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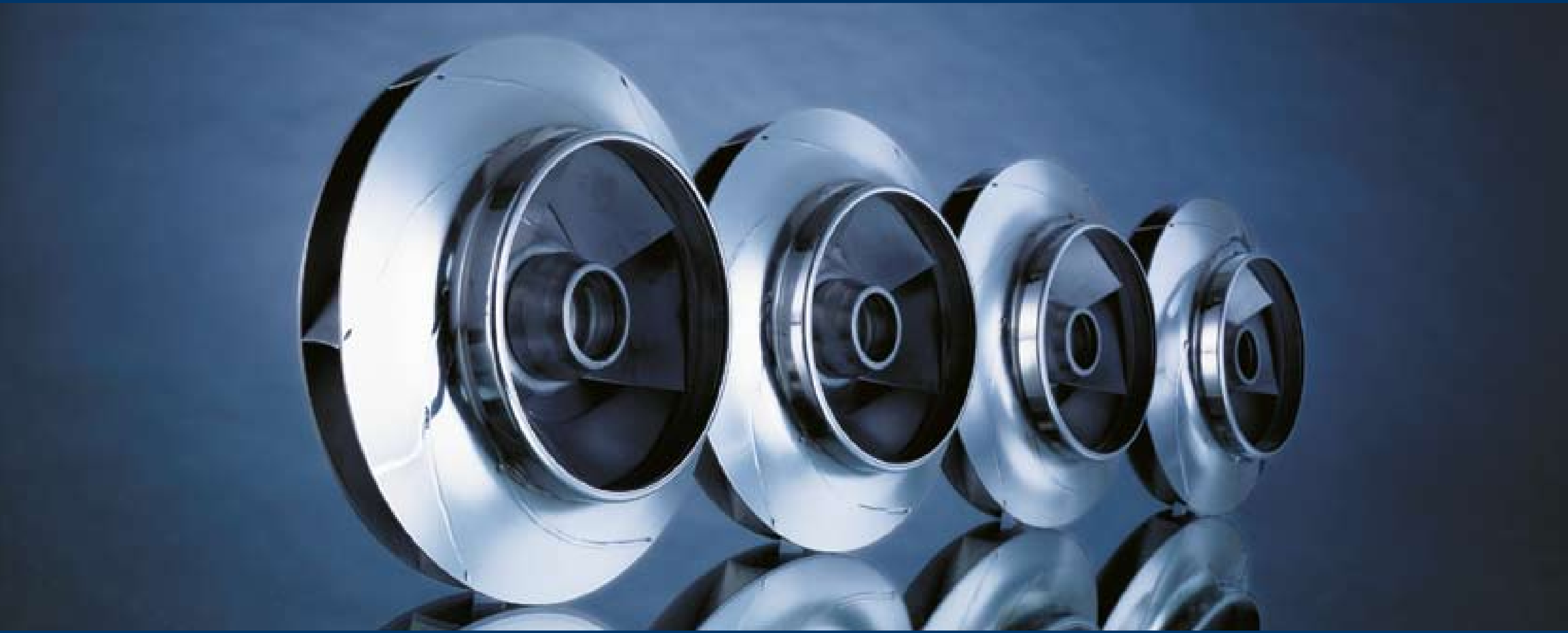

The Centrifugal Pump

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The Centrifugal Pump

Preface

In the Department of Structural and Fluid Mechanics we are happy to present the first English edition of the book: 'The Centrifugal Pump'. We have written the book because we want to share our knowledge of pump hydraulics, pump design and the basic pump terms which we use in our daily work.

'The Centrifugal Pump' is primarily meant as an internal book and is aimed at technicians who work with development and construction of pump components. Furthermore, the book aims at our future colleagues, students at universities and engineering colleges, who can use the book as a reference and source of inspiration in their studies. Our intention has been to write an introductory book that gives an overview of the hydraulic components in the pump and at the same time enables technicians to see how changes in construction and operation influence the pump performance.

In chapter 1, we introduce the principle of the centrifugal pump as well as its hydraulic components, and we list the different types of pumps produced by Grundfos. Chapter 2 describes how to read and understand the pump performance based on the curves for head, power, efficiency and NPSH.

In chapter 3 you can read about how to adjust the pump's performance when it is in operation in a system. The theoretical basis for energy conversion in a centrifugal pump is introduced in chapter 4, and we go through how affinity rules are used for scaling the performance of pump impellers. In chapter 5, we describe the different types of losses which occur in the pump, and how the losses affect flow, head and power consumption. In the book's last chapter, chapter 6, we go trough the test types which Grundfos continuously carries out on both assembled pumps and pump components to ensure that the pump has the desired performance.

The entire department has been involved in the development of the book. Through a longer period of time we have discussed the idea, the contents and the structure and collected source material. The framework of the Danish book was made after some intensive working days at 'Himmelbjerget'. The result of the department's engagement and effort through several years is the book which you are holding.

We hope that you will find 'The Centrifugal Pump' useful, and that you will use it as a book of reference in you daily work.

Enjoy!

Christian Brix Jacobsen
Department Head, Structural and Fluid Mechanics, R&T

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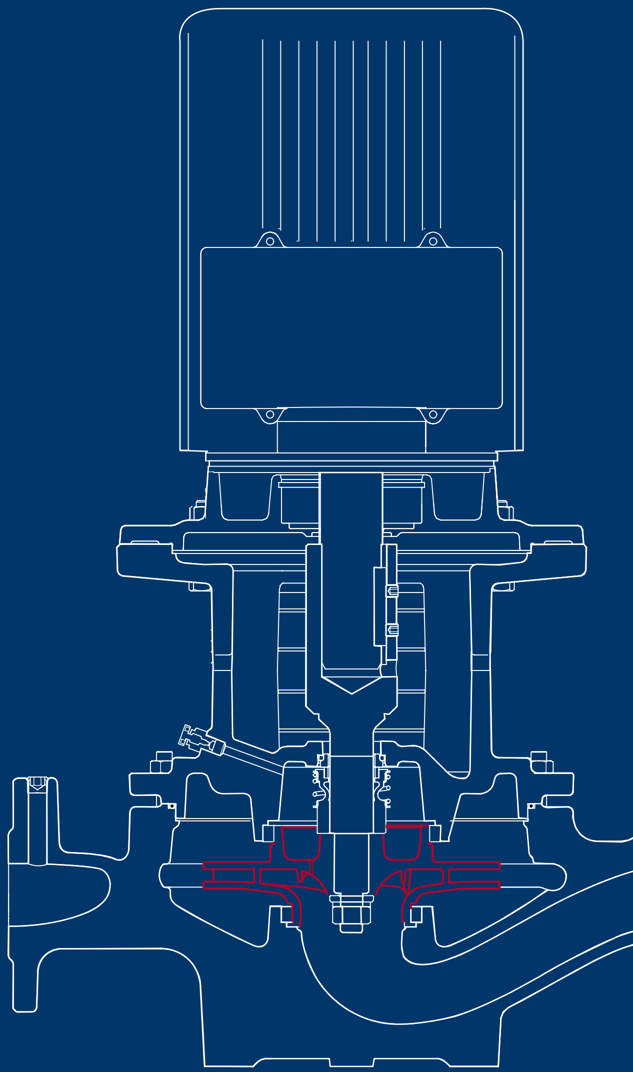
Chapter 1

Introduction to centrifugal pumps



1. Introduction to Centrifugal Pumps

In this chapter, we introduce the components in the centrifugal pump and a range of the pump types produced by Grundfos. This chapter provides the reader with a basic understanding of the principles of the centrifugal pump and pump terminology.

The centrifugal pump is the most used pump type in the world. The principle is simple, well-described and thoroughly tested, and the pump is robust, effective and relatively inexpensive to produce. There is a wide range of variations based on the principle of the centrifugal pump and consisting of the same basic hydraulic parts. The majority of pumps produced by Grundfos are centrifugal pumps.

1.1 Principle of the centrifugal pump

An increase in the fluid pressure from the pump inlet to its outlet is created when the pump is in operation. This pressure difference drives the fluid through the system or plant.

The centrifugal pump creates an increase in pressure by transferring mechanical energy from the motor to the fluid through the rotating impeller. The fluid flows from the inlet to the impeller centre and out along its blades. The centrifugal force hereby increases the fluid velocity and consequently also the kinetic energy is transformed to pressure. Figure 1.1 shows an example of the fluid path through the centrifugal pump.

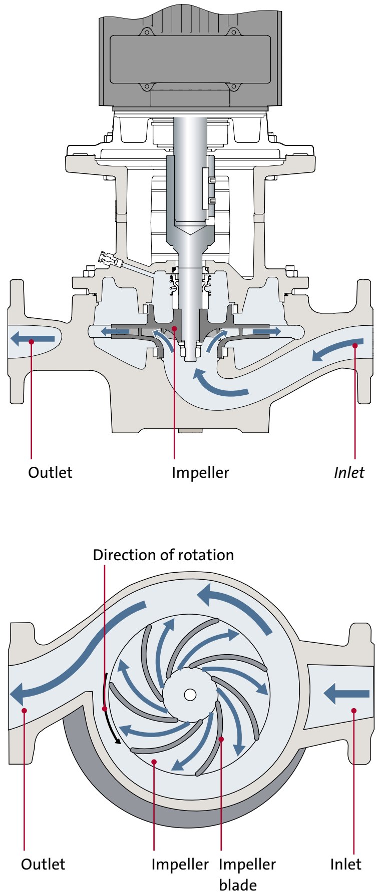


Figure 1.1: Fluid path through the centrifugal pump.

1.2 Hydraulic components

The principles of the hydraulic components are common for most centrifugal pumps. The hydraulic components are the parts in contact with the fluid. Figure 1.2 shows the hydraulic components in a single-stage inline pump. The subsequent sections describe the components from the inlet flange to the outlet flange.

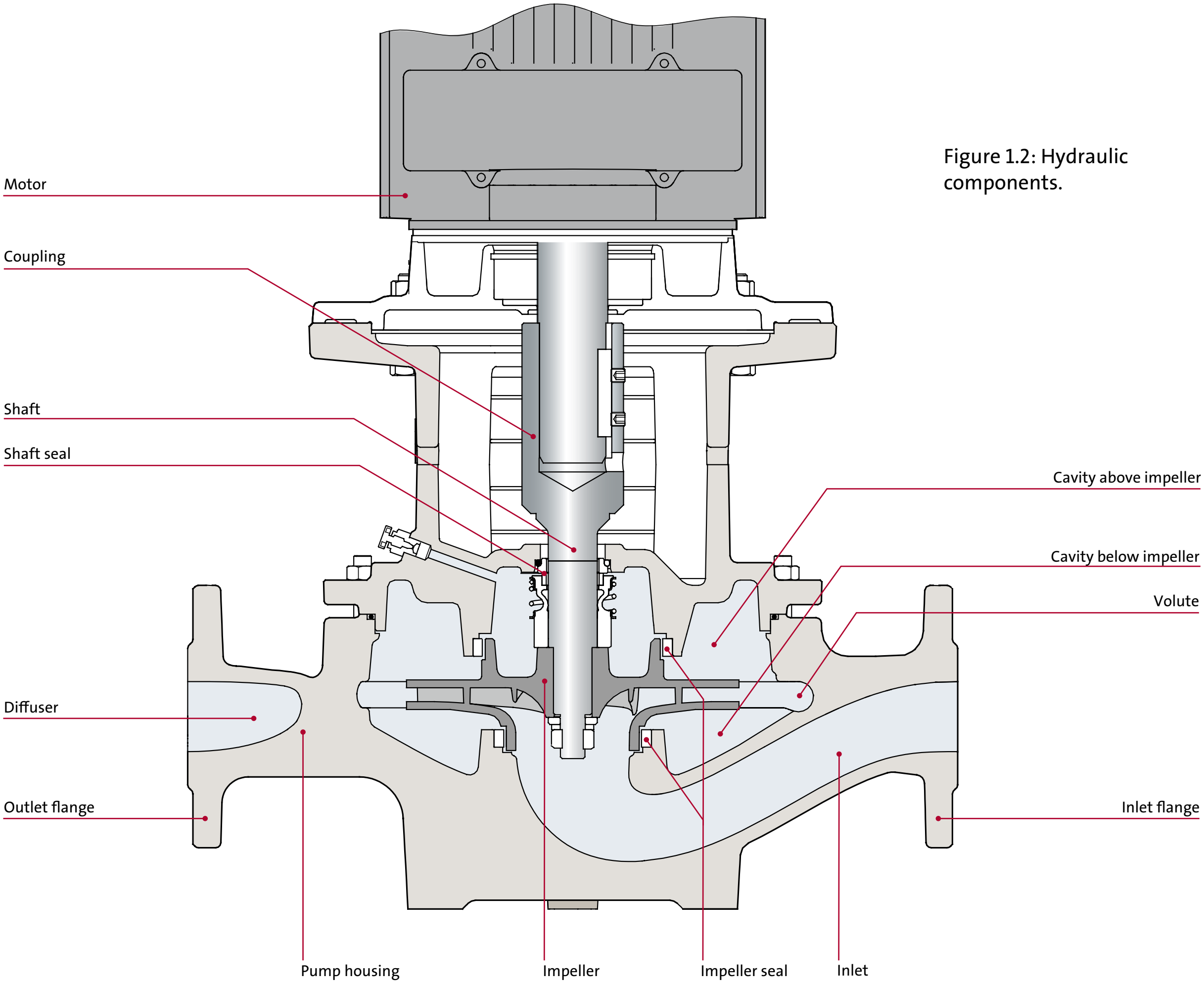


Figure 1.2: Hydraulic components.

1.2.1 Inlet flange and inlet

The pump is connected to the piping system through its inlet and outlet flanges. The design of the flanges depends on the pump application. Some pump types have no inlet flange because the inlet is not mounted on a pipe but submerged directly in the fluid.

The inlet guides the fluid to the impeller eye. The design of the inlet depends on the pump type. The four most common types of inlets are inline, endsuction, doublesuction and inlet for submersible pumps, see figure 1.3.

Inline pumps are constructed to be mounted on a straight pipe – hence the name inline. The inlet section leads the fluid into the impeller eye.

Endsuction pumps have a very short and straight inlet section because the impeller eye is placed in continuation of the inlet flange.

The impeller in doublesuction pumps has two impeller eyes. The inlet splits in two and leads the fluid from the inlet flange to both impeller eyes. This design minimises the axial force, see section 1.2.5.

In submersible pumps, the motor is often placed below the hydraulic parts with the inlet placed in the mid section of the pump, see figure 1.3. The design prevents hydraulic losses related to leading the fluid along the motor. In addition, the motor is cooled due to submersion in the fluid.

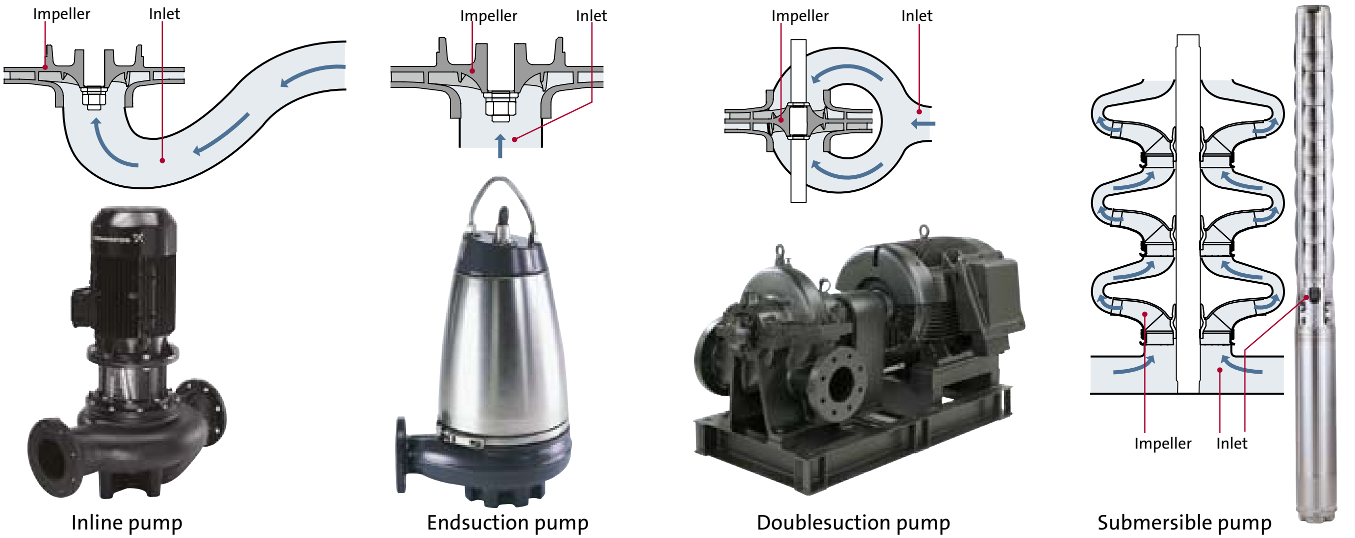


Figure 1.3: Inlet for inline, endsuction, doublesuction and submersible pump.

The design of the inlet aims at creating a uniform velocity profile into the impeller since this leads to the best performance. Figure 1.4 shows an example of the velocity distribution at different cross-sections in the inlet.

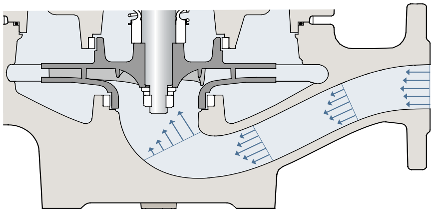


Figure 1.4: Velocity distribution in inlet.

1.2.2 Impeller

The blades of the rotating impeller transfer energy to the fluid there by increasing pressure and velocity. The fluid is sucked into the impeller at the impeller eye and flows through the impeller channels formed by the blades between the shroud and hub, see figure 1.5.

The design of the impeller depends on the requirements for pressure, flow and application. The impeller is the primary component determining the pump performance. Pumps variants are often created only by modifying the impeller.

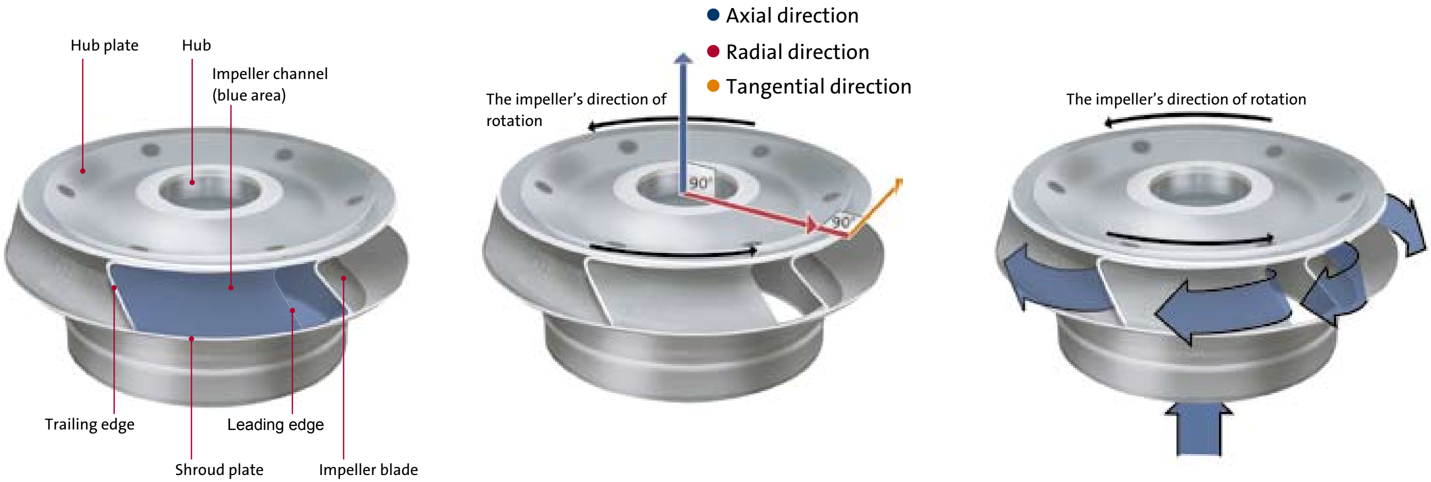


Figure 1.5: The impeller components, definitions of directions and flow relatively to the impeller.

The impeller's ability to increase pressure and create flow depends mainly on whether the fluid runs radially or axially through the impeller, see figure 1.6.

In a radial impeller, there is a significant difference between the inlet diameter and the outlet diameter and also between the outlet diameter and the outlet width, which is the channel height at the impeller exit. In this construction, the centrifugal forces result in high pressure and low flow. Relatively low pressure and high flow are, on the contrary, found in an axial impeller with a no change in radial direction and large outlet width. Semiaxial impellers are used when a trade-off between pressure rise and flow is required.

The impeller has a number of impeller blades. The number mainly depends on the desired performance and noise constraints as well as the amount and size of solid particles in the fluid. Impellers with 5-10 channels has proven to give the best efficiency and is used for fluid without solid particles. One, two or three channel impellers are used for fluids with particles such as waste-water. The leading edge of such impellers is designed to minimise the risk of particles blocking the impeller. One, two and three channel impellers can handle particles of a certain size passing through the impeller. Figure 1.7 shows a one channel pump.

Impellers without a shroud are called open impellers. Open impellers are used where it is necessary to clean the impeller and where there is risk of blocking. A vortex pump with an open impeller is used in waste water application. In this type of pump, the impeller creates a flow resembling the vortex in a tornado, see figure 1.8. The vortex pump has a low efficiency compared to pumps with a shroud and impeller seal.

After the basic shape of the impeller has been decided, the design of the impeller is a question of finding a compromise between friction loss and loss as a concequence of non uniform velocity profiles. Generally, uniform velocity profiles can be achieved by extending the impeller blades but this results in increased wall friction.

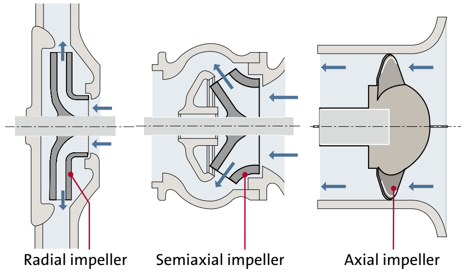


Figure 1.6: Radial, semiaxial and axial impeller.

Figure 1.7: One channel pump.

Figure 1.8: Vortex pump.

1.2.3 Coupling and drive

The impeller is usually driven by an electric motor. The coupling between motor and hydraulics is a weak point because it is difficult to seal a rotating shaft. In connection with the coupling, distinction is made between two types of pumps: Dry-runner pumps and canned rotor type pump. The advantage of the dry-runner pump compared to the canned rotor type pump is the use of standardized motors. The disadvantage is the sealing between the motor and impeller.

In the dry runner pump the motor and the fluid are separated either by a shaft seal, a separation with long shaft or a magnetic coupling.

In a pump with a shaft seal, the fluid and the motor are separated by seal rings, see figure 1.9. Mechanical shaft seals are maintenance-free and have a smaller leakage than stuffing boxes with compressed packing material. The lifetime of mechanical shaft seals depends on liquid, pressure and temperature.

If motor and fluid are separated by a long shaft, then the two parts will not get in contact then the shaft seal can be left out, see figure 1.10. This solution has limited mounting options because the motor must be placed higher than the hydraulic parts and the fluid surface in the system. Furthermore the solution results in a lower efficiency because of the leak flow through the clearance between the shaft and the pump housing and because of the friction between the fluid and the shaft.

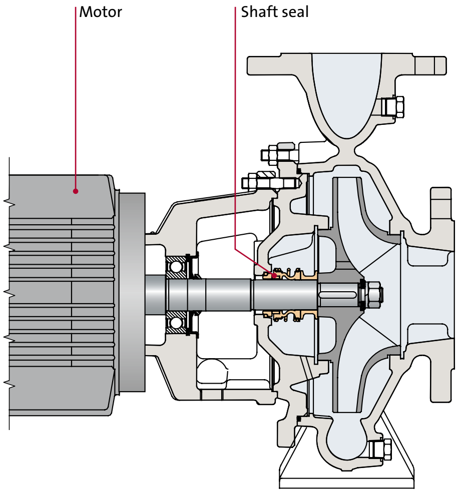


Figure 1.9: Dry-runner with shaft seal.

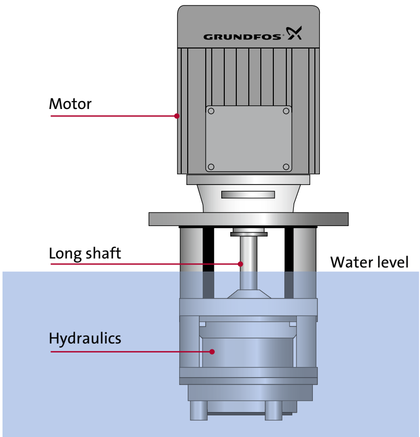
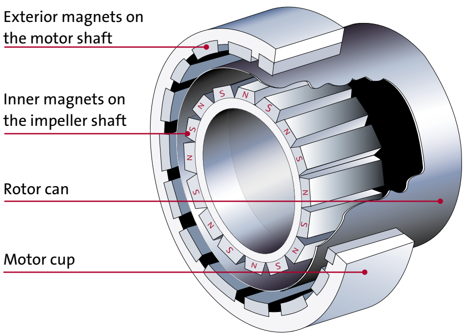


Figure 1.10: Dry-runner with long shaft.



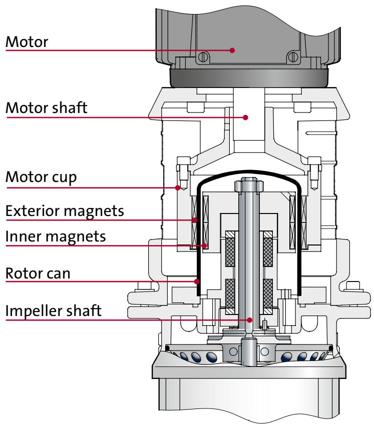


Figure 1.11: Dry-runner with magnet drive.

In pumps with a magnetic drive, the motor and the fluid are separated by a non-magnetizable rotor can which eliminates the problem of sealing a rotating shaft. On this type of pump, the impeller shaft has a line of fixed magnets called the inner magnets. The motor shaft ends in a cup where the outer magnets are mounted on the inside of the cup, see figure 1.11. The rotor can is fixed in the pump housing between the impeller shaft and the cup. The impeller shaft and the motor shaft rotate, and the two parts are connected through the magnets. The main advantage of this design is that the pump is hermitically sealed but the coupling is expensive to produce. This type of sealing is therefore only used when it is required that the pump is hermetically sealed.

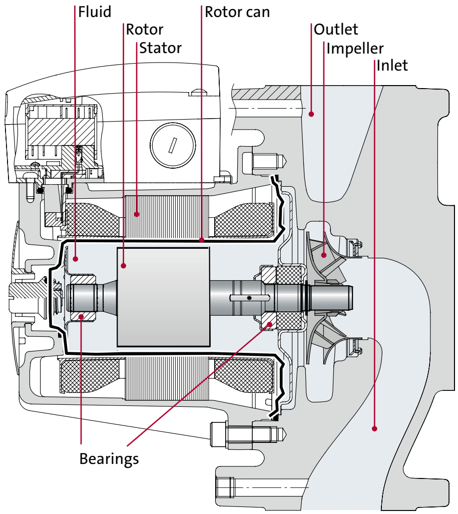


Figure 1.12: Canned rotor type pump.

In pumps with a rotor can, the rotor and impeller are separated from the motor stator. As shown in figure 1.12, the rotor is surrounded by the fluid which lubricates the bearings and cools the motor. The fluid around the rotor results in friction between rotor and rotor can which reduces the pump efficiency.

1.2.4 Impeller seal

A leak flow will occur in the gap between the rotating impeller and stationary pump housing when the pump is operating. The rate of leak flow depends mainly on the design of the gap and the impeller pressure rise. The leak flow returns to the impeller eye through the gap, see figure 1.13. Thus, the impeller has to pump both the leak flow and the fluid through the pump from the inlet flange to the outlet flange. To minimise leak flow, an impeller seal is mounted.

The impeller seal comes in various designs and material combinations. The seal is typically turned directly in the pump housing or made as retrofitted rings. Impeller seals can also be made with floating seal rings. Furthermore, there are a range of sealings with rubber rings in particular well-suited for handling fluids with abrasive particles such as sand.

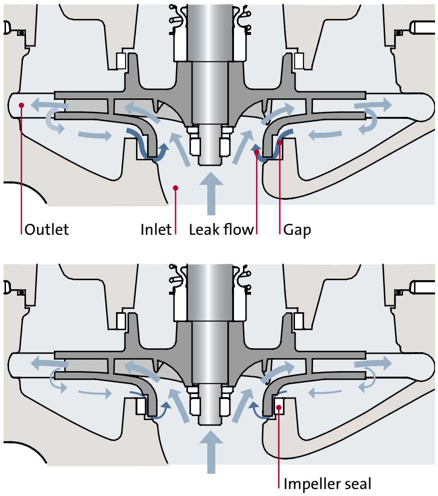
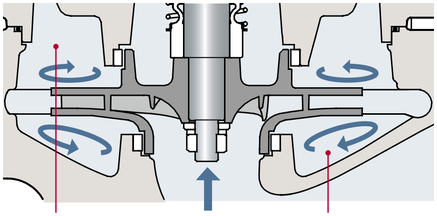


Figure 1.13: Leak flow through the gap.

Achieving an optimal balance between leakage and friction is an essential goal when designing an impeller seal. A small gap limits the leak flow but increases the friction and risk of drag and noise. A small gap also increases requirements to machining precision and assembling resulting in higher production costs. To achieve optimal balance between leakage and friction, the pump type and size must be taken into consideration.



1.2.5 Cavities and axial bearing

The volume of the cavities depends on the design of the impeller and the pump housing, and they affect the flow around the impeller and the pump's ability to handle sand and air.

The impeller rotation creates two types of flows in the cavities: Primary flows and secondary flows. Primary flows are vorticies rotating with the impeller in the cavities above and below the impeller, see figure 1.14. Secondary flows are substantially weaker than the primary flows.

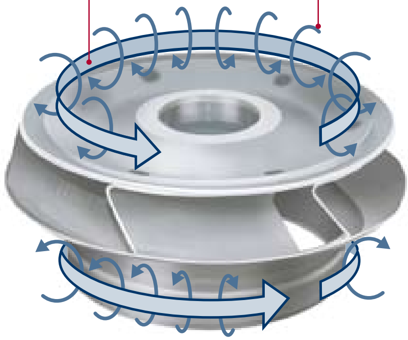


Figure 1.14: Primary and secondary flows in the cavities.

Primary and secondary flows influence the pressure distribution on the outside of the impeller hub and shroud affecting the axial thrust. The axial thrust is the sum of all forces in the axial direction arising due to the pressure condition in the pump. The main force contribution comes from the rise in pressure caused by the impeller. The impeller eye is affected by the inlet pressure while the outer surfaces of the hub and shroud are affected by the outlet pressure, see figure 1.15. The end of the shaft is exposed to the atmospheric pressure while the other end is affected by the system pressure. The pressure is increasing from the center of the shaft and outwards.

The axial bearing absorbs the entire axial thrust and is therefore exposed to the forces affecting the impeller.

The impeller must be axially balanced if it is not possible to absorb the entire axial thrust in the axial bearing. There are several possibilities of reducing the thrust on the shaft and thereby balance the axial bearing. All axial balancing methods result in hydraulic losses.

One approach to balance the axial forces is to make small holes in the hub plate, see figure 1.16. The leak flow through the holes influences the flow in the cavities above the impeller and thereby reduces the axial force but it results in leakage.

Another approach to reduce the axial thrust is to combine balancing holes with an impeller seal on the hub plate, see figure 1.17. This reduces the pressure in the cavity between the shaft and the impeller seal and a better balance can be achieved. The impeller seal causes extra friction but smaller leak flow through the balancing holes compared to the solution without the impeller seal.

A third method of balancing the axial forces is to mount blades on the back of the impeller, see figure 1.18. Like the two previous solutions, this method changes the velocities in the flow at the hub plate whereby the pressure distribution is changed proportionally. However, the additional blades use power without contributing to the pump performance. The construction will therefore reduce the efficiency.

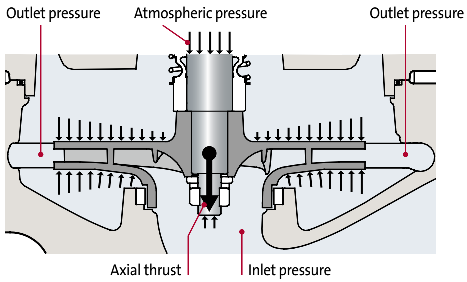


Figure 1.15: Pressure forces which cause axial thrust.

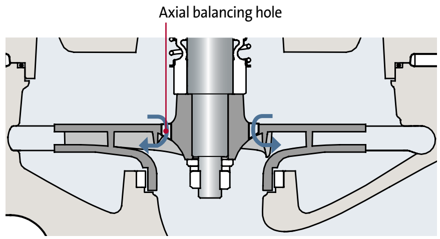


Figure 1.16: Axial thrust reduction using balancing holes.

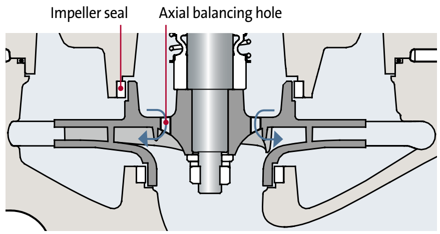


Figure 1.17: Axial thrust reduction using impeller seal and balancing holes.

A fourth method to balance the axial thrust is to mount fins on the pump housing in the cavity below the impeller, see figure 1.19. In this case, the primary flow velocity in the cavity below the impeller is reduced whereby the pressure increases on the shroud. This type of axial balancing increases disc friction and leak loss because of the higher pressure.

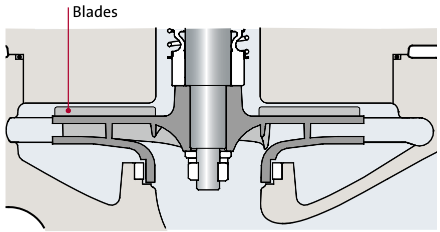


Figure 1.18: Axial thrust reduction through blades on the back of the hub plate.

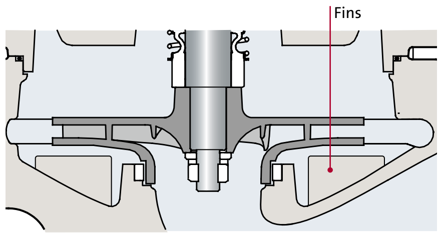
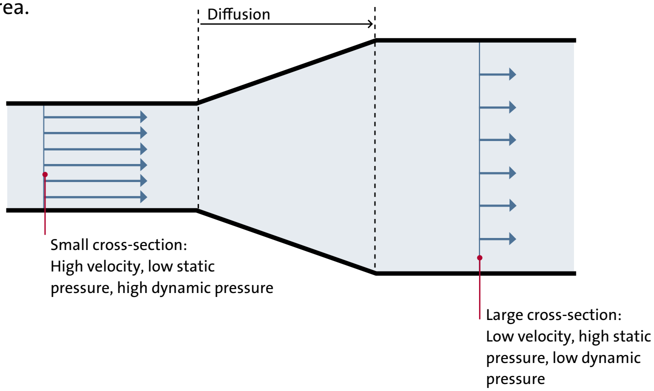


Figure 1.19: Axial thrust reduction using fins in the pump housing.

1.2.6 Volute casing, diffuser and outlet flange

The volute casing collects the fluid from the impeller and leads into the outlet flange. The volute casing converts the dynamic pressure rise in the impeller to static pressure. The velocity is gradually reduced when the cross-sectional area of the fluid flow is increased. This transformation is called velocity diffusion. An example of diffusion is when the fluid velocity in a pipe is reduced because of the transition from a small cross-sectional area to a large cross-sectional area, see figure 1.20. Static pressure, dynamic pressure and diffusion are elaborated in sections 2.2, 2.3 and 5.3.2.

Figure 1.20: Change of fluid velocity in a pipe caused by change in the cross-section area.



The volute casing consists of three main components: Ring diffusor, volute and outlet diffusor, see figure 1.21. An energy conversion between velocity and pressure occurs in each of the three components.

The primary ring diffusor function is to guide the fluid from the impeller to the volute. The cross-section area in the ring diffussor is increased because of the increase in diameter from the impeller to the volute. Blades can be placed in the ring diffusor to increase the diffusion.

The primary task of the volute is to collect the fluid from the ring diffusor and lead it to the diffusor. To have the same pressure along the volute, the cross-section area in the volute must be increased along the periphery from the tongue towards the throat. The throat is the place on the outside of the tongue where the smallest cross-section area in the outlet diffusor is found. The flow conditions in the volute can only be optimal at the design point. At other flows, radial forces occur on the impeller because of circumferential pressure variation in the volute. Radial forces must, like the axial forces, be absorbed in the bearing, see figure 1.21.

The outlet diffusor connects the throat with the outlet flange. The diffusor increases the static pressure by a gradual increase of the cross-section area from the throat to the outlet flange.

The volute casing is designed to convert dynamic pressure to static pressure is achieved while the pressure losses are minimised. The highest efficiency is obtained by finding the right balance between changes in velocity and wall friction. Focus is on the following parameters when designing the volute casing: The volute diameter, the cross-section geometry of the volute, design of the tongue, the throat area and the radial positioning as well as length, width and curvature of the diffusor.

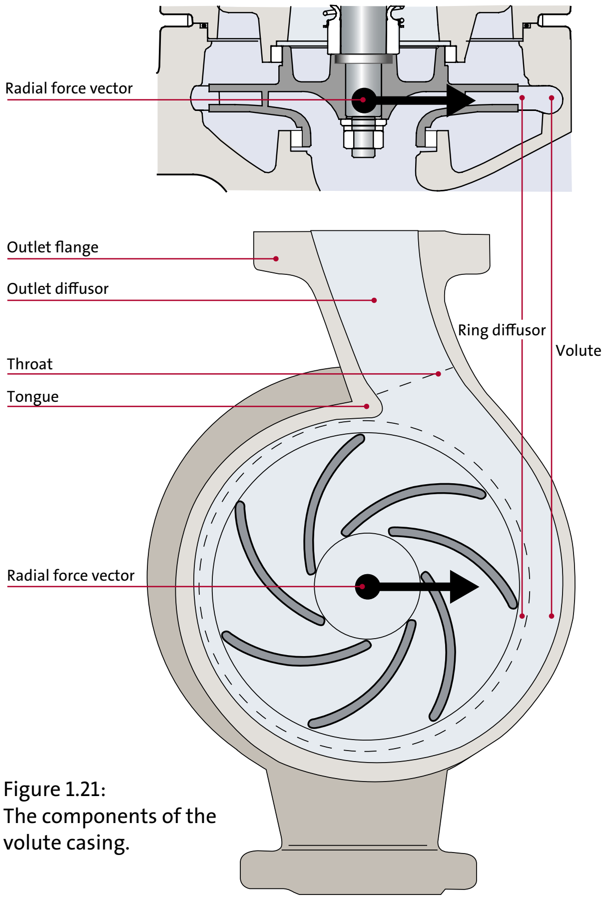


Figure 1.21: The components of the volute casing.

1.2.7 Return channel and outer sleeve

To increase the pressure rise over the pump, more impellers can be connected in series. The return channel leads the fluid from one impeller to the next, see figure 1.22. An impeller and a return channel are either called a stage or a chamber. The chambers in a multistage pump are altogether called the chamber stack.

Besides leading the fluid from one impeller to the next, the return channel has the same basic function as volute casing: To convert dynamic pressure to static pressure. The return channel reduces unwanted rotation in the fluid because such a rotation affects the performance of the subsequent impeller. The rotation is controlled by guide vanes in the return channel.

In multistage inline pumps the fluid is lead from the top of the chamber stack to the outlet in the channel formed by the outer part of the chamber stack and the outer sleeve, see figure 1.22.

When designing a return channel, the same design considerations of impeller and volute casing apply. Contrary to volute casing, a return channel does not create radial forces on the impeller because it is axis-symmetric.

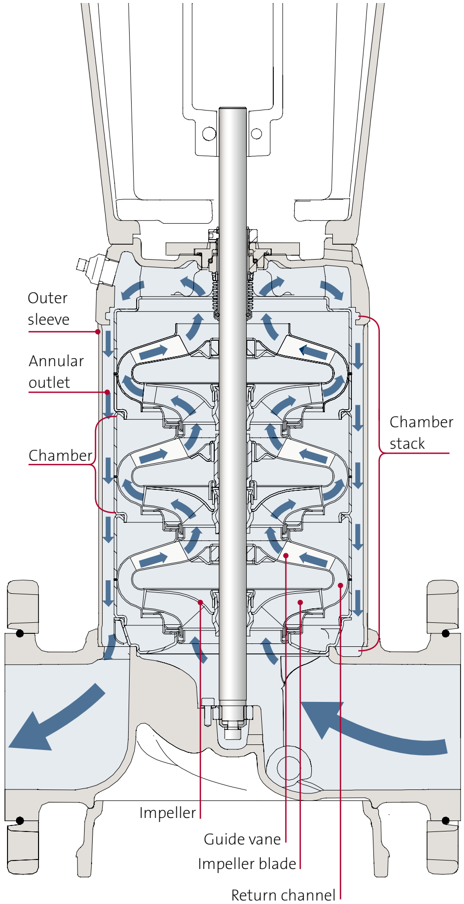


Figure 1.22: Hydraulic components in an inline multistage pump.

1.3 Pump types and systems

This section describes a selection of the centrifugal pumps produced by Grundfos. The pumps are divided in five overall groups: Circulation pumps, pumps for pressure boosting and fluid transport, water supply pumps, industrial pumps and wastewater pumps. Many of the pump types can be used in different applications.

Circulation pumps are primarily used for circulation of water in closed systems e.g. heating, cooling and airconditioning systems as well as domestic hot water systems. The water in a domestic hot water system constantly circulates in the pipes. This prevents a long wait for hot water when the tap is opened.

Pumps for pressure boosting are used for increasing the pressure of cold water and as condensate pumps for steam boilers. The pumps are usually designed to handle fluids with small particles such as sand.

Water supply pumps can be installed in two ways: They can either be submerged in a well or they can be placed on the ground surface. The conditions in the water supply system make heavy demands on robustness towards ochre, lime and sand.

Industrial pumps can, as the name indicates, be used everywhere in the industry and this in a very broad section of systems which handle many different homogeneous and inhomogeneous fluids. Strict environmental and safety requirements are enforced on pumps which must handle corrosive, toxic or explosive fluids, e.g. that the pump is hermetically closed and corrosion resistant.

Wastewater pumps are used for pumping contaminated water in sewage plants and industrial systems. The pumps are constructed making it possible to pump fluids with a high content of solid particles.

1.3.1. The UP pump

Circulation pumps are used for heating, circulation of cold water, ventilation and aircondition systems in houses, office buildings, hotels, etc. Some of the pumps are installed in heating systems at the end user. Others are sold to OEM customers (Original Equipment Manufacturer) that integrate the pumps into gas furnace systems. It is an inline pump with a canned rotor which only has static sealings. The pump is designed to minimise pipe-transferred noise. Grundfos produces UP pumps with and without automatic regulation of the pump. With the automatic regulation of the pump, it is possible to adjust the pressure and flow to the actual need and thereby save energy.

Figure 1.23: UP pumps.

1.3.2 The TP pump

The TP pump is used for circulation of hot or cold water mainly in heating, cooling and airconditioning systems. It is an inline pump and contrary to the smaller UP pump, the TP pump uses a standard motor and shaft seal.

Figure 1.24: TP pump.

1.3.3 The NB pump

The NB pump is for transportation of fluid in district heating plants, heat supply, cooling and air conditioning systems, washdown systems and other industrial systems. The pump is an endsuction pump, and it is found in many variants with different types of shaft seals, impellers and housings which can be combined depending on fluid type, temperature and pressure.

Figure 1.25: NB pump.

1.3.4 The MQ pump

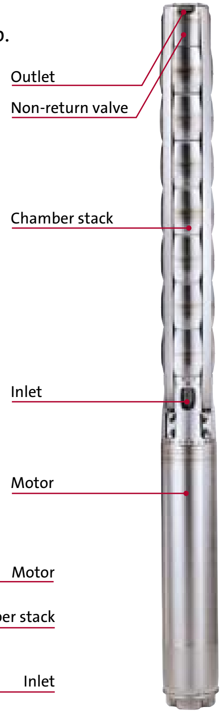
The MQ pump is a complete miniature water supply unit. It is used for water supply and transportation of fluid in private homes, holiday houses, agriculture, and gardens. The pump control ensures that it starts and stops automatically when the tap is opened. The control protects the pump if errors occur or if it runs dry. The built-in pressure expansion tank reduces the number of starts if there are leaks in the pipe system. The MQ pump is self-priming, then it can clear a suction pipe from air and thereby suck from a level which is lower than the one where the pump is placed.

Figure 1.26: MQ pump.

1.3.5 The SP pump

The SP pump is a multi-stage submersible pump which is used for raw water supply, ground water lowering and pressure boosting. The SP pump can also be used for pumping corrosive fluids such as sea water. The motor is mounted under the chamber stack, and the inlet to the pump is placed between motor and chamber stack. The pump diameter is designed to the size of a standard borehole. The SP pump is equipped with an integrated non-return valve to prevent that the pumped fluid flows back when the pump is stopped. The non-return valve also helps prevent water hammer.

Figure 1.27: SP pump.



1.3.6 The CR pump

The CR pump is used in washers, cooling and air conditioning systems, water treatment systems, fire extinction systems, boiler feed systems and other industrial systems. The CR pump is a vertical inline multistage pump. This pump type is also able to pump corrosive fluids because the hydraulic parts are made of stainless steel or titanium.

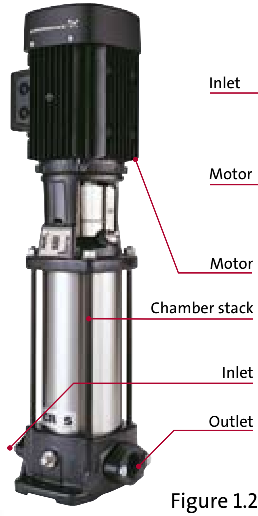


Figure 1.28: CR-pump.

1.3.7 The MTA pump

The MTA pump is used on the non-filtered side of the machining process to pump coolant and lubricant containing cuttings, fibers and abrasive particles. The MTA pump is a dry-runner pump with a long shaft and no shaft seal. The pump is designed to be mounted vertically in a tank. The installation length, the part of the pump which is submerged in the tank, is adjusted to the tank depth so that it is possible to drain the tank of coolant and lubricant.

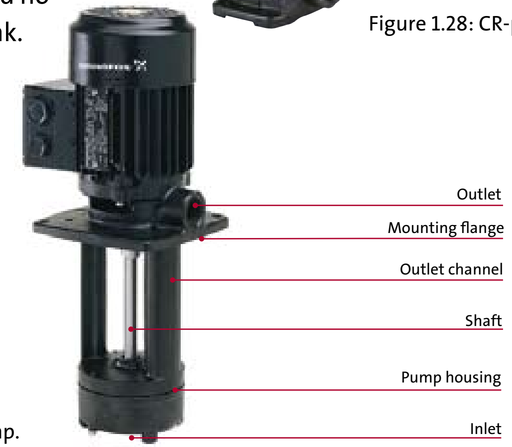


Figure 1.29: MTA pump.

1.3.8 The SE pump

The SE pump is used for pumping wastewater, water containing sludge and solids. The pump is unique in the wastewater market because it can be installed submerged in a waste water pit as well as installed dry in a pipe system. The series of SE pumps contains both vortex pumps and single-channel pumps. The single-channel pumps are characterised by a large free passage, and the pump specification states the maximum diameter for solids passing through the pump.

Figure 1.30: SE pump.

1.3.9 The SEG pump

The SEG pump is in particular suitable for pumping waste water from toilets. The SEG pump has a cutting system which cuts perishable solids into smaller pieces which then can be lead through a tube with a relative small diameter. Pumps with cutting systems are also called grinder pumps.

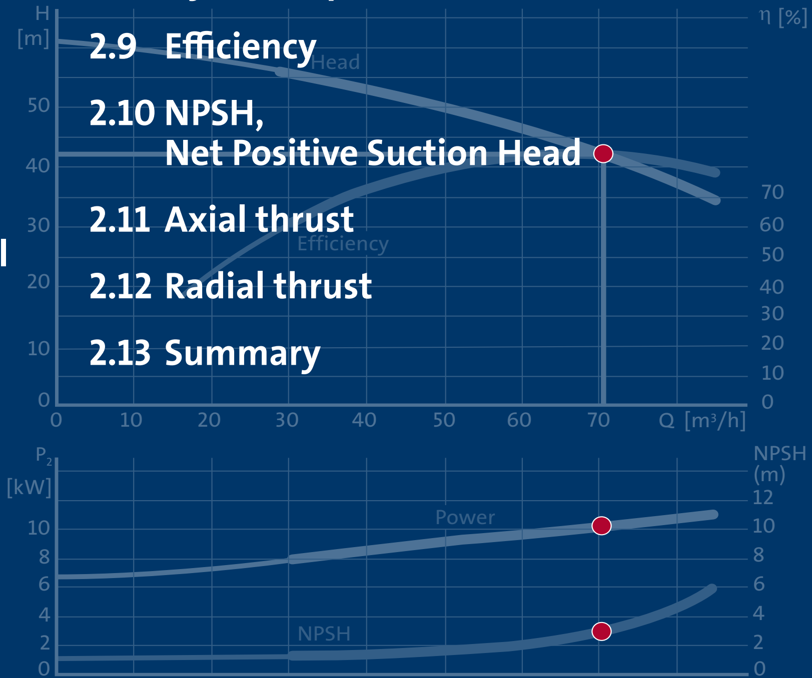
Figure 1.31: SEG pumps.

1.4 Summary

In this chapter, we have covered the principle of the centrifugal pump and its hydraulic components. We have discussed some of the overall aspects connected to design of the single components. Included in the chapter is also a short description of some of the Grundfos pumps.

Chapter 2

Performance curves



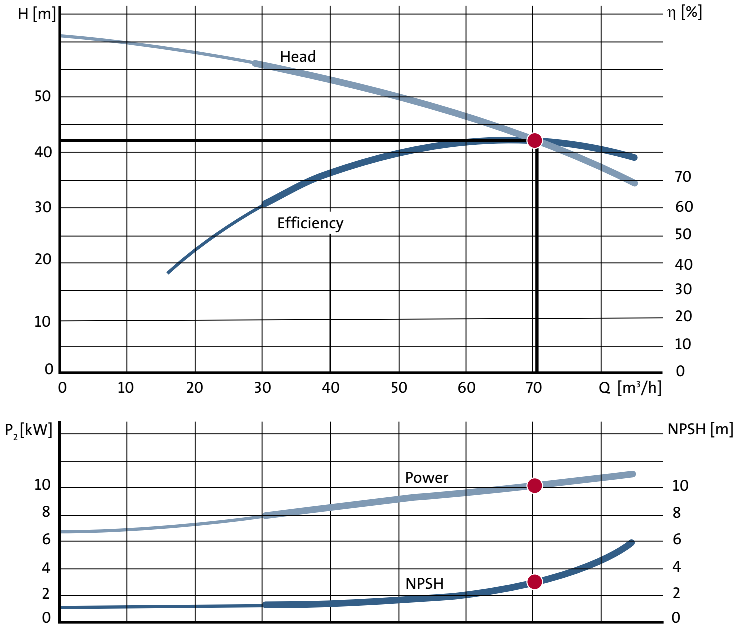
2. Performance curves

The pump performance is normally described by a set of curves. This chapter explains how these curves are interpretated and the basis for the curves.

2.1 Standard curves

Performance curves are used by the customer to select pump matching his requirements for a given application.

The data sheet contains information about the head (H) at different flows (Q), see figure 2.1. The requirements for head and flow determine the overall dimensions of the pump.



Fígure 2.1: Typical performance curves for a centrifugal pump. Head (H), power consumption (P), efficiency (η) and NPSH are shown as function of the flow.

In addition to head, the power consumption (P) is also to be found in the data sheet. The power consumption is used for dimensioning of the installations which must supply the pump with energy. The power consumption is like the head shown as a function of the flow.

Information about the pump efficiency (η) and NPSH can also be found in the data sheet. NPSH is an abbreviation for 'Net Positive Suction Head'. The NPSH curve shows the need for inlet head, and which requirements the specific system have to fullfill to avoid cavitation. The efficiency curve is used for choosing the most efficient pump in the specified operating range. Figure 2.1 shows an example of performance curves in a data sheet.

During design of a new pump, the desired performance curves are a vital part of the design specifications. Similar curves for axial and radial thrust are used for dimensioning the bearing system.

The performance curves describe the performance for the complete pump unit, see figure 2.2. An adequate standard motor can be mounted on the pump if a pump without motor is chosen. Performance curves can be recalculated with the motor in question when it is chosen.

For pumps sold both with and without a motor, only curves for the hydraulic components are shown, i.e. without motor and controller. For integrated products, the pump curves for the complete product are shown.

Figure 2.2.: The performance curves are stated for the pump itself or for the complete unit consisting of pump, motor and electronics.

2.2 Pressure

Pressure (p) is an expression of force per unit area and is split into static and dynamic pressure. The sum of the two pressures is the total pressure:

$$p_{tot} = p_{stat} + p_{dyn} \quad [Pa] \qquad (2.1)$$

where
p_{tot} = Total pressure [Pa]
p_{stat} = Static pressure [Pa]
p_{dyn} = Dynamic pressure [Pa]

Static pressure is measured with a pressure gauge, and the measurement of static pressure must always be done in static fluid or through a pressure tap mounted perpendicular to the flow direction, see figure 2.3.

Total pressure can be measured through a pressure tap with the opening facing the flow direction, see figure 2.3. The dynamic pressure can be found measuring the pressure difference between total pressure and static pressure. Such a combined pressure measurement can be performed using a pitot tube.

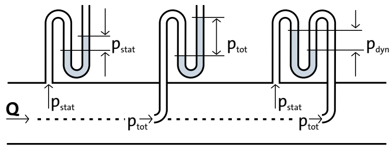


Figure 2.3: This is how static pressure p_{stat}, total pressure p_{tot} and dynamic pressure p_{dyn} are measured.

Dynamic pressure is a function of the fluid velocity. The dynamic pressure can be calculated with the following formula,where the velocity (V) is measured and the fluid density (ρ) is know:

$$p_{dyn} = \frac{1}{2} \cdot \rho \cdot V^2 \quad [Pa] \qquad (2.2)$$

where
V = Velocity [m/s]
ρ = Density [kg/m³]

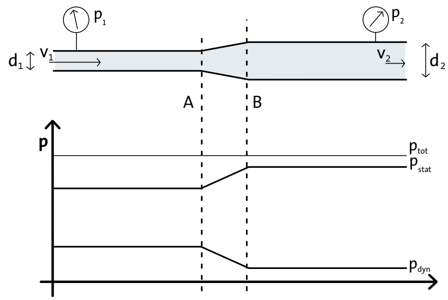


Figure 2.4: Example of conversion of dynamic pressure to static pressure in a diffusor.

Dynamic pressure can be transformed to static pressure and vice versa. Flow through a pipe where the pipe diameter is increased converts dynamic pressure to static pressure, see figure 2.4. The flow through a pipe is called a pipe flow, and the part of the pipe where the diameter is increasing is called a diffusor.

2.3 Absolute and relative pressure

Pressure is defined in two different ways: absolute pressure or relative pressure. Absolute pressure refers to the absolute zero, and absolute pressure can thus only be a positive number. Relative pressure refers to the pressure of the surroundings. A positive relative pressure means that the pressure is above the barometric pressure, and a negative relative pressure means that the pressure is below the barometric pressure.

The absolute and relative definition is also known from temperature measurement where the absolute temperature is measured in Kelvin [K] and the relative temperature is measured in Celsius [°C]. The temperature measured in Kelvin is always positive and refers to the absolute zero. In contrast, the temperature in Celsius refers to water's freezing point at 273.15K and can therefore be negative.

The barometric pressure is measured as absolute pressure. The barometric pressure is affected by the weather and altitude. The conversion from relative pressure to absolute pressure is done by adding the current barometric pressure to the measured relative pressure:

$$p_{abs} = p_{rel} + p_{bar} \quad [\mathrm{Pa}] \tag{2.3}$$

In practise, static pressure is measured by means of three different types of pressure gauges:

- An absolute pressure gauge, such as a barometer, measures pressure relative to absolute zero.
- An standard pressure gauge measures the pressure relative to the atmospherich pressure. This type of pressure gauge is the most commonly used.
- A differential pressure gauge measures the pressure difference between the two pressure taps independent of the barometric pressure.

2.4 Head

The different performance curves are introduced on the following pages.

A QH curve or pump curve shows the head (H) as a function of the flow (Q). The flow (Q) is the rate of fluid going through the pump. The flow is generally stated in cubic metre per hour [m^3/h] but at insertion into formulas cubic metre per second [m^3/s] is used. Figure 2.5 shows a typical QH curve.

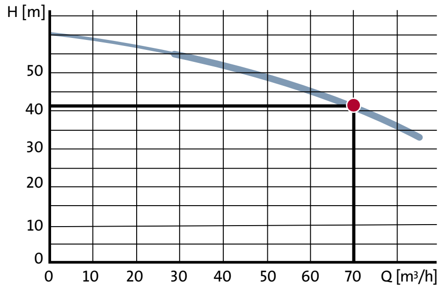


Figure 2.5: A typical QH curve for a centrifugal pump; a small flow gives a high head and a large flow gives a low head.

The QH curve for a given pump can be determined using the setup shown in figure 2.6.

The pump is started and runs with constant speed. Q equals 0 and H reaches its highest value when the valve is completely closed. The valve is gradually opened and as Q increases H decreases. H is the height of the fluid column in the open pipe after the pump. The QH curve is a series of coherent values of Q and H represented by the curve shown in figure 2.5.

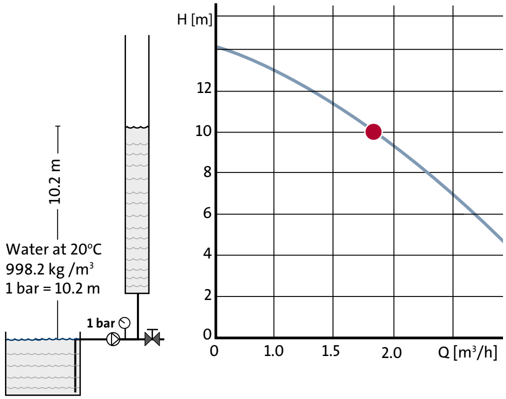


Figure 2.6: The QH curve can be determined in an installation with an open pibe after the pump. H is exactly the height of the fluid column in the open pipe. measured from inlet level.

In most cases the differential pressure across the pump Δp_{tot} is measured and the head H is calculated by the following formula:

$$H = \frac{\Delta p_{tot}}{\rho \cdot g} \quad [m] \qquad (2.4)$$

The QH curve will ideally be exactly the same if the test in figure 2.6 is made with a fluid having a density different from water. Hence, a QH curve is independent of the pumped fluid. It can be explained based on the theory in chapter 4 where it is proven that Q and H depend on the geometry and speed but not on the density of the pumped fluid.

The pressure increase across a pump can also be measured in meter water column [mWC]. Meter water column is a pressure unit which must not be confused with the head in [m]. As seen in the table of physical properties of water, the change in density is significant at higher temperatures. Thus, conversion from pressure to head is essential.

2.5 Differential pressure across the pump - description of differential pressure

2.5.1 Total pressure difference

The total pressure difference across the pump is calculated on the basis of three contributions:

$$\Delta p_{tot} = \Delta p_{stat} + \Delta p_{dyn} + \Delta p_{geo} \quad [Pa] \qquad (2.5)$$

where

Δp_{tot} = Total pressure difference across the pump [Pa]
Δp_{stat} = Static pressure difference across the pump [Pa]
Δp_{dyn} = Dynamic pressure difference across the pump [Pa]
Δp_{geo} = Geodetic pressure difference between the pressure sensors [Pa]

2.5.2 Static pressure difference

The static pressure difference can be measured directly with a differential pressure sensor, or a pressure sensor can be placed at the inlet and outlet of the pump. In this case, the static pressure difference can be found by the following expression:

$$\Delta p_{stat} = p_{stat,\,out} - p_{stat,\,in} \quad [Pa] \qquad (2.6)$$

2.5.3 Dynamic pressure difference

The dynamic pressure difference between the inlet and outlet of the pump is found by the following formula:

$$\Delta p_{dyn} = \frac{1}{2} \cdot \rho \cdot V_{out}^{\;2} - \frac{1}{2} \cdot \rho \cdot V_{in}^{\;2} \quad [Pa] \qquad (2.7)$$

In practise, the dynamic pressure and the flow velocity before and after the pump are not measured during test of pumps. Instead, the dynamic pressure difference can be calculated if the flow and pipe diameter of the inlet and outlet of the pump are known:

$$\Delta p_{dyn} = \frac{1}{2} \cdot \rho \cdot \left(\frac{Q}{\pi/4} \right)^2 \cdot \left(\frac{1}{D_{out}^{\ 4}} - \frac{1}{D_{in}^{\ 4}} \right) \ [Pa] \qquad (2.8)$$

The formula shows that the dynamic pressure difference is zero if the pipe diameters are identical before and after the pump.

2.5.4 Geodetic pressure difference

The geodetic pressure difference between inlet and outlet can be measured in the following way:

$$\Delta p_{geo} = \Delta z \cdot \rho \cdot g \ [Pa] \qquad (2.9)$$

where
Δz is the difference in vertical position between the gauge connected to the outlet pipe and the gauge connected to the inlet pipe.

The geodetic pressure difference is only relevant if Δz is not zero. Hence, the position of the measuring taps on the pipe is of no importance for the calculation of the geodetic pressure difference.

The geodetic pressure difference is zero when a differential pressure gauge is used for measuring the static pressure difference.

2.6 Energy equation for an ideal flow

The energy equation for an ideal flow describes that the sum of pressure energy, velocity energy and potential energy is constant. Named after the Swiss physicist Daniel Bernoulli, the equation is known as Bernoulli's equation:

$$\frac{p}{\rho} + \frac{V^2}{2} + g \cdot z = \text{Constant} \quad \left[\frac{m^2}{s^2}\right] \qquad (2.10)$$

Bernoulli's equation is valid if the following conditions are met:

1. Stationary flow – no changes over time
2. Incompressible flow – true for most liquids
3. Loss-free flow – ignores friction loss
4. Work-free flow – no supply of mechanical energy

Formula (2.10) applies along a stream line or the trajectory of a fluid particle. For example, the flow through a diffusor can be described by formula (2.10), but not the flow through an impeller since mechancial energy is added.

In most applications, not all the conditions for the energy equation are met. In spite of this, the equation can be used for making a rough calculation.

2.7 Power

The power curves show the energy transfer rate as a function of flow, see figure 2.7. The power is given in Watt [W]. Distinction is made between three kinds of power, see figure 2.8:

- Supplied power from external electricity source to the motor and controller (P_1)
- Shaft power transferred from the motor to the shaft (P_2)
- Hydraulic power transferred from the impeller to the fluid (P_{hyd})

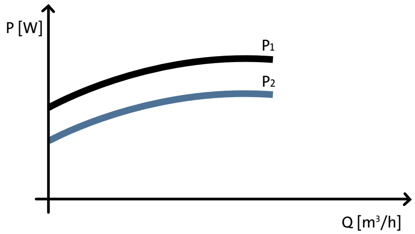


Figure 2.7: P_1 and P_2 power curves.

The power consumption depends on the fluid density. The power curves are generally based on a standard fluid with a density of 1000 kg/m^3 which corresponds to water at 4°C. Hence, power measured on fluids with another density must be converted.

In the data sheet, P_1 is normally stated for integrated products, while P_2 is typically stated for pumps sold with a standard motor.

Figure 2.8: Power transfer in a pump unit.

2.7.1 Speed

Flow, head and power consumption vary with the pump speed, see section 3.4.4. Pump curves can only be compared if they are stated with the same speed. The curves can be converted to the same speed by the formulas in section 3.4.4.

2.8 Hydraulic power

The hydraulic power P_{hyd} is the power transferred from the pump to the fluid. As seen from the following formula, the hydraulic power is calculated based on flow, head and density:

$$P_{hyd} = H \cdot g \cdot \rho \cdot Q = \Delta p_{tot} \cdot Q \quad [W] \qquad (2.11)$$

An independent curve for the hydraulic power is usually not shown in data sheets but is part of the calculation of the pump efficiency.

2.9 Efficiency

The total efficiency (η_{tot}) is the ratio between hydraulic power and supplied power. Figure 2.9 shows the efficiency curves for the pump (η_{hyd}) and for a complete pump unit with motor and controller (η_{tot}).

The hydraulic efficiency refers to P_2, whereas the total efficiency refers to P_1:

$$\eta_{hyd} = \frac{P_{hyd}}{P_2} \quad [\cdot 100\ \%] \tag{2.12}$$

$$\eta_{tot} = \frac{P_{hyd}}{P_1} \quad [\cdot 100\ \%] \tag{2.13}$$

$$P_1 > P_2 > P_{hyd} \quad [W] \tag{2.14}$$

The efficiency is always below 100% since the supplied power is always larger than the hydraulic power due to losses in controller, motor and pump components. The total efficiency for the entire pump unit (controller, motor and hydraulics) is the product of the individual efficiencies:

$$\eta_{tot} = \eta_{control} \cdot \eta_{motor} \cdot \eta_{hyd} \quad [\cdot 100\ \%] \tag{2.15}$$

where

$\eta_{control}$ = Controller efficiency [· 100%]
η_{motor} = Motor efficiency [· 100%]

The flow where the pump has the highest efficiency is called the optimum point or the best efficiency point (Q_{BEP}).

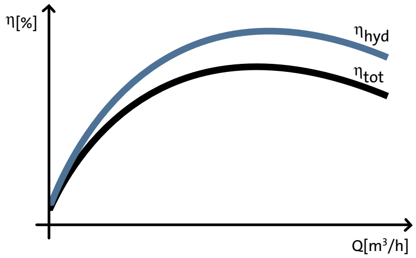


Figure 2.9: Efficiency curves for the pump (η_{hyd}) and complete pump unit with motor and controller (η_{tot}).

2.10 NPSH, Net Positive Suction Head

NPSH is a term describing conditions related to cavitation, which is undesired and harmful.

Cavitation is the creation of vapour bubbles in areas where the pressure locally drops to the fluid vapour pressure. The extent of cavitation depends on how low the pressure is in the pump. Cavitation generally lowers the head and causes noise and vibration.

Cavitation first occurs at the point in the pump where the pressure is lowest, which is most often at the blade edge at the impeller inlet, see figure 2.10.

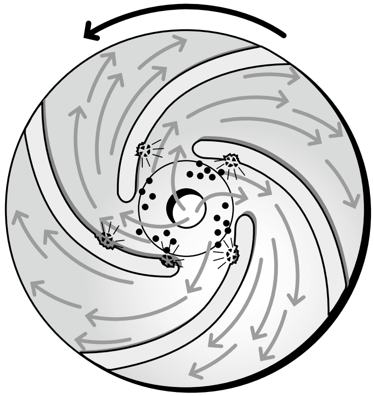


Figure 2.10: Cavitation.

The NPSH value is absolute and always positive. NPSH is stated in meter [m] like the head, see figure 2.11. Hence, it is not necessary to take the density of different fluids into account because NPSH is stated in meters [m].

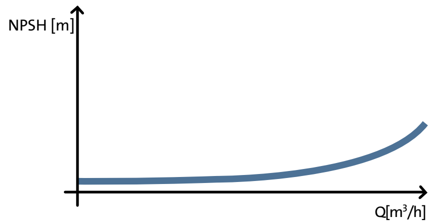


Figure 2.11: NPSH curve.

Distinction is made between two different NPSH values: $NPSH_R$ and $NPSH_A$.

$NPSH_A$ stands for NPSH Available and is an expression of how close the fluid in the suction pipe is to vapourisation. $NPSH_A$ is defined as:

$$NPSH_A = \frac{(p_{abs,tot,in} - p_{vapour})}{\rho \cdot g} \; [m] \qquad (2.16)$$

where

p_{vapour} = The vapour pressure of the fluid at the present temperature [Pa]. The vapour pressure is found in the table "Physical properties of water" in the back of the book.

$p_{abs,tot,in}$ = The absolute pressure at the inlet flange [Pa].

$NPSH_R$ stands for NPSH Required and is an expression of the lowest NPSH value required for acceptable operating conditions. The absolute pressure $p_{abs,tot,in}$ can be calculated from a given value of $NPSH_R$ and the fluid vapour pressure by inserting $NPSH_R$ in the formula (2.16) instead of $NPSH_A$.

To determine if a pump can safely be installed in the system, $NPSH_A$ and $NPSH_R$ should be found for the largest flow and temperature within the operating range.

A minimum safety margin of 0.5 m is recommended. Depending on the application, a higher safety level may be required. For example, noise sensitive applications or in high energy pumps like boiler feed pumps, European Association of Pump Manufacturers indicate a safety factor S_A of 1.2-2.0 times the $NPSH_{3\%}$.

$$NPSH_A > NPSH_R = NPSH_{3\%} + 0.5 \quad [m] \quad \text{or} \qquad (2.17)$$

$$NPSH_A > NPSH_R = NPSH_{3\%} \cdot S_A \quad [m] \qquad (2.17a)$$

The risk of cavitation in systems can be reduced or prevented by:

- Lowering the pump compared to the water level - open systems.
- Increasing the system pressure - closed systems.
- Shortening the suction line to reduce the friction loss.
- Increasing the suction line’s cross-section area to reduce the fluid velocity and thereby reduce friction.
- Avoiding pressure drops coming from bends and other obstacles in the suction line.
- Lowering fluid temperature to reduce vapour pressure.

The two following examples show how NPSH is calculated.

Example 2.1 Pump drawing from a well

A pump must draw water from a reservoir where the water level is 3 meters below the pump. To calculate the $NPSH_A$ value, it is necessary to know the friction loss in the inlet pipe, the water temperature and the barometric pressure, see figure 2.12.

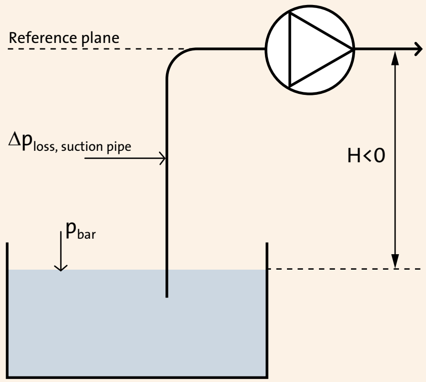


Figure 2.12: Sketch of a system where water is pumped from a well.

Water temperature = 40°C
Barometric pressure = 101.3 kPa
Pressure loss in the suction line at the present flow = 3.5 kPa.
At a water temperature of 40°C, the vapour pressure is 7.37 kPa and ρ is 992.2kg/m³. The values are found in the table ”Physical properties of water” in the back of the book.

For this system, the $NPSH_A$ expression in formula (2.16) can be written as:

$$NPSH_A = \frac{(p_{bar} + \rho \cdot g \cdot H_{geo} - \Delta p_{loss,\,suction\ pipe}) - p_{vapour}}{\rho \cdot g} \ [m] \qquad (2.18)$$

H_{geo} is the water level's vertical position in relation to the pump. H_{geo} can either be above or below the pump and is stated in meter [m]. The water level in this system is placed below the pump. Thus, H_{geo} is negative, H_{geo} = -3m.

The system $NPSH_A$ value is:

$$NPSH_A = \frac{101300\ Pa}{992.2kg/m^3 \cdot 9.81m/s^2} - 3m - \frac{3500\ Pa}{992.2kg/m^3 \cdot 9.81m/s^2} - \frac{7375\ Pa}{992.2kg/m^3 \cdot 9.81m/s^2}$$

$$NPSH_A = 6.3m$$

The pump chosen for the system in question must have a $NPSH_R$ value lower than 6.3 m minus the safety margin of 0.5 m. Hence, the pump must have a $NPSH_R$ value lower than 6.3-0.5 = 5.8 m at the present flow.

Example 2.2 Pump in a closed system

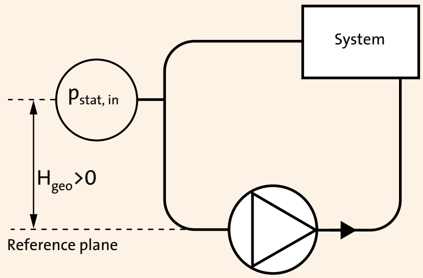


Figure 2.13: Sketch of a closed system.

In a closed system, there is no free water surface to refer to. This example shows how the pressure sensor's placement above the reference plane can be used to find the absolute pressure in the suction line, see figure 2.13.

The relative static pressure on the suction side is measured to be $p_{stat,in}$ = -27.9 kPa_2. Hence, there is negative pressure in the system at the pressure gauge. The pressure gauge is placed above the pump. The difference in height between the pressure gauge and the impeller eye H_{geo} is therefore a positive value of +3m. The velocity in the tube where the measurement of pressure is made results in a dynamic pressure contribution of 500 Pa.

Barometric pressure = 101 kPa
Pipe loss between measurement point ($p_{stat,in}$) and pump is calculated to $H_{loss,pipe}$ = 1m.
System temperature = 80°C
Vapour pressure p_{vapour} = 47.4 kPa and density is ρ = 973 kg/m³, values are found in the table "Physical properties of water".

For this system, formula 2.16 expresses the $NPSH_A$ as follows:

$$NPSH_A = \frac{p_{stat,in} + p_{bar} + 0.5 \cdot \rho \cdot V_1^2}{\rho \cdot g} + H_{geo} - H_{loss,\ pipe} - \frac{p_{vapour}}{\rho \cdot g}\ [m] \qquad (2.19)$$

Inserting the values gives:

$$NPSH_A = \frac{-27900\ Pa + 101000\ Pa + 500\ Pa}{973\ kg/m^3 \cdot 9.81 m/s^2} + 3m - 1m - \frac{47400\ Pa}{973\ kg/m^3 \cdot 9.81 m/s^2}$$

$$NPSH_A = 4.7m$$

Despite the negative system pressure, a $NPSH_A$ value of more than 4m is available at the present flow.

2.11 Axial thrust

Axial thrust is the sum of forces acting on the shaft in axial direction, see figure 2.14. Axial thrust is mainly caused by forces from the pressure difference between the impeller's hub plate and shroud plate, see section 1.2.5.

The size and direction of the axial thrust can be used to specify the size of the bearings and the design of the motor. Pumps with up-thrust require locked bearings. In addition to the axial thrust, consideration must be taken to forces from the system pressure acting on the shaft. Figure 2.15 shows an example of an axial thrust curve.

The axial thrust is related to the head and therefore it scales with the speed ratio squared, see sections 3.4.4 and 4.5.

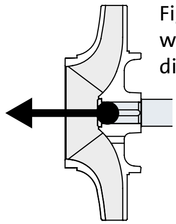


Figure 2.14: Axial thrust work in the bearing's direction.

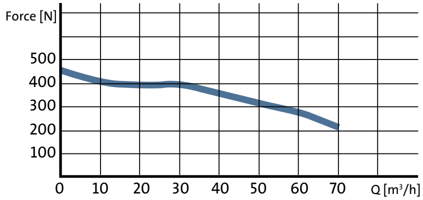


Figure 2.15: Example of a axial thrust curve for a TP65-410 pump.

2.12 Radial thrust

Radial thrust is the sum of forces acting on the shaft in radial direction as shown in figure 2.16. Hydraulic radial thrust is a result of the pressure difference in a volute casing. Size and direction vary with the flow. The forces are minimum in the design point, see figure 2.17. To size the bearings correctly, it is important to know the size of the radial thrust.

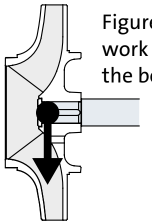


Figure 2.16: Radial thrust work perpendicular on the bearing.

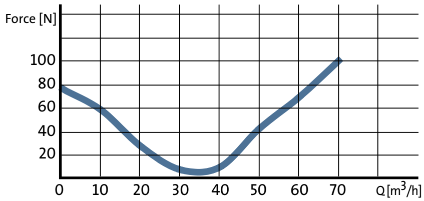


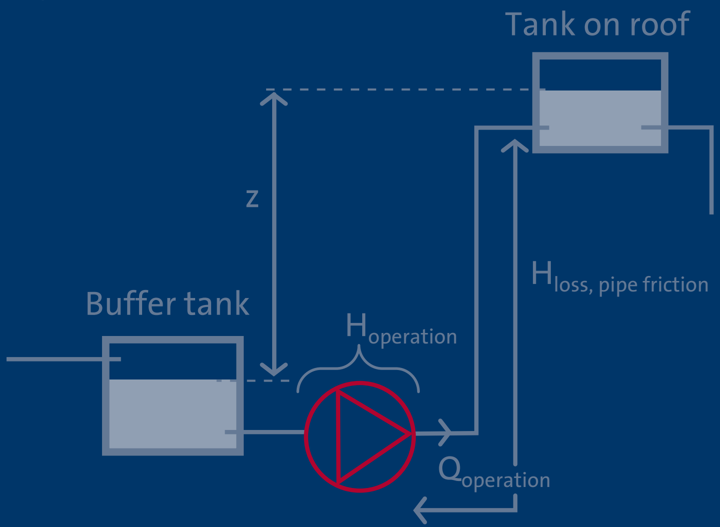
Figure 2.17: Example of a radial thrust curve for a TP65-410 pump.

2.13 Summary

Chapter 2 explains the terms used to describe a pump's performance and shows curves for head, power, efficiency, NPSH and thrust impacts. Furthermore, the two terms head and NPSH are clarified with calculation examples.

Chapter 3

Pumps operating in systems



3. Pumps operating in systems

This chapter explains how pumps operate in a system and how they can be regulated. The chapter also explains the energy index for small circulation pumps.

A pump is always connected to a system where it must circulate or lift fluid. The energy added to the fluid by the pump is partly lost as friction in the pipe system or used to increase the head.

Implementing a pump into a system results in a common operating point. If several pumps are combined in the same application, the pump curve for the system can be found by adding up the pumps' curves either serial or parallel. Regulated pumps adjust to the system by changing the rotational speed. The regulation of speed is especially used in heating systems where the need for heat depends on the ambient temperature, and in water supply systems where the demand for water varies with the consumer opening and closing the tap.

3.1 Single pump in a system

A system characteristic is described by a parabola due to an increase in friction loss related to the flow squared. The system characteristic is described by a steep parabola if the resistance in the system is high. The parabola flattens when the resistance decreases. Changing the settings of the valves in the system changes the characteristics.

The operating point is found where the curve of the pump and the system characteristic intersect.

In closed systems, see figure 3.1, there is no head when the system is not operationg. In this case the system characteristic goes through (Q,H) = (0,0) as shown in figure 3.2.

In systems where water is to be moved from one level to another, see figure 3.3, there is a constant pressure difference between the two reservoirs, corresponding to the height difference. This causes an additional head which the pump must overcome. In this case the system characteristics goes through $(0,H_z)$ instead of (0,0), see figure 3.4.

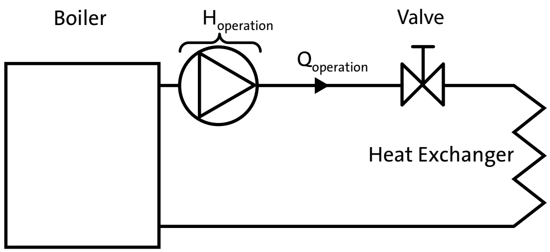


Figure 3.1: Example of a closed system.

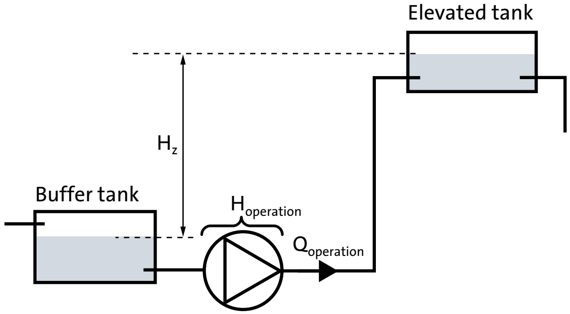


Figure 3.3: Example of an open system with positive geodetic lift.

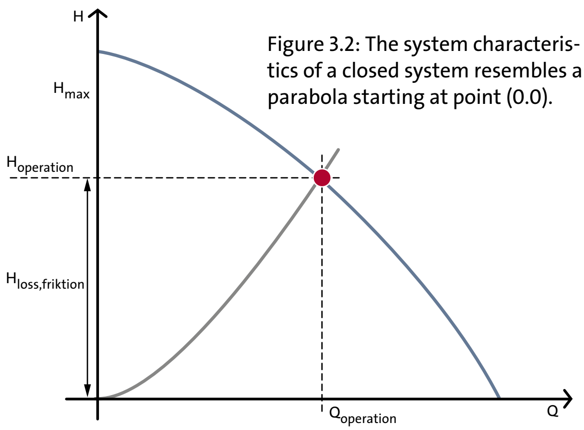


Figure 3.2: The system characteristics of a closed system resembles a parabola starting at point (0.0).

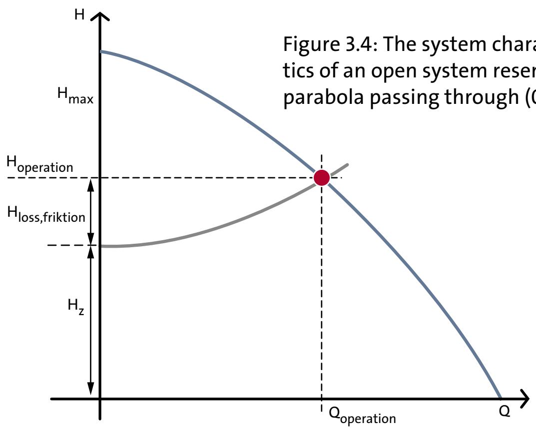


Figure 3.4: The system characteristics of an open system resembles a parabola passing through $(0,H_z)$.

3.2 Pumps operated in parallel

In systems with large variations in flow and a request for constant pressure, two or more pumps can be connected in parallel. This is often seen in larger supply systems or larger circulation systems such as central heating systems or district heating installations.

Parallel-connected pumps are also used when regulation is required or if an auxiliary pump or standby pump is needed. When operating the pumps, it is possible to regulate between one or more pumps at the same time. A non-return valve is therefore always mounted on the discharge line to prevent backflow through the pump not operating.

Parallel-connected pumps can also be double pumps, where the pump casings are casted in the same unit, and where the non-return valves are build-in as one or more valves to prevent backflow through the pumps. The characteristics of a parallel-connected system is found by adding the single characteristics for each pump horisontally, see figure 3.5.

Pumps connected in parallel are e.g. used in pressure booster systems, for water supply and for water supply in larger buildings. Major operational advantages can be achieved in a pressure booster system by connecting two or more pumps in parallel instead of installing one big pump. The total pump output is usually only necessarry in a limited period. A single large pump will in this case typically operate at lower efficiency.

By letting a number of smaller pumps take care of the operation, the system can be controlled to minimize the number of pumps operating and these pumps will operate at the best efficiency point. To operate at the most optimal point, one of the parallel-connected pumps must have variable speed control.

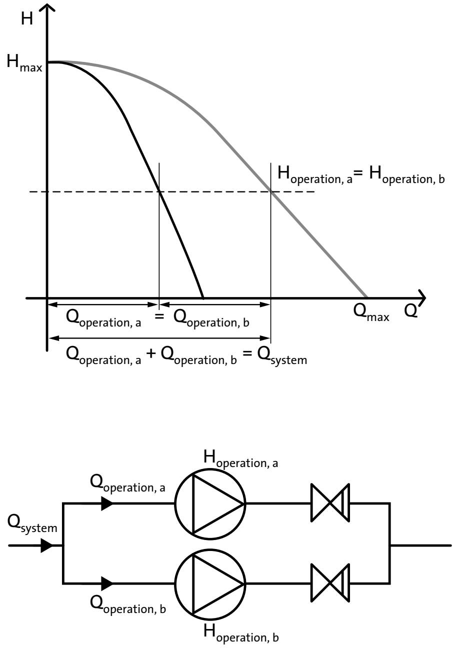


Figure 3.5: Parallel-connected pumps.

3.3 Pumps operated in series

Centrifugal pumps are rarely connected in serial, but a multi-stage pump can be considered as a serial connection of single-stage pumps. However, single stages in multistage pumps can not be uncoupled.

If one of the pumps in a serial connection is not operating, it causes a considerable resistance to the system. To avoid this, a bypass with a non-return valve could be build-in, see figure 3.6. The head at a given flow for a serial-connected pump is found by adding the single heads vertically, as shown in figure 3.6.

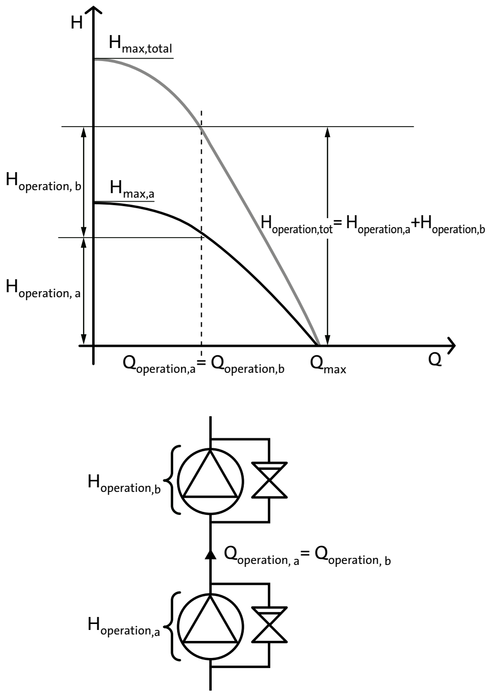


Figure 3.6: Pumps connected in series.

3.4 Regulation of pumps

It is not always possible to find a pump that matches the requested performance exactly. A number of methods makes it possible to regulate the pump performance and thereby achieve the requested performance. The most common methods are:

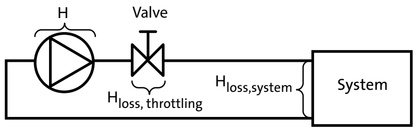
1. Throttle regulation, also known as expansion regulation
2. Bypass regulation through a bypass valve
3. Start/stop regulation
4. Regulation of speed

There are also a number of other regulation methods e.g. control of pre-swirl rotation, adjustment of blades, trimming the impeller and cavitation control which are not introduced further in this book.

3.4.1 Throttle regulation

Installing a throttle valve in serial with the pump it can change the system characteristic, see figure 3.7. The resistance in the entire system can be regulated by changing the valve settings and thereby adjusting the flow as needed. A lower power consumption can sometimes be achieved by installing a throttle valve. However, it depends on the power curve and thus the specific speed of the pump. Regulation by means of a throttle valve is best suited for pumps with a relatve high pressure compared to flow (low-n_q pumps described in section 4.6), see figure 3.8.

Figure 3.7: Principle sketch of a throttle regulation.



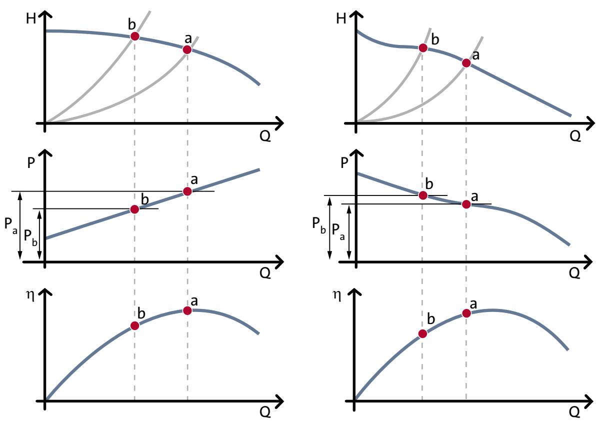


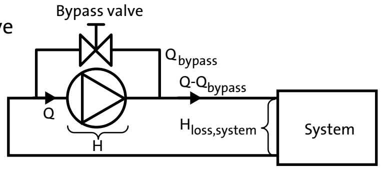
Figure 3.8: The system characteristic is changed through throttle regulation. The curves to the left show throttling of a low n_q pump and the curves to the right show throttling of a high n_q pump. The operating point is moved from a to b in both cases.

3.4.2 Regulation with bypass valve

A bypass valve is a regulation valve installed parallel to the pump, see figure 3.9. The bypass valve guide part of the flow back to the suction line and concequently reduces the head. With a bypass valve, the pump delivers a specific flow even though the system is completely cut off. Like the throttle valve, it is possible to reduce the power consumption in some case. Bypasss regulation is an advantage for pumps with low head compared to flow (high n_q pumps), see figure 3.10.

From an overall perspective neither regulation with throttle valve nor bypass valve are an energy efficient solution and should be avoided.

Figure 3.9: The bypass valve leads a part of the flow back to the suction line and thereby reduces the flow into the system.



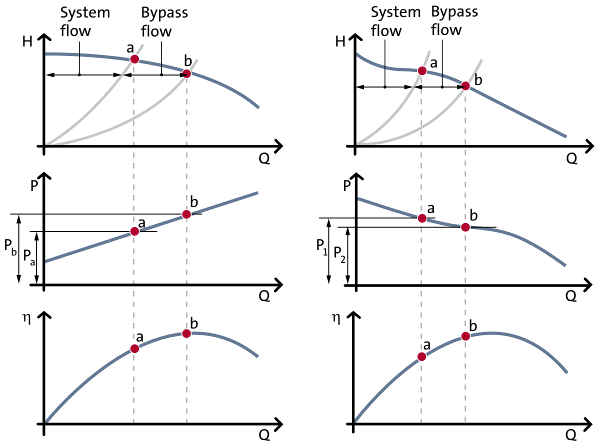


Figure 3.10: The system characteristic is changed through bypass regulation. To the left the consequence of a low-n_q pump is shown and to the right the concequences of a high n_q pump is shown. The operating point is moved from a to b in both cases.

3.4.3 Start/stop regulation

In systems with varying pump requirements, it can be an advantage to use a number of smaller parallel-connected pumps instead of one larger pump. The pumps can then be started and stopped depending on the load and a better adjustment to the requirements can be achieved.

3.4.4 Speed control

When the pump speed is regulated, the QH, power and NPSH curves are changed. The conversion in speed is made by means of the affinity equations. These are futher described in section 4.5:

$$Q_B = Q_A \cdot \frac{n_B}{n_A} \tag{3.1}$$

$$H_B = H_A \cdot \left(\frac{n_B}{n_A}\right)^2 \tag{3.2}$$

$$P_B = P_A \cdot \left(\frac{n_B}{n_A}\right)^3 \tag{3.3}$$

$$NPSH_B = NPSH_A \cdot \left(\frac{n_B}{n_A}\right)^2 \tag{3.4}$$

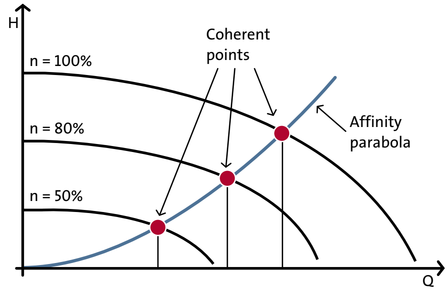


Figure 3.11: Affinity parabola in the QH graph.

Index A in the equations describes the initial values, and index B describes the modified values.

The equations provide coherent points on an affinity parabola in the QH graph. The affinity parabola is shown in figure 3.11.

Different regulation curves can be created based on the relation between the pump curve and the speed. The most common regulation methods are proportional-pressure control and constant-pressure control.

Proportional-pressure control

Proportional-pressure control strives to keep the pump head proportional to the flow. This is done by changing the speed in relation to the current flow. Regulation can be performed up to a maximum speed, from that point the curve will follow this speed. The proportional curve is an approximative system characteristic as described in section 3.1 where the needed flow and head can be delivered at varying needs.

Proportional pressure regulation is used in closed systems such as heating systems. The differential pressure, e.g. above radiator valves, is kept almost constant despite changes in the heat consumption. The result is a low energy consumption by the pump and a small risk of noise from valves.

Figure 3.12 shows different proportional-pressure regulation curves.

Constant-pressure control

A constant differential pressure, independent of flow, can be kept by means of constant-pressure control. In the QH diagram the pump curve for constant-pressure control is a horisontal line, see figure 3.13. Constant-pressure control is an advantage in many water supply systems where changes in the consumption at a tapping point must not affect the pressure at other tapping points in the system.

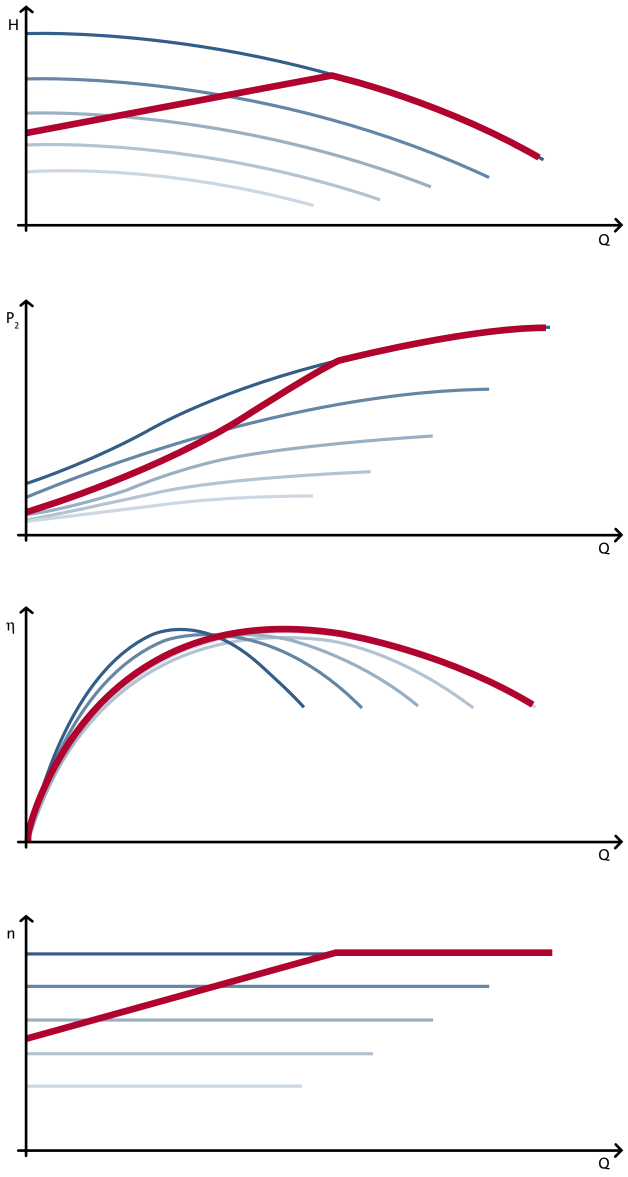


Figure 3.12: Example of proportional-pressure control.

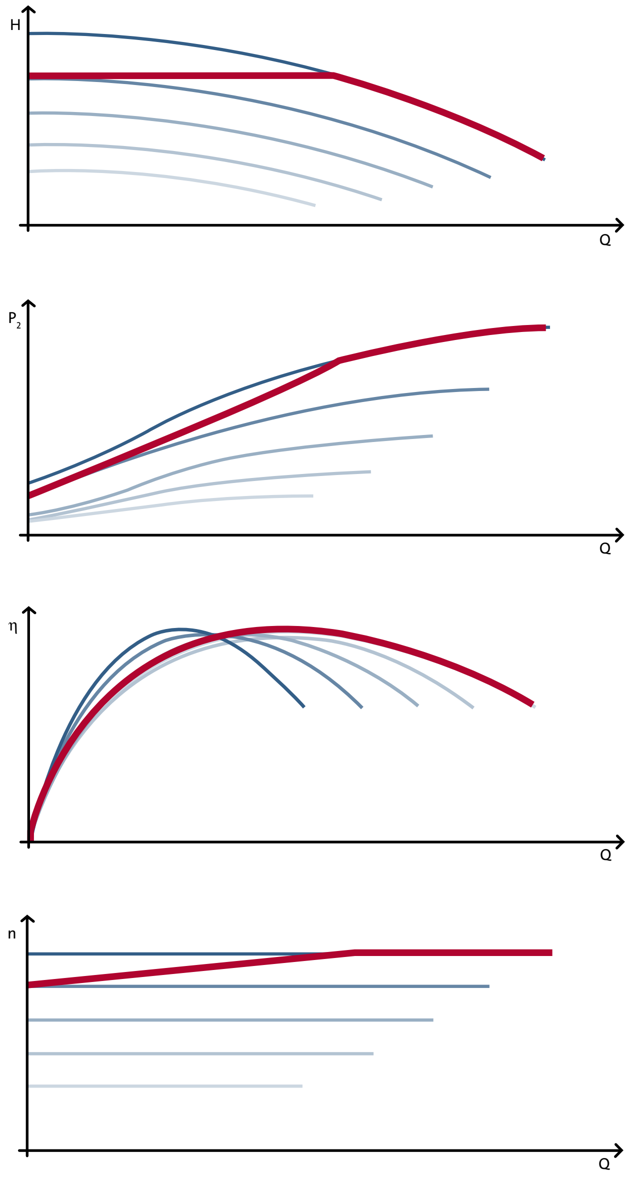


Figure 3.13: Example of constant-pressure control.

3.5. Annual energy consumption

Like energy labelling of refrigerators and freezers, a corresponding labelling for pumps exists. This energy label applies for small circulation pumps and makes it easy for consumers to choose a pump which minimises the power consumption. The power consumption of a single pump is small but because the worldwide number of installed pumps is very large, the accumulated energy consumption is big. The lowest energy consumption is achieved with speed regulation of pumps.

The energy label is based on a number of tests showing the annual runtime and flow of a typical circulation pump. The tests result in a load profile defined by a nominal operating point ($Q_{100\%}$) and a corresponding distribution of the operating time.

The nominal operating point is the point on the pump curve where the product of Q and H is the highest. The same flow point also refers to $P_{100\%}$, see figure 3.14. Figure 3.15 shows the time distribution for each flow point.

The representative power consumption is found by reading the power consumption at the different operating points and multiplying this with the time expressed in percent.

$$P_{L,avg} = 0.06 \cdot P_{100\%} + 0.15 \cdot P_{75\%} + 0.35 \cdot P_{50\%} + 0.44 \cdot P_{25\%} \qquad (3.5)$$

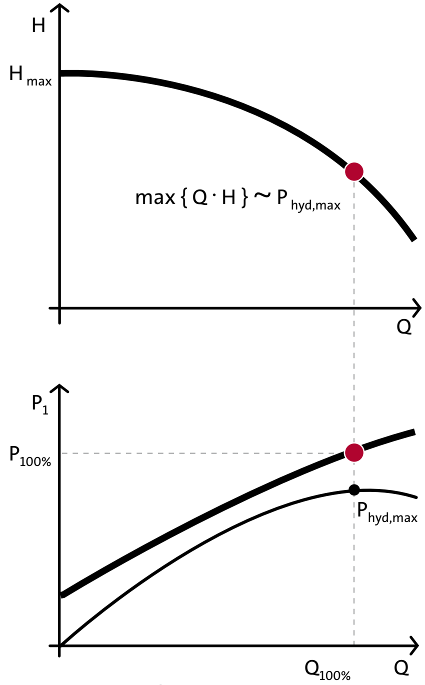


Figure 3.14: Load curve.

Flow %	Time %
100	6
75	15
50	35
25	44

Figure 3.15: Load profile.

3.6 Energy efficiency index (EEI)

In 2003 a study of a major part of the circulation pumps on the market was conducted. The purpose was to create a frame of reference for a representative power consumption for a specific pump. The result is the curve shown in figure 3.16. Based on the study the magnitude of a representative power consumption of an average pump at a given $P_{hyd,max}$ can be read from the curve.

The energy index is defined as the relation between the representative power ($P_{L,avg}$) for the pump and the reference curve. The energy index can be interpreted as an expression of how much energy a specific pump uses compared to average pumps on the market in 2003.

$$EEI = \frac{P_{L,avg}}{P_{Ref}} \quad [-] \tag{3.6}$$

If the pump index is no more than 0.40, it can be labelled energy class A. If the pump has an index between 0.40 and 0.60, it is labelled energy class B. The scale continues to class G, see figure 3.17.

Speed regulated pumps minimize the energy consumption by adjusting the pump to the required performance. For calculation of the energy index, a reference control curve corresponding to a system characteristic for a heating system is used, see figure 3.18. The pump performance is regulated through the speed and it intersects the reference control curve instead of following the maximum curve at full speed. The result is a lower power consumption in the regulated flow points and thereby a better energy index.

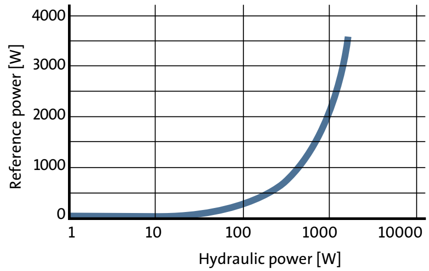


Figure 3.16: Reference power as function of $P_{hyd,max}$.

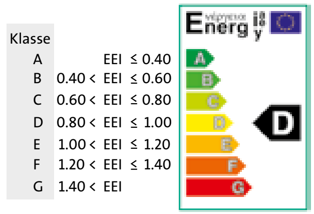


Figure 3.17: Energy classes.

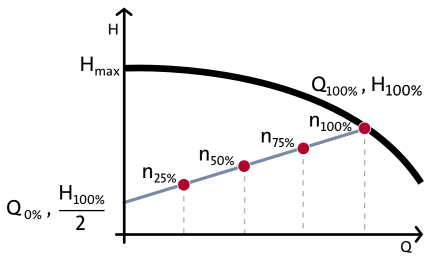


Figure 3.18: Reference control curve.

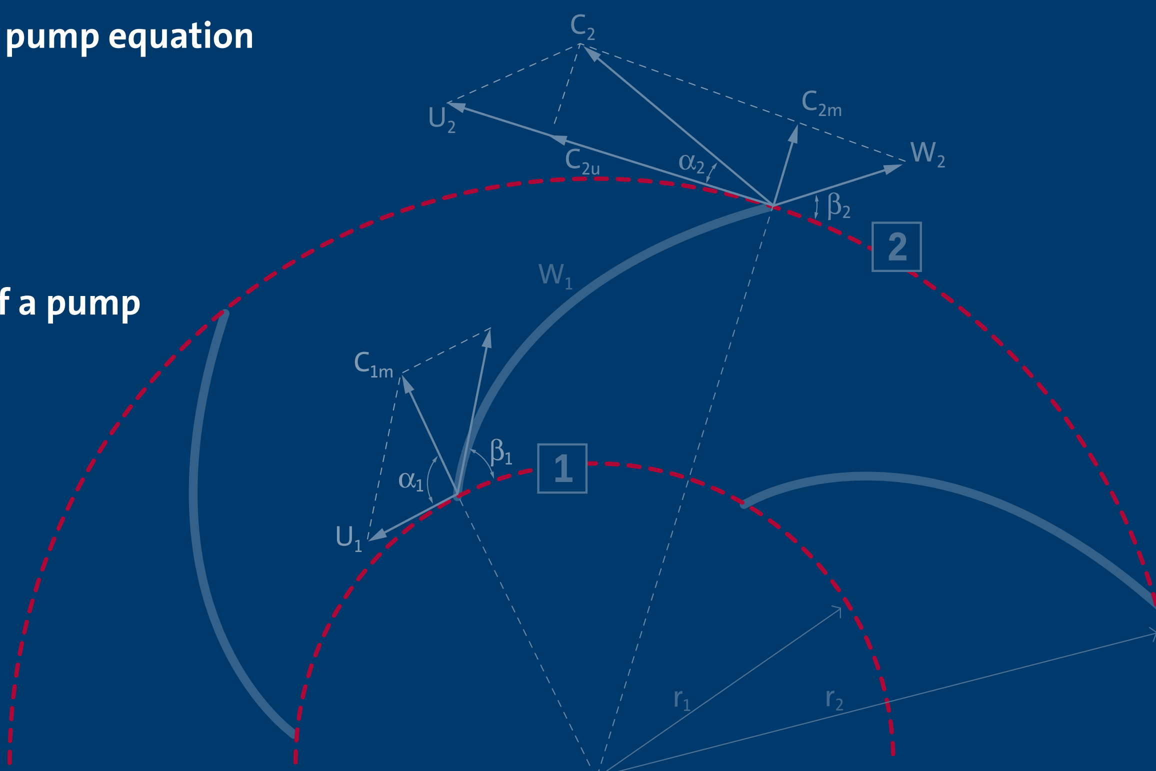
3.7 Summary

In chapter 3 we have studied the correlation between pump and system from a single circulation pump to water supply systems with several parallel coupled multi-stage pumps.

We have described the most common regulation methods from an energy efficient view point and introduced the energy index term.

Chapter 4

Pump theory



4. Pump theory

The purpose of this chapter is to describe the theoretical foundation of energy conversion in a centrifugal pump. Despite advanced calculation methods which have seen the light of day in the last couple of years, there is still much to be learned by evaluating the pump's performance based on fundamental and simple models.

When the pump operates, energy is added to the shaft in the form of mechanical energy. In the impeller it is converted to internal (static pressure) and kinetic energy (velocity). The process is described through Euler's pump equation which is covered in this chapter. By means of velocity triangles for the flow in the impeller in- and outlet, the pump equation can be interpreted and a theoretical loss-free head and power consumption can be calculated.

Velocity triangles can also be used for prediction of the pump's performance in connection with changes of e.g. speed, impeller diameter and width.

4.1 Velocity triangles

For fluid flowing through an impeller it is possible to determine the absolute velocity (C) as the sum of the relative velocity (W) with respect to the impeller, i.e. the tangential velocity of the impeller (U). These velocity vectors are added through vector addition, forming velocity triangles at the in- and outlet of the impeller. The relative and absolute velocity are the same in the stationary part of the pump.

The flow in the impeller can be described by means of velocity triangles, which state the direction and magnitude of the flow. The flow is three-dimensional and in order to describe it completely, it is necessary to make two plane illustrations. The first one is the meridional plane which is an axial cut through the pump's centre axis, where the blade edge is mapped into the plane, as shown in figure 4.1. Here index 1 represents the inlet and index 2 represents the outlet. As the tangential velocity is perpendicular to this plane, only absolute velocities are present in the figure. The plane shown in figure 4.1 contains the meridional velocity, C_m, which runs along the channel and is the vector sum of the axial velocity, C_a, and the radial velocity, C_r.

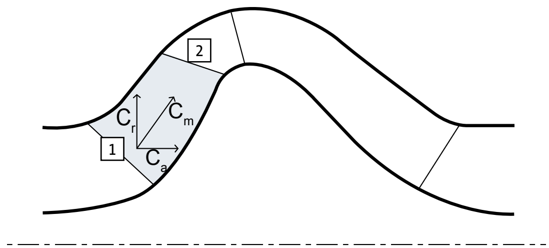
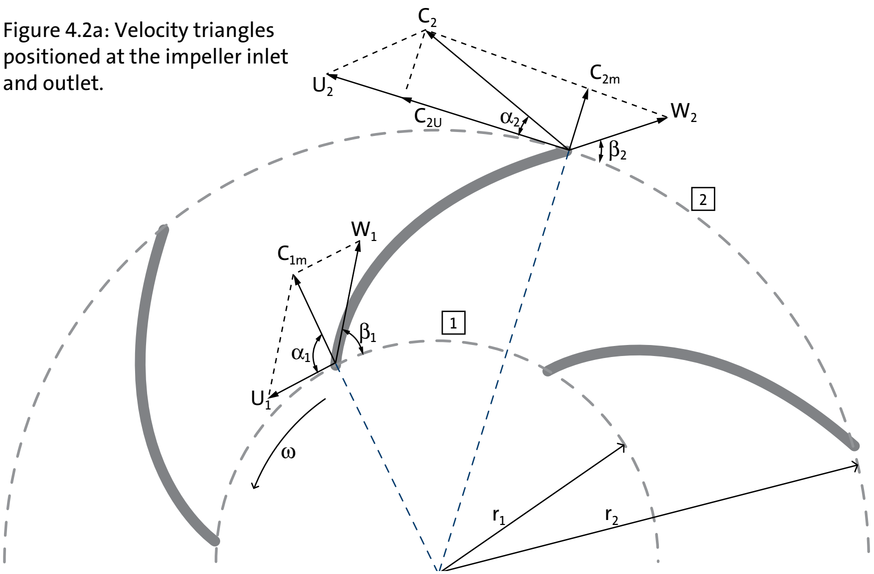


Figure 4.1: Meridional cut.

Figure 4.2a: Velocity triangles positioned at the impeller inlet and outlet.



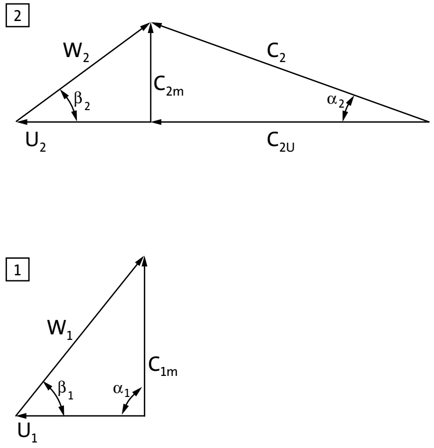


Figure 4.2b: Velocity triangles

The second plane is defined by the meridional velocity and the tangential velocity.

An example of velocity triangles is shown in figure 4.2. Here U describes the impeller's tangential velocity while the absolute velocity C is the fluid's velocity compared to the surroundings. The relative velocity W is the fluid velocity compared to the rotating impeller. The angles α and β describe the fluid's relative and absolute flow angles respectively compared to the tangential direction.

Velocity triangles can be illustrated in two different ways and both ways are shown in figure 4.2a and b. As seen from the figure the same vectors are repeated. Figure 4.2a shows the vectors compared to the blade, whereas figure 4.2b shows the vectors forming a triangle.

By drawing the velocity triangles at inlet and outlet, the performance curves of the pump can be calculated by means of Euler's pump equation which will be described in section 4.2.

4.1.1 Inlet

Usually it is assumed that the flow at the impeller inlet is non-rotational. This means that α_1=90°. The triangle is drawn as shown in figure 4.2 position 1, and C_{1m} is calculated from the flow and the ring area in the inlet.
The ring area can be calculated in different ways depending on impeller type (radial impeller or semi-axial impeller), see figure 4.3. For a radial impeller this is:

$$A_1 = 2\pi \cdot r_1 \cdot b_1 \quad [m^2] \tag{4.1}$$

where
r_1 = The radial position of the impeller's inlet edge [m]
b_1 = The blade's height at the inlet [m]

and for a semi-axial impeller this is:

$$A_1 = 2 \cdot \pi \cdot \left(\frac{r_{1,\,hub} + r_{1,\,shroud}}{2} \right) \cdot b_1 \qquad [m^2] \tag{4.2}$$

The entire flow must pass through this ring area. C_{1m} is then calculated from:

$$C_{1m} = \frac{Q_{impeller}}{A_1} \quad \left[\frac{m}{s}\right] \tag{4.3}$$

The tangential velocity U_1 equals the product of radius and angular frequency:

$$U_1 = 2 \cdot \pi \cdot r_1 \cdot \frac{n}{60} = r_1 \cdot \omega \qquad \left[\frac{m}{s}\right] \tag{4.4}$$

where
ω = Angular frequency [s^{-1}]
n = Rotational speed [min^{-1}]

When the velocity triangle has been drawn, see figure 4.4, based on α_1, C_{1m} and U_1, the relative flow angle β_1 can be calculated. Without inlet rotation ($C_1 = C_{1m}$) this becomes:

$$\tan \beta_1 = \frac{C_{1m}}{U_1} \qquad [-] \tag{4.5}$$

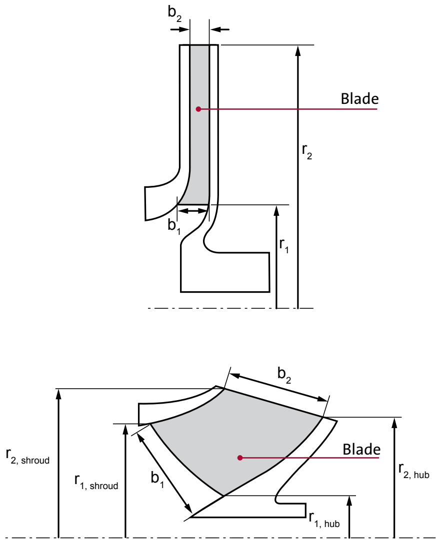


Figure 4.3: Radial impeller at the top, semi-axial impeller at the bottom.

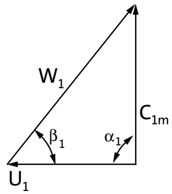


Figure 4.4: Velocity triangle at inlet.

4.1.2 Outlet

As with the inlet, the velocity triangle at the outlet is drawn as shown in figure 4.2 position 2. For a radial impeller, outlet area is calculated as:

$$A_2 = 2\pi \cdot r_2 \cdot b_2 \quad [m^2] \tag{4.6}$$

and for a semi axial impeller it is:

$$A_2 = 2 \cdot \pi \cdot \left(\frac{r_{2,\,hub} + r_{2,\,shroud}}{2}\right) \cdot b_2 \quad [m^2] \tag{4.7}$$

C_{2m} is calculated in the same way as for the inlet:

$$C_{2m} = \frac{Q_{impeller}}{A_2} \quad \left[\frac{m}{s}\right] \tag{4.8}$$

The tangential velocity U is calculated from the following:

$$U_2 = 2 \cdot \pi \cdot r_2 \cdot \frac{n}{60} = r_2 \cdot \omega \quad \left[\frac{m}{s}\right] \tag{4.9}$$

In the beginning of the design phase, β_2 is assumed to have the same value as the blade angle. The relative velocity can then be calculated from:

$$W_2 = \frac{C_{2m}}{\sin\beta_2} \quad \left[\frac{m}{s}\right] \tag{4.10}$$

and C_{2U} as:

$$C_{2U} = U_2 - \frac{C_{2m}}{\tan\beta_2} \quad \left[\frac{m}{s}\right] \tag{4.11}$$

Hereby the velocity triangle at the outlet has been determined and can now be drawn, see figure 4.5.

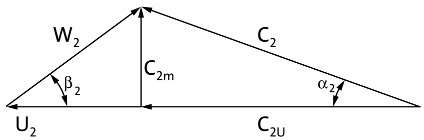


Figure 4.5: Velocity triangle at outlet.

4.2 Euler's pump equation

Euler's pump equation is the most important equation in connection with pump design. The equation can be derived in many different ways. The method described here includes a control volume which limits the impeller, the moment of momentum equation which describes flow forces and velocity triangles at inlet and outlet.

A control volume is an imaginary limited volume which is used for setting up equilibrium equations. The equilibrium equation can be set up for torques, energy and other flow quantities which are of interest. The moment of momentum equation is one such equilibrium equation, linking mass flow and velocities with impeller diameter. A control volume between 1 and 2, as shown in figure 4.6, is often used for an impeller.

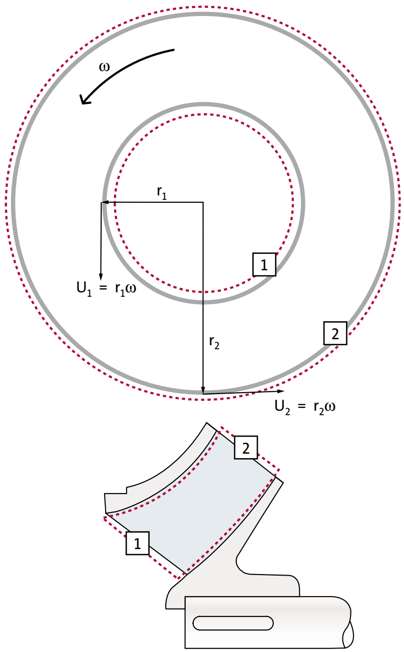


Figure 4.6: Control volume for an impeller.

The balance which we are interested in is a torque balance. The torque (T) from the drive shaft corresponds to the torque originating from the fluid's flow through the impeller with mass flow $\dot{m}=\rho Q$:

$$T = \dot{m} \cdot (r_2 \cdot C_{2U} - r_1 \cdot C_{1U}) \quad [\mathrm{Nm}] \tag{4.12}$$

By multiplying the torque by the angular velocity, an expression for the shaft power (P_2) is found. At the same time, radius multiplied by the angular velocity equals the tangential velocity, $r_2 w = U_2$. This results in:

$$\begin{aligned} P_2 &= T \cdot \omega \quad [\mathrm{W}] \\ &= \dot{m} \cdot \omega \cdot (r_2 \cdot C_{2U} - r_1 \cdot C_{1U}) \\ &= \dot{m} \cdot (\omega \cdot r_2 \cdot C_{2U} - \omega \cdot r_1 \cdot C_{1U}) \\ &= \dot{m} \cdot (U_2 \cdot C_{2U} - U_1 \cdot C_{1U}) \\ &= Q \cdot \rho \cdot (U_2 \cdot C_{2U} - U_1 \cdot C_{1U}) \end{aligned} \tag{4.13}$$

According to the energy equation, the hydraulic power added to the fluid can be written as the increase in pressure Δp_{tot} across the impeller multiplied by the flow Q:

$$P_{hyd} = \Delta p_{tot} \cdot Q \quad [\mathrm{W}] \tag{4.14}$$

The head is defined as:

$$H = \frac{\Delta p_{tot}}{\rho \cdot g} \quad [m] \tag{4.15}$$

and the expression for hydraulic power can therefore be transcribed to:

$$P_{hyd} = Q \cdot H \cdot \rho \cdot g = m \cdot H \cdot g \quad [W] \tag{4.16}$$

If the flow is assumed to be loss free, then the hydraulic and mechanical power can be equated:

$$\begin{aligned} & P_{hyd} = P_2 \\ \Updownarrow \\ & m \cdot H \cdot g = m \cdot (U_2 \cdot C_{2U} - U_1 \cdot C_{1U}) \\ \Updownarrow \\ & H = \frac{(U_2 \cdot C_{2U} - U_1 \cdot C_{1U})}{g} \end{aligned} \tag{4.17}$$

This is the equation known as Euler's equation, and it expresses the impeller's head at tangential and absolute velocities in inlet and outlet.
If the cosine relations are applied to the velocity triangles, Euler's pump equation can be written as the sum of the three contributions:

- Static head as consequence of the centrifugal force
- Static head as consequence of the velocity change through the impeller
- Dynamic head

$$H = \underbrace{\frac{U_2^2 - U_1^2}{2 \cdot g}}_{\text{Static head as consequence of the centrifugal force}} + \underbrace{\frac{W_1^2 - W_2^2}{2 \cdot g}}_{\text{Static head as consequence of the velocity change through the impeller}} + \underbrace{\frac{C_2^2 - C_1^2}{2 \cdot g}}_{\text{Dynamic head}} \quad [m] \tag{4.18}$$

If there is no flow through the impeller and it is assumed that there is no inlet rotation, then the head is only determined by the tangential velocity based on (4.17) where $C_{2U} = U_2$:

$$H_0 = \frac{U_2^2}{g} \quad [m] \tag{4.19}$$

When designing a pump, it is often assumed that there is no inlet rotation meaning that C_{1U} equeals zero.

$$H = \frac{U_2 \cdot C_{2U}}{g} \quad [m] \tag{4.20}$$

4.3 Blade shape and pump curve

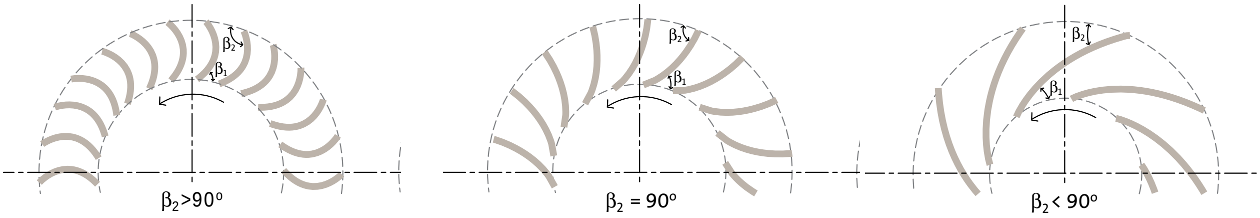


Figure 4.7: Blade shapes depending on outlet angle

If it is assumed that there is no inlet rotation (C_{1U} =0), a combination of Euler's pump equation (4.17) and equation (4.6), (4.8) and (4.11) show that the head varies linearly with the flow, and that the slope depends on the outlet angle β_2:

$$H = \frac{U_2^2}{g} - \frac{U_2}{\pi \cdot D_2 \cdot b_2 \cdot g \cdot \tan(\beta_2)} \cdot Q \quad [m] \tag{4.21}$$

Figure 4.7 and 4.8 illustrate the connection between the theoretical pump curve and the blade shape indicated at β_2.

Real pump curves are, however, curved due to different losses, slip, inlet rotation, etc., This is further discussed in chapter 5.

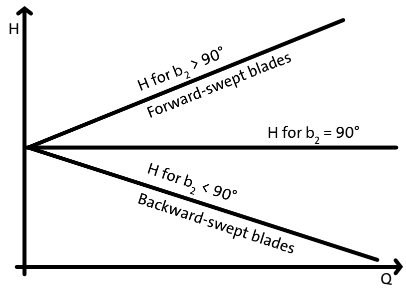
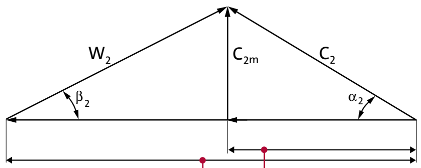


Figure 4.8: Theoretical pump curves calculated based on formula (4.21).

4.4 Usage of Euler's pump equation

There is a close connection between the impeller geometry, Euler's pump equation and the velocity triangles which can be used to predict the impact of changing the impeller geometry on the head.

The individual part of Euler's pump equation can be identified in the outlet velocity triangle, see figure 4.9.



$$H = \frac{1}{g} \cdot U_2 \cdot C_{2U}$$

Figure 4.9: Euler's pump equation and the matching vectors on velocity triangle

This can be used for making qualitative estimates of the effect of changing impeller geometry or rotational speed.

In the following, the effect of reducing the outlet width b_2 on the velocity triangles is discussed. From e.g. (4.6) and (4.8), the velocity C_{2m} can be seen to be inversely proportional to b_2. The size of C_{2m} therefore increases when b_2 decreases. U_2 in equation (4.9) is seen to be independent of b_2 and remains constant. The blade angle β_2 does not change when changing b_2.

The velocity triangle can be plotted in the new situation, as shown in figure 4.10. The figure shows that the velocities C_{2U} and C_2 will decrease and that W_2 will increase. The head will then decrease according to equation (4.21). The power which is proportional to the flow multiplied by the head will decrease correspondingly. The head at zero flow, see formula (4.20), is proportional to U_2^2 and is therefore not changed in this case. Figure 4.11 shows a sketch of the pump curves before and after the change.

Similar analysis can be made when the blade form is changed, see section 4.3, and by scaling of both speed and geometry, see section 4.5.

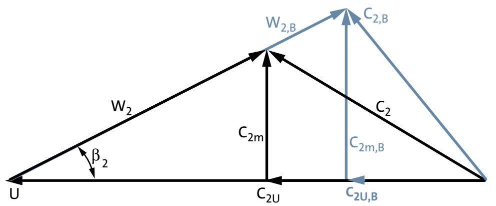


Figure 4.10: Velocity triangle at changed outlet width b_2.

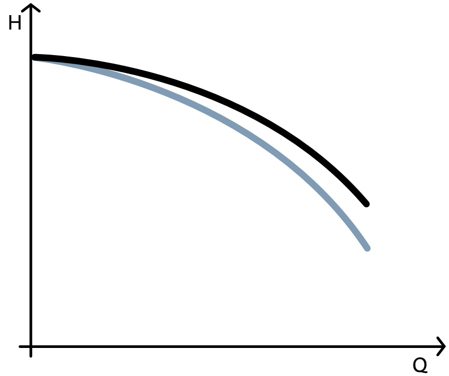


Figure 4.11: Change of head curve as consequence of changed b_2.

4.5 Affinity rules

By means of the so-called affinity rules, the consequences of certain changes in the pump geometry and speed can be predicted with much precision. The rules are all derived under the condition that the velocity triangles are geometrically similar before and after the change. In the formulas below, derived in section 4.5.1, index $_A$ refers to the original geometry and index $_B$ to the scaled geometry.

$$\left.\begin{aligned} Q_B &= Q_A \cdot \left(\frac{n_B}{n_A}\right) \\ H_B &= H_A \cdot \left(\frac{n_B}{n_A}\right)^2 \\ P_B &= P_A \cdot \left(\frac{n_B}{n_A}\right)^3 \end{aligned}\right\} \text{Scaling of rotational speed} \qquad (4.22)$$

$$\left.\begin{aligned} Q_B &= Q_A \cdot \left(\frac{D_B^2 \cdot b_B}{D_A^2 \cdot b_A}\right) \\ H_B &= H_A \cdot \left(\frac{D_B}{D_A}\right)^2 \\ P_B &= P_A \cdot \left(\frac{D_B^4 \cdot b_B}{D_A^4 \cdot b_A}\right) \end{aligned}\right\} \text{Geometric scaling} \qquad (4.23)$$

Figure 4.12 shows an example of the changed head and power curves for a pump where the impeller diameter is machined to different radii in order to match different motor sizes at the same speed. The curves are shown based on formula (4.26).

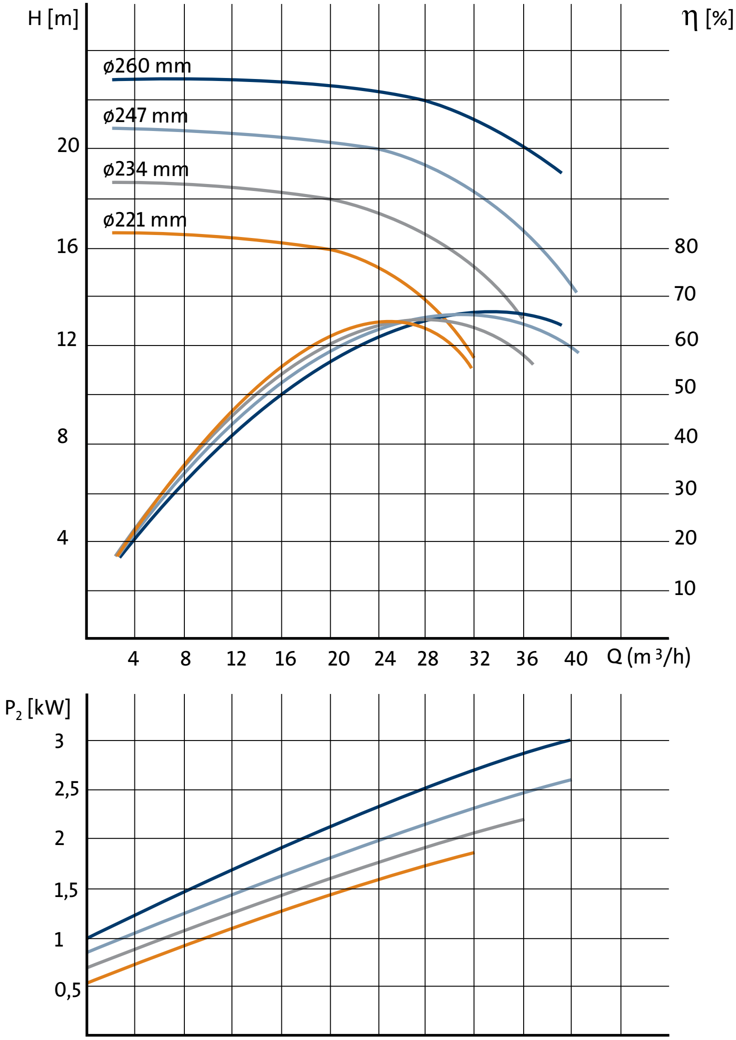


Figure 4.12: Examples of curves for machined impellers at the same speed but different radii.

4.5.1 Derivation of the affinity rules

The affinity method is very precise when adjusting the speed up and down and when using geometrical scaling in all directions (3D scaling). The affinity rules can also be used when wanting to change outlet width and outlet diameter (2D scaling).

When the velocity triangles are similar, then the relation between the corresponding sides in the velocity triangles is the same before and after a change of all components, see figure 4.13. The velocities hereby relate to each other as:

$$\frac{U_B}{U_A} = \frac{C_{m,B}}{C_{m,A}} = \frac{C_{u,B}}{C_{u,A}} \tag{4.24}$$

The tangential velocity is expressed by the speed n and the impeller's outer diameter D_2. The expression above for the relation between components before and after the change of the impeller diameter can be inserted:

$$\frac{U_B}{U_A} = \frac{n_B \cdot D_{2,B}}{n_A \cdot D_{2,A}} \tag{4.25}$$

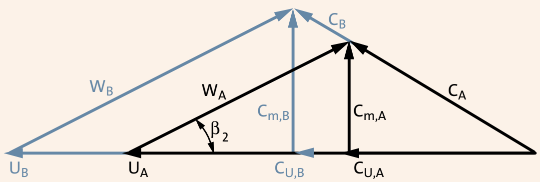


Figure 4.13: Velocity triangle at scaled pump.

Neglecting inlet rotation, the changes in flow, head and power consumption can be expressed as follows:

Flow:

$$Q = A_2 \cdot C_{2m} = \pi \cdot D_2 \cdot b_2 \cdot C_{2m} \tag{4.26}$$

$$\frac{Q_B}{Q_A} = \frac{\pi \cdot D_{2,B} \cdot b_{2,B}}{\pi \cdot D_{2,A} \cdot b_{2,A}} \cdot \frac{C_{2m,B}}{C_{2m,A}} = \frac{\pi \cdot D_{2,B} \cdot b_{2,B} \cdot n_B \cdot D_{2,B}}{\pi \cdot D_{2,A} \cdot b_{2,A} \cdot n_A \cdot D_{2,A}} = \left(\frac{D_{2,B}}{D_{2,A}}\right)^2 \cdot \frac{b_{2,B}}{b_{2,A}} \cdot \frac{n_B}{n_A}$$

Head:

$$H = \frac{U_{2,A} \cdot C_{2U,A}}{g} \tag{4.27}$$

$$\frac{H_B}{H_A} = \frac{U_{2,B} \cdot C_{2U,B} \cdot g}{U_{2,A} \cdot C_{2U,A} \cdot g} = \frac{U_{2,B} \cdot C_{2U,B}}{U_{2,A} \cdot C_{2U,A}} = \frac{n_B \cdot D_{2,B} \cdot n_B \cdot D_{2,B}}{n_A \cdot D_{2,A} \cdot n_A \cdot D_{2,A}} = \left(\frac{D_{2,B}}{D_{2,A}}\right)^2 \cdot \left(\frac{n_B}{n_A}\right)^2$$

Power consumption :

$$P = Q \cdot \rho \cdot U_2 \cdot C_{2U} \tag{4.28}$$

$$\frac{P_B}{P_A} = \frac{Q_B \cdot \rho \cdot U_{2,B} \cdot C_{2U,B}}{Q_A \cdot \rho \cdot U_{2,A} \cdot C_{2U,A}} = \frac{Q_B}{Q_A} \cdot \frac{U_{2,B} \cdot C_{2U,B}}{U_{2,A} \cdot C_{2U,A}} = \frac{Q_B}{Q_A} \cdot \frac{H_B}{H_A} = \left(\frac{D_{2,B}}{D_{2,A}}\right)^4 \frac{b_{2,B}}{b_{2,A}} \cdot \left(\frac{n_B}{n_A}\right)^3$$

4.6 Inlet rotation

Inlet rotation means that the fluid is rotating before it enters the impeller. The fluid can rotate in two ways: either the same way as the impeller (co-rotation) or against the impeller (counter-rotation). Inlet rotation occurs as a consequence of a number of different factors, and a differentation between desired and undesired inlet rotation is made. In some cases inlet rotation can be used for correction of head and power consumption.

In multi-stage pumps the fluid still rotates when it flows out of the guide vanes in the previous stage. The impeller itself can create an inlet rotation because the fluid transfers the impeller's rotation back into the inlet through viscous effects. In practise, you can try to avoid that the impeller itself creates inlet rotation by placing blades in the inlet. Figure 4.14 shows how inlet rotation affects the velocity triangle in the pump inlet.

According to Euler's pump equation, inlet rotation corresponds to C_{1U} being different from zero, see figure 4.14. A change of C_{1U} and then also a change in inlet rotation results in a change in head and hydraulic power. Co-rotation results in smaller head and counter-rotation results in a larger head. It is important to notice that this is not a loss mechanism.

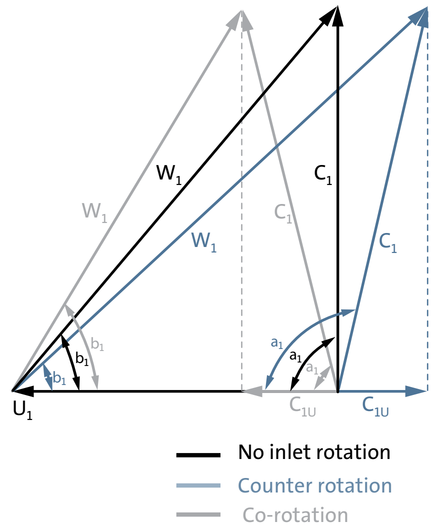


Figure 4.14: Inlet velocity triangle at constant flow and different inlet rotation situations.

4.7 Slip

In the derivation of Euler's pump equation it is assumed that the flow follows the blade. In reality this is, however, not the case because the flow angle usually is smaller than the blade angle. This condition is called slip.

Nevertheless, there is close connection between the flow angle and blade angle. An impeller has an endless number of blades which are extremely thin, then the flow lines will have the same shape as the blades. When the flow angle and blade angle are identical, then the flow is blade congruent, see figure 4.15.

The flow will not follow the shape of the blades completely in a real impeller with a limited number of blades with finite thickness. The tangential velocity out of the impeller as well as the head is reduced due to this.

When designing impellers, you have to include the difference between flow angle and blade angle. This is done by including empirical slip factors in the calculation of the velocity triangles, see figure 4.16. Empirical slip factors are not further discussed in this book.

It is important to emphasize that slip is not a loss mechanism but just an expression of the flow not following the blade.

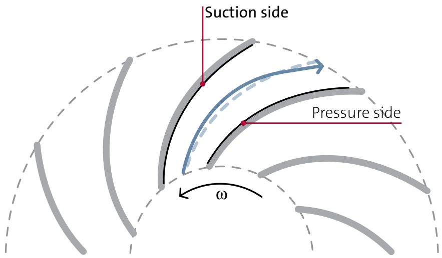
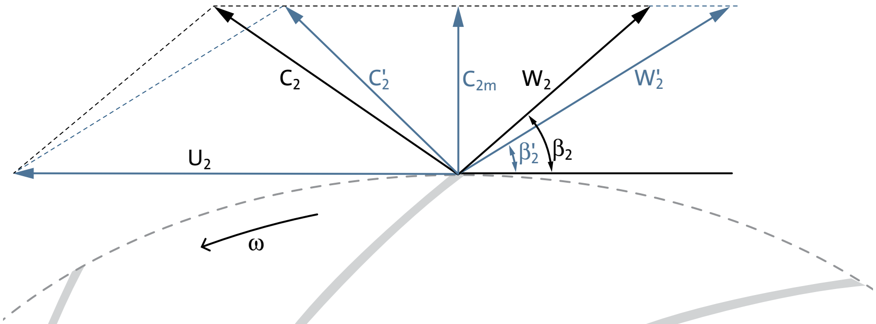


Figure 4.15:
Blade congruent flow line: Dashed line.
Actual flow line: Solid line.



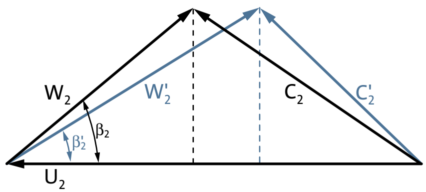


Figure 4.16: Velocity triangles where ' indicates the velocity with slip.

4.8 Specific speed of a pump

As described in chapter 1, pumps are classified in many different ways for example by usage or flange size. Seen from a fluid mechanical point of view, this is, however, not very practical because it makes it almost impossible to compare pumps which are designed and used differently.

A model number, the specific speed (n_q), is therefore used to classify pumps. Specific speed is given in different units. In Europe the following form is commonly used:

$$n_q = n_d \cdot \frac{Q_d^{1/2}}{H_d^{3/4}} \tag{4.29}$$

Where

n_d = rotational speed in the design point [min^{-1}]

Q_d = Flow at the design point [m^3/s]

H_d = Head at the design point [m]

The expression for n_q can be derived from equation (4.22) and (4.23) as the speed which yields a head of 1 m at a flow of 1 m^3/s.

The impeller and the shape of the pump curves can be predicted based on the specific speed, see figur 4.17.

Pumps with low specific speed, so-called low n_q pumps, have a radial outlet with large outlet diameter compared to inlet diameter. The head curves are relatively flat, and the power curve has a positive slope in the entire flow area.

On the contrary, pumps with high specific speed, so-called high n_q pumps, have an increasingly axial outlet, with small outlet diameter compared to the width. Head curves are typically descending and have a tendency to create saddle points. Performance curves decreases when flow increases. Different pump sizes and pump types have different maximum efficiency.

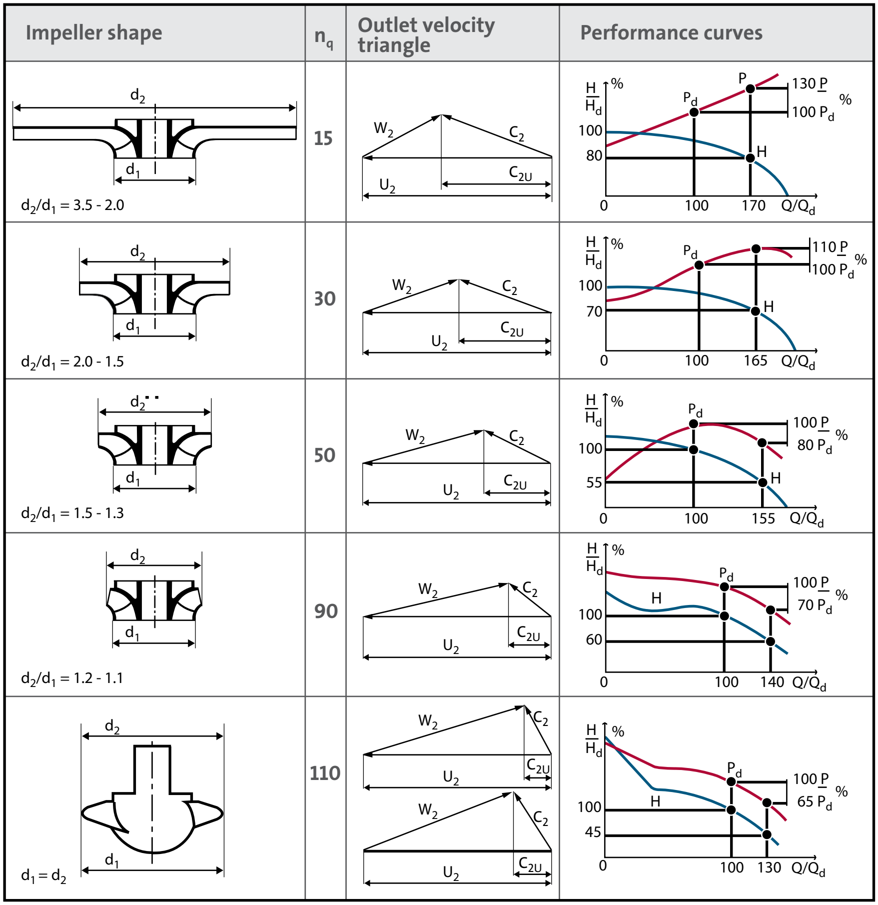


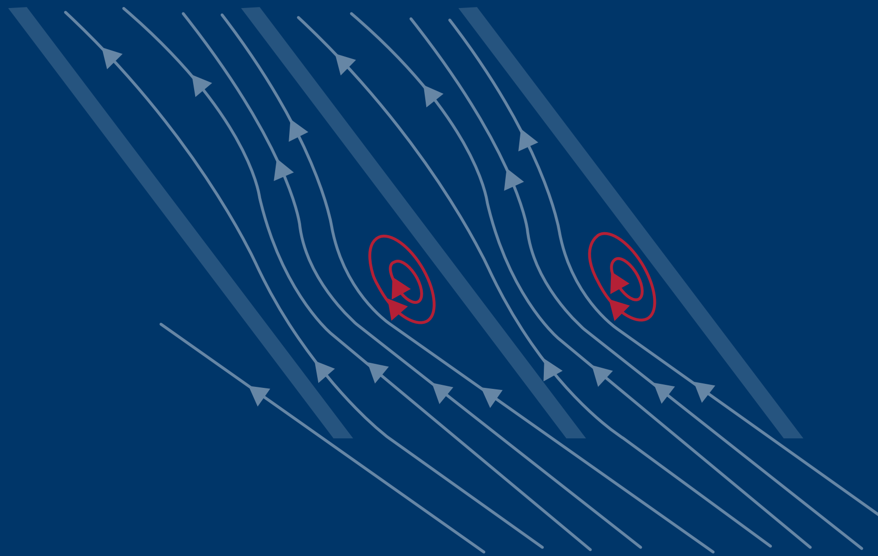
Figure 4.17: Impeller shape, outlet velocity triangle and performance curve as function of specific speed n_q.

4.9 Summary

In this chapter we have described the basic physical conditions which are the basis of any pump design. Euler's pump equation has been desribed, and we have shown examples of how the pump equation can be used to predict a pump's performance. Furthermore, we have derived the affinity equations and shown how the affinity rules can be used for scaling pump performance. Finally, we have introduced the concept of specific speed and shown how different pumps can be differentiated on the basis of this.

Chapter 5

Pump losses



5. Pump losses

As described in chapter 4, Euler's pump equation provides a simple, loss-free description of the impeller performance. In reality, because of a number of mechanical and hydraulic losses in impeller and pump casing, the pump performance is lower than predicted by the Euler pump equation. The losses cause smaller head than the theoretical and higher power consumption, see figures 5.1 and 5.2. The result is a reduction in efficiency. In this chapter we describe the different types of losses and introduce some simple models for calculating the magnitude of the losses. The models can also be used for analysis of the test results, see appendix B.

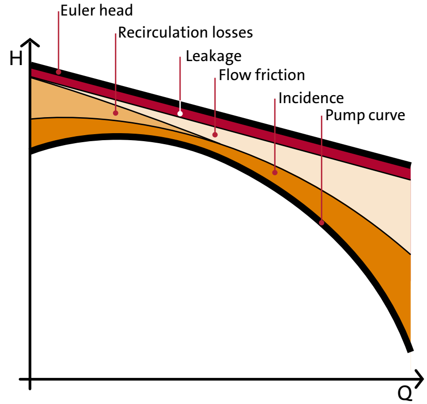


Figure 5.1: Reduction of theoretical Euler head due to losses.

5.1 Loss types

Distinction is made between two primary types of losses: mechanical losses and hydraulic losses which can be divided into a number of subgroups. Table 5.1 shows how the different types of loss affect flow (Q), head (H) and power consumption (P_2).

	Loss	Smaller flow (Q)	Lower head (H)	Higher power consumption (P_2)
Mechanical losses	Bearing			X
	Shaft seal			X
Hydraulic losses	Flow friction		X	
	Mixing		X	
	Recirculation		X	
	Incidence		X	
	Disk friction			X
	Leakage	X		

Chart 5. 1: Losses in pumps and their influence on the pump curves.

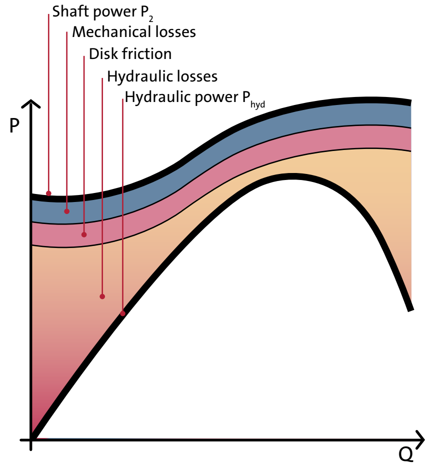


Figure 5.2: Increase in power consumption due to losses.

Pump performance curves can be predicted by means of theoretical or empirical calculation models for each single type of loss. Accordance with the actual performance curves depends on the models' degree of detail and to what extent they describe the actual pump type.

Figure 5.3 shows the components in the pump which cause mechanical and hydraulic losses. It involves bearings, shaft seal, front and rear cavity seal, inlet, impeller and volute casing or return channel. Throughout the rest of the chapter this figure is used for illustrating where each type of loss occurs.

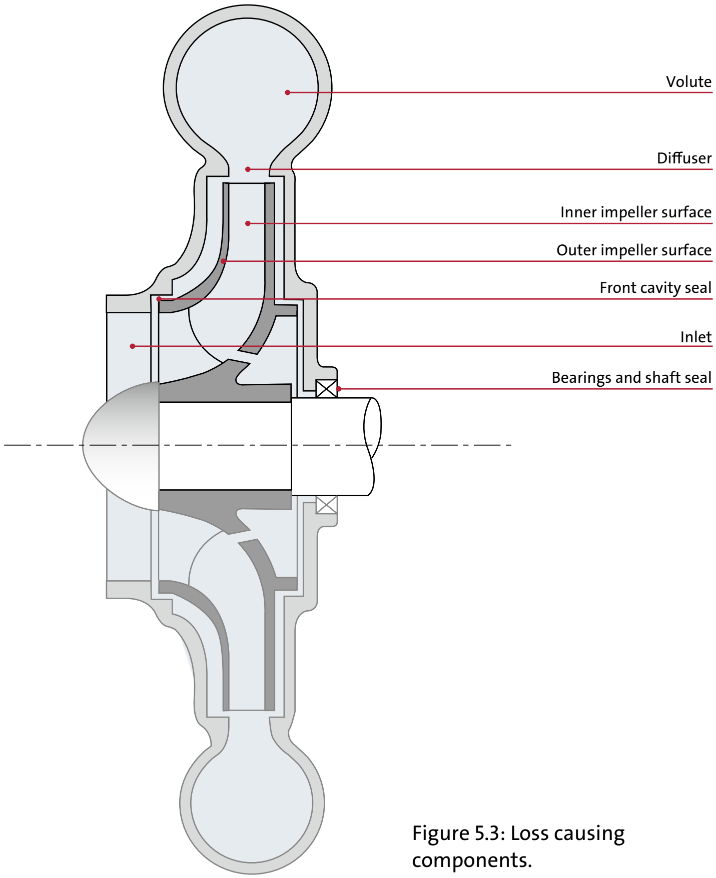
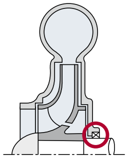


Figure 5.3: Loss causing components.

5.2 Mechanical losses

The pump coupling or drive consists of bearings, shaft seals, gear, depending on pump type. These components all cause mechanical friction loss. The following deals with losses in the bearings and shaft seals.

5.2.1 Bearing loss and shaft seal loss



Bearing and shaft seal losses - also called parasitic losses - are caused by friction. They are often modelled as a constant which is added to the power consumption. The size of the losses can, however, vary with pressure and rotational speed.

The following model estimates the increased power demand due to losses in bearings and shaft seal:

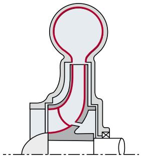
$$P_{\text{loss, mechanical}} = P_{\text{loss, bearing}} + P_{\text{loss, shaft seal}} = \text{constant} \tag{5.1}$$

where

$P_{\text{loss, mechanical}}$ = Increased power demand because of mechanical loss [W]
$P_{\text{loss, bearing}}$ = Power lost in bearings [W]
$P_{\text{loss, shaft seal}}$ = Power lost in shaft seal [W]

5.3 Hydraulic losses

Hydraulic losses arise on the fluid path through the pump. The losses occur because of friction or because the fluid must change direction and velocity on its path through the pump. This is due to cross-section changes and the passage through the rotating impeller. The following sections describe the individual hydraulic losses depending on how they arise.

5.3.1 Flow friction 

Flow friction occurs where the fluid is in contact with the rotating impeller surfaces and the interior surfaces in the pump casing. The flow friction causes a pressure loss which reduces the head. The magnitude of the friction loss depends on the roughness of the surface and the fluid velocity relative to the surface.

Model

Flow friction occurs in all the hydraulic components which the fluid flows through. The flow friction is typically calculated individually like a pipe friction loss, this means as a pressure loss coefficient multiplied with the dynamic head into the component:

$$H_{loss,\,friktion} = \zeta \cdot H_{dyn,\,in} = \zeta \cdot \frac{V^2}{2g} \qquad (5.2)$$

where

ζ = Dimensionless loss coefficient [-]
$H_{dyn,\,in}$ = Dynamic head into the component [m]
V = Flow velocity into the component [m/s]

The friction loss thus grows quadratically with the flow velocity, see figure 5.4.

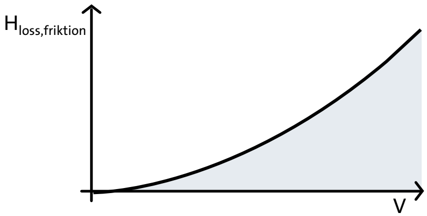


Figure 5.4: Friction loss as function of the flow velocity.

Loss coefficients can be found e.g. in (MacDonald, 1997). Single components such as inlet and outer sleeve which are not directly affected by the impeller can typically be modelled with a constant loss coefficient. Impeller, volute housing and return channel will on the contrary typically have a variable loss coefficient. When the flow friction in the impeller is calculated, the relative velocity must be used in equation (5.2).

Friction loss in pipes

Pipe friction is the loss of energy which occurs in a pipe with flowing fluid. At the wall, the fluid velocity is zero whereas it attains a maximum value at the pipe center. Due to these velocity differences across the pipe, see figure 5.5, the fluid molecules rub against each other. This transforms kinetic energy to heat energy which can be considered as lost.

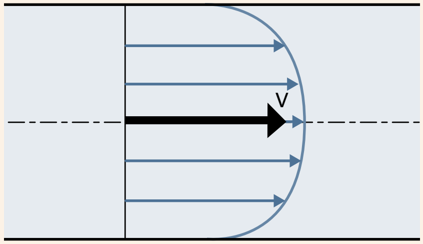


Figure 5.5: Velocity profile in pipe.

To maintain a flow in the pipe, an amount of energy corresponding to the energy which is lost must constantly be added. Energy is supplied by static pressure difference from inlet to outlet. It is said that it is the pressure difference which drives the fluid through the pipe.

The loss in the pipe depends on the fluid velocity, the hydraulic diameter of the pipe, lenght and inner surface roughness. The head loss is calculated as:

$$H_{loss,\,pipe} = f\,\frac{LV^2}{D_h 2g} \qquad (5.3)$$

where

$H_{loss,\,pipe}$ = Head loss [m]
f = Friction coefficient [-]
L = Pipe length [m]
V = Average velocity in the pipe [m/s]
D_h = Hydraulic diameter [m]

The hydraulic diameter is the ratio of the cross-sectional area to the wetted circumference. The hydraulic diameter is suitable for calculating the friction for cross-sections of arbitrary form.

$$D_h = \frac{4A}{O} \qquad (5.4)$$

where

A = The cross-section area of the pipe [m^2]
O = The wetted circumference of the pipe [m]

Equation (5.4) applies in general for all cross-sectional shapes. In cases where the pipe has a circular cross-section, the hydraulic diameter is equal to the pipe diameter. The circular pipe is the cross-section type which has the smallest possible interior surface compared to the cross-section area and therefore the smallest flow resistance.

The friction coefficient is not constant but depends on whether the flow is laminar or turbulent. This is described by the Reynold's number, Re:

$$Re = \frac{VD_h}{\nu} \qquad (5.5)$$

where
ν = Kinematic viscosity of the fluid [m^2/s]

The Reynold's number is a dimensionless number which expresses the relation between inertia and friction forces in the fluid, and it is therefore a number that describes how turbulent the flow is. The following guidelines apply for flows in pipes:

Re < 2300	: Laminar flow
2300 < Re < 500	: Transition zone
Re > 5000	: Turbulent flow.

Laminar flow only occurs at relatively low velocities and describes a calm, well-ordered flow without eddies. The friction coefficient for laminar flow is independent of the surface roughness and is only a function of the Reynold's number. The following applies for pipes with circular cross-section:

$$f_{laminar} = \frac{64}{Re} \qquad (5.6)$$

Turbulent flow is an unstable flow with strong mixing. Due to eddy motion most pipe flows are in practise turbulent. The friction coefficient for turbulent flow depends on the Reynold's number and the pipe roughness.

Figure 5.6 shows a Moody chart which shows the friction coefficient f as function of Reynold's number and surface roughness for laminar and turbulent flows.

Figure 5.6: Moody chart:
Friction coefficient for laminar (circular cross-section) and turbulent flow (arbitrary cross-section). The red line refers to the values in example 5.1.

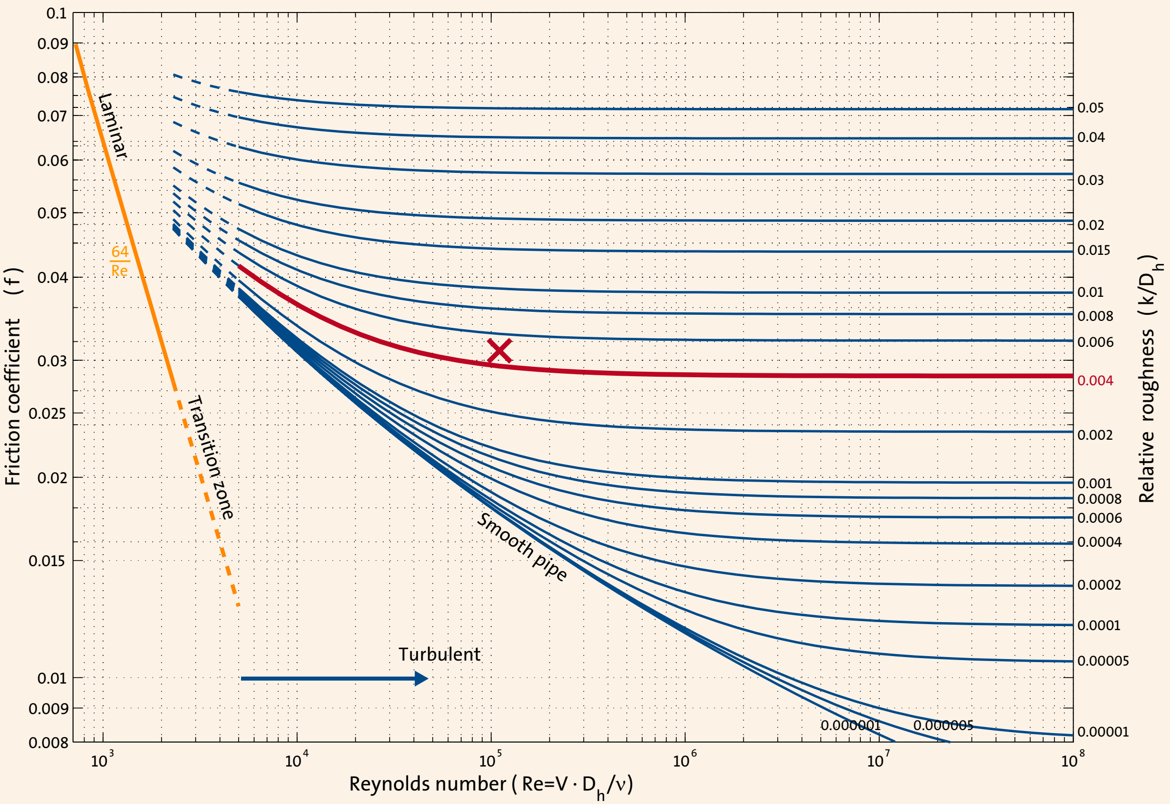


Table 5.2 shows the roughness for different materials. The friction increases in old pipes because of corrosion and sediments.

Materials	Roughness k [mm]
PVC	0.01-0.05
Pipe in aluminium, copper og brass	0-0.003
Steel pipe	0.01-0.05
Welded steel pipe, new	0.03-0.15
Welded steel pipe with deposition	0.15-0.30
Galvanised steel pipe, new	0.1-0.2
Galvanised steel pipe with deposition	0.5-1.0

Table 5.2: Roughness for different surfaces (Pumpeståbi, 2000).

Example 5.1: Calculation of pipe loss

Calculate the pipe loss in a 2 meter pipe with the diameter d=32 mm and a flow of Q=10 m^3/h. The pipe is made of galvanized steel with a roughness of 0.15 mm, and the fluid is water at 20°C.

$$\text{Mean velocity: } V = \frac{Q}{A} = \frac{(10/3600)\ m^3/s}{\frac{\pi}{4}\, 0.032^{2}\ m^2} = 3.45 m/s$$

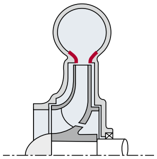
$$\text{Reynolds number: } Re = \frac{VD_h}{\nu} = \frac{3.45 m/s \cdot 0.032 m}{1 \cdot 10^{-6}\ m^2/s} = 110500$$

$$\text{Relative roughness: } k/D_h = \frac{0.15 mm}{32 mm} = 0.0047$$

From the Moody chart, the friction coefficent (f) is 0.031 when Re = 110500 and the relative roughness k/D_h=0.0047. By inserting the values in the equation (5.3), the pipe loss can be calculated to:

$$\text{Pipe loss: } H_{loss,\,pipe} = f\frac{LV^2}{D_h 2g} = 0.031 \frac{2m \cdot (3.45\, m/s)^2}{0.032m \cdot 2 \cdot 9.81\ m/s^2} = 1.2\ m \qquad (5.7)$$

5.3.2 Mixing loss at cross-section expansion



Velocity energy is transformed to static pressure energy at cross-section expansions in the pump, see the energy equation in formula (2.10). The conversion is associated with a mixing loss.

The reason is that velocity differences occur when the cross-section expands, see figure 5.7. The figure shows a diffuser with a sudden expansion beacuse all water particles no longer move at the same speed, friction occurs between the molecules in the fluid which results in a diskharge head loss. Even though the velocity profile after the cross-section expansion gradually is evened out, see figure 5.7, a part of the velocity energy is turned into heat energy instead of static pressure energy.

Mixing loss occurs at different places in the pump: At the outlet of the impeller where the fluid flows into the volute casing or return channel as well as in the diffuser.

When designing the hydraulic components, it is important to create small and smooth cross-section expansions as possible.

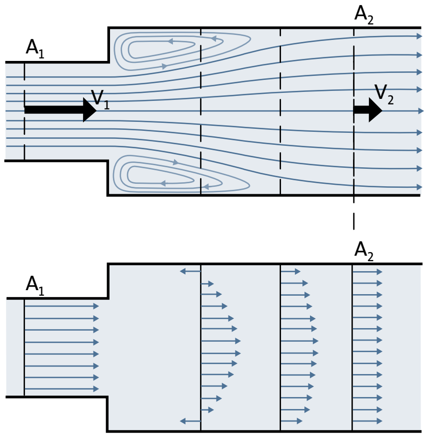


Figure 5.7: Mixing loss at cross-section expansion shown for a sudden expansion.

Model

The loss at a cross-section expansion is a function of the dynamic head into the component.

$$H_{loss,\,expansion} = \zeta \cdot H_{dyn,1} = \zeta \cdot \frac{V_1^{\,2}}{2\,g} \tag{5.8}$$

where

V_1 = Fluid velocity into the component [m/s]

The pressure loss coefficient ζ depends on the area relation between the component's inlet and outlet as well as how evenly the area expansion happens.

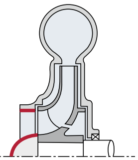
For a sudden expansion, as shown in figure 5.7, the following expression is used:

$$\zeta = \left[1 - \frac{A_1}{A_2}\right]^2 \tag{5.9}$$

where
A_1= Cross-section area at inlet [m^2]
A_2= Cross-section area at outlet [m^2]

The model gives a good estimate of the head loss at large expansion ratios (A_1/A_2 close to zero). In this case the loss coefficient is $\zeta = 1$ in equation (5.9) which means that almost the entire dynamic head into the component is lost in a sharp-edged diffuser.

For small expansion ratios as well as for other diffuser geometries with smooth area expansions, the loss coefficient ζ is found by table lookup (MacDonalds) or by measurements.



5.3.3 Mixing loss at cross-section contraction

Head loss at cross-section contraction occurs as a consequence of eddies being created in the flow when it comes close to the geometry edges, see figure 5.8. It is said that the flow 'separates'. The reason for this is that the flow because of the local pressure gradients no longer adheres in parallel to the surface but instead will follow curved streamlines. This means that the effective cross-section area which the flow experiences is reduced. It is said that a contraction is made. The contraction with the area A_0 is marked on figure 5.8. The contraction accelerates the flow and it must therefore subsequently decelerate again to fill the cross-section. A mixing loss occurs in this process. Head loss as a consequence of cross-section contraction occurs typically at inlet to a pipe and at the impeller eye. The magnitude of the loss can be considerably reduced by rounding the inlet edges and thereby suppress separation. If the inlet is adequately rounded off, the loss is insignificant. Losses related to cross-section contraction is typically of minor importance.

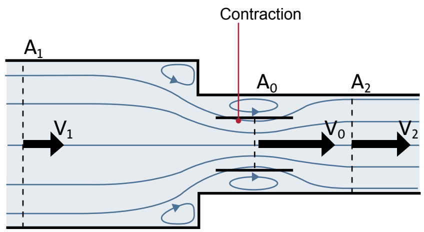


Figure 5.8: Loss at cross-section contraction.

Model

Based on experience, it is assumed that the acceleration of the fluid from V_1 to V_0 is loss-free, whereas the subsequent mixing loss depends on the area ratio now compared to the contraction A_0 as well as the dynamic head in the contraction:

$$H_{loss,\ contraction} = \left[1 - \frac{A_0}{A_2}\right]^2 \cdot \frac{V_0^{\ 2}}{2g} \qquad (5.10)$$

where
V_0 = Fluid velocity in contraction [m/s]
A_0/A_2 = Area ratio [-]

The disadvantage of this model is that it assumes knowledge of A_0 which is not directly measureable. The following alternative formulation is therefore often used:

$$H_{loss,\ contraction} = \zeta \cdot H_{dyn,2} = \zeta \cdot \frac{V_2^{\ 2}}{2g} \qquad (5.11)$$

where
$H_{dyn,2}$ = Dynamic head out of the component [m]
V_2 = Fluid velocity out of the component [m/s]

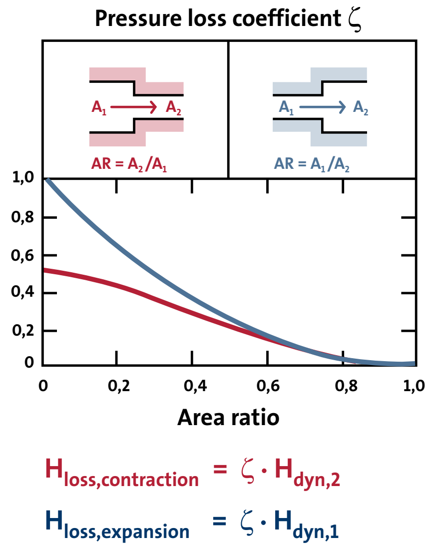
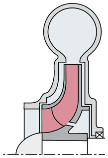


Figure 5.9: Head loss coefficents at sudden cross-section contractions and expansions.

Figure 5.9 compares loss coefficients at sudden cross-section expansions and –contractions as function of the area ratio A_1/A_2 between the inlet and outlet. As shown, the loss coefficient, and thereby also the head loss, is in general smaller at contractions than in expansions. This applies in particular at large area ratios.

The head loss coefficient for geometries with smooth area changes can be found by table lookup. As mentioned earlier, the pressure loss in a cross-section contraction can be reduced to almost zero by rounding off the edges.

5.3.4 Recirculation loss



Recirculation zones in the hydraulic components typically occur at part load when the flow is below the design flow. Figure 5.10 shows an example of recirculation in the impeller. The recirculation zones reduce the effective cross-section area which the flow experiences. High velocity gradients occurs in the flow between the main flow which has high velocity and the eddies which have a velocity close to zero. The result is a considerable mixing loss.

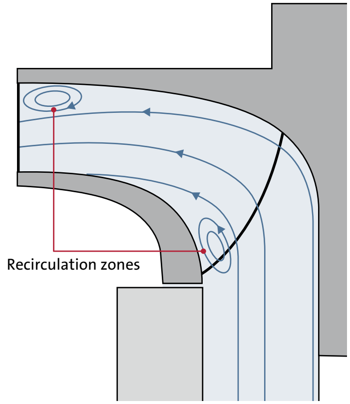


Figure 5.10: Example of recirculation in impeller.

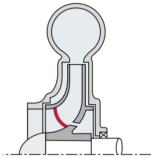
Recirculation zones can occur in inlet, impeller, return channel or volute casing. The extent of the zones depends on geometry and operating point. When designing hydraulic components, it is important to minimise the size of the recirculation zones in the primary operating points.

Model

There are no simple models to describe if recirculation zones occur and if so to which extent. Only by means of advanced laser based velocity measurements or time consuming computer simulations, it is possible to map the recirculation zones in details. Recirculation is therefore generally only identified indirectly through a performance measurement which shows lower head and/or higher power consumption at partial load than predicted.

When designing pumps, the starting point is usually the nominal operating point. Normally reciculation does not occur here and the pump performance can therefore be predicted fairly precisely. In cases where the flow is below the nominal operating point, one often has to use rule of thumb to predict the pump curves.

5.3.5 Incidence loss



Incidence loss occurs when there is a difference between the flow angle and blade angle at the impeller or guide vane leading edges. This is typically the case at part load or when prerotation exists.

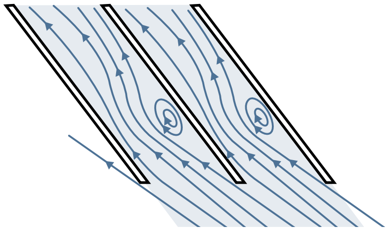


Figure 5.11: Incidence loss at inlet to impeller or guide vanes.

A recirculation zone occurs on one side of the blade when there is difference between the flow angle and the blade angle, see figure 5.11. The recirculation zone causes a flow contraction after the blade leading edge. The flow must once again decelerate after the contraction to fill the entire blade channel and mixing loss occurs.

At off-design flow, incidence losses also occur at the volute tongue. The designer must therefore make sure that flow angles and blade angles match each other so the incidence loss is minimised. Rounding blade edges and volute casing tongue can reduce the incidence loss.

Model

The magnitude of the incidence loss depends on the difference between relative velocities before and after the blade leading edge and is calculated using the following model (Pfleiderer og Petermann, 1990, p 224):

$$H_{loss,\,incidence} = \varphi \, \frac{w_s^{\,2}}{2 \cdot g} = \varphi \, \frac{\left| \vec{w}_1 - \vec{w}_{1,\,kanal} \right|^2}{2 \cdot g} \qquad (5.12)$$

where

φ = Emperical value which is set to 0.5-0.7 depending on the size of the recirculation zone after the blade leading edge.

w_s = difference between relative velocities before and after the blade edge using vector calculation, see figure 5.12.

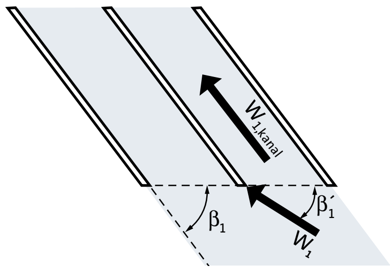


Figure 5.12: Nomenclature for incidence loss model.

Incidence loss is alternatively modelled as a parabola with minimum at the best efficiency point. The incidence loss increases quadratically with the difference between the design flow and the actual flow, see figure 5.13.

$$H_{loss,\ incidence} = k_1 \cdot (Q - Q_{design})^2 + k_2 \qquad (5.13)$$

where

Q_{design} = Design flow [m³/s]

k_1 = Constant [s²/m⁵]

k_2 = Constant [m]

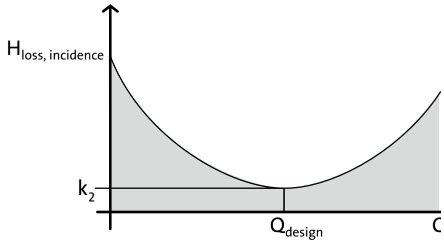
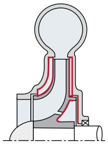


Figure 5.13: Incidence loss as function of the flow.

5.3.6 Disk friction



Disk friction is the increased power consumption which occurs on the shroud and hub of the impeller because it rotates in a fluid-filled pump casing. The fluid in the cavity between impeller and pump casing starts to rotate and creates a primary vortex, see section 1.2.5. The rotation velocity equals the impeller's at the surface of the impeller, while it is zero at the surface of the pump casing. The average velocity of the primary vortex is therefore assumed to be equal to one half of the rotational velocity.

The centrifugal force creates a secondary vortex movement because of the difference in rotation velocity between the fluid at the surfaces of the impeller and the fluid at the pump casing, see figure 5.14. The secondary vortex increases the disk friction because it transfers energy from the impeller surface to the surface of the pump casing.

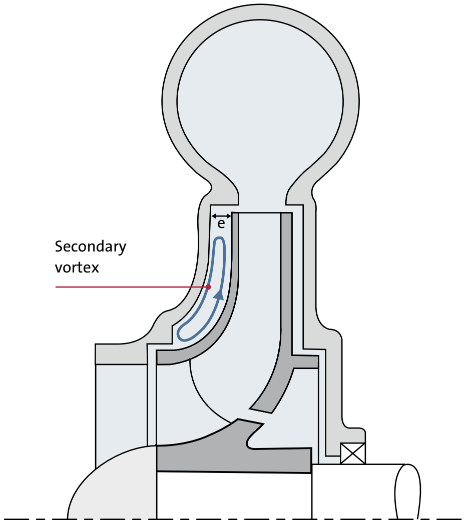


Figure 5.14: Disk friction on impeller.

The size of the disk friction depends primarily on the speed, the impeller diameter as well as the dimensions of the pump housing in particular the distance between impeller and pump casing. The impeller and pump housing surface roughness has, furthermore, a decisive importance for the size of the disk friction. The disk friction is also increased if there are rises or dents on the outer surface of the impeller e.g. balancing blocks or balancing holes.

Model

Pfleiderer and Petermann (1990, p. 322) use the following model to determine the increased power consumption caused by disk friction:

$$P_{loss,\,disk} = k\rho\, U_2^3 D_2\,(D_2 + 5e)$$
$$k = 7.3 \cdot 10^{-4}\left(\frac{2\nu \cdot 10^6}{U_2 D_2}\right)^m \qquad (5.14)$$

where

D_2 = Impeller diameter [m]

e = Axial distance to wall at the periphery of the impeller [m], see figure 5.14

U_2 = Peripheral velocity [m/s]

ν = Kinematic viscosity [m²/s], $\nu = 10^{-6}$ [m²/s] for water at 20°C.

k = Emperical value

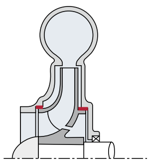
m = Exponent equals 1/6 for smooth surfaces and between 1/7 to 1/9 for rough surfaces

If changes are made to the design of the impeller, calculated disk friction $P_{loss,disk,A}$ can be scaled to estimate the disk friction $P_{loss,disk,B}$ at another impeller diameter or speed:

$$(P_{loss,\,disk})_A = (P_{loss,\,disk})_B \quad \frac{(n^3 D_2^5)_A}{(n^3 D_2^5)_B} \qquad (5.15)$$

The scaling equation can only be used for relative small design changes.

5.3.7 Leakage



Leakage loss occurs because of smaller circulation through gaps between the rotating and fixed parts of the pump. Leakage loss results in a loss in efficiency because the flow in the impeller is increased compared to the flow through the entire pump:

$$Q_{impeller} = Q + Q_{leakage} \quad (5.16)$$

where
$Q_{impeller}$ = Flow through impeller [m³/s], Q = Flow through pump [m³/s] , $Q_{leakage}$ = Leakage flow [m³/s]

Leakage occurs many different places in the pump and depends on the pump type. Figure 5.15 shows where leakage typically occurs. The pressure differences in the pump which drives the leakage flow as shown in figure 5.16.

The leakage between the impeller and the casing at impeller eye and through axial relief are typically of the same size. The leakage flow between guidevane and shaft in multi-stage pumps are less important because both pressure difference and gap area are smaller.

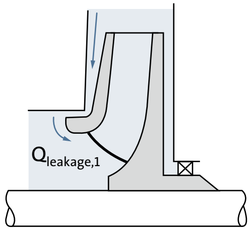
To minimise the leakage flow, it is important to make the gaps as small as possible. When the pressure difference across the gap is large, it is in particular important that the gaps are small.

Model
The leakage can be calculated by combining two different expressions for the difference in head across the gap: The head difference generated by the impeller, equation (5.17) and the head loss for the flow through the gap equation (5.18). Both expressions are necessary to calculate the leak flow.

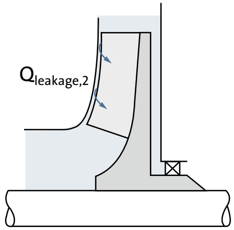
In the following an example of the leakage between impeller eye and pump housing is shown. First the difference in head across the gap generated by the impeller is calculated. The head difference across the gap depends on the static head above the impeller and of the flow behaviour in the cavity between impeller and pump casing:

$$H_{stat,\,gap} = H_{stat,\,impeller} - \omega_{fl}^2 \frac{(D_2^2 - D_{gap}^2)}{8\,g} \quad (5.17)$$

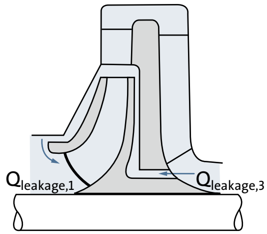
Figure 5.15: Types of leakage



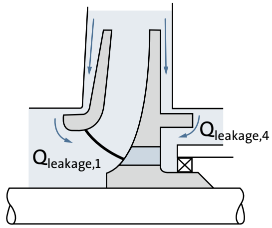
Leakage between impeller eye and pump casing.



Leakage above blades in an open impeller



Leakage between guidevanes and shaft in a multi-stage pump



Leakage as a result of balancing holes

where

ω_{fl} = Rotational velocity of the fluid in the cavity between impeller and pump casing [rad/s]

D_{gap} = Inner diameter of the gap [m]

$H_{stat,\ impeller}$ = Impeller static head rise [m]

The head difference across the gap can also be calculated as the head loss of the flow through the gap, see figure 5.17. The head loss is the sum of the following three types of losses: Loss due to sudden contraction when the fluid runs into the gap, friction loss between fluid and wall, and mixing loss due to sudden expansion of the outlet of the gap.

$$H_{stat,\ gap} = 0.5\ \frac{V^2}{2g} + f\ \frac{L}{s}\frac{V^2}{2g} + 1.0\ \frac{V^2}{2g} \qquad (5.18)$$

where

f = Friction coefficient [-]

L = Gap length [m]

s = Gap width [m]

V = Fluid velocity in gap [m/s]

A_{gap} = Cross-section area of gap [m²]

The friction coefficient can be set to 0.025 or alternatively be found more precisely in a Moody chart, see figure 5.6.

By isolating the velocity V in the equation (5.18) and inserting $H_{stat,gap}$ from equation (5.17), the leakage can be calculated:

$$V = \sqrt{\frac{2gH_{stat,\ gap}}{f\ \frac{L}{s} + 1.5}} \qquad (5.19)$$

$$Q_{leakage} = VA_{gap}$$

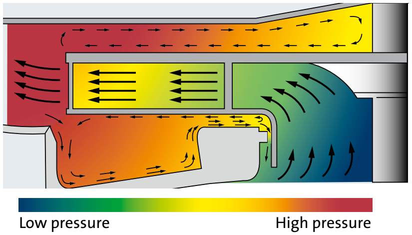
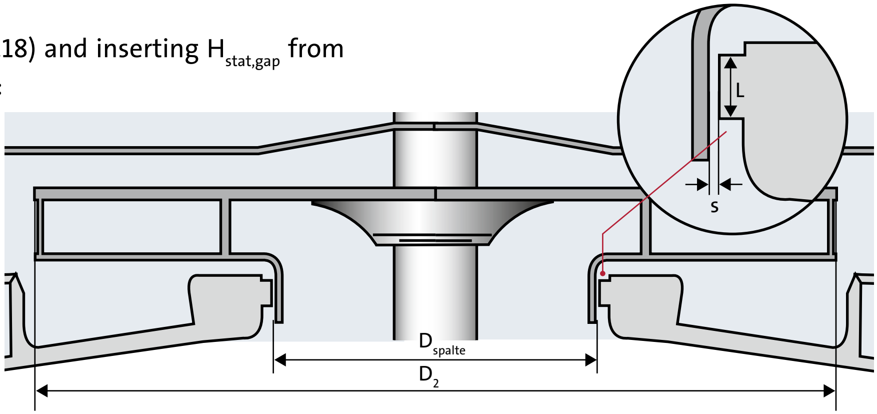


Figure 5.16: The leakage is drived by the pressure difference across the impeller.

Figure 5.17: Pressure difference across the gap through the friction loss consideration.



5.4 Loss distribution as function of specific speed

The ratio between the described mechanical and hydraulic losses depends on the specific speed n_q, which describes the shape of the impeller, see section 4.6. Figure 5.18 shows how the losses are distributed at the design point (Ludwig et al., 2002).

Flow friction and mixing loss are significant for all specific speeds and are the dominant loss type for higher specific speeds (semi-axial and axial impellers). For pumps with low n_q (radial impellers) leakage and disk friction on the hub and shroud of the impeller will in general result in considerable losses.

At off-design operation, incidence and recirculation losses will occur.

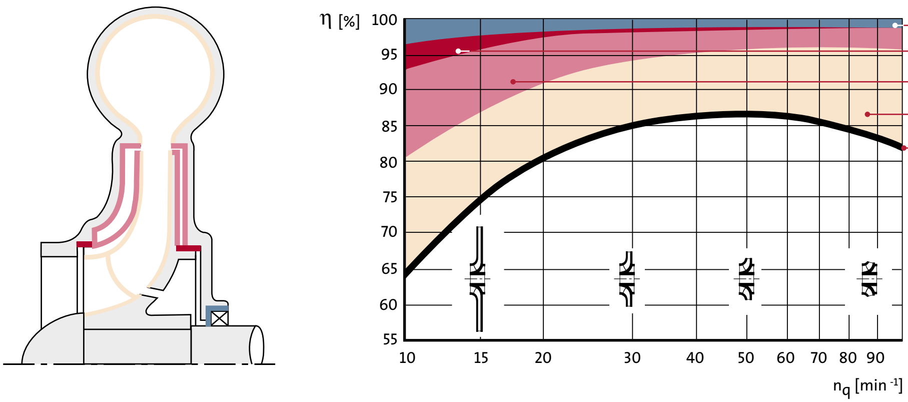


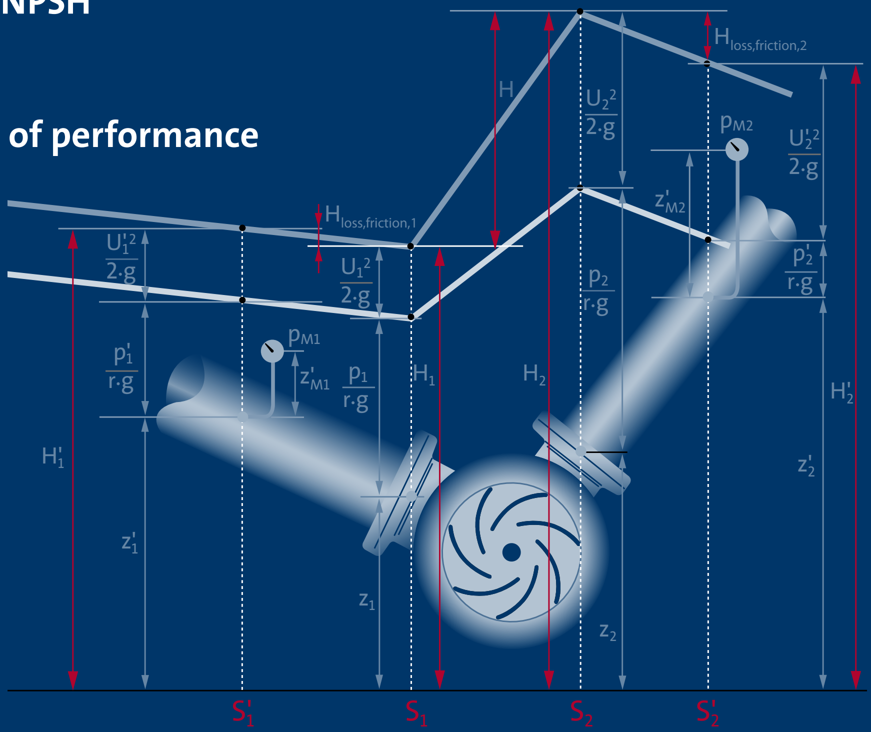
Figure 5.18: Loss distribution in a centrifugal pump as function of specific speed n_q (Ludwig et al., 2002).

5.5 Summary

In this chapter we have described the individual mechanical and hydraulic loss types which can occur in a pump and how these losses affect flow, head and power consumption. For each loss type we have made a simple physical description as well as shown in which hydraulic components the loss typically occurs. Furthermore, we have introduced some simple models which can be used for estimating the magnitude of the losses. At the end of the chapter we have shown how the losses are distributed depending on the specific speeds.

Chapter 6

Pump tests



6. Pump tests

This chapter describes the types of tests Grundfos continuosly performs on pumps and their hydraulic components. The tests are made on prototypes in development projects and for maintenance and final inspection of produced pumps.

6.1 Test types

For characterisation of a pump or one of its hydraulic parts, flow, head, power consumption, NPSH and force impact are measured. When testing a complete pump, i.e. motor and hydraulic parts together, the motor characteristic must be available to be able to compute the performance of the hydraulic part of the pump. For comparison of tests, it is important that the tests are done identically. Even small differences in mounting of the pump in the test bench can result in significant differences in the measured values and there is a risk of drawing wrong conclusions from the test comparison.

Flow, head, power consumption, NPSH and forces are all integral performance parameters. For validation of computer models and failure finding, detailed flow field measurements are needed. Here the velocities and pressures are measured in a number of discrete points inside the pump using e.g. LDA (Laser Doppler Anemometry) and PIV (Particle Image Velocimetry) for velocity, see figure 6.1 and for pressure, pitot tubes and pressures transducers that can measure fast fluctuations.

The following describes how to measure the integral performance parameters, i.e. flow, head, power consumption, NSPH and forces. For characterisation of motors see the Motor compendium (Motor Engineering, R&T). For flow field measurements consult the specialist literature, e.g. (Albrecht, 2002).

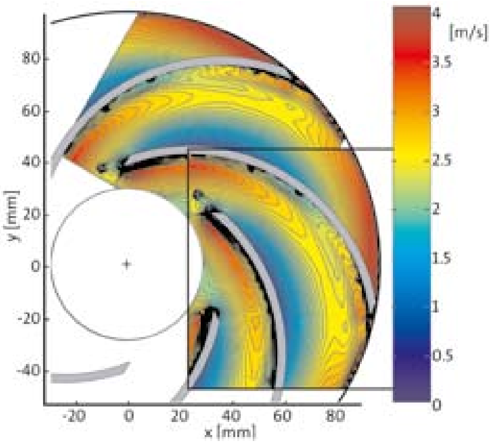


Figure 6.1: Velocity field in impeller measured with PIV.

6.2 Measuring pump performance

Pump performance is usually described by curves of measured head and power consumption versus measured flow, see figure 6.2. From these measured curves, an efficiency curve can be calculated. The measured pump performance is used in development projects for verification of computer models and to show that the pump meets the specification.

During production, the performance curves are measured to be sure they correspond to the catalogue curves within standard tolerances.

Flow, head and power consumption are measured during operation in a test bench that can imitate the system characteristics the pump can be exposed to. By varying the flow resistance in the test bench, a number of corresponding values of flow, differential pressure, power consumption and rotational speed can be measured to create the performance curves. Power consumption can be measured indirectly if a motor characteristic that contains corresponding values for rotational speed, electrical power, and shaft power is available. Pump performance depends on rotational speed and therefore it must be measured.

During development, the test is done in a number of operating points from shut-off, i.e. no flow to maximum flow and in reversal from maximum flow to shut-off. To resolve the performance curves adequately, 10 - 15 operating points are usually enough.

Maintenance tests and final inspection tests are made as in house inspection tests or as certificate tests to provide the customer with documentation of the pump performance. Here 2 - 5 predefined operating points are usally sufficient. The flow is set and the corresponding head, electrical power consumption and possibly rotational speed are measured. The electrical power consumption is measured because the complete product performance is wanted.

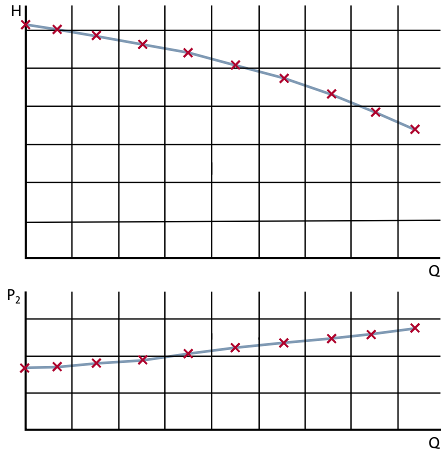


Figure 6.2: Measured head and power curve as function of the flow.

Grundfos builds test benches according to in-house standards where GS241A0540 is the most significant. The test itself is in accordance with the international standard ISO 9906.

6.2.1 Flow

To measure the flow, Grundfos uses magnetic inductive flowmeters. These are integrated in the test bench according to the in-house standard. Other flow measuring techniques based on orifice, vortex meters, and turbine wheels exist.

6.2.2 Pressure

Grundfos states pump performance in head and not pressure since head is independent of the pumped fluid, see section 2.4. Head is calculated from total pressure measured up and down stream of the pump and density of the fluid.

The total pressure is the sum of the static and dynamic pressure. The static pressure is measured with a pressure transducer, and the dynamic pressure is calculated from pipe diameters at the pressure outlets and flow. If the pressure transducers up and down stream of the pump are not located at the same height above ground, the geodetic pressure enters the expression for total pressure.

To achieve a good pressure measurement, the velocity profile must be uniform and non-rotating. The pump, pipe bends and valves affect the flow causing a nonuniform and rotating velocity profile in the pipe. The pressure taps must therefore be placed at a minimum distance to pump, pipe bends and other components in the pipe system, see figure 6.3.

The pressure taps before the pump must be placed two pipe diameters upstream the pump, and at least four pipe diameters downstream pipe bends and valves, see figure 6.3. The pressure tap after the pump must be placed two pipe diameters after the pump, and at least two pipe diameters before any flow disturbances such as bends and valves.

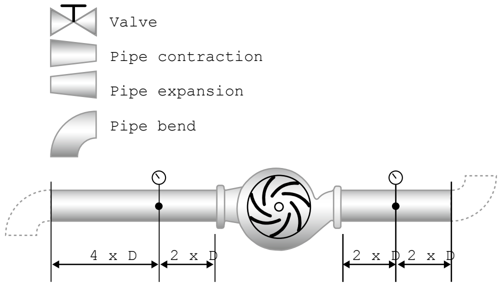


Figure 6.3: Pressure measurement outlet before and after the pump. Pipe diameter, D, is the pipe's internal diameter.

The pressure taps are designed so that the velocity in the pipe affects the static pressure measurement the least possible. To balance a possible bias in the velocity profile, each pressure tab has four measuring holes so that the measured pressure will be an average, see figure 6.4.

The measuring holes are drilled perpendicular in the pipe wall making them perpendicular to the flow. The measuring holes are small and have sharp edges to minimise the creation of vorticies in and around the holes, see figure 6.5.

It is important that the pressure taps and the connection to the pressure transducer are completely vented before the pressure measurement is made. Air in the tube between the pressure tap and transducer causes errors in the pressure measurement.

The pressure transducer measures the pressure at the end of the pressure tube. The measurements are corrected for difference in height Δz between the center of the pressure tap and the transducer to know the pressure at the pressure tap itself, see figure 6.4. Corrections for difference in height are also made between the pressure taps on the pump's inlet and outlet side. If the pump is mounted in a well with free surface, the difference in height between fluid surface and the pressure tap on the pump's outlet side must be corrected, see section 6.2.4.

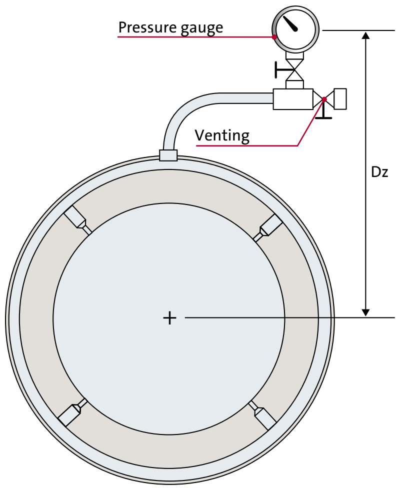


Figure 6.4: Pressure taps which average over four measuring holes.

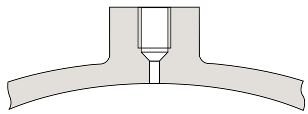
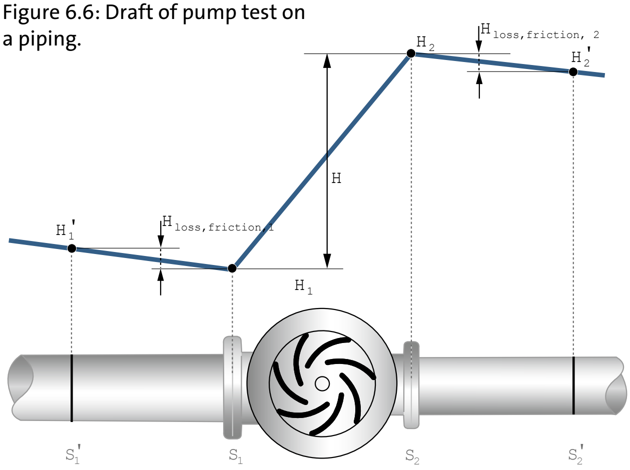


Figure 6.5: Draft of pressure tap.

6.2.3 Temperature

The temperature of the fluid must be known to determine its density. The density is used for conversion between pressure and head and is found by table look up, see the chart "Physical properties of water" at the back of the book.

Figure 6.6: Draft of pump test on a piping.



6.2.4 Calculation of head

The head can be calculated when flow, pressure, fluid type, temperature and geometric sizes such as pipe diameter, distances and heights are known. The total head from flange to flange is defined by the following equation:

$$H = H_2 - H_1 \tag{6.1}$$

Figure 6.6 shows where the measurements are made. The pressure outlets and the matching heads are marked with a ('). The pressure outlets are thus found in the positions S'_1 and S'_2 and the expression for the total head is therefore:

$$H = (H'_2 + H_{loss,\,friction,2}) - (H'_1 - H_{loss,\,friction,1}) \tag{6.2}$$

where $H_{loss,friction,1}$ and $H_{loss,friction,2}$ are the pipe friction losses between pressure outlet and pump flanges.

The size of the friction loss depends on the flow velocity, the pipe diameter, the distance from the pump flange to the pressure outlet and the pipe's surface roughness. Calculation of pipe friction loss is described in section 5.3.1.

If the pipe friction loss between the pressure outlets and the flanges is smaller than 0.5% of the pump head, it is normally not necesarry to take this into consideration in the calculations. See ISO 9906 section 8.2.4 for further explanation.

Figure 6.7: Pump test where the pipes are at an angle compared to horizontal.

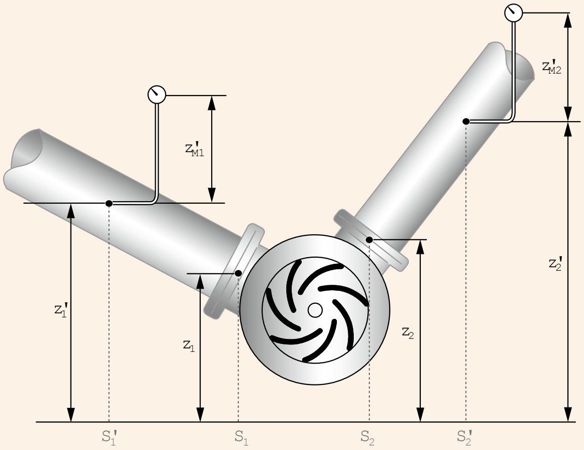
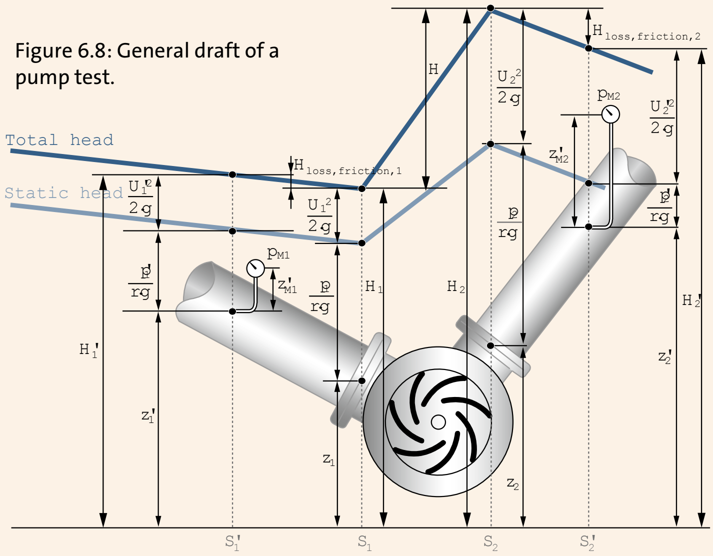


Figure 6.8: General draft of a pump test.



6.2.5 General calculation of head

In practise a pump test is not always made on a horizontal pipe, see figure 6.7. This results in a difference in height between the centers of the pump in- and outlet, z'_1 and z'_2, and the centers of the inlet and outlet flanges, z_1 and z_2 respectively. The manometer can, furthermore, be placed with a difference in height compared to the pipe centre. These differences in height must be taken into consideration in the calculation of head.

Because the manometer only measures the static pressure, the dynamic pressure must also be taken into account. The dynamic pressure depends on the pipe diameter and can be different on each side of the pump.

Figure 6.8 illustrates the basic version of a pump test in a pipe. The total head which is defined by the pressures p_1 and p_2 and the velocities U_1 and U_2 in the inlet and outlet flanges S_1 and S_2 can be calculated by means of the following equation:

$$H = \underbrace{z_2 - z_1}_{\text{Geodetic pressure}} + \underbrace{\frac{p_2 - p_1}{\rho \cdot g}}_{\text{Static pressure}} + \underbrace{\frac{U_2^2 - U_1^2}{2 \cdot g}}_{\text{Dynamic pressure}} \tag{6.3}$$

Using the measured sizes in S'_1 and S'_2, the general expression for the total head is:

$$H = \left[z'_2 + \left(\frac{p'_{M2}}{\rho \cdot g} + z'_{M2} \right) + \frac{U'^2_2}{2 \cdot g} + H_{\text{loss, friction, 2}} \right] - \left[z'_1 + \left(\frac{p'_{M1}}{\rho \cdot g} + z'_{M1} \right) + \frac{U'^2_1}{2 \cdot g} - H_{\text{loss, friction,1}} \right] \tag{6.4}$$

6.2.6 Power consumption

Distinction is made between measurement of the shaft power P_2 and added electric power P_1. The shaft power can best be determined as the product of measured angular velocity ω and the torque on the shaft which is measured by means of a torque measuring device. The shaft power can alternatively be measured on the basis of P_1. However, this implies that the motor characteristic is known. In this case, it is important to be aware that the motor characteristic changes over time because of bearing wear and due to changes in temperature and voltage.

The power consumption depends on the fluid density. The measured power consumption is therefore usually corrected so that it applies to a standard fluid with a density of 1000 kg/m^3 which corresponds to water at 4°C. Head and flow are independent of the density of the pumped fluid.

6.2.7 Rotational speed

The rotational speed is typically measured by using an optic counter or magnetically with a coil around the motor. The rotational speed can alternatively be measured by means of the motor characteristic and measured P_1. This method is, however, more uncertain because it is indirect and because the motor characteristic, as mentioned above, changes over time.

The pump performance is often given for a constant rotational speed. By means of affinity equations, described in section 4.5, the performance can be converted to another speed. The flow, head and power consumption are hereby changed but the efficiency is not changed considerably if the scaling of the speed is smaller than ± 20 %.

6.3 Measurement of the pump's NPSH

The NPSH test measures the lowest absolute pressure at the inlet before cavitation occurs for a given flow and a specific fluid with vapour pressure p_{vapour} , see section 2.10 and formula (2.16).

A typical sign of incipient cavitation is a higher noise level than usual. If the cavitation increases, it affects the pump head and flow which both typically decrease. Increased cavitation can also be seen as a drop in flow at constant head. Erosion damage can occur on the hydraulic part at cavitation.

The following pages introduce the $NPSH_{3\%}$ test which gives information about cavitation's influence on the pump's hydraulic performance. The test gives no information about the pump's noise and erosion damage caused by cavitation.

In practise it is thus not an actual ascertainment of cavitation but a chosen (3%) reduction of the pump's head which is used for determination of $NPSH_R$ - hence the name $NPSH_{3\%}$.

To perform a $NPSH_{3\%}$ test a reference QH curve where the inlet pressure is sufficient enough to avoid cavitation has to be measured first. The 3% curve is drawn on the basis of the reference curve where the head is 3% lower. Grundfos uses two procedures to perform an $NPSH_{3\%}$ test. One is to gradually lower the inlet pressure and keep the flow constant. The other is to gradually increase the flow while the inlet pressure is kept constant.

6.3.1 $NPSH_{3\%}$ test by lowering the inlet pressure

When the $NPSH_{3\%}$ curve is flat, this type of $NPSH_{3\%}$ test is the best suited.

The $NPSH_{3\%}$ test is made by keeping the flow fixed while the inlet pressure $p_{stat,in}$ and thereby $NPSH_A$ is gradually lowered until the head is reduced with more than 3%. The resulting $NPSH_A$ value for the last measuring point before the head drops below the 3% curve then states a value for $NPSH_{3\%}$ at the given flow.

The $NPSH_{3\%}$ curve is made by repeating the measurement for a number of different flows. Figure 6.9 shows the measuring data for an $NPSH_{3\%}$ test where the inlet pressure is gradually lowered and the flow is kept fixed. It is these NPSH values which are stated as the pump's NPSH curve.

Procedure for an $NPSH_{3\%}$ test where the inlet pressure is gradually lowered:

1. A QH test is made and used as reference curve
2. The 3% curve is calculated so that the head is 3% lower than the reference curve.
3. Selection of 5-10 flow points
4. The test stand is set for the seleted flow point starting with the largest flow
5. The valve which regulates the counter-pressure is kept fixed
6. The inlet pressure is gradually lowered and flow, head and inlet pressure are measured
7. The measurements continue until the head drops below the 3% curve
8. Point 4 to 7 is repeated for each flow point

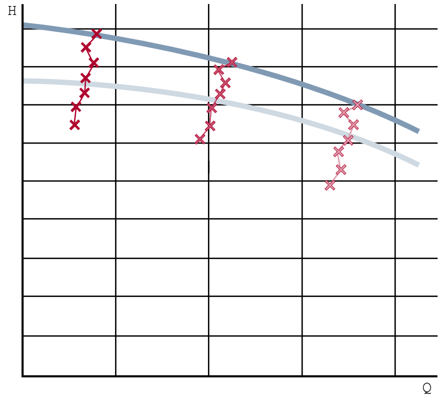
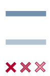


Figure 6.9: $NPSH_A$ measurement by lowering the inlet pressure.



6.3.2 NPSH$_{3\%}$ test by increasing the flow

For NPSH$_{3\%}$ test where the NPSH$_{3\%}$ curve is steep, this procedure is preferable. This type of NPSH$_{3\%}$ test is also well suited for cases where it is difficult to change the inlet pressure e.g. an open test stand.

The NPSH$_{3\%}$ test is made by keeping a constant inlet pressure, constant water level or constant setting of the regulation valve before the pump. Then the flow can be increased from shutoff until the head can be measured below the 3% curve, see figure 6.10. The NPSH$_{3\%}$ curve is made by repeating the measurements for different inlet pressures.

Procedure for NPSH$_{3\%}$ test where the flow is gradually increased

1. A QH test is made and used as reference curve
2. The 3% curve is calculated so that the head is 3% lower than the reference curve.
3. Selection of 5-10 inlet pressures
4. The test stand is set for the wanted inlet pressure
5. The flow is increased from the shutoff and flow, head and inlet pressure are measured
6. The measurements continue until the head is below the 3% curve
7. Point 4-6 is repeated for each flow point

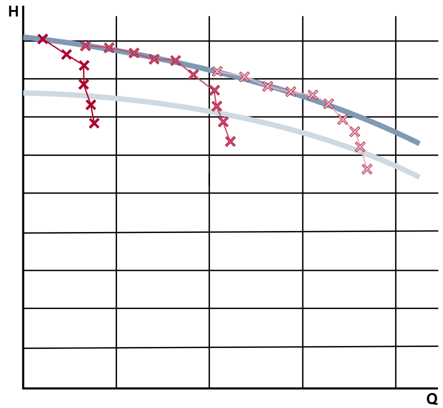


Figure 6.10: NSPH$_A$ measurement by increasing flow.

6.3.3 Test beds

When a closed test bed is used for testing pumps in practise, then the inlet pressure can be regulated by adjusting the system pressure. The system pressure is lowered by pumping water out of the circuit. The system pressure can, furthermore, be reduced with a throttle valve or a vacuum pump, see figure 6.11.

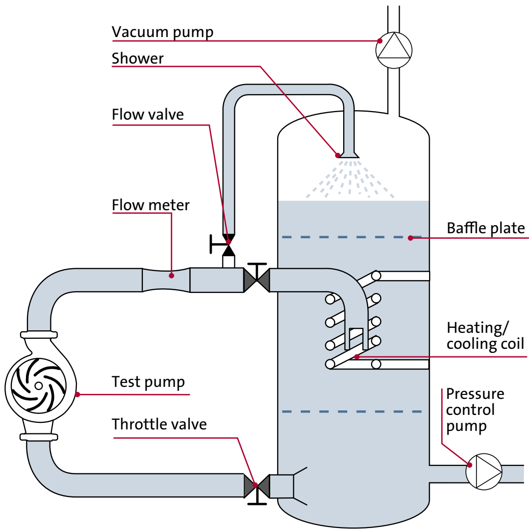


Figure 6.11: Draft of closed test bed for NPSH measurement.

In an open test bed, see figure 6.12, it is possible to adjust the inlet pressure in two ways: Either the water level in the well can be changed, or a valve can be inserted before the pump. The flow can be controlled by changing the pump's counter-pressure by means of a valve mounted after the pump.

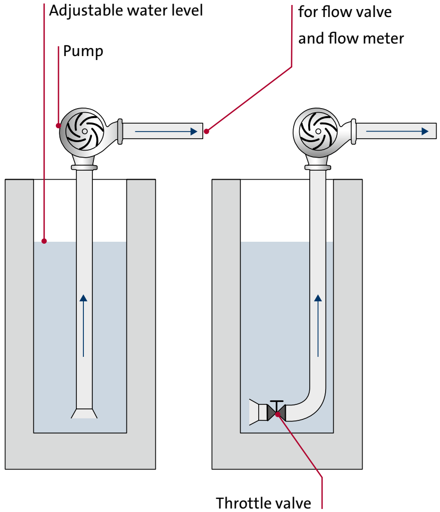


Figure 6.12: Drafts of open test beds for NPSH measurement.

6.3.4 Water quality

If there is dissolved air in the water, this affects the pump performance which can be mistaken for cavitation. Therefore you must make sure that the air content in the water is below an acceptable level before the NPSH test is made. In practise this can be done by extracting air out of the water for several hours. The process is called degasification.

In a closed test bed the water can be degased by lowering the pressure in the tank and shower the water hard down towards a plate, see figure 6.11, forcing the air bubbles out of the fluid. When a certain air volume is gathered in the tank, a part of the air is removed with a vacuum pump and the procedure is repeated at an even lower system pressure.

6.3.5 Vapour pressure and density

The vapour pressure and the density for water depend on the temperature and can be found by table look-up in "Physical properties of water" in the back of the book. The fluid temperature is therefore measured during the execution of an NPSH test.

6.3.6 Reference plane

NPSH is an absolute size which is defined relative to a reference plane. In this case reference is made to the center of the circle on the impeller shroud which goes through the front edge of the blades, see figure 6.13.

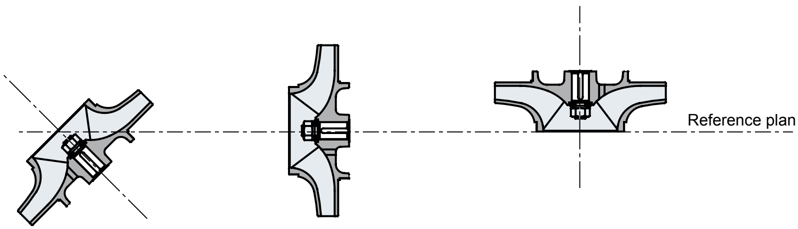


Figure 6.13: Reference planes at NPSH measurement.

6.3.7 Barometric pressure

In practise the inlet pressure is measured as a relative pressure in relation to the surroundings. It is therefore necesarry to know the barometric pressure at the place and time where the test is made.

6.3.8 Calculation of $NPSH_A$ and determination of $NPSH_{3\%}$

$NPSH_A$ can be calculated by means of the following formula:

$$NPSH_A = \frac{p_{stat,in} + p_{bar} + 0.5 \cdot \rho \cdot V_1^2}{\rho \cdot g} + z_{geo} - H_{loss,\,friction,} - \frac{p_{vapour}}{\rho \cdot g} \qquad (6.5)$$

$p_{stat,in}$ = The measured relative inlet pressure
p_{bar} = Barometric pressure
V_1 = Inlet velocity
z_{geo} = The pressure sensor's height above the pump
$H_{loss,friction}$ = The pipe loss between pressure measurement and pump
p_{vapour} = Vapour pressure (table look-up)
ρ = Density (table look-up)

The $NPSH_{3\%}$ value can be found by looking at how the head develops during the test, see figure 6.14. An $NPSH_{3\%}$ value is determined by the $NPSH_A$ value which is calculated from the closest data point above the 3% curve.

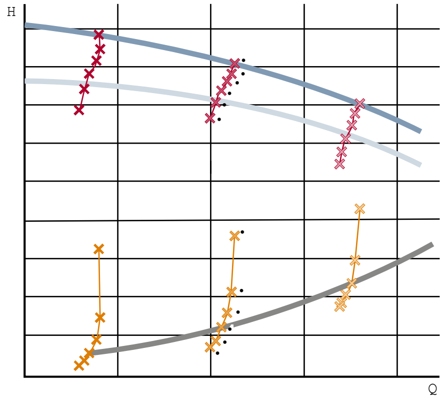
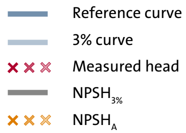


Figure 6.14: Determination of $NPSH_{3\%}$.



6.4 Measurement of force

Measurement of axial and radial forces on the impeller is the only reliable way to get information about the forces' sizes. This is because these forces are very difficult to calculate precisely since this requires a full three-dimensional numerical simulation of the flow.

6.4.1 Measuring system

The force measurement is made by absorbing the forces on the rotating system (impeller and shaft) through a measuring system.

The axial force can e.g. be measured by moving the axial bearing outside the motor and mount it on a dynamometer, see figure 6.15. The axial forces occuring during operation are absorbed in the bearing and can thereby be measured with a dynamometer.

Axial and radial forces can also be measured by mounting the shaft in a magnetic bearing where it is fixed with magnetic forces. The shaft is fixed magnetical both in the axial and radial direction. The mounting force is measured, and the magnetic bearing provides information about both radial and axial forces, see figure 6.16.

Radial and axial force measurements with magnetic bearing are very fast, and both the static and the dynamic forces can therefore be measured.

By measurement in the magnetic bearing, the pump hydraulic is mounted directly on the magnetic bearing. It is important that the fixing flange geometry is a precise reflection in the pump geometry because small changes in the flow conditions in the cavities can cause considerable differences in the forces affecting on the impeller.

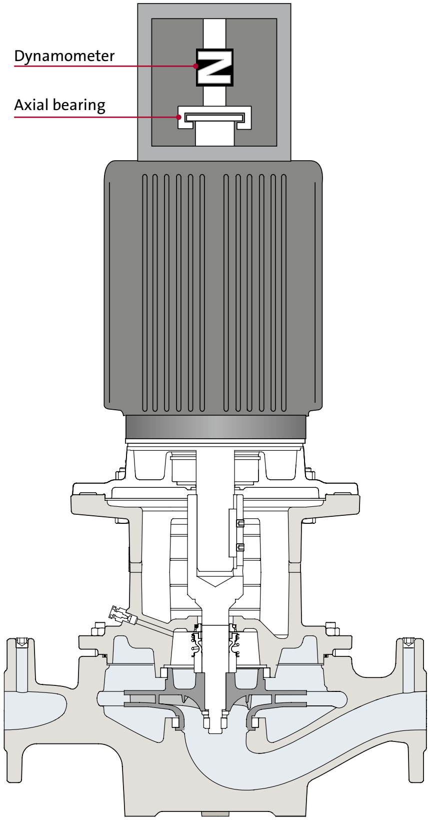


Figure 6.15: Axial force measurement through dynamometer on the bearing.

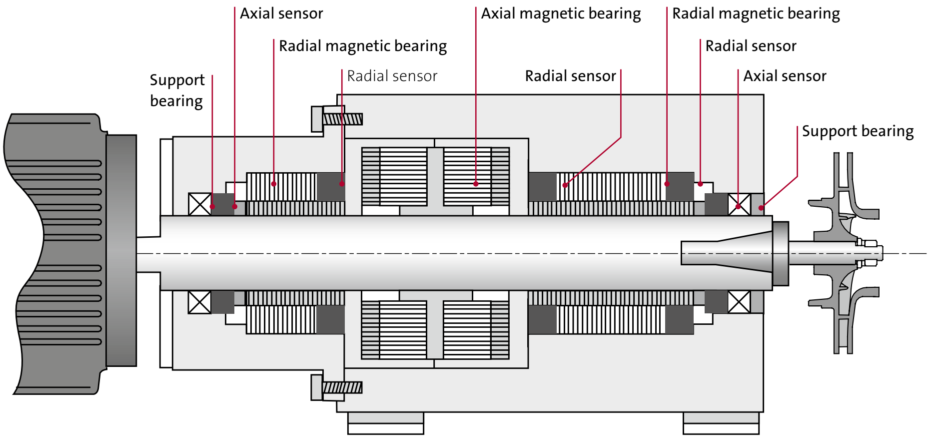


Figure 6.16: Radial and axial force measurement with magnetic bearing.

6.4.2 Execution of force measurement

During force measurement the pump is mounted in a test bed, and the test is made in the exact same way as a QH test. The force measurements are made simultaneously with a QH test.

At the one end, the shaft is affected by the pressure in the pump, and in the other end it is affected by the pressure outside the pump. Therefore the system pressure has influence on the size of the axial force.

If comparison between the different axial force measurement is wanted, it is necesarry to convert the system pressure in the axial force measurements to the same pressure. The force affecting the shaft end is calculated by multiplying the area of the shaft end with the pressure in the pump.

6.5 Uncertainty in measurement of performance

At any measurement there is an uncertainty. When testing a pump in a test bed, the uncertainty is a combination of contributions from the measuring equipment, variations in the test bed and variations in the pump during the test.

6.5.1 Standard demands for uncertainties

Uncertainties on measuring equipment are in practise handled by specifying a set of measuring equipment which meet the demands in the standard for hydrualic performance test, ISO09906.

ISO09906 also states an allowed uncertainty for the complete measuring system. The complete measuring system includes the test beds' pipe circuit, measuring equipment and data collection. The uncertainty for the complete measuring system is larger than the sum of uncertainties on the single measuring instrument because the complete uncertainty also contains variations in the pump during test which are not corrected for.

Variations occuring during test which the measurements can be corrected for are the fluid's characteristic and the pump speed. The correction is to convert the measuring results to a constant fluid temperature and a constant speed.

To ensure a measuring result which is representative for the pump, the test bed takes up more measurements and calculates an average value. ISO09906 has an instruction of how the test makes a representative average value seen from a stability criteria. The stability criteria is a simplified way to work with statistical normal distribution.

6.5.2 Overall uncertainty

The repetition precision on a test bed is in general better than the collected precision. During development where very small differences in performance are interesting, it is therefore a great advantage to make all tests on the same test bed.

There can be significant difference in the measuring results between several test beds. The differences correspond to the overall uncertainty.

6.5.3 Measurement of the test bed's uncertainty

Grundfos has developed a method to estimate a test beds' overall uncertainty. The method gives a value for the standard deviation on the QH curve and a value for the standard deviation on the performance measurement. The method is the same as the one used for geometric measuring instruments, e.g. slide gauge.

The method is outlined in the Grundfos standard GS 241A0540: Test benches and test equipment.

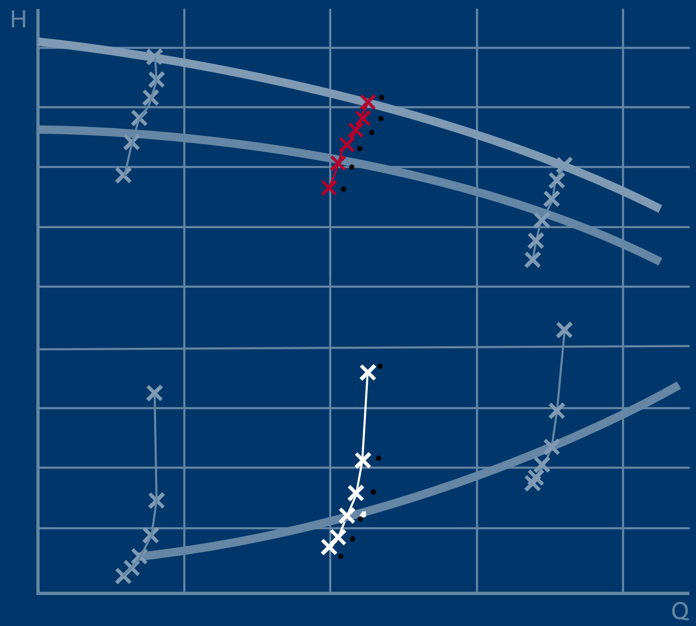
6.6 Summary

In this chapter we have introduced the hydraulic tests carried out on complete pumps and their hydraulic components. We have described which sizes to measure and which problems can occur in connection with planning and execution of a test. Furthermore, we have described data treatment, e.g. head and NPSH value.

Appendix

Appendix A. Units

Appendix B. Check of test results



A. Units

Some of the SI system's units

Basic units

Unit for	Name	Unit
Length	meter	m
Mass	kilogram	kg
Time	second	S
Temperature	Kelvin	K

Additional units

Unit for	Name	Unit	Definition
Angle	radian	rad	One radian is the angle subtended at the centre of a circle by an arc of circumference that is equal in length to the radius of the circle..

Derived units

Unit for	Name	Unit	Definition
Force	Newton	N	$N = kg \cdot m/s^2$
Pressure	Pascal	Pa	$Pa = N/m^2 = kg/(m \cdot s^2)$
Energy, work	Joule	J	$J = N \cdot m = W \cdot s$
Power	Watt	W	$W = J/s = N \cdot m/s = Kg \cdot m^2/s^3$
Impulse			$kg \cdot m/s$
Torque			$N \cdot m$

Conversion of units

Length

m	in (inches)	ft (feet)
1	39.37	3.28
0.0254	1	0.0833

Time

s	min	h (hour)
1	$16.6667 \cdot 10^{-3}$	$0.277778 \cdot 10^{-3}$
60	1	$16.6667 \cdot 10^{-3}$
3600	60	1

Flow, volume flow

m^3/s	m^3/h	l/s	gpm (US)
1	3600	1000	15852
$0.277778 \cdot 10^{-3}$	1	0.277778	4.4
10^{-3}	3.6	1	15.852
0.000063	0.2271	0.063	1

Mass flow

kg/s	kg/h
1	3600
$0.277778 \cdot 10^{-3}$	1

Speed

kg/s	kg/h	ft/s
1	3600	3.28
$0.277778 \cdot 10^{-3}$	1	0.9119
0.3048	1.097	1

Rotational speed

RPM = revolution per minute	s^{-1}	rad/s
1	$16.67 \cdot 10^{-3}$	0.105
60	1	6.28
9.55	0.1592	1

Pressure

kPa	bar	mVs
1	0.01	0.102
100	1	10.197
9.807	$98.07 \cdot 10^{-3}$	1

Temperature

K	°C
1	t(°C) = T - 273.15K
T(Kelvin) = 273.15°C + t	1

Work, energy

J	kWh
1	$0.277778 \cdot 10^{-6}$
$3.6 \cdot 10^{6}$	1

Kinematic viscosity

m^2/s	cSt
1	10^{6}
10^{-6}	1

Dynamic viscosity

Pa · s	cP
1	10^{3}
10^{-3}	1

B. Check of test results

When unexpected test results occur, it can be difficult to find out why. Is the tested pump in reality not the one we thought? Is the test bed not measuring correctly? Is the test which we compare with not reliable? Have some units been swaped during the data treatment?

Typical examples which deviate from what is expected is presented on the following pages. Furthermore, some recommendations of where it is appropriate to start looking for reasons for the deviating test results are presented.

The test shows that the efficiency is below the catalogue curve.

Possible cause	What to examine	How to find the error
Power consumption is too high and/or the head is too low	Decide whether it is the power consumption or the head which deviates	Use one of the three schemes below, scheme 1 -3

Table 1: The test shows that the power consumption for a produced pump lies above the catalogue value but the head is the same as the catalogue curve.

Possible cause	What to examine	How to find the error
The catalogue curve does not reflect the 0-series testen.	Compare 0-series test with catalog curve	If the catalogue curve and 0-series test do not correspond, it can not be expected that the pump performs according to the catalogue curve.
The impeller diameter or outlet width is bigger than on the 0-series	Make a scaling of the test where the impeller diameter D2 is reduced until the power matches most of the curve. If the head also matches the curve, then the diameter on the produced pump is probably too big. Repeat the same procedure with the impeller outlet width b2. Scaling of D2 and b2 is discussed in chapter 4.5	Make sure that the right impeller is tested. Measure the impeller's outlet on the 0-series pump. Adjust impeller diameter and outlet width in the production
Mechanical drag is found	Listen to the pump. If it is noisy, turn off the pump and rotate by hand to identify any friction. Look at the difference of the two power curves. Is it constant, there is probably drag.	Remove the mechanical drag
The motor efficiency is lower than specified.	Separate motor and pump. Test them separately. The pump can be in a test bench with torque meter or with a calibrated motor.	If the pump's power consumption is ok, the motor is the problem. Find cause for motor error.

Table 2: The test shows that the power consumption and head lies below the catalogue curve.

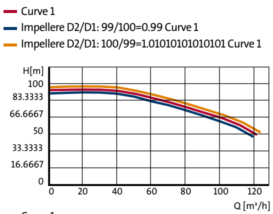
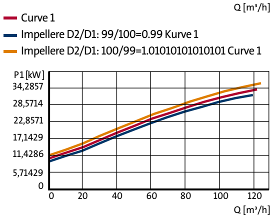
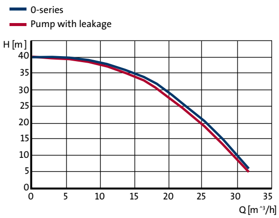
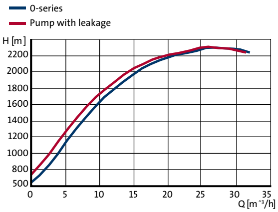
Possible cause	What to examine	How to find the error
Curves have been made at different speeds.	Find the speed for the catalogue curve and the test.	Convert to the same speed and compare again.
The catalogue curve does not reflect the 0-series test.	Compare the 0-series test with the catalogue curve.	If the catalogue curve and 0-series test do not correspond, it can not be expected that the pump performs according to catalogue curve.
The impeller's outlet diameter or outlet width is smaller than on the 0-series test.	Make a scaling of the test where the impeller diameter D_2 is increased until the power matches over most of the curve. If the head also matches over most of the curve, then the diameter on the produced pump is probably too small. Repeat the same procedure with the impeller's outlet width b_2. Scaling of D_2 and b_2 is discussed in chapter 4.5  	Measure the impeller outlet on the 0-series pump. Adjust impeller diameter and outlet width in the production.

Table 3: The power consumption is as the catalogue curve but the head is too low.

Possible cause	What to examine	How to find the error
The catalogue curve does not reflect the 0-series test.	Compare O-series test with catalogue curve.	If the catalogue curve and 0-series test do not correspond, it can not be expected that the pump performs according to the catalogue curve.
Increased hydraulic friction	Compare the QH curves at the same speed. Is the difference developing as a parabola with the flow, there could be an increased friction loss. Examine surface roughness and inlet conditions.	Remove irregularities in the surface. Reduce surface roughness. Remove elements which block the inlet.
Calculation of the head is not done correctly.	Examine the information about pipe diameter and the location of the pressure transducers. Examine whether the correct density has been used for calculation of the head.	Repeat the calculation of the head.
Error in the differential pressure measurement.	Read the test bed's calibration report. Examine whether the pressure outlets and the connections to the pressure transducers have been bleed. Examine that the pressure transducers can measure in the pressure range in question.	If it has been more than a year since the pump has been calibrated, it must be calibrated now. Use the right pressure transducers.
Cavitation	Examine whether there is sufficient pressure at the pump's inlet. See section 2.10 and 6.3)	Increase the system pressure.

Table 3 (continued)

Possible cause	What to examine	How to find the error
Increased leak loss.	Compare QH curves and power curves. If the curve is a horizontal displacement which decreases when the head (the pressure difference above the gap) falls, there could be an increased leak l oss. Leak loss is described in section 5.3.7. Measure the sealing diameter on the rotating and fixed part. Compare the results with the specifications on the drawing. Examine the pump for other types of leak loss.  	Replace the impeller seal. Close all unwanted circuits.

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ISO 9906 Rotodynamic pumps – Hydraulic performance acceptance test-Grades 1 and 2. The standard deals with hydraulic tests and contains instructions of data treatment and making of test equipment.

ISO2548 has been replaced by ISO9906

ISO3555 has been replaced by ISO9906

ISO 5198 Pumps – Centrifugal-, mixed flow – and axial pumps – Hydraulic function test – Precision class

GS 241A0540 Test benches and test equipment. Grundfos standards for contruction and rebuilding of test benches and data loggers.

List of Symbols

Symbol	Definition	Unit
FLOW		
Q	Flow, volume flow	$[m^3/s]$
Q_{design}	Design flow	$[m^3/s]$
$Q_{impeller}$	Flow through the impeller	$[m^3/s]$
Q_{leak}	Leak flow	$[m^3/s]$
$\dot{m}$	Mass flow	[kg/s]
HEAD		
H	Head	[m]
$H_{loss,\{loss\ type\}}$	Head loss in {loss type}	[m]
NPSH	Net Positive Suction Head	[m]
$NPSH_A$	NPSH Available (Net Positive Suction Head available in system)	[m]
$NPSH_R$, $NPSH_{3\%}$	NPSH Required (The pump's net positive suction head system demands)	[m]
GEOMETRIC DIMENSIONS		
A	Cross-section area	$[m^2]$
b	Blade height	[m]
β	Blade angle	[°]
β'	Flow angle	[°]
s	Gap width	[m]
D, d	Diameter	[m]
D_h	Hydraulic diameter	[m]
k	Roughness	[m]
L	Length (gap length, length of pipe)	[m]
O	Perimeter	[m]
r	Radius	[m]
z	Height	[m]
Δz	Difference in height	[m]
PRESSURE		
p	Pressure	[Pa]
Δp	Differential pressure	[Pa]
p_{steam}	The fluid vapour pressure	[Pa]
p_{bar}	Barometric pressure	[Pa]
p_{beho}	Positive or negative pressure compared to p_{bar} if the fluid is in a closed container.	[Pa]
$P_{loss,\{loss\ type\}}$	Pressure loss in {loss type}	[Pa]
EFFICIENCIES		
η_{hyd}	Hydraulic efficiency	[-]
$\eta_{control}$	Control efficiency	[-]
η_{motor}	Motor efficency	[-]
η_{tot}	Total efficiency for control, motor and hydraulics	[-]

Symbol	Definition	Unit
POWER		
P	Power	[W]
P_1	Power added from the electricity supply network	[W]
P_2	Power added from motor	[W]
P_{hyd}	Hydraulic power transferred to the fluid	[W]
$P_{loss,\{loss\ type\}}$	Power loss in {loss type}	[W]
SPEED		
ω	Angular frequency	[1/s]
f	Frequency	[Hz]
n	Speed	[1/min]
VELOCITIES		
V	The fluid velocity	[m/s]
U	The impeller tangential velocity	[m/s]
C	The fluid absolute velocity	[m/s]
W	The fluid relative velocity	[m/s]
SPECIFIC NUMBERS		
Re	Reynold's number	[-]
n_q	Specific speed	
FLUID CHARACTERISTICS		
ρ	The fluid density	$[kg/m^3]$
ν	Kinematic viscosity of the fluid	$[m^2/s]$
MISCELLANEOUS		
f	Coefficient of friction	[-]
g	Gravitational acceleration	$[m/s^2]$
ζ	Dimensionless pressure loss coefficient	[-]

General indices

Index	Definition	Examples
1, in	At inlet, into the component	A_1, C_{in}
2, out	At outlet, out of the component	A_2, C_{out}
m	Meridional direction	C_m
r	Radial direction	W_r
U	Tangential direction	C_{1U}
a	Axial direction	C_a
stat	Static	p_{stat}
dyn	Dynamic	p_{dyn}, $H_{dyn,in}$
geo	Geodetic	p_{geo}
tot	Total	p_{tot}
abs	Absolute	$p_{stat,abs}$, $p_{tot,abs,in}$
rel	Relative	$p_{stat,rel}$
Operation	Operation point	$Q_{operation}$

Physical properties for water

T [°C]	p_{vapour} [10^5 Pa]	ρ [kg/m^3]	ν [10^{-6} m^2/s]
0	0.00611	1000.0	1.792
4	0.00813	1000.0	1.568
10	0.01227	999.7	1.307
20	0.02337	998.2	1.004
25	0.03166	997.1	0.893
30	0.04241	995.7	0.801
40	0.07375	992.3	0.658
50	0.12335	988.1	0.554
60	0.19920	983.2	0.475
70	0.31162	977.8	0.413
80	0.47360	971.7	0.365
90	0.70109	965.2	0.326
100	1.01325	958.2	0.294
110	1.43266	950.8	0.268
120	1.98543	943.0	0.246
130	2.70132	934.7	0.228
140	3.61379	926.0	0.212
150	4.75997	916.9	0.199
160	6.18065	907.4	0.188

Affinity rules

$$\left.\begin{aligned} Q_B &= Q_A \cdot \left(\frac{n_B}{n_A}\right) \\ H_B &= H_A \cdot \left(\frac{n_B}{n_A}\right)^2 \\ P_B &= P_A \cdot \left(\frac{n_B}{n_A}\right)^3 \end{aligned}\right\} \text{Scaling of rotational speed}$$

$$\left.\begin{aligned} Q_B &= Q_A \cdot \left(\frac{D_B^2 \cdot b_B}{D_A^2 \cdot b_A}\right) \\ H_B &= H_A \cdot \left(\frac{D_B}{D_A}\right)^2 \\ P_B &= P_A \cdot \left(\frac{D_B^4 \cdot b_B}{D_A^4 \cdot b_A}\right) \end{aligned}\right\} \text{Geometric scaling}$$

Pictograms


Pump


Valve


Stop valve


Pressure gauge


Heat exchanger

BE > THINK > INNOVATE >

Being responsible is our foundation
Thinking ahead makes it possible
Innovation is the essence

GRUNDFOS®