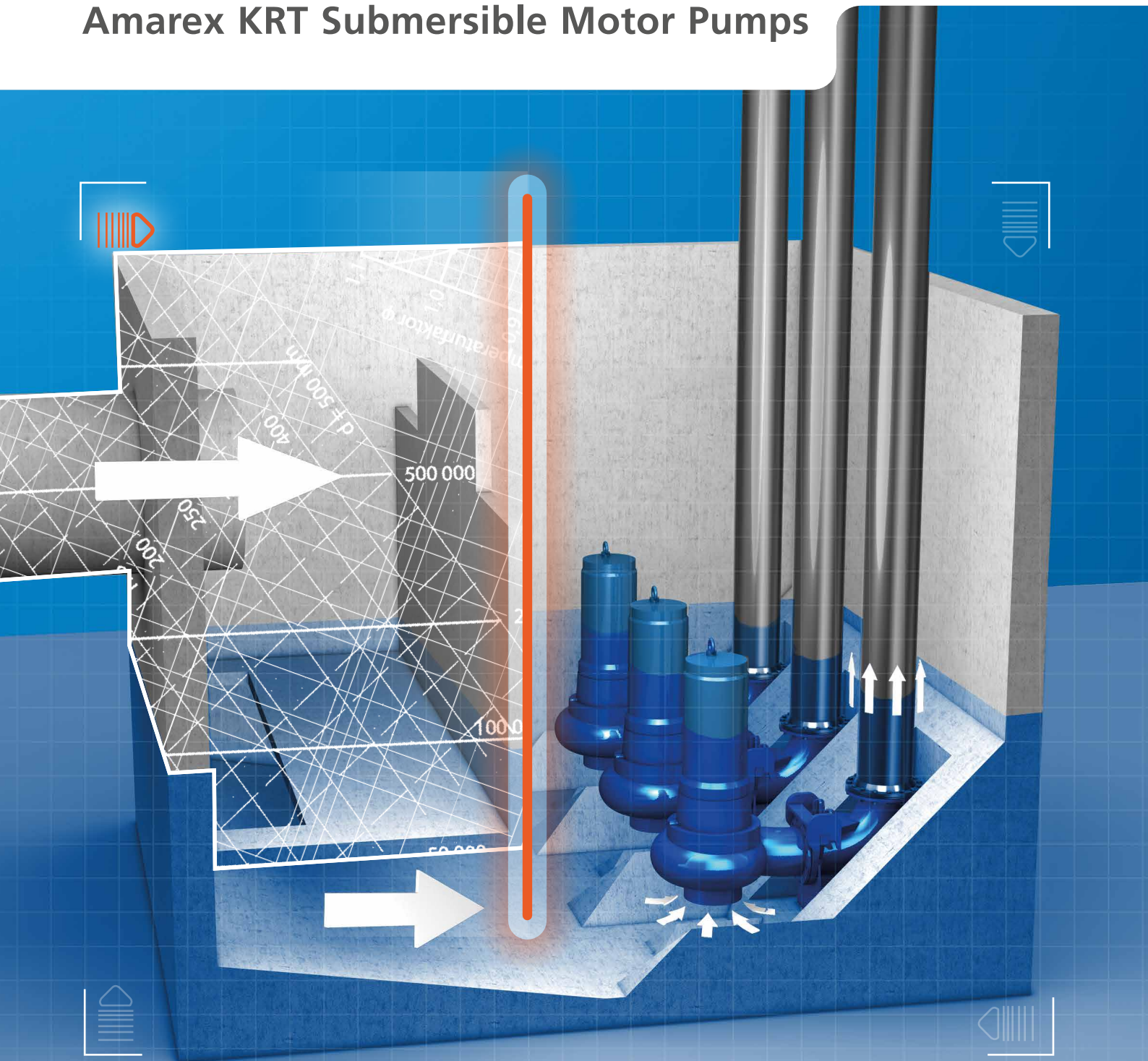**KSB**

# KSB Know-how: Planning Information for Amarex KRT Submersible Motor Pumps

**Dear Partners,**

This brochure has been created to assist consultants and operators in selecting, dimensioning and operating the most suitable submersible motor pump of the Amarex KRT type series for their application. This know-how brochure focuses specifically on KRT pumps with a flow rate > 200 l/s. KSB designed these submersible motor pumps as a safe, reliable and energy-efficient solution for all pumping duties in municipal and industrial waste water applications. One of our aims was to provide maximum versatility by offering the pump in a multitude of materials, with a range of robust sensors and flexible installation options. Specially adapted hydraulic systems with wide free passages achieve optimum reliability and make for the optimum economic handling of diverse fluids. Explosion-proof versions are available for potentially explosive atmospheres. Protection against excessive temperature rise in the motor winding, absolutely water-tight cable entries, a special shaft seal and bearings dimensioned to give long service lives are features which provide for long and trouble-free pump operation.

**The know-how volumes can easily be downloaded or ordered here: www.ksb.com/know-how**

# General Pump Selection

## Planning parameters / design data

When planning or dimensioning a pump or a pumping station, determining the flow rate and the corresponding head is of paramount importance. Although unequivocal assumptions can be made about the losses to be expected when calculating the head, the flow rate is influenced by a number of additional factors which are explained in the following sections.

### Pump flow rate

The flow rate Q, also referred to as the volume flow rate, (measured in l/s or m³/h) is defined as the useful volume of fluid delivered by a pump in a unit of time.

The actual inflow, and thus the required flow rate need to be determined as accurately as possible in order to correctly dimension the pumps (the duty point should lie within the optimal range) as well as the pumping station.

The daily inflow of a waste water pumping station is significantly influenced by the following factors:

- The type of drainage system (combined or separate sewer lines for waste water and stormwater)
- The size and composition of the area served by the treatment plant (residential or industrial area, or a mixture of both)
- The number of inhabitants connected to the waste water system
- The number and type of industrial and commercial areas connected to the waste water system (expressed in population equivalents)

The inflow can be illustrated in a load curve which shows the flow rate of the calculated / typical waste water inflow over a day (see examples, Figs. 1 and 2).[1]

Significant differences might become apparent not only in the 24-hour load curve, but also when comparing the total inflow of workdays with that of public holidays or weekend days. In the event of precipitation, the inflow is expected to rise. This factor is of particular importance in combined sewer systems (which serve to transport both waste water and stormwater to the waste water treatment plant).[2]

The load curve forms an important basis for selecting the type and number of pumps, assigning pumps to different operating ranges / conditions, deciding on the pump drives (e.g. fixed or variable speed) and, last but not least, determining the required duty points of the individual pump sets.
Given the large fluctuations of waste water inflow (24-hour load curve, Fig. 1 and Fig. 2), the factor "time" needs to be taken into account.

[1] Additional literature: ATV-DVWK – A 134
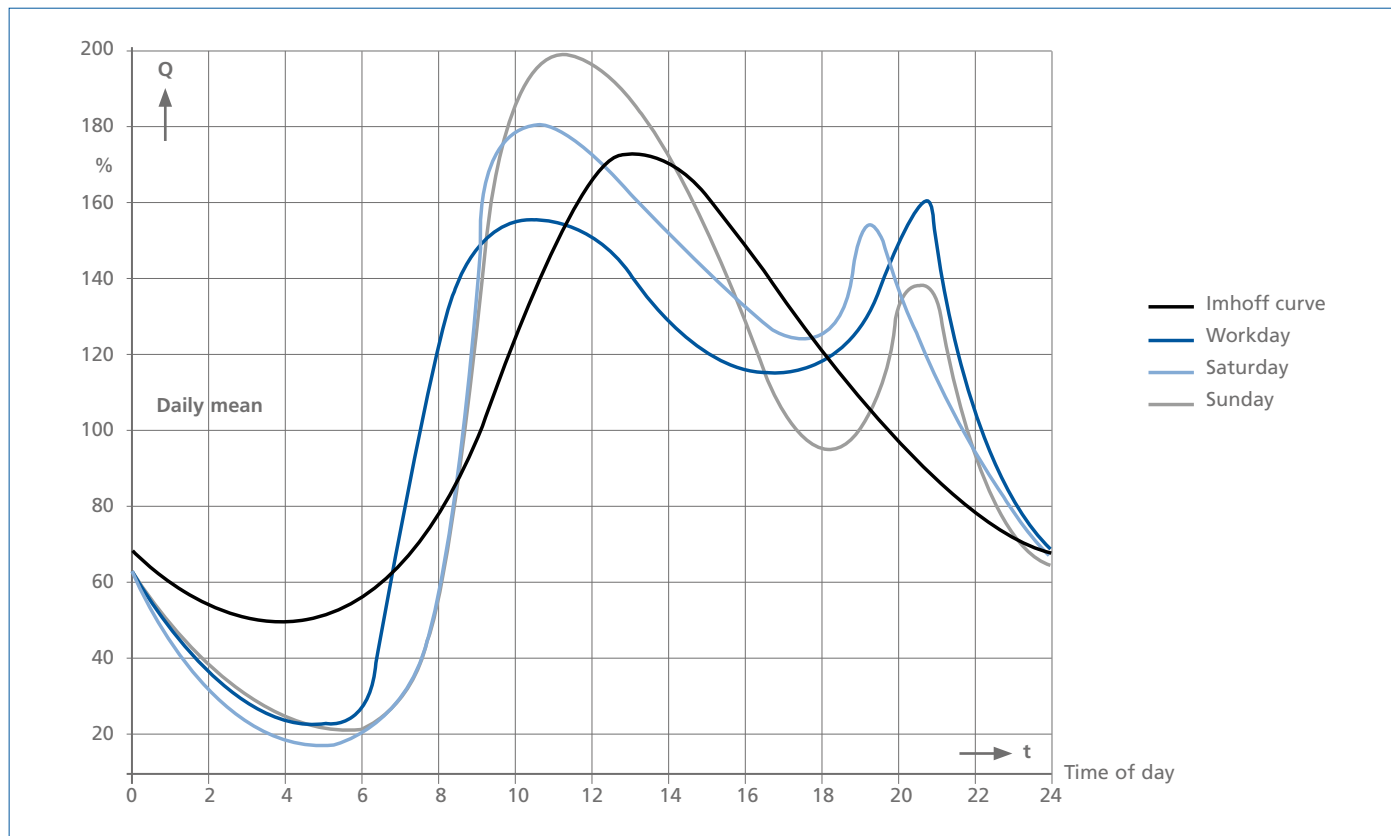[2] Additional literature: ATV-DVWK – A 118

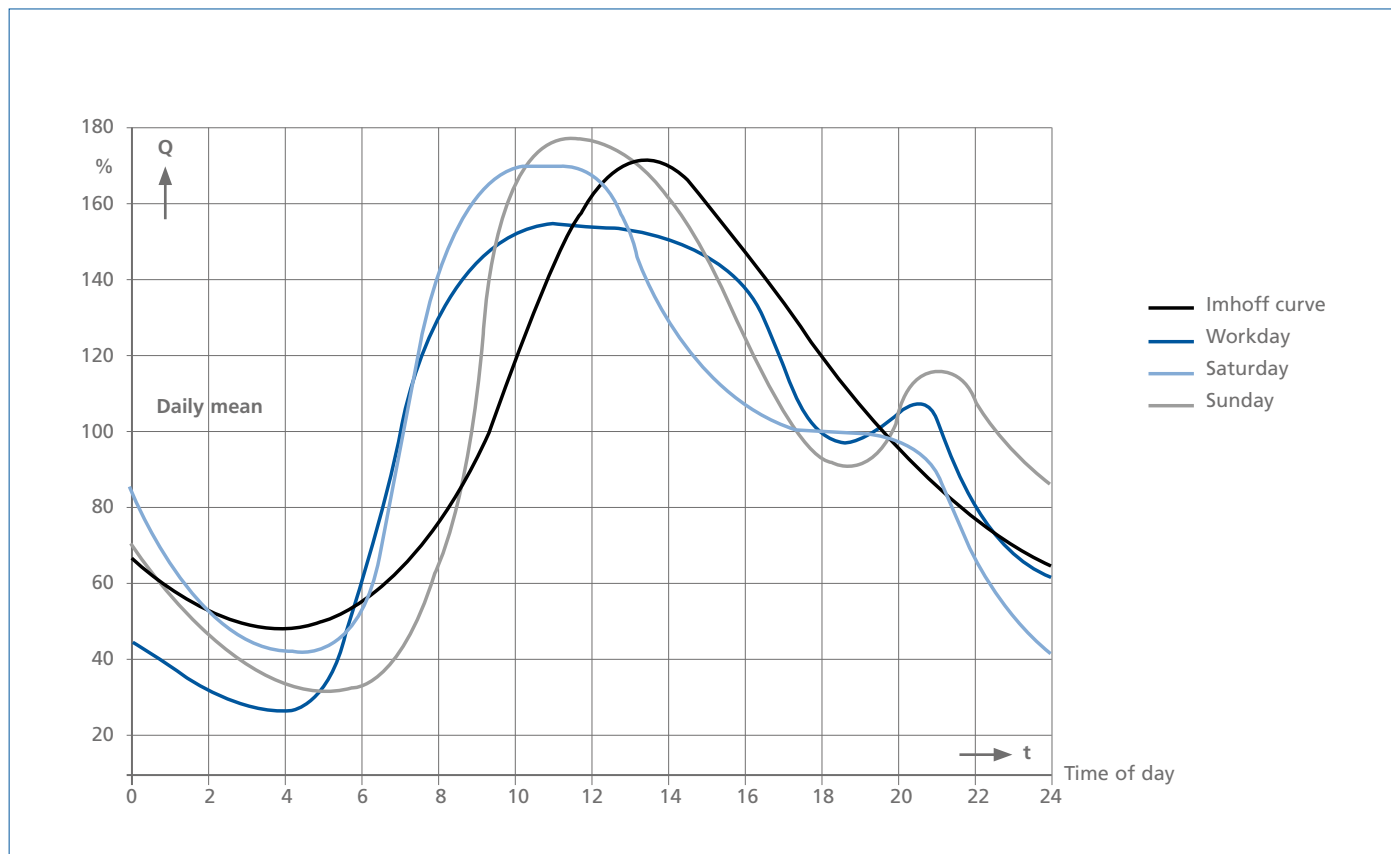Fig. 1: Example of inflow load curves during dry weather, primarily from a residential area



Fig. 2: Example of inflow load curves during dry weather, large proportion of industrial effluents

**Pump head**

The head H developed by a pump (e.g. expressed in [m]) is defined as the useful mechanical energy transferred to the flow per weight of the fluid handled, taking into account the local acceleration due to gravity.

To determine the total head H of a pumping system or a pump, the following information needs to be known:

- The height AOD (Above Ordnance Datum) of the invert of the inlet channel or pump sump floor
- The cut-in and cut-out points of the pumps (which equal the minimum submergence and the maximum permissible fluid level in the pump sump)
- The terrain (distance and heights) between the pumping station and the discharge point
- The valves, fittings and pipes installed, including their nominal diameters DN and their resistance coefficients $\zeta$
- The height AOD of the discharge point

The total head $H_{total}$ is described as follows:

$$H_{total} = z_2 - z_1 + \frac{p_2 - p_1}{\rho \cdot g} + \frac{v_2^2 - v_1^2}{2 \cdot g} + \sum H_L \qquad \text{Formula (1)}$$

**Key:**

| | |
|---|---|
| $H_{total}$ | = Total head [m] |
| $H_L$ | = Head loss [m] |
| $V_1$ | = Flow velocity, suction side [m/s] |
| $V_2$ | = Flow velocity, discharge side [m/s] |
| g | = Acceleration due to gravity = 9.81 m/s² |
| $p_1$ | = Static ambient pressure, suction side [N/m²] |
| $p_2$ | = Static ambient pressure, discharge side [N/m²] |
| $z_1$ | = Height of inlet cross-section, relative to reference level [m] |
| $z_2$ | = Height of outlet cross-section, relative to reference level [m] |
| $\rho$ | = Density of fluid handled [kg/m³] |

For wet-installed submersible motor pumps this equation can be further simplified. Only the height differences, also referred to as geodetic head $H_{geo}$, and the sum of all losses $\sum H_L$ have to be considered.

The total head $H_{total}$ can be described using the following simplified equation[3]:

$$H_{total} = H_{geo} + \sum H_L = H_{geo} + H_{LIND} + H_{VD} \qquad \text{Formula (2)}$$

**Key:**

| | |
|---|---|
| $H_{ges}$ | = Total head, sum of geodetic / static head and total head loss / dynamic head [m] |
| $H_{geo}$ | = Geodetic or static head, measurable difference in height between suction-side and discharge-side water level or AOD [m] |
| $H_L$ | = Total head loss [m] |
| $H_{LIND}$ | = Total head loss of valves, fittings and pipes on the discharge side of the pump – for multiple pump systems the losses of individual lines up to the collecting discharge line [m] |
| $H_{LD}$ | = Head losses of valves, fittings and pipes on the discharge side of the pump in the collecting discharge line [m] |

Fluids with increased kinematic viscosity play a minor role in waste water applications. However, the dry solids content DS and any other substances in the fluids are responsible for an increased power input at the pump shaft. When selecting a motor, an appropriate power reserve must therefore be taken into account.[4]

However, from approx. 40 m²/s upwards, a fluid's kinematic viscosity $\upsilon$ ($\upsilon = \eta / \rho$ [m²/s] oder [cSt]) does have an influence on the pump operating data. The kinematic viscosity is calculated as follows:

$$\upsilon = \frac{\eta}{\rho} \qquad \text{Formula (3)}$$

**Key:**

| | |
|---|---|
| $\upsilon$ | = Kinematic viscosity [m²/s] |
| $\eta$ | = Dynamic viscosity [Pa s] |
| $\rho$ | = Density of fluid handled [kg/m³] |

It is an important parameter in the sludge treatment processes of waste water treatment plants. Affected are the head, flow rate and the power input of a pump.

If centrifugal pumps are used for sludge pumping, the operating point should preferably lie within the full load and overload range of the characteristic curve. The pumps' sensitivity to gas increases in the part-load range. To determine the operating points for sludge transport, a correction factor is applied to the loss values calculated for water.

[3] Additional literature: KSB publication: Selecting Centrifugal Pumps
[4] Additional literature: KSB Centrifugal Pump Lexicon

a) Raw sludge (present in the mechanical treatment stage of a waste water treatment plant, it is directly removed from the primary clarifier; this sludge type has a water content of approx. 97 %)

b) Thickened raw sludge from static thickeners, recirculated sludge, heating sludge, faecal sludge or return activated sludge (present in the sludge treatment process of a waste water treatment plant; this sludge type has a water content of approx 95 - 96 %)

| Nominal pipe diameter DN | 100 | 125 | 150 | 200 | 250 | 300 | Note |
|---|---|---|---|---|---|---|---|
| Correction factor for sludge type a) | 1.7 | 1.6 | 1.5 | 1.4 | 1.3 | 1.2 | of $H_L$ for water |
| Correction factor for sludge type b) | 2.1 | 2.0 | 1.9 | 1.8 | 1.7 | 1.6 | of $H_L$ for water |

Table 1: Correction factors for sludges

## NPSH value

The NPSH value (Net Positive Suction Head) is significant in assessing the suction characteristics of a centrifugal pump. It describes the minimum inlet pressure required by each centrifugal pump for reliable and cavitation-free operation.[5] A distinction must be made between the $NPSH_{3\%}$ value (permissible cavitation which causes the pump head to drop by 3 percent), also referred to as $NPSH_{req}$ (req = required), and the NPSH value of the system, also referred to as $NPSH_{av}$ (av = available).

To be able to document the $NPSH_{3\%}$ value of a pump in the sales literature, pump manufacturers conduct a test run in a test field using a special dry pump installation. In a wet installation this NPSH value is practically impossible to measure. As the $NPSH_{3\%}$ value changes in relation to the flow rate, it is entered as a function of the flow rate **$NPSH_{req}$ = f (Q)**. This value describes the pressure head in [m] required above the fluid's vapour pressure at the impeller inlet. The intersection $P_{s'}$ of the pump shaft axis and a plane situated at right angles to the pump shaft passing through the outer points of the vane leading edge is taken as a reference point for NPSH, see Fig. 3.

The $NPSH_{3\%}$ value can be influenced by the pump manufacturer by defining the impeller type and design (impeller eye diameter, number of vanes and design of the vane leading edge) as well as the rated speed of the pump.
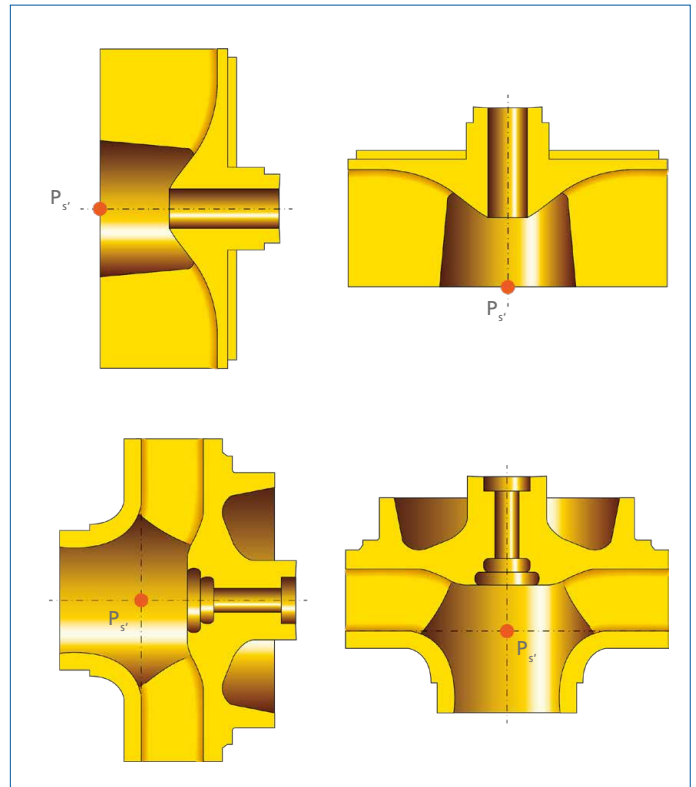


Fig. 3: Position of the reference point $P_{s'}$ for various impeller types (Source: KSB publication: Selecting Centrifual Pumps [01.08])

[5] Additional literature: Brochure Europump 1974
"NPSH bei Kreiselpumpen – Bedeutung, Berechnung, Messung"
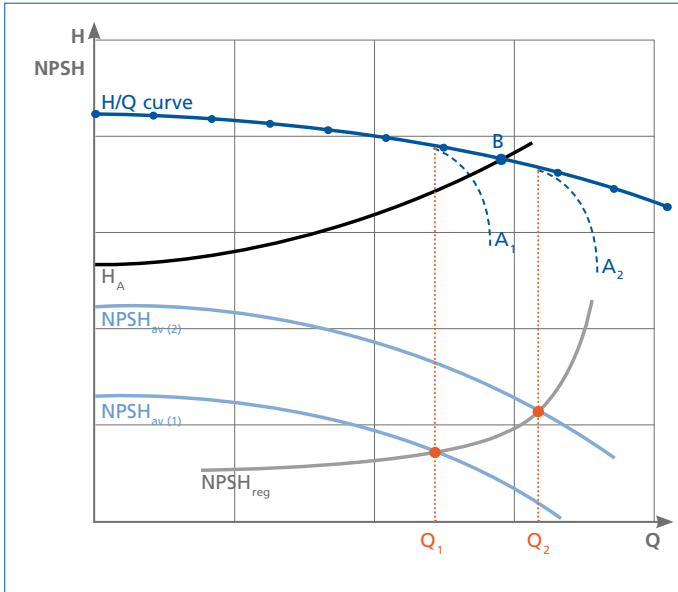(NPSH for centrifual pumps – significance, calculation, measurements)

Fig. 4: Impact of NPSH$_{av}$ on the H/Q curve of the pump
(Source: KSB Centrifugal Pump Lexicon [01.06])



Fig. 5: Impeller showing cavitation damage
(Source: KSB Centrifugal Pump Lexicon [01.06])

The most critical area is the inlet of the vane passage at the impeller, which is the narrowest passage to be passed by the fluid downstream of the suction line for dry-installed pumps and downstream of the pump inlet for wet-installed pumps. The flow around the vane leading edges leads to an unavoidable local pressure reduction in this area. If the pressure falls below the vapour pressure, vapour bubbles develop. These bubbles are transported along with the flow and implode abruptly when the pressure in the vane passage rises again.

The formation and abrupt implosion of vapour bubbles is called cavitation.

Cavitation can lead to severe negative conditions, from a drop in head and efficiency through to flow separation, rough running, vibrations and loud noise emissions to pitting due to erosion on the impeller (see Fig. 5) or internal pump parts. Cavitation is therefore only acceptable within permissible limits. The permissible cavitation for each individual case also depends on the operating conditions, the period of time the pump is operated outside the permissible range, the fluid handled and, particularly, the materials of the wetted parts (especially the impeller).

Cavitation-free pump operation requires the following condition to be met:

$$NPSH_{av} \text{ (System)} \geq NPSH_{req}$$

**where $NPSH_{req} = NPSH_{3\%}$ (Pump) $+ S$**

Formula (4)

**Key:**

| | |
|---|---|
| S | = Safety allowance [m] |
| NPSH$_{av}$ | = Net positive suction head of the system [m] |
| NPSH$_{req}$ | = Suction characteristics of a centrifugal pump [m] |
| NPSH$_{3\%}$ | = Suction characteristics of a pump with permissible cavitation which causes the pump head to drop by 3 percent |

The safety allowance (S) depends on the impeller's design and hydraulic characteristics, and is essentially determined by the pump manufacturer on the basis of the above conditions and the permissible level of cavitation. It is taken into account in the NPSH$_{req}$ value specified.

Formula 4 can be used to calculate the $NPSH_{req}$ value of the system for reference point $P_{s'}$, taking the following influences into account[6]:

$$NPSH_{req} = z_e + \frac{p_e + p_b - p_D}{\rho \cdot g} + \frac{v_e^2}{2 \cdot g} - H_{LS}$$

Formula (5)

**Key:**

| | |
|---|---|
| $H_{LS}$ | = Head loss of suction line [m] |
| $z_e$ | = Difference in height between the reference point $P_{s'}$ and the suction-side fluid level; the value becomes positive if the reference point is below the fluid level. [m] |
| $p_e$ | = Positive pressure (above atmospheric pressure) acting on the suction-side fluid; required for pumping from a tank or when using a booster pump [N/m²]. |
| $p_b$ | = Atmospheric pressure, depending on the pump's installation altitude [N/m²] |
| $p_D$ | = Vaporisation pressure, largely dependent on the fluid temperature (> 40°C) [N/m²] |
| $v_e$ | = Mean flow velocity in the system inlet cross-section, i.e. in the inlet-side tank [m/s] |
| $g$ | = Acceleration due to gravity = 9.81 m/s² |
| $\rho$ | = Density of fluid handled [kg/m³] |

Simplified:

$$NPSH_{av} = z_e + 10 \text{ m}$$

The diagram in Fig. 4 illustrates the point at which the permissible cavitation is exceeded. At the intersection of $NPSH_{av}$ and $NPSH_{req}$ the condition of formula 4 is not fulfilled, i.e. to the right of the intersection the flow rate of the fluid no longer rises and the head drops rapidly. This type of curve is referred to as a "head breakdown curve". Prolonged operation in this condition will damage the pump components (impeller, bearings, shaft seal, etc.). Fig. 5 shows damage to an impeller's vane leading edge (material loss including pitting) due to cavitation. To re-adjust operation to operating point B, $NPSH_{av}$ can be increased (e.g. by increasing the fluid level in the inlet tank).

**Input power**

The input power $P_2$ of a centrifugal pump is the mechanical power delivered by the drive to the pump shaft or coupling. It can be calculated using formula 6.[7]

**Q in m³/h**

Formula (6)

$$P_2 = \frac{Q \cdot H \cdot g \cdot \rho}{1000 \cdot \eta_p} \text{ [kW]}$$

**Q in l/s**

$$P_2 = \frac{Q \cdot H \cdot \rho}{102 \cdot \eta_p} \text{ [kW]}$$

**Q in m³/s**

$$P_2 = \frac{Q \cdot H \cdot \rho}{0.102 \cdot \eta_p} \text{ [kW]}$$

**Q in m³/h and ρ in kg/dm³**

$$P_2 = \frac{Q \cdot H \cdot \rho}{367 \cdot \eta_p} \text{ [kW]}$$

**Key:**

| | |
|---|---|
| Q | = Flow rate [m³/h] or unit as specified above equation |
| H | = Head in [m] |
| $\eta_p$ | = Efficiency of pump or coupling, provides information on a pump's internal hydraulic losses comprising friction losses and shock losses. Both loss types are defined as a function of flow rate. To be specified by a figure between 0 and 1, not in %, e.g. 82.5 % = 0.825 |
| g | = Acceleration due to gravity = 9.81 m/s² |
| $\rho$ | = Density of fluid handled [kg/m³ ] (the value is defined as 1000 kg/m³ for clean water and in documented characteristic curves; the following values must be used for waste water or sludges.) |

**Waste water | municipal, untreated**
with a temperature between 10-40 °C   $\rho$ = 1.030 kg/m³
**Waste water | municipal, treated**
with a temperature between 10-40 °C   $\rho$ = 1.000 kg/m³
**Sludges with a dry solids content of 2 %**
and with a temperature between 0-60 °C   $\rho$ = 1.011 kg/m³
**Sludges with a dry solids content of 4 %**
and with a temperature between 0-60 °C   $\rho$ = 1.021 kg/m³
**Sludges with a dry solids content of 6 %**
and with a temperature between 0-60 °C   $\rho$ = 1.032kg/m³

| | |
|---|---|
| $P_2$ | = Mechanical power absorbed [kW] |

The input power $P_2$ must not be confused with the power available at the drive (referred to as drive rating or rated motor power $P_R$) The rated motor power is indicated by the motor manufacturer on the rating plate.

On submersible motor pumps only the power $P_1$ consumed by the motor can be measured.

This value includes the internal motor losses indicated by the motor efficiency $\eta_M$.

In this case the pump input power can be calculated using formula 7.

[6] Brochure Europump 1974 "NPSH bei Kreiselpumpen – Bedeutung, Berechnung, Messung" (NPSH for centrifual pumps – significance, calculation, measurements)
[7] Additional literature: KSB Centrifugal Pump Lexicon

$$P_2 = P_1 \cdot \eta_M \text{ [kW]} \quad \text{or} \quad P_1 = \frac{P_2}{\eta_M}$$

Formula (7)

**Key:**

$\eta_M$    = Motor efficiency, provides information on a motor's internal mechanical losses (friction losses) and the magnetic losses (clearance gap losses).
To be specified by a figure between 0 and 1, not in %, e.g. 92.7 % = 0.927

$P_1$    = Power consumed by the motor [kW]

$P_2$    = Mechanical power absorbed [kW]

When determining the required motor rating, we recommend adding a minimum power reserve above the pump input power, see table 2. This power reserve will generally compensate for the pump's manufacturing tolerances, fluctuations in the fluid properties (e.g. changes in density, changes in the dry solids content DS and/or the presence of any other substances in the fluid). The motor output power should cover the input power requirements of the installed impeller with its respective diameter for the entire operating range.

For an input power below 5 kW, the power reserve percentage should be agreed between the manufacturer / supplier and the operator.

| Required pump input power | Motor power reserve for mains operation | Motor power reserve for operation on a frequency inverter |
|---|---|---|
| <30 kW | 10 % | 15 % |
| >30 kW | 5 % | 10 % |

Source: Sewatec type series booklet and operating manual

Table 2: Recommended minimum power reserves

## Head loss calculation

The following section provides information on the calculation of heads actually required from pumping systems in a comprehensible manner using example systems.

The pumping task: a (waste) water quantity X [litre] is to be transported from A to B.

The recommended flow velocities for waste water (see next section) are used for preliminary estimation of pipe cross-sections. On the one hand it is important to observe the minimum velocities required to ensure the reliable transport of solids present in the fluid. On the other hand, economic considerations regarding maximum flow velocities must also be taken into account.

The resulting diameters are used to calculate the pressure losses. As already described in section "1.2 Pump head" with equation 5, the head H is calculated only from the static head $H_{geo}$ (measurable difference in height between the suction-side and discharge-side water level or the outlet level of the pipe) and the dynamic head $\Sigma H_L$ (the sum of all losses dependent on the flow rate).

**Head losses**

**Head losses in the piping**

The head loss $H_L$ is calculated for straight pipes as follows[8]:

$$H_L = \lambda \cdot \frac{L}{d} \cdot \frac{v^2}{2 \cdot g}$$

Formula (8)

**Key:**

$\lambda$    = Pipe friction coefficient, dependent on the state of fluid flow which is described by the REYNOLDS number Re.
Laminar flow Re <2320   $\lambda$ = 64/Re,
turbulent flow Re >2320   $\lambda$ = 0.309/(lg Re/7)²

L    = Pipe length [m]

d    = Pipe diameter [m]

v    = Flow velocity in a characteristic cross-sectional area A [m/s]

g    = Acceleration due to gravity = 9.81 m/s²

$H_L$    = Head loss

[8-10] Additional literature: KSB publication: Selecting Centrifugal Pumps

With sufficiently large REYNOLDS numbers (Re > 10 000), $\lambda$ is only dependent on the relative roughness k/D resulting from the absolute roughness k and the pipe diameter D. Formula 9 can be used in accordance with MOODY[9]:

$$\lambda = 0.0055 + \left(0.15 \sqrt[3]{\frac{D}{k}}\ \right)$$

Formula (9)

**Key:**

k = absolute pipe roughness, for cast iron pipes in new condition approx. 100 µm

For practical use, the graph below shows the head losses $H_L$ per 100 m in a straight length of piping. ($H_L$ is calculated using formula 9 as when calculating head loss in valves.)
The values shown in Fig. 6 apply to clean water at 20 °C or to fluids with the same kinematic viscosity for a fully filled cast iron pipe (k = 0.1 mm).

The head losses $H_L$ in Fig. 6 must be multiplied by the following factors:

0.8     for new rolled steel pipes
1.7     for encrusted pipes
1.25     for old, slightly rusty steel pipes.

**Head losses in valves and fittings**

Head loss $H_L$ for valves and fittings is calculated as follows[10]:

$$H_L = \zeta \cdot \frac{\upsilon^2}{2 \cdot g}$$

Formula (10)

**Key:**

$\zeta$ = Loss coefficient (dimensionless)
$\upsilon$ = Flow velocity in a characteristic cross-sectional area A [m/s]
g = Acceleration due to gravity = 9.81 $(m/s)^2$

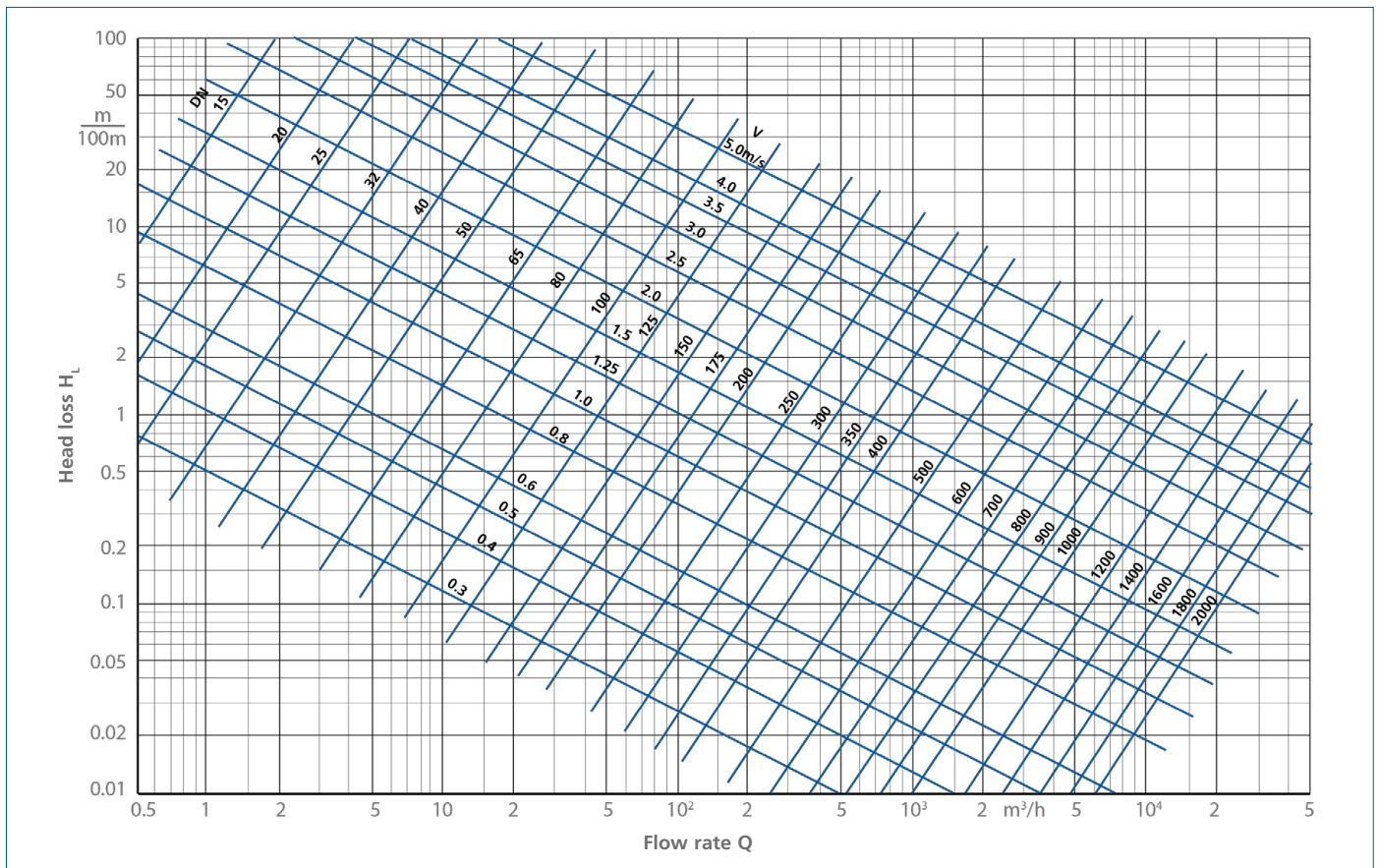Table 3 below lists a selection of the most common valves and fittings used in waste water transport.



Fig. 6: Head losses for straight pipes (cast iron, as-new) from DN 15 to 2,000 and flow rates Q from 0.5 to 50,000 m³/h (flow velocity v in m/s, water at 20 °C)
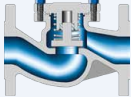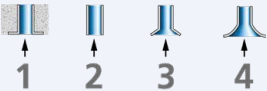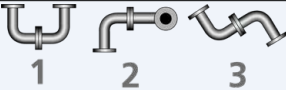Source: KSB publication: Selecting Centrifugal Pumps
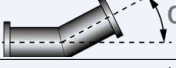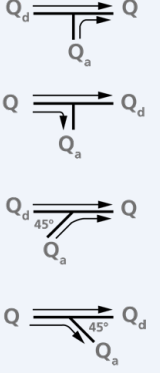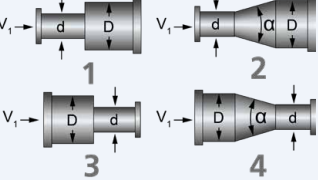
| Valve / Fitting | Description | Resistance coefficient $\zeta$ reference value |
|---|---|---|
| | Slide disc valves | DN 50 $\quad\quad\quad$ $\zeta = 0.45$<br>DN 80 $\quad\quad\quad$ $\zeta = 0.35$<br>DN 100 - DN 1000 $\quad$ $\zeta = 0.3$ |
| | Swing check valves | DN 50 $\quad\quad\quad\quad$ $\zeta = $ min 0.40 - max 2.0<br>DN 80 - DN 250 $\quad$ $\zeta = $ min 0.40 - max 1.8-1.5<br>DN 100 - DN 1000 $\quad$ $\zeta = $ min 0.30 - max 1.4-1.0 |
| | Inlet pipe fittings | Inlet edge $\quad$ Type $\quad$ 1 $\quad\quad$ 2 $\quad\quad$ 3 $\quad\quad$ 4<br>Sharp $\quad\quad\quad$ $\zeta = \quad$ 0.5 $\quad\quad$ 3<br>Chamfered $\quad$ $\zeta = \quad$ 0.25 $\quad$ 0.55 $\quad$ 0.2 $\quad$ 0.05 |
| | Discharge pipe fittings | $\zeta = 1$ downstream of an adequate length of straight pipe with an approximately uniform velocity distribution in the outlet cross-section.<br>$\zeta = 2$ for a very irregular velocity distribution, for example immediately downstream of a pipe fitting or a valve, etc. |
| | Cast iron elbow, 90° | R = DN + 100 $\quad\quad$ $\zeta = 0.5$ for all nominal diameters |
| | Pipe bend, 90° | Nominal diameter $\quad$ 1 $\quad\quad$ 2 $\quad\quad$ 3 $\quad\quad$ 4<br>R = 2 to 4 x DN $\quad$ $\zeta = \quad$ 0.26 $\quad$ 0.23 $\quad$ 0.21 $\quad$ 0.19 $\quad$ 0.18<br>broken $\quad\quad\quad$ $\zeta = \quad$ 0.25 $\quad$ 0.55 $\quad$ 0.2 $\quad$ 0.05<br><br>If the angle is $\quad\quad\quad\quad\quad\quad$ 60° $\quad$ 45° $\quad$ 30° $\quad$ 15°<br>factor $\zeta$ is multiplied by factor $\quad$ 0.85 $\quad$ 0.7 $\quad$ 0.45 $\quad$ 0.3 |
| | Combination of elbows and/or pipe bends | For the combination of single elbows, the $\zeta$ value of these elbows is multiplied by only one factor. $\quad\quad$ 1 $\quad\quad$ 2 $\quad\quad$ 3<br>$\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad$ 1.4 $\quad$ 1.6 $\quad$ 1.8 |
| | Bends | If the angle is $\quad\quad\quad\quad$ 90° $\quad$ 60° $\quad$ 45° $\quad$ 30° $\quad$ 15°<br>$\quad\quad\quad\quad\quad$ $\zeta = $ 1.3 $\quad$ 0.7 $\quad$ 0.35 $\quad$ 0.2 $\quad$ 0.1 |
| | Combinations with 90° bend | Combination $\quad$ 1 $\quad\quad$ 2 $\quad\quad$ 3<br>$\quad\quad\quad$ $\zeta = \quad$ 2.5 $\quad$ 3 $\quad\quad$ 5 |
| | Branch fittings | The loss coefficients $\zeta a$ for the branched-off flow Qa or $\zeta d$ for the main flow Qd = Q - Qa refer to the velocity of the total flow Q. On the basis of this definition, $\zeta a$ or $\zeta d$ may have negative $\zeta$ values; in this case they are indicative of a pressure gain instead of a loss.<br><br>$\quad\quad\quad\quad\quad\quad\quad$ Qa/Q $\quad$ 0.2 $\quad\quad$ 0.4 $\quad\quad$ 0.6 $\quad\quad$ 0.8 $\quad\quad$ 1<br>Branch fitting 1 $\quad$ $\zeta a = $ -0.4 $\quad$ 0.08 $\quad$ 0.47 $\quad$ 0.72 $\quad$ 0.91<br>$\quad\quad\quad\quad\quad\quad$ $\zeta d = $ 0.17 $\quad$ 0.3 $\quad\quad$ 0.41 $\quad$ 0.51 $\quad$ -<br>Branch fitting 2 $\quad$ $\zeta a = $ 0.88 $\quad$ 0.89 $\quad$ 0.95 $\quad$ 1.1 $\quad\quad$ 1.28<br>$\quad\quad\quad\quad\quad\quad$ $\zeta d = $ -0.08 $\quad$ -0.05 $\quad$ 0.07 $\quad$ 0.21 $\quad$ -<br>Branch fitting 3 $\quad$ $\zeta a = $ -0.38 $\quad$ 0 $\quad\quad$ 0.22 $\quad$ 0.37 $\quad$ 0.37<br>$\quad\quad\quad\quad\quad\quad$ $\zeta d = $ 0.17 $\quad$ 0.19 $\quad$ 0.09 $\quad$ -0.17 $\quad$ -<br>Branch fitting 4 $\quad$ $\zeta a = $ 0.88 $\quad$ 0.5 $\quad\quad$ 0.38 $\quad$ 0.35 $\quad$ 0.48<br>$\quad\quad\quad\quad\quad\quad$ $\zeta d = $ -0.08 $\quad$ -0.04 $\quad$ 0.07 $\quad$ 0.2 $\quad\quad$ - |
| | Expansions<br><br>Reductions | $\quad\quad\quad\quad\quad\quad\quad\quad\quad$ d/D $\quad$ 0.5 $\quad$ 0.6 $\quad$ 0.7 $\quad$ 0.8 $\quad$ 0.9<br>Type 1 $\quad\quad\quad\quad\quad\quad$ $\zeta = $ 0.56 $\quad$ 0.41 $\quad$ 0.26 $\quad$ 0.13 $\quad$ 0.04<br>$\quad\quad\quad\quad$ α = 8° $\quad$ $\zeta = $ 0.07 $\quad$ 0.05 $\quad$ 0.03 $\quad$ 0.02 $\quad$ 0.01<br>Type 2 for $\;$ α = 15° $\quad$ $\zeta = $ 0.15 $\quad$ 0.11 $\quad$ 0.07 $\quad$ 0.03 $\quad$ 0.01<br>$\quad\quad\quad\quad$ α = 20° $\quad$ $\zeta = $ 0.23 $\quad$ 0.17 $\quad$ 0.11 $\quad$ 0.05 $\quad$ 0.02<br>Type 3 $\quad\quad\quad\quad\quad\quad$ $\zeta = $ 4.80 $\quad$ 2.01 $\quad$ 0.88 $\quad$ 0.34 $\quad$ 0.11<br>Type 4 for 20 < α < 40° $\;$ $\zeta = $ 0.21 $\quad$ 0.10 $\quad$ 0.05 $\quad$ 0.02 $\quad$ 0.01 |

Table 3: Head losses in valves and fittings (selection)
$\quad\quad\quad$ Source: KSB publication: Selecting Centrifugal Pumps

## Calculation example for a vertical, wet-installed pump in suction head operation

The total head H comprises the static (geodetic) head $H_{geo}$ and the dynamic head $H_L$ resulting from the discharge-side head loss calculation $H_{Ld}$.

The suction-side head loss is not taken into account as the pump draws in the fluid directly through the suction nozzle without any suction-side piping elements.

$$H_{total} = H_{geo} + H_{Ld} \qquad \text{Formula (11)}$$

**Key:**

$H_{total}$ = Total head, sum of geodetic / static head and total head loss / dynamic head [m]

$H_{geo}$ = Geodetic or static head, measurable difference in height between suction-side and discharge-side water level or AOD [m]

$H_{Ld}$ = Head losses of valves, fittings and pipes on the discharge side of the pump in the collecting discharge line [m]

Static head (all data in [m])

| Suction-side water level | Discharge-side water level | Calculation | $H_{geo}$ |
|---|---|---|---|
| 202.5 | 215.6 | $H_{geo} = H_d - H_s$ | 13.10 |

Table 4: Calculation of geodetic head

Design volume flow rate Q in [m³/s] 0.167

The inlet head Ze is the difference between the water level in the suction chamber and the impeller inlet.
The inlet head can also be negative; in this case, pumping is then referred to as suction-lift operation. As a pipe is not installed on the pump's suction side, this cannot occur.

### Pressure loss calculation, discharge side $H_{Ld}$

| No. | Description | DN | Pcs | Length | $\zeta$ | Value | Factor | Calculation | V | $H_{Ld}$ | Note |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Duckfoot bend expansion DN 250/ 300 | 250 | 1 | | 0.030 | | | $H_{Ls} = \zeta \bullet \, v^2/2\,g$ | 3.3939 | 0.018 m | DN 250 / DN 300 |
| 2 | Pipe section, cast iron | 300 | | 4.0 | | 1.5 | 1.00 | $H_{Ls}=Value*Factor*L/100$ | 2.3569 | 0.060 m | |
| 3 | Swing check valve | 300 | 1 | | 1.000 | | | $H_{Ls} = \zeta \bullet \, v^2/2\,g$ | 2.3569 | 0.283 m | |
| 4 | Slide disc valve | 300 | 1 | | 0.300 | | | $H_{Ls} = \zeta \bullet \, v^2/2\,g$ | 2.3569 | 0.085 m | |
| 5 | Pipe bend, 90° R= d | 300 | 1 | | 0.210 | | | $H_{Ls} = \zeta \bullet \, v^2/2\,g$ | 2.3569 | 0.059 m | |
| 6 | Tee, cast iron (90° flow division) | 300 | 1 | | 1.300 | | | $H_{Ls} = \zeta \bullet \, v^2/2\,g$ | 2.3569 | 0.368 m | |
| 7 | Pipe section, cast iron | 300 | | 3.0 | | 1.5 | 1.00 | $H_{Ls}=Value*Factor*L/100$ | 2.3569 | 0.045 m | |
| 8 | Expansion, type 1 a=8° | 300 | 1 | | 0.025 | | | $H_{Ls} = \zeta \bullet \, v^2/2\,g$ | 2.3569 | 0.007 m | DN 300 / DN 400 |
| 9 - 15 | Pipe section, cast iron, old | 400 | | 846.0 | | 0.4 | 1.25 | $H_{Ls}=Value*Factor*L/100$ | 1.3259 | 4.230 m | |
| 16 | Pipe bend, 45° | 400 | 6 | | 0.200 | | 0.70 | $H_{Ls}=Value*Factor* \zeta \bullet \, v^2/2\,g$ | 1.3259 | 0.075 m | |
| 17 | Outlet edge, chamfered | 400 | 1 | | 1.000 | | | $H_{Ls} = \zeta \bullet \, v^2/2\,g$ | 1.3259 | 0.090 m | |
| | | | | | | | | | $H_{Ld}=$ | 5.320 m | |
| Required total head $H_{total}$ calculated using formula 11 = 13.10 m + 5.32 m = 18.420 m | | | | | | | | | | | |

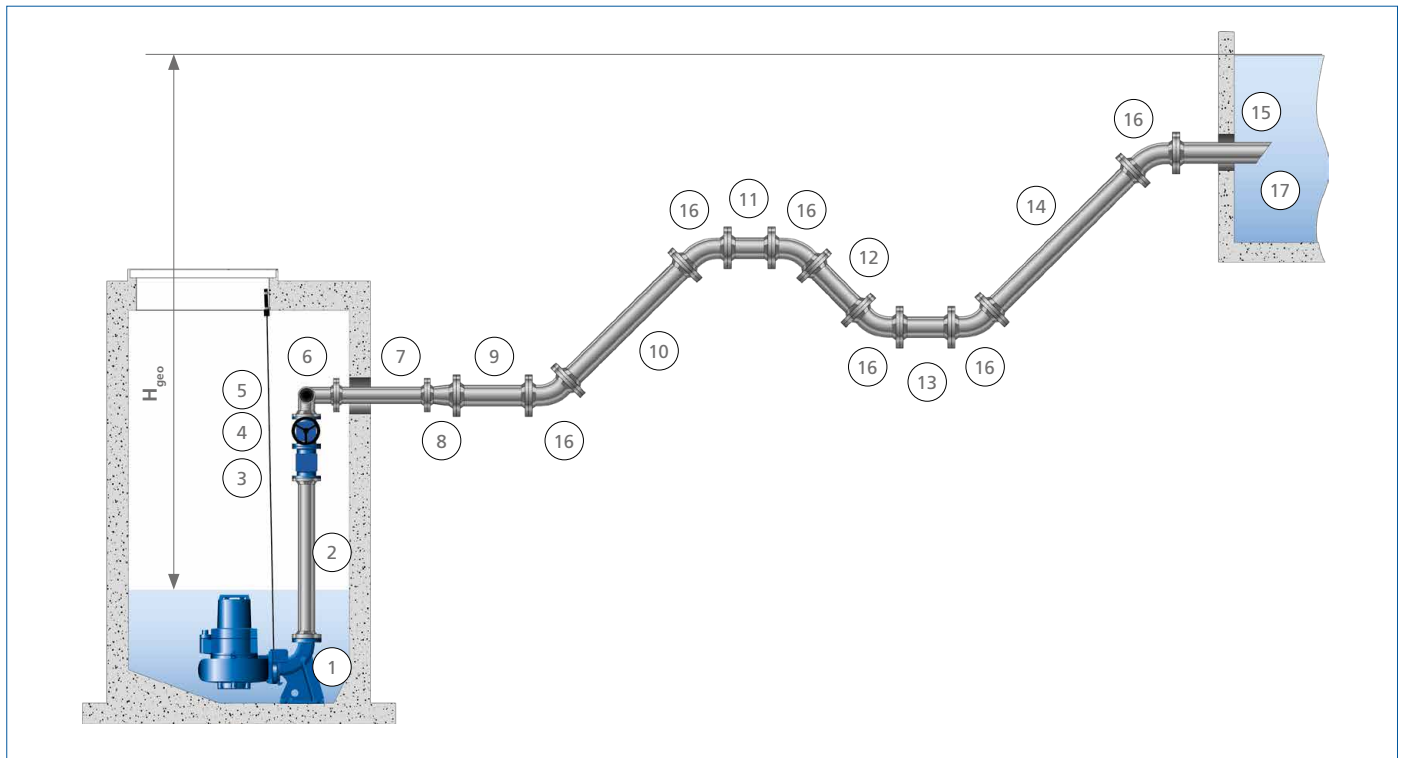Table 5: Calculation of the discharge-side head loss and the total head

Fig. 7: Illustration of piping for calculation example

## Pump selection (hydraulic selection)

Pump selection is primarily determined by the specification of operating conditions – in other words, the operating properties the pumps will be supplied for.

The operating conditions primarily comprise data on the fluid handled (e.g. temperature, density, viscosity, dry solids content, sand content or other substances in the fluid), the expected flow rate and the required head, the suction behaviour and the speed of the centrifugal pump. Also required is information on the drive size and rating, the operating mode, the expected frequency of starts as well as any factors determined by the system or environmental regulations such as the maximum permissible noise emission, permissible vibrations, piping forces and potential explosion hazards (indication of ATEX zones).

The KRT type series with its specific impeller types (free-flow impeller, single-, two-, three-channel impeller and open mixed flow impeller) is specially designed for the particular requirements of waste water transport such as the operating conditions and fluid composition.

For detailed information on matching impellers to a variety of fluids and on the impellers' application limits refer to the section "Configuration and Installation".

## Characteristic curves

When centrifugal pumps are operated at constant speed, their flow rate Q increases as their head H decreases.

In the characteristic head versus flow curve, also referred to as H/Q curve, the head H is plotted against the flow rate Q. Apart from the H/Q curve, a pump is characterised by its efficiency curve, its $NPSH_{req}$ or $NPSH_{3\%}$ curve and its input power curve, all as a function of the flow rate. When selecting a pump, all of these characteristic curves must be considered.

This is illustrated with an example in Fig. 8 showing the characteristic curves for a pump with a three-channel impeller. All hydraulic data has been determined in accordance with the applicable EN ISO 9906 standard, assuming that clean water is handled.

We distinguish between flat and steep curves. On a steep curve, the change in flow rate will be smaller than on a flat curve for an identical change of head. Pumps with steep head versus flow curves are advantageous for flow rate control.
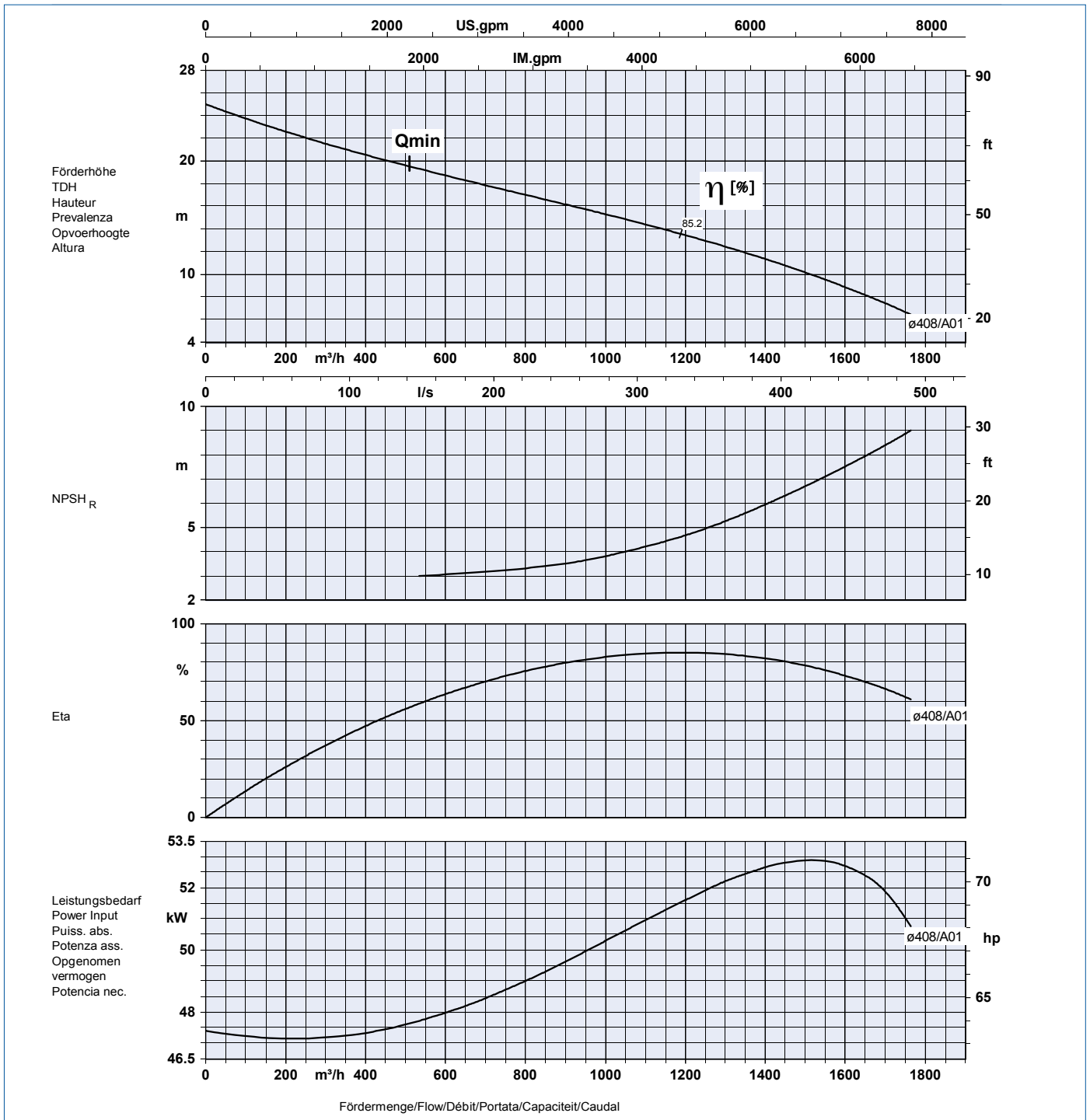
Fig. 8: Characteristic curves for a three-channel impeller with $n_s$ approx. 80 rpm and a pump speed of n = 960 rpm
Source: KSB publication: Selecting Centrifugal Pumps

## Pump characteristic curve

Generally, the curve referred to as the pump characteristic curve is the characteristic head versus flow curve (H/Q curve). As no pump can be operated without losses, the internal hydraulic losses of the pump (friction losses and shock losses) must be subtracted from the theoretical (loss-free) H/Q curve used for selection. The internal hydraulic losses can be defined as a function of the flow rate.

While friction losses rise continuously with increasing flow rates, shock losses rise with both decreasing and increasing flow rates, i.e. when the pump does not run at its rated flow rate or capacity (also referred to as $Q_{shock-free}$).
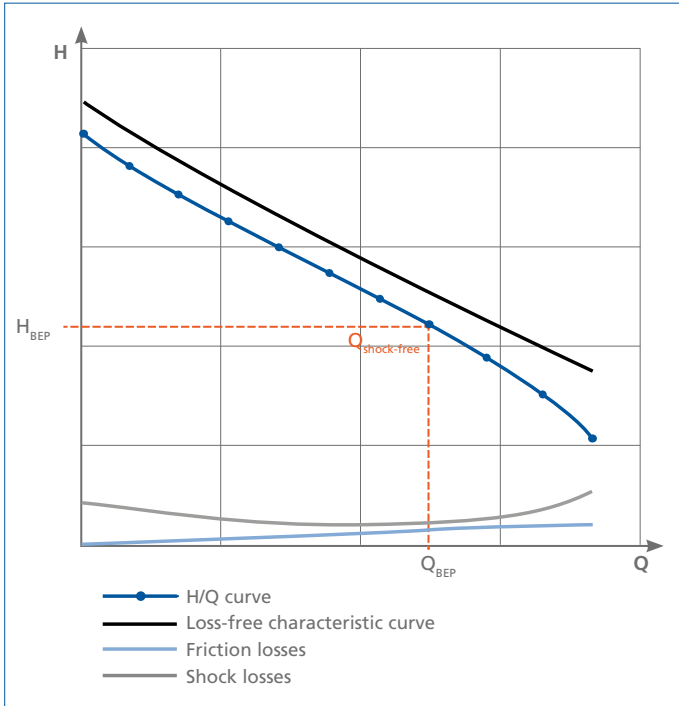Fig. 9 shows a proportional representation of hydraulic losses.

Fig. 9: Characteristic head versus flow curve and its reduction by internal hydraulic losses. Curves plotted in percentage ratios referring to the best efficiency point.
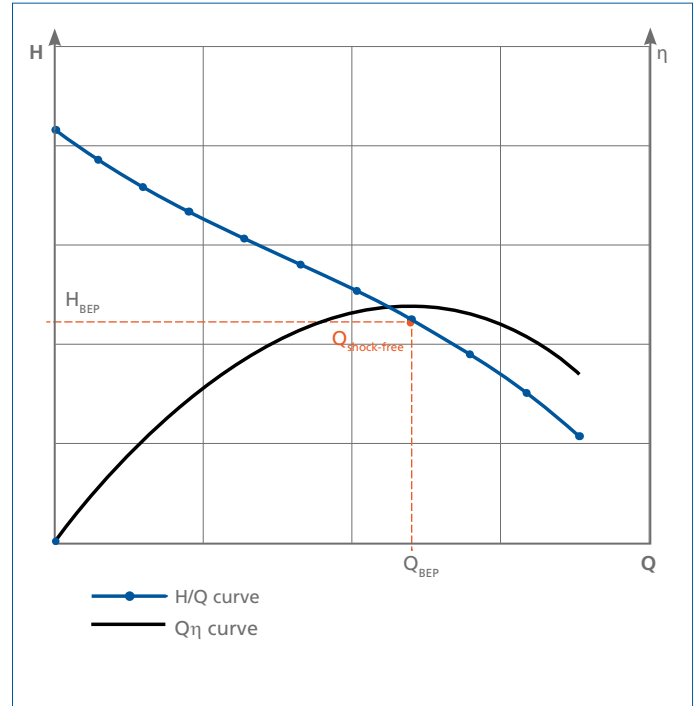


Fig. 10: Efficiency curve $\eta = f(Q)$.
Curves plotted in percentage ratios referring to the best efficiency point.

From zero flow to $Q_{BEP}$ (~$Q_{shock-free}$), the efficiency curve Q-$\eta$ rises to a maximum value and then drops again. The efficiency curve reflects the internal pump losses. The curve has been plotted in Fig. 10.

Fig. 11 and Fig. 12 show the curves of $NPSH_{3\%}$ and the pump input power $P_2$.

The $NPSH_{3\%}$ curve presents the suction characteristics of the pump (see section "1.3 NPSH value"). The pump input power curve is used to determine the required motor rating.
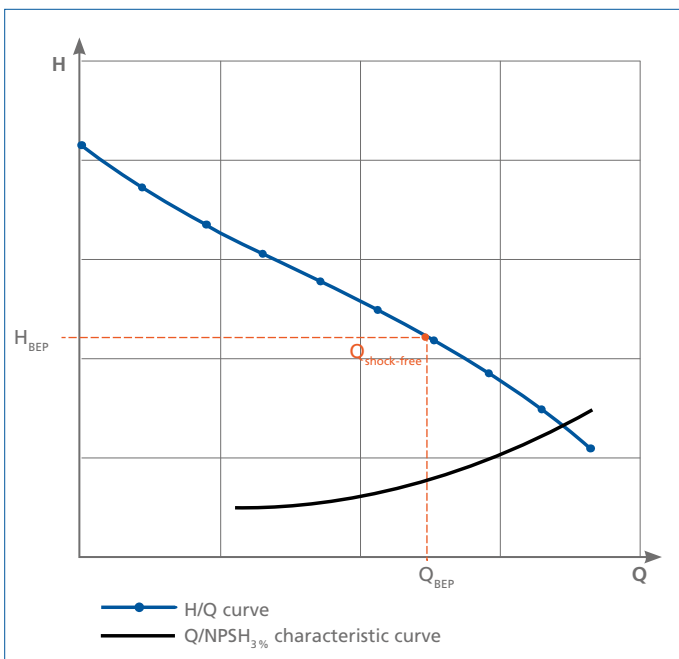


Fig. 11: $NPSH_{3\%}$ characteristic curve, $NPSH_{3\%} = f(Q)$.
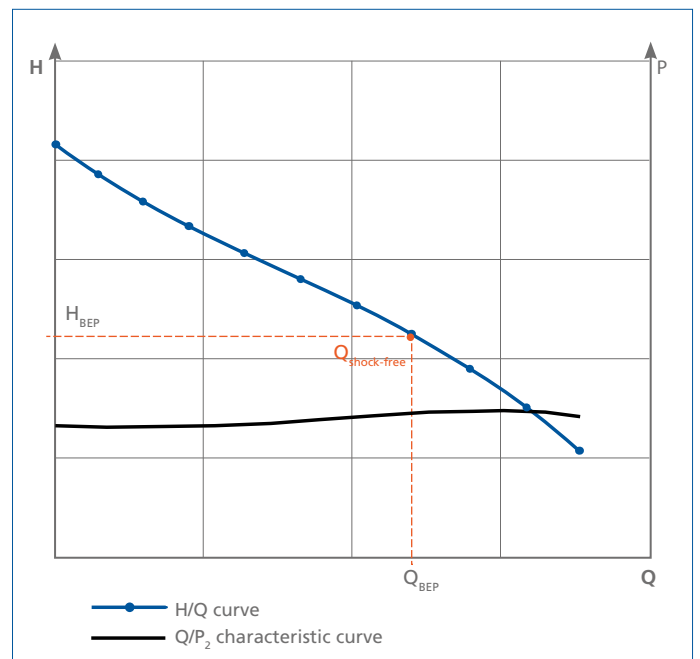Curves plotted in percentage ratios referring to the best efficiency point.



Fig. 12: Electrical pump input power $P_2 = f(Q)$.
Curves plotted in percentage ratios referring to the best efficiency point.

## System characteristic curve

The system characteristic curve is also plotted as a function of the flow rate. As shown in Fig. 13 the system characteristic curve is composed of a constant static component and a dynamic component which is quadratically[11] proportional to the flow rate.

The static head component Hgeo is the measurable difference in height between the suction-side and discharge-side water levels. The dynamic head component comprises all head losses of valves, fittings and pipes that are installed or planned for installation on the suction side and the discharge side of the pump.

## Permissible pump operating limits

Every centrifugal pump comes with recommended application or operating limits. They denote the maximum and minimum points at which the pump can be used for design, system and drive reasons.

Operation within these limits is preliminary for the pump to be able to fulfil its pumping task for the entire duration of its planned service life. The key application and operating limits are discussed in the following sections.

## Operating point

The operating point or duty point of a pumping system is defined by the intersection of the system characteristic curve and pump characteristic curve[12] (Fig.13).

This point defines the resulting head and the corresponding flow rate. If the operating point needs to be changed, either the system characteristic curve or the pump characteristic curve must be changed. For further details refer to the section "Pump operating mode".

## Operating limits $Q_{min}$ and $Q_{max}$

The (hydraulic, mechanical and acoustic) operating characteristics of a centrifugal pump are mainly defined by the position of the operating point with respect to the point $Q_{BEP}$[13].

A pump should be selected whose operating point is, if possible, near the best efficiency point (highest efficiency point on pump characteristic curve) at $Q_{OP}$ approx. 0.8 to 1.2 x $Q_{BEP}$.
Operation within this range will keep down costs for energy and maintenance as well as minimising hydraulic excitation forces. In daily practice, however, certain processes may require the pump set to be run in off-design conditions. The further the operating point from the best efficiency point, the less favourable the approach flow at the impeller vanes and the diffuser (casing). Areas of flow separation will develop which will increasingly interfere with the energy transfer to the fluid. The hydraulic forces (radial and axial forces) will increase, and so will mechanical vibration, noise and also cavitation, leading to visible/audible effects.
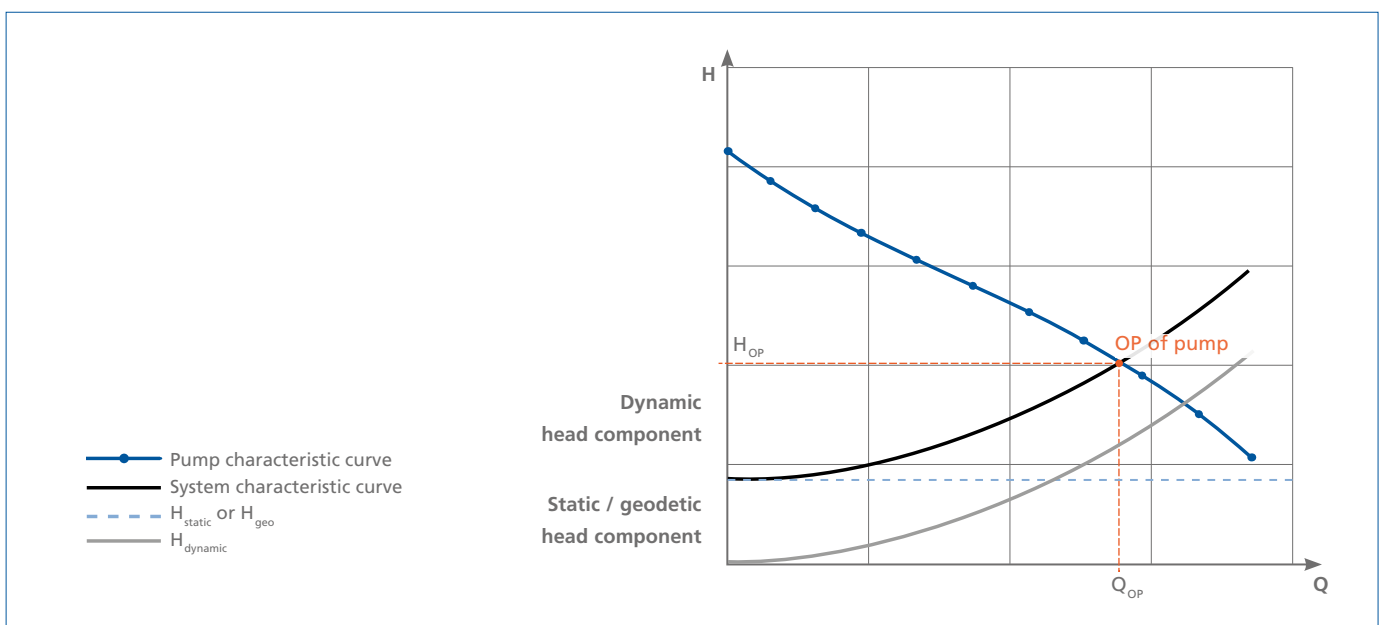


Legend:
- Pump characteristic curve
- System characteristic curve
- $H_{static}$ or $H_{geo}$
- $H_{dynamic}$

Fig. 13: System characteristic curve – the sum of the static and dynamic head components

[11] Note: This only applies if the dependence of the pipe friction on the Reynolds number is not taken into account.
[12] Note: The characteristic head versus flow curve of the pump shall hereinafter be referred to as pump characteristic curve, the commonly used short form.
[13] Note: BEP, also referred to as the pump's best efficiency point, shall hereinafter be used as the commonly used form.
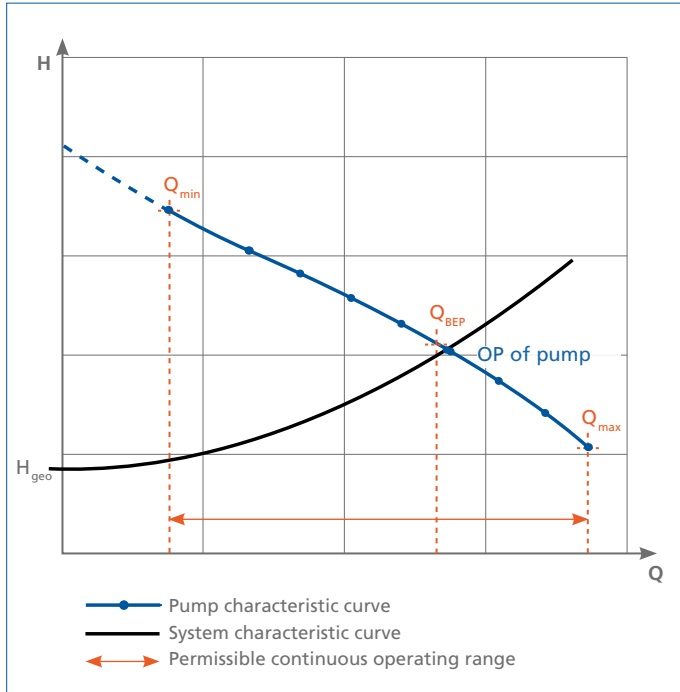
Fig. 14: Operating limits $Q_{min}$ and $Q_{max}$ – illustration of the permissible range for continuous operation of a centrifugal pump ($Q_{min}$ approx. 0.3 x $Q_{BEP}$ and $Q_{max}$ approx. 1.4 x $Q_{BEP}$)
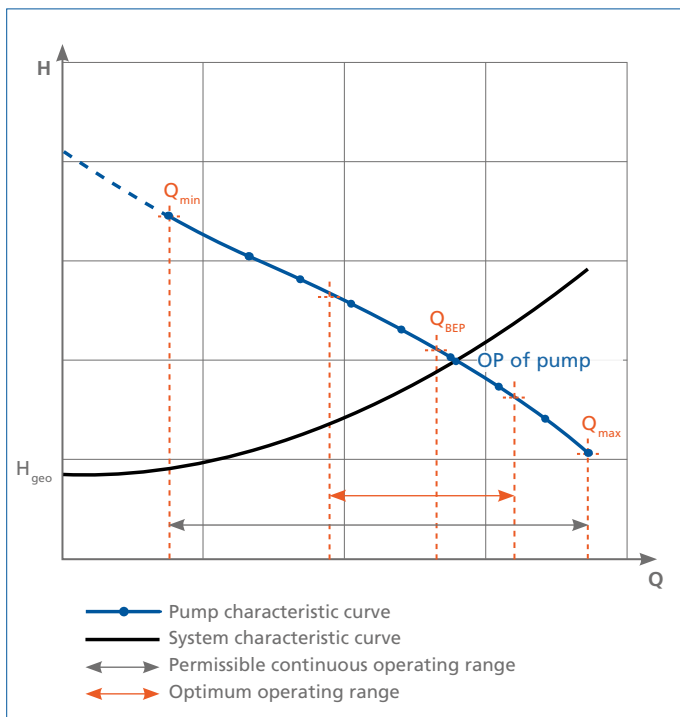


Fig. 15: Preferable or optimum operating range for waste water transport ($Q_{min}$ approx. 0.8 x $Q_{BEP}$, $Q_{max}$ approx. 1.2 x $Q_{BEP}$)

Pump manufacturers define the permissible range for continuous operation of their pumps by specifying the limits $Q_{min}$ and $Q_{max}$ (if no limits are given, the end of the illustrated pump characteristic curve is considered $Q_{max}$). Usually, the permissible operating range indicated lies between approx. 0.3 and 1.4 x $Q_{BEP}$. Stricter limits may however apply for the operating ranges of certain pumps. When operating pump sets outside of the permissible operating range, excessive loads and early wear of pump components must be expected.

**Special aspects of waste water transport**

A centrifugal pump can only be as good as the waste water treatment plant it is integrated in. The pump can only be operated reliably if the peripheral systems of the plant, the fluid to be handled (properties and composition), the control system and the operating mode match the characteristics of the centrifugal pump and its hydraulic system.

Drawing on our experts' practical experience, we have compiled some special aspects of waste water transport. These should be taken into account when planning a waste water system to make sure it has a high operating reliability (low-clogging operation).

- Operating point near best efficiency point. The $Q_{OP}$ approx. 0.8 to 1.2 x $Q_{BEP}$ range is not only the most favourable operating range from an energy point of view, but also the range in which the substances contained in the fluid are transported at higher velocities. The range is marked in Fig. 15. Especially under part-load conditions between $Q_{min}$ and 0.8 x $Q_{BEP}$ the transport of substances in the fluid is more or less restricted due to the low flow rates (low flow velocity). Continuous operation of the pumps in this range can lead to clogging of the impeller. This characteristic of centrifugal pumps is significant when defining the control and operating range and adjusting the operating point accordingly. This applies in particular to speed control, change of geodetic head between cut-in and cut-out fluid level in the pump sump, and parallel operation of two or multiple pumps.

- Make sure the motor has sufficient power reserve. All hydraulic pump data supplied by the manufacturer has been determined with reference to the applicable EN ISO 9906 standard, assuming that clean water is handled. It is impossible for a consultant to predict the waste water's exact properties; the trend is towards ever increasing solids and fibre contents in waste water. The motor power reserves specified in table 2 must be observed.

- Select a suitable impeller. Centrifugal pumps from the KRT pump range and their waste water specific impeller types (free-flow impeller, single-, two-, three-channel impeller and open mixed flow impeller) are specially designed for the particular requirements and operating conditions of waste water transport.

- Pump foundations and pipe mounting. Pumps equipped with channel-type impellers inevitably generate pressure pulsations as the impeller vane passes the casing tongue, regardless of whether it is a single-, two-, or three-channel impeller. This excitation, also referred to as the blade passing frequency, could potentially generate sympathetic vibrations in the piping system. The foundations and pipe mounting must be capable of handling the demands placed on them by these vibrations. The use of rubber expansion joints in the piping system must be avoided as far as possible. (Refer to section "Piping" for further information.)

- Flow velocities in the pipes and the control regime for pump operation on a frequency inverter. In today's systems, pumps are increasingly controlled via frequency inverters. This makes them more energy-efficient and enables continuous waste water treatment processes. The control ranges are generally independent of the design or engineering characteristics of the pump and drives; they always need to be determined individually with consideration given to the minimum flow velocity for handling solids and fibres. Experience in this field has shown that the flow velocity of waste water should not be below approx. 1.5 m/s in vertical discharge lines; a minimum of 0.8 m/s is usually sufficient for horizontal lines. Refer to Fig. 50 "Minimum flow velocities" in section "Piping and valves" for more detailed information on flow velocity. At low pump speeds it must be ensured that any substances in the fluid are transported out of the pump and discharge line. The circumferential speed at the outer diameter D2 of impellers used for waste water applications should not be below 15 m/s.
The start ramps of centrifugal pumps should generally be as short as possible to allow the pumps to quickly reach their highest possible speed. The speed is then reduced to meet the operating point.
The natural frequency range of the system (foundation / pump / piping) must be suppressed at the frequency inverter; the pump must not be operated continuously in this range. In parallel operation all pumps should be operated at the same frequency, if possible, to prevent individual pumps from being pushed towards impermissibly low flow by the others. The swing check valves must open completely at every operating point of the pump to minimise the surface which can be attacked by substances in the fluid and to prevent clogging.

- Approach flow conditions and structural design. Smooth pump operation at the agreed operating data requires hydraulically optimised approach flow conditions (refer to section "Structural Design" for further explanations).

**Pump operating mode**

The pump characteristic curve depends on many factors and can be influenced by a control system.
The pump characteristic curve can be altered by adjusting the impeller diameter or speed, or by changing to parallel or series operation. In this context, system conditions such as fluid levels in the inlet tank, inlet conditions and the NPSH available must also be taken into account. Fluid properties are also of major importance, for example composition, density, viscosity and temperature. These factors will be explained in detail in the following sections.

**Single-pump operation**

The system characteristic curve can be adjusted by altering the static head component. The geodetic head can be altered by means of the fluid level in the suction chamber for example, as illustrated in Fig. 16. The operating range of the pump on the H/Q curve lies between the two operating points "pump cut-in" and "pump cut-out".
For this reason, always plot two operating points on the pump characteristic curve. They mark the limits of the pump's actual operating range.
**Note:** Make sure the operating point is near the best efficiency point, i.e. within the optimum operating range.

**Throttling control or changes in the system characteristic curve**

The system characteristic curve can generally be changed in two ways. The flow resistance in the piping can be increased or decreased. This can be the intended result of adjusting a throttling element or using a different pumping route (pipes with different nominal diameters and lengths), or the unintended consequence of unplanned pipe deposits, corrosion or incrustation.
When closing the gate or throttle valves of a centrifugal pump, some of the pump input power which has been converted into head is intentionally wasted. From the point of view of conserving energy, this is the least favourable control option, and should also be avoided particularly in waste water applications (apart from start-up and shut-down processes) to prevent increased risk of clogging at or downstream of the valves (Fig. 17).
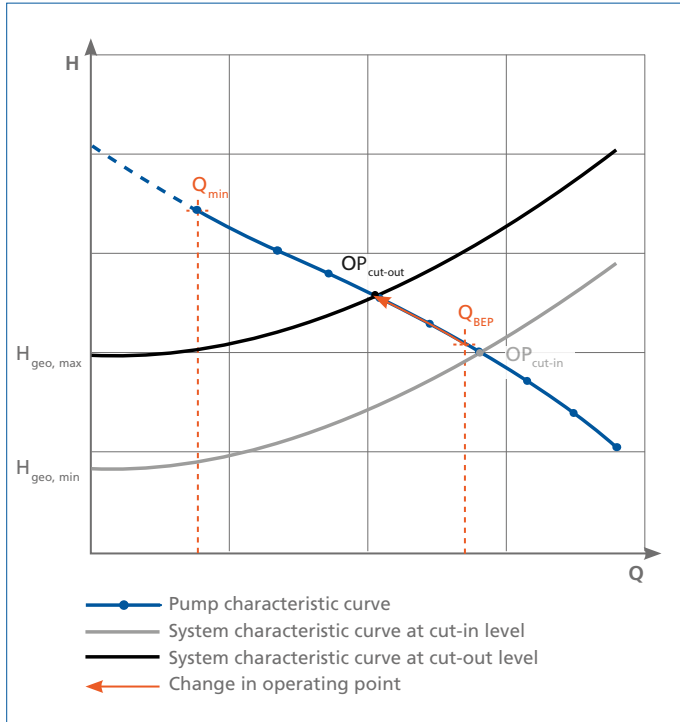
Fig. 16: Shifting the operating point of the pump at fixed speed by altering the static head between the suction-side cut-in and cut-out level
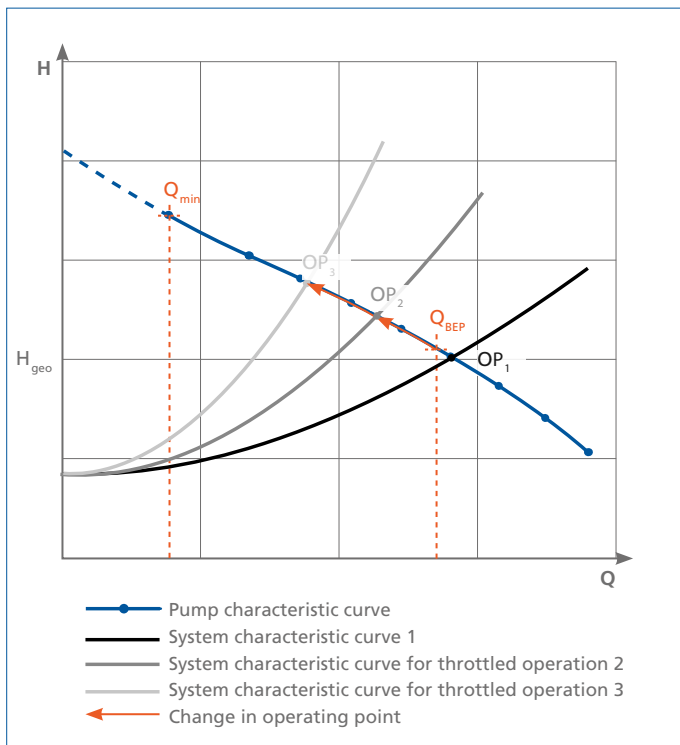


Fig. 17: Shifting the operating point of the pump by changing the pressure losses in the piping, e.g. the nominal diameter of pipes, the pumping route or length, or the deposits and incrustations in the piping

**Note:** Make sure the operating point is near the best efficiency point, i.e. within the optimum operating range. Observe the flow velocities in the pipes and the control regime. For the sake of clarity, Fig. 17 illustrates the system characteristic curve for the throttling control mode for only one geodetic head.

### Adjusting the impeller diameter

A relatively easy and hydraulically very efficient (but irreversible) measure to reduce both the flow rate and the head at an unchanged speed is to adjust the impeller diameter $D_2$ by turning down the impeller or cutting back the impeller vane tips.

As this measure modifies the vane length or the width and angle of the vane trailing edge, its effect on flow rate, head and efficiency depend on the impeller type.

**Note:** For the sake of clarity, Fig. 18 illustrates a system characteristic curve with only one geodetic head for the trimmed impeller.

Pump manufacturers indicate the possible diameter reduction range for their impellers in the form of a selection chart in the product literature / characteristic curve booklets. Within this range the correlation between impeller diameter, flow rate and head can be shown by using formula 12.

$$\frac{Q_1}{Q_2} = \frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \qquad \text{Formula (12)}$$

**Key:**
$Q_{1/2}$ = Flow rate [m³/h]
$H_{1/2}$ = Head [m]
$D_{1/2}$ = Impeller diameter (M)
   Indices: 1 initial value
            2 lower target value

The corresponding values for Q and H are positioned along an imaginary straight line which passes through the origin of the H/Q coordinate system. The new operating point of the pump set is the resulting intersection of the characteristic curve of the pump with reduced (trimmed) impeller diameter and the unchanged system characteristic curve.

**Note:** Make sure the operating point is near the best efficiency point, i.e. within the optimum operating range. Also, the circumferential speed should not be below approx. 15 m/s.

## Speed control

A variable speed centrifugal pump will always produce the flow rate / head which is actually required. Speed control is the most energy-efficient type of control. It also keeps the load on the pump components to a minimum. The correlation of speed, flow rate, head and input power is described by the special affinity laws for centrifugal pumps.

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} \text{ or } \frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \text{ or } \frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \qquad \text{Formula (13)}$$

**Key:**
$Q_{1/2}$ = Flow rate [m³/h]
$H_{1/2}$ = Head [m]
$n_{1/2}$ = Speed [rpm]
$P_{1/2}$ = Mechanical power absorbed [kW]
Indices: 1 initial value
2 lower target value

**Note:** For the sake of clarity, Fig. 19 illustrates the system characteristic curve for variable speed operation for only one geodetic head.

The corresponding values for Q and H are positioned along an imaginary parabola which passes through the origin of the H/Q coordinate system, illustrated by the dashed line in Fig. 19. The new operating point of the pump set is the resulting intersection of the characteristic curve of the pump with reduced speed and the unchanged system characteristic curve. For system characteristic curves with a small static component the new operating point remains near the best efficiency point. The following effect is proportional to the static component of the characteristic curve: Speed reduction will cause the pump to operate with low flow rates in a range of poor part-load efficiencies, and speed increase will cause the pump to operate with poor overload efficiencies.

**Note:** Make sure the operating point is near the best efficiency point, i.e. within the optimum operating range.
When using a control regime with frequency inverter operation, observe the flow velocities in the pipes. Also observe a minimum circumferential speed at the impeller outlet of approximately 15 m/s.
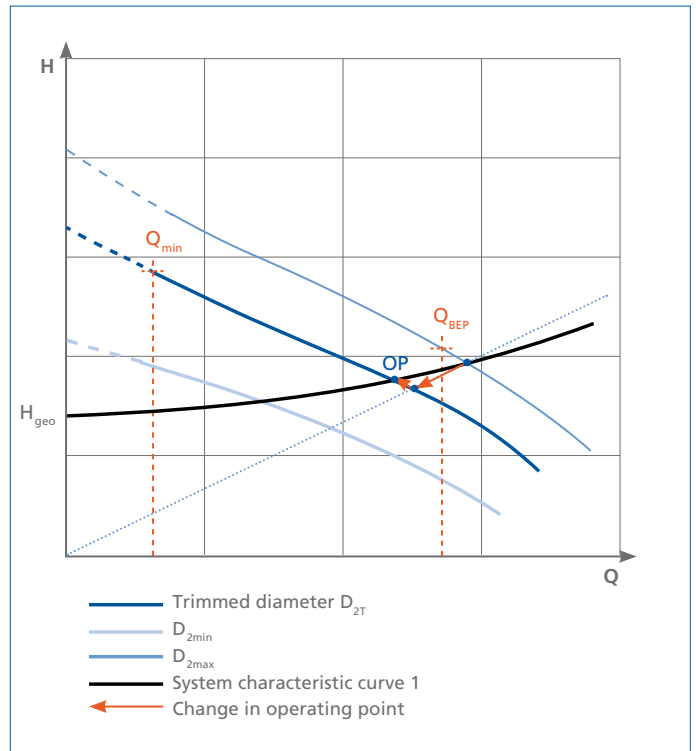


Fig. 18: Trimming or adjusting the impeller diameter to meet the required operating point of the pump
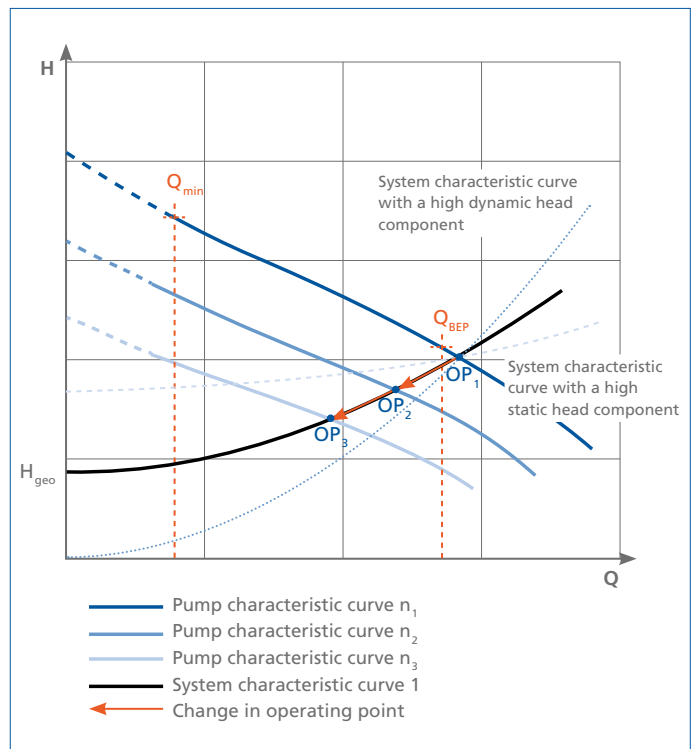


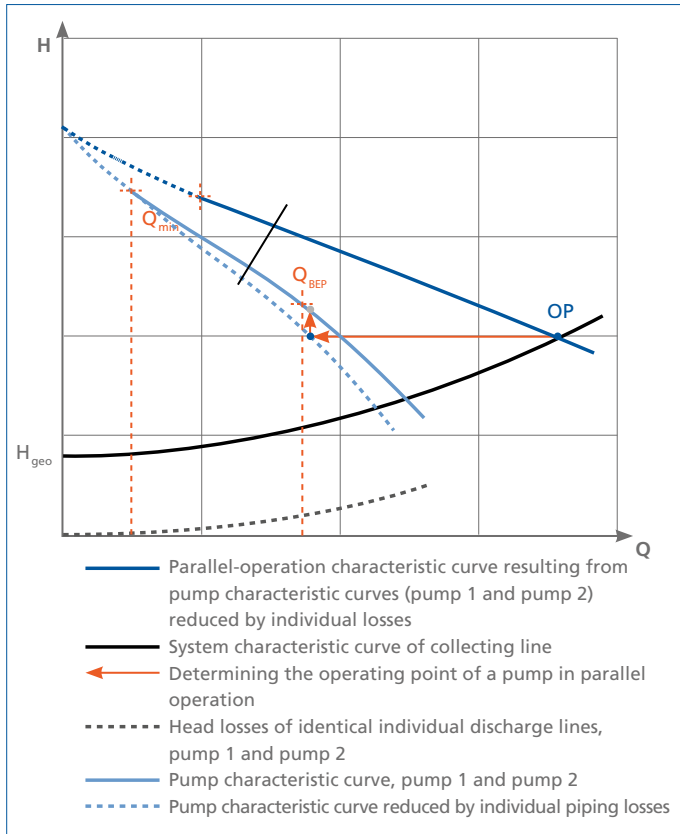Fig. 19: Shifting the operating point of a centrifugal pump by changing the speed

Fig. 20: Parallel operation of two identical centrifugal pumps. Losses of individual lines (head losses up to the collecting line) have been taken into account in the reduced pump characteristic curve.
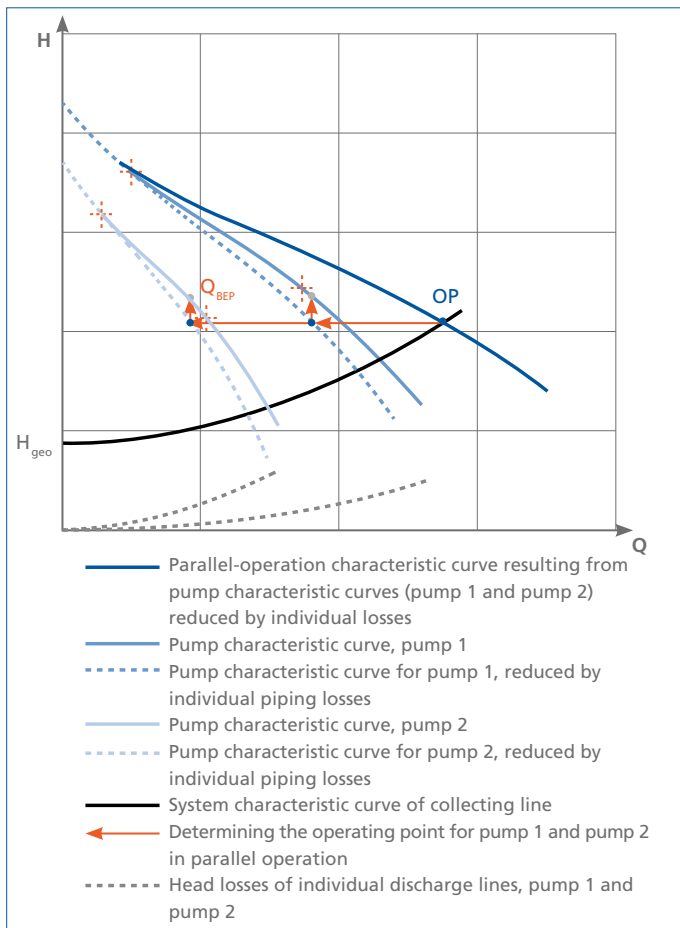


Fig. 21: Parallel operation of two different centrifugal pumps. Losses of individual lines (head losses up to the collecting line) have been taken into account in the reduced pump characteristic curve.

## Parallel operation of pumps of identical sizes

Parallel operation of two or more centrifugal pumps with a shared discharge/collecting line is particularly suitable for flat system characteristic curves. The lower the dynamic head component, which is quadratically proportional to the flow rate, the larger the attainable increase in flow rate. This correlation is illustrated in Fig. 20.

**Note:** For the sake of clarity, Fig. 20 illustrates a system characteristic curve with only one geodetic head for two pumps operated in parallel.

If the pump sets are identical, the total flow rate is composed of equal parts of the individual flow rates of each pump at a given head. The heads of each individual pump must be reduced by the dynamic component of head losses occurring in the individual lines up to the collecting line. The system characteristic curve of the collecting line only contains the head losses downstream of the point where the individual lines join.

**Note:** Make sure the operating point is near the best efficiency point, i.e. within the optimum operating range.
Observe the flow velocities in the pipes.
When selecting a pumping station for parallel operation of two or more identical pump sets, single-pump operation pumping into the collecting line must in certain circumstances be excluded. This is the case if no permissible operating point can be established, i.e. the single-pump characteristic curve does not intersect the characteristic curve of the system.

## Parallel operation of pumps of different sizes

In principle, the parallel operation of two or more centrifugal pumps of different sizes is analogous to that of pumps of identical sizes. The pump sets will work together smoothly if they have stable H/Q characteristic curves (the shut-off head is higher than the head at $Q_{min}$). Preferably, the pumps should have identical shut-off heads.

**Note:** For the sake of clarity, Fig. 21 illustrates a system characteristic curve with only one geodetic head for two pumps operated in parallel.

The total flow rate is composed of the flow rates of the individual pumps at a given head. The heads of each individual pump must be reduced by the dynamic component of head losses occurring in the individual lines up to the collecting line. The system characteristic curve of the collecting line only contains the head losses downstream of the point where the individual lines join.

**Note:** Make sure the operating point is near the best efficiency point, i.e. within the optimum operating range. The pump set with the lower shut-off head can very quickly be pushed towards low flow rates when the total head changes. Observe the flow velocities in the individual pipes.

When selecting a pumping station for parallel operation of two or more different pump sets, single-pump operation pumping into the collecting line must in certain circumstances be excluded. This is the case if no permissible operating point can be established, i.e. the single-pump characteristic curve does not intersect the characteristic curve of the system.

### Series operation

When two identical centrifugal pumps are operated in series (series operation) with a shared collecting line, the total head is composed of the heads of the individual pumps at a given flow rate. This correlation is shown in Fig. 22. Series operation is not recommended for waste water transport.

**Note:** For the sake of clarity, Fig. 22 illustrates a system characteristic curve with only one geodetic head for two pumps in series operation.

### Assigning pumps to different operating ranges/conditions

Pumping stations which are integrated in larger networks, cover a variety of pumping routes at different times and are subject to major fluctuations often require a solution that goes beyond varying the operating modes and control options of one or several identical centrifugal pumps.

Here, assigning different pump sizes with or without different impeller types to different operating ranges/conditions can be a useful approach.

This means the pumping tasks can be divided and the pumps can be assigned accordingly, e.g. to applications with markedly different flow rates (rainfall, day time and night time operation).
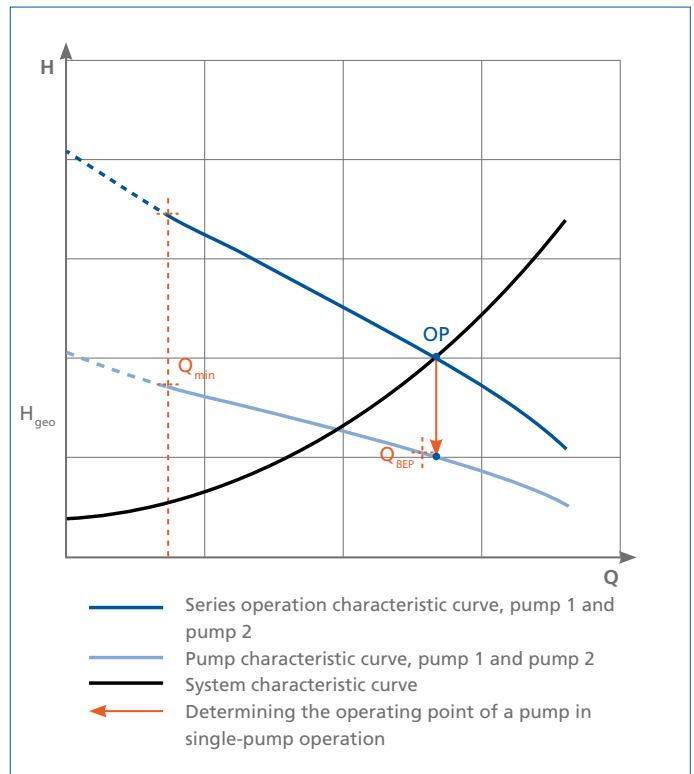


Series operation characteristic curve, pump 1 and pump 2
Pump characteristic curve, pump 1 and pump 2
System characteristic curve
Determining the operating point of a pump in single-pump operation

Fig. 22: Series operation of two identical centrifugal pumps

# Configuration and Installation

## Selecting the optimum impeller geometry

The number of different impeller types for centrifugal pumps in waste water transport exceeds that of any other application, see Fig. 23. Each of these impeller types serves a specific purpose.
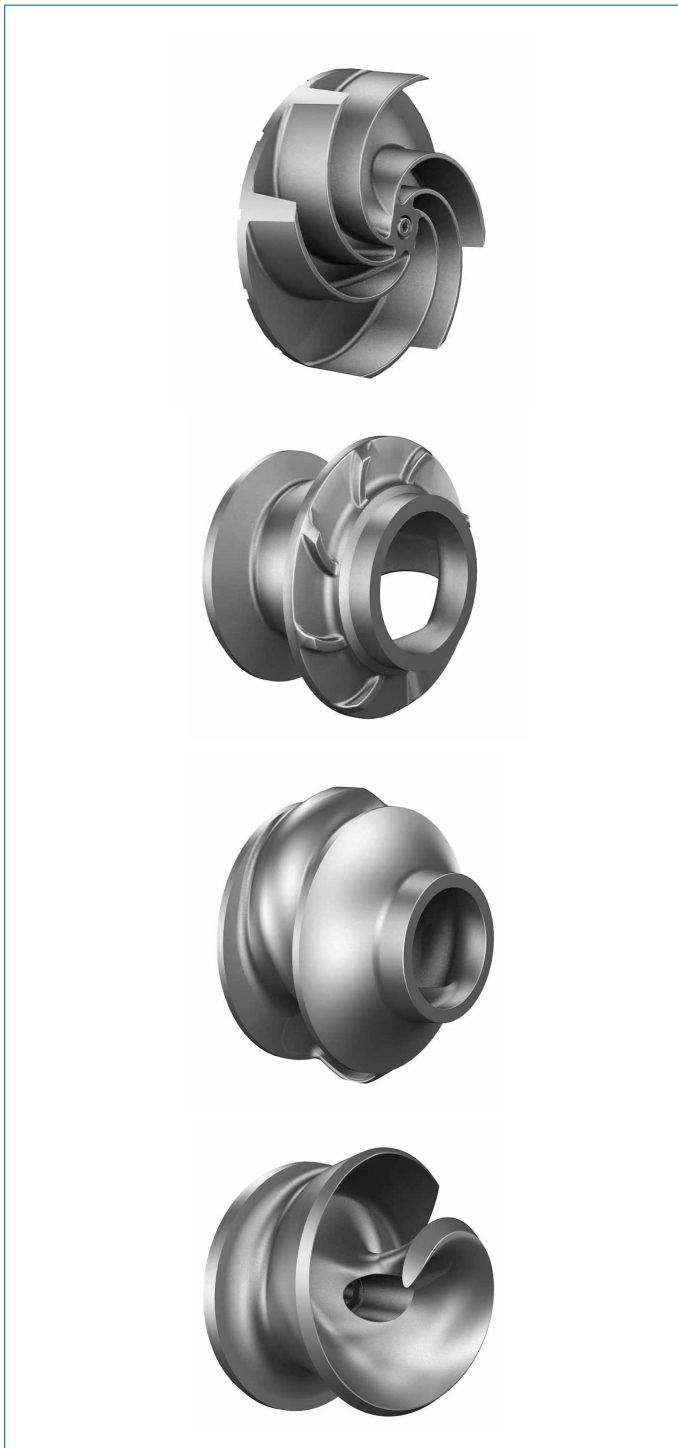


**Free-flow impeller (impeller type F)**
Suitable for the following fluids:
Fluids containing solids and stringy material as well as fluids with entrapped air or gas,
e.g. collecting tanks in pumped drainage applications, sand trap in waste water treatment plants

**Closed single-channel impeller (impeller type E)**
Suitable for the following fluids:
fluids containing solids and stringy material,
e.g. intake structures of waste water treatment plants

**Closed multi-channel impeller (impeller type K)**
Suitable for the following fluids:
contaminated, solids-laden, non-gaseous fluids without stringy material, e.g. pre-treated waste water in waste water treatment plants

**Open, mixed flow impeller (impeller type D)**
Suitable for the following fluids:
Waste water with solid substances and long fibres,
e.g.: sludge holding tank

Fig. 23: Impeller types

The most important criterion in impeller selection is the operating reliability. Further criteria to be considered are the efficiency and the local regulations and/or recommendations for free impeller passages. These exist for instance in Germany (Worksheet ATV DVWK-A 134), in Austria and in the USA.

Key factors to be considered for reliable operation are fibre content such as wet wipes, solid particle size, dry solids content (DS), gas content and sand content. Table 6 shows the expected loads for fluids handled in waste water treatment.

For gas, sand and dry substance content the limits of each impeller type can be defined reasonably accurately; for fibres and other solids in the fluid, quantities are harder to define. The composition of waste water can also change over time. The operator's experience should be considered when selecting an impeller. The application limits of each impeller type specified in Table 7 provides an orientation for decision-making. When fluids with a high gas content and a relatively high fibre content are handled, free flow impellers (F impellers) are preferably used for low volume flow rates and open channel impellers (D impellers) for higher volume flow rates.

To reduce the disadvantage of lower efficiencies achieved with F impellers, and to further increase operating reliability, the impeller and particularly the vane contour of KSB's new $F_{max}$ impellers have been optimised (see Fig. 24).

For waste water with low gas content but relatively high sand content, single channel impellers represent an important alternative to open channel impellers, as increased wear between the impeller and the suction-side contour of the pump casing must be expected with open channel impellers. The $E_{max}$ impeller has been developed in order to avoid hardened deposits and the jamming of fibres in the front and rear clearances between the impeller and the pump casing (see Fig. 25).
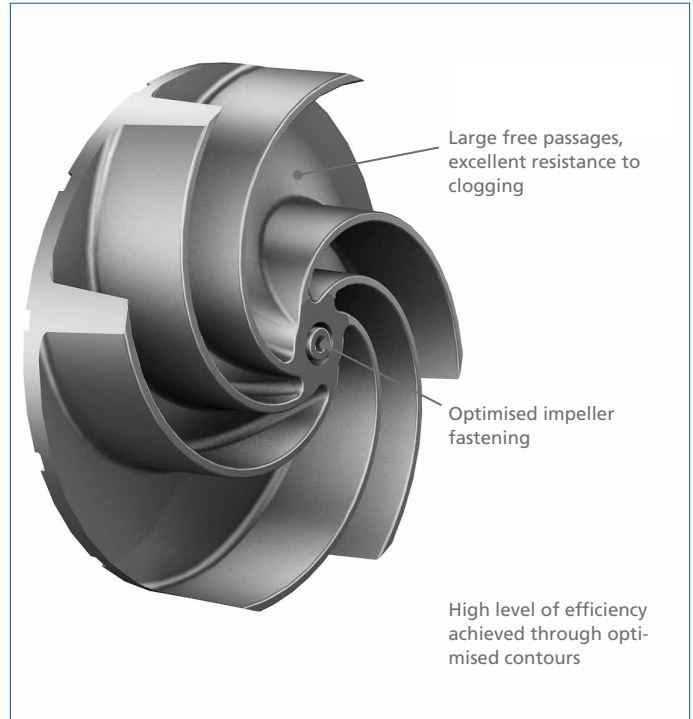


Large free passages, excellent resistance to clogging

Optimised impeller fastening

High level of efficiency achieved through optimised contours
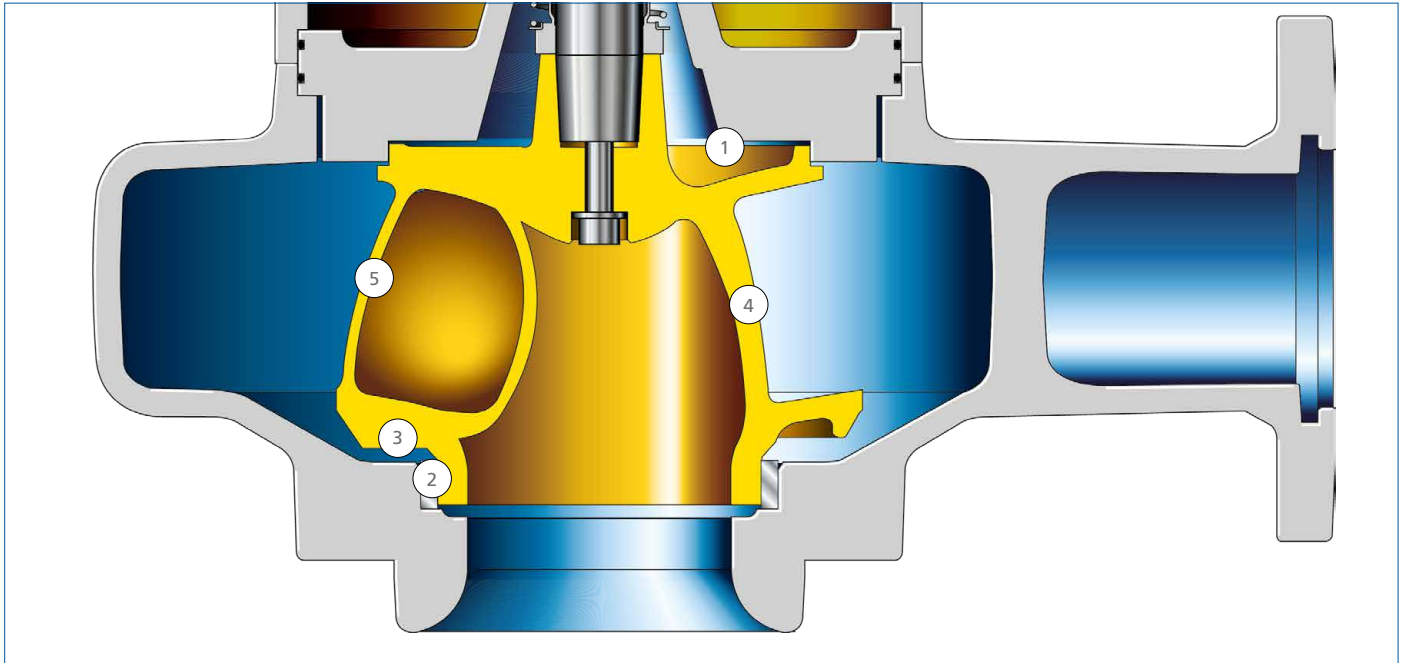
Fig. 24: $F_{max}$ impeller

Fig. 25: Sectional drawing of a pump with $E_{max}$ impeller

**Key:**
1. **Radial sealing of rear clearance between impeller and casing**
2. **Radial casing wear ring on suction side (hydrodynamic bearing)**
3. **Auxiliary vanes on impeller suction side**
4. **Vane leading edge optimised for a better fibre-repellent profile**
5. **Axial vane trailing edge for lower pressure impulses**



Long vanes

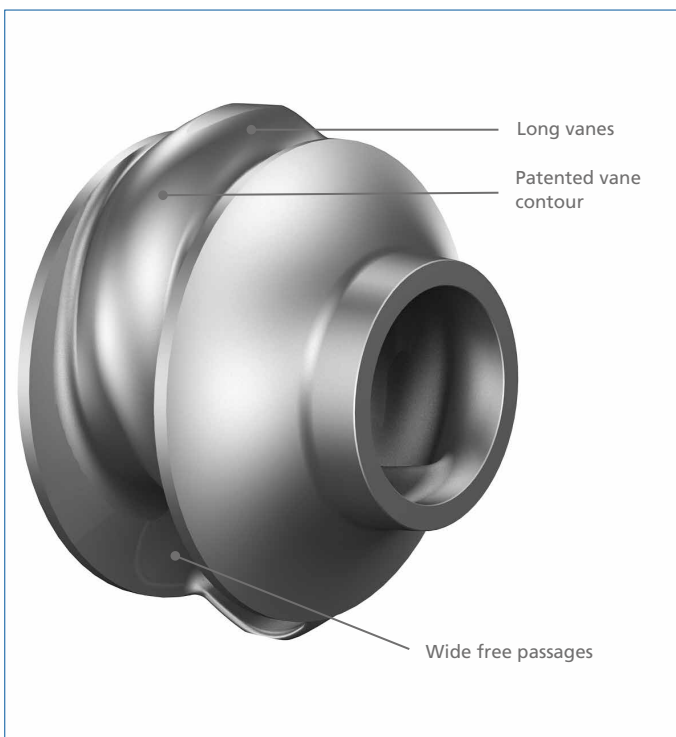Patented vane contour

Wide free passages

Fig. 26: $K_{max}$ impeller

Additional vanes on the suction-side impeller shroud (auxiliary vanes) help reduce leakage flow through the gap at the casing wear ring and thus prevent solids from entering this clearance. Despite the presence of back vanes on the outer diameter, the motor-end clearance between impeller and casing is covered to prevent fibres from entering.

Pre-screened waste water can be expected to involve lower solids loads. Closed channel impellers are best suited for this purpose thanks to better efficiencies.
These impellers have also been optimised as can be seen in the $K_{max}$ impellers (see Fig. 26).

When evaluating efficiencies, it is not sufficient to compare the pumps' best efficiencies. What matters is the efficiency level at the actual operating points. The expected runtime of a pump must be taken into account when considering energy efficiency.

In some applications, the pumps must operate for more than 4,000 hours per year, while the operating time in domestic pumped drainage or stormwater overflow tanks is often well below 100 hours per year.

In addition to operating reliability, the selection of pumps is governed by criteria such as efficiency for continuous-duty pumps and price for pumped drainage applications.

| | Gas content [vol%] | Fibre content | Size of solids | Dry solids content [%] | Sand content* [g/l] |
|---|---|---|---|---|---|
| Stormwater and surface water | – | low | small | – | 0 – 3 |
| Waste water | | | | | |
| – Municipal waste water | | | | | |
| – Domestic waste water | 0 – 2 | medium | medium | – | 0 – 3 |
| – Commercial waste water | 0 – 2 | large | large | – | 0 – 3 |
| – Industrial waste water | 0 – 2 | large | large | 0 – 5 | 0 – 3 |
| Water containing a large proportion of sand | – | – | – | – | 8 – 10 |
| Sludges | | | | | |
| – Activated sludges | 2 – 4 | low | small | 1 – 2 | – |
| – Primary sludge | 2 – 4 | low | small | 2 – 6 | – |
| – Secondary sludge | 2 – 4 | low | small | 2 – 3.5 | – |
| – Thickened sludge | 3 – 6 | low | small | 2 – 5 | 0 – 2 |
| – Stabilised sludge | – | low | small | 5 – 10 | – |
| – Dewatered sludge | – | low | small | 20 – 30 | – |
| – Dried sludge | – | low | small | 30 – 50 | – |

Table 6: Anticipated values of different (waste) water and sludge types

| | Gas content [vol%] | Fibre content | Size of solids | Dry solids content [%] | Sand content [g/l] |
|---|---|---|---|---|---|
| Free-flow impeller (F impeller) | ≤8 | high | large | <8 | ≤10 |
| Closed single-channel impeller (E impeller) | ≤2 | medium | large | ≤6 | ≤6 |
| Open single-channel impeller (D impeller) | ≤4 | high | large | ≤13 | ≤4 |
| Closed two-channel impeller (K impeller) | – | low | medium | ≤3 | ≤4 |
| Open two-channel impeller | 4 | high | medium | 6 | ≤6 |
| Multi-channel impeller (K impeller) | – | low | small | ≤5 | ≤6 |

Table 7: Application limits of different impeller types                * Provided the impeller is made of a suitable material

## Matching materials to the application

Grey cast iron, especially EN-GJL-250, has proven very successful for all main components in waste water applications. This material has several benefits:

Its excellent chemical resistance in neutral and alkaline environments matches its resistance against hydro-abrasive wear. Noteworthy is that the properties of the casting skin which contains large amounts of carbon, iron oxides and silicon oxide provide effective protection against wear and corrosion. Modern designs keep machining to a minimum in order to preserve the casting skin as much as possible.

A further advantage of grey cast iron is its mouldability which allows the implementation of uncompromisingly energy-efficient design solutions. The material's low modulus of elasticity dampens vibrations at the casing parts, bearing bracket, and the duckfoot



Fig. 27: Amarex KRT in material variant 1.4517

bend. At a reasonably low price per kilogram, manufacturers can offer a stable and reliable design.

As a material for the motor housing, grey cast iron has very good thermal conductivity compared to stainless steel, for example. Grey cast iron is the first choice in more than 90 % of applications.

### Pumps for highly abrasive fluids

In areas where large amounts of sand are present in the stormwater and transported to the waste water system, the impeller and even the pump casing might have to be made of a material which is particularly resistant to abrasion. We recommend high-alloy grey cast iron (EN-GJL-HB555). A comparison in a model wear test showed that the linear material loss rate of EN-GJL-250 is 20 times higher than that of EN-GJL-HB555.

Coatings may also prove a viable option for casing parts. Coatings should be selected according to their ability to handle the type and concentration of solids in the fluid. The coating method must be compatible with the component to be coated. Coatings are not recommended for impellers as the coating life would be too short.

### Pumps for corrosive fluids

Duplex stainless steels such as 1.4517 and 1.4593 have proven themselves many times over in applications with corrosive fluids (see Fig. 27). Pure austenitic steel such as 1.4408 is not as wear-resistant and is unsuitable for seawater applications.

Duplex stainless steels with an austenite/ferrite ratio of 1:1 not only offer excellent corrosion resistance, but also outstanding mechanical properties, such as strength and elongation at fracture, as well as adequate wear resistance.
If a very long service life is required, pumps for seawater applications need to be made of a material with a so-called PREN value of 35 or even 38.

The PREN number (Pitting Resistance Equivalent Number) is calculated using the following formula:
PREN = %Cr + 3.3 x % Mo + 16 x % N
PREN is the pitting resistance equivalent number for the elements Cr (chromium), Mo (molybdenum), N (nitrogen). Ferritic-austenitic duplex steels with PREN values > 40 are also referred to as superduplex steels.

The effect of coatings is limited in combination with corrosive fluids such as seawater. Even the slightest damage to the coating will lead to uninhibited spreading of the corrosion process underneath the coating.

Special coatings combined with cathodic protection by anodes have resulted in remarkable resistance. Based on this solution, several pumps with JL1040 casings have been in operation in pumping stations at the Red Sea for more than five years without displaying any visible signs of corrosion. Prerequisites for this solution are proper maintenance and regular anode replacement, every one to two years.
Electric conductivity between the pump and other components with large surfaces such as pipes or grids must be prevented during installation. If this is not ensured, the anode is used up early due to an unfavourable anode/cathode ratio, impairing pump protection.

**Casing wear rings (K and E impellers) / wear plate (D impellers)**
The clearance between the casing wear ring and the impeller with or without an impeller wear ring determines the flow rate of the leakage volume. The flow rate of the leakage volume significantly influences the pump efficiency. If the clearance remains constant, pump efficiency generally remains constant too. If wear causes the clearance to widen, leakage reduces the effective volume flow rate of the pump, thus reducing pump efficiency.

A larger clearance also increases the risk of hardened deposits as described in the section "Selecting the optimum impeller geometry".

As K impellers are normally only used for fluids with a low solids content, EN-GJL-250 is usually an adequate casing ring material. Materials of higher wear resistance are required for more demanding fluids. As an option, KSB offers wear rings made of semi-austenitic cast CrNi steel (VG 434) for K impellers. Material 1.4464 has comparable properties. To ensure higher wear resistance, wear plates used in conjunction with D impellers are offered in EN-GJL-HB555 as an alternative to EN-GJL-250.

**Shaft**
KSB shafts are made of ferritic chrome steel (1.4021) as standard. This material offers good mechanical strength and corrosion resistance in more than 95 % of all applications. Pumps supplied in chrome steel are fitted with shafts made of 1.4462.

## Shaft seal

Sealing the shafts of submersible motor pumps with two mechanical seals in tandem arrangement (Fig. 29) and an internal liquid reservoir has been the established solution for many decades.

As this system does not require any external mechanical seal supply system or other auxiliary systems, its susceptibility to malfunctions is extremely low.

Mechanical seal sizes up to 95 mm are designed as a rubber bellows-type seal with a single spring as standard (see Fig. 28). This bellows-type seal has many advantages. The bellows completely compensate for shaft deflection caused by radial forces in the pump without any relative motion on the shaft. The shaft is thus protected against wear. As shafts for Amarex KRT are made of corrosion-resistant steel and shaft wear does not occur, no shaft protecting sleeve is required for this mechanical seal type.
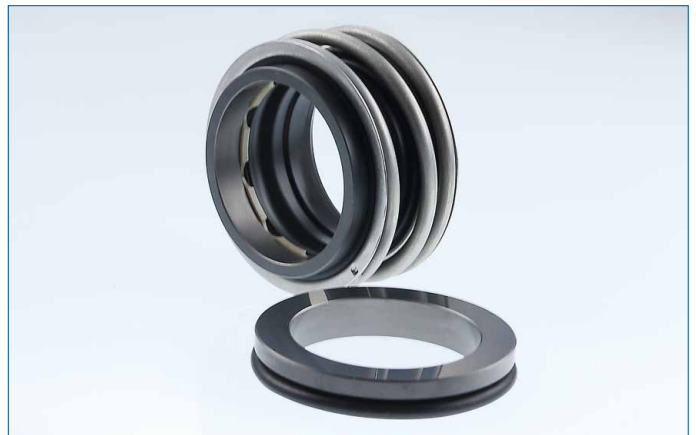


Fig. 28: Rubber bellows-type seal with a single spring

The rubber bellows and the mating ring's O-ring are made of oil-resistant NBR. If a higher resistance is required, FPM is available as an optional material.

Fig. 29: HJ seal type



Fig. 30: KSB mechanical seal type 4STQ

**Key:**
1. **Primary ring**
2. **Mating ring**
   **two stationary seal components -> no shaft wear**
3. **O-ring**
4. **Spiral spring**
   **internal -> no contact with the fluid handled, short spring travel**
5. **Shaft sleeve**



Leakage chamber                    Impeller with tapered fit

Fig. 31: Leakage chamber and impeller with tapered fit

Two stationary mechanical seals in tandem arrangement are used for larger shaft diameters. The stationary mechanical seal design features a primary ring which is fitted in the casing part and pressed against the mating ring by springs. The mating ring is firmly seated on the shaft, ruling out any relative movement between the mechanical seal and the shaft section and eliminating shaft wear. This seal is highly robust and resistant even in extreme applications.

Both the rubber bellows-type seal and the stationary seal are balanced. The contact pressure between the primary ring and the mating ring is only determined by the spring force. It is not influenced by the pressure in the seal chamber.

The inboard seal face combination is SiC/SiC for both mechanical seals. This seal face combination is resistant to chemical and abrasive substances, but offers only limited anti-seizure and dry-running properties. The risk of dry running arises with the presence of a higher proportion of methane in the fluid and with suction lift operation.

In order to prevent dry running and minimise wear on the seal faces, the space between the two mechanical seals arranged in tandem is filled with a quench liquid at the factory. A non-toxic biodegradable paraffin oil is the standard quench liquid used for this purpose. Up to 10-15 % of the quench space remains filled with air to allow the quench liquid to expand when heated. Even if the space behind the inlet bore appears to be completely filled, the design ensures air volume is present. The quench liquid has the task of forming a lubricating film between the seal faces of the mating ring and primary ring of both mechanical seals. This lubrication is a prerequisite for ensuring low wear of the seal faces.
The mechanical seals are cooled by the fluid handled.

A KSB mechanical seal is available for smaller sizes (up to impeller diameter 315) to meet higher corrosion and abrasion requirements. This 4STQ seal (see Fig. 30) consists of a cartridge encompassing two stationary mechanical seals which are also mounted in tandem. This version is extremely robust, and its cartridge design allows the seal to be replaced without removing the discharge cover.

For pumps with rated powers above ~30kW, the design provides for a leakage chamber arranged behind the second mechanical seal (see Fig. 31) in which an optional leakage sensor can be installed.
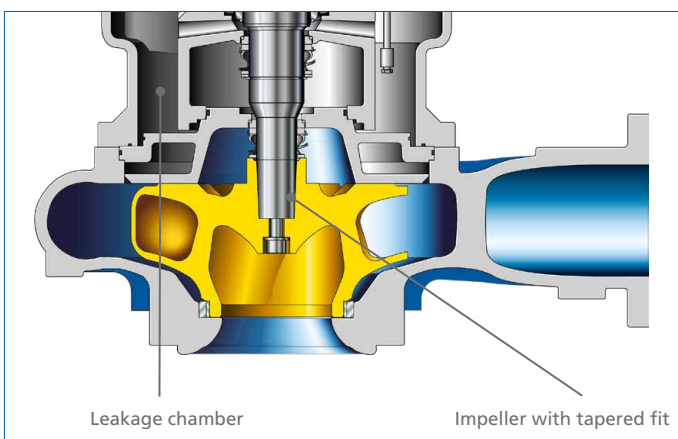
# Rotor and bearings

## Shaft load

It is essential that the shaft and bearings are correctly dimensioned for loads anticipated in the permissible operating range.

Operation under off-design conditions may lead to damage of the shaft and the mechanical seals, and in particular to the bearings. The various reasons for the importance of determining operating limits are explained in detail in the section "General Pump Selection".

Operating the pump against a closed gate valve is an unfortunately common mistake. Apart from high bearing forces, this will also lead to severe shaft deflection as a result of forces acting upon the impeller. Shaft deflection can in turn lead to the impeller rubbing against the casing wear ring. Prolonged operation under these conditions will widen the clearance between the impeller and the casing wear ring.

## Impeller seat

At KSB, impellers with a tapered fit have proven their worth for many years. KSB uses this impeller/shaft connection up to approx. 110 kW (see Fig. 31). The benefits of this impeller seat are as follows:

a) The values achieved during the balancing procedure can be exactly reproduced when the shaft is installed in the pump, as there is no clearance between the impeller and the shaft.
b) Straightforward assembly and dismantling of impeller.
c) This impeller fastening type will not become loose during reverse operation if a swing check valve has not been installed.

For higher power ratings, the torque is still transmitted via a cylindrical fit with two keys.

## Bearings

The Amarex KRT pump is generally fitted with rolling element bearings. As the shaft expands due to the heat generated in the motor, the radial bearing type used here is always a cylindrical roller bearing.

The fixed bearing for drive ratings up to ~65 kW is a deep groove ball bearing. Two angular contact ball bearings are employed for higher ratings (see Fig. 32).

For Amarex KRT with a drive rating of < 65 kW, the bearings are dimensioned in accordance with the "Technical specifications for centrifugal pumps" ISO 5199, with a calculated minimum bearing life of 17,500 hours. This mostly results in a significantly higher bearing life for the actual hydraulic systems and operating points. For pumps with a drive rating of > 65 kW, the assumed minimum calculated bearing life is 25,000 hours.

The bearings are generally grease-lubricated. For pumps with a low and medium power range (< 65 kW) the bearings are greased for life. For pumps with higher ratings (> 65 kW), re-lubrication is frequently required. These pumps come with a re-lubrication system. Refer to the pump operating manual for information on the required grease quality, grease quantity and the maintenance intervals.
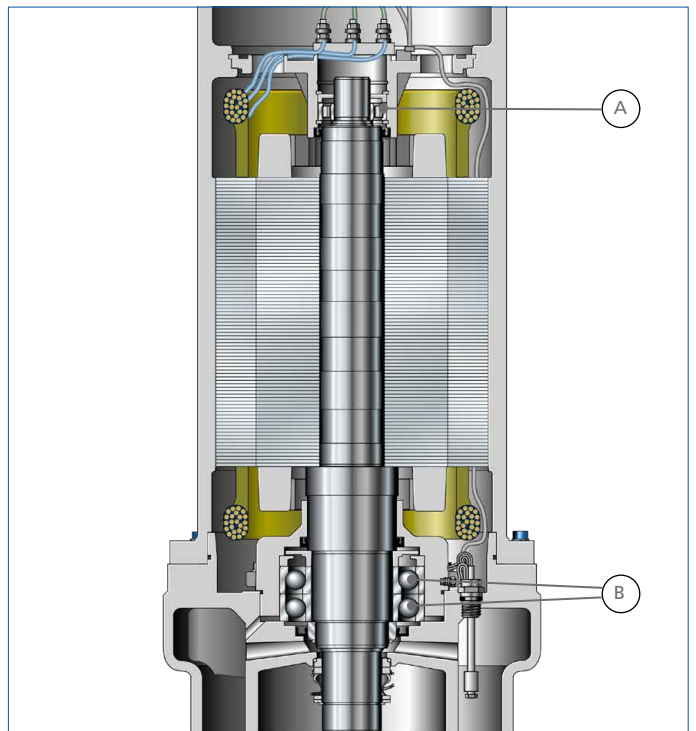


Fig. 32: Bearings for Amarex KRT with ratings > 65 kW

Key:
A) Cylindrical roller bearing
B) Angular contact ball bearing

Fig. 33: Transportable waste water pump

Wet installation with guide wire

Wet installation with guide rail(s)

Fig. 34: Waste water pump in stationary wet installation



Fig. 35: Waste water pump in horizontal and vertical dry installation

## Installation

Transportable pump models must be securely positioned in the pump sump, and suitable lifting equipment is required to install and lift the pump (see Fig. 33).

Like Sewabloc, the Amarex KRT pump in horizontal dry installation (see Fig. 35) is installed on foundation rails. Two variants are available for vertical dry installation. Pumps with an impeller diameter of 315 mm are installed on a duckfoot bend. Larger pumps are installed on two concrete bases (see Fig. 35). In order for the pump feet to stand on a metal support, foundation rails are cast into the concrete bases beforehand.

Stationary wet installation (see Fig. 34) is effected via a duckfoot bend fastened to the tank floor. The duckfoot bend has to be matched to the permissible load of the pump and the mating dimension of the claw fastened to the discharge nozzle of the pump.

The mating dimensions are not standardised. For heavy pumps, duckfoot bends might need to be fastened to the concrete floor with foundation rails (see Fig. 37). The duckfoot bend or foundation rails are fastened with chemical anchors (composite anchor bolts) approved for structural engineering. The chemical anchors need to be set in a concrete floor of sufficient strength (minimum C25/C30 to DIN 1045).

Also ensure that the pump's inlet section is positioned at a sufficient height above the tank floor.
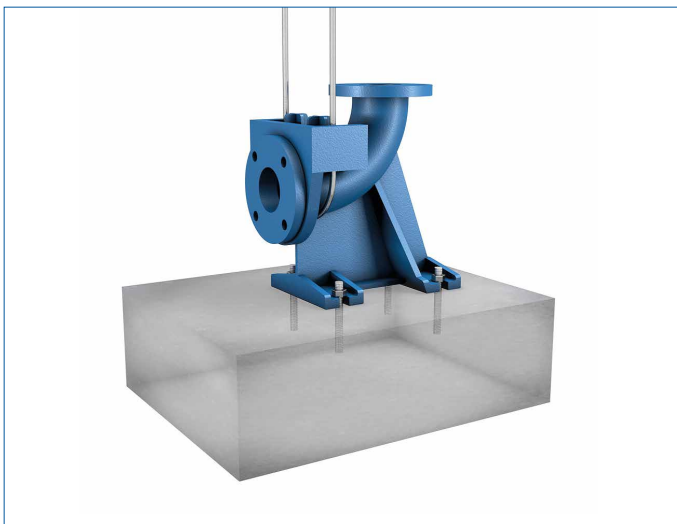


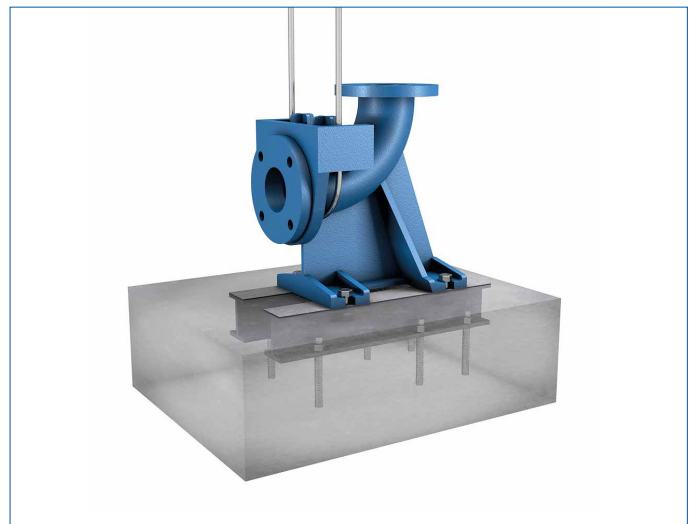Fig. 36: Guide wire arrangement | installation directly on the tank floor



Fig. 37: Guide wire arrangement | installation on foundation rails

Prior to installing the pump, the complete discharge line and the guide arrangement for the pump must be fitted. Refer to the respective section for information on the piping.

Two different guide arrangements are available: the guide wire (see Figs. 38 and 39) and guide rail(s) (see Fig. 40). A comparison (Table 8) shows that the guide wire arrangement has many advantages. KSB therefore offers the guide wire arrangement as a standard solution, with the guide rail option also being available for all sizes.

The upper brackets of the guide arrangement are suitably fastened to a concrete wall or ceiling with chemical anchors (see Fig. 38).

Spacers need to be fitted on guide wire arrangements for large installation depths. For the guide rail arrangement, the installation of intermediate supports every 6 m is recommended. The intermediate support brackets should be either clamped to the discharge line or mounted directly to the wall of the pump sump.
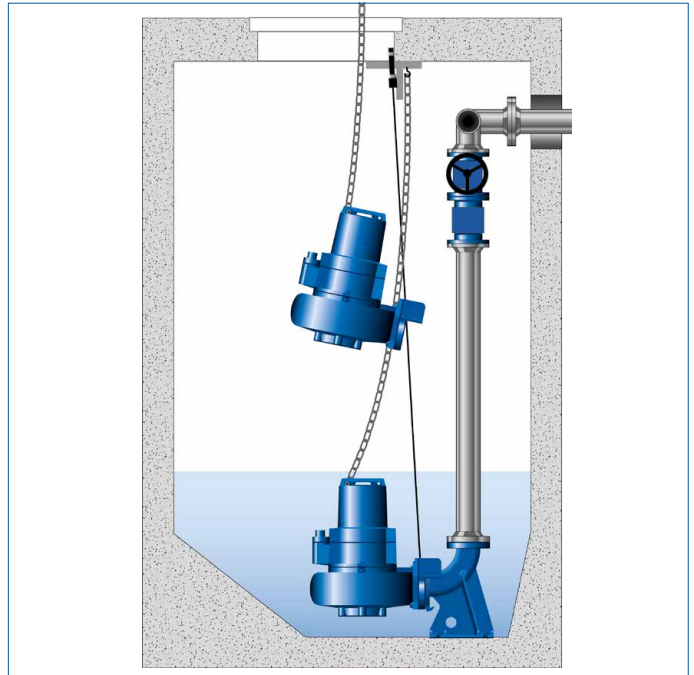


Fig. 38: Guide wire arrangement



Fig. 39: Guide wire arrangement



Fig. 40: Guide rail arrangement

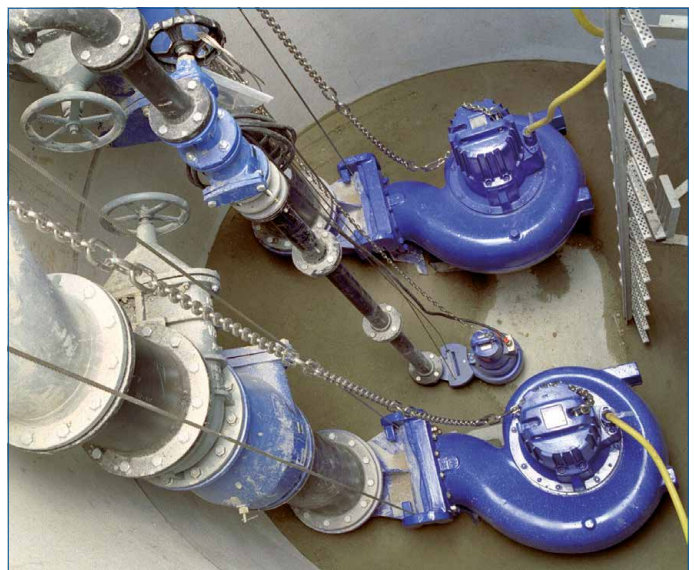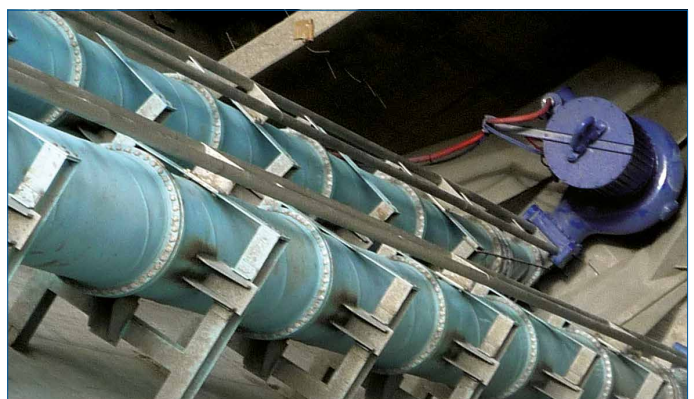| Guide wire arrangement | Guide rail arrangement |
|---|---|
| Easy and convenient to transport | Rails may be difficult to transport |
| Fast to install, saving costs | Inflexible regarding structural deviations |
| Compensation of installation tolerances without additional effort | Very costly for large installation depths |
| Allows installation with inclined angle of up to + 5° | Corrosion problems of guide rails |
| Flexible and reliable adaptation to different installation depths down to 85 m | Corrosion-resistant rails are very costly |
| Guide wire made of corrosion-resistant stainless steel 1.4401 (316) | Problems with sealing (metal-to-metal contact or shearing of rubber gasket) at the duckfoot bend |
| Guide wire included in KSB's scope of supply | Access to valves obstructed by inflexible rails |
| Problems with contamination in cases where powerful flow patterns or floating sludge are present in the pump sump. | Less sensitive to fluids containing fibres |

Table 8: Comparison of guide wire arrangement and guide rail arrangement
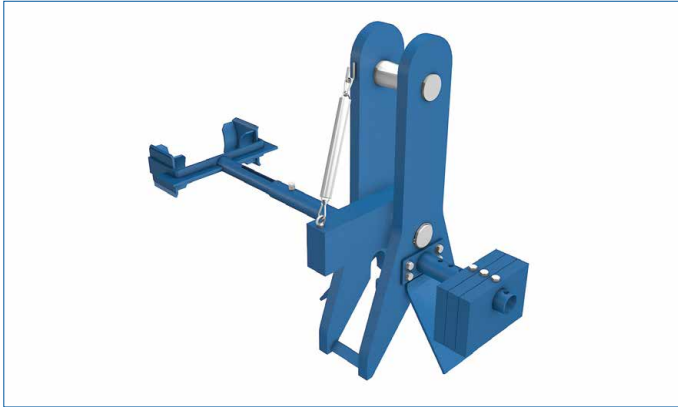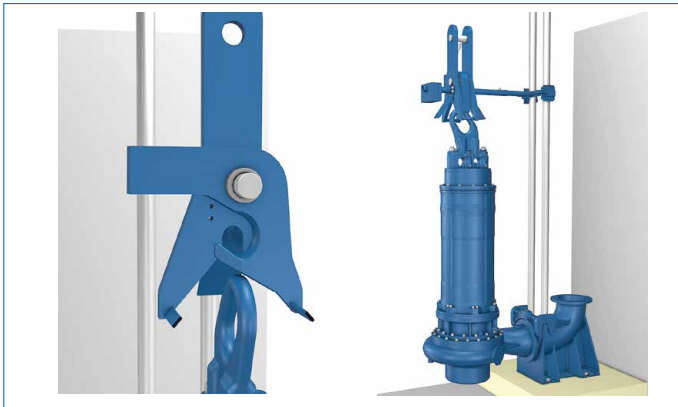
Fig. 41: Pump hooking device components



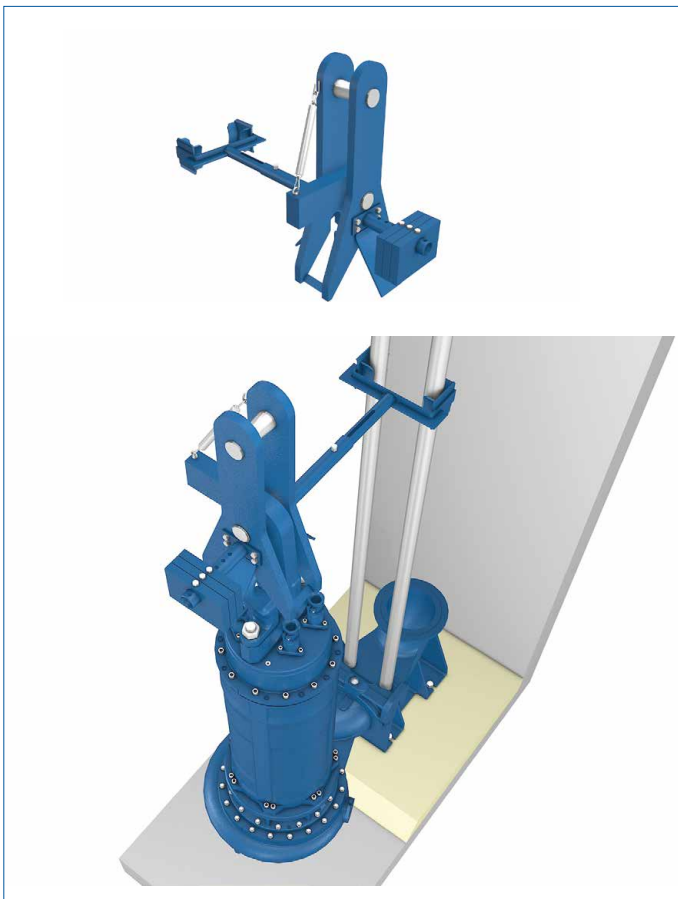Fig. 42: Hooking device components for lowering the pump



Fig. 43: Hooking device components for lifting the pump

For large installation depths, a hooking device can be used instead of a heavy lifting chain for pump lifting and lowering. This device consists of a guiding rod, a counterweight and a hook weight, the actual lifting hook and the mounting plates (see Fig. 41).

The following components are required for lowering and lifting the pump:

**Lowering:** Device with the hook weight (without guiding rod and without counterweight)

**Lifting:** Device with guiding rod and a counterweight (without a hook weight)

For lowering, the hooking device's hook is attached to the bail on the pump and suspended from a crane hook. The pump is then lowered onto the duckfoot bend along the guide rail arrangement. The hook weight opens the hook as soon as the pump is "unloaded". The hooking device can then be pulled up again without the pump via the crane (see Fig. 42).

For lifting the pump, the complete hooking device is required, but without the hook weight. The device is suspended from the crane hook and is lowered down the guide rails via its guiding rod. When the device is above the pump's bail, the hook is pushed open by the weight of the hooking device and automatically hooks to the bail. The device including pump can now be pulled up. (see Fig. 43).

On vertical dry-installed pumps, the inflow is usually via an intake elbow. Amarex KRT pumps with an impeller diameter of ≥ 400 mm and a nominal discharge nozzle diameter of ≥ 150 mm are fitted with an accelerating elbow as standard (see Fig. 35). The accelerating elbow ensures a uniform velocity distribution at the suction-side pump inlet.

**Inspection cover**
Pump casings with nominal discharge nozzle diameters from DN 100 in dry installation are fitted with inspection covers. An inspection cover is generally provided on intake elbows or accelerating elbows in dry installation.

# General Description of the Motor

KRT motors are water-tight, three-phase asynchronous squirrel-cage motors, especially designed for driving submersible motor pumps (Fig. 44 and Fig. 45). The motor is available as a non-explosion-proof or as an explosion-proof model with "flameproof enclosure" type of protection.

Pump and motor have a common pump / motor shaft and form an inseparable unit. No defined electrotechnical standards exist for the special pump motors described below; however, the requirements of DIN EN 60034 are complied with as far as possible.

KSB submersible motor pumps are floodable, non-self-priming close-coupled pump sets, which are normally operated completely submerged. They can be operated non-submerged for a short period of time, during which the permissible motor temperature is monitored by a temperature sensor installed in the motor. The minimum fluid level indicated in the operating manual generally has to be observed.

For applications requiring permanently or partly not submerged operation, a motor design with cooling jacket is available ensuring sufficient motor cooling regardless of the fluid level in the pump sump.

## Motor sizes

Standard rated power (depending on the number of poles):
4 to 480 kW also up to 880 kW

Standard number of poles (depending on the rated power):
2 to 10 poles

Standard rated voltages:
400/690 V, 50 Hz and 460 V, 60 Hz

**Note:**
1. For 60 Hz, rated voltages are available in the range from 200 to 575 V.
2. Higher rated powers and different rated voltages (also high voltage) on request.
3. For the individual motor size refer to the motor data sheets (included with the project documentation or available on request).
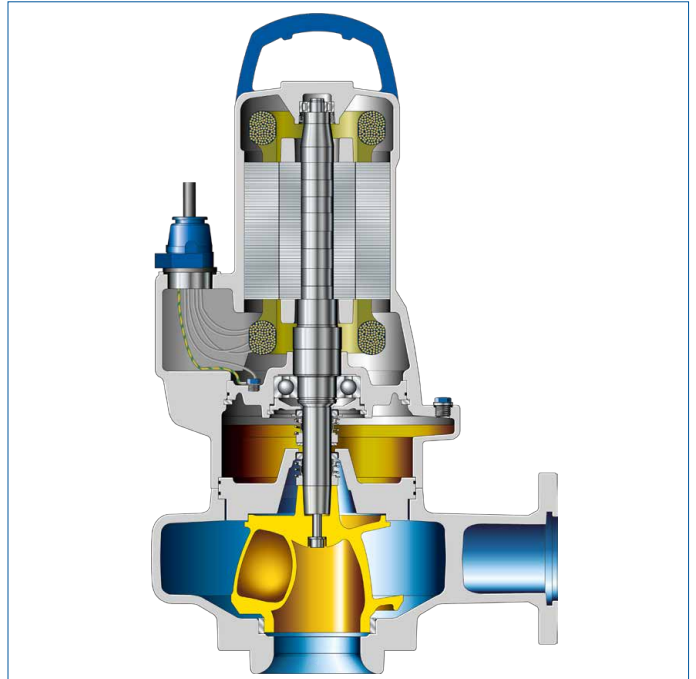


Fig. 44: Sectional drawing of a KRT 4 to 60 kW, without cooling jacket



Fig. 45: Sectional drawing of a KRT > 60 kW, with cooling jacket

## Type of construction

The type of construction of KRT motors is similar to that of
IM V10 (to DIN EN 60034-7).

### Motor installation

The motor and the pump are installed directly in the pump
sump as a close-coupled unit. During operation the motor is
submerged in the fluid to be handled. It can be operated outside
the fluid for a short period of time during which the permissible
motor temperature is monitored by a temperature sensor installed
in the motor. The minimum fluid level indicated in the operating
manual generally has to be observed.

For large submersible motor pumps, the minimum fluid level is
usually determined by hydraulic parameters (e.g. air-entraining
vortices, NPSH).

Fluid temperature: max. 40 °C as standard.

For operation with fluid temperatures > 40 °C and < 60 °C, a
hot water variant is available. Operation with fluid temperatures
> 60 °C on request.

### Sizes

The dimensions for standardised motors to IEC 72 are not
applicable for integrated submersible pump sets. Nevertheless,
the electrical sheet steel laminations correspond with the
so-called IEC dimensions or sizes common on the market.

### Immersion depth

KRT submersible motor pumps can be operated up to an immer-
sion depth of 30 m without any special measures being taken.

## Duty types

KRT motors without a cooling system (installation type S) are
designed for continuous duty S1 (to DIN EN 60034-1) in a sub-
merged condition. If the motor is operated non-submerged and
dry running is detected, a bimetal switch in the motor winding
switches the duty type to S3.

KRT motors with a cooling system (installation types K and D)
are designed for continuous duty S1 (to DIN EN 60034-1) at all
fluid levels.

## Enclosure

KRT motors comply with the requirements of enclosure IP68 to
DIN EN 60034-5, and the pump set complies with the
requirements of enclosure IP68 to DIN EN 60529.

## Type of protection and temperature classes

Explosion-proof Amarex KRT motors comply with the require-
ments of Ex II2G Ex db IIB T3 and, in some cases, T4 type of
protection to DIN EN 60079-0 / DIN EN 60079-1 and they are
suitable for use in zone 1. The following codes are used in the
motor designation: X → T3 and Y → T4.
Permissible surface temperatures: T3 -> 200°C and T4 -> 135°C

## Electrical design data

### Rating

To ensure a long motor life, a mains voltage tolerance of ± 5 %
and mains frequency tolerance of ± 2 % (Design A) in accordance
with DIN EN 60034-1 should be observed.

Other than that, the power output of Amarex KRT motors in a
submerged condition is only restricted by a mains voltage tolerance
of ± 10 % and a mains frequency tolerance of ± 2 % compared
to the rated values.

### Voltage and frequency

KRT motors are fully functional as defined in DIN EN 60034-1
section 12.3 given a mains voltage fluctuation of ± 10 % and a
mains frequency fluctuation of up to -5 % /+ 3 % compared to
the rated values. They correspond with Design B to DIN EN
60034-1.

## Current and starting current

Depending on the motor size, the starting current at the motor terminals equals 4 to 9 times the rated current at rated voltage. For individual starting current values refer to the motor data booklet or project documentation.

■ **Start-up and starting method**
KRT motors can be started up using either the star-delta or the DOL starting method. The winding ends are supplied wired for both star and delta operation as standard.
If the starting current ratio is limited by the operator, KSB will supply calculated starting curves for the motor (M-n curves) to facilitate selection, dimensioning and parameter setting of a soft starter or autotransformer as an alternative to star-delta starting at 400 V mains voltage.

When a soft starter is used, the requirements on electro-magnetic compatibility to EN 50081 and EN 50082 must be met. When selecting a soft starter observe the information provided by the manufacturer and the electrical motor data, especially the rated current.

■ **Run-up time**
The run-up time of Amarex KRT submersible motor pumps on DOL starting at rated voltage should be less than 1.5 seconds.

■ **Permissible locked rotor time**
The permissible locked rotor time at rated voltage equals:
For cold start-up: ≤ 25 s
For warm start-up: ≤ 5 s

■ **Moments**
For complete pump sets such as submersible motor pumps the moments of inertia, start-up and pull-out torque for the motor do not normally need to be defined. However, for an optimum soft starter setting the corresponding curves can be supplied on request.

■ **Configuration**
On KRT motors the six ends of the stator phase windings are always brought out separately, regardless of the starting method.
They can be connected in delta (e.g. 400 V/ 50 Hz or 460 V/ 60 Hz) or star (e.g. 690 V/ 50 Hz) configuration, depending on the voltage.
The motor wiring is included in the project documentation or will be made available on request.

■ **Frequency of starts**
To prevent excessive thermal loads on the rotor as well as mechanical loads on the bearings and electrical loads on the insulation, the defined frequency of starts per hour must not be exceeded (Table 9).
In this context, the correlation between flow rates and pump sump volume needs to be considered.

| Motor rating | Maximum frequency of starts |
|---|---|
| Up to 7.5 kW | 30/h |
| Above 7.5 kW | 10/h |

Table 9: Maximum number of starts at given motor ratings

■ **Direction of rotation**
The shaft must rotate counter-clockwise, seen from the free shaft end at the impeller. Checking the direction of rotation prior to installing the pump is highly recommended (see operating manual).

■ **Individual motor data**
Individual motor data such as the load-dependent efficiencies, power factor, rated current, etc. can be found in the motor data sheets, including load curves, which are provided on request or together with the project documentation.

■ **Name plate**
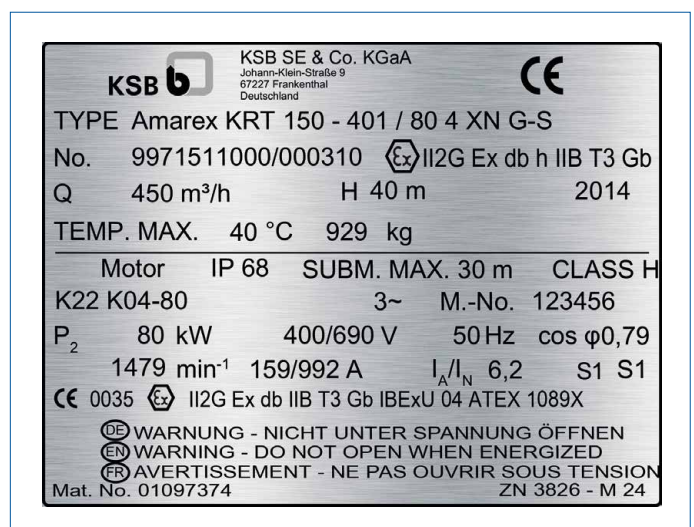A name plate for the complete submersible pump set is shown in Fig. 46.



Fig. 46: Name plate

## KRT motors operated on a frequency inverter

When operating KRT motors on a frequency inverter, observe KSB's notes on operating submersible motor pumps on a frequency inverter or the relevant EUROPUMP publications.

KRT motors are suitable for operation on a frequency inverter. Select any commercially available IGBT frequency inverter with DC link. Any make can be used, even for explosion-proof pumps.
The motor insulation is suitable for pulse voltages of up to 1600 V. Higher pulse voltages are frequent if the rated voltage exceeds 500 V. In this case, fit a dV/dt filter to the frequency inverter or use motors with a special insulation (available on request).

To minimise the risks involved in freely matching frequency inverters to motors, KSB recommends an additional power reserve of 5 %. This power reserve will definitely be able to compensate for any increased losses caused by harmonics in the output voltage of the frequency inverters.

### Selecting a frequency inverter

When selecting a frequency inverter observe the manufacturer's information and the electrical motor data. Consider the rated current of the motor rather than the rated power of the motor, especially for motors with a high number of poles and low cos φ.

### Explosion-proof drives

The following conditions must be met when operating explosion-proof KRT motors on a frequency inverter:

- In steady-state operation, the operating points must be within a range of 50 to 100 % of the rated frequency. Operation above the rated frequency is excluded in the available type test certificates.
- The current limiter of the frequency inverter must be set to 3 x lN as a maximum.
- The thermistor tripping unit must bear the PTB 3.53 – PTC/A conformity mark.

## Design details of the motor

KRT submersible pump sets feature a water-tight, three-phase asynchronous squirrel-cage motor, especially designed and selected to drive submersible motor pumps. Pump and motor have a common pump / motor shaft and form an inseparable unit.

### Stator

The stator of the KRT motor consists of a stator core into which a three-phase current winding made of copper wire has been inserted. High-quality enamelled wire and insulating materials (in the slot and end windings) combined with polyester resin impregnation provide high mechanical and electrical strength.

### Insulation materials and thermal class

The insulation system of the motor is made exclusively with market-proven components from renowned manufacturers. All insulation materials comply with the requirements of thermal class H. A variety of lacing methods is available for the end windings.

Within the rated conditions the motor winding temperatures comply with thermal class F.

### Rotor

The rotor is of the squirrel-cage type. The cage in the rotor core is either made of die-cast aluminium or of soldered copper bars and rings, depending on the motor size.

### Bearings

The drive end and non-drive end are fitted with rolling element bearings. Up to pump size 280 the bearings are greased for life; the bearings of pump sizes 315 and above come with a re-lubrication system. The lubricants used are high-temperature lithium soap base greases. The grease quality is defined in the operating manual.

# Cooling

The motor is primarily cooled with air. Inside the motor the air is circulated in a closed system by fan blades on the two ends of the rotor. The heat generated in the motor is dissipated via the casing to the water which acts as a secondary coolant.

EN 60034-6 code: IC 4 A1 W 8.

# Monitoring equipment

### Overcurrent protection

The motor must be protected from overloading by a thermal time-lag overcurrent relay in accordance with DIN VDE 0660/ IEC 947 (Fig. 47). The relay has to be set to the rated current of the motor as indicated on the name plate (see Fig. 46) or as specified in the project documentation, or as specified following a request.
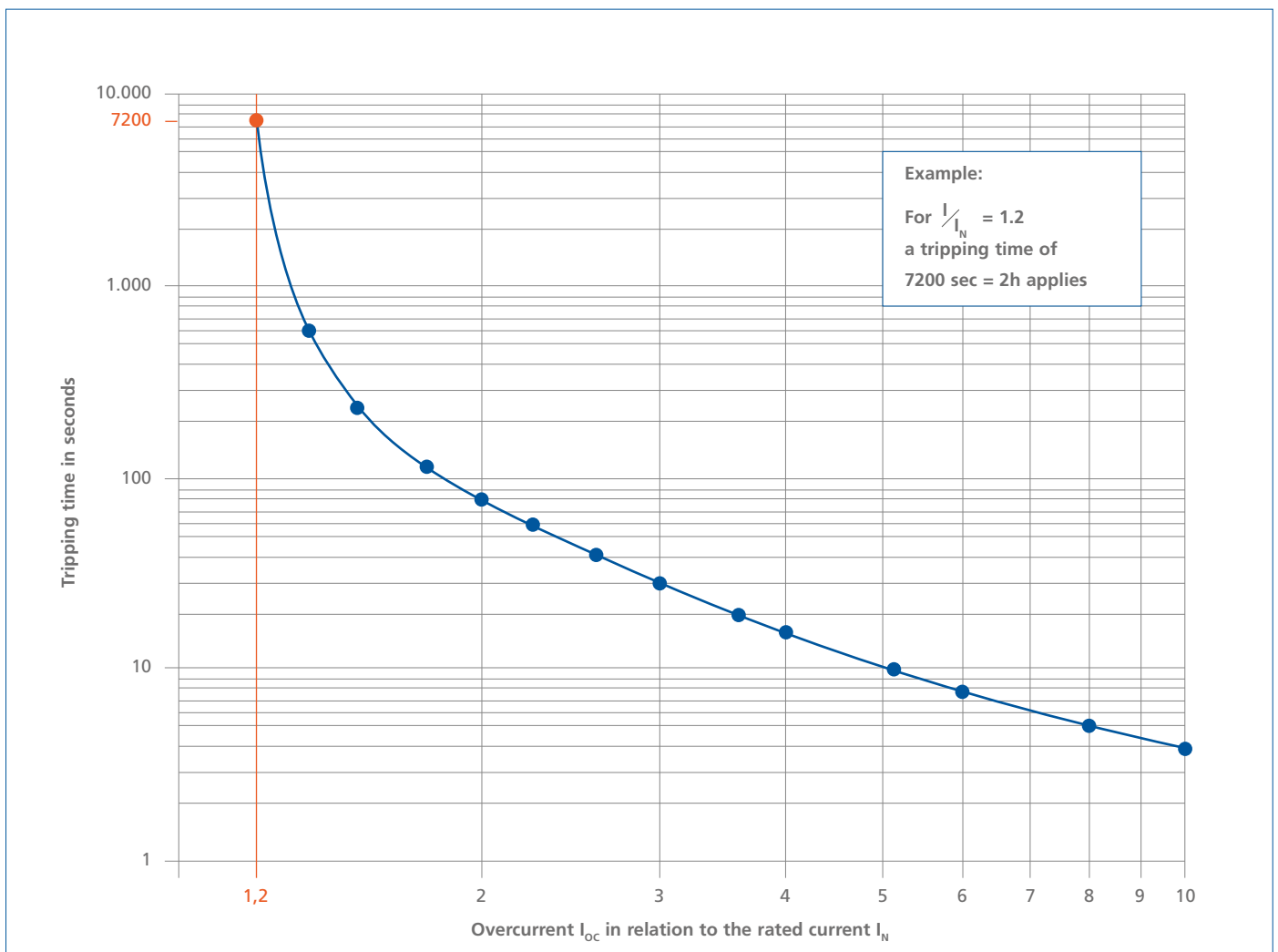


Fig. 47: Tripping curve for class 10 thermal time-lag over-current trips to EN 60947-6-2; VDE 0660-115: 2007-12, low-voltage switchgear and controlgear.

**Anti-condensation heater**

The motors of KRT submersible motor pumps do not require an anti-condensation heater (see section "Leakage protection of the motor"). The water-tight design prevents condensation inside the motor. Any residual humidity from the assembly process is absorbed by silica gel sachets.

**Sensors in the motor pump set**

**Winding temperature monitoring without explosion protection (installation types S and P / wet)**

The windings of KRT motors are protected by a temperature monitoring circuit. Two bimetal switches connected in series or three PTC thermistors connected in series monitor the temperature; they trigger a switch-off signal when the permissible winding temperature limit is reached, and a switch-on signal after the windings have cooled down. The plant engineering consultant must specify how the circuit breakers are controlled.

**Winding temperature monitoring with explosion protection (installation types S and P / wet)**

Explosion-proof KRT motors must be monitored by two independent temperature sensors in the winding.

- Two bimetal switches connected in series monitor the temperature; they trigger a switch-off signal when the permissible winding temperature limit is reached, and a switch-on signal after the windings have cooled down. The bimetal switches protect the motors against overloading and dry running.
- In addition, three PTC thermistors connected in series (with a tripping temperature 20 Kelvin above the maximum winding temperature) are fitted. In the event of a failure of the bimetal temperature sensors, these additional thermistors will trigger a switching operation which trips the pump before the permissible temperature limits for explosion protection at the surface of the motor housing are exceeded.

The connection of fully functional temperature limiters is mandatory for explosion-proof KRT pumps. Automatic reset and start-up is not permitted in this case. For this purpose a commercially available thermistor motor protection relay with manual reset must be integrated in the control circuit of the motor contactor.

**The winding temperature of dry operated submersible motor pumps (installation types K and D) is exclusively monitored by PTC thermistors.**

**Leakage protection of the motor**

A conductive leakage sensor is fitted to check the motor space for integrity. If any leakage penetrates into the motor space a current will flow through the sensor to the earth conductor. The sensor can be combined with any commercially available electrode relays which trigger an alarm when the electrode resistance falls below 6 kΩ. For small pumps, a commercially available 30 mA RCD can be used as an alternative.

**Bearing temperature monitoring**

The temperature of the motor bearings is monitored as follows, depending on the motor size:

- Motor rating > 30 kW: monitoring of pump-end fixed bearing optional
- Motor rating > 60 kW: monitoring of pump-end fixed bearing as standard, monitoring of motor-end radial bearing optional.

Each monitored bearing is protected by a temperature monitoring circuit (Table 10). The temperature is monitored by Pt100 temperature sensors installed in the bearing housing. The sensors change their resistance proportionally with temperature.

| Temperature | Resistance R | Warning |
|:---:|:---:|:---:|
| 20 °C | 107.7 Ω | Test / cold motor |
| 110 °C | 142.1 Ω | Warning |
| 130 °C | 149.8 Ω | Stop |
| Special lubricant | | |
| 130 °C | 149.8 Ω | Warning |
| 150 °C | 157.0 Ω | Stop |

Table 10: Bearing temperature monitoring

Formula for calculating the resistance relative to the temperature:

$$R = 100 \text{ Ω} \cdot \left( 1 + 0.00383 \cdot \frac{T}{°C} \right)$$

Formula (14)

Key:
T    = Temperature in °C
R    = Resistance in Ω

**Mechanical seal monitoring**

The mechanical seals of motors with ratings > 60 kW are moni-
tored as standard. An integrated float switch (NC contact) opens
when water penetrates into the leakage chamber as a result of
defective mechanical seals. The contact may either trigger an
alarm signal or trip the motor.
The NC contact is suitable for a maximum of 250 V AC / 1.5 A.

**Vibration velocity monitoring**

Motors with a rating above 60 kW can be fitted with a sensor at
the upper bearing to monitor the effective vibration velocity. The
sensor transmits an analog 4 - 20 mA measuring signal. It must
be supplied with a voltage of 15 to 32 V DC. Pumps of this vari-
ant are always fitted with a shielded control cable.

For KSB submersible motor pumps with multi-channel impellers
(K impellers), observe the limits in Table 11.

On submersible motor pumps with single-channel impeller (E
impeller), higher vibration velocities of up to 17 mm/s can be
caused by hydraulic imbalance, depending on the operating
point.

| $V_{rms}$ [mm/s] | $V_{rms}$ [inch/s] | $V_{peak}$ [inch/s] | Measuring signal [mA] | Status |
|---|---|---|---|---|
| 9 | 0.35 | 0.50 | 11.2 | Not critical |
| 11 | 0.43 | 0.61 | 12.8 | Warning |
| 14 | 0.55 | 0.78 | 15.2 | Emergency stop |

See also EUROPUMP-Leitfaden / Schwingungen bei Kreiselpumpen [Europump Guidelines / Vibrations of Centrifugal Pumps],
Table A.1 / Flexible installation, vertical)

Table 11: Vibration velocity limits for submersible motor pumps with multi-channel impeller

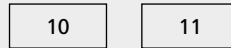# Connection and description of monitoring equipment

**Thermal motor monitoring**

Brief description of the sensors



| 20 | 21 |
| --- | --- |
| 21 | 22 |
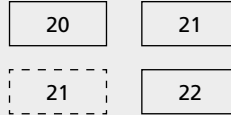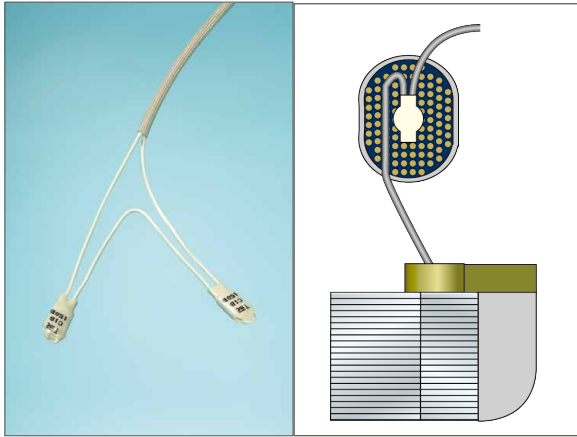
**Bimetal switch**
- Temperature-sensitive miniature contact
- Glued into the motor winding
- Volt-free NC contact; 250 V ~; 2 A

Closed          Temperature OK
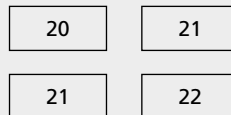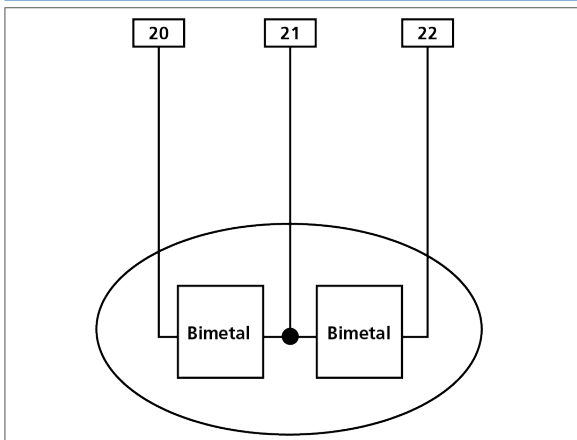Open            Temperature too high

| 10 | 11 |
| --- | --- |

**PTC thermistor**
- Temperature-dependent semiconductor resistor with positive temperature coefficient
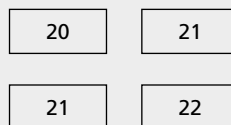- Glued into the motor winding
- Max. voltage 30 V

R < 1250 $\Omega$          Temperature OK
R > 4000 $\Omega$          Temperature too high

Connection for motor ratings up to 4 kW (for submersible motor pump type Amarex N only)



| 20 | 21 |
| --- | --- |
| 21 | 22 |

**Motor versions U / W, non-ATEX-compliant**
- No direct connection to the control circuit required; connect to dummy terminal

| 20 | 21 |
| --- | --- |
| 21 | 22 |

**Motor version Y, ATEX-compliant**
- To be wired directly into the control circuit
- Connection via thermistor motor protection relay with manual reset

Sensors for motor ratings > 4 kW (installation types S and P)



| 20 | 21 |
| --- | --- |
| 10 | 11 |
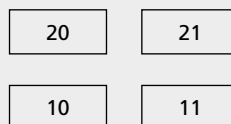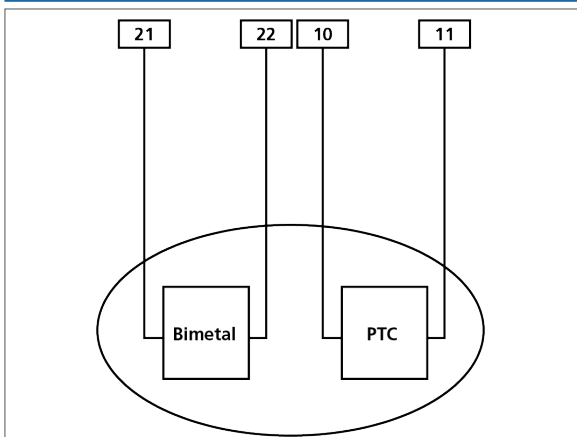
**Motor versions U / W / UN / WN, non-ATEX-compliant**
- No direct connection to the control circuit required; connect to dummy terminal
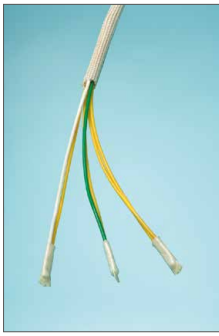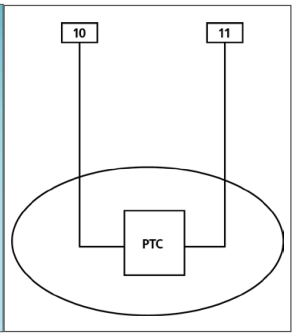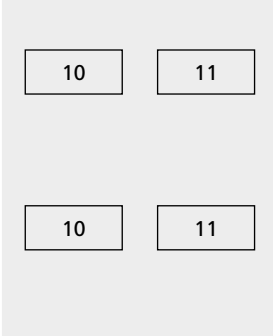
| 20 | 21 |
| --- | --- |
| 10 | 11 |

**Motor versions X / Y / XN, ATEX-compliant**
- To be wired directly into the control circuit
- Connection via thermistor motor protection relay with manual reset

## Thermal motor monitoring

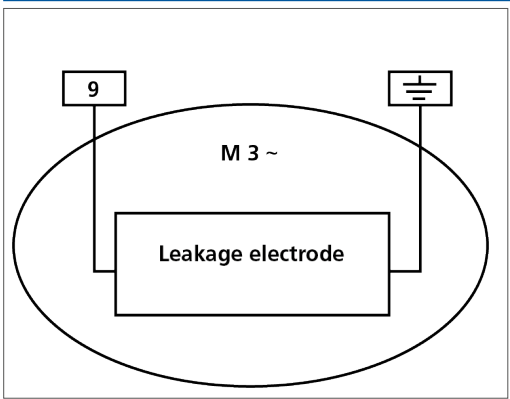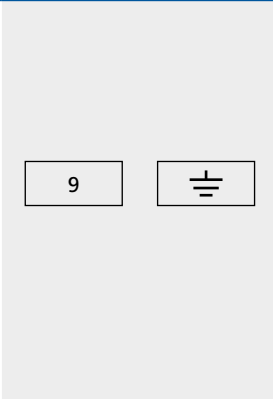### Connection for motor ratings > 30 kW (installation types K and D)

| 10 | 11 |

**Motor version UN, non-ATEX-compliant**
- Connection via thermistor motor protection relay with manual reset

| 10 | 11 |

**Motor version XN, ATEX-compliant**
- Connection via thermistor motor protection relay with manual reset

## Monitoring by leakage sensor (in the motor space)

### Brief description of the sensors

| 9 | ⏚ |

**Leakage sensor**
- Conductive sensor
- Screwed to the lower bearing bracket; for motors > 65 kW additionally at the upper bearing bracket
- AC voltage sensors shall be used to prevent the formation of insulating layers
- Max. voltage 30 V
- To be triggered at a leakage resistance of approx. 6 kΩ

### Connection for all motor types

M 3 ~

Leakage electrode

| 9 | ⏚ |

**Motor versions U / X / Y / W / UN / XN / WN ATEX-compliant or non-ATEX-compliant**
- Connection to an electrode relay with the following parameters:

Sensor circuit    10 - 30 V ~
Tripping current  0.5 mA

## Thermal bearing monitoring

### Brief description of the sensors
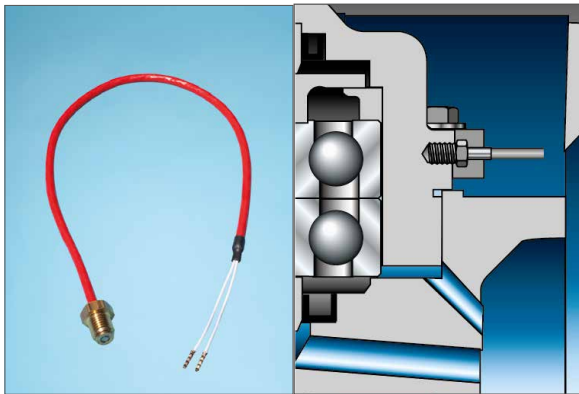


**Pt100 - ball bearings**

- Pt100 resistance thermometer
- M8 thread in the bearing housing
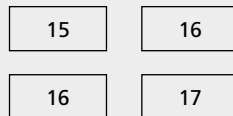- Analog, continuous temperature signal
- Max. voltage 6 V

| 15 | 16 |
|----|----|

Lower bearing

| 16 | 17 |
|----|----|

Upper bearing (motors > 65 kW – optional)

Alert temperature:           130°C
Cut-out temperature:        150°C

### Connection for motor ratings > 30 kW
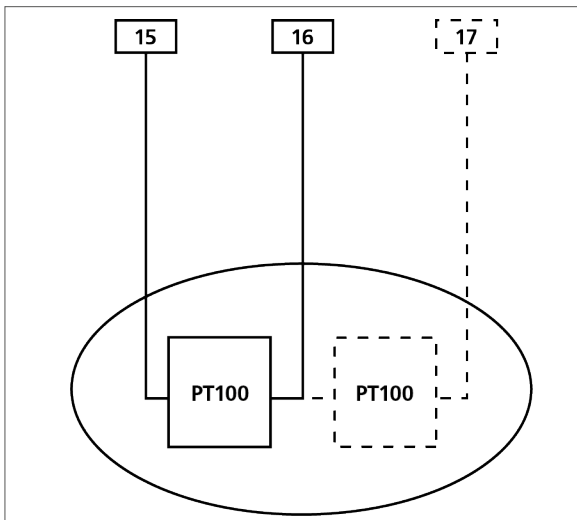


**Motor versions U / X / Y / W**
**ATEX-compliant or non-ATEX-compliant**

- Connection to a Pt100 switching relay with the following parameters:

| 15 | 16 |
|----|----|
| 16 | 17 |

Alert temperature:           110°C
Cut-out temperature:        130°C

**Motor versions UN / XN / WN**
**ATEX-compliant or non-ATEX-compliant**

- Connection to a Pt100 switching relay with the following parameters

| 15 | 16 |
|----|----|
| 16 | 17 |

Alert temperature:           130 °C

Cut-out temperature:        150 °C

## Mechanical seal monitoring via float switch

### Brief description of the sensors



| 3 | 4 |

**Float switch**
- Volt-free NC contact; 250 V ~; 2 A
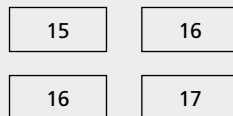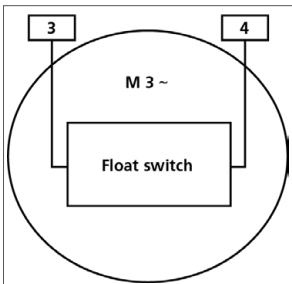
| Closed | Leakage chamber empty |
| Open | Leakage, check mechanical seal |

### Connection for all motor types



| 3 | 4 |

**Motor versions U / X / Y / W / UN / XN / WN**

**ATEX-compliant or non-ATEX-compliant**
- Connection for alarm and cut-out

| Measured variable | Submersible motor pump | Sensor | Recommended tripping device | Material No. |
|---|---|---|---|---|
| Winding temperature | Amarex KRT < 4 kW, non-ATEX-compliant Motor U or W up to pump size 80-210 | Bimetal switch | NC switching contact, no switchgear required | – |
| | Amarex KRT < 4 kW, ATEX-compliant Motor X or Y or Z up to pump size 80-210 | Bimetal switch and | NC switching contact, no switchgear required | – |
| | | Bimetal switch | Restart interlock function to be implemented by customer | – |
| | Amarex KRT > 4 kW, non-ATEX-compliant Motor U or W from pump size 40-250 Installation type P or S without cooling jacket | Bimetal switch | NC switching contact, no switchgear required | – |
| | Amarex KRT > 4 kW, ATEX-compliant Motor X or Y or Z from pump size 40-250 Installation type P or S without cooling jacket | Bimetal switch and | NC switching contact, no switchgear required | – |
| | | PTC | Thermistor motor protection relay (with manual reset / restart interlock) DOLD MK 9163N.12/110 AC 230V | 01 086 253 |
| | Amarex KRT, all motor variants, all pump sizes Installation type D or K (cooling jacket or convection cooling with air) | PTC | Thermistor motor protection relay (with manual reset / restart interlock) DOLD MK 9163N.12/110 AC 230V | 01 086 253 |
| Bearing temperature | KRT > 60 kW (all motor variants / shaft centreline heights K22, K28, K31, K35; K40) | Pt100 | DOLD BA9094.28/100 AC 230V | 01 118 924 |
| Motor leakage | | Leakage sensor | For mains operation: Elettrosonda QNS AC 230V | 11 303 923 |
| | | | For operation on frequency inverter: DOLD MK9151.11/KSB AC 230V (Integrated filter for enhancing EMC behaviour) | 01 085 179 |
| Mechanical seal leakage | KRT > 60 kW (all motor variants / shaft centreline heights K22, K28, K31, K35; K40) | Integrated float switch | NC switching contact, no switchgear required | – |

Table 12: Amarex KRT 50 Hz sensors

1 Long rubber gland
2 Outer cable insulation
3 Individual cores are cast in epoxy resin
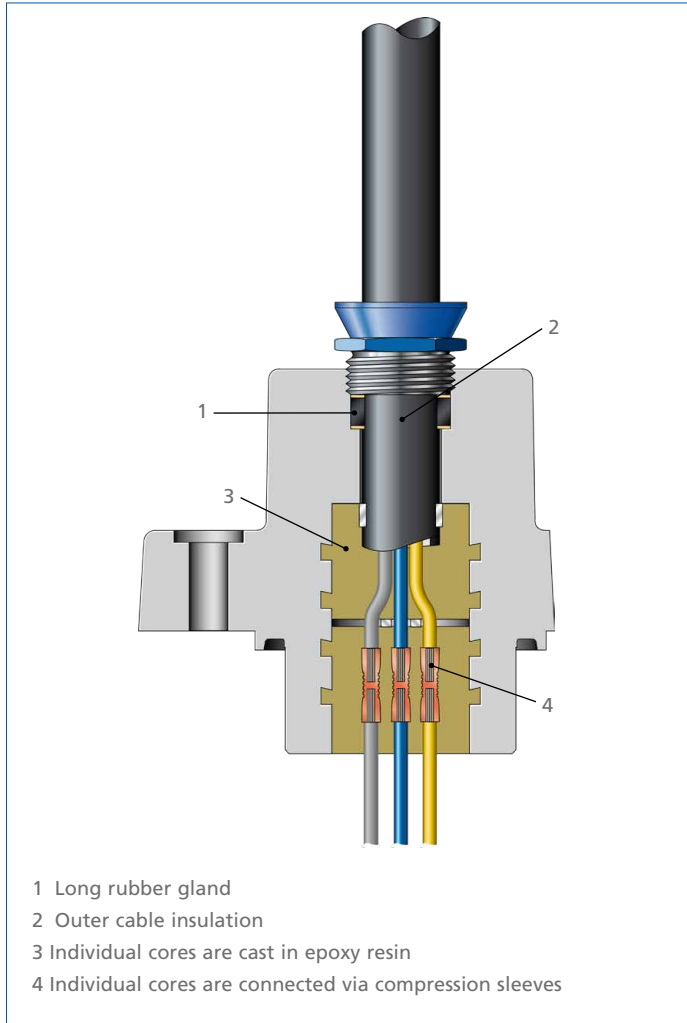4 Individual cores are connected via compression sleeves

Fig. 48: Cable gland of a KRT

# Power and control cables with cable glands

Amarex KRT submersible motor pumps are supplied fitted with flexible cables suitable for use in waste water.
Entering the motor through special cable glands, the cables are connected in the motor as follows:

- For motor ratings < 60 kW with plug-type or crimp connections
- For motor ratings > 60 kW with terminal board and cable shoes.

### Cable gland

The cable glands are absolutely water-tight up to an immersion depth of 30 m, with multiple safety features (Fig. 48):
1. Long rubber gland
2. Resin-embedded cable sheath
3. Individual cores stripped, tinned and sealed in resin

The power and control cables are suitable for use in waste water and are of especially high mechanical strength. Depending on the operating conditions the following cable types are available for selection:

- S1BN8-F / standard
- S07RC4N8-F / shielded version optional
- Tefzel with ETFE cable sheath / optional for chemically aggressive fluids

Number of conductors and cross-sections:
Refer to the project documentation or motor data sheet, or request details.

### Benefit:

Long service life of the motor due to an absolutely water-proof design (see also section "Shaft seal", page 29).
Only high-quality flexible cables are used for submerged operation to ensure reliable continuous operation of submersible motor pumps and a long service life. KSB has developed cables optimised for waste water applications together with a renowned cable manufacturer.

# Power cables

## Rubber-sheathed cable

### Brief description

OZOFLEX-(PLUS) S1BN8-F rubber-sheathed cables have been developed for KSB's standard and explosion-proof pumps. They are designed to flexibly connect KSB submersible motor pumps up to a cable cross-section of 50 mm².

Given the variety and changing composition of waste water, the cables must only be used in areas which can be easily accessed and inspected.

If aggressive water or water of special composition is handled, the resistance of the cables must be checked in each individual case.

The cables can be used indoors, outdoors, in potentially explosive atmospheres, in areas with a high fire risk, in industrial plants, in commercial and in agricultural facilities.

The regulations stipulated in
DIN VDE 0298-300 must generally be observed.

Design based on DIN VDE 0828-16,
VDE-REG.-NO. 7586.

### Technical data

| | |
|---|---|
|  | Stranded class 5 copper conductors to DIN VDE 0295<br><br>EPR insulation<br><br>Inner sheath made of rubber<br><br>Special outer sheath made of synthetic rubber, colour: black |
| **90°c** | Continuously permissible conductor temperature |
| **200°c** | Permissible conductor temperature with short circuit (up to 5 seconds) |
| | Fire protection to DIN EN 50265-2-1 |
| | Resistant to UV radiation, weather and ozone |
| | Oil-resistant to DIN VDE 0473-811-2-1, part 10 |
| | Flexible |
| | Laying and transport temperature: -25 to +80 °C |
| **V** | 0.6 / 1 kV: with green/yellow conductor |
| | For application in water / not drinking water |

# Tefzel cable (TEHSITE)

## Technical data

| | |
|---|---|
| | Stranded class 5 copper conductors to DIN VDE 0295 |
| | TE400 insulation |
| | Inner sheath made of silicone |
| | TE-400 outer sheath, colour: black |
| **135°C** | Continuously permissible conductor temperature |
| **270°C** | Permissible conductor temperature with short circuit (up to 5 seconds) |
| | Fire protection to DIN EN 50265-2-1 |
| | Resistant to UV radiation, weather and ozone |
| | Oil-resistant / general chemical resistance |
| | Flexible |
| | Laying and transport temperature: -25 to +80 °C |
| **V** | 450/750 V: with green/yellow conductor |
| | For application in water / not drinking water |

## Brief description

TEHSITE cables (TEFZEL) are resistant to heat and chemical substances. They are designed to flexibly connect KSB's submersible motor pumps if the temperature of the fluid handled and/or the ambient temperature exceed 60 °C or if high chemical resistance is required.

The scope of application is defined in the corresponding VDE report of 30 November 1983 and its amendment of 14 October 1987.

Due to their design and materials TEHSITE cables are less flexible than rubber-sheathed cables.

The general regulations stipulated in DIN VDE 0298-300 must be observed.

Design based on DIN VDE 0828-16.

# Shielded rubber-sheathed cable

## Brief description

OZOFLEX (FC+) S07RC4N8-F rubber-sheathed cables were developed to flexibly connect submersible motor pumps to frequency inverters. They meet the requirements of the EMC directive and can be supplied up to a cable cross-section of 50 mm².

Given the variety and changing composition of waste water, the cables must only be used in areas which can be easily accessed and inspected.
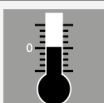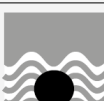
If aggressive water or water of special composition is handled, the resistance of the cables must be checked in each individual case.

The cables can be used indoors, outdoors, in potentially explosive atmospheres, in areas with a high fire risk, in industrial plants, in commercial and in agricultural facilities.

The general regulations stipulated in DIN VDE 0298-300 must be observed.

Design based on DIN VDE 0282, part 16.

## Technical data

| | |
|---|---|
| | Stranded class 5 copper conductors to DIN VDE 0295<br><br>EPR insulation<br><br>Inner sheath made of rubber<br><br>Braided shield made of tinned copper wires Special outer sheath made of synthetic rubber, colour: black |
| 90°C | Permissible conductor temperature |
| 250°C | Permissible conductor temperature with short circuit (up to 5 seconds) |
| | Fire protection to DIN EN 60332-1-2 |
| | Resistant to UV radiation, weather and ozone |
| | Oil-resistant to DIN EN 60811-2-1 |
| | Flexible |
| | Laying and transport temperature: -25 to +80°C |
| V | 450 / 750 V: with green/yellow conductor |
| | For application in water / not drinking water DIN VDE 0282-16 HD 22.16 |

# Quality assurance and test logs

### Standard routine testing

Standard routine tests of KRT motors are conducted to the IEC 60034-1 standard, which comprises:

- Winding resistance test
- Insulation resistance test
- High-voltage test on the winding
- Earth conductor test
- Check of monitoring devices
- No-load current
- Direction of rotation check

### Sample type testing

Sample type testing is performed in accordance with IEC 60034 for each motor type. Type testing includes determining the motor efficiency and starting current and a temperature rise test.

Optionally, pumps can be supplied with a so-called 2.2 certificate of a motor of the same design.

# Piping and Valves

Detailed planning is a prerequisite for proper installation and maintenance, trouble-free operation and a high availability of technical equipment. This also applies to the piping and valves within and outside the pumping station. The discharge lines transport the fluid handled from the pump to the discharge point. The pump and discharge line should be seen as a technical / hydraulic unit. They can be graphically illustrated via the pump characteristic curve and the system characteristic curve, also referred to as system curve.

We will refer to the piping inside the pumping station as inner piping. The discharge lines outside of the pumping station right up to the actual discharge will be referred to as outer discharge lines.

In general, the inner piping of a pumping station consists of suction lines and discharge lines. As KRT pumps are submersible waste water motor pumps usually operated in stationary wet installation, suction lines are not present.

In practice, the outer discharge lines are also referred to as waste water discharge lines or transport lines. They are usually laid underground in frost free conditions. When planning the piping layout, extreme high and low points should be avoided in the waste water discharge lines if possible. If any constraints require such high and low points, additional technical measures must be taken, such as fitting flushing connections and drains at low points, and air valves at high points.

To reliably transport municipal waste water a free passage of 100 mm is recommended through the pump impeller, the valves and the discharge line. The inside diameter of the piping should measure 80 mm as a minimum (see ATV 134).

## Planning the piping

### Dimensioning the piping

Initial key parameters for the dimensioning of discharge lines:
- Flow rate
- Operating pressure

The dimensioning or calculation of the inner (nominal) diameter depends on the:
- Flow velocity

If the nominal diameter is known, the following parameters can be calculated for a specified flow rate:
- Head losses or pipe friction losses

The formula "geodetic head + head losses = manometric head" can be used to determine the
- Operating pressure.

A transient flow analysis can be used, if required, as an approach to calculate the operating pressure, nominal pressure and conduct static piping calculations.

This section explains the selection parameters in further detail.

### Determining the flow velocity and selecting the nominal diameter
- Based on the required or planned flow rate, the nominal diameter is directly dependent on the flow velocity. The pipe friction losses can be derived from the flow velocity.

From an economic point of view (investment and operating costs) the following aspects need to be taken into account when specifying the flow velocities and nominal diameters of the pipes.

Flow velocities below the specified minimum can lead to failures (clogging, etc.). Flow velocities above the specified maximum can also lead to operational failures. In addition, they generate high pipe friction losses, consuming unnecessary energy.
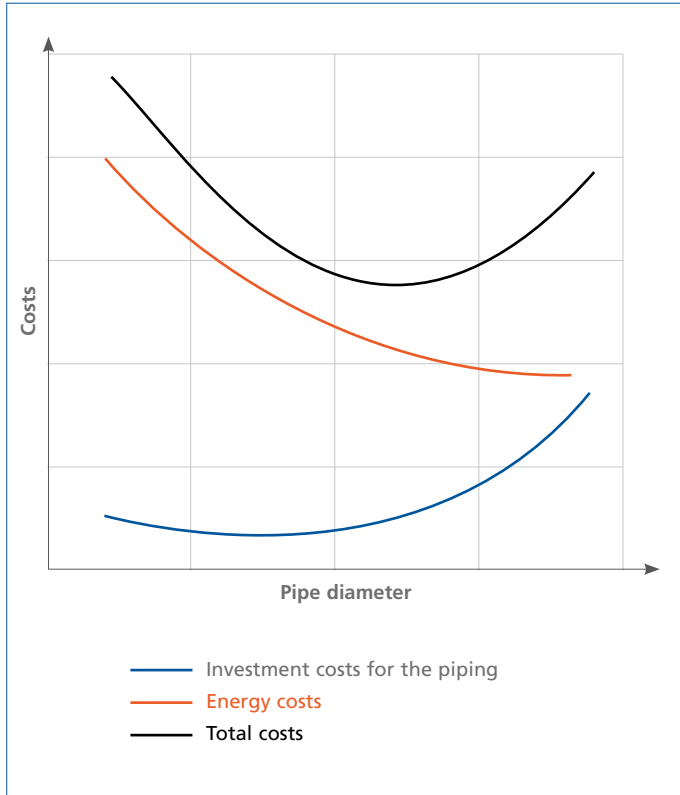
Fig. 49: Cost structure for designing and operating a piping system

**Nominal diameter / inside diameter of the pipes**

The nominal diameter of the discharge line can be derived from the maximum planned flow rate, the piping length and the fluid handled based on the reference values mentioned above.
The selection of an optimum nominal diameter should be based on a profitability analysis, i.e. the investment costs should be compared to the operating costs and the energy costs in particular. Small nominal diameters with high flow velocities will cause high flow losses in longer pipes. As a result, pumps with a higher head and higher pump input power will be required (Fig. 49).

The nominal diameter of the pipes should be equal to or greater than the nominal diameter of the corresponding pump connection. For small pumping stations, the pumps must not only be dimensioned for the inflow; resistance to clogging and the minimum speed must also be considered. If the recommended flow velocity in the vertical discharge line is not reached, measures should be taken to prevent clogging (e.g. a flushing connection). Flow velocities higher than 2.5 m/s should be avoided.

A special case is the drainage of isolated properties (e.g. pumped drainage) that are to be connected to a central waste water system. If pumps with cutters are used, smaller diameter pipes can be selected. This type of drainage is limited to a pump flow rate of approx. Q = 20 m³/h.
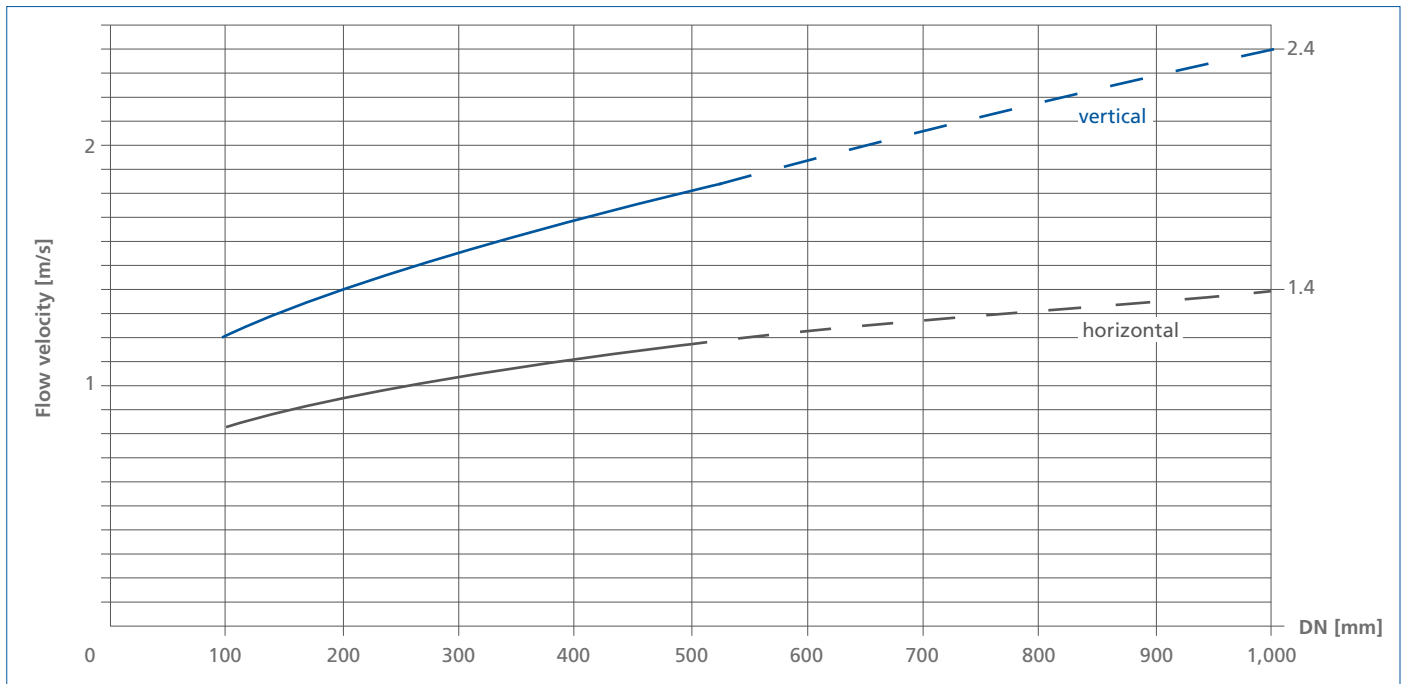


Fig. 50: Minimum flow velocity

## Flow velocities

Compared to drinking water lines, the specific feature of waste water lines is that they need to reliably transport a large variety of substances and fluid compositions such as fibres and mineral solids of various sizes, e.g. sand, grit, stones and organic contaminants.

For this reason, the minimum flow velocities must be observed. The following aspects need to be considered:

- Different flow velocities for vertical and horizontal pipes
- Internal pipe diameters – bearing in mind that larger diameters also require higher flow velocities
- Fluid composition (analysis of fibre content, solids content and particle size)
- Pipe operating modes (intermittent or continuous)
- Total pipe length

Some research was conducted on the transport of substances contained in waste water at the Berlin Technical University. The results are documented in Fig. 50. The nominal diameter range of DN 100 to DN 250 was examined in detail, and the nominal diameter range of DN 500 to DN 1000 was extrapolated. The substances in the fluid handled are those commonly contained in waste water such as fibre, gravel 0/4, gravel 16/32 and sharp-edged granite gravel 2/5.

## Surge analyses

In water supply and waste water systems any change in the operating conditions also leads to dynamic changes of pressure and flow. These transient processes in piping systems for the extraction, treatment, transport and distribution of water have to be taken into account and examined when planning a pumping station as they can cause significant damage to pipes, valves and other internal parts. Transient operating conditions can have many causes and are generally unavoidable. They can be the result of pump failure, pumps starting up or stopping, or cutting in or cutting out in addition to already operating pumps, also of variable speed pumps or of closing/opening valves, control valves, pressure-reducing valves, etc.

Given the complexity of surge pressure, the use of the numerous "proximation" calculation methods should be avoided as their scope of validity is very limited and cannot be applied in general. The use of such methods outside their scope of validity can lead to significant dimensioning errors.

In this context we would like to refer to the DVGW Set of Rules, Worksheet W 303 "Dynamic pressure variations in water supply systems", and to the brochure "KSB Know-how – Water Hammer".

**We recommend consulting experts in this field to assess the surge pressure and define preventive measures.**

## Static piping calculations

The pipes must be able to continuously and reliably withstand the internal and external pressures acting on the system. Unless preventive measures have been taken, this includes the pressures of transient processes (e.g. surge pressure). The required pressure class, the external loads and the material must be considered when calculating the required wall thickness of the piping.

The necessity of conducting static and dynamic piping calculations should be established on a project-by-project basis. The piping calculations can comprise:

### Static calculations

- Determining the primary loads such as the internal pressure, the weight of the pipe, the valves and of all other add-on or internal parts, plus the weight of the water fill.
- Determining the secondary loads such as stresses or forces caused by the impacts of temperature difference in relation to the design temperature
- Determining occasional loads such as wind, ice and snow

### Dynamic calculations

- Determining the dynamic loads
- Conducting a stress analysis
- Determining the vibration behaviour showing the excitation and natural frequencies including an evaluation of the resonance behaviour
- Proof of earthquake stability

**Note:**

Secondary loads (forces and moments), also called "thermal loads", are often larger than primary loads (weight and pressure), especially if the piping is rigidly connected between two anchorage points. This often leads to the permissible force limits in parts of the structure (e.g. wall ducts) being exceeded. Additional preventive measures must be taken, such as fitting bellows or other expansion joints.

The piping layout can also be adjusted to reduce stresses and forces. In a straight piping layout between two wall penetrations, the pipe cannot "give way" unlike an angled piping layout, where stresses and forces are much lower. It is also important to properly select and arrange plain and fixed bearings as well as guides.

## Piping layout

### Inner piping

For installation in the pump sump, the submersible waste water motor pump is supplied with a duckfoot bend, a guide arrangement (guide wire or guide rail) and a lifting chain or rope or a hooking device for lifting and lowering the pump.

The discharge line is connected to the duckfoot bend which is fastened to the floor of the pump sump. The individual discharge line consists of a vertical riser with a horizontal discharge line branching off. If the pumping station consists of several pumps with the same pumping destination, each pump has its own discharge line. For longer pipe lengths the individual discharge lines converge in one or several collecting discharge lines.

The valves are either arranged vertically in the risers in the pump sump (Fig. 52) or horizontally in a separate valve pit (Fig. 54 and Fig. 55). When fitting the valves in the risers, they should be installed in the upper section of the riser, in order that:
- The valves are protected from flooding when installed above the water level.
- The shut-off valves can be easily accessed for operation,

- The check valves can be easily accessed for maintenance purposes.
- A minimum distance to the duckfoot bend is observed to avoid problems when starting up the pump due to air pockets in the discharge line.
- The amount of solids depositing on the swing check valve's disc due to sinking waste water particles is reduced.

The shut-off valves' actuating elements (e.g. handwheel) must be positioned such that pump removal is not obstructed.

Risers above the swing check valve should be kept as short as the site conditions allow to prevent possible solid deposits. Risers must always be integrated into the collecting discharge line in the horizontal axis (Fig. 54). Their integration should be designed to be as hydraulically optimised as possible. Depending on the site conditions, saddle connections, weld-in elbows, Y-pipes and specific connecting angles can be used (Fig. 56).

If the cross-section needs to be increased, select suitable expansion tapers (referred to as "reducers" in piping contexts). Choose the smallest possible diffuser angle. Use eccentric reducers in the riser to allow the pump to be pulled out without being obstructed by pipe fittings.

If an absolutely tight seal is required to guide the discharge line through the wall of the structure, a flanged wall pipe (sometimes referred to as a puddle flange) or a flexible wall duct can be used. When sealing the wall penetration with a modular seal (Fig. 53), make sure that the pipe is correctly centred and fastened in the opening.
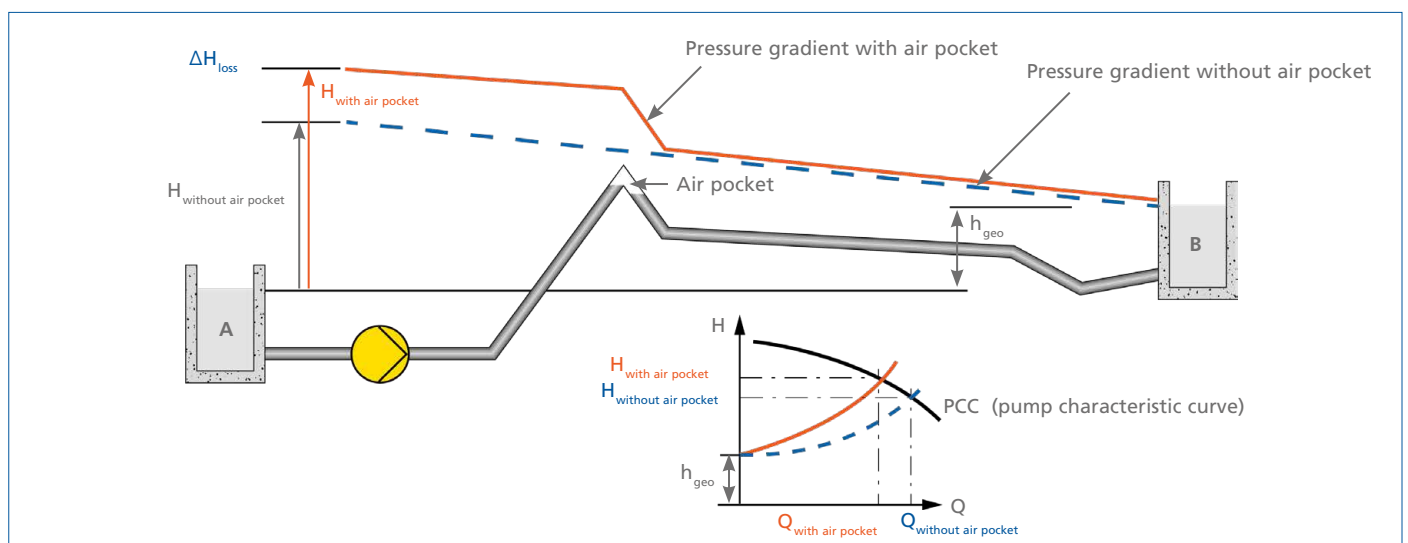


Fig. 51: Pump discharge line with and without air pockets

Expansion and dismantling joints or bellows expansion joints should be installed in the piping as required to allow for proper installation without stresses or strains, compensation of length tolerances and dismantling for any repair work. These tasks can usually also be covered by flanged pipe bends installed in a suitable position in the piping layout.

The number of flanged connections should be kept to a minimum to facilitate assembly and repair of the piping. It should be ensured that flanged connections required for repair and for the installation of prefabricated piping units are provided. On-site welding should be minimised.

The flanged connections should be selected as required for the fluid handled, the chosen pipe material and the maximum system pressure. Gaskets of DN 200 and larger should be selected with a steel insert. When using fasteners made of stainless steel, use bolts of quality V2A and hexagon nuts of quality V4A.

If space is at a premium, the collecting line can also be installed outside the pump sump. A separate structure for the gate valves (also referred to as valve chamber or valve pit) might be a useful option for space or operating reasons (Fig. 54 and Fig. 55).

If high points in the discharge line of the pumping station cannot be avoided, a venting option must be provided. Waste water treatment plants should generally be fitted with a vent as gas emissions are to be expected. If the external piping is laid with a downward slope, an automatic air valve should be fitted as a vacuum breaker at the highest point of the pumping station.

If required, provide drain options and flushing connections (e.g. nozzle, ball valve or Storz coupling with blind cap).

If the pumping station is to be installed as an open structure (tank), frost protection measures must be taken as necessary.



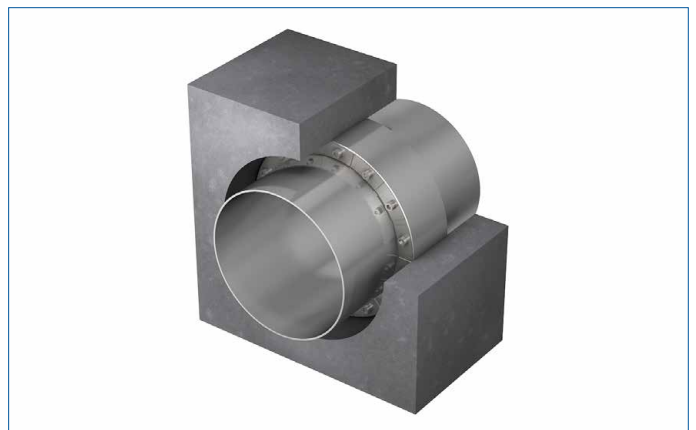Fig. 52: Vertical valve arrangement in the pump sump



Fig. 53: Pipe sealing using a modular seal



Fig. 54: Valve chamber with the piping integrated horizontally

Fig. 55: Valve pit

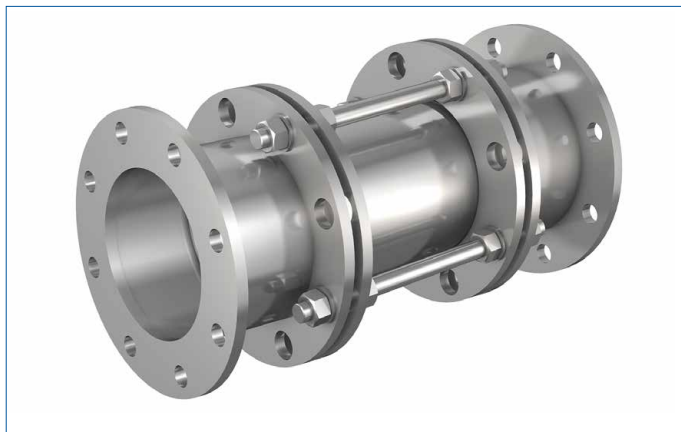Fig. 56: Integration of individual discharge line in flow direction

Fig. 57: Articulated pipe joint for compensating differences in height
of pipes

Fig. 58: Double-clamp sliding support

**Outer piping**

The discharge line from the pumping station should ideally be laid with a rising slope towards the outlet. If the topographic conditions require the piping to be laid with pronounced high and low points, air or gas could accumulate at the high points and sediments at the low points. To prevent this the following options are available:

- When dimensioning the nominal pipe diameter the required minimum flow velocity of the waste water must be taken into account. In accordance with DWA 134, it should be between 0.5 m/s and 1.0 m/s depending on the total daily pumping time. Fig. 50 shows the nominal diameter of waste water pipes as a function of the recommended minimum flow velocity required for transporting substances contained in waste water. The required minimum flow velocity has been determined in pumping tests.

- High points may involve the risk of air and gas accumulations constricting the free pipe cross-section. Air and gas bubbles can enter the piping via pumps, leaks, vent valves, biological and chemical reactions, the intentional introduction of air for flushing or as a result of gas evolution. Larger accumulations of air lead to higher flow resistances and increasing pressure losses which reduce the pump flow rate. (See Fig. 51: Pump discharge line with and without air pockets). An examination should therefore be carried out to see whether self-venting of the high point may occur through the fluid's flow velocity. Air expulsion not only depends on the flow velocity and turbulence of the fluid handled, but also on the pipe inclination and the air pocket size at the beginning of air expulsion. The formation of air or gas accumulations can be avoided by installing automatic air valves at high points. Air valves also serve as a surge pressure protection measure (see Know-how brochure "Water Hammer") and are needed to ensure that the pipe can be completely emptied and unwanted backflow due to the siphoning effect is avoided. The valve is also required for the controlled filling of the pipe so that the displaced air can escape from the pipe.

- Low points involve the risk of waste water particles depositing; these include fibres, greases and sludge, and heavy material such as stone and chips. To prevent this, it is necessary to provide drain and flushing connections at suitable low points which allow the piping to be flushed and drained at regular intervals. Drain and flushing connections must have a shut-off facility, be located in frost-free and easily accessible shaft structures and be fitted with suitable connections for e.g. suction vehicle hoses.
- Inspection holes must be provided at suitable and easily accessible points, e.g. for pipe camera inspection and pipe pigging.

To allow for differential settlement at pumping station/piping interfaces, an articulated pipe joint may be required at the transition from the inner to the outer piping, i.e. at the point immediately upstream of the pumping station's outer wall (Fig. 57).

## Pipe fasteners / brackets

Pipes can be fastened or held in place using the following means:
- Double pipe clamps (Fig. 58)
- Single pipe clamp (Fig. 59)
- Floor support (Fig. 60)
- Wall bracket (Fig. 61)
- Suspended from the ceiling (Fig. 62)
- Saddle bearing (Fig. 63)
- Special structures.

The static piping calculation should be taken into account when selecting fixed or sliding supports.



Fig. 59: Single clamp



Fig. 60: Floor support



Fig. 61: Wall bracket

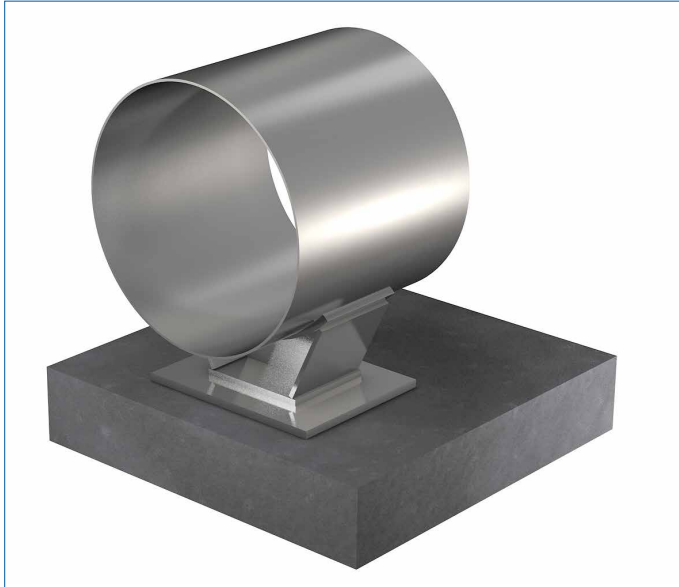

Fig. 62: Suspended from the ceiling

Fig. 63: Saddle bearing

are specially fitted with reinforcing ribs to handle the weight of the pipes, valves and the fluid.

Pipe brackets should be stable and positioned such that the permissible pipe support spacing is not exceeded. The fasteners should be suitable for the pipe and fluid weight. They should also prevent any impermissible loads resulting from additional forces and moments (e.g. wind and snow on pipes laid outdoors or earth loads on buried pipes) as well as any vibrations at the connecting points.

The permissible pipe bracket spacing depends on the permissible deflection and the stresses and loads the material is subjected to. A permissible load is defined as

$$2 \text{ mm} \leq F_{perm} \leq \frac{Da}{60} \leq 10 \text{ mm} \qquad \text{Formula (15)}$$

The following formula allows the pipe support spacing to be roughly calculated:

$$Ls = 16 \times \sqrt{D} \qquad \text{Formula (16)}$$

Key

Ls = Pipe support spacing [m]

D = Pipe diameter [m]

## Fasteners / brackets of pipes in pumping stations with vertical wet-installed KRT submersible motor pumps (Fig. 64)

The vertical riser is directly and tightly connected to the duck-foot bend's flange via a flanged connection. Duckfoot bends are used as anchorage points within the load limits specified (see Table 13 and Fig. 65, Permissible flange loads). Duckfoot bends

| Flange diameter | Forces [N] | | | | Moments [Nm] | | | |
|---|---|---|---|---|---|---|---|---|
| | $F_y$ | $F_z$ | $F_x$ | $\sum F$ | $M_y$ | $M_z$ | $M_z$ | $\sum M$ |
| 50/65 | 1350 | 1650 | 1500 | 2600 | 1000 | 1150 | 1400 | 2050 |
| 80 | 2050 | 2500 | 2250 | 3950 | 1150 | 1300 | 1600 | 2350 |
| 100 | 2700 | 3350 | 3000 | 5250 | 1250 | 1450 | 1750 | 2600 |
| 150 | 4050 | 5000 | 4500 | 7850 | 1750 | 2050 | 2500 | 3650 |
| 200 | 6000 | 5400 | 6700 | 10450 | 3250 | 2300 | 2650 | 4800 |
| 250 | 7450 | 6750 | 8350 | 13050 | 4450 | 3150 | 3650 | 6550 |
| 300 | 8050 | 10000 | 8950 | 15650 | 4300 | 4950 | 6050 | 8900 |
| 350 | 10450 | 9400 | 11650 | 18250 | 7750 | 5500 | 6350 | 11400 |
| 400 | 11950 | 10750 | 13300 | 20850 | 9700 | 6900 | 7950 | 14300 |
| 500 | 14950 | 13450 | 16600 | 26050 | 14450 | 10250 | 11800 | 21300 |
| 600 | 17950 | 16150 | 19900 | 31250 | 20200 | 14400 | 16600 | 29900 |
| 700 | 20800 | 19100 | 22500 | 36600 | 25700 | 17200 | 21000 | 37300 |

Table 13: Permissible flange loads

Impulses generated by the vanes of the rotating impeller in the pump casing cause the discharge line to vibrate. If the excitation frequency of the pump coincides with the natural frequency of the piping, the result will be resonance. Resonance leads to maximum vibration amplitudes and extreme forces acting on the bearings (supports). To rule out resonance definitively, it must be ensured that these two frequencies differ from each other. The minimum difference between the excitation frequency of the pump and the natural frequency of the piping must be higher than 10 % of the value for the excitation frequency.
The excitation frequency cannot usually be changed (e.g. by changing the pump speed). For this reason, it is the natural frequency of the pipe which has to be adjusted. The natural frequency of the piping depends on the mass distribution within the system, i.e. on valve positioning, pipe wall thickness and material, and bracket placement. A well-thought-out fastening scheme will help prevent unfavourable natural frequencies of the piping. The natural frequencies of the piping can be adjusted by changing the position of or adding individual bearings (supports), preferably near valves, outlets, etc. The positioning of valves also has an impact on the natural frequency.

The excitation frequency depends on the speed and the number of impeller vanes and is calculated as follows:

$$EF = \frac{n}{60} \times z \qquad \text{Formula (17)}$$

**Key**

EF = Excitation frequency [Hz]

n = Speed of impeller [rpm]

z = Number of impeller vanes

The bearing (support) positions and a pipe's natural frequency should be determined in the planning phase using various simulation or calculation programs, e.g. via the finite element method (FEM). The natural pipe frequency can also be influenced to a limited extent by varying the wall thickness. Thicker walls cause higher natural frequencies than thinner walls, assuming the same pipe material. Hammer excitation tests followed by vibration measurement are conducted to determine the natural frequency of existing pipes.
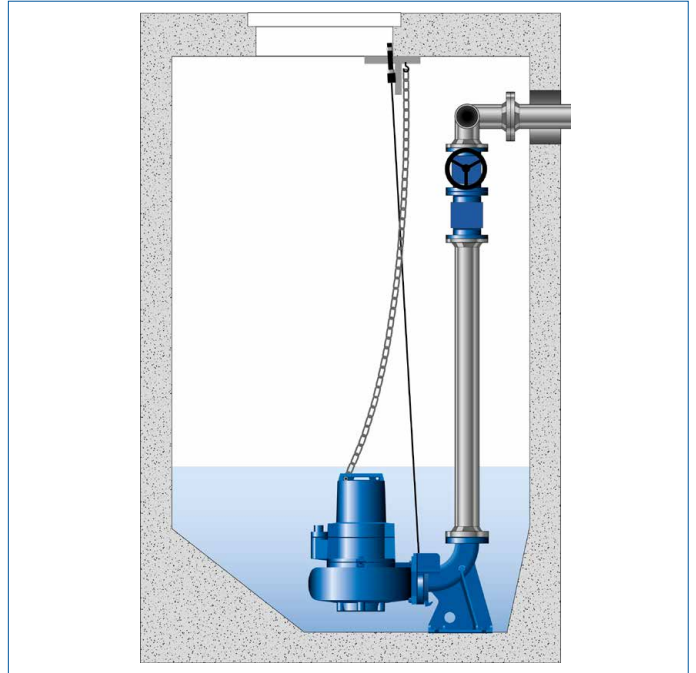


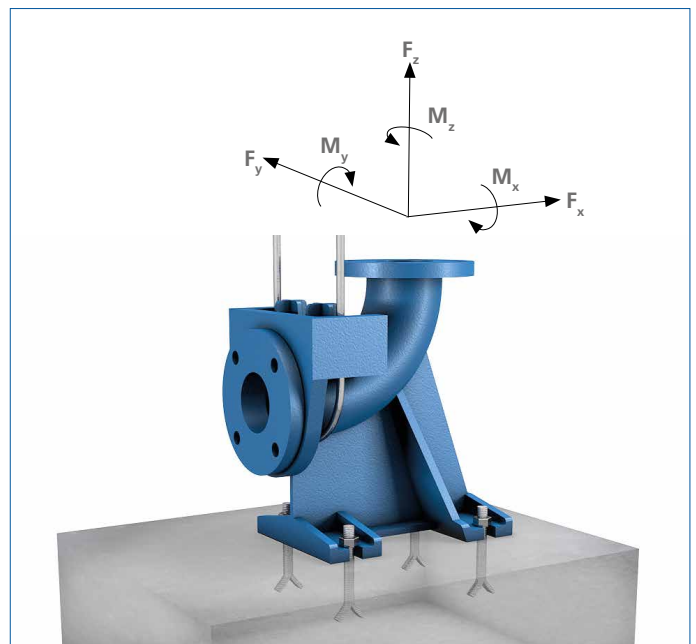Fig. 64: An example of a KRT pumping station
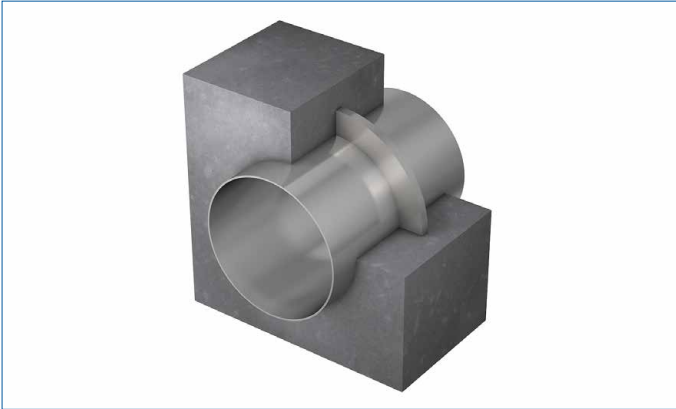


Fig. 65: Permissible flange loads
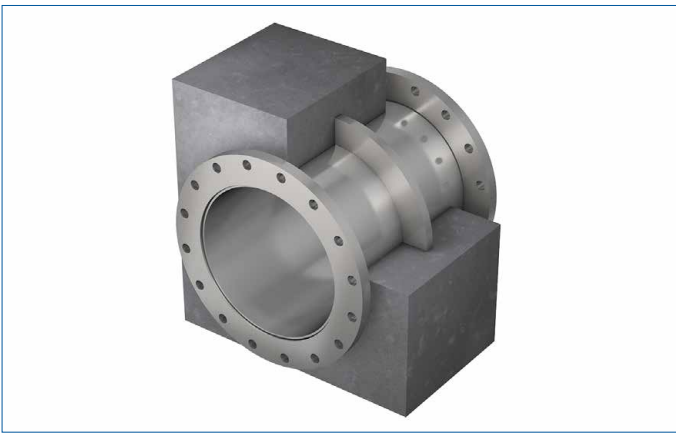
Fig. 66: Wall duct with weld ends



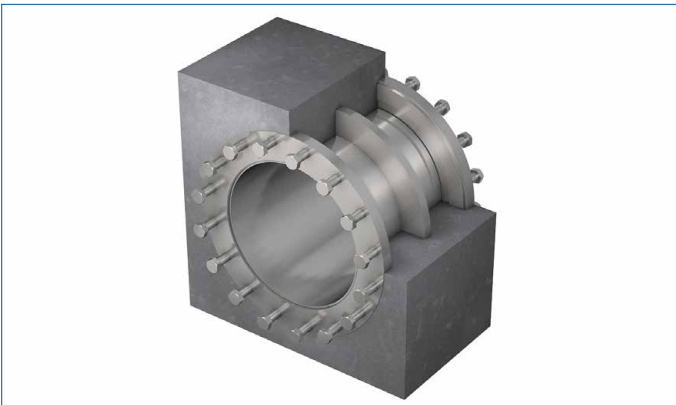Fig. 67: Wall duct with protruding flanges



Fig. 68: Wall duct with flange fitted flush with the wall



Fig. 69: Wall duct, flexible, with modular seal and pipe sleeve

## Wall ducts

If pipes are to be laid through internal and external walls of the structure, one of the following two types of wall ducts must be used. A distinction is made between the following two types:

- Rigid wall ducts
- Flexible wall ducts

### Rigid wall ducts

Rigid wall ducts consist of the pipe transporting the fluid (wall pipe) and a puddle flange which is welded onto the pipe in the centre of the wall penetration and integrated into the wall.
This type of wall duct serves as an anchorage point. The forces at this point should be determined by means of a static piping calculation and verified against the permissible forces for the structure. Rigid wall ducts can be designed as either:

- Wall pipe with weld ends (Fig. 66)
- Wall pipe with flanged ends (Fig. 67, Fig. 68)

Wall pipes with weld ends need to protrude from the wall cladding. In special cases, wall pipes can be retrofitted in a wall penetration. In this case, they must be grouted with a second concrete pour. If this option is chosen, the structural details should be discussed with the construction consultant.
Wall pipes with flanged ends can either protrude from the wall (Fig. 67) or be flush with the wall (Fig. 68).
When wall pipes are fitted flush with the wall, their installation between wall reinforcement and wall cladding requires utmost precision with regard to the height of the pipe axis and the horizontal alignment of the pipes.

### Flexible wall ducts

A flexible wall duct consists of a pipe sleeve with puddle flange; the pipe containing the fluid handled is guided through this flexible wall duct (Fig. 69). The space between the pipe sleeve and the fluid pipe can generally be sealed by either:

- Modular seals (Fig. 70 and Fig. 71)
- Compression-type gasket inserts (Fig. 72)

Instead of using a pipe sleeve, a core hole can be drilled into the reinforced concrete wall. Core holes are generally not used for larger diameters. If axial thrust has to be prevented when using flexible wall ducts, a pipe bearing/bracket must be fitted as an anchorage point (axial thrust protection) in a suitable position upstream of the wall duct (Fig. 73 and Fig. 74).

# Pipe materials

The pipes within the pumping station should preferably be made of steel. With a view to ensuring corrosion resistance, the steel pipes are designed with thick walls and either coated (e.g. hot-dip galvanised or a coating system) or made of stainless steel (material number 1.4571/V4A).

When using cast pipes, the availability of fittings and the weight must be taken into account.

If other materials such as plastic are used, particularly in industrial waste water applications, special attention should be paid to providing sufficient pipe fasteners and separate supports for fittings such as valves.

Possible pipe materials for the inner piping:
- Steel (e.g. coated or galvanised)
- Stainless steel (e.g. 1.4301 or 1.4571)
- PE-HD
- Cast (bitumen-coated or with electrostatic plastic coating)

The selection of material for pipes outside of the pumping station depends on the local conditions (ground, corrosion), design criteria for the structure and piping technology, and economic considerations.

Possible pipe materials for the outer piping:
- Cast (bitumen-coated or with electrostatic plastic coating, lined with cement mortar)
- PE-HD
- GFRP
- Bitumen-coated steel
- Steel lined with cement mortar

**A technically and dimensionally matching joint/adapter must be fitted between the inner and the outer piping of the pumping station.**
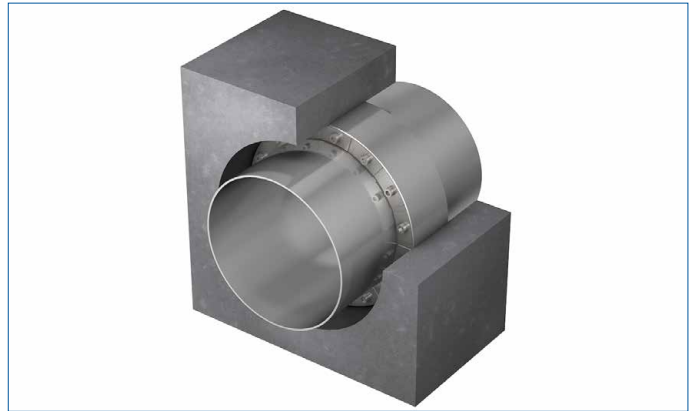


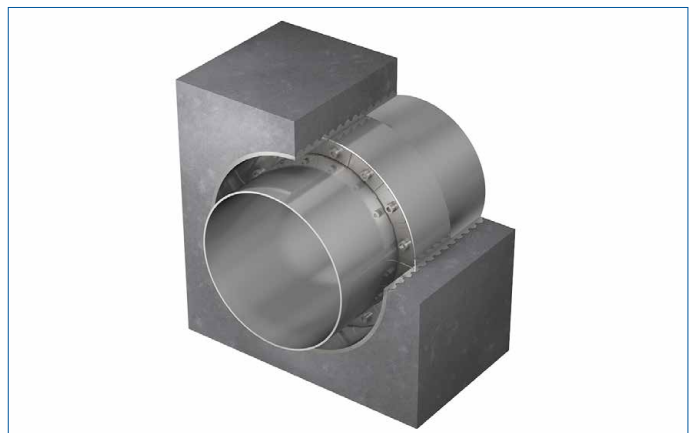Fig. 70: Wall duct, flexible, with modular seal and core hole



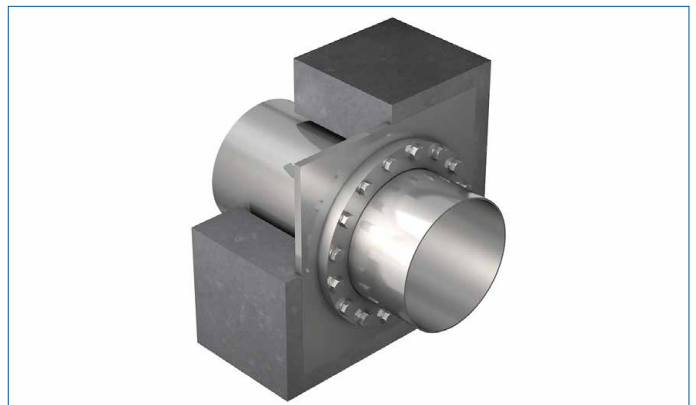Fig. 71: Wall duct, flexible, with modular seal and pipe sleeve



Fig. 72: Wall duct, flexible, with pipe sleeve and compression-type gasket insert



Fig. 73: Axial thrust protection as anchorage point

Fig. 74: Axial thrust protection as anchorage point



Downstream
stabilisation
distance
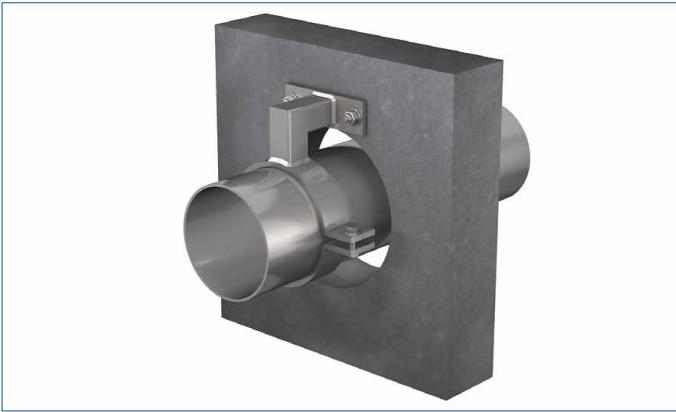
Upstream
stabilisation
distance
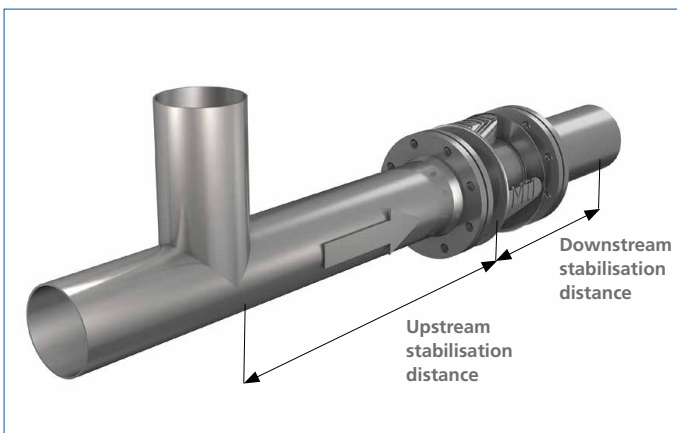
Fig. 75: MID fitted in the pipe with upstream and downstream stabilisation
distance



Downstream
stabilisation
distance

Upstream
stabilisation
distance
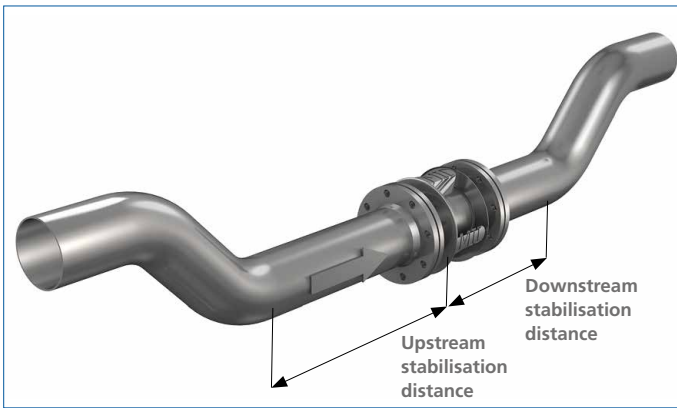
Fig. 76: MID fitted in the inverted siphon pipe with upstream and
downstream stabilisation distance



Fig. 77: Pressure measurement in horizontal pipe

# Fitting measuring devices in the piping

Measuring devices must be fitted in the discharge lines (usually the inner piping) in accordance with the monitoring and control concept.

### Devices fitted directly in the piping
The only measuring devices which are generally fitted in the pipes are
- Magnetic-inductive flow meters (MID)

### Devices fitted at or integrated in the pipes
Devices which are fitted at the pipes measure the following:
- Pressure (pressure gauge or transmitter)
- Flow (flow switch as dry running protection)
- Temperature (uncommon in waste water lines)
- Flow measurements using ultrasonic sensors

### Note
At this point we would like to mention that in practice additional measurements are required for controlling the pumping station, such as the fluid level in the pump sump and possibly also at the outlet. These measurements are not linked with the piping system.

### Notes on the position of MID measuring points
When fitting or positioning MIDs:
- Provide sufficient upstream and downstream flow stabilisation distances. Observe the information provided by the corresponding manufacturer (Fig. 75).
- Select the installation position in accordance with the manufacturer's instructions. For example, provide inverted siphons to completely fill the measuring section of devices which are unsuitable for partially filled pipes (Fig. 76).

### Notes on the position of measurement points for pressure measurement, flow switches and ultrasonic measurement
Always select pressure measurement points on the sides of the pipe at the level of the pipe axis (Fig. 77 and Fig. 78). Measurement points should always be located in pipe sections with a stable flow. Avoid measurement points at reducers, adapters, diversions, fittings, etc.
Also avoid pipe inverts or crowns as contamination and entrapped air can lead to inaccurate results.

# Valve selection

Valves are a functional part of the piping system and are required to implement the pumping process. They perform the following key functions:

- Shutting off and opening the pipe passage
- Preventing backflow
- Controlling the flow
- Admitting air to and venting air from the piping



Fig. 78: Pressure measurement in vertical pipe

The valve industry offers the following valve types for the above functions:

- Gate valves (wedge gate valves, knife gate valves), butterfly valves, globe valves
- Control gate valves (piston valves, spectacle valves, knife gate valves)
- Swing check valves (with lever and weight or with internally mounted hinge pin), check valves (with diaphragm or discs), ball check valves
- Air valves of various designs

## Selection criteria

Key criteria for selecting a valve:

- Fluid handled
- Compatibility of design and function with the fluid handled
- Compatibility of materials with the fluid handled
- Nominal size depending on the flow velocity and the resulting head losses

## Fluids handled

The fluid handled is divided into the following (waste) water types which place special demands on valves:

- Stormwater
- Pre-screened surface water
- Industrial waste water
- Grey water without any stringy substances
- Domestic waste water with stringy substances
- Raw sludge
- Return sludge
- Excess sludge
- Clean waste water without contaminants
- Service water

See table "Suitability of valves for different waste water types" (Table 14) for the selection of valves.

## Valve design requirements

The use of valves in the above fluids can require designs which are suitable for handling fluids containing corrosive and/or abrasive substances, stringy or clogging material, or a mixture of both.

Fluids containing stringy or clogging material, such as domestic waste water, require valve designs which:

- Keep the flow cross-section in the valve as clear as possible
- Prevent clogging when the valve is actuated
- Feature seals with suitable design and materials

## Installation position and direction of flow

The direction of flow and the installation position often depend on the particular design of a valve. In the case of swing check valves and all other check valves, the design dictates the direction of flow.

In many cases, valves are designed for either vertical or horizontal installation.

**Observe the manufacturer's information (e.g. installation and operating instructions) right from the planning stage.**

## Materials

Suitable materials should be selected based on the fluids involved. Materials are usually indicated by the valve manufacturers for each part, such as body, valve disc, seat, seal, shaft / stem, nuts and bolts/screws, etc.

Coated grey cast iron is suitable for surface water / stormwater and municipal waste water. Electrostatic plastic coating can be considered particularly suitable.
Highly abrasive fluids require hard cast materials with a special coating.

Care must also be taken to select suitable elastomers for the seals. EPDM and NBR are generally suitable for municipal waste water. For industrial waste water, Viton (FPM) may be necessary.

Some industrial waste water may require the use of stainless steel valves. Materials for use with industrial waste water should always be selected on a case-by-case basis according to the information provided on substances contained in the fluid handled.

Valve manufacturers or distributors should always be informed of the substances contained in the fluid handled so they can match the valves to the operating conditions.

## Nominal size

As with pipes, the nominal size of valves depends on the flow velocities (see section "Dimensioning the piping", page 51). Usually, pipes and valves have matching nominal sizes.

When selecting the nominal size for a valve, particular attention should be paid to the head losses through the valve.
Swing check valves in particular can have such high loss coefficients that the next size up should be selected for the nominal valve size. The pipe selection has to be adjusted accordingly.

# Valve installation

## Installation position

For the direction of flow and the suitability for vertical and/or horizontal installation, refer to section "Installation position and direction of flow", page 64.

## Installation position

The installation position is determined by:

- The function of the valve
- Functional framework conditions
- Accessibility and operability

These three criteria are of equal importance when selecting an installation position for a valve. They must all be taken into account when determining the best solution for a given project.

**Function of the valve**

A valve's installation position and its location in the piping are determined by the valve's operational function. Information on installation positions and location in the piping is typically found in the manufacturer's operating and installation instructions. System design requirements may also have an influence on the valves' installation position and location in the piping. However, they should not contradict the manufacturers' information.

The following examples serve to illustrate this point:

- The gate valve at the discharge side of the pump serves to shut off the pipe if the pump or swing check valve need to be repaired. It should ideally be located immediately downstream of the pump and swing check valve. However, functional framework conditions (max. fluid level in the pump sump) and the operability may require a different position.

- Swing check valves serve to automatically prevent backflow and are mandatory in pumping stations with multiple pump configurations. They should ideally be located directly downstream of the pump, as mentioned above. This is possible if the fluid handled is not contaminated. In this case, the fluid level and accessibility dictate the location of the valve in the piping. If submersible motor pumps are installed, the maximum water level should never be above the level of the swing check valve. One reason for this is that the absence of back pressure from the water column or weight at the swing check valve could allow static pressure in the pump sump to open the swing check valve and cause fluid to flow through. Another reason is ensuring automatic venting of the pump following re-installation in the filled pump sump as the air in the pump can be discharged into the emptied discharge line as far as the swing check valve and compressed by the static pressure of the water level in the sump. Swing check valves used in waste water applications require special attention. If the discharge line has a long vertical section, the swing check valve must not be installed low down in the pipe. During pump standstill, solid matter in the waste water, e.g. stones, sand, sludge etc. may sink and deposit on the valve disc. Stones falling back down could also damage the valve disc and impair the valve's function, potentially blocking the valve disc. A swing check valve should therefore be fitted upstream of the gate valve at the highest possible point in the waste water discharge line. Should this not be possible for reasons of accessibility, the swing check valve must be installed in a horizontal pipe section.

If necessary, a horizontal pipe section must be planned for in the piping layout, either inside the pumping station or – if too little space is available – in a separate, external valve pit.

- Air valves must always be installed at pipe high points. Accessibility must be ensured with due consideration of the station's construction and layout. The air valves must be installed in a frost-proof location inside buildings to ensure safe operation in winter. If necessary, the discharge line and the air valve must be fitted with trace heating.

**Functional framework conditions**

The installation position can be determined by functional framework conditions. For example:

- Installation criteria provided by the valve manufacturer (vertical/horizontal)
- Routing of the discharge line
- If the discharge line has a long vertical section, the swing check valve must not be installed low down in the vertical pipe axis as its function will be impaired by contaminants being deposited on it (sand, stones, sludge). Stones falling back down could also cause some direct damage. In this case, the swing check valve should be fitted in a horizontal pipe section which has to be planned for in the piping layout.
- Maximum fluid level
- Individual discharge lines joining collecting lines

**Accessibility and operability**

Accessibility for operating staff is an essential factor in ensuring operability and access for maintenance and repair work. Applicable occupational safety and accident prevention regulations need to be considered at the planning stage.

Operability and accessibility can generally be achieved as follows:
- The valve should be located in a part of the structure which is already easily accessible.
- Ladders and platforms should be installed where required.
- The access structure should be designed to ensure easy operability and maintenance.
- For ease of operation, shut-off valves are fitted with extensions, e.g. deck stands with telescopic rods or electric actuators. Additional measures must also be taken to facilitate access for maintenance and repair work, e.g. the provision of maintenance platforms.

Suitable structural solutions for accessibility and operability:
- **Valve pits**
  A prefabricated pit can be provided for the valves so that they are separated from the pump sump. Correct positioning of the access covers can facilitate straightforward installation and replacement.
- **Valve chambers**
  Valve chambers are a practical solution for large pumping stations with large nominal pipe and valve sizes. Valve chambers provide favourable working conditions as they accommodate the complete piping including individual and collecting (manifold) lines as well as all valves and measuring equipment.

# Technical solutions for installing and removing valves

The initial design should provide ease of installation and possible later removal plus maintenance. The installation of expansion and dismantling joints next to valves or groups of valves should be considered.

## Installation without expansion and dismantling joints

Expansion and dismantling joints are generally not required if pipe bends are included in the piping upstream and/or downstream of the valve or valve group. Dismantling the pipe bend will make the corresponding pipe section accessible for valve replacement.

## Installation with expansion and dismantling joints

In many cases, the piping layout obstructs the fitting and dismantling of pipe elements and valves. If this is the case, expansion and dismantling joints should be installed next to the valve or valve group. Expansion and dismantling joints come with a length adjustment range to enable fitting and dismantling of the valve or valve group without transmitting any stresses and strains.

The main types of joints are:
- Expansion and dismantling joints with lockable length adjustment- This type of expansion and dismantling joint can generally be supplied:
  – with continuous threaded rods (Fig. 79)
  – with discontinuous threaded rods (Fig. 80)
- Expansion and dismantling joints without lockable length adjustment (Fig. 81)

Expansion and dismantling joints with discontinuous threaded rods are particularly installation-friendly as the joint length is very easy to adjust. Expansion and dismantling joints without lockable length adjustment can be used as expansion joints as well as dismantling joints. They do not transmit forces into the pipe.



Fig. 79: Expansion and dismantling joint with lockable length adjustment and continuous threaded rods



Fig. 80: Expansion and dismantling joint with lockable length adjustment and discontinuous threaded rods



Fig. 81: Expansion and dismantling joint without lockable length adjustment

Fig. 82: Axial-restraint pipe coupling



Fig. 83: Expansion and dismantling joint with axial-restraint pipe coupling
       and continuous threaded rods

**Pipe coupling as a fitting/dismantling aid (Fig. 82)**
Using a pipe coupling might be a straightforward solution. A
pipe coupling is a steel collar which connects two pipe ends with
a small clearance. For small to medium nominal diameters, the
clearance can be sufficient to allow fitting or dismantling a pipe
section and to replace valves. Pipe couplings are available as
axial-restraint and non-axial-restraint models. An application
example is the axial-restraint expansion and dismantling joint
with two flanges and continuous threaded rods. (Fig. 83)

**Suitability of valves for different waste water types**

| Valve design | Stormwater, pre-screened surface water | Industrial waste water, grey water without any stringy substances | Domestic waste water with stringy substances, raw sludge | Return sludge, excess sludge | Clean water without contamination, service water |
|---|---|---|---|---|---|
| **Shut-off valves** | | | | | |
| **Ball and plug valves** | | | | | |
| Ball valve with reduced port | Partly suitable | Suitable | Not suitable | Partly suitable | Suitable |
| Ball valve with full port | Suitable | Suitable | Suitable | Suitable | Suitable |
| Drain valve | Partly suitable | Suitable | Not suitable | Partly suitable | Suitable |
| Ball/plug valve with tapered seat | Partly suitable | Suitable | Not suitable | Partly suitable | Suitable |
| Ball/plug valve with cylindrical seat | Partly suitable | Suitable | Not suitable | Partly suitable | Suitable |
| **Globe valves** | | | | | |
| Horizontal seat globe valve | Not suitable | Partly suitable | Not suitable | Not suitable | Suitable |
| Slanted seat globe valve | Not suitable | Partly suitable | Not suitable | Not suitable | Suitable |
| Angle globe valve | Not suitable | Partly suitable | Not suitable | Not suitable | Suitable |
| Plunger valve | Not suitable | Partly suitable | Not suitable | Not suitable | Suitable |
| **Metal-seated gate valves** | | | | | |
| Round body gate valve | Partly suitable | Suitable | Partly suitable | Partly suitable | Suitable |
| Oval body gate valve | Partly suitable | Suitable | Partly suitable | Partly suitable | Suitable |
| Slide disc valve | Partly suitable | Suitable | Partly suitable | Partly suitable | Suitable |
| Wedge gate valve | Partly suitable | Suitable | Partly suitable | Partly suitable | Suitable |
| **Soft-seated gate valves** | | | | | |
| Wedge gate valve | Suitable | Suitable | Suitable | Suitable | Suitable |
| Knife gate valve | Suitable | Suitable | Suitable | Suitable | Suitable |
| Penstock valve (sluice gate valve) | Suitable | Suitable | Suitable | Suitable | Suitable |
| **Butterfly valves** | | | | | |
| Centred-disc butterfly valve | Suitable | Suitable | Not suitable | Partly suitable | Suitable |
| Offset-disc butterfly valve | Suitable | Suitable | Not suitable | Partly suitable | Suitable |
| **Diaphragm shut-off valves** | | | | | |
| Straight-through diaphragm valves | Suitable | Suitable | Partly suitable | Suitable | Suitable |
| Weir-type diaphragm valves | Not suitable | Not suitable | Not suitable | Not suitable | Suitable |
| Diaphragm pinch valve (hydraulic or pneumatic) | Suitable | Suitable | Suitable | Suitable | Suitable |
| Diaphragm valve with annular diaphragm | Not suitable | Not suitable | Not suitable | Not suitable | Suitable |

Key:

■ Not suitable
■ Partly suitable
■ Suitable

## Suitability of valves for different waste water types

**Valve design** — **Waste water type**

| Valve design | Stormwater, pre-screened surface water | Industrial waste water, grey water without any stringy substances | Domestic waste water with stringy substances, raw sludge | Return sludge, excess sludge | Clean water without contamination, service water |
|---|---|---|---|---|---|
| **Check valves** | | | | | |
| **Lift check valves** | | | | | |
| Lift check valve, spring-loaded | Not suitable | Not suitable | Not suitable | Not suitable | Suitable |
| Foot valve with inlet and suction strainer | Suitable | Suitable | Not suitable | Not suitable | Suitable |
| Nozzle check valve | Not suitable | Not suitable | Not suitable | Not suitable | Suitable |
| **Swing check valves** | | | | | |
| Swing check valve with/without lever and weight | Suitable | Suitable | Suitable | Suitable | Suitable |
| Multiple swing check valve | Partly suitable | Suitable | Suitable | Not suitable | Suitable |
| Slanted-seat tilting swing check valve with/without lever and weight | Partly suitable | Suitable | Not suitable | Suitable | Suitable |
| Check valve | Suitable | Suitable | Suitable | Suitable | Suitable |
| Ball check valve | Suitable | Suitable | Suitable | Suitable | Suitable |
| Dual-plate check valve | Not suitable | Suitable | Suitable | Not suitable | Suitable |
| Wafer-type swing check valve | Partly suitable | Suitable | Suitable | Suitable | Suitable |
| **Diaphragm check valves** | | | | | |
| Nozzle check valve | Not suitable | Partly suitable | Not suitable | Not suitable | Suitable |
| Diaphragm check valve | Not suitable | Partly suitable | Not suitable | Not suitable | Suitable |
| **Other valves** | | | | | |
| Discharge valve with float switch | Not suitable | Partly suitable | Not suitable | Not suitable | Suitable |
| Air valve | Suitable | Suitable | Not suitable | Not suitable | Suitable |
| Waste water compatible air valve | Suitable | Suitable | Suitable | Suitable | Suitable |
| Safety valve | Partly suitable | Suitable | Not suitable | Not suitable | Suitable |
| Minimum flow valves | Not suitable | Suitable | Suitable | Suitable | Suitable |
| Discharge and end-fitting valve, backflow flap | Suitable | Suitable | Not suitable | Not suitable | Suitable |

**Key:**

- Not suitable (red)
- Partly suitable (yellow)
- Suitable (green)

Table 14: "Suitability of valves for different waste water types"

# Structural Design

The structural requirements of a pumping station are largely determined by its purpose. Alongside purely structural and mechanical requirements, consideration must also be given to hydraulic aspects (fluid dynamics) in the planning and execution of the construction work. The first part of the hydraulic areas to be designed is the inlet upstream of the pumping station, followed by the pump sump area upstream of the pump(s), parts of which may require a special shape, and finally the discharge line or discharge system.

The pump manufacturers' aim is to specify in the product's technical literature the dimensions (e.g. geometry of the building) required for the installation of the centrifugal pumps. The reference values provided here are essential for the planning process, in particular for establishing the main dimensions of the pumping station. Successful planning of a pumping station is a complex task. Apart from specifications such as the minimum clearance between pumps and between pump and floor, or the floor contour, the planning phase also encompasses the design of the area between the intake and the pump(s).

If the requirements regarding intake chamber dimensions, minimum water levels, or the geometry of hydraulic areas within the pumping station are not met, i. e. deviations occur during the planning or construction phase, proper functioning of the entire station can no longer be guaranteed. In such a case, it is irrelevant whether these problems are caused by single or multiple deviations. These modifications or deviations will mean that the conditions required for pump operation are not met and the resultant problems will be reflected in either the operating behaviour or the performance of the centrifugal pump(s). This can be avoided by incorporating the pump manufacturer's hydraulic and mechanical design specifications in the pumping station's design process at an early stage to ensure that the required performance data is achieved and operating malfunctions are avoided.

According to Prosser [5.1], the criteria for unsatisfactory pumping station design can be clearly defined and assessed.

The following aspects of geometrical design can lead to poor sump performance:
1. Undersized control gates and valves
2. Abrupt changes in flow direction (e.g. sharp corners)
3. Submerged high-velocity flow areas (e.g. diffusers with an excessive angle of divergence)
4. Stepped sump floor
5. Weirs with no provision for dissipating the energy of the falling fluid
6. Blunt pillars, piers and guide vanes
7. Any design or mode of operation which leads to asymmetric distribution of the flow in the sump
8. Sump inlet above water level

Items 1, 2, 3, 6 and 7 may cause swirls at the pump's inlet. Air-entraining surface vortices and submerged vortices may form in extreme cases (Fig. 106 and Fig. 107). Items 4, 5 and 8 can lead to air intake in the fluid handled, and items 3, 4 and 5 may cause transient flow conditions within the sump.

The purpose of a pump sump is to provide stored volume and favourable conditions for the flow approaching the pump. The following undesirable hydraulic conditions must be avoided:
- Jets (high-velocity flows) discharging into stagnant or slowly moving fluids (as these form large, transient eddies as downstream wakes)
- Areas of separated flow
- High-velocity flows (v > 2 m/s)
- Transient flow
- Large surface waves
- Free-falling fluids

Observing the above criteria at the planning and construction stage is an important step towards creating a smoothly functioning pumping station.

KSB's geometries compare favourably with those specified by other pump manufacturers and internationally recognised research institutions in their literature. The geometries suggested by KSB in their type series booklets and software tools allow smaller pumping stations to be built and therefore reduce building costs.
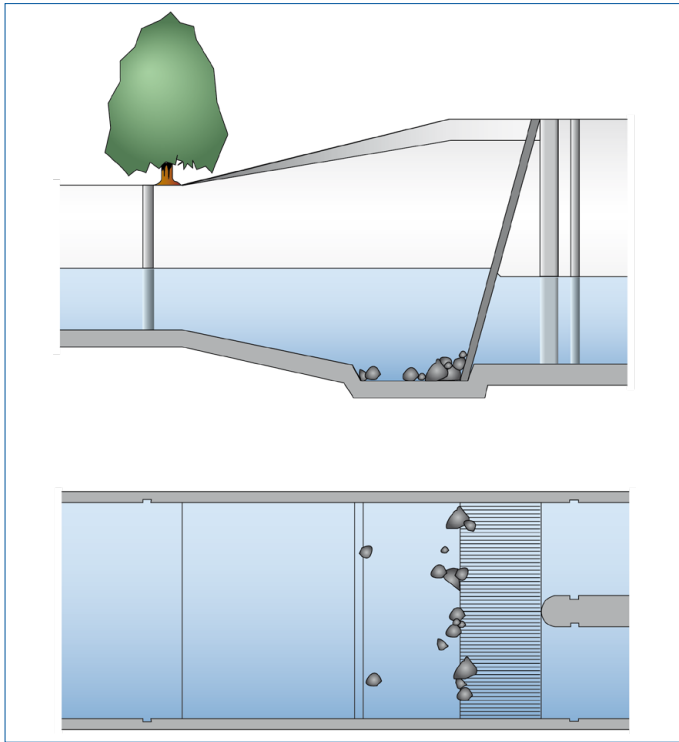
Fig. 84: Screen with automatic cleaning system



Fig. 85: Shingle trap upstream of pumping station

## Screening equipment

The installation of screening equipment is required for trouble-free pump operation: Depending on the type and origin of the fluid handled, it is advisable to install coarse screens (bars spaced between 5 and 30 cm apart) (Fig. 84) and/or fine screens (bars spaced between 5 and 20 mm apart) as well as shingle traps mounted upstream of the screens if needed (Fig. 85). Screens and traps should be cleaned automatically during pump operation using appropriate mechanical equipment (Fig. 84). In stormwater pumping stations or in applications where surface water from rivers, lakes and channels is pumped, the installation of screening equipment is strongly advised.

The fact that river water in particular contains shingle and sediment is often overlooked. Under long-term operation, failure to fit the appropriate screening equipment upstream of a pumping station will lead to sand accumulation and considerable sedimentation in stagnation zones within the system, as well as to increased wear on centrifugal pumps. Mechanical damage to the impeller and other pump parts can also not be ruled out (Fig. 90).

Where the screens are to be accommodated in the pumping station layout is the designer's decision. The screening equipment can be either mounted upstream of the pumping station or sump to prevent coarse material from entering the pumping station, or single screens can be directly integrated into the intake chamber. Selecting the latter option may necessitate longer chambers due to the slightly reduced flow cross-section resulting from integrating the screen. Downstream of the screening equipment the velocity distribution developing across the flow cross-section should be even and therefore favourable for pump operation, provided the screen is largely free from any trapped material (Fig. 84). When establishing the minimum fluid level $t_1$ in the pump sump, it is also necessary to take into account that a screen filled with trapped material creates flow resistance, resulting in different fluid levels upstream and downstream of the screen. The water level downstream of the screen must not fall below the permissible minimum water level $t_1$ for the pump's operating point.

Half of the impeller's free passage should be used as a reference value to determine the permissible maximum distance between the screen bars. This value can be found in the respective pump characteristic curve (see type series booklet or selection software).

To evaluate the screen's influence on the water level directly upstream of the pumps it is advisable to use Hager's simplified calculation [10], if a detailed screen selection procedure is not being undertaken.

Applying this calculation will result in the lowering of the water level downstream of the screen $\Delta H$ as expressed in the equation:

$$\Delta H = \xi_{sc} \cdot \frac{v_0^2}{2g} \qquad \text{Formula (18)}$$

$$\text{where } \xi_{sc} = \beta_{sc} \cdot c_{sc} \cdot \sin \sigma_{sc} \cdot \zeta_{sc} \qquad \text{Formula (19)}$$

**Key:**
$\Delta H$ = Lowering of the water level downstream of the screen [m]
$\xi_{sc}$ = Loss coefficient of screen
$v_0$ = Approach flow velocity upstream of the screen [m/s]
$\beta_{sc}$ = Loss coefficient for screen bar geometry
$c_{sc}$ = Correction factor for the cleaning method and screen clogging
$\sigma_{sc}$ = Inclination angle of screen bars [°]
$\zeta_{sc}$ = Coefficient for screen geometry
$g$ = Acceleration due to gravity = 9.81 m/s²

The following values can be applied for the following different bar shapes (Fig. 87):

| Shape | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|
| $\beta_{sc}$ | 1 | 0.76 | 0.76 | 0.43 | 0.37 | 0.3 | 0.74 |

Table 15: Loss coefficient for the screen bars' geometry

$\overline{L}$ is the length of the screen bar profile and $\overline{d}$ the width. If the ratio of $\overline{L}/\overline{d}$ is approx. 5 and the condition $\overline{b}/\overline{a} > 0.5$ is satisfied, the formula (19) for $\zeta_{sc}$ can be simplified and expressed as follows:

$$\xi_{sc} = \frac{7}{3} \cdot \beta_{sc} \cdot c_{sc} \cdot \sin\sigma_{sc} \cdot \left[\frac{\overline{b}}{\overline{a}} - 1\right]^{\frac{4}{3}} \qquad \text{Formula (20)}$$

**Key:**
$\xi_{sc}$ = Loss coefficient of screen
$\beta_{sc}$ = Loss coefficient for screen bar geometry (Table 2)
$c_{sc}$ = Correction factor for the cleaning method and screen clogging
$\sigma_{sc}$ = Inclination angle of screen bars [°]

The following applies for $c_{sc}$:
$c_{sc}$ = for unobstructed screens
$c_{sc}$ = 1.1 ... 1.3 for mechanically cleaned screens
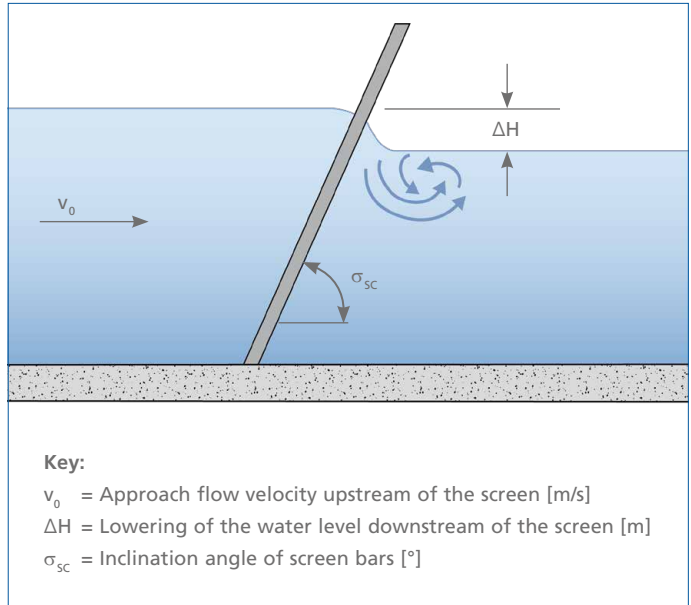$c_{sc}$ = 1.5 ... 2.0 for manually cleaned screens



**Key:**
$v_0$ = Approach flow velocity upstream of the screen [m/s]
$\Delta H$ = Lowering of the water level downstream of the screen [m]
$\sigma_{sc}$ = Inclination angle of screen bars [°]

Fig. 86: Flow through screen, without lowered floor



**Key:**
$\overline{L}$ = Length of screen bar profile [m]
$\overline{d}$ = Width of screen bar [m]
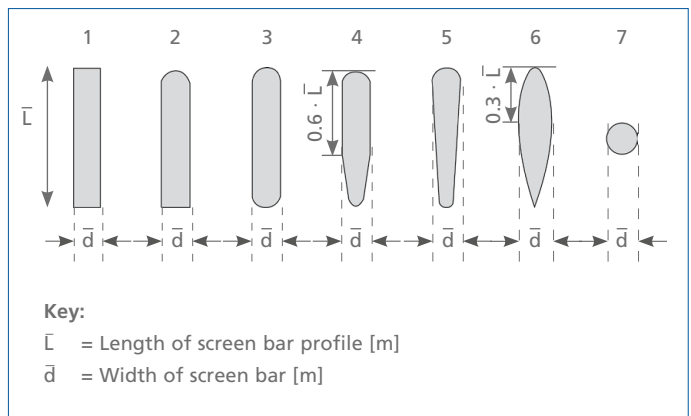
Fig. 87: Shape of screen bars



**Key:**
$v_0$ = Approach flow velocity upstream of the screen [m/s]
$\overline{a}$ = Clearance between screen bars [m]
$\overline{b}$ = Distance between screen centrelines [m]

Fig. 88: Screen layout drawing

Key:

$v_0$   = Approach flow velocity upstream of the screen [m/s]
$\Delta H$ = Lowering of the water level downstream of the screen [m]
$\Delta z$  = Lowering of the floor of the intake structure or channel [m]
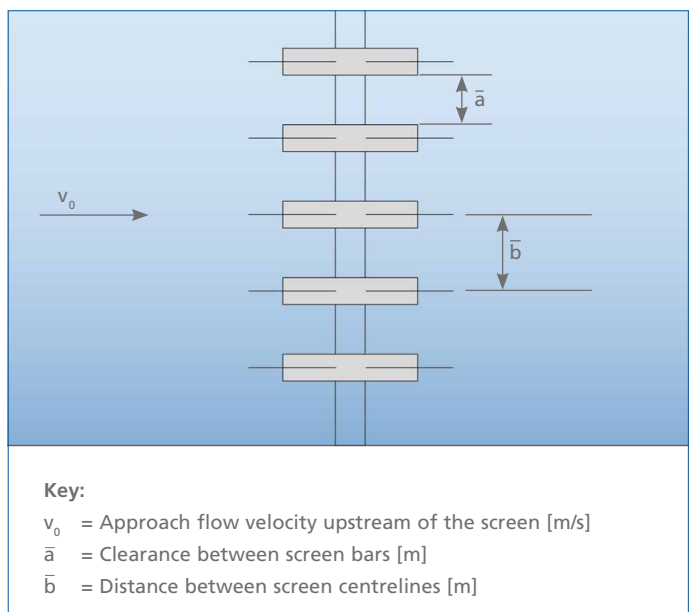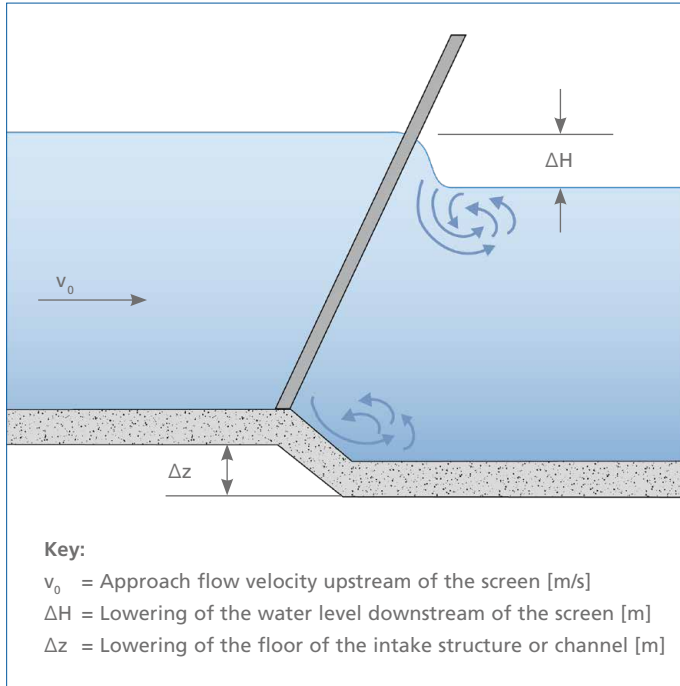
Fig. 89: Flow through screen, with lowered floor



Fig. 90: Wood inside the impeller of a KRT pump

In order to compensate for the losses $\Delta H$ occurring as the flow passes the screen (Fig. 89), the floor of the structure or channel is often lowered by the value $\Delta z$ downstream of the screen:  **$\Delta H = \Delta z$**

The values usually applied for losses through screens range between approx. 5 cm for mechanically cleaned screens to approx. 10 cm for manually cleaned screens.

For detailed screen selection, the method according to Idelchik [11, p. 504 ff] is recommended. This method is most appropriate when the influence of an oblique approach flow to the screen is also to be taken into account or if the screen bars are markedly different from those illustrated in Fig. 87.

Screens are often integrated directly into intake chambers. Each pump thus has its own screen. The distance between the screening equipment and the pump should be at least $4 \cdot D$ (D being the suction nozzle diameter).

If it is assumed that the flow might approach the screen from the side and the influence on the water level downstream of the screen might be difficult to predict, preventive measures are then advisable. Extending the intake chamber wall and positioning the screen in the intake chamber area would then achieve better defined and uniform approach flow conditions for the pump and screen.

Automatic screen cleaning is preferable (Fig. 84). The difference in fluid levels upstream and downstream of the screen can be used to activate the cleaning process. This ensures that the cleaning process is activated as required. Manual cleaning is unfavourable for pump systems in continuous operation, as the fluid level has to be regularly checked and the screen cleaned by operating staff. Timer-controlled cleaning is also not sufficiently reliable.

## Installing benching in pump sumps

As waste water and surface water usually do contain solids, the transport of these solids should also be addressed when planning a pump sump.

When the fluid leaves the inlet line, the flow velocity decreases. Depending on the velocity distribution in the structure, sedimentation and the formation of scum layers might occur. The pump or pumps cannot draw in sediment and transport it out of the structure together with the water.

If the structure has not been fitted with suitable slopes (benching), sedimentation will continue and can eventually change the flow through the structure and/or clog the pump(s). This situation can be prevented by fitting sufficiently large areas of sloping floor and/or benching (corner fillets). The angles can vary depending on the surface finish of the structure. Angles of around 60 degrees are recommended according to Worksheet ATV-DVWK-A 134. However, it should be noted that this will result in relatively high building costs as the sump will need to be much deeper for the same sump fill volume. Smaller angles are sufficient if the surfaces are coated, which reduces the required depth of the structure (see recommendations by the Hydraulic Institute 9.8 of 2018).

If the floor of the pump sump has a very flat design, consideration should be given to providing a flow gulley (possibly by using internal fittings) to ensure sufficient flushing. Varying the cross-section at some points, for example, will increase the flow velocities so that solids/sediment are transported with the fluid. Rule of thumb: Fill all areas of low flow or no flow (dead volume) with concrete to prevent deposits.

To prevent any deposits in low-flow areas, suitably designed benching can be used to close the area between the duckfoot bend of the pump and the wall of the structure (in the direction of the approach flow) (Fig. 92 and Fig. 94). When installing benching in this location, the option of later installation work around the duckfoot bend should be provided for (enough space for aligning and bolting down should be allowed). Coating the concrete contours not only improves the sliding behaviour of solids in the waste water, but also protects the structure against so-called concrete corrosion. Although this issue will not be covered in further detail in this brochure, it is of major significance.



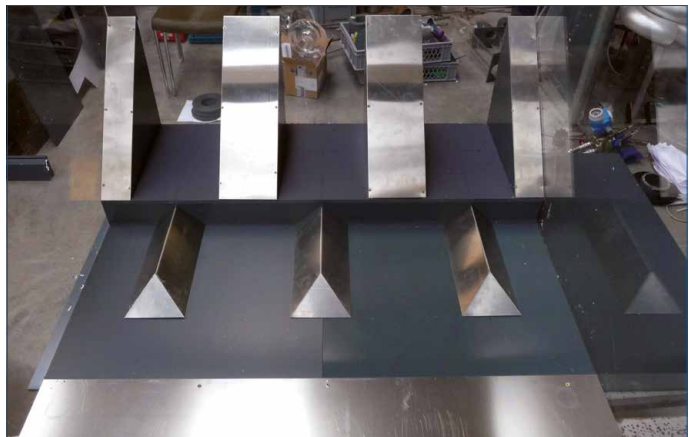Fig. 91: Formation of a layer of scum in the pump sump



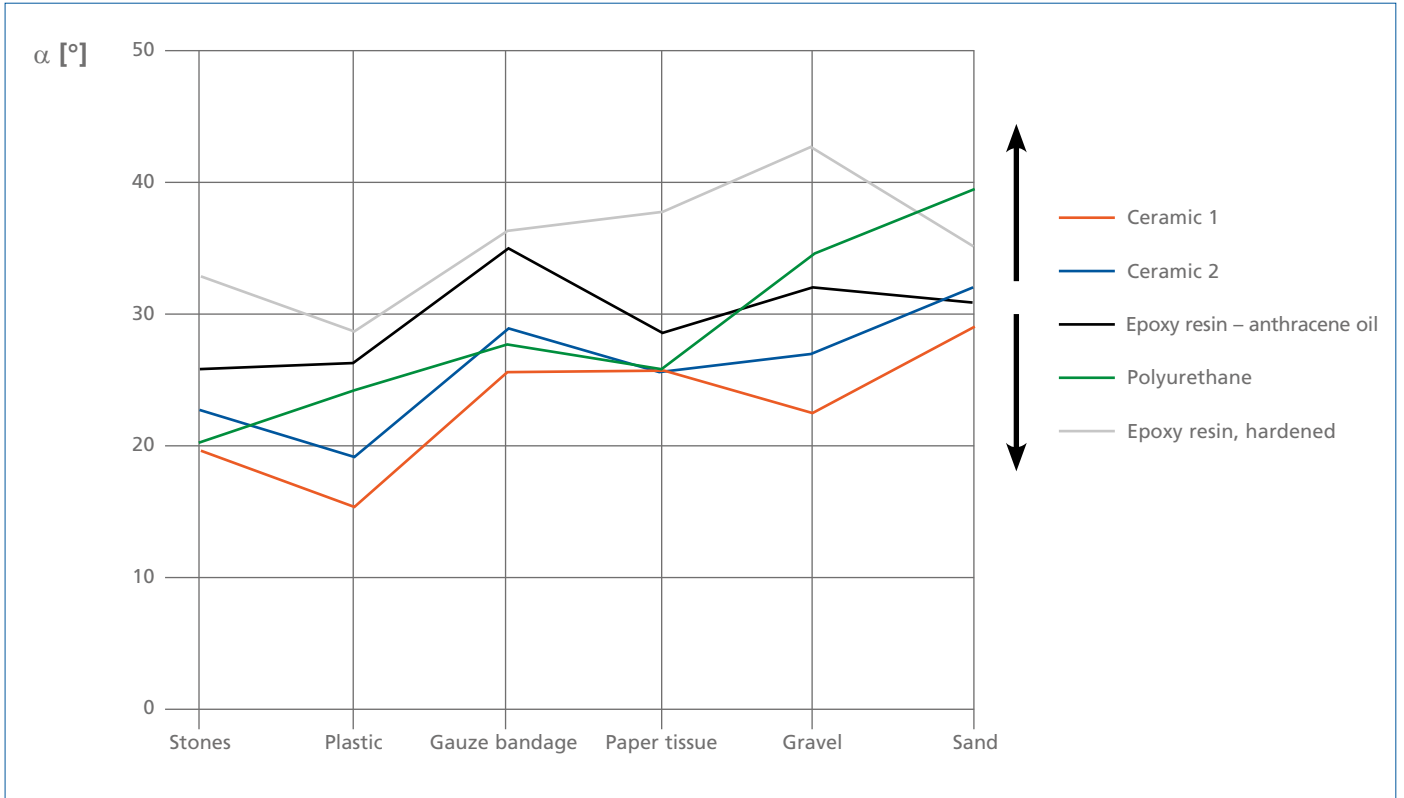Fig. 92: Model for a waste water pumping station with benching and bottom flow splitters

Fig. 93: Sliding angle of various solids contained in waste water combined with various coatings (without considering the flow)
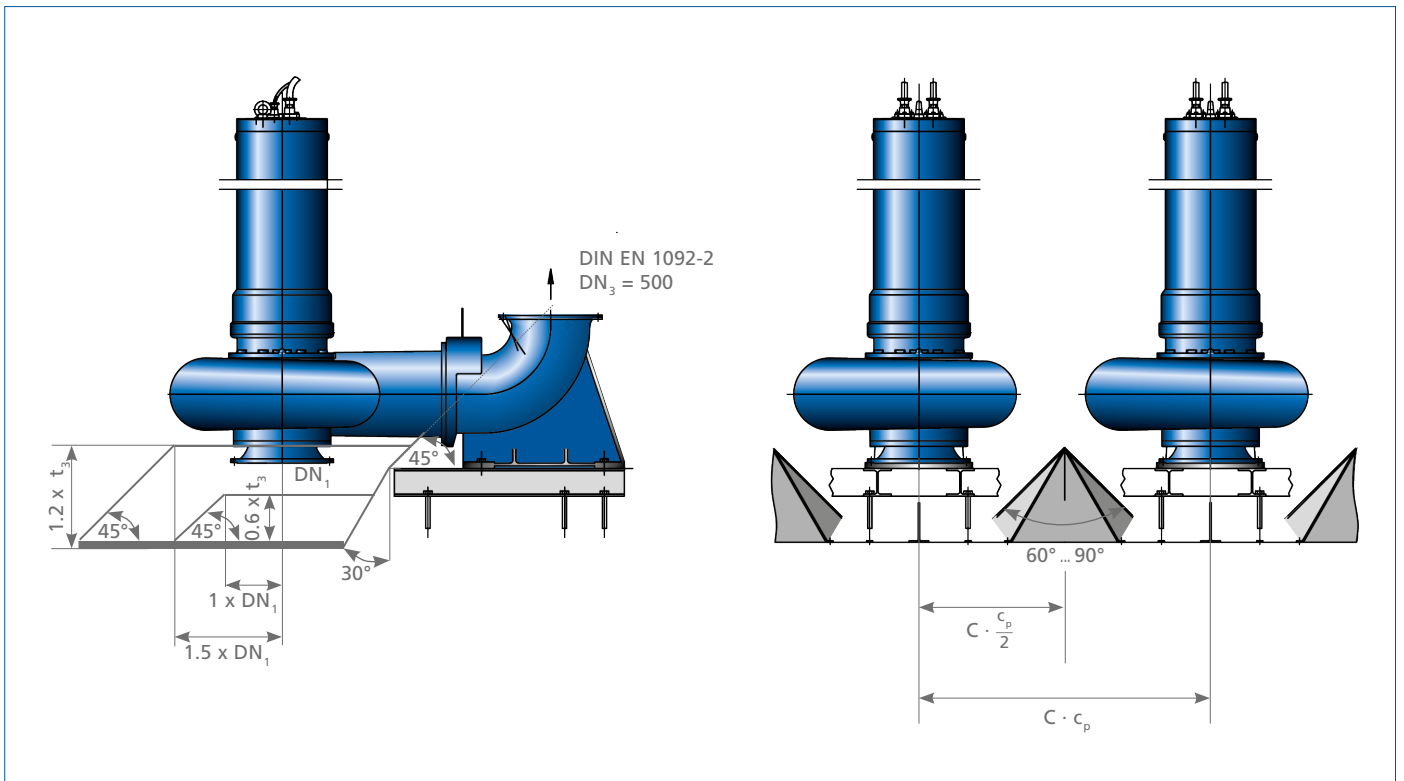


Fig. 94: Geometric design of (bottom) flow splitters

KSB commissioned comprehensive research on this topic at the Berlin Technical University. The results demonstrate the impact of the inclination angle of variously coated benching on the sliding behaviour of individual substances contained in the fluid (Fig. 93).

If established flow velocities in the pump sump are assumed for Fig. 94, it might be possible to reduce the inclination angle without any sedimentation forming in the floor area. This would again influence the costs for the structure. A more precise prediction of the situation in the pump sump can be made by means of a CFD simulation (see section "The significance of CFD simulation", page 86). Surface deterioration (increased roughness) caused by solids contained in the waste water or increased sliding resistance (sticking) of greases and oils were not part of the research conducted. These factors must be estimated depending on the local composition of the waste water.
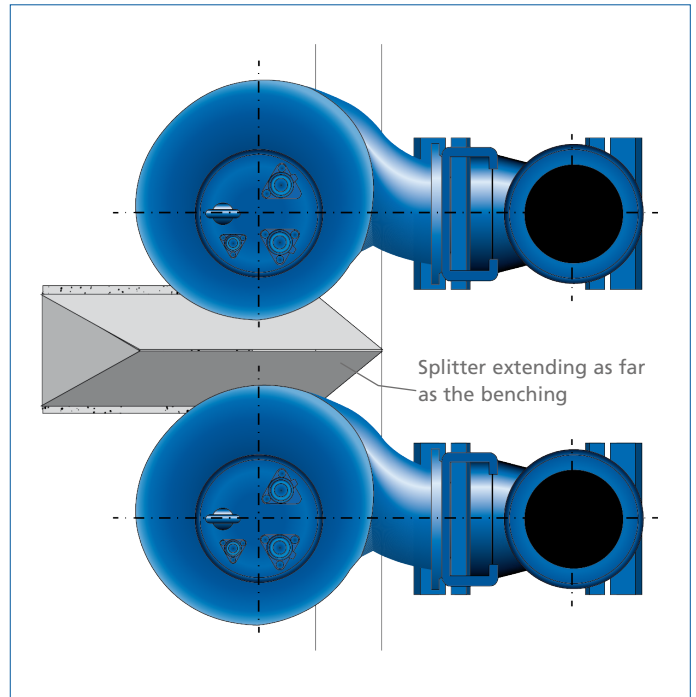


Splitter extending as far as the benching

Fig. 95: Top view – flow splitter between two pumps

## Splitters preventing submerged vortices

When installing pumps, unfavourable approach flow conditions can lead to the formation of submerged vortices, which have a negative effect on the performance and smooth running of the pumps. This can be addressed at the pumping station's planning stage: Fitting (bottom) flow splitters between the pumps will actively prevent the formation of such vortices (Fig. 94 and Fig. 95). Bottom flow splitters fitted underneath the suction cross-section (inlet) will directly influence the formation of vortices in the inlet flow. Flow splitters are intended to prevent the formation of submerged vortices between the pumps. The required geometric dimensions can be derived from the geometry of the selected pump size. The bottom flow splitters must be fitted such that they are absolutely symmetrical in relation to the inlet section of the pump. If not, the approach flow towards the impeller will be asymmetrical, causing well-known effects. The calculated dimensions should then be adjusted to the pump sump's shape or increased as required by the benching and wall contours. This will reduce the occurrence of impermissible low flow velocities in the direct vicinity of the pump and prevent undesirable deposits.

**Definition of variables and dimensions**

| Dimension Variable | Description |
|---|---|
| A | Distance between the centreline of the suction nozzle of the pump and the inlet or the opposite wall |
| $C_{cp}$ | Distance between the centrelines of adjacent suction nozzles/pumps |
| $C_{cw}$ | Distance between side walls and the centreline of the suction nozzle |
| $C_o$ | Opening in the curtain wall or balcony |
| $C_w$ | Width of the stilling basin or balcony |
| $C_b$ | Height of the balcony above the floor of the pump sump |
| Y | Minimum distance from the centreline of the suction nozzle to the outlet of a screen upstream of the suction nozzle |
| $\alpha$ | Inclination angle of the floor upstream of the suction level |

Table 16: Definition of variables and dimensions

The flow splitters can either be made of concrete or of (stainless) sheet steel. An advantage of sheet steel fabrications is that they can be installed after the concrete work and pump installation has been completed. This makes it easier to fit the flow splitters symmetrically with regard to the suction nozzle of the pump.

The position of the flow splitters between the pumps is based on the minimum distances derived from the required maximum volume flow rates of the individual pumps (Fig. 95). The visual asymmetry of this flow splitter arrangement can be ignored. It is caused by the offset position of the inlet section in relation to the volute-shaped pump casing.

The approach flow in relation to the pump installation and the height of the intake are key criteria for deciding on further procedures or selecting a structural design. In addition, the total volume flow rate of the pumping station and/or the maximum individual volume flow rates of the pump(s) should be assessed regarding the necessity of model testing (cf. section "The necessity of model testing", page 82).

The required minimum dimensions for the pumping station can be clearly identified by means of Table 16. The definitions are based on the terminology of the internationally renowned standards of the Hydraulic Institute H.I. 9.8 – 2018 [5.6].

## Dimensions for the pump sump and pump installation

The required minimum dimensions for installing pumps in a pump sump are a function of the maximum volume flow rates of the individual pumps and the maximum number of pumps in the pumping station. The individual volume flow rate of a pump can be used to calculate the required distance between pump and wall, adjacent pumps and the inlet channel or pipe.
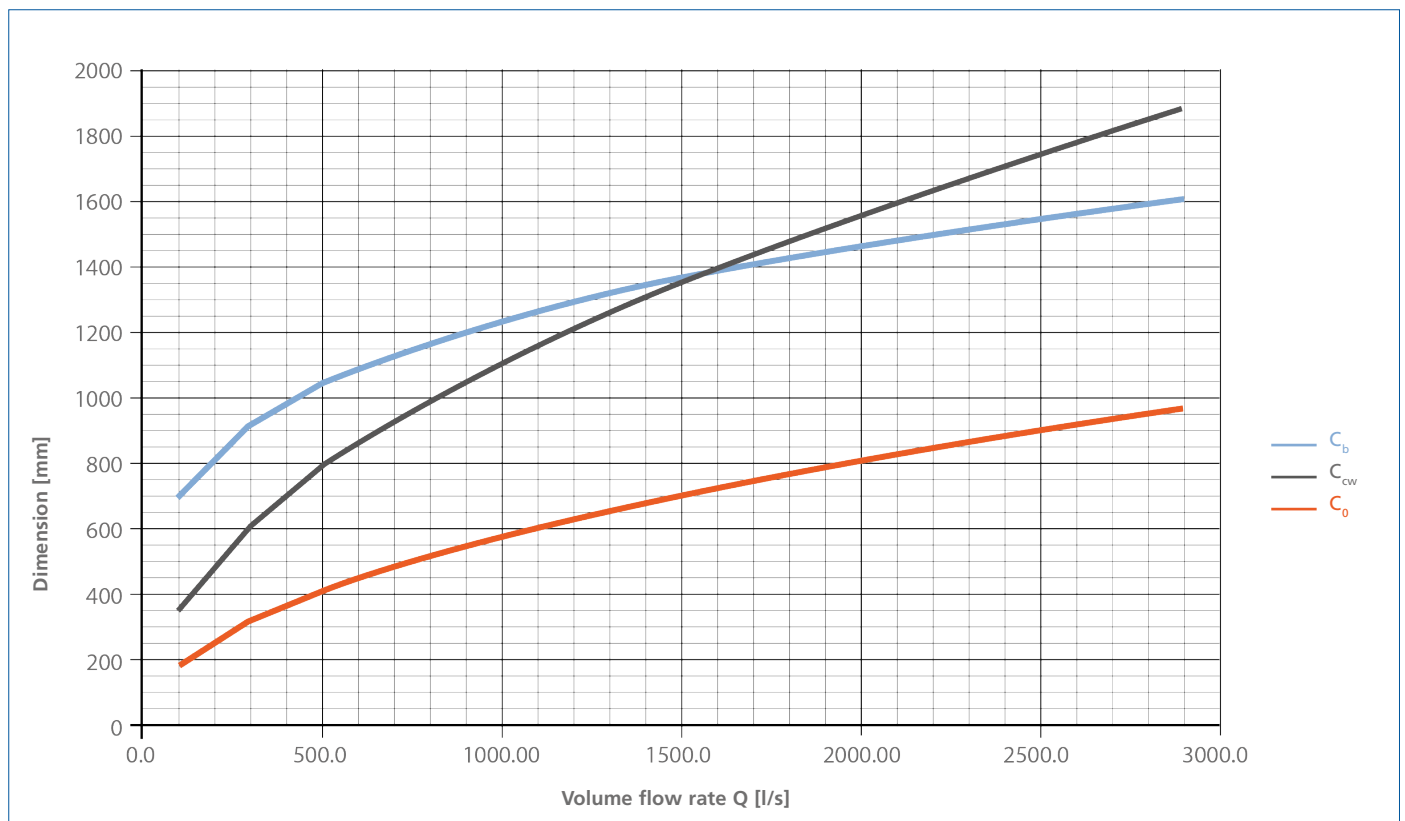


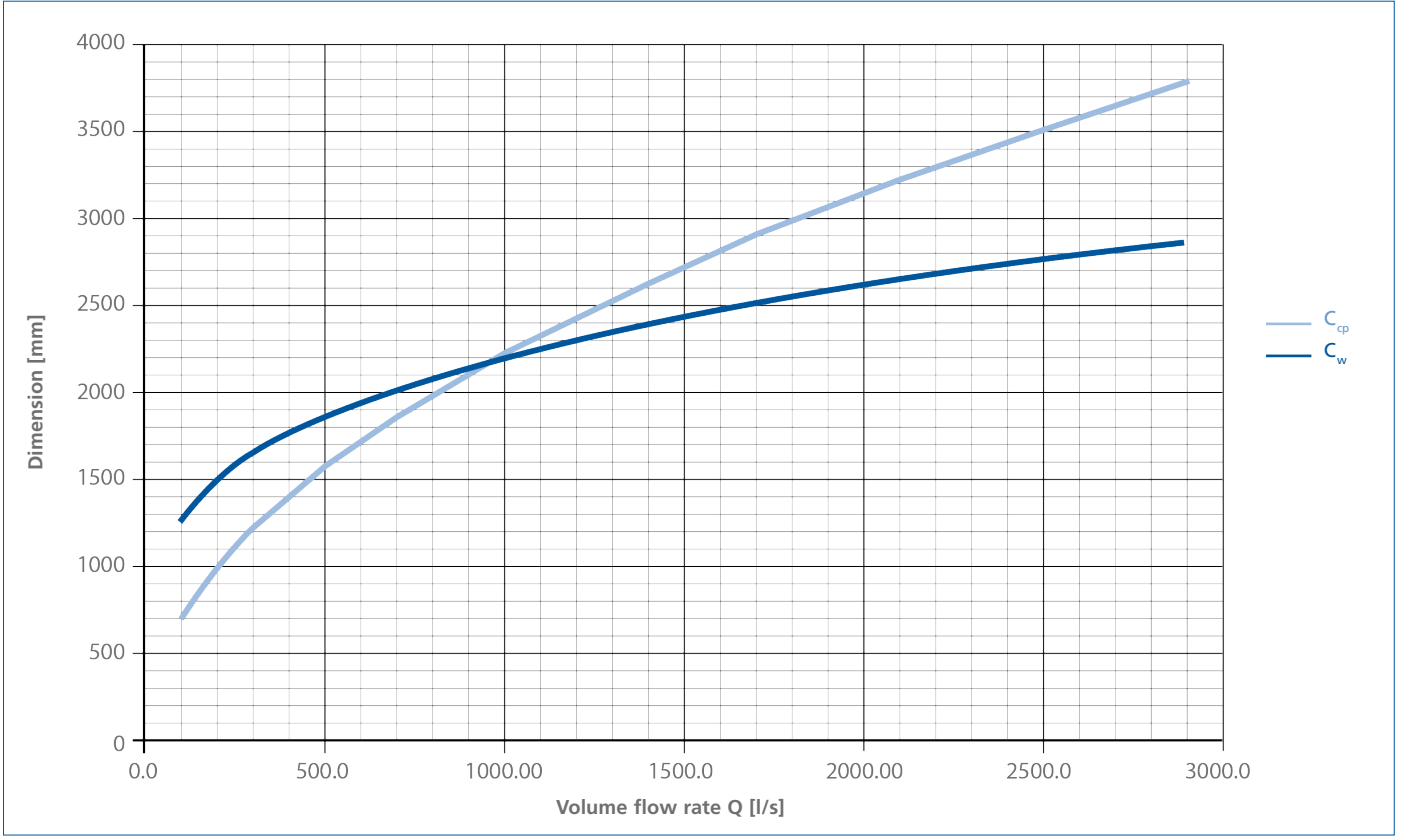Fig. 96: Minimum dimensions for the pump sump
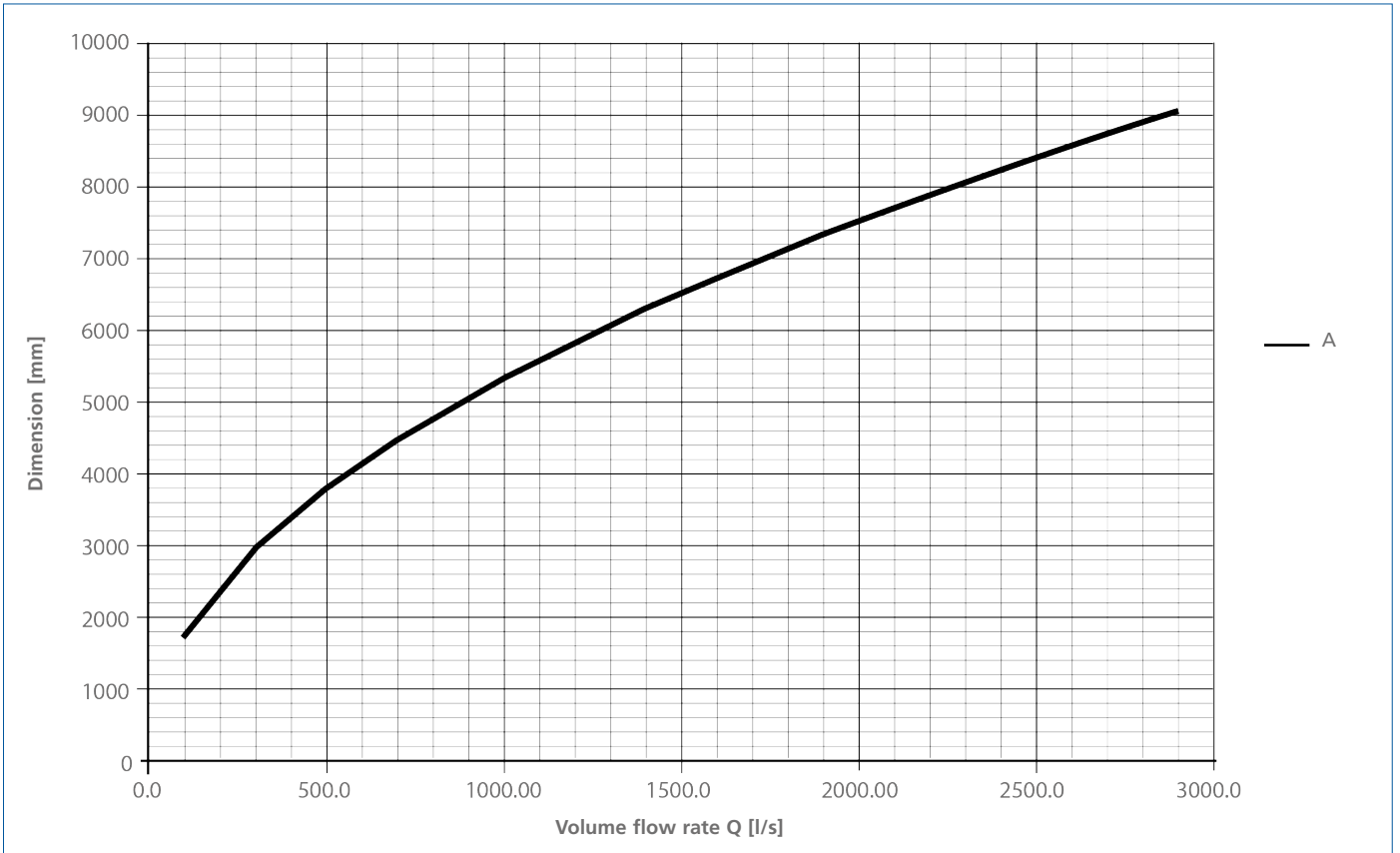
Fig. 97: Minimum dimensions for the pump sump



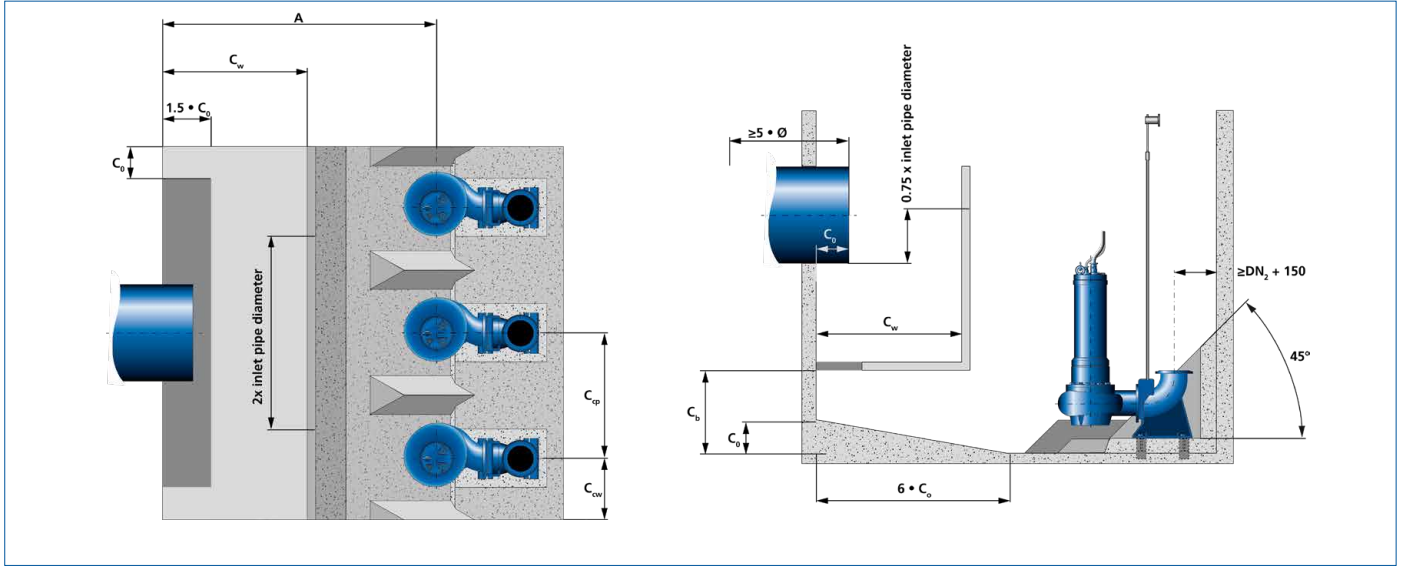Fig. 98: Minimum dimensions for the pump sump

Fig. 99: Example of a KRT (wet installation) with direct approach flow and height difference between pipe invert and pump sump
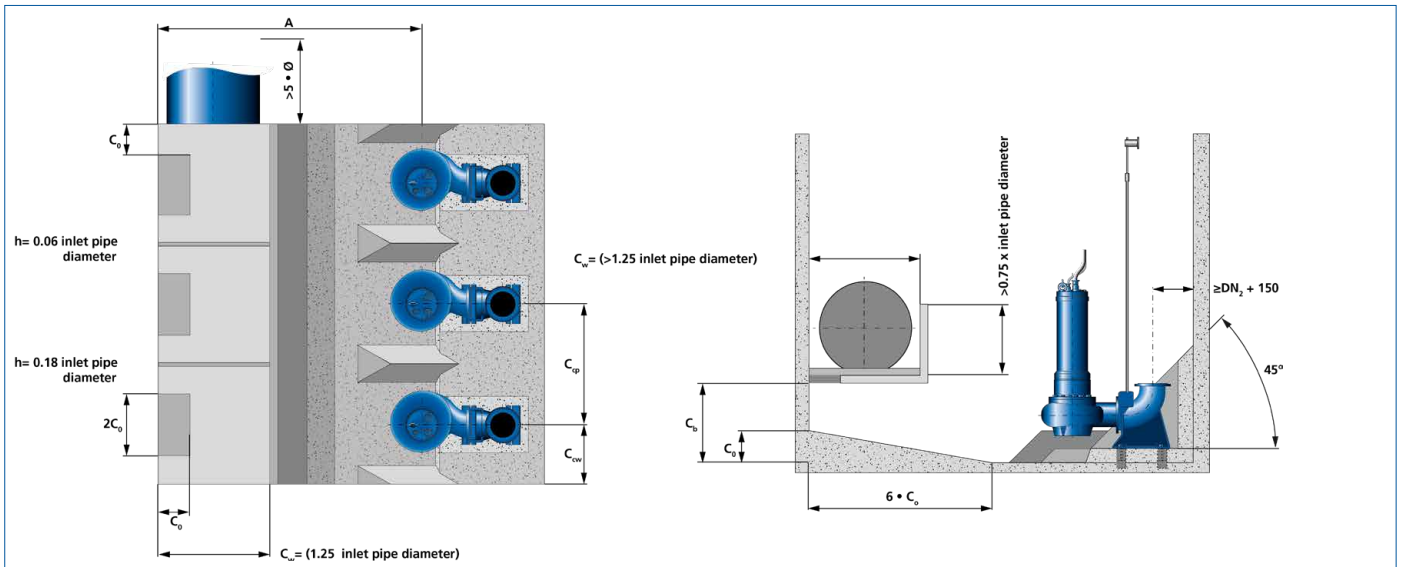


Fig. 100: Example of a KRT (wet installation) with longitudinal approach flow and height difference between pipe invert and pump sump
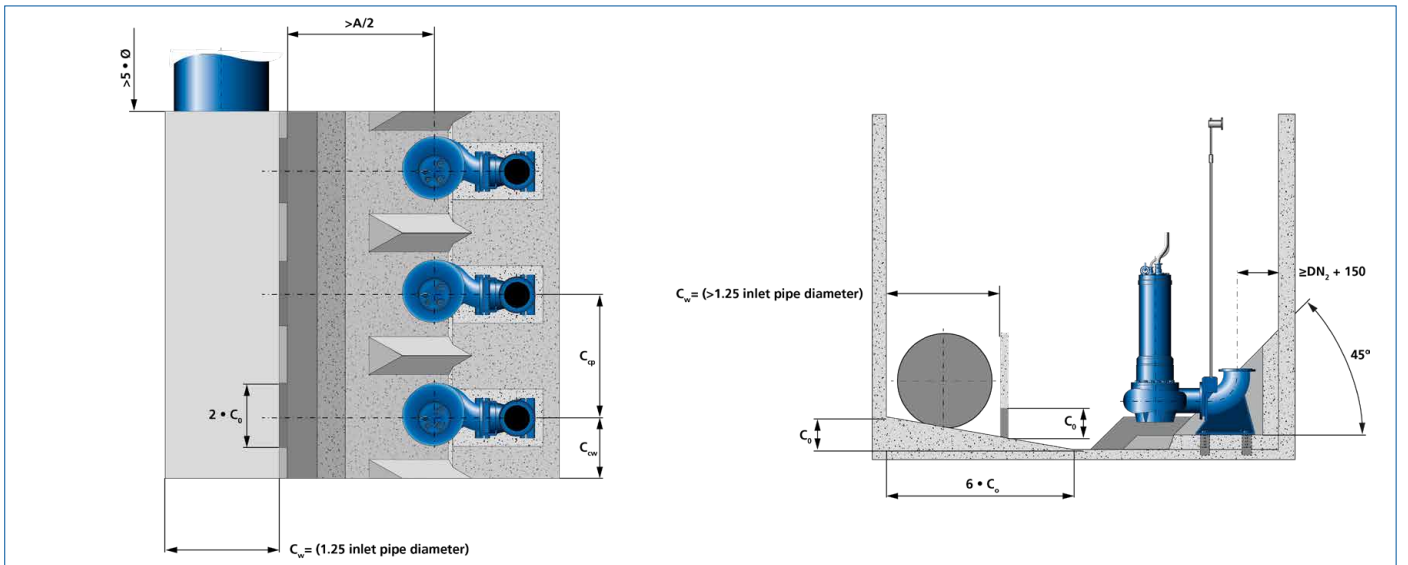


Fig. 101: Example of a KRT (wet installation) with longitudinal approach flow and no height difference between pipe invert and pump sump

Fig. 96 to Fig. 101 provide the required dimensions depending on the flow rate of the individual pumps. These diagrams only apply to a maximum number of five individual pumps. If this number of pumps per pump sump is exceeded, the pump sump dimensions must be validated using CFD and possibly also model testing. If more than five pumps are positioned next to each other, the distribution of the intake impulse in the pump sump and its effects are hard to predict, especially when waste water is being handled. If the flow approaches the pump in the precise direction of the pump installation, the intake impulse should be eliminated by means of a curtain wall with a floor opening. The height difference between the pipe invert and the minimum fluid level in the pump sump can be compensated for by a type of balcony construction. The inflow cross-section must be compatible with the maximum inlet flow velocity of 2.0 m/s. The length of the inlet channel should be at least 5 x the diameter of the inlet pipe to compensate for any negative effects from diversions or fittings upstream of the pump sump. The same applies to longitudinal approach flow in relation to the pump installation (see Fig. 99, Fig. 100 and Fig. 101).

## Wet-installed submersible motor pumps

The position of the inlet channel or pipe is decisive for determining the minimum dimensions of the pump sump. It should be established whether the inlet will be at the same height as the floor of the pump sump, or if an additional height difference has to be overcome (water falling onto a free surface increases the risk of additional air intake in the fluid). The direction of the approach flow in relation to the pump installation must also be determined.

## Pump sumps with major pollution loads

Temporary or even permanent high pollution loads can be handled by today's modern waste water pumping station without any major difficulties. The following conditions need to be fulfilled as a prerequisite:

- The pump sump is correctly dimensioned in size and shape.
- The operating mode will not overload the hydraulic system (e.g. concentrated pollution loads from a stormwater overflow basin causing a brief peak load of a few minutes).
- The pollution load or the fluid are not extreme in their characteristics.



Fig. 102: Submersible motor mixer in the pump sump

Over the last few years, experience has shown that problems with retained pollution loads, solids or sludge occur in no more than 3 % of all pumping stations worldwide. In these cases, the use of a small submersible motor mixer has proven extremely successful (Fig. 102).

This represents one of the most flexible targeted solutions – in terms of both time and location:

With regard to time, the mixer can be switched on briefly prior to the actual pumping process if a longer operating period is not required. This will distribute the total pollution load over the total fluid volume to achieve optimum conditions for pumping. The pollution load will be transported with the fluid right from the outset rather than being retained.

With regard to location, local deposits can be mixed in by pointing the jet core of the mixer directly at them. No action is required for zones in which adequate suspension is provided by the structural design or approach flow.

The mixer can also be positioned to mix in floating sludge so that it can be transported with the fluid.

Key benefits are:

- The mixer can be selected for the specific situation, e.g. the fluid composition, and the size and design of the sump.
- The total sump volume can be covered by one small mixer.
- Flexibility (see above)
- No reduction of the volume flow rate of the pump and complete transport of the substances without any additional cleaning procedures.

## The necessity of model testing

The object of model testing is to simulate the flow of a planned pumping station in a scale model. It helps identify precisely where problematic conditions (vorticity, uneven velocity distribution, etc.) might arise and how to then influence these positively, where necessary. The high transparency of acrylic glass makes this material an excellent choice for the construction of suitable models (Fig. 103). In order to be able to transfer the flow conditions to the full-size structure, dimensionless numbers are applied in the design of the model. These characteristic coefficients describe the forces acting in the flow; they should be as identical as possible for both the model and the full-size structure. The most relevant forces are gravity, as well as those resulting from dynamic viscosity, surface tension and the inertia of the fluid in motion. The dimensionless numbers applied here are as follows:

| | | |
|---|---|---|
| **Reynolds-number** | $Re = \dfrac{v \cdot d}{\nu}$ | Formula (21) |
| **Froude-number** | $Fr = \dfrac{v}{\sqrt{g \cdot l}}$ | Formula (22) |
| **Weber-number** | $We = \dfrac{\rho \cdot v^2 \cdot l}{\sigma}$ | Formula (23) |

**Key:**
$v$ = Flow velocity [m/s]
$d$ = Hydraulic diameter [m]
$\nu$ = Kinematic viscosity [m²/s]
$g$ = Acceleration due to gravity = 9.81 m/s²
$l$ = Characteristic length (in the hydraulic system) [m]
$\rho$ = Density [kg/m³]
$\sigma$ = Surface tension [N/mm²]

As these characteristics are to a degree interdependent, it is impossible to apply them at the same time in a scaled model. It is therefore important to find a compromise which helps achieve the optimum for a given application.

Model testing is absolutely necessary when one or more of the criteria listed below apply to the intake structure or pump sump:

- The concept of the pumping station building deviates from proven layouts as regards dimensions, piping layout, wall spacing, considerable changes in flow direction between inflow into the building and the approach flow to the pump, etc.
- The volume flow rate per pump is higher than 2.5 m³/s or 6.3 m³/s for the entire pumping station.
- The approach flow is asymmetrical and/or not uniform.
- Alternating pump operation in multiple-pump stations involves significant changes in flow direction.
- An existing pump station has already created problems.

### Test set-up

The geometry of the model must correspond with the original structure, taking into account the selected scale and the characteristic coefficients mentioned previously. This applies to the hydraulic part of the building structure and the pumps. Both the structure of the building and the pumps are constructed from transparent material. A model of the impeller is not required as the test aims to simulate only the flow approaching the impeller.

Instead of an impeller, a vortometer (see Fig. 103) is employed whose rotational speed provides information on the development of vortices in the intake.

The flow velocities are measured at reference points across the model pump's entire suction cross-section via Pitot tube or laser. To judge vortex development the fluid surface as well as the wall and floor areas are observed. Vortex intensity in a given flow cross-section is visualised by means of dyes while their size is measured by the swirl angle θ of the vortometer. The following equation is applicable:



Fig. 103: Vortometer

$$\sigma = \tan^{-1}\left(\frac{\pi \cdot d_m \cdot n}{u}\right)$$

Formula (24)

**Key:**
$\sigma$ = Swirl angle [°]
$\pi$ = Pi = 3.14159265359
$d_m$ = Pipe diameter (here the pump's suction pipe) [m]
$n$ = Rotations of vortometer [1/s]
$u$ = Axial flow velocity [m/s]

The surface vortices are classified according to Hecker in six categories (1 = low, 6 = very high, Fig. 106) and the submerged vortices in three categories (Fig. 107).

If one were to consult only the diagrams, these vortex formations would appear relatively harmless. Yet vortex formation observed in model tests gives an idea of what could happen in a real structure. Unlike laboratory situations, a real pumping station rarely deals with clear water and it is difficult to identify vortex action as the source of a problem, especially when submerged vortices are involved.

The criteria which apply to this method of investigation may vary slightly depending on the pump type and the size and design of the pumping station.
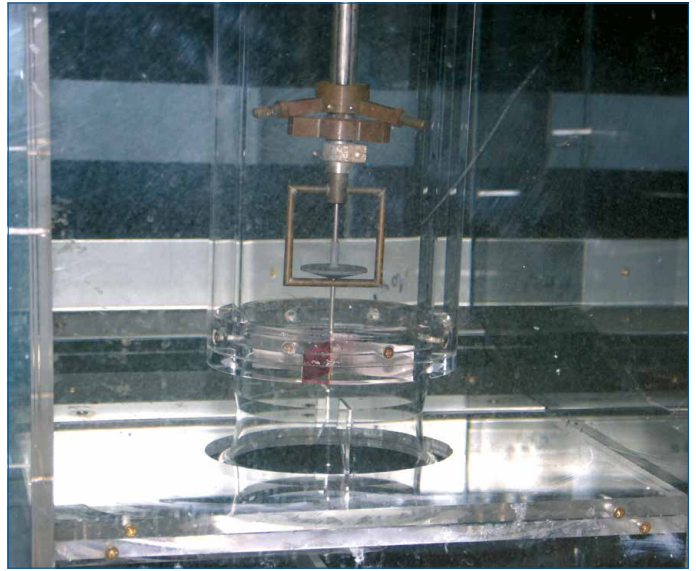
Fig. 104: Laboratory photo of a surface vortex type 6



Fig. 105: Laboratory photo of a surface vortex type 3

**Evaluation of results**

The measurement results should be confirmed by all parties involved before the design is finalised: pumping station designer, end customer, pump manufacturer and the institution which conducted the tests.

Key criteria:

- The mean flow velocity at the defined measurement points of the suction cross-section should not deviate from the mean value by more than 10 %.
- The swirl angle should not exceed 5 degrees. A swirl angle of 6 degrees can be tolerated if it has occurred during less than 10 % of the period of observation.
- Surface vortices may only be accepted up to type 2 and submerged vortices up to type 1. They can be tolerated in exceptional cases if they have occurred during less than 10 % of the period of observation.

**In general the following applies:**

Occurrences that have a minor effect in the model may be considerably more significant in the full-scale structure!

The tests must be concluded with a detailed report on the operating conditions investigated. The vortex formations and operating conditions observed (for the tested fluid levels in the building structure) must be documented on video and made available to the party commissioning the tests.

KSB will support and co-ordinate project-related model testing upon request.
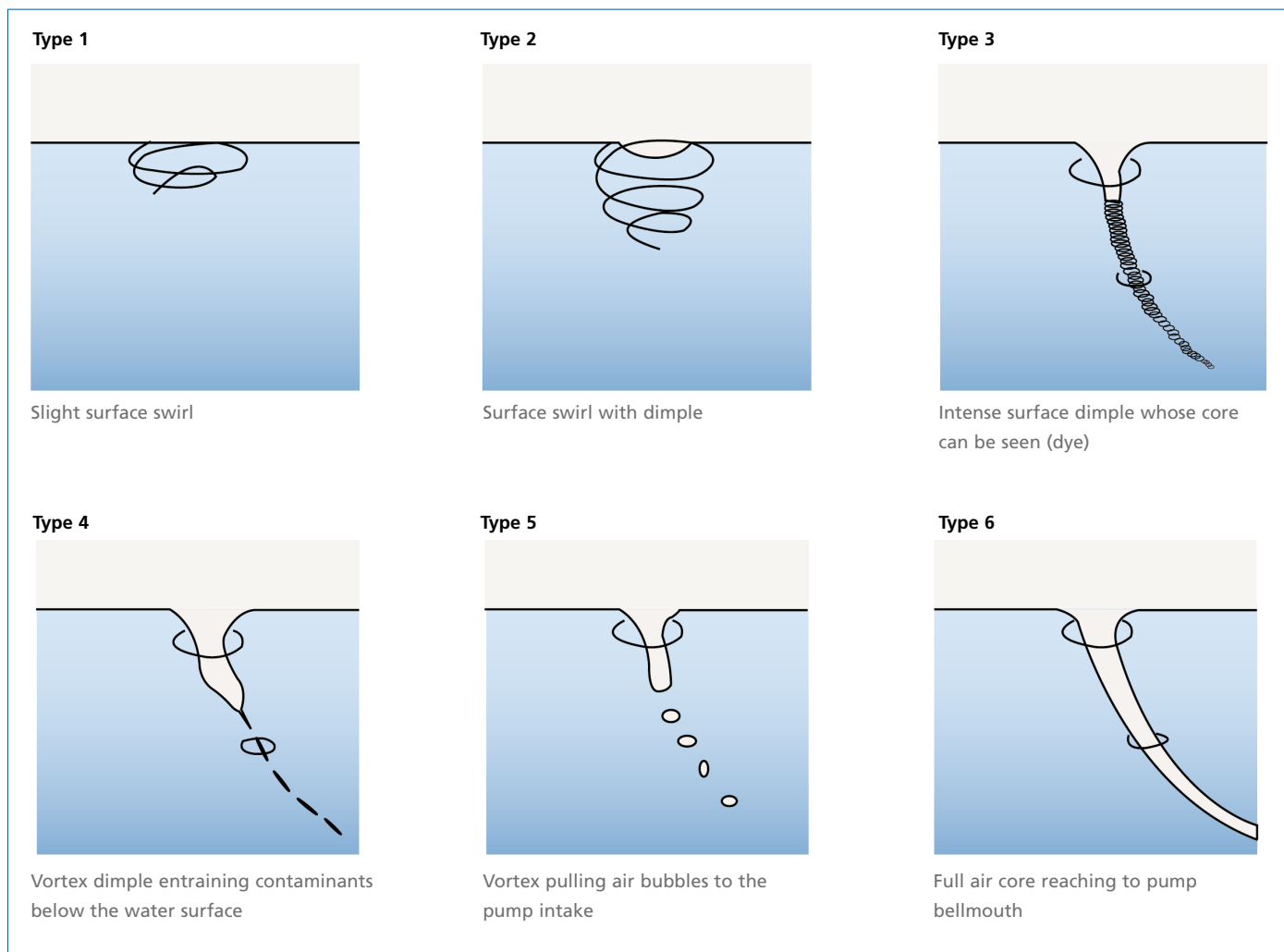
**Type 1**



Slight surface swirl

**Type 2**



Surface swirl with dimple

**Type 3**



Intense surface dimple whose core can be seen (dye)

**Type 4**



Vortex dimple entraining contaminants below the water surface

**Type 5**



Vortex pulling air bubbles to the pump intake

**Type 6**



Full air core reaching to pump bellmouth

Fig. 106: Classification of surface vortices according to Hecker (Types 1 to 6)



**Type 1:**
Slight swirl at floor or wall



**Type 2:**
Vortex at floor or wall



**Type 3:**
Air-entraining vortex at floor or wall



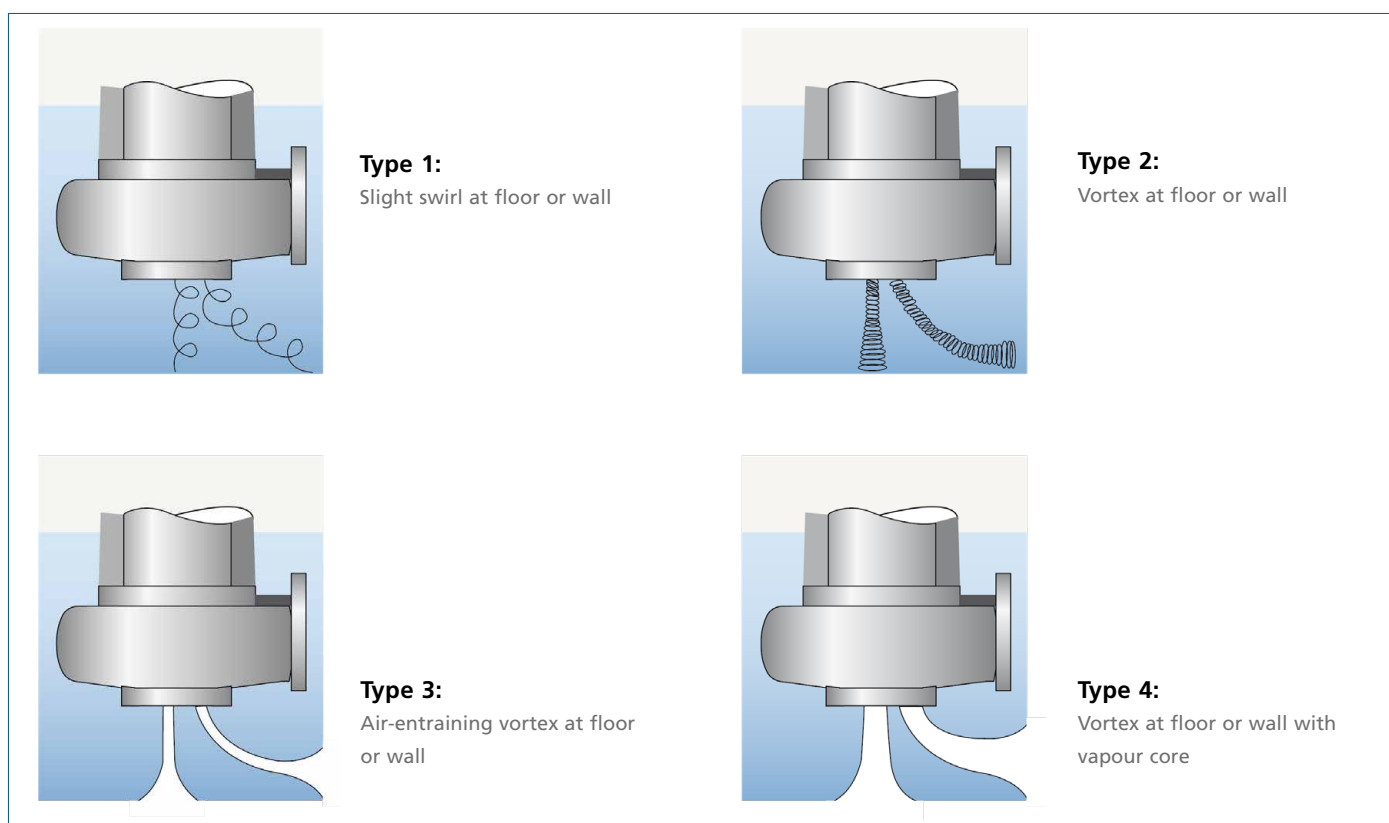**Type 4:**
Vortex at floor or wall with vapour core

Fig. 107: Classification of submerged vortices according to Hecker (Types 1 to 4)
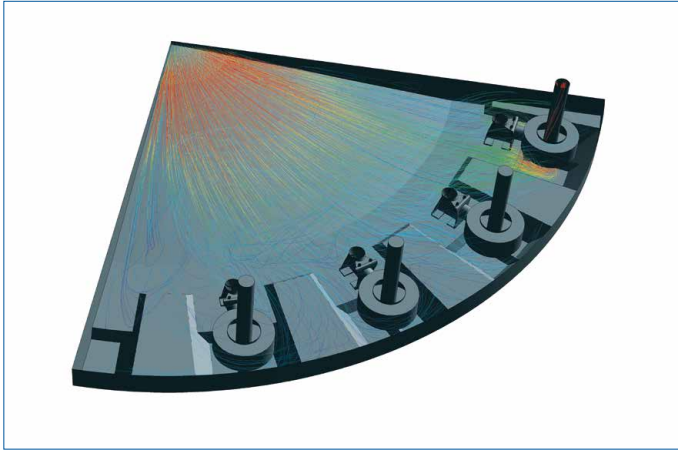
Fig. 108: Flow pattern developing in KRT intake chambers

# The significance of CFD simulation

Numerical flow simulation (Computational Fluid Dynamics = CFD) is becoming increasingly important.
The software specially developed for this purpose is an effective instrument allowing relatively precise predictions of the flow conditions. The time and cost of flow modelling depends on the

- Size of the flow area to be modelled
- Desired geometric resolution
- Computer performance
- Form of presentation (report)
- and scope of results.

### Methodology

The mathematical description of fluid flows is based on the Navier-Stokes equations. They describe the processes at each point of a flow by means of partial differential equations for mass, energy and momentum.

The calculation of each spatial point in a flow is not feasible as this would result in an infinite number of calculations. A grid is generated instead, whose nodes are calculated. The grid model is then processed further to provide information on the pressure and velocity distribution, which can then be subjected to numerical and/or graphical analysis. In modelling, the distances between individual nodes may partly differ; they depend on the flow velocity gradients.

The calculated nodes lie closer together near walls and corners, which are considered as discontinuities from a fluid dynamics point of view. In areas with low velocity gradients it is not a problem to increase the distances. In addition, assumptions on the distribution of turbulence are made at the nodes. The task of a CFD specialist is to choose the "correct" turbulence model. It takes a lot of experience to be able to create an adequate model and to be able to accurately interpret the results obtained.

CFD simulation is perfectly suited to evaluating flows in intake structures and pump sumps, especially as it can also be used to perform a very precise analysis of the influence individual pumps have on the flow pattern in multiple-pump systems. Problems are more often caused by the fact that surface and submerged vortices and asymmetrical approach flows do not always exhibit steady behaviour and are therefore difficult to predict exactly.

At KSB, CFD simulation is a well-established engineering tool that has been used for years. The fact that CFD calculations have been proven to conform well with model testing in past investigations allows more accurate predictions to be made today on potential flow situations and enables pumping stations to be more systematically optimised. In complicated cases, however, physical model testing is preferable to CFD calculations for building structure investigations. In the future, the use of both CFD simulations and model testing will significantly reduce the overall costs of pumping station investigations.
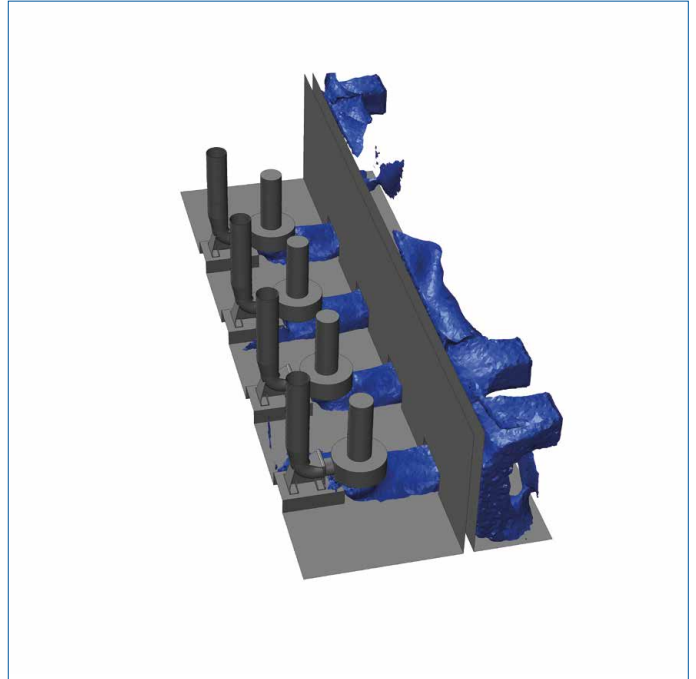


Fig. 109: Flow pattern developing in a KRT pumping station

$$\frac{\partial u}{\partial t} + u \cdot \frac{\partial u}{\partial y} + v \cdot \frac{\partial u}{\partial y} + w \cdot \frac{\partial u}{\partial z} = f_x - \frac{1}{\rho} \cdot \frac{\partial p}{\partial x} + v \cdot \left[ \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right]$$

$$\frac{\partial v}{\partial t} + u \cdot \frac{\partial v}{\partial x} + v \cdot \frac{\partial v}{\partial y} + w \cdot \frac{\partial v}{\partial z} = f_y - \frac{1}{\rho} \frac{\partial p}{\partial y} + v \cdot \left[ \frac{\partial^2}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right]$$

$$\frac{\partial w}{\partial t} + u \cdot \frac{\partial w}{\partial x} + v \cdot \frac{\partial w}{\partial y} + w \cdot \frac{\partial w}{\partial y} = f_z - \frac{1}{\rho} \frac{\partial p}{\partial z} + v \cdot \left[ \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right]$$

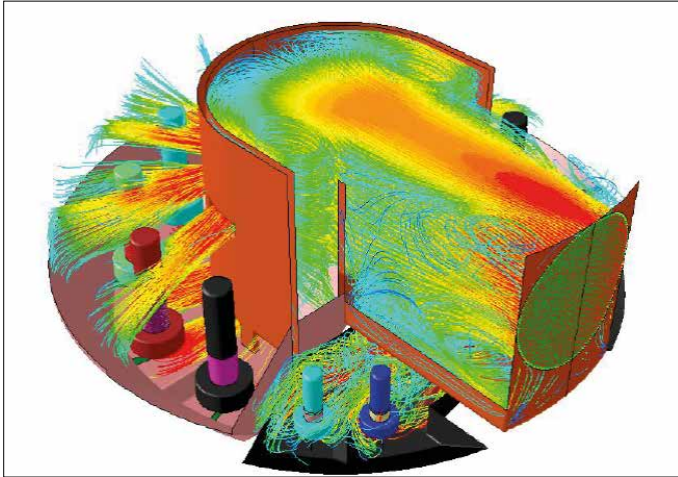Fig. 110: Navier-Stokes equation system for flow description

Fig. 111: Simulation of pumping station with several pumps



Fig. 112: Structural design of a previously calculated KRT pumping station

## Objectives

Model testing is particularly valuable when diagnosing surface vortices and swirl values for various pump inlet positions. The velocity distribution at the impeller level can be examined via thorough testing. Recognising the flow quality of complicated inlet geometries in a model test requires extensive experience. This is where a CFD analysis is advantageous. The flow can be made visible over the entire volume. The flow quality can easily be analysed in areas of constant velocities and cross-sections.

The following severe problems can occur in the geometries:
- Transient flows in the pump area
- Sedimentation in large waste water treatment plants
- Air-entraining vortices and submerged vortices
- Swirling approach flows to the pumps (pre-swirl leads to increased cavitation or changed heads)
- Air intake (here: air transport by the flow)

Transient flows are time-dependent flows. If the flow quality changes over time, the required acceleration forces have to be generated by the pump, which usually causes vibrations. Pumps with a high specific speed are especially susceptible.

Sedimentation is extremely detrimental to the operation of a waste water treatment plant. Removing deposited sediment can be very costly. The risk of sedimentation in the pump chamber can be analysed by examining the velocities near the floor.

Air-entraining vortices caused by the qualitative flow in the chamber can be predicted fairly accurately. An approach flow tangential to the chamber is most likely to generate a vortex in the chamber with an air-entraining vortex at its centre. Swirling flows influence the pump head and the pump input power. They also have an effect on the cavitation characteristics. The air intake cannot be calculated but the transport of air entering the structure with the flow can be estimated by examining the velocity distribution in the volume.

**Summary**

If operational problems are anticipated, every possible option should be employed to analyse these potential problems and thus prevent the costs they could incur. CFD simulation is perfectly suited to evaluating flows in intake structures and pump sumps. The aim is to prevent any operating problems in the water system or waste water treatment plant that is to be built. A logical analysis of the plant under consideration is the basis for efficient model testing and CFD analyses.

At KSB, CFD simulation is a well-established engineering tool that has been used for years. For some pumping stations it makes sense to use a combination of CFD simulation and model testing to optimise the plant and find the best solution.

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