Основы проектирования центробежных насосов. Basic principles for the design of centrifugal pump installations.

STERLING SIHI

Basic Principles for the Design of Centrifugal Pump Installations

$$\frac{Q_x}{Q} = \frac{n_x}{n} \qquad H = (z_2 - z_1) + \frac{p_2 - p_1}{\rho \cdot g} + \frac{u_2^2 - u_1^2}{2g}$$

$$\frac{H_x}{H} = \left(\frac{n_x}{n}\right)^2 \qquad P_u = \rho \cdot Q \cdot g \cdot H$$

$$\frac{H_x}{H} = \left(\frac{D_x}{D}\right)^2 \qquad n_s = n \cdot \frac{(Q/Q_s)^{0.5}}{(H/H_s)^{0.75}}$$

$$\eta_x = 1 - (1 - \eta) \cdot \left(\frac{n}{n_x}\right)^{0.1}$$

$$\Delta H_0 = \frac{\Delta H_I \cdot Q_{II}^2 - \Delta H_{II} \cdot Q_I^2}{Q_{II}^2 - Q_I^2}$$

$$NPSHA = (z_1 - z_D) + \frac{p_1 + p_{amb} - p_v}{\rho \cdot g} + \frac{u_1^2}{2g}$$

$$H_{dyn} = \frac{U_{A2}^2 - U_{A1}^2}{2g} + H_{Jr} \qquad \frac{Q_x}{Q} = \left(\frac{D_x}{D}\right)^2$$

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BASIC PRINCIPLES FOR THE DESIGN OF CENTRIFUGAL PUMP INSTALLATIONS

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Foreword

This is the 7th edition of this Pump Book, and since it was first written it has become an established and respected reference book . It has been completely reworked and enlarged to include the very latest international and European standards. In many cases reference is also made to old standards as well as the new.

This comprehensive edition covers all aspects concerned with the pumping of liquids and is a dependable reference work in the design and operation of pumps and pump systems.

In addition it offers both designers and operators a wealth of detail when considering the whole life costs of a pump installation.

It has been compiled by Sterling SIHI's most experienced and knowledgeable engineers, ensuring it's content should be invaluable to any engineer who is involved with the pumping of liquids.

Our thanks to all those who assisted in its compilation.

Centrifugal Pumps
Vacuum Technology
Engineered Systems
Service Support

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- Side Channel Pumps
- Multistage Ring Section
- Pitot Tube Pumps
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Formulae, Notation and Units

The following is an overview of the most important terms and units which are used in relation to centrifugal pumps and pump installations.

Notation	Terminology	Unit	Other common units
A	Area, cross sectional area	m^2	mm^2, cm^2, dm^2
D	Diameter e.g. impellers pipelines	m	mm
d	Diameter e.g. shafts, shaft sleeves orifices	m	mm
E	Energy	J	kJ
F	Power	N	kN
f	Frequency	Hz	
G	Gravitational force (local)	N	kN
g	Acceleration due to gravity (local)	m/s^2	
H	Total head of pump	m	
J	Moment of inertia	kg m^2	
K	Unit of speed	s^{-1}	
k	Roughness coefficient of pipes	m	mm
l	Length	m	mm, cm, dm
M	Torque	N m	
n	Speed	s^{-1}	min^{-1}
NPSH	Nett positive suction head	m	
P	Power	W	kW, MW
p	Pressure	Pa	hPa, bar
Q	Flowrate	m^3/s	m^3/h, L/s
q	Massflow	kg/s	kg/h, t/h
Re	Reynolds number	dimensionless	

Formulae, Notation and Units (continuation)

S	Suction capacity	m^3/s	m^3/h
T	Thermodynamic temperature	K	
t	1. Temperature Celsius 2. Time	°C s	 min, h, d
U	Mean flow velocity	m/s	
u	Tip speed	m/s	
V	Volume	m^3	dm^3, L
v	Speed	m/s	km/h
y	Specific energy	J/kg	m^2/s^2
z	Altitude Height above datum	m	
ξ	Loss Coefficient	dimensionless	
η	Efficiency		%
λ	Coefficient of friction	dimensionless	
μ	Dynamic viscosity	Pa s	mPa s, N s/m²
ν	Kinematic viscosity	m^2/s	mm^2/s
ρ	Density	kg/m^3	kg/dm^3 kg/L
ω	Angular velocity	rad/s	

Indices

Parameters or measured values are referenced to a specific point or condition by use of indices.

Index	Meaning	Example
A	Referred to pump installation	H_A Total head of installation
abs	Absolute	p_{abs} Absolute pressure
all	Allowable	$n_{max\ all}$ Maximum allowable speed
amb	Ambient	p_{amb} Ambient pressure
B	Balancing	Q_B Balancing flow

Indices (continued)

Index	Meaning	Example
D	Difference, datum level	z_D Height of NPSH datum
dyn	Dynamic	H_dyn Dynamic component of total head
G	Guarantee	Q_G Guaranteed flowrate
g	Slip	v_g Slip speed
geo	Static	H_{geo} Static head
gr	Referred to pump unit	η_gr Efficiency of pump unit
int	Internal	η_int Internal efficiency
J	Loss	H_J Head loss
L	Leakage	Q_L Leakage flowrate
M	1. Pressure gauge, manometer 2. Referred to the fluid in manometer measurement line	p_1M Pressure reading at pump inlet ρ_M Density of fluid in manometer measurement line
m	Referred to mechanical efficiency	$\hbar_\mathrm{m}$ Mechanical efficiency
max	Maximum	n_max Maximum speed
min	Minimum	n_min Minimum speed
mot	Referred to the motor	P_mot Absorbed power of pump drive
N	Nominal	$P_\mathrm{N\ mot}$ Nominal motor power
op	Operating	Q_op Operating flowrate
opt	Optimum	Q_opt Flowrate at optimum efficiency
r	Rated	Q_r Rated flowrate
s	1. Specific 2. Suction	n_s Specific speed $H_\mathrm{s\ geo}$ Static datum suction head
sch	Peak	H_sch Total head at peak flowrate on a characteristic
sp	Specific	n_sp Specific speed
ss	Suction specific	n_ss Specific suction number

Indices (continued)

Index	Meaning	Example
stable	Stable	$Q_{\text{min stable}}$ Minimum continuous stable flowrate
stat	Static	H_{stat} Static head
T	Transmitted	M_T Transmitted torque
t	Total	H_{Jt} Total head loss of installation
thermal	Thermal	$Q_{\text{min thermal}}$ Minimum thermal flowrate
u	Useful	P_u Hydraulic performance
v	Vapour	p_v Vapour pressure
w	Operating	p_w Operating pressure
x	1. Referred to any chosen point in the installation 2. Referred to any chosen value	H_x Head at point x D_x Required impeller Ø for a particular operating point
z	Suction (Positive)	$H_{\text{z geo}}$ Static suction head (pos.)
0	At flowrate $Q = 0$	H_0 Zero flow head
1	Suction side	p_1 Pressure in pump suction
1’	Suction side measurement point	$p_{1'}$ Pressure at suction side measurement point
2	Discharge side	p_2 Pressure in pump discharge
2’	Discharge side measurement point	$p_{2'}$ Pressure at discharge side measurement point
3, 4	Intermediate point	p_3 Pressure at intermediate point
3’, 4’	Intermediate measurement point	$p_{3'}$ Pressure at intermediate measurement point

Note: A hyphen (-) between 2 indices indicates a difference in value at the given point but does not indicate which is the larger.

Example: z_{1-2} = Difference between z_1 and z_2 .

Either $z_1 - z_2$ or $z_2 - z_1$

1 Selection of centrifugal pumps and installations

1.1 Flowrate *Q*

The flowrate Q is the usable flow (volume of liquid per unit of time) discharged by the pump through its outlet branch.

The common units are m³/s, m³/h and l/s.

Flow extracted for other purposes before the outlet branch of the pump must be taken into account in establishing the pump flowrate.

Internal recirculating flows e.g. balance flow Q_B and leakage loss Q_L, are not calculated in Q.

Flowrates are defined as follows:

Notation	Terminology	Definition
Q_{opt}	Optimum flow	Flowrate at point of optimum efficiency
Q_r	Selection flowrate	Flowrate for selection data taking account of all necessary allowances
Q_w	Operating flowrate	Flowrate expected in normal operating conditions.
Q_{max}	Maximum flowrate	Maximum flowrate which can be expected.
Q_{min}	Minimum flowrate	Minimum flowrate which can be expected.
$Q_{max\ all}$ and $Q_{min\ all}$	Maximum or minimum permissible flowrate	The maximum or minimum flowrate which the pump can deliver continuously without damage when operating at the selected speed and liquid for which it has been supplied.
$Q_{min\ stable}$	Minimum stable flowrate	The smallest flowrate at which the pump can operate without exceeding the minimum or maximum limits for reliability, noise or vibration.
$Q_{min\ thermal}$	Minimum continuous thermodynamic flowrate	The smallest flowrate at which the pump can operate without the operational temperature rise deteriorating the pumped liquid.
Q_B	Compensation flow	The flow rate required to operate the axial thrust compensation device.

Q_L	Leakage flow	The flowrate which leaks out through the shaft seals.
Q_1	Suction flowrate	The flowrate from the plant which enters the suction branch of the pump.
Q_2	Discharge flowrate	The flowrate which is delivered to the plant through the discharge branch of the pump.
$Q_{3,4\,...}$	Intermediate flowrate	The flowrate which is drawn from one or more extraction branches from the principal flow.

1.2 Massflow *q*

The massflow *Q* is the usable mass of liquid per unit of time discharged by the pump through its outlet branch.

The common units are kg/s and t/h.

Extracted and circulating massflow and leakage follow the same rules as identified under flowrate.

The relationship between massflow *q* and flowrate *Q* is

$$q = \rho \cdot Q \qquad (\rho = \text{density of liquid})$$

Note:

The terminologies listed above for flowrate can be used with the same meaning for massflow e.g. q_r = selection massflow

1.3 Total head of pump

1.3.1 The total head of a pump *H* is the usable mechanical work transferred by the pump to the pumped liquid and expressed in terms of potential force of the pumped liquid under the local gravitational force.

Heads are defined as follows:

Notation	Terminology	Definition.
H_{opt}	Optimum head	Total head at optimum efficiency.
H_r	Selection head	Total head for selection data taking account of all necessary allowances.
H_0	Shut off head	Total head at zero flow (Q = 0).
H_{sch}	Peak head	Total head at the peak of a head characteristic.

Total head *H* is measured as the increase in the usable mechanical energy *E* of the flowrate per unit weight *G* between the inlet and outlet branch of the pump.

Using the unit of energy N·m and the unit of force N, the energy per unit of weight and hence the total head is expressed in the unit metre.

$$N \cdot m / N = m$$

In spite of this unit the total head should not in principle be interpreted as a height, e.g. the height of a column of liquid.

Under constant speed and flowrate Q, the total head H is independent of the density ρ, but is however dependent on the viscosity ν and inversely proportional to the acceleration due to gravity g.

1.3.2 The total head of the system H_A

The total head of the system H_A is the total head of the pump H required to maintain the flowrate Q in the plant.

In continuous (stable) flow conditions $H = H_A$. During start up $H > H_A$; the differential provides the acceleration of the liquid in the pipeline.

1.3.3 Establishing the total head

In establishing the total head, along with the flowrate, the magnitude of the plant parameters as described in the following sections are required.

1.3.4 Height

The height referred to here is the difference in altitude between the observed point and the datum level in the plant.

The datum level of the plant is any horizontal plane which serves as a reference point from which the height is established. For practical reasons it is advisable to specify an exactly definable level in the plant, for example the floor level on which the pump foundation is mounted. A datum level which refers to the dimensions of the pump e.g. the shaft centre line or suction flange should be avoided.

The height is measured in metres (m).

The heights are defined as follows:

Notation	Terminology	Definition
z	Height	Difference in altitude between the observed point and the datum level. *Note:* The height can be negative when the observed point is below the datum level.

z_1	Height of pump suction.	Height of centre point of pump entry branch.
z_2	Height of pump discharge.	Height of centre point of pump discharge branch.
$z_{1'}$	Height of suction side measurement point.	Height of manometer tapping point in the suction side pipework.
$z_{2'}$	Height of discharge side measurement point.	Height of manometer tapping point in the discharge side pipework.
z_{A1}	Height at entry to plant.	Height of liquid level in the entry section of the plant. If no liquid level is given then equals the height of centre point of pump entry branch.
z_{A2}	Height at discharge to plant.	Height of liquid level in the discharge section of the plant. If no liquid level is given then equals the height of centre point of pump entry branch.
z_D	Height of the (*NPSH*)-datum.	Difference in altitude between the datum level of the plant and the (*NPSH*) reference level.

When the height difference between two levels is given with the index *z,* the points are identified and separated by hyphen.

Notation	Terminology	Definition
$z_{1\text{-}2}$	Height difference between pump inlet and discharge.	$z_{1\text{-}2} = z_2 - z_1$
$z_{1'\text{-}M}$	Height difference between suction side manometer and measurement point tapping.	Datum point = zero or midpoint of manometer.
$z_{2'\text{-}M}$	Height difference between discharge side manometer and measurement point tapping.	Datum point = zero or midpoint of manometer.
$z_{x\text{-}x}$	Height difference.	Height difference between one observed level in the plant and another observed level.

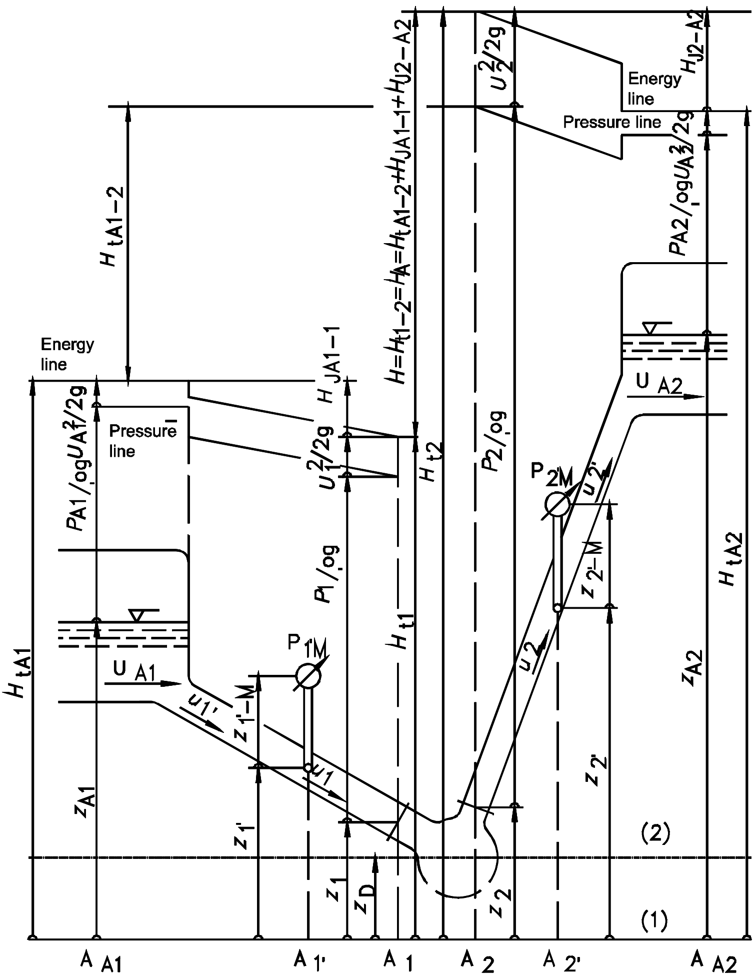


Fig. 1.01 Centrifugal pump installation

(1)=Datum level of plant
(2)=(NPSH)-level

Indices

A1=Suction side of plant
A2=Discharge side of plant

1 =Suction side of pump
1'=Suction side tapping point
2 =Discharge side of pump
2'=Discharge side tapping point

For a number of level differences a special terminology is common as outlined below.

Notation	Terminology	Definition
H_{geo}	Static head	Level difference between inlet and outlet branches. $H_{geo} = z_{A1\text{-}A2} = z_{A2} - z_{A1}$
$H_{z\ geo}$	Static feed head	Height difference between inlet of plant and (*NPSH*) datum level. $H_{z\ geo} = z_{A1\text{-}D} = z_{A1} - z_{D}$
$H_{s\ geo}$	Static suction head	Height difference between inlet of plant which lies below the datum level and the (*NPSH*) datum level. $H_{s\ geo} = z_{A1\text{-}D} = - z_{A1} - z_{D}$

1.3.5 Cross sectional area

The cross section referred to here relates to the size of the branches.

The common unit used is m².

The areas are defined as follows:

Notation	Terminology	Definition
A_1	Entry section of pump	The open cross section of the entry branch of the pump. *Note* For pumps without an entry branch the entry cross section must be defined.
A_2	Discharge section of the pump	The open cross section of the discharge branch of the pump. *Note* For pumps without a discharge branch the discharge cross section must be defined. *Note 2* For in line, submersible and similar pumps with in-built suction lift pipework, the cross section of this pipework can be considered as the discharge cross section.

$A_{1'}$	Cross section of suction side measurement point.	Open cross section of manometer tapping point in the suction side pipework.
$A_{2'}$	Cross section of discharge side measurement point.	Open cross section of manometer tapping point in the discharge side pipework.
A_{A1}	Cross section at entry to plant.	Open cross section in a defined area of the entry of the plant for which the area, head and pressure are known.
A_{A2}	Cross section at discharge to plant.	Open cross section in a defined area of the discharge of the plant for which the area, level and pressure are known.

1.3.6 Flow velocity

The mean axial flow velocity is the ratio of the flowrate at a defined point to the cross section at that point

$$U = \frac{Q}{A} \quad \text{in m/s} \qquad \text{with } Q \text{ in m}^3\text{/s and } A \text{ in m}^2$$

If the cross section is a pipe section with a nominal diameter DN of the pipe, then the velocity U can be related to the flowrate Q in m³/h and the nominal diameter DN with the following equation:

$$U = \left(\frac{18{,}8}{\mathrm{DN}}\right)^2 \cdot Q \quad \text{in m/s} \quad \text{with } Q \text{ in m}^3\text{/h} \quad [18{,}8 \text{ is rounded off value of } 18{,}806319]$$

In this equation the nominal diameter in mm is generally so close to the actual diameter that the result of the calculation is sufficiently accurate.

The flow velocity at various selected cross sections locations are referred to with the following indices:

Notation	Definition
U_1	Flow velocity in the entry branch of the pump.
U_2	Flow velocity in the discharge branch of the pump.
$U_{1'}$ and $U_{2'}$	Flow velocity in the open section of the entry / discharge measurement point. *Note:* only of value if $A_{1'} \neq A_1$ and/or. $A_{2'} \neq A_2$
U_{A1}	Flow velocity in the entry section of the plant.
U_{A2}	Flow velocity in the discharge section of the plant.

1.3.7 Pressure

Pressures are defined as follows:

Notation	Terminology	Definition
p_{amb}	Atmospheric pressure	The mean atmospheric pressure at the pump location.
p_v	Vapour pressure of the pumped media	The absolute pressure at which the pumped media evaporates at a given temperature.

The pressures at various selected cross sections locations are referred to with the following indices:

In contrast to the atmospheric pressure p_{amb} and vapour pressure p_v these are always given as gauge values, (over/under)

Notation	Terminology	Definition
p_1	Pressure at pump entry	Pressure in entry section of pump at level z_1
p_2	Pressure in pump discharge	Pressure in discharge section of pump at level z_2
$p_{1'M}$	Pressure at entry side manometer	Pressure reading on manometer located on entry side of pump at level $z_{1'}$
$p_{2'M}$	Pressure at discharge side manometer	Pressure reading on manometer located on discharge side of pump at level $z_{2'}$
p_{A1}	Pressure at entry to plant	Pressure in entry section A_1. If a level of liquid is present then this is the imposed pressure
p_{A2}	Pressure at outlet of plant	Pressure in discharge section A_2. If a level of liquid is present then this is the imposed pressure

The unit of pressure is the Pascal (Pa). The common unit for pumps and pump installations is the bar.

The calculation to convert the manometer pressures $p_{1'M}$ and $p_{2'M}$ to the entry and discharge pressures at the pump p_1 and/or p_2 can be made using the following equation.

The following equations are valid for pressures in bar and density in kg/dm³

$$p_1 = p_{1'M} + [\rho_M \cdot g \cdot z_{1'\text{-}M} + \rho \cdot g \cdot (z_{1\text{-}1'} + \frac{U_{1'}^2 - U_1^2}{2g} - H_{J1'\text{-}1})] \cdot 10^{-2} \text{ in bar}$$

$$p_2 = p_{2'M} + [\rho_M \cdot g \cdot z_{2'\text{-}M} + \rho \cdot g \cdot (z_{2'\text{-}2} + \frac{U_{2'}^2 - U_2^2}{2g} + H_{J2\text{-}2'})] \cdot 10^{-2} \text{ in bar}$$

Note:

ρ is the density of the pumped media, ρ_M is the density of the liquid in the manometer. If the manometer line is filled with pumped media then $\rho_M = \rho$. If the manometer line is filled with air then the first part of the equation in square brackets can be ignored as $\rho_{air} \ll \rho_{liquid}$.

1.3.8 Head loss

Head loss refers to the loss of mechanical energy between the start and finish of a section of pipework i.e. the pipe losses including the inlet and outlet losses and the losses due to instruments and other fittings. Losses inside the pump itself are not included. The unit of head loss is the metre (m).

The head losses are defined as follows:

Notation	Terminology	Definition
$H_{J\ x\text{-}x}$	Head loss	The difference in potential energy at one selected point to that at another point. *Note:* The loss can be expressed as total energy level, pressure level or velocity level.
$H_{J\ A1\text{-}1}$	Entry head loss	Head loss between entry section of the plant and inlet branch of the pump.
$H_{J\ 2\text{-}A2}$	Discharge head loss	Head loss between discharge branch of pump and outlet section of plant.
H_{Jt}	Total head loss in the installation	Sum of entry and discharge losses $H_{Jt} = H_{J\ A1\text{-}1} + H_{J\ 2\text{-}A2}$

1.3.9 Energy level

The energy level is the usable mechanical energy, i.e. the sum of the static head over the datum level, the pressure level (gauge pressure over atmospheric pressure) and the flow velocity level.

The usable pressure energy of the liquid under static pressure p is referred to as pressure head. The unit of pressure head is the metre (m).

$$\text{Pressure head at point x} = \frac{p_x \cdot 100}{\rho_x \cdot g} \quad \text{in m, with } p \text{ in bar and } \rho \text{ in kg/dm}^3$$

The usable dynamic energy of the pumped media is referred to as velocity head. The unit of velocity head is the metre (m).

$$\text{Velocity head at point x} = \frac{U_x^2}{2\,g} \quad \text{in m, with } U \text{ in m/s}$$

The total energy level at any point is:

$$H_{tx} = z_x + \frac{p_x \cdot 100}{\rho_x \cdot g} + \frac{U_x^2}{2\,g} \quad \text{in m, with } p \text{ in bar and } \rho \text{ in kg/dm}^3$$

Absolute energy level is calculated with pressure head expressed as absolute pressure. Absolute energy level therefore exceeds the energy level by the value of atmospheric pressure.

$$H_{tx\ abs} = H_{tx} + \frac{p_{amb} \cdot 100}{\rho_x \cdot g} \quad \text{in m, with } p \text{ in bar and } \rho \text{ in kg/dm}^3$$

The energy levels are defined as follows:

Notation	Terminology	Definition
H_{t1}	Total energy at pump entry	$H_{t1} = z_1 + \frac{p_1}{\rho \cdot g} + \frac{U_1^2}{2\,g}$
H_{t2}	Total energy at pump discharge	$H_{t2} = z_2 + \frac{p_2}{\rho \cdot g} + \frac{U_2^2}{2\,g}$
H_{tA1}	Total energy at entry to plant	$H_{tA1} = z_{A1} + \frac{p_{A1}}{\rho \cdot g} + \frac{U_{A1}^2}{2\,g}$
H_{tA2}	Total energy at discharge of plant	$H_{tA2} = z_{A2} + \frac{p_{A2}}{\rho \cdot g} + \frac{U_{A2}^2}{2\,g}$

1.3.10 Calculation of total head

The total head or usable mechanical energy of a liquid is in general the sum of head energy, pressure energy and dynamic energy.

Total head values are calculated with the following equations:

Total head of the pump

$$H = H_{t2} - H_{t1} = \underbrace{(z_2 - z_1)}_{\text{Head energy}} + \underbrace{\frac{p_2 - p_1}{\rho \cdot g}}_{\text{Pressure energy}} + \underbrace{\frac{U_2^2 - U_1^2}{2\,g}}_{\text{Dynamic energy}}$$

Total head of the plant

$$H_A = H_{tA2} - H_{tA1} = \underbrace{(z_{A2} - z_{A1})}_{\text{Head energy}} + \underbrace{\frac{p_{A2} - p_{A1}}{\rho \cdot g}}_{\text{Pressure head}} + \underbrace{\frac{U_{A2}^2 - U_{A1}^2}{2\,g}}_{\text{Velocity head}} + \underbrace{H_{JA1\text{-}1} + H_{J2\text{-}A2}}_{\text{Head losses}}$$

The total head of the plant can also be defined as follows:

$$H_A = \underbrace{H_{stat}}_{\text{Static component of total head}} + \underbrace{H_{dyn}}_{\text{Dynamic component of total head}}$$

The static component of the total head is independent of the flowrate and made up of head energy and pressure energy.

$$H_{stat} = (z_{A2} - z_{A1}) + \frac{p_{A2} - p_{A1}}{\rho \cdot g}$$

The dynamic component of the total head is dependent on the flowrate and made up of dynamic energy and head losses.

$$H_{dyn} = \frac{U_{A2}^2 - U_{A1}^2}{2\,g} + H_{Jt}$$

1.3.11 Example of calculation of total head

1st Example

Calculation of total head H of a centrifugal pump

Given: A_1 Entry section of pump DN_1 = DN 125

A_2 Discharge section of pump DN_2 = DN 80

$A_{1'}$ Cross section of suction side measurement point= DN 150

$A_{2'}$ Cross section of discharge side measurement point = DN 125

Heights

$z_1 = 350$ mm $\quad z_2 = 650$ mm

$z_{1'} = 370$ mm $\quad z_{2'} = 700$ mm

$z_{1'\text{-}M} = 140$ mm $\quad z_{2'\text{-}M} = 120$ mm

Pumped media: cold water, $\rho = 1{,}0$ kg/dm³

Required: The total head of the pump H for $Q_r = 100$ m³/h

The following manometer readings are taken:

$p_{1'M} = -0{,}2$ bar (gauge!) $\quad p_{2'M} = 11{,}4$ bar

The manometer lines are filled with the pumped media, i.e. $\rho_M = \rho$

The head loss is calculated as follows:

$H_{J\,1'\text{-}1} = 0{,}007$ m $\quad H_{J\,2\text{-}2'} = 0{,}015$ m

The flow velocity is calculated as $U = (18{,}8/DN)^2 \cdot Q$.

$U_1 = 2{,}26$ m/s $\quad U_2 = 3{,}53$ m/s

$U_{1'} = 1{,}57$ m/s $\quad U_{2'} = 2{,}26$ m/s

The pressure at pump entry p_1 and discharge p_2 are calculated with the equation shown previously:

$$p_1 = -0{,}2 + \left[1 \cdot 9{,}81 \cdot 0{,}14 + 1 \cdot 9{,}81\left(0{,}02 + \frac{1{,}57^2 - 2{,}26^2}{2 \cdot 9{,}81} - 0{,}007\right)\right] \cdot 10^{-2}$$

$p_1 = -0{,}1982$ bar

$$p_2 = 11{,}4 + \left[1 \cdot 9{,}81 \cdot 0{,}12 + 1 \cdot 9{,}81\left(0{,}05 + \frac{2{,}26^2 - 3{,}53^2}{2 \cdot 9{,}81} + 0{,}015\right)\right] \cdot 10^{-2}$$

$p_2 = 11{,}3814$ bar

The total head of the pump is calculated using the results obtained and the equation shown previously:

$$H = (z_2 - z_1) + \frac{p_2 - p_1}{\rho \cdot g} \cdot 10^2 + \frac{U_2^2 - U_1^2}{2\,g}$$

$$H = (0{,}65 - 0{,}35) + \frac{11{,}3814 - (-0{,}1982)}{1 \cdot 9{,}81} \cdot 10^2 + \frac{3{,}53^2 - 2{,}26^2}{2 \cdot 9{,}81} = 118{,}7 \text{ m}$$

The total head of the pump for Q_r = 100 m³/h equals 118,7 m.

2nd Example

Calculation of the total head of a centrifugal pump installation H_A with an open liquid surface in the inlet and outlet section of the installation, where the liquid is open to the atmosphere.

Given: Pumped media: cold water, ρ = 1,0 kg/dm³

A_{A1} = 0,35 m² A_{A2} = 0,14 m²

z_{A1} = 5 m z_{A2} = 48 m

Pressures: $p_{A1} = p_{A2} = p_{amb}$

Head loss for Q = 50 m³/h = 0,0139 m³/s

$H_{J\,A1\text{-}1}$ = 2 m $H_{J\,2\text{-}A2}$ = 8,9 m

Required: The total head of the installation H_A for Q = 50 m³/h = 0,0139 m³/s

Intermediate values:

$U_{A1} = Q/A_{A1}$ = 0,0139 : 0,35 = 0,04 m/s $U_{A2} = Q/A_{A2}$ = 0,0139 : 0,14 = 0,1 m/s

Difference in velocity head $= \dfrac{U_{A2}^2 - U_{A1}^2}{2 \cdot g} = \dfrac{0{,}1^2 - 0{,}04^2}{2 \cdot 9{,}81} = 0{,}00043$ m

Normally the total head of the installation is calculated with the equation shown previously. However in this example the pressures p_{A1} and p_{A2} are equal and therefore the pressure difference is zero; furthermore as the difference in velocity head is negligible, the calculation can be made with the following simplified equation.

$H_A = (z_{A2} - z_{A1}) + H_{J\,A1\text{-}1} + H_{J\,A2\text{-}A2}$ = (48 – 5) + 2 + 8,9 = 53,9 m

The total head of the installation for Q = 50 m³/h equals 53,9 m.

3rd Example

Calculation of the total head H_A of a boiler feed pump installation.

Given: Hot water, t = 160 °C, ρ = 0,9073 kg/dm³, p_v = 6,181 bar (absolute)

Atmospheric pressure p_{amb} = 1,011 bar (absolute)

The feed tank is under the vapour pressure of the hot water.

A_{A1} = 0,35 m² A_{A2} corresponds to DN 150

z_{A1} = 9 m z_{A2} = 14 m

$p_{A1} = p_v - p_{amb}$ = 6,181 – 1,011 = 5,17 bar p_{A2} = 73 bar

The head losses for Q = 130 m³/h = 0,0361 m³/s are

$H_{J\,A1\text{-}1}$ = 2,4 m $H_{J\,2\text{-}A2}$ = 11,3 m

Required: The total head of the installation for Q = 130 m³/h.

Intermediate values:

$U_{A1} = Q/A_1$ = 0,036 : 0,35 = 0,103 m/s

U_{A2} = (18,8/DN)² · Q = (18,8 : 150)² · 130 = 2,042 m/s

The total head of the installation is calculated with the equation shown previously:

$$H_A = (z_{A2} - z_{A1}) + \frac{p_{A2} - p_{A1}}{\rho \cdot g} \cdot 10^2 + \frac{U_{A2}^2 - U_{A1}^2}{2\,g} + H_{J\,A1\text{-}1} + H_{J\,2\text{-}A2}$$

$$= (14 - 9) + \frac{73 - 5{,}17}{0{,}9073 \cdot 9{,}81} \cdot 10^2 + \frac{2{,}042^2 - 0{,}103^2}{2 \cdot 9{,}81} + 2{,}4 + 11{,}3$$

$$H_A = 5\text{ m} + 762{,}1\text{ m} + 0{,}2\text{ m} + 2{,}4\text{ m} + 11{,}3\text{ m} = 781\text{ m}$$

The total head of the plant for Q = 130 m³/h equals 781 m.

1.4 Suction capacity *S*

The suction capacity S is the volume of gas, dependent on suction pressure, which can be handled by the pump per unit of time.

It is characteristic data for pumps handling gas or gas/liquid mixtures e.g. liquid ring vacuum pumps, side channel pumps, or centrifugal pumps with a side channel suction stage which are used for evacuation of suction lines or priming centrifugal pumps and pump installations. The suction capacity and the volume to be evacuated determine the time to achieve a required reduction in pressure.

The common unit for suction capacity is m³/h.

1.5 NPSH in centrifugal pumps

The important term ($NPSH$) used in centrifugal pumps and pump installations stands for “Net Positive Suction Head”.

($NPSH$) is defined as the nett energy level (= total energy level less the vapour pressure level) in the entry section of the pump.

$$(NPSH) = H_{t1} - z_D + \frac{p_{amb} - p_v}{\rho_1 \cdot g}$$

The ($NPSH$)-value is referred to the ($NPSH$)-datum level, whereas the energy level in the entry section of the pump is referred to the datum level of the plant.

The term cavitation is closely connected with ($NPSH$) .

Cavitation indicates the formation of vapour bubbles as a result of a localised drop in static pressure below the vapour pressure of the liquid and the subsequent implosive collapse of the bubbles at increased pressure. This process produces pressure waves with high pressure points. When vapour bubbles occur on or close to a surface, e.g. wall or impeller vanes, the implosion results in so called “Microjets” which impact the surface of the wall / impeller vane with high speed causing heavy wear. This explains the material pitting so characteristic of full cavitation.

The main cause of cavitation in centrifugal pumps is the local pressure reduction at the inlet to the impeller blade channel which is unavoidable as a result of the increase of flow velocity at the entry edge of the impellers and the transmission of energy from the impellers to the liquid stream. Cavitation can also result at other points in the pump at which local pressure drops occur e.g. entry edge of guide vanes, casing webs and spacer rings.

Further causes are either the rising temperature of the pumped media, the reduction in pressure in the entry section of the pump, the increase in the static datum suction head or the reduction in the static suction head.

The indications of cavitation in order of severity are:

a) Formation of individual vapour bubbles or areas of bubbles

   This can only be observed and evaluated by stroboscopic examination of the impeller entry in a specially built copy of the original pump design, which allows internal visibility. This costly procedure would normally only be undertaken when establishing the hydraulic design of highly stressed impellers for large condensate pumps, in order to optimise the pump and impeller entries for the ($NPSH$)-requirements.

b) Drop in the total head compared to that for cavitation free operation with the same flowrate.

The drop is given as a percentage of the total head for cavitation free flow. For multistage pumps it is given as a percentage of the total head of the first stage.

c) Drop in efficiency compared to cavitation free operation.

d) Noise or change in noise compared to cavitation free operation.

The implosion of the vapour bubbles creates a rattling noise similar to pebbles in a concrete mixer.

e) Rough operation, revealed by an increase in vibration compared to cavitation free running.

f) Material damage (wear) to internal parts of pump.

The pitting of the material surface leaves a sponge like appearance.

g) Collapse of total head

The total head of the pump drops completely as the impeller cells become blocked by the cavitation bubbles preventing any energy transmission to the pumped media.

1.5.1 (*NPSH*) required by a pump

The (*NPSH*) value required by a pump, (*NPSHR*), is the minimum amount by which the overall total energy level at the datum level for the (*NPSH*) value has to exceed the vapour pressure head of the pumped liquid, in order to guarantee correct operation of the pump without the effect of cavitation at the selected speed and selected flowrate (or selected total head) for the pumped media for which the selection was made.

(*NPSHR*) is stated by the pump manufacturer and shown as the *NPSHR* / *Q* characteristic curve.

The (*NPSHR*) is based on cavitation tests with clean, cold water, with a deduction of 3% compared to cavitation free operation. This value is referred to as (*NPSH3*). For multistage pumps the term (*NPSH3*) refers to the first stage.

For pumps which have a higher than normal energy transfer than standard pumps, e.g. boiler feed pumps or condensate pumps for large power stations, the value (*NPSHR*) is given considerably higher than (*NPSHR3*), as in these conditions the influence of vibration and material loss take effect sooner. However by selection of suitable cavitation resistant materials for the impeller it is possible to reduce the (*NPSHR*) value and/or extend the life of the impeller.

The cavitation resistance of the impeller material can be estimated from the following table.

Table 1.01 Indices of erosion wear through cavitation

Material	Example	Wear index
Cast iron	EN-JL1040 EN-GJL-250 (GG25)	1,0
Bronze and copper alloys	CC480KCuSn10-Cu	0,5
Cast chrome steel	1.4317 GX4CrNi13-4	0,2
Aluminium bronze	CC333GCuAl10Fe5Ni5-C	0,1
Cast stainless steel	1.4408 GX5CrNiMo19-11-2	0,05
Duplex cast steel	1.4517 GX2CrNiMoCuN25-6-3-3	0,02

The wear index indicates in relative terms the approximate material loss compared to cast iron, wear index 1, for the same (*NPSHA*) and pumped media.

1.5.2 Reduction of (*NPSH*) value when pumping water and hydrocarbons

Operational experience and test bed results show that pumps operate safely and with less cavitation damage when handling gas free hydrocarbons and water at higher temperatures. In these applications therefore, pumps require a lower (*NPSHA*) value than for cold water.

The Hydraulic Institute (HI) has published a chart from which the (*NPSHR*), dependent on temperature and/or vapour pressure, of various hydrocarbons can be read off.

Application of this reduction in (*NPSHR*) is dependent on exactly adhering to the entry conditions and temperatures of the selected duty. These reduced values should therefore only be considered when the purchaser or user has clearly indicated awareness of this requirement.

In accordance with the Standards of Technical Requirements for Centrifugal Pumps class I and II (DIN ISO 9905 and 5199) the use of correction factors for hydrocarbons is not permitted.

1.5.3 Datum level for (*NPSH*) value

The datum level for the (*NPSH*) value is defined as the horizontal plane which passes through the centre of the circle which is described by the outermost points of the leading edge of the impeller blades. For double entry pumps with a shaft which is not horizontal, the higher level impeller inlet is determinant. For pumps with a horizontal shaft, the (*NPSH*) datum level is generally on the shaft centreline.

For pumps with a vertical shaft or one inclined to the vertical, the impeller inlet and therefore the datum level for the (*NPSH*) value cannot be determined externally and has to be given by the manufacturer.

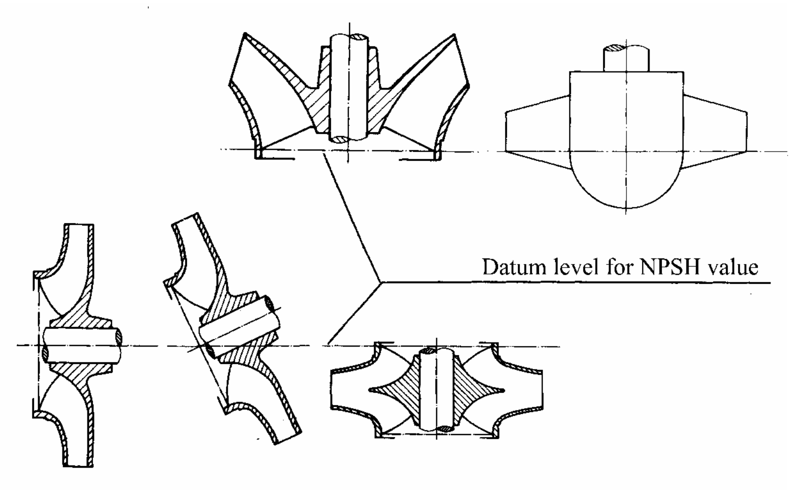


Fig. 1.02 Position of datum level for (*NPSH*) value

1.5.4 Available (*NPSH*) value of an installation (*NPSHA*)

The available (*NPSH*) value of the installation (*NPSHA*) is that which is determined for a specified pumped media and flowrate.

The (*NPSH*) value is determined as follows:

$$(NPSHA) = (z_1 - z_D) + \frac{p_1 + p_{amb} - p_v}{\rho \cdot g} + \frac{U_1^2}{2g}$$

and/or

$$(NPSHA) = (z_{A1} - z_D) + \frac{p_{A1} + p_{amb} - p_v}{\rho \cdot g} + \frac{U_{A1}^2}{2g} - H_{J\,A1\text{-}1}$$

For trouble free operation the following condition

$(NPSHA) \geq (NPSHR)$

must be satisfied.

For reasons of safety and to cover variations in operating conditions, if no other special standards or regulations apply, an additional safety margin of at least 0,5m is applied i.e.:

$(NPSHA) \geq (NPSHR) + 0{,}5$ m

If the application carries a lower (*NPSHA*) value than this, then the following options can be considered.:

- Selection of a pump with lower running speed
- Division of the flowrate between several pumps, or use pumps with double entry impellers, possibly first stage only
- Selection of a pump with a low NPSH suction impeller; for multistage pumps, first stage only
- Installation of a low speed booster pump (primary)
- Installation of a barrel type pump, whereby according to the length of the barrel, the mouth of the first stage impeller and therefore the (*NPSH*) datum level is lowered.

If for reasons associated with the application, operation of the pump in the cavitation region cannot be avoided, e.g. change over operation or overload operation, then the susceptible parts, especially the impeller, in the case of multistage pumps the first stage impeller, are selected from ductile materials which resist the erosion effects of cavitation for longer. Such materials are listed with their wear index in Table 1.01.

1.5.5 Example of the calculation of *(NPSH)* value of the installation

The following relationships are illustrated in Figs.1.03 & 1.04.

Note! If the atmospheric pressure at the installation is not clearly stated in the specification then the pressure corresponding to the altitude can be read off from Chart 13.

In all examples it is assumed that the velocity head $U_{A1}^2/2g$ is negligible.

1st Example

Suction from a closed container

Given: pumped media water, $t = 60\ °C$, $\rho = 0{,}9832\ kg/dm^3$

$p_v = 0.19920$ bar, $p_{amb} = 1{,}025$ bar, $p_{A1} = 0{,}4$ bar

$z_{A1} = -3{,}2$ m $z_D = 0{,}8$ m $H_{J\,A1\text{-}1} = 1{,}8$ m

Required: available (*NPSH*) value of the installation

$$(NPSHA) = (z_{A1} - z_D) + \frac{p_{A1} + p_{amb} - p_v}{\rho \cdot g} - H_{J\,A1\text{-}1}$$

$$(NPSHA) = (-3{,}2 - 0{,}8) + \frac{0{,}4 + 1{,}025 - 0{,}19920}{0{,}9832 \cdot 9{,}81} \cdot 10^2 - 1{,}8 = 6{,}9 \text{ m}$$

The available (*NPSH*) value of the installation (*NPSHA*) equals 6,9 m.

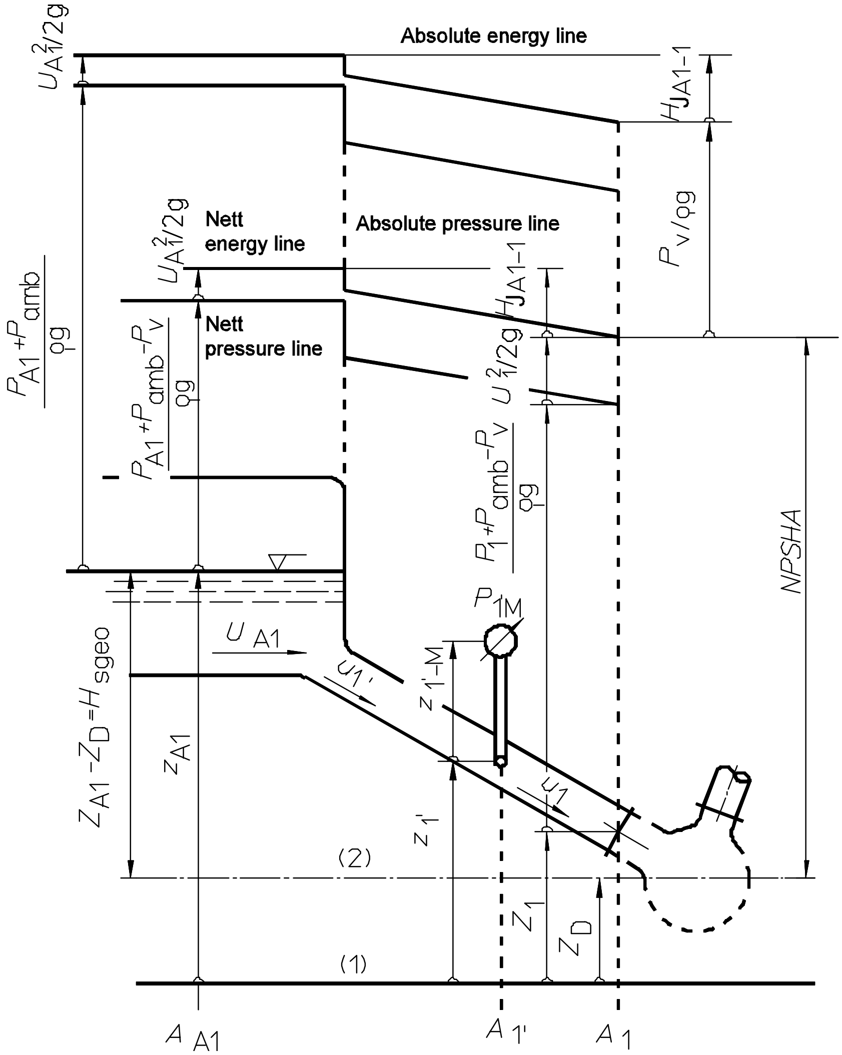


Fig. 1.03 Centrifugal pump installation, fed suction

(1)=Datum level of installation	<u>Indices</u>	1 =Entry side of pump
(2)=(NPSH) datum level	A1=Entry side of plant	1'=Entry side measurement tapping

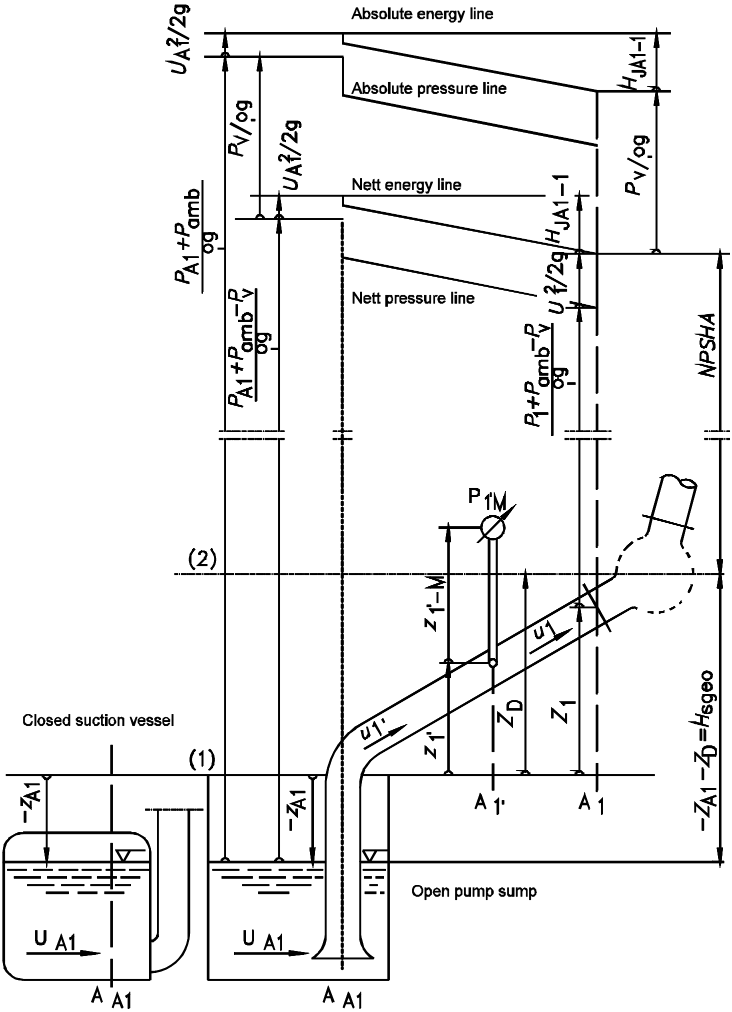


Fig. 1.04 Centrifugal pump installation, suction operation

(1)=Datum level of installation	<u>Indices</u>	1 =Entry side of pump
(2)=(NPSH) datum level	A1=Entry side of plant	1'=Entry side measurement tapping

2nd Example

Suction from an open container

In this case $p_{A1} = 0$ applies, i.e. only atmospheric pressure p_{amb} acts on the liquid level.

Given: pumped media water, $t = 40$ °C, $\rho = 0{,}9923$ kg/dm³

$p_v = 0{,}07375$ bar, $p_{amb} = 1{,}016$ bar

$(NPSHR) = 2{,}9$ m

$(NPSHA) = (NPSHR) + 0{,}5$ m safety addition $= 3{,}4$ m

$H_{J\,A1\text{-}1} = 2{,}7$ m

Required: the highest possible static suction head $H_{s\,geo}$

from $$(NPSHA) = (z_{A1} - z_D) + \frac{p_{amb} - p_v}{\rho \cdot g} - H_{J\,A1\text{-}1}$$

gives $$(z_{A1} - z_D) = (NPSHA) - \frac{p_{amb} - p_v}{\rho \cdot g} + H_{J\,A1\text{-}1}$$

this gives the highest possible static suction head as $H_{s\,geo}$:

$$(z_{A1} - z_D) = 3{,}4 - \frac{1{,}016 - 0.07375}{0{,}9923 \quad 9{,}81} \cdot 10^2 + 2{,}7 = -\,3{,}58 \text{ m}$$

The value is **negative**, therefore the pump can be installed **above** the liquid level in the suction container. The highest possible suction head is $H_{s\,geo} = 3{,}58$ m.

If the pump is to be installed 2000 m above sea level, with all other operating conditions unchanged, the maximum static suction head is given using $p_{amb} = 0{,}795$ bar (see chart 13.01) as:

$$(z_{A1} - z_D) = 3{,}4 - \frac{0{,}795 - 0{,}07375}{0{,}9923 \cdot 9{,}81} \cdot 10^2 + 2{,}7 - 1{,}31 \text{ m}$$

The maximum static suction head $H_{s\,geo}$ at an altitude 2000 m above sea level is only 1,31 m. This demonstrates the large influence of the atmospheric pressure on the static suction head.

3[rd] Example

Feed from a closed container

Given: pumped media water, $t = 140$ °C, $\rho = 0{,}9258$ kg/dm³

$p_v = 3{,}614$ bar, $p_{amb} = 0{,}996$ bar, $p_{A1} = 3{,}0$ bar

$(z_{A1} - z_D) = H_{z\ geo} = 16$ m $\qquad H_{J\ A1\text{-}1} = 15$ m

Required: the available (*NPSH*) value of the installation

$$(NPSHA) = (z_{A1} - z_D) + \frac{p_{A1} + p_{amb} - p_v}{\rho \cdot g} - H_{J\ A1\text{-}1}$$

gives:

$$(NPSHA) = 16 + \frac{3 + 0{,}996 - 3{,}614}{0{,}9258 \cdot 9{,}81} \cdot 10^2 - 15 = 5{,}21 \text{ m}$$

4[th] Example

Feed from a closed container

Given: Pumped media water $t = 160$ °C, $\rho = 0{,}9073$ k/dm³

$p_v = 6{,}181$ bar, $p_{amb} = 1{,}013$ bar, $p_{A1} = 5{,}4$ bar

(*NPSHR*) = 4 m (including safety addition), $H_{J\ A1\text{-}1} = 2$m

Required: the minimum necessary static feed head $H_{z\ geo}$

from $$(NPSHA) = (z_{A1} - z_D) + \frac{p_{A1} + p_{amb} - p_v}{\rho \cdot g} - H_{J\ A1\text{-}1}$$

gives $$(z_{A1} - z_D) = (NPSHA) - \frac{p_{A1} + p_{amb} - p_v}{\rho \cdot g} + H_{J\ A1\text{-}1}$$

In place of (*NPSHA*) in this case (*NPSHR*) is used

from this $$(z_{A1} - z_D) = 4 - \frac{5{,}4 + 1{,}013 - 6{,}181}{0{,}9073 \cdot 9{,}81} \cdot 10^2 + 2 = 3{,}39 \text{ m}$$

The value is **positive**, therefore the pump must be sited **below** the liquid level in the container.

The minimum necessary static feed head is $H_{z\ geo} = 3{,}39$ m.

5^{th} Example

Feed from a closed container under the influence of vapour pressure of the pumped media (saturated mixture).

In this special case, $p_{A1} + p_{amb} = p_v$ and the equation for available (*NPSH*) value of the installation simplifies to:

$$(NPSHA) = (z_{A1} - z_D) - H_{J\ A1\text{-}1}$$

and $(z_{A1} - z_D) = (NPSHA) + H_{J\ A1\text{-}1} + \text{safety addition}$

Given: $(NPSHR) = 1{,}3$ m, safety addition = 0,5 m, $H_{J\ A1\text{-}1} = 0{,}2$ m

Required: minimum necessary static feed head $H_{z\ geo}$

(*NPSHA*) in this case is equal to (*NPSHR*) and therefore gives:

$$(z_{A1} - z_D) = 1{,}3\ \text{m} + 0{,}2\ \text{m} + 0{,}5\ \text{m} = 2{,}0\ \text{m}$$

The value is **positive**, therefore the pump must be sited **below** the liquid level in the container.

The minimum necessary static feed head is $H_{z\ geo}$ = 2,0 m.

1.6 Specific Energy

Specific Energy y is defined in terms of the mass of the pumped media.

It is calculated from the equation

$$y = H \cdot g$$

The unit is

J/kg = N m/kg = W/kg s = m²/s²

Comparison between energy level and specific energy.

Energy level	Notation	Specific Energy	Notation
Total head of pump	H	Of the pump	y
Total head of installation	H_A	Of the installation	y_A
Altitude	z	Of the static head	$z \cdot g$
Pressure head	$p/\rho \cdot g$	Of the pressure	p/ρ
Velocity head	$U^2/2\ g$	Of the speed	$U^2/2$
Head loss	H_J	Of the energy level losses	y_J

1.7 Power, efficiency

1.7.1 The hydraulic power output P_u is the usable power transferred by the pump to the pumped media:

$$P_u = q \cdot g \cdot H = \rho \cdot Q \cdot g \cdot H$$

whereby ρ is the density of the pumped media. If there is a noticeable change in density of the pumped media during the passage through the pump, then the density in the entry section ρ_1 is used.

The SI unit for power is the Watt (W). It is customary to express power in kW. It is determined from the following equation:

$$P_u = \frac{\rho \cdot Q \cdot H}{367} \text{ in kW} \quad \text{with } \rho \text{ in kg/dm}^3,\ Q \text{ in m}^3\text{/h and } H \text{ in m}$$

1.7.2 The absorbed power *P* (power input) is the power input at the coupling or pump shaft. It is higher than the hydraulic power P_u output by the amount of the pump losses.

If the efficiency η of the pump, which reflects the pump losses, is known, the absorbed power of the pump can be calculated from the following equation:

$$P = \frac{P_u}{\eta} = \frac{\rho \cdot Q \cdot H}{367 \cdot \eta} \text{ in kW}$$

with ρ in kg/dm³, Q in m³/h and H in m

η is expressed as a decimal value.

Example:

given: $Q = 50$ m³/h, $H = 54$ m, $\rho = 1{,}0$ kg/dm³, $\eta = 70\ \% = 0{,}70$

$$P = \frac{1{,}0 \cdot 50 \cdot 54}{367 \cdot 0{,}70} = 10{,}5 \text{ kW}$$

The power figures are defined as follows:

Notation	Terminology	Definition
P_u	Pump output power	The usable power transferred to the pumped media by the pump *Note* Also known as pump hydraulic power

P	Absorbed power of pump	The power input at the pump coupling or shaft
P_r	Absorbed power for selected duty	The necessary pump power to meet the selection duty
$P_{J\,m}$	Mechanical power loss of the pump	The power absorbed in the pump to overcome friction losses in bearings and seals

1.7.3 The pump efficiency η is the ratio of the pump hydraulic power output to the absorbed power at the pump coupling or shaft at the operating point.

The pump efficiency is given by the following equation:

$$\eta = \frac{P_u}{P} = \frac{\rho \cdot Q \cdot g \cdot H}{P}$$

and with unit factors:

$$\eta = \frac{\rho \cdot Q \cdot H}{367 \cdot P}$$ with P in kW, ρ in kg/dm³, Q in m³/h and H in m

Example:

given: $Q = 200$ m³/h, $H = 90$ m, $\rho = 1{,}0$ kg/dm³, $P = 64{,}5$ kW

$$\eta = \frac{1{,}0 \cdot 200 \cdot 90}{367 \cdot 64{,}5} = 0{,}76 = 76\ \%$$

The efficiency values are defined as follows:

Notation	Terminology	Definition
η	Pump efficiency	The ratio of the pump output power P_u to the absorbed power P at a particular operating point $\eta = P_u / P$
η_{opt} also η_{BEP} η_{max}	Optimum efficiency	The highest value of pump efficiency for a particular pumped media and pump operating speed

η_m	Mechanical efficiency	The ratio of the absorbed power P of the pump, reduced by the mechanical power loss $P_{J\,m}$ to the absorbed power for a selected operating duty. $\eta_m = (P - P_{J\,m}) / P$ *Note* Depending on the size and performance of the pump, η_m lies between 0,994 and 0,96
η_{int}	Hydraulic efficiency	The ratio of the hydraulic power output P_u to the absorbed power P of the pump, reduced by the mechanical power loss $P_{J\,m}$ $\eta_{int} = P_u / (P - P_{J\,m})$

1.7.4 The installed power P_M is the rated power P_N of the driver. The installed power should be adequate for the entire specified operating range.

Furthermore when assessing the required installed power, certain power additions must be considered. The unavoidable deviations of the actual conditions from the design data of the pump installation and the pumped media, as well as extra power losses e.g. through the shaft seals, material wear etc. need to be included.

If no extreme conditions apply and no special standards or specifications have been laid down, then in practice the following power additions should be included.

a) Power addition for side channel pumps

for P	< 1,5 kW	25%	$P_M \approx 1{,}25 \cdot P$
	1,5 to 4 kW	20%	$P_M \approx 1{,}2 \cdot P$
	> 4 kW	10%	$P_M \approx 1{,}1 \cdot P$

b) Power addition for centrifugal pumps with radial flow impellers

for P	< 1,5 kW	50%	$P_M \approx 1{,}5 \cdot P$
	1,5 to 4 kW	25%	$P_M \approx 1{,}25 \cdot P$
	4 to 7,5 kW	20%	$P_M \approx 1{,}2 \cdot P$
	7,5 bis 40 kW	15%	$P_M \approx 1{,}15 \cdot P$
	> 40 kW	10%	$P_M \approx 1{,}1 \cdot P$

c) Power addition for mixed flow and axial flow pumps

Power additions for this type of pump are especially influenced by the shape of the pump power input curve and are therefore established for each point and operating range.

d) Power addition for pumps with absorbed power > 100 kW

In these cases the additional power must be carefully calculated to avoid over sizing the driver. The efficiency plays a major role and the selection should be matched to the duty data as closely as possible.

1.8 Speed and direction of rotation

1.8.1 Speed

The speed n is is the ratio of number of revolutions per unit of time. For a symmetrical body, e.g. impeller, the unit is 1/s, but the customary unit in English speaking countries is rpm (revolutions per minute).:

Notation	Terminology	Definition
n	Speed	The number of revolutions per unit of time for a uniform rotating body
n_r	Selected speed	The required speed of a pump to meet the duty specification
$n_{max\ all}$	Maximum permitted speed	The maximum speed at which a pump can operate in accordance with the selection data and installation conditions
$n_{min\ all}$	Minimum permitted speed	The minimum speed at which a pump can operate in accordance with the selection data and installation conditions

1.8.2 Direction of rotation of pump

The direction of rotation of the pump refers to the rotation of the impeller/ impellers for which the pump has been designed or selected.

a) Clockwise rotation (sometimes described as direction of rotation right)

   Clockwise rotation of the pump shaft when viewed on the drive end.

   Commonly designated 'cw'.

b) Counterclockwise or anticlockwise rotation (sometimes described as direction of rotation left)

   Counterclockwise rotation of the pump shaft when viewed on the drive end.

   Commonly designated 'ccw'.

For pumps which can be driven from either end, then the direction of rotation is given by means of a sketch.

In all cases pumps are fitted with a direction arrow which enables identification of the direction of rotation from outside the pump and when the pump is stopped.

1.9 Specific speed and characteristic type number

1.9.1 Specific speed n_s

A given operating point with flowrate Q and total head H can be achieved by centrifugal pumps with differently shaped impellers depending on rotational speed. The specific speed n_s serves as the characteristic value for the impeller shape. This term is defined as the rotational speed of an impeller which is geometrically similar in all aspects and which is so dimensioned that for a total head H_s of 1m, the pump delivers a flowrate Q_s of 1 m³/s.

From the pump affinity laws, the following applies:

$$n_s = n \cdot \frac{(Q / Q_s)^{0,5}}{(H / H_s)^{0,75}}$$

in 1/min with n in 1/min, Q in m³/s and H in m

The specific speed refers to the operating data of an impeller of maximum diameter at the point of optimum efficiency. For multistage pumps this refers to the data of the first stage only, whereas it should be noted that different impellers can be fitted in the first stage, e.g. NPSH impellers. For double entry impellers, the data for one side only is used.

The above equations can be simplified with Q_s = 1 m³/s and H_s = 1 m to the following:

$$n_s = n \cdot \frac{Q_{opt}^{0,5}}{H_{opt}^{0,75}} \text{ in 1/min} \quad \text{and/or.} \quad n_s = \frac{n}{60} \cdot \frac{Q_{opt}^{0,5}}{H_{opt}^{0,75}} \text{ in 1/min}$$

with:		and/or.	
	n in 1/min		n in 1/min
	Q_{opt} in m³/s		Q_{opt} in m³/h
	H_{opt} in m		H_{opt} in m

Note: For the specific speed n_s the notation n_q was commonly used.

1.9.2 Characteristic type number *K*

The characteristic type number is given the notation *K*.

The characteristic number *K*, valid for the same conditions as the specific speed is defined by the following equation.

$$K = 2 \cdot \pi \cdot n \cdot \frac{Q_{opt}^{0,5}}{(g \cdot H_{opt})^{0,75}} \quad \text{with } n \text{ in 1/s, } Q_{opt} \text{ in m³/s, } g \text{ in m/s², } H_{opt} \text{ in m}$$

The conversion factor from K to n_s is as follows:

$$\{K\} = \frac{\{n_s\}}{52{,}919}$$

1.9.3 Specific suction number n_{ss}

The specific suction number n_{ss} is based on the specific speed. Instead of the total head H_{opt}, the value (*NPSH3*) as in Q_{opt} is used.

The specific suction number n_{ss} is a characteristic number for the suction ability and the (*NPSH*) status of a centrifugal pump and is defined in the following equation:

$$n_{qs} = n \cdot \frac{Q_{opt}^{0,5}}{(NPSH3)^{0,75}} \text{ in 1/min} \quad \text{and/or } n_{qs} = \frac{n}{60} \cdot \frac{Q_{opt}^{0,5}}{(NPSH3)^{0,75}} \text{ in 1/min}$$

with:		and/or.	
	n in 1/min		n in 1/min
	Q_{opt} in m³/s		Q_{opt} in m³/h
	(*NPSH3)* in m		(*NPSH3)* in m

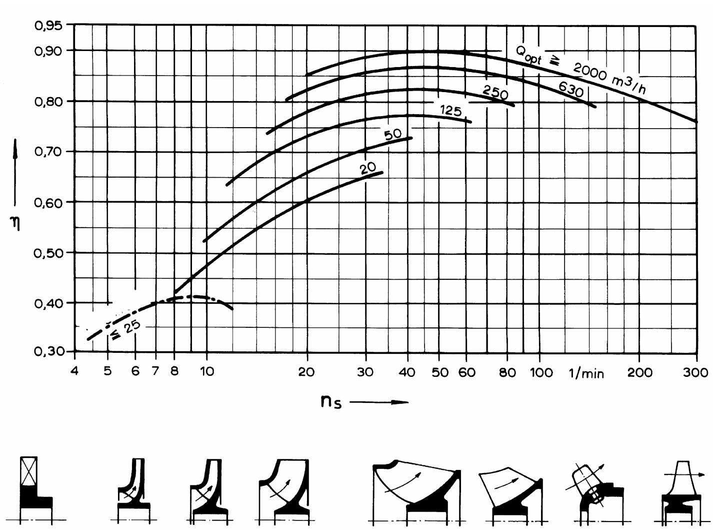
The value of n_{ss} generally lies at 130 1/min for a standard impeller and at 240 1/min for a NPSH (suction stage) impeller. In some cases, values lower than 130 1/min down to 60 1/min arise. In extreme cases for NPSH impellers, it can be up to 380 1/min.

The characteristic value for a pump is dependent on the design of the impeller, the speed, the flowrate, and the acceptable cavitation level.

1.9.4 Specific speed, shape of impeller, efficiency

The specific speed and the shape of the impeller have a substantial influence on the efficiency of the pump. Fig.1.05 shows the relationship between specific speed of various shapes of impellers and the efficiency of the pump.

It should be borne in mind that the absolute size of the pump, determined by the flowrate Q, also has an influence.



Vane wheel impeller

Vane wheel impellers n_s=4 to 12 rpm	Radial flow impellers n_s=8 to 45 rpm	Mixed flow impellers n_s=40 to 160 rpm	Axial flow impellers n_s=100 to 300 rpm

Fig. 1.05 Specific speed, shape of impeller, efficiency

1.10 Pump selection

The selection of a pump is based on the given operating data as specified by the process designer or operator.

The following data are required as a minimum:

- Pumped media

  The media must be completely specified. The composition of a mixture should be clearly given. When applicable, details such as solids content, corrosive and erosive components, non-dissolved gases and dangerous substances e.g. inflammable, poisonous, irritant etc. must be given.

- Operating temperature

  The operating temperature is required for the duty selection. If applicable the maximum and minimum temperatures should be given.

- Physical properties of the pumped media

  For liquids and mixtures which are not common, the temperature dependent properties: density, viscosity and vapour pressure must be given.

- Operating data

  Operating data includes flowrate, inlet and discharge pressures, or total head (*NPSH*) or available head (*NPSHA*) values. If these values are subject to operational variations due to installation conditions, then minimum and maximum values must be specified.

- Standards

  Any standards or special requirements should be included in the enquiry specification.

The enquiry is considerably simplified if a centrifugal pump enquiry sheet complying with standard EN 25 199, DIN ISO 5199 or DIN ISO 9905 is available. This data sheet when complete, contains all the necessary information for the pump selection. The essential data which is required is indicated on the form by a black triangle.

To ensure the selection of the most reliable and efficient pump for the application, as much of this data sheet as possible should be completed with the enquiry.

1.11 Design of the installation and life cycle costs

An important step in the design of an installation is the estimation and optimisation of the life cycle costs of the plant.

The life cycle costs comprise:

- Capital costs
- Operating costs
- Maintenance costs
- Repair costs
- Disposal costs

1.11.1 Capital costs

Capital costs principally cover the cost of the pump unit. Dependent on the operating conditions these costs reflect the size of pump, the materials of construction, the driver specification and any accessories.

To optimise capital costs, the following tips should be considered in the design:

Size of pump	
Flowrate *Q*	Without additional safety.
Total head *H*	Establish the static head H_{stat} and the dynamic head H_{dyn} as exactly as possible. Avoid excessive safety additions.
Speed *n*	Select electric motors with the minimum number of poles and direct drive if possible. For high pressure pumps and pumps with pressure differences exceeding 100 bar, speeds > 3000 or 3600 rpm can often be more economic than large pump size or high number of stages. The selected speed is governed by the mechanical limits of the pump, the shaft seals and the driver. Furthermore the speed can be limited by the (*NPSHA*) value.
Construction of pump	
Material in contact with pumped media	The materials must satisfy the requirements of corrosion, abrasion and erosion resistance and suit the operating pressure and temperature. If there is a choice of suitable materials, an estimate of whether the better and consequently, more expensive material may reduce the maintenance and repair costs.
Shaft seals	The properties and conditions of the pumped media determine the type of seals, unless sealless pumps are selected. Environmental considerations also play a major roll. If there is a choice of suitable solutions, then the life cycle costs of each variant should be assessed.

Size of driver	Assess safety margins in accordance with relevant standards. Only exceed these additional safety margins in exceptional cases after careful examination of operating conditions.
Accessories	
Coupling and base-plate	Wherever possible monobloc construction, in-line pumps and vertical multistage pumps should be selected, however performance limits must be observed.

1.11.2 Operating costs

The largest part of the operating costs are the energy costs. Alongside the correct selection of the pump and driver, (see 1.11.1) the efficiencies of the pump and driver play a major role.

Wherever possible, pumps should be operated in the range of optimum efficiency. For standard pumps with a fixed performance field, certain concessions must be made on efficiency.

Electric motors should be operated within the range of nominal power. If the motor is oversized then the efficiency is reduced and therefore operating costs are increased.

If any adjustment is required to meet changing or varying operating conditions, this should involve as little loss as possible. Chapter 2.2 describes the options.

If there are large changes in the operating points then it is worthwhile estimating the probable operating hours at each point and the energy requirement for each variant. It can for instance be of advantage to divide the total flow of the plant between a number of pumps, possibly of different sizes, each of which are optimised for a particular operating point. This avoids having to strongly throttle or partially load a pump which has been selected for maximum duty.

1.11.3 Maintenance costs

Regular monitoring and maintenance of a pump installation is a necessary requirement for trouble free operation and long operational life.

Experience shows that regular maintenance is a lower cost option than emergency repairs in the event of failure.

1.11.4 Repair costs

If effective routine maintenance is carried out on the pump installation then, as a rule, major repair costs should not arise. Replacement and repair of wearing parts is to be expected and if carried out by skilled personnel can extend the operational life significantly.

When considering the overall economics and efficiency, the down time and loss of production must also be included in the cost calculation.

In the event of excessive repair costs it is advisable to check if the current operating conditions are still the same as those specified for the pump selection.

1.11.5 Recycling costs

Recycling costs can occur for instance when the repair of a pump is no longer economic, or if a plant is taken out of operation and the pump cannot be used elsewhere. Metal components which have not been heavily contaminated can be recycled. Plastics, elastomers and lubricating materials must be disposed of in an environmentally correct manner, which can incur costs. It is therefore advisable when purchasing a pump installation to examine the recycling and environmental aspects of the design.

1.11.6 Calculation of Life Cycle Costs

The Confederation of Pump Manufacturers in the USA (HI = Hydraulics Institute) and Europe (Europump) introduced a specification for establishing and reducing Life Cycle Costs (LCC) in 1999. Interested readers should consult the draft or latest issues of this specification to include in the design of the installation.

2 Operational performance of centrifugal pumps

2.1 Characteristic curves

2.1.1 Pump characteristic curves

For a centrifugal pump driven at constant rotational speed, the head *H*, the power absorbed *P* and hence the efficiency η - as well as the (*NPSHR*), are functions of the flowrate *Q*. The relationship between these different values is represented by characteristic curves. Fig. 2.01 shows an example of four characteristic curves of a single stage centrifugal pump at a rotational speed n = 1450 rpm.

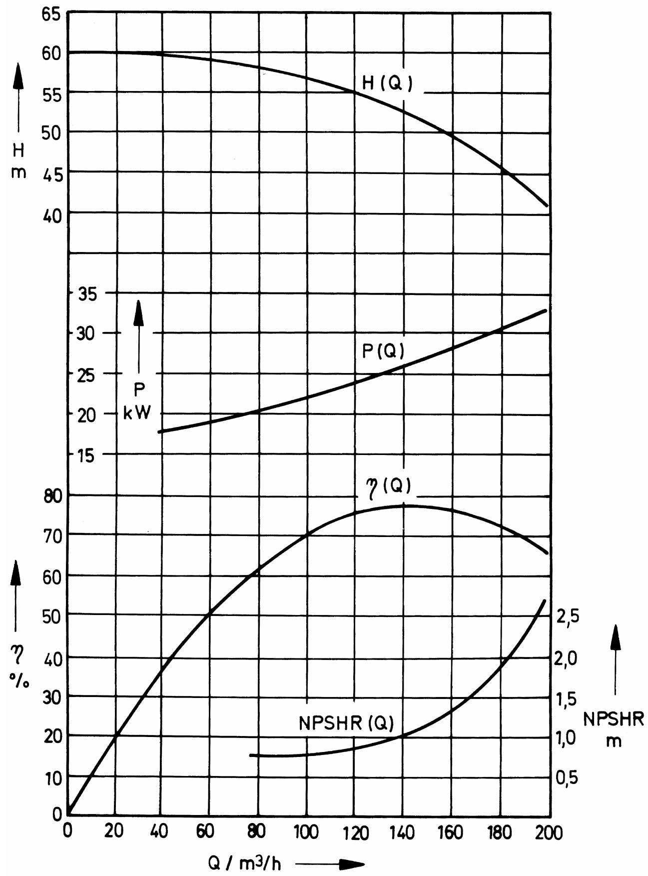


Fig. 2.01 Characteristic curves of a single stage centrifugal pump.

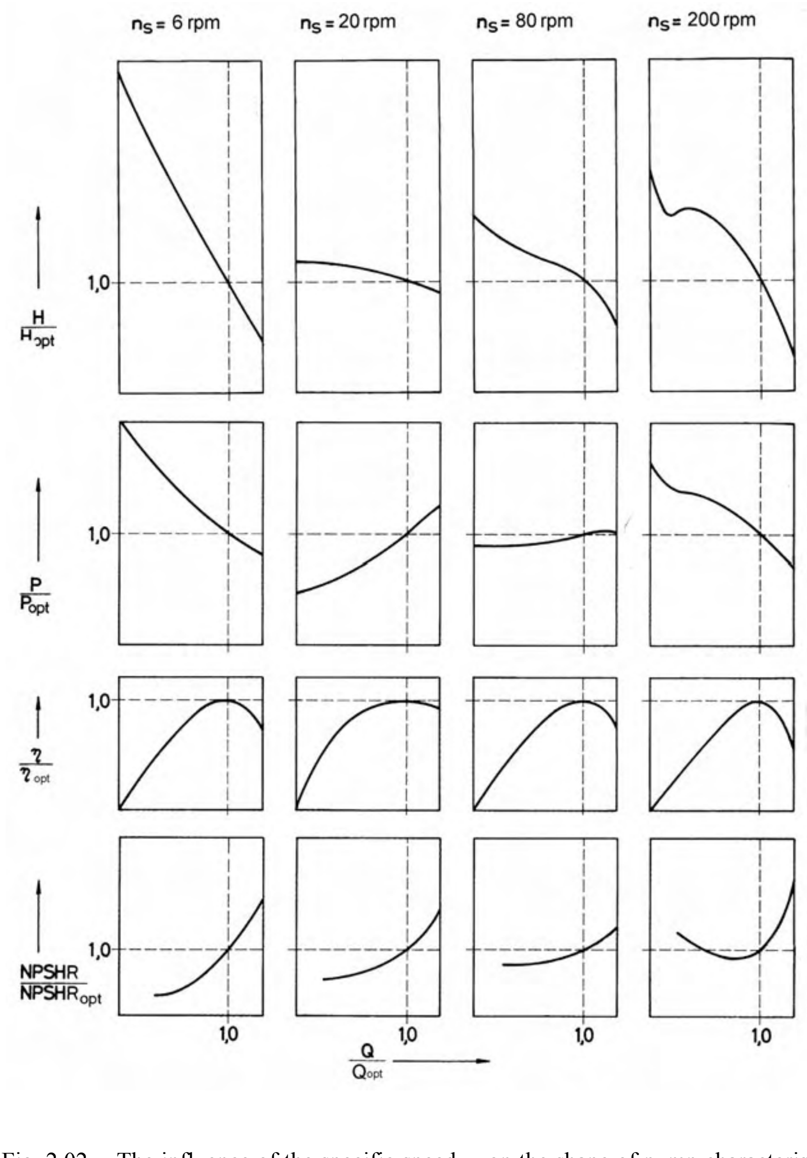


Fig. 2.02 The influence of the specific speed n_s on the shape of pump characteristic curves. (Curves displayed in relative terms)

The **Head / Capacity curve *H(Q)*** also known as the **throttling curve** - represents the relationship between the head of a centrifugal pump and its flowrate.

Generally the head decreases with increasing flowrate. The measure of this fall in head is given in practice by the ratio:

$$\frac{H_0 - H_{opt}}{H_{opt}}$$

this is often termed the 'steepness'.

This steepness ratio is a function of the specific speed and its values are:

for side channel pumps	approx.	1 to 3
for radial flow pumps	approx.	0,10 to 0,25
for mixed flow pumps	approx.	0,25 to 0,80
for axial flow pumps	over	0,80

The steepness ratio is therefore dependent on the type of pump and the shape of the impeller and cannot be selected arbitrarily.

H(Q) curves in which the head decreases with increasing flowrate are described as stable. (Fig. 2.03a, b and c). With stable H(Q) curves, for each value of head there exists only a single value of flowrate.

In contrast, an unstable H(Q) curve is one in which there is a range of flowrates for which the head increases with increasing flowrate. (Fig. 2.03d and e). With an unstable H(Q) curve there may be two or more values of flowrate for each value of head. The peak values on an unstable curve are known as Q_{sch} and H_{sch}. The shape of the characteristic curve in Fig. 2.03e is typical for pumps with high specific speeds, above $n_s \approx 110$ rpm.

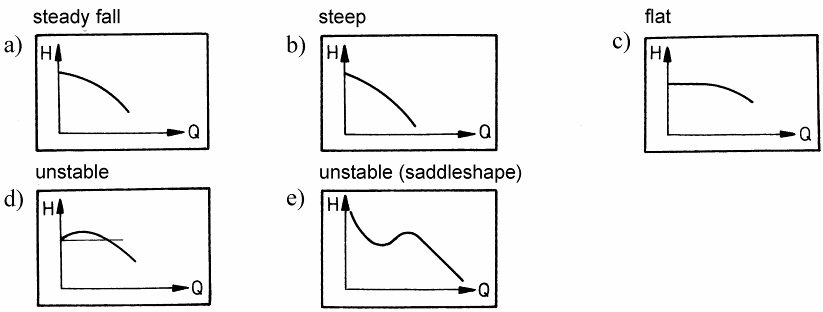


Fig. 2.03

Typical characteristic H(Q) curves for centrifugal pumps.

The shape of the **absorbed power curve *P*(*Q*)** of a centrifugal pump is also a function of the specific speed (see Fig. 2.02).

For side channel pumps, the maximum power absorbed occurs at $Q = 0$. With radial flow pumps the power absorbed increases with increasing flowrate. The maximum power absorbed by mixed flow pumps occurs at approx. Q_{opt} and it falls again at higher flowrates. Axial flow pumps have the maximum power absorbed at $Q = 0$ and this power falls with increasing flowrate.

Consequently, radial flow pumps are generally started with a closed discharge valve, whereas side channel pumps and axial flow pumps are started with the discharge open, to prevent overload of the driver during start up.

The **efficiency curve η(*Q*)** increases from zero with increasing flowrate to a maximum (η_{opt}) and falls as the flowrate increases further. Unless other considerations are the determining parameter for the pump selection, then the pump with optimum efficiency η_{opt} which is as close as possible to the required flowrate Q_r is selected, i.e. $Q_r \approx Q_{opt}$.

The shape of the **necessary - required value of (*NPSH*)-(*NPSHR*) (*Q*) curve** is also largely dependent on the specific speed (see Fig. 2.02).

2.1.2 System characteristic curve

The characteristic curve of the system ***H*$_A$(*Q*)** - also known as pipeline or installation head curve - is the total head requirement as a function of the flowrate. As shown in Fig.1.3.10, the total head of the plant is generally the sum of one component which is independent of the flowrate, the static head

$$H_{stat} = (z_{A2} - z_{A1}) + \frac{p_{A2} - p_{A1}}{\rho \cdot g}$$

and a component which increases with the square of the flowrate, the dynamic head.

$$H_{dyn} = \frac{U_{A2}{}^2 - U_{A1}{}^2}{2 \cdot g} + H_{Jt}$$

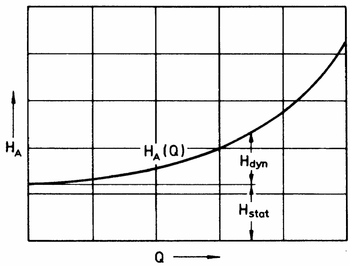


Fig. 2.04 System head curve
(Installation / pipework curve)

In special cases the static head may be zero e.g. in recirculating systems

2.1.3 Interaction between the pump and system

The duty point of a pump is where the total head generated by the pump is equal to the head required by the installation. In other words where the pump characteristic $H(Q)$ and the system characteristic $H_A(Q)$ intersect.

This gives the flowrate Q, which the pump delivers through the system and the corresponding values of absorbed power P, the efficiency η and the necessary (*NPSH*)-value of the pump (*NPSHR*). An important precondition for the operation at the duty point is the requirement laid down in chapter 1.5.4, that the available (*NPSH*)-value of the installation (*NPSHA*) exceeds the necessary (*NPSH*)-value of the pump (*NPSHR*) by at least the safety margin.

As a rule the selection of the pump is determined by the flowrate required; the total head of the system (= head of the pump) is then calculated to suit the specified operating conditions.

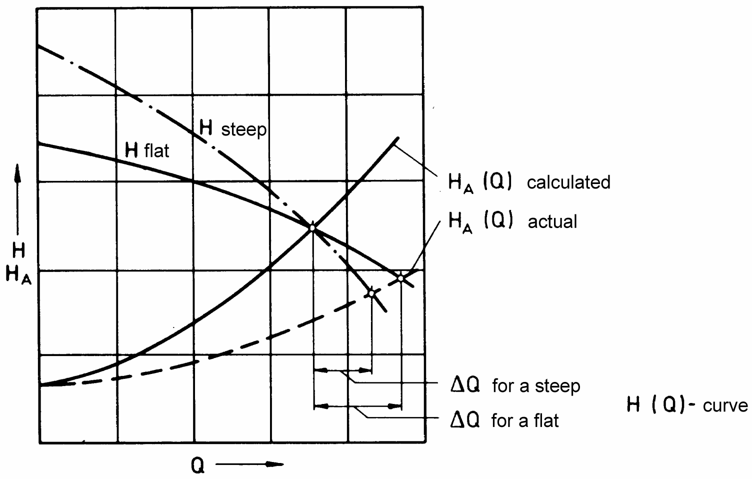


Fig. 2.05 Variation in flowrate due to deviation of the actual $H_A(Q)$ curve from the calculated curve.

When this calculation is based on some assumptions (e.g. assumptions about the surface roughness when calculating head losses in pipelines) and if large safety factors have to be applied, then the effective duty point may differ from the calculated value, see Fig. 2.05.

For steep $H_A(Q)$ curves, the variation in flowrates is less than for flat curves. Conversely, flat $H_A(Q)$ curves have certain advantages over steep ones if the duty point is modified by throttling the delivery pipeline, see chapter 2.2.1.1.

Centrifugal pumps with unstable head curves will operate satisfactorily in systems with ‘fixed’ characteristics at constant rotational speed, provided that the shut off head H_0 is greater than the static component of the system head H_{stat}. In such cases, as in the case of pumps with stable head curves, there is only one intersection of the pump head curve and the system head curve, see (Fig. 2.06, characteristic H_{AI} and H_{AII}).

The unstable shape of the $H_A(Q)$ curves must however be considered if there is an elastic component e.g. a pressure vessel in the system, i.e. if the static component of the head varies. If for example the static head H_{stat} increases, then the characteristic curve changes as shown in Fig. 2.06 from H_{AI} to H_{AII}, H_{AIII} etc. The flowrate reduces until at Q_{IV} the flow ceases suddenly. If operating against a closed discharge non-return valve, (Q = 0) the pump continues to run with a zero flow head H_0. If the static head H_{stat} falls below H_0, the operation resumes with the flowrate corresponding to the intersection of $H(Q)$, $H_A(Q)$.

This process continues at shorter or longer intervals corresponding to the variations in H_{stat} and in certain conditions can lead to variations of the flowrate as well as undesirable pressure surges.

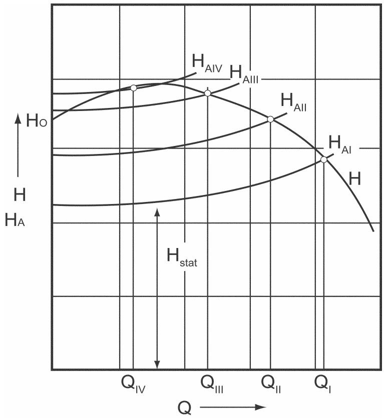


Fig. 2.06 Interaction between unstable $H_A(Q)$ curves and variable plant heads

Where pumps with unstable head curves are to operate in parallel (see chapter 2.4.2 and 2.4.3) or with variable speed drives, the operating conditions must be carefully examined.

2.2 Adapting to various operating conditions

If the installation requires other flowrates, see chapter 2.1.3, there are many possible ways of adapting the operating point to suit. These possibilities may be either changing the system characteristic curve $H_A(Q)$, e.g.

- by throttling (control valve)
- by a bypass (bypass valve)

On changes to the pump characteristic curve *H*(*Q*), e.g.

- by changing the impeller diameter
- by undercutting the angle of the blade tips
- by fitting dummy stages
- by speed variation
  (speed control / adjustment)
- by adjusting the angle of inlet guide vanes upstream of the impeller (pre-swirl)
- by creating pre-swirl immediately upstream of the impeller by means of a directional bypass flow (bypass pre-swirl)
- by adjusting the angle of incidence of the blades (pitch adjustment)

2.2.1 Adjusting the system characteristic

2.2.1.1 Throttling the system by a discharge control valve increases the flow resistance and hence the system head loss and the dynamic head loss H_{dyn}.

The steepness of the system characteristic curve increases so that the intersection with the characteristic curve of the pump occurs at a lower flowrate.

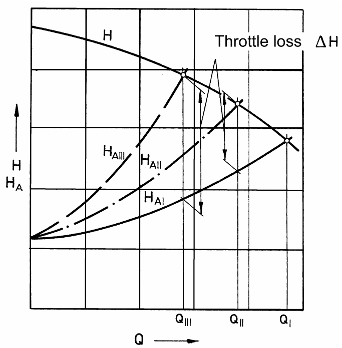


Fig. 2.07 Adjusting the flowrate by throttling

The throttling valve causes energy losses so that continuous operation with a control valve is inefficient. The throttling losses are at a minimum when the pump characteristic *H*(*Q*) is flat (Fig. 2.08). For this reason throttling control is mainly used for radial pumps, as with these pumps the absorbed power reduces along with reducing flowrate, (see Fig. 2.02). Even where throttling control appears an attractive method from the aspect of initial investment costs of the control system, the total economics should be examined, especially in cases of high installed power.

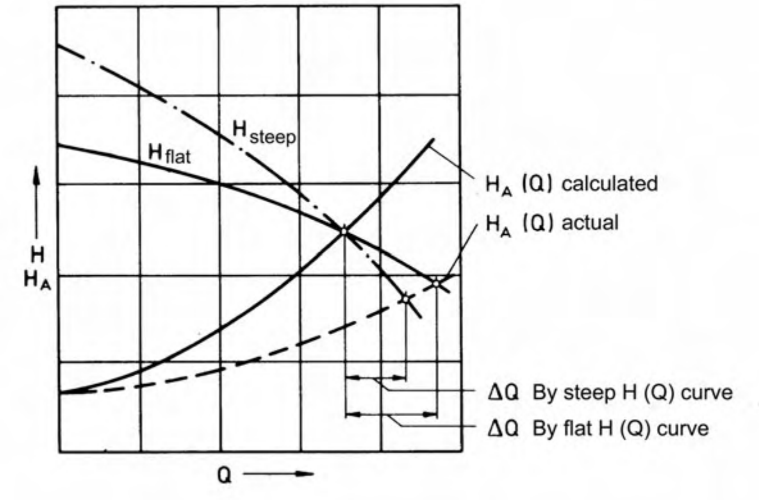


Fig. 2.08 Throttling losses for steep and flat *H*(*Q*) curves

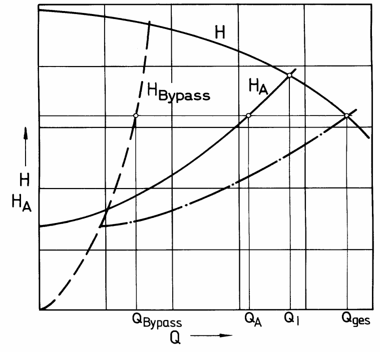
For side channel pumps, mixed flow pumps from $n_s \approx 100$ rpm and for axial flow pumps, it should be remembered that the power absorbed increases with decreasing flowrate. In addition axial flow pumps may move into a range of unstable operation due to the throttling process. This means rough running and an increased noise level, both being features of pumps with high specific speed, so this range of operation is to be avoided for continuous operation.

Fundamentally, throttling should always be done on the delivery side of the pump. Throttling the suction side of the pump will cause a reduction in available (*NPSH*) value of the system (*NPSHA*), (see chapter 1.5.4) and must be avoided due to cavitation danger.

2.2.1.2 With **Bypass** control a recirculation line, parallel to the pump, returns part of the flow from the delivery side of the pump to the suction side. Dependent on the characteristic of the bypass curve, the characteristic curve of the system increases to a larger flowrate.

$Q_{ges} = Q_{Bypass} + Q_A$ (Fig. 2.09)

As a result the flowrate of the pump increases from Q_I to Q_{ges}, and the net useful flow through the system decreases from Q_I to Q_A.



In the case of a high bypass flow, to prevent excessive warming of the pumped liquid, the recirculation should be returned to the suction vessel, not the pump suction pipe.

Fig. 2.09 Flow control by means of a bypass

Flowrate control by means of a bypass is especially recommended for side channel and axial flow pumps, as the power absorbed by the pump decreases with increasing flowrate.

2.2.2 Adjustment of pump characteristic

2.2.2.1 The **reduction of impeller diameter** is a useful method of permanently reducing the flowrate and total head of the pump.

Impeller trim is only possible however with radial flow and to a limited extent, mixed flow impellers. In the case of pumps without a diffuser e.g. volute case pumps, the diameter of the impeller is reduced by machining the diameter D down to D_x, in the case of pumps with a diffuser e.g. multistage split case pumps, the impeller is usually modified by machining down the vanes only to diameter D_x , as shown in Fig.2.10.

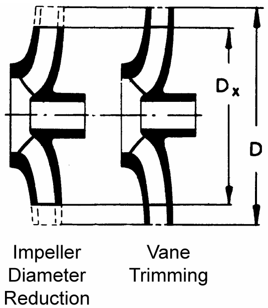


Fig.2.10 Reduction of impeller diameter.

If only a minor adjustment is made to the impeller or vane diameter the following relationships apply to the flowrate and head:

$$Q_x / Q = (D_x / D)^2$$

$$H_x / H = (D_x / D)^2$$

The diameter D_x can be established for the usual linear co-ordinates of the H /(Q) characteristic from the following curves (Fig 2.11):

Draw a straight line through the point Q = 0, H = 0 and the desired duty point Q_x , H_x ; this line intersects the $H(Q)$ curve of the unmodified impeller at point "S". Using the values of Q and H of this point the diameter D_x can be calculated from:

$$D_x \approx D \cdot (Q_x / Q)^{0,5}$$

or

$$D_x \approx D \cdot (H_x / H)^{0,5}$$

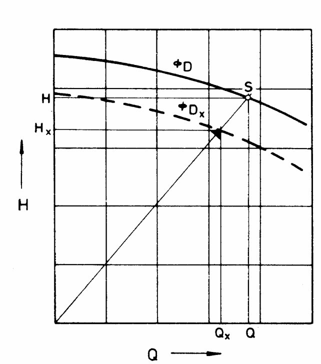


Fig. 2.11 Modifying pump performance by reduction of impeller diameter

The above calculations are not accurate if the performance of the pump needs to be reduced by a large amount and it is recommended that adjustments are made in small steps.

Initially the impeller should be reduced to a diameter larger than the calculated figure D_x, and the pump then tested to determine the final required diameter. For pumps of higher performance this final diameter may need to be established in several steps.

In order to avoid this expensive procedure for pumps which are manufactured in quantity, standard design data such as measured curves for $H(Q)$, $P(Q)$ and $\eta(Q)$ is collated, not only for the unmodified impeller diameter D but also for several impeller trims D_x . This data is often presented in the form of contour curves. In this way the above relationships can also be used for substantial diameter reductions by interpolating between values for different impeller diameters.

For pumps with low specific speed, the loss of efficiency for minor impeller adjustments is small, but pumps with higher specific speed and hence pumps with mixed flow impellers display a noticeable drop in efficiency, even for small adjustments.

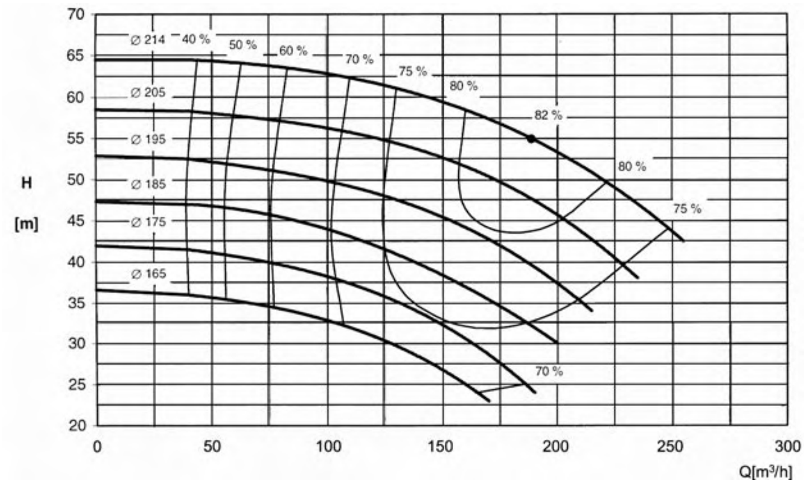


Fig. 2.12

H(*Q*) Characteristic curves for a single stage volute casing pump for maximum, minimum and intermediate impeller diameters.

For multistage ring section pumps the adjustment of operating characteristics can be achieved by a combination of impellers with different diameters. For each size of pump there is a combination of impeller diameters and number of stages which gives a narrow field of different characteristics, without the need for expensive and time consuming impeller diameter adjustments.

This method does not always ensure exactly matching the selection point (*Q* and *H*). It is therefore possible that depending on the characteristic, the flowrate may be larger than required. As a result the total head and power absorbed will rise. Therefore this method should only be used for powers up to approx. 50 kW. For larger powers the efficiency should be checked.

If a combination of different impellers is not applicable, then all the impellers are trimmed to the same diameter.

So that the (*NPSH3*) value is not affected, the first stage impeller is not trimmed.

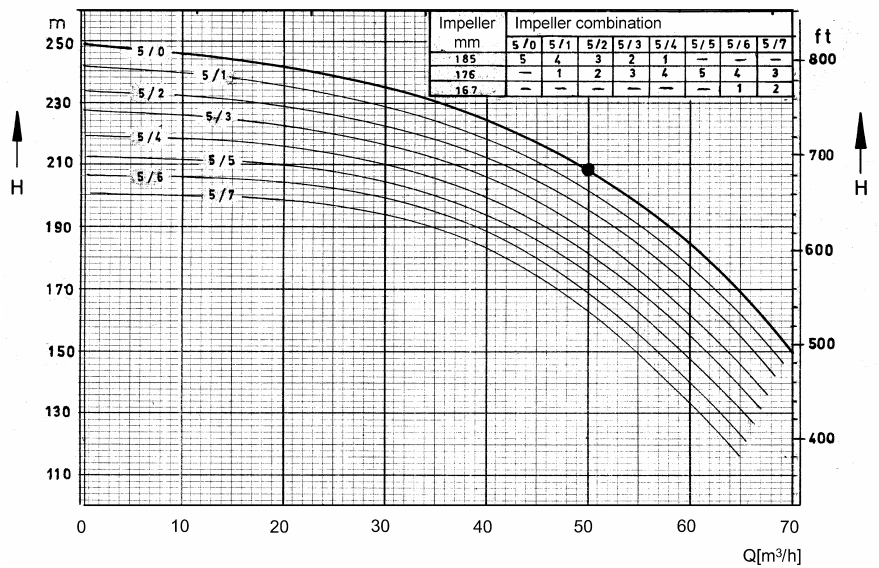


Fig. 2.13

H(*Q*) Characteristics for multistage ring section pumps with various impeller combinations

The impellers of waste water pumps (paddle wheel impellers) and slurry pumps cannot have impeller diameter trims for hydraulic reasons. For these types of pumps the characteristics are adjusted by speed control (see chapter 2.2.2.4).

Impeller adjustment is also not possible for tubular case pumps with axial or mixed flow impellers. Any necessary adjustment is made by an alteration of the blade angles or guide vane angle and/or selection of motor with appropriate speed or with intermediate gearbox. Similarly side channel pumps are not suitable for impeller trimming.

2.2.2.2 The **undercutting of the angle of blade tips** is a means of achieving small increases in the total head generated by centrifugal pumps with radial or mixed flow impellers. In the range of optimum flowrate the *H*(*Q*) characteristic can be raised by approximately 3%. The undercutting has little effect on the shut off head H_0.

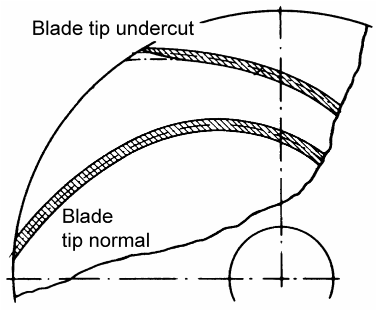


Fig. 2.14 Undercutting impeller tips

2.2.2.3 The adjustment of operating characteristic of multistage split case pumps can be achieved by the **fitting of dummy stages**. According to the requirement, the impeller and guide vanes of one or more stages can be replaced by dummy stages. The total head of the pump is reduced depending on the number of dummy stages.

If at the design stage of an installation it can be foreseen that at a later phase an increased head will be required, then the pump can be supplied for the initial phase with dummy stage(s), which will be replaced by pumping stages for the subsequent phase(s).

2.2.2.4 In the case of **speed variation** of a centrifugal pump, the pump characteristic curve varies in accordance with the pump affinity laws. The following relationships apply for the flowrate and head:

$Q_x / Q = n_x / n$

$H_x / H = (n_x / n)^2$

If the $H(Q)$ curve is known for one speed of rotation n then it can be converted for another speed n_x . The intersection of the new H_x (Q_x) curve with the system curve gives the new duty point.

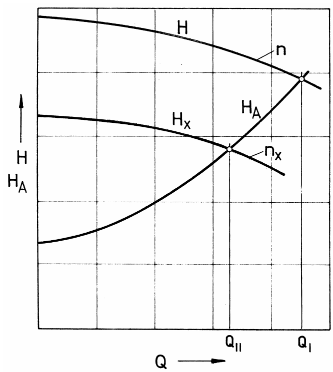


Fig. 2.15 Change of flowrate due to change of speed of rotation.

It should be noted that the (*NPSH3*) value of the pump changes according to the following relationship:

$$(NPSH3)_x / (NPSH3) \approx (n_x / n)^x$$

The first approximation of the conversion of the (*NPSH3*) value can be made with the factor x = 2 providing n_x falls in the range 80 to 120% of the nominal speed *n* and the flowrate Q_x falls in the range 50 to 120% of the optimum flowrate Q_{opt} at the nominal speed *n*.

Furthermore the factor x = 2 is only valid for pumps with a specific speed $n_s \leq 106$ rpm.

If the values of n_x, Q_x and n_q lie outside the above ranges, then the manufacturer should be consulted.

For small changes in speed ($\Delta n / n \leq 0{,}2$), the efficiency remains nearly constant.

For larger changes in speed, the efficiency can be approximated from the following equation:

$$\eta_x \approx 1 - (1 - \eta) \cdot (n / n_x)^{0,1}$$

The following equation applies to the power absorbed:

$$P_x / P \approx \eta / \eta_x \cdot (n_x / n)^3$$

All the above formulae are valid equally for a reduction in speed as for an increase in speed from the original selection speed.

The adjustment of the pump characteristic by **speed control** is the most effective method when the pump flowrate must be matched to continually changing operating conditions.

As the absorbed power reduces by the cube of the speed reduction ratio, the use of speed control is the most efficient method of matching the actual flowrate to the changing operating requirements.

Speed control is especially favoured for plants with a steep system characteristic. This is normally the case when $H_{stat} = 0$ or when the static head component of the total system head is relatively small. In this case the pump, if correctly, selected always works at the optimum and best efficiency.

In contrast, plants with flat system curves operate more in an area of part load efficiency if the speed is reduced.

In such cases it may be better to divide the total flowrate between a multiple of equally sized pumps, or even different sizes, but without speed control. Another alternative is to operate a base load pump at constant speed and to satisfy the peak loads with speed controlled pumps.

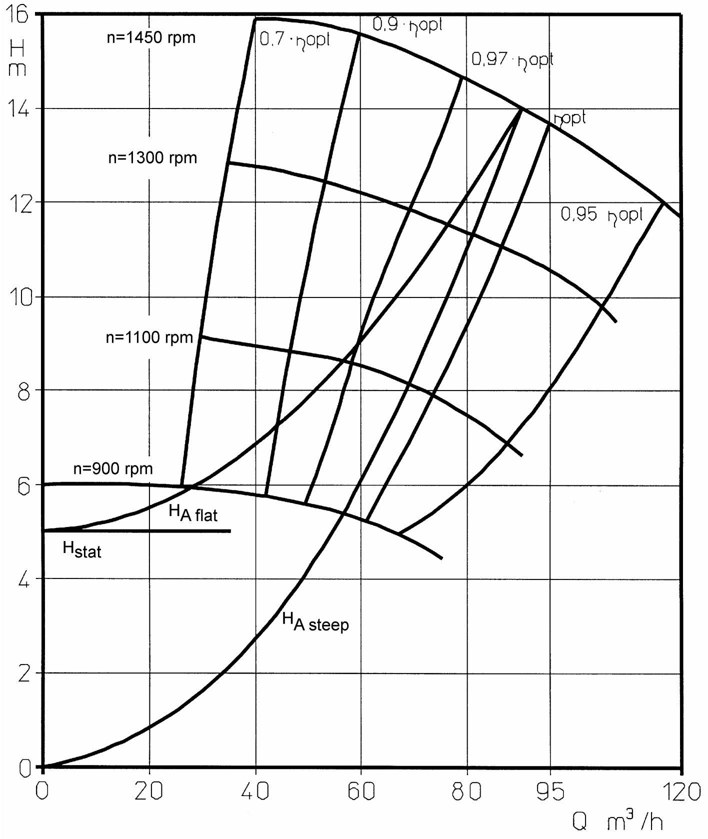


Fig. 2.16 Characteristic of a single stage volute case pump with speed control

The figure also shows the interaction between the pump characteristic and flat and steep system characteristics.

The principal pump drives with speed control are:

- Three phase motor with frequency inverter
- Pole-changing three phase motor
- Three phase motor with mechanical, hydraulic or combination of mechanical and hydraulic speed control.

For further information on drives see section 9.6, “Speed control of electrical drive”.

For larger drive power, e.g. for boiler feed pumps in thermal power stations, steam turbines are often used above 8MW and these can generally be speed regulated.

Speed variation is normally employed when impeller reduction or other methods of matching the operating conditions are not considered viable for hydraulic or economic reasons.

This applies to both waste water pumps and pumps for handling solids, which have specially designed impellers. If such a pump does not meet the required Q and H at direct coupled motor speed, due to the hydraulic design, then the necessary speed to achieve the operating data with the pump in question is calculated. The pump is then driven by belt drive or gearbox at the required increased or decreased speed.

Speed variation is also of advantage if an economic pump selection is not possible with a direct coupled motor.

Operating speeds up to 7000 rpm are common especially for multistage high pressure pumps, e.g. boiler feed pumps and also special pumps e.g. pitot tube pumps. Such pumps are normally gearbox driven. To achieve speed variation a hydraulic speed controller is added. For higher powers steam and gas turbines are also used.

With any change of pump speed, especially increased speed, the hydraulic speed limitation with regard to the (*NPSH*) value and the mechanical speed limitation of the pump and drive must be considered.

In accordance with the formula, an increase in the (*NPSH3*) value as the square of the speed ratio must be calculated and consequently the available (*NPSH*) value of the plant (*NPSHA*) could set a limit on the speed increase.

The mechanical speed limitation is dependent on the limiting values of the bearings, the lubrication, the shaft loading (P/n value), the shaft seals (slip speed v_g) and the material of the impeller.

Table 2.01 Maximum permissible tip speed of impeller dependent on materials.

Material	example	$u_{max\ all}$ in m/s
Grey cast iron	EN-JL1040 EN-GJL-250	40
Spheroidal graphite cast iron Bronze and brass	EN-JS1030 EN-GJS-400-15 2.1050 G-CuSn 10	50
Stainless steels (normal)	1.4008 GX7CrNiMo12-1	95
Stainless steels (special)	1.4317 GX4CrNi13-4	110

Speed reduction generally gives no problems, but with multistage pumps with axial thrust balance (relief devices) the minimum speed for the full effectiveness of the device must be observed.

2.2.2.5 Pre-rotation control utilises the effect of the pre-rotation of the liquid flowing into a centrifugal pump on the $H(Q)$ characteristic. Positive pre-rotation (i.e. pre-rotation in direction of rotation of impeller) in any type of centrifugal pump results in reduction in the $H(Q)$ characteristic compared to inflow without pre-rotation. Negative pre-rotation (i.e. pre-rotation in opposite direction to rotation of impeller) lifts the $H(Q)$ characteristic. The pre-rotation for this type of control is induced by adjusting the angle of incidence of a cascade of inlet guide vanes upstream of the impeller.

The control range for the application of pre-rotation techniques depends principally on the specific speed of the pump. Whilst in the case of radial flow pumps, the influence of pre-rotation is hardly noticeable, its influence increases with increasing specific speed, i.e. in mixed flow pumps and axial flow pumps. For these pumps the point of optimum efficiency η_{opt} is generally achieved at rates of flow where optimum inflow conditions to the impeller exist. By modifying the inflow conditions, this operating point is changed, whilst at the same time the efficiency drops only slightly, so that control by pre-rotation can be considered a low loss method of control.

The use of a flap guide with variable geometry profile (flap diffuser), in place of the normal inlet guide vanes, holds the efficiency nearly constant over an even wider $H(Q)$ range.

The most advantagous use of pre-rotation control is given by the position and shape of the pre-rotation dependent $H(Q)$ and $\eta(Q)$ curves, when only relatively small changes in flowrate are required for large changes in the system head.

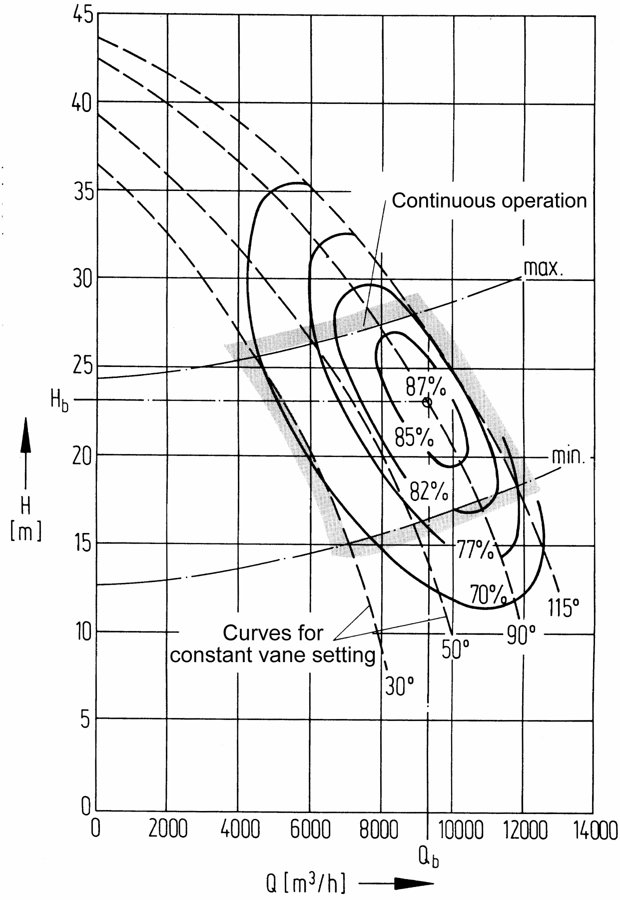


Fig. 2.17 Characteristic field for a tubular casing pump with mixed flow impeller and pre-rotation.

The field shows the maximum limit for continuous operation. Above this limit the pump is likely to operate roughly.

Another method is employed for waste water pumps, whereby a spiral shaped sump is installed. This induces pre-rotation for low inflow rates but virtually no pre-rotation for high inflow rates.

2.2.2.6 Pre-rotation control induced by a bypass represents a substantial improvement compared to normal bypass control (see section 2.2.1.2) by utilising the hydraulic advantages of control by modifying the inflow conditions to the pump impeller (see section 2.2.2.5).

The bypass taken from the delivery side of the pump is not returned to the suction vessel, but is fed into the pump through suitable nozzles in such a way that positive pre-rotation is created upstream of the impeller. The control is not by the position of the nozzles, but only bypass flowrate. A control valve in the bypass line is the only adjustable control device in this arrangement.

2.2.2.7 Control by pitch variation, i.e. adjustment of the angle of the impeller vanes, is only used on mixed flow and axial flow pumps with inserted blades. The control device which may be mechanically, hydraulically or electrically operated is generally so designed that the pitch of the vanes can be changed steplessly whilst the pump is operating. Thereby the flowrate can immediately be matched to frequently changing operating requirements over a wide control range.

A long term adjustment can be achieved in a simpler way with vanes which can be adjusted when the pump is dismantled, but remain fixed in operation.

The most advantageous use of pitch variation control is given by the position and shape of the pitch dependent $H(Q)$ and $\eta(Q)$ curves, when large changes in flowrate are required with only relatively small changes in the system head.

In combination with speed control e.g. gearchange transmission, a large range of control can be achieved. The efficiency remains virtually constant. This is especially the case for plant characteristic with a very small static head component.

Pumps with pitch variation should be started with the blade angle set at the minimum, i.e. with the lowest power requirement.

When a large blade angle is in use, then the minimum overlap of the impeller and pump inlet in respect of cavitation and air pockets should be observed. It may be necessary to conduct trials with models.

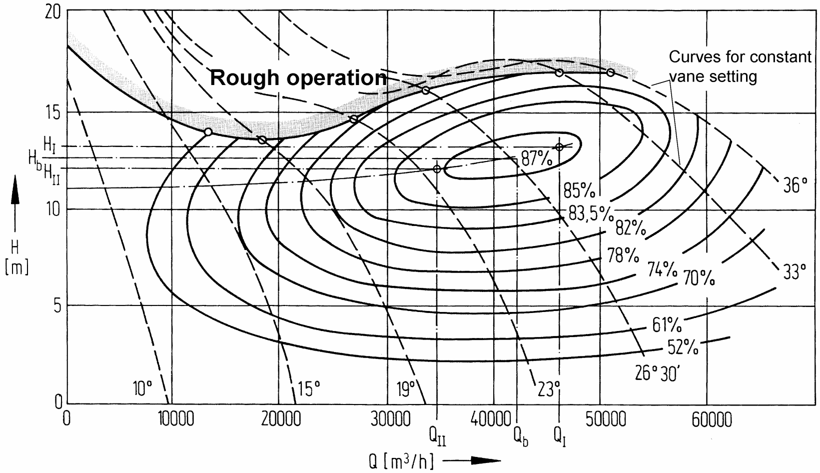


Fig. 2.18 Characteristic field for a tubular casing pump with mixed flow impeller and pitch variation.

The field shows the maximum limit for continuous operation. Above this limit the pump is likely to operate roughly.

The Fig. 2.18 shows that economic operation is possible over a wide range of flowrate by utilising pitch variation.

The example shows that for a vertical tubular casing pump with axial flow impeller, by changing the running blade angle from Q_I to Q_{II} that the flowrate immediately reduces by 25% and the head simultaneously falls by 8% from H_I to H_{II}. In contrast, the efficiency drops by only 1 point compared to the original operating point.

2.3 Effect on the characteristics by installation of an orifice plate.

2.3.1 General

If the $H(Q)$ characteristic is to be matched to a modified duty point in an application where an extremely steep characteristic curve is required to meet control and installation considerations, it cannot normally be achieved by modifying the pump hydraulics only, particularly in the case of mass produced pumps. The only method is to install an orifice plate between the pump discharge branch and delivery pipeline. It should be noted that this constitutes pure throttling in which the losses affect the characteristic curves of the pump directly.

The pressure loss due to an orifice plate follows a square relationship or parabolic form:

$$\Delta p_x = \Delta H_x \cdot \rho \cdot g = \text{const.} \cdot Q_x^2$$

The new characteristic curve of the centrifugal pump achieved by installing the orifice plate, differs from the normal characteristic at every point by the pressure drop Δp_x.

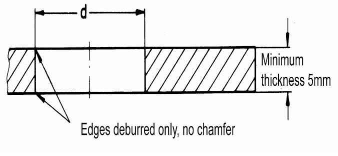


Fig. 2.19 Orifice plate design parameters

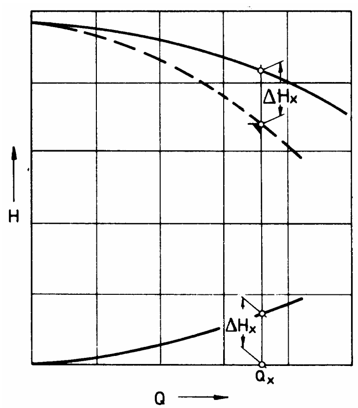
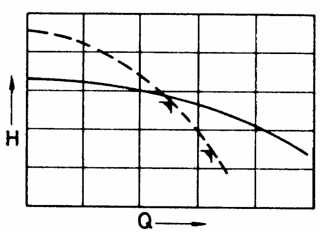


Fig. 2.20 Influence of an orifice plate on the characteristic

As the pressure loss is proportional to the square of the flowrate Q , only two points can be specified and guaranteed for any one orifice plate. All additional points either have to lie on the calculated curve, or can only be achieved by using several orifice plates which have to be changed for each set of points. Limits are therefore set as to the choice of these points.

- The shut off head of the desired $H(Q)$ characteristic cannot exceed the value of the original characteristic curve (Fig. 2.21)
- The slope of the desired $H(Q)$ characteristic cannot be less steep than the original characteristic curve (Fig. 2.22)



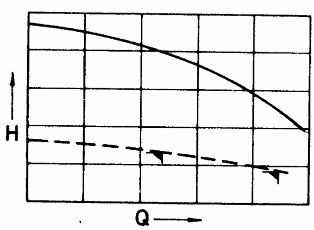


Fig. 2.21 Not achievable characteristic Fig. 2.22 Achievable characteristic

When checking the $H(Q)$ characteristic obtained by using an orifice plate, it should be noted that the measuring point should be no less than 6 x DN (DN = nominal bore of pipeline) downstream of the orifice plate, as the permanent pressure drop is present only after this distance (see also DIN EN ISO5167-1 Code of flow measurement).

2.3.2 Determining the diameter of the orifice plate

Depending on the location of the two guaranteed points, there are in principle two cases to be distinguished in determining the diameter of the orifice plate:

Case I: The shut off head of the pump is not to be modified. In this case the diameter of the orifice plate and hence the pressure drop should be determined so that the characteristic curve achieves the 2nd guarantee point.

The diameter *d* (Fig. 2.19) can be determined by means of the nomogram in table 13.19, following the example.

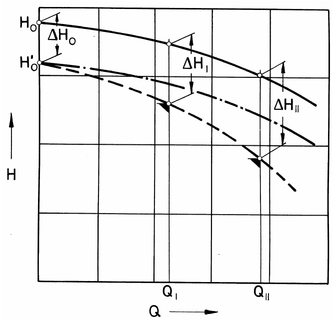


Fig. 2.23 Characteristic field for two operating points

Case II: The required slope of the characteristic curve is expressed by two arbitrary points on the characteristic curve $H(Q)$ (Fig. 2.23), however the limitations quoted in section 2.3.1 should be borne in mind.

Using the following values:

$$\Delta H_0 = \frac{\Delta H_I \cdot Q_{II}^2 - \Delta H_{II} \cdot Q_I^2}{Q_{II}^2 - Q_I^2} \text{ in m}$$

with ΔH_I and ΔH_{II} in m and Q_I and Q_{II} in m³/h

the following results are obtained for:

$\Delta H_0 < 0$: Configuration cannot be achieved, as $H_0' > H_0$ (Fig. 2.21)

$\Delta H_0 = 0$: $H_0' = H_0$, i.e. Case I

$\Delta H_0 > 0$:

The desired characteristic curve can be achieved only by a reduced diameter impeller (or impellers in the case of multistage pumps). After the required impeller diameter has been calculated for the shut off head, $H_0' = H_0 - \Delta H_0$ and the corresponding $H(Q)$ characteristic (see section 2.2.2.1), one proceeds as for case 1. It is immaterial for which of the two guarantee points the diameter of the orifice plate is determined. It is recommended however, that the calculation is made for both points as a check.

2.3.3 Influence of the throttling process on the efficiency

The throttling losses caused by the orifice plate reduces the efficiency. A pump equipped with an orifice plate achieves a lower total head for the same absorbed power.

The efficiency η' of a pump equipped with an orifice plate can be determined as follows:

$$\eta' \approx \eta \left(1 - \frac{\Delta H}{H}\right)$$

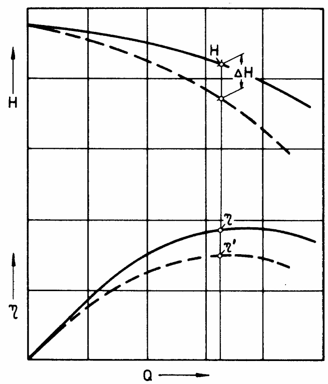
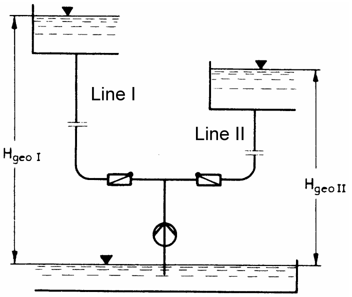


Fig. 2.24 Influence of throttling on the efficiency

2.4 Operation of centrifugal pumps in branched pipelines

2.4.1 Branched delivery pipeline and one centrifugal pump

If one pump supplies several pipelines (e.g. two delivery pipelines as in Fig. 2.25), the combined H_A (Q) of the installation can be determined from the individual component characteristic curves H_{AI} (Q), H_{AII} (Q) etc. by adding the individual flows Q_I , Q_{II} etc. at the same total head, to give the overall rate of flow $Q_I + Q_{II} +$ ... (see Fig. 2.26). It is assumed here that a short suction pipeline as well as a short delivery pipeline is used up to the branch and hence that losses of head in these sections are negligible.
If short pipelines are not used it will be necessary to proceed in accordance with section 2.4.3.



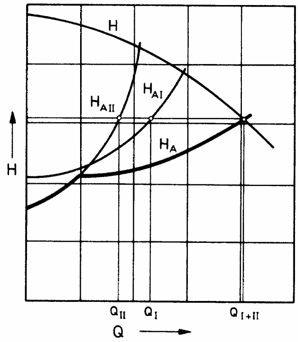


Fig. 2.25

Pump with branched delivery

Fig. 2.26

Characteristic for the installation per Fig. 2.25

The duty point of the pump occurs at the intersection of the $H(Q)$ characteristic with the $H_A(Q)$ characteristic of the system. The horizontal line, (line of equal total head) though the duty point intersects the individual component curves $H_{AI}(Q)$, $H_{AII}(Q)$ etc. at the flowrates of the respective branches of the pipeline. (Fig. 2.26).

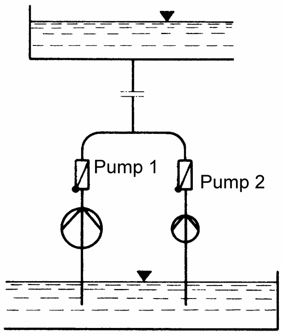
If the static head of the branches is not identical H_{stat} (see example in Fig. 2.25), a non-return valve must be installed in each of the pipeline branches to prevent backflow e.g. one container emptying in to the other when the pump has stopped.

2.4.2 Operation of centrifugal pumps in parallel into a common pipeline

If as shown in Fig. 2.27, two pumps with characteristic curves $H_I(Q)$ and $H_{II}(Q)$ supply a common pipeline having the system characteristic $H_A(Q)$ (assuming that the loss of head of the two pipelines up to the confluence is negligible), a common $H(Q)$ curve is constructed by adding the flows Q_I and Q_{II} of the individual pumps at the same total head (Fig. 2.28).

At the intersection of the characteristic curves $H(Q)$ and $H_A(Q)$ the total flow passing through the pipeline is given. The horizontal line (line of equal head) through this point determines the duty points of the individual pumps on their respective characteristics and hence the component flows Q_I and Q_{II}. Note that these component flows are less than the rates of flow Q_I' and Q_{II}', which would be achieved by each of the pumps operating singly.

This feature is more noticeable with flatter pump characteristic curves and a steeper system characteristic (Fig. 2.29).



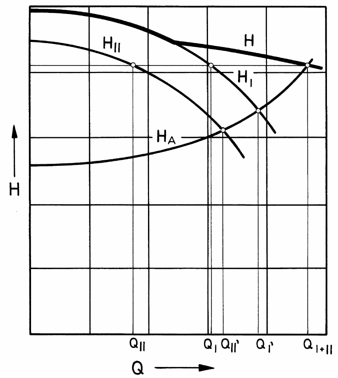


Fig. 2.27 System with two pumps in parallel operation

Fig. 2.28 Characteristic for the plant as Fig 2.27 with a flat system curve and steep pump characteristic

It should be noted that in such cases, when operating in parallel, the individual pumps operate in areas of very poor efficiency and no substantial increase in flowrate can be achieved.

In the case of a very steep system characteristic curve, the flow can be increased more by operating the pumps in series rather than in parallel.

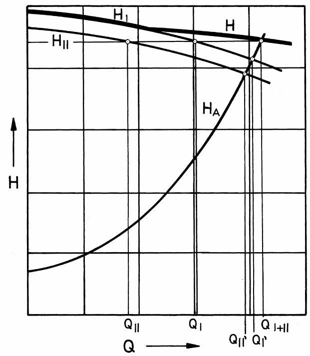


Fig. 2.29 Characteristic for the plant as Fig 2.27 with a steep system curve and flat pump characteristic

Identical pumps with unstable $H(Q)$ characteristics (see also section 2.1.3), generally operate in parallel without trouble in systems with a "fixed" pipeline, bearing in mind that the total head of the 1st pump in the duty point ($Q_{I,II}$; $H_{I,II}$) is smaller than the shut off head of the 2nd pump. The 2nd pump can only be started under this condition (Fig. 2.30).

If both pumps are used in a plant with a system characteristic, as shown with a broken line in this diagram, then this method cannot be used.

The operating parameters should be drawn in all cases, but especially if dissimilar pumps with unstable $H(Q)$ characteristics are used. Only then can it be safely determined whether one of the pumps, or both in the case of parallel operation, is working in an area where flow oscillations or undesirable pressure surges may occur (see section 2.1.3).

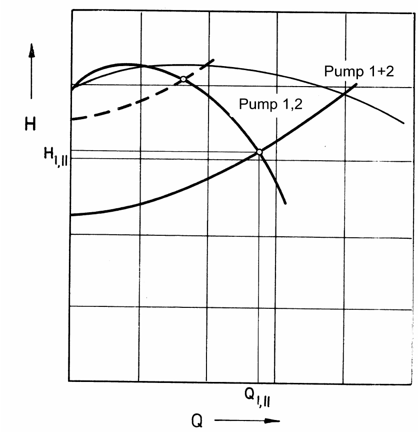


Fig. 2.30

Characteristic for the plant as Fig 2.27 with an unstable $H(Q)$ characteristics

2.4.3 Operation of centrifugal pumps in parallel with separate and common pipeline sections

In this case the flowrates of the individual pumps (in Fig. 2.31 two pumps are shown) add up only at point A, the start of the common section of the pipeline. This means that the total head values of the component flows have to be identical at this point.

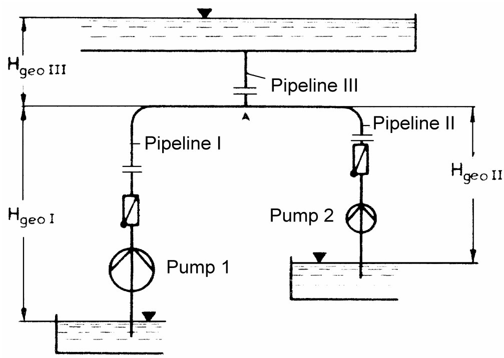
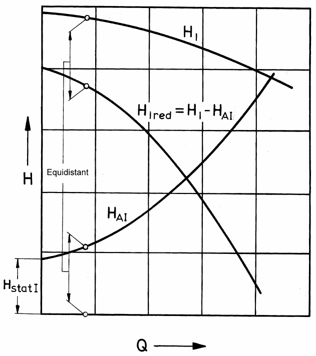


Fig. 2.31 Operation of centrifugal pumps in parallel with separate and common pipeline sections



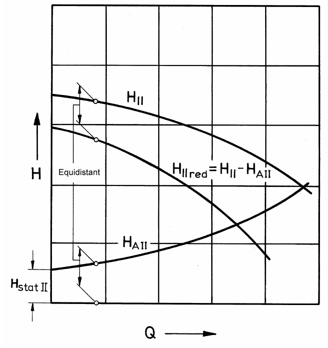


Fig. 2.32 Reduced characteristic curve of pump 1 at point A

Fig.2.33 Reduced characteristic curve of pump 2 at point A

The total head at point A is obtained from the $H_I(Q)$ or $H_{II}(Q)$ characteristics of the individual pumps by reducing them by the head $H_{AI}(Q)$ or $H_{AII}(Q)$ respectively, of the separate sections of the pipeline (Fig. 2.32 and 2.33).

These reduced $H_{I\ red}(Q)$ and $H_{II\ red}(Q)$ characteristics can be added to a common $H_{red}(Q)$ "characteristic", as in section 2.4.2, by adding the flowrates at the same total head. The intersection of this "characteristic curve" with the characteristic curve $H_{AIII}(Q)$ of the system representing the common section of the pipeline establishes the flowrate Q_{I+II}.

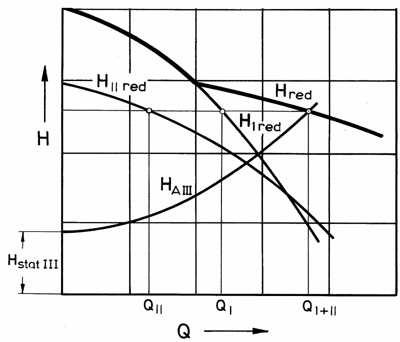


Fig. 2.34 Characteristic for the common section of pipeline

The component flows Q_i and Q_{ii} are obtained at the intersection of the horizontal line with the individual reduced characteristic curves. At these flows Q_i and Q_{ii} respectively of the individual pump, the *H*, *P*, η and (*NPSHR*) can be read off.

2.4.4 Operating centrifugal pumps in series into a common pipeline

When operating centrifugal pumps in series in a system with a common pipeline, the total head values of the individual pumps add up at the same flowrate to give the total head of the series combination.

In the case of unequal pumps, it is expedient to arrange the pump with the lowest (*NPSHR*) as the first in the series.

2.4.5 Operating centrifugal pumps in series with branched pipelines

In contrast to the arrangement of centrifugal pumps in series, in systems with only one pipeline in the arrangement shown in Fig. 2.35 (main pump No1 and booster pump No2) the pumps have different total heads. This can be reduced to that treated in Section 2.4.1, by integrating pump No2 in the pipeline section II, i.e. by defining a system characteristic curve $H_{AIIred}(Q)$, deducting the characteristic curve $H_{AII}(Q)$ from the characteristic curve of pump 2 (Fig. 2.36).

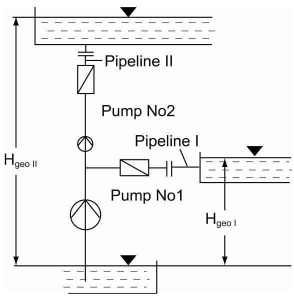


Fig. 2.35 Operating centrifugal pumps in series with branched pipelines

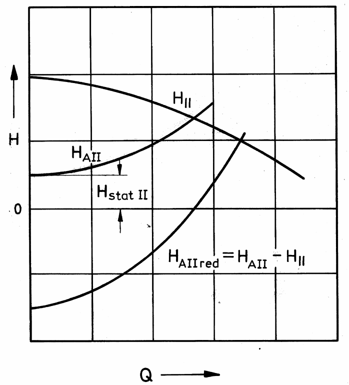


Fig. 2.36 Reduced characteristic curve of pipeline section II of plant

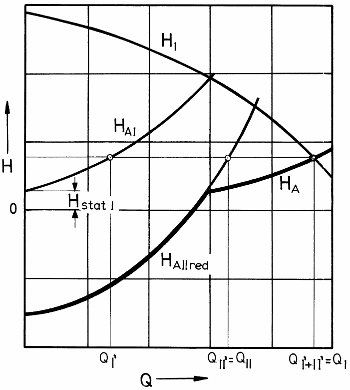


Fig. 2.37 Characteristic curve of the plant per Fig. 2.35

This characteristic curve $H_{AII\,red}$ (Q) is added to the characteristic curve H_{AI} (Q) of the pipeline section I to obtain a common system characteristic H_A (Q) in accordance with section 2.4.1. Its intersection with the $H(Q)$ characteristic of pump No1 gives the duty point of pump No1 (Fig. 2.37). The horizontal line through this duty point intersects the characteristic curves H_{AI} and $H_{AII\,red}$ at the component flows $Q_{I'}$ and $Q_{II'}$. The component flow $Q_{II'}$ is equal to the component flow Q_{II} of Pump No2.

2.5 Start up and run down of centrifugal pumps

Occasionally the conditions during start up and run down of centrifugal pumps may be decisive in the operation of a centrifugal pump installation. In such cases it is important to know the starting torque and start up and run down times of the pump.

2.5.1 The **starting torque M_P** is the torque required by the pump during starting to maintain the running speed reached at each point in time.

$$M_P = 9549 \cdot \frac{P}{n} \qquad \text{in N m} \qquad \text{with } P \text{ in kW, } n = \text{rpm}$$

The value increases from zero when stepped up to the nominal torque. In contrast the absorbed torque transmitted via the coupling M_m is mainly dependent on the driver. Until the nominal torque M_N is reached, the transmitted torque exceeds the starting torque. The difference between these two torque values is the acceleration torque M_b, which increases the speed of the entire rotating mass in the pumpset.

$$M_b = M_m - M_P$$

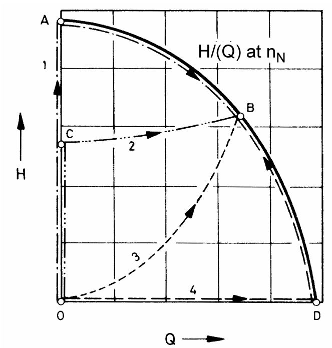


Fig. 2.38 Run up of a centrifugal pump in $H(Q)$ diagram

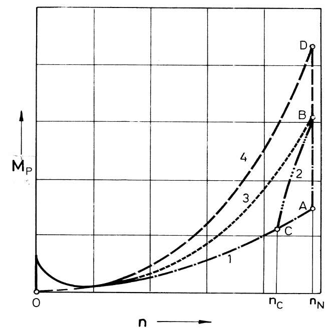


Fig. 2.39 Run up of a centrifugal pump in M_p/(n) diagram

Because in a set of $H(Q)$ curves (Fig. 2.38), a specific rotational speed and a specific torque is associated with every possible duty point, the shape of the starting torque curve is determined by the line along which the operating point of the pump moves from zero to the operating point "B" at nominal speed. Basically four different cases can be distinguished (see Fig. 2.38 and 2.39 for an example of a low specific speed radial flow pump in which the absorbed power P rises with increasing flowrate Q).

1. Starting against a closed valve which is opened after the nominal speed has been reached

Line 0 - A – B (Fig 2.38)

For the section 0 - A the hydraulic torque of the pump increases as the square of the rotational speed. This torque has to be increased to overcome friction in the bearing and shaft seals, which accounts for a relatively high percentage of the starting torque at low rotational speeds. At n=0 the static friction, also known as break-out torque, is particularly high.

Static friction normally amounts to approximately 5 to 10% of the nominal torque. In pumps with overhung impeller arrangements and high inlet pressure its value may reach the order of magnitude of the nominal torque.

For the section A-B the starting torque - now at constant rotational speed - increases or decreases as a function of the shape of the pump power input curve. Side channel pumps and axial flow pumps, where at Q=0 the power input and hence the starting torque is higher than at the duty point, should not be started in this way.

2. Starting with an open valve with a purely static system head acting on the non-return valve

Line 0 - C – B (Fig 2.38)

In this instance the shape of the starting torque curve is identical with the previous case of starting against a closed valve, up to the point C, as only at this point can the non-return valve be opened by the pump delivery pressure and flow commence. The further path of the torque curve is established by constructing intermediate characteristic curves for H (Q) for the section C - B at different rotational speeds (see section 2.2.2.4) and by calculating the torque from the power input. In practice it is often sufficient to make an assumption for the course of the torque to be linear between "C" and "B", whilst the rotational speed for point "C" is calculated from:

$$n_C = n_N \cdot (H_{stat} / H_0)^{0,5} \quad \text{with } n \text{ in rpm, } H \text{ in m}$$

3. Starting with an open valve against a purely dynamic system head

Line 0 – B (Fig 2.38)

This only occurs if the pipeline is very short. The starting torque M_P increases, apart from the torque due to static friction, as the square of the rotational speed from zero to the nominal torque.

If on the other hand the pipeline is very long, the time required to accelerate the mass of water is considerably longer than the starting time for the pump. The mass of water at rest, acts in this case like a closed valve and the starting torque takes a similar course to case 1.

4. Starting with an open valve and an empty delivery pipeline

Line 0 - D – B (Fig 2.38)

Whilst the delivery pipeline remains empty the pump has to provide only a minimal head. If the time taken to fill the pipeline is longer than the start up time of the pump, then the process of starting follows the curve O - D and the torque will increase as the square of the rotational speed. The section D -B again depends on the shape of the pump power input curve. Starting along the curve O-D-B is most suitable for pumps in which the power input decreases with the increasing flowrate. Fig 2.40 shows the starting torque of an axial flow pump for these starting conditions. If pumps are used which have power characteristics that increase with increasing flowrate, then a graphical check should be made to ensure that the driver is not overloaded during the period the pipe is filling.

If the delivery pipeline fills up substantially during the increase of the rotational speed, the course of the torque will resemble case 3.

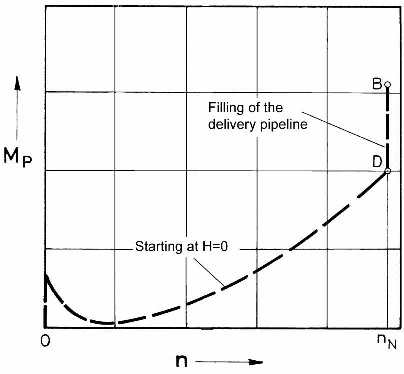


Fig. 2.40 Starting a tubular casing pump with an open valve and an empty delivery pipeline

When starting side channel pumps in a system with an empty delivery pipeline, the valve should be partially closed in order to limit the flow and to prevent cavitation in the pump.

2.5.2 The **start up time** of a centrifugal pump results from the difference between the input torque M_m (torque transmitted via the coupling) and the starting torque M_P, i.e. from the excess torque M_b, which is available for acceleration of the entire rotating mass of the pumpset, from speed $n = 0$ until the nominal rotational speed n_N has been reached.

$$M_b = M_m - M_P$$

Fig. 2.41 shows how the input torque M_m and the starting torque of the pump M_P and therefore the acceleration torque M_b are dependent on the rotational speed.

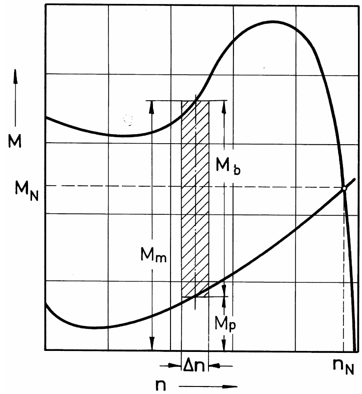
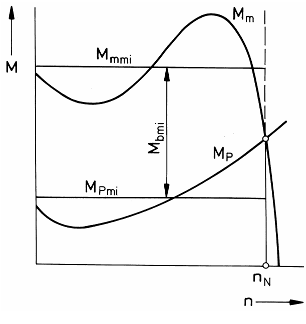


Fig. 2.41 $M(n)$ chart for DOL starting of electric motor and start up of pump as in case 3 (see 2.5.1)

As these interdependent functions are generally not available in analytical form it is usual to derive a mean accelerating torque from the mean input and starting torque.

$$M_{b\,mi} = M_{m\,mi} - M_{P\,mi}$$

Fig. 2.42. shows a sufficiently accurate method of establishing the mean accelerating torque. It can be calculated for example by counting the squares on graph paper for the input and starting torque curves.



The time t_A to reach the nominal speed n_N is calculated from the equation:

$$t_A = \frac{\pi}{30} \cdot \frac{n_N \cdot J_{gr}}{M_{b\,mi}} \quad \text{in secs}$$

Fig. 2.42 Establishing mean accelerating torque

In this equation:

n_N = Nominal speed rpm

J_{gr} = Moment of inertia of entire rotating mass in kg m²

$M_{b\,mi}$ = Mean accelerating torque in N m

The total moment of inertia J_{gr} comprises the moments of inertia of the motor, pump, coupling and where applicable gearbox or belt drive system. It is always referred to the speed of the motor.

If the speed of the motor and pump are different, the moment of inertia J_P (Sum of the moments of inertia of all components rotating at the pump speed n_P) can be interpolated to the motor speed with the following equation;

$$J_{P'} = J_P \cdot \left(\frac{n_P}{n_{Mot}} \right)^2$$

As a result of the favourable torque ratio (M_m : M_P) of centrifugal pumps and the relatively low moment of inertia, the starting time is well within the limits of that of electric motors and so present no problems.

2.5.3 The **run-down time** of a centrifugal pump is found in the same way as the start up time if the starting torque M_P is known as a function of the rotational speed n .

Since during the run down of the pump, as a result of switching off or failure of the driver, the driving torque M_m moves toward zero and the starting torque of the pump becomes a retarding torque M_v.

The run down time, i.e. the time for the pump to come to a stand still after switching off the driver, is calculated from the following equation:

$$t_{aus} = \frac{\pi}{30} \cdot \frac{n_N \cdot J_{gr}}{M_{v\,mi}} \quad \text{in secs}$$

where n_N = Rotational speed rpm

J_{gr} = Total moment of inertia of unit in kg m²

$M_{v\,mi}$ = Mean retarding torque in N m

In general the moment of inertia J_{gr} of the rotating components is small so that centrifugal pumps run down quickly. In very long pipeline systems this may lead, under certain conditions, to pressure surges which exceed the pressure limitations of components in the system. This could lead, in extreme cases, to failure of the pipeline, fittings or instruments, see also section 4.7. Increasing the moment of inertia J_{gr} by the addition of a flywheel can increase the run down time, but whether possible pressure surges are prevented must be checked, in each case.

2.6 Minimum and Maximum Flowrate

2.6.1 Minimum flowrate due to internal power losses

The power loss P_{int} causes a rise in temperature of the pumped liquid. If the normally small heat dissipation through radiation and conduction is neglected, then the heat generated equivalent to the power loss must be removed by the flow of liquid:

$$P_{int} = q \cdot c \cdot \Delta T = \rho \cdot Q \cdot c \cdot \Delta T$$

or in numerical terms:

$$P_{int} = 0{,}278 \cdot \rho \cdot Q \cdot c \cdot \Delta T \qquad \text{in kW}$$

with ρ in kg/dm³, Q in m³/h, c = specific heat in kJ/kg K

c - Values for various liquids see tables 13.16 and 13.17

The internal power loss P_{int} is calculated from the power absorbed by the pump P after deducting the external mechanical power loss P_m and the hydraulic power P_u:

$$P_{int} = P - P_m - P_u = \rho \cdot g \cdot Q \cdot H \cdot \left(\frac{\eta_m}{\eta} - 1\right)$$

The temperature increase is given by:

$$\Delta T = \frac{P - P_{Jm} - P_u}{\rho \cdot Q \cdot c} = \frac{g \cdot H}{c}\left(\frac{\eta_m}{\eta} - 1\right)$$

or in numerical terms:

$$\Delta T = 0{,}01 \cdot \frac{H}{c} \cdot \left(\frac{\eta_m}{\eta} - 1\right) \qquad \text{in K}$$

with H in m, c in kJ/kg K

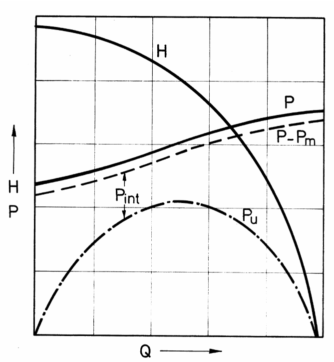


Fig. 2.43 Power loss P_{int} dependent on Q

Fig. 2.43 shows schematically these conditions for a low specific speed centrifugal pump.

Fig. 2.44 shows the curve of the temperature rise ΔT as a function of the flowrate Q. Theoretically, at $Q = 0$ the temperature rise is infinite. If the maximum permissible temperature rise is ΔT_{zul} the flowrate must not drop below the minimum value, also known as the thermal flowrate $Q_{min\ thermal}$. The curve $\Delta T/(Q)$ has to be established point by point and $Q_{min\ thermal}$ has to be read off at the pre-determined value ΔT_{zul} .

The maximum temperature occurs in the outlet branch of the pump where, due to the prevailing pressure, no danger arises from the vaporisation of the pumped liquid.

The temperature rise of the pumped liquid in areas inside the pump where the lowest static pressure occurs is more critical, i.e. in the cavity behind, or in the throttling area of the balance device of multistage pumps, or upstream of the impeller inlet (of the first stage impeller in the case of multistage pumps), if part of the warmed liquid is returned from the delivery side to the suction side of the pump (through a balance or bypass line).

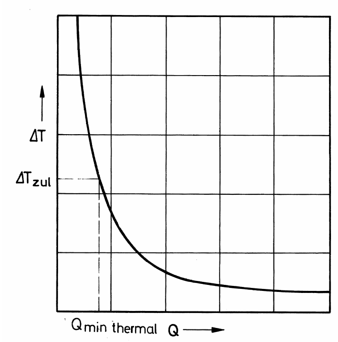


Fig. 2.44 Temperature rise for $Q_{\text{min thermal}}$

This return flow is only acceptable if:

- the internal power loss and hence the temperature rise is small and if it can be assumed that the heat dissipation by radiation from the external balance pipe or bypass line is sufficiently large.
- the difference between the available (*NPSH*) of the system (*NPSHA*) and the required (*NPSH*) value of the pump (*NPSHR*) is sufficiently large.
- it does not cause unsteady flow conditions or vibration

In all other cases flow through the balance pipe or bypass line should be returned to the inlet vessel or container. One further possibility is to cool the liquid which is returned to the suction, but the time lag involved in controlling the coolant can create serious difficulties in cases where the load of the pump frequently varies. It is for this reason that this procedure is not frequently adopted.

2.6.2 Minimum flowrate based on unstable flow conditions

Centrifugal pumps are in most cases selected for a design flowrate, a design total head and design speed which matches the optimum flowrate Q_{opt} .

All hydraulic dimensions such as impeller diameter, discharge diameter, casing cross sectional area, blade angle etc. are fixed for the design data.

If the operating flowrate is smaller than the design or optimum flowrate, then the hydraulic dimensioning no longer matches the actual flowrate. As a result of recirculation and other flow disturbances, this can lead to unstable flow conditions which are manifested by increased vibration and noise levels and can lead to increased bearing loads.

In continuous operation, to prevent pump operating problems and damage, operation below the manufacturer's minimum permissible flowrate $Q_{\text{min stable}}$, which is known as the "minimum stable flowrate" should be avoided.

2.6.3 Maximum flowrate

If the operating flowrate is higher than the selection and/or optimum flowrate, then the points in the preceding section 2.6.2 are relevant, i.e. these operating conditions can also cause unstable flow conditions.

In continuous operation, to prevent pump operating problems and damage, operation above the manufacturer's maximum permissible flowrate $Q_{\text{max stable}}$ which is known as the "maximum stable flowrate" should be avoided.

The maximum permissible flowrate can also be limited by the (*NPSH*) value of the system (*NPSHA*), if the (*NPSH*) value of the pump (*NPSHR*), increasing with the flowrate, becomes larger than the (*NPSHA*). These operating conditions are not suitable for continuous operation. See also section 1.5.4, and 2.1.3.

2.6.4 Recommended limits for minimum and maximum flowrate in continuous operation

The following table gives the recommended limits of $Q_{\text{min stable}}$ and $Q_{\text{max stable}}$, for continuous operation, for different types of pump, dependent on the specific speed n_s, on the assumption that the (*NPSH*) value permits (see section 1.5.4).

When establishing the limits for side channel pumps, axial flow pumps and centrifugal pumps with unstable characteristics, the sections 2.1.1, 2.1.3, and 2.4.2, should be considered.

Pump type	Side channel pumps	Centrifugal pumps	Mixed flow pumps	Axial flow pumps
n_s	4 ... 12	8 ... 45	40 ...160	100 ... 300
$Q_{\text{min stable}}/Q_{\text{opt}}$	0,10 ... 0,64	0,10 ... 0,40	0,60 ... 0,65	≈ 0,75
$Q_{\text{max stable}}/Q_{\text{opt}}$	1,10 ... 1,40	≈ 1,50	≈ 1,35	≈ 1,10

Table 2.02 Recommended limits for continuous operation

The limits for multistage ring section pumps with thrust balance are set much more closely and therefore it is important that the data provided by the manufacturer is observed.

This also applies to canned motor pumps and pumps with a magnetic coupling.

2.6.5 Protection of centrifugal pumps

If the operation of the plant means that the pump may be working against a closed shut off or control valve or similar, it is necessary to protect the pump from operating below the minimum flow. This applies generally for all types of pump, but especially for multistage high pressure pumps with common relief devices against axial thrust. This is important, as with this type of pump, falling below the minimum flow leads to evaporation of the pumped liquid in the spaces in the relief device, which can cause immediate failure of the pump with considerable damage.

There are several possible methods of protection

- Continuous return of the minimum flowrate through a by-pass line with a fixed orifice plate. With this method, the minimum flow must be added to the nominal flow for the pump selection.
- Return of the minimum flowrate through a by-pass line with a fixed orifice plate and an open/shut valve. The valve is opened when the flow in the main pipeline falls below the minimum flowrate.
- Regulation of the minimum flowrate through an automatic by-pass valve, e.g. a free flow non-return valve. The spring operated non-return valve is flow dependent. The setting of the valve, controls the by-pass flow. It is set so that the by-pass line opens as soon as the flow in the main pipeline falls below the minimum flowrate. When the non-return valve is closed the by-pass is fully open. This method is used principally for boiler feed and condensate pumps.
- Regulation of the minimum flowrate through a modulating valve. The modulating valve can be controlled by the flow in the main pipeline, from pressure or temperature difference, or from the absorbed power of the drive motor, however control by the flowrate is the most reliable method.

3 Hydraulic Acceptance tests for Centrifugal Pumps

3.1 Preliminary remarks

Hydraulic acceptance tests are carried out in accordance with fixed regulations which are defined in the appropriate standards.

The codes for the tests simplify the communication between the pump manufacturer or supplier and the purchaser or the nominated acceptance test inspector.

In general they comprise:

- Definitions concerning all parameters which are required to describe the function of a centrifugal pump and to enable the determination of guarantees of the duty point (Q_G, H_G), for the efficiency (η_G) and the necessary *(NPSH)* value *(NPSHR)*.
- Statement of the technical guarantees and their fulfilment.
- Recommendations for the preparations for and the execution of, the acceptance tests for the guarantee values.
- Laying down the methods to be used for the comparison of measured results with the guaranteed values and for the conclusions to be drawn from this comparison.
- Recommendations for drawing up the test report.
- Description of the most important methods of measurement which can be used to confirm the guarantee values.

Terms used in this context like “guarantee” or “acceptance” are to be understood in a technical, but not a legal sense. The term “guarantee” specifies contractual test values, but has no meaning in respect of the rights and obligations which arise if these values are not achieved. The term “acceptance” has no legal meaning in this context. A successful acceptance test does not on its own mean legal acceptance.

More accurate test measurements and closer adherence to the guarantee values involves additional procedures and expenditure. The guarantee values should therefore be limited to those which are essential to ensure trouble free operation of the plant.

3.2 Acceptance tests in accordance with EN ISO 9906

This standard is for the hydraulic acceptance of centrifugal pumps and combines and replaces the following former international standards.:

- ISO 3555 “Centrifugal pumps (Radial, Mixed flow and Axial pumps) - Guidelines for Acceptance Tests - Class B” (corresponds with grade 1 of the new standard)
- ISO 2548 “Centrifugal pumps (Radial, Mixed flow and Axial pumps) - Guidelines for Acceptance Tests - Class C” (corresponds with grade 2 of the new standard)

This standard further replaces the national standard:

- DIN 1944 “Acceptance Tests for Centrifugal Pumps”

There is however an important change in the verification of guarantees, because the uncertainty of measurement must not influence the acceptability of a pump and the tolerances are due to constructional differences only.

New tolerance factors have been introduced to ensure, as far as possible, that a pump which was acceptable under the previous International Standards (ISO 2548 and/or ISO 3555) would also be acceptable under this new International Standard.

The International Standard ISO 5198 “Centrifugal pumps (Radial, Mixed flow and Axial pumps) – Code for hydraulic performance tests – Precision grade” is not to be understood as an acceptance test code. It gives guidance for measurements of very high accuracy and for the thermodynamic method for direct measurement of efficiencies, but it does not recommend verification of guarantees.

3.2.1 General

The International Standard EN ISO 9906 specifies hydraulic performance tests for acceptance of Centrifugal pumps (Radial, Mixed flow and Axial pumps, hereinafter simply designated as “pumps”). For side channel pumps see section 3.3. It is applicable to pumps of any size and to any pumped liquids behaving as clean cold water.

The standard contains two grades of accuracy of measurement:

- Grade 1, for higher accuracy
- Grade 2, for lower accuracy

These grades include different values for tolerance factors, for allowable fluctuations and uncertainties of measurement.

For pumps produced in quantity, whose selection is made from typical performance curves and for pumps with a power input of less than 10 kW, see Table 3.05, for higher tolerance factors.

This Standard is applicable both to a pump itself, without any fittings and to a combination of a pump associated with all or part of its upstream and/or downstream fittings.

If the manufacturer/supplier and the purchaser have not otherwise agreed, the following applies:

- Accuracy to grade 2
- The test is carried out in the manufacturers test area
- The cavitation test *(NPSH)* is not included

Any deviations from the above should be agreed between the manufacturer and purchaser.

Among others, such deviations may be:

- Accuracy according to grade 1
- No negative tolerance factors
- Higher tolerance factors for pumps produced in quantity and for pumps with a power input of less than 10 kW.
- Number of pumps to be tested in the case of an order for several identical pumps
- Checking the behaviour of the pump with respect to bearing temperature, noise levels and vibration during the acceptance test
- Test rig for checking the self priming capability of self priming pumps
- Procedure for predicting the pump performance on the basis of a test with clean cold water
- Scope of the guarantee

  a) Pump without motor or pumpset with motor

  b) Pump with or without pipe section

  c) Guaranteed values for one or more operating points (e.g. flowrate, total head, absorbed power, efficiency, (*NPSHR*))
- Tolerance factors at the operating point and other points, if several operating points are to be guaranteed
- Similar construction of pumps, (e.g. several rotors in the same casing)
- A requirement for cavitation testing *(NPSH)*
- Index for the conversion equation for the (*NPSH)* value

3.2.2 Guarantee

A guarantee point shall be defined by a guarantee flow Q_G and a guarantee head H_G.

The manufacturer/supplier guarantees that under the specified conditions and at the specified speed the measured *H(Q)* curve will pass through a tolerance range which surrounds the guarantee point (see Fig. 3.01).

In addition; at the guaranteed flowrate, one or more of the following quantities may be guaranteed under the specified conditions and the specified speed:

- Efficiency of the pump η_G or the combined efficiency of the pump/driver set $\eta_{gr\,G}$
- The required (*NPSH*) value (*NPSHR*)

Unless otherwise agreed, the guarantee point is valid for clean cold water.

3.2.3 Test speed

Unless otherwise agreed, tests may be carried out at a test speed of rotation within the range 50 to 120% of the specified speed. However, it should be noted that when deviating by more than 20% of the specified speed, the efficiency may be affected.

For *(NPSH)* tests, the test speed of rotation should lie within the range 80 to 120% of the specified speed of rotation, provided that the rate of flow lies within 50 and 120% of the rate of flow corresponding to the maximum efficiency at the test speed of rotation. This is provided that the type number K is less than or equal to 2. For pumps with a type number greater than 2 a special agreement must be reached between the parties.

3.2.4 Tests on pumps for liquids other than clean cold water

The performance of a pump varies substantially with the nature of the liquid being pumped (viscosity, dissolved gas, solids content etc.). Although it is not possible to give general rules whereby performance with clean cold water can be used to predict performance on another liquid, it is often desirable for the parties to agree on empirical rules to suit the particular case and to test the pump with clean cold water. Guidelines for the correction of test results for viscous liquids, see section 4.1.2.2.

The specification of “clean cold water” according to the Standard are defined in the following table:

Table 3.01 Specification of “clean cold water”

Characteristic	Unit	Max value
Temperature	°C	40
Kinematic viscosity	m^2/s	$1.75 \cdot 10^{-6}$
Density	kg/m^3	1050
Non-dissolved solids content	kg/m^3	2.5
Dissolved solids content	kg/m^3	50

The total dissolved and free gas content of the water shall not exceed the saturation volume, corresponding to the following conditions:

- for an open circuit, the saturation volume corresponding to the pressure and temperature in the entry side of the open container
- for a closed circuit, the saturation volume corresponding to the pressure and temperature in the closed container

Pumps for liquids other than clean cold water may be tested for flowrate, head and efficiency with clean cold water if the liquid to be pumped is within the specification in the following table:

The total dissolved and free gas content of the liquid shall not exceed the saturation volume as defined above.

Table 3.02 Characteristics of liquids

Characteristic of liquid	Unit	Min value	Max value
Kinematic viscosity	m^2/s	No limit	$10 \cdot 10^{-6}$
Density	kg/m^3	450	2000
Non-dissolved solids content	kg/m^3	–	5.0

Tests for pumps, for liquids other than those described above shall be subject to special agreement.

If not otherwise agreed, cavitation tests shall be carried out with clean cold water. The necessary *(NPSH)* value *(NPSHR)* is always given for clean cold water.

3.2.5 Conversion of the test results to the guarantee conditions

All test data which is obtained at a speed of rotation n which deviates from the specified speed n_{sp} must be converted to a value for the specified speed n_{sp}. If the pumped liquid deviates from the specified density, then in the calculation of the absorbed power P, allowance must be made.

Providing the test speed falls within the limits outlined in section 3.2.3, then the measured data for flowrate Q, total head H, absorbed power P and efficiency η can be converted by means of the equations:

$$Q_T = Q\,(n_{sp} / n)$$

$$H_T = H\,(n_{sp} / n)^2$$

$$P_T = P\,(n_{sp} / n)^3 \cdot (\rho_{sp} / \rho)$$

$$\eta_T = \eta$$

The measured data for *(NPSH)* value can be converted by means of the equation:

$$(NPSHR)_T = (NPSHR) \cdot (n_{sp} / n)^x$$

As a first approximation for the *(NPSH)* value, the exponent $x = 2$ may be used if the conditions in section 3.2.3 are fulfilled and the physical state of the liquid at the impeller inlet is such that the operation of the pump cannot be affected by gas separation.

If the pump operates near its cavitation limits, or if the deviation of the test speed from the specified speed exceeds the specifications in section 3.2.3, then values for exponent x between 1.3 and 2 are observed and a value must be agreed between the parties to establish the conversion formula.

3.2.6 Values of tolerance factors

Centrifugal pumps are manufactured to permitted tolerances of casting and machining. During manufacturing every pump is subject to deviations from the drawing dimensions.

When comparing the test results with guaranteed values (operation points), tolerances shall be allowed, which cover all the possible deviations in the test pump data from that of a pump (theoretical only) without any manufacturing variations from design.

In the absence of specific agreement the tolerance factors in the following table shall be used for the guarantee points Q_G, H_G.

Table 3.03 Values of tolerance factors

Measurement	Symbol	Grade 1 %	Grade 2 %
Flowrate	t_Q	± 4.5	± 8
Total head of pump	t_H	±3	± 5
Pump efficiency	t_η	– 3	– 5

For pumps produced in series, which are selected from published typical performance curves in catalogues, the following tolerance factors are applicable:

Table 3.04 Tolerance factors for pumps produced in series

Measurement	Symbol	Tolerance factor %
Flowrate	t_Q	± 9
Total head of pump	t_H	± 7
Pump absorbed power	t_P	+ 9
Motor power input	$t_{P\,gr}$	+ 9
Efficiency	t_η	– 7

For pumps with an absorbed power less than 10 kW but more than 1 kW, where friction losses in various mechanical components (bearings, stuffing box packings and mechanical seals) become relatively important and not easily predictable, the tolerance factors in the following table can be applied:

Table 3.05 Tolerance factors for pumps with absorbed power between 1 and 10 kW

Measurement	Symbol	Tolerance factor %
Flowrate	t_Q	± 10
Total head of pump	t_H	± 8

3.2.7 Verification of guarantee

The verification of each guarantee shall be accomplished by comparing the results obtained from the tests (including any measurement uncertainties) with the values guaranteed in the contract (including the associated tolerances).

3.2.7.1 Verification of guarantee on flowrate, head and efficiency

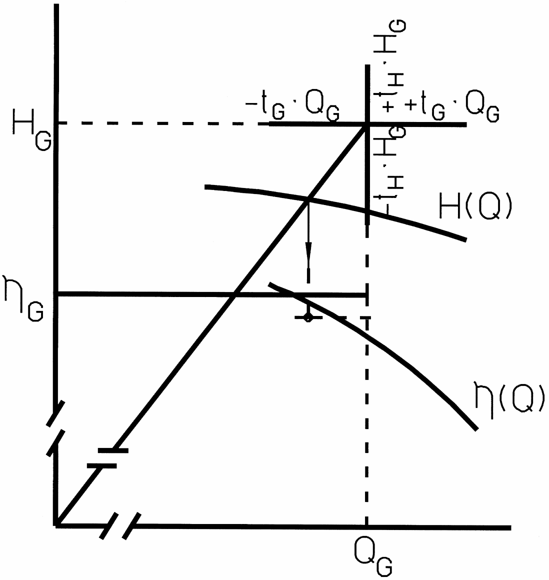


Fig. 3.01 Verification of guarantee on flowrate, head and efficiency

The results of the measurements are converted to values for the specified speed. They are then plotted against the flowrate Q. The curves which best fit the measured points will represent the performance of the pump.

A tolerance cross is drawn through the guarantee point Q_G, H_G with the horizontal line from $Q = Q_G - t_Q \cdot Q_G$ to $Q = Q_G + t_Q \cdot Q_G$ and the vertical line from $H = H_G - t_H \cdot H_G$ to $H = H_G + t_H \cdot\cdot H_G$.

The guarantee value of the flowrate and total head have been met if the QH curve cuts or at least touches the vertical and/or horizontal line (see Fig. 3.01).

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

The guarantee condition on efficiency is within tolerance if the value at this point of intersection is higher than or at least equal to $\eta_G \cdot (1 - t_\eta)$ (see Fig. 3.01).

If the measured values of Q and H are higher than the guaranteed values Q_G and H_G, but within the tolerances, and also the efficiency is within tolerance, the actual absorbed power may be higher than that given in the data sheet.

3.2.8 Obtaining specified characteristics

3.2.8.1 Reduction of impeller diameter

When it appears from the tests that the characteristics of the pump are higher than the specified characteristics, in general a reduction in impeller or vane diameter is carried out (see section 2.2.2.1).

If the difference between the agreed values and the measured values is small, it is possible to avoid a new series of tests by application of proportionality rules (see section 2.2.2.1). Using the proportionality rules, the necessary impeller or vane diameter can be derived with sufficient accuracy.

3.2.8.2 Speed variation

If a pump with a variable speed drive does not meet or exceeds the guarantee values, the test points may be recalculated for a different speed of rotation, provided the maximum allowable continuous speed is not exceeded (see section 2.2.2.4).

3.2.9 Cavitation tests

In most cases cavitation can be detected by the decrease in delivery head at a given flowrate (see section 1.5.1). In the case of multistage pumps, the decrease in head is taken as the decrease in the first stage if this is accessible for measurement. If the head at the first stage cannot be measured then it is calculated by dividing the total head by the number of stages.

Most cavitation tests will be conducted with clean cold water. However cavitation tests with clean cold water cannot accurately predict the behaviour of the pump with other liquids.

Different types of cavitation tests can be carried out according to the requirements or agreed documentary verification.

3.2.9.1 Verification of guaranteed characteristics at a specified (*NPSHA)* value

A simple check may be made to determine the flowrate performance of the pump at the specified (*NPSHA)* value without exploring the cavitation effects.

The pump meets the requirements if the guaranteed pump total head and efficiency are obtained at the specified flowrate and (*NPSHA)*.

3.2.9.2 Verification that the performance of the pump at the specified (*NPSHA)* value is not influenced by cavitation

A check may be made to show that the flowrate of the pump at the specified conditions is not affected by cavitation.

The pump meets the requirement if a test at a higher (*NPSH)* value than the specified (*NPSHA)* value gives the same total head and efficiency at the same flowrate.

3.2.9.3 Determination of (*NPSH3)*

In this test the (*NPSH)* is progressively reduced until the total head (for multistage pumps, of the first stage) drops by 3% at constant flowrate. This (*NPSH)* value is the (*NPSH3)*.

The pump meets the requirements if the measured values, are smaller than or equal to the supplier's required (*NPSHR)*.

For pumps with very low total heads, a larger drop figure can be agreed for the verification.

3.2.9.4 Other cavitation tests

Other cavitation criteria e.g. increase in noise or gas bubble creation and corresponding cavitation tests may be used. In this case special agreement in the contract is necessary. It is necessary to clearly identify the criteria and limit values to be used and set these down.

3.2.10 Determination of the (*NPSH)* required by the pump

For the purpose of the acceptance tests, the (*NPSH)* required is understood to be the head at which the acceptance criteria e.g. max. 3% head reduction, size of gas bubble formation, increase in noise level etc. are achieved compared to operation without cavitation.

3.2.10.1 Tolerance factor for (*NPSHR)*

The maximum permissible difference between measured and guaranteed (*NPSHR)* is

for grade1: $t_{NPSHR} = + 3\%$ or $t_{NPSHR} = + 0{,}15$ m

for grade 2: $t_{NPSHR} = + 6\%$ or $t_{NPSHR} = + 0{,}30$ m

which ever is the greater.

Using the following formula, the guarantee is met if:

$$(NPSHR)_G + (t_{NPSHR} \cdot NPSHR_G) \geq NPSHR_{measured}$$

or

$$(NPSHR)_G + (0{,}15 \text{ m and/or } 0{,}30 \text{ m}) \geq NPSHR_{measured}$$

3.3 Acceptance tests for side channel pumps

Side channel pumps are generally produced in quantity with published typical performance curves.

Although side channel pumps are not specifically covered by the Standard, Sterling SIHI uses the tolerance factors according to appendix A of the Standard for acceptance tests, (see table 3.04).

For all side channel pumps with an absorbed motor power less than 10 kW the tolerance factors according to table 3.05, are applicable.

For side channel pumps with an absorbed motor power less than 1 kW and special designs, the guarantee values and corresponding tolerance factors are to be specially agreed.

4 Special information for designing centrifugal pump installations

4.1 Pumping viscous liquids

4.1.1 Viscosity

Viscosity is a property that is exhibited by all material that is capable of flow (fluids). The range of these fluids is from gases, which are not covered here, through thin hydrocarbons to gelatinous and sticky gels.

Viscosity is the property that generates a resistance (internal friction) to relative movement between adjacent layers. The internal friction manifests itself in a velocity gradient *D* perpendicular to the direction of flow, i.e. adjacent layers have different velocities *v* and in such laminar flow a force *t* acts between these layers in the direction *x*.

The velocity gradient *D* is defined as the ratio of the velocity difference $\Delta v_x = v_{x2} - v_{x1}$ between two positions 1 and 2 and the distance between them Δy:

$$D = \lim_{\Delta y \to 0} \left(\frac{\Delta v_x}{\Delta y}\right) = \frac{dv_x}{dy}$$

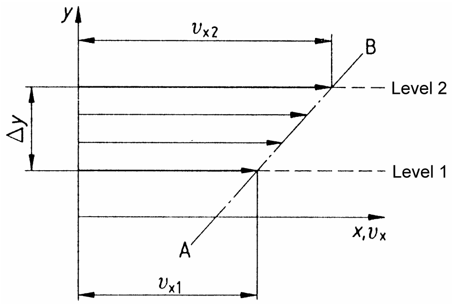


Fig. 4.1.01 Simple diagram showing viscous flow, velocity gradient and shear stress.

By plotting the velocity gradient *D* over the shear stress τ, the fluidity curve of the fluid is generated.

The viscosity curve is obtained by plotting the ratio of shear force / velocity gradient τ/D over the shear force τ or the velocity gradient D.

From the characteristic of the curve the fluidity and / or viscosity properties of the fluid can be read off and the type of fluid differentiated as follows:

4.1.1.1 Newtonian fluids

A Newtonian fluid is an isotropic linear viscous fluid which satisfies the following conditions:

a) Shear stress t and velocity gradient D are directly proportional

b) In the simple shear flow (see fig. 4.1.01) the normal stresses in the directions of the x axis, the y axis and vertical to that are equal.

Examples of Newtonian fluids are water and light oils.

The relationship between the shear stress τ and the velocity gradient D is given as:

$$\tau = \eta \cdot D$$

The proportionality constant η denotes this characteristic property of a liquid and is called the dynamic viscosity. The value of the viscosity is dependent on temperature, i.e. by rising temperature the viscosity reduces.

The ratio of dynamic viscosity η divided by the density ρ is known as the kinematic viscosity ν.

$$\nu = \eta / \rho$$

4.1.1.2 Non-Newtonian fluids

Non-Newtonian fluids are fluids and materials which have non-linear viscosity and materials (e.g. plastics) with linear and non-linear elasticity.

Fluids and materials which have non-linear viscosity are:

- Pseudo-plastic fluids

Non-linear pure viscous fluids, for which the viscosity reduces with increasing velocity gradient (see fig. 4.1.02a).

Examples of pseudo-plastic fluids are fats, molasses, paint, soap, starch and many emulsions.

- Dilatant fluids

Non-linear pure viscous fluids, for which the viscosity increases with increasing velocity gradient (see fig. 4.1.02b).

Examples of dilatant fluids are suspended solids, especially clay / water suspensions and dissolved sugars.

- Plastic materials

The behaviour of this material is characterised by limiting value, i.e. the material only begins to flow above the limit value, (see fig. 4.1.02 c). Below the limit value the material is either not deformed at all or only elastic deformation occurs.

There are several rheological models for this behaviour. The best known is the Bingham model.

An example of a Bingham fluid is tomato ketchup.

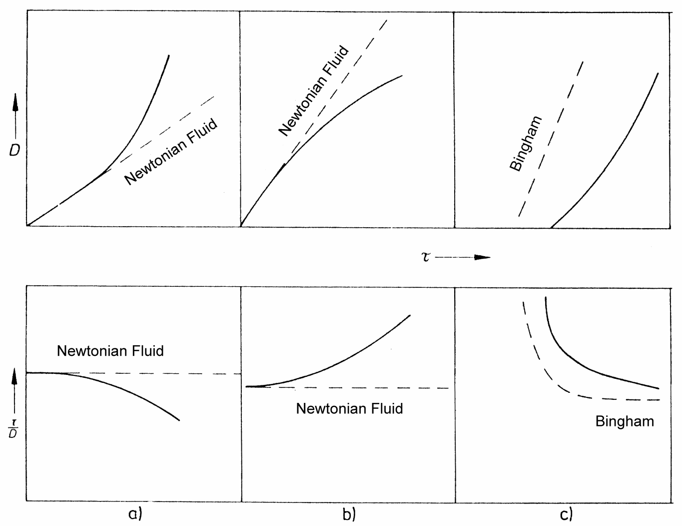


Fig. 4.1.02 Typical flow curves (top) and viscosity curves (bottom)

The flow behaviour of non-Newtonian fluids described above is always independent of time. However flow behaviour can be time dependent and these fluids are known as thixotropic or rheopectic.

Thixotropic is a time dependent flow behaviour in which the viscosity reduces from the stationary value to a lower limit as a result of a constant mechanical force. After removal of the force the viscosity is restored.

An example of a thixotropic fluid is non-drip paint.

Rheopectic is a time dependent flow behaviour in which the viscosity increases from the stationary value to a higher limit as a result of a constant mechanical force. After removal of the force the viscosity is restored.

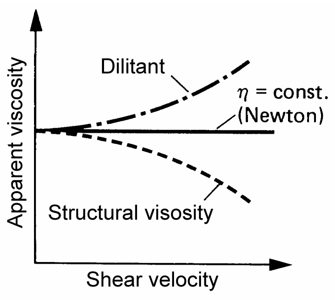


Fig. 4.1.03

Dependence of the viscosity on the shear velocity

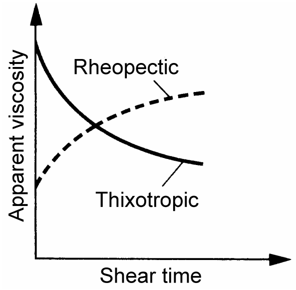


Fig. 4.1.04

Dependence of the viscosity on the shear time

4.1.2 The performance of centrifugal pumps with radial impellers pumping viscous liquids

4.1.2.1 General

The performance of centrifugal pumps will vary when viscous liquids are pumped. For medium and high viscosities, the power requirement increases considerably, whilst the head and to a lesser extent the flowrate, is reduced.

With the aid of diagram fig. 4.1.06, (section 4.1.2.3), the characteristics of a centrifugal pump pumping viscous liquids can be calculated providing the characteristic for pumping water is known. Conversely, the diagram may also be used to select a pump for given requirements.

The correction factors established from the diagram are sufficiently accurate for general application within the limits given. If more accurate values are required, then a test should be performed with the particular liquid.

Due to the considerable loss of efficiency when pumping viscous liquids when using centrifugal pumps, it is recommended that other types of pump be considered (e.g. rotary positive displacement pumps), which could give more economical running costs. The limits for centrifugal pumps are:

For discharge nominal diameter:

< 50 approx 120 to 300 mm²/s

< 150 approx 300 to 500 mm²/s

> 150 approx 800 mm²/s

Limitations and tips on the use of the diagram fig. 4.1.06:

- The diagram should only be used for centrifugal pumps with open or closed radial impellers within their normal *Q-H* range. The diagram must not be used for pumps with mixed flow or axial flow impellers, or for special pumps for viscous or heterogeneous liquids. For side channel pumps use section 4.1.3.
- The diagram should only be used if there is sufficient (*NPSH*) available (*NPSHA*) to prevent the influence of cavitation.
- The diagram can only be used for homogeneous Newtonian fluids. For gelatinous and sludgy liquids, liquids containing fibrous material and other heterogeneous liquids, widely scattered results are obtained in practice, depending on the special properties of the liquid.
- With multistage pumps, the head per stage must be used in the calculation.
- For pumps with double entry impellers, half the flowrate must be used in the calculation.

4.1.2.2 Selection of pump size for a viscous liquid

Approximation of an equivalent operating point for water:

Subscripts	vis	viscous liquid
	w	water
Given :	Q_{vis} in m³/h,	kinematic viscosity ν in mm²/s,
	H_{vis} in m,	ρ_{vis} in kg/dm³
Required:	to determine a suitable pump for which only performance data for water are known: Q_w in m³/h, H_w in m	
	To determine the driver power required: P_{vis} in kW	

The following procedure is used to establish the correction factors from the diagram:

Starting with the flowrate *Q* on the abscissa, move vertically upwards to intersect with the required head *H*, then horizontally (right or left) to intersect with the viscosity ν of the liquid, then vertically again to the intersections with the lines of the correction factors.

To establish the correction factor C_H for the total head, the curve $1{,}0 \cdot Q_{opt}$ is used.

This gives:

$$Q_w \approx \frac{Q_{vis}}{C_Q}, \qquad H_w \approx \frac{H_{vis}}{C_H}, \qquad \eta_{vis} \approx C_\eta \cdot \eta_w$$

Example: Q_{vis} = 100 m³/h, H_{vis} = 29,5 m, ν = 100 mm²/s, ρ_{vis} = 0,90 kg/dm³

The factors established from the diagram are:

$C_H = 0{,}94 \quad C_Q = 0{,}98 \quad C_\eta = 0{,}70$

With these factors the approximation for water is given:

$$Q_w \approx \frac{100 \text{ m}^3/\text{h}}{0{,}98} = 102 \text{ m}^3/\text{h}, \quad H_w \approx \frac{29{,}5 \text{ m}}{0{,}94} = 31{,}4 \text{ m}$$

For the pump to be used $\eta_w = 75\%$

Therefore $\eta_{vis} = 0{,}75 \cdot 75\% = 53\%$

$$P_{vis} \approx \frac{Q_{vis} \cdot H_{vis} \cdot \rho_{vis}}{367 \cdot \eta_{vis}} \approx \frac{100 \cdot 29{,}5 \cdot 0{,}90}{367 \cdot 0{,}53} \text{ kW} \approx 13{,}6 \text{ kW}$$

This procedure is to be considered as an approximation only, as the numerical values for flowrate and total head shown in the diagram apply to water. However in most cases this procedure is accurate enough for preliminary pump selection.

If the flowrate $Q_w < 0{,}9 \cdot Q_{opt}$ or. $> 1{,}1 \cdot Q_{opt}$ then the selection should be checked by the more accurate procedure described in the following section.

4.1.2.3 Establishing the characteristic of a pump for viscous liquids

Conversion of the characteristic for water:

The pumping characteristic for water gives the following: Q_{opt} , H_{opt} and η_{opt} . Starting from these values, the correction factors C_H (for 0,6, 0,8, 1,0 and 1,2 · Q_{opt}), C_Q and C_η can be established from the diagram using the procedure described in section 4.1.2.2.

For the conversion of the performance data it is convenient to use a tabular form, see example.

When drawing the characteristic it should be noted that the zero flow head H_0 remains about constant.

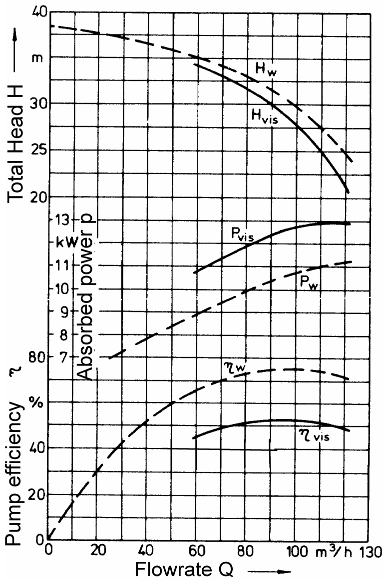


Fig. 4.1.05

Conversion of the characteristic for water

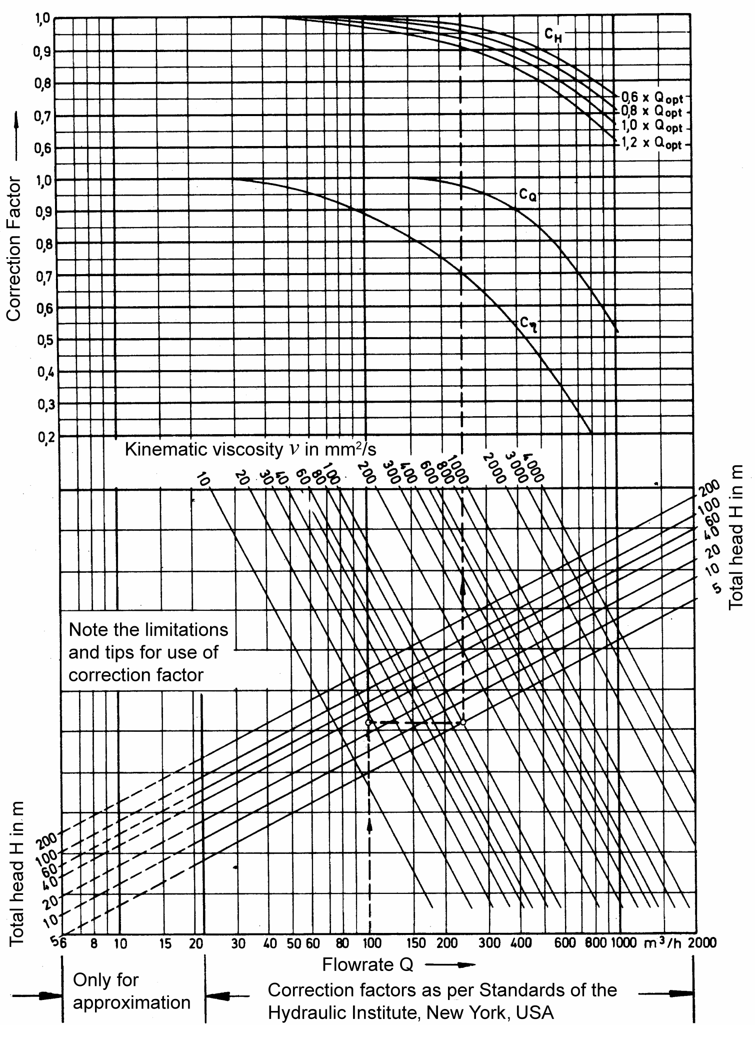


Fig. 4.1.06 Correction factors for Q, H and η for centrifugal pumps with radial impellers, pumping viscous liquids

Example of the conversion of an available pump characteristic for water to a characteristic for pumping a liquid with viscosity ν = 100 mm²/s from fig. 4.1.05.

Table 4.1.01 Conversion of the characteristic in table form (centrifugal pumps)

	$0{,}6 \cdot Q_{opt}$	$0{,}8 \cdot Q_{opt}$	$1{,}0 \cdot Q_{opt}$	$1{,}2 \cdot Q_{opt}$
Flowrate Q_w m³/h	60	80	100	120
Total head H_w m	35	33	29.8	24.5
Efficiency η_w %	65	73	75	71
Kinematic viscosity of the liquid ν mm²/s	100			
Correction factor H_w C_H	0.97	0.96	0.94	0.91
Correction factor Q_w C_Q	0.98			
Correction factor η_w C_η	0.70			
Flowrate $Q_{vis} = C_Q \cdot Q_w$	58.8	78.4	98	117.6
Total head $H_{vis} = C_H \cdot H_w$	34	31.7	28	22.3
Efficiency $\eta_{vis} = C_h \cdot \eta_w$	45.5	51.1	52.5	49.7
Density ρ_{vis} kg/dm³	0.90			
Absorbed power of pump $P_{vis} = \frac{Q_{vis} \cdot H_{vis} \cdot \rho_{vis}}{367 \cdot \eta_{vis}}$ kW	10.8	11.9	12.8	12.9

4.1.3 The performance of side channel pumps when pumping viscous liquids

4.1.3.1 General

The performance of side channel pumps also varies when pumping viscous liquids. However, due to the special internal flow conditions of these pumps, there are substantial differences between the characteristics of these and radial pumps when pumping viscous liquids (see section 4.1.2).

For Sterling SIHI side channel pumps, the characteristics applicable to pumping viscous liquids can be approximated with the aid of the diagram fig. 4.1.07, (section 4.1.3.2), provided that the water characteristic of the pump is known. Conversely, the diagram may also be used to select a pump for given requirements.

Limitations and tips on the use of the diagram:

- The diagram can only be used for homogeneous Newtonian fluids.
- The application limits of the pump e.g. the permissible absorbed power and the required (*NPSH*) value (*NPSHR*), should be considered using the manufacturers data.

4.1.3.2 Selection of pump size for a viscous liquid

Approximation of an equivalent operating point for water:

Subscripts	vis	viscous liquid
	w	water

The following procedure is used to establish the correction factors:

1. $Q_w = Q_{vis} = Q$
2. Q determines the model of pump to be selected and also gives Q_{max}. See table 4.1.02.
3. Starting from the value Q/Q_{max} on the abscissa in fig.4.1.07 the correction factors C_H for total head and C_P for absorbed power of the pump are established.

This gives:

$$H_w \approx \frac{H_{vis}}{C_H}, \quad P_{vis} \approx C_P \cdot \frac{\rho_{vis}}{\rho_w} \cdot P_w$$

The power absorbed figure P_{vis} is only to be considered an approximation. It is therefore recommended in selecting the driver to use a larger power addition figure than shown in section 1.7.4.

Example: $Q_{vis} = 3$ m³/h, $\nu = 150$ mm²/s

$H_{vis} = 60$ m $\rho_{vis} = 0.90$ kg/dm³

For $Q_{vis} = 3$ m³/h a Sterling SIHI pump from the range 3100 is suggested.

This gives: $Q_{max} = 6.2$ m³/h and $Q/Q_{max} = 0.48$

The correction factors: $C_H = 0.83$ and $C_P = 1.47$ are established from the diagram.

With these factors the data for water is given:

$$Q_w = 3 \text{ m}^3\text{/h and } H_w = \frac{60 \text{ m}}{0.83} = 72 \text{ m}$$

The absorbed power of this pump for water is given from the characteristic, $P_w = 1.9$ kW (with $\rho_w = 1.0$ kg/dm³) and from that

$$P_{vis} \approx 1.47 \cdot \frac{0.90 \text{ kg/dm}^3}{1.0 \text{ kg/dm}^3} \cdot 1.9 \text{ kW} = 2.5 \text{ kW}$$

Range	Q_{max} m³/h
1200	3.5
1900	4.4
3100	6.2
3600	8.7
4100	13.5
5100	24.6
6100	38.0

Table 4.1.02
Guide values for the Sterling SIHI side channel pump range

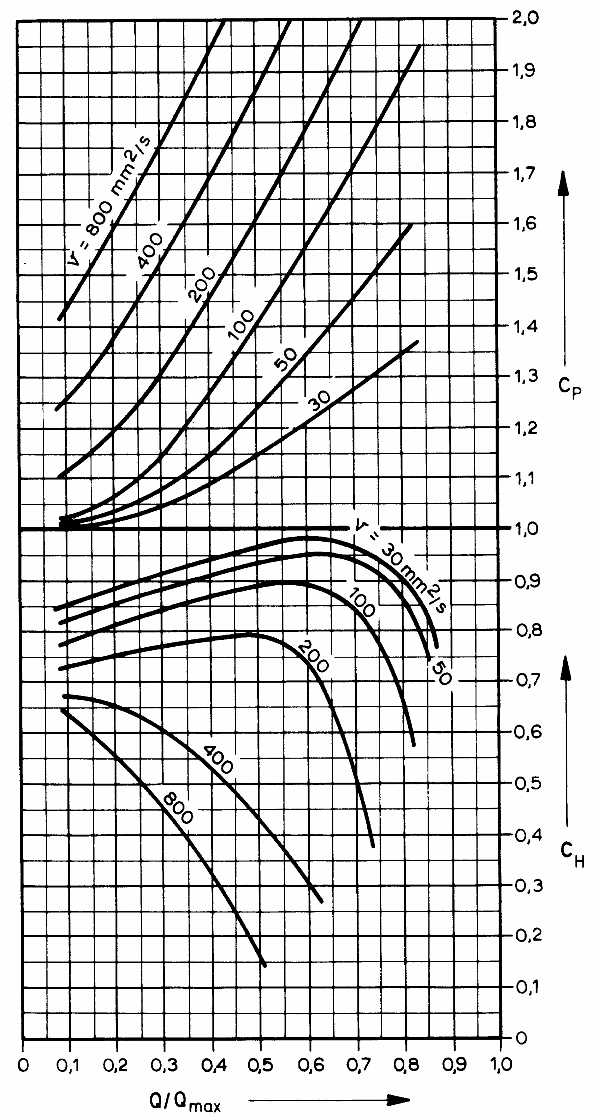
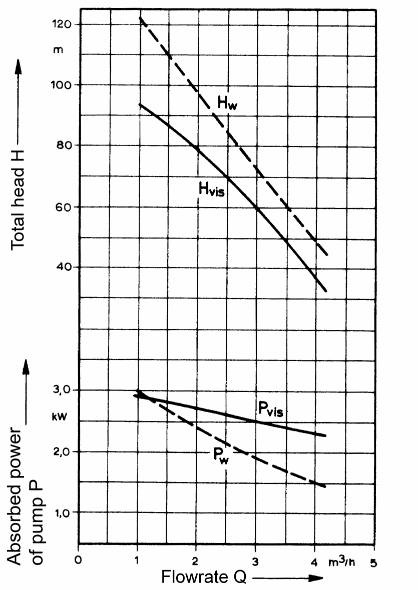


Fig. 4.1.07
Correction factors for the conversion of *H* and *P* for side channel pumps used with viscous liquids



4.1.3.3 Establishing the characteristic of a pump for viscous liquids

Conversion of the characteristic for water.

The conversion of the performance data is carried out in accordance with the procedure outlined in section 4.1.3.2; it is convenient to use a tabular form, see example.

Fig. 4.1.08 Conversion of the characteristic for water

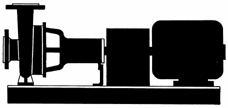
Example: 3-stage Sterling SIHI Pump from the range 3100

Table 4.1.03 Conversion of the characteristic in table form (side channel pump)

Flowrate	$Q_w = Q_{vis} = Q$	m³/h	1	2	3	4
Total head	H_w	m	122	98	72	49
Pump power absorbed	P_w (ρ = 1,0 kg/dm³)	kW	3.0	2.4	1.9	1.5
Q_{max} = 6,2 m³/h	Q/Q_{max}		0.16	0.32	0.48	0.65
Kinematic viscosity of pumped liquid	ν	mm²/s	150			
Correction factor for head	C_H		0.77	0.81	0.83	0.75
Correction factor for power absorbed	C_P		1.08	1.26	1.47	1.74
Total head	$H_{vis} = C_H \cdot H_w$	m	94	79	60	37
Density	ρ_{vis}	kg/dm³	0.90			
Pump power absorbed	$P_{vis} = C_P \cdot \rho_{vis} \cdot P_w$	kW	2.9	2.7	2.5	2.3

4.2 Design of the pump according to the installation

Baseplated pumps



Advantages:

adaptable to a selection of drivers and drive methods

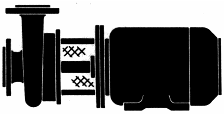
Disadvantages:

space requirement

precise alignment of driver and pump is necessary

cost of baseplate, coupling and guard

Close coupled pumps



Advantages:

reduced space requirement due to compact construction

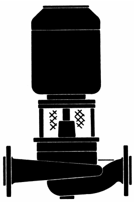
no alignment of pump and driver is necessary

no baseplate, coupling or guard is necessary

Disadvantages:

limited to drive by electric motor, up to a power of approx. 45 kW

Inline Pumps



Advantages:

direct installation in the pipeline is possible,
so minimal space requirement

no alignment of pump and driver is necessary

no baseplate, coupling or guard is necessary

Disadvantages:

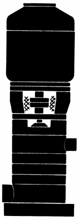
limited to drive by electric motor, up to a power of approx. 45 kW

Multistage Pumps

Advantages:

installation with piping from almost all directions
secondary discharge from one of the stages possible
accessories such as instrumentation, lubrication and seal flushing on the base plate possible
special high temperature installations with feet in plane of axis

Vertical pumps (multi)



Advantages:

minimum space requirement for multistage pumps

no alignment of pump and driver is necessary

no baseplate, coupling or guard is necessary

Disadvantages:

limited to drive by electric motor, up to a power of approx. 55 kW

Vertical pumps

Advantages:

direct installation in the container or sump is possible , so minimal space requirement

suction and delivery line not necessary

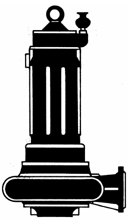
easy installation ready for operation

Disadvantages:

given adequate submersion it’s immediaetly ready for operation

driver must be above flood height

Submersible pumps



Advantages:

direct installation in the sump is possible

suction and delivery line not necessary

given adequate submersion it’s immediately ready for operation
special pump house not required

Disadvantages:

special submersible driver is required

operating temperature limited to 40 to 50 °C

Underwater pumps

Advantages:

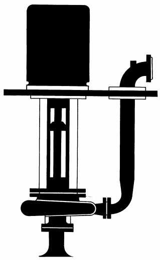
installation in narrow and deep boreholes possible without special drive arrangement

can be installed directly in pipeline as a booster pump

Disadvantages:

limited applications

Sump pumps



Advantages:

direct installation in the container or sump is possible

connections in the base of the container are not necessary removing safety problems for certain fluids

suction and delivery line not necessary

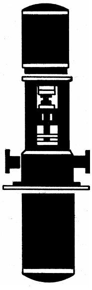
given adequate submersion it's immediately ready for operation

Disadvantages:

limited installation length

fluids with abrasive solids content require a special construction (cantilever) due to the inner bearing design

Canned pumps



Advantages:

by varying the can length and therefore the pump length, the suction head is varied increasing the value of (*NPSHA*)

even with poor suction conditions, no booster pump is nessary, increasing realiability

Disadvantages:

higher capital and installation costs

4.3 Design of suction and inlet pipes

To avoid air and gas pockets, suction pipes must be arranged horizontally or slope continuously upwards towards the pump. They must be completely leak free and be capable of being completely vented. If conical section reducers are necessary, they should be of the eccentric type. Inlet pipework which does not fall vertically to the pump must be arranged horizontally or slope continuously downwards toward the pump.

Sudden changes in the cross sectional area and sharp bends should be avoided. Badly designed suction and inlet pipework (e.g. bends in several planes immediately before the pump inlet) can substantially impair the pump performance.

For double entry pumps, it is essential that the flow into each side of the impeller is equal. For this reason a straight section of pipe, length at least 2x the diameter, is placed between any necessary bend and the suction flange of the pump, to equalise the flow.

If several identical pumps are connected to a common suction or inlet pipe, the pipework should be arranged in such a way that each pump has identical inlet flow conditions.

Right angle branches should be avoided, even where a straight section of pipe can be fitted before the pump. (Fig. 4.3.01).

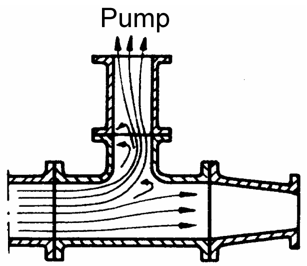


Fig. 4.3.01 Poor branch arrangement

More favourable flow conditions are achieved by swept branch connections, fig. 4.3.02 shows a satisfactory arrangement of inlet pipework for a two pump system.

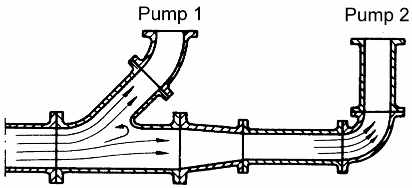


Fig. 4.3.02 Correct pipe arrangement for two similar pumps with a common inlet pipe

The velocity of flow should be kept within the following guidelines:

in the suction line	$U_S \approx$ 1.0 to 2 m/s	max. 3 m/s
in inlet line	$U_Z \approx$ 1,5 to 2,5 m/s	max. 3 m/s

Isolating valves in the suction or inlet pipes must be fully open during operation and should not be used for control or regulation.

Isolating valves in suction pipework should be mounted with the spindle horizontal or vertically downwards, so that air pockets in the spindle cover are avoided. The spindle seal should be adequate to prevent the in leakage of air into the valve.

If the pump is drawing from a sump and a suction strainer and valve cannot be fitted, then a bell mouth suction pipe should be fitted.

The positioning of the suction strainer or bell mouth from the sump floor and walls should be such that the liquid can enter uniformly from all directions, see fig. 4.3.03 and 4.3.04.

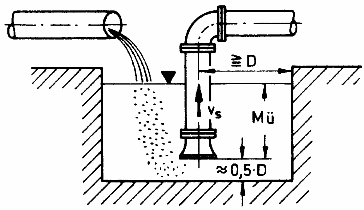


Fig. 4.3.03

Arrangement of a sump with open feed

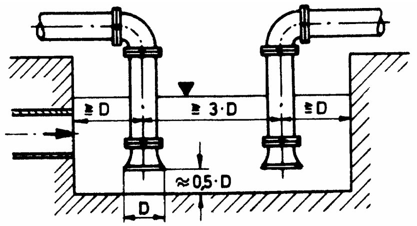


Fig. 4.3.04

Arrangement of a sump with two suction pipes

If the sump supply pipe discharges above the liquid level (as shown in fig. 4.3.03), there is the danger of air entrainment, which can impair the pump performance. The problem can be reduced by increasing the distance between the feed and the suction pipe to allow the air to escape from the liquid, or by providing baffle plates, or by selecting a relatively large immersion depth (Mü) as per fig. 4.3.05, although this may increase installation costs.

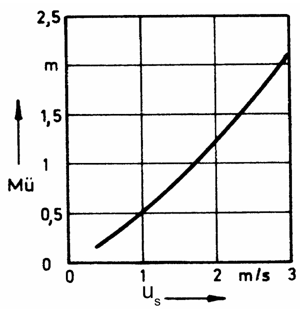


Fig. 4.3.05 Minimum submergence Mü for open feed to the sump according to Fig. 4.3.03

4.4 Design of the suction chambers for vertical pumps

4.4.1 General

The intake chamber of a vertically installed pump should be designed to ensure undisturbed flow to the pump for all operating conditions and all water levels. This is particularly important for pumps with high specific speed (mixed flow and axial flow) as these are more sensitive to irregular inlet flow conditions than centrifugal pumps.

The pump operation will be trouble free if the flow in to the pump impeller is swirl free and there is a uniform velocity profile across the entire cross section of the entry chamber. Furthermore the formation of air entraining vortices, in the intake chamber, must be prevented when operating at minimum liquid levels. If these conditions are not met, the flowrate and efficiency of the pump may be impaired. In severe cases, damage could occur due to vibration and cavitation.

4.4.2 Open intake chambers

If a single pump is installed in an intake chamber, then the principal dimensions may be selected from the guidelines in Fig. 4.4.01 . A channel of uniform cross section at least 5xD should be provided upstream of the pump. The flow velocity in this channel should not exceed 0.5 m/s.

The minimum submergence depth *Mü* is defined as the distance from the lowest edge of the suction bell mouth to the lowest inlet water level (NNW). For the installation of vertical pumps, no general guideline values can be given. This must be determined by the pump manufacturer for each individual application, see Fig. 4.4.01.

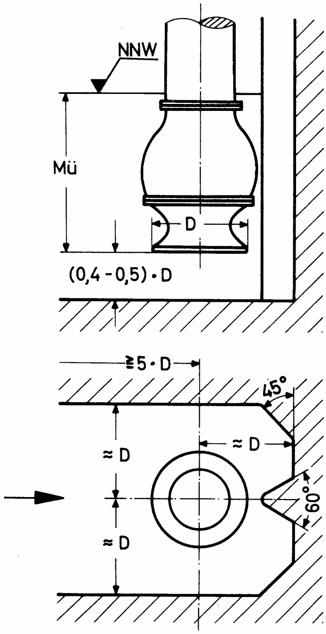
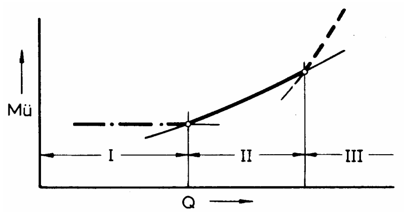


Fig. 4.4.01 Dimensions for intake chamber for a single pump

Fig. 4.4.02

Minimum submergence dependent on flowrate.



In the flow range I, the minimum submergence of pumps in wet installations, with bearings which are lubricated by the pumped liquid, ensures that the lowest bearing is lubricated during start up.

In this case *Mü* is determined by the mechanical design of the pump.

In the flow range II the minimum submergence must prevent the formation of air vortices which could enter the pump to be broken up by the impeller causing severe vibrations which could damage the unit. In this case *Mü* is determined by the flow velocity at the pump inlet.

In the flow range III, the (*NPSH*) required value (*NPSHR*) is the determining parameter. The minimum submergence must ensure that cavitation does not occur at any point in the pump.

If several pumps have to be installed in one intake chamber, separate bays for the individual pumps provide the best solution (Fig. 4.4.03).

If this is not possible, an arrangement similar to Fig. 4.4.04 should be used, whereby the spacing suggested are guidance values only. In difficult cases, baffle plates may have to be installed (Fig. 4.4.05), but their positioning should be agreed with the pump manufacturer.

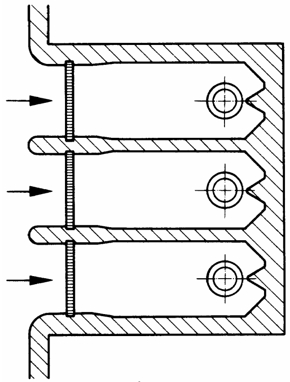


Fig. 4.4.03

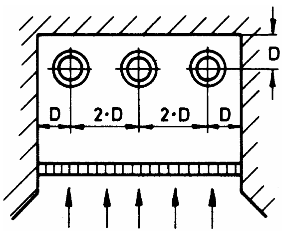


Fig. 4.4.04

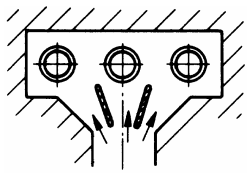


Fig. 4.4.05

Incorrect design of intake chambers:

- In the arrangements in Fig. 4.4.06 and 4.4.07, the liquid enters at one end of the suction chamber. The flow to the individual pumps is unequal and the pumps will affect each other.

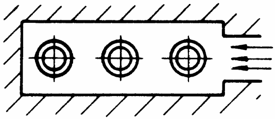


Fig. 4.4.06

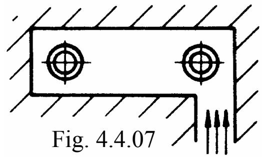


Fig. 4.4.07

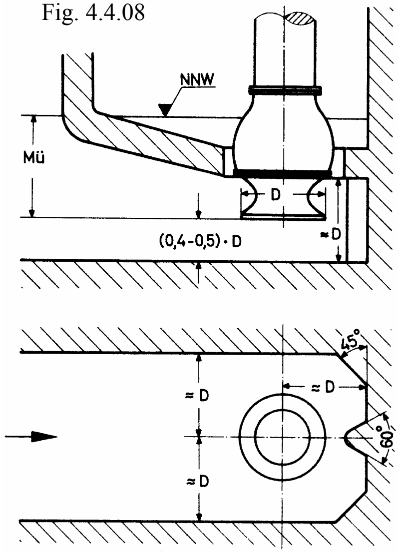
- Several pumps arranged non-symmetrically in one intake chamber
- Sudden expansion or contraction of the supply channel
- Insufficient length of supply channel of uniform cross sectional area
- Steps or pipes in the bed of the intake chamber, immediately before the pump
- Suction inlet too close to the bottom of suction chamber
- The sump supply pipe discharges above the liquid level so that air entrainment can impair the pump performance, see section 4.3.

4.4.3 Covered intake chambers

If for any reason the required length of supply channel ($l \geq 5xD$), which is required for trouble free operation cannot be achieved, then an alternative is to fit a sloping cover to the intake chamber. These covers are very effective in reducing swirl. Guidelines for the dimensions of these covers can be taken from Fig. 4.4.08, but final figures should be agreed with the pump manufacturer.

A cover with an appropriate profile can also provide the necessary acceleration of the inlet flow to achieve a more uniform velocity profile in open intake chambers where site conditions make changes in the angle of the inlet chamber side walls or slope of the bed before the suction bell mouth unavoidable.

Fig. 4.4.08



4.4.4 Inlet elbows

Minimum installation dimensions are achieved by using "turbine type" elbows , which are shaped to accelerate the flow (Fig. 4.4.09 and 4.4.10). If the flow velocity is accelerated by a factor of 4 to 5, then the length of the elbow (inlet section to centre of the pump) of $l_{Kr} \approx$ 4x impeller inlet diameter is sufficient to achieve uniform velocity distribution.

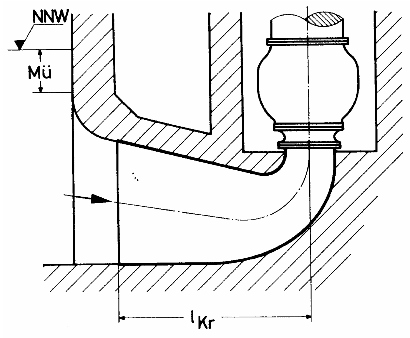
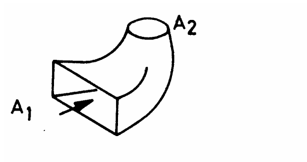


Fig. 4.4.09 Accelerating elbow

The inlet cross sectional area A_E of the elbow should be large enough to ensure that the velocity at inlet is low enough to prevent air entrainment or bubble formation.



A_1 = (4 to 5) x A_2

Fig. 4.4.10 Cross sections

In each case a cost effectiveness assessment should be carried out to compare the higher constructional costs of an inlet elbow compared to an inlet chamber. The design and construction of an inlet elbow is often more complex and can require much deeper excavations.

4.5 Priming centrifugal pumps prior to start up

4.5.1 General

In general, centrifugal pumps have to be filled with liquid prior to starting up (i.e. primed). In installations where the liquid flows to the pump (flooded suction), care must be taken to ensure that the casing of the pump is adequately vented. Priming pumps with a static suction lift can be more difficult. In contrast to positive displacement pumps, standard design of centrifugal pumps if not primed, can produce virtually no suction lift. They are therefore incapable of evacuating the suction line and their own casing and care must be taken using other means to achieve this.

A distinction should be made between self priming centrifugal pumps and non-self priming centrifugal pumps with external priming devices.

4.5.2 Self priming pumps

Self priming is a term used to describe pumps which are capable of priming their suction pipe without the use of external devices, i.e. which are capable of pumping air (gas) if the pump has previously been filled with liquid.

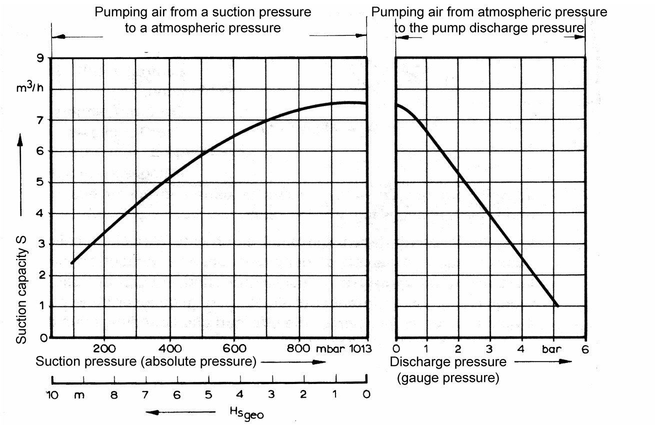


Fig. 4.5.01 Suction capacity of a side channel pump

The best known types are the side channel pump (star vane impeller) and the radial flow centrifugal pump with built in ejector. Fig. 4.5.01. shows the characteristic suction capacity curve of a side channel pump, pumping air. During the suction operation, the pump operates in this range until the liquid enters the pump due to the suction lift created. For a short time a mixture of both gas and liquid are pumped until eventually the full liquid flowrate, for which the pump is designed, is reached. The sequence of operations from priming to full liquid flow, proceeds automatically without any external influence.

The design of the branch connections and the configuration of the pump internals, ensure that when the pump is stopped, the backflow of liquid does not empty the pump completely (siphon out). Sufficient liquid remains in the pump to restart the self priming action at any time, even without a foot valve in the suction line. The "self priming capacity" indicates the maximum suction lift which the pump can re-prime and re-establish full pumping, after stopping and allowing the suction line to drain down.

The self priming feature increases the reliability of operation, particularly where immediate availability is required on intermittent operation, where the pumping is arranged from lower lying wells or vessels or the suction pipe is laid over ground that rises and falls.

As the economic installation of side channel pumps is usually limited to lower flowrates (up to 35 m^3/h), compound self priming pumps are used for higher duties. These are single or multi-stage centrifugal pumps with radial impellers and an integrated side channel stage, which is arranged in parallel with the first or last radial impeller. The radial stages pump the liquid when the efficiency is high and the side channel stage enables the self priming capacity and the pumping of entrained gases.

4.5.3 Non-Self-priming pumps

Where a non-self-priming pump operates under suction lift conditions, the pumping operation can only be started when the pump casing and the suction pipe is filled with liquid. A foot valve arranged in the suction pipe will permit filling from an external source. If this is not possible the pump and suction pipe must be evacuated by means of an external priming pump with the discharge closed by an isolating valve. Liquid ring vacuum pumps are generally used for this operation, although occasionally self priming side channel pumps may be used.

4.5.4 Design of priming pumps

Suction lines rarely consist of a simple vertical pipe, they normally include horizontal and vertical (or sloping) sections. It can be assumed with reasonable accuracy, that the pressure in the entire suction pipe only reduces during the priming of vertical or sloping sections and during priming of horizontal sections it remains constant. Different formulae are therefore used to calculate the priming requirement of horizontal and vertical (or sloping) sections.

For rising (vertical or sloping) pipe sections

$$S \cdot t = 60 \cdot V_{\text{rise}} \cdot \left(2 - \frac{p_E}{p_A - p_E} \ln \frac{p_A}{p_E}\right)$$

For horizontal pipe sections

$$S \cdot t = 60 \cdot V_{\text{horiz}} \left(\ln \frac{p_A}{p_E} + 1\right)$$

with	S in m³/h	=	Suction capacity
	t in min	=	Evacuation time
	V_{rise} in m³	=	Volume of (vertical or sloping) pipe sections
	V_{horiz} in m³	=	Volume of horizontal pipe sections including the centrifugal pump
	p_E in bar	=	Absolute pressure at the priming pump suction branch when the suction pipe is fully primed with liquid
	p_A in bar	=	Absolute pressure in the suction pipe when the evacuation commences

To make allowance for minor leakage, head loss in the priming pump pipework and the influence of entrained gases in the liquid, it is advisable to use only 90% of the priming pump suction capacity in the formula, or to increase the calculated suction capacity required by 10%, when selecting the pump.

An approximation of the size of a suitable priming pump for a centrifugal pump which has to lift water from an open chamber is easily calculated by means of the following equations:

Assumptions: That the suction capacity of the priming pump is constant

The losses mentioned above have been taken into account by the coefficients k_1 and k_2 (see Fig. 4.5.04.)

Atmospheric pressure p_{amb} = 1013 mbar

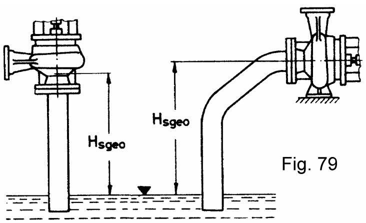


Fig. 4.5.02.

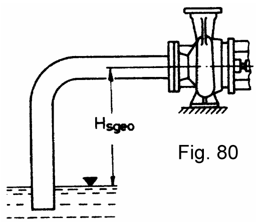


Fig. 4.5.03

a) Suction capacity and priming time

Suction pipe vertically upwards or sloping upwards (Fig. 4.5.02)

$$S \approx \frac{k_1 \cdot V_o}{t} \quad \text{in m}^3/\text{h}$$

or

$$t \approx \frac{k_1 \cdot V_o}{S} \quad \text{in min}$$

Suction pipe with rising and horizontal sections (Fig. 4.5.03)

$$S \approx \frac{k_1 \cdot V_{rise} + k_2 \cdot V_{horiz}}{t} \quad \text{in m}^3/\text{h}$$

or

$$t \approx \frac{k_1 \cdot V_{rise} + k_2 \cdot V_{horiz}}{S} \quad \text{in min}$$

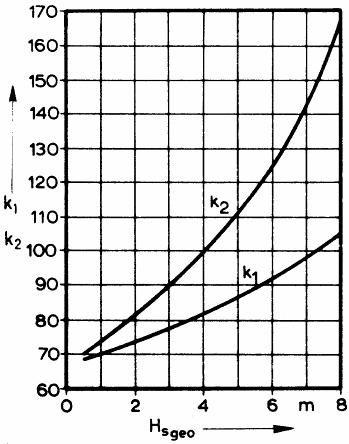


Fig. 4.5.04

Factors to calculate the required suction capacity of priming pumps

k_1 and k_2 see Fig. 4.5.04

V_o in m³ = Volume of the suction line including the centrifugal pump

V_{rise}, V_{horiz}, S and t see previous section

b) Suction pressure

Suction pressure = 1013 – 98 · H_{sgeo} in mba with H_{sgeo} in m

Example:

Given: rising section of the suction pipe with $V_{ansteig}$ = 0.055 m³

horizontal section of the suction pipe including the centrifugal pump with V_{horiz} = 0,17 m³

H_{sgeo} = 7 m

required priming time t = 1 min

A priming pump with approximately the following suction capacity is required:

$$S \approx \frac{k_1 \cdot V_{rise} + k_2 \cdot V_{horiz}}{t} = \frac{97 \cdot 0.055 + 143 \cdot 0{,}17}{1} = 29{,}6 \text{ m}^3/\text{h}$$

Suction pressure = 1013 – 98 · H_{sgeo} = 1013 – 98 · 7 = 327 mbar

A liquid ring vacuum pump with a mean capacity between the suction pressure and atmospheric pressure p_{amb} of $S \approx 42$ m³/h is selected. This satisfies the requirement for a 10% safety margin and the priming time is reduced to

$$t = \frac{29.6 \text{ m}^3/\text{h}}{(42 \text{ m}^3/\text{h} - 10\%) = 37.8 \text{ m}^3/\text{h}} \cdot 1 \text{ min} = 0.78 \text{ min}$$

4.6 Pumping liquid / gas mixtures

4.6.1 General

Whilst centrifugal pumps are primarily selected for pumping liquids, the handling of undissolved gases and vapours cannot be excluded. Air entrainment can for example occur due to insufficient suction bellmouth submergence when pumps draw from open chambers (see section 4.3). Air can also leak in through flange joints in the suction pipework, past valve spindle seals and possibly the pump shaft stuffing box. This air entry is difficult to control and is undesirable, leading to loss of performance and indeed interruption of pump flow.

The requirements of process plant are different in that often the pump is required to handle gases and vapours from the process without loss of function. Pumps which are handling liquids close to their vapour pressure (condensate, liquefied gases etc.) face special demands. The generation and growth of gas or vapour bubbles from the pumped liquid is to be expected when high suction lift or throttling due to a series of fittings in the suction pipe, has to be overcome, or an increase in the temperature of the liquid occurs, due to poor insulation of the suction pipework.

It is therefore important to take into consideration the operational characteristic and application limits of the different types of centrifugal pumps when liquid / gas mixtures are pumped. The effect of the mixture on the pump characteristic is dependent on the relative proportions q_{Gs} of gas to liquid as follows:

$$q_{Gs} = \frac{Q_G}{Q_F}$$

Q_G = Gas flowrate

Q_F = Liquid flowrate

4.6.2 Operational characteristics of non-self priming pumps

Non-self priming pumps are only able to handle a limited amount of gas in the liquid pumped. For centrifugal pumps with radial impellers and standard design, this limit is approximately 5-7% of gas by volume. Pumps with open or unshrouded impellers can handle a higher gas content of up to 10%.

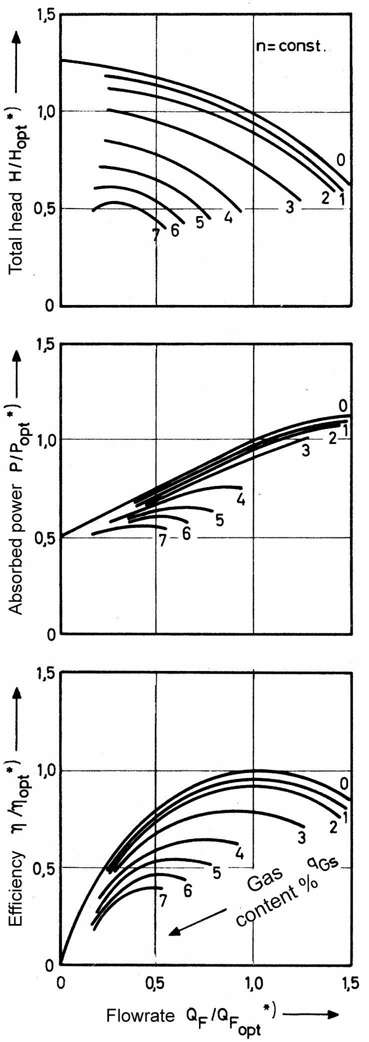


Fig. 4.6.01

The influence of the gas content q_{Gs} on the characteristic of a centrifugal pump with radial impeller

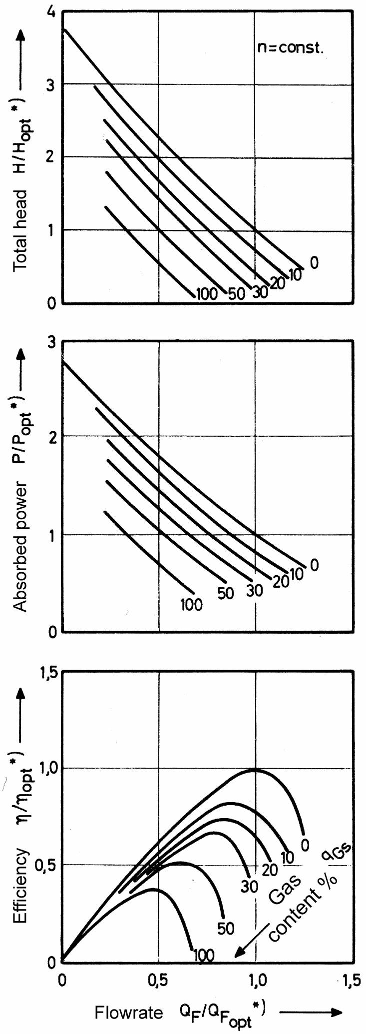


Fig. 4.6.02

The influence of the gas content q_{Gs} on the characteristic of a self priming side channel pump

Q_{Fopt}, H_{opt}, P_{opt}, η_{opt} : data applicable at the point of maximum efficiency at $q_{Gs} = 0$

Fig. 4.6.01 shows the influence of entrained gas on the characteristic of a single stage centrifugal pump with a radial impeller. With a gas content of 7%, the flowrate and delivery head at the point of maximum efficiency are reduced by approximately half. This sensitivity of radial impellers is caused by the fact that a stable, gas filled dead space is created at the hub and the volume of this pocket increases as a function of the gas content and the flow conditions until it fills the impeller entry and leads to collapse of pumped flow. Automatic restarting of the pumping will only occur under certain inlet flow conditions. With small flowrates, instability can start with quite small gas content percentages.

The characteristics of multi-stage pumps can be derived approximately from the performance of the single stage pumps if it is considered that the relative volume of gas reduces by the pressure ratio of the preceding stage. The limiting values for multi-stage pumps are determined by the first stage.

4.6.3 Operational characteristics of side channel pumps

Self priming side channel pumps, which are a special type of self priming centrifugal pump, are capable of pumping large gas flows with the liquid when operating in steady state conditions. In the extreme case, during the evacuation of the suction pipe, self priming side channel pumps handle gas only. Between this condition and that of pumping liquid only, the range of gas / liquid mixtures which occur in practice, can be handled without any external auxiliary equipment. Fig. 4.6.02 shows the influence of the gas content q_{Gs} on the characteristic of a single stage side channel pump. Gas contents of e.g. 10% which would lead to the collapse of the pumped flow of centrifugal pumps with radial impellers, have only a small effect on the characteristics of side channel pumps.

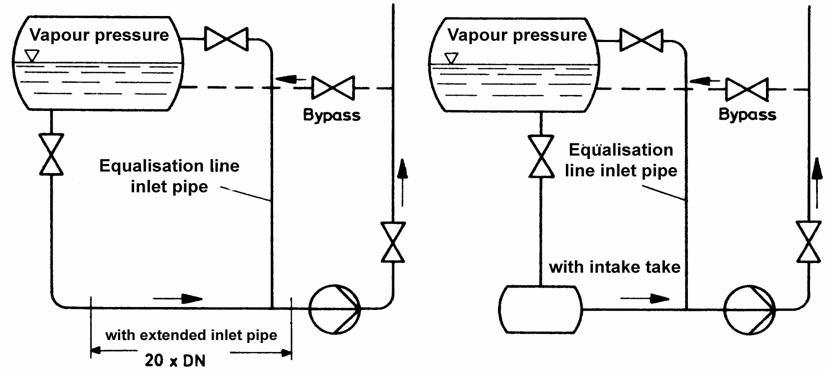
The characteristics of multi-stage side channel pumps can be derived approximately from the performance of the single stage pumps if it is considered that the relative volume of gas reduces by the pressure ratio of the preceding stage.

Compound side channel pumps using a radial impeller for the first stage, follow almost the same principles as pure side channel pumps. As a result of their low *NPSHR* requirement, these pumps are often preferred for applications where liquids are pumped which are close to their vapour pressure (condensate, liquified gas etc.).

To keep the gas or vapour content as low as possible at the pump inlet and so minimise the reduction in performance of the pump, the following recommendations should be observed when designing the installation:

- Head losses in the inlet pipework are to be avoided or reduced to a minimum.
- A gas balancing pipe should be located between the pump inlet and the inlet vessel.
- An extended inlet pipe approximately 20x DN (see Fig. 4.6.03a) or an intake tank (see Fig. 4.6.03b) should be fitted upstream of the pump inlet.

- A bypass connection should return to the inlet tank and not the inlet pipe.
- The inlet pipe should be run as close as possible to the pump level. Horizontal sections, which are sloped down to the pump to aid degassing, should first be brought to the pump level to gain the full *NPSHA* of the plant and so prevent generation of gas as a result of pipe losses.
- The complete installation should be protected by roofing or insulation to prevent heat absorption by radiation.



a) with extended inlet pipe b) with intake tank

Fig. 4.6.03 Schematic of a condensate installation

4.7 Pressure surges (water hammer) in piping systems

4.7.1 General

In systems with long discharge lines, (e.g. industrial and municipal water supplies, refineries and power stations), if the flow is accelerated or decelerated, then pressure fluctuations occur due to the changes in velocity. If these velocity changes take place rapidly, they propagate pressure surges in the pipe system, which originate at the point of disturbance and travel in both directions (direct waves). They are reflected at points of discontinuity, (e.g. changes of cross sectional area, branches, control or isolating valves, pumps or vessels) and depending on the boundary conditions, these reflections (indirect waves) cause negative or positive pressure surges. The combined effect of all these direct and indirect waves produces the prevailing condition at a particular point and time.

These pressure surges added to the maximum working pressure can lead to excessive pressure and stress on the components of the system. In severe cases, this can lead to failure of pipework, fittings or pumps. A minimum pressure surge may, particularly at the highest point of the installation, fall below the vapour pressure of the pumped liquid, causing evaporation and voids in the liquid column. The subsequent pressure increase and reuniting of the separated liquid column can lead to considerable water hammer. The pressure surges resulting from this can also lead to damage or failure of the components of the installation.

For the maximum pressure fluctuation, the JOUKOWSKY pressure surge formula can be used:

$$\Delta p = \rho \cdot a \cdot \Delta U$$

where ρ = density of the pumped liquid

a = velocity of wave front

ΔU = change of velocity of flow in pipe

The full pressure fluctuation, corresponding to the change of velocity ΔU, occurs only if that velocity change takes place in the time

$$t < \text{reflection time } \; t_r = \frac{2 \cdot l}{a}$$

where l = distance between the next discontinuity (reflection point) and the point of disturbance.

The velocity of the wave front *a* is mainly a function of:

- the density and modulus of elasticity of the pumped liquid.
- the dimensions of the pipe (diameter, wall thickness) and the pipe supports.
- the modulus of elasticity of the pipe material.

As a mean value, a = 900 to 1300 m/s for water as the pumped liquid, with cast iron, steel or concrete pipes. Because of the non-linear elasticity behaviour of plastic pipes with time, an approximation value of a = 300 to 500 m/s is applied for the most commonly used plastics.

Further, a knowledge of the rate of change of the velocity Δv is important in the evaluation of pressure increase, pressure surges and possible development of oscillations.

As an example, the closing of gate valves, throttle valves or similar is often used. It can be shown that effective throttling only takes place during the last 10 to 20% of the valve movement. This means that such valves can be closed up to 80 or 90% in as short a time as required, without causing a dangerous increase in pressure. The last part of the movement however must be effected more slowly, to suit the parameters of the pipe system.

Calculation of pressure surges can be very complicated, particularly in complex networks. However it cannot be ignored in long piping systems to determine if surge suppression equipment is necessary.

In most cases the development of water hammer in pipes can be calculated sufficiently accurately using partial similarity calculations, (continuity and movement comparisons). The analysis of these similarity results can be made using the graphical (Schneider - Bergeron) method, or on a computer using the characteristic or impedance method.

4.7.2 Causes of pressure surges

In addition to considering basic data for the calculation of pressure surges, e.g. starting and stopping the pumps, opening and closing of control and isolating valves, changes of pump speed etc., it is also necessary to take into account the unusual demands caused by abnormal and dangerous operating conditions.

4.7.2.1 Interruption of the electrical power supply

- Failure of electrical supply.
- With the loss of drive, the pump runs to a standstill, dependent on the moment of inertia of the rotating parts.
- Failure of the control voltage for the operation of isolating valves can cause incorrect valve actuation.

4.7.2.2 Failures within the installation

- Faulty operation of control or isolating valves.
  Due to failure of the damping system the valve can close suddenly rather than smoothly as designed.
- Air in hydraulic lines.
  Entry or accumulation of air in the hydraulic control lines will prevent normal operation of the control functions.
- Failure of air bleed or venting valves.
- Air locks in the pipework.
- Air escaping from openings.
  The change over from air to water can set up oscillations.

4.7.2.3 Defective components in the installation

- Valve flutter.
- Incorrect installation of isolating valves.
  Turbulence downstream of bends can lead to flow separation in fittings , which can cause oscillations and pressure surges.
- Pipe breaks.

4.7.2.4 Incorrect operation of the pumps

- Priming.
  If insufficient care is taken during filling of the discharge line, a severe pressure surge can occur.
- Dead pipe sections.
  If one section of a branched system is closed off whilst the other side remains connected to the network, unexpected pressure surges may occur.

4.7.3 Preventative measures (pressure surge control)

To prevent unacceptably high and low pressures in the pipe system, the following measures can be taken:

- Design of the discharge line for low flow velocities.
- Increase of the inertial mass and hence the run down time of the pump set by use of a flywheel.

- Installation of equipment to supply liquid to the piping system during starting and stopping, e.g. an air/liquid bladder tank, a suction reservoir, a by-pass back to the suction chamber or a stand pipe at the start of the discharge line.
- Correct selection of the opening and closing times and the closing characteristics of the control and isolating valves.
- Reduction of the reflection time t_r of the system, by using the shortest pipe runs as possible, or where long runs are necessary, by installation of intermediate reflection points, e.g. a surge chamber at the highest point.
- Installation of vacuum relief valves at points in the system where evaporation of the liquid could occur due to low pressure surges.
- Installation of equipment to relieve liquid accumulations, e.g. additional outlets or relief valves.

4.8 Forces and moments on flanges

The pump is connected to the pipework of the installation at the suction (inlet) and discharge (outlet) flanges.
When connecting the pipework, care should be taken to ensure that it imparts as little force as possible onto the pump.
There is a limit to the external forces and moments which the pump flanges and casing can accept. If the forces and moments are too high, there is a danger of distortion and overloading of the pump casing. As a result, the pump impeller can pick up on the casing, or the clearance ring or throttle bush. Furthermore the coupling alignment could be affected, which could lead to failure of the pump bearings or the coupling itself. Additionally there is also a danger of overloading the bolts holding the pump down on the baseplate.
Pump suppliers therefore state maximum allowable forces and moments which can be exerted on the pump flanges using a three dimensional co-ordinate system.

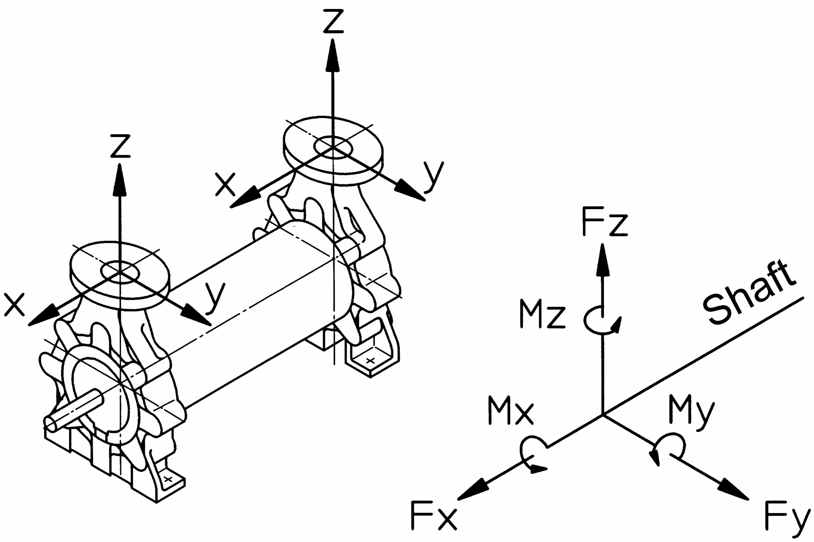


Fig. 4.8.01
Notation of the allowable forces and moments on the pump flange, in the three dimensional co-ordinate system (example, horizontal multistage split case pump).

The design and installation of the pipework should ensure that these maximum allowable values, as stated by the manufacturer, are not exceeded during operation, when under the loads, imparted by operation at maximum over-pressure and operating temperature.
The allowable values for the forces and moments which can be exerted on the pump flanges, can be taken from various references and standards, e.g. ISO 5199 & 9905, dependent on the type of pump construction, materials, type of installation and frame size, regardless of the manufacturer.
Unless otherwise stated, the values for the forces F and moments M are valid for a particular material and maximum operating temperature up to 100 °C.
For other materials and higher operating temperatures, the values can be corrected in relation to their modulus of elasticity ratios as follows:

$$F_t \text{ or } M_t = F_{20°C} \text{ or } M_{20°C} \cdot \frac{E_{t,m}}{E_{20,b}}$$

with: $E_{20,b}$ = Modulus of elasticity of basic material at 20 °C.
$E_{t,m}$ = Modulus of elasticity of selected material at operating temperature.

Table 4.8.01 Modulus of elasticity in kN/mm² for various materials and operating temperatures.

Material	Temperature in °C					
	20	100	200	300	350	400
Grey cast iron	112	110	103	98		
Spheroidal graphite cast iron (ductile iron)	169	159	153	144	139	
Cast steel, non- alloy or low alloy	211	204	196	186	182	177
Chrome steel with approx 12% Cr	216	209	200	190	185	179
Austenitic and austenitic/ferrous steel	200	194	186	179	176	172
Cast tin bronze	100					

4.9 Pumping suspended matter

4.9.1 Suspended matter and stocks

Suspended matter is generally understood to mean cellulose / fibre / water mixtures. Primarily cellulose mixtures concern wood pulp, paper, straw and similar materials. This raw material may be boiled, shredded and bleached according to its nature to create the pulp.

Depending on the required paper product, the raw material is mixed with water, colour, fillers and size in milling machines to produce the stock.

During the production process, the intermediate products are present as suspensions of varying concentrations and consistency.

The stock consistency is defined by the mass ratio of solids within the suspension.

$$\text{Consistency } w_{\text{Solids}} = \frac{m_{\text{Solids}}}{m_{\text{Solids}} + m_{\text{Water}}} \cdot 100 \quad \text{in \% bone dry or \% air dry}$$

$$\text{Consistency } w_{\text{Solids}} = \frac{m_{\text{Solids}}}{m_{\text{Suspension}}} \cdot 100 \quad \text{in \% bone dry or \% air dry}$$

where:

% bone dry = Mass percentage of absolutely dry solids in the suspension .
% air dry = Mass percentage of air dry solids in the suspension.

Air dry solids are defined as containing 12% water, i.e. contain 88% absolutely dry material.

1% bone dry = 1.14% air dry

1% air dry = 0.88% bone dry

The capacity of cellulose and paper production plants is generally given in tons/day, i.e. tons of bone dry or air dry stock per 24 hours. To calculate the required pump capacity from the tons/day, the following formula can be applied:

$$Q = \frac{\text{t}_{\text{bone dry}}}{24\ \text{h}} \cdot \frac{4.17}{w_{\text{Solids}}\ \text{\% bone dry}} \quad \text{in m}^3\text{/h}$$

or

$$Q = \frac{t_{\text{air dry}}}{24\text{ h}} \cdot \frac{3.72}{w_{\text{Solids}}\,\%\text{ air dry}} \qquad \text{in m}^3\text{/h}$$

In the UK and USA the consistency is generally expressed as:

O.D. = Oven Dry Stock (water-free) = i.e. % bone dry (B.D.)

A.D. = Air Dry Stock

For the latter a water content of 10% is defined and is therefore not directly comparable with % air dry as defined above.

O.D. concentration = 0.90 · A.D. concentration

A.D. concentration = 1.11 · O.D. concentration

4.9.2 Air content in suspended matter

When the stock comes into contact with the air, it is not possible to prevent small air bubbles from adhering to the fibres. If the fibre density is very high, this can result in a high air content, which can noticeably reduce the total head of the pump. It is therefore important to ensure, that during the processing of the stock, as little contact as possible takes place with the air and that the entry to the pump should be designed to prevent air being drawn in. The influence of air content on the total head of the pump can be estimated as shown in section 4.6.01.

4.9.3 Pipe friction losses

The flow behaviour of suspended matter in pipelines is vastly different from that of water or other Newtonian fluids. This leads to different characteristics for pipe friction loss as shown in Figs. 4.9.01 and 4.9.02.
Careful examination of the characteristics can reveal different ranges of flow behaviour as indicated in Figs. 4.9.03 and 4.9.04.
These ranges can be described as:

Range 1 - section A-B of curve.
In this range the relationship of the losses and the flow velocity are linear up to the velocity U_1 .

Range 2 - section B-C-D of curve.
In this range the flow losses reduce to point C, then increase again to point D where the curve intersects that of water.

Point D is identified as the point where “drag reduction” commences. The flow velocity at this point is called U_2.

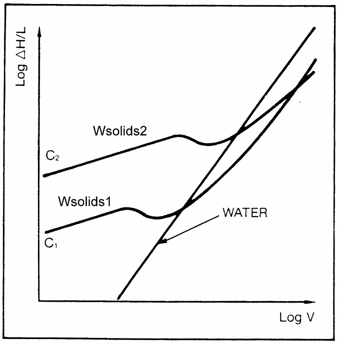


Fig. 4.9.01 Pipe friction loss curve for chemically processed stock (cellulose) $w_{\text{Solids2}} > w_{\text{Solids1}}$

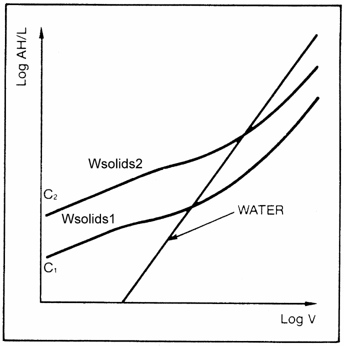


Fig. 4.9.02 Pipe friction loss curve for mechanically processed stock (wood pulp) $w_{\text{Solids2}} > w_{\text{Solids1}}$

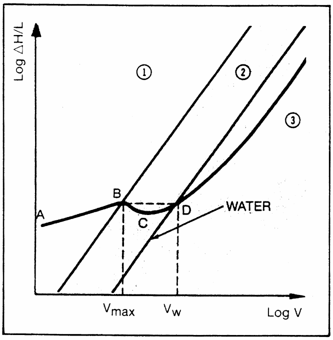


Fig. 4.9.03 Pipe friction loss curve for chemically processed stock

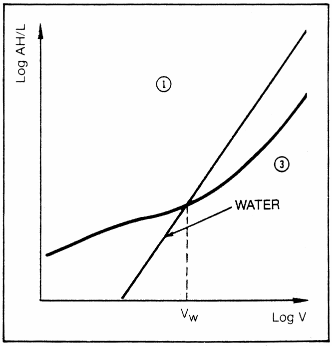


Fig. 4.9.04 Pipe friction loss curve for mechanically processed stock

Range 3 - Section D-E of curve
In this range the friction loss curve for the suspended matter lies under that for water. The reason for this is the so called “drag reduction” phenomena of the flow behaviour of suspended matter.

The guideline values for flow velocity, which are used in practice, of max. 3.1 m/s for suspended matter up to 3% bone dry and max. 2.4 m/s for suspended matter >3% bone dry give the following picture:
For chemically prepared suspensions with consistency between 1.5 and 2% bone dry, the pipe losses lie in the Range 3 (section D-E) and the pipe losses curve for water can be applied in general. For consistencies between 2.5 and 4.5% bone dry, the pipe losses lie in the Range 2 (section B-C-D) and between 5 and 6% in the Range 1 (section A-B).
For mechanically prepared suspensions, the pipe losses, within the above guideline flow velocities, lie in the Range 1 (section A-B), for all consistencies.
The loss of head, which is dependent on other factors as well as the consistency and flow velocity, e.g. the method of preparation of the stock, the temperature and the material of the pipeline, can be calculated by various methods. One of the most commonly used is the TAPPI technical information sheet (TIS) 408-4 which is published by the Technical Association of the Pulp and Paper Industry, Atlanta Georgia, USA and is also available for certain parameters in curve form.

4.9.4 Pumps for handling suspended matter

The best results when handling suspended matter are achieved by centrifugal pumps. For consistencies up to 1.5% bone dry, standard centrifugal pumps can be utilised. Higher consistencies require centrifugal pumps which have been specifically designed to meet the hydraulic and constructional requirements of the suspended matter.
Centrifugal pumps with semi-open impellers, which are distinguished by their ample flow chambers and low flow velocities, can be used for suspension consistencies up to 6% bone dry, without problem and without deviating from the characteristic for water. This is however conditional on the air content being low, not more than 1-2% by volume. If the air content exceeds this value, then the total head of the pump is reduced as described in section 4.9.2.
When selecting a pump for suspended matter, the requirements of the plant in terms of flowrate and total head should be observed as closely as possible. It is not advisable to throttle an oversized pump to meet the plant requirements, as the high flow velocity which occurs at the throttle plate leads to separation of the water and can cause formation of lumps of material. These lead to vibration in the pipeline which in turn is transmitted to the pump.
Adjustment of performance to match operating requirements should be achieved either by impeller trim or, in the case of frequently changing conditions, with bypass control.

The minimum flow should not fall below 25% of the flowrate at the point of optimum efficiency.
It should be noted that pumps for suspended matter will not generate negative pressure, therefore cannot operate with a suction lift. The installation should be such that the stock flows into the pump with adequate flow head. The inlet flow head should be at least high enough to overcome the suction pipe friction head loss, whereby it should not fall below a minimum value of 2m. The inlet pipe should be as short as possible and without bends, i.e. it should be as close as possible to the supply container. The diameter of the pipe should not be less than that of the pump suction flange.
In general rotary positive displacement pumps are installed for suspension consistencies up to 15%. Centrifugal pumps which are used for suspension consistencies > 6 to 8% bone dry, must be equipped with devices (inducers) to ensure an even loading on the impeller.

4.10 Shaft sealing

In general pumps are designed with the shaft passing through to the outside of the casing for the purpose of fitting the shaft bearings and the drive.
At the point where the rotating shaft passes through the stationary casing it must be sealed in order that:

- the liquid which is under pressure is prevented from leaking to the outside,
- air is not drawn into the casing when, (e.g. in suction operation), the seal area is under negative pressure.

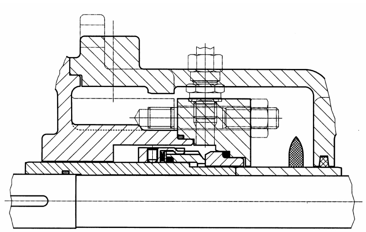
It is clear from this that the shaft seal performs an important function in ensuring the reliable operation of the pump and in preventing environmental damage. This seal must therefore be very carefully selected.
In general there are two types of seal construction:

- seals with a narrow radial gap (bush seal)
- seals with a narrow axial gap (face seal)

4.10.1 Construction of a shaft seal with a narrow radial gap (bush seal)

4.10.1.1 Contact-free shaft seal

The radial gap in a contact-free shaft seal must be sufficient to ensure that it does not pick up, due to shaft deflections and vibrations. It must therefore be set relatively large and consequently the leakage is also large.
Contact-free seals are thus primarily used as throttle rings or sleeves to limit the flow between two chambers under different pressures. Throttling may also be used to reduce the pressure in the chamber after the throttle, or to maintain the pressure in the chamber before the throttle, e.g. to prevent evaporation.



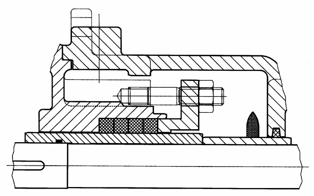
The narrow clearance in the throttle limits the leakage rate and reduces the pressure in the direction of flow. The length of the gap is dependent on the pressure ratios.

Fig. 4.10.01 Throttle in front of the pressure side mechanical seal of a high pressure pump.

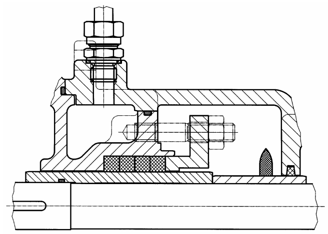
4.10.1.2 Shaft seals with contact

The radial clearance in shaft seals with contact could be reduced to nearly nothing. However as frictional heat is generated by the contact with the rotating shaft, the clearance must be increased sufficiently, to allow a controlled amount of leakage to remove this heat. This flow lubricates the sealing faces and ensures they are not destroyed by the heat generated by dry running. The leakage rate required is relatively high, compared to seals with axial gaps, and so this type of contact seal with radial clearance should only be used for pumps handling environmentally friendly fluids, e.g. drinking water, cooling water, hot water and sea water.
This type of shaft seal is mainly used in the form of a packed gland with packing rings made from asbestos free yarns e.g. cotton, synthetics and PTFE-graphite.

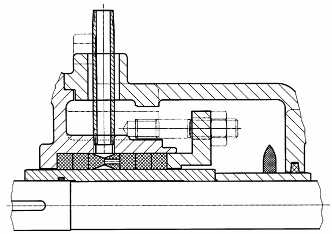
Packed gland seals



Packing rings
Depending on the pressure ratios, the stuffing box will be fitted with 4 to 6 packing rings.
For liquid temperatures up to 110°C without cooling.



Packing rings with cooling
Cooling is by means of a cooling jacket or for intensive cooling a sleeve.
A cooling jacket for liquid temperatures up to 180°C.
Cooling sleeve for liquid temperatures up to 210°C.



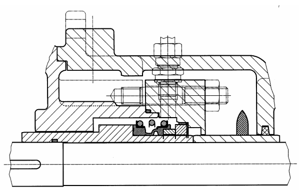
Packing rings with lantern ring
This construction is selected for shaft sealing which operates under vacuum. By supplying the lantern ring with a clean liquid, which is compatible with the pumped liquid (external flush), or the pumped liquid itself (internal flush), the entry of air in to the pump is prevented.
This is important for suction operation and the flushing liquid should be supplied at 1-2 bar.

4.10.2 Construction of a shaft seal with a narrow axial gap (face seal)

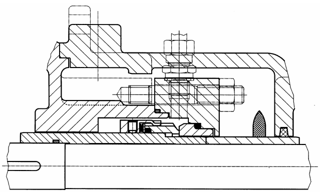
Shaft seals of this type are known as (axial) mechanical or face seals.
With this construction, two sealing faces at right angles to the axis of rotation are pushed against each other and one rotates on the other. In operation a narrow gap is produced between the two finely machined faces and a liquid or gas lubricating film forms. The size of this gap is dependent on a number of factors, amongst others the surface roughness of the sealing faces, which for generally used materials lies between 0.01 and 0.15 µm (arithmetic mean value R_a). To calculate the leakage rate of a mechanical seal, generally for a mean clearance gap of less than 1 µm is considered. This value lies well under that which can be achieved by a shaft seal with a radial gap. Consequently, the leakage rate of a mechanical seal is considerably lower than that of a seal with radial gap.

For the mechanical seal of a centrifugal pump, the following constructions come into consideration:

4.10.2.1 Single seal



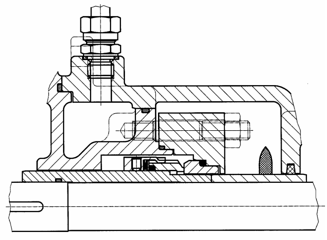
Single seal unbalanced, with rotating spring section and stationary counter face. With this construction the seal faces are subject to the full pressure in the seal housing. This type of seal is therefore restricted to applications with maximum pressures of 10 to 16 bar.



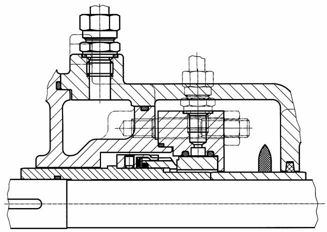
Single seal balanced, with rotating spring section and stationary counter face. The seal faces are unloaded by means of a step in the shaft or shaft sleeve which gives a surface area ratio $k < 1$. This type of seal is suitable for applications with maximum pressures of 20 to 40 bar and, in special designs, even higher.

With this type of seal it is necessary to have a circulation of the pumped media from an area of high pressure, to the seal chamber to remove the frictional heat from the sliding faces, to prevent solids deposition and if necessary to maintain an over-pressure in the seal chamber.

Return of the circulated fluid from the seal chamber to a low pressure area e.g. the pump inlet is also possible.
The circulation can be made either through external piping or internal borings.



Single seal balanced, with cooling jacket
This dead end construction, i.e. without product circulation, is used for hot water pumps for temperatures up to 140°C.



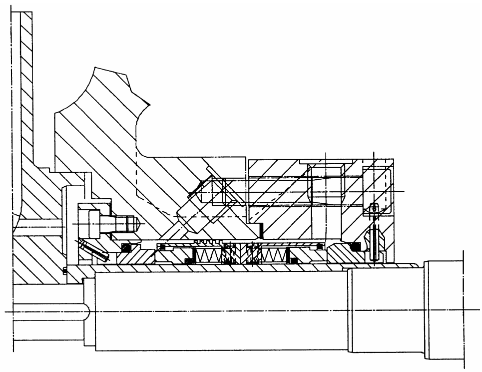
Single seal balanced, with cooling jacket and counter ring cooling
This dead end construction i.e. without product circulation is also used for hot water pumps. With the counter ring cooling it is suitable for water temperatures up to 230°C.
By using a cooling sleeve in place of the cooling jacket and with an external heat exchanger, this mechanical seal is suitable for temperatures up to 311°C.

4.10.2.2 Double mechanical seals

Double mechanical seals are selected when the pumped medium cannot be allowed to leak past the seal for chemical, physical or environmental reasons. Before making the selection however it is advisable to consider whether a leak free pump (magnetic drive or canned motor) may be more suitable for the application.

In addition to special double mechanical seals, in most cases the double seal is made up of two single seals. Depending on the operating pressure and the pumped media, one or both seals can be either balanced or unbalanced.

Double mechanical seals, back to back design



With this construction the two seal faces are positioned opposing each other and so form, with the seal housing, a chamber which is sealed from the product and the atmosphere. An environmentally and product compatible buffer fluid is required to remove the frictional heat and to lubricate the sliding faces. A pressure approximately 2 to 3 bar higher than the sealing pressure is needed.

Fig. 4.10.02 Double mechanical seals in back to back design

This type of seal is considered when no leakage from the pump can be permitted due to explosion, environmental or health dangers or when the pumped liquid can polymerise due to the heat generated by the friction of the sliding faces.
The required circulation of the buffer fluid is ensured by use of a thermo-syphon system or a pumping screw on the seal.

Gas sealed, double mechanical seal
Gas lubrication of double back to back seals can provide the solution if no product compatible liquid is available, or if it is essential to prevent product from entering the seal gap, due to risk for example of deposition of crystals.
The buffer medium is generally an inert gas such as nitrogen with a pressure approximately 2 bar higher than the sealing pressure.

Double mechanical seals, tandem design
With this construction, two single seals are positioned one behind the other.
Whilst the product side seal is lubricated by the product, it is necessary to supply an environmentally suitable quench liquid to the outboard side. In contrast to back to back construction, the quench liquid does not need to be at a higher pressure.

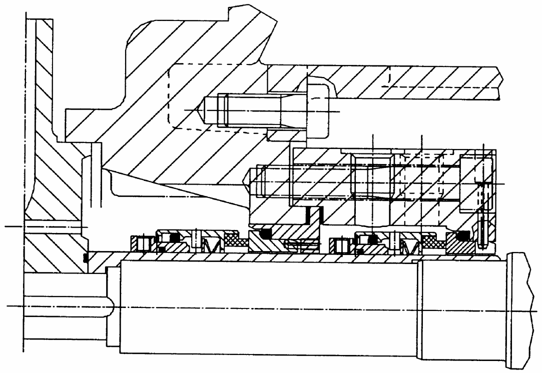


Fig. 4.10.03 Double mechanical seals tandem design

Double mechanical seals with a stationary spring
This design of seal with a rotating counter ring and stationary spring section was specially developed for the standard chemical pump. The design can equally be used as a single seal with quench or as a double seal.

The seal on the outboard side can also use a rotating spring section.

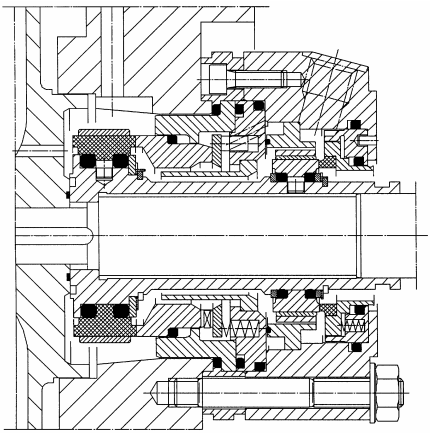


Fig.4.10.04 Double mechanical seal with a stationary spring and pumping screw

The special features of this seal are:

- The springs are protected from contact with product and leakage.
- The seal is especially suitable for products containing abrasive solids.
- The seal is doubly balanced, i.e. it does not open with loss of buffer fluid pressure and is self closing in the event of pressure reversal.
- Is available as a cartridge seal, i.e. pre-assembled, ready for installation, which speeds up and simplifies fitting, shortens fitting time and avoids fitting errors.

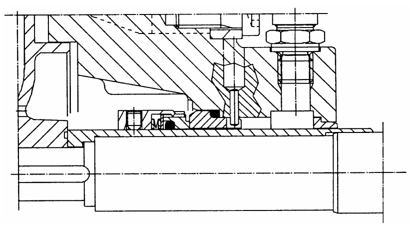
4.10.2.3 Quench installation for single and double mechanical seals

Quench is the term commonly used in seal technology for an arrangement which supplies an external fluid, without pressure, to the outboard side seal faces.
The quench fluid can be:

- Liquids, providing they are easily available, environmentally suitable and not hazardous.
- Steam.
- Gases, primarily inert or dry air.

A liquid quench, absorbs and takes away safely, any leakage and also serves to monitor the leakage rate, by observation of the level of the quench fluid in the vessel.
Steam quench is primarily used for heating the atmospheric side of the seal when media with a high melting point is handled. This prevents solidification of any leakage and blockage, preventing proper seal function.
Gas quench is used as icing protection for cryogenic media, the dry gas preventing ice formation and loss of seal function.

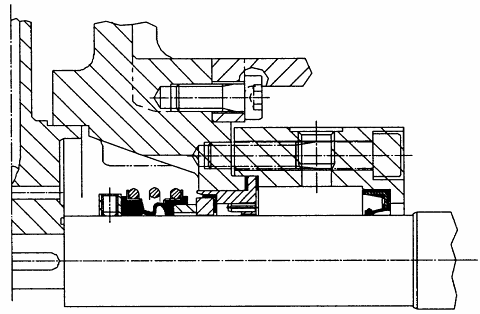
There are various methods of sealing the quench medium:



Throttle bush
With a narrow radial gap, preferred for gas and vapour quench, but less for liquid quench.

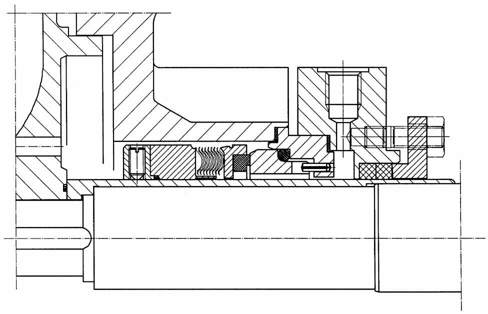
Fig. 4.10.05 Quench with throttle bush

A throttle bush, which must be made of a non-sparking material, can also be installed without a quench, to reduce the leakage in the event of seal failure.



Lip seals
Preferred for all lubricating quench fluids, e.g. oil and water.

Fig. 4.10.06 Quench with lip seal



Outboard gland
Preferred for steam and some liquids. The packing rings must have good dry running properties.

Fig. 4.10.07 Quench with outboard gland

Mechanical seals
These are the preferred arrangement for all circulated quench fluids. The construction is similar to a tandem mechanical seal.

A single mechanical seal with quench, is often installed in place of double mechanical seals when the operating, conditions allow. This simplifies the installation and the operation as instead of a flush under pressure, only a pressure free quench is needed.

4.10.2.4 Selection of mechanical seals

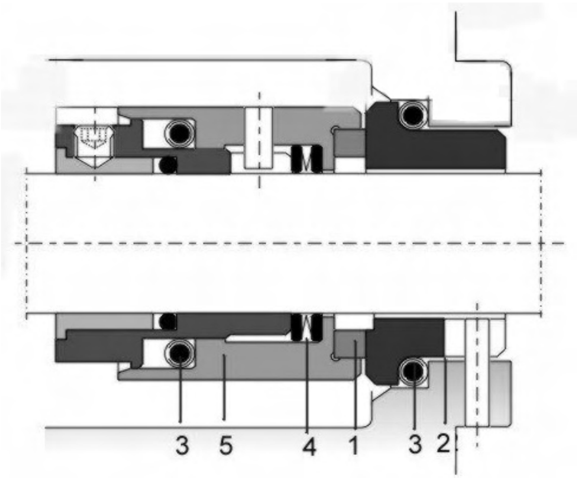
The selection of the mechanical seal is made depending upon the operating conditions of the pump in which it is to be fitted and the pump operating data including speed, shaft or shaft sleeve diameter and medium pressure.
The medium pressure p_A generally lies between the pressure in the pump entry section p_1 and the pressure in the discharge section p_2, therefore:

$$p_1 < p_A < p_2$$

The actual value of the medium pressure depends on the pump construction and the type of axial thrust equalisation and can only be provided by the pump manufacturer.

4.10.2.5 Materials of construction of mechanical seals

The materials of construction of a single mechanical seal are defined by a five position code in accordance with EN 12756. Double mechanical seals with a common spring section, use an eight position code. The first five positions refer to the product side seal, including the spring and the next three positions refer to the sliding face, the stationary face and the secondary seal comprising the outboard seal. If a double mechanical seal is made up of two single seals, then both are individually designated.



1 Seal face
2 Stationary seat
3 Secondary seal
4 Spring
5 Other metallic parts

Fig.4.10.08 Basic construction of a mechanical seal, e.g. Sterling GNZ seal

Table 4.10.01 Material code (extract from EN 12756)

Position 1 / Position 2	Position 3	Position 4 / Position 5
Materials for seal face and stationary seat	Material for secondary seals	Material for spring and other metallic parts
Synthetic carbons A = carbon graphite antimony impregnated B = carbon graphite resin impregnated Metals S = Cast Cr Mo Steel Carbides Q = Silicon carbide U = Tungsten carbide Metal oxides (ceramics) V = Aluminium oxide	Elastomers E = Ethylene propylene rubber (EPDM, EPPM) K = Perfluorocarbon rubber N = Chloroprene rubber (CR) P = Nitrile-butadiene rubber (NBR) V = Fluorocarbon rubber (FPM) Elastomers, wrapped M = Elastomer / PTFE Thermo Elastomers T = PTFE	G = Cr Ni Mo steel M = Nickel alloy Hastelloy ®

Combinations of face / seat materials
Usual material combinations are:

<u>Hard / Soft</u>
This combination has particular emergency running properties:
- Chrome molybdenum steel against carbon graphite - code SB
- Aluminium oxide against resin impregnated carbon graphite - code VB
- Silicon carbide against antimony impregnated carbon graphite - code QA

<u>Hard / Hard</u>
This combination has good wearing properties:
- Silicon carbide against silicon carbide - code QQ
- Tungsten carbide against tungsten carbide - code UU

The primary consideration for the material selection apart from the sliding properties, is the corrosion resistance to the medium handled.

Secondary seal
The materials of the secondary seal are selected for their permitted operating temperature and chemical resistance. Reference data for this is given in section 11.6 “organic materials”.

Spring and other metallic parts
The usual material for these parts is a chrome nickel molybdenum steel e.g. EN material 1.4571, X6 Cr Ni Mo Ti 17 12 2, code G
For higher degree of corrosion resistance a nickel alloy incorporating chrome and molybdenum as Hastelloy type is used e.g. DIN material 2.4610, Ni Mo 16 Cr 16 Ti, code M.

Example of designation using material codes
SBVGG means:

Position 1: seal face	S = Cast Cr Mo Steel
Position 2: stationary seat	B = Resin impregnated carbon
Position 3: secondary seal	V = Fluorocarbon rubber
Position 4: spring	G = 17 12 2 Cr Ni Mo steel
Position 5: other metallic parts	G = 17 12 2 Cr Ni Mo steel

4.11 Leak-free pumps

The increased use of leak free pumps throughout the process industries can largely be attributed to the regulations applied to plants which handle dangerous materials. To meet the stringent regulations and limits, often even minimal leakage is not permissible. As a rotating shaft seal, whether packed gland or mechanical seal, always needs some leakage to lubricate the sliding faces, according to function, so in such cases a seal-less or leak free pump must be installed. Apart from displacement pumps such as diaphragm or peristaltic, centrifugal pumps and side channel pumps with magnetic coupling or canned motors can be considered.

4.11.1 Pumps with magnetic coupling

In a magnetic coupling, the drive power, is transmitted to the pump shaft without contact by permanent magnets, through the shroud which seals the pump casing.The pump shaft does not pass through the casing, instead it is supported inside the pump casing by product lubricated bearings. Therefore there is no need for a rotating shaft seal.

The vast operational experience gained over many years of pumps with magnetic couplings, proves that comparing their efficiency with conventionally sealed pumps, there are no reasons to exclude their consideration. Their development has so advanced, that any restriction on their suitability and performance compared to mechanically sealed pumps should not be expected. In order to compare and categorise the many constructions of magnetic couplings, several references have been produced, e.g. the German VDMA 24279, “Requirements of centrifugal pumps with magnetic couplings and canned motors”.

4.11.1.1 Operating principles of permanent magnetic coupling

Magnetic couplings are contact-free, power transmitting couplings, which consist of one driven set of magnets and one non-driven set, which are separated by a shroud of non-magnetic material, which also serves to isolate the driven machine from the surrounding environment. For use with centrifugal pumps, modular couplings have been developed in which the shroud and the magnets are arranged coaxially, (see Fig. 4.11.01).

An even number of magnetic bars of alternating polarity are arranged around the circumference of both the inner and outer magnets. The magnetic bars are secured by a lightly magnetic steel retaining ring which also connects the magnetic field lines from bar to bar.

The field lines pass from the outer magnetic bar to that lying directly opposite, inside the shroud and then through the inner retaining ring to the neighbouring pole and from there returning from its inner bar to the directly opposite bar in the outer ring, so setting up a magnetic field circuit,(see Fig. 4.11.02). A rare earth alloy, samarium cobalt (SmCo) has become established as the best magnetic material which also shows very good high temperature magnetic properties.

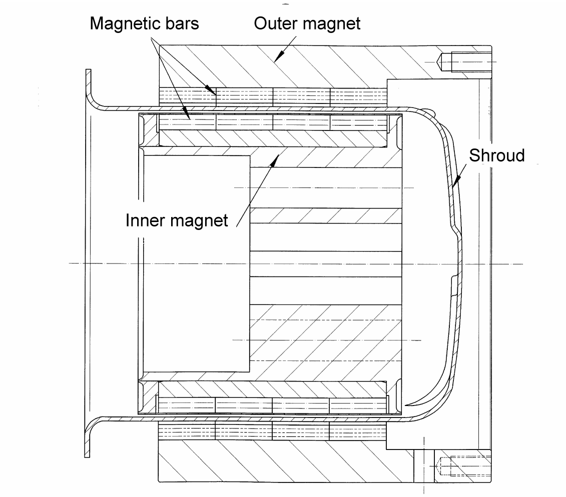
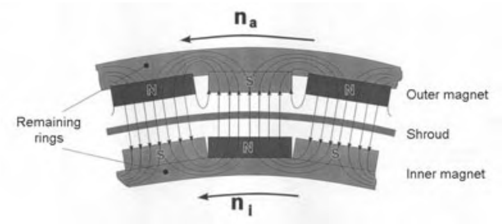


Fig. 4.11.01 Basic construction of a magnetic coupling



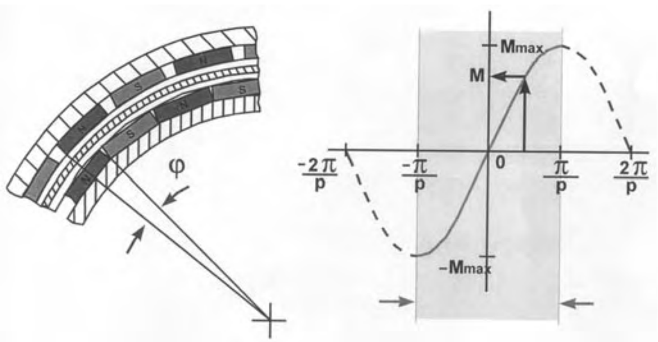


Fig. 4.11.02 Magnetic field lines and slip angle φ
p = number of poles, plates, in circumference.

In order to transmit a torque, the field line between the outer and inner magnets must have a tangential component. This is given by the slip angle φ by which the driven magnet, dependent on load, lags behind the drive magnet. The relationship between the transmitted torque *M* and the slip angle φ is described by a sine wave function. As the slip angle becomes larger an increasing portion of the magnet's volume is located adjacent to one of the same polarity. As like poles repel, the operating condition of the coupling becomes unstable. The transmitted torque reaches a maximum when exactly half the magnet overlaps one of the same polarity. Further slip rotation reduces the transmitted torque to the point where the overlap is full and the poles repel each other. As a consequence of this overload the coupling slips and drive is lost. In normal pump operation the inertia of the rotating parts and the load prevent re-synchronisation of the coupling magnets which is termed "break out".

4.11.1.2 Efficiency of a magnetic coupling

The relative movement of the magnetic field to the shroud induces an electric current as the field is cut by the conductive material of the isolating shroud. As the strength of the magnetic field in the coupling is constant, according to the induction laws the magnitude of the induced voltage is dependent only on the volume and the speed of the conductor in the magnetic field, in other words the rotational speed and the diameter of the shroud. The eddy current losses are proportional to the wall thickness, to the cube of the diameter and to the square of the speed, as well as the conductivity of the shroud material.

These eddy currents heat up the shroud and draw power from the motor additional to the drive power thus reducing the efficiency. The transmitted torque of the coupling is not diminished by the eddy current losses.

The heat generated in the shroud by the eddy currents must be removed by a part of the flow taken from inside the pump. This partial flow also has the function of lubricating the sleeve bearings in the pump.

The partial flow passing through the narrow gap between the inner magnet and the shroud causes a flow pressure loss, the size of which is dependent on the viscosity and density of the pumped media and the speed and geometry of the inner magnet. These flow losses also reduce the efficiency of the coupling.

4.11.1.3 Influence of temperature on transmitted torque

The transmitted torque of a mechanical seal is influenced by the temperature of the magnets and the radial separation between the inner and outer magnets. The influence of temperature is shown in Fig. 4.11.0.3.

The reduction in transmitted torque with rising temperature is made up of a reversible and an irreversible component. The irreversible component is caused by a one off alteration of the magnet which occurs when it is first warmed up. This loss is permanent even after the temperature has reduced to ambient condition.

The reversible component must be taken into account when the coupling is selected. If for example a coupling rated for 100 Nm is installed in an application at 200°C, then the transmitted torque is reduced to 90 Nm.

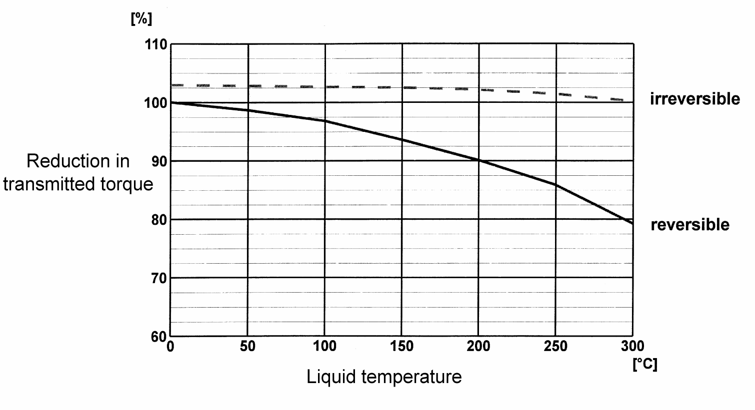


Fig. 4.11.03 Influence of temperature on the transmitted torque of a magnetic coupling with SmCo magnets.

4.11.1.4 Pump shaft bearings

The use of magnetic couplings requires the installation of bearings which can be flushed and lubricated by the pumped medium, as the shaft is not brought outside the pump casing. Widely used for this purpose are ceramic, mostly silicon carbon (SiC) sleeve bearings. The ceramic material is characterised by very low wear rate and a universal chemical resistance.

4.11.1.5 Partial flow for cooling and lubrication

As mentioned above, a partial flow is taken off the main pumped stream in order to cool the metallic shroud and to lubricate the pump shaft bearings. Generally the partial flow is taken off inside the pump from the pressure side, fed through the shroud and the bearings and returned at a suitable position in the low pressure pump inlet chamber, where it mixes in with the main pumped stream.
If the pumped medium is heavily contaminated, the partial flow can be taken off after the pump through a free flow filter.

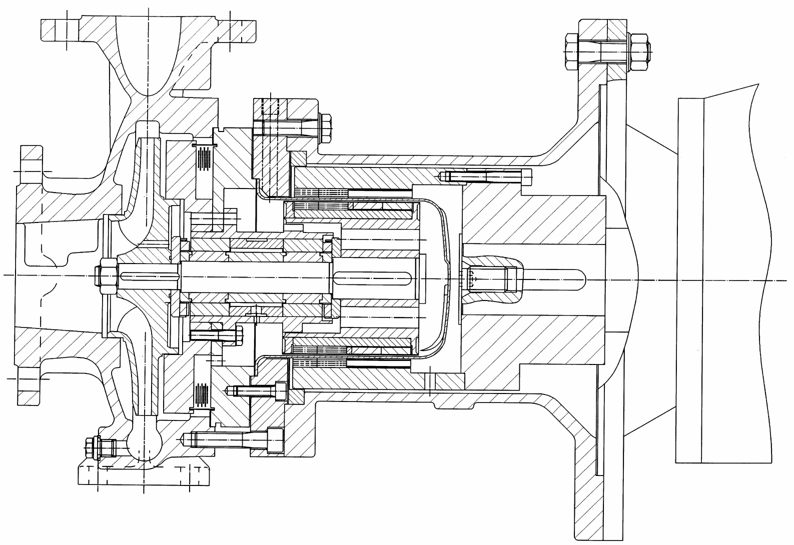


Fig. 4.11.04 Magnetic coupled pumped in monobloc design.

If the pumped medium is not suitable for cooling or lubricating, a liquid which is compatible with the medium, can be fed externally. If the pumped medium must not be contaminated in this way, then a separate external circulation must be established. This will need an auxiliary pump impeller in the magnet housing to circulate the cooling medium through a heat exchanger.
The partial flow must remove all the heat losses which are generated in the magnet chamber, i.e. eddy current and flow losses. Consequently, the partial flow must not exceed a temperature limit which can be established from the following criteria:

1. The vapour temperature of the pumped media or external cooling and lubricating flow must not be exceeded.
2. A chemical reaction (e.g. polymerisation) must not occur.

4.11.1.6 Limits of operation

Current technology based on the use of SmCo magnets allows magnetic couplings to be installed with the following limits:

1. <u>Temperature</u>
   With common pump designs, media with temperatures in the range -20°C to +180°C can be handled. For temperatures up to 300°C, in some cases modification of the bearings and lantern rings is necessary.

For temperatures above 300°C a heat barrier between the pump and coupling, the "dead end" design with a ceramic shroud is used. These will operate with media up to 400°C without external cooling. For temperatures below -20°C, suitable measures need to be taken to ensure that condensation does not freeze when the pump is shut down, preventing the outer magnets from rotating on start up.

2. Viscosity
   Magnetic couplings can be used with media in the range 0.3 to 300 mPa s. For lower viscosity, the suitability of the bearing, particularly the material, should be checked. For liquefied gases and other media with poor lubricating properties, a carbon bearing has been proven in place of the silicon carbide.

3. Pressure
   In theory the shroud can be designed for any nominal pressure. However as increases in pressure require an increase in the wall thickness, which in turn increases the eddy current losses, at pressures above 100 bar, the magnetic coupling becomes uneconomic when compared to a canned motor pump.

4. Torque
   There are no physical limits for the design of high torque transmitting drives. In practice however the requirement for drives over 380 Nm is very small.

4.11.1.7 Selection of magnetic couplings

The magnetic coupling must transmit the torque of the pump, the friction moment of the bearings and the liquid resistance in the gap between the shroud and the inner magnets and the accelerating moment of the motor on start up and under load changes. For a pump with particular performance data, the required magnetic coupling and drive motor can be established as follows:

1. Establish the hydraulic power requirement of the pump with respect to the density and viscosity of the pumped media.

2. Select a magnetic coupling which at least matches the nominal power requirement in item 1. above, at the required speed and anticipated operating temperature.

3. Establish the friction losses for the selected coupling.

4. Check that the nominal power of the selected coupling is higher or at least equal to the sum of the required power item 1. and the friction losses item 3. at the anticipated temperature and if necessary select a size larger.

5. Establish the eddy current losses for the selected coupling.

6. Select an electric motor with a rated power larger than the sum of the required power, the friction losses and the eddy current losses. In addition a safety margin should be allowed, (e.g. DIN ISO 5199)

7. Check the start up conditions. To be sure that the magnetic coupling will transmit the acceleration moment of the motor started direct on line, the ratio of the nominal torque of the coupling to the motor must be at least equal to a value determined from the ratios of the moments of inertia of the driver and driven part.

The correct combination of pump, magnetic coupling and motor can be established by this method. More convenient is to use a software programme which has links to all the necessary databases for pumps, couplings and motors.

4.11.2 Canned motor pump
4.11.2.1 Operating principle
The canned motor pump is a combination of centrifugal pump, and asynchronous three phase motor. The hydraulic part of the pump is directly connected to the drive motor. The pump and motor have a common shaft with the motor rotor lubricated by the pumped fluid. The drive torque is transmitted from stator to rotor by the induction principle. The principle is the same as a normal asynchronous motor except that a "can" of non-magnetic material is positioned in the gap between the stator and rotor and seals off the stator from the liquid flushed rotor.

4.11.2.2 Efficiency of the canned motor
The efficiency of a canned motor is determined primarily by the eddy current losses in the can and by liquid flow losses in the gap between the stator and rotor. Also the width of the gap which is very large (1 to 2 mm) compared to the air gap in a conventional motor influences the efficiency of the drive. As the larger gap requires a stronger magnetic field to transmit the torque and this is generated by the electric current, then invariably, a higher absorbed power is needed for the same mechanical performance.

4.11.2.3 Influence of temperature

The performance of a canned motor is essentially determined by the permissible winding temperature. Therefore for particular temperature ranges, a maximum absorbed current and hence maximum power is established, such that the permissible winding temperature in the stator is not exceeded.

4.11.2.4 Pump shaft bearings

As the shaft of a canned motor pump is not brought outside the casing, the shaft is supported in medium lubricated bearings. Mostly ceramic sleeve bearings e.g. silicium carbide are used and arranged either side of the motor rotor. The impellers are therefore an overhung arrangement.

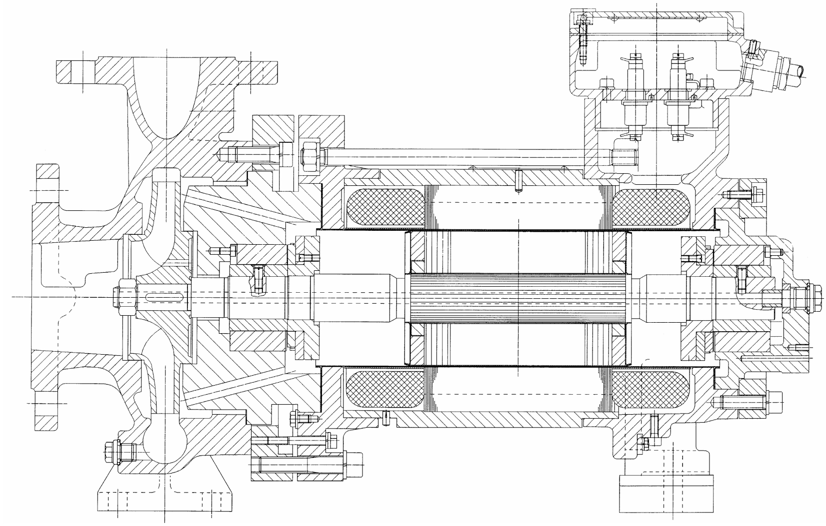


Fig. 4.11.05 Canned motor pump

4.11.2.5 Partial flow for cooling and lubrication

With canned motors, the partial flow not only removes the heat from the can caused by the eddy currents and lubricates the sleeve bearings, but also cools the stator. Due to the hermetic sealing it is not possible to fit a cooling fan on the non-drive end of the motor, so the partial flow must also remove the heat generated by the copper losses in the stator.

The partial flow must therefore not exceed a certain temperature which is established from the following criteria:

1. The boiling point of the pumped media must not be exceeded.
2. A chemical reaction, e.g. polymerisation must not occur.
3. A high partial flow temperature must not cause a high surface temperature of the pump unit in cases where this must be avoided (explosion hazard).

4.11.2.6 Limits of operation

1. Temperature:
   As indicated above, the performance of a canned motor is essentially determined by the permissible winding temperature. This varies greatly with the type of winding . In all cases the cooling provided by the partial flow must ensure that the permitted temperature is not exceeded. With carefully designed windings and cooling arrangement of the partial flow, very high media temperatures can be allowed.

2. Viscosity:
   The limiting values for viscosity lie between 0.3 and 300 mPa s, at the lower viscosities the suitability of the bearings and bearing materials must be considered. High viscosity leads to increased flow losses in the gap between the rotor and can.

3. Pressure:
   In principal the canned motor can be used for all nominal pressures. Even for high pressures a relatively thin can wall has sufficient strength as it is supported by the stator winding. That does mean however that the construction of the stator housing must be sufficiently robust to accept these loads.

4. Performance:
   There are virtually no limits for the construction of extremely powerful drives. Most important with high power drive is the removal of the heat losses. In practice there is little requirement for drives over 100 kW.

4.11.2.7 Protection for canned motor pumps

If a canned motor pump is to be installed in an explosion hazard area, then special requirements for the operation must be met as described in section 9.5.6.
The level and temperature controls which are required for such installations are basically also recommended for other installations where no explosion protection is required.

The use of level control will avoid the possibility of the pump running dry, which can cause considerable damage to the bearings and the can.
The use of temperature control will ensure that the pump unit is shut down if the permissible maximum temperature is exceeded, so that it is only used within safe operating ranges.

Similarly the use of a minimum temperature setting can prevent the unit from being operated at too low a temperature, when for example the media temperature is above the ambient temperature.

Additionally it is recommended that canned motor pumps are protected from over-current and thermal overload as described in section 9.4.8.

4.12 Pumping liquefied gases

4.12.1 General

The term liquefied gas is a general description applied to gases which can be changed from the gas to liquid state at near ambient temperatures and at relatively low pressures. A stricter definition as in standard DIN 51622 identifies hydrocarbon gases such as propane, propene, butane, butene and mixtures which are produced as by-products of oil refineries and natural gas plants. The liquefied gases propane, butane and their mixtures are primarily used as household, vehicle and industrial fuels. Propene and butene are raw ingredients used in the plastics industry.
In order to utilise the liquefied gases for household and industrial purposes distribution systems requiring pumps are set up as typically shown in Fig. 4.12.01.

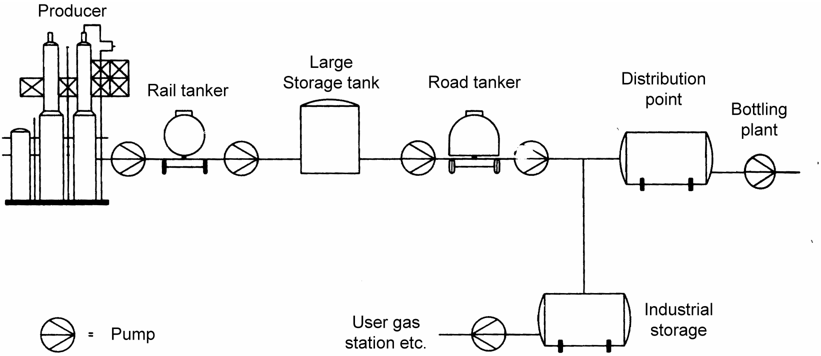


Fig. 4.12.01 Distribution system for LPG

Special regulations apply for the safe handling and storage of liquefied gases, which also put special requirements on the pumps to ensure the media is kept below its boiling point.
Important are:

- A good NPSH performance
- Gas/liquid mixture handling ability
- Reliable shaft sealing

To satisfy these requirements so called combination pumps with side channel and centrifugal stages were developed.

4.12.2 Construction of liquefied gas pumps

4.12.2.1 Side channel combination pumps

The most common construction of side channel combination pumps is the horizontal ring section pump with axial suction. The side channel stages are located at the discharge side in series after a centrifugal stage. This side channel combination uses the best pumping characteristics of each type of impeller to complement the other. The primary centrifugal stage allows low (*NPSHR*) values to be achieved (Fig. 4.12.02). The following side channel stages enable considerable gas/liquid mixture flows to be handled and for the pump to be self priming.

3 Side channel pump

2 Centrifugal pump

1 Side channel pump with low NPSH primary stage

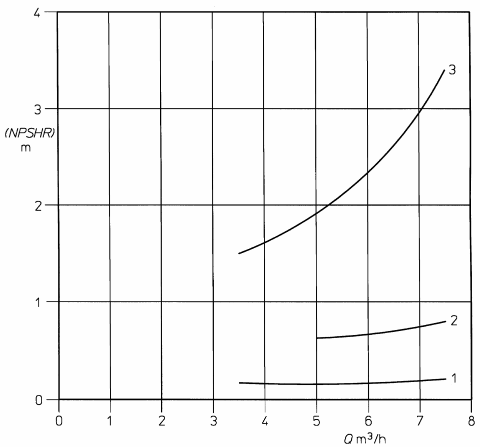


Fig. 4.12.02 (*NPSHR*) - comparison of different pump constructions.

Unlike other types of pump construction, the side channel combination pump requires a suction head of less than 0.5m for flow rates up to Q=35 m^3/h. This makes the plant installation much simpler and requires lower capital investment. The combination with side channel stages at the discharge ensures that the flow does not collapse even when gas is evolved.
Multi-stage side channel combination pumps are used for flow rates Q up to 35 m^3/h and total heads up to H= 350 m, in accordance with DIN 24 254.
The steps in the performance grid are so close that a pump can be selected very close to the required data. The number of stages required in the pump is determined by the operating point. The steep Q/H characteristic has advantages for control purposes.

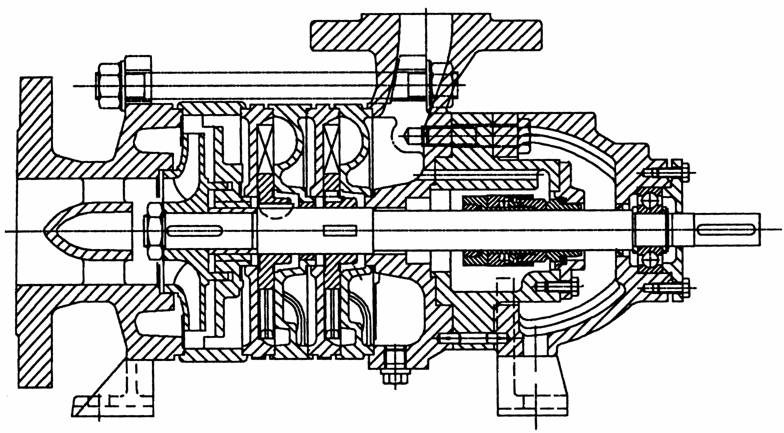


Fig. 4.12.03 Side channel pump with internally flushed mechanical seal

4.12.2.2 Side channel combination pumps with retaining stage

A variation on the side channel combination pump is the incorporation of a retaining stage. This is normally positioned between the primary centrifugal stage and the first side channel stage. Its function is to ensure that when stopped the pump retains sufficient volume of liquid to enable it to be restarted at any time. Furthermore the connections on the housing, enable the installation of a level control device to prevent running of the pump with insufficient liquid (dry running protection).

4.12.2.3 Centrifugal combination pumps

For economical pumping of larger flows, without giving up the self priming and gas handling abilities, the centrifugal combination pump was created. This features a main flow centrifugal stage with an integral side channel stage. The axial suction and special (*NPSH*) impeller at the first stage, ensures that the lowest suction head when handling liquefied gases can be met. The performance range extends to capacities up to $Q = 220\ m^3/h$ and heads up to $H = 250$ m.

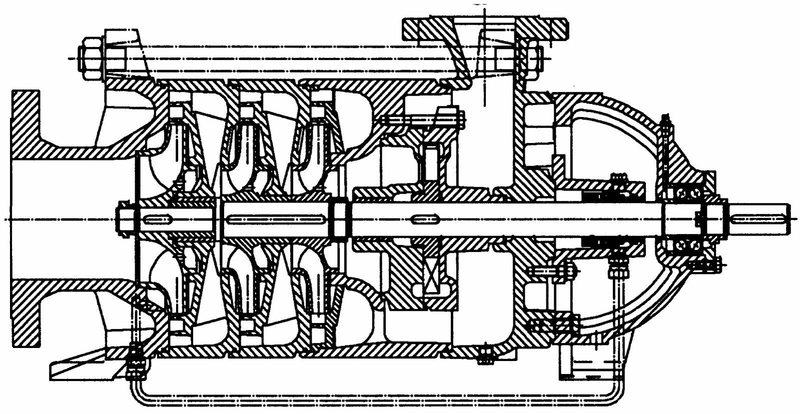


Fig. 4.12.04 Centrifugal combination pump with single mechanical seal

4.12.3 Pumping liquefied gases in suction lift operation

4.12.3.1 Pumping with partial flow - bypass

Liquefied gases are mainly stored in permanent storage tanks, the size of which is dependent on the usage and serviceability requirements. With safety in mind, these are mostly in earth mounds or underground. Extraction from the base of the tank to provide a flooded suction for the pump operation is possible, but in practice not adopted for safety reasons. The only possible extraction position from the upper dome of the tank, presents special problems for pumps in suction operation with a volatile medium.

A possible solution without having to install moving devices in the pressurised container is shown by the vehicle-LPG filling station example, see Fig. 4.12.0.5.

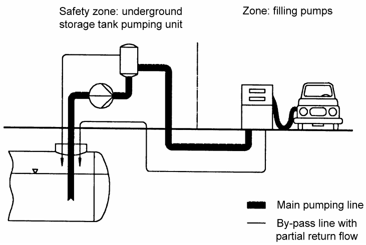


Fig. 4.12.0.5. Vehicle-LPG filling station

The installation comprises the underground storage tank, the ground level side channel combination pump, a gas separator, the main pumping line and the by-pass line with partial return flow. The advantage of this system lies in its simplicity.

This is made possible by the characteristic ability of the side channel combination pump to handle large quantities of vapour and their low (*NPSH*). By exhausting the vapour and the re-evaporation in the suction line, heat is drawn from the liquid. The resulting pressure difference compared to the constant pressure in the storage tank causes the liquid to rise up the pump suction pipe and the pumping commences. At the same time the high energy of the partial flow returned to the tank in the vapour phase produces a differential pressure, sufficient to overcome the initial suction lift and stabilise the pumping operation. The optimum parameters for, as far as possible bubble free pumping are for butane H_s < 4m and a tank capacity V < 20 m^3. When handling propane or propane/butane mixtures, the tank capacity can be increased to V < 200 m^3. The larger the storage tank capacity, the more significant the gas content in the liquid flow.

4.12.3.2 Pumping with vertical tank pumps

For storage tanks with diameters not exceeding 6m, vertical tank pumps with externally mounted motors can be installed. These pumps are mounted in the domed flange of the tank. The first stage, preferably a low (*NPSH*) stage, is located at the bottom of the tank on an extension designed for the tank, so that it operates with the suction immersed in the liquid. See Fig. 4.12.06.

Depending on the performance range, the following pump constructions can be considered:

- Side channel combination pumps for flows up to Q = 35 m^3/h
- Centrifugal pumps with a low (*NPSH*) first stage for flows up to 100 m^3/h
- Booster pumps to increase the inlet pressure.

All pumps are driven with magnetic couplings and are therefore totally leak-free. They are designed to meet the latest safety regulations and environmental considerations. With booster pumps a prime consideration is to have as few moving and, where possible, wear resistant parts inside the storage tank, to extend the time intervals between routine maintenance.
The low (*NPSH*) stage is mounted inside the storage tank. The main pressure increase is achieved by an externally mounted pump.

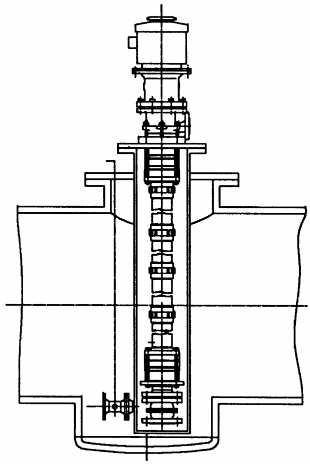


Fig. 4.12.06 Vertical tank pump

4.12.3 Shaft seals

Pumps for liquefied gas can be fitted with various types of shaft seals dependent on the safety standards for the installation:

- single balanced mechanical seal
- single balanced mechanical seal with throttle bush
- double mechanical seals in tandem or back to back construction
- seal free design with magnetic coupling

The most frequently used are pumps with single balanced mechanical seal or pumps with seal free design with magnetic coupling.

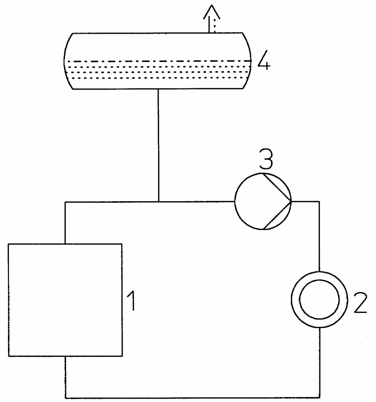
4.13 Pumping hot liquids

4.13.1 General

High demands are made on pumps which are to be used for pumping hot liquids in heating and heat transfer installations.

The pump must:

- Have a high level of operational reliability, as loss of the pump means shutdown of the entire plant.
- Be suitable for the temperature of the media, without the need for external cooling of the pump bearings and mechanical seals. An external cooling system not only increases the capital investment for the additional cooling pipework and control, but also the operating costs of removing the heat and providing cooling water.



1 Heating unit
2 Consumer
3 Circulation pump
4 Expansion tank

Fig. 4.13.01 Schematic of a heat transfer installation

Heat transfer liquids can be:

- water for temperatures in the range t = 75 to 230°C and for specific applications up to 300°C

- heat transfer oils, mineral based, e.g. Mobiltherm, Essotherm and similar for temperatures from t = 100 to 340°C

- synthetic heat transfer liquids, e.g. Diphyl and Dowtherm for temperatures from t = 100 to 400°C.

For handling these heat transfer media, single stage volute casing pumps, which have been specially developed for the application, are mainly installed.
Because of the variation of physical properties of the heat transfer liquids, e.g. vapour pressure, the design of hot water pumps is different to that for heat transfer oils or equally, synthetic heat transfer liquids.

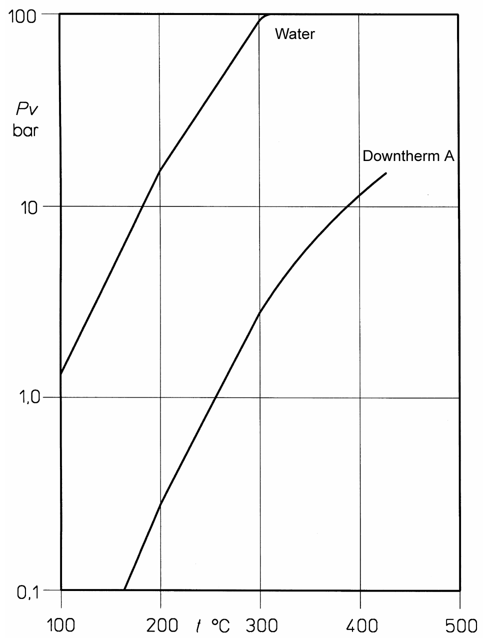


Fig. 4.13.02 Comparison of the vapour pressure curve of water with a synthetic thermal transfer liquid.

4.13.2 Pumping hot water

4.13.2.1 Water up to 110°C

This application is most typical for hot water heating systems.
For circulations up to a flowrate of approx. $Q = 60$ m^3/h and heads up to $H = 10$m the first route is to use a canned motor pump, i.e. a heating circulation pump, sealless with media flooded rotor.
For higher flowrates or heads, conventional pumps with uncooled stuffing box or mechanical seal are installed (see section 4.13.3).
Standard PN10 water pumps to EN733 are typical for these applications.

The pumps can be baseplated, or for space saving, of monoblock or in-line design.

4.13.2.2 Water from 110 up to 160°C

For heating systems with inlet temperatures up to 160°C it is similarly possible to install standard PN10 water pumps to EN733 / DIN 24 255 with uncooled balanced mechanical seals. Internal circulation, ensures that the mechanical seal is lubricated by the pumped medium thus preventing dry running. In an installation where the minimum temperature difference above the vapour temperature ($\Delta t_{min} \geq 10°C$ above boiling point) cannot be guaranteed, then a throttle device must be installed in the circulation line to maintain sufficient pressure in the seal housing so as to prevent vaporisation.

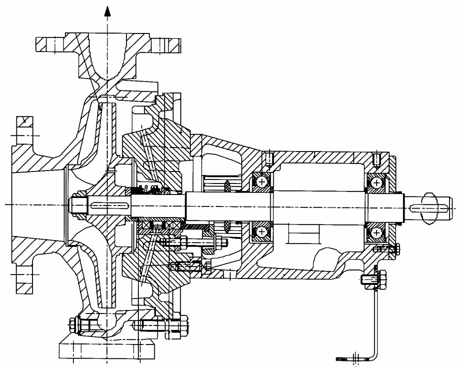


Fig. 4.13.03 Conventional pump with uncooled mechanical seals

4.13.2.3 Water from 160 up to 230°C

Even in this temperature range it is possible to run the pump uncooled. However it is necessary to fit the mechanical seal at the “cold” drive end of the pump and to protect the seal housing by a heat barrier, as shown in Fig. 4.13.04. Even with inlet temperatures up to 230°C with this design, the temperature in the mechanical seal will not exceed 100°C.

4.13.2.4 Water from 230 up to 311°C

In this temperature range the requirements outlined in the introduction, which are placed on an uncooled design, can no longer be met, so that constructions with cooled mechanical seals must be selected.

Up to a temperature of 290°C, the method of positioning the mechanical seal in the “cold end” is used with a cooled housing cover and counter face. A cooling sleeve between the pump and the mechanical seal housing, reduces thermal transmissions.

For water temperatures >290 < 311°C an integral pumping ring in the mechanical seal, sets up a circulating flow through an external heat exchanger.
This construction is shown in Fig. 4.13.05

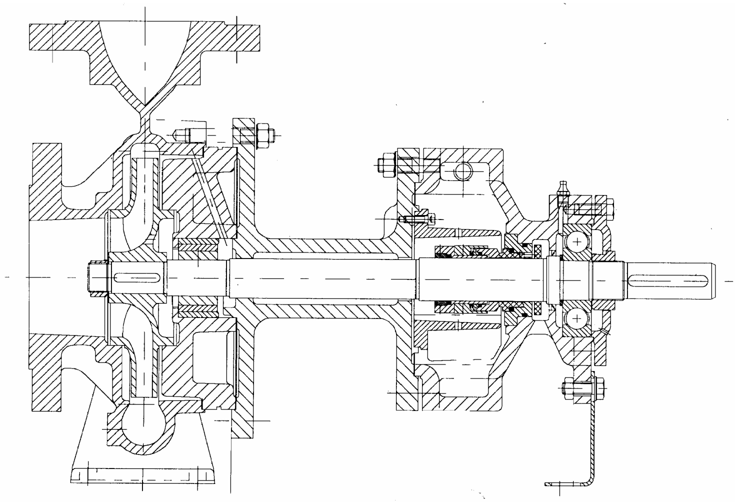


Fig. 4.13.04 Pump with uncooled seal at the "cold end"

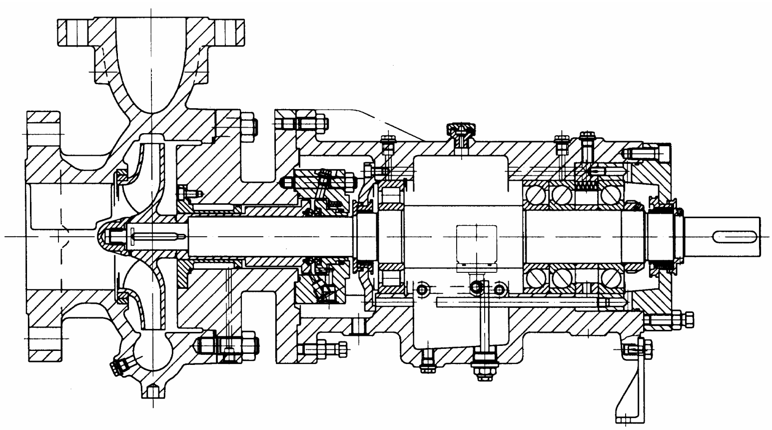


Fig. 4.13.05 Pump with cooling sleeve and cooled mechanical seal.

4.13.2.5 Materials for hot water circulating pumps

The selection of materials for hot water circulation pumps is primarily dependent on temperature and pressure. When selecting the pump and materials, the permitted operating pressure of the pump and the required operating pressure and temperature in the specification should be considered. The selection of flange ratings (PN) is made in accordance with the permitted pressure/temperature regulations for the material.

The following diagrams show the permitted pressure/temperature limits for different materials or groups of materials at different nominal pressures, when handling hot water.

Using the diagrams Figs. 4.13.06 to 08 and table 4.13.01, a suitable material can be selected, for a particular operating or design pressure and temperature.

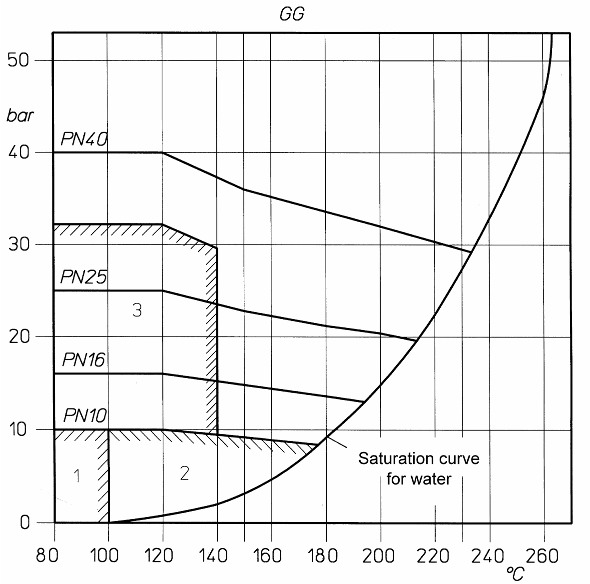


Fig. 4.13.06 Pressure/temperature limits for a flange in cast iron (GG25 to EN-GJL-250) according to EN1092-2.

1. Limit for pumps in chemical industry installations according to VDMA 24 276
2. Limits for circulation pumps DN≤ 200 according to EN 12953-2 and TRD 108
3. Limits for feed pumps (according to TRD 108)
   (TRD = German technical regulations for steam boilers)

Hot water in heating and hot water plants does not generally represent a corrosion risk due to its pre-treatment. The material selection is therefore only dependent on temperature and pressure.
Pumping hot water in power station systems, does require consideration of the chemical nature as well as the temperature and pressure. With an alkaline operation with pH > 9.5 and for pressure up to 63 bar, there is no corrosion danger. For lower pH values, higher operating pressures or neutral pH or a combination of conditions, the use of ferritic chrome steel with at least 12% Cr is recommended.

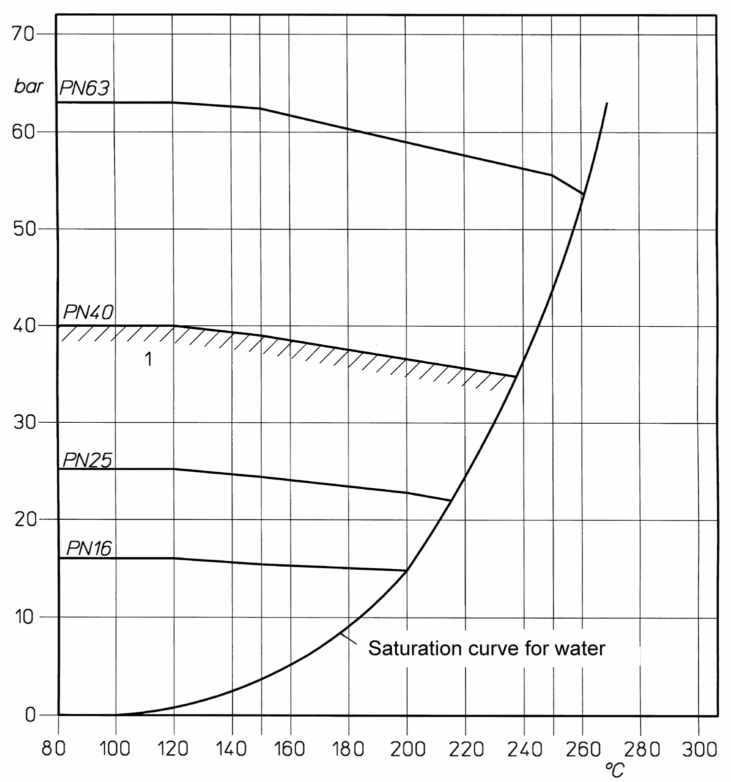


Fig. 4.13.07 Pressure/temperature limits for a flange in ductile (SG) iron (GGG40 to EN-GJS-400-15) according to EN1092-2.

1. Limits for circulation pumps and feed pumps (according to TRD 108)

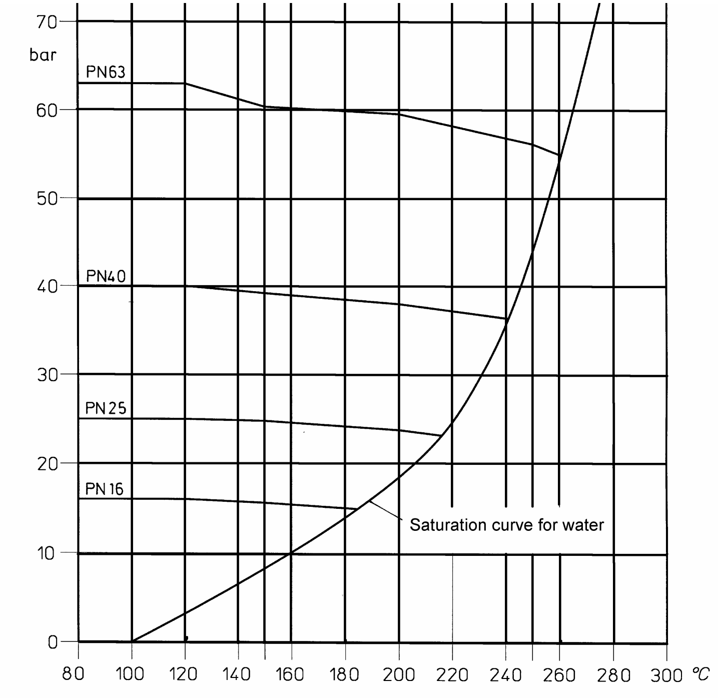


Fig. 4.13.07 Pressure/temperature limits for a flange in cast steel (GS-C25 to 1.0619, GP240GH) according to EN1092-1.

Table 4.13.01 Materials for hot water circulating pumps t > 230 to 311 °C
(Example of a pump range for boiler circulation)

Max. operating data		Flange	Material
$t_{max.}$ °C	$p_{max.}$ bar [1]	PN [2]	
255	50	63	1.4008
270	63	63	1.7706 [3]
280	70	100	1.4931 [3]
290	80	100	1.4317 - V1 [4]
311	110	160	1.4317 - V2 [5]

1) Max. housing pressure $p_{max.}$ = entry pressure + zero flow head
2) Flange to EN 1092-1
3) Cast steel to TRD 103
4) V1 = Heat treatment stage 1: 0,2%- tensile strength = 550 N/mm²
5) V2 = Heat treatment stage 2: 0,2%- tensile strength = 830 N/mm²

4.13.3 Pumping heat transfer oils and synthetic heat transfer liquids up to 400 °C

Heat transfer installations and their circulation pumps are subject to standards and guidelines.

Pumps which have a shaft seal mounted at the "cold end" drive side and a thermal barrier, with the bearing housing thermally separate or a moveable seal cartridge meet these standards for inlet temperatures up to 350°C. The pumps can be baseplated as shown in Fig. 4.13.09, or for space saving, of monobloc or in-line design.

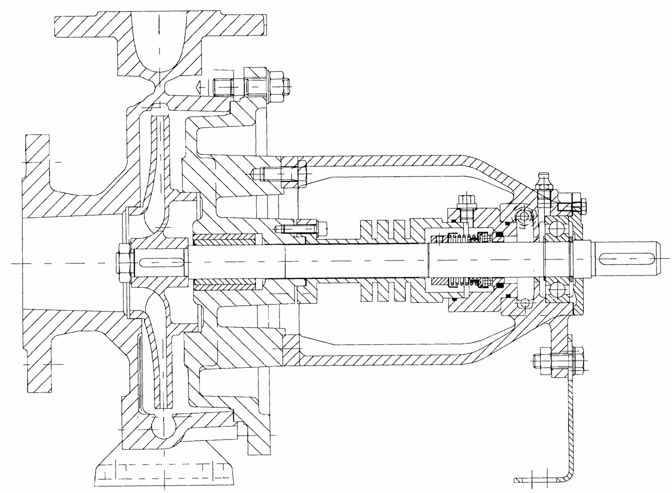


Fig. 4.13.09 Heat transfer oil pump with mechanical seal to EN 733

To meet increasingly stringent standards, reflecting concern about environmental pollution and requirements for safety, the sealless pump is being used more and more for this application. In particular for synthetic heat transfer liquids, which are classified as health hazards, the use of magnetically coupled and canned motor pumps is widespread. Such pumps are suitable for inlet temperatures up to 400°C. For more information see section 4.11 "Leak-free pumps".

Fig. 4.13.10 shows a heat transfer oil pump with a heat barrier and magnetic coupling in a compact monoblock construction.

The materials for the pump casing need to be durable.

Ductile cast iron with spheroidal graphite e.g. 400-18-L to ISO 1083 (GGG-40.3 to DIN 1693) for inlet temperatures up to 350°C and heat stable cast steel e.g. GP240GH to EN 10213-2 (GS-C 25 to DIN 17 245) for inlet temperatures up to 400°C, have proven reliable and are widely used.

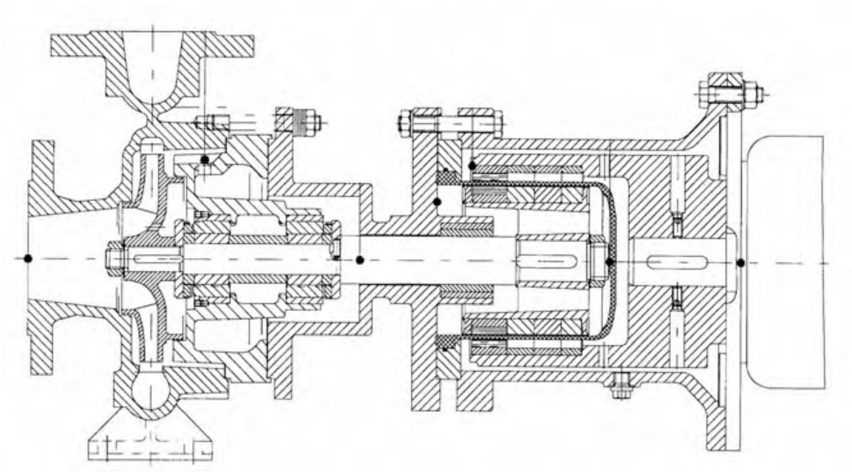


Fig.4.13.10 Heat transfer oil pump with a heat barrier and magnetic coupling in a compact monoblock construction

Table 4.13.02 Properties of various heat transfer liquids

Heat transfer liquids	Temperature range °C	Density kg/dm³	Vapour pressure mbar	Viscosity mm²/s
organic base Glycol (water-free)	20 200	1.114 1.028	< 0.1 50	150 2.1
mineral oil base naphtha base	40 250	0.861 0.725	0.02 5000	8.4 0.60
mineral oil base paraffin base	150 320	0.781 0.670	0.1 40	2.3 0.60
synthetic base Isomer mixture	150 400	0.968 0.774	19.3 5300	1.1 0.30

5 Vibration and noise

5.1 Vibration and smooth running

The reference to smooth running, is a judgement of the mechanical vibrations, less a judgement of the noise, although the two are closely connected as cause and effect.

With rotating machines and liquid flow, small yet detectable vibrations, which increase with speed and throughput, are unavoidable. The smoothness of operation of a centrifugal pump therefore only demands additional attention when the vibration levels exceed a certain level.

The benchmark for judgement of vibration is its effective velocity v_{eff} expressed in m/s or mm/s.

The generally acceptable limits for vibration velocity are set out in Table 5.01

Speed n rpm	Maximum effective vibration velocity v_{eff} (mm/s) dependent on the shaft height of the pump h_1	
	$h_1 < 225$ mm	$h_1 > 225$ mm
$n < 1800$	2.8 mm/s	4.5 mm/s
$n > 1800$ to 4500	4.5 mm/s	7.1 mm/s

Table 5.01 Maximum acceptable vibration velocity of pumps
(Limits set according to DIN ISO 5199 and DIN ISO 9905)

This table is valid for horizontal centrifugal pumps with multibladed impellers. For centrifugal pumps with special impellers, which are common with waste water pumps, then higher limiting values than in Table 5.01 may be permitted if stated by the manufacturer.

The values are measured on or near the bearing housing, at the design speed (±5%) and the design flowrate (±5%) in cavitation free operation.

For vertical pumps, the measurements are taken at the upper drive flange for pumps with rigid couplings and at or near the upper pump bearing housing for pumps with flexible couplings.

Irrespective of the type of bearing, (roller or journal), at the design speed (±5%) and the design flowrate (±5%) in cavitation free operation, the limit value $v_{eff} = 7.1$ mm/s should not be exceeded.

One of the assumptions that the above limit values are not exceeded is that all the main rotating parts of the pump and the coupling and possibly the complete assembled pump rotor, are balanced in accordance with ISO 1940-1.

The smooth running of a centrifugal pump which was acceptable during commissioning, can deteriorate during operation due to imbalance resulting from:

- asymmetric build up of deposits
- asymmetric corrosion
- asymmetric erosion due to abrasive solids in the pumped liquid
- asymmetric material loss due to cavitation
- foreign body jammed in impeller
- damage to one or more impeller blades

Other factors which can influence the smooth running of the pump can include an increase in bearing play or clearances or incorrect assembly after repair or inspection. Excessive forces transmitted from the pipework can also impact on smooth running.

5.2 Noise

5.2.1 General concepts

Sound
Pressure oscillations in the air are denoted by sound. Human hearing can detect air vibrations and pressure waves in the range 16 to 16000 Hz. Infrasound below 16 Hz and ultrasound above 16000 Hz are outside the audible range. As well as in air and other gases, sound waves can be propagated in solids and liquids. Sound transmitted through solids or liquids cannot be detected by the human ear unless they are transformed into airborne sound. In the following discussions therefore sound refers to airborne sound.

Tone, timbre, noise
Depending on the source of the noise, the pressure oscillations manifest themselves in various ways described as :

Tone: a sinusoidal (single) frequency within the audible range
Timbre: a mixture of basic and higher frequencies which have a simple numerical relationship to each other.
Noise: a term for multiple frequency sound, often simply used for machinery derived sound.

Sound pressure, sound pressure level

The sound pressure p is the oscillating pressure which is superimposed on the atmospheric pressure as a result of the sound oscillations.

For the measurement of sound pressure level L, the logarithmic ratio of the sound pressure p to a reference level p_o has been established;

$$L_p = 20 \cdot \lg \frac{p}{p_0} \qquad \text{in dB}$$

with p in µbar and $p_0 = 2 \cdot 10^{-4}$ µbar (reference sound pressure)

The value of sound pressure is non-dimensional, but in honour of the inventor of the first functional telephone, Graham Bell, it is quoted in the unit of Bel, or the more manageable deciBel (dB).

Weighted sound pressure level

In order to relate sound measurements to the actual audible sensitivity of the ear, the sound pressures at low and high frequencies are differently weighted by the incorporation of filters in the measuring instrument. These weighted sound pressure levels are designated A, B, C & D to denote the filter used and expressed with the units dB(A), dB(B), dB(C) & dB(D) respectively.

The standardised weighted curve A (filter characteristic), roughly takes account of the ears sensitivity for moderate engineering noise and therefore dB(A) is the weighting which should be used for evaluation of noise in the workplace.

Sound power, sound power level

The sound power W (unit Watt) is the quotient of the emitted sound energy and associated duration and proportional to the square of the sound pressure. The sound power level is given by:

$$L_w = 10 \cdot \lg \frac{W}{W_0} \qquad \text{in dB}$$

with W in Watt and $W_0 = 10^{-12}$ W (reference sound power)

5.2.2 Sound (noise) measurement, units

The procedures for measuring sound are defined in standards such as EN 12639. The sound pressure level is measured by a sound pressure meter on a hypothetical measurement surface S. This surface follows the actual surface of the machine at a distance of 1m, normally in the simple geometric form of a rectangular geometric solid whose six faces are parallelograms. The A-sound pressure level is averaged over the measuring surface S and if necessary corrected for the influence of background noise and reflections is referred to as the sound pressure level L_{pA} and expressed in dB(A).

As a measure of the noise emitted by the machine to the surrounding air, under defined positional conditions, the A-weighted sound power level L_{wA} is used. It can be determined from the sum of the sound pressure level at the measurement surface L_{pA} and the value of L_S for the measurement surface:

$$L_{wA} = L_{pA} + L_S \quad \text{in dB(A)} \qquad \text{where } L_S = 10 \cdot \lg\frac{S}{S_0} \quad \text{in dB}$$

S = measurement area in m² and S_0 = 1 m² (reference area).

For similar types of machines, where the measurement surface value does not differ by more than 1 dB, it is usually sufficient to compare the sound pressure level at the surface directly.

For machines where the sound pressure level at the measurement surface has been determined for which the measurement surface value differs by more than 1 dB, the A-sound power level should be used for comparison.

For more accurate investigations, a sound level measurement covering the entire frequency range is inadequate. The range must be subdivided into individual sound bands, as only by this method can clear conclusions be drawn as to the source of the sound and the appropriate measure to reduce it. For such sound analyses, filters are used which only allow the passage of a defined band. The entire range of frequencies is divided into octaves and the sound pressure level is noted for each mid point frequency. These mid point frequencies have the following series:

63 Hz - 125 Hz - 250 Hz - 500 Hz - 1000 Hz - 2000 Hz - 4000 Hz - 8000 Hz

5.2.3 Noise emissions of centrifugal pumps

In centrifugal pumps, mechanical energy is transferred to the pumped liquid through the shaft and impellers, resulting in periodic pressure oscillations. This is because the number of impeller and guide vanes is finite and is due to turbulent flow, friction and vortex shedding from boundary layers. The pump casing and attached pipework are excited by these oscillations, which are transmitted to the surrounding air and manifested as noise.

A particularly distinctive noise is generated if the required (*NPSHR)* value of the pump is not achieved and cavitation ensues, (see section 1.6).

The unit of measurement for the noise generated by a pump and emitted to the environment is the A-weighted sound pressure level L_{WA} in Decibel, dB(A).

Investigation and measurements of the sound pressure levels emitted by the vast range of centrifugal pumps, shows that it is primarily dependent on the design and power absorbed.

For a few selected designs, within a specific performance range, the expected A-weighted sound pressure level can be calculated from the following formula.

Side channel pumps $2\ kW \le P \le 40\ kW$

$L_{WA} = 67 + 12{,}5 \lg P / P_0$ dB(A)

Single stage volute or ring section pumps with multi-impellers

$10\ kW \le P \le 2000\ kW$

$L_{WA} = 66 + 13{,}5 \lg P / P_0$ dB(A)

Multi-stage ring section pumps $10\ kW \le P \le 2000\ kW$

$L_{WA} = 78 + 8{,}5 \lg P / P_0$ dB(A)

where

P = Absorbed power in kW and $P_0 = 1$ kW

Measurement surface value

For the designs indicated above and stated performance range, the measurement surface value L_S is given by the following equation with a tolerance of ± 1 dB.

$L_S = 23 + \lg P / P_0 - 3 \lg n / n_0$ [dB]

where

P = Absorbed power in kW and $P_0 = 1$ kW

n = Pump speed in rpm and $n_0 = 1$ rpm

valid for speed range: 300 rpm $\le n \le$ 3000 rpm

The addition of several sound sources, the following rules apply:

Table 5.02 Addition of sources with equal sound levels

Number of sound sources of equal sound level	2	3	4	5	6	7	8
Additional sound level dB	3	4,8	6	7	7,8	8,5	9

Table 5.03 Addition of two sound sources with different sound levels

Difference of sound levels $L_2 - L_1$ dB	0	4	8	12	16	20	24
Additional sound level ΔL dB	3	1,5	0,6	0,3	0,1	0,04	0

$L_2 \ge L_1$, resulting sound level $L_R = L_2 + \Delta L$

5.3 Noise protection measures

Noise protection measures are necessary when the noise emitted by an installation reaches an unacceptable level in the surrounding area or exceeds limits set by the appropriate authorities.

The most important primary preventative measure to avoid the generation of noise is the selection of the correct type and size of pump, so that it works at or near the point of optimum efficiency.

This presupposes that the H_AQ characteristics of the system have been determined as accurately as possible. Excessive safety factors lead to the selection of an unnecessarily oversized pump, which therefore operates under partial load condition, resulting in higher noise levels.
Further primary measures to minimise the flow noise and thus the mechanical resonance include the following:

- Avoid operation in the region of cavitation
- Selecting a low pump speed
- Selection giving low flow velocities in the connecting pipework
- Use of low noise fittings
- Locating fittings at a distance from the pump flanges
- Avoiding sudden changes in cross section of the pipework
- Use of long radius bends
- Careful alignment of pump, motor and coupling
- Mounting the baseplate on vibration dampers
- Connecting pipework to the pump with flexible bellows
- Use of vibration reducing pipework mountings and bushes when passing through walls etc.

Noise reduction is also achieved by the use of low noise motors, which have unidirectional cooling fans. If a gearbox drive is necessary, then this should also be of a low noise construction.
If satisfactory results are not obtained by the above measures, or the sound level is basically too high because of the design and absorbed power of the pump, then secondary measures must be adopted. This means the use of insulation and silencing materials.
A distinction is drawn between active measures to reduce the noise emission at source (sound reduction) and passive measures to reduce the effect on people working in the area (personnel protection).
Active measures in the main, comprise of sound reflecting or sound absorbing walls or complete enclosures. If maintenance access is required during operation, then the enclosure will be fitted with lockable service hatches or access doors.
Passive measures include the provision of sound proofed cabins and control rooms or ear protection.

To obtain the best sound reduction, it is necessary to have a sound / frequency analysis to decide on the most suitable protection measures to adopt.

5.4 Noise and sound pressure level

Table 5.04 Noise sources and sound pressure level

Noise source	Sound pressure level L_{pA} dB(A)	Noise source	Sound pressure level L_{pA} dB(A)
Jet aircraft	105 - 135	Motor car	77
Pneumatic hammer	90 - 105	Pumpset 10 kW	70
Pumpset 950 kW	89	Pumpset 2 kW	64
Pumpset 500 kW	87	Radio and television /	
Pumpset 100 kW	81	living room	55 - 65
Motor bus	79 - 83	Conversation	40 - 60
HGV	78 - 84	Quiet room	30 - 40
Pumpset: centrifugal pump and motor n = 3000 rpm			

The above sound pressure levels are average values.

The pain threshold for human hearing is ca. 130 dB(A).

Risk of hearing damage may occur at 85 dB(A), with continuous exposure over an 8 hour day for one year.

Table 5.05 Guide for noise nuisance limits measured at the nearest dwelling house (0.5 m from an open window)

Type of buildings in area	By day	By night
Mainly industrial	70 dB(A)	70 dB(A)
Mainly commercial buildings	65 dB(A)	50 dB(A)
Mixture of commercial and dwellings	60 dB(A)	45 dB(A)
Predominantly dwellings	55 dB(A)	40 dB(A)
Exclusively dwellings	50 dB(A)	35 dB(A)
Hospitals and nursing homes.	45 dB(A)	35 dB(A)

Night is defined as the eight hours between 2200 and 0600.

6 Head losses in pipework and fittings

6.1 Head losses H_J in straight pipework

6.1.1 General

For pipes with circular cross-section, and which are completely filled, the head loss H_J can be calculated according to DARCY-WEISBACH as follows:

$$H_J = \lambda \cdot \frac{l}{D} \cdot \frac{U^2}{2\,g}$$

where λ = coefficient of friction l = length of pipe

D = diameter of pipe U = mean flow velocity

The coefficient if friction λ is dependent on and determined by, the similarity law on the dimensionless Reynolds number:

$$Re = \frac{D \cdot U}{\nu}$$

For $Re < 2320$, i.e. laminar flow, the HAGEN-POISSEULLE law applies, regardless of the roughness of the pipe bore:

$$\lambda = \frac{64}{Re}$$

For $Re > 2320$, i.e. turbulent flow, which is generally the case in most conditions, the value of the coefficient of friction λ

- for the extreme case of a hydraulically smooth pipe, where λ is only dependent on Re and can be determined from:

  $$\frac{1}{\sqrt{\lambda}} = 2 \cdot \lg \left(\frac{Re \cdot \sqrt{\lambda}}{2.51}\right)$$

  The internal surface roughness of the pipe is within the boundary layer and has no influence.

- for the extreme case of a hydraulically rough pipe, where λ is only dependent on the internal roughness and diameter of the pipe and can be determined from:

  $$\frac{1}{\sqrt{\lambda}} = 1.14 \div 2.0 \cdot \lg \frac{k}{D}$$

  where k = internal surface roughness of the pipe

  The surface roughness of the pipe penetrates the boundary layer to some extent and affects the main flow.

For the usual pipe materials, diameters and flow velocities, there is a relationship between hydraulically smooth and hydraulically rough. For this transition range, the coefficient of friction λ has the following equation according to PRANDTL-COLEBROOK:

$$\frac{1}{\sqrt{\lambda}} = -2 \cdot \lg \left(\frac{2.51}{Re \cdot \sqrt{\lambda}} - \frac{k}{D} \cdot \frac{1}{3.71}\right)$$

In this range, the Reynolds number *Re* and the condition of the pipe expressed as absolute roughness *k* or relative roughness *k/D*, will affect the magnitude of the coefficient of friction λ.

For pipes with non-circular cross section, the diameter of an equivalent circular pipe which would have the same head losses for the same flow velocity, roughness and length can be calculated from:

$$D_{\text{equivalent}} = \frac{4\,A}{U} \quad \text{in mm}$$

where A = cross sectional area in mm²

U = wetted circumference in mm

This equivalent formula is also valid for open channels, on the assumption that the free surface of the liquid does not affect the resistance, (which is not entirely correct). This diameter can also be used for calculating the Reynolds number *Re*.

6.1.2 Determination of head loss H_J

6.1.2.1 Reynolds number *Re*

The determination of the Reynolds number in order to establish whether laminar or turbulent flow is present is only necessary for viscous liquids. In all other cases turbulent flow can be assumed.

Given:

D = diameter of the pipe in mm
(the nominal bore DN can be used)

U = mean flow velocity in m/s

Q = flowrate in m³/h

ν = kinematic viscosity in mm²/s

the Reynolds number *Re* can be calculated from

$$Re = \frac{D \cdot U}{\nu} \cdot 10^3 \qquad \text{or} \qquad Re = \frac{354 \cdot Q}{D \cdot \nu} \cdot 10^3$$

6.1.2.2 Head loss H_J for laminar flow (Re < 2320)

$$H_J = \lambda \frac{l}{D} \cdot \frac{U^2}{2\,g} \cdot 10^3 \qquad \text{in m}$$

where $\lambda = \dfrac{64}{Re}$

L = length of pipe in m

D = diameter of pipe in mm (= DN)

U = mean flow velocity in m/s

6.1.2.3 Head loss H_J for turbulent flow (Re > 2320)

The determination of λ using the PRANDTL-COLEBROOK equation is very time consuming. It is therefore more expedient to calculate the head loss H_J with the help of table 6.03 derived from PRANDTL-COLEBROOK.

Table 6.03 was established using the following parameters

- an absolute inner surface roughness k = 0.1 mm (new cast pipe with internal bitumen coating)
- a kinematic viscosity ν = 1.236 mm²/s. This is the value for pure water at 12°C. The values obtained are sufficiently accurate for water and other liquids of similar viscosities at normal ambient temperature.
- a pipe length of l = 100 m

If the absolute inner surface roughness of the pipe k differs greatly from the 0.1 mm (guideline values see table 6.01), then the head loss value obtained from table 6.03 must be multiplied by a correction factor f as follows:

$H_J = H_{J\,(k\,=\,0.1)} \cdot f$ $\qquad$ $H_{J\,(k\,=\,0.1)}$ established from table 6.03

f established from table 6.02

The correction factors in table 6.02 also enable an estimate of the change in head loss H_J which might be expected in a pipe after several years in service.

It should be noted that if the inner surface of a pipe becomes encrusted, then the reduced diameter resulting from this, is important.

If the kinematic viscosity of the liquid ν differs greatly from 1.236 mm²/s, then the head loss value H_J obtained from table 6.03 must be corrected as described in section 6.1.3.

Material and type of pipe	Condition of pipe	k in mm
new (ductile) cast iron	bitumen coated not bitumen coated cement lined	0.1 to 0.15 0.25 to 0.15 0.025
used (ductile) cast iron	evenly corroded slightly to heavily encrusted cleaned after several years in service	1 to 1.5 1.5 to 3 1.5
new seamless steel	rolled or drawn	0.02 to 0.05
new welded steel		0.04 to 0.1
new coated steel	zinc plated bitumen coated cement lined galvanised	0.1 to 0.15 0.05 0.025 0.01
used steel	evenly corroded slightly encrusted medium encrusted heavily encrusted	0.15 0.15 to 0.4 1.5 2 to 4
new concrete	commercial grade smooth commercial grade medium commercial grade rough	0.3 to 0.8 1 to 2 2 to 3
used concrete	after several years use	0.2 to 0.3
centrifugally spun concrete		0.25
drawn or extruded copper, brass, aluminium, plastic or glass	new used	to 0.01 to 0.03

Table 6.01 Roughness values k for various materials and conditions of pipe

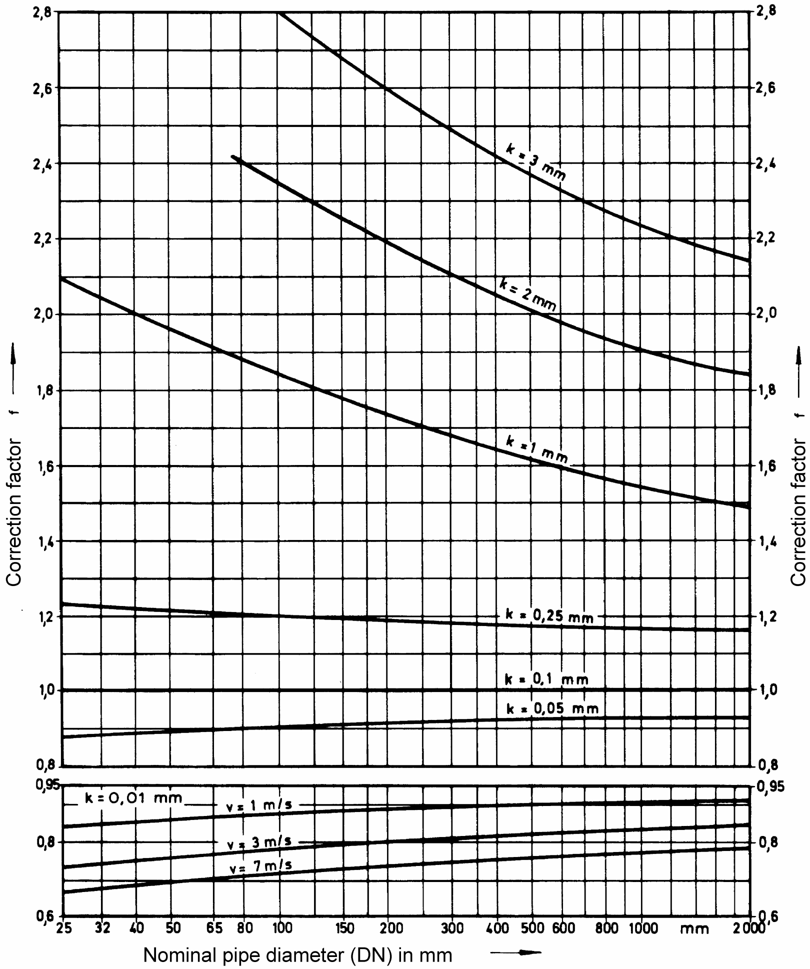


Table 6.02 Correction factor for roughness value $k \neq 0.1$ mm
For $k = 0.05$ to 3mm, the difference in the correction values due to flow velocity are negligible, so average values are shown. However for $k = 0.01$mm, the effect of flow velocity must be taken into account.

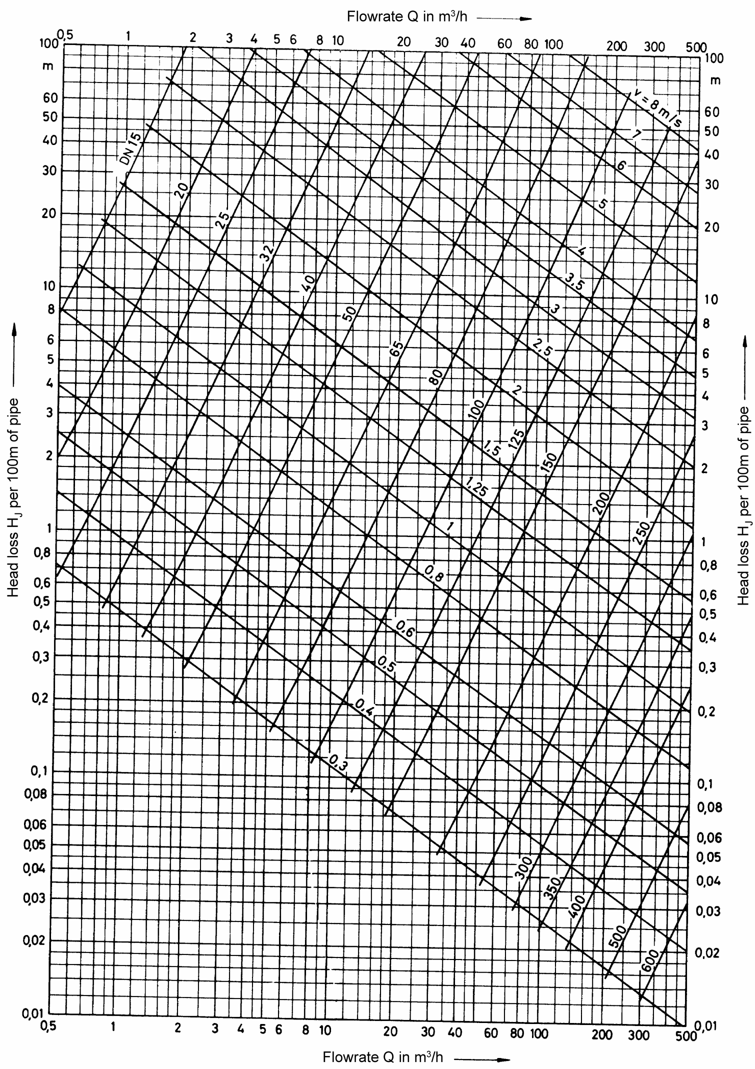


Table 6.03a Head loss per 100m of pipe using PRANDTL-COLEBROOK for k = 0.1mm and turbulent flow ν = 1.236 mm²/s.

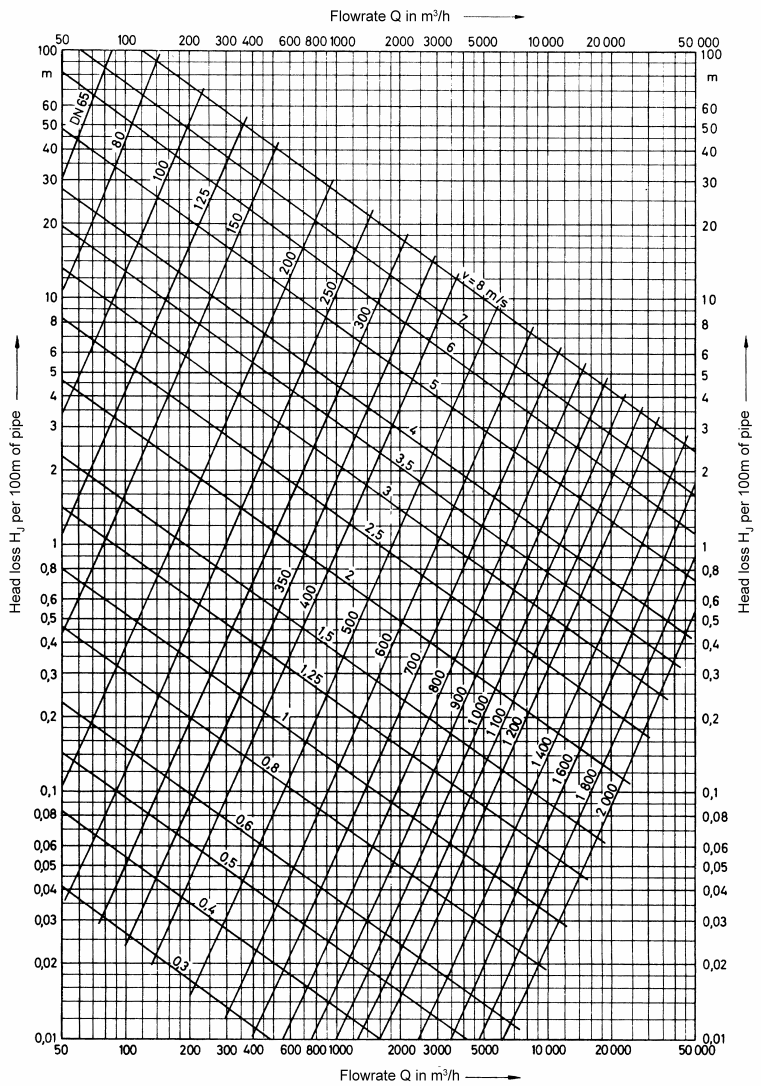


Table 6.03b Head loss per 100m of pipe using PRANDTL-COLEBROOK for k = 0.1mm and turbulent flow ν = 1.236 mm²/s.

Example:

Given: pure water with $t = 10$ °C ($\nu = 1.30$ mm²/s), $Q = 360$ m³/h

new steel pipe bitumen coated DN 200, $l = 400$ m

Reynolds number:

$$Re = \frac{354 \cdot 360}{200 \cdot 1.30} \cdot 10^3 = 4.9 \cdot 10^5 \qquad \text{i.e. turbulent flow}$$

From table 6.03 is given: $H_J = 4.5$ m/100 m, $U = 3.2$ m/s

From table 6.01 a roughness value for new steel pipe, bitumen coated $k = 0.05$ mm is given and from table 6.02 a correction value f = 0.91 so the head loss is obtained from:

$$H_{J\,(k = 0.05)} = 4.5\text{ m} \cdot 0.91 = 4.1\text{ m/100 m}$$

or for the entire pipe length $l = 400$ m:

$$H_J = 4.1\text{ m} \cdot \frac{400\text{ m}}{100\text{ m}} = 16.4\text{ m}$$

6.1.3 Correction of head loss H_J for liquids with a kinematic viscosity $\nu \neq$ 1.236 mm²/s

Assumption of turbulent flow with $Re > 2320$.

The correction is carried out in three steps a) to c). The subscript *'x'* denotes the values for the given kinematic viscosity.

a) $Q = Q_x \cdot \dfrac{1.236}{\nu}$ in m³/h with Q in m³/h and ν in mm²/s

b) from table 6.03, H_J is evaluated as described in section 6.1.2 for the flowrate Q and the given nominal bore of pipe.

c) $H_{J\,x}$ is then obtained from:

$$H_{Jx} = H_J \cdot \left(\frac{\nu}{1.236}\right)^2 \quad \text{in m per 100 m pipe}$$

with H_J in m per 100 m pipe and ν in mm²/s

Example:

Given: Liquid with $\nu = 20$ mm²/s, $Q = 150$ m³/h

New cast pipe, bitumen coated ($k = 0.1$ mm), DN 100, $l = 50$ m

$$Re = \frac{354 \cdot 150}{100 \cdot 20} \cdot 10^3 = 2.65 \cdot 10^4 \qquad = \text{turbulent flow}$$

a):

$$Q = 150 \text{ m}^3/\text{h} \cdot \frac{1.236 \text{ mm}^2/\text{s}}{20 \text{ mm}^2/\text{s}} = 9.3 \text{ m}^3/\text{h}$$

from table 6.03

$$H_J = 0.13 \text{ m per 100 m pipe}$$

and the required value

$$H_{Jx} = 0.13 \text{ m} \cdot \left(\frac{20 \text{ mm}^2/\text{s}}{1.236 \text{ mm}^2/\text{s}}\right)^2 = 34 \text{ m per 100 m pipe}$$

or for the entire pipe length $l = 50$ m:

$$H_{Jx} = 34 \text{ m} \cdot \frac{50 \text{ m}}{100 \text{ m}} = 17 \text{ m}$$

The flow velocity is obtained from table 6.03 for $Q = 150$ m³/h and DN100 as $U = 5$ m/s.

6.2 Head loss H_J in valves and fittings

$$H_J = \zeta \cdot \frac{U^2}{2\,g} \quad \text{in m}$$

with U = average flow velocity across the reference section in m/s

ζ = friction coefficient of the fitting

or. $$H_J = \Sigma\,\zeta \cdot \frac{U^2}{2\,g} \quad \text{in m,}$$

if the friction coefficients ζ of all the fittings are summed, which is only correct if all have the same nominal bore.

The following tables summarise the friction coefficients of the most commonly used valves and fittings. Using known values for U and ζ, the value H_J can easily be obtained from table 6.04.

Valves and fittings

Straight through valves - fully open

Globe valves vertical spindle

Cast body, DN 25 to 200	$\zeta = 2.5$
Forged body, DN 25 to 50	$\zeta = 6.5$

Inclined seat, full flow valve with inclined stem

DN	25	32	40	50	65	80	100	125 to 200
ζ	1.7	1.4	1.2	1.0	0.9	0.8	0.7	0.6

Angle valve - fully open   DN 25 to 200: $\zeta = 2.0$

Non-return valve

Flat seat valve, DN 25 to 200	$\zeta = 3.5$
Inclined seat valve, DN 50 to 200	$\zeta = 2.0$

Foot valve with strainer

DN	50 to 80	100 to 350
$U = 1$ m/s	$\zeta = 4.1$	$\zeta = 3$
$U = 2$ m/s	$\zeta = 3.0$	$\zeta = 2.25$

Combination of foot valves

DN	400	500	600	700	800	1000	1200
ζ	7.0	6.1	5.45	4.95	4.55	4.05	3.9

Butterfly and shut off valves - fully open

DN		400	600	800	1000	1200	1500
ζ for	PN 2.5			0.08	0.06	0.05	0.13
	PN 6			0.16	0.30	0.25	0.22
	PN 10	0.48	0.33	0.50	0.45	0.41	0.37
	PN 16	1.20	0.85	0.73	0.63		

Non-return valves, without lever and weight

DN		200	300	500	600	700	800	1000	1200
ζ for	U =1m/s	2.95	2.90	2.85	2.70	2.55	2.40	2.30	2.25
	U =2m/s	1.30	1.20	1.15	1.05	0.95	0.85	0.80	0.75
	U =3m/s	0.76	0.71	0.66	0.61	0.54	0.46	0.41	0.36

If the check valves are fitted with lever and weight, the loss coefficients can amount to a multiple of the values given. A rough estimate can be obtained by application of the following factors: for $U = 1$m/s$\rightarrow$ f = 4, for $U = 2$m/s$\rightarrow$ f = 3 and for $U = 3$m/s$\rightarrow$ f = 2.

Swing type non-return valves, with lever and weight

U	1 m/s	1.5 m/s	2 m/s	≥ 2.5 m/s
ζ	8	3	1.3	0.7

Flap valves

The coefficient of friction of flap valves depends on the flow velocity, the design and the weight of the flap. For this reason reliable figures can only be given by the manufacturer. For rough approximation, the following values can be used: ζ = 1.0 to 1.5.

Back flow preventer (HYDRO - STOP)

DN		50	100	150	200	250	300	400
ζ for	U=2 m/s	5	6	8	7.5	6.5	6	7
	U=3m/s	1.8	4	4.5	4	4	1.8	3.4
	U=4m/s	0.9	3	3	2.5	2.5	1.2	2.2

Gate valve, flat type fully open

DN	100	200	300	400	500	600 to 800	900 to 1200
ζ	0.18	0.16	0.14	0.13	0.11	0.10	0.09

Gate valve, oval and round type fully open

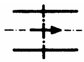
DN	100	200	300	400	500	600 to 800	900 to 1200
ζ	0.22	0.18	0.16	0.15	0.13	0.12	0.11

Needle valve, fully open

with reference to the smallest cross section ζ = 0.5 to 0.8

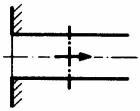
Fittings

The reference cross section for the flow velocity is always denoted by



Inlet, also the outlet from a container into a pipe

sharp edged

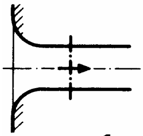


very sharp $\zeta = 0.5$

slight chamfer $\zeta = 0.25$

full chamfer $\zeta = 0.2$

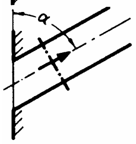
rounded entry



smoothness ζ = 0.06 to 0.05

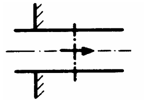
normal $\zeta = 0.05$

sharp edged
with angle α



α	45°	60°	75°
ζ	0.8	0.7	0.6

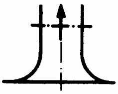
projecting sharp edged



very sharp $\zeta = 3$
slight chamfer $\zeta = 0.6$

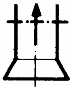
Pump intake

bell mouthed



$\zeta = 0.05$

conical

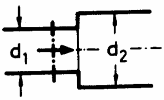


$\zeta = 0.20$

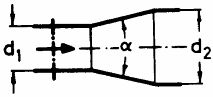
Outlet, outlet loss $\zeta = 1$

The flow velocity in the outlet cross section is the determining parameter

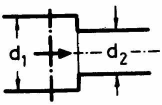
Changes in cross section



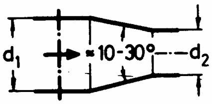
d_1/d_2	0.5	0.6	0.7	0.8	0.9
ζ	0.56	0.46	0.24	0.13	0.04



d_1/d_2		0.5	0.6	0.7	0.8	0.9
ζ for	α=8°	0.12	0.09	0.07	0.04	0.02
	α=16°	0.19	0.14	0.09	0.05	0.02
	α=25°	0.33	0.25	0.16	0.08	0.03



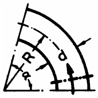
d_1/d_2	1.2	1.4	1.6	1.8	2.0
ζ	0.10	0.22	0.29	0.33	0.35



d_1/d_2	1.2	1.4	1.6	1.8	2.0
ζ	0.02	0.05	0.10	0.17	0.26

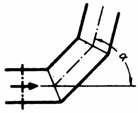
Bends

swept



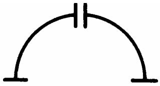
α		45°		60°		90°	
surface		smooth	rough	smooth	rough	smooth	rough
ζ for	R=d	0.14	0.34	0.19	0.46	0.21	0.51
	R=2d	0.09	0.19	0.12	0.26	0.14	0.30
	R≥5d	0.08	0.16	0.10	0.20	0.10	0.20

fabricated

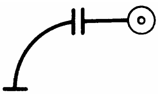


α	45°	60°	90°
Number of weld seams	2	3	3
ζ	0.15	0.2	0.25

90° bends in series



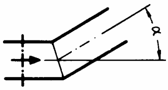
$2 \cdot \zeta_{90°}$



$3 \cdot \zeta_{-90°}$

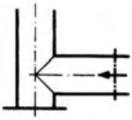
$4 \cdot \zeta_{90°}$

Elbow bends

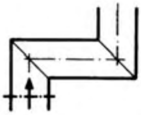


α	45°		60°		90°	
surface	smooth	rough	smooth	rough	smooth	rough
ζ	0.25	0.35	0.50	0.70	1.15	1.30

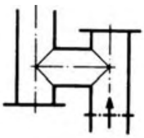
Combination with 90° elbows



$\zeta = 2.5$



$\zeta = 3$

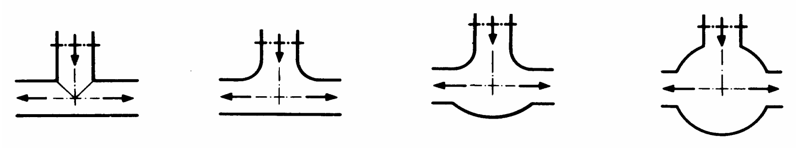


$\zeta = 5$

Expansion joints

Pipe expansion joints, with/without guide tube	$\zeta = 0.3 / 2.0$
"Lyra" expansion section smooth	$\zeta = 0.7$
"Lyra" expansion bellows	$\zeta = 1.4$

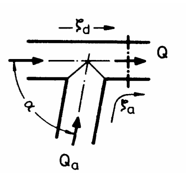
Tee-pieces, with flow separation



sharp edged	rounded with straight base	spherical with concave neck	spherical
$\zeta = 1;3$	$\zeta = 0.7$	$\zeta = 0.9$	$\zeta = 2.5$ to 4.9

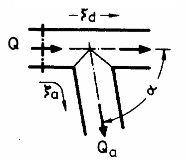
Branches, main line and branch of same nominal bore

Confluence



		$Q_a = 0$	$Q_a = 0.5\,Q$	$Q_a = 0.8\,Q$	$Q_a = Q$
$\alpha=90°$	ζ_d	0.04	0.35	0.5	-
	ζ_a	-	0.3	0.7	0.9
$\alpha=45°$	ζ_d	0.04	0.2	0.1	-
	ζ_a	-	0.15	0.35	0.4

Divergence



		$Q_a = 0$	$Q_a = 0.5\,Q$	$Q_a = 0.8\,Q$	$Q_a = Q$
$\alpha=90°$	ζ_d	0.04	0.01	0.2	-
	ζ_a	-	0.9	1.1	1.3
$\alpha=45°$	ζ_d	0.04	0.02	0.2	-
	ζ_a	-	0.4	0.35	0.5

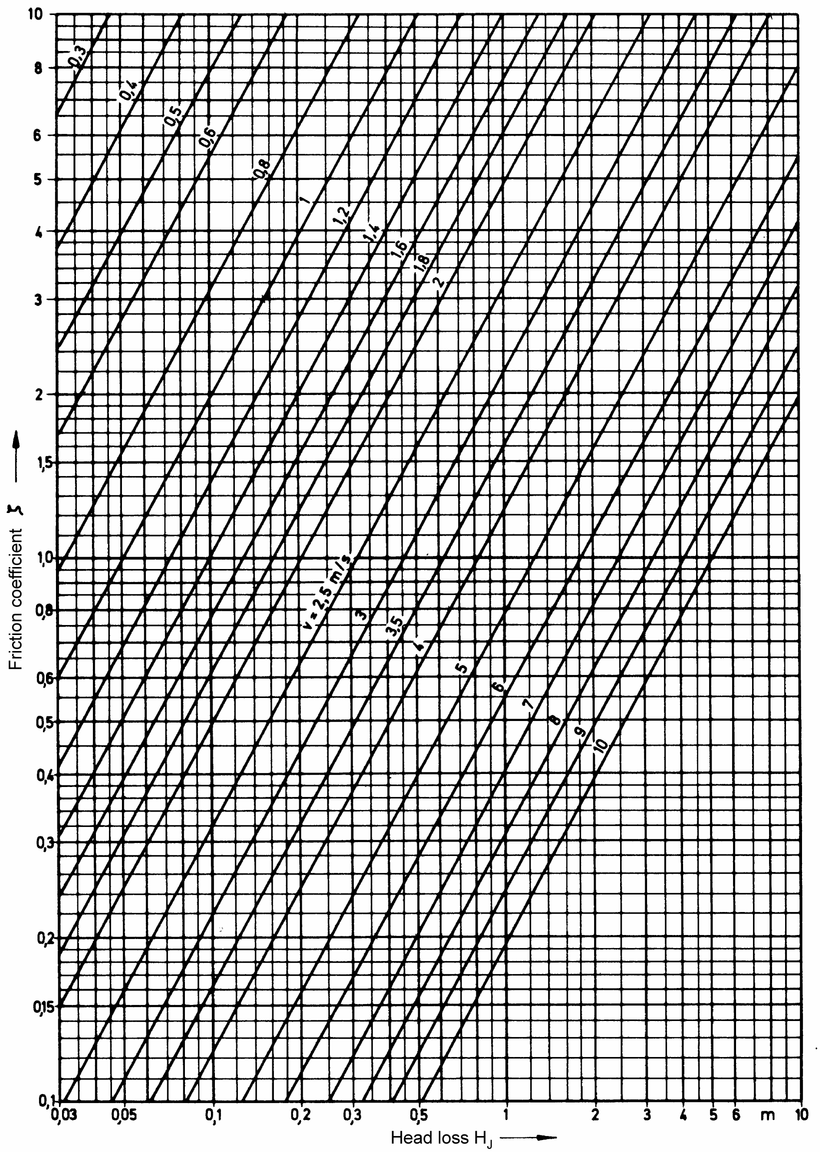


Table 6.04 Determination of the head loss $H_J = \zeta \cdot \dfrac{U^2}{2\,g}$

7 Flanges

7.1 Flanges to European and German Standards

7.1.1 Definitions

PN nominal pressure

Standard EN 1333 defines PN as an alpha-numeric designation which relates the mechanical and dimensional properties of a pipe system component. The letters PN are followed by a dimensionless whole number, which indirectly correlates with the magnitude of the design pressure, in bar, of the connections at a temperature of 20°C.

DN nominal bore

Standard EN ISO 6708 defines DN as an alpha-numeric designation for the size of pipe system components, which is used for reference purposes. The letters DN are followed by a dimensionless whole number, which indirectly correlates with the magnitude of the bore or the outer diameter, in millimetre, of the connections.

Pressure / temperature rating

The pressure / temperature rating defines the permissible pressure (surge free) at various temperatures. The relationship is dependent on the nominal pressure rating (PN) and the materials of the flange. The temperature refers to that of the pumped liquid.
The pressure / temperature ratings are given in standard flange tables.
However the pressure / temperature rating of the flange may not necessarily apply to the entire pipe system. The pressure / temperature rating of fittings, instruments, gaskets and pumps can restrict the pressure / temperature rating of the flange connection. Data from the manufacturer should therefore always be taken into account.

7.1.2 Flanges to European EN, (DIN EN, BS EN) Standards

Cast iron flanges to EN 1092-2

The standard covers the following nominal pressure ratings

PN 2,5	PN 6	PN 10	PN 16	PN 25	PN 40

Flanges normally raised face type B

Ductile iron (SG) flanges to DIN EN 1092-2

The standard covers the following nominal pressure ratings

PN 10	PN 16	PN 25	PN 40	PN 63

Flanges normally raised face type B

Cast steel flanges to EN 1092-1

The standard covers the following nominal pressure ratings

PN 6	PN 10	PN 16	PN 25	PN 40	PN 63	PN 100

Flanges normally raised face type B

7.1.3 Flanges to German DIN Standards

Cast iron flanges

The standard covers the following nominal pressure ratings

PN 2,5 DIN 2530	PN 6 DIN 2531	PN 10 DIN 2532	PN 16 DIN 2533	PN 25 DIN 2534	PN 40 DIN 2535

Flanges normally raised face type C

Note! These standards are no longer applicable for new equipment. EN 1092-2 should be used.

Cast steel flanges

The standard covers the following nominal pressure ratings

PN 16 DIN 2543	PN 25 DIN 2544	PN 40 DIN 2545	PN 64 DIN 2546	PN 100 DIN 2547
PN 160 DIN 2548	PN 250 DIN 2549	PN 320 DIN 2550	PN 400 DIN 2551	

Flanges normally raised face, PN 16 to PN 40 type C

PN 64 to PN 400 type E

Note: The standards for PN16 to PN100 are no longer applicable for new equipment. EN 1092-1 should be used.

7.2 Flanges to American ANSI Standards

7.2.1 Definitions

Class

The flanges are categorised by class. The classes have similar meaning to the European nominal pressure rating, although this term is not used. The figure given after the class refers to a pressure in psig at a given temperature, according to the material, which lies between 65 and 650°C.

However for steel flanges to ANSI B 16.5, the class is comparable with nominal pressure rating.

NPS

The expression NPS = nominal pipe size is comparable with the European term nominal bore (DN)

NPS is the nominal diameter in inches and is applicable to connecting fittings.

Pressure / Temperature (P/T) rating

The pressure / temperature rating defines the permissible pressure at various temperatures. The relationship is dependent on the pressure class and the materials of the flange and for cast flanges on the nominal pipe size (NPS).

The rating is independent of the pumped liquid. The temperature refers to that of the pipe system, although it can be assumed to be the same as that of the pumped liquid.

The pressure / temperature ratings are given in standard flange tables. However the pressure / temperature rating of fittings, instruments, gaskets and pumps according to data from the manufacturer should always be taken into account.

7.2.2 Cast iron flanges to ANSI B 16.1

The standard covers the following nominal pressure classes

Class 25 (≈ PN 2,5) 1)	Class 125 (≈ PN 10) 1)	Class 250 (≈ PN 25) 1)	Class 800 (≈ PN 40) 1)

1) These values are not given in Standard ANSI B 16.1, i.e. they are not official and serve only for comparison.

Flanges CL.25 and 125FF, CL.250 and 800RF

7.2.3 Cast steel flanges to ANSI B 16.5

The standard covers the following nominal pressure classes

Class						
150 PN 20	300 PN 50	400 PN 68	600 PN 100	900 PN 150	1500 PN 250	2500 PN 420

Flanges normally raised face type RF

7.3 Flanges to International ISO Standards

7.3.1 Cast iron flanges to ISO 7005-2

The standard covers the following nominal pressure ratings

Preferred series (Series 1):

PN 10	PN 16	PN 20	PN 50

Optional series (Series 2):

PN 2,5	PN 6	PN 25	PN 40

Flanges: Normally raised face type RF

Exception: PN 20, without raised face FF (flat face)

7.3.2 Ductile iron (SG) flanges to ISO 7005-2

The standard covers the following nominal pressure ratings

Preferred series (Series 1):

PN 10	PN 16	PN 20	PN 50	PN 110	PN 150	PN 260	PN 420

Optional series (Series 2):

PN 2,5	PN 6	PN 25	PN 40

Flanges normally raised face type B

7.3.3 Cast steel flanges to ISO 7005-1

The standard covers the following nominal pressure ratings

Preferred series (Series 1):

PN 10	PN 16	PN 20	PN 50	PN 110	PN 150	PN 260	PN 420

Optional series (Series 2):

PN 2,5	PN 6	PN 25	PN 40

Flanges normally raised face type B

7.4 Test pressure

All components of a pump installation which are subject to pressure, i.e. pumps, pipework, fittings etc. must pass a hydrostatic test. This is to confirm the mechanical strength of the component but also primarily the leak tightness.

The designated test pressure should generally be applied for a period of 30 minutes.

The test pressure is generally set at between 1.3 to 1.5 times the maximum operating pressure, or the maximum permissible operating pressure, or the nominal pressure rating of the flanges.

Clean cold water is used for the hydrostatic test.

8 Instrumentation for monitoring centrifugal pump installations

Instrumentation is installed in pump installations to monitor and control the various operating parameters such as flowrate, pressure, power absorbed, speed and temperature. In most cases the data is read directly at the point of measurement, but with appropriate equipment can be transmitted to a remote central control room, where it can be displayed, recorded and if necessary acted upon.
The following instrumentation is generally used to monitor centrifugal pump installations.

8.1 Pressure measurement

The instruments for pressure measurement are designated according to their operating principle as:

- liquid filled manometers (columns) and dead weight gauges
- mechanical instruments (dial gauges)
- electronic instruments (transducers)

8.1.1 Manometers

Liquid filled manometers are available as U-tube or single column (with reservoir) or concentric tube.
Liquid filled manometers are simple, accurate and reliable and advantageously can also measure negative pressures. The liquid column is normally provided by mercury (p_{max} = 2.5 bar) or tetrabromomethane (p_{max} = 0.2 bar). They cannot be used for remote monitoring.
Liquid filled manometers are generally not used for monitoring processes, rather as a scientific instrument for research and testing.
The dead weight piston and cylinder manometer is the most accurate method for measuring higher pressures. They are therefore principally used for standardisation and calibration of mechanical gauges.

8.1.2 Mechanical instruments

Dial gauges are used to measure static pressure in the form of absolute pressure, positive or negative pressure, or differential pressure.
Due to their robustness and ease of use, these are widely installed.
These use a flexible tube or plate spring such as the “Bourdon” type. These are used for example to measure the total head on larger oil or cooling water pumps and utilise a bent tube, sealed at one end which is under internal pressure.

The tendency of this tube to straighten against the measured pressure provides the means of moving the indicating needle.
For higher accuracy, precision gauges of class 0.6 are used. These must be accurate to ± 0.6% of their full scale range. For general monitoring purposes, gauges of class 2 (accuracy to ± 2% of their full scale range) are used. The device should be selected such that it operates within its most accurate range, i.e. > 40% of its full scale range.

Diaphragm pressure gauges utilise the deflection of a diaphragm which is clamped around its circumference and is subject to the measured pressure on one side only. The disadvantage of this type is the sensitivity to overload and the high gearing of the pointer, due to the small movement of the diaphragm. Their advantage is in ease of use especially for measurement of pressure differences. Furthermore the use of a diaphragm makes it ideal for measurement of sticky or aggressive fluids.

8.1.3 Electronic pressure transducers and displays

Electronic pressure measurement based on piezo-resistance or inductance or strain gauges are used to measure static and dynamic pressure changes. As well as absolute pressures above and below atmospheric, they can measure pressure differences. Their range covers 0.1 to 4000 bar with a measurement class of 0.1. They are also ideal for measurement of pressure peaks which may occur with surges.

8.2 Flow measurement

Flow measurement instruments can be:

- flowrate measurement, or
- volumetric meters

8.2.1 Flowrate measurement

Flowrate measurement instruments include:

- flowrate / pressure differential devices
- magnetic inductive instruments
- ultrasonic devices
- flotation devices

Flow measurement using pressure differentials is a universally applicable method. It utilises the principal that throttling a stream flow will cause a differential in pressure. Throttling devices may be an orifice plate, nozzle, venturi tube or venturi nozzle.

The standard orifice plate is a flat disc with a concentric hole with a sharp edge on the upstream side. With a nozzle, the inlet is rounded and is extended in the direction of flow. The standard venturi is similar to the nozzle on the inlet side, but has a diffuser form outlet to reduce the permanent pressure drop. The dimensions, geometry and installations are all laid down in various standards and hence these devices are often referred to as standard orifices etc.

The measurement devices for the differential pressure can be liquid filled manometers or electronic differential pressure instruments.

Magnetic induction flow measurement, depends on the Faraday law of induction and is the standard method for conductive liquids. The conductivity must be at least 20 µS/cm. This value can be reduced to 0,05 µS/cm with special designs. This allows this method to be used for distilled water (conductivity < 5 µS/cm) and for boiler feed water (conductivity 0,2 µS/cm). A homogenous magnetic field, vertical to the direction of flow, induces a voltage in the conductive liquid, which is proportional to the flow velocity. This system has the advantage that no additional pressure drop is caused by the instrument. Furthermore it is suitable for pulpy and muddy liquids and with the selection of suitable materials or coatings, can be used for foodstuffs, acids and alkalis.

Ultrasonic flow measurement is mostly used when the conductivity of the liquid is so poor as to make the magnetic induction method impossible, e.g. mineral oil. A piezo electric head is located at an angle to the flow and alternately transmits and receives ultrasound signals through the flow. The passage through the liquid takes a different time on the in and out direction and from this the flow velocity and hence the flowrate can be calculated.

For monitoring low flowrates up to ca. 120 m^3/h, flotation type instruments are widely used. In these instruments floats are lifted by the force of the flow, up vertical measuring columns to a point where the weight of the float is balanced. This point is a measure of the flow velocity and from this the flowrate can be read off a calibrated scale on the column.

8.2.2 Volumetric flow meters

Volumetric meters include the following:

- turbine type
- gear wheel type
- rotary piston type

For centrifugal pumps, the turbine type flow meter is most widely used. With this instrument the speed of rotation of the turbine is proportional to the flow velocity. It is suitable for hot and cold liquids and in materials suitable for corrosive liquids. A condition of all volumetric flow meters is that the liquid is clean.

The gear wheel type meter is especially suitable for mineral oil products, up to a viscosity of $1 \cdot 10^5$ mPa·s and also for milk, fruit juices, acids and alkalis. Using the positive displacement principle the wheels are turned by the pressure differential and shown on the indicator. For very high viscosity the wheels can be heated.

The rotary piston type is used for similar applications and an eccentrically mounted piston is rotated by the flow.

8.2.3 Flow indicators

For the monitoring of service liquids, such as cooling water, barrier liquids and lubricating oil, it is not necessary to have a quantitative flow measurement. It is however often advisable for the safe operation of the plant to use a flow indicator to monitor the supply. Such devices use a rotating ball or a flag to give a visual flow check. They can also use electrical contacts to give a loss of flow signal.

8.3 Power measurement

The power requirement of a centrifugal pump can generally be measured in two ways:

- torque measurement
- electrical power measurement

8.3.1 Torque measurement

8.3.1.1 Torque measurement with strain gauges

This measurement is made on torque transmitting shafts between the pump and driver.
For this reason it is generally only used on test beds.

8.3.1.2 Torque measurement with eddy current sensors

This method relies on the principle that the permeability of the magnetic field lines is altered by mechanical strain. The sensor head, fixed adjacent to the pump shaft, generates a magnetic field which penetrates the shaft and induces an electrical voltage in the secondary coils of the sensor head, which is proportional to the torque. The measurement is contact free with no negative influence.
This method can only be used if there is an accessible section of shaft for positioning the sensor on the drive side of the pump.

8.3.2 Measurement of electrical power

The power absorbed by the electrical motor is measured and from the efficiency of the motor, the power absorbed by the pump can be derived. The motor efficiency is taken from IEC guidelines and figures given by the manufacturer.
With DC drives it is sufficient to measure the current and voltage to calculate the electrical power.
For three phase motors the power in any two phases is measured with two wattmeters and the total power obtained by their sum. More sophisticated meters will read both phases and calculate the total power giving a single reading.
If there is a gearbox between the pump and motor, a means of determining the loss of efficiency must be established.

8.4 Speed measurement

The speed of a centrifugal pump can be measured in several ways:

- mechanical tachometer
- impulse transmitter
- eddy current generators
- slip meters

Mechanical tachometers, generally hand held, measure the rotations of a counter over a period of time, and give the running speed of a machine. The measurement is usually made in the countersunk free end of the shaft.

Impulse transmitters can be inductive or optical and usually give the reading of impulses per unit of time with an electronic counter. The impulses can be generated for example by a shaft mounted gearwheel with a suitable number of teeth.
The eddy current meter operates electronically using the linear proportionality of a DC or AC generated current to the speed. The generator is directly coupled to the pump shaft.
For centrifugal pumps which are driven by AC and 3-phase motors, the speed can be calculated from the supply frequency and the slip speed of the motor. A coil mounted at a suitable point on the motor casing is the transmitter and a moving coil meter acts as indicator. The particular advantage of this method is that no free shaft end is needed, e.g. with canned rotor motors.

8.5 Temperature measurement

Temperature measurement with centrifugal pumps is usually made with direct contact instruments. This means there must be good heat exchange between the object whose temperature is to be measured and the sensing device, but conversely that heat is not transferred to the outside.
Contact thermometers can be divided into:

- mechanical contact thermometers
- electrical contact thermometers

Mechanical contact thermometers are mostly liquid filled glass tubes or springs with a range of -200°C to 800°C depending on the liquid (usually alcohol, toluene or mercury).

<u>and</u>

Metal expansion thermometers, either as expanding bar or bimetal thermometer with a range between -50°C and 1000°C.

Electrical direct contact thermometers are familiar in the form of resistance devices. They rely on the linear proportionality of the electrical resistance of platinum with change of temperature. They are almost exclusively known as Pt100 (100Ω at 0°C). and have a range of -200°C to 750°C.
Thermocouples depend for operation on the fact that two dissimilar metals when welded or soldered together generate a small voltage when subject to change of temperature. This voltage is extremely small, but proportional to the change in temperature and is dependent on the pairing of metals. The range is -270°C to 1770°C according to the metal pair.

Resistance thermometers and thermocouples have the advantage that the measured value is given directly as an electrical signal and can therefore be recorded, transmitted or used for control.

8.6 Vibration measurement

The measurement is made of the vibration velocity, (see section 5.1). It is made radially in the area of the bearing or on the shaft.
Vibration monitoring is primarily for indicating the overall reliable operation of the centrifugal pump. If the limit values given in table 5.01 are exceeded for a period of time, then the cause should be investigated and the problem rectified.
Continuous monitoring would normally only be carried out on pumps of high capital value or with high consequential costs of breakdown. For most centrifugal pumps, periodic checks with portable equipment are adequate.

8.7 Level measurement

Liquid level measurement in tanks, receivers and sumps is necessary for the switching on and off the pumps and to prevent dry running. For this purpose mostly pressure, contact or reed switches are used or similar transmitters to give an electrical signal to operate monitoring and control gear.

9 Fundamentals of electrical drives

Electric motors are the most widely used drive for centrifugal pumps.
At the forefront are three phase induction motors which are available in a practically unlimited performance range.
Single phase alternating current motors are generally only used where three phase supply is not available.
Direct current motors are installed for emergency equipment such as emergency lubricating oil pumps which are necessary to operate a plant or bring it safely to standstill in the event of failure of the main supply. Such motors are generally operated from battery packs, (UPS).

9.1 Electrical supply

9.1.1 Current and voltage

An electric current is the movement of negatively charged particles (electrons) carrying an electric charge through metallic conductive material, semiconductors, liquids (electrolyte) and gases. The conventionally agreed direction of electrical flow from positive to negative poles is in fact in the opposite direction to the actual electron flow.
The electrical potential (voltage) is the motive power for the current flow in an electrical circuit. The flow of an electric current represents the continuous and constant or alternating movement of charged particles in a conductor.

9.1.2 Direct current supply (DC)

An electric current with a constant direction of flow is designated as direct current. DC generators are used to generate direct current from non-electrical sources. As mains power supplies are universally alternating current, DC supply is normally obtained from alternating current and three phase supplies using rectifiers or in the case of emergency equipment (e.g. standby oil pumps) from battery packs (accumulators).
The most commonly used DC supplies are 110, 220, 440, 550 and 600 Volt (V) and for low power 24 to 60 Volt.

9.1.3 Alternating current (AC) and three phase supply

An electric current whose direction and strength varies periodically, usually following a sine curve, between positive and negative limit values, is designated as alternating current.
An alternating voltage is induced in a coil which rotates in a homogenous magnetic field. One complete rotation of the coil of 360° induces one complete alternating voltage cycle. Continuation of this effects the “electromagnetic generation” of alternating current.
Unlike direct current the magnitude and direction of AC is cyclic.

The simplest form of AC supply is the single phase system, however in practice multiphase systems are preferred. The standard supply is 3-phase.
Three phase is the most important type of energy supply of the electrical generation industry. It is formed by three single phase currents which are displaced by 120° from each other in the time cycle and are carried by three conductors, or four conductors in a three phase and neutral system. Alongside the line voltage (delta voltage) there is available the star voltage (between line and neutral star point) which is smaller than the line voltage by a factor √3. Power stations generate almost exclusively three phase supply. Lighting systems, small motors and power tools use single phase which is taken between a three phase line and neutral.
Three phase low voltage systems are formed by three main lines L1, L2, L3 and the neutral line N. The neutral point is connected at the star point of the generator. Two lines only or a single line and neutral form a single phase alternating current system.
The voltage between the lines (L1, L2, L3) is the supply voltage U_L; the voltage between the line and neutral is the star voltage U.

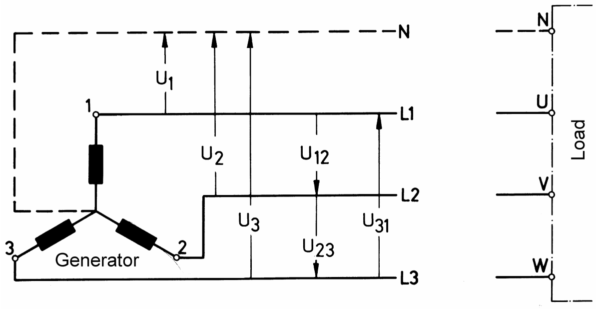


Fig. 9.01 Three phase system

Alternating currents are generated with various frequencies (cycles per second). The frequency f carries the unit Hertz (Hz). The European grid supply system operates on a stable 50Hz supply. Some exceptions include local supply systems and railway systems.
Outside Europe, systems with 60Hz supply are also common, although some countries have parts with 50Hz and some with 60Hz. In cases of doubt the end user should be consulted.

Table 9.01 Frequencies outside of Europe

		Exception
Africa	50 Hz	Liberia: 60 Hz
America	60 Hz	Barbados, Chile, Jamaica, Paraguay, Uruguay: 50 Hz Bolivia, Guyana, Haiti: 50/60 Hz
Asia	50 Hz	Korea, Philippines, Saudi-Arabia, Taiwan: 60 Hz Japan: 50/60 Hz
Australia and New Zealand	50 Hz	

Electrical supplies are divided into three ranges, very low voltage up to 42 V, low voltage to 1000 V and high voltage > 1000 V. Very low voltages are used in toys (up to 24 V) and safe area working lights e.g. in vessels. Low voltage and high voltage are the main supplies for electrical drives.

Table 9.02 Low voltage three phase supplies

Frequency	Rated voltage to DIN IEC 38 (tolerance ± 10%)	Permitted voltage range to DIN VDE 0530 or DIN IEC 34-1 (additional tolerance of ± 5%)
50 Hz	230V Δ / 400V Y	220...240V Δ / 380...420V Y
	400V Δ / 690V Y	380...420V Δ / 660...725V Y
60 Hz	460V Y	440...480V Y
	460V Δ	440...480V Δ

Table 9.03 High voltage selection

50 Hz		3 kV		6 kV	6.6 kV		10 kV	
60 Hz	2.4 kV		4.8 kv			6.9 kV		12 kV

9.2 Electric motors

Electric motors convert electrical energy in to mechanical work. Electric motors utilise the power generated by a magnetic field on an electrical conductor and the torque produced. Motors consist of a fixed stator and a rotor which are separated by an air gap.

Dependent on the available or chosen supply, the following types of motor are available:

- DC motors
- Single phase AC motors
- Three phase motors

9.2.1 Direct current motors

DC motors are available as:

- Shunt wound motors
- Series wound motors
- Compound wound motors
- Externally excited shunt wound motors (permanent magnet).

For driving centrifugal pumps with DC, shunt wound motors are exclusively used.
In DC shunt wound motors, the field (stator) and armature (rotor) windings are connected in parallel to the mains supply.
Compared to other types of DC motors, the shunt wound has the advantage that the speed is almost independent of the load.
Centrifugal pumps with DC shunt wound motors are generally limited to special applications such as vehicles, ships and for driving emergency equipment (e.g. standby oil pumps). The DC supply is normally obtained from DC generators, rectifiers or battery packs (accumulators).

9.2.2 Single phase alternating current motors

Single phase induction motors are built for low powers up to approximately 5 kW. They are connected to a single phase mains supply, most commonly 230 V. Single phase motors can be connected to a three phase supply, utilising the star voltage (U).
As the single phase supply does not give the motor a defined direction of rotation, it will not start from rest.
To give the motor a defined direction of rotation, an auxiliary winding (starter winding) is fitted in the stator and supplied via a capacitor, with a current which is displaced from the stator phase. In this way a rotating magnetic field in a defined direction is generated.
A running capacitor is as a rule, suitable for centrifugal pumps. The starting torque produced is 0.3 to 0.4 times the rated torque. If this is insufficient then starting and running capacitors can be fitted. This produces a starting torque of 1.5 to 1.8 times the rated torque. The starting capacitor is switched out by a centrifugal switch as the motor gets up to speed.

9.2.3 Three phase motors

Three phase induction motors are constructed as both low voltage and high voltage. The upper power limit for low voltage motors is approximately 800 kW. High voltage motors start from about 160 kW, but they are predominantly used in the range 1 to 11 MW.

Three phase motors are available as:

- Asynchronous squirrel cage induction motors
- Asynchronous slip ring motors
- Synchronous motors

9.2.3.1 Three phase asynchronous squirrel cage induction motors

This type of motor, by far dominates the market for centrifugal pump drives. The operating characteristics as a rule meet the requirements and the simplicity of mechanical construction can hardly be bettered. The rotor does not need a power supply and hence no commutator, slip rings or brushes and apart from the bearings there are no wearing parts.

The most commonly used machines are standardised with respect to power output and dimensions for a particular type of construction and the degree of protection and thus the process of designing the pump drive is simplified.

The electrical parts of the motor consist of the stator and the rotor. The stator is constructed from iron laminations with slots which carry the three phase windings. The rotor has slots which carry copper or aluminium conductor bars. These are connected at the ends by short circuiting rings forming a cage. This leads to the commonly used term “squirrel cage motor”. The ends of the stator windings can be connected in star connection (Y) or delta connection (Δ) or be connected to a star-delta starter. See section 9.4.1.

If the stator windings are connected to a mains supply with fixed voltage and frequency, then a magnetic field is set up which rotates relative to the fixed stator at the synchronous speed

$$n_{\text{syn}} = \frac{f}{p} \cdot 60 \quad \text{in rpm}$$

where f = Supply frequency in Hz

p = Number of poles in stator winding

This rotating field induces a voltage in the conductors, in the rotor winding and a current, whose magnitude depends on the resistance in the circuit. The rotating field and the rotor current are the prerequisites for generating a torque. This means that the rotor cannot match the speed of rotation of the field, as the generation of rotor current is dependent on the conductors cutting the lines of force of the field. The rotor therefore runs more slowly (asynchronously) than the synchronous speed.

This differential, the slip s is expressed as a % of the synchronous speed.

$$s = \frac{n_{syn} - n}{n_{syn}} \cdot 100 \quad \text{in \%}$$

The motor speed (asynchronous speed) is therefore given by:

$$n = n_{syn}\,(1 - \frac{s}{100}) \qquad \text{in rpm}$$

Table 9.04 Synchronous speed n_{syn} in rpm at f = 50 and 60 Hz for different numbers of poles

No of poles	2	4	6	8	10	12	14	16	18
50 Hz	3000	1500	1000	750	600	500	428	375	333
60 Hz	3600	1800	1200	900	720	600	514	450	400

With increasing load on the motor, a higher rotor current is required, the slip increases and the speed reduces. The slip required at rated power is dependent on the size of motor and reduces as the motor size increases. At no-load the motor only has to overcome small internal losses, for which a low torque is adequate. Under light load, the motor speed is therefore closer to the synchronous speed.

Table 9.05 Slip s_N for various rated powers (guideline values)

Rated power kW	1	10	100	1000
Slip s_N %	6 - 9	3 - 4	0.7 - 1.6	0.5 - 0.8

The lower value is for 2 pole motors and the higher value for 8 pole.

The exact rated speed and rated power are obtained from the manufacturer's data.

To accelerate asynchronous squirrel cage induction motors from rest n = 0 up to the rated speed n_N with a given load, the magnitude of the starting torque and the shape of the torque speed / curve are determining factors. The shape of the curve is mainly determined by the design of the rotors, principally the rotor conductor bars. The large number of different constructions and designations can basically be reduced to three types of rotor: round bar, deep bar and double bar rotors.

Dependent on the number of poles and the frame size, three phase motors are available with different rotor classes and corresponding torque curves. Since the driving torque of centrifugal pumps increases with the square of the speed, the torque (rotor) class of the motor generally has little consequence for starting the pump. There is always sufficient accelerating torque M_{bmi} available.

M_A	Starting torque
M_K	Pull-out torque
M_L	Load torque
M_m	Motor torque
M_N	Rated torque
M_S	Pull-up torque
n_N	Rated speed
n_{syn}	Synchronous speed

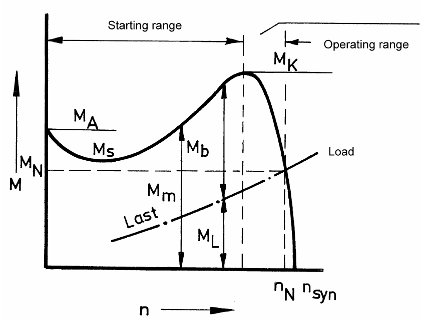


Fig. 9.02
Characteristic torque speed / curve of asynchronous squirrel cage induction motors

9.2.3.2 Special construction three phase asynchronous squirrel cage induction motors

For a number of applications, special constructions in which the motor and pump form a compact monoblock unit have been developed.
These include:

- Submersible motors
- Canned motors
- Underwater motors

The submersible pump utilises the liquid in which the pump unit is immersed as a cooling medium. The motors are sealed and filled with air or oil. They are constructed with sufficient safety that they can still operate when they are not totally immersed in the cooling medium. They are used to drive vertical submersible pumps e.g. waste water and sewage pumps and cooling pumps.

Canned motor pumps have a wetted rotor and a dry stator. The corrosion resistant thin walled can, seals off the stator windings from the pumped liquid. The canned motor and pump form a leak free sealless pump unit. Canned motor pumps are used as chemical pumps and heating circulation pumps. See section 4.11.2.

The underwater pump has wetted rotor and stator. The motor is completely filled with liquid, either water or oil. They are used to drive pumps in narrow bore holes and wells, (4, 6, 8 inch bore).

9.2.3.3 Three phase asynchronous slip ring motors

These motors are selected instead of squirrel cage motors when

- a high starting torque is required
- a low starting current is required

As these attributes are not relevant to centrifugal pumps, the use of slip ring motors is not necessary.

Slip ring motors are used in some cases when a centrifugal pump requires speed regulation combined with high power output. The use of a series of rectifiers with the slip ring motor can provide an economic solution. See also section 9.6.1.3.

The slip ring rotor has a three phase winding, with the same number of pairs of poles and similar construction to the stator. The windings are connected through slip rings with variable resistance. The starting and run up torque can be adjusted by varying the total resistance of the rotor circuit. Because of the slip ring construction, this asynchronous motor loses its simplicity and reliability. It is more expensive than a squirrel cage motor and the brushes and slip ring require maintenance. If the motor is accessible, the wear which takes place on the slip rings and brushes can be avoided with a short circuit / brush lifting device. The motors can also be fitted with wear monitoring and micro-switching equipment.

9.2.3.4 Three phase synchronous motors

The use of synchronous motors for centrifugal pump drives is a question of capital cost, the supply (power factor correction) and the type of application for the pump. The investment required for the starting, synchronisation, excitation and control is only to a limited extent dependent on motor size and becomes less significant as the size of motor increases. Synchronous motors cost between 10 to 40% more than the equivalent asynchronous motor. The advantages of a synchronous motor in compensation, include the possibility for power factor correction and higher efficiency, so that an economic benefit can be realised for higher power centrifugal pumps above 8 to 10 MW.

Synchronous generators can also be used as synchronous motors. Such machines are found in pumped storage power generation plants as pump drives and generators.

9.3 Construction of electric motors

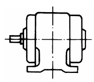
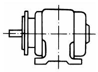
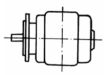
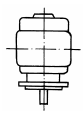
9.3.1 Type of construction

The types of construction and mounting arrangements of electric motors are standardised in Europe by IEC 34-7, (BS4999 pt 107).
The designation follows two code systems IM (International Mounting) which are:

- Code I — an alpha numeric system for motors with bearing end shields and one shaft end
- Code II — a numerical code with wide ranging application including code I

The most commonly used types of construction for centrifugal pump drives are listed in the following table.

Table 9.06 Types of construction of electrical machines

Code I	Code II	Fig.	Construction
Machines with horizontal shaft			
IM B3	IM 1001		2 bearing end shields, housing with foot, foot mounted.
IM B35	IM 2001		2 bearing end shields, housing with foot, flanged bearing shield drive side, foot mounted, additional flange mounting.
IM B5	IM 3001		2 bearing end shields, housing without foot, flanged bearing shield drive side, flange mount. It is recommended that this type of construction is used up to frame size 200 (37kW, 2-pole; 30kW, 4-pole). For higher powers construction IM B35 can be used.
Machines with vertical shaft			
IM V1	IM 3011		2 bearing end shields, flanged bearing shield drive side, flange mount drive side down. With this construction a protective cover is recommended. For explosion protection motors this is a regulation.

9.3.2 Protection class of electrical equipment

9.3.2.1 IP code

The IP code (International Protection) describes the protection measures to prevent contact by persons and protection against ingress of solids and liquids, according to IEC 34-5, EN 60529 (BS 4999 Pt 105).

The protection covers the following:

- Protection afforded against contact by persons with live and moving parts inside the housing and prevention of entry of solid bodies.

- Protection of the equipment from entry of water.

The class of protection is designated by an alpha numeric code of two letters IP and two digits.

Table 9.07 First digit defines protection against contact and foreign bodies

First digit	Protection against contact and foreign bodies
0	No special protection
1	Protection against entry of foreign bodies larger than 50 mm, but no protection against intentional entry
2	Protection against entry of solid foreign bodies of greater diameter than 12mm. Protection against contact by fingers with live or moving parts.
3	Protection against entry of solid foreign bodies of greater diameter than 2.5mm. Protection against contact by tools or wires.
4	Protection against entry of solid foreign bodies of greater diameter than 1mm. Protection against contact by tools or wires.
5	Protection against harmful build up of dust. The penetration of dust is not prevented, but does not accumulate in sufficient amount to impair operation of the machine. Complete protection against contact with live or moving parts.
6	Dust tight protection. Complete protection against contact.

Table 9.08 Second digit defines protection against water.

Second digit	Protection against water
0	No special protection
1	Protection against water droplets falling vertically, which must have no harmful effect.
2	Protection against water droplets falling vertically, which must have no harmful effect. The motor housing may by tipped up to 15° from its normal position without harmful effect (water dropping at an angle).
3	Protection against water falling at any angle up to 60°, which must have no harmful effect. Sprayed water.
4	Protection against water sprayed at any angle, which must have no harmful effect. Splashed / sprayed water.
5	Protection against water sprayed from a nozzle at any angle, which must have no harmful effect. Hosed water.
6	Protection against water flooded over the machine, e.g due to heavy seas, which must not enter the housing in sufficient quantity to have harmful effect.
7	Protection against water when the machine is immersed at a specified pressure and time. Water must not enter the housing in sufficient quantity to have harmful effect.
8	The housing is designed for permanent immersion in water under conditions defined by the manufacturer.

The standard protection class for three phase motors is IP55.

9.3.2.2 IK code

The IK code describes the degree of protection of the housing against external mechanical impacts, according to EN 50102.
The class of protection is designated by an alpha numeric code of two letters IK and two digits.

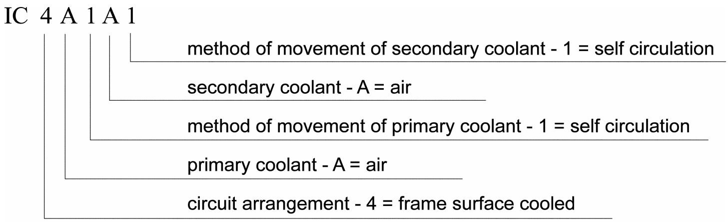
Table 9.09 Degree of protection against mechanical impact.

IK-Code	Impact energy
01	0.15 J
02	0.20 J
03	0.37 J
04	0.50 J
05	0.70 J
06	1 J
07	2 J
08	5 J
09	10 J
10	20 J

The standard protection class for three phase motors is IK 08.

9.3.3 Method of cooling

The method of cooling is defined in accordance with IEC 34-6, EN 60034-6, (BS4999 pt 106). The class of cooling is designated by an alpha numeric code of two letters IC and further digits and letters.
Example:



Simplified form:
If there is no danger of ambiguity, the symbol A for air can be dropped, so the shortened form would be:

IC411.

The standard method of cooling three phase induction motors with protection class IP55 is system IC411. This is a surface cooled motor with air cooled, smooth or ribbed housing and a shaft mounted fan.

9.3.4 Insulation

The insulation of the windings is classified in temperature classes in accordance with IEC 34-1

Table 9.10 Temperature rise (ΔT) and maximum temperature at the hottest point in the winding (T_{max}) in accordance with IEC 34-1

Insulation	ΔT measured by resistance method	T_{max} with a cooling media temperature 40 °C
Class B	80 K	125 °C
Class F	105 K	155 °C
Class H	125 K	180 °C

As a rule three phase motors are fitted with class F insulation.

9.4 Installation and operation of electric motors

9.4.1 Ratings

9.4.1.1 Power

Motors are selected from tables indicating the measured power output.
This is the available power output at the motor shaft and must equate to the power required by the pump. See also section 1.7.4.

9.4.1.2 Power absorbed, efficiency and power factor

Three phase asynchronous motors

The power absorbed by an electric motor is the product of the electric current drawn from the line and the voltage applied to it. Due to the losses, it is higher than the output power. A motor consumes a combination of active and reactive power. This produces a phase displacement between the voltage and the current, in which the current lags behind the voltage by an angle φ. For the calculation of the power consumption, therefore only the active current, which is in phase with the voltage, is taken into account. The active power is given by:

$$P_w = \frac{I \cdot \cos\varphi \cdot U \cdot \sqrt{3}}{1000} \quad \text{in kW}$$

The current drawn can be calculated from the active power

$$I = \frac{P_w \cdot 1000}{U \cdot \cos\varphi \cdot \sqrt{3}} \quad \text{in Ampere (A)}$$

The motor efficiency η is the ratio of the output power P_M (mechanical power at the shaft) to the absorbed power P_w (active power).

$$\eta = \frac{P_M}{P_w} \cdot 100 \qquad \text{in \%}$$

The output power at the shaft is therefore.

$$P_M = \frac{I \cdot \cos\varphi \cdot U \cdot \sqrt{3} \cdot \eta}{1000 \cdot 100} \quad \text{in kW}$$

For the current drawn the following applies.

$$I = \frac{P_M \cdot 1000 \cdot 100}{\cos\varphi \cdot \eta \cdot U \cdot \sqrt{3}} \quad \text{in Ampere (A)}$$

The product of the line voltage and measured current gives the apparent power P_s, a purely mathematical figure, as U and I occur at different instances.

$$P_s = \frac{I \cdot U \cdot \sqrt{3}}{1000} \quad \text{in Kilovoltampere (kVA)}$$

For the establishment of the magnetic field, i.e. magnetisation of the motor, power is required which is not converted into mechanical energy. There is only a continual interchange between the field winding and the mains supply. This so called reactive power P_b, is calculated by subtraction of the active and apparent power.

$$P_b = P_s^2 - P_M^2 = \frac{I \cdot U \cdot \sin\varphi \cdot \sqrt{3}}{1000} \quad \text{in Kilovoltampere reactive (kvar)}$$

The factor " cos φ " which appears in the calculation of active power is known as the power factor. It represents the ratio of the active power to the apparent power.

$$\cos\varphi = \frac{P_w}{P_s}$$

The power factor cos φ is therefore a measure of the part of the apparent power which is converted into a different form of energy and is therefore a consumption factor.

In general it can be said that efficiency and power factor

- increase with increasing motor power
- decrease with decreasing motor power

This must be taken into account when selecting the motor. If too high a safety factor is built in to the power requirement of the pump, then the motor is run continuously under part load, the consequence is it runs with low efficiency and power factor.

The values in the following table for efficiency and power factor are guidance values only and may vary from one manufacturer to another.
The output power of a motor can be estimated using these values based on the current drawn and the voltage. Measurement of the current drawn during operation in order to monitor the pump unit, therefore merely serves to check whether a motor is operating within its design range. This can be important if there is a danger of the absorbed power of the pump exceeding the design figures due to unexpected factors, such as changes in operating duty or wear in the pump. This allows measures to be taken to prevent the permitted power of the motor being exceeded before the motor protection system trips and operational disruption occurs.

Table 9.11 Power factor cos φ for various rated powers (guide values)

Rated power kW	1	10	100	1000
Power factor cos φ	0.81 - 0.84	0.84 - 0.85	0.86 - 0.88	0.89 - 0.93

The lower value is for 4-pole and the higher value for 2-pole electric motors.

Table 9.12 Variation in power factor cos φ under part load

1/2 full load	3/4	**4/4**	5/4	1/2 full load	3/4	**4/4**	5/4
0.86	0.90	**0.92**	0.92	0.69	0.79	**0.83**	0.84
0.85	0.89	**0.91**	0.91	0.67	0.77	**0.82**	0.83
0.83	0.88	**0.90**	0.90	0.66	0.76	**0.81**	0.82
0.80	0.86	**0.89**	0.89	0.65	0.75	**0.80**	0.81
0.78	0.85	**0.88**	0.88	0.63	0.74	**0.79**	0.80
0.76	0.84	**0.87**	0.87	0.61	0.72	**0.78**	0.80
0.75	0.83	**0.86**	0.86	0.59	0.71	**0.77**	0.79
0.73	0.81	**0.85**	0.86	0.58	0.70	**0.76**	0.78
0.71	0.80	**0.84**	0.85	0.56	0.69	**0.75**	0.78

The values for 4/4 full load = rated power as published by the manufacturer. The part load figures are averages.

According to EN 60034 the following tolerances apply: $\frac{1 - \cos \varphi}{6}$

within limits: min 0.02 max 0.07

Table 9.13 Efficiency η_N in % for various rated powers (guide values)

Rated power kW 2-pole and 4-pole	1.1	11	110	1000
Efficiency η_N %	eff 2 77 eff 1 84	eff 2 89 eff 1 91	eff 3 95	eff 3 97

eff = Standard eff. eff 2 = Improved eff. eff 1 = High eff

Table 9.14 Variation in efficiency η_N in % under part load

1/2 full load	3/4	**4/4**	5/4	1/2 full load	3/4	**4/4**	5/4
96	97	**97**	96.5	81	82	**82**	80.5
95	96	**96**	95.5	80	81	**81**	79.5
93.5	95	**95**	94.5	79	80	**80**	78.5
92.5	94	**94**	93.5	77	79.5	**79**	77.5
91.5	93	**93**	92.5	75.5	78.5	**78**	76.5
91	92	**92**	91.5	74	77.5	**77**	75
90	91	**91**	90	73	76	**76**	74
89	90	**90**	89	72	75	**75**	73
88	89	**89**	88	71	74	**74**	72
87	88	**88**	87	70	73	**73**	71
86	87	**87**	86	68	72	**72**	70
85	86	**86**	85	67	71	**71**	69
84	85	**85**	83.5	66	70	**70**	68
83	84	**84**	82.5	65	69	**69**	67
82	83	**83**	81.5	64	67.5	**68**	66

The values for 4/4 full load = rated power as published by the manufacturer. The part load figures are averages.

According to EN 60034 the following tolerances apply

for $P_N \leqq 50$ kW $\quad -0.15\,(1-\eta)$

for $P_N > 50$ kW $\quad -0.1\,(1-\eta)$

Where η is expressed as a decimal.

9.4.2 Installation requirements

The rated powers of motors are based on defined installation conditions.
They are valid for an ambient temperature up to 40°C and an altitude up to 1000m above sea level.

For other installation conditions the power output must be de-rated according to the following table.

Table 9.15 Correction factors for height above sea level (AH) and ambient temperature (KT)

Height above sea level (AH)	Ambient temperature (KT) in °C					
	< 30	30-40	45	50	55	60
1000	1.07	1.00	0.96	0.92	0.87	0.82
1500	1.04	0.97	0.93	0.89	0.84	0.79
2000	1.00	0.94	0.90	0.86	0.82	0.77
2500	0.96	0.90	0.86	0.83	0.78	0.74
3000	0.92	0.86	0.82	0.79	0.75	0.70
3500	0.88	0.82	0.79	0.75	0.71	0.67
4000	0.82	0.77	0.74	0.71	0.67	0.63

For extreme climatic conditions, when e.g. the ambient temperature is < – 40°C or relative humidity is > 95%, then anti-condensation heaters are recommended. This ensures an average motor temperature is maintained to prevent starting problems arising. Furthermore loss of winding insulation integrity, due to condensation is prevented.

For conditions where relative humidity of 90 to 100 % can be expected over long periods, then special tropicalised insulation is essential.

9.4.3 Effect of changes in supply voltage and frequency on the operation of three phase asynchronous induction motors

9.4.3.1 Changes in supply voltage with constant frequency

Starting torque and pull out torque, vary as the square of the voltage and the starting current is approximately proportional.
According to EN 60 034-1 a voltage tolerance of ± 5% is allowed (range A). IEC 38 allows a tolerance of ± 10% for supply voltages 230, 400 and 690V.
An additional tolerance of ± 5% can be used according to EN 60 034 if the permitted temperature rise, according to the class is allowed to be exceeded by 10K.
Changes in the rated values are given in table 9.16.

9.4.3.2 Changes in supply frequency with constant voltage

The absolute values of the starting torque and pull out torque vary as the inverse of the square of the frequency and the rated speed is approximately proportional to the frequency. In general frequency variations of ± 5% are permissible. Changes in the rated values are given in the following table.

Table 9.16 Effect of changes in voltage and frequency on rated values

Rated value	Voltage 110% of rated value	Voltage 90% of rated value	Frequency 105% of rated value	Frequency 95% of rated value
Starting and pull out torque	Increase 21%	Decrease 19%	Decrease 10%	Increase 11%
Synchronous speed	Unchanged	Unchanged	Increase 5%	Decrease 5%
Full load speed	Increase 1%	Decrease 1.5%	Increase 5%	Decrease 5%
Full load efficiency	Increase 0.5 to 1 point	Decrease 2 points	Slight increase	Slight decrease
Power factor at full load	Decrease 3 points	Increase 1 point	Slight increase	Slight decrease
Starting current	Increase 10 to12%	Decrease 10 to 12%	Decrease 5 to 6%	Increase 5 to 6%
Full load current	Decrease 7%	Increase 11%	Slight decrease	Slight increase
Temperature	Decrease 3 - 4 K	Increase 6 - 7 K	Slight decrease	Slight increase

All values are for guidance only. Exact values obtainable from manufacturer.

The use of motors for larger variations in frequency is limited. In the case of a 50Hz motor connected to a 60Hz supply with unchanged voltage, the starting and pull out torque are reduced to approximately 70%. If a 60Hz motor is used on a 50Hz supply, the power output must be considerably reduced to re-establish the original magnetic conditions. If not, the increase in absorbed current will cause damage to the winding insulation due to overheating.

9.4.3.3 Simultaneous changes in supply voltage and frequency

If the voltage and frequency vary simultaneously and in the same ratio and sense, e.g. from 400V 50Hz to 460V 60Hz, then the magnetic conditions are unaltered, if the effect of resistance is neglected. The motor will produce the normal torque with approximately the same stator and rotor currents.

The rated speed and power vary in proportion to the frequency. This is valid for approximately ± 20% of the rated frequency. However in such cases the manufacturer should be consulted, as the increased output may be limited by the additional heat generation.

9.4.4 Torque class (rotor class) of three phase asynchronous squirrel cage induction motors

Three phase asynchronous squirrel cage induction motors have different torque characteristics depending on the form and cross section of the rotor cage bars, as described in section 9.2.3.1. the cross section form on its own does not define the absolute starting torque of the motor. The manufacturer's data will therefore define the rotor class from which the maximum torque against which the motor can be safely started, can be determined. The following figure illustrates the range of characteristics for a motor of given rotor class, which can be reliably started direct-on-line against a load of up to 90% of the rated torque (broken line).

As the load torque of centrifugal pumps increases as the square of the speed ($M \sim n^2$) and the break-out torque is normally only 5 to 10% of the rated torque, then the rotor class is generally not of importance for centrifugal pump drives.

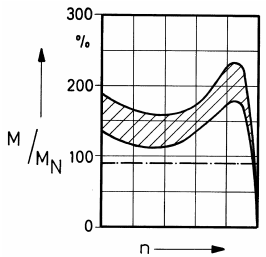


Fig. 9.03 Example of torque characteristic.

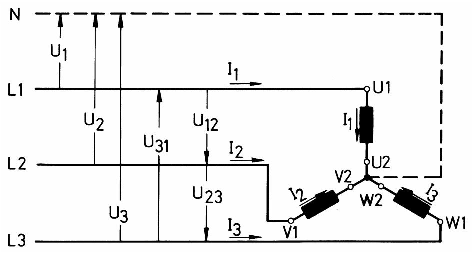
9.4.5 Connection of three phase asynchronous squirrel cage induction motors to mains supply

Three phase motors are connected to the three outer conductors, L1, L2 & L3. Generally the windings may be connected (linked) by one of two methods: Star or Delta.

9.4.5.1 Star connection Y

The ends of the windings U2, V2 & W2 are connected together. The phase voltages in the windings are equal to the star voltages (phase voltages) of the supply and the currents in the windings are equal to the currents in the supply.
If the three phases are equally loaded (symmetrically loaded) then the sum of the currents in the windings at any moment is zero.

Star connection Y



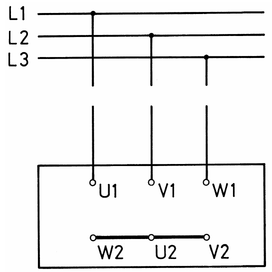


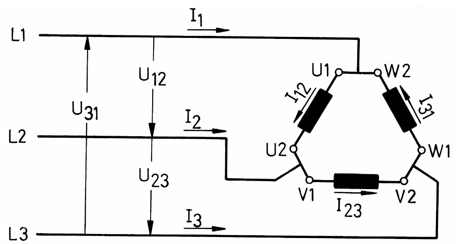
Fig. 9.04 Wiring principle

Fig.9.05 Connection diagram

9.4.5.2 Delta connection Δ

The end of one phase winding is connected to the beginning of the next. The phase voltages in the windings are equal to the line voltage of the supply and the winding currents are linked and together compose the line current.
If the three phases are equally loaded (symmetrically loaded) then the sum of the linked currents in the windings at any moment is zero.

Delta connection Δ



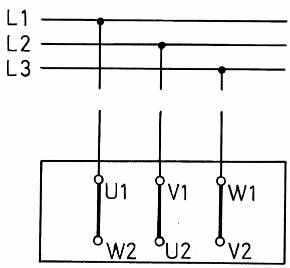


Fig. 9.06 Wiring principle

Fig.9.07 Connection diagram

The method of starting the motor influences the method of connection when the motor is running. The following table shows the possible running connections for squirrel cage motors depending on the winding and supply voltage.
For star delta starting, the running connection must be delta.

Table 9.17 Running connection for three phase squirrel cage motors

Winding voltage	Operating voltage at 50 Hz	For direct starting or slip ring rotor	For star delta starting
230 Δ / 400 Y	230 400	230 Δ 400 Y	230 Δ —
400 Y 400 Δ	400	400 Y 400 Δ	— 400 Δ
400 Δ / 690 Y	400 690	400 Δ 690 Y	400 Δ —

The direction of rotation of the motor is the same as that of the magnetic field. If it is required to reverse the direction of rotation of the magnetic field and hence the rotation of the motor, it is sufficient to interchange the connections of two phases at the terminal box of the motor.

9.4.6 Starting three phase asynchronous squirrel cage induction motors

9.4.6.1 Direct on line starting

Direct on line starting is the ideal, as it is the simplest. The motor is connected directly to the full supply voltage through a contactor. At the moment of connection a starting current of 4 to 7 times the rated current flows.

As a result the voltage in the mains conductors drops and other users may be affected. For this reason the electrical supply authorities set limits to the size of individual motors (ca. 4kW) which can be started direct on line. However even if low voltage motors are connected to independent factory supply systems, if these have a high voltage connection from the mains then this must be considered at the planning stage.

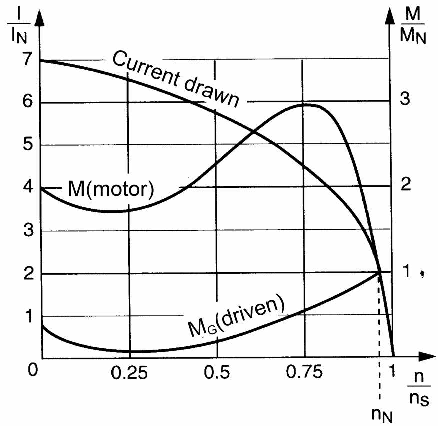


Fig. 9.08 Typical characteristic of direct on line starting

9.4.6.2 Star-Delta (YΔ) starting

If it is necessary to restrict the starting current due to supply limitations, the YΔ starting method is a possibility, but even with this in some circumstances a power limit may be set. In this case high power motors may be started with series resistors or auto-transformers to limit the current, although this does represent additional investment, so is not often used.

For YΔ starting, YΔ contactors or a corresponding combination of simple contactors are used, to which all six ends of the windings are connected. The start takes place in the star connection. With this the phase voltage is only 1/√3 of the line voltage in contrast to the delta connection (running connection) in which the full line voltage is applied to the phase. As the torque is approximately proportional to the square of the voltage, it is reduced to roughly 1/3 of the value in the running connection. The current is also reduced correspondingly to 1/3 of the direct on line value. As the motor runs up to speed it is switched to the delta connection and a current surge occurs depending on load, which may exceed the current limitation.

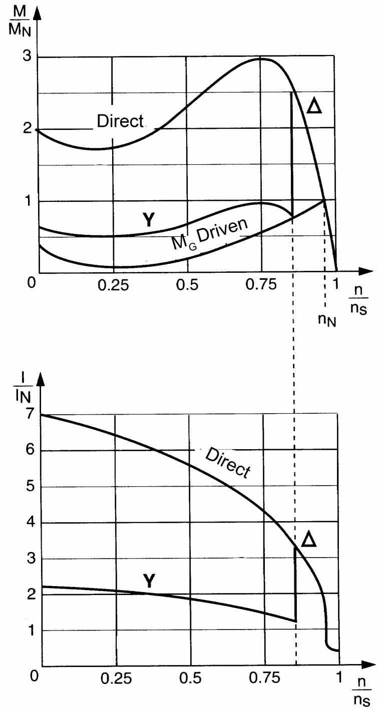


Fig. 9.09 Example of characteristics for YΔ starting

9.4.6.3 Soft starting

Soft starting is achieved using electronic control (semiconductors).
This method allows the magnitude of the starting current to be selected according to the starting requirement. The run up time can be programmed and/or a limit set on the current value.

After the run up is completed and the full voltage is applied, the soft start can be disconnected by a contactor to reduce losses.
The soft start can also be used as a run down control. The run down of the pump can be controlled to meet the requirements and e.g. prevent water hammer in the pipework.

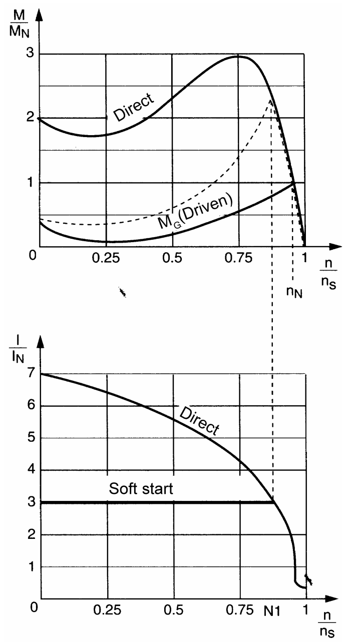


Fig. 9.10 Example of characteristic for soft starting

9.4.6.4 Starting high voltage motors

Providing the electrical supply authorities permit, the most economic method is to use squirrel cage motors with direct on line starting. For high power motors which exceed the limits, it should be examined if start up with a high voltage frequency inverter is possible. By controlling the speed, it is also possible to limit the start up current to the rated value.

9.4.7 Types of duty

The type of duty for a motor affects its thermal behaviour and hence its load capacity, thus influencing the selection and design of a suitable machine. The type of duty should be described as accurately as possible at the time of purchase. To simplify this and ensure better understanding between supplier and user, the many types of duty are defined into 10 categories (S1 to S10) according to EN 60034-1.

For centrifugal pumps normally only the category “continuous duty” (S1) is considered. The rated performance given in manufacturer’s data sheets are for continuous duty.
Continuous duty is defined as a sufficient period of operation under constant load that a thermal equilibrium is reached. For this type of duty, the motor selection (rating) can be made according to the power requirement of the pump, taking into account any safety factors required by the standards or by experience.
It is indicated on the rating plate by the term “continuous running duty” or the abbreviation S1. If there is no marking, then continuous duty running can be assumed.

9.4.8 Motor protection against over-current and thermal overload

Motor protection is usually given by thermal delay overload protection, built into the motor starter.
This is a current dependent device and protects against overheating. Overheating can occur as a result of overload, un-symmetrical current take up, loss of phase, excess starting frequency or jammed rotor.
Apart from this, the motor can be protected by thermistors built into the windings and connected to a trip device.
This type of device is temperature dependent and protects the motor against excessive heating of the windings, due to e.g. strongly varying load or rapid switching on and off.
For pole-change motors with two separate windings, double the number of thermistors are required.
Fuses and trips are not motor protection devices but protect the supply equipment from high short circuit currents.

9.5 Explosion protection

9.5.1 General

In the process industries, particularly chemical industry and refineries, gases, vapours, mists and dust clouds can be present, which when mixed with the oxygen in the atmosphere may form an explosive mixture. Under certain conditions this mixture may detonate or explode.

The composition and concentration of the mixture (ease of ignition) and the energy of the ignition source (e.g. electrical spark or overheating of a motor) have a major influence on this. The results of such an explosion, are frequently substantial material damage and alongside that possible loss of life. For this reason explosion protection measures are required by regulation.

Explosion/hazardous areas are defined as those in the normal course of local and operational conditions, where an explosive mixture of gases, vapours, mist or dust can accumulate in hazardous quantities. This can apply, equally to open air sites as to enclosed spaces.

Explosion protection, embraces a very wide range of applications and the regulations covering the protection of electrical equipment are correspondingly extensive.
The following regulations form a basis only, as in individual cases all the relevant regulations must be complied with.

The European Standards EN 50 014 to EN 50 020, EN 50 028 and EN 50 029 define the construction and test requirements for different types of protection.

EN 50 014	The general requirements for the construction and testing of electrical apparatus for use in a hazardous area.

Areas endangered by explosives are not classified as hazardous areas, but if a hazardous area also contains explosives, then the above regulations apply.

DIN VDE 0166	The requirements for the construction and testing of electrical apparatus in areas containing explosives.

For methane danger in mines other regulations apply in conjunction with the mines authority.

DIN VDE 0118	The requirements for the installation of electrical equipment in underground mines.

According to the “Statutory Regulations Covering the Installation of Electrical Equipment in Locations Subject to Explosion Hazard (Elex V)” the equipment may only be used in such areas if the following conditions are satisfied:

- They must be approved for the gases and vapours which are present.
- They must be subject to a detailed inspection by the manufacturer ensuring that they are in accordance with the type approval.
- They must bear the appropriate mark and data as laid down by the regulatory body

Type approval is given by the relevant local authority under the jurisdiction of the principal regulatory authority. All parties concerned have a duty to observe the regulations laid down in these approval documents.

In order to classify the explosion protection requirements of a particular pump (e.g. a canned motor pump) it is not sufficient to only consider the pumping data and properties of the pumped media. Other sources of hazard may exist in the same area which must be taken into account, e.g. if a pump is handling acetone (temperature class T1) in an area where ethyl ether (temperature class T4) is present.

The decision as to whether a particular location, either in the open or in an enclosed space, is to be considered as hazardous within the meaning of the regulations has to be made by the user, or in the case of doubt, by the appropriate regulating authority, (e.g. factory inspectorate). The authority will determine the necessary protective measures to be taken to avert danger of explosion.

9.5.2 Designation of hazardous area zones

Hazardous areas are classified into zones according to the likelihood, duration and frequency of an explosive atmosphere occurring.

Zone 0 comprises areas in which an explosive atmosphere is present continuously or for long periods, (e.g. inside vessels containing inflammable liquids or gases).

In Zone 0, only equipment especially designed for this may be used. Electric motors regardless of protection class are not permitted.

Zone 1 comprises areas where an explosive atmosphere can be expected to occur occasionally.

Electric motors used in this zone must be explosion protected to either explosion proof “d” or increased safety “e” standard.

Zone 2 comprises areas in which explosive atmospheres are expected to occur only occasionally and then for a short period.

Motors with explosion proof “d” protection and increased safety “e” may be used. In many cases standard three phase squirrel cage induction motors may also be used.

9.5.3 Gas groups

Flammable gases and liquids are classified into groups by the minimum gap through which an explosion may be propagated under defined experimental conditions and/or the minimum ignition current.

Table 9.18 Example of the explosion group classification of gases and vapours.

Group	Flammable gas or vapour
II A	Acetone, ammonia, benzene, benzole, butane, diesel fuel, kerosene, acetic acid, fuel oil, hexane, methanol, propane, toluene
II B	Ethanol, ethylene oxide, ethyl ether, towns gas
II B + H_2	as II B + hydrogen
II B + CS_2	as II B + carbon disulphide
II B + $C_2 H_2$	as II B + hydrogen sulphide
II C	Acetylene

9.5.4 Temperature class

An explosive atmosphere can be ignited simply by the normal heat generation of electrical equipment. To prevent this the maximum surface temperature of the machine must be kept below the ignition temperature of the flammable atmosphere. The ignition temperature of a flammable atmosphere is defined as the lowest temperature of a surface, which results in combustion of the mixture in contact with it.
The ignition temperature of liquids and gases is determined according to EN 50 014 / DIN51 794. They are categorised in temperature classes T1 to T6.

The maximum surface temperature is the highest temperature which may be reached in operation, under the most unfavourable conditions by any parts or surfaces of the equipment, which is able to cause ignition in contact by an explosive atmosphere. The most unfavourable conditions also include known overload and fault conditions. For pumpsets, not only the surface of the electrical driver but also that of the pump itself must be considered.

The safety criteria relating to flammable gases and vapours e.g. combustion point, ignition temperature, temperature class, explosion class are all defined and described in the regulations and publications of the regulatory authority of the country.

Table 9.19 Temperature classification

Temperature class	Maximum permissible surface temperature °C	Ignition temperature of the flammable material °C
T1	450	> 450
T2	300	> 300 < 450
T3	200	> 200 < 300
T4	135	> 135 < 200
T5	100	> 100 < 135
T6	85	> 85 < 100

9.5.5 Comparison of Standards

The European Standards EN 50 014 to 50 020 which came into force in May 1978, are compared with the previous German standard DIN VDE 0170 / 0171 which were still used by manufactures up to 1.5.1988 and which are still valid for operation.

Table 9.20 Explosion Protection Designations to EN 50 014 / 50 020:

EN 50 014 / 50 020			DIN VDE 0170/0171	
Explosion protection		Group II	Explosion protection	(Ex)
Type of protection	Explosion proof Increased safety	d e	Type of protection	d e
Temperature class		T1 T2 T3 T4 T5 T6	Ignition group	G1 G2 G3 G4 G5 --
Explosion group		IIA IIB IIB + H_2 IIB+ CS_2 IIB+C_2H_2 IIC	Explosion class	1 2 3a 3b 3c 3n
Example of designation	EEx e II T3 EEx de IIC T4		(Ex) e G3 (Ex) d 3n G4	

Example: EEx de IIC T4

E: Motor to European standard

Ex: Explosion protection

de: Combination of explosion proof and increased safety

II: Group of electrical apparatus for places with potentially explosive atmospheres, other than mines, susceptible to firedamp.

C: The highest class of the maximum experimental safe gap for type of protection Ex d. This class is suitable for all gases and vapours.

T4: Temperature class 4 (max. surface temperature 135 °C)

Certification is provided by the appropriate regulatory body for each country.

9.5.6 Explosion protection in canned motor pumps

For canned motor pumps, the authorities require additional safety measures to be observed as well as the normal explosion proof motor regulations.
In the certification under the section "Special Conditions", the additional conditions for the operation of such pumps in hazardous areas are set out.

The following points are covered:

- For reasons of safety, the rotor casing must be always filled with the pumped liquid. Therefore it is necessary to fit the unit with a liquid level indicator, or an equally reliable alternative method to ensure that the motor can only run with an adequate level of liquid.

- To prevent unacceptable temperatures being reached in the cooling / lubricating stream, temperature probes must be fitted. These must ensure that the maximum permissible temperature allowable by the explosion protection requirements is not exceeded due to the operating conditions of the pump.

9.5.7 Explosion protection to European standards

The directive 94/9/EC published by the European Community covered the harmonisation of the regulations of the member states for equipment and protection systems for operation in explosion hazard areas and in doing so changed the basis for explosion protection.

This directive also known as ATEX 100a came into effect on the 23rd May 1999 with a transition period until 30th June 2003.

This is a further step toward uniform safety standards in the European Community.
The most obvious sign of this standard is the CE mark which explosion protected equipment will also have to carry and which is a condition for the free movement of goods within the European Community.

The term explosion protection has been greatly extended in its importance and in future non-electrical ignition risks will also be considered. The manufacturer will have to ensure the suitability of the entire unit for safe installation in an explosion hazardous area.

The effect of the 94/9/EC directive on pumps and manufacturers is described in a document produced by EUROPUMP, (the Association of European Pump Manufacturers).

9.6 Speed control of electrical drives

9.6.1 Controlling the motor

The following motors can be considered for speed control:

- Three phase asynchronous squirrel cage induction motors
- Three phase asynchronous slip ring induction motors
- Three phase synchronous motors
  - → Reluctance motors
  - → Permanent magnet motors

DC motors have very little use in driving centrifugal pumps.

The rotational speed of a three phase induction motor is given by:

Rotor speed $$n = \frac{f}{p \cdot (1 - s)}$$

with f = supply frequency p = number of poles s = slip

Therefore there are three ways of controlling the speed of a three phase motor:

- Changing the number of poles
- Changing the frequency
- Changing the slip

9.6.1.1 Pole change motors

Pole-change motors, enable a stepwise alteration of speed at constant supply frequency. Pole changing is generally confined to three phase squirrel cage induction motors.

The simplest method of switching, the "Dahlander switching" as here the winding of one phase, distributed over several slots remains together; however it only permits a speed change ratio of 1:2.

With two windings, three or four speeds can be obtained.

Table 9.21 The most common pole change motor configurations

Number of windings	Type of winding	Number of poles	Synchronous speed at 50Hz in rpm
1	Dahlander switching	4 / 2 8 / 4	1500 / 3000 750 / 1500
2	Separate windings	6 / 4	1000 / 1500
3	Separate windings but with 750/1500 rpm in Dahlander switching	8 / 6 / 4	750 / 1000 / 1500

As a rule, the windings are designed so that the torque remains practically constant at all speeds. For pump drives, however it is possible to have motors with windings which match the torque requirements of the pump, i.e. rising as the square of the speed. These motors are often referred to as ventilator drives.

9.6.1.2 Frequency change

Frequency control is achieved using a frequency converter. These take two forms:

- voltage source converter
- current source converter

With the voltage source converter, the speed is changed by alterations to the frequency f by application of the appropriate voltage pulse U. The supply is created with fixed frequency and voltage (e.g. 3 AC 50/60Hz, 380 to 690 V) in a three phase system with variable frequency and voltage (3 AC 0 to 200Hz, from 0 V to rated supply voltage).
With the current source converter, the speed is changed by alterations to the frequency f by application of appropriate load dependent, current pulses. They are generated from a three phase voltage with fixed frequency and amplitude (e.g. 3 AC 50/60Hz) a three phase system with variable frequency (3 AC 0 to 50/60Hz) and load dependent current.
For centrifugal pumps and fans, converters with a general U/f characteristic are satisfactory. Pump sets can be continuously speed regulated with minimum losses. For centrifugal pumps which are known to have a square relationship torque/speed curve ($M \approx n^2$), then no torque or performance loss need be expected for norm and transnorm-motors, compared to normal mains operation. For this reason conversion to converter speed regulation after installation, is normally not a problem.
When utilising speed control, the upper mechanical speed limit of the motor, as well as the mechanical and hydraulic limitations of the pump must be considered. Similarly the manufacturers minimum recommended speed of the pump, due to hydraulic loads must be observed.

Frequency converters can be considered for the following :

Three phase asynchronous squirrel cage induction motors
These motors are the most widely used for frequency converter drives. All motors of this type, up to frame size 250 and mains supply 500V can be used with speed regulation. Motors from size 280 and above, or with mains supply 500V, need specially protected bearings and sometimes, depending on size, special insulation.

When selecting the motor, the required torque at the maximum speed must be taken into consideration.
Smaller motors up to 7.5 kW are supplied with integral converters. These have the advantage of no external wiring between the motor and converter and so have a reduced danger of interference emissions.
High voltage motors can also be supplied with high voltage frequency converters.

Reluctance motors
Reluctance motors are three phase motors, with synchronous behaviour in which the windings are fixed to the stator. The rotor with stamped poles does not carry an excitation voltage. The induced voltage in the stator windings, is caused by the changes in magnetic resistance (reluctance) caused by the rotor turning. The synchronous running is achieved by a special rotor design.
Reluctance motors share dimensions in common with the frames of standard motors and are available in constructions IM B3, IM B5 and IM V1. They are manufactured in protection class IP55 and cooling class IC 411, as 4 pole motors in frame sizes 71 to 160L and ratings 0.17 to 8.5 kW. They can be driven with frequency converters in the range 50 to 200 Hz giving speeds from 1500 to 6000 rpm.

Permanent magnet motors
These are permanently excited brushless self starting three phase synchronous motors. The rotor has a squirrel cage for asynchronous starting and a permanent magnet rotor for synchronous operation. The permanent magnets are made of ferritic material or rare earth alloy as in magnetic couplings.
Permanent magnet motors share dimensions in common with the frames of standard motors and are available in constructions IM B3, IM B5 and IM V1. They are manufactured in protection class IP44 and cooling class IC 411, as 2, 4 & 6 pole motors in frame sizes 71 to 160 and ratings 0.3 to 5.5 kW. The 2 pole version can be driven with frequency converters in the range 50 to 300Hz giving speeds from 1500 to 18000 rpm. The maximum frequency for 4 and 6 pole motors is 200Hz.

9.6.1.3 Changing the slip

This system of speed control is used for three phase asynchronous motors with slip ring rotors.
These motors operate with either a cascade of converters, a pulse resistance or with a double (rotating) field supply.

For centrifugal pumps, generally only sub-synchronous converter cascade is considered and mainly for high power, boiler feed pumps measured in MW up to 25 MW.
The speed control is achieved by altering the rotor resistance. The slip is steplessly controlled by the addition of external resistances connected to the rotor winding through the slip rings. The typical speed control range lies between 1: 1.3 and 1: 5.
The slip power P_{slip} taken by the slip rings when using a cascade is converted and fed back to the mains through an inverter. The current converter is sized for the maximum slip power P_{slip} the slip power is fed back through the cascade.

9.6.2 Variable speed coupling

Apart from controlling the speed of the drive, it is possible to regulate the speed of the pump with a variable speed coupling. These can be mechanical or electromechanical devices. Hydraulic couplings known as hydrodynamic or fluid couplings can be used, also in combination with a turbo-gearbox to give a gear regulated coupling.

Such geared couplings are used for example, in power stations to match the speed and control of the boiler feed pump, to the requirement.

Fluid couplings use a transmission fluid, usually a turbine oil, to transmit the torque between the drive and driven shafts. The pumping wheel on the driving shaft converts the mechanical power into hydraulic power by accelerating the transmission fluid which is in turn reverted to mechanical power by a turbine wheel on the driven shaft. If the speed of the driving and driven wheels is equal, then no torque is generated and no power transmitted. Therefore for transmission of power there must always be a slip between the driving and driven wheels, i.e. the driver is higher speed than the driven. The slip and therefore the speed regulation can be adjusted by altering the fill of the coupling housing by means of an adjustable scavenge pipe. Stepless speed control in the range 4:1 up to maximum 5:1 is possible.

The efficiency of a fluid coupling is very good, as the slip when transmitting the rated power is very small. The losses in the coupling made up of slip power and mechanical losses must be removed by an oil cooler. The greater the speed regulation the greater the slip and the poorer the efficiency of the drive, but this is within acceptable limits.

9.7 Selection tables for 3 phase asynchronous squirrel cage induction motors

The IEC (International Electrotechnical Commission) published with IEC 72 recommendations for the dimensions of electrical machines. These recommendations cover the dimensions of the housings, flanges and shaft ends independently from each other. The dimensions refer to the shaft centre line height (H) from 56 to 315 mm. This dimension (H) also defines the frame size of the motor.

As a result of the IEC recommendations, standards have been developed in individual countries for the most widely used motor types. These standards combine the frame size and rated power dependent on protection class and speed. This allows the space requirement of the drive to be fixed at the planning stage, knowing only the power, speed and protection class. Additional frame sizes for (trans-norm) motors of higher power with correspondingly higher shaft heights from 355 to 450mm, extend this standard.

Table 9.22 Overview of DIN standard squirrel cage induction motors, surface cooled

Construction	Protection class	DIN	Selection table
IM B 3	IP 44 or above EEx e II Increased safety EEx d IIC Explosion proof	42 673-1 42 673-2 42 673-3	9.23 9.24 9.25 9.26 9.27 9.28
IM B 35, B5 and V1	IP 44 or above EEx e II Increased safety EEx d IIC Explosion proof	42 677-1 42 677-2 42 677-3	as above *note:* IM B 35 to 315 L IM B 5 to 200 L

The values given in tables 9.23 to 9.25 for efficiency, power factor and rated current are guideline values only and exact data should be obtained from the motor manufacturer.

Table 9.23 Three phase asynchronous, surface cooled, squirrel cage induction motors, IP55, frame sizes 80 to 315L (norm motors)

Frame size	3000 rpm, 2-pole 50Hz				1500 rpm, 4-pole 50Hz			
	Rated power kW	Efficiency % [1)]	Power factor cos φ	Rated current Amp at 400 V	Rated power kW	Efficiency % [1)]	Power factor cos φ	Rated current Amp at 400 V
80	0.75	74	0.83	1.8	0.55	71	0.79	1.4
	1.1	76	0.84	2.5	0.75	74	0.79	1.9
90 S	1.5	78	0.82	3.4	1.1	74	0.81	2.7
90 L	2.2	80	0.85	4.7	1.5	74	0.81	3.6
100 L	3	83.5	0.85	6.1	2.2	80	0.82	4.9
					3	81.5	0.83	6.4
112 M	4	85.5	0.88	7.7	4	84	0.83	8.3
132 S	5.5	84.5	0.85	11.1	5.5	86	0.81	11.4
	7.5	86	0.86	14.7				
132 M					7.5	87.5	0.82	15.1
160 M	11	87	0.85	21.4	11	88.5	0.84	21.4
	15	88.5	0.87	28.2				
160 L	18.5	90	0.85	34.7	15	90	0.84	28.5
180 M	22	92	0.88	39	18.5	90.5	0.83	35
180 L					22	91	0.84	41
200 L	30	92	0.89	53	30	92	0.86	55
	37	93	0.89	65				
225 S					37	93	0.87	66
225 M	45	94	0.89	78	45	93	0.87	80
250 M	55	94	0.91	93	55	94	0.87	97
280 S	75	95	0.90	128	75	95	0.86	132
280 M	90	95	0.91	150	90	95	0.86	160
315 S	110	95	0.90	186	110	95	0.86	194
315 M	132	95	0.90	225	132	95	0.87	230
315 L	160	95	0.91	265	160	96	0.87	275
	200	96	0.92	325	200	96	0.87	345

Note 1. See the following page

Table 9.24 Three phase asynchronous, surface cooled, squirrel cage induction motors, IP55, frame sizes 315 to 450 (trans-norm motors)

Frame size	3000 rpm, 2-pole 50Hz				1500 rpm, 4-pole 50Hz			
	Rated power kW	Efficiency %	Power factor cos φ	Rated current Amp at 400 V	Rated power kW	Efficiency %	Power factor cos φ	Rated current Amp at 400 V
315	250	96	0.90	415	250	96	0.88	425
	315	97	0.91	520	315	96	0.88	540
355	355	97	0.90	590	355	96	0.87	610
	400	97	0.91	660	400	96	0.87	690
	500	97	0.91	820	500	97	0.88	850
400	560	97	0.91	910	560	97	0.88	950
	630	97	0.91	1020	630	97	0.88	1060
	710	97	0.91	670	710	97	0.89	690 [1)]
450	800	97	0.91	760	800	97	0.88	780 [1)]
	900	97	0.92	840 [2)]	900	97	0.88	880 [1)]
	1000	97	0.93	920 [2)]	1000	97	0.89	970 [1)]

1) Energy saving motors with European efficiency clarification according to EU. / . CEMEP (CEMEP = European Committee of Manufacturers of Electrical Machines and Power Electronics)

2 and 4 pole electric motors in the power range 1.1 to 90kW are supplied, based on the EU / CEMEP agreement as efficiency class "eff 2" (improved efficiency) or, "eff 1" (high efficiency).

The efficiencies listed in Table 9.24 correspond to efficiency class "eff 3 " (standard efficiency) motors of class "eff 2 " and "eff 1" should be requested from the supplier.

2) Measured at 690 V

Table 9.25 Three phase asynchronous, surface cooled, squirrel cage induction motors, IP55, frame sizes 160 to 450 (norm and trans-norm motors)

Frame size	1000 rpm, 6-pole 50Hz				750 rpm, 4-pole 50Hz			
	Rated power kW	Efficiency %	Power factor cos φ	Rated current Amp at 400 V	Rated power kW	Efficiency %	Power factor cos φ	Rated current Amp at 400 V
160 M	7.5	86	0.74	17	5.5	83.5	0.73	13
160 L	11	87.5	0.74	25	7.5	85.5	0.72	18
180 L	15	89.5	0.77	32	11	87	0.75	24
200L	18.5	90	0.77	39	15	87.5	0.78	32
	22	91	0.77	46				
225 S					18.5	89	0.79	38
225 M	30	92	0.77	61	22	90	0.79	45
250 M	37	92	0.86	68	30	92	0.82	58
280 S	45	93	0.86	81	37	93	0.82	70
280 M	55	93	0.86	99	45	93	0.83	84
315 S	75	94	0.86	134	55	93	0.82	104
315 M	90	94	0.86	160	75	94	0.83	138
315 L	110	95	0.86	194	90	94	0.83	166
	132	95	0.86	235	110	94	0.83	206
	160	95	0.86	280	132	94	0.82	245
315	200	96	0.87	345	160	95	0.82	295
	250	96	0.87	430	200	95	0.82	370
355	315	96	0.87	540	250	96	0.82	460
	400	96	0.87	690	315	96	0.82	580
400	450	96	0.86	780	355	96	0.82	650
	500	96	0.87	860	400	96	0.82	730
	560	97	0.87	960	450	96	0.82	820
450	630	97	0.86	1100	500	96	0.81	920
	710	97	0.87	710	560	96	0.81	1040
	800	97	0.87	790	630	96	0.81	1160

Table 9.26 Three phase asynchronous, surface cooled, squirrel cage induction motors, explosion protection EEx e II, increased safety, frame sizes 90 to 355 (norm and trans-norm motors)

Frame size	3000 rpm		1500 rpm		1000 rpm	
	Rated kW for temperature class					
	T1, T2	T3	T1, T2	T3	T1, T2	T3
90 S 90 L	1.3 1.85	1.3 1.85	1 1.35	1 1.35	0.65 0.95	0.65 0.95
100 L	2.5	2.5	2 2.5	2 2.5	1.3	1.3
112 M	3.3	3.3	3.6	3.6	1.9	1.9
132 S 132 M	4.6 6.5	4.6 5.5	5 6.8	5 6.8	2.6 3.5 4.8	2.6 3.5 4.8
160 M 160 L	9.5 13 16	7.5 10 12.5	10 13.5	10 13.5	6.6 9.7	6.6 9.7
180 M 180 L	19	15	17 20	15 17.5	 13.2	 13.2
200 L	25 31	20 24	27	24	16.5 20	16.5 20
225 S 225 M	 38	 28	33 40	30 36	 27	 27
250 M	47	36	50	44	33	33
280 S 280 M	64 76	47 58	68 80	58 70	40 50	40 46
315 S 315 M	95 112	68 80	100 120	84 100	68 82	64 76
315 L	135 165	100 125	135 165	115 135	98 120 135	92 110 125
315	200 255	150 190	200 245	170 215	175 215	160 200
355	300 335 400	220 25ß 300	275 315 400	240 275 350	275 340	250 315

Table 9.27 Three phase asynchronous, surface cooled, squirrel cage induction motors, explosion protection EEx de IIC, explosion proof, frame sizes 80 to 315 (norm motors)

Frame size	3000 rpm	1500 rpm	1000 rpm	750 rpm
	Rated kW for temperature class T1 toT4			
80	0.75 1.1	0.55 0.75	0.37 0.55	
90 L	1.5 2.2	1.1 1.5	0.75 1.1	0.37 0.55
100 L	3	2.2 3	1.5	0.75 1.1
112 M	4	4	2.2	1.5
132 S	5.5 7.5	5.5 7.5	3	2.2
132 M			4 5.5	3
160 M	11 15	11	7.5	4 5.5
160 L	18.5	15	11	7.5
180 M	22	18.5		
180 L		22	15	11
200 L	30 37	30	18.5 22	15
225 S	45	37		18.5
225 M		45	30	22
250 M	55	55	37	30
280 S	75	75	45	37
280 M	90	90	55	45
315 S	110	110	75	55
315 M	132 160	132 160	90 110 132	75 90 110
315 L	200	200	160	132

Table 9.28 Three phase asynchronous, surface cooled, squirrel cage induction motors, explosion protection EEx de IIC, explosion proof, frame sizes 355 to 450 (trans-norm motors)

Frame size	3000 rpm	1500 rpm	1000 rpm	750 rpm
	Rated kW for temperature class T1 toT4			
355 M	250	225	200	160
		250		
355 L	315	280	250	200
		315		
400 S	355	355	280	250
400 M	400	400	315	280
400 L		450	355	315
450 M	450	500	400	355
450 L	500	560	450	400
	560	630	500	450

10 Water

10.1 Natural water, drinking water and industrial process water

Water is the molecule H_2O in the liquid state.

The density of pure water at 0°C is 0,9998 kg/dm³. If the temperature falls below 0°C, water freezes to form solid ice or to be precipitated as snow. The density of ice at 0°C is only 0,91674 kg/dm³. This volumetric expansion of approximately 9% as water freezes is the cause of water filled pipes and tanks bursting due to frost.

At a temperature of 4°C and atmospheric pressure 1013 mbar the density of water is 1,0000 kg/dm³. The boiling point at atmospheric pressure 1013 mbar is 100°C. The boiling point rises and falls with atmospheric pressure.

Pure water does not occur naturally. Water normally contains various dissolved substances depending on its origin (underground or surface). These may be salts, free acids or even gases and they will influence the quality and properties of the water. The solubility is dependent on temperature and so materials can either be deposited or dissolved. Furthermore, raw water can often contain undissolved matter of an organic or inorganic nature in the form of a suspension which may be removed by settling.

Drinking water and industrial water (commercial, industrial, agricultural or similar uses) require certain quality levels. In most cases treatment of the raw water is necessary.

An analysis is required in order to be able to assess any corrosive properties.

10.2 Important parameters for assessing water condition

10.2.1 Hydrogen ion concentration, pH value

The molecules of the dissolved chemical constituents in an aqueous solution are partly divided into opposite charged particles. These are known as ions, the division is called dissociation and the solution, which contains them, is an electrolyte.
Further:

- positively charged ions are cations
- negatively charged ions are anions

The ions of metals and hydrogen usually carry a positive charge e.g.

$[Ca^{2+}]$, $[Na^{+}]$, $[Mg^{2+}]$, $[H^{+}]$

Acidic and hydroxyl ions have a negative charge e.g.

$[Cl^{-}]$, $[HCO_3^{-}]$, $[CO_3^{2-}]$, $[SO_4^{2-}]$, $[NO_3^{-}]$, $[OH^{-}]$

The concentration of hydrogen ions determines the nature of an aqueous solution, i.e. whether it is acidic, alkaline or neutral. In practice the range of H^- or OH^- ions concentration, are considered to lie between 1 to 10^{-14} mol/l. Since these numerical values are cumbersome for general use e.g. graphical presentation, the value for the hydrogen ion concentration [H^-] expressed as negative logarithm to base 10 is designated as the value "pH"

$$pH = -\lg [H^+] \quad \text{or.} \quad [H^+] = 10^{-pH} \text{ mol/l}$$

The product of the hydrogen ion concentration and the hydroxyl-ion concentration, the dissociation constant $K_w = [H^+] \cdot [OH^-]$, is the same for all aqueous solutions, hence equally for acids, alkalis and salt solutions and is principally dependent on temperature. At the reference temperature 25°C its value is ca. 10^{-14} (mol/l). If the value of [H^+] increases, the value of [OH^-] must decrease and vice versa.

Solutions, which have an equal hydrogen ion and hydroxyl-ion concentration, behave as neutral. This condition exists for example with chemically pure water. At the reference temperature 25°C the concentration of hydrogen and hydroxyl ions are both 10^{-7} mol/l. Chemically pure water at 25°C with a pH value 7 behaves as neither acid nor alkaline.

A solution whose hydrogen ion concentration is greater than the hydroxyl ion concentration is acidic. The pH value is therefore lower than that of a neutral solution at the same temperature.

Example:

Hydrogen ion concentration = 0,1 mol/l (10^{-1} mol/l) $\rightarrow$ pH value = 1

A solution whose hydrogen ion concentration is lower than the hydroxyl-ion concentration is alkaline (sometimes referred to as basic). The pH value is therefore greater than that of a neutral solution at the same temperature.

Example:

Hydrogen ion concentration = 0,000000000001 mol/l (10^{-12} mol/l) $\rightarrow$ pH value = 12

The well known pH scale with its classification ranges from 0 to 14 and refers to a solution at a reference temperature of 25°C.

pH value	Chemical reaction	pH value	Chemical reaction
0 to 3	Strongly acidic	8 to 10	Weakly alkaline
4 to 6	Weakly acidic	11 to 14	Strongly alkaline
7	Neutral		

Outside of this range, aqueous solutions with pH < 0 are ultra acidic (e.g. 20% hydrochloric acid is pH = – 0.3) and those with pH > 14 are ultra alkaline (e.g. 50% potassium hydroxide is pH = 14.5).

The dissociation constant K_w varies with increasing temperature. The number of hydrogen ions $[H^+]$ and also hydroxyl ions $[OH^-]$ increase but remain in the same ratio.

It is often thought that such an aqueous solution must be corrosive. For this reason boiler feed pumps are often given a mild alkaline dosing to lift the pH value before the pump inlet.

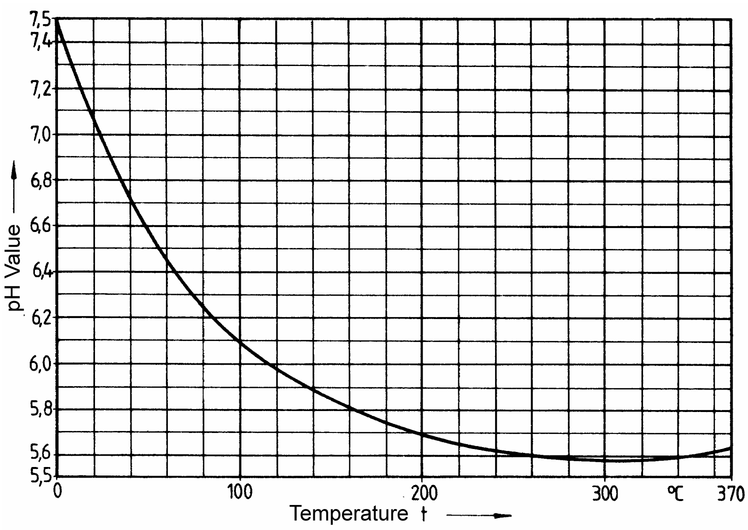


Fig. 10.01 pH value of chemically pure water dependent on temperature

10.2.2 Water hardness

The hardness of water refers to the content of alkaline mineral ions (calcium, magnesium, strontium and barium). The concentration of calcium and magnesium is principally of importance, but in special cases, e.g. seawater, then strontium and barium must also be considered.

Calcium and magnesium salts are found mainly as compounds of carbonate, bicarbonate, sulphate, nitrate and chloride.

The term "water hardness" has historical roots in the effect of calcium ions on fatty acid soaps used in washing, but has no scientific merits.

Instead of using this term, accurate reference should be made to the concentration of the alkaline mineral ions.

The following table compares the preferred new designation and the old terms:

New designation	Old term
Total alkaline minerals	Total hardness
Calcium ions	Calcium hardness
Magnesium ions	Magnesium hardness
Carbonate ions in the alkaline minerals	Temporary hardness
Non-carbonate ions in the alkaline minerals	Permanent hardness

The hardness of water is expressed as the concentration of alkaline mineral ions (DIN 38 409 part 6) e.g. $c\ (Ca^{2+} + Mg^{2+}) = 5$ mmol/l.

The previously commonplace units such as mval/l, °d etc. should be avoided. However in order to be able to utilise existing tables and textbooks, the following table compares and gives conversion factors for the obsolete units:

Hardness	$c(Ca^{2+}+Mg^{2})$ in mmol/l	in mval/l	$CaCO_3$ in ppm	°d	°e	°f
$c(Ca^{2+}+Mg^{2})$ in mmol/l	1	2	100	5.6	7.0	10.0
in mval/l	0.5	1	50	2.8	3.51	5.0
$CaCO_3$ in ppm	0.01	0.02	1	0.056	0.07	0.10
1 degree °d (German)	0.1786	0.357	17.85	1	1.25	1.786
1 degree °e (English)	0.1425	0.285	14.29	0.7999	1	1.429
1 degree °f (French)	0.10	0.20	10.00	0.5599	0.70	1

Table 10.01 Comparison of current unit of hardness mmol/l and previous commonplace units

For washing powder usage, the water hardness is categorised as follows

Hardness	Hardness in mmol/l	Total hardness in °d
1 (soft)	up to 1.3	up to 7
2 (medium hard)	1.3 to 2.5	7 to 14
3 (hard)	2.5 to 3.8	14 to 21
4 (very hard)	over 3.8	over 21

Table 10.02 Categorisation of water hardness for washing powder usage.

Apart from this there is a generally accepted classification of water hardness

Hardness ($Ca^{2+} + Mg^{2+}$) in mmol/l	Total hardness	Description
up to 0.7	up to 4	very soft
0.7 to 1.4	4 to 8	soft
1.4 to 2.1	8 to 12	medium hard
2.1 to 3.2	12 to 18	rather hard
3.2 to 5.4	18 to 30	hard
over 5.4	over 30	very hard

Table 10.03 Generally accepted classification of water hardness

10.2.3 Carbon dioxide

All naturally occurring water contains carbon dioxide in its free form and as its anions.

The chemically bound form (CO_3^{2-}, HCO_3^{-}) is found as either calcium carbonate (lime) or magnesium carbonate (magnesia).

The free carbon dioxide (CO_2) is mostly found as dissolved gas. As part of this, the associated free carbonic acid, causes hydrogen carbonate to be held in solution. The free carbonic acid is not aggressive. The proportion of free carbon dioxide, which is contained in the water above the equilibrium concentration of the associated free carbonic acid, is termed excess or aggressive (compared to lime).

Water which has a concentration of free carbon dioxide equal to the equilibrium concentration of the free associated carbonic acid, i.e. just sufficient to hold the calcium hydrogen carbonate in solution, is in lime / carbonic acid equilibrium. It is neither aggressive to lime, nor is lime deposited.

10.2.4 Degree of acidity and alkalinity of water

10.2.4.1 Degree of acidity of water

The degree of acidity of water (K_S) is the quotient of the number of hydroxium ions which a given volume of water can absorb, until a specified pH value is reached, to the volume of this portion:

$$K_S = n\,(H_3O^+) \,/\, V\,(H_2\,O) \qquad \text{SI unit:} \quad mol/m^3$$

common unit: mmol/l

The value is to a small extent dependent on the temperature and ion strength of the water.

The degree of acidity up to a pH value 8.2 ($K_{S\ 8.2}$) is defined when this pH is achieved through the addition of ions, generally hydrochloric acid, with a known content (e.g. c(HCl) = 0,1 mol/l).

This pH value is derived from carbonic acid. The pH value 8.2 is an aqueous hydrogen carbonate solution with more than 1 mmol/l HCO_3 ions, a temperature of 25°C and an ion strength 10 mmol/l.
Previously the degree of acidity $K_{S\ 8.2}$ was designated as the p-value.
The degree of acidity up to a pH value 4.3 ($K_{S\ 4.3}$) is defined when this pH is achieved through the addition of ions, generally hydrochloric acid, with a known content.

This pH value is derived from carbon dioxide solution. The pH value 4.3 is an aqueous carbon dioxide solution with a hydrogen carbonation concentration in mmol/l of 1% of the dissolved carbon dioxide, a temperature of 25°C and an ion strength 10 mmol/l.

Previously the degree of acidity $K_{S\ 4.3}$ was designated *m* value.

The previously commonly used term of temporary hardness °d can be derived from the $K_{S4.3}$ value as follows

Temporary hardness °d = $K_{S\ 4.3} \cdot 2.8$

10.2.4.2 Degree of alkalinity of water

The degree of alkalinity of water (K_B) is the quotient of the number of hydroxyl ions which a given volume of water can absorb, until a specified pH value is reached, to the volume of this portion:

$K_B = n\,(OH^-) / V\,(H_2\,O)$	SI unit:	mol/m³
	common unit:	mmol/l

The degree of alkalinity up to a pH value 8,2 ($K_{B\ 8,2}$) is defined when this pH is achieved through the addition of hydroxyl ions, generally sodium hydroxide, with a known content (e.g. c (NaOH) = 0,1 mol/l).

The previously commonly used term free carbon dioxide (CO_2) can be derived from the $K_{B\ 8,2}$ as follows:

free CO_2 in mg/l = $K_{B\ 8.2} \cdot 44$

The degree of alkalinity up to a pH value 4,3 ($K_{B\ 4.3}$) is defined when this pH is achieved through the addition of hydroxyl ions, generally sodium hydroxide, with a known content.

As with the degree of acidity, this pH value is defined in the same way using carbonic acid/water.

10.2.5 Electrical conductivity of water

The electrical conductivity of water gives a measure of the dissolved salts or dissociated materials.

It is expressed in Siemens per metre (S/m). Common units are: mS/m (= 10^{-3} S/m) and µS/cm (= 10^{-4} S/m).

Industrial waters generally have a conductivity of up to 100 mS/m. By full demineralisation, a conductivity level < 0,05 mS/m can be achieved. The accuracy of measurement of the electrical conductivity has special importance in the assessment and control of the purity of boiler feed water, steam turbine condensate and purified water.

10.3 Analysis of water

The corrosion characteristics of a particular grade of water can be assessed from the following table 10.4.

In assessing the corrosion risk, the nature and concentration of dissolved substances, which act as natural inhibitors by causing the formation of a protective coating, are of importance, e.g. phosphates, silicates and aluminium compounds.

When investigating the occurrence of corrosion, the concentration of materials taken into solution after the point of supply can be very important, e.g. copper ions or other corrosion products from the components installed in the plant. It is then necessary to do a further water analysis nearer the point of corrosion.

Table 10.04 Coverage of a basic water analysis

Description of sample: Source of sample:		Date of sample:	
Parameter	Unit	Parameter	Unit
Water temperature	°C	Chloride ions	mol/m³
pH value		Nitrate ions	mol/m³
Electrical conductivity	mS/m	Sulphate ions	mol/m³
Degree of acidity to pH=4,3	mol/m³	Total iron	mol/m³
Degree of alkalinity to pH=8.2	mol/m³	Filtered iron	mol/m³
Total alkaline minerals	mol/m³	Total manganese	mol/m³
Calcium ions	mol/m³	Oxygen	g/m³
Magnesium ions	mol/m³	Oxidised manganese	g/m³

10.3.1 Judging the likelihood of corrosion of non-alloy and low alloy ferrous materials (per DIN 50 930)

The corrosion behaviour of a material is not only dependent on the properties of the material and the water, but is also influenced by the operating conditions. In practice the properties of the material and the water and similarly the operating conditions are neither fully known nor steady enough to make an exact judgement. It is therefore normally only possible to make an estimation of the probable corrosion effect of the water on the metallic material.

10.3.1.1 Formation of homogeneous coating

An estimation of the anticipated build up of protective coating can only be made for temperatures under approximately 30°C. The following is therefore only valid for cold water.

The formation of protective coatings is promoted under the following conditions:

$c\ (O_2)$	> 3 g/m³,	$c\ (Ca^{2+})$	> 0.5 mol/m³
pH	> 7.0	$K_{S\,4,3}$	> 2 mol/m³

Moreover, the flow conditions are decisive for the formation of protective coatings. These can only occur in flowing water, if the flow is interrupted, the coating is destroyed. On the other hand, high flow velocities can prejudice the formation of a protective coating. For flow velocities in excess of 4 m/s, erosion effects occur.

10.3.1.2 Uniform surface corrosion

Uniform corrosion always occurs in acidic water. The corrosion rate can be greatly reduced by surface coatings, so that the rate of material loss is technically acceptable.

In the event of failure of the protective coating, the corrosion rate is dependent on the concentration of oxidising agents, $c\ (O_2)$, $c\ (NO_3^-)$ and $c\ (H^+)$, in the presence of sulphate reducing bacteria also $c\ (SO_4^{2-})$) and the flow velocity. The corrosion rate for cold water can be neglected when:

$c\ (O_2) < 0.1$ g/m³ and $K_{B\,8.2} < 0.05$ mol/m³

In warm water the $c\ (O_2)$ level is lower.

With $c\ (O_2) < 0.02$ g/m³ no corrosion elements are present.

10.3.1.3 Uneven surface corrosion

Pitting and holing corrosion occurs when there is no protective coating or incomplete coating and is generally the case in water containing oxygen. The scale of the localised attack depends on many factors, especially the geometry and dimensions of the component, the condition of the material surface and the conditions when the corrosion began. The rate of corrosion is reduced when:

$$S_1 = \frac{c\ (Cl^-) + 2\ c\ (SO_4^{2-})}{K_{S\,4.3}} \text{ is} < 1$$

For water with a relatively low value of $K_{S\,4.3}$ and a high quotient value it is possible for uneven corrosion to make the transition to uniform corrosion.

In water with depleted oxygen, c (O_2) < 0,1 g/m³ no protective coatings are formed to cause uneven corrosion.

10.3.2.4 Judging the analysis of individual component values

Chloride (Cl^-)

Chloride ions are found in almost all natural waters, rainwater and wastewater. Their concentration varies according to local conditions within a wide range (barely mg/l to ca. 250 mg/l). If the concentration of chloride ions is greater than 150 mg/l there is the danger of pitting corrosion. The softer the water and the lower the carbonate level, the greater the attack of the chloride ions on metals.

Sulphate (SO_4^{2-})

Normal water contains mostly 10 to 30 mg/l of sulphate ions. Water containing more than 250 mg/l is corrosive to iron. Spheroidal graphite (SG) iron and steel suffer pitting. Sulphate rich water is also damaging to concrete. This occurs with concentrations above 150 to 200 mg/l (SO_4^{2-}).

Nitrate (NO_3^-)

Nitrate ions are found in various concentrations in underground and surface water. Normal values lie around ca. 10 mg/l NO_3^-. Concentrations over 20 to 50 mg/l will attack iron if the water is soft even if it contains carbon dioxide promoting the formation of, a protective rust coating.

Nitrite (NO_2^-)

Nitrite is virtually never found in pure water. Dirty water may contain 0.1 to 2 mg/l, moorland water 0.1 to 1 mg/l. The presence of nitrite generally indicates impurities from animal and human waste.

Ammonium-nitrogen (NH_4^+, N)

Ammonium-nitrogen is found in many surface and underground water sources and in all household and often industrial wastewater. The nature in which the ammonium-nitrogen is found, whether as NH_4^+ ions or NH_4 OH and/or NH_3^+, depends on the pH value of the water. Iron is attacked at concentrations over 20 mg/l. Copper and copper alloys corrode in the presence of ammonia and ammonium salts and should be avoided.

Iron (Fe)

Iron appears in various forms in water: as undissolved iron (II) and iron (III) ions, as dissolved colloidal or organic compounds, apart from complex compounds, especially in waste water.

The iron content can be specified as:

- Total iron i.e. the sum of dissolved and undissolved iron
- Total dissolved iron i.e. the sum of iron (II) and iron (III) compounds

Iron is invariably chemically corrosive. With a content over 0,2 mg/l an ochre sludge is deposited which increasing with rising Fe and O_2 content can be deposited in pipework with low flow velocity. Pumps are less affected, but only mechanical cleaning is effective.

Manganese (Mn^{2+})

Manganese is invariably chemically corrosive. With content greater than 0.1 mg/l it leads to deposition like iron.

Oxidisation of Mn VII to II

The capacity for oxidisation is a measure of the impurities in a water source. It is expressed in g/m³ O_2.

Typically occurring values:

pure underground water	0.7 to 2 g/m³
pure surface water	2.5 to 7.5 g/m^3
dirty water	5 to 38 g/m³
extremely dirty water (moorland water)	> 75 g/m³

The danger of corrosion due to organic materials occurs when, e.g. deposition occurs (danger of pitting) or hydrogen sulphide compounds are formed. In moorland water, the corrosive effects are caused by naturally occurring acids which are extremely aggressive to iron.

The previously common method of water analysis using potassium permanganate, can be related to the oxidisation capacity from the following:

$$1 \text{ g/m}^3\ O_2 = 3.95 \text{ mg/l } KMnO_4$$

Solids content

Borehole and surface water usually contain sand. Borehole water may contain up to 0.3 mg/l sand. Industrial water depending on the application may contain various solids from metallic scale, cinders, flue dust to slag granules.

The use of side channel pumps for liquids with a solids content is not recommended due to the narrow clearances.

The maximum solids content for radial, axial and mixed flow pumps is between 130 and 200 mg/l. This content however will cause considerable wear within the pump at points of high velocity.

10.4 Other natural waters

Mineral water is a natural water from natural or man-made wells containing at least 1000 mg of dissolved salts or 250 mg free carbon dioxide per kilogram of water.

Brine is a natural salt laden water containing at least 14 g of salt, mainly sodium chloride, per kilogram of water. Mineral waters in which the salt content has been made more concentrated by the removal of water is also referred to as brine, e.g. refinery or evaporator discharges. In the broadest sense, all waters with dissolved salts are referred to as brine.

The majority of mineral waters and all brines are extremely aggressive to materials. The material selection will depend on the analysis.

Thermal water is a water which, regardless of its mineral content, occurs naturally at a temperature above 20°C. An analysis is usually necessary to determine if the water will be aggressive to materials.

Mine drainage water often contains considerable quantities of metal salts and oxides. The selection of materials will be dependent on the analysis and the pH value whereby a considerable variation may be expected.

Sea water has approximately the following % by weight of dissolved salts, (mostly sodium chloride and magnesium chloride):

Oceans	3.3 to 3.75 %
North Sea	3.2 %
Baltic Sea	0.6 to 1.9 %
Mediterranean Sea	3.8 %
Dead Sea	21.7 %

Depending on the salt content, the electrical conductivity lies between 2100 and 5200 mS/m with a mean value of 4200 mS/m. In the Persian Gulf it is 7200 mS/m.

Seawater is aggressive due to its oxygen content and the conductivity producing salts and because the chloride content prevents formation of an effective protective layer.

Brackish water is a mixture of fresh water and seawater in a river estuary, which varies widely in content with the changing tides and levels and can rise in conductivity to ca. 3200 mS/m with often aggressive properties. The material selection will depend on the analysis.

10.5 Wastewater

Wastewater is the effluent produced by household, industrial and commercial use and the run off of storm water from cultivated and enclosed land. The material most usually used for pumps is cast iron. Copper alloys can be attacked by the presence of ammonia.

Commercial and industrial effluents have widely varying compositions The material selection will depend on the analysis.

10.6 Treated water in heating systems and steam power plant

10.6.1 Hot water

Water heating systems are classified as:

- Warm water heating with temperatures up to 110°C
- Hot water heating systems with temperatures above 110°C up to ca. 200°C
- Hot water heating systems from which hot water and saturated steam are extracted for manufacturing with temperatures over 110°C up to ca. 230°C

Domestic water heating systems are filled and topped up from the drinking water supply, without treatment. To avoid problems of lime scale deposits and oxygen corrosion it is recommended that an inhibitor is added to the system on first filling and that this is checked periodically for continued effectiveness. Under these conditions, a cast iron pump may be installed.

In larger installations, some method of softening the filling and top up water is necessary. This does however mean that no protective coating will form on the iron surfaces and the water will become aggressive if free oxygen or carbon dioxide are present. Oxygen especially will cause serious corrosion (pitting). A de-aeration system or use of a chemical to combine with the oxygen is essential if the system is to operate efficiently.

10.6.2 Preparation processes

The following processes are common for hot water systems:

Water softening

Several processes are available for softening according to the application requirement.

- De-carbonisation
- Chemical precipitation
- Ion exchange

The precipitation by the addition of lime, soda, sodium hydroxide or tri-sodium phosphate involves the removal of the sludge containing the hardness causing substance.

Demineralisation

There are two processes available

- Partial demineralisation, a de-carbonisation and softening process.
- Full demineralisation, for the removal of all ions contained in the water. This is designated as demineralised and de-ionised.

10.6.3 Boiler feed water

All water used in hot water - steam cycle systems in steam turbine power stations is prepared in line with standards and regulations.

In particular, for boiler feed water, regulations are valid for pressures up to 68 bar.

For boiler feed water different chemical processes are categorised:

Alkaline process

The alkaline process with oxygen free water predominates for circulation vessels with up to 68 bar pressure rating, as this is the only possible method for boilers with water containing salts. Corrosion can only be avoided when sufficient alkali is present. In most cases solid material is used to provide the alkalinity as volatile media is insufficient on its own.

Boiler feed water, which is free of salts, can be treated with volatile alkaline material if the conductivity is limited (measured conductivity after a strongly acidic cation exchanger < 3 µS/cm). It is recommended that a pH value > 7 in the boiler water is maintained. With conductivity in the boiler water > 3 µS/cm, the pH value should be fixed > 9.5 by additional solid alkaline material.

Neutral and combined processes

The neutral and combined processes assume salt free feed water with a conductivity after a strongly acidication exchanger of < 0.2 µS/cm. After dosing with oxidising material such as oxygen O_2 or hydrogen peroxide H_2O_2, with a pH value over 6.5 a protective coating of magnetite and oxides is formed.

Whereas with the neutral process a pH value of > 6.5 is maintained, with a combined process the pH value is adjusted between 8 and 8.5 with volatile alkaline medium. This is generally restricted to vessels with through flow.

10.6.4 Condensate

Condensate is produced in the steam - water cycle in a steam turbine power station. In principle the condensate has the same quality as the feed water, but in practice there is an increase in the salts content, which is not desirable in other parts of the system. Condensate is generally therefore passed through a clean up installation with ion exchange or full demineralisation to achieve the original feed water quality.

10.6.5 Cooling water

Cooling water must not cause deposition or precipitation on the cooling surfaces and must not be corrosive. Therefore, the following limits should be observed:

Total alkaline minerals	0.7 to 1.4 mmol/l
Iron	< 0.3 mg/l
Capacity for oxidisation	< 6 g/m³ O_2
Salts content (cooling tower operation)	max. 3000 mg/l
Chloride content	max. 600 mg Cl^-/l
Sulphate content	max. 400 mg SO_4^{--}/l

If the cooling system contains austenitic materials, the chloride content must be even lower, due to the danger of pitting.

Micro-organisms such as bacteria algae and fungi can cause a damaging type of deposition.

These organisms can be combated by dosing with chlorine. However to prevent corrosion, the chlorine addition should not exceed 2 mg free Cl_2 /l. A high shock dose treatment is only permitted for a short time.

However, for environmental reasons it is preferable to use bactericides or biocides rather than chlorine.

Bactericides in the right concentration restrict the growth and division of bacteria leading to their clear up.

Biocides involve the use of environmentally friendly chemicals to kill the organisms. They work through enzyme activity on the organism. Due to the ability of micro-organisms to mutate in the environment, it is necessary to carry out complete and effective control of the cooling system to reduce the damaging activity of bacteria and organisms.

In contrast to the use of chlorine, the use of biocides and bactericides have no corrosive effect on the cooling system.

If river water, brackish water or seawater is used for cooling, the presence of sand must be expected.

The design of the cooling water pump must allow for this. A solids content up to 200 mg/l is as a rule no problem, but higher contents may cause considerable wear, which even specially selected wear resistant materials and design measures may not reduce.

10.7 Material selection for various waters

The following table gives the materials of construction, grouped together, which are commonly used for various types of water

Table 10.05 Material groups for material selection as per table 10.07

Material group	Material	Standard
D*)	Grey cast iron	EN 1561 DIN 1691 ASTM A 48 and A 278
	Spheroidal graphite or nodular cast iron (ductile iron)	EN 1563 DIN 1693 ASTM A 395 and A 536
	Cast steel	EN 10213-2 DIN 17 245 ASTM A 216
E	Copper / tin / zinc alloy (Bronze and brass)	EN (1982) DIN 1705 ASTM B 584
F	Austenitic cast iron (Ni-Resist - Types)	EN (none) DIN 1694 ASTM A 436 and A 439
G	Martensitic stainless steel	EN (none) DIN 17 445, E DIN 17445 ASTM A 217 and A 743
H	Austenitic stainless steel	EN 10213-4 DIN 17 445, E DIN 17445 ASTM A 351, A 743, A 744
L	Austenitic ferritic stainless steels, Duplex	EN 10213-4 E DIN 17445 and SEW 410 ASTM A 890
R	Copper aluminium alloys (Aluminium bronze)	EN (1982) DIN 1714 and DIN 17 665 ASTM B 150

See the next page*)

*) the materials in group “D”
– grey cast iron
and
– spheroidal graphite iron
are shown in the following table with limiting values:

Table 10.06 Limits for cast iron

Regulation	Application	Max operating pressure bar	Max operating temperature °C	DN_1 max
Grey cast iron DIN EN 1561: EN-GJL-250, EN-JL1040 previous designation: DIN 1691: GG-25, 0.6025				
EN 12953-2 TRD 108	Circulation pumps	10	183	200
TRD 108	Feed pumps	32	140	-
VDMA 24 276	Pumps in chemical installations	10	100	-
Spheroidal graphite iron DIN EN 1563: EN-GJS-400-18, EN-JS1020 previous designation: DIN 1693: GGG-40.3, 0.7043				
TRD 108	Circulation pumps and feed pumps	40	250	-

Abbreviations: TRD = Technical regulations for boilers
VDMA = German machinery manufacturers association
DN_1 = Nominal bore of pump suction

For installations above the limiting values, tougher materials should be selected.
Such as:
Cast steel to DIN EN 10213-2 (material group D)
e.g.. GP240GH, 1.0619; previous designation GS-C25
Martensitic cast steel to DIN 17 445 (material group G)
e.g.. GX7CrNiMo12-1, 1.4008; previous designation G-X8 CrNi 13

Table 10.07 Material selection for handling various waters

Boiler feed water		
alkaline process pH > 7	G	
pH > 9,5	D D/G G	
neutral and combined process pH > 6,5	G	
pH > 8,5	D D/G G	
Bore hole water	D D/E E	2
Brackish water	F L	
Cooling tower water salt content < 3000 mg/l	D D/E	
salt content > 3000 mg/l	F H L R	6
Demineralised water	G H	
Drinking water	D D/E	
Effluent domestic	D H	
industrial	D H L	1
Fire hydrant water	D D/E	
Fully de-ionised water	G H	
Mine drainage water	H L	3
Heating water	D D/E	
Hot water	D D/G G	4
Partly de-ionised water	D G H	
Rain water / surface water	D D/E	
River water, neutral	D D/E	
aggressive	H L R	
Sea water cold	E F H L R	6
warm and hot	F L R	6
Steam turbine condensate	D D/G	
Very pure water	H	5
Water with:		
Chloride 10 000 to 100 000 ppm, pH > 3	L	6
Chloride 10 000 to 100 000 ppm, pH < 3	L	6

**) D/E and D/G means: D = casing material, /E and /G = impeller material

Tips for table 10.07

1. The material selection is dependent on the actual composition of the effluent and a full analysis should be made.
2. Borehole water often contains aggressive carbon dioxide and very little oxygen. In this case no protective layer is formed.
3. The material selection is dependent on the actual composition of the mine water and a full analysis should be made.
4. Heating water is normally treated so that no corrosion is likely with common materials. The material selection is therefore based in the first instance on the operating temperature and the maximum operating pressure. See also section 4.13.2.5 "materials for hot water circulation pumps."
5. The purest water is defined by triple distilled water with a salts content 1 to 2 ppm and conductivity 0,5 to 1 µS/cm.
6. For high chloride contents, materials with great resistance to pitting are necessary. This can be identified from a material index as follows:

   Index WS = % Cr + 3,3 · % Mo

   For materials which contain nitrogen, the beneficial effect of this on resistance to pitting can be considered from the following:

   Pitting Index PI = % Cr + 3,3 · % Mo + 16 · % N

   Materials with a material index or pitting index greater than 32 can be considered as broadly pitting resistant. The well known austenitic-ferritic steels (Duplex) achieve an even better index in excess of 40.

10.8 Properties of various waters

Pure water

	Conductivity µS/cm	Salts content mg/kg	Gases mg/kg	
			O_2	CO_2
Distilled water	2	< 20		
Triple distilled water	0.5 - 1	1 - 2		
Fully Demineralised water 1)	0.5	0.2 - 0.5	0.01 - 0.03	0.5 – 1.0
De-ionised water 2)	0.1 - 1.0	< 0.1 (Chloride)		

1) Boiler feed water for high pressure boilers, t=250 to 325 °C
2) Primary cooling water in a reactor power station, t=220 to 350 °C

Water, heavy water, hydrogen isotopes

Isotope	Description	Percentage of natural hydrogen	Symbol	Molecule
^1H	Protium light water	99,984 %	H	H_2O Water density max at 4 °C $\rho = 0.999\ 973$ kg/dm³ Fp = 0 °C, Kp = 100 °C
^2H	Deuterium heavy water	0,016 %	D	D_2O heavy water Density max at 11.6 °C $\rho = 1.1057$ kg/dm³ Fp = +3.8 °C, Kp = 101.4 °C
^3H	Tritium super heavy water	10^{-15} %	T	radioactive half life 12.3 years

Salt content of sea water (average for the oceans)

NaCl	29.60 g/l
$MgCl_2$	3.80 g/l
$MgSO_4$	2.25 g/l
$CaSO_4$	1.38 g/l

Apart from the above salts, seawater contains molecules of potassium, bromides, strontium, borate and fluorides. The oxygen content varies between 0 and 8.5 mg/l. Carbon dioxide is found in considerable quantities in sea water and maintains the pH value nearly constant at 7.8 to 8.3.

11 Materials and material selection

11.1 Materials

The materials which are suitable for pump construction are listed in the following overview 11.2 to 11.8.

The material overview 11.2 for iron and non-ferrous materials is based on the European Standard.

Additionally the superseded comparable national standards DIN and SEW are listed. This is intended to ease the comparison with the old national and the new European standards.

Materials which do not currently have a European standard are listed under the still valid DIN and SEW standards.

Where comparable, the American standards ASTM are also listed. It should be noted however that comparable materials are not similar in all details. Both the chemical analysis and mechanical properties may vary, but generally this should have no importance with respect to their use for pump materials. In borderline cases an exact comparison should be made.

Abbreviations

AISI	American Iron and Steel Institute
ASTM	American Society for Testing and Materials
DIN	German Standards Institute
EN	European Standard published by the European Standards Committee CEN
SEW	Steel and Iron Material Publications published by the German Iron & Steel Association
UNS	Unified Numbering System (USA)

Designation

A	Elongation at failure ($L_0 = 5\ d_0$)
KV	Notch impact (ISO-V test) at 20°C, or given temperature
R_m	Tensile strength
$R_{p\ 0,2}$	0,2% - Elongation limit at 20°C, for some materials the 0,1% limit or the 1% limit is given

11.2 Material overview - Cast iron and steel

European Standard - EN			German Standard - DIN			USA - Standard	
Material	Short name or description	Standard	Material	Short name or description	Standard	ASTM	UNS
Grey cast iron (flake)							
EN-JL2030	EN-GJL-HB195 1)	1561	0.6022	GG-190 HB	1691	A 48 30	F 12101
EN-JL1040	EN-GJL-250 2)	1561	0.6025	GG-25	1691	A 278 30	F 12401
Spheroidal graphite cast iron							
EN-JS1015	EN-GJS-350-22-LT	1563	0.7033	GGG-35.3	1693-1		
EN-JS1020	EN-GJS-400-18	1563	0.7043	GGG-40.3	1693-1	A 395 2)	
EN-JS1030	EN-GJS-400-15	1563	0.7040	GGG-40	1693-1	A 536 60-40-18 1)	
Austenitic cast iron							
			0.6656	GGL-NiCuCr 15 6 3	1694	A 436 1b	
			0.7661	GGG-NiCr 20 3	1694	A 439 D-2B	
Abrasion resistant cast iron							
EN-JN2029	EN-GJN-HV 520	12513	0.9620	G-X260 NiCr 4 2	1695	A 532 IB NiCr-LC	
EN-JN2049	EN-GJN-HV 600	12513	0.9630	G-X300 CrNiSi 9 5 2	1695	A 532 ID Ni-Hi-Cr	
High alloy cast iron (silicium)							
			(023 0) 6)	G-X70 Si 15		A 518 1	
			(024 0) 6)	G-X90 SiCr 15 5		A 518 3	
Low temperature cast steel							
1.5422	G18Mo5	10213-3				A 352 LCB	J 03003
1.5638	G9Ni14	10213-3				A 352 LC3	J 31500
Heat resisting cast steel							
1.0619	GP240GH	10213-2	1.0619	GS-C25	17 245	A 216 WCB	J 03002

1) for internal casings, impellers, guide vanes etc. 2) for casings subject to pressure 6) in house material code

Material number	Chemical analysis (Sample from melt analysis) % by weight						Mechanical properties (guide values)				Density
	C	Cr	Ni	Mo	Cu	Other	R_m N/mm²	$R_{p\,0,2}$ N/mm²	A %	KV J (t°C)	ca. kg/dm³
Grey cast iron (flake)											
EN-JL2030	3.3					Si 2.0	250 - 350	165-228 3)	HB30=120-195 7)		7.20
EN-JL1040	3.3					Si 2.0	250 - 350	165-228 3)			7.20
Spheroidal graphite cast iron											
EN-JS1015	min. 3.0					Si max. 2.5	350	220	22	12 (-40°)	7.1
EN-JS1020	min. 3.0					Si max. 2.5	400	250	18	12 (-20°)	7.1
EN-JS1030	3.6					Si 2.2	400	250	15		7.1
Austenitic cast iron											
0.6656	max. 3.0	2.5-3.5	13.5-17.5		5.5-7.5		190				7.4
0.7661	max. 3.0	2.5-3.5	18.0-22.0				390	210	7		7.4
Abrasion resistant cast iron											
EN-JN2029	2.5-3.0	1.5-3.0	3.0-5.5			Si max. 0.8		Vickers-		HV min. 520	7.7
EN-JN2049	2.5-3.5	8-10	4.5-6.5			Si max. 2.5		hardness		HV min. 600	7.7
High alloy cast iron (silicium)											
(023 0) 6)	0.70				1.0	Si 15.0	120				7.1
(024 0) 6)	0.90	5.0			1.0	Si 15.0	120				7.1
Low temperature cast steel											
1.5422	0.15-0.20			0.45-0.65			440-790	240	23	27 (-45°)	7.85
1.5638	0.06-0.12		3.0-4.0				500-650	360	20	27 (-90°)	7.85
Heat resisting cast steel											
1.0619	0.18-0.23					Mn 0.5-1.2	420-600	240	22	27 (RT)	7.85

1 N/mm² equals 1 Mpa 3) 0.1% elongation limit 6) in house material code 7) for reference wall thickness from 40 to 80 mm

11.2 Material overview - Cast steel (cont.)

European Standard - EN			German Standard - DIN			USA - Standard	
Material number	Short name or description	Standard	Material number	Short name or description	Standard	ASTM Standard, Grade, Type	UNS
Stainless steel							
Martensitic steel							
1.4008	GX7CrNiMo12-1	5)	1.4008	G-X8 CrNi 13	17 445	A 217, 743 CA 15	J 91150
1.4317	GX4CrNi13-4	"				A 743 CA 6NM	J 91540
Austenitic steel							
1.4308	GX5CrNi19-10	5)	1.4308	G-X6 CrNi 18 9	17 445	A 351,743,744 CF 8	
1.4309	GX2CrNi19-11	"				A 351,743,744 CF 3	J 92900
1.4408	GX5CrNiMo19-11-2	"	1.4408	G-X6 CrNiMo 18 10	17 445	A 351,743,744 CF 8M	J 92900
1.4409	GX2CrNiMo19-11-2	"				A 351,743,744CF 3M	J 92800
1.4552	GX5CrNiNb19-11	"	1.4552	G-X5 CrNiNb 18 9	17 445	A 744 CF 8C	
1.4581	GX5CrNiMoNb19-11-2	"	1.4581	G-X5 CrNiMoNb 18 10	17 445		
Full austenitic steel							
1.4458	GX2NiCrMo28-20-2	5)					
(238 0) 6)	GX4NiCrCuMo30-20-4	"				A 743, 744 CN 7M	
(238 1) 6)	GX1NiCrMoCuN25-20-6	"				A 743, 744 CN 3MN	
(238 2) 6)	GX2CrNiMoN20-18-6	"				A743,744 CK 3MCuN	
Austenitic - ferritic (Duplex) steel							
1.4468	GX2CrNiMoN25-6-3	5)	1.4468	G-X3 CrNiMoN 26 6 3	SEW 410		
1.4469	GX2CrNiMoN26-7-4	"	1.4469	G-X2 CrNiMoN 25 7 4	SEW 410	A 890 5A (CE 3MN)	J 93404
1.4470	GX2CrNiMoN22-5-3	"				A 890 4A (CD 3MN)	J 92205
1.4517	GX2CrNiMoCuN25-6-3-3	"	1.4517	G-X3CrNiMoCuN 26 6 3 3	SEW 410	A 890 1B(CD4MCuN)	

5) Draft DIN 17445 published April 1996, recommendation for EN 6) In house material code

Material number	Chemical analysis (Sample from melt analysis) % by weight						Mechanical properties (guide values)				Density
	C	Cr	Ni	Mo	Cu	Other	R_m N/mm²	$R_{p\,0,2}$ N/mm²	A %	KV J (t°C)	ca. kg/dm³
Stainless steel											
Martensitic steel											
1.4008	max. 0.10	12.0-13.5	1.0-2.0	0.20-0.50			590	440	15	27 (RT)	7.7
1.4317	max. 0.06	12.0-13.5	3.5-5.0	max. 0.70			760	550	15	50 (RT)	7.7
Austenitic steel											
1.4308	max. 0.07	18.0-20.0	8.0-11.0				440	175-200 4)	30	60 (RT)	7.88
1.4309	max. 0.030	18.0-20.0	9.0-12.0			Nmax. 0.20	440	185-210 4)	30	80 (RT)	7.88
1.4408	max. 0.07	18.0-20.0	9.0-12.0	2.0-2.5			440	185-210 4)	30	60 (RT)	7.9
1.4409	max. 0.03	18.0-20.0	9.0-12.0	2.0-2.5		Nmax. 0.20	440	195-220 4)	30	80 (RT)	7.9
1.4552	max. 0.07	18.0-20.0	9.0-12.0			Nb 8x%C	440	175-200 4)	25	40 (RT)	7.88
1.4581	max. 0.07	18.0-20.0	9.0-12.0	2.0-2.5		Nb 8x%C	440	185-210 4)	25	40 (RT)	7.9
Full austenitic steel											
1.4458	max. 0.03	19.0-22.0	26.0-30.0	2.0-2.5	max. 2.0	Nmax 0.20	430	165-190 4)	30	60 (RT)	8.0
(238 0) 6)	max. 0.06	19.0-22.0	27.5-30.5	2.0-3.0	3.0-4.0		430	170-195 4)	35	60 (RT)	8.0
(238 1) 6)	max. 0.02	19.0-21.0	24.0-26.0	6.0-7.0	0.5-1.5	N 0.10-0.25	480	210-235 4)	30	60 (RT)	8.0
(238 2) 6)	max.0.025	19.5-20.5	17.5-19.5	6.0-7.0	0.5-1.0	N 0.18-0.24	500	260-285 4)	35	50 (RT)	7.9
Austenitic - ferritic (Duplex) steel											
1.4468	max. 0.03	24.5-26.5	5.5-7.0	2.5-3.5		N 0.12-0.25	650	480	22	50 (RT)	7.7
1.4469	max. 0.03	25.0-27.0	6.0-8.0	3.0-5.0	max. 1.30	N 0.12-0.22	650	480	22	50 (RT)	7.7
1.4470	max. 0.03	21.0-23.0	4.5-6.5	2.5-3.5		N 0.12-0.20	600	420	20	30 (RT)	7.7
1.4517	max. 0.03	24.5-26.5	5.0-7.0	2.5-3.5	2.75-3.5	N 0.12-0.22	650	480	22	50 (RT)	7.7

1 N/mm² equals 1 Mpa 4) 1% elongation limit 6) in house material code

11.2 Material overview - Cast steel (cont.)

European Standard - EN			German Standard - DIN			USA - Standards	
Material number	Short name or description	Standard	Material number	Short name or description	Standard	ASTM Standard, Grade, Type	UNS
Stainless steel for pressure vessels							
Austenitic steel							
1.4308	GX5CrNi19-10	10213-4	1.4308	G-X6 CrNi 18 9	17 445	A 351,743,744 CF 8	
1.4309	GX2CrNi19-11	10213-4				A 351,743,744 CF 3	J 92500
1.4408	GX5CrNiMo19-11-2	10213-4	1.4408	G-X6 CrNiMo 18 10	17 445	A 351,743,744 CF 8M	J 92900
1.4409	GX2CrNiMo19-11-2	10213-4				A 351,743,744 CF 3M	
1.4552	GX5CrNiNb19-11	10213-4	1.4552	G-X5 CrNiNb 18 9	17 445	A 744 CF 8C	
1.4581	GX5CrNiMoNb19-11-2	10213-4	1.4581	G-X5 CrNiMoNb 18 10	17 445		
Full austenitic steel							
1.4458	GX2NiCrMo28-20-2	10213-4				A 351,743,744 CN7M	
Austenitic - ferritic (Duplex) steel							
1.4469	GX2CrNiMoN26-7-4	10213-4	1.4469	G-X2 CrNiMoN 25 7 4	SEW 410	A 890 5A (CE 3MN)	J 93404
1.4470	GX2CrNiMoN22-5-3	10213-4				A 890 4A (CD 3MN)	J 92205
1.4517	GX2CrNiMoCuN25-6-3-3	10213-4	1.4517	G-X3 CrNiMoCuN 26 6 3 3	SEW 410	A 890 1B (CD4MCuN)	
Cast steel for special applications							
			1.4034	G-X46 Cr 13		A 743 CA 40	J 91150
			1.4059	G-X22 CrNi 17	17 445	A 743, 744	J 91803
			1.4138	G-X120 CrMo 29 2	SEW 410	A 743, 744	
			1.4316	G-X2 CrNiSi 18 15			
1.4931	GX23CrMoV12-1	10213-2	1.4391	G-X22 CrMoV 12 1	17 245		
1.7706	G17CrMoV5-10	10213-2	1.7706	GS-17 CrMoV 5 11	17 245		

Material number	Chemical analysis (Sample from melt analysis) % by weight						Mechanical properties (guide values)				Density
	C	Cr	Ni	Mo	Cu	Other	R_m N/mm²	$R_{p\,0,2}$ N/mm²	A %	KV J (t°C)	ca. kg/dm³
Stainless steel for pressure vessels											
Austenitic steel											
1.4308	max. 0.07	18.0-20.0	8.0-11.0				440-640	175,200 4)	30	60 (-196°)	7.88
1.4309	max. 0.03	18.0-20.0	9.0-12.0			Nmax. 0.20	440-640	185,210 4)	30	70 (-196°)	7.88
1.4408	max. 0.07	18.0-20.0	9.0-12.0	2.0-2.5			440-640	185,210 4)	30	60 (-196°)	7.9
1.4409	max. 0.03	18.0-20.0	9.0-12.0	2.0-2.5		Nmax. 0.20	440-640	195,220 4)	30	70 (-196°)	7.9
1.4552	max. 0.07	18.0-20.0	9.0-12.0			Nb 8xC	440-640	175,200 4)	25	40 (RT)	7.88
1.4581	max. 0.07	18.0-20.0	9.0-12.0	2.0-2.5		Nb 8xC	440-640	185,210 4)	25	40 (RT)	7.9
Full austenitic steel											
1.4458	max. 0.03	19.0-22.0	26.0-30.0	2.0-2.5	max. 2.0	Nmax. 0.20	430-630	165,190 4)	30	60 (-196°)	8.0
Austenitic - ferritic (Duplex) steel											
1.4469	max. 0.03	25.0-27.0	6.0-8.0	3.0-5.0	max. 1.30	N 0.12-0.22	650-850	480	22	35 (-70°)	7.7
1.4470	max. 0.03	21.0-23.0	4.5-6.5	2.5-3.5		N 0.12-0.20	600-800	420	20	30 (RT)	7.7
1.4517	max. 0.03	24.5-26.5	5.0-7.0	2.5-3.5	2.75-3.5	N 0.12-0.22	650-850	480	22	35 (-70°)	7.7
Cast steel for special applications											
1.4034	0.45	15.0					Rockwell hardness HRC 55				7,7
1.4059	0.20-0.27	16.0-18.0	1.0-2.0				780-980	590	4		7.7
1.4138	0.90-1.30	27.0-29.0		2.0-2.5							7.7
1.4316	max. 0.03	18.0-19.0	14.0-15.0	max. 0.30	max. 0.30	Si 4.3-4.7	540-700	250	30	55 (RT)	7.9
1.4931	0.20-0.26	11.3-12.2	max. 1.0	1.0-1.2		V 0.25-0.35	740-960	550	15	27 (RT)	7.7
1.7706	0.15-0.20	1.2-1.5		0.9-1.1		V 0.25-0.35	590-780	440	15	27 (RT)	7.8

1 N/mm² equals 1 Mpa   4) 1% elongation limit

11.2 Material overview - Steel

European Standard - EN			German Standard - DIN			USA - Standard	
Material number	Short name or description	Standard	Material number	Short name or description	Standard	ASTM Standard, Grade, Type	UNS
Heat treatable steel							
Stainless steel							
1.1191	C45E	10083-1	1.1191	C 45 E (Ck 45)	17 200	A 29, 108 1045	
1.7225	42CrMo4	10083-1	1.7225	42 CrMo 4	17200	A 322 4140	G 41400
Non-alloy quality steel							
1.0501	C35	10083-2	1.0501	C 35	17 200	A 519, 576 1035	G 10350
1.0503	C45	10083-2	1.0503	C 45	17 200	A 576 1045	G 10450
Stainless steel Martensitic steel							
1.4021	X20Cr13	10088-3	1.4021	X20Cr13	17 440	A 276, 473 420	S 42000
1.4112	X90CrMoV18	10088-3	1.4112	X90 CrMoV 18	SEW 400		
1.4122	X39CrMo17-1	10088-3	1.4122	X35 CrMo 17	SEW 400		
1.4313	X3CrNiMo13-4	10088-3	1.4313	X4 CrNi 13 4	SEW 400	A 182 F6NM	
Austenitic steel							
1.4306	X2CrNi19-11	10088-3	1.4306	X2CrNi19-11	17 440	A 276, 403 304L	S 30403
1.4404	X2CrNiMo17-12-2	10088-3	1.4404	X2CrNiMo17-12-2	17 440	A 182, 276, 403 316L	S 31603
1.4541	X6CrNiTi18-10	10088-3	1.4541	X6CrNiTi18-10	17 440	A 182, 276, 403 321	S 32100
1.4571	X6CrNiMoTi17-12-2	10088-3	1.4571	X6CrNiMoTi17-12-2	17 440	A 182, 276, 403 316Ti	
Austenitic-ferritic (Duplex) steel							
1.4462	X2CrNiMoN22-5-3	10088-3	1.4462	X2 CrNiMoN 22 5 3	SEW 400	A 182 F 51 A 276 318 LN	S 31803
1.4507	X2CrNiMoCuN25-6-3	10088-3					

Material number	Chemical analysis (Sample from melt analysis) % by weight						Mechanical properties (guide values)				Density
	C	Cr	Ni	Mo	Cu	Other	R_m N/mm²	$R_{p\,0,2}$ N/mm²	A %	KV J (t°C)	ca. kg/dm³
Heat treatable steel											
Stainless steel											
1.1191	0.42-0.50	max. 0.40	max. 0.40	max. 0.10			630-780	370	17	25 (RT)	7.85
1.7225	0.38-0.45	0.90-1.20		0.15-0.30			900-1100	650	12	35 (RT)	7.85
Non-alloy quality steel											
1.0501	0.32-0.39						550-700	320	20		7.85
1.0503	0.42-0.50						630-780	370	17		7.85
Stainless steel Martensitic steel											
1.4021	0.16-0.25	12.0-14.0					700-850	500	13	25 (RT)	7.7
1.4112	0.85-0.95	17.0-19.0		0.90-1.30		V 0.07-0.12					7.7
1.4122	0.33-0.45	15.5-17.5	max. 1.0	0.80-1.30			750-950	550	12	14 (RT)	7.7
1.4313	max. 0.05	12.0-14.0	3.5-4.5	0.30-0.70		N 0.020	900-1100	800	12	50 (RT)	7.7
Austenitic steel											
1.4306	max. 0.03	18.0-20.0	10.0-12.0			Nmax. 0.11	460-680	180- 215[4)]	45	100 (RT)	7.9
1.4404	max. 0.03	16.5-18.5	10.0-13.0	2.0-2.5		Nmax. 0.11	500-700	200- 235[4)]	40	100 (RT)	7.98
1.4541	max. 0.08	17.0-19.0	9.0-12.0			Ti 5xC	500-700	190- 225[4)]	40	100 (RT)	7.9
1.4571	max. 0.08	16.5-18.5	10.5-13.5	2.0-2.5		Ti 5xC	500-700	200- 235[4)]	40	100 (RT)	7.98
Austenitic - ferritic (Duplex) steel											
1.4462	max. 0.03	21.0-23.0	4.5-6.5	2.5-3.5		N 0.10-0.22	650-880	450	25	100 (RT)	7.8
1.4507	max. 0.03	24.0-26.0	5.5-7.5	2.7-4.0	1.0-2.5	N 0.15-0.30	700-900	500	25	100 (RT)	7.8

1 N/mm² equals 1 Mpa   4) 1% - elongation limit

11.2 Material overview - Non-ferrous metals

European Standard - EN			German Standard - DIN			USA - Standard	
Material number	Short name or description	Standard	Material number	Short name or description	Standard	ASTM Standard, Grade, Type	UNS
Copper, tin, zinc alloy (cast tin bronze and gunmetal)							
CC480K	CuSn10-Cu	1982	2.1050	G-CuSn10	1705	B 427 C 91600	C 91600
CC491K	CuSn5Zn5Pb5-C	1982	2.1096	G-CuSn5ZnPb	1705	B 584 C 92200	C 92200
Copper, tin, alloy (special brass)							
			2.0550	CuZn40Al 2	17 660		
Copper, aluminium alloy (Aluminium bronze)							
			2.0966	CuAl 10Ni5Fe4	17 665	B 150 C 63000	C 63000
CC333G	CuAl10Fe5Ni5-C	1982	2.0975	G-CuAl 10Ni	1714		
Nickel alloy with molybdenum and chrome							
			2.4610	NiMo16Cr16Ti	17 744	B 574 N 06455	N 06455
			2.4617	NiMo28	17 744	B 335 N 10665	N 10665
			2 4685	G-NiMo 28		A 494 N-12MV	
			2.4686	G-NiMo 17Cr		A 494 CW-12MW	
			2.4819	NiMo16Cr15W	17 744	B 574 N 10276	N 10276
Titanium and Zirconium							
			3.7031	G-Ti 2	17 865	B 367 Gr. C-2	
			3.7255	Ti 3 Pd	17 851/60	B 265 Gr. 7	
			(852 0) [6]	G-Zr-Nb		B 752 Gr. 705C	
			(853 0) [6]	Zr-Nb		B 551 R 60705	R 60705

6) In house material code

Material number	Composition, alloy components, permitted levels in brackets % by weight							Mechanical properties R_m N/mm²	$R_{p\,0,2}$ N/mm²	A %	Density ca. kg/dm³
Copper, tin, zinc, alloy (cast tin bronze and gunmetal)											
	Cu	Sn	Zn	Pb	Al	Fe	Ni				
CC480K	88.0-90.0	11.0	max. 0.5	max. 1.0	max. 0.01	max. 0.2	max. 2.0	250	130	18	8.7
CC491K	83.0-87.0	4.0-6.0	4.0-6.0	4.0-6.0	max. 0.01	max. 0.3	max. 2.0	200	90	13	8.7
Copper, tin, alloy (special brass)											
2.0550	56.5-59.0	max. 0.5	30.3-40.5	max. 0.8	1.3-2.3	max. 1.0	max. 2.0	510	230	12	8.1
Copper, aluminium alloy (Aluminium bronze)											
2.0966	75.7-85.5				8.5-11.0	2.0-5.0	4.0-6.0	640	270	15	7.5
CC333G	76.0-83.0	max. 0.1	max. 0.50	max. 0.03	8.5-10.5	4.0-5.5	4.0-6.0	600	250	13	7.6
Nickel alloy with molybdenum and chrome											
	Ni	Cr	Mo	W	Fe	Mn	C				
2.4610	56.8-72.0	14.0-18.0	14.0-18.0		max. 3.0	max. 1.0	max. 0.01	700	280	35	8.6
2.4617	64.5-74.0	max. 1.0	26.0-30.0		max. 2.0	max. 1.0	max. 0.01	745	325	40	9.2
2.4685	min. 62.0	max. 1.0	26.0-30.0		max. 7.0	max. 1.0	max. 0.03	530-650	275	15	9.1
2.4686	58.7-72.0	14.0-18.0	14.0-17.0		max. 3.0	max. 1.0	max. 0.02	540	250	12	8.5
2.4819	50.7-63.5	14.5-16.5	15.0-17.0	3.0-4.5	4.0-7.0	max. 1.0	max. 0.015	700	280	35	8.9
Titanium und Zirconium											
	Ti	Pd	Zr + Hf	Hf	Nb	Fe	O				
3.7031	98.8-99.9					max. 0.2		350	280	15	4.5
3.7255	98.7-99.8	0.15-0.25				max. 0.25		460-590	320	18	4.5
(852 0) [6)]			95.6-99.9	max. 4.5	2.0-3.0		max. 0.3	483	345	12	6.5
(853 0) [6)]			95.5-99.9	max. 4.5	2.0-3.0		max. 0.18				

1 N/mm² equals 1 MPa 6) in house material code

11.3 US materials with AISI designation

AISI	Equivalent material		
	Material No.	Short name	Standard
Austenitic steel			
301	1.4310	X10CrNi18-8	EN 10088-3
304	1.4301	X5CrNi18-10	"
304 L	1.4306	X2CrNi19-11	"
304 LN	1.4311	X2CrNiN18-10	"
305	1.4303	X4CrNi18-12	"
308	1.4303	X4CrNi18-12	"
316	1.4401	X5CrNiMo17-12-2	"
"	1.4436	X3CrNiMo17-13-3	"
316 Cb	1.4580	X6CrNiMoNb17-12-2	"
316 L	1.4404	X2CrNiMo17-12-2	"
316 LN	1.4406	X2CrNiMoN17-11-2	"
"	1.4429	X2CrNiMoN17-13-3	"
316 Ti	1.4571	X6CrNiMoTi17-12-2	"
317 L	1.4438	X2CrNiMo18-15-4	"
321	1.4541	X6CrNiTi18-10	"
347	1.4550	X6CrNiNNb18-10	"
Austenitic ferritic steel			
329	1.4460	X3CrNiMoN27-5-2	EN 10088-3
Ferritic and martensitic steel			
403	1.4000	X6Cr13	DIN 17440
405	1.4002	X6CrAl13	"
410	1.4006	X12Cr13	EN 10088-3
420	1.4021	X20Cr13	"
430	1.4016	X6Cr17	DIN 17440
430 F	1.4104	X14CrMoS17	EN 10088-3
430 Ti	1.4510	X3CrTi17	DIN 17440
431	1.4057	X17CrNi16-2	EN 10088-3
440 C	1.4125	X105CrMo17	SEW 400
Heat treatable steel			
C 1035	1.0501	C35	EN 10083-2
C 1045	1.0503	C45	"
4140	1.7225	42CrMo4	EN 10083-1

11.4 US materials according to ASTM standards

ASTM	Comparible standards	
	Standard	Material group
A 29	EN 10083-1	Heat treatable steel
48	EN 1561	Grey cast iron
108	EN 10083-1	Heat treatable steel
182	EN 10088-3	Stainless steel
216	EN 10213-2	Heat resisting steel
217	DIN 17 445	Cast stainless steel, martensitic
276	EN 10088-3	Stainless steel
278	EN 1561	Grey cast iron
322	EN 10083-1	Heat treatable steel
351	EN 10213-4	Cast stainless steel, austenitic
352	EN 10213-3	Low temperature cast steel
395	EN 1563	Spheroidal graphite cast iron
403	EN 10088-3	Stainless steel
436	DIN 1694	Austenitic grey cast iron
439	DIN 1694	Austenitic spheroidal graphite cast iron
473	EN 10088-3	Stainless steel
494	none	Cast nickel alloy with molybdenum and chrome
519	EN 10083-2	Heat treatable steel
532	EN 12513	Abrasion resistant cast irons
536	EN 1563	Spheroidal graphite cast iron
576	EN 10083-1, -2	Heat treatable steel
743	DIN 17 445	Cast stainless steel,
744	DIN 17 445	Cast stainless steel, austenitic
890	SEW 410	Cast stainless steel (Duplex)
B 150	DIN 17 665	Copper, aluminium alloy
265	DIN 17 851 / 60	Titanium alloy sheet and roll
335	DIN 17 744	Nickel alloy with molybdenum
367	DIN 17 865	Cast titanium
427	EN 1982	Copper, tin, zinc alloy
494	see A 494	
574	DIN 17 744	Nickel alloy with molybdenum and chrome
584	EN 1982	Copper,tin,zinc alloy

11.5 Materials by trade name

Trade name	Comparable material		
	Material-number	Short name	Standard
Austenitic cast iron			
Ni-Resist 1b	0.6656 –	GGL-NiCuCr 15 6 3 Type 1b	DIN 1694 ASTM A 436
Ni.Resist D-2B	0.7661 –	GGG-NiCr 20 3 Type D-2B	DIN 1694 ASTM A 439
Abrasion resistant cast iron			
Ni-Hard 2	EN-JN2029 –	EN-GJN-HV520 Class I Type B Ni-Cr-LC	EN12513 ASTM A 532
Ni-Hard 4	EN-JN2049 –	EN-GJN-HV600 Class I Type D Ni-Hi-Cr	DIN 1695 ASTM A 532
Cast stainless steel			
Alloy 20	1.4458 –	GX2NiCrCuMo28-20-2 CN 7M	EN 10213-4 ASTM A 351 A 744
Sterling K 26	see Alloy 20		
Sterling R 48	1.4517 – – –	GX2CrNiMoCuN25-6-3-3 CD 4MCu Grade 1A (ACI CD 4MCu) Grade 1B (ACI CD 4MCuN)	EN 10213-4 ASTM A 351 A 744 ASTM A 890 ASTM A 890
Nickel alloy with molybdenum and chrome			
Hastelloy B-1	2.4685 –	G NiMo28 N 12 MV	EN/DIN none ASTM A 494
Hastelloy B-2	2.4617 –	NiMo28 N 10665	DIN 17 744 ASTM B 335
Hastelloy C-1	2.4686 –	G-NiMo 17Cr CW 12 MW	EN/DIN none ASTM A 494
Hastelloy C-276	2.4819 –	NiMo16Cr15W N 10276	DIN 17 744 ASTM A574
Sterling R 52	see Hastelloy B-1		
Sterling R 53	see Hastelloy C-1		
Sterling R 55	No comparable material Analysis: Ni 58%, Cr 22%, Cu 4%, Si 4% and W 1%		

11.6 Organic materials overview

Abbreviation ISO 1043 ISO 1629	Abbreviation EN 12756 (Mech seals)	Name	Trade name
Elastomers			
CR	N	Chlorobutadiene elastomer old name: Chloroprene rubber	Baypren (Bayer) Neoprene (Du Pont)
EPDM	E	Ethylene propylene rubber	Ketan (DSM) Nordel (Du Pont)
FPM also FCM CFM, FKM	V	Fluorocarbon elastomer	Viton (Du Pont)
NBR HNBR	P X	Acrylonitrile butadiene elastomer old name nitrile rubber Hydrogenated NBR	Perbunan (Bayer) Therban, Zetpol
None	K	Perfluoroelastomer	Kalrez (Du Pont)
Plastics			
Thermoplastic			
PTFE	T	Polytetrafluoroethylene	Teflon (Du Pont)
PE PE - LD PE - HD PE-UHMW		Polyethylene Polyethylene low density ρ = 0.918to 0.95 g/cm³ Polyethylene high density ρ = 0.95 to 0.96 g/cm³ Ultra high molecular PE	 Hostalen (Hoechst) Lupolen (BASF) Hostalen GUR (Hoechst) RCH 1000
PEEK		Polyether-etherketone	Victrex PEEK (ICI)
PFA		Perfluoralkoxyalkane	Hostaflon PFA (Hoechst) Teflon PFA (Du Pont)
PP		Polypropylene	Hostalen PP (Hoechst) Novolen (BASF)
PPS		Polyphenylene-sulphide	Ryton (Phillips)
PVDF		Polyvinylidenefluoride	Diflor 2000 (Hüls) Kynar (Elf)
Thermosetting			
EP		Epoxide resin	Araldit (Ciba Geigy)
VE		Vinyl ester resin	
Natural rubber			
NR		Natural rubber - soft rubber. hot vulcanised rubber with 1.8 to 2.5% sulphur hardness (25 to 100) Shore A - hard rubber. hot vulcanised with 30 to 50% sulphur	 Ebonit

11.6 Organic materials properties

Abbreviation ISO 1043 ISO 1629	EN 12756 (Mech seals)	Usable temperature range Chemical resistance Application
Elastomers		
CR	N	Usable temperature range – 40 to + 100 °C for special compounds: – 55 to + 150 °C Chemically resistant against (e.g.): cold water, seawater, phosphoric acid Not resistant against (e.g.): kerosene, benzene, nitric acid, sulphuric acid, acetic acid Application: O-rings, bellows
EPDM	E	Usable temperature range: – 50 °C to + 120 °C for special hot water compounds: to + 200 °C Chemically resistant against (e.g.): cold water, hot water, seawater, caustic soda, phosphoric acid, hydrochloric acid, acetic acid (10%, <60 °C) Not resistant against (e.g.): petrol and diesel fuel, kerosene, benzene, liquified gases, mineral oil, nitric acid, sulphuric acid Application: O-rings, bellows
FPM	V	Usable temperature range: – 25 °C to + 200 °C for special compounds: – 45 °C to + 260 °C Chemically resistant against (e.g.): effluent (pH>3 <10), cold water, hot water, seawater, petrol and diesel fuel, kerosene, liquified gases, mineral oil, phosphoric acid, sulphuric acid Not resistant against (e.g.): nitric acid, acetic acid, caustic soda, benzene Application: O-rings, bellows
NBR HNBR	P X	Usable temperature range: – 30 to + 100 °C Usable temperature range: – 40 to + 150 °C Chemically resistant against (e.g.): effluent (pH>6<10), cold water, hot water, seawater, petrol and diesel fuel, kerosene, liquified gases, mineral oil, caustic soda (>50%), phosphoric acid (cold concentrated), HNBR is hydrolysis resistant Not resistant against (e.g.): benzene, nitric acid, sulphuic acid, acetic acid Application: O-rings, bellows, membranes
None	K	Usable temperature range: to + 260 °C Chemically resistant as PTFE Elastic properties as FPM

11.6 Organic materials properties (cont)

Abbreviation ISO 1043 ISO 1629	EN 12756 (Mech seals)	Usable temperature range Chemical resistance Application
Thermoplastic		
PTFE	T M_1 M_2 Y1 / Y2	Usable temperature range: – 200 to + 260 °C Chemically resistant against practically all chemicals and water Application: pure PTFE:- for corrosion resistant coating - for encapsulation of O-rings FPM, double PTFE encapsulation EPDM, double PTFE encapsulation PTFE, glass fibre reinforced and PTFE, carbon fibre reinforced: - for pump impellers - for shaped seals and bellows
Thermoplastic		
PE PE-LD PE-HD PE-UHMW		Usable temperature range: – 50 to + 80 °C – 50 to + 90 °C Chemically resistant against (e.g.): alkali, salt solutions, inorganic acids (reducing and weakly oxidising), organic acids, esters, ketones Not resistant against (e.g.): aliphatic and aromatic hydrocarbons and chlorinated hydrocarbons Application: pump casings and impellers for plastic pumps Usable temperature range: – 30 to + 80 °C Chemically resistant against strongly abrasive and corrosive liquids Application: pump casings and impellers for plastic pumps
PEEK		Usable temperature range: – 40 to + 160 °C Chemically resistant against (e.g.): nearly all inorganic and organic chemicals Not resistant against (e.g.): hydrochloric acid, fuming nitric acid, concentrated sulphuric acid Application: nearly always reinforced, e.g. carbon fibre reinforced PEEK-CF30 for impellers of side channel pumps
PFA		Usable temperature range: – 100 to + 180 °C Chemical resistance as PTFE Application: corrosion resistant coatings and linings

11.6 Organic materials properties (cont)

Abbreviation ISO 1043	ISO 1629	Usable temperature range / Chemical resistance / Application
PP		Usable temperature range: 0 to + 90 °C Chemically resistant against e.g.: aqueous solutions of inorganic salts, weak inorganic acids and alkalis Not resistant against (e.g.): strongly oxidising acids, aliphatic and aromatic hydrocarbons and halogenated hydrocarbons Application: pump casings and impellers for plastic pumps
PPS		Usable temperature range: – 40 to + 200 °C Chemically resistant against (e.g.): concentrated caustic soda, concentrated hydrochloric and sulphuric acid, dilute nitric acid and solvents, to + 180°C Not resistant against (e.g.): concentrated nitric acid Application: nearly always glass fibre reinforced, e.g. PPS-GF40 parts for plastic pumps
PVDF		Usable temperature range: – 20 to + 120 °C Chemically resistant against (e.g.): dissolved salts, alkalis, acids, aliphatic and aromatic hydrocarbons and chlorinated hydrocarbons Not resistant against (e.g.): oleum, high temperature ketones, esters, organic amines Application: pump casings and impellers for plastic pumps
Thermosetting		
EP		Usable temperature range: + 10 to + 80 °C Chemically resistant against (e.g.): dilute acids and alkalis, chlorinated hydrocarbons, toluene Not resistant against (e.g.): concentrated acids and alkali, ammonia, esters, ketones, acetone Application: parts for plastic pumps
VE		Usable temperature range: – 40 to + 120 °C Chemically resistant against (e.g.): dilute acids and alkalis and bleaches Not resistant against (e.g.): concentrated acids and alkalis, hydrofluoric acid, chromic acid, ammonia, solvents Application: glass fibre reinforced, parts for plastic pumps

11.6 Organic materials properties (cont)

Abbreviation ISO 1043	ISO 1629	Usable temperature range Chemical resistance Application
Natural rubber		
NR		Usable temperature range: soft rubber: – 40 to + 65 °C hard rubber: – 40 to + 80 °C Chemically resistant against (e.g.): acids, alkali, hot water Not resistant against (e.g.): fuels, mineral oils, solvents Application: soft rubber: (Shore hardness 40 to 60) wear resistant coating and lining of solids pumps Conditional on: solids are not sharp edged and particle size < 6,4 mm maximum tip speed 27 m/s hard rubber: corrosion resistant coating and lining of pumps and pump parts

11.7 Fibre reinforced materials

Fibre reinforcement improves the mechanical and thermal properties of plastics. With PTFE for example, it will prevent plastic cold forming (pre-forming) under the influence of load at room temperature.

The common reinforcing materials are carbon, glass and synthetic fibres.

Fibre reinforced plastics are designated according to DIN 7728 Pt 2 according to the type of fibre, as follows:

Abbreviation	Material
CFK	Carbon fibre reinforced plastic
GFK	Glass fibre reinforced plastic
SFK	Synthetic fibre reinforced plastic

The material can be defined in more detail according to DIN 7728 Pt 1 by including in the abbreviation, the proportion of reinforcing material

Example:

PTFE-GF 30 → PTFE with 30% glass fibre reinforcement
PPS-GF 40 → Polyphenylene sulphide with 40% glass fibre reinforcement

11.8 Ceramic materials

Material	Composition	Application
Silica ceramic materials		
Porcelain	Al_2O_3 30 to 35% SiO_2 rest	chemically resistant pump parts (not suitable for hydrofluoric acid)
Glass Borax aluminium glass	SiO_2 65.3% Al_2O_3 3.5% B_2O_3 15.0% BaO. ZnO rest	alkali free instrument glass for chemically resistant pump housings (not suitable for hydrofluoric acid)
Oxide ceramic materials		
Aluminium oxide (corundum) Al_2O_3 type Al_2O_3-SiO_2 type Zirconium dioxide	 Al_2O_3 99.7% Al_2O_3 96 to 97.5% ZrO_2 > 90% Y_2O_3. MgO rest	chemically resistant pump parts, mechanical seal faces for pressures <25 bar (not suitable for hydrofluoric acid) Mechanical seal code: V Chemically resistant shroud material for magnetic coupling pumps
None-oxide ceramic materials		
Carbon resin impregnated e.g. with phenolic resin metal impregnated e.g. with antimony	C > 99.7%	Bearing sleeves, mechanical seal faces good chemical resistance, installation limit 150 °C, pressure to 25 bar Mechanical seal code: B Chemical resistance not as good as resin impregnated, but suitable for higher temperatures Mechanical seal code: A
Silicon carbide S-SiC, sintered (pressure free) SiSiC, sintered (reaction)	 SiC > 98% SiC > 90% + Si	Bearing sleeves and journalss, lubricated by pumped liquid, mechanical seal faces Mechanical seal code: Q

11.9 Material selection

After the mechanical properties, the corrosion resistance of the material to the pumped liquid is the determining criterion in the selection.

The following table gives guidelines to the groups of materials which have proven suitable in practice. It must be remembered however that the corrosion resistance and hence the material selection can be greatly affected by the temperature, concentration, impurity content, abrasive solids and the flow velocity and distribution of the liquid.

11.9.1 Tips for notes 1) to 8) and material selection table

1. Pressure retaining pump parts made of grey cast iron (flake) are not permitted in chemical plants for handling dangerous media.
2. Spheroidal graphite cast iron may only be used in quality EN-GJS-400-18 (previous designation GGG 40.3) with a maximum perlite content 5%.
3. Material groups in brackets, e.g. (D) have restricted application.
4. There is pitting danger with this aqueous solution. Preference should be given to materials with high molybdenum content in material groups H and K.
5. In the presence of water, hydrochloric acid (HCl) will form with the danger of pitting, crevice and stress corrosion.
6. If material group D is selected, then only spheroidal graphite cast iron is suitable.
7. The data is only valid for pure phosphoric acid produced by the dry thermal process. For acid produced by wet process, a full analysis of the acid must be made for material selection.
8. When pumping nitric acid, molybdenum free materials with high silicon content have an advantage (material number 1.4306 and 1.4361).

Fp = Flow or melting point at standard pressure 101,325 Pa.

Bp = Boiling point at standard pressure 101,325 Pa. For other reference pressures, the figure is given as /hPa.

Warning! ***<u>General Danger Note</u>*** **Warning!**

Some of the liquids in the material tables are extremely corrosive, poisonous, on inhalation and skin contact irritant, flammable, environmentally dangerous and carcinogenic. If not handled correctly they may also cause considerable damage to the pump and plant. Furthermore there are health and injury dangers to persons and possible loss of life.

Table 11.01 Compilation of materials in groups for the selection table 11.9

Material group	Material	Standard
D	Grey cast iron (flake) 1)	EN 1561 DIN 1691 ASTM A48 and 278
	Spheroidal graphite or nodular cast iron (ductile iron) 2)	EN 1563 DIN 1693 ASTM A395 and 536
	Cast steel	EN 10213-2 DIN 17 245 ASTM A216
H	Austenitic stainless steel with molybdenum	EN 10213-4 DIN 17 445, E DIN 17445 ASTM A351, 743 and 744
K	Full austenitic stainless steel (Alloy 20 type)	EN 10213-4 E DIN 17445 ASTM A743 and 744
L	Austenitic ferritic stainless steels, Duplex	EN 10213-4 E DIN 17445 and SEW 410 ASTM A890
M	Nickel alloy with molybdenum (Hastelloy B type)	EN none DIN 17 744 ASTM A494 and B335
N	Nickel alloy with molybdenum and chrome (Hastelloy C type)	EN none DIN 17 744 ASTM B574
T	Titanium and titanium alloys	EN none DIN 17851,17860 und17865 ASTM B265 and B367
U	Zirconium and zirconium alloys	EN and DIN none ASTM B551 and B752
W	High alloy cast iron (cast silicium)	EN and DIN none ASTM A518
Z	Hard rubber coating	none

1) and 2) see previous page

11.9 Materials of Construction For Pumping Various Liquids

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
A							
Acetaldehyde	to 100	20	CH_3CHO	HKL MN TU		– 123.5	20.5
Acetic Acid	< 5	20	CH_3COOH	(D) HKL MN TU			
	to 70	to boiling		HKL MN TU			
Acetic Acid Glacial	100	to 100		HKL MN TU		16.7	118
	100	boiling		MN TU		16.7	118
Acetic Anhydride		20	$(CH_3CO)_2O$	HKL MN TU		– 74.2	138.9
Acetone	to 100	40	CH_3COCH_3	D HKL MN TU		– 95	56.3
Acetyl Chloride		20	CH_3COCl	HKL MN U	5)	– 112	51
Allyl Alcohol	to 100	20	$CH_2{=}CHCH_2OH$	D HKL MN TU		– 129	97
Allyl Chloride		20	$CH_2{=}CHCH_2Cl$	KL MN TU	5)	– 136.4	45.7
Alum	sat'd	to 90	$KAl(SO_4)_2 \cdot 12\,H_2O$	(H)KL MN			
Aluminium Chloride	25	20	$AlCl_3$	KL M U	4)		
	40	20		M U	4)		
Aluminium Nitrate	30	20	$Al(NO_3)_3 \cdot 9\,H_2O$	HKL MN T			
Aluminium Sulphate	sat'd	20 to 60	$Al_2(SO_4)_3$	HKL MN TU			
	10	to boiling		KL TU			
	sat'd	to boiling		KL			

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Ammonia		20	NH_3	D HKL MN U		– 77.7	– 33.4
Ammonia, Aqueous Sol.	10	20	NH_4OH	D HKL MN TU			
	30	20		D HKL MN TU			
Ammonium Acetate	10	90	$NH_4(C_2H_3O_2)$	HKL M			
Ammonium Bicarbonate	20	20	NH_4HCO_3	D HKL			
Ammonium Difluoride	sat'd	20	NH_4HF_2	KL	4)		
Ammonium Bromide	5	50	NH_4Br	HKL MN U Z	4)		
Ammonium Carbonate	20	20	$(NH_4)_2CO_3$	HKL			
Ammonium Chloride	sat'd	50	NH_4Cl	HKL MN TU	4)		
	sat'd	to boiling		KL MN TU	4)		
Ammonium Nitrate	all	to boiling	NH_4NO_3	HKL MN TU			
Ammonium Sulphate	10	80	$(NH_4)_2SO_4$	HKL MN TU			
	40	80		KL MN TU			
Aniline		to 100	$C_6H_5NH_2$	HKL N TU		– 6.2	184.4
Anisole		20	$C_6H_5OCH_3$	D HKL MN TU		– 37.2	153.8
Arsenic Acid	to 70	65	$H_3AsO_4 \cdot \frac{1}{2} H_2O$	KL MN U			
B							
Barium Chloride	cold sat'd	20	$BaCl_2$	HKL MN TU	4)		
	cold sat'd	100		KL MN TU	4)		

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Barium Hydroxide	all	20	$Ba(OH)_2$	D HKL MN TU			
Barium Nitrate	all	to boiling	$Ba(NO_3)_2$	HKL MN TU			
Benzaldehyde		20	C_6H_5CHO	HKL MN U		– 56	178.9
Benzene		20	C_6H_6	D HKL MN TU		5.5	80.2
Benzoic Acid	1	to boiling	C_6H_5COOH	HKL MN TU			
Benzyl Alcohol		20	$C_6H_5CH_2OH$	HKL MN U		– 15.3	205.4
Benzyl Chloride		20	$C_6H_5CH_2Cl$	HKL MN	5)	– 39.2	179.4
Borax	sat'd	to boiling	$Na_2B_4O_7 \cdot 10\ H_2O$	HKL MN TU			
Boric Acid	4	to boiling	H_3BO_3	(H)KL MN TU			
Bromine	dry	20	Br	MN U		– 8.3	58.8
Butadiene (1,3-B.)		20	$CH_2{=}CHCH{=}CH_2$	HKL M		– 108.9	– 4.5
Butanol (n-B.)		20	C_4H_9OH	(D) HKL MN TU		– 89.3	117.8
Butanone		20	$CH_3CH_2COCH_3$	HKL MN TU		– 86.7	79.7
Butene (1-B.)		20	$C_2H_5CH{=}CH_2$	HKL MN TU		– 185.4	– 6.3
Butyl Acetate (n-B.)		20	$CH_3COOC_4H_9$	(D) HKL MN TU		– 76.8	126.5
Butylene Glycol	to 100	40	$HO(CH_2)_4OH$	(D) HKL MN TU		19	235
Butyric Acid	100	100		KL N TU			
Butyric Acid (n-B.)	100	20	$CH_3(CH_2)_2COOH$	HKL MN TU		– 5.3	163.3

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
C							
Cadmium Chloride	all	20	$CdCl_2$	HKL MN	4)		
Calcium Bisulphite	25	20	$Ca(HSO_3)_2$	HKL N TU			
	25	boiling		KL TU			
Calcium Chloride	all	20	$CaCl_2$	D HKL MN TU Z	4)		
	all	boiling		KL N TU			
Calcium Hydroxide	0.15	20	$Ca(OH)_2$	D HKL MN T			
	15	20		D HKL MN T			
	30	100		N T			
Calcium Hypochlorite	to 3	30	$Ca(ClO)_2$	HKL N TU Z	4)		
	25	60		N TU Z	4)		
Calcium Nitrate	all	20	$Ca(NO_3)_2$	(H)KL MN T			
Carbolic Acid	10% H_2O	to 90		HKL U			
Carbon Disulphide		20	CS_2	D HKL MN TU		– 111.9	46.3
Carbon Tetrachloride	dry	20	CCl_4	D HKL	5)	– 23.2	76.6
	wet	20		L N TU	4)		
Carbonic Acid		20	H_2CO_3	(D) HKL MN TU			
Caustic Soda	*see Sodium Hydroxide*						
Chlorobenzene (Mono-C.)		20	C_6H_5Cl	(D) HKL MN TU	5)	– 45.6	131.8

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Chloric Acid	20	18	$HClO_3$	N			
Chlorine	dry	20	Cl_2	(D) HKL		– 101	– 34.5
	wet	20		KL N T Z	5)		
Chlorine Water	1 g/l Cl_2	20	$Cl_2 + H_2O$	HKL N Z	4)		
	sat'd	20		N Z			
Chloroacetic Acid (Mono-C.)	50	20	$CH_2ClCOOH$	MN TU W			
	50	90		MN TU (W)			
Chloroform		to 50	$CHCl_3$	(D) HKL MN TU	5)	– 63.5	61.3
Chlorosulphuric Acid	100	20	HSO_3Cl	H MN T	5)	– 80	152
	10	20		MN T	4)		
Chrome Alum	sat'd	to 55	$KCr(SO_4)_2 \cdot 12\ H_2O$	(H)KL W			
	sat'd	90		KL W			
Chromic Acid	10	to boiling	H_2CrO_4	K TU W			
	30	100		TU W			
	47	60		N W			
	75	20		K N			
	75	93		(N)			
	with SO_3	20		K W			
Citric Acid	1	20	$C_6H_8O_7$	HKL MN TU			

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
	10	20		HKL MN TU			
	10	100		HKL MN TU			
	30	20		HKL MN TU			
	30	50		KL MN TU			
	50	20		HKL MN TU			
	50	boiling		K MN U			
Copper Chloride	20	20	$CuCl_2$	N T Z	4)		
	20	80		T Z	4)		
Copper Sulphate	sat'd	to boiling	$CuSO_4$	(H)KL N TU			
Cresol (m-C.)		> 40	$CH_3C_6H_4OH$	(D) HKL MN TU		12.3	202.3
Cumene		20	$C_6H_5CH(CH_3)_2$	HKL MN TU		– 99.5	159.3
Cyclohexane		20	C_6H_{12}	D HKL MN TU		6.6	80.8
Cyclohexanol		50	$C_6H_{11}OH$	(D) HKL MN TU		24.9	161.2
Cyclohexanone		20	$C_6H_{10}O$	(D) HKL MN TU		– 30	155.7
D E							
Dibutyl Ether		20	$(C_4H_9)_2O$	D HKL MN TU		– 95	141
Diethyl Ether		20	$(C_2H_5)_2O$	(D) HKL MN TU		– 116.3	34.6
Diethylene Glycol	to 100	20	$C_4H_{10}O_3$	(D) HKL MN TU		– 8	245

Liquid	Conditions of liquid *w* %	*t* °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Dioxane (1.4-D.)	to 100	30	$C_4H_8O_2$	D HKL MN TU		11.1	101.4
Ethanol	to 100	to boiling	C_2H_5OH	(D) HKL MN TU		– 114.1	78.4
Ethyl Acetate		20	$CH_3COOC_2H_5$	HKL MN TU		– 83.6	77.2
Ethyl Chloride		20	CH_3CH_2Cl	HKL MN TU	5)	– 136.4	12.3
Ethyl Formate		20	$HCOOC_2H_5$	(D) HKL MN TU		– 79.4	54.3
Ethylbenzene		20	$C_6H_5C_2H_5$	D HKL MN TU		– 95	136.2
Ethylene Chlorhydrin		20	$ClCH_2CH_2OH$	HKL MN TU	5)	– 70	129.5
Ethylene Dichloride		to boiling	CH_2ClCH_2Cl	HKL MN TU	5)	–35.7	83.5
Ethylene Glycol	bis 100	20	CH_2OHCH_2OH	(D) HKL MN TU		– 13	197.3
F G							
Fatty Acids	*see Oleic Acid and Stearic Acid*						
Fatty Alcohols	Octadecanol		$C_{18}H_{37}OH$	HKL MN TU		58	$210/_{20}$
Ferric Chloride	15	25	$FeCl_3$	N T Z	4)		
	45	25		N T Z	4)		
Ferric Sulphate	30	20	$Fe_2(SO_4)_3$	HKL N TU			
Ferrous Chloride	all	40	$FeCl_2$	(K)(L) MN TU Z	4)		
Ferrous Nitrate	all	to boiling	$Fe(NO_3)_2 \cdot 6\ H_2O$	HKL N T			
Ferrous Sulphate	10	20	$FeSO_4$	(H)KL MN TU			
Formaldehyde	40	to boiling	CH_2O	HKL MN TU	4)		

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Formamide	to 100	20	CH_3NO	(D) HKL MN TU		2.5	210.5
Formic Acid	100	20	HCOOH	HKL MN TU		8.4	100.7
	50	70		(K)L N U			
	100	90		K(L) U		8.4	100.7
Furfural		20	$C_5H_4O_2$	(D) (H)KL MN TU		– 36.5	161.8
Gallic Acid	hot sat'd	to boiling	$C_6H_2(OH)_3COOH$	HK N			
Glaubers Salt	*see Sodium Sulphate*						
Glycerol	to 100	30	$C_3H_5(OH)_3$	(D) HKL MN TU		17.9	289.9
H							
Hexane		20	C_6H_{14}	(D) HKL MN TU		– 95.4	68.8
Hydrobromic Acid	50	20	HBr	M Z	4)		
Hydrochloric Acid	0.5	20	HCl	K MN TU Z	4)		100
	0.5	80		MN TU Z	4)		100
	5	20		K MN TU W Z	4)		101
	5	80		M TU W Z	4)		101
	10	20		MN TU W Z	4)		103
	10	80		M U W Z	4)		103
	20	20		MN TU W Z	4)		108
	20	80		M U W Z	4)		108

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
	30	20		MN U W Z	4)		90
	30	80		M U Z	4)		90
	37	20		M U Z	4)		60
	37	boiling		M U Z	4)		60
Hydrocyanic Acid	dil.	20	HCN	(D) HKL MN TU			
Hydrofluoric Acid	< 40	20	HF	MN Z			
	> 60	20		(D) M			
Hydrofluorosilicic Acid	sat'd	20	H_2SiF_6	(H)KL N			
Hydrogen Peroxide	30	20	H_2O_2	HKL MN TU			
L M							
Lactic Acid	< 5	to 100	$CH_3CH(OH)COOH$	KL N TU W			
	25	20		HKL MN TU W			
	50	20		HKL N TU W			
	50	80		HKL N TU W			
	80	20		HKL MN TU W			
	80	120		KL T			
	100	25		HKL MN TU W		18	$119/_{16}$
Lead Acetate	20	to boiling	$Pb(C_2H_3O_2)_2 \cdot 3\ H_2O$	HKL MN TU			
Lithium Bromide	sat'd	20	$LiBr$	HKL N	4)		

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Lithium Chloride	10	to 80	$LiCl$	KL MN TU	4)		
Magnesium Chloride	all	20	$MgCl_2$	HKL MN TU W	4)		
	< 5	to boiling		KL MN TU W	4)		
	> 5	to boiling		KL N TU W	4)		
Magnesium Sulphate	sat'd	to boiling	$MgSO_4$	(H)KL TU			
Maleic Acid	50	20	HOOCCH=CHCOOH	HKL MN TU			
Manganese Chloride	20	to 100	$MnCl_2$	K N TU W Z	4)		
Manganese Sulphate	30	20	$MnSO_4$	(D) HKL MN TU			
Mercuric Chloride	0.1	20	$HgCl_2$	HKL N TU	4)		
	0.7	20		KL N TU			
Mercuric Nitrate	5	20	$Hg_2(NO_3)_2 \cdot 2\,H_2O$	HKL N			
Methanol	bis 100	to 80	CH_3OH	(D) HKL MN TU		– 97.7	64.7
Methyl Acetate		25	CH_3COOCH_3	HKL MN		– 98.2	57
Methylene Chloride		20	CH_2Cl_2	(D) HKL MN T	5)	– 96	39.9
Methylpentanone		20	$CH_3COCH_2CH(CH_3)_2$	HKL MN TU		– 80	114.5
Mine Water	*see Chapter 10 „Water" Table 10.07*						
Mixed Acids							
50% H_2SO_4 + 50% HNO_3		50		HK N T			85
		85		K			85

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
75% H_2SO_4 + 25% HNO_3		50		(D) K N W			105
		90		K W			105
20% H_2SO_4 + 15% HNO_3		50		HK			110
		80		K			110
70% H_2SO_4 + 10% HNO_3		50		(D) HK W			113
		90		K W			113
30% H_2SO_4 + 5% HNO_3		90		HK			110
		110		HK			110
15% H_2SO_4 + 5% HNO_3		100		K			107
2% H_2SO_4 + 1% HNO_3		100		HK			100
N O							
Nickel Chloride	to 30	20	$NiCl_2$	KL MN TU Z	4)		
Nickel Nitrate	to 10	20	$Ni(NO_3)_2 \cdot 6\,H_2O$	HKL TU			
Nickel Sulphate	all	20	$NiSO_4$	HKL N TU			
Nitric Acid 8)	10	20	HNO_3	1.4309 N TU		– 12	100
	10	60		1.4309 N TU		– 12	100
	37	20		1.4309 N TU		– 35	111
	37	boiling		1.4361 U W		– 35	111

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
	53	20		1.4309 N TU		– 24	118
	53	boiling		1.4361 U W		– 24	118
	68	20		1.4361 N TU		– 34	121.5
	68	boiling		1.4361 TU W		– 34	121.5
	99	20		1.4361 TU W		– 43	88
	99	boiling		1.4361 T W		– 43	88
Nitrobenzene		20	$C_6H_5NO_2$	D HKL MN T		5.7	210.9
Oleic Acid		150	$C_{18}H_{34}O_2$	HKL MN		13.2	$260/_{53}$
		180		KL MN		13.2	$260/_{53}$
		300		K N		13.2	$260/_{53}$
Oleum	11% free SO_3	20	$H_2SO_4 + SO_3$	(D) HKL MN	6)		167
	11% free SO_3	100		KL MN			167
Oleum	60% free SO_3	20		HKL			66
	60% free SO_3	60		KL			66
Oxalic Acid	5	20	HOOC-COOH	HKL MN TU			
	5	100		HKL MN U			
	10	20		HKL MN TU			
	10	100		HKL N U			
	25	to 100		HKL N U			

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
	50	to 100		KL N			
P							
Phenol	dry	45	C_6H_5OH	D HKL MN U		40.9	181.9
	dry	100		HKL MN U		40.9	181.9
Phosphoric Acid 7)	< 10	20	H_3PO_4	HKL MN TU			
	10	100		HKL MN U			102
	50	70		HKL MN U			112
	50	110		KL MN U			112
	75	20		HKL MN U			135
	75	80		HKL MN			135
	75	135		L			135
	85	20		HKL MN			154
	85	90		HKL			154
Phthalic Acid	7	85	$C_6H_4(COOH)_2$	KL TU			
Phthalic Anhydride		180	$C_6H_4(CO)_2O$	HKL MN		131	285
Potassium Bromide	sat'd	20	KBr	HKL MN TU	4)		
Potassium Carbonate	20	to 80	K_2CO_3	(D) HKL MN TU			
Potassium Chlorate	sat'd	20	$KClO_3$	(D) HKL N TU	4)		
Potassium Chloride	all	to 80	KCl	(H)KL MN TU Z	4)		

Liquid	Conditions of liquid *w* %	*t* °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Potassium Cyanide	sat'd	20	KCN	(D) HKL MN			
Potassium Dichromate	sat'd	20	$K_2Cr_2O_7$	D HKL N TU			
Potassium Ferricyanide	10	to 80	$K_3[Fe(CN)_6]$	HKL MN TU			
Potassium Ferrocyanide	10	to 80	$K_4[Fe(CN)_6] \cdot 3\ H_2O$	(H)KL MN U			
Potassium Hydroxide	to 50	to 80	KOH	(D) (H)KL U			
Potassium Hypochlorite	13% free Cl_2	20	KClO	N T Z	4)		
Potassium Iodide	10	20	KI	(H)KL MN TU Z	4)		
Potassium Nitrate	sat'd	20	KNO_3	HKL N T			
Potassium Permanganate	sat'd	to boiling	$KMnO_4$	(D) HKL			
Potassium Sulphate	sat'd	20	K_2SO_4	HKL MN TU			
Propionic Acid	to 100	20	CH_3CH_2COOH	HKL T		– 20.7	140.9
Pyridine	to 100	20	C_5H_5N	D HKL MN TU		– 41.7	115.4
S							
Silver Nitrate	all	20	$AgNO_3$	HKL N TU			
Soda Ash	*see Sodium Carbonate*						
Sodium Bicarbonate	to 8	to boiling	$NaHCO_3$	(D) HKL MN TU			
Sodium Bisulphate	15	35	$NaHSO_4 \cdot H_2O$	KL MN TU			
Sodium Bisulphite	38	to boiling	$NaHSO_3$	HKL N T			
Sodium Bromide	all	20	NaBr	(D) (H)KL MN TU	4)		

Liquid	Conditions of liquid *w* %	*t* °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Sodium Carbonate	all	20	Na_2CO_3	(D) HKL MN TU			
Sodium Chlorate	all	20	$NaClO_3$	(H)KL MN TU W	4)		
Sodium Chloride	< 3	20	$NaCl$	(D) HKL MN TU	4)		
	> 3 to 26	20		HKL MN TU	4)		
	> 3 to 28	120		KL MN TU	4)		
Sodium Chlorite	30	20	$NaClO_2$	KL N T W	4)		
	+ 2% NaOH	60		(H)KL N T W	4)		
Sodium Cyanide	sat'd	20	$NaCN$	(H)KL N T			
Sodium Dichromate	all	20	$Na_2Cr_2O_7$	D HKL N U			
Sodium Hydroxide	20	20	$NaOH$	D HKL MN TU			
	50	50		(D) HKL MN TU			
	50	90		KL MN TU			
Sodium Hypochlorite	20 g/l Cl_2	40	$NaOCl$	(H)(K) N T W Z	4)		
	120 g/l Cl_2	20		T W Z	4)		
Sodium Nitrate	sat'd	to boiling	$NaNO_3$	(D) HKL N TU			
Sodium Phosphates							
monobasic	to 10	25	$NaH_2PO_4 \cdot H_2O$	HKL N			
dibasic	to 6	20	$Na_2HPO_4 \cdot 2\,H_2O$	(D) HKL MN TU			
tribasic	to 10	to 100	$Na_3PO_4 \cdot 12\,H_2O$	(D) HKL MN TU			

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Sodium Silicates	all	20	$Na_2O \cdot x\ SiO_2$	(D) HKL MN TU			
Sodium Sulphate	cold sat'd	20	$Na_2SO_4 \cdot 10\ H_2O$	HKL MN TU			
	cold sat'd	100		HKL MN TU			
Sodium Sulphide	20	20	Na_2S	HK M TU			
	20	boiling		K M TU			
Sodium Sulphite	2	20	Na_2SO_3	HK MN TU			
	+ free SO_2	20		KL MN TU			
Sodium Thiosulphate	30	to 80	$Na_2S_2O_3$	HKL MN TU			
Stannic Chloride	24	20	$SnCl_4$	N TU W Z			
Stannous Chloride	sat'd	50	$SnCl_2$	HL N TU W Z	4)		
	sat'd	boiling		N TU W	4)		
Stearic Acid		150	$C_{18}H_{36}O_2$	HKL MN		70	$184/_2$
		250		K N		70	$184/_2$
Sulphur monochloride		20	S_2Cl_2	(D) HKL N T	5)	– 76	138.1
Sulphuric Acid	5	20	H_2SO_4	HKL MN TU			101
	5	100		K MN TU W			101
	10	20		HKL MN TU W			102
Sulphuric Acid	10	100	H_2SO_4	TU			102
	20	20		KL MN TU W			104

Liquid	Conditions of liquid w %	t °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
	20	100		U			104
	50	20		KL MN TU W			122
	50	120		M U			122
	70	20		KL MN TU W			164
	70	80		M U W			164
	70	160		U W			164
	85	20		(D) HKL MN T W			230
	85	50		MN W			230
	85	80		M W			230
	85	150		W			230
	98	20		D HKL MN T W		10	327
	98	50		HKL MN W		10	327
	98	80		MN W		10	327
	98	150		W		10	327
Sulphurous Acid		20	H_2SO_3	KL MN TU			
T							
Tetrachloroethen		20	C_2Cl_4	(D) HKL MN TU		– 22.2	121.2
Tetrahydrofuran	to 100	20	C_4H_8O	HKL MN TU		– 108.5	65.5
Titanium Tetrachloride		20	$TiCl_4$	N TU		– 24.3	136.5

Liquid	Conditions of liquid *w* %	*t* °C	Formula	Material selection 3)	Refer to	Fp °C	Bp °C
Toluene		20	$C_6H_5CH_3$	(D) HKL N T		– 95.2	110.7
Trichloroethene		20	$CHCl{=}CCl_2$	D HKL	5)	– 86.5	87.3
		80		HKL MN TU	5)	– 86.5	87.3
Tricresyl Phosphate	pure	80	$(C_6H_4CH_3)_3PO_4$	HKL N T		– 25	410
Triethylamine	to 100	20	$(C_2H_5)_3N$	HKL		– 114.8	89.6
U V W							
Urea	all	25	CH_4N_2O	(D) (H)KL TU			
Vinyl Acetate		20	$CH_3COOCH{=}CH_2$	HKL N		– 100	72
Vinyl Chloride		20	$CH_2{=}CHCl$	(D) HKL N T		– 153.8	– 13.4
Water	*see Chapter 10 "Water" Table 10.07*						
X Z							
Xylene (m-X.)		20	$C_6H_4(CH_3)_2$	D HKL N T		– 47.9	139.2
Zinc Chloride	sat'd	20	$ZnCl_2$	HKL MN TU W Z	4)		
	sat'd	45		K MN TU W Z	4)		
	sat'd	boiling		K U W			
Zinc Sulphate	30	20	$ZnSO_4$	HKL N T			

12 Dimensional units

12.1 General

In most European countries, the use of “statutory units” is established in law. In all commercial and official transactions the legal units must be used. For countries within the Common Market these units are largely harmonised. For commercial and official transactions with countries outside these limits, other units may be used.

For the European Union the SI units (International System of Units) is used.

The basic units for the system were defined at the General Conference for Weights and Measures.

Table 12.01 SI Basic units

Measure	Name	Symbol
Length	metre	m
Mass	kilogramme	kg
Time	second	s
Electrical current	Ampere	A
Thermodynamic temperature	Kelvin	K
Amount of substance	mol	mol
Luminous intensity	Candela	cd

The derived SI units are coherent i.e. without using any numerical factor differing from 1, are derived from the basic unit. The derived unit may be named after the basic unit or may have a special name

Units outside the SI system can be used providing they are published in the appropriate country standard.

Decimal fractions and multiples of units are prefixed with a symbol as shown in Table 12.02.

The prefix and prefix symbol is only used in conjunction with the unit.

Table 12.02 Prefix and prefix symbol for decimal fractions and multiples of units

Prefix	Symbol	Factor	Name of Factor (European)
Yokto	y	10^{-24}	Quadrillionth
Zepto	z	10^{-21}	Trilliardth
Atto	a	10^{-18}	Trillionth
Femto	f	10^{-15}	Billiardth
Pico	p	10^{-12}	Billionth
Nano	n	10^{-9}	Milliardth
Micro	µ	10^{-6}	Millionth
Milli	m	10^{-3}	Thousandth
Centi	c	10^{-2}	Hundredth
Deci	d	10^{-1}	Tenth
Deca	da	10^{1}	Ten
Hecto	h	10^{2}	Hundred
Kilo	k	10^{3}	Thousand Tsd.
Mega	M	10^{6}	Million Mio.
Giga	G	10^{9}	Milliard Mrd. 1)
Tera	T	10^{12}	Billion Bio. 1)
Peta	P	10^{15}	Billiard
Exa	E	10^{18}	Trillion
Zetta	Z	10^{21}	Trilliard
Yotta	Y	10^{24}	Quadrillion

1) in the USA: 10^{9} = 1 Billion, 10^{12} = 1 Trillion

The prefix symbol is placed immediately in front of the basic unit symbol with no space. The prefix symbol and the basic unit symbol form the symbol of a new unit. An exponent of the basic unit is also classed as a prefix.

Prefixes may only be used singly.

12.2 Dimensional Units and Conversions

Contents

12.2.1 Length

SI basic unit name: metre symbol: m in formulae: l

Definition: the metre is the distance which light travels in a vacuum in 1/299,792,458 of a second. (17th.General Conference, 1983)

1 m = 10 dm = 100 cm = 1000 mm

1 µm = 10^{-3} mm

1 km = 1000 m

In air and sea travel, the international unit is:

1 international nautical mile (sm) = 1852 m

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 inch (in) = 1´´ = 25.40 mm

1 thousandth (thou) = 0.0254 mm

1 mikroinch (µin) = 0.0254 µm

1 foot (ft) = 1´ = 12´´ = 0.3048 m

1 yard (yd) = 3´ = 36´´ = 0.9144 m

1 mile (statute mile) = 1609.34 m

1 n mile (nautical mile) = 1852 m

12.2.2 Area

Derived SI unit name: square metre symbol: m² in formulae: A

1 m² = 100 dm² = 10^4 cm² = 10^6 mm²

For the area of land and large floor space the unit is:

1 Ar (a) = 100 m²

1 Hectare (ha) = 100 a = 10^4 m²

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 square inch (sq in, in²) = 6.4516 cm²

1 square foot (sq ft, ft²) = 0.0929 m²

1 square yard (sq yd, yd²) = 0.8361 m²

1 acre = 4840 yd² = 4046.86 m² = 40.4686 a

12.2.3 Volume

Derived SI unit name: cubic metre symbol: m³ in formulae: V

1 m³ = 1000 dm³ = 10^6 cm³

Commonly used unit outside of SI name: litre symbol: l or L

1 l = 1 dm³ = 1000 cm³ = 100 cl = 1000 ml = 1 L 1 hl = 100 L

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 cubic inch (cu in, in^3) = 16.3871 cm^3

1 cubic yard (cu yd, yd^3) = 0.76456 m^3

Measurement of liquids:

United Kingdom (UK)	United States (US)
1 fluid ounce (fl oz) = 0.028413 L	1 fluid ounce (fl oz) = 0.029574 L
1 gill = 5 fl oz = 0.14207 L	1 gill = 4 fl oz = 0.11829 L
1 pint (pt) = 4 gills = 0.56826 L	1 liquid pint (liq pt) = 4 gills = 0.47318 L
1 quart (qt) = 2 pt = 1.13652 L	1 liquid quart (liq qt) = 2 liq pt = 0.94635 L
1 gallon (gal) = 4 qt = 4.5461 L	1 gallon (gal) = 4 liq qt = 3.7854 L
1 barrel (bbl) = 36 gal = 163.6 L	1 liquid barrel (liq bbl) = 119.24 L
	1 barrel crude oil = 42 gal = 159 L
1 L = 0.220 Imp.gal	1 L = 0.2642 US gal

To differentiate these liquid measures, the UK units are referred to as (Imperial) Imp. and the USA units as US, (e.g. Imp. gal or US gal).

12.2.4 Mass

SI basic unit name: kilogramme symbol: kg in formulae: *m*

Definition: the kilogramme is the unit of mass and is equal to the mass of the International kilogramme prototype. (1st. General Conference 1889) and (3rd. General Conference 1901)

Commonly used unit outside of SI name: gramme symbol: g
name: tonne symbol: t

1 kg = 1000 g 1 g = 10^{-3} kg = 1000 mg = 10^6 µg

The SI basic unit kilogramme (kg) is not used with prefixes as more than one is not allowed.

1 t = 1000 kg = 1 Mg 1 dt = 100 kg

Units of mass are also used commercially as the “weight” (wt) of goods.

The Imperial units in the United Kingdom (UK) and the United States (USA):

Avoirdupois abbreviated: av (commercial use)

1 grain (gr) = 64.799 mg	1 ounce (oz) = 16 dram = 28.3495 g
1 dram = 1.77184 g	1 pound (lb) = 16 oz = 0.45359 kg
	1 kg = 2.205 lb

To differentiate from the unit of force (see 12.2.14) if necessary the suffix “m” can be added (e.g. lbm).

United Kingdom (UK)	United States (USA)
1 hundredweight (cwt)	1 hundredweight (cwt)
or long cwt (cwt l) = 50.8023 kg	or short cwt (cwt sh) = 45.3592 kg
1 ton (tn)	1 ton (tn)
or long ton (tn l) = 1.01605 t	short ton (tn sh)=2000 lb=0.90718 t
	1 long ton (tn l)=2240 lb=1.01604 t

12.2.5 Amount of substance

SI basic unit name: molar mass symbol: mol in formulae: *n*

Definition: the molar mass is the amount of substance in a system (e.g. the portion of a chemical molecule), which has as many individual particles as the atoms contained in 0.012 kilogrammes of the carbon nucleus ^{12}C. In using the molar mass, the particles must be specified and may be atoms, molecules, ions, electrons and other particles or groups of such particles of exactly specified composition (14th. General Conference, 1971)

The molar mass is a property not a realisable unit. It exists alongside the mass and volume and cannot be measured, but must be calculated from the other properties.

$1\ \mu mol = 10^{-6}\ mol$ $1\ mmol = 10^{-3}\ mol$

12.2.5.1 Molar proportion

Derived SI unit symbol: mol/mol in formulae: *x*

The substance proportion of a component A is the quotient of the molar mass of substance $n(A)$ and total mixture n.

$$x(A) = \frac{n(A)}{n}$$

As the numerator and denominator have the same unit, the result is purely numerical, but always < 1.

12.2.5.2 Molar concentration

Derived SI unit symbol: mol/m³ in formulae: *c*

$1\ mol/m^3 = 1\ mmol/l$

The substance concentration is the quotient of the molar mass of the dissolved component (A) and the volume of the solution (L).

$$c(\text{A}) = \frac{n(\text{A})}{V(\text{L})}$$

The data on substance concentration is mainly given for aqueous solutions of ionic substances.

12.2.5.3 Mass proportion

Derived SI unit symbol: kg/kg in formulae: *w*

Common units:

g/g

g/100 g = mass proportion in %

mg/g = mass proportion in ‰

μg/g = mass proportion in ppm (parts per million)

ng/g = mass proportion in ppb (parts per billion)

ppm = parts per million = parts per 10^6 parts

ppb = parts per billion = parts per 10^9 parts

The mass proportion of a component (A) is the quotient of the mass of this component and the mass of the total mixture (*m*).

$$w(\text{A}) = \frac{m(\text{A})}{m}$$

The expression mass proportion in % is used for the composition of an aqueous solution, for the solids content of a liquid and the composition of a metallic alloy.

12.2.5.4 Mass concentration

Derived SI unit symbol: kg/m³ in formulae: β

1 kg/m³ = 1000 g/m³ = 1 g/l

1 g/m³ = 1000 mg/m³

The mass concentration is the quotient of the amount of the dissolved substance (A) and the volume of the solution (L).

$$\beta(\text{A}) = \frac{m(\text{A})}{V(\text{L})}$$

The expression mass concentration is e.g. used for data on the gas or vapour content of a liquid or gas volume.

12.2.6 Density

Derived SI unit symbol: kg/m³ in formulae: ρ

1 kg/dm³ = 1 kg/L = 1000 kg/m³

The density is the quotient of the mass and the volume of a body.

$\rho = m/V$

The density of a body is influenced by many factors, e.g. the chemical composition, the physical state and especially for liquids and gases, the temperature and pressure.

Unless otherwise indicated, density is given at an ambient temperature of 20°C and atmospheric pressure of 1.013 bar.

For mineral oil products, the density is often quoted at a temperature of 15°C.

The Imperial units in the United Kingdom (UK) and the United States (USA):

Specific Weight in formulae: *W*

1 pound per cubic foot (lb/ft³) = 0.01602 kg/dm³

1 pound per gallon (lb/UK gal) = 0.09978 kg/dm³

(lb/US gal) = 0.1198 kg/dm³

12.2.7 Relative density

Unit: 1 in formulae: *d*

The relative density is the ratio of the density ρ of a substance or mixture to the density ρ_0 of a reference substance under conditions defined for both.

$d = \rho / \rho_0$

Relative density is primarily used for gases. The reference mostly used is dry air under standard conditions $\rho_L = 1.2930$ kg/m³.

In the United Kingdom (UK) and the United States (USA) this is referred to as "specific gravity" (sp.gr.).

For liquids the reference is water at 60°F (15.56°C) with $\rho = 0.9991$ kg/dm³. Other reference temperatures are 39.2°F (4°C, maximum density of water $\rho_{Water} = 1.000$ kg/dm³) and 68 °F (20 °C, $\rho_{Water} = 0.9983$ kg/dm³).

12.2.8 Time

SI basic unit name: second symbol: s in formulae: *t*

The second is the duration of 9,192,631,770 periods of the radiation corresponding to the transition between the two hyperfine levels of the ground state of the atom of caesium ^{133}Cs (13th. General Conference, 1967).

The duration which the definition assigns to the caesium radiation was carefully chosen to make it impossible, by any existing experimental evidence, to distinguish the second from the old unit based on the earth's motion, although according to relativity theory, its magnitude may be dependent on the movement of the observer.

The term time is also used as duration or period between two events.

1 minute (min) = 60 s 1 hour (h) = 60 min = 3600 s

1 day (d) = 24 h = 1440 min = 86,400 s

1 year (a) = 365 d or 366 d = 8,760 h or 8,784 h

The units min, h, d and a are not used with prefixes.

1 millisecond (ms) = 10^{-3} s   1 microsecond (µs) = 10^{-6} s

1 nanosecond (ns) = 10^{-9} s

12.2.9 Velocity

Derived SI unit   symbol: m/s   in formulae: v

Velocity is the quotient of the distance travelled s in uniform motion to the time taken t, i.e. $v = s/t$.

1 m/s = 3.6 km/h   1 km/h = 1 / 3.6 m/s = 0.2778 m/s

In air and sea travel, the international unit is:

1 knot (kn) = 1 nautical mile per hour (sm/h) = 1.852 km/h

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 foot per second (ft/s) = 0.3048 m/s

1 mile per hour (mph) = 1.60934 km/h

12.2.9.1 Velocities with special names

Stream velocity

Generally the mean axial stream velocity also known as velocity of flow is given.

Unit: m/s   in formulae: U

The stream velocity is the quotient of the flowrate at the considered point to the cross sectional area.

$$U = \frac{Q}{A}$$ in m/s   with Q in m³/s and A in m²

With the flowrate Q in m³/h and the nominal diameter DN in mm the velocity of flow can be calculated from:

$$U = (18.8/\mathrm{DN})^2$$

For Imperial and USA units the following apply:

$U = (0.7005/\text{NPS})^2$ in ft/s

with NPS (nominal pipe size) in inches, Q in Imp.gpm

$U = (0.6391/\text{NPS})^2$ in ft/s

with NPS in inches, Q in US gpm

Circumferential speed

Unit: m/s in formulae: u

The circumferential speed is the speed of a point which describes a circle, e.g. a point on the blade of an impeller.

The tip speed of an impeller is:

$$u = \frac{D \cdot \pi \cdot n}{1000 \cdot 60}$$ in m/s with D = impeller Ø in mm and n = speed in rpm

Sliding speed

Unit: m/s in formulae: v_g

The sliding speed is the circumferential speed of the sliding face of a mechanical seal or protective shaft sleeve, relative to the stationary face of the seal or the stuffing box packing.

$$v_g = \frac{d \cdot \pi \cdot n}{1000 \cdot 60}$$ in m/s with d = outer Ø of the shaft sleeve in mm and n = speed in rpm

The sliding speed of a mechanical seal is given for the mean of the face Ø d_m.

$$d_m = \frac{D + d}{2}$$ in mm with D = outer Ø of the sliding face in mm and d = inner Ø of the sliding face in mm

12.2.10 Frequency

Derived SI unit unit: Hz (Hertz) in formulae: f

The frequency is the quotient of the number of repeats of the same operation to the time taken.

$f = 1/t$ 1 Hz = 1/s

the figure 2π times the frequency is the circular or angular frequency ω.

$\omega = 2\pi \cdot f$.

In the United Kingdom (UK) and the United States (USA):

cycles per second: cps or c/s   1cps = 1 c/s = 1 Hz

12.2.11 Rotational speed

The rotational speed is the rotational frequency and for a uniformly rotating body is the quotient of the number of rotations to the time taken.

Unit: 1/s or s^{-1} in formulae: n

also 1/min or min^{-1}

$1\ s^{-1} = 60\ 1/min$

In the United Kingdom (UK) and the United States (USA):

- revolutions per second: r/s or rps = 1 s^{-1}
- revolutions per minute: r/min or rpm = 1/min

The relationship between the speed n and the angular velocity ω is

$$\omega = 2\pi \cdot n$$

12.2.12 Flowrate (Volumetric flow)

Derived SI unit   unit: m^3/s in formulae: Q

For a uniform flow, the volumetric flowrate is the quotient of the pumped volume V to the time taken t, i.e. $Q = V/t$

Units outside of SI and conversion factors

	m^3/h	L/s	L/min	m^3/s
1 m^3/h	1	0.278	16.67	$2.778 \cdot 10^{-4}$
1 L/s	3.6	1	60	$1 \cdot 10^{-3}$
1 L/min	0.06	0.01667	1	$1.667 \cdot 10^{-5}$

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 gallon per minute (gal/min or gpm)

Imp. gpm or UK gpm = 0.07577 L/s = 0.2728 m^3/h

US gpm = 0.06309 L/s = 0.2271 m^3/h

1 cubic foot per second (cu ft/sec or cusec or ft^3/sec) = 28.32 L/s = 101.9 m^3/h

1 m^3/h = 3.665 Imp.gal = 4.403 Usgal

1 m^3/s = 35.329 cu ft/sec

12.2.13 Mass flow

Derived SI unit unit: kg/s in formulae: q

For a uniform flow, the mass flowrate is the quotient of the pumped mass m to the time taken t, i.e. $q = m/t$.

1 t/h = 0.2778 kg/s

The relationship between mass flowrate q and volumetric flowrate Q is:

$q = \rho \cdot Q$ with ρ = density of the pumped media

Conversion of mass flowrate q to volumetric flowrate Q

$$Q\,[\mathrm{m^3/h}] = \frac{3.6 \cdot q\,[\mathrm{kg/s}]}{\rho\,[\mathrm{kg/dm^3}]} \qquad \text{or } Q\,[\mathrm{m^3/h}] = \frac{q\,[\mathrm{t/h}]}{\rho\,[\mathrm{kg/dm^3}]}$$

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 pound per second (lb per s or lb/s) = 0.4536 kg/s = 1.633 t/h

Conversion of mass flowrate q (lb/s) to volumetric flowrate Q (gpm).

$$Q\,[\text{UK gpm}] = \frac{6.0 \cdot q\,[\mathrm{lb/s}]}{\text{sp gr}} \qquad \text{or } Q\,[\text{US gpm}] = \frac{7.2 \cdot q\,[\mathrm{lb/s}]}{\text{sp gr}}$$

12.2.14 Force

Derived SI unit with special name and symbol

Name: Newton symbol: N in formulae: F

$$1\ \mathrm{N} = 1\,\frac{\mathrm{J}}{\mathrm{m}} = 1\,\frac{\mathrm{kg \cdot m}}{\mathrm{s^2}}$$

Force is the cause of movement of free bodies or the deformation of fixed bodies.

1 Newton is the force which will accelerate a mass m = 1 kg with a = 1 m/s².

$F = m \cdot a$

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 pound force (lbf) = 4.44822 N

1 poundal (pdl) = 0.138255 N

1 poundal is the force which will accelerate a mass m = 1 lb with a = 1 ft/s².

A special case is:

Gravitational force

Unit: N in formulae: F_G or G

Gravitational force is the product of mass m and the acceleration due to gravity g.

$$F_G = m \cdot g$$

In most cases the value of g is 9.81 m/s².

For an exact calculation, the local value can be calculated from:

$$g = 9.7803(1 + 0.0053 \cdot \sin^2 \varphi) - 3 \cdot 10^{-6} \cdot h \qquad \text{in m/s}^2$$

with φ = geographic latitude in °

and h = geographic altitude = height above sea level in m

12.2.15 Torque, moment

Derived SI unit unit: N·m or Nm in formulae: M

$$1\ \text{Nm} = \frac{\text{kg} \cdot \text{m}}{\text{s}^2}$$

Torque is a measure of the force required to produce rotation of a rigid body. The value of the torque is the product of the force F and the distance r at right angles of its line of action from the point of rotation i.e. $M = F \cdot r$.

The required torque for a pump at a defined operating point and the available torque of the motor are calculated as follows:

$$M = 9549 \cdot \frac{P}{n} \quad \text{in Nm} \qquad \text{with } P \text{ in kW,} \qquad n \text{ in rpm}$$

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 lb.ft. = 1.356 Nm

1 lb.ft. = a force F = 1lb applied at a distance r of 1 ft

Calculation of the torque: $$M = 5{,}250 \cdot \frac{\text{hp}}{\text{rpm}} \qquad \text{in lb.ft.}$$

with hp = horse power and rpm = revolutions per minute

12.2.16 Stress

The application of external forces on an elastic body induces internal reaction forces which try to reverse the changes in the shape of the body.

The stress is expressed as the reaction force per unit area.

Unit: MPa other unit: 1 N/mm² = 1 MPa

Tensile stress	$\sigma = F/A$	in N/mm² = MPa
Compression stress	$\tau = F/A$	in N/mm² = MPa

with the reaction force F in N and cross sectional area A in mm²

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 psi = 0.006894760 MPa psi = pound per square inch

1000 psi = 1 ksi = 6.894760 MPa ksi = kilopound per square inch

1 N/mm² = 1 Mpa = 145.04 psi

12.2.17 Pressure

Derived SI unit with special name and symbol

Name: Pascal symbol: Pa in formulae: p

$$1 \text{ Pa} = 1 \frac{\text{N}}{\text{m}^2} = 1 \frac{\text{kg}}{\text{m} \cdot \text{s}^2}$$

Pressure is the quotient of a force F acting at right angles on a surface to the area A of the surface, i.e. $p = F / A$

1 Pa = 1000 mPa = 10^6 µPa = 1 N/m²

1 hPa = 100 Pa 1 kPa = 1000 Pa 1 MPa = 10^6 Pa = 1 N/mm²

Commonly used unit outside of SI: name: bar symbol: bar

It has proved sensible to have the unit bar roughly equivalent to the atmospheric pressure.

1 bar = 0.1 MPa = 10^5 Pa

1 bar = 1000 mbar = 10^6 µbar 1 mbar = 1 hPa 1µbar = 0.1 Pa

1 MPa = 10 bar

Conversion factors for various units of pressure

	bar	mbar	Pa	hPa	MPa
bar	1	1000	10^5	1000	0.1
mbar	10^{-3}	1	100	1	10^{-4}
Pa	10^{-5}	0.01	1	0.01	10^{-6}
hPa	10^{-3}	1	100	1	10^{-4}
Mpa	10	10^4	10^6	10^4	1

Pressure data as defined by DIN 1314 edition 02/77 "Pressure, basics and units" as follows:

- absolute pressure p_{abs}
- pressure difference $\Delta p = p_1 - p_2$
- pressure difference, as single measurement $p_{1,2}$
- atmospheric pressure p_{amb}
- atmospheric pressure difference, $p_e = p_{abs} - p_{amb}$

The **atmospheric pressure** or **air pressure**, is the pressure exerted by the atmosphere as a result of its gravitational weight. The mean air pressure at sea level is 1013.2 hPa = 1.0132 bar. With weather changes this can vary between 930 and 1070 hPa. At higher altitudes the air pressure reduces (see section 13, table 13.01).

The **absolute pressure** is the pressure compared to that in a total vacuum.

According to DIN EN 12723 for centrifugal pumps, only atmospheric pressure p_{amb} and the vapour pressure p_v are given as absolute pressure. All other pressures are given as over pressure (over p_{amb}).

In vacuum technology absolute pressure is always used.

The difference between two pressures is designated **pressure difference**, or when it is a single measure, **differential pressure**, e.g. across an orifice plate. The stage pressures in a multistage pump or the pressure loss in a fitting are pressure differences.

The difference between an absolute pressure p_{abs} and the local (absolute) atmospheric pressure p_{amb} is the **atmospheric pressure difference** $p_e = p_{abs} - p_{amb}$, according to DIN 1314 this is called **over pressure**.

The over-pressure takes a positive value when the absolute pressure is higher than atmospheric pressure.

The over-pressure takes a negative value when the absolute pressure is lower than atmospheric pressure.

Consequently as according to DIN EN 12723 for centrifugal pumps, only atmospheric pressure and the vapour pressure are given as absolute pressure, the use of the word "over" is superfluous and the index "e" can also be omitted.

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 pound force per square inch (psi, lbf/in²) = 6894.76 Pa = 0.0689 bar

1 pound force per square foot (psf, lbf/ft²) = 47.8803 Pa = 0.4788 mbar

1 poundal per square foot (pdl/ ft²) = 1.48816 Pa = 14.8816 µbar

Atmospheric pressure: 14.696 psi = 2116.22 psf = 1.013 bar

Absolute pressure and over pressure are designated:

- Absolute pressure: ...a , ...A e.g. psia, PSIA
- Over pressure: ....g , ...G e.g. psig, PSIG (g or G gauge = manometer)

12.2.18 Work, Energy

Derived SI unit with special name and symbol

Name: Joule symbol: J in formulae: *E* or *W*

$$1\ \mathrm{J} = 1\ \mathrm{N} \cdot \mathrm{m} = 1\ \mathrm{W} \cdot \mathrm{s} = 1\ \frac{\mathrm{m}^2 \cdot \mathrm{kg}}{\mathrm{s}^2}$$

Mechanical work is done when a body is moved by a force acting on it. If the force vector F and the movement vector s have the same direction, and the force is constant over the entire movement, then $E = F \cdot s$

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 foot pound force (ft lbf) = 1.35582 J 1 J = 0.73756 ft lbf

1 foot poundal (ft pdl) = 0.04214 J 1J = 23.730 ft pdl

12.2.19 Power

Derived SI unit with special name and symbol

Name: Watt symbol: W in formulae: P

$$1\ \text{W} = 1\ \frac{\text{J}}{\text{s}}$$

Power is the quotient of the work done ΔW and the time taken Δt, i.e. $P = \Delta W / \Delta t$

1 kW = 1000 W 1 MW = 1000 kW GW = 1000 MW

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 horsepower (hp) = 0.7457 kW 1 kW = 1.341 hp

12.2.20 Viscosity

12.2.20.1 Dynamic viscosity

Derived SI unit unit: Pa · s in formulae: μ

$1\ \text{mPa} \cdot \text{s} = 10^{-3}\ \text{Pa} \cdot \text{s}$ the unit mPa.s has the same value as the previous unit cP (centipoise).

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 pound force second per square foot (lbf. sec/ft²) = 47.8803 Pa s

= 47,880.26 mPa s

12.2.20.2 Kinematic viscosity

Derived SI unit unit: m²/s in formulae: ν

$1\ \text{mm}^2/\text{s} = 10^{-6}\ \text{m}^2/\text{s}$ the unit mm²/s has the same value as the previous unit cSt (centistoke).

Conversion of dynamic and kinematic viscosity:

$$\nu = \frac{\eta}{\rho}$$

The Imperial units in the United Kingdom (UK) and the United States (USA):

1 square foot per second (ft²/sec) = 0.092903 m²/s

= 92,903.04 mm²/s

12.2.21 Temperature

SI basic unit name: Kelvin symbol: K in formulae: T

Definition: the unit Kelvin is 1 / 273.16 part of the thermodynamic temperature of the triple point of water.

The triple point is a fixed point in the phase diagram (p-T diagram) of a pure chemical substance at which the solid, liquid and gas phases are in equilibrium at the specified temperature and pressure. It is the intersection of the vapour pressure, melting point and sublimation curves.

The triple point of pure water is 273.16 K (= 0.01 °C) and 610.628 Pa and serves as the datum point of the international temperature scale. (13th. General Conference, 1967)

Derived SI unit with special name and symbol

Name: degree Celsius symbol: °C (1°C = 1K) in formulae: t

Given the Celsius temperature $t = T - T_0$ with T_0 = 273.15 K the unit degree Celsius is used for the Kelvin. A difference in two Celsius temperatures is also given in degree Celsius.

The Imperial units in the United Kingdom (UK) and the United States (USA):

Fahrenheit scale in place of Celsius

unit: °F in formulae: t_F

Rankine scale in place of Kelvin

unit: °R in formulae: T_R

Table 12.03 Common temperature scales

	Fahrenheit (°F)	Rankine (°R)	Celsius (°C)	Kelvin (K)
Boiling point of water	+ 212 °F	671.67 °R	+ 100 °C	373.15 K
Melting point of ice	+ 32 °F	491.67 °R	± 0 °C	273.15 K
Zero point of Fahrenheit scale	± 0 °F	459.67 °R	– 17 7/9 °C	255.37 K
Absolute zero	– 459.67 °F	0 °R	– 273.15 °C	0 K

Further fixed points of the Celsius scale:

Boiling point of oxygen: – 182.97 °C

Solidification of gold: 1063 °C

Explanation of fixed points:

Boiling point of pure water at normal pressure 1.013 bar.

Melting point of ice at normal pressure 1.013 bar

Zero point on Fahrenheit scale:

The lowest temperature achieved by Fahrenheit with an ice ammonium chloride mixture, winter 1708/09.

Absolute zero:

Temperature at which the atoms in a crystal stop oscillating

Table 12.04 Conversion of various temperature scales

Conversion from	to	Formula
°C	°F	t [°F] = 1.8 · t [°C] + 32
	K	T [K] = t [°C] + 273.15
	°R	T [°R] = 1.8 · t [°C] + 491.67
°F	°C	t [°C] = (t [°F] – 32) : 1.8
	K	T [K] = (t [°F] + 459.67) : 1.8
	°R	T [°R] = t [°F] + 459.67
K	°C	t [°C] = T [K] – 273.15
	°F	t [°F] = 1.8 · T [K] – 459.67
	°R	T [°R] = 1.8 · T [K]
°R	°C	t [°C] = (T [°R] – 491.67) : 1.8
	°F	t [°F] = T [°R] – 459.67
	K	T [K] = T [°R] : 1.8

Temperature difference: 1K = 1 °C = 1.8 °F = 1.8 °R

Table 12.05 Conversion table °F to °C

°F	°C	°F	°C	°F	°C	°F	°C	°F	°C
- 165	- 109.44	- 30	- 34.44	15	- 9.44	45	7.22	75	23.89
- 160	- 106.67	- 28	- 33.33	16	- 8.89	46	7.78	76	24.44
- 155	- 103.89	- 26	- 32.22	17	- 8.33	47	8.33	77	25.00
- 150	- 101.11	- 24	- 31.11	18	- 7.78	48	8.89	78	25.56
- 145	- 98.33	- 22	- 30.00	19	- 7.22	49	9.44	79	26.11
- 140	- 95.56	- 20	- 28.89	20	- 6.67	50	10.00	80	26.67
- 135	- 92.78	- 18	- 27.78	21	- 6.11	51	10.56	81	27.22
- 130	- 90.00	- 16	- 26.67	22	- 5.56	52	11.11	82	27.78
- 125	- 87.22	- 14	- 25.56	23	- 5.00	53	11.67	83	28.33
- 120	- 84.44	- 12	- 24.44	24	- 4.44	54	12.22	84	28.89
- 115	- 81.67	- 10	- 23.33	25	- 3.89	55	12.78	85	29.44
- 110	- 78.89	- 8	- 22.22	26	- 3.33	56	13.33	86	30.00
- 105	- 76.11	- 6	- 21.11	27	- 2.78	57	13.89	87	30.56
- 100	- 73.33	- 4	- 20.00	28	- 2.22	58	14.44	88	31.11
- 95	- 70.56	- 2	- 18.89	29	- 1.67	59	15.00	89	31.67
- 90	- 67.78	0	- 17.78	30	- 1.11	60	15.56	90	32.22
- 85	- 65.00	+ 1	- 17.22	31	- 0.56	61	16.11	91	32.78
- 80	- 62.22	2	- 16.67	32	0.00	62	16.67	92	33.33
- 75	- 59.44	3	- 16.11	33	+ 0.56	63	17.22	93	33.89
- 70	- 56.67	4	- 15.56	34	1.11	64	17.78	94	43.44
- 65	- 53.89	5	- 15.00	35	1.67	65	18.33	95	35.00
- 60	- 51.11	6	- 14.44	36	2.22	66	18.89	96	35.56
- 55	- 45.56	7	- 13.89	37	2.78	67	19.44	97	36.11
- 50	- 45.56	8	- 13.33	38	3.33	68	20.00	98	36.67
- 45	- 42.78	9	- 12.78	39	3.89	69	20.56	99	37.22
- 40	- 40.00	10	- 12.22	40	4.44	70	21.11	100	37.78
- 38	- 38.89	11	- 11.67	41	5.00	71	21.67	101	38.34
- 36	- 37.78	12	- 11.11	42	5.56	72	22.22	102	38.89
- 34	- 36.67	13	- 10.56	43	6.11	73	22.78	103	39.45
- 32	- 35.56	14	- 10.00	44	6.67	74	23.33	104	39.98

°F	°C	°F	°C	°F	°C	°F	°C	°F	°C
105	40.55	235	112.78	365	185.00	495	257.22	660	348.89
110	43.33	240	115.56	370	187.78	505	262.78	670	354.44
115	46.11	245	118.33	375	190.55	510	265.56	680	360.00
120	48.89	250	121.11	380	193.33	515	268.33	690	365.56
125	51.67	255	123.89	385	196.11	520	271.11	700	371.11
130	54.44	260	126.67	390	198.89	525	273.89	710	376.67
135	57.22	265	129.44	395	201.67	530	276.67	720	382.22
140	60.00	270	132.22	400	204.44	535	279.44	730	387.78
145	62.78	275	135.00	405	207.22	540	282.22	740	393.33
150	65.56	280	137.78	410	210.00	545	285.00	750	398.89
155	68.33	285	140.55	415	212.78	550	287.78	760	404.44
160	71.11	290	143.53	420	215.56	555	290.55	770	410.00
165	73.89	295	146.11	425	218.33	560	293.33	780	415.56
170	76.67	300	148.89	430	221.11	565	296.11	790	421.11
175	79.44	305	151.67	435	223.89	570	298.89	800	426.67
180	82.22	310	154.44	440	226.67	575	301.67	810	432.22
185	85.00	315	157.22	445	229.44	580	304.44	820	437.78
190	87.78	320	160.00	450	232.22	585	307.22	830	443.33
195	90.55	325	162.78	455	235.00	590	310.00	840	448.89
200	93.33	330	165.56	460	237.78	595	312.78	850	454.44
205	96.11	335	168.33	465	240.55	600	315.56	860	460.00
210	98.89	340	171.11	470	243.33	610	321.11	870	465.56
215	101.67	345	173.89	475	246.11	620	326.67	880	471.11
220	104.44	350	176.67	480	248.89	630	332.22	890	476.67
225	107.22	355	179.44	485	251.67	640	337.78	900	482.22
230	110.00	360	182.22	490	254.44	650	343.33	910	487.78

Interpolation values

°F	1	2	3	4	5	6	7	8	9
°C	0.56	1.11	1.67	2.22	2.78	3.33	3.89	4.44	5.00

12.2.22 Electrical current

SI basic unit name: Ampere symbol: A in formulae: I

Definition: the Ampere is the strength of a constant electrical current. which flowing through two straight. parallel conductors of infinite length. of negligible circular cross section. in a vacuum. at a separation of 1 metre. generates a force between the conductors of 2×10^{-7} Newton per metre length. (General Conference 1946). adopted by the (9th. General Conference 1948).

The electrical strength is the quotient of the quantity of electricity dQ flowing in the time dt and the time taken dt

$$I = \mathrm{d}Q / \mathrm{d}t$$

12.2.23 Electrical voltage

Derived SI unit with special name and symbol

Name: Volt symbol: V in formulae: U

$$1\mathrm{V} = 1 \frac{J}{\mathrm{C}}$$ Note: C (Coulomb) = Unit of electrical flow Q

One Volt is the electrical potential between two points of a homogeneous equal temperature single conductor in which a constant electrical current of 1 Ampere dissipates a power of 1 Watt i.e. 1V = 1W/A.

The electrical potential is the cause of electric current in a circuit.

12.2.24 Electrical resistance

Derived SI unit with a special name and symbol

Name: Ohm symbol: Ω in formulae: R

$$1\ \Omega = 1 \frac{\mathrm{V}}{\mathrm{A}}$$

Electrical resistance is the property of a substance to prevent the flow of electricity when the conductive elements are subjected to an electrical field or potential.

The alternating current resistance is that which arises in an AC circuit as a result of the frequency ω of the AC current dependent relationship $Z = U/I$.

12.2.25 Electrical conductance

Derived SI unit with special name and symbol

Name: Siemens symbol: S in formulae: *G*

Conductance is the reciprocal of resistance

$$1\ \mathrm{S} = \frac{1}{\Omega}$$

$$1\mu\mathrm{S} = 10^{-6}\ \mathrm{S} \qquad 1\mathrm{mS} = 10^{-3}\ \mathrm{S}$$

12.2.26 Electrical conductivity

Derived SI unit unit: Siemens/m = S/m in formulae: γ,σ or k

$$1\ \mathrm{S/m} = 10^3\ \mathrm{mS/m} = 10^4\ \mu\mathrm{S/cm}$$

Conductivity is a material property dependent on temperature. With its increase. the current rises for a given Voltage.

When assessing water quality. the electrical conductivity is used as a measure of the concentration of ionisable material.

12.2.27 Conversion of Imperial UK and USA Units to Metric Units

Imperial UK and USA		Metric Unit		Conversion factor
Length				
Foot Inch Microinch	ft in µin	Metre Millimetre Micrometre	m mm µm	0.3048 25.4 0.0254
Area				
Square inch Square foot	in^2 ft^2	Square centimetre Square metre	cm^2 m^2	6.4516 0.0929
Volume				
Cubic foot Gallon (US) Gallon (UK)	ft^3 US gal UK gal	Litre Litre Litre	L L L	28.3168 3.78541 4.54609
Weight				
Pound	lb	Kilogramme	kg	0.45359
Force				
Pound force	lbf	Newton	N	4.44822
Pressure				
Pound force per square inch	lbf/in^2	Kilopascal	kPa	6.895
Torque				
Pound foot	lb ft	Newton metre	Nm	1.356
Density				
Pound mass per cubic foot	lbm/ft^3	Kilogramme per cubic metre	kg/m^3	16.018
Flowrate				
Gallon (US) per minute Gallon (UK) per minute	USgpm UKgpm	Cubic metre per hour Cubic metre per hour	m^3/h m^3/h	0.2271 0.2728
Velocity				
Foot per second	ft/s	Metre per second	m/s	0.3048
Power				
Horsepower	hp	Kilowatt	kW	0.746

13 Tables

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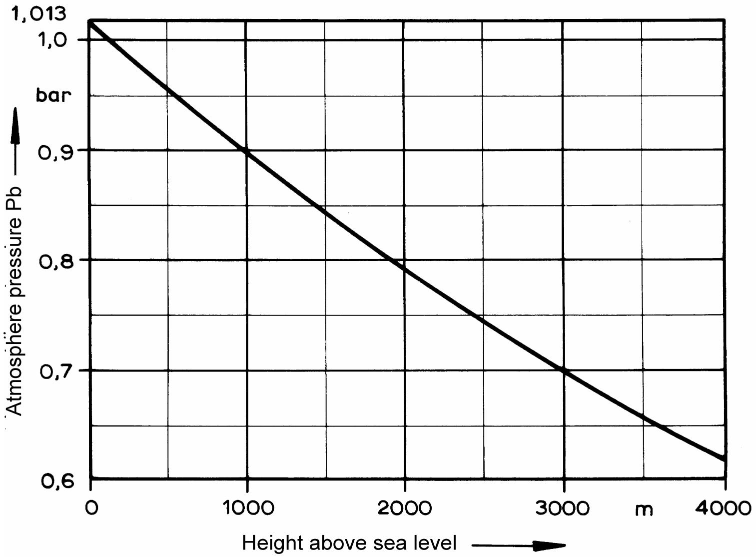

Table 13.01 Atmospheric pressure p_{amb} dependent on height above sea level, assuming 1.013 bar sea level

Climatic variations can be up to ± 3% from the nominal value. In extreme weather it can vary by + 5.6% to – 8.2%.

The mean atmospheric pressure above sea level can also be calculated from the following formula up to a height of 11 km.

$$p_{amb\ H} = p_{amb\ NN} \left(\frac{288 - 6.5\ H}{288} \right)^{5.255}$$

with p_{amb} in bar
and H in km

From the above formula $p_{amb\ H}$

H km	1	1.5	2	2.5	3	3.5	4	5
$p_{amb\ H}$ bar	0.898	0.845	0.795	0.747	0.701	0.657	0.616	0.540

Table 13.02 Acceleration due to gravity g

The local acceleration due to gravity is dependent on the latitude φ and the height h above sea level. There may also be local variations for geological reasons.

For a latitudes ca. 44 to 54° the mean figure is $g = 9.81$ m/s².

For other latitudes and for more exact calculations, the following formula may be used:

Acceleration due to gravity at sea level

$$g_{NN} = 9.7803\ (1+0.0053 \cdot \sin^2 \varphi)\ [\mathrm{m/s^2}]$$

Local acceleration due to gravity

$$g_h = g_{NN} - 3 \cdot 10^{-6} \cdot h\ [\mathrm{m/s^2}]$$

with φ in degrees and h in m

Acceleration due to gravity g_{NN} for various latitudes

φ °	0	2	4	6	8
0	9.7803	9.7804	9.7806	9.7809	9.7813
10	9.7819	9.7825	9.7833	9.7842	9.7852
20	9.7864	9.7876	9.7889	9.7903	9.7917
30	9.7933	9.7949	9.7965	9.7982	9.8000
40	9.8017	9.8035	9.8053	9.8070	9.8090
50	9.8107	9.8125	9.8142	9.8159	9.8176
60	9.8192	9.8207	9.8222	9.8236	9.8249
70	9.8261	9.8272	9.8282	9.8291	9.8299
80	9.8306	9.8311	9.8316	9.8319	9.8321
90	9.8321				

Table 13.03 Vapour pressure p_v in bar and density ρ in kg/dm³ of saturated water vapour

°C	bar	kg/dm³	°C	bar	kg/dm³	°C	bar	kg/dm³
0	0.00611	0.9998	31	0.04491	0.9954	62	0.2184	0.9821
1	0.00657	0.9999	32	0.04753	0.9951	63	0.2286	0.9816
2	0.00706	0.9999	33	0.05029	0.9947	64	0.2391	0.9811
3	0.00758	0.9999	34	0.05318	0.9944	65	0.2501	0.9805
4	0.00813	1.0000	35	0.05622	0.9940	66	0.2615	0.9799
5	0.00872	1.0000	36	0.05940	0.9937	67	0.2733	0.9793
6	0.00935	1.0000	37	0.06274	0.9931	68	0.2856	0.9788
7	0.01001	0.9999	38	0.06624	0.9931	69	0.2984	0.9782
8	0.01072	0.9999	39	0.06991	0.9927	70	0.3116	0.9777
9	0.01147	0.9998	40	0.07375	0.9923	71	0.3253	0.9770
10	0.01227	0.9997	41	0.07777	0.9919	72	0.3396	0.9765
11	0.01312	0.9997	42	0.08198	0.9915	73	0.3543	0.9759
12	0.01401	0.9996	43	0.08639	0.9911	74	0.3696	0.9753
13	0.01497	0.9994	44	0.09100	0.9907	75	0.3855	0.9748
14	0.01587	0.9993	45	0.09582	0.9902	76	0.4019	0.9741
15	0.01704	0.9992	46	0.10086	0.9898	77	0.4189	0.9735
16	0.01817	0.9990	47	0.10612	0.9894	78	0.4365	0.9729
17	0.01936	0.9988	48	0.11162	0.9889	79	0.4547	0.9723
18	0.02062	0.9987	49	0.11736	0.9884	80	0.4736	0.9716
19	0.02196	0.9985	50	0.12335	0.9880	81	0.4931	0.9710
20	0.02337	0.9983	51	0.12961	0.9876	82	0.5133	0.9704
21	0.02485	0.9981	52	0.13613	0.9871	83	0.5342	0.9697
22	0.02642	0.9978	53	0.14293	0.9866	84	0.5557	0.9691
23	0.02808	0.9976	54	0.15002	0.9862	85	0.5780	0.9684
24	0.02982	0.9974	55	0.15741	0.9857	86	0.6011	0.9678
25	0.03166	0.9971	56	0.16511	0.9852	87	0.6249	0.9671
26	0.03360	0.9968	57	0.17313	0.9846	88	0.6495	0.9665
27	0.03564	0.9966	58	0.18147	0.9842	89	0.6749	0.9658
28	0.03778	0.9963	59	0.19016	0.9837	90	0.7011	0.9652
29	0.04004	0.9960	60	0.19920	0.9832	91	0.7281	0.9644
30	0.04241	0.9957	61	0.2086	0.9826	92	0.7561	0.9638

Table 13.03 continuation

°C	bar	kg/dm³	°C	bar	kg/dm³	°C	bar	kg/dm³
93	0.7849	0.9630	124	2.2504	0.9396	155	5.433	0.9121
94	0.8146	0.9624	125	2.3210	0.9388	156	5.577	0.9111
95	0.8453	0.9616	126	2.3933	0.9379	157	5.723	0.9102
96	0.8769	0.9610	127	2.4675	0.9371	158	5.872	0.9092
97	0.9094	0.9603	128	2.5435	0.9362	159	6.025	0.9082
98	0.9430	0.9596	129	2.6215	0.9354	160	6.181	0.9073
99	0.9776	0.9589	130	2.7013	0.9346	161	6.339	0.9063
100	1.0133	0.9581	131	2.7831	0.9337	162	6.502	0.9053
101	1.0500	0.9574	132	2.8670	0.9328	163	6.667	0.9043
102	1.0878	0.9567	133	2.9528	0.9320	164	6.836	0.9033
103	1.1267	0.9559	134	3.041	0.9311	165	7.008	0.9024
104	1.1668	0.9552	135	3.131	0.9302	166	7.183	0.9013
105	1.2080	0.9545	136	3.223	0.9294	167	7.362	0.9003
106	1.2504	0.9537	137	3.317	0.9285	168	7.545	0.8994
107	1.2941	0.9529	138	3.414	0.9276	169	7.731	0.8983
108	1.3390	0.9515	139	3.513	0.9268	170	7.920	0.8973
109	1.3852	0.9515	140	3.614	0.9258	171	8.114	0.8963
110	1.4327	0.9507	141	3.717	0.9250	172	8.311	0.8953
111	1.4815	0.9499	142	3.823	0.9241	173	8.511	0.8942
112	1.5832	0.9491	143	3.931	0.9232	174	8.716	0.8932
113	1.5832	0.9484	144	4.042	0.9223	175	8.924	0.8921
114	1.6362	0.9476	145	4.155	0.9214	176	9.137	0.8911
115	1.6906	0.9468	146	4.271	0.9205	177	9.353	0.8901
116	1.7465	0.9460	147	4.389	0.9195	178	9.574	0.8891
117	1.8039	0.9453	148	4.510	0.9186	179	9.798	0.8879
118	1.8628	0.9445	149	4.510	0.9177	180	10.027	0.8869
119	1.9233	0.9437	150	4.760	0.9168	181	10.259	0.8858
120	1.9854	0.9429	151	4.889	0.9158	182	10.496	0.8848
121	2.0492	0.9421	152	5.021	0.9149	183	10.738	0.8837
122	2.1145	0.9412	153	5.155	0.9140	184	10.983	0.8826
123	2.1816	0.9404	154	5.293	0.9130	185	11.233	0.8815

Table 13.03 continuation

°C	bar	kg/dm³	°C	bar	kg/dm³	°C	bar	kg/dm³
186	11.488	0.8804	217	21.896	0.8442	248	38.449	0.8021
187	11.747	0.8794	218	22.324	0.8429	249	39.108	0.8006
188	12.010	0.8783	219	22.758	0.8416	250	39.776	0.7992
189	12.278	0.8771	220	23.198	0.8403	251	40.452	0.7977
190	12.551	0.8760	221	23.645	0.8391	252	41.137	0.7962
191	12.829	0.8749	222	24.099	0.8378	253	41.831	0.7947
192	13.111	0.8739	223	24.560	0.8365	254	42.534	0.7932
193	13.398	0.8727	224	25.027	0.8352	255	43.246	0.7916
194	13.690	0.8715	225	25.501	0.8339	256	43.967	0.7901
195	13.987	0.8704	226	25.982	0.8326	257	44.697	0.7886
196	14.289	0.8693	227	26.470	0.8313	258	45.437	0.7870
197	14.596	0.8681	228	26.965	0.8300	259	46.185	0.7855
198	14.909	0.8670	229	27.467	0.8286	260	46.943	0.7839
199	15.226	0.8659	230	27.976	0.8273	261	47.711	0.7824
200	15.549	0.8647	231	28.493	0.8260	262	48.488	0.7808
201	15.877	0.8635	232	29.016	0.8246	263	49.275	0.7792
202	16.210	0.8624	233	29.547	0.8233	264	50.071	0.7775
203	16.549	0.8612	234	30.086	0.8219	265	50.877	0.7760
204	16.893	0.8600	235	30.632	0.8206	266	51.693	0.7744
205	17.243	0.8588	236	31.186	0.8192	267	52.519	0.7727
206	17.598	0.8576	237	31.747	0.8178	268	53.355	0.7711
207	17.959	0.8565	238	32.317	0.8164	269	54.202	0.7694
208	18.326	0.8552	239	32.893	0.8150	270	55.058	0.7678
209	18.699	0.8540	240	33.478	0.8136	271	55.925	0.7661
210	19.077	0.8528	241	34.071	0.8122	272	56.802	0.7644
211	19.462	0.8516	242	34.672	0.8108	273	57.689	0.7627
212	19.852	0.8503	243	35.281	0.8094	274	58.587	0.7610
213	20.249	0.8491	244	35.898	0.8080	275	59.496	0.7593
214	20.651	0.8479	245	36.523	0.8065	276	60.415	0.7576
215	21.060	0.8467	246	37.157	0.8050	277	61.346	0.7558
216	21.475	0.8454	247	37.799	0.8036	278	62.287	0.7541

Table 13.03 continuation

°C	bar	kg/dm³	°C	bar	kg/dm³	°C	bar	kg/dm³
279	63.239	0.7523	308	96.036	0.6951	337	140.64	0.6198
280	64.202	0.7505	309	97.361	0.6929	338	142.42	0.6166
281	65.176	0.7487	310	98.70	0.6906	339	144.23	0.6135
282	66.162	0.7469	311	100.05	0.6884	340	146.05	0.6102
283	67.158	0.7452	312	101.42	0.6861	341	147.89	0.6069
284	68.167	0.7433	313	102.80	0.6838	342	149.76	0.6036
285	69.186	0.7415	314	104.20	0.6814	343	151.64	0.6002
286	70.218	0.7396	315	105.61	0.6791	344	153.54	0.5967
287	71.261	0.7378	316	107.04	0.6767	345	155.45	0.5932
288	72.315	0.7359	317	108.48	0.6743	346	157.39	0.5896
289	73.382	0.7340	318	109.93	0.6718	347	159.35	0.5859
290	74.461	0.7321	319	111.40	0.6694	348	161.33	0.5821
291	75.551	0.7302	320	112.89	0.6669	349	163.33	0.5783
292	76.654	0.7282	321	114.39	0.6644	350	165.35	0.5744
293	77.769	0.7263	322	115.91	0.6627	351	167.39	0.5704
294	78.897	0.7243	323	117.44	0.6593	352	169.45	0.5662
295	80.037	0.7223	324	118.99	0.6567	353	171.54	0.5619
296	81.189	0.7204	325	120.56	0.6541	354	173.64	0.5575
297	82.355	0.7183	326	122.14	0.6514	355	175.77	0.5529
298	83.532	0.7163	327	123.73	0.6487	356	177.92	0.5482
299	84.723	0.7143	328	125.35	0.6460	357	180.09	0.5433
300	85.927	0.7122	329	126.98	0.6432	358	182.29	0.5382
301	87.144	0.7101	330	128.63	0.6404	359	184.51	0.5329
302	88.374	0.7081	331	130.29	0.6376	360	186.75	0.5275
303	89.617	0.7059	332	131.97	0.6347	365	198.33	0.4960
304	90.873	0.7038	333	133.67	0.6318	370	210.54	0.4518
305	92.144	0.7017	334	135.38	0.6289	Kritische Daten: [1]		
306	93.427	0.6995	335	137.12	0.6259	374.15	221.20	0.3155
307	94.725	0.6973	336	138.87	0.6229			

1) Values from publication "Values of water and water vapour in SI units" by E. Schmidt and U. Grigull. 4. edition 1989

Fig 13.04 Vapour pressure p_v of various liquids

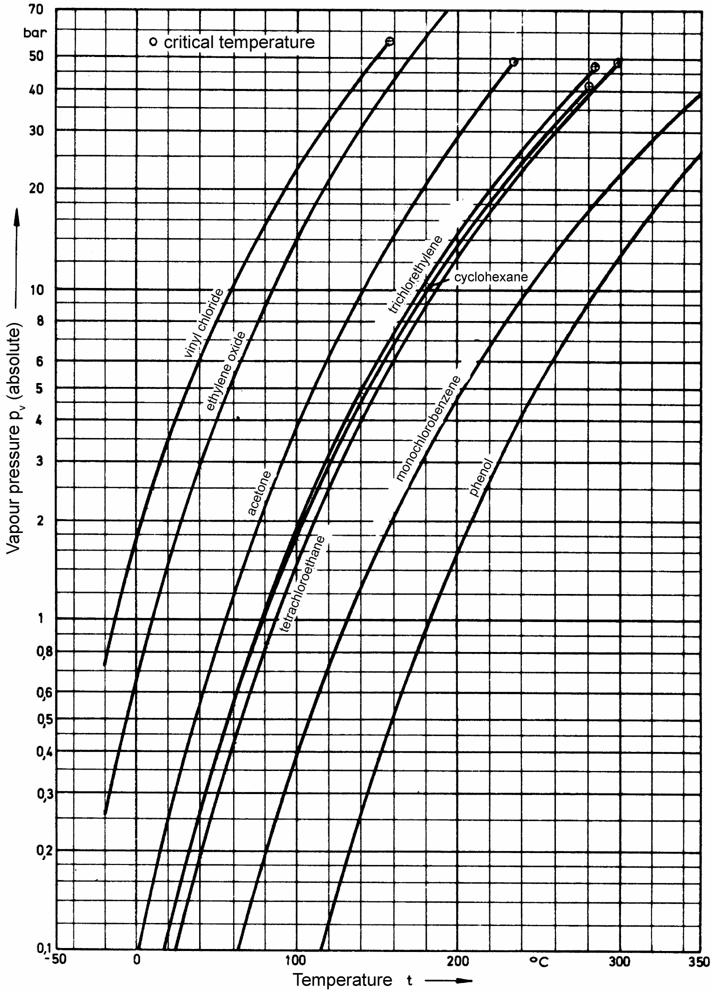

Fig 13.05 Vapour pressure p_v of various liquids

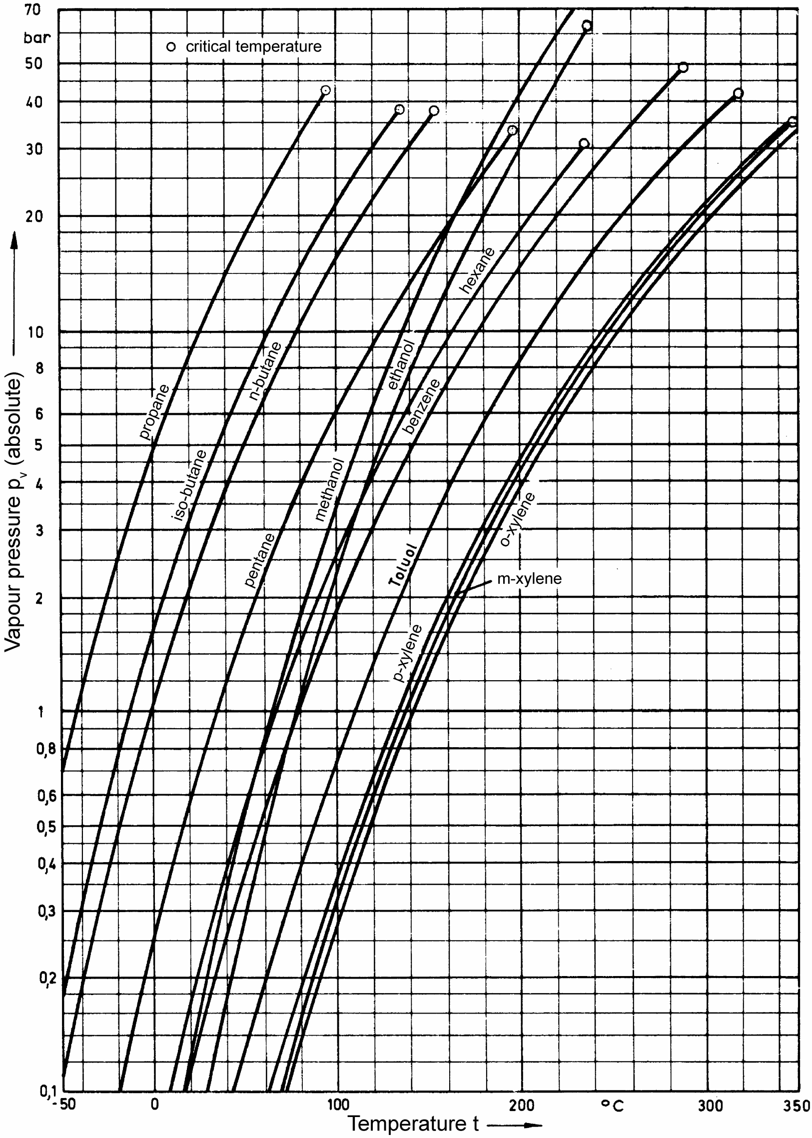

Table 13.06 Density ρ (kg/dm³) of pure liquids and its variation with the temperature t (°C)

Chemical Name	– 100	– 75	– 50	– 25	0	20	50	100	150	200	250
A											
Acetic Acid						1.049	1.018	0.960	0.896	0.827	0.736
Acetic Anhydride					1.105	1.082	1.044				
Acetone		0.983	0.868	0.840	0.812	0.791	0.756				
Acetonitrile				0.829	0.803	0.783	0.750				
Acetyl Chloride						1.104					
Allyl Alcohol				0.891	0.870	0.854	0.827	0.774	0.711	0.628	0.502
Allyl Chloride						0.938					
Ammonia		0.731	0.702	0.671	0.639	0.610	0.563	0.457			
Aniline					1.039	1.022	0.996	0.951			
Anisole						0.955					
B											
Benzaldehyde					1.062	1.046					
Benzene						0.879	0.847	0.793	0.731	0.661	0.561
Benzyl Alcohol					1.061	1.045	1.022				
Benzyl Chloride					1.103						
Bromine					3.208	3.140					
Butadiene (1,2-B.)		0.766	0.734	0.701	0.676	0.652	0.614	0.542	0.420		
Butadiene (1,3-B.)	0.754	0.728	0.701	0.673	0.646	0.621	0.582	0.501	0.320		
Butane (n-B.)	0.690	0.670	0.649	0.626	0.601	0.579	0.542	0.468	0.290		
Butane (iso-B.)				0.610	0.584	0.559	0.520				
Butanol (n-B.)	0.910	0.884	0.866	0.846	0.823	0.807	0.780	0.737	0.686	0.627	0.534

Chemical Name	– 100	– 75	– 50	– 25	0	20	50	100	150	200	250
Butanone					0.826	0.803					
Butene (1-B.)	0.712	0.691	0.668	0.645	0.619	0.592	0.558	0.477			
Butene (cis 2-B.)			0.700	0.675	0.643	0.620	0.585	0.506	0.365		
Butyl Acetate (n-B.)						0.882					
Butylene Glycol						1.017					
Butyric Acid (n-B.)					0.977	0.958	0.927			0.774	0.699
C D											
Carbon Disulphide	1.432	1.398	1.362	1.328	1.292	1.262					
Carbon Tetrachloride				1.685	1.633	1.594	1.534	1.434	1.322	1.189	0.980
Chlorine	1.717	1.659	1.598	1.538	1.469	1.409	1.314	1.123	0.809		
Chlorobenzene (Mono-C.)					1.128	1.106	1.074	1.019	0.960	0.896	
Chloroform		1.660	1.618	1.565	1.526	1.490	1.433	1.315	1.208	1.062	0.790
Chlorosulphonic Acid						1.753					
Cresol (m-C.)						1.034	1.009				
Cumene						0.862					
Cyclohexane						0.779	0.750	0.700	0.645	0.578	0.482
Cyclohexanol							0.925				
Cyclohexanone						0.947	0.920				
Dibutyl Ether						0.770					
Diethyl Ether	0.842	0.816	0.790	0.764	0.736	0.714	0.676	0.611	0.518		
Chemical Name	– 100	– 75	– 50	– 25	0	20	50	100	150	200	250
Diethylene Glycol					1.133	1.126	1.098	1.087	1.014		
Dioxane (1.4-D.)						1.034	1.008				

Chemical Name	– 100	– 75	– 50	– 25	0	20	50	100	150	200	250
E											
Epichlorohydrin				1.245	1.225	1.186	1.142	1.078	1.004	0.928	0.823
Ethanol	0.890	0.868	0.845	0.823	0.807	0.790	0.763	0.716	0.649	0.557	
Ethyl Acetate					0.924	0.901	0.864	0.797	0.721	0.621	0.330
Ethyl Chloride			0.995	0.955	0.924	0.892	0.846	0.750	0.670		
Ethyl Formate					0.948	0.923	0.883	0.811	0.726	0.602	
Ethylbenzene		0.951	0.929	0.907	0.885	0.866	0.840	0.797			
Ethylene Chlorhydrin						1.210					
Ethylene Dichloride			1.360	1.325	1.287	1.258	1.210	1.135	1.060		
Ethylene Glycol					1.128	1.115	1.090	1.054	1.016	0.974	0.922
F G H											
Fatty Alcohol (Octadecanol)							0.814				
Formaldehyde						0.815					
Formamide						1.112					
Formic Acid						1.220	1.184				
Furfural					1.181	1.160	1.128				
Glycerol					1.272	1.260	1.242	1.209	1.172	1.132	1.094
Heptane		0.761	0.741	0.721	0.701	0.684	0.658	0.612	0.560	0.495	0.388
Hexane	0.760	0.742	0.721	0.700	0.678	0.659	0.631	0.580	0.520	0.438	
Hexanol				0.856	0.833	0.820	0.805	0.762	0.710	0.656	0.595
Hydrogen Bromide			2.132	2.020	1.896	1.791	1.625				
Hydrogen Chloride	1.235	1.158	1.080	1.008	0.920						
Hydrogen Cyanide					0.715	0.688					

Chemical Name	– 100	– 75	– 50	– 25	0	20	50	100	150	200	250
Hydrogen Fluoride			1.123	1.063	1.002	0.962					
Hydrogen Sulphide			0.980	0.924	0.866						
M N											
Methanol	0.897	0.875	0.854	0.832	0.811	0.792	0.765	0.714	0.650	0.553	
Methyl Acetate					0.959	0.934	0.894	0.822	0.734	0.610	
Methyl Chloride	1.140	1.090	1.050	0.990	0.960	0.921	0.859	0.770			
Methyl Formate					1.003	0.975	0.929	0.845	0.740	0.566	
Methylcyclohexane	0.870	0.850	0.828	0.808	0.787	0.769	0.742	0.697			
Methylene Chloride	1.540	1.480	1.460	1.390	1.362	1.336	1.270	1.190	1.070	0.920	
Methylpentanone						0.813					
Nitrobenzene						1.203	1.174				
O P											
Oleic Acid							0.886				
Paraldehyde						0.994	0.956	0.899			
Pentane	0.737	0.715	0.693	0.670	0.646	0.626	0.596	0.533	0.460		
Petrol (Extraction-P.)			0.775	0.755	0.735	0.720	0.690	0.650			
Phenol							1.050				
Propane	0.646	0.619	0.591	0.561	0.529	0.501	0.450				
Propanol	0.911	0.887	0.865	0.842	0.820	0.804	0.779	0.733	0.674	0.592	0.453
Propene	0.671	0.642	0.615	0.586	0.555	0.523	0.465				
Propionic Acid					1.015	0.993				0.786	0.703
Propylene Glycol					1.054	1.040	1.016	0.974	0.932		
Pyridine				1.028	1.003	0.983	0.953	0.901			

Chemical Name	– 100	– 75	– 50	– 25	0	20	50	100	150	200	250
S T											
Stannic Chloride						2.224	2.165				
Stearic Acid								0.839			
Sulphur monochlorid						1.678					
Tetrachloroethene				1.692	1.656	1.621	1.570	1.485	1.402	1.308	1.190
Tetrahydrofuran						0.889					
Toluene		0.956	0.933	0.910	0.885	0.867	0.839	0.793	0.737	0.672	0.595
Trichloroethene	1.670	1.630	1.585	1.540	1.500	1.463	1.415	1.328	1.240	1.130	0.980
Tricresyl Phosphate						1.180					
Triethylamine					0.746	0.726	0.699	0.652			0.400
V X											
Vinyl Acetate						0.930	0.892				
Vinyl Chloride	1.085	1.050	1.015	0.980	0.945	0.911	0.863	0.745	0.510		
Xylene (m-X.)				0.902	0.881	0.866	0.838	0.795	0.744	0.683	0.615

The values for the density ρ are taken from reference literature. They are sufficiently accurate for necessary calculations e.g. of mass rate flow or rate of flow, pressure and pump power input.

Intermediate values can be calculated by linear interpolation.

$$\rho = \rho_1 + \frac{t - t_1}{t_2 - t_1} \cdot (\rho_2 - \rho_1)$$

Table 13.07 Density ρ (kg/dm³) of aqueous solutions as a function of concentration (w: 1 to 12%) and temperature

Chemical Name	t °C	1 %	2 %	3 %	4 %	5 %	6 %	7 %	8 %	9 %	10 %	12 %
A												
Acetic Acid	0		1.003								1.018	
	20	1.000	1.001	1.003	1.004	1.006	1.007	1.008	1.010	1.011	1.013	1.015
	60		0.985								0.994	
Acetone	20										0.985	
	60										0.966	
Alum	19	1.008	1.017	1.027	1.037	1.047	1.057					
Aluminium Chloride	18	1.008	1.016		1.034		1.053		1.071		1.090	1.109
Aluminium Nitrate	18	1.007	1.014		1.031		1.047		1.064		1.081	1.099
Aluminium Sulphate	19	1.009	1.019		1.040		1.061		1.083		1.105	1.129
Ammonia	15		0.990			0.978					0.959	
	20	0.994	0.990	0.985	0.981	0.977	0.973	0.969	0.965	0.961	0.958	0.950
	50		0.979			0.966					0.944	
Ammonium Acetate	18	1.001	1.003		1.007		1.012		10.16		1.020	1.024
Ammonium Bromide	18	1.004	1.010		1.022		1.033		1.045		1.057	
	25	1.003	1.008		1.020		1.031		1.043		1.055	1.067
Ammonium Carbonate	15	1.003	1.006		1.013		1.020		1.027		1.034	1.040
Ammonium Chloride	20	1.001	1.005		1.011		1.017		1.023		1.029	1.034
Ammonium Nitrate	20	1.002	1.006		1.015		1.023		1.031		1.040	1.048
Ammonium Oxalate	15	1.004	1.009	1.014	1.019	1.024	1.029	1.035				

Chemical Name	*t* °C	1 %	2 %	3 %	4 %	5 %	6 %	7 %	8 %	9 %	10 %	12 %
Ammonium Sulphate	20	1.004	1.010		1.022		1.034		1.046		1.057	1.069
Arsenic Acid	15	1.006	1.012		1.026		1.040		1.054		1.068	1.083
B												
Barium Acetate	18	1.006	1.013		1.028		1.043		1.059		1.075	1.091
Barium Chloride	20		1.016		1.034		1.053		1.072		1.092	1.113
Barium Hydroxide	18	1.013	1.018		1.025		1.037		1.055		1.077	
Barium Nitrate	18	1.007	1.015		1.032		1.049		1.067		1.086	
Borax	15	1.008	1.018	1.027								
Boric Acid	20	1.002	1.006	1.009	1.014							
C												
Cadmium Chloride	20		1.016		1.034		1.052		1.072		1.092	1.112
Calcium Chloride	20	1.007	1.015		1.032		1.049		1.066		1.084	1.102
Calcium Hydroxide	20		1.013		1.025		1.037		1.049		1.061	
Calcium Nitrate	18		1.014		1.029		1.045		1.061		1.077	1.094
Chloric Acid	18	1.004	1.010		1.022		1.034		1.047		1.059	1.072
Chrome Alum, green	15	1.007	1.016		1.034		1.052		1.070		1.089	1.109
violet	15	1.009	1.018		1.038		1.057		1.077			
Chromic Acid	15	1.006	1.014		1.030		1.045		1.060		1.076	1.093
Citric Acid	18		1.007		1.015		1.022		1.030		1.038	1.046
Copper Chloride	20	1.007	1.017		1.036		1.056		1.076		1.096	1.116
Copper Sulphate	20	1.009	1.019		1.040		1.062		1.084		1.107	1.131
E												
Ethanol	15	0.997	0.995	0.994	0.992	0.990	0.989	0.987	0.986	0.984	0.983	0.981

Chemical Name	t °C	1 %	2 %	3 %	4 %	5 %	6 %	7 %	8 %	9 %	10 %	12 %
	50		0.984			0.979					0.970	
	80		0.968			0.962					0.952	
F G												
Ferric Chloride	20	1.007	1.015		1.032		1.049		1.067		1.085	1.104
Ferric Nitrate	18	1.007	1.014		1.030		1.047		1.064		1.081	1.099
Ferric Sulphate	17.5	1.007	1.016		1.033		1.050		1.067		1.084	1.103
Ferrous Chloride	18	1.008	1.017		1.035		1.054		1.073		1.092	1.113
Ferrous sulphate	18	1.009	1.018		1.038		1.058		1.079		1.100	1.122
Formic Acid	20	1.002	1.004	1.007	1.009	1.012	1.014	1.017	1.020	1.022	1.025	1.030
	40		0.997			1.003					1.015	
Glycerol	0					1.013					1.025	
	20	1.001	1.003	1.005	1.008	1.010	1.013	1.015	1.017	1.020	1.022	1.027
H												
Hydrobromic Acid	20	1.005	1.012		1.027		1.042		1.057		1.072	1.088
Hydrochloric Acid	0		1.012			1.027					1.052	
	20	1.003	1.008	1.013	1.018	1.023	1.028	1.033	1.038	1.043	1.047	1.057
	100		0.968			0.984					1.009	
Hydrocyanic Acid	18	0.998	0.996		0.993		0.990		0.986		0.982	
Hydrofluoric Acid	20		1.005		1.012		1.021		1.028		1.036	1.043
Hydrofluorsilicic Acid	17.5	1.007	1.015		1.031		1.048		1.065		1.082	1.100
Hydrogen Peroxide	18	1.002	1.006		1.013		1.020		1.028		1.035	1.043

Chemical Name	t °C	1 %	2 %	3 %	4 %	5 %	6 %	7 %	8 %	9 %	10 %	12 %
L												
Lactic Acid	20		1.003									
Lead Acetate	18	1.006	1.014		1.029		1.045		1.061		1.077	1.094
Lithium Bromide	20	1.006	1.013		1.028		1.043		1.059		1.075	1.091
Lithium Chloride	20	1.004	1.010		1.022		1.033		1.044		1.056	1.068
M												
Magnesium Chloride	20		1.015						1.065			
	30		1.012						1.062			
Magnesium Sulphate	20		1.019		1.039		1.060		1.082		1.103	1.126
Manganese Chloride	18	1.007	1.015		1.032		1.050		1.068		1.086	1.105
Manganese Sulphate	15	1.009	1.019		1.039		1.060		1.081		1.103	1.125
Mercuric Chloride	20	1.007	1.015	1.024	1.032	1.041	1.050					
Methanol	15.6	0.997	0.996	0.994	0.992	0.991	0.989	0.987	0.986	0.984	0.983	0.980
N												
Nickel Chloride	20	1.008	1.018		1.037		1.057		1.078		1.099	1.121
Nickel Nitrate	20	1.007	1.015		1.033		1.050		1.069		1.088	1.107
Nickel Sulphate	18	1.009	1.020		1.042		1.063		1.085		1.109	1.133
Nitric Acid	20	1.004	1.010	1.015	1.020	1.026	1.031	1.037	1.043	1.049	1.054	1.066
	100		0.969			0.983					1.008	
O												
Oleum (% stands for % free SO_3)	35		1.827		1.835		1.843		1.850		1.857	1.863
Oxalic Acid	17.5	1.004	1.008	1.013	1.018	1.023	1.028	1.033	1.038	1.042		

Chemical Name	*t* °C	1 %	2 %	3 %	4 %	5 %	6 %	7 %	8 %	9 %	10 %	12 %
P												
Phosphoric Acid	20	1.004	1.009	1.014	1.020	1.025	1.031	1.036	1.042	1.047	1.053	1.065
Potassium Bromide	20	1.005	1.013		1.028		1.043		1.058		1.074	1.090
Potassium Carbonate	20	1.007	1.016		1.035		1.053		1.072		1.090	1.110
Potassium Chlorate	18	1.005	1.011	1.018	1.025	1.031	1.038					
Potassium Chloride	20	1.005	1.011		1.024		1.037		1.050		1.063	1.077
Potassium Cyanide	15	1.004	1.009		1.019		1.030		1.040		1.051	1.061
Potassium Dichromate	20	1.005	1.012	1.019	1.026	1.034	1.041	1.048	1.055	1.063	1.070	1.086
Potassium Ferricyanide	20	1.003	1.009		1.020		1.031		1.043		1.054	1.066
Potassium Ferrocyanide	20	1.005	1.012		1.026		1.040		1.054		1.068	1.082
Potassium Hydroxide	0		1.019			1.048					1.096	
	15	1.008	1.018	1.027	1.036	1.045	1.054	1.064	1.073	1.082	1.092	1.111
	50		1.055			1.032					1.077	
Potassium Iodide	20	1.006	1.013		1.028		1.044		1.060		1.076	1.093
Potassium Nitrate	20	1.005	1.011		1.023		1.036		1.049		1.063	1.076
Potassium Permanganate	15	1.006	1.013	1.020	1.027	1.034	1.041					
Potassium Sulphate	20	1.006	1.015	1.023	1.031	1.039	1.048	1.056	1.065	1.073	1.082	
S												
Silver Nitrate	20	1.007	1.015		1.033		1.051		1.069		1.088	1.108
Sodium Bicarbonate	18	1.006	1.013	1.021	1.028	1.035	1.043	1.051	1.058			
Sodium Bromide	20	1.006	1.014		1.030		1.046		1.063		1.080	1.098
Sodium Carbonate	15	1.010	1.020	1.031	1.041	1.052	1.062	1.073	1.083	1.094	1.105	1.127
	25	1.007	1.018	1.028	1.038	1.048	1.059	1.069	1.080	1.090	1.101	1.122

Chemical Name	t °C	1 %	2 %	3 %	4 %	5 %	6 %	7 %	8 %	9 %	10 %	12 %
Sodium Chlorate	18	1.005	1.012		1.026		1.040		1.054		1.068	1.083
Sodium Chloride	0						1.046		1.062		1.077	1.094
	20	1.005	1.013		1.027		1.041		1.056		1.071	1.086
Sodium Dichromate	15		1.013		1.027		1.041		1.056		1.070	1.084
Sodium Hydrogensulphate	20	1.006	1.014		1.029		1.045		1.061		1.077	1.094
Sodium Hydrogensulphite	15		1.019	1.032	1.042	1.051	1.062		1.084		1.105	1.125
Sodium Hydroxide	0		1.024			1.060					1.117	
	20	1.010	1.021	1.032	1.043	1.054	1.065	1.076	1.087	1.098	1.109	1.131
	100		0.980			1.012					1.064	
Sodium Hypochlorite	20						1.016					
Sodium Nitrate	20	1.005	1.012		1.025		1.039		1.053		1.067	1.082
Sodium Nitrite	20	1.005	1.011		1.024		1.038		1.052		1.065	1.078
Sodium Phosphates												
monobasic	25	1.005	1.012		1.027		1.042		1.058		1.073	
dibasic	18	1.009	1.020	1.031	1.043	1.055	1.067					
tribasic	15	1.009	1.019	1.030	1.041	1.052	1.062	1.074	1.085	1.096	1.108	
Sodium Silicates												
$Na_2O \cdot 2{,}06\ SiO_2$	20	1.007	1.016		1.035		1.054		1.073		1.093	1.113
$Na_2O \cdot 3{,}36\ SiO_2$	20	1.006	1.014		1.030		1.047		1.065		1.083	1.101
Sodium Sulphate	10	1.009	1.018		1.037		1.056		1.075			
	25	1.006	1.015		1.033		1.052		1.070		1.089	1.108
Sodium Sulphide	18	1.010	1.021		1.044		1.067		1.091		1.115	1.139
Sodium Sulphite	19	1.008	1.017		1.036		1.056		1.075		1.095	1.115

Chemical Name	t °C	1 %	2 %	3 %	4 %	5 %	6 %	7 %	8 %	9 %	10 %	12 %
Sodium Thiocyanate	18	1.004	1.009		1.020		1.030		1.041		1.052	
Sodium Thiosulphate	20	1.007	1.015		1.032		1.048		1.065		1.083	1.100
Stannic Chloride	15	1.007	1.015		1.031		1.047		1.064		1.081	1.099
Stannous Chloride	15	1.007	1.015		1.031		1.047		1.064		1.081	1.099
Sulphuric Acid	0		1.015			1.036					1.074	
	20	1.005	1.012	1.018	1.025	1.032	1.039	1.045	1.052	1.059	1.066	1.080
	100		0.971			0.989					1.021	
Sulphurous Acid	15.5	1.004	1.009		1.019		1.029		1.039		1.049	
Z												
Zinc Chloride	20		1.017		1.035		1.053		1.072		1.090	1.109
Zinc Sulphate	20		1.019		1.040		1.062		1.084		1.107	1.131

Table 13.08 Density ρ (kg/dm³) of aqueous solutions as a function of concentration (w: 15 to 65 %) and temperature

Chemical Name	t °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
A												
Acetaldehyde	19	1.003			0.997			0.986		0.971		0.947
Acetic Acid	0		1.034		1.049		1.062		1.073		1.081	
	20	1.020	1.026	1.033	1.038	1.044	1.049	1.053	1.058	1.061	1.064	1.067
	60		1.004		1.011		1.018		1.023		1.028	
Acetone	20		0.972		0.958		0.941		0.921		0.899	
	60		0.947		0.929		0.907		0.884		0.859	
Aluminium Nitrate	18	1.126	1.175		1.281							
Aluminium Sulphate	19	1.164	1.226	1.292								
Ammonia	15	0.941	0.925	0.910	0.895	0.880	0.865	0.849	0.832	0.815	0.796	0.776
	20	0.940	0.923	0.907	0.892							
	50	0.924	0.906	0.888	0.870	0.853	0.835	0.818				
Ammonium Acetate	18	1.030	1.039	1.048	1.057							
Ammonium Bromide	25	1.086	1.119		1.190		1.270					
Ammonium Carbonate	15	1.051	1.068	1.084	1.100	1.116	1.129					
Ammonium Chloride	20	1.043	1.057	1.070								
Ammonium Iodide	18	1.102	1.141	1.183	1.227	1.275	1.326	1.385	1.442			
Ammonium Nitrate	20	1.061	1.083		1.128	1.151	1.175		1.226	1.252	1.279	

Chemical Name	t °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
	40		1.073		1.116		1.163		1.213		1.266	
	100		1.041		1.082		1.127		1.175		1.227	
Ammonium Sulphate	20	1.087	1.115	1.144	1.172	1.200	1.228		1.283			
Ammonium Thiocyanate	18	1.033	1.045	1.057	1.065		1.087		1.111			
Arsenic Acid	15	1.105	1.145	1.187	1.233	1.283	1.337	1.396	1.460	1.530	1.607	1.690
B												
Barium Acetate	18	1.116	1.160	1.207	1.255	1.307	1.361					
Barium Chloride	20	1.145	1.203	1.266								
Barium Hydroxide	15	1.142	1.213	1.287	1.360							
C												
Cadmium Chloride	20	1.143	1.199	1.260	1.327	1.401	1.483	1.575	1.676			
Calcium Chloride	– 30			1.263	1.309							
	– 10	1.138	1.199	1.253	1.298							
	20		1.178	1.228	1.282	1.337	1.396					
Calcium Bisulphite	20		1.040									
Calcium Hydroxide	20	1.092	1.126	1.161	1.199							
Calcium Nitrate	18	1.119	1.164	1.211	1.259	1.311	1.366	1.423				
Chloric Acid	18	1.092	1.127									
Chrome Alum, green	15	1.140	1.193	1.251	1.315	1.383	1.456	1.533	1.615			

Chemical Name	t °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
Chromic Acid	15	1.119	1.163	1.210	1.260	1.313	1.371	1.435	1.505	1.581	1.663	
Citric Acid	18	1.058	1.079	1.101	1.124	1.148	1.172	1.197	1.222			
Copper Chloride	20	1.149	1.205									
Copper Sulphate	20	1.167										
D												
Diethylene Glycol	– 10		1.043				1.076				1.104	
	20		1.034				1.062				1.088	
	140		0.953				0.975				0.993	
E												
Ethanol	– 20						0.963		0.944		0.923	
	0	0.980	0.976	0.971	0.965	0.958	0.949	0.940	0.930	0.918	0.907	0.896
	15	0.977	0.971	0.964	0.957	0.948	0.939	0.929	0.918	0.907	0.895	0.884
	80	0.942	0.932	0.922	0.910	0.899	0.887	0.875	0.863	0.851	0.839	0.827
Ethylene Glycol	– 40										1.110	
	– 20					1.070	1.079	1.087	1.096			
	0		1.033				1.061				1.090	
	40		1.019				1.042				1.063	
	100		0.980				1.000				1.018	

Chemical Name	*t* °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
F												
Ferric Chloride	20	1.133	1.182	1.234	1.291	1.353	1.417	1.485	1.551			
Ferric Nitrate	18	1.127	1.175	1.228								
Ferric Sulphate	17.5		1.181	1.241	1.307	1.376	1.449	1.528	1.613	1.703	1.798	
Ferrous Chloride	18	1.144	1.200	1.260								
Ferrous Sulphate	18	1.156	1.214									
Formic Acid	20	1.037	1.049	1.061	1.073	1.085	1.096	1.109	1.121	1.132	1.142	1.154
	40		1.037		1.059		1.080					
G												
Glycerol	0	1.039	1.052	1.065	1.079	1.093	1.108	1.122	1.136	1.150	1.164	1.178
	20	1.035	1.047	1.060	1.073	1.086	1.099	1.113	1.126	1.140	1.154	1.168
	60	1.018	1.030	1.043	1.054	1.067	1.079	1.093	1.115	1.118	1.131	1.144
	100		1.o25				1.053				1.140	
H												
Hydrobromic Acid	20	1.113	1.158	1.206	1.258	1.315	1.377	1.445	1.517	1.595	1.679	1.768
Hydrochloric Acid	0	1.080	1.107	1.134	1.161	1.188						
	20	1.073	1.098	1.124	1.149	1.174	1.198					

Chemical Name	*t* °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
	100	1.033	1.058	1.081	1.103	1.122						
Hydrocyanic Acid	18	0.972	0.958	0.943	0.925	0.908	0.892	0.876	0.860	0.844	0.826	0.809
Hydrofluoric Acid	20	1.054	1.070	1.087	1.102	1.116	1.128	1.142	1.155			
Hydrofluorsilicic Acid	17.5	1.127	1.173	1.222	1.273							
Hydrogen Peroxide	0	1.059	1.080	1.100	1.121	1.143	1.165	1.187	1.209	1.232		
	18	1.054	1.073	1.092	1.112	1.133	1.154	1.175	1.197	1.219	1.242	1.265
L												
Lactic Acid	20	1.020			1.060				1.120			
Lead Acetate	18	1.197	1.166	1.217	1.271	1.330	1.399					
Lead Nitrate	18	1.145	1.203	1.266	1.329							
Lithium Bromide	20	1.117	1.162	1.210	1.263	1.320	1.384	1.454				
Lithium Chloride	20	1.085	1.115	1.146	1.179	1.217	1.254					
M												
Magnesium Chloride	20	1.128	1.176	1.226	1.279							
	30	1.124	1.171	1.221	1.272							
Magnesium Sulphate	20	1.160	1.220	1.283								
Manganese Chloride	18	1.134	1.185	1.240	1.299							
Manganese Nitrate	18	1.124	1.172	1.223	1.278	1.337	1.399	1.466	1.538	1.615		
Manganese Sulphate	15	1.160	1.221	1.286	1.357							

Chemical Name	t °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
Methanol	– 40								0.955		0.937	
	– 20			0.980	0.970		0.956		0.942		0.923	
	0	0.978	0.972	0.967	0.961	0.954	0.946	0.937	0.929	0.919	0.909	0.898
	15.6	0.975	0.968	0.961	0.954	0.946	0.937	0.928	0.919	0.909	0.898	0.887
N												
Nickel Chloride	20	1.155	1.215	1.280	1.353							
Nickel Nitrate	20	1.138	1.169	1.249	1.311	1.377						
Nickel Sulphate	18	1.171										
Nitric Acid	0	1.093	1.126	1.159	1.195	1.231	1.265	1.299	1.332	1.364	1.393	1.420
	20	1.084	1.115	1.147	1.180	1.214	1.246	1.278	1.310	1.339	1.367	1.391
	100	1.034	1.60	1.086	1.112	1.138	1.164	1.188	1.212	1.234	1.255	1.275
O												
Oleum (% stands for %free SO_3)	15		1.920		1.957		1.979		2.009		2.020	
	35	1.872	1.892	1.913	1.928	1.944	1.958	1.966	1.973	1.977	1.974	1.965
P												
Phosphoric Acid	20	1.082	1.113	1.146	1.181	1.216	1.254	1.293	1.355	1.379	1.426	1.475
Potassium Bisulphate	18	1.110	1.150	1.193								
Potassium Bromide	20	1.116	1.160	1.208	1.259	1.315	1.375					

Chemical Name	t °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
Potassium Carbonate	– 10			1.249	1.302	1.361	1.417					
	20		1.190		1.298	1.355	1.414	1.476	1.540			
Potassium Chloride	20	1.097	1.133									
Potassium Cyanide	15	1.077										
Potassium Ferricyanide	20	1.085	1.115									
Potassium Ferrocyanide	20	1.105										
Potassium Hydroxide	0	1.146	1.195	1.246	1.299	1.352	1.407	1.465	1.526			
	15	1.140	1.188	1.239	1.291	1.344	1.399	1.456	1.514			
	50	1.123	1.171	1.221	1.272	1.325	1.379	1.435	1.494			
Potassium Iodide	20	1.119	1.166	1.217	1.271	1.331	1.396	1.467	1.546			
Potassium Nitrate	20	1.097	1.133									
Propanol	– 20		0.990						0.937			
	0	0.983	0.979	0.972	0.963	0.953	0.943	0.933	0.923	0.913	0.903	0.893
	30	0.973	0.964	0.954	0.944	0.933	0.923	0.912	0.902	0.891	0.881	0.870
Propylene Glycol	– 20						1.053	1.059				
	0			1.030	1.036	1.042	1.044	1.052				
	20			1.024	1.028	1.033	1.038	1.043				
	100			0.979	0.980	0.982	0.984	0.986				

Chemical Name	*t* °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
S												
Silver Nitrate	20	1.139	1.194	1.255	1.321	1.303	1.474		1.668		1.916	
Sodium Bisulphate	20	1.119	1.161									
Sodium Bisulphite	15	1.156	1.203	1.244	1.284	1.323						
Sodium Bromide	20	1.126	1.175	1.227	1.284	1.346						
Sodium Carbonate	20	1.159	1.215	1.274								
	30	1.153	1.209	1.267	1.327							
Sodium Chlorate	18	1.105	1.145	1.187	1.231	1.278	1.329					
Sodium Chloride	– 10	1.120	1.162	1.205								
	20	1.109	1.148	1.189								
Sodium Dichromate	15	1.105			1.207	1.244	1.279	1.312	1.342			
Sodium Hydroxide	0	1.174	1.229	1.285	1.340	1.393	1.443	1.492	1.540			
	20	1.164	1.219	1.274	1.328	1.380	1.430	1.478	1.525			
	100	1.117	1.170	1.223	1.275	1.326	1.375	1.422	1.469			
Sodium Hypochlorite	20				1.230							
Sodium Nitrate	20	1.104	1.143	1.184	1.226	1.270	1.318					
Sodium Sulphide	18	1.176										
Sodium Sulphite	19	1.145										
Sodium Thiosulphate	20	1.127	1.174	1.223	1.274	1.327	1.383					

Chemical Name	*t* °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
Sodiumsilicates												
$Na_2 \cdot 2{,}06\ SiO_2$	20	1.145	1.200	1.260	1.321	1.385	1.450					
$Na_2 \cdot 3{,}36\ SiO_2$	20	1.130	1.179	1.235	1.290							
Stannic Chloride	15	1.126	1.173	1.223	1.278	1.337	1.403	1.475	1.555	1.644	1.742	1.851
Stannous Chloride	15	1.126	1.174	1.227	1.284	1.346	1.415	1.490	1.573	1.666	1.770	1.887
Sulphuric Acid	0	1.112	1.150	1.191	1.232	1.275	1.317	1.363	1.411	1.462	1.515	1.571
	20	1.102	1.139	1.178	1.219	1.260	1.303	1.348	1.395	1.445	1.498	1.553
	30	1.097	1.134	1.172	1.212	1.253	1.295	1.340	1.387	1.437	1.490	1.545
	50	1.086	1.122	1.159	1.198	1.238	1.281	1.325	1.372	1.422	1.474	1.528
	75	1.070	1.106	1.142	1.180	1.220	1.263	1.301	1.353	1.402	1.454	1.507
	100	1.054	1.089	1.125	1.163	1.203	1.245	1.289	1.335	1.384	1.435	1.487
Z												
Zinc Chloride	20	1.137	1.187	1.238	1.293	1.352	1.417	1.489	1.568	1.655	1.749	1.851
Chemical Name	*t* °C	15 %	20 %	25 %	30 %	35 %	40 %	45 %	50 %	55 %	60 %	65 %
Zinc Sulphate	20	1.168	1.232	1.304	1.378							

The values for the density ρ are taken from reference literature. They are sufficiently accurate for necessary calculations e.g. for the mass or the volume flow, pressure and pump power input.

Intermediate values can be calculated by linear interpolation.

for intermediate values of concentration w: $\rho = \rho_1 + \dfrac{w - w_1}{w_2 - w_1} \cdot (\rho_2 - \rho_1)$

for intermediate values of temperature t: $\rho = \rho_1 + \dfrac{t - t_1}{t_2 - t_1} \cdot (\rho_2 - \rho_1)$

13.09 Density ρ (kg/dm³) of aqueous solutions as a function of concentration (w: 70 to 100%) and temperature

Chemical Name	t °C	70 %	75 %	80 %	85 %	90 %	95 %	96 %	97 %	98 %	99 %	100 %
A												
Acetaldehyde	19	0.923			0.854							0.783
Acetic Acid	0	1.087		1.090		1.086						
	20	1.069	1.070	1.070	1.069	1.066	1.061	1.059	1.057	1.055	1.052	1.050
	60	1.030		1.029		1.023						
Acetone	20	0.876		0.850		0.822						
	40	0.855		0.828		0.799						
Ammonium Nitrate	100	1.283		1.342		1.408						
	160								1.422		1.433	
	200								1.406		1.412	
Arsenic Acid	15	1.781										
D												
Diethylene Glycol	– 20			1.132								
	0			1.120								1.135
	20			1.098								1.122
	140			1.014								1.028
E												
Ethanol	– 50	0.926		0.902		0.872						
	– 20	0.901		0.877		0.851						
	0	0.884	.0872	0.860	0.848	0.835	0.821					0.806
	15	o.872	0.860	0.848	0.835	0.822	0.809	0.906	0.803	0.800	0.797	0.794
	80	0.814	0.801	0.789	0.776	0.762	0.749					0.735

Chemical Name	*t* °C	70 %	75 %	80 %	85 %	90 %	95 %	96 %	97 %	98 %	99 %	100 %
Ethylene Glycol	– 40			1.136								
	0			1.113								1.128
	40			1.085								1.098
	100			1.039								1.054
F G H												
Formic Acid	20	1.166	1.177	1.186	1.195	1.204	1.214	1.216	1.217	1.218	1.220	1.221
Glycerol	0			1.221								1.273
	20	1.181	1.195	1.209	1.222	1.235	1.248	1.251	1.253	1.256	1.259	1.261
	60	1.158	1.171	1.184								1.238
	100			1.155								1.210
Hydrocyanic Acid	18	0.792	0.775	0.758		0.724						0.691
Hydrogen Peroxide	18	1.290	1.315	1.341	1.367	1.393	1.420					1.447
M N O												
Methanol	– 60	0.932		0.910		0.863						
	– 40	0.917		0.894		0.845						
	0	0.887	0.875	0.863	0.851	0.838	0.824	0.821	0.819	0.816	0.813	0.810
	15.6	0.875	0.863	0.850	0.837	0.824	0.810	0.807	0.804	0.802	0.799	0.796
Nitric Acid	0	1.444	1.465	1.485	1.502	1.517	1.529					1.550
	20	1.413	1.434	1.452	1.469	1.483	1.493	1.495	1.497	1.501	1.506	1.513
	50	1.369	1.387	1.404	1.420	1.432	1.443					1.461
	100	1.294	1.311	1.326	1.339	1.351	1.363					1.382
Oleum (% stands for % of free SO_3)	15	2.018		2.008		1.990						1.984
	35	1.955	1.941	1.925	1.908	1.889	1.866	1.861	1.855	1.849	1.843	1.837

Chemical Name	t °C	70 %	75 %	80 %	85 %	90 %	95 %	96 %	97 %	98 %	99 %	100 %
P												
Phosphoric Acid	20	1.526	1.579	1.633	1.689	1.746	1.807	1.819	1.832	1.844	1.857	1.870
Propanol	– 20	0.903										0.838
	0	0.884	0.874	0.863	0.853	0.843	0.832					0.819
	20	0.868	0.858	0.847	0.837	0.827	0.816					0.804
	30	0.860	0.850	0.839	0.829	0.819	0.808					0.796
Propanol (Iso-P.)	20	0.860	0.848	0.836	0.824	0.812	0.799					0.785
S Z												
Stannic Chloride	15	1.971										
Sulphuric Acid	0	1.629	1.687	1.748	1.801	1.836	1.855					1.851
	20	1.611	1.669	1.727	1.779	1.814	1.834	1.836	1.836	1.836	1.834	1.831
	50	1.584	1.642	1.697	1.747	1.783	1.804					1.801
	100	1.542	1.597	1.649	1.697	1.733	1.757					1.758
Zinc Chloride	20	1.962										

The values for the density ρ are taken from reference literature. They are sufficiently accurate for necessary calculations e.g. for the mass or the volume flow, pressure and pump power input.

Intermediate values can be calculated by linear interpolation.

for intermediate values of concentration w: $$\rho = \rho_1 + \frac{w - w_1}{w_2 - w_1} \cdot (\rho_2 - \rho_1)$$

for intermediate values of temperature t: $$\rho = \rho_1 + \frac{t - t_1}{t_2 - t_1} \cdot (\rho_2 - \rho_1)$$

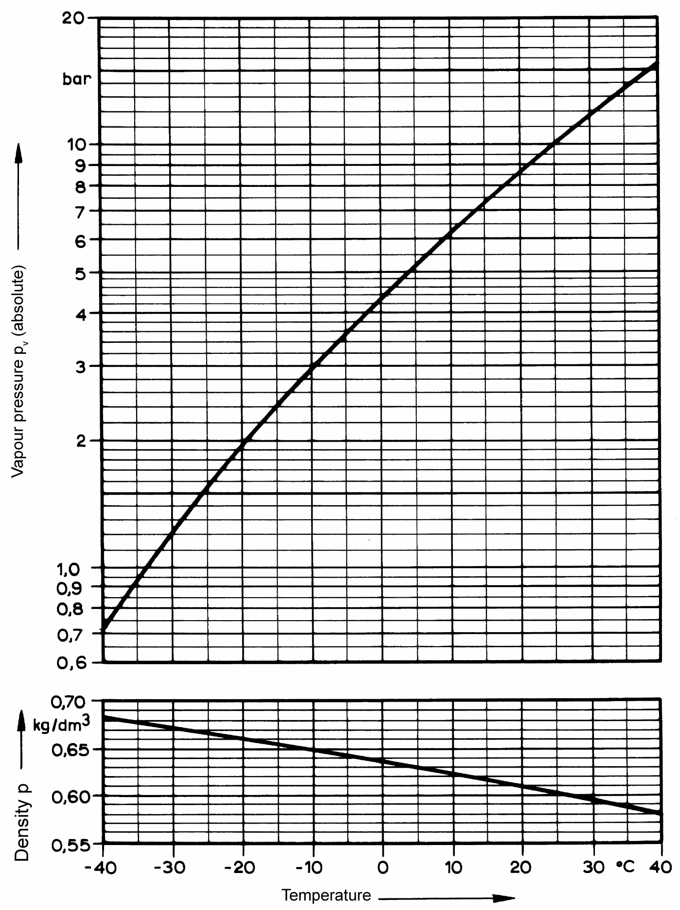

Fig 13.10 Density ρ and vapour pressure p_v of ammonia dependent on temperature

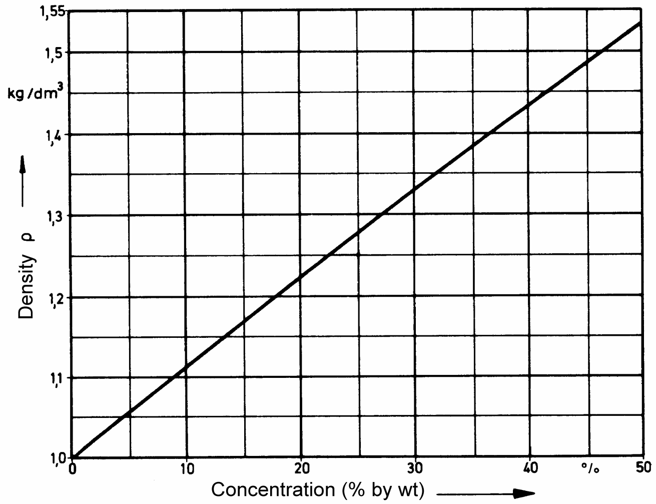

Fig 13.11 Aqueous caustic soda solution NaOH density ρ at *t* = 15°C dependent on concentration

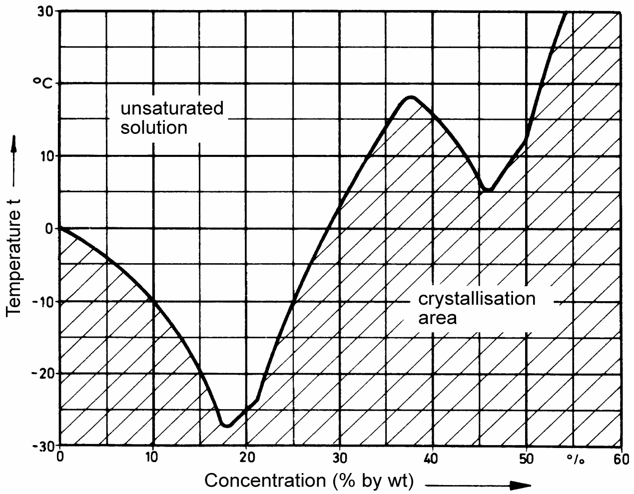

Fig 13.12 Aqueous caustic soda solution NaOH, crystallisation dependent on temperature and concentration

Table 13.13 Various liquid densities at 20°C

Name	Density kg/dm³
Alcohol (non-aqueous)	0.79
Antifreeze-aqueous mixture	
23 Vol %	1.03
38 Vol %	1.04
54 Vol %	1.06
Beer	1.02 - 1.04
Crude oil	
Arabia	0.85
Iran	0.835
Kuwait	0.87
Libya	0.83
Trinidad	0.885
Venezuela	0.935
Diesel fuel	0.81 - 0.85
Fuel oil	
EL	0.83 - 0.85
L	0.86 - 0.91
M	0.92 - 0.99
S (Bunker C)	0.95 - 0.1
Gear oil	0.92
Hydraulic oil	0.875
Jet fuel	
JP-3	0.74 - 0.78
JP-4	0.75 - 0.80
JP-5	0.79 - 0.84
Kerosene	0.78 - 0.82
Linseed oil	0.93
Machine oil	
light	0.88 - 0.90
medium	0.91 - 0.935

Name	Density kg/dm³
Mercury	13.546
Milk	1.2 - 1.05
Olive oil	0.915 - 0.919
Petroleum	0.79 - 0.82
Petroleum spirit	
light-	0.68 - 0.72
normal-	0.72 - 0.74
super-	0.75 - 0.78
Plant oils	0.904 - 0.974
Quench oil	0.9 - 0.93
Rape seed oil	0.91
Salt solution	
5%	1.035
15%	1.109
28% (saturated)	1.189
Sea water	1.02 - 1.045
Sugar solution	
10%	1.04
20%	1.08
40%	1.18
60%	1.28
Tar	1.22 - 1.24
Tar oil	
Brown coal	0.88 - 0.92
Pit coal	0.9 - 1.1
Transformer oil	0.88
Turbine oil	0.91
Turpentine	0.86 - 0.87
Wine	0.99 - 1.0

Table 13.14 Density of Solids

Name	Density kg/dm³	Name	Density kg/dm³
Aluminium	2.70	Marble	2.5 - 2.8
Aluminium oxide	3.8 - 3.9	Masonry	
Amber	1.0 - 1.1	brick	1.4 - 1.6
Basalt	2.9	quarry stone	2.5
Borax	1.75	sandstone	2.0
Brass	8.1 - 8.6	Nickel	8.35 - 8.90
Bronze	8.7	Nickel-Silver	8.5
Carbon		Paper	0.7 - 1.2
antimony impregnated	2.1 - 2.3	Platinum	21.15
resin impregnated	1.75 - 1.83	PP	0.90 - 0.907
Ceramic	2.25 - 2.5	PTFE	2.0 - 2.3
Chromium	7.14	Pumice stone	0.4 - 0.9
Clay		Rock salt	2.15
fresh	2.6	Sand	
dry	1.8	dry	1.4 - 1.6
Coal	1.35	wet	to 2.0
Coke	1.6 - 1.9	Silicon carbide	3.05 - 3.21
Concrete	1.8 - 2.5	Silver	10.42-10.53
Copper	8.63 - 8.80	Snow, loose	
Cork	0.24	dry	0.125
Diamond	3.5	wet	to 0.95
Fluro Polymer	1.85 - 2.22	Steel	7.85
Glass	2.4 - 2.6	Tin	7.2
Gold	19.25	Titanium	4.5
Granite	2.50 - 3.05	Wood, air dry	
Gypsum	2.32	hickory	0.6 - 0.9
Ice, at 0 °C	0.9167	oak	0.7 - 1.0
Lead	11.34	pine	0.35 - 0.6
Lignite	1.2 - 1.5	Zinc	6.95 - 7.15
Limestone	2.3 - 3.2	Zirconium oxide	5.56

Table 13.15 Kinematic viscosity ν of water in 10^{-6} m²/s dependent on pressure and temperature

Pressure	Temperature in °C						
bar	0	25	50	75	100	150	200
5	1.791	0.8928	0.5537	0.3877	0.2942	0.1991	
10	1.790	0.8925	0.5536	0.3877	0.2942	0.1991	
20	1.786	0.8919	0.5536	0.3878	0.2944	0.1993	0.1555
30	1.783	0.8912	0.5535	0.3879	0.2945	0.1994	0.1556
40	1.779	0.8906	0.5534	0.3880	0.2946	0.1996	0.1558
50	1.776	0.8900	0.5534	0.3881	0.2948	0.1997	0.1559
60	1.773	0.8894	0.5533	0.3882	0.2949	0.1999	0.1561
70	1.770	0.8888	0.5533	0.3883	0.2951	0.2001	0.1562
80	1.767	0.8882	0.5532	0.3884	0.2952	0.2002	0.1564
90	1.763	0.8876	0.5532	0.3885	0.2953	0.2004	0.1565
100	1.760	0.8871	0.5531	0.3886	0.2955	0.2005	0.1567
150	1.745	0.8843	0.5529	0.3890	0.2962	0.2013	0.1574
200	1.731	0.8816	0.5527	0.3895	0.2969	0.2020	0.1582

Table 13.16 Isobaric specific heat c_p of water in kJ/kgK, dependent on pressure and temperature

Pressure	Temperature in °C						
bar	0	25	50	75	100	150	200
20	4.218	4.178	4.177	4.186	4.213	4.307	4.486
40	4.207	4.172	4.173	4.182	4.208	4.301	4.475
80	4.186	4.161	4.164	4.173	4.199	4.288	4.453

Table 13.17 Specific heat c_p of various liquids in kJ/kgK, at constant pressure, dependent on temperature

Name	Temperature in °C					
	– 75	– 50	– 25	0	20	50
Ammonia		4.45	4.53	4.61	4.74	5.08
Butane	2.052	2.119	2.194	2.278	2.480	2.713
Butene	1.959	2.022	2.102	2.186	2.273	2.437
Carbon dioxide		1.80	2.04	2.28		
Propane	2.119	2.202	2.630	2.456	2.592	2.840
Propene	2.123	2.177	2.435	2.590	2.784	3.286

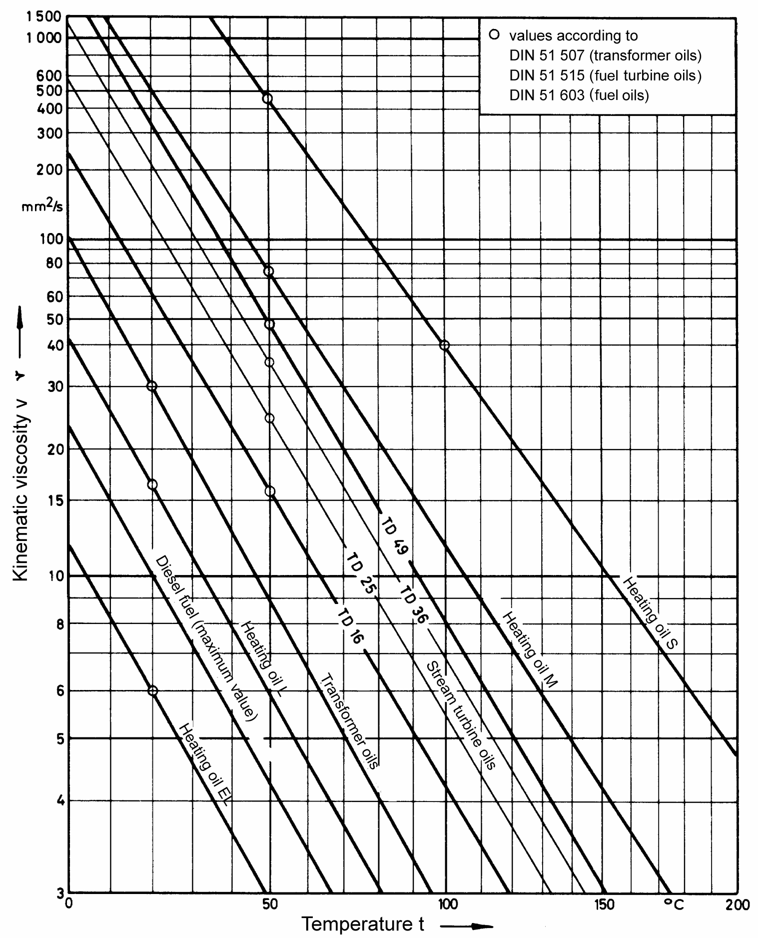

Table 13.18 Kinematic viscosity ν of various mineral oil distillates

The values shown are mean values and therefore for guidance only.
The scales chosen give straight lines for the viscosity - temperature relationship, which means it is possible to interpolate the relationship for other mineral oil distillates not shown.

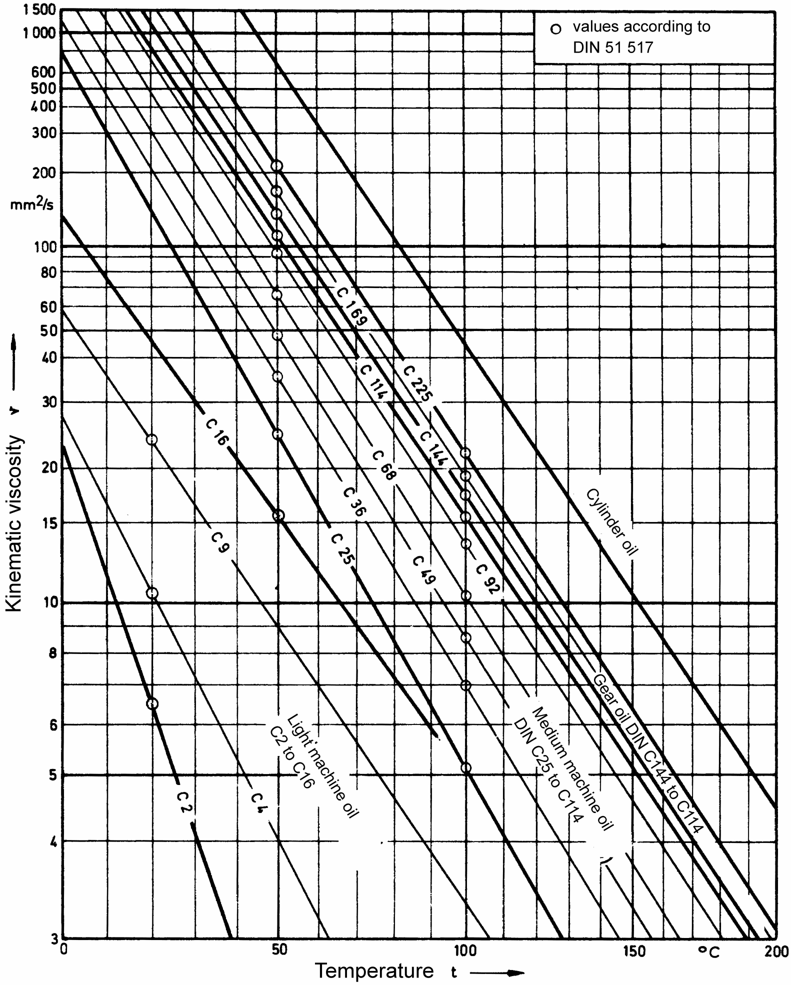

Table 13.18 (cont.) Kinematic viscosity ν of various mineral oil distillates

Given the viscosity for two temperatures, the points are marked and a straight line drawn between them. If only one figure is known, this is plotted and the straight line drawn at a similar slope to its nearest neighbour.

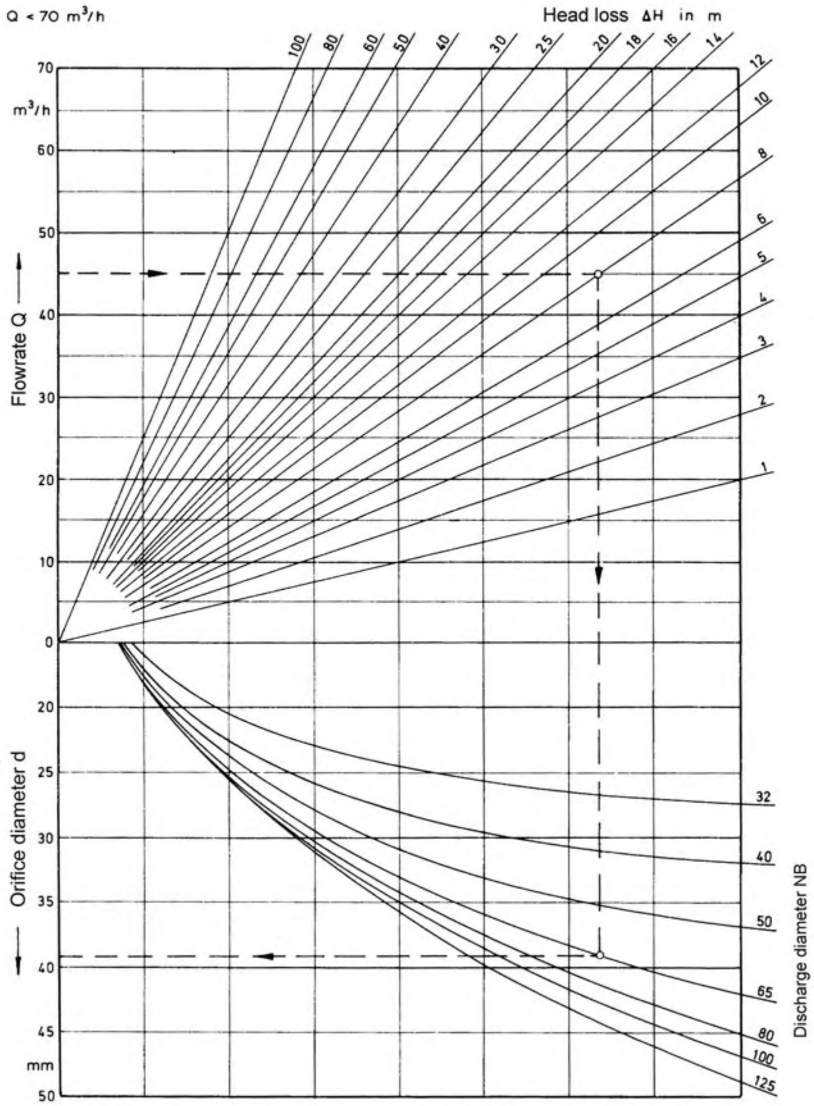

Table 13.19 Determining the orifice diameter for throttling

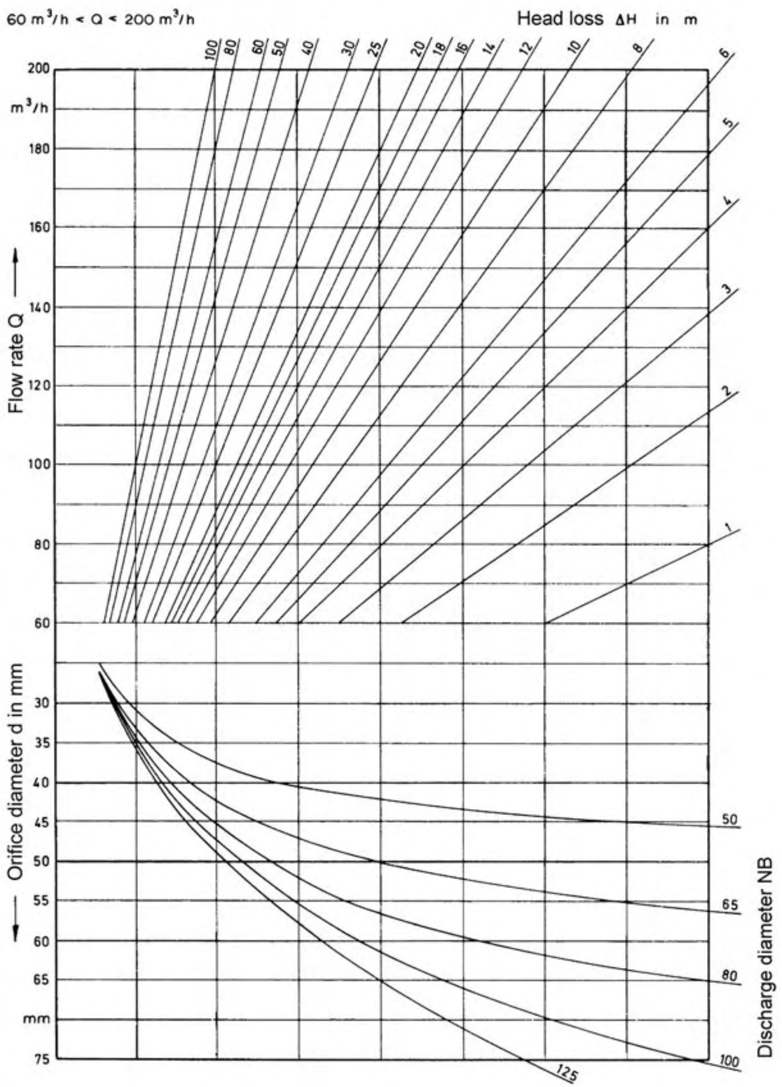

Table 13.19 (cont.) Determining the orifice diameter for throttling

Table 13.20 Standards for Centrifugal Pumps (Extract)

Standard	Edition	Title
Pump Ranges, Nominal Duty Points, Main Dimensions		
EN 733	08.95	End-suction centrifugal pumps, rating 10 bar with bearing bracket (EN 733, 1994)
EN 734	08.95	Side channel pumps PN 40 (EN 734, 1994)
EN 735	08.95	Overall dimensions of rotodynamic pumps Tolerances (EN 735, 1994)
DIN EN 22858	07.93	End-suction centrifugal pumps, rating 16 bar (ISO 2858: 1975, EN 22858: 1993)
VDMA 24252	04.91	End-suction centrifugal pumps, rating 10 bar (washing machine pumps) with bearing bracket
VDMA 24253	02.71	End-suction centrifugal pumps, strengthened singleflow, single entry
Technical Specifications		
DIN ISO 5199	12.00	Technical specifications for centrifugal pumps, Class II (mainly for Pumps acc. to EN 22858)
DIN ISO 9905	03.97	Technical specifications for centrifugal pumps, Class I (similar to API 610 Standard)
DIN ISO 9908	04.98	Technical specifications for centrifugal pumps, Class III (mainly for Pumps acc. to EN 733)
VDMA 24279	04.93	Technical specifications for centrifugal pumps, magnetic couplings and canned motors
API 610	08.95	Centrifugal Pumps for Petroleum, Heavy Duty Chemical and Gas Industry Service (8th. Edition)
Guarantee requirements and Tests		
EN ISO 9906	08.02	Rotodynamic pumps - Hydraulic performance acceptance tests - Grade 1 and 2
DIN 24273	01.98	Pumps and pump sets for liquids, materials and testing installation
VDMA 24276	05.94	Liquid pumps for chemical installations, quality requirements
DIN 1944	10.68	Acceptance tests. Replaced by EN ISO 9906

Table 13.20 Continuation

Standard	Edition	Title
Accessories		
DIN EN 23661	08.94	End-suction centrifugal pumps, baseplate and installation dimensions (ISO 3661: 1977, EN 23661: 1993)
DIN 24259 - 1	03.79	Pumps and baseplates general dimensions
DIN EN 12756	13.01	Mechanical seals; mounting dimensions, principal dimensions, designation and material codes
Various		
DIN EN 809	10.98	Pumps and pumpsets with safety requirements
EN 12262	02.99	Rotodynamic pumps - Technical documents - terms, delivery range, layout
DIN EN 12462	10.98	Biotechnical performance requirements for pumps
DIN EN 12723	a) 09.00	Liquid pumps - general terms for pumps and installations - definitions, quantities, letter symbols and units
DIN 24296	01.98	Pumps and pump sets, spare parts

Explanations:

API	American Petroleum Institute
DIN	German Standard German Institute for Standardization
EN	European Standard European Committee for Standardization
ISO	International Standard International Organization for Standardization
VDMA	Deutscher Maschinen- und Anlagenbau e.V. (German Machinery and Plant Association)

Hydrochloric Acid (20% HCl)	– 0.3	< 0	ultra acidic
Hydrochloric Acid (3,6% HCl)	0.1	0 to 3	strongly acidic
Sulphuric Acid (4,9% H_2SO_4)	0.3		
Hydrochloric Acid (0,36% HCl)	1.1		
Sulphuric Acid (0,49% H_2SO_4)	1.2		
Hydrochloric Acid (0,036% HCl)	2.0		
Lemon juice	2.3		
Acetic Acid (6% CH_3-COOH)	2.4		
Vinegar	3.1	3 to 7	weakly acidic
Fruit juice	3.5		
Wine	3.8		
Marshy Water	4.0		
Beer	5.0		
Boric Acid (0,2% H_3BO_3)	5.2		
Milk	6.3		
Water, chemically pure	7.0	7	neutral
Seawater	8.1	7 to 11	weakly alkaline
Sodium Bicarbonate (0.42% $NaHCO_3$)	8.4		
Soap - suds	8.7		
Borax (1.9% $Na_2B_4O_7$)	9.2		
Ammonia. aqueous (0.017% NH_3)	10.6		
Soda Ash (0.53% Na_2CO_3)	11.3	11 to 14	strongly alkaline
Ammonia. aqueous (1.7% NH_3)	11.6		
Potassium Hydroxide (0.056% KOH)	12.0		
Sodium Hydroxide (0.04% NaOH)	12.0		
Lime Water. saturated	12.3		
Sodium Hydroxide (0.4% NaOH)	13.0		
Sodium Hydroxide (4% NaOH)	14.0		
Potassium Hydroxide (50% KOH)	14.5	> 14	ultra alkaline

Table 13.21 pH-values of various liquids

The practical scale of pH-values extends from 0 to 14. In some special cases also pH-values <0 and >14 are used.
The percentages quoted in brackets are percentage mass-proportions.

Table 13.22 Greek Alphabet

Α	α	Alpha	Ι	ι	Iota	Ρ	ρ	Rho
Β	β	Beta	Κ	κ	Kappa	Σ	σ	Sigma
Γ	γ	Gamma	Λ	λ	Lambda	Τ	τ	Tau
Δ	δ	Delta	Μ	μ	Mu	Υ	υ	Ypsilon
Ε	ε	Epsilon	Ν	ν	Nu	Φ	ϕ	Phi
Ζ	ζ	Zeta	Ξ	ξ	Xi	Χ	χ	Chi
Η	η	Eta	Ο	ο	Omikron	Ψ	ψ	Psi
Θ	θ	Theta	Π	π	Pi	Ω	ω	Omega

Table 13.23 Roman numerals

I	II	III	IV	V	VI	VII	VIII	IX
1	2	3	4	5	6	7	8	9
X	XX	XXX	XL	L	LX	LXX	LXXX	XC
10	20	30	40	50	60	70	80	90
C	CC	CCC	CD	D	DC	DCC	DCCC	CM
100	200	300	400	500	600	700	800	900
M								
1000								

Example: MCMLXXXVII = 1000 + 900 + 80 + 7 = 1987

Table 13.24 Old units

Length	
Chain (UK; US)	20,1168 m
Elle (yard, Prussia)	0,6669 m
Furlong (UK; US)	201,168 m
Foot (Prussia)	0,31385 m
Line (US)	2,54 mm
Line (Prussia)	2,18 mm

Area	
Acre (UK; US)	4047 m²
Mile of Land (UK)	2,5899 km²
Morgen (Germany)	2500 m²
Square foot (Germany)	0,0985 m²
Square rod (Prussia)	14,1843 m²
Rood (UK)	1011,71 m²

Volume	
Anker (Prussia)	34,35 L
Bushel (UK)	36,348 L
Cord (US)	3,62 L
Bucket (Prussia)	68,7 L
Barrel (Prussia)	2,29 hL
Load (UK)	2907,8 L

Weight	
Bale (US)	226.796 kg
Box (US)	11,34 kg
Hundredweight (cwt)(UK)	50,8 kg
Pound (Germany)	500 g
Stone (UK)	6,35 kg
Zentner (Germany)	50 kg

C

D

E

T

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