**Technical information** 

### Selecting Centrifugal Pumps





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1 Nomenclature

А	m <sup>2</sup>	Area
A	m	Distance between measuring point and pump
		flange
a	m, mm	Width of a rectangular elbow
В	m, mm	Vertical distance from suction pipe to floor
Cv	gpm	Flow coefficient for valves, defined as the flow
- •	or	of water at 60 °F in US gallons/minute at a
		pressure drop of 1 lb/in <sup>2</sup> across the valve
cD		Resistance coefficient of a sphere in water flow
c <sub>T</sub>	(%)	Solids content in the flow
D	m (mm)	Outside diameter; maximum diameter
DN	(mm)	Nominal diameter
d	m (mm)	Inside diameter; minimum diameter
ds	m (mm)	Grain size of solids
d <sub>50</sub>	m (mm)	Mean grain size of solids
F	N	Force
f		Throttling coefficient of an orifice
f <sub>H</sub>		Conversion factor for head (KSB system)
fQ		Conversion factor for flow rate (KSB system)
fη		Conversion factor for efficiency (KSB system)
g	m/s <sup>2</sup>	Gravitational constant = $9.81 \text{ m/s}^2$
H	m	Head; discharge head
Hgeo	m	Geodetic head
Hs	m	Suction lift
H <sub>s geo</sub>	m	Vertical distance between water level and pump
0		reference plane for suction lift operation
H <sub>z geo</sub>	m	Vertical distance between pump reference plane
0		and water level for positive inlet pressure
		operation
$H_{L}$	m	Head loss
$H_0$	m	Shut-off head (at $Q = 0$ )
Ι	А	Electric current (amperage)
Κ		Dimensionless specific speed, type number
k	mm, µm	Mean absolute roughness
k		Conversion factors $k_Q$ , $k_H$ , $k_\eta$ (HI method)
k <sub>v</sub>	m³/h	Metric flow factor for valves, defined as the
		flow of water at 20 °C in cubic metres per hour
		at a pressure drop of 1 bar
L	m	Length of pipe
Ls	m	Straight length of air-filled pipe
М	Nm	Moment
NPSH <sub>r</sub>	m	NPSH required by the pump
NPSH <sub>a</sub>	m	NPSH available
Ns		Specific speed in US units
n	$\min^{-1}(rpm)$	Speed of rotation
	$s^{-1}$ (rev/s)	
nq	min <sup>-1</sup>	Specific speed in metric units
Р	kW (W)	Power; input power

pe		Pressure in suction or inlet tank	Indice	s,
PN	(bar)	Nominal pressure	а	A
Δp	bar (Pa)	Pressure rise in the pump; pressure differential		t
		$(Pa \equiv N/m^2)$	Bl	F
р	bar (Pa)	Pressure (Pa $\equiv$ N/m <sup>2</sup> = 10 <sup>-5</sup> bar)	d	C
p <sub>b</sub>	mbar (Pa)	Atmospheric pressure (barometric)		с
$p_L$	bar (Pa)	Pressure loss		t
p <sub>v</sub>	bar (Pa)	Vapour pressure of fluid pumped	dyn	Γ
Q	m <sup>3</sup> /s, m <sup>3</sup> /h	Flow rate / capacity (also in litre/s)	,	p
q <sub>air</sub>	%	Air or gas content in the fluid pumped	Е	A
$Q_{\text{off}}$	m³/h	Flow rate at switch-off pressure		S
Qon	m <sup>3</sup> /h	Flow rate at start-up pressure	Е	A
R	m (mm)	Radius		n
Re		Reynolds number	e	A
S	m	Submergence (fluid level above pump);		S
		immersion depth		0
S	mm	Wall thickness	f	F
s'	m	Difference of height between centre of pump im-	Н	ŀ
		peller inlet and centre of pump suction nozzle	in	F
Т	Nm	Torque	K	F
t	°C	Temperature	L	F
U	m	Length of undisturbed flow	m	N
U	m	Wetted perimeter of a flow section	max	N
VB	m <sup>3</sup>	Suction tank volume	min	N
$V_N$	m <sup>3</sup>	Useful volume of pump sump	Ν	N
v	m/s	Flow velocity	opt	0
W	m/s	Settling velocity of solids	opt	e
у	mm	Travel of gate valve; distance to wall	Р	F
Z	1/h	Switching cycle (frequency of starts)	р	F
Z		Number of stages	r r	F
z <sub>s,d</sub>	m	Height difference between pump discharge and	-	p
-,		suction nozzles	s	r (
			-	t
α	0	Angle of change in flow direction; opening angle	s	F
δ	0	Angle of inclination	stat	S
ζ		Loss coefficient	sys	F
η	(%)	Efficiency	- ) -	iı
η	Pa s	Dynamic viscosity	t	F
λ		Pipe friction factor		р
ν	m <sup>2</sup> /s	Kinematic viscosity	V	۲ ۱
Q	kg/m <sup>3</sup>	Density	W	F
ε	$N/m^2$	Shear stress	z	F
$\tau_{\rm f}$	N/m <sup>2</sup>	Shear stress at yield point	0	Ē
φ	1 () 111	Temperature factor; opening angle of a butter-	Ũ	t
Ť		fly valve; $\cos \varphi$ : power factor of asynchronous	1, 2, 3	
		motors	-, <b>-</b> , 3	it
ψ		Head coefficient (dimensionless head generated	I, II	N
Ψ		by impeller)	-,	a
		o, impener/		a

#### Indices, Subscripts

a	At outlet cross-section of the system; branching off
Bl	Referring to orifice bore
d	On discharge side; at dis-
u	charge nozzle; flowing
	through
dyn	Denoting dynamic com-
uyn	ponent
E	At the narrowest cross-
L	section of valves (Table 5)
Е	At suction pipe or bell-
2	mouth inlet
e	At inlet cross-section of
	system, e.g. in suction
	or inlet tank
f	Referring to carrier fluid
Н	Horizontal
in	Referring to inlet flow
K	Referring to curvature
L	Referring to losses
m	Mean value
max	Maximum value
min	Minimum value
Ν	Nominal value
opt	Optimum value; at best
1	efficiency point (BEP)
Р	Referring to pump
р	Referring to pressure
r	Reduced, for cutdown im-
	peller or impeller vanes
s	On suction side; at suc-
	tion nozzle
s	Referring to solids
stat	Static component
sys	Referring to system /
	installation
t	Referring to impeller
	prior to trimming
V	Vertical
w	Referring to water
Z	Referring to viscous fluid
0	Basic position, referred
	to individual sphere
1, 2, 3	Consecutive numbers;
	items
I, II	Number of pumps oper-
	ated

## 2

#### 2 Pump Types

Typical selection criteria for centrifugal pumps are their design data (flow rate or capacity Q, discharge head H, speed of rotation n and NPSH), the properties of the fluid pumped, the application, the place of installation and the applicable regulations, specifications, laws and codes. KSB offers a broad range of pump types to meet the most varied requirements.

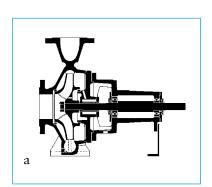
Main design features for classification are:

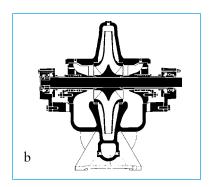
 the number of stages (singlestage / multistage),

- the position of the shaft (horizontal / vertical),
- the pump casing (radial, e.g. volute casing / axial, e.g. tubular casing),
- the number of impeller entries (single entry / double entry),
- the type of motor (dry motor / dry rotor motor, e.g. submerged motor / wet rotor motor, e.g. canned motor, submersible motor).

These features usually determine what a pump type or series looks like. An overview of typical designs according to classification features is given below (Table 1 and Figs. 1a to 1p). Other pump classification features include:

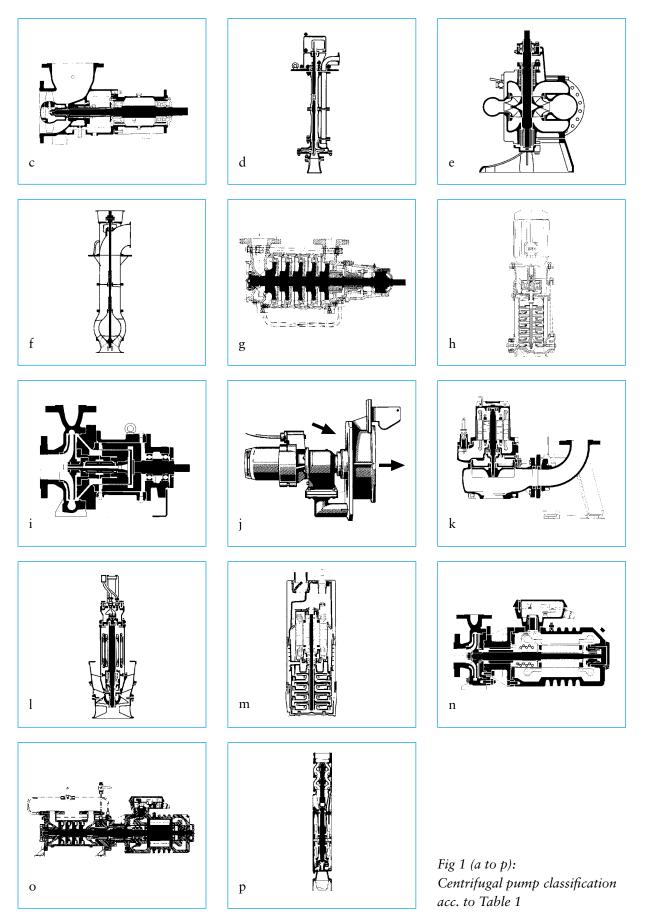
- the mode of installation, which is dealt with in section 7.1,
- the nominal diameter (for the pump size, as a function of the flow rate),
- the rated pressure (for the wall thickness of casings and flanges),
- the temperature (for example for the selection of cooling equipment for shaft seals),
- the fluid pumped (abrasive, aggressive, toxic fluids),
- the type of impeller (radial flow / axial flow depending on the specific speed),
- the self-priming ability,
- the casing partition, the position of the pump nozzles, an outer casing, etc.





#### Table 1: Centrifugal pump classification

Number of stages	Sin	Single stage					Multistage	
Shaft position	Ho	Horizontal			rtica	ıl	Horiz.	Vertic.
Casing design	Ra	dial	Axial	Radial Axial		Stage casing		
Impeller entries	1	2	1	1	2	1	1	1
Motor type, Fig. 1 Dry (standardized) motor Magnetic drive Submerged dry rotor motor (See 3.3.2) Wet rotor motor	a i	b	c j	d k	e	f 1	g	h m
(See 3.3.2)	n						0	p



9

3

#### **Selection for Pumping Water**

This section applies mainly to pumping water; the particularities of pump selection for other media are treated in sections 4, 5 and 6.

#### 3.1 Pump Data 3.1.1 Pump Flow Rate

The pump flow rate or capacity Q is the useful volume of fluid delivered to the pump discharge nozzle in a unit time in m<sup>3</sup>/s (l/s and m<sup>3</sup>/h are also used in practice, as are GPM in the US). The flow rate changes proportionally to the pump speed of rotation. Leakage flow as well as the internal clearance flows are not considered part of the pump flow rate.

#### 3.1.2 Developed Head and Developed Pressure of the Pump

The total developed head H of a pump is the useful mechanical energy in Nm transferred by the pump to the flow, per weight of fluid in N, expressed in Nm/N = m (also used to be called "metres of fluid")<sup>1)</sup>. The head develops proportionally to the square of the impeller's speed of rotation and is independent of the density  $\varrho$  of the fluid being pumped. A given centrifugal pump will impart the same head H to various fluids (with the same kinematic viscosity  $\nu$ ) regardless of their density  $\varrho$ . This statement applies to all centrifugal pumps.

The total developed head H manifests itself according to Bernoulli's equation (see section 3.2.1.1) as

- the pressure head H<sub>p</sub> proportional to the pressure difference between discharge and suction nozzles of the pump,
- the geodetic head z<sub>s,d</sub> (Figs. 8 and 9), i.e., the difference in height between discharge and suction nozzles of the pump and
- the difference of the kinetic energy head  $(v_d^2-v_s^2)/2g$  between the discharge and suction nozzles of the pump.

The pressure rise  $\Delta p$  in the pump (considering the location of the pressure measurement taps according to section 7.3!) is determined solely by the pressure head H<sub>p</sub> along with the fluid density  $\varrho$  according to the equation

 $\Delta p = \varrho \cdot g \cdot [H \cdot z_{s,d} \cdot (v_d^2 \cdot v_s^2)/2g]$ (1)

#### where

- Q Density of the fluid being pumped in kg/m<sup>3</sup>
- g Gravitational constant 9.81 m/s<sup>2</sup>
- H Total developed head of the pump in m
- z<sub>s,d</sub> Height difference between pump discharge and suction nozzles in m (see Figs. 8 and 9)

- $v_d$  Flow velocity in the discharge nozzle = 4 Q/ $\pi d_d^2$  in m/s
- $v_s$  Flow velocity in the suction nozzle = 4 Q/ $\pi d_s^2$  in m/s
- Q Flow rate of the pump at the respective nozzle in m<sup>3</sup>/s
- d Inside diameter of the respective pump nozzle in m
- $\Delta p$  Pressure rise in N/m<sup>2</sup> (for conversion to bar: 1 bar =  $100\,000$  N/m<sup>2</sup>)

High-density fluids therefore increase the pressure rise and the pump discharge pressure. The discharge pressure is the sum of the pressure rise and the inlet pressure and is limited by the strength of the pump casing. The effect of temperature on the pump's strength limits must also be considered.

#### 3.1.3 Efficiency and Input Power

The input power P of a pump (also called brake horsepower) is the mechanical power in kW or W taken by the shaft or coupling. It is proportional to the third power of the speed of rotation and is given by one of the following equations:

<sup>&</sup>lt;sup>1)</sup> In the US, the corresponding units are ft-lbf/lbm, i. e. 1 foot head = 1 footpound-force per pound mass; the numerical value of head and specific work are identical.

Р	$= \frac{\varrho \cdot g \cdot Q \cdot H}{\eta} \text{ in } W$	$= \frac{\varrho \cdot g \cdot Q \cdot H}{1000 \cdot \eta} \text{ in } kW$	$= \frac{\varrho \cdot Q \cdot H}{367 \cdot \eta} \text{ in kW} $ (2)
wh	ere		
Q	Density in kg/m <sup>3</sup>	in kg/m <sup>3</sup> in m <sup>3</sup> /s	in kg/dm <sup>3</sup> in m <sup>3</sup> /h
Q	Flow rate in m <sup>3</sup> /s	in m <sup>3</sup> /s	in m <sup>3</sup> /h

g Gravitational constant =  $9.81 \text{ m/s}^2$ 

H Total developed head in m

 $\eta$  Efficiency between 0 and <1 (not in %)

The pump efficiency  $\eta$  is given with the characteristic curves (see section 3.1.6).

The pump input power P can also be read with sufficient accuracy directly from the characteristic curves (see section 3.1.6) for density  $\varrho = 1000 \text{ kg/m}^3$ . For other densities  $\varrho$ , the input power P must be changed in proportion.

Pumping media which are more

viscous than water (see section 4) or have high concentrations of entrained solids (see section 6) will require a higher input power. This is, for example, the case when pumping sewage or waste water, see section 3.6.

The pump input power P is linearly proportional to the fluid density Q. For high-density fluids the power limits of the motor (section 3.3.3) and the torque limits (for the loading on coupling, shaft and shaft keys) must be considered.

#### 3.1.4 Speed of Rotation

When using three-phase current motors (asynchronous squirrel-

cage motors to IEC standards) the following speeds of rotation are taken as reference for pump operation:

Table 2:	Reference	speeds	of rotation
----------	-----------	--------	-------------

Number of poles	2	4	6	8	10	12	14	
Frequency		Reference speeds for the characteristic curve documen- tation in min <sup>-1</sup> (rpm)						
For 50 Hz	2900	1450	960	725	580	480	415	
For 60 Hz	3500	1750	1160	875	700	580	500	

In practice the motors run at slightly higher speeds (which depend on the power output and on the make) [1], which the pump manufacturer may consider for the pump design and selection when the customer agrees. In this case, the affinity laws described in section 3.4.3 are to be applied. The characteristic curves of submersible motor pumps and submersible borehole pumps have already been matched to the actual speed of rotation.

When using motor speed controllers (for example phase angle control for ratings up to a few kW or, in most other cases, frequency inverters), gearboxes, belt drives or when using turbines or internal combustion engines as drivers, other pump speeds are possible.

#### 3.1.5 Specific Speed and Impeller Type

The specific speed  $n_q$  is a parameter derived from a dimensional analysis which allows a comparison of impellers of various pump sizes even when their operating data differ (flow rate  $Q_{opt}$ , developed head  $H_{opt}$ , rotational speed n at the point of best efficiency  $\eta_{opt}$ ). The specific speed can be used to classify the optimum impeller design (see Fig. 2) and the corresponding pump characteristic curve (see section 3.1.6, Fig. 5).

 $n_q$  is defined as the theoretical rotational speed at which a geometrically similar impeller would run if it were of such a size as to produce 1 m of head at a flow rate of 1 m<sup>3</sup>/s at the best efficiency point. It is expressed in the same units as the speed of rotation. The specific speed can be made a truly dimensionless characteristic parameter while retaining the same numerical value by using the definition in the right-hand version of the following equation [2]:

"mixed flow" (diagonal) and

eventually axial exits (see Fig. 2).

The diffuser elements of radial

pump casings (e.g. volutes) be-

come more voluminous as long

axial exit of the flow is possible

(e.g. as in a tubular casing).

as the flow can be carried off

radially. Finally only an

$n_q = n \cdot \frac{\sqrt{Q_{opt}/1}}{(H_{opt}/1)^{3/4}}$	= 33	$63 \cdot n \cdot \frac{\sqrt{6}}{(g)}$	$\frac{\overline{Q_{opt}}}{(H_{opt})^{3/4}}$	(3)
where $Q_{opt}$ in m <sup>3</sup> /s H <sub>opt</sub> in m n in rpm n <sub>q</sub> in metric uni	ts n <sub>q</sub>	Dimensi	= Flow rate at η = Developed hea = Pump speed onless parameter ional constant 9.8	

For multistage pumps the developed head  $H_{opt}$  at best efficiency for a **single** stage and for doubleentry impellers, the optimum flow rate  $Q_{opt}$  for only **one** impeller half are to be used.

As the specific speed  $n_q$  increases, there is a continuous change from the originally radial exits of the impellers to

	c	1
Approximate	reference	values
rippioninate	1010101100	raraeo.

n <sub>a</sub> up	to approx. 25	Radial high head impeller
1 -	to approx. 40	Radial medium head impeller
up	to approx. 70	Radial low head impeller
up	to approx. 160	Mixed flow impeller
approx. from	140 to 400	Axial flow impeller (propeller)

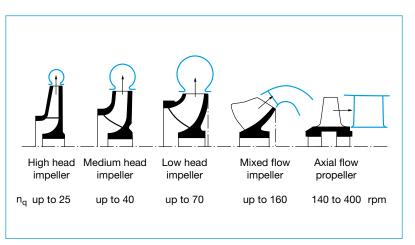


Fig. 2: Effect of the specific speed  $n_q$  on the design of centrifugal pump impellers. The diffuser elements (casings) of single stage pumps are outlined.

Using Fig. 3 it is possible to determine n<sub>q</sub> graphically. Further types of impellers are shown in Fig. 4: Star impellers are used in self-priming pumps. Peripheral impellers extend the specific speed range to lower values down to approximately  $n_q = 5$ (peripheral pumps can be designed with up to three stages). For even lower specific speeds, rotary (for example progressive cavity pumps with  $n_q = 0.1$  to 3) or reciprocating positive displacement pumps (piston pumps) are to be preferred.

The value of the specific speed is one of the influencing parameters required when converting the pump characteristic curves for pumping viscous or solids-laden media (see sections 4 and 6).

In English-language pump literature the true dimensionless specific speed is sometimes designated as the "type number K". In the US, the term N<sub>s</sub> is used, which is calculated using gallons/min (GPM), feet and rpm. The conversion factors are:

$$K = n_q / 52.9 N_s = n_q / 51.6$$
 (4)

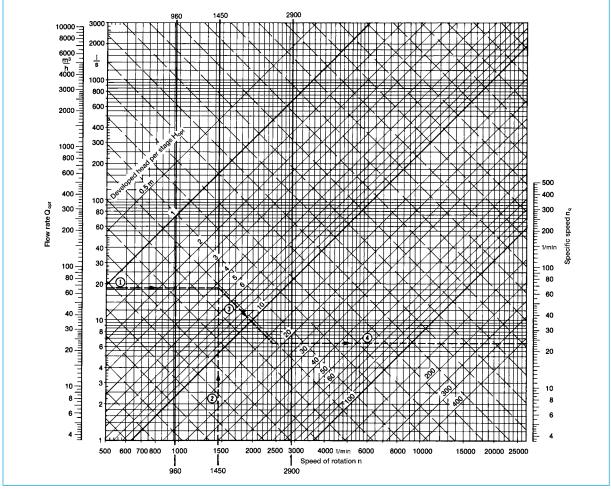
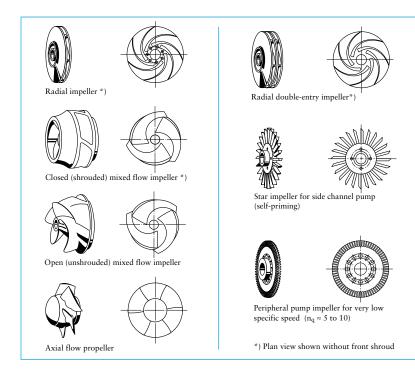


Fig. 3: Nomograph to determine specific speed  $n_q$  (enlarged view on p. 80) Example:  $Q_{opt} = 66 \ m^3/h = 18.3 \ l/s; n = 1450 \ rpm, H_{opt} = 17.5 \ m.$  Found:  $n_q = 23$  (metric units).



#### 3.1.6 Pump Characteristic Curves

Unlike positive displacement pumps (such as piston pumps), centrifugal pumps deliver a variable flow rate Q (increasing with decreasing head H) when operating at constant speed. They are therefore able to accommodate changes in the system curve (see section 3.2.2). The input power P and hence the efficiency  $\eta$  as well as the NPSH<sub>r</sub> (see section 3.5.4) are dependent on the flow rate.

Fig. 4: Impeller types for clear liquids

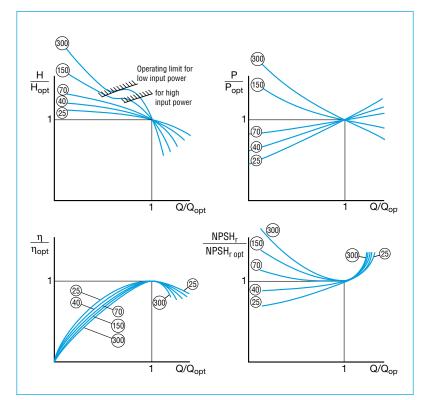


Fig. 5: Effect of specific speed  $n_q$  on centrifugal pump characteristic curves (Not drawn to scale! For NPSH<sub>r</sub>, see section 3.5.4).

The relationship of these values is shown graphically in the pump characteristic curves, whose shape is influenced by the specific speed  $n_q$  and which document the performance of a centrifugal pump (see Fig. 5 for a comparison of characteristics and Fig. 6 for examples). The head curve of the pump is also referred to as the H/Q curve.

The H/Q curve can be steep or flat. For a steep curve, the flow rate Q changes less for a given change of developed head  $\Delta$ H than for a flat curve (Fig. 7). This can be advantageous when controlling the flow rate.

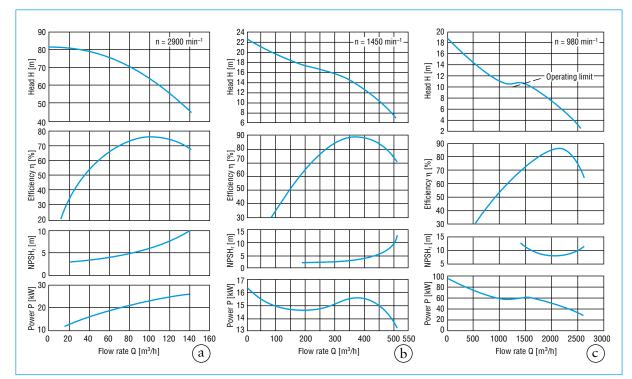


Fig. 6: Three examples of characteristic curves for pumps of differing specific speeds. a: radial impeller,  $n_q \approx 20$ ; b: mixed flow impeller,  $n_q \approx 80$ ; c: axial flow impeller,  $n_q \approx 200$ . (For NPSH<sub>r</sub> see section 3.5.4)

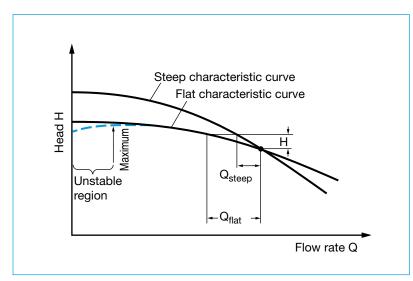


Fig. 7: Steep, flat or unstable characteristic curve

H/Q characteristics normally have a stable curve, which means that the developed head falls as the flow rate Q increases. For low specific speeds, the head H may – in the low flow range – drop as the flow rate Q decreases, i.e., the curve is unstable (shown by the dash line in Fig. 7). This type of pump characteristic curve need only be avoided when two intersections with the system curve could result, in particular when the pump is to be used for parallel operation at low flow rates (see section 3.4.4) or when it is pumping into a vessel which can store energy (accumulator filled with gas or steam). In all other cases the unstable curve is just as good as the stable characteristic.

Unless noted otherwise, the characteristic curves apply for the density  $\varrho$  and the kinematic viscosity  $\nu$  of cold, deaerated water.

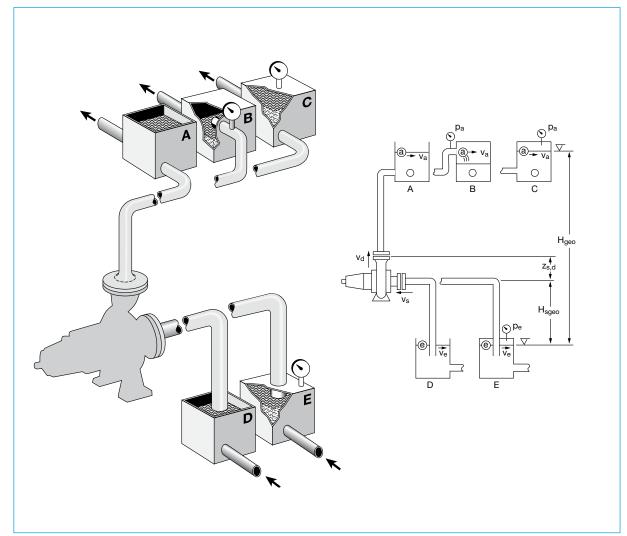


Fig. 8: Centrifugal pump system with variously designed vessels in suction lift operation

*A* = *Open tank with pipe ending below the water level* 

- *B* = Closed pressure vessel with free flow from the pipe ending above the water level
- *C* = *Closed pressure vessel with pipe ending below the water level*
- *D* = *Open suction/inlet tank*
- *E* = *Closed suction/inlet tank*

 $v_a$  and  $v_e$  are the (usually negligible) flow velocities at position (a) in tanks A and C and at position (e) in tanks D and E. In case B,  $v_a$  is the non-negligible exit velocity from the pipe end at (a).

# 3.2System Data3.2.1System Head3.2.1.1Bernoulli's Equation

Bernoulli's equation expresses the equivalence of energy in geodetic (potential) energy, static pressure and kinetic energy form. The system head  $H_{sys}$  for an assumed frictionless, inviscid flow is composed of the following three parts (see Figs. 8 and 9):

• H<sub>geo</sub> (geodetic head) is the difference in height between the liquid level on the inlet

and discharge sides. If the discharge pipe ends above the liquid level, the centre of the exit plane is used as reference for the height (see Figs 8B and 9B).

 (p<sub>a</sub> - p<sub>e</sub>)/(Q · g) is the pressure head difference between the inlet and outlet tank, applic-

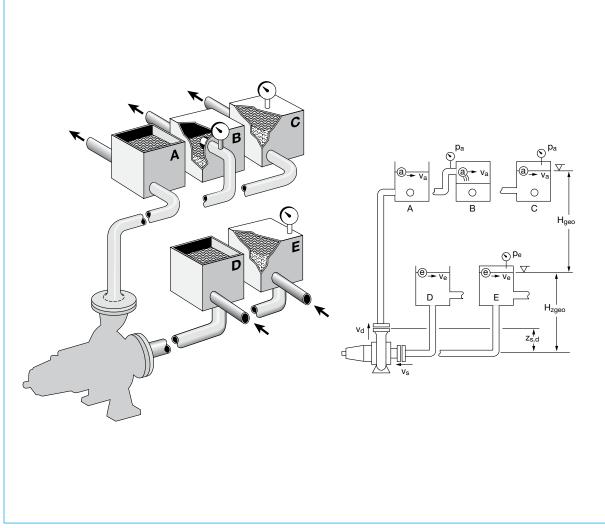


Fig. 9: Centrifugal pump system with variously designed vessels in suction head (positive inlet pressure) operation. Legend as in Fig. 8.

able when at least one of the tanks is closed as for B, C or E (see Figs. 8B, C, E, 9B, C, E).

•  $(v_a^2 - v_e^2)/2g$  is the difference in the velocity heads between the tanks.

For a physically real flow, the friction losses (pressure head losses) must be added to these components:

 $\Sigma H_L$  is the sum of the head losses (flow resistance in the

piping, valves, fittings, etc in the suction and discharge lines as well as the entrance and exit losses, see section 3.2.1.2), and is referred to as the system pressure loss.

The sum of all four components yields the system head H<sub>sys</sub>:

$$H_{sys} = H_{geo} + (p_a - p_e) / (\varrho \cdot g) + (v_a^2 - v_e^2) / 2g + \Sigma H_L$$
(5)

where all the heads H are in m, all the pressures p are in Pa (1 bar = 100 000 Pa), all velocities v are in m/s, the density  $\varrho$  is in kg/m<sup>3</sup>, the gravitational constant is g = 9.81 m/s<sup>2</sup>. The difference of the velocity heads can often be neglected in practice. When at least one tank is closed as for B, C or E (see Figs. 8B, C, E, 9B, C, E), Eq. 5 can be simplified as

 $H_{sys} \approx H_{geo} + (p_a - p_e)/(\varrho \cdot g) + \sum H_L$ (6)

and further simplified when both tanks are open as for A and D (see Figs. 8A, D and 9A, D) as

$$H_{svs} \approx H_{geo} + \Sigma H_L$$

#### 3.2.1.2 Pressure Loss Due to Flow Resistances

The pressure loss  $p_L$  is caused by wall friction in the pipes and flow resistances in valves, fittings, etc. It can be calculated from the head loss  $H_L$ , which is independent of the density  $\varrho$ , using the following equation:

$$p_{\rm L} = \varrho \cdot g \cdot H_{\rm L} \tag{8}$$

#### where

(7)

- Q Density in kg/m<sup>3</sup>
- g Gravitational constant 9.81 m/s<sup>2</sup>
- $H_L\,$  Head loss in m
- pL Pressure loss in Pa (1 bar = 100 000 Pa)

#### 3.2.1.2.1 Head Loss in Straight Pipes

The head loss for flow in straight pipes with circular cross-sections is given in general by

$$H_{L} = \lambda \cdot \frac{L}{d} \cdot \frac{v^{2}}{2g}$$
 (9)

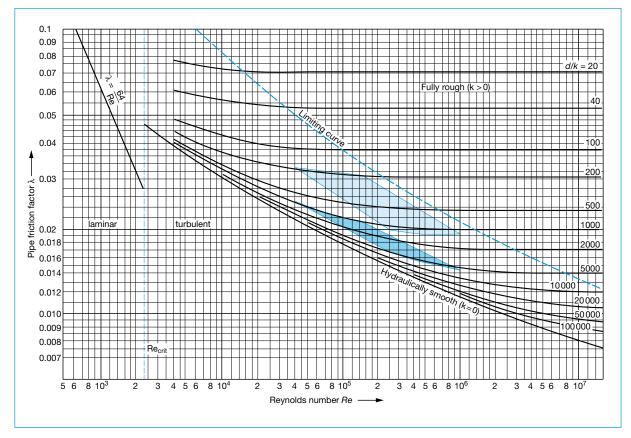
where

- $\lambda$  Pipe friction factor according to Eqs. (12) to (14)
- L Length of pipe in m
- d Pipe inside diameter in m

v Flow velocity in m/s (=  $4Q/\pi d^2$  for Q in m<sup>3</sup>/s)

g Gravitational constant 9.81 m/s<sup>2</sup>

Fig. 10: Pipe friction factor  $\lambda$  as a function of the Reynolds number Re and the relative roughness d/k (enlarged view on p. 81)



For pipes with non-circular cross-sections the following applies:

where

- A Cross-sectional flow area in  $m^2$
- U Wetted perimeter of the cross-section A in m; for open channels the free fluid surface is not counted as part of the perimeter.

Recommended flow velocities

for cold water	
Inlet piping	0.7-1.5 m/s
Discharge piping	1.0-2.0 m/s
for hot water	
Inlet piping	0.5-1.0 m/s
Discharge piping	

The pipe friction factor  $\lambda$  has been determined experimentally and is shown in Fig. 10. It varies with the flow conditions of the liquid and the relative roughness d/k of the pipe surface. The flow conditions are expressed according to the affinity laws (dimensional analysis) using the Reynolds' number Re. For circular pipes, this is:

 $\operatorname{Re} = \mathbf{v} \cdot \mathbf{d} / \boldsymbol{\nu}$ 

(11)

#### where

- v Flow velocity in m/s
- $(= 4Q/\pi d^2 \text{ for } Q \text{ in } m^3/s)$
- d Pipe inside diameter in m
- $\nu$  Kinematic viscosity in m<sup>2</sup>/s (for water at 20 °C exactly  $1.00 \cdot (10)^{-6}$  m<sup>2</sup>/s).

For non-circular pipes, Eq. 10 is to be applied for determining d.

For hydraulically smooth pipes (for example drawn steel tubing

or plastic pipes made of polyethylene (PE) or polyvinyl chloride (PVC)) or for laminar flow, λ can be calculated:

In the laminar flow region (Re <2320) the friction factor is independent of the roughness:

$\lambda = 64/\text{Re}$	(12)

For turbulent flow (Re > 2320) the test results can be represented by the following empirical relationship defined by Eck (up to Re <  $10^8$  the errors are smaller than 1%):

$$\lambda = \frac{0.309}{(\lg \frac{\text{Re}}{7})^2}$$
 (13)

In Fig. 10 it can be seen that the pipe friction factor depends on another dimensionless parameter, the relative roughness of the pipe inner surface d/k; k is the average absolute roughness of the pipe inner surface, for which approximate values are given in Table 3. Note: both d and k must be expressed in the same units, for example mm!

As shown in Fig. 10, above a limiting curve,  $\lambda$  is dependent only on the relative roughness d/k. The following empirical equation by Moody can be used in this region:

#### $\lambda = 0.0055 + 0.15/\sqrt[3]{(d/k)}$ (14)

For practical use, the head losses  $H_L$  per 100 m of straight steel pipe are shown in Fig. 11 as a function of the flow rate Q and pipe inside diameter d. The values are valid only for cold, clean water or for fluids with the same kinematic viscosity, for completely filled pipes and for an absolute roughness of the pipe inner surface of k = 0.05 mm, i.e., for new seamless or longitudinally welded pipes. (For the pipe inside diameters, see Table 4).

The effect of an increased surface roughness k will be demonstrated in the following for a frequently used region in Fig. 11 (nominal diameter 50 to 300 mm, flow velocity 0.8 to 3.0 m/s). The dark-shaded region in Fig. 11 corresponds to the similarly marked region in Fig. 10 for an absolute roughness k = 0.05 mm. For a roughness increased by a factor 6 (slightly incrusted old steel pipe with k = 0.30 mm), the pipe friction factor  $\lambda$  (proportional to the head loss H<sub>L</sub>) in the lightly shaded region in Fig. 10 is only 25% to 60% higher than before.

For sewage pipes the increased roughness caused by soiling must be taken into consideration (see section 3.6). For pipes with a large degree of incrustation, the actual head loss can only be determined experimentally. Deviations from the nominal diameter change the head loss considerably, since the pipe inside diameter enters Eq. (9) to the 5th power! (For example, a 5% reduction in the inside diameter changes the head loss by 30%). Therefore the nominal diameter may not be used as the pipe inside diameter for the calculations!

<u>k</u>

	1							T		
Pipe material	Cond	ition of pipe in	iterior	1 m	5	10	50	100	500 100	$5000 \ 10^4$
Steel	new,	seamless	skin acid-cleaned galvanized							
		longitudin- ally welded,	skin bituminized galvanized cemented							
		riveted								
	used,	moderately r slight incrust heavy incrust after cleaning	ation tation							
Asbestos cement Heavy clay (drainage) Concrete Spun concrete	new new new,	unfinished with smooth unfinished								
Reinforced concrete All concretes	new, used,	with smooth with smooth with smooth	finish							
Metal pipes Glass, plastic Rubber tubing Wood	new, new	drawn not embrittle	ed	-						
Masonry	after	long exposure	to water							
		k in mm	→ 0	0.001	0.00	5 0.01	0.05	0.1	0.5 1	5 10

*Table 3: Approximate average roughness height k (absolute roughness) for pipes* 

Table 4: Inside diameter d and wall thickness s in mm and weight of typical commercial steel pipes and their	
<i>water content in kg/m to ENV 10 220 (formerly DIN ISO 4200). D = outside diameter, s = wall thickness</i>	

		All dimer Seamless	nsions in m	ım Welded		Seamless weight in		Welded j weight in	
DN	D	s *	d	s **	d	Pipe	Water	Pipe	Water
15	21.3	2.0	17.3	1.8	17.7	0.952	0.235	0.866	0.246
20	26.9	2.0	22.9	1.8	23.3	1.23	0.412	1.11	0.426
25	33.7	2.3	29.1	2.0	29.7	1.78	0.665	1.56	0.692
32	42.4	2.6	37.2	2.3	37.8	2.55	1.09	2.27	1.12
40	48.3	2.6	43.1	2.3	43.7	2.93	1.46	2.61	1.50
50	60.3	2.9	54.5	2.3	55.7	4.11	2.33	3.29	2.44
65	76.1	2.9	70.3	2.6	70.9	4.71	3.88	5.24	3.95
80	88.9	3.2	82.5	2.9	83.1	6.76	5.34	6.15	5.42
100	114.3	3.6	107.1	3.2	107.9	9.83	9.00	8.77	9.14
125	139.7	4.0	131.7	3.6	132.5	13.4	13.6	12.1	13.8
150	168.3	4.5	159.3	4.0	160.3	18.2	19.9	16.2	20.2
200	219.1	6.3	206.5	4.5	210.1	33.1	33.5	23.8	34.7
250	273.0	6.3	260.4	5.0	263.0	41.4	53.2	33.0	54.3
300	323.9	7.1	309.7	5.6	312.7	55.5	75.3	44.0	76.8
350	355.6	8.0	339.6	5.6	344.4	68.6	90.5	48.3	93.1
400	406.4	8.8	388.8	6.3	393.8	86.3	118.7	62.2	121.7
500	508.0	11.0	486.0	6.3	495.4	135	185.4	77.9	192.7
600	610.0	12.5	585.0	6.3	597.4	184	268.6	93.8	280.2

\* above nominal diameter DN 32 identical to DIN 2448

\*\* above nominal diameter DN 25 identical to DIN 2458

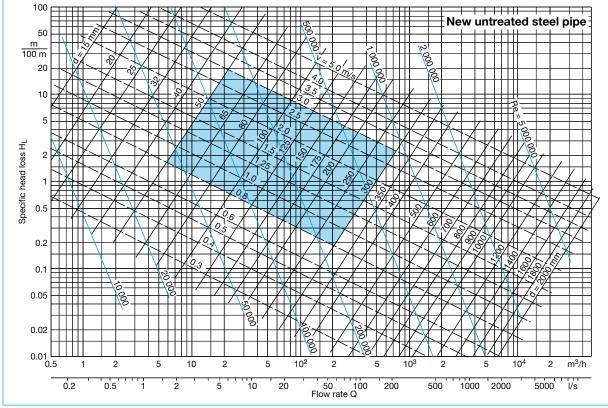


Fig. 11: Head losses  $H_L$  for new steel pipes (k = 0.05 mm) (enlarged view on p. 82)

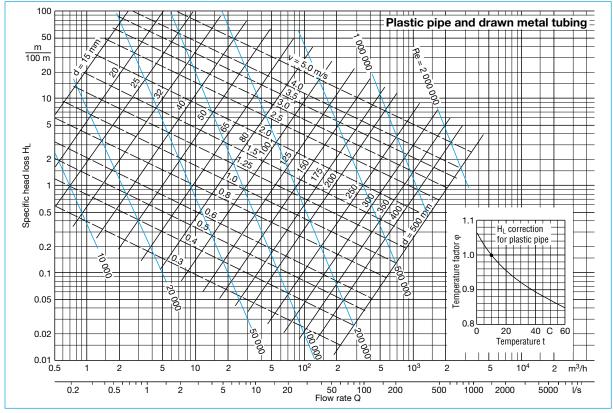
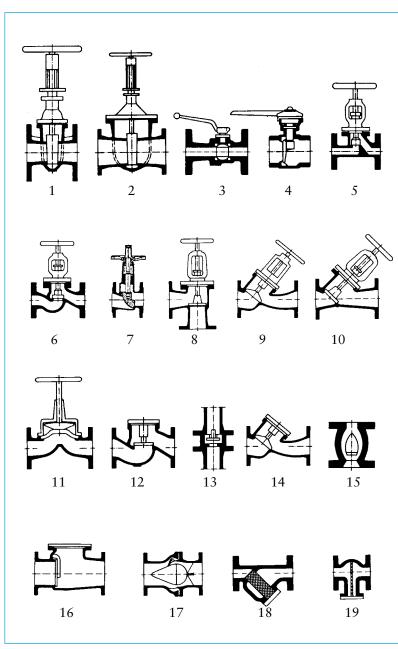


Fig. 12: Head losses  $H_L$  for hydraulically smooth pipes (k = 0) (enlarged view on p. 83). For plastic pipe when  $t \neq 10$  °C multiply by the temperature factor  $\varphi$ .

The head losses  $H_L$  in plastic (for example PE or PVC) pipes or smooth drawn metal piping are very low thanks to the smooth pipe surface. They are shown in Fig. 12 and valid for water at 10 °C. At other temperatures, the loss for plastic pipes must be multiplied with a temperature correction factor indicated in Fig. 12 to account for their large thermal expansion. For sewage or other untreated water, an additional 20-30% head loss should be taken into consideration for potential deposits (see section 3.6).



*Fig.* 13: *Schematic representation of the valve designs listed in Table 5* 

#### 3.2.1.2.2 Head Loss in Valves and Fittings

The head loss in valves and fittings is given by

$$H_{\rm L} = \zeta \cdot v^2 / 2g$$
 (15)

where

- $\zeta$  Loss coefficient
- v Flow velocity in a characteristic cross-section A (for example the flange) in m/s
- g Gravitational constant 9.81 m/s<sup>2</sup>

Tables 5 to 8 and Figures 13 to 15 contain information about the various loss coefficients  $\zeta$  for valves and fittings for operation with cold water.

The minimum and maximum in Table 5 bracket the values given in the most important technical literature and apply to valves which have a steady approach flow and which are fully open. The losses attributable to straightening of the flow disturbances over a length of pipe equivalent to 12 x DN downstream of the valve are included in the  $\zeta$  value in accordance with VDI/VDE 2173 guidelines. Depending on the inlet and exit flow conditions, the valve models used and the development objectives (i.e. inexpensive vs. energy-saving valves), the loss values can vary dramatically.

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r .	Type of valve / titting		De- sign	Loss 15	Loss coefficient $\zeta$ for DN 15  20  25  32	icient ( 25	ς tor D  32	= <mark>4</mark> 0	50	65	80	100	125	150	200	250	300	400	500	600	800	1000	Comment
<b>v</b> , -	Slide disc valves (d <sub>F</sub> = DN)	min max		$0.1 \\ 0.65$	↓ 9.0	0.55	5 0.5	0.5	0.45	6.0	0.35	0.3										$0.1 \\ 0.3$	$\int For d_{\rm E} < DN$
	dy gate valves	min max	2						0.25 0.32		-		$0.21 \\ 0.26$	$0.19 \\ 0.25$	$0.18 \\ 0.23$	$0.17\\0.22$	0.16	$0.15 \\ 0.19$	$0.13 \\ 0.18$	$0.12 \\ 0.16$	$0.11 \\ 0.15$		$\int cf. \text{ footnote } ^1)$
<b>H</b> $\smile$	lug valves	min max	ŝ	$0.10 \\ 0.15$	<b>0.1</b> 0	0.09	60.0 6	9 0.08	0.08	0.07	0.07	0.06	0.05	0.05	0.04	0.03	0.03	0.02					For $d_{\rm E} < DN$ $\zeta = 0.4$ to 1.1
н	<sub>ly</sub> PN 2.5 10	min max	4					$\begin{array}{c} 0.90\\ 1.20\end{array}$			$\begin{array}{c} 0.26\\ 0.70\end{array}$	$0.20 \\ 0.62$	$0.14 \\ 0.56$		0.09 0.42		0.37		0.33	0.33	0.3		
-	valves PN 16 25	min max							$2.50^{*}$	* 2.30*	1.55 * 2.10*	1.55         1.30         1.08         0.84           2.10*         1.90*         1.70*         1.50*	$^{+}_{* 1.70*}$	$^{0.84}{* 1.50*}$	$^{0.75}_{* 1.30}$	0.56	0.48 0.90	$0.40 \\ 0.83$	<b>▲</b> 0.76	0.71	0.67;	0.67* 0.63*	* Also for PN 40
$\sim$	Globe valves, forged	min max	5			6.0 6.8	$\downarrow\downarrow\downarrow$		· 6.0 · 6.8														
<u> </u>	Globe valves, cast	min max	9	3.0 6.0	$\downarrow\downarrow\downarrow$												· 3.0 · 6.0						$\zeta = 2$ to 3 can be achieved for optimized valve
$\overline{}$	Compact valves	min max		$0.3 \\ 0.3$	0.9 0.9	$0.6 \\ 1.9$	<b>0</b> .6	1.0	1:1	1.9 ▲	2.2	2.2	2.3	2.5	. 1.1 2.5								
~	Angle valves	min max	8	2.0 3.1	$\downarrow\downarrow\downarrow$	$   \overline{ }$	♦ 3.1	3.4	3.8	4.1	4. 4.	4.7	5.0	5.3	5.7	6.0	6.3	• 2.0 6.6					
r .	Y-valves	min max	6	1.5 2.6	$\downarrow\downarrow\downarrow$												• 1.5 • 2.6						
<b>U</b> )	Straight-through valves	min max	10	$0.6 \\ 1.6$	$\downarrow\downarrow\downarrow$													• 0.6 • 1.6					
Ι	Diaphragm valves	min max	11	0.8 2.7	$\downarrow\downarrow\downarrow$									• 0.8 • 2.7									
r s	Non-return valves, straight seat	min max	12	3.0 6.0	$\downarrow\downarrow\downarrow$										3.0 6.0								
2 F	valves,	min max	13	3.2 3.4	<b>▲</b> 6.00	3.5	3.6		. 3.2 4.2	3.7 5.0	5.0 6.4	7.3 8.2	4.3 6.6	↓↓			. 4.3 4.6						Axially expanded as from DN 125
r s	Non-return valves, slanted seat	min max	14	2.5 3.0	4.2	2.2	2.1	2.0	1.9	1.7	1.6	1.5	¥				1.5 . 3.0						
щ		min max	15						$1.0 \\ 3.0$	6.0	0.8	0.7	0.6	0.5	0.4	0.4	• 0.4 3.0	(7.0)	(6.1)	(5.5)	(4.5)	(4.0)	() In groups
<b>U</b> )	Swing check valves	min max	16	0.5 3.0	$\downarrow\downarrow\downarrow$		Ţ	♦ 0.5	0.4	↓							• 0.4	0.3	↓			$0.3 \\ 3.0$	Swing check valves with- out levers and weights <sup>2</sup> )
ł	Hydrostops $v = 4 m/s$ v = 3 m/s v = 2 m/s		17						$\begin{array}{c} 0.9\\ 1.8\\ 5.0 \end{array}$			3.0 4.0 6.0		3.0 4.5 8.0	2.5 4.0 7.5	2.5 4.0 6.5	$\begin{array}{c} 1.2\\ 1.8\\ 6.0\end{array}$	2.2 3.4 7.0					
	Filters		18					2.8	↓ .								2.8						In clean condition
ر م	Strainers		۲y	_				1.U															

Round elbow	α		15°		30°		45°		60°		90°	
į			Surf smooth		Surf smooth		Surf smooth		Surf smooth		Surf smooth	
	ζ for	R = 0	0.07	0.10	0.14	0.20	0.25	0.35	0.50	0.70	1.15	1.30
	ζ for	R = d	0.03	_	0.07	_	0.14	0.34	0.19	0.46	0.21	0.51
	$\zeta$ for	R = 2 d	0.03	-	0.06	-	0.09	0.19	0.12	0.26	0.14	0.30
	$\zeta$ for	R 5 d	0.03	-	0.06	-	0.08	0.16	0.10	0.20	0.10	0.20
Welded bend	Number ferential	of circum- welds	_	_	_	_	2	_	3	_	3	_
	ζ		-	-	-	-	0.15	-	0.20	-	0.25	-

#### *Table 6: Loss coefficients* $\zeta$ *in elbows and bends*

Note: For the branch fittings in Table 7 and the adapters of Table 8, one must differentiate between the **irreversible** pressure loss (reduction in pressure)

$$p_{\rm L} = \zeta \cdot \varrho \cdot v_1^2 / 2 \tag{16}$$

where

- p<sub>L</sub> Pressure loss in Pa
- $\zeta$  Loss coefficient
- $\varrho$  Density in kg/m<sup>3</sup>

v Flow velocity in m/s

and the **reversible** pressure change of the frictionless flow according to Bernoulli's equation (see 3.2.1.1):

$$p_2 - p_1 = \varrho \cdot (v_1^2 - v_2^2)/2$$
 (17)

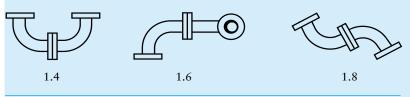
For accelerated flows (for example a reduction in the pipe diameter),  $p_2-p_1$  is always negative, for decelerated flows (e.g. pipe expansion) it is always positive. When calculating the net pressure change as the arithmetic sum of  $p_L$  and  $p_2-p_1$ , the pressure losses from Eq. 16 are always to be subtracted.

Often the so-called  $k_v$  value is used instead of the loss coef-

#### Table 7: Loss coefficients $\zeta$ for fittings

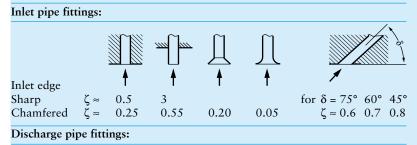
Combinations of pipe bends (elbows):

The  $\zeta$  value of the single 90° elbow should not be doubled, but only be multiplied by the factors indicated to obtain the pressure loss caused by the combination of elbows illustrated.



#### **Expansion** joints:

Bellows-type expansion joint	
with/without guide pipe	$\zeta \approx 0.3/2.0$
Compensation tube bend	$\zeta \approx 0.6$ to 0.8
Creased compensation tube bend	$\zeta \approx 1.3$ to 1.6
Bellows-type compensation tube bend	$\zeta \approx 3.2$ to 4



 $\approx 1$  downstream of an adequate length of straight pipe with an approximately uniform velocity distribution in the outlet cross-section

≈ 2 for a very irregular velocity distribution, for example immediately downstream of a pipe fitting or a valve, etc.

Continued on next page

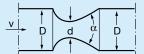
Expansion			Cont	traction		
$\begin{array}{c} \downarrow \\ \underline{v_1} \\ \underline{v_1} \\ \underline{i} \\ \underline$	α			$\begin{array}{c c} \downarrow \\ d \\ \uparrow \end{array}$	$\begin{array}{c} \uparrow \\ \downarrow \\$	↓ d
Type I	II		III		IV	
Туре	d/D	0.5	0.6	0.7	0.8	0.9
Ι	ζ ≈	0.56	0.41	0.26	0.13	0.04
$ \begin{array}{c} \alpha = 8^{\circ} \\ \text{II for} \begin{cases} \alpha = 15^{\circ} \\ \alpha = 20^{\circ} \end{array} \end{array} $	ζ ≈	0.07	0.05	0.03	0.02	0.01
II for $\alpha = 15^{\circ}$	ζ ≈	0.15	0.11	0.07	0.03	0.01
$\alpha = 20^{\circ}$	ζ ≈	0.23	0.17	0.11	0.05	0.02
III	ζ≈	4.80	2.01	0.88	0.34	0.11
IV for $20^{\circ} < \alpha < 40^{\circ}$	ζ ≈	0.21	0.10	0.05	0.02	0.01

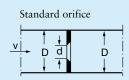
#### *Table 8: Loss coefficients* $\zeta$ *for adapters*

#### Table 7 (continued)

#### Flow meters:

Short Venturi tube  $\alpha$  = 30°





#### $\zeta$ is referred to the velocity v at diameter D.

-						
Diameter ratio d/D	= 0.30	0.40	0.50	0.60	0.70	0.80
Area ratio $m = (d/D)$	$(0)^2 = 0.09$	0.16	0.25	0.36	0.49	0.64
Short Venturi tube	$\zeta \approx 21$	6	2	0.7	0.3	0.2
Standard orifice	$\zeta\approx 300$	85	30	12	4.5	2

Water meters (volume meters)  $\zeta \approx 10$ 

For domestic water meters, a max. pressure drop of 1 bar is specified for the rated load. In practice, the actual pressure loss is seldom higher.

Branch fittings (of equal diameter)

#### Note:

The loss coefficients  $\zeta_a$  for the branched-off flow  $Q_a$  or  $\zeta_d$  for the main flow  $Q_d = Q - Q_a$  refer to the velocity of the total flow Q in the branch. On the basis of this definition,  $\zeta_a$  or  $\zeta_d$  may have negative values; in this case, they are indicative of a pressure **gain** instead of a pressure **loss**. This is not to be confused with the reversible pressure **changes** according to Bernoulli's equation (see notes to Tables 7 and 8).

$Q_a/Q =$	0.2	0.4	0.6	0.8	1
$Q_d \xrightarrow{\qquad} Q_a$	$\begin{array}{ll} \zeta_a\approx -0.4\\ \zeta_d\approx & 0.17 \end{array}$	0.08 0.30	0.47 0.41	0.72 0.51	0.91 -
$Q \xrightarrow{\qquad} Q_d$	$\begin{array}{l} \zeta_a \approx  0.88 \\ \zeta_d \approx -0.08 \end{array}$	0.89 -0.05	0.95 0.07	1.10 0.21	1.28 -
$Q_d \xrightarrow[Q_a]{45^\circ} Q_a$	$\begin{array}{l} \zeta_a\approx -0.38\\ \zeta_d\approx  0.17 \end{array}$	0 0.19	0.22 0.09	0.37 -0.17	0.37
$Q = \overbrace{Q_{d}}^{45^\circ} Q_{d}$	$\begin{array}{l} \zeta_a \approx  0.68 \\ \zeta_d \approx -0.06 \end{array}$	0.50 -0.04	0.38 0.07	0.35 0.20	0.48 -

ficient  $\zeta$  when calculating the pressure loss for water in valves:

$$p_L = (Q / k_v)^2 \cdot Q / 1000$$
 (18)

#### where

- Q Volume rate of flow in  $m^3/h$  (!)
- $\rho$  Density of water in kg/m<sup>3</sup>

p<sub>L</sub> Pressure loss in bar (!)

The  $k_v$  value is the flow rate in  $m^3/h$  which would result from a pressure drop of 1 bar through the valve for cold water. It correlates with the pressure loss  $p_L$  in bar with the flow rate Q in  $m^3/h$ . The notation  $k_{vs}$  is used for a fully open valve.

Conversion for cold water:

$$\zeta \approx 16 \cdot d^4 / k_v^2$$
 (19)

where

d Reference (nominal) diameter of the valve in cm (!)

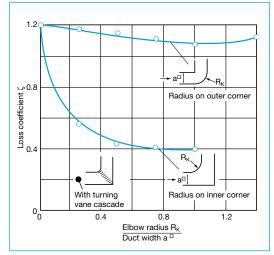


Fig. 14: Effect of rounding off the inner and outer side of elbows in square ducts on the loss coefficient  $\zeta$ 

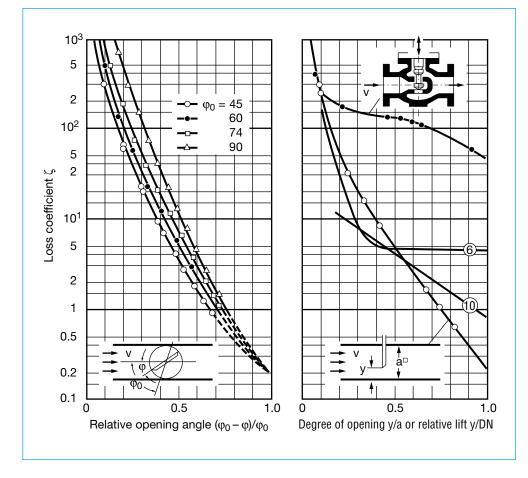


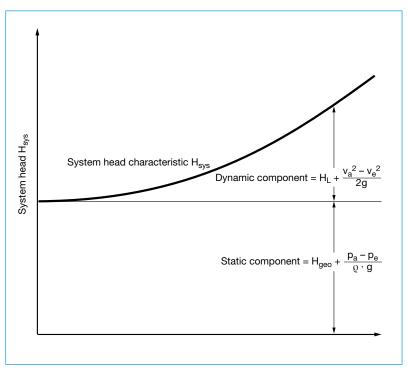
Fig. 15: Loss coefficients  $\zeta$  of butterfly valves, globe valves and gate valves as a function of the opening angle or degree of opening (The numbers designate the types illustrated in Fig. 13)

#### 3.2.2 System Characteristic Curve

The system characteristic curve plots the head  $H_{sys}$  required by the system as a function of the flow rate Q. It is composed of the so-called "static" and "dynamic" components (see Fig. 16)<sup>3</sup>.

The static component consists of the geodetic head  $H_{geo}$  and the pressure head difference  $(p_a-p_e)/(\varrho \cdot g)$  between the inlet

Fig. 16: System characteristic curve  $H_{sys}$  with static and dynamic components



<sup>&</sup>lt;sup>3</sup> One must be careful to distinguish between this use of "static" and "dynamic" components and the precisely defined "static head" and "dynamic head" used in fluid dynamics, since the "dynamic component" of the system head curve consists of both "static head" (i.e. pressure losses) and "dynamic head" (i.e. velocity or kinetic energy head).

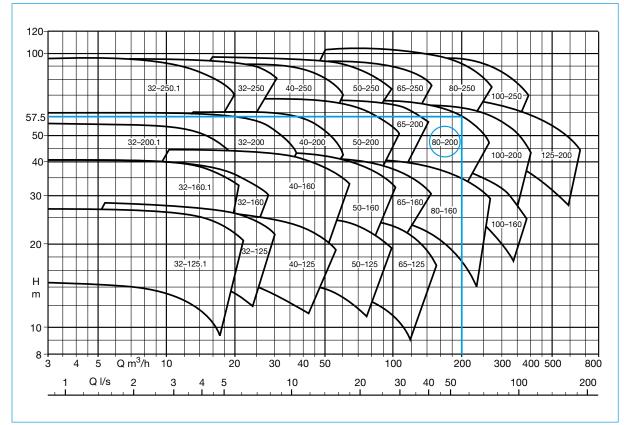


Fig. 17: Selection chart for a volute casing pump series for n = 2900 rpm (First number = nominal diameter of the discharge nozzle, second number = nominal impeller diameter)

and outlet tanks, which are independent of the flow rate. The pressure head difference is zero when both tanks are open to the atmosphere.

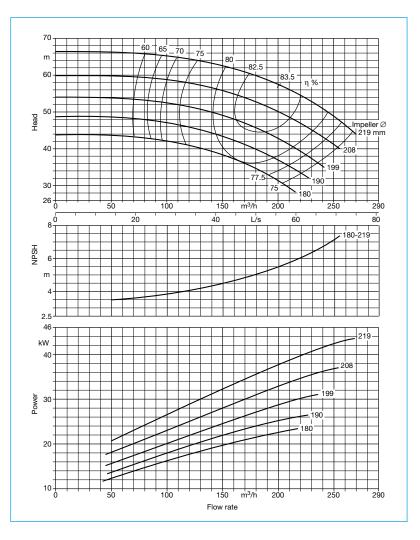
The dynamic component consists of the head loss  $H_L$ , which increases as the square of the flow rate Q (see section 3.2.1.2), and of the change in velocity head  $(v_a^2 - v_e^2)/2g$  between the inlet and outlet cross-sections of the system. Two points are sufficient to calculate this parabola, one at Q = 0 and one at any point Q > 0.

For pipe systems connected one after the other (series connection) the individual system curves  $H_{sys1}$ ,  $H_{sys2}$  etc. are plotted as functions of Q, and the heads for each flow rate are added to obtain the total system curve  $H_{sys} = f(Q)$ .

For branched piping systems the system curves  $H_{sys1}$ ,  $H_{sys2}$ , etc. of the individual branches between the flow dividers are each calculated as functions of Q. The flow rates Q<sub>1</sub>, Q<sub>2</sub>, etc. of all branches in parallel for each given head  $H_{sys}$  are then added to determine the total system curve  $H_{sys} = f(Q)$  for all the branches together. The sections before and after the flow dividers must be added as for a series connection.

## 3.3Pump Selection3.3.1Hydraulic Aspects

The data required for selecting a pump size, i.e. the flow rate Q and the head H of the desired operating point are assumed to be known from the system characteristic curve; the electric mains frequency is also given. With these values it is possible to choose the pump size, the speed of rotation and, if necessary, the number of stages, from the selection chart in the sales literature (see Figs. 17 and 19). Further details of the chosen pump such as the efficiency  $\eta$ , the input power P, the required  $NPSH_r$  (see section 3.5.4) and the reduced impeller diameter D<sub>r</sub> can then be determined from



*Fig. 18: Complete characteristics of a centrifugal pump* 

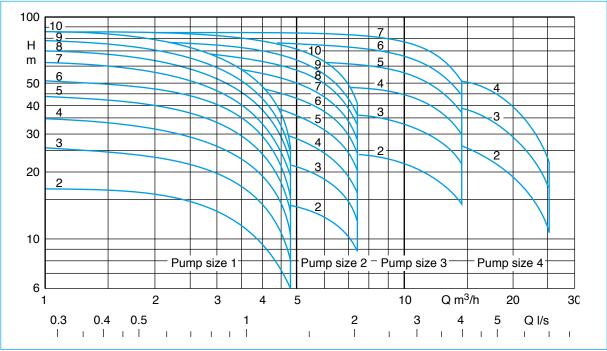


Fig. 19: Selection chart for a multistage pump series for n = 2900 rpm

the individual characteristic curve (for example see Fig. 18).

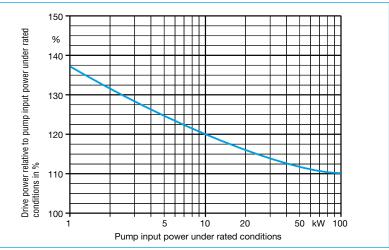
If there are no specific reasons for doing otherwise, a pump should be selected so that the operating point lies near its best efficiency point  $Q_{opt}$  (= flow rate at which efficiency is highest, BEP). The limits  $Q_{min}$ and  $Q_{max}$  (for example due to vibration behaviour, noise emission as well as radial and axial forces) are given in the product literature or can be determined by inquiry [1].

To conclude the selection, the NPSH conditions must be checked as described in section 3.5.

A multistage pump is chosen using the same general procedure; its selection chart shows the number of stages in addition to the pump size (Fig. 19).

For pumps operating in series (one after the other) the developed heads  $H_1$ ,  $H_2$ , etc. of the individual characteristic curves must be added (after subtracting any head losses which occur between them) to obtain the total characteristic H = f(Q).

For pumps operating in parallel, the individual characteristics  $H_1$ ,  $H_2$ , etc. = f(Q) are first reduced by the head losses occurring up to the common node (head loss  $H_L$  calculation according to section 3.2.1.2) and plotted versus Q. Then the flow rates Q of the reduced characteristics are added to produce the effective characteristic curve of a "virtual" pump. This characteristic interacts with the system curve  $H_{sys}$  for the rest of the system through the common node.



*Fig. 20: Drive power as a function of rated pump input power at the operationg point* 

Example as per ISO 9905, 5199 and 9908 (Class I, II and III)

#### 3.3.2 Mechanical Aspects

When selecting a pump the mechanical aspects require attention in addition to the hydraulics. Several examples are:

- the effects of the maximum discharge pressure and temperature of the fluid pumped on the operating limits,
- the choice of the best shaft sealing method and cooling requirements,
- the vibration and noise emissions,
- the choice of the materials of construction to avoid corrosion and wear while keeping in mind their strength and temperature limits.

These and other similar requirements are often specific to certain industries and even to individual customers and must be addressed using the product literature [1] or by consulting the design department.

## 3.3.3Motor Selection3.3.3.1Determining Motor Power

Operation of a centrifugal pump is subject to deviations from rated speed and fluctuations in the flow volume handled, and, consequently, changes in the operating point (see section 3.4.1). In particular if steep power curves are involved (see Figs. 5 and 6), this may result in a higher required pump input power P than originally specified. For practical purposes, a safety allowance is therefore added when the appropriate motor size is selected. Safety allowances may be specified by the purchaser, or laid down in technical codes, see Fig. 20. The safety allowances stipulated by individual associations are shown in the relevant type series literature [1] or the customer's specification.

When energy-saving control methods are used (e.g., speed control systems), the maximum power peaks which may possibly occur must be taken into account.

If a pump is selected for a product with a density lower than that of water, the motor power required may have to be determined on the basis of the density of water (for example, during the performance test or acceptance test in the test bay).

Typical efficiencies  $\eta$  and power factors cos  $\varphi$  of standardized IP 54 motors at 50 Hz are shown in Fig. 21, and the curves of efficiency  $\eta$  and power factor cos  $\varphi$  as a function of relative motor load P/PN in Fig. 22.

Table 9 lists types of enclosure that provide protection of electric motors against ingress of foreign objects or water, and of persons against accidental contact.

The specific heat build-up in both electric motors and flexible couplings during start-up as well as the risk of premature contactor wear limit the frequency of starts. Reference values for the maximum permissible number of starts Z are given in table 10, unless otherwise specified.

Submersible motor pumps (Figs. 1j to 1m) are ready-assembled pump units whose motors need not be selected individually [7]. Their electrical characteristics are given in the type series literature. The motor is filled with air and can be operated submerged in the product handled thanks to a – in most cases – double-acting shaft seal with a paraffin oil barrier.

Table 9: Types of enclosure for electric motors to EN 60 529 and DIN/VDE 0530, Part 5

The type of protective enclosure is indicated l	by the IP code as follows:
Code letters (International Protection)	IP
First digit (0 to 6 or X if not applicable)	Х
Second digit (0 to 8 or X if not applicable)	Х
Alternatively letters A, B, C, D and H, M, S,	W – for special purposes only.

Alterna	tivel	y letters A, B, C, D and H, M, S,	w – for special purposes only.		
Key to digits:		Protection of electrical equipment against ingress of solid objects	Protection of persons against accidental contact by		
First digit	0 1 2 3 4 5	<pre>(not protected) &gt; 50 mm in dia. &gt; 12.5 mm in dia. &gt; 2.5 mm in dia. &gt; 1.0 mm in dia. protected against dust (limited ingress permitted, no harmful deposits)</pre>	(not protected) back of the hand finger tool wire wire		
	6	totally protected against dust Protection against ingress of wa	wire ter with harmful consequences		
Second digit	0 1 2 3 4 5 6 7 8	(not protected) vertical dripwater dripwater up to 15° from the ve sprays (60° from the vertical) sprays (all directions) low-pressure jets of water strong jets of water (heavy sea) temporary flooding permanent flooding			

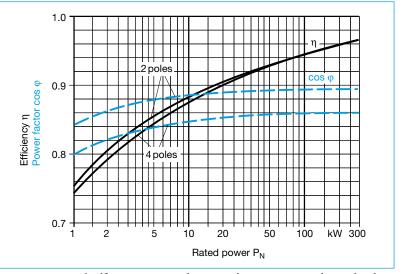


Fig. 21: Typical efficiencies  $\eta$  and power factors  $\cos \phi$  of standardized motors, IP 54 enclosure, at 50 Hz as a function of motor power  $P_N$ 

Motor installation	Dry	Wet (submersible motors)		
Motors up to 4 kW	15	30		
Motors up to 7.5 kW	15	25		
Motors up to 11 kW	12	25		
Motors up to 30 kW	12	20		
Motors above 30 kW	10	10		

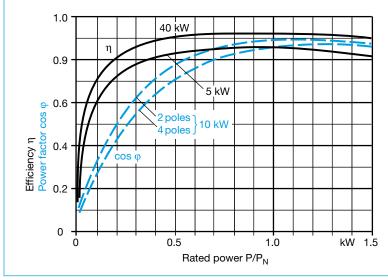


Fig. 22: Curve of efficiency  $\eta$  and power factor  $\cos \phi$  of standardized IP 54 motors plotted over relative motor power P/P<sub>N</sub>

Submersible borehole pumps, which are mostly used for extracting water from wells, are another type of ready-assembled units whose motors need not be selected individually (Fig. 1p). On these pumps, the rotor and the windings are immersed in water [7]. Their electrical characteristics and permissible frequency of starts are indicated in the type series literature [1].

#### 3.3.3.2 Motors for Seal-less Pumps

Seal-less pumps are frequently used for handling aggressive, toxic, highly volatile or valuable fluids in the chemical and petrochemical industries. They include magnetic-drive pumps (Fig. 1f) and canned motor pumps (Figs. 1n and 1o). A mag-drive pump is driven by a primary magnetic field rotating outside its flameproof enclosure and running in synchronization with the secondary magnets inside the enclosure [12]. The primary component in turn is coupled to a commercial dry driver. The impeller of a canned motor pump is mounted directly on the motor shaft, so that the rotor is surrounded by the fluid pumped. It is separated from the stator windings by the can [7].

Seal-less pump sets are generally selected with the help of computerized selection programs, taking into account the following considerations:

- The rotor is surrounded by the fluid pumped, whose kinematic viscosity υ (see section 4.1) must be known, as it influences friction losses and therefore the motor power required.
- Metal cans or containment shrouds (for example made of 2.4610) cause eddy current losses, resulting in an increase in the motor power required. Non-metal shrouds in magdrive pumps do not have this effect.

- The vapour pressure of the fluid pumped must be known, so as to avoid bearing damage caused by dry running when the fluid has evaporated. It is advisable to install monitoring equipment signalling dry running conditions, if any.
- Data on specific fluid properties such as its solids content and any tendency to solidify or polymerize or form incrustations and deposits, need to be available at the time of selection.

#### 3.3.3.3 Starting Characteristics

The pump torque Tp transmitted by the shaft coupling is directly related to the power P and speed of rotation n. During pump start-up, this torque follows an almost parabolical curve as a function of the speed of rotation [10], as shown in Fig. 23. The torque provided by the asynchronous motor must, however, be higher so as to enable the rotor to run up to duty speed. Together with the voltage, this motor torque has a direct effect on the motor's current input, and the latter in turn on heat build-up in the motor windings. The aim, therefore, is to prevent unwanted heat buildup in the motor by limiting the run-up period and/or current inrush [2] (see also Table 11).

Starting method	Type of equipment	Current input (mains load)	Run-up time	Heat build- up in motor during start-up	Mechani- cal loading	Hydraulic loading	Cost relation	Recommended motor designs	Comments
D. o. l.	Contactor (mecha- nical)	4–8 · I <sub>N</sub>	Approx. 0.5–5 s	High	Very high	Very high	1	All	Mostly limited to ≤4 kW by energy supply companies
Star- delta	Contactor combi- nation (mecha- nical)	<sup>1</sup> / <sub>3</sub> of d. o. l. values	Approx. 3–10 s	High	Very high	Very high	1.5–3	All; canned mo- tors and sub- mersible motors subject to a major drop in speed during switchover	lated for motors
Reduced voltage	Autotrans- former, mostly 70% tap- ping	0.49 times the d. o. l. values	Approx. 3–10 s	High	High	High	5–15	All	No currentless phase during switchover (gradually re- placed by soft starters)
Soft start	Soft starter (power electro- nics)	Continuous- ly variable; typically 3 · I <sub>N</sub>	Approx. 10–20 s	High	Low	Low	5-15	All	Run-up and run- down continu- ously variable via ramps for each individual load appllication; no hydraulic surges
Fre- quency inverter	Frequency inverter (power electro- nics)	1 · I <sub>N</sub>	0–60 s	Low	Low	Low	Approx. 30	All	Too expensive to use solely for run- up and run-down purposes; better suited for open- or closed-loop control

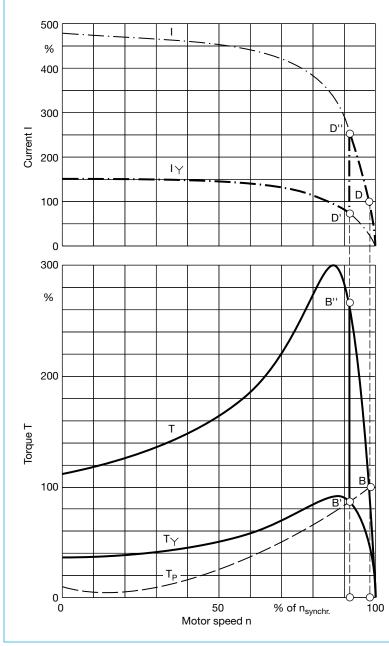
Table 11: Starting methods for asynchronous motors

In the case of d.o.l. starting (where the full mains voltage is instantly applied to the motor once it is switched on), the full starting torque is instantly available and the unit runs up to its duty speed in a very short period of time. For the motor itself, this is the most favourable starting method. But at up to 4-8 times the rated current, the starting current of the d.o.l. method places a high load on the electricity supply mains, particularly if large motors are involved, and may cause problematic voltage drops in electrical equipment in their vicinity. For motor operation on public

low-voltage grids (380 V), the regulations laid down by the energy supply companies for d.o.l. starting of motors of 5.5 kW and above must be complied with. If the grid is not suitable for d.o.l starting, the motor can be started up with reduced voltages, using one of the following methods:

**Star-delta starting** is the most frequent, since most inexpensive, way of reducing the starting current. During normal operation, the motor runs in delta, so that the full mains voltage (for example 400 V) is applied to the motor windings. For start-up, however, the windings are star-connected, so that the voltage at the windings is reduced by a factor of 0.58 relative to the mains voltage. This reduces the starting current and torque to one third of the values of d.o.l. starting, resulting in a longer start-up process.

The motor runs up in star connection beyond pull-out torque up to the maximum speed of rotation at point B' in Fig. 23. Then, switchover to delta is effected and the motor continues to accelerate up to rated speed. During the switchover period of approx. 0.1 s, the current supply to the motor is interrupted and the speed drops. On pump sets with a low moment of inertia (canned motors and submersible motors), this speed reduction may be so pronounced that switchover to delta may result in almost the full starting current being applied after all, same as with d.o.l. starting. An **autotransformer** also serves to reduce voltage at the motor windings and – unlike star-delta starting – allows selection of the actual voltage reduction. A 70% tapping of the transformer, for instance, will bring down the start-up torque and current supplied by the mains to 49%



of the values for d.o.l. starting. The fact that current supply is never interrupted is another advantage of autotransformers.

Soft starters are used for electronic continuous variation of the voltage at the motor windings in accordance with the dimmer principle. This means that the start-up time and starting current can be freely selected within the motor's permissible operating limits (heat losses due to slip!). Special constraints regarding the frequency of starts (contrary to Table 10) have to be heeded [1].

**Frequency inverters** (usually for open- or closed-loop control) provide a soft starting option without the need for any additional equipment. For this purpose, the output frequency and voltage of the frequency inverter (see section 3.4.3) are increased continuously from a minimum value to the required value, without exceeding the motor's rated current.

Fig. 23: Starting curve for current I and torque T of squirrel-cage motors in star-delta connection ( $\Upsilon$  = star connection;  $\Delta$  = delta connection; P = pump)

#### 3.4 Pump Performance and Control [4], [6], [8] 3.4.1

#### **Operating Point**

The operating point of a centrifugal pump, also called its duty point, is given by the intersection of the pump characteristic curve (see section 3.1.6) with the system characteristic curve (see section 3.2.2). The flow rate Q and the developed head H are both determined by the intersection. To change the operating point either the system curve or the pump curve must be changed.

A system characteristic curve for pumping water can only be changed

- by changing the flow resistance (for example, by changing the setting of a throttling device, by installing an orifice or a bypass line, by rebuilding the piping or by its becoming incrusted) and/or
- by changing the static head component (for example, with a different water level or tank pressure).

A **pump characteristic curve** can be changed

- by changing the speed of rotation (see section 3.4.3),
- by starting or stopping pumps operated in series or parallel (see sections 3.4.4 or 3.4.5),
- for pumps with radial impellers, by changing the impeller's outside diameter (see section 3.4.6),
- for pumps with mixed flow impellers, by installing or

changing the setting of installed pre-swirl control equipment (see section 3.4.8),

• for axial flow (propeller) pumps, by changing the blade pitch setting (see section 3.4.9).

Please note: the effect of these measures for changing the characteristic curve can only be predicted for non-cavitating operation (see section 3.5).

#### 3.4.2 Flow Control by Throttling

Changing the flow rate Q by operating a throttle valve is the simplest flow control method not only for a single adjustment of the flow rate but also for its continuous control, since it requires the least investment. But it is also the most energy wasting method, since the flow energy is converted irreversibly to heat.

Fig. 24 illustrates this process: by intentionally increasing the system resistance (for example by throttling a valve on the

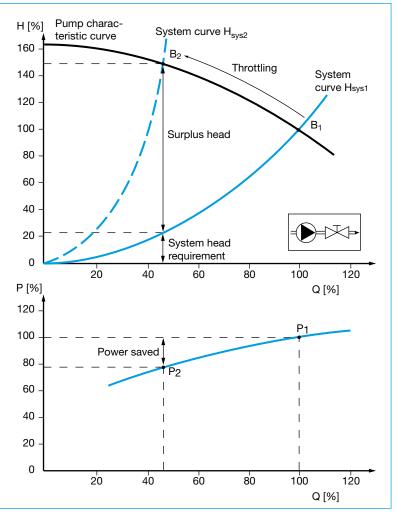


Fig. 24: Change of the operating point and power saved by throttling a pump whose power curve has a positive slope

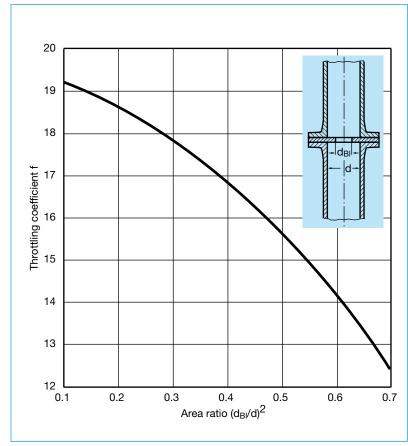


Fig. 25: Orifice plate and its throttling coefficient f

pump discharge side) the original system curve H<sub>sys1</sub> becomes steeper and transforms into  $H_{sys2}$ . For a constant pump speed, the operating point  $B_1$  on the pump characteristic moves to  $B_2$  at a lower flow rate. The pump develops a larger head than would be necessary for the system; this surplus head is eliminated in the throttle valve. The hydraulic energy is irreversibly converted into heat which is transported away by the flow. This loss is acceptable when the control range is small or when such control is only seldom needed. The power saved is shown in the lower part of the figure; it is only moderate compared with the large surplus head produced.

The same is principally true of the installation of a fixed, **sharp-edged orifice plate** in the discharge piping, which can be justified for low power or short operating periods. The necessary hole diameter  $d_{Bl}$  of the orifice is calculated from the head difference to be throttled  $\Delta$ H, using the following equation:

$$d_{Bl} = f \cdot \sqrt{Q/\sqrt{g \cdot \Delta H}}$$
 (20)

where

- $d_{BI}$  Hole diameter of the orifice in mm
- f Throttling or pressure drop coefficient acc. to Fig. 25
- Q Flow rate in m<sup>3</sup>/h

- g Gravitational constant 9.81 m/s<sup>2</sup>
- $\Delta H$  Head difference to be throttled in m

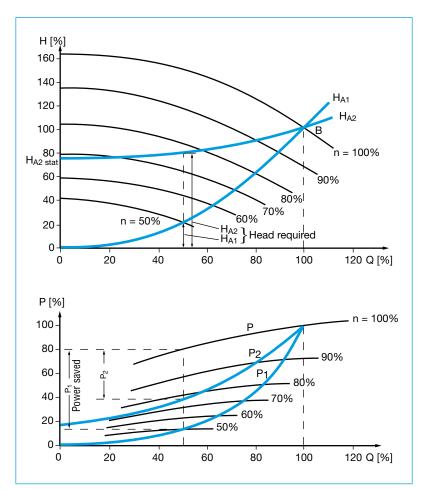
Since the area ratio  $(d_{Bl}/d)^2$ must be estimated in advance, an iterative calculation is necessary (plotting the calculated vs. the estimated diameter  $d_{Bl}$  is recommended so that after two iterations the correct value can be directly interpolated, see example calculation 8.20).

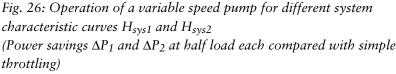
#### 3.4.3 Variable Speed Flow Control

At various speeds of rotation n, a centrifugal pump has different characteristic curves, which are related to each other by the affinity laws. If the characteristics H and P as functions of Q are known for a speed  $n_1$ , then all points on the characteristic curve for  $n_2$  can be calculated by the following equations:

$Q_2 = Q_1 \cdot n_2/n_1$	(21)
$H_2 = H_1 \cdot (n_2/n_1)^2$	(22)
$\mathbf{P}_2 = \mathbf{P}_1 \cdot (\mathbf{n}_2/\mathbf{n}_1)^3$	(23)

Eq. (23) is valid only as long as the efficiency  $\eta$  does not decrease as the speed n is reduced. With a change of speed, the operating point is also shifted (see section 3.4.1). Fig. 26 shows the H/Q curves for several speeds of rotation; each curve has an intersection with the system characteristic H<sub>sys1</sub>. The operating point B moves along this system curve to smaller flow rates when the speed of rotation is reduced.





If the system curve is a parabola through the origin as for  $H_{sys1}$ in the example, the developed head H according to Eq. (22) is reduced to one fourth its value and the required driving power in Eq. (23) to one eighth its value when the speed is halved. The lower part of Fig. 26 shows the extent of the savings  $\Delta P_1$ compared with simple throttling.

If the system curve is a parabola with a large static head component as for  $H_{sys2}$ , it is possible that the pump characteristic at reduced speed has no intersec-

tion with it and hence, that no operating point results; the lower speed range is then of no use and could be eliminated. The potential power savings  $\Delta P_2$ at a given flow rate Q are less than for the system curve H<sub>sys1</sub> as shown in the lower part of the diagram [4]. The improvement compared with throttling decreases as the static head component H<sub>sys,stat</sub> increases (i.e., for a lower dynamic head component H<sub>sys,dyn</sub>).

Variation of the speed usually means varying the electrical driving frequency, which must be considered when choosing the motor. The expenditure for variable speed drives is not low, but it is amortized quickly for pumps which are used often and which are frequently required to run at reduced flows with small static head component H<sub>sys,stat</sub> [8]. This is particularly the case for pumps in heating systems.

#### 3.4.4 Parallel Operation of Centrifugal Pumps

Where one pump is unable to deliver the required flow Q at the operating point, it is possible to have two or more pumps working in parallel in the same piping system, each with its own non-return valve (Fig. 27). Parallel operation of pumps is easier when their shutoff heads H<sub>0</sub> are all equal, which is the case for identical pumps. If the shutoff heads H<sub>0</sub> differ, the lowest shutoff head marks the point on the common H/Q curve for the minimum flow rate Q<sub>min</sub>, below which no parallel operation is possible, since the nonreturn valve of the pump with smaller shutoff head will be held shut by the other pump(s).

During parallel pumping it must be kept in mind that after stopping one of two identical centrifugal pumps (Fig. 27), the flow rate  $Q_{single}$  of the remaining pump does not fall to half of  $Q_{parallel}$ , but rather increases to more than half. The remaining pump might then immediately run at an operating point  $B_{single}$  above its design point, which must be considered when checking the NPSH values (see section 3.5) and the drive power (see section 3.1.3). The reason for this behaviour is the parabolic shape of the system characteristic  $H_{sys}$ . For the same reason, the reverse procedure of taking a second identical pump on line does not double the flow rate  $Q_{single}$  of the pump that was already running, but rather increases the flow rate less than that:

$$Q_{parallel} < 2 \cdot Q_{single}$$
 (24)

This effect when starting or stopping one additional pump is more intense when the system curve is steeper or when the pump characteristic is flatter. As long as both pumps I and II are running, the total flow rate Q<sub>parallel</sub> is the sum of Q<sub>I</sub> and Q<sub>II</sub>, i.e.:

$$Q_{\text{parallel}} = Q_{\text{I}} + Q_{\text{II}}$$
 (25)

To compute the characteristic curve for parallel operation see section 3.3.1.

Starting or stopping individual pumps operated in parallel does save energy, but it allows only a stepped control of the flow rate. For continuously variable control, at least one of the pumps must be fitted with a variable speed drive or a control valve must be installed in the common discharge piping [4].

If centrifugal pumps running at fixed speeds and having unstable characteristics (see Fig. 7 in section 3.1.6) are run in parallel, difficulties can arise when bringing another pump online.

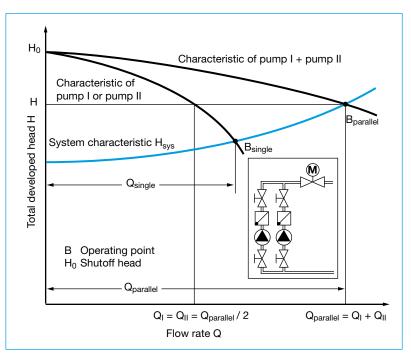
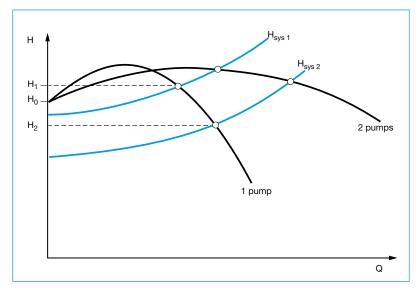


Fig. 27: Parallel operation of 2 identical centrifugal pumps with stable characteristic curves

The problems occur when the developed head  $H_1$  of the pump running is larger than the shutoff head (i.e., the developed head at Q = 0) of the pump to be started; the second pump is unable to overcome the pressure on its non-return valve (Fig. 28, System curve  $H_{sys1}$ ). Pumps with unstable characteristics are not suitable for such a low flow operation. (For a lower system curve  $H_{sys2}$  they would be perfectly able to operate properly since the developed head  $H_2$ of the pump running is lower than the shutoff head  $H_0$  of the pump to be started).



*Fig. 28: Parallel operation of 2 identical centrifugal pumps with unstable characteristics* 

## 3.4.5 Series Operation

In series operation, the pumps are connected one after the other so that the developed heads can be added for a given flow rate. This means that the discharge pressure of the first pump is the inlet pressure for the second pump, which must be considered for the choice of shaft seal and for the strength of the casing. For this reason multistage pumps are usually used for such applications (except for the hydraulic transport of solids, see section 6). They do not pose these shaft sealing problems.

## 3.4.6 Turning Down Impellers

If the flow rate or developed head of a radial or mixed flow centrifugal pump are to be reduced **permanently**, the outside diameter D of the impeller should be reduced. The reduction should be limited to the value for which the impeller vanes still overlap when viewed radially. The documentation of the pump characteristics (Fig. 18) usually shows curves for several diameters D (in mm).

Impellers made from hard materials, such as those used for solids-handling pumps, or from stainless steel sheet metal, as well as single vane impellers (Fig. 43) and star or peripheral pump impellers **cannot** be turned down. (The same is true for under-filing as described in section 3.4.7). For multistage pumps, usually only the vanes but not the shrouds of the impellers are cut back. It is sometimes possible to simply remove the impeller and diffuser of one stage of a multistage pump and replace them with a blind stage (two concentric cylindrical casings to guide the flow) instead of cutting back the impeller vanes. Impellers with a noncylindrical exit section are either turned down or have only their blades cut back as specified in the characteristic curve literature (for example, as shown in Fig. 29).

If the impeller diameter only needs to be reduced slightly, a rule of thumb can be applied. An exact calculation cannot be made, since the geometrical similarity of the vane angle and exit width are not preserved when turning down the impeller. The following approximate relationship exists between Q, H and the impeller diameter D to be found (averaged, if required):

 $(D_t/D_r)^2 \approx Q_t/Q_r \approx H_t/H_r$  (26)

where subscript t designates the condition before the reduction

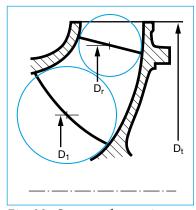


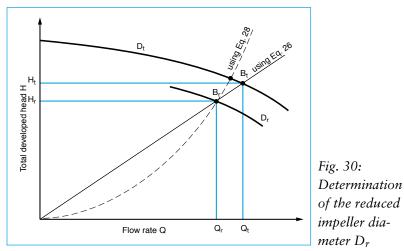
Fig. 29: Contour for cutting back the vanes of an impeller with a mixed flow exit

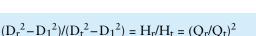
of the impeller outer diameter and index r the condition after the reduction. The required (average) reduced diameter results as:

$$D_r \approx D_t \cdot \sqrt{(Q_r/Q_t)} \approx D_t \cdot \sqrt{(H_r/H_t)}$$
(27)

The parameters needed to determine the reduced diameter can be found as shown in Fig. 30: in the H/Q curve (linear scales required!) a line is drawn connecting the origin (careful: some scales do not start at zero!) and the new operating point  $B_r$ . The extension of the line intersects the characteristic curve for full diameter Dt at the point Bt. In this way the values of Q and H with the subscripts t and r can be found, which are used with Eq. (27) to find the desired reduced diameter D<sub>r</sub>.

The ISO 9906 method is more accurate, but also more involved through the consideration of the average diameter D<sub>1</sub> of the impeller leading edge (subscript 1), valid for  $n_q < 79$  and for a change of diameter < 5%, as long as the vane angle and the impeller width remain constant. Thus using the nomenclature of Figs. 29 and 30:





A solution is only possible when  $D_1$  is known and when a parabola  $H \sim Q^2$  is drawn through the reduced operating point  $B_r$  (with  $H_r$  and  $Q_r$ ), not a line as

#### 3.4.7 Under-filing of Impeller Vanes

A small, permanent increase of the developed head at the best efficiency point (up to 4 - 6%) can be achieved for radial impellers by filing the back sides of the backward-curved vanes, i.e., by sharpening the vanes on the concave side, as shown in Fig.

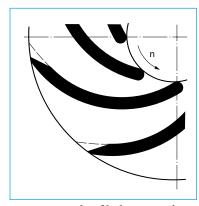


Fig. 31: Under-filed vanes of a radial impeller

in Fig. 30, which intersects the base H/Q curve for diameter  $D_t$  at a different point  $B_t$  (with different  $H_t$  and  $Q_t$ ).

31. The shutoff head does not change. This method is suitable for minor final corrections.

## 3.4.8 Pre-Swirl Control of the Flow

For tubular casing pumps with mixed flow impellers, the pump characteristic can be influenced by changing the pre-rotation in the impeller inlet flow. Such pre-swirl control equipment is often fitted to control the flow rate. The various characteristic curves are then shown in the product literature labelled with the control setting (Fig. 32).

#### 3.4.9

(28)

# Flow Rate Control or Change by Blade Pitch Adjustment

The characteristic curves of axial flow (propeller) pumps can be altered by changing the setting of the propeller blade pitch. The setting can be fixed and firmly bolted or a device to change the blade pitch during operation can be used to control the flow rate. The blade pitch angles are

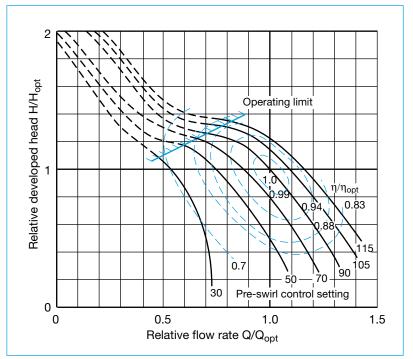


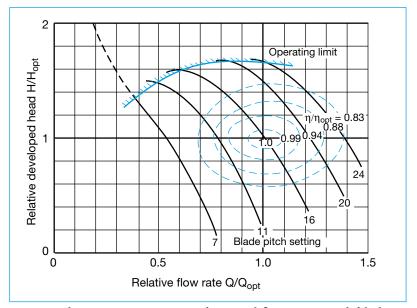
Fig. 32: Characteristic curve set of a centrifugal pump with pre-swirl control equipment,  $n_q \approx 160$ 

shown in the product literature with their respective characteristic curves (see Fig. 33).

## 3.4.10 Flow Control Using a Bypass

The system characteristic curve can be made steeper by closing a throttle valve, but it can also be made flatter by opening a bypass in the discharge piping as shown in Fig. 34. The pump operating point moves from B1 to a larger flow rate B<sub>2</sub>. The bypass flow rate is controlled and can be fed back into the inlet tank without being used directly. From the point of view of saving energy, this type of control only makes sense when the power curve falls for increasing pump flow rates ( $P_1 >$ P<sub>2</sub>), which is the case for high specific speeds (mixed and axial flow propeller pumps).

For these types of pumps, controlling the flow by pre-swirl control or by changing the blade pitch is even more economical, however. The expenditure for a bypass and control valve is not small [4]. This method is also suitable for preventing pumps from operating at unacceptably low flow rates (see operating limits in Figs. 5 and 6c as well as in Figs. 32 and 33).



*Fig. 33: Characteristic curve set of an axial flow pump with blade pitch adjustment,*  $n_q \approx 200$ 

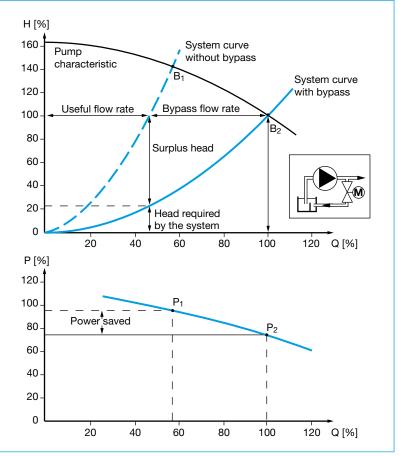


Fig. 34: Characteristic curves and operating points of a pump with a falling power curve and flow control using a bypass. (For a radial flow pump the power curve would increase towards the right and this type of control would cause an increase in power input, see Fig. 5).

## 3.5 Suction and Inlet Conditions [3]

NPSH = Net Positive Suction Head

## 3.5.1 The NPSH Value of the System: NPSH<sub>a</sub>

The NPSH<sub>a</sub> value is the difference between the total pressure in the centre of the pump inlet and the vapour pressure  $p_v$ , expressed as a head difference in m. It is in certain respects a measure of the probability of vaporization at that location and it is determined only by the operating data of the system and the type of fluid. The vapour pressure of water and other liquids are shown in Table 12 and in Fig. 35 as a function of the temperature.

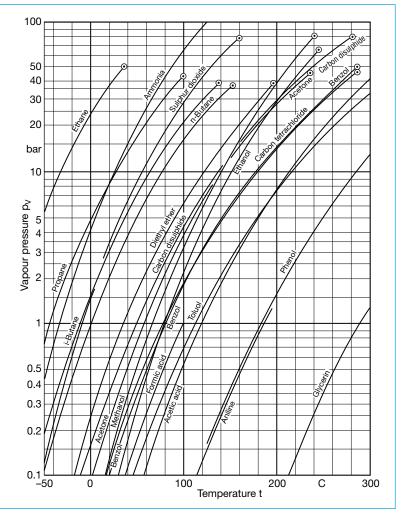


Fig. 35: Vapour pressure  $p_v$  of various fluids as a function of the temperature t (for an enlarged view see page 84)

Table 12: Vapour pressure  $p_v$ , density  $\varrho$  and kinematic viscosity v of water at saturation conditions as a function of the temperature t

: °C	p <sub>v</sub> bar	Q kg/m <sup>3</sup>	$\frac{\nu}{\mathrm{mm}^{2}/\mathrm{s}}$	t °C	p <sub>v</sub> bar	0 kg/m <sup>3</sup>	$\frac{\nu}{\text{mm}^2/\text{s}}$	t °C	p <sub>v</sub> bar	Q kg/m <sup>3</sup>	$\frac{\nu}{\text{mm}^2/\text{s}}$
0	0.00611	999.8	1.792	61	0.2086	982.6		145	4.155	921.7	
1	0.00656	999.9		62	0.2184	982.1		150	4.760	916.9	
2	0.00705	999.9		63	0.2285	981.6					
3	0.00757	1000.0		64	0.2391	981.1		155	5.433	912.2	
4	0.00812	1000.0		65	0.2501	980.5		160	6.180	907.4	0.1890
5	0.00872	1000.0		66	0.2614	980.0					
6	0.00935	999.9		67	0.2733	979.4		165	7.008	902.4	
7	0.01001	999.9		68	0.2856	978.8		170	7.920	897.3	
8	0.01072	999.8		69	0.2983	978.3					
9	0.01146	999.7		70	0.3116	977.7	0.413	175	8.925	892.1	
10	0.01227	999.6	1.307					180	10.027	886.9	0.1697
				71	0.3253	977.1					
11	0.01311	999.5		72	0.3396	976.6		185	11.234	881.4	
12	0.01401	999.4		73	0.3543	976.0		190	12.553	876.0	
13	0.01496	999.3		74	0.3696	975.4					
14	0.01597	999.2		75	0.3855	974.8		195	13.989	870.3	
15	0.01703	999.0		76	0.4019	974.3		200	15.550	864.7	0.1579
16	0.01816	998.8		77	0.4189	973.7		200	15.550	001.7	0.1572
17	0.01936	998.7		78	0.4365	973.0		205	17.245	858.7	
18	0.02062	998.5		79	0.4547	972.5		210	19.080	852.8	
19	0.02196	998.4		80	0.4736	971.8	0.365	210	17.000	052.0	
20	0.02337	998.2	1.004	00	0.4750	271.0	0.505	215	21.062	846.6	
20	0.02337	<i>))0.2</i>	1.001	81	0.4931	971.3		213	23.202	840.3	0.1488
21	0.02485	997.9		82	0.5133	970.6		220	25.202	040.5	0.1400
22	0.02485	997.7		82	0.5342	969.9		225	25.504	834.0	
22 23	0.02808	997.5		83 84	0.5557	969.9 969.4		223	27.979	827.3	
23 24	0.02982	997.2		85	0.5780	968.7		230	21.979	827.5	
24 25	0.02982	997.0		86	0.6010	968.1		235	30.635	820.6	
23 26	0.03360	996.7		87	0.6249	967.4		233	33.480	820.8	0.1420
20 27	0.03564	996.4		88	0.6495	966.7		240	33.400	015.0	0.1420
27 28	0.03384	996.4 996.1		88 89	0.6493	966.7 966.0		245	36.524	806.5	
28 29	0.04004	995.8					0.226				
29 30		995.8 995.6	0.801	90	0.7011	965.3	0.326	250	39.776	799.2	
30	0.04241	995.6	0.801	01	0 72 01	0647		255	42.247	701.0	
31	0.04491	995.2		91 02	0.7281	964.7		255	43.247	791.8	0 1 2 2 0
		993.2 994.9		92	0.7561	964.0		260	46.944	784.0	0.1339
32	0.04753			93	0.7849	963.3		2.5			
33	0.05029	994.6		94	0.8146	962.6		265	50.877	775.9	
34 25	0.05318	994.2		95	0.8452	961.9		270	55.055	767.9	
35	0.05622	993.9		96	0.8769	961.2		275	50.407	750 4	
36	0.05940	993.5		97	0.9095	960.4		275	59.487	759.4	
37	0.06274	993.2		98	0.9430	959.8		280	64.194	750.7	0.1279
38	0.06624	992.9		99	0.9776	959.0					
39	0.06991	992.6	0.670	100	1.0132	958.3	0.295	285	69.176	741.6	
40	0.07375	992.2	0.658					290	74.452	732.3	
				102	1.0878	956.8					
41	0.07777	991.8		104	1.1668	955.5		295	80.022	722.7	
42	0.08198	991.4		106	1.2504	954.0		300	85.916	712.5	0.1249
43	0.08639	991.0		108	1.3390	952.6					
44	0.09100	990.6		110	1.4327	951.0		305	92.133	701.8	
45	0.09582	990.2						310	98.694	690.6	
46	0.10085	989.8		112	1.5316	949.6					
47	0.10612	989.3		114	1.6361	948.0		315	105.61	679.3	
48	0.11162	988.9		116	1.7465	946.4		320	112.90	667.1	0.1236
49	0.11736	988.5		118	1.8628	944.8					
50	0.12335	988.0	0.553	120	1.9854	943.1	0.2460	325	120.57	654.0	
00	0.12000	200.0	0.000					330	128.64	640.2	
51	0.12960	987.7		122	2.1144	941.5					
51	0.12960	987.2		124	2.2503	939.8		340	146.08	609.4	0.1245
52 53	0.13813	987.2 986.7		126	2.3932	938.2		350	165.37	572.4	
	0.14293 0.15002	986.7 986.2		128	2.5434	936.5		550	103.37	372 <b>.4</b>	
54				130	2.7011	934.8		360	186.74	524.4	0.1260
55	0.15741	985.7		130		20110					
56	0.16509	985.2		132	2.8668	933.2		370	210.53	448.4	
~~ <b>'</b>	0.17312	984.7		132	3.0410	933.2 931.4		274.2	225.60	2260	0 1 4 0 0
					5.0410	751.4		374.2	223.60	326.0	0.1490
57 58	0.18146	984.3			3 2224	9796					
		984.3 983.7 983.2	0.474	136 138	3.2224 3.4137	929.6 927.9			o of sea wa		

## 3.5.1.1 NPSH<sub>a</sub> for Suction Lift Operation

For suction lift operation (Fig. 8) the pump is installed above the suction-side water level. The value of NPSH<sub>a</sub> can be calculated from the conditions in the suction tank (index e) as follows (see Fig. 36)

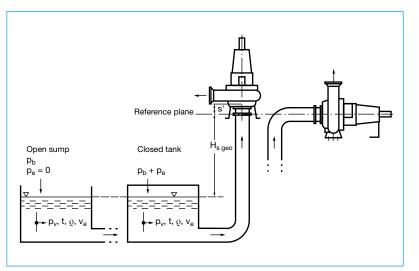


Fig. 36: Calculation of the NPSH<sub>a</sub> for suction lift operation for horizontally or vertically installed pumps

NPSH<sub>a</sub> =  $(p_e + p_b - p_v)/(\varrho \cdot g) + v_e^2/2g - H_{L,s} - H_{s \text{ geo}} \pm s'$  (29)

#### where

- pe Gauge pressure in suction tank in N/m<sup>2</sup>
- pb Absolute atmospheric pressure in N/m<sup>2</sup> (Table 13: consider effect of altitude!)
- $p_v$  Vapour pressure in N/m<sup>2</sup> (in Table 12 as absolute pressure!)
- Q Density in kg/m<sup>3</sup>
- g Gravitational constant, 9.81 m/s<sup>2</sup>
- ve Flow velocity in the suction tank or sump in m/s
- $H_{L,s}$  Head loss in the suction piping in m
- $H_{s \text{ geo}}$  Height difference between the fluid level in the suction tank or sump and the centre of the pump inlet in m
- s' Height difference between the centre of the pump inlet and the centre of the impeller inlet in m

Table 13: Influence of the altitude above mean sea level on the annual average atmospheric pressure and on the corresponding boiling point (1 mbar = 100 Pa)

Altitude above mean sea level	Atmospheric pressure p <sub>b</sub>	Boiling point
m	mbar	°C
0	1013	100
200	989	99
500	955	98
1000	899	97
2000	795	93
4000	616	87
6000	472	81

For cold water and open sump (Fig. 36, on the left) at sea level this equation can be simplified with sufficient accuracy for most practical purposes to

NPSH<sub>a</sub> =  $10 - H_{L,s} - H_{s \text{ geo}} \pm s'$  (30)

The correction using s' is only necessary when the centre of the impeller inlet (which is the decisive location for cavitation risk) is not at the same height as the centre of the pump inlet (= reference plane). In Fig. 36,  $H_{s \text{ geo}}$  must be "lengthened" for the pump on the left by the value s' (i.e., same sign for  $H_{s \text{ geo}}$ and s'!). When s' is unknown, it can usually be estimated with enough accuracy by examining the pump's outline drawing.

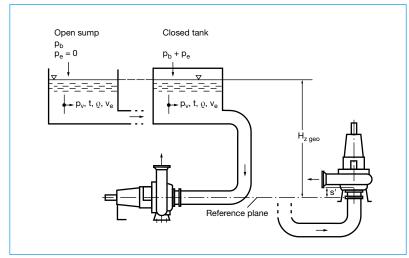


Fig. 37: Calculation of the NPSH<sub>a</sub> for suction head operation for horizontally or vertically installed pumps

#### 3.5.1.2 NPSH<sub>a</sub> for Suction Head Operation

For operation with positive inlet pressure (also called "suction

head operation"), the pump is installed below the liquid level. Eqs. (29) and (30) change by replacing  $-H_s$  geo with  $+H_z$  geo and then read:

NPSH<sub>a</sub> = 
$$(p_e + p_b - p_v)/(\varrho \cdot g) + v_e^2/2g - H_{L,s} + H_{z \text{ geo}} \pm s'$$
 (31)

where

H<sub>z geo</sub> Height difference between the fluid level in the inlet tank and the centre of the pump inlet in m

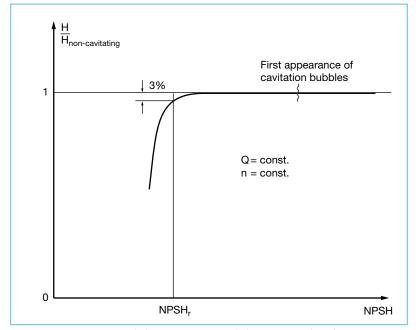


Fig. 38: Experimental determination of the NPSH<sub>r</sub> for the criterion  $\Delta H = 0.03 H_{non-cavitating}$ 

For cold water and open tanks (Fig. 37, on the left) at sea level this equation can also be simplified for all practical purposes to:

NPSH<sub>a</sub> = 
$$10 - H_{L,s} + H_{z \text{ geo}} \pm s$$
 (32)

The comments on s' as outlined in section 3.5.1.1 apply analogously.

#### 3.5.2 The NPSH Value of the Pump: NPSH<sub>r</sub>

When the inlet pressure drops, cavitation bubbles start to develop in the pump long before their effects become apparent in the hydraulic performance. One must therefore accept the presence of a small amount of cavitation bubbles in order to operate economically. The permissible amount of cavitation can be defined with certain criteria. Often a head drop of 3% resulting from cavitation is accepted. Fig. 38 shows how this point is identified: At a constant flow rate and constant speed of rotation, the NPSH<sub>a</sub> of the test loop is reduced until the pump's discharge head has fallen by 3%. Other criteria for the cavitation limit can also be used, such as the increase in sound level due to cavitation, the amount of material erosion or a certain reduction in pump efficiency. To avoid impermissible cavitation conditions, a minimum NPSH value is required, which is shown (in units of m) in the  $NPSH_r$  curves below the H/Q

characteristics (see Fig. 18). The reference plane is the centre of the impeller inlet (Fig. 39), which can vary by the height s' from the reference plane of the system, for example for vertical pumps (see Figs. 36 and 37).

So as to avoid exceeding the given cavitation limit, it is necessary that

 $NPSH_a > NPSH_r$  (33)

Fig. 40 shows this graphically at the intersection of the NPSH<sub>a</sub> and NPSH<sub>r</sub> curves. If the NPSH requirement is not fulfilled, the developed head will quickly decrease to the right of the intersection (i.e. at larger flow rates), which produces a "cavitation breakdown curve". Prolonged operation under these conditions can damage the pump.

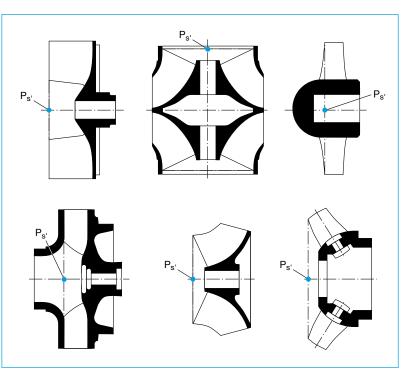


Fig. 39: Position of the reference point  $P_{s'}$  for various impellers

## 3.5.3 Corrective Measures

The numerical values of  $NPSH_a$  and  $NPSH_r$  are based on the fixed design geometry of the system and of the pump, which cannot be changed after the fact, and on the particular operating point. It follows that a subsequent improvement of the NPSH<sub>a</sub> > NPSH<sub>r</sub> condition in an installed centrifugal pump system is only possible with major design and financial expenditure for the pump or the system. Options include: increasing H<sub>z</sub> geo or reducing H<sub>s</sub> geo (by mounting the tank at

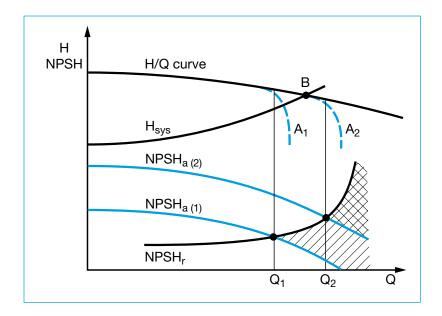


Fig. 40: "Cavitation breakdown curves"  $A_1$  and  $A_2$  of the H/Q curve in the case of insufficient NPSH<sub>a</sub>: An NPSH deficit exists in the singly hatched (case 1) and cross-hatched regions (case 2). After increasing NPSH<sub>a</sub>(1) to NPSH<sub>a</sub>(2), the pump's useful operating range is increased from Q<sub>1</sub> to Q<sub>2</sub> and the operating point B can now be reached.

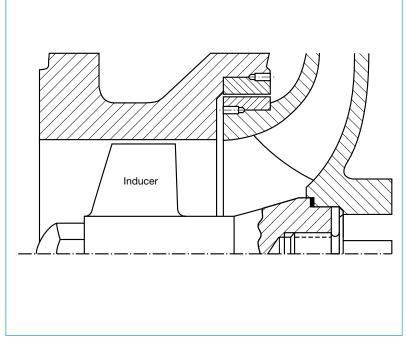
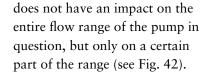


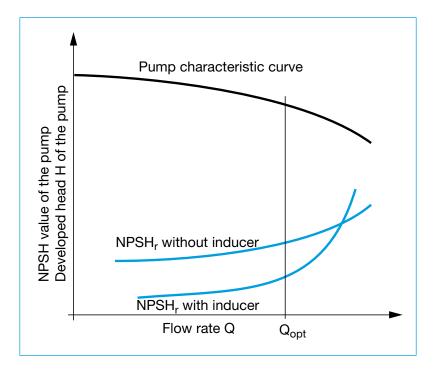
Fig. 41: Sectional drawing of a pump with an inducer (detail)

a higher level or installing the pump at a lower point), minimizing the pressure losses in the inlet piping  $H_{L,s}$  or replacing the pump. In the latter case, using a special low-NPSH suction-stage impeller or installing an inducer (propeller in front of the impeller, Fig. 41) can keep the costs of the improvement within limits (a rebuild of the pump is unavoidable, however). It must be kept in mind that the NPSH<sub>r</sub> reduction by the inducer



The resistance to cavitation erosion can be increased by choosing more suitable (and more expensive) materials for the impeller, in particular for larger pump sizes.

In one special case, the elimination of an NPSH problem is quite simple: For closed flow loops (for example in a heating system), the system pressure can simply be increased to improve the NPSH<sub>a</sub>, as long as the system is designed to cope with the higher pressure.



*Fig.* 42: *Effect of an inducer on the NPSH<sub>r</sub>* 

## 3.6 Effect of Entrained Solids

If the water to be pumped (for example, domestic waste water, rainwater or mixtures) contains small amounts of entrained solids, special impeller and pump types are used (for example with cleaning covers or special shaft seals) [1].

Fig. 43 shows the most common impeller designs for these types of waste water. For pumping sludge, non-clogging channel impellers can be used up to 3% solids content, single vane impellers up to 5%, free flow impellers up to 7% and worm type impellers for even higher concentrations. Since single vane impellers cannot be turned down to adjust the operating point (see section 3.4.6), this type of pump is often driven using a belt drive (see Fig. 59g).

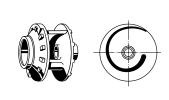
Allowances added to the drive power are not shown in Fig. 20, but rather in the product literature [1], since they depend not only on the drive rating but also on the impeller design and specific speed. For example, for **single vane impellers** pumping domestic waste water or sewage the following **power reserves** are recommended:

up to	7.5 kW	approx. 30%
		$(\geq 1 kW)$
from	11–22 kW	approx. 20%
from	30-55 kW	approx. 15%
above	55 kW	approx. 10%

When assessing the head losses in the piping (see section 3.2.1.2), adequate allowances have to be added [1].

To avoid blockages in the pipes for waste water with high solids concentrations, a minimum flow velocity of 1.2 m/s in horizontal pipes and 2 m/s in vertical pumps should be maintained. (Exact values can only be determined experimentally!). This is of particular importance for variable speed drives [1].

## **Impeller Types for Pumping Waste Water**



Front view shown without shroud

Fig. 43a: Closed single vane impeller for waste water containing solid or stringy substances

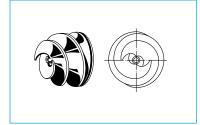


Fig. 43d: Worm type impeller for waste water containing coarse, solid or stringy substances or for sludge with up to 5 to 8% solids content

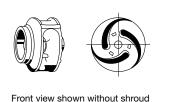


Fig. 43b: Closed non-clogging

channel impeller for sludge or non-gassing liquids containing solids without stringy components

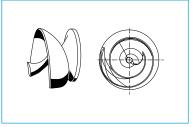


Fig. 43e: Diagonal impeller for waste water containing solid, stringy or coarse substances

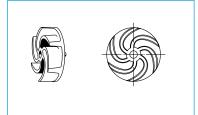


Fig. 43c: Free flow impeller for fluids with coarse or stringy solids and gas content

## 4 Special Issues when Pumping Viscous Fluids 4.1

## The Shear Curve

Viscosity is that property of a fluid by virtue of which it offers resistance to shear. Fig. 44 shows this process. In a fluid, a plate with a wetted surface area A is moved with speed  $v_0$  parallel to a stationary wall at a distance  $y_0$ .

The movement requires that the resistance force F be overcome, which can be expressed as a shear stress  $\tau = F/A$ . If the wall distance y<sub>0</sub>, the velocity v<sub>0</sub> or the type of fluid is changed, then the shear stress also changes in proportion to the velocity v<sub>0</sub> or inversely proportional to the distance y<sub>0</sub>. The two easily identified parameters v<sub>0</sub> and y<sub>0</sub> are combined to yield the shear gradient v<sub>0</sub>/y<sub>0</sub>.

Since the viscosity of the fluid causes a shear stress  $\tau$  not only at the walls, but rather at every distance from the wall within the fluid, the definition of the rate of shear is generalized as  $\partial v/\partial y$  (change of velocity per change of distance). Just as for the shear stress  $\tau$ , it is not the same for all wall distances y. During an experiment, pairs of values  $\tau$  and  $\partial v/\partial y$  are measured and can be plotted as a function, the so-called shear curve (Fig. 45).

When the shear curve is a straight line going through the origin:

$$\tau = \eta \cdot \partial v / \partial y \tag{34}$$

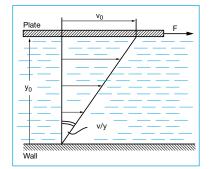


Fig. 44: Velocity profile between a plane wall and a moving parallel plate.

- F = Towing force
- $v_0 = Towing speed$
- $y_0 = Distance to wall$
- $\partial v / \partial y = Rate of shear$

the constant factor of proportionality  $\eta$  is referred to as the dynamic viscosity with the units Pa s. Fluids with this type of curve (for example water or all mineral oils) are normally viscous or Newtonian fluids, for which the laws of hydrodynamics apply without restriction. If the shear curve is not a straight line through the origin, the fluid is a non-Newtonian fluid, for which the laws of hydrodynamics apply only in a limited fashion. One must therefore strictly differentiate between these two cases.

Since the quotient of dynamic viscosity  $\eta$  and density  $\varrho$  is often used in fluid dynamic relation-

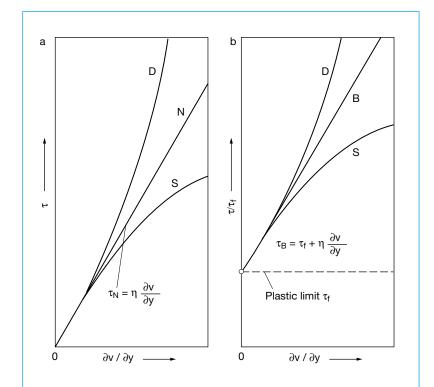


Fig. 45: Overview of the shear behaviour of viscous fluids a without, b with a plastic shear limit  $\tau_f$ N Newtonian, B Bingham, S pseudo-plastic, D dilatant fluids

ships it is defined as the kinematic viscosity

 $\nu = \eta/\varrho \tag{35}$ 

where

- $\nu$  Kinematic viscosity in m<sup>2</sup>/s
- η Dynamic viscosity in Pa s
  (= kg/sm)
- Q Density in kg/m<sup>3</sup> (for numerical values see Fig. 48)

For water at 20°C,  $v = 1.00 \cdot 10^{-6} \text{ m}^2/\text{s}$ . For further numerical values see Table 12. The units centistokes = mm<sup>2</sup>/s, degrees Engler °E, Saybolt seconds S" (USA) and Redwood seconds R" (UK) are no longer used and can be converted to m<sup>2</sup>/s using Fig. 46.

Independently of the discussion above, viscosity varies with

Fig. 46: Conversion between various units of kinematic viscosity v

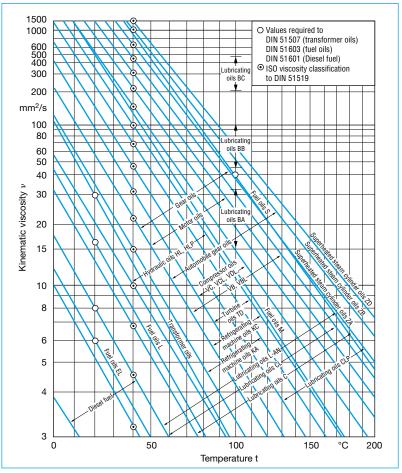
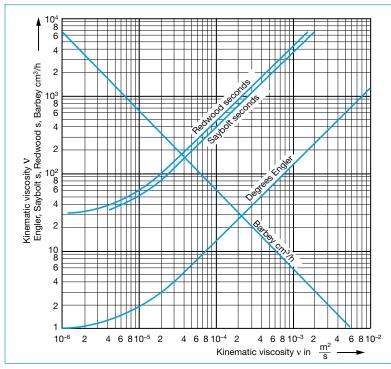
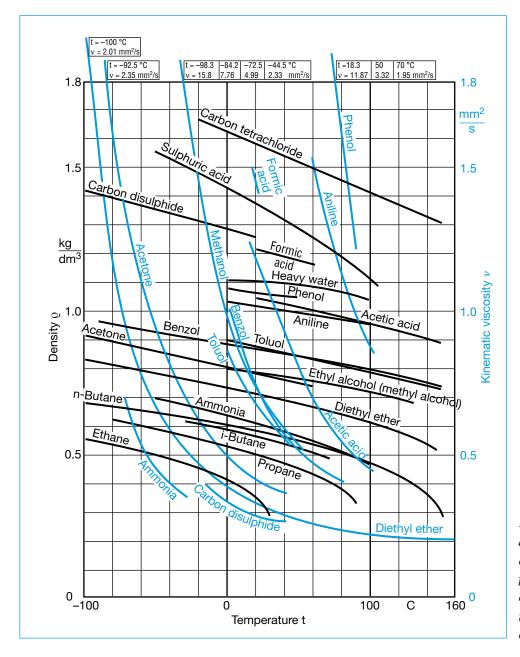


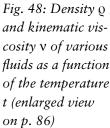
Fig. 47: Kinematic viscosity v of various mineral oils as a function of the temperature (enlarged view on page 85)



temperature: at higher temperatures almost all liquids become "thinner"; their viscosity decreases (Figs. 47 and 48).

The dynamic viscosity  $\eta$  can be measured for all liquids using a rotating viscometer to determine the shear curve. A cylinder rotates with a freely chosen speed in a cylindrical container filled with the liquid in question. The required driving torque is measured at various speeds along with the peripheral speed, the size of the wetted area and the distance of the cylinder from the wall.





## 4.2 Newtonian Fluids

## 4.2.1 Influence on the Pump Characteristics

The characteristic curves of a centrifugal pump (H,  $\eta$  and P as functions of Q) only start to change perceptibly at viscosities above  $\nu > 20 \cdot 10^{-6} \text{ m}^2/\text{s}$  and only need to be corrected with empirical conversion factors above this limit. The **two** 

most well known methods are that described in the Hydraulic Institute (HI) Standards and that of KSB. Both methods use diagrams containing the conversion factors which are applied in a similar manner, but differ in that the KSB method not only includes the parameters Q, H and  $\nu$  but also the significant influence of the specific speed  $n_q$  (see section 3.1.5). The HI method (Fig. 49) is based on measurements with  $n_q = 15$  to 20 and gives the same numerical results as the KSB method (Fig. 50) in this narrow range. The KSB method is based on measurements with  $n_q$  from 6.5 to 45 and viscosity up to  $v_z = 4000 \cdot 10^{-6} \text{ m}^2/\text{s}$ . The use of both diagrams is explained with the examples shown in them [9].

The flow rate Q, the total developed head H and the ef-

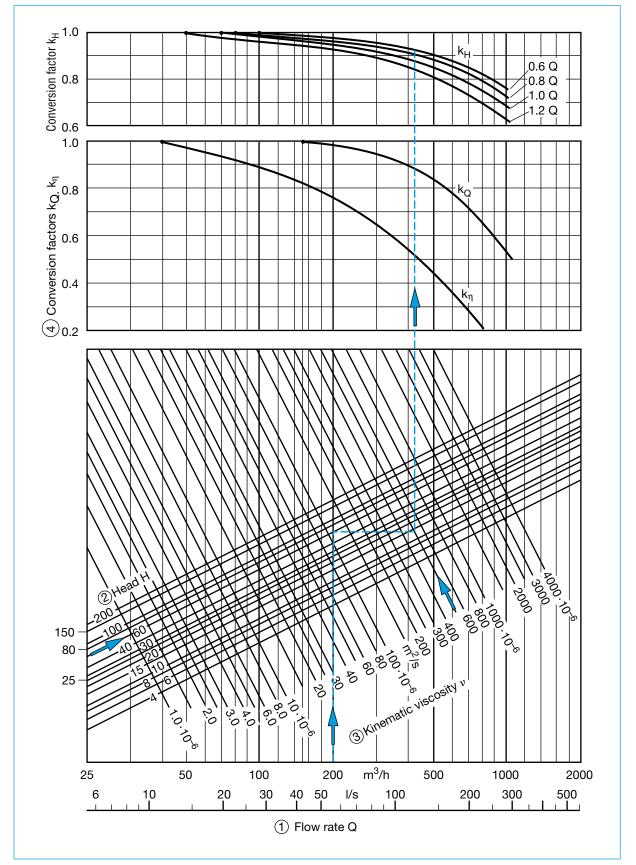


Fig. 49: Determination of the conversion factors k using the Hydraulic Institute method. Example shown for  $Q = 200 \text{ m}^3/\text{h}$ , H = 57.5 m,  $v = 500 \cdot 10^{-6} \text{ m}^2/\text{s}$ 

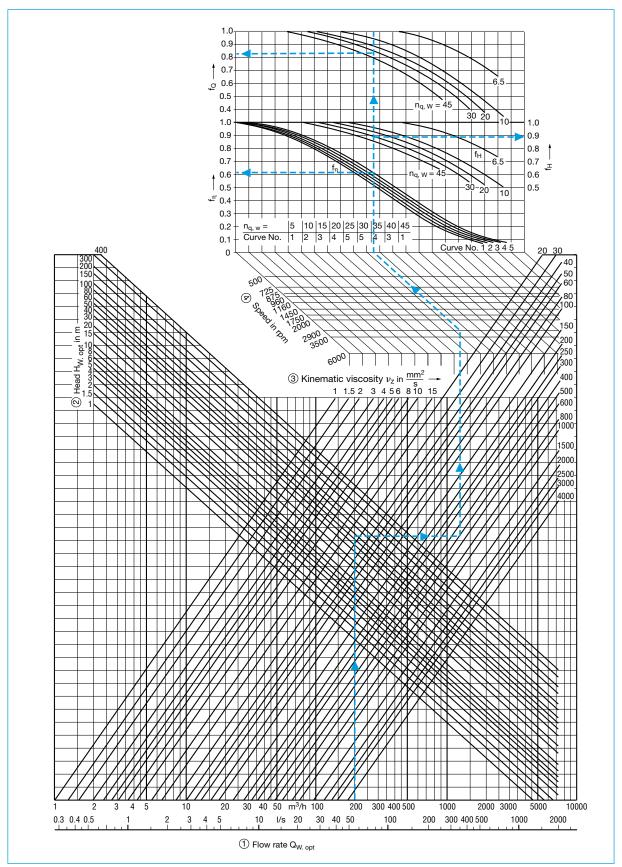


Fig. 50: Determination of the conversion factors f using the KSB method. Example shown for  $Q = 200 \text{ m}^3/\text{h}$ , H = 57.5 m, n = 1450 rpm,  $v = 500 \cdot 10^{-6} \text{ m}^2/\text{s}$ , n = 2900 rpm,  $n_q = 32.8$ 

ficiency  $\eta$ , which are known for a single-stage centrifugal pump operating with water (subscript w), can be converted to the values for operation with a viscous medium (subscript z) as follows:

$Q_z = f_Q \cdot Q_w$	(36)
$H_z = f_H \cdot H_w$	(37)
$\eta_z = f_\eta \cdot \eta_w$	(38)

The factors f are designated with k in the HI method; both are shown graphically in Figs. 49 and 50. In Fig. 50 the speed of rotation n of the pump must be considered in the diagram and the specific speed  $n_q$  of the pump impeller must be known, for example from Fig. 3 or Eq. 3.

With these factors the known performance for water can be converted to reflect operation with a viscous fluid. The conversion is valid for the range

$$0.8 Q_{opt} < Q < 1.2 Q_{opt}$$
 (39)

A simple calculation can thus be done for three flow rates, with a single exception:

At Q = 0.8 Q<sub>opt</sub>, H<sub>z</sub> =  $1.03 \cdot f_H \cdot H_w$  applies (but H<sub>z</sub> never is > H<sub>w</sub>!).

At flow rate Q = 0, simply set  $H_z = H_w$  and  $\eta_z = \eta_w = 0$ . A worksheet or spreadsheet calculation as shown in Fig. 51 can simplify the conversion.

After the power is calculated at the three flow rates (in the flow range according to Eq. 39) using

#### How to Find the Operating Point

Given:			
Flow rate	Qw		m³/h
Total developed head	H <sub>w</sub>		m
Speed of rotation	n		1/min
Kinematic viscosity	ν <sub>z</sub>		m²/s
Density	Qz		kg/m <sup>3</sup>
Gravitational constant	g	9.81	m/s <sup>2</sup>

#### Calculation

earea	anon						
Q/Q <sub>opt</sub>	=	0	0.8	1.0	1.2	-	
Q <sub>w</sub> fi	rom the	0				m³/h	
н !!	oump cha- acteristic					m	
	or 4 points	0				-	
n <sub>q, w</sub> fr 3	rom section .1.5	-	-		-	1/min	
f <sub>Q, w</sub> fr	rom Fig. 50	I				-	
f <sub>H, w</sub>		-				-	
f <sub>η, w</sub>		-				-	
$Q_z = Q_z$	w ∙ f <sub>Q, w</sub>	0				m³/h	
$H_z =$		$= H_w$	$=$ $H_{W} \cdot f_{H, W} \cdot 1,03$	<u>H</u> <sub>w</sub> ⋅ f <sub>H, w</sub>	H <sub>w</sub> .f <sub>H,w</sub>		
		ŧ	1)	+	+	m	Ļ
$\eta_z = \eta_w$	$f_{\eta, w}$	0					
$\overline{P_{z} \!=\! \frac{\varrho_{z} \cdot g \cdot H_{z} \cdot \varrho_{z}}{\eta_{z} \cdot 1000 \cdot 3600}}$		$\mathbf{X}$				kW	

 $^{1)}$  If  $H_{Z}$  becomes larger than  $H_{W},$  it should be set to  $H_{W}.$ 

 $^{2)}$  These are four values on the Hz/Q and Q/ $\eta_z$  curve and three points on the Q/Pz curve. Plot versus Q.

Fig. 51: Spreadsheet for calculating the pump characteristics for a viscous fluid using the KSB method (enlarged view on p. 87)

$$P_z = \varrho_z \cdot g \cdot H_z \cdot Q_z / 1000 \eta_z$$
(40)

where

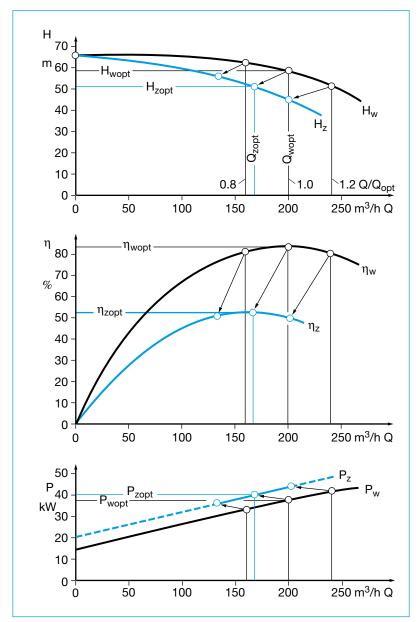
 $\varrho_z$  Density in kg/m<sup>3</sup>

#### $Q_z$ Flow rate in m<sup>3</sup>/s

- g Gravitational constant 9.81 m/s<sup>2</sup>
- $H_z$  Total developed head in m
- $\eta_z\;\; Efficiency \; between 0 \; und 1$
- $P_z$  Power in kW (!)

all the characteristic curves can be plotted over  $Q_z$  using the 3 or 4 calculated points, as shown in Fig. 52 on page 54.

For the inverse problem, i.e. when the operating point for the viscous fluid is known and the values for water are to be found (for example when choosing a suitable pump for the requested operating point), the water values are estimated and the solu-



*Fig. 52: Conversion of the pump characteristics for water to that for a viscous fluid* 

tion is approached iteratively using  $f_Q$ ,  $f_H$  and  $f_\eta$  in two (or sometimes three) steps.

For specific speeds above  $n_q \approx 20$ the more realistic KSB method results in smaller power requirements; below this limit the calculated required driving power according to HI is too small [9]!

#### 4.2.2 Influence on the System Characteristics

Since the laws of fluid dynamics retain their validity for all Newtonian fluids, the equations and diagrams for calculating the pipe friction factor and the loss coefficients for valves and fittings are also applicable to viscous media. One must simply substitute the kinematic viscosity of the viscous liquid  $v_z$  for the water viscosity  $v_w$  when calculating the Reynolds number Re =  $v \cdot d/\nu$ . This yields a lower Reynolds number, and a larger friction factor  $\lambda_z$  results from Fig. 10. (Note: the influence of the wall roughness can now often be ignored because of the larger boundary layer thickness in the flow.) All of the pressure losses in the pipes, valves and fittings calculated for water in accordance with section 3.2.1.2 are to be increased by the ratio  $\lambda_z/\lambda_w$ .

Fig. 53 is also suitable for general practical use: the diagram provides a fast way of determining the pipe friction factor  $\lambda_z$  as a function of the flow rate Q, pipe inside diameter d and kinematic viscosity  $v_z$ . It must be kept in mind, however, that the coefficient  $\lambda_w$  for water in this diagram is only valid for hydraulically smooth pipes (i.e. not for rough-surfaced pipes)! The corresponding  $\lambda_w$  can be used to calculate the ratio  $\lambda_z/\lambda_w$ .

Since the static component of the system characteristic curve  $H_{sys}$  (Fig. 16) is not affected by viscosity, the "dynamic" component of the system characteristic for water can be redrawn as a steeper parabola for a viscous fluid.

## 4.3 Non-Newtonian Fluids 4.3.1 Influence on the Pump Characteristics

Since the local velocity gradients in all the hydraulic components of a pump are not known, a cal-

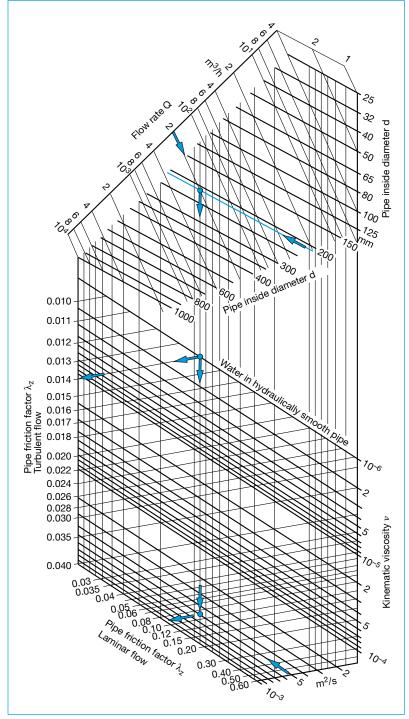


Fig. 53: Finding the pipe friction factor  $\lambda_z$  for viscous liquids Example:  $Q = 200 \text{ m}^3/h$ ; d = 210 mm;  $v_z = 5 \cdot 10^{-4} \text{ m}^2/\text{s}$ 

culation of the influence of non-Newtonian fluids on the pump characteristics is not generally possible. Only for a limited number of special fluids, such as fibre pulp, is a prediction based on knowledge gained during years of experience with this fluid feasible. The selection of a suitable pump must therefore be done by the design department.

#### 4.3.2 Influence on the System Characteristics

When the shear curves are not straight lines of constant linear viscosity, one must divide them into sections and determine the coefficient (= stiffness number) and the exponent n (= structural number) for each section individually (easiest when plotted on double-logarithmic scales). Using a special diagram (analogous to Fig. 10), which shows the pipe friction factors  $\lambda_z$  as a function of the generalized Reynolds number Ren for various exponents n, the value of  $\lambda_z$  can be read and the system curve H<sub>sys</sub> determined for a particular flow rate Q. Since this process is very laborious, in particular because of the need for multiple iterations, it cannot be recommended for general use.

Just as for the pump characteristics, in most cases diagrams with a narrow range of application based on experience with a particular fluid are used to find the head loss  $H_L$ . The more the application differs from the particular conditions of the diagram, the more uncertain will the head loss analysis become, so that in such cases the experience of the design department must be tapped.

# 5

## 5 Special Issues when Pumping Gas-laden Fluids

Unlike dissolved gases, a nondissolved gas in a liquid (expressed as a volume percentage) can change the design parameters, the characteristic curve and the general performance of a centrifugal pump dramatically, as shown in Fig. 54 using a nonclogging impeller pump as an example. The gas content may be caused by the production process itself but also by leaking flanges or valve stems in the suction line or from air-entraining vortices in an open sump inlet when the water level is too low (see section 7.2).

In the centrifugal force field of an impeller, the gas bubbles tend to accumulate in certain locations and to disturb the flow there. This effect is reinforced

- the more the pump operates at a reduced flow rate, since the lower velocities exert less carrying force on the gas,
- the smaller the impeller inlet diameter is, since the throttling effect of the gas volume is increased,
- the lower the specific speed n<sub>q</sub> of the pump impeller is, and
- the lower the speed of rotation of the pump is.

These effects cannot be calculated. When significant gas

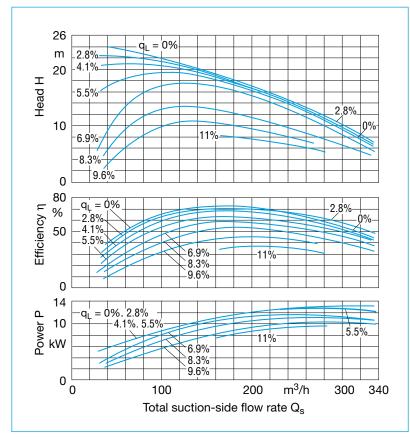


Fig. 54: Influence of non-dissolved air on the operation of a nonclogging impeller pump when pumping pre-treated sewage (open three-channel impeller, D = 250 mm, n = 1450 rpm,  $n_q = 37$ )  $q_{air} = Gas$  volume in suction piping as % of the mixture.

volumes are expected in the pumpage, the following measures can be useful:

- A sufficiently large settling tank in the suction line can allow the gas to separate out of the fluid and thus mitigate its disturbing effects.
- Pipes which are used to fill an open inlet sump must end below the liquid level so that there is no free fall of water that might entrain air bubbles in the tank. In addition, a baffle can prevent the entry of vortices in the suction piping (see Figs. 64 and 65).
- Low-flow operation of the main pump can be prevented by installing a special partload pump. When this pump is only needed occasionally, it is advantageous to use a self-priming pump (whose efficiency is lower, though).
- A gas removal line in front of the impeller hub requires a vacuum system, is only of limited use for large gas quantities and disturbs the normal operation of the pump.
- In the pump, open impellers (see Fig. 4) with few vanes are advantageous, as is the installation of an inducer (Fig. 41).
   Without any special precautions, non-clogging impeller pumps (Fig. 43) can pump up to 3%vol and free flow impellers up to 6 to 7%vol of gas.
- If a large gas content is to be expected under normal operating conditions, side channel pumps or water-ring pumps (positive displacement principle) operate more reliably.

## 6 Special Issues When Pumping Solids-laden Fluids 6.1

## **Settling Speed**

Solids (which are heavier than water) are most easiest to pump when their settling speed is lowest and their flow velocity highest. Because of the large number of influencing parameters, the settling speed can only be calculated based on simplifying assumptions: the settling speed of a single sphere in an unlimited space (subscript 0) results from force equilibrium as

$$w_{s0} = \sqrt{\frac{4 \text{ g } d_s}{3 \text{ c}_D} \cdot \frac{\varrho_s - \varrho_f}{\varrho_f}}$$
(41)

where

 $w_{s0}$  Settling speed in m/s

- g Gravitational constant 9.81 m/s
- d<sub>s</sub> Diameter of sphere in m
- $c_D \ \, \text{Resistance coefficient of the} \\ sphere \ \, \text{dependent on } Re_s$
- $\varrho_s$  Density of the solid in kg/m<sup>3</sup>
- $\varrho_f$  Density of the fluid in kg/m<sup>3</sup>

$$Re_s = w_{s0} \cdot d_s / \nu_f$$
 (42)

where

 $v_{\rm f}$  Kinematic viscosity of the liquid in Pa s

The settling speed  $w_{s0}$  is shown graphically in Fig. 55.

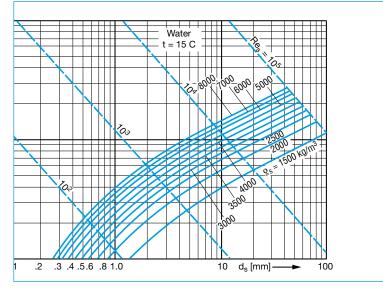


Fig. 55: Settling speed  $w_{s0}$  of individual spherical particles (spherical diameter  $d_s$ ) in still water

The solids concentration:

$$c_{\rm T} = Q_{\rm s} / (Q_{\rm s} + Q_{\rm f})$$
 (43)

has a large effect, where

- cT Flow-based solids concentration (transport concentration)
- $Q_s$  Flow rate of the solid in  $m^3/s$
- $Q_f$  Flow rate of the fluid in  $m^3/s$

The solids concentration and the boundary effect of the pipe walls reduce the settling speed considerably because of the mutual repulsion of the particles, approximately according to the following empirical relationship

$$w_s = w_{s0} \cdot (1 - c_T)^5$$
 (44)

The effect of an irregular particle shape cannot be estimated; the shape may differ substantially from that of a sphere.

The effect of the particle size distribution can also hardly be calculated. Fig. 56 shows an example of the distribution of

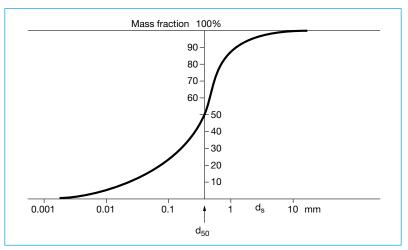


Fig. 56: Example of a particle size distribution

particle sizes d<sub>s</sub> plotted logarithmically for the portion which passed through a sieve of a given mesh size. Transported solids are almost always composed of particles of various sizes, so that the size distribution has a more or less distinct S-shape. To simplify the analysis, it can be assumed that the particle size for

50% mass fraction, designated  $d_{50}$ , is representative of the mixture. This assumption is the most important source of disparities in the planning phase.

After all these assumptions and gross approximations, no exact predictions of the effects of solids on the flow behaviour, the system curve, the total developed head and the efficiency of pumps are to be expected. The design and selection of pumps for solids transport must therefore be left to experts who have sufficient experience with similar cases. Even then, experiments are often necessary to attain a measure of certainty. Only certain general tendencies can be stated.

#### 6.2 Influence on the Pump Characteristics

The solids behave differently under the influence of the centrifugal force field in an impeller than the carrier fluid (usually water) does. The solids cross the streamlines or collide with and rub against the walls of the flow passages. They thus reduce the head H produced in the impeller by the difference  $\Delta$ H.

Experimental data exist on the effects of the particle diameter

 $d_s$ , the concentration  $c_T$  and the density  $\varrho_s$  of the solids as well as the specific speed  $n_q$ . The

empirical relationship for the relative head reduction  $\Delta$ H/H is approximately

$$\Delta H/H = c_T / \psi \cdot \sqrt[3]{Re_s} \cdot (11.83/n_q)^3 \cdot (\varrho_s/\varrho_f - 1)$$
(45)

## where

- $c_T \ \ \, Transport \ \, concentration \ \, according \ to \ \, Eq. \ \, 43$
- $\Psi$  Head coefficient of the pump; here approx. = 1
- Res Reynolds number of the solids flow according to Eq. 42

 $n_q$  Specific speed of the pump according to Eq. 3

- $Q_s$  Density of the solid in kg/m<sup>3</sup>
- $\rho_f$  Density of the fluid in kg/m<sup>3</sup>

When conveying solids hydraulically, the pump characteristic curve needs to be shown as developed pressure  $\Delta p$  versus flow rate Q, not as developed head H, since the average density  $\rho_m$ of the solids / water mixture (in contrast to pumping clean water) is not constant. As simplifications, the geodetic head difference  $z_{s,d}$  between the pump inlet and discharge as well as the velocity head difference  $(c_d^2 - c_s^2)/2$  g are ignored, i.e., the static head is set to equal the total head  $(H_p \approx H)$ :

## $\Delta p = \varrho_m \cdot g \cdot (H - \Delta H)$ (46)

where

- Qm Average density of the solids/ water mixture given by Eq. 47 in kg/m<sup>3</sup>
- g Gravitational constant 9.81 m/s<sup>2</sup>
- H Total developed head in m
- $\Delta$ H Head reduction according to Eq. 45 in m

 $\Delta p$  Pressure in N/m<sup>2</sup>

(to convert to bar: 1 bar = 100 000 N/m<sup>2</sup>)

The average density of the mixture is given by  $o_{\rm m} = c_{\rm T} \cdot o_{\rm s} + (1 - c_{\rm T}) \cdot o_{\rm w}$ 

(47)

where

- $\varrho_m$  Average density in kg/m<sup>3</sup>  $\varrho_s$  Density of the solid in
- kg/m<sup>3</sup>
- $\varrho_w$  Density of water in kg/m<sup>3</sup>
- cT Transport concentration according to Eq. 43

Since the pressure rise in the pump is the product of the density and the developed head (which is reduced when transporting solids), two independent influences are at work in Eq. 46: the increased average density due to the presence of the solids, and the reduced developed head  $(H - \Delta H)$ . Both changes are caused by the solids concentration, but they have opposite effects, since the density raises the pressure while the head deficit decreases it. Therefore, no general prediction can be made as to whether the pump pressure rise will be higher or lower than the water curve when the solids concentration increases. Heavy, small-grained solids (for example ores) are likely to

produce an increase, while light, large solids (for example coal) and low specific speeds tend to decrease the pressure.

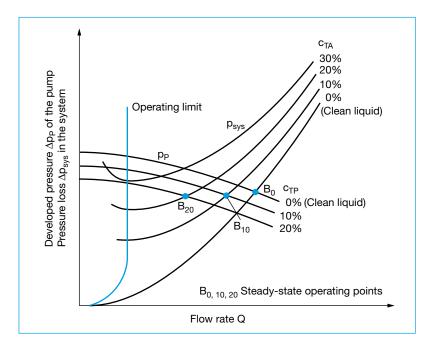
## 6.3 Influence on the System Characteristics

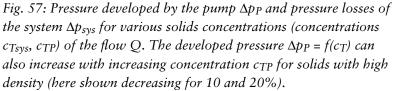
When the flow velocity drops, solids tend to settle to the bottom of horizontal pipe runs and collect on the pipe wall. The flow resistance increases and the free flow passage becomes smaller, so that despite the decreasing flow rate, the flow resistance can actually increase. This results in the unusual shape of the system curves as shown in Fig. 57. The minimum in the curves measured at various concentrations is a sure sign that a solids accumulation is taking place and that the pipes

will soon be clogged. The curve minimum is therefore generally considered to be the lower limit of operation. Exact predictions are only possible with sufficient experience or by experiment.

## 6.4 Operating Performance

Fig. 57 shows the typical behaviour of a centrifugal pump transporting solids through a horizontal pipe: with increasing concentration, the intersection of the pump and system characteristic curves shifts to ever lower flow rates, so that the lower limit of operation could be exceeded. To avoid this, a control system must intervene promptly. Throttle valves would be subject to high wear, however, so only a change of rotational speed remains as a feasible control





method for the hydraulic transport of solids. Speed control has an additional advantage: when the head developed by the pump impeller drops as the impeller wears, it is possible to compensate by merely increasing the speed.

In vertical pipes, the settling of the solids poses much greater risk, since the pipe can suddenly become plugged if the flow falls below the minimum required, even if only due to stopping the pump.

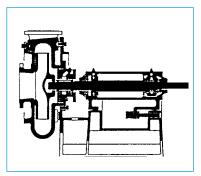
The high erosion rates when pumping granular solids are the decisive parameter for the design of the pumps used. An example of their typical robust design is shown in Fig. 58. The risk of erosion also limits the permissible operating range to near Q<sub>opt</sub>.

High solids concentrations put constraints on the use of centrifugal pumps; the limit values can only be found by experience.

The above considerations should have convinced the reader that the selection of pumps for hydraulic solids transport is risky without a solid base of experience and should be left to experts who do this frequently!

## 6.5 Stringy, Fibrous Solids

If long, stringy solids are present in the flow, problems can occur, in particular for axial flow (propeller) pumps, when these materials (plant fibres, plastic sheets and rags for example) are caught on the propeller blade leading edge and accumulate

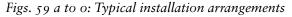


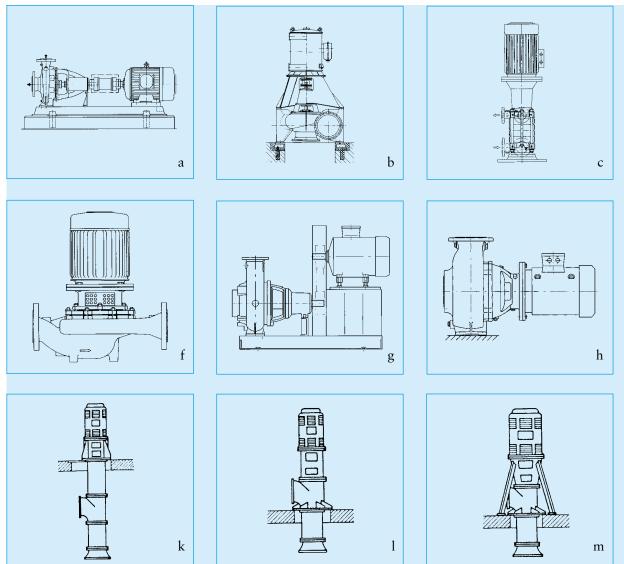
*Fig. 58: Typical centrifugal pump for the hydraulic transport of solids* 

there. The consequences are an increasing head loss and power input, which continue until the driving motor must be stopped due to overloading.

The problem can be solved by slanting the leading edges of the propeller blades backwards by shifting the individual profile planes during blade design, just as for a backswept airfoil. During operation, the fibres can slide along the blade leading edge until they are shredded in the clearance gap at the outside diameter of the propeller and flushed out. These self-cleaning blades are called "ECBs" (= ever clean blades) [5].

Untreated municipal sewage often contains textiles which tend to form braids and plug impellers with multiple vanes or other flow-dividing devices. Single vane impellers, worm type (screw) impellers, or free flow impellers (see Fig. 43) are the better choice for these applications.

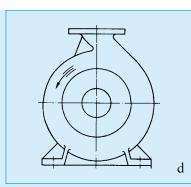


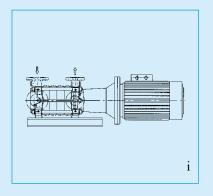


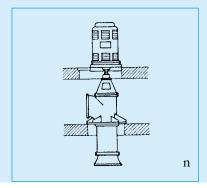
## 7 The Periphery 7.1 Pump Installation Arrangements

Pump installation arrangements are design features in which pumps of the same type (in general of the same series) may differ. Figures 59 a to o provide typical examples of the most frequent installation arrangements for horizontal and vertical centrifugal pumps [1].

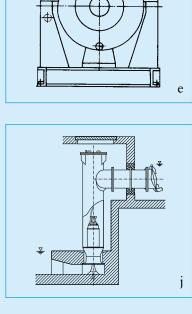
The major parameters classifying the pump installation arrangement are:

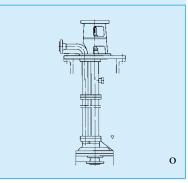






- the position of the shaft, i. e. horizontal or vertical (see Figs. a and b, also i and c or h and f),
- the arrangement of the feet, i. e. underneath or shaft centreline (see Figs. d and e),
- the mode of installation of the pump set, i. e. with or without foundation (see Figs. b and f),
- the arrangement of the drive, i. e. on its own or a common baseplate or flanged to the pump (see Figs. g, a, h and i),
- the weight distribution of the pump and drive,





- the arrangement of the discharge nozzle on tubular casing pumps (see Figs. k, l, m and n),
- the environment of the pump casing, i. e. dry or wet (see Figs. b and o).

## 7.2 Pump Intake Structures 7.2.1 Pump Sump

Pump sumps (or suction tanks) are designed to collect liquids and be intermittently drained if the mean inlet flow is smaller than the pump flow rate. The sump or tank size depends on the pump flow rate Q and the permissible frequency of starts Z of the electric motors, see section 3.3.3.1.

The useful volume  $V_N$  of the pump sump is calculated using:

$$V_N = Q_{in} \cdot \frac{Q_m - Q_{in}}{Q_m \cdot Z} \quad \textbf{(48)}$$

where

- Z Max. permissible frequency of starts per hour
- Q<sub>in</sub> Inlet flow in m<sup>3</sup>/h
- $Q_m = (Q_{on} + Q_{off}) / 2$
- Qon Flow rate at switch-on pressure in m<sup>3</sup>/h
- Qoff Flow rate at switch-off pressure in m<sup>3</sup>/h
- V<sub>N</sub> Useful volume of pump sump including potential backwash volume in m<sup>3</sup>

The maximum frequency of starts occurs when the flow rate  $Q_m$  is twice the incoming flow  $Q_{in}$ . The max. frequency of starts per hour is therefore:

 $Z_{max} = Q_m/4V_N$ 

With dirty liquids, solids must be prevented from being deposited and collecting in dead zones and on the floor. Walls arranged at a 45°, or better still 60° angle, help prevent this (Fig. 60).

## 7.2.2 Suction Piping

The suction pipe should be as short as possible and run with a gentle ascending slope towards the pump. Where necessary, eccentric suction piping as shown in Fig. 61 should be provided (with a sufficient straight length of pipe upstream of the pump  $L \ge d$ ) to prevent the formation of air pockets. If, on account of the site conditions, fitting an elbow immediately upstream of the pump cannot be avoided, an accelerating elbow (Fig. 62) helps to achieve a smooth flow. For the same reason, an elbow with multiple turning vanes (see Fig. 63) is required in front of double-entry pumps or pumps with mixed flow (or axial flow) impellers unless this is impossible because of the nature of the medium handled (no stringy, fibrous solids, see 6.5).

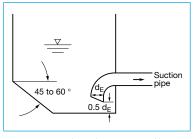


Fig. 60: Inclined sump walls to prevent deposits and accumulation of solids

The suction and inlet pipes in the suction tank or pump sump must be sufficiently wide apart to prevent air from being entrained in the suction pipe; positive deflectors (Figs. 64 and 65) should be provided, if required. The mouth of the inlet pipe must always lie below the liquid level, see Fig. 65.

If the suction pipe in the tank or pump sump is not submerged adequately because the liquid level is too low, rotation of the medium might cause an airentraining vortex (hollow vortex) to develop. Starting as a funnel-shaped depression at the surface, a tube-shaped air cavity forms within a short period of time, extending from the surface to the suction pipe. This will cause the pump to run very unsteadily and the output to decrease. The required minimum submergence (minimum depth

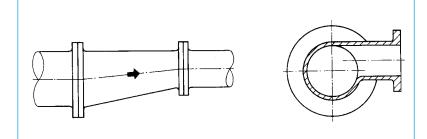


Fig. 61: Eccentric reducer and branch fitting to avoid air pockets

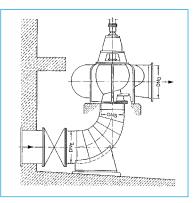


Fig. 62: Flow-accelerating elbow upstream of a vertical volute casing pump with high specific speed

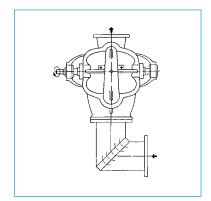


Fig. 63: Intake elbow with multiple turning vanes upstream of a double-entry, horizontal volute casing pump (plan view)

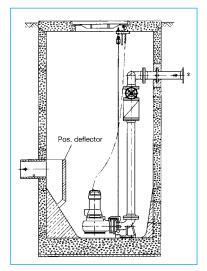
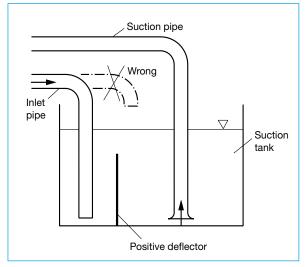
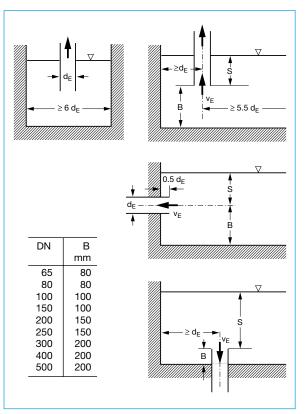


Fig. 64: Installation of a positive deflector in the intake chamber of a submersible motor pump



*Fig. 65: Piping arrangement in the suction tank / pump sump to prevent air entrainment* 

Fig. 66: Clearances between wall and suction pipe in the suction tank or pump sump according to relevant German regulations.  $S_{min}$ , as shown in Fig. 67. 2 suction pipes arranged side by side require a distance of  $\geq 6 d_E$ .



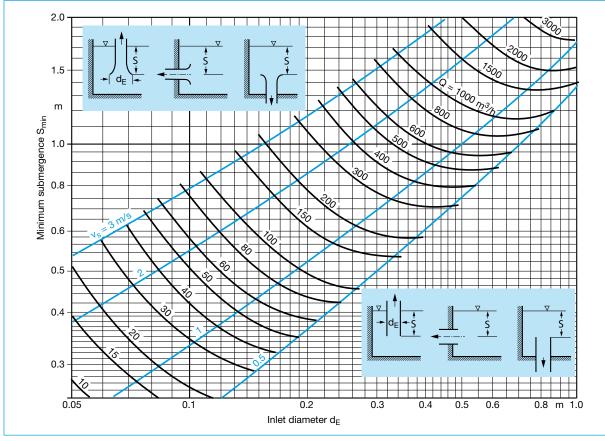


Fig. 67: Minimum submergence  $S_{min}$  of horizontal and vertical suction pipes (with and without entry nozzle) required for suction tanks to avoid hollow vortices (to Hydraulic Institute standards)

of immersion) is specified in Fig. 67, the minimum clearance between suction pipes and walls / sump floor in Fig. 66. (Special measures must be taken for tubular casing pumps, see 7.2.3).

The minimum submergence  $S_{min}$  can be read from Fig. 67 as a function of the intake diameter  $d_E$  (this is the pipe inside diameter of straight, flangeless pipes) or, where available, the inlet diameter of the entry nozzle and the flow rate Q. It can also be calculated according to the following equation given by the Hydraulic Institute:

$$S_{\min} = d_E + 2.3 \cdot v_s \cdot \sqrt{\frac{d_E}{g}}$$
 (50)

where

- S<sub>min</sub> Minimum submergence in m v<sub>s</sub> Flow velocity
- =  $Q/900 \pi d_E^2$  in m/s, recommended 1 to 2 m/s but never exceeding 3 m/s
- Q Flow rate in  $m^3/h$
- g Gravitational constant 9.81 m/s<sup>2</sup>
- d<sub>E</sub> Inlet diameter of suction pipe or entry nozzle in m

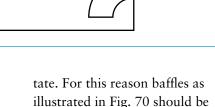


At flow velocities of 1 m/s, the minimum submergence levels specified by the relevant German regulations agree well with the data given above [13].

 $\nabla$ 

Wherever the required minimum submergence cannot or not always be ensured, measures as shown in Figs. 68 and 69 have to be taken to prevent air-entraining vortices. Irrespective of the aspects mentioned before, it should be checked whether the submergence levels also meet the NPSH<sub>a</sub> requirements laid down in 3.5.2.

Round tanks with tangential inlet pipes are special cases but used frequently. The liquid discharged via the inlet pipe causes the contents of the tank to ro-



provided.

Suction pipe

Raft

## 7.2.3 Intake Structures for Tubular Casing Pumps [1]

For tubular casing pumps, the minimum submergence and the design of the intake chamber are of particular importance because impellers with high specific speeds react very sensitively to uneven inlet flows and airentraining vortices.

Fig. 71 shows the arrangement of suction pipes in intake chambers of tubular casing pumps.

Refer to Fig. 72 for the minimum water level required for open, unlined intake chambers

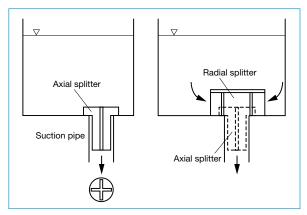
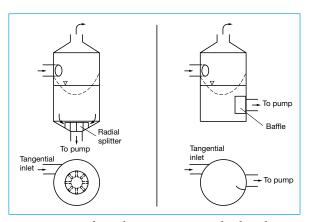


Fig. 69: Use of swirl preventers



*Fig.* 70: Use of swirl preventers in cylindrical tanks to ensure smooth flow to pump

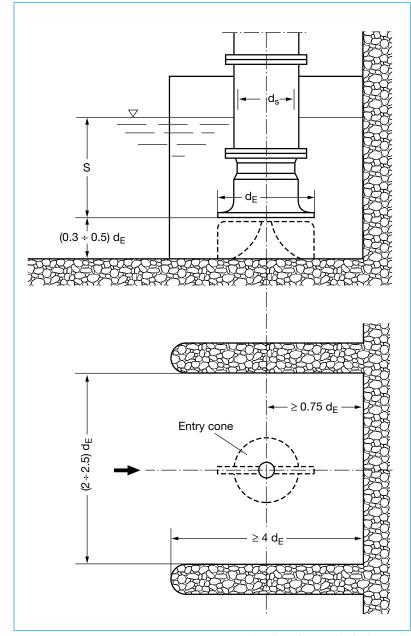


Fig. 71: Suction pipe arrangement in intake chambers of tubular casing pumps.  $S_{min}$  as shown in Fig. 72

 $d_E \approx (1.5 \div 1.65) \ ds$ 

2 suction pipes arranged side by side require a distance of > 3  $d_E$ 

with and without entry cones or calculate it using the following equation:

$$S_{\min} = 0.8 \ d_E + 1.38 \cdot v_s \cdot \sqrt{\frac{d_E}{g}}$$
(51)

where S<sub>min</sub> Minimum submergence in m

 $v_s$  Flow velocity

 $= Q / 900 \pi d_E^2$  in m/s

- Q Flow rate in  $m^3/h$
- g Gravitational constant 9.81 m/s<sup>2</sup>
- d<sub>E</sub> Inlet diameter of bellmouth in m

Lined or covered intake chambers or Kaplan intake elbows are more expensive, but allow pump operation at lower submergence levels [1].

Irrespective of the aspects mentioned before, it should be checked whether the submergence levels also meet the NPSH<sub>a</sub> requirements laid down in 3.5.2.

## 7.2.4 Priming Devices

Most centrifugal pumps are not self-priming; i. e. their suction pipes and suction-side casings must be deaerated prior to start-up unless the impeller is arranged below the liquid level. This often inconvenient procedure can be avoided by installing a foot valve (functioning as a non-return valve) at the suction pipe mouth (Fig. 73). Deaerating is then only necessary prior to commissioning and after a long period of standstill.

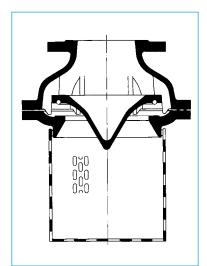
A closed suction tank (static tank) serves the same purpose, in particular when contaminated liquids are handled (it does, however, increase the flow losses and therefore reduces the  $NPSH_a$ ). The tank is under negative pressure and mounted upstream of the pump suction nozzle (Fig. 74). Prior to commissioning it must be filled with liquid. When the pump is started up, it empties the tank, and the air in the suction or siphon pipe is drawn into the suction tank across the apex until the liquid to be pumped follows. After the pump has been stopped, the tank is refilled with liquid via the discharge pipe either manually or automatically. The air stored in the tank escapes into the suction pipe.

The suction tank volume  $V_B$  depends on the suction pipe volume and the suction lift capacity of the pump:

$$V_{B} = d_{s}^{2} \frac{\pi}{4} \cdot L_{s} \cdot \frac{p_{b}}{p_{b} - QgH_{s}}$$
(52)

where

- $V_B$  Suction tank volume in m<sup>3</sup> d<sub>s</sub> Inside diameter of the air-
- filled inlet pipe in m
- L<sub>s</sub> Straight length of air-filled piping in m
- pb Atmospheric pressure in Pa (≈ 1 bar = 100 000 Pa)
- Q Density of the liquid handled in kg/m<sup>3</sup>
- g Gravitational constant 9.81 m/s<sup>2</sup>
- $H_s$  Suction lift of pump in m according to equation



*Fig. 73: Foot valve (cup valve) with suction strainer* 

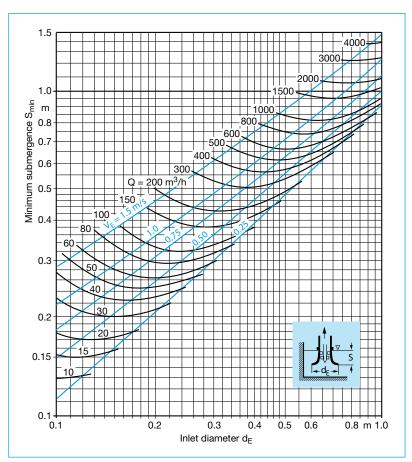


Fig. 72: Minimum submergence  $S_{min}$  of tubular casing pump suction pipe to avoid hollow vortices

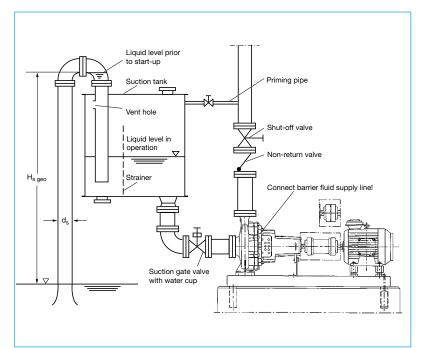


Fig. 74: Suction tank arrangement

$$H_s = H_{s \text{ geo}} + H_{L,s}$$
 (53)

where

- H<sub>s geo</sub> Vertical distance between water level and pump reference plane for suction lift operation in m, see Fig. 36
- $H_{L,s}$  Head loss in the suction piping in m (refer to 3.2.1.1).

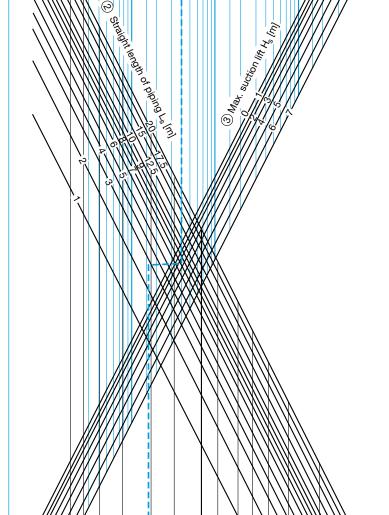
As  $H_{L,s}$  is in most cases notably smaller than  $H_{s \text{ geo}}$ , Eq. 53 can be neglected and  $H_s$  equated with  $H_{s \text{ geo}}$ . In this case, Fig. 75 provides a much faster way of finding the required tank size.

For safety reasons the suction tank volume should be multiplied by a factor of 2 to 2.5, or by a factor of up to 3 in the case of smaller pumping stations. The liquid pressure must never reach its specific vaporization pressure at any point in the system.

#### 7.3 Arrangement of Measurement Points

In order to achieve a certain accuracy in pressure and velocity measurement, the flow must be smooth and regular at the measuring points. Therefore, undisturbed straight lengths of piping need to be arranged upstream and downstream of the measurement point(s), as shown in Fig. 76 and indicated in Table 14. All pipe components which may impede a straight, parallel and non-swirling flow of liquid are considered a disturbance.

Relevant German regulations (VdS – Association of German



(4) Suction tank volume 0.03 0.05 0.1 0.2 0.3 0.5

30 50

100 200300 500 1000 I

1 1.5 2

3 5

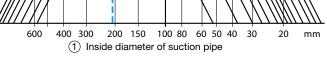


Fig. 75: Graph to determine the size of the suction tank. Follow the numbers from (1) to (4) for selection. A safety factor of 3.0 has already been considered in the above diagram (head losses  $H_{L,s}$  in the suction piping were neglected).

Table 14: Minimum values for undisturbed straight lengths of piping
at measurement points in multiples of the pipe diameter D

Source	Distan	ce from	Undistur	bed	
	pump flange		pipe length		
	A <sub>s</sub> /D	A <sub>d</sub> /D	U <sub>s</sub> /D	U <sub>d</sub> /D	
VdS 2092-S	0.5	1.0	2.5	2.5	In-service measurement
ISO 9906	2.0 2.0		$5 + n_q / 53$	-	Acceptance test measurement

60 m<sup>3</sup>

10 15 20 30

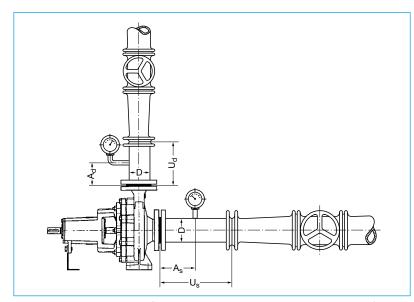


Fig. 76: Arrangement of pressure measurement points up- and downstream of the pump

Property Insurance Companies) stipulate pipe lengths in multiples of the pipe diameter for in-service measurements, while **ISO 9906** specifies pipe lengths for acceptance test measurements. The data from both sources are listed in Table 14.

If the required straight pipe lengths cannot be provided, the measuring results are likely to be less accurate. Consequently, pump flanges are not suitable as measurement points.

The pressure measuring points should consist of a 6 mm diameter hole and a weld socket to fit the pressure gauge. Even better still are annular measuring chambers with four drilled holes spread evenly across the circumference.

## 7.4 Shaft Couplings

In centrifugal pump engineering, both rigid and flexible shaft couplings are used. Rigid couplings are mainly used to connect shafts in perfect alignment, since the smallest degree of misalignment will cause considerable stress on the coupling and on the adjacent shaft sections.

Flexible couplings to DIN 740 are elastic, slip-free connecting elements between drive and pump which accommodate axial, radial and angular misalignment and damp shock loads. Flexibility is usually achieved by the deformation of damping and rubber-elastic spring elements whose life is governed to a large extent by the degree of misalignment. Fig. 77 shows two of the most common types of flexible shaft coupling. Fig. 78 shows a spacer coupling between a volute casing pump and drive. It permits removal of the pump rotating assembly without having to dismantle the suction and discharge piping or move the pump casing or drive (back pull-out design).

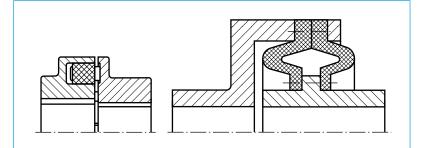


Fig. 77: Flexible (left) and highly flexible coupling

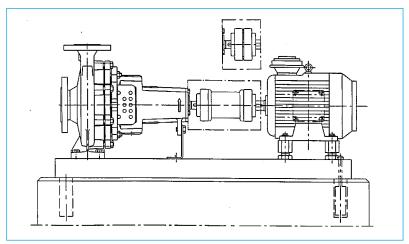


Fig. 78: Pump with spacer coupling compared with normal coupling

## 7.5 Pump Nozzle Loading

A centrifugal pump mounted on the foundation should not be used as an anchorage point for connecting the piping. Even if the piping is fitted to the nozzles without transmitting any stresses or strains, forces and moments, summarized as nozzle loading, will develop under actual operating conditions (pressure and temperature) and as a result of the weight of the liquid-filled piping. These cause stresses and deformation in the pump casings, and above all changes in coupling alignment, which, in turn, may affect the pump's running characteristics, the service life of the flexible elements in the shaft coupling, as well as the bearings and mechanical seals. For this reason limits have been defined for permissible nozzle loading [1].

As the loading profile for each pump nozzle is made up of three different forces and moments, it is not possible to specify theoretical nozzle loading limits for all conceivable combinations. Therefore, operators either need to check whether the nozzle loading imposed by the system is still within the pump's permissible limits, or have to contend with the considerably reduced general limits specified in several national and international standards and codes (EURO-PUMP brochure "Permissible flange forces and moments for centrifugal pumps", 1986; API 610; ISO 5199).

Fig. 79 shows the permissible nozzle loading for single-stage volute casing pumps to ISO 5199 (solid line for pumps on grouted baseplate, broken line for pumps on non-grouted baseplates).

## 7.6

## National and International Standards and Codes

A series of national standards and other technical codes have been introduced in Germany since the early sixties which govern the dimensions, manufacture, design, procurement and use of centrifugal pumps. Many of the requirements laid down have been included in European and international standards and codes. Drawn up by both operators and manufacturers, these are now wellestablished in virtually all sectors of industry using or producing pumps. The most important standards are tabulated in Fig. 80 on page 70.

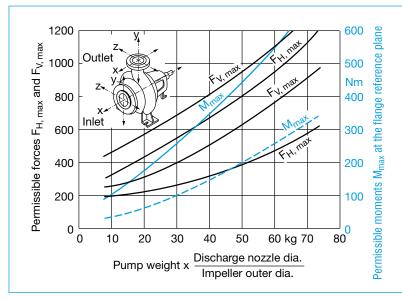


Fig. 79: Permissible moments  $M_{max}$  at the flange reference plane, as well as permissible forces  $F_{H,max}$  (at x,z plane) and  $F_{V,max}$  (in y direction) to ISO 1599 for single-stage volute casing pumps made of ferritic cast steel or nodular cast iron at room temperature. Lower numerical values apply to austenitic cast steel, lamellar graphite cast iron or higher temperatures.

				_		
			DIN EN 12056-4 Gravity drain-	age systems inside build- Bingse Part 4: Sewage Part 4: Sewage Part 4: and dimen- sioning sioning	tum Institute	API 610 Centrifugal Pumps for Petrochemical and Natural Gas Industries
		DIN EN 12262 Centrifugal pumps; Technical do- cumentation, terms, scope of supply, quality			American Petroleum Institute	API 682 Shaft Sealing Systems for Centrifugal and Rotary Pumps
			DIN EN 12050 Sewage lift- ing units for	the disposal of water from buildings and premises; Design and testing prin- ciples	4	4 0 0 0 ° L
		DIN 24420-1 Spare parts lists; General DIN 24420-2 Spare parts lists; Form and struc- ture of text field				
		DIN 1988-5 Technical spe- cifications for drinking wa- ter installa- tions, pres- sure boost- ing and redu- cing systems	DIN EN 806-1 and -2 Technical	specifica- tions for drinking water sys- tems	Γ	ISO 9908 Technical specifica- tions for pumps – Class III Class III
Codes and Specifications		DIN ISO 9905 (Class I) DIN ISO 5199 (Class II) DIN ISO 5199 (Class II) DIN ISO 9908 (Class III) DIN ISO 9908 Centrifugal pumps; Technical specifica- tions	SIION			ISO 51 99 Technical specifica- tions for centrifugal pumps – Class II
Codes and S	VDMA 24292 Liquid pumps; pumps; for pumps for pumps for pumps for pump for safety for safety instructions	DIN 24296 Pumps and pump units for liquids; Spare parts; Selection and procure- ment		EN 809 Pump and pump units for liquids; General safety re- quirements		ISO 9905 Technical specifica- tions for centrifugal pumps - Class I Class I
	VDMA 24 279 24 279 Dentrifugal nical require- ments, Mag- metic drive and cantor pumps	DIN 24273 Pumps and pump units for liquids; Materials and product tests		EN 1151 Circulators with input powers up to 200 W for heating sys- tems and service water heating sys- tems for domestic	use – Re- auirements.	testing, marking
	VDMA 24276 Liquid Liquid chemical plant – Duality spe- cifications for materials and compo- nents	DIN EN 12639 Liquid pumps and pump units - Noise measure- ment - Test (3sses 2 and 3		EN 12639 Liquid pumps and pump units; Noise measure- ment; Test clas- ses 2 and 3		
	VDMA 24261-1 Pumps; Designations based on function and design features; Centrifugal pumps	DIN EN 12723 Liquid pumps – Ge- neral terms for pumps and installa- tions; Defini- tions; vari- ables, sym- bols and	nuits	EN 12162 Liquid pumps – Safetyre- quirements – Procedure for hydro- static test- ing		
		DIN EN 24250 Centrifugal pumps; Nomen- clature and component numbers				ISO 5198 Centrifugal mixed flow and axial pumps – Code for hydraulic performance tests; Precision grade
		DIN EN ISO 9906 Rotodynamic pumps – Hydraulic performance acceptance test – Grades 1 and 2				ISO 9906 Rotodynamic pumps – Hydraulic performance acceptance tests – and 2 and 2
		DIN EN 12756 Mechanical seals				ISO 3069 End suction centrifugal pumps – pumps – for mechani- cal seals and for soft packing
ssories		DIN 24299-1 Pump name- plates; General specifica- tions	23661 23661	End suction centrifugal pumps; pumps; and installa- tion dimen- sions		ISO 3661 End suction centrifugal pumps – pamps – and installa- tion dimen- sions
Dimensional Standards: Pumps and Accessories		DIN 24259-1 Pumps; Baseplates for machin- ery; Dimen- sions	22858 22858	End suction centrifugal ing 16 bar); Designation, rated per- formance, main dimen- sions		ISO 2858 End suction centrifugal pumps (rat- ing 16 bar) – Designation, rated per- formance and dimen- sions
nal Standards:		DIM EN 735	DIN EN / 35 Overall	dimensions of centritu- gal pumps; Tolerances		
Dimensio	VDMA 24253 Centrifugal pumps with lined assing (lined pumps); single entry; single entry; single entry; single entry; single entry; single entry; that axial init; fated powers, dimensions	thiftugal age pumps ed of Dink EN 724	DIN EN 734 Side channel	pumps PN 40; Rated per- formance, main dimen- sions, designation		
	VDMA 24252 Centrifugal Dumps with wear plates, (wash water pumps), with bearing Designatio, rated powers, dimensions	DIN 24251 Multistage centritugal Multistage pumps with heads up to 1000 at a rated speed of 1500 rpm	UIN EN / 33 End suction	centrifugal pumps (rat- pumps (rat- with baring PN 10); with baraing bracket; Rated per- formance, main dimen- sions, system system		
Scope of Application and Responsibilities	VDMA German Engineering Federation Pump Committee	DIN German Standards Institute Mechanical Engineering Standards Committee, Pumps		CEN Comité Euro- péen de Nor- malisation European Standaards Coordination Committee TC 197 Pumps		[ISO] Inter- national Organiza- tion for Standardi- Techn. Techn. Techn. Pumps Pumps
cope of / nd Respc	vic of Germany	Federal Reput		Europe	ימנוסווק	Morldwide
ars					enoiter	motol

Fig. 80: National and international standards and codes for centrifugal pumps (last update: 2005)

## 8. Calculation Examples

The consecutive numbers of the calculation examples in this chapter are identical to the numbers of the respective equations

in the text. For example, the application dealt with in exercise 8.3 refers to Equation (3).

## 8.1 Pressure Differential

Given: A volute casing pump Etanorm 80-200 with characteristic curves as per Fig. 18, speed of rotation n = 2900 rpm, impeller diameter  $D_2 = 219$  mm, operating point at the point of best efficiency:  $Q = 200 \text{ m}^3/\text{h}$ , H = 57.5 m,  $\eta$  = 83.5%, water temperature t =  $20 \,^{\circ}$ C, density  $\rho = 998.2 \text{ kg/m}^3$ . Nominal nozzle diameters  $DN_d = 80$ ;  $DN_s = 100$ ; inside nozzle diameter  $d_d = 80 \text{ mm}, d_s = 100 \text{ mm}$ [1]. Height difference between suction and discharge nozzles  $z_{s,d} = 250 \text{ mm}$ , Fig. 8.

## 8.2 Input Power

Given: The data as per exercise 8.1. Sought: The input power P.

## 8.3 Specific Speed

Given: The data as per 8.1; the specific speed  $n_q$  is calculated using Eq. (3)

Sought: The pressure differen-
tial between the discharge and
suction sides indicated by the
pressure gauges.

(Taking  $z_{s,d} = 250$  mm into account presupposes that the pressure gauges are fitted exactly at the respective nozzle levels to keep the same difference in height. If they are mounted at the same level,  $z_{s,d}$  must be set to zero. Refer to paragraph 7.3 and ISO DIS 9906 for the correct location of the pressure measurement taps).

## Flow velocities

 $v_d = 4 Q / \pi d_d^2 = 4 \cdot (200/3600) / \pi 0.08^2 = 11.1 m/s$  $v_s = 4 Q / \pi d_s^2 = 4 \cdot (200/3600) / \pi 0.10^2 = 7.08 m/s.$ 

According to Eq. (1) the pressure differential is:  $\Delta p = \varrho \cdot g \cdot [H - z_{s,d} - (v_d^2 - v_s^2) / 2g]$   $= 998.2 \cdot 9.81 \cdot [57.5 - 0.250 - (11.1^2 - 7.08^2)/(2 \cdot 9.81)]$  = 524 576 Pa = 5.25 bar

se	According to Eq. (2) the input power is:
	$\mathbf{P} = \mathbf{\varrho} \cdot \mathbf{g} \cdot \mathbf{Q} \cdot \mathbf{H} / \mathbf{\eta}$
	$= 998.2 \cdot 9.81 \cdot (200/3600) \cdot 57.5/0.835$
	= 37 462 W = 37.5 kW

$$\begin{split} n_{q} &= n \cdot \sqrt{Q_{opt}} / H_{opt}^{3/4} = 2900 \cdot \sqrt{(200/3600)} / 57.5^{3/4} \\ &= 2900 \cdot 0.236 / 20.88 = 32.8 \text{ rpm} \\ \text{or} \\ &= 333 \cdot (n/60) \cdot \sqrt{Q_{opt}} / (gH_{opt})^{3/4} \\ &= 333 \cdot 48.33 \cdot \sqrt{(200/3600)} / 9.81 \cdot 57.5^{3/4} \\ &= 333 \cdot 48.33 \cdot 0.236 / 115.7 = 32.8 \text{ (dimensionless)} \end{split}$$

#### 8.5 Bernoulli's Equation

Given: A centrifugal pump system as shown in Fig. 8 with tanks B and D, designed for a flow rate of Q = 200 m<sup>3</sup>/h for pumping water at 20 °C. The discharge-side tank is under a pressure of 4.2 bar (positive pressure), the suction tank is open to atmosphere,  $v_e \approx 0$ . The geodetic difference in height is 11.0 m; the welded discharge piping has a nominal diameter of DN 200 (d = 210.1 mm acc. to Table 4). The system head loss is 3.48 m.

#### 8.9 Head Loss in Pipes

**Given:** The data as per 8.1 and: suction pipe DN 200, d = 200.1 mm according to Table 4, length = 6.00 m, average absolute roughness k = 0.050 mm. Sought: The system head H<sub>sys</sub>. Eq. (5) gives:  $H_{svs} = H_{geo} + (p_a - p_e)/(o \cdot g) + (v_a^2 - v_e^2)/2g + \Sigma H_L$ where Density  $\rho = 998.2 \text{ kg/m}^3$  according to Table 12 Pressure in tank B:  $p_a = 4.2 \text{ bar} = 420\ 000 \text{ Pa}$ Pressure in tank D:  $p_e = 0$  $(p_a - p_e)/(q \cdot g) = 420\ 000/(998.2 \cdot 9.81) =$ 42.89 m  $v_a = 4 \text{ Q} / (3600 \cdot \pi \cdot d^2) = 4 \cdot 200/(3600 \cdot \pi \cdot 0.2101^2)$ = 1.60 m/s $(v_a^2 - v_e^2)/2g = (1.60^2 - 0)/(2 \cdot 9.81) =$ 0.13 m 11.00 m  $H_{geo} =$  $\Sigma H_L =$ 3.48 m  $H_{sys} =$ 57.50 m

**Sought:** The head loss H<sub>L</sub> according to Fig. 11 or Eq. (9).

The diagram in Fig. 11 gives:  $H_L = 1.00 \cdot 6.00/100 = 0.060$  m The calculation according to Fig. 10 would be more complex and involved, but also **absolutely necessary for other roughness values**. Relative roughness d / k = 210.1 / 0.050 = 4202 According to Eq. (11), the Reynolds number is Re = v · d / v where  $v = 1.00 \cdot 10^{-6}$  m<sup>2</sup>/s,

$$v = Q/A = (Q/3600) \cdot 4/(\pi d^2) = (200/3600) \cdot 4/(\pi \cdot 0.2101^2)$$
  
= 1.60 m/s,

Re = v · d /  $\nu$  = 1.60 · 0.2101 / 10<sup>-6</sup> = 3.37 · 10<sup>5</sup>.

From Fig. 10, d / k =  $4202 \rightarrow \lambda = 0.016$ .

Eq. **(9)** gives

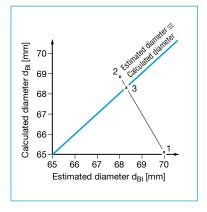
 $H_{L} = \lambda (L / d) \cdot v^{2} / 2g$ = 0.016 \cdot (6.00 / 0.2101) \cdot 1.60^{2} / 2 \cdot 9.81 = 0.060 m

#### 8.15 Head Loss in Valves and Fittings

#### 8.20 Orifice Plate

Given:

The pump described in exercise 8.1 is provided with a welded discharge pipe DN 80, the inside diameter being d = 83.1 mm. The discharge head is to be constantly throttled by  $\Delta H = 5.00$  m.



Given	

Given:	
The suction pipe described in 8.9, including	
a slide disc valve DN 200,	
a 90° elbow with smooth surface and $R = 5 d$ ,	
a foot valve DN 200	
and a reducer DN 200 / DN 100 according to Table 8,	
type IV with an opening angle of $\alpha = 30^{\circ}$ .	
<b>Sought:</b> The head losses H <sub>L</sub> .	
According to Table 5, the loss coefficient of the slide disc	:
valve is	$\zeta = 0.20$
Acc. to Table 6, the loss coefficient of the 90° elbow is	$\zeta = 0.10$
Acc. to Table 5, the approx. loss coefficient of	
the foot valve is	$\zeta = 2.0$
Acc. to Table 5, the loss coefficient of the reducer is	$\zeta = 0.21$
The total of all loss coefficients is	$\Sigma \zeta = 2.51$
Eq. (15) then gives the following head loss:	
$H_L = \Sigma \zeta \cdot v^2 / 2 g = 2.51 \cdot 1.60^2 / (2 \cdot 9.81) = 0.328 m$	

Sought: The inside diameter of the orifice plate d<sub>Bl</sub>.

#### Eq. (20) gives

 $d_{Bl} = f \cdot \sqrt{Q / \sqrt{(g \cdot \Delta H)}}$  with f according to Fig. 25.

As an iterative calculation is necessary,  $d_{Bl}$  is estimated in the first instance, and this value is compared with the calculated diameter. If the two values differ, the value selected for the second estimate lies between the initially estimated and calculated diameters.

The following is calculated first of all:

$$\sqrt{Q/\sqrt{g \cdot \Delta H}} = \sqrt{200/\sqrt{9.81 \cdot 5.0}} = 5.34 \text{ m}$$

First estimate  $d_{Bl} = 70 \text{ mm}$ ;  $(d_{Bl} / d)^2 = 0.709$ ; f = 12.2; Result:  $d_{Bl} = 12.2 \cdot 5.34 = 65.1 \text{ mm}$ 

Second estimate  $d_{Bl} = 68 \text{ mm}; (d_{Bl} / d)^2 = 0.670; \text{ f} = 12.9;$ Result:  $d_{Bl} = 12.9 \cdot 5.34 = 68.9 \text{ mm}$ 

Third estimate  $d_{Bl} = 68.4$ ;  $(d_{Bl} / d)^2 = 0.679$ ; f = 12.8; Result:  $d_{Bl} = 12.8 \cdot 5.34 = 68.4$  mm

For a faster solution, it is recommended to plot the calculated versus the corresponding estimated diameters in a diagram so that the third estimate already provides the final result in the intersection of connecting line and diagonal, see adjacent diagram.

#### 8.21 Change of Speed

#### Given:

The pump speed as per 8.1 (operating data with index  $_1$ ) is to be reduced from  $n_1 = 2900$  rpm to  $n_2 = 1450$  rpm.

**Sought:** The data for flow rate Q<sub>2</sub>, discharge head H<sub>2</sub> and driving power P2 after change of speed.

Eq. (21) gives

 $Q_2 = Q_1 \cdot (n_2/n_1) = 200 \ (1450 / 2900) = 100 \text{ m}^3/\text{h}$ Eq. (22) gives  $H_2 = H_1 \cdot (n_2/n_1)^2 = 57.5 \cdot (1450 / 2900)^2 = 14.4 \text{ m}$ Eq. (23) gives  $P_2 = P_1 \cdot (n_2/n_1)^3 = 37.5 \cdot (1450 / 2900)^3 = 4.69 \text{ kW},$ 

on the assumption that the efficiency is the same for both speeds.

Sought: The reduced diameter D<sub>r</sub> and the discharge head H<sub>r</sub> at BEP

#### 8.27 Turning Down Impellers

#### Given:

The flow rate of the pump at BEP described in 8.1, i.e.  $Q_t = 200 \text{ m}^3/\text{h}$ , is to be reduced to  $Q_r = 135 \text{ m}^3/\text{h}$  by turning down the original impeller diameter  $D_t = 219 \text{ mm}$ .

#### 8.29 NPSH<sub>a</sub> for Suction Lift Operation

#### Given:

The centrifugal pump system described in exercise 8.5 plus the following data: place of installation 500 m above M.S.L.;  $H_{L,s}$  (refer to exercises 8.9 and 8.15) = 0.39 m;  $H_{s \text{ geo}}$  = 3.00 m;  $v_e \approx 0$ . The pump described in 8.1 is installed horizontally with an open suction tank, as shown in Fig. 36. According to Fig. 18, the pump's NPSH<sub>r</sub> is 5.50 m at a flow rate of Q = 200 m<sup>3</sup>/h. Eq. (27) gives  $D_r \approx D_t \cdot \sqrt{(Q_r / Q_t)} = 219 \cdot \sqrt{(135 / 200)} = 180 \text{ mm}$ Eq. (26) gives  $H_r \approx H_t \cdot (Q_r / Q_t) = 57.5 \cdot 135 / 200 = 38.8 \text{ m}$ 

after turning down the impeller ( $H_t = 57.5 \text{ m}$ ).

Question: Is NPSH<sub>a</sub> sufficient?

According to Eq. (29), NPSH<sub>a</sub> =  $(p_e + p_b - p_v)/(\varrho \cdot g) + v_e^2/2g - H_{L,s} - H_{s geo} \pm s'$ where Gauge pressure in suction tank  $p_e = 0$ Atmospheric pressure  $p_b = 955$  mbar = 95 500 Pa acc. to Table 13 Vapour pressure p<sub>v</sub> = 0.02337 bar = 2337 Pa acc. to Table 12 Density o = 998.2 kg/m<sup>3</sup> acc. to Table 12  $(p_e + p_b - p_v)/(\rho \cdot g) = (0 + 95\ 500 - 2337)/(998.2 \cdot 9.81) = 9.51\ m$  $v_e^2/2g$ = 0 = 0.39 m H<sub>L,s</sub> = 3.00 mH<sub>s geo</sub> s' = 0, as the centre of the impeller inlet is at the same height as the centre of the pump inlet. **NPSH**<sub>a</sub> = 6.12 m

With an NPSH<sub>r</sub> of 5.50 m,

NPSH<sub>a</sub> is larger than NPSH<sub>r</sub> in this case and therefore sufficient.

#### 8.31 NPSH<sub>a</sub> for Suction Head Operation

Given: The pump system described in exercise 8.29 is be operated in suction head operation with a closed tank as shown in Fig. 37. The system data are as follows: place of installation 500 m above M. S. L.;  $H_{L,s}$  (refer to exercises 8.9 and 8.15) = 0.39 m;  $H_{z \text{ geo}} = 2.00$  m;  $ve \approx 0$ . The pump described in 8.1 is installed horizontally with a closed suction tank, as shown in Fig. 37. According to Fig. 18, the pump's NPSH<sub>r</sub> is 5.50 m at a flow rate of Q = 200 m<sup>3</sup>/h.

#### Question: Is NPSH<sub>a</sub> sufficient?

<b>\</b>			
According to Eq. (31)			
$NPSH_a = (p_e + p_b - p_v) / $	$(q \cdot g) + v_e^2 / 2g$	$g - H_{L,s} + H_z$	$geo \pm s'$
where			
Gauge pressure in			
suction tank pe	= -0.40 bar	$= -40\ 000\ $	Pa
Atmospheric pressure ph	, = 955 mbar	= 95 500 Pa	a acc. to Table 13
Vapour pressure p <sub>v</sub>	= 0.02337 ba	r = 2337 Pa	acc. to Table 12
Density Q	= 998.2 kg/m	3	acc. to Table 12
$(p_e + p_b - p_v) / (o \cdot g)$			
$= (-40\ 000 + 95\ 500 - 2)$	2337) / (998.2 •	9.81)	= 5.43 m
ve <sup>2</sup> /2g			= 0
H <sub>L,s</sub>			= 0.39 m
H <sub>z geo</sub>			= 2.00 m
s' = 0, as the centre of the	ne impeller inlet	t is at	
the same height as t	he centre of the	e pump inlet.	
		NPS	$H_a = 7.04 \text{ m}$
With an NPSH <sub>r</sub> of $5.50$	m,		

### NPSH<sub>a</sub> is larger than NPSH<sub>r</sub> in this case and therefore sufficient.

#### 8.36 Pump Characteristics When Pumping Viscous Fluids

Given: A mineral oil with a density of  $\varrho_z = 0.897 \text{ kg/m}^3$ and a kinematic viscosity of  $\nu_z = 500 \cdot 10-6 \text{ m}^2/\text{s}$  is to be pumped by the centrifugal pump described in 8.1; characteristic curves according to Fig. 19.

**Sought:** The characteristics for discharge head, efficiency and input power when pumping this viscous fluid, using the spread-sheet calculation as per Fig. 51.

The data for handling water (index w) are required first to find the conversion factors:

Elemente et DED	$\cap$	200 - 3/1
Flow rate at BEP	Qw, op	$t = 200 \text{ m}^3/\text{h}$
Head at BEP	H <sub>w, op</sub>	t = 57.5 m
Optimum efficiency	$\eta_{w, opt}$	= 0.835
Power	P <sub>w, opt</sub>	= 37.5 kW
Speed	n	$= 2900 \text{ min}^{-1}$
Specific speed (as per exercise 8.3)	nq	= 32.8
Kinematic viscosity	$\nu_z$	$= 500 \cdot 10^{-6} \text{ m}^2/\text{s}$
Density of mineral oil	Qz	$= 897 \text{ kg/m}^3$
	( )	00 ( 0 ( 2

The three conversion factors  $f_Q = 0.84$ ,  $f_H = 0.88$ ,  $f_\eta = 0.62$  are taken from Fig. 51.

The calcul	lation	is	continued	using	the	table	below:

Q/Q <sub>opt</sub>	0	0.8	1.0	1.2	
Qw]	0	160	200	240	m³/h
$H_w$ Fig. 19	66.5	62.0	57.5	51.0	m
$ \left. \begin{array}{c} Q_w \\ H_w \\ \eta_w \end{array} \right\} \begin{array}{c} \text{refer to} \\ \text{Fig. 18} \end{array} $	0	0.81	0.835	0.805	
$ \begin{array}{c} Q_z = Q_w \cdot f_Q \\ H_z \end{array} $	0	134.4	168	201.6	m³/h
Hz	$= H_w$	= $1.03 \text{ H}_{\text{w}} \cdot f_{\text{H}}$	$= H_w \cdot f_H$	$= H_{w} \cdot f_{H}$	
	66.5	56.2	50.6	44.9	m
$\eta_z = \eta_w \cdot f_\eta$	0	0.502	0.518	0.499	
$P_z = \varrho_z \cdot Hz \cdot Q_z$	/ $(\eta_z \cdot 36)$	57)			
	÷	36.8	40.1	44.3	kW

To calculate the power  $P_z$ , the values for flow rate  $Q_z$  in m<sup>3</sup>/h and the density  $\varrho_z$  in kg/m<sup>3</sup> are inserted into the equation.

These calculated points can be used to plot the characteristic curve for a viscous fluid, cf. Fig. 52 and Fig. 18 (this chart is applicable for handling water with an impeller diameter of 219 mm).

#### 8.45 Head Reduction for Hydrotransport

**Given:** Grit with a density of  $Qz = 2700 \text{ kg/m}^3$  and an average particle size of  $d_s = 5 \text{ mm}$  is to be pumped in cold water (kinematic viscosity  $v_f = 1.00 \cdot 10^{-6} \text{ m}^2/\text{s}$ ) at a concentration of  $c_T = 15\%$  with a centrifugal pump (hydraulic data as per 8.1, specific speed  $n_q = 33$ , head coefficient  $\Psi = 1.0$ ).

#### 8.47 Average Density

**Given:** Hydrotransport as described in exercise 8.45.

**Sought:** The average density  $Q_m$  and its effect on the pump discharge pressure; will it rise or fall?

#### 8.48 Pump Sump

Given: The pump sump for a pump as per 8.1 with the following data:

Inlet flow  $Q_{in} = 120 \text{ m}^3/\text{h}$ 

Flow rate at switch-on pressure  $Q_{on} = 220 \text{ m}^3/\text{h}$  and

Flow rate at switch-off pressure  $Q_{off} = 150 \text{ m}^3/\text{h}$ 

The maximum permissible number of start-ups of a pump unit is given in Table 10 (section 3.3.3.1, dry motor with P > 30 kW, in this case Z = 10/h). **Sought:** The head reduction  $\Delta$ H/H at H = 57.5 m.

According to Fig. 55, the settling speed  $w_{s0}$  of a single sphere under the conditions described above is 0.5 m/s. Thus, the Reynolds number is  $\text{Re}_s = w_{s0} \cdot \text{d}_s / \nu_f = 0.5 \cdot 0.005 / 1.0 \cdot 10 - 6 = 2500$ .

The head reduction is calculated using Eq. (45):

$$\begin{split} \Delta H/H &= c_T / \psi \cdot \sqrt[3]{\text{Re}_s} \cdot (11.83/n_q)^3 \cdot (\varrho_s/\varrho_f - 1) \\ &= (0.15 / 1.0) \cdot \sqrt[3]{2500} \cdot (11.83 / 33)^3 \cdot (2700 / 1000 - 1) \\ &= 0.15 \cdot 13.6 \cdot 0.0461 \cdot 1.70 = 0.16 \end{split}$$

 $\Delta H = 0.16 \cdot 57.5 = 9.2 \text{ m}$ 

Under the above conditions the pump discharge head of  $H_{w, opt} = 57.7$  m would be reduced by 16%, i.e. 57.5 - 9.2 = 48.3 m.

According to Eq. (47) the average density is  $\varrho_m = c_T \cdot \varrho_s + (1 - c_T) \cdot \varrho_f$ where

 $\varrho_f \equiv \varrho_w = 998,2 \text{ kg/m}^3 \text{ for water at } 20 \text{ °C}$  $\varrho_m = 0.15 \cdot 2700 + 0.85 \cdot 998.2 = 1253 \text{ kg/m}^3$ 

The pressure differential according to equation (46)

 $\Delta_{\mathbf{p}} = \varrho_{\mathbf{m}} \cdot \mathbf{g} \cdot (\mathbf{H} - \Delta \mathbf{H})$ 

 $= 1253 \cdot 9.81 \cdot (57.5 - 9.2) = 593\ 700\ Pa = 5.94\ bar$ 

This is higher than the discharge pressure for handling water  $(\Delta p = 5.25 \text{ bar})$  as per exercise 8.1. Hence, the characteristic curve  $\Delta p = f(Q)$  has increased by 13% for hydrotransport of solids.

**Sought:** The useful volume  $V_N$  of the pump sump according to equation **(48)** (all flow rates in m<sup>3</sup>/h):

$$V_{N} = Q_{in} \cdot (Q_m - Q_{in}) / (Q_m \cdot Z)$$

where

 $Q_m = (Q_{on} + Q_{off}) / 2 = (220 + 150) / 2 = 185 \text{ m}^3/\text{h}$ 

 $V_N = 120 \cdot (185 - 120) / (185 \cdot 10) = 4.22 \text{ m}^3/\text{h}$ 

#### 8.50 Minimum Submergence

**Given:** The vertical unflanged suction pipe according to 8.9 and Fig. 8D, inside pipe diameter d = d<sub>E</sub> = 210.1 mm at a flow rate of Q = 200 m<sup>3</sup>/h.

#### 8.52 Suction Tank Volume

Given: A centrifugal pump system, data according to 8.1 and 8.9, including a suction tank as per Fig. 74. The straight length of the air-filled suction pipe DN 200 (inside diameter  $d_s =$ 210.1 mm according to Table 4) is  $L_s = 3.00$  m, with  $H_{s \text{geo}}$ = 2.60 m (= vertical distance between pump reference plane and water level for positive inlet pressure operation). The atmospheric pressure  $p_b = 989 \text{ mbar} =$ 98900 Pa; density of the water at 20° C = 998.2 kg/m<sup>3</sup>, vapour pressure  $p_v = 2337$  Pa.

8

**Sought:** The minimum submergence  $S_{min}$  in the open suction tanks. The flow velocity  $v_s$  in the suction pipe inlet is

$$v_s = Q/A = (Q/3600)/(\pi \cdot d_E^2/4) = (200/3600) \cdot (\pi \cdot 0.2101^2/4) = 1.60 \text{ m/s}$$

Eq. (50) gives the minimum submergence as

$$\begin{split} S_{\min} &= d_E + 2.3 \cdot v_s \cdot \sqrt{d_E} / g \\ &= 0.2101 + 2.3 \cdot 1.60 \cdot \sqrt{0.2101 / 9.81} \\ &= 0.75 \text{ m.} \end{split}$$

The same result can be obtained faster from the diagram in Fig. 67.

Fig. 66 provides the required distance to the wall with > 0.21 m, the channel width with > 1.26 m and the distance to the floor with > 0.150 m.

Sought: The volume of the suction tank according to Eq. (52):

 $V_{B} = (d_{s}^{2} \pi / 4) \cdot L_{s} \cdot p_{b} / (p_{b} - \varrho \cdot g \cdot H_{s})$ 

The suction lift Hs is defined by Eq. (53):

 $H_s = H_s geo + H_{L,s}$ 

Given is  $H_{s \text{ geo}} = 2.60$  m, the suction pipe head loss  $H_{L,s}$  is to be calculated from  $H_{L,s1}$  and  $H_{L,s2}$  as follows:

1) Head loss  $H_{L,s}$  of the pipe as per 8.9:

$$H_{L,s1} = \lambda \cdot (L / d_s) \cdot v_s^2 / 2g$$

where

- $\lambda = 0.016$  from 8.9
- $L = H_{s geo} = 2.6 \text{ m} (\text{not } 3.0 \text{ m} \text{ because the elbow length is taken}$ into account in  $H_{L,s2}$ )
- $d_s = 0.2101 \text{ m}.$
- $v_s = 1.60 \text{ m}$  from exercise 8.9.

 $H_{L,s1} = 0.016 \cdot (2.60 / 0.2101) \cdot 1.60^2 / (2 \cdot 9.81) = 0.026 \text{ m}$ 

 $H_{L,s2}$  covers the 180° elbow (2 x 90° elbow according to Table 6 as in 8.15) and inlet pipe fittings according to Table 7.

Loss coefficient  $\zeta$  of 180° elbow (factor 1.4) =  $1.4 \cdot 0.10$  = 0.14

Loss coefficient  $\zeta$  of inlet pipe fitting (broken inlet edge) = 0.20

 $H_{L,s2} = \Sigma \zeta \cdot v_s^2 / 2g = (0.14 + 0.20) \cdot 1.60^2 / (2 \cdot 9.81) = 0.044 \text{ m}$ 

3) The total head loss  $H_{L,s} = H_{vs1} + H_{L,s2} = 0.026 + 0.044 = 0.070$  m

and therefore

 $H_s = H_{s \text{ geo}} + H_{L,s} = 2.60 + 0.07 = 2.67 \text{ m}$ 

The example shows that the head loss  $H_{L,s}$  (= 0.070) can be neglected for short suction pipes, since  $H_{s \text{ geo}}$  (2.60 m) is considerably higher. This simplifies the calculation. The volume of the suction tank  $V_B$ can be calculated using Eq. (52) or can simply be determined using the graphs of Fig. 75 (provided the head loss  $H_{L,s}$  is neglected). 
$$\begin{split} &V_B = (d_s^{\ 2}\pi \ / \ 4) \cdot L_s \cdot p_b \ / \ (p_b - \varrho g H_s) \\ &= (0.2101^2 \cdot \pi / 4) \cdot 3.0 \cdot 98 \ 900 \ / \ (98 \ 900 - 998.2 \cdot 9.81 \cdot 2.67) \\ &= 0.141 \ m^3 \end{split}$$

The chosen tank size is 2.8 times the volume of 0.40  $\rm m^3$  (cf. example in Fig. 75).

Check

The lowest pressure is = $p_b - QgH_s$	= 72 828 Pa
The vapour pressure is 0.02337 bar	= 2337 Pa

This means the pressure does not fall below vapour pressure during venting.

#### 9. Additional Literature

- [1] Product literature (KSB sales literature)
- [2] KSB Centrifugal Pump Lexicon
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- [4] Gebäudetechnik von KSB. Pumpenregelung und Anlagenautomation. Planungshinweise. (Building Services Products from KSB. Pump Control and Plant Automation. Planning Information). KSB publication No. 2300.024 (2005)
- [5] Bernauer J., M. Stark, W. Wittekind: Improvement of Propeller Blades Used for Handling Liquids Containing Fibrous Solids.
   KSB Technische Berichte 21e (1986), pp. 16 – 21
- [6] Bieniek K., Gröning N.: Controlling the Output of Centrifugal Pumps by Means of Electronic Speed Control. KSB Technische Berichte 22e (1987), pp. 16 – 31
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- [9] Holzenberger K.: A Comparison of Two Conversion Methods Applied to the Characteristics of Centrifugal Pumps While Pumping Viscous Liquids. KSB Technische Berichte 25e (1988), pp. 45 – 49
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- [11] Kosmowski I., Hergt P.: Förderung gasbeladener Medien mit Hilfe von Normal- und Sonderausführungen bei Kreiselpumpen (Pumping Gas-laden Fluids by Standard and Special Design Centrifugal Pumps). KSB Technische Berichte 26 (1990), pp. 14 – 19
- [12] Schreyer H.: Glandless Chemical Pump with Magnetic Drive. KSB Technische Berichte 24e (1988), pp. 52 – 56

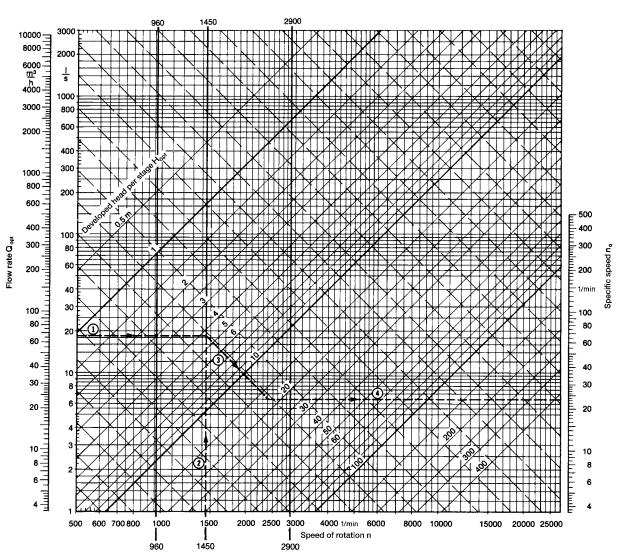
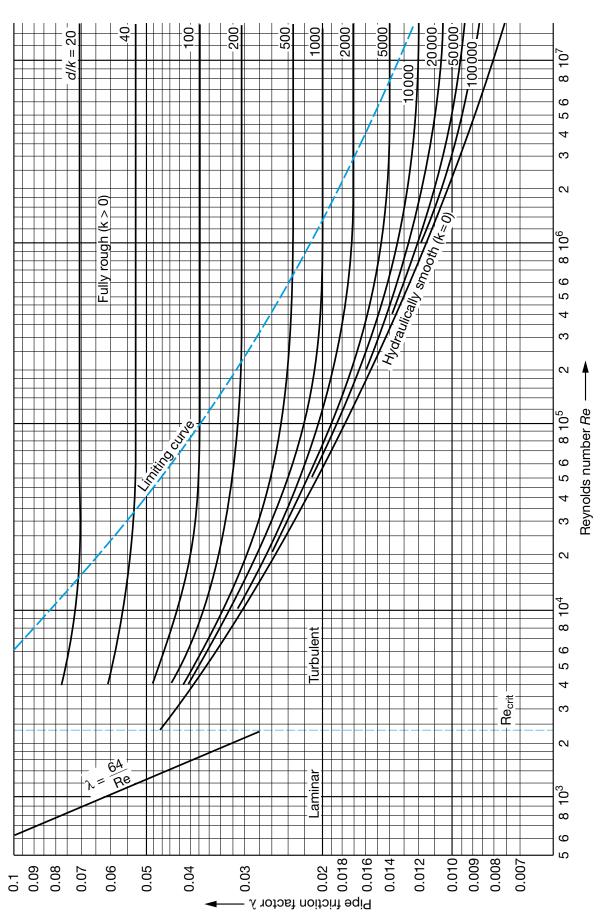


Fig. 3: Nomograph to determine the specific speed  $n_q$ *Example:*  $Q_{opt} = 66 \ m^3/h = 18.3 \ l/s; \ n = 1450 \ rpm, \ H_{opt} = 17.5 \ m.$  Found:  $n_q = 23 \ (metric \ units).$ 

10.

**Technical Annex** 





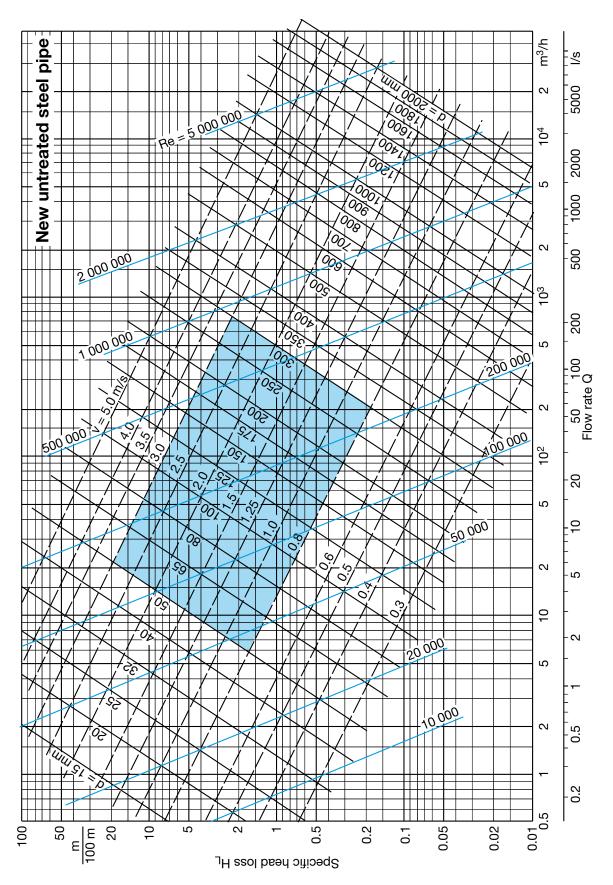
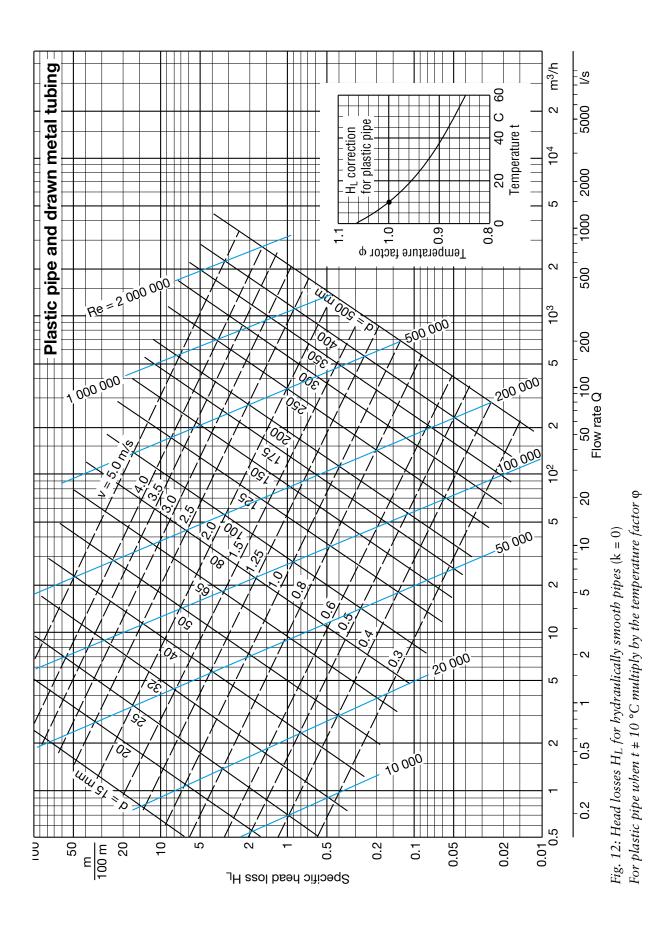


Fig. 11: Head losses  $H_L$  for new steel pipes (k = 0.05 mm)



# 

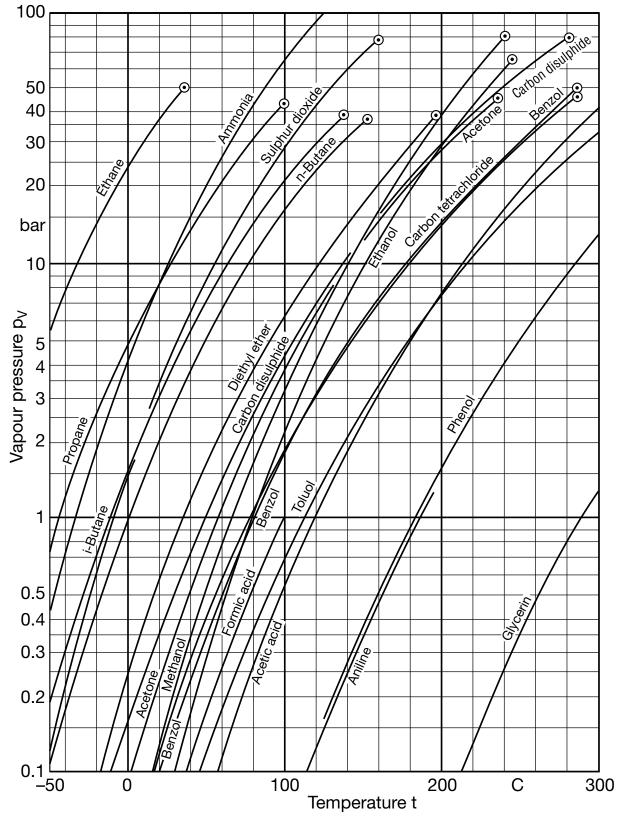
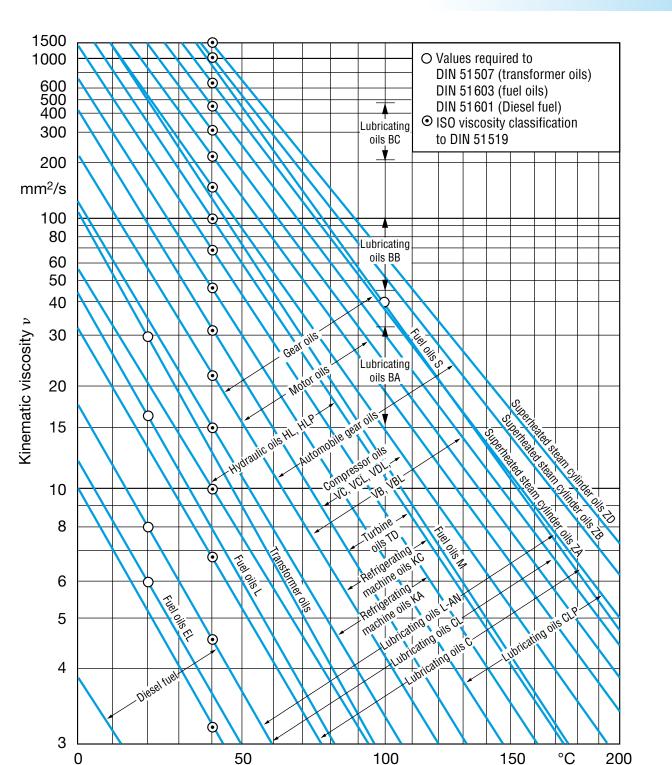


Fig. 35: Vapour pressure  $p_v$  of various fluids as a function of the temperature t



Temperature t

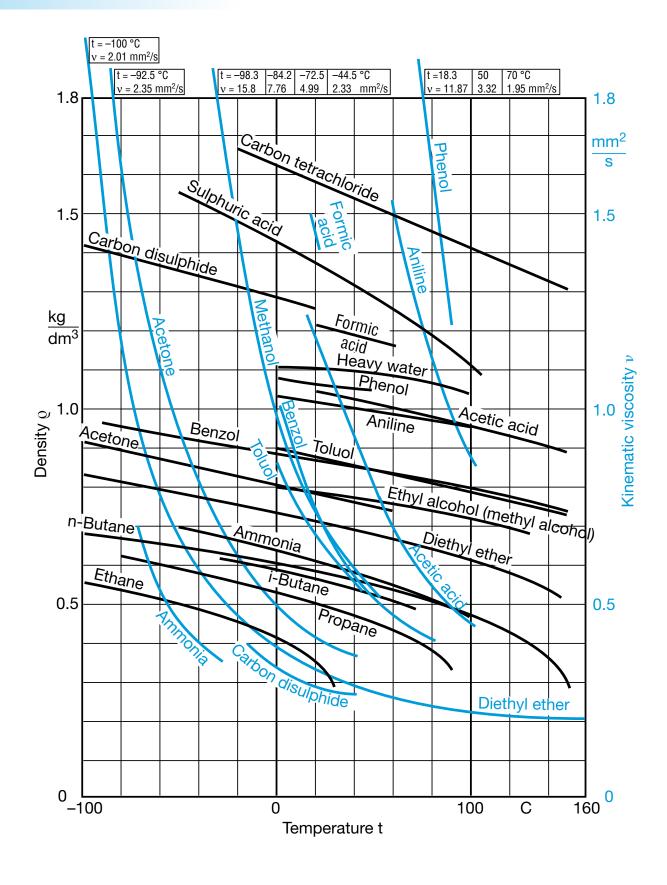


Fig. 48: Density  $\varrho$  and kinematic viscosity v of various fluids as a function of the temperature t

# How to Find the Operating Point

# Given:

Flow rate	Q <sub>w</sub>		m³/h
Total developed head	H <sub>w</sub>		m
Speed of rotation	n		1/min
Kinematic viscosity	νΖ		m²/s
Density	Qz		kg/m <sup>3</sup>
Gravitational constant	g	9.81	m/s <sup>2</sup>

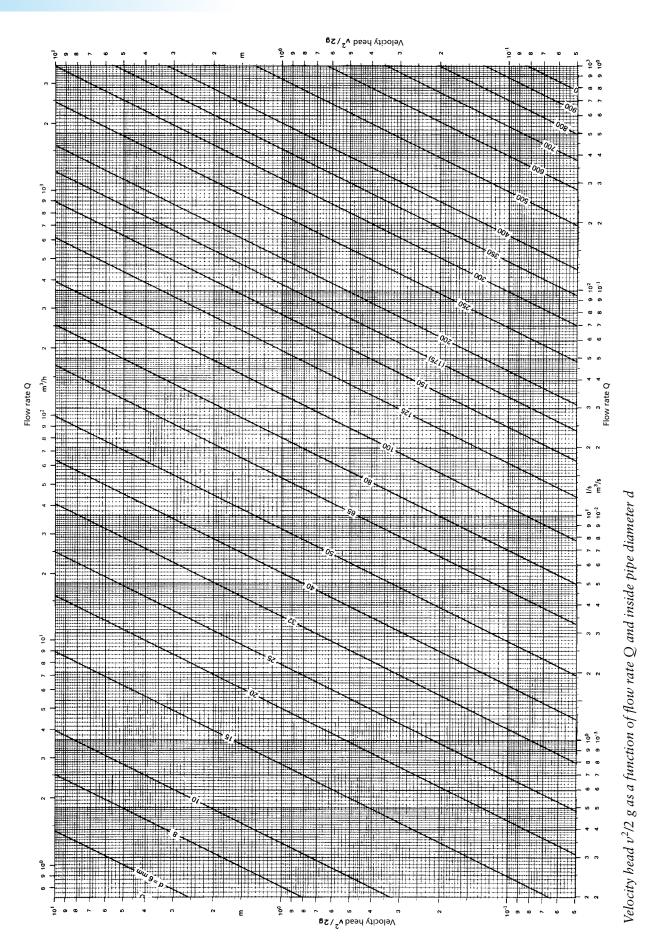
## Calculation

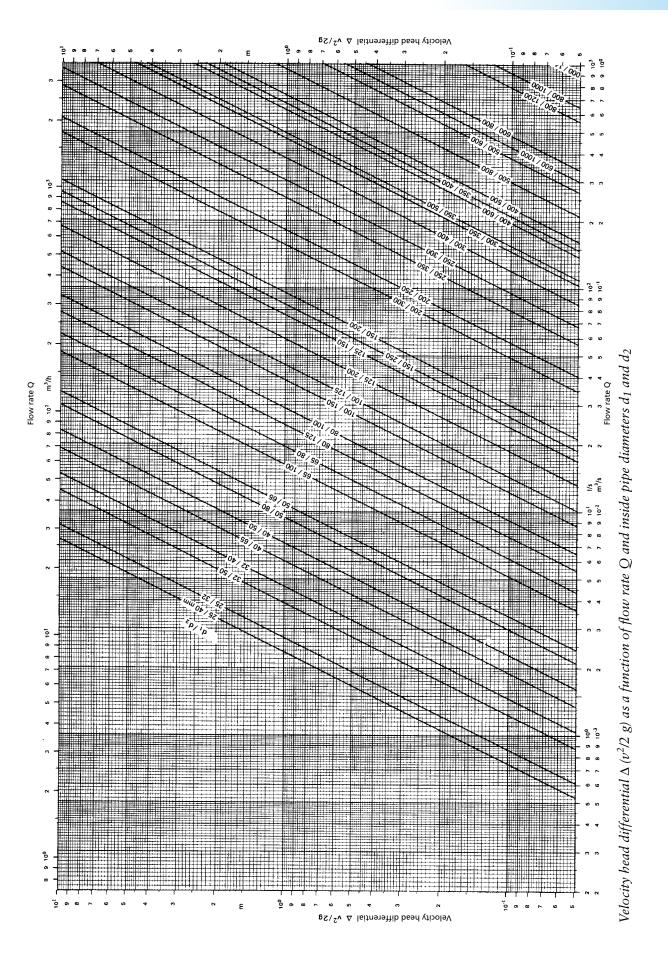
Q/Q <sub>o</sub>	pt =	0	0.8	1.0	1.2	-	
Q <sub>w</sub>	from the	0				m³/h	
H <sub>w</sub>	pump cha- racteristic					m	
$\eta_w$	for 4 points	0				-	
n <sub>q, w</sub>	from section 3.1.5	-	_		-	1/min	
f <sub>Q, w</sub>	from Fig. 50	-			•	-	
f <sub>H, w</sub>		_				-	
$f_{\eta, W}$		_				-	
$Q_z =$	$Q_{w} \cdot f_{Q, w}$	0				m³/h	
$H_z =$		= H <sub>w</sub>	$=$ $H_{W} \cdot f_{H,W} \cdot 1,03$	$H_{W} \cdot f_{H, W}$	$H_{W} \cdot f_{H,W}$		
		+	1) *	+	*	m	> 2)
	η <sub>w</sub> ∙ f <sub>η, w</sub>	0					( -,
$P_z = \frac{6}{1}$	$\frac{Q_z \cdot g \cdot H_z \cdot Q_z}{\eta_z \cdot 1000 \cdot 3600}$	$\mathbf{X}$				kW	

 $^{1)}$  If  $H_{Z}$  becomes larger than  $H_{W},$  it should be set to  $H_{W}.$ 

 $^{2)}$  These are four values on the  $H_z/Q$  and  $Q/\eta_z$  curve and three points on the  $Q/P_z$  curve. Plot versus Q.

Fig. 51: Spreadsheet for calculating the pump characteristics for a viscous fluid using the KSB method





# **11. Excerpt of Important Units for Centrifugal Pumps**

Physical	Sym-	Units			Units not	Recom-	Comments
dimension	bol	SI unit	s	Other units (not complete)	to be used any longer	mended units	
Length	1	m	Metre	km, dm, cm, mm, µm		m	Base unit
Volume	V	m <sup>3</sup>		$dm^3$ , $cm^3$ , $mm^3$ , litre (1 l = 1 dm <sup>3</sup> )	cbm, cdm	m <sup>3</sup>	
Flow rate, capac- ity, volume flow	Q, V	m <sup>3</sup> /s		m <sup>3</sup> /h, l/s		l/s and m³/s	
Time	t	s	Second	s, ms, μs, ns, min, h, d		s	Base unit
Speed of rotation	n	1/s		1 /min (rpm)		1 /s, 1 /min	
Mass	m	kg	Kilogram	g, mg, μg, metric ton (1 t = 1000 kg)	Pound, hundred- weight	kg	Base unit The mass of a commercial commodity is described as weight.
Density	Q	kg/m <sup>3</sup>		kg/dm <sup>3</sup>		kg/dm <sup>3</sup> und kg/m <sup>3</sup>	The term "spezifice gravita" must no longer be employed, because it is ambiguous (see DIN 1305).
Mass moment of inertia	J	kg m <sup>2</sup>				kg m <sup>2</sup>	Mass moment, 2. order
Mass rate of flow	ṁ	kg/s		t/s, t/h, kg/h		kg/s and t/s	
Force	F	Ν	Newton (= kg m/s <sup>2</sup> )	kN, mN, μN,	kp, Mp,	Ν	1  kp = 9.81  N. The weight force is the product of the mass m by the local gravitational constant g.
Pressure	р	Ра	Pascal (= N/m <sup>2</sup> )	bar (1 bar=10 <sup>5</sup> Pa)	kp/cm <sup>2</sup> , at, m w.c., Torr,	bar	1 at = $0.981$ bar = $9.81 \cdot 10^4$ Pa 1 mm Hg = $1.333$ mbar 1 mm w.c. = $0.098$ mbar
Mechanical stress (strength)	σ, τ	Ра	Pascal (= N/m <sup>2</sup> )	N/mm <sup>2</sup> , N/cm <sup>2</sup>	kp/cm <sup>2</sup> ,	N/mm <sup>2</sup>	1 kp/mm <sup>2</sup> = 9.81 N/mm <sup>2</sup>
Bending mo- ment, torque	М, Т	N m			kp m,	N m	1 kp m = 9.81 N m
Energy, work, quantity of heat	W, Q	J	Joule (= N m = W s)	kJ, Ws, kWh, 1 kW h = 3600 kJ	kp m kcal, cal, WE	J und kJ	1 kp m = 9.81 J 1 kcal = 4.1868 kJ
Total head	Н	m	Metre		m l. c.	m	The total head is the work done in $J = N$ m applied to the mass unit of the fluid pumped, referred to the weight force of this mass unit in N.
Power	Р	W	Watt (= J/s = N m/s)	MW, kW,	kp m/s, PS	kW	1 kp m/s = 9.81 W 1 PS = 736 W
Temperature difference	Т	K	Kelvin	°C	°K, deg.	К	Base unit
Kinematic viscosity	ν	m²/s			St (Stokes), °E,	m²/s	1 St = $10^{-4} \text{ m}^2/\text{s}$ 1 cSt = 1 mm <sup>2</sup> /s
Dynamic viscosity	η	Pas	Pascal second (= N s/m <sup>2</sup> )		P (Poise)	Pa s	1 P = 0.1 Pa s
Specific speed	nq	1				1	$n_q = 333 \cdot n \cdot \frac{\sqrt{Q_{opt}}}{(g H_{opt})^{3/4}}$ Sl units (m und s)

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