2

Steam Turbines and Condensers

2.1 INTRODUCTION

Steam turbine is a type of turbomachine which is an assembly of nozzles and blades. It converts a part of the energy of high temperature and high pressure steam into mechanical energy (or shaft work). The operation of steam turbine completely depends on the dynamic action of the steam expanding in nozzles. The steam turbines are used for the generation of electricity in steam power plants varying from 1 MW to 1500 MW capacity. These are also used for marine propulsion. The steam turbines operate at very high speed (up to about 40,000 rpm) and are able to give efficiency about 40% which is higher than the other power producing devices.

The steam expands in a turbine from high pressure to low pressure (below the atmospheric pressure). Steam cannot be exhausted to atmosphere at a pressure lower than the atmospheric pressure. This is made possible by using an additional unit called condenser. A steam condenser is a closed vessel in which vacuum is maintained and the exhausted steam is condensed by extraction of heat. About 50% to 60% of the heat energy associated with steam is lost in a steam condenser. Cooling water supplied to the condenser for the condensation of steam becomes hot. The cooling towers and the cooling ponds are used to cool this hot water coming out of the condenser so that the cooled water can be reutilized in the condenser again.

2.2 STEAM NOZZLE AND ITS GENERAL FLOW ANALYSIS

A nozzle is a duct of smoothly varying cross section by means of which pressure energy of steadily flowing fluid is converted into kinetic energy. The fluid enters the nozzle with a relatively small velocity and high pressure. As it flows through the nozzle, its pressure falls and a part of the enthalpy of steam gets converted into kinetic energy. The amount of energy which converts into kinetic energy depends on the pressure ratio and the type of expansion. Generally, nozzles are designed to obtain isentropic expansion which imparts maximum conversion. The velocity increases from entrance to the exit of the nozzle.

The applications of nozzles are: (i) In steam and gas turbines for power generation, (ii) In aviation purposes such as propulsion of jet engines and rocket, (iii) In injectors and ejectors, and (v) For flow measurements, etc.

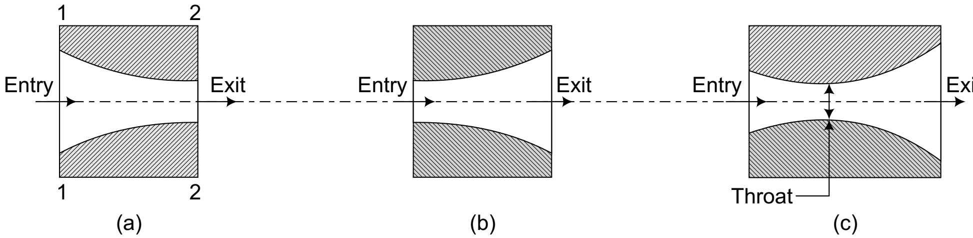


Fig. 2.1 Types of Nozzles

The nozzles are of three types:

(i) Convergent Nozzle If the cross section of the nozzle decreases continuously from the entrance to the exit, it is called convergent nozzle shown in Figure 2.1 (a).

(ii) Divergent Nozzle If the cross section of the nozzle increases continuously from the entrance to the exit, it is called divergent nozzle shown in Figure 2.1 (b).

(iii) Convergent-divergent Nozzle If the cross section of the nozzle first decreases and then increases, it is called convergent-divergent nozzle shown in Figure 2.1 (c).

In nozzles the flow is considered adiabatic (i.e., $q = 0$) and no work is done on or by the fluid (i.e., $w = 0$). Consider a nozzle (Figure 2.1a) in which section 1-1 is the entrance of the nozzle and section 2-2 is the exit of the nozzle. Let the flow of fluid be initially at pressure p_1, velocity V_1, and enthalpy h_1 expands through the nozzle. Applying steady flow energy equation, we have,

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \quad (\because q = 0 \text{ and } w = 0)$$

The entering velocity of steam V_1 is small as compared with velocity V_2. Thus, neglecting V_1, we have,

$$h_1 = h_2 + \frac{V_2^2}{2}$$

$$V_2 = \sqrt{2 \times (h_1 - h_2)}$$

Generally, enthalpy is expressed in kJ/kg, then the exit velocity in m/s is given by,

$$V_2 = \sqrt{2 \times 1000 \times (h_1 - h_2)}$$

$$= 44.72 \times \sqrt{(h_1 - h_2)} \text{ m/s} \qquad (2.1)$$

When the nozzle efficiency (η_{nozzle}) is also considered, then the exit velocity is given by,

$$V_2 = 44.72 \times \sqrt{\eta_{nozzle} \times (h_1 - h_2)} \text{ m/s} \qquad (2.2)$$

The general thermodynamic relation for flow of gas through a duct is given by,

$$\frac{dA}{A} = \frac{dV}{V} (M^2 - 1) \qquad (2.3)$$

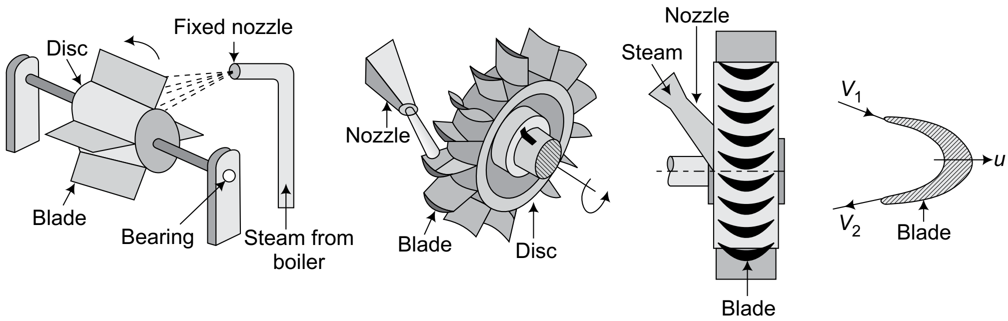


Fig. 2.2 Principle of Impulse Turbine

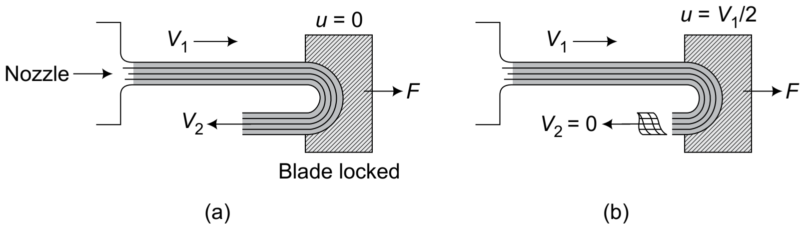


Fig. 2.3 Jet Action on Blade

About 2000 years ago, the first steam turbine was made by Hero of Alexandria which worked on pure reaction principle. There was no provision for driving any device by it, but it was a type of toy (Figure 2.4). It consisted of a hollow ball mounted on between two pivots. The ball was provided with two converging tubes (nozzles). The steam generated in a cauldron placed below the ball was supplied to the ball through one pivot. The steam was expanded through the nozzles to atmospheric pressure which results in a reactive force on the ball and made it to rotate between the pivots.

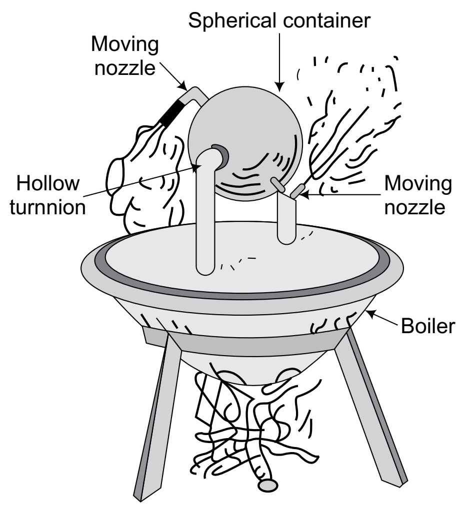


Fig. 2.4 Hero's Turbine (Principle of Reaction Turbine)

2.4 CLASSIFICATION OF STEAM TURBINES

On the basis of the principles of operation (i.e., mode of steam action) the steam turbines can be divided into two types: (i) Impulse turbine, and (ii) Reaction turbine (or impulse-reaction turbine).

(i) Impulse Turbine In impulse turbines the steam expands only in the nozzles. It means the pressure drop (enthalpy drop) takes place only in nozzles and not in moving blades. The moving blades only deflect the steam through an angle. An impulse turbine works on the principle of impulse, means the kinetic energy of steam is used to exert a force on the moving blades. This is achieved by having the symmetrical blades, means the cross sectional area of blades is constant as shown in Figure 2.5. Thus, steam pressure remains constant while it flows through the moving blades.

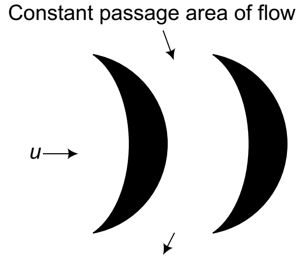


Fig. 2.5 Impulse Blade

A simple impulse turbine consists of a nozzle or a set of nozzles, a rotor (or runner) mounted on a shaft, one set of moving blades fixed to the runner, and a casing. A set (or row) of nozzles and moving blades makes a stage. The schematic view and the flow of steam through a simple impulse turbine are shown in Figure 2.6 in which the pressure and velocity variation have also been illustrated. It can be seen from the figure that the pressure drop takes place only in nozzles. So, the complete expansion of steam from the steam chest pressure to the condenser pressure takes place only in one set of nozzles and it leaves with a high

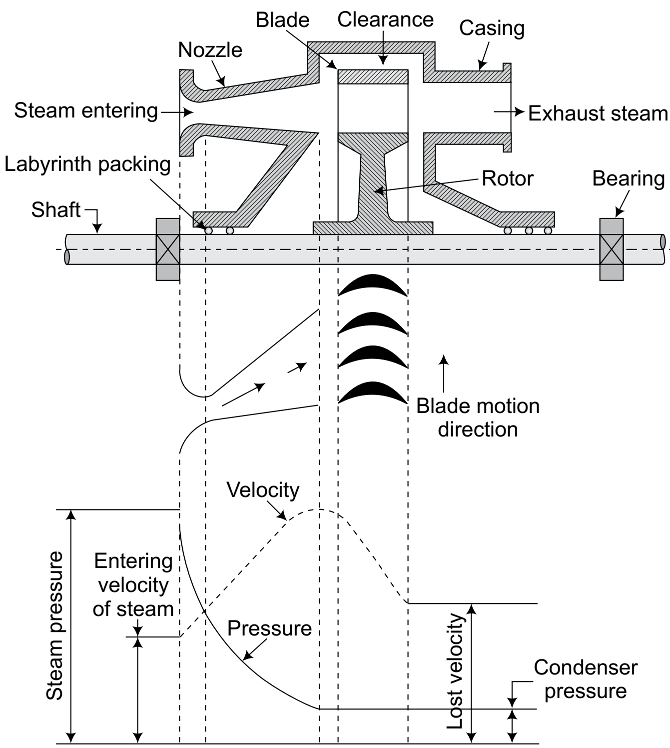


Fig. 2.6 Schematic View of Simple Impulse Turbine

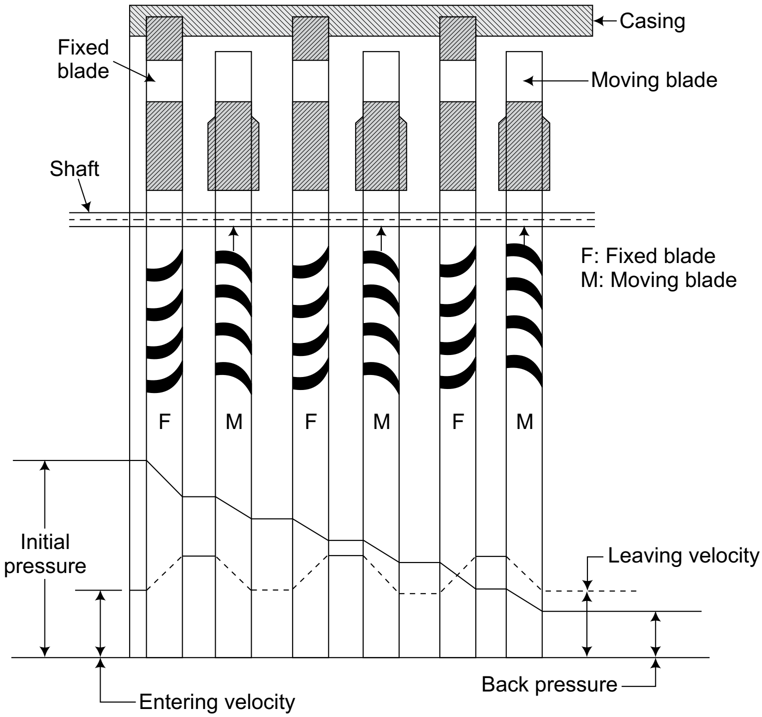


Fig. 2.8 Schematic View of a Reaction Turbine

the total initial available kinetic energy. The example of an impulse reaction turbine is Parson's reaction turbine.

The steam turbine can also be classified on the following basis:

1. On the basis of direction of flow: (i) Radial flow turbine, and (ii) Axial flow turbine.
2. On the basis of method of compounding: (i) Velocity compounded turbine, (ii) Pressure compounded turbine, and (iii) Pressure-velocity compounded turbine.
3. On the basis of number of stages: (i) Single-stage turbine, and (ii) Multistage turbine.
4. On the basis of position of shaft: (i) Horizontal shaft turbine, and (ii) Vertical shaft turbine.
5. On the basis of pressure of steam at the inlet: (i) Low pressure steam turbine, and (ii) High pressure steam turbine.
6. On the basis of exhaust condition of steam: (i) Condensing turbine, and (ii) Non-condensing turbine.
7. On the basis of their use or service.

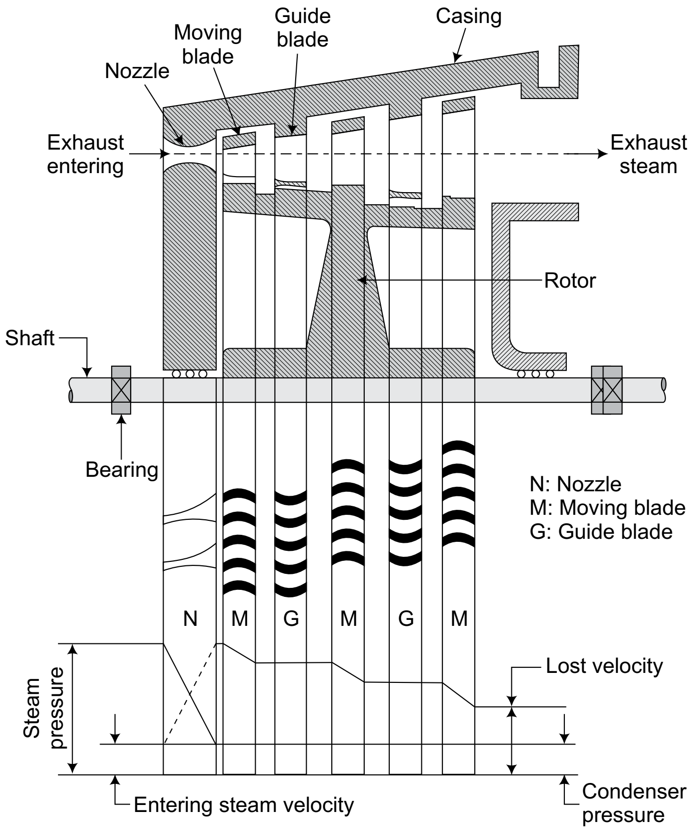


Fig. 2.9 Velocity Compounding (The Curtis Turbine)

gets converted into kinetic energy. There is no pressure drop either in moving blades or the guide blades, means it remains constant as in the case of simple impulse turbine.

The high velocity steam from the nozzles enters the first row of moving blades where its velocity reduces slightly by giving up its kinetic energy to the rotor. It then enters the first row of guide blades which direct it to the second row of moving blades. A slight drop in velocity occurs in the guide blades due to friction. When the steam passes through the second row of moving blades again there will be slight drop in its velocity due to giving up some more kinetic energy to the rotor. Then it is again directed by the second row of guide blades to the third row of moving blades where further velocity drop takes place, and finally, it leaves the rotor axially with a reduced velocity. The pressure and velocity variation is shown in Figure 2.9.

Therefore, the kinetic energy of the steam gained in the nozzles is successively absorbed in stages by the rows of moving blades and gets converted into mechanical energy. Finally, the steam is exhausted from the last row of the moving blades. The example of velocity compounded impulse turbine is the Curtis turbine named after its designer. The velocity compounded turbines are used as drives for centrifugal compressors, centrifugal pumps, small generators, and for driving feed pumps in large power plants.

Advantages of Velocity Compounded Turbines Due to relatively fewer numbers of stages (2 to 3 only) its initial cost is low. It is easy to start, requires less space, and is reliable. The turbine and its casing are not subjected to high pressure as all the expansion of steam takes place in the nozzles, so these need not be manufactured of high strength. Due to low pressure inside the turbine the leakage losses are less.

Disadvantages of Velocity Compounded Turbines Due to very high steam initial velocity the frictional losses are more which causes low efficiency. The ratio of blade velocity to steam velocity is not optimum for all the wheels which also results in low efficiency of the turbine. Therefore, the efficiency of the turbine decreases with the increase in the number of stages. The power developed in each successive blade row decreases with the increase in the number of rows and all the rows require same space, material, and initial cost. So, all stages are not used with equal economy.

2.7.2 Pressure Compounding (The Rateau Turbine)

In this type of turbine the whole pressure drop from the steam chest pressure to the condenser pressure divides into a series of smaller pressure drops across many stages of impulse turbine, thus it is called pressure compounded impulse turbine.

It has a number of simple impulse turbines in a series on the same shaft. Each of the simple impulse turbines consists of one row of nozzles and one row of moving blades which is termed a stage of the turbine. The rows of moving blades are mounted on the rotor or wheel which is keyed to the turbine shaft in series. The rows of fixed blades (nozzles) are fitted into the diaphragm which separates the two stages and is fixed to the turbine casing. The pressure drop is divided up equally between all the nozzles rows. The pressure and velocity variations are shown in Figure 2.10.

The steam from the boiler enters the first row of nozzles where its pressure reduces partially and its velocity increases. Then it passes over the first row of moving blades where nearly all its velocity is absorbed and its pressure remains constant. The exhaust steam from the first row of moving blades enters the next nozzles row. The steam again partially expands and its velocity again increases which gets absorbed in the second row of moving blades, and so on.

This turbine was invented by Professor Rateau, so it is known as Rateau turbine. In this turbine, the ratio of blade velocity to steam velocity remains constant, so this is the most efficient type of impulse turbine. But it is very costly as it has large number of stages. As the steam pressure decreases from one stage to the next stage, the specific volume of the steam increases, therefore, the blade height increases towards the low pressure side of the turbine.

2.7.3 Pressure-Velocity Compounding

It is a combination of pressure and velocity compounding. The total drop in steam pressure is carried out in two stages and the velocity obtained in each stage is also compounded. This type of arrangement for two rotors is shown in Figure 2.11.

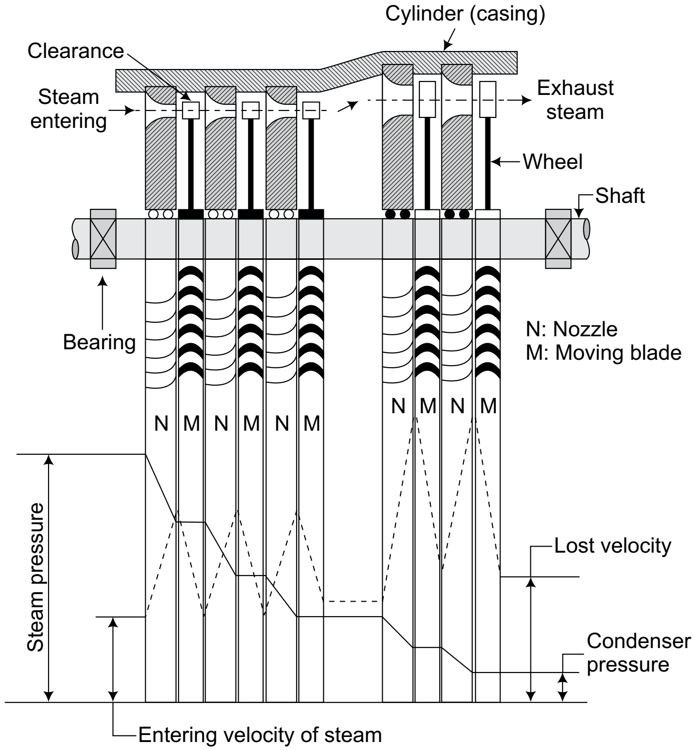


Fig. 2.10 Pressure Compounding (The Rateau Turbine)

In the first row of nozzles, there is a drop in pressure of the steam but its velocity increases. The pressure remains constant in the two rows of moving blades of the first rotor and in the first row of guide blades (fixed blades). But there is a velocity drop in the moving blades row. In the second set of nozzles the remaining pressure drop occurs and velocity increases. The pressure remains constant in the two rows of moving blades of the second rotor and in the second row of guide blades. But the velocity drops in the moving blades of the second rotor.

So, in each rotor, velocity drop is achieved by the many rows of moving blades and hence its velocity is compounded. And the total pressure drop occurs into two small pressure drops in two nozzles rows, hence, it is pressure compounded. This method of compounding is used in Moore and the Curtis turbines. This arrangement results in a more compact turbine than a pressure compounded turbine, but its efficiency is relatively lower. The specific volume of steam is higher in the second stage, so, the blade height is greater than the first stage.

2.8 VELOCITY DIAGRAMS FOR IMPULSE TURBINE

To evaluate the force on the blades and the power developed by a turbine, it is necessary to determine the rate of change of momentum of steam across the moving blades. The steam should

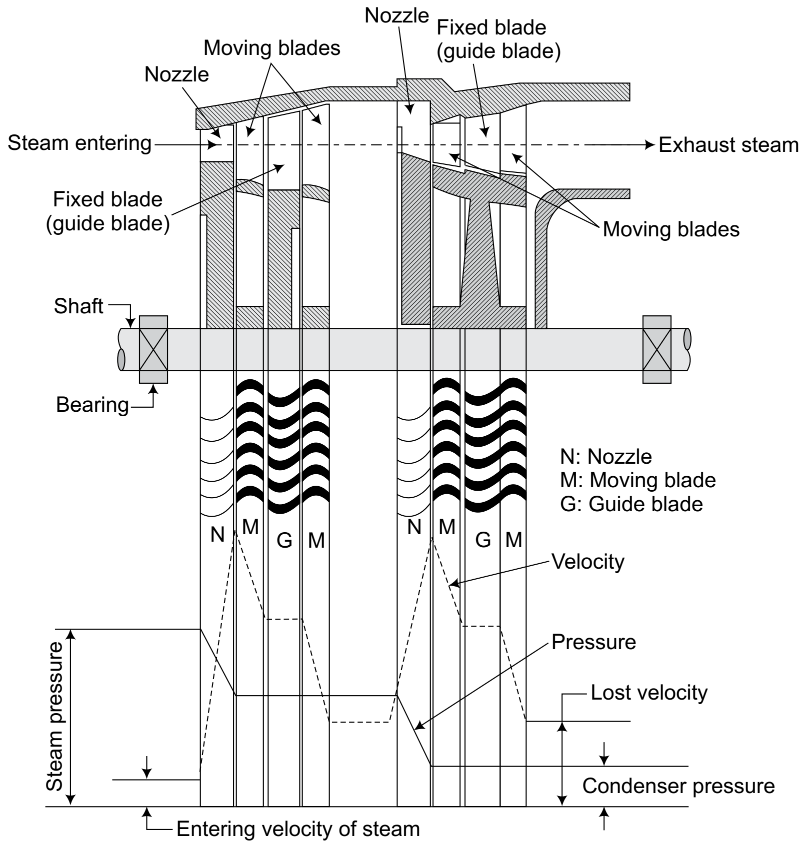


Fig. 2.11 Pressure-Velocity Compounding

enter and leave the blade without any shock for which the inlet and outlet angles of the moving blades should be evaluated. In order to fulfil these objectives, it is essential to draw the vector diagrams at the inlet and outlet of the moving blades showing the variations of velocity of steam during its flow through the blade passage.

The following notations are used for the velocity diagrams:

Suffixes 1 and 2 denote the inlet and outlet conditions respectively, for moving blades as shown in Figure 2.12 and Figure 2.13.

u = Tangential or circumferential velocity of blades which remains constant at inlet and outlet of moving blades because of small blades height, i.e., $u_1 = u_2 = u$

V_1 and V_2 = Absolute velocity of steam at the inlet and outlet respectively

V_{w1} and V_{w2} = Velocity of whirl at the inlet and outlet respectively, (i.e., tangential component of V_1)

V_{f1} and V_{f2} = Velocity of flow at the inlet and outlet respectively, (i.e., axial component of V_1 and V_2 respectively)

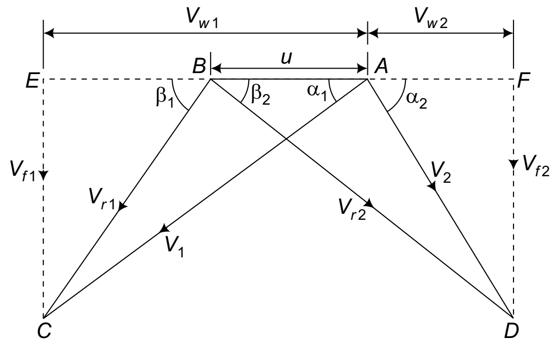


Fig. 2.13 Combined Velocity Diagram for Impulse Turbine

The value of K varies from 0.85 to 0.9 and for smooth blades $K = 1$, means $V_{r1} = V_{r2}$.

The procedure for drawing the combined velocity diagram is given below:

1. Draw horizontal line AB equal to blade velocity u to some suitable scale.
2. Draw a line AC at an angle α_1 with AB. Cut $AC = V_1$.
3. Join B and C which represents the relative velocity V_{r1} at inlet. The blade inlet angle β_1 is measured and its value is evaluated.
4. From point C draw a perpendicular CE on AB produced. CE represents flow velocity (axial velocity) at inlet and AE represents whirl velocity (tangential velocity) at inlet.
5. From point B draw a line BD at an angle β_2 (blade outlet angle).

   Cut $BD = V_{r2} = KV_{r1}$. Join A and D which represents the absolute velocity (V_2) at outlet. The angle α_2 is measured.
6. From point D draw a perpendicular DF on BA produced. DF represents flow velocity (axial velocity) at outlet and AF represents whirl velocity (tangential velocity) at outlet.

   Thus, velocity triangles get completed.

2.8.1 Force and Work Done on Turbine Blades and Efficiency

1. Force in the tangential direction (F_t).

$$F_t = \text{mass} \times \text{acceleration in tangential direction}$$

$$= \text{mass per second} \times \text{change in velocity in tangential direction}$$

$$F_t = m \times (V_{w1} - V_{w2}) \text{ N}$$

(when V_{w1} and V_{w2} are in opposite direction)

$$F_t = m \times (V_{w1} + V_{w2}) \text{ N}$$

(when V_{w1} and V_{w2} are in same direction)

Thus general expression for tangential force becomes,

$$F_t = m \times (V_{w1} \pm V_{w2}) \text{ N} \tag{2.6}$$

2.9.1 Degree of Reaction (R)

The degree of reaction is defined as the ratio of enthalpy drop in the moving blades to the sum of the enthalpy drops in the fixed and the moving blades, i.e., in a stage. Figure 2.16 shows the mounting of fixed and moving blades with $h - s$ diagram in which points 1, 2, and 3 indicate the actual conditions of steam.

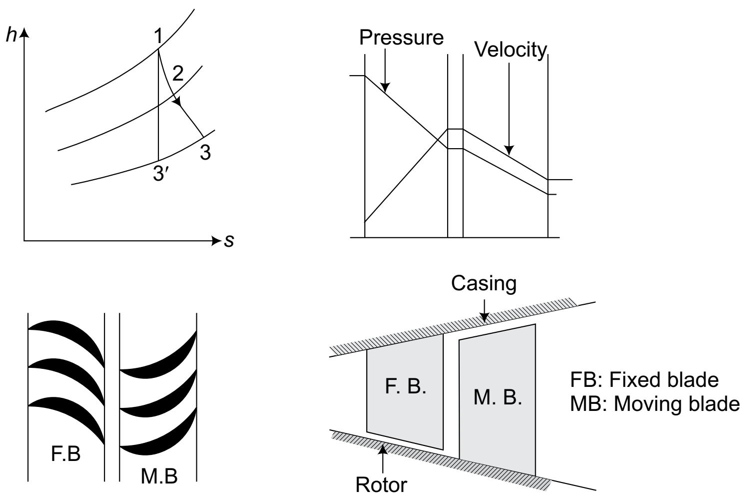


Fig. 2.16 Isentropic Expansion with Pressure-Velocity Distribution in a Reaction Turbine Stage

The degree of reaction can be given by,

$$R = \frac{(\Delta h)_{\text{moving blades}}}{(\Delta h)_{\text{fixed blades}} + (\Delta h)_{\text{moving blades}}}$$

$$= \frac{(\Delta h)_m}{(\Delta h)_f + (\Delta h)_m} = \frac{h_2 - h_3}{h_1 - h_3} \tag{2.19}$$

In pure impulse turbine, degree of reaction is zero and in pure reaction turbine it is one.

The enthalpy drop in moving blades is given by,

$$\Delta h_m = \frac{V_{r2}^2 - V_{r1}^2}{2} \tag{2.20}$$

And enthalpy drop in fixed blades is given by,

$$\Delta h_f = \frac{V_1{}^2 - V_2{}^2}{2} \tag{2.21}$$

The total enthalpy drop in a stage = work done by the steam in the stage, i.e.,

$$\Delta h_m + \Delta h_f = u(V_{w1} + V_{w2}) \tag{2.22}$$

$$R = \frac{V_{r2}^2 - V_{r1}^2}{2u(V_{w1} + V_{w2})} \tag{2.23}$$

From Figure 2.15, we have,

$$V_{r2} = V_{f2} \operatorname{cosec} \beta_2,$$

$$V_{r1} = V_{f1} \operatorname{cosec} \beta_1,$$

and $$(V_{w1} + V_{w2}) = V_{f1} \cot \beta_1 + V_{f2} \cot \beta_2$$

The velocity of flow remains constant, i.e.,

$$V_{f1} = V_{f2} = V_f$$

So, degree of reaction can be given by,

$$R = \frac{(V_{f2} \operatorname{cosec} \beta_2)^2 - (V_{f1} \operatorname{cosec} \beta_1)^2}{2uV_f (\cot \beta_1 + \cot \beta_2)}$$

$$R = \frac{V_f (\operatorname{cosec} \beta_2 - \operatorname{cosec} \beta_1)^2}{2u(\cot \beta_1 + \cot \beta_2)}$$

$$R = \frac{V_f}{2u} \left[\frac{(\cot^2\beta_2 + 1) - (\cot^2\beta_1 + 1)}{(\cot \beta_1 + \cot \beta_2)} \right]$$

$$R = \frac{V_f}{2u} \left[\frac{(\cot^2\beta_2 - \cot^2\beta_1)}{(\cot \beta_1 + \cot \beta_2)} \right]$$

$$\therefore \quad R = \frac{V_f}{2u} (\cot \beta_1 + \cot \beta_2) \tag{2.24}$$

If the reaction turbine is designed for 50% reaction then Eq. (2.24) can be written as,

$$\frac{1}{2} = \frac{V_f}{2u} (\cot \beta_1 + \cot \beta_2)$$

$$u = V_f (\cot \beta_1 + \cot \beta_2)$$

The blade velocity u can also be given by,

$$u = V_f (\cot \alpha_1 + \cot \beta_1) = V_f (\cot \beta_2 + \cot \alpha_2)$$

Therefore, $\beta_1 = \alpha_2$, and $\beta_2 = \alpha_1$

It means that the moving blades and fixed blades 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.

2.9.2 Blade or Diagram Efficiency of a Reaction Turbine

For determining the efficiency the following assumptions are made:

(i) The degree of reaction is 50%,

(ii) The fixed and moving blades are of the same shape, and

Substituting the values of W and Δh in blade efficiency expression, we have,

$$\eta_b = \frac{W}{\Delta h}$$

$$= \frac{V_1^2\,(2\rho\cos\alpha_1 - \rho^2)}{(V_1^2/2)\,(1 + 2\rho\cos\alpha_1 - \rho^2)}$$

$$\eta_b = \frac{2\rho\,(2\cos\alpha_1 - \rho)}{1 + 2\rho\cos\alpha_1 - \rho^2}$$

$$= \frac{(4\rho\cos\alpha_1 - 2\rho^2)}{(1 + 2\rho\cos\alpha_1 - \rho^2)} \qquad (2.28)$$

For maximum efficiency,

$$\frac{d\eta_b}{d\rho} = 0,$$

i.e.,

$$\frac{d}{d\rho}\left[\frac{(4\rho\cos\alpha_1 - 2\rho^2)}{(1 + 2\rho\cos\alpha_1 - \rho^2)}\right] = 0$$

$$\frac{(1 + 2\rho\cos\alpha_1 - \rho^2)\,(4\cos\alpha_1 - 4\rho) - (4\rho\cos\alpha_1 - 2\rho^2)\,.\,(2\cos\alpha_1 - 2\rho)}{(1 + 2\rho\cos\alpha_1 - \rho^2)} = 0$$

$$4\,(\cos\alpha_1 - \rho)\,(1 + 2\rho\cos\alpha_1 - \rho^2) - 4\rho\,(\cos\alpha_1 - \rho)\,(2\cos\alpha_1 - \rho) = 0$$

$$4\,(\cos\alpha_1 - \rho)\,[(1 + 2\rho\cos\alpha_1 - \rho^2) - \rho\,(2\cos\alpha_1 - \rho)] = 0$$

$$4\,(\cos\alpha_1 - \rho)\,[1 + 2\rho\cos\alpha_1 - \rho^2 - 2\rho\cos\alpha_1 + \rho] = 0$$

$$4\,(\cos\alpha_1 - \rho)\,.\,[1] = 0$$

$$\therefore \qquad \rho = \cos\alpha_1 \qquad (2.29)$$

Substituting the value of ρ from equation (2.29) in equation (2.28), the value of maximum efficiency is given by,

$$(\eta_b)_{\max} = \frac{(4\cos^2\alpha_1 - 2\cos^2\alpha_1)}{(1 + 2\cos^2\alpha_1 - \cos^2\alpha_1)}$$

$$\therefore \qquad (\eta_b)_{\max} = \frac{2\cos^2\alpha_1}{(1 + \cos^2\alpha_1)} \qquad (2.30)$$

2.10 LOSSES IN STEAM TURBINES

The steam turbine losses tend to decrease the efficiency and work output of a turbine. The various losses in steam turbines are given below.

Residual Velocity Loss The steam leaves the turbine with certain absolute velocity (V_1). The energy loss due to absolute exit velocity of steam is equivalent to ($V_1^2/2$). In a single-stage impulse turbine, the residual velocity loss may be about 10% to 12% which can be reduced by using multistage turbine.

Loss Due to Friction and Turbulence Friction loss occurs in nozzles, turbine blades, and between steam and rotating disc. The friction loss in the nozzle is taken into account by nozzle efficiency. The loss due to friction and turbulence is about 10%.

Losses Due to Leakage Some amount of steam leaks from the chamber without doing useful work. The total leakage loss is about 1% to 2%. The steam leakage occurs: (i) between the turbine shaft and the bearings, (ii) at the clearance between the diaphragm and the stationary disc in the case of impulse turbines, (iii) at the blade tips in the case of reaction turbines, and (iv) leakage of steam through the labyrinth glands.

Loss Due to Mechanical Friction and Bearing This loss occurs due to friction between shaft and bearing which can be reduced by a proper lubricating system.

Losses in Regulating Valves The steam enters the turbine through the stop and regulating (governor) valves. In these valves, the steam gets throttled and the pressure of steam at the entry to the turbine is less than the boiler pressure. Although throttling is a constant enthalpy process, the enthalpy drop available for work in the turbine is reduced. The magnitude of this loss may be of the order of 5% to 10%.

Losses Due to Wetness of Steam The condition of steam after last stage of the turbine is wet. The velocity of water particles is less than that of steam. Therefore, the water particles have to be dragged along with the steam which results in loss of the kinetic energy of steam.

Radiation Loss Turbines are heavily insulated to avoid any heat loss to the surroundings, so this loss is negligible.

2.11. EFFICIENCIES AND REHEAT FACTOR

Due to various internal losses in the stage of steam turbine the work available is less than that due to isentropic expansion. This is schematically presented on the $h - s$ diagram as shown in Figure 2.17. The process $1 - 2s$ shows the isentropic expansion in the nozzle and 1-2 shows the actual path of the process which is due to friction loss.

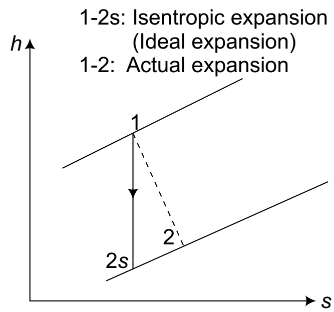


Fig. 2.17 Expansion of Steam in Single Stage on $h - s$ Diagram

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

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

$$= \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{\Delta h_{1-2}}{\Delta h_{1-2s}} \tag{2.31}$$

Figure 2.18 shows the expansion of steam through a four-stage turbine.

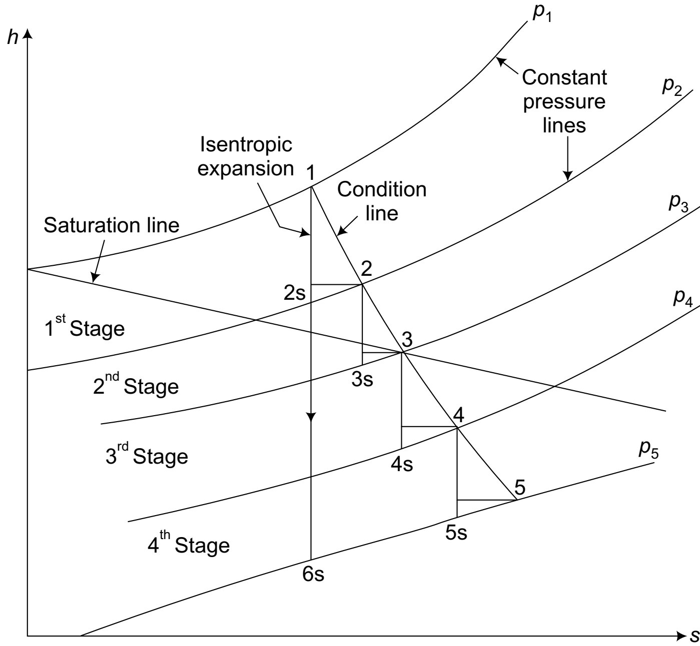


Fig. 2.18 Expansion of Steam in Multistage Turbine on $h - s$ Diagram

The isentropic heat drops are shown by points $1 - 2s$, $2 - 3s$, $3 - 4s$ and $4 - 5s$, whereas actual heat drops are shown by points 1-2, 2-3, 3-4, and 4-5. The curve joining the points 1, 2, 3, 4, and 5 is called the condition line. The heat drops Δh_{1-2}, Δh_{2-3}, Δh_{3-4}, Δh_{4-5}, are the actual heat drops which get converted into useful work. The heat drops Δh_{1-2s}, Δh_{2-3s}, Δh_{3-4s} and Δh_{4-5s} are the isentropic heat drops. The sum of the isentropic heat drops is called cumulative heat drop which is denoted by Δh_c. The heat drop Δh_{1-6s} is called direct isentropic heat drop (or Rankine heat drop) which is denoted by Δh_{ise}. The constant pressure lines on the $h - s$ diagram diverges as we move from left to right, so the cumulative heat drop is always greater than Rankine heat drop.

Reheat Factor (RF) It is defined as the ratio of the cumulative heat drop to the direct isentropic heat drop (or Rankine heat drop). It can be given by,

$$RF = \frac{\text{Cumulative isentropic heat drop}}{\text{Rankine heat drop}} \tag{2.32}$$

Referring to Figure 2.18,

$$RF = \frac{\Delta h_{1-2s} + \Delta h_{2-3s} + \Delta h_{3-4s} + \Delta h_{4-5s}}{\Delta h_{1-6s}}$$

$$= \frac{\Delta h_c}{\Delta h_{ise}} \tag{2.33}$$

As the cumulative heat drop is always greater than Rankine heat drop, so the value of reheat factor is always greater than unity. The reheat factor depends on turbine efficiency, initial pressure, superheat, exit pressure, and number of stages in a given pressure range. The reheat factor is greater if the number of stages are more for a given pressure range and lower the stage efficiency. The lower value of reheat factor is always desirable. The value of reheat factor generally lies in the range of 1.02 to 1.06.

Internal Efficiency It is defined as the ratio of the internal work done (actual work done) in the turbine to the direct Rankine work done. It is given by,

$$\eta_i = \frac{\Delta h_{1-2} + \Delta h_{2-3} + \Delta h_{3-4} + \Delta h_{4-5}}{\Delta h_{1-6s}}$$

$$= \frac{\Delta h_i}{\Delta h_{ise}} \tag{2.34}$$

If the stage efficiency remains same for all the stages then we have,

$$\eta_s = \frac{\Delta h_{1-2}}{\Delta h_{1-2s}} = \frac{\Delta h_{2-3}}{\Delta h_{2-3s}} = \frac{\Delta h_{3-4}}{\Delta h_{3-4s}} = \frac{\Delta h_{4-5}}{\Delta h_{4-5s}}$$

$$\eta_s = \frac{\Delta h_{1-2} + \Delta h_{2-3} + \Delta h_{3-4} + \Delta h_{4-5}}{\Delta h_{1-2s} + \Delta h_{2-3s} + \Delta h_{3-4s} + \Delta h_{4-5s}}$$

$$= \frac{\Delta h_i}{\Delta h_c} \tag{2.35}$$

or

$$\Delta h_i = \eta_s \,.\, \Delta h_c \tag{2.36}$$

But

$$\Delta h_c = RF \times \Delta h_{ise} \quad \text{(From Eq. 2.33)}$$

∴

$$\Delta h_i = \eta_s \,.\, (RF \times \Delta h_{ise}) \tag{2.37}$$

Substituting equation (2.37) in equation (2.34), we have,

$$\eta_i = \frac{\Delta h_i}{\Delta h_{ise}} = \frac{\eta_s \,.\, (RF \times \Delta h_{ise})}{\Delta h_{ise}}$$

$$= \eta_s \times RF \tag{2.38}$$

2.12 GOVERNING OF STEAM TURBINES

The purpose of governing is to maintain the speed of the turbine fairly constant irrespective of load on the turbine. The methods of governing which are used in steam turbines are: (1) Throttle governing, (2) Nozzle control governing, (3) By-pass governing, (4) Combination of throttle and nozzle governing, and (5) Combination of throttle and bypass governing.

2.12.1 Throttle Governing

The main parts of a simple throttle governing system are shown schematically in Figure 2.19. The turbine stop valve '*A*' is used to start and stop the turbine by its opening or closing with the help

2.12.2 Nozzle Control Governing

In the nozzle control governing, the steam supply from the main valve is divided into two, three, or more lines. Each line feeds a set of nozzles controlled by valves. The steam supply to one or more of the nozzle sets is cut-off by the valves operated by the governor when the load on the turbine decreases.

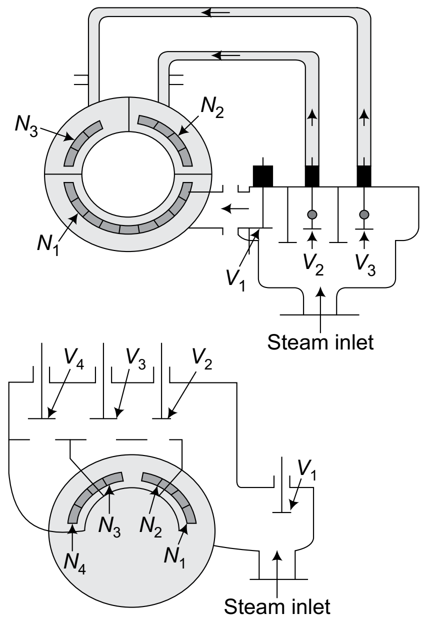


Fig. 2.20 Schematic View of Nozzle Control Governing

Figure 2.20 shows arrangements for nozzle control governing, where nozzles are divided into three sets N_1, N_2, and N_3 each controlled by valves V_1, V_2, and V_3 respectively and four sets N_1, N_2, N_3, and N_4 each controlled by valves V_1, V_2, $V_{3,}$ and V_4 respectively. This method is suitable for medium and large steam turbines. It is generally employed at the first stage of turbine and is not practical for multistage impulse turbines.

2.12.3 Bypass Governing

In this method of governing, the extra quantity of steam from the first stage nozzle box of the turbine is bypassed into the latter stages as shown in Figure 2.21.

The main valve and the bypass valve are under the control of speed governor. The total amount of steam entering the turbine passes through the main valve. For all loads greater than the economic load, the bypass valve gets opened which allows the steam to pass from the first stage nozzle box into the steam belt and thus into the nozzles of the downstream stages. When the load reduces the bypass valve closes.

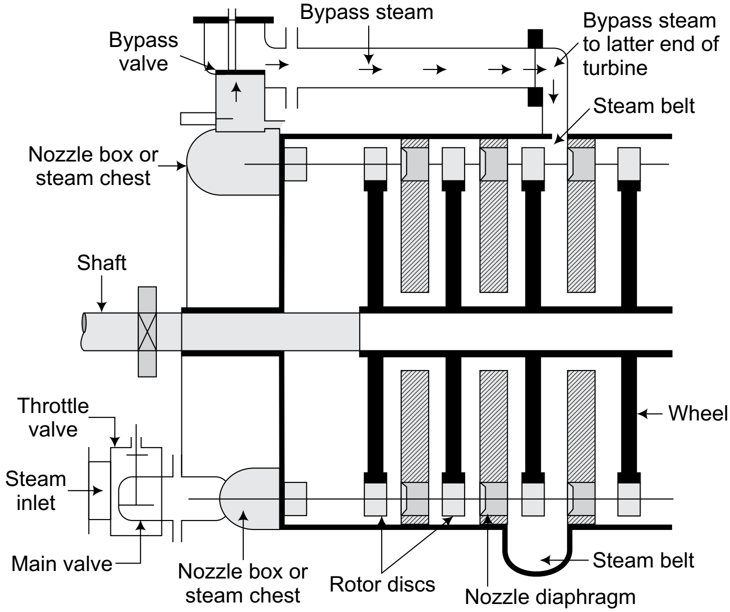


Fig. 2.21 Schematic View of Bypass Governing

Example 2.1 Steam flows from the nozzles of a single row impulse turbine with a velocity of 500 m/s in a direction which is inclined at an angle of 15° to the peripheral velocity. Steam comes out of the moving blades with an absolute velocity of 100 m/s and the direction at 120° with the direction of blade motion. The blades are equiangular and steam flow rate is 7.5 kg/s. Calculate (i) the power developed, (ii) power loss due to friction, and (iii) blade efficiency. Solve the problem analytically.

Solution Refer Figure 2.22.

$$CB = V_{f1} = V_1 \sin \alpha_1$$

$$= 500 \times \sin 15^\circ = 129.41 \text{ m/s}$$

$$CD = V_1 \cos \alpha_1 - u$$

$$= 500 - \cos 15^\circ - u = (482.96 - u) \text{ m/s}$$

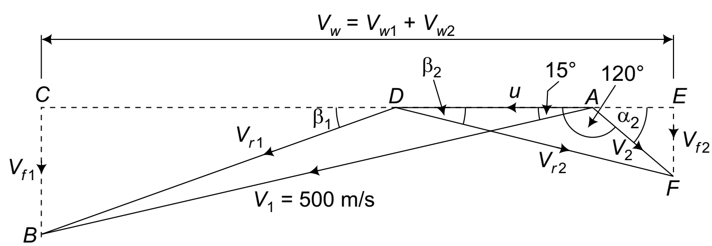


Fig. 2.22

Example 2.2 In a De-Laval turbine the steam enters the wheel through a nozzle with a velocity of 550 m/s and at an angle of 20° to the direction of motion of the blade. The blade speed is 250 m/s and the exit angle is 24°. By following the graphical method, find (i) the inlet angle of moving blade, (ii) exit velocity of steam and its direction, (iii) power developed per kg of steam, and (iv) the diagram efficiency.

Solution Following are the steps to draw the velocity diagram:

1. Choosing scale, 1 cm = 50 m/s

   $\therefore$ $u = (200/50) = 4$ cm

   and $V_1 = (500/50) = 10$ cm

2. Draw $AB = u = 4$ cm. Draw line AC at 20° ($\because \alpha_1 = 20°$), cut this line $V_1 = 10$ cm.
3. Join BC which represents V_{r1}. Measure BC and multiply it by scale to get V_{r1}.
4. Draw line at B at 24° ($\because \beta_2 = 24°$). With B as centre and radius BC swing an arc meeting line through B at D. Join BD which represents V_{r2}.
5. Join AD which represents V_2.
6. From points C and D draw perpendiculars meeting line AB produced at E and F respectively. CE represents V_{f1}, and DF represents V_{f2}. AE represents V_{w1}, and AF represents V_{w2}.

This completes the velocity diagram as shown in Figure 2.23.

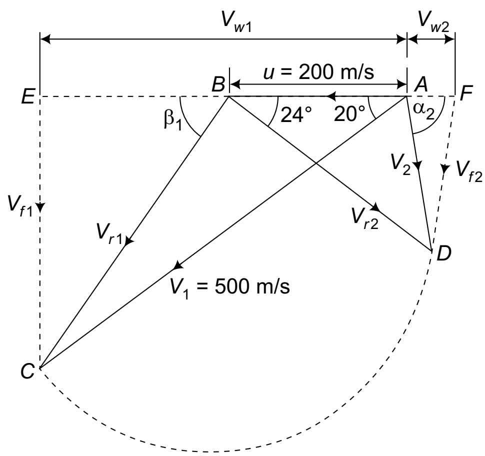


Fig. 2.23 (Not to scale)

(i) Inlet angle of the moving blade (β_1): By measurement from the velocity triangle,

we found, $\angle EBC = \beta_1 = 32°$

(ii) Exit velocity of steam (V_2) and its direction (α_2): By measurement from the velocity triangle,

we get, $AD = 3.2$ cm

$$= \frac{1 \times 200 \times (420 + 25)}{1000} = 89 \text{ kW}$$

(iv) Diagram efficiency: By measurement from the velocity triangle,

$$V_w = (V_{w1} + V_{w2})$$

$$= (420 + 25) = 445 \text{ m/s}$$

Diagram efficiency, $$\eta_b = \frac{2 \times u \times V_w}{V_1^2}$$

$$= \frac{2 \times 200 \times 445}{450^2} = 0.8790 \text{ or } 87.90\%$$

(v) Axial thrust per kg of steam (F_a): By measurement from the velocity triangle, we get,

$$EC = 3.1 \text{ cm}$$

$\therefore$ $$V_{f1} = 3.1 \times 50 = 155 \text{ m/s}$$

and $$FD = 2 \text{ cm}$$

$\therefore$ $$V_{f2} = 2 \times 50 = 100 \text{ m/s}$$

$$F_a = m \times (V_{f1} - V_{f2})$$

$$= 1 \times (155 - 100) = 50 \text{ V}$$

Example 2.4 In a stage of an impulse turbine provided with a single row wheel, the mean diameter of blade ring is 0.80 m and the speed of rotation is 3000 rpm. The steam issues from the nozzle with a velocity of 300 m/s and the nozzle angle is 20°. The blades are equiangular and blade coefficient is 0.85. Determine the power developed in the blades when the axial thrust on the blade is 150 N. Also determine the diagram efficiency.

Solution Refer Figure 2.25.

$$u = \pi \times d \times (N/60)$$

$$= \pi \times 0.80 \times (3000/60) = 125.66 \text{ m/s}$$

$$V_{w1} = V_1 \cos \alpha_1$$

$$= 300 \times \cos 20° = 281.91 \text{ m/s}$$

$$BE = V_{w1} - u = 281.91 - 152.66$$

$$= 156.25 \text{ m/s}$$

$$CE = V_{f1} = \sqrt{V_1^2 - V_{w1}^2}$$

$$= \sqrt{(300)^2 - (281.91)^2} = 102.59 \text{ m/s}$$

$$\tan \beta_1 = \frac{CE}{BE} = \frac{102.59}{156.52} = 0.6554$$

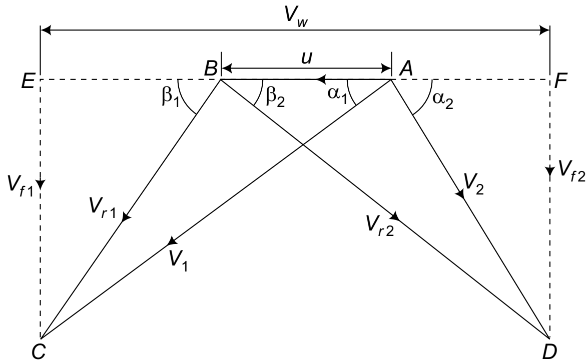


Fig. 2.25

$$\beta_1 = 33.20°$$

$$\therefore \quad \beta_1 = \beta_2 = 33.20°$$

$$V_{r1} = \sqrt{V_{f1}^2 + BE^2}$$

$$= \sqrt{(102.59)^2 + (165.25)^2} = 186.92 \text{ m/s}$$

But $$K = 0.85 = \frac{V_{r2}}{V_{r1}},$$

$$\therefore \quad V_{r2} = 0.85 \times 186.92 = 158.88 \text{ m/s}$$

$$\therefore \quad V_{w2} = V_{r2} \cos \beta_2 - u$$

$$= 158.88 \times \cos (33.20) - 125.66 = 7.29 \text{ m/s}$$

$$V_{f2} = V_{r2} \times \sin \beta_2$$

$$= 158.88 \times \sin (33.20°) = 86.99 \text{ m/s}$$

$$F_a = m \times (V_{f1} - V_{f2})$$

$$\therefore \quad m = \frac{F_a}{(V_{f1} - V_{f2})}$$

$$= \frac{150}{(156.52 - 86.99)} = 2.16 \text{ kg/s}$$

$$\text{Blade power} = \frac{m \times u \times (V_{w1} + V_{w2})}{1000}$$

$$= \frac{2.16 \times 125.66 \times (281.91 + 7.29)}{1000}$$

$$= 78.49 \text{ kW}$$

$$\eta_b = \frac{2 \times u \times V_w}{V_1^2}$$

$$= \frac{2 \times 125.66 \times (281.91 + 7.29)}{300^2}$$

$$= 0.8076 \text{ or } 80.76\%$$

Example 2.5 Steam at 350 m/s is supplied to the single stage impulse turbine through a nozzle. The nozzle angle is 25°. The mean diameter of the blade rotor is 1.0 m and it has a speed of 2000 rpm. Find the blade angles if there is no axial thrust. If the steam flow rate is 5 kg/s and the blade coefficient is 0.85 then determine the power developed. Also determine the blade efficiency and energy lost in the blade friction.

Solution Refer Figure 2.25.

$$u = \pi \times d \times (N/60)$$

$$= \pi \times 1.0 \times (2000/60) = 104.72 \text{ m/s}$$

$$V_{w1} = V_1 \cos \alpha_1 = 350 \times \cos 25°$$

$$= 317.21 \text{ m/s}$$

$$BE = V_{w1} - u = 317.21 - 104.72$$

$$= 212.49 \text{ m/s}$$

$$CE = V_{f1} = V_{f2} = \sqrt{V_1^2 - V_{w1}^2}$$

$$= \sqrt{(350)^2 - (317.21)^2} = 147.91 \text{ m/s}$$

$$V_{r1} = \sqrt{V_{f1}^2 + BE^2}$$

$$= \sqrt{(147.91)^2 + (212.49)^2} = 258.90 \text{ m/s}$$

$$\tan \beta_1 = \frac{CE}{BE} = \frac{147.91}{212.49}$$

$$= 0.6961$$

$$\therefore \quad \beta_1 = 34.84°$$

$$K = 0.85 = \frac{V_{r2}}{V_{r1}}$$

$$\therefore \quad V_{r2} = 0.85 \times 258.90 = 220.06 \text{ m/s}$$

$$\sin \beta_2 = \frac{V_{f2}}{V_{r2}} = \frac{157.91}{220.06} = 0.6721$$

$$\therefore \quad \beta_2 = 42.23°$$

$$\therefore \quad V_{w2} = V_{r2} \cos \beta_2 - u$$

$$= 220.06 \times \cos(42.23) - 104.72 = 58.22 \text{ m/s}$$

$$\text{Power developed} = \frac{m \times u \times (V_{w1} + V_{w2})}{1000}$$

$$= \frac{5 \times 104.72 \times (317.21 + 58.22)}{1000} = 196.58 \text{ kW}$$

Blade efficiency,
$$\eta_b = \frac{2 \times u \times V_w}{V_1^2}$$

$$= \frac{2 \times 104.72 \times (317.21 + 58.22)}{350^2}$$

$$= 0.6419 \text{ or } 64.19\%$$

Power loss due to friction

$$= \frac{m \times \left(V_{r1}^2 - V_{r2}^2\right)}{2 \times 1000}$$

$$= \frac{5 \times (258.90^2 - 220.06^2)}{2 \times 1000}$$

$$= 46.51 \text{ kW}$$

Example 2.6 In a stage of Parson's reaction turbine the mean diameter of the blade ring is 80 cm and its speed is 1500 rpm. The steam velocity at the inlet of moving blades is 165 m/s and the outlet blade angle is 20°. The mass flow rate of steam is 6.5 kg/s and the stage efficiency is 85%. Determine (i) blade inlet angle, (ii) power developed, and (iii) available isentropic enthalpy drop.

Solution Refer Figure 2.26.

$$u = \pi \times d \times (N/60)$$

$$= \pi \times 0.80 \times (1500/60) = 62.83 \text{ m/s}$$

In Parson's reaction turbine,

$$V_{f1} = V_{f2} = V_f,$$

$$\beta_1 = \alpha_2 \quad \text{and} \quad \beta_2 = \alpha_2$$

$$V_{w1} = V_1 \cos \alpha_1$$

$$= 165 \times \cos 20° = 155.05 \text{ m/s}$$

$$V_{f1} = V_{f2} = V_f = V_1 \sin \alpha_1$$

$$= 165 \times \sin 20° = 56.43 \text{ m/s}$$

$$BE = V_{w1} - u$$

$$= 155.05 - 62.83 = 92.22 \text{ m/s}$$

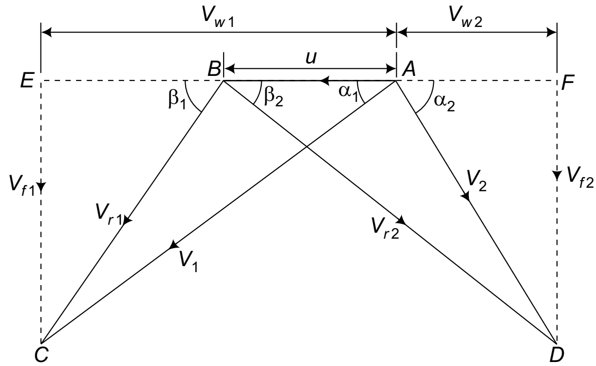


Fig. 2.26

$$\tan \beta_1 = \frac{CE}{BE} = \frac{56.43}{92.22} = 0.6119$$

$$\beta_1 = \alpha_2 = 31.46°$$

$$\tan \alpha_2 = \frac{V_{f2}}{V_{w2}}$$

or
$$V_{w2} = \frac{V_{f2}}{\tan \alpha_2} = \frac{56.43}{\tan (31.46)} = 92.23 \text{ m/s}$$

$$P = \frac{m \times u \times (V_{w1} + V_{w2})}{1000}$$

$$= \frac{6.5 \times 62.83 \times (155.05 + 92.23)}{1000}$$

$$= 100.99 \text{ kW}$$

$$\eta_s = \frac{\text{Work done by turbine}}{\text{Isentropic enthalpic drop}}$$

$$= \frac{P}{\Delta h_{isn}}$$

∴
$$\Delta h_{isn} = \frac{P}{\eta_s} = \frac{100.99}{0.85}$$

$$= 118.81 \text{ kW}$$

Example 2.7 In a stage of impulse turbine the blade angles are equal and the nozzle angle is 20^{o}. The velocity coefficient of the blade is 0.85. Find the maximum blade efficiency. If the actual blade efficiency is 90 per cent of maximum blade efficiency, determine the blade speed ratio.

Solution $\beta_1 = \beta_2$, $\alpha_1 = 20°$, and $K = 0.85$

1. **Condenser** It is a closed vessel heat exchanger in which exhaust steam from the turbine is condensed. During the condensation process the steam gives up heat to circulating cooling water.
2. **Condensate Extraction Pump** It is a pump required to extract the condensate from the condenser to the hot well.
3. **Air Extraction Pump** It is a pump to remove air and non-condensable gases. Sometimes a single pump known as wet air pump is used to remove both air and the condensate.
4. **Hot Well** It is a sump provided between the condenser and the boiler. In a hot well the condensate pumped by the condensate extraction pump is collected.
5. **Boiler Feed Pump** It is a pump used to force the condensate from hot well into the boiler. This is done by increasing the pressure of the condensate above the boiler pressure.
6. **Condenser Cooling Water Pump** It is a pump used to maintain the circulation of cooling water through the condenser.
7. **Make Up Water Pump** It is a pump used to compensate the water lost due to evaporation during cooling of hot water in the cooling tower. It circulates the required amount of cooling water into the condenser from the river or some other water source.
8. **Cooling Tower or Spray Pond** A cooling tower is used to re-cool the hot water coming out of the condenser. The hot water is cooled by evaporative cooling in which heat is rejected to the atmospheric air. The use of cooling tower becomes essential when there is shortage of cooling water, and it has to be used over and over again in the condenser.
9. **Relief Valve** It is used to relieve the steam from the condenser when the condenser does not work properly by which the plant becomes non-condensing.

2.15 TYPES OF STEAM CONDENSERS

The steam condensers are broadly classified into two types, namely, jet condenser (or mixing type condenser), and surface condenser (or non-mixing type condenser).

1. Jet Condensers or Mixing Type Condensers In jet condensers there is direct contact between the exhaust steam and cooling water. The temperature of the condensate is same as that of the cooling water leaving the condenser. Heat exchange occurs by direct conduction between the steam and water. If the cooling water is not pure and free from harmful impurities then the condensate cannot be reused as feed water to the boilers. Due to loss of condensate and high power requirement by the pump these condensers are rarely used in modern steam power plants.

Jet condensers are of the following types:

(i) Low-level jet condensers: These are of the following two types
   (a) Parallel flow type: The steam and cooling water flow in the same direction.
   (b) Counter flow type: The steam and cooling water flow in the opposite directions.

(ii) High-level jet condensers, and

(iii) Ejector jet condensers.

2. Surface Condensers or Non-mixing Type Condensers In surface condensers, there is no direct contact between the exhaust steam and the cooling water. The steam surrounds the tubes fitted in the condenser shell and the cooling water circulates through these tubes. The steam gets condensed due to the heat transfer to cooling water by conduction and convection. The condensate collected from these condensers is reused as feed water in the boiler. Thus, these condensers are most suitable for modern steam power plants and chemical industries. These are generally used where a large quantity of inferior water is available and better quality of feed water is to be supplied to the boiler. So, these condensers are universally used in marine engines where seawater is used for cooling purposes. The only drawback of these condensers is its high initial cost but is recovered by the saving in running cost.

The surface condensers may be classified according to:

(i) Direction of condensate flow and tube arrangement:
   (a) Down flow type,
   (b) Central flow type, and
   (c) Inverted flow type.

(ii) The number of water passes: single pass, double pass, or multi pass, and

(iii) Evaporative type.

2.15.1 Low-level Jet Condensers

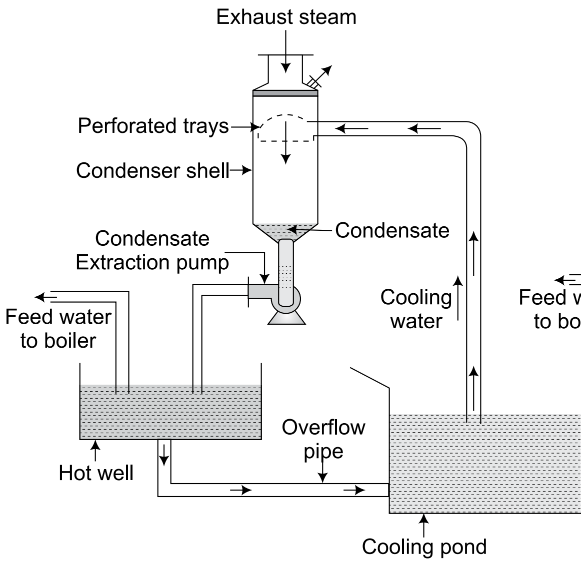
Low-level jet condenser is placed at low levels such that vacuum inside the condenser draws the cooling water into it from the cooling water source. Parallel flow and counterflow low-level jet condensers are shown in Figure 2.30.

In counterflow jet condenser the water and steam flows in opposite directions but in parallel flow both flow in the same direction. In both the cases the cooling water enters at the top of the condenser and passes through the perforated trays so that it breaks into sprays and increases the heat transfer rate by providing more contact surface area.

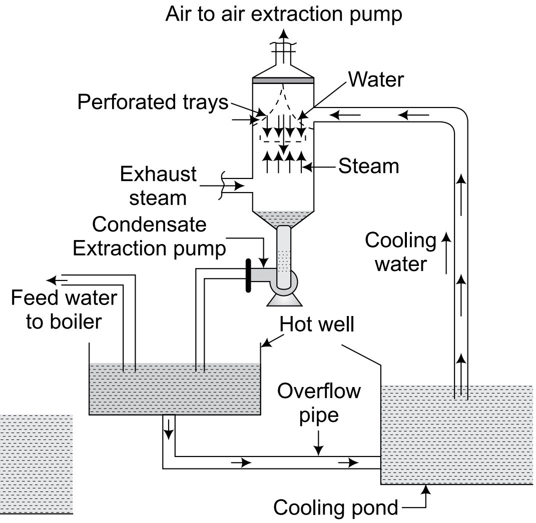
When the steam comes into contact with cooling water it gets condensed. The extraction of air is done from the top of the condensers. The vacuum created in the condenser is sufficient to draw the cold water from the cooling pond. The pressure causing the water flow from the cooling pond (or cold tank) to the condenser top is given by $(p_a - p_c)$, where p_a is the atmospheric pressure and p_c is the condenser pressure. The condensate is extracted by the extraction pump and is discharged to the hot well. The excess amount of condensate from hot well flows into the cooling pond by an overflow pipe and the remaining water is pumped to the boiler as feed water. These condensers have the disadvantage of flooding the turbine if water extraction pump fails due to any reason.

2.15.2 High-level Jet Condensers

A high-level jet condenser is also called barometric condenser because the condenser shell is placed above the hot well by more than barometric height of water column of 10.363 m. A long tailpipe, more than 10.363 m in length is attached between the bottom of the condenser and the



(a) Parallel flow type low-level jet condenser



(b) Counter flow type low-level jet condenser

Fig. 2.30 Low-level Jet Condensers

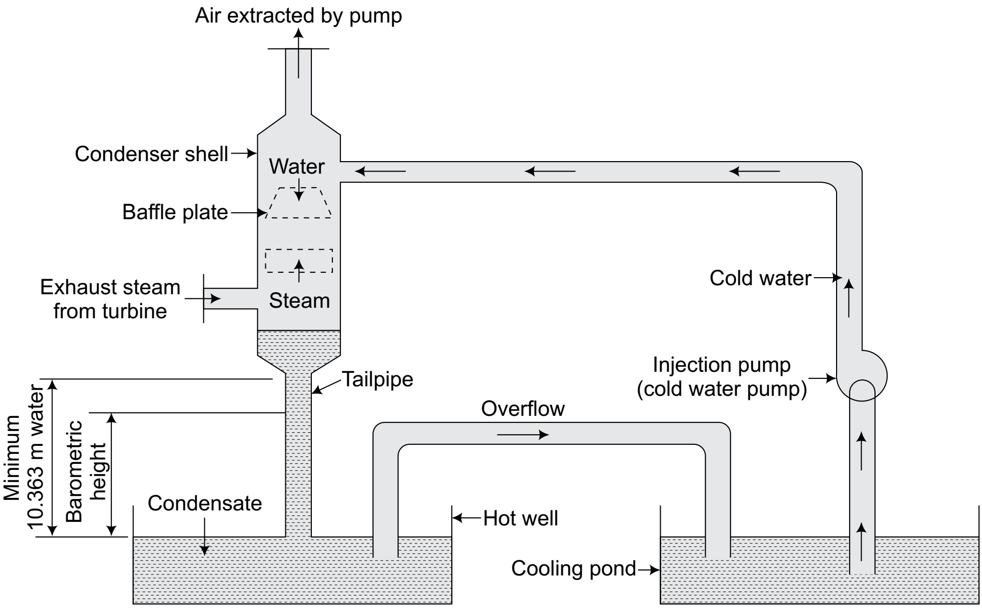


Fig. 2.31 High-level Jet Condenser

hot well. The pressure at the bottom of the pipe is equal to atmospheric pressure, whereas at its top in the condenser shell vacuum pressure is maintained. It avoids the rise of water in the tail pipe and water extraction pump is also not required. The condensate and water from the condenser go down to the hot well under the gravity and maintain a water leg in the tail pipe depending upon the vacuum in the condenser. As the height of the shell is large so, an injection pump is required to pump water to the top of the shell. A schematic view of high jet condenser is shown in Figure 2.31.

The working and other details of this condenser are similar to low-level counter flow jet condenser. Its drawbacks are high costs and loss of vacuum between the turbine and the condenser. It is used where sufficient head required for tailpipe is available.

2.15.3 Ejector Condenser

A schematic view of an ejector condenser is shown in Figure 2.32. It is suitable for moderate vacuum only.

The cooling water under the head of about 5 m to 6 m enters the condenser from its top. It is discharged through a series of convergent nozzles in which the potential energy get converted into the kinetic energy and thus partial vacuum is created. The exhaust steam enters the condenser through a non-return valve. It condenses through the mixing with cooling water and thus vacuum further increases. The condensate after passing through the series of convergent nozzles passes through the divergent nozzle. When it passes through the divergent nozzle the kinetic energy of condensate reconverts into potential energy and thus a higher pressure than the atmospheric pressure is obtained which forces out the condensate to the hot well. Thus, in this condenser no air extraction pump is required. A non-return valve is fitted at the exhaust steam pipe to the condenser to prevent any rush of water from the hot well to the turbine in the case of cooling water failure.

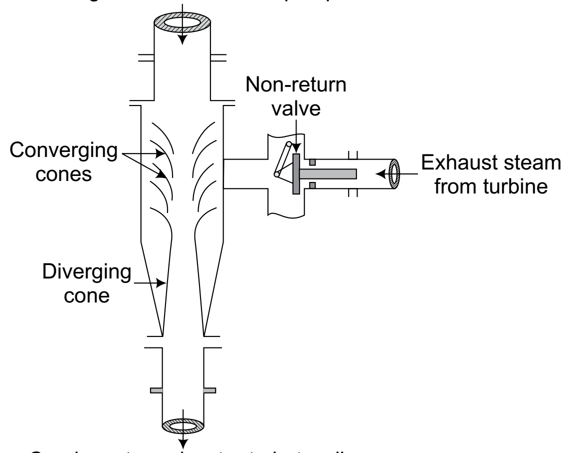


Fig. 2.32 Ejector Condenser

2.15.4 Surface Condenser

A schematic view of the surface condenser is shown in Figure 2.33.

It consists of an airtight cylindrical shell closed at each end. A number of parallel brass water tubes are fitted in the tube plates which are fixed between each cover head and the shell. The space between the tube plates and the cover head is known as water box. A baffle plate partitions the water box into two sections. The cooling water enters the shell at the lower half section and flows in one direction. Then it returns in the opposite direction through the upper half section of

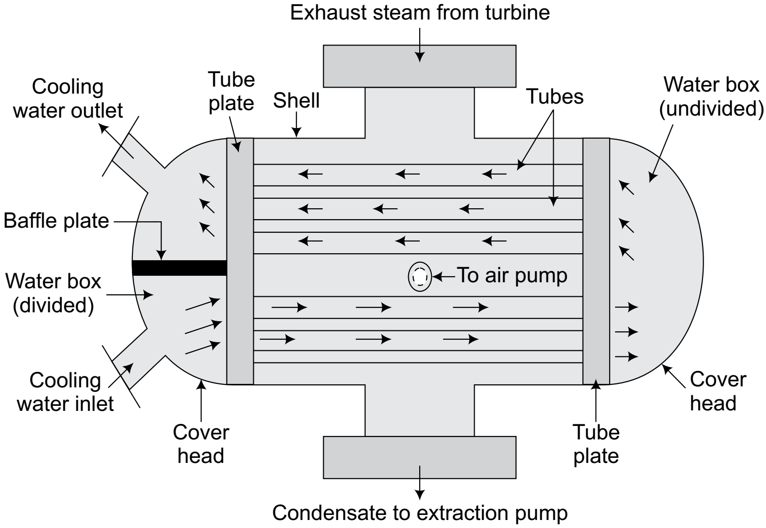


Fig. 2.33 Surface Condenser

the condenser and finally, it leaves through the outlet. In this type of condenser the water traverses two times, so it is called the two-pass condenser. The exhaust steam from the turbine enters at the top of the condenser and surrounds the tubes through which cooling water flows under force. The steam condenses when it comes in contact with the cold surface of the tubes. The water gets warmed in the condenser is discharged into the cooling tower, river, or cooling pond. The condensate is taken out from the condenser by a condensate extraction pump. The air is removed by an air extraction pump.

Down Flow Surface Condenser Figure 2.34 (a) shows a sectional view of a down flow surface condenser which is also known as dry vacuum type condenser. The steam enters at the top and flow downwards over the tubes carrying cooling water. A section of tubes near the air pump suction is screen off by providing a baffle plate. This screen is also called air cooler which helps in extracting air at a lower temperature than the condensate. The low temperature of air reduces its volume and hence the size of the pump gets reduced by as much as 50 per cent.

Central Flow Surface Condenser Figure 2.34 (b) shows a sectional view of a central flow surface condenser in which the air extraction pump is provided at the centre and the air is extracted from the centre. The exhaust steam and air enter from the top and flow radially towards the centre by passing over the entire periphery of the tubes. The condensate is extracted at the bottom by condensate extraction pump. This condenser is an improvement over the down flow type as steam has an access to the whole periphery of the cooling tubes.

Inverted Flow Surface Condenser In the inverted flow type surface condenser the air suction pump is provided at the top. So, the steam entering from the bottom of the condenser flows upwards. The condensed steam collects in the bottom section from where it is extracted with the help of a condensate extraction pump.

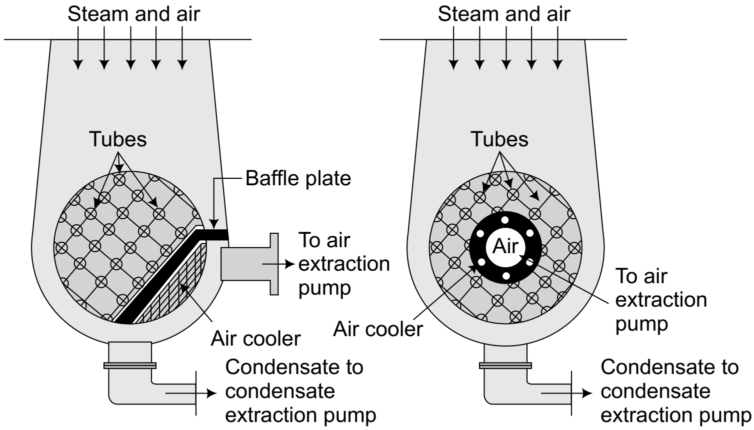


Fig. 2.34 (a) Down Flow Surface Condenser (b) Central Flow Surface Condenser

2.15.5 Evaporative Condenser

A schematic arrangement of the evaporative condenser is shown in Figure 2.35.

These are used where there is scarcity of water. The exhaust steam from the turbine enters a coiled finned pipe. The water from the cooling pond is pumped by means of a pump to a horizontal header which is provided with spray nozzles. The sprayed water forms thin film over the pipe surface and gets evaporated in passing over the pipe under a small partial pressure and thus cools the steam inside the pipes. The air is drawn over the surface of finned pipe with the help of induced draft fan to increase the evaporation of cooling water which further increases the condensation of steam in pipes. Eliminators are provided to prevent the exit of water vapors with the leaving heated air. The cooling water gets collected into the cooling pond. In the cooling pond the water lost due to evaporation is replenished by the addition of required amount of cold make up water.

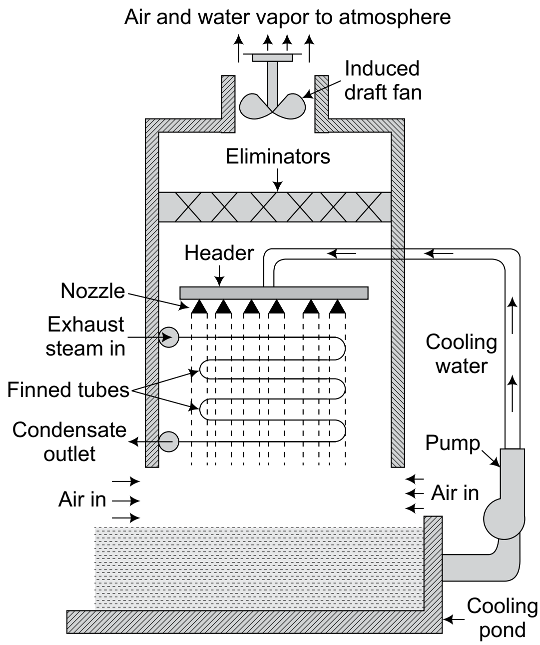


Fig. 2.35 Evaporative Condenser

The arrangement of this type of condenser is simple, cheap, and requires small quantity of cooling water thus capacity of water circulating pump is small. The vacuum maintained in this type of condenser is not as high as in the case of surface condensers. These condensers are used in small power plants and are extensively used in refrigeration plant units.

2.16 REQUIREMENTS OF A GOOD SURFACE CONDENSER

The following are the requirements of a good surface condenser:

1. The steam should enter the condenser with least possible resistance for its easy flow.
2. For effective condensation the steam should be well distributed in the vessel and there should be minimum pressure drop.
3. The circulating cooling water should flow through the tubes with least friction. The rise in temperature of cooling water should be limited to 10°C for obtaining better thermal efficiency.
4. There should be no undercooling of the condensate, so the steam should lose only its latent heat to the circulating water. This is made possible by regulating the quantity of circulating water in such a way that its exit temperature is same as the saturation temperature of steam.
5. To obtain maximum heat transfer rate the tubes should be made of high thermal conductivity material. The water should flow through tubes and steam outside so that the outer surface of the tubes does not get deposited with sediments. If the cooling water is dirty then sediments will get deposited inside the tubes. These sediments can be cleaned by motor driven brushes after removing the end cover plates.
6. There should be no leakage of air from the condenser.
7. Minimum energy should be spent to extract the air from the condenser. This is achieved by fitting a baffle plate at the coolest section where air pump is fitted. This arrangement reduces the specific volume of air and thus reduces the size of the pump.

2.17 ADVANTAGES AND DISADVANTAGES OF CONDENSERS

2.17.1 Jet Condensers

The following are the advantages and disadvantages of jet condensers.

Advantages

1. Due to more intimate mixing of steam and cooling water it requires less quantity of circulating water for the condensation of steam.
2. Due to direct mixing it requires less building space.
3. The arrangement of jet condenser is simple in construction and low in cost. Its maintenance cost is also low.
4. Low-level jet condenser does not require cooling water pump. In barometric and ejector condensers there is no need of condensate extraction pump.

Disadvantages

1. There is wastage of condensate.
2. If the condensate is to be used as feed water then the cooling water should be pure and free from any harmful impurities.

3. In the barometric condenser use of long pipe increase the cost of the condenser.
4. In the low-level jet condenser if the condensate extraction pump fails then there is greater possibility of flooding of the engine.
5. In the case of barometric condenser a vacuum loss of about 1 to 1.5 cm of Hg occurs due to leakage in the long exhaust pipe line.
6. Vacuum more than 66 cm of Hg cannot be achieved. This is because the dissolved air in the cooling water gets liberated at low pressures.
7. The air extraction pump needs high power which may be about double the power required by a surface condenser.

2.17.2 Surface Condensers

The following are the advantages and disadvantages of surface condensers.

Advantages

1. A high vacuum can be achieved (as much as 73.5 cm of Hg) and thus gives greater plant efficiency.
2. Since the cooling water and steam do not mix, the condensate is recovered and can be used as feed water to the boiler. Due to this advantage, these condensers are used in all steam power plants.
3. Since the cooling water and steam do not mix, any kind of cooling water can be used. This results in considerable reduction in the cost of water softening plant.
4. The chances of vacuum loss are minimized.
5. It requires much less power to run the air extraction pump and for water pumping.
6. It ideally suits high capacity plants.
7. It requires less quantity of make up water (about 4% to 5%).

Disadvantages

1. The system is bulky and requires large floor area.
2. It requires high capital cost and maintenance cost.
3. It requires more cooling water.

2.18 COMPARISON OF JET AND SURFACE CONDENSERS

Jet Condenser	Surface Condenser
1. Cooling water and steam mix together.	1. Cooling water and steam do not mix together.
2. Condensate contains impurity, so it cannot be reused as feed water.	2. Condensate is pure and free from any harmful impurities, so it can be reused as feed water.
3. It requires less quantity of cooling water for condensation of steam due to direct mixing of water with steam.	3. It requires more quantity of cooling water for condensation of steam since there is no mixing.

4. The cooling water should be free from impurities.	4. The cooling water of any quality can be used.
5. Vacuum efficiency is low. Thus, not suitable for high capacity plants.	5. Vacuum efficiency is high. Thus, more suitable for high capacity plants.
6. The system design is simple, compact, and economical.	6. The system design is complicated, bulky, and costly.
7. It requires small floor area.	7. It requires large floor area.
8. Maintenance is simple and less costly.	8. Maintenance is costly and requires skilled worker.
9. It requires more power for air pump.	9. It requires less power for air pump.
10. It requires more power for water pumping.	10. It requires less power for water pumping.
11. It does not require condensate extraction pump.	11. It requires condensate extraction pump.
12. Vacuum more than 66 cm of Hg cannot be achieved.	12. A high vacuum as much as 73.5 cm of Hg can be achieved.

2.19 CONDENSER VACUUM AND ITS MEASUREMENT

The pressure inside the condenser is below the atmospheric pressure and is called vacuum pressure. The vacuum in the condenser is the difference between the barometric pressure and the absolute pressure in the condenser. The difference between the barometer reading and the vacuum gauge reading gives the absolute pressure in the condenser as shown in Figure 2.36.

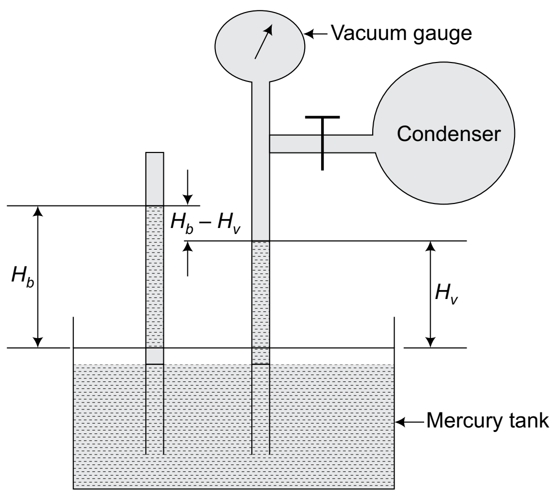


Fig. 2.36 Vacuum Measurement

Let the vacuum in the condenser be equivalent to H_v cm of mercury which is measured by vacuum gauge and the barometric pressure is equivalent to H_b cm of mercury.

Therefore, absolute pressure = $(H_b - H_v)$ cm of Hg

We know that 76 cm of Hg = 1.01325 bar

Therefore, pressure equivalent to 1 cm of Hg = (1.01325/76) = 0.0133322 bar

If it is desired to know the absolute pressure in the condenser then the barometric reading and gauge reading must be known. If barometric reading is 76 cm of Hg and gauge reading is 68 cm of Hg then the absolute pressure in the condenser will be

$$= (76 - 68) = 8 \text{ cm of Hg}$$

$$= 8 \times 0.0133322 = 0.10666 \text{ bar.}$$

The barometric head H_b varies according to atmospheric conditions, and hence the absolute pressure in the condenser is a function of the barometric pressure.

If the barometric reading (atmospheric pressure) is 76 cm of Hg at sea-level then the corrected vacuum in the condenser is given by,

$$= 76 - \text{absolute pressure in the condenser in cm of Hg}$$

$$= 76 - (H_b - H_v) \tag{2.39}$$

2.20 SOURCES OF AIR IN THE CONDENSER, ITS EFFECTS AND REMOVAL

Sources of Air Leakage into the Condenser Following are the sources of air leakage into the condenser:

1. As the pressure inside the condenser is below atmospheric, so the air tends to leak into it through the joints, through the vents from the atmospheric relief valve and other accessories.
2. The air dissolved in exhaust steam is liberated at low pressure.
3. The dissolved air in the water is also carried with the cooling water in jet condensers which gets liberated at low pressure. The quantity of air carried with cooling water is about 0.005% of the cooling water.

In a properly maintained surface condenser for a steam turbine plant the amount of air leakage is about 5 kg per 10,000 kg of steam condensed, whereas for steam engine plants the air leakage is about 15 kg per 10,000 kg of steam.

Effects of Air on the Performance of Condenser The presence of air in the condenser affects its performance in the following ways:

1. It increases the condenser pressure or back pressure of the turbine (or engine) and reduces the work output.
2. Due to lower thermal conductivity of air the rate of heat transfer also lowers, thus the surface area of the condenser tubes is to be increased for a condenser duty.
3. The presence of air decreases the partial pressure of steam, so at low pressure the steam has more latent heat. To remove this increased amount of heat more amount of cooling water is to be supplied.

4. The air presence causes reduction in the rate of condensation of steam because the extraction of heat by circulating water is partly from steam and partly from air.
5. An air extraction pump is required to maintain vacuum in the condenser. In spite of shielding some quantity of steam escapes with the air which reduces the amount of the condensate. Moreover, the condensate gets under-cooled, so more heat is to be supplied to the feed water in the boiler.
6. Air present in the condenser increases the corrosive action.

Procedure to Check Air Leakage Run the plant till the pressure and temperature in the condenser become steady. Then isolate the condenser from the rest of the plant by shutting off the steam supply and stopping the air extraction and condensate pumps. If the condenser has any leakage then the readings of vacuum gauge and thermometer will fall.

Methods to Check Air Leakage The point of air leakage can be located by any one of the following methods:

(i) **Soap Bubble Method** The condenser is filled with air at pressure and then its effect on soap water is noted at the locations where infiltration is likely to occur. At the point of air leakage bubbles can be seen.

(ii) **Candle Flame Method** When the condenser is under vacuum then the large leakages can be detected by passing a candle flame over suspected locations.

(iii) **Peppermint Oil Method** Peppermint oil is applied over suspected locations when the condenser is working. If any leakage is there then oil fumes will enter into the condenser and there will be oil odour in the discharge from the condenser.

2.21 DALTON'S LAW OF PARTIAL PRESSURES

The law states that the total pressure exerted by a non-reactive mixture of gases or a mixture of gas and vapor is equal to the sum of partial pressure of the individual constituents of the mixture at the same temperature (T).

The partial pressure of each constituent of the mixture is the pressure exerted by the constituent taken separately in the same volume (V) of the vessel as that of mixture and at the same temperature. This is explained by taking the following example as shown in Figure 2.37.

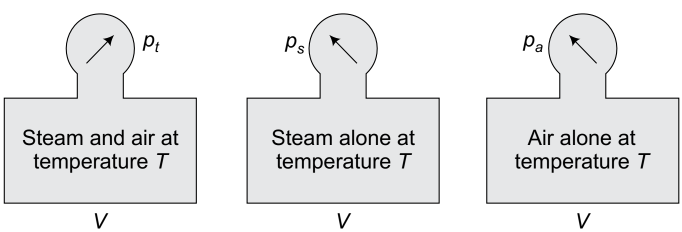


Fig. 2.37 Dalton's Law of Partial Pressure

The total pressure (p_t) in the condenser or vessel is the sum of the partial pressures of steam (p_s) and air (p_a).

According to Dalton's law of partial pressure,

$$p_t = p_s + p_a$$

or

$$p_a = p_t - p_s$$

The vacuum gauge reads the total pressure in the condenser, i.e., p_t. The steam pressure p_s can be known from the steam table corresponding to condenser temperature. As p_t and p_s are known thus p_a can be determined.

2.22 VACUUM EFFICIENCY

The vacuum efficiency is a measure of the degree of perfection of maintaining a desired vacuum in the condenser. The ideal vacuum means the vacuum due to steam alone when air is absent. Under such a condition total pressure in the condenser will reach to the pressure of steam corresponding to the saturation temperature of the steam.

Let

p_s = Saturation pressure of steam in bar corresponding to the temperature of water entering the condenser,

p_t = Total pressure of air and steam

$= p_s + p_a$, and

p_b = Barometric pressure or atmospheric pressure

$$\text{Ideal vacuum} = p_b - p_s \tag{2.40}$$

$$\text{Actual vacuum} = p_b - p_t$$

$$= p_b - (p_a + p_s) \tag{2.41}$$

The vacuum efficiency (η_{vacuum}) is defined as the ratio of actual vacuum recorded by the vacuum gauge to the ideal vacuum.

$$\eta_{\text{vacuum}} = \frac{\text{Actual vacuum recorded by gauge}}{\text{Ideal vacuum}}$$

$$= \frac{p_b - (p_a + p_s)}{p_b - p_s} = \frac{p_b - p_t}{p_b - p_s} \qquad \text{`(2.42)}$$

If there is no air leakage into the condenser then $p_a = 0$ and hence the vacuum efficiency becomes 100%. The vacuum efficiency depends upon the effectiveness of the air cooling and the rate at which the air is removed by the air pump. Generally, the vacuum efficiency is about 98% to 99%.

2.23 CONDENSER EFFICIENCY

In an ideal condenser the steam should give only its latent heat to the circulating water so that temperature of the condensate becomes equal to the saturation temperature corresponding to the condenser pressure. It means there should be no undercooling of the condensate. Maximum temperature to which the cooling water can be raised is the condensate temperature. The condenser efficiency ($\eta_{\text{condenser}}$) is then defined as the ratio of actual rise in the temperature of the cooling water to the maximum possible rise, i.e.,

$$\eta_{\text{condenser}} = \frac{T_2 - T_1}{T_s - T_1} \tag{2.43}$$

where T_2 = Outlet temperature of cooling water, T_1 = Inlet temperature of cooling water, and T_s = Saturation temperature corresponding to condenser temperature.

Example 2.9 The following readings are recorded during a test on a steam condenser: vacuum in condenser = 71.5 cm of Hg, barometer reading = 76.5 cm of Hg, mean temperature of condenser = 33°C, hot well temperature = 29°C, inlet temperature of cooling water = 9°C, outlet temperature of cooling water = 26.5°C. Calculate (i) corrected vacuum to standard barometer, (ii) vacuum efficiency, (iii) under-cooling of the condensate, and (iv) condenser efficiency.

Solution

(i)

$$\text{Corrected vacuum} = 76 - (H_b - H_g)$$
$$= 76 - (76.5 - 71.5) = 71 \text{ cm of Hg}$$

(ii) The absolute pressure of steam corresponding to condenser temperature of 33°C (from steam table)

$$= 0.05029 \text{ bar}$$
$$= (0.05029/0.0133322) = 3.772$$

$$\therefore \qquad \eta_{\text{vacuum}} = \frac{\text{Actual vacuum recorded by gauge}}{\text{Ideal vacuum}}$$
$$= \frac{71.5}{76.5 - 3.772} = 0.9831 \text{ or } 98.31\%$$

(iii) Under cooling of condensate = (condenser temperature – hot well temperature)

Therefore, under-cooling of the condensate = (33 – 29) = 4°C

(iv) Absolute condenser pressure = (barometric reading – vacuum reading)

$$= (76 - 71.5) = 4.5 \text{ cm of Hg}$$
$$= 4.5 \times 0.0133322 = 0.05999 \text{ bar}$$

Saturation temperature corresponding to 0.05999 bar = 35.85°C (from steam table)

Therefore, the maximum temperature to which cooling water can be raised is 35.85°C.

$$\therefore \qquad \eta_{\text{condenser}} = \frac{T_2 - T_1}{T_s - T_1} = \frac{26.5 - 9}{35.85 - 9}$$

$$= 0.6518 \text{ or } 65.18\%$$

Example 2.10 In a condenser, vacuum gauge reads 71.5 cm of Hg while barometer reads 75.5 cm of Hg. The temperature of condenser is 25°C. Determine (i) the pressure of steam and air, (ii) mass of air per kg of steam, and (iii) corrected vacuum to standard barometer, and (iv) vacuum efficiency.

Solution

(i) Total absolute pressure (p_t) in the condenser

$$= (H_b - H_v) = (75.5 - 71.5) = 4 \text{ cm of Hg}$$

$$= 4 \times 0.0133322 = 0.05332 \text{ bar}$$

(ii) Partial pressure of steam (p_s) at condensate temperature of 25°C

$$p_s = 0.03166 \text{ bar (From steam table)}$$

Partial pressure of air,

$$p_a = p_t - p_s = 0.05332 - 0.03166$$

$$= 0.02166 \text{ bar}$$

Specific volume of steam = 43.402 m^3/kg. Air will also occupy the same volume.

Applying ideal gas equation,

$$p_a\, v_a = m_a\, R_a\, T_a$$

$$(0.02166 \times 10^5) \times 61.02 = m_a \times 287 \times (25 + 273)$$

$$\therefore \qquad m_a = 1.094 \text{ kg/kg of steam}$$

(iii) $$\text{Corrected vacuum} = 76 - (H_b - H_g) = 76 - (75.5 - 71.5)$$

$$= 72 \text{ cm of Hg}$$

(iv) From steam table, the absolute pressure of steam corresponding to condenser temperature of 25°C

$$= 0.03166 \text{ bar} = (0.03166/0.0133322)$$

$$= 2.375$$

$$\therefore \qquad \eta_{\text{vacuum}} = \frac{\text{Actual vacuum recorded by gauge}}{\text{Ideal vacuum}}$$

$$= \frac{71.5}{75.5 - 2.37}$$

$$= 0.9777 \text{ or } 97.77\%$$

Example 2.11 A condenser deals with 900 kg of steam per hour with a dryness fraction of 0.9. The temperature of condenser is 40°C. The air associated with the steam in the condenser is 200 kg/hour. Determine the vacuum reading while barometer reads 75.5 cm of Hg. Correct this vacuum to a standard barometer reading of 76 cm of Hg.

Solution From steam table, partial pressure of steam at 40°C,

$$p_s = 0.07375 \text{ bar, and}$$

Volume of one kg of steam at 40°C,

$$v_s = 19.55 \text{ m}^3$$

$$\therefore \quad \text{Total volume of steam} = m \times (x \times v_s) = 900 \times (0.9 \times 19.55)$$

$$= 15835.5 \text{ m}^3.$$

Air will also occupy the same volume.

$$p_a \, v_a = m_a \, R_a \, T_a$$

$$p_a \times 15835.5 = 200 \times 287 \times (40 + 273)$$

$$\therefore \quad p_a = 1134.55 \text{ N/m}^2 = 0.011345 \text{ bar}$$

$$\text{Total condenser pressure} = 0.07375 + 0.011345 = 0.085095 \text{ bar}$$

$$\text{or} \quad = (0.085095/0.0133322) = 6.38 \text{ cm of Hg}$$

$$\therefore \quad \text{Vacuum reading} = 75.5 - 6.38 = 69.12 \text{ cm of Hg}$$

$$\text{Corrected vacuum} = 76 - (75.5 - 69.12) = 69.62 \text{ cm of Hg}$$

Example 2.12 The following data are recorded during a test on a steam condenser: vacuum in condenser = 71 cm of Hg, barometer reading = 76.5 cm of Hg, mean temperature of condenser = 35°C, hot well temperature = 28°C, inlet temperature of cooling water = 8.5°C, outlet temperature of cooling water = 25.6°C, condensate collected = 1900 kg/hour, and cooling water quantity = 59,500 kg/hour. Calculate (i) corrected vacuum to standard barometer, (ii) vacuum efficiency, (iii) under-cooling of condensate, (iv) condenser efficiency, (v) quality of steam entering the condenser, and (iv) mass of air present per m^3 of the condenser volume and per kg of uncondensed steam.

Solution

(i) $$\text{Corrected vacuum} = 76 - (H_b - H_v)$$

$$= 76 - (76.5 - 71) = 70.5 \text{ cm of Hg}$$

(ii) The absolute pressure of steam corresponding to the condenser temperature of 35°C (from steam table)

$$= 0.05622 \text{ bar} = (0.05622/0.0133322)$$

$$= 4.22 \text{ cm of Hg}$$

$$\therefore \qquad \eta_{\text{vacuum}} = \frac{\text{Actual vacuum recorded by gauge}}{\text{Ideal vacuum}}$$

$$= \frac{71}{76.5 - 4.22} = 0.9823 \text{ or } 98.23\%$$

(iii) Under cooling of condensate = (condenser temperature – hot well temperature)

$$= (35 - 28) = 7°\text{C}$$

(iv) Absolute condenser pressure = (barometric reading – vacuum reading)

$$= (76.5 - 71) = 5.5 \text{ cm of Hg}$$

$$= 5.5 \times 0.0133322 = 0.07333 \text{ bar}$$

Saturation temperature corresponding to 0.05999 bar = 39.11°C (from steam table)

Therefore, the maximum temperature to which cooling water can be raised is 35.85°C

$$\therefore \qquad \eta_{\text{condenser}} = \frac{T_2 - T_1}{T_s - T_1} = \frac{25.6 - 8.5}{39.11 - 8.5}$$

$$= 0.5586 \text{ or } 55.86\%$$

(v) Heat lost by steam = Heat gained by water (neglecting heat received by air)

$$\therefore \qquad m_s\,(h_1 + x_1\,h_{fg1} - h_c) = m_w\,c_{pw}\,(T_2 - T_1) \qquad (h_c \text{ is } h_f \text{ value of condensate at } 28°\text{C})$$

$$1900 \times (166.89 + x_1 \times 2407.21 - 117.3) = 59500 \times 4.2 \times (25.6 - 8.5)$$

$$\therefore \qquad x_1 = 0.914$$

(vi)
$$p_a = p_t - p_s = 0.07333 - 0.05622$$

$$= 0.01711 \text{ bar}$$

To determine the mass of air per m^3 of the condenser at 0.01711 bar pressure and at 35°C applying ideal gas equation,

$$p_a\,v_a = m_a\,R_a\,T_a$$

$$0.01711 \times 10^5 \times 1 = m_a \times 287 \times (35 + 273)$$

$$\therefore \qquad m_a = 0.01936 \text{ kg/m}^3$$

The volume of one kg mass of uncondensed steam is given by v_s which is the specific volume of saturated steam at 35°C

So, volume of one kg of uncondensed steam = 25.25 m^3/kg (from steam table)

The mass of air in one kg of uncondensed steam is calculated by applying ideal gas equation,

$$0.01911 \times 10^5 \times 25.25 = m_a \times 287 \times (35 + 273)$$

$$\therefore \qquad m_a = 0.489 \text{ kg}$$

2.24 MASS OF COOLING WATER REQUIRED IN A CONDENSER

In a jet condenser the exhaust steam and the cold water mix directly, so the resulting temperature of the condensed steam and water is the same at its outlet. But in a surface condenser the temperature of the condensate and the circulating water may differ.

Let m = Quantity of circulating water in kg/h, m_s = Quantity of steam condensed in kg/h, T = Temperature of the wet exhaust steam in °C, T_c = Temperature of the condensate in °C, h_f = Sensible heat or enthalpy of water at T°C, h_{fg} = Latent heat or enthalpy of evaporation of steam at T°C, x = Dryness fraction of exhaust steam, T_1 = Inlet temperature of cooling water in °C, T_2 = Outlet temperature of cooling water in °C, h_c = Sensible heat or enthalpy of condensate, c_w = Specific heat of cooling water.

Now for a condenser,

Heat given up by steam = Heat absorbed by cooling water

For a surface condenser,

$$m_s \times (h_f + xh_{fg} - h_c) = c_{pw} \times m \times (T_2 - T_1)$$

$$\therefore \qquad m = \frac{m_s \times (h_f + xh_{fg} - h_c)}{c_{pw} \times (T_2 - T_1)} \tag{2.44}$$

For a jet condenser,

$$T_2 = T_c$$

$$\therefore \qquad m = \frac{m_s \times (h_f + xh_{fg} - h_c)}{c_{pw} \times (T_c - T_1)} \tag{2.45}$$

Example 2.13 A 175 kW steam engine consumes 9 kg of steam per kWh. The back pressure of the engine and the condenser pressure are equal to 0.15 bar. The temperature of the cooling water at the inlet and outlet are 15°C and 28°C respectively. The temperature of the condensate is 29°C. Determine the quantity of cooling water required per hour if the steam exhausted to the condenser is dry saturated.

Solution From steam table (Table A-1.2) at 0.15 bar,

$$h_f = 225.91 \text{ kJ/kg}, \qquad h_g = 2599.1 \text{ kJ/kg},$$

$$h_{fg} = (2599.1 - 225.91) = 2373.19 \text{ kJ/kg}$$

Cooling water/kg of steam,

$$m = \frac{(h_f + xh_{fg} - h_c)}{c_{pw} \times (T_2 - T_1)}$$

$$\therefore \qquad m = \frac{(225.91 + 1 \times 2373.19 - 4.186 \times 28)}{4.186 \times (28 - 15)}$$

$$= 45.61 \text{ kg/h/kg of steam}$$

Total steam used per hour

$$= 175 \times 9 = 1575 \text{ kg}$$

∴ Total circulating water used per hour

$$= 45.61 \times 1575 = 71835.75 \text{ kg}$$

2.25 AIR PUMP

Air pump maintains the required vacuum in the condenser by removing air or air and the condensate both. Commonly, dry air pumps and wet air pumps are used for this purpose. Dry air pump removes air only, whereas the wet air pump removes the mixture of condensate, water, and other gases. These pumps may be reciprocating type or rotary type. Here Edward's air pump which is of reciprocating type is discussed.

Edward's Air Pump Edward's air pump consists of a piston which reciprocates in the pump barrel. The piston is flat on its upper surface and conical at the bottom. The barrel has a ring of ports near its lower end through which it communicates with the condenser. The top of the barrel has cover in which head or delivery valves are provided as shown in Figure 2.38.

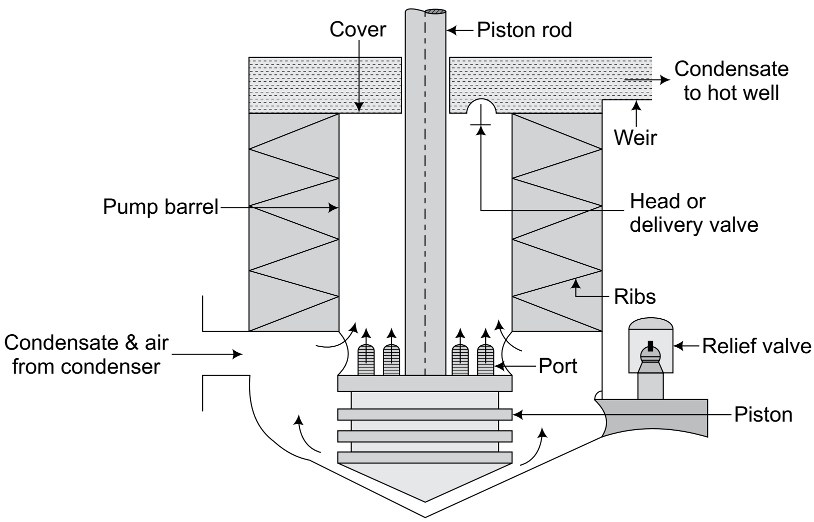


Fig. 2.38 Edward's Air Pump

When the piston moves in the downward direction a partial vacuum is produced in the barrel above the piston due to which the delivery valve closes. The air and water vapors from the condenser rush into the space above the piston through the ports. Further, motion of the piston causes its conical end to displace the condensate rapidly through the ports. Thus, water vapors, air, and the condensate fills the space above the piston.

When the piston moves in the upward direction the charge of water vapors, air, and the condensate get compressed to a pressure slightly above the atmospheric pressure. This opens the head valves through which the charge flows over the weir to the hot well. The weir maintains a sufficient head of water over head valves and thus seals them against any air leakage.

A relief valve is provided at the base of the barrel to release the charge if the pressure exceeds the atmospheric pressure due to any reason.

These pumps are generally used for low vacuum requirements. Its main advantage over other pumps is that it does not have any inaccessible valves.

2.26 COOLING TOWERS AND COOLING PONDS

Cooling water is supplied to the condenser for the condensation of steam. In this process the cooling water becomes hot. To cool this hot water coming out of the condenser the cooling towers and the cooling ponds are used so that the cooled water can be reutilized in the condenser again.

2.26.1 Cooling Towers

The cooling towers are very useful when there is scarcity of both the water and the land. It is an artificial device by which the hot water coming out of the condenser is cooled effectively. By using cooling towers the cooling water requirement is reduced and only makeup water is to be supplied. The principle of cooling the water is similar to that of the evaporative condenser. Some water about 1% goes into air in the form of water vapor by absorbing its latent heat of vaporization from the remaining water and thus causes the reduction in the water temperature. The cooling towers reduce the cooling water demand about 75 times but it is achieved at the expense of large capital, land, and operational costs. The types of cooling towers on the basis of the draught (method of air circulation) are:

(i) Natural draught cooling towers

(ii) Mechanical draught cooling towers: (a) Forced draught cooling towers, and (b) Induced draught cooling towers.

(i) Natural Draught Cooling Towers The schematic view of natural cooling tower is shown in Figure 2.39. The hot water from the condenser is pumped to a height of about 8 m to 12 m which enters the tower and then sprayed over the woodwork and trays. The water in the form of sprays meets the air entering from the bottom of the tower which is open to the atmosphere. Hot water gives up its heat to the air and gets cooled. The hot air along with some water vapor leaves the tower at top and the cooled water falls down in the form of rain and gets collected in the pond at the bottom of the tower. The cooled water from the pond is again supplied to the condenser. No fan is used in natural draught cooling towers. The bottom of the tower is kept open through which air enters into the tower. The airflow is maintained due to pressure difference caused by the difference in density between the hot air inside the tower and the outside atmospheric air. The water vapors leaving the tower along with the air is prevented by using water eliminators. Still some loss of water will be there which will be compensated by adding fresh water called make up water. The reduction in temperature of the water is called range. The disadvantage of this type of tower is that to produce large natural draught the tower will be very high.

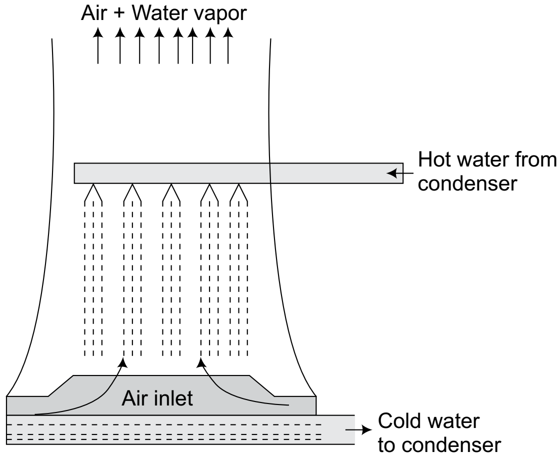
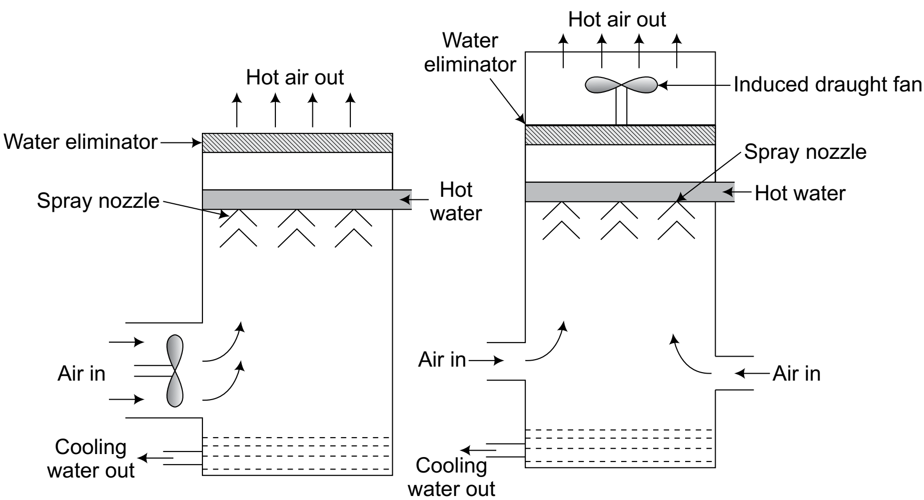


Fig. 2.39 Schematics of Natural Draught Cooling Tower

(ii) Mechanical Draught Cooling Towers In the mechanical draught cooling towers the air is circulated with the help of a mechanical device like a fan or a blower. Depending on the position of the fan or blower these are classified as forced draught or induced draught cooling towers. When the fan is installed at the bottom of the tower it is known as forced draught cooling tower as shown in Figure 2.40 (a). When the fan is installed at the top of the tower it is called induced draught cooling tower as shown in Figure 2.40 (b).

Hot water coming from the condenser enters the tower from its top and is sprayed through the nozzles. The sprayed water meets with the air going upwards. Eliminators are provided at the



(a) Forced draught cooling tower (b) Induced draught cooling tower

Fig. 2.40 Schematics of Mechanical Draught Cooling Tower top to prevent the escaping of water droplets with air leaving from the top. Cooling towers are generally hyperbolic in shape and are made of steel, concrete, or timber. The induced draught cooling towers are generally used in large capacity power plants. In the case of forced draught tower power requirement is high and maintenance cost of the fan is high. The induced draught tower occupies less space as the fan drives are installed at the top of the tower. In the induced draught tower the air is drawn by the fan from all the sides of the tower through the openings at low velocity and thus the cooling effect is obtained across the entire cross section of the tower. Also it handles warm air, so there will be no freezing problems during winter season as in the case of forced draught cooling towers.

2.26.2 Cooling Ponds

It is the simplest method of removing heat from the hot water by discharging it through a pipeline into a large open pond exposed to the atmosphere. The water is cooled by blowing air over the surface of the pond and mixing with cold water of the pond. The heat from the hot water will be transferred to the air by evaporation and convection processes. The loss of water by evaporation and wind blowing over the cooling pond is about 2% to 3%. To reduce the area of pond the water is sprayed into the air over the pond surface by water spray nozzles. For effective cooling these nozzles are fitted at a height of about 1 m to 2.5 m above the surface of water. Even after the use of spray nozzles the area required for cooling is large due to the evaporative cooling process. The factors which affect the dissipation of heat from the cooling pond are: area and depth of pond, temperature of water entering the pond, relative humidity, air velocity, atmospheric temperature and pressure, and solar radiation.

The cooling ponds are classified into two categories, namely, non-directed flow type and directed flow type. The non-directed flow type cooling pond is shown in Figure 2.41 (a) and directed flow type is shown in Figure 2.41 (b).

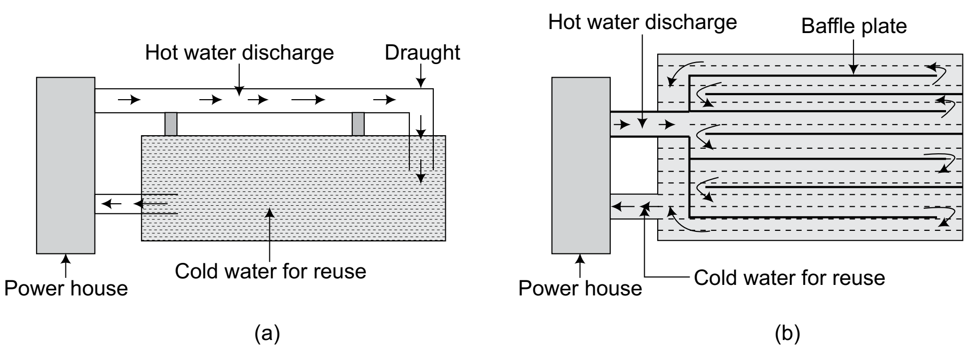


Fig. 2.41 Cooling Ponds

In the case of non-directed flow type ponds, the hot water is discharged into the open pond. But in the case of directed flow type cooling ponds, the pond is divided into a number of channels by providing baffle plates which direct the flow of hot water. The cooling effect in directed flow cooling ponds will be more due to thorough mixing of hot and cold water streams.

On the basis of spray nozzles arrangement, the cooling ponds may also be classified as single deck or double deck ponds. In a single deck cooling pond, the spray nozzles are arranged at the same elevation, whereas in double deck system the nozzles are arranged at different elevations. The double deck cooling ponds are more efficient than single deck system.

The disadvantages of cooling ponds are: (i) Large area is required for cooling, for example, the surface area required in a cooling pond is about 30 times the size of cooling tower for the same duty, (ii) The loss of water due to air blow is large and it is unprotected against dust, and (iii) There is no control over the temperature of cooled water, and its cooling efficiency is low.

The cooling ponds are suitable only for small capacity steam plants where land is easily and cheaply available.

EXERCISE

(i) Objective Type Questions

1. Nozzle is designed for:
- (a) maximum pressure and maximum discharge
- (b) maximum pressure at outlet
- (c) maximum discharge
- (d) none of the above

2. For supersonic flow, the nozzle should be:
- (a) convergent
- (b) convergent-divergent
- (c) constant area duct
- (d) divergent

3. For subsonic flow, the nozzle should be:
- (a) divergent
- (b) convergent-divergent
- (c) convergent
- (d) constant area duct

4. Velocity of steam at the exit of nozzle is given by:
- (a) $44.72 \sqrt{\eta_{\text{nozzle}} (h_1 - h_2)}$
- (b) $4.472 \sqrt{\eta_{\text{nozzle}} (h_1 - h_2)}$
- (c) $4472 \sqrt{\eta_{\text{nozzle}} (h_1 - h_2)}$
- (d) $0.4472 \sqrt{\eta_{\text{nozzle}} (h_1 - h_2)}$

5. In the case of nozzle, the whole of friction loss is assumed:
- (a) between inlet and throat
- (b) between throat and exit
- (c) between inlet and outlet
- (d) none of the above

6. Moore and Curtis turbine is:
- (a) velocity compounded turbine
- (b) pressure compounded turbine
- (c) reaction turbine
- (d) pressure-velocity compounded turbine

7. Parson's turbine is:
(a) impulse turbine (b) impulse-reaction turbine
(c) velocity compounded turbine (d) pressure compounded turbine

8. Curtis turbine is:
(a) velocity compounded turbine
(b) pressure compounded turbine
(c) pressure-velocity compounded turbine
(d) reaction turbine

9. Rateau turbine is:
(a) velocity compounded turbine
(b) pressure compounded turbine
(c) pressure-velocity compounded turbine
(d) reaction turbine

10. De-Laval turbine is:
(a) simple impulse turbine (b) reaction turbine
(c) pressure compounded turbine (d) velocity compounded turbine

11. The pressure on the two sides of the moving blades of a reaction turbine is:
(a) same (b) higher at inlet (c) lower at inlet (d) none of the above

12. In case of impulse turbine:
(a) enthalpy drops only in moving blades
(b) enthalpy drops only in fixed blades
(c) enthalpy drops in moving and fixed blades
(d) none of the above

13. The steam consumption in reciprocating steam engine in comparison to steam turbine is:
(a) equal (b) less (c) more (d) none of the above

14. Stage efficiency is also called:
(a) gross efficiency (b) blade efficiency
(c) diagram efficiency (d) mechanical efficiency

15. Stage efficiency of steam turbine is:
(a) $\eta_{nozzle} \times \eta_{blade}$ (b) $\eta_{nozzle}/\eta_{blade}$ (c) $\eta_{blade}/\eta_{nozzle}$ (d) none of the above

16. In case of reaction turbine, the enthalpy drops in:
(a) moving blades only (b) fixed blades only
(c) both fixed and moving blades (d) none of the above

17. Degree of reaction is the ratio of:
(a) enthalpy drop in moving blade to the sum of enthalpy drop in moving and fixed blades
(b) sum of enthalpy drop in moving and fixed blades to enthalpy drop in moving blade

(c) enthalpy drop in fixed blade to sum of enthalpy drop in moving and fixed blades
(d) none of the above

18. Degree of reaction for Parson's turbine is:
(a) 50% (b) 60% (c) 70% (d) 80%

19. Reheat factor is defined as:
(a) ratio of isentropic enthalpy drop to cumulative enthalpy drop
(b) ratio of cumulative isentropic heat drop to Rankine heat drop
(c) ratio of Rankine heat drop to cumulative isentropic heat drop
(d) none of the above

20. The value of reheat factor generally varies from:
(a) 0.6 to 1.06 (b) 0.02 to 0.06 (c) 0.8 to 1.08 (d) 1.02 to 1.06

21. The value of reheat factor increases if the number of stages are:
(a) less (b) more (c) same (d) none of the above

22. Internal efficiency is the ratio of:
(a) useful heat drop to heat supplied
(b) adiabatic heat drop to heat supplied
(c) total useful heat drop to adiabatic heat drop
(d) none

23. The number of stages in pressure compounded steam turbine in comparison to velocity compounded is:
(a) more (b) less (c) equal (d) none of the above

24. The efficiency of pressure compounded steam turbine in comparison to velocity compounded is:
(a) more (b) less (c) equal (d) none of the above

25. If η_s is constant for all stages then reheat factor is given by:
(a) $RF = \eta_s/\eta_i$ (b) $RF = \eta_i/\eta_s$ (c) $RF = \eta_s \times \eta_i$ (d) none of the above

26. Maximum blade or diagram efficiency of an impulse turbine is:
(a) $[1 + C] \cos \alpha$ (b) $[1 + KC] \cos \alpha$ (c) $\cos \alpha$ (d) $[1 + KC] (\cos \alpha)/2$

27. The maximum rate of work done for a single stage impulse turbine is:
(a) u^2 (b) $2u$ (c) $3u^2$ (d) $2u^2$

28. For maximum blade efficiency of a single stage impulse turbine:
(a) $\rho = \cos \alpha_1/2$ (b) $\rho = \cos^2\alpha_1/2$ (c) $\rho = \cos \alpha_1$ (d) $\rho = \cos^2\alpha_1$

29. The ratio of actual vacuum to the ideal vacuum is termed:
(a) condenser efficiency (b) boiler efficiency
(c) nozzle efficiency (d) vacuum efficiency

30. The ratio of actual temperature rise to the maximum possible rise is termed:
(a) condenser efficiency (b) boiler efficiency
(c) nozzle efficiency (d) vacuum efficiency

31. The cooling system used for big power plants:
(a) spray pond (b) hyperbolic cooling tower
(c) mechanical draught cooling tower (d) natural draught cooling

32. The ratio of sensible and evaporation cooling in wet cooling tower is:
(a) 20:80 (b) 40:60 (c) 40:80 (d) 60:80

33. The total absolute pressure in a condenser is equal to the:
(a) atmospheric pressure – gauge pressure
(b) atmospheric pressure + gauge pressure
(c) barometric pressure – vacuum pressure
(d) barometric pressure + vacuum pressure

34. Actual vacuum in a condenser is equal to the:
(a) barometric pressure – actual pressure
(b) barometric pressure + actual pressure
(c) atmospheric pressure – gauge pressure
(d) atmospheric pressure + gauge pressure

35. Pure cooling water is used in:
(a) surface condenser (b) jet condenser
(c) in both jet and surface condensers (d) none of the above

36. The rotor speed in steam turbines can be reduced by:
(a) velocity compounding (b) pressure compounding
(c) velocity-pressure compounding (d) all of the above

37. The vacuum efficiency of a condenser is about:
(a) 60% (b) 70% (c) 80% (d) 90%

38. Cooling water requirement in a surface condenser is:
(a) about 50 times of steam condensed (b) about 40 times of steam condensed
(c) about 30 times of steam condensed (d) about 20 times of steam condensed

39. Thermal efficiency of steam power plant increases by the employment of surface condenser because:
(a) average temperature of heat addition is increased
(b) average temperature of heat addition is reduced
(c) average temperature of heat rejection is reduced
(d) none of the above

40. Edward's air pump:
(a) removes air, vapor, and the condensed water
(b) removes air only
(c) removes air and vapor
(d) removes vapor only

41. For Parson's reaction turbine, the maximum efficiency is given by:
(a) $\dfrac{\cos^2\alpha_1}{1 + \cos^2\alpha_1}$ (b) $\dfrac{2\cos^2\alpha_1}{1 + \cos^2\alpha_1}$ (c) $\cos^2\alpha_1$ (d) $\dfrac{1 + \cos^2\alpha_1}{\cos^2\alpha_1}$

42. For maximum blade efficiency for Parson's reaction steam turbine the speed ratio is equal to:
(a) $\cos^2\alpha$ (b) $\cos^2\alpha/2$ (c) $2\cos^2\alpha$ (d) $\cos\alpha$

ANSWERS

1. (c)	2. (d)	3. (c)	4. (a)	5. (b)	6. (d)	7. (b)	8. (a)	9. (b)	10. (a)
11. (b)	12. (b)	13. (c)	14. (a)	15. (a)	16. (c)	17. (a)	18 (a)	19. (b)	20. (d)
21. (b)	22. (c)	23. (a)	24. (a)	25. (a)	26. (d)	27. (d)	28. (a)	29. (d)	30. (a)
31. (b)	32. (a)	33. (c)	34. (a)	35. (b)	36. (d)	37. (d)	38. (a)	39. (c)	40. (a)
41. (b)	42. (d)								

(ii) Review Questions

1. Describe with neat sketches the compounding of steam turbine. Why compounding of steam turbine is necessary?

2. What is degree of reaction? Prove that for a turbine with 50% degree of reaction the guide blades and moving blades are identical.

3. Explain the various losses that occur in a steam turbine.

4. Discuss the advantages of steam turbine over steam engine.

5. Define the terms related to steam turbine: (i) Speed ratio, (ii) Blade velocity coefficient, (iii) Diagram efficiency, (iv) Stage efficiency, and (v) Internal efficiency.

6. How are the steam turbines classified?

7. Explain the term reheat factor as applied to steam turbine.

8. Derive the condition for maximum efficiency of an impulse turbine and show that the maximum efficiency is $\cos^2\alpha$ where α is the angle at which the steam enters the blades.

9. Give the comparisons between impulse and reaction turbines.

10. Explain the working principles of impulse and reaction turbines with neat sketches.

11. Derive an expression for maximum diagram efficiency of a reaction turbine.

12. Derive an expression for maximum efficiency of an impulse turbine.

13. Describe the various types of cooling towers with neat sketches.

14. Define (i) Vacuum efficiency, and (ii) Condenser efficiency.

15. Give the requirements of a modern surface condenser.

16. What do you mean by a steam condenser? Explain its functions.

17. Write short notes on

(i) Cooling ponds, (ii) Cooling towers, (ii) Edward's air pump.

18. What are the main types of condensers? State the advantages of a surface condenser over a jet condenser.

19. Differentiate clearly between the jet and surface condensers.

20. Explain the governing methods of steam turbines with neat sketches.

21. Explain the sources of air in the condenser, its effects, and removal in detail.

(iii) Numerical Problems

1. The rotor of an impulse turbine is 0.26 m diameter and runs at 20,500 rpm. The nozzle angle is 20° and issues a steam jet with a velocity of 910 m/s. The steam mass flow rate through the turbine nozzles is 2.0 kg/s. Draw the velocity diagram and determine (i) tangential force on blades, (ii) axial force on blades, (iii) power developed by the turbine, and (iv) blade efficiency.

[**Ans.** F_t = 2228 N, F_a = 0, Power developed = 621.6 kW, and blade efficiency = 75%]

2. Steam issues from the nozzles of a De-Laval turbine with a velocity of 1200 m/s. The nozzle angle is 20°, the mean blade velocity is 400 m/s, and the inlet and outlet angles of blade are equal. The rate of mass flow through the turbine is 900 kg/hr. Determine (i) the blade angles, (ii) the relative velocity of steam entering the blade, (iii) the tangential force on the blade, (iv) the axial force on the blade, (v) the power developed, (vi) the blade efficiency, and (vii) energy lost due to friction in blade per kg of steam. Take velocity of coefficient as 0.8.

[**Ans.** (i) 29.4°, (ii) 840 m/s, (iii) 328 N, (iv) 20 N, (v) 131.2 kW, (vi) 73%, and (vii) 127 kJ/ kg.]

3. A single row impulse turbine develops 135 kW at a blade speed of 175 m/s using steam mass flow rate of 2 kg/s. Steam leaves the nozzle at 400 m/s. Steam leaves the turbine blade axially. Determine the nozzle angle, blade angle at entry and at exit, assuming no shock and velocity coefficient of the blade is 0.9. [**Ans.** 15.35°, 26.67°, 34.55°]

4. In a simple impulse turbine the nozzles are inclined at 20° to the direction of motion of the moving blades. The steam leaves the nozzles at 375 m/s. The blade speed is 165 m/s. Find suitable inlet and outlet angles for the blades in order that the axial thrust is zero. The relative velocity of steam as it flows over the blade is reduced by 15% by friction. Also determine the power developed for a flow rate of 10 kg/s. [**Ans.** 34.4°, 41.6°, 532 kW]

5. Steam enters the blade row of an impulse turbine with a velocity of 600 m/s at an angle of 25° to the plane of rotation of the blades. The mean blade speed is 255 m/s. The blade angle on the exit side is 30°. The blade friction coefficient is 10%. Determine (i) the angle of the blade on the entry side, (ii) the diagram efficiency, (iii) the power developed per kg of steam, and (iv) axial thrust per of steam per second.

[**Ans.** 41.5°, 83.65, 150.45 kW, – 90 N/kg/s]

6. In a stage reaction turbine the mean diameter of the rotor is 1.4 m. The speed ratio is 0.7. Determine the blade inlet angle if the blade outlet angle is 20°. The rotor speed is 50 rps. Also find the diagram efficiency. Find the percentage increase in diagram efficiency and rotor speed if the rotor is designed to run at the best theoretical speed, the exit angle is 20°. [**Ans.** 20°, 90.5%, 3.65%, 67.08 rps]

7. The vacuum reading of a condenser is 70.5 cm of Hg when the barometer reads 76 cm of Hg and the condensate temperature is 31°C. Calculate vacuum efficiency. [**Ans.** 97.06%]

8. The vacuum in a surface condenser is found to be 70.5 cm of Hg when the barometer reads 76 cm of Hg. The cooling water enters the condenser at 20°C and leaves at 36.5°C. Determine the condenser efficiency. [**Ans.** 87%]

9. A 180 kW steam engine consumes 9.5 kg of steam per kWh. The back pressure of the engine and the condenser pressure are equal to 0.15 bar. The temperature of the cooling water at the inlet and outlet are 18°C and 34°C respectively. The temperature of the condensate is 35°C. Determine the quantity of cooling water required per hour if the steam exhausted to the condenser is dry saturated. [**Ans.** 62568 kg]

10. In a condenser the following data were recorded: vacuum = 70 cm of Hg, barometer = 76 cm of Hg, mean temperature of the condenser = 35°C, hot well temperature = 29°C, mass of cooling water = 45500 kg/hour, inlet temperature = 16.5°C, outlet temperature = 31°C. Determine (i) the mass of air present per unit condenser volume, (ii) dryness fraction of steam entering the condenser, (iii) vacuum efficiency, (iv) the condensate undercooling, and (v) condenser efficiency. [**Ans.** 0.02686 kg/m^3, 94%, 97.51%, 6°C, 78.37%]

5. Steam enters the blade row of an impulse turbine with a velocity of 600 m/s at an angle of 25° to the plane of rotation of the blades. The mean blade speed is 255 m/s. The blade angle on the exit side is 30°. The blade friction coefficient is 10%. Determine (i) the angle of the blade on the entry side, (ii) the diagram efficiency, (iii) the power developed per kg of steam, and (iv) axial thrust per of steam per second.
[**Ans.** 41.5°, 83.65, 150.45 kW, – 90 N/kg/s]

6. In a stage reaction turbine the mean diameter of the rotor is 1.4 m. The speed ratio is 0.7. Determine the blade inlet angle if the blade outlet angle is 20°. The rotor speed is 50 rps. Also find the diagram efficiency. Find the percentage increase in diagram efficiency and rotor speed if the rotor is designed to run at the best theoretical speed, the exit angle is 20°. [**Ans.** 20°, 90.5%, 3.65%, 67.08 rps]

7. The vacuum reading of a condenser is 70.5 cm of Hg when the barometer reads 76 cm of Hg and the condensate temperature is 31°C. Calculate vacuum efficiency. [**Ans.** 97.06%]

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