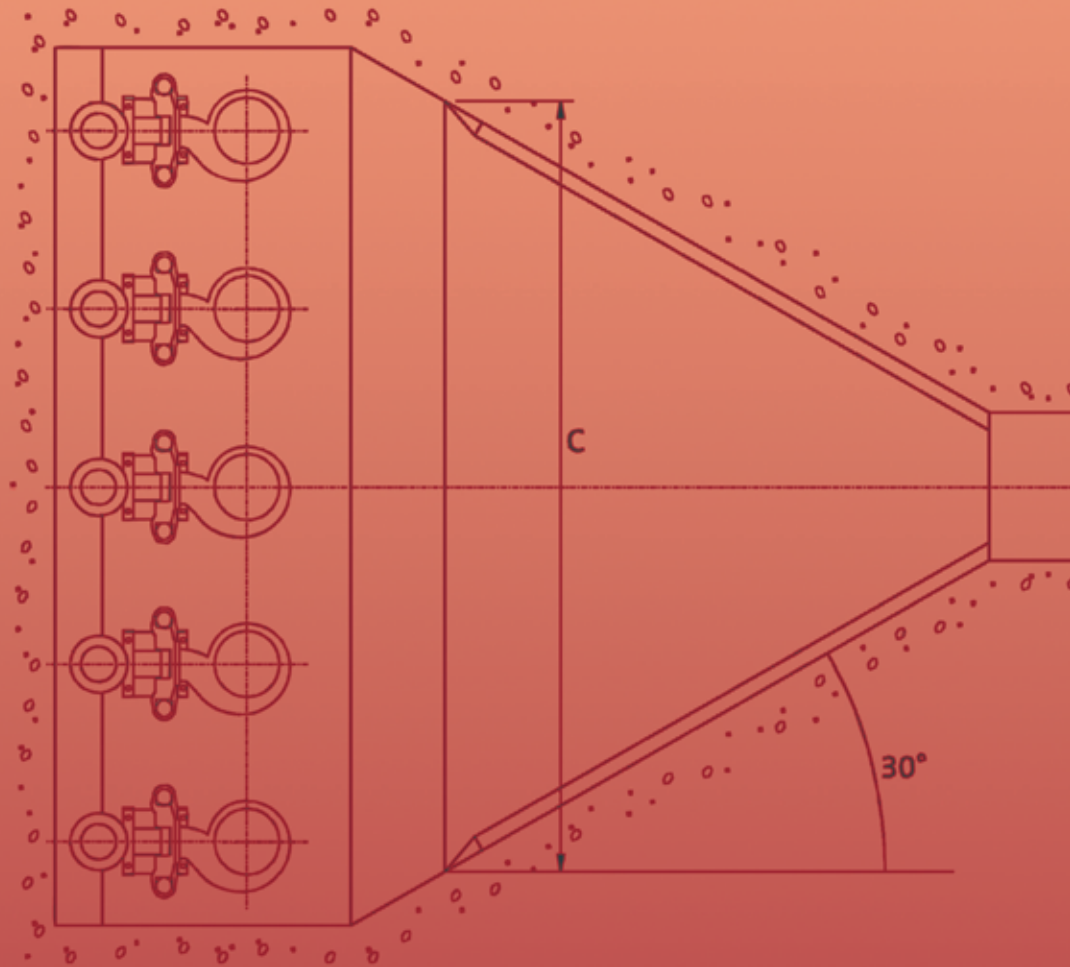


GRUNDFOS WASTEWATER



## THE SEWAGE PUMPING HANDBOOK

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GRUNDFOS 

# **THE SEWAGE PUMPING HANDBOOK**



## Foreword

The use of submersible pumps in sewage and drainage pumping applications has increased greatly in the last decades since they entered the market. The introduction of heavy-duty submersible pumps with motor power ratings exceeding 500 kW has also made them available for central municipal pumping duties. The good service record and high quality standard attained by these pumps has all but excluded the use of conventional pumps in municipal service.

By the same token, the special characteristics of submersible pumps have also required the development of new knowledge on their implementation, such as the design of pumping stations. This work has been advanced by both pump manufacturers and municipal engineers.

The intention of this book is to bring the newest information on both submersible pumps and pumping stations to the use of all concerned professional people in a concise form. The book is divided into Sections according to the related topics.

Basic pump theory is described in Section 1, providing a reference background for the assessment of pump performance. Submersible pump design

and construction is described in Section 2. Pump performance is dealt with in Section 3, offering methods for the calculation of pump performance in various installations. Factors affecting pump selection are also discussed. Section 4 offers information on pump testing. Basic design of pumping stations is discussed in Section 5, offering design information for both large and small applications. Continuous regulation of submersible pump operation by frequency control is described in Section 6. The concept of whole-life cost for pumps and pumping installations is presented in Section 7. Matters relating to pump commissioning are presented in Section 8, whereas pump operation and servicing is described in Section 9. Section 10 deals with pumping station control and monitoring. Appendix A offers information on the hydraulic characteristics of common pipe components for pipeline loss calculations. Appendix B presents a method for the determination of sewage pumping station capacity and pump starting frequency.

One objective of the book has been to make the contents easy to read and comprehend. The presentation is therefore enhanced with a large number of illustrations, providing examples of and complementary information on the matter at hand.





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# 1 Pump Theory

This section is a primer of fluid pumping theory and provides the reader with the theoretical background knowledge essential for deeper understanding of the pumping process.

## 1.1 The Head Equation

Figure 1 shows part of continuous fluid flow in a duct. Between the two observation sections 1 and 2 no energy is transferred to or from the fluid and the flow is assumed to be frictionless. Thus the total energy of the fluid relative to a horizontal reference plane T at the two sections must be equal. The total energy comprises components for potential energy, pressure energy and kinetic energy, and for a fluid particle with a mass  $m$  the energy at the observation sections is as follows:

Section	1	2
Potential Energy	$mgh_1$	$mgh_2$
Pressure Energy	$mg \frac{p_1}{\rho g}$	$mg \frac{p_2}{\rho g}$
Kinetic Energy	$\frac{1}{2}mv_1^2$	$\frac{1}{2}mv_2^2$

where  $\rho$  is the fluid density and  $g$  the acceleration of gravity.

For a flow without losses the total energy in section 1 and 2 will be equal, thus

$$mgh_1 + mg \frac{p_1}{\rho g} + \frac{1}{2}mv_1^2 = mgh_2 + mg \frac{p_2}{\rho g} + \frac{1}{2}mv_2^2$$

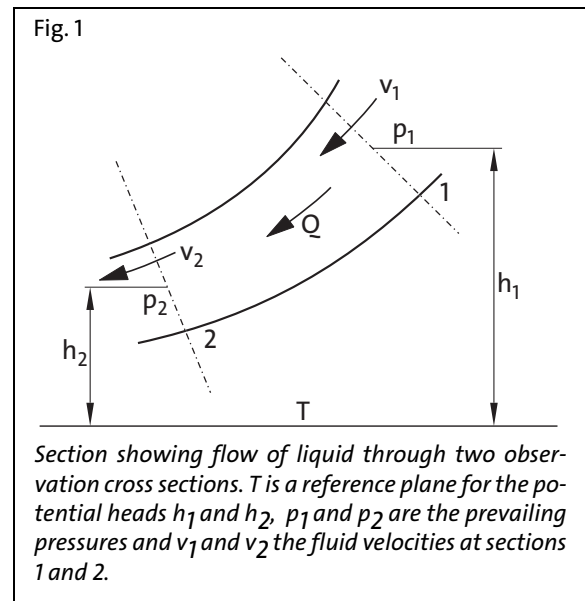
Dividing both sides of the equation with the term  $mg$  it is obtained

$$h_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} = h_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} \quad (1)$$

This equation is called Bernoulli's equation after the engineer who first derived it. The terms of the equation are expressed as heads, and the terms

are consequently called static head, pressure head and kinetic head, respectively.

The equation is essential for fluid mechanics and can be used to account for many hydrodynamic phenomena, such as the decrease in pressure that accompanies a reduction in a flow cross section area. In this case the fluid velocity increases, and for the total head to remain constant and assuming the potential head remains unchanged, the pressure term or static head, must decrease.



### 1.1.1 Flow with Losses or Addition of Energy

If there are losses in the flow between section 1 and section 2 in Figure 1, the head equation 1 can be written

$$h_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} = h_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + H_r \quad (2)$$

where  $H_r$  is the head loss.

If energy is added to the flow by placing a pump between section 1 and section 2 in Figure 1, the equation 2 can be written

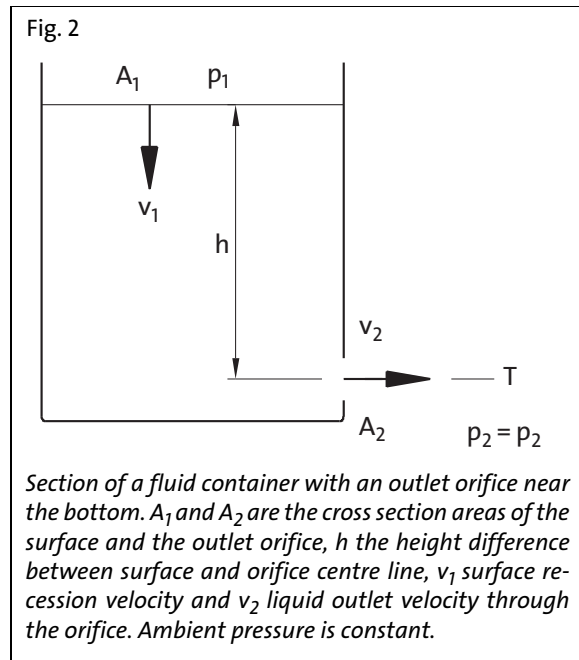
$$h_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} + H = h_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + H_r \quad (3)$$

where  $H$  is the pump total head.

## 1.1.2 Fluid Flowing from a Container

An example of the application of the Bernoulli equation is the calculation of the flow rate of a fluid flowing freely from an open container.

Figure 2 shows an open container with an outlet orifice near the bottom. For practical purposes the area  $A_1$  is assumed much larger than the orifice area  $A_2$ , and the atmospheric pressure  $p_1$  in the container is equal to that outside the orifice,  $p_2$ .



Choosing the centre line of the orifice as the reference plane T, the term  $h_2$  is equal to zero and  $h_1$  equal to  $h$ . Because  $A_1$  is much larger than  $A_2$ , the kinetic head  $\frac{v_1^2}{2g}$  can be assumed as zero. Thus the head equation 1 can be written

$$h = \frac{v_2^2}{2g} \quad (4)$$

whence

$$v_2 = \sqrt{2gh} \quad (5)$$

For volume flow without losses is obtained

$$q_1 = A_2 \sqrt{2gh} \quad (6)$$

To accommodate for losses present, a flow coefficient  $\mu$  is added to equation 6, whence

$$q_1 = \mu A_2 \sqrt{2gh} \quad (7)$$

The flow coefficient  $\mu$  is dependent on the shape of the orifice, and can be obtained from text books on the subject. If the fluid level in the container is allowed to recede, the level height  $h$  will change, which will have to be accommodated for in calculations.

## 1.2 The Basic Pump Equation

The basic pump equation is used to calculate and design geometrical shapes and dimensions of centrifugal pumps. The basic pump equation is also used to deduce the pump Q/H curve.

A pump impeller vane and its associated velocity vectors are shown in Figure 3.

- $v$  = absolute fluid velocity
- $w$  = velocity relative to the vane
- $u$  = perimeter velocity
- $v_u$  = tangential component of absolute velocity
- $v_m$  = radial component of absolute velocity

The relative velocity is parallel to the vane at any given point.

$$\text{Also } v_{u1} = v_1 \cdot \cos \alpha_1 \text{ and } v_{u2} = v_2 \cdot \cos \alpha_2$$

Assuming the flow to be without losses and the number of vanes infinite ( $\infty$ ), the familiar basic equation of pump theory can be derived using the laws of mechanics. This relationship is known as the Euler equation and is expressed as:

$$H_{t\infty} = \frac{1}{g}(u_2 v_{u2} - u_1 v_{u1}) \quad (8)$$

where the index t refers to a flow without losses and  $\infty$  refers to the assumption of infinite number of vanes ensuring complete fluid direction.

In an actual pump neither of these assumptions can be satisfied, as friction losses are always present, and the finite number of vanes will not direct the flow entirely in the direction of the vane.

The reduction in head caused by losses in the flow is taken into account by the hydraulic efficiency  $\eta_h$ , and the reduction due to the deviation of the flow from ideal angle  $\beta_2$  is accounted for by a vane coefficient  $k$ . With these modifications, the Euler equation for an actual pump reads as follows:

$$H = \frac{\eta_h}{g}(ku_2v_{u2} - u_1v_{u1}) \quad (9)$$

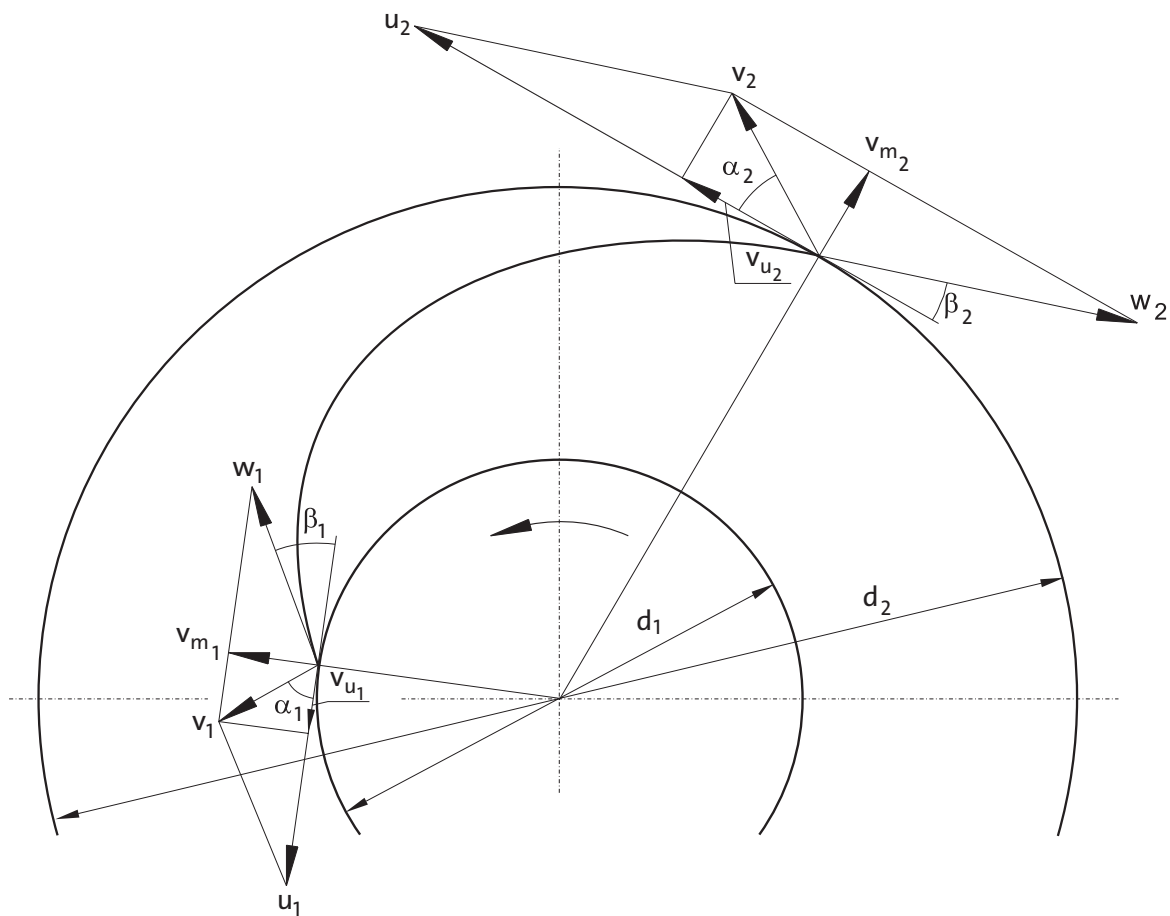
It can be shown that both  $\eta_h$  and  $k$  are less than unity. They will not be discussed in further detail here.

Centrifugal pumps are normally designed with  $\alpha_1 = 90^\circ$ , whence  $v_{u1} = 0$ .

Thus the basic pump equation is simplified to

$$H = k\eta_h \frac{u_2v_{u2}}{g} \quad (10)$$

Fig. 3



*Pump impeller vane with the velocity triangles at leading and trailing edges. Fluid absolute velocity  $v$ , relative velocity  $w$ , vane perimeter velocity  $u$ , liquid absolute velocity tangential component  $v_u$  and radial component  $v_m$ .*

## 1.3 Pump Curve and Losses

The ideal head obtained from the Euler equation is independent of the volume rate of flow  $Q$ . If the  $Q/H_{t\infty}$  curve is plotted,  $H_{t\infty}$  is indicated by a straight line. The real  $Q/H$  curve is derived from this by subtracting the effects of the finite number of vanes and various other losses that occur within the pump. Please refer to Figure 4.

### 1.3.1 The Effect of Finite Number of Vanes

As noted earlier, a finite number of vanes decreases the head by the vane factor  $k$ . Taking this into account, the theoretical head  $H_t$  is obtained. It can be written:

$$H_t = kH_{t\infty} \quad (11)$$

$H_t$  is not perfectly linear because the vane coefficient is slightly dependent on the volume rate of flow  $Q$ . The head reduction from  $H_{t\infty}$  to  $H_t$  is not caused by flow losses, but by deviation of the fluid from the ideal flow angles due to the finite number of vanes.

### 1.3.2 Friction Losses $H_f$

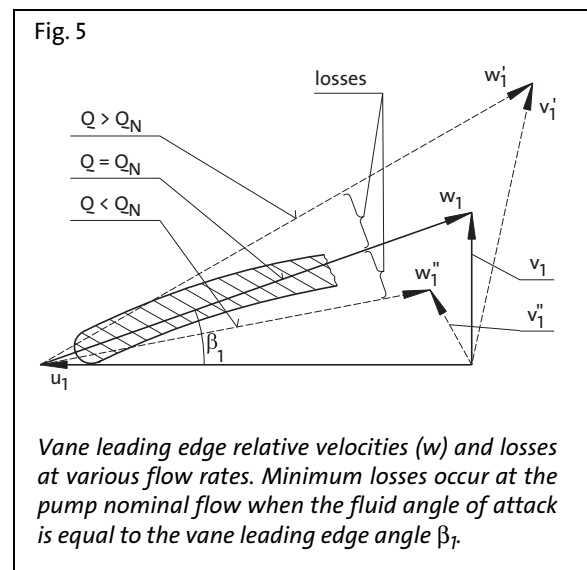
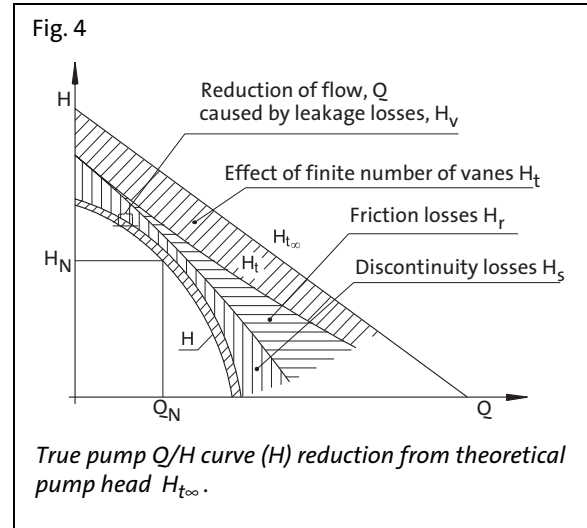
Friction losses occur as the fluid flows through the passages of the impeller and the pump casing. They increase approximately with the square of the flow rate  $Q$ .

### 1.3.3 Discontinuity Losses $H_s$

Discontinuity losses are generated in the following areas:

- At the vane leading edge where the fluid hits the vane tip. The loss is smallest at the pump design point, where the fluid contacts the vane at the vane angle  $\beta_1$ . The losses increase with increasing deviation of the contact angle from the vane angle  $\beta_1$ , see Figure 5.
- At the vane trailing edge losses occur due to eddies shed by the vane. These increase approximately with the square of the flow rate.
- In the pump casing at flow rates other than the design value, when the flow velocity at the casing differs from that at the impeller perimeter. The effect is shown in Figure 6. The velocity differences create turbulence leading to losses, growing with increasing difference of actual flow from design flow.

The effects of the discontinuity losses are shown in Figure 4.



### 1.3.4 Leakage Losses $H_v$

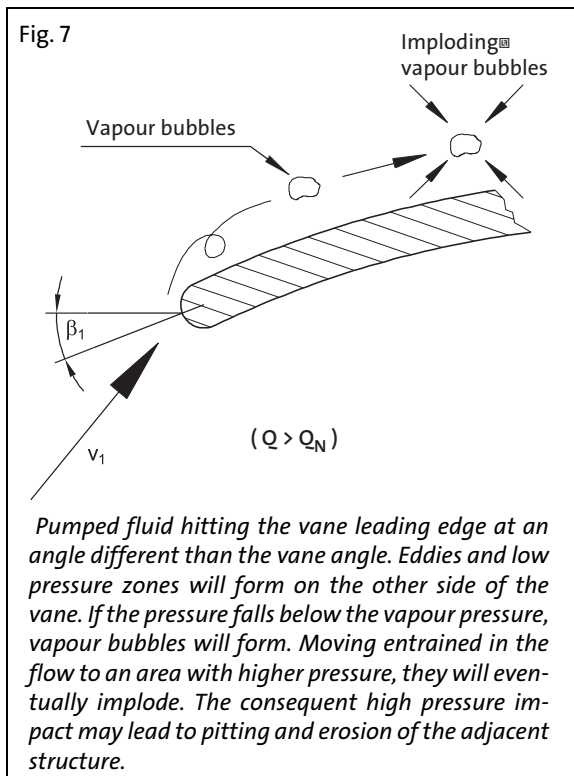
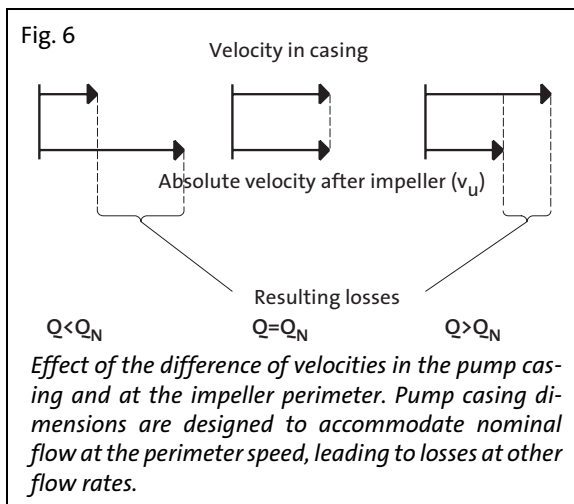
Leakage losses occur at the clearance between impeller and pump casing. Even if the clearance is kept as small as possible, a small backflow passes from the high pressure area at the impeller rim to the low pressure area of the impeller eye. Thus the flow through the impeller is slightly larger than the flow out of the pump casing, and the pump head is met with a reduced flow, the difference being the leakage loss  $H_v$ . The effect of the leakage loss is shown in Figure 4. As the pump wears out, this loss will increase.

### 1.3.5 Other Losses

There are further losses in a centrifugal pump, not affecting the Q/H curve, but that will increase the motor shaft power requirement. These include:

- friction losses at the impeller outside surfaces
- shaft seal friction losses
- bearing friction losses

For submersible pumps, the last two items are included in the motor losses.



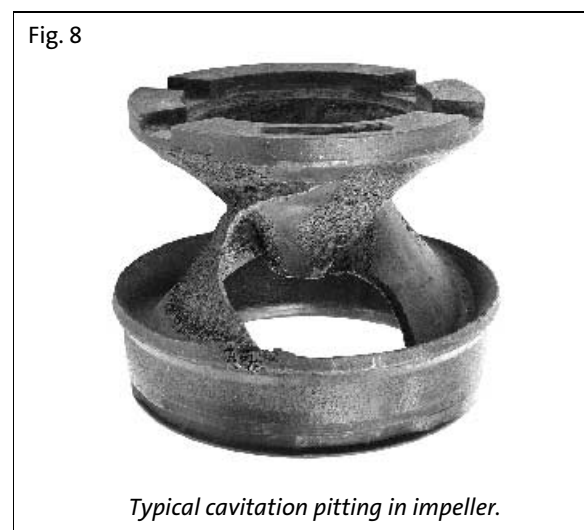
## 1.4. Cavitation and NPSH

Cavitation is caused by the formation and collapse of vapour bubbles in a liquid. Vapour bubbles form when the local static pressure in a flowing liquid decreases to or below the liquid vapour pressure at ambient temperature. When the bubble, or void, moves with the flow to an area with a higher pressure, it will rapidly collapse. The implosion causes a transitory, extremely high local shock wave in the fluid. If the implosion takes place near a surface, the pressure shock will, if occurring repeatedly, eventually erode the surface material.

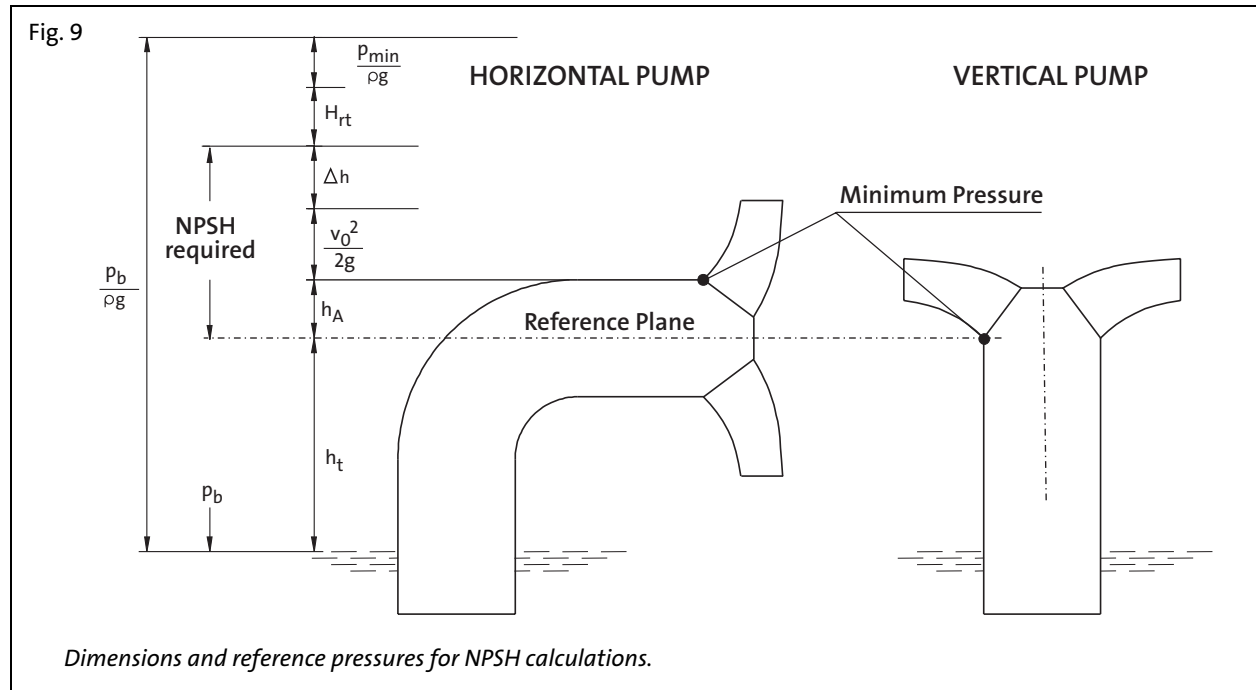
The cavitation phenomenon will typically occur in centrifugal pumps at a location close to the impeller vane leading edge, see Figure 7. Cavitation may also lower the pump Q/H curve and efficiency. A cavitating pump emits a typical rattling noise, like sand being pumped through the pump. No pump material will completely withstand cavitation, so care should be exercised if the pump operating conditions present a risk of cavitation.

Wear marks from cavitation typically occur locally and consist of deep pittings with sharp edges. The pittings can be several millimetres deep, please refer to Figure 8.

Normally pump curves published for submersible pumps are drawn so that a pump in normal submerged installation will not cavitate as long as the duty point is on the allowed section of the Q/H curve.







If the submersible pump is installed dry with a suction pipe, the installation must be checked for cavitation. In these cases the concept of NPSH is used.

## 1.4.1 Definition of NPSH

NPSH is the acronym for Net Positive Suction Head. The following pressure heads are used for the calculation of NPSH:

- $h_t$  = inlet geodetic head
- $h_A$  = height difference between reference plane and tip of vane leading edge.
- $H_{rt}$  = flow losses in inlet pipe
- $\frac{v_0^2}{2g}$  = pressure drop caused by inlet velocity
- $\Delta h$  = local pressure drop at vane leading edge
- $p_b$  = ambient pressure at liquid level
- $p_{min}$  = minimum static pressure in pump
- $p_v$  = liquid vapour pressure at prevailing temperature

The pressure heads are shown in Figure 9.

In order to avoid cavitation, the minimum static pressure in the pump ( $p_{min}$ ) must be larger than the liquid vapour pressure, or

$$p_{min} > p_v$$

Figure 10 shows the principle of static liquid pressure distribution in inlet pipe, pump and pressure pipe of a dry pump installation.

## 1.4.2 Reference Plane

The reference plane is the plane on which the NPSH calculations are performed. It is the horizontal plane through the centre point of the circle described by the tip of the vane leading edge. For horizontal pumps the reference plane coincides with the shaft centre line. For vertical pumps the location of the reference plane is stated by the pump manufacturer.

## 1.4.3 Required NPSH

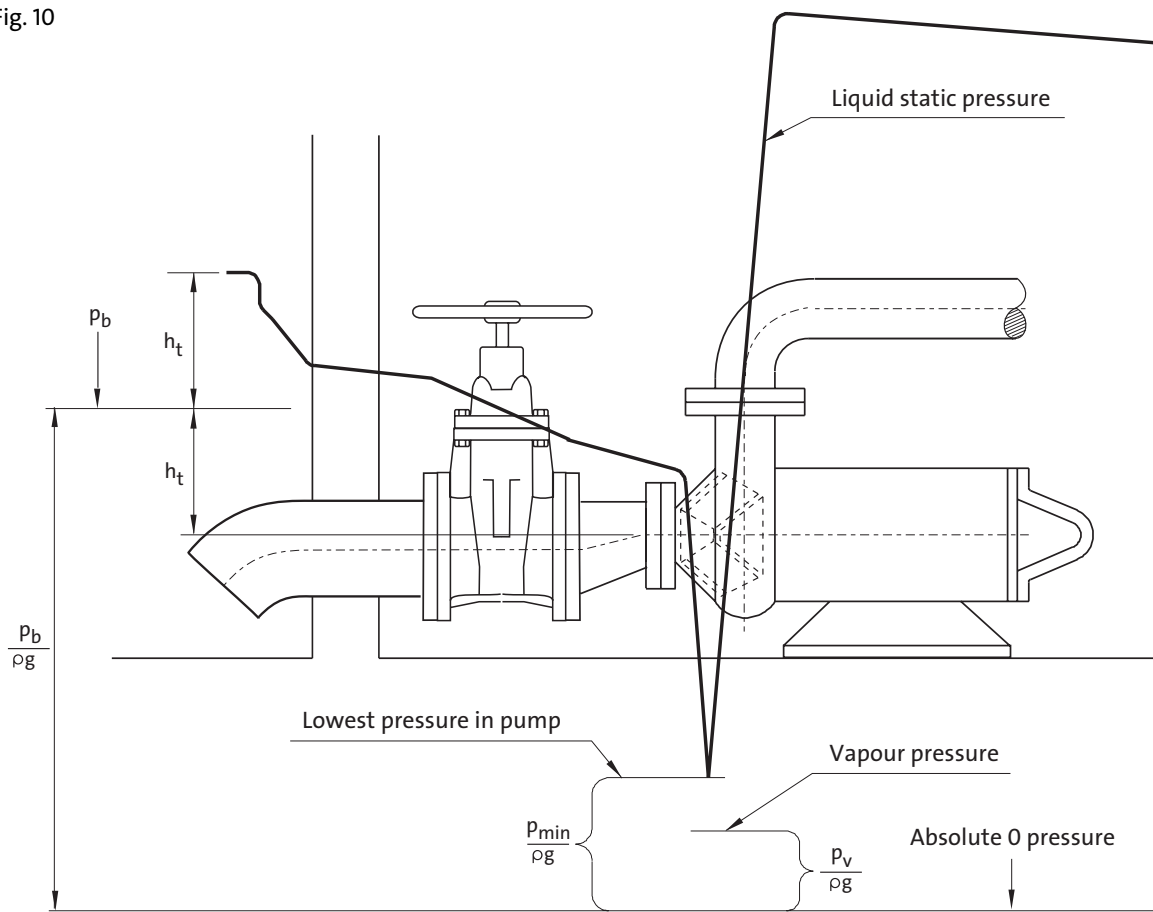
The required NPSH is obtained from the following equation:

$$NPSH_{required} = h_A + \frac{v_0^2}{2g} + \Delta h \quad (12)$$

This is also called the pump NPSH value. It can be presented as a function of flow as shown in Figure 11. It is independent of temperature and type of liquid being pumped. The pump manufacturer is required to state NPSH as a numeric value or curve.

Any pump will actually have different NPSH-values, depending of definition of occurrence, as can

Fig. 10



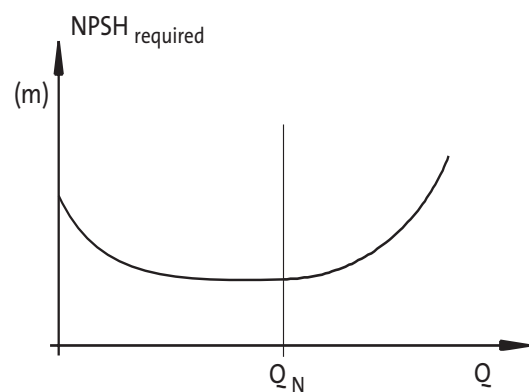
Pressure variation in a dry pump installation. Distribution of static liquid pressure in inlet pipe, pump and pressure pipe.

be seen in Figure 12. According to the testing standards used by pump manufacturers, the  $NPSH_r$  is defined as the situation where pump head is decreased by 3% due to cavitation. This value is defined as  $NPSH_3$ .

Light cavitation can be harmless to the pump if the vapour bubbles do not implode near the pump structural parts, such as the impeller vane.

The difference between the various NPSH values is greater in pumps with impellers with few vanes. Thus single-vane impellers have the greatest differences in NPSH values with the difference being caused by the  $NPSH_3$  curve dropping, and the tests thus giving too favourable readings. Therefore an  $NPSH_r$  curve based on the 3% rule of the standard is a poor base for a cavitation risk assessment in pumps with few vanes. The  $NPSH_r$

Fig. 11



Typical variation of required NPSH with pump flow rate.

# 1 Pump Theory

curve published by a pump manufacturer should in principle guarantee that no damages will occur in the pump if the pump is operated above it. This is especially the case for wastewater pumps, which have a low number of impeller vanes. The problem is that there is no accurate way of testing and establishing such an NPSH value.

## 1.4.4 Available NPSH

The available NPSH indicates the pressure available for the pump suction under the prevailing conditions. This may be called the pumping station NPSH.

$$\text{NPSH}_{\text{available}} = \frac{p_b}{\rho g} - H_{rt} - h_t - \frac{p_v}{\rho g} \quad (13)$$

The term  $h_t$  is positive when the reference plane is above the liquid surface and negative if below it. Available NPSH is determined by the pumping station designer.

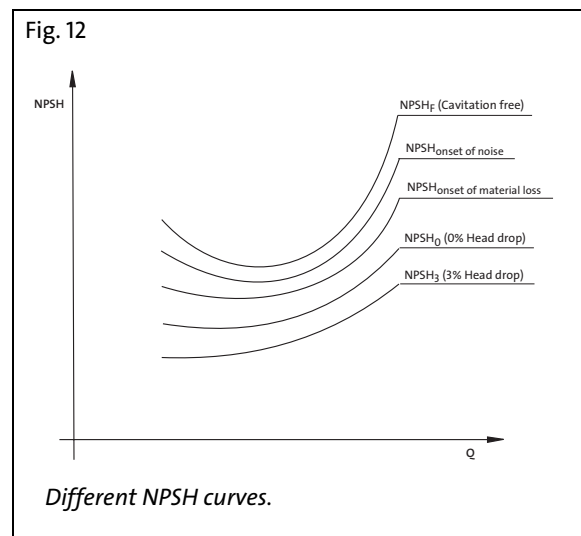


Figure 13 shows vapour pressure for water as a function of water temperature.

Figure 14 shows atmospheric pressure as function of elevation above sea level.

Fig. 13

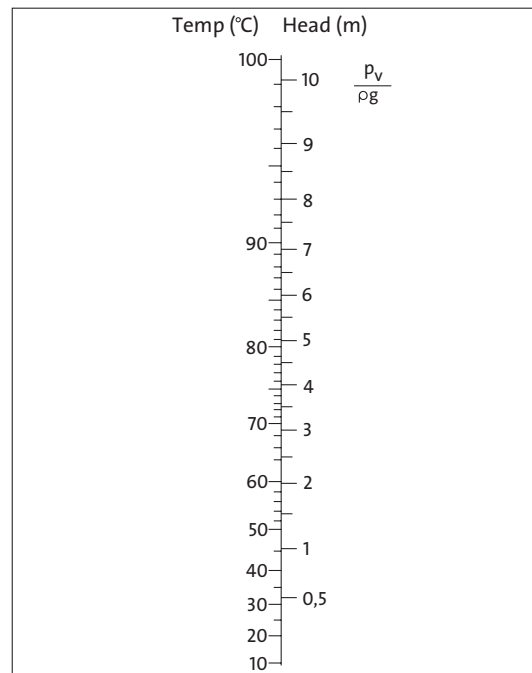
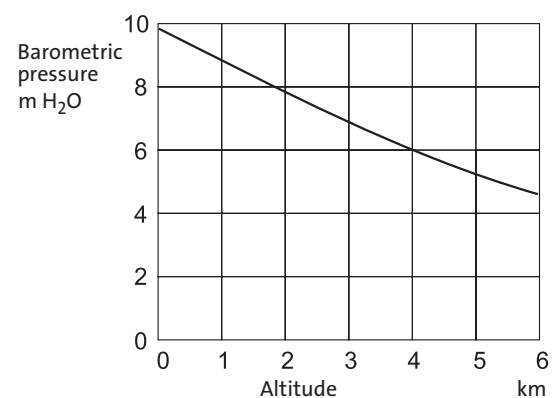


Fig. 14



### 1.4.5 NPSH Safety Margin

$$\text{NPSH}_{\text{available}} \geq \text{NPSH}_{\text{required}} + \text{Safety margin}$$

The NPSH margin must be sufficient to allow for variations in a situation where the real conditions may differ from those theoretically calculated. The suction pipe flow losses may be inaccurately estimated and actual pump operation point may differ from the theoretical because of variations in the Q/H curve and inexact pressure pipe resistance calculations. Harmful cavitation may occur earlier than expected, or at greater NPSH values than  $\text{NPSH}_3$  (Figure 12). Manufacturing technical variations of the shape of the vane leading edge may affect cavitation behaviour. The required NPSH may also be affected by the inlet pipe shape.

For horizontally installed pumps with straight suction pipes, a safety margin of 1 to 1,5 m is suitable.

For vertically installed pumps the safety margin should be set at 2 to 2,5 m, provided that a reducing bend is used before the pump inlet. Bend centreline curvature radius should be at least  $D_1 + 100$  mm, where  $D_1$  is the diameter of the larger opening.

The matter of NPSH, safety margins and measuring methods for NPSH are discussed in details in the EUROPUMP publication "NPSH FOR ROTODYNAMIC PUMPS, REFERENCE GUIDE" (1997).

### 1.4.6 Damming up of Suction Wells

In practical installations situations may arise, where the liquid level on the suction side rises and the pump head decreases so that the pump duty point moves to a sector where  $\text{NPSH}_r > 10$  m. No cavitation will occur, however, since  $\text{NPSH}_{\text{available}}$  will also rise and still be larger than  $\text{NPSH}_{\text{required}}$ . Typical installations where this situation will arise are dry dock drainage pumping, sewer blockage situations and drainage pumping with varying suction liquid levels.

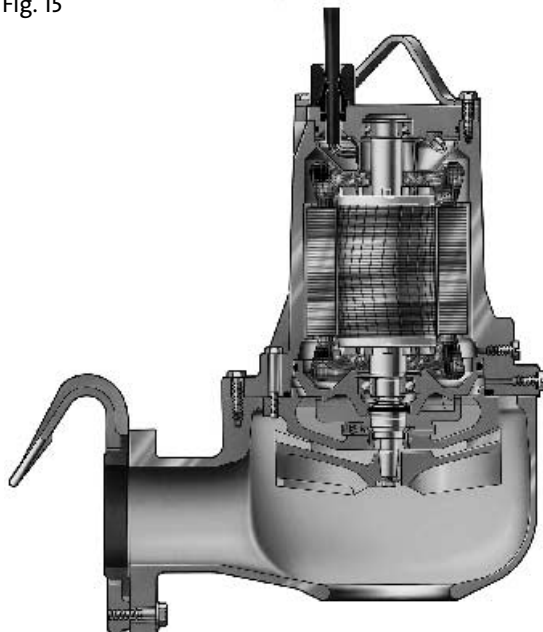
## 2 Pump Construction

This section describes the construction of modern electric submersible pumps. Various designs and the main parts of the pumps are discussed as well as topics concerning pump operation and maintenance. The study is limited to pumps for municipal sewage, drainage and raw water.

### 2.1 General

The submersible pump is a unit combining a pump and an electrical motor to an enclosed unit, suitable for submersible installation in a wet well holding the liquid to be pumped. The submersible pump may be connected to the pressure piping with a special baseplate connection at the bottom of the wet well for ease of installation and removal, or it can be installed connected with a flexible hose or other arrangements with riser pipes. Power to the unit is fed through one or more flexible cables, supplied with the pump in lengths suitable for the installation.

Fig. 15



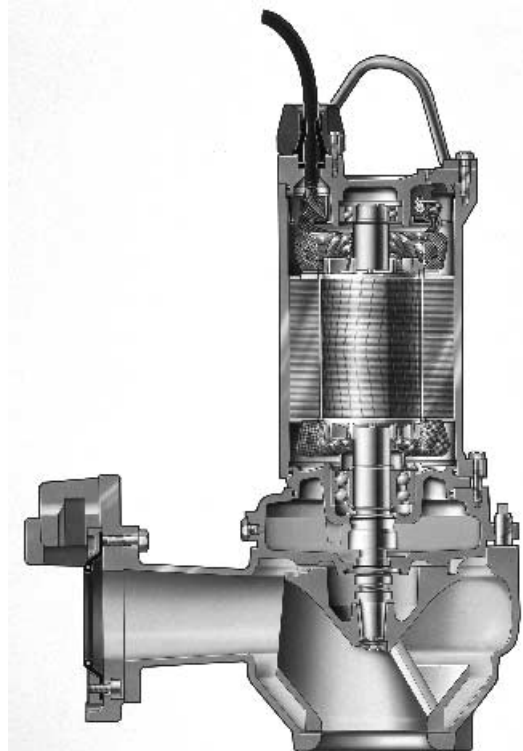
*Section of a GRUNDFOS 2,4 kW submersible pump showing details of motor and pump. The pump is fitted with a guide shoe for use with a submerged baseplate in the wet well, facilitating easy pump installation and removal.*

Many submersible pumps can also be installed dry like conventional pumps. This type of installation ensures uninterrupted operation of the installation in case of flooding of the dry well.

Submersible pumps are available for a number of applications with different requirements, and different designs for various special applications have been devised.

A submersible pump comprises a waterproof motor and matching pump components. The pump components include the impeller, the pump casing and the required connection parts for different installation alternatives. These include a guide shoe for submersible installation onto a matching connecting baseplate, a stand for porta-

Fig. 16



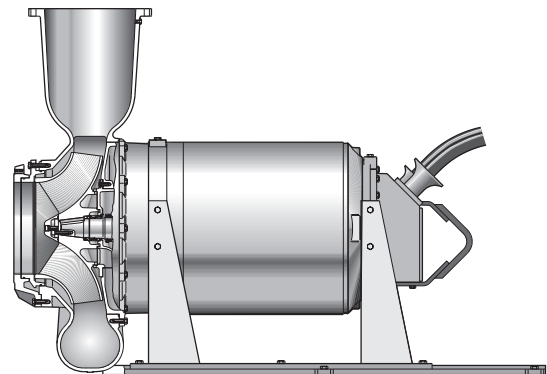
*Section of a GRUNDFOS 17 kW submersible pump showing details of motor and pump. The pump is fitted with a guide shoe for use with a submerged baseplate in the wet well, facilitating easy pump installation and removal. The pump casing is adjustable with set screws for maintenance of impeller suction clearance.*

ble pumps and the necessary connection flanges, stand for dry-installed pumps and seat ring for column-installed pumps.

The motor is a dry squirrel-cage electric motor matching a range of pump parts for various duties. Motor and pump have a common shaft with the bearings and shaft seals housed in the motor. The motor also includes watertight cable inlets and a handle for lifting the unit.

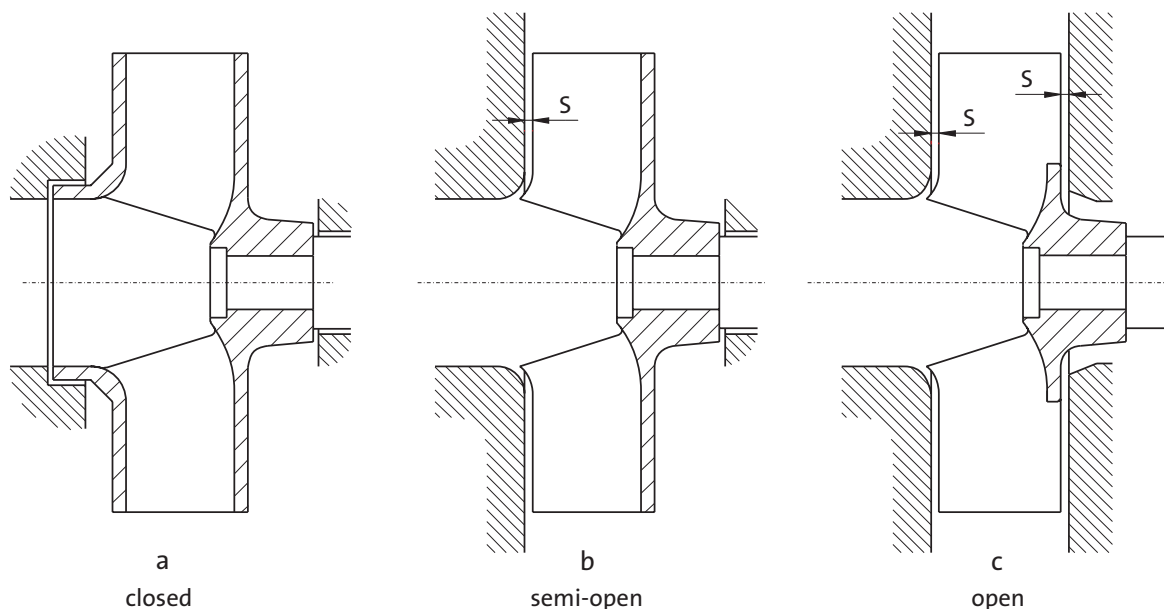
Figure 15 shows a modern small submersible sewage pump and Figure 16 a medium-sized submersible sewage pump. Submersible sewage pumps are available with motors rated from under 1 up to 500 kW for duties ranging from light portable use to large city sewerage system main pumping installations. A submersible pump for dry installation is shown in Figure 17.

Fig. 17



*Section and outline of a 160 kW submersible pump. The pump is intended for horizontal dry installation and connects with integral flange joints to both suction and pressure pipework. The submersible design permits flooding of the installation without risk of damage to the pump.*

Fig. 18



*Different impeller designs. The closed impeller has integral shrouds on both sides of the vanes, whereas the semi-open impeller incorporates only one shroud on the back side. An open impeller consists of only a hub and vanes, relying on close clearances (s) to the pump casing for its function.*

### 2.2 Pump

The pump comprises the impeller and the pump casing as well as ancillary devices and fittings.

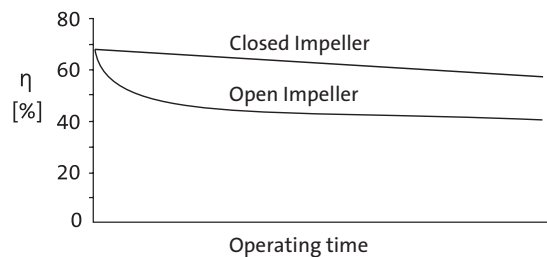
#### 2.2.1 Impellers

Submersible pumps are fitted with different impeller designs depending on intended use. The various impellers can be classified as

- impellers for sewage pumps
- impellers for macerating pumps
- propellers for axial pumps

Impellers can also be classified according to construction as closed, semi-open or open impellers. These are shown in Figure 18. Semi-open impellers and open impellers rely on the close clearance between impeller and pump casing (about 0,5 mm) for their function. The efficiency of these impellers is very sensitive to wear and will decrease rapidly as the clearance increases. Figure 19 shows the effect on pump efficiency from wear on closed and open impellers. The open and semi-open impellers are also susceptible to impurities becoming jammed between impeller and suction plate, slowing down or completely stopping the pump.

Fig. 19



Test results from a comparison of the effect of wear on pump efficiency for different impeller types.

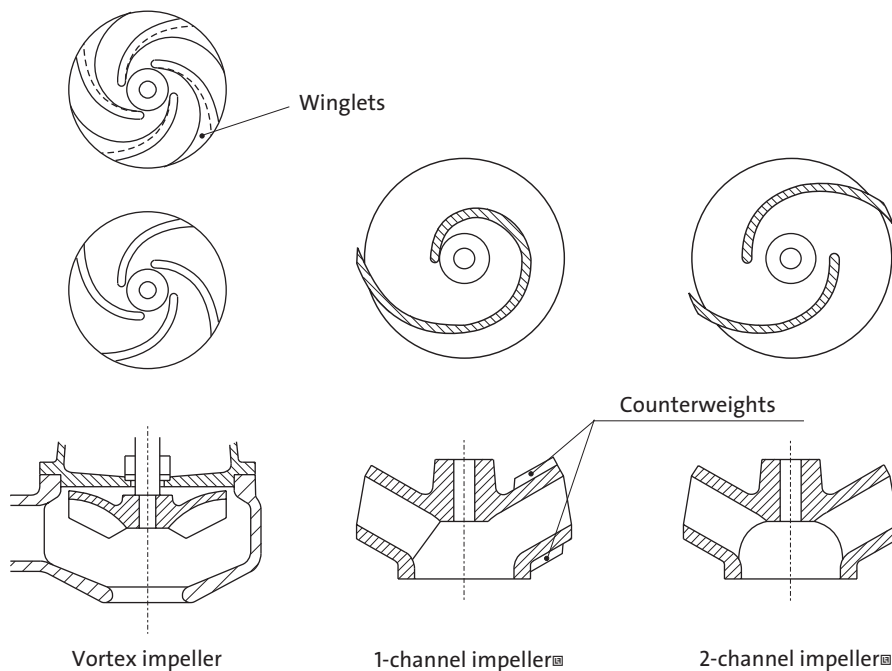
#### Impellers for Sewage Pumps

In order to avoid pump blockage, or clogging, special impellers have been developed for sewage pumping. These include single-channel impellers, double-channel impellers and vortex impellers. The design principles of these are shown in Figure 20. For very large sewage pumps, impellers with a multitude of vanes may also be used.

#### Free Passage

The concept of free passage is of special relevance to sewage pumps. It refers to the ability of the pump to let solids in the pumped liquid pass

Fig. 20



Impeller types for sewage pumps.

through, and thus to the capacity of being non-clogging. The dimension of the free passage usually refers to the largest spherical object that may pass through the impeller and the casing openings. If the free passage is described with two numbers, it refers to the largest oblong object that can pass crosswise through the pump.

The ability of a pump to operate without clogging relates strongly to the free passage, as can be seen in the diagram in Figure 21. Normally a free passage of 80 mm will be sufficient for unscreened sewage in small and medium-sized pumps. In larger pumps (flow > 100 l/s) the minimum free passage should be at least 100 mm.

Free passage alone does not ensure good properties against clogging in a pump. Impeller and vane geometry must also have features that prevent blockages. Pumps from different manufacturers have varying qualities in this respect. There are cases where a pump clogging problem has been solved by changing pump make, even with the pumps having equal free passages, same number of vanes and pump speed. The tendency of sewage to choke a pump may vary from one location to another, with "easy" and "difficult" pumping stations. The design of the sewer line leading up to the pumping station is important for the function of the pumps, because they must be able to handle any agglomeration of solids originating there. Real conditions in sewage systems cannot be simulated in laboratories, and the good properties of the Grundfos sewage pumps against clogging are based on long-term practical experience.

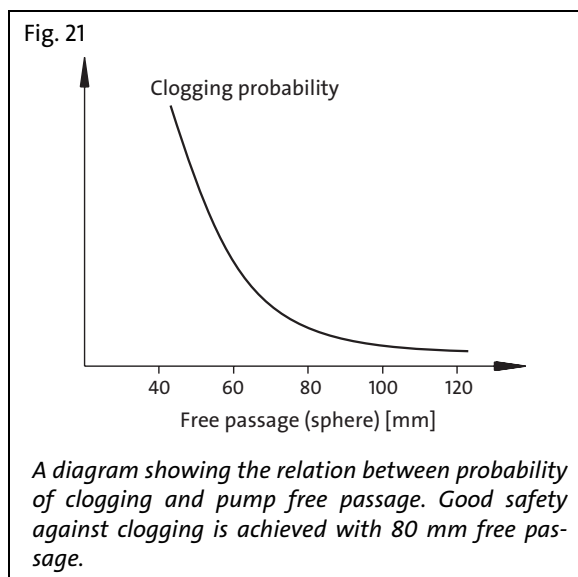
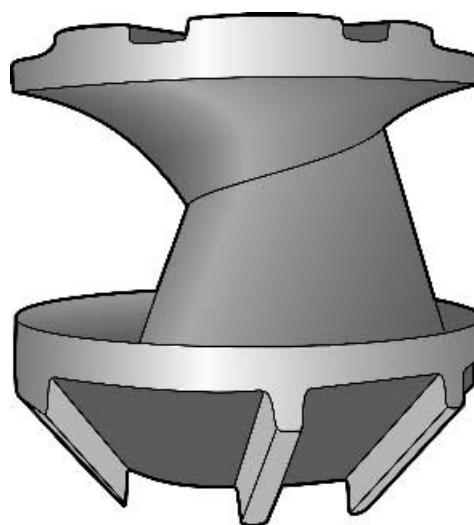


Fig. 22



A GRUNDFOS S-1 single-channel impeller for sewage use. The impeller is semi-axial in design with one long continuous vane, ensuring good properties against clogging. The asymmetric design requires the casting to include counterweight masses to facilitate static and dynamic balancing of the impeller

## Single-channel Impellers

A single-channel impeller is shown in Figure 22. The single vane is designed as long as possible for best efficiency within the limits set by the requirement of free passage. The impeller having only one route of passage for the pumped liquid ensures good inherent characteristics against clogging. The asymmetric shape requires the impeller to include integral counterweights for balance. Highest attainable efficiency is 70...75%.



Fig. 23



*A GRUNDFOS S-2 double-channel impeller. Good properties against clogging are achieved with recessed vane leading edges and a semi-axial design. The symmetric design is inherently in balance.*

### Double-channel Impellers

A double-channel impeller is shown in Figure 23. The inherent difficulty with double-channel impellers is that long, fibrous impurities may enter both channels and get caught by the vane leading edges, causing the pump to clog. This situation can be alleviated by good vane leading edge design, and this can be found only by development work under real conditions in difficult pumping stations. With the right design and a free passage of at least 100 mm, double-channel impellers can be designed to handle unscreened sewage without clogging. Highest attainable efficiency is 80...85% for double-channel impellers.

### Three- and Four-channel Impellers

In very large pumps impellers with three or four vanes can be used and still have a free passage of at least 100 mm and an impeller with good properties against clogging. Also for these impellers the design of the vane leading edge is decisive. Highest attainable efficiency is 82...86% for these impellers.

Fig. 24



*A GRUNDFOS SuperVortex impeller. The design includes patented vane winglets. The winglets prevent the formation of secondary eddies over the vane edges, greatly improving pump efficiency.*

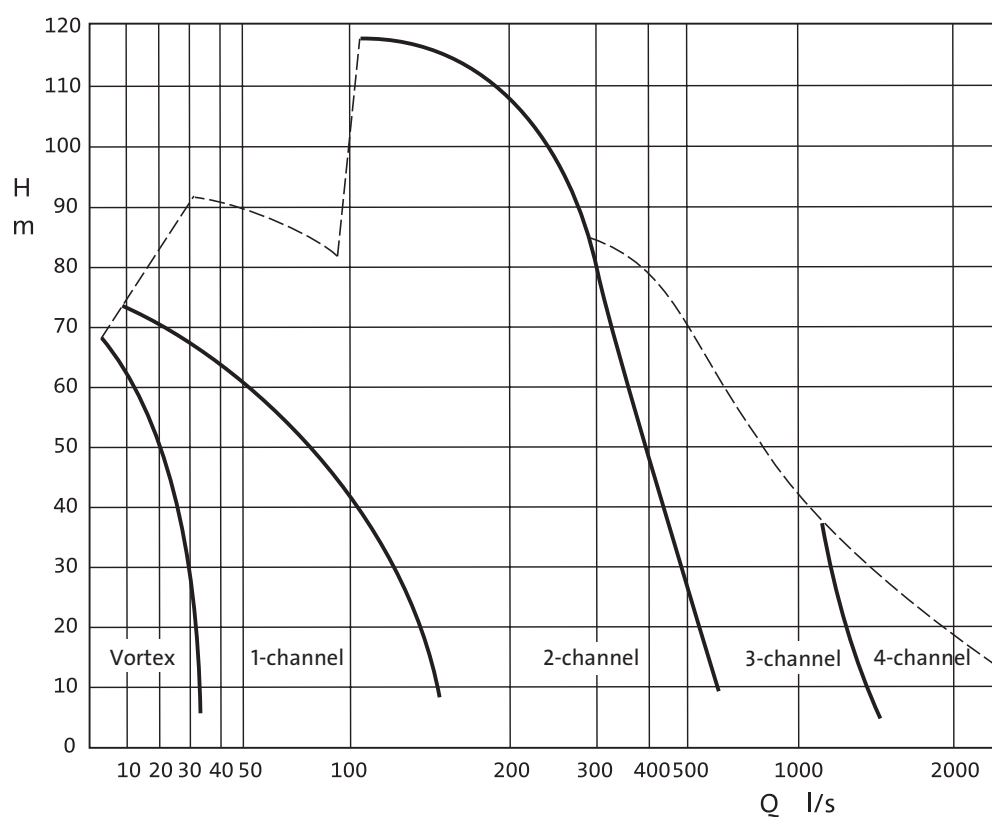
### Vortex Impellers

The principle of the vortex impeller is to induce a strong vortex in the open pump casing. The pumping action of the vortex pump is therefore indirect, with the impeller being situated outside the main liquid flow. Vortex impeller pumps have inherently excellent properties against clogging, and the pumps run very smoothly. The use of small vortex impeller pumps for sewage is increasing largely due to improved design and efficiency in later years. They are also used as sand separation pumps in sewage treatment plants. A vortex impeller is shown in Figure 24. Highest attainable efficiency is around 50% for vortex impellers. It is essential to note that the efficiency in the flow range 3...15 l/s of vortex pumps is roughly equal to that of single-channel pumps.

### Flow and Head (Q/H) Ranges for Different Impeller and Submersible Pump Types

Figure 25 shows the typical application areas for different sewage pump and impeller types of the GRUNDFOS range. It can be seen that with increasing flow and pump size the number of vanes of the impeller also increases. The diagram also shows the Q/H area for which submersible pumps are available for sewerage use. The largest pump in the Grundfos range has a motor of 520 kW power.

Fig. 25



Flow and head (Q/H) ranges for different impeller types.

Fig. 26



GRUNDFOS Grinder pump. The macerating unit is made of hardened stainless steel.

## 2 Pump Construction

### Impellers for Macerating Pumps

For installations with very small amounts of sewage, macerating pumps have been developed. Typical applications are pumping stations for single homes, small developments or camping areas. The required flow is very small, sometimes less than 1 l/s, but the total head may be high because of long and narrow rising mains. The flow for a macerating pumps are typically 1...5 l/s, with heads as high as 50 m.

In macerating pumps the solids are shredded into small fragments of around 10 mm, which makes it possible to use rising mains of small dimensions, usually DN 40...DN 80. For the very small flows from single pumping stations even smaller piping will be used, in order to attain a flow velocity of at least 0,5 m/s.

Macerating pumps may not be used for sewage with sand content, since the shredding unit is very susceptible to wear. Where macerating pumps are considered for larger installations comprising several buildings, it is always advisable make a technical and economic comparison with a solution based on conventional pumps.

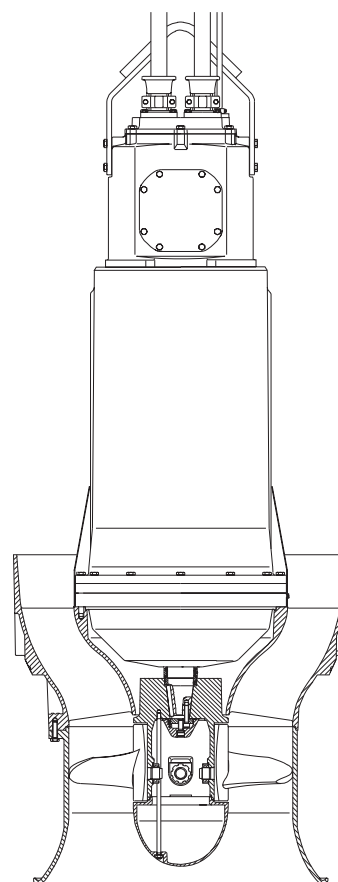
Figure 26 shows a GRUNDFOS macerating pump. Outside of the impeller is a macerating unit with sharp cutting elements installed. The macerating unit is made of hardened stainless steel.

### Propellers for Axial Pumps

Axial pumps, using the submersible motors from sewage pumps, have been developed by many manufacturers. Figure 27 shows a GRUNDFOS axial-flow pump with adjustable-pitch propeller. The design incorporates trailing fixed vanes that transform the rotating movement of the water into pressure energy, increasing pump efficiency. Propeller pumps are normally column-installed.

Propeller pumps are used for stormwater and flood water pumping, drainage, irrigation and raw water pumping as well as for effluent pumping in sewage treatment plants. Propeller pumps are not suitable for unscreened sewage because of risk of clogging. Small and medium-sized propeller pumps are not suitable for sewage treatment plant internal circulation pumping of e.g. return sludge, since they may clog and get jammed by the fibres present in these fluids. Highest attainable efficiency for propeller pumps is 75...85%.

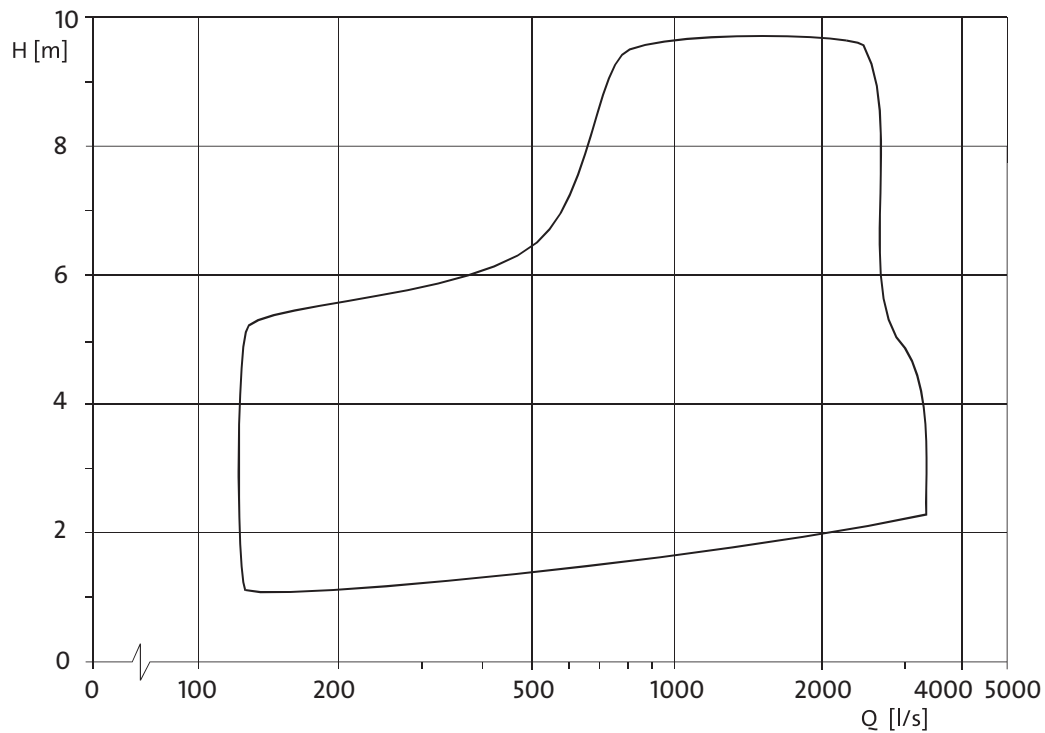
Fig. 27



*GRUNDFOS propeller pump. Blade angle is adjustable for best efficiency.*

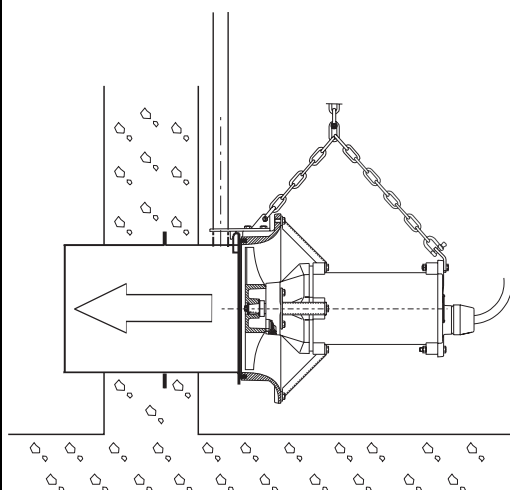
The operating range (Q/H area) of the GRUNDFOS propeller pumps is shown in Figure 28. Part of the range is also covered by channel-impeller pumps for column installation, which may be a more suitable choice for many applications. The final choice between these should be based on desired duty point and application. The pump manufacturer should be consulted in the selection process in difficult projects.

Fig. 28



GRUNDFOS propeller pump flow and head (Q/H) range.

Fig. 29



Submersible recirculation pump for sewage treatment plant. The pump is lowered in place along guide rails.

For recirculation pumping in sewage treatment plants special axial pumps have been developed as shown in Figure 29. They are intended to operate at very low heads, only 0,3... 1,0 m, and high flow rates, up to 2000 l/s. These pumps are designed non-clogging with back-swept blades, large (10 mm) clearance between blade tips and casing, and by omitting the lead vanes. Highest attainable efficiency for circulation axial pumps is 35...50%. The delivery loss at the exit of the thimble is significant for the head. Additional head can be attained by using a conical thimble, lowering losses.

## 2 Pump Construction

Fig. 30



### Impeller Auxiliary Vanes

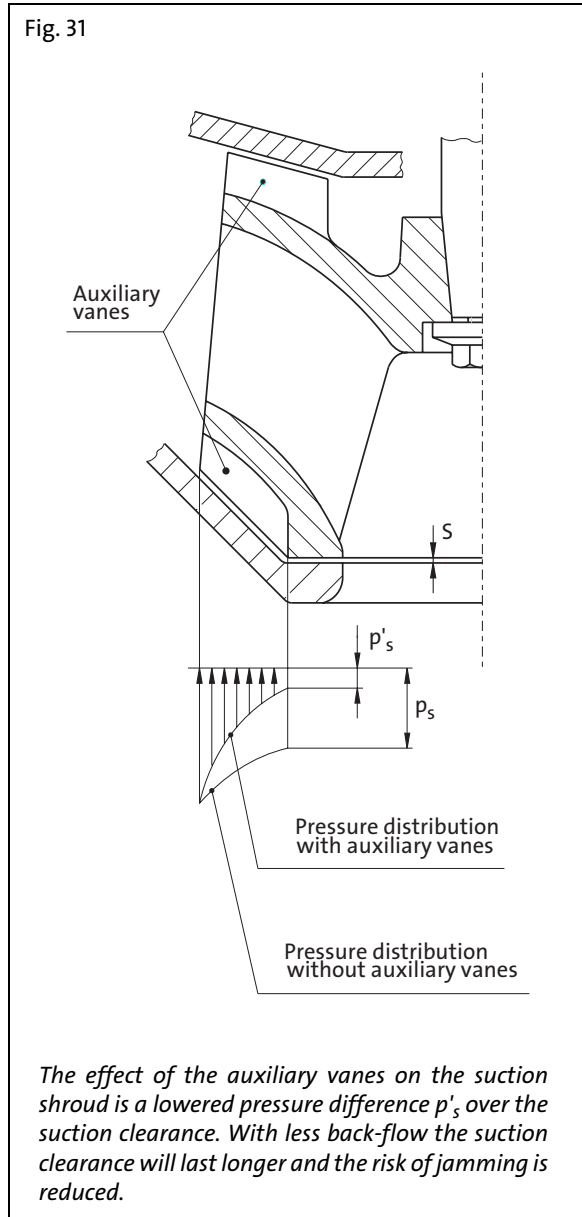
Auxiliary vanes on the outside of the shrouds are an important feature of the impellers of small sewage pumps. The auxiliary vanes increase the velocity of the flow of fluid in the space between the impeller and the pump casing. Figure 31 shows the location of auxiliary vanes on a single-channel pump impeller.

Auxiliary vanes assist the pump operation by performing the following functions:

- Decrease axial loads on bearings, particularly if semi-open impellers are being used
- Reduce impeller and casing wear at the suction clearance
- Prevent the wedging of fibres in the suction clearance
- Prevent fibres and rags from wrapping around the pump shaft behind the impeller.

The use of auxiliary vanes extending to the shroud perimeter is not possible on large impellers, since at high flow rates they could cause a pressure drop below the vapour pressure of the liquid, leading to cavitation. Large pumps are, however, less prone to jamming because of high motor torque. Auxiliary vanes are therefore not included on the inlet side of large impellers.

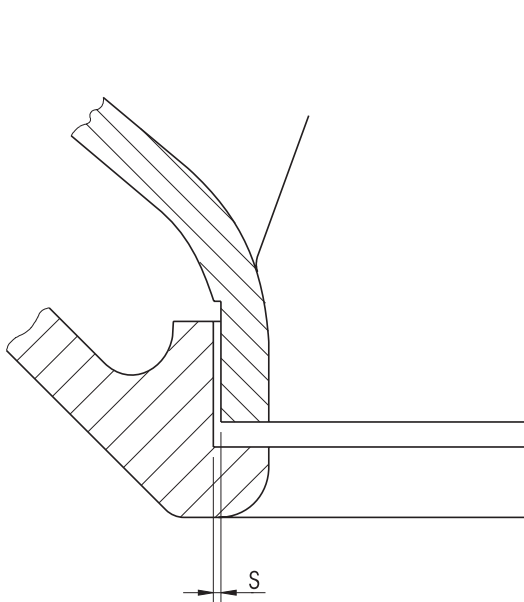
Fig. 31



### Suction Clearance

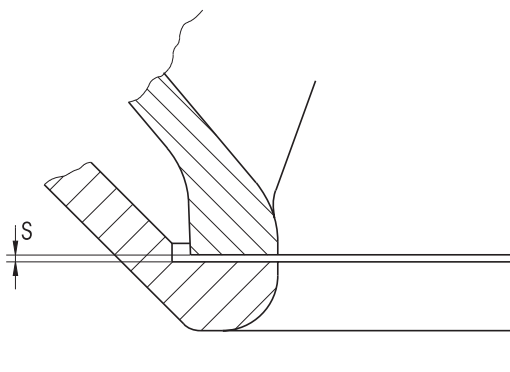
The clearance between the impeller and the pump casing should be as small as possible in order to reduce leakage losses. The suction clearance is in the order of 0,5...1,0 mm for most centrifugal pumps. The clearance can be designed cylindrical or axial as shown in Figures 32 and 33.

Fig. 32



*Cylindrical suction clearance. The design is susceptible to jamming, since fibres that get wedged in the space between impeller and casing may accumulate and drag down the pump. In case of wear, a wear ring on the suction cover and the impeller need to be replaced or re-machined.*

Fig. 33



*Axial suction clearance. This design is less prone to jamming, since drag forces will remove wedged material towards the pump suction. The clearance can be made adjustable for ease of maintenance and wear compensation.*

Pump performance and efficiency over time are dependent on the suction clearance being kept within specified limits. The lowering effect of the suction clearance on pump efficiency and head can be calculated with the following empirical equation:

$$\Delta\eta \approx \Delta H \approx \sqrt{K^2 + K} - K \quad (14)$$

where

$$K = 0,008 \cdot s^2 \cdot \frac{H}{Q}$$

Q= flow [l/s]

H= head [m]

s= clearance [mm]

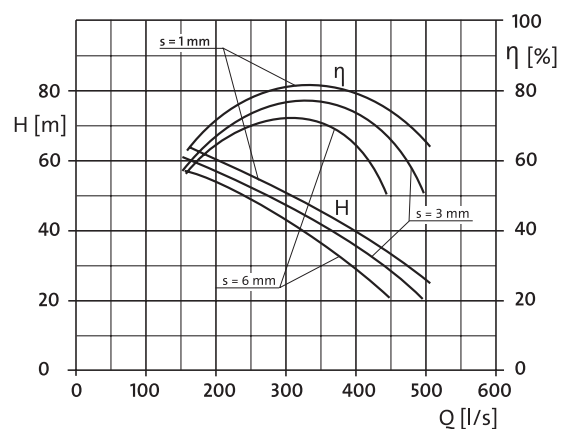
$\Delta\eta$  and  $\Delta H$  are proportional.

For semi-open impellers the effect is increased by the factor 1,5.

Figure 34 shows the results of a test where a pump was operated with varying suction clearance.

If the suction clearance widens to 2...3 mm for impellers without auxiliary vanes and to 4...5 mm for impellers with auxiliary vanes, it is necessary to restore the clearance in order to retain pump performance. If the suction clearance is made adjustable, this procedure is easily done by service technicians in the field, whereas a pump with

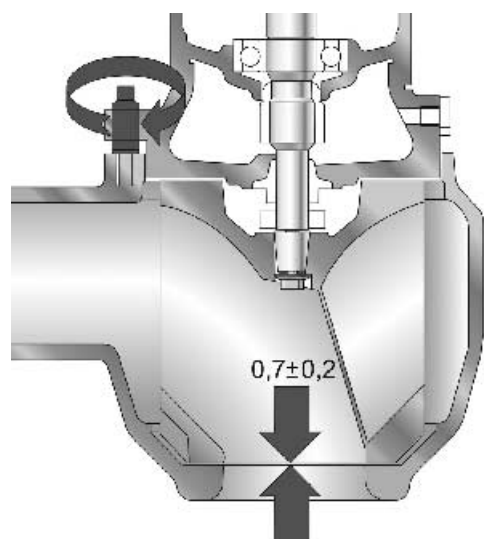
Fig. 34



*Effect of different suction clearance dimensions on pump curve and efficiency.*

## 2 Pump Construction

Fig. 35



*Suction clearance adjustment system with three set screws.*

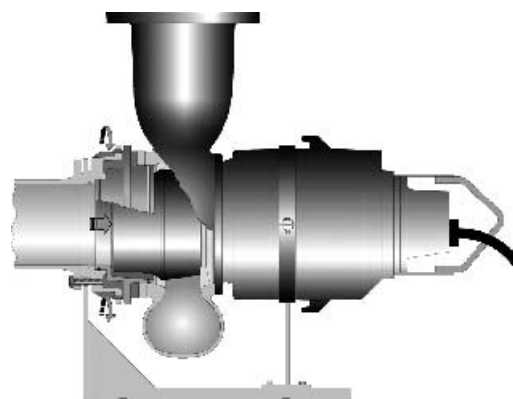
fixed impeller suction clearance will have to be brought to the shop for overhaul, or worse, scrapped for high costs of spare parts and work.

In pumps with adjustable axial suction clearance performance can always be guaranteed by checking and adjusting the suction clearance during routine maintenance. Figure 35 shows a submersible pump design, where the suction clearance is adjusted with the help of three set screws.

For dry-installed pumps GRUNDFOS has developed a patented design (SmartTrim) allowing the suction clearance to be adjusted and restored without the need of removing the pump or opening pipe connections. Adjustment does not affect pipe connections or require re-alignment of these. Figure 36 shows the principle. The adjustment is done by first closing the clearance and then backing up the adjustment screws 1 mm, after which the suction cover is tightened against the set screws with the fastening bolts.

The adjustment margin on the GRUNDFOS pumps is 10...15 mm, depending on pump size. It is dimensioned to last the lifetime of the impeller.

Fig. 36



*Suction clearance external adjustment system on dry-installed pumps.*

### Impeller Attachment

The attachment of the impeller onto the shaft must be both reliable and easy to dismantle. Removal is necessary for shaft seal maintenance, and for impeller replacement if the pump is used for pumping abrasive materials. The impeller may have either a cylindrical or tapered fit onto the shaft end.

A shaft joint tapered to the right angle is easy to dismantle. The tapered joint is additionally tightened with a screw, which makes the joint free of play and rigid.

The joint is keyed for transmission of torque. Solid impeller mounting is a key component in pump operational reliability, and great care should always be exercised when the impeller is dismantled. It is good practice to always use a torque wrench when setting the impeller screw. The pump manufacturer issues correct tightening torque information and possible recommendation of screw lubricant in each case.



## 2.3 Motors

### 2.3.1 General

Submersible pump motors are squirrel-cage electric motors wound for regular three-phase or single-phase alternating current supply. Single-phase pump motors are available only for small pumps (2 kW or less). Motors are available for 50 or 60 Hz supply and a number of voltages. The motors are built for submersible operation, Class IP 68 according to IEC. The electrical features of submersible motors are described in detail later in this book.

The submersible pump is a fixed combination of a motor and a pump with a common shaft and bearings. The motor is short-coupled to the pump, and some of the pump parts, such as the volute cover may be integral with the motor attachment flange. For best results the pump and motor are designed together, with one motor frame fitting a range of pump parts for different duties and different operational ranges by the same manufacturer. Motor and pump sections are selected and designed so as to exclude overloading at any duty point on the pump curve.

Submersible motors are normally air-filled. Small motors (1,5 kW and less) are also made oil-filled. The oil used in these is low-viscosity oil used also in transformers, in order to keep the rotor friction losses as small as possible. The growing losses and lower efficiency prevent manufacturers from making larger oil-filled motors. Oil-filled motors are cheaper than air-filled motors because of smaller number of parts.

### 2.3.2 Explosion-proof Motors

Submersible pumps are available in explosion-proof versions for use in environments where the pumped liquid or ambient atmosphere may contain explosive gases. This condition may exist, for example, at or near petrochemical works but a space can also be defined as explosive elsewhere, if deemed necessary for safety reasons.

The principle of explosion-proof motors is their safety against causing a potentially explosive atmosphere from igniting. The following two alternative technical solutions are available to fill the requirement:

- The motor is designed so that the enclosure

can withstand any internal explosive blaze and prevent it from spreading into the explosive surroundings. This is referred to as Class D.

- The motor is designed so that no sparking or high temperatures may occur inside the motor. This is referred to as Class E.

An explosion-proof motor is designed and built according to the rules set forth by international governing bodies (for example, Euronorm 50014 and 50018). The requirements for class D motors are detailed, involving among others the selection and gauge of construction materials, casing joint design and manufacturing tolerances, motor interior volume utilization as well as strength of the structure and fasteners. The essential requirement for the joints is that the mating surfaces have to be longer, as they are supposed to serve as “extinguishing” gaps. Certification and approval of a design is always subject to extensive tests, where the actual ability to withstand internal explosions, is determined.

Class E explosion-proof motors do not require extensive structural modifications, but are tested for internal temperature rise at certain loads. Also internal sparking must be prevented by adequate gaps between rotating and stationary parts.

Usually explosion-proof motors are based on the regular designs of a manufacturer, and form a complement to these. The power characteristics are normally not altered, and the pump parts are common for both. The structural requirements on explosion-proof motors make these more expensive than regular motors.

### 2.3.3 Motor Cooling

Mechanical and electric losses in the motor are converted to heat, which must be dissipated. In a regular submersible pump motor (see Figure 14) the heat is transferred from the stator casing to the liquid through submersion. For cooling purposes it is normally sufficient if the motor is submerged to about half of the motor depth. The liquid level may be pumped down all the way for short periods without risk of overheating the motor.

A motor operating in water this way is in fact very efficiently cooled, since cooling continues after the motor has stopped. Thus it is possible to allow frequent starts and stops of submersible motors,



## 2 Pump Construction

which is beneficial for the design of pumping installations.

### Allowed Water Temperature

Cooling of submersible motors relies on the pumped liquid, either by submersion or otherwise. Water temperature is therefore essential. Usually the motors are rated for +40 °C liquids. Higher liquid temperatures may be allowed, but then the pump selection should be referred to the pump manufacturer. Also the cavitation risk must be assessed for higher temperatures with an NPSH analysis, because of higher vapour pressure of the liquid.

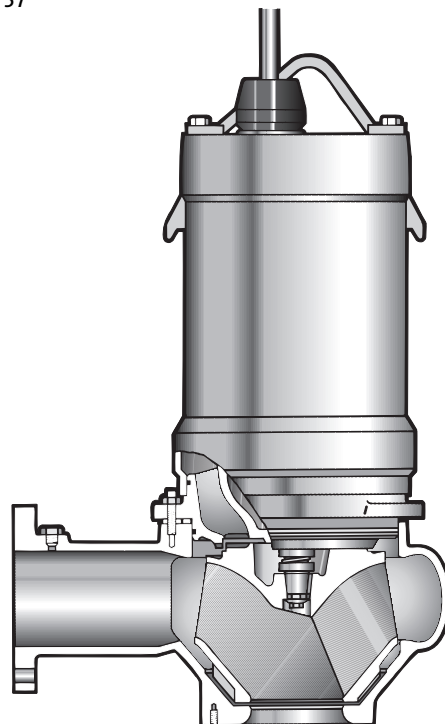
### Submersible Motor Cooling in Dry Installations

Many submersible motors are installed dry for various reasons. Adequate cooling of these motors must be ensured, and it can be arranged in a number of ways:

With a **cooling water jacket** that encases the motor or parts thereof. Part of the pumped liquid is diverted through channels from the pump casing into the cooling jacket where it recirculates after the casing has filled up. The water enters the space behind the impeller through a filtering clearance (about 0,5 mm) and is circulated by the auxiliary vanes on the back of the shroud around the motor stator housing inside the jacket. Excess heat is conveyed to the water through forced convection, ensuring efficient cooling. The principle is shown in Figure 37. The usage of a filtering clearance and wide enough cooling channels has ensured that the system is non-clogging also in practice. A cooling jacket is often optional on small and medium-sized pumps for dry installations, whereas very large pumps are often equipped with a cooling jacket as standard regardless of installation method.

In some cases, where the liquid being pumped is unsuitable for circulation in the water jacket, external cooling water may be used. In these cases the pump is modified with external water connections in the jacket and by plugging the entrance channels from the pump casing. A safety circuit is necessary to protect the pump from overheating due to disruption of the external cooling water supply.

Fig. 37

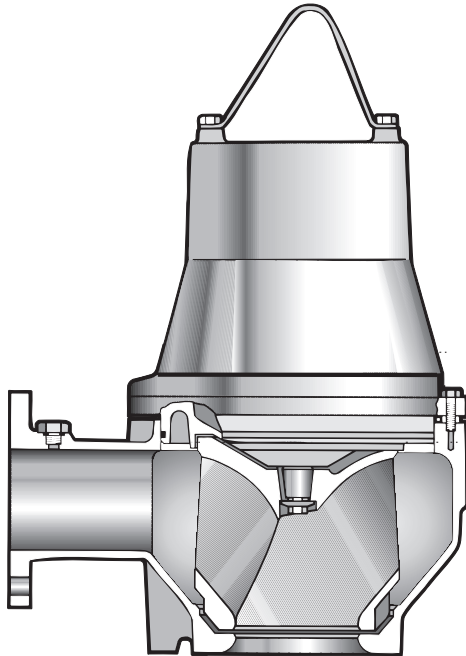


*A dry-installed GRUNDFOS submersible pump with motor cooling jacket. Part of the pumped liquid is filtered through a gap of about 0,5 mm and remains recirculating in the cooling jacket, circulated by the pumping action of the impeller back shroud auxiliary vanes. Efficient cooling is provided by heat convection from the stator through dissipation into the pumped liquid.*

With **thick stator housing walls**. This design, suitable for small submersible pumps, employs a thickened stator housing that conveys the heat from the stator to the pumped liquid. In this construction the stator housing flange may contact the pumped liquid directly or through the oil housing cover flange. The flange can be shaped with a recess or channel for good contact with the liquid. The stator casing may also be made of aluminium in dry-installed pumps to further enhance heat dissipation. Figure 38 shows the construction.

For dry-installed pumps only a cooling water jacket offers equal or even superior motor cooling to submergence. Other motors may have to be down rated for dry installations, limiting the selection of pump components from the matching range.

Fig. 38



*A GRUNDFOS submersible pump suitable for dry or submerged installation. The thick-walled lower section of the motor serves as a heat conduit to the pumped liquid. The stator casing may be made of aluminium to further enhance the effect.*

With an **internal cooling circuit**, where a cooling fluid, e.g. glycol, is circulated by a separate small impeller on the pump motor shaft. The pump incorporates a heat exchanger between pump housing and the motor, where the cooling fluid yields heat into the pumped liquid. System complexity may pose problems.

### 2.3.4 Motor Tightness

Water intrusion in the motor leads invariably to damage, or, if detected by safety devices, at least to pump outage. The chief requirement and design consideration of submersible pump motors is therefore complete integrity against leakage. Motor tightness is ensured with good design and continuous quality control including tests during manufacturing.

All submersible motor joints are machined to fit, and O-rings are used throughout. The O-rings are

renewed each time a joint is opened for service to ensure tightness.

The electrical cable inlet to the motor must be reliably watertight. A good design uses compressible rubber grommets that match both cable and the inlet recess. The grommet is compressed to prescribed tightness by the shape of the matching parts when assembled. A cable clamp external to the sealing carries all outside tensile loads on the cable, preventing pulling at the seal.

The possibility of water intrusion through the cable is a reality. If the cable free end is allowed to be submerged, water may travel by capillary action between the copper strands of the leads to the motor. This action is enhanced by the temperature changes of the motor, and water may this way enter an otherwise undamaged motor. The condition can arise in new pumps that have been stored outdoors prior to installation with the cable free end unprotected.

Most pump manufacturers deliver their pumps with protecting sleeves on the cable free ends. Warning labels are attached to warn the storage and installation personnel of the risk of submerging the cable free end.

Securing submersible motor tightness requires special knowledge and special tooling, and it is therefore advisable to return the pump to an authorized shop for repairs. Pump manufacturers offer training and special tools to their customers. For owners of large numbers of submersible pumps an in-house authorized shop may be warranted.

### Shaft Seals

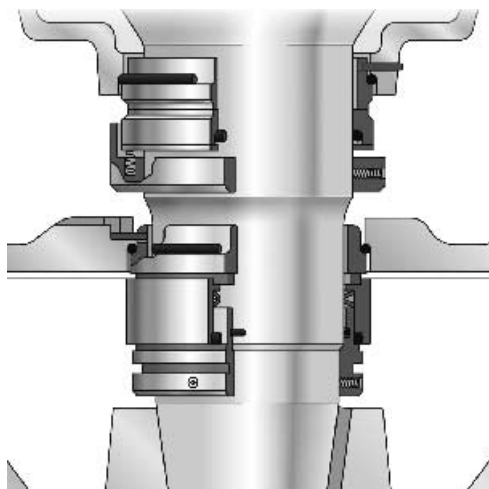
The shaft seal, providing safety against leakage of the pumped liquid into the motor, is one of the most important elements in a submersible pump.

Modern submersible pumps almost exclusively use a shaft sealing arrangement with double mechanical seals separated by an oil-filled chamber. The arrangement, developed and refined over the years, provides adequate protection against leakage and motor damage in most cases.

Figure 39 shows a mechanical seal arrangement used in submersible pumps. There is a lower or primary seal and an upper or secondary seal. The

## 2 Pump Construction

Fig. 39



*A GRUNDFOS double mechanical seal with primary and secondary seals.*

seals, being separated by an oil bath, operate under different conditions. This is reflected in their construction with different materials. Both seals comprise two contacting slip rings, one stationary and one rotating with the shaft. The rings are pressed against each other by spring force and, for the primary seal, in addition by the pump pressure.

Sealing between the slip rings is based on the extremely smooth and flat contact surfaces of the slip rings. The surfaces are in such close contact that no or only a very minute leakage can pass between them. The flatness and smoothness of the rings are in the magnitude of 0,0005 mm and the faces are finished by lapping. The slip rings seal against the stationary seat or shaft with O-rings. The material of the O-rings is selected to withstand high temperature and the corrosive and dissolving action of the seal oil and the impurities in the pumped liquid.

Notches in the stationary slip rings of the primary seal secure them in the seat against turning. The rotating rings are locked similarly with drive pins. Spring clips or washers keep the stationary rings in their seats during abnormal pressure situations.

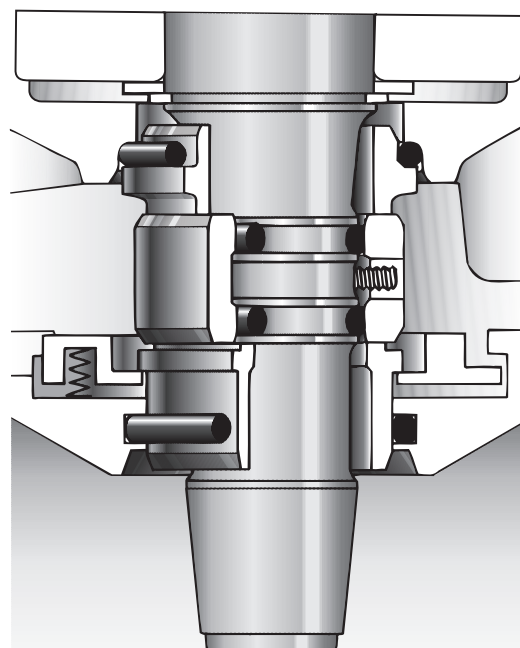
The material of the primary seal faces is normally hard, because of the abrasive action of the

pumped liquid. The material used today is silicon carbide (SiC), which has a hardness around 2000 on the Vickers scale and ranks next to the diamond. The silicon carbide rings can be either solid or converted. Converted carbide rings are sintered to SiC to a depth of approximately 1 mm, leaving the ring interior unchanged. SiC also has very good resistance against corrosion, and can be used in all wastewater and dewatering applications.

If the secondary seal is oil lubricated, a combination of materials may be used. A stationary ring of a softer material with good friction properties in combination with a hard rotating ring provides for low seal rotation resistance. The oil lubrication protects the seal against wear. Modern secondary seals normally have faces of silicon carbide and carbon.

Modern submersible pumps utilize mechanical seals custom-designed for the purpose. Good designs have been developed by most major manufacturers. A proprietary design combining primary and secondary seal is shown in Figure 40.

Fig. 40



*A GRUNDFOS integrated double mechanical seal.*

All mechanical seals used in submersible pumps must allow rotation in either direction, since pumps frequently get started in the wrong direction or may be turned backwards by back-flowing water in installations without check valves.

All submersible pumps with double mechanical seals have an oil space between the seals. The oil serves the following functions vital to the function of the seals and the pump:

- Seal lubrication, especially of the secondary seal
- Seal cooling
- Emulsification of possible leakage water, thus rendering it less harmful
- Seal condition monitoring. By checking the seal oil during maintenance, the seal condition and rate of leakage can be estimated.

Overfilling of the seal oil chamber should be avoided in order for the oil to be able to absorb leakage water by emulsification and to prevent possible overpressure due to thermal expansion of the oil. The pump manufacturer provides information on oil quantity and filling and monitoring methods.

In special applications, where the pumped liquid contains very fine materials, the primary seal may open due to material build-up on the slip ring faces. In these cases it may be warranted to arrange for continuous external flushing of the seal. These installations are always considered separately for each case by manufacturer and customer.

Mechanical seal life expectancy cannot be determined theoretically or even by lab tests. Performance over time is also difficult to predict. Seal life varies greatly from case to case, with service records from a few years to over 15 years reported.

## 2.3.5 Motor Bearings

### Bearing loads

The submersible pump bearings carry the combined load of the pump and motor as exerted on the common shaft. The following forces act on the bearings, either radially or axially:

- hydrodynamic radial force
- hydrodynamic axial force
- magnetic radial force
- the weight of the rotating parts

The significant forces acting on the bearings are the hydrodynamic forces.

The hydrodynamic radial force is the resultant of the pressure distribution at the impeller perimeter in various relative positions to the pump casing. The radial force is dependent on a number of design factors as well as on the pump operating point.

The hydrodynamic axial force is the resultant of the forces induced by the impeller diverting the flow from axial suction to radial discharge, and from the pressure difference between suction and pressure side of the impeller. The axial force is also strongly related to the pump flow and operating point.

### Bearings

Rolling bearings are used throughout in submersible pump motors. Ball bearings are used for their ability to carry both axial and radial loads. In very large motors a combination of ball bearings and roller bearings are used because of the large forces on the components.

To allow for heat expansion of the shaft and for manufacturing tolerances, the shaft upper bearing is allowed axial movement, whereas the lower bearing is locked axially.

Bearing selection is governed by international standards with regard to bearing life. According to the pump standard ISO 5199, "Bearing rating life (B10)" should be at least 17500 hours.

Submersible pump bearings are normally lubricated for life at the pump factory, using special grease suitable for the high operating temperatures allowed in submersible motors.

## 2 Pump Construction

### 2.3.6 Motor Protection Devices

Submersible motors are equipped with various protection devices for prevention of damages for the following reasons:

- overheating
- water intrusion
- seal failure
- bearing failure
- winding insulation deterioration

Some protection devices are standard issue whereas others may be available as extra equipment on request only. Large pumps need better protection devices because of the greater economic values of these pumps.

The protection devices can be divided into internal devices with sensors inside the motor and external devices in the pump motor control panel.

#### Internal Protection Devices

The following protective devices are mounted inside the motor:

- **Thermal switches** in stator windings. These are normally bimetal miniature switches that open at a fixed, preset temperature, please refer to Figure 41. Three switches, one in each phase, are used in three-phase motors. The switches are connected in series in the control circuit, which is wired to stop the motor when opening. The switches reset upon cooling, closing the circuit.

Fig. 41



*Thermal switch. The unit consists of a miniature bi-metal switch that opens according to the switch temperature rating. The switch is connected in the control panel to break the current in case the motor is overheating.*

ing the circuit, making re-start of the pump possible. Thermal switches in the windings protect the motor against overheating from insufficient cooling, and are especially important in pumps depending on submergence for cooling.

- Water intrusion into the sealed motor can be monitored with a **moisture switch** that reacts to excess moisture. Normally the moisture switch is connected in series with the thermal switches in a circuit that disconnects the circuit breaker coil and stops the motor upon opening. Figure 42 shows a moisture switch that operates when the humidity reaches 100%. The moisture switch is non-reversing and does not reset after tripping. In a common circuit with moisture and thermal switches, it can be determined which device has opened, since only the thermal switches close again after cooling. The motor must be opened and dried out before any attempts to restart it after the moisture switch has tripped.
- Water intrusion into the sealed motor past the shaft seals can be monitored with a **leakage detector** sensor in the seal oil chamber. Regular motor oils used as seal oil in submersible pumps can form an emulsion with up to 30% water content. The leakage detector either reacts on a water content exceeding 30% (con-

Fig. 42



*A GRUNDFOS moisture switch. The unit consists of a number of moisture-sensitive disks stacked onto an actuating rod, and a micro switch. The hygroscopic disks expand upon contact with excess moisture, pulling the actuating rod. A cam at the rod end trips the micro switch and breaks the circuit. The unit is non-reversing and must be replaced after use.*



ductive detectors) or monitors the water content continuously (capacitive detectors). The latter may be calibrated to trip at any water content and used to indirectly observe primary seal condition by monitoring water intrusion over time (leakage rate). Leakage detectors are usually not standard, but available as extra equipment.

- Water intrusion into the sealed motor through capillary flow through the supply cable before pump installation can be prevented by fitting a tight **protective sleeve** over the cable free end at the factory. The sleeve is not removed until the cable is connected at the control panel.
- The condition of the bearings and/or bearing grease can be monitored with **thermal sensors** in the bearing bracket. These are installed close to the bearing outer race, and calibrated to register bearing temperature. Thermal sensors are extra equipment.

### External Protection Devices

The following protective devices are mounted in the motor control panel:

- **Short-circuit protection** is accomplished by means of fuses, circuit breakers or electronic motor protectors. Fuses and circuit breakers should be dimensioned to withstand the motor starting current, but the rating must not exceed that of the supply cable or switchgear. Where fuses are used, these should be of the slow type.
- Overload protection is required in a sudden overload situation, such as when the impeller

develops operational difficulties or gets jammed, when the pump becomes clogged or during loss of phase in the mains supply. Overload protection is frequently provided by overload relays coupled to the motor contactors. These consist of ambient temperature-compensated bimetal elements, that trip the current to the contactor coils in case the current exceeds the set specified value. **Overload relays** provide good protection against loss of phase in the supply. The overload relay should be set according to the motor nominal current. When star delta start is used, the current through the overload relay is reduced by the factor 0,58 ( $1/\sqrt{3}$ ), which must be taken into account when setting the relay. Figure 43 shows an overload relay.

- The stator winding insulation is monitored by an **automatic resistance measuring device** that measures the resistance between the phases and between phases and earth each time the pump stops. Alarm levels for resistance can be set, preventing damages short circuits and damages to windings.

Fig. 43



*Thermal overload relay. The relay connects to the motor contactor and breaks the current in case of the electric load exceeding the set value.*

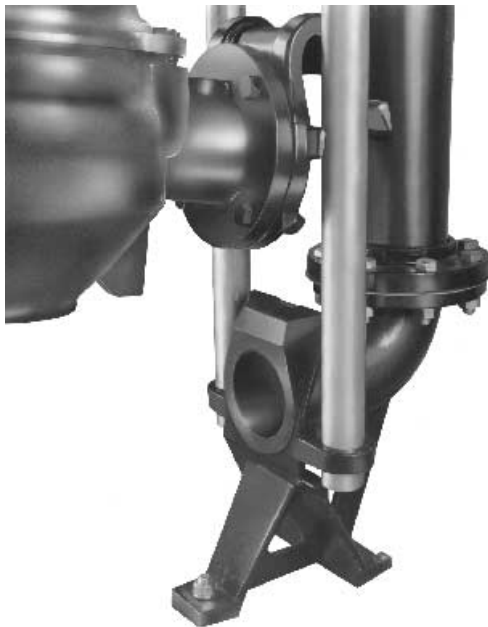
### 2.4. Pump Connection

A submersible pump, when installed submerged, is connected only to the discharge pipe. For fixed installations a self-connecting baseplate arrangement is normally used.

#### Submersible Baseplate

The concept of the submersible baseplate has been developed over the years for use with submersible pumps. The arrangement allows for the pump to be lowered into the pump well and firmly connect to the discharge pipework without the need of the operating personnel to enter the well. Likewise the pump can safely be hoisted from the well for service. The system includes rails or pipes that guide the pump down onto the baseplate. A special flange, or guide shoe, on the pump discharge mates with the joining surfaces of the baseplate for a firm connection. Well-designed systems are made to precision and have machined surfaces and rubber seal rings for a

Fig. 44



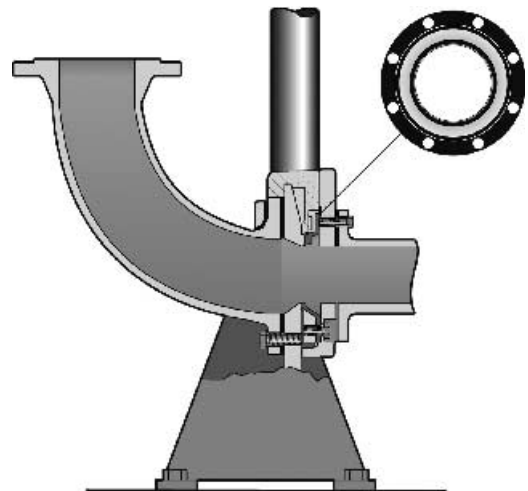
*A GRUNDFOS submersible baseplate. When seated, the weight of the pump keeps it firmly in place. Precision-machined connecting surfaces and a rubber disk seal ensures tightness. Clearance between the guide shoe and the rails ensures unobstructed hoisting of the pump even in fouled conditions*

sturdy and tight connection. The pump is kept in place by its own weight. Figure 44 shows a submersible pump baseplate and guide rails.

Figure 45 shows a flexible seal designed in a way that the seal action is further enhanced by pump pressure, ensuring a tight connection at all times.

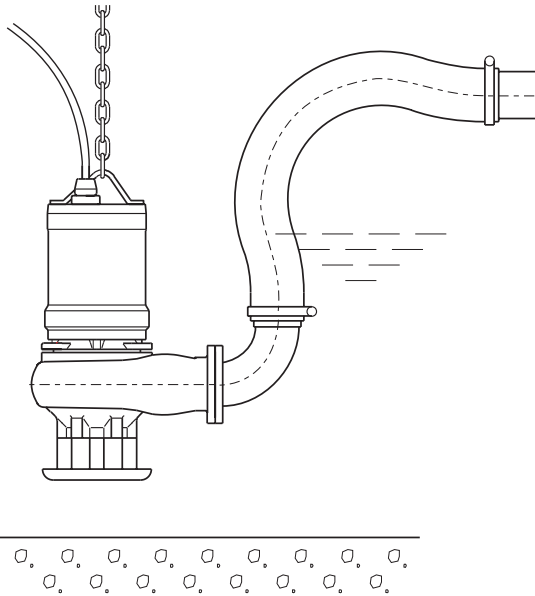
Some pump manufacturers offer conversion kits for the connection of pumps to older baseplates or as replacement pumps to some other manufacturer's baseplate. Thus the upgrading or conversion of existing pumping stations may be done with a minimum of work and costs.

Fig. 45



*Flexible seal between pump pressure flange and connector. The seal is designed in a way that the seal action is further enhanced by pump pressure, ensuring a tight connection at all times.*

Fig. 46



*Submersible pump on stand with hose connection. This installation version is used for temporary or shifting installations.*

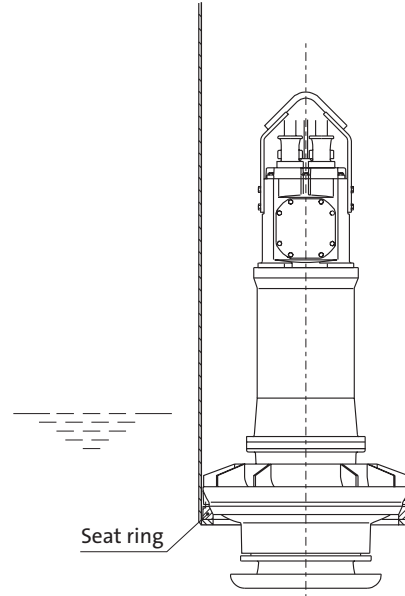
### Hose Connection

Figure 46 shows a submersible pump installation with hose connection. It may be used for temporary installations or in applications where the pump is shifted around the wet well for sludge pumping.

### Column Installation

The concept of column installation of submersible pumps has been developed during the past few years. The pump is lowered into a vertical pipe or column, where the circular pump casing fits onto a seat ring installed at the lower end of the column, please refer to Figure 47. The pump stays in place by its own weight and from the pressure force from the pumping action. The pump casing is special-designed for the installation, and is fitted with trailing vanes. The seat ring is conical, ensuring a tight connection between pump and column. The tight connection and dowels prevent the pump from spinning loose at start-up

Fig. 47



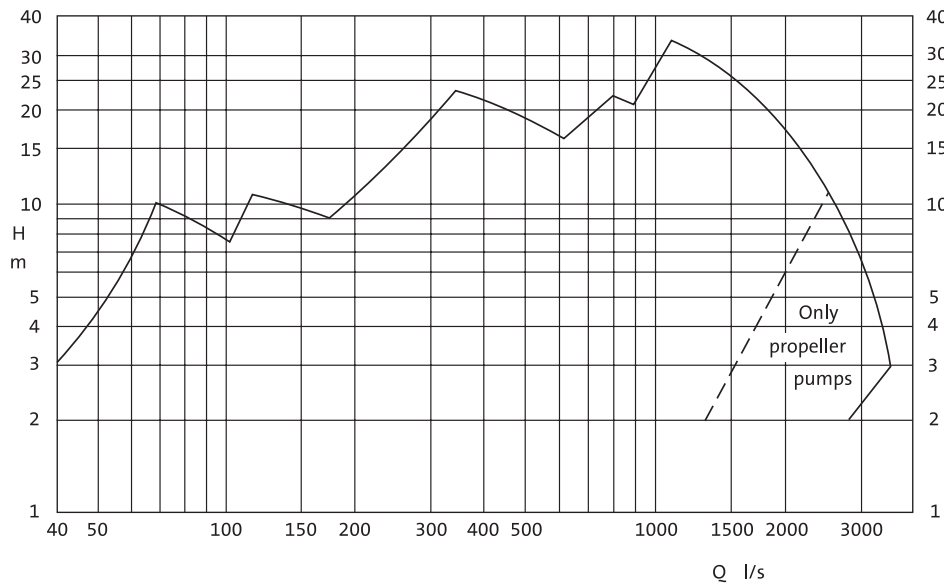
*Pump column installation. Pump rests on conical seat ring installed at the bottom of the column*

Column installation is ideal for submersible propeller pumps, but also for sewage pumps intended for large flows and low to moderate heads. Figure 48 shows the Q/H area on which column-installed Grundfos pumps are available. For this range column installation is likely to lead to lower investment costs, but each project should be assessed individually. Column-installed pumps have the same efficiency as pumps intended for other installation modes, but the pump curves will differ slightly because of the open pump casing. Column installation is very suitable for return sludge pumping in sewage treatment plants. The column pipe may be made of stainless steel or hot dip galvanized steel.

For seawater installations a column made of stainless steel may create a strong galvanic element between pump and column, leading to pump corrosion. Especially galvanized pump parts will rapidly corrode from the galvanic action of the large cathode area of the column surrounding



Fig. 48



GRUNDFOS column-installed pump flow and head (Q/H) range.

the pump. A lifting chain left in place, for instance, will have to be made of stainless steel. The cast iron pump should be protected by sacrificing anodes that are replaced at regular intervals. Painting the column with a paint layer of at least 200 µm thickness prevents the cathode surface from forming and thus pump corrosion.

## 2.5 Construction Materials, Corrosion and Wear

### 2.5.1 Corrosion Resistance

Cast iron is the main construction material in submersible sewage pumps, with fasteners and hardware made of stainless steel. The pump shaft is either made entirely of stainless steel or protected against contact with the pumped media. Where the pump or baseplate includes fabricated steel parts, these are hot-dip galvanized. These materials will last for decades in regular sewerage duty.

In cases, where the pumped liquid contains industrial effluents, the corrosion resistance of cast iron may not be sufficient, especially for parts subject to fast flow velocities, such as impellers and pump casings, which will be subjected to erosion corro-

sion. In these applications the natural corrosion layer, providing the underlying material with natural protection becomes scrubbed away, leading to rapid corrosion. The use of stainless materials for these vulnerable parts may be warranted.

Corrosion in seawater is dependent on a number of factors, such as salinity, oxygen content, pollution and temperature, and the right material selection must be considered for each case. Sacrificial zinc anodes may offer protection against corrosion in certain cases.

The supply cable sheath material must be able to withstand oils and other pollutants present in sewage. Other rubber parts, such as O-rings are usually made of Nitrile or Neoprene for resistance against sewage, oil and chemicals.

Submersible pumps are also available made entirely of stainless steel for use in highly corrosive liquids, such as process industry effluents. Stainless steel submersible pumps are 3...4 times as expensive as pumps made of regular materials. In difficult applications the pump manufacturer may not be able to guarantee the corrosion properties for a specific case, but will cooperate with the client to find the right solution for the case.

### 2.5.2 Wear Resistance

The sand content in sewage is on average between 0,002 and 0,003 % (in volume). The content may periodically, e.g. during heavy rainfall and snow melting be much higher in areas with combined sewage and stormwater drain systems. Regular cast iron will last in most applications for years, but special material may have to be considered for highly abrasive effluents, such as sewage treatment plant sand trap pumping.

### 2.5.3 Abrasive Liquids

Pump performance in an abrasive liquid is strongly dependent on the content of abrasives in the liquid. The standard abrasive is common quartz or silicon sand, to which the following can be applied directly.

The sand content is either expressed as volume or weight content, which are related as follows:

$$p_m \approx 3 \cdot p_v \quad (15)$$

where  $p_m$  is weight content and  $p_v$  is volume content in %. Thus  $p_v = 5\%$  equals  $p_m = 15\%$ .

With increased sand content the density of the liquid/sand mixture increases. Since required pump power is directly related to the density of the pumped liquid, required power will have to be checked separately in each case to ensure pump performance, whenever liquids with high sand content are being pumped. For sand trap pumps in sewage treatment plants, a power reserve of 30% has proven adequate.

The density of a mixture water and sand can be written

$$\rho = 1 + 0,007p_m \quad (16)$$

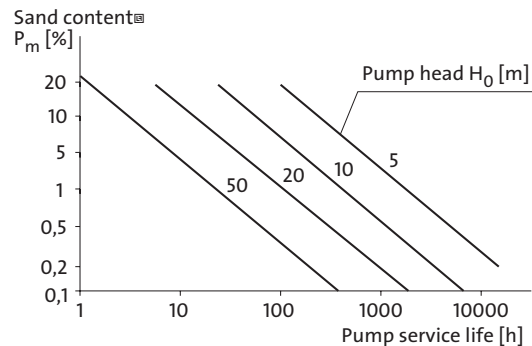
where  $p_m$  is expressed in %.

Thus, if  $p_m = 15\%$ ,  $\rho = 1,1 \text{ kg/l}$ .

The following factors affect the wear of a pump:

- sand content
- sand quality
- pump material
- pump head
- type of impeller

Fig. 49



*Pump wear rate as a function of sand content and pump head.  $H_0$  is pump head at  $Q=0$ . Wear rate is expressed as expected service life of a cast iron impeller and is strongly dependent of sand content and pump head. The graph is based on experiments, and can be used generally.*

Figure 49 is a diagram showing the relations between the pump wear rate and the sand content and pump head. High sand contents in the liquid will have a dramatic effect on pump service life. The effect of the sand content is exacerbated by high pump head.

Pump wear can be minimized using suitable wear-resistant materials and through appropriate design. For best results, materials with a hardness over 500 HB should be used. The difficult machineability of hard materials, such as Nihard and some alloyed irons, may require special impeller and pump casing designs where machining is minimized.

The use of submersible pumps in abrasive environments must be considered separately on a case by case basis and using sound engineering judgment.

## 3 Pump Performance

Pump performance is the result of the interaction between pump and rising main or pressure pipeline. An introduction to pump selection and the calculation of rising main resistance characteristics are presented.

### 3.1 Pump Head

#### 3.1.1 Submersible Pumps

In the following the concept of head is applied to submersible pumps. For practical reasons the pressure in the pump well, or lower well, is assumed to be equal to the pressure prevailing in the receiving, or upper container. Should these containers be under different pressure, the pressure difference would have to be taken into account. The difference in atmospheric pressure can also be disregarded in all practical installations, since the difference in atmospheric pressure between a receiving container situated, for instance, 100 m above the pump well is only 0,001 bar or 0,01 m of water.

Figure 50 shows how the head is defined in a submersible pump installation. The following units are used:

$H$  = pump total head (m)

$H_{st}$  = pump static head (m)

$H_d$  = pump dynamic head (m)

$H_{geod}$  = geodetic head (m)

$H_j$  = pipeline losses (m)

$p_L$  = atmospheric pressure in pump well

$p_U$  = atmospheric pressures in upper container

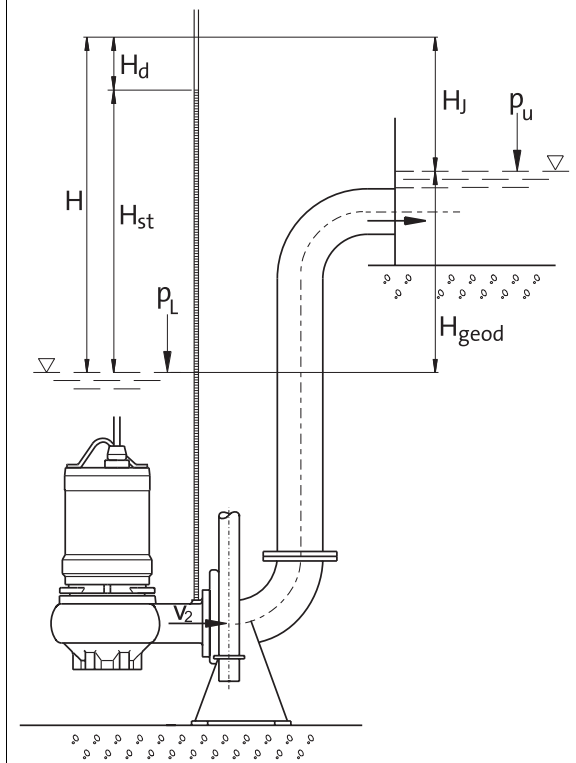
$v_2$  = flow velocity at outlet (m/s)

$g$  = acceleration of gravity (9,81 m/s<sup>2</sup>)

If an observation pipe is installed at the pump outlet flange, the pumped liquid will rise in it to a height  $H_{st}$  from the well level. This height represents the pump static head. In addition, the liquid has a velocity  $v_2$  at the pump discharge, which can be converted to pressure or dynamic head  $H_d$  with the following equation:

$$H_d = \frac{v_2^2}{2g} \quad (17)$$

Fig. 50



Head components in submersible pump installations.

The sum of the static head and the dynamic head is the pump total head, thus

$$H = H_{st} + H_d \quad (18)$$

According to international agreement (Standard ISO 2548), the total head  $H$  according to equation 18 is used when plotting characteristic curves for submersible pumps.

The total head  $H$  is thus available to pump the liquid through the rising main. The pressure or head required to pump a given flow through a pipeline is made up by the geodetic head and the flow losses. Thus can be written:

$$H = H_{geod} + H_j \quad (19)$$

The geodetic head  $H_{geod}$  is the actual physical difference in height between the liquid levels in the pump well and the receiving container. Pipeline flow losses consist of pipe friction losses, local losses from various fittings in the pipeline

(elbows, valves, etc.) and the outlet loss at the point of discharge.

Losses due to liquid flow in the well to the pump are considered as pump losses in submersible pump installations. If a suction pipe is installed before the pump, it will have to be taken into account when calculating pipeline losses.

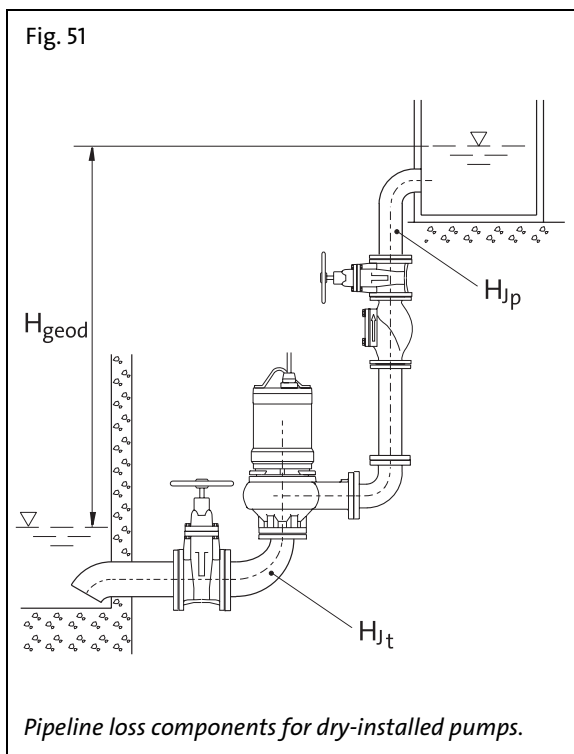
### 3.1.2 Dry-installed Pumps

When calculating heads of dry-installed pumps, the situation before the pump will also have to be considered. Figure 51 illustrates the situation.

In this case it is assumed that the suction well and the receiving container are open to the atmosphere and that the pressure at the liquid surfaces is constant. Thus the pump head is the sum of the geodetic head and the flow losses in the suction and pressure pipelines. Thus

$$H = H_{\text{geod}} + H_{\text{Jt}} + H_{\text{Jp}} \quad (20)$$

where  $H_{\text{Jt}}$  represents flow losses in the suction pipeline and  $H_{\text{Jp}}$  flow losses in the pressure pipeline.

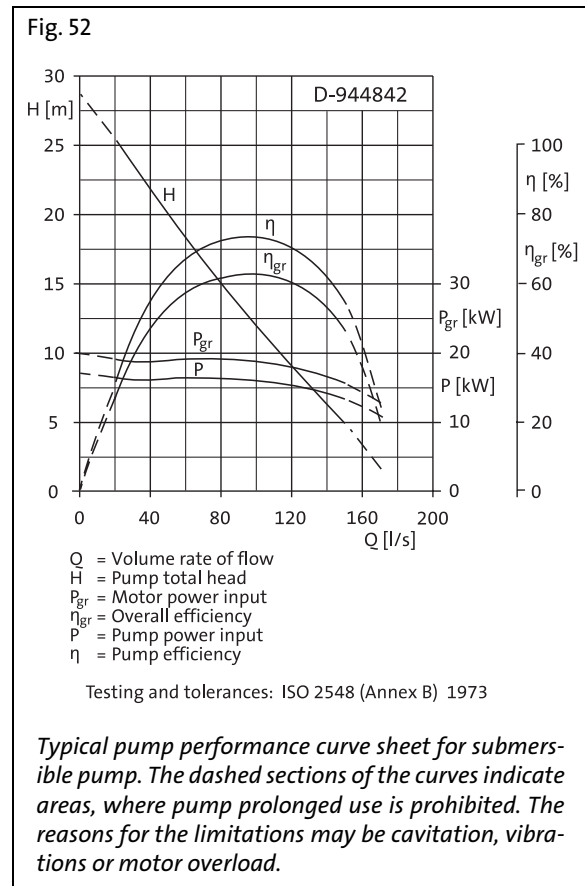


## 3.2 Pump Performance Curves

Centrifugal pump characteristics are normally presented as a set of curves, where the data has been established through the testing of the pumps or assessed by the manufacturer for e.g. a special impeller diameter. For submersible pumps the following important information is normally plotted as curves against the flow rate  $Q$ :

- $H$  head curve
- $\eta$  efficiency curve(s)
- $P$  power curves

Figure 52 shows a typical pump performance curve sheet with information important for the user.



### 3.2.1 H Curve

The head or  $H$  curve gives the pump total head as a function of the flow  $Q$ . The curve may contain additional information on pump usage, such as limits due to cavitation, vibration or motor overload.

## 3 Pump Performance

### 3.2.2 Efficiency Curves

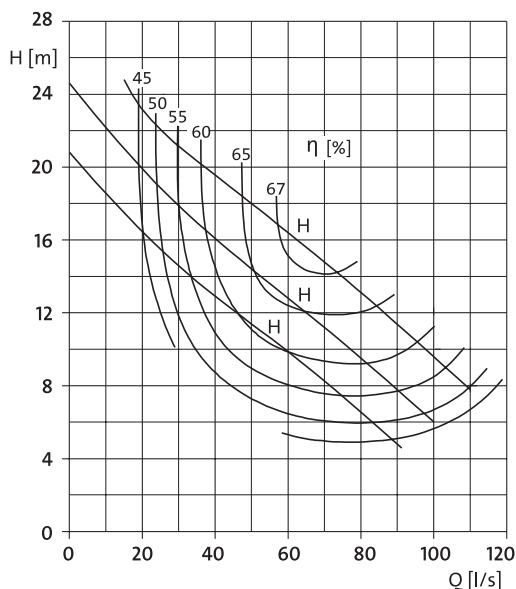
Pump efficiency  $\eta$  is also a function of the flow rate  $Q$ . The efficiency can be indicated as a ratio or percentage. For submersible pumps both the pump efficiency  $\eta$  and the overall efficiency  $\eta_{gr}$  are defined, where  $\eta_{gr}$  includes motor losses. It is important to distinguish between these definitions for efficiency, especially when comparing pump performance. The losses leading to the pump efficiency are discussed in Section 1 of this book. Thus it can be written:

$$\eta_{gr} = \eta_{mot} \cdot \eta \quad (21)$$

where  $\eta_{mot}$  is motor efficiency.

The efficiency can also be marked on the head curve, with numbers indicating different efficiency values. If several head curves for various impeller diameters are plotted in the same graph, these markings can be connected to form isograms, or operating areas with the same efficiency. The pump performance diagram will then assume its typical appearance as shown in Figure 53.

Fig. 53



*Pump performance sheet with curves for different impeller diameters, where the efficiency is indicated as points directly on the head curves. Impeller selection is facilitated by connecting the points with the same efficiency.*

### 3.2.3 Power Curves

The pump required power is also a function of the flow rate  $Q$ . Figure 52 contains both the pump power curve and the motor power curve. The motor power is the electric power drawn by the motor and measured at the cable junction box at the motor. According to international standards on pump testing the pump power is designated  $P$  and the power absorbed by the motor  $P_{gr}$ . The required power can also be calculated using the equation

$$P = \frac{\rho Q g H}{\eta} \quad (22)$$

where

$P$  = power (W)

$\rho$  = liquid density ( $\text{kg/m}^3$ )

$Q$  = volume flow ( $\text{m}^3/\text{s}$ )

$g$  = acceleration of gravity ( $9,81 \text{ m/s}^2$ )

$H$  = pump head (m)

$\eta$  = efficiency

### 3.2.4 NPSH Curve

Since NPSH calculations are performed only for dry-installed pumps, the NPSH curve is not usually included on submersible pump data sheets. It will be provided by the manufacturer on request if cavitation is feared in a dry installation, or if otherwise required by the client.

Results from tests performed with clean water are applicable as such on normal municipal sewage and most industrial effluents, since the low solids content in sewage (less than 0,05%) does not significantly affect pump performance.

### 3.3 Pipe Losses and Rising Main Characteristic Curves

In the following the theory for calculation of flow losses in pipelines is presented. Practical calculations can be made with the help of the detailed instructions with calculation diagrams and nomograms presented in Appendix A, or with a computer program.

Flow velocities used in sewage pumping are high enough to ensure uniform turbulent flow in the piping. Flow losses therefore increase with the square of the flow velocity. The flow loss of a rising main is the sum of the friction loss of the pipeline constituent parts and the local losses from the various components and fittings.

#### 3.3.1 Friction Losses

Pipe friction losses depend on the following factors :

- pipe length
- pipe internal diameter
- flow velocity
- pipe wall relative roughness
- fluid kinematic viscosity.

A dimensionless relation, Reynold's number is introduced:

$$Re = \frac{vD}{\nu} \quad (23)$$

where

Re = Reynold's number

v = flow velocity (m/s)

D = pipe internal diameter (m)

$\nu$  = kinematic viscosity (m<sup>2</sup>/s)

The kinematic viscosity for water is dependent on temperature:

t °C	0	20	40	60	100
$\nu \cdot 10^{-6} \text{ m}^2/\text{s}$	1,78	1,00	0,66	0,48	0,30

The equation for pipeline losses can be written:

$$H_{Jp} = \lambda \frac{l v^2}{D^5 g} \quad (24)$$

where

$H_{Jp}$  = pipeline loss (m)

$\lambda$  = friction factor

l = pipeline length (m)

v = flow velocity (m/s)

g = acceleration of gravity (9,81 m/s<sup>2</sup>)

D = pipeline internal diameter (m)

Obtaining the friction factor  $\lambda$  from the diagram in Figure 54, equation 24 can be solved. Surface roughness values (mm) presented in the following table can be used:

Pipe material	k new	k old
Plastic	0,01	0,25
Drawn steel	0,05	1,0
Welded steel	0,10	1,0
Drawn stainless steel	0,05	0,25
Welded stainless steel	0,1	0,25
Cast iron	0,25	1,0
Bituminized cast iron	0,12	
Asbestos cement	0,025	0,25
Concrete	0,3...2,0	

The surface of an old pipe material becomes rougher from erosion. Corrosion and sediment layers forming on the pipe surface may decrease the pipe diameter, also leading to higher flow losses.

The effect of pipe diameter change can be calculated with the following relation:

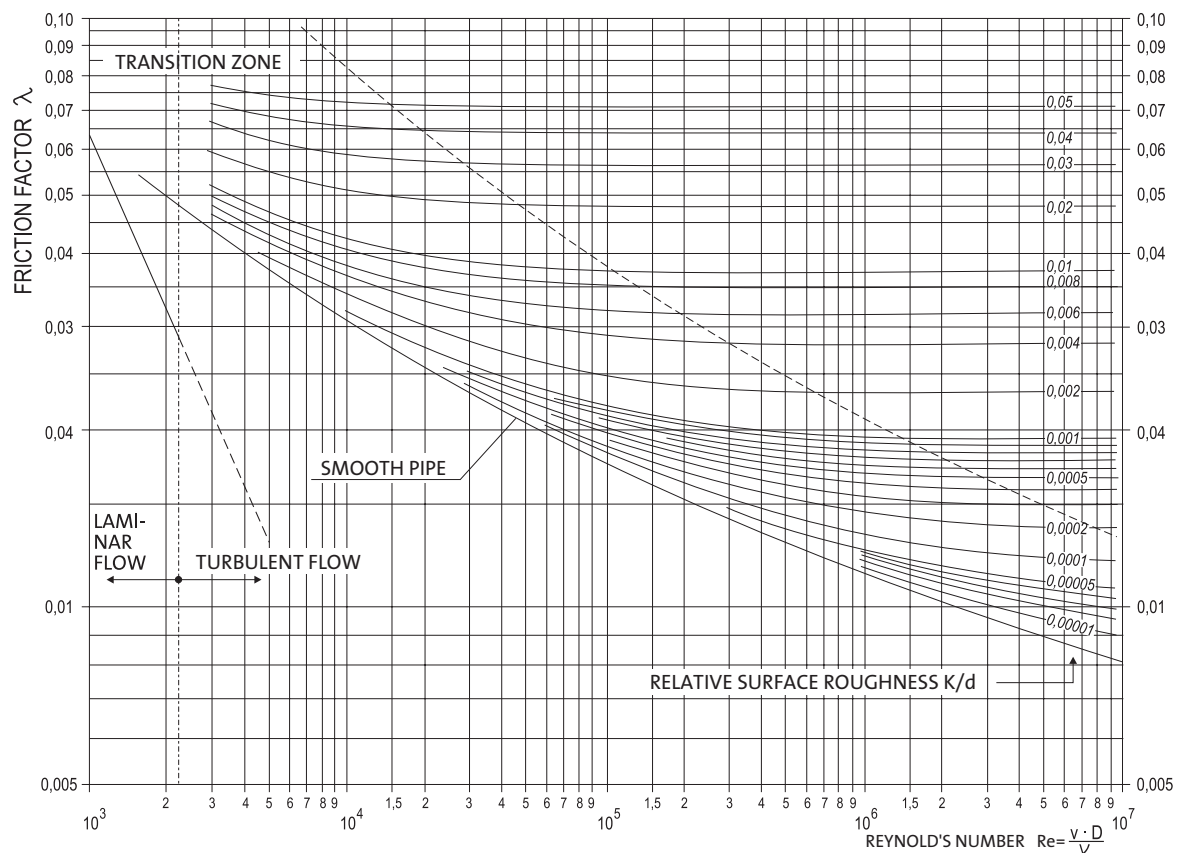
$$H'_{Jp} \approx H_{Jp} \left( \frac{D'}{D} \right)^5 \quad (25)$$

Thus an increase of pipe diameter from, for instance, 100 mm to 108 mm decreases the flow loss by 30%.

Equation 25 is sufficiently accurate for practical purposes when comparing flow losses in rising mains of different diameter, particularly since accurate surface roughness values are seldom available.

### 3 Pump Performance

Fig. 54



Moody diagram for establishing the friction factor  $\lambda$ . The value of  $\lambda$  is obtained using Reynold's number and the relative roughness number  $k/D$  as parameters, where  $D$  is pipe internal diameter in mm and  $k$  equivalent surface roughness in mm. Completely turbulent flow can be assumed in wastewater applications.

Rising main flow losses are frequently calculated with the help of proprietary computer programs, also available from some pump manufacturers. These programs may also suggest some pump selections from the manufacturer's range to best suit the purpose. It is advisable to take a cautious view on the pump selection suggested by a program only, and always contact the pump manufacturer in dubious cases.

The rising main is sometimes divided into two separate parallel pipelines. They have the same length but may have different diameters or be made of different materials. The distribution of flow between the two lines and the ensuing losses in these lines can be difficult to determine. Grundfos has developed a method for this, where the two lines are substituted with a single virtual rising main. An equivalent diameter is determined for this so that the resulting flow losses are

equal to the resultant losses of the two true rising mains.

The equivalent diameter is calculated with the following equations:

A. Both rising mains have the same diameter

$$D_e = 1,3 \cdot D \quad (26)$$

where  $D$  = diameter of the two parallel rising mains

B. The rising mains have different diameters

$$D = (D_1^{2,65} + D_2^{2,65})^{0,3774} \quad (27)$$

where  $D_1$  and  $D_2$  are the different diameters of the parallel rising mains.



The volume rates of flow for the two rising mains are calculated with the following equations:

A. Both rising mains have the same diameter

$$Q_1 = \frac{Q}{2} \quad (28)$$

B. The rising mains have different diameters

$$Q_1 = \left( \frac{D_1}{D_e} \right)^{2,65} \cdot Q \quad (29)$$

$$Q_2 = Q - Q_1 \quad (30)$$

The equations above are valid for turbulent flow, which is normal for water pumping. The equations require that both pipelines have the same surface roughness.

### 3.3.2 Local Losses

Changes in pipeline internal diameter and shape, bends, valves, joints, etc. as included in the rising main cause additional losses that comprise both a friction and turbulence component. The following equation is used to calculate the losses:

$$H_{Jn} = \zeta \frac{v^2}{2g} \quad (31)$$

where

$H_{Jn}$  = local loss (m)

$\zeta$  = local resistance factor

$v$  = flow velocity (m/s)

$g$  = acceleration of gravity (9,81 m/s<sup>2</sup>)

Local resistance factors for different pipeline elements and fittings are presented in Appendix A. The friction loss of these are not included in the local resistance factor, but is calculated as part of the rising main friction loss by including their length and internal diameter when calculating pipeline length.

Pipe expansion discontinuity loss can be calculated using the Borda equation:

$$H_{Jn} = \frac{(v_1 - v_2)^2}{2g} \quad (32)$$

where

$H_{Jn}$  = local loss (m)

$v_1$  = flow velocity 1 (m/s)

$v_2$  = flow velocity 2 (m/s)

$g$  = acceleration of gravity (9,81 m/s<sup>2</sup>)

If the pipe expansion is designed with a conical section with an expansion angle of 10°, the loss is reduced to 40% of the value calculated with equation 32. This fact is important when expanding the pipe section right after the pump pressure flange, where the flow velocity can be quite high. By designing the transition with a 10° gradual expansion joint, energy can be saved. In a contracting pipe section the losses are much smaller, and the conical section can be built much shorter.

Losses in a section with velocity reduction are generally much greater than in section with increasing velocity.

The final component of pipeline loss is the outlet loss at the end of the rising main. If no expansion is provided, the loss equals the velocity head or  $v^2/2g$ .

Loss coefficients for different valves are provided by the manufacturers. Guide values for the most common valves used in sewage installations are presented in Appendix A.

### 3.3.3 Rising Main Characteristic Curve

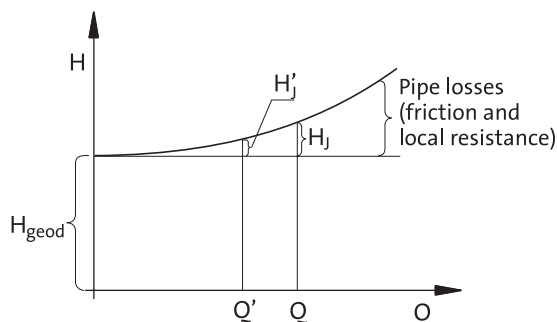
In sewage installations the pump sump and the delivery well are open to the atmosphere, and the rising main characteristic curve will contain the geodetic head and the flow losses only. Figure 55 shows the general shape of the characteristic resistance curve for a pipeline. Since the flow is turbulent at the flow velocities in consideration, it can be assumed that the flow loss varies in proportion to the square of the flow rate. Thus, if the flow loss at one flow rate is calculated with the method described above, the other points of the curve are obtained sufficiently exactly with the following equation:

$$H'_J = H_J \left( \frac{Q'}{Q} \right)^2 \quad (33)$$



### 3 Pump Performance

Fig. 55



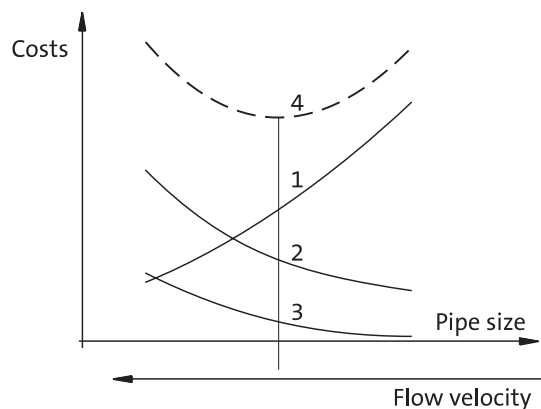
Characteristic resistance curve for a pipeline. Pipe losses ( $H_j$ ) are plotted against flow rate ( $Q$ ) and added to the geodetic head, which is constant.

### 3.4 Rising Main Size

Rising main size is selected based on the following factors:

- economy
- required internal diameter for the application
- required smallest flow velocity for the application.

Fig. 56



1. Procurement cost of rising main
2. Procurement cost of pumping station
3. Energy costs
4. Total costs

The relation of key cost components for a pumping installation as related to rising main size. With costs having opposite relations to pipeline size and flow velocity, an optimum can be found.

### 3.4.1 Economy

The economy of an installation is made up by both procurement costs and operational costs during its lifetime. A number of installation and operational costs are directly dependent on rising main size, and will react to changes in pipeline size as follows:

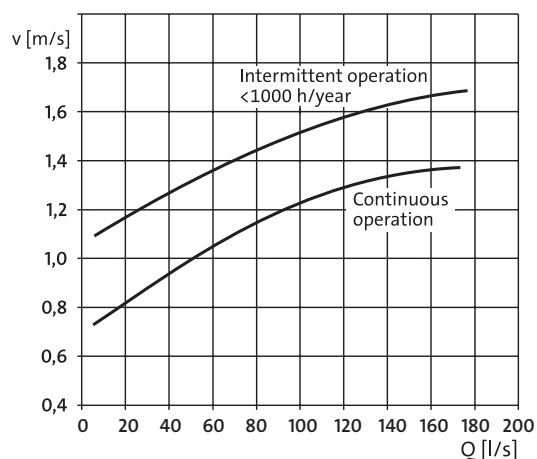
With decreased pipeline diameter

- Piping and pipework component procurement prices will decrease.
- Pumping station procurement cost will increase due to increased flow losses with consequent requirement for larger pumps and control equipment. Costs for increased electrical supply systems, such as substations may increase significantly.
- Operating costs will increase due to higher energy costs because of pipeline losses.

With different costs having opposite relations to rising main size, an optimal pipeline size may be found. Figure 56 shows the relation. The selection of an optimal pipeline diameter may be based on Figure 57, which shows the optimum flow velocity for different installations, and is based on several studies.

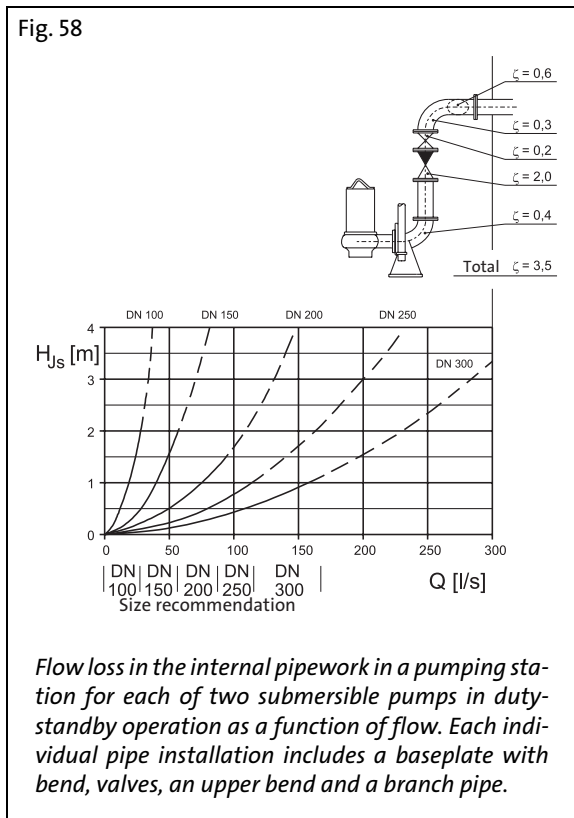
Where possible, a more detailed study can, and should, be conducted.

Fig. 57



Guideline values for economically optimal flow velocities for submersible pump installations. The figure is based on a study of submersible pump installations using geodetic head, pipeline length, yearly operation hours and energy cost as parameters.

Pumping station internal piping should be selected so as to minimize component costs without unduly increasing the flow losses in the station. Figure 58 shows the flow loss in the internal pipework in a pumping station with two pumps in duty-standby operation as well as the economical pipe dimensions, based on several studies.



### 3.4.2 Free Passage for Solids

For untreated municipal sewage the smallest allowable free passage of the rising main is generally 100 mm in order to allow passage of solids without clogging. In pumping stations with small flows the internal pipework may have a free passage of 80 mm, especially when the pump free passage is also 80 mm.

### 3.4.3 Avoiding Settling of Solids and Sludge

If the flow velocity in a rising main is too low, sand or sludge may have time to settle, which increases the risk of clogging. Settled sludge may harden and form a crust on the pipeline wall that permanently decreases the diameter, leading to increased flow losses. Larger sludge clots moving with the flow may block bends or other fittings in the rising main.

For municipal sewage a minimum flow velocity of 0,7 m/s is recommended. Where only domestic sewage is pumped, the minimum flow velocity may be as low as 0,5 m/s, but if sand is found in the sewage, this lower value is not endorsed. In installations with varying flow, e.g. where frequency converters are used, the flow velocity may temporarily be lower.

Where settling is known to occur, flushing out of the system with all pumps running simultaneously at intervals is recommended. The shape of the pipeline is also important, and sedimentation is likely to occur in rising mains having a pronounced low, such as pipelines laid underneath waterways, e.g. a river. In these cases it is recommended to select a higher flow velocity.

### 3.4.4. Water Hammer

Oscillating pressure waves are generated in a liquid being pumped through a pipeline during starting and stopping of the pumps. This phenomenon is called water hammering, and, if severe, may lead to pipeline and equipment damage. The severity of the phenomenon is dependent on a number of variables, such as change of velocity during the reflection cycle, pipe material characteristics as well as liquid characteristics.

When the liquid is accelerated or decelerated, a transitory pressure wave oscillates back and forth until dampened. The oscillating frequency can be calculated with the following equation:

$$\mu = \frac{2L}{a} \quad (34)$$

where

$\mu$  = reflection cycle duration, during which the pressure wave oscillates back and forth once (s)

$L$  = pipeline length (m)

$a$  = pressure wave velocity (m/s)

### 3 Pump Performance

Pressure wave velocities in clean water in pipes of different materials can be obtained from the following table:

Pipe material	Velocity (m/s)
Steel	900...1300
Cast iron	1000...1200
Reinforced concrete	1000...1200
Plastic	300...500

Sewage and sludge often contain insoluble air or gas, which has a significant effect on the pressure wave velocity, as can be seen from the following table, where the pressure wave velocity is expressed as a function of the quantity of insoluble air in the liquid:

Head = 15 m	
Quantity of insoluble air as volumetric ratio	Velocity ratio of pressure wave
0	1,0
10 <sup>-6</sup>	1,0
10 <sup>-5</sup>	0,96
10 <sup>-4</sup>	0,73
10 <sup>-3</sup>	0,32
10 <sup>-2</sup>	0,11

Dissolved air has no practical effect on pressure wave velocity.

The pressure transient resulting from a change in the flow velocity within a reflection cycle can be calculated with the following equation:

$$\Delta h = \pm \frac{a \cdot \Delta v}{g} \quad (35)$$

where

$\Delta h$  = pressure change (m)

$a$  = pressure wave velocity (m/s)

$\Delta v$  = flow velocity change (m/s) during one reflection cycle

$g$  = acceleration of gravity (9,81 m/s<sup>2</sup>)

Since it is difficult to establish the change in flow velocity when the pump starts or stops, exact calculations of the pressure transient cannot be easily performed. Only if, for example, a valve is closed within the reflection cycle, and the flow velocity change  $\Delta v$  is equal to the flow velocity  $v$ , can the pressure change be accurately calculated. Because the pressure fluctuates symmetrically, the pressure may fall below vapour pressure, causing cavitation with resulting high pressure transients and noise. Potential locations for these are pump, valve and pipeline high point. The high grade vacuum may also cause the pipeline to collapse.

In sewage pumping the water hammer pressures induced during pump stop are higher than those induced at pump starting. In theoretical computations the objective is to calculate the water retardation immediately after pump stop and the pressure transient induce at that instance. The most uncertain and significant factors to find out are the pump flow resistance and lowest pressure generated in the pump, after the supply current has been cut off. This information is not readily available from the pump manufacturers.

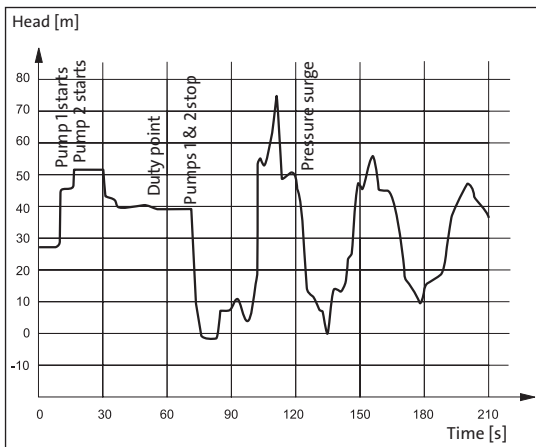
Another uncertain factor is the air or gas content in the water or pipeline. The solution here is to analyze for different concentrations in order to find out the effect of the gas content.

Figure 59 shows the outcome when observing the water hammer phenomenon in a twin pump installation. The following is noted:

- Measured reflection cycle duration is 45 seconds. Theoretical calculations for a fluid without gas or air indicated a duration of only 12...20 seconds. The difference between the two values shows that gas is present in the water.
- Immediately after pump stop the pressure in the pipeline falls to vacuum. Because the pressure was measured at the discharge flange, the pressure inside the pump must have been even lower. It is likely that the pressure inside the pump fell below cavitation pressure (-10 m).

Significant for the water hammer phenomenon is that it cannot be heard, since the pressure surge is fairly slow, but it can be observed with a pressure

Fig. 59



Rising main length 3000 m  
Pipe dia DN 400, PVC  
Pump combined flow 140 l/s  
Flow velocity 1,3 m/s  
Gedotic head 32,4 m  
Check valves DN200  
Pressure gauge at pump delivery flange

*Water hammer sequence pressure measurements as function of time.*

rising main will prevent the pressure from dropping inside the pump. Dimension of by-pass pipe should be selected one size smaller than pump pressure flange size.

- Using heavier pipe components that will withstand water hammer pressure. Vacuum transients may be more critical to the pipeline and equipment than pressure surges.

gauge. Noise will be emitted only in case of cavitation or if a valve closes rapidly.

Water hammer is not a common problem in sewage installations. Theoretical description of the problem is difficult because of the large number of unknown entities.

### 3.4.5 Avoiding Water Hammer

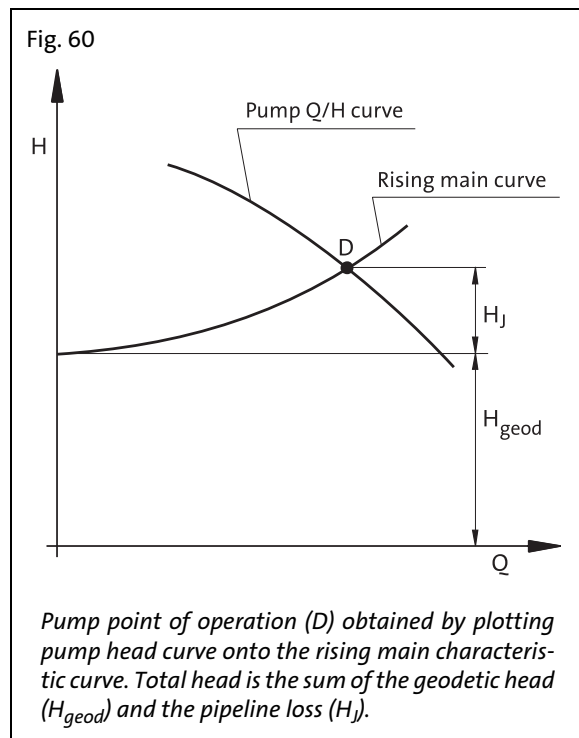
If water hammer occurs in sewage installations, the situation can be alleviated with one or several of the following measures:

- Preventing of simultaneous stopping of two or more pumps.
- Installing automatic valves with closing times of 20...30 seconds instead of regular check valves. Pump stops after valve has closed.
- Stopping pumps slowly with frequency control.
- Using soft start equipment also for stopping of the pumps. Complete control of stopping sequence not always possible.
- Installing automatic air relief valves at points where negative pressure occurs.
- In cases of cavitation in pump during the stopping cycle, the installation of a by-pass suction line with check valve from the wet well to the

### 3.5 Pump Duty Point

#### 3.5.1 Single Pump Operation

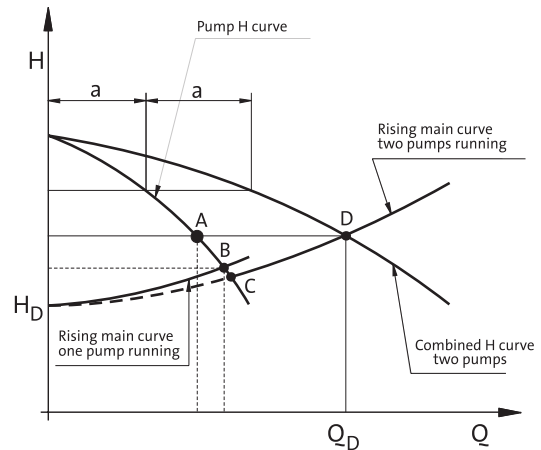
By adding the geodetic head ( $H_{\text{geod}}$ ) and the piping loss ( $H_J$ ), the rising main head is obtained. The geodetic head is a constant independent of the flow, whereas the losses increase with approximately the square of the flow rate  $Q$  (see Figure 55). If a pump head curve drawn to the same scale is overlaid or plotted on the rising main characteristic curve, the point of operation will be the intersection of the curves. At this point the pump head equals the head required by the rising main. The pump flow rate  $Q$  can then be read directly off the diagram as illustrated in Figure 60.



#### 3.5.2 Parallel Operation, Identical Pumps

Parallel operation is the situation where the combined flow of two or more pumps is directed into the same rising main. The shape of the characteristic curve for the rising main will change slightly with the different numbers of pumps operating, since each pump has its own discharge line up to the common point, and the rising main constitution will therefore vary.

Fig. 61



Operating points for two identical pumps operating singly (B) or in parallel (D). Since the pipeline resistance increases with the flow rate, the combined output of two pumps ( $Q_D$ ) is always less than two times the output of a single pump. For practical purposes a single pump can be assumed to have the operating point C.

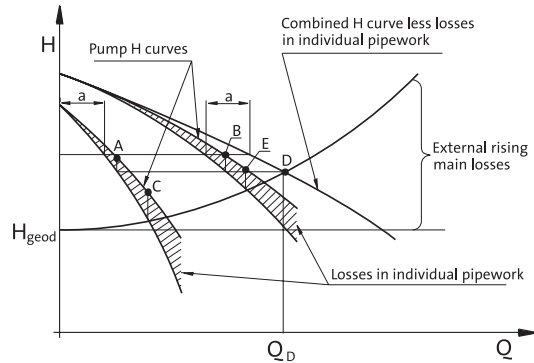
Assuming two identical pumps with identical separate pipework combined by a branch or header to the rising main operating in parallel, we obtain the characteristic rising main curve as illustrated in Figure 61. The duty point for both pumps is obtained by plotting the sum of two pump head curves at constant head onto the rising main characteristic curve for two pumps.

#### 3.5.3 Parallel Operation, Different Pumps

When calculating the point of operation for two different pumps operating in parallel, different characteristics for the separate pipework up to the header should be assumed. The following method for obtaining the points of operation can be used.

The losses for each pump in their separate pipework before the common header are checked first. These can be plotted in the graph as reductions of the heads, reducing the pump curves. The combination of these reduced curves at constant head gives the combined head curve for the pumps. The intersection of this curve and the rising main characteristic curve is the combined point of operation. By drawing backwards from this point at

Fig. 62



Operating points for two different pumps discharging into a common rising main. Losses in the individual pipework are reduced from the pump head curves plotted to scale in the graph. The combined output curve is obtained using the reduced head curves, giving the combined operating point D. The individual operating points are A and B. For the pumps operating singly, the points of operation will be C and E, respectively.

constant head to the reduced pump curves, the individual pump operating points can be read at the original pump curves straight above these intersections. Likewise, the individual operation points with the pumps running singly are obtained by reading from the head curves above the intersection of the rising main curve and the reduced pump curves. The method is illustrated in Figure 62.

### 3.5.4 Serial Operation

It is possible to connect a number of pumps in series in order to increase head. The combined head is obtained by adding the individual heads at constant flow. The complexity of the arrangement makes it warranted only in rare instances, and it is nearly always advisable to use a larger pump from the manufacturer's range that can do the job alone.

Submersible pumps can be connected in series only if the boosting pumps are installed dry, thus making them different from the lead pump.

Another risk involving pumps connected in series is the possible failure of the lead pump, which may lead to cavitation in the booster pump due to

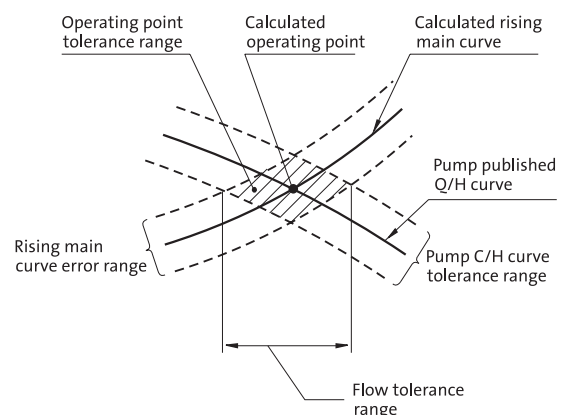
increased suction losses or loss of suction head. The designer should design the pumping plant so that serial connection of pumps can be avoided, and make certain that there are pumps available for the intended duty point.

### 3.5.5 True Duty Point

The true pump duty point will almost always differ from calculated. The reason for this is the inaccuracies in all numeric methods for calculation of rising main losses, as well as the tolerances allowed in the published pump performance curves. Furthermore, the properties of the pump will change with use due to wear, and corrosion or sedimentation will change the rising main with age. Figure 63 shows the relation between performance tolerances. Pump inherent performance tolerances are discussed in detail in Section 4 of this book.

If the duty point is located on the low flow segment of the pump Q/H curve, and the rising main characteristic curve is steep, the flow tolerance range can be very large in proportion to designed duty point. This fact should be taken into account when selecting the pump.

Fig. 63



Tolerance area for the duty point. The true duty point for an installation may lie within the shaded area limited by the allowed tolerance zones for pump head curve and rising main characteristic curve. Pump output Q may vary largely.



### 3.6 Sludge Pumping

Sludge of varying consistence is frequently being pumped by submersible pumps in sewage treatment plant duty. With increasing solids content in the sludge, the rising main flow losses will increase while pump performance decreases. When selecting a pump for sludge duty these two factors must be considered. The effect is illustrated in principle in Figure 64.

The situation is complicated by the fact that not enough is known about the behaviour of sludge in centrifugal pumps. Treatment plant sludge may have high gas content, either dissolved or entrained, and this will have a profound effect on centrifugal pumps. As a rule, sludge with high solids content also has a high content of gas, which will lower pump performance significantly. In extreme cases the pump will stop pumping when the separated gas accumulates in the impeller eye, preventing it from developing the necessary centrifugal force.

As a precaution when pumping dense sludge, the pump should be placed as low as possible, to ensure positive suction head. The use of long suction pipes should also be avoided, since the pressure drop in these is also increased by the solids content.

With a sludge solids content less than 1%, it is usually safe to assume that the rising main character-

istic curve is the same as for water. With greater solids contents the characteristic curve will be higher, but lack of data on the sludge makes the establishment of a correct curve difficult. Another practical problem is the fact that the solids content of the liquid in the pump and rising main can momentarily considerably exceed the mean or design value. In thicker sludges the pump motor cooling may become a problem, depending on cooling method.

Generally submersible sewage pumps are suitable for pumping of treatment plant sludges with a solids content of maximum 3%. These sludges include primary sludge, return sludge and excess sludge, whereas for denser sludges, such as thickened sludge and digested sludge, positive displacement pumps are preferred. For these thicker liquids the pumped volumes are relatively small.

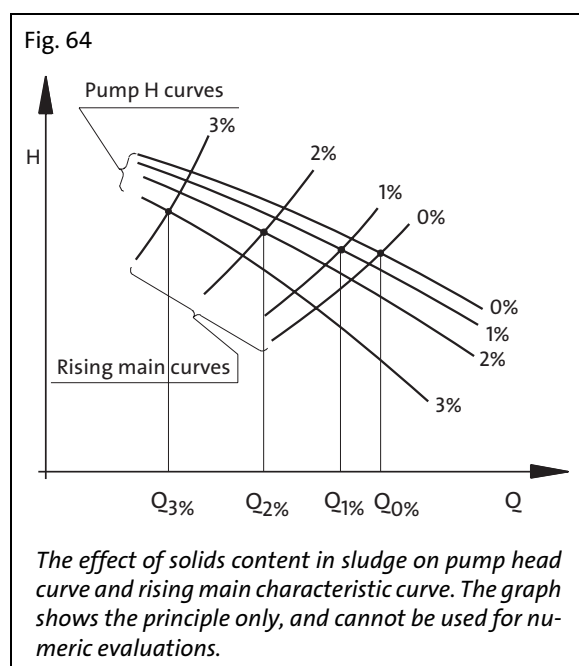
Propeller pumps are not recommended in sewage treatment plant duty because of the risk of clogging. For return sludge pumping a channel type pump in vertical column installation is a good solution.

### 3.7 Complex Rising Mains

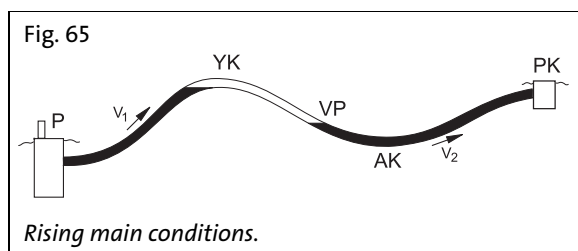
Long transfer sewer lines frequently have complex profiles, with low and high turning points. Air or gas trapped at the high turning points increases pump head, whereas the low turning point increases the risk of sedimentation. There are cases where a selected pump has proven to be inadequate, and cases of sedimentation are also known. Exact forecasting of sewer main performance is difficult because of the intermittent pumping action of the pumps. Water in the main may move as little as 100 metres during one pumping cycle, and air or gas in the pipeline will not be removed and the flow will not stabilize in that period.

#### 3.7.1 What Goes on in a Complex Rising Main?

In Figure 65 the section YK-VP contains air. As the pump starts, the liquid level VP begins to slowly rise and the air pressure in the section YK-VP increases and a flow develops from point VP and to point PK ( $v_2$ ). As the pump stops, the flow from VP to PK continues for some time, slowly decreasing. Because the duration of flow from VP to PK is







longer than pump running time, the maximum flow velocity  $v_2$  is slower than  $v_1$ . The low velocity of  $v_2$  and the rising section after point AK may increase the risk of sedimentation. The air or gas in the section YK-VP prevents the siphoning effect from forming, leading to increased geodetic head.

The exact location of point VP is difficult to estimate accurately. If the amount of air were constant, the location of point VP could be calculated as a function of time. In practice the amount of air in the pipe will change, and the location of point VP cannot be calculated. If YK is located lower than PK, the air could in theory be removed with an automatic air valve. If point YK is situated higher than PK, the air will flow back into the pipe after the pump has stopped. Automatic air valves are prone to clog up in sewage. A solution could be a hand-operated air valve that is opened at certain intervals according to information on air or gas accumulation gathered over time.

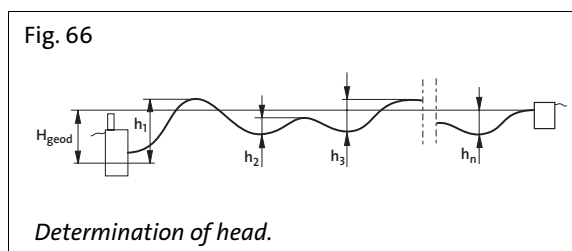
### 3.7.2 Determination of Head

For a rising main with a profile similar to Figure 66, the pump total head is difficult to estimate exactly. An estimation of magnitude can be made, however. The minimum Head ( $H_{\min}$ ) is determined with the rising main completely filled and the maximum head ( $H_{\max}$ ) as a situation with all downward sloping sections air or gas-filled. Thus

$H_{\min} = H_{\text{geod}} + \text{pipe flow friction losses for total length of rising main}$

$H_{\max} = h_1 + h_2 + h_3 + \dots + h_n + \text{pipe flow friction losses for total length of rising main}$

Real total head is a value between the maximum and minimum value. A useful estimation may be the mean value of  $H_{\max}$  and  $H_{\min}$ .



### 3.7.3 Pipe Size and Flow Velocity

As noted above, the air or gas collected in the rising main will even the flow velocity in the following section, causing lower flow velocity in the low points of the pipeline. This gives reason to choose a rising main of a dimension small enough to ensure that the flow velocity does not fall too low. Minimum pipe dimension is DN 100, however.

A smaller pipe has also a smaller volume, meaning that the water moves a longer distance with each pumping cycle, increasing the flow velocities at the low point of the pipeline. From an odour point of view a smaller pipe dimension is better, since the sewage stays a shorter time in the rising main. A higher flow velocity may also carry out some of the air in the pipe. In these cases the dimensioning flow velocity ( $v_1$ ) should be at least 0,8 m/s, in more difficult cases even higher.

### 3.7.4 Choice of Pump

In a complex rising main the true head may differ considerably from calculated. If the calculated duty point is situated near either end of the pump Q/H curve allowed section, this pump should not be considered. A pump with a Q/H curve passing above the calculated duty point should also be considered, since it offers security of choice and the flow velocity increases.

### 3.7.5 Confirming Measurements

Since true duty point may differ considerably in cases with a complex rising main, it may be useful to measure the volume rate of flow a few weeks after pumping station commissioning, using the volumetric method. Comparing measured values with calculated will show deviations and indicate true state of the rising main. The measurements can be repeated a few times during the first year of operation, since gas or air content in the rising main may change.

### 3 Pump Performance

Control measurements are necessary after commissioning. All rising mains laid out in difficult terrains require careful planning and site-specific considerations and technical solutions.

#### 3.8 Duty Point Evaluation for Parallel Pumping Stations

The combined output of two or more pumping stations discharging at different points into the same common rising main may be determined using a graphical method. The method is described below.

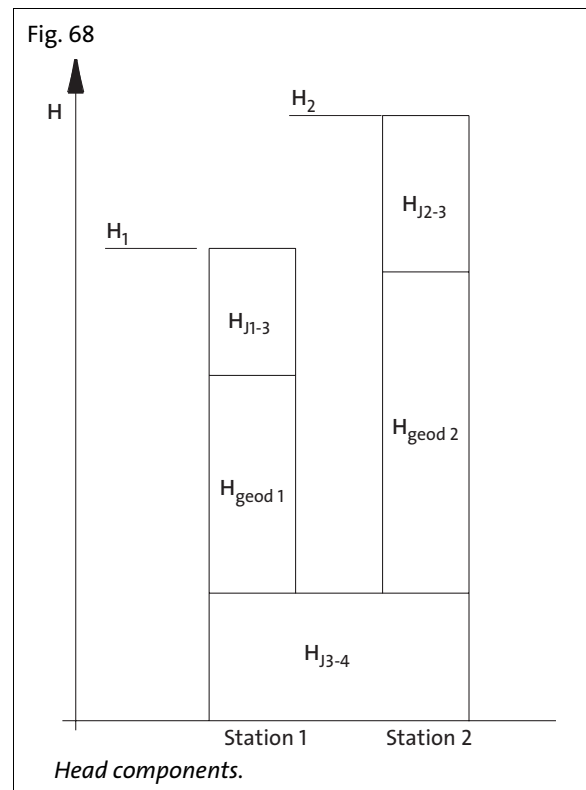
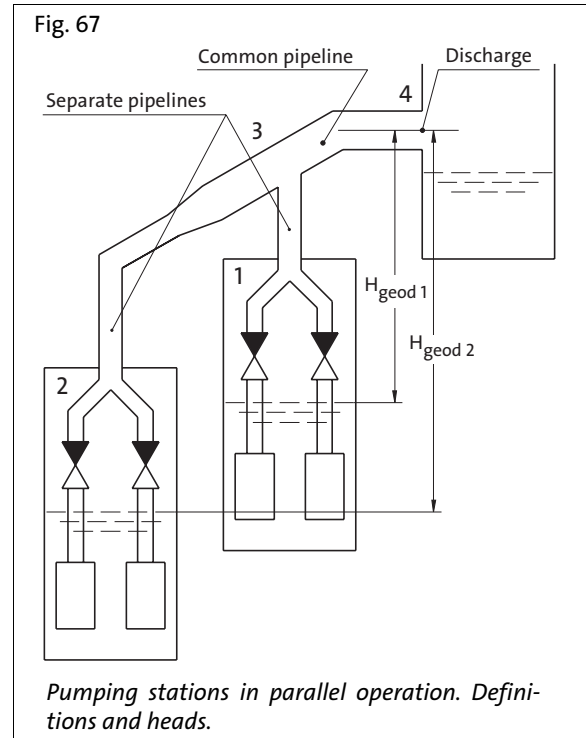
Figure 67 presents graphically the situation where two pumping stations operate in parallel discharging into a common rising main. When both pumping stations are operating, the pump operating points are governed by the pressure at the junction point 3, where the outputs of the pumping stations merge in the common rising main. The total heads for the individual pumping stations can be separated into components as is shown in Figure 68. The heads comprise the following components.

- $H_{J\ 3-4}$  = Pipe loss in the common main between sections 3 and 4
- $H_{geod\ 1}$  = Geodetic head for pumping station 1
- $H_{geod\ 2}$  = Geodetic head for pumping station 2
- $H_{J\ 1-3}$  = Pipe loss in separate portion of rising main between points 1 and 3
- $H_{J\ 2-3}$  = Pipe loss in separate portion of rising main between points 1 and 3

The loss in the common main  $H_{J\ 3-4}$  is equal for both pumping stations.

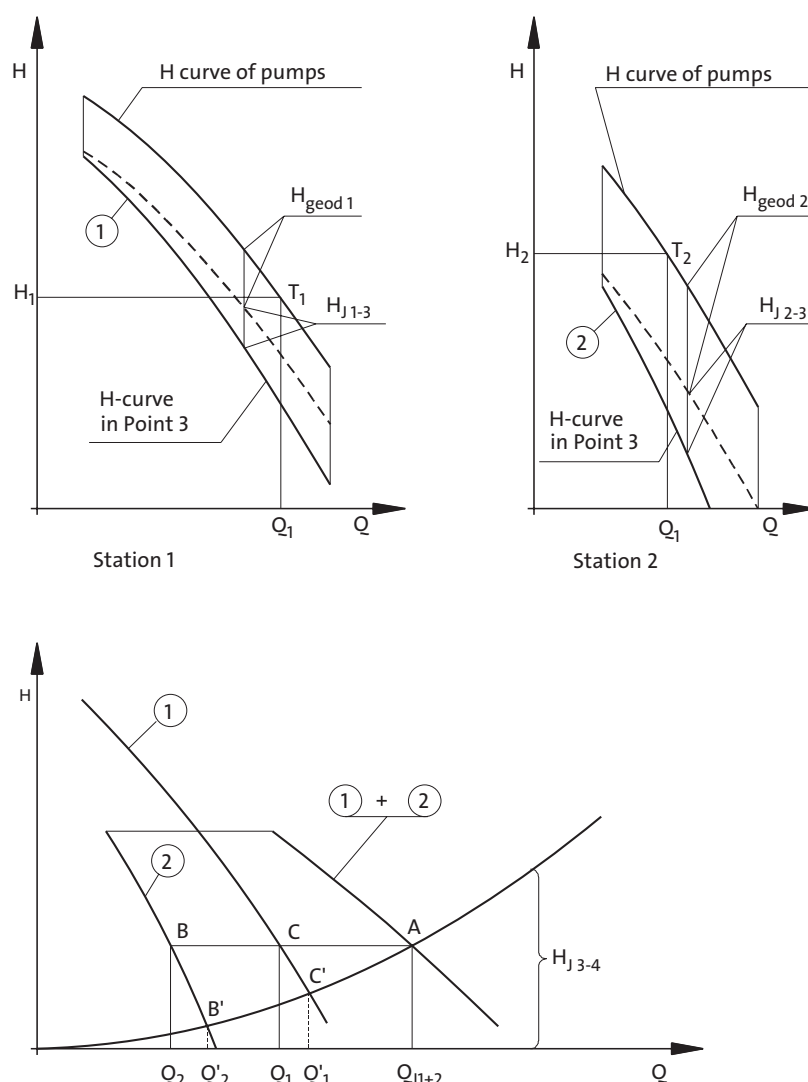
The combined output of two pumping stations is graphically determined by following the steps of the procedure shown in Figure 69:

1. The geodetic head  $H_{geod}$  and the pipe loss in the separate portion  $H_J$  are subtracted from each of the pumping station H curves. The H curve is taken for one pump or two pumps operating, as the case may be. The pipe loss  $H_J$  is also determined accordingly.
2. The head loss curve  $H_{J\ 3-4}$  for the common main is plotted



3. The reduced H curves 1 and 2 obtained in step 1 is plotted onto the head loss curve both combined and separately (1+2).

Fig. 69



*Establishing operating points for pumping stations discharging into a common rising main.*

4. The intersection point A between the combined pumping stations H curve 1+2 and the head loss curve  $H_{J\ 3-4}$  represents the combined output  $Q_{1+2}$  at the discharge point.
  5. A horizontal line is plotted through point A, intersecting the separate head curves 1 and 2 in points C and B respectively. The corresponding flow rates at these points,  $Q_1$  and  $Q_2$ , represent the pumping station individual outputs.
  6. Plotting the individual outputs  $Q_1$  and  $Q_2$  onto the individual head curves for each pumping station, the operating point for each pump is obtained as the intersection points T1 and T2.
- The operating points for the pumping stations

working singly are the intersection points C' and B' of the reduced individual head curves 1 and 2 and the head loss curve  $H_{J\ 3-4}$  as plotted in step 3 above.

The procedure can be extended for installations with even more pumping stations in a common main. Working out the various operating points becomes, however, an arduous task. Large sewerage systems comprise collection wells and gravity sewer sections, breaking the network into separate pressurised sections that each can be determined exactly. It is therefore unlikely that very complex combined calculations will have to be performed.

# 4 Testing of Pumps

Actual pump performance is established or confirmed through testing. Tests may be conducted for a number of different reasons and at various locations. In order to reach unambiguous results, testing standards have been developed and agreed upon. The standards require controlled and calibrated testing circumstances and are therefore generally not applicable for field or site testing.

A pump manufacturer tests his pumps at the works for both production development purposes and quality control. Corroborated tests may be required to confirm that pump performance is according to the terms of purchase or to settle performance disputes.

Field testing of pumps in actual installations will not yield exact data on pump performance, because the precision of the testing arrangements cannot meet the terms of the testing standards. These tests provide useful information on pumps and pumping stations, however, and may be used for pumping station monitoring, if performed periodically or if suitably automated.

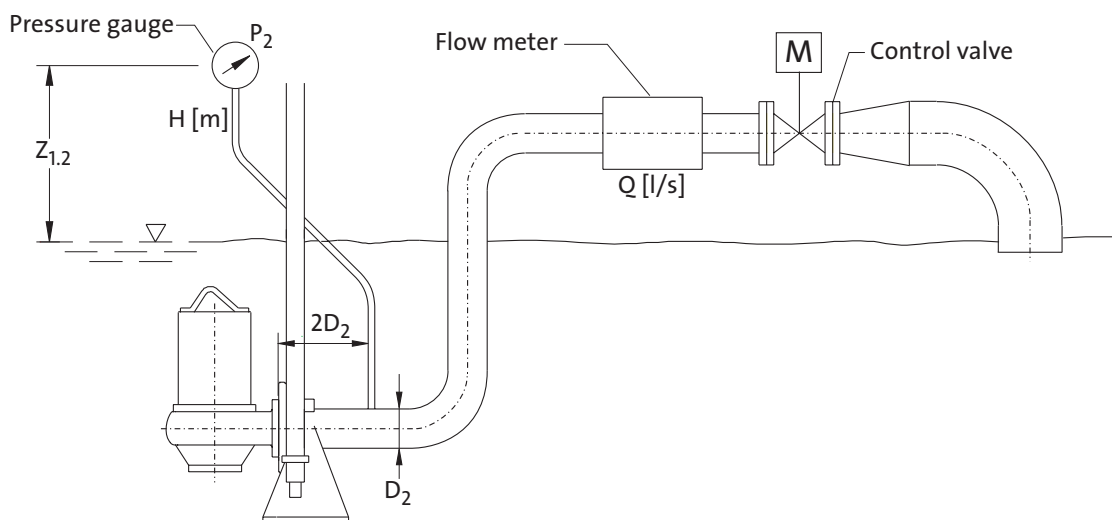
Testing standards are conventions that are agreed for use as a gauge for pump performance evaluation. The presentation below offers methods for their interpretation. It has been kept brief for clarity, and therefore it can and should be used with the appropriate standard text as immediate reference.

## 4.1 Testing Arrangements

### 4.1.1 Production Testing

Testing of submersible pumps under controlled circumstances requires a testing facility built and calibrated to the standards governing the testing. Testing facilities in a production line should also be designed for efficient handling and connection of the pumps so as not to slow down the production process. Figure 70 shows the principle of a submersible pump testing facility. The test rig includes the necessary pipework and instruments for pressure and flow rate measurement. The testing facility may also include various measurements recording devices as well as computing equipment for the processing and presentation of the measured data. For pump head the total head, including both the static and dynamic component is used.

Fig. 70



*Principle of a submersible production pumps testing facility, where water is circulated. All pipework is designed to provide ideal and known operating conditions for the pressure gauge and flow meter for unambiguous readings. Obtained data are fed into computer for speedy results and evaluation service.*

Pump total head was established in equation 18 and can be written:

$$H = \frac{p_2}{\rho g} + Z_{1,2} + \frac{v_2^2}{2g} + H_J \quad (35)$$

where

$\frac{p_2}{\rho g}$  = pressure gauge reading changed to head

$Z_{1,2}$  = pressure gauge height above water level

$\frac{v_2^2}{2g}$  = pump dynamic head at pressure measurement point

$$v_2 = \frac{4 \cdot Q}{\pi \cdot D_2^2}$$

$H_J$  = Head losses between point of measure and pump flange (calculated).

According to the testing standards, the point of measure of pressure shall be at a distance  $2 \times D_2$  from the pump pressure flange. The distance shall constitute of a straight pipe section.

#### 4.1.2 Field Testing, Duty Point

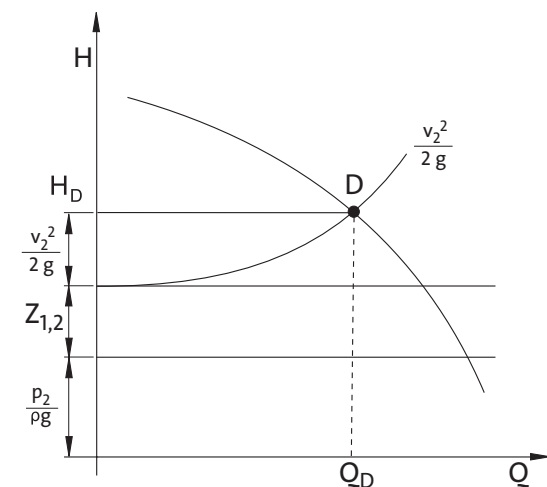
Testing of pumps in actual installations is useful when information on pump performance with reasonable accuracy is required, or when pumping station performance over time is monitored.

The pump flow rate can be accurately estimated with the volumetric method, where the change of wet well level in a pumping station with known dimensions is measured against time. If the incoming flow in the pumping station cannot be stemmed for the duration of the measurements, the effect of this must be checked separately with the pumps stopped. The pump flow rate can then readily be calculated.

Pump static pressure is measured with a pressure gauge connected to the submersible pump near the outlet flange. The pump total head can then be calculated using equation 35 ( $H_J = 0$ ). The test arrangement is shown in Figure 72.

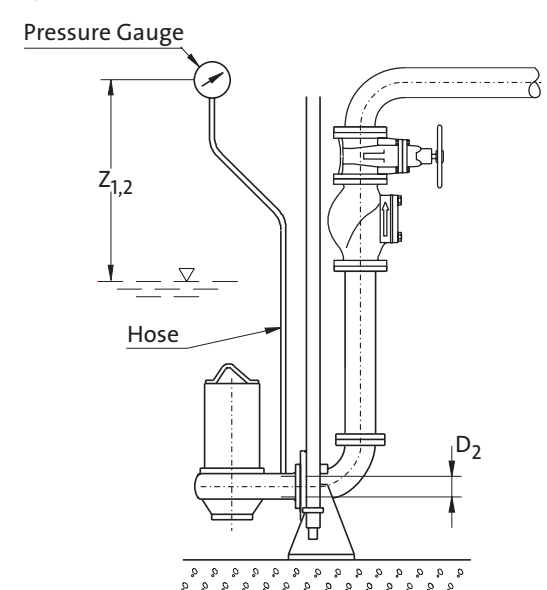
If an accurate head curve for the pump is avail-

Fig. 71



Using a pump head curve sheet for operating point estimation. Measured static head and dynamic head function against flow rate are plotted. Pump operating point D is obtained graphically.

Fig. 72



Field testing of a submersible pump. The pressure gauge is connected to the pump pressure flange with a flexible hose. The height of the gauge above the water level in the well during the event,  $Z_{1,2}$ , is recorded. Pump outlet diameter  $D_2$  is used for calculation of dynamic head if flow rate is known. Air in the flexible hose must be removed after the pump is started.

able, the pump operating point can be determined without an estimation of the flow rate. The sum of the static head and the gauge height  $Z_{1,2}$  is

## 4 Testing of Pumps

measured onto the H axis of the pump head curve sheet. The function of the dynamic head against a number of flow rates is then plotted onto the sheet. Pump operating point will be in the intersection of this curve and pump head curve. The principle is shown in Figure 71.

Site conditions do not fulfil the testing rig requirements of the testing standards. The results can therefore not be used for pump acceptance tests.

A pressure gauge connected to the piping of a pumping station can be used for an approximate determination of the pump duty point. The height of the pressure gauge above the water level in the pump suction sump, the calculated pressure losses between pump flange and point of measure as well as the dynamic head ( $v^2/2g$ ) shall be added to the reading. For dry-installed pumps the losses in the suction piping shall be deducted. These losses are usually minimal.

### 4.2 Acceptance Tests

Pump acceptance testing is the procedure with which a pump is confirmed to have the properties set out in the manufacturer's sales literature or contract specifications. Acceptance tests can be routinely carried out by the manufacturer as part of the manufacturing process or may be executed in the presence of the customer or his representative.

The testing standards contain two major principles:

- The pump is tested at a duty point agreed upon at the time of purchase.
- The pump is tested at any point along the curve published for the pump. This practice is intended for pumps in serial production, and the allowed tolerances are larger than those for custom-made pumps.

In the case of pumps in serial production, the pump manufacturers perform production tests of their pumps either at a multitude of points along the curve or at three distinct selected points. These three points are selected at both ends of the allowed portion of the pump curve, and at a point at the middle of the curve.

The essence of the testing standards is to govern how the tests should be conducted technically and what are the allowed performance tolerances, unless otherwise agreed. The testing standards do not regulate actions to be taken in case a pump fails to perform according to the tolerances, or the consequences thereof. The parties should separately agree on these issues at time of purchase or later.

#### 4.2.1 Testing Standards

The purpose of the testing standards is to define in detail how the tests are performed and how the test results shall be technically compared with the guaranteed values. The content of the standards is mainly as follows:

- terms, definitions and symbols
- organizations of tests
- test arrangements
- measuring uncertainties
- verification of guarantees

Unless otherwise agreed, the testing standards stipulate the following guarantee values to be compared:

#### Testing Standard ISO 9906 (Grade 1 and 2)

Pumps over 10 kW power:

- Q/H duty point
- Efficiency  $\eta$  or  $\eta_{gr}$

Pumps less than 10 kW power:

- Q/H duty point
- Efficiency  $\eta$
- Motor power input  $P_{gr}$  (over the range of operation)

Pumps produced in series with selection made from typical performance curves (Annex A):

- Q/H duty point
- Efficiency  $\eta$
- Pump power input P
- Motor power input  $P_{gr}$

#### Testing Standard ISO 2548 (Class C)

- Q/H duty point
- Efficiency  $\eta$  or  $\eta_{gr}$

Pumps produced in series with selection made from typical performance curves (Annex B):

- Q/H duty point
- Motor power input  $P_{gr}$

### Testing Standard ISO 3555 (Class B)

- Q/H duty point
- Efficiency  $\eta$  or  $\eta_{gr}$

These standards contain performance values for the tolerances of the measured variables.

If guarantee values are required in specifications or sales contracts, the following variables according to the testing standards are suitable:

- Q/H duty point
- Efficiency  $\eta_{gr}$  or  $\eta$

The desired duty point and testing standard to be used must also be specified.

The testing standards do not require testing of the pump NPSH<sub>r</sub> value unless specifically requested. NPSH tests are difficult and time-consuming, and do not provide absolute information on the possibilities of cavitation, as can be seen in Section 1.4.3. The benefits of NPSH tests are thus questionable. The testing standard ISO 9906 provides tolerance factors for the NPSH<sub>r</sub> value. In the testing standards ISO 2548 and ISO 3555 no tolerance factors for NPSH<sub>r</sub> are provided.

### Testing Standard ISO 9906 Grade 1 and 2

The new testing standard ISO 9906 was published in 2000 and is scheduled to replace the older testing standards ISO 2548 and ISO 3555.

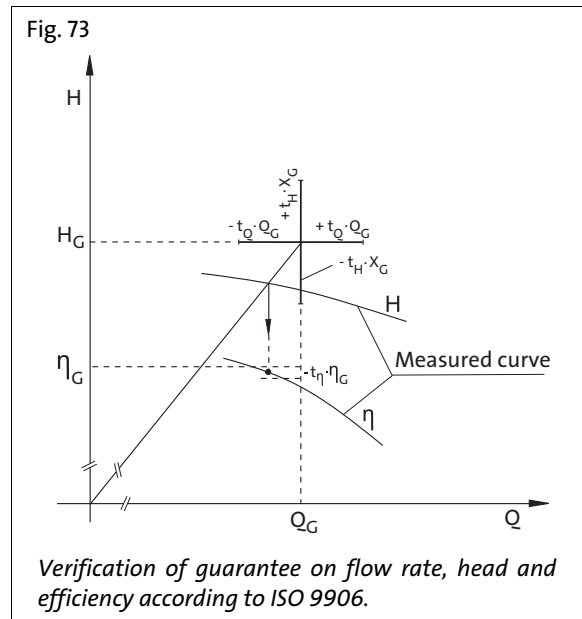
Grade 1 requires higher accuracy, whereas Grade 2 allows larger tolerances. Because sewage pumps usually operate in intermittent duty, Grade 2 is suitable for these pumps. Grade 1 is intended for the testing of fine-tuned process pumps in continuous duty. For the verification of guaranteed values, a crosshair method is used. The principle is shown in Figure 73.

The verification principle as shown in Figure 73 works in the following way:

A tolerance cross with the horizontal line  $\pm t_Q \cdot Q_G$  and the vertical line  $\pm t_H \cdot H_G$  is drawn through the guarantee point  $Q_G, H_G$ .

The guarantee on the head and flow rate has been met if the measured Q/H curve cuts or at least touches the vertical and/or horizontal line.

The efficiency shall be derived from the measured



Q/H curve where it is intersected by the straight line passing through the specified duty point  $Q_G, H_G$  and the zero of the Q, H axes and from where a vertical line intersects the  $\eta$  curve.

The guarantee condition on efficiency is within tolerance if the efficiency value at this point of intersection is higher or at least equal to  $\eta_G (1-t_\eta)$ .

### Testing Standards ISO 2548 (Class C) and ISO 3555 (Class B)

The ISO 2548 (Class C) corresponds generally to the standard ISO 9906 Grade 2 and the ISO 3555 (Class B) to the standard ISO 9906 Grade 1. The standard ISO 2548 is suitable for sewage pumps.

In these standards an elliptic graphic method for the verification of guaranteed values is used. The principle is shown in Figure 74. The efficiency verification is performed in the same way as in standard ISO 9906.

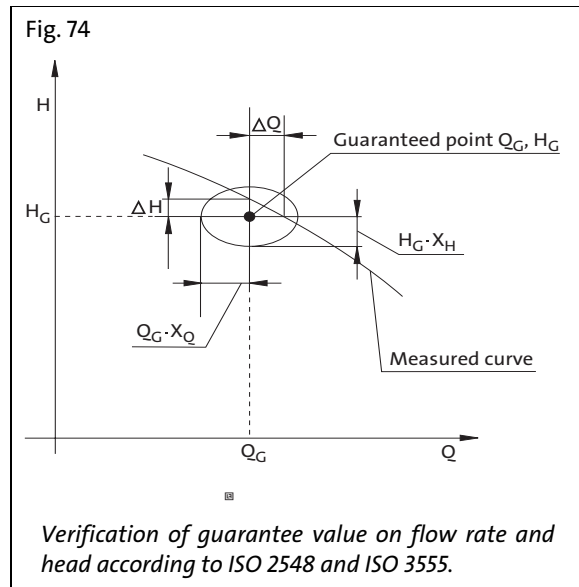
The verification principle as shown in Figure 74 works in the following way:

An elliptic tolerance zone with semi-axis  $Q_G X_Q$  and  $H_G X_H$  is drawn with the guaranteed duty point  $Q_G, H_G$  as centre point.

The guarantee on the head and flow rate has been met if the measured Q/H curve cuts or at least touches the ellipse.



## 4 Testing of Pumps



The efficiency shall be derived from the measured Q/H curve where it is intersected by the straight line passing through the specified duty point  $Q_G$ ,  $H_G$  and the zero of the Q, H axes and from where a vertical line intersects the  $\eta$  curve.

The guarantee condition on efficiency is within tolerance if the efficiency value at this point of intersection is higher or at least equal to  $\eta_G (1-t_\eta)$ .

Grundfos has developed an application method of the elliptic tolerance zones of the ISO 2548 standard, making it more convenient to use in numeric calculations. The method uses the slope of the tangent to the Q/H curve at the point of examination, and makes it possible to numerically determine  $H_{\min}$  and  $H_{\max}$  at guaranteed rate of flow, so that the ellipse condition is met.

### Other Testing Standards

Many countries have issued national standards equivalent to the ISO standards. In the U.S. a national testing standard issued by Hydraulic Institute is frequently used. This standard differs from the ISO standards with regard to the tolerance system.

### Allowed True Performance Deviations

The maximum possible deviation from the guaranteed duty point is composed of inaccuracies of the measurement technology and the allowed tolerances. The testing standards specify the accuracy requirements of the measuring instruments and guide values for the allowed tolerances.

The maximum possible true deviation from desired volume of flow is also dependent on the shape of the rising main characteristic curve and location of duty point on the pump Q/H curve. According to the standards ISO 9906, Grade 2 and ISO 2548, the deviation near the optimal point may be  $\pm 3 \dots 10\%$ , depending on main curve shape. For the standard ISO 9906, Grade 1 and ISO 3555, the corresponding deviation is  $\pm 2 \dots 6\%$ . If the duty point is in the low volume flow range, and the rising main characteristic curve is flat, the deviations above may be much greater.

The normal tolerances for pump efficiency according to the ISO testing standards are as follows:

- ISO 9906, Grade 2     -5%
- ISO 2548               -5%
- ISO 9906, Grade 1   -3%
- ISO 3555               -2,8%

These are proportional values, not percentage points.

For sewage pumping the  $\pm$  tolerances of the standards ISO 9906, Grade 2 and ISO 2548 are quite acceptable. They are also compatible with normal production variations in manufacturing. More stringent requirements may cause extra costs in production and delivery delays. The published curves of sewage pumps are also based on these standards, a fact stated on these curves.

Sometimes a condition stipulating that negative tolerances are unacceptable may be raised by customers. Problems and misunderstandings are a common consequence of this, with difficulties for both manufacturer and customer. The ISO testing standards do not recognize asymmetric tolerance systems, and the published curves of the manufacturers are based on the symmetric tolerance systems of the ISO standards. If the customer finds that a true volume flow lesser than indicated by the published curve is unacceptable, a better solution than requesting no-negative tolerances would be to increase the required volume flow by  $3 \dots 10\%$  and then select a pump based on this figure.

Requiring no-negative tolerances for the pump efficiency does not make much sense, since pump manufacturers would then be forced to lower the published nominal figures. Too low, conservative figures would not be representative of the majority of the pumps and lead to misunderstandings and confusion.

## 5 Pumping Stations

The working environment for submersible pumps, regardless of size, is the pumping station. Pumping station design and construction is decisive for the performance of the pumps, and care and diligence should therefore be exercised whenever specifying them. The following is a primer of pumping station design offering hints and advice for the design engineer and the operator of pumping stations. Some aspects of pump operation and interaction with the pumping station pipework is also discussed.

### 5.1 Pumping Station Basic Design

The decisive factor for pumping station operation is a good hydrodynamic design. A bad pumping station design may lead to pump malfunction, uneconomical pumping and frequent needs for pumping station service and cleanout.

Modern sewage pumping stations are designed for pumping of unscreened sewage, and the design criteria for these differs from those for clean water. In the following the design and special requirements of sewage and stormwater pumping stations are discussed.

#### 5.1.1 Wet Well Volume and Surface Area

The wet well effective volume should be of correct size. Too large a volume may lead to accumulation of sludge in the well, whereas too small a volume leads to too frequent starting and stopping of the pumps. The use of modern submersible pumps, with high allowable starting frequency, leads to smaller and more efficient pumping station designs.

The effective sump volume is the volume between pump start and stop levels, and it can be determined with the use of nomograms as a function of allowed starting frequency. A method for the calculation of the effective sump volume is presented in Appendix B of this book.

In reality the incoming volume in a pumping station varies greatly over time, and the mean starting frequency will therefore be lower than theoretical.

In a good design the start and stop levels should be relatively close to each other for the following reasons:

- Pump starting frequency becomes high enough to prevent sludge and impurities from settling onto well floor.
- Pumping station inlet should stay low relative to the liquid level in the wet well.

A guideline maximum value for the effective volume height in small pumping stations is approx. 1 m and 2 m in larger pumping stations.

The effective volume can be substituted with the wet well surface area using the following equation:

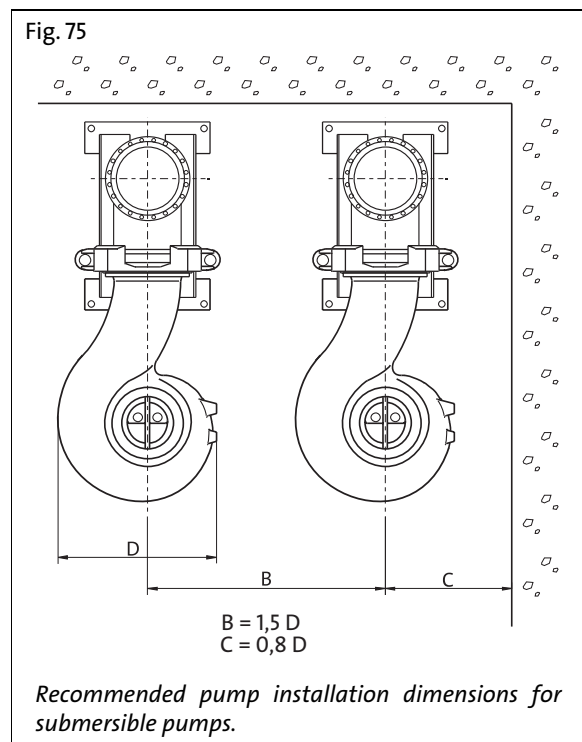
$$A_W = \frac{Q}{20} \quad (36)$$

where

$A_W$  = wet well surface area in  $m^2$

$Q$  = pumping station total flow rate, l/s

For small pumping station flow rates, however, the surface area will be limited by the physical dimensions of the pumps when submersible pumps are used. The surface area will then be



## 5 Pumping Stations

larger than obtained with equation 36. Recommended pump installation dimensions are shown in Figure 75.

For larger flows the direction of approach towards the pumps should be square on. If the flow comes from behind, the submersible baseplates disturb the flow causing eddies to form. These impede pump operation, lowering pump performance and efficiency and increasing the risk of cavitation and pump vibrations.

### 5.1.2 Pumping Station Inlet Pipe

Location and size of the pumping station inlet pipe is important for the function of the pumping station. Problems encountered in operation of the pumps are frequently caused by bad inlet pipe design.

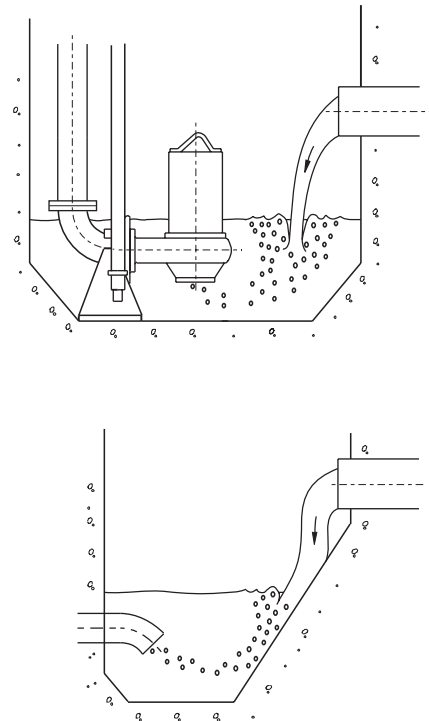
An inlet pipe located too high relative to the liquid surface or with a high flow velocity may cause entrainment of air and the formation of eddies in the water when splashing into the well. Air mixed into sewage water has a tendency to remain because of the possibility of air bubbles to adhere to the solid particles present. A separate calming chamber may therefore not alleviate the situation at all.

Inlet fall height should always be minimized and should not exceed 1 m with the water level down, regardless if the pumping station has a separate calming chamber or not. The effect of a high inlet fall height cannot be effectively alleviated with baffles.

Air entrained in the water has a tendency to remain inside the pump impeller, where the centrifugal force makes it accumulate around the impeller hub. This may lead to increased power requirement and lowered performance and efficiency. The risk of cavitation and pump vibrations also increases. If the amount of air in the pump is very high, the pump may cease to function altogether.

Air is frequently a problem for pumps drawing directly from aeration basins in treatment plants, because of the high content of air. If a pump is placed in an aeration basin it should be placed as low as possible, with the suction pipe near the bottom.

Fig. 76



*Inlet locations to be avoided. Too high an inlet fall height may lead to entrained air reaching the pump inlet directly or along the bench surface, with consequent operational problems in the pumps.*

Location of the inlet pipe should be as remote as possible from the pump inlets. Figure 76 shows designs to be avoided.

Flow velocity at the inlet should not exceed 1,2 m/s so as to avoid the formation of eddies in the wet well.

### 5.1.3 Wet Well Floor Shape

The shape of the wet well floor is important for the functioning of a sewage pumping station. A good design prevents bottom sedimentation, but may also assist in the prevention of scum formation and the accumulation of flotsam on the surface. The following principles should be recognized in a good bottom design:

All corners should be benched to a minimum bench angle of 45°, in small pumping station the bench angle may be as high as 60°. The angle may be smaller, if the section is flushed by a flow.

Bottom area should be minimized and liquid volume below pump stop level should be kept to a minimum.

Minimizing the bottom area and the residual volume, the flow velocities near the inlets will increase, flushing out possibly settled sludge. A surface area decreasing with the falling water level leads to less accumulation of surface debris.

### 5.1.4 Stop Levels

The start and stop levels are specified at the design stage. They should always be checked for function and possibly altered at commissioning in order to secure good operation.

The stop level should be as low as possible, so that the flow velocity increases toward the end of the working cycle. Limits for the stop level are set by the required motor cooling submergence or by the level when air becomes sucked into the pump intake. The latter level cannot always be foreseen, but must be confirmed at trials during pumping station commission.

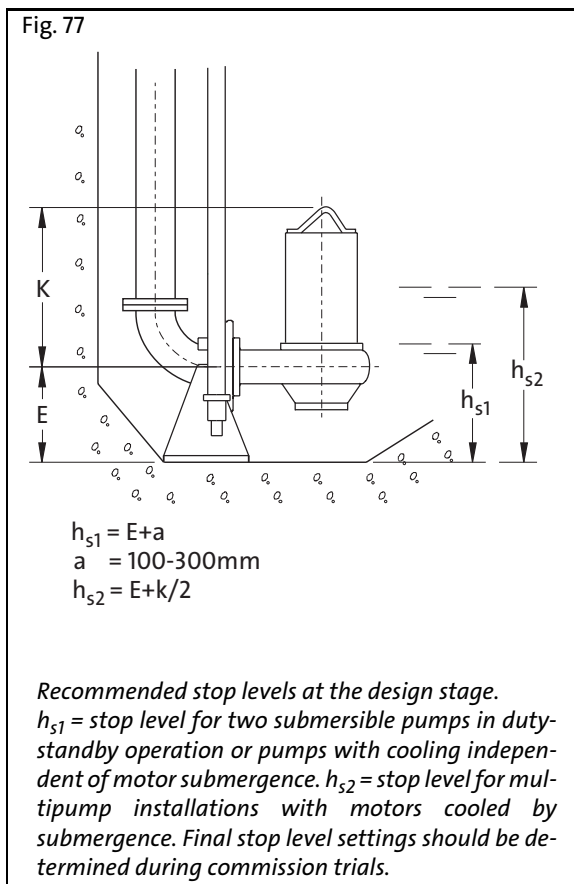
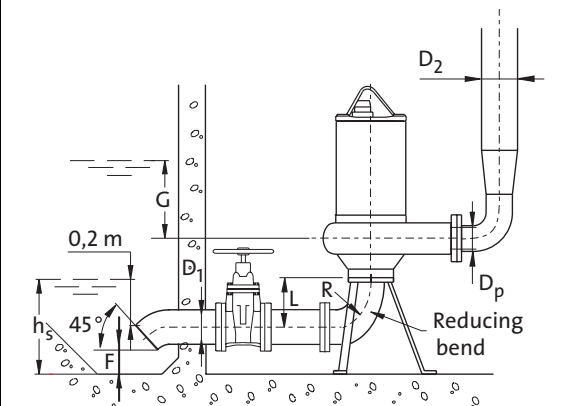


Fig. 78



*Recommended installation dimensions for vertical dry-installed submersible pumps.  $F = 0,5 \cdot D_p$ ,  $v_{1max} = 2,0 \text{ m/s}$ ,  $G = D_p$ ,  $L \geq D_1 + 100 \text{ mm}$ ,  $R \approx L$ .*

In pumping stations with two submersible pumps in duty-standby configuration the stop level can normally be set below the motor even if the motor is cooled chiefly by submergence, see Figure 77. The identical pumps are selected so as to be able to cope with the pumping station flow alone, and the risk of the liquid level remaining for long near the stop level is slight. Submersible pumps also have protective devices against overheating that stop the pump in case of inadequate cooling conditions.

In pumping stations with a multitude of submersible pumps running under varying conditions, the stop level must be set so that the pump motors always have enough submergence for adequate cooling. Pumps with cooling water jackets or other means of heat dissipation independent of submergence are preferred in such installations.

The stop level setting for dry-installed pumps is dependent on the suction pipe inlet height, shape and flow velocity. 200 mm above the suction pipe inlet is a good rule-of-thumb for this height, and useful for the designer. The shape of the suction pipe inlet is important, and good designs are shown in Figures 78 and 79. For this inlet shape a provisional pump stop level height can be calculated using the following equation:

$$h_s = 0,04\sqrt{Q} + 0,2 \quad (37)$$

## 5 Pumping Stations

where

$h_s$  = stop level height, m

$Q$  = pump flow rate, l/s

In pumping stations with several different stop levels, such as in frequency-controlled installations, it is important to program the control sequence to pump down to lowest stop level at least once per day to clean out the bottom.

### 5.1.5 Start Levels

If the wet well surface area  $A_W$  is dimensioned using equation 36, the first start level in a pumping station with two submersible pumps in duty-standby configuration can be set 1 m above the stop level. Where small inflows are encountered, the start level may be lower. The second start level, can be set 0,2...0,3 m above the first.

In pumping stations with more than two pumps the starting levels should be considered from case to case. If the pumps have a common stop level, a suitable design would be with the first start level 1 m above stop level and the following start levels at 0,3 m intervals from this. If the pump stop levels are staggered should the start levels be set at or near equal intervals.

In pumping stations with dry-installed pumps the starting levels have to be set above the pump casing in order to ensure that the casings fill up and the pumps start pumping. For vertical pumps, this height may be considerable and should be set with a margin according to Figure 78.

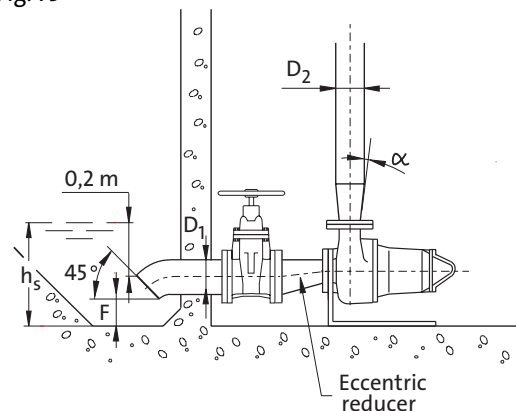
Horizontal pumps do not normally require special considerations for the start levels, if the suction pipe is designed to prevent air pockets from forming, see Figure 79.

### 5.1.6 Suction Pipe Dimension and Design

Design and dimensioning of the suction pipe is important, with bad designs possibly causing vibrations, lowered pump efficiency and risk of cavitation.

The suction pipe should be dimensioned so that the flow velocity does not exceed 2,0 m/s for vertical pumps and 2,5 m/s for horizontal pumps. When new, bigger pumps are installed in old pumping stations, these figures may have to be exceeded. The situation must then be considered

Fig. 79



*Recommended installation dimensions for horizontal dry-installed submersible pumps.  $F = 0,5 \cdot D_1$ ,  $v_{1max} = 2,5$  m/s.*

from case to case. Wider NPSH safety margins may be warranted.

Recommended suction pipe inlet designs are shown in Figures 78 and 79. The downward suction exerts a cleansing flow on the pumping station floor, and is less prone to suck air from the surface.

In vertical pumps the suction pipe will have to turn 90° to reach the pump suction cover. The bend before the pump suction inlet is crucial for the function of the pump, since it causes the flow to be irregular. Too sharp a bend may cause impeller cavitation, lower pump efficiency and cause vibrations. If the pump suction inlet is smaller than the suction piping, a reducing bend should be used, minimizing the interference. Figure 78 shows suction bend dimension recommendations.

The contraction of the straight inlet pipe to a horizontal pump should be eccentric so as to avoid air from collecting and possibly blocking the impeller.

An inlet design with unfavourable flow characteristics may cause a pressure drop large enough to spend the available NPSH and lead to pump cavitation. The recommended NPSH margin should be observed in installations where the suction pipe geometry gives reason to concern. The concept of cavitation and NPSH and recommendations for NPSH margins are presented in detail in Section 1 of this book.

### 5.1.7 Pumping Station Internal Pipework

The internal pressure pipework in a pumping station should be selected for a flow velocity of 2...3 m/s. Especially if the sewage contains sand should the flow velocity be at least 2 m/s, in order for the sand to be carried with the flow out of the pump. In frequency-controlled installation this requirement may cause problems at low frequencies. Figure 58 in Section 3 shows size recommendations and losses typical for piping. The pipework should have a dimension of at least 100 mm but can be 80 mm in small pumping stations, provided that the pump free passage is 80 mm.

The use of flexible joints in the internal pipework is not recommended, since most pipe vibrations are pressure-induced by the flowing liquid, and cannot be avoided by the use of flexible joints. When installing flexible joints the pipe is cut, and the section will be subject to a separating force with a magnitude of pump pressure  $\times$  area. The pressure near the pump is pulsating at a frequency determined by pump speed and number of impeller channels, causing the piping and joint to vibrate. The vibration is more pronounced when flexible joints are installed. Flexible joints are also susceptible to damage.

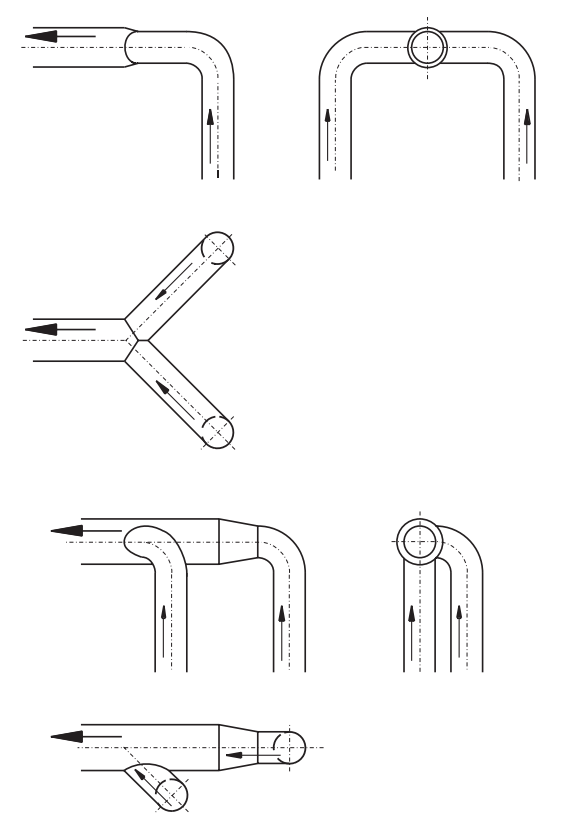
The pressure pipework is normally expanded after the pump, and in order to save energy these transition pieces should have a conical shape with a maximum shank angle  $\alpha$  of  $10^\circ$ . Please see Figure 79.

For vertical dry-installed pumps and submersible vortex pumps the check valve should be installed as far away as possible from the pump in order to alleviate possible problems with air in the pump at start-up.

For horizontally installed large pumps, where the shaft bearings include separate bearings for radial and axial forces, the check valve must not be placed directly in the vertical pipe from the pump delivery flange. Possible shocks from a rapidly closing valve may pound at the pump hard enough to gradually damage the radial bearings.

In multipump installations the pump pressure pipes should be joined by a branch designed to prevent settling of solids during pump stoppage into the individual pipes, which may lead to valve blockage. Good branch designs are shown in Figure 80.

Fig. 80



*Pressure pipework branch designs. The design should emphasize smooth transition and prevent sludge in rising main from settling on valves in pump risers when the pumps are stopped.*

### 5.1.8 Flushing Devices

Pumping station flushing devices consist of a remote-controlled by-pass valve mounted on the submersible pump before the connection to the pipework. When the valve is opened, the pump flow is directed back into the wet well, agitating the liquid and causing settled sludge and scum to disperse. The suspended matter will then be pumped out with the liquid when the flushing valve is closed.

The flushing valve should be of the normally closed type (e.g. a spring-loaded pneumatic device) so that in the case of a malfunction the pumping action will be able to proceed.

In a correctly dimensioned and shaped pumping station wet well flushing devices are normally not needed. They serve a function in old, large wet



## 5 Pumping Stations

wells and in special situations, where the sewage contains large amounts of e.g. grease. A flushing device can also be retrofitted without changes to wet well structures. The flushing devices are brand specific and detailed information is available from the pump manufacturer.

### 5.1.9 Odour Problems in Pumping Stations

A sewage pumping station may cause odour problems in its immediate environment. Many factors affect the situation, such as pumping station location, sewage quality, situation before the pumping station and wet well dimensions and design. If the pumping station is fed by another, remote pumping station, the sewage transfer time between the pumping stations may be so extensive that the sewage turns septic by anaerobic action. Septic sewage produces Hydrogen-Sulphide ( $H_2S$ ) that, apart from being toxic, also creates a typical foul odour.

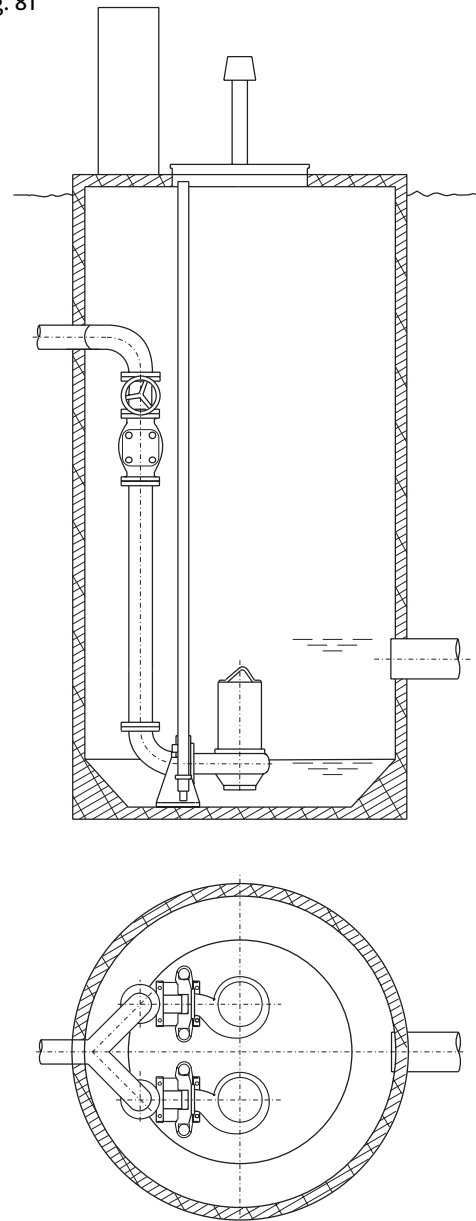
The occurrence of odour problems is practically impossible to predict. In case of severe problems, they may be attempted to be corrected by the following measures:

- Lowering start and stop levels, in order to cut the retention time in the wet well and prevent sludge from forming.
- Installing a submerged inlet bend in the wet well, in order to convey the incoming sewage below the surface, thus preventing aerosols from forming.
- Installing air filters in the wet well ventilators.
- Dosing odour-preventing chemicals into the sewer upstream from the pumping station.

### 5.1.10 Pumping Station Design Examples

Wet well design will depend on pumping station size and flow volume. Figures 81...84 show principles for wet well design for various cases and pumping station sizes. A pumping station with submersible pumps for large flows can be designed according to Figure 83. If the pumps require it, the stop level can be set at height  $h_{s2}$ . The flow velocity  $v_D$  in the expanding section of the wet well must be high enough to avoid sludge settling. A suitable value for  $v_D$  is 0,1...0,3 m/s when the liquid is at stop level. The dimension D can be calculated using the relation

Fig. 81



*Pumping station design for submersible pumps and relatively small flows ( $Q = 4...50$  l/s). The preferred cross section of small pumping stations is circular, which minimizes liquid surface area and avoids corners where sludge could accumulate. Minimum diameter 1,5...2 m to facilitate service workovers.*

$$D = \frac{Q}{1000 \cdot v_D \cdot C} \quad (38)$$

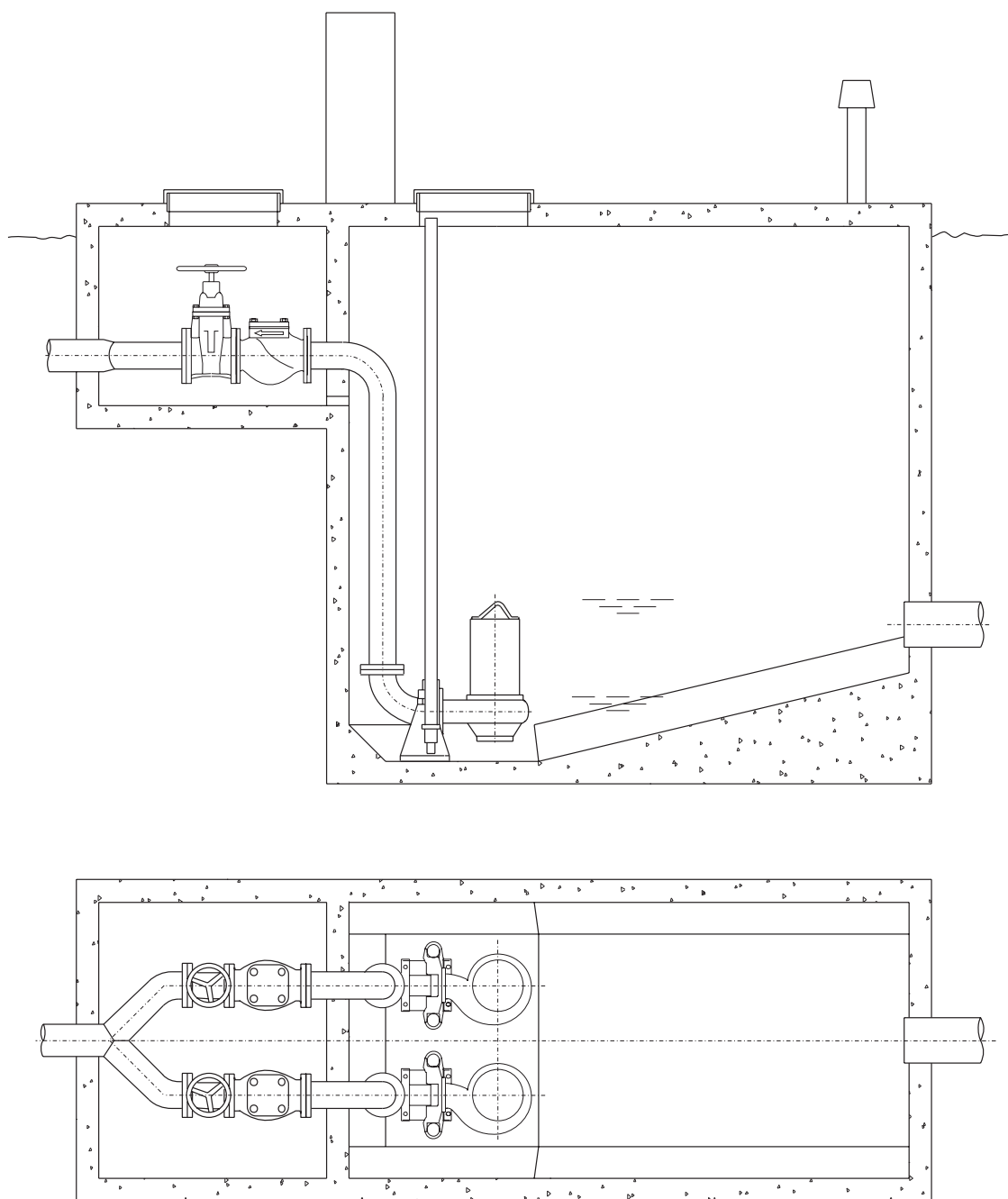
where

$Q$  = pumping station flow rate, l/s



$v_D$  = flow velocity in expanding section,  
0,1...0,3 m/s  
D, C = pumping station dimensions, m

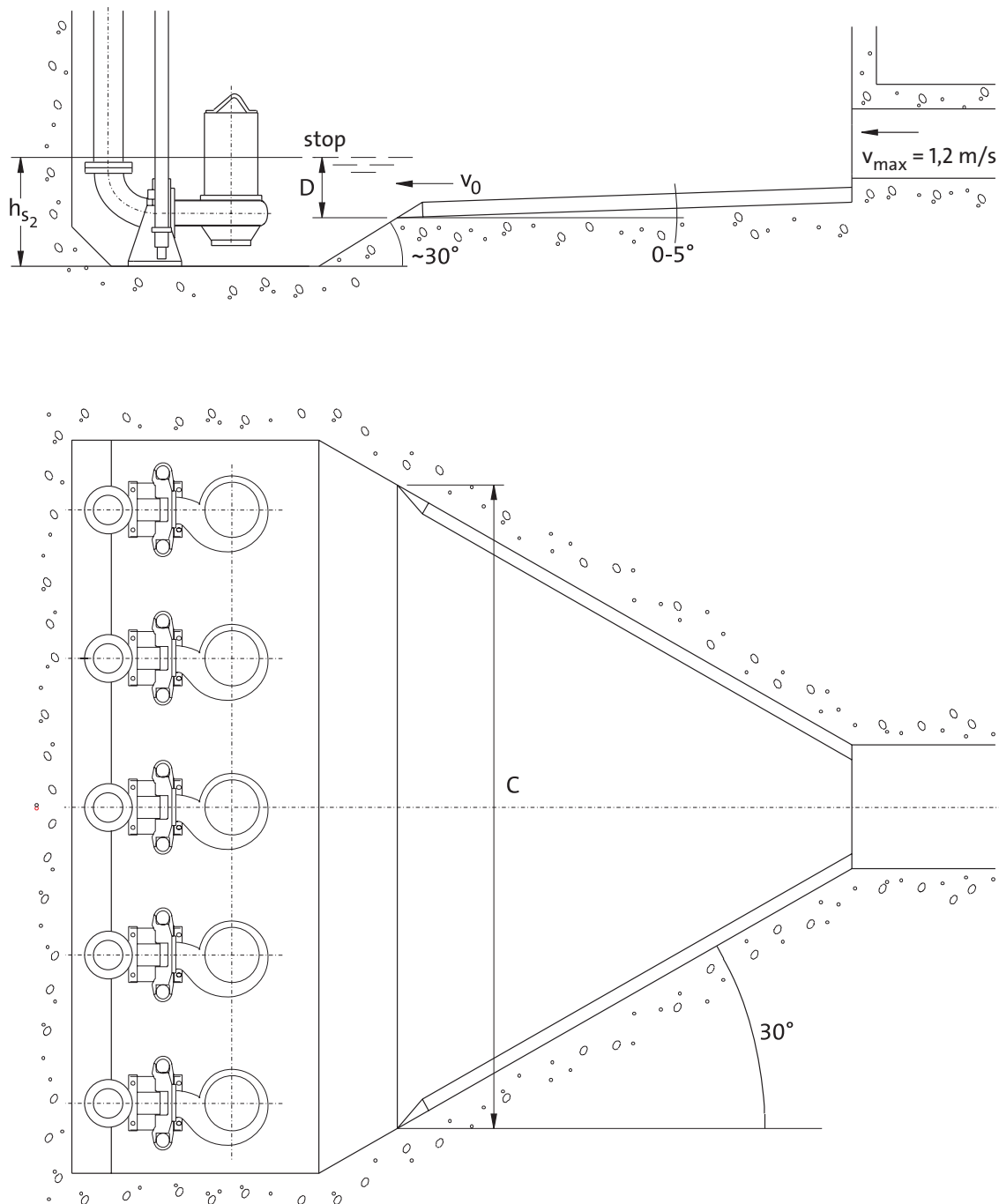
Fig. 82



*Pumping station design for two submersible pumps and moderate flows ( $Q = 50...2000$  l/s). The elongated wet well shape is an important feature that places the inlet pipe away from the pumps and prevents the build-up of sludge on the wet well floor.*

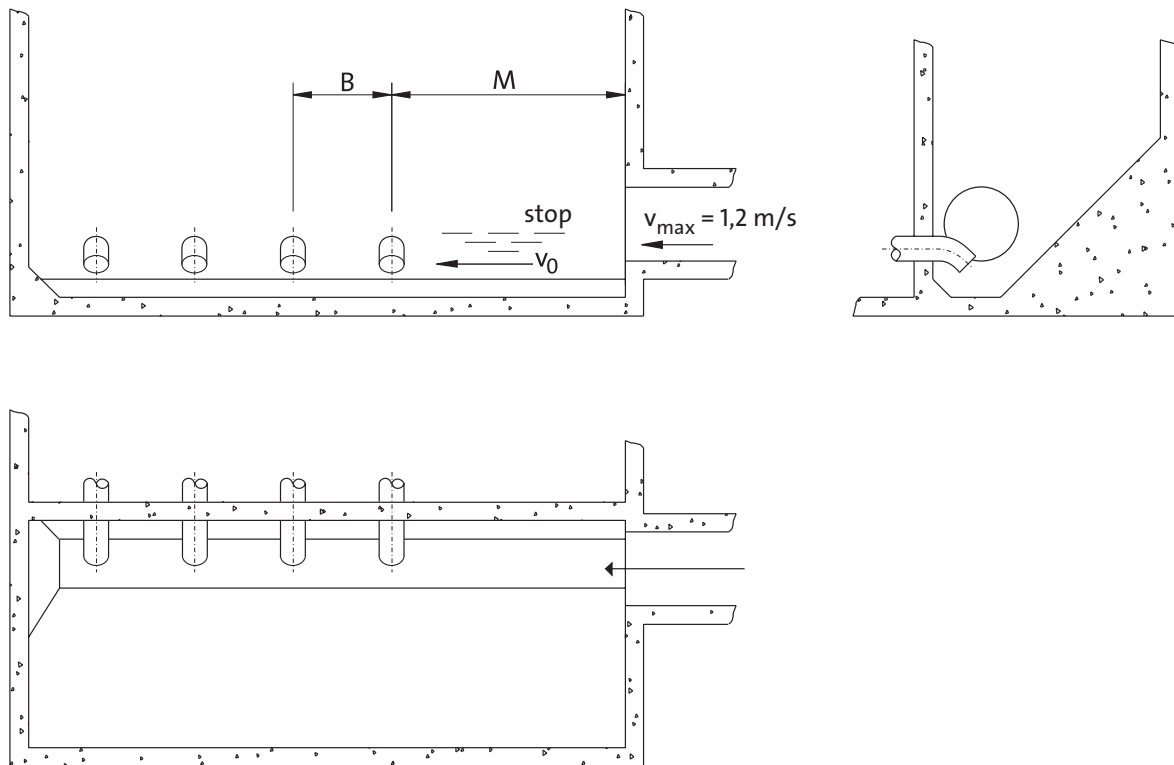
## 5 Pumping Stations

Fig. 83



*Pumping station design for several submersible pumps and large flows. If the pumps are depending on submergence for cooling, the stop level  $h_{s2}$  is chosen accordingly.*

Fig. 84



Wet well design for a pumping station with multiple dry-installed pumps. Flow velocity across the suction bends  $v_0 = 0,3 \dots 0,4$  m/s with the liquid at stop level. Pump internal distance  $B$  can be selected as for submersible pumps, whereas the distance  $M$  should be selected according to inlet fall height, and should ensure an even flow at the suction inlets.

### 5.1.11 Dry-installed Pump Positions

For dry installation most manufacturers can offer pumps for both vertical and horizontal installation. Usually a pump in horizontal position offers advantages, such as:

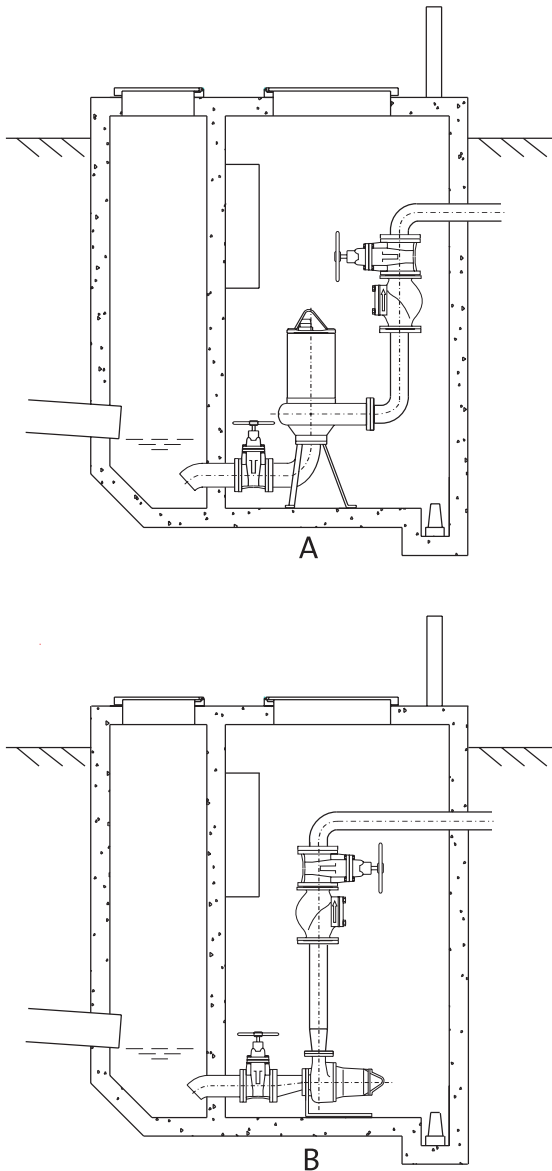
- simplified piping with less bends
- suction flow to the impeller is even
- lower pump position.

For larger pumps the NPSH safety margin requirement may not be met for pumps in vertical position, because of pump location and greater margin requirement, whereas a horizontal pump will be acceptable. All possible pump duty points must be considered when performing NPSH cal-

culations for installations where more than one pump is operated simultaneously.

Large pumps for horizontal installation are fitted with slide bars for easy removal of pump motor from the pump housing. Please refer to Figure 17 in Section 2.

Fig. 85



*Wet well-dry well pumping station outlines. Pumps can be installed vertically (A) or horizontally (B). Submersible pump construction is protected against accidental flooding of the dry chamber. A separate sump pump is provided in the dry well for drainage of leakage water. The pumping station control panel can be installed on the top or in the dry well space above flood level.*

## 5.2 Package Pumping Stations

### 5.2.1 Out-of-doors Pumping Stations

Package pumping stations are made ready at a factory for installation on site. The material used is glass-fibre reinforced plastic (GRP) or, for smaller pumping stations, Polyethylene (PE), and the stations are made complete with all internal pipework and other components in place. Thus the installation is reduced to the excavation of the site, laying of a foundation and connecting the station to the incoming sewer and rising main, and connection of the control panel to the power supply and possible telemetry connections.

The buoyancy of the pumping station when empty requires it to be anchored to a foundation or concrete slab, which also may be prefabricated and matched to the pumping station foundation bolts. The concrete slab mass can be calculated using the following equation:

$$M_B = 2000 \cdot V_G \quad (39)$$

where

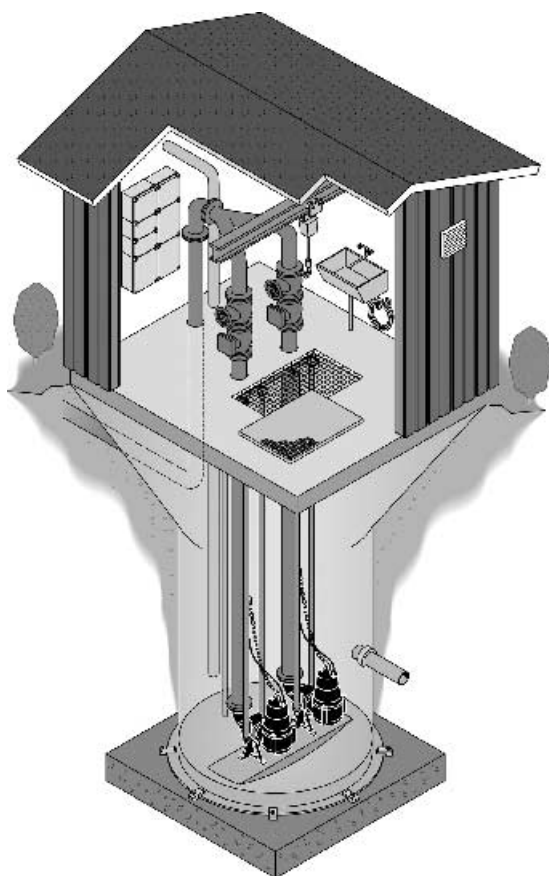
$M_B$  = concrete mass (kg)

$V_G$  = volume of pumping station below water table ( $m^3$ )

The pumping station must be vented to prevent the build-up of toxic or explosive gases. If there is risk of freezing, the upper part of the pumping station can be insulated.

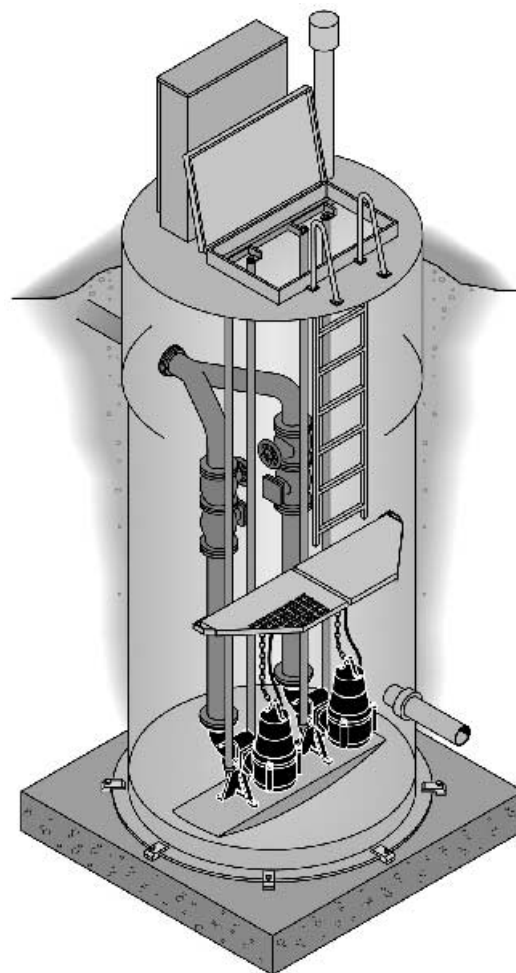
Packaged pumping stations are fitted with access covers that may be made of aluminium or galvanized steel and moulded into the structure. The internal pipework can be either cast iron or thin-walled stainless steel with fabricated bends and branches. Valves should be cast iron and suitable for use in either horizontal or vertical position. Figures 86...88 show typical package pumping station arrangements.

Fig. 86



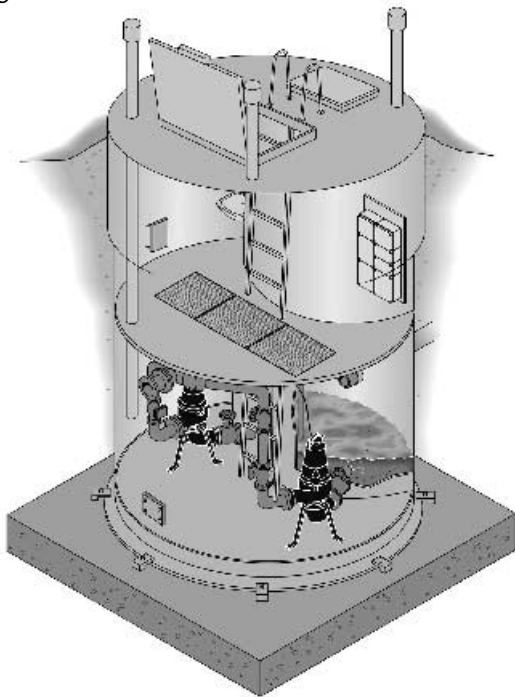
*Package pumping station with separate above-ground service building. Wet well collar serve as foundation for the building.*

Fig. 87



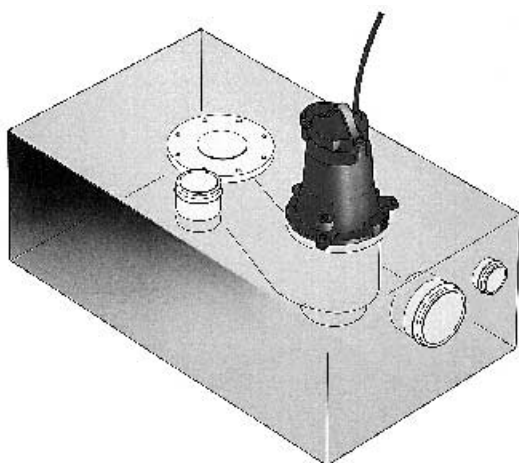
*Typical package pumping station. The wet well is complete with folding work platforms for valve access and service.*

Fig. 88



A package wet well-dry well pumping station. Round wet well shape adds strength and facilitates manufacturing. Dry-installed submersible pumps are safe against flooding and are easy and clean to maintain. Intermediate platform offers access to control panel, mounted above flood level.

Fig. 89



Pumping station for small flows. The pump may be integrated into the structure of the container and can easily be removed. Air-tight construction is suitable for indoor installation, and the unit can be installed near the effluent source.

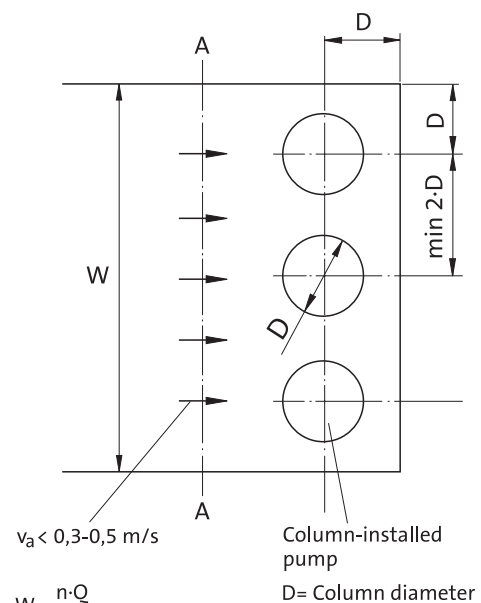
### 5.2.2 Indoor Pumping Stations

Pumping stations for very small capacities can be installed indoors, for instance in basements of buildings close to the source of the effluent. These may be designed as containers with the pumps integrated or mounted externally. Figure 89 shows a typical arrangement.

### 5.3 Pumping Stations with Column-installed Pumps

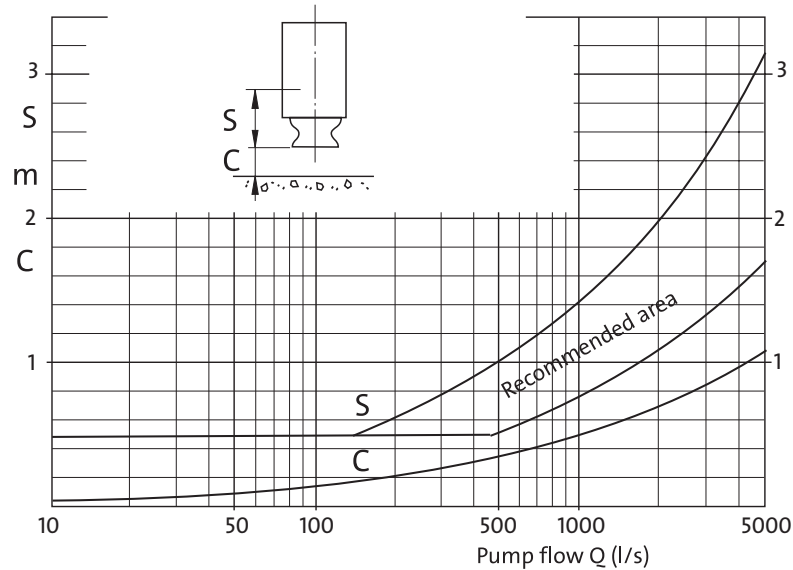
Pumping stations with column-installed pumps typically have large pumping capacity, and especially axial pumps are sensitive to the conditions in the suction chamber. Figure 90 shows recommended distances between pumps and between pumps and wall sections. It is very important that the flow feeding the pumps is even and that the flow velocity at this point does not exceed 0,5 m/s.

Fig. 90



Pump distance and suction flow velocity recommendations for column-installed pumps.

Fig. 91



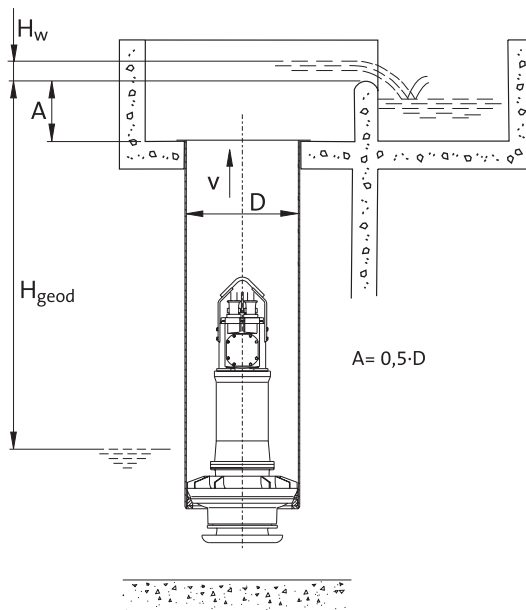
*Pump immersion depth recommendation according to proposed CEN (draft) standard.*

Pump immersion must be sufficient in order for suction vortices to be avoided. Figure 91 shows a dimensioning recommendation diagram according to the proposed new CEN standard (draft). The appearance of suction vortices is still impossible to completely predict beforehand. Pump characteristics and flow conditions in the suction chamber influence the development of suction vortices. The suction chamber shape may induce vortices

to appear unexpectedly. A vortex can be prevented by placing a float on top of it, if possible.

Pumps are frequently installed in columns in a manner where the water exits straight upwards through the column. In these cases the pump head and energy use can be changed by the top design. A good working design is shown in Figure 92. The pump column is terminated well below

Fig. 92



$$H \approx H_{\text{geod}} + \frac{v^2}{2g} + H_w$$

$$H_w = 0,6 \cdot (Q/b)^{0,7} \quad \text{weir edge rounded}$$

$$H_w = 0,8 \cdot (Q/b)^{0,7} \quad \text{weir sharp-edged}$$

$$g = 9,81 \text{ m/s}^2$$

$$v = \text{flow velocity [m/s]}$$

$$Q = \text{flow [m}^3/\text{s]}$$

$$b = \text{weir width [m]}$$

$$A = 0,5 \cdot D$$

*Top outlet design and head determination for column-installed pumps.*



the weir, allowing the flow to smoothen before flowing over the weir. In this design the pump head can be calculated with workable accuracy using the equations in Figure 92. Losses in the column can be disregarded in practical examinations.

### 5.4 Pumping Station Dimension Selection

Pumping station dimensioning is based on the expected incoming flow, which usually must be estimated without the use of collected data. Guidance values cannot always be applied, since flow rates depend on a great number of variables. Figures are available from the sewerage systems designers or, less accessible, from technical literature on the subject. The possibility of sewerage system future expansions must also be considered as reserve capacity or flexibility in pump size installation. Sewage pumping station incoming flow is also typically greatly varying with time, both in short and long cycle.

Incoming flow estimation always starts with the analysis of the possible constituent parts. These are normally classified as

- residential sewage
- industrial effluent
- stormwater (rain and melting snow)
- leakage water

Of these, leakage water is water entering the sewerage system from ground water leaks, leaking water mains or stormwater inadvertently entering a separate sewage system through manholes or other entrances, such as worksite excavations.

To correctly dimension a pumping station, the type of use must be known. Sewerage systems are classified as

- sewers for regular sewage, receiving domestic and/or industrial effluents only
- stormwater sewers, handling stormwater only
- combined sewers, handling both regular sewage and stormwater in various proportions.

#### 5.4.1 Regular Sewage Pumping Stations

Flow rate estimation of residential sewage is normally based on population numbers. The flow rate varies in daily and weekly cycles, the variation being in the range of 0,5...1,5 times average flow. Industrial effluent must be estimated on a case by case basis, depending on the plant type in question. The amount of leakage water present depends on a number of variables, such as water table level, local rainfall and soil characteristics and general condition of the subterranean pipework. It can be estimated as units per pipeline length unit, e.g. kilometre or as a ratio related to quantity of the sewage.

The possibility of flooding at the pumping station with consequent environmental damage must be taken into account. For this reason sewage pumping stations have two pumps in duty-standby configuration, with each pump capable of handling peak flow. Thus flooding will not occur in situations when one pump is out of order or being shut down for service. If pumping station capacity is based on two pumps operating in parallel, a third pump should be provided as standby. Estate or other private pumping stations may be equipped with a single pump, since the incoming sewage flow can easily be controlled by restricting the usage of facilities.

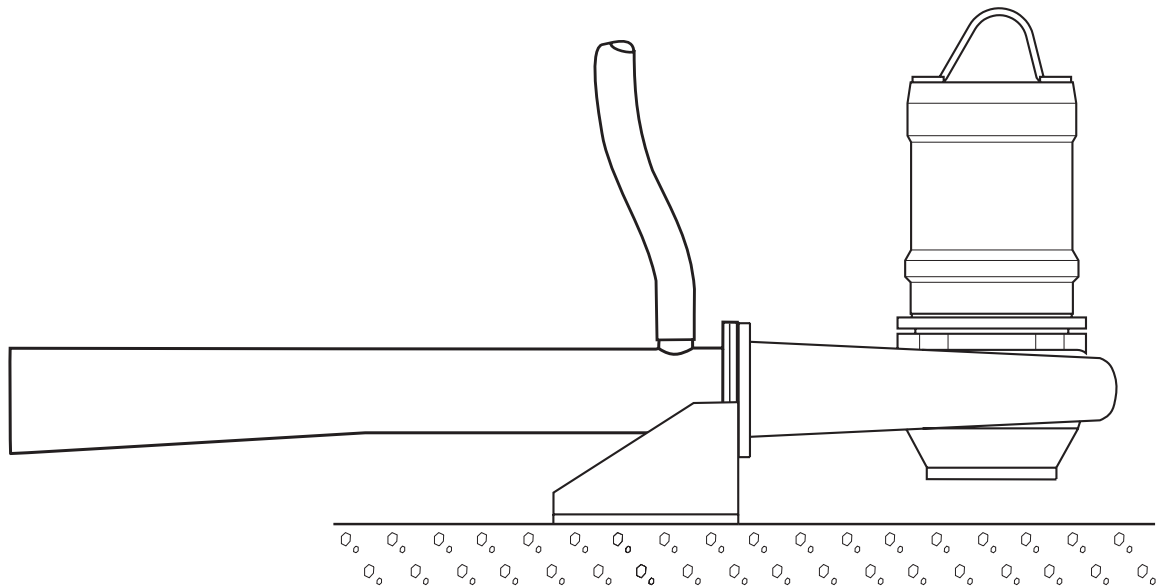
#### 5.4.2 Stormwater Pumping Stations

Rain water flow rates are considerably larger than other stormwater sources, such as melting snow. Dimensioning of the system is based on the largest anticipated amount that will reach the pumping station. This amount may not necessarily be the most severe torrent, for stormwater sewers are allowed to flood under heavy rain circumstances because of the relative harmlessness of rainwater. The design values are also affected by flood tolerance of the area and the type of urban environment in question. Leakage water addition must also be considered.

Controlled flooding at the pumping station is arranged with overflow weirs, discharging in a suitable direction, such as a ditch or canal.

Stormwater pumping stations do not have the same required reliability factor as sewage pumping stations, and they can be designed to handle the maximum flow with all pumps running in parallel duty.

Fig. 93

*Ejector and pump for retention basins.*

### 5.4.3 Combined Sewage Pumping Stations and Retention Basins

The dimensioning flow rate for combined sewage pumping stations is the sum of the estimated sewage, stormwater and leakage water flow rates. Reliability requirement is the same as for sewage pumping stations, making it necessary to dimension them with at least one pump as standby. Combined sewage pumping stations combine in an unfavourable way the properties of regular sewage and stormwater pumping stations, and their use is therefore discouraged.

In conjunction with both stormwater and combined sewage pumping stations, retention basins may be used for temporary storage of incoming sewage that exceeds the installed pumping capacity. As the flow decreases (such as after heavy rainfall), the basin is emptied by pumping or through gravity sewer, and normal pumping station operation can be resumed. Retention basins may also be used to even out fluctuations of the incoming sewage flow to a treatment plant. Essential in a retention basin is to prevent solids from settling onto the basin floor when emptying. This can be accomplished by designing the basin shape “self-cleaning” or by agitating and mixing the basin content. Special ejector mixers have been developed by pump manufacturers,

consisting of an ejector drawing air from the surface combined with a submersible pump. The design is shown in Figure 93.

The air provided by the ejector makes the mixing more effective at low water depths. The required pumping power can be estimated at approximately 70 W per m<sup>2</sup> basin bottom area. The ejectors should be placed and directed so that the jets flush the solids towards the basin drain.

### 5.5 Pump Selection

#### 5.5.1 Pump Selection Based on Pump Curves

The pumps for a pumping station project are initially selected using the methods described in Chapter 3 of this book. It is good practice to consider a number of pumps from a manufacturer's range that have curves passing near the desired operation point.

Pumps having curves both above and below the initial requirement should be included, because other considerations, such as pump efficiency and cost may be economically decisive factors.

The intermittent character of sewage pumping station operation allows a wide margin for the pump selection, giving the designer freedom of choice beyond a fixed nominal point of operation. Theoretically calculated operating points are uncertain in any case, since actual head may vary due to changing start and stop levels brought forth by programmed level control, pump wear and tolerances in pipeline and pump characteristic curves.

For instance, a pump with a head curve higher than originally desired may offer better overall economy, especially if the pipeline characteristic curve is flat, or the dynamic losses are small compared to geodetic head.

The pumping station designer is therefore well advised to select a pump from the manufacturer's standard range and to refrain from requesting exactly tuned pumps. The use of standard pumps will also simplify pump spare part service and later pump replacement, if needed.

The pump selection should be checked so as to make sure that the operating point under any circumstances does not fall outside allowable range of the pump curve. Operating range restrictions can be imposed for a number of reasons such as risk of cavitation or vibration, or overloading. The following should be checked:

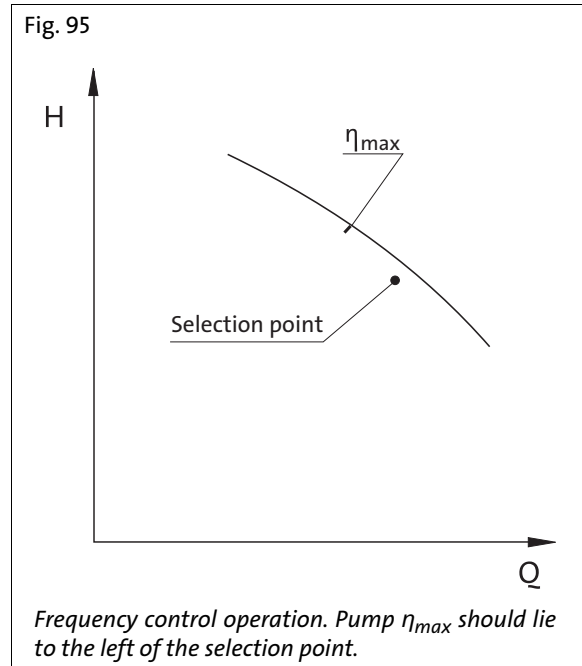
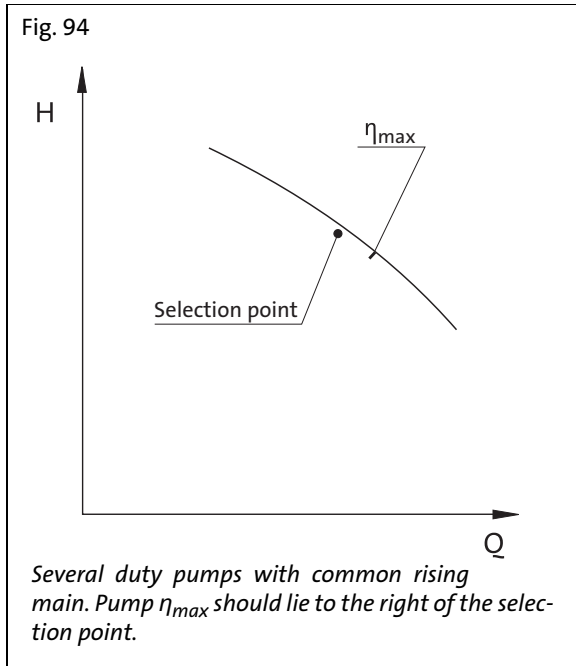
- Single pump duty points in installations with several pumps pumping in parallel into common rising main. Duty points for situations with one, two and more up to and including all pumps operating in parallel.

- Effect of liquid level variations on pump duty point. Liquid level may vary in suction sump as well as in discharge reservoir. If the pump duty points moves into the cavitation area ( $NPSH_r > 10\text{ m}$ ) because of rising suction level and thus increased suction head, the pump can normally be used without consequences, because the  $NPSH_A$  will increase correspondingly. Pump cavitation is thus prevented, and only pump power requirement and available motor power must be confirmed. It is recommended that the pump manufacturer is consulted in uncertain cases. It is especially important to check all possible level combinations for propeller pumps, since these have very narrow allowed Q/H bands, because of strongly varying power curves.

#### 5.5.2 Observing Pump Efficiency

With larger pumps, the pump efficiency becomes increasingly important for the pump selection. When warranted, whole life costs calculations should be performed for a number of alternatives. Please refer to Section 7. All duty points at different duty situations should be taken into account. The following three different cases should be examined separately:

- A. Two pumps installed in duty/stand-by configuration, or all pumps have separate rising mains.  
In these cases the pumps have only one duty point (if the variations in suction liquid level are disregarded), and the situation is fairly easy from an efficiency point of view. Pump selection should not be based on having a pump Q/H curve passing near the desired duty, if the pump best efficiency point falls far away from it. Another pump with a Q/H curve passing above the desired duty point, but with a far better efficiency may be found in the same price range, and be a far better choice.
- B. Several duty pumps with common rising main.  
In this case the pumps may have several duty points, depending on number of pumps in operation. Normally the duty point is selected for the situation when all duty pumps are in operation at the same time. In order for the pump efficiency to be as good as possible with less pumps running, the pump should be chosen so that pump best efficiency point lies to the right of the main duty point, please see Figure 94.



- C. Pumps used with frequency control.  
In order for the efficiency to be acceptable also at low frequencies, and for the  $Q_{\min}$  to be small enough, the best efficiency point should lie to the left of the main duty point. Please refer to Figure 95.
- D. Several duty pumps with common rising main and frequency control.  
For this case it is likely that the best choice is a pump with the best efficiency point coinciding as closely as possible with the main duty point.

Frequently more than one pump, even from the same manufacturer, may be considered for a desired duty point. One alternative may offer lower costs but have lower efficiency than another. The decision between these pumps should in principle be based on a whole life cost analysis. This assessment will frequently have to be performed by the customer or his consultant, since the pump manufacturer usually does not have all relevant information. The position of the buyer may also be significant, since a contractor may stress purchasing price over operating costs, whereas the owner will look at total costs.

Unfortunately, the pump manufacturer frequently has to select and offer pumps with very little or no information on the project, and pump selection may therefore not be optimal, or it may

even be incorrect. In particular, information on the following items is important:

- Will more than one pump use a common rising main? In this case the rising main characteristic curve or the number of duty pumps and value of geodetic head are needed.
- Information on frequency control usage.
- For pumps in column installation information on nature of liquid is needed, in order to ascertain the possibility of using axial propeller pumps.

### 5.5.3 Number of Pumps

Sewage pumping station pumps are selected so that at least one pump always is on standby. Especially in larger pumping stations, the number of pumps should be selected so as to optimize pump usage and investment cost. The cost of pumping capacity, or pump power in kW, decreases with increased pump unit size. On the other hand, the requirement of one standby unit will increase the cost of redundancy if few very large units are used. Installation costs are therefore almost constant for a given capacity, regardless of number of pumps used to meet the requirement, at least within a reasonable range. Likewise, the energy cost will remain almost constant, if the pumps considered can run near the optimal operating point.

Factors affecting pump number selection may also be the requirement of even or continuous output, which is easier to accomplish with a large number of pumps.

Unless special requirements are put forth, the optimal number of pumps for most small to medium size pumping stations is two.

In pumping stations with several pumps it is normally good practice to select identical pumps only. In some special cases where the incoming flow fluctuates randomly and to a great extent, e.g. as a consequence of rain storms, it may be sensible to install larger pumps that run only during peak situations.

The effect on investment cost by variations in the number of pumps installed for a given pumping requirement may vary from one manufacturer to another, since pump size increments are different for different manufacturers. Thus increasing the number of pumps may lead to a more favourable installation for one manufacturer and to a more expensive one for another. Where a multitude of pumps are required for the operation of a pumping station, the final number could be left open for the bidding manufacturers to decide within given limits. Thus a larger number of bidders are likely to be able to offer competitively.

## 5.6 Special Considerations

### 5.6.1 Pump Vibrations

Most sewage pumps vibrate, at least to some extent. Vibrations are caused by residual mechanical unbalance of the rotating parts, pressure pulsations incited by the impeller vanes and the hydrodynamic radial force caused by the fluid mass rotating with single-vane impellers. Vortex pumps vibrate much less, since they do not induce pulsating pressures. For pumps with volute casings the residual imbalance is negligible as compared with the other vibration factors. Improving already good balancing procedures by the manufacturer does not have a measurable effect on vibrations in the pump.

Sewage pump impellers (except for vortex impellers) induce higher rates of vibrations than impellers for clean water, because of the small number of vanes and large channels. Pump installation method also has an important impact on the vibration level. A submersible pump resting on a baseplate stays in place by its own weight only, which increases vibrations as compared to a fixed installation. A vertically installed dry pump may vibrate more than a horizontally installed pump because of a different support structure. The suction bend required may also enhance vibration levels.

Vibration prediction and calculation information for sewage pumps is available in the EuroPump publication "Guide to Forecasting the Vibrations of Centrifugal Pumps", 1992 EuroPump. The values presented in this book are guide lines and valid when measured at the main bearing closest to the pump impeller. Any vibration velocity above 10 mm/s (RMS) measured at this point indicates an abnormal situation in the pump. The reason can be clogging of pump, operation outside allowed section of pump Q/H curve, severe cavitation, and high content of air in liquid or damaged impeller. Possible mechanical imbalance can be controlled by running the pump out of water, when the vibration reading should be less than 2 mm/s (RMS).

Vibration frequency for pumps with volute pump casings equals rotational speed times number of impeller vanes. If the pump or piping is supported in such a way that the natural frequency of these

elements is near the exciting frequency from the pump, the system resonance will increase vibrations. In these cases the support structures must be stiffened. For a frequency controlled pump the system may vibrate more at some frequency because of resonance.

The pressure pulsation induced by the pump moves ahead in the pressure piping with the liquid flow several metres, causing vibrations in the pipe wall. Normal piping vibration levels are below 10 mm/s (RMS). Greater levels may lead to pipe failure. The reason can be inadequate pipe support or resonance.

Column-installed pumps have low vibration levels because of trailing vanes in the casing, effectively dampening the pressure pulsation. A single-vane impeller may still cause pressure pulsations because of the strong hydrodynamic forces induced.

### 5.6.2 Pump Noise

Pumping station noise level is affected by the following elements:

- pump vibration noise
- piping vibration noise caused by pressure pulsation from the pump or other transmitted vibrations
- flow in piping. Pipe bends, tee branches and valves cause disturbances in the flow, emitting noise
- pumping station acoustic characteristics
- inlet stream in wet well
- pump cavitation.

The sound level in the pumping stations composed by all the above constituents, and sound information on the pump alone is not very useful, and cannot be accurately measured on site. Noise emitted from the piping is usually decisive because of the large vibrating emission surface. Correct pump sound level measurement would require the pump to be located in sound insulated space, with the piping on the outside. There are no standards available for allowed sewage pump sound levels. Sound level measuring procedures for submerged pump units is difficult to define because of the practical difficulties involved.

Pumping station noise is not a common problem. A pumping station built in connection with a residential or office building may in some cases cause

noise problems. In severe situations the piping and dry-installed pump motors can be clad with sound-proofing insulation.



## 6 Frequency-controlled Sewage Pumps

### 6.1 General

The reasons for using frequency control are in principle as follows:

- levelling of flow for process-related technical reasons
- energy savings made possible by favourable rising main characteristic curve.

Process-related technical reasons for frequency control are found in the following applications:

- return sludge pumping applications
- recirculation pumps in nitrogen reduction processes
- treatment plant incoming pumping stations.

Alternatively the incoming flow can be regulated by increasing the number of pumps in the final pumping station. With smarter pump control and increased starting frequency the output can be levelled. Treatment plant basin and channel design can also be used to level the flow in the plant. In other pumping stations the use of frequency control should be considered only if major energy savings can be expected.

Frequency control brings energy savings only if the rising main is long and the geodetic portion of total head is less than 40%. In installations with a high geodetic head the energy consumption is bound to rise with frequency control, since the pump duty point will move to a section of the pump Q/H curve, where pump efficiency is lower. Losses are experienced in the frequency control unit and it lowers the pump motor efficiency. Whenever a frequency controller is being considered for the objective of energy saving only, the pay-back period of the investment in the control system should be calculated separately. For this the variations in flow, pump efficiency at different frequencies must be known. The latter is also dependent on the rising main characteristic curve. Also the efficiencies of the frequency controller and the pump motor at various frequencies must be known.

Frequency control increases risk of pump clogging. Should frequency control be selected, a number of conditions and facts should be consid-

ered when designing the pumping station and choosing the pumps.

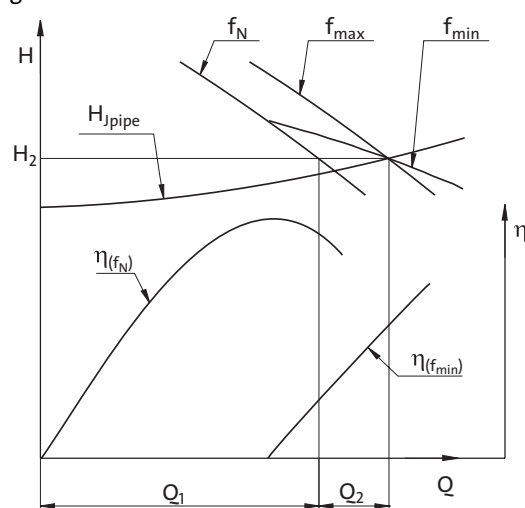
#### 6.1.1 Pump Motor Selection

The supply current modulated by frequency controllers is not perfectly sine wave shaped, causing the motor efficiency to decrease slightly. Normally, however, considering pump duty point and operation conditions, the pump standard motor can be used, provided that nominal supply frequency (50 or 60 Hz) is not exceeded. The choice of motor should be confirmed with the pump manufacturer, since he will have complete information on the motor power and heating characteristics.

#### 6.1.2 Maximum Frequency

In installations with one frequency-controlled pump and one or more unregulated pumps in parallel operation, output control will be irregular at the point where a fixed-speed pump is added to or taken off duty, unless the frequency-controlled pump is allowed to operate at higher than nominal frequency at that point. Typically the required over frequency is 53 Hz for 50 Hz pumps. The situation is shown in Figure 96 below.

Fig. 96



Parallel operation of speed-controlled pumps. Only one pump is speed-regulated at the time.  $Q_1$  = Nominal output of one pump,  $Q_2$  = Regulated output of one pump at minimum speed. Regulated output of one pump is extended to  $Q_1 + Q_2$ , preventing repeated starting and stopping of pumps at full speed.  $f_N$  = Nominal frequency (50 or 60 Hz),  $f_{max}$  = Maximum frequency,  $f_{min}$  = Minimum frequency,  $\eta$  = efficiency.



If all pumps in a parallel installations are frequency-controlled, the situation with irregular control will not arise, and regulation above nominal frequency will not be necessary.

If regulation above nominal is needed, this must be stated on all inquiries, in order for the pump manufacturer to be able to allow for it in the pump and motor selection process. In some cases the use of a standard 60 Hz pump for a 50 Hz installation with frequency control is favourable, but this solution should be weighed against following drawbacks:

- If the frequency controller is out of order and the pump can be run at nominal 50 Hz only, the output of a 60 Hz pumps falls to 50...80% of that of a 50 Hz pump, depending on rising main characteristics and friction losses.
- The motor is likely to have special windings because of the differences in voltages in standard 60 Hz pumps as compared to standard 50 Hz pumps, which will impede future spare parts service.
- Pump efficiency may be lower, since some 60 Hz pumps are converted from 50 Hz by reducing impeller diameter.

### 6.1.3 Minimum Frequency and Minimum Performance

Allowed minimum frequency for a specific pump is often inquired. A comprehensive answer to this question requires information on the installation and rising main, since the shifting of duty point when reducing operating frequency is dependent on pump Q/H curve shape and rising main characteristics. It is recommended that the minimum frequency is worked out from required minimum flow with the help of pump Q/H curve and rising main characteristics.

Pumping sewage at too low flow (too low frequency) may lead to excessive pump wear from sand or other abrasive matter remaining in the pump instead of being pumped out with the liquid. Too small a flow may also lead to clogging of the pump. If the geodetic portion of pump head is high (above 40%), the duty point will move to the left of the pump Q/H curve, where pump efficiency is lower, and energy costs may increase. As a general rule, minimum performance can be limited to 25% of flow at best efficiency at nominal frequency ( $Q_{opt}$ ).

If the rising main characteristic curve is steeply rising, and several pumps may be running simultaneously, it may be necessary to define several minimum pump flow levels depending of number of pumps in use at the same time. The pumps must then be controlled by a suitable, programmable logic device. A minimum frequency determined for the maximum number of pumps running simultaneously, and then used in all situations, with varying minimum performance as result, depending on pumps in use.

### 6.1.4 Pump Frequency Curves

The pump Q/H curves for different frequencies are necessary in order to determine pump performance at various speeds against a given rising main characteristic curve. Minimum frequency must be determined and also pump efficiency at various frequencies. Pump curves for different frequencies are easily drawn based on affinity rules calculations, but possible limitations on curve usage can be determined by pump manufacturer alone. It makes sense to request frequency curves from the manufacturer, with Q, H and  $\eta$  for different frequencies at e.g. 5 Hz intervals.

Frequency curves can be calculated based on the affinity rules using the following equations:

$$Q' = Q \left( \frac{f'}{f} \right) \quad (40)$$

$$H' = H \left( \frac{f'}{f} \right)^2 \quad (41)$$

$$P' = P \left( \frac{f'}{f} \right)^3 \quad (42)$$

$$\eta' = \eta \quad (43)$$

$$NPSH_R' = NPSH_R \left( \frac{f'}{f} \right)^2 \quad (44)$$

The above equations are valid simultaneously for the change of a given Q/H duty point and with an accuracy acceptable for practical considerations.

### 6.1.5 Pump Clogging

In frequency control operation the pump clogging risk increases for the following reasons :

- Liquid level in pumping station stays stationary from accurate pumping control, leading to accumulation of debris on the surface and/or pit bottom. These may be larger than the free passage of the pump, blocking the impeller at pump-out.
- The pumps run continuously for too long periods, preventing the backwash at stopping from clearing pump from debris accumulations.
- The pumps stop slowly as controlled by the frequency controller, preventing the backwash from clearing pump from debris accumulations. Smooth starting also prevents cleansing action.
- Low speed in combination with rising main curves and losses may lead to complicated pump internal flow patterns, increasing clogging susceptibility.

Frequency control installations differ from each other and a comprehensive prediction of pump behaviour is impossible. The risk of clogging may be reduced with the following actions:

- After pump start-up the frequency is controlled in a manner that the suction well level falls steadily and reaches pump stop level in an hour under normal conditions, after which pump is stopped.
- As above, in addition pump is programmed to run at nominal speed for 30 seconds before stopping at stop level. This increases flow in pump sump and rising main, flushing out possible debris accumulations.
- Pump is programmed for 1...2 flush-out sequences per hour by increasing pump speed to nominal for 20 seconds each, after which pump is stopped without frequency control or the frequency is lowered as quickly as possible to minimum frequency, where it is allowed to remain for approximately 20 seconds, after which normal operation is resumed.
- Pump is programmed to run in the wrong direction for some time before each start. This will remove any beginning clogging remaining in the pump from the previous running interval. The frequency when running the pump backwards should be lower than nominal in order to avoid vibrations, i.e. 30 Hz for a 50 Hz pump.

- Using an automatic valve in lieu of non-return valve, and programming it to remain open for some time at certain intervals after the pump has stopped, allowing the back flow to wash out the pump.

When the pump stops suddenly, the amount of water in the rising main continues to flow, efficiently flushing the freely rotating impeller, clearing possible beginning clogging.

The measures above can be accommodated in the pumping station planning stage but taken into use only if necessary.

The use of frequency control for return sludge and recirculation pumps does not increase the risk of clogging. This is also the case for the final pumping station before the sewage treatment plant, if the sewage screening takes place prior to these pumps.

### 6.1.6 EMC Cable Requirement

According to the EU Council Directive on Electromagnetic Compatibility (EMC), frequency controller manufacturers may require that pump motor cables are replaced with EMC compatible and approved cables. The requirement of EMC cables for submersible pumps may complicate manufacturing and add to costs. EMC cables also complicate pump handling, because these cables are less supple than regular supply cables. For submersible pumps the choice of frequency controller may be dictated by requirement of EMC cables or not. Use of EMC cables may be avoided with the use of emission suppressing filters in the frequency controller.

### 6.1.7 Bearing Currents

In some instances frequency control causes interference currents through the bearings of large, air-cooled squirrel-cage motors, causing bearing damage. It is likely that submersible motors are less susceptible to these currents, because they are well grounded by the pipework and submergence in water, and thus protected. This assumption is supported by the Grundfos experience that submersible motors have been free from bearing damages to date. Adding insulation to the bearings would require extensive redesigning of the motors and add to costs.

### 6.1.8 High Tension

For supply voltages above 500 V, frequency control may cause too high voltage fluctuations for standard motors. In these cases the motors may have to be redesigned with special winding insulation and insulated bearings. The use of voltages higher than 500 V are therefore discouraged in combination with frequency control.

### 6.1.9 Explosion-proof Motors

In frequency control the motors may operate at a higher temperature than normally. Thus an explosion proof certification of a motor at nominal frequency may be void for frequency control operation. The ex-proof certification of a pump cable is likely not to be valid for an EMC cable.

If an explosion-proof motor is intended for frequency control, this must be clearly stated in the inquiry documents, in order for the manufacturer to be able to correctly assess pump and motor suitability. Adding frequency control to an existing installation also warrants contacting the manufacturer for clearance.

### 6.1.10 Guaranteed Values

The essential requirement of pump performance is that pump volume rate of flow matches the specified demand and that pump energy costs are under control.

In order to secure total output, the guaranteed duty point should be according to parallel operation of pumps. If the rising main characteristic curve is flat (high  $H_{\text{geod}}$ ) or each pump has its own individual rising main, the same duty point is also suitable for pump efficiency guarantee evaluation. On the other hand, if the rising main characteristic curve is steep or if the geodetic head is fluctuating, the determination of a rational guarantee point for pump efficiency gets difficult. The guarantee duty point of  $\eta$  may be different from that of volume rate of flow and head. It makes sense to separately agree what duty point shall be used for evaluation of pump efficiency. This point could be the maximum efficiency point or the intersection point of pump Q/H curve at nominal frequency and the rising main characteristic curve. A duty point at other frequency than nominal may also be chosen as guarantee point. It should be noted that, according to the test standards, test pump speed may differ by  $\pm 20\%$  from

specified pump installation duty speed, and that test bed results are converted to duty point data using the affinity law equations.

### 6.1.11 Tests with Frequency Controller (String Tests)

If the overall efficiency of the combination of pump and frequency controller shall be verified, the situation of the pump manufacturer is difficult. This situation requires exact information on frequency controller efficiency and pump motor efficiency at modulated current, when voltage alternation is different from unmodulated sine wave. These data are device-specific and almost impossible to get accurate information on beforehand, and they must therefore be postulated. The test standard also do not specify tolerance values for total efficiencies measured under these conditions. The frequency controller must also be made available in advance to the pump manufacturer for testing, further complicating things and adding costs. String tests are of little practical value.

### 6.1.12 Collaboration with the Pump Manufacturer

Designing and executing an installation with frequency control of pumps is much more complicated than simple fixed-speed pump installations. The close collaboration between pump manufacturer and client is therefore important already at the planning stage. Guaranteed duty points and testing standard usage should also be agreed upon beforehand at contract negotiations whenever possible.

# 7 Pump Whole-life Cost Evaluation

The pump selection process should comprise a pump life time cost evaluation, including the estimation of all costs of acquiring, operating and maintaining the pumping plant over its forecasted life span. The importance of life cycle cost evaluations and comparative calculations increases with increasing pumping plant size. For instance, the energy costs of operating a mid-size (30 kW) sewage pumps over three years are equivalent to the original pump procurement costs.

## 7.1 General

Pumping plant whole-life costs are needed for project financial and investment feasibility calculations. For instance, in a pumping station renovation project, where old pumps are replaced with new ones, the chief investment assessment criterion is life cycle cost evaluation. Correct long-term calculations will have to take into account predicted energy cost changes, inflation and interest rates in addition to pump life cycle costs. These calculations require financial and project management skills in addition to solid pump knowledge.

Life cycle cost calculations are commonly used for comparison of pumps during purchasing. The alternatives to consider are either different makes or different models from the same manufacturer. In these comparisons the financial elements normally have the same proportional magnitude for the various alternatives. As future changes in energy costs and maintenance labour costs are difficult to forecast, it makes good sense to simplify the comparative calculations to comprise life cycle costs calculations at present-day cost level, without financial analysis. Thus the analysis can be based on two different approaches:

- The whole-life costs are calculated for the different alternatives at present-day cost level, and compared.
- A comparison based on the most inexpensive alternative is performed, where the pay-back periods for those alternatives with lower operating and maintenance costs are calculated.

It should be noted that these methods have fairly large error margins for energy and maintenance costs because these items are based on forecasts, such as future pumped volume and wear rate estimations.

The decision may also be based on reasons of principle or commercial grounds. Environmental aspects may stress energy use and costs. If the pumps are part of general contract and purchased by a contractor, purchase price alone may be decisive.

## 7.2 Calculation Period

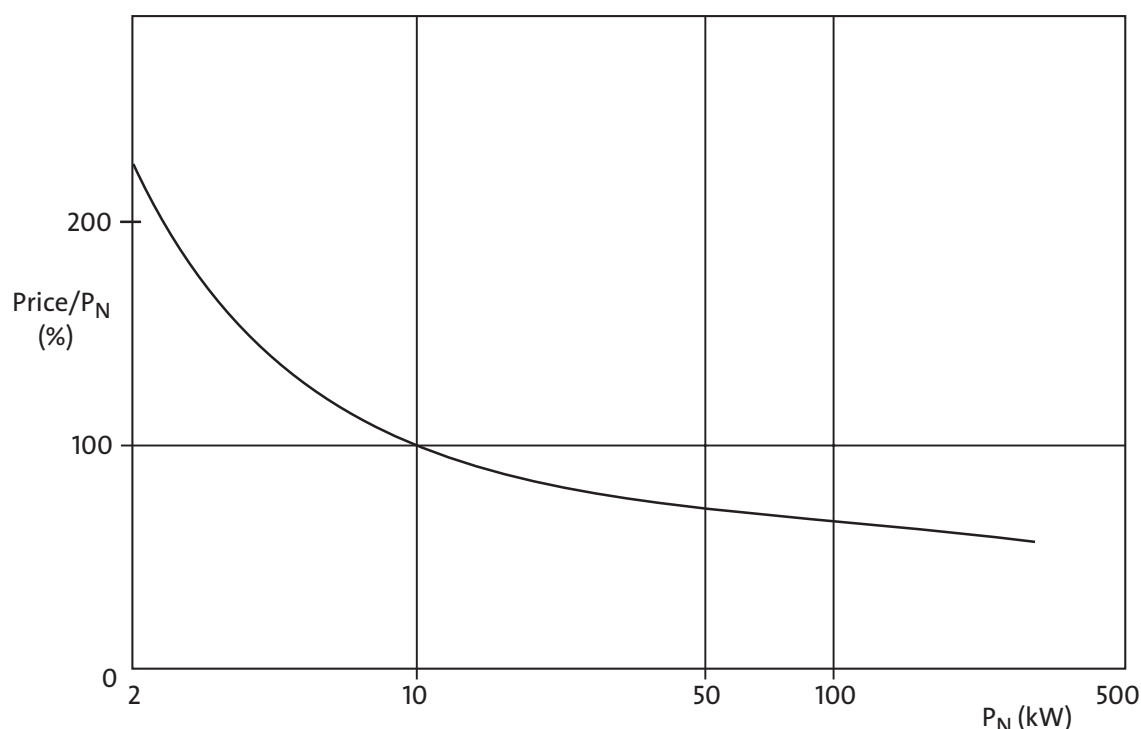
The useful life time of modern sewage pumps is in the magnitude of 25 years. A pumping station may become due for renovation much earlier, e.g. if changes in the neighbourhood lead to increased pumping needs or zoning measures call for its abolition or relocation. Also the unavailability of spare parts may cause early pump obsolescence. A suitable period for whole-life economic calculations is therefore 8...10 years from commission.

## 7.3 Investment Costs

Pump purchase prices are obtained from the manufacturers by inquiry or negotiations. Final price may also include other commercial and purchasing costs, such as transportation. Also the effects of different pump specifications on other acquisition costs must be considered. A larger motor may, for instance, require a frequency converter or mains supply fuse of higher rating, adding investment costs.

Figure 97 shows the proportional effect of pump size on pump cost for pumps of 1500 l/min nominal speed. A pump with lower nominal speed will generally be more expensive than a pump of the same rating running at higher speed, because of larger size. The figure shows that for small pumps of less than 10 kW rating, purchase price will be decisive for the life cycle costs.

Fig. 97



Effect of pump size on pump specific cost Price/ $P_N$  when pump nominal speed is 1500 1/min relative to 10 kW pump.

## 7.4 Energy Costs

The energy requirement is correctly calculated using the efficiency ( $\eta_{gr}$ ), since efficiency is subject to the manufacturer's guarantee according to the testing standards, whereas the information on power is not. It is important that the testing standard to be used is agreed upon at this stage, since different testing standards have different tolerances for pump efficiency, which can affect the efficiency figures reported by the manufacturer. Please refer to Section 4, Testing of Pumps, for more information.

### 7.4.1 Efficiency Over Time

With the exception of vortex pumps, the pumping efficiency of sewage pumps deteriorates over time because the clearance between impeller and suction cover widens from wear. This change should be taken into account when performing energy usage calculations. Based on tests and experience, the following efficiency reductions

factors can be used when performing calculations:

- Closed impeller with adjustable suction clearance: -1,5% ( $\eta_{gr}$  points)
- Semi-open impeller with adjustable impeller clearance: -3,0% ( $\eta_{gr}$  points)
- Closed impeller without adjustable clearance: -3,0% ( $\eta_{gr}$  points)
- Semi-open impeller without adjustable impeller clearance: -5,0% ( $\eta_{gr}$  points)

The efficiency reduction factors above suggest that in practise sewage is pumped with pumps having significantly lower efficiency than new pumps. The higher values for pumps without clearance adjustment possibilities are based on the fact that these pumps run longer intervals between clearance restorations, because parts have to be replaced in shop conditions. The effect is greater on pumps with semi-open impellers, since these pumps wear faster and the efficiency

## 7 Pump Whole-life Cost Evaluation

is more sensitive to changes in the clearance (see section 2.2.1 Impellers).

### 7.4.2 Energy Usage Calculations

The energy calculation can be performed using two different methods:

- Appraisal using yearly pumped water volume, first computing specific energy, using the following equation:

$$E_{sp} = \frac{g \cdot H}{\eta_{gr} \cdot 3600} [\text{kWh/m}^3] \quad (45)$$

where

H = pump head at duty point [m],

g = 9,81 [m/s<sup>2</sup>],

$\eta_{gr}$  = overall efficiency (pump + motor) at duty point [decimal value],

Liquid density is assumed to be 1000 kg/m<sup>3</sup>.

The energy consumption is calculated using specific energy and estimated yearly pumped volume.

- Appraisal based on operating hours, first computing power at guaranteed efficiency, using the following equation:

$$P_{gr} = \frac{g \cdot Q \cdot H}{\eta_{gr} \cdot 1000} [\text{kW}] \quad (46)$$

where

Q = pump volume flow at duty point [l/s],

H = pump head at duty point [m],

g = 9,81 [m/s<sup>2</sup>],

$\eta_{gr}$  = overall efficiency (pump + motor) at duty point [decimal value],

Liquid density is assumed to be 1000 kg/m<sup>3</sup>.

The energy consumption is calculated using the power obtained and estimated yearly

operating hours. If the pump Q/H curve passes above the desired duty point the pump output will be higher and consequently, the operating hours fewer. This must be considered when using the pump operating hour method.

The energy calculation methods are fairly simple when the pump is operated in a single duty point. The situation gets more complicated with pumps in parallel duty and if the pump is used with frequency converter. In parallel duty, calculations should be performed separately for the different duty points, and then by approximating the pumped volumes or operating hours accruing in each of these.

With frequency converter the pump has an infinite number of duty points. A duty point, representative of the average pump duty should be selected for the calculations in these cases. Another uncertainty factor when calculating the energy consumption for frequency-controlled pumps is the fact that the overall efficiency of the system is difficult to accurately determine. For comparative calculations the pump efficiency without frequency controller may be used.

## 7.5 Maintenance Costs

Normally submersible pumps are recommended routine maintenance on a yearly basis. Maintenance includes seal oil control, motor insulation control with resistance meter, suction clearance check and, if necessary, adjustment, and general surface inspection. Most manufacturers recommend very similar routines. Distinctions between pumps from different manufacturers are most clear in the possibilities of maintaining and restoring pump efficiency.

If a pump is equipped with an adjustable suction clearance mechanism, the costs for maintaining pump efficiency does not add to costs, since the adjustment can be performed during normal routine maintenance on site. If, on the other hand, pump efficiency maintenance requires that spare parts be used or the pump brought to shop, the costs for these measures will have to be taken into account when the whole-life costs of the pump are calculated.



## **7.6 Cooperation With Pump Suppliers**

Whole-life cost calculations and comparisons are seldom completely unambiguous, and it is therefore reasonable and fair to perform these openly and in cooperation with the pump vendors, at least when considering a pump from that supplier. This way possible misunderstandings can be avoided, and suggestions and alternatives as proposed by the supplier can be taken into account for best possible selection.

## **7.7 Life Cycle Cost Publication**

The pump manufacturers associations Europump (Europe) and Hydraulic Institute (USA) have jointly published a guide for pump life cycle cost (LCC) evaluation:

PUMP LIFE CYCLE COSTS  
A GUIDE TO LCC ANALYSIS  
(ISBN 1-880952-58-0)

This publication deals with the complete pumping system from the design stage, existing pumping systems and examples of implemented improvements.



# 8 Commissioning

During pump commissioning, the following items should be inspected:

- Check duty point(s) using pressure gauging and possibly flow metering, using the volumetric method, for comparison of these with projected values and to confirm that true duty point lies within allowed limits of pump Q/H curve.

In long rising mains with several high and low points on the way to the discharge point, the true situation may take some time to stabilize. The measurements should therefore be repeated after some time after commissioning to confirm duty point.

- Check pump operation for vibrations and noise. Check for signs of cavitation.
- Compare start and stop levels with projected values and adjust if necessary. Lowest possible stop level for dry-installed pumps should be found by trials, observing suction of air into inlet pipe.

For submersible pumps combined with large volume rates of flow, check for surface vortices at low level. Adjust stop level where necessary.

## 9 Operation and Service

Submersible sewage pumps should be subjected to routine inspection and service yearly. The scheduled maintenance should be performed on site and includes

- Oil check and replacement when necessary.
- Suction clearance (impeller/pump housing) inspection and adjustment, if the clearance has widened to 2 mm or more from wear. For pumps without adjustment possibility the restoration of the suction clearance and pump performance requires installation of new parts.
- Motor insulation resistance metering at control panel.
- Checking of lifting chain and lifting eyes and lugs.
- General pump inspection and operation control.

The pump operator's manual provides thorough information on maintenance.

Routine maintenance can be performed by the owner or by contract service company. Impeller replacement should be possible on site during maintenance, if necessary. Shaft seal replacement and other work on the watertight pump motor enclosure should always be referred to an authorized workshop.

Spare part availability is not a problem for submersible pumps made by recognized manufacturers. The pump manufacturing series are long and parts kept in stock both for assembly of new pumps and for spare parts service. Pre-stocking of spare parts is normally not warranted.

### 9.1 Safety

The most important risk factors associated with the operation of sewage pumps relate to the following:

- electricity
- lifting and handling of pumps
- hot surface temperatures of dry-installed pumps
- handling of pump parts during service and repairs
- incidents of fire and explosions in hazardous environments
- health risks from human contact with sewage.

The following international standards deal with pump and pumping safety issues:

- EN 809 (1998)  
Pumps and pump units for liquids – common safety requirements
- prEN 13386 (1999)  
Liquids pumps – submersible pumps and pump units – particular safety requirements (proposal 2002)

# 10 Pumping Station Control and Condition Monitoring

All sewage pumping stations, either working individually or as a part of a sewage network comprised of several pumping stations, should be reliably controlled in order to provide safe and efficient operation. Modern electronic control technology offers possibilities to design and build versatile control and condition monitoring systems to reduce long-term operation costs and to increase operational reliability.

Unreliable sewage pumping stations represent an ecological as well as economical risk in the form of waste water overflowing into the environment or basements of buildings. Reliability is therefore the prime concern in design of a control unit for waste water pumping station.

This chapter describes the sensors the reliable pumping station control is based on, different control methods concentrating on the modern state-of-the-art control technology and finally a network level remote control and monitoring system and its future possibilities combining Internet and WAP technology.

## 10.1 Local Control Methods

Local control is always needed at site at the pumping station to control the operation of the pumps. The local control unit can be built to different technical levels according to requirements of control features as well as costs.

### 10.1.1 Manual Control Units

Manual control is the simplest control method. It consists merely of a switch (normally manual-off-auto) with the necessary relays and circuit breakers to start and stop the pumps. Manual control is normally never used as the primary pump control, but is used as a backup control method during malfunctions of the regular controls and during pump repair and maintenance work to check the pump operation. Possibility for manual control should always exist.

### 10.1.2 Relay-based Control Units

In case pump condition monitoring is not required, automatic relay-based control units for local control purposes can be used. Relay-based controls are simple units with fixed or adjustable start and stop levels. They may include sequencing of multiple pumps, or this may be accomplished with additional pump sequencing units.

In case continuous level measurement is used, these control units may have freely adjustable start and stop levels and local level display. In most cases, however, relay-based control units use preset or manually adjustable level switches, such as float switches.

Relay-based control units are both easy to use and reliable due to simplicity of design. They are suitable for small or secondary pumping stations where little or no operational flexibility is required.

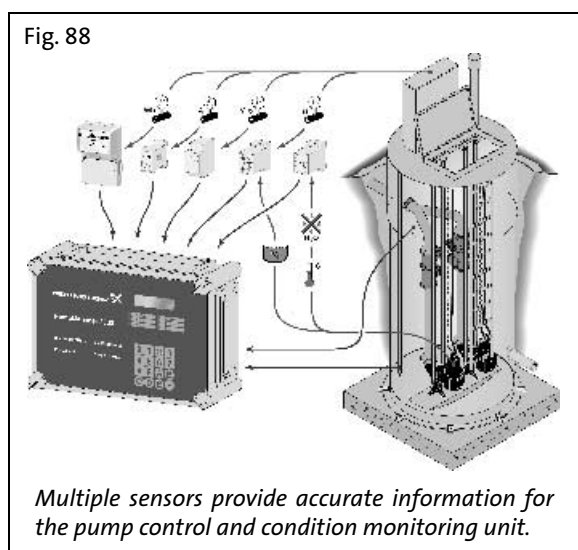
### 10.1.3 Programmable Logic Controllers

Pump control units based on programmable logic controllers (PLCs) offer extensive possibilities for pump condition monitoring, data logging and analysis as well as flexible pump control. Designing of a good pump control unit based on PLCs is demanding and always requires solid knowledge of the operation and requirements of a sewage pumping station, in addition to programming skills. Selection of control and measurement signals, pump and pumping station analysis and choice of level measurement sensors are among the things that has to be considered.

## 10.2 Sensors for Pump Control and Condition Monitoring

The pump control unit, simple relay-based or versatile PLC, requires different sensors to gather information from the operation of the pumps as well as the whole pumping station as shown in Figure 88. Such sensors provide information on wet well water level, current consumption of the pump, condition of the pump primary seal as well as motor coil insulation and so on.

Fig. 88



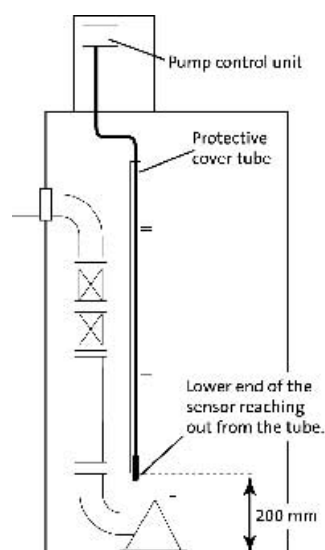
### 10.2.1 Wet Well Water Level Sensors

The basic information required by any automated pump control system is the water level in the wet well of the pumping station. There are multiple ways to provide this information as there are different types of information available as well. Depending on the sensor, the current water level is given as continuous analog signal or on/off information as the water level passes certain, normally preset, height positions.

By experience, **pressure transmitters** offer the most reliable and economic way for continuously measuring the water level in sewage applications. Especially, a piezo-resistive pressure sensor, either embedded in a stainless steel enclosure or integrated in a sealed liquid-filled rubber construction, is excellent for use in waste water. Pressure transmitters provide continuous analog current (0...20 mA or 4...20 mA) or voltage (0...45 mV) signal proportional to the water level.

The function of pressure transmitters is sensitive to sedimentation, but this can be avoided by correctly installing the transmitter inside a protective pipe as shown in Figure 89.

Fig. 89



*Correct installation of the level transmitter is essential. The completely sealed transmitter is suspended in the wet well and its piezo-resistive element transmits the level signal to the control unit. The sensor is used with electronic pump control units and provides continuous level reading.*

**Ultrasonic devices** are the only choice if the level sensors cannot be in contact with the liquid. Modern ultrasonic sensors have built-in programmable functions for various operating conditions and ranges. Ultrasonic sensors are also quite expensive.

Ultrasonic sensors are normally accurate and reliable. On the other hand, in waste water applications steam and foam on the liquid surface may cause level indication errors or complete loss of echo, which can lead to interrupted level monitoring. Problems arising from such situations can be avoided with the installation of backup devices for the most vital functions, such as a float switch for high level alarm.

Some earlier level sensors were based on **capacitive sensors**. This kind of sensor is also installed in

the wet well and consists of a rubber or plastic bladder connected to a vertical pipe and containing a reference liquid. A wire is suspended in the pipe and connected to a signal transmitter. The level of the reference liquid in the pipe rises and falls with the level of the content in the wet well. The capacitance of the wire-pipe element changes accordingly and the signal is transformed in the transmitter to a suitable signal for the pump control unit.

Capacitive devices are normally reliable but exposed to malfunctioning due to accumulation of sediments on the bladder at the bottom of the wet well. The device also needs careful installation and more service than pressure transmitters.

**Float switches** have been used for level control in waste water pumping applications for many years. They provide the simplest means for level control at fixed levels, but do not provide any possibility for continuous level control. In multiple float switch installations there is always a risk for the control wires getting entangled in each other or the pump cables. Float switches, together with a relay-based control unit, are used today mainly in small installations.

On the other hand, due to their simplicity and reliability float switches are even today quite often used as a backup or emergency level control system in larger units, too. This usage provides emergency operation in case of the main level measuring equipment failing.

**A level bell** is another very simple level sensing device. It consists of an upside-down positioned plastic cone with air tube between the upper end of the cone and the controller. As the water level reaches the cone, air inside the cone and the tube is compressed causing pressure against a switch in the controller. As the pressure rises, the switch contacts and the pump starts. With this kind of devices the pump is usually stopped after a pre-set delay.

### 10.2.2 Current Sensor

The pump input current is monitored by a current transformer through which one of the three mains phase cables is routed. Each pump requires one current transformer for adequate monitoring reliability. Current transformers provide an analog signal (0...20 mA or 4...20 mA) proportional to the pump input current.

### 10.2.3 kWh Meter

Pumping stations with a modern electronic controller should always be equipped with a kWh meter, which has a potential free pulse output. As the meter provides a certain amount of pulses for each kilowatt hour used by the pumping station, the energy consumption can be monitored.

### 10.2.4 Phase Failure Relay

All the three mains phases are connected to the phase failure relay. This device provides an alarm signal in case loss of power in the pumping station.

### 10.2.5 SARI 2 Monitoring Device

The Grundfos SARI 2 is a combined monitoring device for motor insulation resistance and seal oil water content. The motor insulation resistance is measured between one of the mains phases and ground when the pump is stopped and disconnected from the mains supply. Low insulation resistance indicates moisture inside the motor, which could lead to the motor burning and expensive repair work.

In case the pump is equipped with a Grundfos OCT 1 oil condition transmitter, the SARI 2 also continuously monitors the water content in the seal oil chamber. As the primary shaft seal wears in time and water leaks into the oil chamber, the OCT 1 probe indicates the water content in the seal oil. This information is routed to the SARI 2 monitoring device, which raises an alarm.

Fig. 90



*Grundfos SARI 2 monitoring device. The SARI 2 is mounted on a DIN rail in the control panel. The SARI 2 is intended for both stand-alone alarm use and for interfacing with a remote monitoring system.*

### 10.2.6 ASM 3 Alarm Status Module

All the Grundfos submersible pumps are equipped with an internal moisture switch as well as temperature switches embedded in each motor coil. These safety devices are connected in series, and if any one of them trips, the controller stops the pump and raises a safety device failure alarm. With the ASM 3 module these two alarms (moisture or overheat) can be separated to provide accurate alarm information.

Fig. 91



*Grundfos ASM 3 alarm status module.*

Fig. 92



*The GRUNDFOS PumpManager electronic pump control and condition monitoring unit. The unit controls all pumping station functions with software stored in a PLC circuitry. Operational parameters are set using the keypad and the LCD display of the unit. Pumping station data logged and calculated by the unit are readable on the display by entering codes on the keypad or from a remote monitoring system. A scanning function allows the operator to get all important data with a minimum of keystrokes.*

## 10.3 Pump Control Units

A modern and versatile pump control unit is based on the use of microprocessors and control software. The unit is likely to be a PLC with a built-in application software for pump control and condition monitoring. The user interfaces with the unit to access necessary control parameters, such as start and stop levels that can easily be checked and adjusted. The complete control unit consists of the electronic controller and an array of auxiliary equipment, such as level sensor, current transformers, and phase voltage relays, etc., forming an integrated package. Figure 92 shows an intelligent electronic pump control unit.

### 10.3.1. Control Features

The main parameter to measure is the water level in the wet well. A continuous level indicator is always used in this type of control unit. Several types of sensors are available, such as a sealed pressure transformer and ultrasonic devices.

The pump control sequence is normally quite simple. In a regular duty-standby application, the pre-set operation levels are stop level, start level and second start level. The duty pump starts when the water in the wet well reaches the start level, and stops when the water has been pumped down to the stop level. The duty pump is alternated at each cycle in order to ensure even distribution of usage and wear between the pumps. The standby pump starts at the second start level in a situation where the incoming flow is larger than the capacity of one pump. If more than one standby pump is installed, these may be started at the same level simultaneously or at adjustable intervals, or at different levels.

All the running pumps are stopped simultaneously when the level reaches the stop level or at adjustable intervals. In some multi-pump installations all pumps may have different start and stop levels. This, however, makes pump condition monitoring calculations more complicated and less reliable.

In some cases a separate overflow pump with different characteristics may be installed to handle large flows. This pump does not participate in the sequencing and should be controlled by a separate unit independently from the other pumps.



**Flow measurement** is possible without a separate flow meter. It is done according to the volumetric method, where the changes of levels in a wet well of known dimensions are measured against time. The unit software calculates both incoming and pumped flows with the same accuracy as that of a magnetic flow meter. The volumetric method is also the basis for the measurement of the **pump capacity**, which is continuously measured as a running average of ten latest pump actions.

In case of an **overflow** from a pumping station, it should be possible to estimate the volume accurately and unambiguously in order for the operator to handle possible claims of damage. When the incoming flow at the time and the duration of overflow is known, the volume is estimated by the unit software for authority reporting.

**Pump motor current** measuring is necessary for protection and condition monitoring purposes. With adjustable over and under current limits, the unit is set to protect the pump motor in abnormal situations. In case the input current rises above the over current limit, where a burnout of the motor becomes a risk due to possible pump failure or clogging, the pump is automatically stopped. Together with embedded thermal relays or electronic motor protectors in the pump motor starter this offers a very reliable motor protection. An abnormally low input current indicates that the pump is not pumping normally, which may be due to pump impeller wear or failure or gas entrapment.

**Pump running hours and number of starts** are information needed for scheduling of pump maintenance. These are also important information in verifying the pumping station operational design and when determining the correct start and stop levels during the commissioning.

All the features described above are available in the GRUNDFOS PumpManager control and condition monitoring unit, and readable from the interface display. This enables the motor control panel to be simple without separate ammeters, hour counters and sequencing relays, features that are all incorporated within the PLC.

With the control units entirely controlling the pumps by software, it is simple to embed special features within the program. Thus it is possible, for example, to let the pumping station **pump**

**below the stop level** to allow pump snoring (controlled dry running) at given intervals. This is to prevent sludge accumulation and cake formation on the wet well surface. In this manner possible pump blockages can be avoided with savings in unscheduled maintenance costs. This has also been found, by experience, to be an effective way to prevent odour problems.

Another possible advantageous operational feature is to let the pump starting level intentionally fluctuate around its setting; this is to prevent a sludge rim from forming on the wet well wall at the start level.

The pump control unit is also programmed to indicate all operation failures in the station, such as alarms for high level, low level, pump power failure, and other alarms based on settings of parameter limits.

### 10.3.2 Condition Monitoring Features

The pump control unit performs **automatic pump condition monitoring** based on the parameters logged and analyzed. When the rate at which the water level in the wet well rises and falls during the pumping cycle is monitored, **pumping capacity** of each pump can be calculated. The unit then compares these values to the pump nominal performance data stored in the memory and raises an alarm in case the performance is outside set tolerance limits.

The benefit of such a system lies in its capability to give early warning for slowly developing defects that ultimately could lead to sudden and unexpected pump failure and consequent environmental damage. Also developing pressure pipe work problems can be detected by closely analyzing pump performance. Another benefit of such a system is the monitoring of the operation from the economical point of view, where maintenance actions can be planned and executed according to need. This finally leads the pump servicing from repair-on-failure to preventive and even predictive maintenance.

### 10.3.3 Parameters and Signals

The pump control unit needs a number of parameters in order to operate as required. The parameters are entered into the unit based on actual pumping station dimensions and units taken from the plans or measured at site. For calibration



either actual dimensions or percentages of reference values can be used. Values to be entered are usually various operating levels such as pump start and stop levels, low and high level alarm as well as overflow levels, which all correspond to certain water level in the wet well. Other parameters usually required are wet well dimensions and pump nominal values for the input current and capacity, which are available from the pump data sheets.

Several signals are necessary for the pump control to operate as planned. These are either digital or analog. Digital signals are either input or output signals and indicate an ON or OFF status. Necessary digital input signals are pump running or standby indication from the circuit breakers as well as potential free contact signals from the phase voltage relay and energy meter when available. Digital output signals are needed for starting and stopping the pumps.

Analog input signals from additional sensors are used for various continuous measurements. These signals are, for example, pump motor winding and bearing temperature measurements, pump sealing house oil condition information, data from an additional flow meter or frequency converter etc. The use of these signals may require an additional extension card and special version of the application software.

### 10.3.4 Data Logging and Analysis

The pump control unit must have sufficient memory capacity to log and analyze data over a certain period of time. The unit has to log at least the hours run, number of pump starts and incidents of abnormal pump motor current. The unit has to analyze and calculate flow, pump capacity and overflow from the logged data. The logged data can be collected and further analyzed by downloading the data at intervals to a portable PC with suitable software, or continuously by an automatic remote control system.

Even in case the pump control unit operates as an outstation of a network level control and monitoring system, it must have memory capacity to store the logged and analyzed data for several days. This is due to the fact that vital data must not be lost even during possible communication breakdown between the outstation and control center.

### 10.3.5 User Interface

To access the data and to enter parameters, the operator needs to interface to the pump control unit. The interface must be at least a small LCD display and a keypad. Using the keypad the user must be able to enter all necessary parameters and read logged and calculated data. The use of such interface must be simple and logical.

Normally some helpful features like an automatic scanning function, makes the routine checking of the data easy and fast. Separate LED signal lights are used for indicating alarms and pump running status.

## 10.4 Remote Control and Monitoring System

Waste water pumping stations are designed to incorporate extra capacity in case of too large an inflow or a pump failure. This reserve volume does not, however, prevent overflow in case the fault goes unnoticed for a longer period. Scheduled maintenance visits alone cannot prevent all contingencies possible in pumping stations; therefore systems for **remote control and condition monitoring** as well as for **alarm relaying** have been developed.

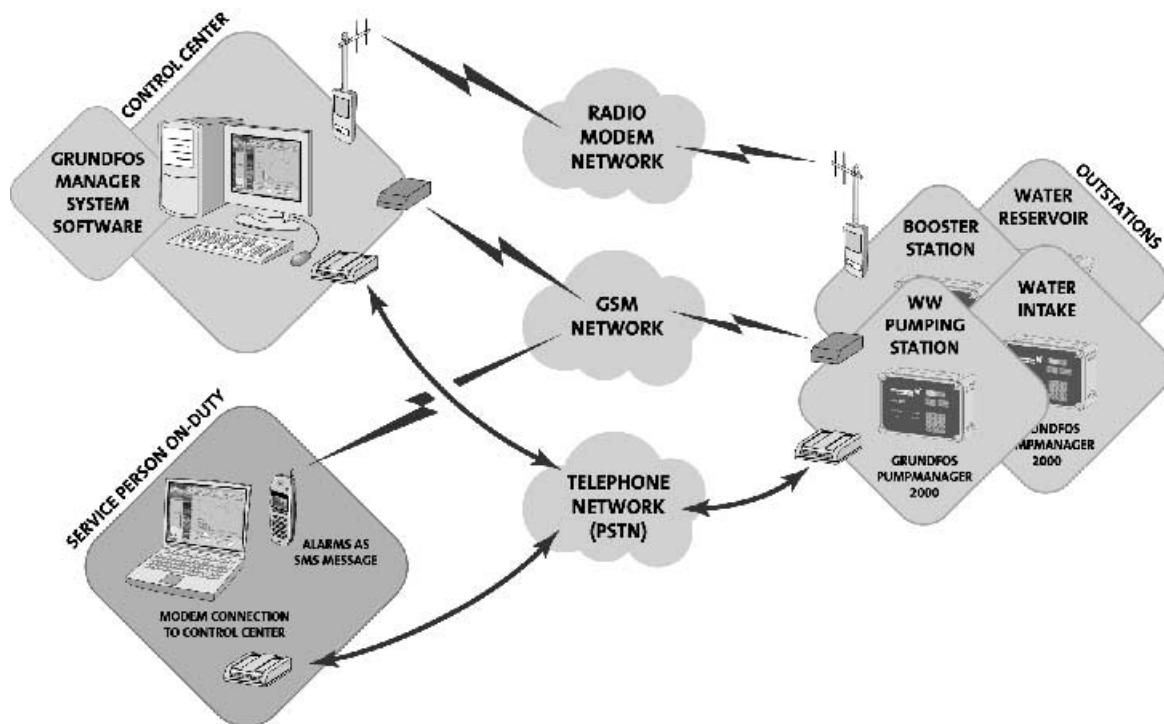
The visible and audible alarms located outside of the pumping station used in the earlier years have been developed into sophisticated and decentralized remote control systems. These latest systems consist of PLC based pump control units controlling and monitoring the local processes in the pumping stations. These control units also function as telemetry outstations and are remotely connected to a central computer where special network level administration software is running.

### 10.4.1 Different Levels for Remote Control

The modern pump control units enable the remote control and condition monitoring system to be tailored according to the features required by the customer in comparison to the available investment funds.

In case a very simple **automatic alarm transfer** is preferred, the control unit can be equipped with a GSM modem, with which the alarms generated by the control unit will be transferred to the GSM phone of the person on-duty as an SMS message.

Fig. 93



*The GRUNDFOS solution for the network level remote administration. The local control units at the pumping stations are connected via normal telephone (PSTN), radio, GSM modem network or any combination of them to the central control station. Also fixed (leased) line cable pairs can be used, but are nowadays getting rare.*

Such a system offers highly increased operational reliability with modest investment costs since there is no control center at all. On the other hand, the modern control unit is capable of utilizing the whole SMS message by adding the most important logged and analyzed information into the alarm message. Such information could be the pump running hours, number of starts, energy consumption, pumping station inflow and pump capacities, for example. In case such a system creates automatic reports and transfers them to the person on duty on a weekly basis even without any alarm situations, the normal driving around to the pumping stations can largely be avoided.

In case a **network level remote control and monitoring system** is introduced, there are several ways to build the communication link between the outstations and the control center as described in the following sections.

### 10.4.2 Software and Hardware

The control center consists of a standard PC workstation, a printer for report printing and the specially designed administration software. The user interface of the software must be mouse-controlled and menu-driven for flexible and easy use. Depending on system configuration, theoretically an unlimited number pump control units can be controlled and monitored by a single control station. Practically the number is limited to about 200 outstations by the time required to gather the observation data from the outstations during night time.

The central control station performs remote control and monitoring, by which real time conditions at the pumping stations can be viewed at any time. This feature largely substitutes for site visits by the operating personnel. Pumps can be started and stopped; levels and other parameters can be changed, and so forth. The system gathers all pumping station observations on a daily basis and stores the data into the databases from

which e.g. the pump and flow data for many years backwards can be monitored for further analysis, if desired. The software also generates numeric and graphic reports on flows, pump data, alarms and other parameters. Figure 93 shows a remote control arrangement.

## 10.4.3 Data Transmission

Although the pump control units operate completely independent, the transmission of data is crucial for the remote control systems to work. The time needed for data transfer can be decreased, if the pump control unit performs all data analysis locally and stores the results in its memory. Only the calculated results, instead of all logged data, need to be transferred to the control center. This also enables the pump control units to operate independently without having to be constantly connected to the control center.

The results can also be stored at the outstation for some period of time, usually one week, before it is automatically sent as a package. This is an important feature in case there are indefinite breakdowns in the communication link.

Data transmission is always configured to suit the individual needs. The communication link must be flexible and normally the public switched telephone network, radio modems, GSM modems or any combination of them can be used. Also fixed cable pairs can normally be used, but they have lately got quite rare due to increasing monthly fees and uncertain reliability. A modem is needed in both ends of the communication link to modulate the data for transfer. The choice between the different transfer methods must be made by the customer keeping in mind the building costs, data transmission costs and features both required and offered by each method.

In general, radio modems and fixed cable pair (leased lines) are used over short distances and always in case there is a need for continuous communication such as with control loops between fresh water reservoir and water intake station. If connected through the public telephone network, the pumping station and the central control station can be situated at a practically unlimited distance from each other. The public telephone network also makes it possible to authorize third parties, such as equipment vendors and service

companies, to access a pumping station for special purposes.

Modern GSM telecom technology offers an attractive solution for remote control and monitoring for far-off outstations with long distances. GSM often offers the best alternative for retrofit installations, since PSTN line access installation afterwards is expensive and availability may be limited. It is a clear trend that GSM modem connections are getting increasingly popular in the future.

## 10.4.4 Alarm Transfer

Alarms raised at an outstation are transferred to the control center, where all incoming alarms are stored in the database. The administration software running on the control center computer includes an automatic categorization of alarms as well as a calendar of the service personnel on duty, according to which it transfers the alarm to the right person at the right time (in case the alarm is categorized to be transferred). Occasionally, the control center computer is also equipped with a separate alarm printer, which prints out all the alarms for later analysis.

The alarms are normally transferred to the GSM phone of the service person on duty as an SMS (text) message. The message may include, in addition to the alarm text and the station name, more detailed information on the pump status (running/off/failure), station inflow, pumped volume during the day, other active alarms (which are stated as not to be transferred), etc.

Another way to transfer the alarms is by pager. The control center computer creates the alarm report text, contacts the pager operator and sends the message to be displayed on the pager. Typically the message contains coded information on the station identity and type of alarm issued.

If so required, the control center can also transfer alarms via voice message. A text corresponding to the alarm and stored vocally on the computer hard disk, is retrieved by the software and used for transmission over telephone to the operator. The control center can be programmed to call different numbers, until the alarm is acknowledged by an operator.

### 10.4.5 System Integration

A wastewater pumping control and monitoring system can be integrated into some other control system, such as a treatment plant control system, or an integrated water company control system, if combined control is preferred.

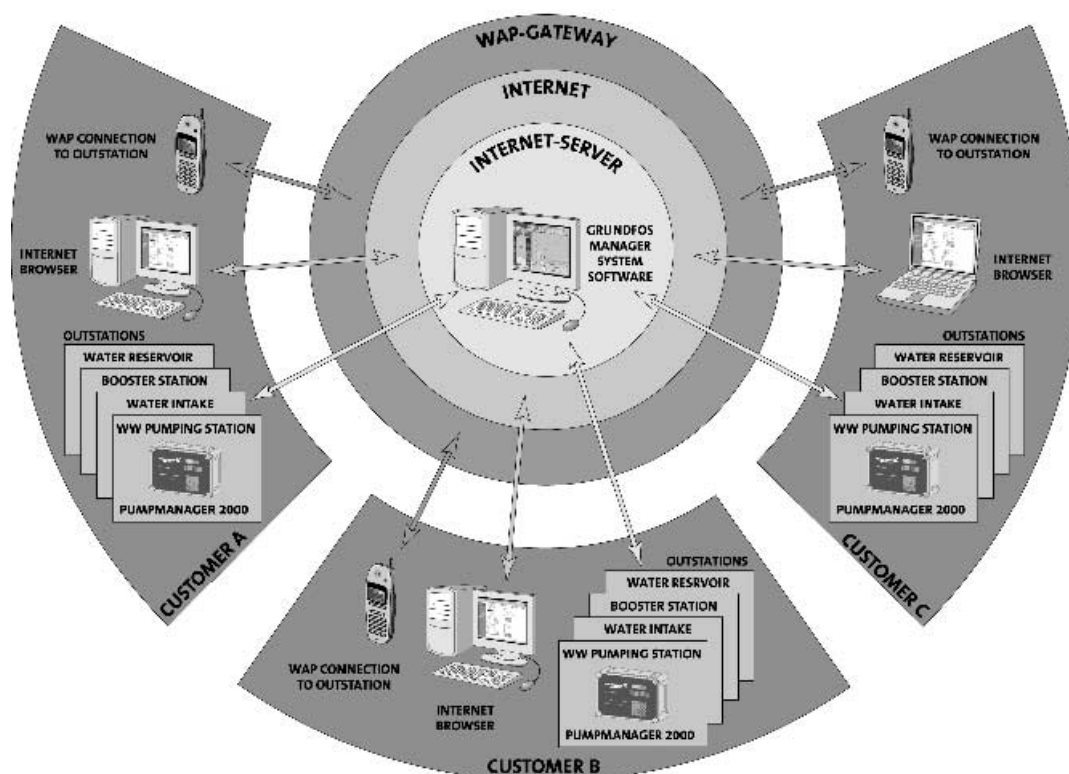
Integration does not mean that all systems run in the same computer and with the same software. It is normally useful to pick the best system for each application and to combine them at a suitable level. This could, for example, mean common software for alarm transfer and reporting. To make integration possible, the systems should be designed using standard procedures such as PC operating systems, standard data transmission and signal input and output protocols.

### 10.5 Internet & WAP Based Remote Control and Monitoring

Alarm messages transferred to the service persons as SMS messages are purely one-way information. If the service person would have the possibility to control the system and change some vital parameters from his mobile phone while being on the field, the total flexibility by the means of a mobile control center can be achieved.

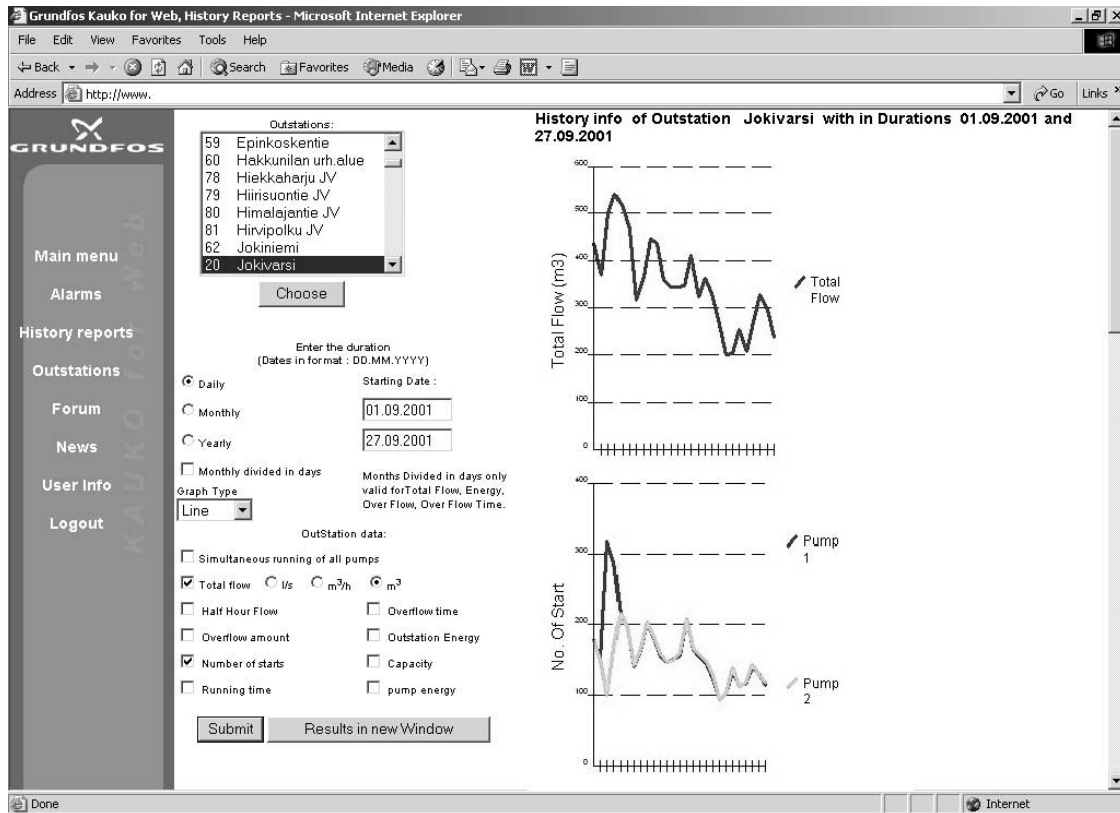
The latest improvements in remote control and monitoring techniques involve Internet and WAP technology to overcome the limitations of traditional monitoring systems described above. The Internet/WAP control and monitoring systems also enable remote monitoring to be offered as service to the municipalities. Figure 94 shows an Internet or WAP based remote control arrangement.

Fig. 94



*The GRUNDFOS model for the Internet/WAP based control and monitoring system offers completely mobile control center from the WAP cellular phone. In addition, the system utilizes Internet for data storage and the customers can monitor the outstations and create reports from any computer with Internet access. The system also enables the whole remote control and monitoring to be offered as a contract service.*

Fig. 95



From the Internet the customers can browse the history data of their own outstations independent on the location. Information can also easily be shared inside the organisation – operation personnel, duty persons, decision-makers as well as sewage system designers and engineers.

The Internet based control and monitoring system allows historical data from the outstations to be viewed and reported from multiple locations, thus enabling the information to be used wherever needed. After typing in the user identification the operation personnel, persons on duty, decision-makers, sewage system engineers, etc. are able to browse detailed historical data from the outstations for years backwards e.g. from their own office computers. Figure 95 shows the interface page of the Grundfos Web based control service.



## Symbols

A	area
A <sub>W</sub>	wet well surface area
a	pressure wave velocity
D	pipe internal diameter
F <sub>a</sub>	axial force
F <sub>r</sub>	radial force
f <sub>H</sub>	head measurement uncertainty factor
f <sub>max</sub>	maximum frequency
f <sub>min</sub>	minimum frequency
f <sub>N</sub>	nominal frequency
f <sub>Q</sub>	flow rate measurement uncertainty factor
g	acceleration of gravity, 9,81 m/s <sup>2</sup>
H	pump total head (head)
H <sub>0</sub>	head at zero flow rate
H <sub>d</sub>	dynamic head
H <sub>f</sub>	head friction losses
H <sub>G</sub>	guaranteed head
H <sub>geod</sub>	geodetic head
H <sub>J</sub>	head loss in pipeline
H <sub>Jn</sub>	local head loss
H <sub>Jp</sub>	head loss in pressure pipeline
H <sub>Jt</sub>	head loss in suction pipeline
H <sub>max</sub>	maximum allowable head
H <sub>min</sub>	minimum allowable head
H <sub>r</sub>	head loss
H <sub>rt</sub>	head losses in inlet pipe
H <sub>s</sub>	head discontinuity losses
H <sub>st</sub>	static head
H <sub>t</sub>	theoretical head
H <sub>t∞</sub>	ideal head
H <sub>v</sub>	head leakage losses
h	height
h <sub>A</sub>	height difference between reference plane and tip of vane leading edge
h <sub>s</sub>	pump stop level
h <sub>t</sub>	inlet geodetic height
I	electric current
I <sub>l</sub>	line current
I <sub>ph</sub>	phase current
k	coefficient
L	length
l	length

M <sub>B</sub>	concrete mass, kg
MH	head measurement uncertainty
MQ	flow rate measurement uncertainty
m	mass
NPSH	net positive suction head
n	rotational speed
n <sub>N</sub>	nominal rotational speed
P	pump power input
P <sub>gr</sub>	motor power input
p	pressure
p <sub>b</sub>	ambient pressure at liquid level
p <sub>L</sub>	atmospheric pressure in pump well
p <sub>m</sub>	sand content by weight
p <sub>min</sub>	minimum static pressure in pump
p <sub>U</sub>	atmospheric pressure in receiving well
p <sub>v</sub>	liquid vapor pressure, sand content by volume
Q	volume rate of flow
Q <sub>0</sub>	volume rate of flow at zero head
Q <sub>I</sub>	volume rate of flow, one pump
Q <sub>II</sub>	volume rate of flow, two pumps
Q <sub>in</sub>	incoming flow
Q <sub>N</sub>	pump nominal flow rate
q	volume flow
Re	Reynold's number
S	curve slope
T	cycle duration
t	pump running time
t <sub>H</sub>	head tolerance factor
t <sub>Q</sub>	flow rate tolerance factor
U <sub>I</sub>	line voltage
U <sub>ph</sub>	phase voltage
u	perimeter velocity
V	volume
V <sub>G</sub>	pumping station volume below water table, m <sup>3</sup>
V <sub>H</sub>	effective wet well volume
V <sub>h</sub>	effective wet well volume
v	true fluid velocity
v <sub>2</sub>	flow velocity at pump outlet
v <sub>m</sub>	radial component of true velocity
v <sub>u</sub>	tangential component of true velocity
w	velocity relative to the vane
X <sub>H</sub>	head tolerance factor
X <sub>Q</sub>	flow rate tolerance factor
Z	starting frequency

$Z_{1,2}$	pressure gage height above water level
$Z_{lmax}$	maximum pump starting frequency
$Z_{llmax}$	maximum pump starting frequency
$Z_{max}$	maximum pump starting frequency
$\beta$	vane edge angle
$\Delta h$	local pressure drop at vane leading edge
$\Delta h$	pressure (head) change
$\Delta H_L$	lower allowable head deviation
$\Delta H_T$	allowable head deviation
$\Delta H_U$	upper allowable head deviation
$\Delta Q_L$	lower allowable flow rate deviation
$\Delta Q_T$	allowable flow rate deviation
$\Delta Q_U$	upper allowable flow rate deviation
$\Delta v$	flow velocity change
$\zeta$	local resistance factor
$\eta$	pump efficiency
$\eta_{gr}$	overall efficiency
$\eta_h$	hydraulic efficiency
$\eta_{mot}$	motor efficiency
$\lambda$	friction factor
$\mu$	reflection cycle duration
$\nu$	kinematic viscosity
$\rho$	fluid density



# Nomogram for head losses in bends, valves etc.

**GRUNDFOS** 

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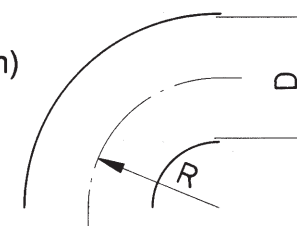
Q = Flow, l/s  
D = Pipe inner diameter, mm  
v = Flow velocity, m/s  
 $\zeta$  = Loss coefficient  
 $H_j$  = Head losses, m

Example  
Q = 12 l/s  
D = 100 mm  
v = 1,55 m/s  
 $\Sigma \zeta = 5$   
 $H_j = 0,6$  m

## Guide values for head losses in bends, valves, etc .

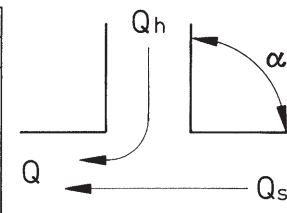
- Pipe bend 90 °, R/D = 1,5
- Discharge loss
- Swing check valve
- Ball check valve
- Gate valve

$\zeta$   
0,3  
1,0 (pipe without expansion)  
1...2  
0,7...1,2  
0,2



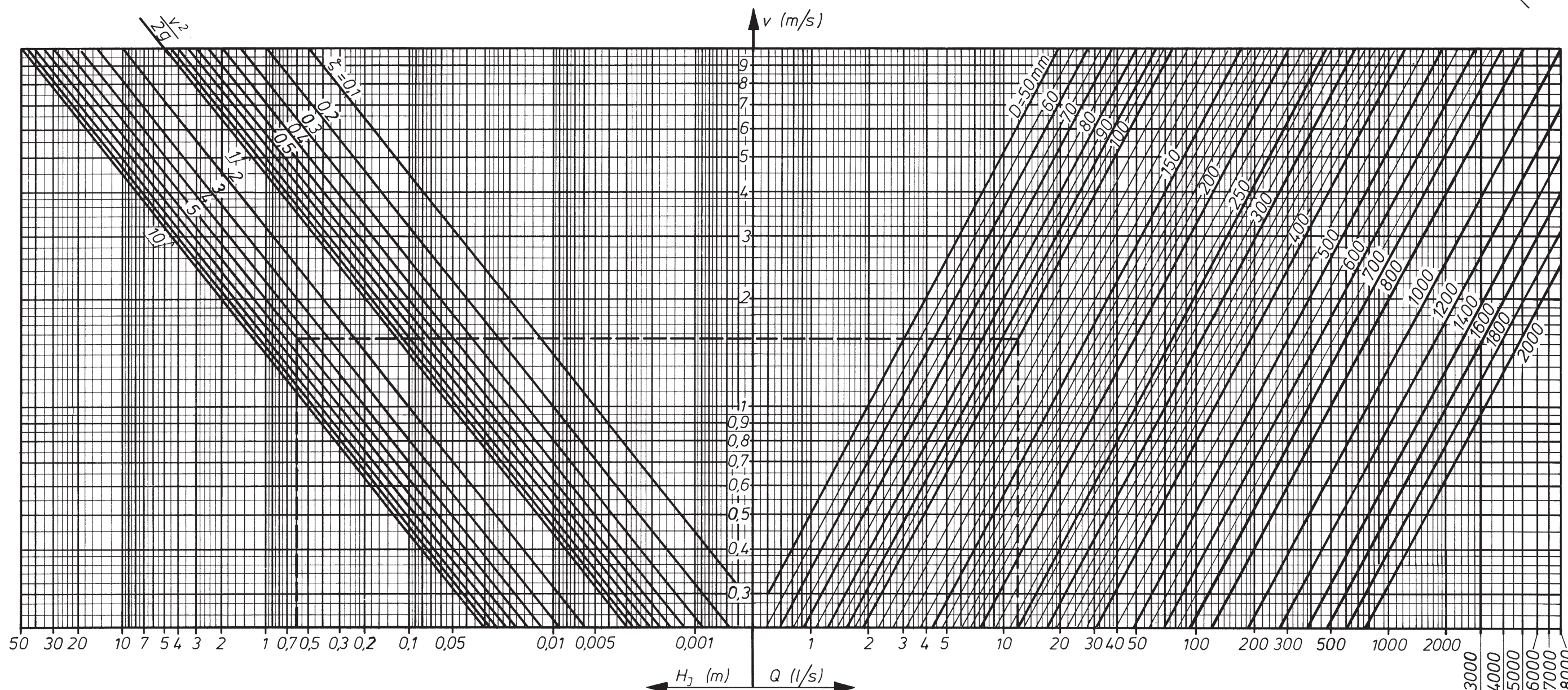
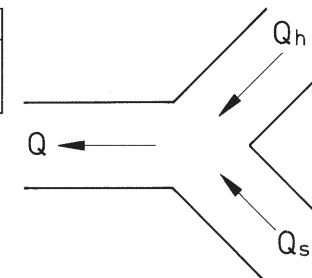
### • T-piece

Q <sub>H</sub> /Q	$\alpha = 90^\circ$		$\alpha = 45^\circ$	
	$\zeta_h$	$\zeta_s$	$\zeta_h$	$\zeta_s$
0,0	-1,00	0,04	-0,90	0,04
0,2	-0,40	0,17	-0,38	0,17
0,4	0,08	0,30	0,00	0,19
0,6	0,47	0,41	0,22	0,09
0,8	0,72	0,51	0,37	-0,17
1,0	0,91	0,60	0,37	-0,54



### • Y-piece

Q <sub>H</sub> /Q	$\zeta_h$	$\zeta_s$
0,0	-	0,6
0,5	0,3	0,3
1,0	0,6	-





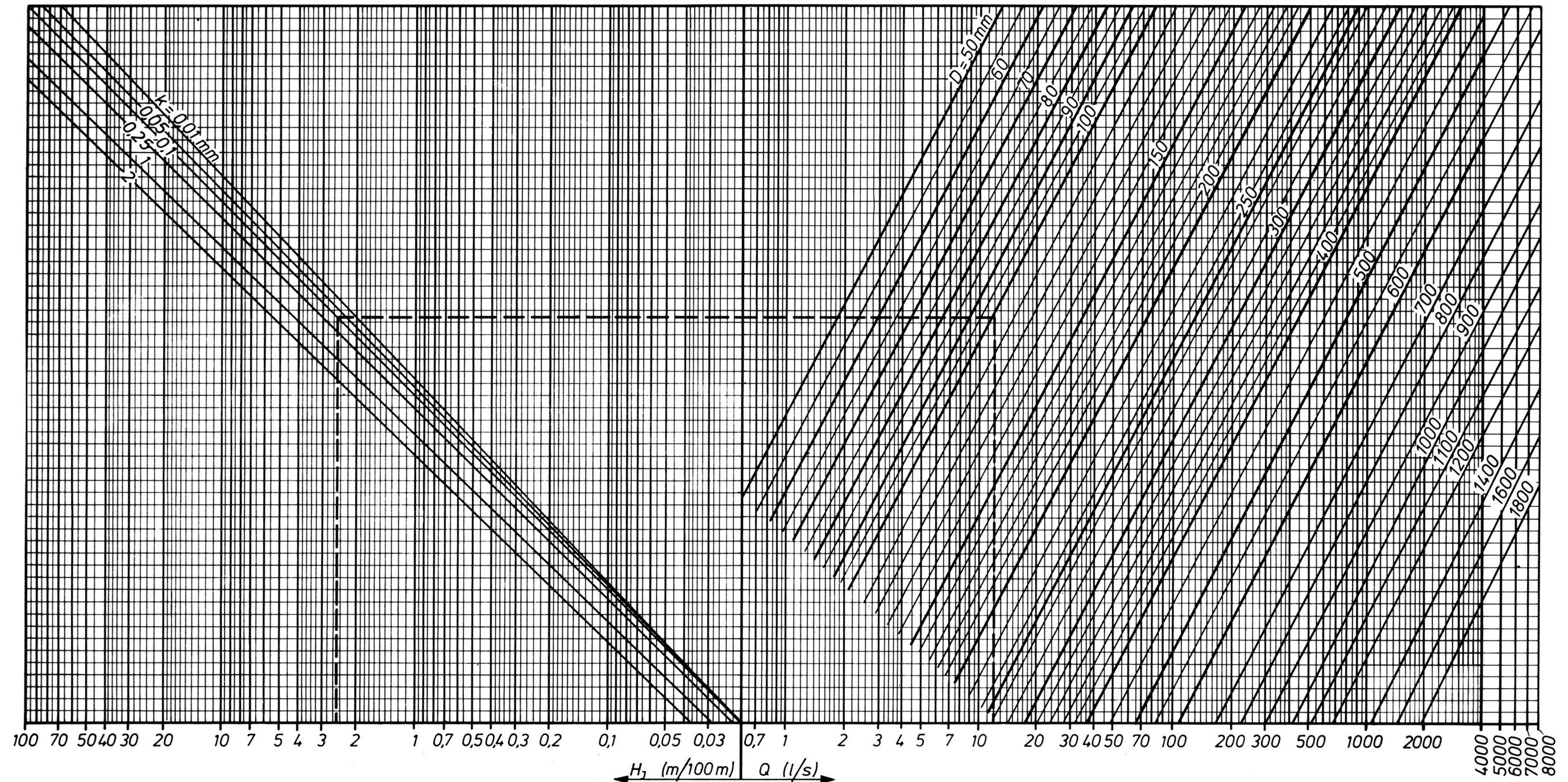
# Pipe loss nomogram for clean water 20 ° C

## Guide values for surface roughness (k) for pipes

pipe material	new pipe k (mm)	old pipe k (mm)
plastic	0,01	0,25
drawn steel	0,05	1,0
welded steel	0,1	1,0
drawn stainless steel	0,05	0,25
welded stainless steel	0,1	0,25
cast iron	0,25	1,0
galvanized steel	0,15	
bituminized cast iron	0,12	
concrete	0,3...2,0	
asbestos cement	0,025	

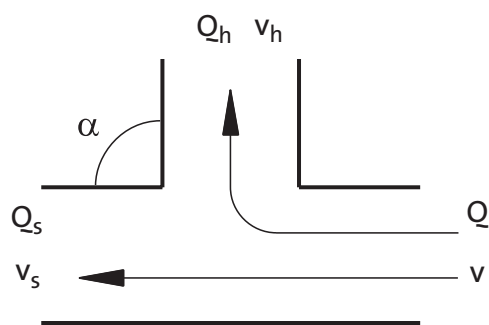
Q = Flow, l/s  
D = Pipe inner diameter, mm  
k = Surface roughness, mm  
H<sub>J</sub> = Pipe losses, m/100 m

Example  
Q = 12 l/s  
D = 100 mm  
k = 0,1 mm  
H<sub>J</sub> = 2,5 m/ 100 m

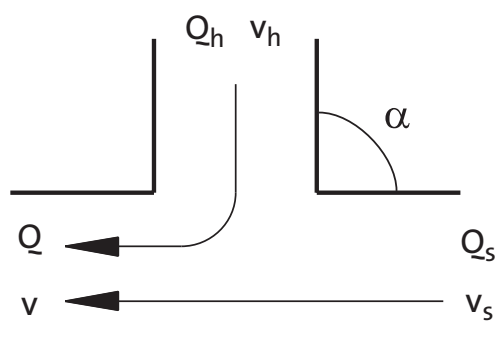


## Local Resistance Factors

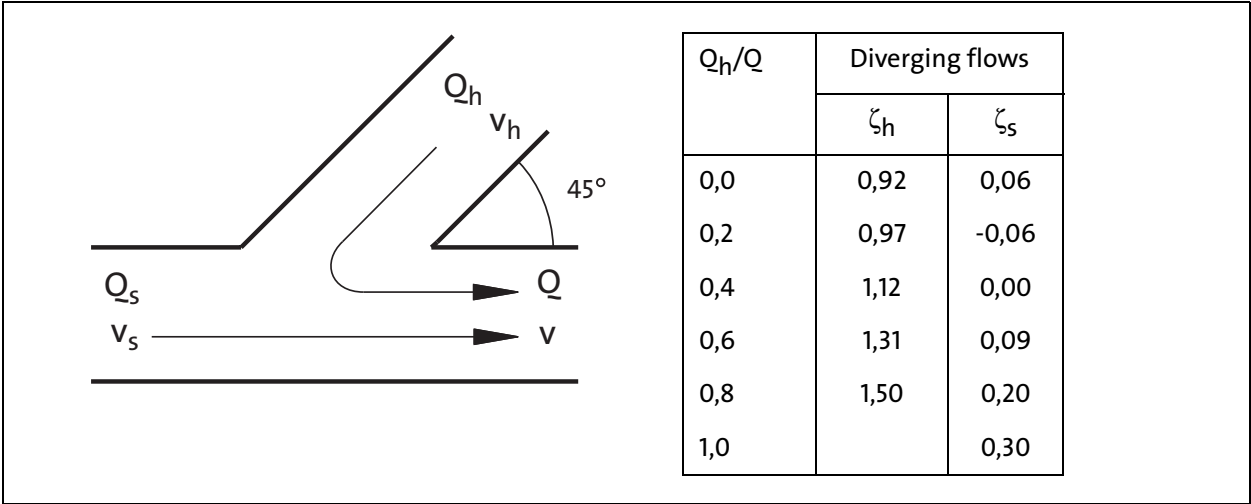
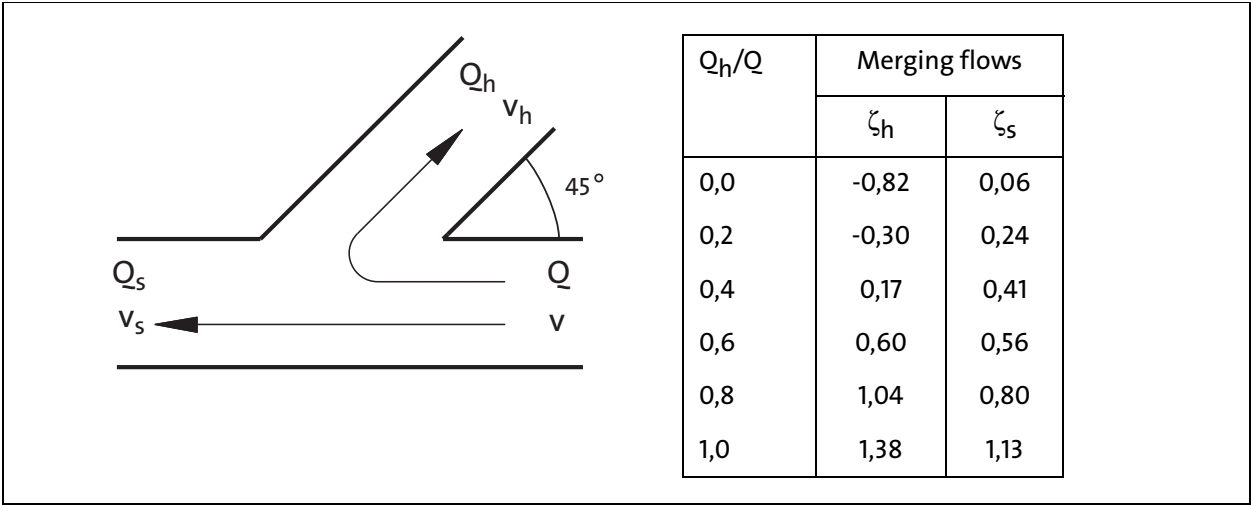
### Branches



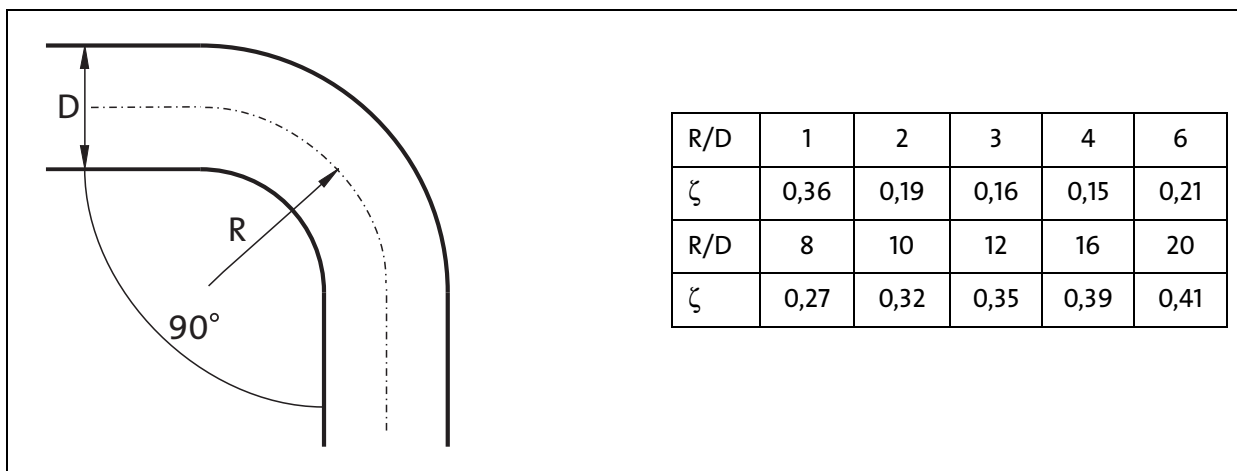
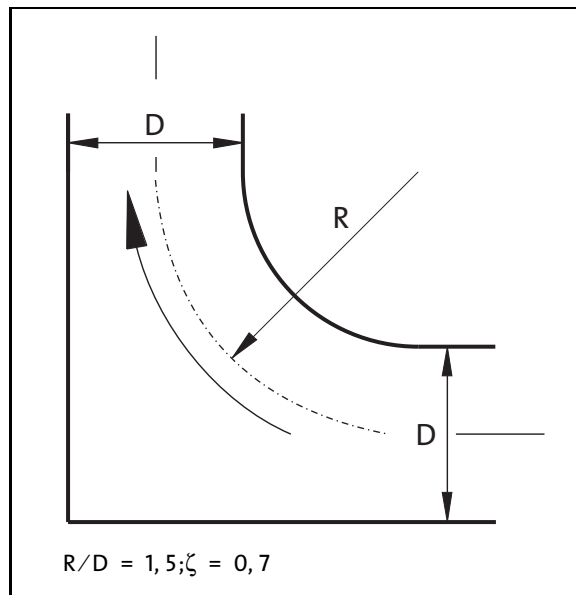
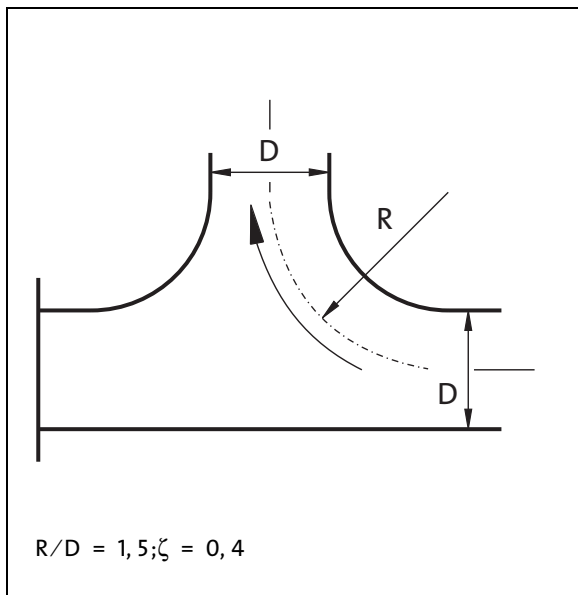
$Q_h/Q$	Diverging flows			
	$\alpha = 90^\circ$		$\alpha = 45^\circ$	
	$\zeta_h$	$\zeta_s$	$\zeta_h$	$\zeta_s$
0,0	0,95	0,04	0,90	0,04
0,2	0,88	-0,08	0,68	-0,06
0,4	0,89	-0,05	0,50	-0,04
0,6	0,95	0,07	0,38	0,07
0,8	1,10	0,21	0,35	0,20
1,0	1,28	0,35	0,48	0,33

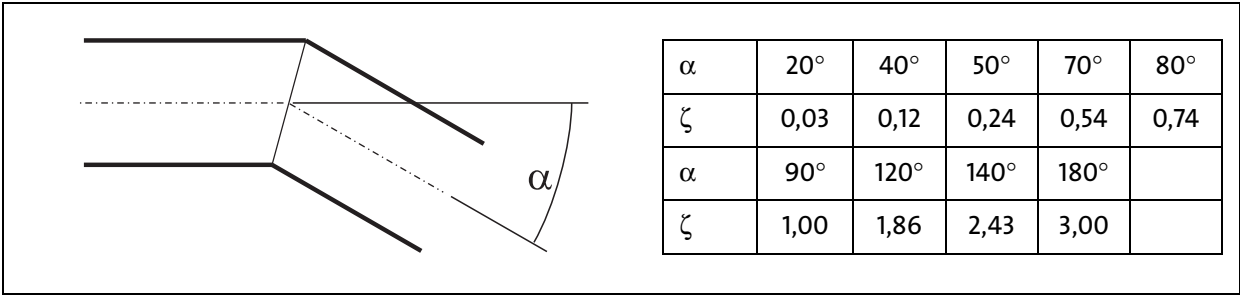
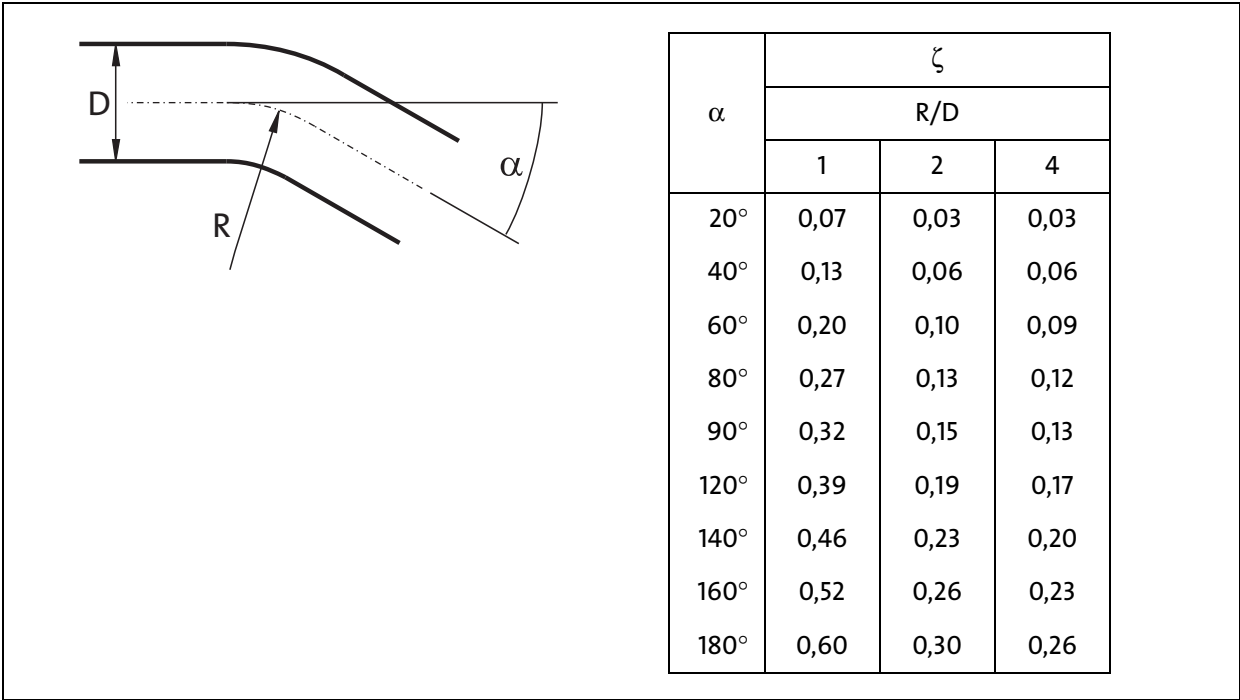


$Q_h/Q$	Merging flows			
	$\alpha = 90^\circ$		$\alpha = 45^\circ$	
	$\zeta_h$	$\zeta_s$	$\zeta_h$	$\zeta_s$
0,0	-1,00	0,04	-0,90	0,04
0,2	-0,40	0,17	-0,38	0,17
0,4	0,08	0,30	0,00	0,19
0,6	0,47	0,41	0,22	0,09
0,8	0,72	0,51	0,37	-0,17
1,0	0,91	0,60	0,37	-0,54

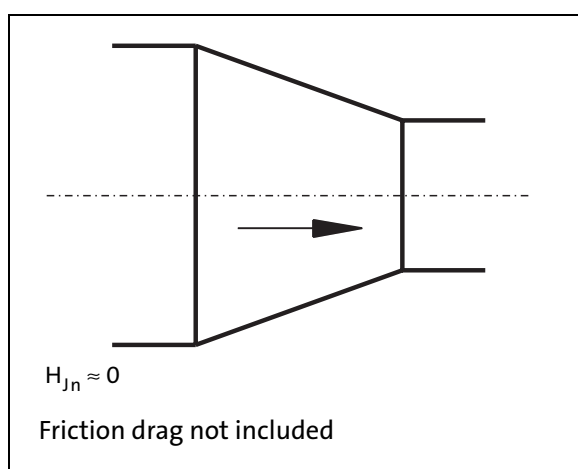
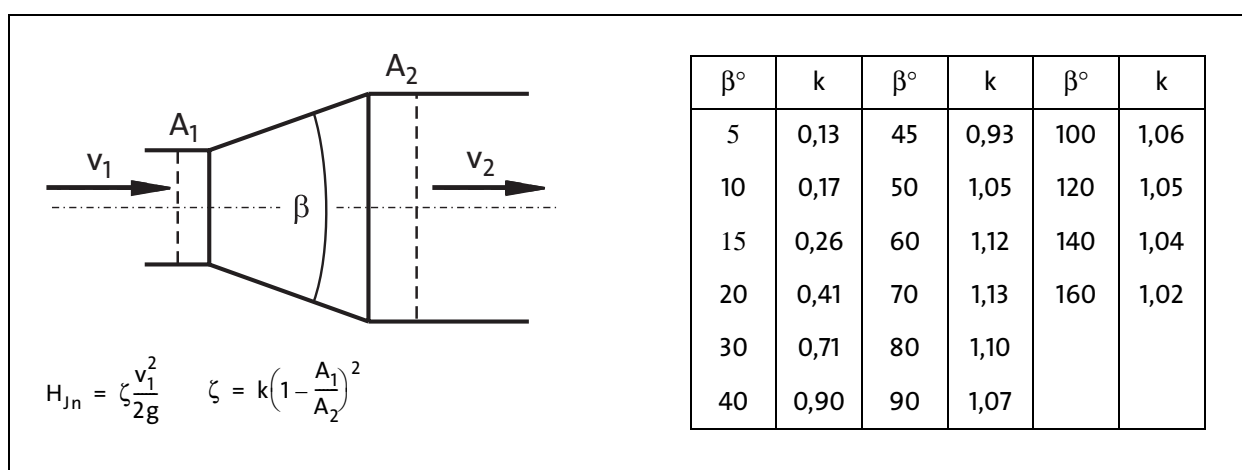
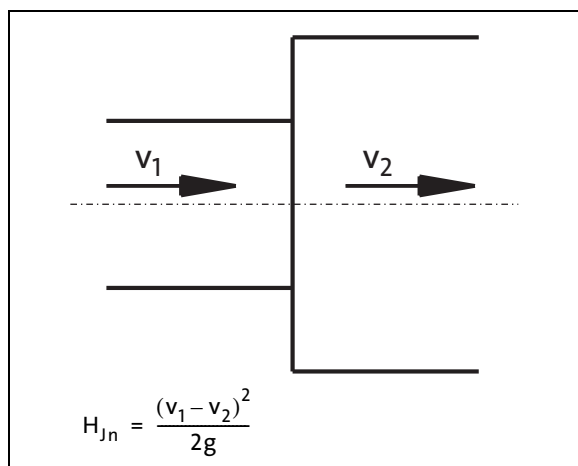


## Bends

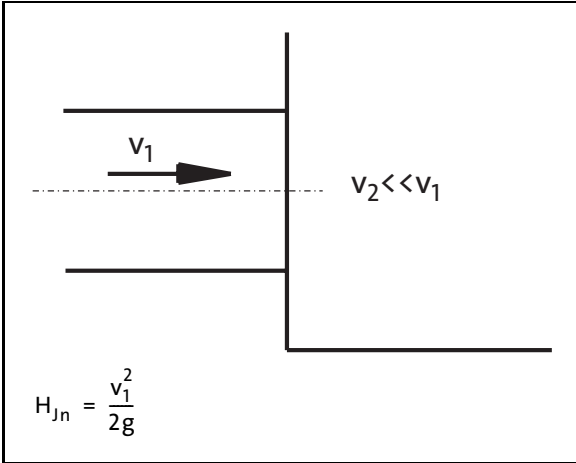
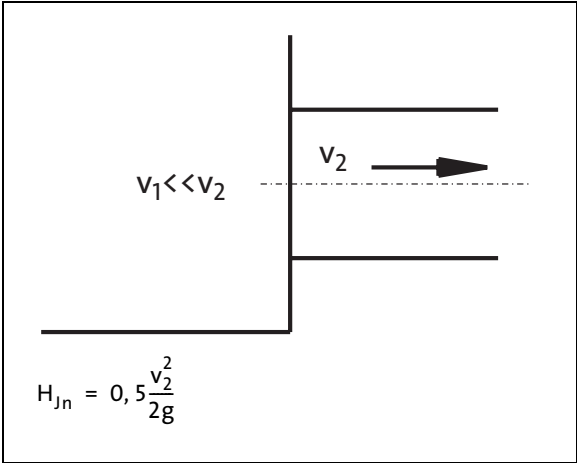
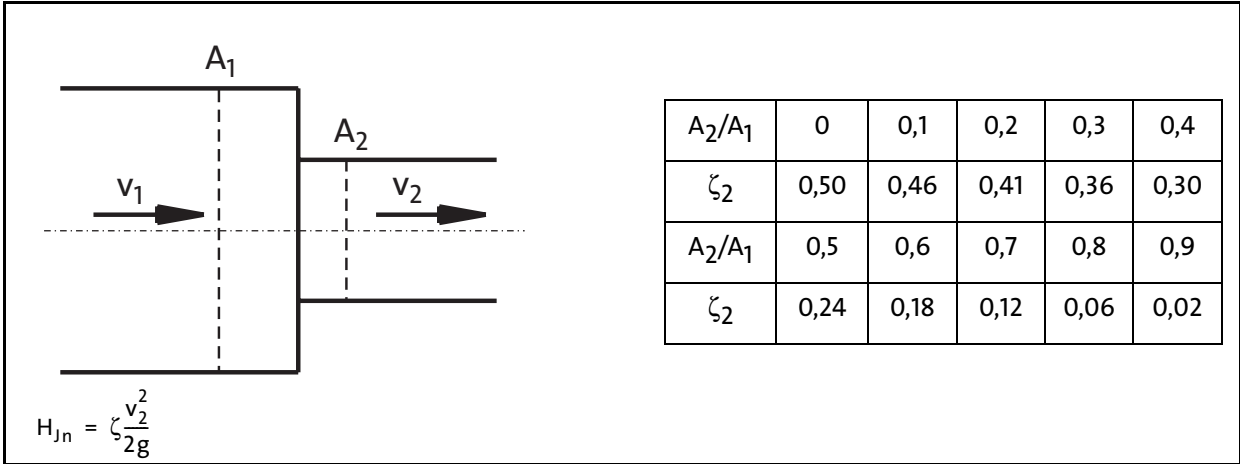




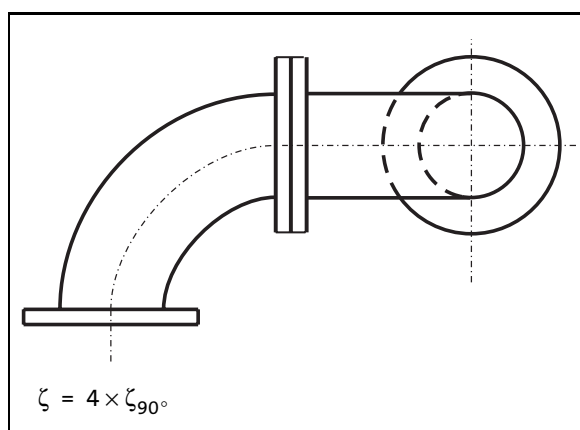
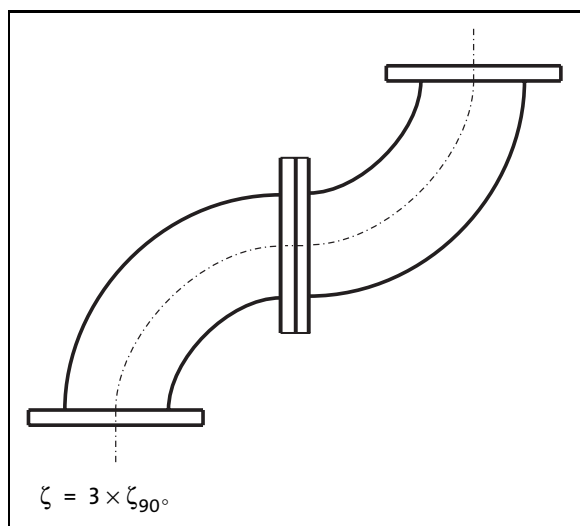
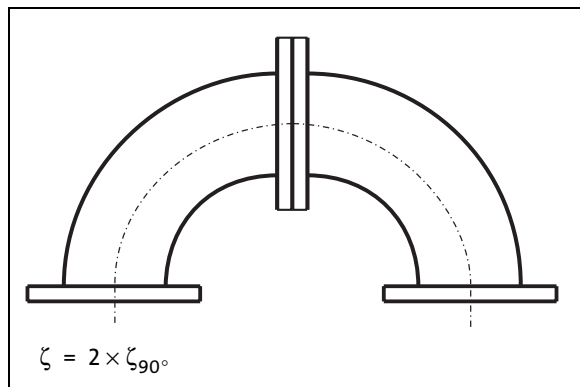
## Expansions and Contractions



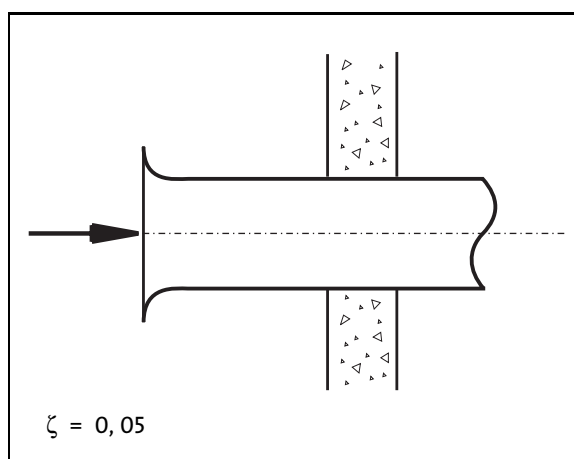
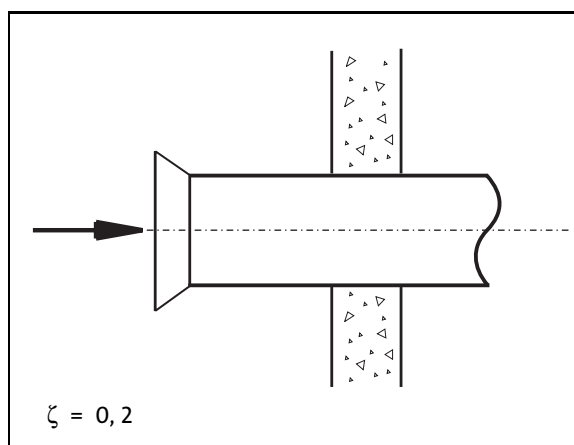
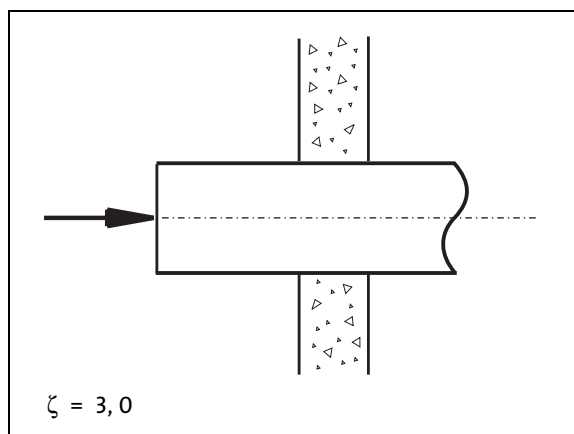




## Bend Combinations

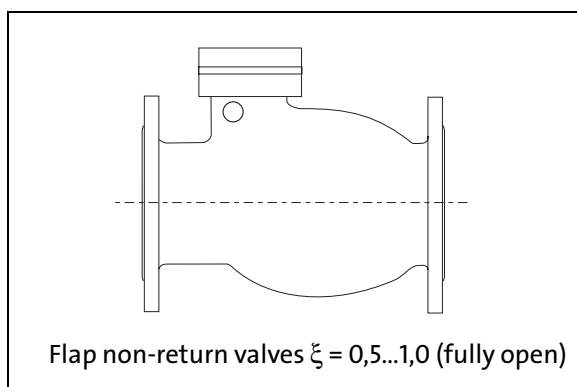
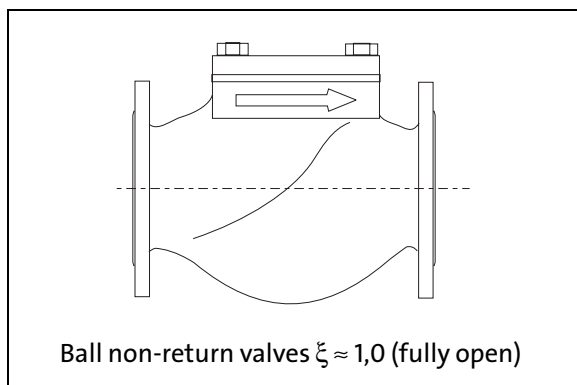
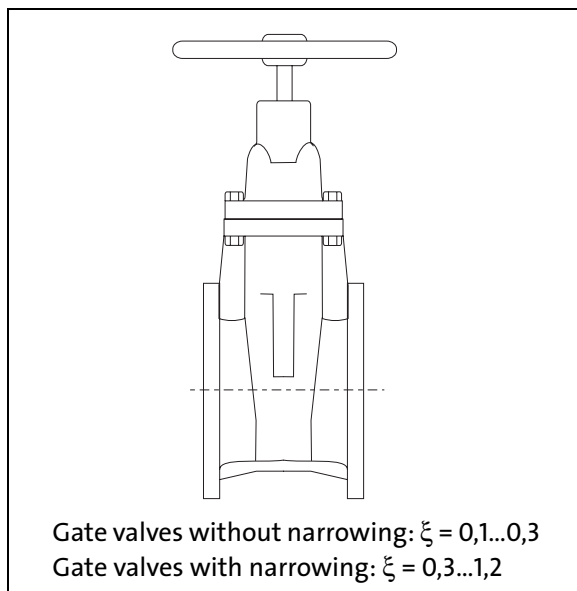


## Suction Inlets



### Valves

$\xi$ -values depend strongly on shape. Factory values should be used when available.



$\xi$ -values above are valid for fully open valves. In partly open position,  $\xi$  may be 1,5-2 times as high. Depending on shape and position, a certain minimum flow velocity through the valve is required for it to be regarded as fully open. Exact information on each valve is available from the manufacturer or supplier.

## Pumping Station Starting Frequency and Pumping Capacity

In a pumping station the water volume comprises the volume below the lowest pump stop level and the pumpable volume above this level, fluctuating with pump usage and water incoming flow rate. The starting frequency of the pumps depends on the available pumpable volume and the incoming flow rate.

The following different cases are investigated:

- single pump pumping station
- pumping station with two pumps in duty-standby operation
- pumping station with more than two pumps.

### Single Pump Station

Incoming water during one unit of time (cycle) can be expressed as:

$$V = Q_{in} \cdot T \quad (B1)$$

where

$Q_{in}$  = incoming flow rate

$T$  = duration of cycle

The same volume must be removed by the pump during the cycle, whence

$$V = Q \cdot t \quad (B2)$$

where

$Q$  = pump capacity

$t$  = pump running time

Combining equations B1 and B2 is obtained

$$t = \frac{Q_{in} T}{Q} \quad (B3)$$

When the pump is stopped, the volume between the start and stop levels  $V_h$  fills up during the time  $T - t$ , whence

$$V_h = Q_{in} \cdot (T - t) = Q_{in} \cdot T - Q_{in} \cdot t \quad (B4)$$

Substituting with the expression B3 for  $t$  in equation B4:

$$V_h = Q_{in} T - Q_{in} \frac{Q_{in} T}{Q} \quad (B5)$$

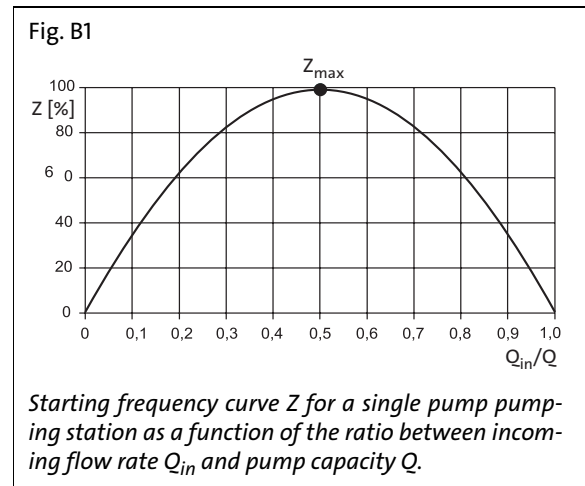
Solving equation B5 for  $T$  is obtained:

$$T = \frac{V_h Q}{Q_{in} Q - Q_{in}^2} \quad (B6)$$

Starting frequency is the inverse value of  $T$ , hence:

$$Z = \frac{Q_{in} Q - Q_{in}^2}{V_h Q} \quad (B7)$$

The starting frequency  $Z$  is a function of the ratio  $Q_{in}/Q$  and is shown in Figure B1.



Differentiating the equation B7 over  $Q_{in}$  is obtained:

$$\frac{dZ}{dQ_{in}} = \frac{Q - 2Q_{in}}{V_h Q} \quad (B8)$$

Equation B8 equals 0 when  $Q_{in} = \frac{1}{2}Q$

Substituting  $Q_{in} = \frac{1}{2}Q$  to equation B7:

$$Z_{max} = \frac{Q}{4 \cdot V_h} \quad (B9)$$

## Appendix B

From this the pumping station capacity  $V_h$  is obtained:

$$V_h = \frac{Q}{4 \cdot Z_{\max}} \quad (\text{B10})$$

The solution to equation B10 is shown graphically in Figure B2.

In practice there may be situations where the incoming flow to a pumping station is very small and only momentary, for instance in pumping stations serving a few households only. In such cases the selected pump capacity should be selected much larger, in order to attain high enough a flow velocity in the rising main to prevent sedimentation. In this situation the  $Q_{\text{in}}/Q$  ratio remains small, and the  $Z_{\max}$  value is not reached at all or very seldom only.

Fig. B2

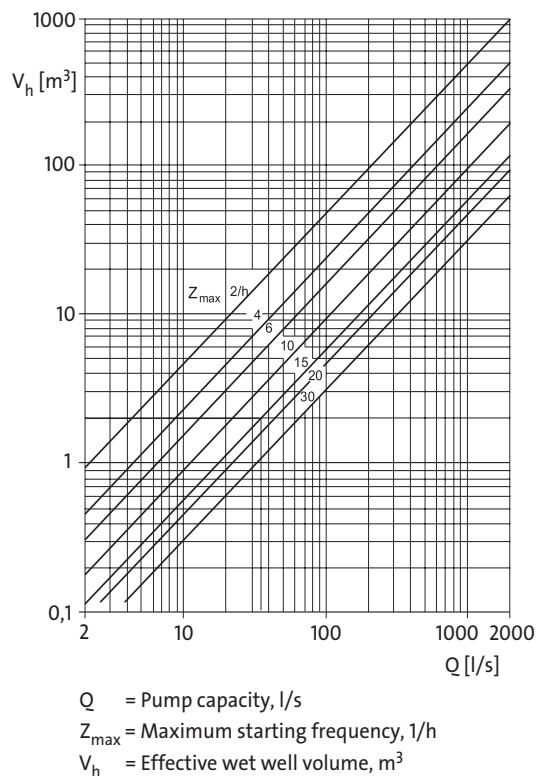
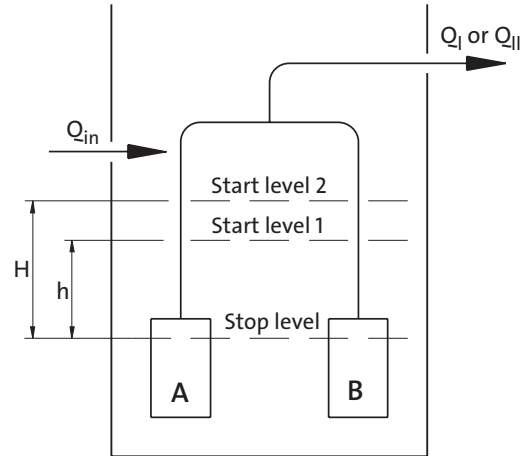


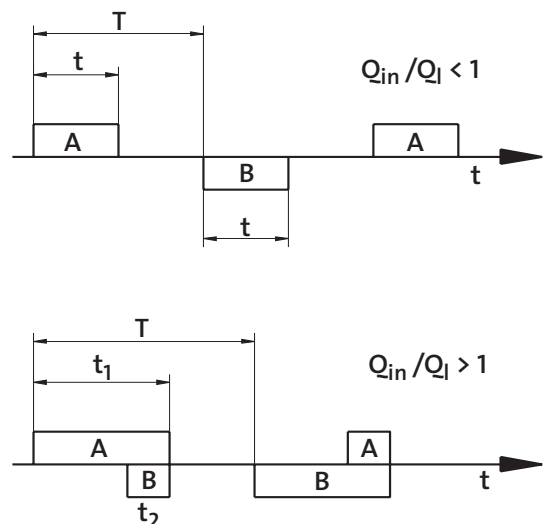
Diagram for the determination of the effective wet well volume  $V_h$  for a single pump pumping station.

Fig. B3



Pumping station with two pumps in alternating duty. The lead pump starts when the water level rises to start level 1. If the incoming flow exceeds the capacity of one pump, the lag pump will start at start level 2. Pumps alternate between lead and lag positions with each running cycle.

Fig. B4



Operation time diagram of the duty and standby pumps in a pumping station for an incoming flow ( $Q_{\text{in}}$ ) both smaller and larger than the capacity of one pump ( $Q_l$ ).

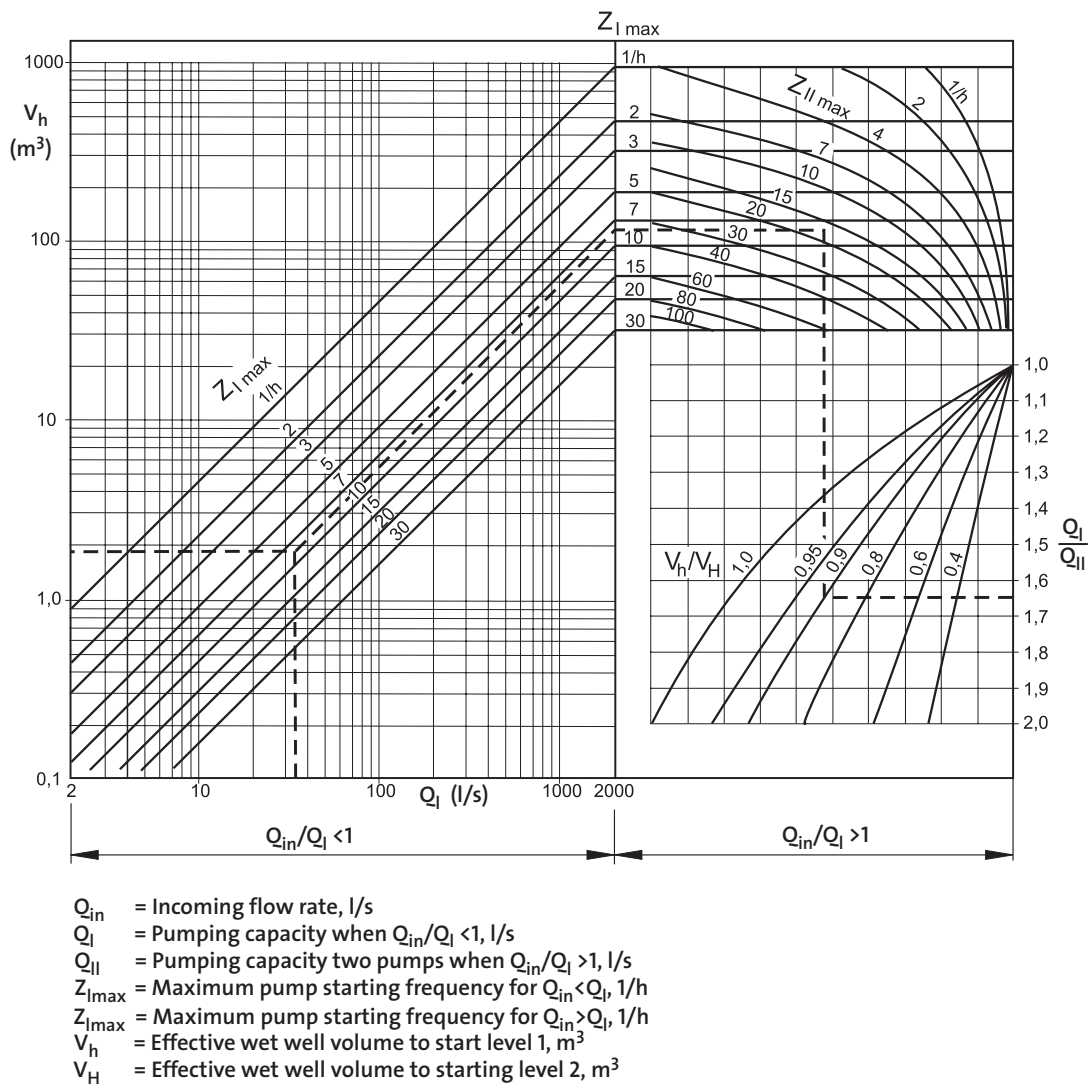
## Two Pumps in Duty-Standby Configuration

The principle of operating a pumping station with two identical pumps is shown in Figure B3. The pumps assume alternately the positions of duty (lead) and standby (lag) pump with each running cycle. When the water level in the wet well reaches the first start level the duty pump starts. The water level is pumped down to the stop level, and the pump stops, allowing the water level to rise again to the first start level, completing the cycle.

The duty pump alone is able to handle most regular incoming flow situations, and the standby pump will start only if the incoming flow rate ( $Q_{in}$ ) is larger than the capacity of one pump ( $Q_l$ ), in which case the water level continues to rise to the second start level, starting the standby pump. If the combined capacity of two pumps ( $Q_{II}$ ) is larger than the incoming flow, all pumps stop when the water reaches the stop level.

Figure B4 shows a time diagram of the running cycle of two pumps in alternating duty, further explaining the principle.

Fig. B5



Nomogram for the determination of the effective wet well volume  $V_h$  and the starting frequency  $Z$  for a pumping station with two pumps in duty-standby configuration.

$$Q_{in} < Q_I$$

Equations B9 and B10 can be used in the situation where the incoming flow is smaller than the capacity of one pump for the calculation of starting frequency for each pump. With two pumps starting alternately, the expressions are divided by two, whence

$$Z_{lmax} = \frac{Q_I}{8 \cdot V_h} \quad (B11)$$

$$V_h = \frac{Q_I}{8 \cdot Z_{lmax}} \quad (B12)$$

The solution to equations B11 and B12 are shown graphically in Figure B5.

$$Q_{in} > Q_I$$

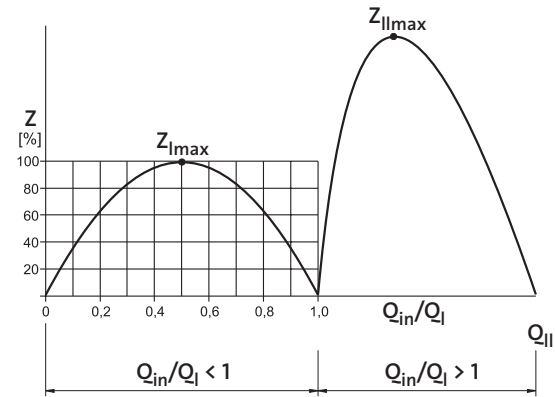
In the case when the incoming flow is larger than the capacity of one pump, two additional factors must be considered. These are the ratio of the pumping station capacity to the first start level,  $V_h$ , and the second start level,  $V_H$ , and the combined capacity of the pumps  $Q_{II}$ . The following equation for the starting frequency can then be derived:

$$Z_{II} = \left[ \frac{Q_I(V_H - V_h)}{Q_{in}^2 - Q_I Q_{in}} + \frac{Q_{II} V_H}{Q_{in} Q_{II} - Q_{in}^2} \right]^{-1} \quad (B13)$$

The expression for  $Z_{IImax}$  can be solved by differentiation, but the expression is very complex. A graphic presentation of the solution is presented in Figure B5.

Figure B6 shows the relation between starting frequency and the  $Q_{in}/Q_I$  ratio. The starting frequency rises sharply at conditions requiring parallel duty. The diagram shows a marked peak value  $Z_{IImax}$ .

Fig. B6



Starting frequency curve  $Z$  for one pump and two pumps in pumping station with two pumps in duty-standby configuration as function of the ratio between incoming flow rate  $Q_{in}$  and pump capacity  $Q_I$ .

The diagram in Figure B7 shows the effect of the ratio  $V_h/V_H$  on  $Z$  for constant  $V_H$  and varying  $V_h$ . In this case the ratio  $Q_{II}/Q_I$  is 1,6. The conclusion from Figure B7 is, that  $Z_{IImax}$  is reduced and  $Z_{lmax}$  increased with lower start level 1.

The diagram in Figure B8 shows the effect of the ratio  $Q_{II}/Q_I$  on  $Z$  for a constant  $V_h/V_H$  ratio of 0,8. Increasing rising main losses, decreasing  $Q_{II}/Q_I$  also decreases  $Z_{IImax}$ .

If the pumps are selected so that one pump can handle all incoming flows,  $Z_{IImax}$  loses significance.

### Pumping Stations with more than Two Pumps

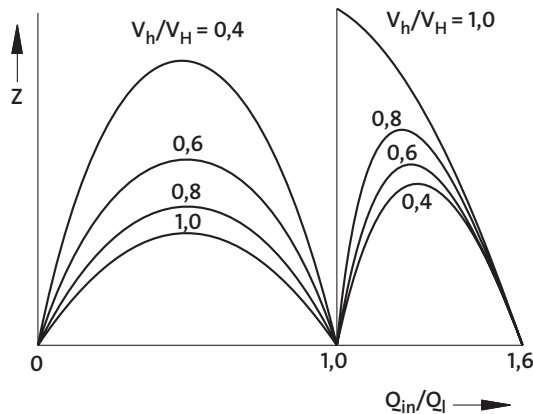
Pumping stations with a multitude of pumps can be divided into the following two design categories:

- Stations with common stop level for all pumps
- Stations with different or stepped stop levels for each pump

The starting cycle of the pumps are normally alternated between the pumps in order to ensure even distribution of wear.

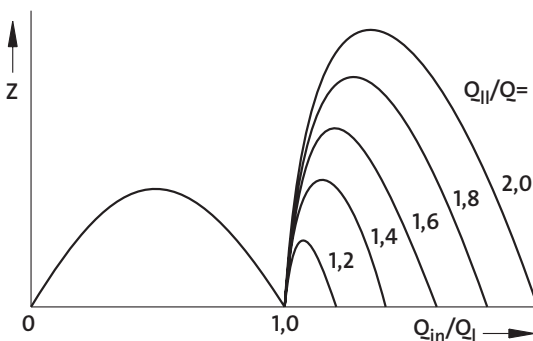


Fig. B7



Starting frequency curves for different  $V_h/V_H$  ratios with constant  $V_H$  and a  $Q_{II}/Q_I$  ratio of 1,6.

Fig. B8



Starting frequency curves for different  $Q_{II}/Q_I$  ratios and a  $V_h/V_H$  ratio of 0,8.

## Pumping Station Capacities and Starting Frequencies

With several pumps installed in a pumping station, the starting frequency changes dramatically with variations in the incoming flow. The starting frequency will vary between zero and peak values, of which there are several.

Great flow fluctuations are typical for sewage pumping, and it becomes impossible and also rather unnecessary to numerically calculate starting frequencies for each pump. With the aid of design nomograms total pumping capacities and average starting frequencies, on which pumping station further design in all practical cases can be based, can be determined.

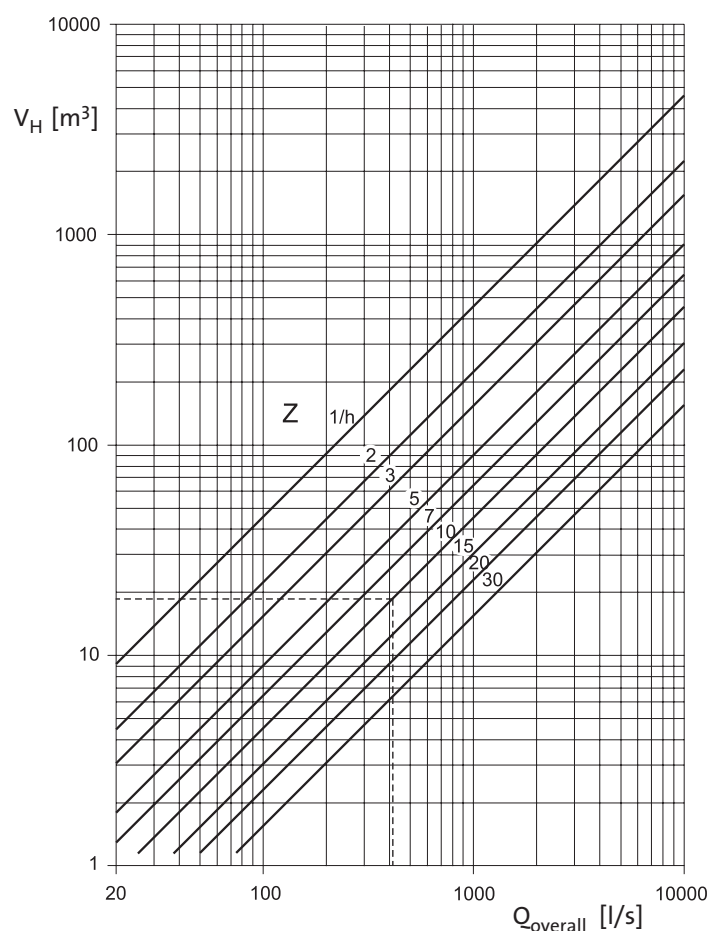
For the different design categories the following nomograms can be used.

## Common Stop Level

Figure B9 shows a diagram from which  $V_H$  or  $Z$  can be selected as functions of overall flow rate  $Q_{\text{overall}}$ .

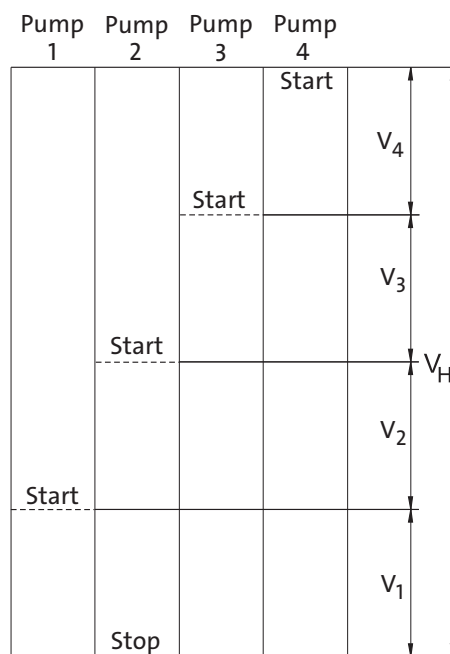
For both of these categories it is good practice to divide the total pumping volume ( $V_H$ ) by the starting levels at approximately equal intervals if all the pumps are identical. If the pumps have different capacities, the pumping volume may be divided into intervals proportional to the pump capacities. The use of modern electronic level control equipment facilitates the optimization of the start levels either manually or automatically.

Fig. B9



$Q_{\text{overall}}$  = Flow rate, l/s  
 $Z$  = Approx. average starting frequency of pump, 1/h  
 $V_H$  = Pumping volume to top level, m<sup>3</sup>

Starting frequency nomogram for pumping station with more than two pumps and common stop level.



For similar pumps:  
 $V_1 \approx V_2 \approx V_3 \approx \text{etc.}$

## Stepped Stop Levels

Figure B10 shows a diagram from which  $V_H$  or  $Z$  can be selected as functions of overall flow rate  $Q_{\text{overall}}$ .

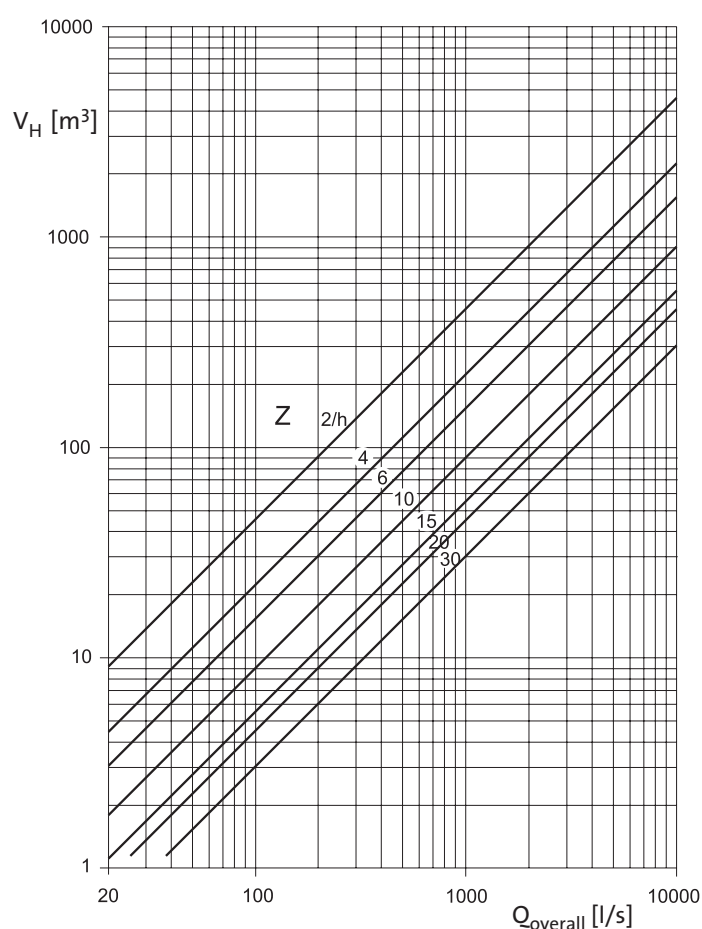
## Recommended Starting Frequencies

Pump and control equipment operation and wear is significantly related to the number of starts and stops over the long period, such as a year, since very high starting frequencies can be allowed in the short term. If peak starting frequencies are used for dimensioning, the occurrence of these must be investigated. As shown earlier, the peak starting frequency for one pump  $Z_{\text{I max}}$  may never be attained in reality. Likewise, the peak starting frequency for two pumps in parallel operation  $Z_{\text{II max}}$  is usually much higher (1,5...2 times) than the  $Z_{\text{I max}}$  value and only occurs occasionally.

Starting frequencies selection should be checked against pump and control equipment manufacturers' recommendations. The following guidelines for mean allowable starting frequencies for submersible pumps may be used:

Pump power	Allowable Z
0...5 kW	25 1/h
5...20 kW	20 1/h
20...100 kW	15 1/h
100...400 kW	10 1/h

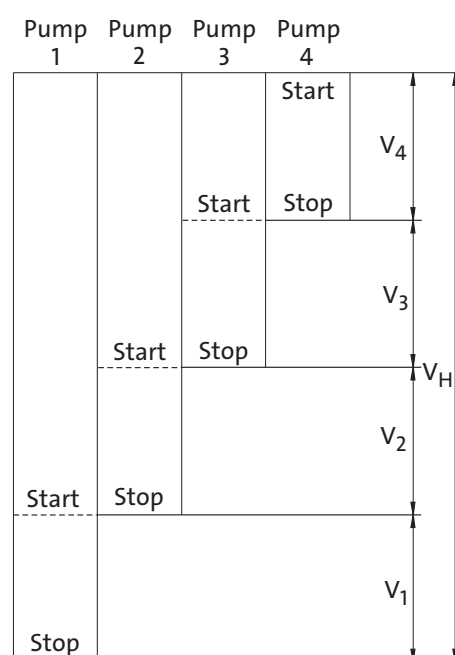
Fig. B10



$Q_{\text{overall}}$  = Flow rate, l/s

Z = Approx. average starting frequency of pump, 1/h

$V_H$  = Pumping volume to top level, m<sup>3</sup>



For similar pumps:  
 $V_1 \approx V_2 \approx V_3 \approx \text{etc.}$

*Starting frequency nomogram for a pumping station with more than two pumps and stepped stop levels.*

BE > THINK > INNOVATE >

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