Manual for the Design of Pipe Systems and Pumps

engineering for a better world

GEA Mechanical Equipment

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Preface

Archimedes - the ingenious scientist of the ancient world - recognized the functionality of pumps as early as in the middle of the 3rd cent. B.C. Through the invention of the Archimedean screw, the irrigation of the fields became much more effective.
2200 years later GEA Tuchenhagen is building high-tech pumps for hygienic process technology giving the process lines the optimal impetus.

Selecting the right pump to serve the purpose is not always that easy and requires special knowledge. GEA Tuchenhagen has set up this Manual for giving support in finding out the optimal pump design. Special attention was given to produce a Manual that is interesting and informative for everybody, from the competent engineer to the layman.
The contents is self-explanatory and built up one after the other. Simplifications were partly accepted and profound theories dispensed with. We hope that this Manual will give you an extended comprehension of this subject and will help you solving any problems that might occur.

Formula, Units, Designation

Formula		Designation	SI - Unit
B		Operating point	-
D		Impeller diameter	mm
DN or d		Nominal width of the pipe or pump port	mm
g		Acceleration of the fall = 9.81 m/s^2	m/s^2
H		Flow head	m
H_A		Flow head of the system	m
H_{geo}		Geodetic flow head	m
$H_{s,geo}$		Geodetic suction head	m
$H_{d,geo}$		Geodetic pressure head	m
$H_{z,geo}$		Static suction head	m
H_z		Flow head viscous medium	m
H_v		Pressure drops	m
$H_{v,s}$		Pressure drops, suction side	m
$H_{v,d}$		Pressure drops, delivery side	m
K_H		Correction factor for the flow head	-
K_Q		Correction factor for the flow rate	-
K_h		Correction factor for the efficiency	-
k		Pipe roughness	mm
l		Pipe length	m
n		Speed	rpm.
$NPSH_{req.}$		NPSH (pump)	m
$NPSH_{avl}$		NPSH (system)	m
P		Power consumption	kW
P_z		Power consumption viscous medium	kW
p		Pressure	bar
p_a		Pressure at the outlet cross section of a system	bar
p_b		Air pressure / ambient pressure	bar
p_D		Vapour pressure of pumped liquids	bar
p_e		Pressure at the inlet cross section of a system	bar
Q		Flow rate	m^3/h
Q_z		Flow rate viscous medium	m^3/h
Re		Reynolds number	-
v		Flow speed	m/s
v_a		Flow speed at the outlet cross section of a system	m/s
v_e		Flow speed at the inlet cross section of a system	m/s
ζ	(Zeta)	Loss value	-
η	(Eta)	Efficiency of the pump	-
η_z	(Eta)	Efficiency of the pump for viscous medium	-
λ	(Lambda)	Efficiency value	-
ν	(Ny)	Kinematic viscosity	mm^2/s
η	(Eta)	Dynamic viscosity	Pa s
ρ	(Rho)	Density	t/m^3

2 Introduction

The requirements made on process plants steadily increase, both regarding the quality of the products and the profitability of the processes. Making liquids flow solely due to the earth's gravitational force is today unthinkable. Liquids are forced through pipes, valves, heat exchangers, filters and other components, and all of them cause an increased resistance of flow and thus pressure drops.

Pumps are therefore installed in different sections of a plant. The choice of the right pump at the right place is crucial and will be responsible for the success or failure of the process. The following factors should be taken into consideration:

1. Installation of the pump

2. Suction and delivery pipes

3. The pump type chosen must correspond to product viscosity, product density, temperature, system pressure, material of the pump, shearing tendency of the product etc.

4. The pump size must conform to the flow rate, pressure, speed, suction conditons etc.

As a manufacturer and supplier of centrifugal pumps and positive displacement pumps we offer the optimum for both applications.
Generally spoken, the pump is a device that conveys a certain volume of a specific liquid from point A to point B within a unit of time.
For optimal pumping, it is essential before selecting the pump to have examined the pipe system very carefully as well as the liquid to be conveyed.

2.1 Pipe systems

Pipe systems have always special characterstics and must be closely inspected for the choice of the appropriate pump. Details as to considerations of pipe systems are given in Chapter 6, "Design of pumps".

2.2 Liquids

Each liquid possesses diverse characteristics that may influence not only the choice of the pump, but also its configuration such as the type of the mechanical seal or the motor. Fundamental characteristics in this respect are:

- Viscosity (friction losses)
- Corrodibility (corrosion)
- Abrasion
- Temperature (cavitation)
- Density
- Chemical reaction (gasket material)

Besides these fundamental criteria, some liquids need special care during the transport. The main reasons are:

- The product is sensitive to shearing and could get damaged, such as yoghurt or yoghurt with fruit pulp
- The liquid must be processed under highest hygienic conditions as practised in the pharmaceutical industry or food industry
- The product is very expensive or toxic and requires hermetically closed transport paths as used in the chemical or pharmaceutical industry.

2.3 Centrifugal or positive displacement pump

Experience of many years in research and development of pumps enables GEA Tuchenhagen today to offer a wide range of hygienic pumps for the food and beverage industry as well as the pharmaceutical and chemical industry.

We offer efficient, operationally safe, low-noise pumps for your processes and this Manual shall help you to make the right choice.

The first step on the way to the optimal pump is the selection between a centrifugal pump or a positive displacement pump. The difference lies on one hand in the prin-ciple of transporting the liquid and on the other hand in the pumping characteristic. There are two types of centrifugal pumps: "non-self priming" and "selfpriming".
Centrifugal pumps are for most of the cases the right choice, because they are easily installed, adapted to different operating parameters and easily cleaned. Competitive purchase costs and reliable transport for most of the liquids are the reason for their steady presence in process plants. Restrictions must be expected in the following cases:

- with viscous media the capacity limit is quickly reached,
- the use is also restricted with media being sensitive to shearing,
- with abbrasive liquids the service life of the centrifugal pump is reduced because of earlier wear.

2.4 GEA Tuchenhagen® VARIFLOW Programme

The GEA Tuchenhagen®-VARIFLOW Pump Programme conforms to today's requirements made on cleanability, gentle product handling, efficiency and ease of maintenance.
The different technical innovations made on the pumps for the optimization of cleanability have been EHEDG-certified.

2.5 ➤ Applications

The new GEA Tuchenhagen®-VARIFLOW pumps are preferably used in the brewing and beverage industry as well as in dairies and in process plants for chemical, pharmaceutical and health care products where highest hygienic standards are set. They are used in these industries mainly as transfer pumps, ClP supply pumps and booster pumps.

Fig. 1 - GEA Tuchenhagen®-VARIFLOW Centrifugal Pump, Type TP

2.6 ➤ Capacity range

The GEA Tuchenhagen®-VARIFLOW series is designed for flow rates up to 220 m^3/h and flow heads up to 92 m liquid column

2.7 ➤ Design

Pumps of the GEA Tuchenhagen®-VARIFLOW series are non-self priming, one-stage centrifugal pumps with single bent vanes.

- All pumps of the GEA Tuchenhagen®-VARIFLOW series are driven by standard motors of type IM B35.

- A spiral housing is used as guiding device for the GEA Tuchenhagen®-VARIFLOW series . It provides high efficiency so that the operating costs can be lowered.
  Besides high efficiency, special emphasis has been given to gentle product handling. After the product leaves the impeller, it is gently discharged in flow direction via the spiral housing.

- GEA Tuchenhagen®-VARIFLOW pumps have been certified according to EHEDG and 3A.

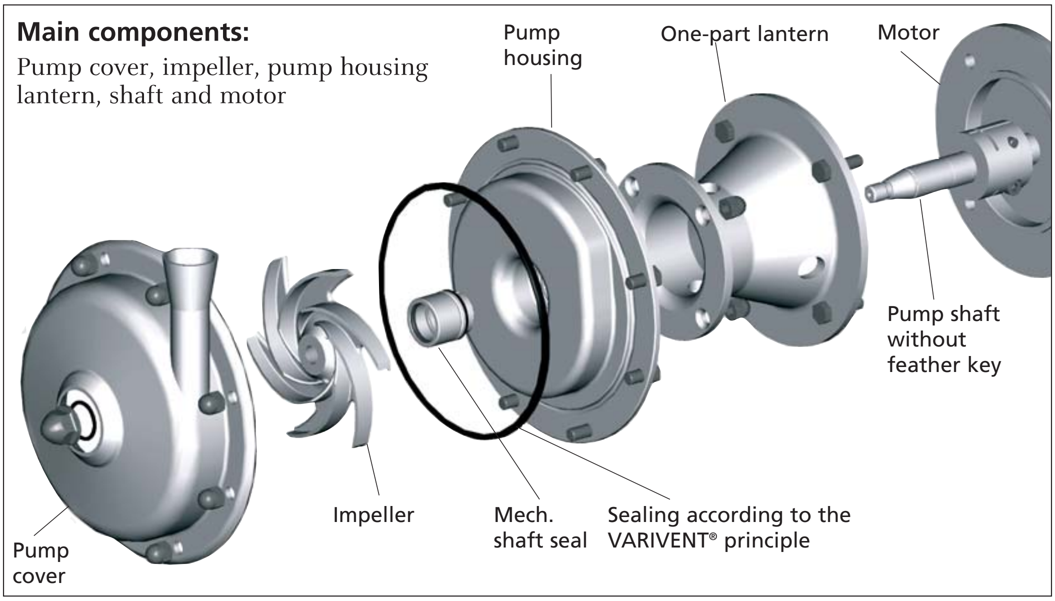


Fig 2 - GEA Tuchenhagen®-VARIFLOW, TP

2.8 Special features

- All parts in stainless steel, product wetted components are made of AISI 316L (1.4404).
- High efficiency
- Gentle product handling
- Low noise
- Ease of maintenance
- Excellent hygienic properties.

2.9 Connection fittings

- Threaded joint as per DIN 11851 (Standard)
- VARIVENT® flange connection
- Aseptic flanges as per DIN 11864-2
- Aseptic union as per DIN 11864-1
- Other marketable connections according to BS, SMS, RJT, Tri-Clamp
- Metric and Inch diameters

2.10 Accessories and Options

- Mechanical seals in different materials Carbon/Silicon carbide or Silicon carbide/Silicon carbide
- Different designs as single acting, single with flush (Quench) or double acting
- FDA approved soft seals: EPDM and FPM
- Stainless steel protection hood, mobile baseframe, drainage valve
- Adjustable calotte type feet frame
- Inducer

2.11 ➤ Self-priming centrifugal pumps

The GEA Tuchenhagen self-priming pump of the TPS series are used for conveying aggressive, clean liquids that are free of abrasive constituents. Pumps of the TPS series are horizontal, self-priming pumps. They stand out by their sturdy construction and high operational reliability and are preferably used in the food processing and luxury food industry as a CIP return pump.

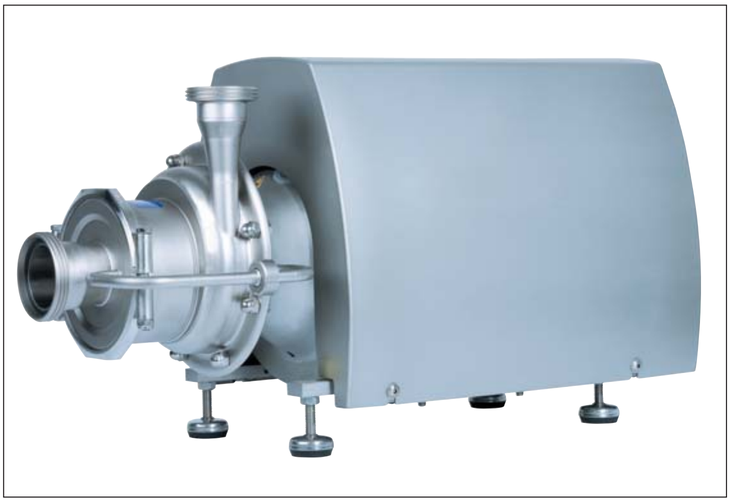


Fig. 3 - Self-priming centrifugal pump, type TPS

2.12 ➤ Rotary lobe pumps

GEA Tuchenhagen rotary lobe pumps of the VPSH and VPSU series are used whenever viscous, sensitive or solids-containing liquids must be gently transferred.
Type VPSH is used in hygienic applications of all kinds.
Type VPSU has been designed especially for high aseptic requirements that are standard in sterile areas.
The special design of the Skimitar rotors and the pump's design enable the pump to convey a wide range of media: from low-viscous media to products with a viscosity of up to 1,000,000 mPas or even media with suspended solids.
Due to the shape of the Skimitar rotors, a particularly high efficiency is achieved.

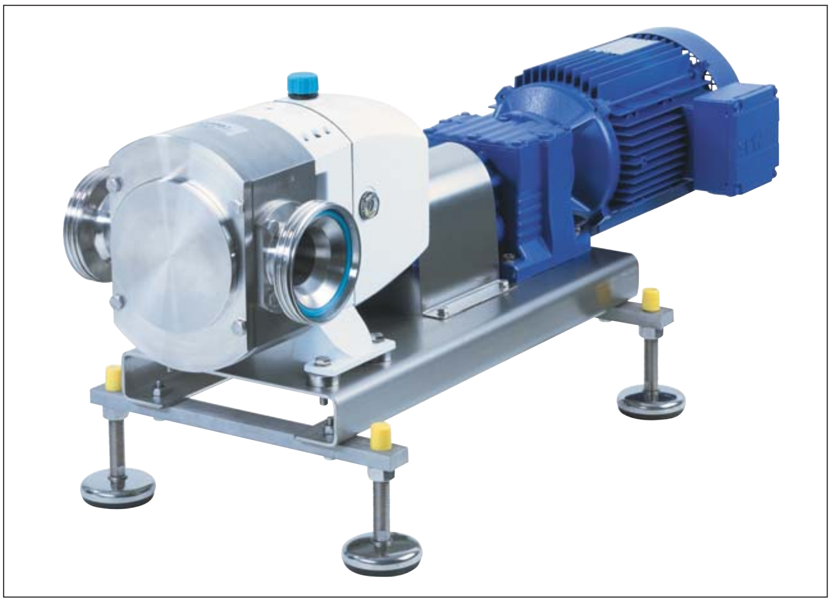


Fig. 4 - Rotary lobe pump VPSH

3 Physical Fundamentals

Fluids - a subject matter of this Manual - include liquids, gases and mixtures of liquids, solids and gases. All these fluids have specific characteristics that will be explained in this chapter.

3.1 Density

Density (ρ = Rho) - former specific weight - of a fluid is its weight per unit volume, usually expressed in units of grams per cubic centimeter (g/cm^3).
Example: If weight is 80 g in a cube of one cubic centimeter, the density of the medium is 80 g/cm^3. The density of a fluid changes with the temperature..

3.2 Temperature

Temperature (t) is usually expressed in units of degrees centigrade (°C) or Kelvin (K). The temperature of a fluid at the pump inlet is of great importance, because it has a strong effect on the suction characteristic of a pump.

3.3 Vapour pressure

The vapour pressure (p_D) of a liquid is the absolute pressure at a given temperature at which the liquid will change to vapour. Each liquid has its own specific point where it starts to evaporate. Vapour pressure is expressed in bar (absolute).

3.4 Viscosity

Viscosity of a medium is a measure of its tendency to resist shearing force.
Media of high viscosity require a greater force to shear at a given rate than fluids of low viscositiy.

3.5 Dynamic and kinematic viscosity

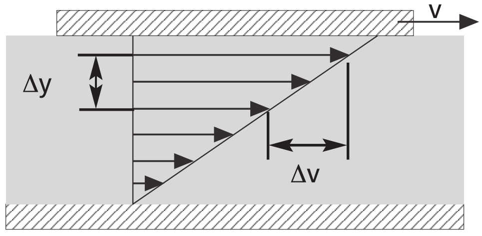
One has to distinguish between kinematic viscosity (ν = Ny) and dynamic viscosity (η = Eta). Centipoise (cP) is the traditional unit for expressing dynamic viscosity. Centistokes (cSt) or Millipascal (mPa) express the kinematic viscosity.

$$\text{Ratio: kinematic viscosity} = \frac{\text{dynamic viscosity}}{\text{density}}$$

Viscosity is not constant and thus depending on external factors. The viscous behaviour of media is more clearly expresed in effective viscosity or shearing force. The behaviour of viscous fluids varies.
One distinguishes between Newtonian and Non-Newtonian fluids.

3.6 Fluid behaviour

The flow curve is a diagram which shows the correlation between viscosity (η) and the shear rate (D). The shear rate is calculated from the ratio between the difference in flow velocity of two adjacent fluid layers and their distance to eachother.



$$D = \frac{\Delta v}{\Delta y}$$

Fig. 5 - Shear rate

The flow curve for an ideal fluid is a straight line. This means constant viscosity at all shear rates. All fluids of this characteristic are "Newtonian fluids". Examples are water, mineral oils, syrup, resins.

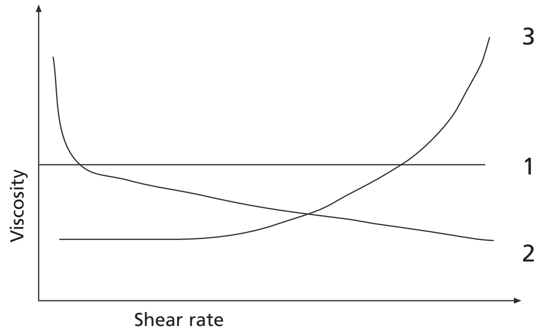


Fig. 6 - Flow curves

Fluids that change their viscosity in dependence of the shear rate are called "Non-Newtonian fluids". In practice, a very high percentage of fluids pumped are non-Newtonian and can be differentiated as follows:

Intrinsically viscous fluids

Viscosity decreases as the shear rate increases at high initial force. This means from the technical point of view that the energy after the initial force needed for the flow rate can be reduced. Typical fluids with above described characteristics are a.o. gels, Latex, lotions.

Dilatent fluids

Viscosity increases as the shear rate increases. Example: pulp, sugar mixture

Thixotropic fluids

Viscosity decreases with strong shear rate (I) and increases again as the shear rate decreases (II). The ascending curve is however not identical to the descending curve. Typical fluids are a.o. soap, Ketchup, glue, peanut butter

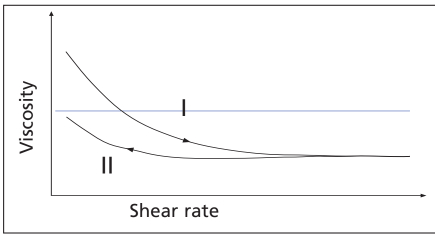


Fig. 7 - Thixotropic fluids

4 Hydraulic Fundamentals

Pumps shall produce pressure. Fluids are conveyed over a certain distance by kinetic energy produced by the pump.

4.1 Pressure

The basic definition of pressure (p) is the force per unit area. It is expressed in this Manual in Newton per square meter ($N/m^2 = Pa$).

$$1 \text{ bar} = 10^5 \frac{N}{m^2} = 10^5 \text{ Pa}$$

4.2 Atmospheric pressure

Atmospheric pressure is the force exerted on a unit area by the weight of the atmosphere. It depends on the height above sea level (see Fig. 8). At sea level the absolute pressure is approximately 1 bar = 10^5 N / m^2.

Gage pressure uses atmospheric pressure as a zero reference and is then measured in relation to atmospheric pressure. Absolute pressure is the atmospheric pressure plus the relative pressure.

Height above sea level m	Air pressure p_b bar	Boiling temperature °C
0	1,013	100
200	989	99
500	955	98
1,000	899	97
2,000	795	93

Fig. 8 - Influcence of the topographic height

4.3 Relation of pressure to elevation

In a static liquid the pressure difference between any two points is in direct proportion to the vertical distance between the two points only.

The pressure difference is calculated by multiplying the vertical distance by density.

In this Manual different pressures or pressure relevant terms are used. Here below are listed the main terms and their definitions:

Static pressure	Hydraulic pressure at a point in a fluid at rest.
Friction loss	Loss in pressure or energy due to friction losses in flow
Dynamic pressure	Energy in a fluid that occurs due to the flow velocity.
Delivery pressure	Sum of static and dynamic pressure increase.
Delivery head	Delivery pressure converted into m liquid column.
Differential pressure	Pressure between the initial and end point of the plant.

4.4 Friction losses

The occurance of friction losses in a pipe system is very complex and of essential importance when selecting the pump. Friction losses in components caused by the flow in the pipe system (laminar flow and turbulent flow) are specified by the pump manufacturer.

There are two different types of flow
Laminar flow is characterized by concentric layers moving in parallel down the length of the pipe, whereby highest velocity is found in the centre of the pipe, decreasing along the pipe wall (see Fig. 9). Directly at the wall the velocity decreases down to zero. There is virtually no mixing between the layers. The friction loss is proportional to the length of the pipe, flow rate, pipe diameter and viscosity.

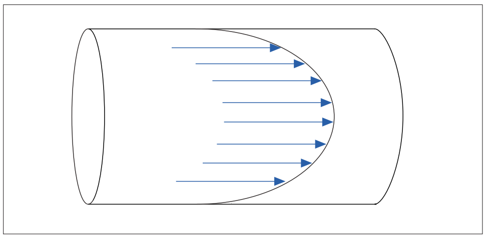


Fig. 9 - Laminar flow

In case of **turbulent flow** strong mixing takes place between the layers whereby the the velocity of the turbulences is extremely high.
Turbulent flow occurs mainly in low viscous fluids and is characterised by higher friction losses. The friction losses behave proportional to the length of the pipe, square flow rate, pipe diameter and viscosity.

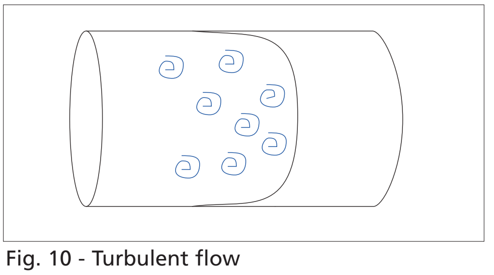


Fig. 10 - Turbulent flow

4.5 Reynolds number

In transition between laminar flow and turbulent flow there is a multitude of so called „mixed flows“. They are characterised by a combination of properties of the turbulent flow and the laminar flow. For determination and simple computing of the specific characteristics the Reynolds number was introduced. This dimensionless number is the ratio of fluid velocity multiplied by pipe diameter, divided by kinematic fluid viscosity.

$$Re = v \times DN / \nu$$

Re = Reynolds number
v = Fluid velocity (m/s)
DN = Pipe diameter
ν = Kinematic fluid viscosity

General:

Laminar flow - if $Re < 2320$
Turbulent flow - if $Re \geq 2320$

5 Technical Fundamentals

This Manual helps carrying out the optimal design of centrifugal pumps. We show you how to proceed to find the right pump.

5.1 Installation

Install the pump in close vicinity to the tank or to another source from which the liquid will be pumped. Make sure that as few as possible valves and bends are integrated in the pump's suction pipe, in order to keep the pressure drop as low as possible. Sufficient space around the pump provides for easy maintenance work and inspection. Pumps equipped with a conventional base plate and motor base should be mounted on a steady foundation and be precisely aligned prior commissioning.

5.2 Pipe connection

GEA Tuchenhagen pumps are equipped with pipe connections that are adaped to the flow rate. Very small pipe dimensions result in low cost on one hand, but on the other hand put the safe, reliable and cavitation-free operation of the pump at risk.
Practical experience has shown that identical connection diameters on a short suction pipe are beneficial, however, always keep an eye on the fluid velocity. Excepted thereof are long suction pipes with integrated valves and bends. In this case the suction pipe should be by one size larger, in order to reduce the pressure drop.

The pipes connected to the pump should always be supported in a way that no forces can act on the pump sockets. Attention must be paid to thermal expanson of the pipe system.
In such a case, expansion compensators are recommended.
As long as the pump is mounted on calotte-type feet, the pump will be able to compensate slight pipe length expansions.
If the pump is rigid mounted on to a base plate, compensation must be ensured by the pipe system itself, using pipe bends or suitable compensators.

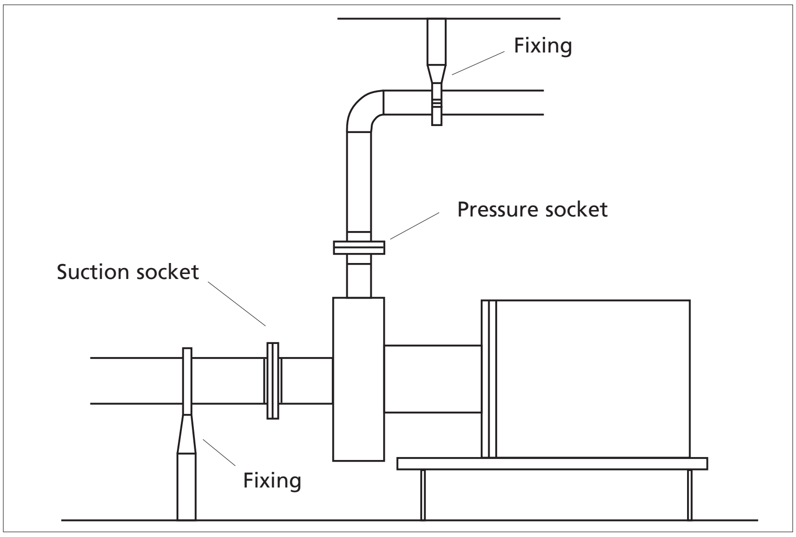


Fig. 11 - Pipe support

5.3 Suction pipe

It is important for most of the pumps - but especially for non-selfpriming centrifugal pumps - that no air is drawn into the pump - as otherwise this would impair the pump performance. In the worst case the pump would stop pumping. Therefore the tanks should be designed and constructed in a way that no air-drawing turbulences occur. This can be avoided by installing a vortex breaker into the tank outlet.
The locaton of the pump as well as the connection of the suction pipe must not cause the formation of air bubbles. When planning the suction pipe, sufficient length must be provided upstream the pump. This section should be in length at least five times the diameter of the inlet socket (Fig. 12).

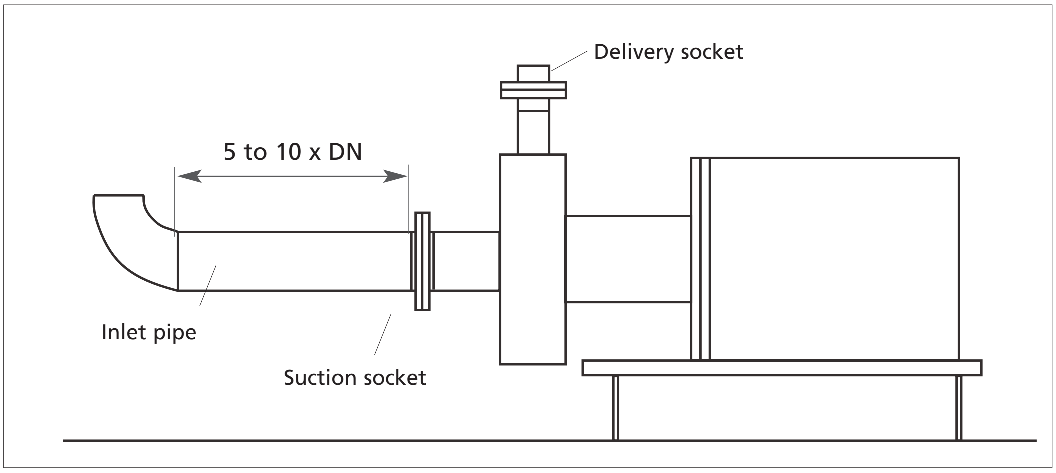


Fig. 12 - Distance to the inlet socket

5.4 Delivery pipe

Normally valves, heat exchangers, filters und other components are installed in the delivery pipe. The flow head results from the resistance of the components, the pipe and the geodetic difference. Flow rate and flow head can be influenced via the control fittings installed in the delivery pipe.

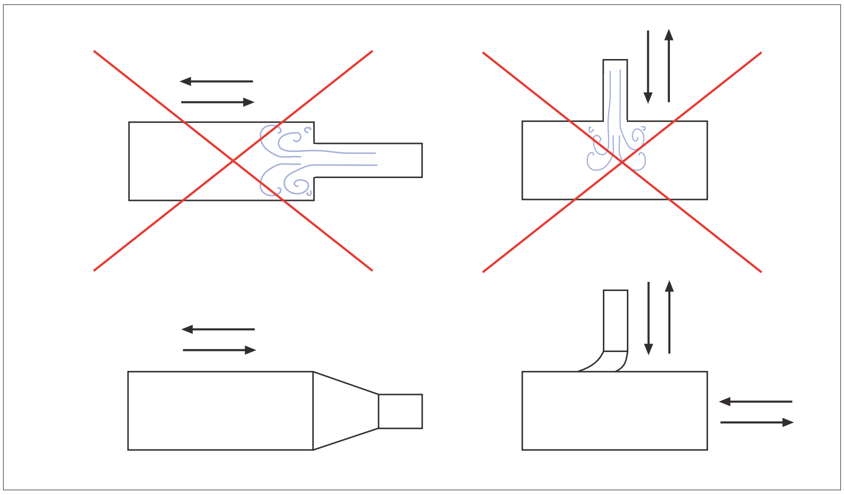


Fig. 13 - Right and wrong connection of a pipe

5.5 NPSH

NPSH (Net Positive Suction Head) is the international dimension for the calculation of the supply conditions.

For pumps the static pressure in the suction socket must be above the vapour pressure of the medium to be pumped. The NPSH of the pump is determined by measurements carried out on the suction and delivery side of the pump. This value is to be read from the pump characteristic curve and is indicated in meter (m). The NPSH is in the end a dimension of the evaporation hazard in the pump inlet socket and is influenced by the vapour pressure and the pumped liquid. The NPSH of the pump is called NPSH **req**uired, and that of the system is called NPSH **av(ai)l**able. The $NPSH_{avl}$ should be greater than the $NPSH_{req}$ in order to avoid cavitation.

$$NPSH_{avl} > NPSH_{req}$$

For safety reasons another 0.5 m should be integrated into the calculation, i.e.:

$$NPSH_{avl} > NPSH_{req} + 0.5m$$

5.6 Suction and supply conditions

Troublefree operation of centrifugal pumps is given as long as steam cannot form inside the pump; in other words: if cavitation does not occur. Therefore, the pressure at the reference point for the NPSH must be at least above the vapour pressure of the pumped liquid. The reference level for the NPSH is the centre of the impeller so that for calculating the $NPSH_{avl}$ according to the equation below, the geodetic flow head in the supply mode ($H_{z,geo}$) must be set to positive and in the suction mode ($H_{s,geo}$) to negative

$$NPSH_{avl} = \frac{p_e + p_b}{\rho \times g} - \frac{p_D}{\rho \times g} + \frac{v_e^2}{2g} - H_{v,s} + H_{s,geo}$$

p_e	=	Pressure at the inlet cross section of the system
p_b	=	Air pressure in N/m² (consider influence of height)
p_D	=	Vapour pressure
ρ	=	Density
g	=	Acceleration of the fall
v_e	=	Flow speed
$H_{v,s}$	=	Sum of pressure drops
$H_{s,geo}$	=	Height difference between liquid level in the suction tank and centre of the pump suction socket

At a water temperature of 20 °C and with an open tank the formula is simplified:

$$NPSH_{avl} = 10 - H_{v,s} + H_{z,geo}$$

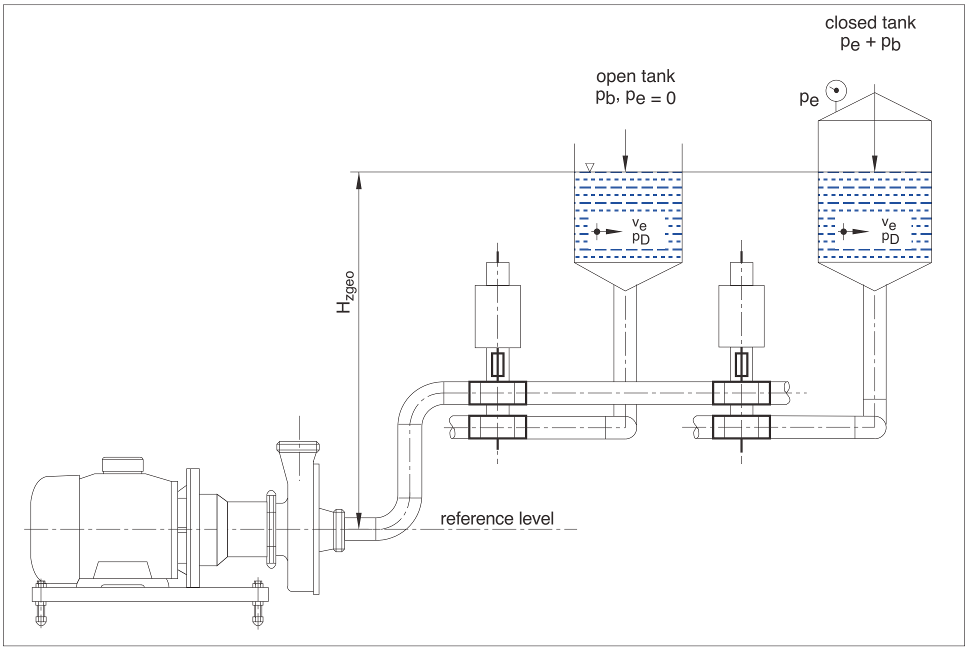


Fig. 14 - Pumping system

5.7 Cavitation

Cavitation produces a crackling sound in the pump. Generally spoken is cavitation the formation and collapse of vapour bubbles in the liquid. Cavitation may occur in pipes, valves and in pumps. First the static pressure in the pump falls below the vapour pressure associated to the temperature of a fluid at the impeller intake vane channel. The reason is in most of the cases a too low static suction head. Vapour bubbles form at the intake vane channel. The pressure increases in the impeller channel and causes an implosion of the vapour bubbles. The result is pitting corrosion at the impeller, pressure drops and unsteady running of the pump. Finally cavitation causes damage to the pumped product.

Cavitation can be prevented by :

1. Reducing the pressure drop in the suction pipe by a larger suction pipe diameter, shorter suction pipe length and less valves or bends
2. Increasing the static suction head and/or supply pressure, e.g. by an upstream impeller (Inducer)
3. Lowering the temperature of the pumped liquid

5.8 Q-H characteristic diagram

Before designing a pump, it is important to ascertain the characteristic curve of the plant that allows you to select the right pump by help of the pump characteristic curve

The operating performance of centrifugal pumps is rarely represented in the form of tables, but mainly in the form of characteristic curves (Fig. 15). These pump characteristic curves are measured at line machines at constant speed and show the flow rate (Q in m3/h) and the flow head (liquid column in m) of the pump. The flow head H of a pump is the effective mechanical energy transferred by the pump to the pumped liquid, as a function of the weight force of the pumped liquid (in m liquid column). It is independent of the density (r) of the pumped liquid; that means a centrifugal pump transfers liquids regardless of the density up to the same flow head. However, the density must be taken into account for the determination of the power consumption P of a pump.
The actual flow head of the pump is determined by the flow rate H_A of the plant, which consists of the following components:

$$H_A = H_{geo} + \frac{p_a - p_e}{\rho \times g} + \frac{v_a^2 - \overset{0}{v_e^2}}{2g} + \Sigma H_v$$

H_{geo} geodetic flow head = the difference in height to overcome between the liquid level of the suction and the delivery side

$\frac{p_a - p_e}{\rho \times g}$ difference of pressure heights between liquid level of the suction and delivery side with closed tanks

$\frac{v_a^2 - \overset{0}{v_e^2}}{2g}$ speed difference (can be neglected in practice)

ΣH_V sum of pressure drops (pipe resistances, resistance in fittings and formed parts in suction and delivery pipes)

5.9 ➤ Flow rate

The flow rate (Q) accrues from the requirements of the process plant and is expressed in m^3/h or GPM (Gallons per minute).

5.10 ➤ Flow head

A decisive factor in designing a pump is the flow head (H), that depends on:

- the required flow head (for instance of a spray ball of 10 to 15 m; equal to 1.0 to 1.5 bar),
- difference in the pressure height of a liquid level on the delivery side and suction side,
- the sum of pressure drops caused by pipe resistance, resistance in components, fittings in the suction and delivery pipe.

5.11 ➤ Plant characteristic curve

The graphical representation of the flow head of a plant (H_A) in dependance of the flow rate (Q) is the characteristic curve of a pipe or plant. It consists of a static portion that is is independent of the flow rate and a dynamic portion in square with rising flow rate .

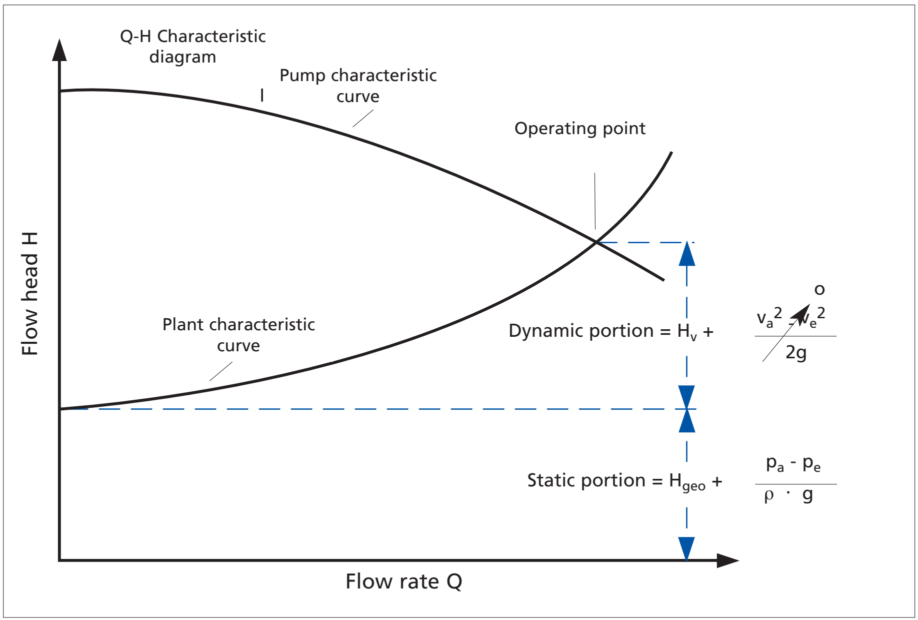


Fig. 15 - Q-H Characteristic diagram

5.12 ➤ Operating point

The operating point of a pump is the intersection of a pump characteristic curve with the plant characteristic curve.

5.13 Pressure drops

Essential for the design of a pump are not only the NPSH, flow head and flow rate, but also pressure drops.

Pressure drops of a plant may be caused by pressure drops in:
• the pipe system,
• installed components (valves, bends, inline measurement instruments),
• installed process units (heat exchangers, spray balls).

Pressure drops H_v of the plant can be determined by help of tables and diagrams.
Basis are the equations for pressure drops in pipes used for fluid mechanics that will not be handled any further.
In view of extensive and time-consuming calculation work, it is recommended to proceed on the example shown in Chapter 6.1. The tables in Chapter 8.2 and 8.3 help calculating the equivalent pipe length.
The data is based on a medium with a viscosity ν = 1 mm^2/s (equal to water).
Pressure drops for media with a higher viscosity can be converted using the diagrams in the annexed Chapter 8.5.

5.14 Theoretical calculation example

Various parmeters of the pipe system determine the pump design. Essential for the design of the pump is the required flow head. In the following, the three simplified theoritical calculation examples shall illustrate the complexity of this subject before in Chapter 6 the practical design of a pump is handled.

H_v	=	Pressure drop
$H_{v,s}$	=	Total pressure drop - suction pipe
$H_{v,d}$	=	Total pressure drop - delivery pipe
$H_{s,geo}$	=	Geodetic head - suction pipe
$H_{z,geo}$	=	Geodetic head - supply pipe
$H_{d,geo}$	=	Geodetic head - delivery pipe
$H_{v,s}$	=	Pressure drop - suction pipe
$H_{v,d}$	=	Pressure drop - delivery pipe
p	=	Static pressure in the tank

Attention:
Pressure in the tank or supplies in the suction pipe are negative because they must be deducted from the pressure drop. They intensify the flow.

Example 1 - Negative supply

$H_{d,geo}$ = 25 m
$H_{v,d}$ = 10 m
$H_{s,geo}$ = 6 m (suction pressure)
$H_{v,s}$ = 3 m

$H_{v,d}$ = $H_{d,geo} + H_{v,d}$ = 25 m + 10 m = 35 m
$H_{v,s}$ = $H_{s,geo} + H_{v,s} + p$ = 6 m + 3 m + 0 m = 9 m

H_v = $H_{v,d} + H_{v,s}$ = 35 m + 9 m = 44 m

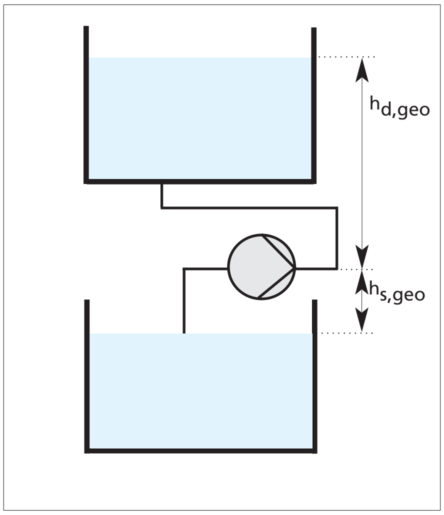


Fig. 16 - Negative supply

Example 2 - Supply under atmospheric pressure

$H_{d,geo}$ = 10 m
$H_{v,d}$ = 5 m
$H_{z,geo}$ = - 3 m (supply pressure)
$H_{v,s}$ = 2 m

$H_{v,d}$ = $H_{d,geo} + H_{v,d}$ = 10 m + 5 m = 15 m
$H_{v,s}$ = $H_{z,geo} + H_{v,s} + p$ = -3 m + 2 m + 0 m = -1 m

H_v = $H_{v,d} + H_{v,s}$ = 15 m - 1 m = 14 m

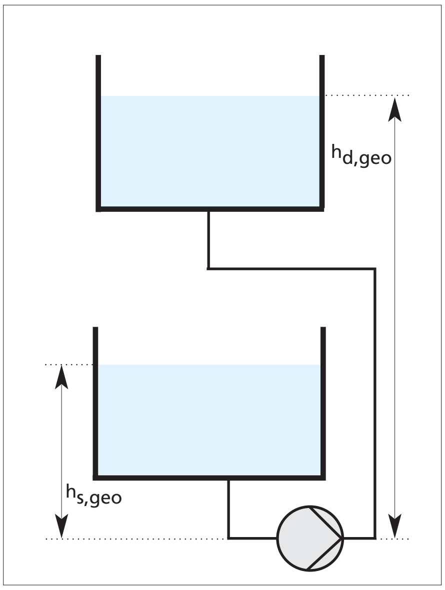


Fig. 17 - Supply under atmospheric pressure

Example 3 - Supply from pressure tank

$H_{d,geo}$ = 15 m
$H_{v,d}$ = 3 m
$H_{z,geo}$ = -2 m
$H_{v,s}$ = 1 m
p = 8 m

$H_{v,d}$ = $H_{d,geo} + H_{v,d}$ = 15 m + 3 m = 18 m
$H_{v,s}$ = $H_{z,geo} + H_{v,s} + p$ = -2 m + 1 m + (-8 m) = -9 m

H_v = $H_{v,d} + H_{v,s}$ = 18 m + (-9 m) = 9 m

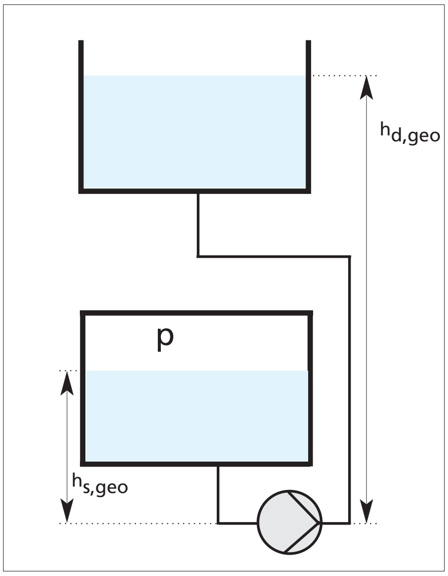


Fig. 18 - Supply from pressure tank

6 Design of Centrifugal Pumps

By help of the example below and the annexed summarised diagrams and tables all the centrifugal pumps can be designed. The tables contain Tuchenhagen specific valves and pipe fittings. For the calculation of pressure drops in a plant, the conversion principle of the measured friction factor (ζ) of valves and fittings in metre equivalent pipe length is applied.

6.1 Practical calculation example

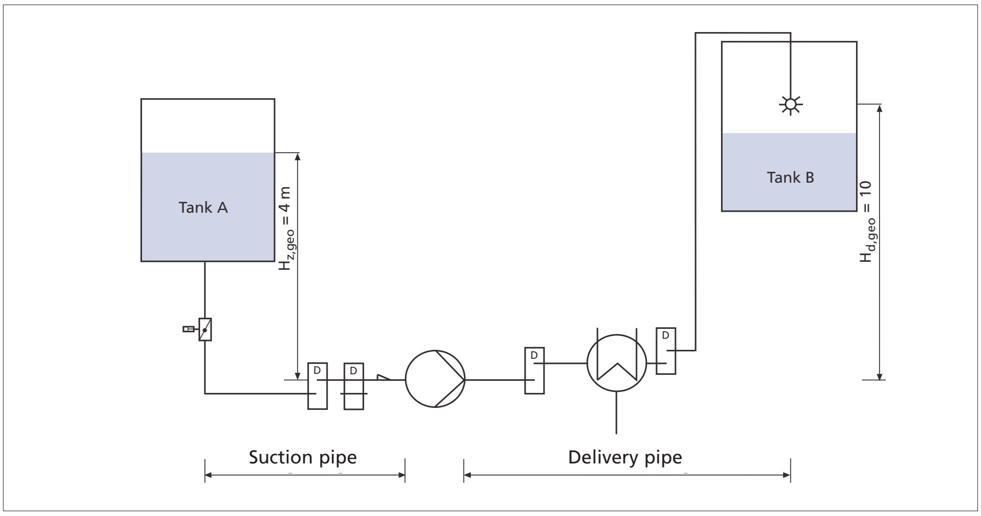


Fig. 19 - Pressure drop in a plant

6.1.1 Calculation

Pressure drop of the plant

$$H_A = H_{geo} + \frac{p_a - \overset{0}{\cancel{p_e}}}{\rho \times g} + \frac{v_a^2 - \overset{0}{\cancel{v_e^2}}}{2 \times g} + \Sigma H_v$$

$$H_{geo} = H_{d,geo} - H_{z,geo} = 10\ \text{m} - 4\ \text{m} = \underline{6\ \text{m}}$$

$$\Sigma H_v = H_{v,s} + H_{v,d}$$

$H_{v,s}$

1 Tank outlet	= 0.8 m eqv.pipe"
1 Double seat valve DN 65 flow through (seat)	= 22.5 m "
1 Double seat valve DN 65 flow through (housing).	= 2.9 "
1 Reducer DN 65	= 0.2 m
5 Bends 90° DN 65	= 5 x 0.6 m
10 m pipe DN 65	<u>10.0 m</u> see Page 37
	Σ = 40.2 m Page 36 (and 40 - 44)

Pressure drop H_v at 24 m³/h DN 65

$$40{,}2 \times \frac{6.5\ \text{m}}{100\ \text{m}} = 2{,}62$$

$H_{v,s}$ = <u>2.6 m</u>

$H_{v,d}$

1 Double seat valve DN 50 flow through (seat)	=10.5 m eqv. pipe
1 Normal valve DN 50 flow through (seat)	= 2.2 m "
10 Bends 90° DN 50	= 10 x 0.45 m "
20 m pipe DN 50	<u>20.0 m</u> see Page 37
	Σ = 37.2 m Page 36 (and 40 - 44)

Pressure drop H_v at 24 m³/h DN 50

$$37{,}2 \times \frac{25.0\ \text{m}}{100\ \text{m}} = 7{,}44$$

Heat exchanger at 24 m³/h	= 12.0 m
Spray ball at 24 m³/h	= <u>5.0 m</u>
	24.4 m =>

$H_{v,d}$ = <u>24.4 m</u>

$H_A = H_{geo} + H_{v,s} + H_{v,d}$
= 6 m + 2.6 m + 24.4 m
H_A = <u>33 m</u>

6.1.2 ➤ Explanations

The flow rate is 24 m³/h. Components and process units are installed in the pipe between Tank A to be emptied and Tank B to be filled. As already mentioned before, it is essential to install the pump as close as possible to the tank to be emptied.
Between Tank A and the pump are located a butterfly valve and two double seat valves as well as one reducer and 5 bends and finally 10 m pipe in DN 65.

In the pipe from the pump up to Tank B (20 m in DN 50) are installed a double seat valve, a single seat valve, one heat exchanger and one spray ball. The difference in elevation of the liquid level in Tank A to Tank B is 6 m. Now the metre equivalent pipe length must be determined for each component installed. For this purpose see the standard tables for pressure drops on Page 37 and 38. The outcome is in total 40.18 m on the suction side. This value is converted into the corresponding pressure drop (H) of the pipe, cross section DN 65. According to the table, the pressure drop is 6.5 m per 100 m at a flow rate of 24 m³/h and with a pipe DN 65. Based on 40.18 m, the pressure drop ($H_{v,s}$) is 2.61 m. Downstream the pump, the liquid must be conveyed in length equivalent pipe of 37.2 m in total. The pressure drop of a pipe in DN 50 is according to the table 25 m per 100 m. Based on 37.2 m, the pressure drop is 7.4 m. In addition, on the delivery side there is a heat exchanger with a pressure drop of 12 m (at 24 m³) as well as a spray ball at the end of the pipe with a pressure drop of 5 m.
In total the pressure drop on the delivery side ($H_{v,d}$) is 24.4 m.

The sum of pressure drops on the suction side ($H_{v,s}$), on the delivery side ($H_{v,d}$) and the geodetic flow head (H_{geo}), result in a total pressure drop (H_A) of 33.0 m that must be compensated by the pump.

6.1.3 ➤ Calculation of the NPSH

The next step is the calculation of the NPSH of the plant that finally complete the parameters needed for the design of your pump.
The calculation of the NPSH takes only the suction pipe into consideration.

$$NPSH_{avl} = \frac{p_e + p_b}{\rho \times g} - \frac{p_D}{\rho \times g} + \cancelto{0}{\frac{v_e}{2g}} - H_{v,s} + H_{z,geo}$$

=	10 m	- 2.0 m	- 2.6 m	+ 4 m	= 9.4 m
		Vapour pressure at 60°C from page 38	Flow head	static suction head	$NPSH_{avl}$

$$NPSH_{avl} = 9.4 \text{ m must be above the } NPSH_{pump}$$

The calculated NPSH of the plant is 9.4 m and must be above that of the pump.
Using this data now available, the plant characteristic curve can be ascertained.

6.2 Characteristic curve interpretation

The flow rate, flow head, the required motor power, the NPSH and efficiency of the pump are indicated in the pump characteristic.
On the example shown below it is explained how a pump characteristic is to interprete.

Values ascertained so far (from Chapter 6.1):

Flow rate	=	24.0 m³/h
Req. flow head	=	33.0 m
$NPSH_{avl}$	=	9.4 m

These are the relevant values for finding out the optimal pump by use of diagrams.

Step 1

The first diagram to be used is the Q/H Diagram (Fig. 20 - the diagram of a TP 2030). First the intersection point of the flow rate (24 m³/h) with flow head (33 m) should be made out. The intersection point is located in the area of the impeller of 160 mm in diameter.

Step 2

The pump efficiency (η) is read in Fig. 20 and amounts to approximately 57 %.

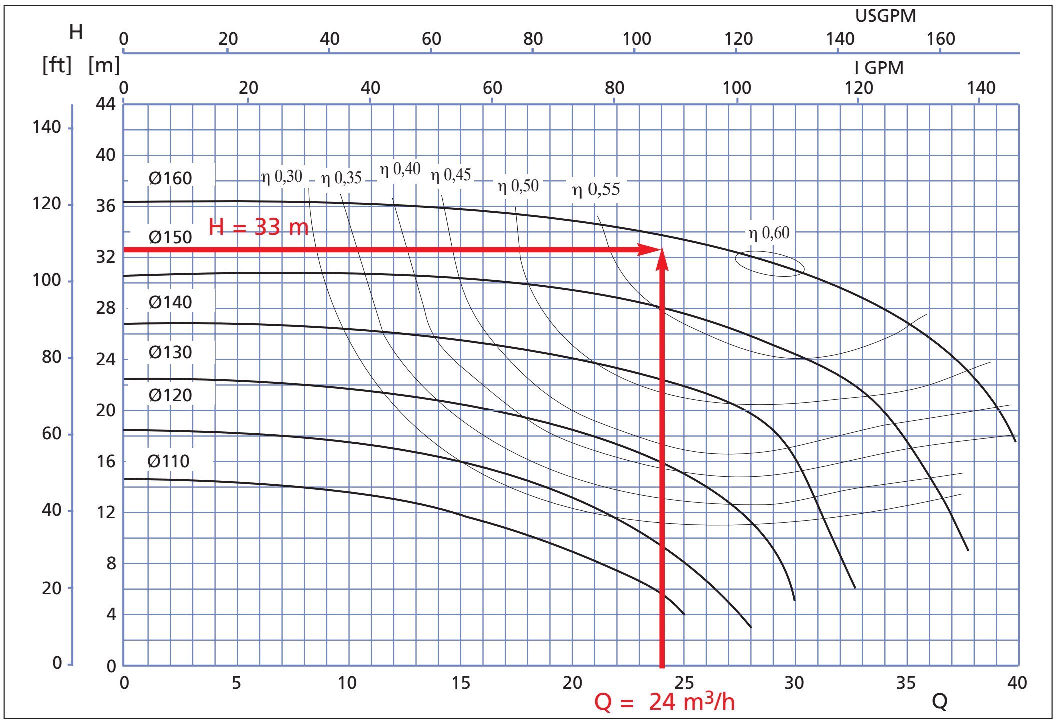


Fig. 20

Step 3

The NPSH/Q Diagram (Fig. 21) shows the $NPSH_{req}$, that amounts to 1.9 m.

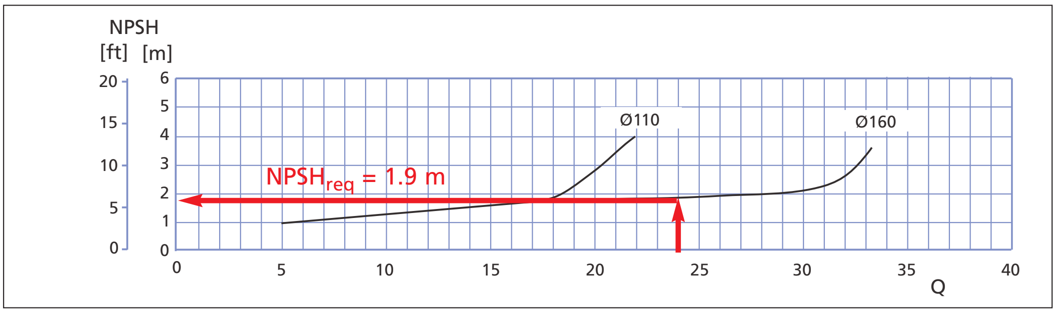


Fig. 21

Step 4

The impeller diameter of 160 mm is required in order to read out the required motor power in the Q/P Diagram (Fig. 22). Accordingly, at a flow rate of 24 m³/h the motor power is 3.7 kW. Fluctuations in volume and pressure must be expected in the plant and consequently fluctuations of the operating point, that causes variation of the power consumption P of the pump. This is the reason why in principle an increased factor of 5% is fixed.
The result is that the motor size should be at least to 4 kW (the required 3.7 kW plus increased safety). The next larger sized standard motor has 4 kW and should therefore be selected.

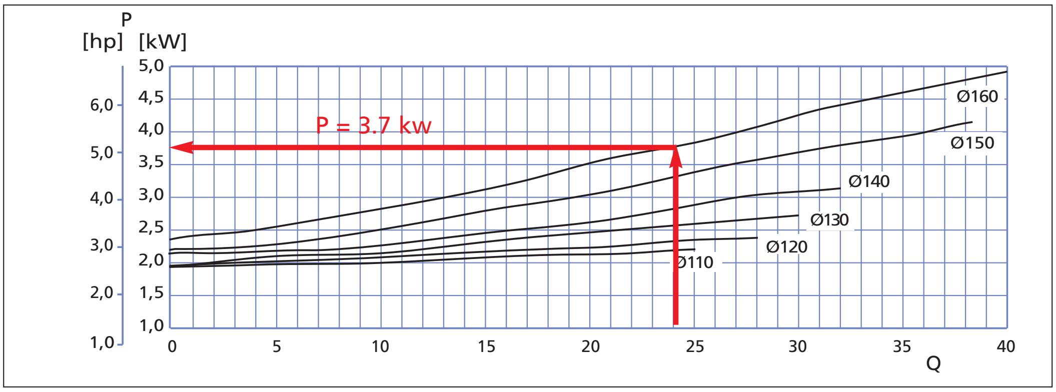


Fig. 22

The power consumption of a pump can also be calculated using the formula

$$P = \frac{\rho \times Q \times H}{\eta \times 367}$$

and using the diagrams, the missing parameters for the optimal pump design are made available.

The required flow rate of 24 m³/h and the specified flow head of 33 m require the use of the pump TP 2030 with an impeller diameter 160 mm and 4 kW motor capacity at
n = 2,900 rpm and 50 Hz.
The efficiency of this pump is about 57 % and the NPSH of the pump (1.9 m) does not exceed the NPSH of the plant (9.4 m > 1.9 + 0.5 m) so that cavitation does not occur.
Accordingly, the pump is suitable for the application in question.

6.3 Modification

**In the previous example the pump design took place in four steps.
In practice, however, pumps are used at different operating points. These may be pumping of viscous media, temperature changes or systems with integratation of pressurised tanks.**

6.3.1 Throttling

Changes in the flow head of a system H_A (throttling) are realised in practice by increasing or reducing the resistance on the delivery side of the pump, e.g. by installing a throttling valve. In this case the operating point is always located on the intersection of the plant characterstic curve with the pump characteristic curve.

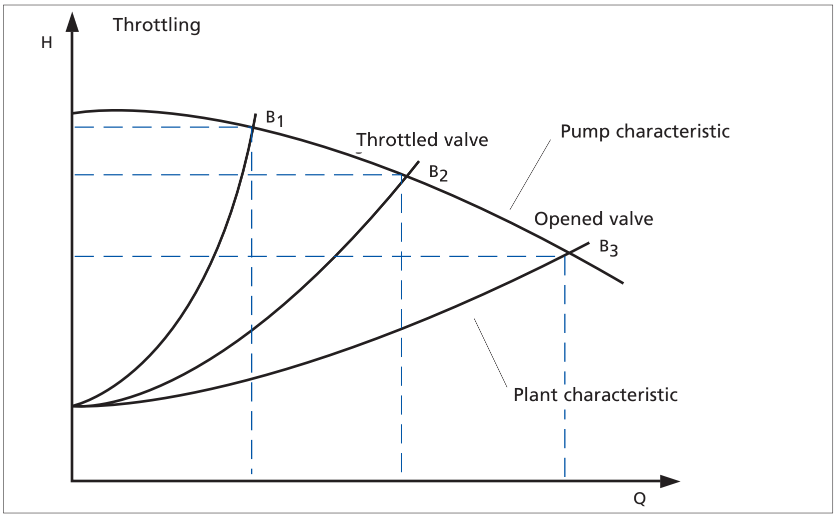
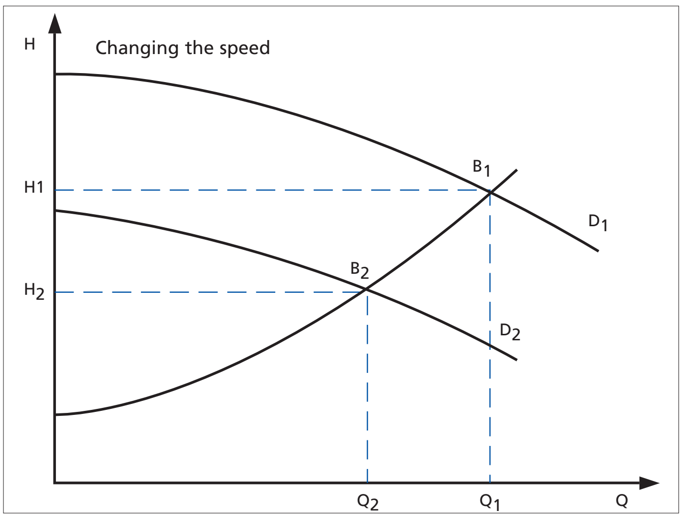


Fig. 23 - Throttling

6.3.2 Changing the speed

Changing the speed (n) causes a change of the operating point and thus of the flow rate (Q) and the flow head (H). For this purpose a frequency converter or a pole changing motor is needed. In spite of the high purchase costs for a frequency converter, its use is in view of the operating costs the clearly more favourable alternative to the throttling process with a throttling valve. Speed control is used, if different operating points shall be achieved, e.g. for product and cleaning liquid.



$$Q_2 = \frac{n_2}{n_1} \times Q_1$$

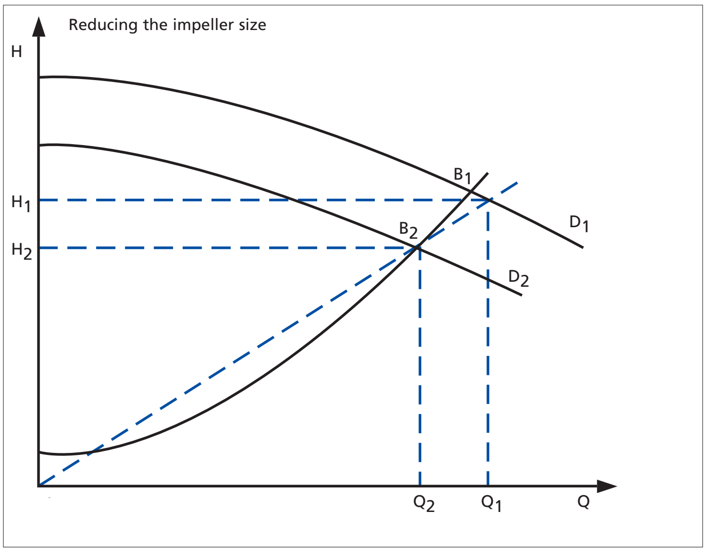
$$H_2 = \left[\frac{n_2}{n_1}\right]^2 \times H_1$$

$$P_2 = \left[\frac{n_2}{n_1}\right]^3 \times P_1$$

Fig. 24 - Changing the speed

6.3.3 Reducing the impeller size

Tuchenhagen offers for each pump different impeller sizes. It may happen that the best efficiency point of the impeller is located between two characteristic curves. The impeller will then be turned to size in order to obtain the required diameter.
This is both the most simple and favourable method.



$$\left[\frac{D_1}{D_2}\right]^2 \approx \frac{Q_1}{Q_2} \approx \frac{H_1}{H_2}$$

Fig. 25 - Reducing the impeller size

6.3.4 Operation in parallel

Two pumps can be operated in parallel, if the desired operating point cannot be reached with only one pump. In such a case the flow of the two pumps are added while the flow head remains unchanged.

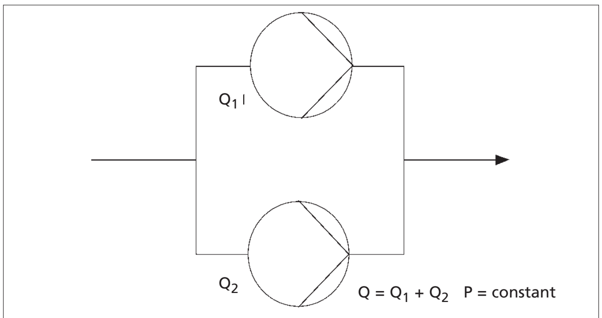


Fig. 26 - Operation in parallel

6.3.5 Operation in series

If the required flow head cannot be achieved by one pump only, two pumps are connected in series. Thus the flow head is doubled at constant flow rate.

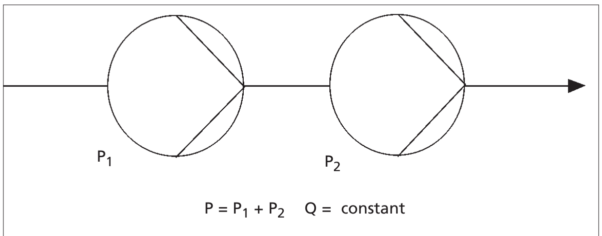


Fig. 27 - Operation in series

6.4 Pumping of viscous media

In the previous example (Chapter 6.1) water served as pumping medium. In practice media other than water are conveyed. In this respect viscosity is a factor that must be taken into account for the calculation and design of the pump.

Conveying liquids of higher viscosity (ν) at constant speed (n), reduce the flow rate (Q), flow head (H) and the efficiency (η) of the pump, while power consumption P_z of the pump (see Fig. 28) increases tt the same time. According to the method of approxima-tion, (6.4.2) the suitable pump size can be determined, starting from the operating point for viscous liquids via the operating point for water. The pump's power consumption depends on the efficiency of the complete unit.

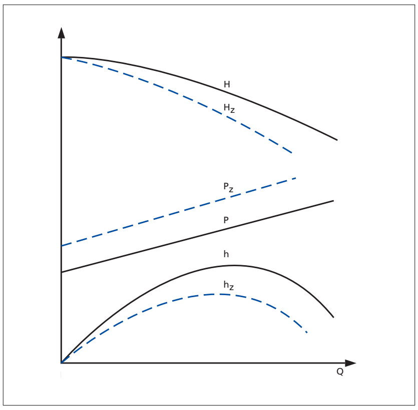


Fig. 28

Annexed are tables used for the determination of pressure drops in dependence of viscosity and pipe diameter. In this connection it is worthwhile to menti-on that the pressure drop in dependence of viscosity is irrelevant for centrifugal pumps and can therefore be neglected. Centrifugal pumps are suitable for liquids up to a viscosity of 500 mm²/s.
If it is the question of pumping viscous media such as quarg, butter or syrup, rotary piston pumps will be used due to their higher efficiency in this respect.

6.4.1 Correction for high viscosities

The following page shows an example that explains the calculation and design of a pump used for viscous media. Decisive in this connection are the correction factors for the flow head (K_H), flow rate (K_Q) and the pump efficiency (K_η).

The correction factors are found in the diagram on page 31, by proceeding in the following steps:

1. Find out the kinematic viscosity of the medium in mm²/s
2. Determine product of Q x √H (m³/h √m)
3. Set up a vertical at the intersection of Q x √H with the corresponding viscosity
4. Reading the intersections with the three correction lines at the vertical
5. Enter these values into the equations and calculate the corrected value

On the basis of the obtained values, the pump can be desigend by means of the pump characteristic for water, (see Chapter 6.2).

6.4.2 Calculation of correction factors

Pumping medium:	Oil		
Flow rate:	Q	=	24 m3/h
Flow head:	H	=	33 m
Viscosity:	ν	=	228 mm²/s
Density:	ρ	=	0.9 t/m3
Efficiency:	η	=	0.55 %

A vertical is set up cutting K_H, K_Q and K_η at the intersection of the horizontal viscosity line coming from the left side with the diagonal Q x √H line.
From each of the newly created intersections, a horizontal leads to the right hand side, on to the correction factors. The reading is: $K_Q = 0.83$, $K_H = 0.84$, $K_\eta = 0.47$

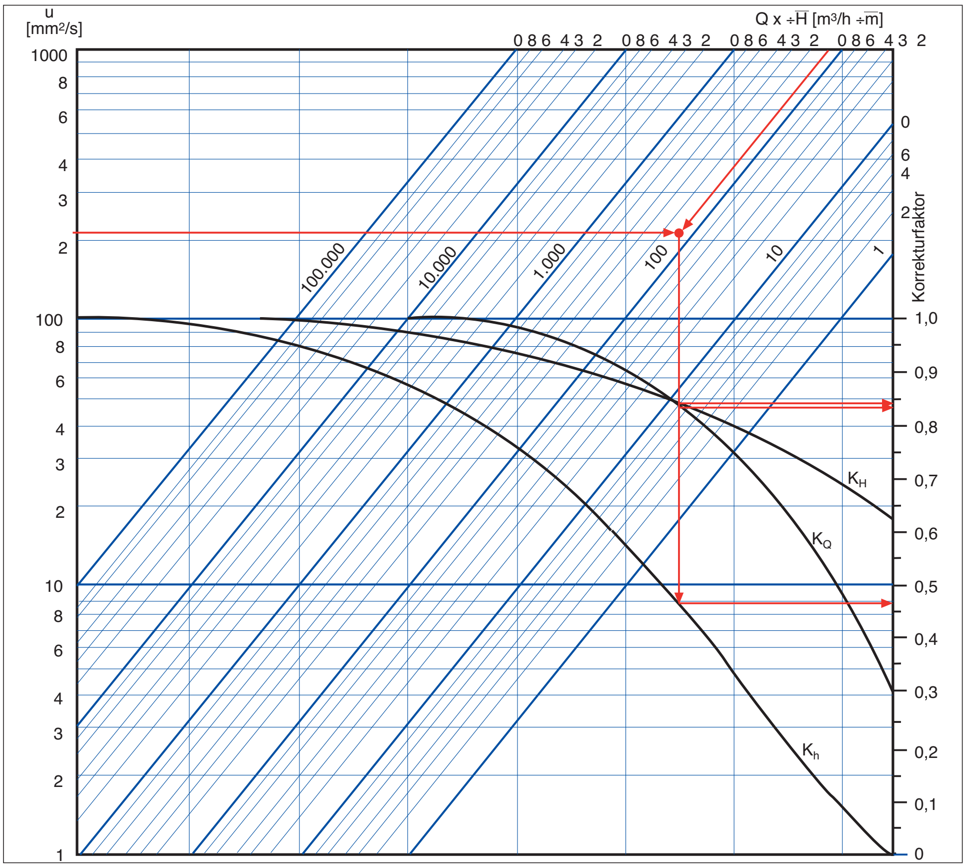


Fig. 29 - Digram correction factors

The pump should be designed for the following pump data based on water:

$$Q = \frac{Q_z}{K_Q} = \frac{24}{0.83} = 28.9 \text{ m}^3/\text{h}; \qquad H = \frac{H_z}{K_H} = \frac{33}{0.84} = 39.29 \text{ m}$$

Fill into the formula for the power consumption (P_z), the efficiency (η) from the „water flow head diagram“.

$$\text{power consumpt. } P_z = \frac{Q_z \times H_z \times \rho}{367 \times K_\eta \times \eta}$$

$$= \frac{24 \times 33 \times 0{,}9}{367 \times 0.47 \times 0.55} = 7.52 \text{ kW}$$

Higher accuracy is achieved by repeating the procedure with the data obtained.
Result: After correction using the factors K_Q, K_H and K_η, a pump must be selected for pumping oil and a flow head of 24 m3/h that is capapable of achieving 29 m3/h and 39 m flow head. The required motor power is at least 7.5 kW.

7 Design of Rotary Lobe Pumps

7.1 Fundamentals

GEA Tuchenhagen Rotary Lobe Pumps of the VPSU and VPSH series are rotating positive displacement pumps. Two rotors with two lobes each rotate in the pump housing in opposite direction creating a fluid movement through the pump.
The rotors do neither come in contact with each other nor with the pump housing.

A positive pressure difference is generated between the pump's delivery and suction sockets when the liquid is conveyed. A part of the pumped medium flows back from the delivery side to the suction side through the gap between the two rotors and the pump housing. The flow rate - theoretically resulting from the volume of the working areas and the pump speed - is reduced by the volume of the back flow. The back flow portion rises with increasing delivery pressure and decreases as the product viscosity rises.
The capacity limits of rotary lobe pumps are usually revealed when rating the pump. They are reached, if one of the parameters needed for the pump design cannot be determined (e.g. speed), or if the NPSH of the pump is above or equal to that of the plant. In such a case the next bigger pump size should be selected for safety reasons.

Rotary lobe pumps, type VPSH and VPSU are positive displacement pumps. Pumping against a closed delivery side will result in an intolerable rise of pressure that can destroy the pump or other parts of the plant. If pumping against a closed delivery side cannot be excluded to the full extent, safety measures are to be taken either by suitable flow path control or by the provision of safety or overflow valves. GEA Tuchenhagen offers overflow valves of the VARIVENT® System, series "Q" that are either ready mounted (including by-pass from the delivery to the suction side) or as a loose part for installa-tion on site.

7.2 Pump rating conditions

In the following the rating of GEA Tuchenhagen rotary lobe pumps, type VPSH and VPSU shall be explained on an example.

For the selection of a pump that suits the specific application the following parameters should be known:

Product data
- Medium
- Temperature
- Product viscosity
- Density
- Portion of particles
- Particle size

Pumping data
- Volume flow Q in l/min.
- Differential pressure Δp in bar
- NSPH of the plant in bar

7.3 ➤ Example

Product data

- Medium: Yeast
- Temperature: t = 10°C
- Viscosity: η = 100 cP
- Density: ρ = 1,000 kg/m^3
- Portion of particles none
- Particle size none
- Single acting mechanical seal in the pairing carbon against stainless steel

Pumping data

- Q = 300 l/min.
- Δp = 5 bar
- NPSH: 0.4 bar abs.

Selection of the pump size

The characteristic diagram (Fig. 30) may be used as a first step for the preliminary selection of the pump type in question. For this purpose the required flow rate and the known viscosity of the product are to be entered. According to this diagram, the VPSH 54 is the suitable pump size. Now the pump can be examined more closely using the detailed characteristic diagram (Fig. 31) shown on the next page.

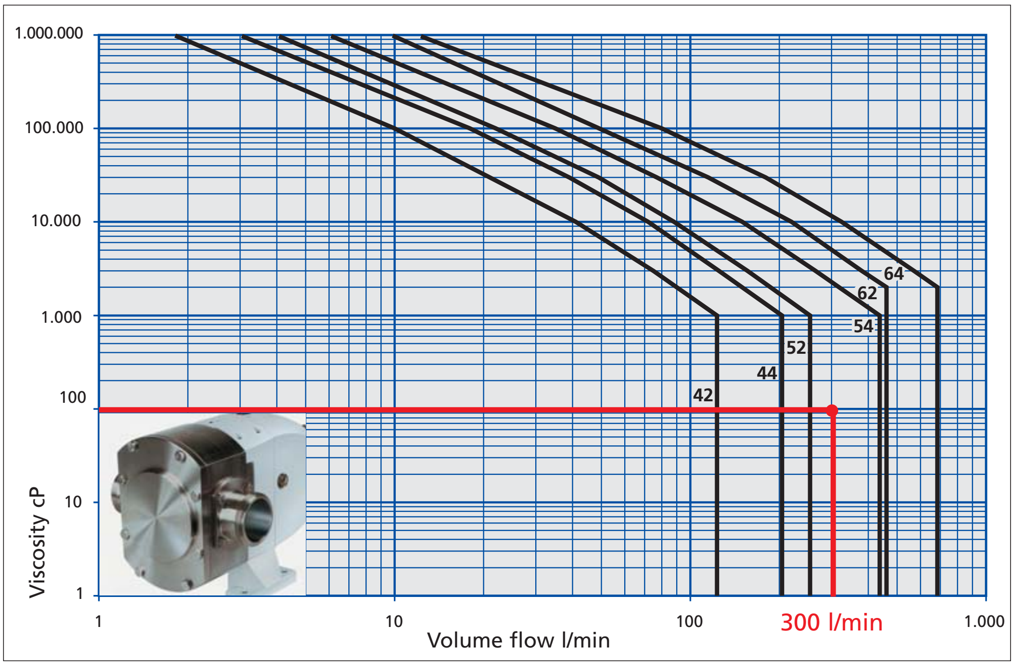


Fig. 30

7.4 Rating the pump

①. In Diagram 1 of Fig. 31 follow the horizontal line marking the viscosity of 100 cP until it intersects with the line for differential pressure Δp = 5 bar.
With higher viscosities than 1,000 cP the line intersects with the zero point of Diagram 2 (zero-bar line) (Fig. 31).

②. As of the the point of intersection of the viscosity line with the differential pressure line a vertical line is drawn upwards until it intersects with the zero-bar line (abscissa) of Diagram 2 (Fig. 31).

③. Starting at this point of intersection, draw a line that is in parallel to the pressure curves.

④. At the point of intersection of this line with the required volume flow rate (300 l/min) draw a vertical line upwards.

⑤. The necessary operating speed of the pump can then be read at the scale (715 min^{-1})

⑥. Use Diagram 3 (Fig. 31) to check whether the maximum differential pressure (8 bar) was not exeeded at the procut temperature. The pump shall be operated within the pressure/temperature range under the curve p/t.

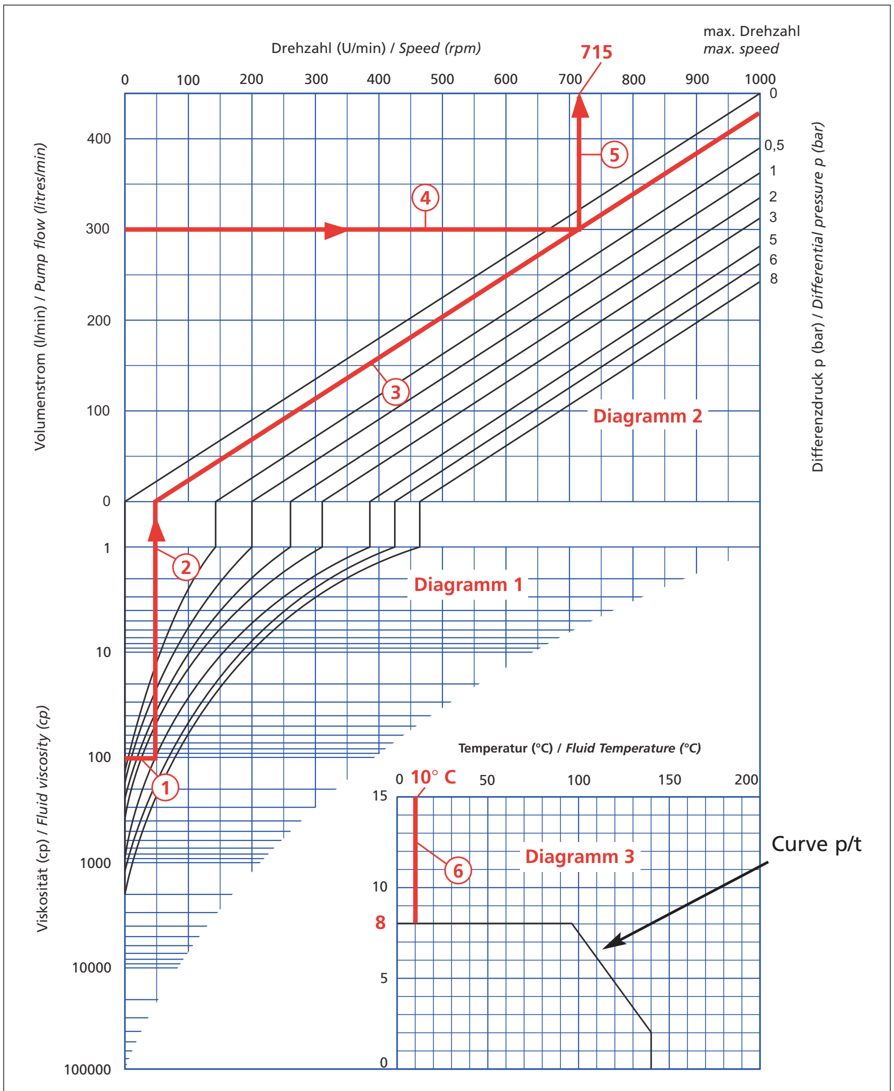


Fig. 31

⑦. In diagram 4 (Fig. 32), first the viscosity factor required for calculating the capacity is determined by drawing a vertical line upwards from the point of intersection of viscosity with the characteristic curve

⑧. and by reading the corresponding viscosity factor f on the scale of the abscissa (2.8)

From the table in Diagram 4 (Fig. 32) find out factor S according to the selected gasket.

The required motor power is calculated as follows:

$$P\ (W) = \left(\frac{p}{0.61} + f + S\right) \times 0.265 \times N$$

$$P\ (W) = \left(\frac{5}{0.61} + 2{,}8 + 1\right) \times 0.265 \times 715$$

$$= \mathbf{2.27\ kW}$$

The required drive torque can now be calculated as follows:

$$M\ (Nm) = \frac{P\ (W) \times 9.56}{N}$$

$$M\ (Nm) = \frac{2270 \times 9.56}{715} = \mathbf{30.35\ Nm}$$

⑨. From Diagram 5 (Fig. 32) the required pressure at the suction port ($NSPH_{req}$) can be read (intersection pump speed / viscosity).
The pressure prevailing at the suction port ($NPSH_{avl}$) should always be 0.1 bar higher than the required pressure in order to prevent cavitation.

7.5 ➤ Result

The result of the pump design is a VPSH 54 pump with a 3.0 kW motor and a pump speed of 750 rpm.

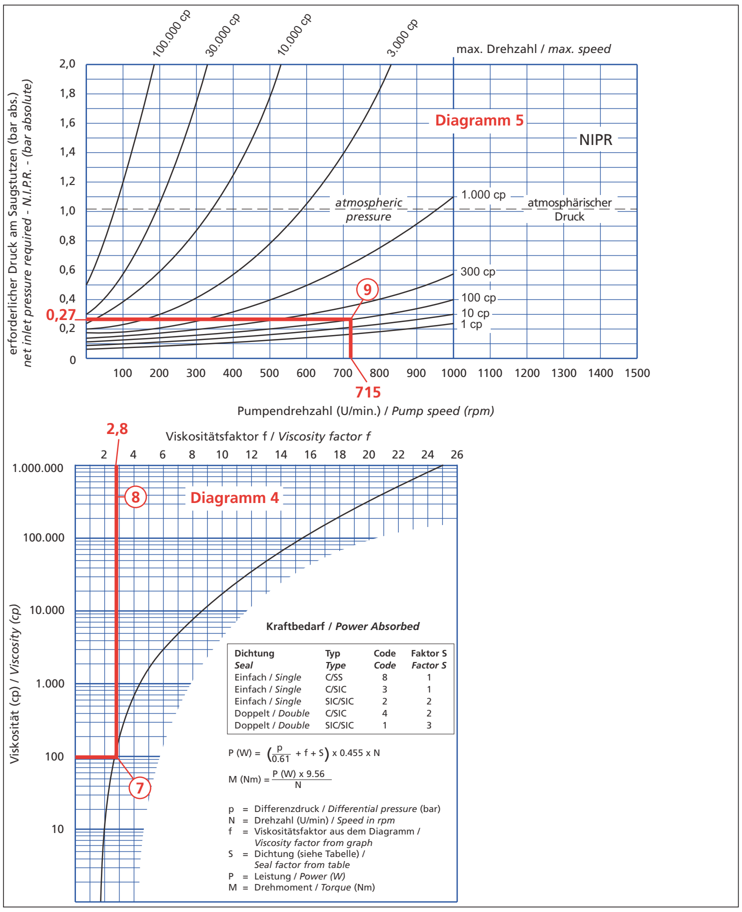
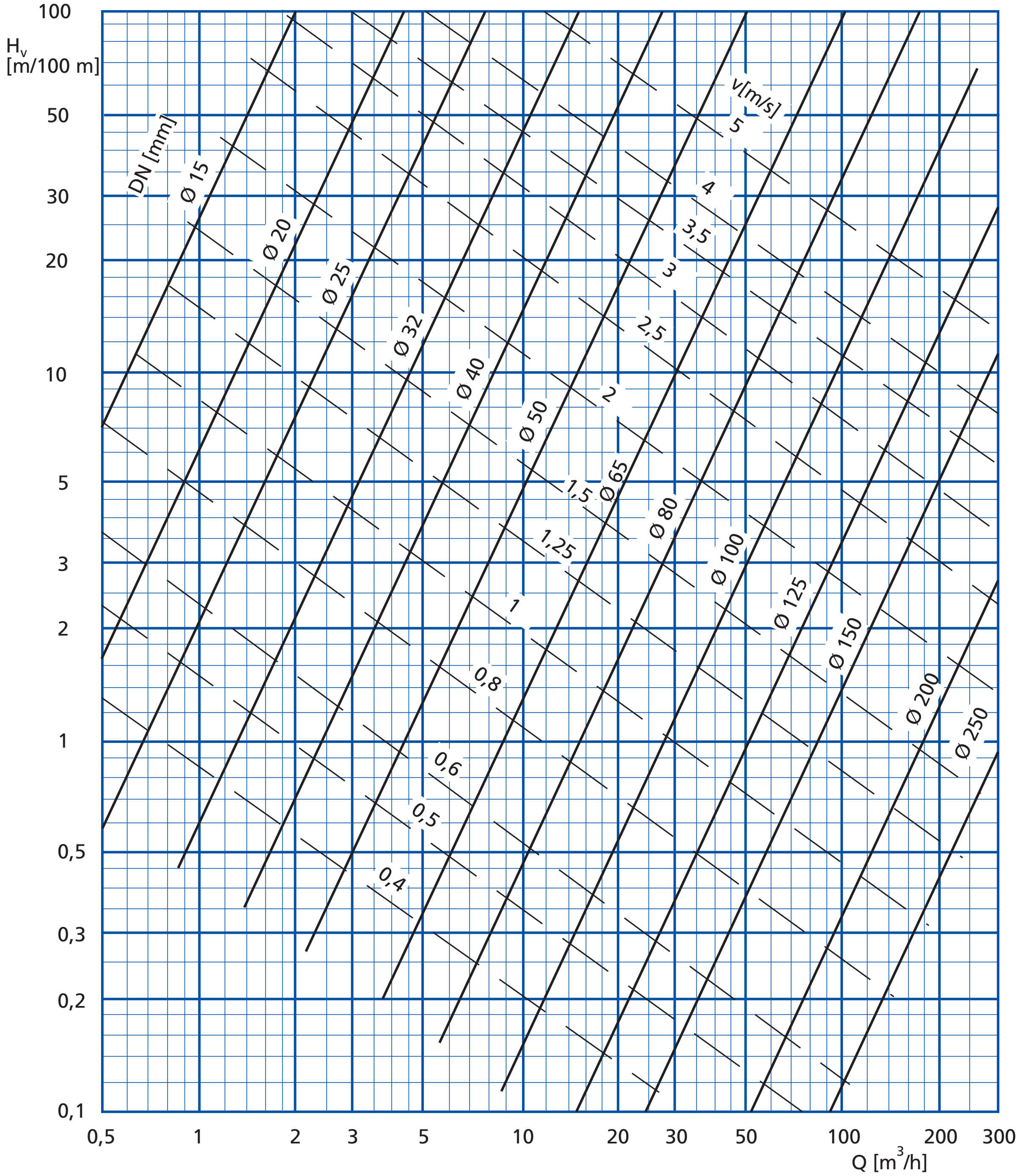


Fig. 32

8.1 Diagram for the calculation of pressure drops

Pressure drops H_v per 100 m pipe length for stainless steel pipes with a surface roughness of k = 0.05 and media with 1 mm²/s viscosity (= water) (accuracy ± 5%)



Pipe diameter (beverage pipe)

Metric								
DN	25	32	40	50	65	80	100	125
inside Ø [mm]	26	32	38	50	66	81	100	125

	Inch OD							Inch IPS			
DN	1"	1 1/2"	2"	2 1/2"	3"	4"		2"	3"	4"	6"
inside Ø [mm]	22	35	47,5	60	73	97,5		57	85	110	162

8.2 Pressure drops of fittings in metre equivalent pipe length

Fitting	Nominal Diameter in mm								
	25	32	40	50	65	80	100	125	150
ζ = 0.05 Reducer Tee	0.05	0.07	0.09	0.12	0.17	0.20	0.28	0.40	0.48
ζ = 0.15 Bend 45°	0.14	0.20	0.27	0.35	0.50	0.60	0.85	1.20	1.40
ζ = 0.25 Bend 90° Expansion Butterfly valve Inlet (Tank outlet)	0.25	0.35	0.45	0.60	0.80	1.00	1.35	1.90	2.4
ζ = 0.90 Tee	0.90	1.20	1.60	2.00	3.00	3.70	5.20	7.00	8.80
ζ = 1.30 Tee	1.20	1.80	2.30	3.00	4.30	5.40	7.40	10.00	12.50
ζ = 1.5 Reflux valve	1.40	2.10	2.70	3.50	5.00	6.30	8.50	11.50	14.50

Applies to: Pipe roughness k = 0.05 mm
Flow speed v = 1-3 m/s (error >10% deviation in speed)
(Accuracy ± 5%)

8.3 Pressure drops of valves in metre equivalent pipe length

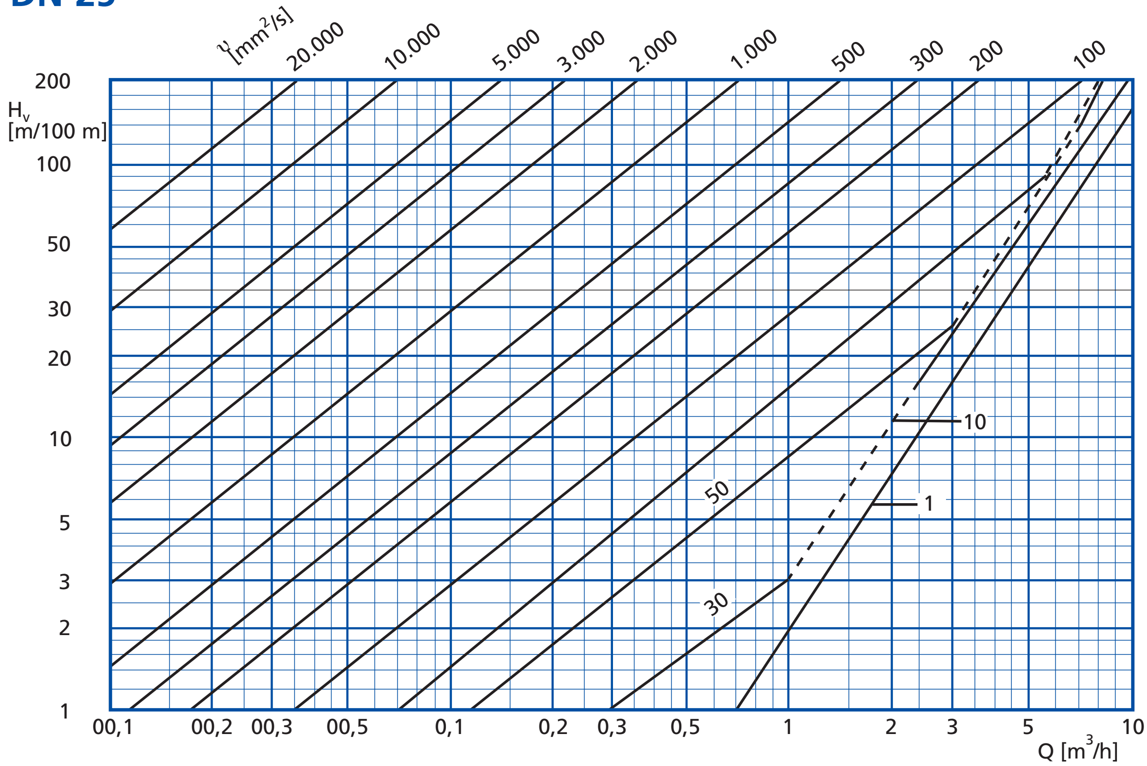
Valve	DN 25	40	50	65	80	100	125	Inch OD 1	1 1/2	2	2 1/2	3	4
Type D													
I to II	1.13	1.3	1.49	2.91	3.64	2.59	5.08	0.85	0.98	1.25	1.58	1.82	2.2
III to IV	0.86	1.23	1.32	2.85	3.55	2.8	4.96	0.76	1.02	1.12	1.52	1.82	2.4
III to I	3.76	8.83	18.41	22.51	20.79	18.58	24.05	2.92	6.18	9.69	15.18	29	36.65
I to III	4.09	5.69	10.48	10.5	23.41	18.29	23.65	3.38	4.06	10.31	7	13.49	27.13
Type R													
I to II	0.97	1.3	1.49	2.91	3.73	2.59	5.08		0.98	1.25	1.58	1.82	2.2
III to IV	0.95	1.59	1.86	4.39	4.79	3.62	6.5		1.13	1.46	2.48	2.8	3.18
III to I	3.33	10.37	17.03	47.28	24.91	18.44	28.15		16.49	11.49	11.82	14.52	19.3
I to III	3.4	10.11	18.85	21.59	25.48	17.24	28.1		19.1	11.56	10.95	15.21	19.25
Type RN													
I to II			3.15	3.41	5.35	4.33	9.5			2.1	2.6	4.27	4.65
III to IV			1.82	3.16	4.68	3.62	6.5			1.46	2.48	2.8	3.32
III to I			15.83	15.05	25.07	18.53	26.65			11.93	10.69	14.54	20.22
I to III			17.37	16.04	28.87	18.4	24.69			12.25	9.72	16.03	24.34
Type B													
I to II				2.91	3.64	2.59	5.08				1.58	1.82	2.19
III to IV				3.64	4.76	3.76	6.25				1.92	2.71	3.37
III to I				16.38	25.19	20.89	27.29				7.64	14.93	18.86
I to III				17.43	27.28	19.54	27.17				7.33	15.49	17.79
Type BN													
I to II				3.41	5.35	4.33	9.5				2.6	4.27	4.65
III to IV				3.12	4.71	3.93	8.82				1.92	2.71	3.37
III to I				16.05	25.43	20.12	28.15				7.88	14.93	17.7
I to III				17	28.17	20.01	28.1				5.79	15.76	17.79
Type N													
I to II	0.91	1.62	2.19	3.34	2.21	2.72		0.77	1.50	1.44	1.88	2.45	2.42
III to IV	0.67	0.85	1.64	2.58	1.48	2.14		0.53	0.91	0.78	1.2	1.33	1.82
III to I	5.44	10.96	22.67	28.56	36.8	25.93		3.47	5.63	19.25	19.83	29.84	26.43
I to III	2.03	3.86	6.36	10.15	16.06	19.27		1.59	2.76	4.87	5.95	11.84	17.98
Type NL													
I to III	1.77	3.3	3.76	7.03	10.12	11.43	11.61	11.61	2.23	3.2	5.18	6.95	9.58
III to I	1.6	2.73	4.42	6.16	11.08	12.81	11.55	11.55	1.67	3.82	2.84	6.44	10.58
Type W													
I to II	1.05	1.48	1.57	2.36	4.14	3.16	4.54	4.54	1.73	1.88	2.09	2.34	2.56
I to III	1.23	9.36	16.02	23.68	20.68	41.49	21.37	21.37	6.09	21.58	19.61	35.24	39.69
III to I	1.19	3.59	7.05	9.37	23.7	25.06	19.67	19.67	3.77	8.56	8.43	18.93	23.54
III to V	2.58	3.27	6.89	10.11	24.37	24.68	20.09	20.09	3.29	8.91	8.83	21.75	25.05
V to III	6.53	9.39	20.31	26.32	39.06	30.47	30.50	30.50	8.35	27.01	20.68	29.51	28.25
Type X													
I to II	0.85	1.73	1.29	1.86	3.53	2.28	4.37	4.37	0.84	1.12	1.48	2.1	2.01
I to III	2.77	3.11	8.71	7.41	17.95	19.33	18.50	18.50	1.99	5.44	5.07	12.85	15.29
III to I	2.95	3.86	9.3	13.08	34.84	29.64	35.24	35.24	3.02	6.28	8.57	18.66	22.93
III to V	2.93	4.65	9.57	11.31	23.29	28.94	35.79	35.79	2.99	6.31	8.8	16.44	25.39
V to III	2.61	3.05	8.12		18.49	20.42	19.46	19.46	1.94	4.88	5.12	13.4	16.21
V to VI	0.73	1.49	1.01	2.27	3.06	4.38	2.58	2.58	0.76	0.88	1.27	1.31	1.6
Type Y													
I to II	0.78	1.18	1.25	1.81	3.52	2.21	4.17	4.17	0.86	1.19	1.5	2.12	2.05
I to III	3.73	12.09	10.49	7.41	35.77	59.77	23.11	23.11	7.81	7.98	8.67	21.44	59.92
III to I	4	10.23	10.8	13.08	51.81	57.02	37.48	37.48	8.11	8.15	11.97	29.14	55.33
III to V	3.59	5.39	6.22	11.31	17.42	35.96	20.35	20.35	2.83	6.51	11.27	23	32.9
V to III	2.18	2.43	5.34	7.09	18.2	20.86	19.60	19.60	1.83	3.89	5	10.14	18.51
V to VI	0.9	1.31	1.37	1.88	3.57	2.35	3.89	3.89	0.84	1.17	2.02	2.24	2.49

8.4 Vapour pressure table for water

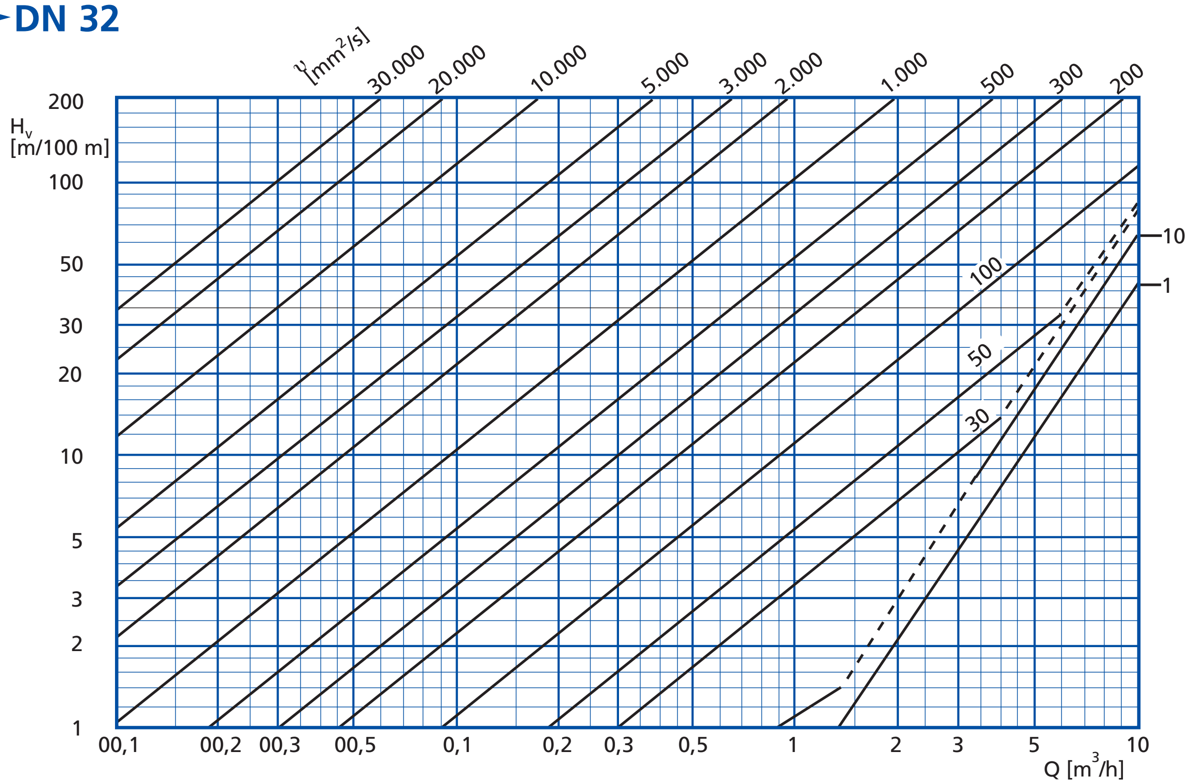
t °C	T K	p_D bar	ρ kg/dm³	t °C	T K	p_D bar	ρ kg/dm³
0	273.15	0.00611	0.9998	61	334.15	0.2086	0.9826
1	274.15	0.00657	0.9999	62	335.15	0.2184	0.9821
2	275.15	0.00706	0.9999	63	336.15	0.2286	0.9816
3	276.15	0.00758	0.9999	64	337.15	0.2391	0.9811
4	277.15	0.00813	1.0000	65	338.15	0.2501	0.9805
5	278.15	0.00872	1.0000	66	339.15	0.2615	0.9799
6	279.15	0.00935	1.0000	67	340.15	0.2733	0.9793
7	280.15	0.01001	0.9999	68	341.15	0.2856	0.9788
8	281.15	0.01072	0.9999	69	342.15	0.2984	0.9782
9	282.15	0.01147	0.9998	70	343.15	0.3116	0.9777
10	283.15	0.01227	0.9997	71	344.15	0.3253	0.9770
11	284.15	0.01312	0.9997	72	345.15	0.3396	0.9765
12	285.15	0.01401	0.9996	73	346.15	0.3543	0.9760
13	286.15	0.01497	0.9994	74	347.15	0.3696	0.9753
14	287.15	0.01597	0.9993	75	348.15	0.3855	0.9748
15	288.15	0.01704	0.9992	76	349.15	0.4019	0.9741
16	289.15	0.01817	0.9990	77	350.15	0.4189	0.9735
17	290.15	0.01936	0.9988	78	351.15	0.4365	0.9729
18	291.15	0.02062	0.9987	79	352.15	0.4547	0.9723
19	292.15	0.02196	0.9985	80	353.15	0.4736	0.9716
20	293.15	0.02337	0.9983	81	354.15	0.4931	0.9710
21	294.15	0.02485	0.9981	82	355.15	0.5133	0.9704
22	295.15	0.02642	0.9978	83	356.15	0.5342	0.9697
23	296.15	0.02808	0.9976	84	357.15	0.5557	0.9691
24	297.15	0.02982	0.9974	85	358.15	0.5780	0.9684
25	298.15	0.03166	0.9971	86	359.15	0.6011	0.9678
26	299.15	0.03360	0.9968	87	360.15	0.6249	0.9671
27	300.15	0.03564	0.9966	88	361.15	0.6495	0.9665
28	301.15	0.03778	0.9963	89	362.15	0.6749	0.9658
29	302.15	0.04004	0.9960	90	363.15	0.7011	0.9652
30	303.15	0.04241	0.9957	91	364.15	0.7281	0.9644
31	304.15	0.04491	0.9954	92	365.15	0.7561	0.9638
32	305.15	0.04753	0.9951	93	366.15	0.7849	0.9630
33	306.15	0.05029	0.9947	94	367.15	0.8146	0.9624
34	307.15	0.05318	0.9944	95	368.15	0.8453	0.9616
35	308.15	0.05622	0.9940	96	369.15	0.8769	0.9610
36	309.15	0.05940	0.9937	97	370.15	0.9094	0.9602
37	310.15	0.06274	0.9933	98	371.15	0.9430	0.9596
38	311.15	0.06624	0.9930	99	372.15	0.9776	0.9586
39	312.15	0.06991	0.9927	100	373.15	1.0133	0.9581
40	313.15	0.07375	0.9923	102	375.15	1.0878	0.9567
41	314.15	0.07777	0.9919	104	377.15	1.1668	0.9552
42	315.15	0.08198	0.9915	106	379.15	1.2504	0.9537
43	316.15	0.08639	0.9911	108	381.15	1.3390	0.9522
44	317.15	0.09100	0.9907	110	383.15	1.4327	0.9507
45	318.15	0.09582	0.9902	112	385.15	1.5316	0.9491
46	319.15	0.10086	0.9898	114	387.15	1.6362	0.9476
47	320.15	0.10612	0.9894	116	389.15	1.7465	0.9460
48	321.15	0.11162	0.9889	118	391.15	1.8628	0.9445
49	322.15	0.11736	0.9884	120	393.15	1.9854	0.9429
50	323.15	0.12335	0.9880	124	397.15	2.2504	0.9396
51	324.15	0.12961	0.9876	130	403.15	2.7013	0.9346
52	325.15	0.13613	0.9871	140	413.15	3.6850	0.9260
53	326.15	0.14293	0.9866	150	423.15	4.7600	0.9168
54	327.15	0.15002	0.9862	160	433.15	6.3020	0.9073
55	328.15	0.15741	0.9857	170	443.15	8.0760	0.8973
56	329.15	0.16511	0.9852	180	453.15	10.2250	0.8869
57	330.15	0.17313	0.9846	190	463.15	12.8000	0.8760
58	331.15	0.18147	0.9842	200	473.15	15.8570	0.8646
59	332.15	0.19016	0.9837	250	523.15	40.5600	0.7992
60	333.15	0.19920	0.9832	300	573.15	87.6100	0.7124

8.5 Pressure drops depending on viscosity

DN 25

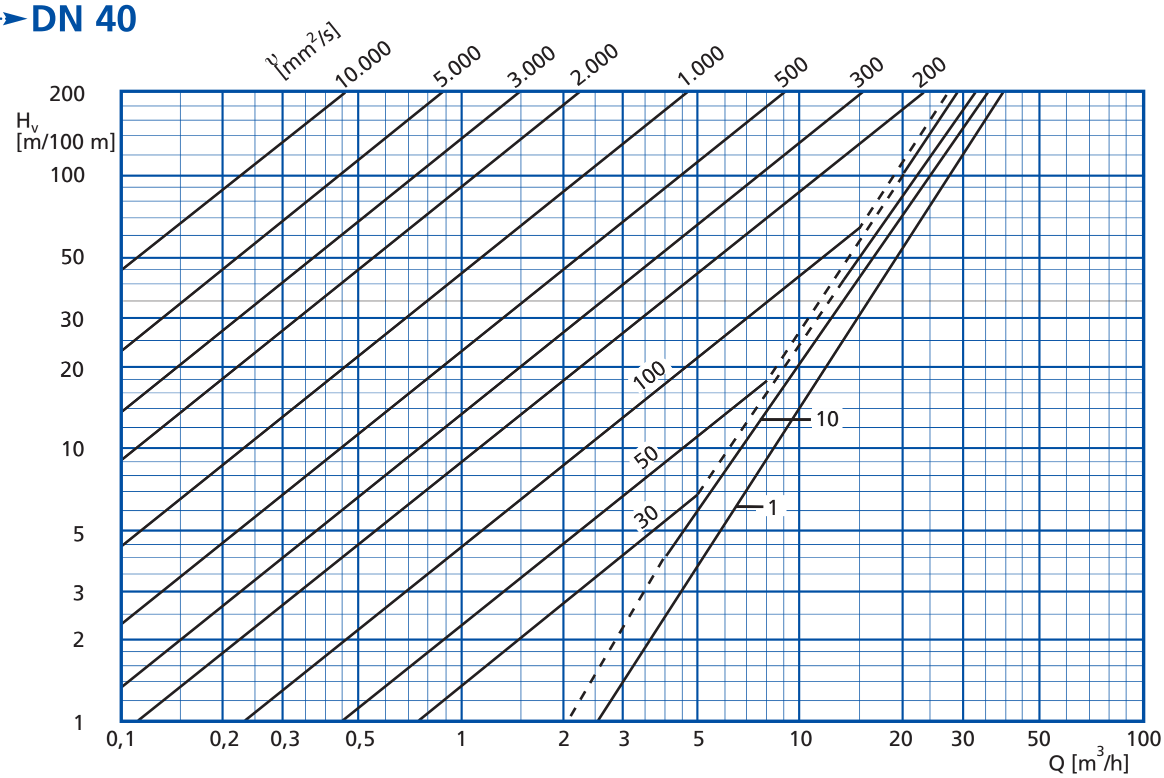


DN 32

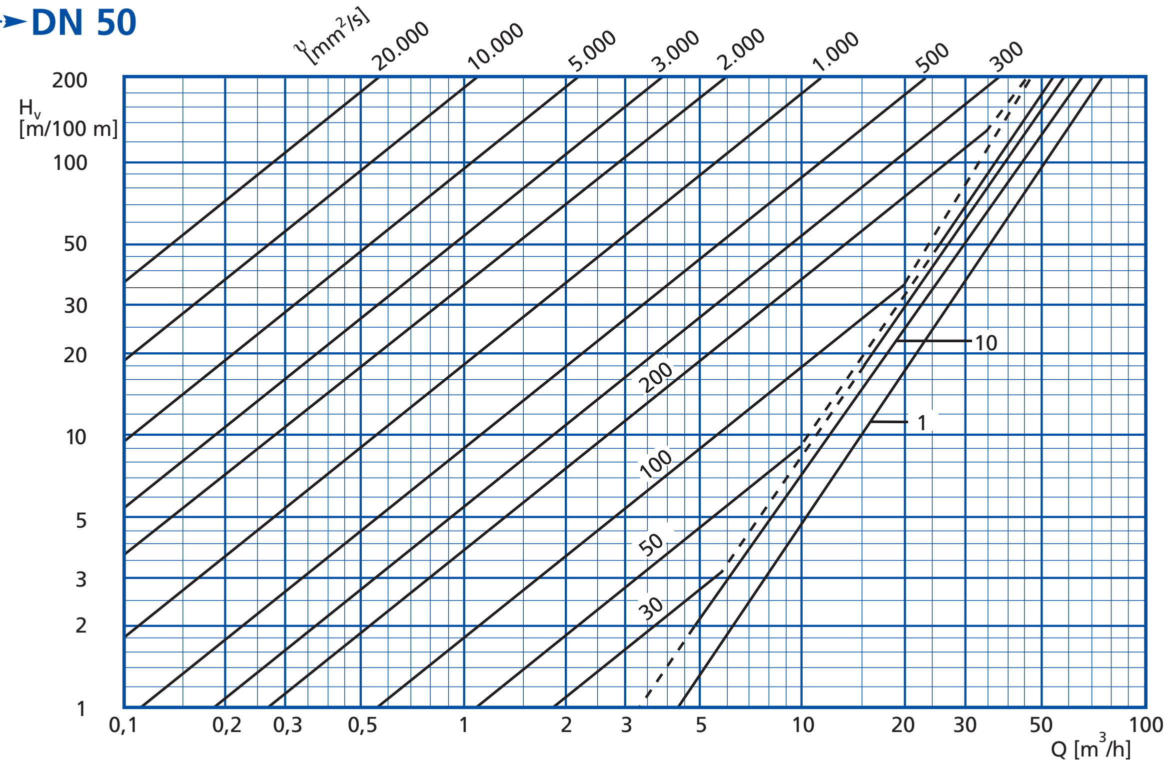


------ Transition range from laminar to turbulent flow (Re: ≈ 1.400 - ≈ 3.500)
(Accuracy ± 5 %)
Pressure drop H_v per 100 m pipe length (k = 0.05)

DN 40

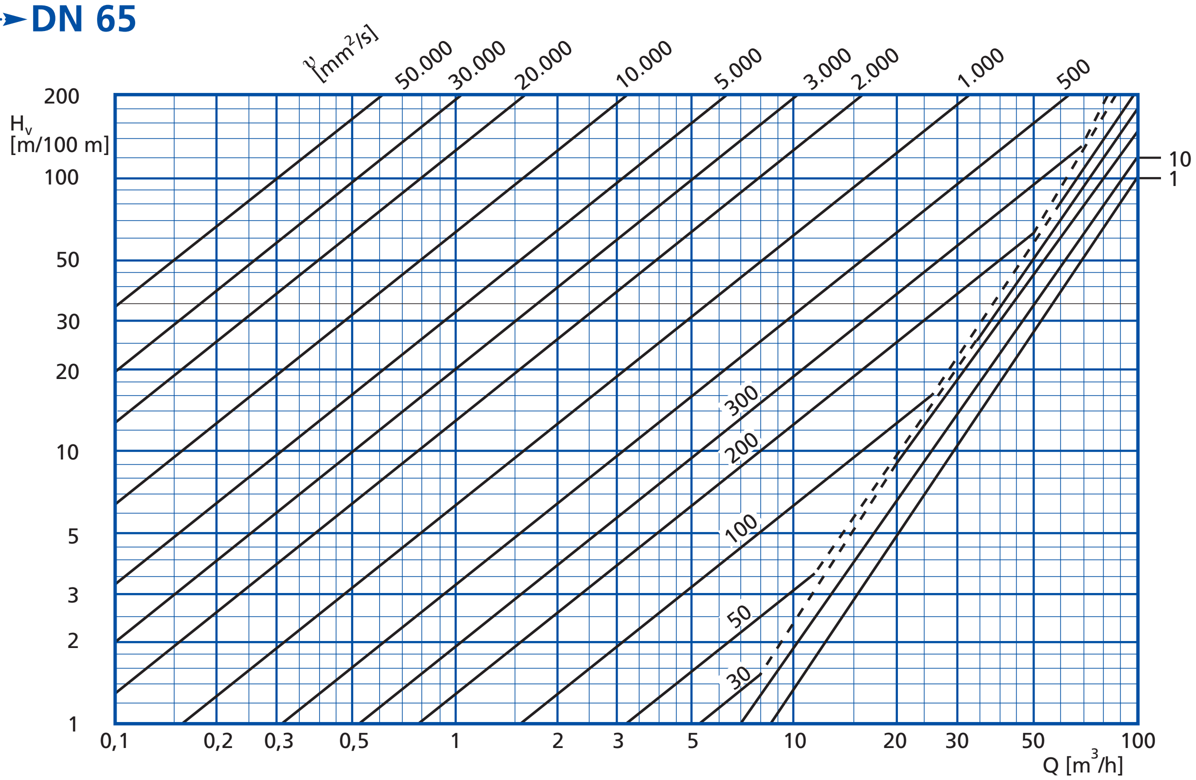


DN 50

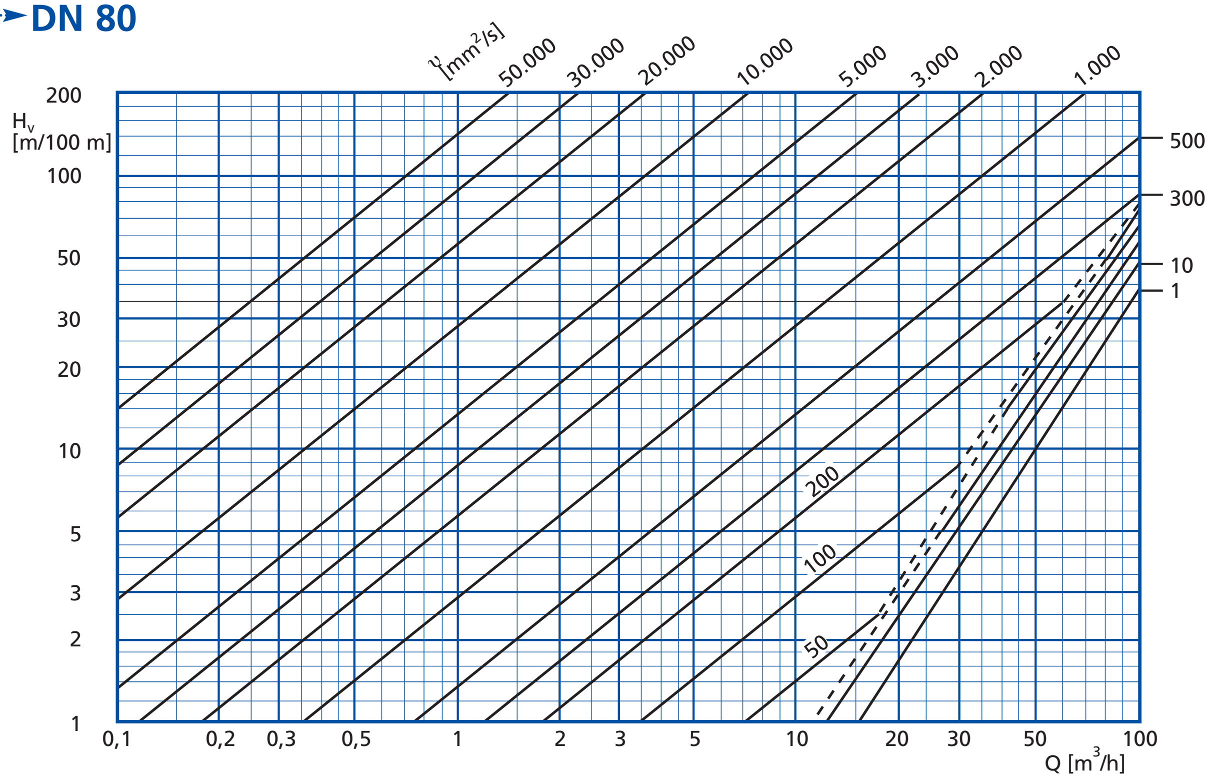


------ Transition range from laminar to turbulent flow (Re: ≈ 1.400 - ≈ 3.500)
(Accuracy ± 5 %)
Pressure drop H_v per 100 m pipe length (k = 0.05)

DN 65

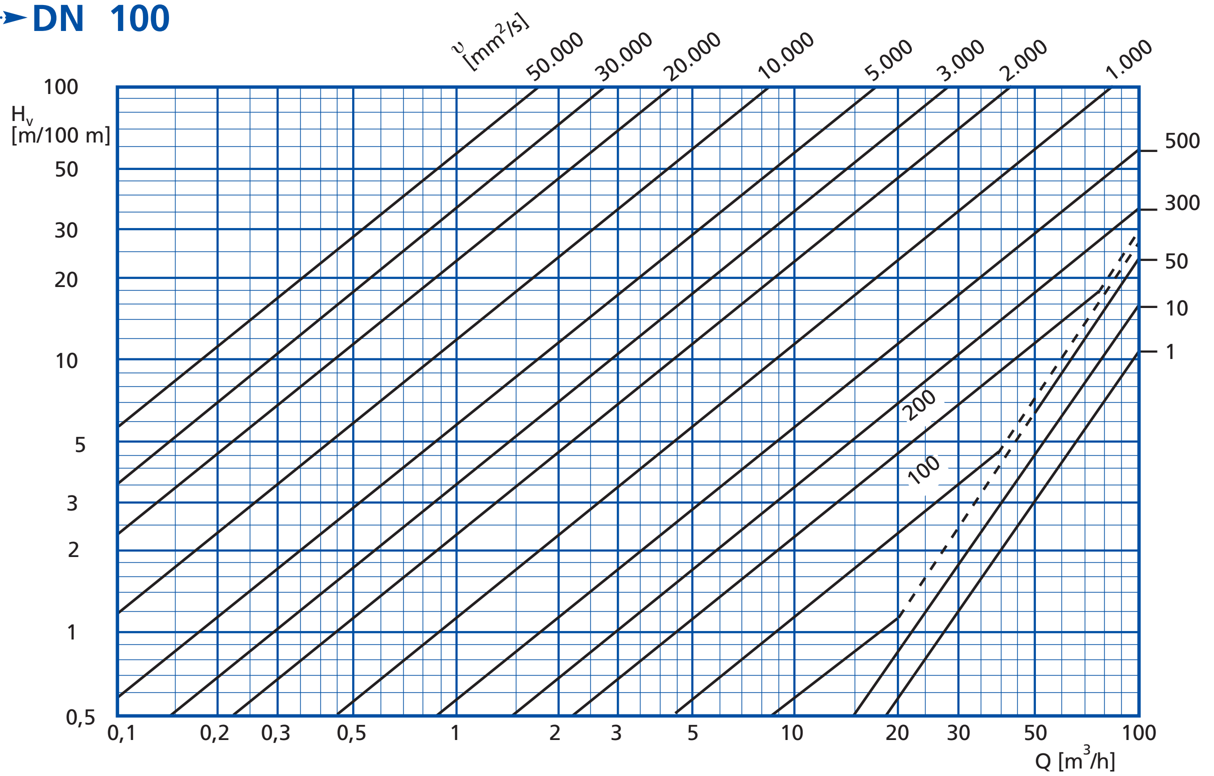


DN 80

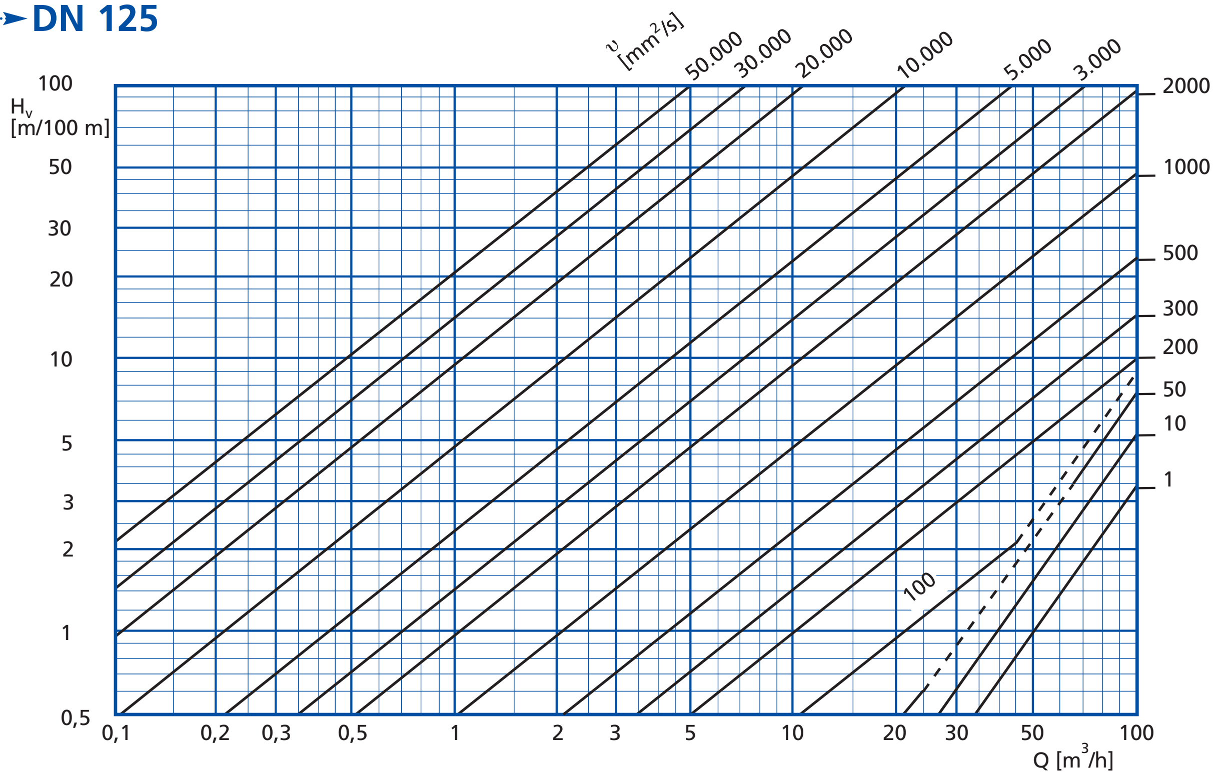


------ Transition range from laminar to turbulent flow (Re: ≈ 1.400 - ≈ 3.500)
(Accuracy ± 5 %)
Pressure drop H_v per 100 m pipe length (k = 0.05)

DN 100

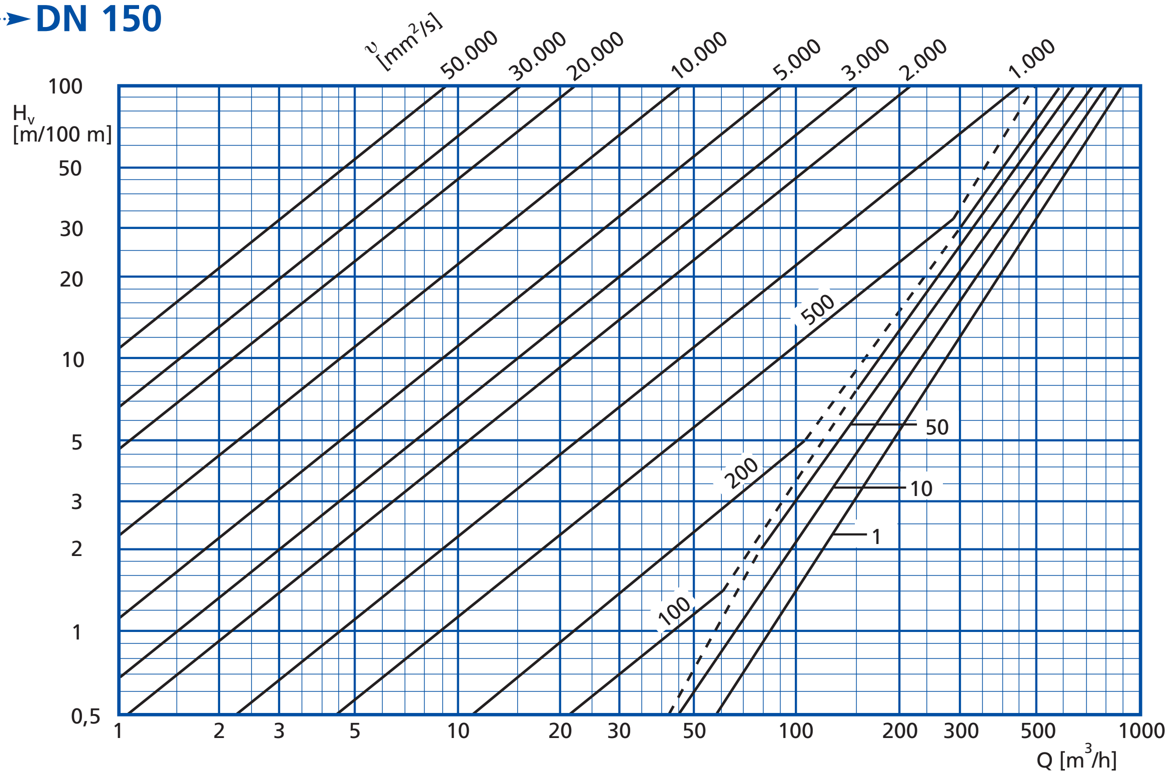


DN 125

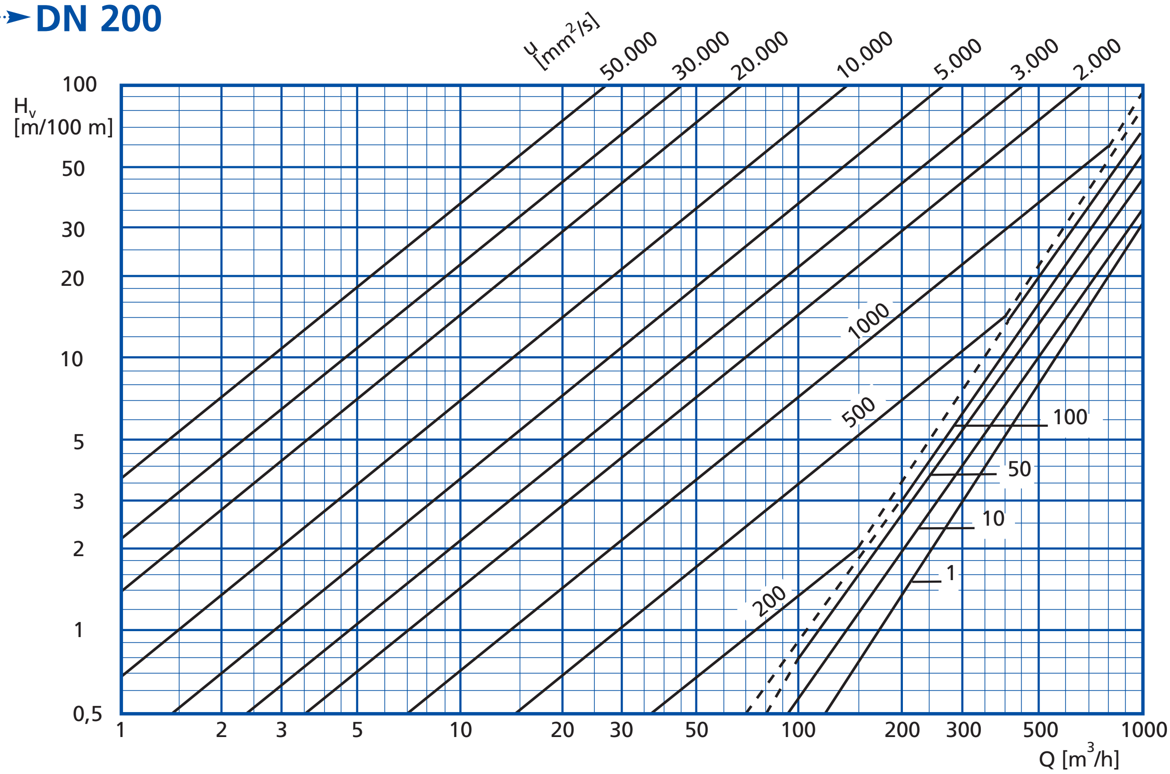


------ Transition range from laminar to turbulent flow (Re: ≈ 1.400 - ≈ 3.500)
(Accuracy ± 5 %)
Pressure drop H_v per 100 m pipe length (k = 0.05)

DN 150



DN 200



------ Transition range from laminar to turbulent flow (Re: ≈ 1.400 - ≈ 3.500)
(Accuracy ± 5 %)
Pressure drop H_v per 100 m pipe length (k = 0.05)

8.6 SI - Units

Legal units (Abstract for centrifugal pumps)

Designation	Formula symbols	Legal units (the unit listed first should be used)	not admitted units	Conversion
Length	l	m km, cm, mm		base unit
Volume	V	m^3 cm^3, mm^3, (Liter)	cbm, cdm	
Flow rate	Q	m^3/h		
Volumetric flow	V	m^3/s, l/s		
Time	t	s (second) ms, min, h, d		base unit
Speed	n	1/min 1/s		
Mass	m	kg (Kilogram) g, mg, (Tonne)	pound, centner	base unit
Density	ρ	kg/m^3 kg/dm^3, kg/cm^3		
Force	F	N (Newton = kg m/s^2) kN, mN	kp, Mp	1 kp = 9.81 N
Pressure	p	bar (bar = N/m^2) Pa	kp/cm^2, at, m WS, Torr,	1 bar = 10^5 Pa = 0.1 MPa 1 at = 0.981 bar = 9.81 x 10^4Pa 1 m WS = 0,98 bar
Energy, Wort, Heat amount	W, Q	J (Joule = N m = W s) kJ, Ws, kWh,	kp m kcal, cal	1 kp m = 9.81 J 1 kcal = 4.1868 kJ 1 kWh = 3600 kJ
Flow head	H	m (Meter)	m Fl.S.	
Power	P	W (Watt = J/s = N m/s) MW, kW	kp m/s, PS	1 kp m/s = 9.81 W; 1 PS = 736 W
Temperature, t-difference	T	K (Kelvin) °C	°K, grd	base unit
Kinematic viscosity	ν	m^2/s mm^2/s	St (Stokes), °E,…	1St = 10^{-4} m^2/s 1 cSt = 1 mm^2/s Approximation: mm^2/s = (7.32 x °E - 6.31/°E) $\nu = \frac{\eta}{\rho}$
Dynamic viscosity	η	Pa s (Pascal seconds = N s/m^2)	P (Poise), …	1P = 0.1 Pa s

8.7 Conversion table of foreign units

Designation	Unit	Unit code		British		USA	
Length	1 inch	in		25.4	mm	25.4	mm
	1 foot	ft	= 12 in	0.3048	m	0.3048	m
	1 yard	yd	= 3 ft	0.9144	m	0.9144	m
	1 mile	mi	= 1.760 yd	1.6093	km	1.6093	km
	1 nautical mile	mi		1.8532	km	1.8532	km
Surface	1 square inch	sq in		6.4516	cm^2	6.4516	cm^2
	1 square foot	sq ft		929.03	cm^2	929.03	cm^2
	1 square yard	sq yd		0.8361	m^2	0.8361	m^2
	1 acre			4,046.86	m^2	4.046,86	m^2
	1 square mile	sq mi		2.59	km^2	2.59	km^2
Volume	1 cubic inch	cu in		16.387	cm^3	16.387	cm^3
	1 cubic foot	cu ft		28.3268	dm^3	28.3268	dm^3
	1 register ton	RT	=100 cu ft	2.8327	m^3	2.8327	m^3
	1 British shipping ton		= 42 cu ft	1.1897	m^3	-	
	1 US shipping ton		= 40 cu ft	-		1.1331	m^3
	1 gallon	gal		4.5460	dm^3	3.7854	dm^3
	1 US oil-barrel (crude oil)		-			0.159	m^3
Mass & weight	1 ounce	oz (avdp)		28.3495	g	28.3495	g
	1 pound	lb		0.4536	kg	0.4536	kg
	1 stone			6.3503	kg	6.3503	kg
	1 ton			1,016.047	kg	-	
Density	1 pound per cubic foot	lb/cu ft		0.0160	kg/dm^3	0.0160	kg/dm^3
	1 pound per gallon	lb/gal		0.09978	kg/dm^3	0.1198	kg/dm^3
Flow rate	1 gallon per minute	gpm		0.07577	l/s	0.06309	l/s
	1 cubic foot per second	cusec		28.3268	l/s	28.3268	l/s
Force	1 ounce (force)	oz		0.2780	N	0.2780	N
	1 pound (force)	lb		4.4438	N	4.4438	N
	1 short ton	shtn		8.8964	kN	8.8964	kN
Pressure	$1\ \frac{\text{pound (force)}}{\text{square foot}}$	$\frac{\text{lb (force)}}{\text{sq ft}}$		47.88025	Pa	47.88025	Pa
	$1\ \frac{\text{pound (force)}}{\text{square inch}}$	$\frac{\text{lb (force)}}{\text{sq in}}$	, psi	68.9476	m bar	68.9476	m bar
Work, Energy, Heat amount	1 foot-pound	ft lb		1.3558	J	1.3558	J
	1 Horse power Hour	Hp h		2.6841	MJ	2.6841	MJ
Power	$1\ \frac{\text{foot-pound (av)}}{\text{per second}}$	$\frac{\text{ft lb}}{\text{s}}$		1.3558	W	1.3558	W
	1 Horse power (Hp)			0.7457	kW	0.7457	kW
Dynamic viscosity	$1\ \frac{\text{pound (mass)}}{\text{foot x second}}$	$\frac{\text{lb (mass)}}{\text{ft s}}$		1.4882	Pa s	1.4882	Pa s

8.8 Viscosity table (guideline values)

	Product	Density ρ	Viscosity η in CPs	Temp °C t	Viscous behaviour type*
Reference	Water	1	1		N
Bakery products	Egg	0.5	60	10	N
	Emulsifier		20		T
	Melted butter	0.98	18	60	N
	Yeast slurry (15%)	1	180		T
	Lecithine		3,250	50	T
	Batter	1	2,200		T
	Frosting	1	10,000		T
Chemicals	Glycerin 100%	1.26	624	30	
	Glycerin 100%	1.26	945	20	
	Glycerin 45%	1.11	5	20	
	Glycerin 80%	1.21	62	20	
	Glycerin 90%	1.23	163	25	
	Glycerin 95%	1.25	366	25	
	Caustic soda 20%	1.22	7	20	
	Caustic soda 40%	1.52	39	20	
	Caustic soda 50%	1.51	20	40	
	Caustic soda 50%	1.52	38	30	
	Nictric acid 10%	1.05	1	20	
Food	Apple pulp		10,020	20	
	Pear pulp		4,000	70	T
	Honey	1.5	2,020	45	
	Mashed potatoes	1	20,000		T
	Ketschup	1.11	560	60	T
	Magarine emulsion		26	50	
	Mayonnaise	1	5,000	25	T
	Nut core		9,500	20	
	Prune juice	1	60	50	T
	Mustard		11,200	20	
Fats & oils	Peanut oil	0.92	42	40	N
	Linseed oil	0.93	30	40	N
	Corn oil	0.92	30		N
	Olive oil	0.91	84	20	
	Vegetable oil	0.92	5	150	N
	Lettuce oil		85	20	
	Lard	0.96	60	40	N
	Soybean oil	0.95	36	40	N
Meat products	Meat emulsion	1	22,000	5	T
	Ground beef fat	0.9	11,000	15	T
	Pork fat (slurry)	1	650	5	T
	Animal fats	0.9	43	40	N
	Pet food	1	11,000	5	T

***Viscous behaviour type**
N = Newtonian
T = Thixotropic

8.8 Viscosity table (continued)

	Product	Density ρ	Viscosity η in CPs	Temp °C t	Viscous behaviour type
Reference	Water	1	1		N
Beverages and concentrates	Apple juice concentrate		7	20	
	Apple wine concentrate	1.3	300	20	
	Beer	1	1	5	N
	Coca Cola	1	1	40	
	Cola-Konzentrat		25	20	
	Egg liqueur		620	20	
	Strawberry syrup		2,250	40	
	Fruit liqueur		12	20	
	Coffee extract 30% i.Tr.		18	20	
	Yeast concentrate (80%)		16,000	4	T
	Herb liqueur		3	20	
	Orange concentrate		1,930	20	
	Orange juice concentrate	1.1	5,000	5	T
	Currant juicesaft		2	20	
Cosmetics, soaps	Face cream		10,000		T
	Hair gel	1.4	5,000		T
	Hand soap		2,000		T
	Shampoo		5,000		T
	Toothpaste		20,000		T
Dairy products	Buttermilk		8	20	
	Cream for churning, acid		550	20	
	Skimmilk, acid		140	20	
	Cottage cheese	1.08	225		T
	Yogurt		1,100		T
	Cacao milkdrink		7	20	
	Cheese		30	70	T
	Evaporated milk 77%	1.3	10,000	25	N
	Evaporated milk 10%		45	20	
	Evaporated milk 7,5%		12	20	
	Evaporated milk, sweetened		6,100	20	
	Concentrated skimmilk		100	20	
	Milk	1.03	1	15	N
	Cream	1.02	20	4	N
	Acid cream		32	20	
	Whole milk	1.03	2	20	
	Yoghurt		900	20	
Confectionary	Hot fudge	1.1	36,000		T
	Cacao butter		42	40	
	Cacao mass		4,000	20	
	Caramel	1.2	400	60	
	Chocolate	1.1	17,000	50	T
	Chocolate coating		2,600	40	
	Toffee	1.2	87,000		T
	Sugar syrup 50%		15	20	
	Sugar syrup 56%	1.27	32	20	
	Sugar syrup 64%	1.31	120	20	

8.9 Mechanical seals (recommendation)

			Material						
Medium	Concentration %	Temp. C°	Carbon/Sic	Sic/Sic	EPDM	FKM	Standard seal	Rinsed seal (Quenched)	Note
Alcohol: ethanol			X		X		X		
Alcohol: butanol			X		X		X		
Alcohol: methanol			X		X		X		
Pineapple juice			X			X	X		
Apple juice, pulp, wine				X	X			X	
Apple juice, acidic				X	X			X	
Apricot juice			X		X		X		
Beer				X	X	X		X	
Beer yeast, wort				X		X		X	
Blood			X			X	X		
Butter			X		X	X	X		
Buttermilk			X		X	X	X		
Egg liqueur			X			X		X	
Egg yolk			X			X		X	
Ice cream			X			X	X		
Peanut oil			X			X	X		
Fat, fatty alcohol			X			X	X	X	
Fatty acids		150		X		X	X		
Fish glue / oil / meal				X		X		X	
Fruit pulp			X		X		X		
Gelatine			X		X	X		X	heated
Glucose			X		X	X	X		
Hair shampoo			X			X		X	
Body lotion			X			X	X		
Honey			X			X		X	
Hop mash				X		X	X		
Coffee extract				X	X	X		X	
Cacao butter - oil				X	X	X			
Mashed potatoes				X	X		X	X	
Potato starch			X			X		X	
Cheese, cheese cream				X	X	X		X	
Ketchup (tomatoe extract)			X		X		X		
Adhesives: vegetable			X		X	X			
Adhesives: synthetic			X	X		X		X	
Adhesives: animal glue			X	X		X		X	
Adhesives: cellulose			X	X	X			X	
Carbon dioxide			X			X		X	
Coco oil			X			X	X	X	
Lactose (milk/sugar solution)			X	X		X	X		
Limonades, alcoholfree beverages			X	X	X		X		
Limonades, syrup			X	X		X	X	X	

8.9 Mechanical seals (continued)

Medium	Concentration %	Temp. °C	Material						Note
			Carbon/Sic	Sic/Sic	EPDM	FKM	Standard seal	Rinsed seal (Quenched)	
Corn oil			X			X	X		
Mayonnaise			X				X	X	
Marmelade			X	X		X	X		
Melasse				X	X	X	X		
Milk		<80	X		X		X		
Milk		<140	X		X	X		X	
Whey			X		X			X	
Caustic soda	<2		X		X		X		
Caustic soda	<20			X	X			X	
Caustic soda	<10	80°	X	X	X		X	X	
Olive oil			X			X	X	e	
Orange juice			X			X	X		
Vegetable oil			X			X	X		
Rape oil			X		X	X	X		
Cane sugar solution				X		X		X	
Beet mash			X			X	X		
Juice (solution)				X		X	X	X	
Cream			X			X	X		
Lettuce oil			X			X	X		
Nitric acid	<2		X			X	X		
Nictric acid	<60	<65		X		(X)	X	X	PTFE
Brine	<5			X		X	X		
Black liquor				X		X			
Lard			X			X	X		
Soap solution			X			X	X		
Mustard				X		X		X	
Soybean oil			X			X	X		
Tomato juice			X			X	X		
Walöl			X			X	X		
Water		<140	X		X		X		
Wine				X			X	X	
Wine brandy			X		X		X		
Citrus fruit juice			X			X		X	
Sugar solution	>10			X		X		X	
Sugar solution	<10		X			X	X		
Sugar cane			X	X		X		X	
Sugar beet juice			X			X		X	

8.10 Pump data sheet

Pumping medium:

Temperature: °C

Density: kg/dm³

Viscosity: mm²/s

Solids content / Grain size: %/mm

Flow rate: m³/h

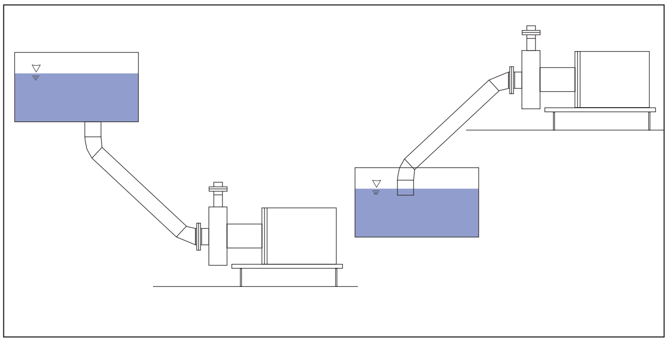
Flow head: m

Plant data	**Suction side**	**Delivery side**
Pipe - Ø	DN	DN
Pipe length	m	m
Ø and number of bends 45°	DN /	DN /
Ø and number of bends 90°	DN /	DN /
Ø and number of Tees Direction of flow..............	DN /	DN /
Ø and number of expansions	DN /	DN /
Ø and number of butterfly valves	DN /	DN /
Ø and number of single seat valves with seat rinsing	DN /	DN /
Ø and number of double seat valves with seat rinsing Direction of flow..............	DN /	DN /
Ø and number of valve housings straight through flow	DN /	DN /
Geodetic heights, pump towards liquid level	m	m
Pressure on to liquid level	barG	barG
Other pipe installations		

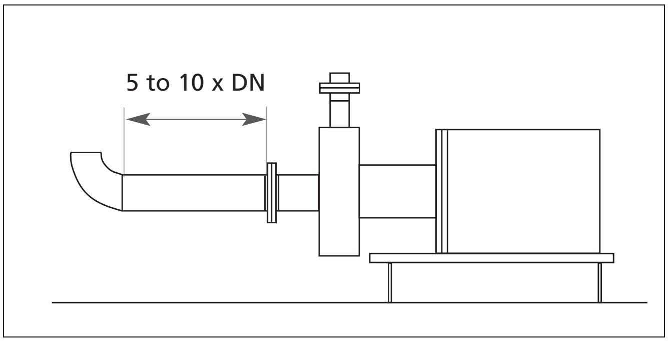
Other fittings generating flow resistance
e.g. heat exchanger, filter
(with indication of differential pressures
at nominal flow rate) bar at m³/h bar at m³/h

Power supply Voltage V Frequency Hz

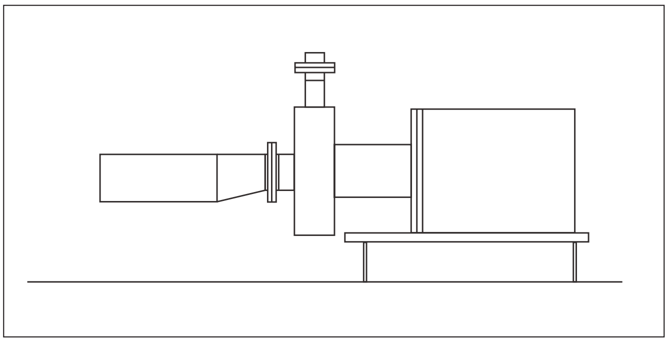
8.11 Assembly instructions



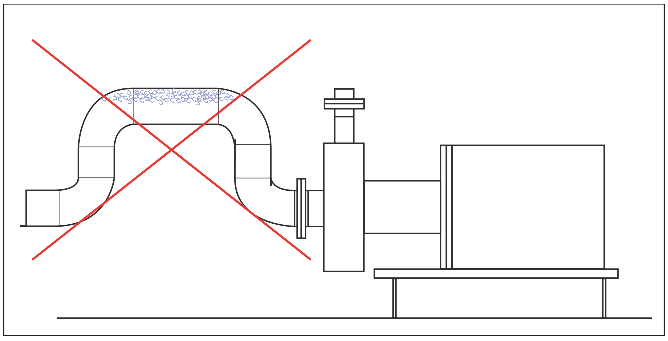
The suction pipe should be placed steadily ascending to the pump, the supply pipe steadily descending to the pump.



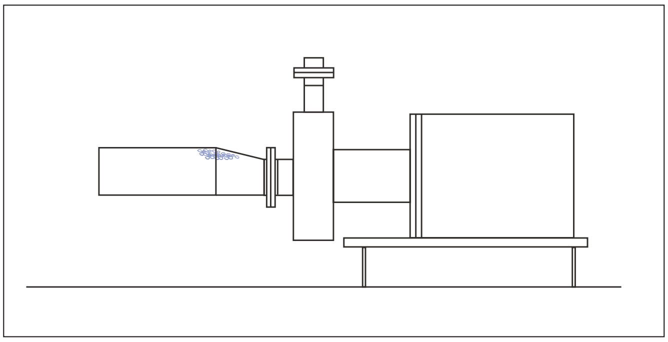
Never install a pipe bend directly upstream the pump. The distance should be the five to tenfold in diameter of the inlet socket.



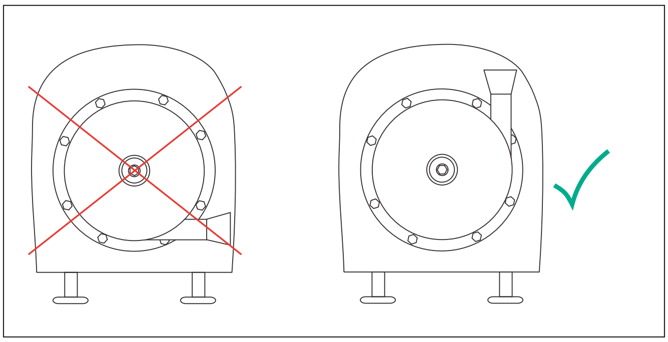
The cone of a conical suction pipe upstream the pump should be acutely conical in order to avoid deposits.



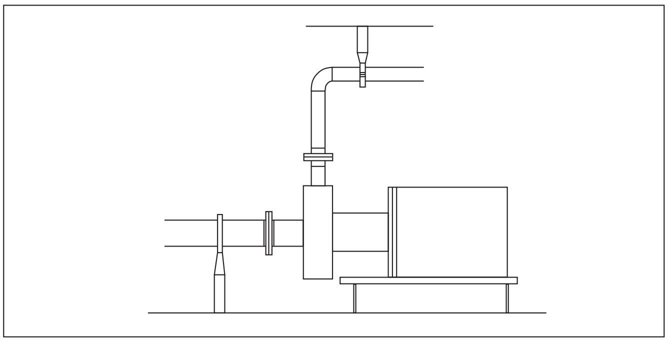
Avoid air cushions.



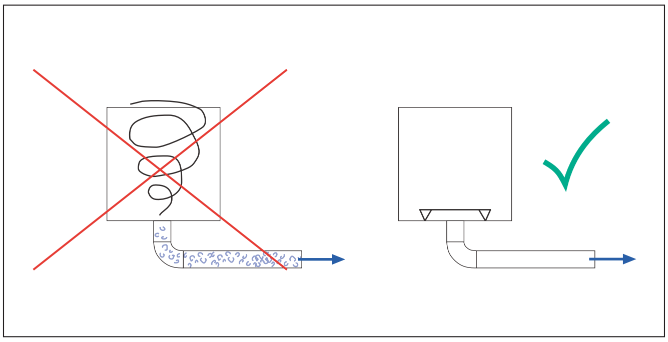
A conical suction pipe upstream the pump with top cone prevents soiling on one hand, on the other hand it leads to the formation of air cushions



The pump's delivery socket should be directed straight upward.



The pump should be adequately relieved from pipe forces acting on the pump.



Connecting the pump to a tank, air drawing-off vortex should be avoided..

Notes

Notties

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