Department of Hydraulic and Water Resources Engineering

KOIT, Wollo University

Lecture Notes

Course Code: ***WRIE3154***

Course Title: ***Basics of Hydropower Engineering***

Chapter 4: HYDRAULIC TURBINES

Target Group: ***G3_WRIE***

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CHAPTER 4: HYDRAULIC TURBINES

4.1 General

4.2 Classification

4.3 Characteristics of Turbines

4.4 Procedure in Preliminary Selection of Turbines

4.5 Turbine Scroll Case

4.6 Draft Tubes

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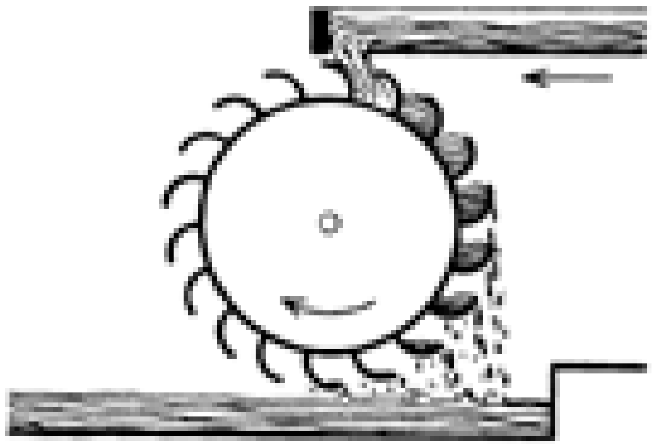
Definition:

- **Hydropower Engineering** refers to the technology involved in **converting the pressure energy and kinetic energy** of water into more **easily used electrical energy.**
- The prime mover in the case of hydropower is a **water wheel or hydraulic turbine** which transforms the energy of the **water into mechanical energy**.

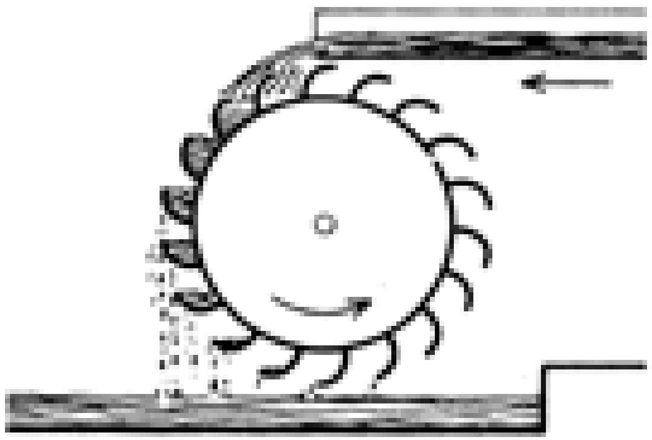
Water Wheels

- **Types of water wheels are based upon where the water strikes it**
  - **Pitch back – water drops from top and is deflected backwards to fall back towards the dam/river**
  - **Overshot – shoots over the top onto the wheel; the usual kind**
  - **Breast shot – strikes about 50% to 80% of height of the near side of the wheel**
  - **Undershot – pushes underneath and need not be more than immersed in a stream**
- **Waterwheels turn slowly compared with turbines**
  - **one to fifty rpm**

Pitchback



PitchBack (90%)



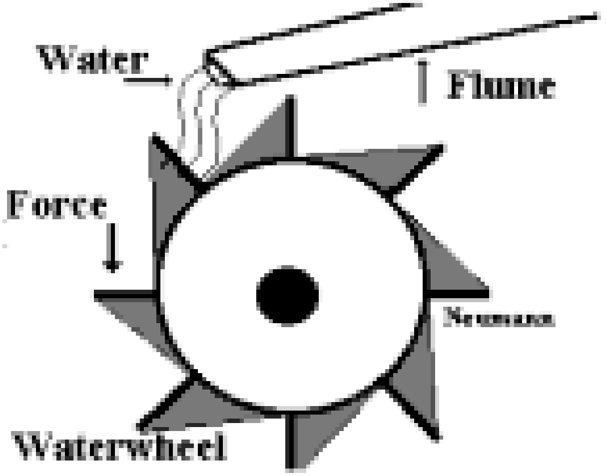
OverShot (70%)

- Note the difference in direction of the water flow
- A containing surround structure could force the water against the wheel as it falls and increase the weight of the water in the wheel

Overshot

- The water flows across the top of the wheel, pushing it forward, but also partially filling the buckets so that the weight pushes downward to turn the wheel
- The inertia of the water helps turn the wheel only slightly since it doesn't flow very fast
  - A very fast flow would be needed to get kinetic energy

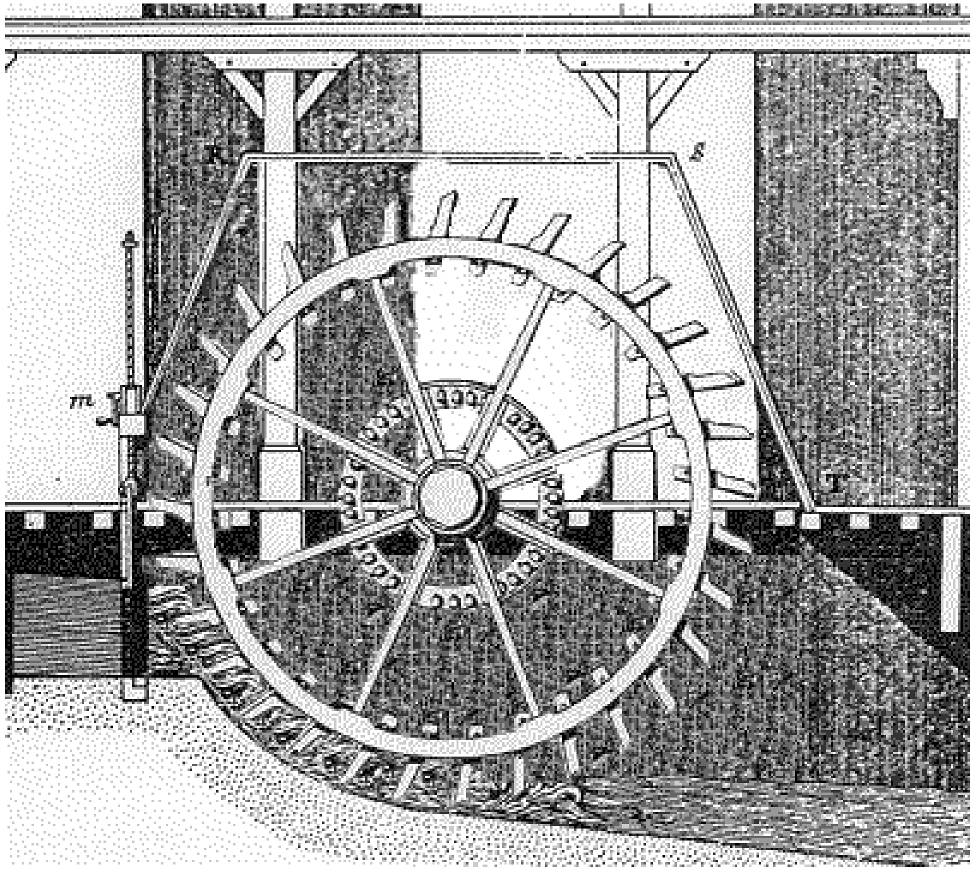
Figure 5



Overshot Waterwheel

Breast shot

- Note the contoured channel or surround at the bottom of the wheel that holds the water into the wheel



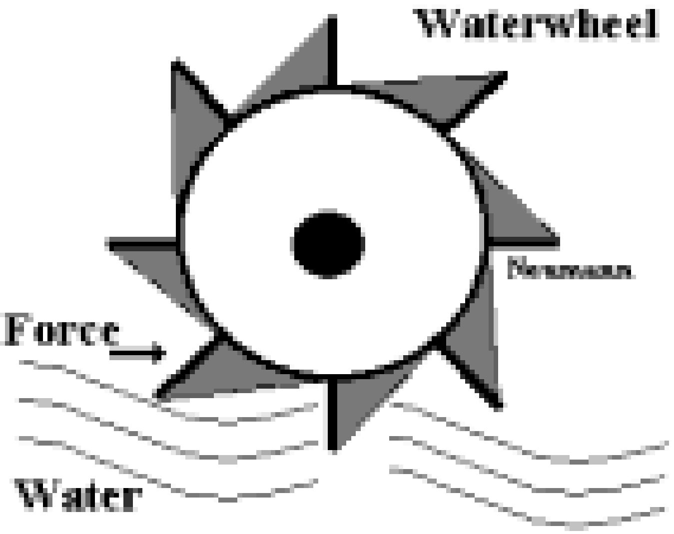
Water Flow

- The water strikes the wheel about mid-way up so the inertia and the weight of the water push the wheel around

Undershot

- The undershot wheel is simply placed in a stream with the bottom of the wheel pushed by the current
- Works well where there is little depth and no head
- Inefficient, but works where others won’t
- Can be on a small boat anchored in a stream

Figure 4



Undershot Waterwheel

Turbines:

- The turbine is made to convert hydraulic energy (potential and kinetic) into rotational mechanical energy on the turbine shaft.
- The flow discharge is controlled by an aperture mechanism just in front of the turbine runner.
- The rotating part of the turbine or water wheel is often referred to as the *runner*.
- The shaft is directly connected to an electric generator that further converts the mechanical energy into electric energy.

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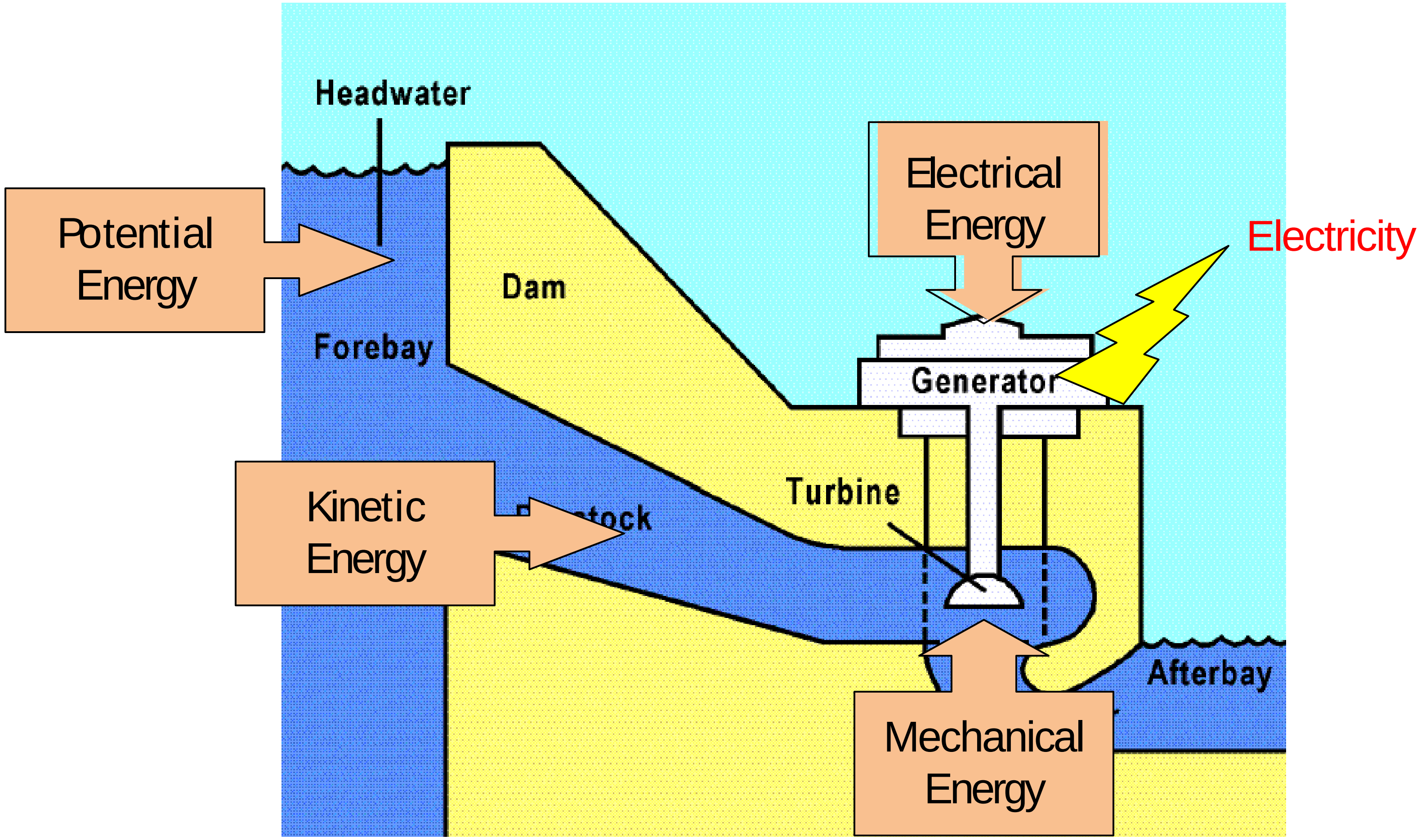
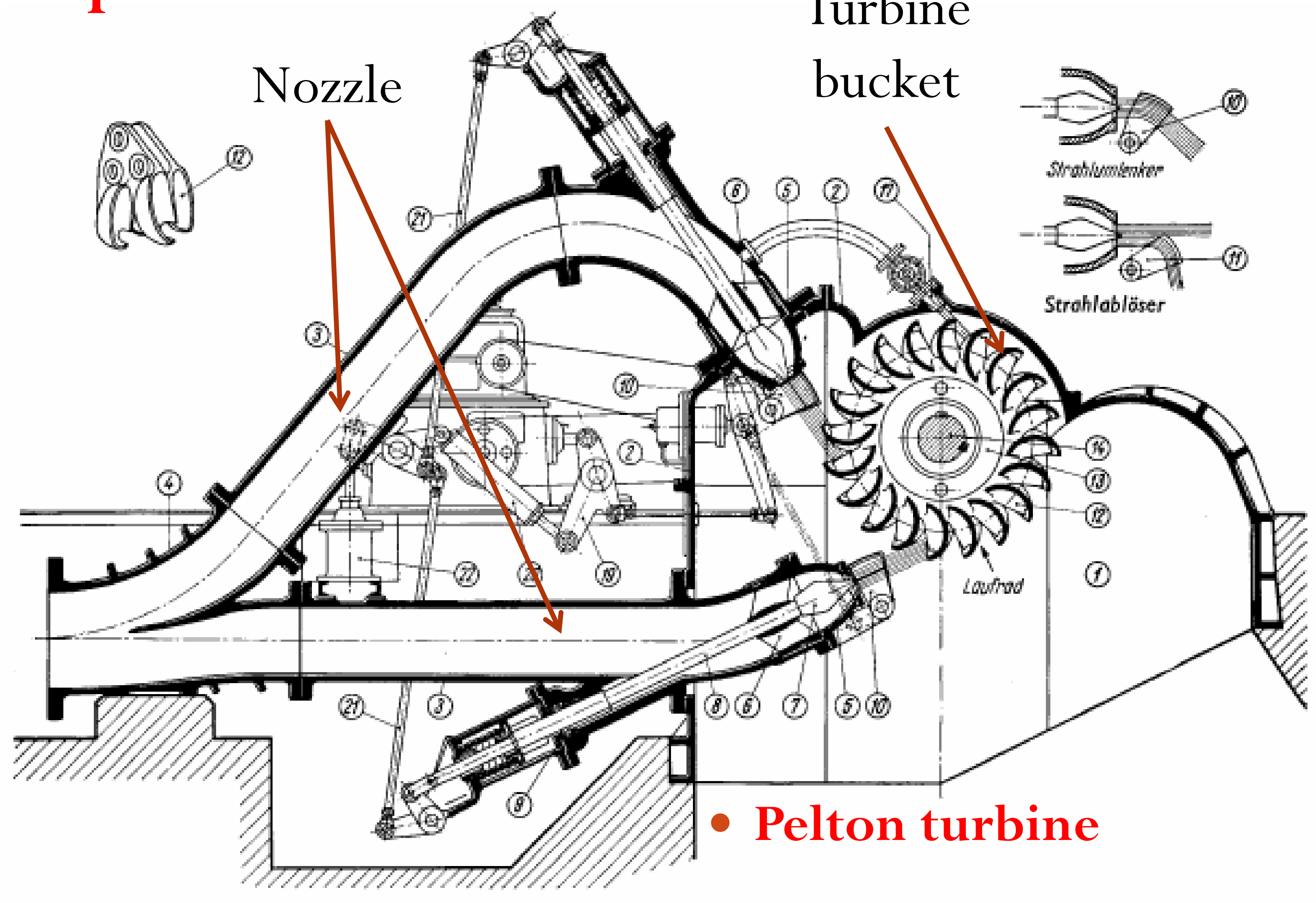


Turbine types

- A hydro turbine is usually tailor made in order to fit a particular net hydraulic head and a design flow discharge. However, there are two different categories of turbines, - they are based on two different principles of energy conversion:
  - **Impulse turbines**
  - **Reaction turbines**
- **Impulse Turbines**
  - Where all the hydraulic energy entering the turbine is converted into velocity energy in the stationary parts in front of the runner. The runner of the impulse turbine is only partly filled with water, and the runner operates in nearly atmospheric pressure.
  - The Pelton Turbine is the most commonly used type in this category, and it is used for the highest heads.

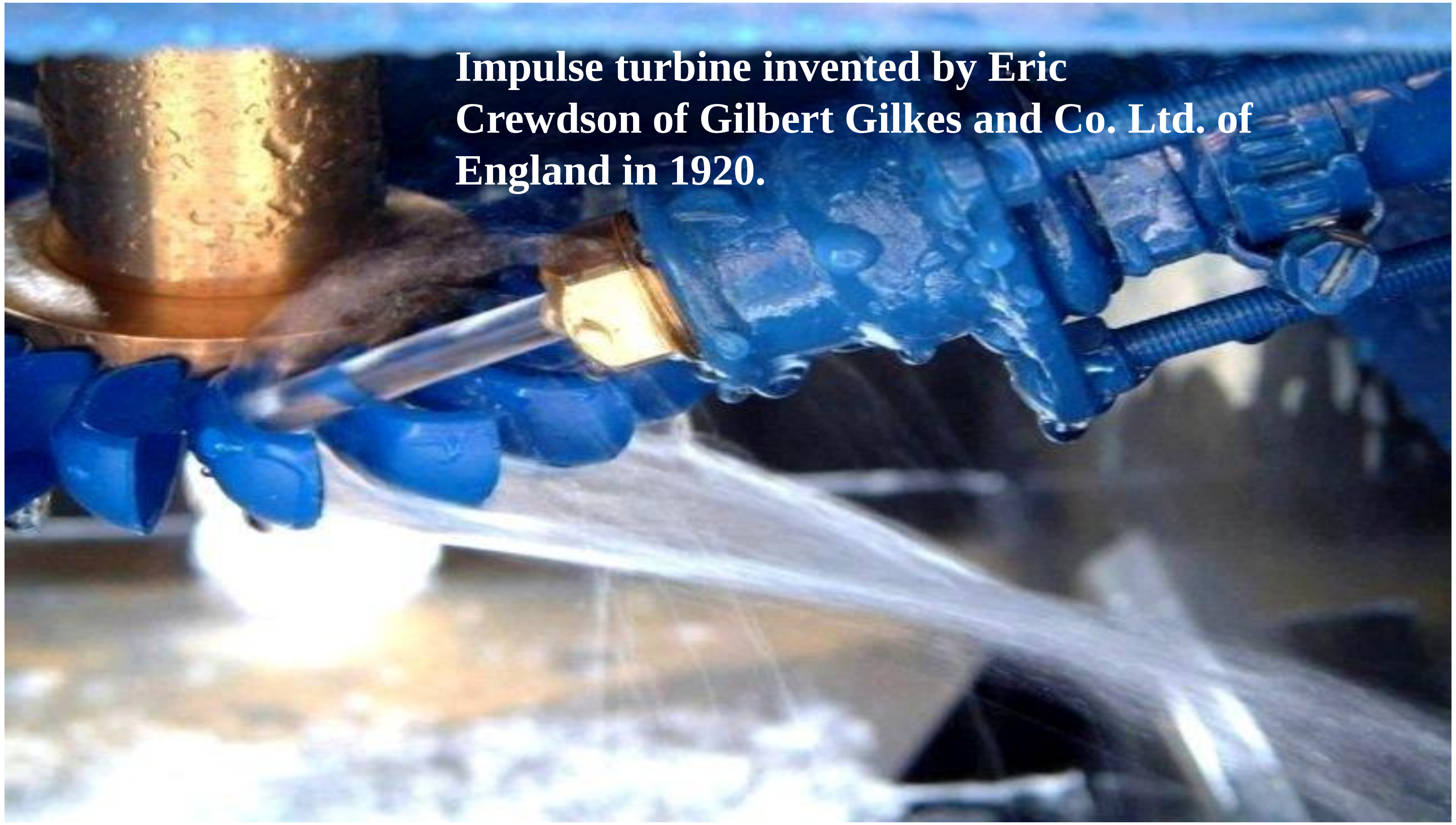
Impulse turbine



- **Pelton turbine**

Turgo Impulse turbine

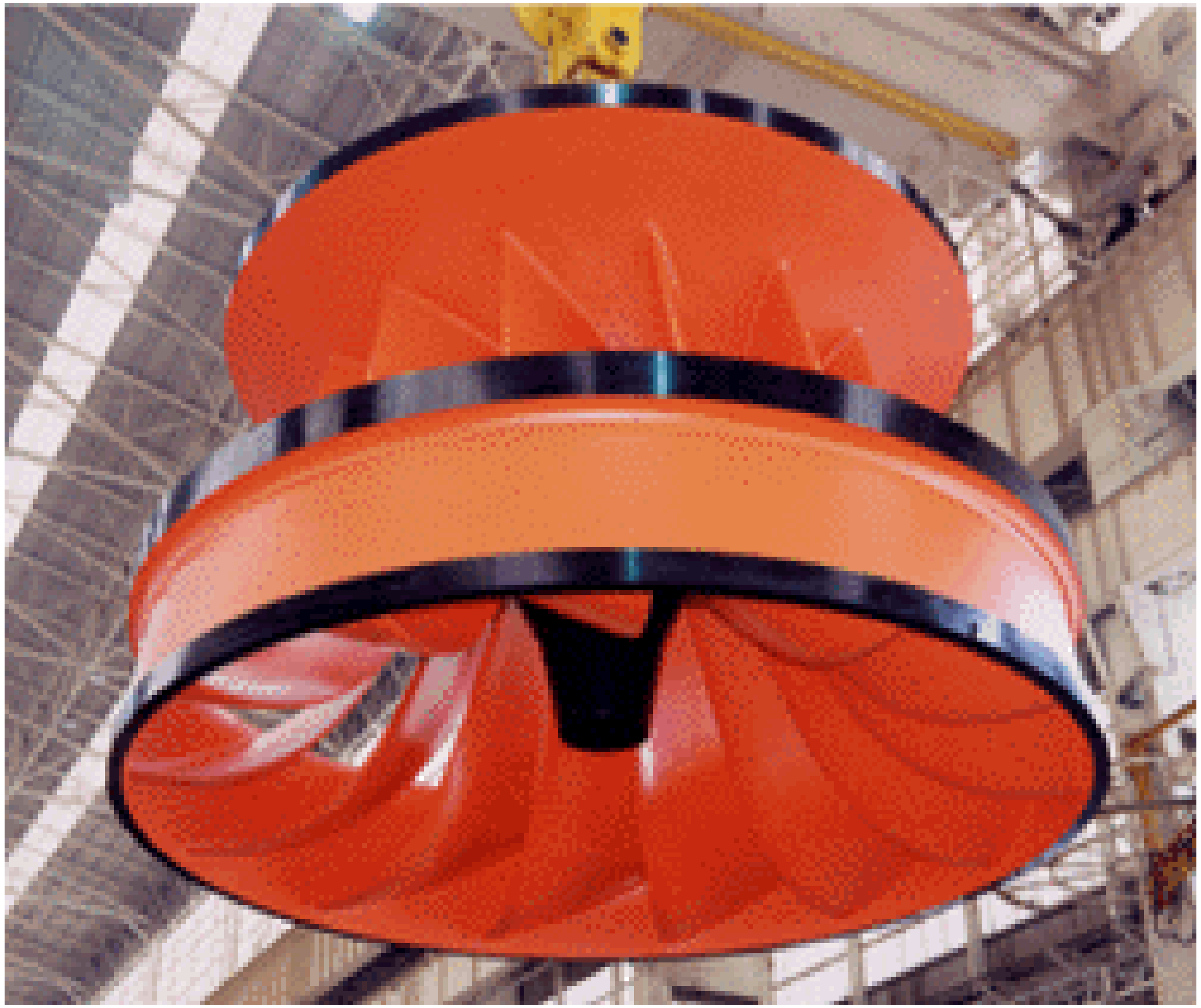
- **The turbine is designed so that the jet of water strikes the buckets at an angle to the face of the runner and the water passes over the buckets in an axial direction before being discharged at the opposite side.**



Turbine types

- **Reaction turbines**
  - In reaction turbines only a part of the inlet hydraulic energy is converted into velocity energy in the stationary turbine parts. Thus, the conversion of hydraulic to mechanical energy in the runner can be divided into two:
    - The impulse action caused by the change of velocity direction from the runner inlet to the outlet, and
    - The reaction contribution caused by the pressure drop through the runner.
  - The pressure drop is obtained because the runner is completely filled with water. In the draft tube diffuser (Connecting the outlet of the turbine runner to the tail race) some of the velocity energy at the runner outlet is converted to potential energy.
  - Francis, Kaplan and Bulb turbines belong to this group.

Francis Turbine

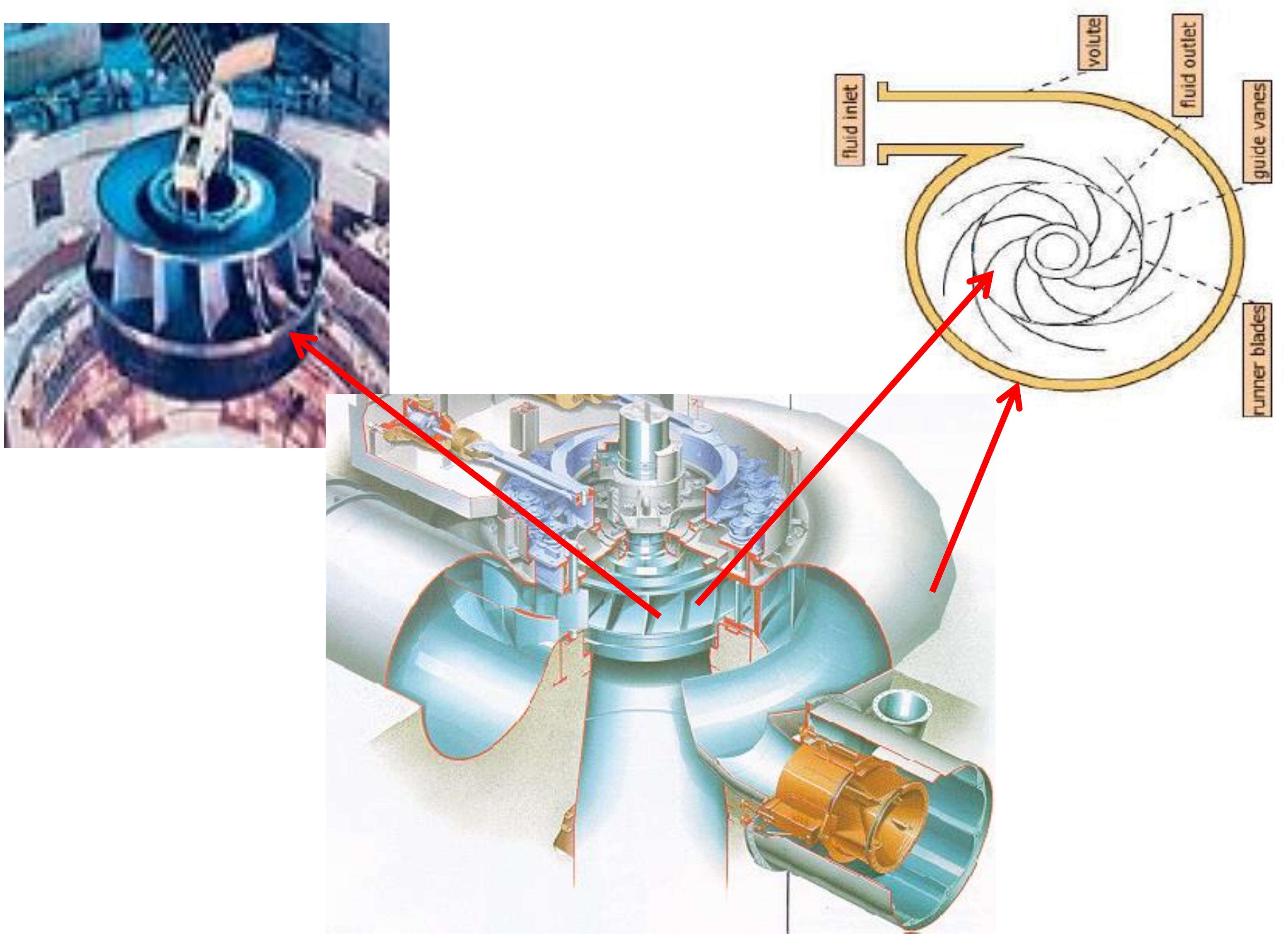


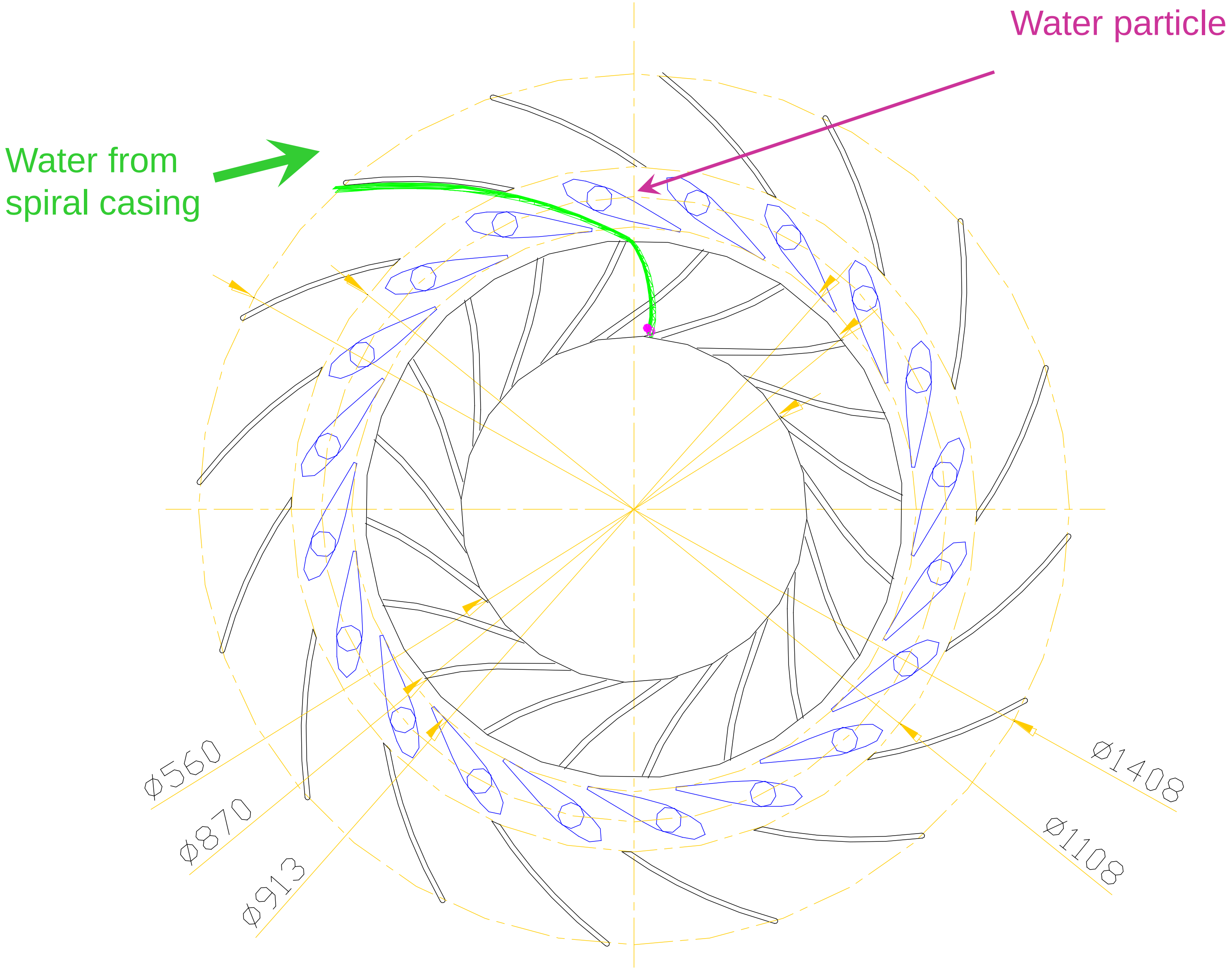
- Water flows through the runner to spin the turbine

Francis turbines

- It is a reaction turbine developed by an English born American Engineer, Sir J.B. Francis.
- The water enters the turbine through the outer periphery of the runner in the radial direction and leaves the runner in the axial direction, and hence it is called 'mixed flow turbine'.
- It is a reaction turbine and therefore only a part of the available head is converted into the velocity head before water enters the runner.
- The pressure head goes on decreasing as the water flows over the runner blades.
- The static pressure at the runner exit may be less than the atmospheric pressure and as such, water fills all the passages of the runner blades.
- The change in pressure while water is gliding over the blades is called 'reaction pressure' and is partly responsible for the rotation of the runner.
- A Francis turbine is suitable for medium heads (45 to 400 m) and requires a relatively large quantity of water.

The Francis Turbine: Developed by James B. Francis





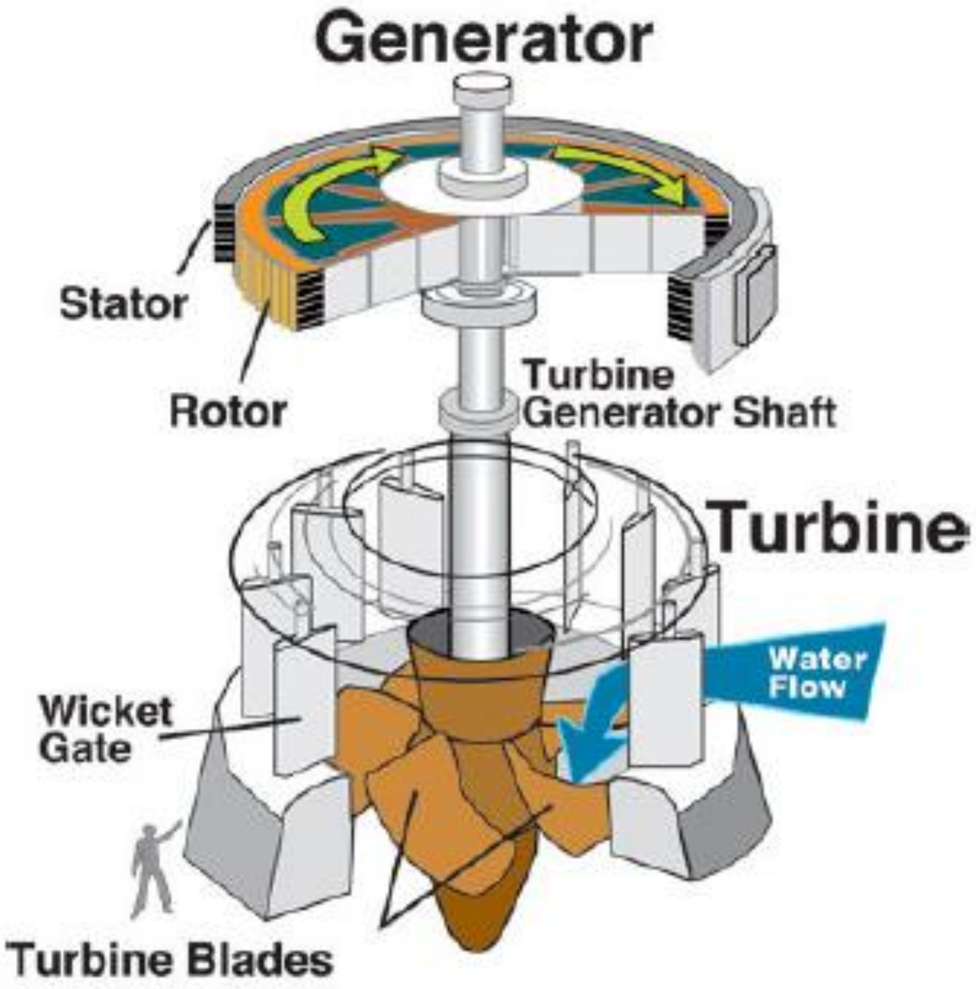
R a d i a l v i e w

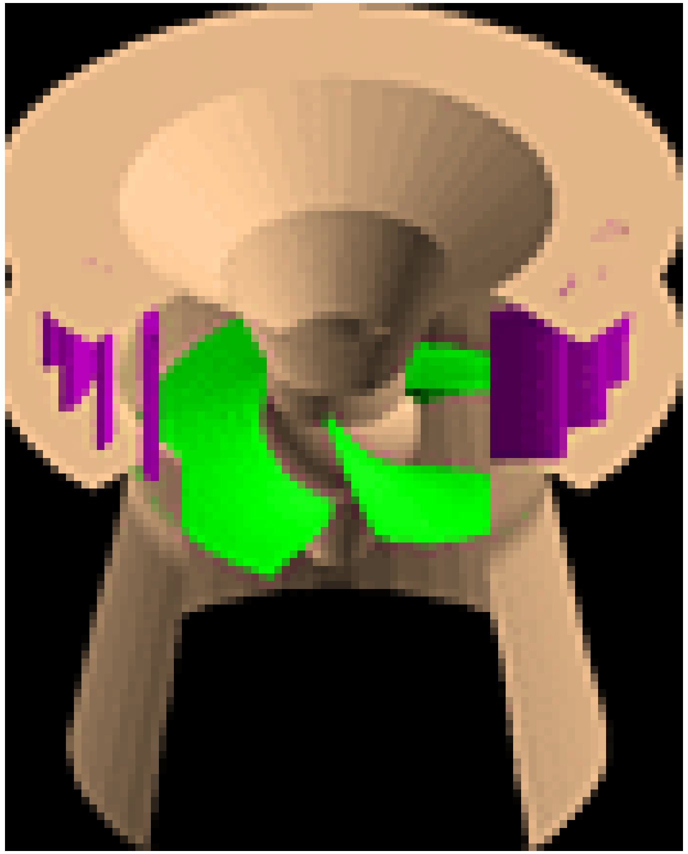
runner guide vanes and stay vanes

Kaplan Turbine

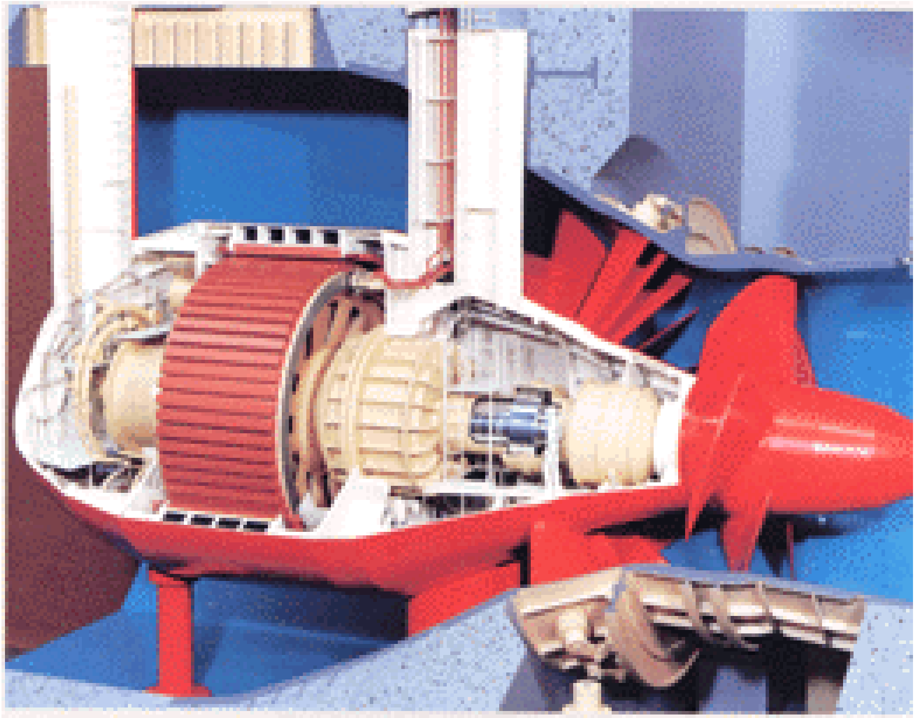
- Kaplan turbine runner

Major Parts of A Kaplan Turbine





Bulb Turbine



Ampair's Aquair UW submersible propeller turbine

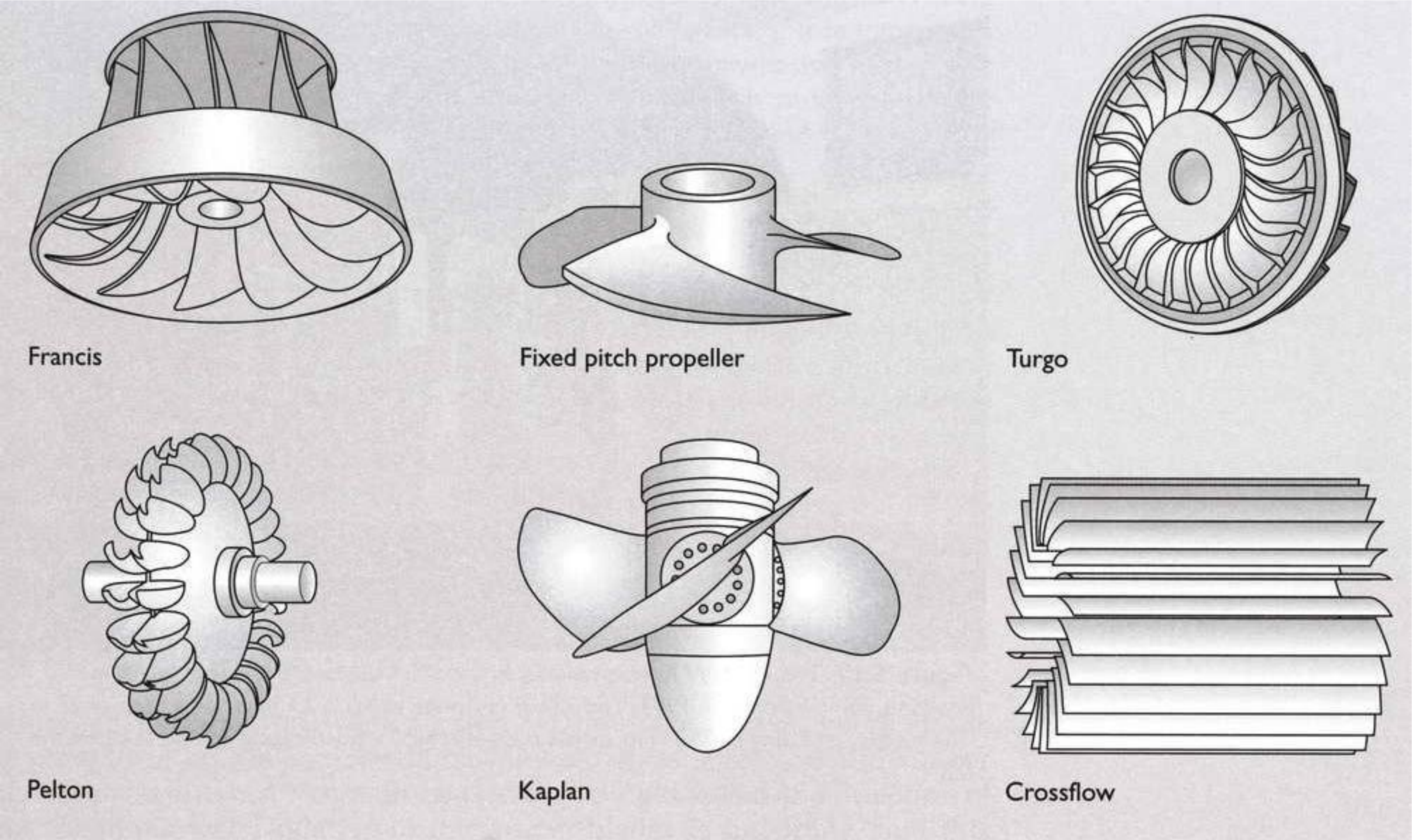
- Small versions of the bulb turbine can be lowered into a stream by hand to power a remote home

Classification of Turbines

- Turbines may also be classified according to the main direction of flow of water in the runner as:
  - Tangential flow turbine (Pelton wheel)
  - Radial flow turbine (Francis)
  - Mixed flow turbine (modern Francis)
  - Axial flow turbine of fixed blade (Propeller) or movable blade (Kaplan or bulb) type.

- Furthermore, turbines may be classified based on head, discharge, speed, specific speed.

Types of Turbine Runners



Classification based on head and discharge:

- **Head:**
  - **Low head, 1.5-15m, reaction-Propeller**
  - **Medium head, 16-70m, reaction-Kaplan**
  - **High head, 71-500m, reaction- Francis**
  - **Very high head, >500m, Impulse-Pelton**
- **Discharge:**
  - **Low discharge, Impulse- Pelton**
  - **Intermediate discharge, Reaction-Francis**
  - **High discharge, Reaction-Kaplan**

Turbine selection criteria

- The usual practice is to base selection on the annual energy output of the plant and the least cost of that energy for the particular scale of hydropower installation.
- In a theoretical sense, the energy output, E, can be expressed mathematically as plant output or annual energy in a functional relation as follows:
  - $E = F(h, q, TW, d, n, H_s, P_{max})$
- Where :
  - h = net effective head
  - q = plant discharge capacity
  - TW = tail water elevation
  - d = diameter of runner
  - n = generator speed
  - H_s = turbine setting elevation above tail water
  - P_{max} = maximum output expected or desired at plant.

- Generally the selection shall be based on: **Available head, Available discharge, Power demand fluctuation and Cost**
- For small head, the discharge requirement is high, requiring bigger turbines; thus costly. For larger head, the discharge requirement is low, requiring smaller turbines; thus cheaper.
- The choice of a suitable hydraulic prime-mover depend upon various considerations for the given head and discharge at a particular site of the power plant. The type of the turbine can be determined if the head available, power to be developed and speed at which it has to run are known to the engineer beforehand.
- The following factors have the bearing on the selection of the right type of hydraulic turbine:
  - I. Rotational Speed – Generator
  - II. Specific Speed;
  - III. Maximum Efficiency;

I. Rotational speed

- ***Turbine or synchronous speed*: Since turbine and generator are fixed, the rated speed of the turbine is the same as the speed of the generator.**
- **In all modern hydraulic power plants, the turbines are directly coupled to the generator to reduce the transmission losses. This arrangement of coupling narrows down the range of the speed to be used for the prime-mover. The generator generates the power at constant voltage and frequency and, therefore, the generator has to operate at its synchronous speed. The synchronous speed of a generator is given by**

$$N = 60\frac{f}{p}$$

- **Where: *N* speed rpm; *f*- frequency of the generator (usually 50 *hz* or 60 *hz*), *p*- number of pair of poles of the generator**
- ***f and p* are constants thus *N* is constant.**

- **Problems associated with the high speed turbines are the danger of cavitation and centrifugal forces acting on the turbine parts which require robust construction.**
- **No doubt, the overall cost of the plant will be reduced by adopting higher rotational speed as smaller turbine and smaller generator are required to generate the same power. The construction cost of the power house is also reduced.**
- **The ratio of the peripheral speed , *v*, of the bucket or vanes at the nominal diameter, D, to the theoretical velocity of water under the effective head, H, acting on the turbine is called the speed factor or peripheral coefficient , ϕ.**

$$\phi = \frac{v}{\sqrt{2gH}} = \frac{\omega r}{\sqrt{2gH}} = \frac{\pi DN}{60\sqrt{2gH}} = \frac{DN}{84.6\sqrt{H}}$$

- The following table suggests appropriate values of ϕ, which give the highest efficiencies for any turbine, the head & specific speed ranges and the efficiencies of the three main types of turbine.

Type of runner	ϕ	N_s	H (m)	Efficiency (%)
Impulse	0.43 – 0.48	8-17 17 17-30	>250	85-90 90 90-82
Francis	0.6 – 0.9	40 – 130 130-350 350-452	25-450	90-94 94 94-93
Propeller	1.4-2.0	380-600 600-902	< 60	94 94-85

In general:

- Pelton turbines are used for high heads & low discharges
- Francis types are used for medium & high head plants
- Propeller & Kaplan are used for low head plants with large discharges

II. Specific Speed

Runner	Specific Speed N_s (?)		
	Slow	Medium	Fast
Pelton	4-15	16-30	31-70
Francis	60-130	151-250	251-400
Kaplan	300-400	451-700	701-1100

$$N_s = \frac{N\sqrt{P}}{h^{5/4}}$$

$$N_s = \frac{1750}{h^{1/2}} \quad for\ 18 < h < 300m$$

$$N_s = \frac{1475}{h^{1/3}} \quad for\ h < 18m$$

Specific speed: is a speed at which a turbine is running to produce 1kW power through a head of 1m.

- **The turbine specific speed is a quantity derived from dimensional analysis. For a specific turbine type (Francis, Kaplan, Pelton), the turbine efficiency will be primarily a function of specific speed.**
- **Neglect, for the moment, the effect of head losses on the turbine power. The power will then be given by:** $P = \eta \gamma Q H \ldots$ (1)
- **Obviously the turbine should have as large an efficiency η as possible. In general, η will depend on the specific geometrical configuration of the turbine system, as well as the flow rate Q , the head H, and the turbine rotation rate N.**
- **For specific values of Q , H, and N, an optimum geometrical design would exist which would optimize the turbine efficiency. Determination of this optimum design would be performed using either experimental methods or (more recently) numerical CFD simulations, and work of this type has led to the development of the Francis, Kaplan, and Pelton designs of common hydroelectric use.**

- **Alternatively, given a specific turbine design (i.e., Francis, Kaplan, Pelton), one would anticipate that there would be a specific set of operating conditions Q, *H, and N which would optimize the turbine efficiency.***
- **The basic concept of the *turbine specific speed* is to identify the optimum operating conditions for a given turbine design. This identification process can be developed via simple dimensional analysis, coupled with an in viscid (i.e., ideal) model of fluid mechanics.**
- **Say that we have developed an optimized turbine design (i.e., maximized *η) for the specific conditions* Q_1, H_1, *and* N_1. *This design would have associated it a characteristic* size D_1 *(Eg. the turbine* runner diameter).**
- **It is not important to precisely connect D_1 *to some actual dimension of the* turbine; the length D_1 *is simply meant to represent* the overall size (or scale) of the turbine.**

- Since the design is optimized for the specific conditions, we can state that $$P_1 = \eta_{opt}\gamma Q_1 H_1 \ldots \; (2)$$
- where η_{opt} is the optimum (i.e., maximized) efficiency.
- Say we **change the head to some new value H_2 we want to estimate the corresponding conditions Q_2 and N_2 which will maintain the optimum efficiency of the turbine. Or, perhaps, we scale the turbine to a new characteristic size D_2: what are the corresponding new values of N_2, Q_2, H_2 which maintain η_{opt}?**
- If the conditions at state 2 give the same efficiency as state 1, then Eq. (2) would imply that $$\frac{P_2}{P_1} = \frac{H_2 Q_2}{H_1 Q_1} \ldots \; (3)$$
- The volumetric flow rate Q *will be proportional to a* characteristic velocity in the turbine, *V*, *times a characteristic* flow area. The flow area, in turn, would be proportional to the square of the characteristic size, D^2. *Therefore,* $$\frac{Q_2}{Q_1} = \frac{V_2 D_2^2}{V_1 D_1^2} \ldots \; (4)$$

- **If we neglect viscous effects (i.e., friction losses), Bernoulli's equation would show that $V^2 = 2\,gH$ …*(5)* and this implies that**

$$\frac{Q_2}{Q_1} = \frac{D_2^2}{D_1^2}\left(\frac{H_2}{H_1}\right)^{0.5} \quad \ldots (6)$$

- **Therefore, for the *same turbine design operating at* the two *optimized states 1 and 2, we would expect* that**

$$\frac{P_2}{P_1} = \frac{D_2^2}{D_1^2}\left(\frac{H_2}{H_1}\right)^{1.5} \quad \ldots (7)$$

- **Now consider the rotation speed of the turbine, *N*. If *R represents the radius of the turbine runner and U* the velocity at this radius, then *N = U / R, in radians* per s. For two turbines of the same design, one would expect that $U_2/U_1 = V_2/V_1$ *and* $R_2/R_1 = D_2/D_1$.**

- **Using again the head relation for the characteristic velocity *V*, *Eq. (5), we get***

$$\frac{N_2}{N_1} = \frac{D_1}{D_2}\left(\frac{H_2}{H_1}\right)^{0.5} \quad \ldots (8)$$

- The size ratio D_2/D_1 *can be eliminated between* Eqs. (7) and (8), and after rearranging,

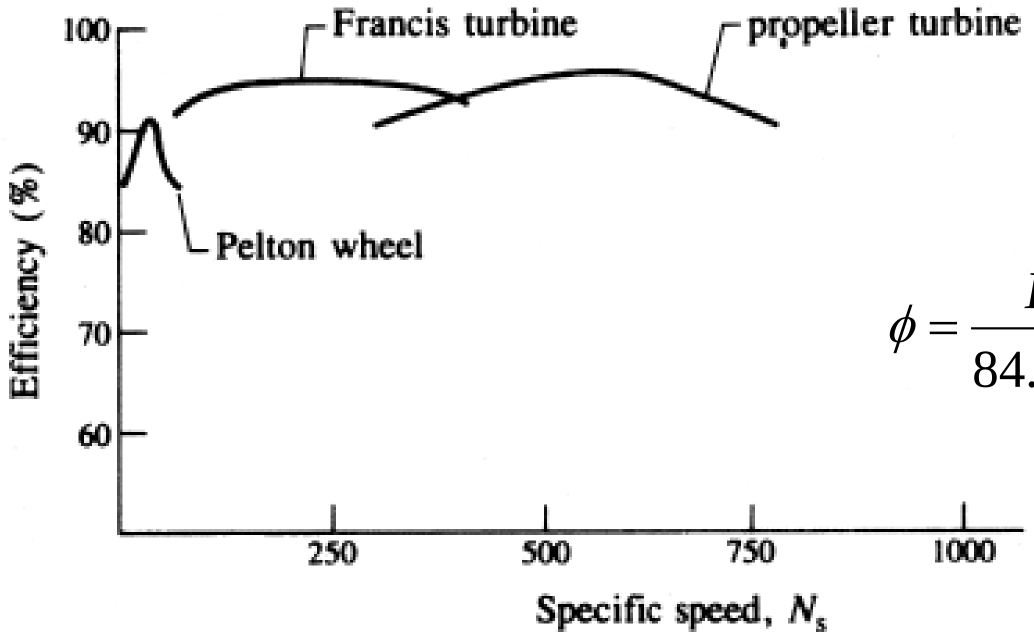
$$\frac{N_1(P_1)^{0.5}}{(H_1)^{1.25}} = \frac{N_2(P_2)^{0.5}}{(H_2)^{1.25}} = const. = N_s \ldots (9)$$

- The quantity N_s *is referred to as the specific speed of* the turbine. Understand that N_s, *as defined above*, is not a dimensionless quantity – we would need to appropriately include ***ρ and g to cancel out the units.***
- The important point of Eq. (9) is that a turbine operating at it's optimum design conditions would have ***a constant value of N_s.***

- **Meaning of specific speed**: Any turbine, with identical geometric proportions, even if the sizes are different, will have the same specific speed. If the model had been refined to get the optimum hydraulic efficiency, all turbines with the same specific speed will also have an optimum efficiency.
- In all modern power plants, it is common practice to select a high specific speed turbine because it is more economical as the size of the turbo-generator as well as that of power house will be smaller.
- Suppose generator of a given power runs at **120 rpm or at 800 rpm and say available head is 200 meters,** if the power developed in a single unit at 120 rpm is 60000 kW the required specific speed of the runner will be 39.08 rpm (?).
- Now if the same power is developed at 800 rpm in two runners, the required specific speed of the runner will be 260.54 rpm (?).
- The above calculations show that the required power can be developed either with one impulse turbine (Pelton) or two reaction turbines (Francis).

III. Maximum Efficiency

- The maximum efficiency, the turbine can develop, depends upon the type of the runner used.
- In case of impulse turbine, high specific speed is not conducive to efficiency, since the diameter of the wheel becomes relatively large in proportion to the power developed so that the bearing tend to become too large.



$$\phi = \frac{DN}{84.6\sqrt{H}} = \frac{DN_s H^{3/4}}{84.6\sqrt{P}}$$

- In most hydropower design problems one would typically know beforehand the available head H and the total available flow-rate Q.
- Then the power P produced by the turbine, assuming an efficiency of 1 and no head losses, could be estimated from Eq. (2).
- The figure above could then be used, in conjunction with the head H and the estimated power P, to determine the corresponding rotational speeds of Pelton, Francis, and Kaplan turbines operating at their maximum efficiency, i.e.,

$$N = \frac{N_s H^{1.25}}{P^{0.5}} \ldots \quad (10)$$

- An alternative approach is to specify beforehand the desired rotation rate N of the turbine. The power produced by the three turbine types, operating at optimum efficiency, would then be obtained by:

$$P = \frac{N_s^2 H^{2.5}}{N^2} \ldots \quad (11)$$

- Q could then be calculated from Eq. (2), and the number of required turbine units would be obtained from the total available flow rate divided by the flow through a single turbine.

- The low specific speed of reaction turbine is also not conducive to efficiency. The large dimensions of the wheel at low specific speed contribute disc friction losses.
- The leakage loss is more as the leakage area through the clearance spaces becomes greater and the hydraulic friction through small bracket passages is larger. These factors tend to reduce the efficiency as small values of specific speed are approached.

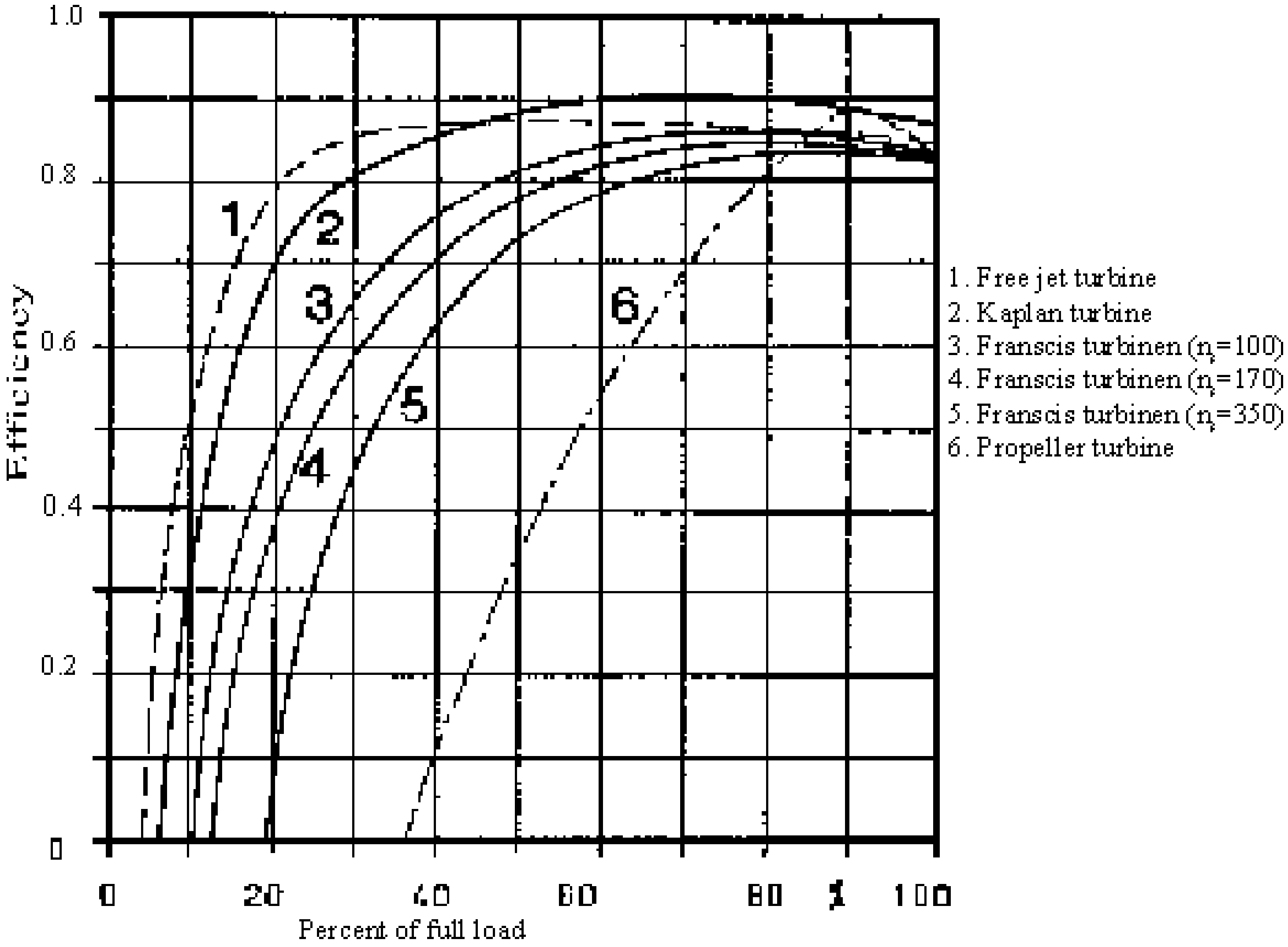
Procedure in preliminary selection of Turbines

- From design Q and H, calculate approximate P that can be generated , $P = \eta\, \gamma\, Q\, H$
- Calculate N (or assume) & compute N_s. From this, the type of turbine can be suggested $N = 120\dfrac{f}{p}$
- Calculate ϕ from: $\phi = \dfrac{DN}{84.6\sqrt{H}}$
- If ϕ is found to be too large, either N can be increased or more units may be adopted.
- For approximate calculations of runner diameter; the following empirical formula may be used (Mosony) $D = a\left(\dfrac{Q}{M}\right)^{1/3}$
  - Where D is in m; Q in m^3/s; N in rpm
  - a = 4.4 for Francis & propeller; a = 4.57 for Kaplan.
- $D = \dfrac{7.1\sqrt{Q}}{(N_s + 100)^{1/3} H^{1/4}}$ for propeller, H in m

- Nominal diameter, D , of pelton wheel

$$D = 38\sqrt{\frac{H}{N}} \qquad d_j = 0.542\sqrt{\frac{Q}{H}}$$

  - (d_j is diameter of the jet for N=0.45)
- Jet ratio given by $m = D/d_j$, is important parameter in design of pelton wheels.
- Number of buckets, $n_b = 0.5\ m + 15$ (good for $6 < m < 35$)
- It is not uncommon to use a number of multiple jet wheels mounted on the same shaft so as to develop the required power.
- Hydraulic turbines (runner) is designed for optimum speed and maximum efficiency at design head. But in reality, head and load conditions change during operation and it is extremely important to know the performance of the unit at other heads. This is furnished by manufacturer's curve.

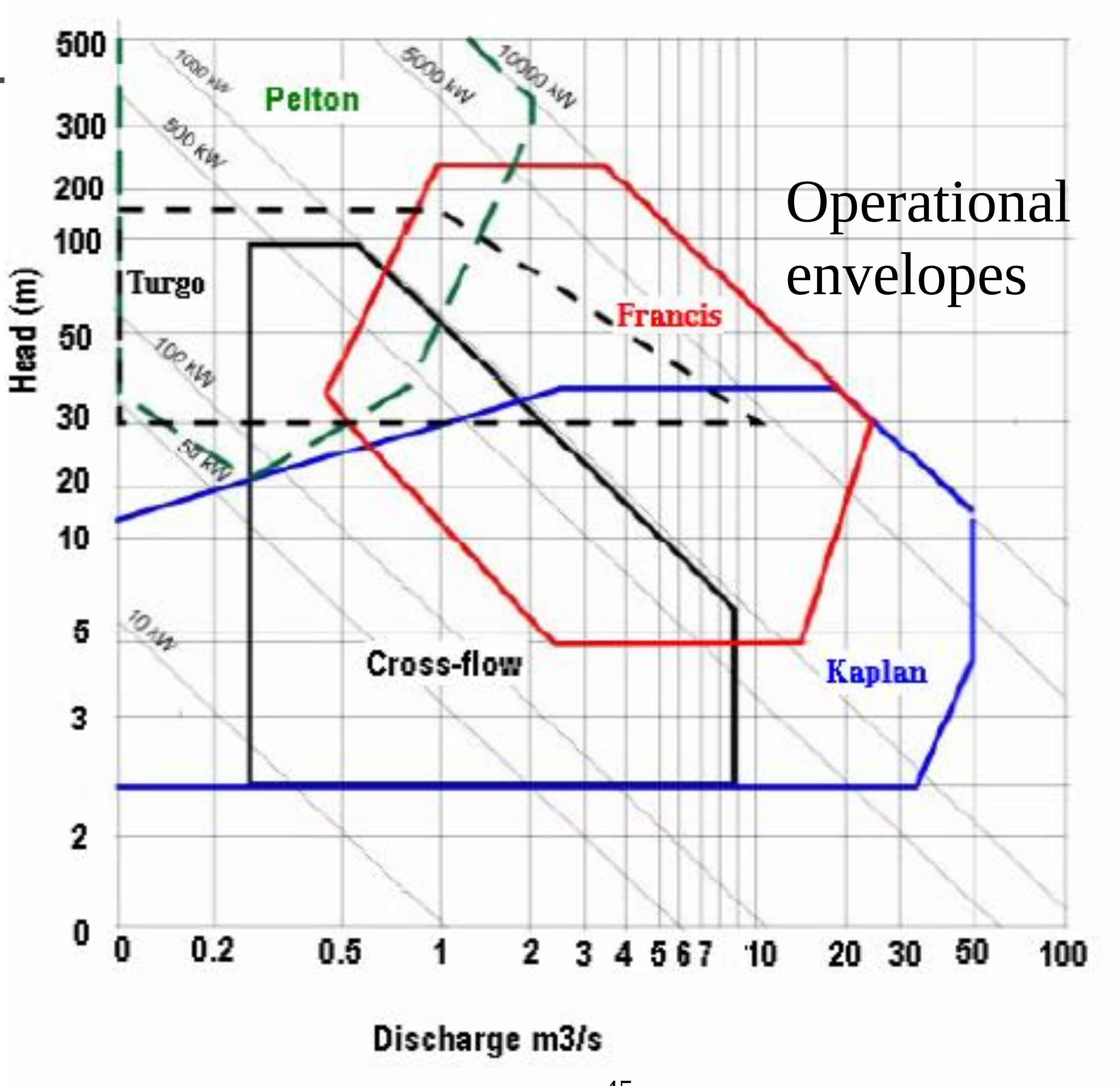

1.0
0.8
0.6
0.4
0.2
0
Efficiency
1
2
3
4
5
6
0
20
40
60
80
100
Percent of full load
1. Free jet turbine
2. Kaplan turbine
3. Franscis turbinen (n_s=100)
4. Franscis turbinen (n_s=170)
5. Franscis turbinen (n_s=350)
6. Propeller turbine

Limits of use of turbine types

- For practical purposes there are some definite limits of use that need to be understood in the selection of turbines for specific situations.
- **Impulse turbines** normally have most economical application at heads above 300m, but for small units and cases where surge protection is important, impulse turbines are used with lower heads.
- For **Francis turbines** the units can be operated over a range of flows from approximately 50 to 115% best-efficiency discharge. The approximate limits of head range from 60 to 125% of design head.
- **Propeller turbines** have been developed for heads from 5 to 60m but are normally used for heads less than 30m. For fixed blade propeller turbines the limits of flow operation should be between 75 and 100% of best-efficiency flow.

- **Kaplan units** may be operated between 25 and 125% of the best-efficiency discharge. The head range for satisfactory operation is from 20 to 140% of design head.
- **Operational Envelopes**
  - The rated flow and the net head determine the set of turbine types applicable to the site and the flow environment
  - Suitable turbines are those for which the given rated flow and net head plot within the operational envelope

Operational envelopes



Determination of number of units

- Normally, it is most cost effective to have a minimum number of units at a given installation. However, multiple units may be necessary to make the most efficient use of water where flow variation is great.
- Factors governing selection of number of units:
  - Space limitations by geology or existing structure.
  - Transportation facility
  - Possibility of on site fabrication
    - Field fabrication is costly and practical only for multiple units where the cost of facilities can be spread over many units.
    - Runners may be split in two pieces, completely machined in the factory and bolted together in the field. This is likewise costly, and most users avoid this method because the integrity of the runner cannot be assured.

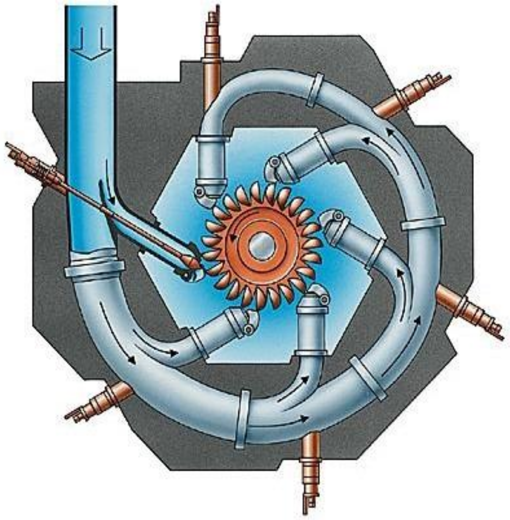
- The current trend is to have small number of units having larger sizes, as studies have found out that larger sized units have a better efficiency.
- Ex.1 Installed capacity needed: 1500 MW
  - This could have a number of alternatives on the number of units selection
    - 3 +1 units of 500 MW capacity
    - 5 +1 units of 300 MW capacity
    - 8 +0 units of 200 MW capacity
    - 1+1 unit of 1500 MW capacity
    - etc

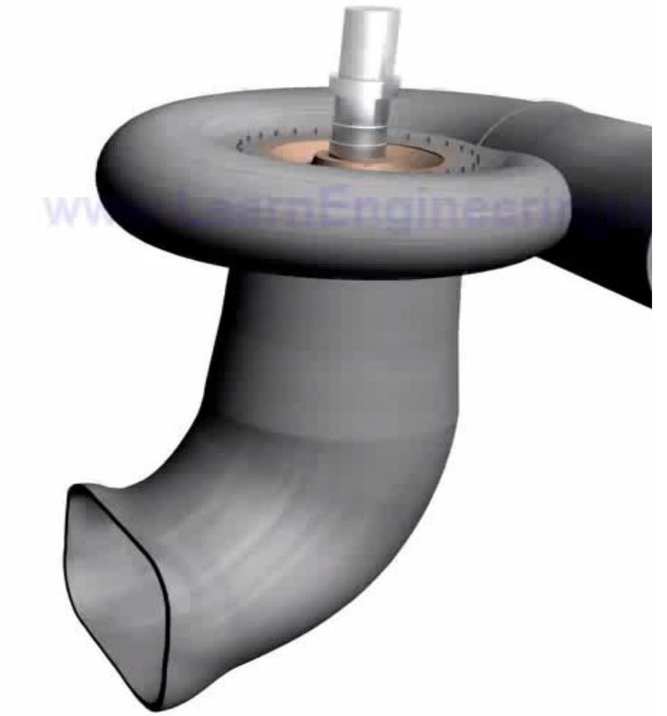
Turbine Scroll Case

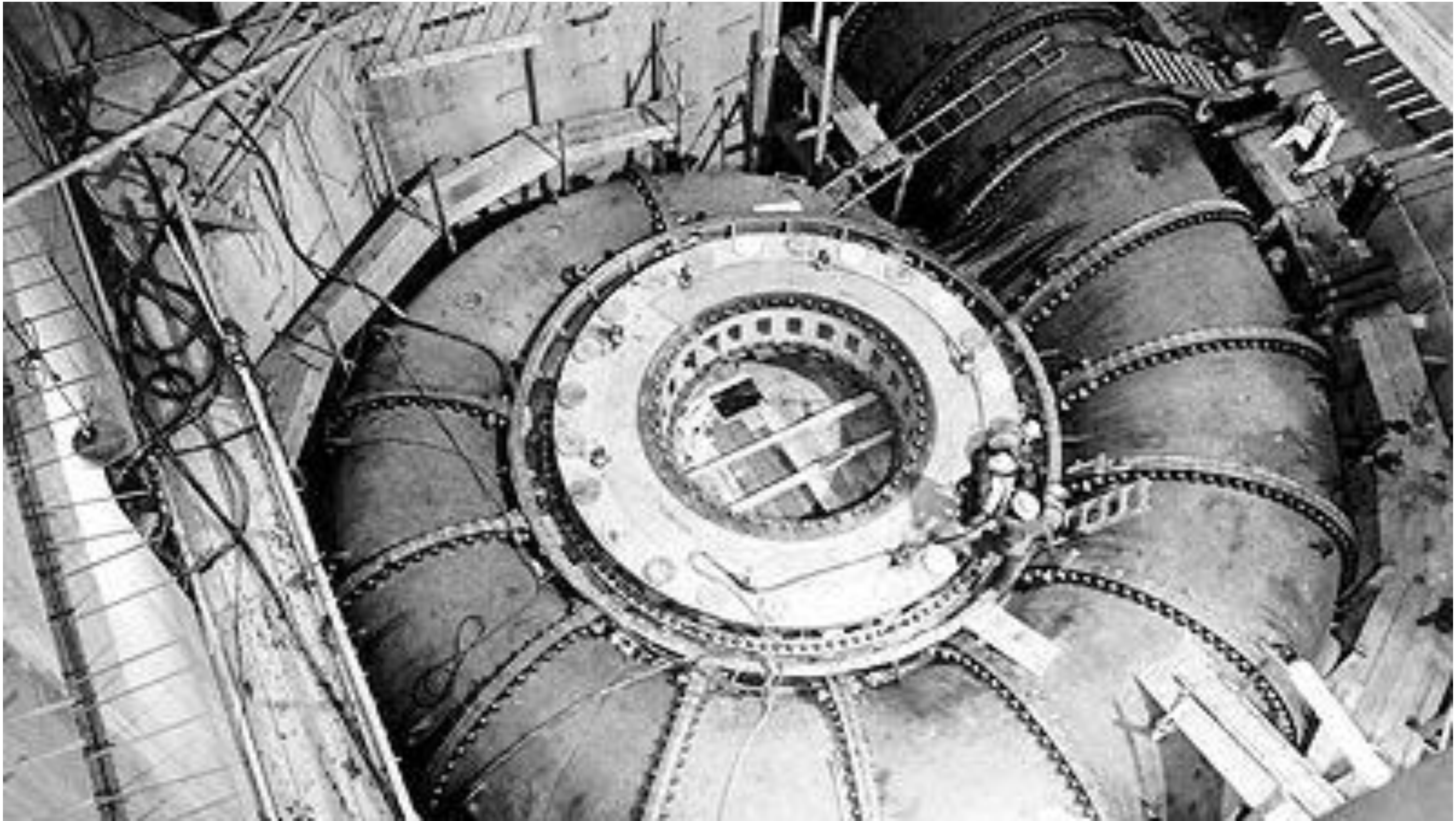
- A scroll case is the conduit directing the water from the intake or penstock to the runner in reaction type turbine.

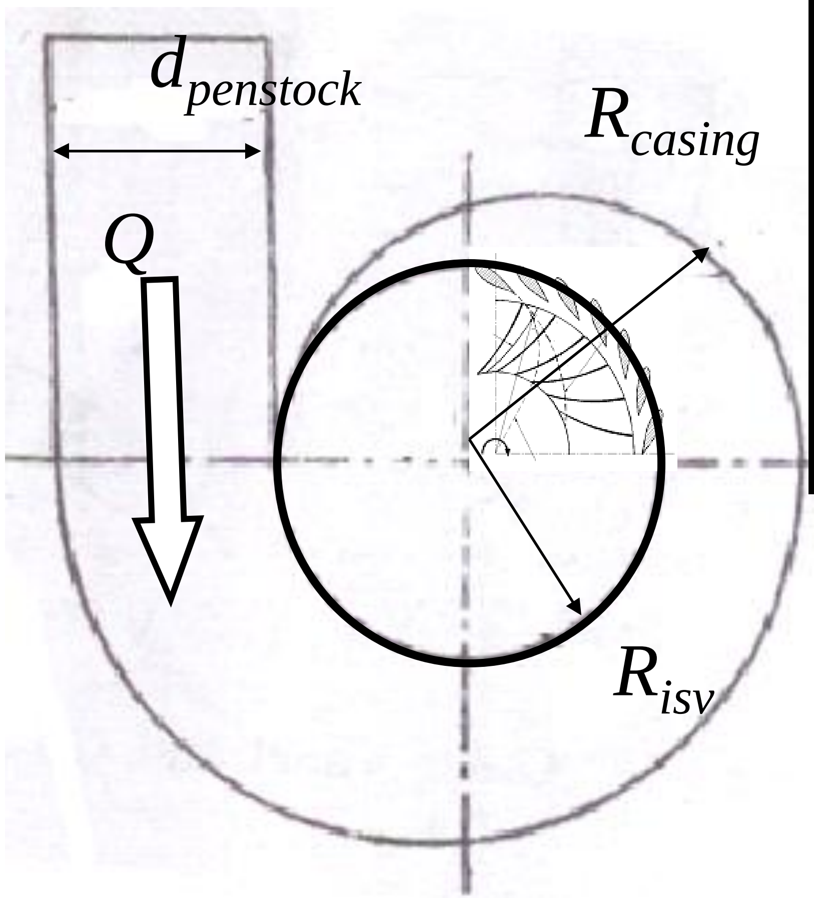
- In impulse wheels a casing is usually provided only to prevent splashing of water & lead water to the tail race.

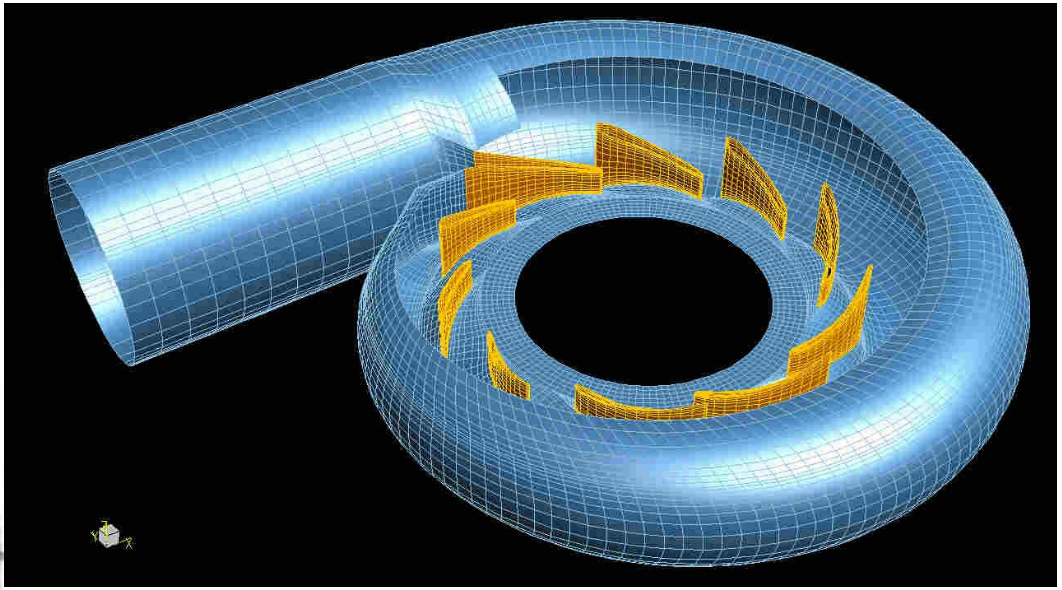
- A spiral shaped scroll case of the correct geometry ensures even distribution of water around the periphery of the runner with the minimum possible eddy formations.



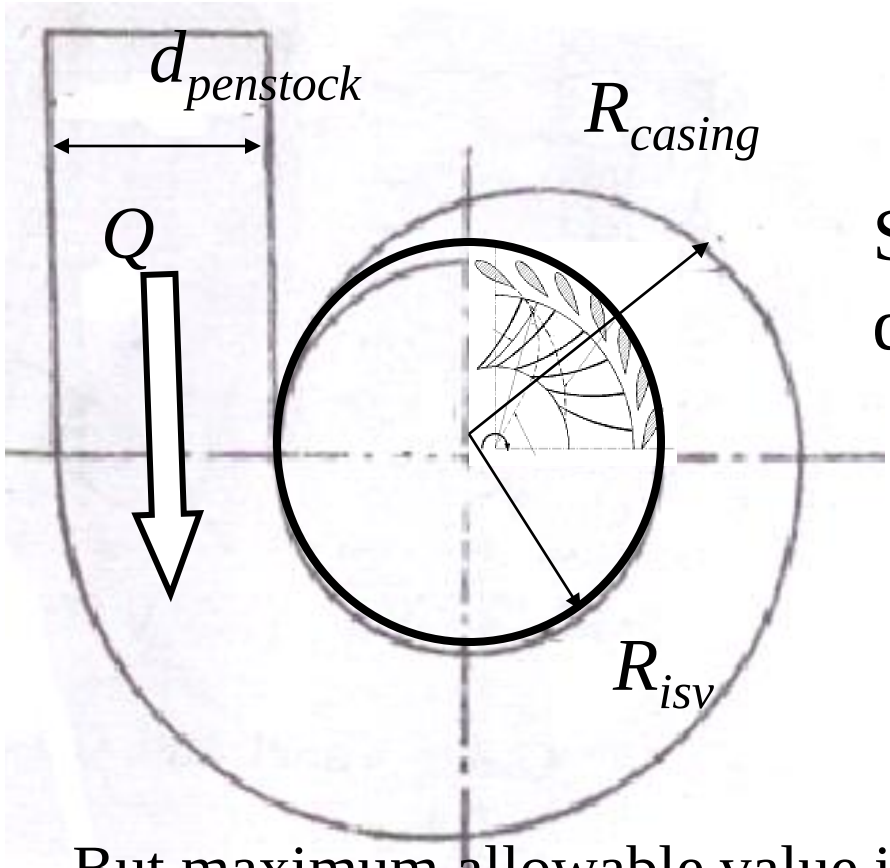








How to select Q ?



Select a suitable value of discharge per unit: Q

$$Q = V_{penstock} \frac{\pi}{4} d^2_{penstock}$$

But maximum allowable value is 10 *m/s*

Maximum allowable head loss in Penstock =2 to 4% of available head

At any angle θ, the radius of casing is:

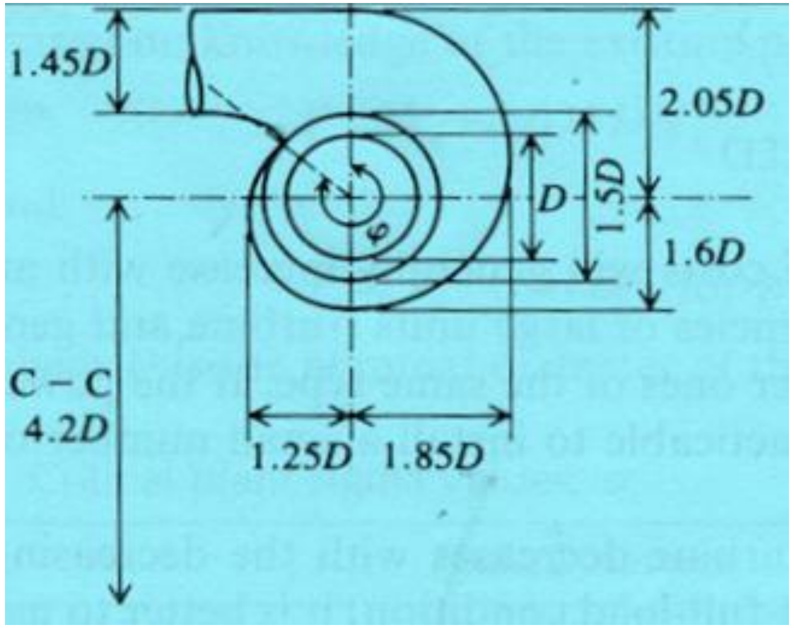
$$R_{\text{casing}} = R_{isv} + \frac{\kappa\theta}{2\pi} d_{penstock} \qquad\qquad Q_\theta = Q\frac{\theta}{2\pi}$$

A full spiral is generally recommended for high head 300m, semi-spiral is recommended for low head < 50m.

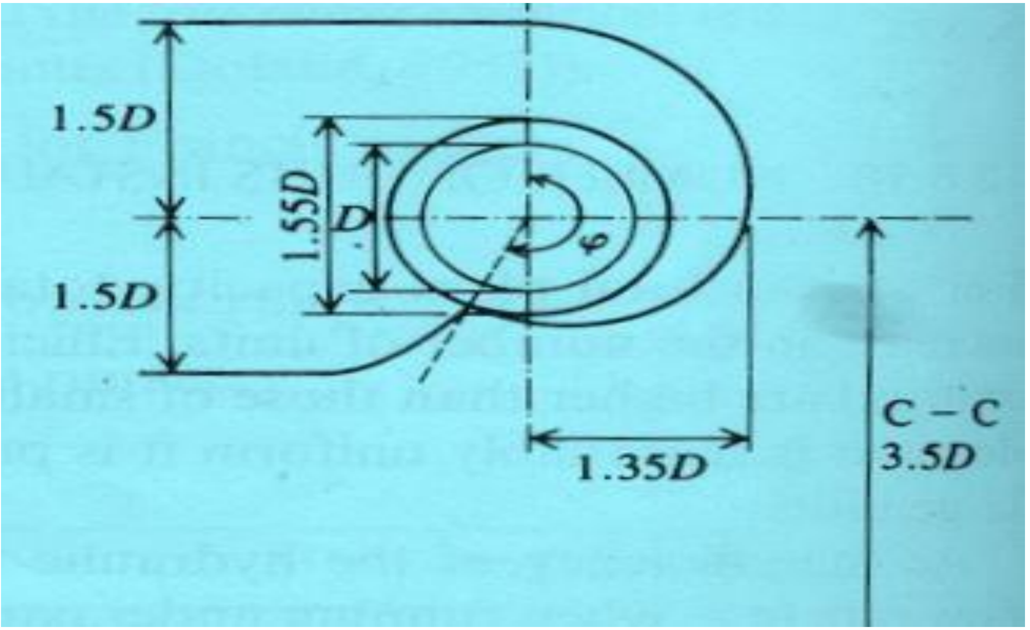
In general κ =1.0, however corrected using CFD.

Turbine Scroll Case

- Recommended dimensions of scroll casings



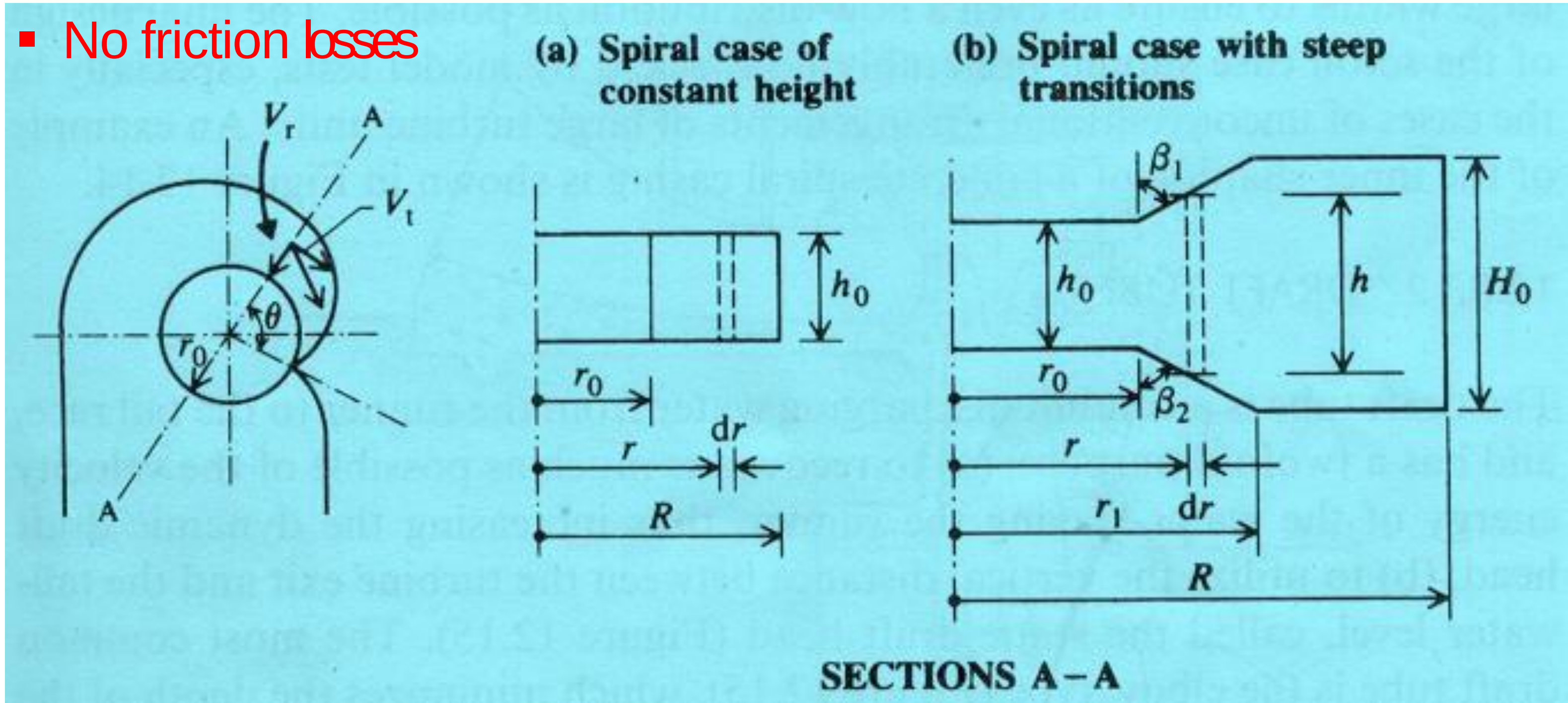
a) *Francis turbine with steel spiral*



b) *Propeller turbine with partial spiral*

Turbine Scroll Case

- The design of the shape of the spiral case is governed by the flow requirements.
- Initial investigation should be based on assumptions:
  - Spiral case of constant height
  - An evenly distributed flow in to the turbine
  - No friction losses



- The discharge in section of spiral case defined by angle θ is

$$q = \frac{Q\theta}{2\pi}$$

Q is the total discharge to the runner

$$V_t = \frac{k}{r}$$

k is a vortex strength & given by: $k = 30\frac{\eta g H}{N\pi}$

- The discharge through the strip ***dq*** is given by

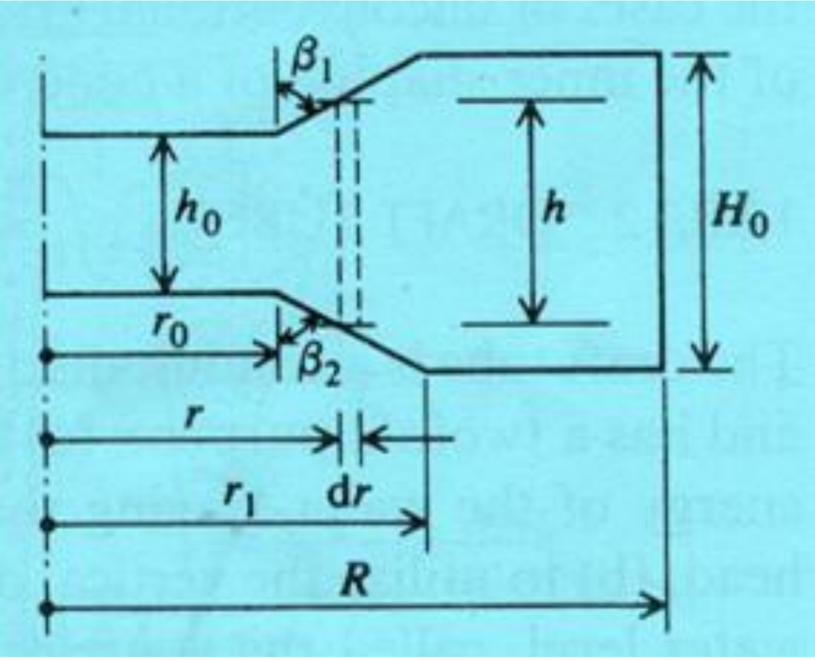
$$dq = V_t h_0 dr$$

$$dq = \frac{k}{r} h_0 dr$$

$$q = \int dq = \int_{r_0}^{R} k h_0 \frac{dr}{r} = \frac{Q\theta}{2\pi} \text{ or } \ln\frac{R}{r} = \frac{Q\theta}{2\pi k h_0}$$

Turbine Scroll Case

- The most economical design of a power station substructure and the narrowest spiral case can be obtained by choosing a rectangular section adjoining the guide vanes (entrance ring) by step transition (symmetrical or asymmetrical)



$$h = h_0 + \alpha(r - r_0)$$

$$\alpha = \cot\beta_1 + \cot\beta_2$$

$$\frac{Q\theta}{2\pi k} = \int_0^{r_1} h\frac{dr}{r} + \int_{r_1}^{R} H_0\frac{dr}{r}$$

$$\frac{Q\theta}{2\pi k} = h_0 - \alpha r_0 \ln\left(\frac{r_1}{r_0}\right) + H_0 - h_0 + H_0 \ln\left(\frac{R}{r_1}\right)$$

Turbine Scroll Case

- Knowing r_1 from $r_1 = \left(\frac{H_0 - h_0}{\alpha}\right) + r_0$ the value of R defining the shape of the spiral case can be determined.

- Shape at various θ is determined by assuming existence of uniform velocity equal to entrance velocity,

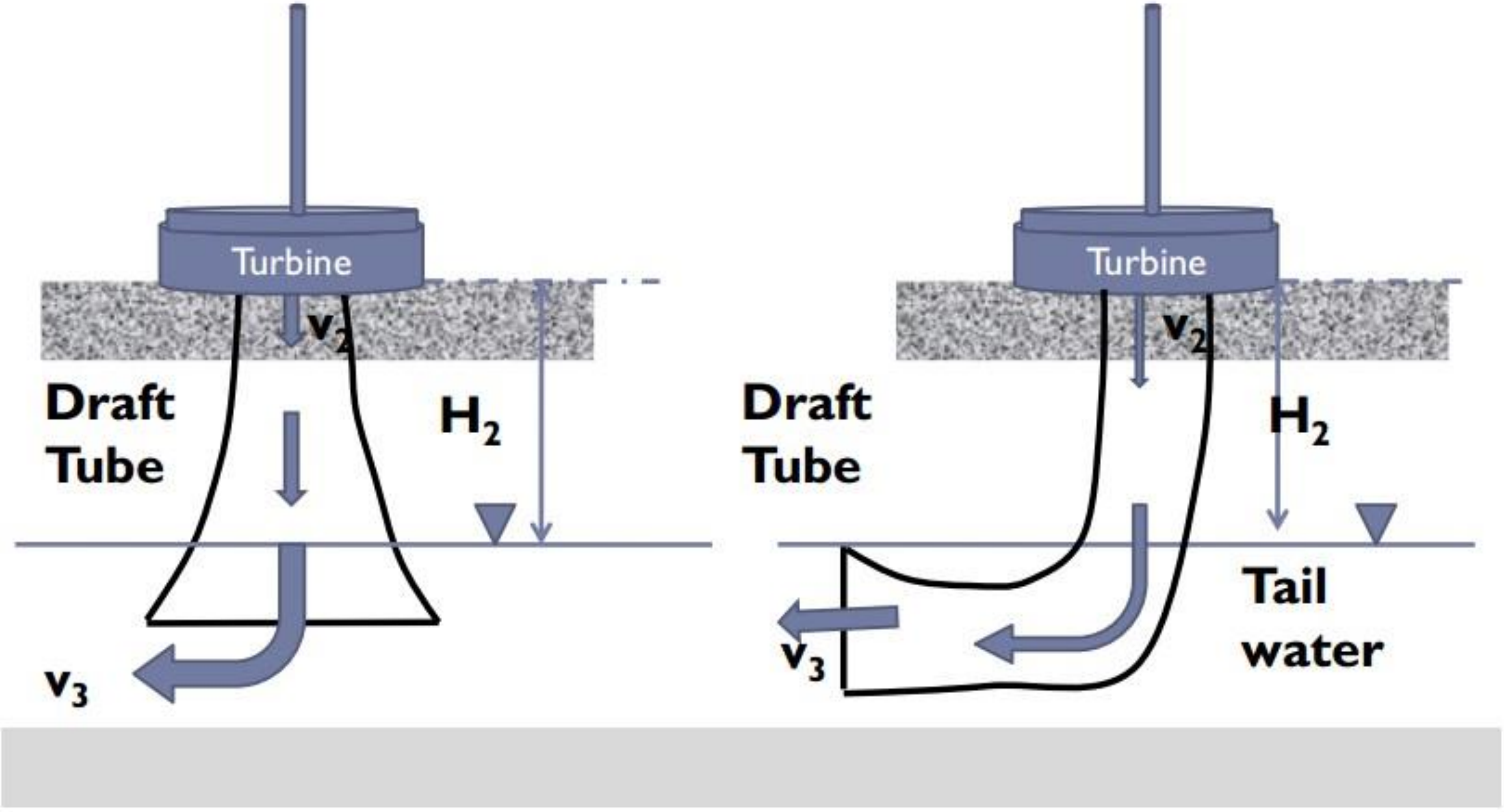
$$v_0 \cong 0.2\sqrt{2gH} \;\; and \; q_i = \frac{Q\theta_i}{2\pi}$$

- Area of cross-section at angle θ_i

$$A_i = \frac{q_i}{v_0} = 0.18\frac{Q\theta_i}{\sqrt{H}}$$

Draft Tubes

- A draft tube is a conduit discharging water from the turbine runner to the tailrace.
- Two types: Elbow & Conical/Vertical



- A draft tube provided:
  - To recover as much as possible of the velocity energy of the water leaving the runner, which otherwise would have gone to waste as an exit loss, thus increasing the dynamic draft head.
  - To utilize the vertical distance between the turbine exit and the tail-water level, called the static draft head. In other words, to allow the turbine to be set at higher elevation without losing the advantage of elevation difference.

Draft Tubes

A draft tube provided:

- In the absence of a draft tube, the energy at the turbine outlet, as counted from the tail water elevation, would be:

$$E_2 = H_2 + \frac{v_2^2}{2g}$$

- In the presence of a draft tube, the energy at the end of the draft tube, will be:

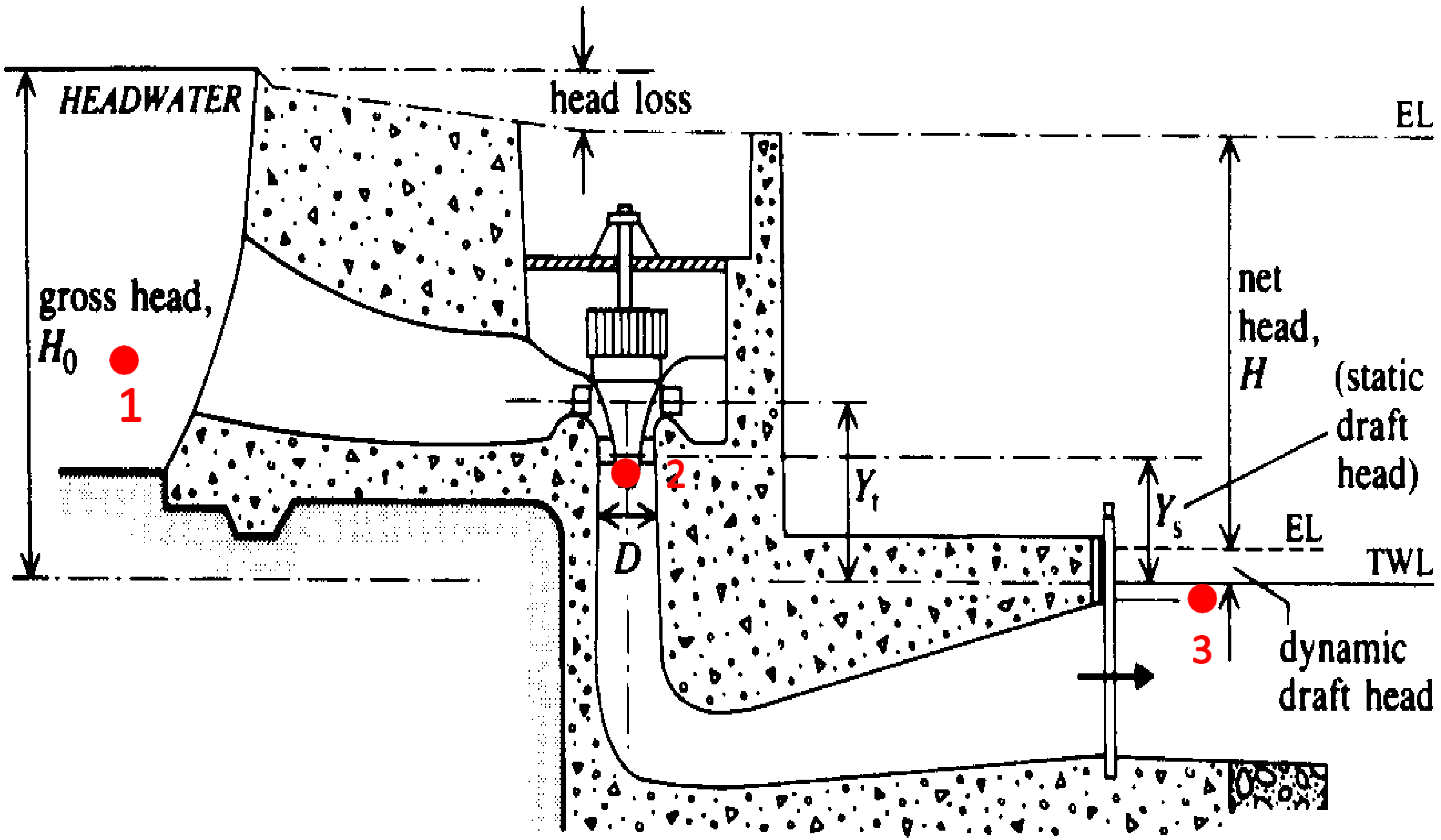
$$E_3 = \frac{v_3^2}{2g}$$

- The difference between E_2 and E_3

$$E_2 - E_3 = H_2 + \frac{v_2^2}{2g} - \frac{v_3^2}{2g}$$

This equation suggests that the draft tube allows the turbine to utilize:

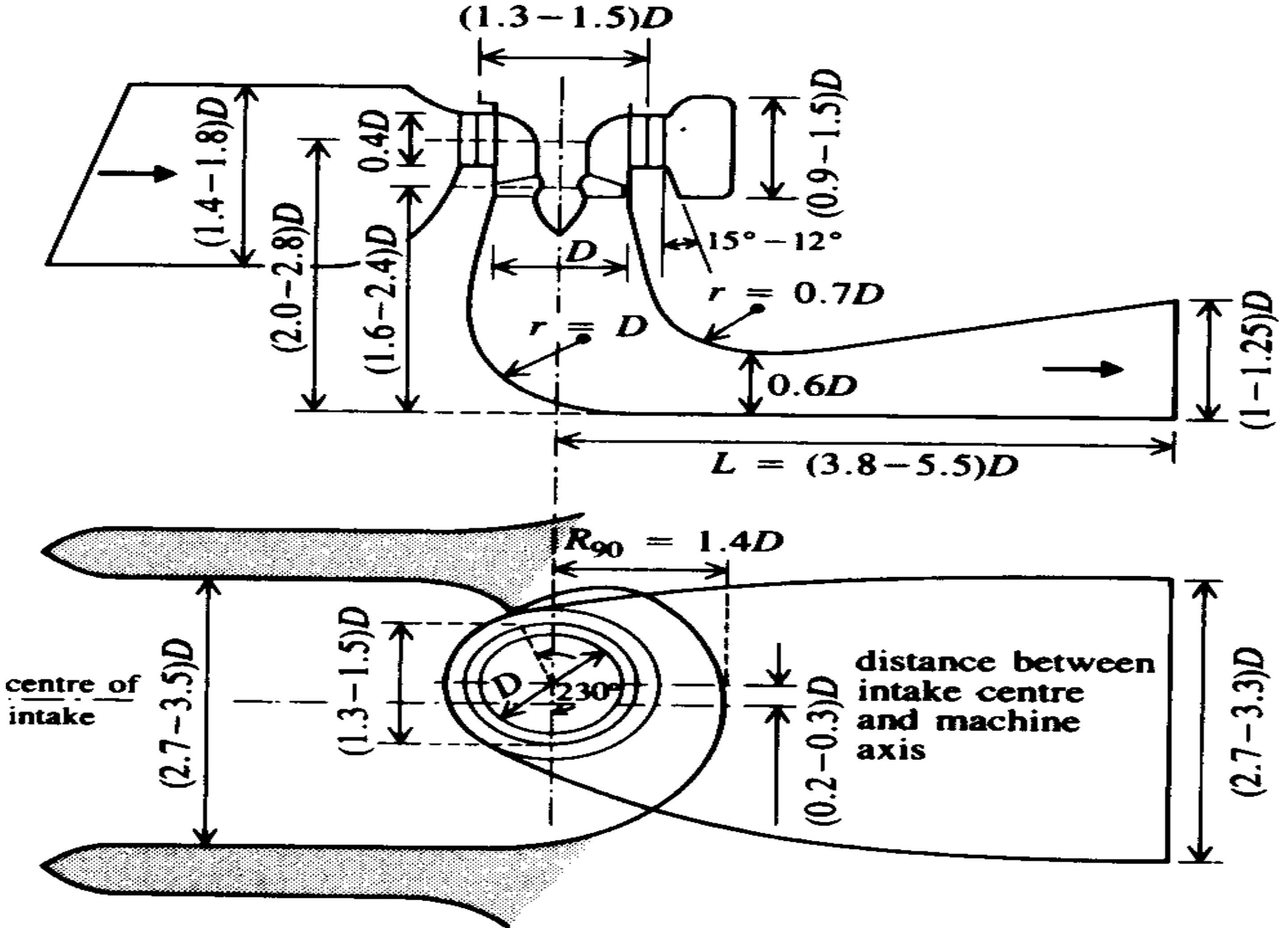
- *All of the energy due to the elevation Y_s of the runner above the tail water level, &*
- *Most of the kinetic energy of the water that leaves the turbine, because, owing to the increase in area, the velocity v_3 is much lower than v_2.*



HEADWATER
head loss
EL
gross head,
H_0
1
2
net
head,
H
(static
draft
head)
Y_t
Y_s
EL
TWL
D
3
dynamic
draft head

Draft Tubes

- Recommended dimensions of an elbow-type draft tube *(after Mosonyi)*



Draft Tubes

- Writing Bernoulli's equation between section at the turbine exit (2) and section at the draft tube exit (3), neglecting the energy lost in the draft tube,

$$H_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g} = 0 + 0 + \frac{v_3^2}{2g}$$

  - Thus the pressure under the runner is negative as shown below

$$\frac{p_2}{\gamma} = -\left[H_2 + \left(\frac{v_2^2}{2g} - \frac{v_3^2}{2g} \right) \right]$$

- That means, a vacuum is maintained under the runner

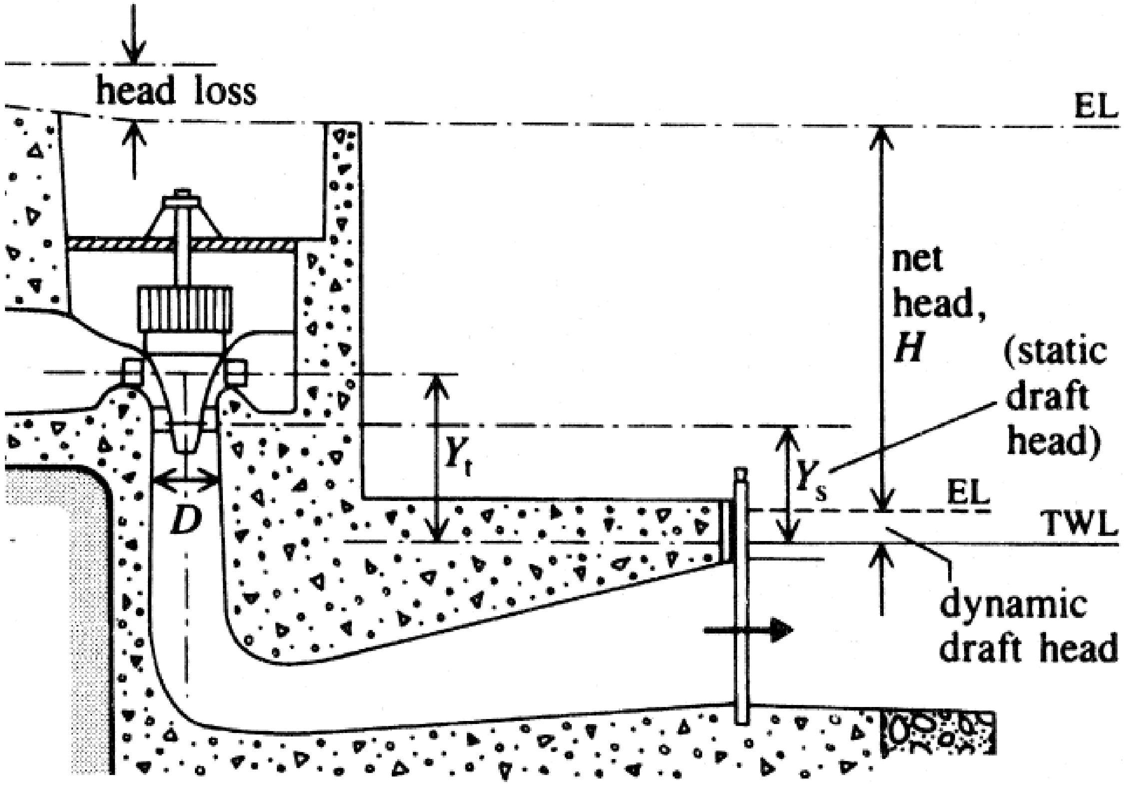
- In order to avoid Cavitation: $\frac{P_2}{\gamma} > \frac{P_v}{\gamma}$

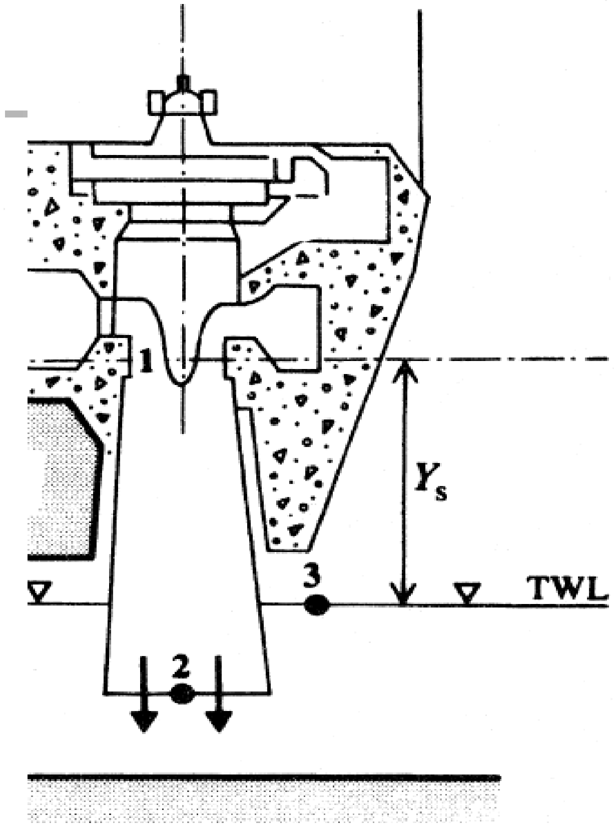
Cavitation

- **A reduced pressure under the blades (or buckets) of a turbine runner may lead to cavitation – phenomenon detrimental to the turbine. The term cavitation basically refers to the ability of cold water to boil under low pressure.**
- **Under a normal absolute barometric pressure of 1 bar water starts to boil at 100 ^{o}C. However, when the pressure drops to 0.033 bar (which is called the critical pressure, P_{cr}) it may begin to bubble at 25 ^{o}C, that is, at normal river water temperature. When the pressure under a runner approaches P_{cr}, the water in the stream starts boiling, giving rise to cavities (known as cavitation bubbles) filled with water vapor.**
- **The boundary between the low pressure zone immediately under the blades (or buckets) and the high pressure zone in the stream above the runner follows an extremely unstable pattern.**
- **The cavitation bubbles find themselves from time to time in the high pressure zone. As a result, the vapor instantly condenses and a cavitation bubble collapses. As this takes place, an enormous pressure develops at the bubble centre, which spreads quickly in an explosion-like manner. A series of such micro-explosions following one another at very short intervals causes a good deal of noise and vibration in the turbine and may provoke the runner blades into pitting.**

Cavitation and turbine setting

- **Cavitation results in pitting, vibration and reduction in efficiency and is certainly undesirable. Runners most seriously affected by cavitation are of the reaction type, in which the pressures at the discharge ends of the blades are negative and can approach the vapor pressure limits.**
- **Cavitation may be avoided by suitably designing, installing and operating the turbine in such a way that the pressures within the unit are above the vapor pressure of the water.**
- **Turbine setting or draft head, *Ys* (*Figs Next slide*), is the most critical factor in the installation of the reaction turbines.**





- The cavitation characteristic of a hydraulic machine is defined by the cavitation coefficient or plant sigma (σ), given by:

$$\sigma = \left(\frac{Ha - H_v - Y_s}{H} \right)$$

  - where Ha – Hv = Hb, is the barometric pressure head (at sea level and 20°C, Hb = 10.1 m), and H is the effective head on the runner.

- The maximum permissible turbine setting Y_{smax} (elevation above tail-water to the centre line of the propeller runners, or to the bottom of the Francis runners) can be written as:

$$Y_{smax} = H_b - \sigma_c H \quad \text{(Thoma's formula)}$$

  - ✓ σ_c is the minimum (critical) value of at which cavitation occurs (usually determined by experiments).
  - ✓ If Y_s is *-ve the runner must be* set below the TW

	Francis runners					Propeller runners	
N_s	75	150	225	300	375	600	750
σ_c	0.025	0.10	0.23	0.40	0.64	0.8	1.5

Table:- Critical plant sigma value of σ_c

Cavitation in Turbine & Turbinesetting

$$\boxed{\sigma_c = 0.0432\left(\frac{N_s}{100}\right)^2}$$

for Francis

$$\boxed{\sigma_c = 0.28 + 0.0024\left(\frac{N_s}{100}\right)^3}$$

for propeller

- The preliminary calculation for the elevation of the distributor above the TWL, Y_t is:

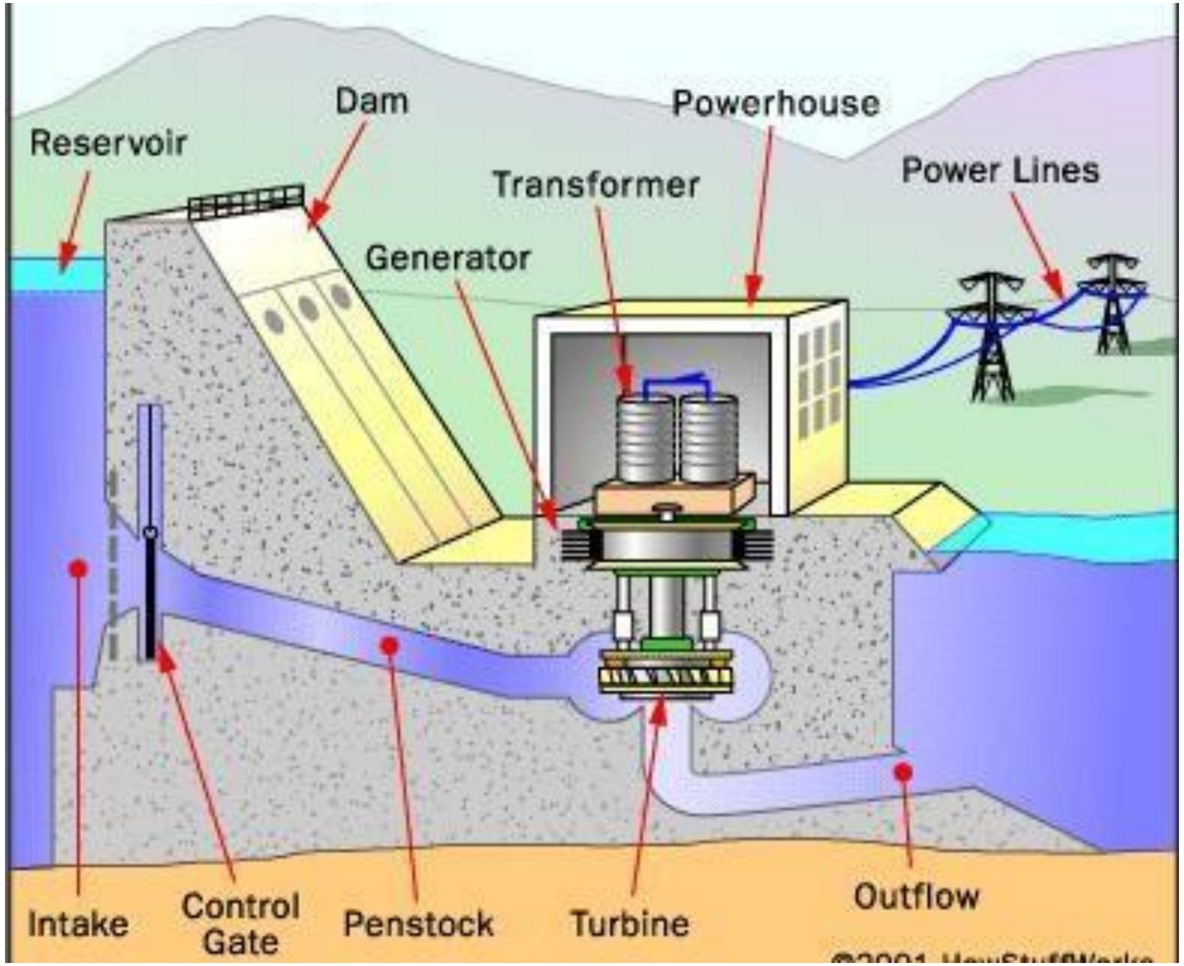
$$Y_t = Y_s + 0.025DN_s^{0.34}$$ *For Francis*

$$Y_t = Y_s + 0.025D$$ *For propeller*

Where: D is the nominal diameter of the runner

Generator and Turbine Control

- Generators is the machine that transform mechanical energy into electrical energy.

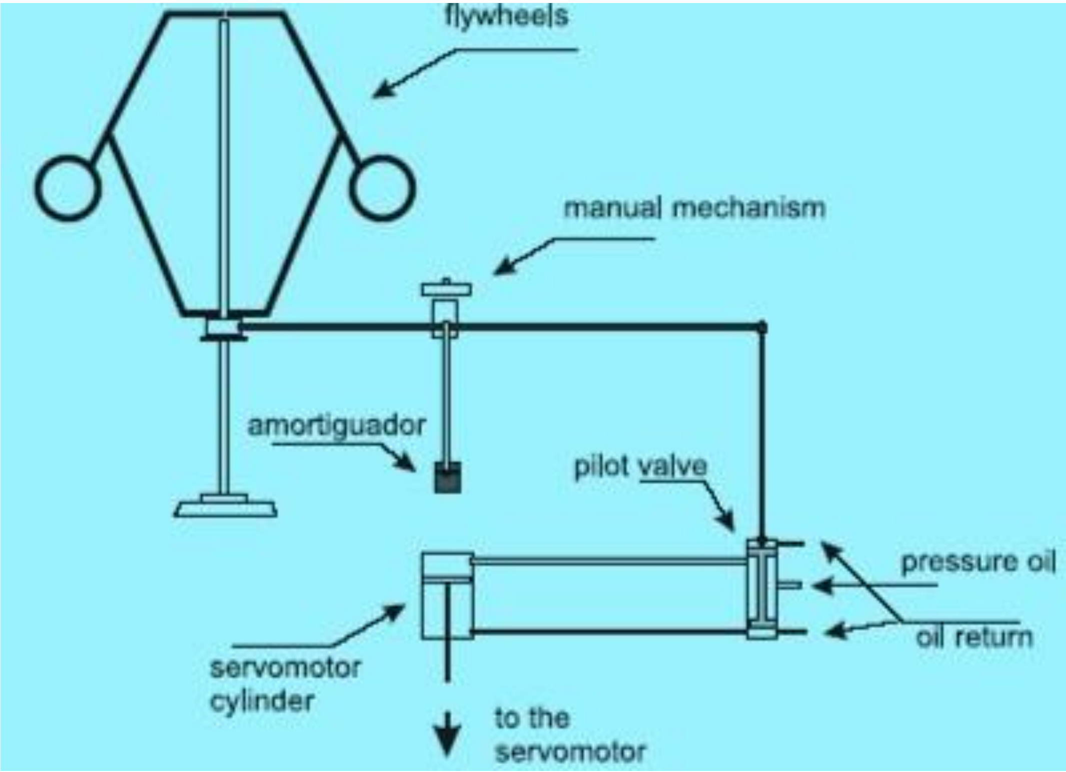


- Turbines are designed for a certain net head and discharge.

- Any deviation from these parameters must be compensated for, by opening or closing control devices such as the wicket-vanes or gate valves to keep constant, either the outlet power, the level of the water surface in the intake or the turbine discharge.

- The parameter to be controlled is the runner speed, which controls the frequency

- Two approaches to control the runner speed:
  - Controlling the water flow to the turbine or by keeping the water flow constant and
  - Adjusting the electric load by an electric ballast load connected to the generator terminals



Thank you!!