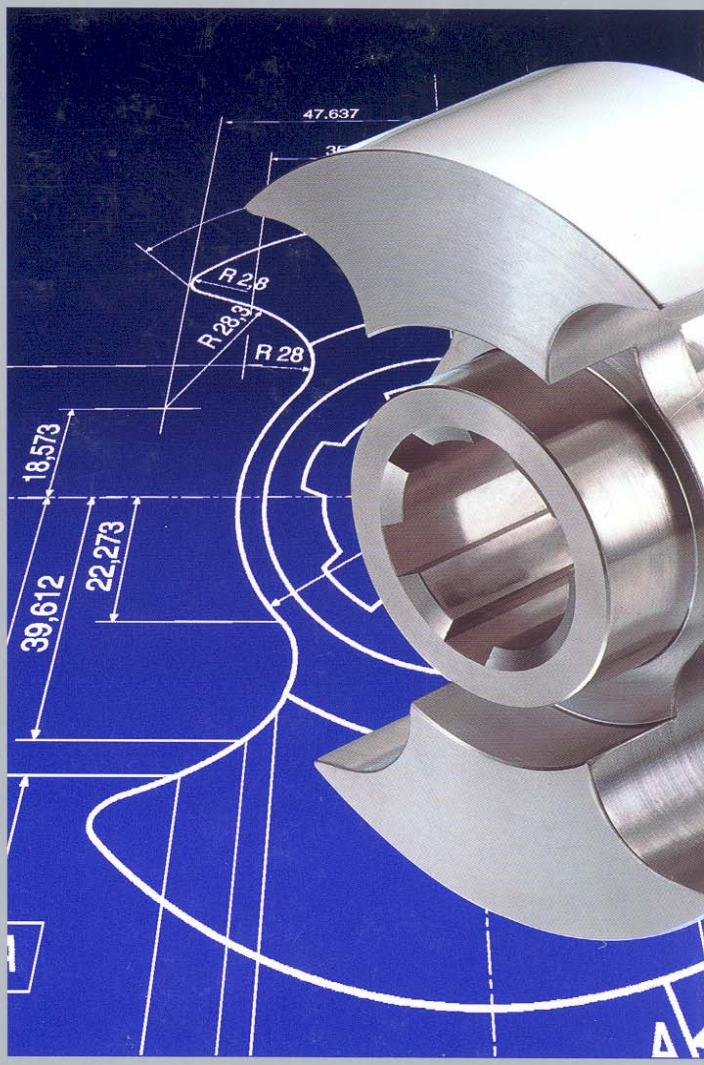
ABC of Pumps



Fristam
PUMPEN

For better understanding of the following chapters, we firstly will define and explain the technical terms relating to pump technology used in this brochure. The reader will find these terms in alphabetical order in the index. Measures and conversion formulae are summarised in a table.

Flow rate [m≈/h]

The flow rate is the effective volume flowing per unit of time through the discharge connection of a pump.

In order to optimize the pump design, the flow rate must be accurately determined.

Total head [m]

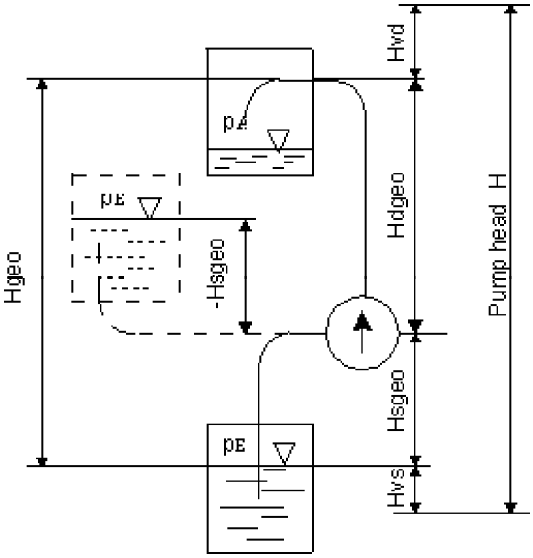
The total head is the effective mechanical energy transferred by a pump to the fluid as a function of the weight force of the fluid.

The total head results as follows:

$H = H_{geo} + H_V + p$

It consists of:

- the difference in height to overcome between the suction side and the discharge side of an installation.
  $H_{geo} = H_{dgeo} \pm H_{sgeo}$
- the friction loss resulting from pipe walls, fittings and valves within the plant.
  $H_V = H_{VS} + H_{Vd}$
- the pressure difference
  $p = p_A \pm p_E$

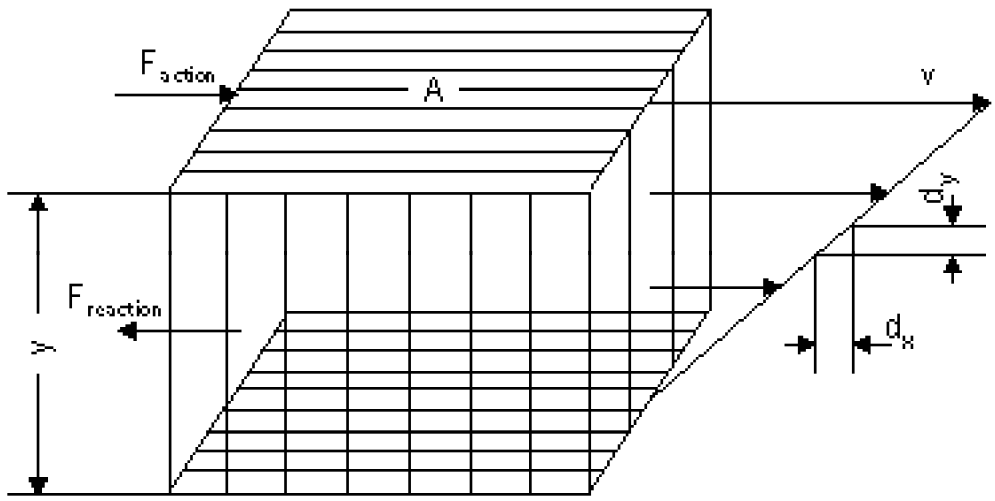


Power consumption

The power consumption is the total energy transferred by the pump to the discharge flow.

dynamic viscosity η

Looking at two parallel plates with the surface A and the distance y, displaced against each other as a result of a force F_{action} with a velocity v, a force $F_{reaction}$ opposes to this displacement and increases with increasing dynamic viscosity of the medium between the two plates.



The ratio of F to A is called shear stress τ.

$$\tau = \frac{F}{A}$$

The shear stress τ increases in proportion to the shear velocity D and the dynamic viscosity η.

$$\tau = D \times \eta$$

The ratio of v to y is defined as shear velocity D.

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

Thus the resulting dynamic viscosity η:

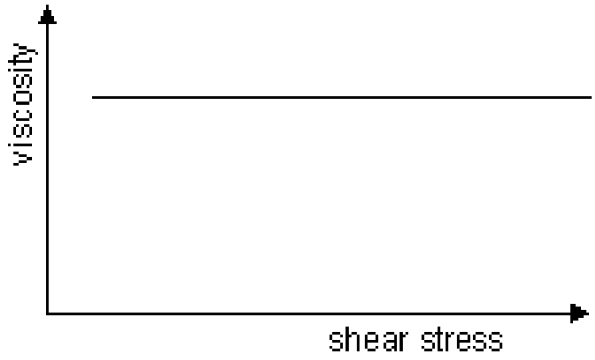
$$\eta = \frac{\tau}{D}$$

Thus, the dynamic viscosity η is a characteristic parameter of the fluid concerned and depends on the temperature. Therefore the viscosity is always indicated together with the corresponding temperature.

Flow behaviour of fluids

Ideal viscous flow behaviour:

Fluids with an ideal viscous flow behaviour are called Newtonian fluids. They are viscous fluids with linear molecules. They show a proportional flow behaviour.



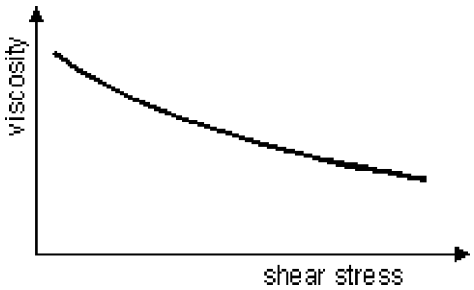
Typical Newtonian fluids are: water, salad oil, milk, sugar solutions, honey.

Pseudoplastic flow behaviour:

The flow behaviour of fluids depends on their physicochemical properties. Adding a filling agent to a pure solvent, will increase the viscosity and change the flow behaviour.

With increasing shear stress, in general the viscosity of highly molecular products in solutions and melts tends to decrease.

Such a flow behaviour is called pseudoplastic.

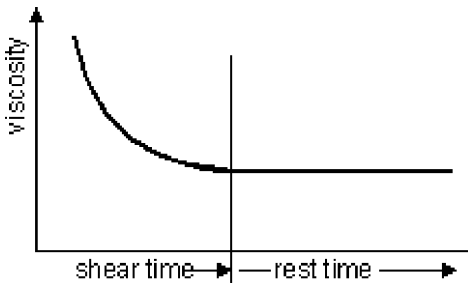


Examples of pseudoplastic flow behaviour:
condensed milk, orange juice

Irreversible flow behaviour:

Fluids deformed under applied shear stress in a way that the structure after the destructive phase (shear time) can not be restored show an irreversible flow behaviour.

The result is a permanent, shear time dependent change of viscosity.



Example for irreversible flow behaviour:
Yoghurt

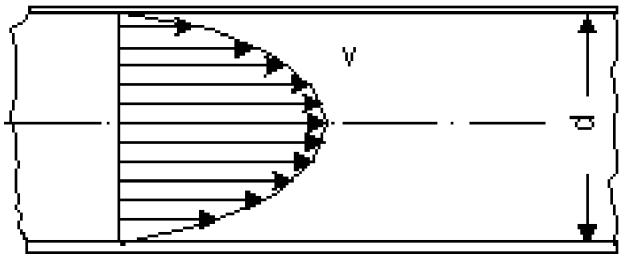
Types of flow

Depending on the Reynolds number, the flow passing through a pipe shows specific, typical flow patterns with different physical properties.

In this context the generation of a laminar or turbulent flow is of particular concern.

Laminar flow

In case of a laminar flow, the particles move in a streamline form and parallely to the pipe axis without being mixed.



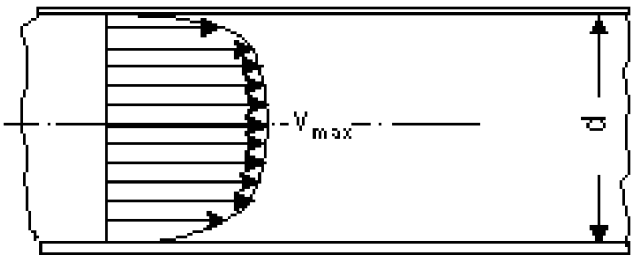
The roughness of the inside wall of pipes has no effect on the friction loss.

You will find a laminar flow mainly with high viscous fluids.

The loss of head changes linearly with the flow velocity.

Turbulent flow

In case of a turbulent or vortical flow the particles are mixed because of the movement along the pipe axis and an additional, transverse movement..



The roughness of the pipe inside has great effect on the friction loss.

Turbulent flows are mainly found with water or fluids similar to water.

The loss on pump head varies by square of the flow velocity.

Reynolds number

The Reynolds number describes the correlation between the flow velocity v, the viscosity η and the inner diameter of the pipe *d*.

The Reynolds number has no dimension.

$$Re = \frac{v \times d_i \times \rho}{\eta}$$

Flow velocity	v	[m/s]
Viscosity	η	[Pa s]
Inner pipe diameter	d_i	[mm]
Density	ρ	[kg/dm≈]

With a Reynolds number of 2320 the laminar flow passes to a turbulent flow.

Laminar flow < Re_{krit} = 2320 < turbulent flow

Example:
In one second, 2 litres of acetic acid passes through a pipe with a nominal bore of 50 mm.
The acetic acid has a kinematic viscosity of η = 1.21 mPa s = 0.00121 Pa s
and a density of 1.04 kg/dm≈.

Is the flow laminar or turbulent?

The average flow velocity amounts to:

$$v = \frac{Q}{A} = \frac{Q}{\frac{d^2 \times \pi}{4}} = \frac{2 \times 1000}{\frac{50^2 \times \pi}{4}} = 1.02\ m/s$$

Q [l/s]
d [mm]
v [m/s]

Thus the calculated Reynolds number is:

$$Re = \frac{v \times d \times \rho}{\eta} = \frac{1.02 \times 50 \times 1.04}{0.00121} = 43634$$

The Reynolds number exceeds the critical Reynolds number Re_{krit}=2320. The flow is turbulent.

NPSH value [m]

NPSH is the abbreviation for **N**et **P**ositive **S**uction **H**ead

Besides the flow rate Q and the pump head H, the NPSH value is one of the most important characteristic parameter of a centrifugal pump.

NPSH value of the pump

The NPSH value of the pump depends on the design and speed of the pump. The higher the speed of the pump, the higher the NPSH value will be.
The NPSH value is measured on a pump test stand and cannot be modified without supplementary means.

NPSH value of the plant

The NPSH value of the plant depends on the loss of head including the losses in fittings and apparatus in the line of the plant, and should be always checked by **calculation**.

p_E = pressure at the inlet cross section of the plant [bar]

p_A = pressure at the outlet cross section of the plant [bar]

p_D = vapour pressure of the fluid at the middle of the suction connection of the pump [bar]

p_b = air pressure at the installation site of the pump [bar]

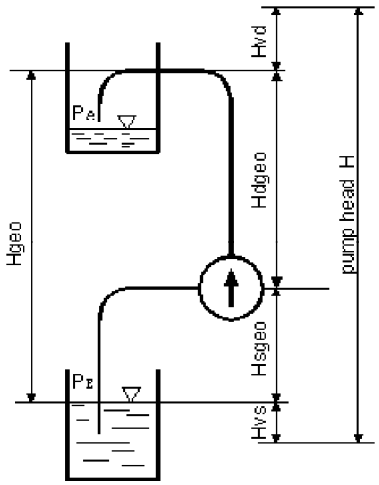
H_{VS} = loss of head of the suction line, from the inlet cross section of the plant to the inlet cross section of the pump [m]

H_{sgeo} = geodetic suction height (negative, in case of flooded suction) [m]

ρ = density of the fluid [kg/m≈]

v_E = inlet flow velocity [m/s]

$$NPSH = \frac{p_E + p_b - p_D}{\rho \times g} + \frac{v_E^{\leq}}{2g} + H_{sgeo} - H_{VS}$$



In order to ensure a correct operation of the pump the following condition must be given:

$$\mathbf{NPSH_{plant} > NPSH_{pump}}$$

Boiling fluids with a velocity up to 0,3 m/s are a special case.

In this case: $p_E = p_D$; as $\frac{v_E^2}{2g}$ and H_{VS} become negligible resulting in:

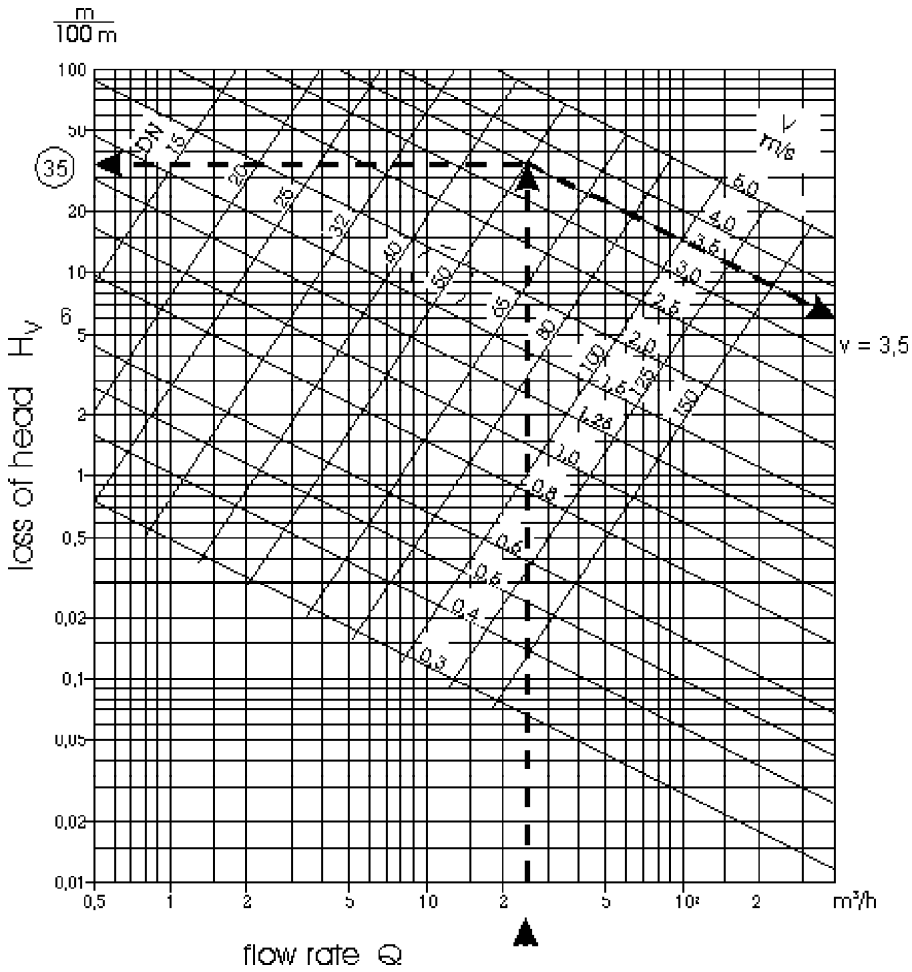
$$\mathbf{NPSH_{plant} = H_{sgeo}}$$

Already during design of the plant and piping layout in front of and behind the pump, losses can be limited when considering:

- the pipe diameter is sufficiently dimensioned,
- less fittings are used,
- fittings with low friction loss are selected,
- short pipe runs are planned.

Loss of head in straight pipe runs

The diagram shows the loss of head for straight pipe runs as a function of a pipe length of 100 m and a given flow velocity v depending on the flow rate and the pipe diameter.



Example:

Flow rate	Q = 25 m≈/h
Pipe diameter	d = 50 mm
From the diagram results:	
Flow velocity	v = 3.5 m/s
Loss of head	**H_v = 35 m/100 m**

Loss of head caused by fittings

The loss of head in fittings can be determined almost exactly when using adequate pipe lengths.

The loss of head in a fitting is considered equal to a straight pipe with corresponding length.

This calculation is valid only for water and fluids similar to water.

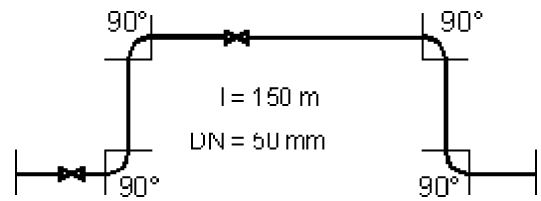
With the same diameter of pipes and fittings we can simplify the calculation.

Equivalent pipe lengths in meter for fittings
(valid for Re ⊕ 100,000 and roughness k ∪ 0.04 mm)

Diameter DN [mm]	25	40	50	65	80	100	125	150
Standard valve	2.1	4.6	7.5	11.0	14.0	20.0	26.0	37.0
Free-flow valve	0.7	0.9	1.2	1.5	1.8	2.2	2.9	3.7
90° valve	3.2	5.0	7.2	10.0	13.0	17.0	23.0	31.0
Elbow 90° R = 4 x d	0.5	0.6	0.7	0.9	1.2	1.5	1.9	2.3
Elbow 90° R = 3 x d	0.7	0.9	1.1	1.4	1.7	2.2	2.8	3.5
Tee (diffusion)	1.6	2.0	2.5	3.3	4.0	4.8	5.7	6.8
Tee (junction)	2.0	2.6	3.2	4.4	5.8	7.5	10.0	13.0

Example:

Flow rate	Q =	25 m≈/h
Straight pipe length	l =	150 m
Diameter	DN =	50 mm
Elbow 90°		4 pieces
Free-flow valves		2 pieces



from diagram (page 16): $v = 3.5$ m/s

$H_V = 35$ m/100 m pipe length

from table :

equivalent pipe length 4 elbows:	$l_{bend} = 1.1 \infty 4 = 4.4$m
equivalent pipe length 2 free-flow valves:	$l_{slide} = 1.2 \infty 2 = 2.4$ m
straight pipe length:	$l_{pipe} = 150.0$ m
total pipe length	$l_{total} = 156.8$ m

loss of head:

$$H_{total} = H_V \times l_{total}$$

$$H_{total} = \frac{35}{100} \times 156.8 = 52 \text{ m}$$

with laminar flow (high viscosities) the loss of head Δp_V can be calculated using the Hagen-Poiseuille formula:

$$\Delta p_V = \frac{v_M \times 32 \times \eta \times l}{d^2 \times 10^5}$$

v_M=Q/A [m/s]

η [Pa s = kg/m s]

l [m]

d [m]

Δp_V [bar]

$H_V \cup 10 \infty \Delta p_V$

Once the required total head has been calculated, the pump type can be selected and the required pump can be sized by means of the ***Fristam*** pump curves.

Pump type selection

The viscosity of the fluid is an important parameter for the pump selection and leads us to the right decision.

Fristam centrifugal pumps are equipped with open impellers which are suitable for the transfer of liquids with viscosities up to 1000 mPa s.

Centrifugal pumps have the following features:

- pulsation free transfer without alteration of flow rate and total head.
- high reliability in operation due to low number of moving parts.
- high operating speed, directly coupled to high-speed electric motors.
- small dimensions and therefore low space requirement.
- low operating costs.
- excellent performance control by speed adjustment.

Fristam positive displacement pumps

Usually it is recommended to use positive displacement pumps for low flow rates and high pressures. Even though the viscosity of the fluid would not require its use, because a centrifugal pump would work under these conditions with a very low efficiency rate.

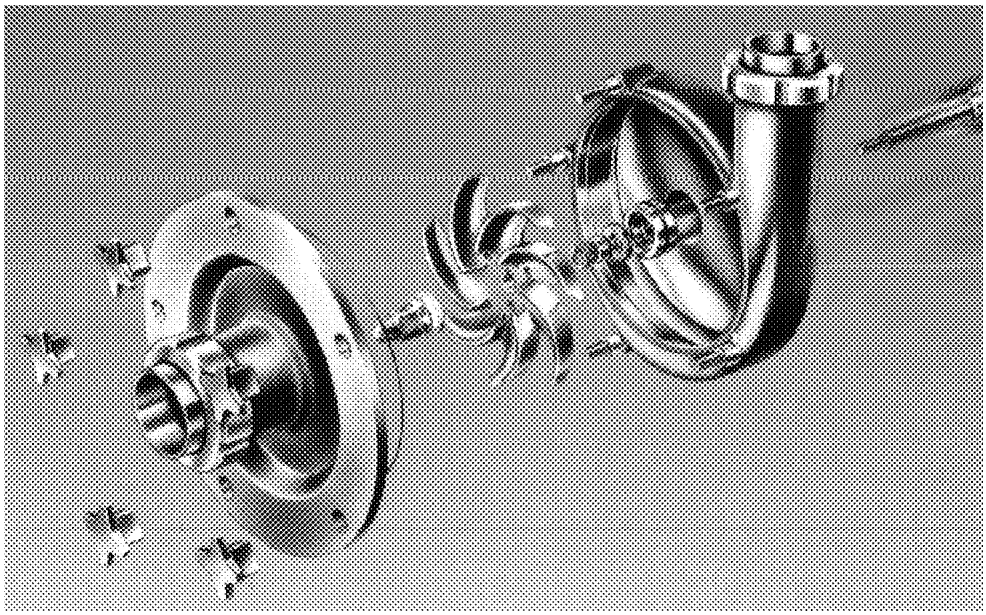
Note:

In the following chapters the various pump types, centrifugal and positive displacement are described with tips for the correct selection of the pump size. Each general pump type description is followed by a section dealing with the correct use of the pump curves provided.

Features of the centrifugal pump

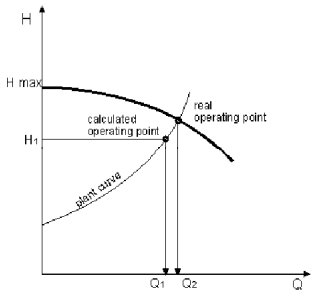
Centrifugal pumps are fluid-kinetic machines designed for power increase within a rotating impeller. Therefore it is also called the hydrodynamic pumping principle.

According to this principle, the fluid is accelerated through the impeller. In the outlet connection of the centrifugal pump, the resulting increase in speed is converted into delivery head.



Q/H curve

In centrifugal pumps the delivery head H depends on the flow rate Q. This relationship, also called pump performance, is illustrated by curves.



During a bench test, the pump is operated at constant speed and the values Q and H are determined for the various operating points. In order to allow a comparison between the various pump types these measurements are carried out using only water as liquid. With these operating points a Q/H curve be drawn connecting the points on the graph.

Once the flow rate Q is defined and the delivery head H is calculated, the operating point of the plant can be determined. Usually the operating point is not on the Q/H curve of the pump. Depending on the required delivery head, the centrifugal pump will find its operating point when the plant curve and pump curve meet. The flow rate rises from Q_1 to Q_2.

The required operating point is obtained by adapting the pump to the specified operating conditions.

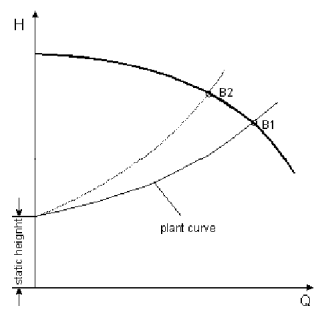
This can be done by the following actions:

- throttling the flow
- correcting the diameter of the impeller
- Adjusting the speed of the drive

Throttling the flow

Partially closing a throttle valve or mounting an orifice plate into the discharge pipe of the pump will increase the pressure drop. The plant curve is shifted.

The operating point B1 (intersection point between pump curve and plant curve) moves on the pump curve to B2.



Note: throttling reduces the overall efficiency.

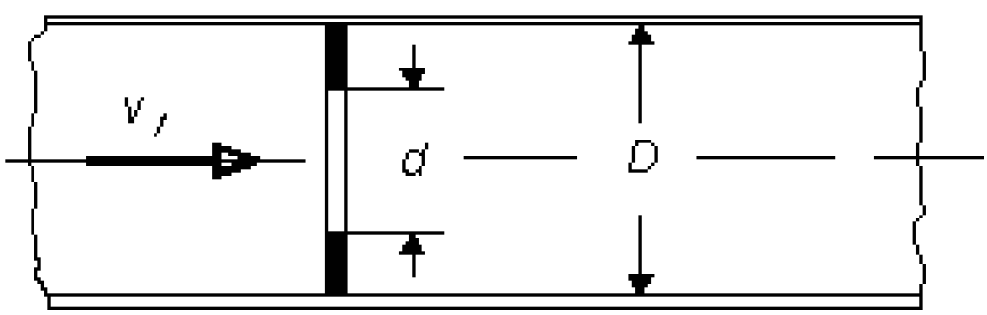
A throttle control or a mounted orifice plate is the less expensive control regarding the investment expenses. In case of significant power requirement, an economic appraisal is highly recommended.

Orifice plate calculation

The friction loss in an orifice plate can be calculated easily:

$$\Delta p_v = \zeta \times \frac{\rho}{2} v_1^2 \times 10^{-5}$$

ρ [kg/m≈]
v_1 [m/s]
Δp_v [bar]



See the values ζ stated in the table below.

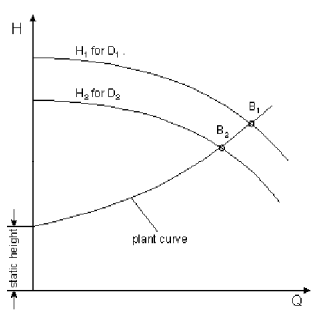
Aperture ratio $m = (d/D)^2$	Resistance value ζ
0.05	800
0.1	250
0.2	50
0.3	20
0.4	4

Calculation:

- take the figure stated in the table for d, see table ζ ,calculate Δp_V .
- if Δp_V varies from the required value, take new value for d and calculate once more Δp_V .

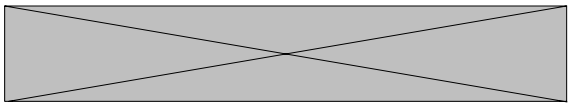
Correction of the impeller diameter

A correction of the impeller diameter is to be favoured when a permanent reduction of flow rate or differential head is required. The performance of the pump is adjusted towards the duty point by reducing the impeller diameter.



The operating point is shifted from B1 to B2. This is the point where the new pump curve meets the plant curve.

The required impeller diameter can be determined easily using following formulae:

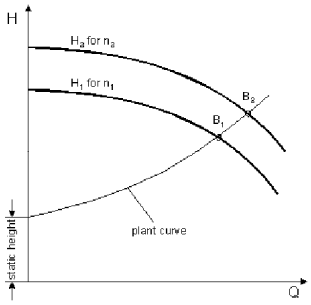


N = power consumtion
D = impeller diameter
Q = flow rate
H = total head

Note: the efficiency of the pump decreases with increasing correction.

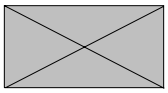
Pump speed control

A great number of various operating points can be set continuously, when modifying the pump speed using a variable speed drive or frequency inverter. The operating point moves on the pump curve from B to B2.

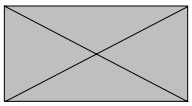


Considering the overall efficiency, this is the best way of flow control. Using a variable-speed drive or a frequency inverter additional costs can arise and should be evaluated in an economic appraisal.

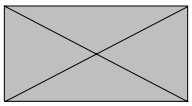
The flow rate changes linearly to the speed.



The total head changes with the square of speed.

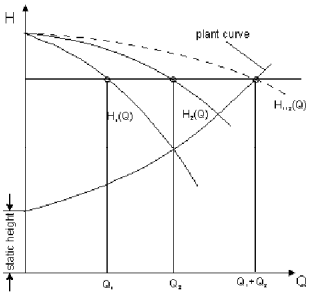


The power consumption changes with the third power of the speed.



Parallely connected pumps

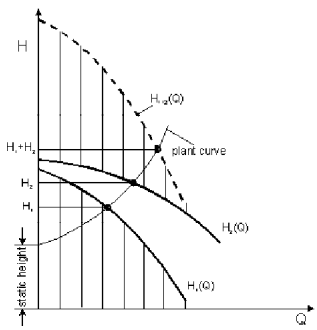
In the case of pumps connected in parallel the fluid flows are added with corresponding delivery head. This applies to pumps even with different Q/H curve.



Pumps connected in series

A multistage centrifugal pump performs as single stage pumps connected in series.

Note:
A stationary pump in a system creates a considerable pressure drop. Therefore it is recommendable to install a by pass around pumps which are connected in series.



The overall performance curve of centrifugal pumps connected in series can be calculated by adding the differential head of each pump at the relevant flow rate .

Cavitation

Cavitation can be recognised by a strongly increased noise level of the pump with a simultaneous reduced flow rate.

What causes cavitation in centrifugal pumps ?

The lowest pressure point in a pump occurs at the inlet of the pump impeller. Due to local pressure reduction part of the fluid may evaporate generating small vapour bubbles. These bubbles are carried along by the fluid and implode instantly when they get into areas of higher pressure. These implosions can create local pressure peaks up to 100.000 bar.

If a pump is cavitating over longer periods, the impeller, the pump housing and cover will wear out. The surface is typically perforated and pitted.

How to avoid cavitation?

Vapour pressure

We should ensure that at all points of the pump, the fluid pressure is higher than the vapour pressure at the corresponding temperature. Take the pressure stated in the vapour-pressure-table of the product to be transfered.

The NPSH value of the plant must be at least 0.5 m higher than the NPSH value of the pump.

For a safe and cavitation free operation the following formular is valid:

$NPSH_{plant} > NPSH_{Pump} + 0.5$ m

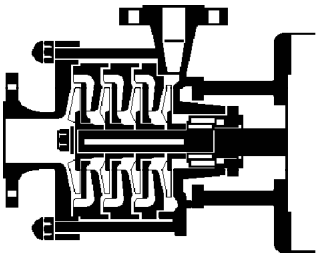
The vapour pressure of the product is dependent on the temperature and will rise with increasing temperature.

If the product is pumped at different temperatures the maximum vapour pressure should be used to determine the NPSH value of the plant.

Centrifugal pump types

The ***Fristam*** centrifugal pump range consists of following pump types:

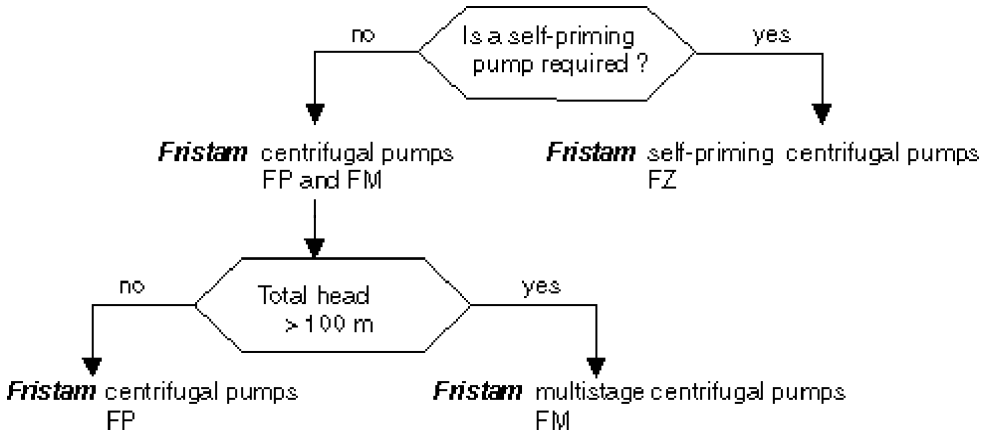
- ***Fristam*** centrifugal pump **FP**
  The design principle of the ***Fristam*** centrifugal pump FP with open impeller and optimised volute guarantees shear sensitive handling of and minimum heat transfer to the product. Viscosities up to 1000 mPa are no problem. The fluid may contain air or gas, may be homogeneous or contain additives. Low NPSH values make it possible to use the pump also under unfavourable conditions. The ***Fristam*** centrifugal pump FP is designed as a pump for flooded suction and fully suitable for CIP and SIP application.



- ***Fristam*** multistage centrifugal pump **FM**
  The centrifugal pump FM is designed as a multistage pump especially developed for high delivery heads. The centrifugal pump FM can be used for difficult pressure conditions such as feed pump for filters, heat exchangers and fillers, as well as for recirculation and as booster pump in membrane filtration and reverse osmosis plants.

- ***Fristam*** self-priming centrifugal pump **FZ**
  The centrifugal pump FZ works on the water ring-side channel principle. Impellers with radial blades transfer the pressure energy to the liquid. Close clearances make it possible to obtain an excellent suction performance. Thus it is possible to pump gaseous products and to deaerate the suction line. This ensures also an optimum drain of the plant..

Selection criteria



The selection between the pump types FP and FM also depends on the required flow rate.

Centrifugal pump FP

Selecting the correct size

Example:

Flow rate Q_A = 90 m≈/h

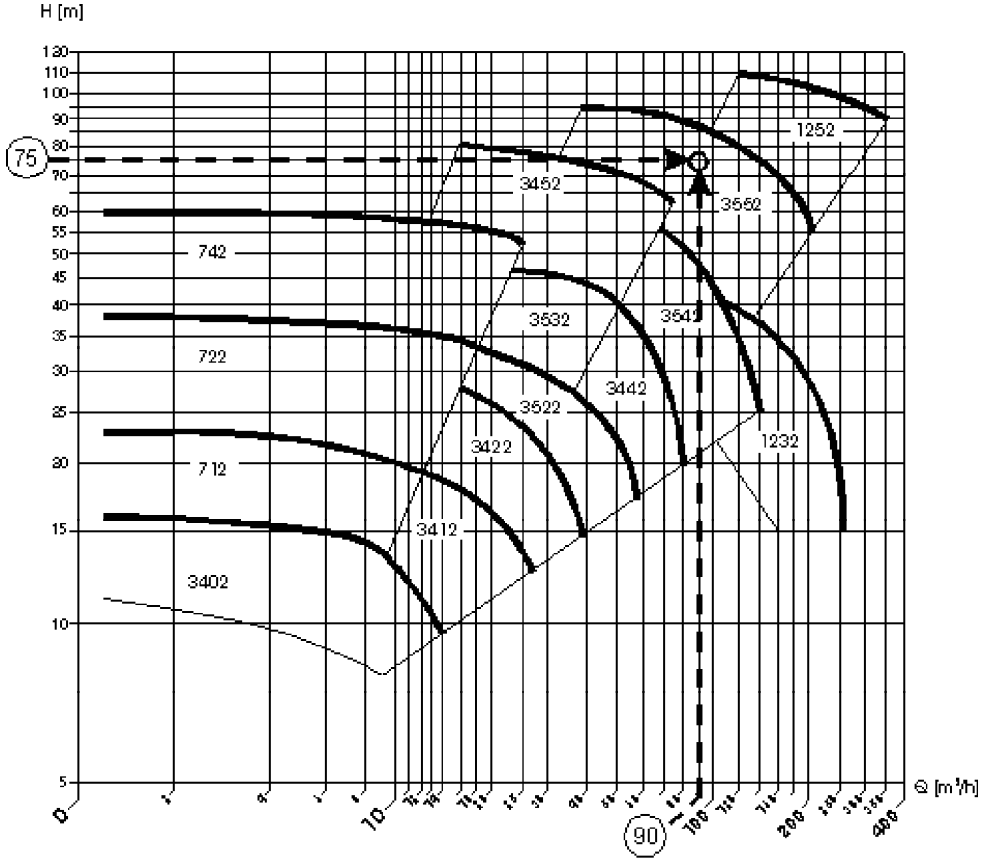
Total head H_A = 75 m

Step 1:

Select the pump size.

FP sizes

Size selection



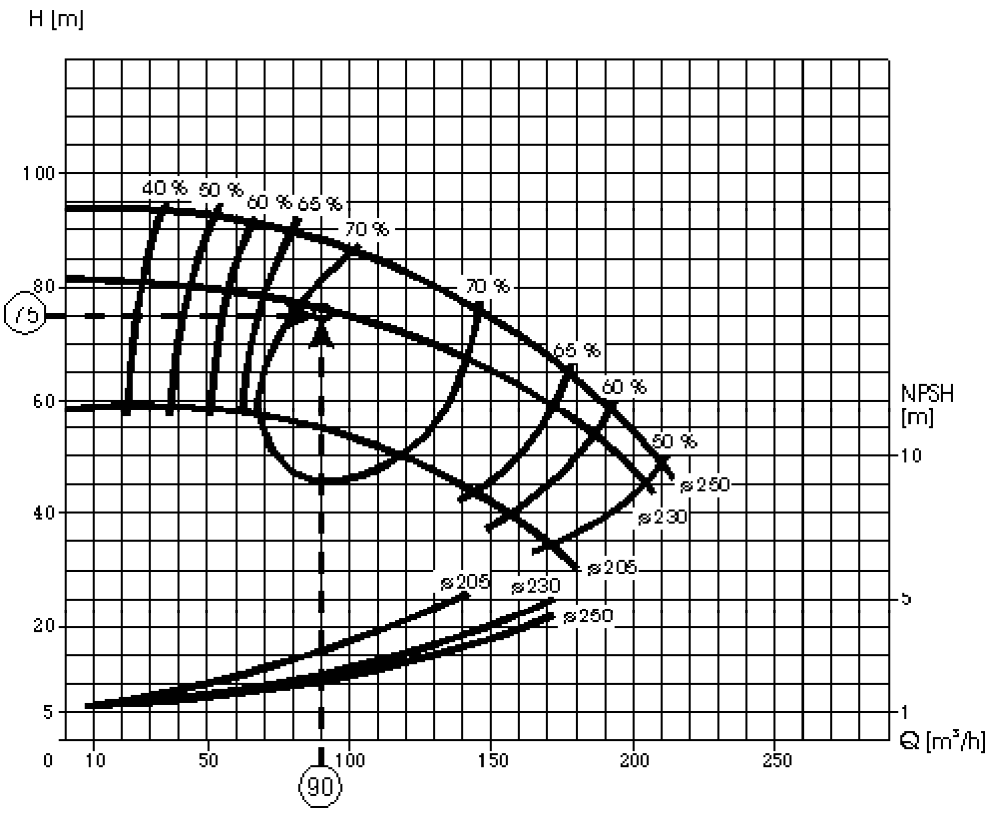
Selected pump size: **FP 3552**

Centrifugal pump FP performance diagram

Step 2:
Enter the operating point of your plant into the pump diagram.

If the duty point is not exactly on the pump curve, the performance of the pump can be adjusted by throttling the flow, reducing the impeller diameter or adjusting the output speed of the drive. (see page 21–23)

FP 3552

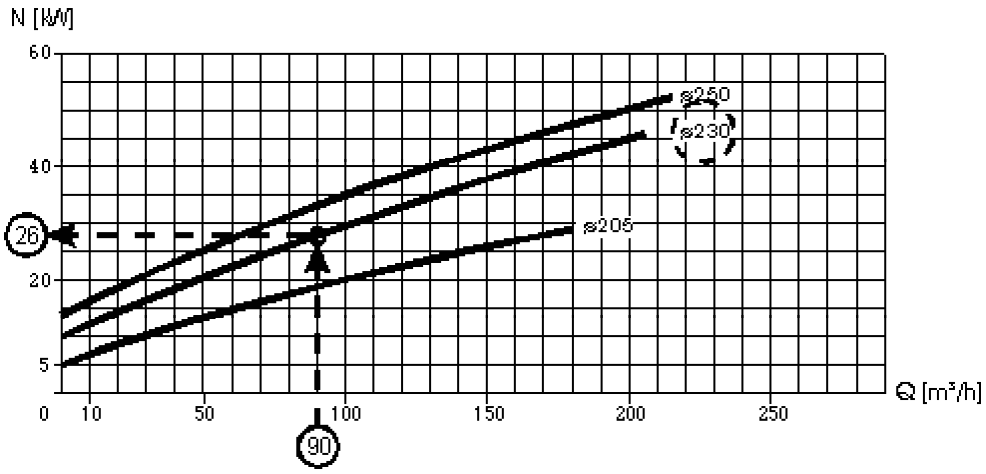


Impeller diameter resulting from the diagram = 230 mm

Power consumption of the pump

Step 3:
Find the power consumption of the pump at the point in the diagram where the power curve of the impeller used meets the design flow rate.

Select the motor with the next higher power rating.



Power consumption according to the diagram: N = 26 kW
selected motor: 30.0 kW

pump efficiency

Step 4:
Check the efficiency

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

Q [m≈/h]
H [m]
N [kW]
ρ [kg/dm≈]

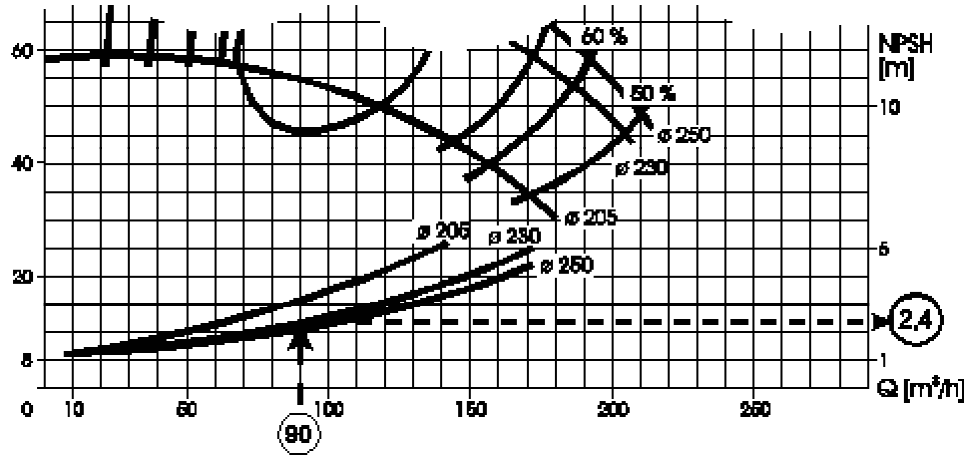
$$\eta = \frac{90 \times 75 \times 1}{367 \times 26}$$

ρ_{water} = 1 kg/dm≈

$$\eta = 0.7 \Rightarrow 70\%$$

Step 5: **Check NPSH value**

Check if $NPSH_{plant} > NPSH_{Pump}$



Resulting NPSH value of the pump from the diagram = 2.4 m

Self- priming centrifugal pump FZ

Selecting the correct size

Example:

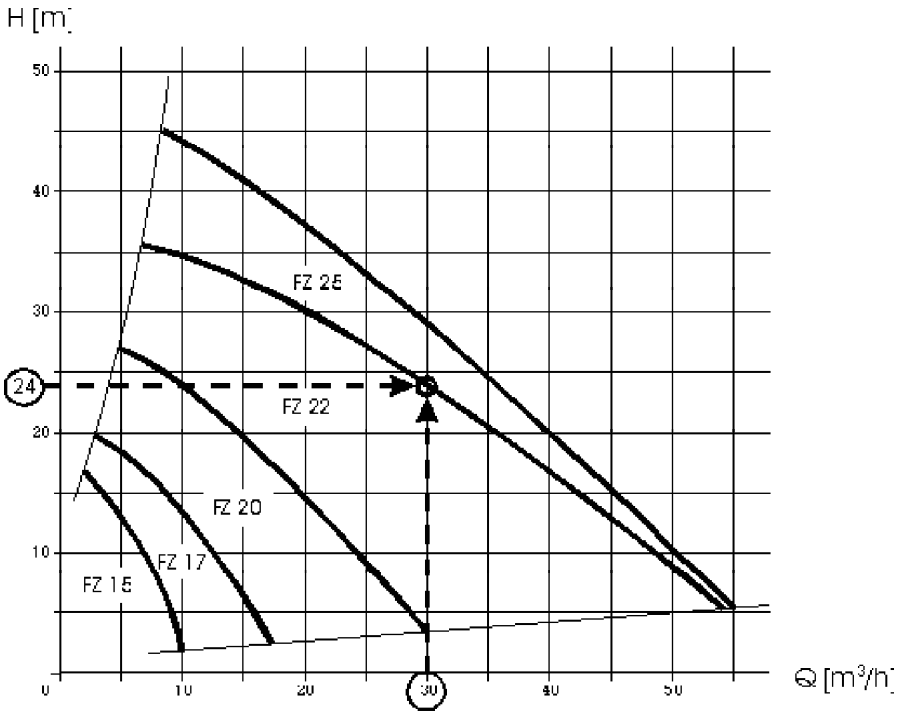
Flow rate	Q_A= 30 m≈/h
Total head	H_A= 24 m

Size selection

Step 1:

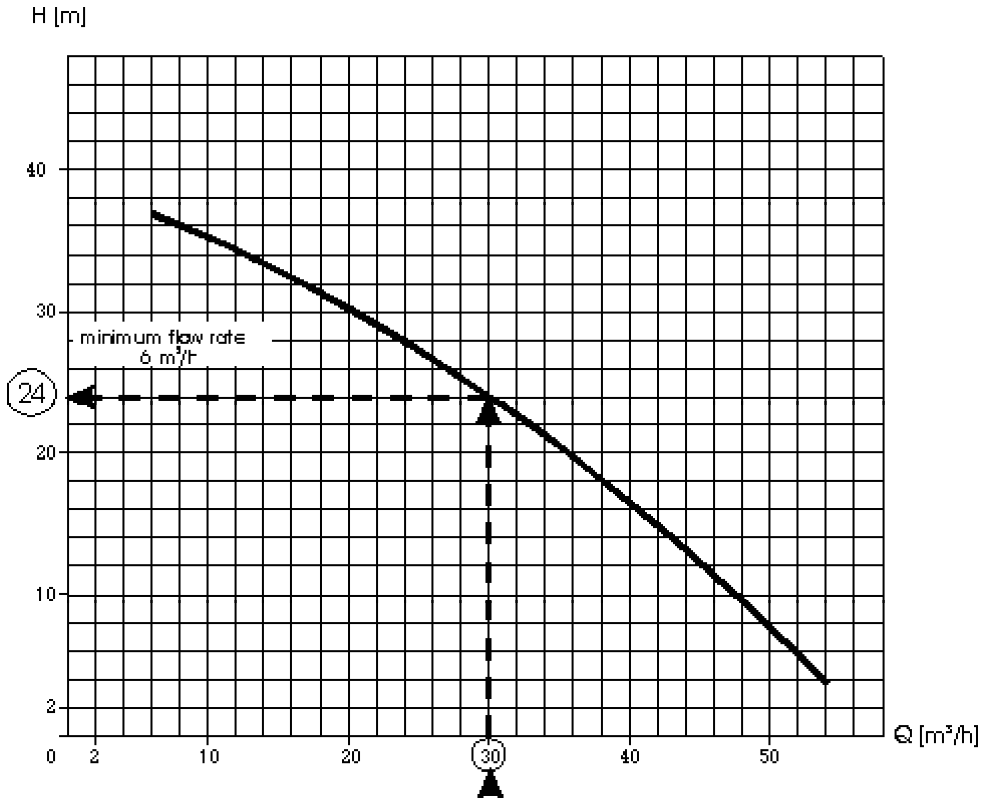
select the pump size whose curve is above to the operating point of the plant.

FZ sizes



Selected pump size: **FZ 22**

Power consumption of the pump

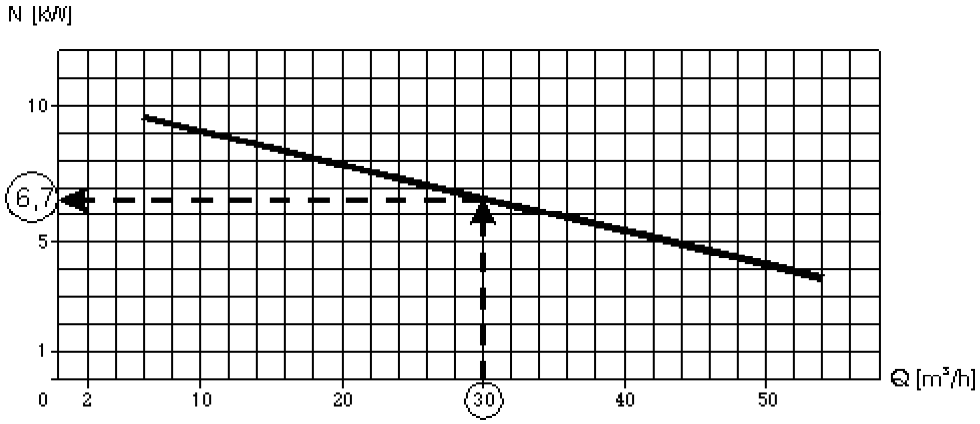


Note:

The performance of FZ pumps can be adjusted to the required operating point only by throttling the flow (see page 21/22) or variation of the speed (see page 23). It is **not possible** to modify the impeller diameter.

Step 2:

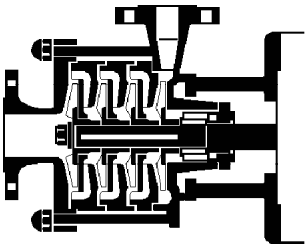
Find the power consumption of the pump at the point in the diagramm where the power curve meets the design flow rate. Select the motor with the next higher power rating.



From the diagram: N = 6.7 kW, selected motor: 7.5 kW

Multistage centrifugal pump FM

The selection is carried out the same way as single-stage centrifugal pumps FP are selected (See page 28).



Features and operating principle of a positive displacement pump

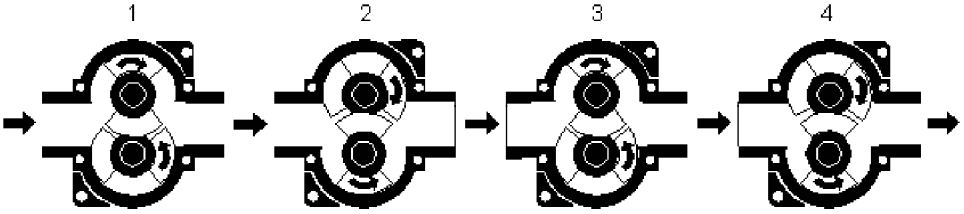
Positive displacement pumps are hydrostatic machines. They operate with a positive transfer and should not work against a closed system.

All rotary pumps are designed after the same principle. Two rotors are arranged on parallel shafts and driven by an external synchronous gear box.

The rotors rotate in opposite directions to each other. Small radial and axial clearences assure that they have no contact with each other, or the pump body. The rotors are designed to form a barrier between the suction and pressure side of the pump in any position. The sealing is only maintained by narrow gaps. There are no additional seals or valves.

The increasing cavity between the rotors on the suction side is filled with the product. The product is displaced in a circumferential direction and discharged on the pressure side as the cavity between the rotors is collapsing. This generates a constant flow from the suction to the discharge side of the pump.

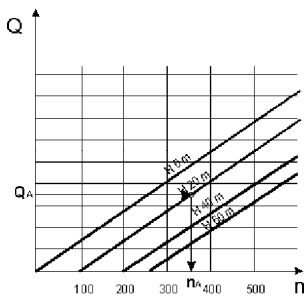
Operating scheme



1	2	3	4
Suction and filling of the upper chamber and simultaneous displacement of the product present in the lower chamber	Displacement of the product present in the upper chamber and simultaneous suction and filling of the lower chamber	Discharge of the product present in the upper chamber and simultaneous displacement of the product present in the lower chamber	Displacement of the product present in the upper chamber and simultaneous discharge of the product present in the lower chamber

Rotary pumps ensure a gentle fluid transfer with minimum stress or damage to the product.

Performance curve
(only for water)



With positive displacement pumps the flow rate Q is linear dependent on the pump speed n.

On a test stand the flow rate is determined for various speeds and total head. In order to allow a comparison between the various pump designs and types, these tests are always carried out with water.

Once the flow rate Q and the total head H have been determined, a pump speed n that corresponds to this operating point will result from the diagram.

The positive displacement pump is usually operated with a fixed speed drive. The flow rate is constant.

Pump speed

The flow rate can be adjusted to the various operating conditions by changing the pump speed.

Viscosity

The viscosity of the product must be always taken into consideration for the design and selection of the pump type.

Fluids with higher viscosities require more time to enter the displacement chamber. In those cases the pump speed must be adjusted accordingly to avoid cavitation which reduces the volumetric efficiency and increases the wear. A pump operating with cavitation creates a considerable noise level.

Clearance losses

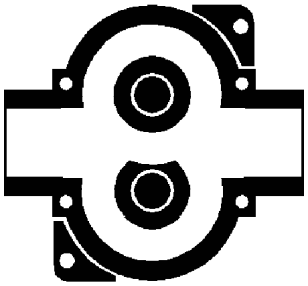
Regardless of the low clearance between the rotor and the pump body, a slip from the pressure side back to the suction side will be generated when waterlike products are transfered.

In case of circumferential piston pumps the slip stops at a product viscosity of about 200 mPa s and at about 500 mPa s in the case of rotary lobe pumps.

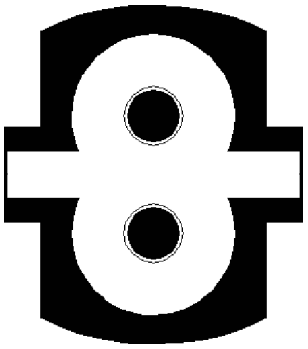
Designs

Fristam supplies two different positive diplacement pump designs depending on the application.

- ***Fristam*** circumferential piston pumps **FK** and **FKL**
  The circumferential piston pumps type FK and FKL have a very narrow clearance in the pump chamber and a gland sealing allover. Due to these design features circumferential piston pumps have an outstanding suction performance and are suitable for high differential heads.



- ***Fristam*** rotary lobe pumps **FL**
  Due to the gland/line sealing, rotary lobe pumps type FL are mainly used for flooded suction conditions. They reach slightly lower differential heads than the circumferential piston pumps especially at low viscous products, but can run at higher speeds.



Design versions

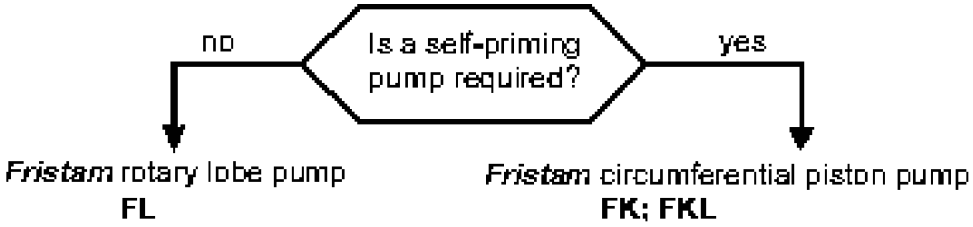
Circumferential piston pumps and rotary lobe pumps can be used for hot products
- up to approx. 90 °C using **rotors with standard dimensions**
- up to approx. 150 °C using **rotors with high temperature dimensions.**

They are suitable for automatic cleaning (CIP process) and sterilisation (SIP process).

The pumps can be supplied with horizontal or vertical ports. Various types of connections such as flanges, clamps or different threads are available.

Selection of design

The design selection depends amongst other:

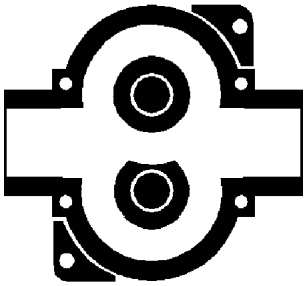


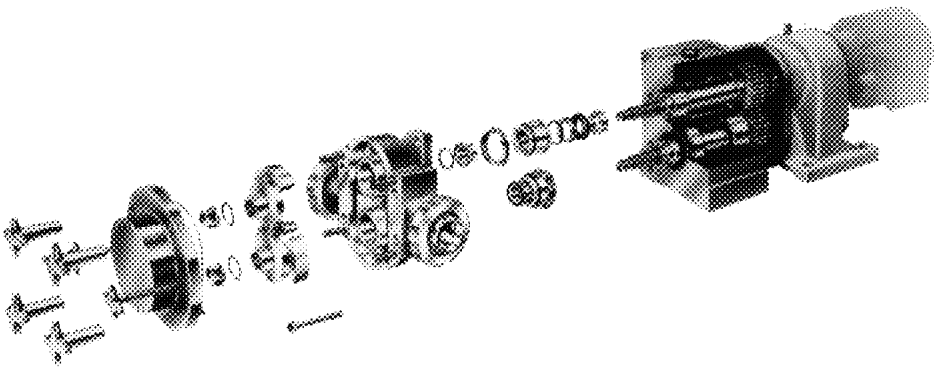
An additional selection criteria is the difference in pressure performance of the various types.

- ***Fristam*** rotary lobe pumps **FL**, maximum total head 120 m (12 bar)
- ***Fristam*** circumferential piston pump **FK**, maximum total head 200 m (20 bar)
- ***Fristam*** circumferential piston pump **FKL**, maximum total head 250 m (25 bar)

Circumferential piston pumps FK, FKL

The Fristam circumferential piston pumps are manufactured with very close clearances. Thus they can generate a small vacuum in the suction pipeline. Due to the atmospheric pressure or system pressure the product is forced into the pump chambers.



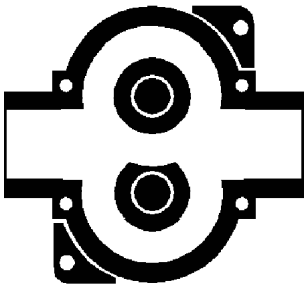


Example:

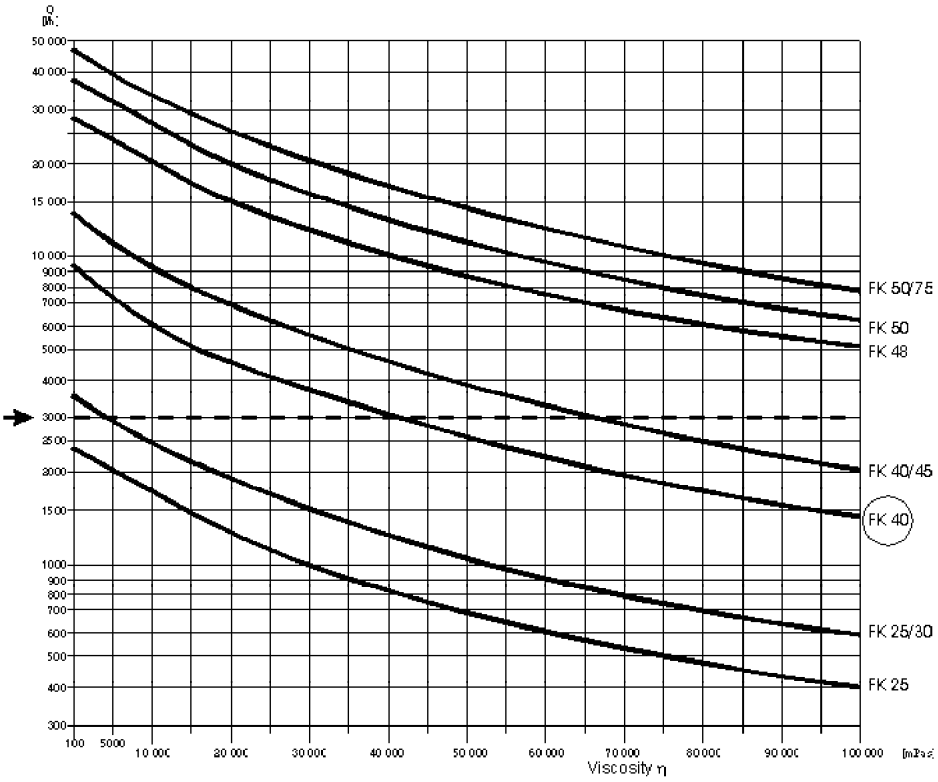
Type selection

Flow rate Q = 3000 l/h
Total head H = 120 m

Pump to be used for products with different viscosities.

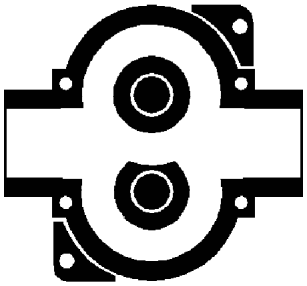


FK pump basic selection diagram



for case 1: water
case 2: 10 mPa s | selected: **FK 40**
case 3: 10,000 mPa s ≠

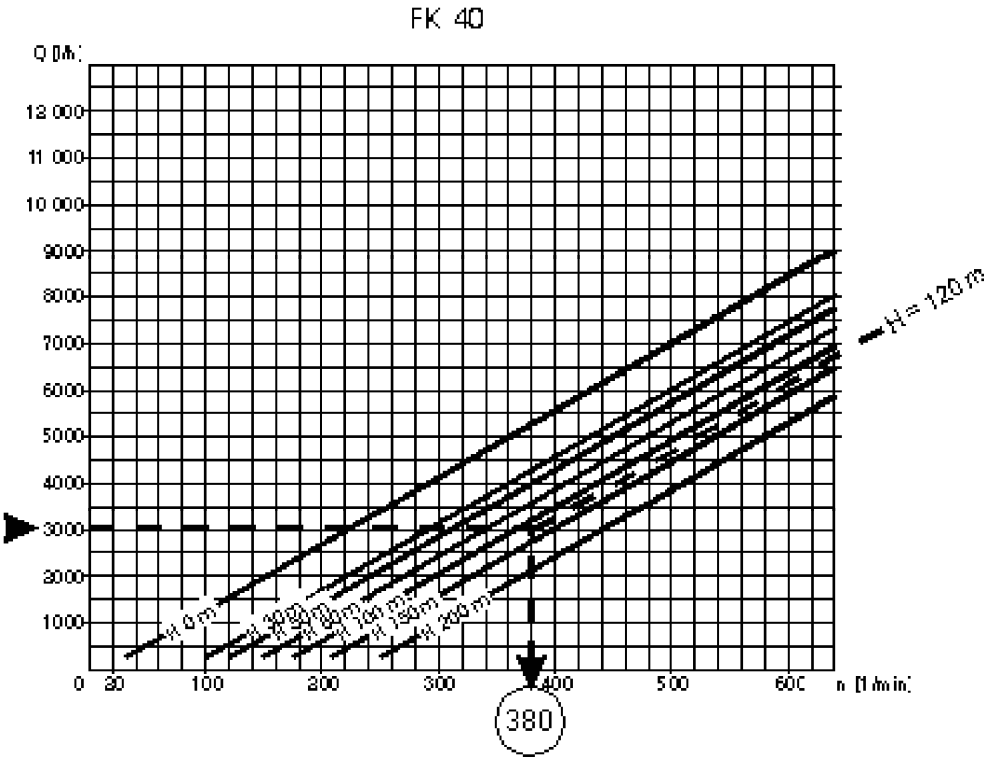
Case 1: viscosity η ∪ 1 mPa s



Example:

Q = 3000 l/h
H = 120 m ∪ p = 12 bar
η = 1 mPa s

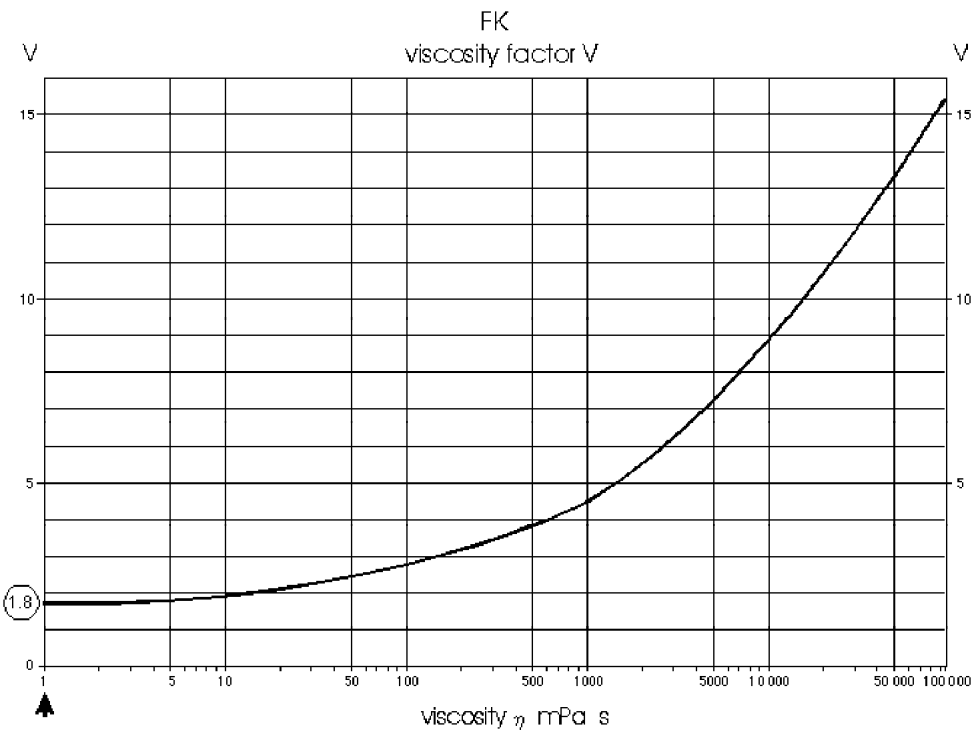
Step 1:
read speed n [1/min]



resulting from the diagram: speed n = 380 1/min

Step 2:
define viscosity factor

Case 1: viscosity
$\eta \cup$ 1 mPa s

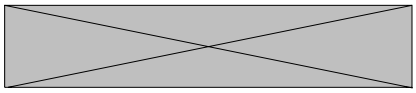


Viscosity factor V = 1.8

Case 1: viscosity $\eta \cup$ 1 mPa s

Step 3:

Calculate the power N [kW] required for the pump drive.



p = pressure [bar]

V = viscosity factor

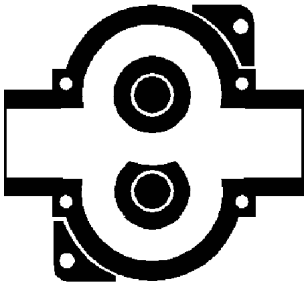
n = speed [1/min], stated in the diagram

C = flow rate/revolution [l/rev.]

RK	25	25/30	40	40/45	48	50	50/75
C	0,07	0,11	0,26	0,36	0,77	1,1	1,37

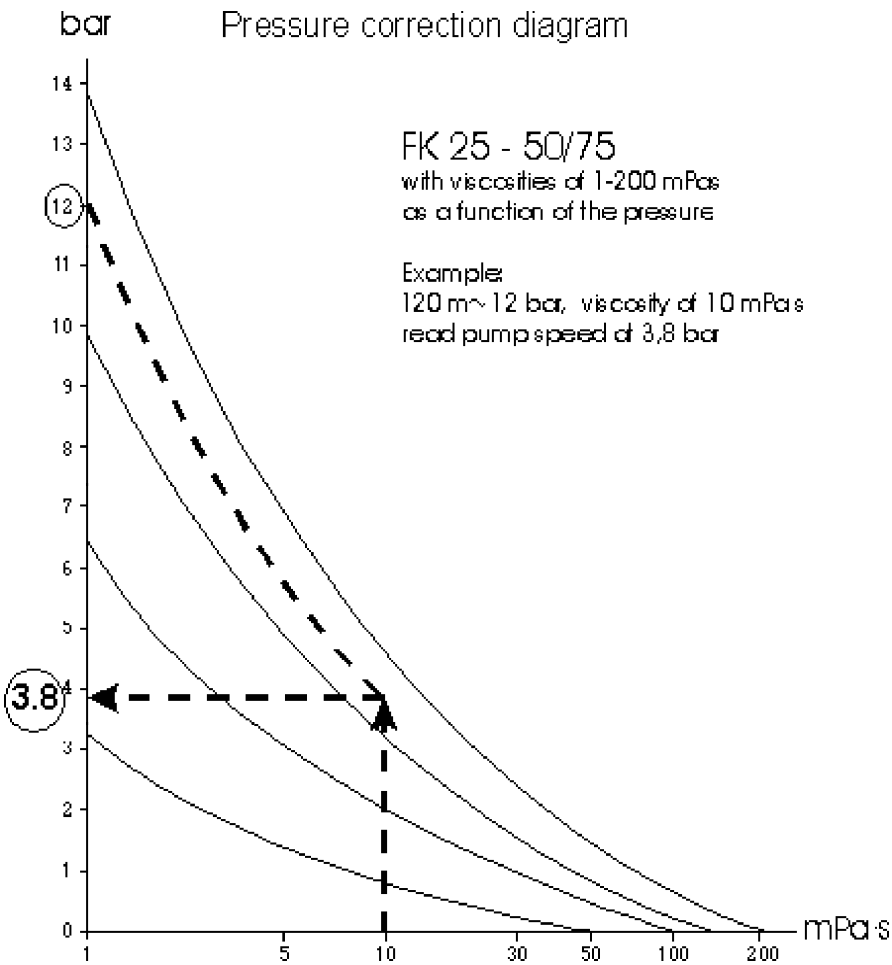
Example: $$N = \frac{(2 \times 12 + 1.8) \times 360 \times 0.26}{1000} = 2.5\ \text{kW}$$

Case 2:
viscous product
η up to 200 mPa s



Example:
Q = 3000 l/h
H = 120 m ∪ p = 12 bar
η = 10 mPa s

Step 1:
speed correction

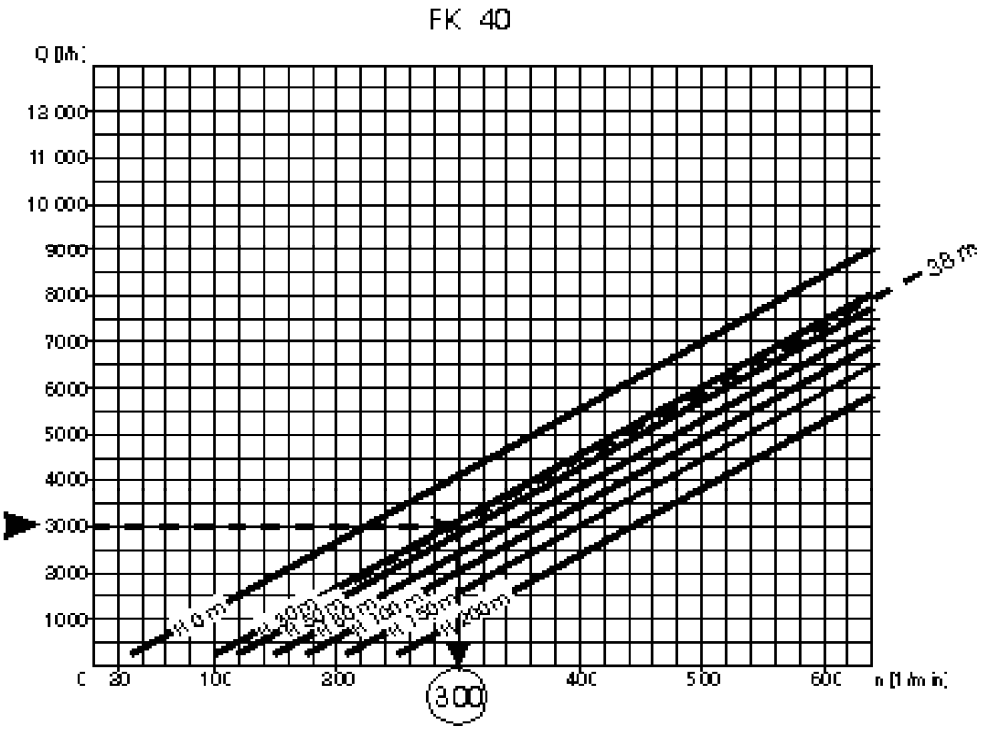


stated in the diagram: p = 3.8 bar.

Define now the speed required for the corrected pressure.

Case 2:
viscous product
η up to 200 mPa s

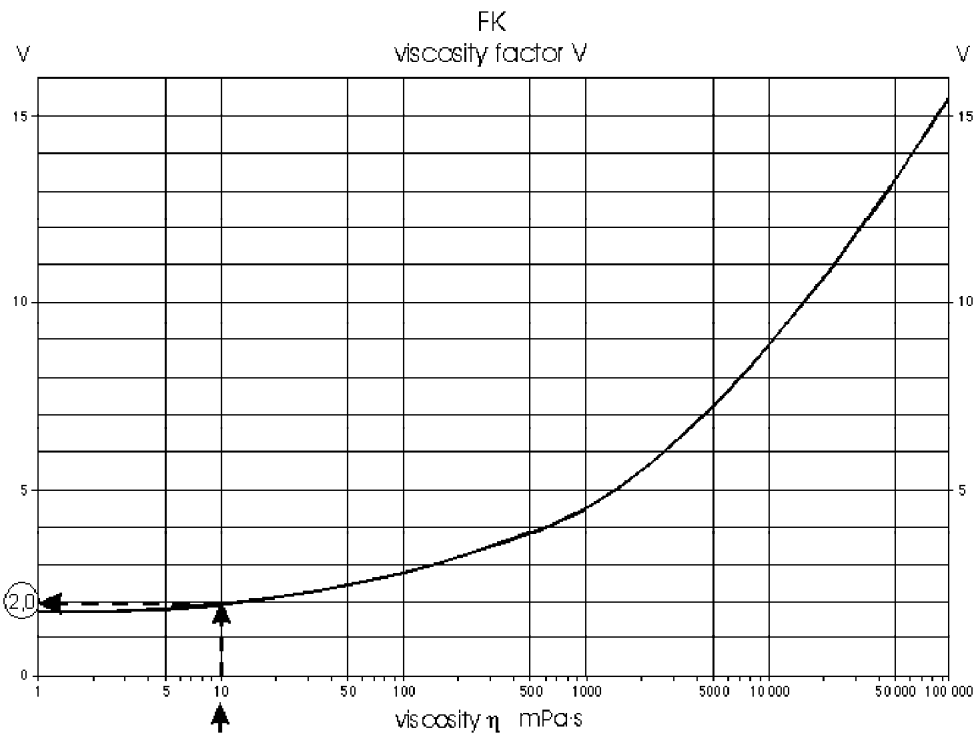
Step 2:
read speed n [1/min]



stated in the diagram: speed n = 300 1/min

Step 3:
define viscosity factor

Case 2:
viscous product
η up to 200 mPa s



stated in the diagram: viscosity factor V = 2.0

Case 2: viscous product η up to 200 mPa s

Step 3:

Calculate power consumption N [kW] to select the pump drive.

$$N = \frac{(2 \times p + V) \times n \times C}{1000}$$

p = pressure in bar ∪ H/10

V = viscosity factor

n = speed with H = 38 m

C = flow rate/revolution [l/rev.]

RK	25	25/30	40	40/45	48	50	50/75
C	0.07	0.11	0.26	0.36	0.77	1.1	1.37

Example :

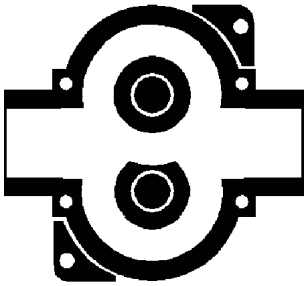
$$N = \frac{(2 \times 12 + 2) \times 300 \times 0.26}{1000} = 2.03 \text{ kW}$$

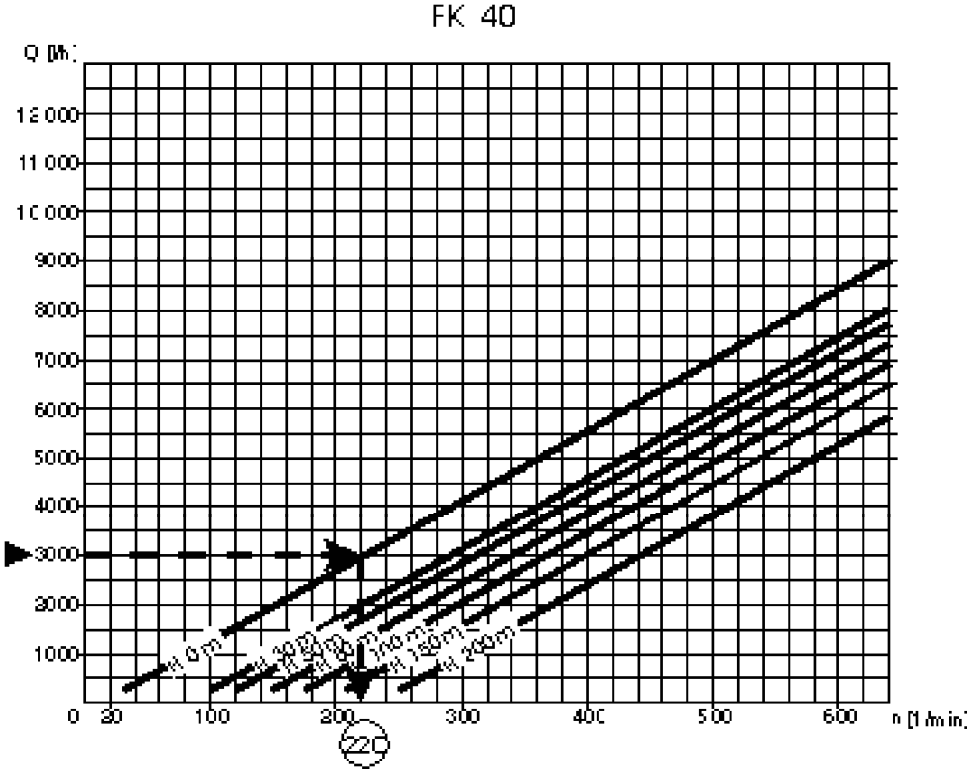
Example:

Q = 3000 l/h
H = 120 m ∪ p = 12 bar
η = 10,000 mPa s

Step 1:
read speed with H = 0, η > 200 mPa s

Case 3:
viscous product
η = 200–100,000 mPa s

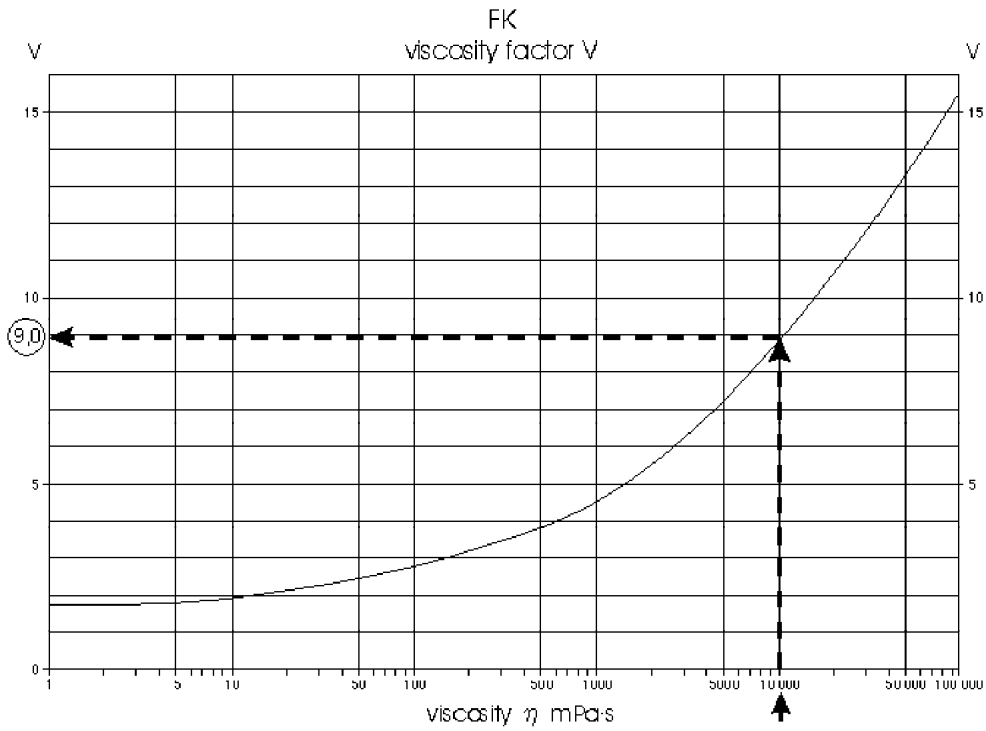




Stated in the diagram: n = 220 1/min

Case 3:
viscous product
η = 200–100,000 mPa s

Step 2:
read viscosity factor V.



stated in the diagram: V = 9.0

Case 3:
viscous product
η = 200–100,000 mPa s

Step 3:
Calculate the absorbed power N [kW] to select the pump drive.

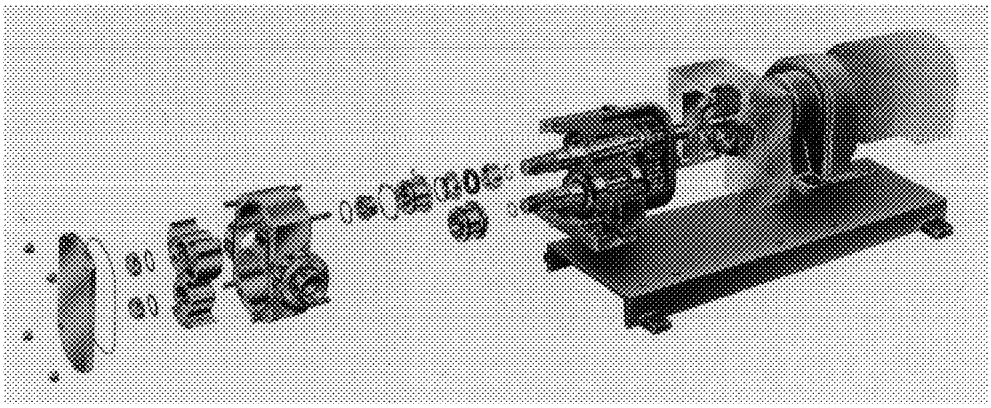
$$N = \frac{(2 \times p + V) \times n \times C}{1000}$$

p = pressure [bar] ∪ H/10
V = viscosity factor
n = speed [1/min], stated in the diagram
C = flow rate/revolution [l/rev.]

Example:

$$N = \frac{(2 \times 12 + 9.0) \times 220 \times 0.26}{1000} = 1.9\ \text{kW}$$

Rotary lobe pump FL

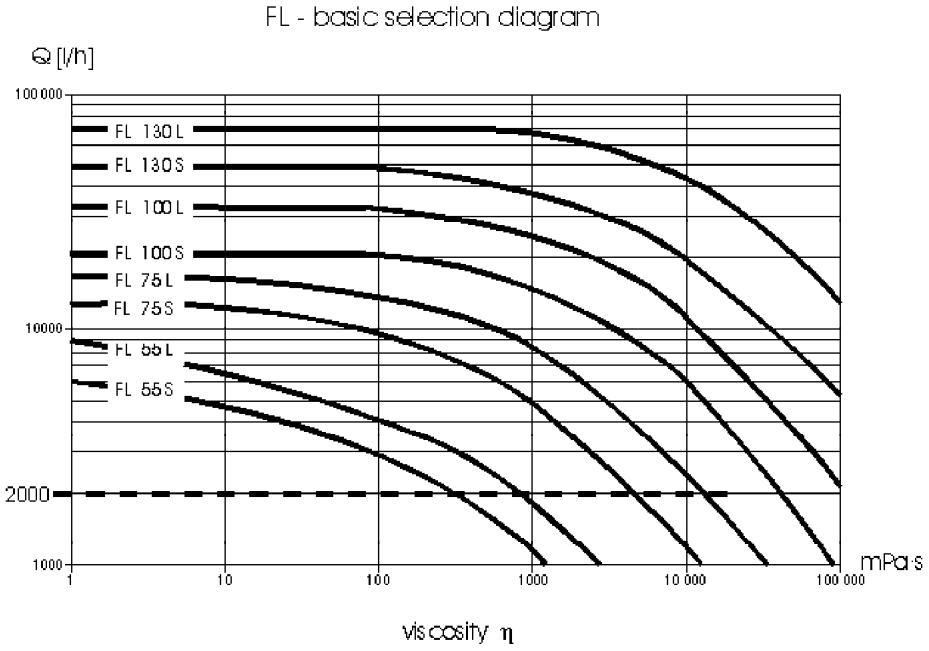


Example:

Flow rate	Q = 2000 l/h
Total head	H = 60 m

Selection

Pump to be used for products with different viscosities.

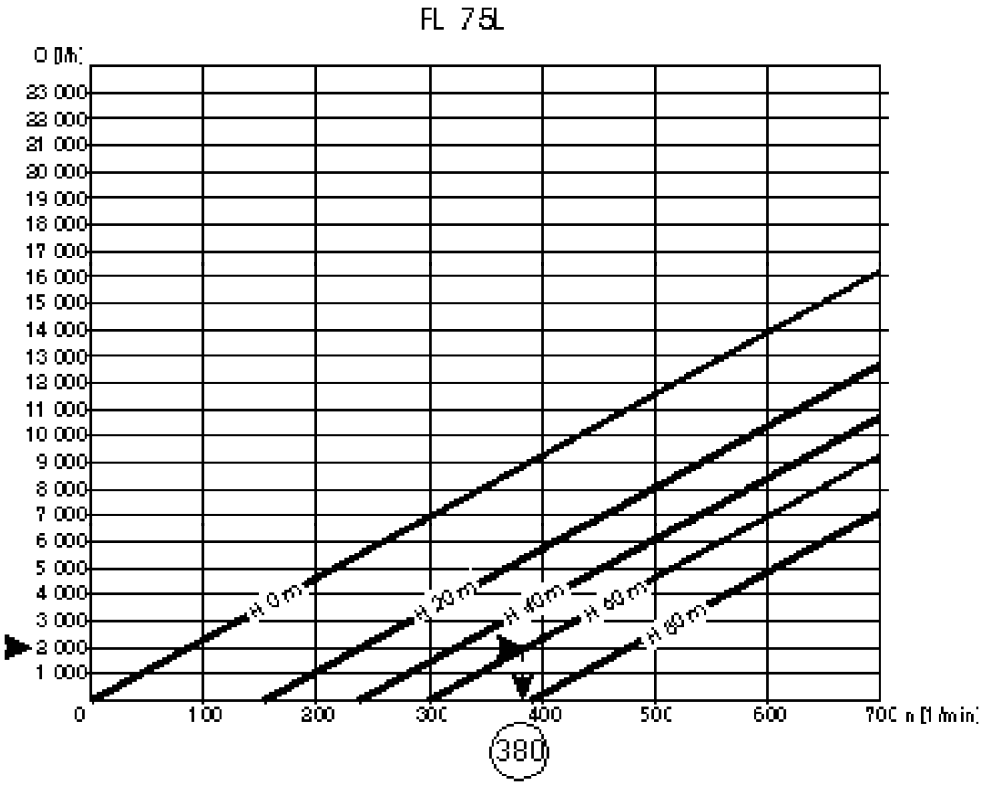


for case number 1: water
case number 2: 10 mPa s | selected: **FL 75 L**
case number 3: 10,000 mPa s ≠

Flow rate	Q = 2000 l/h
Total head	H = 60 m
	η = 1 mPa s (water).

Case 1: viscosity η ∪ 1 mPa s

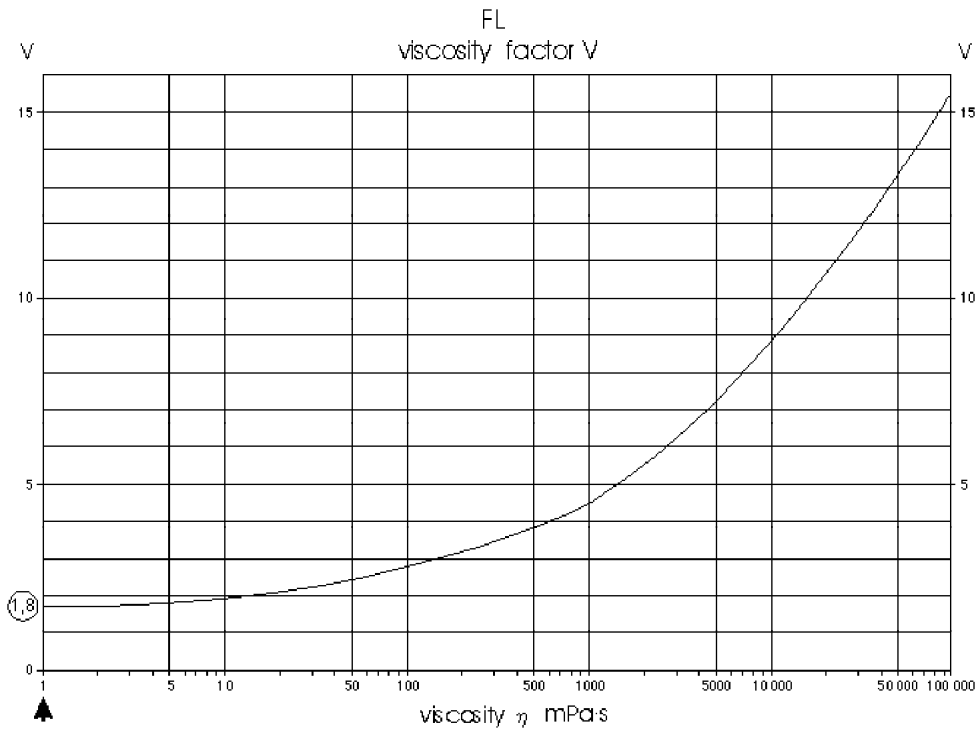
Step 1:
read speed n [1/min] .



read: speed n = 380 1/min

Case 1: viscosity
$\eta \cup$ 1 mPa s

Step 2:
define viscosity factor



stated in the diagram: viscosity factor V = 1.8

Step 3:

Calculate the absorbed power N [kW] to select the pump drive.

Case 1: viscosity
η ∪ 1 mPa s

$$N = \frac{(2 \times p + V) \times n \times C}{1000}$$

p = pressure in bar ∪ H/10
V = viscosity factor
n = speed with H = 0
C = flow rate/revolution [l/rev.]

FLF	55S	55L	75S	75L	100S	100L	120S	120L
C	0.106	0.152	0.283	0.389	0.69	1.07	1.80	2.54

Example:

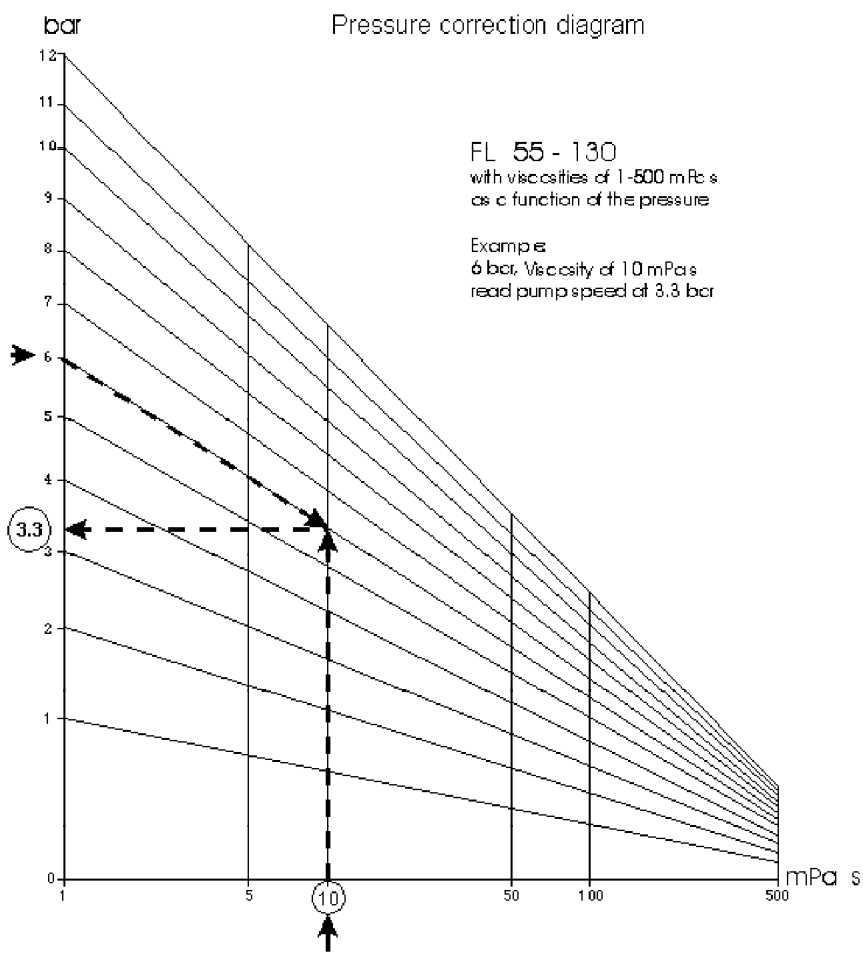
$$N = \frac{(2 \times 6 + 1.8) \times 380 \times 0.389}{1000} = 2.04 \text{ kW}$$

Case 2:
viscous product
η up to 500 mPa s

Flow rate	Q = 2000 l/h
Total head	H = 60 m
Viscosity	η = 10 mPa s

Step 1:
define correction for the speed

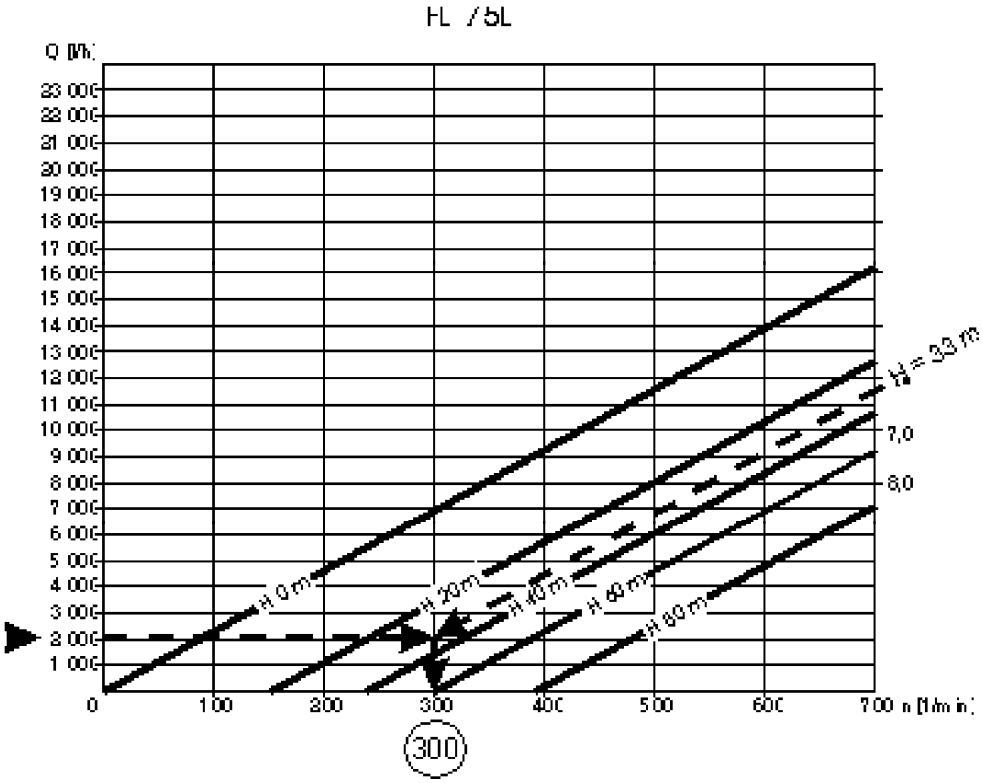
H = 60 m ∪ 6 bar



read speed at H = 33 m (equal to p = 3.3 bar)

Step 2:
read speed at H = 33 m (∪ 3.3 bar)

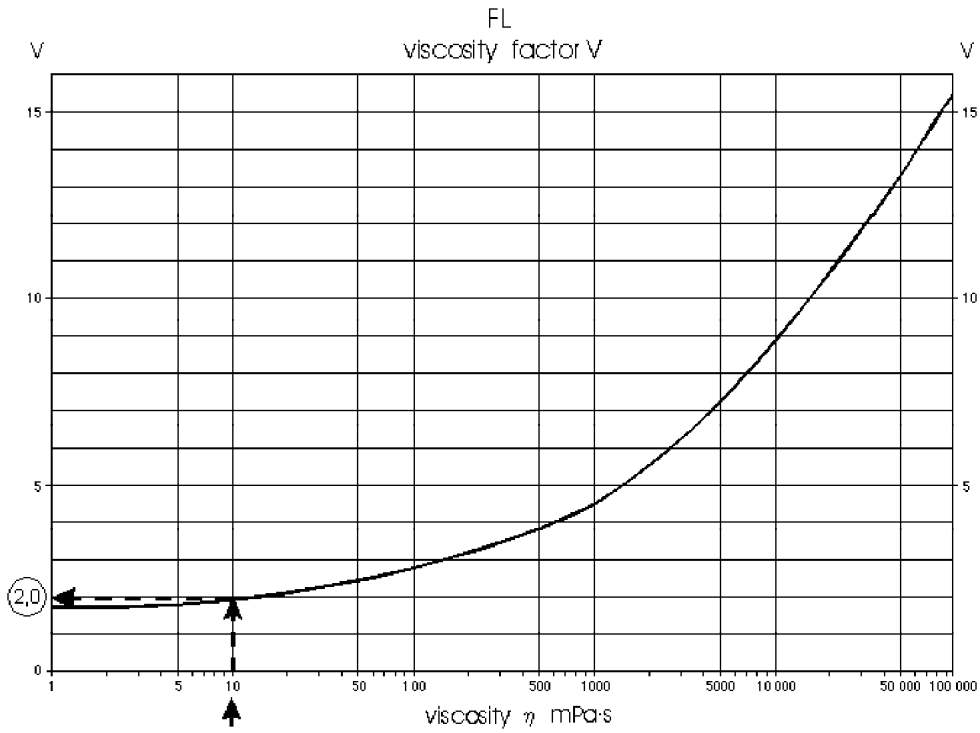
Case 2:
viscous product
η up to 500 mPa s



read: speed n = 300 1/min

Case 2: viscous product η up to 500 mPa s

Step 3:
define viscosity factor



stated in the diagram: viscosity factor V = 2.0

Case 2:
viscous product
η up to 500 mPa s

Step 4:
Calculate absorbed power N [kW] to select the pump drive.

$$N = \frac{(2 \times p + V) \times n \times C}{1000}$$

p = pressure in bar ∪ H/10
V = viscosity factor
n = speed at H = 0
C = flow rate/revolution [l/rev.]

FLF	55S	55L	75S	75L	100S	100L	120S	120L
C	0.106	0.152	0.283	0.389	0.69	1.07	1.80	2.54

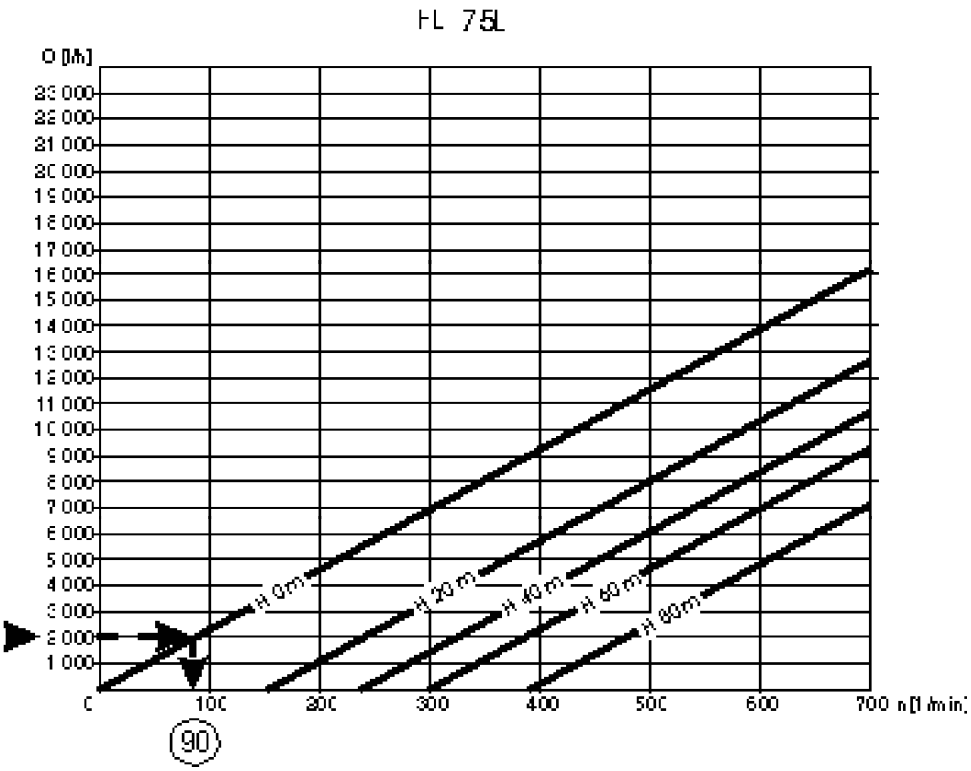
Example :

$$N = \frac{(2 \times 6 + 2) \times 300 \times 0.389}{1000} = 1.63 \text{ kW}$$

Case 3:
viscous product
η = 500–100,000 mPa s

Flow rate	Q = 2000 l/h
Total head	H = 60 m (∪ 6 bar)
viscosity	η = 10,000 mPa s

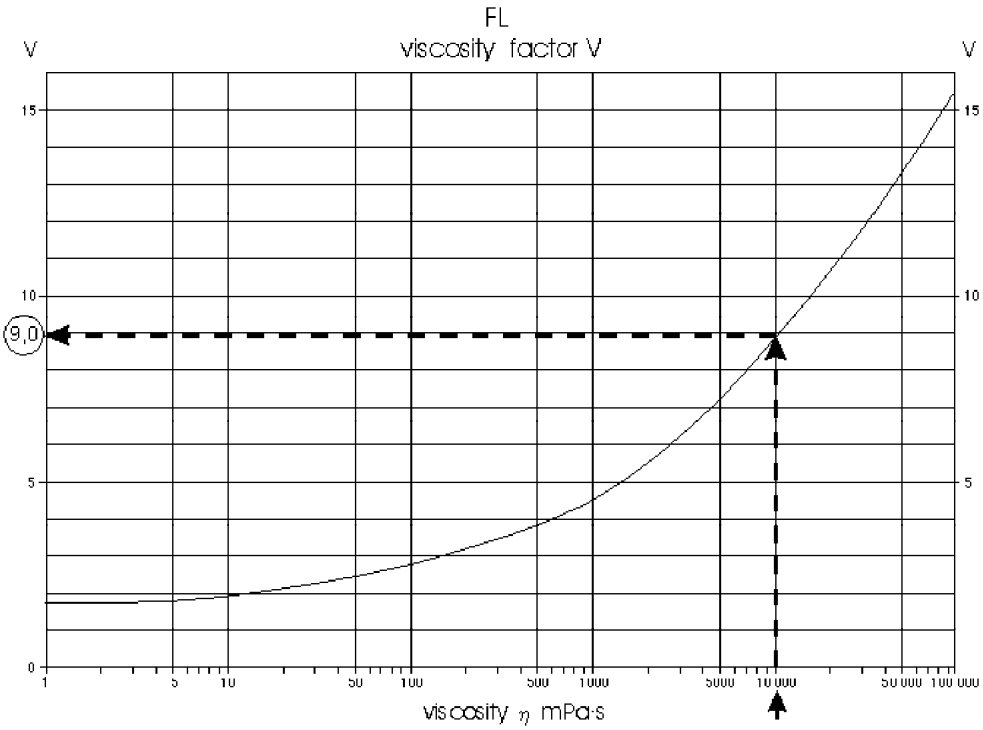
Step 1:
read speed at H = 0 m, as η > 500 mPa s



read: n = 90 1/min

Step 2:
define viscosity factor

Case 3:
viscous product
η = 500–100,000 mPa s



Viscosity factor V = 9.0

Case 3:
viscous product
η = 500–100,000 mPa s

Step 3:
calculate absorbed power N [kW].

$$N = \frac{(2 \times p + V) \times n \times C}{1000}$$

p = pressure in bar ∪ H/10
V = viscosity factor
n = speed at H = 0
C = flow rate/revolution [l/rev.]

FLF	55S	55L	75S	75L	100S	100L	120S	120L
C	0.106	0.152	0.283	0.389	0.69	1.07	1.80	2.54

Example :

$$N = \frac{(2 \times 6 + 9) \times 90 \times 0.389}{1000} = 0.74 \text{ kW}$$

Mechanical seal

Mechanical seals are devices to seal machines between rotating parts (shafts), and stationary parts (pump housing).

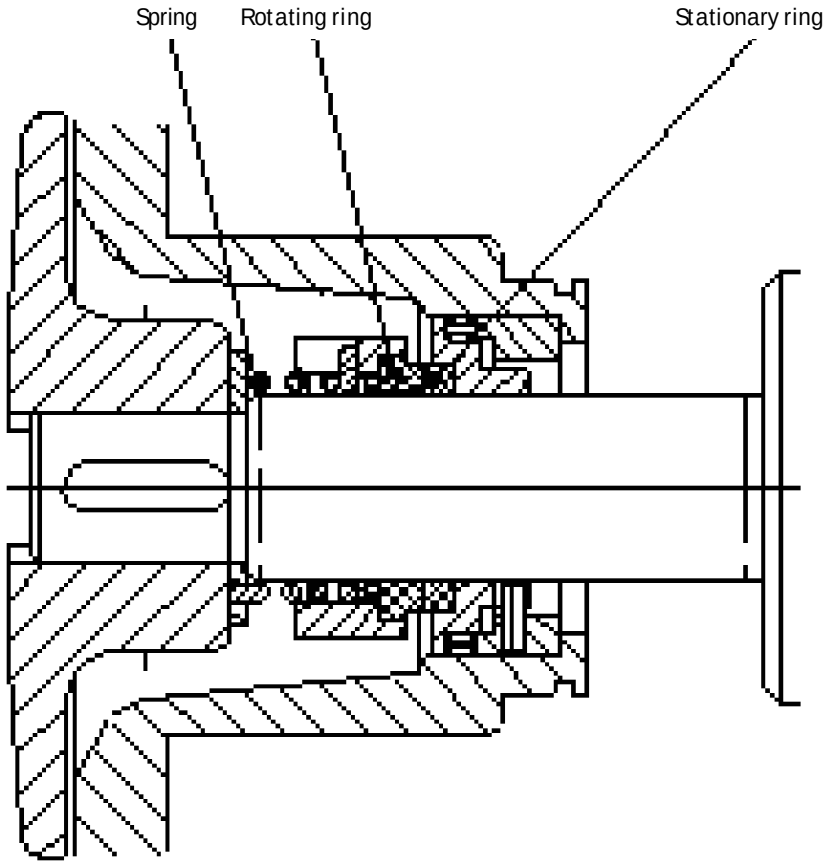
There are two types of mechanical seals:

- single mechanical seals
- double mechanical seals

Single mechanical seals

Mechanical seals consist of two surfaces which slide against each other The surfaces are pressed together by a spring. Between these two surfaces a fluid film is generated by the pumped product.

This fluid film prevents that the mechanical seal touches the stationary ring. The absence of this fluid film will result in frictional heat and the destruction of the mechanical seal (dry run of the pump).



The spring is in the product. The product pressure acts additional to the spring on the rotating seal part.

Therefore standard mechanical seals are used only for a pressure up to 10 bar. For higher pressures, balanced mechanical seals are used.

Double mechanical seals

In this case two mechanical seals are arranged in series. The inboard or, "primary seal" keeps the product in the pump housing. The outboard or, "secondary seal" prevents leakage of the flush liquid into the atmoshere.

The double mechanical seals can be provided by ***Fristam*** in two different arrangements:

- Back to Back
- Face to Face

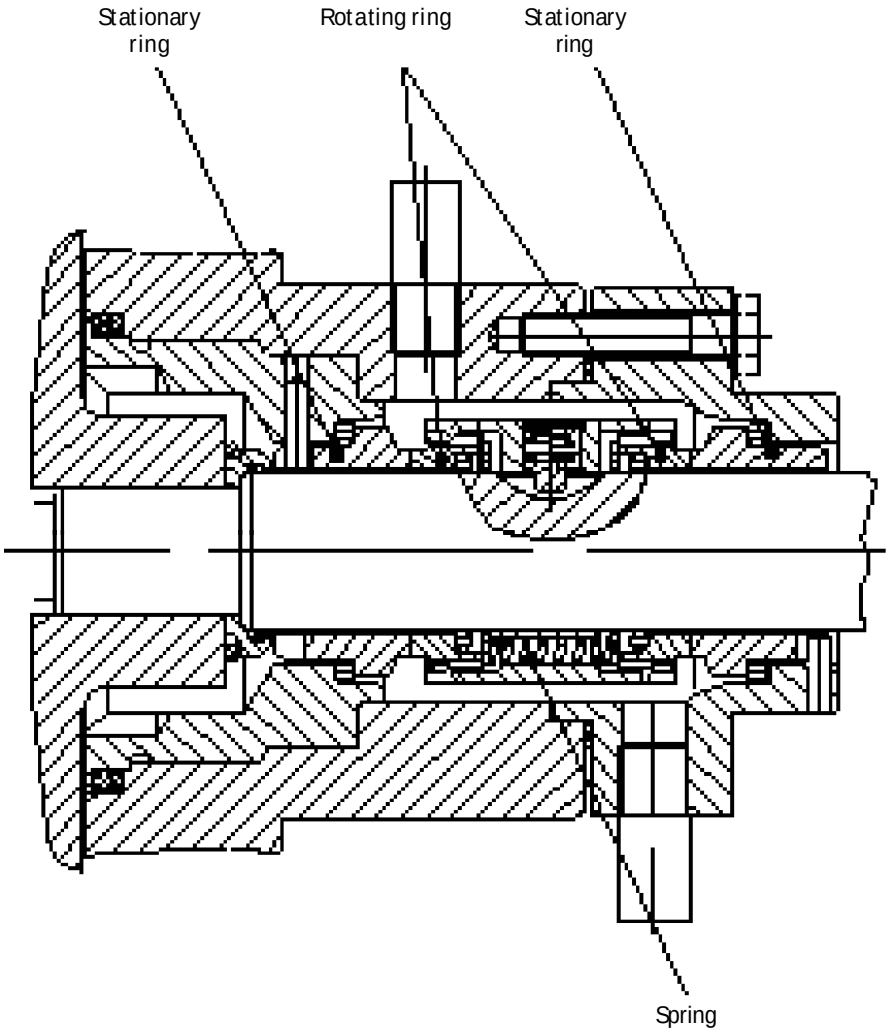
These mechanical seal arrangements are used,

- if a fluid product leakage needs to be avoided,
- when aggressive media are used or at high pressures and temperatures,
- for many polymerising, sticky media and media which tend to sedimentation,
- for vacuum applications.

"Back to Back" arrangement

Two rotating seal rings are arranged facing away from each other "Back to Back". The lubricating film is generated by the barrier fluid.

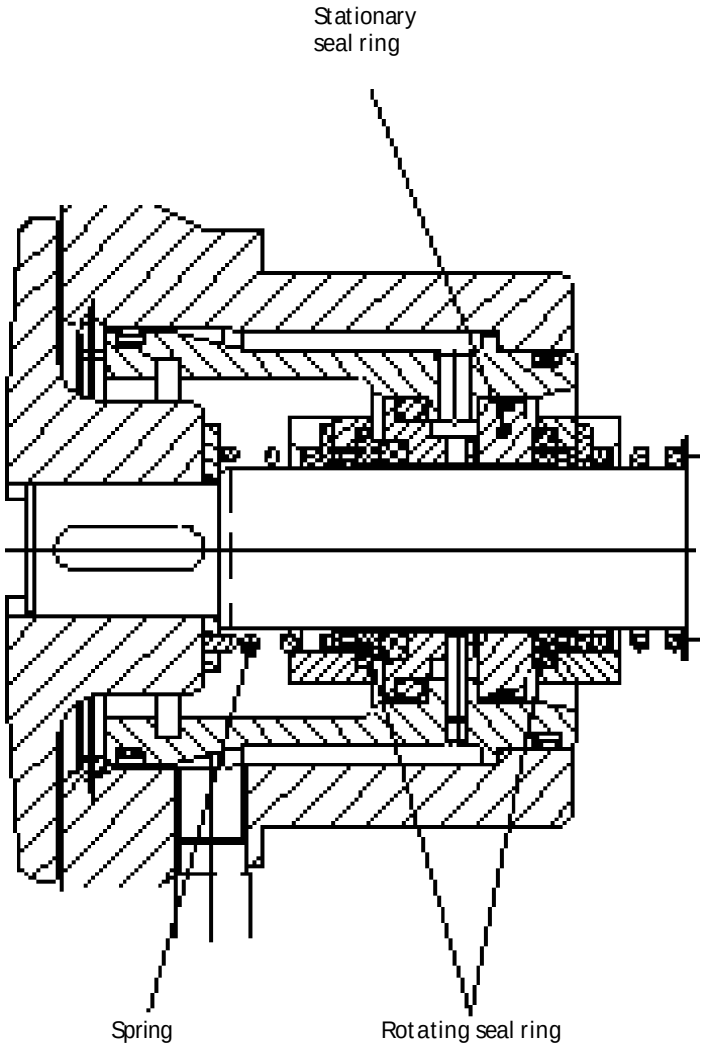
The barrier pressure in the case of a "Back to Back" arrangement should be 1.5 up to 2.0 bar above the product pressure in the seal area.



Mechanical seals with a "Back to Back" arrangement are mainly used in the chemical industry. In case of a leakage, the barrier liquid penetrates the product.

"Face to Face" arrangement

The spring loaded rotary seal faces are arranged face to face and slide from the opposite direction to one or two stationary seal part(s).



The mechanical seals with a "Face to Face" arrangement are often used in the food industry in particular for products which tend to stick and for vacuum applications. The barrier pressures are very low (0.2 bar). In the case of leakage the product penetrates the barrier liquid.

In the case of hot products the barrier liquid also acts as a cooling agent for the mechanical seal.

Mechanical seal designs

Fristam has many years of experience in the manufacture of mechanical seals and is able to provide the best mechanical seal for any application.

Standard mechanical seals in conformity with DIN 24 960 can be mounted without problems.

Seal face materials

Material	Properties
1. Carbons	
synthetic resin, impregnated	Good antifrictional properties, high temperature stability. Chemical stability is to be tested.
2. Metals	
Chromium-Nickel-Molybdenum	Good chemical stability.
3. Metal carbides	
3.1 Tungsten carbide	Low thermal conductivity, but high hardness and wear resistance.
3.2 Silicon carbide	Hardness greater than with tungsten carbide, outstanding chemical stability, good antifrictional properties and thermal conductivity, but very brittle.
4. Ceramics	High quality aluminium oxide, high wear resistance, good chemical stability, low thermal conductivity, sensitive to thermal shocks

Elastomers

Material	Range of temperatures	Properties
1. Nitrile	−30 up to + 100°C	Resistant to water, vapour, mineral and vegetable shortening (fat) and oils, alcohol, salt solutions. Not resistant to aromatic and chlorine hydrocarbons, acids and alkaline solutions.
2. EPDM	− 50 up to + 150°C	Good thermal properties, can be used for alcoholic solutions, diluted acids and concentrated alkaline solutions. Not resistant to mineral and vegetable shortening (fat) and oils, and hydrocarbons.
3. Viton (FKM)	−25 up to + 200°C	Good thermal resistivity, water resistant, vapour resistant, resistance to mineral and vegetable fat and oils, to alcohol, to acids and alkaline solutions, salt solutions. Not resistant to ketones such as acetone and ester.
4. PTFE	−20 up to + 200°C	Best chemical and thermal resistivity to all aggressive liquids, elasticity ensured through use of Viton-caoutchouc or EP- core material.

The transfer of hygienic, high quality products requires a clean pump. Therefore at the end of a production process the pumps must be cleaned immediately. The plant needs to be clean and free from germs before starting a new production cycle.

Cleaning

Cleaning is the operation to remove all traces of product from the plant. A properly cleaned surface is free from visible, perceptible or chemically detectable dirt deposits (residue).

CIP

The standard cleaning process for plants is the **CIP** - (**C**leaning **i**n **P**lace). This implies cleaning without dismantling of the plant by means of CIP fluids.

In the food industry a CIP process requires the following steps:

- preliminary rinsing with water
- flushing with alkaline solution
- intermediate rinsing with water
- flushing with acid
- rinsing with clean water

CIP plant cleaning

To clean the unit efficiently a turbulent flow of the CIP fluid is required. A minimum flow velocity in pipes is usually 2 m/s.

Viscous fluids are often transferred by positive displacement pumps at low flow velocities. In order to obtain the flow rate required for CIP it may be necessary to fit an additional cleaning pump such as a centrifugal pump.

CIP compatible design

Fristam pumps are fully CIP capable. They are characterised by:

- welded and ground joints
- round edges and angles
- smooth joining
- no narrow gaps and dead legs
- O-rings immersed in the pump housing
- smooth, nonporous internals with a high surface finish

SIP After CIP cleaning an additional sterilisation in place process (SIP) may be required when highly sensitive products are handled, inactivating any micro-organisms which might be still present in the pump.

The sterilisation can be carried out by means of chemicals, hot water or steam. In the dairy industry the sterilisation temperature is approximately 145° C.

Units of measure

Term	Symbol	Unit used in practice	Coherent unit
Flow rate	Q	m^3/h	l/h
Total head	H	m	m
NPSH-value	NPSH	m	m
Power consumption	N	kW	Nm/s
Pump efficiency	η	%	–
Speed	n	1/min	1/s
Pressure	p	Pa	N/m^2
Density	ρ	kg/dm^3	kg/m^3
Fluid flow velocity	v	m/s	m/s
Local gravitational acceleration	g	m/s^2	m/s^2

Used indices:

geo = total discharge head
S = suction side
D = discharge side
E = inlet cross section
A = discharge cross section
V = loss

Conversion

Term	Unit	
Pressure p	Pascal	Pa, 100 kPa=1 bar
Total head	Meter	1 m = 0.1 bar *)
Dynamic viscosity	mPa s	1 mPa s = 1 cP

*) valid only for water

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