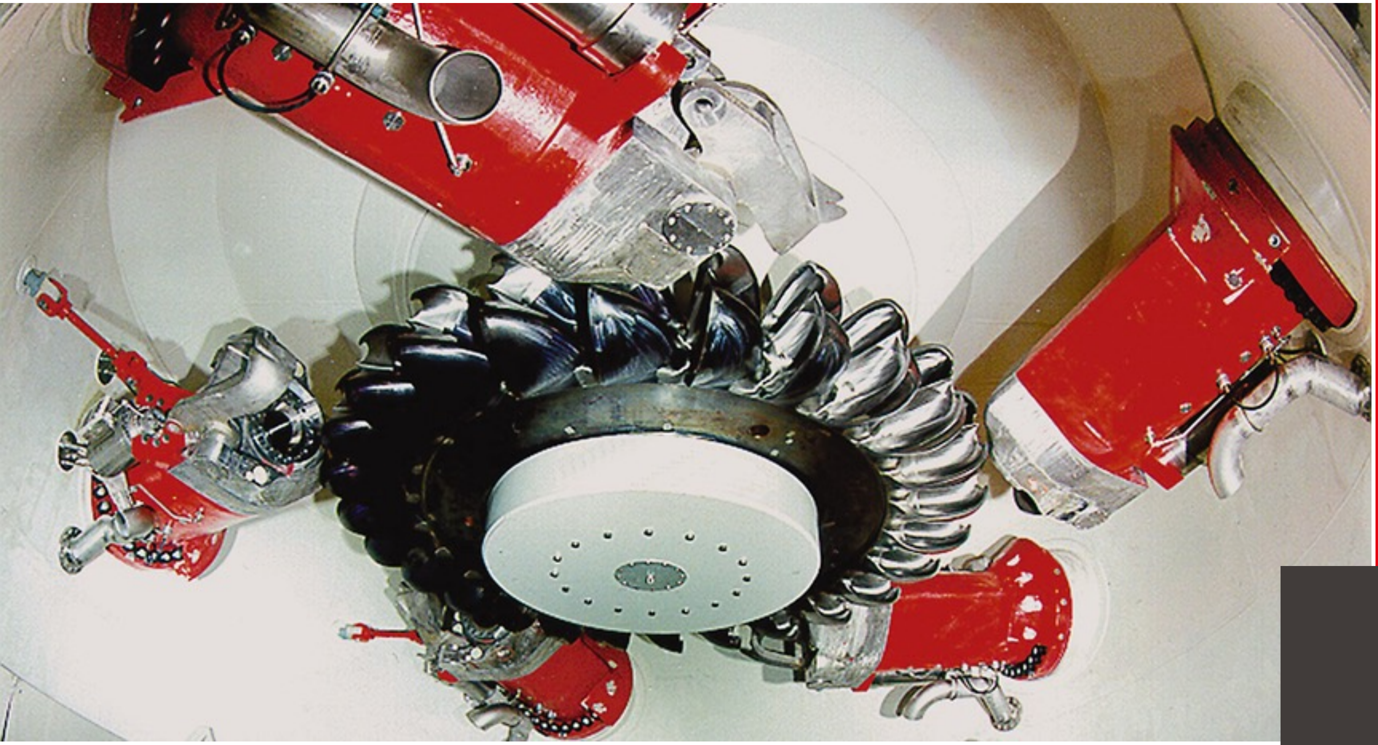
EPFL



Chapter 8: Hydraulic Turbines

ME-342 Introduction to turbomachinery

École polytechnique fédérale de Lausanne

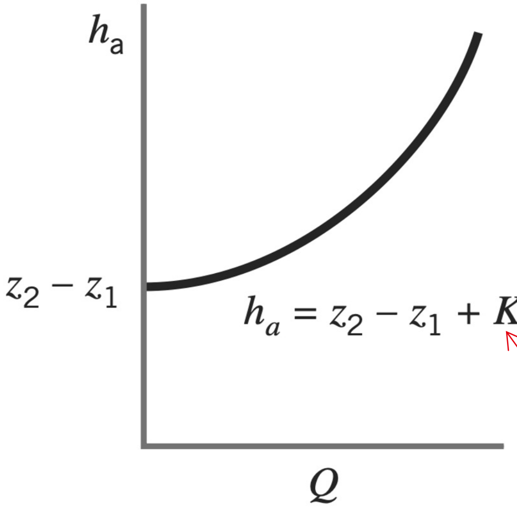
Prof. Eunok Yim, HEAD-lab.

Last week

- **K vs. K_L**

Loss coefficient for Minor loss

System equation



$$h_a = z_2 - z_1 + KQ^2$$

An arbitrary constant

Exercise

Pump water with 0.014 m^3/s the required NPSH is 4.5 m specified pump manufacturer. The water temperature is 30 ºC, 101.3 kPa. The loss occurs mainly due to the filter at inlet with K_L = 20. Friction loss is neglected. Pipe is with diameter of 10 cm. Determine the max height z the pump can be located without the cavitation.

$$\mathrm{NPSH_A} = \frac{p_{atm}}{\gamma} - \Delta z - \sum h_L - \frac{p_v}{\gamma}$$

$\sum h_L = K_L \frac{V^2}{2g}$

Cavitation occurs when $NPSH_R = NPSH_A$

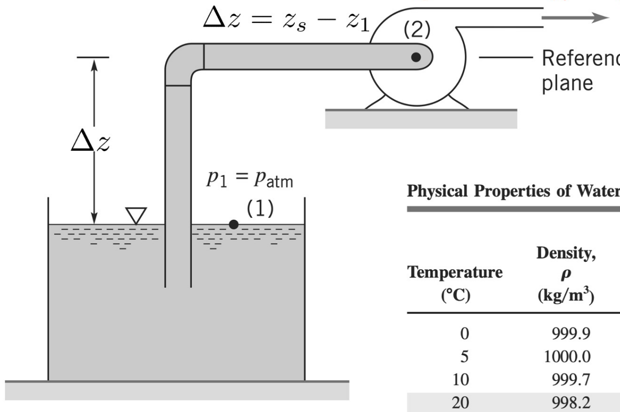
$$NPSH_A = NPSH_R = 4.5\,m = \frac{101.3\times10^3\ Pa}{9.765\times10^3\ N/m^3} - \Delta z - 20\cdot\underbrace{\frac{V^2}{2g}}_{=3.23} - \frac{4.243\times10^3}{9.765\times10^3}$$

$$V = \frac{Q}{A} = \frac{0.014}{\pi\cdot0.05^2} = 1.78\ m/s$$

$$= 10.37 - \Delta z - 3.23 - 0.43$$

$$\Delta z = 10.37 - 4.5 - 3.23 - 0.43$$

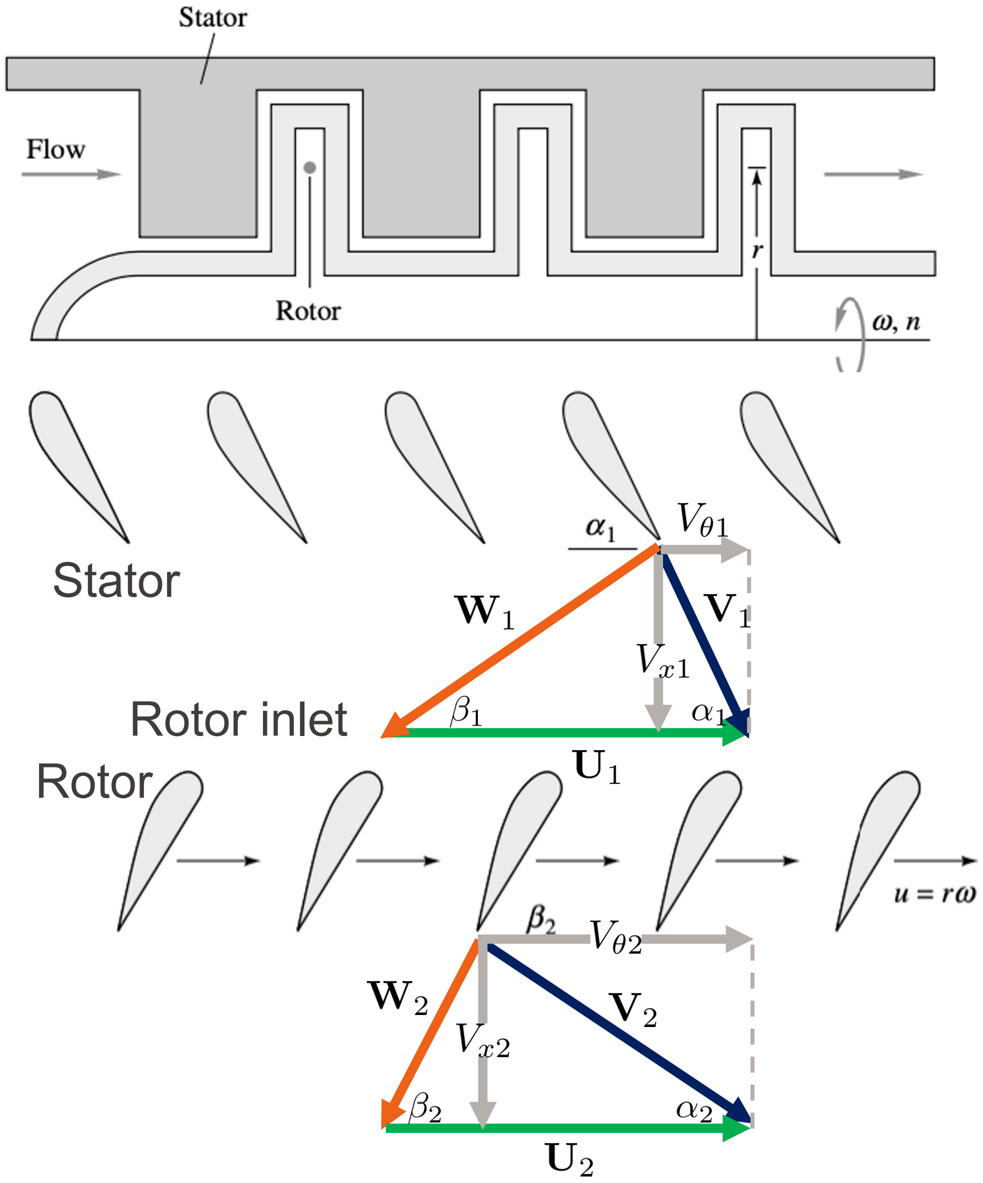
$$= 2.21\ m$$



Physical Properties of Water (SI Units)[a]

Temperature (°C)	Density, ρ (kg/m^3)	Specific Weight[b], γ (kN/m^3)	Dynamic Viscosity, μ ($N\cdot s/m^2$)	Kinematic Viscosity, ν (m^2/s)	Surface Tension[c], σ (N/m)	Vapor Pressure, p_v [N/m^2(abs)]
0	999.9	9.806	1.787 E − 3	1.787 E − 6	7.56 E − 2	6.105 E + 2
5	1000.0	9.807	1.519 E − 3	1.519 E − 6	7.49 E − 2	8.722 E + 2
10	999.7	9.804	1.307 E − 3	1.307 E − 6	7.42 E − 2	1.228 E + 3
20	998.2	9.789	1.002 E − 3	1.004 E − 6	7.28 E − 2	2.338 E + 3
30	995.7	9.765	7.975 E − 4	8.009 E − 7	7.12 E − 2	4.243 E + 3
40	992.2	9.731	6.529 E − 4	6.580 E − 7	6.96 E − 2	7.376 E + 3

Last week - Axial pump



Since the stator is fixed, ideally the absolute velocity V_1 is parallel to the trailing edge of the blade

$$V_{x1} = V_{x2} = V_x = \frac{Q}{A} = \mathrm{const}$$

The ideal head expressed stator angle α_1 and rotor angle β_2

$$h_i = \frac{U_2 V_{\theta 2} - U_1 V_{\theta 1}}{g}$$

Strictly speaking, this applies only to a single streamtube of radius *r*, but it is a good approximation for very short blades if *r* denotes the average radius.

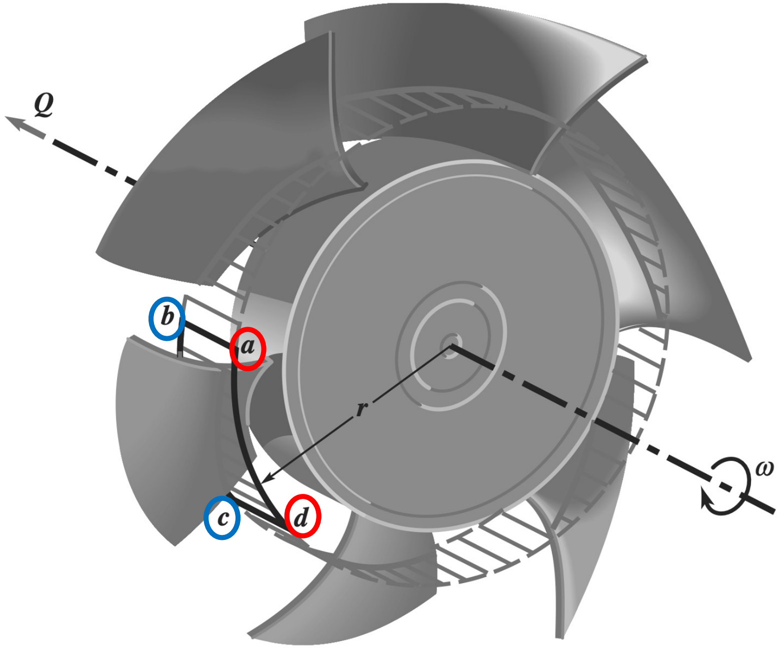
Recall - Chapter 5

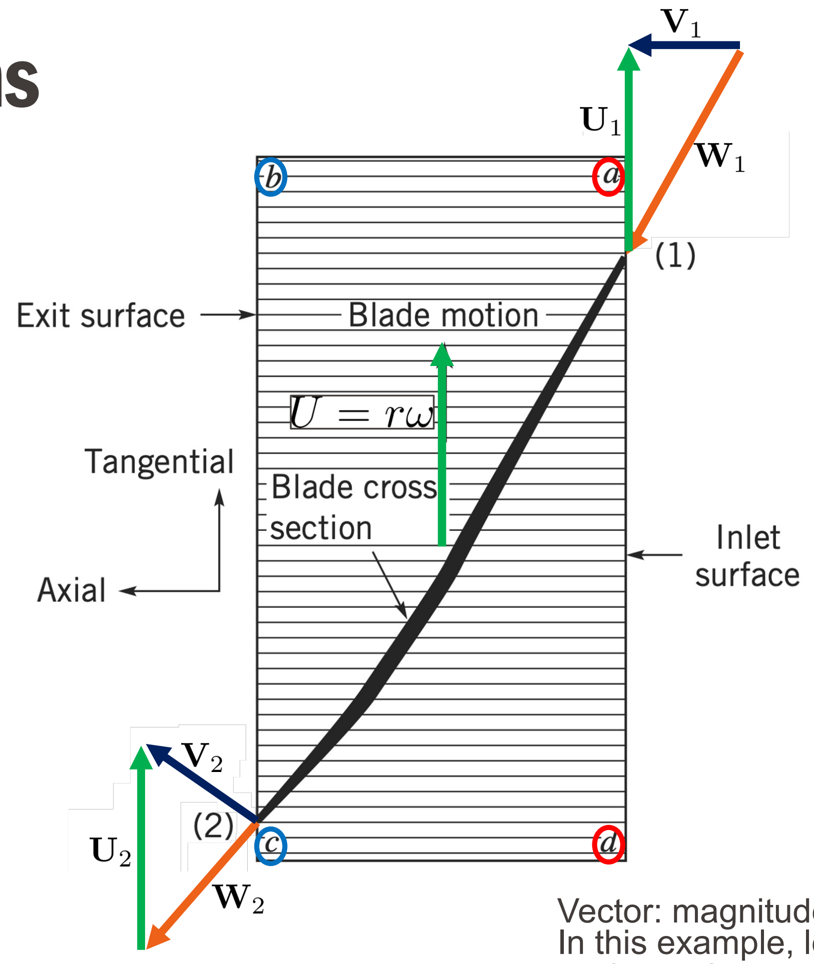
Basic Energy Considerations

Chap. 5 | E. Yim

INTRODUCTION TO TURBOMACHINERY

- **Velocity diagram (fan)**



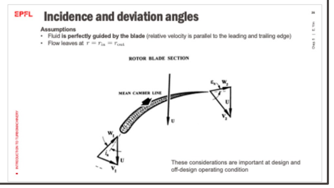


Vector: magnitude and angle
In this example, let's assume we know the magnitude of **W**

Assumptions

- Fluid is **perfectly guided by the blade** (relative velocity is parallel to the leading and trailing edge)*
- Flow leaves at $r = r_{\text{in}} = r_{\text{out}}$

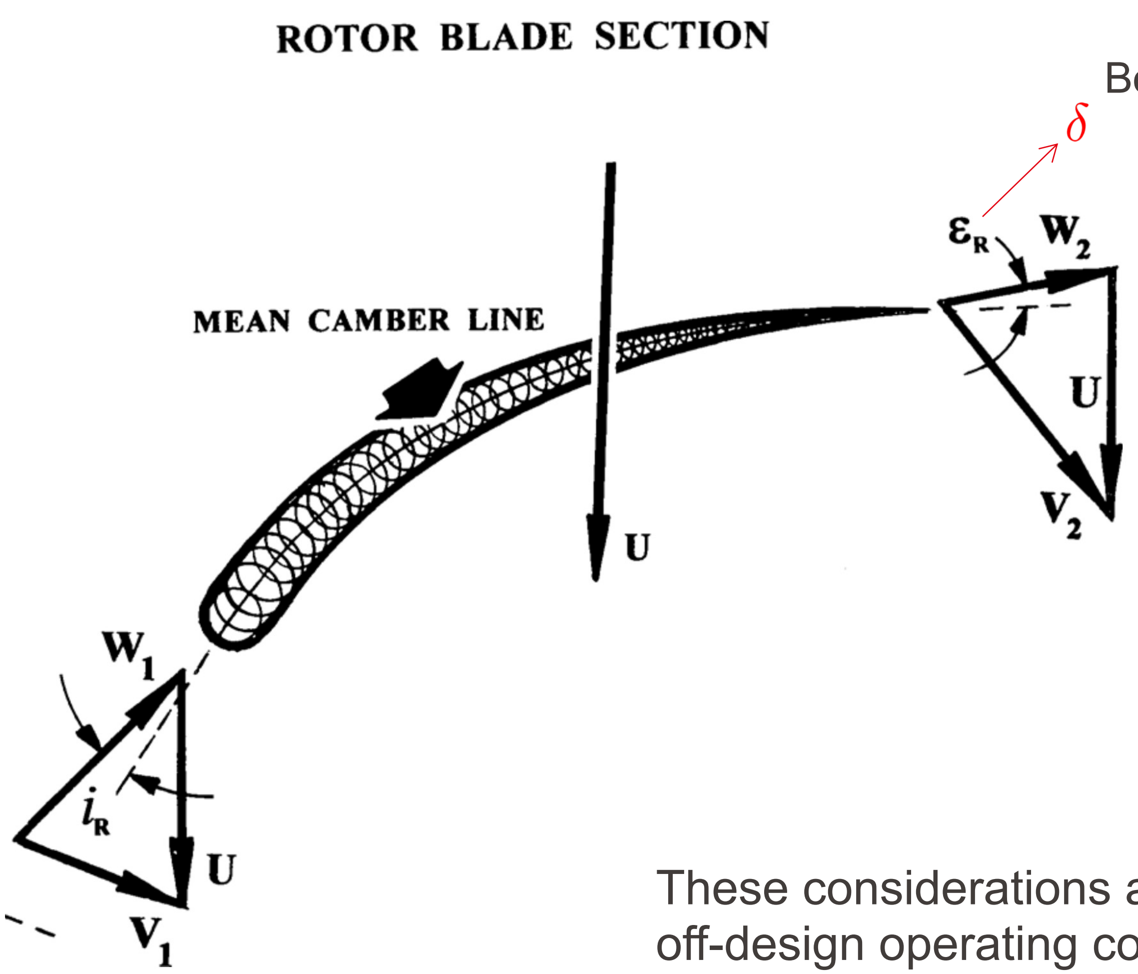
*sometimes leading edge (inlet) **V** is prescribed



Incidence and deviation angles

Assumptions

- Fluid **is perfectly guided by the blade** (relative velocity is parallel to the leading and trailing edge)
- Flow leaves at $r = r_{\text{in}} = r_{\text{out}}$



Both ε and δ are used..

These considerations are important at design and off-design operating condition

Incidence and deviation angles

The **incidence** is the difference between the inlet flow angle (α) and the blade **inlet** angle (α'):

$$i = \alpha_1 - \alpha_1'$$

The **deviation** is the difference between the exit flow angle and the blade **exit** angle:

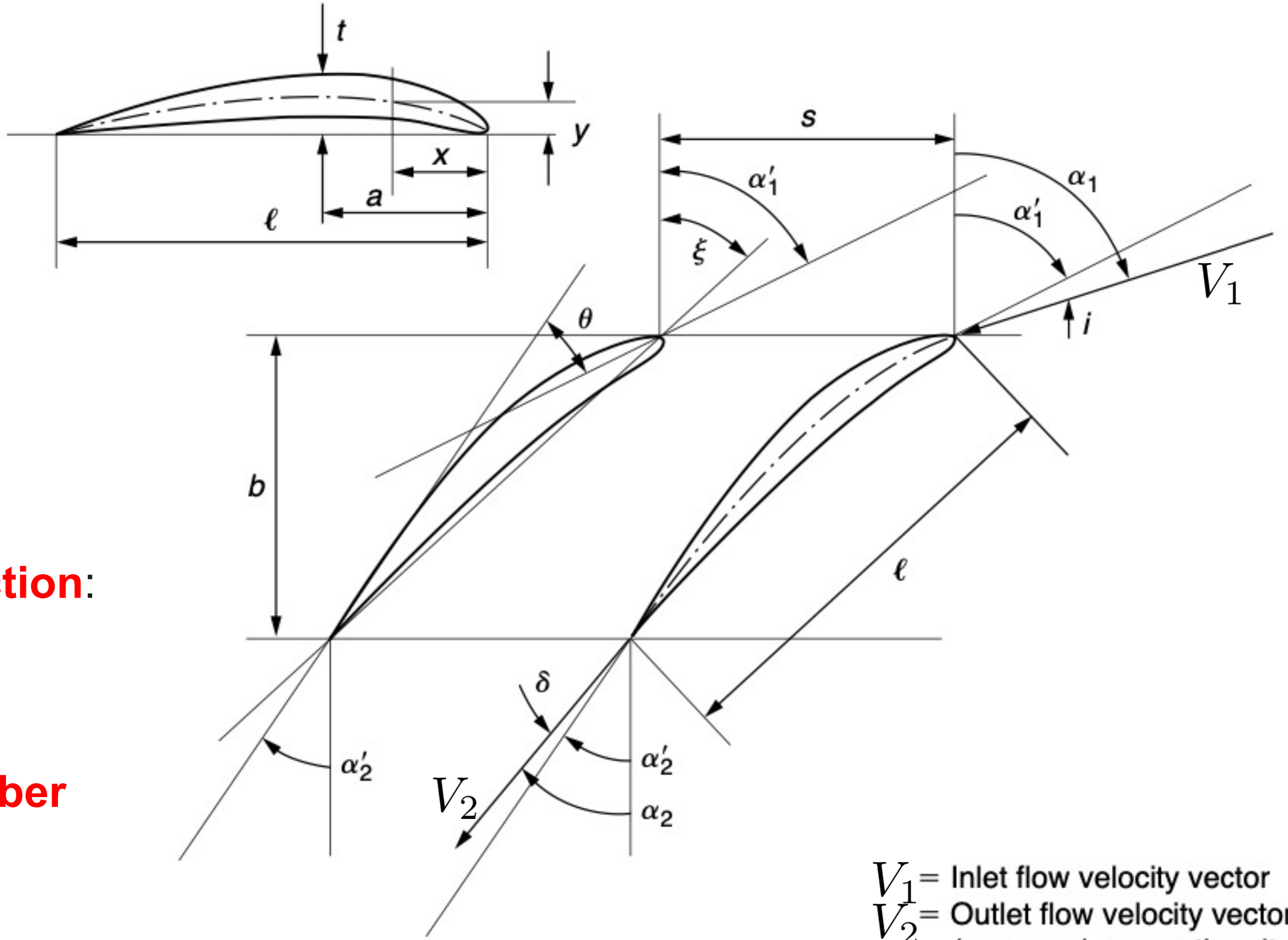
$$\delta = \alpha_2 - \alpha_2'$$

The change in angle of the flow is called **deflection**:

$$\varepsilon = \alpha_1 - \alpha_2$$

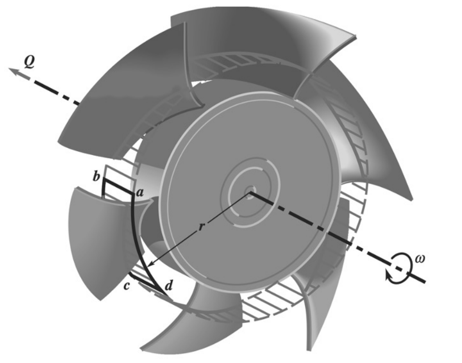
The change in angle of the blade is called **camber angle**:

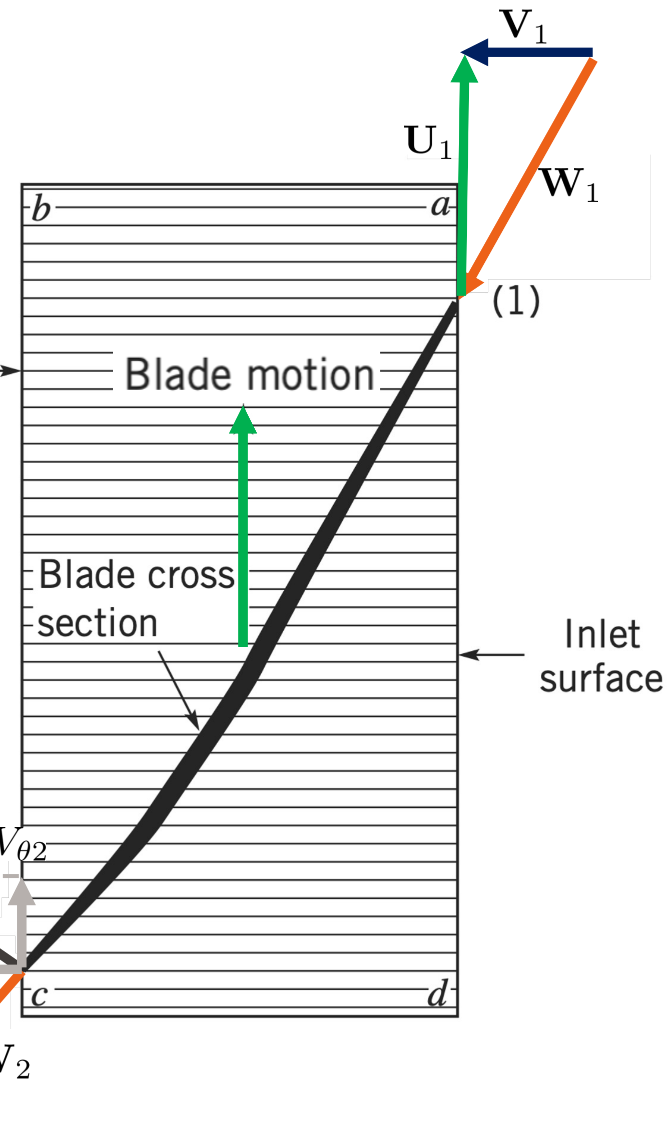
$$\theta = \alpha_1' - \alpha_2'$$

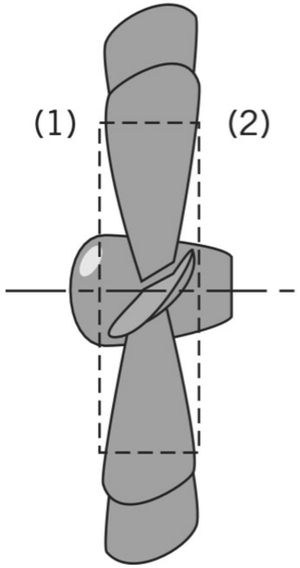


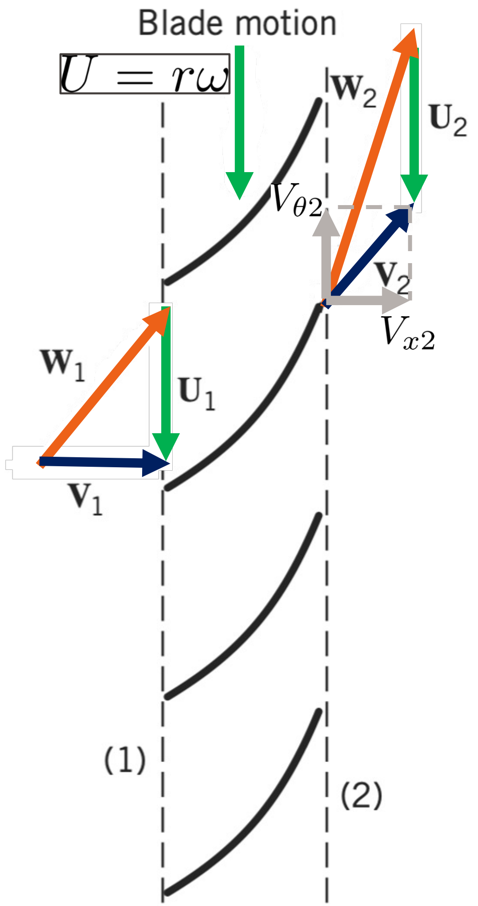
V_1 = Inlet flow velocity vector
V_2 = Outlet flow velocity vector (averaged across the pitch)

Actung! Pump vs. turbine





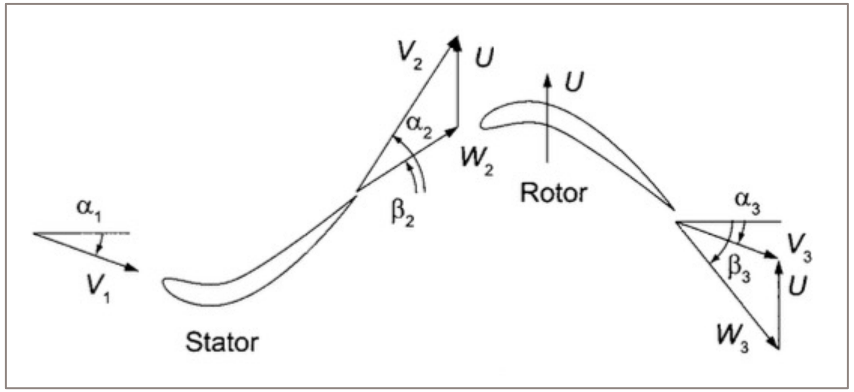


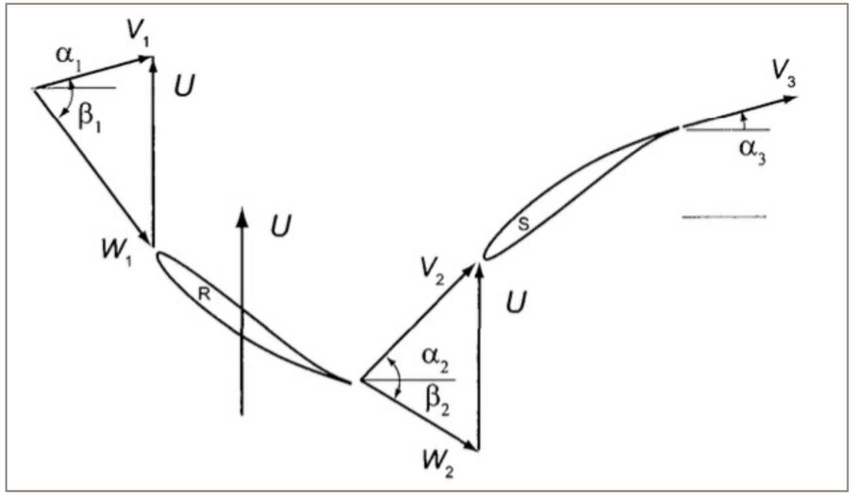


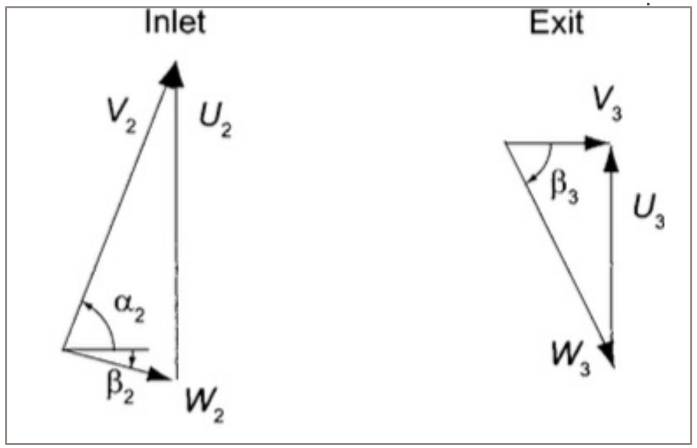
$$w_{\text{shaft}} = \frac{\dot{W}_{\text{shaft}}}{\rho Q} = U_2 V_{\theta 2} - U_1 V_{\theta 1}$$

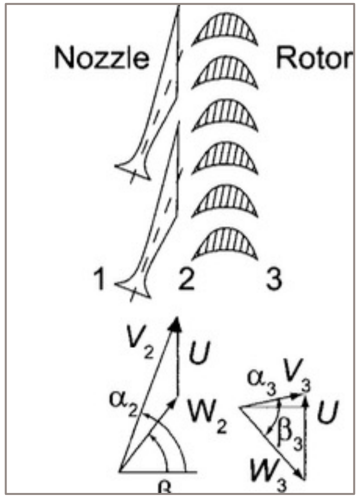
- The direction argument is valid only when $V_{\theta 1} = 0$
- For nonzero $V_{\theta 1} \neq 0$, compare the magnitudes of $U_1 V_{\theta 1}$ and $U_2 V_{\theta 2}$
  - For $U_1 = U_2$, compare only V_θ

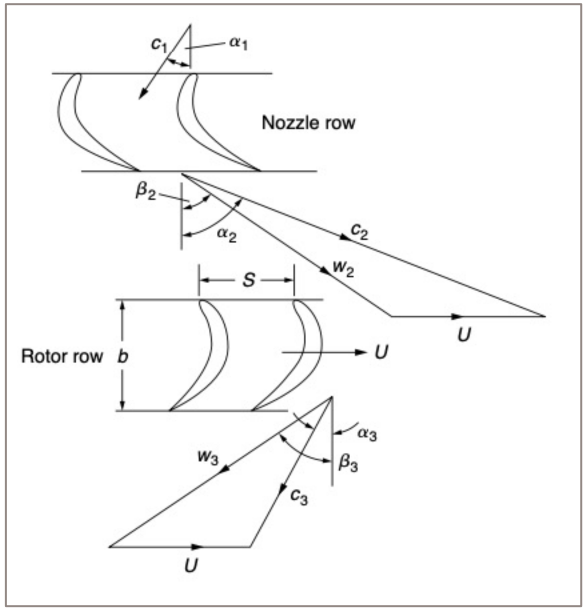
Pump or turbine? – when inlet $V_{\theta 1}$ is non-zero

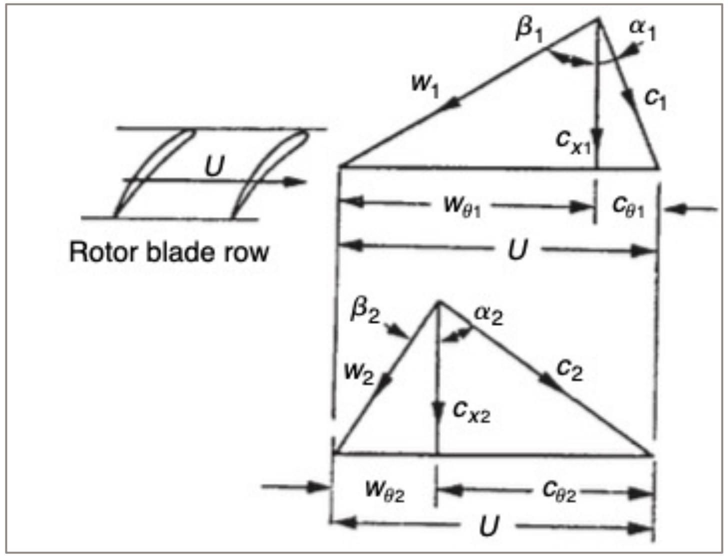


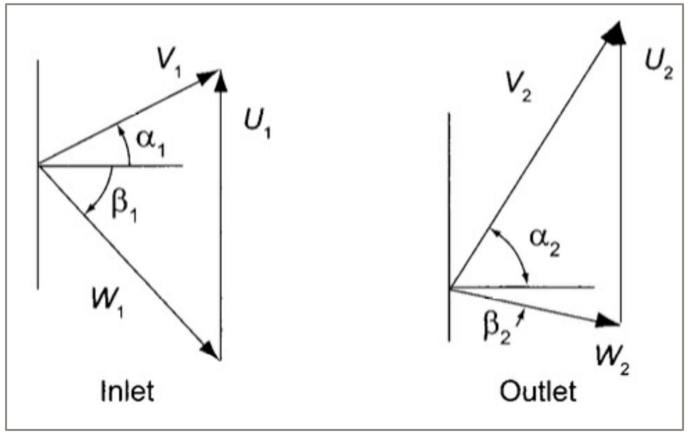












Commonly used notations

Relative	**W**	**w**
Rotation	**U**	**u**
Absolute	**V**	**c**

Hydraulic Turbines

Turbines

Impulse or **reaction** : No matter the working fluid, turbines can be broadly classified into two types based on the mechanism of the fluid interaction

Impulse turbine: the force on the blades is produced solely by turning the fluid, without appreciable pressure drop in the blade passage, with all of the pressure drop occurring in a fixed nozzle.

Reaction turbine: some of the fluid-vane force is from fluid turning and some of the force is a reaction to acceleration of the fluid relative to the vane. In reaction blading, a pressure drop occurs in both a fixed nozzle and the moving vane.

Turbine blading is characterized by the **degree of reaction (R)**, which is the ratio of the drop in static pressure (or enthalpy) across the moving blade to the overall drop in static pressure (or enthalpy) across the fixed nozzle plus the moving blade. Impulse turbines have $R = 0$ while reaction turbines typically have $0.1 < R < 0.7$.

Degree of reaction or Reaction (R)

Change in static enthalpy across the **rotor** divided by the static enthalpy change across the entire **stage**

(static) Enthalpy: internal energy and flow work

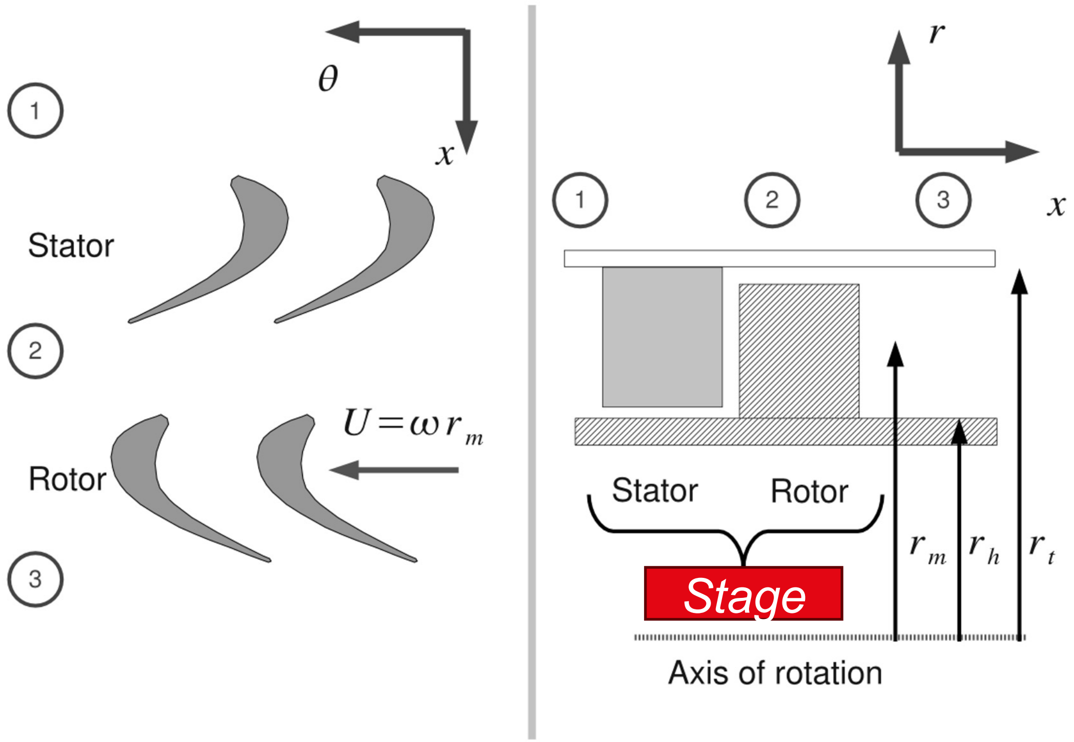
$$\check{h} = \check{u} + \frac{p}{\rho} \quad \text{[J/kg]}$$

$\mathrm{Pa} \cdot \ \mathrm{m}^3/\mathrm{kg} \equiv \ \mathrm{J/kg}$

Stagnation enthalpy: sum of enthalpy, kinetic energy and potential energy

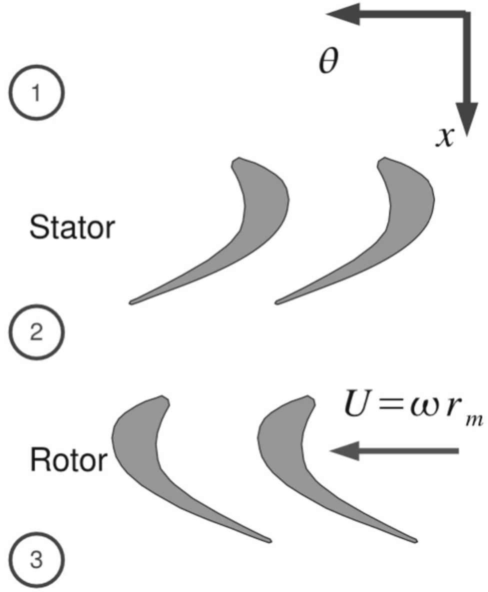
$$\check{h}_0 = \check{h} + \frac{V^2}{2} + gz$$

Achtung! $\check{h}$ is enthalpy, not head h or h_a



Degree of reaction or Reaction (R)

This concept is much used in axial flow machines as a measure of the relative proportions of energy transfer obtained by static and dynamic pressure change.



(static) Enthalpy: internal energy and flow work

$$\check{h} = \check{u} + \frac{p}{\rho} \qquad \text{[J/kg]}$$

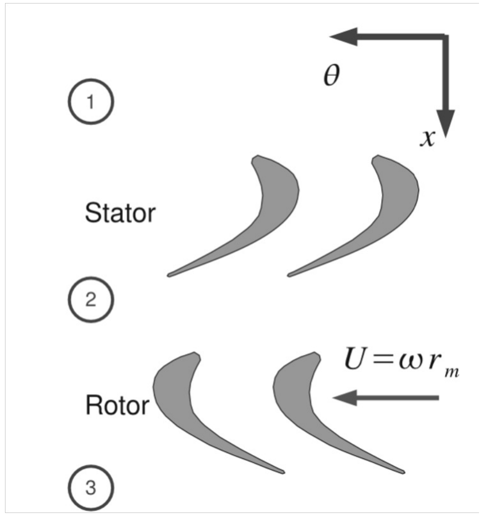
$$R = \frac{\text{energy change due to, or resulting from, static pressure change in the \textbf{rotor}}}{\text{total energy change for a \textbf{stage}}}$$

$$R = \frac{\text{static enthalpy change in \textbf{rotor}}}{\text{\textbf{stage} static enthalpy change}}$$

$$= \frac{\check{h}_2 - \check{h}_3}{\check{h}_1 - \check{h}_3}$$

If no internal energy is changed, incompressible, $\boxed{R \simeq \frac{p_2 - p_3}{p_1 - p_3}}$

Degree of reaction or Reaction (R)



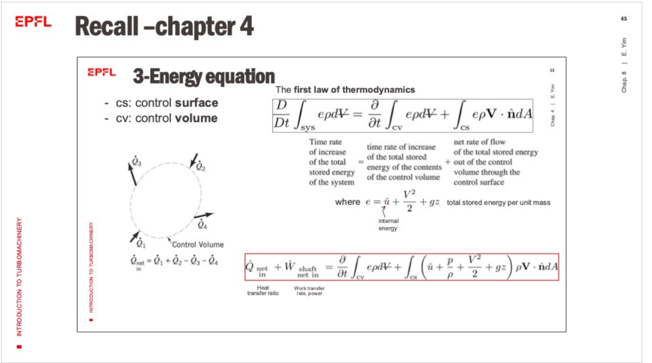
$$\boxed{R \simeq \frac{p_2 - p_3}{p_1 - p_3}}$$

$$w_f = -w_{\text{shaft}}$$

$$R = \frac{\frac{1}{2}\left(V_3^2 - V_2^2\right) - w_{\text{shaft}}}{\frac{1}{2}\left(V_3^2 - V_1^2\right) - w_{\text{shaft}}} = \frac{\frac{1}{2}\left(V_3^2 - V_2^2\right) - \frac{1}{2}\left(V_3^2 - V_2^2 + U_3^2 - U_2^2 - \left(W_3^2 - W_2^2\right)\right)}{\frac{1}{2}\left(V_3^2 - V_1^2\right) - \frac{1}{2}\left(V_3^2 - V_2^2 + U_3^2 - U_2^2 - \left(W_3^2 - W_2^2\right)\right)}$$

$$\boxed{R = \frac{U_2^2 - U_3^2 + W_3^2 - W_2^2}{V_2^2 - V_1^2 + U_2^2 - U_3^2 + W_3^2 - W_2^2}}$$

- If $U_2=U_3$ (axial machine), $W_3=W_2$ → R=0
- If $V_1=V_2$ → R=1



Recall –chapter 4

3-Energy equation

- cs: control **surface**
- cv: control **volume**

The **first law of thermodynamics**

$$\frac{D}{Dt}\int_{\text{sys}} e\rho d\mathcal{V} = \frac{\partial}{\partial t}\int_{\text{cv}} e\rho d\mathcal{V} + \int_{\text{cs}} e\rho \mathbf{V}\cdot\hat{\mathbf{n}}dA$$

Time rate of increase of the total stored energy of the system = time rate of increase of the total stored energy of the contents of the control volume + net rate of flow of the total stored energy out of the control volume through the control surface

where $e = \check{u} + \frac{V^2}{2} + gz$ total stored energy per unit mass

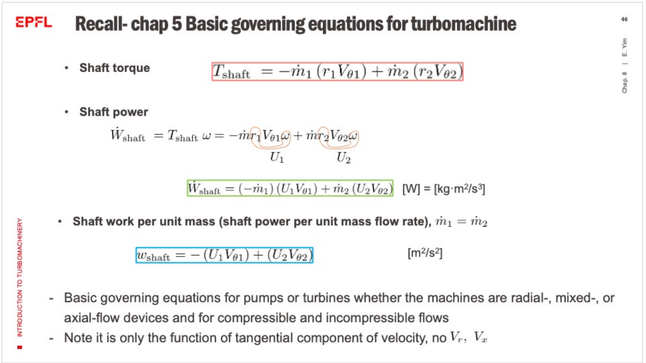
internal energy

Control Volume

$\dot{Q}_{\substack{\text{net}\\ \text{in}}} = \dot{Q}_1 + \dot{Q}_2 - \dot{Q}_3 - \dot{Q}_4$

$$\dot{Q}_{\substack{\text{net}\\ \text{in}}} + \dot{W}_{\substack{\text{shaft}\\ \text{net in}}} = \frac{\partial}{\partial t}\int_{\text{cv}} e\rho d\mathcal{V} + \int_{\text{cs}}\left(\check{u} + \frac{p}{\rho} + \frac{V^2}{2} + gz\right)\rho\mathbf{V}\cdot\hat{\mathbf{n}}dA$$

Heat transfer ratio · Work transfer rate, power



Recall- chap 5 Basic governing equations for turbomachine

- **Shaft torque** $T_{\text{shaft}} = -\dot{m}_1\left(r_1 V_{\theta 1}\right) + \dot{m}_2\left(r_2 V_{\theta 2}\right)$
- **Shaft power**

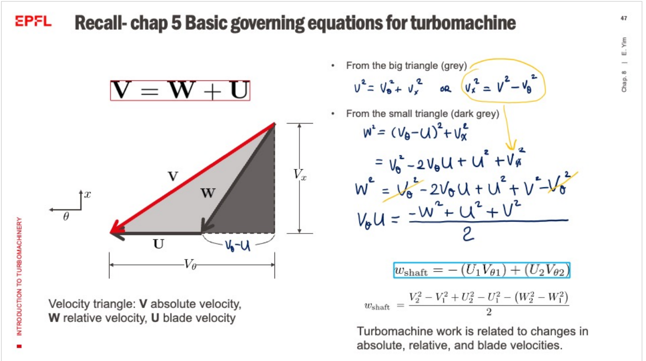
$$\dot{W}_{\text{shaft}} = T_{\text{shaft}}\,\omega = -\dot{m}\underbrace{r_1 V_{\theta 1}\omega}_{U_1} + \dot{m}\underbrace{r_2 V_{\theta 2}\omega}_{U_2}$$

$$\dot{W}_{\text{shaft}} = \left(-\dot{m}_1\right)\left(U_1 V_{\theta 1}\right) + \dot{m}_2\left(U_2 V_{\theta 2}\right) \quad [\text{W}] = [\text{kg}\cdot\text{m}^2/\text{s}^3]$$

- **Shaft work per unit mass (shaft power per unit mass flow rate),** $\dot{m}_1 = \dot{m}_2$

$$w_{\text{shaft}} = -\left(U_1 V_{\theta 1}\right) + \left(U_2 V_{\theta 2}\right) \quad [\text{m}^2/\text{s}^2]$$

- Basic governing equations for pumps or turbines whether the machines are radial-, mixed-, or axial-flow devices and for compressible and incompressible flows
- Note it is only the function of tangential component of velocity, no $V_r,\ V_x$



Recall- chap 5 Basic governing equations for turbomachine

$$\mathbf{V} = \mathbf{W} + \mathbf{U}$$

V W U V_x V_θ $V_\theta - U$ x θ

Velocity triangle: **V** absolute velocity, **W** relative velocity, **U** blade velocity

- From the big triangle (grey)

$V^2 = V_\theta^2 + V_x^2$ or $V_x^2 = V^2 - V_\theta^2$

- From the small triangle (dark grey)

$W^2 = (V_\theta - U)^2 + V_x^2$

$= V_\theta^2 - 2V_\theta U + U^2 + V_x^2$

$W^2 = \cancel{V_\theta^2} - 2V_\theta U + U^2 + V^2 - \cancel{V_\theta^2}$

$V_\theta U = \dfrac{-W^2 + U^2 + V^2}{2}$

$$w_{\text{shaft}} = -\left(U_1 V_{\theta 1}\right) + \left(U_2 V_{\theta 2}\right)$$

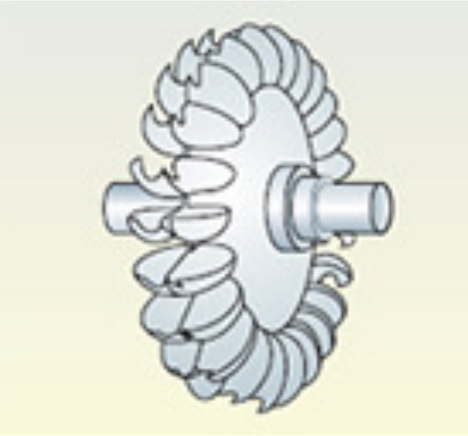
$$w_{\text{shaft}} = \frac{V_2^2 - V_1^2 + U_2^2 - U_1^2 - \left(W_2^2 - W_1^2\right)}{2}$$

Turbomachine work is related to changes in absolute, relative, and blade velocities.

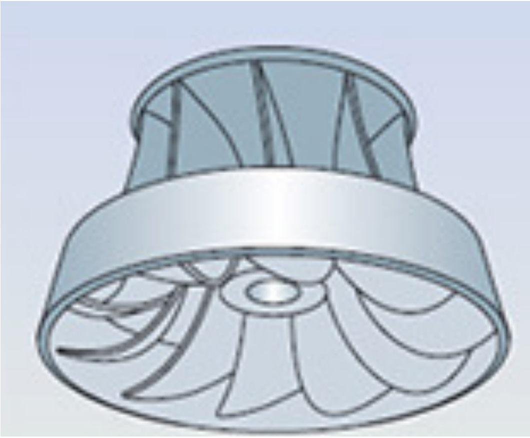
$$w_{\text{shaft}} = \frac{V_3^2 - V_2^2 + U_3^2 - U_2^2 - \left(W_3^2 - W_2^2\right)}{2}$$

Types of hydraulic turbines

- Impulse (Action) turbines: Pelton turbines, R~0
- Reaction turbines: R~[0.1, 0.7]
  Francis turbines (radial et axial), Kaplan turbines (axial)
  Propeller turbines (similar to Kaplan turbines with fixed pitch)

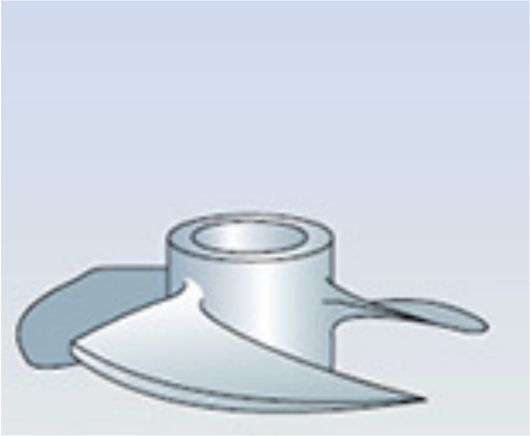


Pelton



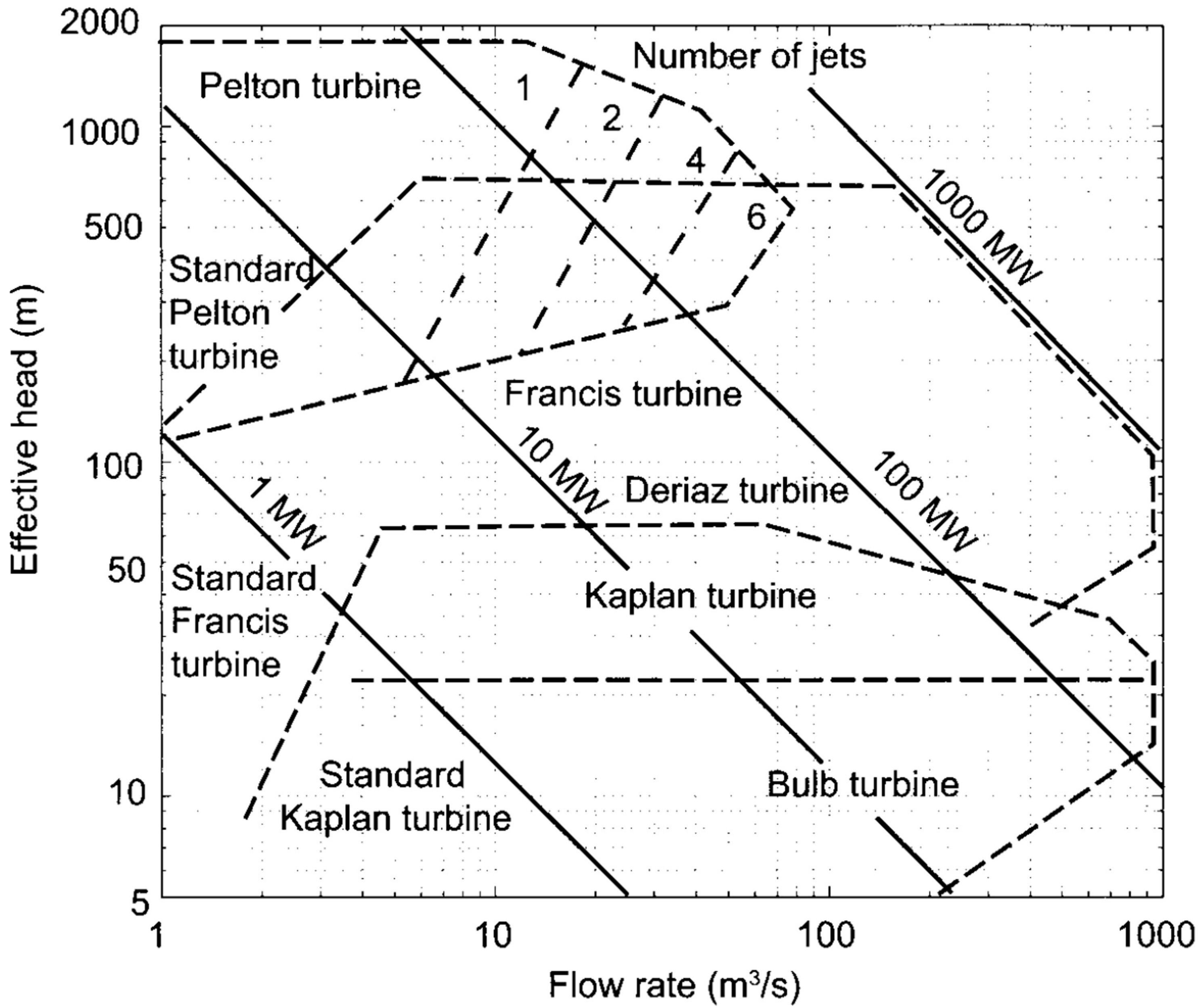
Francis

Kaplan & Bulb



Fixed pitch propeller

Types of hydraulic turbines



Power specific speed

For hydraulic turbines, the rotor diameter D is eliminated between the flow coefficient and the power coefficient to obtain the power-specific speed

Power specific speed (Hydraulic turbines)

$$\boxed{N_s' = \frac{\omega\sqrt{\dot{W}_{\text{shaft}}/\rho}}{(gh_a)^{5/4}}}$$

Specific speed

$$\boxed{N_s = \frac{\omega\sqrt{Q}}{(gh_a)^{3/4}}}$$

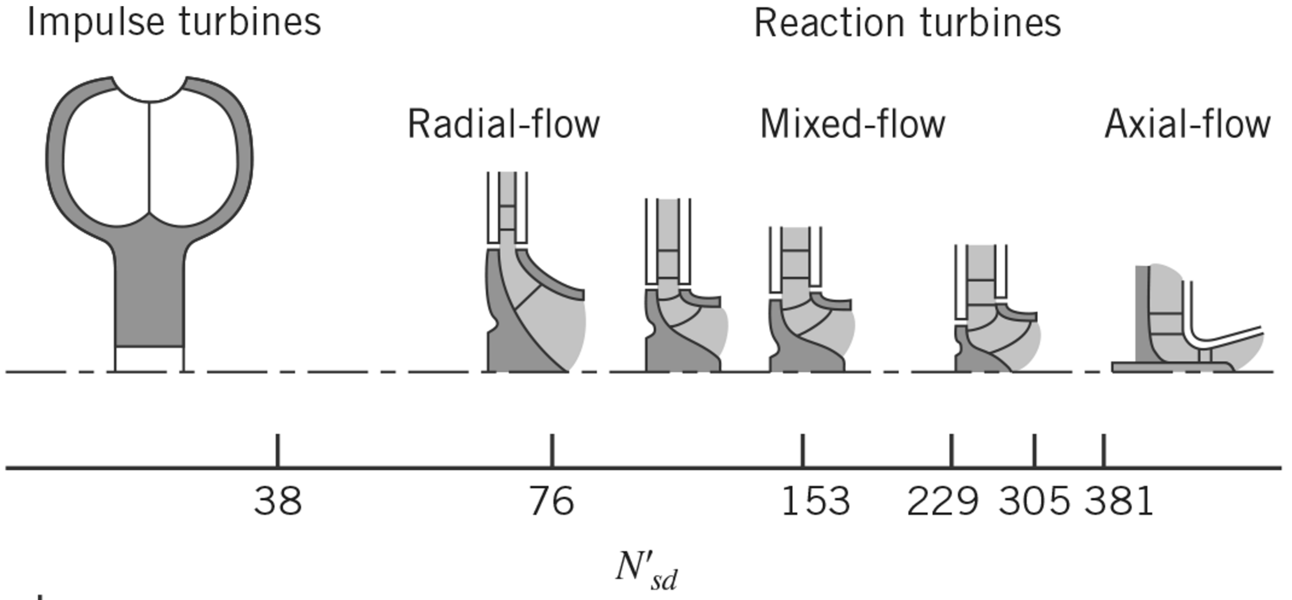
Commonly used, but not dimensionless, definition of power specific speed $N_{sd}' = \frac{\omega(\text{rpm})\sqrt{\dot{W}_{\text{shaft}}(\text{W})}}{[h_a(\text{m})]^{5/4}}$

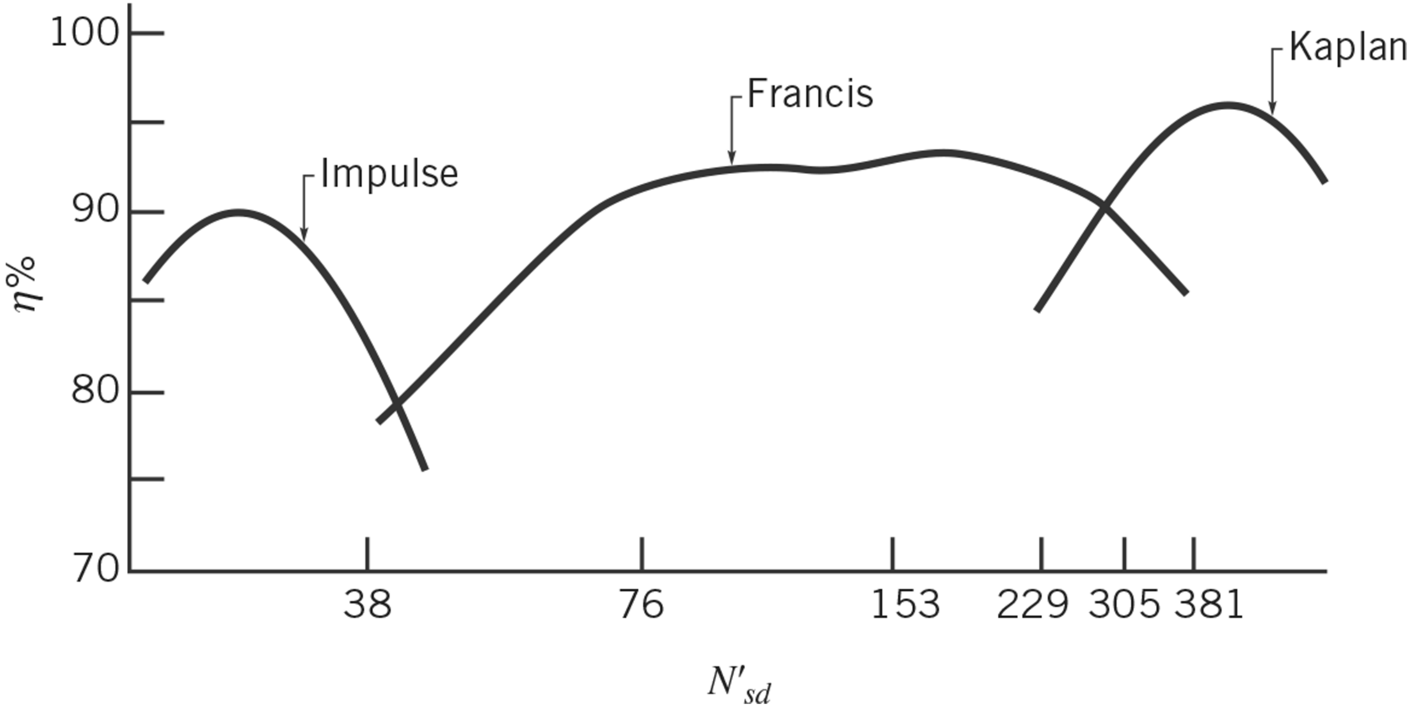
In hydraulic machine, the actual head (in pump) h_a is commonly called '**effective or net head**'.
The elevation head (physical difference between upper reservoir's surface and the one of the lower one) is called '**gross head**', h_g.

$$h_a = h_g - h_L$$

- Power gained by/extracted from the fluids, $P_f = \gamma Q h_a$
- Efficiency pump $\eta = \frac{P_f}{\dot{W}_{\text{shaft}}} \longrightarrow \dot{W}_{\text{shaft}} = P_f/\eta = \gamma Q h_a/\eta$
- Efficiency turbine $\eta = \frac{\dot{W}_{\text{shaft}}}{P_f} \longrightarrow \dot{W}_{\text{shaft}} = \eta P_f = \eta\gamma Q h_a$

Turbines

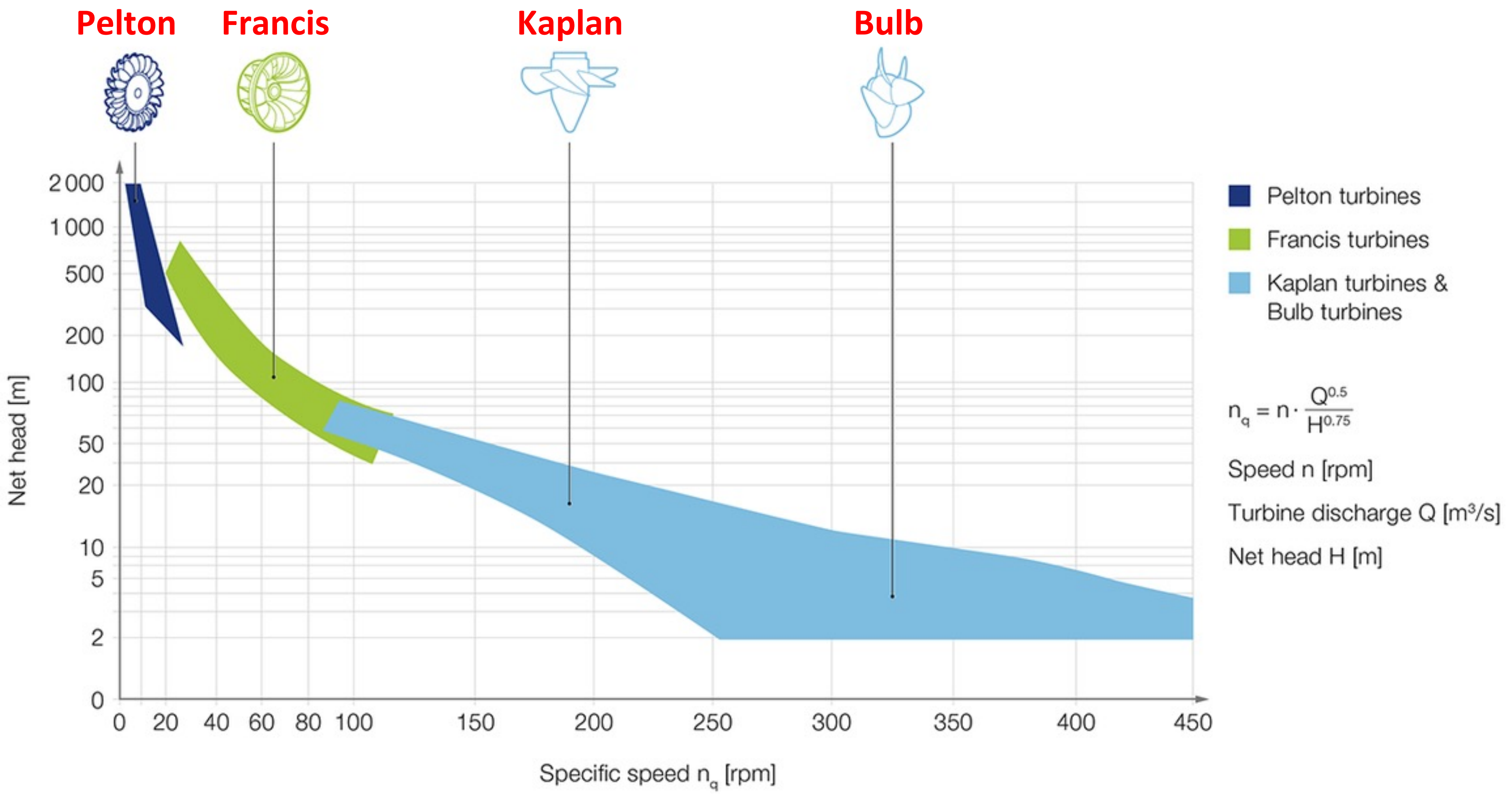




Type		N'_s	η %
Pelton wheel			
	Single jet	0.02 – 0.18	88 – 90
	Twin jet	0.09 – 0.26	89 – 92
	Three jet	0.10 – 0.30	89 – 92
	Four jet	0.12 – 0.36	86
Francis			
	Low-speed	0.39 – 0.65	90 – 92
	Medium-speed	0.65 – 1.2	93
	High-speed	1.2 – 1.9	93 – 96
	Extreme-speed	1.9 – 2.3	89 – 91
Kaplan turbine		1.55 – 5.17	87 – 94
Bulb turbine		3 – 8	

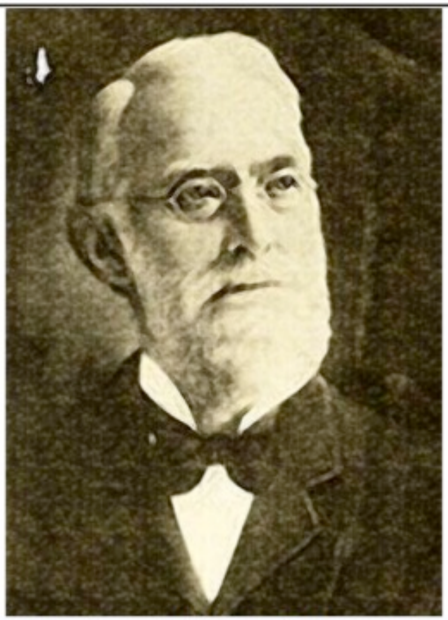
Turbines

- Classification of turbine types as a function of the head and unit specific speed

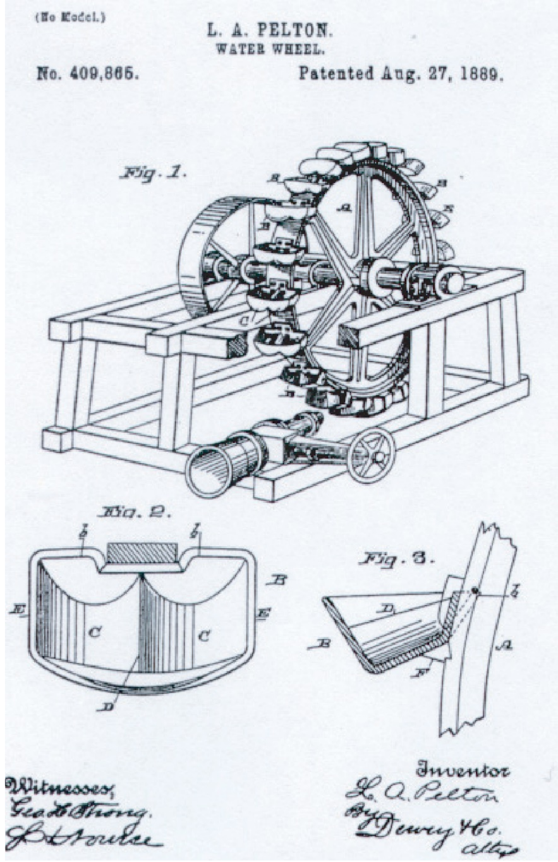


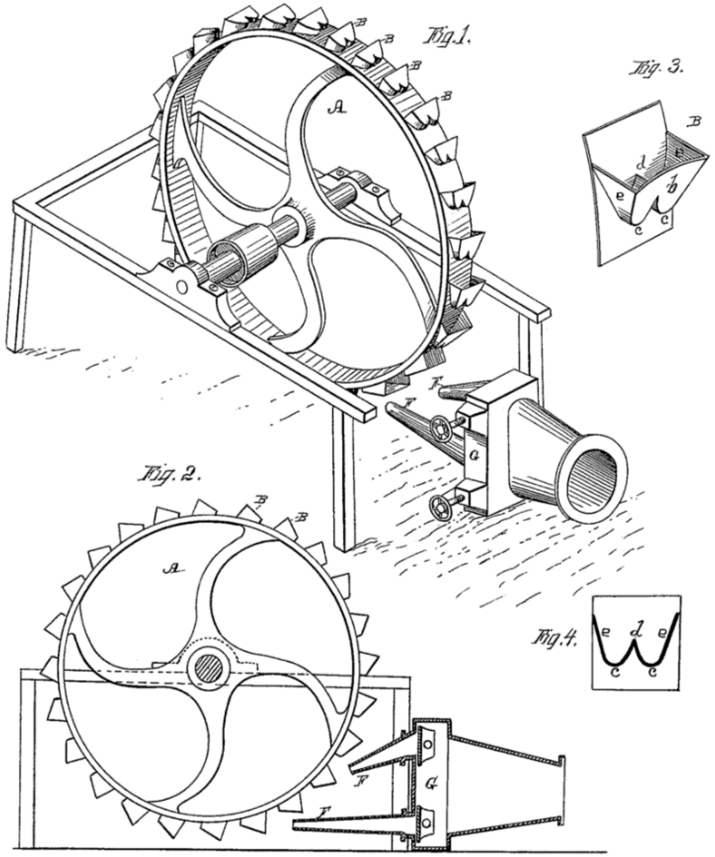
Impulse turbines – Pelton turbine

- **Impulse-type Turbines**
  - Among several types of impulse turbines, the so-called Pelton turbine is the most used
  - Patented by L. A. Pelton in 1889
  - The rotor is made of several buckets and the motion is obtained by high-speed jet(s)



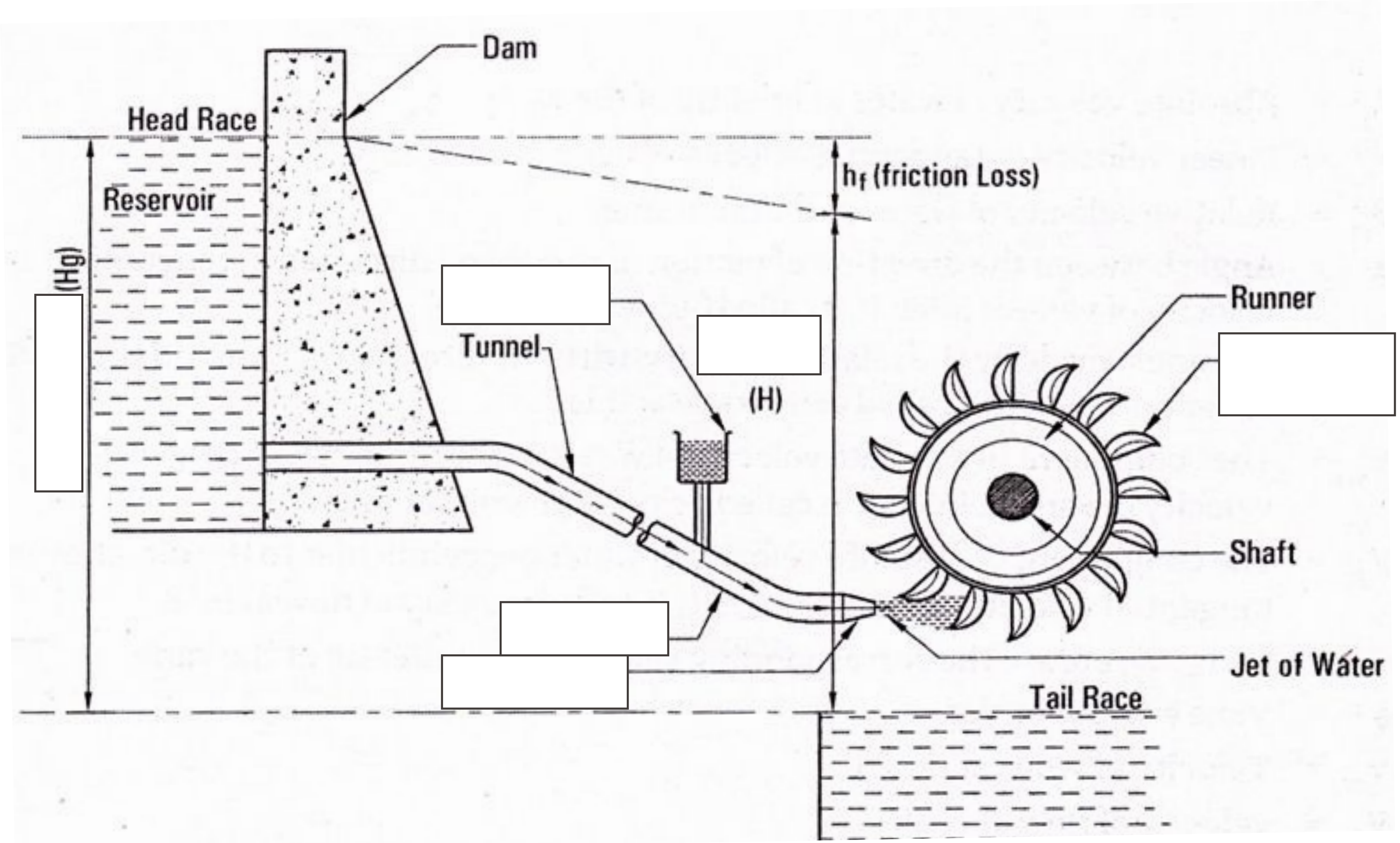
Lester Allan Pelton
(Sept. 5, 1829 – Mar. 14, 1908)
Pelton made his living as a carpenter and a millwright. He created the most efficient form of impulse water turbine.





Pelton turbines

- Pelton wheel is not typical turbomachines (no axial flow, no radial flow), tangential flow
    - Pelton turbines are the most used turbines in **Switzerland** in hydropower generation

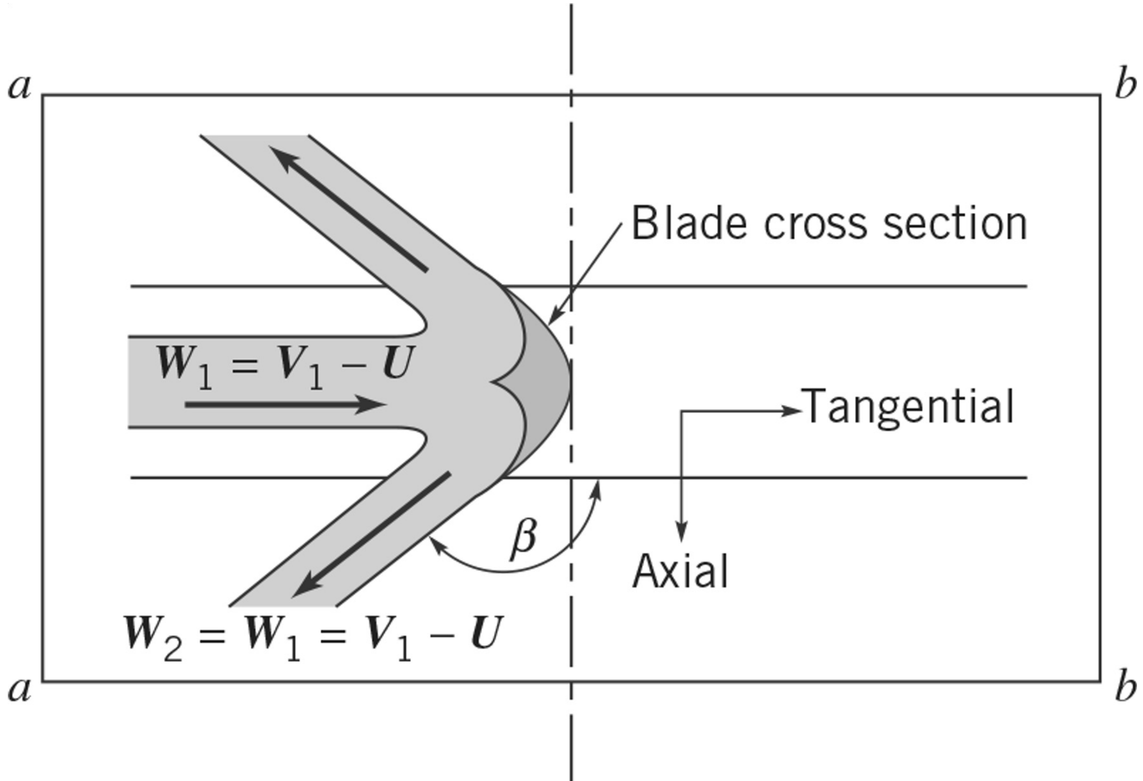


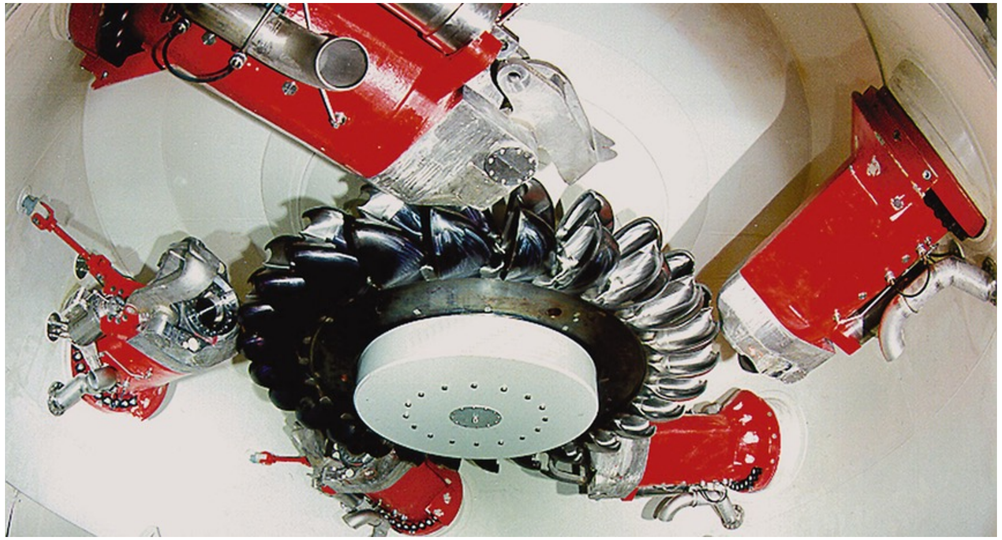
Pelton turbines

- The water enters and leaves the control volume surrounding the wheel as free jets (at **atmospheric** pressure)
- A person riding on the bucket would note that the speed of the water does not change as it slides across the buckets (assuming viscous effects are negligible)

  → the magnitude of the relative velocity does not change, but its direction does.

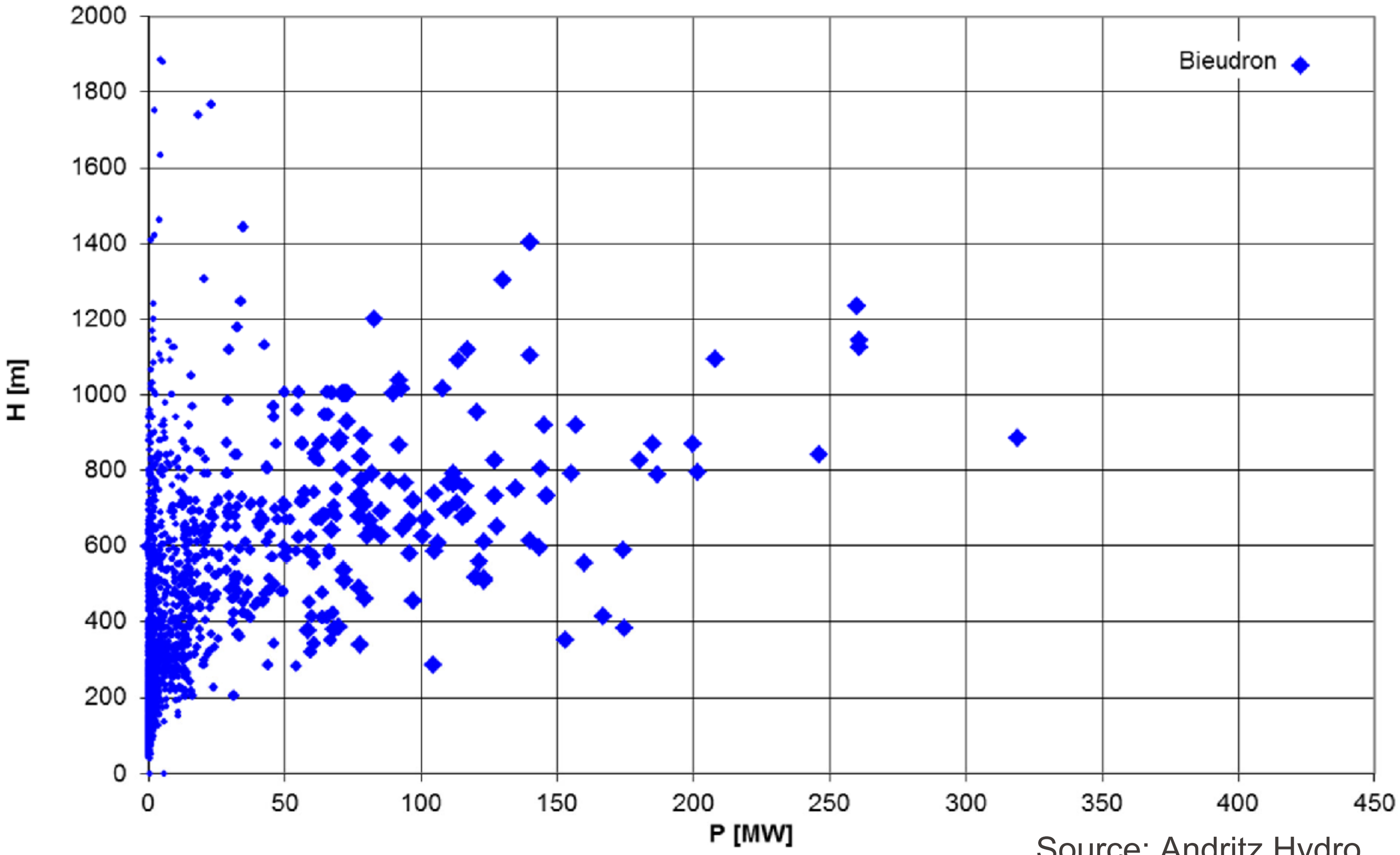
- The change in direction of the velocity of the fluid jet causes a torque on the rotor, resulting in a power output from the turbine





Pelton turbines

- Suitable for high heads (100's meter range):

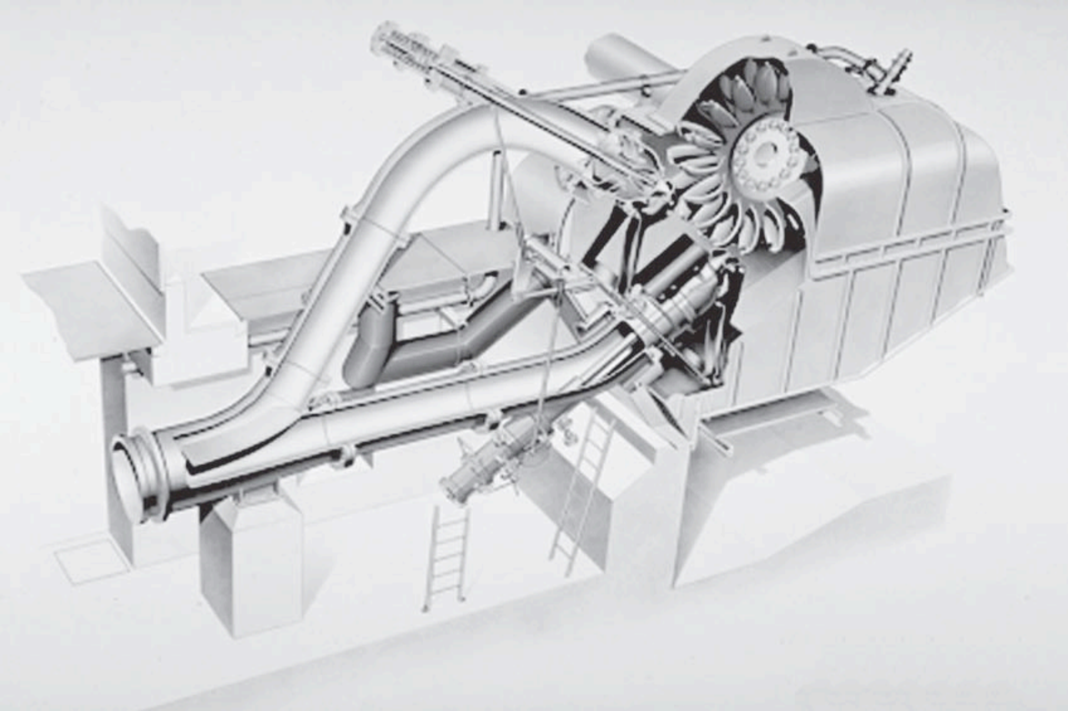


Source: Andritz Hydro

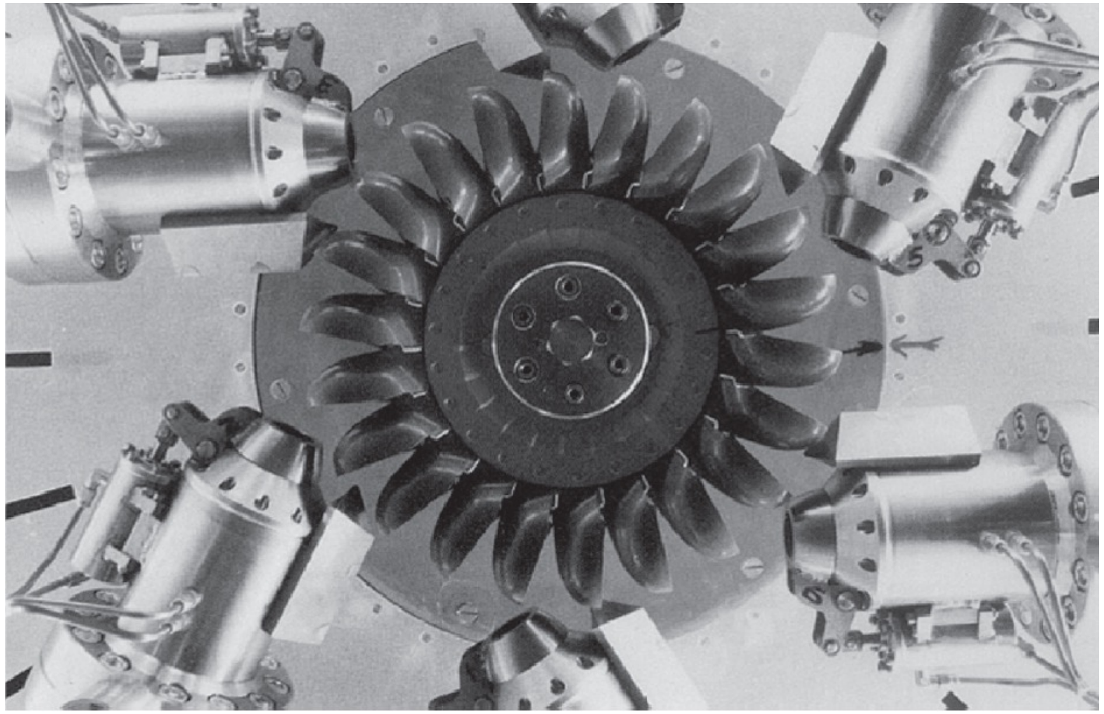
Pelton turbines

- Types of Pelton Turbines:

Horizontal shaft: 1, 2 or 3 Jets

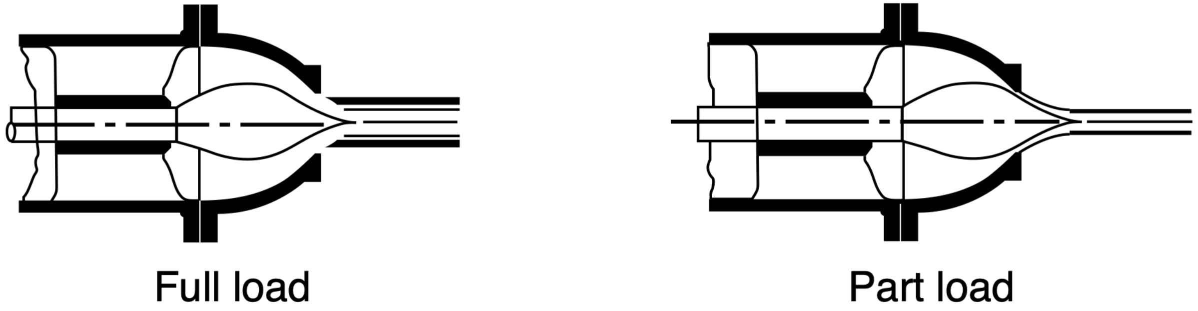


Vertical shaft: 1 to 6 Jets

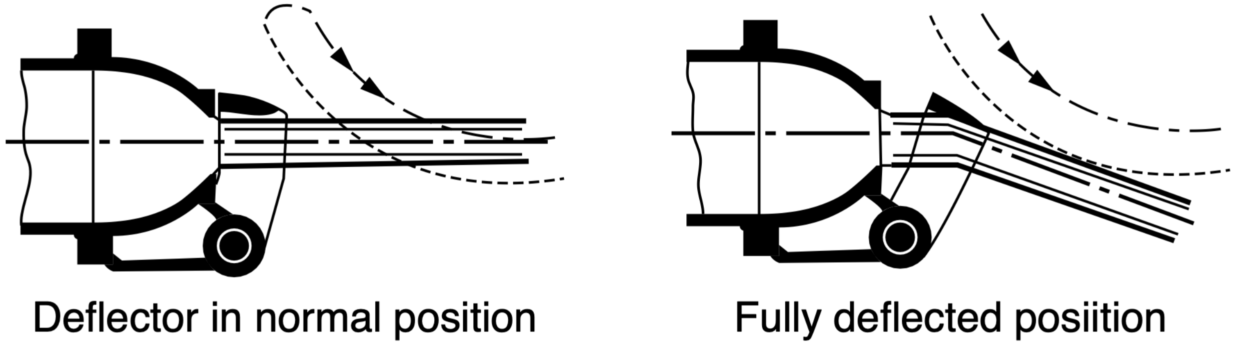


Pelton turbines

- **The nozzle:**
  - Equipped with a needle for flowrate control (e.g. valve)
  - Actioned by a servomotor, which may be internal or external
  - Equipped with a deflector for emergency stops



(a)

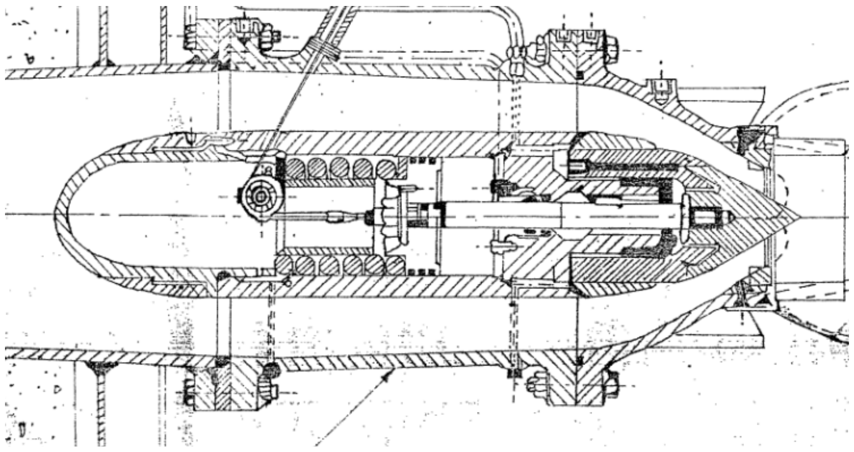


(b)

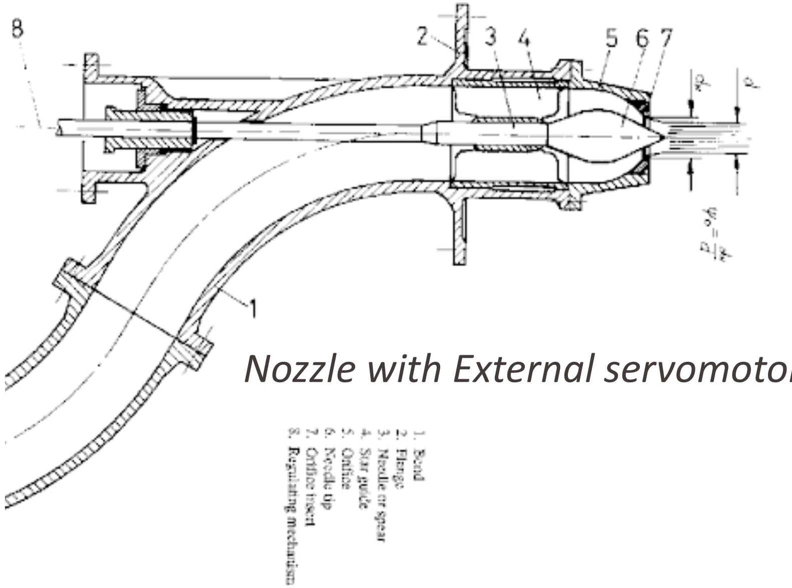
Pelton turbines

- **The nozzle:**
  - Key element of the design: must produce a jet of a “high quality”
    - – Straight jet
    - – Minimum atomization (formation of small droplets)

What happens when the nozzle closes suddenly?



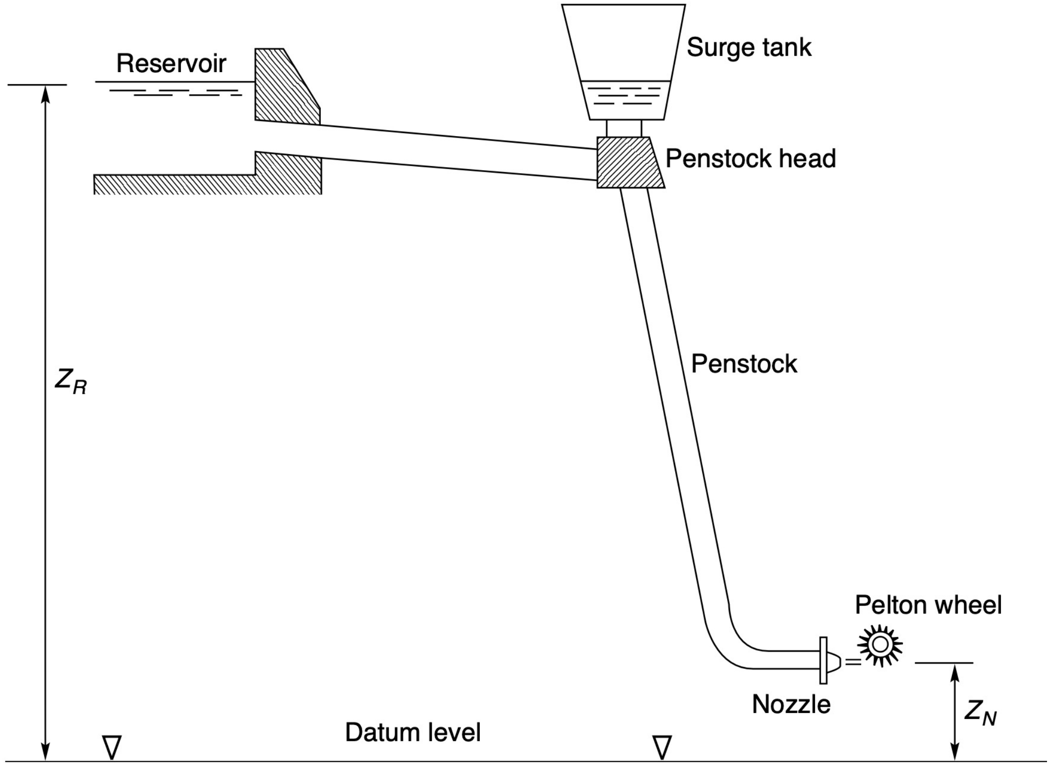
Nozzle with Internal servomotor

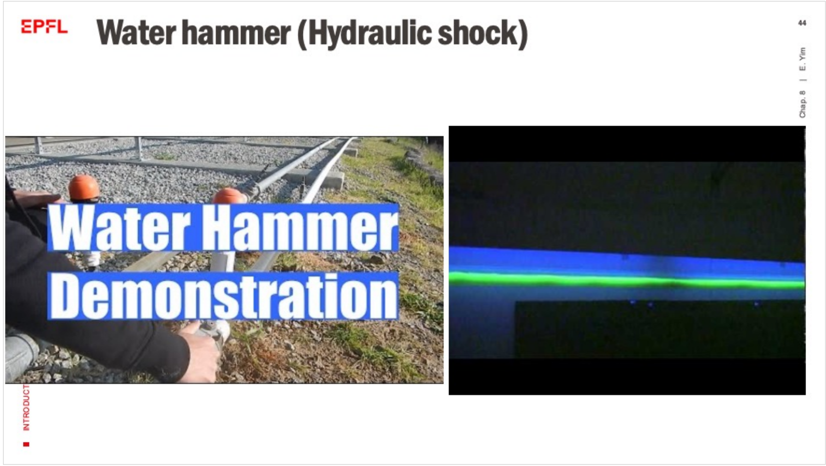


Nozzle with External servomotor

Pressure surges

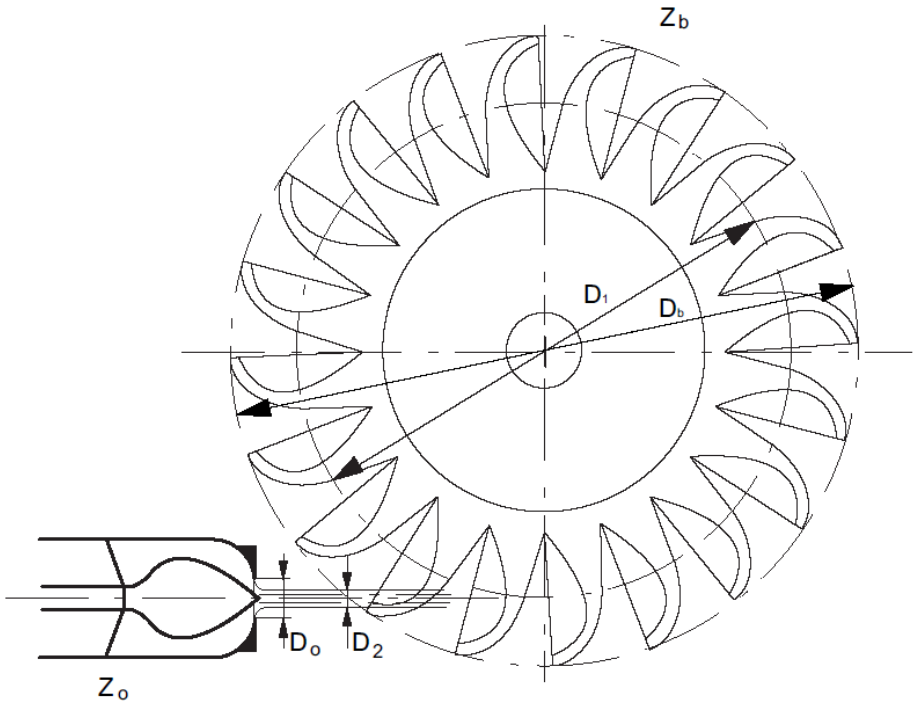
- Spear valve must move slowly: sudden reduction in flow rate may result in serious damage from **pressure surges** (called water hammer)
- If the spear valve closes quickly: all the kinetic energy of the water in the penstock would be absorbed by the elasticity of the supply pipeline (penstock) and the water, creating very large stresses, which would reach their greatest intensity at the turbine inlet where the pipeline is already heavily stressed.
- The surge chamber has the function of absorbing and dissipating some of the pressure and energy fluctuations created by too rapid a closure of the needle valve.

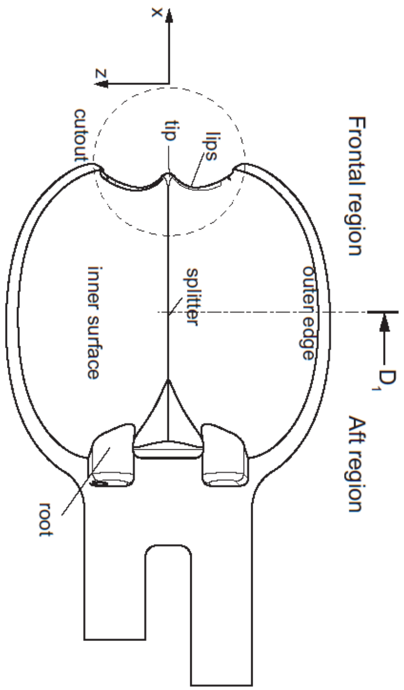


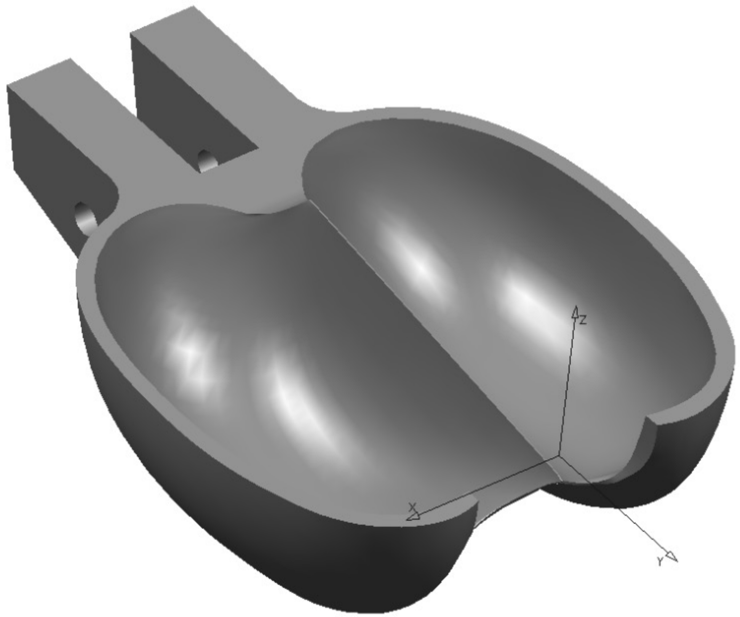


Pelton turbines

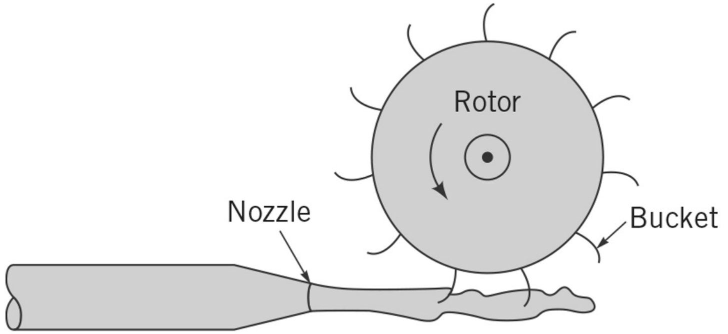
- The buckets:
  - Must deviate “smoothly” the flow and allow for its evacuation with a minimum interference with each other



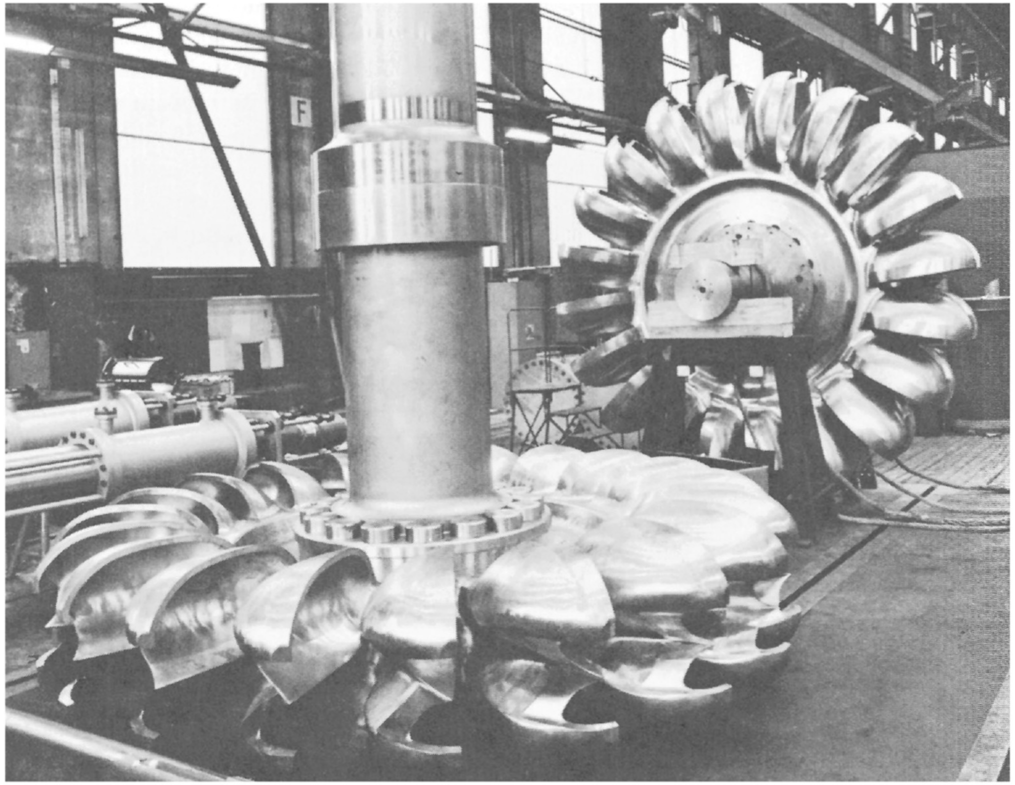




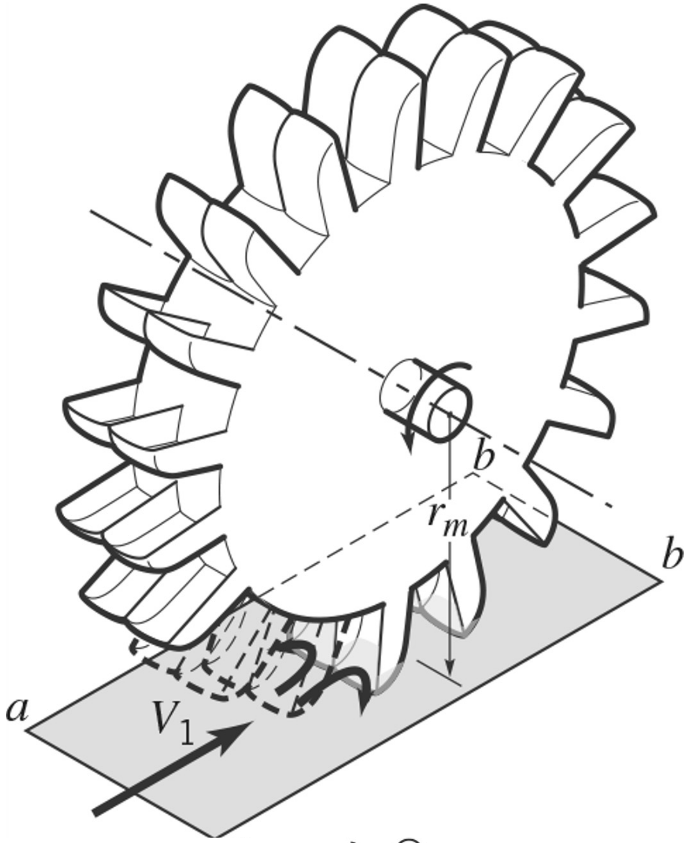
Pelton turbines

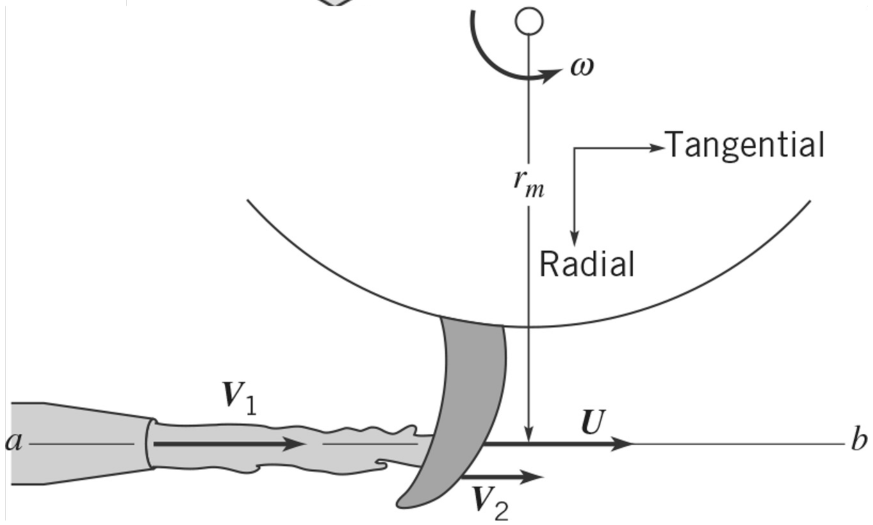


(*a*)

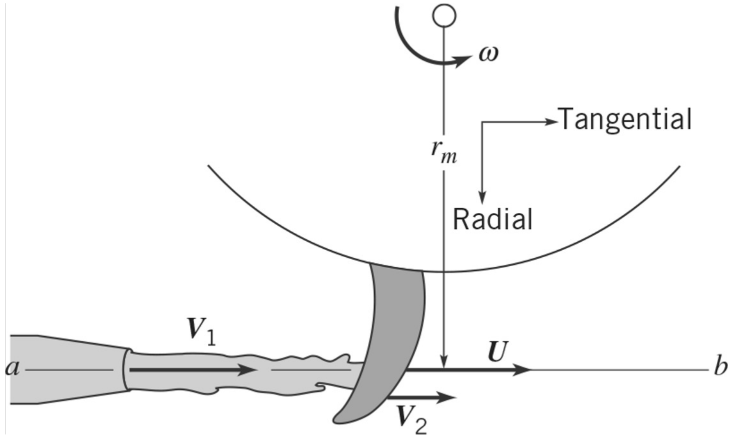

Courtesy of Voith Hydro, York, PA.

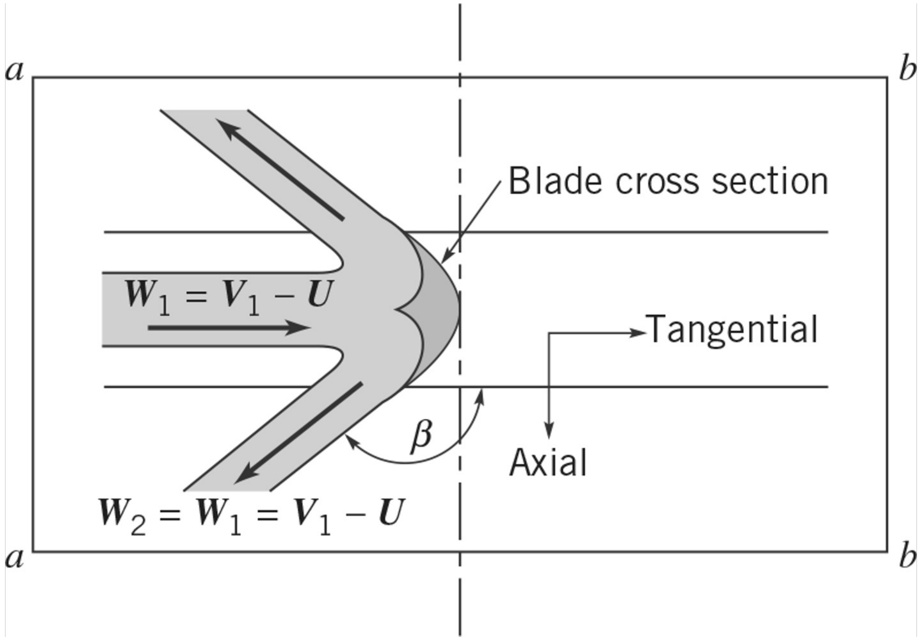
(*b*)





Pelton turbines





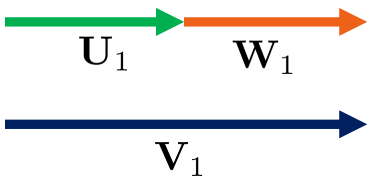
Assumptions

- no radial component of flow
- $W_2 \simeq W_1$ (otherwise $W_2 = C_v W_1$ with $C_v < 1$ velocity coefficient) → exercise 2

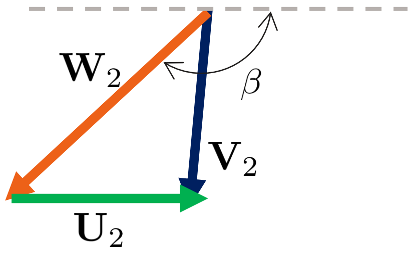
$$\boxed{T_{\text{shaft}} = -\dot{m}_1 \left(r_1 V_{\theta 1}\right) + \dot{m}_2 \left(r_2 V_{\theta 2}\right)}$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m},\ r_1 = r_2 = r_m,\ \mathbf{U}_1 = \mathbf{U}_2 = U$$

$$V_{\theta 1} = V_1 = W_1 + U$$



$$V_{\theta 2} = W_2 \cos\beta + U$$



$$\boxed{\dot{W}_{\text{shaft}} = T_{\text{shaft}}\,\omega = \dot{m} U \left(U - V_1\right)\left(1 - \cos\beta\right)}$$

If $V_1 > U$ (jet impacting bucket), $\dot{W}_{\text{shaft}} < 0$ the turbine extracts power from the fluid

Pelton turbines

$$\boxed{\dot{W}_{\text{shaft}} = \dot{m}U\,(U - V_1)\,(1 - \cos\beta)}$$

- Effect of β : maximum when β = 180° (cos π=-1)

$$\dot{W}_{\text{shaft}} = 2\dot{m}(U^2 - UV_1)$$

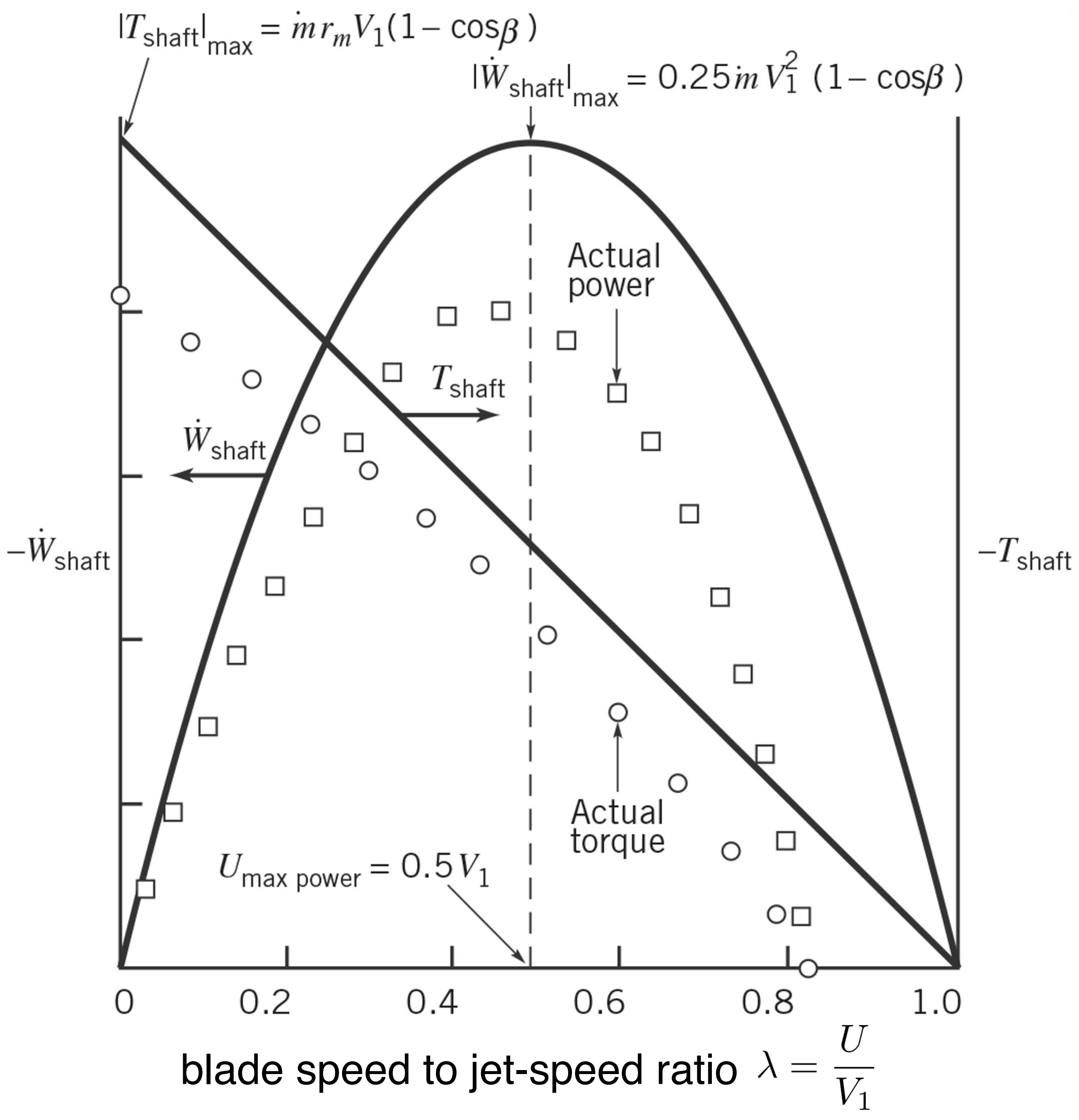
Maximum power
with respect to U

$$\boxed{U_{\text{max power}} = \frac{V_1}{2}}$$

A bucket speed one-half the speed of the fluid coming from the nozzle gives maximum power.

$$\boxed{T_{\text{shaft}} = \dot{m}r_m\,(U - V_1)\,(1 - \cos\beta)}$$

- Shaft torque = 0 when U = V_1

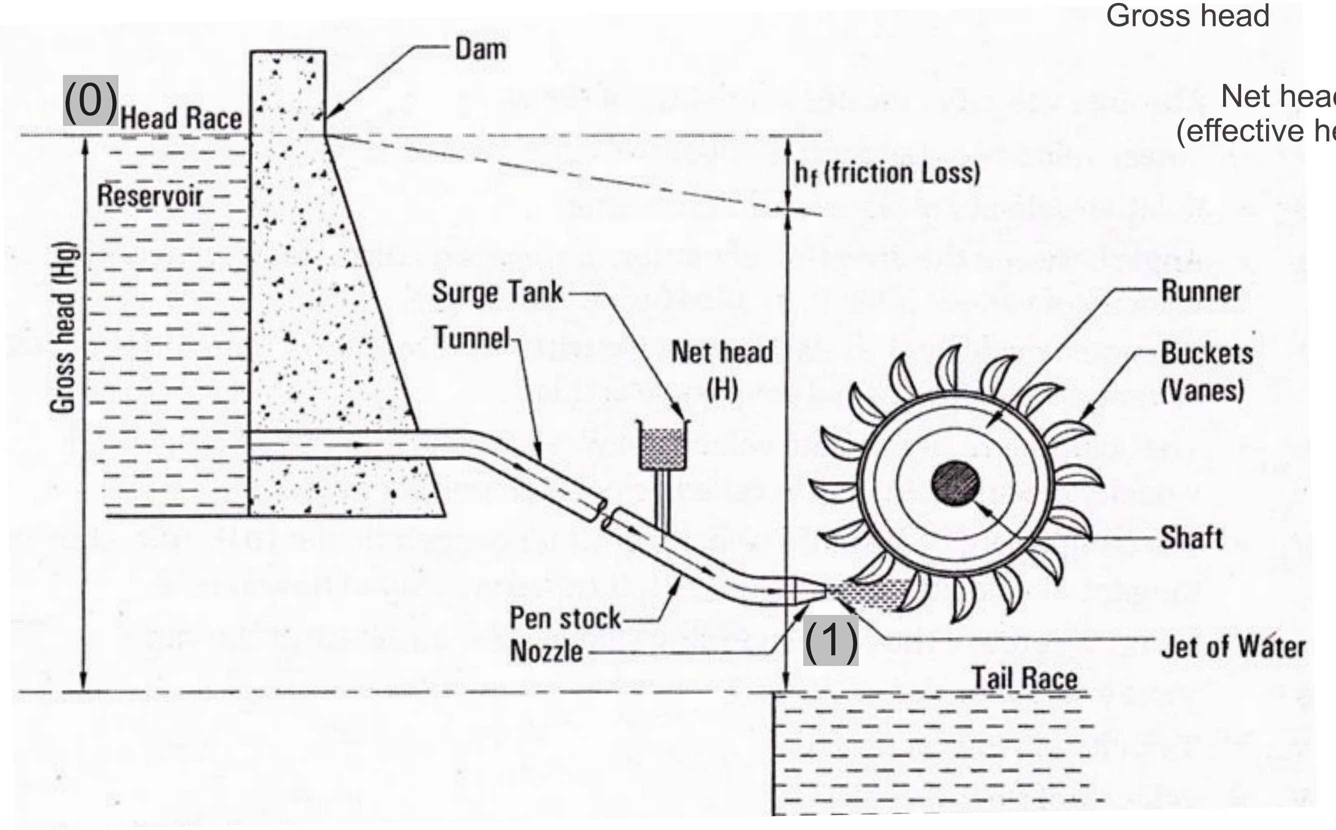


Pelton turbines – Nozzle speed

$$\frac{p_{\text{atm}}}{\gamma} + z_0 = \frac{p_{\text{atm}}}{\gamma} + \frac{V_1^2}{2g} + z_1 + \sum h_L$$

Gross head

Net head h_a
(effective head)



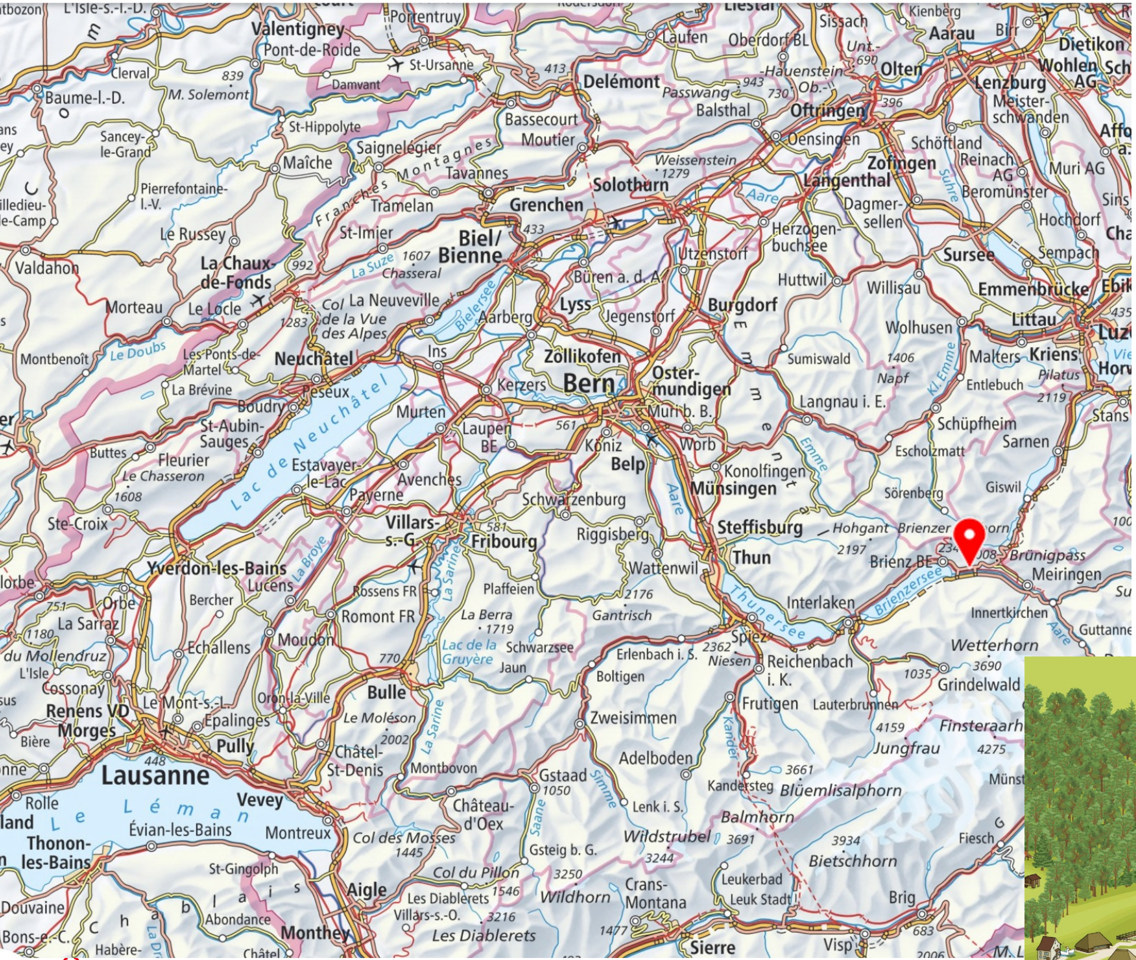
Or $c_N V_1 = V_{\text{nozzle}}$ Nozzle coefficient, c_N <1

Tygun's empirical formula for the number of buckets (Z) in the wheel

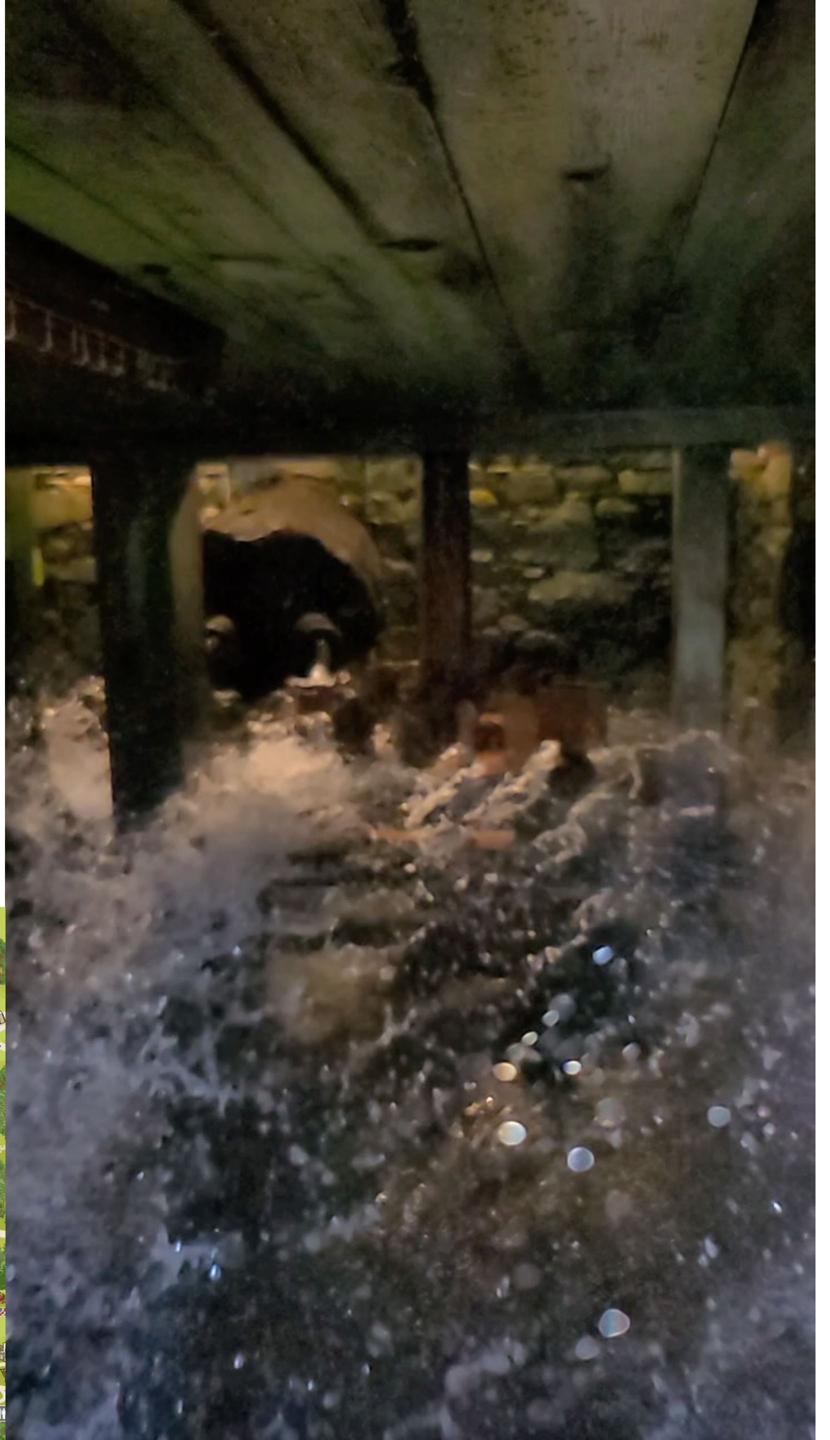
$$Z = \frac{D_{\text{wheel}}}{2D_{\text{nozzle}}} + 15$$

Pelton turbines in Switzerland

SwitzerlandMobility



Ballenberg, Swiss Open-Air Museum

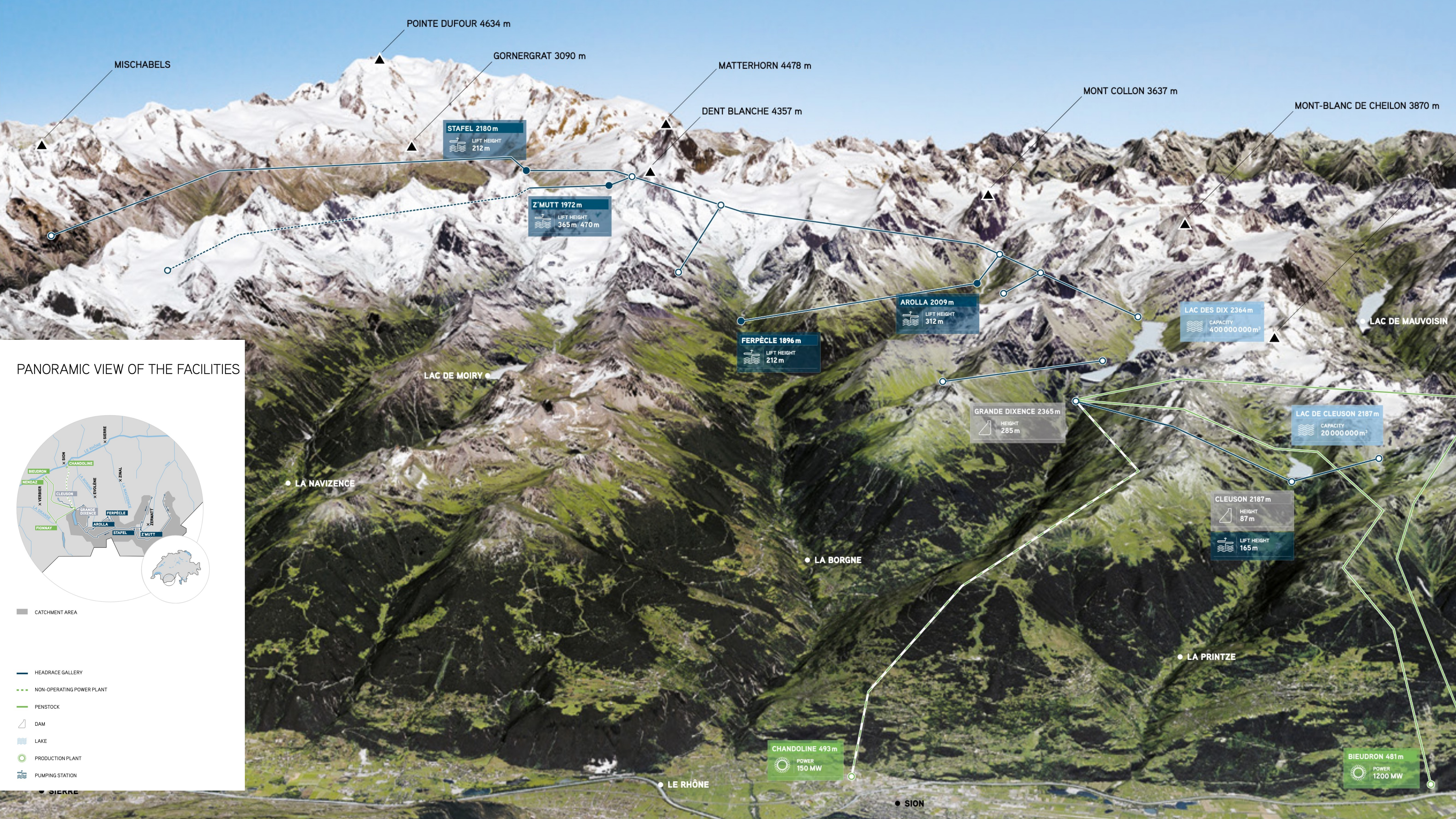


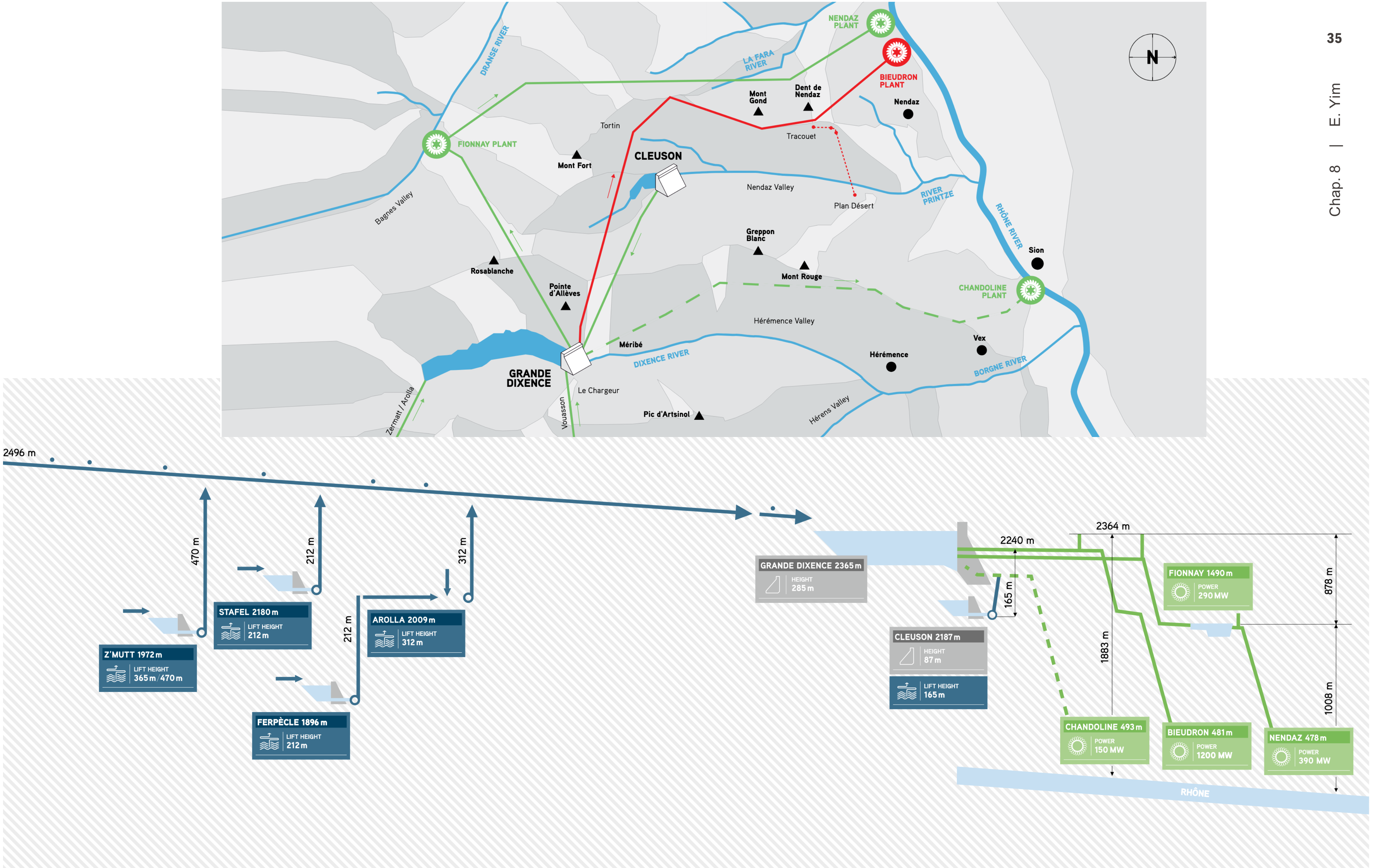
INTRODUC

Pelton turbines in Switzerland

SwitzerlandMobility

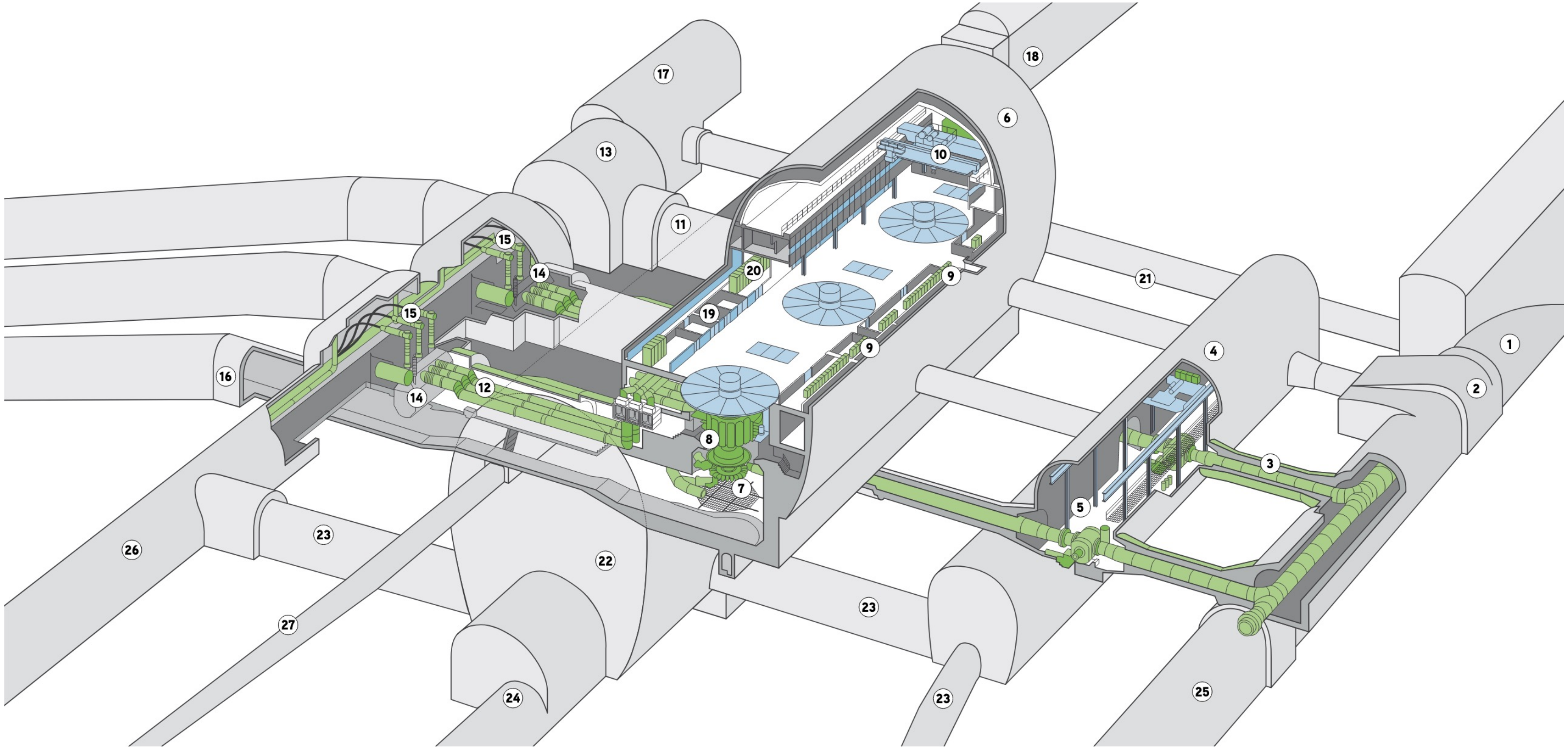
PANORAMIC VIEW OF THE FACILITIES




NENDAZ PLANT
BIEUDRON PLANT
FIONNAY PLANT
CHANDOLINE PLANT
DRANSE RIVER
LA FARA RIVER
RIVER PRINTZE
RHÔNE RIVER
DIXENCE RIVER
BORGNE RIVER
N
Mont Gond
Dent de Nendaz
Nendaz
Tortin
Tracouet
Mont Fort
CLEUSON
Nendaz Valley
Plan Désert
Bagnes Valley
Greppon Blanc
Mont Rouge
Sion
Rosablanche
Pointe d'Allèves
Hérémence Valley
Méribé
Hérémence
Vex
GRANDE DIXENCE
Le Chargeur
Pic d'Artsinol
Hérens Valley
Zermatt / Arolla
Vouasson
2496 m
470 m
212 m
312 m
Z'MUTT 1972 m
LIFT HEIGHT 365 m / 470 m
STAFEL 2180 m
LIFT HEIGHT 212 m
FERPÈCLE 1896 m
LIFT HEIGHT 212 m
AROLLA 2009 m
LIFT HEIGHT 312 m
GRANDE DIXENCE 2365 m
HEIGHT 285 m
2364 m
2240 m
165 m
CLEUSON 2187 m
HEIGHT 87 m
LIFT HEIGHT 165 m
1883 m
878 m
1008 m
FIONNAY 1490 m
POWER 290 MW
CHANDOLINE 493 m
POWER 150 MW
BIEUDRON 481 m
POWER 1200 MW
NENDAZ 478 m
POWER 390 MW
RHÔNE

BIEUDRON POWER PLANT
OVERVIEW

1 PENSTOCK
2 DISTRIBUTOR
3 UNIT BRANCH (25 m³/s, PRESSURE 190 BAR)
4 VALVES CHAMBER
5 SPHERICAL VALVE (210 TONS PER UNIT)
6 MAIN CAVERN
7 423 MW PELTON TURBINE
8 465 MVA GENERATOR
9 CONTROL ROOM
10 250 TONS OVERHEAD CRANE
11 BUSBAR TUNNELS
12 21 kV 15'000 A BUSBARS
13 TRANSFORMER CELL
14 465 MVA THREE-PHASE TRANSFORMERS
15 230/410 kV ONE-PHASE CABLE OUTLET
16 TAILWATER BRANCH
17 MAINTENANCE AND STORAGE BUILDING
18 COOLING WATER RESERVOIR
19 16/0.4 kV AUXILIARY TRANSFORMERS
20 230/400 V SWITCHBOARDS
21 EMERGENCY TUNNEL
22 ASSEMBLY SITE
23 CONNECTING TUNNELS
24 ACCESS TUNNEL TO PLANT
25 ACCESS TUNNEL TO DISTRIBUTOR
26 TUNNEL FOR 410 kV CABLES AND ACCESS TO TRANSFORMERS
27 VENTILATION TUNNEL



Grande
Dixence

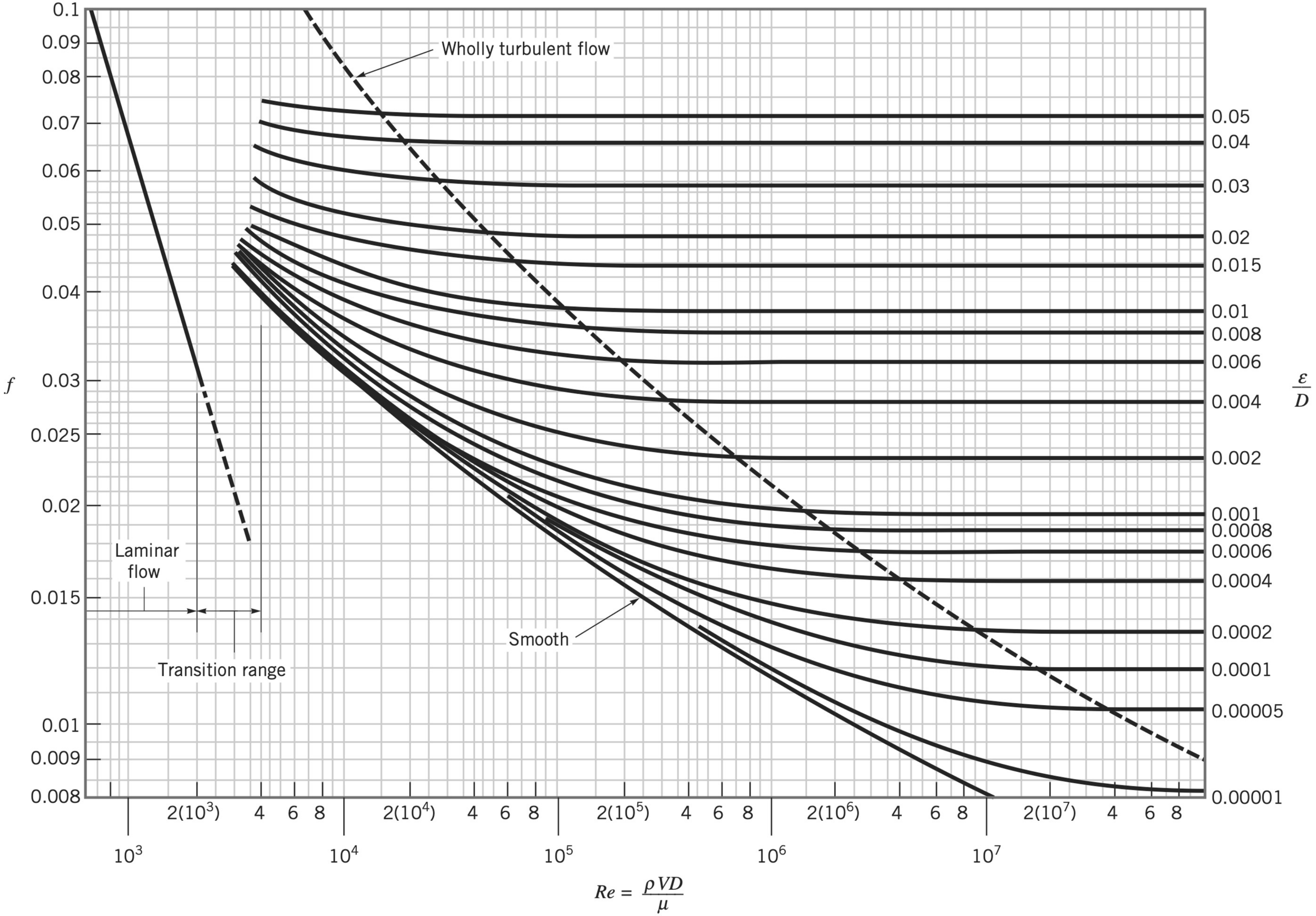
Exercise

A Pelton wheel operates with a gross head of 530 m and a flow rate of 9 m^3/s. The penstock length is 880 m, its diameter is 1.2 m, and its RMS roughness is 0.12 mm. The minor loss can be ignored. The hydraulic efficiency is η_h = 0.84, and the shaft speed is 650 rpm. Water kinematic viscosity is 1.02×10^{-6} m^2/s and density is 998 kg/m^3.

Find (a) the effective head and the power delivered by the turbine and (b) the specific speed and from it the recommended number of jets and the number of buckets in the wheel. The nozzle coefficient is C_N = 0.97, and the ratio of the blade speed to the discharge velocity is $\lambda = U/V_1$ = 0.45.

EPFL

Moody diagram – Chapter 3



Exercise

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Type		N_s'	η %
Pelton wheel			
	Single jet	0.02 – 0.18	88 – 90
	Twin jet	0.09 – 0.26	89 – 92
	Three jet	0.10 – 0.30	89 – 92
	Four jet	0.12 – 0.36	86
Francis			
	Low-speed	0.39 – 0.65	90 – 92
	Medium-speed	0.65 – 1.2	93
	High-speed	1.2 – 1.9	93 – 96
	Extreme-speed	1.9 – 2.3	89 – 91
Kaplan turbine		1.55 – 5.17	87 – 94
Bulb turbine		3 – 8	

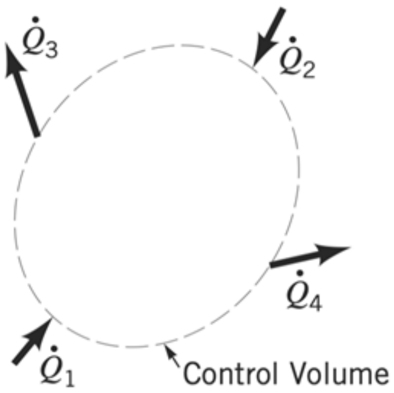
Appendix

Water hammer (Hydraulic shock)

Recall –chapter 4

3-Energy equation

- cs: control **surface**
- cv: control **volume**



$$\dot{Q}_{\substack{\text{net}\\\text{in}}} = \dot{Q}_1 + \dot{Q}_2 - \dot{Q}_3 - \dot{Q}_4$$

The **first law of thermodynamics**

$$\boxed{\frac{D}{Dt}\int_{\text{sys}} e\rho d\forall = \frac{\partial}{\partial t}\int_{\text{cv}} e\rho d\forall + \int_{\text{cs}} e\rho \mathbf{V}\cdot\hat{\mathbf{n}}dA}$$

Time rate of increase of the total stored energy of the system = time rate of increase of the total stored energy of the contents of the control volume + net rate of flow of the total stored energy out of the control volume through the control surface

where $e = \check{u} + \frac{V^2}{2} + gz$ total stored energy per unit mass

internal energy

$$\boxed{\dot{Q}_{\substack{\text{net}\\\text{in}}} + \dot{W}_{\substack{\text{shaft}\\\text{net in}}} = \frac{\partial}{\partial t}\int_{\text{cv}} e\rho d\forall + \int_{\text{cs}}\left(\check{u} + \frac{p}{\rho} + \frac{V^2}{2} + gz\right)\rho\mathbf{V}\cdot\hat{\mathbf{n}}dA}$$

Heat transfer ratio

Work transfer rate, power

INTRODUCTION TO TURBOMACHINERY

Recall- chap 5 Basic governing equations for turbomachine

- **Shaft torque**

$$\boxed{T_{\text{shaft}} = -\dot{m}_1 (r_1 V_{\theta 1}) + \dot{m}_2 (r_2 V_{\theta 2})}$$

- **Shaft power**

$$\dot{W}_{\text{shaft}} = T_{\text{shaft}}\, \omega = -\dot{m} \underbrace{r_1 V_{\theta 1} \omega}_{U_1} + \dot{m} \underbrace{r_2 V_{\theta 2} \omega}_{U_2}$$

$$\boxed{\dot{W}_{\text{shaft}} = (-\dot{m}_1)(U_1 V_{\theta 1}) + \dot{m}_2 (U_2 V_{\theta 2})}$$ [W] = [kg·m^2/s^3]

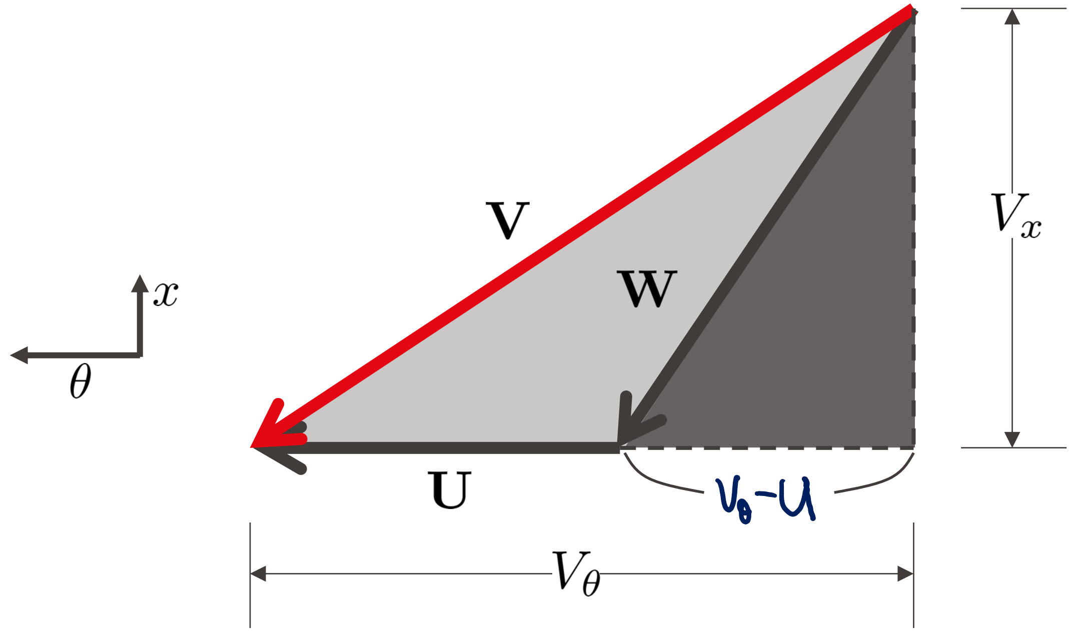
- **Shaft work per unit mass (shaft power per unit mass flow rate),** $\dot{m}_1 = \dot{m}_2$

$$\boxed{w_{\text{shaft}} = -(U_1 V_{\theta 1}) + (U_2 V_{\theta 2})}$$ [m^2/s^2]

- Basic governing equations for pumps or turbines whether the machines are radial-, mixed-, or axial-flow devices and for compressible and incompressible flows
- Note it is only the function of tangential component of velocity, no $V_r,\ V_x$

Recall- chap 5 Basic governing equations for turbomachine

$$\boxed{\mathbf{V} = \mathbf{W} + \mathbf{U}}$$



Velocity triangle: **V** absolute velocity, **W** relative velocity, **U** blade velocity

- From the big triangle (grey)

$$V^2 = V_\theta^2 + V_x^2 \quad \text{OR} \quad V_x^2 = V^2 - V_\theta^2$$

- From the small triangle (dark grey)

$$W^2 = (V_\theta - U)^2 + V_x^2$$

$$= V_\theta^2 - 2V_\theta U + U^2 + V_x^2$$

$$W^2 = \cancel{V_\theta^2} - 2V_\theta U + U^2 + V^2 - \cancel{V_\theta^2}$$

$$V_\theta U = \frac{-W^2 + U^2 + V^2}{2}$$

$$\boxed{w_{\text{shaft}} = -(U_1 V_{\theta 1}) + (U_2 V_{\theta 2})}$$

$$w_{\text{shaft}} = \frac{V_2^2 - V_1^2 + U_2^2 - U_1^2 - \left(W_2^2 - W_1^2\right)}{2}$$

Turbomachine work is related to changes in absolute, relative, and blade velocities.