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## Units of measure and conversion

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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 [m³/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.
  \[
  H_{geo} = H_{geo}^{o} \pm H_{geo}^{d}
  \]

- the friction loss resulting from pipe walls, fittings and valves within the plant.
  \[
  H_v = H_{VS} \pm H_{vd}
  \]

- 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.
Looking at two parallel plates with the surface $A$ and the distance $y$, displaced against each other as a result of a force $F_{\text{action}}$ with a velocity $v$, a force $F_{\text{reaction}}$ 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 $\tau$.

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

The shear stress $\tau$ increases in proportion to the shear velocity $D$ and the dynamic viscosity $\eta$.

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

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

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

Thus the resulting dynamic viscosity $\eta$:

$$\eta = \frac{\tau}{D}$$
Thus, the dynamic viscosity $\eta$ 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.

---

**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.
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 parallelly 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.

The Reynolds number describes the correlation between the flow velocity $v$, the viscosity $\eta$ and the inner diameter of the pipe $d$.

The Reynolds number has no dimension.

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

Row velocity $v$ [m/s]
Viscosity $\eta$ [Pa s]
Inner pipe diameter $d$ [mm]
Density $\rho$ [kg/dm$^3$]

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 $\eta = 1.21$ mPa s = 0.00121 Pa s and a density of 1.04 kg/dm$^3$.

Is the flow laminar or turbulent?

The average flow velocity amounts to:

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

$Q$ [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 Net Positive Suction Head.

Besides the flow rate \( Q \) and the pump head \( H \), the NPSH value is one of the most important characteristic parameters 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.

\[
p_e = \text{pressure at the inlet cross section of the plant [bar]}
\]

\[
p_a = \text{pressure at the outlet cross section of the plant [bar]}
\]

\[
p_o = \text{vapour pressure of the fluid at the middle of the suction connection of the pump [bar]}
\]

\[
p_b = \text{air pressure at the installation site of the pump [bar]}
\]

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

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

\[
\rho = \text{density of the fluid [kg/m}^3\text{]}
\]

\[
v_c = \text{inlet flow velocity [m/s]}
\]

\[
\text{NPSH} = \frac{p_e + p_o - p_b}{\rho \times g} + \frac{v_c^2}{2g} + H_{geo} - H_{vs}
\]
In order to ensure a correct operation of the pump the following condition must be given:

\[ N_{\text{PSH plant}} > N_{\text{PSH pump}} \]

Boiling fluids with a velocity up to 0.3 m/s are a special case.

In this case: \( p_E = p_D \) as \( \frac{v_s^2}{2g} \) and \( H_{vs} \) become negligible resulting in:

\[ N_{\text{PSH plant}} = H_{\text{spec}} \]
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:
Row rate \( Q = 25 \text{ m}^3/\text{h} \)
Pipe diameter \( d = 50 \text{ mm} \)
From the diagram results:
Row velocity \( v = 3.5 \text{ m/s} \)
Loss of head \( H_v = 35 \text{ m/100 m} \)

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 \( Pe \geq 100,000 \) and roughness \( k \leq 0.04 \text{ mm} \))

<table>
<thead>
<tr>
<th>Diameter DN [mm]</th>
<th>25</th>
<th>40</th>
<th>50</th>
<th>65</th>
<th>80</th>
<th>100</th>
<th>125</th>
<th>150</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard valve</td>
<td>2.1</td>
<td>4.6</td>
<td>7.5</td>
<td>11.0</td>
<td>14.0</td>
<td>20.0</td>
<td>28.0</td>
<td>37.0</td>
</tr>
<tr>
<td>Free-flow valve</td>
<td>0.7</td>
<td>0.9</td>
<td>1.2</td>
<td>1.5</td>
<td>1.8</td>
<td>2.2</td>
<td>2.9</td>
<td>3.7</td>
</tr>
<tr>
<td>90° valve</td>
<td>3.2</td>
<td>5.0</td>
<td>7.2</td>
<td>10.0</td>
<td>13.0</td>
<td>17.0</td>
<td>23.0</td>
<td>31.0</td>
</tr>
<tr>
<td>Elbow 90° R = 4 \times d</td>
<td>0.5</td>
<td>0.6</td>
<td>0.7</td>
<td>0.9</td>
<td>1.2</td>
<td>1.5</td>
<td>1.9</td>
<td>2.3</td>
</tr>
<tr>
<td>Elbow 90° R = 3 \times d</td>
<td>0.7</td>
<td>0.9</td>
<td>1.1</td>
<td>1.4</td>
<td>1.7</td>
<td>2.2</td>
<td>2.8</td>
<td>3.5</td>
</tr>
<tr>
<td>Tee (diffusion)</td>
<td>1.6</td>
<td>2.0</td>
<td>2.5</td>
<td>3.3</td>
<td>4.0</td>
<td>4.8</td>
<td>5.7</td>
<td>6.8</td>
</tr>
<tr>
<td>Tee (junction)</td>
<td>2.0</td>
<td>2.6</td>
<td>3.2</td>
<td>4.4</td>
<td>5.8</td>
<td>7.5</td>
<td>10.0</td>
<td>13.0</td>
</tr>
</tbody>
</table>
Loss of head
calculation

Example:

Flow rate \( Q = 25 \text{ m}^3/\text{h} \)
Straight pipe length \( l = 150 \text{ m} \)
Diameter \( DN = 50 \text{ mm} \)
Elbow 90° \( 4 \text{ pieces} \)
Free-flow valves \( 2 \text{ pieces} \)

from diagram (page 16):
\( v = 3.5 \text{ m/s} \)
\( H_v = 35 \text{ m/100 m pipe length} \)

from table:
- equivalent pipe length 4 elbows: \( l_{\text{bend}} = 1.1 \times 4 = 4.4 \text{ m} \)
- equivalent pipe length 2 free-flow valves: \( l_{\text{slide}} = 1.2 \times 2 = 2.4 \text{ m} \)
- straight pipe length: \( l_{\text{pipe}} = 150.0 \text{ m} \)
- total pipe length \( l_{\text{total}} = 156.8 \text{ m} \)

loss of head:

\[
H_{\text{total}} = H_v \times \frac{l_{\text{total}}}{100}
\]

\[
H_v = \frac{3.5}{100} \times 156.8 = 52 \text{ m}
\]

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

\[
\Delta p_v = \frac{\nu \times 32 \times \eta \times l}{d^4 \times 10^3}
\]

\( H_v \cup 10 \approx \Delta p_v \)
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.

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

\[
\begin{align*}
\text{Viscosity } \eta &< 1000 \text{ mPa s} \\
\text{Viscosity } \eta &> 1000 \text{ mPa s}
\end{align*}
\]

**Fristam centrifugal pumps**

*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.
Centrifugal pumps

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 Q₁ to Q₂.
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

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.

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

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

| $\rho$ | [kg/m$^3$] |
| $v_1$ | [m/s] |
| $\Delta P_v$ | [bar] |
**Centrifugal pumps**

See the values $\zeta$ stated in the table below.

<table>
<thead>
<tr>
<th>Aperture ratio $m = (d/D)^2$</th>
<th>Resistance value $\zeta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05</td>
<td>600</td>
</tr>
<tr>
<td>0.1</td>
<td>250</td>
</tr>
<tr>
<td>0.2</td>
<td>50</td>
</tr>
<tr>
<td>0.3</td>
<td>20</td>
</tr>
<tr>
<td>0.4</td>
<td>4</td>
</tr>
</tbody>
</table>

Calculation:

- take the figure stated in the table for $d$, see table $\zeta$, calculate $\Delta p_v$.
- if $\Delta p_v$ varies from the required value, take new value for $d$ and calculate once more $\Delta p_v$.

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:

\[
N = \text{power consumption} \\
D = \text{impeller diameter} \\
Q = \text{flow rate} \\
H = \text{total head}
\]

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

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.
**Centrifugal pumps**

**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?

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 transferred.

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 formula is valid:

\[ \text{NPSH}_{\text{plant}} > \text{NPSH}_{\text{pump}} + 0.5 \text{ m} \]

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.
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.
Centrifugal pumps

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

Selecting the correct size

Example:

Flow rate $Q_A = 90 \text{ m}^3/\text{h}$

Total head $H_A = 75 \text{ m}$

Step 1:
Select the pump size.

Selected pump size: **FP 3552**
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)

Impeller diameter resulting from the diagram = 230 mm
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: $N = 26 \text{ kW}$
selected motor: $30.0 \text{ kW}$

Step 4:
Check the efficiency

$$\eta = \frac{Q \times H \times \rho}{367 \times N}$$
\begin{align*}
Q & \text{[m$^3$/h]} \\
H & \text{[m]} \\
N & \text{[kW]} \\
\rho & \text{[kg/dm$^3$]}
\end{align*}

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

$\rho_{\text{water}} = 1 \text{ kg/dm$^3$}$

$$\eta = 0.7 \Rightarrow 70\%$$
Step 5:
Check if $NPSH_{plant} > NPSH_{pump}$

Check NPSH value

Resulting NPSH value of the pump from the diagram = 2.4 m
Self-priming centrifugal pump FZ

Selecting the correct size

Example:
Flow rate \( Q_A = 30 \text{ m}^3/\text{h} \)
Total head \( H_A = 24 \text{ m} \)

Step 1:
Select the pump size whose curve is above to the operating point of the plant.

FZ sizes

Selected pump size: **FZ 22**
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 diagram 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).
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 clearances 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.

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

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 transferred.

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.
Fristam supplies two different positive displacement 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.

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.
Positive displacement pumps

Selection of design
The design selection depends amongst other:

- **Fristam** rotary lobe pump
  - FL, maximum total head 120 m (12 bar)

- **Fristam** circumferential piston pump
  - FK, maximum total head 200 m (20 bar)
  - FKL, maximum total head 250 m (25 bar)

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:

Flow rate $Q = 3000 \text{ l/h}$
Total head $H = 120 \text{ m}$

Pump to be used for products with different viscosities.

**FK pump basic selection diagram**

- For case 1: water
- For case 2: 10 mPa s
- For case 3: 10,000 mPa s

Selected: **FK 40**
Example:

\[ Q = 3000 \text{ l/h} \]
\[ H = 120 \text{ m} \]
\[ \eta = 1 \text{ mPa s} \]

Step 1:
read speed \( n \) [1/min]

resulting from the diagram: speed \( n = 380 \text{ 1/min} \)
Step 2:
define viscosity factor

Case 1: viscosity
\[ \eta \cup 1 \text{ mPa s} \]

Viscosity factor \( V = 1.8 \)
Positive displacement pumps

Case 1: viscosity \( \eta \leq 1 \text{ mPa s} \)

Step 3:
Calculate the power \( N \text{ [kW]} \) required for the pump drive.

\[ p = \text{pressure [bar]} \]
\[ V = \text{viscosity factor} \]
\[ n = \text{speed [1/min], stated in the diagram} \]
\[ C = \text{flow rate/revolution [l/rev.]} \]

<table>
<thead>
<tr>
<th>FC</th>
<th>25</th>
<th>25/30</th>
<th>40</th>
<th>40/45</th>
<th>48</th>
<th>50</th>
<th>50/75</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>0.07</td>
<td>0.11</td>
<td>0.26</td>
<td>0.36</td>
<td>0.77</td>
<td>1.1</td>
<td>1.37</td>
</tr>
</tbody>
</table>

Example:
\[
N = \frac{\left(2 \times 12 + 1.8\right) \times 380 \times 0.26}{1000} = 2.5 \text{ kW}
\]
Example:
Q = 3000 l/h
H = 120 m \cup p = 12 \text{ bar}
\eta = 10 \text{ mPa s}

Step 1:
speed correction

Pressure correction diagram

FK 25 - 50/75
with viscosities of 1-200 mPa s
as a function of the pressure

Examples:
120 m \sim 12 \text{ bar}, viscosity of 10 mPa s
read pump speed of 3.8 bar

stated in the diagram: p = 3.8 \text{ bar.}

Define now the speed required for the corrected pressure.

Case 2:
viscous product
\eta \text{ up to } 200 \text{ mPa s}
Positive displacement pumps

Case 2: viscous product

\( \eta \) 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
\( \eta \) up to 200 mPa s

stated in the diagram: viscosity factor \( V = 2.0 \)
Positive
displacement
pumps

Case 2: viscous product \( \eta \) 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 \( \cup \) H/10

\( V \) = viscosity factor

\( n \) = speed with H = 38 m

\( C \) = flow rate/revolution [l/rev.]

<table>
<thead>
<tr>
<th>FK</th>
<th>25</th>
<th>25/30</th>
<th>40</th>
<th>40/45</th>
<th>48</th>
<th>50</th>
<th>50/75</th>
</tr>
</thead>
<tbody>
<tr>
<td>( C )</td>
<td>0.07</td>
<td>0.11</td>
<td>0.26</td>
<td>0.36</td>
<td>0.77</td>
<td>1.1</td>
<td>1.37</td>
</tr>
</tbody>
</table>

Example:

\[
N = \frac{(2 \times 12 + 2) \times 300 \times 0.26}{1000} = 2.03 \text{ kW}
\]
Example:

\[ \begin{align*}
Q &= 3000 \text{ l/h} \\
H &= 120 \text{ m} \\
\eta &= 10,000 \text{ mPa s}
\end{align*} \]

\[ \begin{align*}
\eta &= 12 \text{ bar}
\end{align*} \]

**Step 1:**
read speed with \( H = 0, \eta > 200 \text{ mPa s} \)

---

**Case 3:**
viscous product
\[ \eta = 200–100,000 \text{ mPa s} \]

Stated in the diagram: \( n = 220 \text{ 1/min} \)
Positive displacement pumps

Case 3: viscous product \( \eta = 200-100,000 \text{ mPa s} \)

Step 2: read viscosity factor \( V \).

\[ \text{stated in the diagram: } V = 9.0 \]
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]$\cup H/10$
$V =$ viscosity factor
$n =$ speed [1/min], stated in the diagram
$C =$ flow rate/revolution [l/rev]

Example:

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

Case 3:
viscous product
$\eta = 200–100,000 \text{ mPa s}$
Positive displacement pumps

Rotary lobe pump FL

Example:
Flow rate \( Q = 2000 \text{l/h} \)
Total head \( H = 60 \text{ m} \)

Selection
Pump to be used for products with different viscosities.

![FL - basic selection diagram](image)

for case number 1: water
for case number 2: 10 mPa s
for case number 3: 10,000 mPa s

selected: FL 75 L
Positive displacement pumps

Row rate \( Q = 2000 \text{ l/h} \)
Total head \( H = 60 \text{ m} \)
\( \eta = 1 \text{ mPa s (water).} \)

Step 1:
read speed \( n \) [1/min].

Case 1: viscosity
\( \eta \cup 1 \text{ mPa s} \)

read: speed \( n = 380 \text{ 1/min} \)
Positive displacement pumps

Step 2: define viscosity factor

Case 1: viscosity $\eta \cup 1 \text{ mPa s}$

Stated in the diagram: viscosity factor $V = 1.8$
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 in bar $\cup$ $\times 10$
$V =$ viscosity factor
$n =$ speed with $H = 0$
$C =$ flow rate/revolution [l/rev]

<table>
<thead>
<tr>
<th>FLF</th>
<th>55S</th>
<th>55L</th>
<th>75S</th>
<th>75L</th>
<th>100S</th>
<th>100L</th>
<th>120S</th>
<th>120L</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C$</td>
<td>0.106</td>
<td>0.152</td>
<td>0.283</td>
<td>0.289</td>
<td>0.69</td>
<td>1.07</td>
<td>1.80</td>
<td>2.54</td>
</tr>
</tbody>
</table>

Example:

$$N = \frac{(2 \times 6 + 1.8) \times 360 \times 0.389}{1000} = 2.04 \text{ kW}$$
Case 2: Viscous product
\( \eta \) up to 500 mPa s

<table>
<thead>
<tr>
<th>Row rate</th>
<th>Q = 2000 l/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total head</td>
<td>H = 60 m</td>
</tr>
<tr>
<td>Viscosity</td>
<td>( \eta = 10 \text{ mPa s} )</td>
</tr>
</tbody>
</table>

Step 1:
Define correction for the speed

\[ H = 60 \text{ m } \cup \text{ 6 bar} \]

Pressure correction diagram

\[ FL \ 55 - 130 \]
with viscosities of 1-500 mPa s
cr as a function of the pressure

Examples:
6 bar: Viscosity of 10 mPa s
read pump speed of 3.3 bar

Read speed at \( H = 33 \text{ m} \) (equal to \( p = 3.3 \text{ bar} \))
Step 2:
read speed at \( H = 33 \text{ m} \) (\( \cup 3.3 \text{ bar} \))

Case 2:
viscous product
\( \eta \) up to 500 mPa s

read: speed \( n = 300 \text{ 1/min} \)
Positive
displacement
pumps

Case 2:
viscous product
$\eta$ up to 500 mPa s

Step 3:
define viscosity factor

stated in the diagram: viscosity factor $V = 2.0$
Step 4: Calculate absorbed power $N \,[\text{kW}]$ to select the pump drive.

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

$p = \text{pressure in bar} \cup \text{H}/10$
$V = \text{viscosity factor}$
$n = \text{speed at } H = 0$
$C = \text{flow rate/revolution} \,[\text{l/rev}]

<table>
<thead>
<tr>
<th></th>
<th>FLF</th>
<th>55S</th>
<th>55L</th>
<th>75S</th>
<th>75L</th>
<th>100S</th>
<th>100L</th>
<th>120S</th>
<th>120L</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C$</td>
<td>0.106</td>
<td>0.152</td>
<td>0.283</td>
<td>0.289</td>
<td>0.69</td>
<td>1.07</td>
<td>1.80</td>
<td>2.54</td>
<td></td>
</tr>
</tbody>
</table>

Example:

$$N = \frac{(2 \times 6 + 2) \times 300 \times 0.389}{1000} = 1.63 \,[\text{kW}]$$
Positive displacement pumps

Case 3: Viscous product

- Flow rate: $Q = 2000 \text{ l/h}$
- Total head: $H = 60 \text{ m (} \approx 6 \text{ bar)}$
- Viscosity: $\eta = 10,000 \text{ mPa s}$

**Step 1:**

Read speed at $H = 0 \text{ m}$, as $\eta > 500 \text{ mPa s}$

[Graph showing relationship between flow rate and speed]

Read: $n = 90 \text{ 1/min}$
Positive displacement pumps

Step 2:
define viscosity factor

Case 3:
v viscous product
\[ \eta = 500–100,000 \text{ mPa} \text{ s} \]

Viscosity factor \( V = 9.0 \)
Positive displacement pumps

Case 3: viscous product

\( \eta = 500-100,000 \text{ mPa s} \)

Step 3: calculate absorbed power \( N \) [kW].

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

- \( p \) = pressure in bar \( \times \frac{1}{10} \)
- \( V \) = viscosity factor
- \( n \) = speed at \( H = 0 \)
- \( C \) = flow rate/revolution [l/rev.]

Example:

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

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).
Mechanical seal selection

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 atmosphere.

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.
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.
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 selection

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.

<table>
<thead>
<tr>
<th>Material</th>
<th>Properties</th>
<th>Seal face materials</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Carbons</td>
<td>Good antifrictional properties, high temperature stability. Chemical stability is to be tested.</td>
<td></td>
</tr>
<tr>
<td>synthetic resin, impregnated</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2. Metals</td>
<td>Good chemical stability.</td>
<td></td>
</tr>
<tr>
<td>Chromium-Nickel-Molybdenum</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3. Metal carbides</td>
<td>Low thermal conductivity, but high hardness and wear resistance.</td>
<td></td>
</tr>
<tr>
<td>3.1 Tungsten carbide</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.2 Silicon carbide</td>
<td>Hardness greater than with tungsten carbide, outstanding chemical stability, good antifrictional properties and thermal conductivity, but very brittle.</td>
<td></td>
</tr>
<tr>
<td>4. Ceramics</td>
<td>High quality aluminium oxide, high wear resistance, good chemical stability, low thermal conductivity, sensitive to thermal shocks</td>
<td></td>
</tr>
<tr>
<td>Elastomers</td>
<td>Material</td>
<td>Range of temperatures</td>
</tr>
<tr>
<td>-------------</td>
<td>----------</td>
<td>-----------------------</td>
</tr>
<tr>
<td>1. Nitrile</td>
<td>-30 up to +100°C</td>
<td>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.</td>
</tr>
<tr>
<td>2. EPDM</td>
<td>-50 up to +150°C</td>
<td>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.</td>
</tr>
<tr>
<td>3. Viton (FKM)</td>
<td>-25 up to +200°C</td>
<td>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.</td>
</tr>
<tr>
<td>4. PTFE</td>
<td>-20 up to +200°C</td>
<td>Best chemical and thermal resistivity to all aggressive liquids, elasticity ensured through use of Viton-caoutchouc or EP-core material.</td>
</tr>
</tbody>
</table>
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 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).

The standard cleaning process for plants is the CIP - (Cleaning in Place). 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

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.

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
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

<table>
<thead>
<tr>
<th>Term</th>
<th>Symbol</th>
<th>Unit used in practice</th>
<th>Coherent unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate</td>
<td>Q</td>
<td>m³/h</td>
<td>l/h</td>
</tr>
<tr>
<td>Total head</td>
<td>H</td>
<td>m</td>
<td>m</td>
</tr>
<tr>
<td>NPSH-value</td>
<td>NPSH</td>
<td>m</td>
<td>m</td>
</tr>
<tr>
<td>Power consumption</td>
<td>N</td>
<td>kW</td>
<td>Nm/s</td>
</tr>
<tr>
<td>Pump efficiency</td>
<td>η</td>
<td>%</td>
<td>–</td>
</tr>
<tr>
<td>Speed</td>
<td>n</td>
<td>1/min</td>
<td>1/s</td>
</tr>
<tr>
<td>Pressure</td>
<td>p</td>
<td>Pa</td>
<td>N/m²</td>
</tr>
<tr>
<td>Density</td>
<td>ρ</td>
<td>kg/dm³</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Fluid flow velocity</td>
<td>v</td>
<td>m/s</td>
<td>m/s</td>
</tr>
<tr>
<td>Local gravitational acceleration</td>
<td>g</td>
<td>m/s²</td>
<td>m/s²</td>
</tr>
</tbody>
</table>

**Used indices:**
- geo = total discharge head
- S = suction side
- D = discharge side
- E = inlet cross section
- A = discharge cross section
- V = loss

### Conversion

<table>
<thead>
<tr>
<th>Term</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure p</td>
<td>Pascal</td>
</tr>
<tr>
<td>Total head</td>
<td>Meter</td>
</tr>
<tr>
<td>Dynamic viscosity</td>
<td>mPa s</td>
</tr>
</tbody>
</table>

- 1 bar = 100 kPa

*) valid only for water
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Fristam N.V.

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Pompes Fristam S.N.C.
Noisy-le-Sec-Cedex

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Pune

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Fristam Italia S.r.l.
Gallarate

Japan
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Tokyo

Netherlands
Fristam B.V.
Eist (gld)

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Fristam Pumps NZ Limited
Cambridge

Peoples Republic of China
Fristam Pumps (Shanghai) Co. Ltd.
Shanghai

Poland
Fristam Polska Sp.z o.o.
Warsaw

Russian Federation
Fristam Pumpen, OOO
Moscow

Scandinavia
Fristam Pumper A/S
Saeby

South/East Asia
Fristam Pumps (S.E.A.) Pte. Ltd.
Singapore

Spain/Portugal
Fristam Iberica S.L.
Porqueres (Girona)

USA/Canada
Mexico
South America
Fristam Pumps USA,
Limited Partnership
Middleton, WI