Chapter 6. Hydraulic pumps

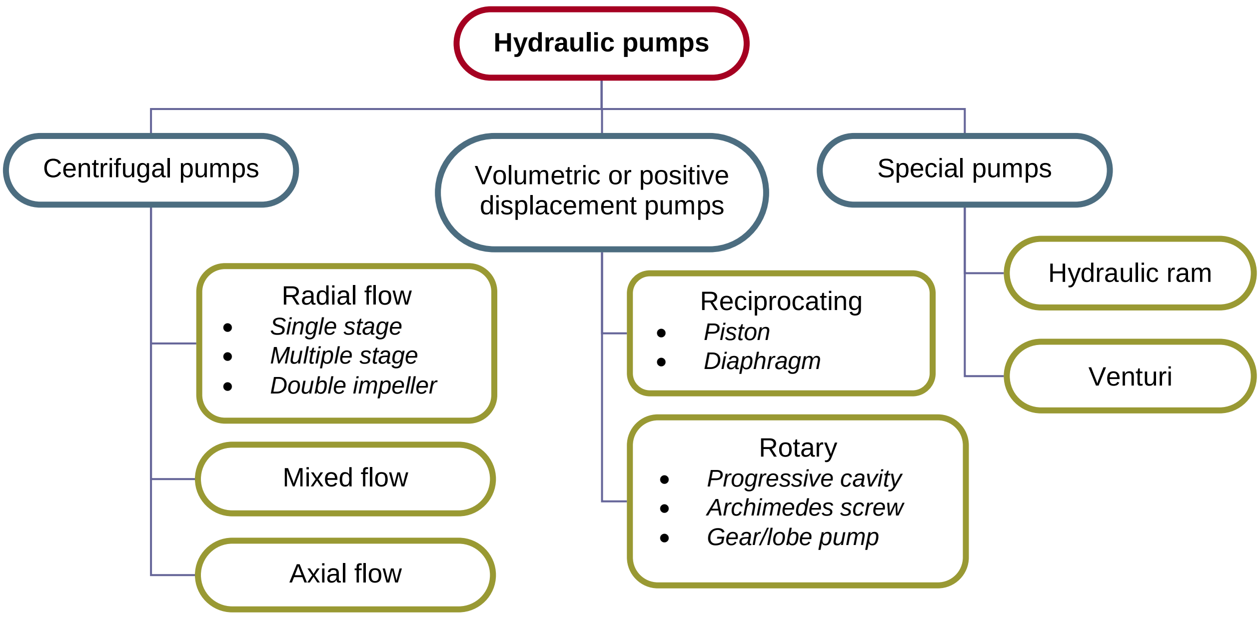
What you will learn
Understand the basic on hydraulic pumps and the specific way how the centrifugal pumps are working.

Why
Centrifugal pumps are an essential part of most of water supply systems. The good understanding of their characteristics and functioning is essential to assure that the best option is selected and ensure efficiency and sustainability.

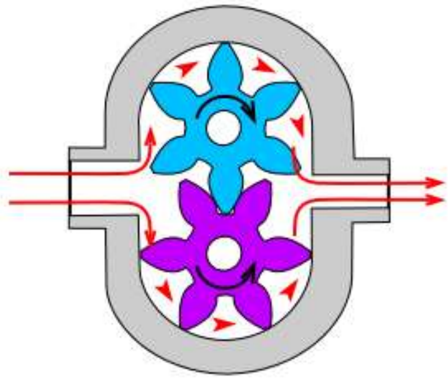
Duration of chapter 6
3 to 4 hours

*6.1. Types of hydraulic pumps **

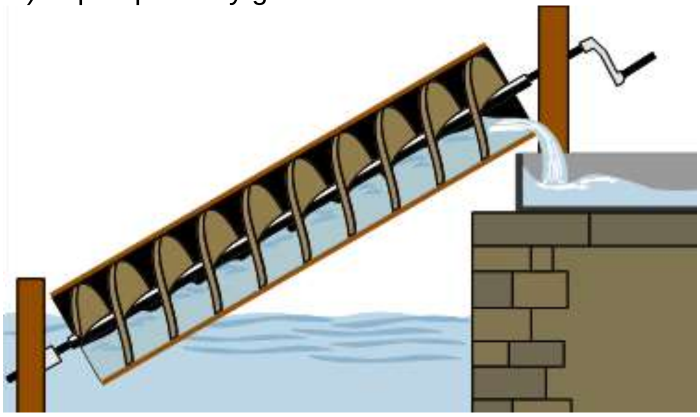
The role of a pump is to transform mechanical power into hydraulic power by increasing the pressure of the pumped fluid or forcing a flow. A huge variety of pumps exists for a wide range of specific applications. They are usually subdivided in three categories, the centrifugal pumps are kinetic machine in which energy is continuously transmitted to the fluid by a rotating impeller, the positive displacement pump moves the fluids by trapping a fixed amount and forcing it mechanically into the discharge and the special pumps are the types not fitting in the two first categories.



In water systems, most of the pumps used for transport are centrifugal (with the exception of rotary progressive pumps used for low flow and relatively high head specification like borehole pumping with solar power) and for dosing (in treatment) are reciprocating pumps (piston or diaphragm depending on the fluid) where a small accurate flow is needed at a high pressure. Special pumps include the hydraulic ram, which is a machine working without motor, using the energy of a part of the water, released at the pump level, to raise the rest of the water at a higher level; and the Venturi pump using the Venturi effect (see §3.2) to pump mainly gas.

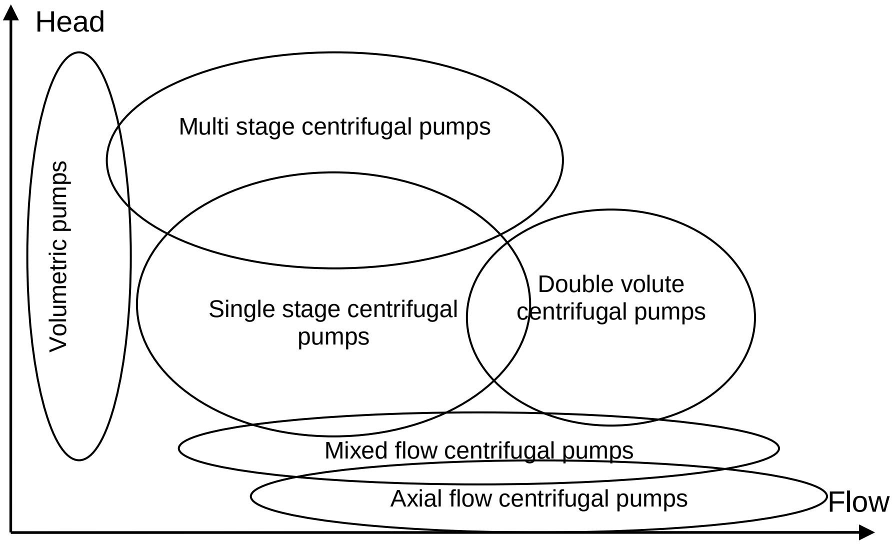


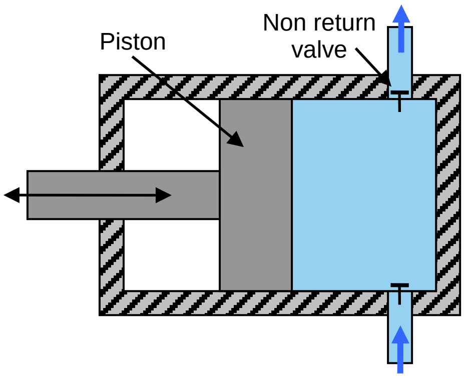
Gear pump



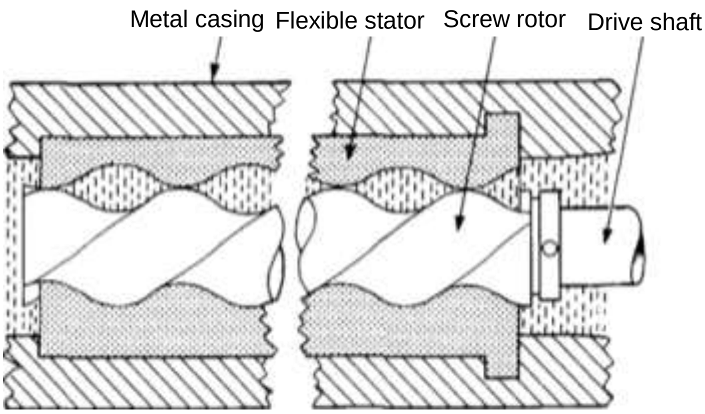
Archimedes screw

Centrifugal pumps are usually divided in three groups characterized by their shape (radial, mixed and axial flow), the radial are themselves subdivided in three groups (single stage, multistage and double impeller or volute) defining their working range. The following figure represent the range were those different pumps are used.





Piston pump



Progressive cavity pump

In some cases, additional characteristics should be specified such as:

- Position of the shaft (horizontal or vertical)
- Location of the pump (dry or submersible)
- Closed or open impeller
- Type of liquid to be pump (clean water, sewage, abrasive…)
- For boreholes (with specifically small diameter)

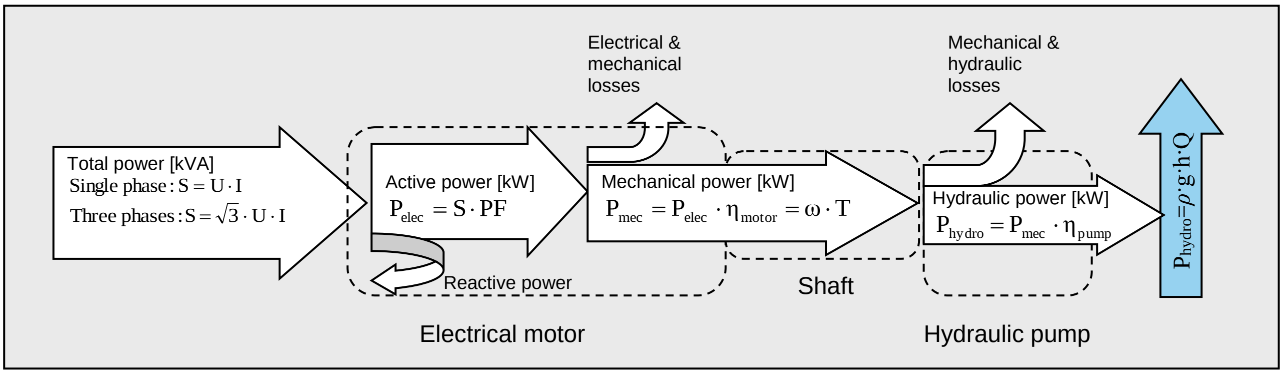
- Position in the system (suction, boosting)
- Number of poles (2,4 or 6 defining the rotation speed)
- Type of seal (mechanical or packing)
- Self priming
- Etc

In this chapter we will first study the power and efficiency, a common aspect for all pumps, and then we will deepen mainly the cases of centrifugal pumps as they are the most common pumps for water supply and have quite special characteristics depending on many factors correlated.

6.2. Power and efficiency *

The pump will transform the mechanical power into hydraulic power by increasing the pressure of a certain quantity of water. This increase of pressure is known as the manometric head, or simply the head of the pump written h (expressed in meter). This mechanical power is usually provided by an electrical motor but can also come directly from human action (hand pump), from the wind or combustion engine. This chapter will mainly focus on the pump driven by electrical motor, but similarity can be done for combustion engine; however, the efficiency is much lower.

The next figure represents the flow of energy (power) with the different losses from the electrical motor through the shaft to the pump where it "powers" the water.



The transmitted power is decreased at each step and can be listed as follow from the biggest to the smallest:

- **Electrical power**: is the electrical power consumed by the motor.

S: total (apparent) power [VA]
U: electrical tension [V]
I: electrical intensity [A]
P_{elec}: active (true) power [W]
PF: power factor or cos phi

$$\text{Single phase}: S = U \cdot I$$
$$\text{Three phases}: S = \sqrt{3} \cdot U \cdot I \quad \textbf{Eq. 6-1}$$
$$P_{elec} = S \cdot PF = S \cdot \cos(\varphi)$$

- **Mechanical power**: is the power transferred from the motor to the pump by the shaft.

P_{mec}: mechanical power [W]
η_m: motor efficiency [-]
ω : rotation speed (rad/s)
T : Torque [N/m]

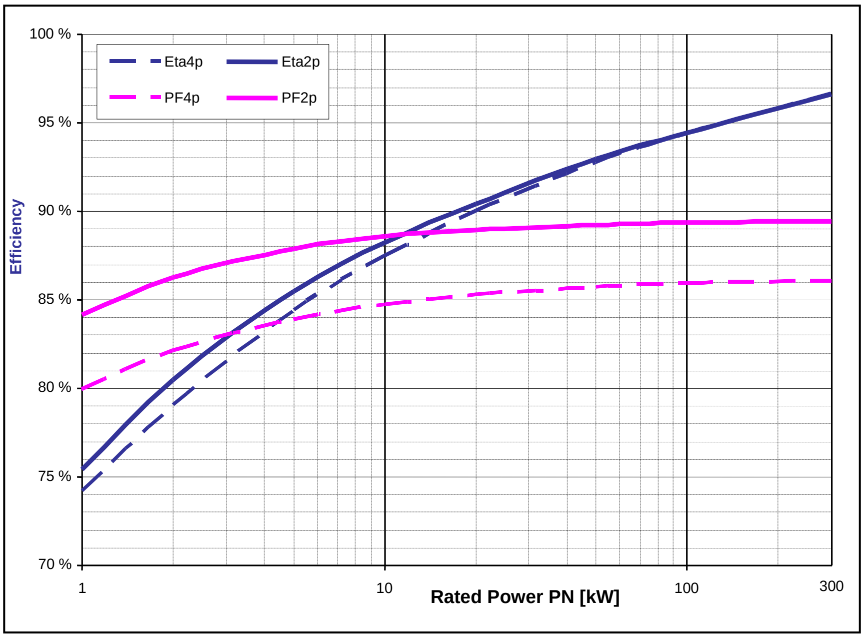
$$P_{mec} = P_{elec} \cdot \eta_m = \omega \cdot T \quad \textbf{Eq. 6-2}$$

- **Hydraulic power**: is the power transmitted to the water in the pump.

P_{hdro}: hydraulic power [W] η_p: pump efficiency [-] ρ: water density [kg/m3] g: earth gravity [m2/s] h: manometric head [m] Q: flow [m3/s]	$$P_{hydro} = P_{mec} \cdot \eta_p = \rho \cdot g \cdot h \cdot Q$$ $$P_{mec} = \frac{\rho \cdot g \cdot h \cdot Q}{\eta_p}$$ **Eq. 6-3**

Thus, the whole system can be written as in the next equation, allowing estimating the necessary current. NB According to Eq. 6-1 the $\sqrt{3}$ should be removed for single-phase motors.	$$S = \sqrt{3} \cdot U \cdot I = \frac{\rho \cdot g \cdot h \cdot Q}{PF \cdot \eta_m \cdot \eta_p}$$ **Eq. 6-4**

Rough evaluation of the pump and motor efficiency and power factor:



For electrical motor:

The efficiency of a motor at its rated power mainly depends on its size and number of poles (2p or 4p, cf §6.3).
The power factor (PF) varies according to the same values.

In the attached chart expectable efficiency for commercial motors are given according to these two values.

It is possible to ask for higher efficiency motors, but they are more expensive especially for small power.

Fig 6- 1 Efficiency versus rated power

For centrifugal pumps:

The efficiency at the optimal working point of a centrifugal pump varies a lot. It mainly depends on the flow and the specific speed (cf. §6.5).

In the attached chart expectable efficiency for commercial pumps are given according to these two values.

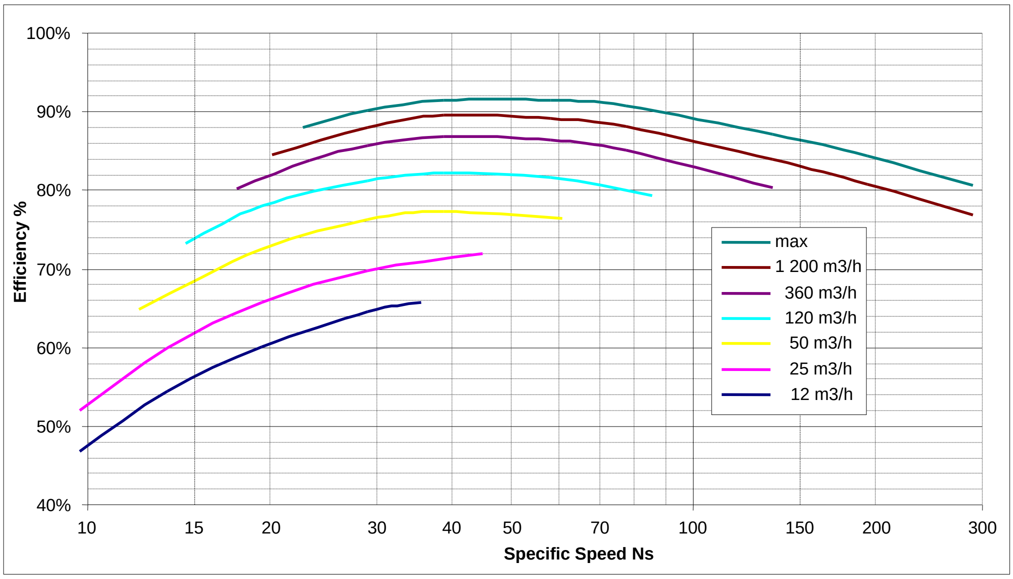


Fig 6- 2 Efficiency versus specific speed

6.3. Electrical motor rotation speed

The big majority of electrical motors used to pump water are of asynchronous type (induction or squirrel cage). A magnetic field is rotating in the stator at a speed depending on the number of pair of poles and frequency of the power (50 or 60 Hz), this is called the synchronous speed. The rotor (rotating part of the motor) is driven at a slightly lower speed depending on the torque applied on it. The difference between the actual rotation speed and the synchronous speed is called the slip and is usually quite small, from 2% for big motors to 10% for small motors (single phase). The rotation speed for motor is measured in rotation per minutes and can be calculated according to the following equation:

n: rotation speed [RPM]
f: frequency of power [Hz]
nbPoles: number of pair of poles
slip in RPM

$$n = 60 \cdot \frac{2 \cdot f}{nbPoles} - slip$$

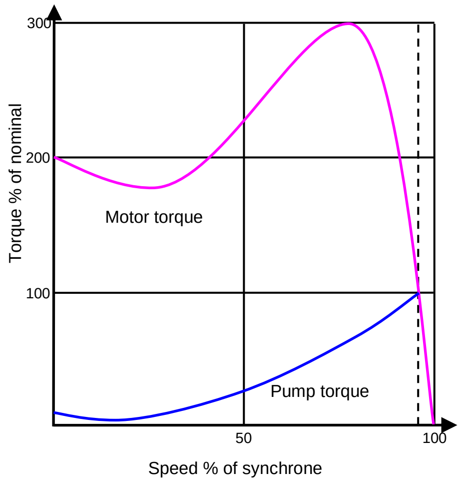
Eq. 6-5

n [RPM]	**Number pair of poles**		
Frequency	2	4	6
50 Hz	2 900	1 450	960
60 Hz	3 500	1 750	1 160

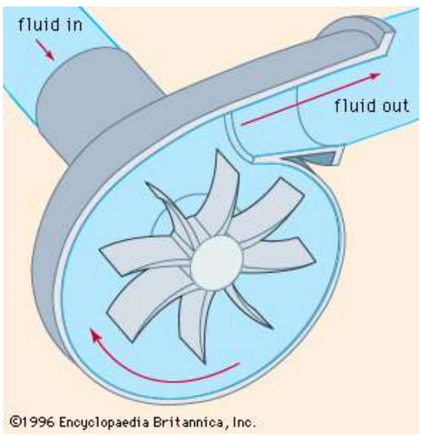
The torque provided by a motor varies according to its speed. A stopped motor (3 phase squirrel cage) will have a starting torque of twice its nominal torque. It will then decrease a bit and then, it will rise up to a maximum called "breakdown torque" around three times its nominal torque. After that, it has almost a linear behaviour between the torque and the slip up to the synchronous speed where the rotor turns as fast as the magnetic field in the stator.
Therefore the rotation speed of a motor will slightly change according to the load, for a motor with a nominal speed of 2900 rpm the following speed can be expected :

Load	0%	50%	100%	200%
Speed	≈3 000	2 950	2 900	2 800

NB if the torque is higher than its nominal value for more than a short time, the motor will start to overheat and will burn after a while. This can be checked with its actual speed if it is lower than its nominal speed.



6.4. Characteristics of centrifugal pumps *



In a (radial) centrifugal pump, the water is introduced at the centre of a chamber in which an impeller is giving a rotating movement to the water forcing it to the periphery of the chamber and increasing its pressure. There, the water is collected in a volute and then through the outlet.
The larger the impeller is or faster it turns, the higher will be the pressure. The flow going through the pump will change dramatically according to the difference of pressure between the inlet and the outlet. This difference of pressure is called the head of the pump.

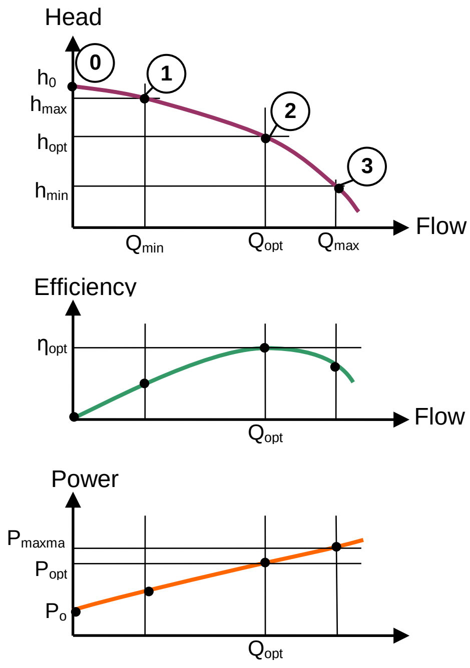
Head versus flow :

Below, the typical behaviour for a radial centrifugal pump is described. For axial pump see § 6.5.

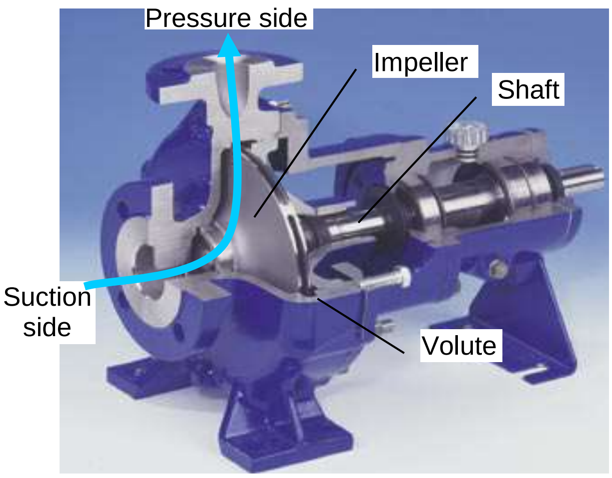
0 A specific pump can increase the pressure only of a certain value (h_0), once this value reached, the water will not move out any more and it will work as a "washing machine" (water is staying inside the pump). If the pump works too long in such conditions, the water will heat and may damage the pump. As there is no flow, the efficiency is nil; the power consumption is at its minimum.

1 If the head is decreased, rapidly the flow will increase to reach conditions where it is safe to run the pump; however, it will still be with low efficiency. This point is usually given as the Q_{min}, h_{max} point.

2 By further decreasing the pressure, the flow will continue to increase as well as the efficiency, until it reaches the optimal point (Q_{opt}, h_{opt}, η_{opt}, P_{opt}) where the water path is well aligned with the blades of the impeller, this is the working point for which the pump was designed.



HQ characteristic of radial centrifugal pump

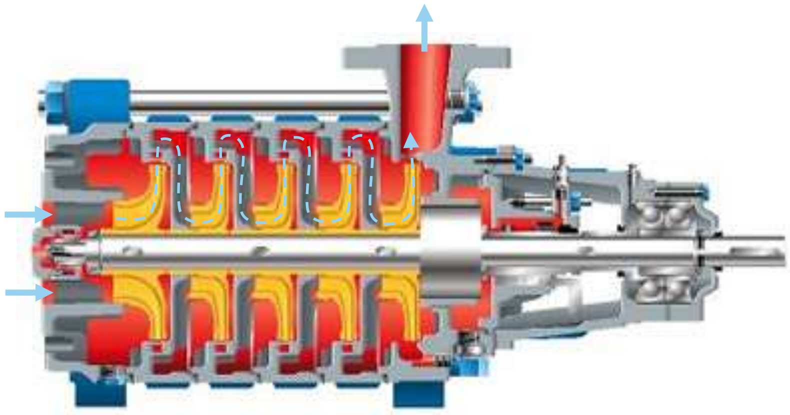


Single stage pump (Etanorm © KSB)

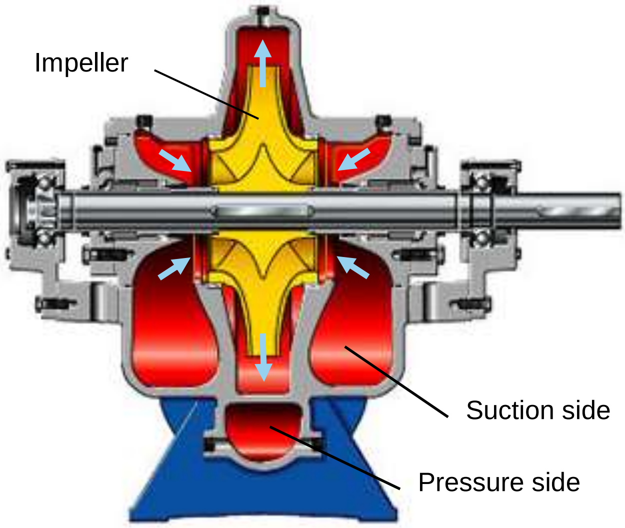
3 Decreasing further the pressure, the flow will still increase but not as fast as before the optimal point, the efficiency will start to decrease, until the maximum permissible flow and minimum permissible head is reached Q_{max}, h_{min} The power request is at its highest level.

Maximum power demand will be at the lowest head and biggest flow

Consequences of having a smaller head might be quite serious and will be explained in § 6.9.



Multi stage pump
(5 stages Multitec © KSB)



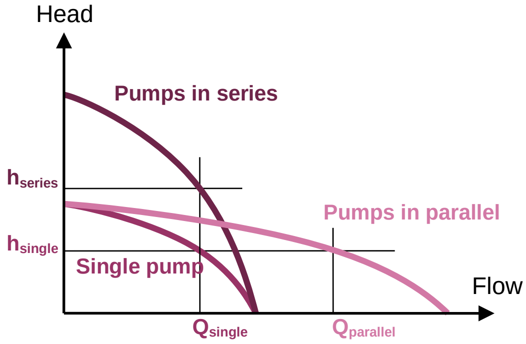
Double volute pump (Omega © KSB)

If a high head is needed, several impellers can be placed in series, summing there characteristics in the same way as pumps in series.

If a large flow is needed, two impellers can be placed in "mirror" two double the flow with the same head in the same way as two pumps in parallel

When centrifugal pumps are connected in series, the head is added for a given flow as represented in the next graph. The efficiency is the average of the initial ones. It is preferable to put in series pumps with the same nominal flow to be able to let both of them work at the optimum flow.

When centrifugal pumps are connected in parallel, the flow is then added for a given head as represented in the next graph. It is preferable to put in parallel pumps with the same nominal head to be able to let both of them work at the optimum flow.

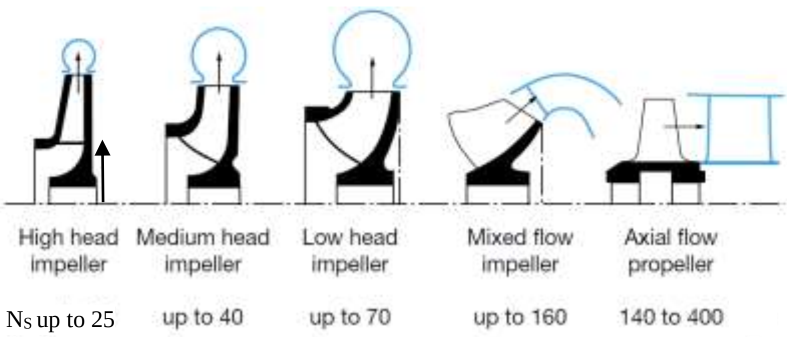


6.5. *Specific speed (N_S)*

Impellers of centrifugal pumps can be classified according to their proportion. Two impellers with similar proportions but different sizes will have similar characteristics. This proportion is expressed thanks to a number called *Specific speed* (N_S) and defined as per equation 6-6.

$$N_S = n \cdot \frac{(Q)^{0.5}}{(h)^{0.75}} \quad \textbf{Eq. 6-6}$$

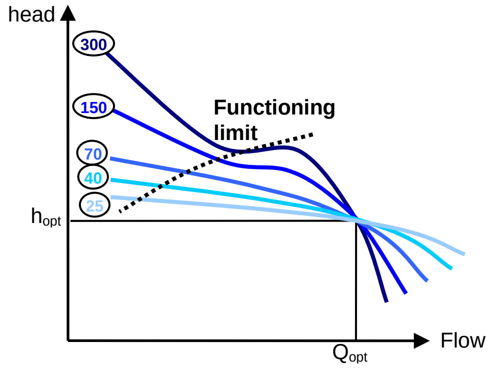
N_S: Specific speed
n: rotation speed [rpm]
Q: nominal flow [m^3/s]
h: nominal head [m]



NB the specific speed should always be calculated for one impeller. In case of multiple-stage pump, the head should be divided by the number of stages and in case of double-volute pumps, the flow has to be divided by two.

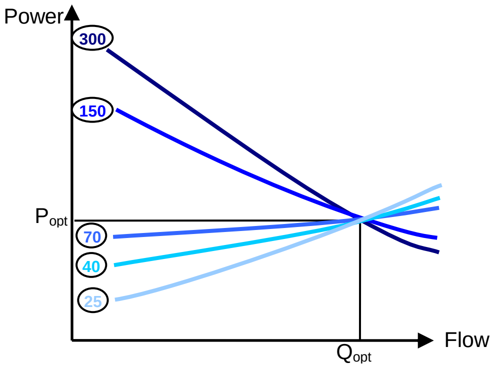
The following charts show the comparison of characteristics of pump with different specific speed.

High head impellers (N_S up to 25) have their outlet diameters more than 4 times bigger than their inlet diameters and have a narrow "channel" (where the water is flowing inside the impeller). Their efficiency are quite bad (cf §6.2) as there is many friction losses in the narrow channel, but the efficiency curves is quite round; it decreases not too quickly around the nominal point. Their HQ curves are very flat, making them interesting to keep a constant pressure with different flow, as in a distribution system – for example for a boosting pump for a house, its maximum pressure is limited and will not decrease too much when many taps are open.



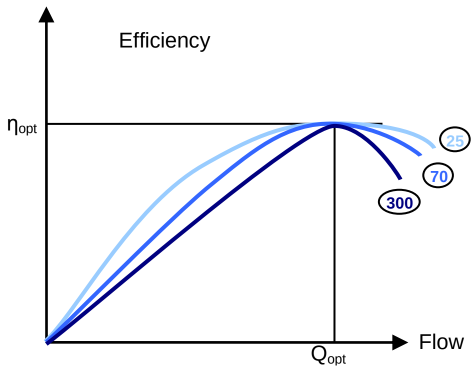
The H-Q curve, is very flat for small N_S and become steeper for bigger one.

Low head impellers or Francis vanes (N_S up to 70) still have a radial operation (water going out of the impeller perpendicularly to the axe) but the outlet diameter is only 1.5 to 2 times the inlet diameter. Its large channel reduces the friction losses giving a very good efficiency. The HQ curve becomes steeper making it interesting to keep a steady flow with head variations as in transport system – for example, a borehole pump, where the level might change with the seasons but the flow should be as steady as possible.



The P-Q curve is increasing for radial pump and decreasing for mixed and axial flow pumps.

Axial flow impellers have an axial operation, pushing the water as a motor boat propeller, their outlet and inlet diameter are equal. This working mode increases the sensibility to hydraulic losses (vortexes) decreasing also slowly the efficiency. This sensibility reduces also the working range and the efficiency curve is quite sharp, it decreases quickly around the nominal point. Problems of cavitation and vortex appear for high head and low head. Unlike the radial pump, the axial pumps will demand less power for a big flow than for a small one.



The η-Q curve is round for small N_S and become sharper for bigger one.

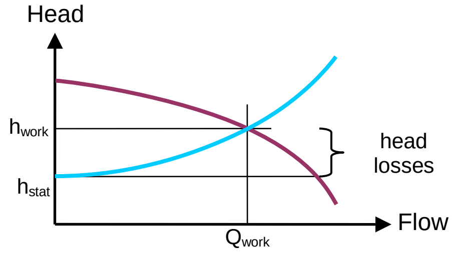
Mixed flow impellers are a mix between the axial and the radial pumps, with properties in between as it can be seen for the curves with a N_S of 150.

For a given head and flow, lower specific speed pump will be larger and more expensive than higher specific speed pump, but cavitation, vibrations and wears will be lessened.

Action	Effect	Result on the N_s
Use 1450 instead of 2900 rpm motor	Reducing the velocity	Divided by 2
Use double volute (impeller) pump	Reduce flow per impeller	Reduced by 30%
Use two stages pump	Reduce head per impeller	Increased by 68%
Trim an impeller	Reduce head & flow	Increasing slightly

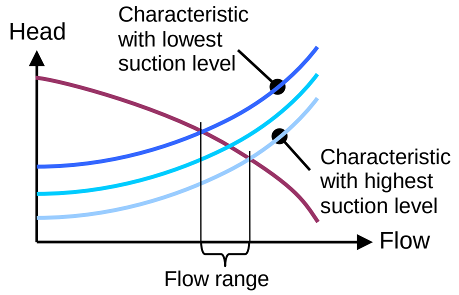
6.6. *Defining the duty point and working range **

As the pump, the water system (pipes & fittings) on which the pump is working has its HQ characteristic; the working point will be the intersection of these two characteristics as shown in the next chart.
The system characteristic is composed of the addition of the static head and the head losses for the different flows:
The **static head** is the difference of altitude in meter between the average level of the downstream and upstream free water surface. On the suction side, the length of the pipe below the water level should not be taken into consideration.
The **head losses** depend on all linear and punctual losses. In this calculation, the length of the pipe below the water has to be taken into consideration and calculated as seen in chapter 4.

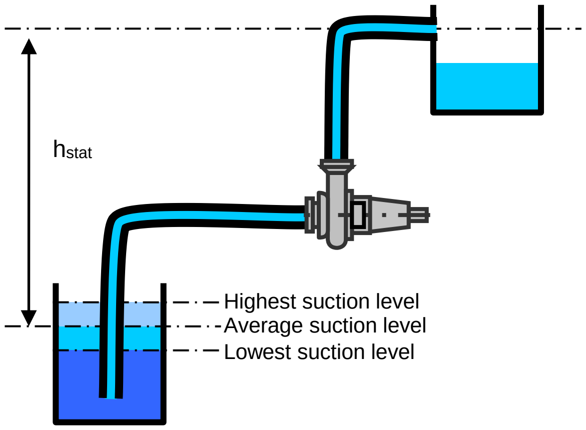


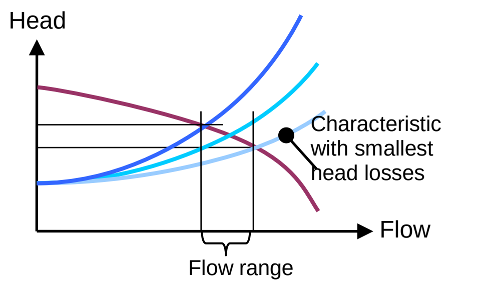
Working point is the intersection between water system and pump characteristics.

The hydraulic condition of the system might change with time: the water level at the suction (or at delivery) might vary, modifying the static head. The temperature of the water might change with hot and cold season or the roughness of pipes might change with time, modifying the head losses.



Flow range with variation of suction level

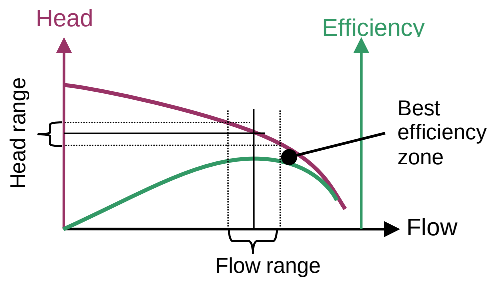




Flow range with variation of friction losses

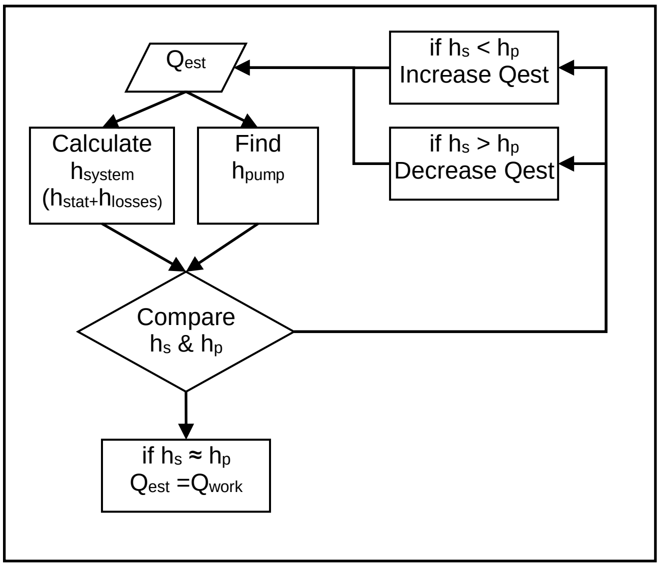
If these changes are significant, a working range should be defined for the pump. It should be checked that the pump is working safely throughout the whole range.

Duty point: once the head range (flow range) is found, the duty point can be selected in the middle of the range. This duty point should be as close as possible to the nominal flow of the pump where the efficiency is at its best.
If the flow range is too far from the nominal point, adjustment of the duty point should be done.



It is good to adapt the pumping working hours in order to have the duty point as close as possible to the optimal point of the pump.

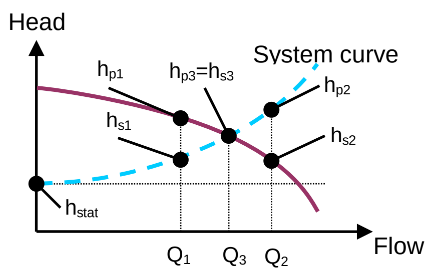
Necessary iteration to find the working point:
To calculate the head losses, the flow should be known, but the flow will be only known once the actual head losses are defined.
Therefore, a flow should be first estimated then the head of the system (static head plus losses) can be calculated and compare to the head of the pump. If head of the system is smaller, a bigger flow should be taken and vice versa, if the head of the system is bigger, a smaller flow should be taken.
Two or three iterations are usually enough to find a precise result.



Concretely, this can be done as follow:

1) Estimate a flow (Q_1), usually chosen as the nominal flow of the selected pump and find the corresponding head on the HQ pump characteristic (h_{pump})
2) Find the total head of the system for this flow, either by
   a. using graph in the annexes for head losses for pipe under pressure to estimate laminar friction losses and add punctual losses to them $$h_{LP} = h_L + k_P \cdot \frac{v^2}{2g}$$
   b. using formulas from chapter 4 to estimate λ, and therefore $$h_{LP} = \left(\lambda \cdot \frac{L}{D} + k_P\right)\frac{v^2}{2g}$$

   Then add the static head to the head losses to find h_{system}
3) Compare h_s & h_p
   a. If $h_s \approx h_p$ you have found the working flow, the process is completed.
   b. If $h_s > h_p$ you have you have to take a smaller flow
   c. If $h_s < h_p$ you have you have to take a bigger flow (in the example Q_2)
4) Place h_{stat} and h_{s1} on the pump characteristic to sketch the system curve as seen in § 4.9
5) Calculate the new total head for the system with Q_2.and plot it in the system curve. Find the intersection between the system and the pump characteristic, this should be the working flow (Q_3).
6) Check that the head of the system equal the head of the pump for Q_3, if it is not precise enough (3%) make additional iteration.



6.7. *Adjustment to duty point*

Pumps used for water supply systems are usually too small to be custom made. Therefore, to fit a given duty point, standard pumps must usually be adjusted. As represented in the next figures, standard pumps are made at a certain interval of flow and head covering a certain working range.

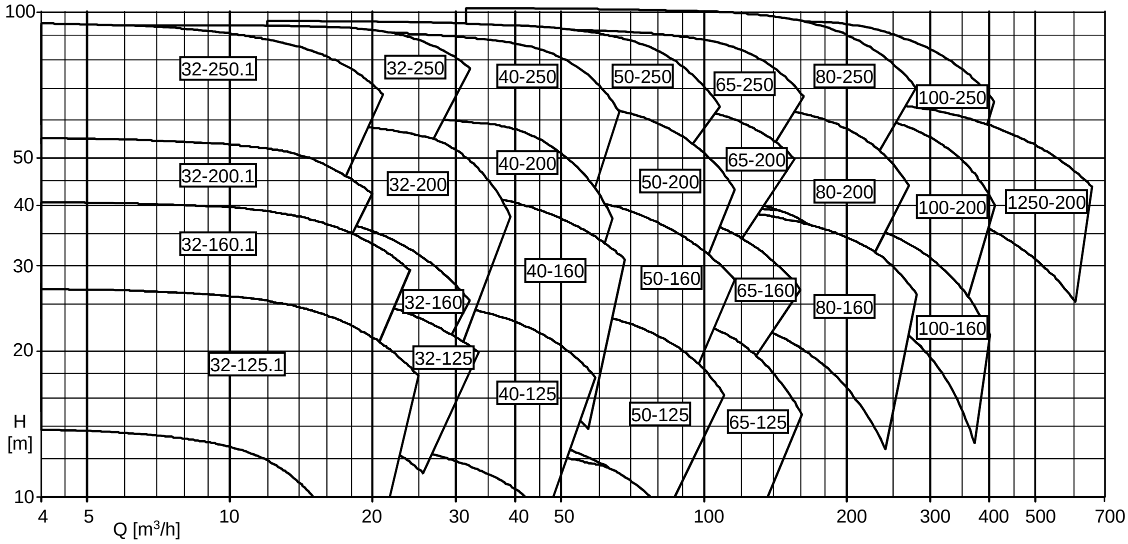
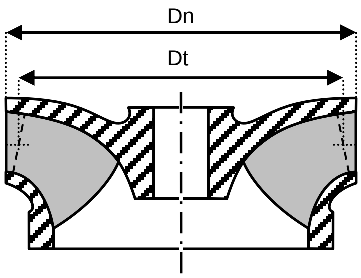


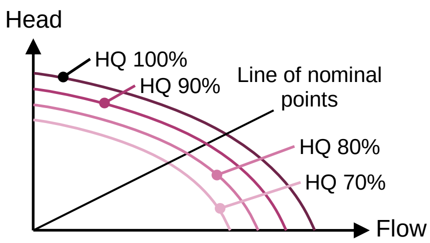
Fig 6- 3 Example of working range (KSB Etanorm pumps)

Trimming

To reduce permanently the head and the flow of an impeller, its external diameter (outlet of the blades) can be trimmed, reducing the outlet velocity of the fluid. In order not to affect performance, it can be done generally to a maximum of 80% of the initial diameter.

This method to adjust the duty point keeps a good efficiency, but it has to be done in the factory and is obviously irreversible and not possible with all types of impeller.





$$Q_t = Q_n \cdot \left(\frac{D_t}{D_n} \right)^2$$

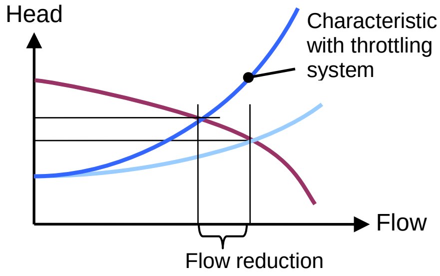
$$h_t = h_n \cdot \left(\frac{D_t}{D_n} \right)^2$$

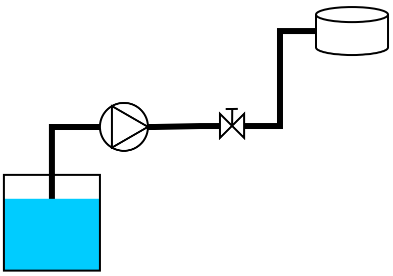
$$P_t = P_n \cdot \left(\frac{D_t}{D_n} \right)^4$$

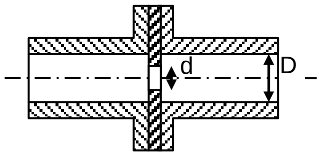
Eq. 6-7

Throttling

For **radial centrifugal pumps,** when the flow is slightly too high, it is possible to adjust it with a throttling system (orifice plate and or a globe valve). This will increase the losses on the system making its characteristics steeper.







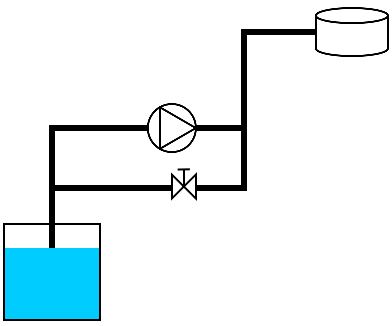
Installation of an orifice plate to be designed according to § 4.6

This is done for radial centrifugal pumps as it will decrease the needed power and save a bit of energy (cf §6.5).
In this case, the flow can even be smaller than the nominal flow of the pump.
This system has the big advantage of being adjustable and simple to install, well adapted when there is a big variation of the inlet level.

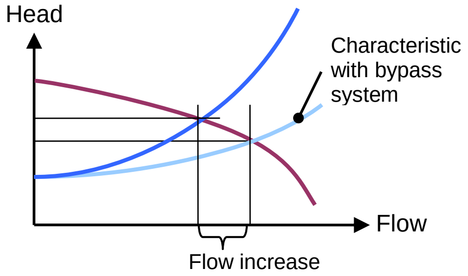
Bypassing

For **mixed or axial flow pump**, when the flow is slightly too small, it is possible to adjust it with a bypass system. This will make its characteristics less steep, increasing the flow.

This is done for mixed or axial flow pumps as it will decrease the needed power and save a bit of energy (cf §6.5).
In this case, the flow can even be higher than the nominal flow of the pump.
This system has the big advantage of being adjustable and simple to install, well adapted when there is a big variation of the inlet level.



This system should not be used for radial centrifugal pumps.



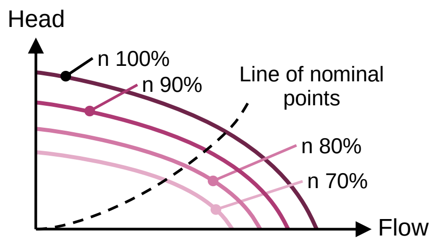
Speed reduction

By reducing the rotation speed of the pump, its flow is modified by the ratio of the speeds, its head by the square of this ratio and the power by the cube.
The nominal point is then changed accordingly and the efficiency for this new nominal point is almost not affected.
This is clearly the best way to regulate a pump but not the easiest to implement.

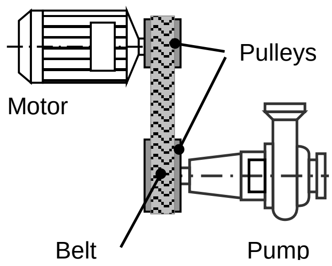
$$Q_2 = Q_1 \cdot {n_2}/{n_1}$$
$$h_2 = h_1 \cdot \left({n_2}/{n_1} \right)^2$$
$$P_2 = P_1 \cdot \left({n_2}/{n_1} \right)^3$$

Eq. 6-8

Q: flow [m^3/s]
h: head
P: power [W]
n_1: initial rotation speed
n_2: reduced rotation speed

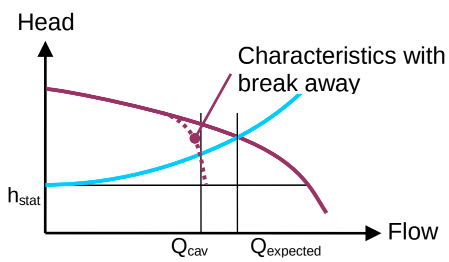


There are two ways to reduce the rotation speed:
The first way is by adding pulleys and a belt between the motor and the pump; it is rather simple, but not very good for bearings. It is less and less used.
The second one is to use an electronic device that will reduce the power frequency, thus the rotation speed according to Eq 6-5. These devices become cheaper but are still quite expensive for big power. As integrated electronic devices, they cannot be repaired and might be sensitive to low quality of power. Therefore, it is important to have a supplier locally available to change them when they are broken.

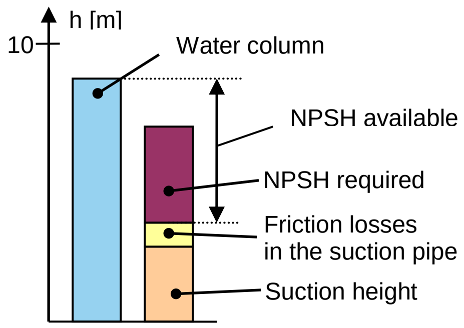


6.8. Cavitation in a pump and NPSH

The phenomenon of cavitation should absolutely be avoided in a pump; as seen in the chapter 2.3, if in any place in the pump the pressure drops below vapour pressure, steam bubbles will be created and when they will implode, it will destroy the impeller, the pump casing and or the delivery pipe. On top of this, the characteristic curve will be affected, falling suddenly, reducing the actual pumped flow; this phenomenon is known as breakaway as represented in the attached figure.



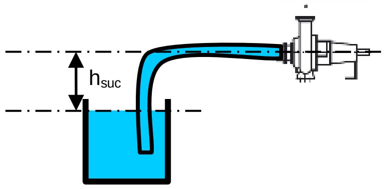
Three main factors will have effects on the decrease of the pressure in the pump: the suction height, the head losses in the suction pipe and the geometry of the pump (directly linked to the NPSH$_r$). The addition of this three "heights" should not exceed the water column height (as defined in §2.4), depending on the temperature and the altitude.
Thus for a given water system, the difference between the water column and the suction height plus the friction losses will give the maximum "height" available for the pump. This height is known as the **Net Positive Suction Head (NPSH) available**.



$NPSH_a$: NPSH available [m]
P_a: minimum atm pressure [Pa]
P_v: vapour pressure [Pa]
h_{LP}: hydraulic losses [m]
h_{suc}: suction height [m]

$$NPSH_a = \frac{P_a - P_V}{\rho \cdot g} - h_{LP} \pm h_{suc}$$

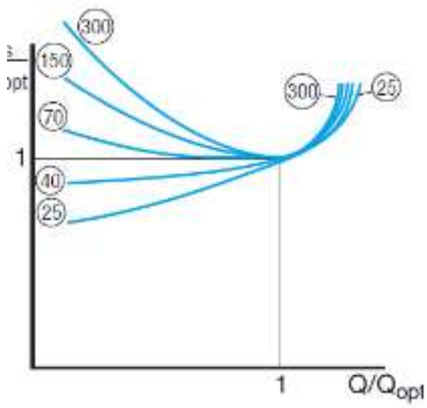
Eq. 6-9



The suction height is considered positive if the pump is placed below the water level.

The "height" needed by the pump is called **NPSH required**. As said, it depends on the geometry, as in a thinner channel the velocity of the water will be higher, thus the pressure lower, increasing the risk of cavitation. The rotation speed is therefore also very important, a small two pair of poles pump (n~2 900 rpm) will be more subject to cavitation than a bigger and more expensive four pair of poles pump (n~1 450 rpm).

As radial centrifugal pump have a good behaviour on cavitation at flow lower than their nominal points, axial flow pumps face cavitation due to formation of vortex in this situation as represented in the attached chart.



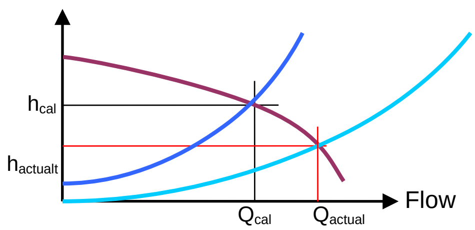
The calculation of the NPSH available should be done for the minimum and maximum suction water level, considering maximum losses in the suction pipe and minimum losses in the delivery pipe. If the NPSH available found is lower than the NPSH required by the pump, it should be considered to lower the elevation of the pump or to use an other type of pump.
Submersible pumps might also be subject to cavitation.

6.9. Consequences of working over the maximum flow *

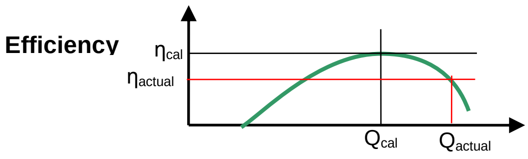
Most of the pumped used in the water supply system are radial centrifugal pumps (Ns<100). Therefore, it is essential not to over estimate losses and head as the following might happen.

By having, a calculated head higher than the actual head, the actual flow will be bigger.

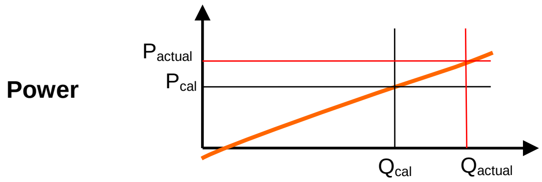
Head



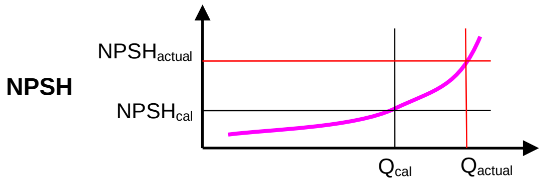
The calculated flow is usually at the optimal point, therefore with a bigger flow, the actual efficiency will be lower and energy will be wasted



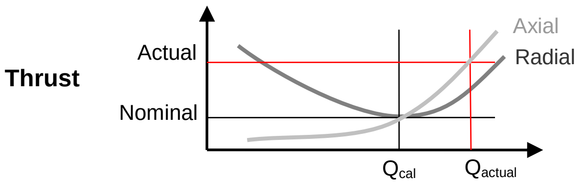
As the power will be higher than the rated one, if the nominal power of the motor is not big enough, there is a risk of overloading and over heating the motor and other electrical devices (genset, transformer, weirs, etc).



With the increased flow, the NPSH required will be higher, and if the suction head is too high, pump will have cavitation, wasting energy and damaging the impeller and pipes



Out of the optimal point, the axial and radial thrust are increased, this accelerate the wearing of the bearings and increase vibrations



With the increased velocity, the water hammer effect will be stronger and might affect the system, especially with steel pipes.

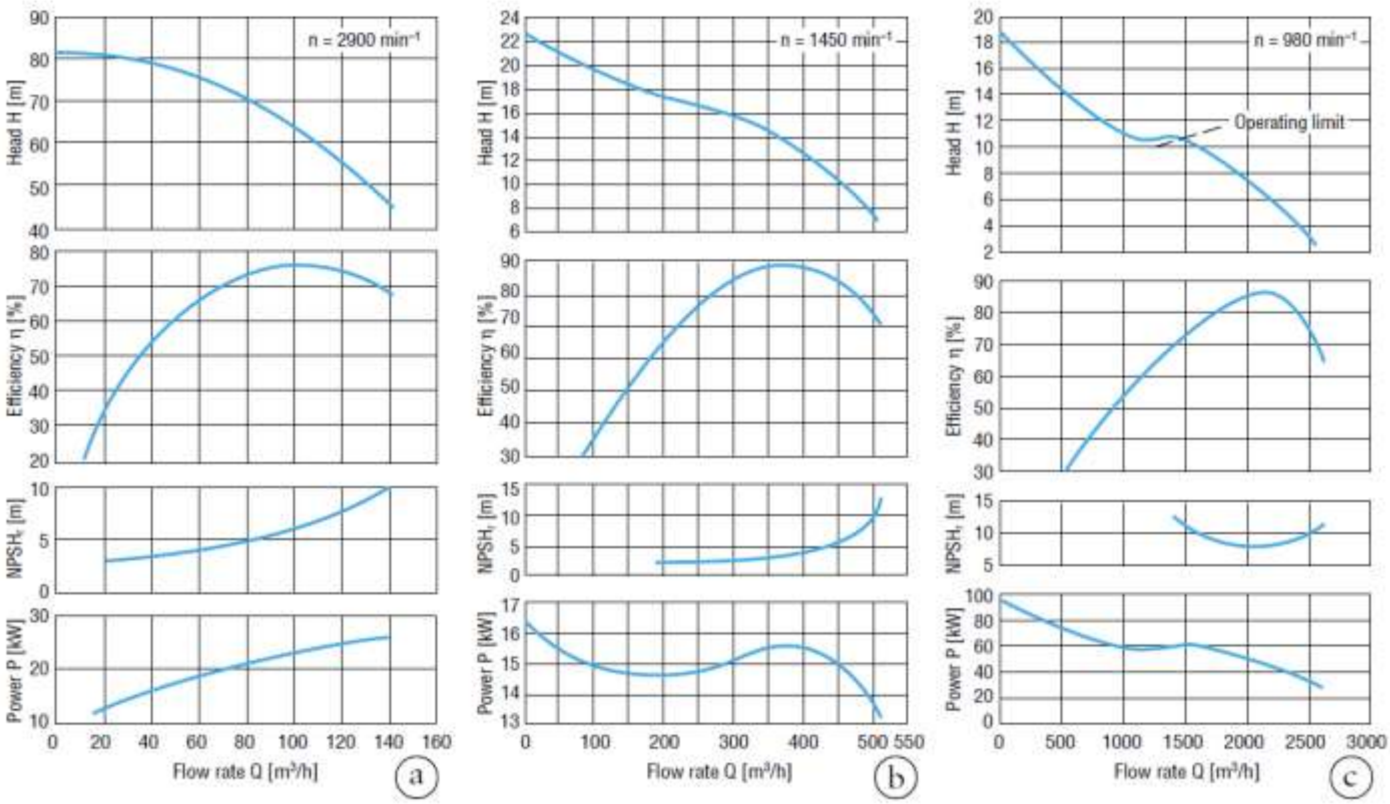
Therefore, to avoid these problems, a centrifugal pump has to be designed at its average working point and checked over the full range. When feasible, the desired flow should be adapted to existing pump nominal flow and the daily working hours adapted accordingly. There is no values that should be increased to be on the "safe side" but the suction height, which should be as small as possible to avoid cavitation.

References for this chapter

Grundofs : Pump Handbook :
KSB : Selecting Centrifugal Pumps

Basic exercises

1. For a pump of 50 m^3/h at 40 m head and with a 2 pair of poles motor (N_S≈20), what is the expected hydraulic power, pump efficiency, mechanical power, motor efficiency and power factor, active and total electrical power?
2. What are the nominal head, flow, efficiency, NPSH and power for the pumps a, b, and c?



3. What are the expected minimum and maximum flow for the pumps a, b, and c?
4. For the following system, what will be the duty point with the pump a, what is the power consumption?

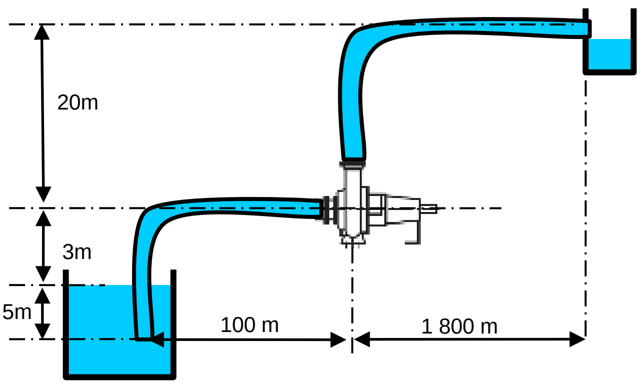
   Elevation : 200 masl

   Temperature 20°c

   All pipes are of new GI, DN 175

   Total punctual friction losses:

   k_p = 15 for suction part
   k_p = 5 for delivery part



Intermediary exercises

5. What is the number of pair of poles and the slip value of the motor for the pumps a, b, and c?
6. What is the specific speed for the pumps a, b, and c?
7. If we want to adjust the working point of the pump used in exercise 4 to the nominal point with a throttling system, what should be the size of the orifice, what would be the power consumption?
8. With the same situation as in exercise 4, what would be the new flow and power consumption, if we decrease the rotation speed, so that it reaches 80% of its initial value?
9. If the system is working as in exercise 4, what is the maximum suction height? And what is the maximum suction height if it is working as in exercise 7?

Advanced exercises

10. For the system used in exercise 7 (with a throttling system), knowing that the intake water level varies of plus or minus 2 meters, what would be the max and min flow, what would be the max suction height?
11. With a frequency inverter, what should be the percent speed reduction to adjust the system to the nominal point? What is the corresponding flow?