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T. de Colombel | November 2023

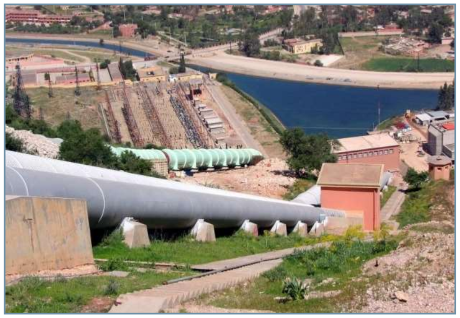
Hydraulic Design for Francis Turbines

GE VERNOVA – Hydro Main Products

Turbines

Generators

Control System & BOP



Hydro Mechanical

+100 years of Hydraulic Turbines in France and all over the world

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Hydraulic Laboratory in Grenoble, France

1917

1948 NEYRPIC
Avenue de Beauvert, GRENOBLE - Téléphone 44.55.30

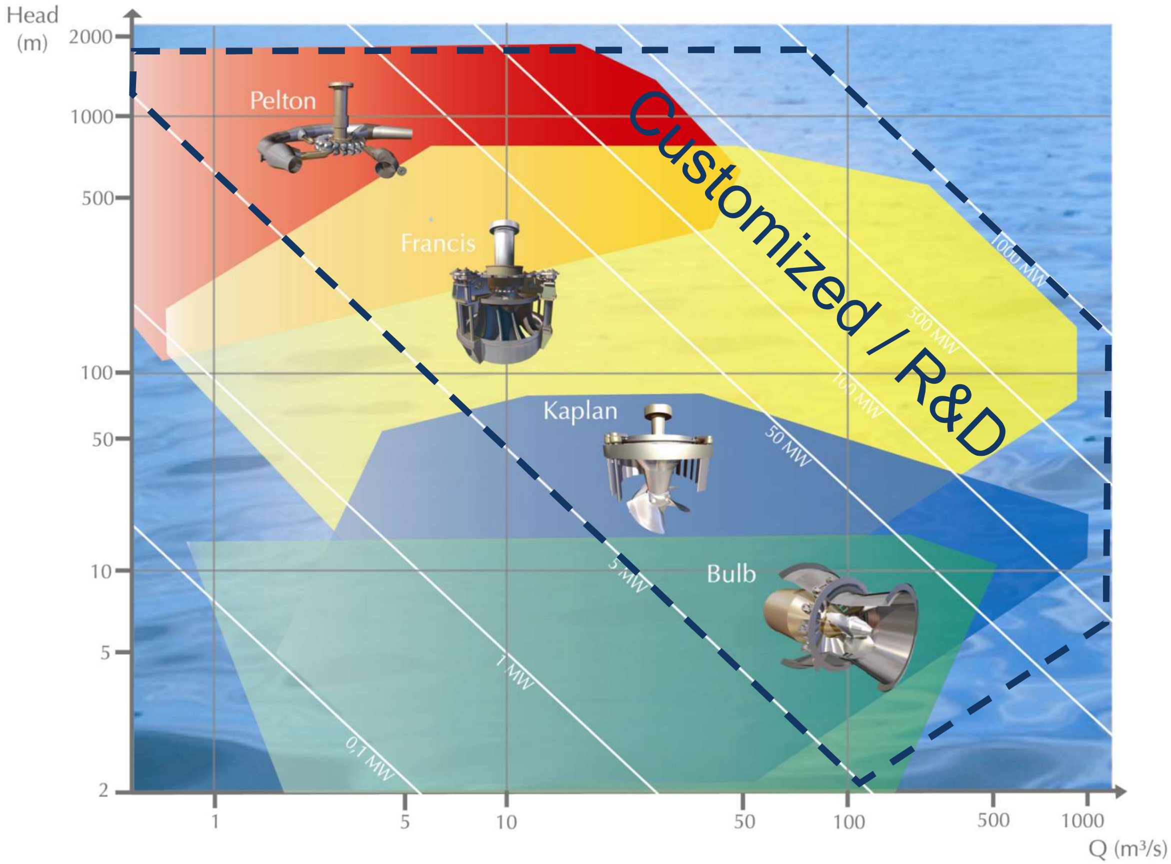
1990 GEC ALSTHOM

2000 ALSTOM

2015

2022 GE VERNOVA

Main Types of Hydraulic Turbines

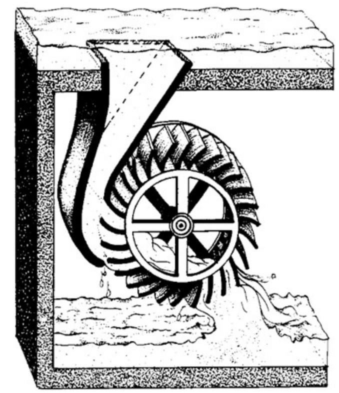


Less common types of turbines

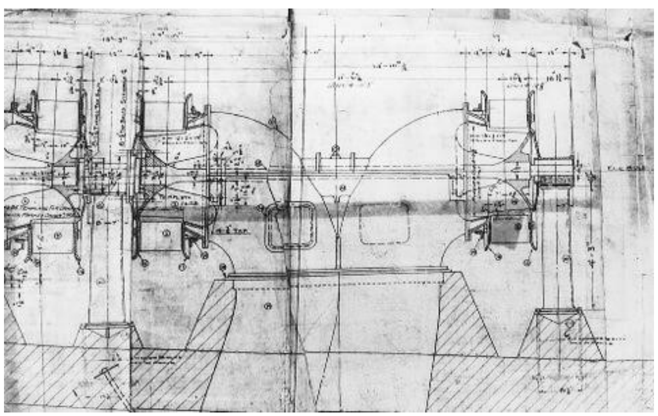
DERIAZ

TURGO

BANKI



CAMEL BACK



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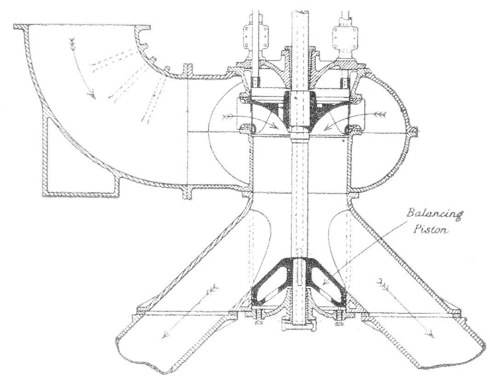

How to design a Francis turbine?

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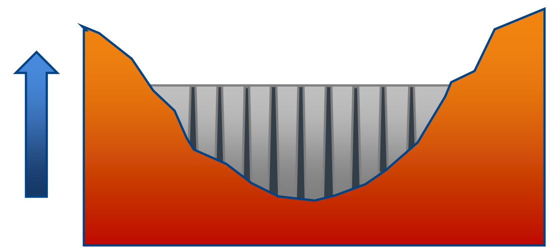
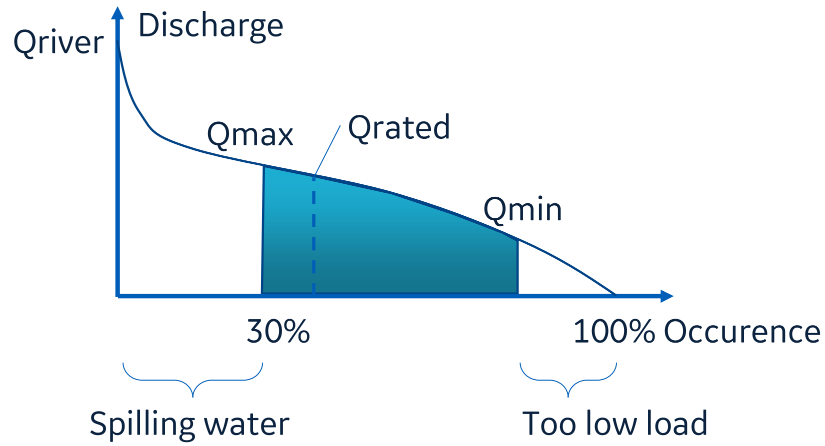
First Dimensioning

First Dimensioning

For a given (new) site, one has to estimate…

- **Max Head →** the higher the dam, the more the potential energy… but the more expensive the **civil work**
- **Max Discharge →** Determined based on the Flow Duration Curve. The higher the max discharge, the higher the max output… but the more expensive the **hydromechanical** supply





First Dimensioning

Number of units will be chosen:

- Less units (more output per unit) → Cheaper!
- For small installations… at least 2 units for maintenance purpose
- For big installations… the individual unit size is the limit (roughly ~ 10m diameter for rotor size)
- Excavation can be the limit for low head installations
- Road access, existing infrastructures… are also to be considered

Examples

Gavet, France
(2 x 50MW)

Three Gorges, China
(32 x 700MW)

First Dimensioning

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Once max head + max discharge (per unit) are chosen

→ **Rotation Speed (n)** is to be chosen

→ directly linked to the Velocity Triangles (inlet and outlet of the runner)

The higher the speed, the less poles in the generator… the cheaper (more compact), but:

→ **electrical limit** = output per pole

→ **mechanical limit** = stresses in the rotor at runaway

Introducing the specific speed (n_s) of the design:
(P in kW or in HP)

$$n_s = \frac{n \cdot P}{H_n^{5/4}}$$

First Dimensioning

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Turbine size defined by the flow speed...

A rough estimation for a Francis = outlet diameter (D_{LP}) of the turbine set so that V ~ 12 to 15 m/s

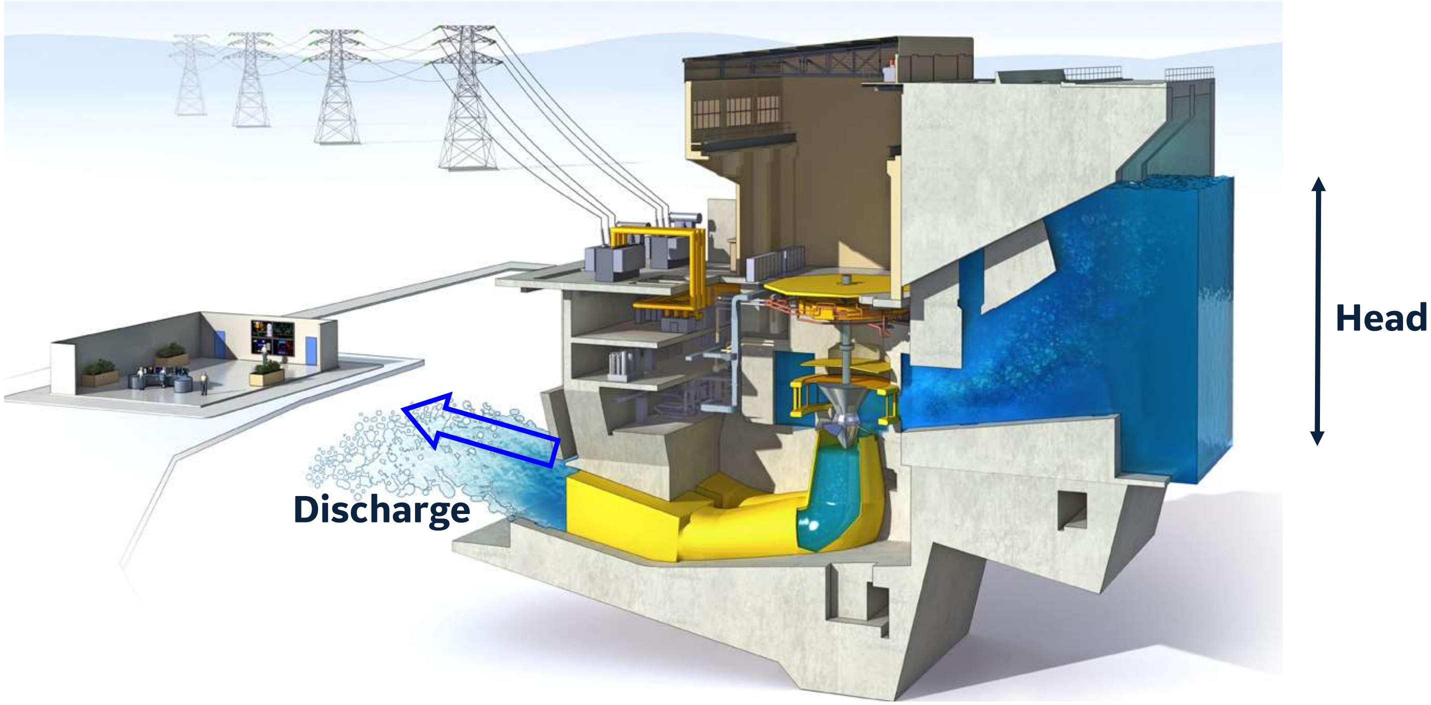
$$V = Q_{max} / \frac{\pi \cdot {D_{LP}}^2}{4}$$

The higher the flow speed, the higher the need for counterpressure on downstream side (higher NPSH) to avoid **cavitation** at the outlet.

For a given output / discharge, the smaller the unit, the lower (altitude) the setting must be. Once again, a tradeoff between **civil work** and **hydromechanical** supply.

From the river to the turbine

Geographic locations / Water data / Tailwater level / head losses in penstocks…



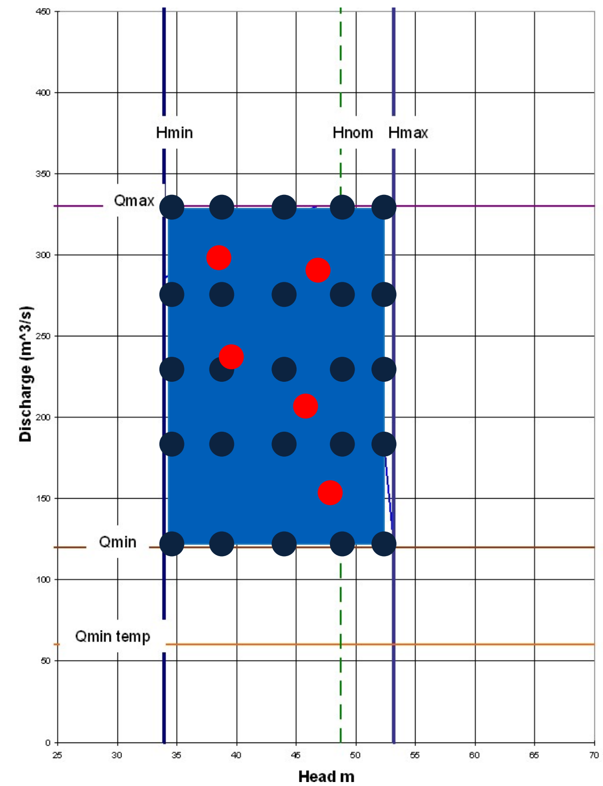
From the river to the turbine

Contract and guaranties come from :

- Site characteristics (discharge/head)
- Economic optimization
- Smooth behavior
- Specific requirements of the customer (fish friendly/erosion…)

Leading to specific constraints :

- Efficiency
- Maximum Output
- Cavitation margins
- Runaway speed
- Level of pressure fluctuations
- Transient behavior (overpressure and overspeed)



Introduction to an efficiency hill chart

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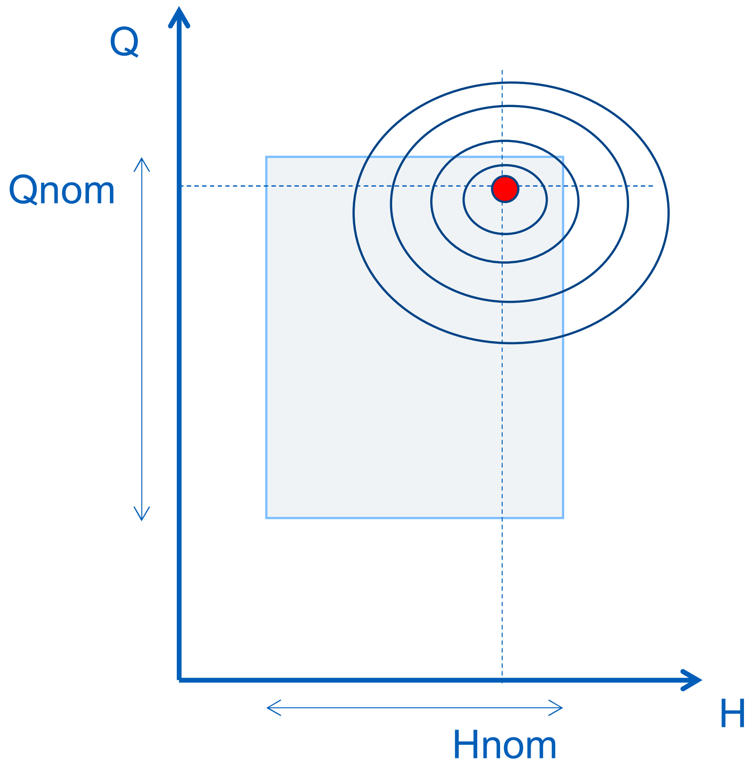

Efficiency → Power

$$P \sim \eta * H * Q$$

Adapting the hill chart
→ Maximizing the **W**eighted **A**verage **E**fficiency
→ Maximizing the power plant productivity

From the river to the turbine

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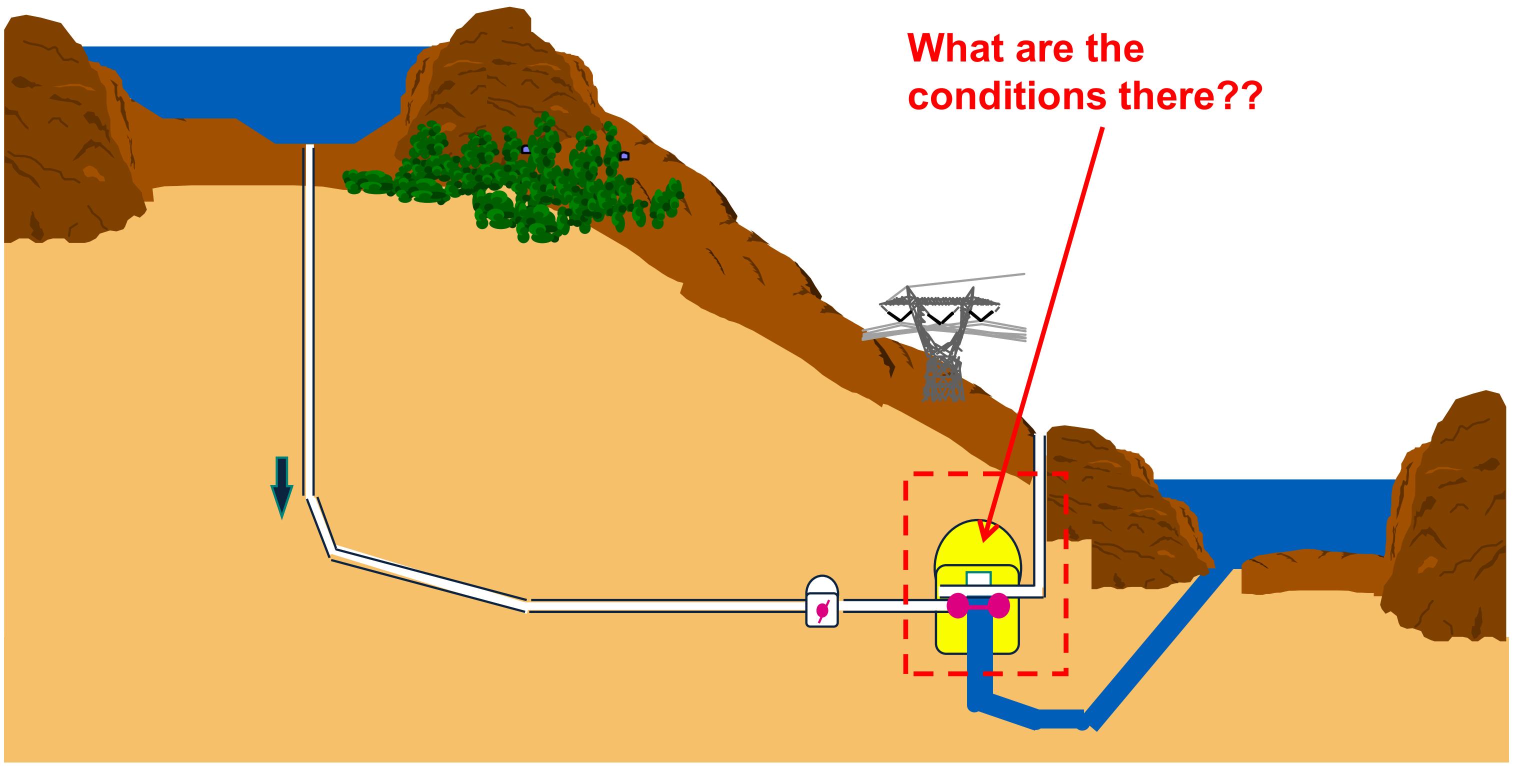
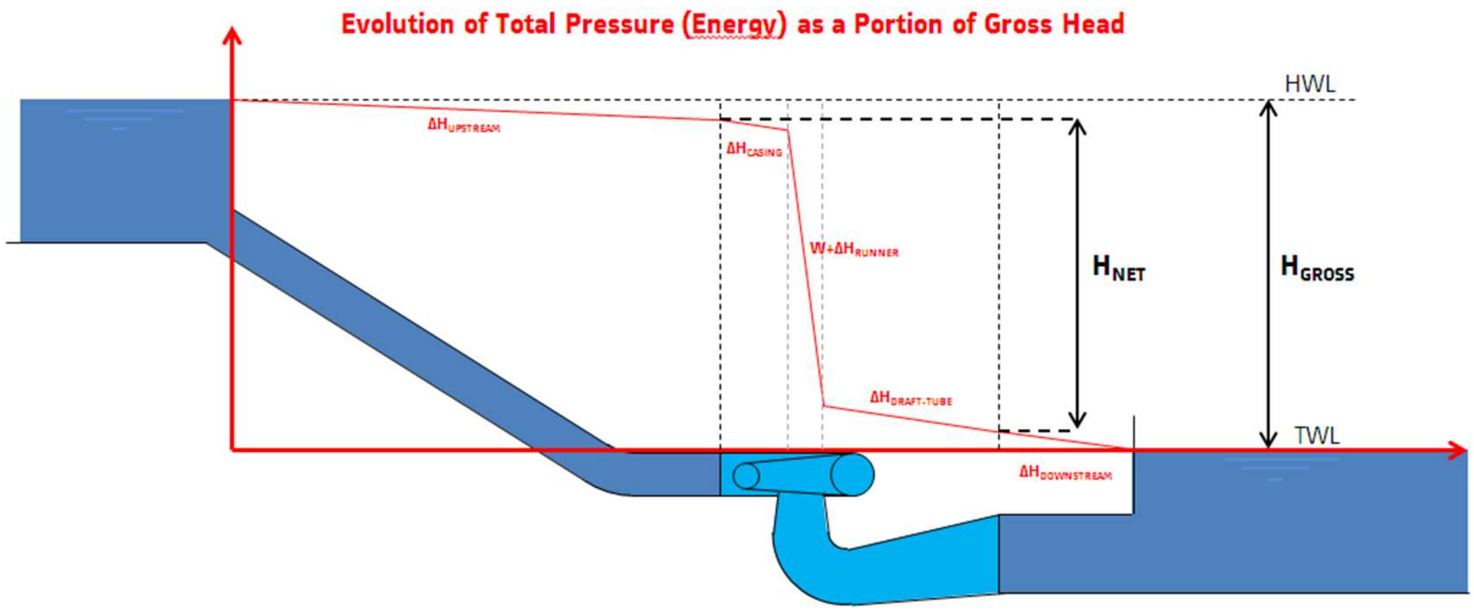
Environment of the Turbine



Energy Evolution in a hydraulic circuit



Different types of energy losses

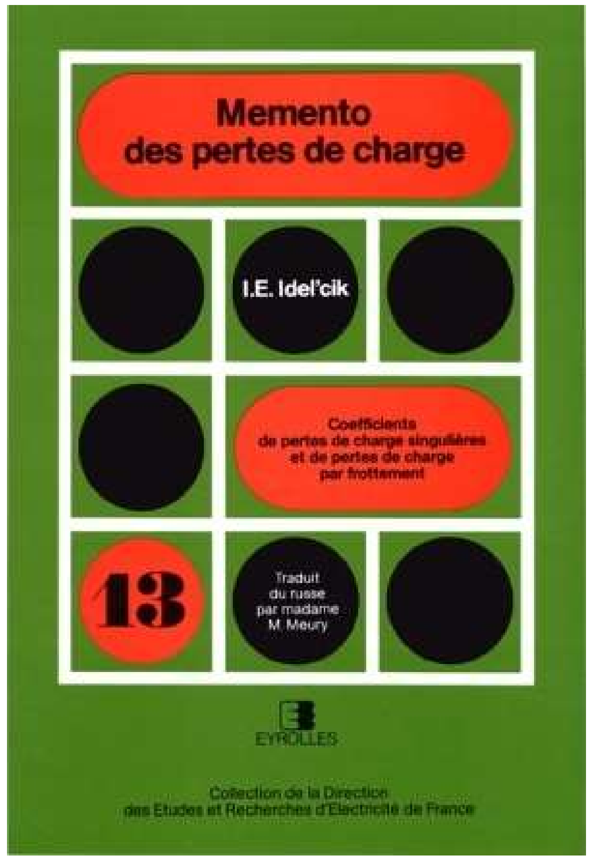
> Head Losses

- Linear losses (friction in tube) DH ~ k * V^2 (flow speed in the tube)
  with k ~ L (length)
- Singular losses
  - Convergent / Divergent
    → Controlling water passage: notion of **section law**
  - Incidence / Wake
    → Shaping the blades: notion of **adaptation**
  - Flow separation (can be seen as a local section law modification)

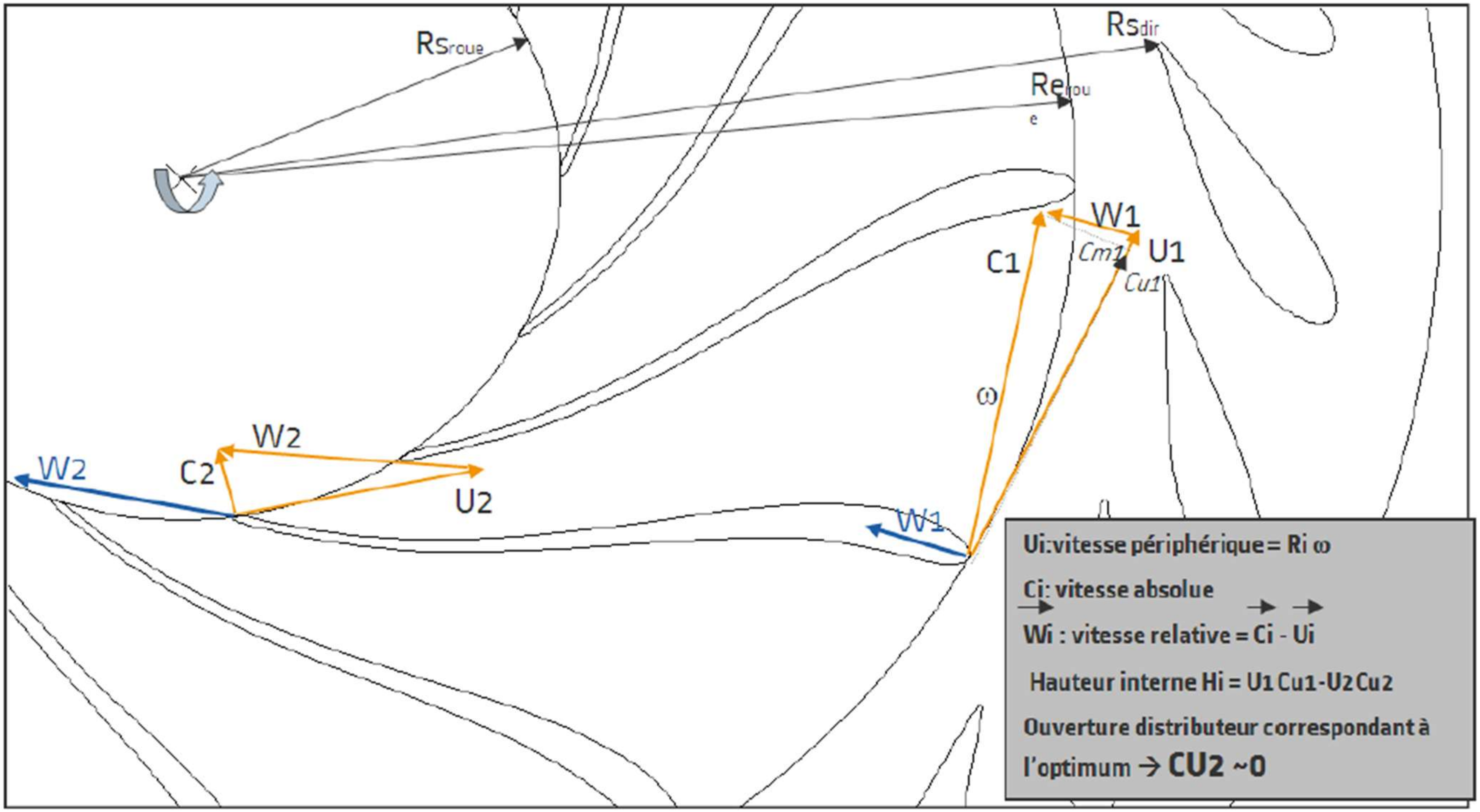
> Volumetric Losses

> Other friction Losses

Bible for head loss coefficient “Idel’cik”



Runner Design: Velocity triangles



Two equations daily used by a hydraulic engineer

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→ **Euler**

E/g = U1.Cu1 – U2.Cu2 = η*H
introduction of "internal head" = Work
Hi = E/g = U1.Cu1 – U2.Cu2
η = Hi/H

→ **Bernoulli**

Definition of "Total Pressure"

$P \quad + \rho g z \quad + \frac{1}{2}\rho V^2$ (in Pa)

pressure, potential, kinetic

$P/\rho g \quad + z \quad + V^2/2g$ (in mWC)

→ constant on a stream line of a perfect fluid (no head loss)

in real life, between A and B :

$$P(A) + \rho g z(A) + \frac{1}{2}\rho V(A)^2 = P(B) + \rho g z(B) + \frac{1}{2}\rho V(B)^2 + \Delta Ptot(A \rightarrow B)$$

Back to the environment of the Turbine

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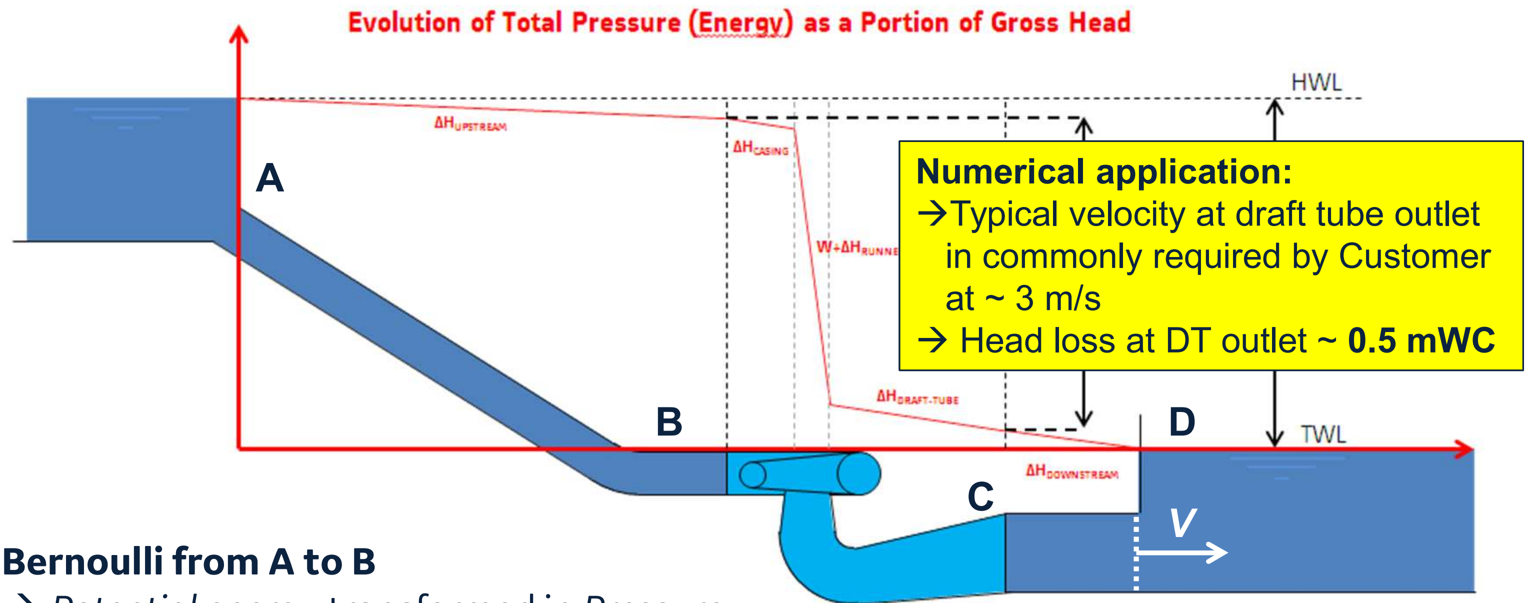

Bernoulli from A to B

→ *Potential energy* transformed in *Pressure*

→ DH in penstock in $k*L*V^2$
"the longer the more losses"
"the smaller the diameter the more losses"

→ Economical optimum

Bernoulli from C to D

→ *Pressure* in *Potential*

→ Friction losses (in kLV^2)
+ Sudden enlargement
$DH = V^2/(2g)$

Components overview

Different parts of the turbine

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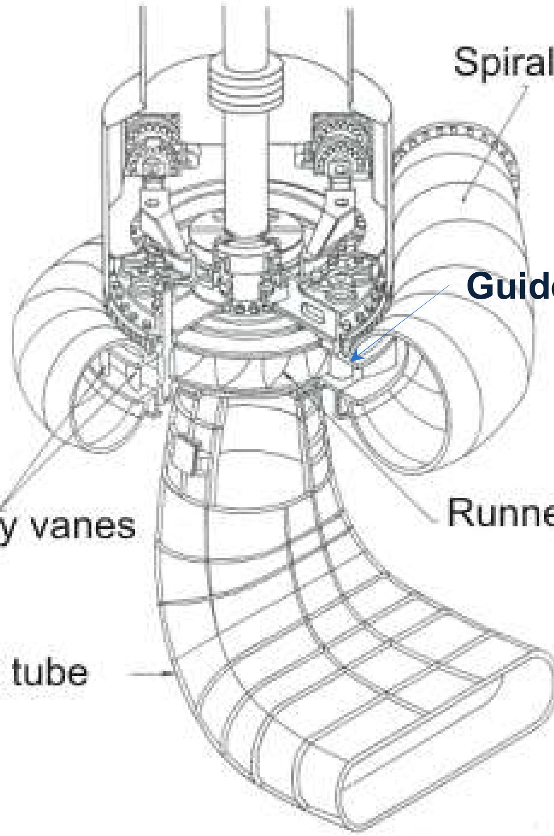
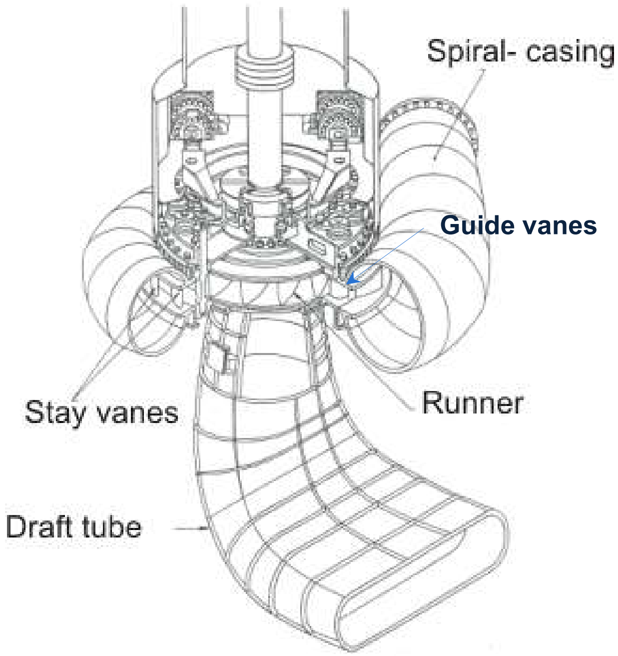
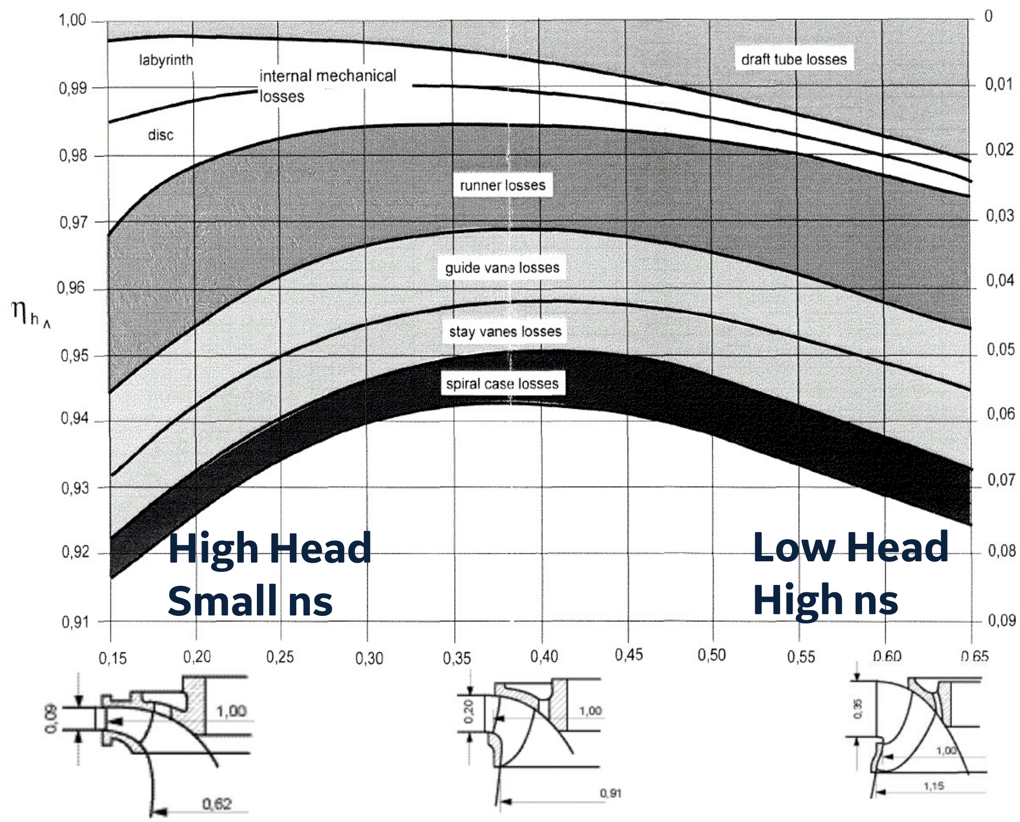


Loss distribution in a Francis turbine





What is the main head loss in the hydraulic circuit?

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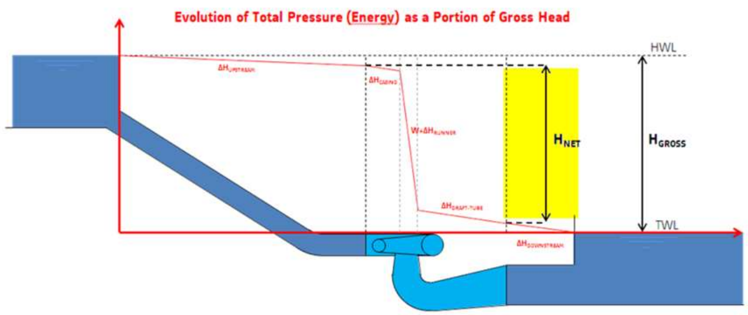
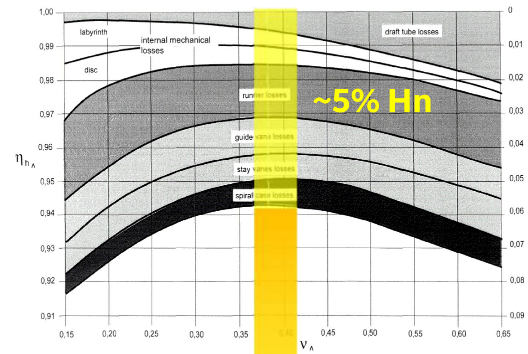




Consequences

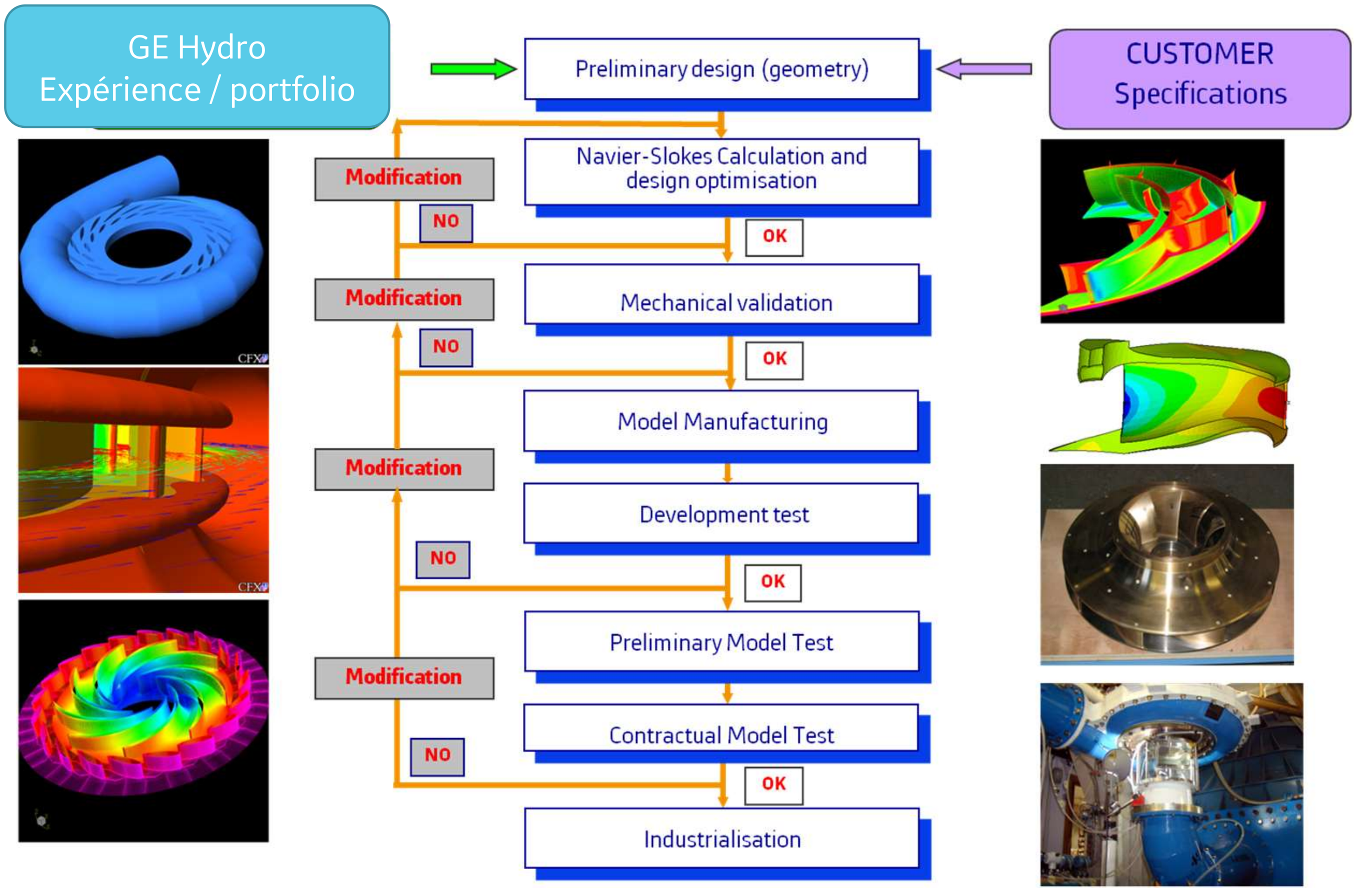
→ Disconnection between upstream and downstream *"the draft tube doesn't see the head"*

→ At the first order, the runner itself grants a good flow distribution on the upstream side

WORK recovered by the runner

~ 95% Hn
(~ 90% Hg)

Hydraulic design process



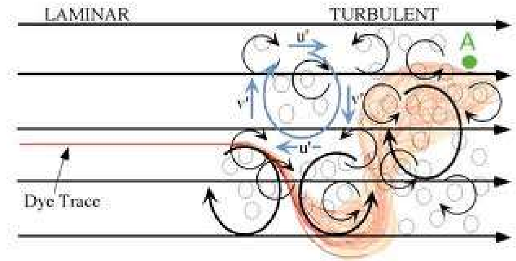
Main design tools : CFD

Resolving Navier Stokes equations : $\rho\left(\frac{\partial \mathbf{v}}{\partial t}+\mathbf{v}\cdot\nabla\mathbf{v}\right)=-\nabla p+\mu\nabla^2\mathbf{v}+\mathbf{f}.$

Nonlinear partial differential equations
Turbulence (time dependent chaotic behavior)
No analytics solution → One of the Millenium Prize Problems (US$1,000,000)

Numerical simulation :

- Discretisation of equation in discritized space
- Significantly different mixing-length scales : Full resolution is not reachable for industrial flows
- Use of statistical model



Lots of uncertainties during the design of hydraulic component

It's very important to validate performances on the test-rig

→ Especially for off-design points (start-up, runaway, low output)

→ Unless very close to a tested reference

Performance validation by Model tests

Similitude rules

Full geometric similitude
Minimum Reynolds number condition
Step up formula

Prototype :

- Head/Discharge
- Dimensions
- Instrumentation not easy
- Operating point not chosen

Defined in International Standard IEC 60193 and 62097

Model Scale :

- Small Scale (D~350mm)
- Test rig is designed for measurements with adapted sensors
- Choice of the measure and functionment points

Step up formula and scale effect

$$Q_p = Q_m \left(\frac{n_p}{n_m}\right)\left(\frac{D_p}{D_m}\right)^3$$

$$H_p = H_m \left(\frac{g_p}{g_m}\right)^{-1}\left(\frac{n_p}{n_m}\right)^2\left(\frac{D_p}{D_m}\right)^2$$

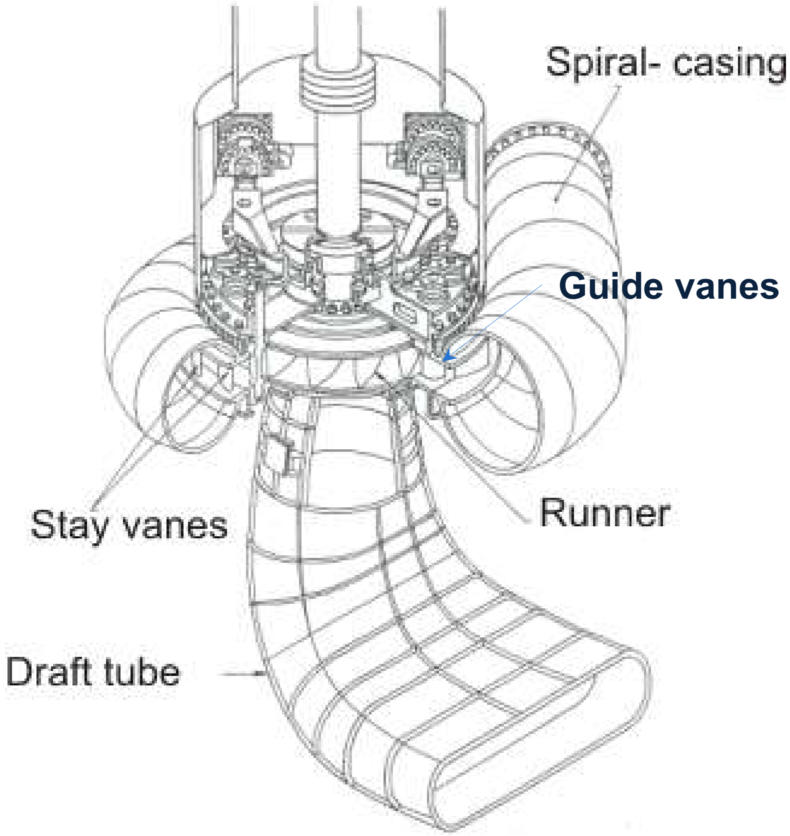
$$T_p = T_m \left(\frac{n_p}{n_m}\right)^2\left(\frac{D_p}{D_m}\right)^5$$

$$\eta_p = \eta_m + \Delta\eta$$

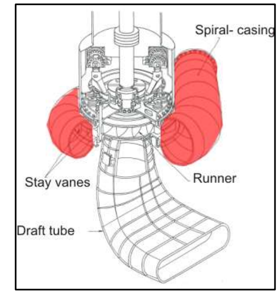
Hydraulic Design

Let's start!

→ Reminder of lexicon



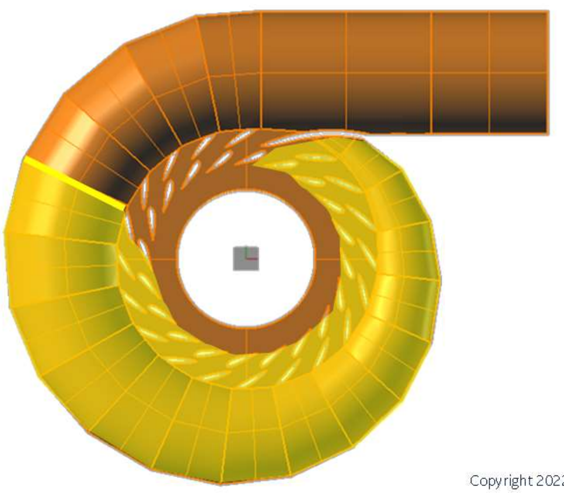
Spiral Casing

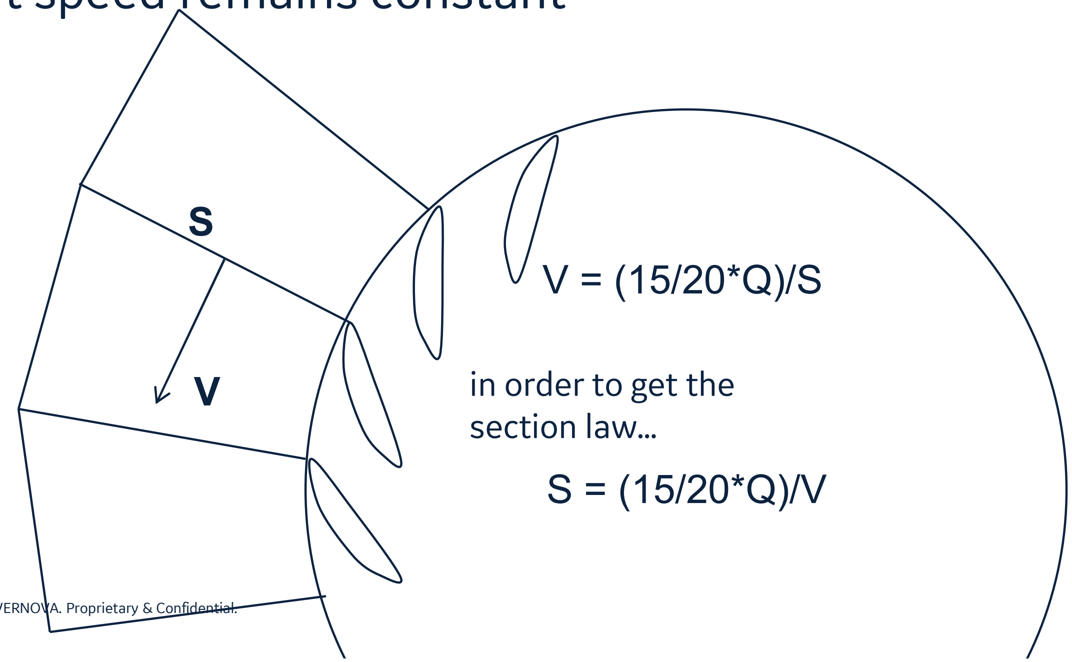


- Ensure an even flow distribution around the runner
- Hypothesis = runner grants a homogenous flow distribution

  → spiral casing should guarantee a good distribution of **speed** and **flow angles**

  → Section law that will make sure that speed remains constant in the different sections





Spiral Casing

- Hypothesis 1: uniform flow distribution around the runner
- Hypothesis 2: uniform flow distribution in sections

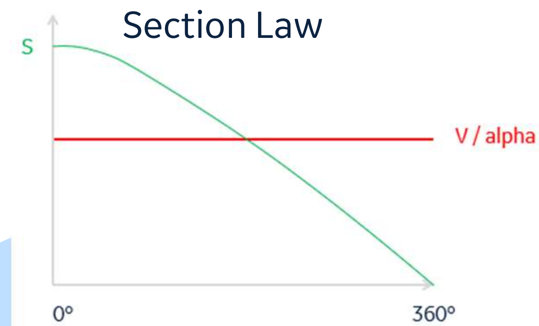
Vradial = Q / S1

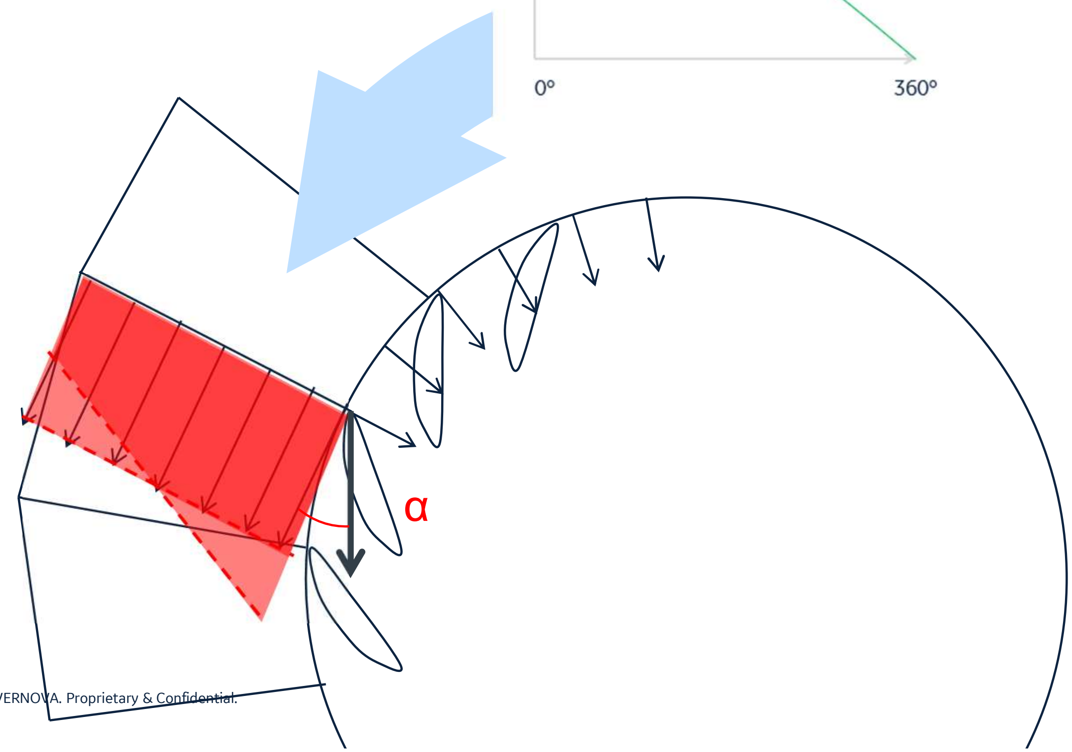
Vtang= n/ntotal * Q / S2

→ α = atan(Vradial/Vtang)
= atan(n/ntotal*S2/S1)
independent from Q!

Flow Angle at Stay Vane inlet
is supposed to be constant
(one profile for all stay vanes)

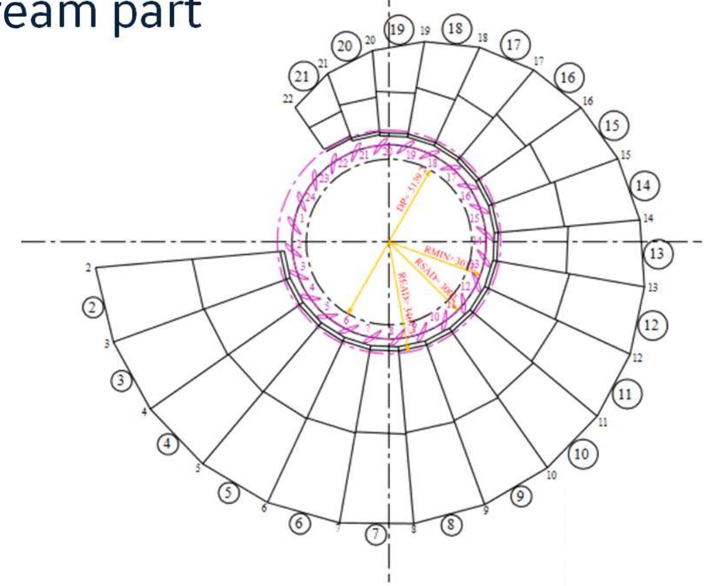
→ Thickness of the Stay Vane
makes it tolerant to angle
variation

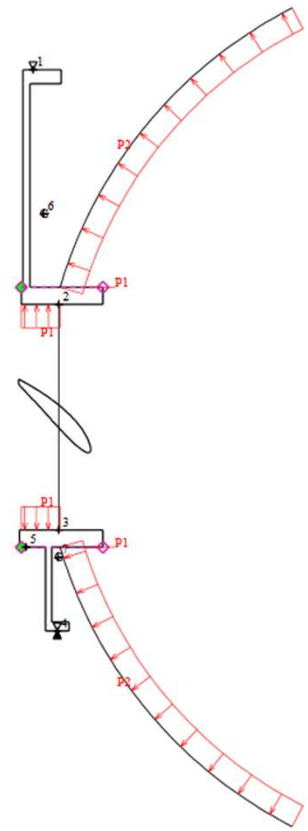


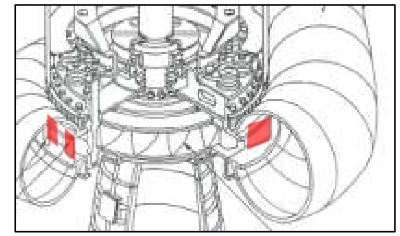


Stay Vane

- Mechanical function to stand the hydraulic pressure in the upstream part

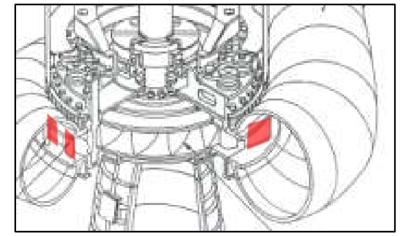






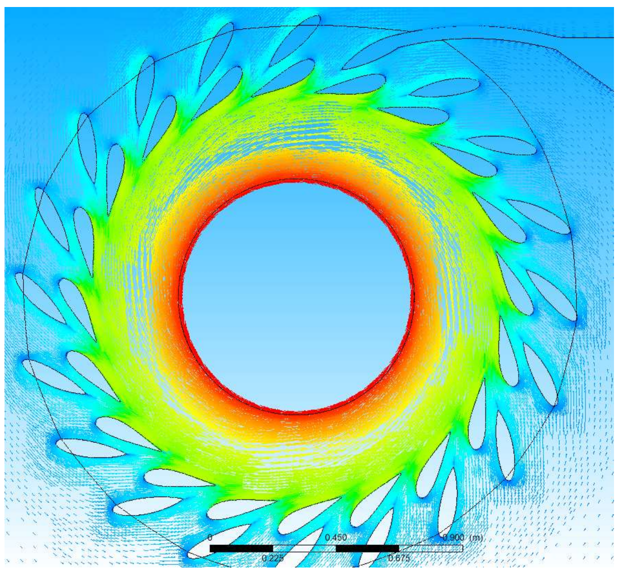
→ **Thickness, stiffness, inertia**

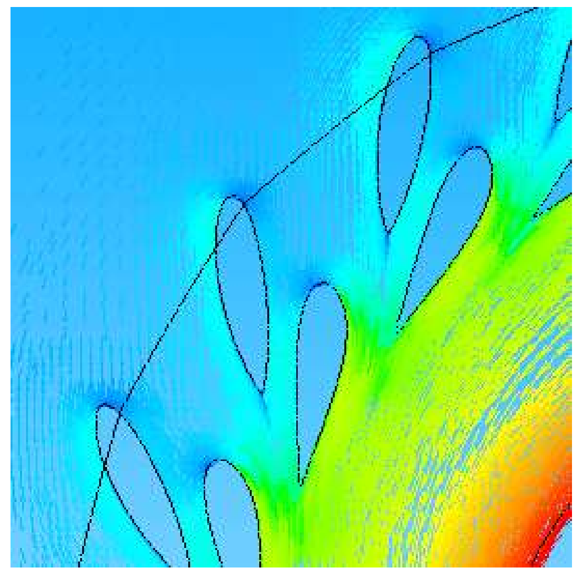
Stay Vane



- Since it *has to be there*, it is designed to be “hydraulic friendly” ☺

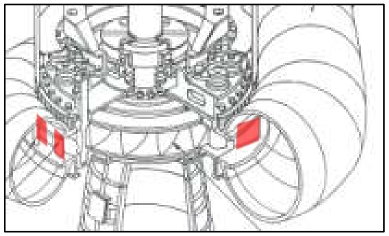
- Thick leading edge to tolerate angle variation (even if small)
  Thin trailing edge to minimize losses (+ Von Karman issues…)

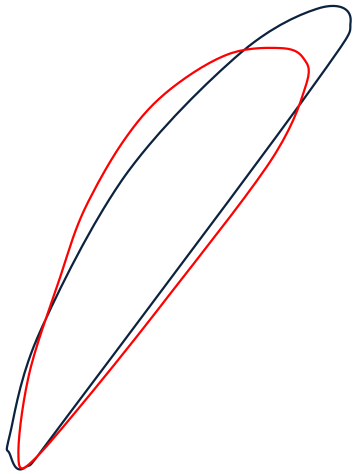


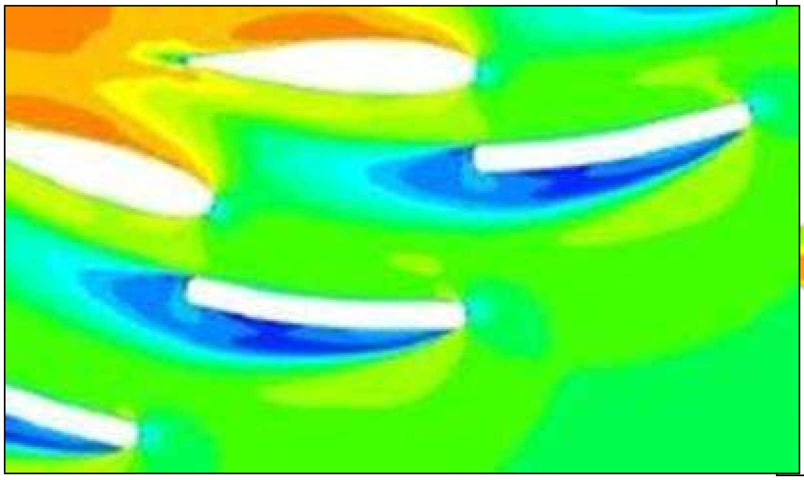


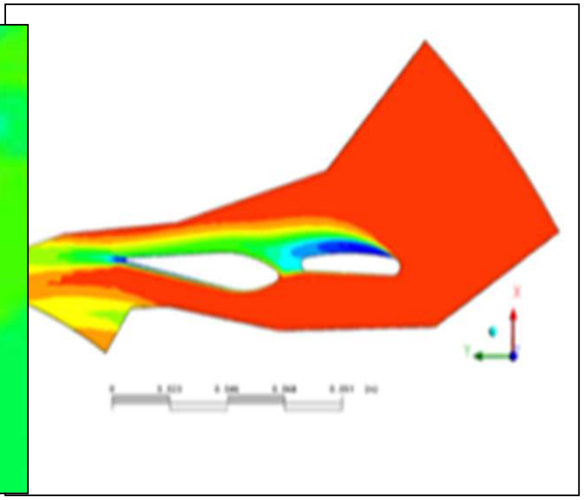
Stay Vane

- These two designs are more or less **equivalent** for mechanical purpose... but what consequences on the hydraulic?

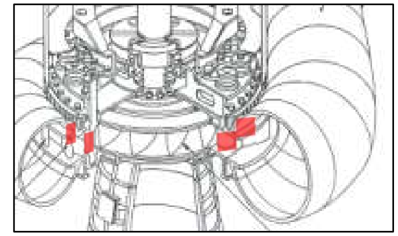






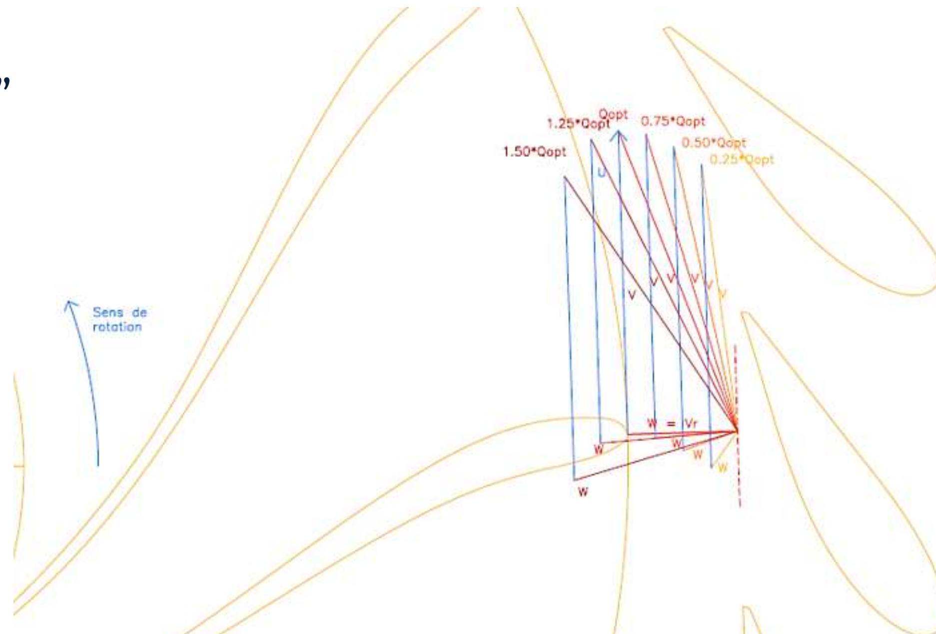


Distributor



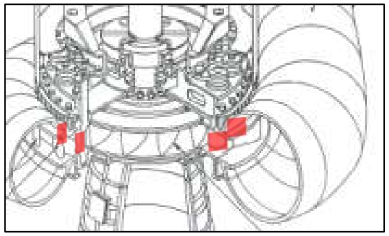
- For a simple regulated turbine (like a Francis),
  it is the only device to regulate the discharge / power / load
- Its function is to deliver **a flow angle** to the runner

- it is not strictly speaking a “tap”
  since main head loss still
  comes from the runner



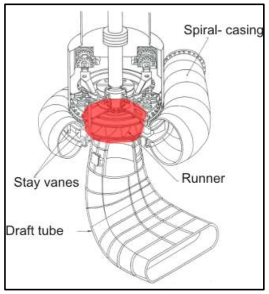
Distributor

- Designed to **stand the head** (especially high head machines) and to **accept various flow angles**
- Flow angle coming from stay vane is constant
- But guide vane angle is not constant (from 0 to 30-40°)



- Leading edge needs to be tolerant = thick
- Trailing edge, again, needs to be thin (wake and Karman)

Runner



- Transform **kinetic and pressure energy** in **mechanical energy**
- Designed for an optimal Head and an optimal Discharge

Available Energy (Hydraulic)

Ph = ρ *g Hn Q*

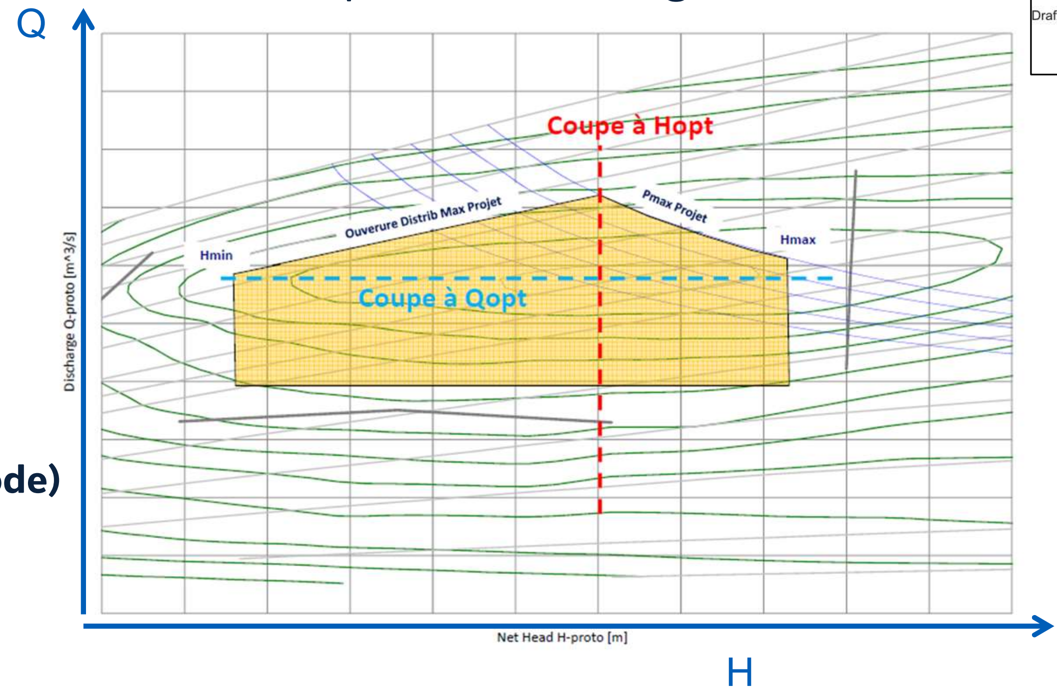
Transformed Energy (mechanic)

*Pm = w*T*

*Rotationnal Speed * Torque*

Definition of Efficiency (turbine mode)

η = *Pm / Ph*



Runner

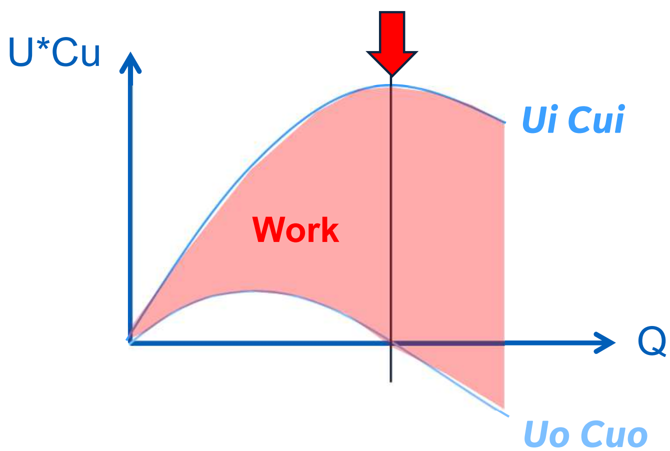
Available Energy

- Δ(Ptot) – Head losses out of Turbine (Inlet + Outlet) = $\rho\, g\, Hn$

Energy transferred to the Shaft by the Runner

- Moment of Momentum theorem (fluid volume in the runner)
- Mass Energy transferred E (i for Inlet / o for Outlet)

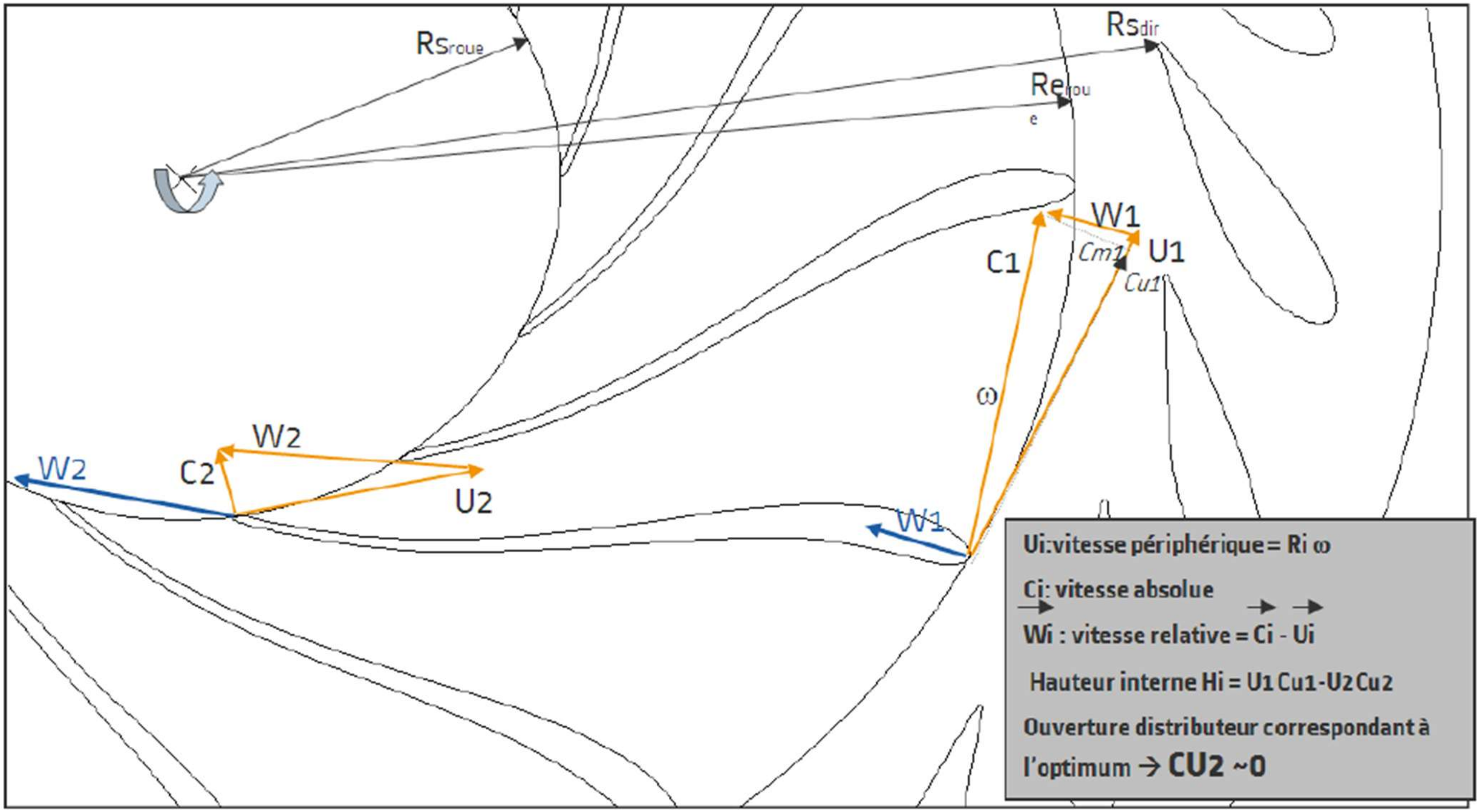
$$E = Ui\ Cui - Uo\ Cuo$$



U = r*ω
C absolute velocity
Cu : tangential component

<u>Optimal:</u>
UCu @inlet maximal
UCu @outlet null

Runner Design: Velocity triangles



Hydraulic design - Runner

A second usefull equation : Euler

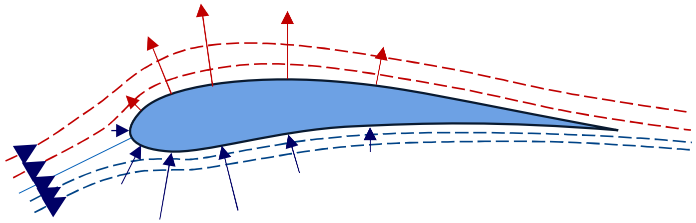
→ **Euler** $H_i = \frac{U_{HP}.Cu_{HP} - U_{LP}.Cu_{LP}}{g}$

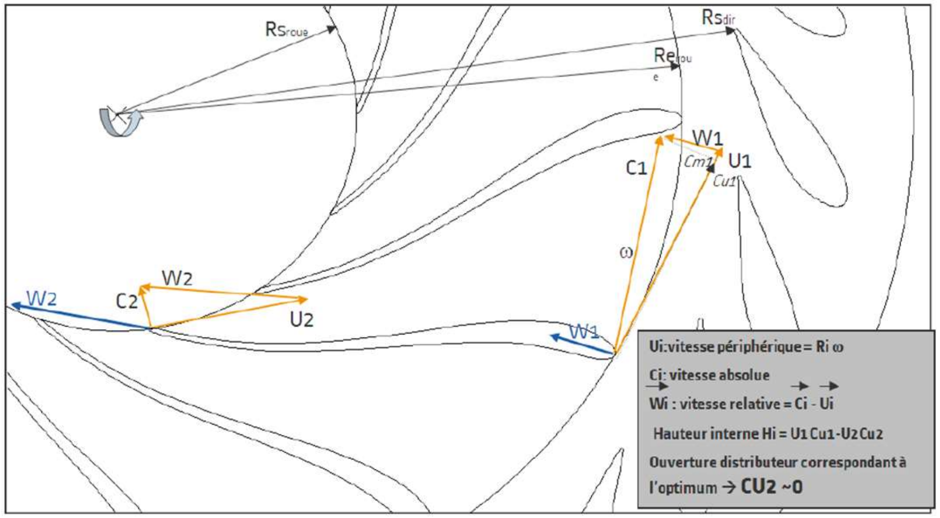
$$\eta_H = \frac{H_i}{H} \text{ (in turbine mode)}$$

The internal head Hi is a function of the angle evolution

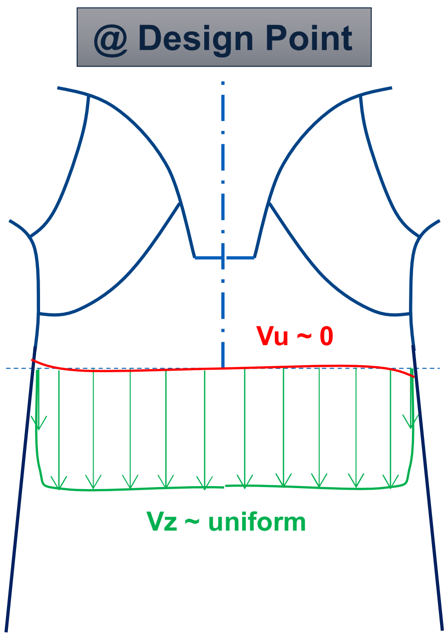
E = gHi may also be written

$\int_{blades}$ *(p pressure side– p suction side) ds*





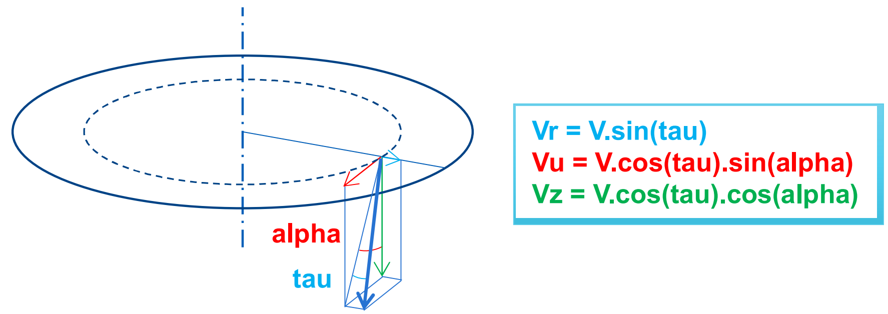
Velocity Profiles at runner outlet



Axi-symmetry hypothesis on velocity profiles

→ Velocity profiles can be described *on a radius*

→ *Cylindrical* projection is the most appropriated



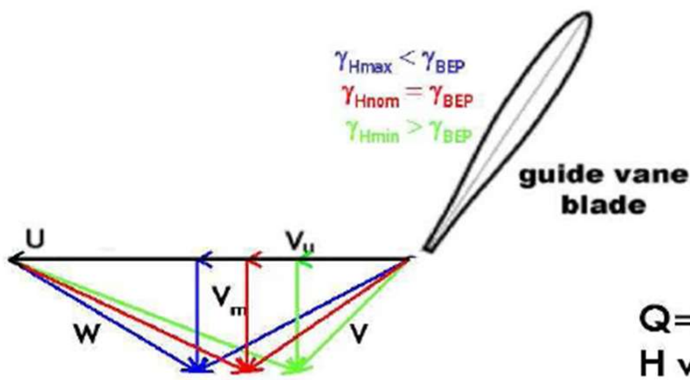
If considered on a diameter… (e.g. *x* axis)

→ *Vz* is an <u>even</u> function of *x*

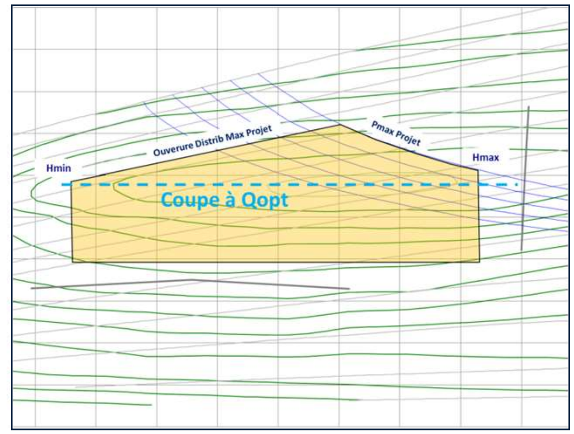
→ *Vu* and *Vr* are <u>odd</u> functions of *x*

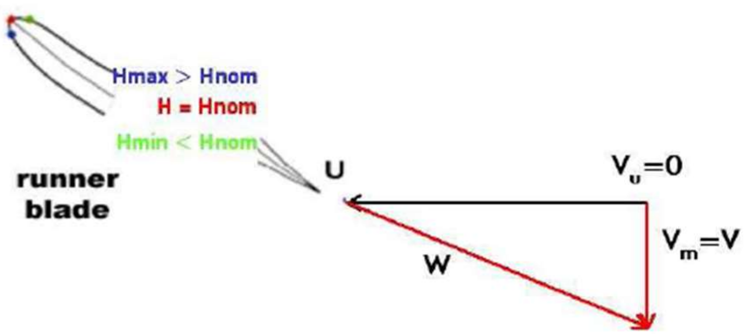
→ "theoretically" on unit axis (r = 0) **Vu = Vr = 0**

Cut at Qopt → Head influence



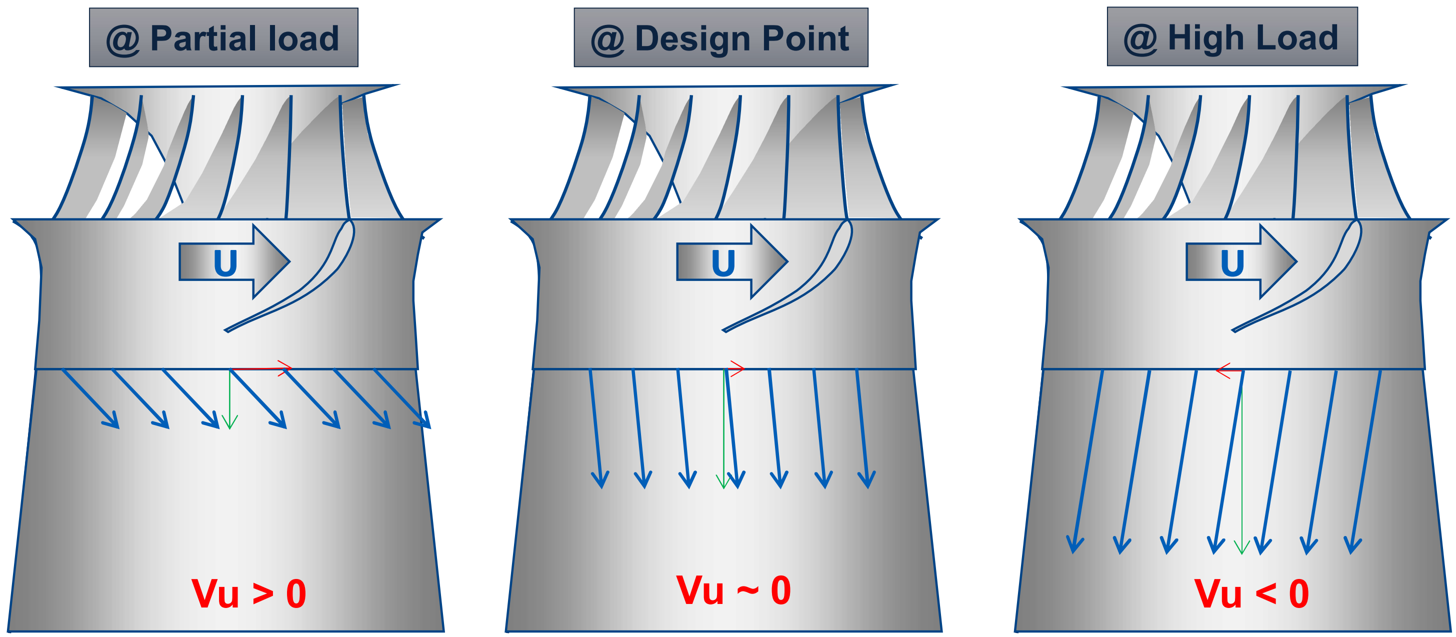
Q=cste=Q_{BEP}
H variable





"The Draft tube does not see the head"

Velocity Profiles at runner outlet



Cut at Hopt → Discharge influence

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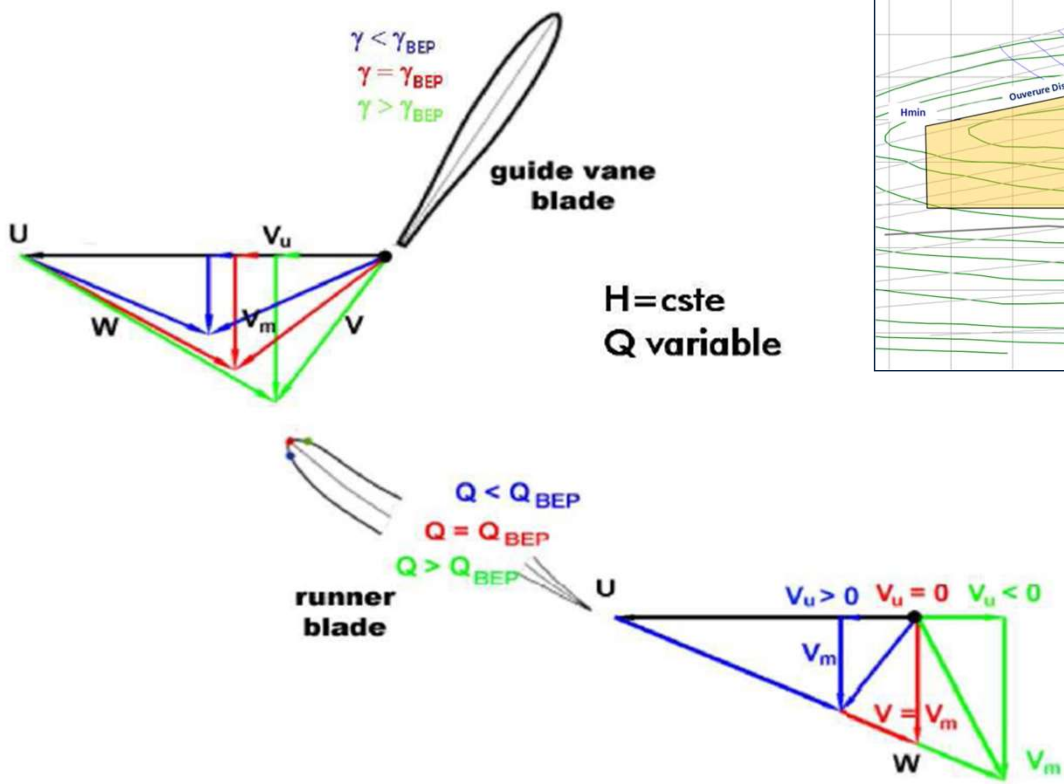
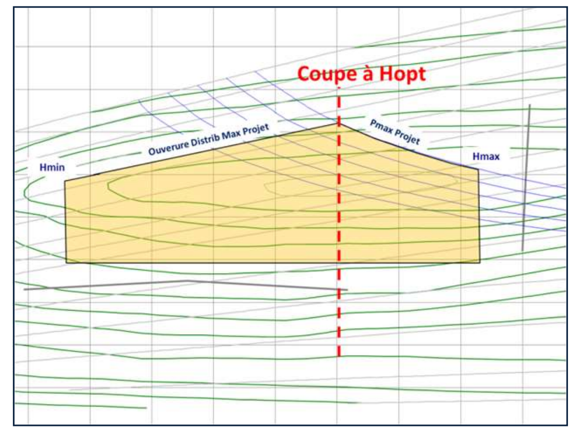
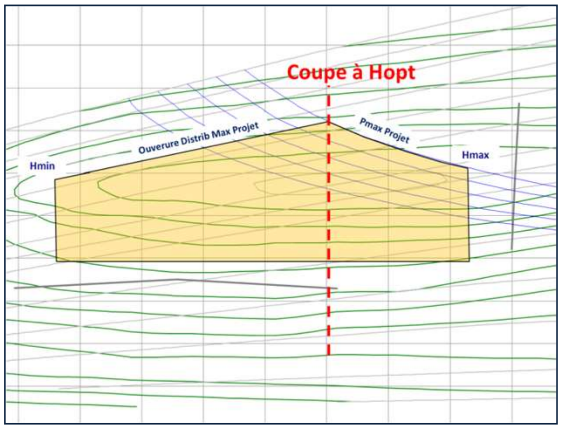




How to design a runner?



Inlet adaptation

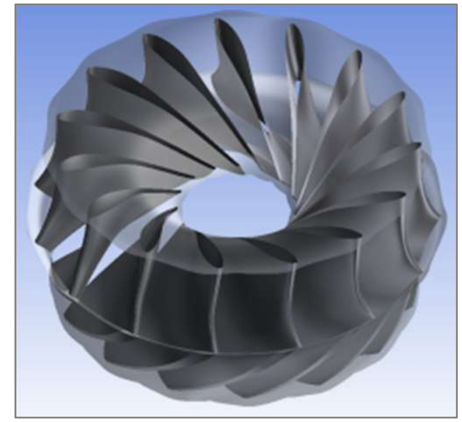
- Designed for optimal head
- Inlet cavitation limits

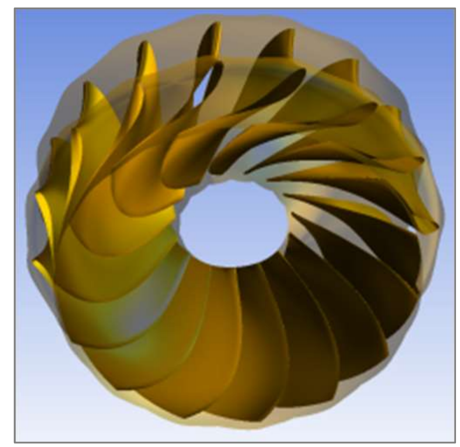
Blade loading

- Ensure a proper “meridian equilibrium”

Outlet adaptation

- Designed to deliver Cu = 0 (nominal discharge)
- Ensure good margin with outlet cavitation





Runner Design: Meridian Channel

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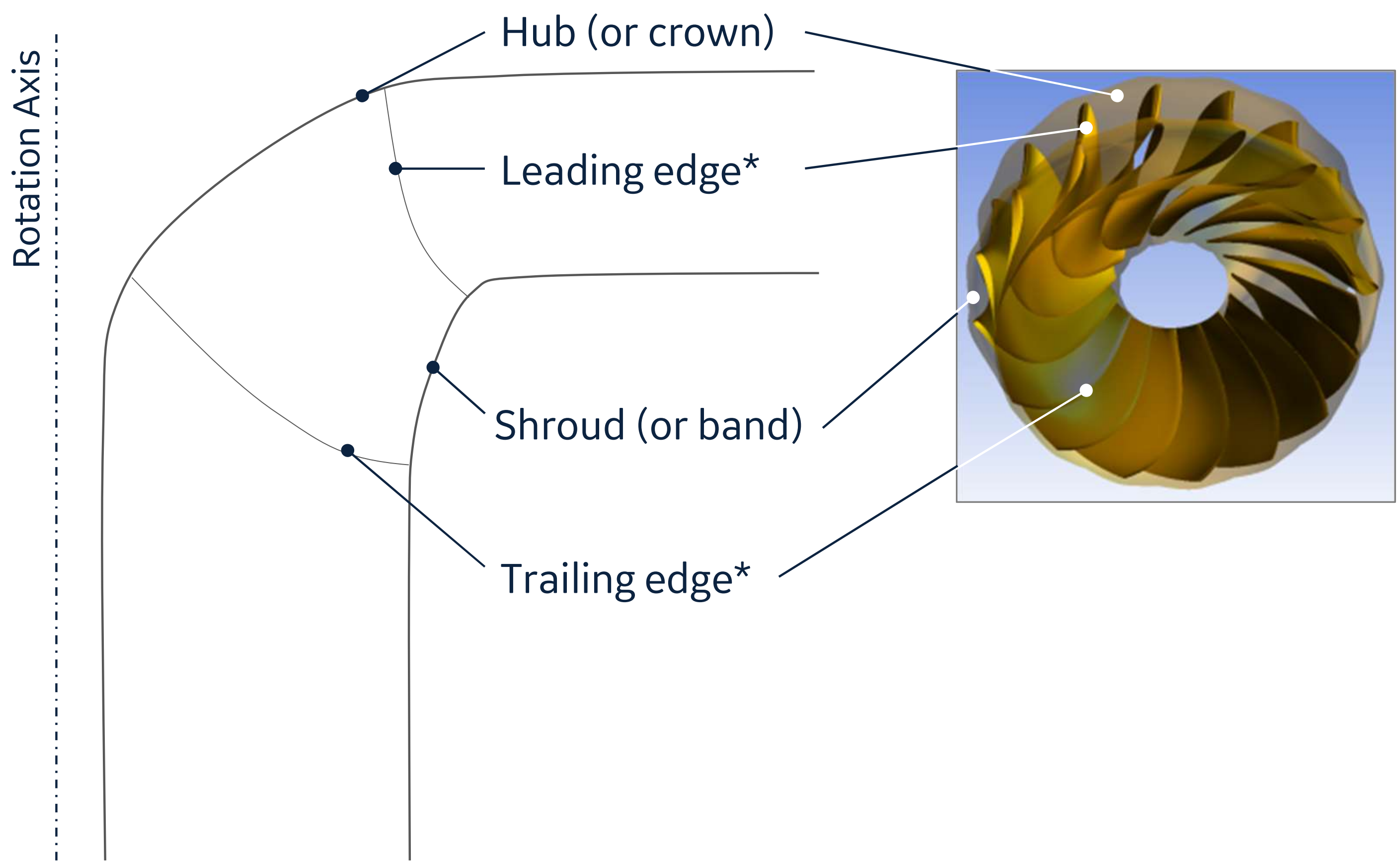

* projection in 2D

Runner Design: Meridian Channel

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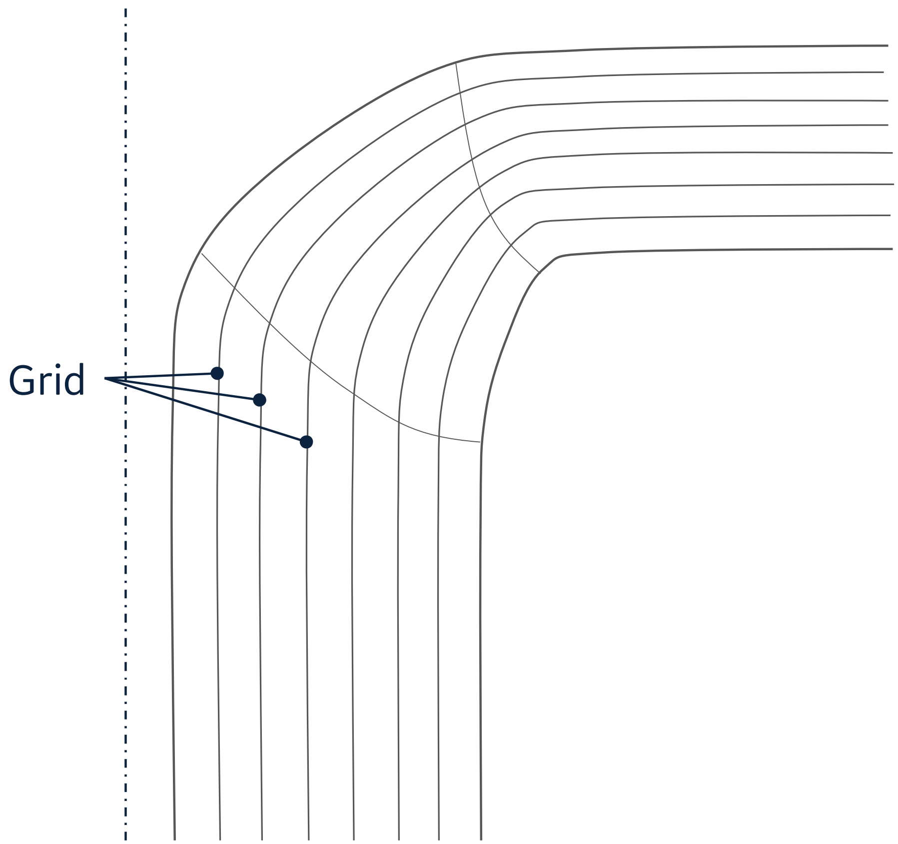
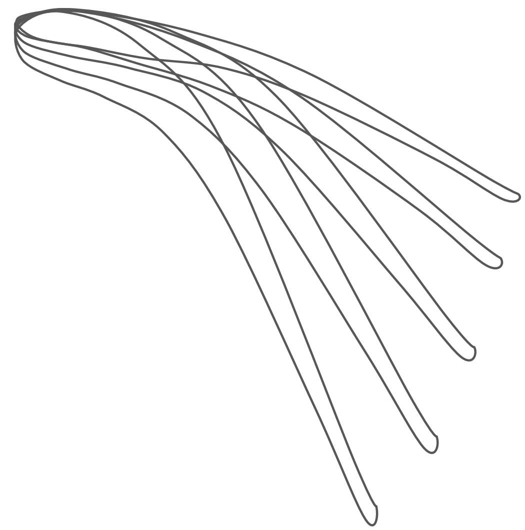
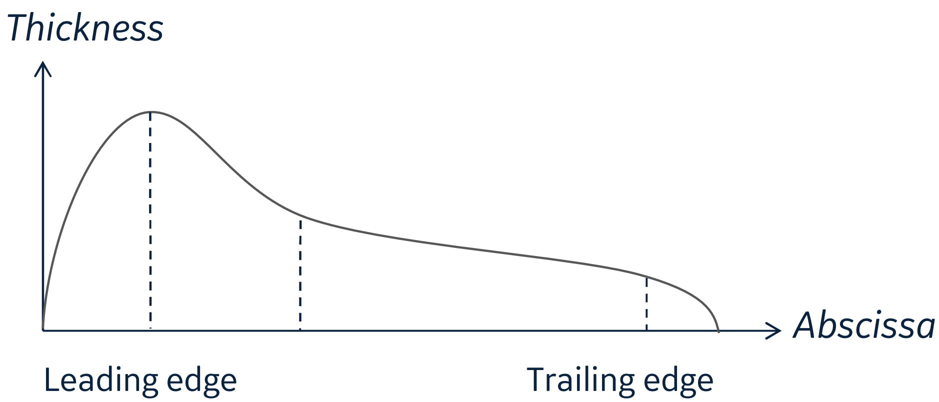


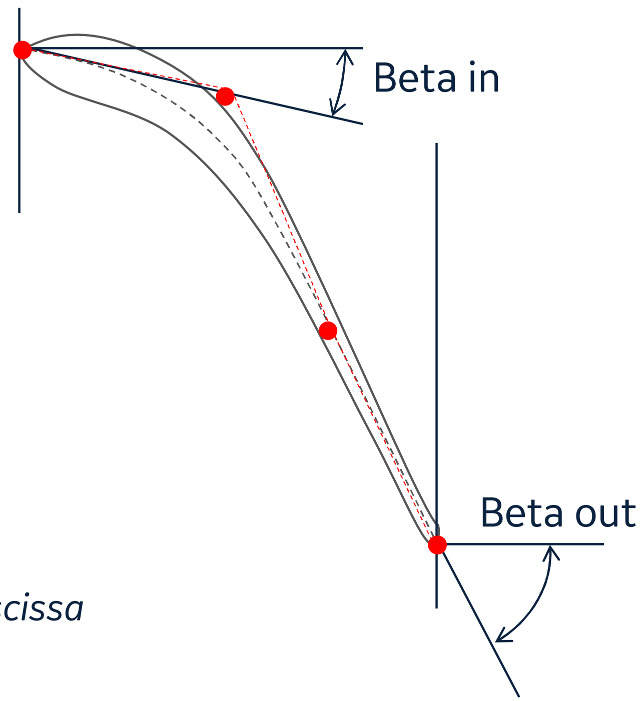
Elementary turbines



Runner Design: Elementary turbines

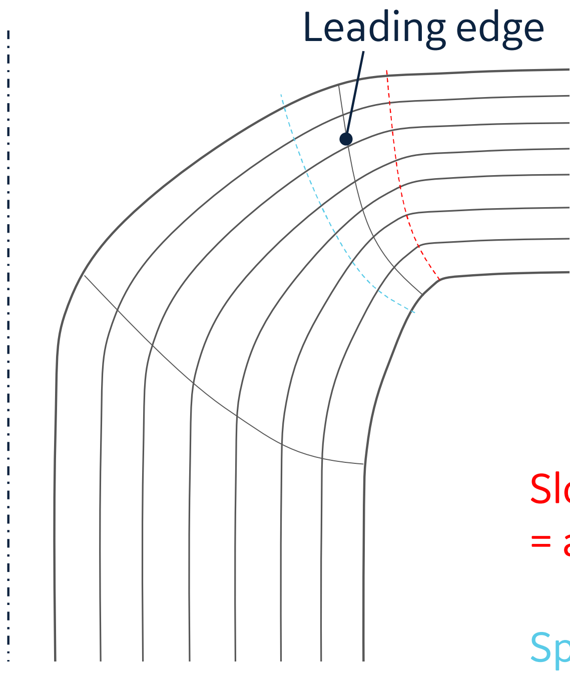
Each elementary turbine defined by:
- 4 poles for curvature (6 coordinates)
- 3 abscissa + 3 thickness values

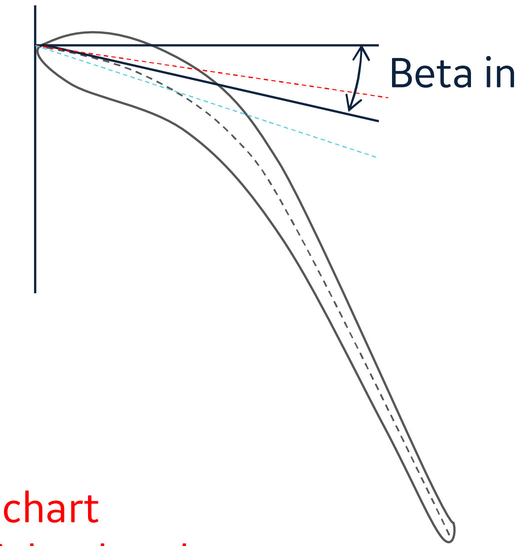




For a typical Francis runner: 13 elementary turbines
→ Up to 156 degrees of freedom!

Runner Design: Head adaptation



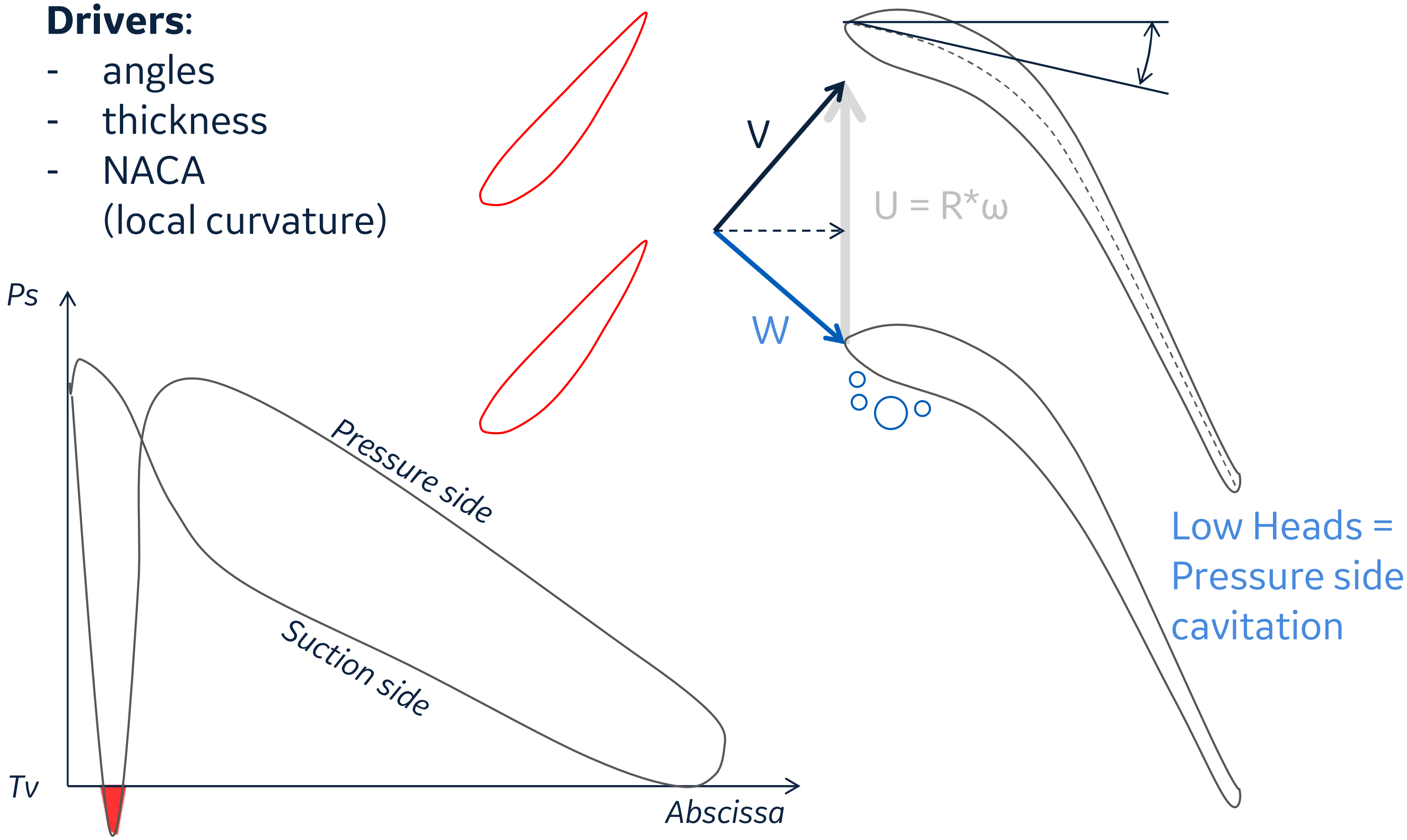


Slow down hill chart
= adapted to higher heads

Speed up hill chart
= adapted to lower heads

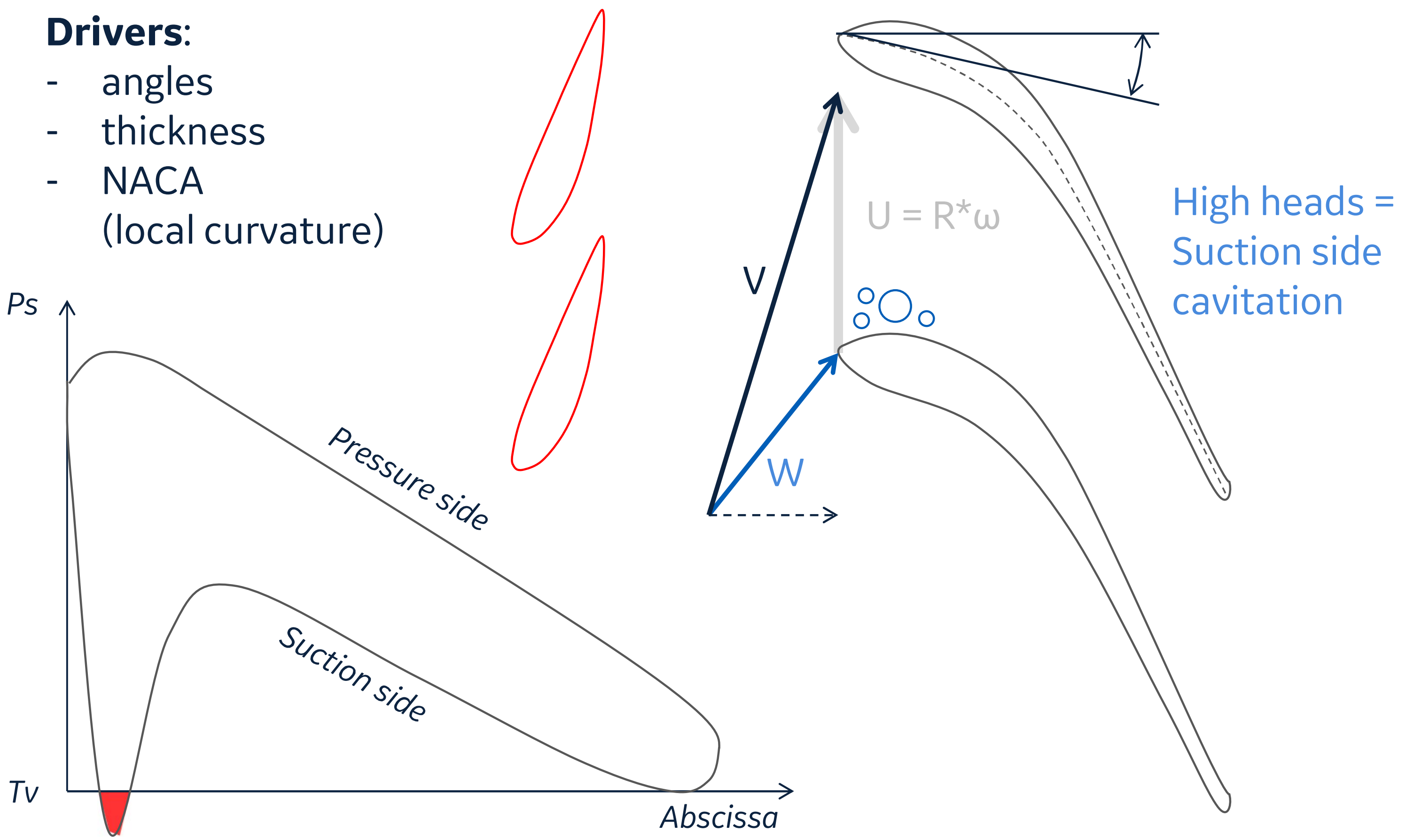
Runner Design: Inlet Cavitation limits

Drivers:
- angles
- thickness
- NACA (local curvature)



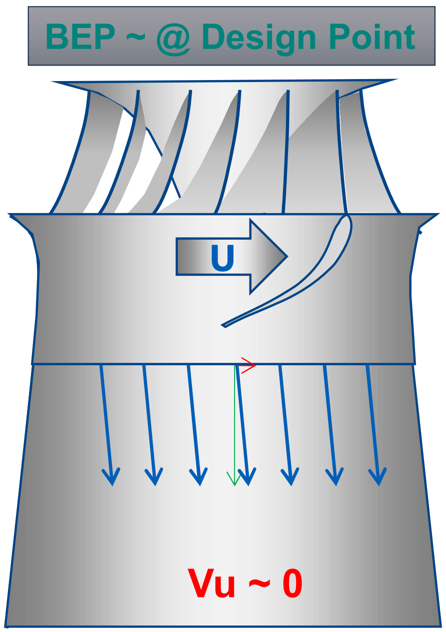
Runner Design: Inlet Cavitation limits

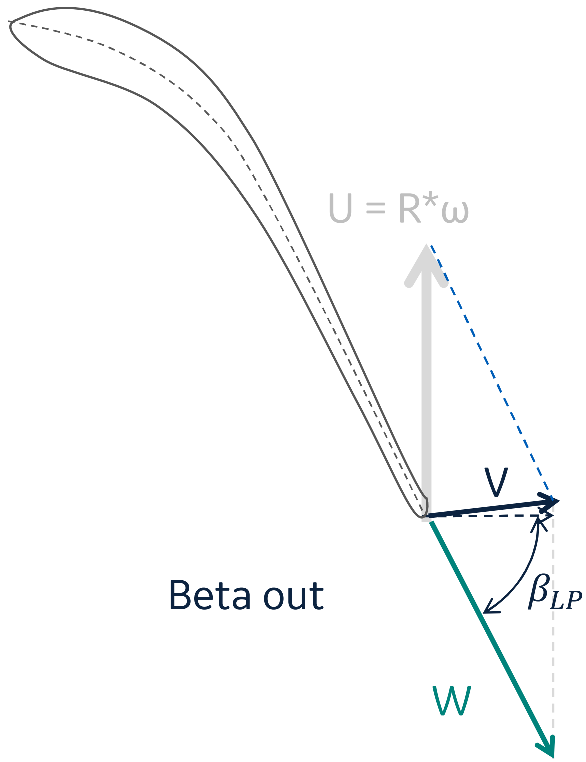
Drivers:
- angles
- thickness
- NACA
  (local curvature)

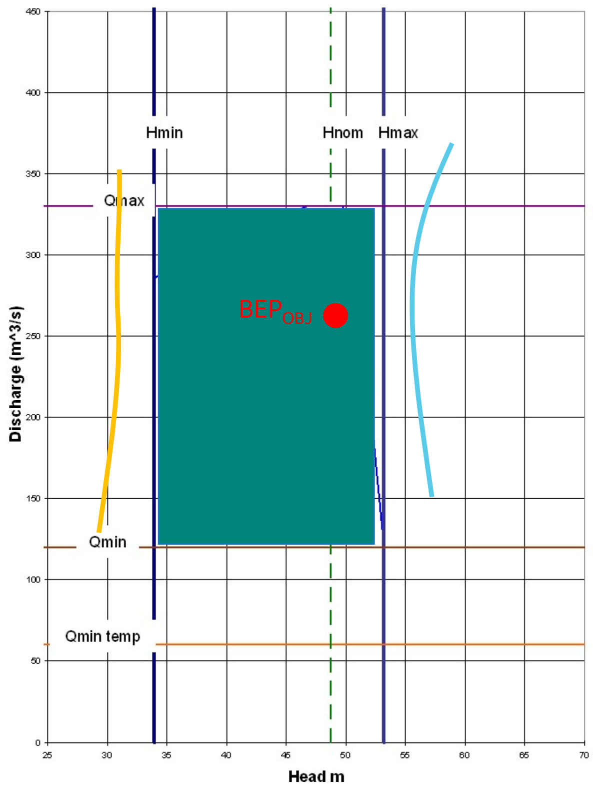


How to design a Francis runner ?

Discharge adaptation

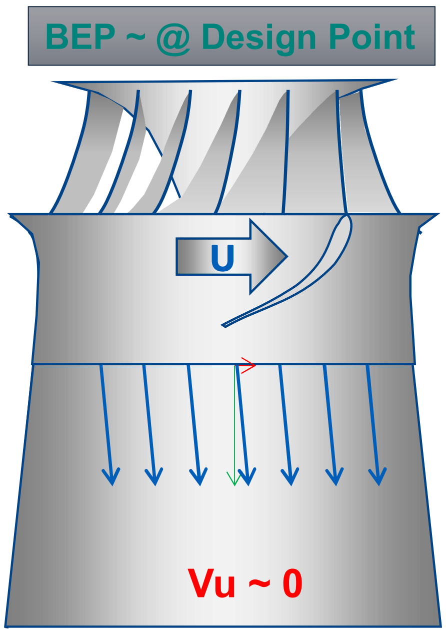


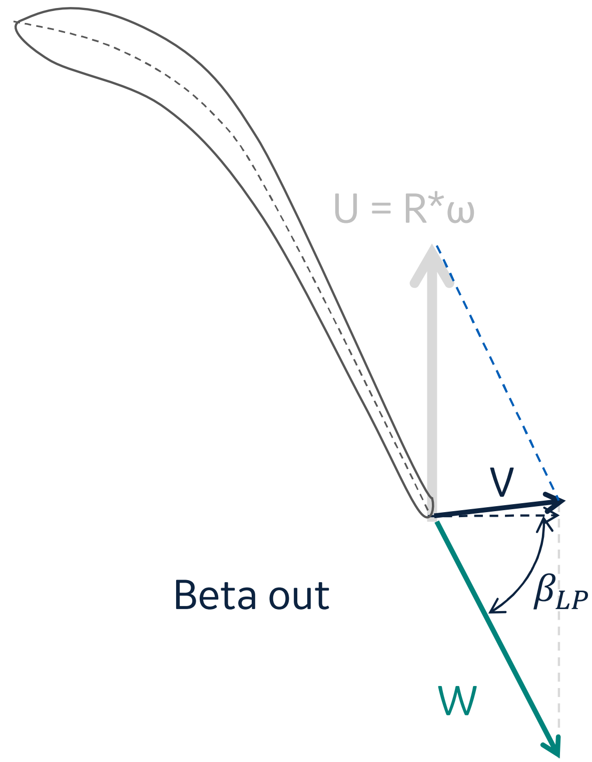


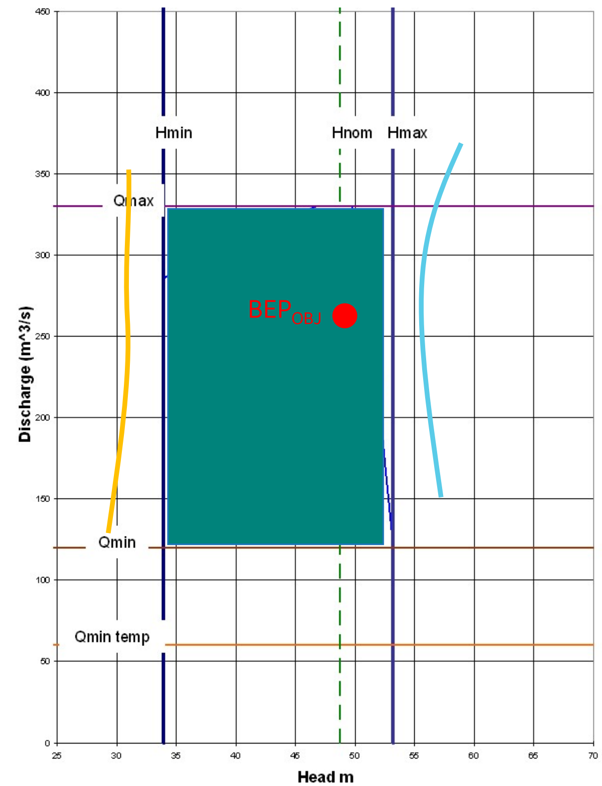


How to design a Francis runner ?

Discharge adaptation

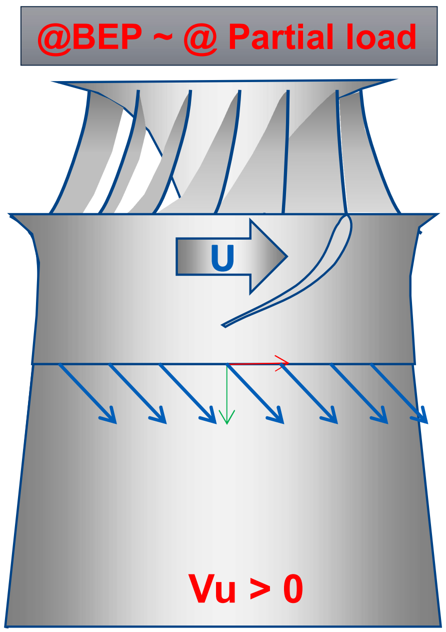


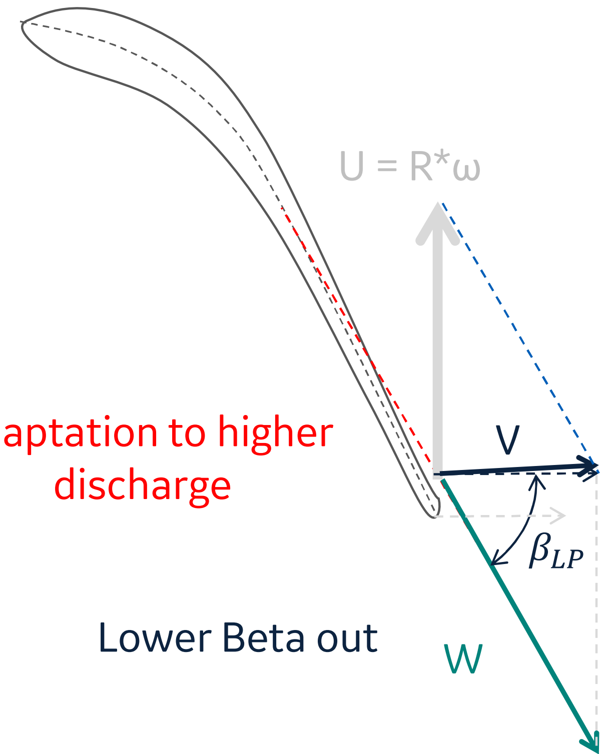




How to design a Francis runner ?

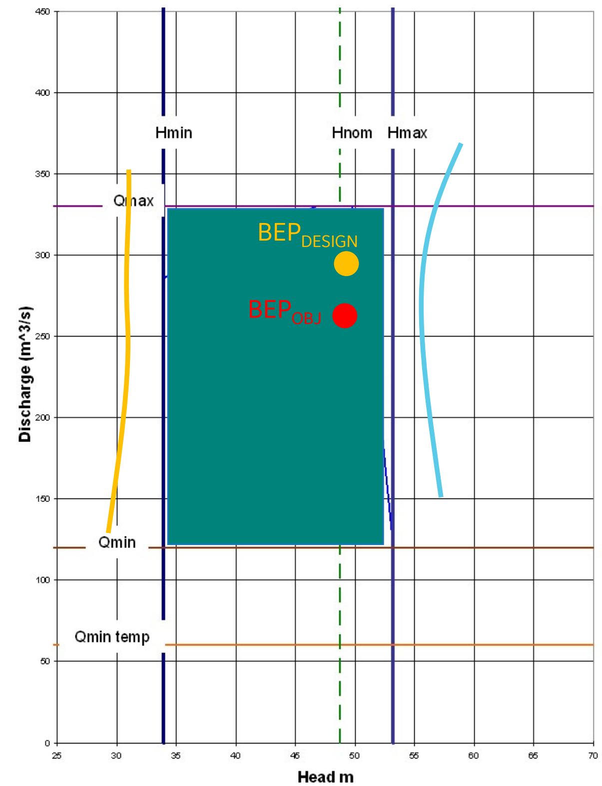
Discharge adaptation





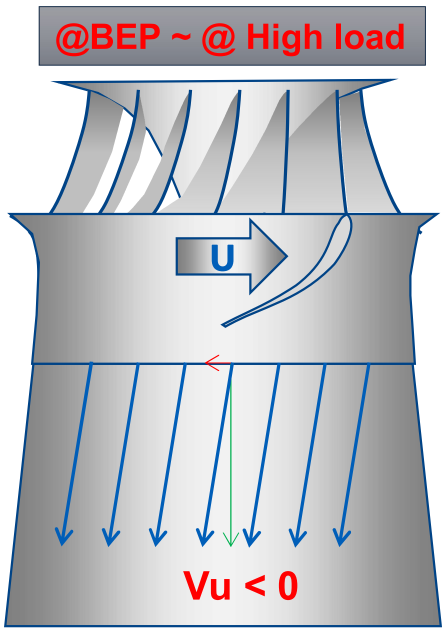
Adaptation to higher discharge

Lower Beta out

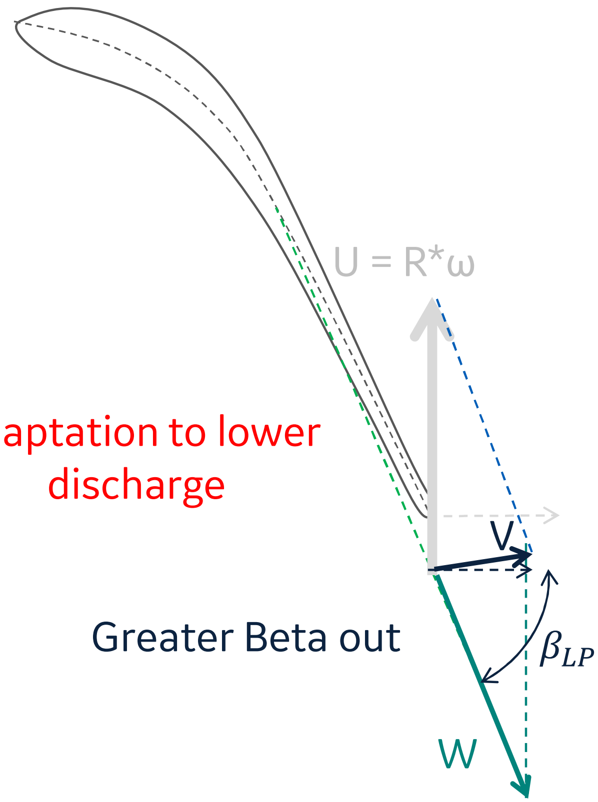


How to design a Francis runner ?

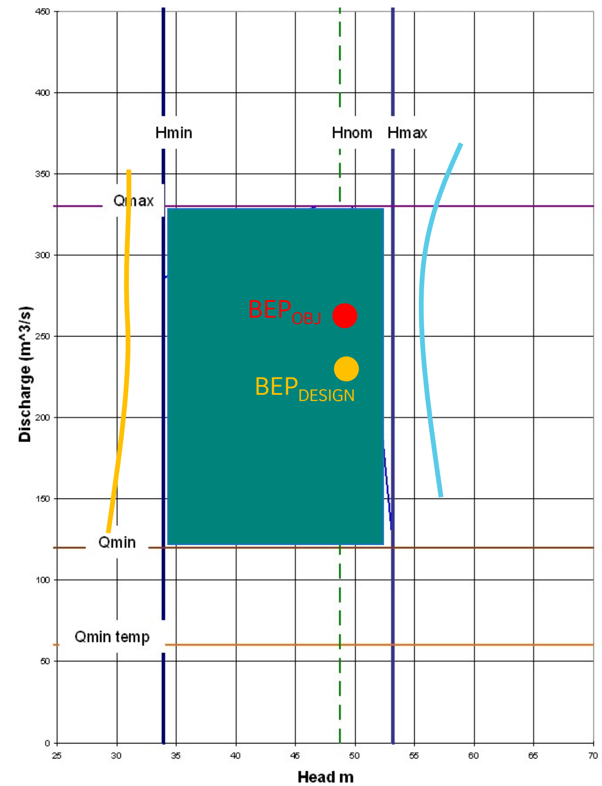
Discharge adaptation



Adaptation to lower discharge



Greater Beta out



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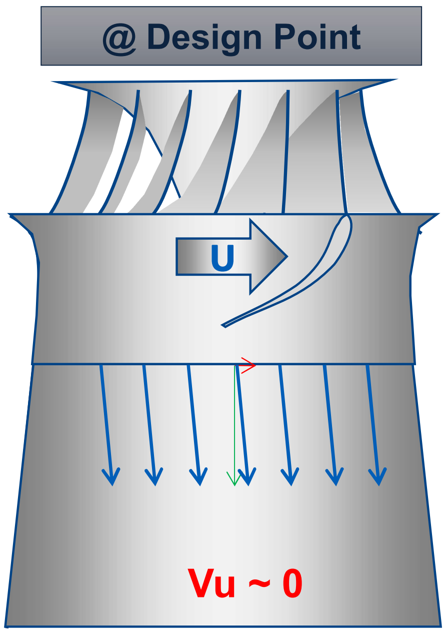
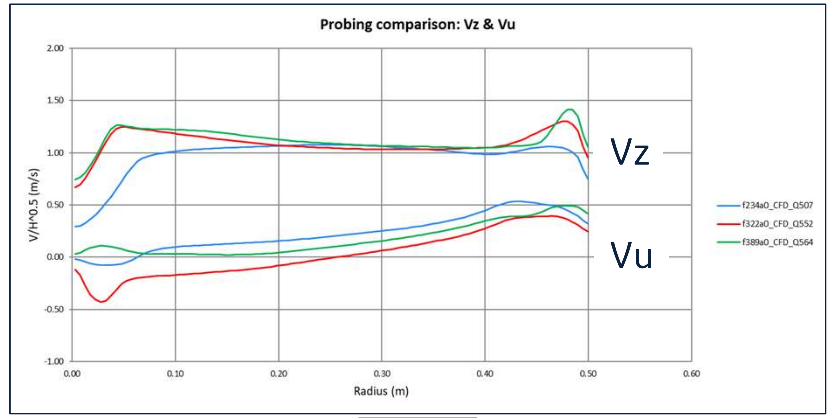
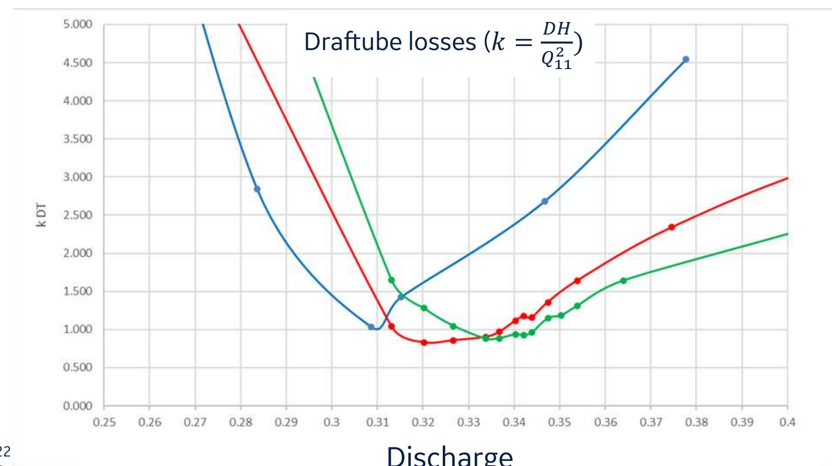
How to design a Francis runner ?

Discharge adaptation, one step further : How to be sure that the DT losses are minimum ?



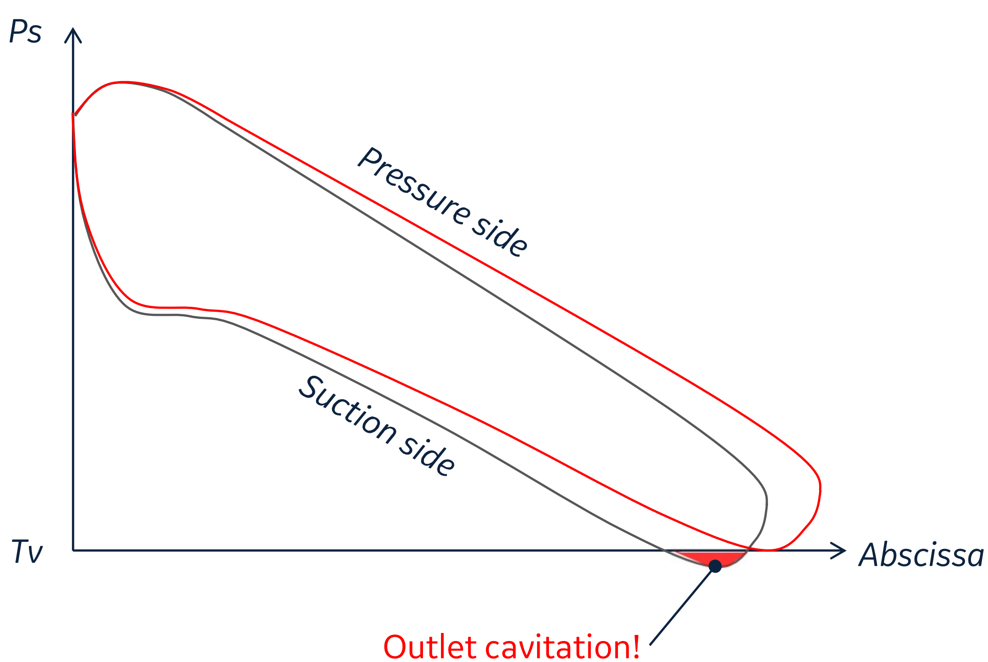


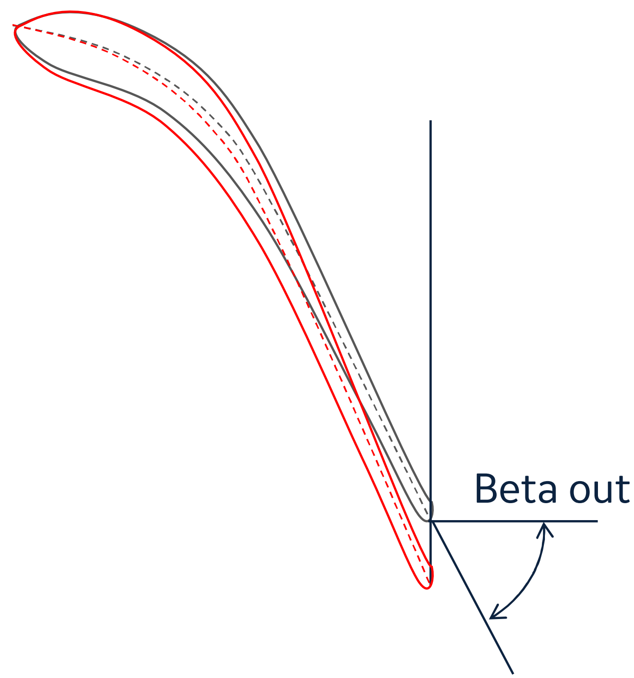


For different probing shape, losses in DT (and η) can change a lot

Runner Design: Outlet Cavitation

Drivers:
- local curvature
- blade length
- number of blades





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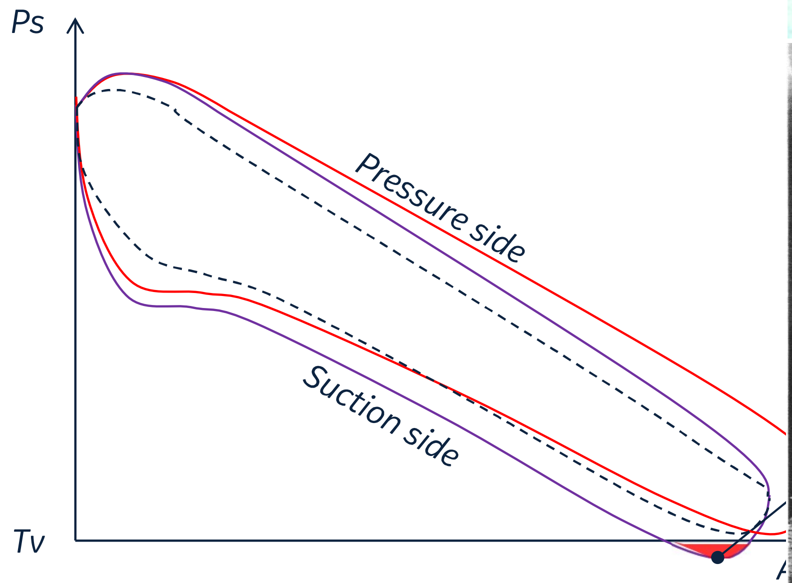
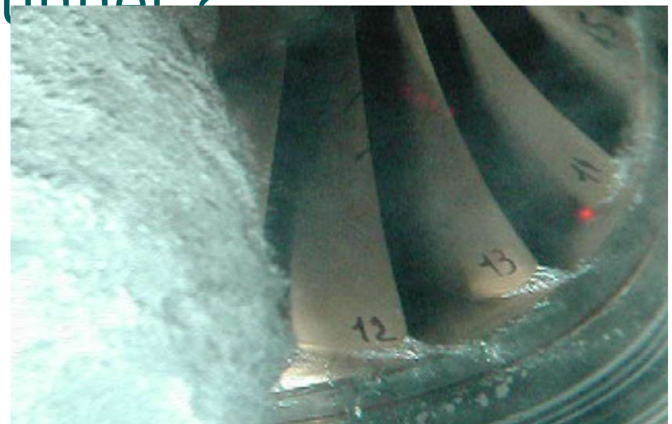
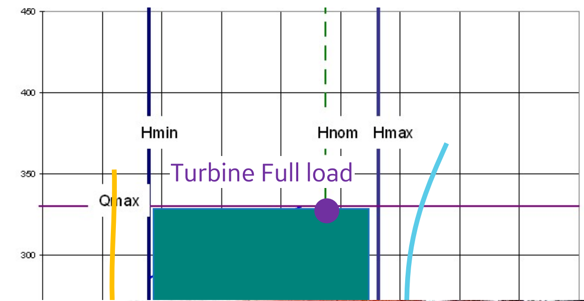
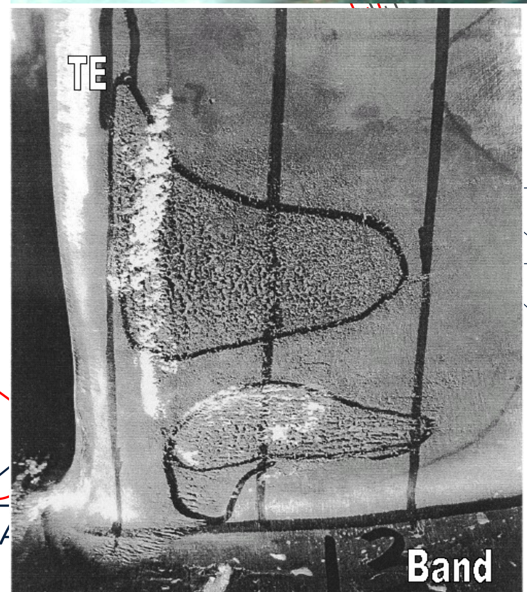
Outlet Cavitation limits

Drivers:
- local curvature
- blade length
- number of blades









Runner Design: Conclusion

Objectives:
1. Minimize head losses to maximize efficiency (productivity)
→ Short blades, low number or blades

2. Cavitation free in the whole domain, and especially to maximum discharge
→ Long blades, high number of blades

3. Cheap solution
→ Small runner… against 1 & 2

→ **OPTIMIZATION**

Note: In tendering (bidding before contract), the “natural” tendency is to sell the *best* efficiency with the *cheapest* supplies… then during execution stage this optimization might be difficult

Real Velocity Profiles (from Test Rig)

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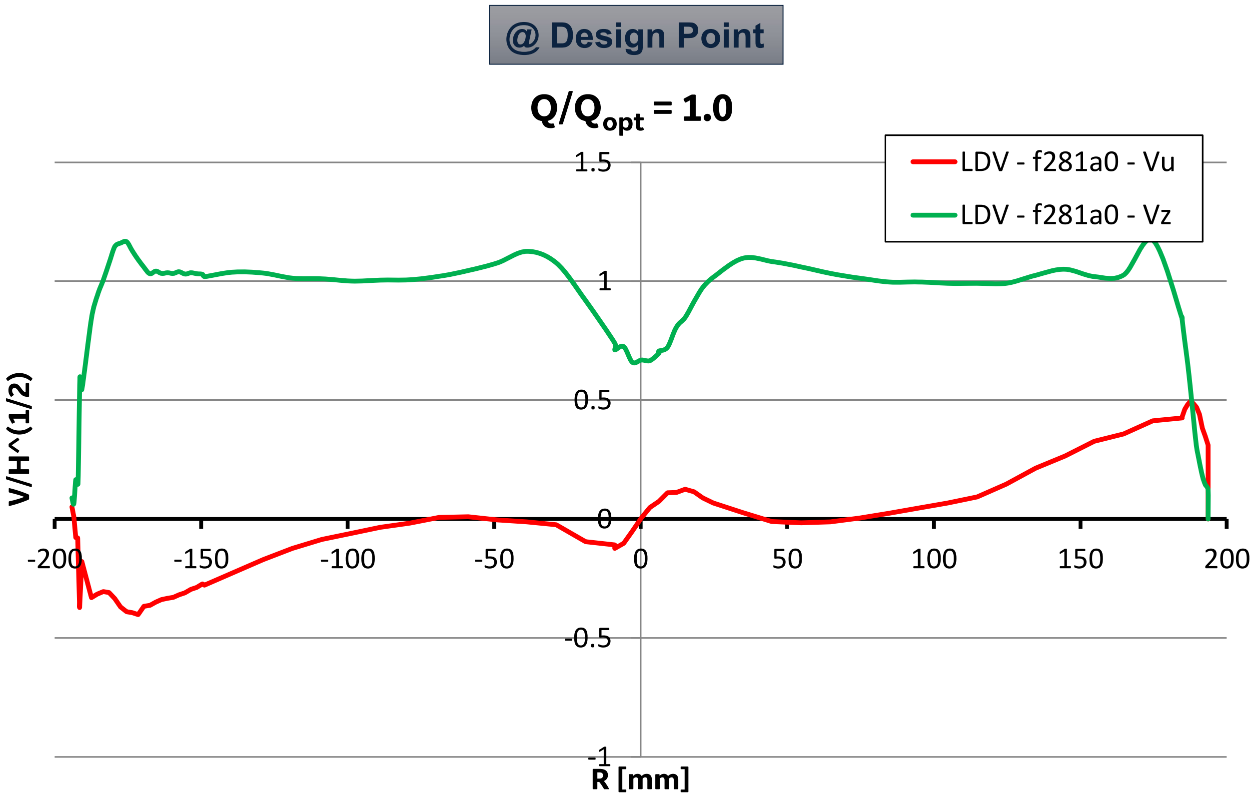
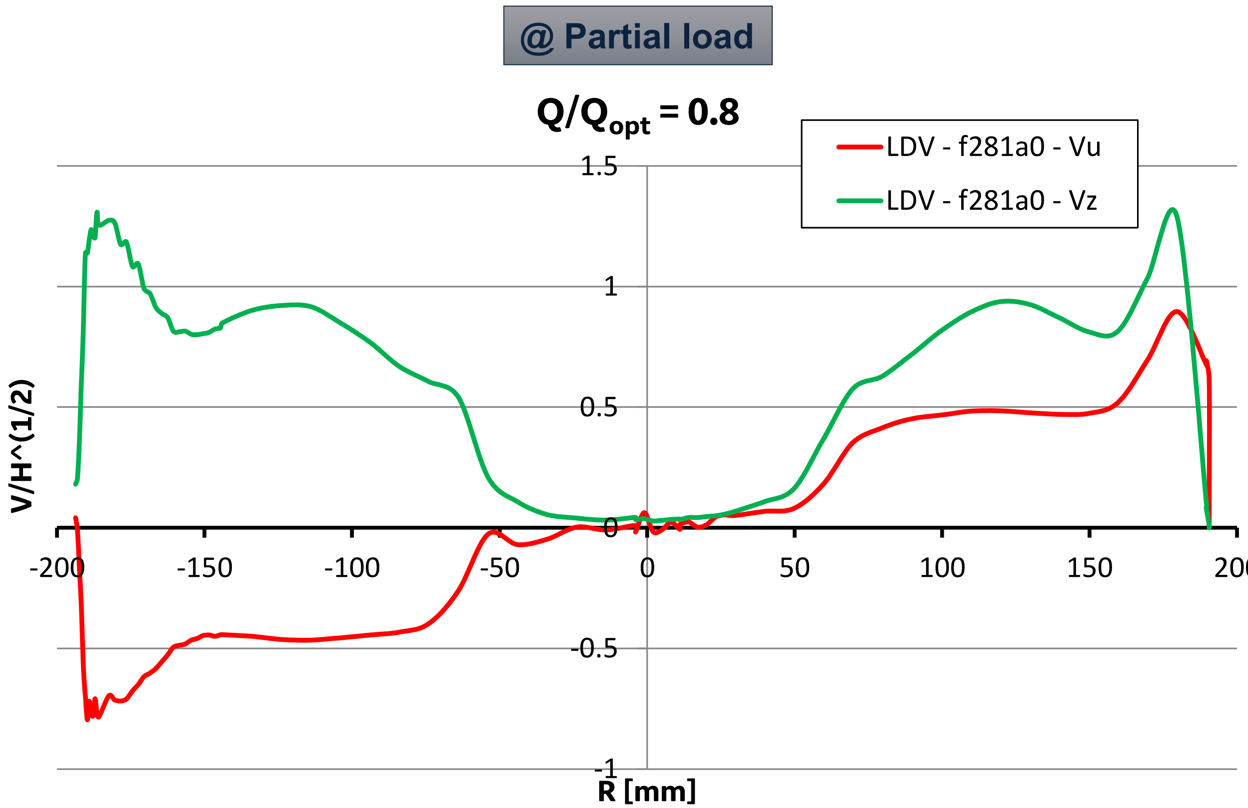
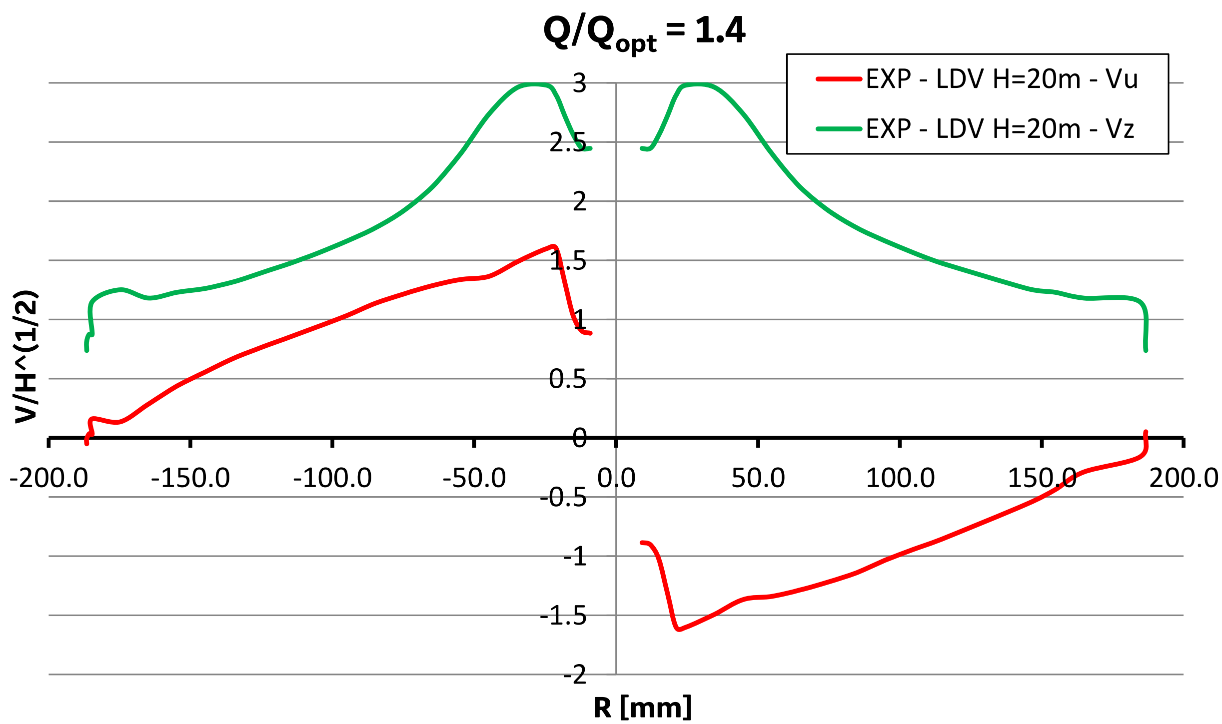


Real Velocity Profiles (from Test Rig)

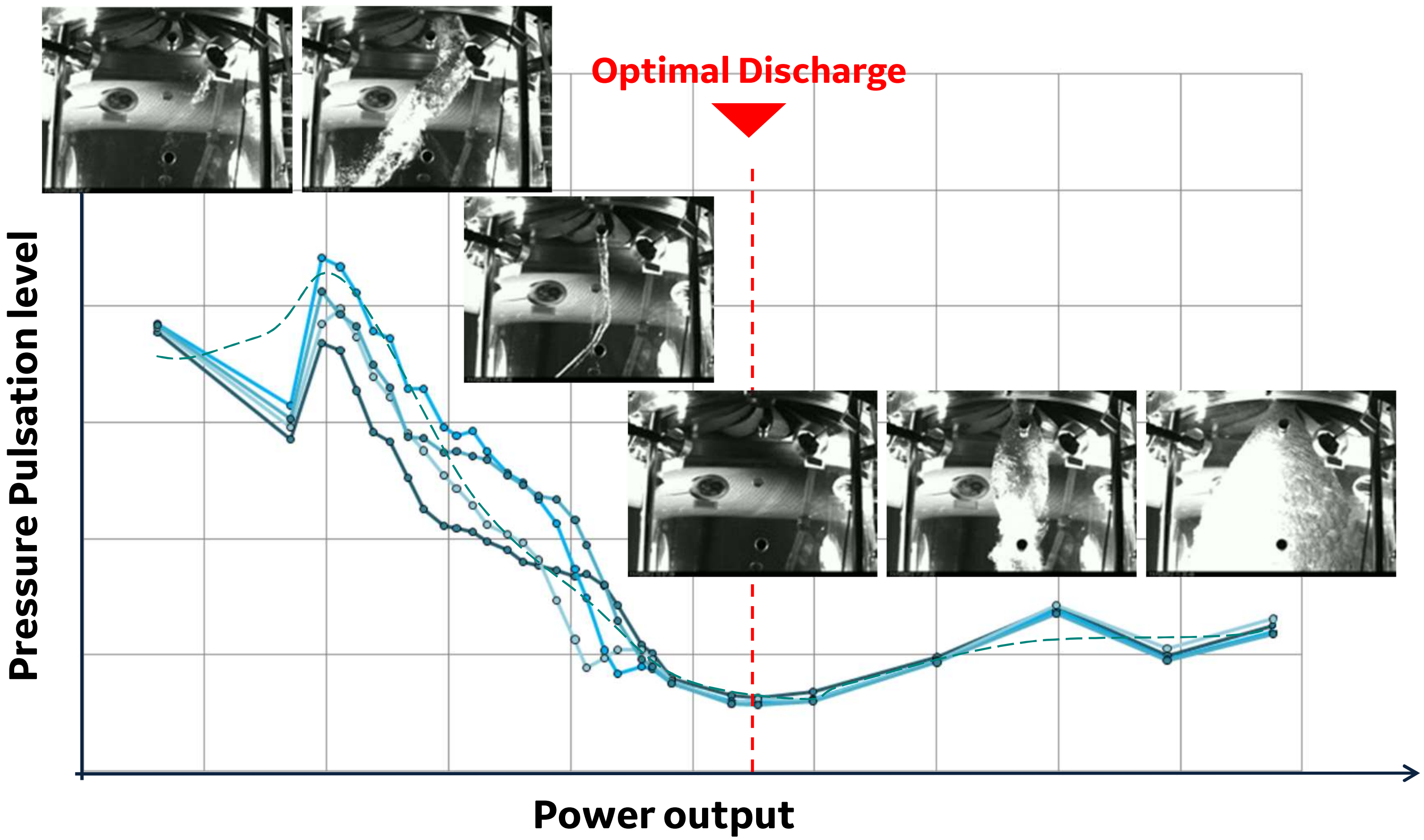


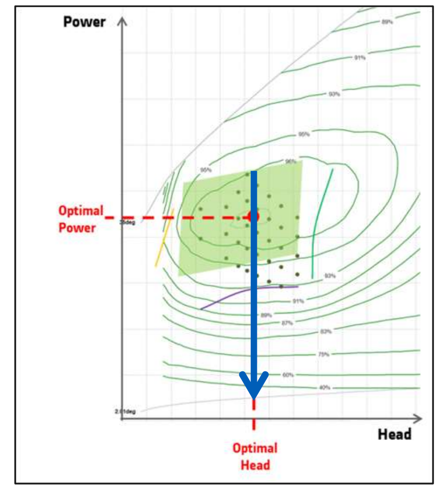
Real Velocity Profiles (from Test Rig)

@ High load

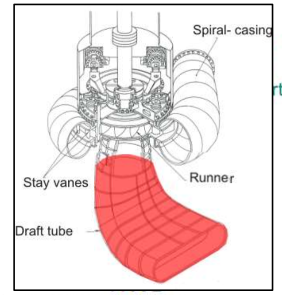


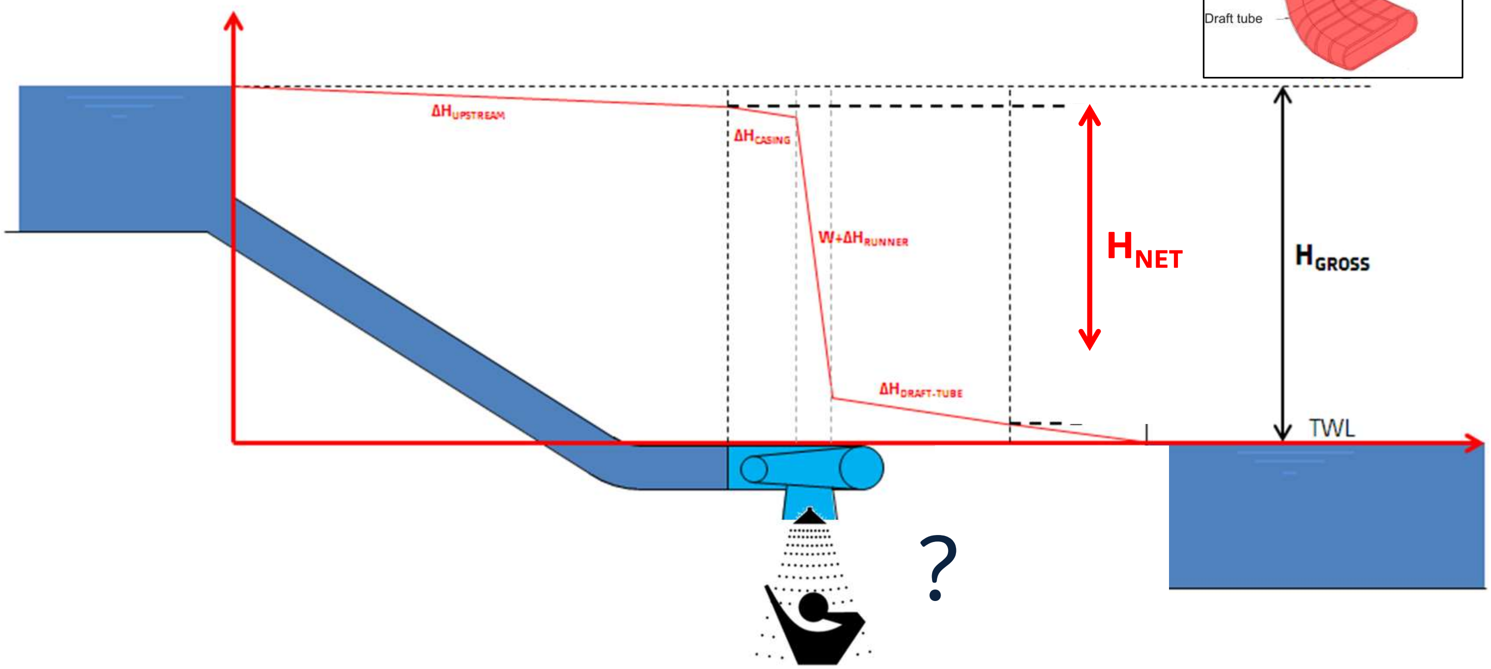
Flow behaviour at Runner outlet





Draft Tube what's is it for?

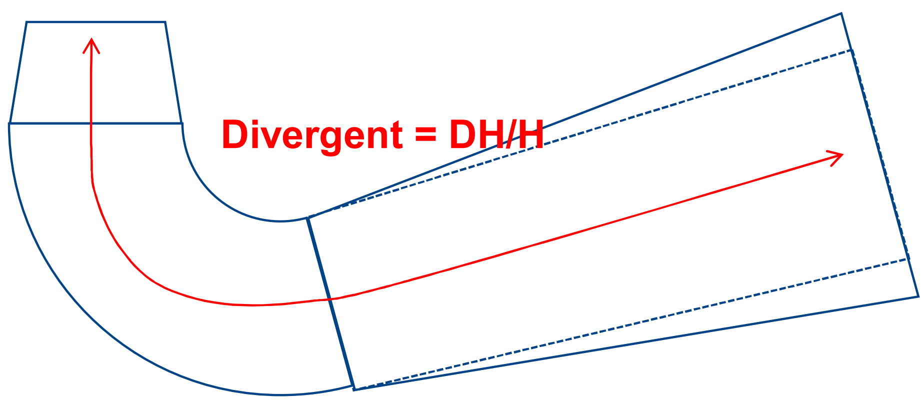




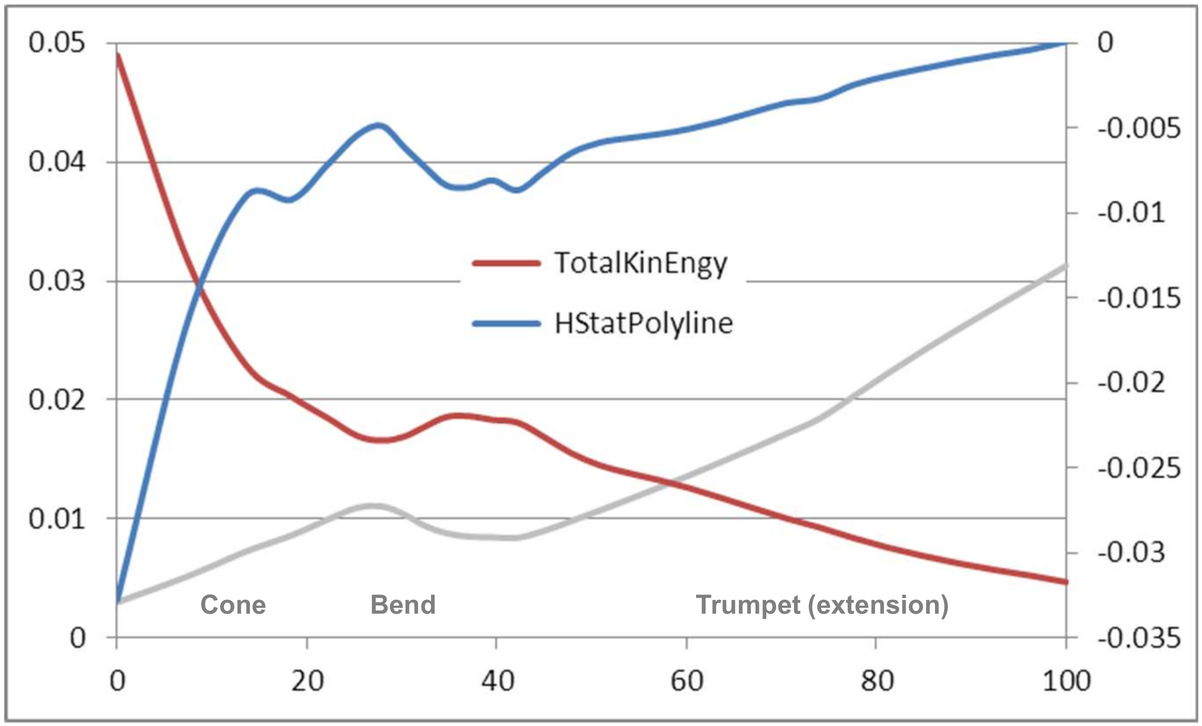
- **Net Head** Hn $= Hb - \Delta H(A \rightarrow 1) - \Delta H(2 \rightarrow B)$
  $= Hb - kV_1^2/(2g) - \mathbf{V_2}^2/(2g)$

Draft Tube helps increasing the net head

- This is why Customers usually impose a **maximum flow speed** at draft tube outlet – otherwise the turbine supplier would tend to minimize the outlet section in order to increase its efficiency

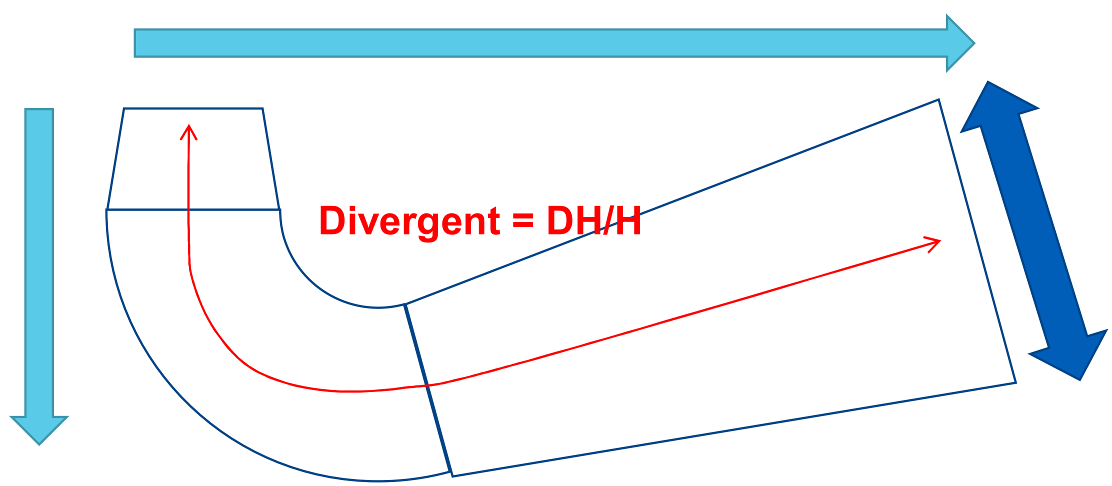


Kinetic energy / static pressure evolution along the draft tube midline curvature



What's a good Draft Tube like?

- To be the most divergent (Customer view point)
- To have the smallest head loss (Turbine supplier view)
- To be the most compact possible
  (Customer / Civil Work / Turbine supplier view....)



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Conclusion on Hydraulic Design

Conclusion

- Design process can be difficult and **iterative** if specific requirements (e.g. high head and/or discharge variation)
- High number or **degrees of freedom**
  → makes the process difficult to automate (optimal design)
- Different levels of **trade-off** required at tendering and execution stages
  - Mechanical vs Hydraulic
  - Efficiency vs Cavitation
  - Efficiency vs Cost
- High added value of **experimented** hydraulic designers
- **No “off the shelf”** product: for each project, a specific design

Elements to consider during a hydraulic design

- **Efficiency** (WAE) / Output (different heads)
  → Checked on Model (step-up as per IEC 60193 / 62097) or Prototype (60041)
- **Cavitation** (erosion pitting as per guarantees, IEC 60609)
  → Cavitation margin (sigmaP/sigma-1 >…) on model test
  → Eroded volume on proto after 8000 hours (for ex.)
- **Pressure Pulsations** (limits set by customer)
  → Pressure pulsation level in Casing / Distributor / Draft tube… on model test
  → Pressure pulsation level on proto… but rather vibration (shaft line) and torque fluctuation (at generator outlet)
- **Runaway** (limit can be critical in case of refurbishment!)
  → Checked on model test… usually not on prototype!
- **Torque on Guide Vanes**
  → Checked on model test, dimensioning with margin on proto

CFD vs Experimental: **Pressure Pulsations**

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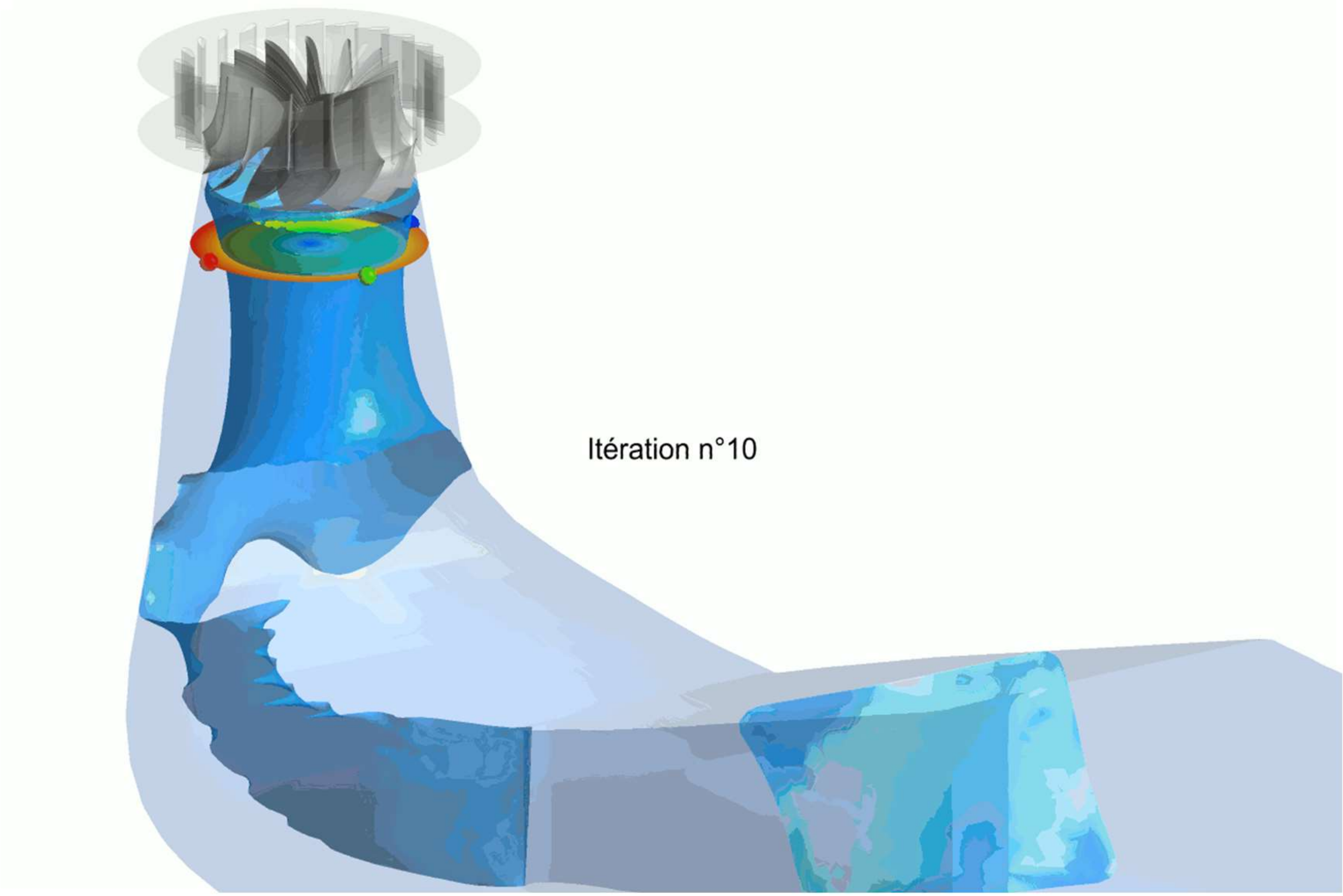


On the mechanical side

Mechanical impacts on hydraulic design

- Hydraulic profiles must be compatible with mechanical stresses
- Max allowable stresses will depend on the material (carbon steel, stainless steel –different types–) and on the manufacturing type
- <u>What kind of stresses?</u>

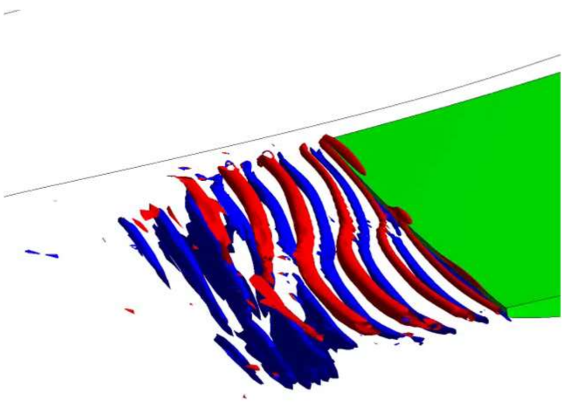
→ **Static head** (max when unit is stopped): Casing, Stay Vanes, Guide Vanes

→ **Transient head** (during starts & stops, in particular emergency stops): "H+deltaH" → May affect the whole turbine

→ **Centrifugal forces**: Runner at runaway

→ **Fatigue phenomena**:

→ Von Karman vortices (all profiles)
→ Part load Vortex Rope (runner + draft tube)
→ Rotor Stator Interaction (distributor + runner)



Mechanical impacts on hydraulic design

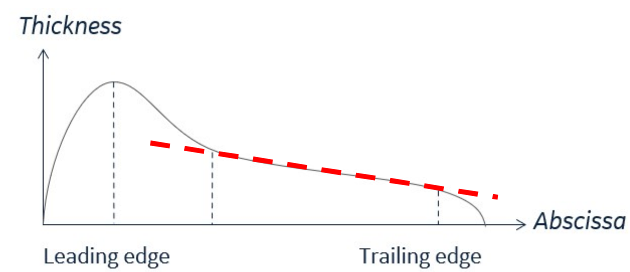
- Spiral case → no blades, limited impact on hydraulic design
- Stay vanes → meant for mechanical purpose, **inertia** is needed
- Guide vanes → need for hydraulic tolerance, but also **inertia** (especially H+deltaH when complex set-up with several units on the same penstock)
- Runner → need to stand Hmax, H+deltaH, runaway (centrifugal force), **fatigue** linked to Inter Blade Vortices, Part Load Vortex Rope
- Draft tube → it “does see the head”, but still… elements that are not embedded may suffer **fatigue**, especially because of Part Load Vortex Rope

Manufacturing considerations

Manufacturing types for the runner

- **Welded fabricated** → Band and Crown come in forged pieces, Blades are casted, a welding in needed on each side of the blade

→ Blades quality might not be optimal especially in the low thickness areas. A specific thickness ***slope*** is required to grant a good quality during the cooling down.



→ Welded zone is the filet radius (blade vs crown or band), which is precisely the area of stress concentration, some ***residual stresses*** are to be considered: max allowable stress is lower compared with other manufacturing types

→ Classical for big units (some even have to be welded at site!)

+ Good finish quality (surface finish)

– Expensive, lower admissible stress

Runner manufacturing
Welded Fabricated with separate elements

Element pre-machining

Element assembling

Pre-heating & welding

Final machining

Runner manufacturing
Welded Fabricated with separate elements

Runner at site
in one piece

Or manufactured closed to the site. Ex. Three Gorges runner

Manufacturing types for the runner

- **Casted Monobloc →** One unique cast for the whole runner

→ Typically used for small units, where no access in between blades for welding… but is has the same problem as the blade casting (average quality)

→ AND the surface finish can hardly be as good all over the hydraulic shapes

+ Cheap and quick to manufacture

– Poor quality (shape and surface)

Manufacturing types for the runner

- **Forged blocks →** Runner machined in one or several blocks

→ Applicable for medium runner where the surface is fully reachable for machining

→ Bi-blocks can be a solution when a welding is possible in the middle of the blades

+ Excellent material quality (forging)
+ 3D shape very well respected
+ No residual stress in the fillet radius

– Expensive, limited in size

Runner manufacturing
Bi-block with welding in the middle of the blades

- **1650mm diam.**
- **ns ~ 260 HP**

Runner manufacturing
Bi-block with welding in the middle of the blades

- **2700mm diam.**
- **ns ~ 110 HP**

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