

Contents

Sr. no.	Description	Page no.	Sr. no.	Description	Page no.
1	Symbols, units, designations	1	8	General	19
2	Design	1	8.1	National & International standards for centrifugal pumps	19
2.1	Pump capacity	1	8.2	Shaft deflection	21
2.2	Pump head	1	8.3	Improving the NPSH requirements	21
2.3	System head	1	8.4	Impeller types	22
2.4	Speed	1	8.5	Pump types	23
2.5	Selecting the pump size	3	8.6	Pump installation arrangements	24
2.6	Calculating the power consumption	3	8.7	Pump sump configuration	25
2.6.1	Pump power input	3	8.8	Suction pipe layout	25
2.6.2	Calculating the drive rating	3	8.9	Shaft coupling	27
2.7	Pump characteristics curve	3			
2.8	System characteristic (piping characteristic)	4	9	Technical data	28
2.9	Operating point	4	9.1	Vapour pressure pd and density of water	28
2.10	Parallel operation of centrifugal pumps	4	9.2	Vapour pressure pd of various liquids	29
			9.3	Density of various liquids at atmospheric pressure	30
3	Suction characteristics	5	9.4	Extract of main legal units for centrifugal pumps	31
3.1	NPSH required	5	9.5	Conversion of British units and U.S. units	32
3.2	NPSH available	5	9.6	Graph for calculating flow velocity	34
			9.7	Graph for calculating velocity head	35
4	Pressure losses Pv	6	9.8	Graph for calculating velocity head differential	36
4.1	Head losses Hv in Straight pipes	6	9.9	Graph for calculating head losses	37
4.2	Head losses Hv in plastic pipes	8	9.10	Graph for calculating conversion factors for viscous liquids	38
4.3	Head losses Hv for viscous liquids	8	9.11	Graph for calculating conversion factors for viscous liquids	39
4.4	Head losses Hv in valves & fittings	9	9.12	Graph for calculating specific speed	40
			-	Schedule for calculating the operating point or pump	
5	Changing the pump performance	13			
5.1	Changing the speed	13			
5.2	Trimming the impeller	13			
6	Handling viscous liquids	14			
7	Typical selection examples	15			
7.1	Selecting the pump size	16			
7.2	Calculating the power consumption	16			
7.2.1	Pump power input	16			
7.2.2	Calculating the drive rating	16			
7.3	Calculating the drive rating	16			
7.3.1	Suction lift from open/closed tank	16			
7.3.2	Positive suction operation from open/closed tank	17			
7.3.3	Positive suction operation from closed tank at vapour pressure	18			
7.4	Changing the speed	18			
7.5	Trimming the impeller	18			
7.6	Handling viscous liquids	18			
7.6.1	Calculating the operating point	18			
7.6.2	Establishing the pump size	19			

1 Symbols, units and designation

A	m ²	Area
a	mm	Width
b ₂	m	Impeller outlet width
D	mm (m)	Impeller diameter, pipe diameter
DN	mm	Nominal bore of pipe
d	mm	Smallest inner diameter
F	N	Force
f _H	-	Conversion factor for head
f _Q	-	Conversion factor for flow rate
f _η	-	Conversion factor for efficiency
g	m/s ²	Gravitational constant
H	m	Head
H _A	m	System head
H _{geo}	m	Static head
H ₀	m	Shut-off head
H _{s geo}	m	Static suction lift
H _{z geo}	m	Static positive suction head
H _v	m	Head loss
H _{v,s}	m	Head loss - suction side
ΔH	m	Differential head
K	1	Coefficient
k	mm	Absolute roughness
L	m	Length of pipe
n	1/min.	Speed
NPSH _{req.}	m	NPSH required
NPSH _{av}	m	NPSH available
n _q	1/min.	Specific speed
P	kW	Pump power input
p	bar (N/m ²)	Pressure
p _b	bar (N/m ²)	Barometric pressure
p _D	bar (N/m ²)	Vapour pressure of liquid
p _v	bar (N/m ²)	Pressure loss
ΔQ	l/s (m ³ /hr.)	Differential capacity
Q	l/s (m ³ /hr.)	Capacity / flow rate
Q _{min.}	l/s (m ³ /hr.)	Minimum flow rate
R	mm	Radius
Re	1	Reynolds number
U	m	Circumference
v	m/s	Flow velocity
y	mm	Stroke
Z	1/h	Switching frequency
Z _{s,d}	m	Height differential between pump suction and discharge nozzles
ζ	-	Loss coefficient
η	-	Pump efficiency
λ	-	Pipe friction coefficient
μ	1	Correction coefficient
ν	m ² /s	Kinematic viscosity
ρ	kg/m ³ (kg/dm ³)	Density
φ	1	Temperature factor
φ	°	Opening angle

Indices

a	at outlet cross section of the system/branching off
B	at operating point
d	at discharge nozzle of pump/flowing through
e	at inlet cross section of plant/branching off
G	for Cast Iron
geo	geodetic
K	for plastic
s	suction side, at suction nozzle of pump
opt	at best efficiency point
R	radial
sch	for Sulphuric acid
W	for water
Z	for viscous liquids
1, 2, 3	consecutive numbers, items

2 Design

2.1 Pump capacity

The capacity Q is the external volume flow per unit of time in m³/s (l/s and m³/h are also commonly used). Balance water, leakage water etc. do not count as part of the capacity.

2.2 Pump head

The head H of a pump is the useful mechanical energy transmitted by the pump to the medium handled, related to the weight of the medium, expressed in m. It is independent of the density ρ of the medium handled, i.e. a centrifugal pump will generate the same head H for all fluids irrespective of the density ρ. The density ρ determines the pressure within the pump.

$$p = \rho \cdot g \cdot H$$

and influences the pump power input P.

2.3 System head

The total head of the system H_A is made up of the following (see figs. 1 & 2) :

□ H_{geo}, Static head = height difference between the suction and discharge fluid levels. If the discharge pipe emerges above the liquid level, then H_{geo} is referred to the center line of the outflow section.

□ $\frac{p_a - p_e}{\rho \cdot g}$ the pressure head difference between the suction and discharge fluid levels in closed tanks.

□ ΣH_v, the sum of all pressure head losses (pipe friction, friction in valves, fittings etc. in suction and discharge pipes).

□ $\frac{v_a^2 - v_e^2}{\rho \cdot g}$, the difference in the velocity heads in the tanks.

$$\text{The system head } H_A = H_{geo} + \frac{p_a - p_e}{\rho \cdot g} + \frac{v_a^2 - v_e^2}{\rho \cdot g} + \Sigma H_v.$$

In practice the difference between the velocity heads can be ignored, leaving

For closed tanks

$$H_A = H_{geo} + \frac{p_a - p_e}{\rho \cdot g} + \Sigma H_v.$$

For open tanks $H_A \approx H_{geo} + \Sigma H_v$.

2.4 Speed

With three-phase motor drives (asynchronous squirrel cage motor) the approximate pump speeds are as follows :

	No. of poles						
	2	4	6	8	10	12	14
Frequency	Reference speeds in curve documentation in 1/min.						
at 50 Hz	2900	1450	960	725	580	480	415
at 60 Hz	3500	1750	1160	875	700	580	500

In practice, however, motors usually run at slightly higher speeds which-upon consent of the customer - are taken into account by the pump manufacturer at the design stage (see section 7.4).

Different speeds are possible using a speed adjustment device, gearbox or belt drive.

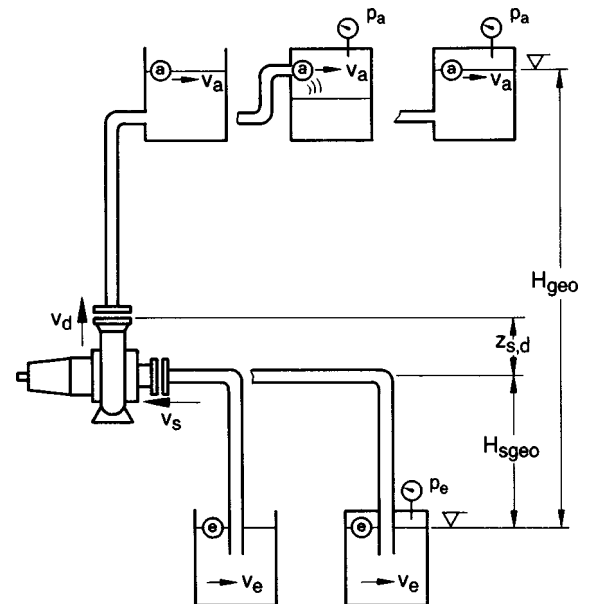
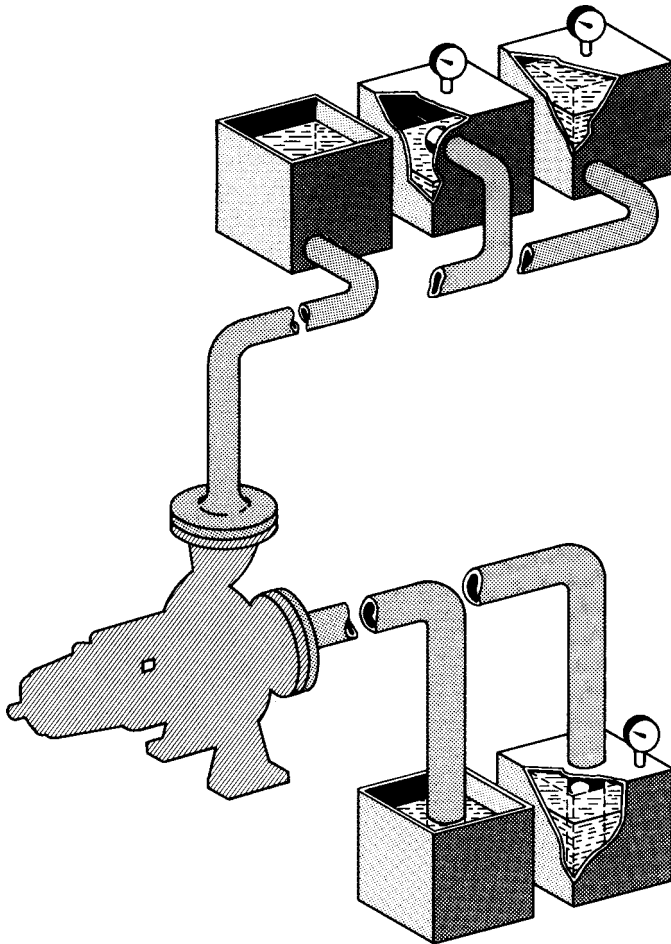


Fig. 1 : Pumping system with suction lift

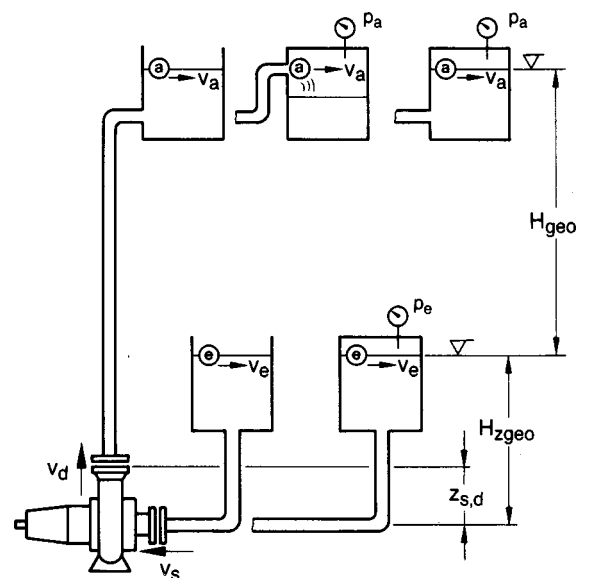
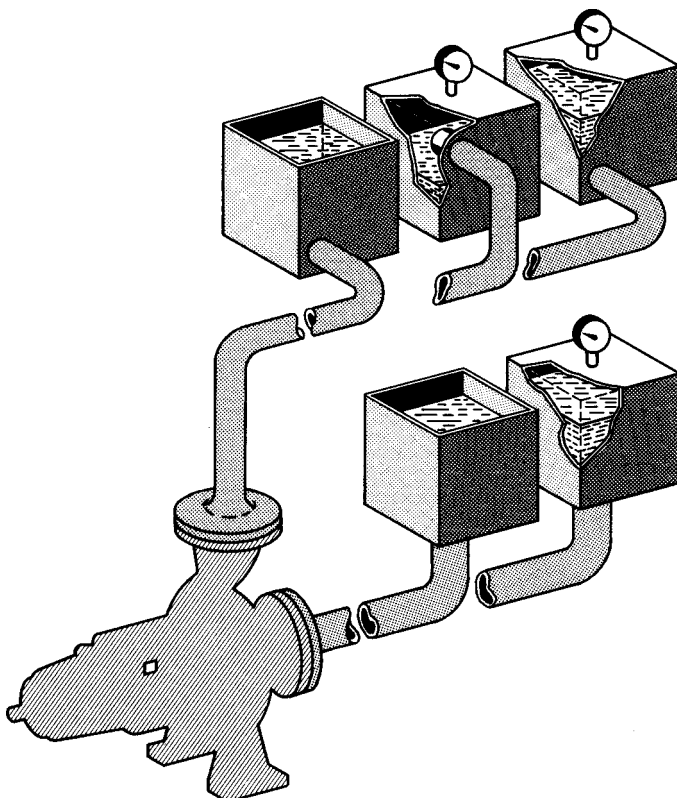


Fig. 2 : Pumping system with positive suction

2.5 Selecting the pump size (see 7.1)

The data needed for selecting the pump size - capacity Q and head H at the required duty point - is known, as is the mains frequency. The pump size and speeds can be determined from the performance chart (also called selection chart) (see 8.0 Fig. 26); then the other parameters of the pump selected, such as efficiency η , input power P and NPSH, can be established from the appropriate individual curve (see 8.0, Fig. 3).

Unless there is a particular reason to the contrary, arrange the operating point near Q_{opt} . (b.e.p.).

For pumps handling viscous liquids see sections 6 and 7.6.2.

2.6 Calculating the Power consumption

2.6.1 Pump power input (see annexure 7.2.1)

The pump power input P of a centrifugal pump is the mechanical energy at the pump coupling or pump shaft absorbed from the drive. It is determined using the following equation.

$$P = \frac{\rho \cdot g \cdot Q \cdot H}{1000 \cdot \eta} \text{ in kW}$$

with ρ in kg/dm^3
 g in m/s^2
 Q in l/s
 H in m
 η between 0 and 1

or another equation which is still used.

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

with ρ in kg/dm^3
 g in m/s^2
 Q in m^3/hr
 H in m
 367 conversion factor

The pump power input P in kW can be directly read with sufficient accuracy off the characteristic curves (see 2.7) where the density $\rho = 1000 \text{ kg/dm}^3$. The pump power input P must be converted (see 7.2.1) for other densities ρ .

2.6.2 Calculating the Drive rating (see example under 7.2.2)

Since it is possible that the system volume flow, and thus the operating point, will fluctuate, which could mean an increase in the pump power input P , it is standard practice to use the following safety margins when determining the motor size, unless the customer specifies otherwise :

up to 7.5 kW approx. 20%
 from 7.5 to 40 kW approx. 15%
 from 40 kW approx. 10%

If extreme volume flow fluctuations are expected, the motor size must be selected with reference to the maximum possible pump capacity on the characteristics curves, taking the following into consideration:

- impeller diameter required
- condition $NPSH_{av} > NPSH_{req}$. (see 3.2)
- permissible P/n values for the bearings

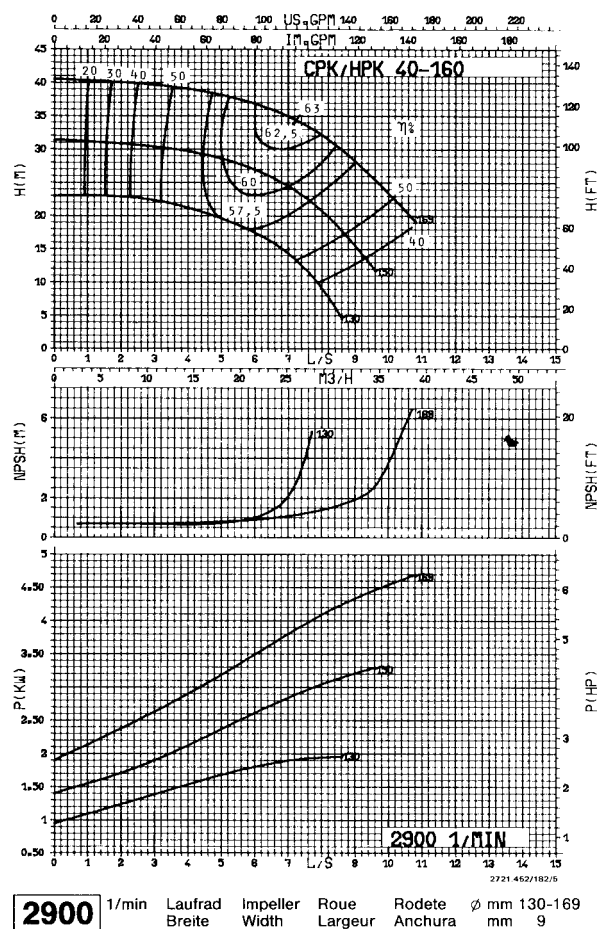
Handling liquids with a high proportion of solids, as well as handling pulp, means using special pumps and/or special impellers.

2.7 Pump characteristic curve

In contrast to positive displacement pumps (e.g. reciprocating pumps) at constant speed ($n = \text{const.}$) centrifugal pumps have a capacity Q which will increase if the head decreases. They are thus capable of self-regulation. The pump power input P , and therefore the efficiency η , plus the $NPSH_{req}$, depend on the capacity.

The behavior and relationship of all these variables are shown by the curves (see fig. 3) which thus illustrate the operating characteristics of a centrifugal pump.

The characteristic curves apply to the density ρ and kinematic viscosity ν of water, unless stated otherwise.



The duty conditions determine which is the more favorable - a flat or a steep curve. With a steep curve the capacity changes less than with a flat curve under the same differential head conditions ΔH (see fig. 4). The steep curve thus possesses better control characteristics.

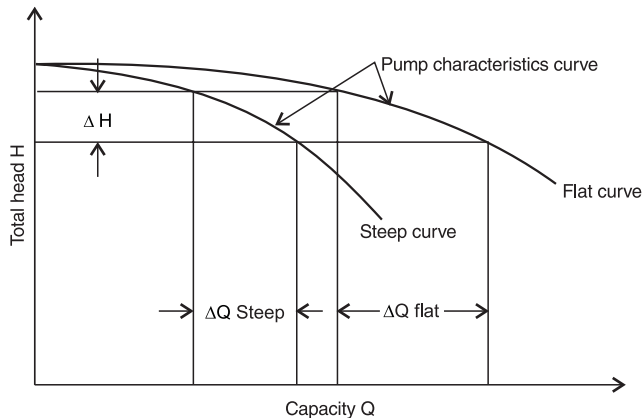


Fig. 4 : Steep and flat curve pump characteristics curve

2.8 System characteristics (piping characteristics)

The system head H_A is plotted against the capacity Q to give the system curve (piping curve) (fig. 5). This curve is made up of the static and dynamic characteristics of the installation.

The static part consists of the static head H_{geo} , which is independent of the capacity, and the difference in pressure head between the system inlet and outlet section $\frac{p_a - p_e}{\rho \cdot g}$.

The latter does not apply with open tanks (see fig. 1 and 2).

The dynamic part consists of the head loss H_v , which increases quadratically with the capacity (see 4.1) and the difference in the velocity head between the system inlet and outlet section $\frac{V_a^2 - V_e^2}{2g}$.

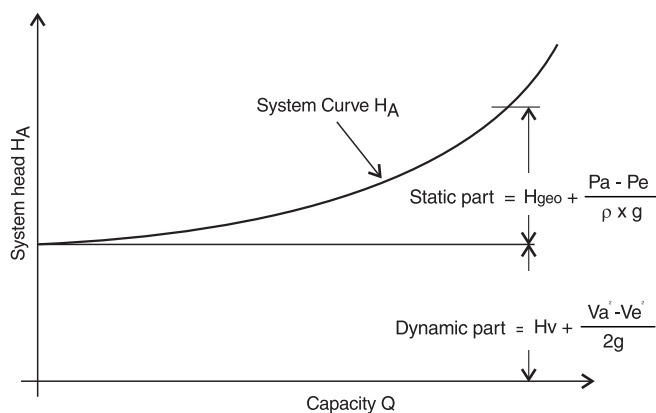


Fig. 5 : System (piping) characteristics

2.9 Operating point

Every centrifugal pump will establish an operating point B which is the point of intersection between the pump curve (QH curve) and the system curve H_A , i.e. the operating point B (and with it the capacity Q and the head H) can with radial impellers generally only be changed by altering speed n (see 5.1), the impeller diameter D (see 5.2) or by modifying the system characteristics H_A , always assuming this does not increase the risk of cavitation (see figs 6 and 7).

The only practical ways to modify the system characteristics when handling solid free, normal viscosity liquids are to increase or reduce the piping friction (i.e. by opening or closing the valve, changing the piping diameter, incrustations etc.) or to alter the static part (e.g. by increasing or reducing the tank pressure or the water level).

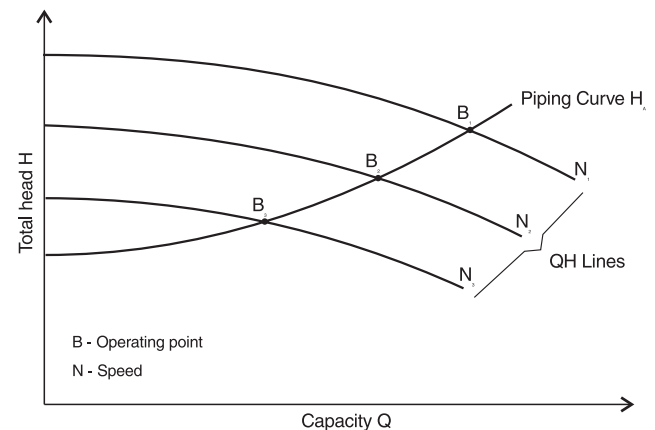


Fig. 6 : Changing the position of the operating point from B_1 to B_2 on the system curve H_A by rising the pump speed N_1 to N_2 .

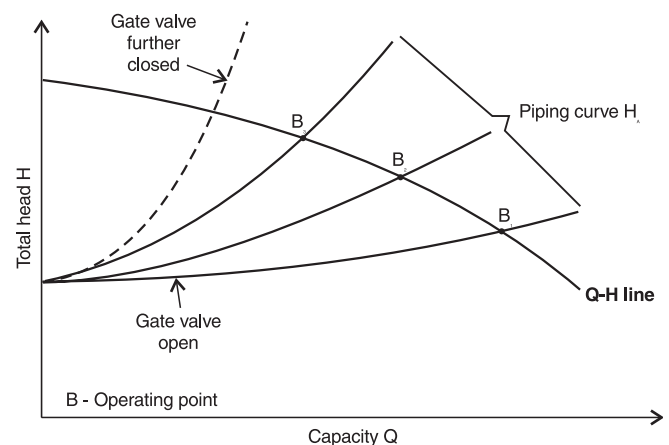


Fig. 7 : Changing the position of the operating point from B_1 to B_2 on the QH line by progressively closing the valve.

2.10 Parallel operation of centrifugal pumps

Where one pump is unable to deliver the required capacity Q at the operating point B , it is possible to have two or more pumps working in parallel in the same piping system. The pumps should preferably (for economic operation) be of the same type (see 8.5 pump types) and have the same shut-off head.

In the example (fig.8) each pump is designed for $0.5 \times Q$ at the same head.

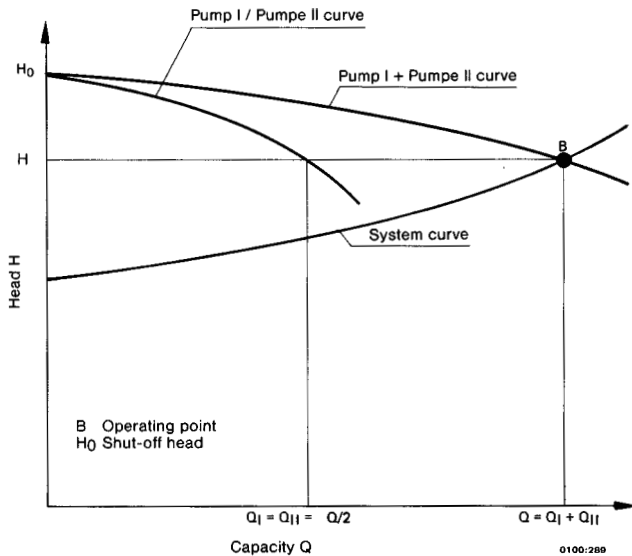


Fig. 8 : Parallel operation of two similar centrifugal pumps with the same shut-off head

Fig. 9 shows an alternative solution : two pumps with the same shut-off head H_0 but different capacities Q_1 and Q_2 pumping at a given operating point B in one piping system. Q_1 of pump 1 and Q_2 of pump 2 combine to produce the total capacity Q_1+Q_2 at the same head H.

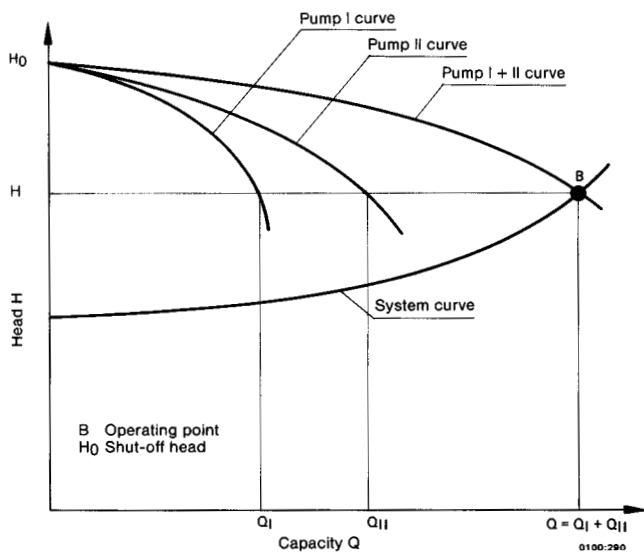


Fig. 9 : Parallel operation of 2 pumps with the same shut-off head H_0

3 Suction characteristics

3.1 NPSH required ($NPSH_{req.}$)

Centrifugal pumps will only operate satisfactorily if there is no buildup of vapour (cavitation) within the pump. Therefore the pressure head at the NPSH datum point must exceed the vapour pressure head of the medium handled. The NPSH datum point is the impeller centre, i.e. the point of intersection between the pump shaft center line and the plane at right angles to the pump shaft and passing through the outer points of the vane inlet edge.

The $NPSH_{req.}$ is the value required by the pump and is expressed in meters on the pump characteristics curves. The value often includes a safety margin of 0.5m.

3.2 NPSH available ($NPSH_{av}$)

The datum point for the $NPSH_{av}$ is the centre of the pump's suction nozzle. With standard, horizontal volute casing pumps the centrelines of the suction nozzle and impeller are on the same level (Figs. 10 & 11), i.e. the geodetic height is 0. However, if there is a difference of geodetic height (e.g. with vertical pumps) it has to be taken into account.

$NPSH_{av}$ is calculated as follows :

a) Suction lift operation; the pump is above the liquid level (Fig. 10).

$NPSH_{av}$ is defined as :

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

However, with a cold liquid, e.g. water, and an open tank,

i.e. $p_b = 1 \text{ bar} (= 10^5 \text{ N/m}^2)$

$p_e = 0 \text{ bar}$

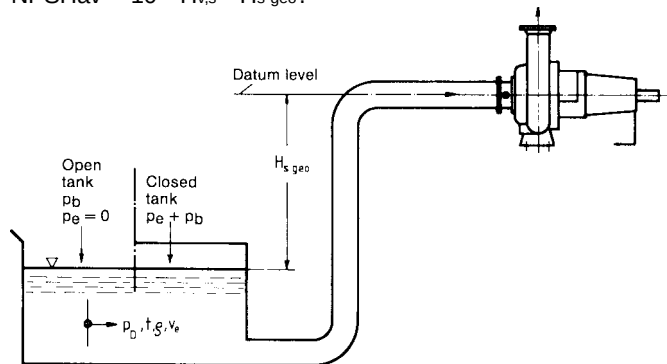
$\rho = 1000 \text{ kg/m}^3$

$g = 10 \text{ m/s}^2$ (incl. 2% error on 9.81 m/s^2)

$v_e^2/2g$ = can be eliminated because of the negligible velocity head in the tank.

The following simplified version is used in practice :

$$NPSH_{av} = 10 - H_{v,s} - H_{s,geo}$$



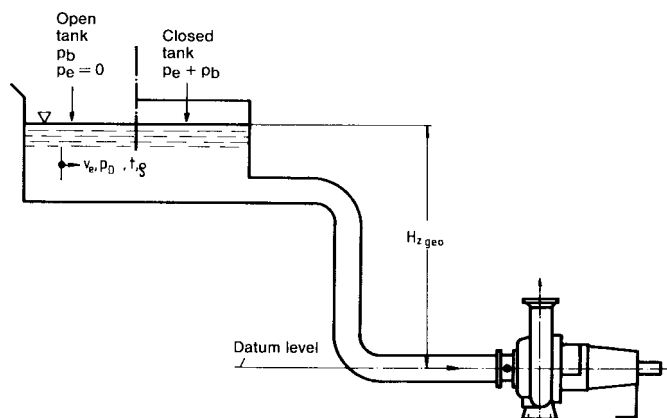
b) Suction head operation; the pump is below the liquid level (fig. 11)

$NPSH_{av}$ is defined as :

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

The following equation is used in practice, assuming the same conditions as in a) :

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



In all cases the following is a prerequisite for cavitation-free operation

$$NPSH_{av} \geq NPSH_{req}$$

4 Pressure loss Pv

The pressure loss P_v is the pressure differential arising as a result of wall friction and internal friction in piping runs, fittings, valves and fittings etc.

$$p_v = \frac{\lambda \cdot U \cdot L}{4A} \times \frac{\rho \cdot v^2}{2}$$

Where P_v = Pipe friction loss
 λ = pipe friction coefficient
 U = wetted periphery of section A through which the fluid flows
 L = Length of pipe
 ρ = Density of the medium pumped
 v = Flow velocity across a section A characteristics of the pressure loss

Straight lengths of circular cross-section piping are defined by the following equation :

$$p_v = \frac{\lambda \cdot L}{D} \times \frac{\rho \cdot v^2}{2}$$

Where D = bore of pipe

The pipe friction coefficient λ varies with the state of flow of the medium and the internal surface finish of the pipeline through which the medium is flowing. The state of the flow is determined by the REYNOLDS number (model laws).

$$Re = \frac{v \cdot D}{\nu}$$

For non-circular sections

$$Re = \frac{v \cdot 4A}{\nu \cdot U}$$

Where ν = kinematic viscosity

Table 1 : Mean peak-to-valley heights k (absolute roughness)

Material	Condition of pipe interior	1	5	10	50	100	500	1000	5000	10 ⁴
Steel	new, seamless, skin acid-cleaned									
	galvanized									
	straight-welded, skin bituminized									
	galvanized cemented									
	riveted									
Cast iron	used, moderately rusty									
	slight incrustation									
	heavy incrustation									
	after cleaning									
Asbestos-cement	new, with skin bituminized									
	galvanized									
	cemented									
	used, moderately rusty									
Heavy-clay (drain.)	slight incrustation									
	heavy incrustation									
	after cleaning									
Concrete	new, unfinished									
	with smooth finish									
	Spun concrete									
	new, unfinished									
Reinforced concrete	with smooth finish									
	new, with smooth finish									
	used, with smooth finish									
All concretes	drawn									
	new, not embrittled									
	new									
	after long exposure to water									
Masonry										

1) Non-ferrous metals, light alloys

λ can be calculated for smooth bore pipes (new rolled steel pipes):
in the region of laminar flow in the pipe ($Re < 2320$) the friction coefficient is :

$$\lambda = \frac{64}{Re}$$

in the region of turbulent flow in the pipe ($Re > 2320$) the test results can be represented by an empirical equation by ECK :

$$\lambda = \frac{0.309}{\left(\lg \frac{Re}{7}\right)^2}$$

In the region of $2320 < Re < 10^8$ the deviations are less than 1%.

Fig. 12 shows, that λ is solely dependent on the parameter D/k at relatively high REYNOLDS numbers; k/D is the "relative roughness", obtained from the "absolute roughness" k and the pipe bore wall surface roughness (coarseness).

According to MOODY the following applies

$$\lambda = 0.0055 + \frac{0.15}{\sqrt[3]{\frac{D}{K}}}$$

Table 1 gives rough approximations of k .

4.1 Head losses H_v in straight pipes

Fig. 13 gives the losses of head H_v per 100 m of straight pipe run for practical usage. The head losses H_v in this context are calculated according to :

$$H_v = \zeta \cdot \frac{v^2}{2 \cdot g}$$

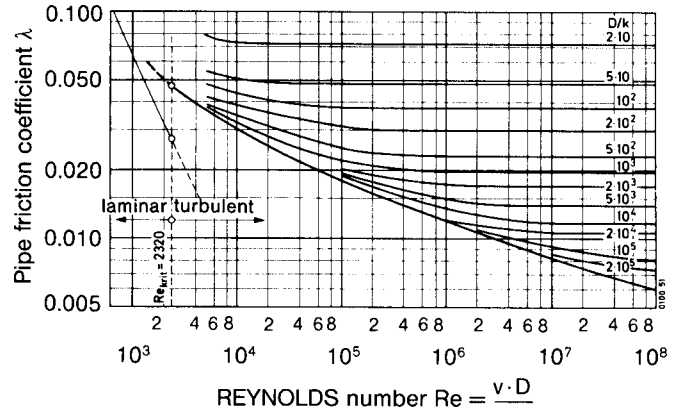


Fig. 12 : Pipe friction coefficient λ in function of REYNOLDS number and of relative wall roughness D/k

where

- ζ loss coefficient
- v flow velocity
- g gravittational constant

The values in fig. 13 apply to clean water at 20°C and to fluids of equal kinematic viscosity, assuming the piping is completely filled, and consists of new Cast Iron pipes, with an internal bitumen coating ($k=0.1\text{mm}$). The head losses H_v of fig. 13 should be multiplied by :

- 0.8 for new rolled steel pipes
- 1.7 for pipes with incrustations (the reduced pipe cross section due to the incrustations is the determining factor),
- 1.25 for old slightly rusty steel pipes

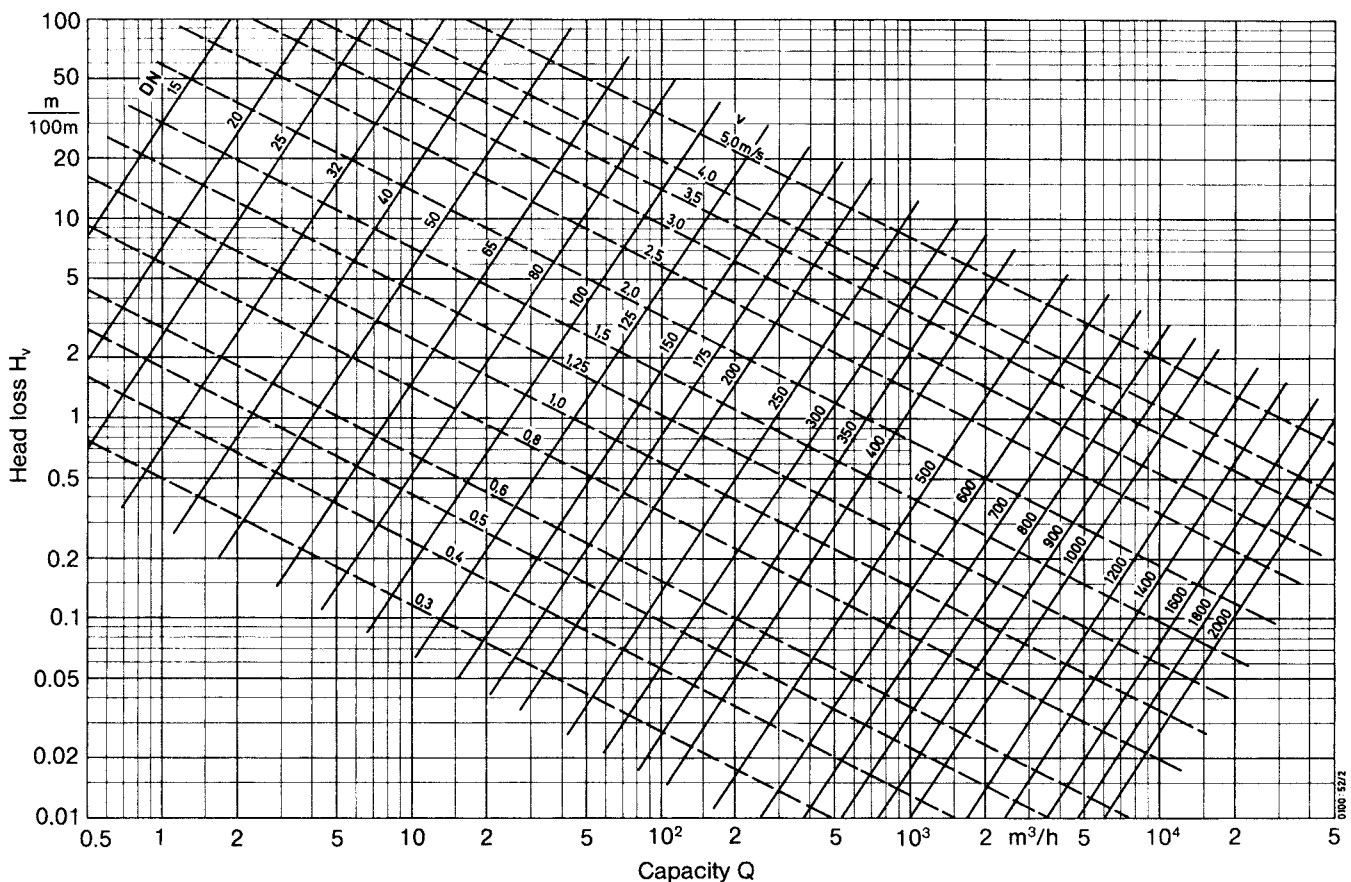


Fig. 13 : Head losses in straight pipes (cast iron pipe, new condition) from 'DIN 15 to 2000 mm and for capacities Q from 0.5 to 50000 m^3/hr . (flow velocity v in m/s , nominal bore in mm , water at 20 °C)

In case of pipes with very heavy intrusions, the actual head loss can only be determined by experiments. Deviations from the normal diameter have a profound effect on the head loss. e.g. an actual bore of 0.95 times the nominal bore (i.e. only a slightly bore reduction) pushes up the head loss up H_v to 1.3 times the "as new" loss. New rubber hoses and rubber lined canvas hoses have H_v values approximately equal to those indicated in fig. 13.

How to use fig. 13 - an example

Assuming a rate of flow $Q=140 \text{ m}^3/\text{hr.}$ and a new Cast Iron pipe, inside diameter $D=150 \text{ mm}$, we obtain; head loss $H_v = 3.25 \text{ m}/100 \text{ m}$ pipe length, flow velocity $v=2.2 \text{ m/s}$.

4.2 Head losses H_v in plastic pipe

Head losses in plastic pipes H_{vk} . The head losses of PVC and polyethylene "hard" and "soft" (drawn) plastic pipes are approximately equal. For the practical calculation of H_{vk} , the respective head losses for Cast Iron pipes H_{vG} (fig. 13) should be multiplied by the correction coefficients μ of fig. 14, which are dependent on the flow velocity v . The head losses evaluated in this way apply to water at a temperature of 10°C .

If the water temperature is other than 10°C , these head losses must in addition be multiplied by a temperature factor ϕ (Fig. 15). Thus,

$$H_{vk} = H_{vG} \cdot \mu \cdot \phi$$

where,

H_{vk} head losses in plastic pipes,

H_{vG} head losses in cast iron pipes according to Fig. 13

μ correction coefficient according to Fig. 14

ϕ temperature factor according to Fig. 15

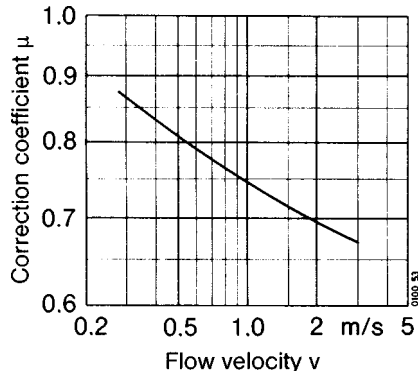


Fig. 14 : Correction coefficient μ for conversion of head losses in a cast iron pipe at 20°C water temperature to values in a plastic pipe at 10°C water temperature; plotted in function of flow velocity v .

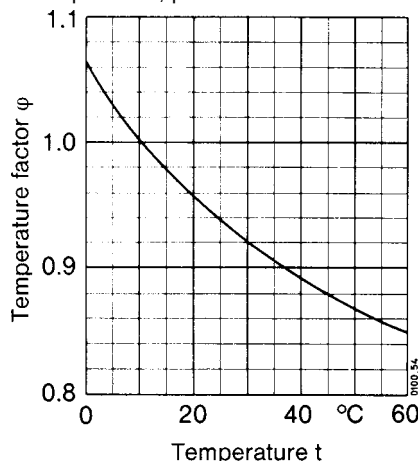


Fig. 15 : Temperature factor ϕ for calculation of head losses in plastic pipes at water temperatures between 0 and 60°C

Increments of 20 to 30% should be added for sewage or untreated water

4.3 Head losses H_v for viscous liquids in straight pipes

The head loss of a viscous fluid (subscript FI) can be ascertained for practical purpose with the aid of fig. 16., after having obtained the head loss for cold water (20°C , $v = 10^{-6} \text{ m}^2/\text{s}$) (subscript W) from fig. 13 :

$$H_{vFI} = \frac{\lambda_{FI} \cdot H_{vW}}{\lambda_W}$$

See viscosity for conversion of viscosity values.

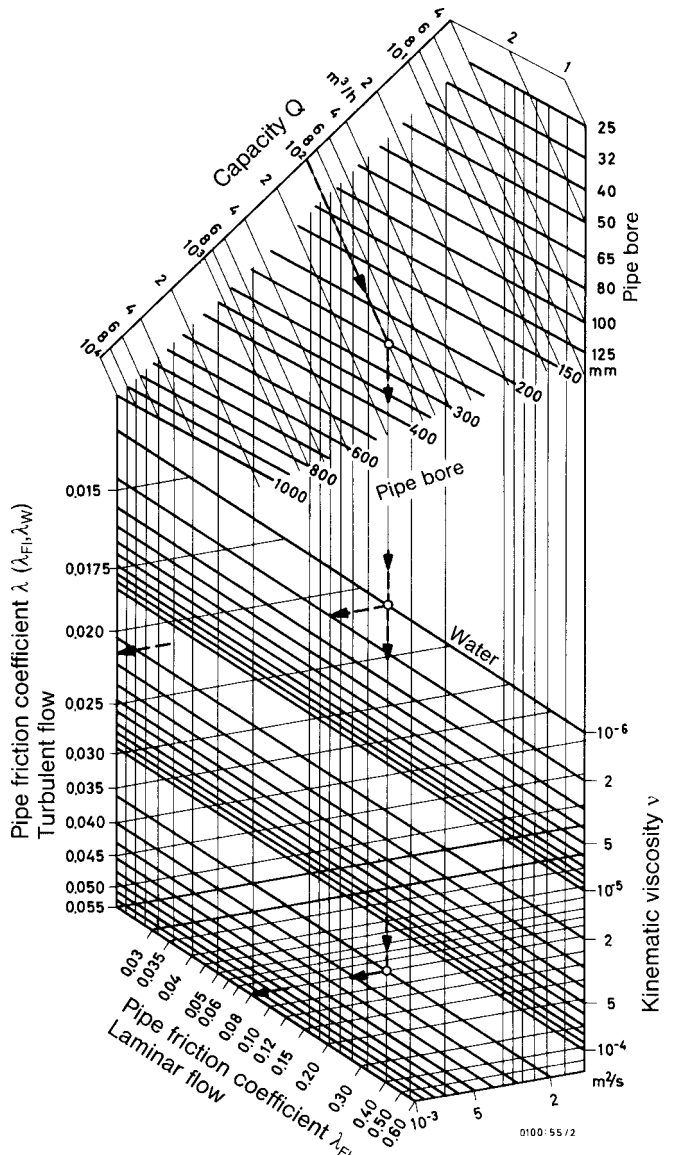


Fig. 16 : Resistance coefficients λ for flow of viscous fluids in straight pipes

How to use figures 16 - an example :

Given : capacity $Q = 100 \text{ m}^3/\text{hr.}$, new Cast Iron pipe, inside diameter $D = 250 \text{ mm}$, kinematic viscosity $\nu = 2 \times 10^{-4} \text{ m}^2/\text{s}$. Found in figs. 13 : $H_{vW} = 0.14 \text{ m}/100 \text{ m}$.

It follows from figure 16 that : $\lambda_{FI} = 0.08$, $\lambda_W = 0.021$.

$$\text{Thus, } H_{vFI} = \frac{0.08 \times 0.14 \text{ m}}{0.021 \times 100 \text{ m}}$$

One quite common viscous fluid is cellulose (pulp pumping), the viscosity of which depends on the velocity, since the material in question is "non-NEWTONian". Fig. 17 a through 17 f offer reference values for the head losses H_v per 100m length of straight Steel pipe run plotted against capacity Q ($H_v = f(Q)$; nominal bore : 100, 150, 200, 250, 300 and 350 mm) for conveying unbleached sulfite cellulose at 15°C , 26°SR

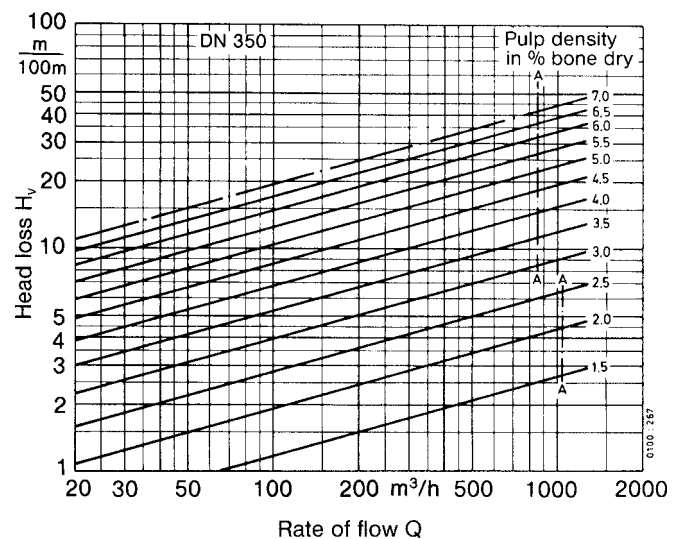
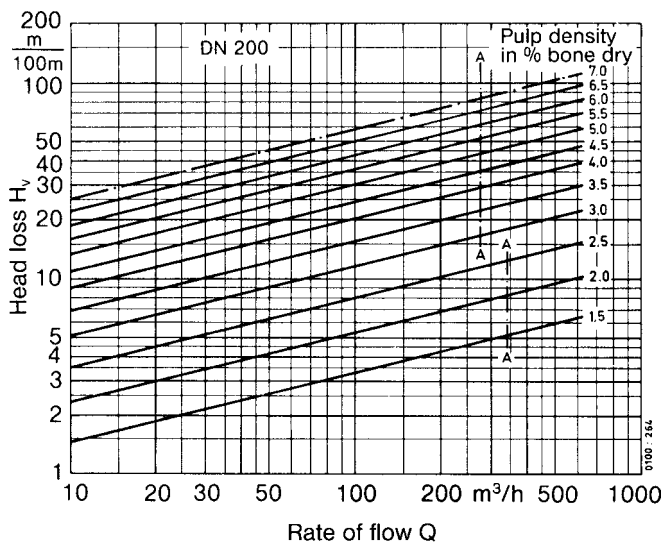
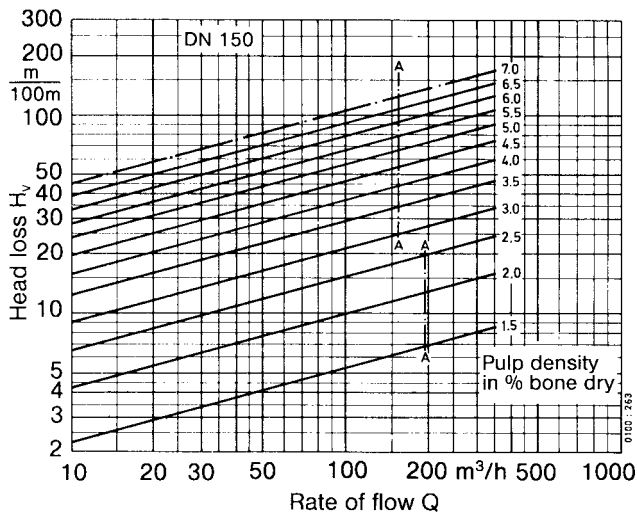
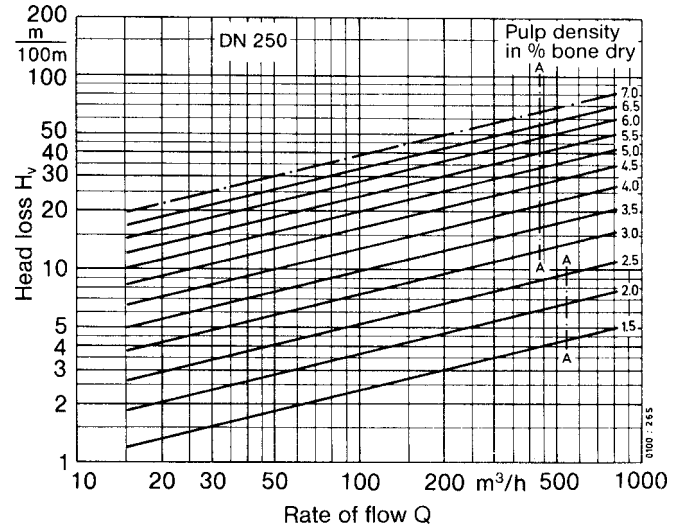
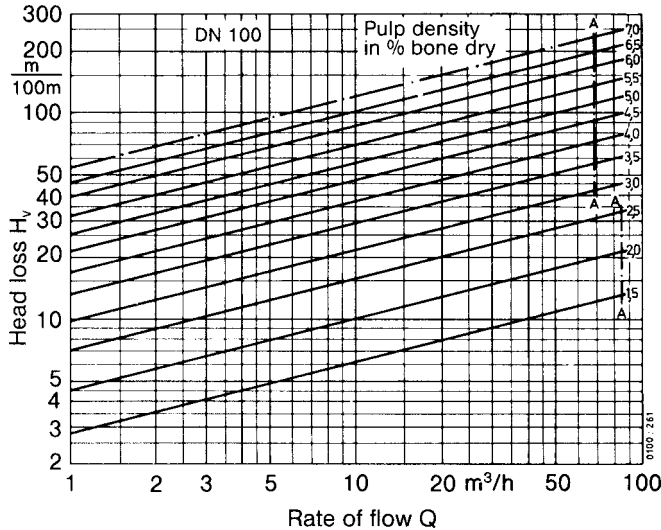
(grinding state, °SR - Schopper-Riegler degree of freeness) and with a pulp density (pulp pumping) of 1.5 to 7% bone dry.

If the pump slurry concerned differs from that used for purpose of plotting the curves of Fig. 17, then the values obtained from Fig. 17 should be multiplied by the following factors :

$K = 0.9$ for bleached sulphite - sulphate cellulose, waste paper pulp

$K = 1.0$ for boiled (digested) wood pulp

$K = 1.4$ for white and brown raw wood pulp.



Furthermore, the head loss obtained from fig. 17, and if necessary corrected by one of the factors listed above, should be corrected additionally if the pulp slurry concerned is at a temperature higher than 15 °C. In this case, 1% of the head loss value which applies to 15 °C should be deducted for every 2° C of temperature difference. In the case of plastic pipes, the H_{VK} value is obtained by multiplying the H_V value for steel pipes by 0.9.

The head loss value is reduced even further if fillers such as kaolin (China Clay) are contained in the plup slurry concerned. For an 18% kaolin content, the head loss value will decrease by 12%, and for a 26.5% kaolin content, it will decrease by 16%.

4.4 Head losses H_V in valves and fittings

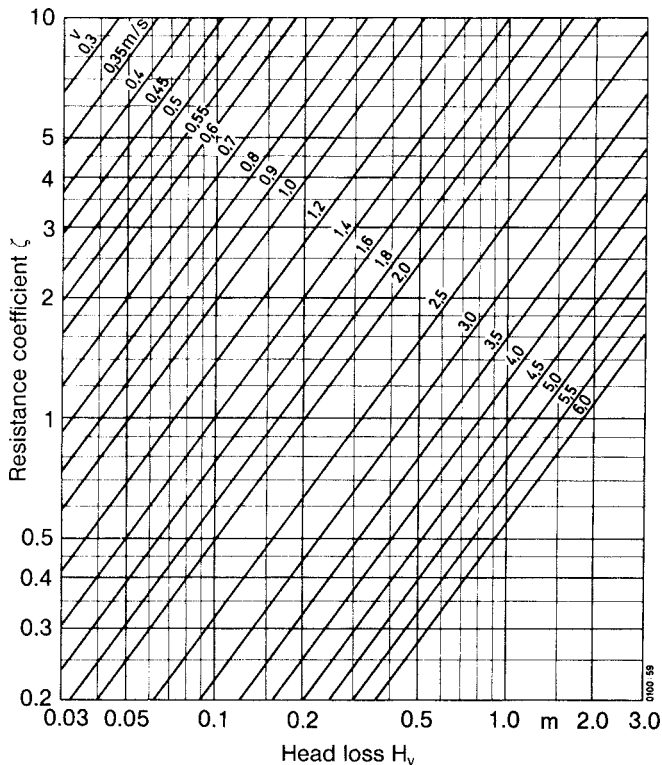


Fig. 18 : Determination of head losses H_V in valves and fittings; flow velocity v relating to the actual cross-sectional area through which the fluid flows

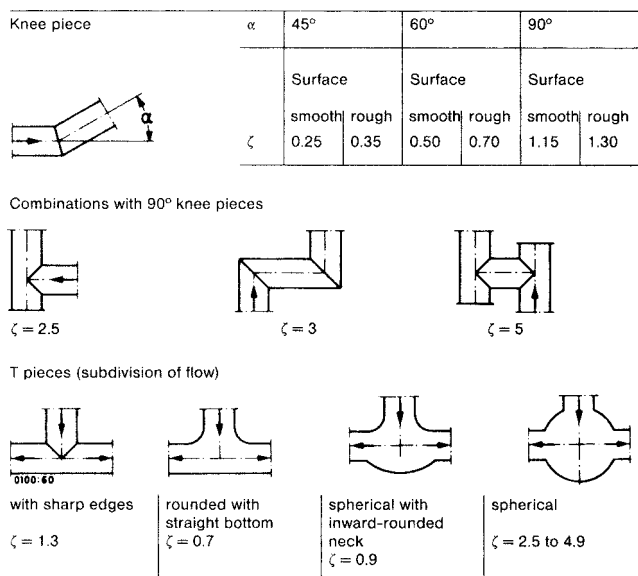


Fig. 19 : Illustration of fittings with related loss coefficient ζ

For pressure losses in valves and fittings the following equation applies :

$$p_v = \zeta \cdot \frac{\rho \cdot v^2}{2}$$

Where,

ζ loss coefficient

ρ density of pumped medium

v flow velocity across a section A which is characteristics of the head loss.

Tables 2 to 4 and figs. 18 to 24 give details of the individual loss coefficients ζ and head losses H_V in valves and fittings for operation with water

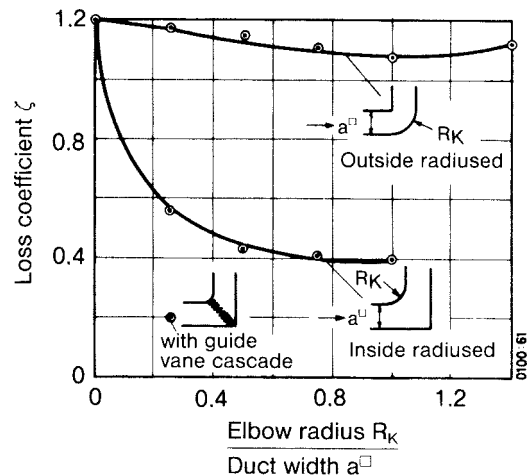


Fig. 20 : Influence of rounding off of concave and convex side on the loss coefficient of elbow with quadratic cross section

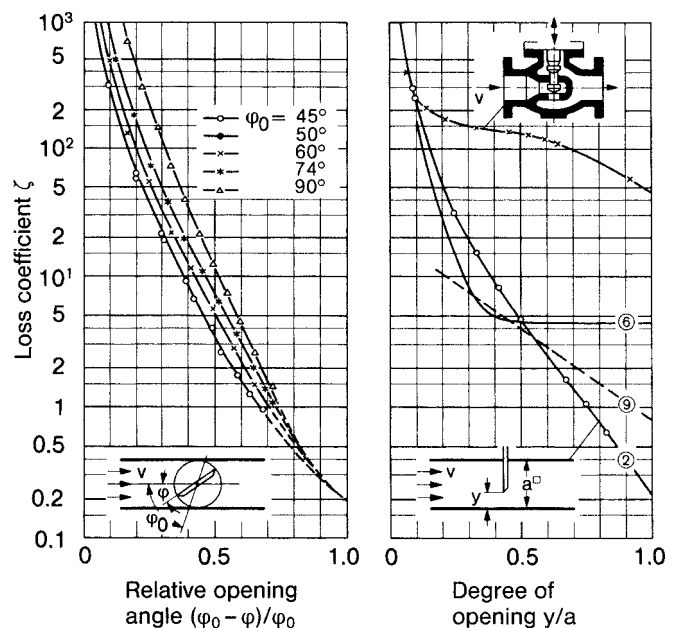
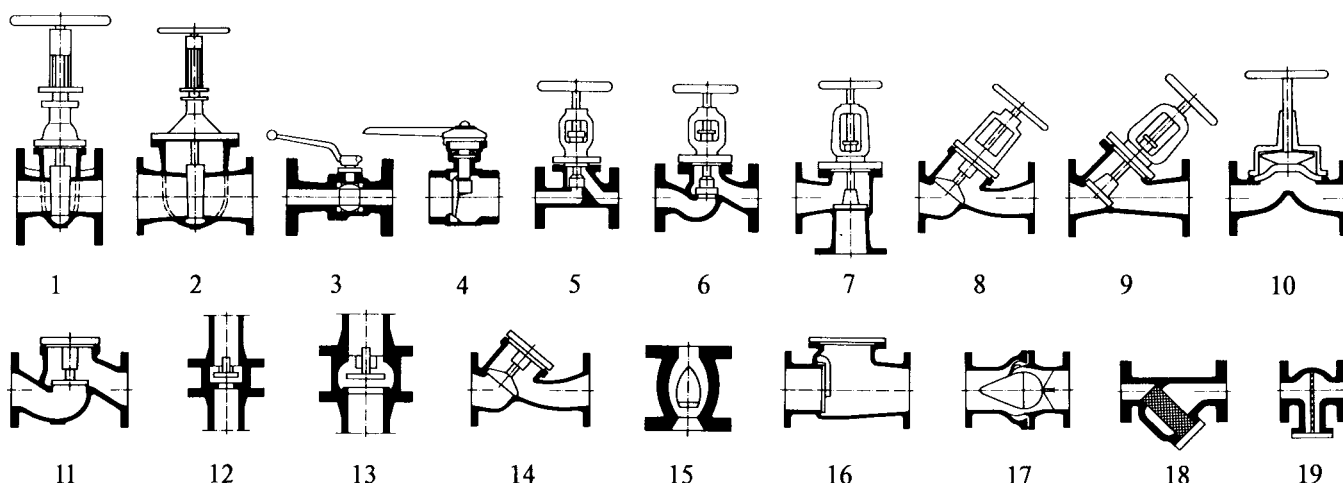


Fig. 21 : Loss coefficient of butterfly valves, globe and gate valves in function of opening angle or degree of opening (position numbers according to Table 2, design)

Table 2 : Loss coefficients ζ of valves and fittings (referred to the velocity of flow in the adjoining cross-section DN)

Type of valve/fitting		Design ³ Loss coefficient ζ for DN =																Remarks				
		15	20	25	32	40	50	65	80	100	125	150	200	250	300	400	500	600	800	1000		
Shut-off valves	Flat gate valves (dE = DN) min. max.	0.1 0.65	← 0.6		0.55 0.5	0.5	0.45 0.4	0.4 0.35	0.35 0.3	← 0.3									→ →	0.1 0.3	For dE<DN see footnote 1)	
	Round body gate valves (dE = DN) min max						0.25 0.32	0.24 0.31	0.23 0.30	0.22 0.28	0.21 0.26	0.19 0.25	0.18 0.23	0.17 0.22	0.16 0.20	0.15 0.19	0.13 0.18	0.12 0.16	0.11 0.15	0.11 0.14		
	Cocks (dE = DN) min. max.	0.10 0.15	0.10 ←	0.09 ←	0.09 ←	0.08	0.08 0.07	0.07 0.07	0.06 0.06	0.06 0.06	0.05 0.05	0.05 0.04	0.04 0.03	0.03 0.03	0.03 →	0.02 0.15					For dE<DN ζ = 0.4 to 1.1	
	Swing type valvePN≥2.5 PN≤ 40					0.90	0.76 0.60	0.60 0.50	0.42 0.42	0.36 0.36	0.30 1.50	0.25 1.20	0.20 1.00	0.16 0.92	0.15 0.83	0.13 0.76	0.12 0.71	0.11 0.67	0.11 0.63			
	Valves, Forged min. max.			6.0 6.8	← ←	6.8 6.8	→ →															
	Valves, cast min. max.	3.0 3.0	← ←												→ →	3.0 6.0						ζ = 2 to 3 possible for optimised valve
	Anglee valves min. max.	2.0 3.1	← ←	→ →	3.1 3.1	3.4 3.4	3.8 3.8	4.1 4.1	4.4 4.4	4.7 4.7	5.0 5.0	5.3 5.7	6.0 6.0	6.3 6.3	2.0 6.6							
	Slanted seat valvesmin. max.	1.5 2.6	← ←												→ →	1.5 2.6						
	Full-bore valves min. max.	0.6 1.6	← ←														→ →	0.6 1.6				
	Diaphragm valve min. max.	0.8 2.2	← ←										→ →	0.8 2.2								
	Non-return valve straight-seat min. max.	3.0 6.0	← ←										→ →	3.0 6.0								
	Backflow preventers	Non-return valve axial min. max.	3.2 3.4	← ←			→ →	3.2 4.2	3.7 5.0	5.0 6.4	7.3 8.2											
Non-return valve axially expanded min. max											4.3 4.6	← ←	→ →	4.3 4.6								
Non-return valve slanted seat min. max.		2.5 3.0	2.4 ←			2.0 2.0	1.9 1.9	1.7 1.7	1.6 1.6	1.5 1.5	← ←	→ →	→ →	1.5 3.0								
Foot valve min. max.							1.0 3.0	0.9 ←	0.8 ←	0.7 ←	0.6 ←	0.5 →	0.4 →	0.4 3.0	0.4 (7.0)	0.4 (6.1)	0.4 (5.5)	(4.5) (4.0)				() in groups
Swing type check valve min. max.		0.5 2.4	2.3 2.3	2.3 2.2	2.2 2.1	0.5 2.1	0.4 2.0	← 2.0	1.8 1.9	1.8 1.8	1.7 1.7	1.6 1.5	1.5 1.5	1.4 1.4	1.3 1.3	1.2 1.2	1.2 1.2	1.1 1.1	1.0 1.0			
Hydrostops v = 4m/s v = 3 m/s v = 2 m/s							0.9 1.8 5.0				3.0 4.0 6.0	2.5 4.0 8.0	2.5 4.0 7.5	1.2 1.8 6.5	1.2 1.8 6.0	2.2 3.4 7.0						
Filters						2.8	←							→	2.8						in clean conditions	
Screens						1.0	←							→	1.0							



The minimum and maximum values listed in Table 2 includes figures taken from the most pertinent trade literature and apply to fully open valves and fittings under uniform conditions of flow. The losses attributable to flow disturbances in a length of pipe equalling ca 12 x DN downstream of the valve or fitting are also included in those values (cf VDI/VDE guideline 2173). Nonetheless, the actual values are subject to wide variance, depending on the conditions of inflow and outflow, the model in question, and the design objectives.

Table 3 : Loss coefficients for fittings

Elbows :

Cast elbows 90°, R = D + 100 mm,
all nominal size $\zeta \approx 0.5$

Pipe bends 90°, R = 2 to 4 x D

Nominal size DN	50	100	200	300	500
ζ	0.26	0.23	0.21	0.19	0.18

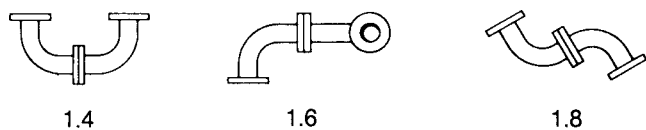
If the deflection angle only amounts to the above ζ values should be multiplied by	60°	45°	30°	15°
	0.85	0.7	0.45	0.3

Knee pieces :

Deflection angle	90°	60°	45°	30°	15°
ζ	1.3	0.7	0.35	0.2	0.1

Combinations of elbows and pipe bends :

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



Expansion joints :

Bellows expansion joint with / without

Guide pipe	$\zeta \approx 0.3/0.2$
Smooth bore pipe harp bend	$\zeta \approx 0.6$ to 0.8
Creased pipe harp bend	$\zeta \approx 1.3$ to 1.6
Corrugated pipe harp bend	$\zeta \approx 3.2$ to 4

Inlet pipe fittings

Inlet edge sharp	$\zeta \approx 0.5$	3			for $\delta = 75^\circ$ 60° 45°
chamfered	$\zeta \approx 0.25$	0.55	0.20	0.05	$\zeta \approx 0.6$ 0.7 0.8

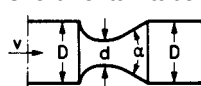
Discharge pieces

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

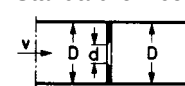
$\zeta = 2$ in the case of very unequal velocity distribution, e.g. immediately downstream of an elbow, a valve etc.

Loss coefficients of flow meters :

short venturi tube $\alpha = 30^\circ$



Standard orifice plate



ζ is related to the velocity v at diameter D.

Diameter ratio d/D	=	0.30	0.40	0.50	0.60	0.70	0.80
--------------------	---	------	------	------	------	------	------

Aperture ratio $m = (d/D^2)$	=	0.09	0.16	0.25	0.36	0.49	0.64
------------------------------	---	------	------	------	------	------	------

Short venturi tube $\zeta =$	21	6	2	0.7	0.3	0.2
Standard orifice plate $\zeta =$	300	85	30	12	4.5	2

Water meters (volumetric meters) $\zeta = 10$

In the case of domestic water meters, a max. pressure drop of 1 bar is prescribed for the rated load, and in practice the actual pressure loss is seldom below this figure.

Branch pieces : (Branch of equal bore)

The resistance coefficients ζ_a for the diverted flow Q_a or ζ_d respectively for the main flow $Q_d = Q - Q_a$ related to the velocity of the total flow Q in the nozzle.

On the basis of that definition, ζ_a and/or ζ_d may taken on negative values, in which case they are indicative of pressure loss. Not to be confused with reversible pressure changes according to BERNOULLI's equation (cf. annotation to Table 4).

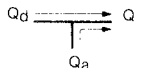
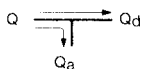
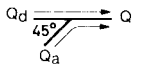
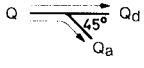
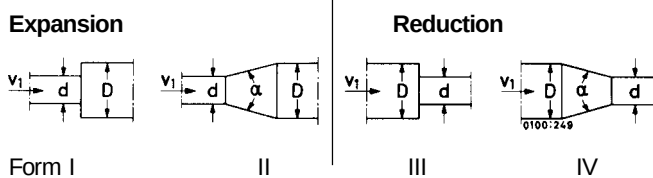
$Q_a/Q =$	0.2	0.4	0.6	0.8	1
	$\zeta_a \approx -0.4$ $\zeta_d \approx 0.17$	0.08 0.30	0.47 0.41	0.72 0.51	0.91 -
	$\zeta_a \approx 0.88$ $\zeta_d \approx -0.08$	0.89 -0.05	0.95 0.07	1.10 0.21	1.28 -
	$\zeta_a \approx -0.38$ $\zeta_d \approx 0.17$	0 0.19	0.22 0.09	0.37 -0.17	0.37 -
	$\zeta_a \approx 0.68$ $\zeta_d \approx -0.06$	0.50 -0.04	0.38 0.07	0.35 0.20	0.48 -

Table 4 : Pressure change coefficients in transition piece for arrangements illustrated in Fig. 14.

A coefficient in accordance with the values in the table below applies to each of the illustrated shapes of transition pieces/reducers. If the pressure rises across the transition pieces in the direction of flow (divergent section), is positive, and if the pressure drops (reducer), is negative.

Coefficients :



Form	d/D	=	0.5	0.6	0.7	0.8	0.9
I		$\zeta \approx$	0.56	0.41	0.26	0.13	0.04
II for	$\alpha = 8^\circ$	$\zeta \approx$	0.07	0.05	0.03	0.02	0.01
	$\alpha = 15^\circ$	$\zeta \approx$	0.15	0.11	0.07	0.03	0.01
	$\alpha = 20^\circ$	$\zeta \approx$	0.23	0.17	0.11	0.05	0.02
III		$\zeta \approx$	4.80	2.01	0.88	0.34	0.11
IV for $20^\circ < \alpha < 40^\circ$		$\zeta \approx$	0.21	0.10	0.05	0.02	0.01

Note : In the case of branch pieces as per Table 3 and transition pieces as per Table 4, differentiation is made between irreversible pressure loss (=pressure reduction)

$$p_v = \zeta \cdot \frac{\rho \cdot v_1^2}{2}$$

on the one hand and **reversible pressure changes** involving frictionless flow as per BERNOULLI's equation (fluid dynamics).

$$p_2 - p_1 = \frac{\rho}{2} (v_1^2 - v_2^2)$$

on the other. In the case of accelerated flow, e.g. through a pipe constriction $p_2 - p_1$ negative. Conversely, it is positive in pipe expansions. By contrast, the pressure losses ascertained by way of the loss coefficients ζ are **always negative**, if the overall pressure change is calculated as the arithmetic sum of p_v and $p_2 - p_1$.

In the case of water transport through valves and fittings, the loss coefficients ζ is occasionally neglected in favour of the so-called k_v -value :

$$p_v = \left(\frac{Q}{k_v} \right)^2 \times \frac{\rho}{1000}$$

where,

Q volume flow in m^3/hr .

ρ density of water in kg/m^3 (effective temperature vapour pressure, Table 1).

p_v pressure loss in bar

The k_v -value (m^3/hr .) represents the volume flow of cold water ($\rho = 1000 \text{ kg/m}^3$) at $p_v = 1$ bar through a valve or fitting; it therefore gives the relationship between the pressure loss p_v in bar and the volume flow Q in m^3/hr .

Conversion :

$$\zeta \approx 16 \cdot \frac{d^4}{k_v^2}$$

where

d reference diameter (nominal diameter) of the valve or fitting in cm.

5 Changing the pump performance

5.1 Changing the speed

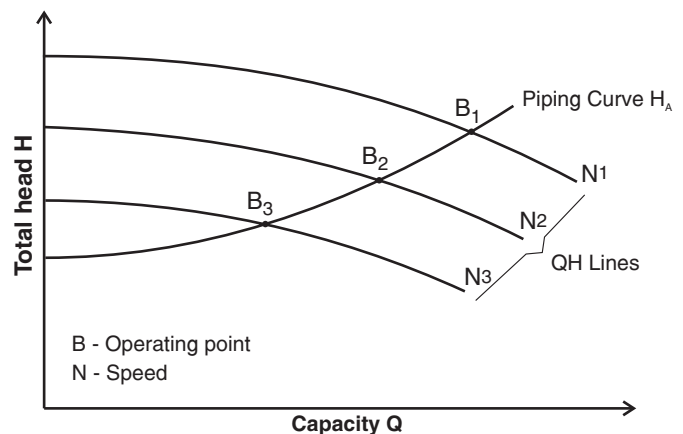
The same centrifugal pump has different characteristics curves for different speeds; these curves are interconnected by the similarity law. If the values for Q_1 , H_1 and P_1 are known at speed n_1 , then the new values for n_2 will be as follows :

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

$$H_2 = \left(\frac{n_2}{n_1} \right)^2 \cdot H_1$$

$$P_2 = \left(\frac{n_2}{n_1} \right)^3 \cdot P_1$$

A change in the speed also causes the operating point to shift (see 2.9) Fig. 22 plots three QH curves for the speeds n_1 , n_2 and n_3 , each curve is intersected by the system curve H_A at points B_1 , B_2 and B_3 respectively. The operating point will move along the system characteristics H_A from B_1 to B_3 when the speed is changed as indicated.



5.2 Trimming the impeller

Permanently reducing the output of a centrifugal pump operating at constant speed (see Fig. 23) entails reducing the impeller diameter D . The characteristic curve booklets contain the pump curves of selected impeller diameters in mm.

When trimming radial flow impellers (see 8.4) (trimming is not a geometrically similar reduction of an impeller since the outlet width normally remains constant), the relationship between Q , H and impeller diameter D is :

$$\left(\frac{D_1}{D_2} \right)^2 \approx \frac{Q_1}{Q_2} \approx \frac{H_1}{H_2} \quad D_2 \approx D_1 \cdot \sqrt{\frac{Q_2}{Q_1}} \approx D_1 \cdot \sqrt{\frac{H_2}{H_1}}$$

The actual diameter can be determined as follows (see Fig. 23) :
Run a line in the QH graph (linear graduation) passing from the point of origin (take into consideration with curves with a suppressed point of origin) through the new operating point B₂ and intersection at B₁ the full diameter curve D₁. The Q and H values 1 and 2 can then be plotted and used in the equation to be obtain the approximate diameter D₂.

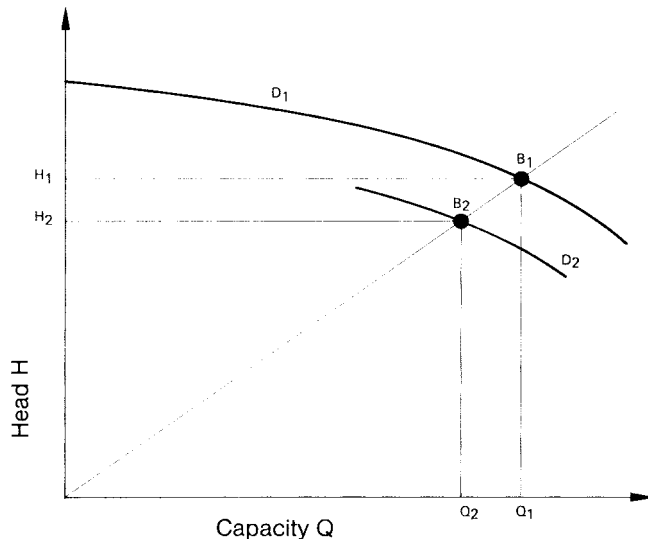


Fig. 23 : Influence of impeller diameter

6 Handling viscous liquids

As the viscosity ν of the medium handled increases (at constant speed) the capacity Q, head H and efficiency η fall; at the same time the pump power input P rises. The best efficiency point shifts to smaller flow rates. The operating point B_w drops to B_z (see fig. 24).

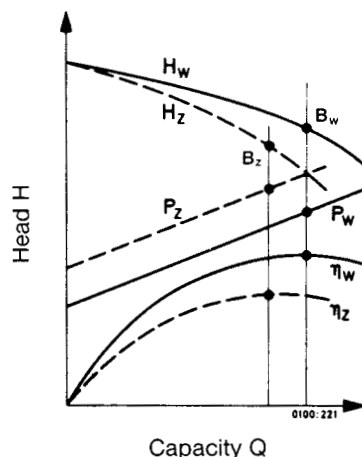


Fig. 24 : Change in operating point when handling viscous liquids (Z) and water (W)

The standard operating point for water B_w with Q_w, H_w and η_w (w = water) is converted to the viscous liquid operating point B_z with Q_z, H_z and η_z (Z = viscous liquid) using the conversion factors for viscous liquids f_Q, f_H and f_η (see Figs. 25a and 25b).

This conversion process can be used

- to convert from B_w to operating B_z using Fig. 25a (see 7.6.1)
- and to select the appropriate pump size from the given operating point B_z via the operating point B_w using Fig. 25b (see 7.6.2).

The conversion is valid for

- single-stage volute casing pumps with radial flow impellers (see 8.4)
- specific speeds n_q of 6 to 45 1/min. (see 7.6.1 and 9.12),
- kinematic viscosities below $22 \times 10^{-6} \text{ m}^2/\text{s}$ are normally disregarded).

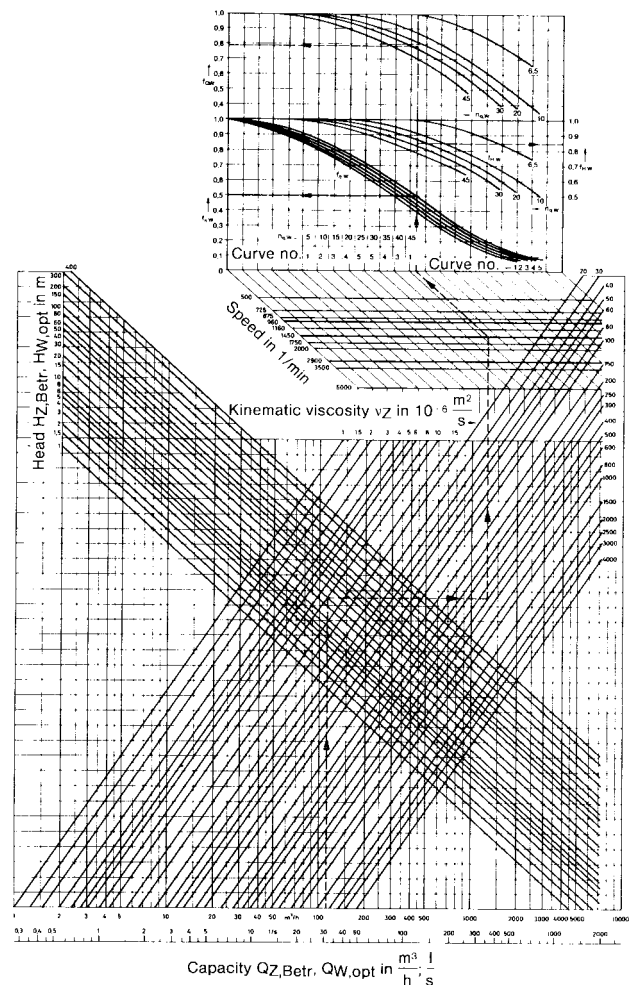


Fig. 25b : Determining the conversion factors $f_{Q,W}$, $f_{H,W}$ and $f_{\eta,W}$ for handling viscous liquids (enlarged version see 9.11), if the operating point for handling water liquid is given.

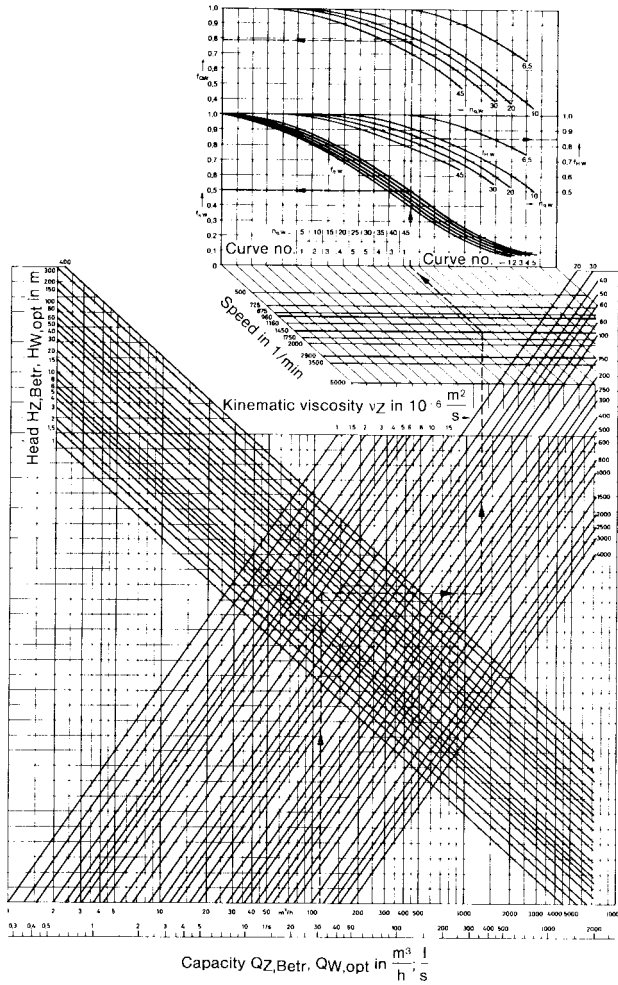
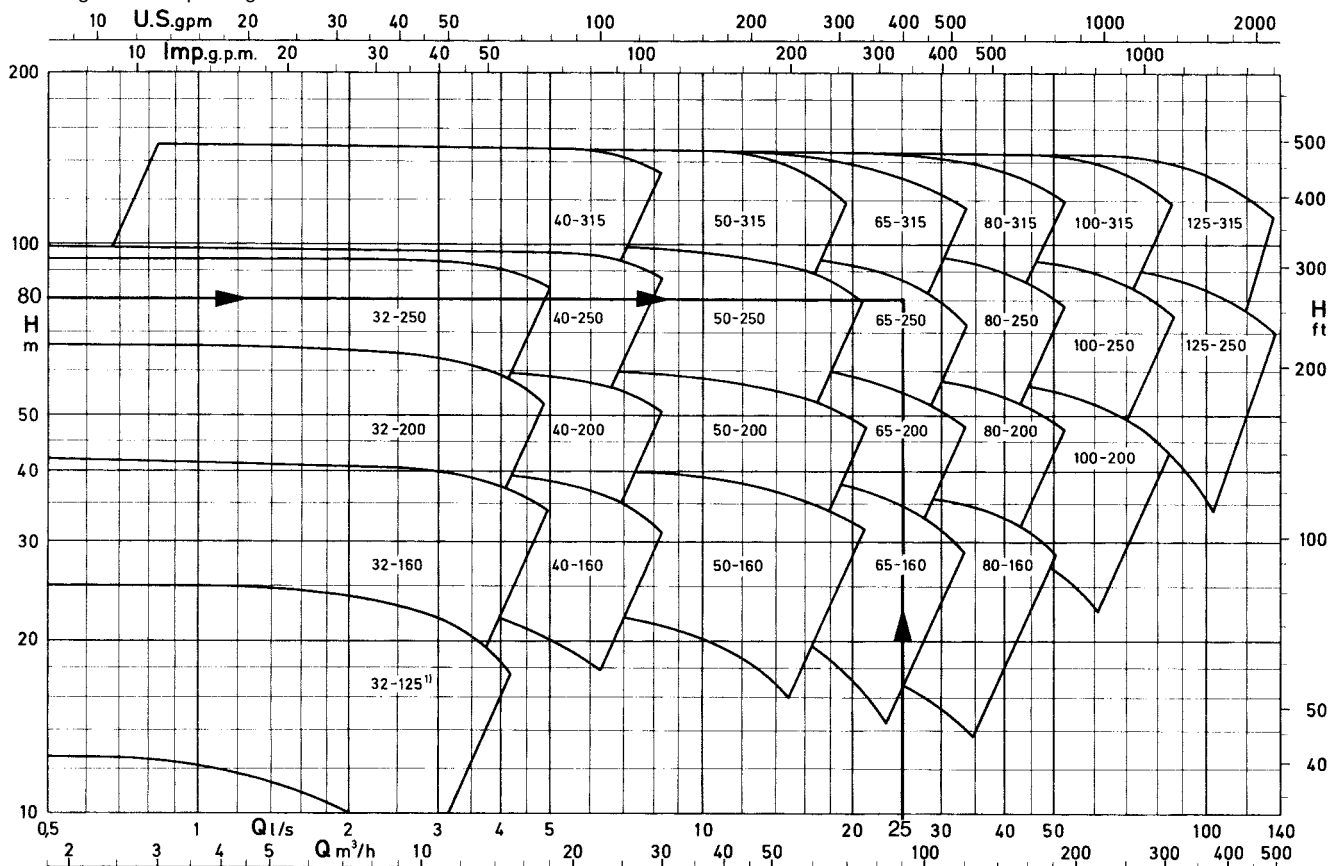


Fig. 25b : Determining the conversion factors $f_{Q,Z}$ and $f_{H,Z}$ for handling viscous liquids (enlarged version see 9.11), if the operating point for handling viscous liquid is given



7 Typical selection example

7.1 Selecting the pump size (see 2.5)

□ The following variables are known

$Q = 25 \text{ l/s} (= 90 \text{ m}^3/\text{h})$

$H = 80 \text{ m}$

Frequency 50 Hz

Medium 60% sulphuric acid (index s)

Density $\rho_s = 1.5 \text{ kg/dm}^3$

Temperature $t_s = 20^\circ\text{C}$

Kinematic viscosity $\nu_s = 3.8 \times 10^{-6} \text{ m}^2/\text{s}$ (can be disregarded, see 6)

(ρ_s and ν_s taken from standard reference tables)

The pump selected for this particular liquid is a CPK series standardized chemical pump.

Technical data and characteristic curves for the CPK are given in the characteristic curve booklet and selection booklet (Figs. 26 and 27 are extracts).

□ Selecting the size of the pump

Using the CPK/HPK characteristic curve booklet for 50Hz the selection charts give the following pump selections for the specified operating data :

CPK 65-250 at $n = 2900 \text{ 1/min.}$ and

CPK 150-250 at $n = 1450 \text{ 1/min.}$

The CPK 65-250 is selected for reasons of economy.

7.2 Calculating the power consumption

7.2.1 Pump input power

Using the known variables and pump selection from 7.1 the power input is calculated as follows

$$P = \frac{\rho_s \cdot g \cdot Q \cdot H}{1000 \cdot \eta} = \frac{1.5 \cdot 9.81 \cdot 25 \cdot 80}{1000 \cdot \eta} = 43.3 \text{ kW}$$

with

$\rho_s = \text{kg/dm}^3$

$g \text{ in m/s}^2$

$Q \text{ in l/s}$

$H \text{ in m}$

$P \text{ in kW}$

or alternative frequency used in practice;

$$P = \frac{\rho_s \cdot Q \cdot H}{367 \cdot \eta} = \frac{1.5 \cdot 90 \cdot 80}{367 \cdot 0.68^{1)}} = 43.5 \text{ kW}$$

with

$\rho_s = \text{kg/dm}^3$

$Q \text{ in m}^3/\text{h}$

$H \text{ in m}$

$P \text{ in kW}$

The pump power input P can also be established with sufficient accuracy from Fig. 27.

P is interpolated as = 29 kW for water, the value for sulphuric acid is :

$$P = 29 \cdot (\rho_s / \rho_{\text{water}}) = 29 \cdot (1.5/1) = 43.5 \text{ kW}$$

1) Efficiency from fig 27 incorporated

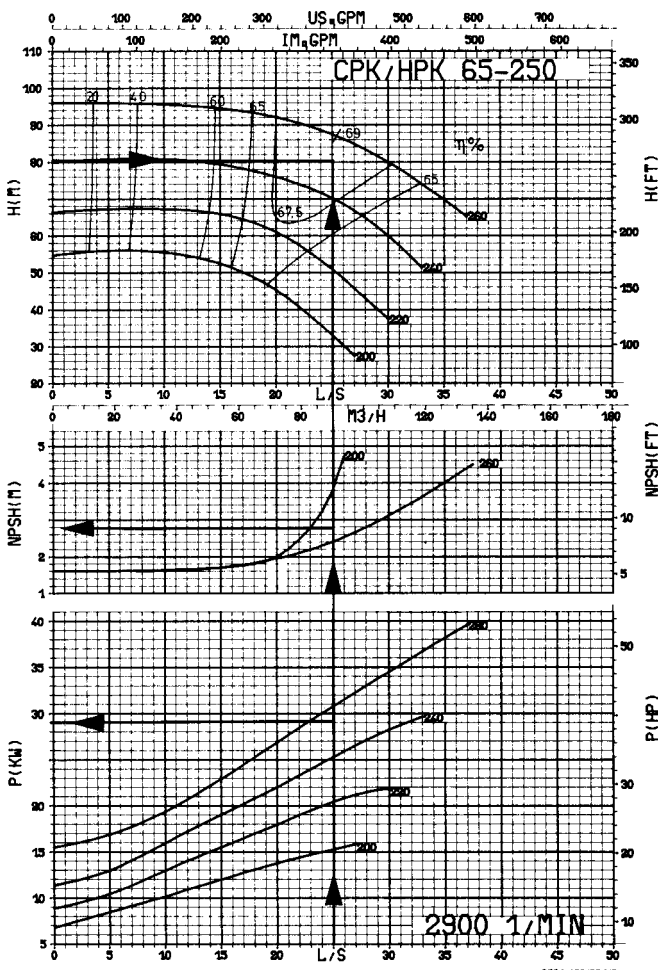


Fig. no. 27 : Characteristic curves CPK 65-250

7.2.2 Calculating the drive rating (see 2.6.2)

Taking the pump power input P (see 7.2.1)

□ a 10% safety margin is added to the 43.3 kW at the operating point

So the drive rating must be at least 47.6 kW

□ the selection is a standard 55 kW motor, 2 pole, IP 55/IP44 type B3

□ P/n value must be checked (see selection booklet, section Technical data).

If the operating point temporarily changes to higher flow rate, the motor rating must be increased accordingly, if necessary up to the maximum possible pump power consumption.

A recheck of the P/n value then becomes important as a criterion for the bearing bracket.

7.3 Calculating the NPSHav (see 3.2)

To achieve cavitation-free operation of the pump the limit of maximum possible suction lift $H_{s \text{ geo max.}}$ or the minimum required suction head $H_{Z \text{ geo min.}}$ must be adhered to.

7.3.1 Suction lift from open/closed tank

Here the pump is above the liquid level (see Fig. 10).

Selected pump is CPK 65-250, technical data see 7.1.

Calculation of $H_{s \text{ geo max}}$ is based on the following pump data

$\rho = 1500 \text{ kg/m}^3$

$p_b = 1 \text{ bar} = 1.10^5 \text{ N/m}^2$

$p_d = 0.0038 \text{ bar} = 0.0038 \cdot 10^5 \text{ N/m}^2$

(from reference table)

(60% Sulphuric acid at 20° C)

$H_{v,s} = 1.5 \text{ m}$ (estimated from fig. 13 for 10m suction pipe DN 100, including fittings and valves) can be disregarded because negligible

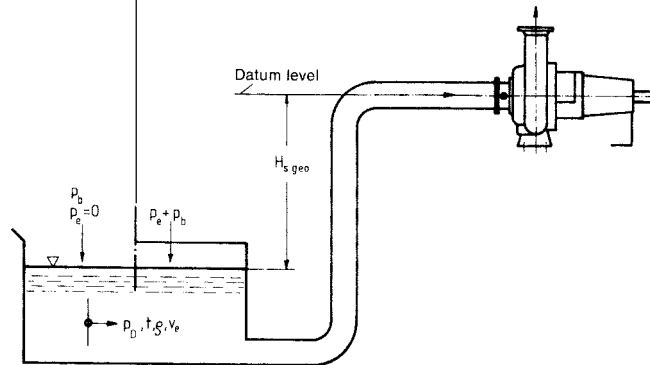
$NPSH_{\text{req.}} = 3.3 \text{ m}$ (interpolated from fig. 27 incl. 0.5m safety margin)

Open tank

Given : $p_e = 0$ bar

Closed tank

Given : $p_e + p_b = 1.5$ bar = $1.5 \cdot 10^5$ N/m²



$$H_{s \text{ geo, max}} = \frac{p_e + p_b - p_D}{\rho_s \cdot g} - H_v, s - \text{NPSHreq. (acc. to 3.2 with NPSHreq. = NPSHav)}$$

$$H_{s \text{ geo, max}} = \frac{0 + 1 \cdot 10^5 - 0.0038 \cdot 10^5}{1500 \cdot 9.81} - 1.5 - 3.3$$

$$= 6.77 - 1.5 - 3.3$$

$$= 1.97 \text{ m}$$

With $H_{s \text{ geo, max}} = 1.97$ m, $\text{NPSHav} = \text{NPSHreq.} = 3.3$ m; therefore $\text{NPSHav} > \text{NPSHreq.}$ requirement is satisfied.

$$H_{s \text{ geo, max}} = \frac{1.5 \cdot 10^5 - 0.0038 \cdot 10^5}{1500 \cdot 9.81} - 1.5 - 3.3$$

$$= 10.17 - 1.5 - 3.3$$

$$= 5.37 \text{ m}$$

With $H_{s \text{ geo, max}} = 5.37$ m, $\text{NPSHav} = \text{NPSHreq.} = 3.3$ m; therefore $\text{NPSHav} > \text{NPSHreq.}$ requirement is satisfied.

7.3.2 Positive suction operation from Open/Closed tank

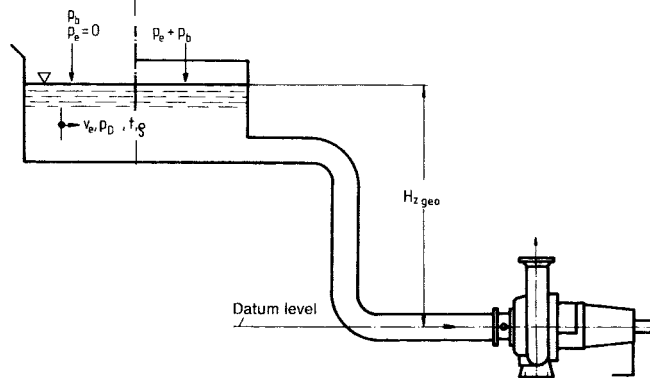
Here the pump is below the liquid level (see fig. 11). Selected pump is a CPK 65-250, technical data see 7.1 to 7.3.1.

Open tank

Given : $p_e = 0$ bar

Closed tank

Given : $p_e + p_b = 1.5$ bar = $1.5 \cdot 10^5$ N/m²



$$H_{z \text{ geo, min.}} = \text{NPSH}_{\text{req.}} + H_{v, s} - \frac{p_e + p_b - p_D}{\rho_s \cdot g}$$

$$H_{s \text{ geo, max}} = 3.3 + 1.5 - \frac{0 + 1 \cdot 10^5 - 0.0038 \cdot 10^5}{1500 \cdot 9.81}$$

$$= 1.5 + 3.3 - 6.77$$

$$= -1.97 \text{ m}$$

$$H_{s \text{ geo, max}} = 3.3 + 1.5 - \frac{1.5 \cdot 10^5 - 0.0038 \cdot 10^5}{1500 \cdot 9.81}$$

$$= 3.3 + 1.5 - 10.17$$

$$= -5.37 \text{ m}$$

Negative heads $-H_{z \text{ geo}}$ are suction lift heads $+H_{s \text{ geo}}$ of the same value. The **minus** sign in the result tells us that the centrifugal pump, with an open or closed tank, could draw roughly the absolute amounts as in example 7.3.1 where the requirement $\text{NPSHav} > \text{NPSHreq.}$ is just about satisfied. This requirement would be more than satisfied in example 7.3.2 with a positive static suction head (as shown in the diagram).

7.3.3 Positive suction operation from closed tank at vapour pressure

(Internal tank pressure = Vapour pressure of liquid,
i.e. $p_e + p_b = p_d$)

The pump is below the liquid level (see fig. 11).

The selected pump is a CPK 65-250, see 7.1 for technical data. See 7.3.1 for system and pump data required to calculate

$$H_{z \text{ geo min.}} = NPSH_{\text{req.}} + H_{v,s} - \frac{p_e + p_b - p_d}{\rho \cdot g}$$

$$= 3.3 + 1.5 - 0$$

$$= 4.8 \text{ m}$$

From 4.8 m upwards ($H_{z \text{ geo, min}}$) the condition $NPSH_{\text{av}} > NPSH_{\text{req.}}$ is fulfilled.

7.4 Changing the speed (see 5.1)

The CPK 65-250 selected in 7.1 but with the following performance data (present duty : index1, new duty index2)

$$Q_1 = 25 \text{ l/s } (=90 \text{ m}^3/\text{h})$$

$$H_1 = 70 \text{ m}$$

$$\text{at } n_1 = 2900 \text{ 1/min.}$$

and $D_1 = 240 \text{ mm}$ (impeller diameter)

is driven by a 55 kW three-phase motor with a nominal speed (n_2) of 2965 1/min. The higher speed shifts the operating point, without considering the system characteristic H_A , as follows to :

$$Q_2 = \frac{2965}{2900} \times 25 = 25.56 \text{ l/s } (= 92.02 \text{ m}^3/\text{hr.})$$

$$H_2 = \left(\frac{2965}{2900} \right)^2 \times 70 = 73.2 \text{ m}$$

If this increase is not acceptable, the original duty can be restored by e.g. reducing the impeller diameter (see 7.5)

7.5 Trimming of impeller

The unacceptably high pump output (see 7.4) caused by the higher motor speed is rectified as follows by trimming the impeller (present duty : index 1, new duty : index 2).

4 points on the new characteristic curve can be established using the calculation chart below

$n_{q,w}$ from graph in 9.12	27				1/min
$f_{q,w}$ from fig. 25a or	0.78				-
sect. 9.10,	0.83				-
page 41	0.49				-
$Q/Q_{\text{opt.}}$	0	0.8	1.0	1.2	-
Q_w from charact.	0	24.8	31	37.2	l/s
curve booklet for	25	21.6	20	18.2	m
4 points on	0	0.74	0.78	0.73	-
curve					
$Q_z = Q_w \cdot f_{q,w}$	0	19.3	24.2	29	l/s
$H_z =$	$=H_w$	$=H_w \cdot f_{H,w}$	$=H_w \cdot f_{H,w}$	$=H_w \cdot f_{H,w}$	
	25	18.5	16.6	15.1	m
$n_z = n_w \cdot f_{n,w}$	0	0.36	0.38	0.36	-
$P_z = \frac{\rho \cdot z \cdot g \cdot H_z \cdot Q_z}{n_z \cdot 1000}$		8.7	9.3	107	kW

Actual (now) :

$$Q_1 = 25.56 \text{ l/s}$$

$$H_1 = 73.2 \text{ m}$$

$$D_1 = 240 \text{ mm}$$

Desired

$$Q_2 = 25 \text{ l/s } (= 122.4 \text{ m}^3/\text{hr.})$$

$$H_2 = 70 \text{ m}$$

i.e.

$$D_2 = D_1 \times \sqrt{\frac{Q_2}{Q_1}} = 240 \times \sqrt{\frac{25}{25.56}} = 237 \text{ mm}$$

Turning the impeller down from 240 mm (D_1) to 237 mm (D_2) restores the original duty given in 7.4.

It is however, standard practice not to make such minor changes (less than 5mm) to the impeller diameter.

7.6 Handling viscous liquids (see 6)

7.6.1 Calculation of the operating point

The product is a mineral oil with a kinematic viscosity ν_z of $500 \times 10^{-6} \text{ m}^2/\text{s}$ and density $\rho_z = 0.897 \text{ kg}/\text{dm}^3$.

We know the characteristic curve and operating data of a pump handling water, where :

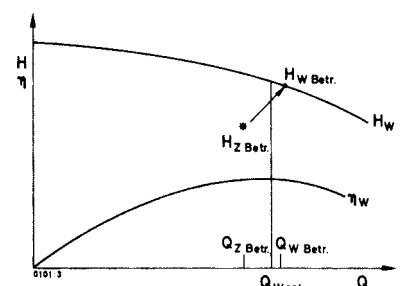
$$Q_w = 34 \text{ l/s}$$

$$H_w = 18 \text{ m}$$

$$n = 1450 \text{ 1/min.}$$

To obtain the new data for mineral oil, the pump data at the BEP must also be calculated and the following additional information must be known.

Capacity	Q_w	31 ¹⁾	l/s
Head	$H_{w, \text{opt}}$	20 ¹⁾	m
Efficiency	$\eta_{w \text{ opt}}$	0.78 ¹⁾	-
Speed	n	1450	1/min.
Kinematic viscosity	ν_z	500×10^{-6}	$\text{m}^2/\text{sec.}$
Density	ρ_z	0.897	kg/dm^3
Gravitational constant	g	9.81	$\text{m}^2/\text{sec.}$



Calculation in graphic form

These values mean 4 points on Q_H and Q_{η} lines plus 3 points on the Q_{Pz} line are established. Plotted over Q (see fig. 28)

7.6.2 Establishing the pump size

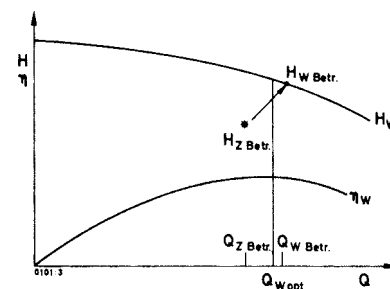
The product is mineral oil, we are looking for the size of the pump capable of meeting the following operating data:

Capacity	$Q_{z, \text{Betr}}$	31	l/s
Head	$H_{z, \text{Betr}}$	20	m
Kinematic viscosity	ν_z	$500 \cdot 10^{-6}$	m^2/sec
Density	ρ_z	0.897	kg/dm^3

Use the following calculation table to convert to operating data with water and thereby find the appropriate pump size.

n selected	1450	1/min.
$n_{q,w}^{3)}$ from graph in 9.12	27	1/min.
$f_{Q,z}$ from fig. 25b or	0.8	-
$f_{H,z}$ section 9.11, page 42	0.86	-
$Q_{w, \text{Betr}} = \frac{Q_{z, \text{Betr}}}{f_{Q,z}}$	38.8	l/s
$H_{w, \text{Betr}} = \frac{H_{z, \text{Betr}}}{f_{H,z}}$	23.3	m

³⁾ where $\left. \begin{array}{l} Q_{z, \text{Betr}} = Q_{\text{opt.}} \\ H_{z, \text{Betr}} = H_{\text{opt.}} \end{array} \right\} \text{ approx.}$



Calculation in graphic form

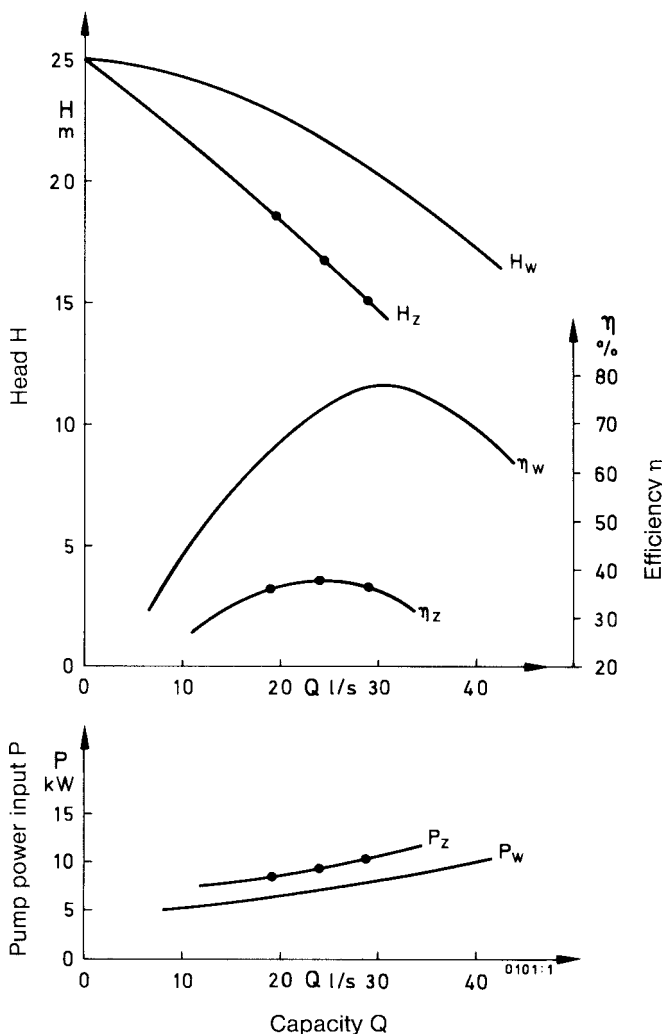


Fig. no. 28 : Characteristics curve for both water (W) and viscous liquids (Z) (see 7.6.1)

The definitive operating data when handling water are thus;

$$Q_{w, \text{Betr}} = Q_w = 38.8 \text{ l/s} (= 139.7 \text{ m}^3/\text{h})$$

$$H_{w, \text{Betr}} = H_w = 23.3 \text{ m}$$

Based on these data a suitable pump is selected from the sales documents selection chart. Using the curve thus established, follow section 7.6.1 to establish 4 points on the new characteristic curve.

These 4 points can be used to establish the curve to be expected for handling mineral oil, see Fig. 28.

8 General

8.1 National and International standards for Centrifugal pumps

A series of national standards have been introduced in Germany since the early sixties governing the manufacture, design, procurement and use of centrifugal pumps. These standards are drawn up by both operators and manufacturers and are now established in virtually all sectors of industry using and producing pumps (see Fig. 29, page 23).

This is particularly true of DIN 24 256 "End suction centrifugal pumps (PN 16) (chemical pumps)" which even in its first edition was virtually identical to the international standard ISO 2858 "End suction centrifugal pumps (rating 16 bar) - Designation, nominal duty point and dimensions".

These two standards occupy a central position because they form the basis for a range of standards already in existence and under preparation covering centrifugal pumps, accessories, guidelines and specifications.

Scope of application and Responsibilities		Dimensional Standards - Pumps				Accessories		Guidelines & specifications															
Federal Republic of Germany	VDMA Association of German Engineering Pump committee	VDMA 24253 Centrifugal pumps with armoured casing (armoured pumps); single-flow, single stage with axial inlet; duties, principal dimensions	DIN 24 251 Drainage pumps with heads up to 1000 m	DIN 24 252 Centrifugal pumps with wear plates PN 10; duties principal dimensions	DIN 24 254 Side channel pumps PN 40; designation, nominal duties principal dimensions	DIN 24 255 End suction centrifugal pumps PN 16 with bearing bracket, designation, nominal duties, principal dimensions	DIN 24 256 End suction centrifugal pumps PN 16 with bearing bracket, designation, nominal duties, principal dimensions	DIN 24 259 T.2 Machinery base-selection for centrifugal pumps to DIN 24 256, dimensions, classifications	DIN 24 299 T.1 Pump name plates; general specifications	DIN 24 960 Mechanical seals; shaft seal chamber, principal dimensions, designations and material codes	DIN 1944 Acceptance test for centrifugal pumps	DIN 24 250 Centrifugal pumps; nomenclature and numbers of components parts	DIN 24 260 Centrifugal pumps and centrifugal pump installations, terms, symbols, units	DIN 45 635 T.24 Noise measurements in machinery, airborne noise measurements enveloping surface method, liquid pumps.	DIN 24 293 Centrifugal pumps; technical documentation, terms, scope of supply, execution	DIN 24 295 Pump and pumpsets for liquids, safety requirements	VDMA 24 273 Pumps Instructions for procurement, Data sheet for materials and manufacturing acceptance tests.	VDMA 24 275 Connection dimensions for centrifugal pumps; admissible deviations and tolerances	VDMA 24 296 Centrifugal pumps, procurements, testing, supply and despatch specifications	VDMA 24 297 Centrifugal pumps technical requirements, specifications	DIN 24 420 Spares list		
	Europe*	CEN European Standards Coordinating Committee																					
International																							
Worldwide	ISO International organization for Standardization TC 115/ Pumps	ISO 2858 End suction centrifugal pumps (rating 16 bar) - Designation, nominal duty point and dimensions	ISO 3661 End suction centrifugal pumps Baseplates and installation dimensions	ISO 3069 End suction centrifugal pumps Dimensions of cavities for mechanical seals and for soft packing	ISO 2548 Centrifugal mixed flow and axial pumps code for acceptance tests - Class II (former C)	ISO 3555 Centrifugal mixed flow and axial pumps - Code for acceptance tests - Class I (former class B)	ISO 5198 Centrifugal mixed flow and axial pumps - Code for hydraulic performance tests - Precision class	ISO 5199 Technical specifications for centrifugal pumps class II															

*12 EC and EFTA member countries
Fig. 29 : Chart of German and International standards for centrifugal pumps, accessories, guidelines and specification (as of February 1990)

The high degree of similarity between DIN 24 256 and ISO 2858 means that a series of national standards and draft standards such as :

DIN 24 259 "Pump baseplate"
DIN 24 960 "Mechanical seals; shaft seal chamber, principal dimensions, designations and material codes"

VDMA 24 297 "Centrifugal pumps; technical requirements, specifications"

need minor or no changes in content even after the publication of the corresponding ISO standard.

8.2 Shaft deflection

Shaft deflection is principally caused by radial forces resulting from the hydraulic thrust in the impeller plane generated by the interaction between the impeller and pump casing (or diffuser). The magnitude and the direction of thrust changes with the rate of flow and affects the shaft and bearings.

The pump maker can favourably influence these hydraulic radial forces by selecting the right casing (see Fig.s 30 and 31).

This guarantees conformity with the specified maximum permissible shaft deflection (e.g. API 610 or ISO) and also means cost-effective sizing of shafts, especially seals and bearings.

The radial thrust F_R can be calculated with the help of the equation

$$F_R = K \cdot \rho \cdot g \cdot H \cdot D_2 \cdot b_2$$

Where,

F_R Radial thrust
 K Radial thrust coefficient according to fig. 31
 ρ Density of the pumped medium
 g Gravitational constant
 H Head
 D_2 Impeller outside diameter
 b_2 Impeller outlet width

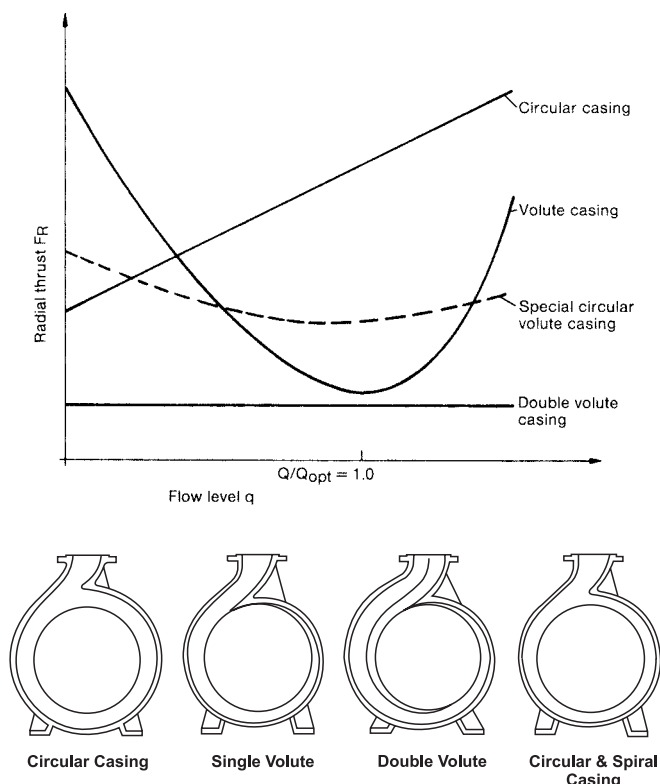


Fig. 30 : Radial thrust in centrifugal pumps with various casing types

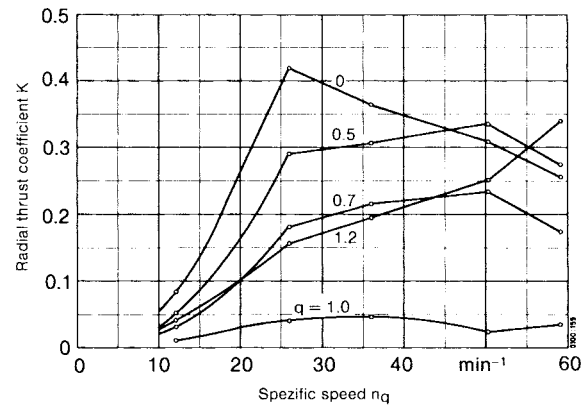


Fig. 31 : Magnitude of the radial thrust coefficient K for volute casing pumps as a function of the specific speed n_q and the pump flow level $q = Q/Q_{opt}$.

8.3 Improving the NPSH requirement.

It is possible in special cases to reduce the NPSH requirement of a pump to approx. 50-60% of the original level by fitting an inducer in front of the impeller, for example when a plant is extended and the available NPSH is inadequate or where economic factors prevent the available NPSH being increased (by rising the suction tank) or a lower speed larger-sized pump (with lower NPSH requirement) being fitted.

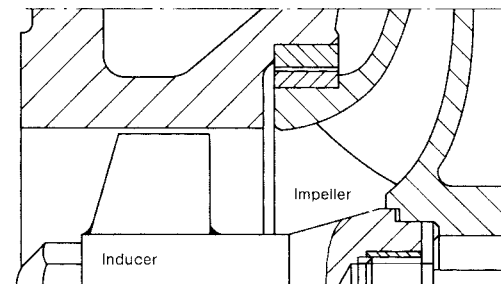
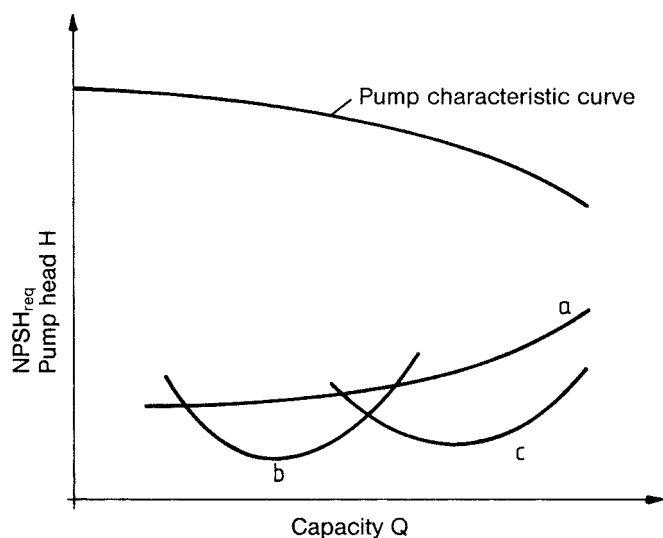


Fig. 32 : Centrifugal pump fitted with inducer

It must be noted that the reduction in the NPSH requirement applies only to a particular section of the flow range and not the complete range of the pump concerned.



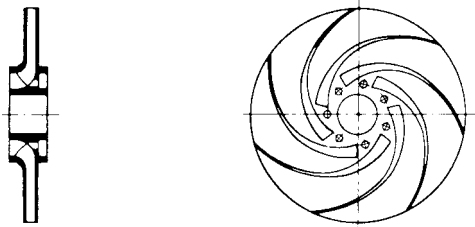
a = NPSHreq. - without inducer
b = NPSHreq. - with inducer A
c = NPSHreq. - with inducer B
A and B different types of inducer

Fig. 33 : NPSH requirement with and without inducer plotted against the capacity

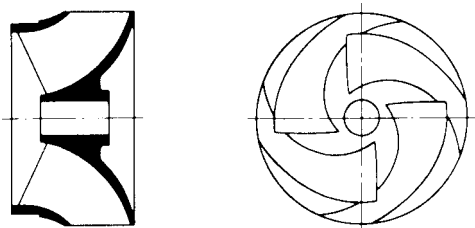
8.4 Impeller types

8.4.1 Vaned Impellers

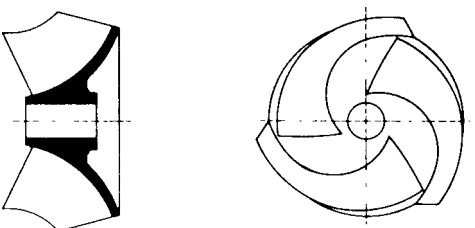
Centrifugal pumps handling clean products have standard impellers fitted with vanes. Such impellers go from the radial flow type through the mixed flow type for higher flow rates up to axial flow impeller for high flow rates and low heads.



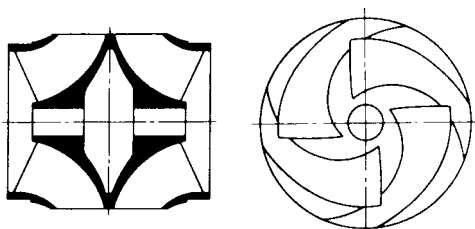
Radial flow impeller *)



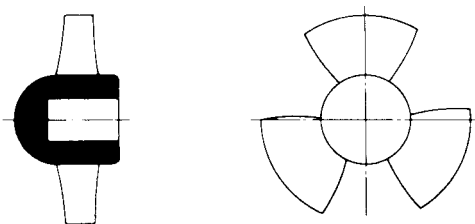
Mixed flow impeller *) closed



Mixed flow impeller open



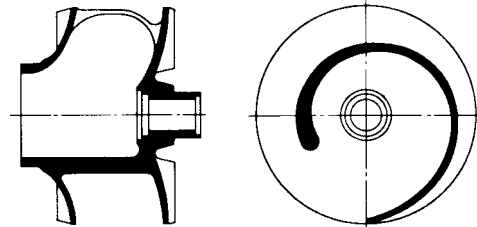
Mixed flow impeller *) closed, double entry



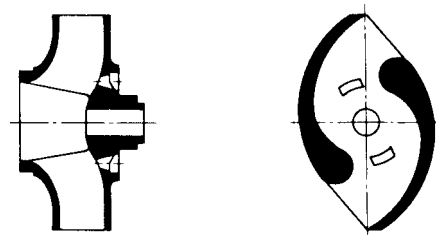
Axial flow impeller

8.4.2 Non-clogging impellers

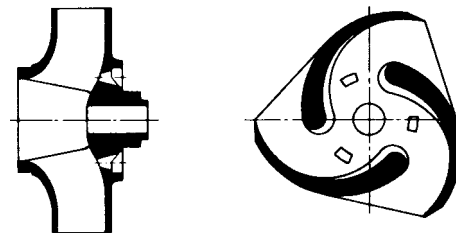
Large clearance impellers are used on pumps handling contaminated liquids containing solids, the single vane impeller has an unrestricted passageway from inlet to outlet (so-called free passage**).



Single vane impeller *) closed



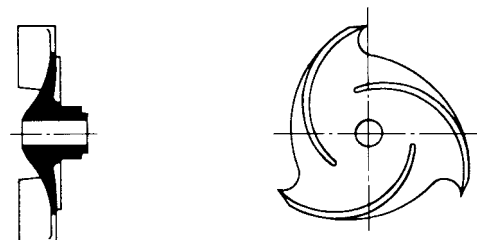
Two passage impeller *) closed



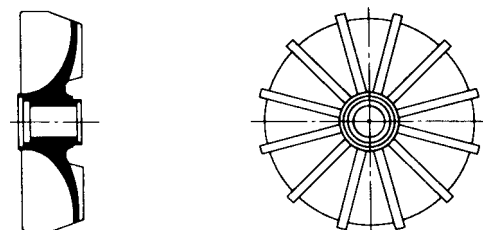
Three passage impeller *) closed

8.4.3 Special impellers

For contaminated and gaseous liquids.



Three vane impeller open



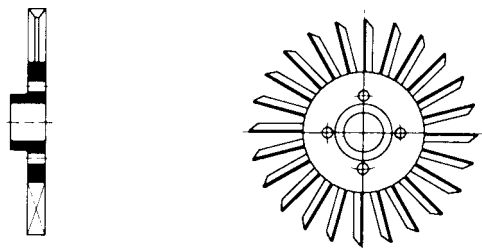
Free flow impeller

*) Front view with coverplate removed

**) Single vane impellers are also available with slightly reduced passage for greater efficiency

8.4.4 Star wheels

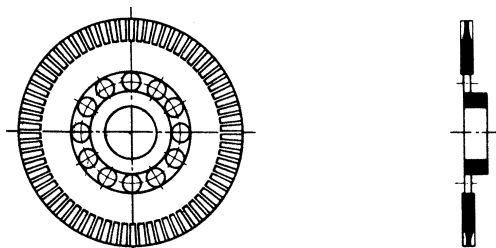
Mainly used in self-priming pumps handling clean media.



Star wheel for side channel pump

8.4.5 Peripheral impellers

Used for clean media, low flow rates and high heads



Peripheral impeller

8.5 Pump types (typical examples)

Figs. 34 to 39 show the various main design features

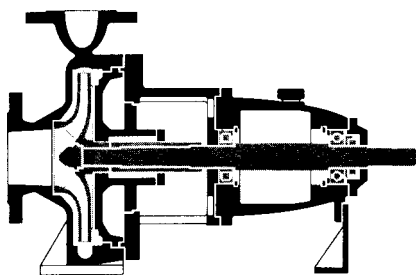


Fig. 34 : Single entry, single stage, overhung e.g. standard chemical pump

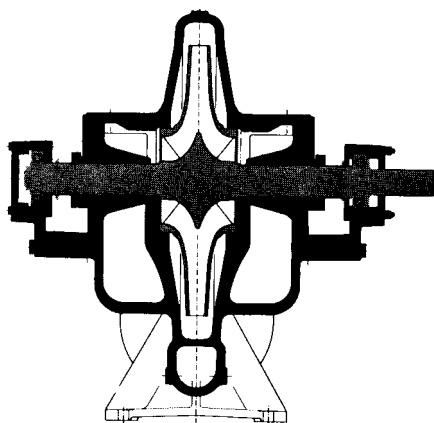


Fig. 35 : Double entry, suction and discharge side bearings, e.g. pipeline pump

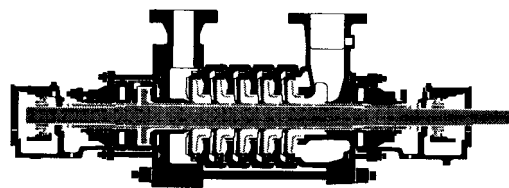


Fig. 36 : Multistage, suction and discharge side bearings, e.g. ring section high pressure centrifugal pump

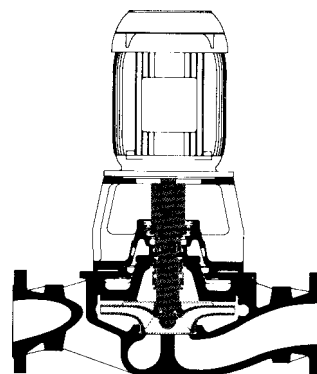


Fig. 37 : Close-coupled, e.g. in-line pump

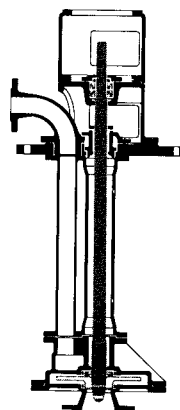


Fig. 38 : Vertical shaft-driven sump pump, e.g. submersible chemical pump

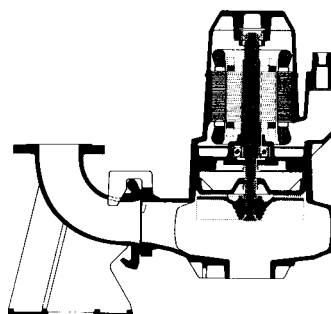


Fig. 39 : Submersible close-coupled pump, e.g. sewage pump

8.6 Pump installation arrangements

The factors which determine how a pump is installed are :

- the position of the shaft, i.e. horizontal or vertical
- the position of the feet, i.e. underneath or shaft centreline
- the arrangement on the drive,
- the weight distribution of the pump and drive (see figs. 40 & 41)

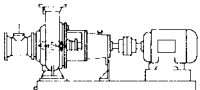
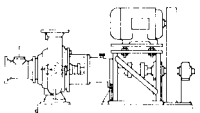
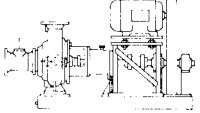
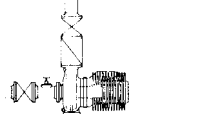
	Shaft	Feet	Drive	Remarks
	Horizontal	Underneath	Coaxial with coupling or gearbox	Common baseplate
	Horizontal	Centreline	Coaxial with coupling or gearbox	Common baseplate
	Horizontal	Underneath	With parallel axis above pump, belt drive	Compact, simple speed variation
	Horizontal	Underneath	with parallel axis above pump with belt drive and outboard bearing or jackshaft	Compact, simple speed variation
	Horizontal	Underneath	Close-coupled, forming a water tight unit with pump	fully submersible

Fig. 40 : Examples of horizontal installation

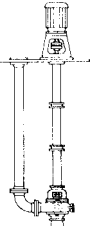
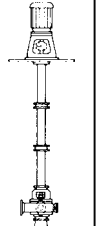
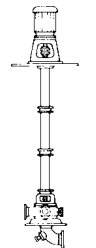
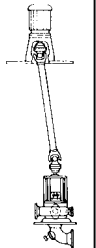
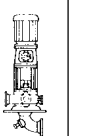
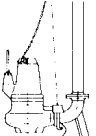
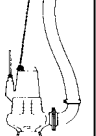
Alternative installation			Shaft	Feet	Drive	Remarks
a	b	c				
			Vertical	-	Above ground on drive stool	Wet installation a) surface level discharge pipe
			Vertical	Soleplate beneath discharge nozzle	a) above ground on drive stool b) above ground on drive stool through cadran shaft c) below surface on drive stool	Dry installation
			Vertical	a) automatic engagement with claw b) on support foot	Submersible close-coupled unit	Wet installation a) permanent b) portable

Fig. 41 : Examples of vertical mounting

8.7 Pump sump configuration

Pump sumps are designed to receive liquids and be intermittently drained. The sump size depends on the capacity Q and permissible start-up frequency Z of the pump set. i.e. the electric motor.

The start-up frequencies of any dry motors are as follows :

Motor rating	Start-up frequency
Up to 7.5 kW	max. 15/h
Up to 30 kW	max. 12/h
Above 30 kW	max. 10/h

Start-up frequency is calculated using :

$$Z = \frac{3600 \cdot Q_{zu} (Q_m - Q_{zu})}{V_N \cdot Q_m}$$

Where Z = no. of starts per hr.

Q_{zu} = inlet flow in l/s

$Q_m = (Q_e + Q_a)/2$

Q_e = Capacity at switch-on pressure in l/s

Q_a = Capacity at switch-off pressure in l/s

V_N = useful volume of pump sump including possible flowback volume in l

The maximum start-up frequency occurs when $Q_m = 2 \times Q_{zu}$ i.e. when the capacity Q_m is twice the incoming flow Q_{zu} . The max. start-up frequency is therefore :

$$Z_{max} = \frac{900 \cdot Q_m}{V_N}$$

With the dry liquids, solids must be prevented from being deposited and collecting in dead zones and on the floor. 45° walls, or better still 60° walls, help prevent this (see Fig. 42).

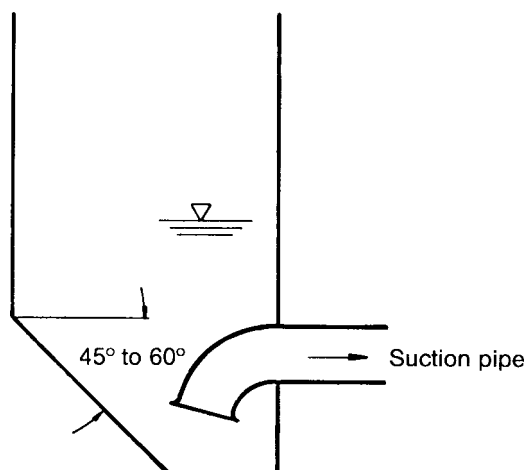


Fig. 42 : Inclined sump walls to prevent solids from being deposited and collecting

8.8 Suction pipe layout

The suction pipe should be as short as possible and run with a gentle slope up to the pump. The suction pipe and inlet pipe must be sufficiently wide apart to prevent air entrainment in the suction pipe. Furthermore the mouth of the inlet pipe must always lie below the liquid level (see Fig. 43.)

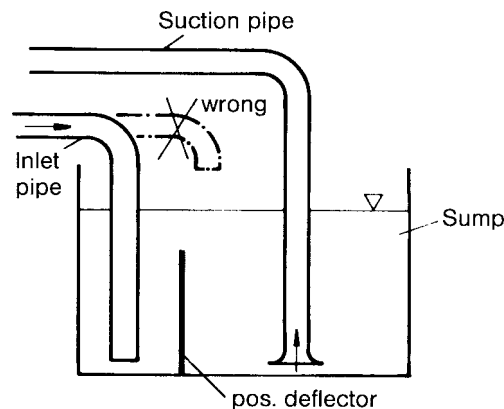


Fig. 43 : Piping arrangement to prevent air entrainment

The medium handled must cover the suction pipe inlet to a suitable depth, otherwise rotation of the liquid could cause air-entering vortices (hollow vortices) to form; starting with a funnel-shaped depression at the liquid surface, a tube shaped air cavity forms instantaneously, extending from the surface to the suction pipe.

By ensuring that the medium handled always has a suitable level (see Figs. 44 and 45) or by taking measures to prevent vortices (see Figs 46 to 48) this can be prevented, which is more important, the higher the flow rate is.

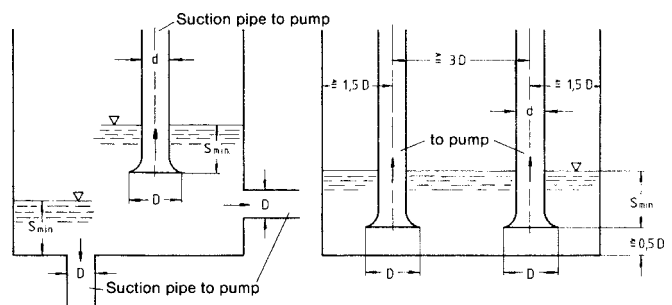


Fig. 44 : Arrangement of pipes in the suction tank (sump) to prevent vortices

The minimum liquid cover S_{min} in m must be the velocity heads plus a 0.1m safety margin for non-uniform velocity distribution. The maximum flow velocity v_s in the suction pipe or inlet pipe should not exceed 3 m/s; we recommend 1 to 2 m/s.

$$S_{min} = \frac{v_s^2}{2g} + 0.1$$

with v_s flow velocity in m/s

S_{min} minimum liquid cover in m

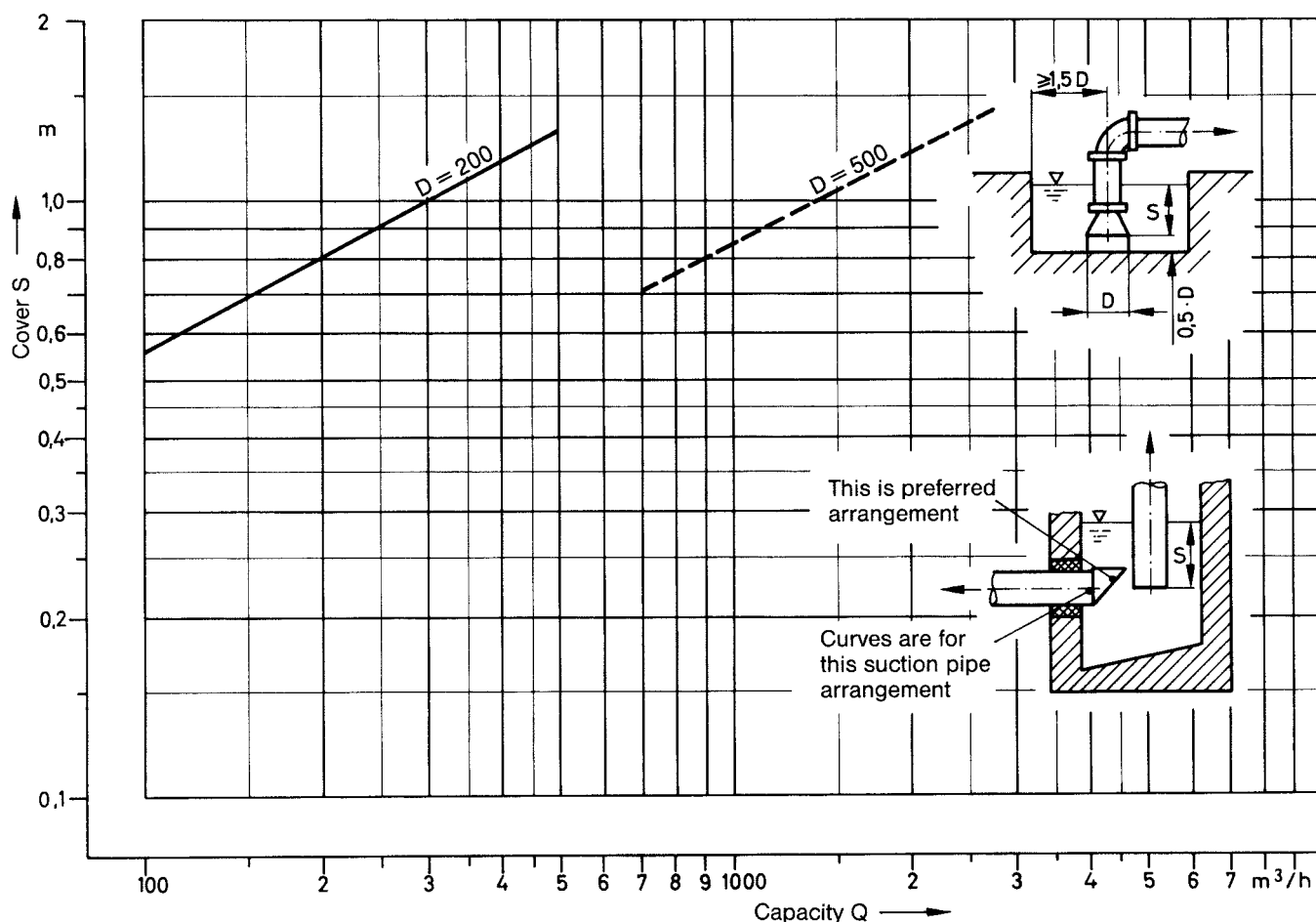


Fig. 45 : Liquid cover S as a function of the piping bore DN and capacity Q

Fig. 45 shows the interdependence between liquid cover S , piping bore DN and capacity Q . The values obtained give sufficient protection against vortices. The graph can be used for the suction pipe layout illustrated.

Figs. 46 and 47 show typical arrangements used to prevent air-entraining inlet vortices where the minimum liquid cover is either not available or cannot be ensured.

Fig. 48 shows a special arrangement which is frequently used - a round tank with a tangential inlet pipe which causes the contents to rotate.

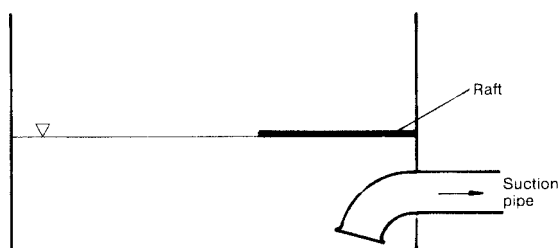


Fig. 46 Raft to prevent formation of vortices

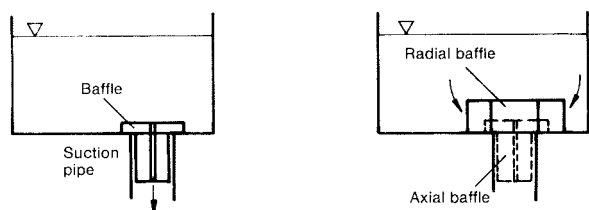


Fig. 47 : Use of swirl-preventing baffles

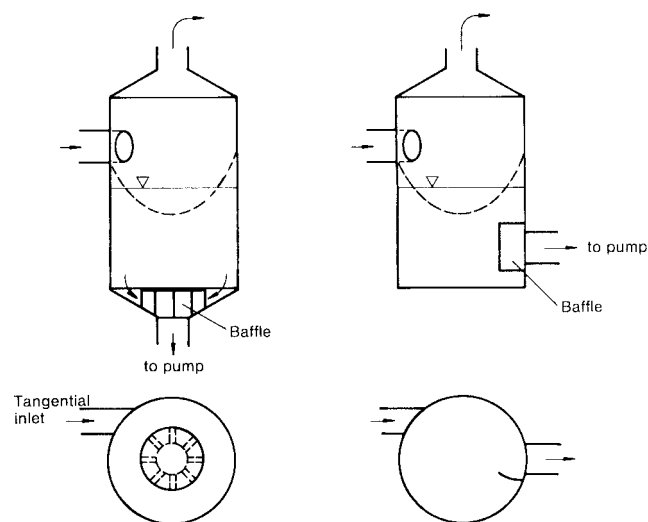


Fig. 48 : Use of baffles in the tank to ensure disturbance-free flow to pump.

8.9 Shaft couplings

Shaft couplings used with centrifugal pumps can be divided into rigid & flexible types. Rigid couplings are mainly used to connect shafts in perfect alignment. The smallest degree of misalignment will cause considerable stress on the coupling and on the shafts. The following types are used.

- Sleeve couplings
- Muff couplings,
- Serrated couplings,
- Split couplings (DIN 115),
- Face plate couplings (DIN 758, DIN 759),
- Flange couplings.

Flexible couplings to DIN 740 are elastic, slip-free connecting elements between drive and driven machine which accommodate axial, radial and angular misalignment (fig. 49) and damp shock loads. The flexibility is usually achieved by the deformation of damping and rubber-elastic spring elements whose life is governed to a large extent by the degree of misalignment.

Fig. no 50 shows the most common types of flexible couplings. Fig. no. 51 shows a spacer coupling between a pump and drive; its function is to permit removal of the pump rotating assembly without disturbing the casing or drive (back-pull out design).

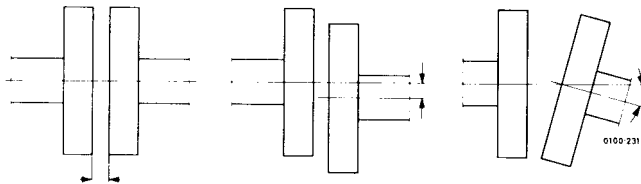


Fig. 49 : Misalignemnt

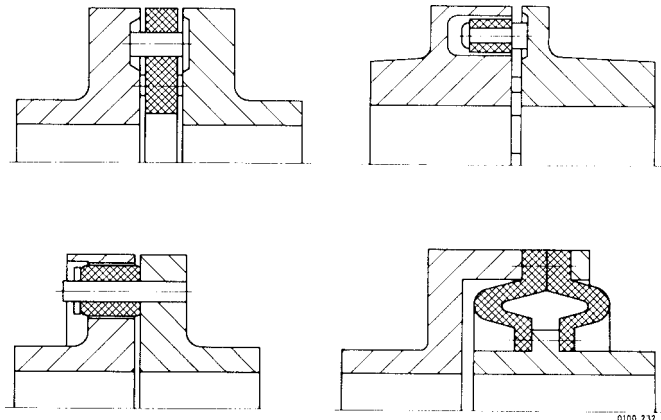


Fig. 50 : Typical coupling

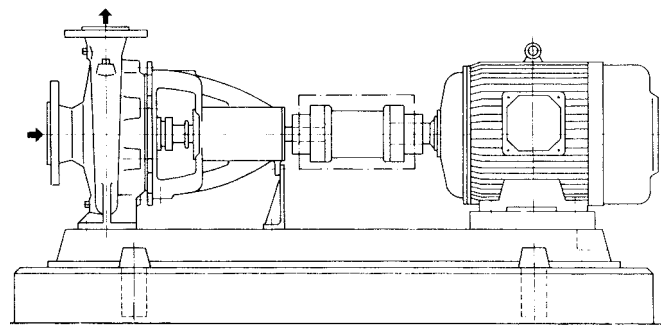


Fig. 51 : Pump with spacer coupling

9 Technical Data

9.1 Vapour pressure p_D and Density ρ of water

t °C	T K	p_D bar	ρ kg/dm ³	t °C	T K	p_D bar	ρ kg/dm ³	t °C	T K	p_D bar	ρ kg/dm ³
0	273.15	0.00611	0.9998	61	334.15	0.2086	0.9826	150	423.15	4.760	0.9168
1	274.15	0.00657	0.9999	62	335.15	0.2184	0.9821	155	428.15	5.433	0.9121
2	275.15	0.00706	0.9999	63	336.15	0.2286	0.9816	160	433.15	6.181	0.9073
3	276.15	0.00758	0.9999	64	337.15	0.2391	0.9811	165	438.15	7.008	0.9024
4	277.15	0.00813	1.0000	65	338.15	0.2501	0.9805	170	443.15	7.920	0.8973
5	278.15	0.00872	1.0000	66	339.15	0.2615	0.9799	175	448.15	8.924	0.8921
6	279.15	0.00935	1.0000	67	340.15	0.2733	0.9793	180	453.15	10.027	0.8869
7	280.15	0.01001	0.9999	68	341.15	0.2856	0.9788	185	458.15	11.233	0.8815
8	281.15	0.01072	0.9999	69	342.15	0.2984	0.9782	190	463.15	12.551	0.8760
9	282.15	0.01147	0.9998	70	343.15	0.3116	0.9777	195	468.15	13.987	0.8704
10	283.15	0.01227	0.9997	71	344.15	0.3253	0.9770	200	473.15	15.55	0.8647
11	284.15	0.01312	0.9997	72	345.15	0.3396	0.9765	205	478.15	17.243	0.8588
12	285.15	0.01401	0.9996	73	346.15	0.3543	0.9760	210	483.15	19.077	0.8528
13	286.15	0.01497	0.9994	74	347.15	0.3696	0.9753	215	488.15	21.060	0.8467
14	287.15	0.01597	0.9993	75	348.15	0.3855	0.9748	220	493.15	23.198	0.8403
15	288.15	0.01704	0.9992	76	349.15	0.4019	0.9741	225	498.15	25.501	0.8339
16	289.15	0.01817	0.9990	77	350.15	0.4189	0.9735	230	503.15	27.976	0.8273
17	290.15	0.01936	0.9988	78	351.15	0.4365	0.9729	235	508.15	30.632	0.8205
18	291.15	0.02062	0.9987	79	352.15	0.4547	0.9723	240	513.15	33.478	0.8136
19	292.15	0.02196	0.9985	80	353.15	0.4736	0.9716	245	518.15	36.523	0.8065
20	293.15	0.02337	0.9983	81	354.15	0.4931	0.9710	250	523.15	39.776	0.7992
21	294.15	0.02485	0.9981	82	355.15	0.5133	0.9704	255	528.15	43.246	0.7916
22	295.15	0.02642	0.9978	83	356.15	0.5342	0.9697	260	533.15	46.943	0.7839
23	296.15	0.02808	0.9976	84	357.15	0.5557	0.9691	265	538.15	50.877	0.7759
24	297.15	0.02982	0.9974	85	358.15	0.5780	0.9684	270	543.15	55.058	0.7678
25	298.15	0.03166	0.9971	86	359.15	0.6011	0.9678	275	548.15	59.496	0.7593
26	299.15	0.03360	0.9968	87	360.15	0.6249	0.9671	280	553.15	64.202	0.7505
27	300.15	0.03564	0.9966	88	361.15	0.6495	0.9665	285	558.15	69.186	0.7415
28	301.15	0.03738	0.9963	89	362.15	0.6749	0.9658	290	563.15	74.437	0.7323
29	302.15	0.04004	0.9960	90	363.15	0.7011	0.9652	300	573.15	85.927	0.7122
30	303.15	0.04241	0.9957	91	364.15	0.7281	0.9644	305	578.15	92.144	0.7017
31	304.15	0.04491	0.9954	92	365.15	0.7561	0.9638	310	583.15	98.700	0.6906
32	305.15	0.04753	0.9951	93	366.15	0.7849	0.9630	315	588.15	106.61	0.6791
33	306.15	0.05029	0.9947	94	367.15	0.8146	0.9624	320	593.15	112.89	0.6669
34	307.15	0.05318	0.9944	95	368.15	0.8453	0.9616	325	598.15	120.56	0.6541
35	308.15	0.05622	0.9940	96	369.15	0.8769	0.9610	330	603.15	128.63	0.6504
36	309.15	0.05940	0.9937	97	370.15	0.9094	0.9602	340	613.15	146.05	0.6102
37	310.15	0.06274	0.9933	98	371.15	0.9430	0.9596	350	623.15	165.35	0.5743
38	311.15	0.06624	0.9930	99	372.15	0.9776	0.9586	360	633.15	186.75	0.5275
39	312.15	0.06991	0.9927	100	373.15	1.0133	0.9581	370	643.15	210.54	0.4518
40	313.15	0.07375	0.9923	102	375.15	1.0878	0.9667	374.15	647.30	221.2	0.3154
41	314.15	0.07777	0.9919	104	377.15	1.1668	0.9552				
42	315.15	0.08198	0.9915	106	379.15	1.2504	0.9537				
43	316.15	0.08639	0.9911	108	381.15	1.3390	0.9522				
44	317.15	0.09100	0.9907	110	383.15	1.4327	0.9507				
45	318.15	0.09582	0.9902	112	385.15	1.5316	0.9491				
46	319.15	0.10086	0.9898	114	387.15	1.6362	0.9476				
47	320.15	0.10612	0.9894	116	389.15	1.7465	0.9460				
48	321.15	0.11162	0.9889	118	391.15	1.8628	0.9445				
49	322.15	0.11736	0.9884	120	393.15	1.9854	0.9429				
50	323.15	0.12335	0.9880	122	395.15	2.1145	0.9412				
51	324.15	0.12961	0.9876	124	397.15	2.2504	0.9396				
52	325.15	0.13613	0.9871	126	399.15	2.3933	0.9379				
53	326.15	0.14293	0.9866	128	401.15	2.5435	0.9362				
54	327.15	0.15002	0.9862	130	403.15	2.7013	0.9346				
55	328.15	0.15741	0.9857	132	405.15	2.8670	0.9328				
56	329.15	0.16511	0.9852	134	407.15	3.041	0.9311				
57	330.15	0.17313	0.9846	136	409.15	3.223	0.9294				
58	331.15	0.18147	0.9842	138	411.15	3.414	0.9276				
59	332.15	0.19016	0.9837	140	413.15	3.614	0.9258				
60	333.15	0.19920	0.9832	145	418.15	4.155	0.9214				

9.2 Vapour pressure po of various liquids

Temperature	Ethane C ₂ H ₆	Acetone (CH ₃) ₂ CO	Ammonia NH ₃	Ethyl Alcohol C ₂ H ₅ OH	n-Butane C ₄ H ₁₀	i-Butane C ₄ H ₁₀	Benzol C ₆ H ₆	Aniline C ₆ H ₅ NH ₂	Ether C ₂ H ₅ OC ₂ H ₅	Formic acid CH ₂ O ₂	Acetic acid C ₂ H ₄ O ₂	n-Propane C ₃ H ₈	Methyl Alcohol CH ₄ O	Sulphurous acid H ₂ SO ₃	Carbon disulphide CS ₂	Toluol C ₇ H ₈	Carbon tetrachloride CCl ₄
-50	223	5.517	0.00319	0.409		0.103			0.0127			0.707		0.1157			
-45	228	6.574		0.545								0.890		0.1598			
-40	233	7.776			0.718				0.0255			1.115		0.2157			
-35	238	9.129		0.932								1.379		0.2883			
-30	243	10.65	0.0149	1.195		0.294	0.483		0.050			0.167		0.3805	0.0335		
-25	248	12.34		1.516								2.017		0.4942			
-20	253	14.23	0.0293	1.902		0.469	0.748		0.0883			0.423		0.6355	0.0609		0.0129
-15	258	16.31		2.363								2.889		0.8071			0.0180
-10	263	18.59	0.0516	2.909		0.691	1.103		0.150			3.405		1.014	0.1047		0.0246
-5	268	21.10		3.549								4.015		1.2611			0.0330
0	273	23.76	0.0856	4.294	0.0159	1.039	1.613	0.0354	0.247		0.0044	4.684	0.0381	1.554	0.1697		0.0439
5	278	26.86	0.115	5.157					0.311			5.453		1.899			0.0576
10	283	30.16	0.1542	6.149	0.0306	1.50	2.201	0.0606	0.389	0.0245	0.0085	6.339	0.0699	2.302	0.2648	0.017	0.0746
15	288	33.76	0.196	7.283					0.481			7.298		2.768			0.0956
20	293	37.75	0.246	8.572	0.0568	2.069	3.119	0.0996	0.589	0.0419	0.0156	8.334	0.1227	3.305	0.3996	0.0298	0.1213
25	298	42.15	0.306	10.03					0.716			9.489		3.9197			0.1527
30	303	47.07	0.377	11.67	0.1008	2.824	4.232	0.1578	0.864	0.0688	0.0275	10.80	0.2068	4.619	0.5848	0.0489	0.1907
35	308		0.462	13.498								12.21		5.411			0.2349
40	313		0.562	15.54	0.1722	3.765	5.609	0.2412		0.1097	0.0464	13.73	0.336	6.303	0.8306	0.0748	0.2876
45	318		0.681	17.81					1.228			15.45		7.303			0.3499
50	323		0.817	20.33	0.2836	4.98	7.257	0.3589	1.702	0.1696	0.0754	17.26	0.528	8.417	1.1466	0.121	0.4228
55	328																0.5057
60	333		1.118					0.0075	2.306	0.2549	0.1186	20.89	0.809		1.549	0.1863	0.6010
65	338																0.7078
70	343		1.55		0.6979	8.14	11.719	0.7301	3.061	0.3733	0.1812	25.79	1.195			0.2689	0.8266
75	348																
80	353		2.08		1.047	10.20		1.0052	3.99	0.533	0.269	31.38	1.729		2.700	0.3818	1.1169
85	358											34.12					
90	363		2.76		1.531	12.55		1.355	5.121	0.7439	0.3915	36.58	2.445			0.5369	1.4828
95	368											39.91					
100	373		3.60		2.184	15.40		1.795	6.478	1.0159	0.556		3.384		4.333	0.7354	1.9505
105	378																
110	383		4.65		3.045	18.34		2.331	8.092				4.595			0.9924	2.5164
115	388																
120	393		5.89		4.159	21.77		2.984	9.992		1.059		6.131		6.999	1.267	3.1911
125	398																
130	403		7.38		5.572	25.69		3.766	12.209		1.423		8.050			1.740	3.956
135	408																
140	413		9.15					4.694	14.768		1.885				10.399	2.2457	4.95
145	418																
150	423		11.28						17.711		2.499					2.824	6.073

9.2 Density ρ of various liquids at atmospheric pressure

Temperature	Ethane C ₂ H ₆	Acetone (CH ₃) ₂ CO	Ammonia NH ₃	Ethyl Alcohol C ₂ H ₅ OH	n-Butane C ₄ H ₁₀	i-Butane C ₄ H ₁₀	Benzol C ₆ H ₆	Aniline C ₆ H ₅ NH ₂	Ether C ₂ H ₅ OC ₂ H ₅	Formic acid CH ₂ O ₂	Acetic acid C ₂ H ₄ O ₂	n-Propane C ₃ H ₈	Methyl Alcohol CH ₄ O	Sulphurous acid H ₂ SO ₃	Carbon disulphide CS ₂	Toluol C ₇ H ₈	Carbon tetrachloride CCl ₄	Heavy Water D ₂ O
-100	223	0.5589	0.920		0.6900				0.842					1.432				
-90	183	0.5479			0.6827											0.9697		
-80	193	0.5367			0.6744							0.6240				0.9604		
-70	203	0.5250			0.6663							0.6134				0.9509		
-60	213	0.5125			0.6577							0.6025				0.9419		
-50	223	0.4999	0.868	0.695	0.6492				0.790			0.5910		1.555	1.362	0.9327		
-40	233	0.4850	0.855		0.6400							0.5793				0.9234		
-30	243	0.4700			0.6306	0.6156						0.5680		1.509		0.9141		
-20	253	0.4526	0.832		0.6210	0.6052						0.5555				0.9049	1.670	
-10	263	0.4339			0.6107	0.5940						0.5430		1.460		0.8956		
0	273	0.4117	0.812	0.636	0.6008	0.5835	0.9001	1.039	0.736			0.5300	0.810	4.435	1.292	0.8863	1.630 (1.105)	
10	283	0.3865			0.7990	0.5898	0.5718	0.8920				0.5160	0.801			0.8769		1.107
20	293	0.3502	0.791	0.609	0.7902	0.5788	0.5590	0.8790	1.022	0.714	1.220	1.049	0.5015	0.792	1.380	1.262	0.6877	1.585
30	303	0.2860			0.7815	0.5665	0.5462	0.8675				0.4860	0.783			0.8583		1.105
40	313		0.765		0.7726	0.5546	0.5340	0.8576		1.192	1.028	0.4690	0.774			0.8489	1.545	1.100
50	323		0.756	0.561	0.7634	0.5422	0.5198	0.8460	0.996	0.676	1.184	1.018	0.4500	0.765		0.8395		
60	333		0.740		0.7546	0.5284	0.5052	0.8357		1.169	1.003	0.4328	0.755			0.8301	1.505	1.090
70	343				0.7452	0.5148	0.4900	0.8248				0.4090	0.746			0.8205		
80	353				0.7357	0.5003		0.8145				0.980	0.736			0.8110	1.460	1.070
90	363				0.7260	0.4848		0.8041				0.3230	0.725			0.8012		
100	373			0.458	0.7158	0.4680		0.7927	0.951	0.611		0.960	0.714	1.110		0.7914	1.420	1.040
110	383				0.7048	0.4492		0.7809					0.702			0.7813		
120	393				0.6927	0.4272		0.7692					0.691			0.7710		
130	406				0.6791	0.4006		0.7568					0.678			0.7608		
140	413					0.3602	0.7440									0.7501		
150	423				0.2900		0.7310		0.518		0.896					0.7392	1.310	

9.4 Extract of important legal units for centrifugal pumps

Physical dimension	Formula symbol	Legal units SI - Units		Further legal units (not complete)	No longer authorised	Recomm. units	Remarks
Length	l	m	Meter	km, dm, cm mm		m	Basic unit\
Volume	v	m ³		dm ³ , cm ³ , mm ³ litre (1l = 1dm ³)	cbm, cdm	m ³	
Capacity, volume flow	Q V	m ³ /s		m ³ /h, l/s		l/s and m ³ /s	
Time	t	s	Second	s, ms, ns... min., h, d		s	Basic unit
Rotat. speed	n	1/s		1/min.		1/min.	
Mass	m	kg	Kilogram	g, mg, ton ton (1t = 1000 kg)	pound honoured weight	kg	Basic weight The mass of commercial commodity is described as weight
Density	ρ	kg/dm ³		kg/dm ³	kg/dm ³ and kg/m ³		The designation "specific gravity" must no longer be employed, because it is ambiguous (see DIN 1305)
Moment of inertia	J	kg-m ²				kg-m ²	Moment of inertia 2.grade
Mass flow	m	kg/s	t/s, t/h, kg/h		kg/s and t/s		
Force	F	N	Newton (= kg m/s ²)	kN, mN,	kp, Mp	N	1 kp = 9.81 N. The weight force is the product of mass m by the local gravitational g
Pressure	p	Pa	Pascal (1bar = 10 ⁵ Pa)	bar m WS, Torr	kp/cm ² , at,	bar	1 at = 0.981 bar = 9.81 · 10 ⁴ Pa 1 mm Hg = 1.333 mbar 1 mm WS = 0.098 mbar
Mechanical Stress (strength)		Pa	Pascal (= n/m ²)	N/mm ² ,	kp/cm ²	N/mm ²	1 kp/mm ² = 9.81 N/mm ²
Bending Moment torque	M T	N m			kp m,	N m	1 kp m = 9.81 J
Energy, work, Quantity of heat	W Q	J	Joule (= N m = W s)	kJ, W s, kW h 1 kW h = 3600 kJ	kp m kcal, cal, WE	J and kJ	1 kp m = 9.81 J 1 kcal = 4.1868 kJ
Head	H	m	Meter		m.l.c.	m	The head is the work done in J = N m applied to the mass unit of the medium pumped, related to the weight force of this mass unit N.
Power	P	W	Watt (= J/s = N m/s)	MW, kW	kp m/s, PS	kW	1 kp m/s = 9.81 W 1 PS = 736 W
Temperature difference	T	K	Kelvin	°C	°K, dge.	K	Basic unit
Kinematic viscosity	ν	m ² /s			St (Stokes) 0E...	m ² /s	1 St = 10 ⁻¹ m ² /s 1cSt = 1 mm ² /s
Dynamic viscosity	n	Pa s	Pascal- second (= N s/m ²)		P (Poise)	Pa s	1 P = 0.1 Pa s
Specific speed	nq	1				1	nq = 333.n. $\frac{Q_{opt.}}{(g.H_{opt})^{3/4}}$ in SI-units (m and s)

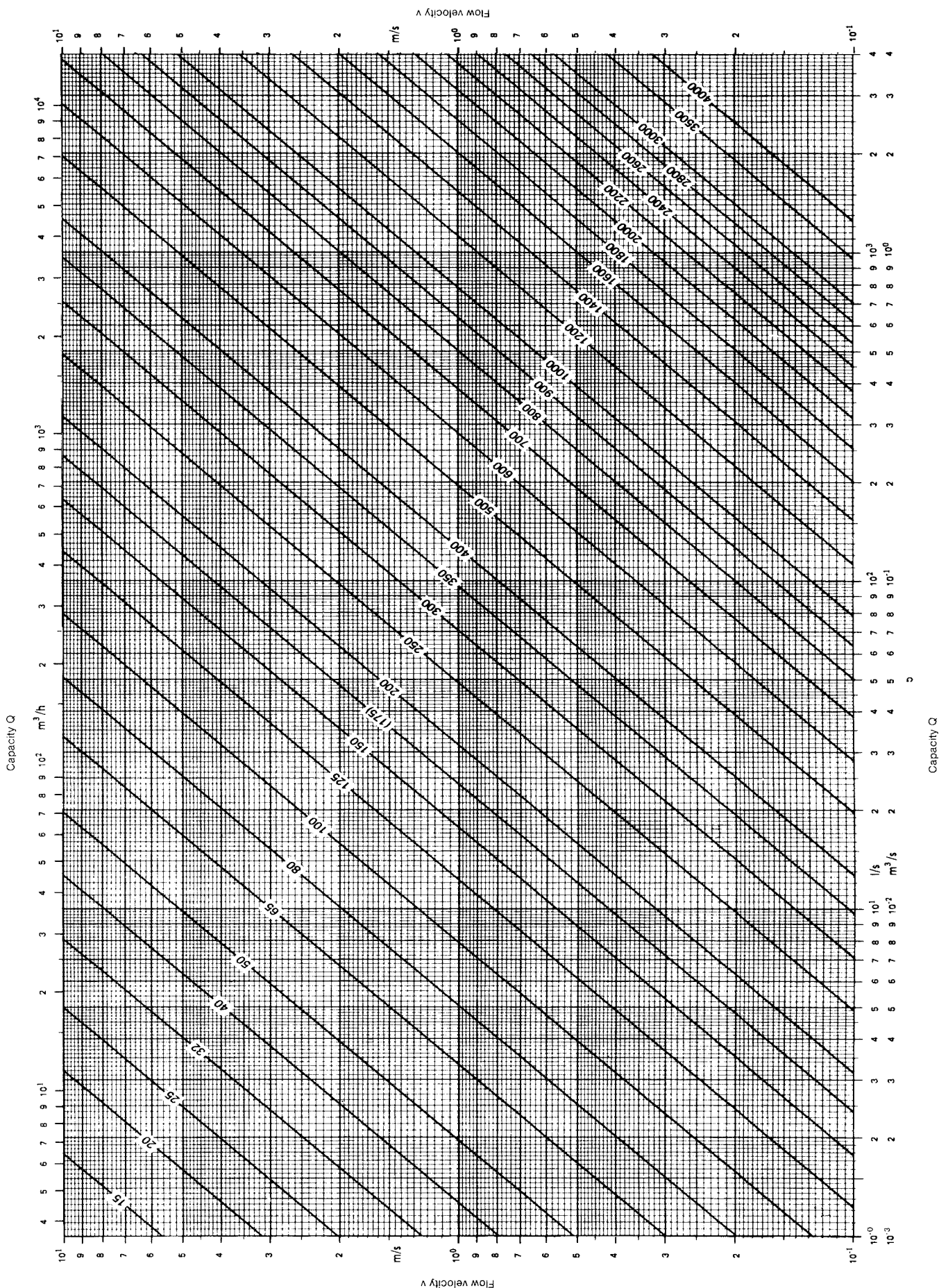
9.5 Conversion of British and U.S. Units

			British		U.S.	
Length	1 mil		25.4	μm	25.4	μm
	1 point		0.3528	mm	0.3528	mm
	1 line		0.635	mm	0.635	mm
	1 inch	(in)	25.4	mm	25.4	mm
	1 hand		10.1	cm	10.16	cm
	1 link	(li)	20.1168	cm	20.1168	cm
	1 span		22.86	cm	22.86	cm
	1 foot	(ft) = 12 in	0.3048	m	0.3048	m
	1 yard	(yd) = 3 ft = 36 in	0.9144	m	0.9144	m
	1 fathom	(fath) = 2 yd	1.8288	m	1.8288	m
	1 rod	(rd)	5.0292	m	5.0292	m
	1 chain	(ch)	20.1168	m	20.1168	m
	1 furlong	(fur)	201.168	m	201.168	m
	1 mile	(mi)				
	(statute mile)	= 1760 yd	1.6093	km	1.6093	km
	1 nauticle mile		1.8532	km	1.8532	km
Area	1 circular mil		506.709	μm ²	506.709	μm ²
	1 circular inch		5.067	cm ²	5.067	cm ²
	1 square inch	(sq in)	6.4516	cm ²	6.4516	cm ²
	1 square link	(sq li)	404.687	cm ²	404.687	cm ²
	1 square foot	(sq ft)	929.03	cm ²	929.03	cm ²
	1 square yard	(sq yd)	0.8361	m ²	0.8361	m ²
	1 square rod	(sq rd)	25.2929	m ²	25.2929	m ²
	1 square chain	(sq ch)	404.686	m ²	404.686	m ²
	1 rood		1011.7124	m ²	1011.7124	m ²
	1 acre		4046.86	m ²	4046.86	m ²
	1 square mile	(sq mi)	2.59	km ²	2.59	km ²
Volume	1 cubic inch	(cu in)	16.387	cm ³	16.387	cm ³
	1 board foot	(fbm)	2.3597	dm ³	2.3597	dm ³
	1 cubic foot	(cu ft)	28.3268	dm ³	28.3268	dm ³
	1 cubic yard	(cu yd)	0.7646	m ³	0.7646	m ³
	1 register ton	(RT) = 100 cu ft	2.8327	m ³	2.8327	m ³
	1 british shipping ton	= 42 cu ft	1.1897	m ³	-	-
	1 US shipping ton	= 40 cu ft.	-	m ³	-	m ³
	Basic unit gallon for fluids		59.1939	mm ³	61.6119	mm ³
	1 minimum	(min.)	1.1839	cm ³	-	-
	1 fluid scruple		3.5516	cm ³	-	-
	1 fluid drachm	(fl. dr.)	-	cm ³	3.6967	cm ³
	1 fluid dram	(fl. oz)	-	cm ³	118.2948	cm ³
	1 fluid ounce	(fl. oz)	28.4131	cm ³	118.2948	cm ³
	1 gill	(gi)	142.065	cm ³	0.9464	dm ³
	1 pint	(liq pt)	0.5683	dm ³	0.9464	dm ³
	1 quart	(liq qt)	1.1365	dm ³	-	-
	1 pottle		2.2730	dm ³	-	-
	1 gallon	(gal)	4.5460	dm ³	3.7854	dm ³
	1 peck		9.0922	dm ³	-	-
	1 bushel		36.3687	dm ³	-	-
	1 US oil-barrel (for crude oil)	-	-	0.159	m ³	-
	1 quarter		0.291	m ³	-	-
	1 chaldron		1.3093	m ³	-	-
Basic unit bushel for dry goods	1 dry pint	(drypt)	-		0.5506	dm ³
	1 dry quart	(dry at)	-		1.1012	dm ³
	1 peck	(pk)	-		8.8098	dm ³
	1 bushel	(bu)	36.3687	dm ³	35.2329	dm ³
	1 dry barrel	(bbl)	-		0.1156	m ³
Mass and weight Avoirdupois system (trade and sommerce weights)	1 grain	(gr)	64.7989	mg	64.7989	mg
	1 dram	(dr avdp)	1.7718	g	1.7718	g
	1 ounce	(oz advp)	28.3495	g	28.3495	g
	1 pound	(lb)	0.4536	kg	0.4536	kg
	1 stone		6.3503	kg	-	-
	1 quarter		12.7006	kg	-	-
	1 cental		-		45.3592	kg
	1 short hundredweight	(sh cwt)	-		45.3592	kg
	1 hundredweight	(cwt)	50.8024	kg	-	kg
	1 long hundredweight	(l cwt)	-		50.8024	kg
	1 short ton	(sh tn)	-		907.1849	kg
	1 ton	(l tn)	1016.0470	kg	-	-
	1 long ton	(l tn)	-		1016.0470	kg
	1 pennyweight	(dwt)	1.5552	g	1.5552	g
	1 troy ounce	(oz tr)	31.1035	g	32.1035	g
	1 troy pound	(lb t)	-		0.3732	kg

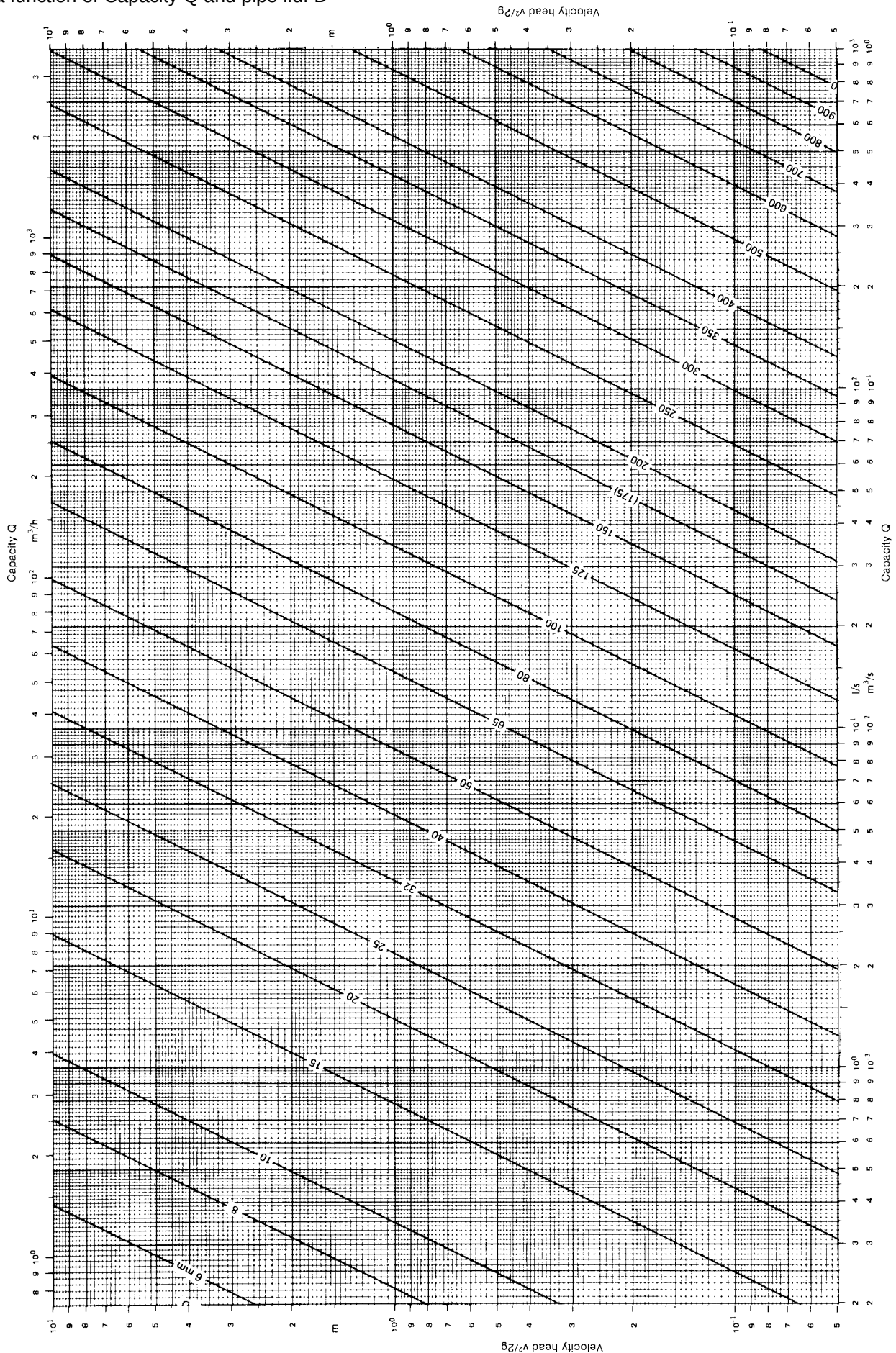
			British		U.S.	
Density	1 ounce (av) per cubic foot	(oz/cu. ft)	0.0010	kg/dm ³	0.0010	kg/dm ³
	1 pound per cubic foot	(lb/cu ft)	0.0160	kg/dm ³	0.0160	kg/dm ³
	1 ounce (av) per cubic inch	(oz/cu in)	1.7300	kg/dm ³	1.7300	kg/dm ³
	1 pound per cubic inch	(lb/cu in)	27.6799	kg/dm ³	27.6799	kg/dm ³
	1 short ton per cubic yard	(shn/cu yd)	-		1.1865	kg/dm ³
	1 long ton per cubic yard	(ltn/cu yd)	-		1.3289	kg/dm ³
	1 pound per gallon	(lb/gal)	0.09978	kg/dm ³	0.1198	kg/dm ³
Velocity	1 foot per second	(ft/s)	0.3048	m/s	0.3048	m/s
	1 foot per minute	(ft/min)	0.00508	m/s	0.00508	m/s
	1 yard per second	(yd/s)	0.9144	m/s	0.9144	m/s
	1 yard per minute	(yd/min)	0.01524	m/s	0.01524	m/s
Capacity (rate of volume flow)	1 gallon per second		4.5460	l/s	3.7854	l/s
	1 gallon per minute	(gpm)	0.07577	l/s	0.06309	l/s
	1 cubic foot per second	(cusec)	28.3268	l/s	28.3268	l/s
	1 cubic yard per second		0.7646	m ³ /s	0.7646	m ³ /s
Mass flow	1 ounce per second	(oz/s)	28.3495	g/s	28.3495	g/s
	1 ounce per minute	(oz/min)	0.4725	g/s	0.4725	g/s
	1 pound per sec	(lb/s)	0.4536	kg/s	0.4536	kg/s
	1 pound per minute	(lb/min)	0.00756	kg/s	0.00756	kg/s
	1 short ton per hour	(shn/h)	-		0.2520	kg/s
	1 ton per hour		0.2822	kg/s	-	
	1 long ton per hour	(ltn/h)	-		0.2822	kg/s
Force (weight force)	1 ounce (force)	(oz)	0.2780	N	0.2780	N
	1 pound (force)	(lb)	4.4483	N	4.4483	N
	1 short ton (force)	(shn)	8.8964	kN	8.8964	kN
	1 long ton (force)	(ltn)	9.9640	kN	9.9640	kN
Pressure	1 $\frac{\text{pound(force)}}{\text{square foot}}$	$\frac{\text{lb(force)}}{\text{sq foot}}$	47.88025	Pa	47.88025	Pa
	1 $\frac{\text{pound(force)}}{\text{square inch}}$	$\frac{\text{lb(force)}}{\text{sq inch}}$ (psi)	68.9476	mbar	68.9476	mbar
	1 $\frac{\text{short ton (force)}}{\text{square inch}}$	$\frac{\text{sh tn (force)}}{\text{sq inch}}$	137.8951	bar	137.8951	bar
	1 inch H ₂ O	(in H ₂ O)	2.4909	mbar	2.4909	mbar
	1 foot H ₂ O	(ft H ₂ O)	29.8907	mbar	29.8907	mbar
	1 inch HG	(in HG)	33.8663	mbar	33.8663	mbar
Mechanical Stress	1 $\frac{\text{pound (force)}}{\text{square inch}}$	$\frac{\text{lb (force)}}{\text{sq in}}$	0.006895	N/mm ²	0.006895	N/mm ²
	1 $\frac{\text{short ton(force)}}{\text{square inch}}$	$\frac{\text{sh tn (force)}}{\text{sq in}}$	13.78951	N/mm ²	13.78951	N/mm ²
Work, energy, quantity of heat, internal (intrinsic) energy and enthalpy	1 foot-pound	(ft lb)	1.3558	J	1.3558	J
	1 horse power hour	(Hp h)	2.6841	MJ	1.0558	kJ
	1 British Thermal Unit	(BTU)	1.0558	kJ	1.0558	kJ
Power (heat flow)	1 foot-pound (av) per second	$\frac{(\text{ft lb})}{\text{s}}$	1.3558	W	1.3558	W
	1 Horse power (Hp)	(Hp h)	0.7457	kW	0.7457	kW
	1 Brithish Thermal unit per second	(BTU/s)	1.0558	kW	1.0558	kW
Dynamic viscosity	1 $\frac{\text{pound (mass)}}{\text{foot x second}}$	$\frac{\text{lb (mass)}}{\text{ft x s}}$	1.4882	Pa s	1.4882	Pa s
	1 $\frac{\text{pound (force) x second}}{\text{square foot}}$	$\frac{\text{lb (force) s}}{\text{sq ft}}$	47.8803	Pa s	47.8803	Pa s

Temperature		Conversion of temperature points :	Conversion of temperature points :
$T = \frac{5}{9} t_F + 255.37;$	$t = \frac{5}{4} (t_F - 32);$	$\Delta T = \Delta t = \frac{5}{9} \Delta t_F$	
$T = \frac{5}{4} t_R + 273.15;$	$t = \frac{5}{4} t_R$	$\Delta T = \Delta t = \frac{5}{4} \Delta t_R$	

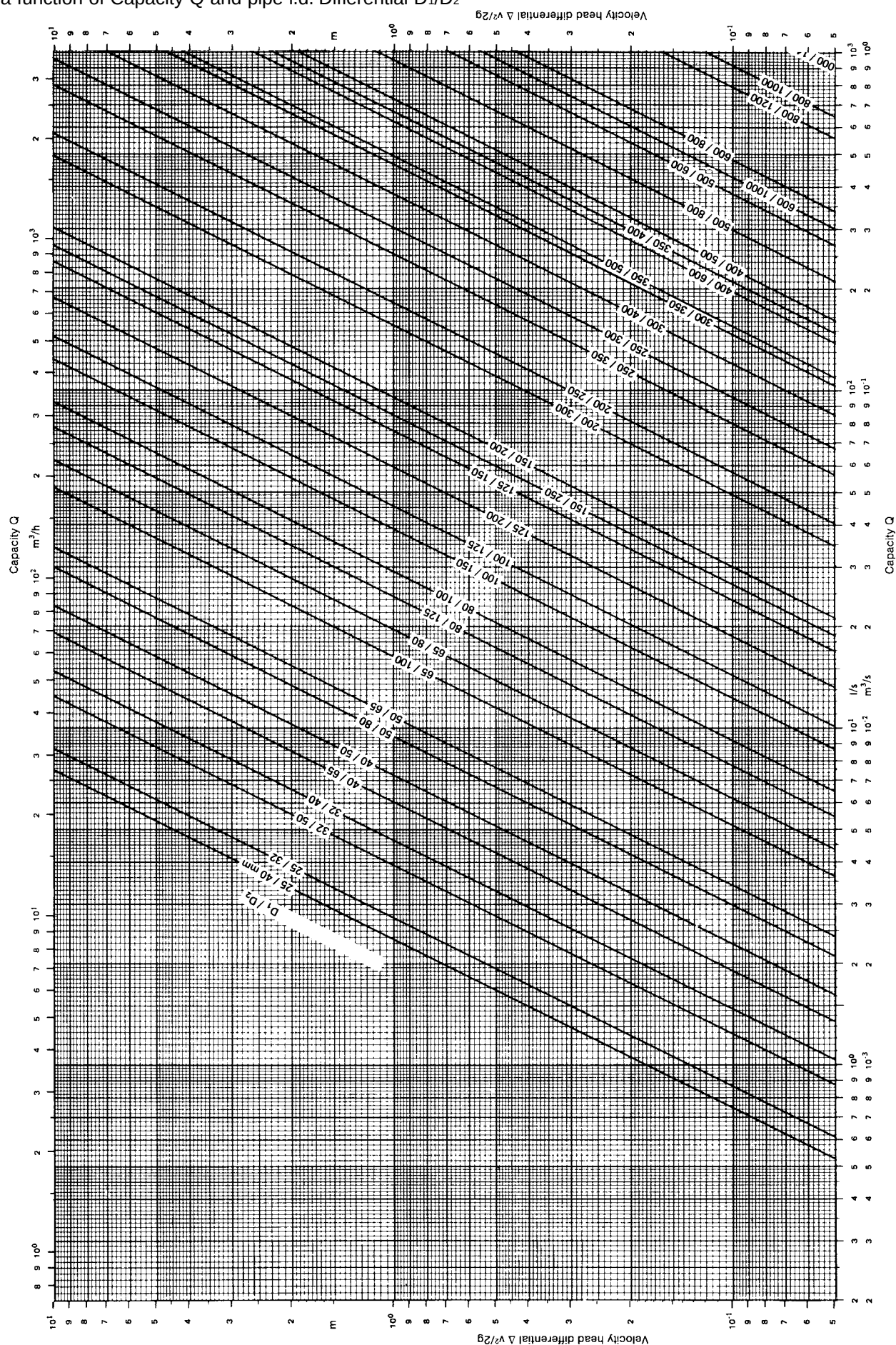
Where : T = thermodynamic temperature in K
 t_F = Fahrenheit temperature in °F
 t_R = reamur temperature in °R
 t_C = Celcius temperature in °C



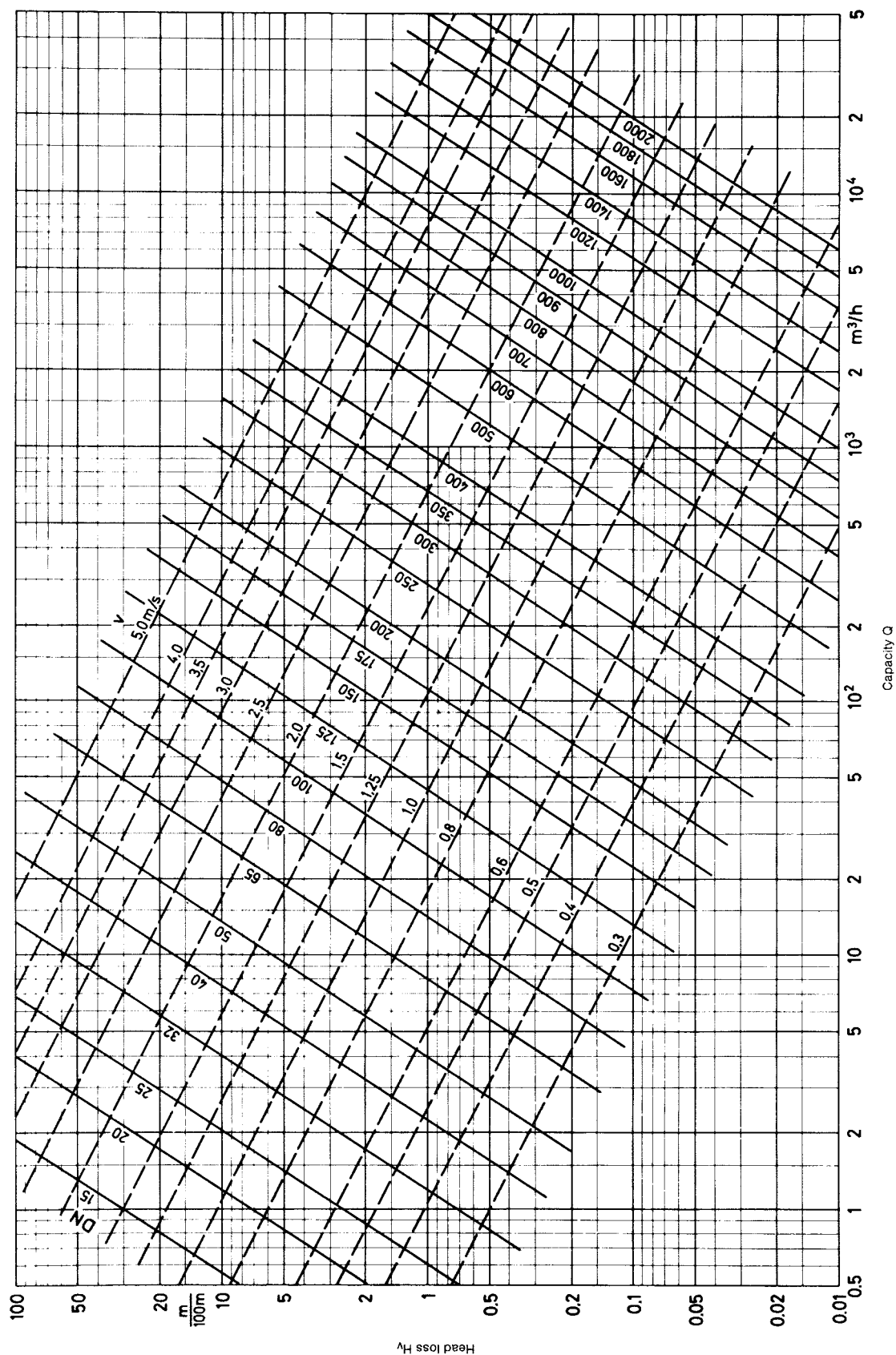
9.8 Graph for calculating velocity head $v^2/2g$
as a function of Capacity Q and pipe i.d. D



9.8 Graph for calculating velocity head differential $\Delta v^2/2g$ as a function of Capacity Q and pipe i.d. Differential D_1/D_2



9.9 Graph for calculating Head Loss H_v
as a Function of i.d. of pipe D, Flow velocity v and Capacity Q



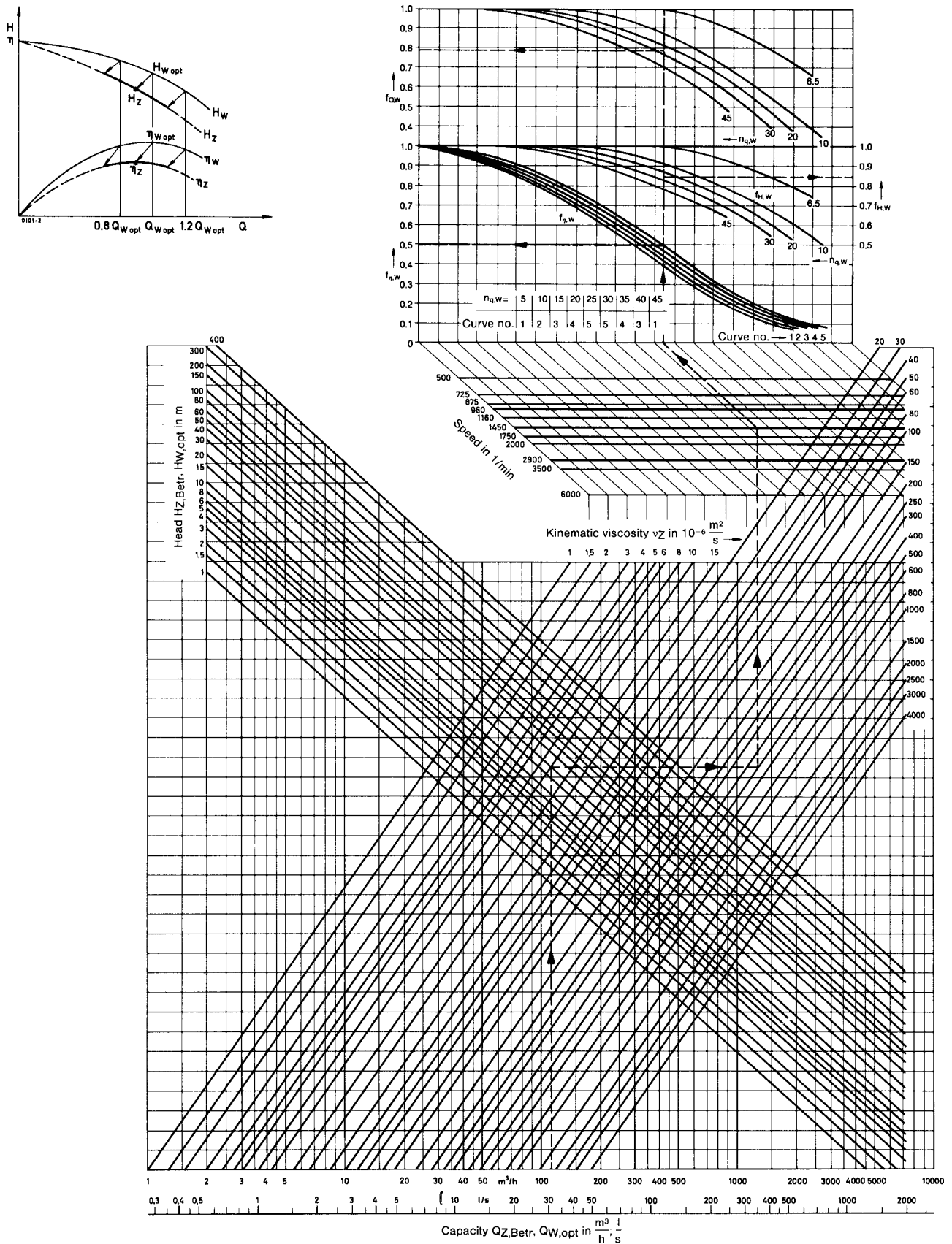
9.10 Graph for calculating conversion factors $f_{Q,W}$ and $f_{H,W}$ and $f_{\eta,W}$ for viscous liquids

Available : data for operation with water

Required : data for operation with viscous liquid

Calculation example : see page 21

Calculation chart : see page 44

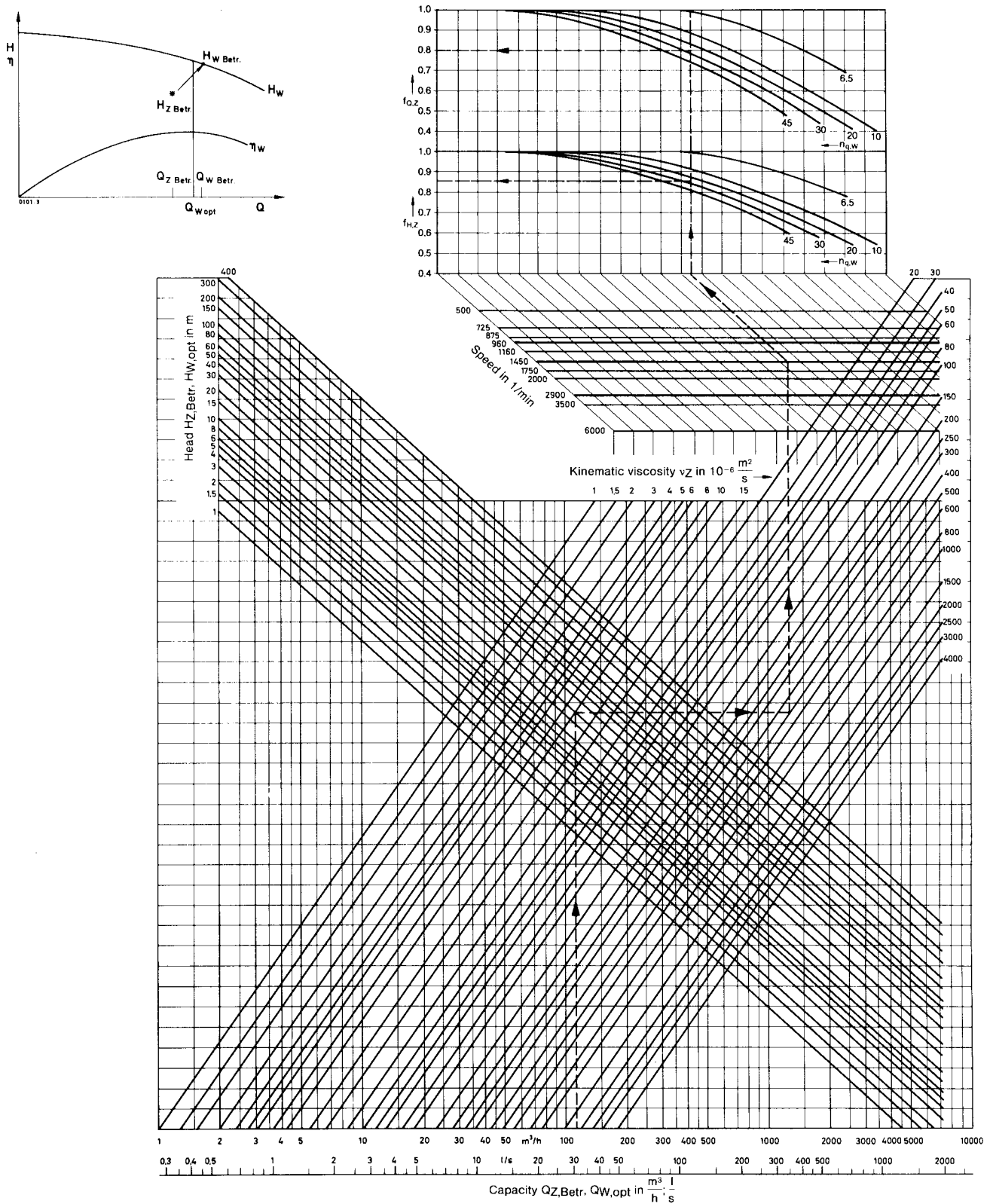


9.11 Graph for calculating conversion factors $f_{Q,Z}$ and $f_{H,Z}$ for viscous liquids

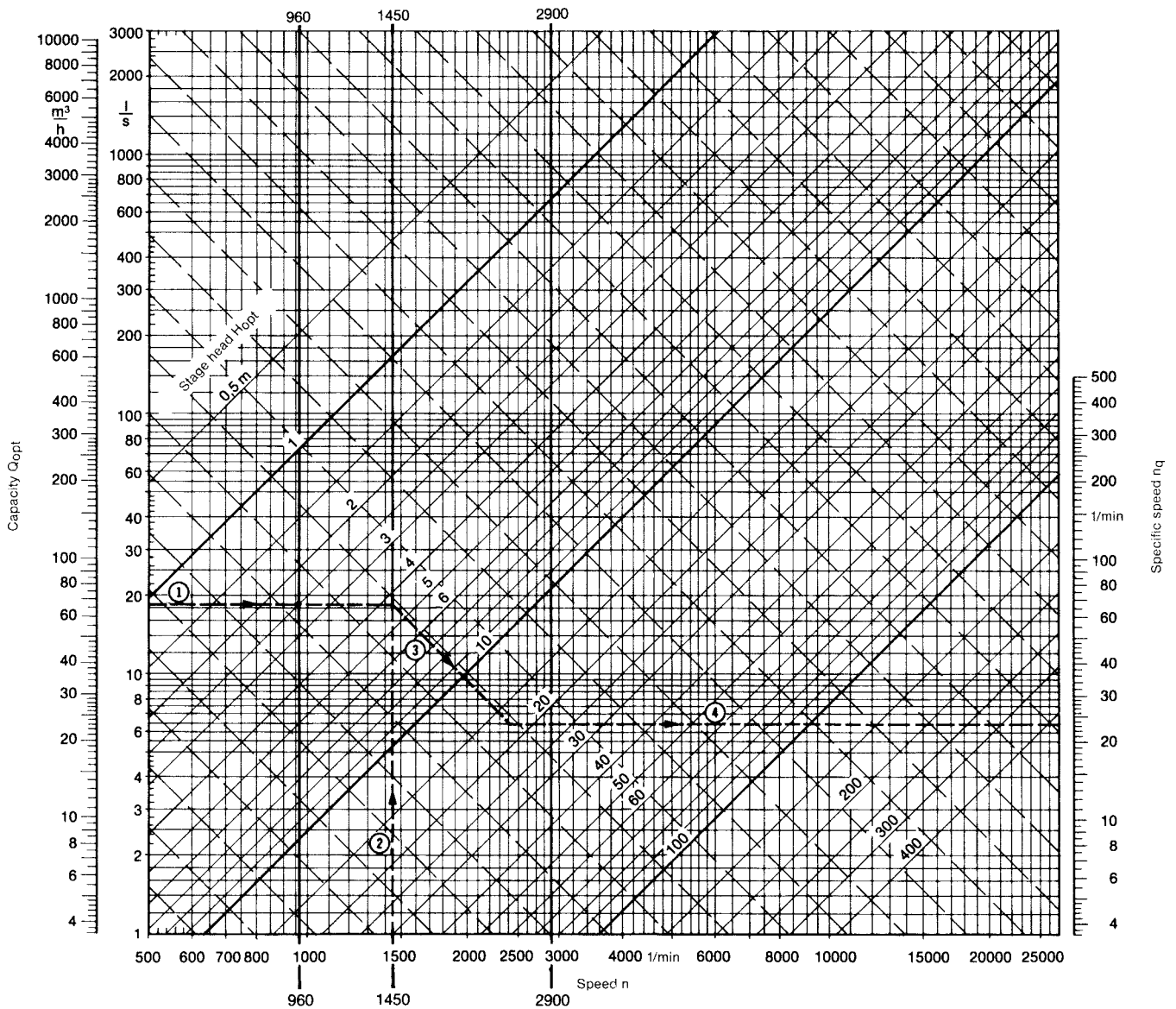
Available : data for operation with viscous liquid

Required : data for operation with water

Calculation chart : see page 44



9.12 Graph for calculating Specific Speed n_q



Equations	Units				
	$Q_{opt.}$	$H_{opt.}$	n	n_q	$g=9.81$
$n_q = n \cdot \frac{\sqrt{Q_{opt.} / 1}}{(H_{opt.} / 1)^{3/4}}$	m^3/s	m	$1/min.$	$1/min.$	
$n_q = 333 \cdot n \cdot \frac{\sqrt{Q_{opt.}}}{(H_{opt.} \cdot g)^{3/4}}$	m^3/s	m	$1/s$	1	m/s^2
$n_q = 5.55 \cdot n \cdot \frac{\sqrt{Q_{opt.}}}{(H_{opt.} \cdot g)^{3/4}}$	m^3/s	m	$1/min.$	1	m/s^2

All equations give numerically equal results

With multistage pumps use the stage head.

With double-entry impeller pumps use only half the capacity

Example : $Q_{opt.} = 66m^3/hr. = 18.3 l/s;$ $n = 1450 1/min.;$ $H_{opt.} = 17.5m$ Established : $n_q = 23 1/min.$

Type series		Quotation no.	
Rated Speed	1/min.	Item no.	

Schedule for calculating the Operating point and pump size for Handling viscous liquids.

Operating point

Available data :

Capacity	Q_w	l/s
Head	H_w	m
Speed	n	1/min.
Kinematic viscosity	ν_z	m^2/sec
Density	ρ_z	kg/dm^3
Gravitational constant	g	m^2/s

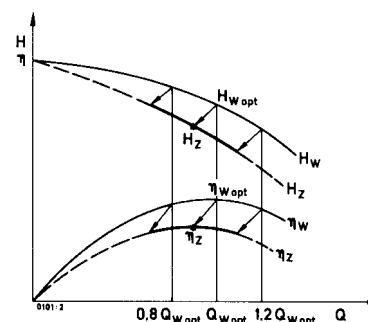
To determine the new operating data it is also necessary to calculate the data at b.e.p.

Capacity	$Q_{w, opt.}$	l/s
Head	$H_{w, opt.}$	m
Efficiency	$\eta_{w, opt.}$	-

Procedure

n _{q,w} from graph in 9.12						1/min
f _{q,w}	from fig. 25a or sect. 9.10, page 41					-
f _{h,w}						-
f _{n,w}						-
Q/Q _{opt.}		0	0.8	1.0	1.2	-
Q _w	from charact. curve booklet for 4 popints on curve					l/s
H _z						m
η _w		0				-
Q _z = Q _w .f _{q,w}		0				l/s
H _z =		= <u>H_w</u>	= <u>H_w.f_{h,w}</u>	= <u>H_w.f_{h,w}</u>	= <u>H_w.f_{h,w}</u>	m
n _z = η _w .f _{n,w}		0				-
P _z = $\frac{\rho_z.g.H_z.Q_z}{\eta_z.1000}$						kW

These values mean 4 points on Q_H and Q_{η} lines plus 3 points on the Q_P line are established. Plotted over Q (see fig. 28)



Pump size

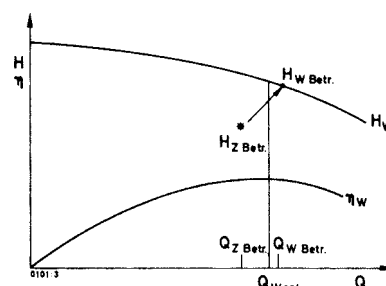
Available data :

Capacity	$Q_z, Betr$	l/s
Head	$H_z, Betr$	m
Kinematic viscosity	ν_z	m^2/sec
Density	ρ_z	kg/dm^3

Procedure :

n selected		1/min.
$n_{q,w}^{3)}$ from graph in 9.12		1/min.
$f_{Q,z}$ from fig. 25b or section 9.11, page 42		-
$f_{H,z}$		-
$Q_{w, Betr} = \frac{Q_z, Betr}{f_{Q,z}}$		l/s
$H_{w, Betr} = \frac{H_z, Betr}{f_{H,z}}$		m

³⁾ where $Q_z, Betr = Q_{opt.}$
 $H_z, Betr = H_{opt.}$ } approx.



Calculation in graphic form

Notes

[illegible]

[illegible]