6- Water Turbines
Impulse Water Turbine (Pelton Turbine)

PELTON Turbine

1 pen stock
2 nozzle
3 spear
4 runner with buckets
5 protective casing
6 deflector
Pelton Turbines

- Pelton turbines are pure impulse turbine
- A jet of fluid is delivered by a nozzle at a high velocity on the buckets.
- These buckets are fixed on the periphery of a circular wheel (also known as runner).
- The impact of water on the buckets causes the runner to rotate and thus develop mechanical energy.
- The buckets deflect the jet through an angle 160° and 165° in the same plane as the jet.
Pelton turbine runner
B/d = 3;  
T/d = 0.8 – 0.95;  
E/d = 0.85;  
e/d = 0.35;  
L/d = 2.5 – 2.8;  

Typical Pelton turbine bucket dimensions
Why use multi-jets?

- Usually Pelton turbine is used at high heads.
- With a decrease in head more water is needed to achieve the same power.
- A nozzle with a large area is required. However the maximum jet diameter is set by the size of the bucket on the runner being used and it is one-third the width of the bucket.
The area of a single nozzle can no longer be increased to maintain the same power output, the next approach is to increase the effective nozzle area by increasing the number of jets.

Six-jet vertical shaft Pelton turbine, horizontal section.
• Regulation of the Pelton Turbine is done by closing the nozzle using properly shaped ‘spear’.

• The movement of the spear while regulating the water flow has to be very slow otherwise the deceleration of the water in the often very long pen-stock becomes too high creating pressure shocks and burst the pen-stock or the nozzle.

• If sudden down regulation is demanded a deflector acts first.
Basic Relationships

With an impulse turbine such as the Pelton turbine, the pressure at the bottom of the penstock creates a jet of water with velocity,

\[ C_1 = K_N \sqrt{2gH} \]

Where:

- \( C_1 \) = jet velocity (m/s)
- \( K_N \) = nozzle velocity coefficient

\[
= \frac{\text{actual velocity at nozzle exit}}{\text{Spouting velocity at nozzle exit}} = \frac{C_1}{C_o}
\]
The spouting (ideal) velocity is given by,

\[ C_o = \sqrt{2gH} \]

Values of \( K_N \) are normally around 0.98 to 0.99.

The jet diameter required can be determined by:

\[ \frac{Q}{n_j} = \frac{\pi d^2}{4} C_0 \Rightarrow d = \frac{54}{H^{1/4}} \sqrt{\frac{Q}{n_j}} \]

Where:

- \( d \) = jet diameter (cm)
- \( Q \) = total flow through turbine (m\(^3\)/s)
- \( n_j \) = number of nozzles
The minimum number of equally sized jets that would be required to develop the design power

\[ n_j = \frac{2900Q}{\sqrt{Hd^2}} \]

For horizontal axis turbines, a maximum of two nozzles is commonly used.

The maximum number of nozzles for a Pelton turbine used in micro-hydropower plants is commonly four, arranged around the runner on a vertical shaft.

Up to six are used in large hydropower plants.
Considering one jet impinging on a bucket, the appropriate velocity diagram is shown below.

- The jet velocity at entry is $C_1$
- the blade speed is $U$
- that the relative velocity at entry is $W_1 = C_1 - U$. 
At exit from the bucket one half of the jet stream flows leaving with a relative velocity $W_2$ and at an angle $\beta_2$ to the original direction of flow.

From the velocity diagram the much smaller absolute exit velocity $C_2$ can be determined.

The specific work from Euler’s turbine equation becomes,

$$\Delta W = U_1 C_\theta_1 - U_2 C_\theta_2$$
Where:

\[ U_1, U_2 = \text{blade speed} \]

\[ C_\theta = \text{the velocity component mutually perpendicular to both the axis of rotation and radius vector } r. \]

For the Pelton turbine,

\[
U_1 = U_2 = U, C_{\theta_1} = C_1 \text{ and } C_{\theta_2} = (U + W_2 \cos \beta_2)
\]

\[
\Delta W = U\left((U + W_1) - (U + W_2 \cos \beta_2)\right)
\]

\[
\Delta W = U(W_1 - W_2 \cos \beta_2)
\]
The effect of friction on the fluid flowing inside the bucket will cause the relative velocity at outlet to be less than the value at inlet.

Writing \[ W_2 = KW_1 \] , where \( K < 1 \) , then,

\[
\Delta W = UW_1(1 - K \cos \beta_2) = U(C_1 - U)(1 - K \cos \beta_2)
\]

An **efficiency of the runner** \( \eta_R \) can be defined as the specific work done divided by the incoming kinetic energy, i.e.
\[ \eta_R = \Delta W \frac{1}{2} C_1^2 \]
\[ = 2U(C_1 - U)(1 - K \cos \beta_2)/C_1^2 \]
\[ \eta_R = 2\nu(1 - \nu)(1 - K \cos \beta_2) \]

- Where the **blade speed to jet speed ratio**, \( \nu = \frac{U}{C_1} \)

- In order to find optimum efficiency, differentiate the above equation with respect to the blade speed ratio,

\[ \frac{d\eta_R}{d\nu} = 2 \frac{d}{d\nu} \left( \nu - \nu^2 \right)(1 - K \cos \beta_2) \]
\[ = 2(1 - 2\nu)(1 - K \cos \beta_2) = 0 \]

Therefore, the maximum efficiency of the runner occurs when \[ \nu = 0.5 \], i.e. \[ U = \frac{C_1}{2} \], Hence

\[ \eta_{R_{\text{max}}} = \frac{(1 - K \cos \beta_2)}{2} \]
Considering the linear velocity ratio to be equal to 0.45, runner speed can therefore be expressed as:

\[
\frac{U}{C_1} = 0.45, \quad \text{where} \quad C_1 = K_N \sqrt{2gH}
\]

Blade speed is given by:

\[
U = \frac{\pi DN}{60} \left[ \frac{m}{s} \right]
\]

Where:

\[
N = \text{runner speed (rev/min)}
\]

\[
D = \text{runner, pitch circle diameter (m)}
\]
Solving for $N$ using above equation and blade speed ratio to the jet speed gives,

\[
\frac{\pi DN}{60} = 0.45,
\]

\[
N = \frac{60 \times 0.45 \times K_N \sqrt{2gH}}{\pi \times D}
\]

taking $K_N$ of 0.99 gives:

\[
N = \frac{38\sqrt{H}}{D}
\]
The number of buckets “$n_b$” to ensure efficient operation can be estimated by the following equation.

\[ n_b = \frac{m}{2} + 15 \]

The equation is expressed in terms of a parameter called the jet or diameter ratio “$m$”. This is defined as:

\[ m = \frac{D}{d} \]

- To ensure that a runner is large enough to accommodate the optimum number of buckets, the diameter ratio can be used.
- The minimum value of this ratio has been found from experience to be about 6. Conventionally this ratio is usually in range of 10-20.
Example:

A four jet Pelton runner with a pitch circle diameter of 150mm is to be designed to generate 10 kW under a head of 20 m. Determine:

a. What jet diameter would be required to generate the desired power

b. Whether this jet size reasonable
Soln.

Assume:

- Turbine efficiency of about 75 %
- Generator efficiency of about 80 %
- A coupling efficiency of 90 %
- An overall efficiency of about 50 %
\[ P = 9.81 \times \eta \times H \times Q \]

\[ Q = \frac{P}{9.81 \times \eta \times H} = \frac{10}{9.81 \times 0.5 \times 20} = 0.1 \frac{m^3}{s} \]

The required jet diameter becomes:

\[ d = \frac{54}{H^{1/4}} \sqrt[4]{n_j} = \frac{54}{20^{1/4}} \sqrt{\frac{0.1}{4}} = 4 \text{cm} \]

The diameter ratio, \( m = D/d = 15/4 = 3.8 \).

This is much \textbf{too small}, implying a nozzle which is too large.
Example: Pelton Turbine

A Pelton turbine is driven by two jets, generating 4.0MW at 375 rev/min. The effective head at the nozzles is 200 m of water and the nozzle velocity coefficient, $K_N=98$. The axes of the jets are tangent to a circle 1.5m in diameter. The relative velocity of the flow across the buckets is decreased by 15 percent and the water is deflected through an angle of 165°. Neglecting bearing and windage losses, determine:

1) the runner efficiency;
2) the diameter of each jet;
3) the power specific speed(shape number).
Solution
Main Parts of Radial Reaction Turbomachine

- The main components of a radial turbomachine are:

1. **Casing**: The water from the penstocks enter the casing which is of spiral shape. The area of cross section of the casing goes on decreasing gradually.

   The casing completely surrounds the runner of the turbine. The casing is spiral in shape to facilitate water flow at constant velocity throughout the circumference of the runner. The casing is usually made of concrete, cast steel or plate steel.
2. **Guide Vanes:** The stationary guide vanes are fixed on a stationary circular which surrounds the runner. **The guide vanes allow the water to strike the vanes fixed on the runner without shock at the inlet.** This fixed guide vanes are followed by adjustable guide vanes. The cross-sectional area between the adjustable vanes can be varied flow control a part load.

3. **Runner:** It is a circular wheel on which a series of radial curved vanes are fixed. **The water passes into the rotor where it moves radially through the rotor vanes and leaves the rotor blades at a smaller diameter.** Later, the water turns through $90^\circ$ into the draft tube.
4. Draft Tube

- Although external to the rotating parts of the turbine the draft tube is an important part of the hydraulic machine.

- The draft tube is a conical diffuser with around 7° divergence which reduces the exit kinetic energy in the departing fluid and therefore increases the efficiency of the machine as a whole.

In order to avoid cavitation (small es), the elbow-type draft tube has a preference over straight diverging type fo draft tubes.
Velocity Triangles and Work Done

• The inlet and outlet velocity triangles for a runner are shown below.

• The specific work done is given by:

\[ Y = U_1 C_{w1} - U_2 C_{w2} \]
Radial Flow Turbine Losses

• The losses in terms of energy balance through the turbine is given by;

\[ P = P_m + P_r + P_c + P_l + P_s \]

Where:
- \( P_m \) = mechanical power loss
- \( P_r \) = runner power loss
- \( P_c \) = casing and draft tube loss
- \( P_l \) = leakage loss
- \( P_s \) = shaft power output
- \( P \) = water power available

\[ P_c + P_l + P_r \Rightarrow \text{Hydraulic Power Loss} \]
Runner power loss $Pr$ is due to friction, shock at the impeller entry and flow separation. It results in a head loss $hr$ associated with the flow rate through the runner of $Qr$.

$$P_r = \rho g Q_r h_r$$

Leakage power loss $P_l$ is caused by a flow rate $q$ leaking past the runner and therefore not being handled by the runner. Thus

$$Q = Q_r + q$$

And with a total head $Hr$ across the runner, the leakage power loss becomes

$$P_l = \rho g q H_r$$
• Casing power loss Pr is due to the friction eddy and flow separation losses in the casing and the draft tube.

• If this head loss is hr then

\[
P_c = \rho g Q h_c
\]

• The total energy balance of the above equation becomes

\[
\rho g Q H = P_m + \rho g (h_r Q_r + h_c Q + H_r q + P_s)
\]

• Thus

\[
\eta_0 = \frac{P_s}{\rho g Q H}
\]
Example: Francis Turbine

- An inward flow Francis turbine, having an overall efficiency of 86 %, hydraulic efficiency of 90 %, and radial velocity of flow at inlet \( \frac{0.28 \sqrt{2gH}}{\eta_H} \). The turbine is required to develop 5000 kW when operating under a net head of 30m, The specific speed(metric) is 270, assume guide vane angle is 300, find

1. Rpm of the wheel
2. The diameter and the width of the runner at inlet
3. The volume flow rate, and
4. The theoretical inlet angle of the runner vanes
Solution