L8 – Industrial Pumps

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Topics of the lecture

- Centrifugal pump types
- Operating principles and issues
- Pump selection criteria

History

First pumping devices date back to 4000 BC: mainly volumetric pumps

- First centrifugal pump proposed by Papin 1705
- First multistage pump patented by Gwynne 1851
- Improved by Reynolds 1875 who designed guide vanes and return vanes

EPFL

Centrifugal Pumps

Centrifugal pumps are widely used to move water from one location to another or circulate water within a circuit. They are also used to provide high-pressure jetting of water or other liquids or chemicals.

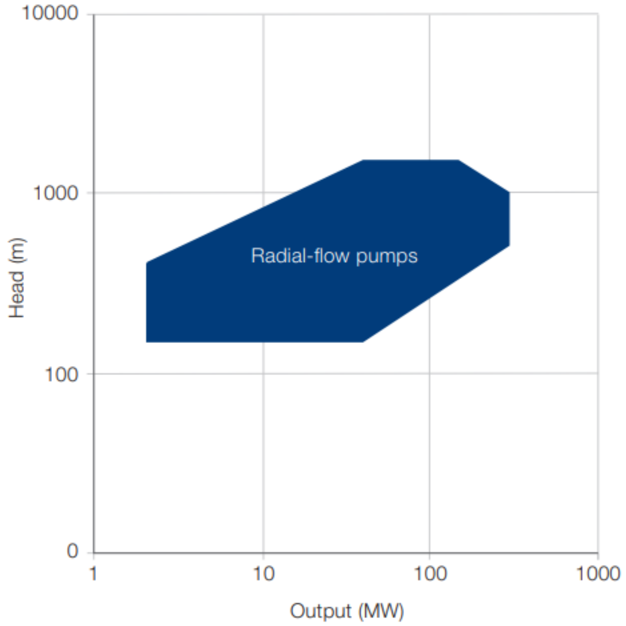
Classification	
Flow orientation: - Radial - Mixed-flow - Axial	**Number of stages:** - Single stage - Two stage - Multistage

Pumps types are very numerous and with specific characteristics depending on the use, the liquid or multiphase mixing, temperature and the required energy.

Centrifugal Pumps

Radial flow pumps as Storage pumps

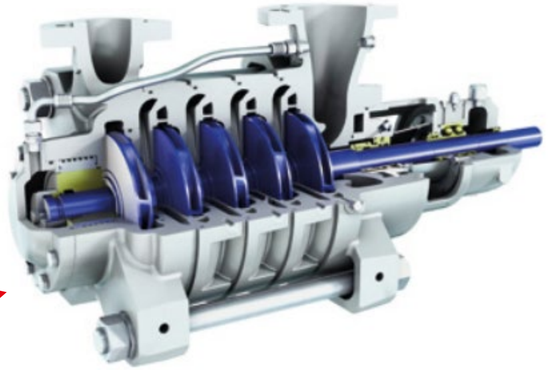
Pumps for storage applications are mainly of the radial-flow type. Depending on the application conditions the construction, can be a single or double-flow, single- and multi-stages

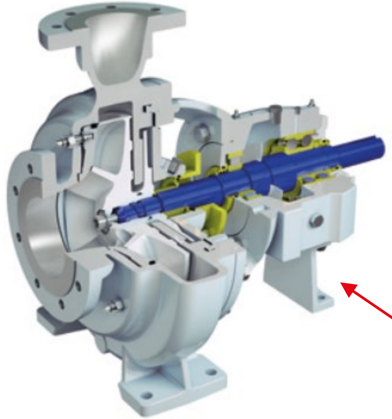


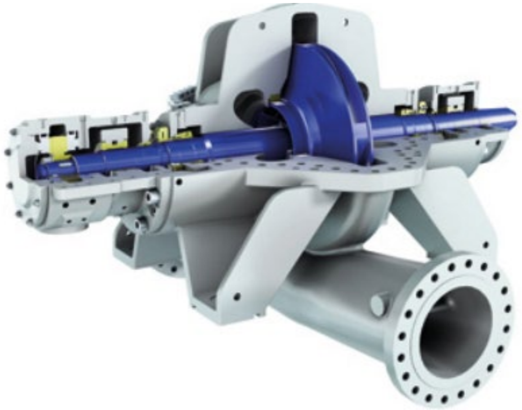
VOITH
Engineered Reliability

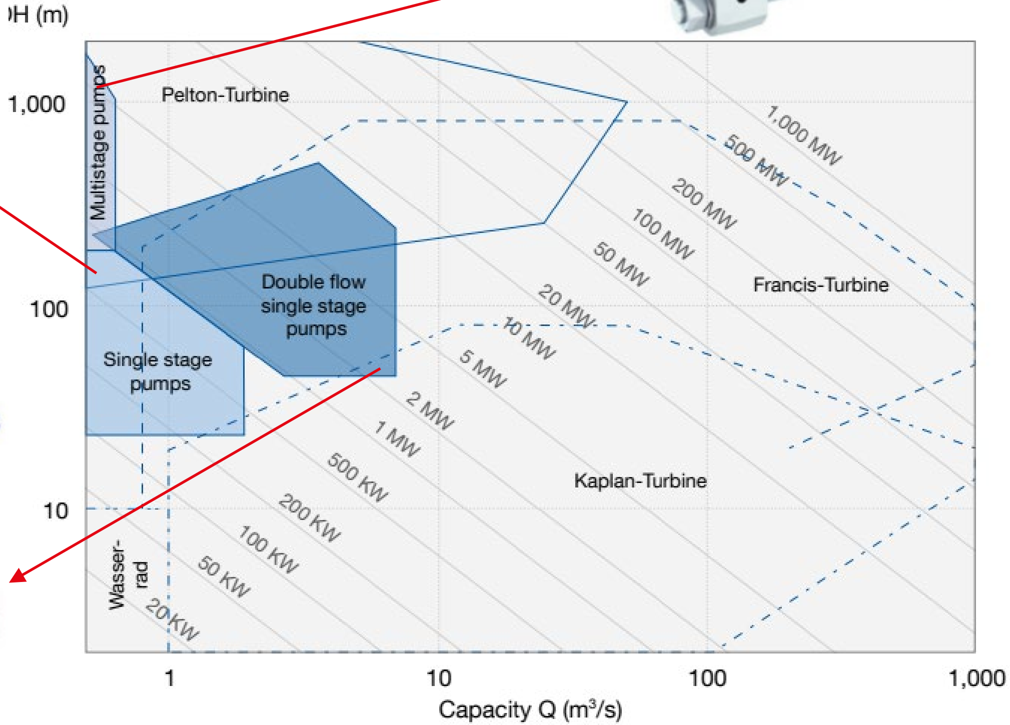
Centrifugal Pumps

Radial flow pumps as Storage pumps





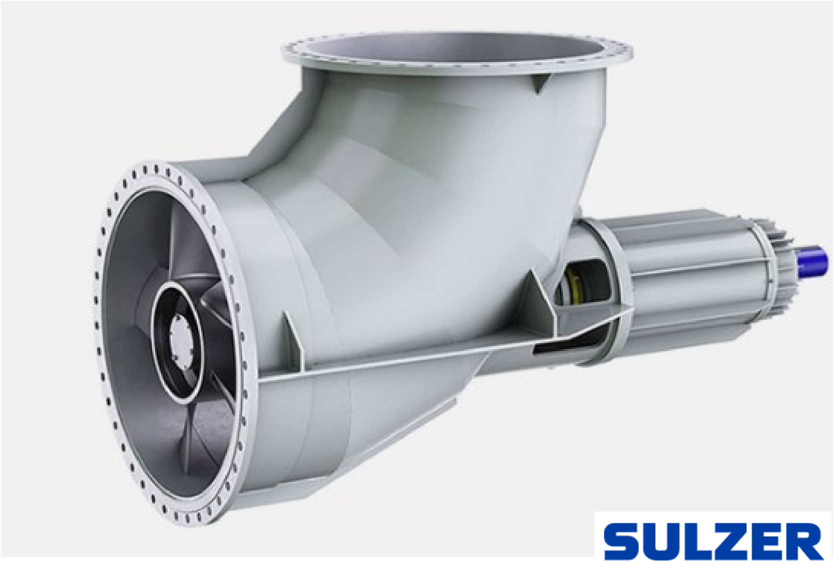




Centrifugal Pumps

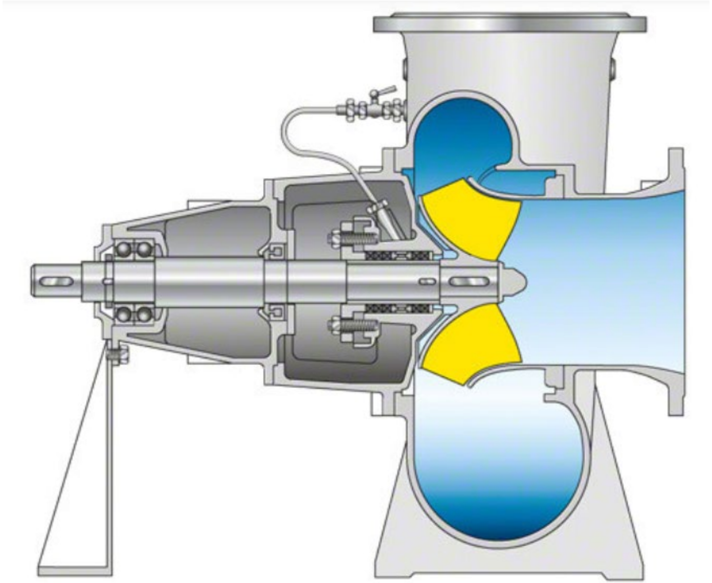
Mixed-flow and axial flow pumps

Axial flow pumps



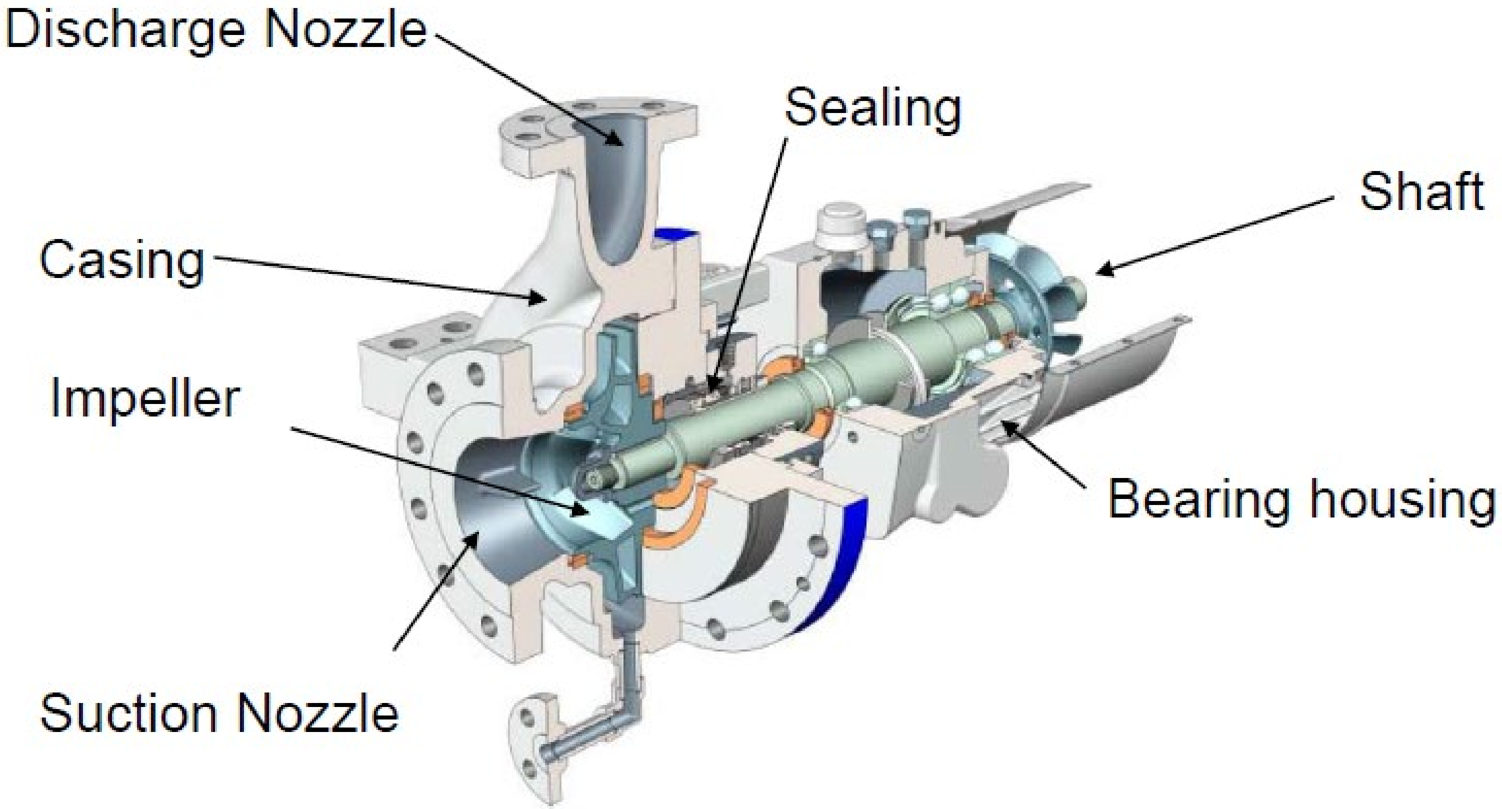
Low head pumping applications, typically up to 15m

Mixed-flow pumps

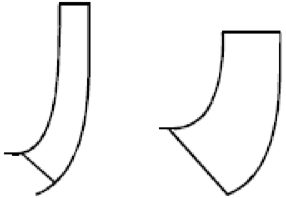
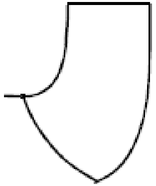
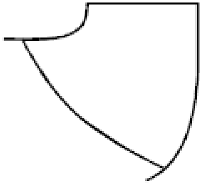


Small- Medium head pumping applications, typically up to 60m

Main Components

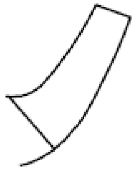
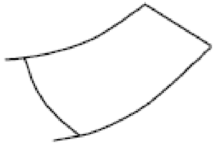
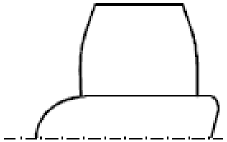


Impeller Characteristics

n_q	Type	Impeller shape	$H_{st,opt}$	ψ_{opt}	η_{opt} [%]	
7÷30			800 m (1200)	1 ÷ 1,2	40 ÷ 88	Below $n_q < 10$ Only for pumps with small dimensions
50	Radial pumps		400 m	0,9	70 ÷ 92	In most cases $H_{st,opt} < 250$ m
100			60 m	0,65	60 ÷ 88	$n_q = 100$ is about the highest limit for radial pumps

$$n_q = \frac{n \cdot \sqrt{Q}}{\left(\frac{H}{z_s}\right)^{3/4}}$$

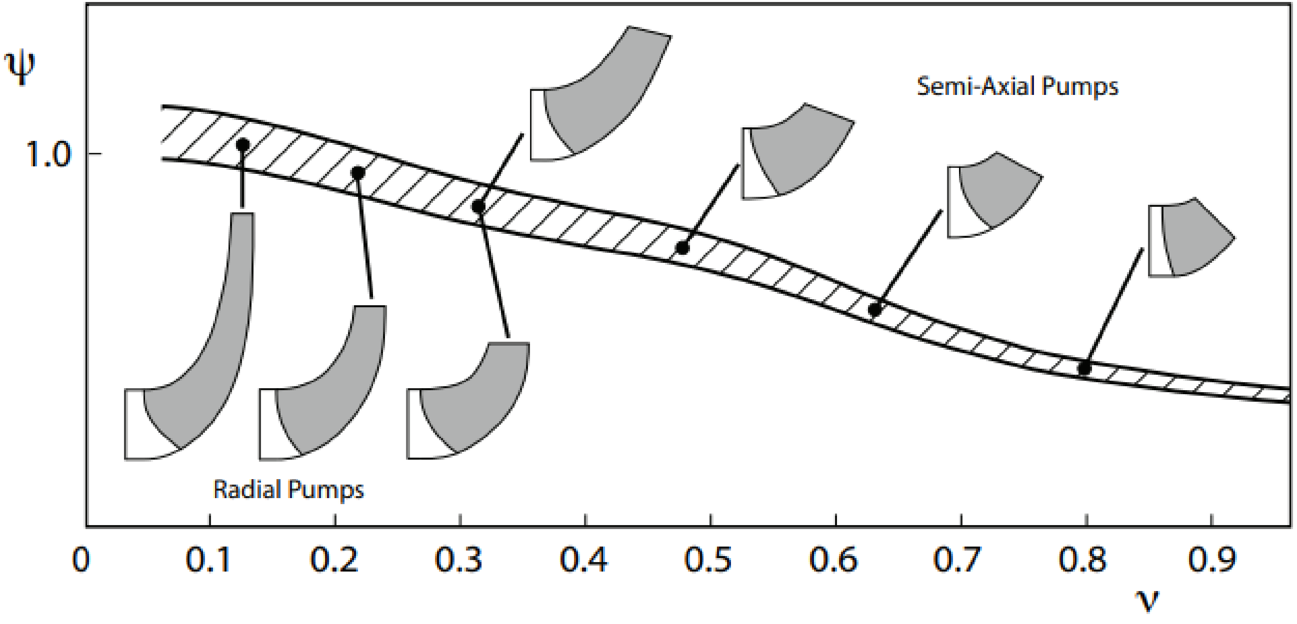
Impeller Characteristics

n_q	Type	Impeller shape	$H_{st,opt}$	ψ_{opt}	η_{opt} [%]	
35	Semi-axial pumps		100 m	1	70 ÷ 90	For $n_q < 50$ Almost always multistage above $n_q > 75$ rarely multistage
160			20 m	0,4	75 ÷ 90	for $n_q > 100$ only single stage
160 to 400	Axial pumps		2 to 15 m	0,4 to 0,1	70 ÷ 88	Flow rates up to 60 m^3/s only single stage

$$n_q = \frac{n \cdot \sqrt{Q}}{\left(\frac{H}{z_s}\right)^{3/4}}$$

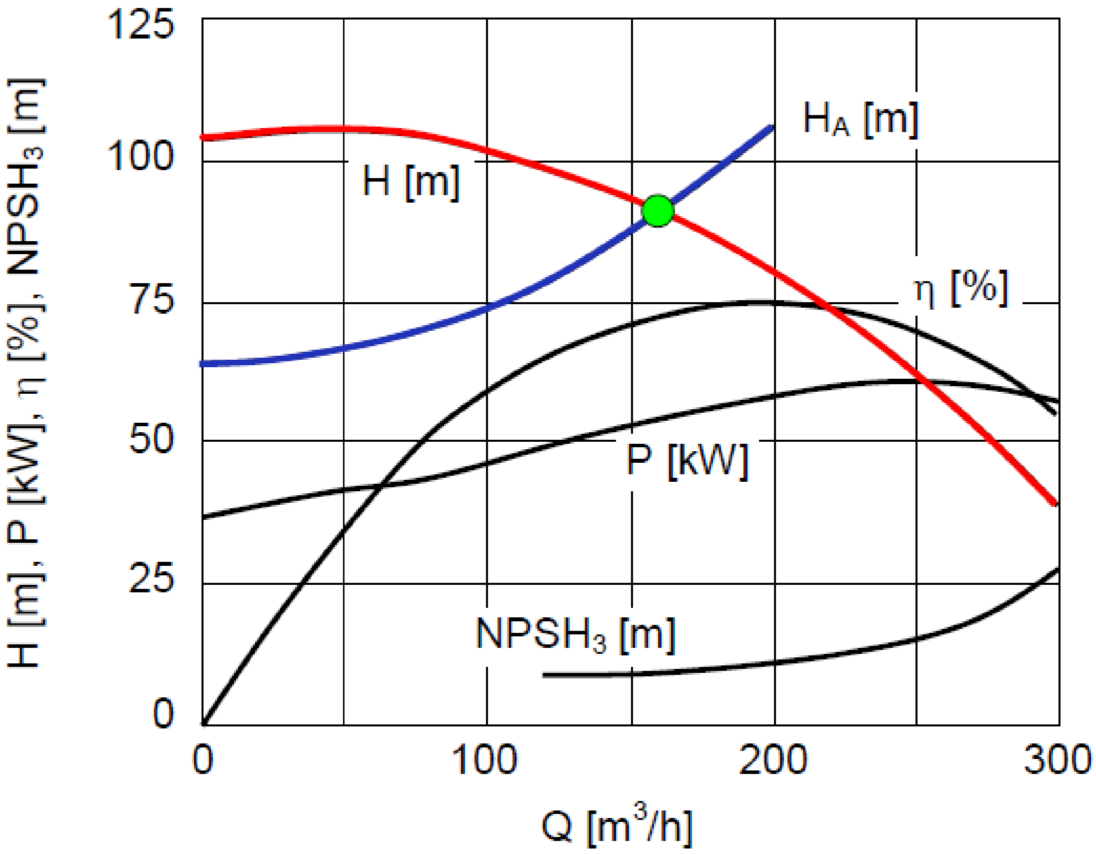
Characteristics of a pump

$$\psi = \frac{E}{\frac{U^2}{2}}$$



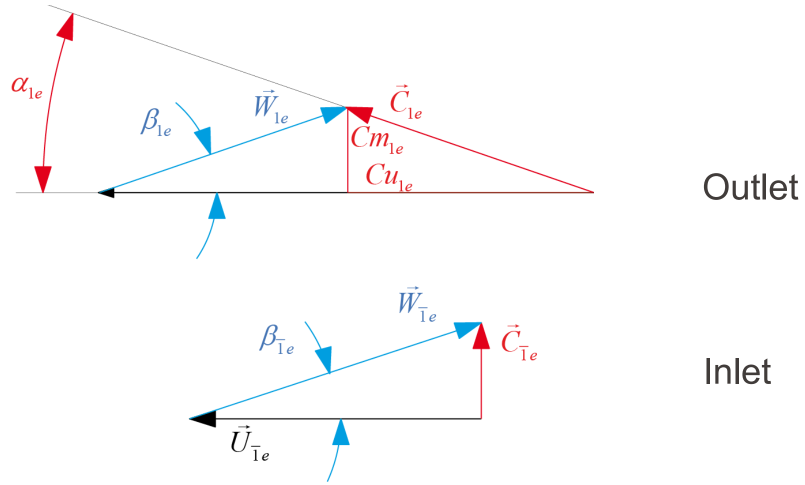
$$\nu = \frac{\varphi^{\frac{1}{2}}}{\psi^{\frac{3}{4}}} = \frac{\omega}{\pi^{\frac{1}{2}} 2^{\frac{3}{4}}} \times \frac{|Q|^{\frac{1}{2}}}{E^{\frac{3}{4}}}$$

Characteristics of a pump



From L3: Pump Impeller Characteristic curve

Hypothesis: Swirl Free Inlet Flow→ axial flow at the inlet section

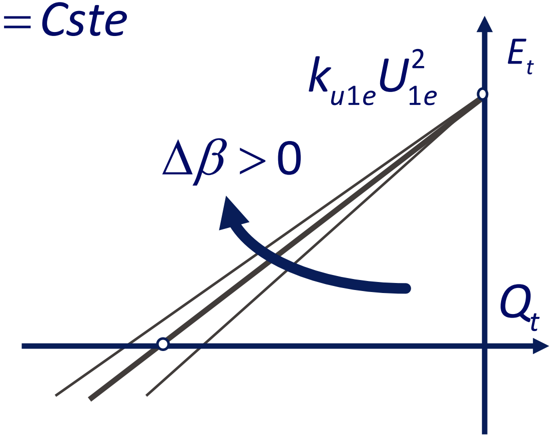


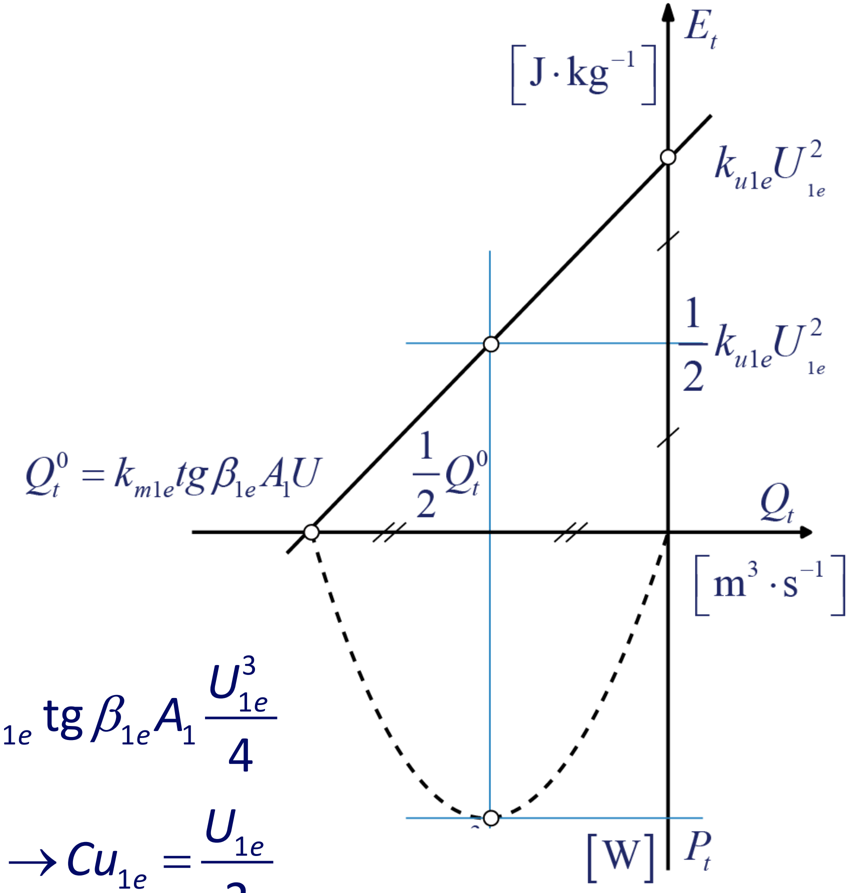
From Euler equation:

$$E_t = k_{u1e} U_{1e}^2 - \left[k_{u1e} \frac{1}{\mathrm{tg}\,\beta_{1e}} + \underbrace{\cancel{k_{u\bar{1}e} \frac{R_{\bar{1}e}}{R_{1e}} \frac{A_1}{A_{\bar{1}e}} \frac{1}{\mathrm{tg}\,\alpha_{\bar{1}e}}}}_{=0 \text{ If } \alpha_{\bar{1}e} = \frac{\pi}{2}} \right] U_{1e} \frac{Q_t}{A_1}$$

From L3: Pump Impeller Characteristic curve Swirl Free Inlet Flow

$\omega = Cste$

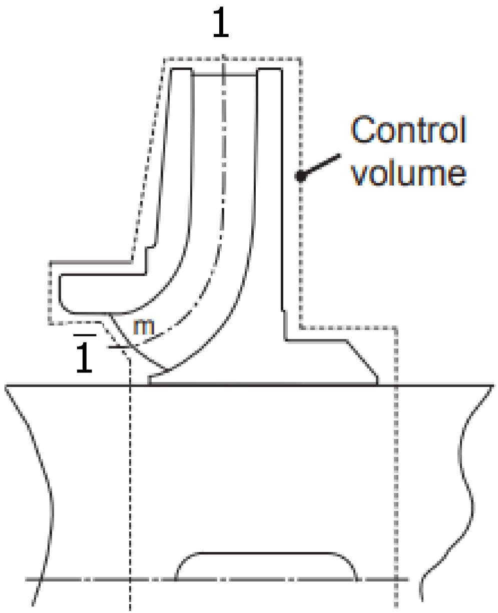


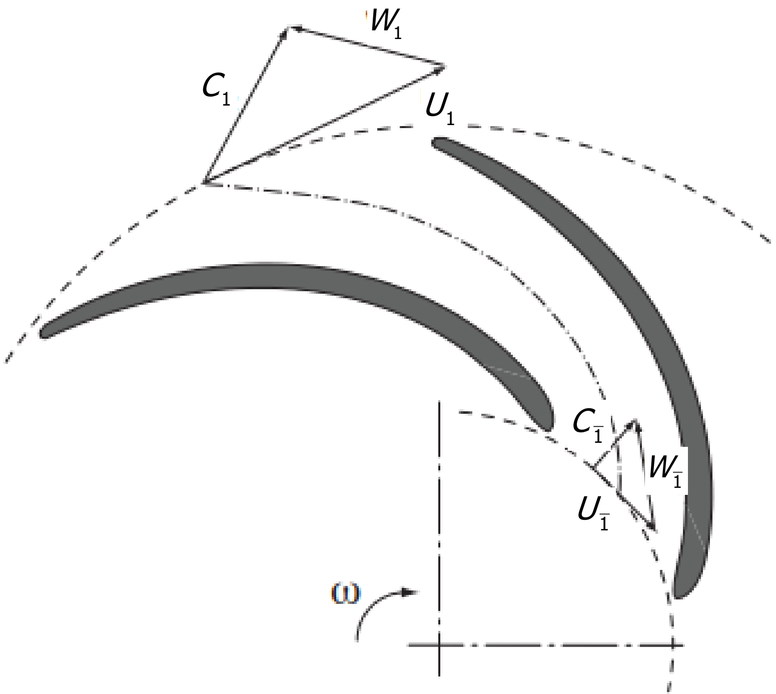


Optimal discharge condition: $P_t = P_t^{\min} = \rho k_{u1e}\, \mathrm{tg}\, \beta_{1e} A_1 \dfrac{U_{1e}^3}{4}$

$$E_t^{\min} = k_{u1e}\frac{U_{1e}^2}{2} \rightarrow Cu_{1e} = \frac{U_{1e}}{2}$$

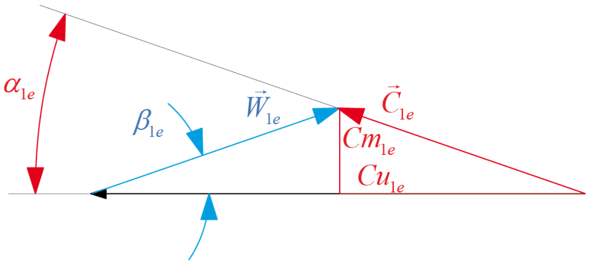
Pump Impeller Characteristic curve

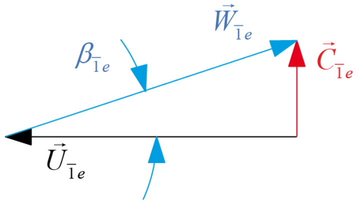




Radially uniform outflow: $E_t = U_{1e}^2 - \left[\frac{1}{\mathrm{tg}\,\beta_{1e}}\right] U_{1e} \frac{Q_t}{A_1}$

Pump Impeller Characteristic curve





$$E_t = Cu_{1e}U_{1e} = U_{1e}\left[U_{1e} - \frac{Cm_{1e}}{\operatorname{tg}\beta_{1e}}\right]$$

Ideally, no-slip condition: $\beta_1 = \beta_{b1}$
In practice, $\beta_1 < \beta_{b1}$ because we have a finite number of blades and it reduces the impeller efficiency

$$Cu_{1e,no-slip} - Cu_{1e} = (1-\mu)U_{1e}$$

Slip factor or deviation angle μ due to:

- Velocity differences between blade suction and pressure surface
- Coriolis force creates secondary flow from suction to pressure side
- The outlet pressure profile (which impose the same pressure at the pressure and suction side at the trailing edge of the blade) affects flow the distribution towards the trailing edge of the blade of the impeller

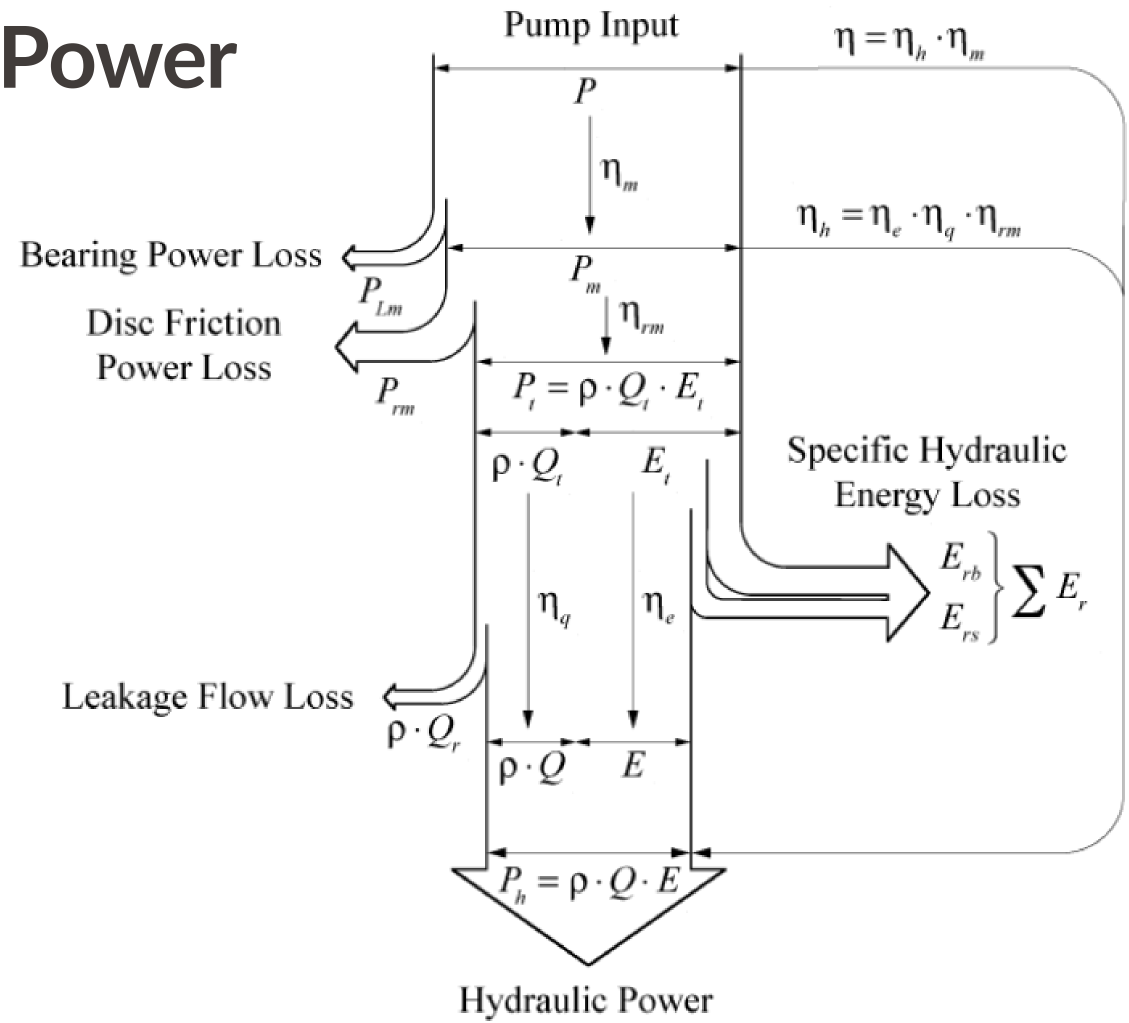
From L2: Pump Power Balance

- Supplied Energy

$$E_t = \frac{P_t}{\rho Q_t}$$

- Resisting Torque

$$P_t = \vec{\omega} \cdot \vec{T}_t$$

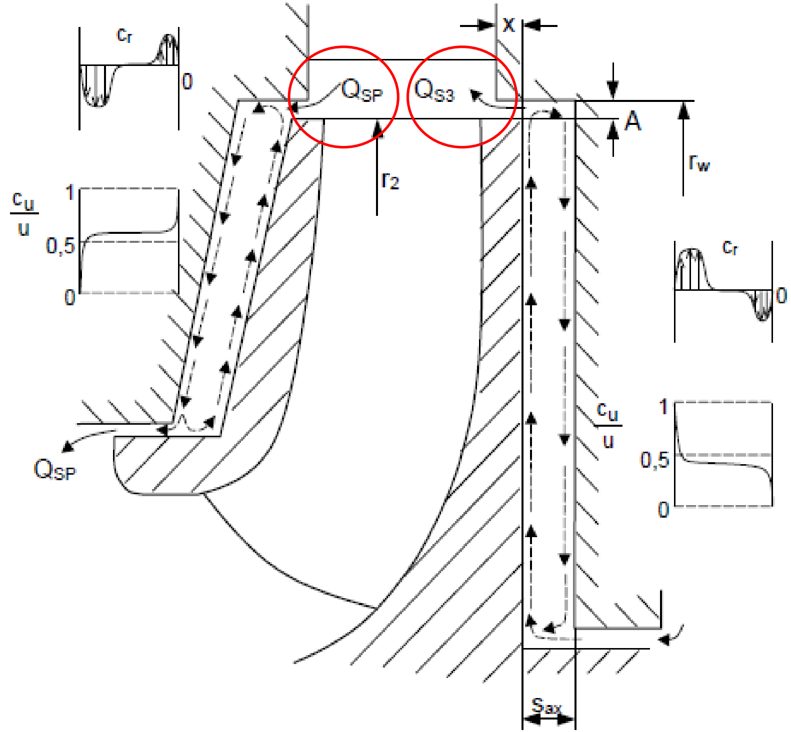


Leakage flow losses

Part of the discharge circulates in the impeller and the side rooms

Flow rate depends on clearance, rotation in the side room, surface roughness of side room and gap

Depending on pump design leakage could be radially inward or outward for multistage pumps



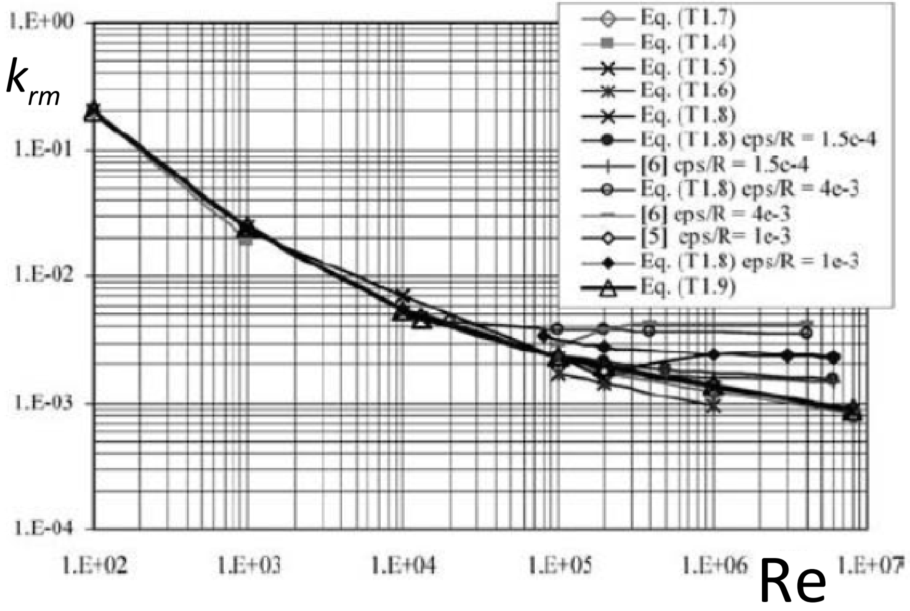
Disk friction losses

Drag losses due to rotation of impeller sidewalls in the fluid

Losses depend on surface roughness, leakage flow rate and swirl at side room inlet

Friction coefficients obtained from statistical data

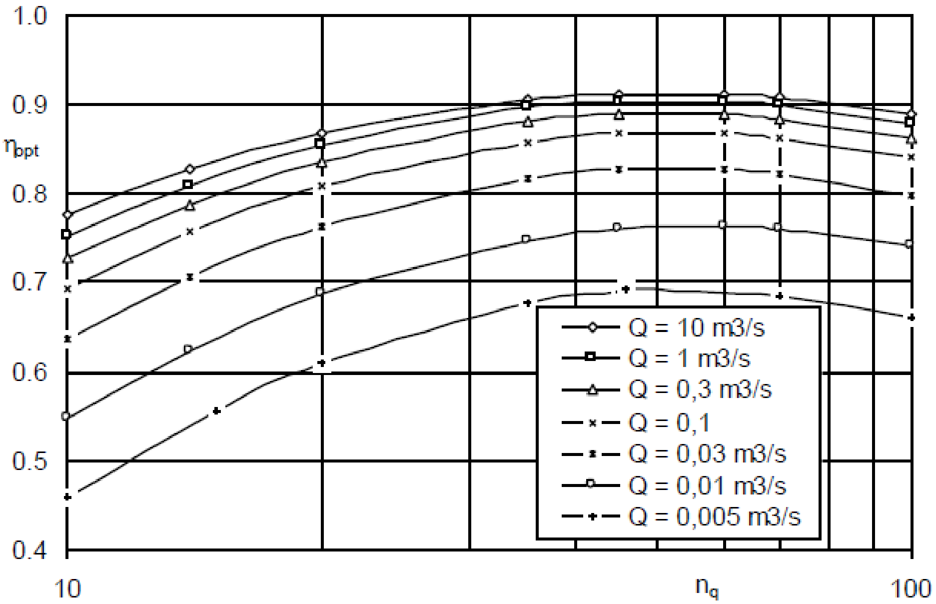
$$P_{rm} \propto k_{rm} = f\left(\mathrm{Re}\right)$$



Pump Efficiency

Efficiency depends on:

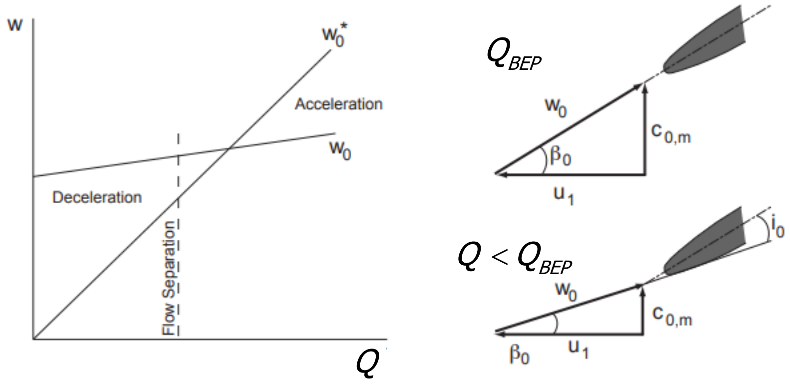
- Pump type
- Size
- Specific speed
- Liquid viscosity
- Rotational speed
- Pump execution (Surface roughness)



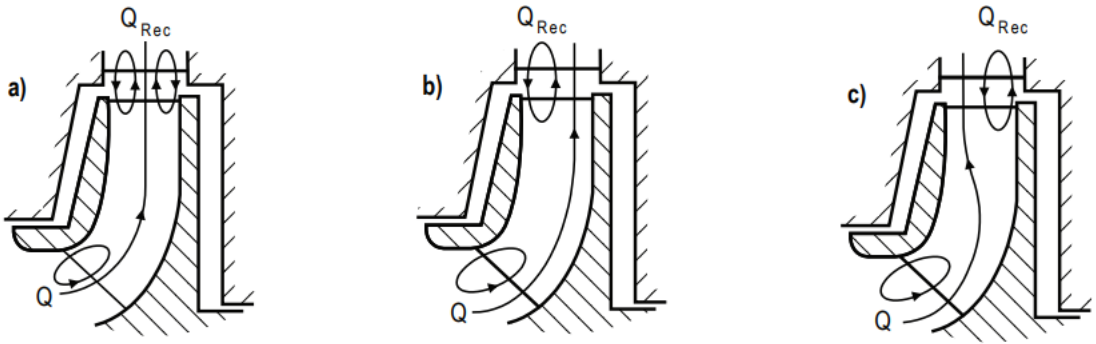
Maximum attainable efficiency for a single stage volute pump acc. to [Guelich2004]

Partial load

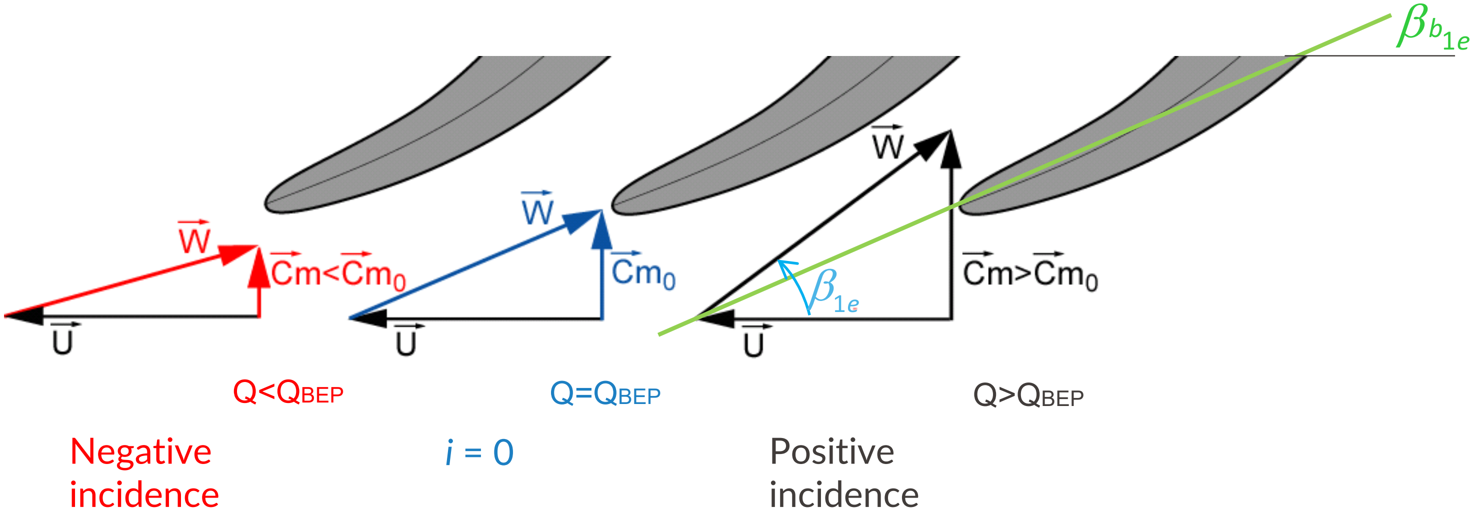
Part load recirculation causes
- Increased vibration levels
- Detoriated suction performance
- Increased pressure pulsation values
- Performance curve instabilities



Recirculation occurs where the flow is detached and strong gradients perpendicular to the main flow are present



From L6: Relative flow angle incidence

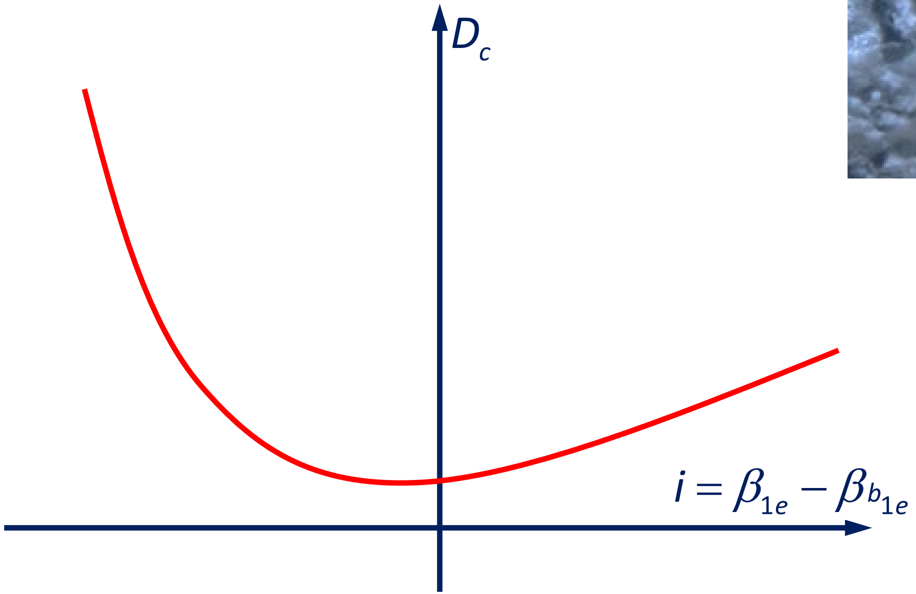


Nomenclature:

$\beta_{b,1e}$	(°)	Blade Pitch Angle
β_{1e}	(°)	Relative Flow Angle
W	(m/s)	Relative Flow Velocity
U	(m/s)	Tangential Flow Velocity
Cm	(m/s)	Meridional component of the flow absolute velocity

Flow Incidence: $i = \beta_{1_e} - \beta_{b_{1e}}$

From L6: Relative flow angle incidence Effect on cavitation onset



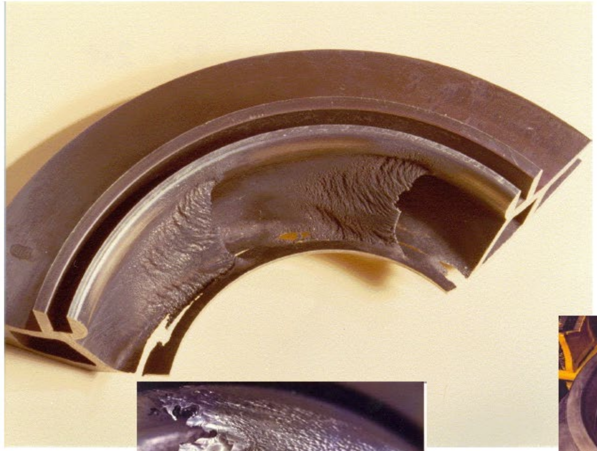
Main Cavitation issues

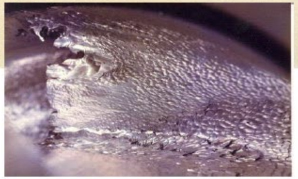
Different cavitation types can be distinguished:

- Bubble cavitation
- Cloud cavitation
- Vortex cavitation

Main problems due to cavitation and bubbles collapse:

- Noise
- Vibrations
- Material removal -> erosion





Main Cavitation issues

Volute

Diffuser

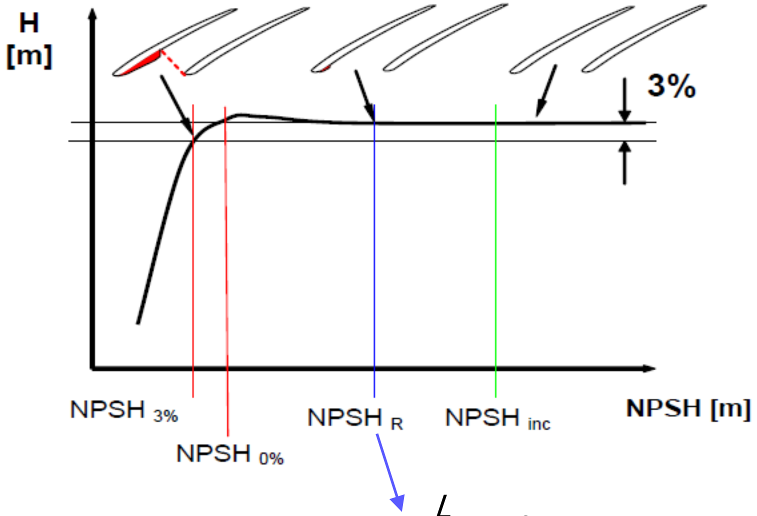
Cavitation Criteria

Net Positive Suction Energy:

$$NPSE \approx \frac{p_a}{\rho} - \frac{p_v}{\rho} - gh_s = g \times NPSH$$

Thoma Number:

$$\sigma_{TH} = \frac{NPSE}{E}$$



$$\frac{L_c}{D_1} = 1\%$$

NPSH3% : 3% head impairment

Large vapor cavities for NPSH3% at part load

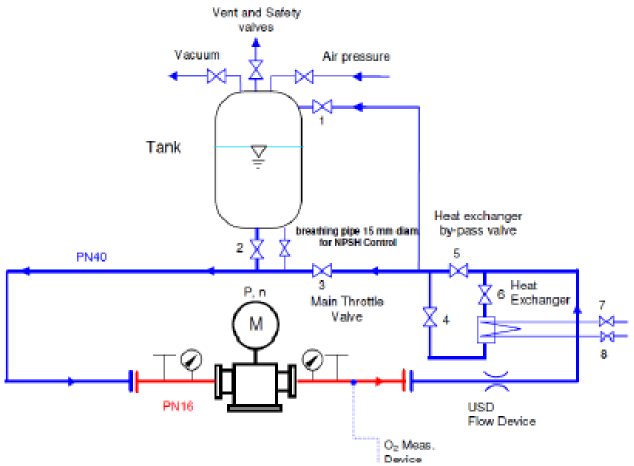
Nomenclature:

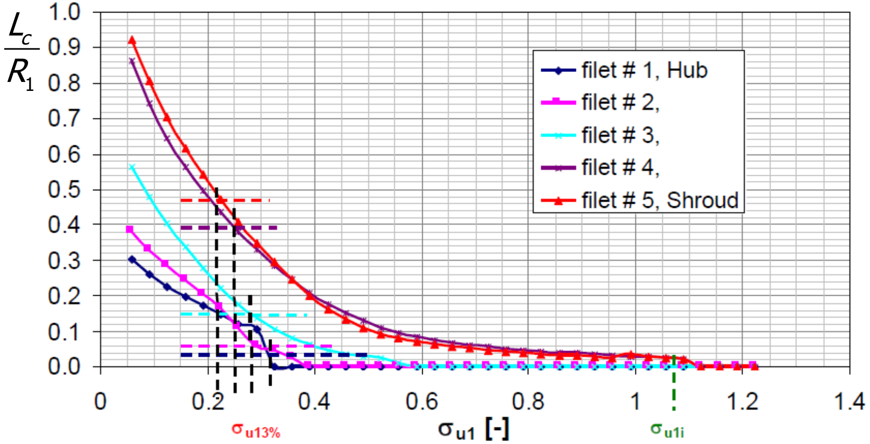
NPSH	(m)	Net Positive Suction Head
Lc	(m)	Cavitation Length

Cavitation Criteria

3% head drop evaluation

- 3% head drop evaluation in closed test loops by controlled variation of suction pressure
- Evaluation of cavitation inception in special model pumps with optical access to impeller
- Predicted cavity length as a function of cavitation coefficient Cavity evolution



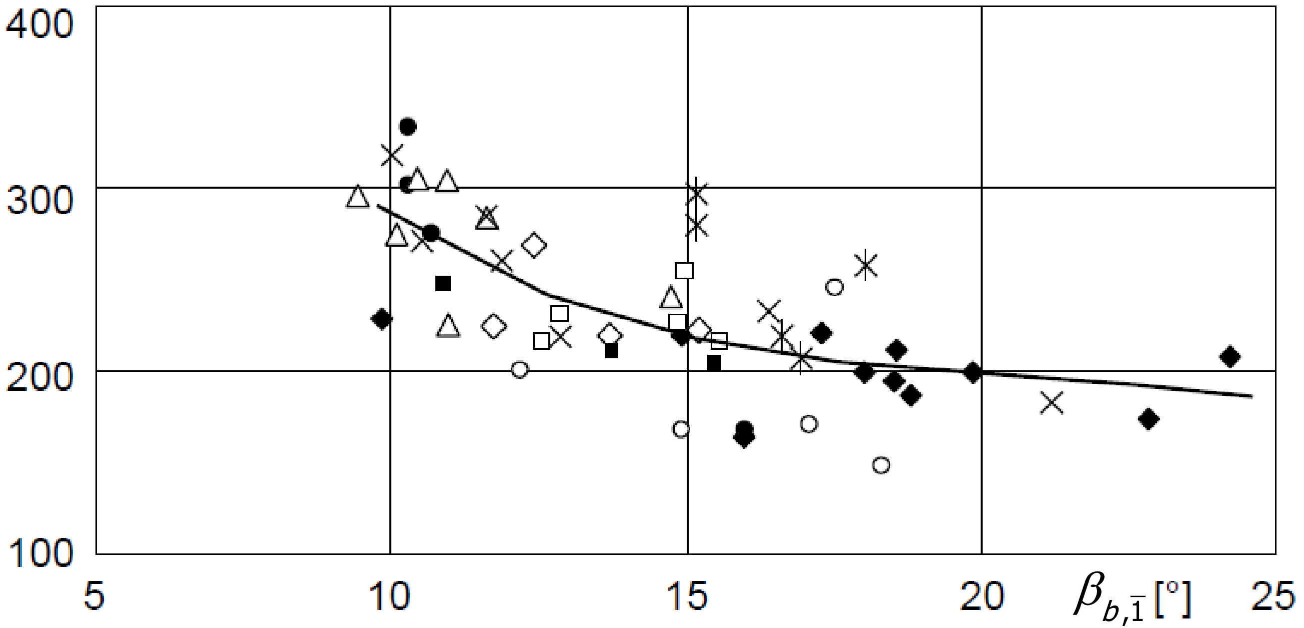


$$\sigma_{U_1} = \frac{2 \cdot g \cdot NPSH}{U_1}$$

Cavitation Criteria

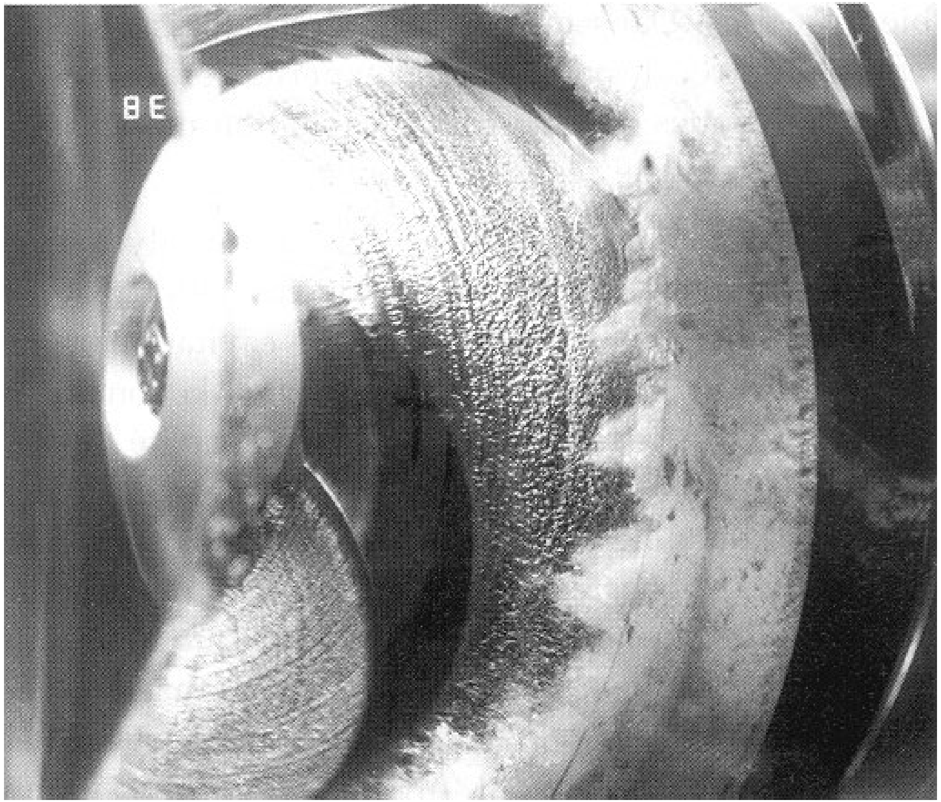
Suction specific speed

$$n_{ss} = \frac{n \cdot \sqrt{Q}}{(NPSH)^{3/4}}$$

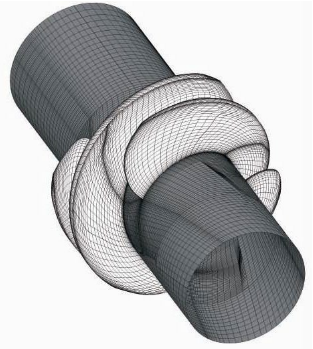


Axial Inducer

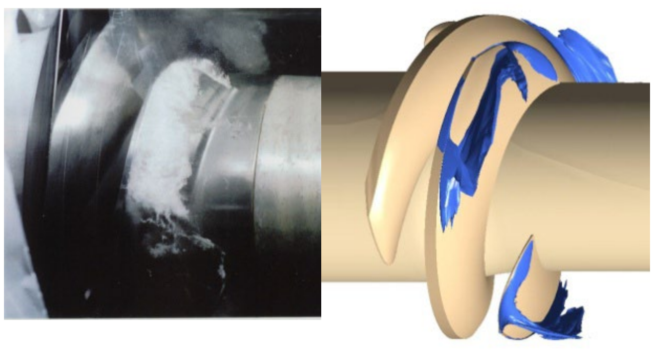
If the pressure is still to low, it is possible to add an axial inducer to pressurized the inlet section of the main pump and avoid cavitation in the main machine



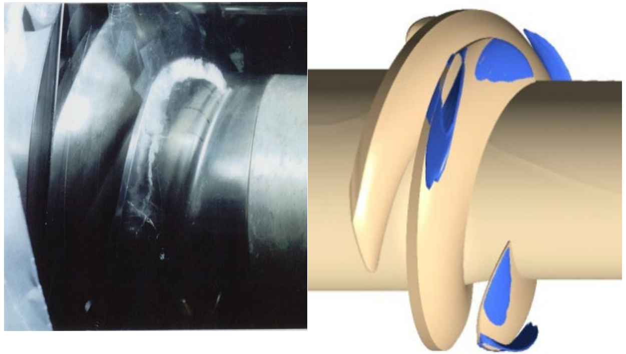
Axial Inducer



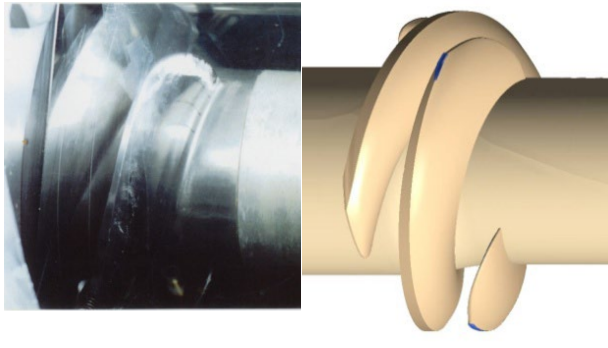
$Q/Q_n = 80\%$



$Q/Q_n = 100\%$



$Q/Q_n = 120\%$



Hydraulic Forces

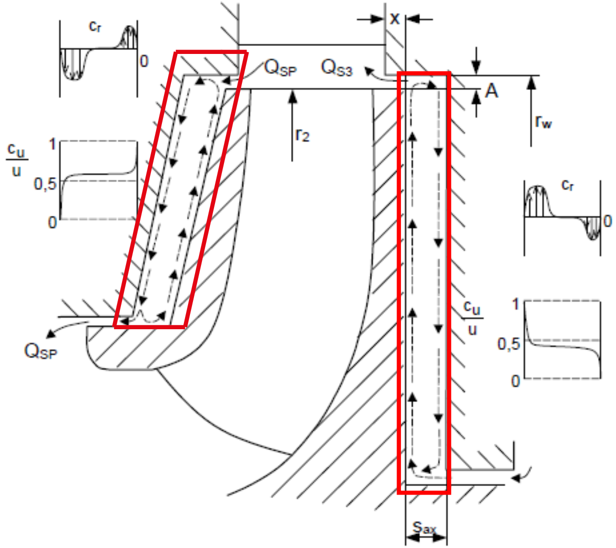
Axial Thrust

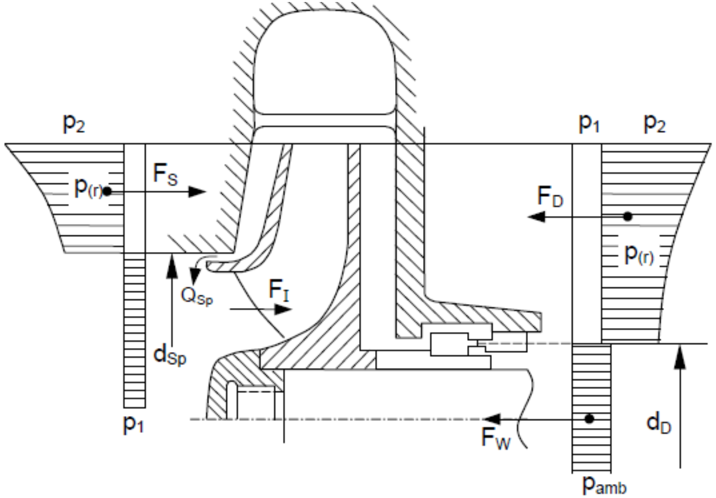
Axial thrust caused by the rotational and radial flow in hub and shroud side rooms which yield pressure distribution between outer and inner diameter.

Resulting thrust consists of

- Force on hub $F_D = 2\pi \int p(r) r dr$
- Force on shroud $F_s = 2\pi \int p(r) r dr$
- Impulse force due to flow redirection in impeller F_I
- Unbalanced shaft force F_W

→ $F_{ax} = F_D - F_S - F_I + F_W$





Hydraulic Forces
Axial Thrust

Depending on pump type, different approaches to balance axial thrust:

- Single stage pumps:
  - Balance holes
  - Back vanes

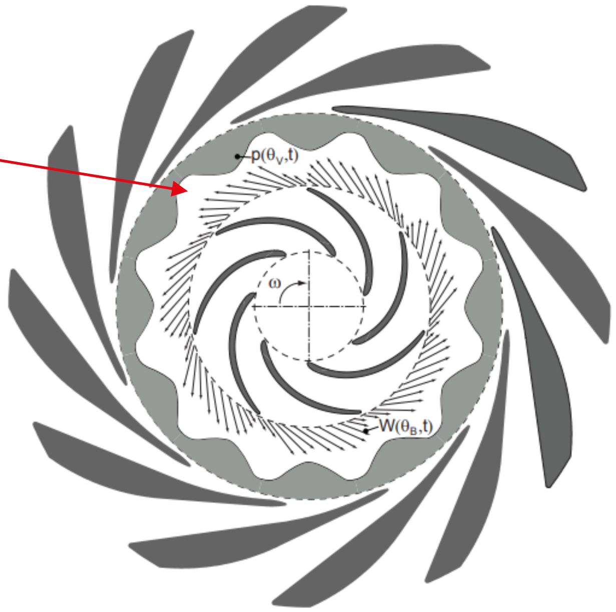
- Multistage pumps:
  - Balance piston
  - Balance disk
  - Back-to back impellers

Hydraulic Forces
Radial Forces

- Stationary forces due to
  - Non-uniform side room flow
  - Pressure distribution in clearances

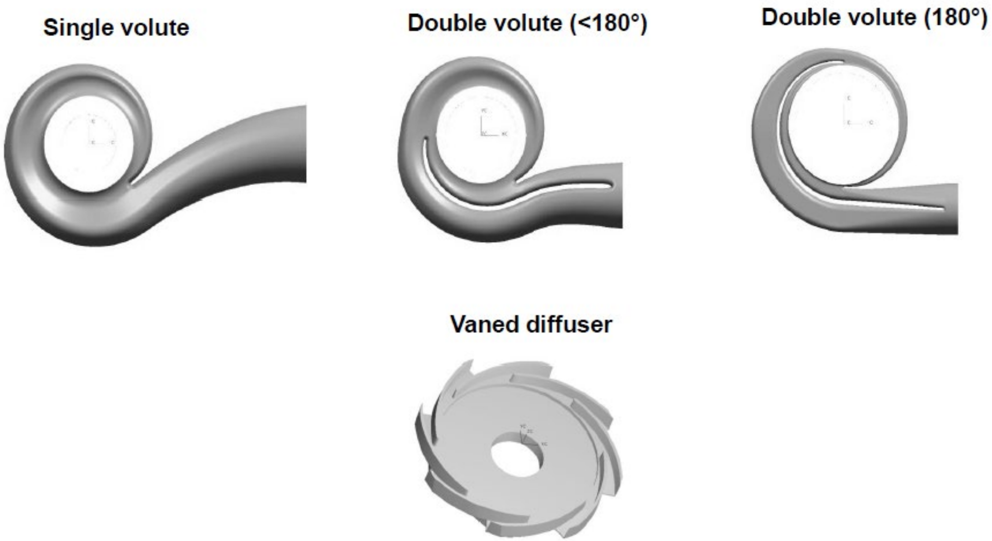
- Dynamic forces are due to Rotor-Stator Interaction and excite vibrations.
  RSI are due to the interaction between the pressure field resulting at the outlet of the impeller blades and the pressure field at the diffuser vanes

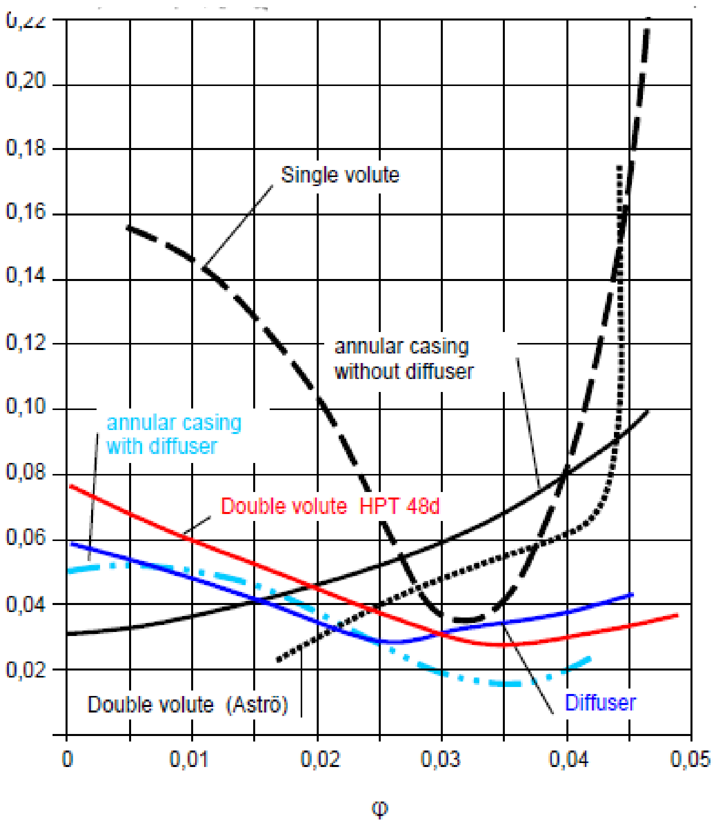
Diffuser vanes



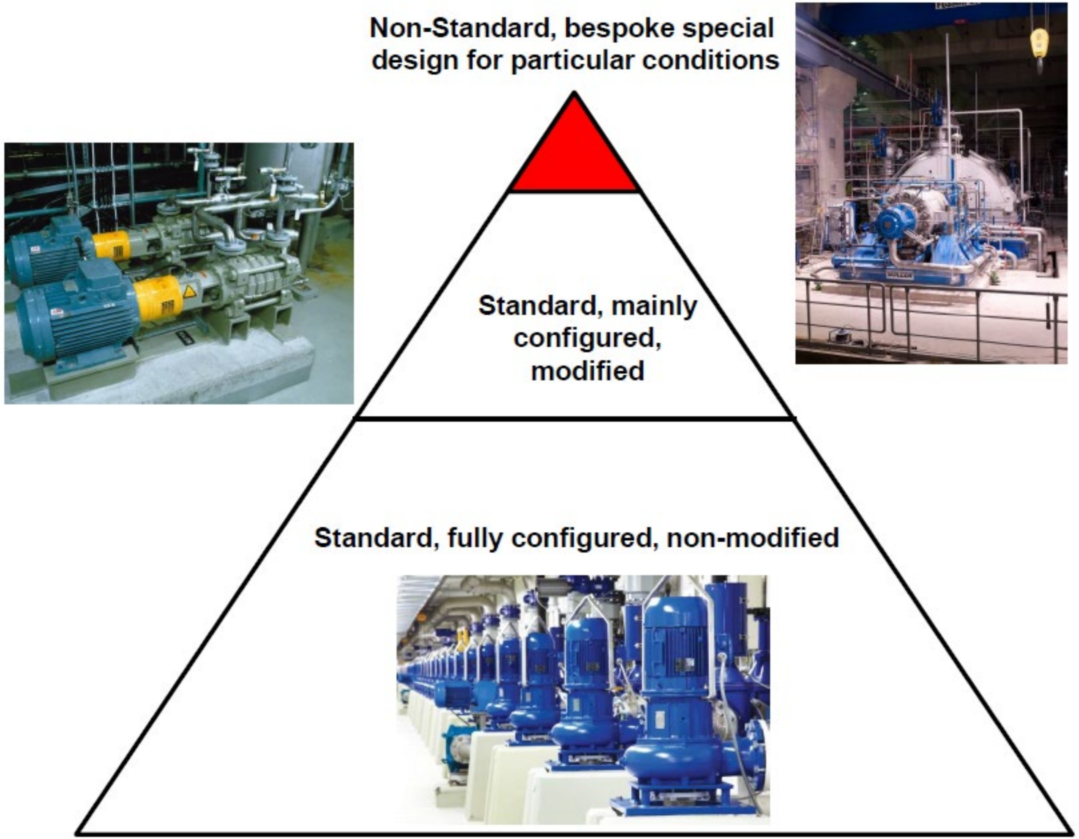
Hydraulic Forces
Radial Forces

- Different radial forces for:
  - Single volute
  - Double volute
  - Vaned Diffuser





Selection of pumps



Selection of pumps

Partial load limit (minimum discharge)

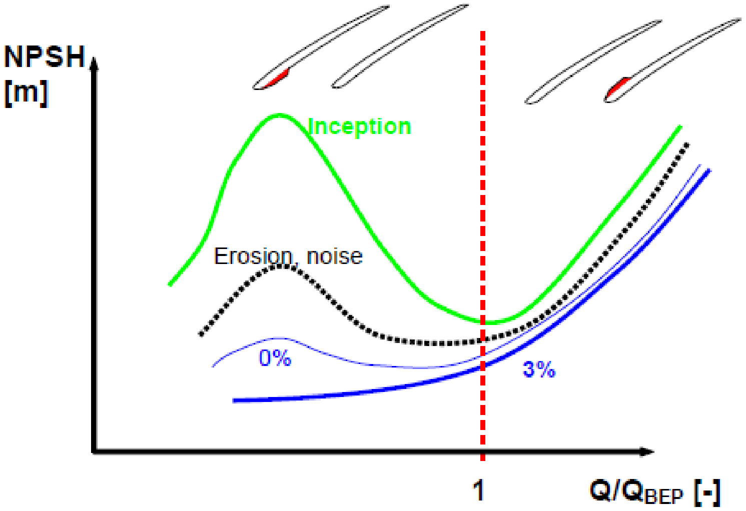
→ Increased vibrations due to

- Dynamic axial & radial loads
- Pressure pulsations

→ Cavitation

→ Motor power (axial pumps)

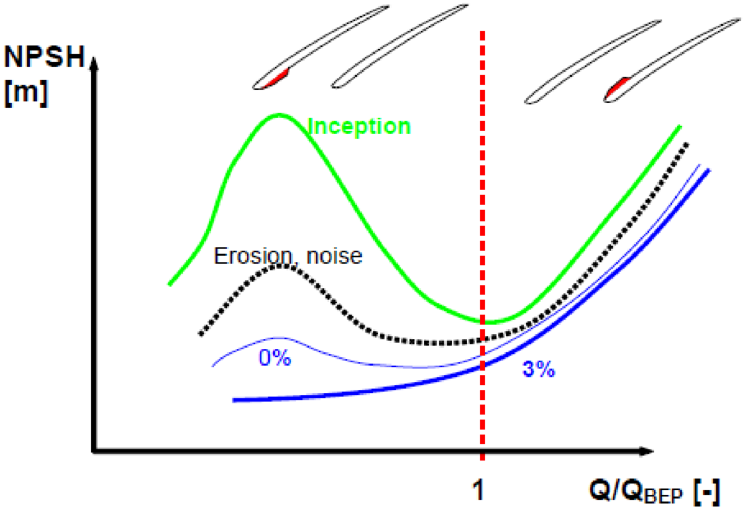
→ Thermal minimum discharge causing vaporization of liquid in the pump: $Q_{th} = \dfrac{P}{\rho C_p \Delta T}$

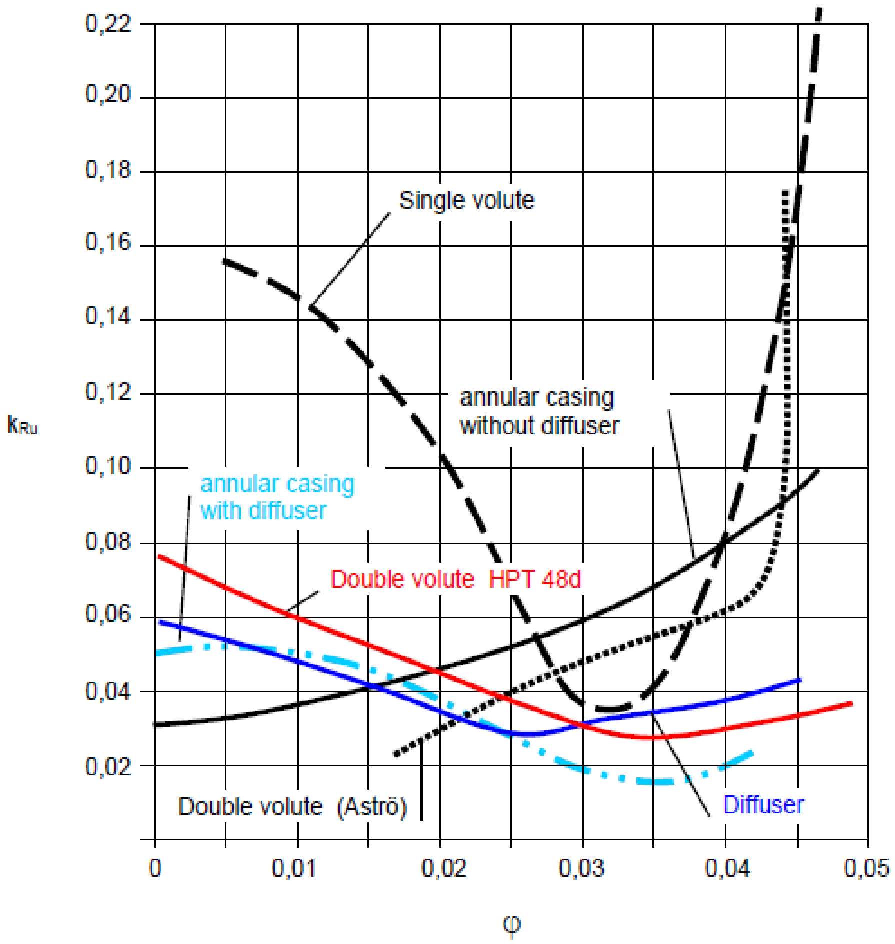


Selection of pumps

Full load limit (maximum discharge)

→ Cavitation
→ Increased loads
→ Motor power (radial pumps)

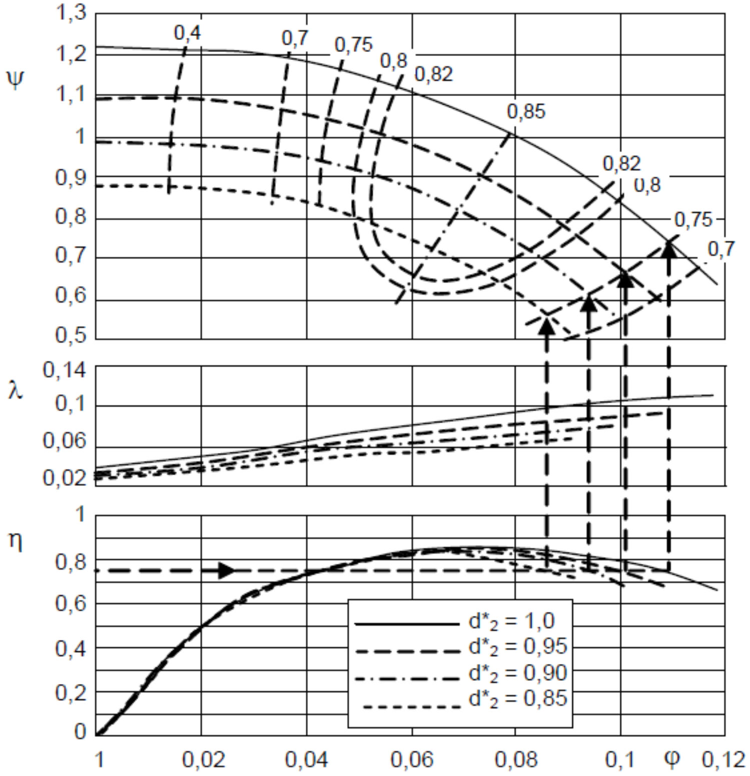


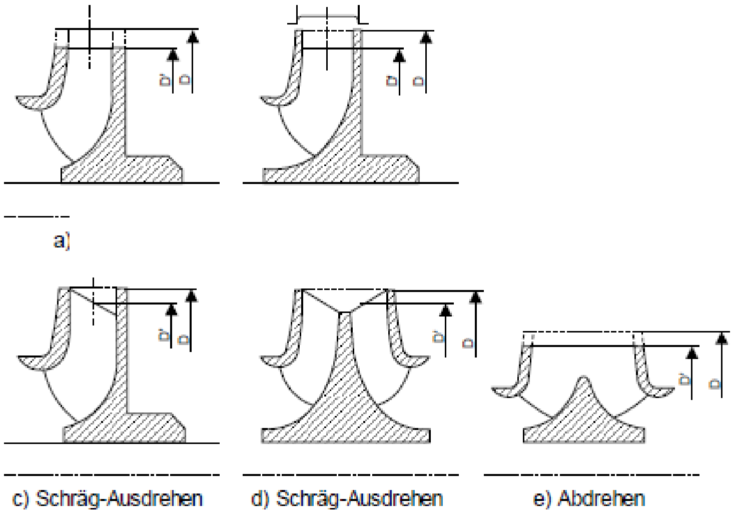


Selection of pumps

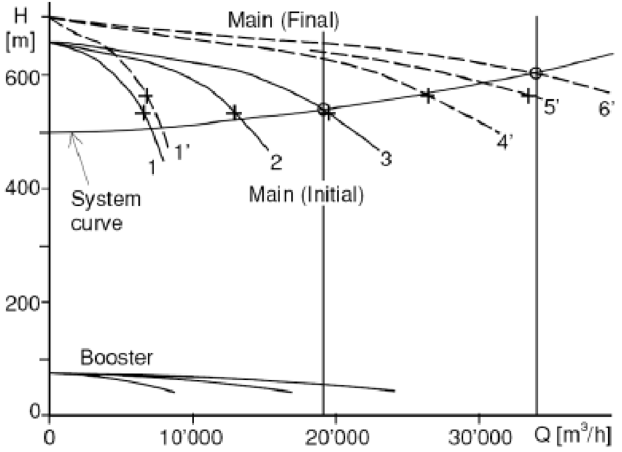
Small size standard pump

Adaptation to required flow and head by trimming of impeller outer diameter

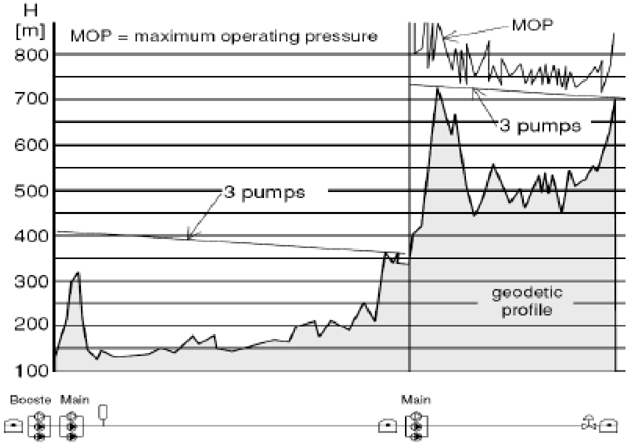




Selection of pumps
Large size pump



Parameters:
- Types of design
- Hydraulics set up
- Adapted Design
- Data for a selection
- Pumps selection method



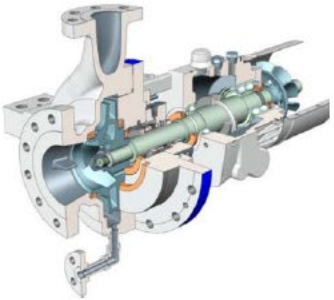
Data Requirements:
- Duty conditions
  - Flow rate
  - Head
  - NPSH available
  - Water temperature
- Water Quality
  - Corrosion
  - Abrasion
- Site conditions
  - Altitude
  - Environmental conditions
- Network description
  - Network profiles
  - Modes of operation

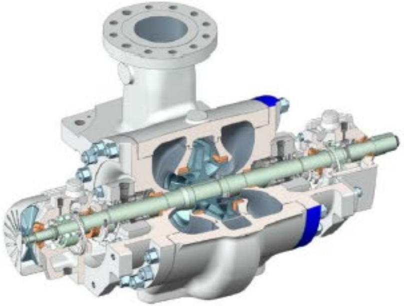
Selection of pumps

Hydraulic set-up

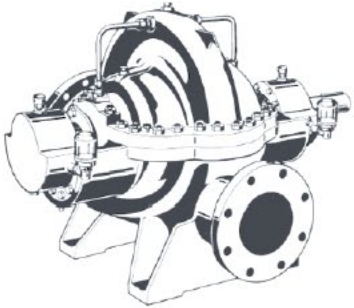
End Suction | Double Suction | 2 Stage Back-to-back

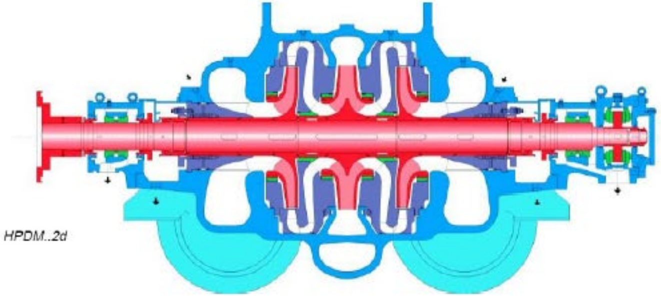
Radial Split





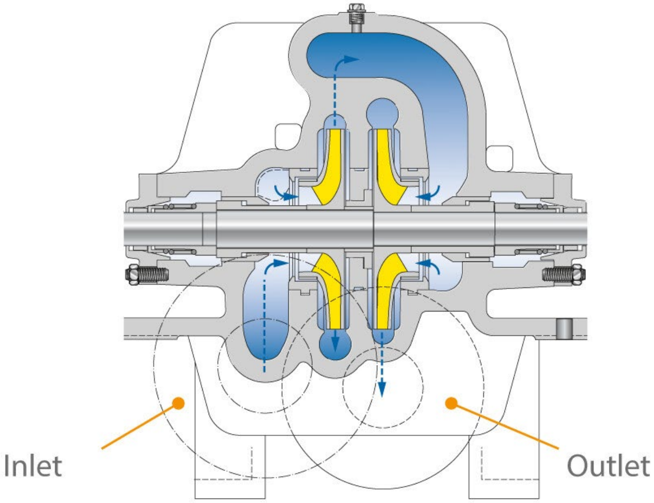
Axial Split



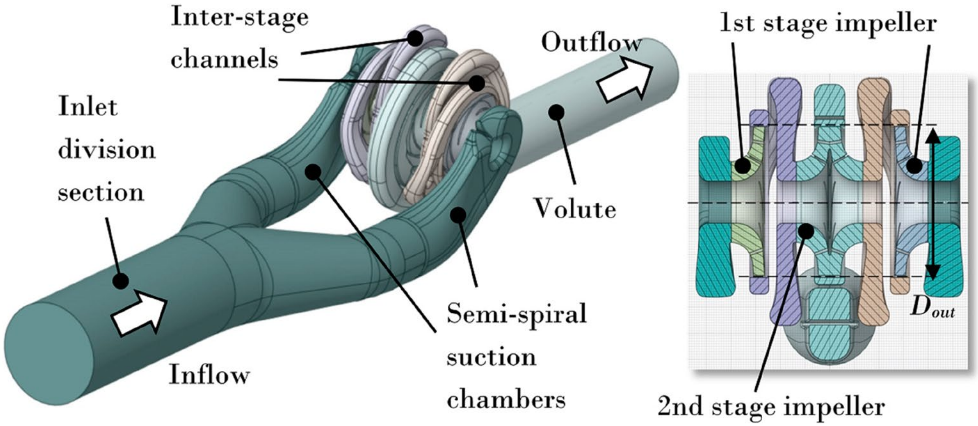


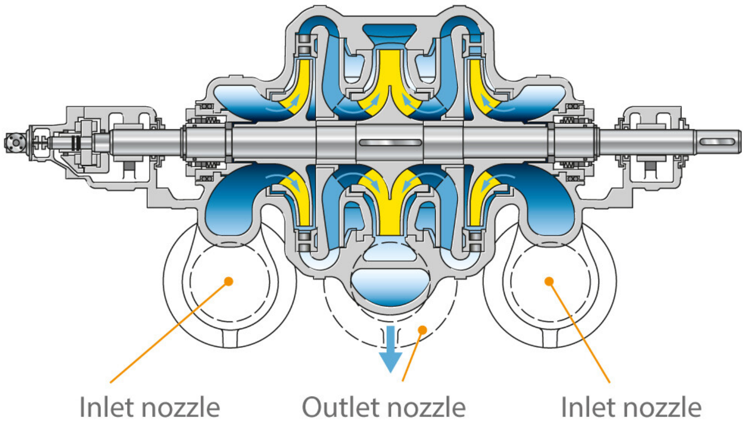
Selection of pumps
Hydraulic set-up

2-stage single suction, also called back-to-back



Single or multistage double suction



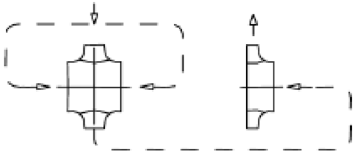


Selection of pumps

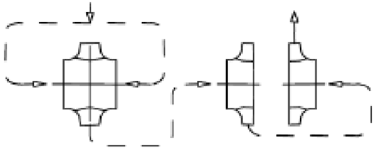
Hydraulic set-up

There exist also multistage pumps which have on or multiple stages with double suctions and the others with single suctions, here some examples:

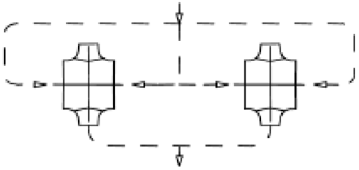
2 stages, double suction
1st stage



3 stages, double suction
1st stage



2 stages, double suction

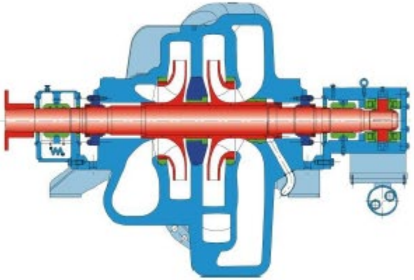


Selection of pumps

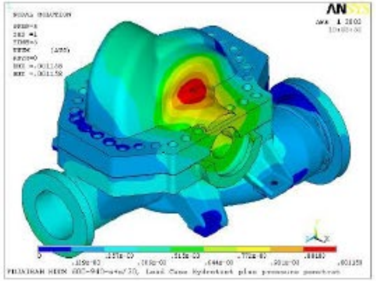
Adapted design

Hydraulic design

Mechanical design



Finite elements analysis



Model

Pattern construction

Final Testing

Selection of pumps

Large size pump

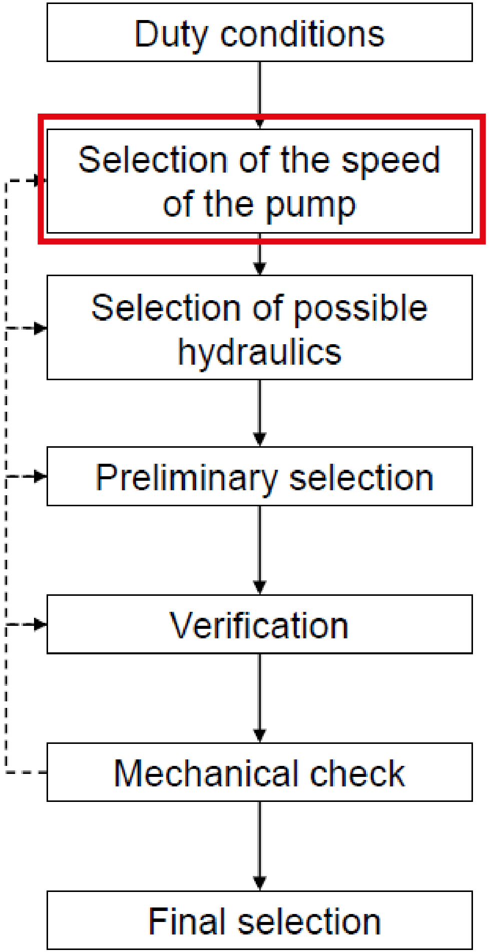
Iterative process

STEP 1: choice of the rotational speed

The pump speed is limited by the NPSH available.
Suction Specific Speed:

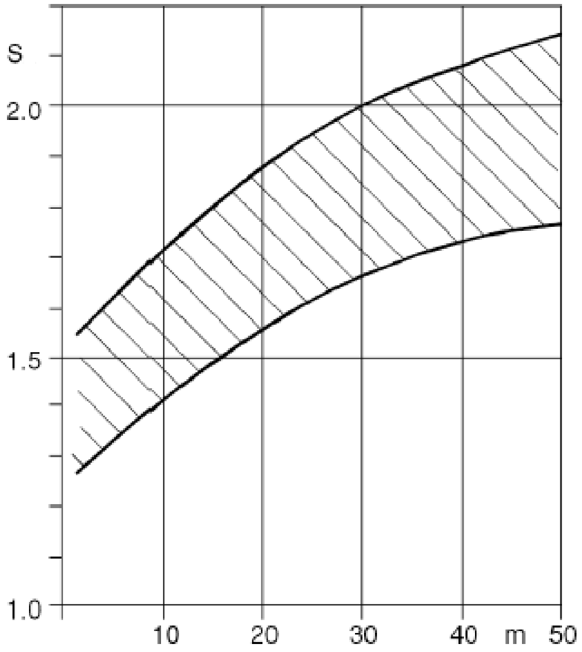
$$n_{ss} = \frac{n \cdot \sqrt{Q}}{(NPSH)^{3/4}} = [200 - 220]$$

$$NPSH_{3\%} = \frac{NPSH}{S}$$

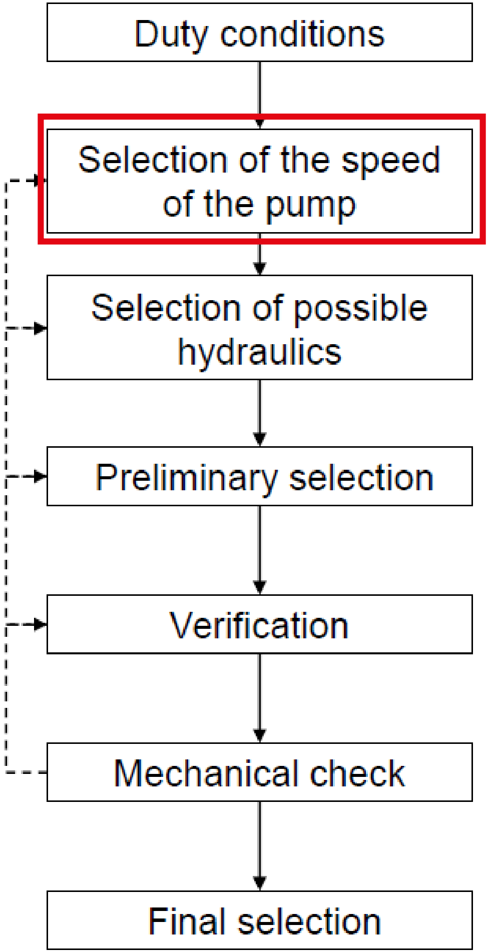


Selection of pumps
Large size pump

Iterative process
STEP 1: choice of the rotational speed



Safety margin in relationship to the NPSH - 3 % of head impairment



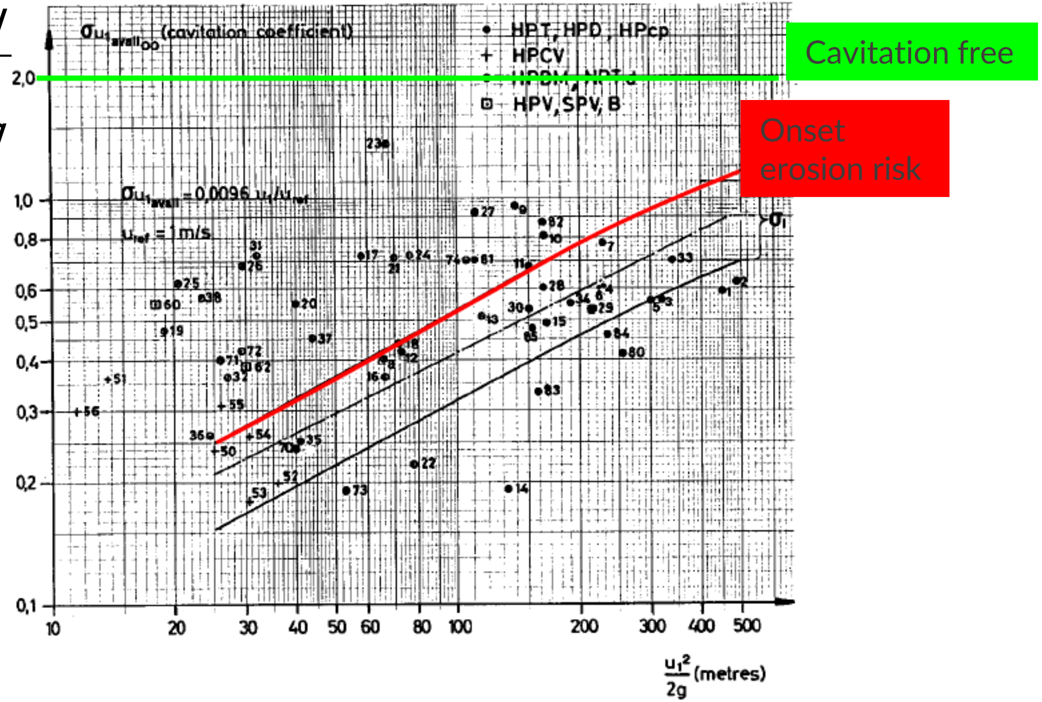
Selection of pumps

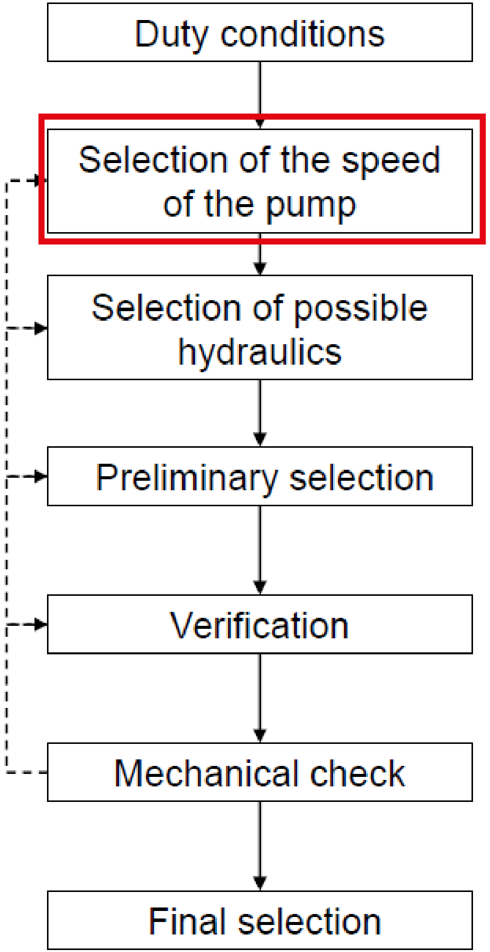
Large size pump

Iterative process

STEP 1: choice of the rotational speed

$$\sigma = \frac{NPSH}{U_1^2 / 2g}$$





Selection of pumps

Large size pump

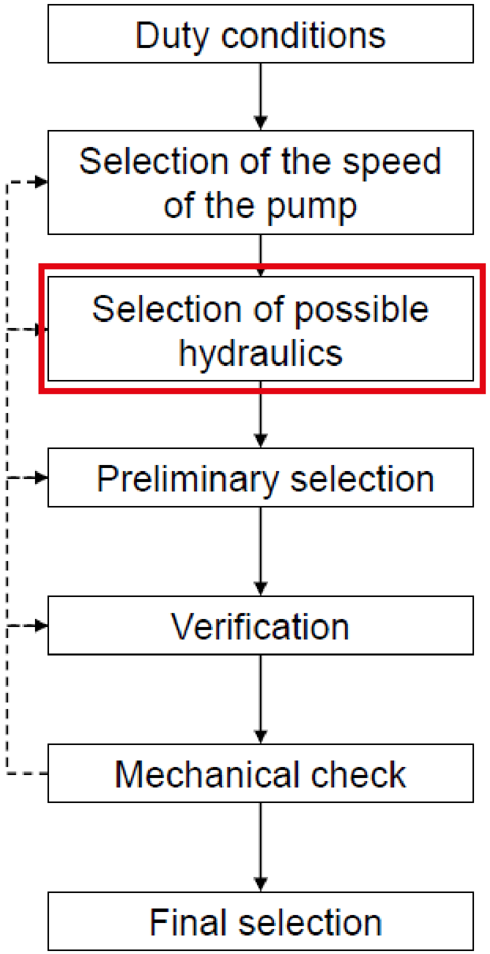
Iterative process

STEP 2: Number of stages

Specific speed of the pump depending on the discharge, head and number of stages:

$$n_q = \frac{n \cdot \sqrt{Q}}{\left(\frac{H}{z_s}\right)^{3/4}}$$

Typical range: [12-80]

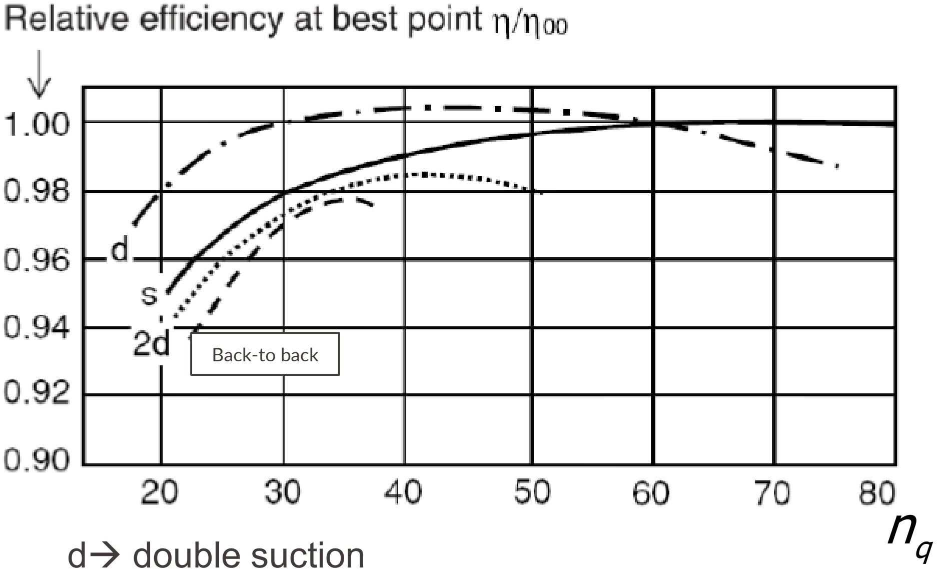


Selection of pumps

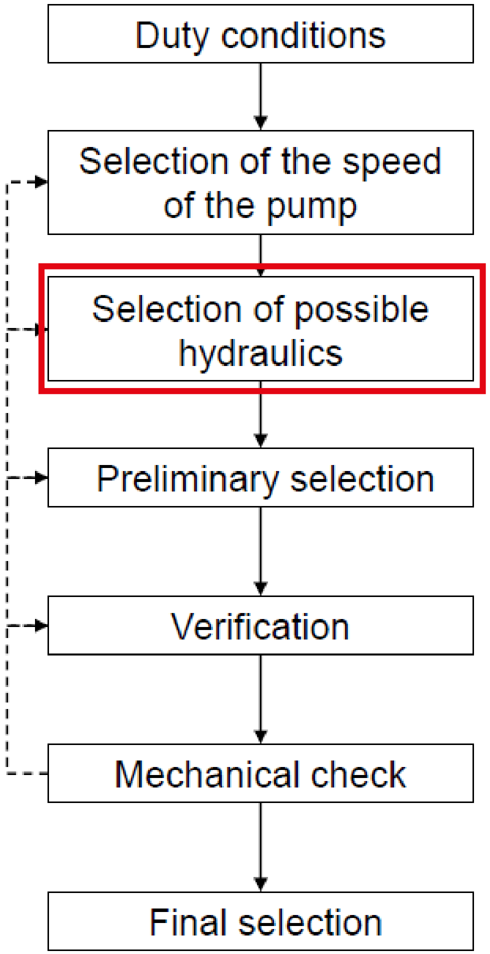
Large size pump

Iterative process

STEP 2: Number of stages



d→ double suction
s→ single suction

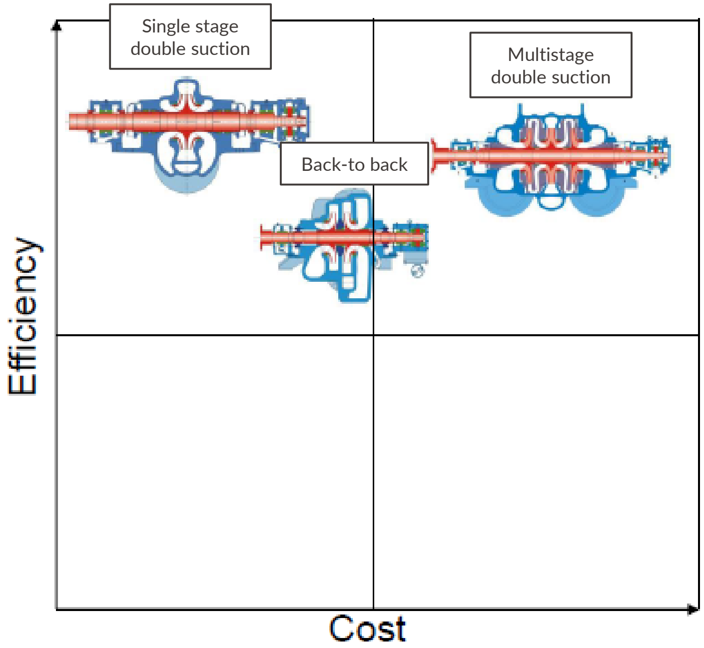


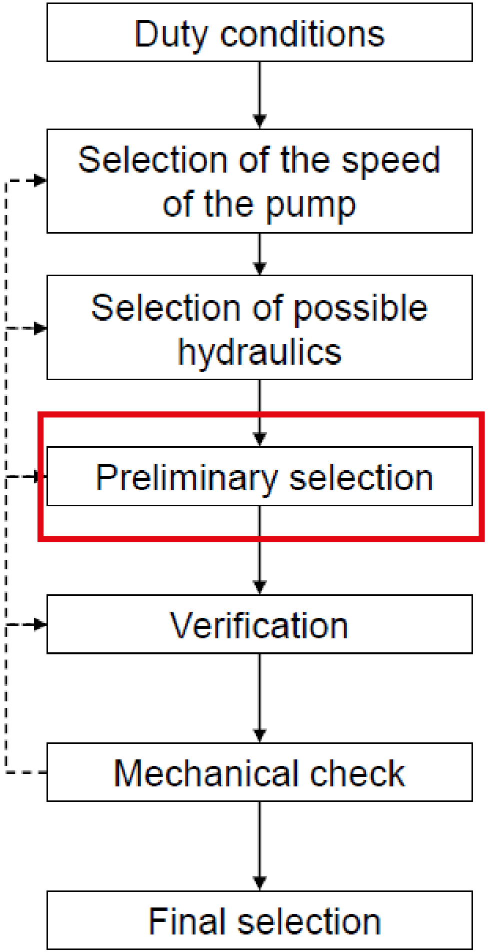
Selection of pumps

Large size pump

Iterative process

STEP 3: Preliminary selection of the design





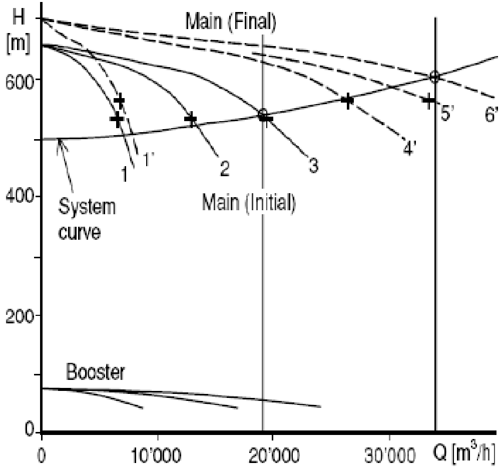
Selection of pumps

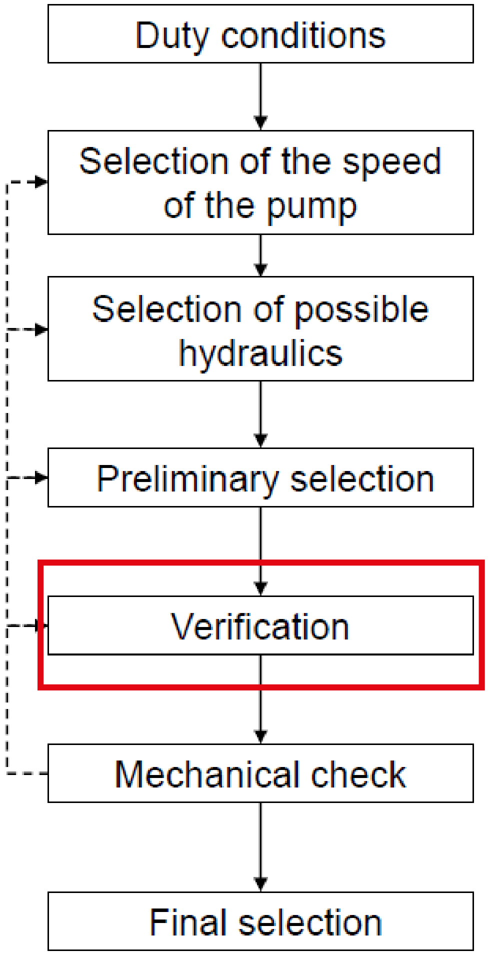
Large size pump

Iterative process

STEP 4: Verification

During operation, the pumping system is experiencing head variations, changes in head losses, partial and full load operations: →NPSH, efficiency , power limits and vibrations need to be verified through the full operating range of the pump.





Selection of pumps

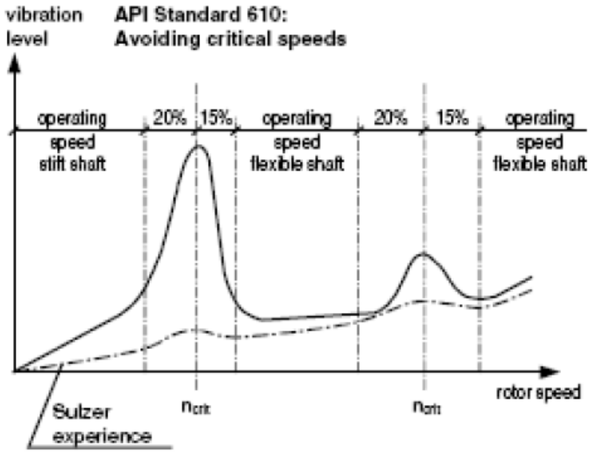
Large size pump

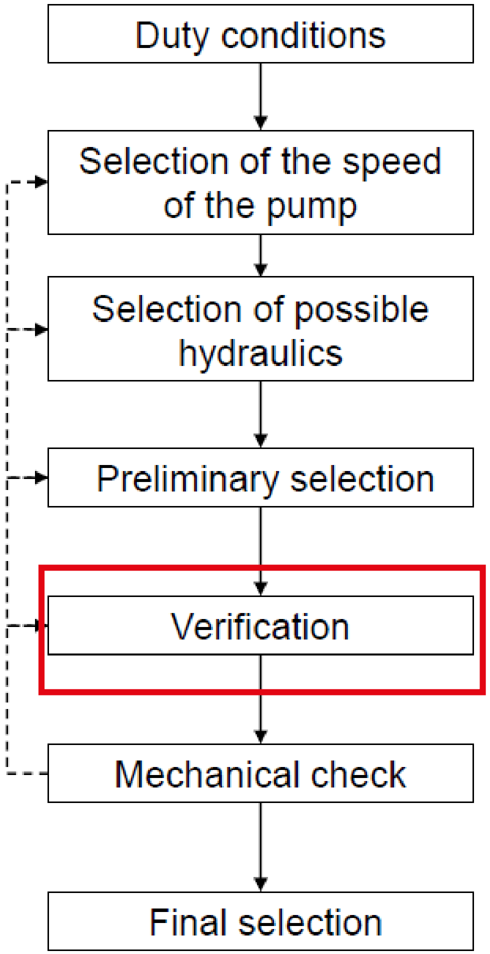
Iterative process

STEP 5: Mechanical check

Typical mechanical verifications:

- Axial and radial loads,
- Shaft deflection,
- Fatigue on the impeller blades,
- Life time of the bearings



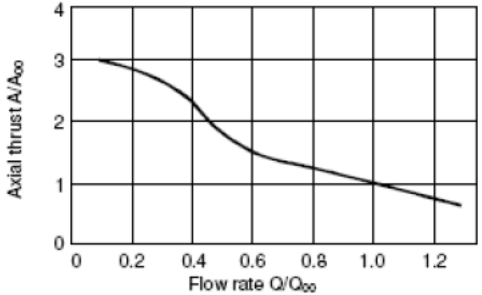


Selection of pumps

Large size pump

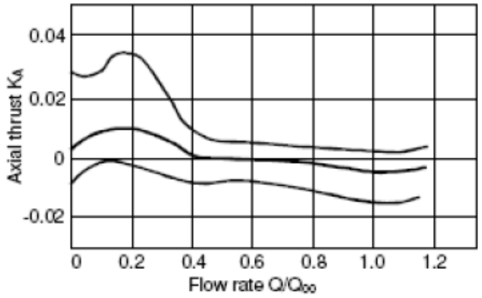
Iterative process

STEP 5: Mechanical check



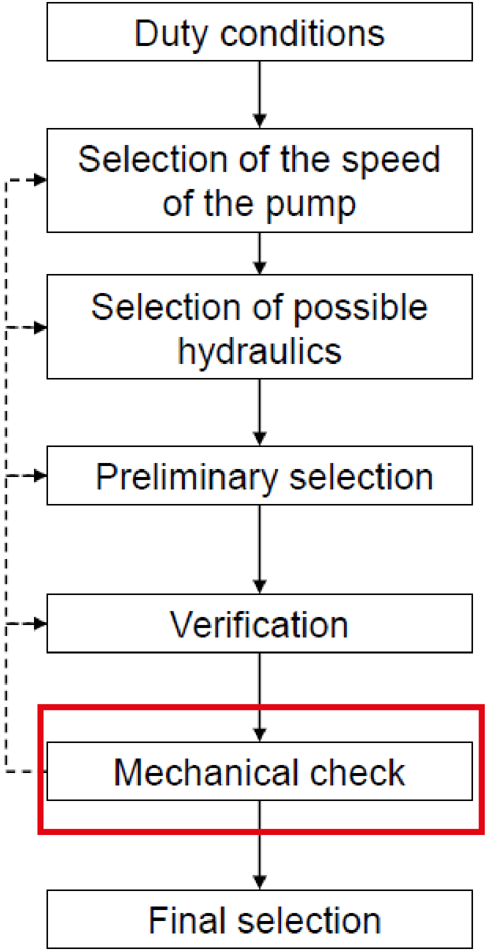
a Dimensionless axial thrusts as a function of the flow rate for back-to-back s+s arrangement

Back-to back



b Dimensionless axial thrusts as a function of the flow rate for two stage, double-flow 2d arrangement

Multistage double suction



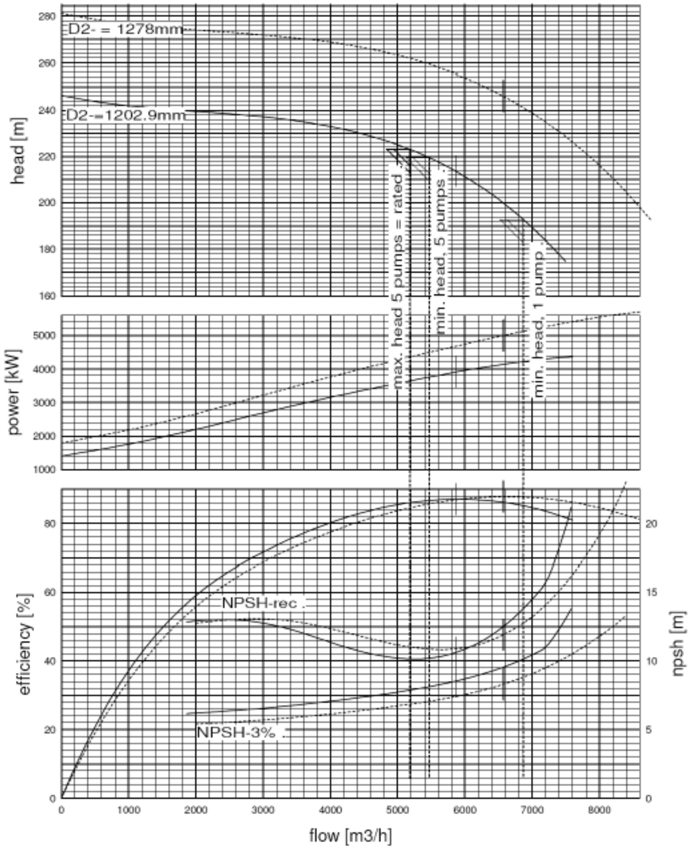
EPFL

Selection of pumps

Large size pump

Iterative process

STEP 5: Selected pump, Final characteristic curve



- Duty conditions
- Selection of the speed of the pump
- Selection of possible hydraulics
- Preliminary selection
- Verification
- Mechanical check
- **Final selection**

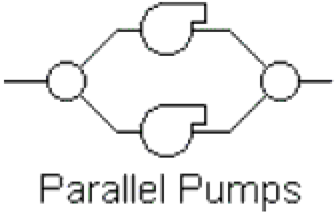
Interaction between pumps and the system

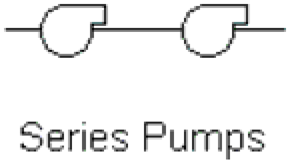
Pumping operating modes

There are 3 possible configurations: SERIAL, PARALLEL, COMBINED (SERIAL AND PARALLEL)

The choice depends on:

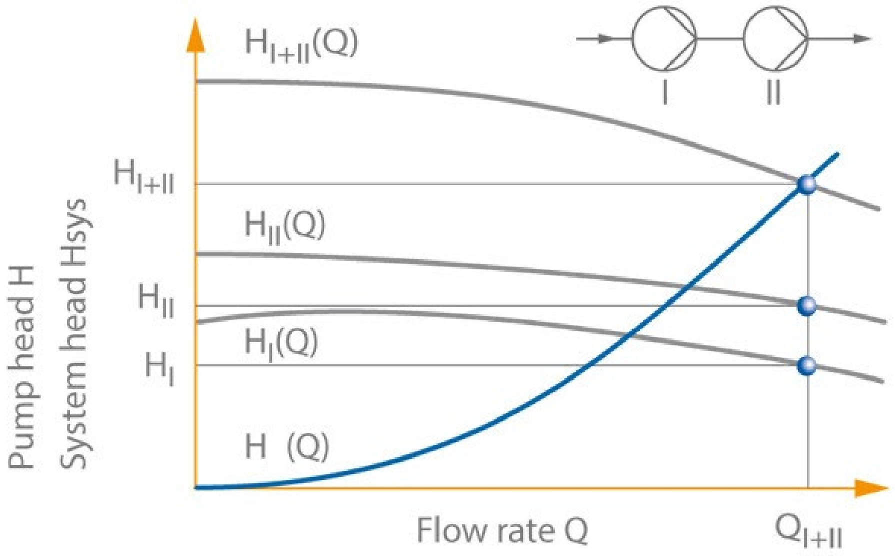
- Pumped fluid (viscosity)
- Continuous and intermittent operation
- Pressure limit / Size / type of piping
- Topography
- Required duty conditions
- Evaluation criteria
- Foreseen changes in the operating conditions





Interaction between pumps and the system
Pumping operating modes

Serial Configuration:
Best choice in case of low head applications



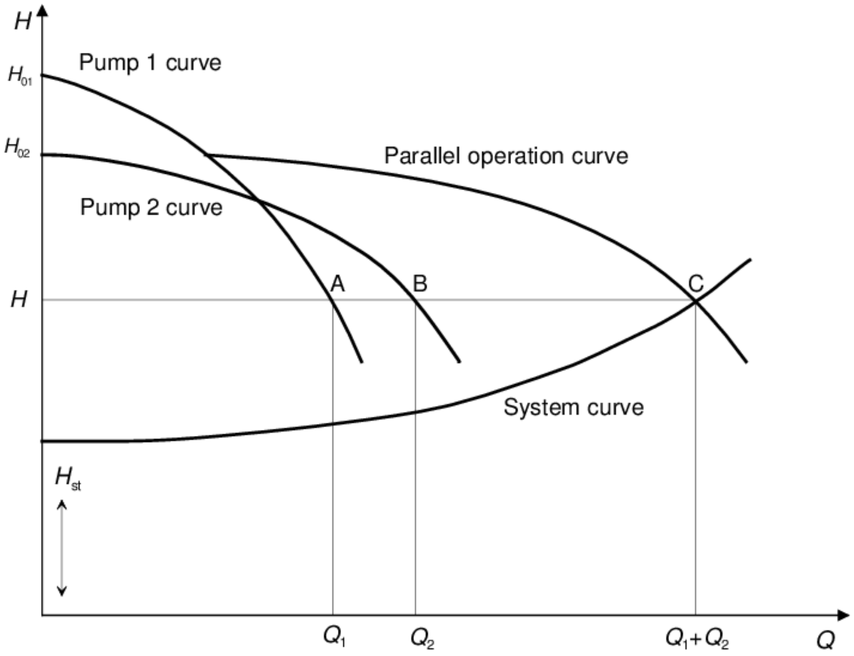
Offers a better operating flexibility and a better efficiency

A variable speed drive may allow a greater flexibility but in case higher speeds are required NPSH to be checked

Interaction between pumps and the system

Pumping operating modes

Parallel Configuration:
Best choice in case of high head applications



- Offers a better operating flexibility
- A variable speed drive may allow a greater flexibility but in case higher speeds are required NPSH to be checked

Interaction between pumps and the system

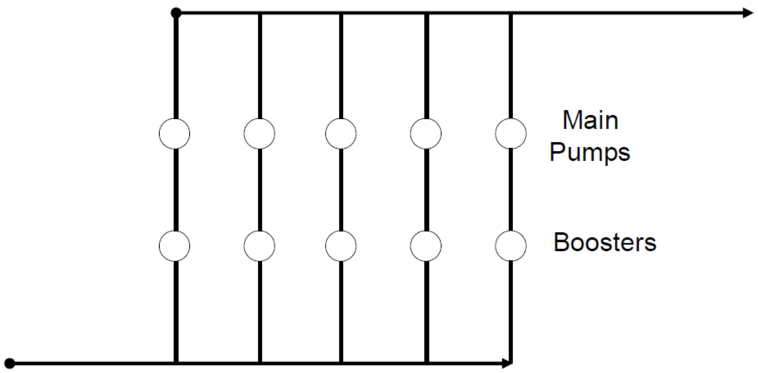
Pumping operating modes

Parallel Configuration:

→ A combination of pumps in series and in parallel, with a fixed or variable drive speed, is the best choice for complex systems

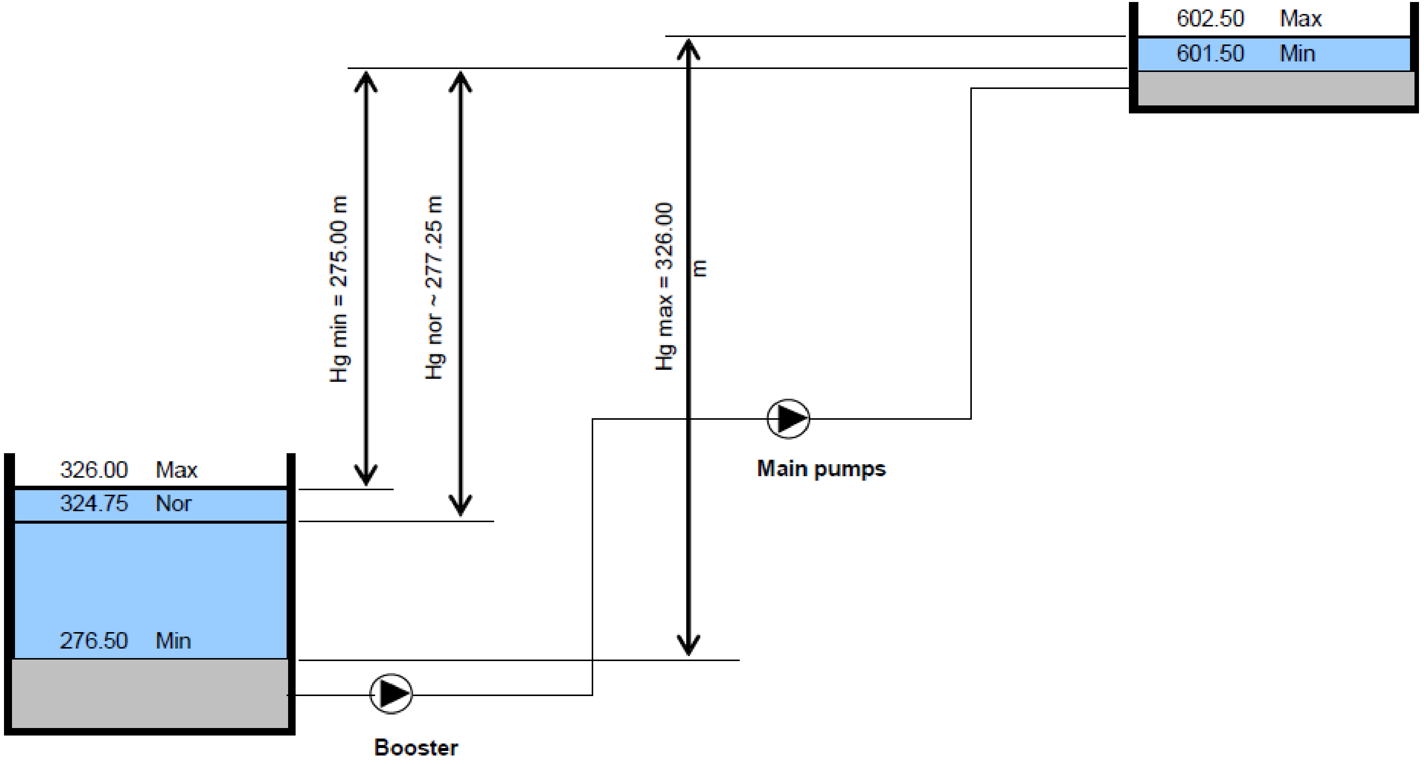
→ It is the combination that offers the largest operating flexibility together with the minimum problems related to the limitation due to cavitation

Example: use of booster pumps to pressurize the inlet section of the main pump



Interaction between pumps and the system

Pumping operating modes

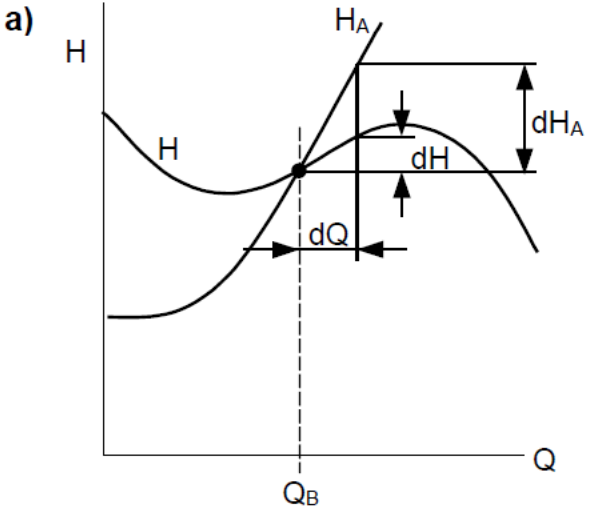


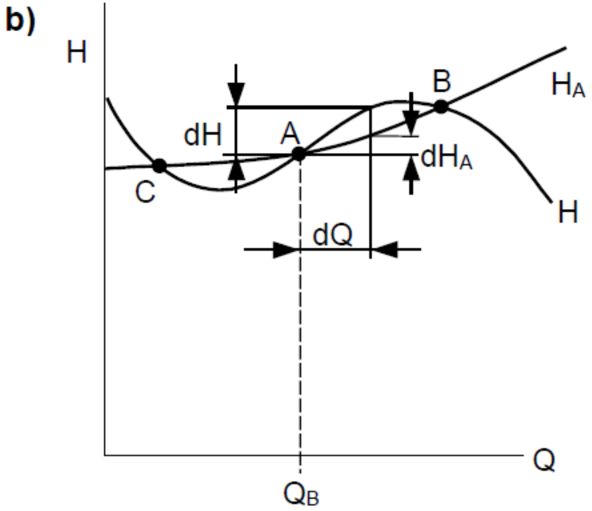
Interaction between pumps and the system

Pumping operating modes

Operational stability:

- Pump statically stable if the pump curve has a steeper gradient as the system curve (case a)
- Pump statically unstable when more than one intersection of pump and system curve is possible (case b)



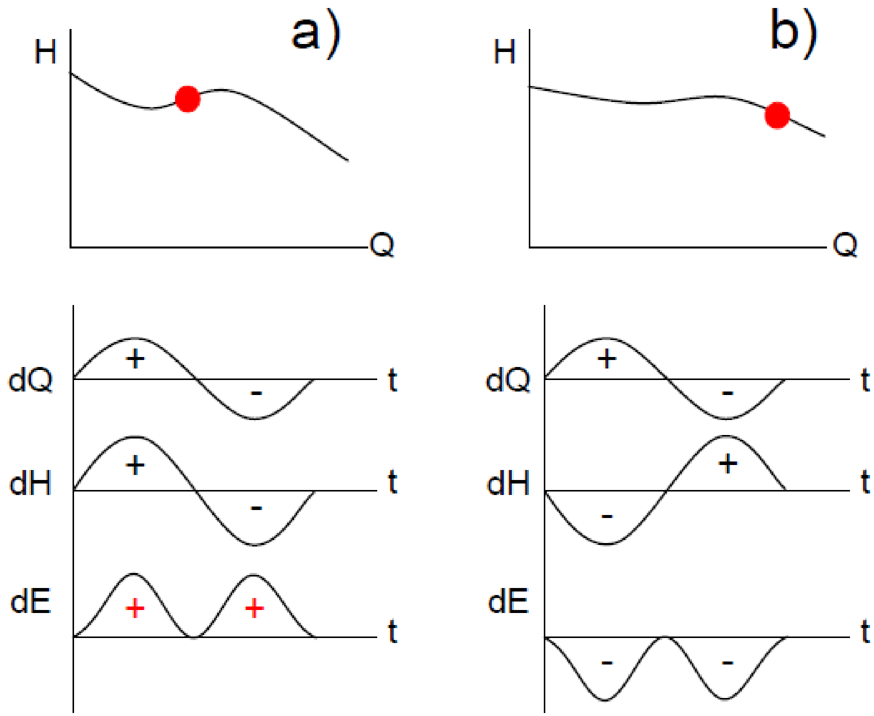


Interaction between pumps and the system

Pumping operating modes

Operational instabilities:

- Self excited flow oscillations can occur when:
  - Performance curve has positive slope
  - Compressible element in the system
- Power input : $dP = \rho \cdot g \cdot dQ \cdot dH$
  - *dH* and *dQ* on positive slope in phase -> amplifier (a)
  - On negative slope out of phase, -> damping element (b)
- Dynamic instabilities lead to low frequency pressure fluctuations



Interaction between pumps and the system
Pumping operating modes

Variable speed:
- Variable speed operation increases the flexibility of the operation of the pumps
- Increase the global efficiency significantly
- Ensures an operation close to the best efficiency conditions of the pump
- For network with variable operating conditions, the payback may be reached rapidly

The speed variation can be obtained with different drives:
  - Electrical motor with full size frequency converter
  - Diesel or gas engine
  - Gas turbine