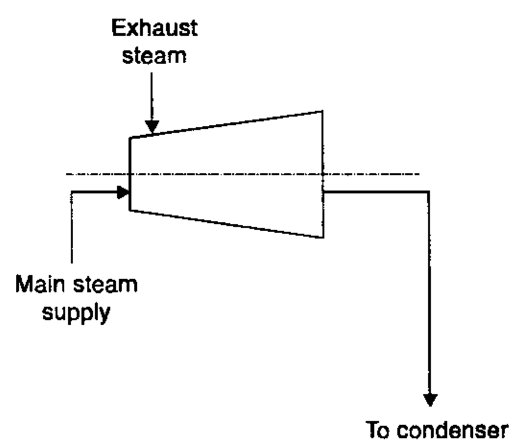

Fig. 3.104. Mixed pressure turbine.

☞**Example 3.13.** *Steam with absolute velocity of 300 m/s is supplied through a nozzle to a single stage impulse turbine. The nozzle angle is 25°. The mean diameter of the blade rotor is 1 metre and it has a speed of 2000 r.p.m. Find suitable blade angles for zero axial thrust. If the blade velocity co-efficient is 0.9 and the steam flow rate is 10 kg/s, calculate the power developed.*

Solution. Absolute velocity of steam entering the blade, $C_1 = 300$ m/s

Nozzle angle, $\alpha = 25°$

Mean diameter of the rotor blade, $D = 1$ m

Speed of the rotor, $N = 2000$ r.p.m.

Blade velocity co-efficient, $K = 0.9$

Steam flow rate, $\dot{m}_s = 10$ kg/s

Power developed :

Blade speed, $$C_{bl} = \frac{\pi DN}{60} = \frac{\pi \times 1 \times 2000}{60} = 105 \text{ m/s.}$$

— With the above data (*i.e.*, $C_1 = 300$ m/s, $C_{bl} = 105$ m/s and $\alpha = 25°$) draw triangle *LMS* (Fig. 3.105). From *S* draw perpendicular *SP* on *LM* produced. Measure C_{r_1}.

— From *S* draw a line parallel to *LP* ($\because\ C_{f_1} = C_{f_0}$) and from point *M* draw an arc equal to $C_{r_0} (= 0.9\, C_{r_1})$ to get the point of intersection *N*. Complete the triangle *LMN*. From *N* draw perpendicular *NQ* on *PL* produced to get C_{f_0}.

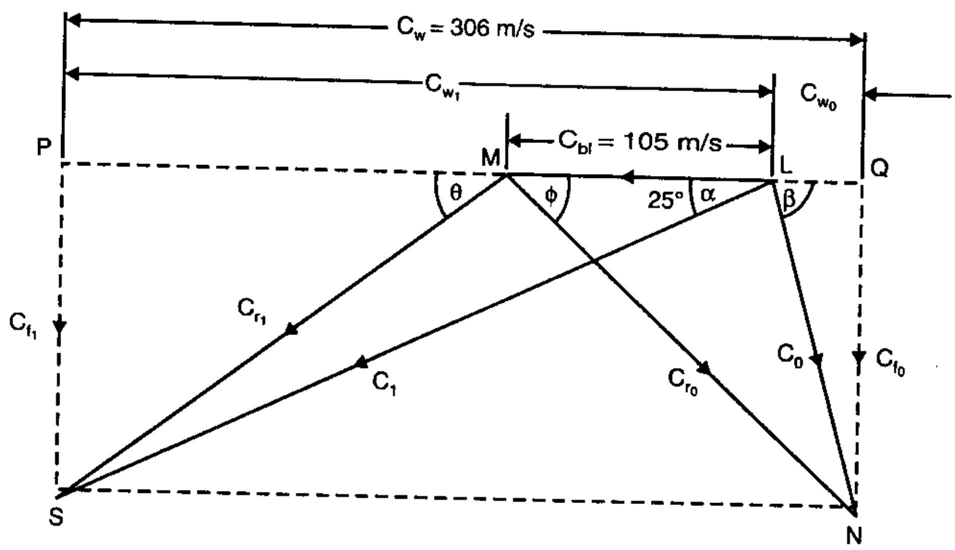

Fig. 3.105

Measure θ and φ (the blade angles) from the velocity diagram.

(*i*) θ = **37°** and ϕ = **42°**. **(Ans.)**

(*ii*) *Power developed,*

$$P = \frac{\dot{m}_s(C_{w_1} + C_{w_0}) \times C_{bl}}{1000} = \frac{10 \times 306 \times 105}{1000} = \mathbf{321.3\ kW.} \quad \textbf{(Ans.)}$$

☞**Example 3.14.** *In an impulse turbine (with a single row wheel) the mean diameter of the blades is 1.05 m and the speed is 3000 r.p.m. The nozzle angle is 18°, the ratio of blade speed to steam speed is 0.42 and the ratio of the relative velocity at outlet from the blades to that at inlet is 0.84. The outlet angle of the blade is to be made 3° less than the inlet angle. The steam flow is 10 kg/s. Draw the velocity diagram for the blades and derive the following :*

(*i*) *Tangential thrust on the blades* (*ii*) *Axial thrust on the blades*
(*iii*) *Resultant thrust on the blades* (*iv*) *Power developed in the blades*
(*v*) *Blading efficiency.*

Solution. Mean diameter of the blades, D = 1.05 m

Speed of the turbine, N = 3000 r.p.m.

Nozzle angle, α = 18°

Ratio of blade speed to steam speed, ρ = 0.42

Ratio, $\frac{C_{r_1}}{C_{r_0}} = 0.84$

Outlet blade angle, $\phi = \theta - 3°$

Steam flow rate $\dot{m}_s$ = 10 kg/s

Blade speed, $C_{bl} = \frac{\pi DN}{60} = \frac{\pi \times 1.05 \times 3000}{60} = 164.5$ m/s

But $\rho = \dfrac{C_{bl}}{C_1} = 0.42$ (given)

$\therefore$ $C_1 = \dfrac{C_{bl}}{0.42} = \dfrac{164.5}{0.42} = 392$ m/s

With the data, $C_1 = 392$ m/s ;

$\alpha = 18°$, complete ΔLMS

$\theta = 30°$ (on measurement)

$\therefore$ $\phi = 30° - 3 = 27°$

Now complete the ΔLMN by taking $\phi = 27°$ and $C_{r_0} = 0.84\ C_{r_1}$.

Finally complete the whole diagram as shown in Fig. 3.106.

(*i*) **Tangential thrust on the blades**

$$= \dot{m}_s(C_{w_1} + C_{w_0}) = 10 \times 390 = \mathbf{3900\ N.}\quad \textbf{(Ans.)}$$

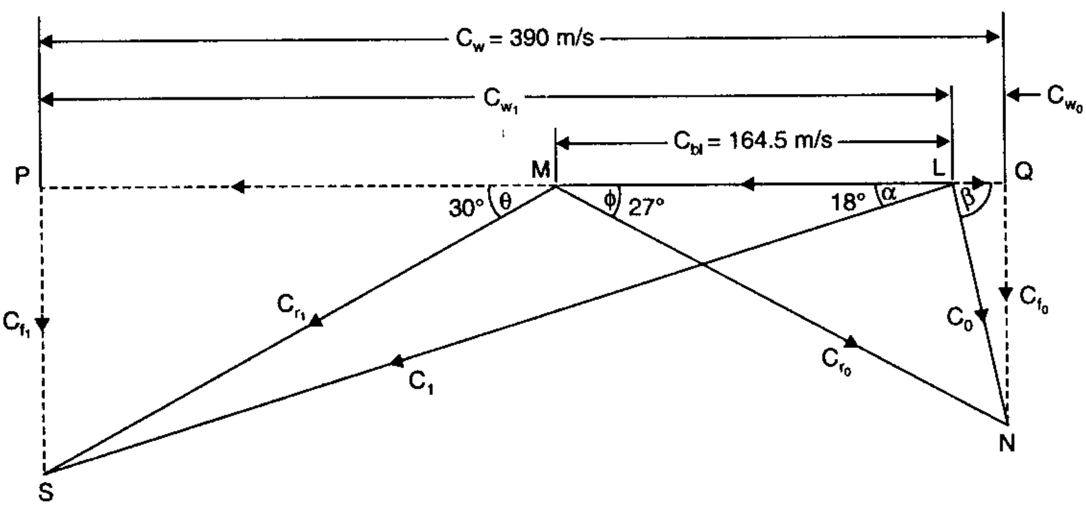

Fig. 3.106

(*ii*) **Axial thrust**

$$= \dot{m}_s(C_{f_1} - C_{f_0}) = 10\ (120 - 95) = \mathbf{250\ N.}\quad \textbf{(Ans.)}$$

(*iii*) **Resultant thrust**

$$= \sqrt{(3900)^2 + (250)^2} = \mathbf{3908\ N.}\quad \textbf{(Ans.)}$$

(*iv*) **Power developed**

$$= \frac{\dot{m}_s(C_{w_1} + C_{w_0}) \times C_{bl}}{1000} = \frac{10 \times 390 \times 164.5}{1000} = \mathbf{641.55\ kW.}\quad \textbf{(Ans.)}$$

(*v*) **Blading efficiency**

$$\eta_{bl} = \frac{2\,C_{bl}\,(C_{w_1} + C_{w_0})}{C_1^{\,2}} = \frac{2 \times 164.5 \times 390}{392^2} = \mathbf{83.5\%.}\quad \textbf{(Ans.)}$$

Example 3.15. *In a stage of impulse reaction turbine provided with single row wheel, the mean diameter of the blades in 1 m. It runs at 3000 r.p.m. The steam issues from the nozzle at a velocity of 350 m/s and the nozzle angle is 20°. The rotor blades are equiangular. The blade friction factor is 0.86. Determine the power developed if the axial thrust on the end bearing of a rotor is 118 N.*

Solution. Mean diameter of the blades, $D = 1$ m
Speed of the turbine, $N = 3000$ r.p.m.
Velocity of steam issuing from the nozzle, $C_1 = 350$ m/s
Nozzle angle, $\alpha = 20°$
Blade angles, $\theta = \phi$
Blade friction factor, $K = 0.86$
Axial thrust $= 118\ N$

Blade, velocity, $$C_{bl} = \frac{\pi DN}{60} = \frac{\pi \times 1 \times 3000}{60} = 157 \text{ m/s}$$

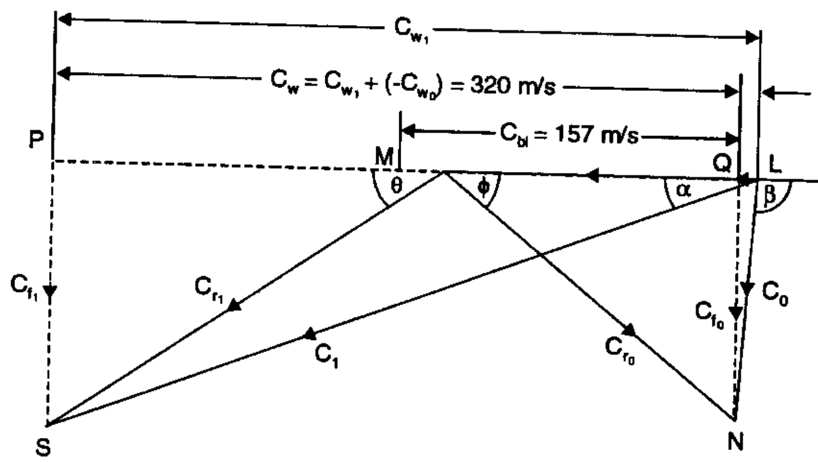

Fig. 3.107

— With the data, $C_{bl} = 157$ m/s, $C_1 = 350$ m/s, $\alpha = 20°$ draw the ΔLMS (Fig. 3.107).

By measurement, $\theta = 35°$

Since the *blades are equiangular,* $\theta = \phi = 35°$

— Now with $\phi = 35°$ and $C_{r_0} = 0.86\ C_{r_1}$, complete the ΔLMN.

On measurement ; $C_{f_1} = 122$ m/s, $C_{f_0} = 102.5$ m/s

Also, axial thrust $\dot{m}_s(C_{f_1} - C_{f_0}) = 118$

$$\therefore \quad \dot{m}_s = \frac{118}{C_{f_1} - C_{f_0}} = \frac{118}{(120 - 102.5)} = 6.74 \text{ kg/s}$$

Further in this case, $C_w = C_{w_1} + C_{w_0} = C_{w_1} + (-C_{w_0}) = 320$ m/s

Now, *power developed,* $$P = \frac{\dot{m}_s(C_{w_1} + C_{w_0}) \times C_{bl}}{1000} \text{ kW}$$

$$= \frac{6.74 \times 320 \times 157}{1000} = \mathbf{338.6\ kW.} \quad \textbf{(Ans.)}$$

☞**Example 3.16.** *The first stage of an impulse turbine is compounded for velocity and has two rows of moving blades and one ring of fixed blades. The nozzle angle is 15° and the leaving angles of blades are respectively, first-moving 30°, fixed 20° ; second-moving 30°. The* *velocity of steam leaving the nozzle is 540 m/s. The friction loss in each blade row is 10% of the relative velocity. Steam leaves the second row of moving blades axially.*

Find : (*i*) *Blade velocity* (*ii*) *Blade efficiency*

(*iii*) *Specific steam consumption.*

Solution. Refer Fig. 3.108.

Nozzle angle, $\alpha = 15° \; ; \; \alpha' = 20°$

$\beta' = 90°$ [since the steam leaves the blades *axially*]

$\phi = \phi' = 30°$

Velocity of steam leaving the nozzle, $C_1 = 540$ m/s

$$\frac{C_{r_0}}{C_{r_1}} = 0.9$$

$$\frac{C_1'}{C_0} = 0.9$$

$$\frac{C_{r_0}'}{C_{r_1}'} = 0.9$$

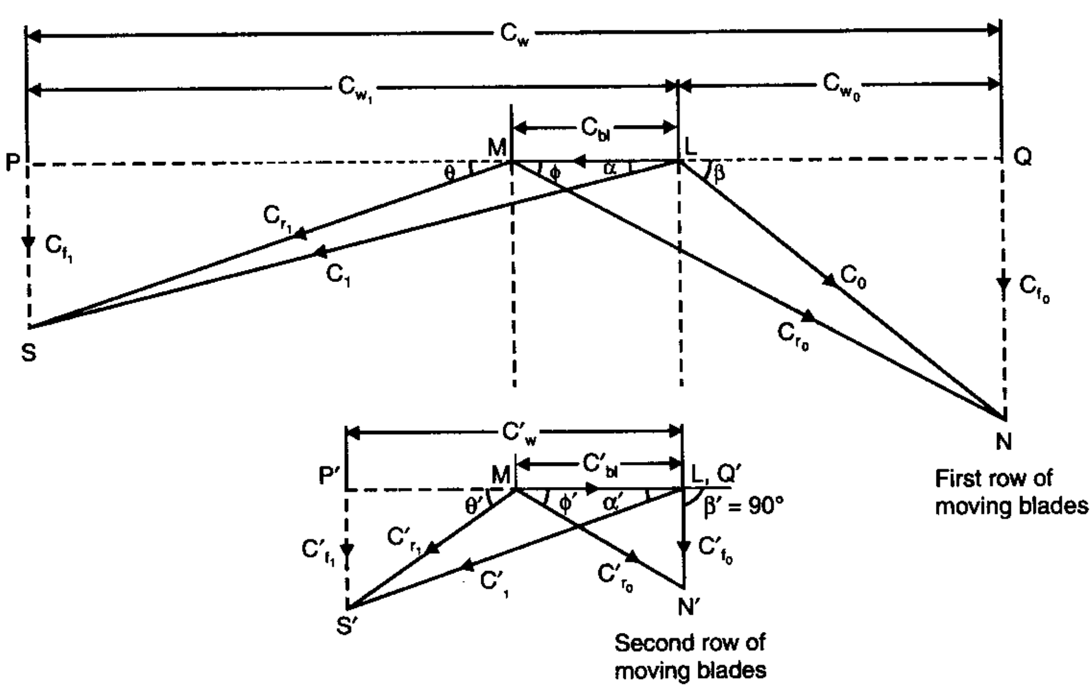

Fig. 3.108

Second row of moving blades :

The velocity triangles should be drawn starting from the second row of moving blades.

The procedure is as follows : Refer Fig. 3.108.

— Draw *LM* to any convenient scale (say 3 cm) as C_{bl} is not known.

— Draw $\phi' = 30°$ and draw perpendicular through the point L (or Q') to LM as $\beta' = 90°$. This meets the line MN' at N'. This completes the outlet triangle LMN'.

— Measure $C_{r_0}' = MN' = 3.5$ cm

$$C_{r_1}' = \frac{C_{r_0}}{0.9} = \frac{3.5}{0.9} = 3.9 \text{ cm.}$$

— Draw an $\angle\alpha' = 20°$ and draw an arc of radius 3.9 cm with centre at M to cut the line LS' at S'. Join MS'. This completes the inlet velocity $\Delta LMS'$. Measure $LS' = C_1' = 6.5$ cm.

First row of moving blades :

The following steps are involved in drawing velocity triangle for first row of moving blades.

— Draw
$$LM = C_{bl} \ (= 3 \text{ cm})$$

$$C_0 = \frac{C_1'}{0.9} = \frac{6.5}{0.9} = 7.22 \text{ cm.}$$

— Draw $\angle\phi = 30°$, through the point M to the line LM.

— Draw an arc of radius 7.22 cm with centre L. This arc cuts the line LN at point N. Join MN. This completes the outlet ΔLMN.

— Measure
$$C_{r_0} = MN = 9.7 \text{ cm}$$

$$MS = C_{r_1} = \frac{C_{r_0}}{0.9} = \frac{9.7}{0.9} = 10.8 \text{ cm}$$

— Draw an $\angle\alpha = 15°$, through a point L. Draw an arc of radius of 10.8 cm with centre at M. This arc cuts the line LS at S. Join MS. This completes the inlet velocity triangle.

Measure LS from the velocity triangle

$$LS = 13.8 \text{ cm} = C_1 = 540 \text{ m/s.}$$

The scale is now calculated from the above.

$\therefore$
$$\text{Scale 1 cm} = \frac{540}{13.8} = 39.1 \text{ m/s}$$

Measure the following distances from the velocity diagram and convert into velocities :

(*i*)
$$C_{bl} = LM = 3 \text{ cm} = 3 \times 39.1 = \mathbf{117.3 \text{ m/s.}} \quad \textbf{(Ans.)}$$
$$C_w = PQ = 18.8 \text{ cm} = 18.8 \times 39.1 = 735.1 \text{ m/s}$$
$$C'_w = P'Q' = 6.2 \text{ cm} = 6.2 \times 39.1 = 242.4 \text{ m/s}$$

(*ii*) **Blade efficiency,**
$$\eta_{bl} = \frac{2\,C_{bl}(C_w + C_w')}{C_1^2}$$
$$= \frac{2 \times 117.3(735.1 + 242.4)}{(540)^2} = \mathbf{0.786 \text{ or } 78.6\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Specific steam consumption, m_s :**

$$1 = \frac{m_s(C_w + C'_w)C_{bl}}{3600 \times 1000} = \frac{m_s(735.1 + 242.4) \times 117.3}{3600 \times 1000}$$

$\therefore$
$$m_s = \frac{3600 \times 1000}{(735.1 + 242.4) \times 117.3} = \mathbf{31.39 \text{ kg/kWh.}} \quad \textbf{(Ans.)}$$

Example 3.17. *The following data relate to a compound impulse turbine having two rows of moving blades and one row of fixed blades in between them.*

The velocity of steam leaving the nozzle	= 600 m/s
Blade speed	= 125 m/s
Nozzle angle	= 20°
First moving blade discharge angle	= 20°
First fixed blade discharge angle	= 25°
Second moving blade discharge angle	= 30°
Friction loss in each ring	= 10% of relative velocity.

Find : (i) Diagram efficiency

(ii) Power developed for a steam flow of 6 kg/s.

Solution. Refer Fig. 3.109.

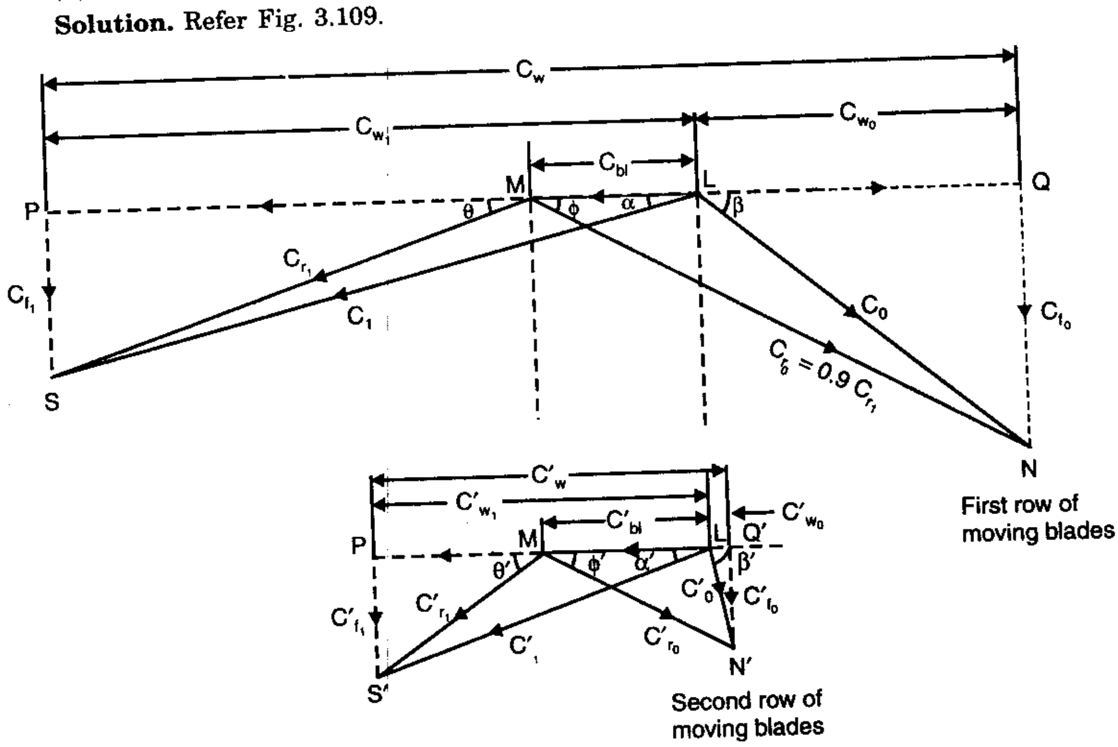

Fig. 3.109

First row of moving blades :

To draw velocity triangles for first row of moving blades the following procedure may be followed :

Select a suitable scale.

— Draw LM = blade velocity (C_{bl}) = 125 m/s.

— Make $\angle MLS$ = nozzle angle, α = 20°.

— Draw LS = velocity of steam leaving the nozzle = 600 m/sec.

— Join MS to complete the inlet triangle LMS.

— Make $\angle LMN$ = outlet angle of first moving blades = 20°

and cut $MN = 0.9\ MS$, since $K = 0.9$.

— Join LN to complete the outlet ΔLMN.

Second row of moving blades :

The velocity triangles for second row of moving blades may be drawn as follows :

— Draw LM = blade velocity (C_{bl}) = 125 m/sec.

— Make $\angle MLS'$ = outlet angle of fixed blade = 25°

and cut $LS' = 0.9\ LN$. $(\because\ K = 0.9)$

— Join MS'. The inlet velocity triangle LMS' is completed.

and cut $MN' = 0.9\ MS'$ $(\because\ K = 0.9)$

— Join LN'. The outlet velocity triangle is completed.

The following required data may now be scaled off from the diagram :

$$C_w = C_{w_1} + C_{w_0} = PQ = 845 \text{ m/s}$$

$$C_w' = P'Q' = 280 \text{ m/s.}$$

(*i*) **Diagram efficiency,** $$\eta_{bl} = \frac{2C_{bl}(C_w + C_w')}{C_1^2}$$

$$= \frac{2 \times 125(845 + 280)}{(600)^2} = \mathbf{0.781 \text{ or } 78.1\%.} \quad \textbf{(Ans.)}$$

(*ii*) **Power developed,** $$P = \frac{\dot{m}_s(C_w + C_w')}{1000} C_{bl}$$

$$= \frac{6(845 + 280) \times 125}{1000} = \mathbf{843.75\ kW.} \quad \textbf{(Ans.)}$$

Reaction Turbines

Example 3.18. *The following data refer to a particular stage of a Parson's reaction turbine :*

Speed of the turbine	*= 1500 r.p.m.*
Mean diameter of the rotor	*= 1 metre*
Stage efficiency	*= 80 per cent*
Blade outlet angle	*= 20°*
Speed ratio	*= 0.7*

Determine the available isentropic enthalpy drop in the stage.

Solution. Mean diameter of the rotor, $D = 1$ m

Turbine speed, $N = 1500$ r.p.m.

Blade outlet angle, $\phi = 20°$

Speed ratio, $$\rho = \frac{C_b}{C_1} = 0.7$$

Stage efficiency, $\eta_{stage} = 80\%$

Isentropic enthalpy drop :

Blade speed, $$C_{bl} = \frac{\pi DN}{60} = \frac{\pi \times 1 \times 1500}{60} = 78.54 \text{ m/s}$$

But $$\rho = \frac{C_{bl}}{C_1} = 0.7 \text{ (given)}$$

$\therefore$ $$C_1 = \frac{C_{bl}}{0.7} = \frac{75.54}{0.7} = 112.2 \text{ m/s}$$

In *Parson's Turbine,* $\alpha = \phi$.

With the above data known, the velocity diagram for the turbine can be drawn to a suitable scale as shown in Fig. 3.110.

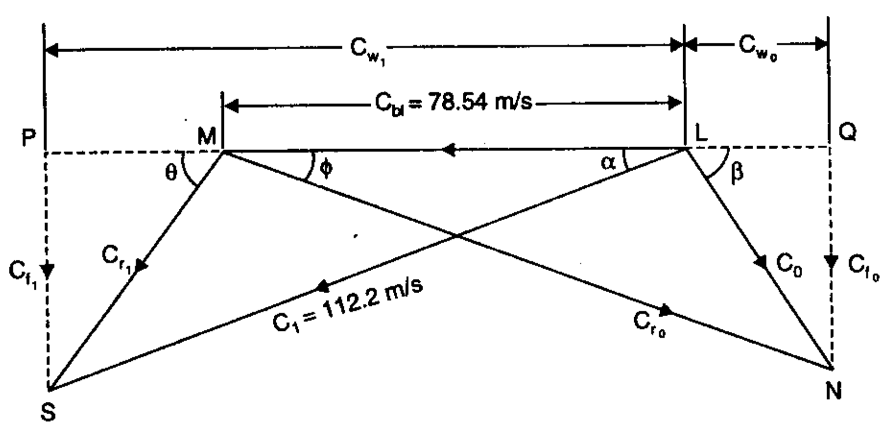

Fig. 3.110

By measurement (from the diagram),

$$C_{w_1} = 106.5 \text{ m/s} \; ; \; C_{w_0} = 27 \text{ m/s}$$

$$\eta_{stage} = \frac{C_{bl}(C_{w_1} + C_{w_0})}{h_d} \quad \text{where } h_d = \text{Isentropic enthalpy drop.}$$

i.e., $$0.8 = \frac{78.54(106.25 + 27)}{h_d \times 1000}$$

$\therefore$ $$h_d = \frac{78.54(106.25 + 27)}{0.8 \times 1000} = \mathbf{13.08 \text{ kJ}}$$

Hence, *isentropic enthalpy drop* = **13.08 kJ/kg. (Ans.)**

Example 3.19. *In a reaction turbine, the blade tips are inclined at 35° and 20° in the direction of motion. The guide blades are of the same shape as the moving blades, but reversed in direction. At a certain place in the turbine, the drum diameter is 1 metre and the blades are 10 cm high. At this place, the steam has a pressure of 1.75 bar and dryness 0.935. If the speed of this turbine is 250 r.p.m. and the steam passes through the blades without shock, find the mass of steam flow and power developed in the ring of moving blades.*

Solution. Refer Fig. 3.111.

Angles, $\alpha = \phi = 20°$, and $\theta = \beta = 35°$

Mean drum diameter, $D_m = 1 + 0.1 = 1.1$ m

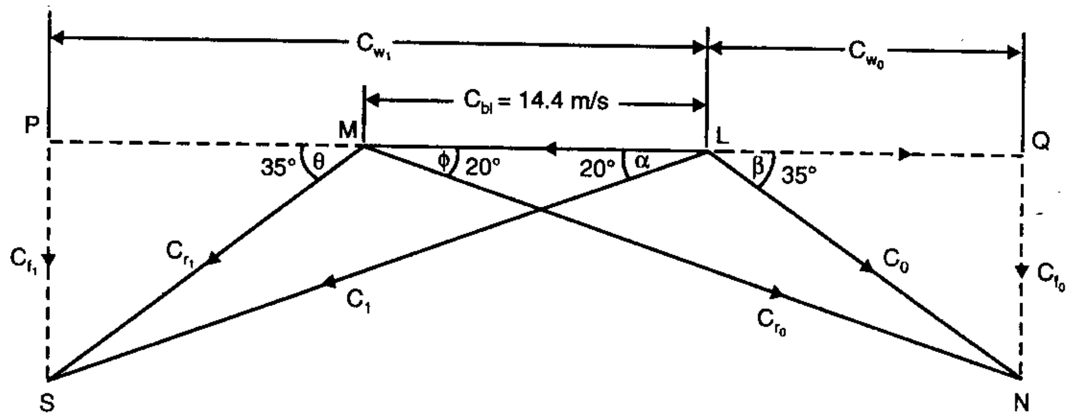

Fig. 3.111

Area of flow $= \pi D_m h$, where h is the height of blade.

$= \pi \times 1.1 \times 0.1 = 0.3456 \text{ m}^2$

Steam pressure $= 1.75$ bar

Dryness fraction of steam, $x = 0.935$

Speed of the turbine, $N = 250$ r.p.m.

Rate of steam flow, $\dot{m}_s$:

Power developed, P :

Blade speed, $$C_{bl} = \frac{\pi DN}{60} = \frac{\pi \times 1.1 \times 250}{60} = 14.4 \text{ m/s}$$

With the above given data the velocity diagram can be drawn to a suitable scale as shown in Fig. 3.111.

By measurement (from diagram),

$$C_{w_1} = 30 \text{ m/s} \; ; \; C_{w_0} = 15.45 \text{ m/s} \; ; \; C_{f_1} = C_{f_0} = 10.8 \text{ m/s}$$

From steam tables corresponding to 1.75 bar pressure,

v_g = Specific volume of dry saturated steam

$= 1.004 \text{ m}^3/\text{kg}$

$x = 0.935$

∴ Specific volume of wet steam $= xv_g = 0.935 \times 1.004 = 0.938 \text{ m}^3/\text{kg}$

Mean flow rate is given by,

$$\dot{m}_s = \frac{\text{Area of flow} \times \text{Velocity of flow}}{\text{Specific volume of steam}}$$

$$= \frac{0.3456 \times 10.8}{0.938} = 3.98 \text{ kg/s}.$$

Power developed,

$$P = \frac{\dot{m}_s(C_{w_1} + C_{w_0})C_{bl}}{1000}$$

$$= \frac{3.98(30 + 15.45) \times 14.4}{1000} = \textbf{2.6 kW}. \quad \textbf{(Ans.)}$$

Example 3.20. *In a reaction turbine, the fixed blades and moving blades are of the same shape but reversed in direction. The angles of the receiving tips are 35° and of the discharging tips 20°. Find the power developed per pair of blades for a steam consumption of 2.5 kg/s, when the blade speed is 50 m/s. If the heat drop per pair is 10.04 kJ/kg, find the efficiency of the pair.*

Solution. Angles of receiving tips, $\theta = \beta = 35°$

Angles of discharging tips, $\alpha = \phi = 20°$

Steam consumption, $\dot{m}_s = 2.5$ kg/s

Blade speed, $C_{bl} = 50$ m/s

Heat drop per pair, $h_d = 10.04$ kJ/kg

Power developed = ?

Efficiency of the pair = ?

Refer Fig. 3.112.

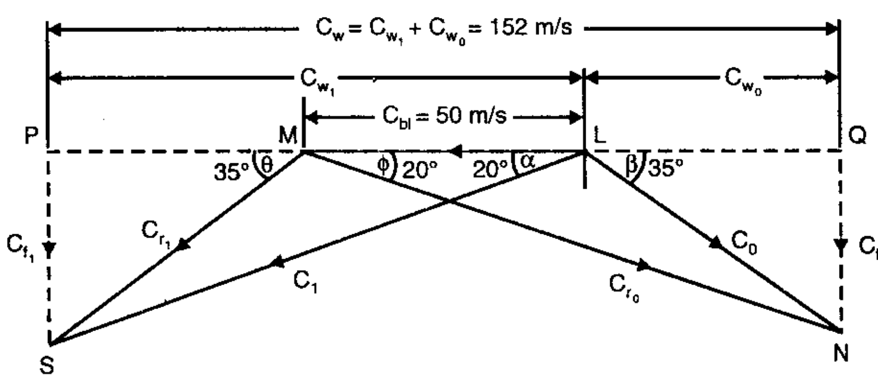

Fig. 3.112

$$NS = PQ = 152 \text{ m/s}$$

Work done per pair per kg of steam

$$= (C_{w_1} + C_{w_0})\, C_{bl} = 152 \times 50 = 7600 \text{ Nm/kg of steam.}$$

$$Power/pair = \frac{\dot{m}_s (C_{w_1} + C_{w_0}) C_{bl}}{1000} = \frac{2.5 \times 7600}{1000} = \mathbf{19\ kW.} \quad \textbf{(Ans.)}$$

$$Efficiency = \frac{\text{Work done per pair per kg of steam}}{h_d}$$

$$= \frac{7600}{10.04 \times 1000} = 0.757 = \mathbf{75.7\%.} \quad \textbf{(Ans.)}$$

☞**Example 3.21.** *A twenty-stage Parson turbine receives steam at 15 bar at 300°C. The steam leaves the turbine at 0.1 bar pressure. The turbine has a stage efficiency of 80% and the reheat factor 1.06. The total power developed by the turbine is 10665 kW. Find the steam flow rate through the turbine assuming all stages develop equal power.*

The pressure of steam, at certain stage of the turbine is 1 bar abs., and is dry and saturated. The blade exit angle is 25° and the blade speed ratio is 0.75. Find the mean diameter of the rotor of this stage and also the rotor speed. Take blade height as 1/12th of the mean diameter. The thickness of the blades may be neglected.

Solution. Number of stage $= 20$

Steam supply pressure $= 15$ bar, 300°C

Exhaust pressure $= 0.1$ bar

Stage efficiency of turbine, $\eta_{stage} = 80\%$

Reheat factor $= 1.06$

Total power developed $= 10665$ kW

Steam flow rate, m_s :

Steam pressure at a certain stage $= 1$ bar abs., $x = 1$

Blade exit angle $= 25°$

Blade speed ratio, $\rho = \dfrac{C_{bl}}{C_1} = 0.75$

Height of the blade, $h = \dfrac{1}{12} D$ (mean dia. of rotor)

(*i*) **Steam flow rate, $\dot{m}_s$:**

Refer Fig. 3.113.

Isentropic drop, $(\Delta h)_{isentropic} = h_1 - h_2 = 3040 - 2195 = 845$ kJ/kg

$\eta_{overall} = \eta_{stage} \times$ Reheat factor $= 0.8 \times 1.06 = 0.848$

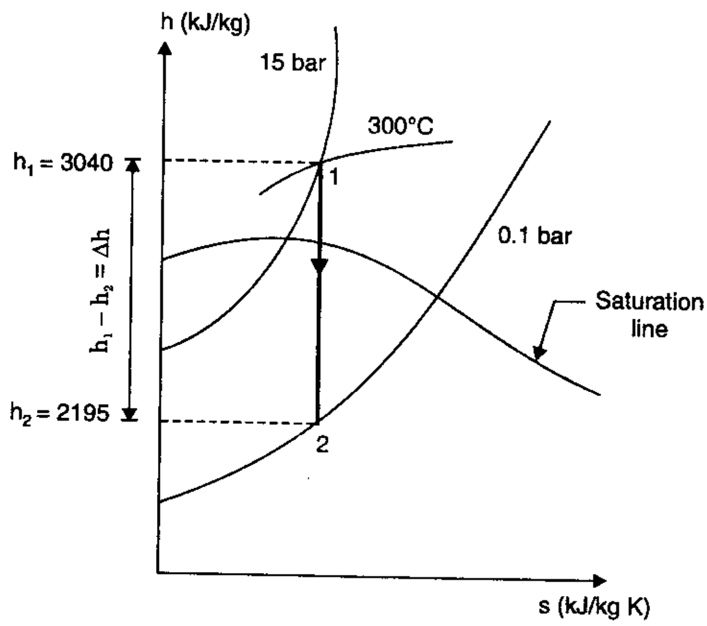

Fig. 3.113

Work done $=$ Actual enthalpy drop

$= (\Delta h)_{isentropic} \times \eta_{overall}$

$= 845 \times 0.848 = 716.56$ kJ/kg

Work done per stage per kg $= \dfrac{716.56}{20} = 35.83$ kJ

Also, total power $=$ no. of stages $\times$ $\dot{m}_s$ $\times$ work done/kg stage

$\therefore \qquad 10665 = 20 \times \dot{m}_s \times 35.83$

$\therefore \qquad \dot{m}_s = \dfrac{10665}{20 \times 35.83} = \mathbf{14.88\ kg/s.}$ **(Ans.)**

(*ii*) **Mean diameter of rotor, D :**

Rotor speed, N :

Refer Fig. 3.114.

Work done per kg per stage $= C_{bl} \times C_w = C_{bl}\,(2C_1 \cos 25° - C_{bl})$

Also, $\dfrac{C_{bl}}{C_1} = 0.75$ ...(Given)

$\therefore \qquad C_1 = \dfrac{C_{bl}}{0.75} = 1.33C_{bl}$

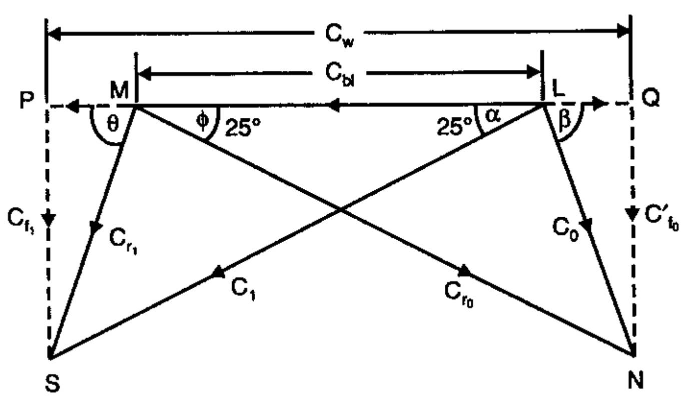

Fig. 3.114

i.e., Work done per kg per stage

$= C_{bl}\,(2 \times 1.33C_{bl} \times 0.906 - C_{bl})$

$= 1.41\ C_{bl}^2$ ...(*ii*)

Equation (*i*) and (*ii*), we get

$1.41\ C_{bl}^2 = 35.83 \times 1000$

$\therefore \qquad C_{bl}^2 = \dfrac{35.83 \times 1000}{1.41}$ *i.e.,* $C_{bl} = 159.41$ m/s

$\therefore \qquad C_1 = 159.41 \times 1.33 = 212$ m/s

From Fig. 3.114

$C_{f_1} = C_1 \sin \alpha = 212 \sin 25° = 89.59$ m/s

v_g = Specific volume at 1 bar when steam is dry and saturated

= 1.694 m³/kg (from steam tables)

Mass flow rate, $\dot{m}_s = \dfrac{\pi D h C_{f_1}}{v}$

$\therefore \qquad 14.88 = \dfrac{\pi \times D \times \left(\dfrac{D}{12}\right) \times 89.59}{1.694}$ or $D^2 = \dfrac{14.88 \times 1.694 \times 12}{\pi \times 89.59}$

$$\therefore \qquad D = \mathbf{1.036\ m.} \quad \textbf{(Ans.)}$$

Now, $$h = \frac{D}{12} = \frac{1.036}{12} = 0.086 \text{ m} = \mathbf{8.6\ cm.} \quad \textbf{(Ans.)}$$

Also, $$C_{bl} = \frac{\pi DN}{60}$$

$$\therefore \qquad N = \frac{C_{bl} \times 60}{\pi D} = \frac{159.41 \times 60}{\pi \times 1.036} = \mathbf{2938.7\ r.p.m.} \quad \textbf{(Ans.)}$$

Example 3.22. *The following data relate to a stage of reaction turbine :*

Mean rotor diameter = 1.5 m ; speed ratio = 0.72 ; blade outlet angle = 20° ; rotor speed = 3000 r.p.m.

(i) Determine the diagram efficiency.

(ii) Determine the percentage increase in diagram efficiency and rotor speed if the rotor is designed to run at the best theoretical speed, the exit angle being 20°.

Solution. Mean rotor diameter, $D = 1.5$ m

Speed ratio, $$\rho = \frac{C_{bl}}{C_1} = 0.72$$

Blade outlet angle $= 20°$

Rotor speed, $N = 3000$ r.p.m.

(This example solved purely by calculations (Fig. 3.115) is not drawn to scale.)

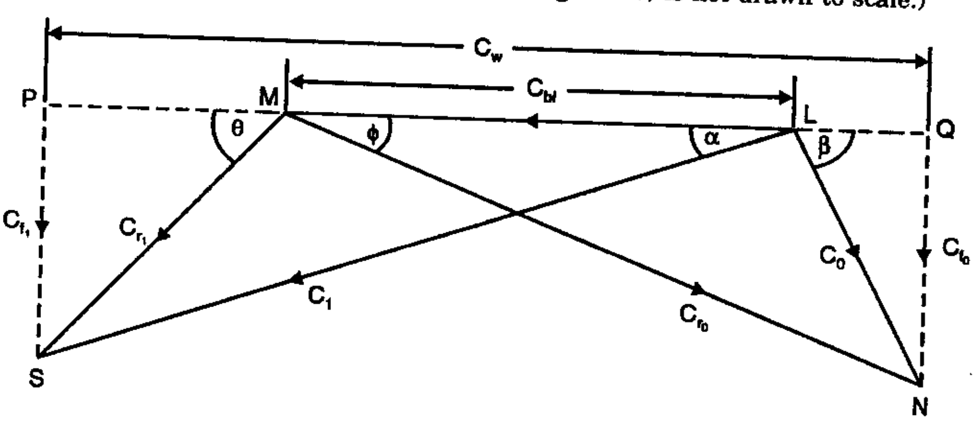

Fig. 3.115

(*i*) **Diagram efficiency :**

Blade velocity, $$C_{bl} = \frac{\pi DN}{60} = \frac{\pi \times 1.5 \times 3000}{60} = 235.6 \text{ m/s}$$

Speed ratio, $$\rho = \frac{C_{bl}}{C_1} = 0.72$$

$$\therefore \qquad C_1 = \frac{C_{bl}}{0.72} = \frac{235.6}{0.72} = 327.2 \text{ m/s}$$

Assuming that velocity triangles are symmetrical,

$$\alpha = \phi = 20°$$

From the velocity ΔLMS, $$C_{r_1}^{\ 2} = C_1^{\ 2} + C_{bl}^{\ 2} - 2C_1 C_{bl} \cos \alpha \qquad \ldots(i)$$

$$C_{r_1} = \sqrt{(327.2)^2 + (235.6)^2 - 2 \times 327.2 \times 235.6 \cos 20^\circ}$$

$$= 100\ \sqrt{10.7 + 5.55 - 14.48} = 133 \text{ m/s}$$

i.e., $C_{r_1} = 133$ m/s

Work done per kg of steam $= C_{bl}C_w = C_{bl}(2C_1 \cos \alpha - C_{bl})$

$= 235.6\ (2 \times 327.2 \cos 20^\circ - 235.6) = 89371.3$ N-m.

Energy supplied per kg of steam

$$= \frac{C_1^2 + C_{r_0}^2 - C_{r_1}^2}{2}$$

$$= \frac{2C_1^2 - C_{r_1}^2}{2} \qquad (\text{as } C_1 = C_{r_0})$$

$$= \frac{2 \times (327.2)^2 - (133)^2}{2} = 98215.3 \text{ N-m}$$

$\therefore$ *Diagram efficiency* $= \dfrac{89371.3}{98215.3} = 0.91 =$ **91%. (Ans.)**

(*ii*) **Percentage increase in diagram efficiency :**

For the best diagram efficiency (maximum), the required condition is

$$\rho = \frac{C_{bl}}{C_1} = \cos \alpha$$

$$\therefore \quad C_{bl} = C_1 \cos \alpha = 372.2 \cos 20^\circ = 307.46 \text{ m/s}$$

For this blade speed, the value of C_{r_1} is again calculated by using eqn. (*i*),

$$C_{r_1} = \sqrt{(327.2)^2 + (307.46)^2 - 2 \times 327.2 \times 307.46 \times \cos 20^\circ}$$

$$= 100\ \sqrt{10.7 + 9.45 - 18.906} = 111.5 \text{ m/s}$$

Diagram efficiency

$$= \frac{2C_{bl}(2C_1 \cos \alpha - C_{bl})}{(C_1^2 + C_{r_0}^2 - C_{r_1}^2)}$$

$$= \frac{2 \times 307.46(2 \times 327.2 \cos 20^\circ - 307.46)}{(327.2)^2 + (327.2)^2 - (111.5)^2} = 0.937 \text{ or } 93.7\%$$

Percentage increase in diagram efficiency

$$= \frac{0.937 - 0.91}{0.91} = \mathbf{0.0296 \text{ or } 2.96\%.} \quad \textbf{(Ans.)}$$

The best theoretical speed of the rotor is given by,

$$C_{bl} = \frac{\pi DN}{60} \quad \therefore \quad N = \frac{60\, C_{bl}}{\pi D} = \frac{60 \times 307.46}{\pi \times 1.5} = \mathbf{3914.7 \text{ r.p.m.}} \quad \textbf{(Ans.)}$$

☞**Example 3.23. (Impulse reaction turbine).** *The following data relate to a stage of an impulse reaction turbine :*

Steam velocity coming out of nozzle = 245 m/s ; nozzle angle = 20° ; blade mean speed = 145 m/s ; speed of the rotor = 300 r.p.m. ; blade height = 10 cm ; specific volume of steam at nozzle outlet and blade outlet respectively = 3.45 m³/kg and 3.95 m³/kg ; H.P. developed by the turbine = 390 ; efficiency of nozzle and blades combinedly = 90% ; carry over co-efficient = 0.82.

Find : (*i*) *The heat drop in each stage* ; (*ii*) *Degree of reaction* ; (*iii*) *Stage efficiency*.

Solution. Steam velocity coming out of nozzle, $C_1 = 245$ m/s

Nozzle angle, $\alpha = 20°$

Blade mean speed, $C_{bl} = 145$ m/s

Speed of the rotor, $N = 3000$ r.p.m.

Blade height, $h = 10$ cm $= 0.1$ m

Specific volume of steam at nozzle outlet or blade inlet, $v_1 = 3.45$ m³/kg

Specific volume of steam at blade outlet, $v_0 = 3.95$ m³/kg

H.P. developed by the turbine = 390 H.P.

Efficiency of nozzle and blades combinedly = 90%

Carry over co-efficient, $\psi = 0.82$

Blade speed, $C_{bl} = \dfrac{\pi DN}{60}$

$$\therefore \quad D = \frac{60\,C_{bl}}{\pi N} = \frac{60 \times 145}{\pi \times 3000} = 0.923 \text{ m}$$

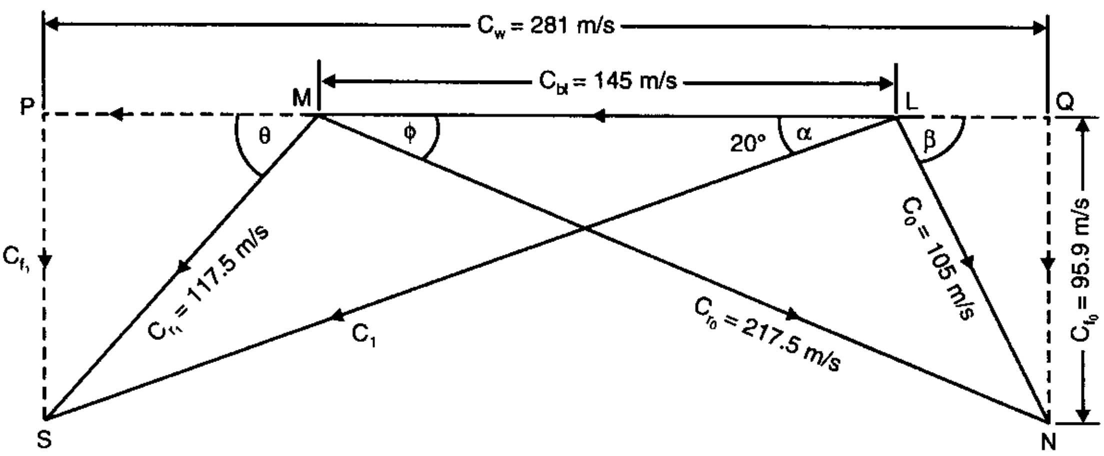

Fig. 3.116

Mass flow rate, $\dot{m}_s = \dfrac{C_{f_1} \times \pi Dh}{v_1} = \dfrac{C_1 \sin\alpha \times \pi Dh}{v_1}$

$$= \frac{245 \times \sin 20° \times \pi \times 0.923 \times 0.1}{3.45} = 7.04 \text{ kg/s}$$

Also $\dot{m}_s = \dfrac{C_{f_0}\,\pi Dh}{v_0}$

$$\therefore \quad C_{f_0} = \frac{\dot{m}_s v_0}{\pi Dh} = \frac{7.04 \times 3.95}{\pi \times 0.923 \times 0.1} = 95.9 \text{ m/s}$$

The H.P. is given by, $\text{H.P.} = \dfrac{\dot{m}_s \times C_{bl} \times C_w}{g \times 75}$

$$390 = \frac{7.04 \times 145 \times C_w}{9.81 \times 75}$$

$$\therefore \quad C_w = \frac{390 \times 9.81 \times 75}{7.04 \times 145} = 281 \text{ m/s}$$

Now draw velocity triangles as follows :

Select a suitable scale (say 1 cm = 25 m/s)

— Draw LM = blade velocity = 145 m/s ; $\angle MLS$ = nozzle angle = 20°
 Join MS to complete the inlet ΔLMS.

— Draw a perpendicular from S which cuts the line through LM at point P.
 Mark the point Q such that $PQ = C_w$ = 281 m/s.

— Draw a perpendicular through point Q and the point N as QN = 95.9 m/s.
 Join LN and MN to complete the *outlet* velocity triangle.

From the velocity triangles ;

$$C_{r_1} = 117.5 \text{ m/s}$$

$$C_{r_0} = 217.5 \text{ m/s}$$

$$C_0 = 105 \text{ m/s.}$$

(*i*) **Heat drop in each stage, $(\Delta h)_{stage}$:**

Heat drop in *fixed blades* $(\Delta h_f) = \dfrac{C_1^2 - \psi C_0^2}{2gJ\,\eta_{nozzle}}$, where ψ is a carry over coefficient

$$= \frac{(245)^2 - 0.82 \times (105)^2}{2 \times 9.81 \times 427 \times 0.9} = 6.76 \text{ kcal/kg}$$

Heat drop in *moving blades* (Δh_m)

$$= \frac{C_{r_0}^2 - C_{r_1}^2}{2gJ\,\eta_{nozzle}} = \frac{(217.5)^2 - (117.5)^2}{2 \times 9.81 \times 427 \times 0.9} = 4.44 \text{ kcal/kg}$$

Total heat drop in a stage,

$$(\Delta h)_{stage} = \Delta h_f + \Delta h_m$$

$$= 6.76 + 4.44 = 11.2 \text{ kcal/kg.}$$

(*ii*) **Degree of reaction, R_d :**

$$R_d = \frac{\Delta h_m}{\Delta h_m + \Delta h_f} = \frac{4.44}{4.44 + 6.76} = \mathbf{0.396.} \quad \textbf{(Ans.)}$$

(*iii*) **Stage efficiency, η_{stage} :**

Work done per kg of steam

$$= \frac{C_{bl} \times C_w}{gJ} = \frac{145 \times 281}{9.81 \times 427} = 9.726 \text{ kcal}$$

$$\therefore \quad \eta_{stage} = \frac{\text{Work done per kg of steam}}{\text{Total heat drop in a stage}}$$

$$= \frac{9.726}{11.2} = \mathbf{0.868 \text{ or } 86.8\%.} \quad \textbf{(Ans.)}$$

3.20. STEAM CONDENSERS

3.20.1. Introduction

*A **steam condenser** is a device or an appliance in which steam condenses and heat released by steam is absorbed by water*. It serves the following *purposes* :

1. It *maintains a very low back pressure* on the exhaust side of the piston of the steam engine or turbine. Consequently, the steam expands to a greater extent which results in an *increase in available heat energy* for converting into mechanical work. The shaded area in Fig. 3.117. (*i.e.*, area 44′5′5) shows the increase in work obtained by fitting a condenser to a non-condensing engine. *The thermal efficiency of a condensing unit therefore is higher than that of non-condensing unit for the same available steam.*

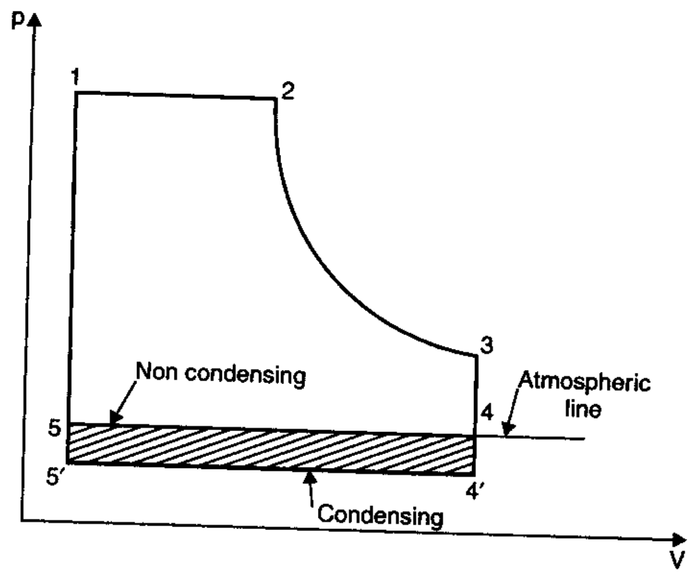

Fig. 3.117

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

3.20.2. Vacuum

***Vacuum** is sub-atmospheric pressure.* It is measured as the pressure depression below atmospheric. The condensation of steam in a closed vessel produces a partial vacuum by reason of the great reduction in the volume of the low pressure steam or vapour. The back pressure in steam engine or steam turbine can be lowered from 0.013 to 0.2 bar abs. or even less. Since the steam engines are intermittent flow machines and as such cannot take the advantage of a very low vacuum, therefore, for most steam engines the exhaust pressure is about 0.2 to 0.28 bar abs. On the other hand, in steam turbines, which are continuous flow machines, the back pressure may be about 0.025 bar abs.

3.20.3. Organs of a Steam Condensing Plant

A steam condensing plant mainly consists of the following *organs/elements* :

1. Condenser (to condense the steam).
2. Supply of cooling (or injection) water.

3. Wet air pump (to remove the condensed steam, the air and uncondensed water vapour and gases from the condenser ; separate pumps may be used to deal with air and condensate).
4. Hot well (where the condensate can be discharged and from which the boiler feed water is taken).
5. Arrangement for recooling the cooling water in case surface condenser is employed.

3.20.4. Classification of Condensers

Mainly, condensers are of two types : (1) Jet condensers, (2) Surface condensers.

In ***jet condensers,*** *the exhaust steam and water come in direct contact with each other* and temperature of the condensate is the same as that of cooling water leaving the condenser. The cooling water is usually sprayed into the exhaust steam to cause rapid condensation.

In ***surface condensers,*** *the exhaust steam and water do not come into direct contact.* The steam passes over the outer surface of tubes through which a supply of cooling water is maintained. There may be single-pass or double-pass. In single-pass condensers, the water flows in one direction only through all the tubes, while in two-pass condenser the water flows in one direction through the tubes and returns through the remainder.

A jet condenser is simpler and cheaper than a surface condenser. It should be installed when the cooling water is cheaply and easily made suitable for boiler feed or when a cheap source of boiler and feed water is available. *A surface condenser is most commonly used because the condensate obtained is not thrown as a waste but returned to the boiler.*

3.20.5. Jet Condensers

These condensers may be classified as :

(*a*) Parallel-flow type

(*b*) Counter-flow type

(*c*) Ejector type.

Parallel flow and counter flow condensers are further sub-divided into two types : (*i*) Low level type (*ii*) High level type.

In *parallel-flow type* of condenser, both the exhaust steam and cooling water find their entry at the top of the condenser and then flow downwards and condensate and water are finally collected at the bottom.

In *counter-flow type,* the steam and cooling water enter the condenser from opposite directions. Generally, the exhaust steam travels in upward direction and meet the cooling water which flows downwards.

Low level jet condenser (Parallel-flow) :

In the Fig. 3.118 is shown a line sketch of a low level parallel-flow condenser. The exhaust steam is entering the condenser from the top and cold water is being sprayed on its way. The baffle plate provided in it ensures the proper mixing of the steam and cooling water. An extraction pump at the bottom discharges the condensate to the hot well from where it may be fed to the boiler if the cooling water being used is free from impurities. A separate dry pump may be incorporated to maintain proper vacuum.

Low level jet condenser (Counter-flow) :

Refer Fig. 3.119. *L, M* and *N* are the perforated trays which break up water into jets. The steam moving upwards comes in contact with water and gets condensed. The condensate and water mixture is sent to the hot well by means of an extraction pump and the air is removed by an air suction pump provided at the top of the condenser.

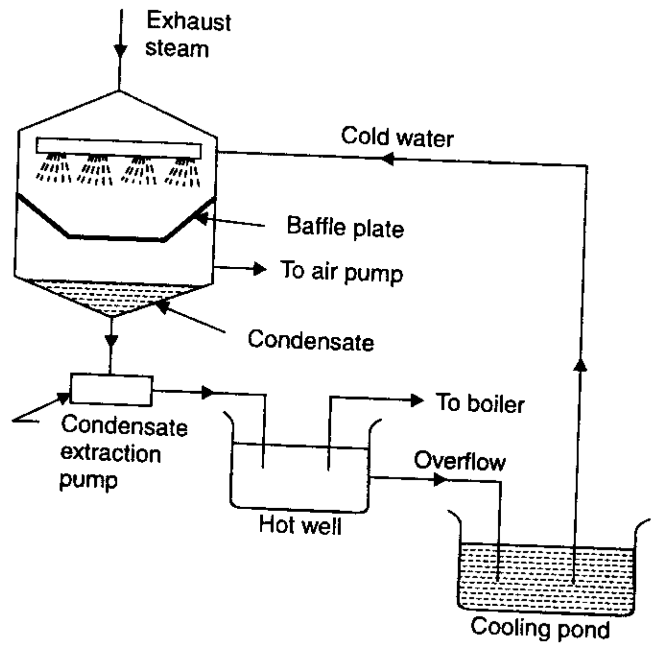

Fig. 3.118. Low Level jet Condenser (Parallel Flow).

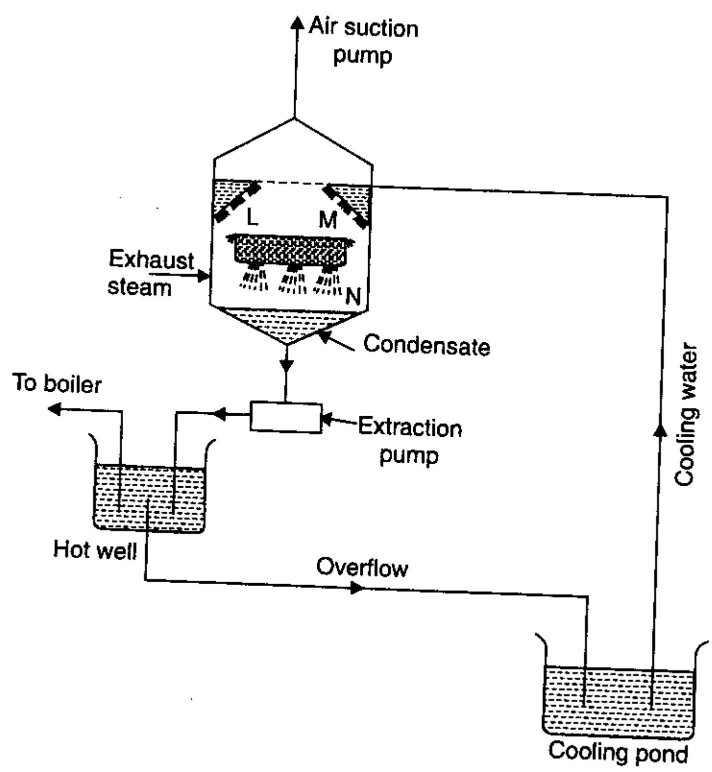

Fig. 3.119. Low level jet Condenser (Counter Flow).

High level jet condenser (Counter-flow type) :

In Fig. 3.120 is shown a high level counter-flow jet condenser. It is also called *barometric condenser*. In this case the shell is placed at a height about 10.363 metres above hot well and thus the necessity of providing an extraction pump can be obviated. However provision of own injection pump has to be made if water under pressure is not available.

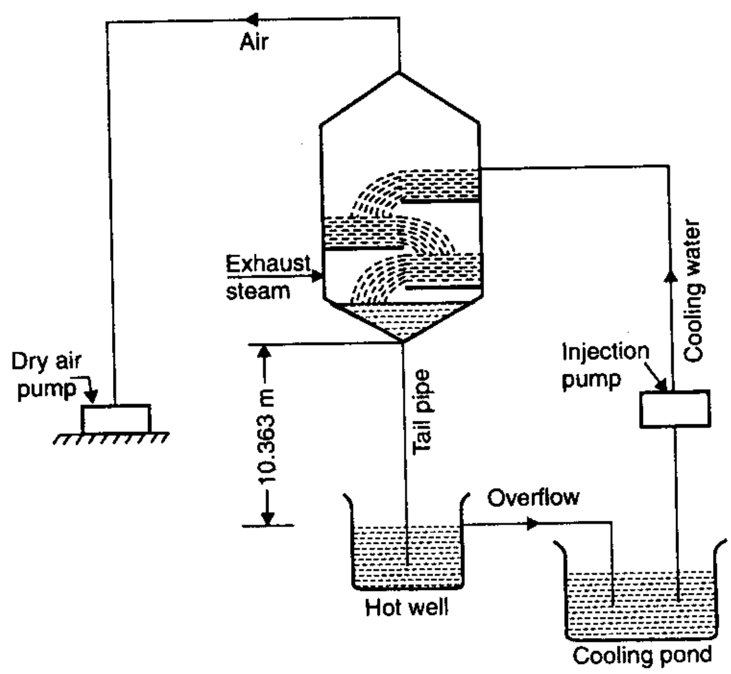

Fig. 3.120. High level jet Condenser (Counter Flow).

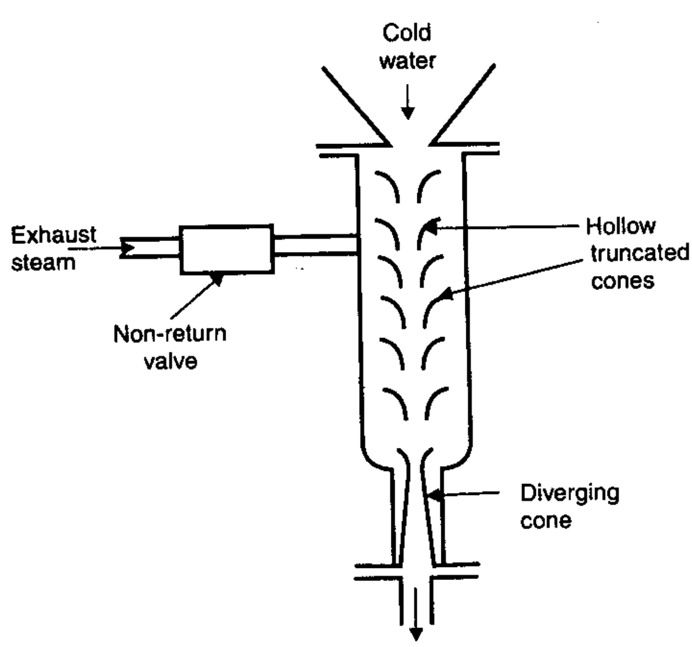

Fig. 3.121. Ejector Condenser.

Ejector condenser :

Fig. 3.121 shows the schematic sketch of an ejector condenser. Here the exhaust steam and cooling water mix in hollow truncated cones. The cold water having a head of about 6 metres flow down through the number of cones and as it moves its velocity increases and drop in pressure results. Due to this decreased pressure exhaust steam along with associated air is drawn through the truncated cones and finally lead to diverging cone. In the diverging cone, a portion of kinetic energy gets converted into pressure energy which is more than the atmospheric so that condensate consisting of condensed steam, cooling water and air is discharged into the hot well. The exhaust steam inlet is provided with a non-return valve which does not allow the water from hot well to rush back to the engine in case a failure of cooling water supply to condenser.

3.20.6. Surface Condensers

Most condensers are generally classified on the direction of flow of condensate, the arrangement of the tubing and the position of the condensate extraction pump. The following is the main classification of surface condensers :

(*i*) Down flow type
(*ii*) Central flow type
(*iii*) Inverted flow type
(*iv*) Regenerative type
(*v*) Evaporative type.

(*i*) Down flow type

In Fig. 3.122 is shown a *down flow type* of surface condenser. It consists of a shell which is generally of cylindrical shape ; though other types are also used. It has cover plates at the ends and furnished with number of parallel brass tubes. A baffle plate partitions the water box into two sections. The cooling water enters the shell at the lower half section and after travelling through the upper half section comes out through the outlet. The exhaust steam entering shell from the top flows down over the tubes and gets condensed and is finally removed by an extraction pump. Due to the fact that steam flows in a direction right angle to the direction of flow of water, it is also called *cross-surface condenser.*

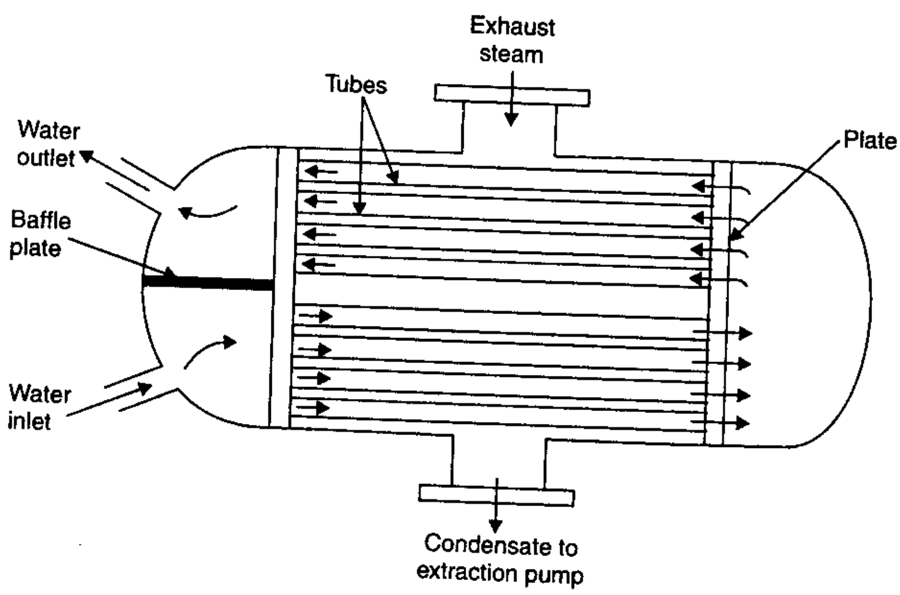

Fig. 3.122. Down Flow Type.

(*ii*) **Central flow type**

Refer Fig. 3.123. In this type of condenser, the suction pipe of the air extraction pump is located in the centre of the tubes which results in radial flow of the steam. The better contact between the outer surface of the tubes and steam is ensured, due to large passages the pressure drop of steam is reduced.

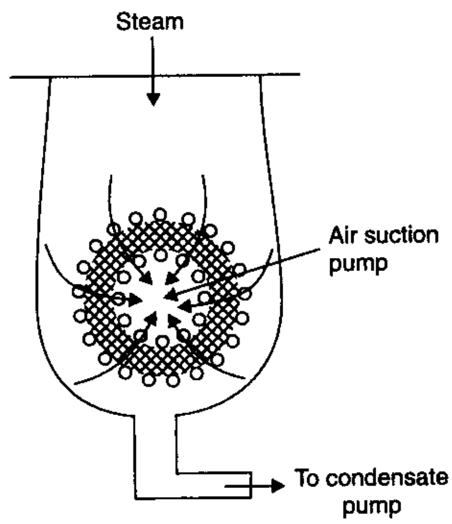

Fig. 3.123. Central flow type.

(*iii*) **Inverted flow type**

This types of condenser has the air suction at the top, the steam after entering at the bottom rises up and then agains flows down to the bottom of the condenser, by following a path near the outer surface of the condenser. The condensate extraction pump is at the bottom.

(*iv*) **Regenerative type**

This type is applied to condensers adopting a regenerative method of heating of the condensate. After leaving the tube nest, the condensate is passed through the entering exhaust steam from the steam engine or turbine thus raising the temperature of the condensate, for use as feed water for the boiler.

(*v*) **Evaporative type**

Fig. 3.124 shows the schematic sketch of an evaporative condenser. The underlying principle of this condenser is that when a limited quantity of water is available, its quantity needed to condense the steam can be reduced by *causing the circulating water to evaporate under a small partial pressure*.

The exhaust steam enters at the top through gilled pipes. The water pump sprays water on the pipes and descending water condenses the steam. The water which is not evaporated falls into the open tank (cooling pond) under the condenser from which it can be drawn by circulating water pump and used over again. The evaporative condenser is placed in open air and finds its application in *small size plants*.

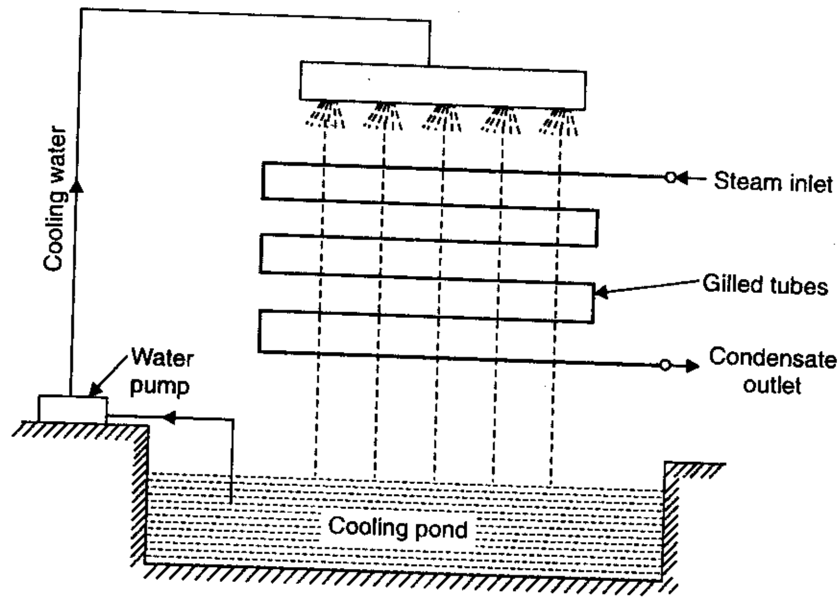

Fig. 3.124. Evaporative condenser.

Advantages of surface condenser :

1. Slightly better vacuum can be obtained.
2. There is no need to treat the condensate chemically before being supplied to the boiler.
3. High vacuum (about 73.5 cm Hg) can be obtained. This increases the thermal efficiency of the plant.
4. Condensate can be reused for boiler feed water.
5. Low pumping power requried.
6. Less chances of losing vacuum as the drop in vacuum has got no effect on water supply.
7. Cooling water of even poor quality can be used because the cooling water does not come in direct contact with steam.

Disadvantages :

1. High initial cost.
2. Requires large floor area.
3. High maintenance cost.
4. More head is necessary in order to obtain sufficient head on hot well pump.
5. Proper cleaning of cooling water is necessary, otherwise it creates the problem of accumulation of dust inside the condenser tubes.

3.20.7. Reasons for Inefficiency in Surface Condensers

The *various reasons for inefficiency in surface condensers* are discussed below :

1. The pressure inside the condenser is less than atmospheric, and in order to obtain the maximum work from unit mass of steam, the pressure should be as low as possible. The pressure in the condenser also depends upon the amount of air. Owing to high vacuum pressure in the condenser it is impossible to prevent air from leaking in through the joints thereby increasing the pressure in the condenser and thus limiting the amount of work

done by unit mass of steam in the engine or turbine. Air leakage also results in lowering the partial pressure of steam and temperature. This means that *latent heat increases* and therefore *more cooling water* is required and the undercooling of the condensate is likely to be more severe with a resulting lower overall efficiency.

2. One of the main causes of poor performance in surface condensers is the pressure drop which occurs as the steam flows over the tubes ; this pressure drop, by increasing the volume of the steam, tends to destroy the vacuum. The decrease in vacuum results in less amount of work done by unit mass of steam.
3. The heat conduction is through the brass tube walls. This conduction of heat is not perfect and results in less efficiency.
4. On examining the heat balance sheets of steam engine plants, it will be found that more than one-half of the heat supplied by fuel is rejected to the condenser cooling water ; this is chief loss of steam plant and is the cause of its low overall efficiency.
5. Steam entering the condenser with high resistance.
6. Circulating water passing through the condenser with high friction and at a velocity not consistent with high efficiency.
7. Undercooling of condensate.
8. Air extraction from hottest section and with comparatively large amount of water vapour.

3.20.8. Comparison Between Jet and Surface Condensers

S. No.	Jet Condenser	Surface Condenser
1.	Low manufacturing cost.	High manufacturing cost.
2.	Lower up keep.	Higher upkeep.
3.	Requires small floor space.	Requires large floor space.
4.	The condensate cannot be used as feed water in the boilers unless the cooling water is free from impurities.	Condensate can be reused as feed water as it does not mix with the cooling water.
5.	More auxiliary power required.	Less auxiliary power needed.

3.20.9. Selection of Condenser

The selection of condenser depends on the following conditions :

1. **The first cost.** The first cost of jet condenser may be about one-fourth than that of an equivalent surface condenser.

2. **The maintenance cost.** The maintenance cost of jet condensers is lower than that of surface condensers.

3. **The space available.** The amount of floor space and the head room available are not actually the deciding factors. Surface condensers require more floor space than the jet condensers.

4. **The quantity of cooling water.** The quantity, quality and source of cooling water are the important considerations in the selection of the condenser where the water supply is limited, an artificial cooling system can be installed. For artificial cooling the cooling ponds and cooling towers are used.

5. **The type of boiler feed-water available.** This is an important factor in the selection of condenser. The surface condenser recovers the distilled condensate for boiler feed-water whereas in case of jet-condenser it is not.

3.20.10. Sources of Air in Condensers

The main sources of air found in condensers are given below :

1. There is a leakage of air from atmosphere at the joints of the parts which are internally under a pressure less than that of atmosphere. The quantity of air that leaks in can be reduced to a great extent if design and making of the vacuum joints are undertaken carefully.
2. Air is also accompanied with steam from the boiler into which it enters dissolved in feed water. The quantity of air depends upon the treatment which the feed water receives before it enters the boiler. However, the amount of air which enters through this source is relatively small.
3. In jet condensers, a little quantity of air accompanies the injection water (in which it is dissolved).

Note. (*i*) In jet condensers, the quantity of air dissolved in injection water is about 0.5 kg/10000 kg of water.

(*ii*) In surface condensers of reciprocating steam engines, the air leakage is about 15 kg/10000 kg of steam whereas in surface condensers of well designed and properly maintained steam turbine plants the air leakage is about 5 kg/10000 kg of steam.

3.20.11. Effects of Air Leakage in Condenser

The following are the effects of air leakage in a condenser :

1. **Lowered thermal efficiency.** The leakage air in the condenser results in increased back-pressure on the primemover which means there is loss of heat drop and consequently thermal efficiency of steam power plant is lowered.

2. **Increased requirement of cooling water.** The leaked air in the condenser lowers the partial pressure of steam which means a lowered saturation temperature of steam. As the saturation temperature of steam lowers, its latent heat increases. So it will require increased amount of cooling water for increased latent heat.

3. **Reduced heat transfer.** Air has poor thermal conductivity. Hence leaked air reduces the rate of heat transfer from the vapour, and consequently it requires surface of the tubes of a surface condenser to be increased for a given condenser capacity.

4. **Corrosion.** The presence of air in the condenser increases the corrosive action.

3.20.12. Method for Obtaining Maximum Vacuum in Condensers

Following are some of the methods used to obtain *maximum possible vacuum in condensers used in modern steam power plants.*

1. **Air pump.** Air pumps are provided to maintain a desired vacuum in the condenser by extracting the air and other non-condensable gases. They are usually classified as : (*a*) ***Wet air pumps*** which remove a *mixture of condensate and non-condensable gases.* (*b*) ***Dry air pump*** which removes the *air only.*

2. **Steam air ejector.** When a wet air pump (also called extraction pump) is employed then use is made of steam air ejectors to remove air from the mixture. The operation of the ejector consists in utilising the viscous drag of a high velocity steam jet for the ejection of air and other non-condensable gases from a chamber ; it is chiefly used for exhausting the air from steam condensers. In the case of ejectors used for steam plants where a high vacuum pressure is maintained in the condenser, it is necessary to use two, or perhaps three ejectors in series to obtain maximum vacuum.

3. **De-aerated feed water.** The de-aeration of feed water helps both in maintaining better vacuum in the condenser and controlling corrosion of the steel shell and piping of the steam power plant.

4. **Air tight joints.** The various joints of the steam power plant are rendered air-tight by suitable packing materials etc. at the joints of piping etc. and these are maintained as such by proper inspection from time to time.

3.20.13. Vacuum Measurement

The term *vacuum* in case of a condenser means *pressure below atmospheric pressure.* It is generally expressed in mm of mercury. The vacuum is measured by means of a vacuum gauge (Fig. 3.125). Usually for calculation purpose the vacuum gauge reading is corrected to standard barometric reading 760 mm as follows.

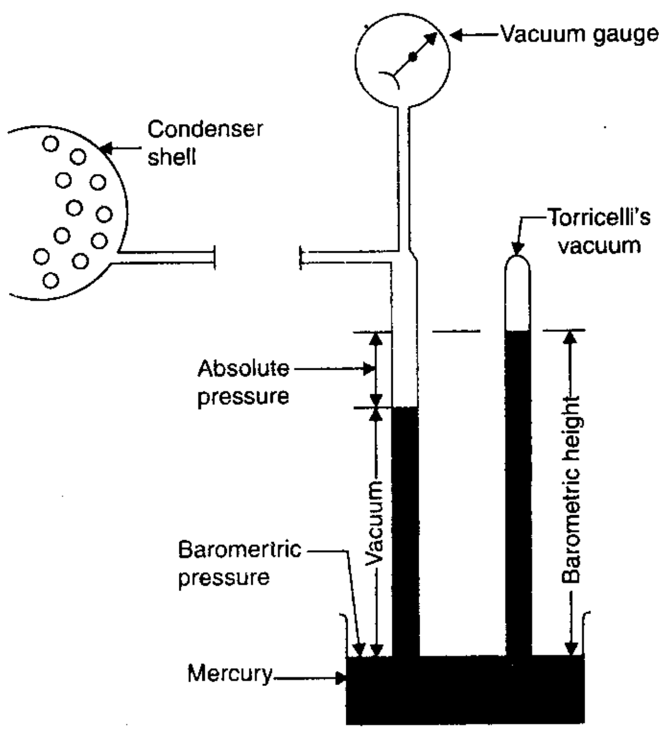

Fig. 3.125. Measurement of vacuum.

Corrected vacuum in mm of Hg = (760 – Absolute pressure in mm of Hg)
= 760 – (Actual barometric height – Actual vacuum)

Also, 760 mm of Hg = 1.01325 bar

$$\therefore \quad \text{mm of Hg} = \frac{1.01325}{760} = 0.001333 \text{ bar}$$

3.20.14. Vacuum Efficiency

It is defined as the ratio of the actual vacuum to the maximum obtainable vacuum. The latter vacuum is obtained when there is only steam and no air is present in the condenser.

$$\text{Vacuum efficiency} = \frac{\text{Actual vacuum}}{\text{Maximum obtainable vacuum}}$$

$$= \frac{\text{Actual vacuum}}{\text{Barometer pressure} - \text{Absolute pressure of steam}} \quad ...(3.73)$$

Note. In case of the absolute pressure of steam corresponding to the temperature of condensate being equal to the absolute pressure in the condenser, the efficiency would be 100%. Actually some quantity of air is also present in the condenser which may leak in and be accompanied by the entering steam. The vacuum efficiency, therefore, depends on the amount of air removed by the air pump from the condenser.

3.20.15. Condenser Efficiency

It is defined *as the ratio of the difference between the outlet and inlet temperatures of cooling water to the difference between the temperature corresponding to the vacuum in the condenser and inlet temperature of cooling water, i.e.,*

$$\text{Condenser efficiency} = \frac{\text{Rise in temperature of cooling water}}{\begin{bmatrix}\text{Temp. corresponding to vacuum}\\ \text{in the condenser}\end{bmatrix} - \begin{bmatrix}\text{Inlet temp. of}\\ \text{cooling water}\end{bmatrix}}$$

or

$$= \frac{\text{Rise in temperature of cooling water}}{\begin{bmatrix}\text{Temp. corresponding to the absolute}\\ \text{pressure in the condenser}\end{bmatrix} - \begin{bmatrix}\text{Inlet temp. of}\\ \text{cooling water}\end{bmatrix}} \quad ...(3.74)$$

3.20.16. Determination of Mass of Cooling Water

Let m_w = Mass of cooling water required in kg/h,

m_s = Mass of steam condensed in kg/h,

t_s = Saturation temperature of steam corresponding to the condenser vacuum in °C,

t_c = Temperature of the condensate leaving the condenser,

t_{w_1} = Temperature of cooling water at inlet in °C,

t_{w_2} = Temperature of cooling water at outlet in °C,

c_{pw} = Specific heat of water at constant pressure,

x = Dryness fraction of steam entering the condenser, and

h_{fg} = Latent heat of 1 kg of steam entering the condenser.

Now, heat lost by steam $= m_s[xh_{fg_2} + c_{pw}(t_s - t_c)]$ kJ/kg

and heat gained by water $= m_w \times c_{pw}(t_{w_2} - t_{w_1})$

If all heat lost by steam is gained by cooling water, then

$$m_s[x h_{f_g} + c_{pw}(t_s - t_c)] = m_w \times c_{pw}(t_{w_2} - t_{w_1})$$

$$\therefore \quad m_w = \frac{m_s\,[xh_{fg} + c_{pw}(t_s - t_c)]}{c_{pw}(t_{w_2} - t_{w_1})} \text{ kg/h} \quad ...(3.75)$$

Eqn. (3.75) applies to *surface condenser only.*

In a jet condenser, since cooling water and steam mix together, therefore the condensate temperature will be same as that of outlet temperature of cooling water (*i.e.*, $t_c = t_{w_2}$). Thus quantity of cooling water, m_w in case of jet condenser is found by the following expression :

$$m_w = \frac{m_s\,[xh_{fg} + c_{pw}(t_s - t_{w_2})]}{c_{pw}(t_{w_2} - t_{w_1})} \text{ kg/h} \quad ...(3.76)$$

[In MKS unit, the value of c_{pw} is taken as unity]

3.20.17. Heat Transmission through Walls of Tubes of a Surface condenser

In case of surface condenser, *the rate of heat transmission varies approximately with square root of the water velocity in tubes.* It thus follows that an increase of heat flow could be obtained by increasing the velocity of flow of water ; but this, in turn, would require a larger amount of energy to circulate the water on account of the corresponding increase in resistance. The following formula is sometimes used for calculating the rate of heat transmission through the walls of the tubes.

Let m_s = Mass of steam used in kg/h,

h = Total heat of 1 kg of steam entering the condenser,

t_m = Mean temperature difference causing heat flow across the tube surface in °C,

t_{w_1} = Temperature of entering cooling water in °C,

t_{w_2} = Temperature of leaving cooling water in °C,

t_s = Temperature of entering steam in °C,

t_c = Temperature of condensate when leaving in °C,

h_{fc} = Total heat of condensate when leaving,

A = Total surface area of condenser tubes in m^2, and

K = Heat transmission coefficient.

(The value of K must be obtained experimentally for the tubes used and for the cooling water velocity in the tubes ; it is a function of both of these factors).

Then,
$$K = \frac{m_s (h - h_{fc})}{t_w A} \quad \text{...(3.77)}$$

where
$$t_m = \frac{(t_c - t_{w_1}) - (t_s - t_{w_2})}{\log_e \dfrac{t_c - t_{w_1}}{t_s - t_{w_2}}} \quad \text{...(3.78)}$$

This equation is due to Grashof and gives approximate result only. It does not hold for all types of surface condensers and modifications of the equation have been made to suite particular types. Eqn. (3.78) may be applied to the *contra-flow* conditions.

If the pressure drop in the condenser is nil, $t_s = t_c$

then eqn. (3.78) may be written as
$$t_m = -\frac{t_{w_2} - t_{w_1}}{\log_e \dfrac{(t_s - t_{w_1})}{(t_s - t_{w_2})}} \quad \text{...(3.79)}$$

For a *cross-flow* condition,
$$t_m = \frac{(t_{w_2} - t_{w_1})}{\log_e \left[\dfrac{d}{d - \left(\dfrac{t_{w_2} - t_{w_1}}{t_s - t_c} \right) \log_e \left(\dfrac{t_s - t_{w_1}}{t_s - t_{w_2}} \right)} \right]} \quad \text{...(3.80)}$$

where 'd' is diameter of condenser tubes.

If the value of the heat transmission coefficient K is known and the value of t_m obtained from Eqns. (3.78), (3.79), or (3.80), the necessary area of the heating surface of the tubes can now be obtained from Eqn. (3.77).

3.20.18. Methods of Cleaning Condenser Tubes

In surface condensers, leaky tube ends and fouling of tubes both inside and outside may give troubles. These troubles are indicated by gradually falling vacuum. This could be offset by increasing the speed of the air as well as increasing the speed of circulating pumps. However, *for better efficiency the tubes must be cleaned.* The cleaning should be thorough.

The inside of the condenser may be usually cleaned with a hose pipe using ordinary city supply of water. A nozzle is fitted to the hose end so that the jet of water may penetrate deep into the condenser. If the water fails to clean out the tubes, soft rubber plugs may be shot through the tubes under air or water pressure. Fig. 3.126 shows the various types of plugs used for cleaning of the condenser tubes.

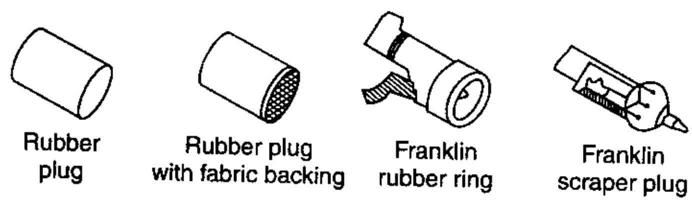

Fig. 3.126. Plugs used for cleaning condenser tubes.

Where water is moderately poor or muddy a built-in tube cleaning equipment may also be used.

Once the life of condenser tube (average 5 years and 15 years in exceptional cases) is finished, the tubes must be replaced.

3.21. COOLING PONDS AND COOLING TOWERS

3.21.1. Introduction

Even in today's many efficient steam plants at least half the energy input must be rejected as unavailable in the steam-turbine exhaust. *The main steam condenser performs the dual function of removing this rejected energy from the plant cycle and keeping the turbine back pressure at the lowest possible level.* The rejected energy inevitably must be returned to the atmosphere. The main condenser does this by transferring the latent heat of the exhaust steam to water exposed to the atmosphere. This water we call circulating, or cooling, water. The cooling water requirement in an open system is about 50 times the flow of steam to the condenser. Even with closed cooling system using cooling towers, the requirement for cooling water is also considerably large as 5 to 8 kg/kWh. This means a 1000 MW station will require about 100 thousand tons of circulating water per day even with the use of cooling towers. Thus the source of cooling water chosen should be able to supply the required quantity of cooling water. The cooling water supply is made by the following sources :

1. River or sea.
2. Cooling ponds.
3. Spray ponds.
4. Cooling towers.

3.21.2. River or Sea

When the power station situated on the bank of a river or lake or other natural source of water, the condenser water supply can be obtained directly from there, the water being discharged to the waste or back to the source (Fig. 3.127).

However, it is not always possible to locate central stations near the source of supply and it becomes necessary to use the same water over and over again by cooling it in a suitable device. Such devices as commonly employed in power stations are the *cooling ponds* and *cooling towers*. Sometimes a combination of natural supply and artificial cooling may be adopted.

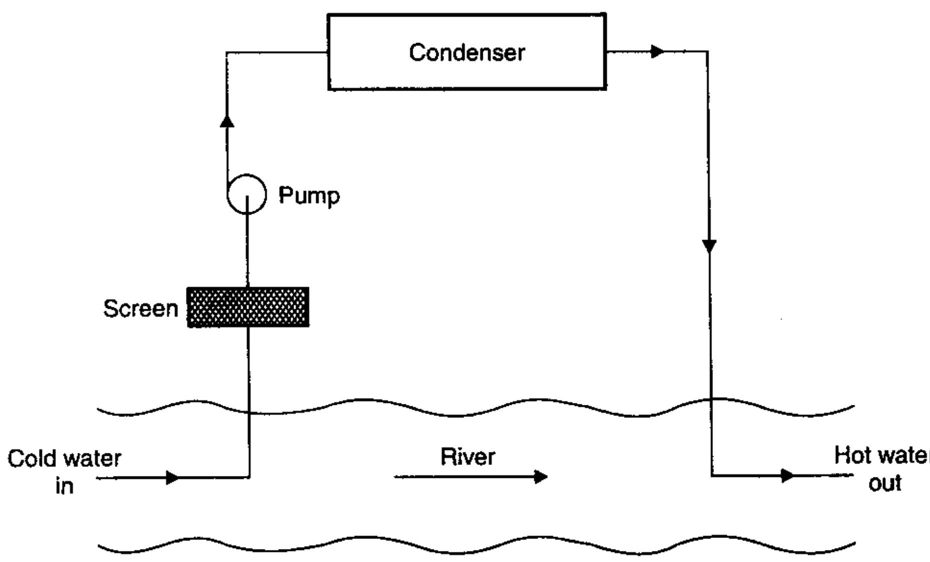

Fig. 3.127. River water cooling system.

3.21.3. Cooling Ponds

The cooling ponds are the simplest of the device for recooling of the cooling water. A cooling pond consists of a large, shallow pool into which the hot water is allowed to come in contact with the atmospheric air. Fig. 3.128 shows a *non-directed flow* naturally cooling pond and Fig. 3.129 illustrates a *directed flow* natural cooling pond. Its cooling effect is more than the non-directed flow pond as water comes in contact with air at lower temperature.

In a cooling pond no spray or other cooling device is employed thus keeping the operating expenses quite low. The cooling pond is suitable where sufficient supply of circulating water is not available. In case the amount of cooling water to be cooled is very large, as in the case of big power plants, the cooling ponds will have to be made very large thus making it prohibitive. For large power plants therefore, spray ponds should be used. The depth of cooling pond should be around one metre.

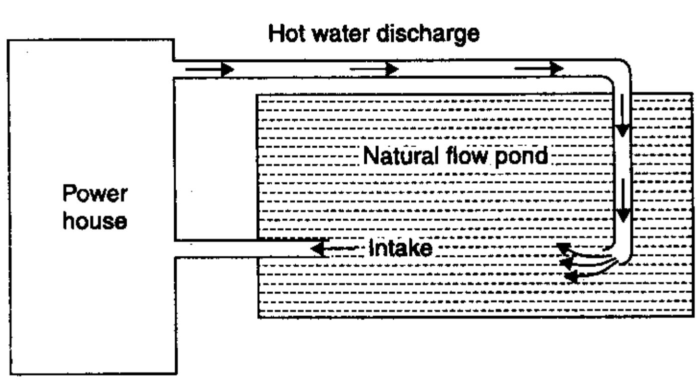

Fig. 3.128. Non-directed flow natural cooling pond.

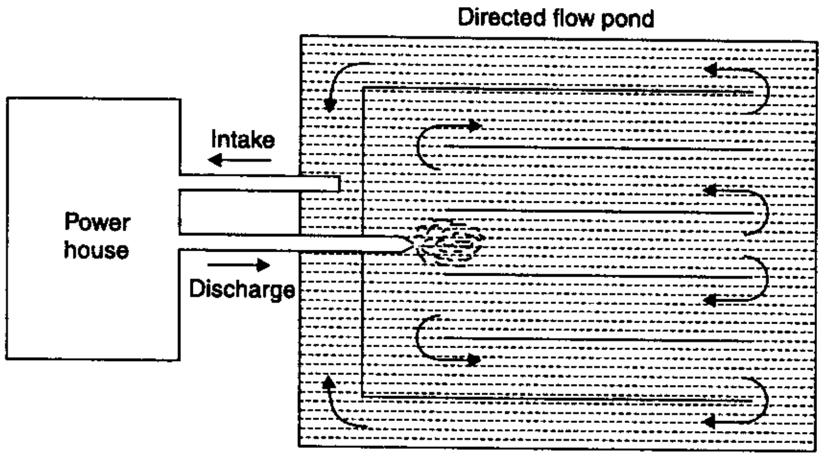

Fig. 3.129. Directed flow natural cooling pond.

3.21.4. Spray Ponds

Refer Fig. 3.130. In this system warm water received from the condenser is sprayed through the nozzles over a pond of large area and cooling effect is mainly due to *evaporation* from the surface of water. In this system sufficient amount of water is lost by evaporation and windage.

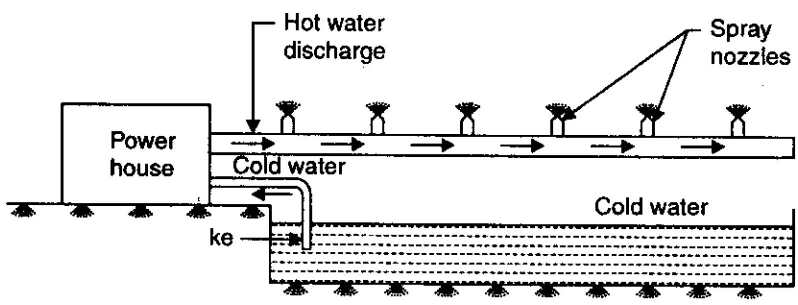

Fig. 3.130. Spray pond.

The spacing of the nozzles in a spray pond depends upon the design and size of the nozzles. Centrifugal nozzles of 50 mm size are usually spaced about 3 metres from centre to centre but the nozzles of large size may be set proportionally farther apart. Nozzles may be mounted in groups of four or five.

Disadvantages of cooling and spray ponds

1. A considerably large area required for cooling.
2. High spray losses (due to evaporation and windage).
3. No control over the temperature of cooled water.
4. Low cooling efficiency (as compared with cooling tower).

3.21.5. Cooling Towers

In power plants the hot water from condenser is cooled in cooling tower, so that it can be reused in condenser for condensation of steam. *In a cooling tower water is made to trickle down drop by drop so that it comes in contact with the air moving in the opposite direction. As a result of this some water is evaporated and is taken away with air. In evaporation the heat is taken away from the bulk of water, which is thus cooled.*

Factors affecting cooling of water in a cooling tower are :

1. Temperature of air.
2. Humidity of air.
3. Temperature of hot air.
4. Size and height of tower.
5. Velocity of air entering tower.
6. Accessibility of air to all parts of tower.
7. Degree of uniformity in descending water.
8. Arrangement of plates in tower.

Cooling towers may be *classified*, according to the *material* of which these are made, as follows :

(*a*) Timber,

(*b*) Concrete (Ferro-concrete, multideck concrete hyperbolic) and

(*c*) Steel duct type.

(*a*) **Timber towers.** Timber towers are rarely used due to the following disadvantages :

(*i*) Due to exposure to sun, wind, water, etc., timber rots easily.

(*ii*) Short life.

(*iii*) High maintenance charges.

(*iv*) The design generally does not facilitate proper circulation of air.

(*v*) Limited cooling capacity.

(*b*) **Concrete towers.** The concrete towers possess the following *advantages* :

(*i*) Large capacity sometimes of the order of 5×10^3 m^3/h.

(*ii*) Improved draft and air circulation.

(*iii*) Increased stability under air pressure.

(*iv*) Low maintenance.

(*c*) **Steel duct type.** Duct type cooling towers are rarely in case of modern power plants owing to their small capacity.

The cooling towers may also be *classified* as follows :

1. Natural draught cooling towers
2. Mechanical draught cooling towers :
   (*i*) Forced draught cooling towers (*ii*) Induced draught cooling towers.

1. **Natural draught cooling tower.** In this type of tower, the hot water from the condenser is pumped to the troughs and nozzles situated near the bottom. Troughs spray the water falls in the form of droplets into a pond situated at the bottom of the tower. The air enters the cooling tower from air openings provided near the base, rises upward and takes up the heat of falling water. A concrete hyperbolic cooling tower is shown in Fig. 3.131. This tower has the following *advantages* over mechanical towers :

(*i*) Low operating and maintenance cost.

(*ii*) It gives more or less trouble free operation.

(*iii*) Considerable less ground area required.

(*iv*) The towers may be as high as 125 m and 100 m in diameter at the base with the capability of withstanding winds of very high speed. These structures are more or less self-supported structures.

(*v*) The enlarged top of the tower allows water to fall out of suspension.

The main *drawbacks* of this tower are listed below :

(*i*) High initial cost.

(*ii*) Its performance varies with the seasonal changes in DBT (dry bulb temperature) and R.H. (relative humidity) of air.

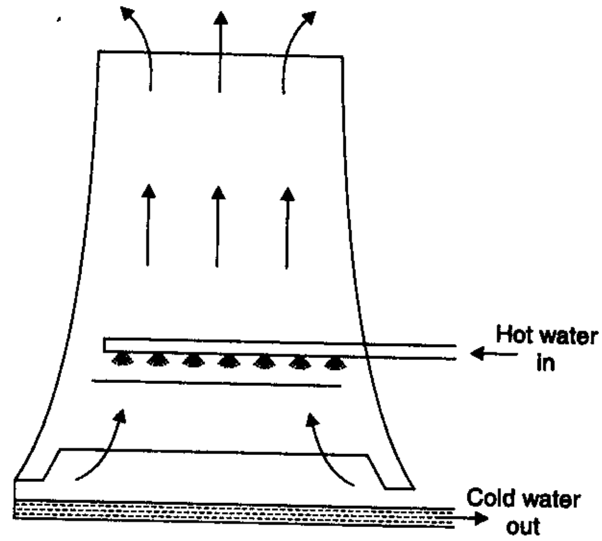

Fig. 3.131. Hyperbolic cooling tower.

While initial cost may be higher, the saving in fan power, longer life and less maintenance always favour for this type of tower. It is also more favourable over mechanical draught cooling towers as central station size increases.

2. **Mechanical draught cooling towers.** In these towers the draught of air for cooling the tower is produced mechanically by means of *propeller fans*. These towers are usually built in cells or units, the capacity depending upon the number of cells used.

Fig. 3.132 shows a *forced draught cooling tower.* It is similar to natural draught tower as far as interior construction is concerned, but the sides of the tower are closed and form an air and water tight structure, except for fan openings at the base for the inlet of fresh air, and the outlet at the top for the exit of air and vapours. There are hoods at the base projecting from the main portion of the tower where the fans are placed for forcing the air, into the tower.

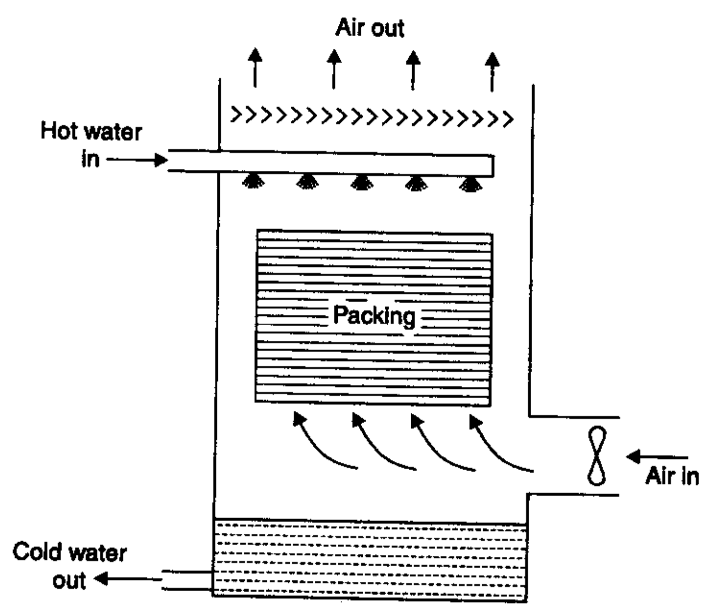

Fig. 3.132. Forced draught cooling tower.

Fig. 3.133 shows an *induced draught cooling tower*. In these towers, the fans are placed at the top of the tower and they draw the air in through louvers extending all around the tower at its base.

Comparison of forced and induced draught towers

Forced Draught Cooling Towers

Advantages :

1. More efficient (than induced draught).
2. No problem of fan blade erosion (as it handles dry air only).
3. More safe.
4. The vibration and noise are minimum.

Disadvantages :

1. The fan size is limited to 4 metres.
2. Power requirement high (approximately double that of induced draught system for the same capacity).
3. In the cold weather, ice is formed on nearby equipments and buildings or in the fan housing itself. The frost in the fan outlet can break the fan blades.

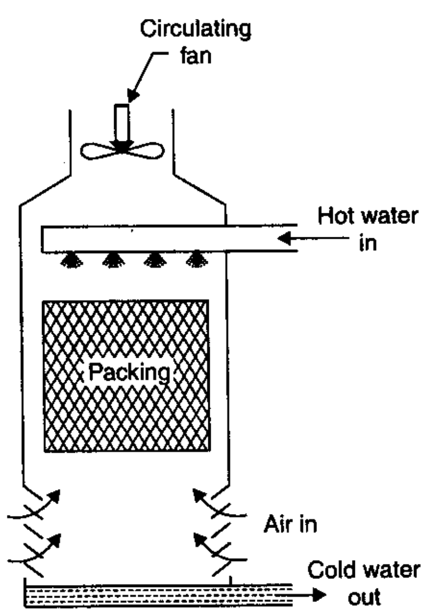

Fig. 3.133. Induced draught cooling tower.

Induced draught cooling towers

Advantages :

1. The coldest water comes in contact with the driest air and warmest water comes in contact with the most humid air.
2. In this tower, the recirculation is seldom a problem.
3. Lower first cost (due to the reduced pump capacity and smaller length of water pipes).
4. Less space required.
5. This tower is capable of cooling through a wide range.

Disadvantages :

1. The air velocities through the packings are unevenly distributed and it has very little movement near the walls and centre of the tower.

2. Higher H.P. motor is required to drive the fan comparatively. This is due to the fact that the static pressure loss is higher as restricted area at base tends to choke off the flow of higher velocity air.

Comparison between Natural and Mechanical Draught Towers

Mechanical draught towers

Advantages :

1. These towers require a small land area and can be built at most locations.
2. The fans give a good control over the air flow and thus the water temperature.
3. Less costly to install than natural draught towers.

Disadvantages :

1. Fan power requirements and maintenance costs make them more expensive to operate.
2. Local fogging and icing may occur in winter season.

3.21.6. Dry Cooling Towers

The necessity of using dry cooling towers has been felt due to the following reasons :

(*i*) *Natural water resources* can be used for dissipating heat in power plants but there are some limitations to do so, as discussed earlier. Moreover there is *potential thermal pollution problems in this system.*

(*ii*) In *evaporative cooling system* (cooling towers) due to continuous evaporation of water, the concentration of impurities goes on increasing and so the disposal of tower blow down may become a serious problem at some sites. Besides this make up water for tower use is limited in some areas.

Thus the use of dry cooling towers in power plants in the only alternative. The *dry system rejects the heat directly to the atmosphere which is the largest heat sink available.*

Following are two basic types of closed cooling water systems for power plants utilising dry cooling principle.

1. Direct system
2. Indirect system.

1. Direct System. Refer Fig. 3.134. In direct system the exhaust steam from the turbine is passed on to the cooling coils where it is condensed directly by *means of air*. Cooling coils constitute air cooled condenser. This system has the disadvantage that *large ducts are needed to convey the exhaust steam to the exchanger coils, to minimise the pressure drop.* This places a practical limit on the size of the unit. This system is limited to maximum generator unit sizes of 200 MW to 300 MW.

2. Indirect System. Refer Fig. 3.135. In this case, the exhaust steam is condensed in a spray condenser by means of circulating water. The major part of the water from the condenser flows back to the cooling coils and an amount equal to the exhaust steam from the turbine is directed back to the boiler feed water circuit. Some energy recovery is possible by using hydraulic turbine (*T*) to drive the circulating water pumps. This system is applicable to all large unit sizes.

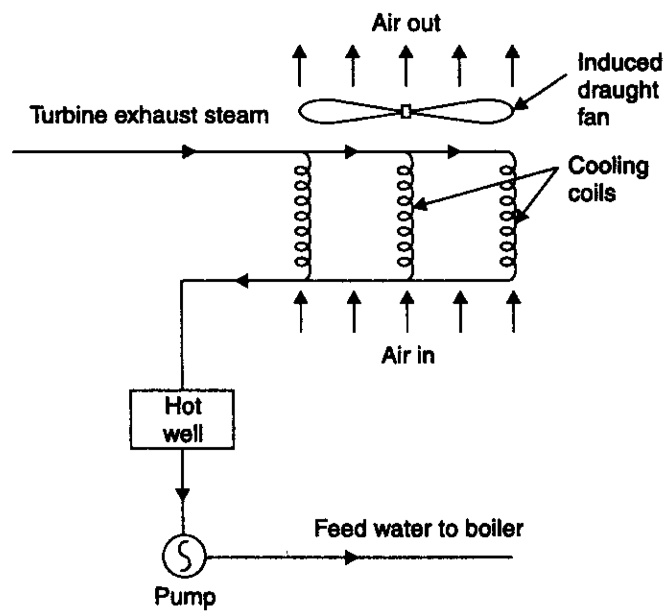

Fig. 3.134. Direct system.

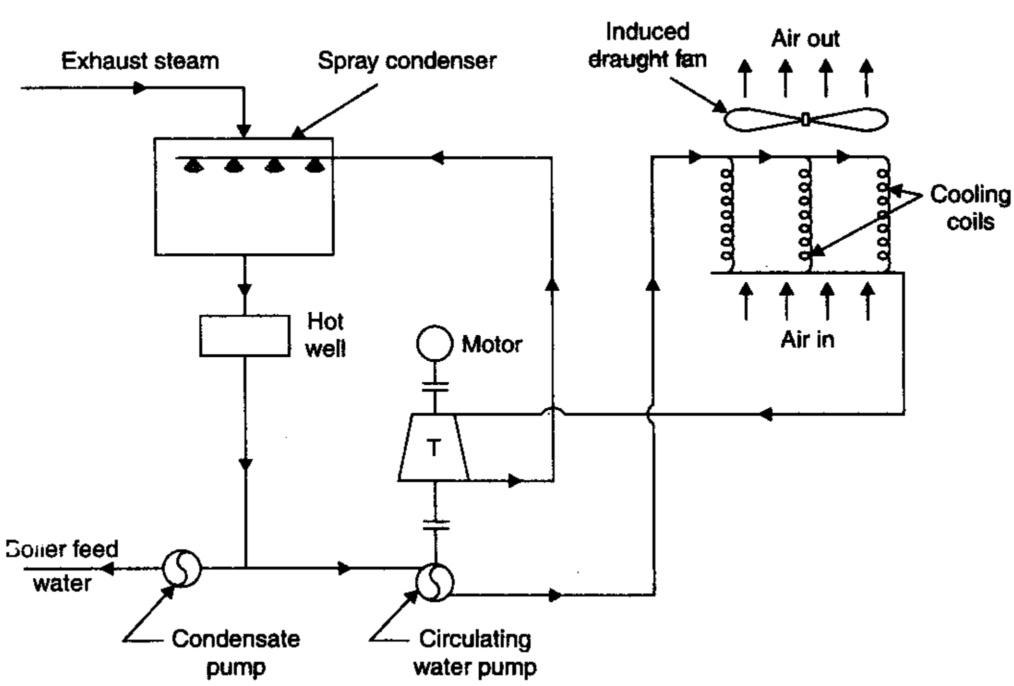

Fig. 3.135. Indirect system.

Advantages of dry cooling system :

1. It solves the problem of thermal pollution of water and as the circulating water does not come into direct contact with the cooling air, there is no evaporative loss of water as in wet type tower.

2. This system eliminates water supply as a significant plant site selection factor. Power plants can be sited closer to the load centre.

3. There is minimum air pollution.

4. From environmental view point, the dry towers offer several desirable features *e.g. no fog, no blowdown treatment, no windage loss of water, no evaporative loss of water* and *no thermal discharge to water source.*

Disadvantages :

1. Their performance is limited by dry bulb temperature and so turbine exhaust temperatures are much higher resulting in a substantial loss of turbine efficiency, most critical in warm climates.

2. Due to low heat transfer co-efficient, dry cooling towers require enormous volumes of air large surface areas and are less effective at high natural air temperatures.

3.21.7. Maintenance of Cooling Towers

The regular maintenance of cooling towers is very essential to achieve the desired cooling and to reduce the depreciation costs. The maintenance of cooling towers includes the following :

1. The fans, motors housings etc. should be inspected from time to time.

2. Motor bearings should be greased and gear boxes oiled. Any unusual noise or vibrations in them should be corrected immediately.

3. At least once in a year motor's gear boxes should be checked for structural weakness.

4. The circulating water should be tested for hardness and should be kept free from impurities to avoid scale formations and to avoid corrosive action of water.

5. The water spraying nozzles should be inspected regularly for clogging.

Example 3.24. *In a surface condenser a section of the tubes near to the air pump suction is screened off so that the air is cooled to a temperature below that of the condensate, separate extraction pumps being provided to deal with air and condensate respectively. 5448 kg of steam are condensed per hour and the air leakage is 4.54 kg/h. The temperature of the exhaust steam is 31°C, the temperature of the condensate is 27°C, and the temperature at the air pump suction is 21.1°C. Assuming a constant vacuum throughout the condenser, find :*

(*i*) *The mass of steam condensed per hour in the air cooler ;*

(*ii*) *The volume of air in m³/h to be dealt with by the air pump ;*

(*iii*) *The percentage reduction in necessary air pump capacity following the cooling of the air.*

Solution. From steam tables, pressure of steam at 31°C,

$$p_s = 0.045 \text{ bar abs.}$$

and partial pressure of steam at air pump suction at 21.1°C,

$$p_s = 0.025 \text{ bar abs.}$$

Partial pressure of air, $p_a = 0.045 - 0.025 = 0.02$ bar abs.

Also, at air pump suction ; $T = 21.1 + 273 = 294.1$ K

(*i*) Using characteristic equation of gas for 4.54 kg of air,

$$p_a V = mRT$$

$$V = \frac{mRT}{10^4 p_a} = \frac{4.54 \times 287 \times 294.1}{0.02 \times 10^5} = 191.6 \text{ m}^3/\text{h}.$$

As per Dalton's law this is also the volume of the steam mixed with air at the air pump suction.

Specific volume of steam at partial pressure of 0.025 bar = 54.25 m³/kg

$\therefore$ *Mass of steam at air pump suction / h* $= \frac{191.6}{54.25}$ **= 3.53 kg. (Ans.)**

(*ii*) **Volume of air/h to be dealt = 191.6 m³/h. (Ans.)**

(*iii*) Temperature at the pump *without air cooling*,

$$T = 27 + 273 = 300 \text{ K}.$$

Partial pressure of steam at 27°C = 0.0357 bar (From steam tables).

Then, partial pressure of air = 0.045 – 0.0357 bar = 0.0093 bar

Again, using the characteristic gas equation to 4.54 kg of air at partial pressure of 0.0093 bar,

$$V = \frac{mRT}{p_a} = \frac{4.54 \times 287 \times 300}{0.0093 \times 10^5} = 420.3 \text{ m}^3/\text{h}.$$

Percentage reduction in air pump capacity due to air cooling

$$= \frac{420.3 - 191.6}{420.3} = \mathbf{0.544 \text{ or } 54.4\%}. \quad \textbf{(Ans.)}$$

☞ **Example 3.25.** *A surface condenser deals with 13625 kg of steam per hour at a pressure of 0.09 bar. The steam enters 0.85 dry and the temperature at the condensate and air extraction pipes is 36°C. The air leakage amounts to 7.26 kg / hour. Determine (i) the surface required if the average heat transmission rate is 3.97 kJ / cm² per second ; (ii) the cylinder diameter for the dry air pump, if it is to be single-acting at 60 r.p.m. with a stroke to bore ratio of 1.25 and volumetric efficiency of 0.85.*

Solution. Corresponding to 0.09 bar (from steam tables)

$$h_f = 183.3 \text{ kJ/kg}, \; h_{fg} = 2397.7 \text{ kJ/kg}$$

$$\text{Heat extracted/sec. from steam} = \frac{13625}{3600} [(h_f + xh_{fg}) - h_{f_1}]$$

$$= \frac{13625}{3600} [(183.3 + 0.85 \times 2397.7) - 4.184 \times 36]$$

$$= 3.785 [(183.3 + 2038) - 150.62]$$

$$= 7837.5 \text{ kJ/s}.$$

(*i*) **Surface required** $= \frac{7837.5}{3.97}$ **= 1974.18 cm². (Ans.)**

(*ii*) Quantity of air leakage/min. into the condenser $= \frac{7.26}{60} = 0.121$ kg

Partial pressure of steam at 36°C (from steam tables)

$$p_s = 0.0595 \text{ bar}$$

But,
$$p = p_a + p_s = 0.09 \text{ bar}$$

$\therefore$
$$p_a = 0.09 - 0.0595 = 0.0305 \text{ bar}$$

Using characteristic gas equation,

$$p_a V = mRT$$

$$0.0305 \times 10^5 \times V = 0.121 \times 287 \times (273 + 36)$$

or
$$V = \frac{0.121 \times 287 \times 309}{0.0305 \times 10^5} = 3.52 \text{ m}^3$$

$$\text{Capacity of air pump/min} = \frac{3.52}{0.85} = 4.14 \text{ m}^3 \text{ or } 4.14 \times 10^6 \text{ cm}^3$$

$$\text{Capacity of the pump/stroke} = \frac{4.14 \times 10^6}{60} = 69000 \text{ cm}^3$$

Also $\pi/4 \times d^2 \times 1.25 \times d = 69000$

where d is the diameter of cylinder for dry air pump

or $$d^3 = \frac{69000 \times 4}{\pi \times 1.25} = 70282.8$$

or $$d \simeq \mathbf{41.3\ cm.} \quad \textbf{(Ans.)}$$

☞ **Example 3.26.** *A surface condenser deals with 13,000 kg of steam per hour. The leakage air in the system amounts to 1 kg per 2700 kg of steam. The vacuum in the air pump suction is 705 mm of mercury (barometer 760 mm of Hg) and temperature is 34.6°C.*

Determine the discharging capacity of the wet air pump which removes both air and condensate in m³ per minute, taking the volumetric efficiency of the pump as 90%.

If the air pump is single-acting and runs at 60 r.p.m. and piston stroke is 1.25 times the diameter of the pump, find the dimensions of the wet air pump.

Solution. From steam tables, corresponding to 34.6°C,

$$p_s = 0.055 \text{ bar}$$

The combined pressure of steam and air in the condenser,

$$p = 760 - 705 = 55 \text{ mm of Hg} = 55 \times 0.001333 = 0.0733 \text{ bar}$$

Partial pressure of air, $$p_a = p - p_s$$
$$= 0.0733 - 0.055 = 0.0183 \text{ bar}$$

$$\left(\because \ 1 \text{ mm of Hg} = \frac{1.01325}{760} = 0.001333 \text{ bar}\right)$$

Mass of air leakage in the condenser/min.,

$$m_a = \frac{13000}{2700 \times 60} = 0.0802 \text{ kg.}$$

Volume of air leakage in the condenser/min.,

$$V_a = \frac{m_a R T_a}{p_a} = \frac{0.0802 \times 287 \times (34.6 + 273)}{0.0183 \times 10^5} = 3.869 \text{ m}^3$$

Mass of steam condensed/min., $m_s = \dfrac{13000}{60}$ kg

Volume of condensate/min. $= \dfrac{13000}{60 \times 1000} = 0.2167 \text{ m}^3$ [$\because$ Density of water = 1000 kg/m³]

$\therefore$ Volume of mixture (air + condensate) actually discharged/min.

$$= 3.869 + 0.2167 = 4.0857 \text{ m}^3$$

$\therefore$ Discharging capacity of the air pump/min.

$$= 4.0857 \times \frac{100}{90} = 4.539 \text{ m}^3$$

$\therefore$ Discharging capacity of the air pump/stroke

$$= \frac{4.539}{60} \times 10^6 = 75650 \text{ cm}^3$$

$$\pi/4\, d^2 \times 1.25\, d = 75650 \qquad [d = \text{dia. of the cylinder for the pump}]$$

$$d = \left(\frac{75650 \times 4}{\pi \times 1.25}\right)^{1/3} = \mathbf{42.55\ cm.} \quad \textbf{(Ans.)}$$

Piston stroke, $$l = 1.25\, d = 1.25 \times 42.55 = \mathbf{53.2\ cm.} \quad \textbf{(Ans.)}$$

Example 3.27. *To check the leakage of air in a condenser, the following procedure is adopted. After ruming the plant to reach the steady conditions the steam supply to the condenser and the air and condensate pump are shut down, thus completely isolating the condenser. The temperature and vacuum readings are noted at shut down and also after a period of 10-minutes. They are 39°C and 685 mm Hg and 28°C and 480 mm Hg respectively. The barometer reads 750 mm Hg. The effective volume of the condenser is 1.5 m³. Determine (i) quantity of air leakage into the condenser during the period of observation ; (ii) the quantity of water vapour condensed during the period.*

Solution. At shut down :

From steam tables, corresponding to $t_s = 39°C$:

$$p_s = 0.07 \text{ bar} = \frac{0.07}{0.001333} = 52.5 \text{ mm Hg}$$

and $$v_g = 20.53 \text{ m}^3/\text{kg}$$

The combined pressure of steam and air in the condenser,

$$p = p_a + p_s = 750 - 685 = 65 \text{ mm Hg}$$

$$\therefore \quad p_a = p - p_s = 65 - 52.5 = 12.5 \text{ mm Hg} = 12.5 \times .001333 = 0.0167 \text{ bar}$$

Now, mass of air in 1.5 m³

$$m_a = \frac{p_a V_a}{RT_a} = \frac{0.0167 \times 10^5 \times 1.5}{287 \times (273 + 39)} = 0.028 \text{ kg}$$

and mass of steam in 1.5 m³

$$m_s = \frac{1.5}{20.53} = 0.073 \text{ kg}$$

After 10 minutes, observed duration,

From steam tables, corresponding to $t_s = 28°C$

$$p_s = 0.0378 \text{ bar} = \frac{0.0378}{0.001333} = 28.36 \text{ mm Hg}, \; v_g = 36.69 \text{ m}^3/\text{kg}$$

Total pressure in the condenser,

$$p = 750 - 480 = 270 \text{ mm Hg}$$

$\therefore$ Air pressure, $$p_a = p - p_s = 270 - 28.36 = 241.64 \text{ mm Hg}$$
$$= 241.64 \times 0.001333 = 0.322 \text{ bar}$$

Mass of air, $$m_a = \frac{p_a V_a}{RT_a} = \frac{0.322 \times 10^5 \times 1.5}{287 \times (273 + 28)} = 0.559 \text{ kg.}$$

Mass of steam, $$m_s = \frac{1.5}{36.69} = 0.0408 \text{ kg}$$

$\therefore$ **Air leakage** in 10 minutes period

$$= (0.559 - 0.028) = \mathbf{0.531 \text{ kg.}} \quad \textbf{(Ans.)}$$

and **steam condensed** in 10 minutes period

$$= (0.073 - 0.0408) = \mathbf{0.0322 \text{ kg.}} \quad \textbf{(Ans.)}$$

Example 3.28. *A jet condenser is required to condenser 5000 kg of steam per hour. 350 m³ of injection water are used per hour. Initial temperature of the cooling water is 27°C. The volume of air at atmospheric pressure dissolved in injection water is 5% of the volume of water. The amount of air entering the condenser with steam is 1 kg for every 3500 kg of steam. The vacuum in the air pump suction is 686 mm. When the barometer records 760 mm and the temperature of the condensate is 34.6°C, determine the suction capacity of the air-pump in m³/min to remove air and water from the condenser. Assume volumetric efficiency of pump as 85%.*

Solution. Total pressure in the condenser,

$$p = p_a + p_s = (760 - 686) = 74 \text{ mm Hg}$$

$$74 \times 0.001333 = 0.0986 \text{ bar}$$

Partial pressure of steam corresponding to 34.6°C = 0.055 bar (From steam tables)

$\therefore$ Partial pressure of air, $p_a = p - p_s$

$$= 0.0986 - 0.055 = 0.0436 \text{ bar}$$

Mass of air entering per minute with steam $= \dfrac{5{,}000}{3500 \times 60} = 0.0238$ kg

Volume of air entering per minute with injection water $= \dfrac{350 \times 5}{100 \times 60} = 0.292$ m^3.

Mass of this volume of air $= \dfrac{p_a V_a}{RT_a}$

$$= \frac{1.01325 \times 10^5 \times 0.292}{287 \times (273 + 27)} = 0.343 \text{ kg.}$$

Total weight of air entering the condenser per minute

$$= 0.0238 + 0.343 = 0.3668 \text{ kg}$$

Now, volume of this air at 0.045 bar and 34.6°C,

$$V = \frac{m_a R T_a}{p_a} = \frac{0.3668 \times 287 \times (273 + 34.6)}{0.0436 \times 10^5} = 7.427 \text{ m}^3$$

Volume of condensate/min $= \dfrac{5000}{60 \times 1000} = 0.083$ m^3

Volume of injected water/min. $= \dfrac{350}{60} = 5.83$ m^3

Hence, total volume to be handled $= 7.427 + 0.083 + 5.83 = 13.34$ m^3

$\therefore$ Suction capacity of the air pump $= \dfrac{13.34}{0.85} =$ **15.69 m^3. (Ans.)**

Example 3.29. *In a condenser air pump and water pump are separately installed. Steam enters the condenser at 41.5°C and the condensate is removed at 37.6°C. The quantity of air infiltrating into the condenser through various zones is 6 kg/h. Determine :*

(i) The volume of air handled by the air pump.

(ii) The quantity handled by a combined air and condensate pump at 39°C.

Make suitable assumptions and list all such assumptions.

Solution. Pressure of steam corresponding to 41.5°C = 0.08 bar.

Air pump suction point

Temperature = 37.6°C

Pressure (p_s) corresponding to 37.6°C = 0.065 bar

$\therefore$ Partial pressure of air = 0.08 – 0.065 = 0.015 bar

$\therefore$ *Volume of air,* $V_a = \dfrac{mRT}{p_a} = \dfrac{6 \times 0.287 \times (273 + 37.6) \times 1000}{0.015 \times 10^5}$

$= $ **356.57 m^3/h. (Ans.)**

Assumptions :

1. Pressure inside the condenser is uniform.
2. Air is removed at the same temperature as that of condensate.
3. The pressure due to air at the entry of steam is neglected.

If condensate and air are to be removed by the same pump, the air is to be removed at 39°C.

Partial pressure of steam at 39°C = 0.07 bar

∴ Partial pressure of air = 0.08 – 0.07 = 0.01 bar

$$\therefore \quad \textit{Volume of air} = \frac{6 \times 0.287 \times (273 + 39) \times 1000}{0.01 \times 10^5} = \mathbf{537.26\ m^3/h.} \quad \textbf{(Ans.)}$$

Example 3.30. *During a trial on a steam condenser, the following observations were recorded :*

Condenser vacuum	*680 mm kg*
Barometer reading	*764 mm kg*
Mean condenser temperature	*36.2°C*
Hot well temperature	*30°C*
Condensate formed per hour	*1780 kg*
Circulating cooling water inlet temperature	*20°C*
Circulating cooling water outlet temperature	*32°C*
Quantity of cooling water	*1250 kg / min.*

Determine :

(i) Condenser vacuum corrected to standard barometer.
(ii) Vacuum efficiency.
(iii) Undercooling of condensate.
(iv) Condenser efficiency.
(v) Condition of steam as it enters the condenser.
(vi) Mass of air present per kg of condensed steam.

Assume : R for air = *0.287 kJ / kg K*
Specific heat of water = *4.186 kJ / kg K.*

Solution. (*i*) **Condenser vacuum corrected to standard barometer**

= Standard barometric pressure – (barometric pressure – gauge pressure)
= 760 – (764 – 680) = 676 mm of Hg

But 1 mm of Hg = 1.333×10^{-3} bar

∴ 676 mm of Hg = $676 \times 1.333 \times 10^{-3}$ = **0.9011 bar. (Ans.)**

(*ii*) **Vacuum efficiency :**

From steam tables saturation pressure corresponding to 36.2°C = 0.06 bar.

$$\therefore \quad \textit{Vacuum efficiency} = \frac{\text{Condenser vacuum}}{\text{Barometer reading} - \text{Pressure of steam}}$$

$$= \frac{680 \times 1.333 \times 10^{-3}}{764 \times 1.333 \times 10^{-3} - 0.06}$$

= **0.9458 or 94.58%. (Ans.)**

(*iii*) **Undercooling of condensate :**

= Condensate temperature – Hot well temperature
= 36.2 – 30 = **6.2°C. (Ans.)**

(*iv*) **Condenser efficiency :**

Absolute condenser pressure

= Barometric pressure – Vacuum reading

= 764 – 680 = 84 mm of Hg

= $84 \times 1.333 \times 10^{-3} = 0.1119$ bar

Saturation temperature corresponding to 0.1119 bar (from steam tables) ≃ 48°C.

$$\therefore \quad \textit{Condenser efficiency} = \frac{\text{Actual cooling water temperature rise}}{\text{Maximum possible temperature rise}}$$

$$= \frac{32 - 20}{48 - 20} = 0.4286 = \mathbf{42.86\%.} \quad \textbf{(Ans.)}$$

(*v*) **Condition of steam entering the condenser :**

Absolute condenser pressure = 0.1119 bar.

For this pressure, $h_f = 200.1$ kJ/kg, $h_{fg} = 2387.2$ kJ/kg

Enthalpy of condensate corresponding to hot well temperature of 30°C

= 125.75 kJ/kg.

Also, heat lost by steam = heat gained by water

$$m_s \, [(h_f + x \, h_{fg}) - h_{\text{hot-well}}] = m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

$$= \frac{1780}{60} \, [(200.1 + x \times 2387.2) - 125.75] = 1250 \times 4.186 \times (32 - 20)$$

$$(200.1 + x \times 2387.2) - 125.75 = 2116.5$$

Solving, $x = 0.855$

Hence, *condition of steam entering the condenser* = **0.855. (Ans.)**

(*vi*) **Mass of air present :**

Specific volume of steam at mean condensate temperature of 36.2°C

= 23.74 m³/kg

Partial pressure of steam at 36.2°C = 0.06 bar

∴ Partial pressure of air = 0.1119 – 0.06 = 0.0519 bar

∴ *Mass of air present per kg of uncondensed vapour,*

$$m_a = \frac{p_a V}{RT} = \frac{0.0519 \times 10^5 \times 23.74}{0.287 \times 1000 \times (273 + 36.2)} = \mathbf{1.3884 \text{ kg.}} \quad \textbf{(Ans.)}$$

Example 3.31. *A primemover uses 15000 kg of steam per hour and develops 2450 kW. The steam is supplied at 30 bar and 350°C. The exhaust from the primemover is condensed at 725 mm Hg when barometer records 755 mm Hg. The condensate temperature from the condenser is 31°C and the rise of temperature of circulating water is from 8°C to 18°C. Determine : (i) The quality of steam entering the condenser, (ii) The quantity of circulating cooling water and the ratio of cooling.*

Assume that no air is present in the condenser and all mechanical drive losses are negligible.

Solution. Quantity of steam used, $m_s = 15000$ kg/h

Pressure of steam = 30 bar, 350°C

Vacuum reading = 725 mm Hg

Barometer reading = 755 mm Hg

Condensate temperature = 31°C

Rise of temperature of cooling water, $t_{w_2} - t_{w_1} = 18 - 8 = 10°C$.

(*i*) **Quality of steam entering the condenser, x :**

Condenser pressure $= 755 - 725 = 30$ mm Hg

$= 30 \times 1.333 \times 10^{-3} = 0.0399$ bar

From steam table :

At 0.0399 bar : $h_f = 121.5$ kJ/kg, $h_{fg} = 2432.9$ kJ/kg

At 30 bar, 350°C : $h_g = 3115.3$ kJ/kg

The work done in the turbine = 2450 kW or kJ/s ...(*i*)

Enthalpy drop in the turbine/sec.

$$= \frac{15000}{3600}\,[3115.3 - (121.5 + x \times 2432.9)] \text{ kJ/s} \quad ...(ii)$$

Since mechanical drive losses are negligible, the expressions (*i*) and (*ii*) are equal.

$$\therefore \quad 2450 = \frac{15000}{3600}\,[3115.3 - (121.5 + x \times 2432.9)]$$

or
$$\frac{2450 \times 3600}{15000} = 3115.3 - 121.5 - x \times 2432.9$$

$$588 = 3115.3 - 121.5 - 2432.9x$$

$$\therefore \quad x = \frac{3115.3 - 121.5 - 588}{2432.9} = \mathbf{0.988.} \quad \textbf{(Ans.)}$$

(*ii*) **Quantity of circulating water, m_w :**

Ratio of cooling :

Heat lost by condensing steam = Heat gained by cooling water

$$m_s\,[h_f + x\,h_{fg} - h_c] = m_w \times c_{pw} \times (t_{w_2} - t_{w_1})$$

or
$$m_w = \frac{m_s\,[h_f + x\,h_{fg} - h_c]}{c_{pw} \times (t_{w_2} - t_{w_1})}$$

$$= \frac{15000\,[121.5 + 0.988 \times 2432.9 - 4.186 \times 31]}{4.186 \times 10}$$

$$= \mathbf{858375\ kg/h.} \quad \textbf{(Ans.)}$$

[where c_{pw} = specific heat of water = 4.186 kJ/kg K]

$$\therefore \quad \textit{Ratio of cooling} = \frac{m_w}{m_s} = \frac{858375}{15000} = \mathbf{57.225\ kg/kg.} \quad \textbf{(Ans.)}$$

☞ **Example 3.32.** *A surface condenser is required to deal with 20000 kg of steam per hour, and the air leakage is estimated at 0.3 kg per 1000 kg of steam. The steam enters the condenser dry saturated at 38°C. The condensate is extracted at the lowest point of the condenser at a temperature of 36°C. The condensate loss is made up with water at 7°C. It is required to find the saving in condensate and the saving in heat supplied in the boiler, by fitting a separate air extraction pump which draws the air over an air cooler. Assume that the air leaves the cooler at 27°C. The pressure in the condenser can be assumed to remain constant.*

Solution. The mass of air per kg of steam at entry = 0.3/1000 = 0.0003 kg

At 38°C : Saturation pressure = 0.06624 bar and

$$v_g = 21.63 \text{ m}^3/\text{kg (From steam tables)}$$

For 1 kg of steam, the volume is 21.63 m^3, and this must be the volume occupied by 0.0003 kg of air when exerting its partial pressure,

i.e., Partial pressure of air $= \dfrac{m_a R_a T}{V} = \dfrac{0.0003 \times 0.287 \times (273 + 38) \times 10^3}{21.63 \times 10^5} = 1.2 \times 10^{-5}$ bar

This is negligibly small and may be neglected.

Condensate extraction :

At 36°C : Saturation pressure = 0.0594 bar, v_g = 23.97 m^3/kg.

The total pressure in the condenser is 0.06624 bar, hence

$$0.06624 = 0.0594 + p_a$$

$$\therefore \quad p_a = 0.00684 \text{ bar}$$

The mass of air removed per hour is

$$\frac{20000 \times 0.3}{1000} = 6 \text{ kg/h}$$

Hence the volume of air removed per hour is

$$\frac{mRT}{p} = \frac{6 \times 0.287 \times (273 + 36) \times 10^3}{0.00684 \times 10^5} = 778 \text{ m}^3/\text{h}$$

The mass of steam associated with the air removed is therefore given by

$$\frac{778}{23.97} = 32.45 \text{ kg/h.}$$

Separate extraction :

At 27°C : Saturation pressure $= 0.03564$ bar, v_g = 38.81 m^3/kg

The air partial pressure $= 0.06624 - 0.03564 = 0.0306$ bar

$\therefore$ The volume of air removed $= \dfrac{mRT}{p} = \dfrac{6 \times 0.287 \times (273 + 27) \times 10^3}{0.0306 \times 10^5} = 168.9$ m^3/h

$\therefore$ Steam removed $= \dfrac{168.9}{38.81} = 4.35$ kg/h.

Hence, *the saving in condensate by using the separate extraction method*

$= 32.45 - 4.35 =$ **28.1 kg/h. (Ans.)**

The *saving in heat to be supplied in the boiler*

$= 28.1 \times 4.186\ (36 - 7) =$ **3411 kJ/h. (Ans.)**

Example 3.33. *For the data of example 3.32, calculate :*

(*i*) *The percentage reduction in air pump capacity by using the separate extraction method.*

(*ii*) *If the temperature rise of cooling water is 5.5 K, calculate the mass flow of cooling water required.*

Solution. Capacity of air pump without air cooler = 778 m^3/h

Capacity of air pump with air cooler = 168.9 m^3/h.

$\therefore$ *Percentage reduction in capacity* $= \left(\dfrac{778 - 168.9}{778}\right) \times 100 =$ **78.3%. (Ans.)**

Fig. 3.136 shows the system to be analysed. Let suffixes *s*, *a*, and *c* denote steam, air and condensate respectively.

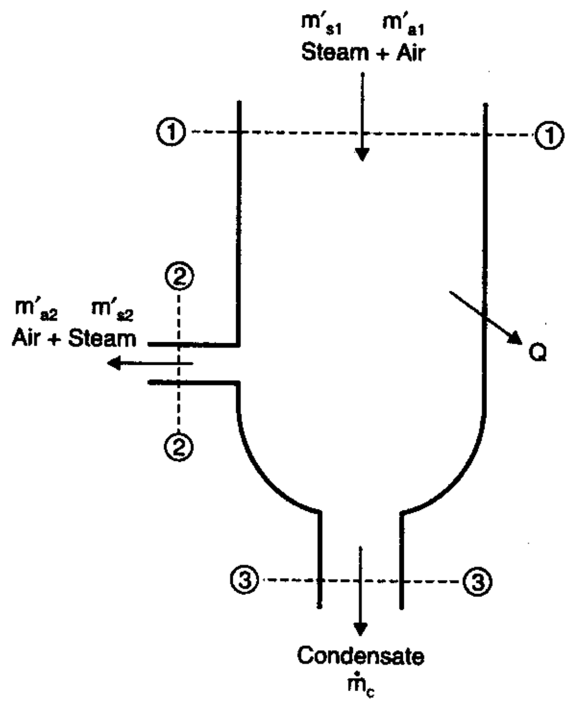

Fig. 3.136

Applying the steady flow energy equation and neglecting changes in kinetic energy, we have

$$Q = (\dot{m}_{s_1} h_{s_1} + \dot{m}_{a_1} h_{a_1}) - (\dot{m}_{s_2} h_{s_2} + \dot{m}_{a_2} h_{a_2}) - \dot{m}_c h_c$$

$$\dot{m}_{a_1} = \dot{m}_{a_2} = 6 \text{ kg/h} ; \dot{m}_{s_2} = 4.35 \text{ kg/h}$$

$$\dot{m}_c = 20000 - 4.35 = 20000 \text{ kg/h approximately.}$$

Also $\quad h_{a_1} - h_{a_2} = c_p (T_1 - T_2)$

$\therefore \quad Q = 20000 \times 2570.1 + 6 \times 1.005 (38 - 27) - 4.35 \times 2550.3 - 20000 \times 150.9$

[where $h_c = h_f$ at 36°C = 150.9 kJ/kg]

i.e. Heat rejected $= 48.37 \times 10^6$ kJ/h

The *mass of cooling water required* for a 5.5 K rise in temperature

$$= \frac{48.37 \times 10^6}{4.186 \times 5.5} = \mathbf{2.1 \times 10^6 \text{ kg/h.}} \quad \textbf{(Ans.)}$$

☞ **Example 3.34.** *The following data relate to a two pass surface condenser :*

Steam condensed	*15400 kg / h*
Temperature of cooling water when it enters the condenser	*15°C*
Temperature of cooling water when it leaves the condenser	*30°C*
The vacuum in the condenser	*675 mm of Hg*
Barometer reading	*755 mm of Hg*

Temperature of the condensate	*32°C*
Quality of exhaust steam	*0.92*
Water velocity in the tubes	*2.6 m/s*
Outside diameter of the tubes	*2.8 cm*
Thickness of the tubes	*0.03 cm*
Heat transfer co-efficient (U)	*3.35 kJ/h/cm²/°C.*

Determine : (i) Area of the tube surface required ; (ii) Number of tubes ; (iii) Length of tubes.

Solution. Absolute pressure in the condenser

$$= 750 - 675 = 75 \text{ mm Hg} = 75 \times 0.001333 = 0.1 \text{ bar}$$

From steam tables saturation temperature corresponding to 0.1 bar pressure is 45.8°C. (At 0.1 bar ; $h_f = 191.8$ kJ/kg ; $h_{fg} = 2392.8$ kJ/kg).

Mean temperature difference,

$$t_m = \frac{t_{w_2} - t_{w_1}}{\log_e \frac{(t_s - t_{w_1})}{(t_s - t_{w_2})}} = \frac{(30-15)}{\log_e \frac{(45.8-15)}{(45.8-30)}} = \frac{15}{\log_e \frac{30.8}{15.8}} = 22.5°\text{C}$$

Heat extracted per kg of steam

$$= (h_f + x h_{fg}) - 1 \times 4.184 \times 32$$
$$= (191.8 + 0.92 \times 2392.8) - 133.9 = 2259.3 \text{ kJ/kg.}$$

Now, heat lost by steam = heat gained by water

$$15400 \times 2393.2 = m_w \times c_{pw} \times (t_{w_2} - t_{w_1}) = m_w \times 4.184 \times (30 - 15)$$

$\therefore$ Mass of circulating water, $m_w = \dfrac{15400 \times 2259.3}{4.184 \times 15} = 554385$ kg/h.

(*i*) **Area of tube surface required, A :**

$$A = \frac{Q}{U t_m}$$

where Q = Heat lost by steam or heat gained by water

$$= m_w \times c_{pw} \times (t_{w_2} - t_{w_1}) = 554385 \times 4.184 \times (30 - 15)$$
$$= 34793202 \text{ kJ/h}$$

U = Heat transfer co-efficient

$$= 3.35 \text{ kJ/h/cm}^2/°\text{C}$$

$$\therefore \quad A = \frac{34793202}{3.35 \times 22.5} = 461601 \text{ cm}^2.$$

(*ii*) **Number of tubes :**

$$\text{Total water required/sec.} = \frac{m_w \times 1000}{3600}$$
$$= \frac{554385 \times 1000}{3600} \qquad [\because \text{ 1 kg occupies 1000 cm}^3 \text{ volume}]$$
$$= 153996 \text{ cm}^3$$

$$\text{Water required/sec./tube} = \frac{\pi}{4} \times (2.8 - 2 \times 0.03)^2 \times (2.6 \times 100) = 1533 \text{ cm}^3$$

$\therefore$ Number of tubes $= \dfrac{2 \times m_w \times 1000}{3600 \times 1533} = \dfrac{2 \times 554385 \times 1000}{3600 \times 1533} = 200.$

(*iii*) **Length of tubes, l :**

Surface area per tube $= \dfrac{A}{200} = \dfrac{461601}{200} = 2308$

i.e., $\pi \times d \times l = 2308$

$\pi \times 2.8 \times l = 2308$

$l =$ **262.4 cm. (Ans.)**

Example 3.35. *A steam condenser consists of 3000 brass tubes of 20 mm diameter. Cooling water enters the tubes at 20°C with a mean flow rate of 3000 kg/s. The heat transfer coefficient on the inner surface is 11270 W/m²°C, and that for condensation on the outer surface is 15500 W/m²°C. The steam condenses at 50°C, and the condenser load is 230 MW. The latent heat of steam is 2380 kJ/kg. Assuming counter flow arrangement, calculate the tube length per pass if two tube passes are used.*

Solution. *Given :* $N_p = 3000$ per pass ; $d = 20$ mm $= 0.02$ m ; $t_{c1} = 20$°C ; $\dot{m}_c = 3000$ kg/s ; $h_i = 11270$ W/m²°C ; $h_o = 15500$ W/m²°C, $t_{h1} = t_{h2} = 50$°C ; condenser load = 230 MW (or 230 × 10^3 kW) ; $h_{fg} = 2380$ kJ/kg ; number of passes = 2.

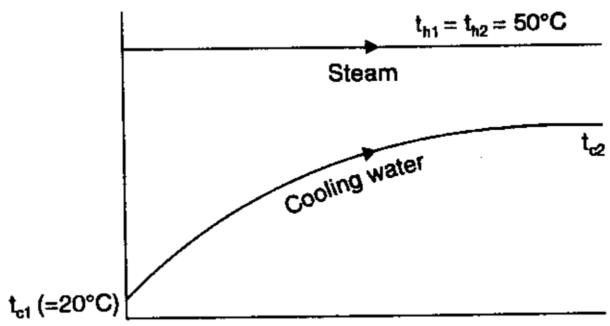

Fig. 3.137

Tube length per pass, L :

Assuming the tubes to be thin, the overall heat transfer coefficient :

$$U_o = \frac{1}{\dfrac{1}{h_i} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{11270} + \dfrac{1}{15500}} = 6525.4 \text{ W/m}^2\text{°C}$$

Heat exchanger load $= \dot{m}_c \, c_{pc} \, (t_{c2} - t_{c1})$

i.e., $230 \times 10^3 = 3000 \times 4.187 \, (t_{c2} - 20)$

$\therefore$ Water outlet temperature, $t_{c2} = 38.31$°C

Log-mean temperature difference,

$$\theta_m = \frac{\theta_1 - \theta_2}{\ln (\theta_1/\theta_2)} = \frac{(t_{h1} - t_{c1}) - (t_{h2} - t_{c2})}{\ln \left[\dfrac{t_{h1} - t_{c1}}{t_{h2} - t_{c2}}\right]}$$

$$= \frac{(50 - 20) - (50 - 38.31)}{\ln \left[\dfrac{50 - 20}{50 - 38.31}\right]} = \frac{18.31}{0.9425} = 19.43\text{°C}$$

Now,
$$Q = U_o A \theta_m$$
$$230 \times 10^6 = 6525.4 \times (\pi\, dL) \times (2N_p) \times 19.43$$
$$= 6525.4 \times (\pi \times 0.02 \times L) \times (2 \times 3000) \times 19.43$$

$\therefore$
$$L = \frac{230 \times 10^6}{6525.4 \times \pi \times 0.02 \times (2 \times 3000) \times 19.43} = \textbf{4.812 m.} \quad \textbf{(Ans.)}$$

Example 3.36. *A two-pass surface condenser is required to handle the exhaust from a turbine developing 15 MW with specific steam consumption of 5 kg/kWh. The condenser vacuum is 660 mm of Hg when the barometer reads 760 mm of Hg. The mean velocity of water is 3 m/s, water inlet temperature is 24°C. The condensate is saturated water and outlet temperature of cooling water is 4°C less than the condensate temperature. The quality of exhaust steam is 0.9 dry. The overall heat transfer coefficient based on outer area of tubes is 4000 W/m²°C. The water tubes are 38.4 mm in outer diameter and 29.6 mm in inner diameter. Calculate the following :*

(*i*) *Mass of cooling water circulated in kg/min,*

(*ii*) *Condenser surface area,*

(*iii*) *Number of tubes required per pass, and*

(*iv*) *Tube length.*

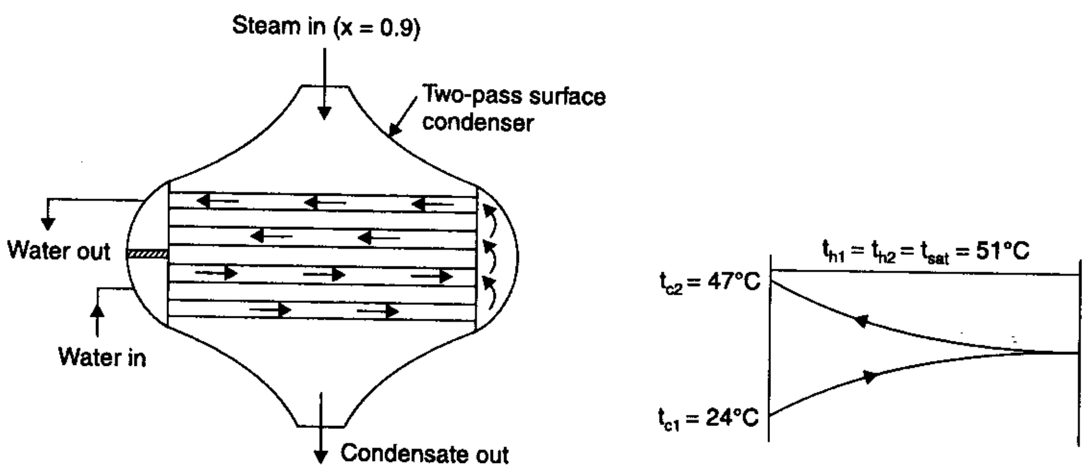

Fig. 3.138. A two-pass surface condenser.

Solution. *Given :* $d_i = 29.6$ mm $= 0.0296$ m ; $d_o = 38.4$ mm $= 0.0384$ m ; $U = 4000$ W/m²°C ; $V = 3$ m/s ; $t_{c1} = 24$°C ; x (dryness fraction) $= 0.9$.

The pressure of the steam in the condenser,
$$p_s = \frac{760 - 660}{760} \times 1.01325 = 0.133 \text{ bar}$$

The properties of steam at $p_s = 0.133$ bar, from steam table, are :
$$t_{sat} = 51°\text{C} \; ; \; h_{fg} = 2592 \text{ kJ/kg}$$

$\therefore$
$$t_{c_2} = 51 - 4 = 47°\text{C}$$

The steam condensed per minute,
$$\dot{m}_s \; (= \dot{m}_h) = \frac{(15 \times 1000) \times 5}{60} = 1250 \text{ kg/min}$$

(*i*) **Mass of cooling water circulated per minute, $\dot{m}_w$ (= $\dot{m}_c$) :**

Heat lost by steam = Heat gained by water

$$\dot{m}_h \times (x \,.\, h_{f_g}) = \dot{m}_c \times c_{pc} \times (t_{c_2} - t_{c_1})$$

$$1250 \times (0.9 \times 2592) = \dot{m}_c \times 4.187\,(47 - 24)$$

$$\therefore \quad \dot{m}_c\ (= \dot{m}_w) = \mathbf{30280\ kg/min.} \quad \textbf{(Ans.)}$$

(*ii*) **Condenser surface area, A :**

$$Q = \frac{\dot{m}_s \times (x \,.\, h_{f_g})}{60} = U A\, \theta_m \quad \text{...}(i)$$

where

$$\theta_m = \frac{\theta_1 - \theta_2}{\ln(\theta_1/\theta_2)} = \frac{(t_{h1} - t_{c1}) - (t_{h2} - t_{c2})}{\ln[(t_{h1} - t_{c1})/(t_{h2} - t_{c2})]}$$

$$= \frac{(51 - 24) - (51 - 47)}{\ln[(51 - 24)/(51 - 47)]} = \frac{(27 - 4)}{\ln(27/4)} = 12.04°\text{C}$$

Substituting the values in eqn. (*i*), we get

$$\frac{1250}{60} \times (0.9 \times 2592 \times 10^3) = 4000 \times A \times 12.04 \quad \text{or} \quad A = \mathbf{1009.1\ m^2}. \quad \textbf{(Ans.)}$$

(*iii*) **Number of tubes required per pass, N_p :**

$$\dot{m}_w = \left(\frac{\pi}{4} d_i^2 \times V \times \rho\right) \times N_p$$

$$\frac{30280}{60} = \frac{\pi}{4} \times (0.0296)^2 \times 3 \times 1000 \times N_p$$

or

$$N_p = \frac{30280 \times 4}{60 \times \pi \times (0.0296)^2 \times 3 \times 1000} = 244.46 \text{ say } \mathbf{245} \quad \textbf{(Ans.)}$$

(Total number of tubes required, $N = 2N_p = 2 \times 245 = 490$)

(*iv*) **Tube length, L :**

$$A = (\pi\, d_o\, L) \times (2N_p)$$

$$1009.1 = \pi \times 0.0384 \times L \times (2 \times 245)$$

or

$$L = \frac{1009.1}{\pi \times 0.0384 \times 2 \times 245} \simeq \mathbf{17.1\ m} \quad \textbf{(Ans.)}$$

3.22. FEED WATER TREATMENT

For steam power plants water is one of the most important raw materials. In most of the cases, water used for steam power plants contains impurities which must be treated before use. All natural waters-even rain, snow, hail, treated municipal supplies contain impurities in one form or the other.

3.22.1. Classification of Impurities in Water

The impurities in water may be classified as follows :

1. Visible impurities

(*i*) *Microbiological growth.* Presence of micro-organisms is always undesirable as they may produce *clogging troubles*.

(*ii*) *Turbidity and sediments. Turbidity* is the suspended insoluble matter whereas *sediments* are the coarse particles which settle down in stationary water, both are objectionable.