Pump Handbook

All you need to know...

if Pumps are the question…

Alfa Laval is an acknowledged market leader in pumping technology, supplying centrifugal and positive displacements pumps worldwide to the dairy, food, beverage and pharmaceutical industries.

Contents

5.0 Pump Materials of Construction

Describes the materials, used in the construction of the Alfa Laval pump portfolio.

6.0 Pump Sealing

Describes the principle of pump sealing and illustrates the different sealing arrangements used on Alfa Laval pump ranges.

7.0 Pump Sizing

Describes how to size an Alfa Laval pump from product/fluid and performance data given.

8.0 Pump Specification Options

Gives descriptions of the various specification options available for the Alfa Laval pump ranges.

9.0 Motors

Describes electric motors, including information on motor, methods of starting, motors for hazardous environments and speed control.

10 Cleaning Guidelines

Provides cleaning guidelines for use in processes utilising CIP systems.

11 Compliance

Describes some of the international standards and guidelines applicable to Alfa Laval pump ranges.

12 Installation Guide

Covers guidelines relating to pump installation, system design and pipework layout.

13 Troubleshooting

Offers possible causes and solutions to most common problems.

14 Technical Data

Includes a summary of nomenclature and formulas used in this handbook.

15 Glossary of Terms

Explains the various terms found in this handbook.

This chapter gives a short introduction
of the Pump Handbook.

1.0

Introduction
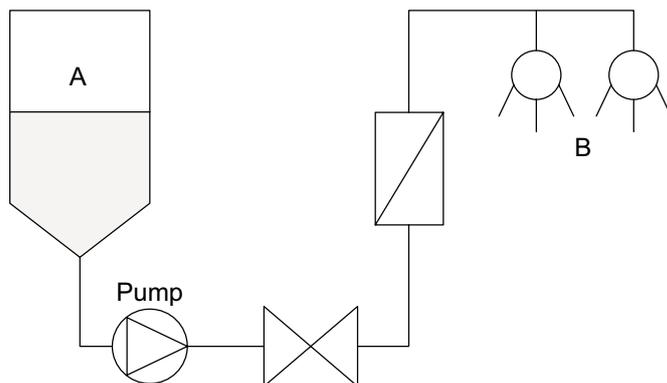
Fig. 1.1a Typical pump installation

1.1 What is a Pump?

There are many different definitions of this but at Alfa Laval we believe this is best described as:

'A machine used for the purpose of transferring quantities of fluids and/or gases, from one place to another'

This is illustrated above transferring fluid from tank A to spray nozzles B.

Pump types generally fall into two main categories - Rotodynamic and Positive Displacement, of which there are many forms as shown in Fig. 1.1b on the following pages.

The Rotodynamic pump transfers rotating mechanical energy into kinetic energy in the form of fluid velocity and pressure. The Centrifugal and Liquid Ring pumps are types of rotodynamic pumps, which utilise centrifugal force to transfer the fluid being pumped.

The Rotary Lobe pump is a type of positive displacement pump, which directly displaces the pumped fluid from pump inlet to outlet in discrete volumes.

Pump Classifications

					Pumps		

Positive Displacement

- **Rotor**
  - **Multi-Rotor**
    - Screw
    - Circumferential Piston
    - Gear
      - Internal
      - External
    - Rotary Lobe
    - Alfa Laval Rotary Lobe
  - **Single Rotor**
    - Piston
    - Archimedean Screw
    - Flexible Member
    - Peristaltic
    - Vane
    - Progressing Cavity

Reciprocating

- **Diaphragm**
  - Simplex
  - Multiplex
- **Plunger**
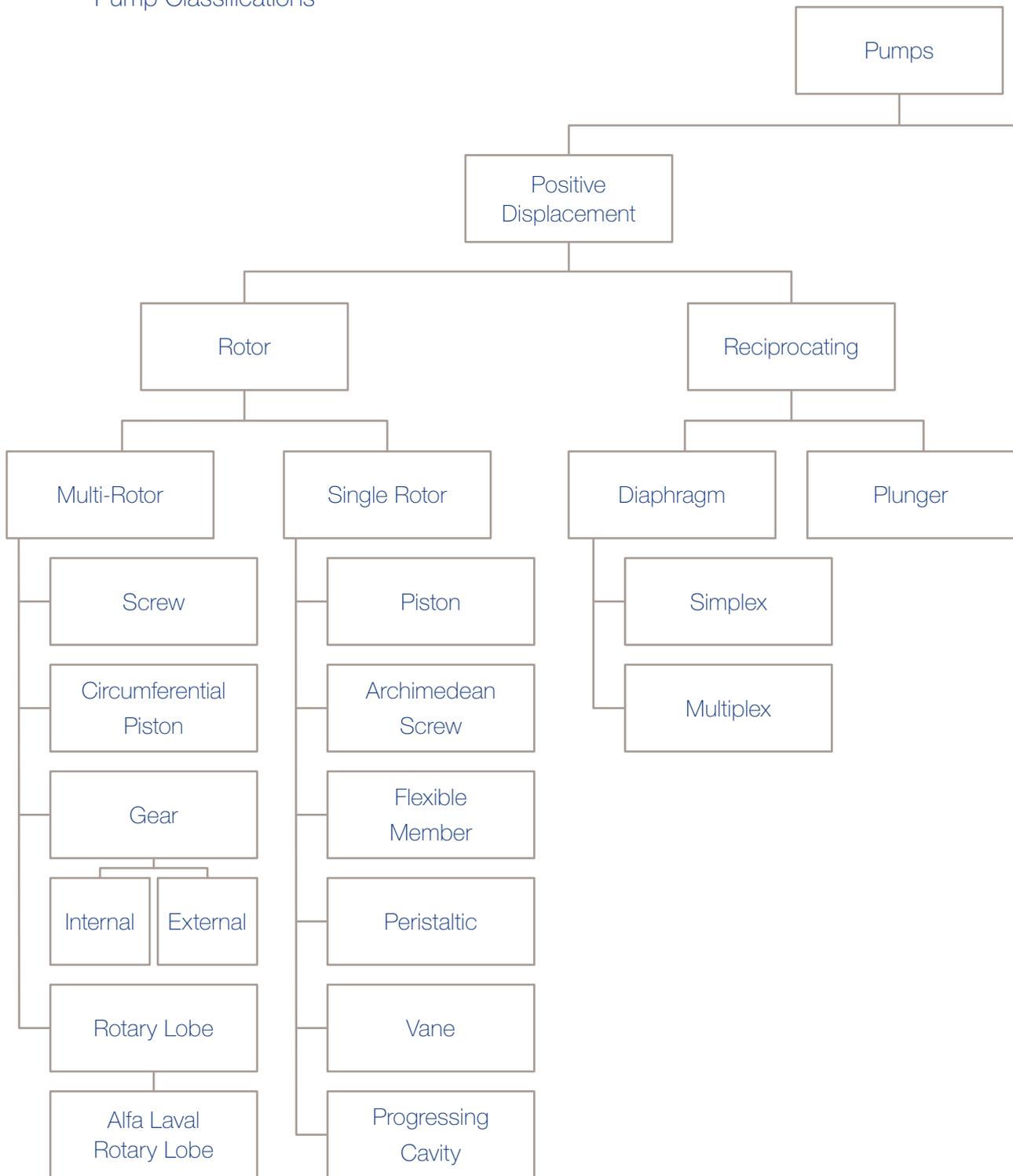
Fig. 1.1b Pump classifications

Rotodynamic

Multi-Stage

Single Stage

End Suction

Double Entry

Process

Rubber Lined

Submersible

General

Alfa Laval
Centrifugal and
Liquid Ring
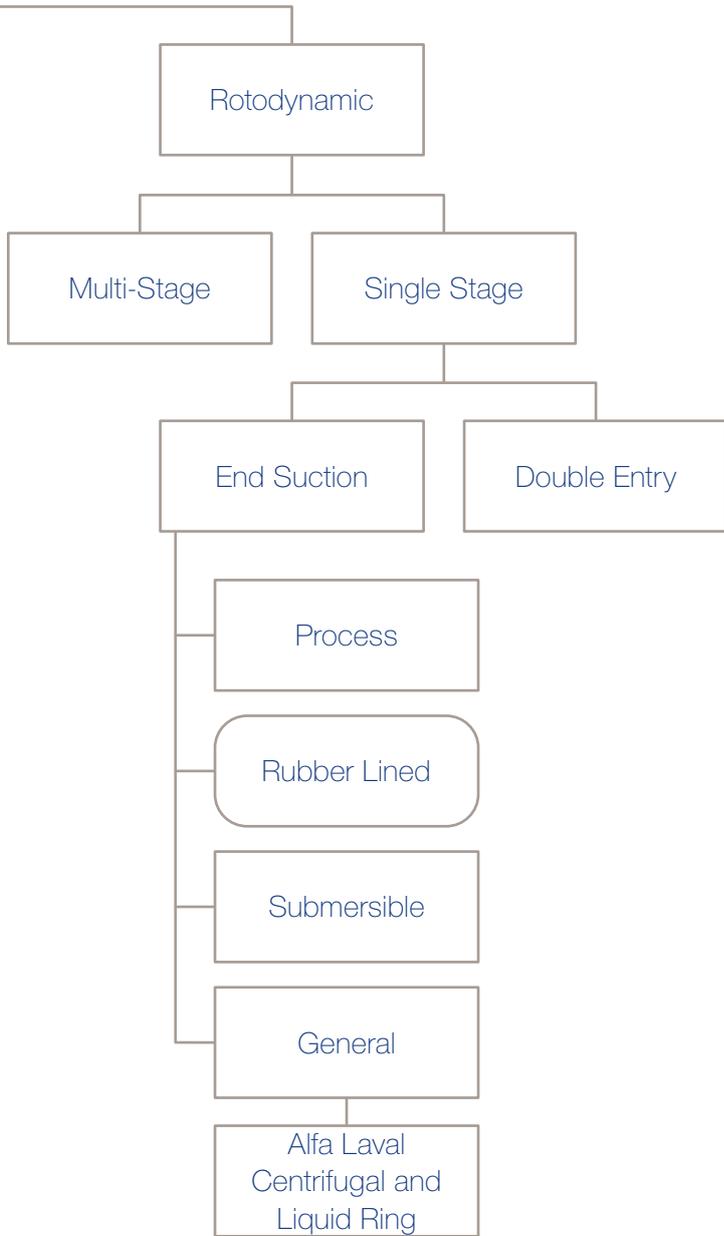
This chapter explains the terminology and theory of pumping applications, including explanations of rheology, flow characteristics, pressure and NPSH.

In order to select a pump two types of data are required:

- Product/Fluid data which includes viscosity, density/specific gravity, temperature, flow characteristics, vapour pressure and solids content

- Performance data which includes capacity or flow rate, and inlet/discharge pressure/head

Different fluids have varying characteristics and are usually pumped under different conditions. It is therefore very important to know all relevant product and performance data before selecting a pump.

2.0

Terminology and Theory

2.1 Product/Fluid Data

2.1.1 Rheology

The science of fluid flow is termed 'Rheology' and one of its most important aspects is viscosity which is defined below.

2.1.2 Viscosity

The viscosity of a fluid can be regarded as a measure of how resistive the fluid is to flow. It is comparable to the friction of solid bodies and causes a retarding force. This retarding force transforms the kinetic energy of the fluid into thermal energy.

The ease with which a fluid pours is an indication of its viscosity. For example, cold oil has a high viscosity and pours very slowly, whereas water has a relatively low viscosity and pours quite readily. High viscosity fluids require greater shearing forces than low viscosity fluids at a given shear rate. It follows therefore that viscosity affects the magnitude of energy loss in a flowing fluid.

Two basic viscosity parameters are commonly used, absolute (or dynamic) viscosity and kinematic viscosity.

Absolute (or Dynamic) Viscosity

This is a measure of how resistive the flow of a fluid is between two layers of fluid in motion. A value can be obtained directly from a rotational viscometer which measures the force needed to rotate a spindle in the fluid. The SI unit of absolute viscosity is mPas in the so-called MKS (metre, kilogram, second) system, while in the CGS (centimetres, grams, seconds) system this is expressed as 1 centipoise (cP) where 1 mPas = 1 cP. Water at 1 atmosphere and 20° C (68° F) has the value of 1 mPas or 1 cP. Absolute viscosity is usually designated by the symbol μ.

Kinematic Viscosity

This is a measure of how resistive the flow of a fluid is under the influence of gravity. Kinematic viscometers usually use the force of gravity to cause the fluid to flow through a calibrated orifice, while timing its flow. The SI unit of kinematic viscosity is (mm^2/s) in the so-called MKS (metre, kilogram, second) system, while in the CGS (centimetres, grams, seconds) system this is expressed as 1 centistoke (cSt), where 1 mm^2/s = 1 cSt. Water at 1 atmosphere and 20° C (68° F) has the value of 1 mm^2/s = 1 cSt. Kinematic viscosity is usually designated by the symbol ν.

Relationship between Absolute and Kinematic Viscosity

Absolute and Kinematic viscosity are related by:

$$\nu = \frac{\mu}{\rho}$$

Where ρ is the fluid density (see section 2.1.3)

In the CGS system this translates to:

$$\text{Kinematic Viscosity (cSt)} = \frac{\text{Absolute Viscosity (cP)}}{\text{Specific Gravity}}$$

or

Absolute Viscosity (cP) =
Kinematic Viscosity (cSt) x SG

A viscosity conversion table is included in section 14.3.10.
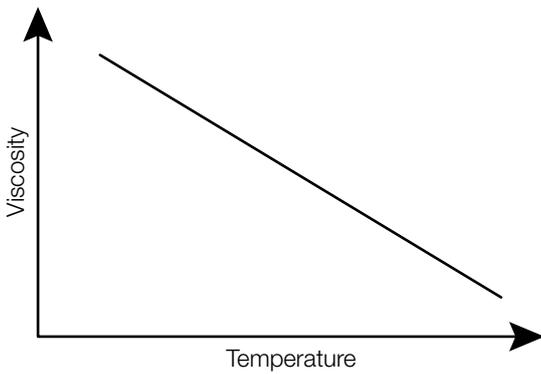
Fig. 2.1.2a Viscosity variation with temperature
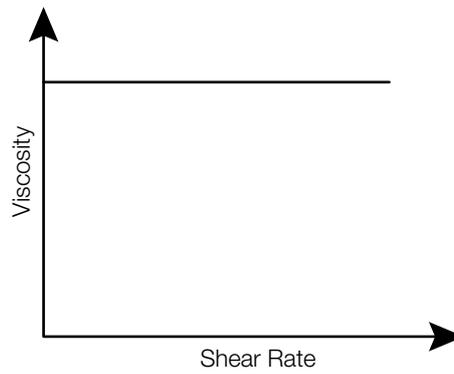
Fig. 2.1.2b Newtonian fluids

Viscosity Variation with Temperature

Temperature can have a significant effect on viscosity and a viscosity figure given for pump selection purposes without fluid temperature is often meaningless - viscosity should always be quoted at the pumping temperature (Fig 2.1.2a). Generally, viscosity falls with increasing temperature and more significantly, it increases with falling temperature. In a pumping system it can be advantageous to increase the temperature of a highly viscous fluid to ease flow.

Newtonian Fluids

In some fluids the viscosity is constant regardless of the shear forces applied to the layers of fluid. These fluids are named Newtonian fluids. At a constant temperature, the viscosity is constant with change in shear rate or agitation (Fig. 2.1.2b).

Typical fluids are:

- Water
- Beer
- Hydrocarbons
- Milk
- Mineral Oils
- Resins
- Syrups
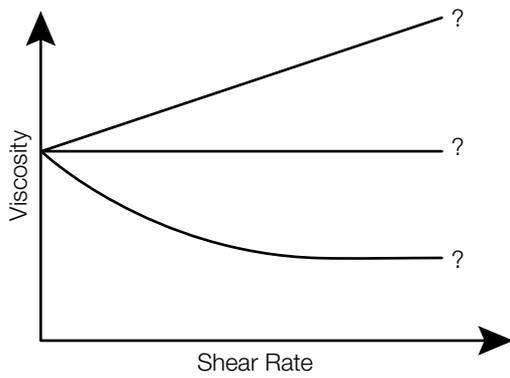
Fig. 2.1.2c Viscosity against Shear Rate

Non-Newtonian Fluids

Most empirical and test data for pumps and piping systems has been developed using Newtonian fluids across a wide range of viscosities. However, there are many fluids which do not follow this linear law, these fluids are named Non-Newtonian fluids (Fig. 2.1.2c).

When working with Non-Newtonian fluids, we use Effective Viscosity to represent the viscous characteristics of the fluid as though it was Newtonian at that given set of conditions (shear rate, temperature). This effective viscosity is then used in calculations, charts, graphs, and 'handbook' information.

It is not always obvious which type of viscous behaviour a fluid will exhibit, and consideration must be given to the shear rate that will exist in the pump under pumping conditions. It is not unusual to find the effective viscosity as little as 1% of the value measured by standard instruments.
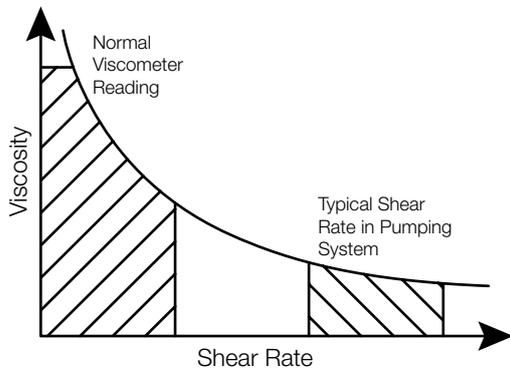
Fig. 2.1.2d Viscosity against Shear Rate
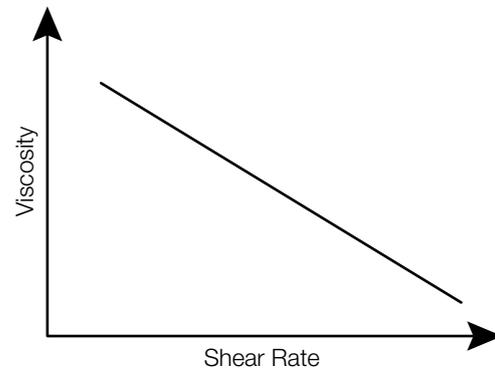
Fig. 2.1.2e Pseudoplastic fluids

Types of Non-Newtonian Fluids

There are a number of different types of Non-Newtonian fluids, each with different characteristics. Effective viscosity at set conditions will be different, depending on the fluid being pumped (Fig. 2.1.2d). This can be better understood by looking at the behaviour of viscous fluids with changes in shear rate as follows:

Pseudoplastic Fluids

Viscosity decreases as shear rate increases, but initial viscosity may be so high as to prevent start of flow in a normal pumping system (Fig. 2.1.2e).

Typical fluids are:

- Blood
- Emulsions
- Gums
- Lotions
- Soap
- Toothpaste
- Yeast
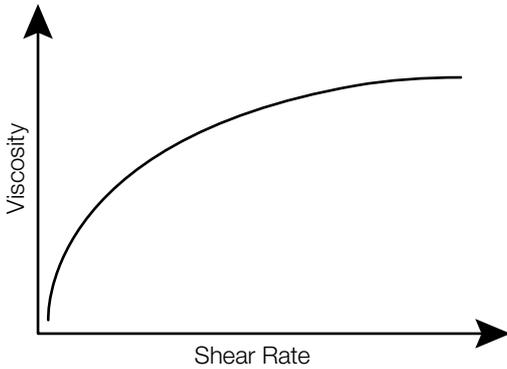
Fig. 2.1.2f Dilatant fluids

Fig. 2.1.2g Thixotropic fluids

Dilatant Fluids

Viscosity increases as shear rate increases (Fig. 2.1.2f).

Typical fluids are:

- Clay Slurries
- Paper Coatings

Thixotropic Fluids

Viscosity decreases with time under shear conditions. After shear ceases, the viscosity will return to its original value - the time for recovery will vary with different fluids (Fig. 2.1.2g).

Typical fluids are:

- Cosmetic Creams
- Dairy Creams
- Greases
- Stabilised Yoghurt
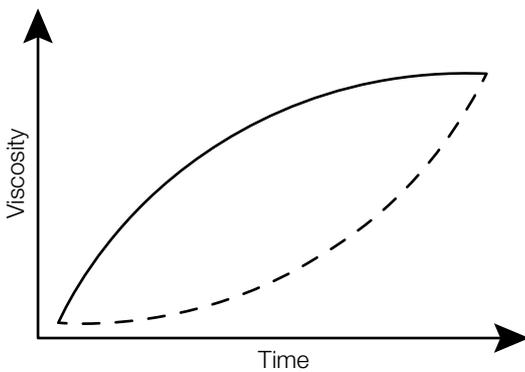
Fig. 2.1.2h Anti-thixotropic fluids

Fig. 2.1.2i Rheomalactic fluids

Anti-thixotropic Fluids

Viscosity increases with time under shear conditions. After shear ceases, the viscosity will return to its original value - the time for recovery will vary with different fluids (Fig. 2.1.2h). As the name suggests anti-thixotropic fluids have opposite rheological characteristics to thixotropic fluids.

Typical fluid is:

- Vanadium Pentoxide Solution

Rheomalactic Fluids

Viscosity decreases with time under shear conditions but does not recover (Fig. 2.1.2i). Fluid structure is irreversibly destroyed.

Typical fluids are:

- Natural Rubber Latex
- Natural Yoghurt
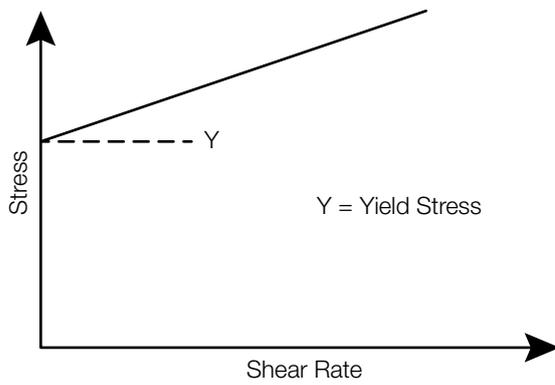
Fig. 2.1.2j Plastic fluids

Plastic Fluids

Need a certain applied force (or yield stress) to overcome 'solid-like structure', before flowing like a fluid (Fig. 2.1.2j).

Typical fluids are:

- Barium X-ray Meal
- Chocolate
- Tomato Ketchup

It should be noted that some fluids would have both thixotropic and pseudoplastic behaviour.
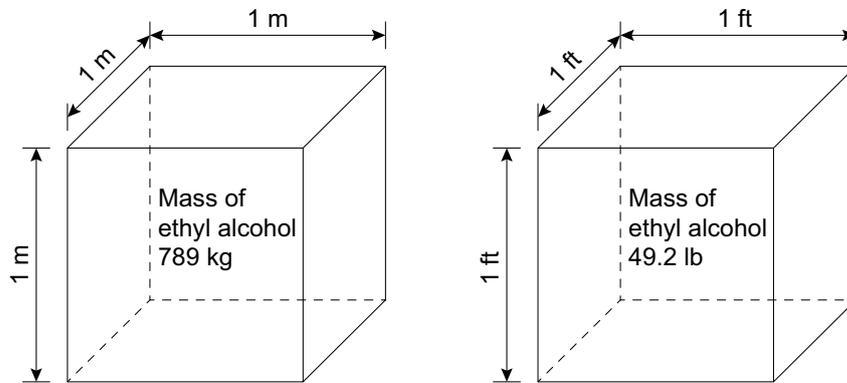
Fig. 2.1.3a Density

2.1.3 Density

The density of a fluid is its mass per unit of volume, usually expressed as kilograms per cubic metre (kg/m^3) or pounds per cubic foot (lb/ft^3) (Fig. 2.1.3a). Density is usually designated by the symbol ρ.

1 m^3 of ethyl alcohol has a mass of 789 kg

i.e., density = 789 kg/m^3

1 ft^3 of ethyl alcohol has a mass of 49.2 lb

i.e., density = 49.2 lb/ft^3

Density in gases varies considerably with pressure and temperature but can be regarded as constant in fluids.

2.1.4 Specific Weight

The specific weight of a fluid is its weight per unit volume and is usually designated by the symbol γ. It is related to density as follows:
$\gamma = \rho \times g$

Where g is gravity

The units of weight per unit volume are N/m^3 or lbf/ft^3

Standard gravity is as follows:
$g = 9.807 \ m/s^2$
$g = 32.174 \ ft/s^2$

The specific weight of water at 20° C (68° F) and 1 atmosphere is as follows:

$\gamma = 9790 \ N/m^3 = 62.4 \ lbf/ft^3$

Note:
Mass should not be confused with weight. Weight is the force produced from gravity acting on the mass.
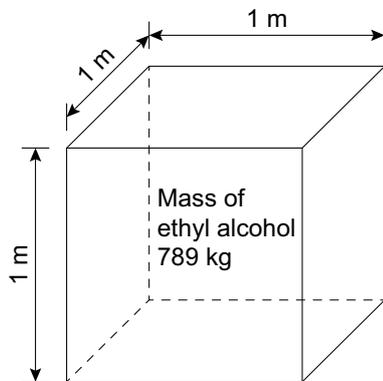
Fig. 2.1.5a Specific gravity
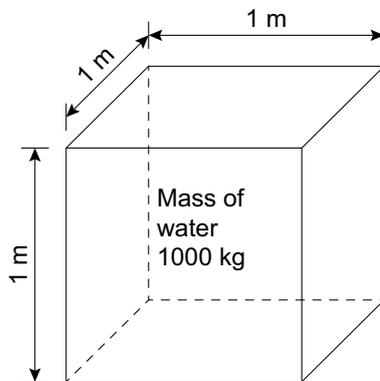
2.1.5 Specific Gravity

The specific gravity of a fluid is the ratio of its density to the density of water. As this is a ratio, it does not have any units of measure (Fig. 2.1.5a).

$1 \ m^3$ of ethyl alcohol has a mass of 789 kg
- its density is 789 kg/m^3
$1 \ m^3$ of water has a mass of 1000 kg
- its density is 1000 kg/m^3

Specific Gravity of ethyl alcohol is:
$$\frac{789 \ kg/m^3}{1000 \ kg/m^3} = 0.789$$

or

$1 \ ft^3$ of ethyl alcohol has a mass of 49.2 lb
- its density is 49.2 lb/ft^3
$1 \ ft^3$ of water has a mass of 62.4 lb
- its density is 62.4 lb/ft^3

Specific Gravity of ethyl alcohol is:
$$\frac{49.2 \ lb/ft^3}{62.4 \ lb/ft^3} = 0.789$$

This resultant figure is dimensionless, so the Specific Gravity (or SG) is 0.789.

2.1.6 Temperature

The temperature of the fluid at the pump inlet is usually of most concern as vapour pressure can have a significant effect on pump performance (see section 2.1.8). Other fluid properties such as viscosity and density can also be affected by temperature changes. Thus, a cooling of the product in the discharge line could have a significant effect on the pumping of a fluid.

The temperature of a fluid can also have a significant effect on the selection of any elastomeric materials used.

A temperature conversion table is given in section 14.3.11.

Temperature is a measure of the internal energy level in a fluid, usually expressed in units of degrees Centigrade (°C) or degrees Fahrenheit (°F).

2.1.7 Flow Characteristics

When considering a fluid flowing in a pipework system it is important to be able to determine the type of flow. The connection between the velocity and the capacity of a fluid (similar to water) in different tube sizes is shown in table 14.6.

Under some conditions the fluid will appear to flow as layers in a smooth and regular manner. This can be illustrated by opening a water tap slowly until the flow is smooth and steady. This type of flow is called laminar flow. If the water tap is opened wider, allowing the velocity of flow to increase, a point will be reached whereby the stream of water is no longer smooth and regular but appears to be moving in a chaotic manner. This type of flow is called turbulent flow. The type of flow is indicated by the Reynolds number.

Velocity

Velocity is the distance a fluid moves per unit of time and is given by equation as follows:

In dimensionally consistent SI units

Velocity V $= \dfrac{Q}{A}$

Where:

V = Fluid Velocity (m/s)
Q = Capacity (m^3/s)
A = Tube Cross Sectional Area (m^2)

Other convenient forms of this equation are:

Velocity V $= \dfrac{Q \times 353.6}{D^2}$

Where:

V = Fluid Velocity (m/s)
Q = Capacity (m^3/h)
D = Tube Diameter (mm)

or

Velocity V $= \dfrac{Q \times 0.409}{D^2}$

Where:

V = Fluid Velocity (ft/s)
Q = Capacity (US gal/min)
D = Tube Diameter (in)

or

Velocity V $= \dfrac{Q \times 0.489}{D^2}$

Where:

V = Fluid Velocity (ft/s)
Q = Capacity (UK gal/min)
D = Tube Diameter (in)

Fluid velocity can be of great importance especially when pumping slurries and fluids containing solids. In these instances, a certain velocity may be required to prevent solids from settling in the pipework, which could result in blockages and changes in system pressure as the actual internal diameter of the pipe is effectively decreased, which could impact on pump performance.
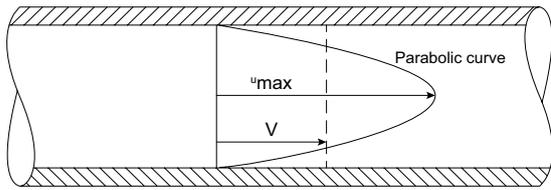
umax = Maximum Velocity
V = Velocity
Fig. 2.1.7a Laminar flow
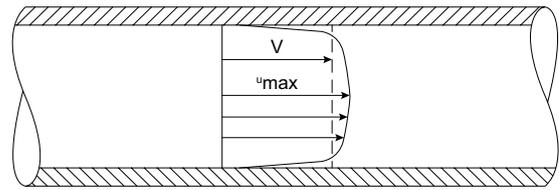
umax = Maximum Velocity
V = Velocity
Fig. 2.1.7b Turbulent flow

Laminar Flow

This is sometimes known as streamline, viscous or steady flow. The fluid moves through the pipe in concentric layers with the maximum velocity in the centre of the pipe, decreasing to zero at the pipe wall.

The velocity profile is parabolic, the gradient of which depends upon the viscosity of the fluid for a set flowrate as shown in Fig. 2.1.7a.

Turbulent Flow

This is sometimes known as unsteady flow with considerable mixing taking place across the pipe cross section. The velocity profile is more flattened than in laminar flow but remains fairly constant across the section as shown in Fig. 2.1.7b. Turbulent flow generally appears at relatively high velocities and/or relatively low viscosities.

Transitional Flow

Between laminar and turbulent flow there is an area referred to as transitional flow where conditions are unstable and have a blend of each characteristic.

This is a ratio of inertia forces to viscous forces, and as such, a useful value for determining whether flow will be laminar or turbulent.

Reynolds Number (Re)

Reynolds number for pipe flow is given by equation as follows:

In dimensionally consistent SI units

Re $= \dfrac{D \times V \times \rho}{\mu}$

Where:

D = Tube Diameter (m)
V = Fluid Velocity (m/s)
ρ = Density (kg/m^3)
μ = Absolute Viscosity (Pas)

Other convenient forms of this equation are:

Re $= \dfrac{D \times V \times \rho}{\mu}$

Where:

D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
ρ = Density (kg/m^3)
μ = Absolute Viscosity (cP)

or

Re $= \dfrac{21230 \times Q}{D \times \mu}$

Where:

D = Tube Diameter (mm)
Q = Capacity (l/min)
μ = Absolute Viscosity (cP)

or

Re $= \dfrac{3162 \times Q}{D \times \nu}$

Where:

D = Tube Diameter (in)
Q = Capacity (US gal/min)
ν = Kinematic Viscosity (cSt)

or

Re $= \dfrac{3800 \times Q}{D \times \nu}$

Where:

D = Tube Diameter (in)
Q = Capacity (UK gal/min)
ν = Kinematic Viscosity (cSt)

Since Reynolds number is a ratio of two forces, it has no units. For a given set of flow conditions, the Reynolds number will not vary when using different units. It is important to use the same set of units, such as show on previous page, when calculating Reynolds numbers.

Re less than 2300   -   Laminar Flow
                              (viscous force dominates
                              - high system losses)

Re in range 2300 to 4000 -   Transitional Flow
                              (critically balanced forces)

Re greater than 4000   -   Turbulent Flow
                              (inertia force dominates
                              - low system losses)

Where transitional flow occurs, frictional loss calculations should be carried out for both laminar and turbulent conditions, and the highest resulting loss used in subsequent system calculations.
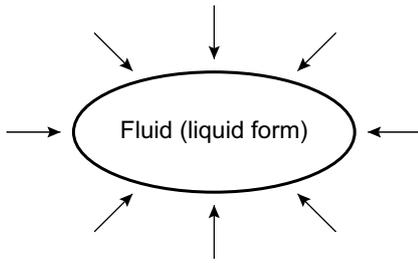
Pvp = Vapour pressure
(external pressure required to maintain as a fluid)
Fig. 2.1.8a Vapour pressure

2.1.8 Vapour Pressure

Fluids will evaporate unless prevented from doing so by external pressure (Fig. 2.1.8a). The vapour pressure of a fluid is the pressure (at a given temperature) at which a fluid will change to a vapour and is expressed as absolute pressure (bar a or PSIA) - see section 2.2.2. Each fluid has its own vapour pressure/temperature relationship. In pump sizing, vapour pressure can be a key factor in checking the Net Positive Suction Head (NPSH) available from the system (see section 2.2.4).

Temperature	Vapour pressure (bar)
0° C (32° F)	0.006 bar a (0.087 PSIA)
20° C (68° F)	0.023 bar a (0.334 PSIA)
100° C (212° F)	1.013 bar a (14.7 PSIA)

Water will boil (vaporise) at a temperature of:

- 0° C (32° F) if Pvp = 0.006 bar a (0.087 PSIA)

- 20° C (68° F) if Pvp = 0.023 bar a (0.334 PSIA)

- 100° C (212° F) if Pvp = 1.013 bar a (14.7 PSIA) (atmospheric conditions at sea level)

In general terms Pvp:

- Is dependent upon the type of fluid

- Increases at higher temperature

- Is of great importance to pump inlet conditions

- Should be determined from relevant tables

The Pvp for water at various temperatures is shown in section 14.4.

2.1.9 Fluids Containing Solids

It is important to know if a fluid contains any particulate matter and if so, the size and concentration. Special attention should be given regarding any abrasive solids with respect to pump type and construction, operating speed, and shaft seals.

Size of solids is also important, as when pumping large particles, the pump inlet should be large enough for solids to enter the pump without 'bridging' the pump inlet. Also, the pump should be sized so the cavity created in the pump chamber by the pump elements is of sufficient size to allow satisfactory pump operation.

Concentration is normally expressed as a percentage by weight (W/W) or volume (V/V) or a combination of both weight and volume (W/V).
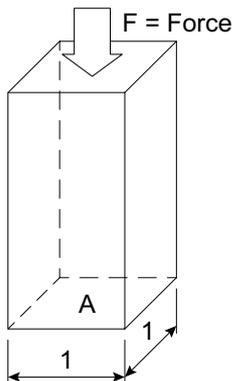
Fig. 2.2.2a Pressure

2.2 Performance Data

2.2.1 Capacity (Flow Rate)

The capacity (or flow rate) is the volume of fluid or mass that passes a certain area per time unit. This is usually a known value dependent on the actual process. For fluids the most common units of capacity are litres per hour (l/h), cubic metres per hour (m^3/h) and UK or US gallons per minute (gal/min). For mass the most common units of capacity are kilogram per hour (kg/h), tonne per hour (t/h) and pounds per hour (lb/h).

2.2.2 Pressure

Pressure is defined as force per unit area:
$$P = \frac{F}{A}$$

Where F is the force perpendicular to a surface and A is the area of the surface (Fig. 2.2.2a).

In the SI system the standard unit of force is the Newton (N) and area is given in square metres (m^2). Pressure is expressed in units of Newtons per square metre (N/m^2). This derived unit is called the Pascal (Pa). In practice Pascals are rarely used and the most common units of force are bar, pounds per square inch (lb/in^2) or PSI, and kilogram per square centimetre (kg/cm^2).

Conversion factors between units of pressure are given in section 14.3.5.

Different Types of Pressure

For calculations involving fluid pressures, the measurements must be relative to some reference pressure. Normally the reference is that of the atmosphere and the resulting measured pressure is called gauge pressure. Pressure measured relative to a perfect vacuum is called 'absolute pressure'.

Atmospheric Pressure

The actual magnitude of the atmospheric pressure varies with location and with climatic conditions. The range of normal variation of atmospheric pressure near the earth's surface is approximately 0.95 to 1.05 bar absolute (bar a) or 13.78 to 15.23 PSI gauge (PSIG). At sea level the standard atmospheric pressure is 1.013 bar a or 14.7 PSI absolute (bar a or PSIA).

Gauge Pressure

Using atmospheric pressure as a zero reference, gauge pressure is the pressure within the gauge that exceeds the surrounding atmospheric pressure. It is a measure of the force per unit area exerted by a fluid, commonly indicated in units of bar g (bar gauge) or PSIG (PSI gauge).

Absolute Pressure

Is the total pressure exerted by a fluid. It equals atmospheric pressure plus gauge pressure, indicated in units of bar a (bar absolute) or PSIA (PSI absolute).

Absolute Pressure =
Gauge Pressure + Atmospheric Pressure

Vacuum

This is a commonly used term to describe pressure in a pumping system below normal atmospheric pressure. This is a measure of the difference between the measured pressure and atmospheric pressure expressed in units of mercury (Hg) or units of PSIA.

14.7 PSIA = 760 mm Hg (30 in Hg)

0 PSIA = 0 mm Hg (0 in Hg)

Inlet (Suction) Pressure

This is the pressure at which the fluid is entering the pump. The reading should be taken whilst the pump is running and as close to the pump inlet as possible. This is expressed in units of absolute bar a (PSIA) or gauge bar g (PSIG) depending upon the inlet conditions.

Outlet (Discharge) Pressure

This is the pressure at which the fluid leaves the pump. Again, this reading should be taken whilst the pump is running and as close to the pump outlet as possible. The reading is expressed in units of gauge bar (PSIG).

Differential Pressure

This is the difference between the inlet and outlet pressures. For inlet pressures above atmospheric pressure the differential pressure is obtained by subtracting the inlet pressure from the outlet pressure. For inlet pressures below atmospheric pressure the differential pressure is obtained by adding the inlet pressure to the outlet pressure. It is therefore the total pressure reading and is the pressure against which the pump will have to operate. Power requirements are to be calculated on the basis of differential pressure (Fig. 2.2.2b).

Example: Inlet Pressure above Atmospheric Pressure

Outlet

4 bar g (58 PSIG)   5.013 bar a (72.7 PSIA)

0 bar g (0 PSIG)   1.013 bar a (14.7 PSIA)

0 bar a (0 PSIA)

Inlet

1.5 bar g (21.75 PSIG)   2.513 bar a (36.45 PSIA)

0 bar g (0 PSIG)   1.013 bar a (14.7 PSIA)

Differential

Differential = 4 - 1.5 = 2.5 bar
or
= 58 - 21.75 = 36.25 PSI
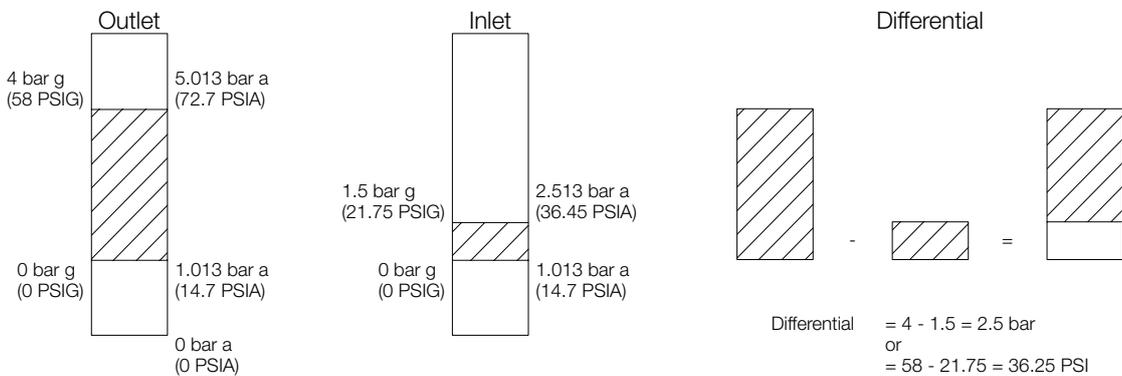
Example: Inlet Pressure below Atmospheric Pressure

Outlet

4 bar g (58 PSIG)   5.013 bar a (72.7 PSIA)

0 bar g (0 PSIG)   1.013 bar a (14.7 PSIA)

0 bar a (0 PSIA)

Inlet

1.013 bar a (14.7 PSIA)

0 bar g (0 PSIG)   0.5 bar a (7.25 PSIA)

0 bar a (0 PSIA)

Differential

Differential = 4 + (1.013 - 0.5) = 4.513 bar
or
= 58 + (14.7 -7.25) = 65.45 PSI
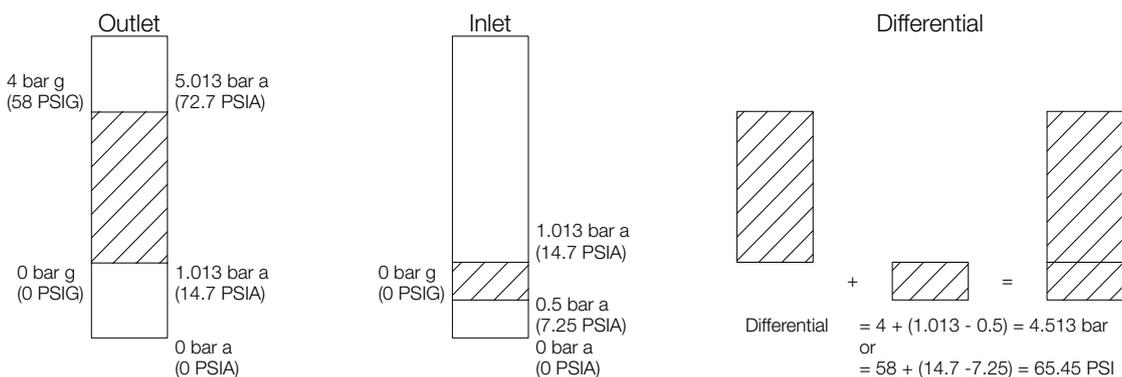
Fig. 2.2.2b Differential pressure
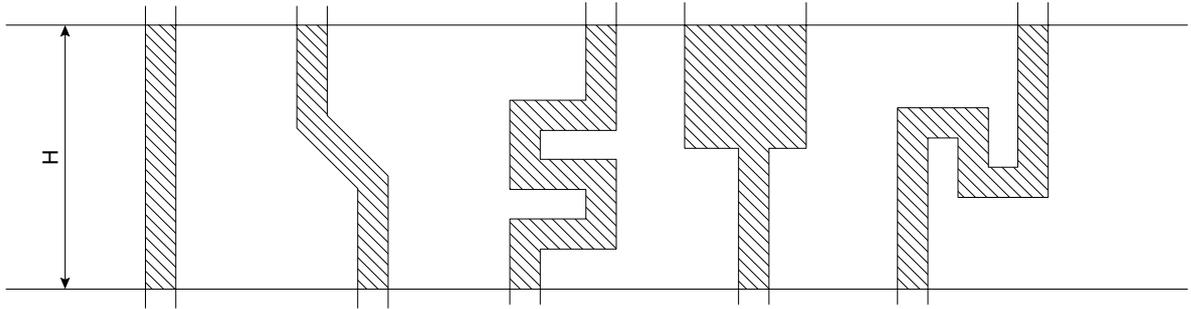
Fig. 2.2.2c Relationship of pressure to elevation

The Relationship Between Pressure and Elevation

In a static fluid (a body of fluid at rest) the pressure difference between any two points is in direct proportion only to the vertical distance between the points. The same vertical height will give the same pressure regardless of the pipe configuration in between (Fig. 2.2.2c).

The relationship of elevation equivalent to pressure is commonly referred to as 'head'.

This pressure difference is due to the weight of a 'column' of fluid and can be calculated as follows:

In dimensionally consistent SI units

Static Pressure (P) = ρ x g x h

Where:

P = Pressure/Head (Pa)
ρ = Fluid Density (kg/m^3)
g = Gravity (m/s^2)
h = Height of Fluid (m)

Other convenient forms of this equation are:

Static Pressure (P) = $\dfrac{h \times SG}{10}$

Where:

P = Pressure/Head (Pa)
SG = Specific Gravity (bar)
h = Height of Fluid (m)

or

Static Pressure (P) = $\dfrac{h \times SG}{2.31}$

Where:

P = Pressure/Head (Pa)
SG = Specific Gravity (PSI)
h = Height of Fluid (ft)
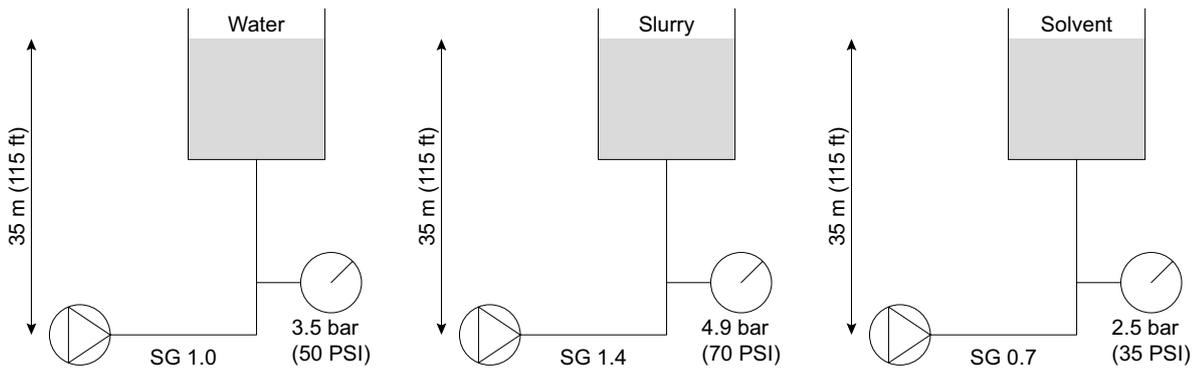
Fig. 2.2.2d Relationship of elevation to pressure

A pump capable of delivering 35 m (115 ft) head will produce different pressures for fluids of differing specific gravities (Fig. 2.2.2d).

A pump capable of delivering 3.5 bar (50 PSI) pressure will develop different amounts of head for fluids of differing specific gravities (Fig. 2.2.2e).

The following are terms commonly used to express different conditions in a pumping system which can be expressed as pressure units (bar or PSI) or head units (m or ft).

Flooded Suction

This term is generally used to describe a positive inlet pressure/head, whereby fluid will readily flow into the pump inlet at sufficient pressure to avoid cavitation (see section 2.2.3).
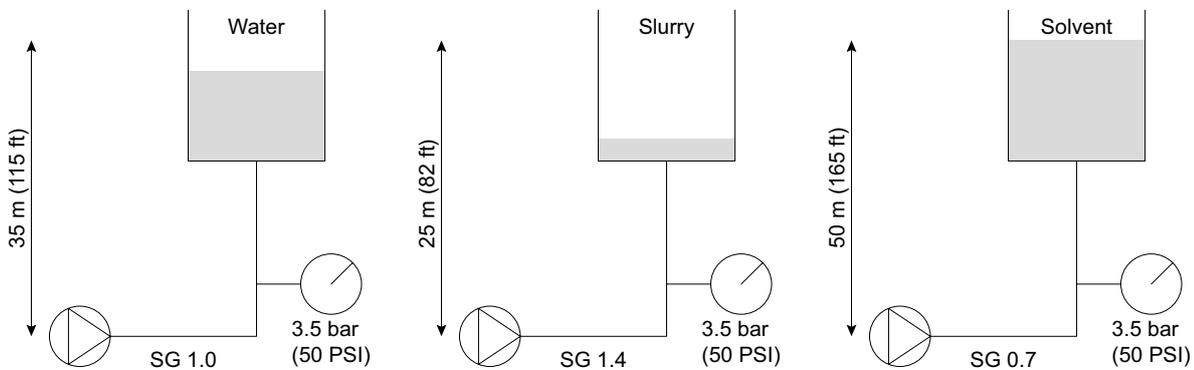
Fig. 2.2.2e Relationship of elevation to pressure

Static Head

The static head is a difference in fluid levels.

Static Suction Head

This is the difference in height between the fluid level and the centre line of the pump inlet on the inlet side of the pump.

Static Discharge Head

This is the difference in height between the fluid level and the centre line of the pump inlet on the discharge side of the pump.

Total Static Head

The total static head of a system is the difference in height between the static discharge head and the static suction head.

Friction Head

This is the pressure drop on both inlet and discharge sides of the pump due to frictional losses in fluid flow.

Dynamic Head

This is the energy required to set the fluid in motion and to overcome any resistance to that motion.
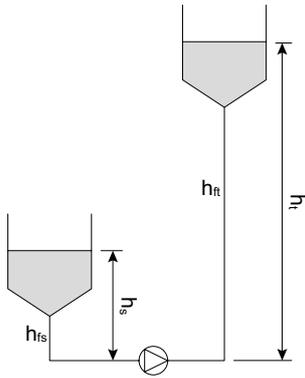
Fig. 2.2.2f Flooded suction and open discharge tanks
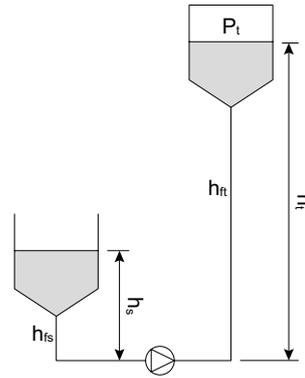
Fig. 2.2.2g Flooded suction and closed discharge tanks

Total Suction Head

The total suction head is the static suction head less the dynamic head. Where the static head is negative, or where the dynamic head is greater than the static head, this implies the fluid level will be below the centre line of the pump inlet (i.e., suction lift).

Total Discharge Head

The total discharge head is the sum of the static discharge and dynamic heads.

Total Head

Total head is the total pressure difference between the total discharge head and the total suction head of the pump. The head is often a known value. It can be calculated by means of different formulas if the installation conditions are specified.

Total Head H = $H_t - (\pm H_s)$

Total Discharge Head H_t = $h_t + h_{ft} + P_t$

Total Suction Head H_s = $h_s - h_{fs} + (\pm P_s)$

Where:

H = Total head

H_s = Total suction head

H_t = Total discharge head

h_s = Static suction head

h_t = Static discharge head

h_{fs} = Pressure drop in suction line

h_{ft} = Pressure drop in discharge line

P_s = Vacuum or pressure in a tank on suction side

P_t = Pressure in a tank on discharge side

In general terms:

$p > 0$ for pressure

$p < 0$ for vacuum

$p = 0$ for open tank

$h_s > 0$ for flooded suction

$h_s < 0$ for suction lift
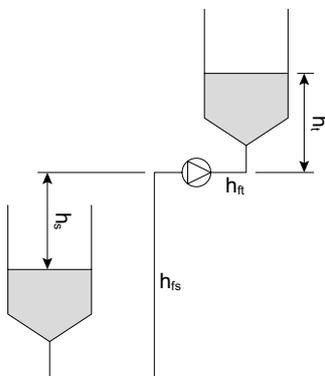
Fig. 2.2.2h Suction lift and open discharge tanks
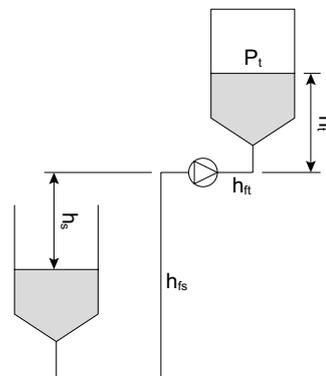
Fig. 2.2.2i Suction lift and closed discharge tanks

Pressure Drop

Manufacturers of processing equipment, heat exchangers, static mixers etc., usually have data available for pressure drop. These losses are affected by fluid velocity, viscosity, tube diameter, internal surface finish of tube and tube length.

The different losses and consequently the total pressure drop in the process are, if necessary, determined in practice by converting the losses into equivalent straight length of tube which can then be used in subsequent system calculations.

For calculations on water like viscosity fluids, the pressure drop can be determined referring to the Pressure Drop Curve (see section 14.5) as shown in Example 1. For higher viscosity fluids, a viscosity correction factor is applied to the tube fittings by multiplying the resultant equivalent tube length by the figures shown in table 2.2.2a below - see Example 2.

> Pressure drop is the result of frictional losses in pipework, fittings and other process equipment etc.

Viscosity - cP	1 - 100	101 - 2000	2001 - 20,000	20,001 - 100,000
Correction Factor	1.0	0.75	0.5	0.25

Table 2.2.2a
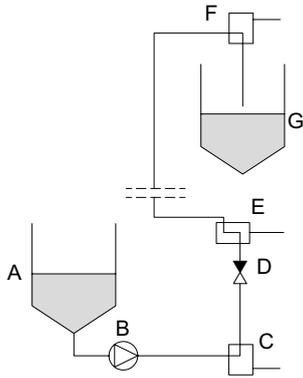
Fig. 2.2.2j Example

Example 1 Process:

Pumping milk from tank A to tank G
Q = 8 m³/h (35 US gal/min) (Fig. 2.2.2j).

Tubes, valves, and fittings:

A: Tank outlet dia. 63.5 mm (2.5 in)

A-B: 4 m (13 ft) tube dia. 63.5 mm (2.5 in)

A-B: 1 off bend 90° dia. 63.5 mm (2.5 in)

B-C: 20 m (66 ft) tube dia. 51 mm (2 in).

C: Unique SSV standard ISO 51 mm

C-E: 15 m (49 ft) tube dia. 51 mm (2 in)

B-E: 3 off bend 90° dia. 51 mm (2 in)

D: Non-return valve type LKC-2, 51 mm (2 in)

E: Unique SSV standard ISO 51 mm

E-F: 46 m (151 ft) tube dia. 38 mm (1.5 in)

E-F: 4 off bend 90° dia. 38 mm (1.5 in)

F: Seat valve type SRC-W-38-21-100

The pressure drop through the tubes, valves and fittings is determined as equivalent tube length, so that the total pressure drop can be calculated.

The conversion into equivalent tube length is carried out by reference to chapter 14.7. This results in the following equivalent tube length for the different equipment as shown in the following tables:

Equipment		Equivalent ISO Tube Length (m)		
		38 mm	51 mm	63.5 mm
A	Tank outlet			1 (estimated)
A-B	Tube			4
A-B	Bend 90°			1 x 1
B-C	Tube		20	
C-E	Tube		15	
C-E	Unique SSV standard ISO		11	
B-E	Bend 90°		3 x 1	
D	LKC-2 non-return valve		12	
E	Unique SSV standard ISO		11	
E-F	Tube	46		
E-F	Bend 90°	4 x 1		
F	Unique SSV standard ISO	6		
Total		56	72	6

Table 2.2.2b

Equipment		Equivalent ISO Tube Length (ft)		
		1.5 in	2 in	2.5 in
A	Tank outlet			3 (estimated)
A-B	Tube			13
A-B	Bend°			1 x 3
B-C	Tube		66	
C-E	Tube		49	
C-E	Unique SSV standard ISO		36	
B-E	Bend°		3 x 3	
D	LKC-2 non-return valve		39	
E	Unique SSV standard ISO		36	
E-F	Tube	151		
E-F	Bend°	4 x 3		
F	Unique SSV standard ISO	20		
Total		183	235	19

Table 2.2.2c

As viewed from the prior tables the pressure drop through the different equipment corresponds to the following equivalent tube length.

38 mm (1.5 in) tube: Length = 56 m (184 ft)

51 mm (2 in) tube: Length = 72 m (236 ft)

63.5 mm (2.5 in) tube: Length = 6 m (20 ft)

The pressure drop through 100 m of tube for sizes 38 mm, 51 mm and 63.5 mm is determined by means of the following curve, also shown in 14.5 (Fig. 2.2.2k).

Pressure drop (m) Pressure drop in 100 m ISO/DIN tube (water at 20º C)

A = 25 mm   D = DN40   G = 63.5 mm   J = DN80   M = DN125
B = DN25   E = 61 mm   H = DN65   K = 101.6 mm N = DN150
C = 38 mm   F = DN50   I = 76 mm   L = DN100

Note: A, C, E, G, I and K refer to ISO Tube - B, D, F, H, J, L, M and N refer to DIN Tube
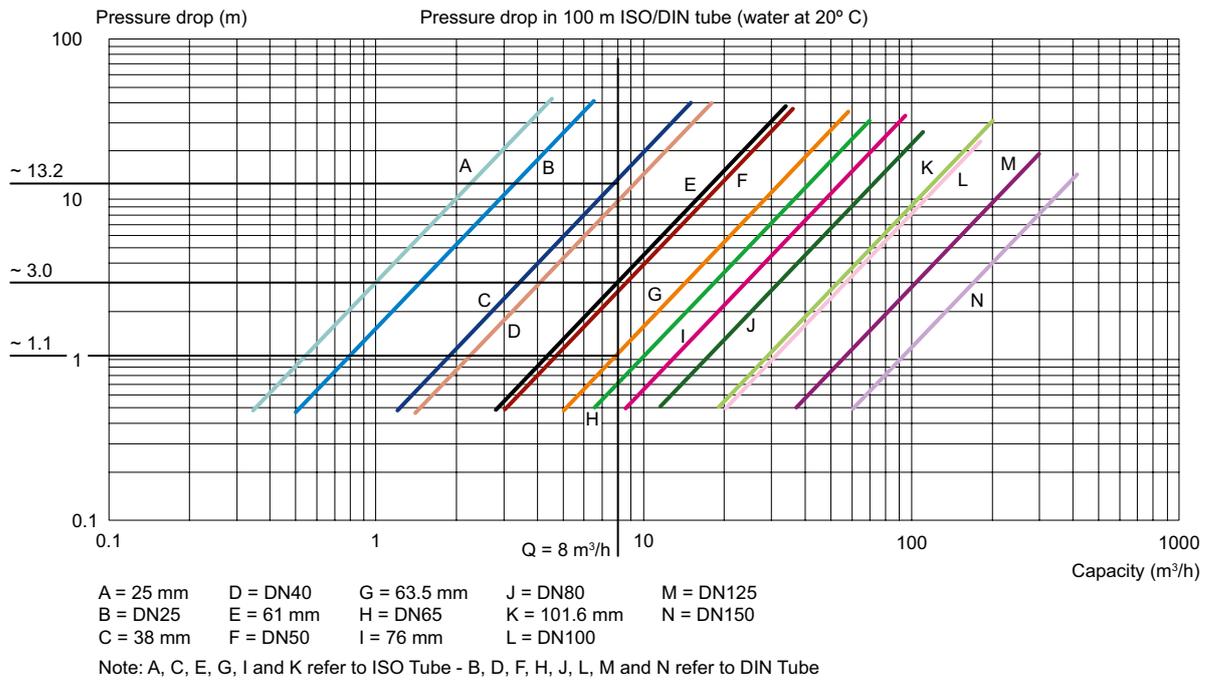
Fig. 2.2.2k Pressure drop curve

The total pressure drop ΔH in the process is consequently calculated as follows:

or

38 mm: $\Delta H = \dfrac{56 \times 13.2}{100} = 7.39$ m

1.5 in: $\Delta H = \dfrac{183 \times 43}{328} = 24.0$ ft

51 mm: $\Delta H = \dfrac{72 \times 3.0}{100} = 2.16$ m

2 in: $\Delta H = \dfrac{235 \times 10}{328} = 7.2$ ft

63.5 mm: $\Delta H = \dfrac{6 \times 1.1}{100} = 0.07$ m

2.5 in: $\Delta H = \dfrac{19 \times 4}{328} = 0.2$ ft

ΔH = 7.39 + 2.16 + 0.07 = 9.62 m ≈ 9.6 m (≈ 1 bar)

ΔH = 24.0 + 7.2 + 0.2 = 30.7 ft ≈ 31.4 ft (≈ 14 PSI)

Example 2 Process:

Pumping glucose with a viscosity of 5000 cP from a flooded suction through discharge pipeline as follows.

Tubes, valves and fittings:

30 m (98 ft) tube dia. 51 mm (2 in)

20 m (66 ft) tube dia. 76 mm (3 in)

2 off Non-return valves 51 mm (2 in)

6 off Bend 90° dia. 51 mm (2 in)

4 off Bend 90° dia. 76 mm (3 in)

3 off Tee (out through side port) 51 mm (2 in)

The pressure drop through the tubes, valves and fittings is determined as equivalent tube length so that the total pressure drop can be calculated.

For the pipe fittings the conversion into equivalent tube length is carried out by reference to tables 14.7. This results in the following equivalent tube length for the different fittings as shown below:

Fittings	Equivalent ISO Tube Length (m)	
	51 mm	76 mm
Non-return valve	2 x 12	
Bend 90°	6 x 1	
Bend 90°		4 x 1
Tee	3 x 3	
Total	39	4

Table 2.2.2d

Fittings	Equivalent ISO Tube Length (ft)	
	2 in	3 in
Non-return valve	2 x 39	
Bend 90°	6 x 3	
Bend 90°		4 x 3
Tee	3 x 10	
Total	126	12

Table 2.2.2e

As viewed from the prior tables the pressure drop through the different fittings corresponds to the following equivalent tube length.

Tube dia. 51 mm (2 in): Length = 39 m (128 ft)

Tube dia. 76 mm (3 in): Length = 4 m (13 ft)

Applying the viscosity correction factor from table 2.2.2a for 5000 cP the equivalent tube length is now:

Tube dia. 51 mm (2 in):
Length = 39 m (126 ft) x 0.5 = 19.5 m (64 ft)

Tube dia. 76 mm (3 in):
Length = 4 m (12 ft) x 0.5 = 2 m (7 ft)

These figures of 19.5 m (64 ft) and 2 m (7 ft) would be added to the straight tube lengths given as shown above, and subsequently used in calculating the discharge pressure at the flow rate required.

Tube dia. 51 mm (2 in):
30 m (98 ft) + 19.5 m (63 ft) = 49.5 m (162 ft)
+
Tube dia. 76 mm (3 in):
20 m (66 ft) + 2 m (7 ft) = 22 m (72 ft)

The friction losses in a pipework system are dependent upon the type of flow characteristic that is taking place. The Reynolds number (Re) is used to determine the flow characteristic, see section 2.1.7.

Friction Loss Calculations

Since laminar flow is uniform and predictable it is the only flow regime in which the friction losses can be calculated using purely mathematical equations. In the case of turbulent flow, mathematical equations are used, but these are multiplied by a co-efficient that is normally determined by experimental methods. This co-efficient is known as the Darcy friction factor (f_D).

The Miller equation given below can be used to determine the pressure loss due to friction for both laminar and turbulent flow in a given length of pipe (L).

In dimensionally consistent SI units

$$Pf = \frac{f_D \times L \times \rho \times V^2}{D \times 2}$$

Where:

P_f = Pressure Loss due to Friction (Pa)
f_D = Darcy Friction Factor
L = Tube Length (m)
D = Tube Diameter (m)
V = Fluid Velocity (m/s)
ρ = Fluid Density (kg/m³)

Other convenient forms of this equation are:

$$Pf = \frac{5 \times SG \times f_D \times L \times V^2}{D}$$

Where:

Pf = Pressure Loss due to Friction (bar)
f_D = Darcy Friction Factor
L = Tube Length (m)
D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
SG = Specific Gravity

or

$$Pf = \frac{0.0823 \times SG \times f_D \times L \times V^2}{D}$$

Where:

Pf = Pressure Loss due to Friction (PSI)
f_D = Darcy Friction Factor
L = Tube Length (ft)
D = Tube Diameter (in)
V = Fluid Velocity (ft/s)
SG = Specific Gravity

For laminar flow, the Darcy friction factor (f_D) can be calculated directly from the equation:

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

For turbulent flow, the Darcy friction factor (f_D) has to be determined by reference to the Moody diagram (see section 14.8). It is first necessary to calculate the relative roughness designated by the symbol E.

Where:

$$E = \frac{k}{D}$$

k = relative roughness which is the average heights of the pipe internal surface peaks (mm)

D = internal pipe diameter (mm)

The relative roughness of pipes varies with diameter, type of material used and age of the pipe. It is usual to simplify this by using a relative roughness (k) of 0.045 mm, which is the absolute roughness of clean commercial steel or wrought iron pipes as given by Moody.

2.2.3 Cavitation

Cavitation is an undesirable vacuous space in the inlet port of the pump normally occupied by fluid. The lowest pressure point in a pump occurs at the pump inlet - 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.

If cavitation occurs this will result in loss of pump efficiency and noisy operation. The life of a pump can be shortened through mechanical damage, increased corrosion, and erosion when cavitation is present.

The term cavitation is derived from the word cavity, meaning a hollow space.
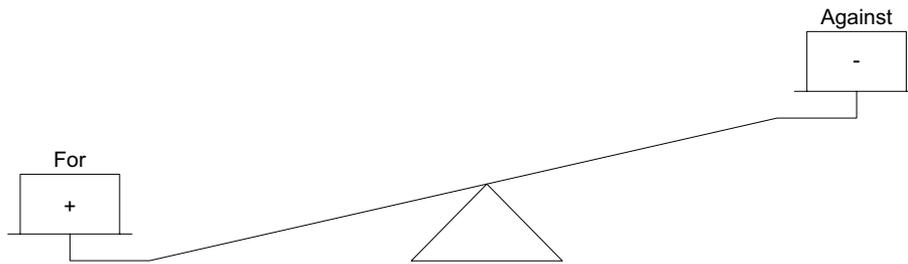
Fig. 2.2.4a NPSH balance

When sizing pumps on highly viscous fluids care must be taken not to select too higher pump speed so as to allow sufficient fluid to enter the pump and ensure satisfactory operation.

For all pump application problems, cavitation is the most commonly encountered. It occurs with all types of pumps, centrifugal, rotary, or reciprocating. When found, excessive pump speed and/or adverse suction conditions will probably be the cause and reducing pump speed and/or rectifying the suction condition will usually eliminate this problem.

Cavitation should be avoided at all costs.

2.2.4 Net Positive Suction Head (NPSH)

In addition to the total head, capacity, power and efficiency requirements, the condition at the inlet of a pump is critical. The system on the inlet side of the pump must allow a smooth flow of fluid to enter the pump at a sufficiently high pressure to avoid cavitation (Fig. 2.2.4a).

This is called the Net Positive Suction Head, generally abbreviated NPSH.

Pump manufacturers supply data about the net positive suction head required by their pumps (NPSHr) for satisfactory operation. When selecting a pump, it is critical the net positive suction head available (NPSHa) in the system is greater than the net positive suction head required by the pump.

For satisfactory pump operation:

NPSHa > NPSHr

N.I.P.A. > N.I.P.R.

NPSHa is also referred to as N.I.P.A. (Net Inlet Pressure Available) and NPSHr is also referred to as N.I.P.R. (Net Inlet Pressure Required).

A simplified way to look at NPSHa or N.I.P.A. is to imagine a balance of factors working for (static pressure and positive head) and against (friction loss and vapour pressure) the pump.

Providing the factors acting for the pump outweigh those factors acting against, there will be a positive suction pressure.

NPSHa = or N.I.P.A.	Pressure action on surface of liquid (Pa)	±	Static suction head (h_s)	-	Pressure drop (h_{fs})	-	Vapour pressure (Pvp)

+ve -ve +ve -ve
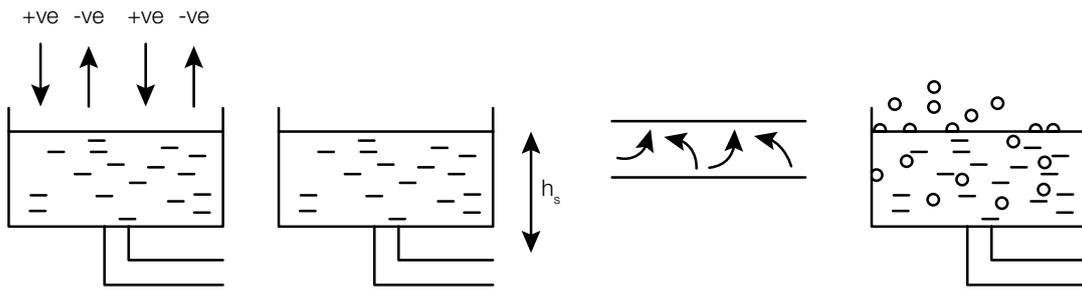
Fig. 2.2.4b NPSH calculation

The value of NPSHa or N.I.P.A. in the system is dependent upon the characteristic of the fluid being pumped, inlet piping, the location of the suction vessel, and the pressure applied to the fluid in the suction vessel. This is the actual pressure seen at the pump inlet. It is important to note, it is the inlet system that sets the inlet condition and not the pump. It is calculated as shown above in Figure 2.2.4b.

It is important the units used for calculating NPSHa or N.I.P.A. are consistent i.e., the total figures should be in m or ft.

For low temperature applications the vapour pressure is generally not critical and can be assumed to be negligible.

NPSHa or N.I.P.A. $= Pa \pm h_s - h_{fs} - Pvp$

Where:

Pa = Pressure absolute above fluid level (bar)
h_s = Static suction head (m)
h_{fs} = Pressure drop in suction line (m)
Pvp = Vapour pressure (bar a)

or

Where:

Pa = Pressure absolute above fluid level (PSI)
h_s = Static suction head (ft)
h_{fs} = Pressure drop in suction line (ft)
Pvp = Vapour pressure (PSIA)
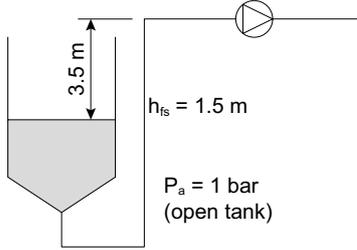
Fig. 2.2.4c Example 1
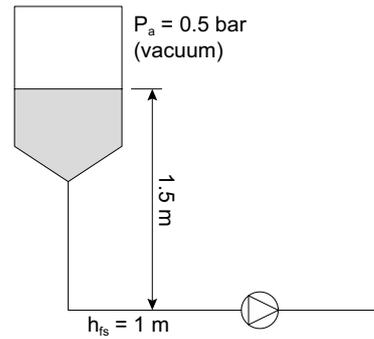
Fig. 2.2.4d Example 2

Example 1 Process:

Water at 50° C (122° F)

Pa	= Pressure Absolute above Fluid Level (1 bar = 10 m) (14.7 PSI = 33.9 ft)
h_s	= Static Suction Head (3.5 m) (11.5 ft)
h_{fs}	= Pressure Drop in Suction Line (1.5 m) (5 ft)
Pvp	= Vapour Pressure (0.12 bar a = 1.2 m) (1.8 PSIA = 4 ft)

NPSHr of pump selected = 3.0 m (10 ft)

$$NPSHa = Pa - h_s - h_{fs} - Pvp \qquad = Pa - h_s - h_{fs} - Pvp$$
$$= 10 - 3.5 - 1.5 - 1.2 \ (m) \quad or \quad = 33.9 - 11.5 - 5 - 4 \ (ft)$$
$$= 3.8 \ m \qquad\qquad = 13.4 \ ft$$

As NPSHa is greater than NPSHr, no cavitation will occur under the conditions stated (Fig. 2.2.4c).

Example 2 Process:

Water at 75° C (167° F)

Pa	= Pressure Absolute above Fluid Level (0.5 bar = 5 m) (7 PSI = 16 ft)
h_s	= Static Suction Head (1.5 m) (5 ft)
h_{fs}	= Pressure Drop in Suction Line (1.0 m) (3 ft)
Pvp	= Vapour Pressure (0.39 bar a = 3.9 m) (5.7 PSIA = 13 ft)

NPSHr of pump selected = 3 m (10 ft)

$$NPSHa = Pa + h_s - h_{fs} - Pvp \qquad = Pa + h_s - h_{fs} - Pvp$$
$$= 5 + 1.5 - 1 - 3.9 \ (m) \quad or \quad = 16 + 5 - 3 - 13 \ (ft)$$
$$= 1.6 \ m \qquad\qquad = 5 \ ft$$

As NPSHa is less than NPSHr, cavitation will occur under the conditions stated (Fig. 2.2.4d).
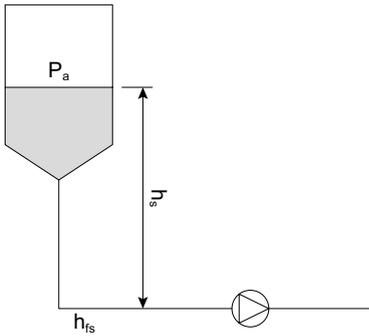
Fig. 2.2.4e Example 3

Example 3 Process:
Glucose at 50° C (122° F)

Pa   = Pressure Absolute above Fluid Level (1 bar = 10 m)
                              (14.7 PSI = 33.9 ft)

h_s = Static Suction Head (1.5 m)
                              (5 ft)

h_{fs} = Pressure Drop in Suction Line (9 m)
                              (29.5 ft)

Pvp   = Vapour Pressure (assumed negligible = 0 m)
                              (0 ft)

NPSHr of pump selected = 3 m (10 ft)

NPSHa   = Pa + h_s - h_{fs} - Pvp   = Pa + h_s - h_{fs} - Pvp
                 = 10 + 1.5 - 9 - 0 (m)   or   = 32.8 + 5 - 29.5 - 0 (ft)
                 = 2.5 m   = 8.2 ft

As NPSHa is less than NPSHr, cavitation will occur under the conditions stated (Fig. 2.2.4e).
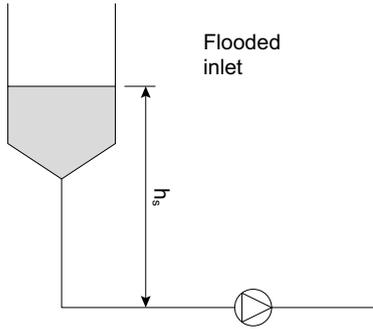
Fig. 2.2.4f Positive effect
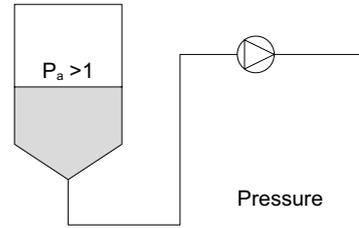
Fig. 2.2.4g Positive effect

From the NPSHa formula it is possible to check and optimise the conditions which affect NPSHa.

The effects are shown in Fig. 2.2.4f - Fig. 2.2.4k
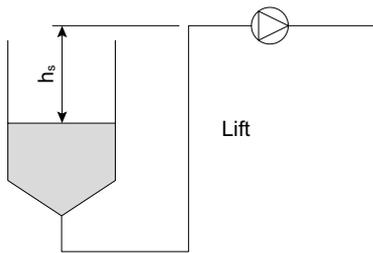
Fig. 2.2.4h Negative effect
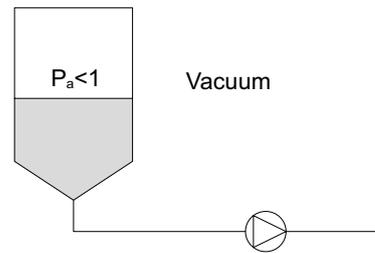
Fig. 2.2.4i Negative effect
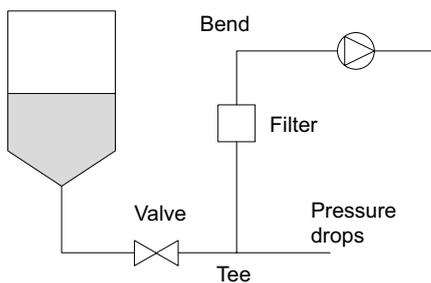
Fig. 2.2.4j Negative effect
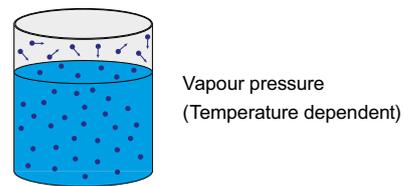
Fig. 2.2.4k Negative effect

Suggestions for avoiding cavitation:

- Keep pressure drop in the inlet line to a minimum i.e., length of line as short as possible, diameter as large as possible, and minimal use of pipe fittings such as tees, valves etc.

- Maintain a static head as high as possible

- Reduce fluid temperature, although caution is needed as this may have an effect of increasing fluid viscosity, thereby increasing pressure drop

2.2.5 Pressure 'Shocks' (Water Hammer)

The term 'shock' is not strictly correct as shock waves only exist in gases. The pressure shock is really a pressure wave with a velocity of propagation much higher than the velocity of the flow, often up to 1400 m/s for steel tubes. Pressure waves are the result of rapid changes in the velocity of the fluid in especially in long runs of piping.

The following causes changes in fluid velocity:

- Valves are closed or opened

- Pumps are started or stopped

- Resistance in process equipment such as valves, filters, metres, etc.

- Changes in tube dimensions

- Changes in flow direction

The major pressure wave problems in process plants are usually due to rapidly closed or opened valves. Pumps, which are rapidly/ frequently started or stopped, can also cause some problems.

When designing pipework systems, it is important to keep the natural frequency of the system as high as possible by using rigid pipework and as many pipework supports as possible, thereby avoiding the excitation frequency of the pump.

Effects of pressure waves:

- Noise in the tube

- Damaged tube

- Damaged pump, valves, and other equipment

- Cavitation

Velocity of propagation

The velocity of propagation of the pressure wave depends on:

- Elasticity of the tubes

- Elasticity of the fluid

- The tubes support

When for example, a valve is closed, the pressure wave travels from the valve to the end of the tube. The wave is then reflected back to the valve. These reflections are in theory continuing but in practice the wave gradually attenuates cancelled by friction in the tube.

A pressure wave as a result of a pump stopping is more damaging than for a pump starting due to the large change in pressure which will continue much longer after a pump is stopped compared to a pump starting. This is due to the low fluid velocity which results in a relatively small damping of the pressure waves.

A pressure wave induced as a result of a pump stopping can result in negative pressure values in long tubes, i.e., values close to the absolute zero point which can result in cavitation if the absolute pressure drops to the vapour pressure of the fluid.

Precautions

Pressure waves are caused by changes in the velocity of the liquid in especially long runs of tube. Rapid changes in the operating conditions of valves and pump are the major reasons to the pressure waves and therefore, it is important to reduce the speed of these changes.

There are different ways to avoid or reduce pressure waves which are briefly described as follows:
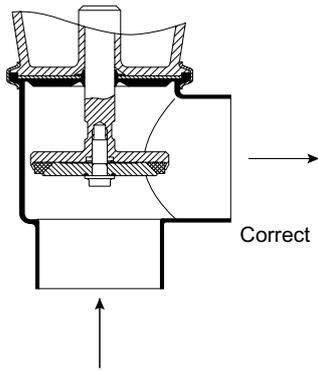
Fig. 2.2.5a Correct flow direction through seat valve
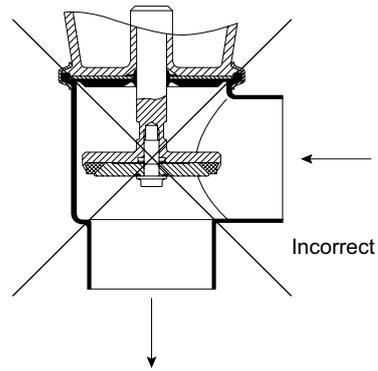
Fig. 2.2.5b Incorrect flow direction through seat valve

Correct flow direction

Incorrect flow direction through valves can induce pressure waves particularly as the valve functions. With air-operated seat valves incorrect direction of flow can cause the valve plug to close rapidly against the valve seat inducing pressure waves (Fig. 2.2.5a) and Fig. 2.2.5b specify the correct and incorrect flow direction for this type of valve.

Correct flow directions in the process plant can reduce or even prevent pressure wave problems.
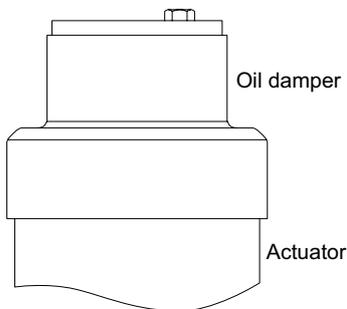
Fig. 2.2.5c Oil damper for seat valve

Damping of valves

The pressure wave induced by a seat valve can be avoided or minimised by damping the movement of the valve plug. The damping is carried out by means of a special damper (see Fig. 2.2.5c).

Speed control of pumps

Speed control of a pump is a very efficient way to minimise or prevent pressure waves. The motor is controlled by means of a soft starter or a frequency converter so that the pump is:

- Started at a low speed which is slowly increased to duty speed

- Stopped by slowly decreasing from duty speed down to a lower speed or zero

The risk of power failure should be taken into consideration when using speed control against pressure waves.

Equipment for industrial processes

There is various equipment available to reduce pressure waves such as:

- Pressure storage tanks

- Pressure towers

- Damped or undamped non-return valves

These however, may not be suitable for hygienic processes and further advice may be required before they are recommended or used in such installations.

This chapter gives an overview of the pump technologies available from Alfa Laval and how to determine which pumps applies within various application areas.

Sustainable, hygienic and efficient pumps

As demand on processes increases, major factors evolve to cater for an ever-growing population. The quality of products and process profitability with an increasing necessity for sustainability and "green" initiatives, adds pressure for the correct selection of a pump to the customer.

As a recognised market leader in pumping technology, Alfa Laval has been at the forefront of supplying sustainable, hygienic and efficient pumps to multiple processes and applications for many years.

The pump is a critical part within a process and must be able to carry out various duties under differing conditions whilst returning economical value to the user.

Some example conditions to consider:

- Transfer several types of fluids/products

- Gentle treatment of the fluids/products

- Overcome different losses and pressure drops in the system

- Supply hygienic and long-lasting operation

- Optimal energy efficiency for sustainable use

- Ensure easy and safe installation, operation and maintenance

Common pump issues can be:

- Incorrect type of pump for the intended application

- Incorrect design of the pump

- Incorrect selection of the pump according to duty conditions, product data etc.

- Incorrect selection of shaft seals

- Incorrect choice of motor drives

3.0

Pump Selection

3.1 General Application Guide

The table shown below gives a general guide as to the various pump technology within Alfa Laval that may be needed to suit the required application (Fig. 3.2a).

General Requirements	Centrifugal	Self-Priming Centrifugal	Rotary Lobe	Circumferential Piston	Twin Screw
Product/Fluid Requirements					
Max. viscosity	800 cP	200 cP	1000000 cP	1000000 cP	1000000 cP
Max. pumping temperature	140° C (284° F)	140° C (284° F)	200° C (392° F)	150° C (392° F)	150° C (392° F)
Min. pumping temperature	- 10° C (14° F)	- 10° C (14° F)	- 20° C (-4° F)	- 20° C (-4° F)	- 20° C (-4° F)
Ability to pump abrasive products	Not recommended	Not recommended	Fair/Moderate	Fair	Fair/Moderate
Ability to pump fluids holding air or gases	Not recommended	Recommended	Fair	Moderate	Recommended
Ability to pump shear sensitive media	Fair	Not recommended	Recommended	Recommended	Recommended
Ability to pump solids in suspension	Fair	Not recommended	Recommended	Recommended	Recommended
CIP/SIP capability (sanitary)	Recommended	Recommended	Recommended	Recommended	Recommended
Dry running capability (when fitted with flushed/quench mechanical seals)	Recommended	Recommended	Recommended	Recommended	Recommended
Self-draining capability	Recommended	Recommended	Recommended	Recommended	Recommended

Performance Requirements					
Max. capacity - m³/hr	520	110	115	157	138
Max. Capacity - US (United States) gal/min	2290	484	506	691	608
Max. discharge pressure - bar	20	5.5	20	40	16
Max. discharge pressure - psig	290	80	290	580	232
Ability to vary flow rate	Fair	Fair	Recommended	Recommended	Recommended
Suction lift capability (primed - wet)	Recommended	Recommended	Recommended	Recommended	Recommended
Suction lift capability (unprimed - dry)	Not recommended	Recommended	Fair	Fair	Fair
Drive Availability					
Electric motor - direct coupled	Yes	Yes	No	No	Yes
Electric motor - geared reducer	No	No	Yes	Yes	Yes
Electric motor - integrated inverter (upon request)	Yes	Yes	Yes	Yes	Yes
Compliance with International Standards and Guidelines					
3-A	Yes	Yes	Yes	Yes	Yes
FDA (Food and Drug Administration)	Yes	Yes	Yes	Yes	Yes
EHEDG (European Hygienic Equipment Design Group)	Yes	No	Yes	Yes	Yes
United States Pharmacopeia (USP)	Yes	Yes	Yes	No	No

Table 3.1a

Alfa Laval Pump Ranges

```
                                    ┌─────────────────────┐
                                    │  Alfa Laval pumps   │
                                    └─────────────────────┘
                                              │
                        ┌─────────────────────┤
                ┌─────────────────┐
                │   Centrifugal   │
                └─────────────────┘
                        │
        ┌───────────────┼───────────────┐
┌─────────────┐  ┌─────────────┐  ┌─────────────┐
│Standard duty│  │Demanding duty│ │  With air   │
└─────────────┘  └─────────────┘  └─────────────┘
        │                │                │
   ┌────┴────┐           │                │
┌────────┐ ┌────────┐ ┌────────┐      ┌────────┐
│ SolidC │ │i-Series│ │  LKH   │      │  LKH   │
└────────┘ └────────┘ └────────┘      │ Prime  │
      │                   │           └────────┘
   ┌──────┐               │
   │FM/GM │               │
   └──────┘               │
                  ┌────────┼────────┬──────────┐
            ┌────────┐ ┌────────┐ ┌────────────┐ ┌────────┐
            │ LKHex  │ │LKH Evap│ │LKH UltraPure│ │ LKHI  │
            └────────┘ └────────┘ └────────────┘ └────────┘
```

Fig. 3.2a Pump ranges

Positive displacement

Standard duty

Demanding duty

Versatile

OptiLobe

OS Twin Screw

SRU

SX/SX UltraPure

DuraCirc/
DuraCirc Aseptic

LKHPF

LKH
Multi-Stage
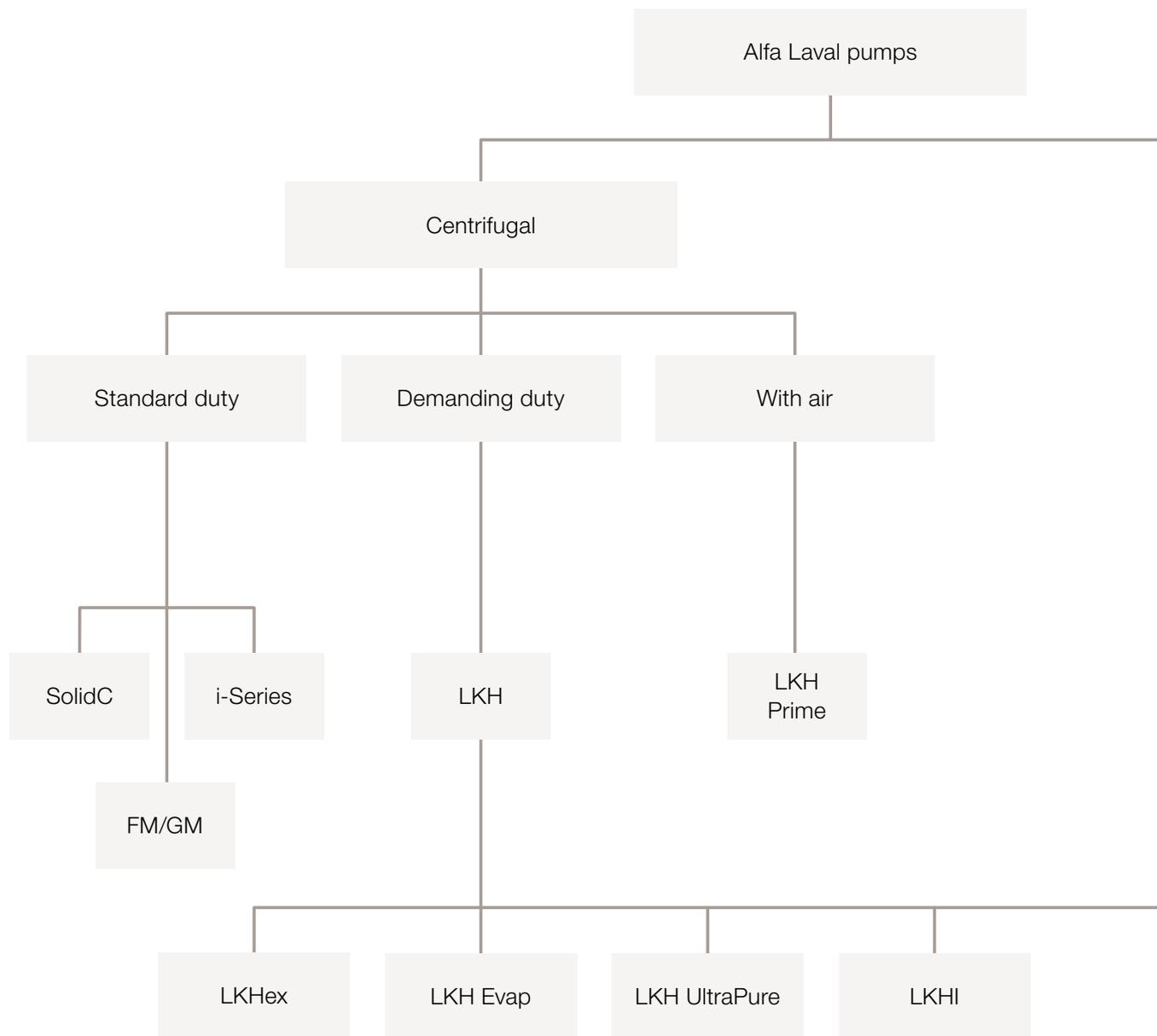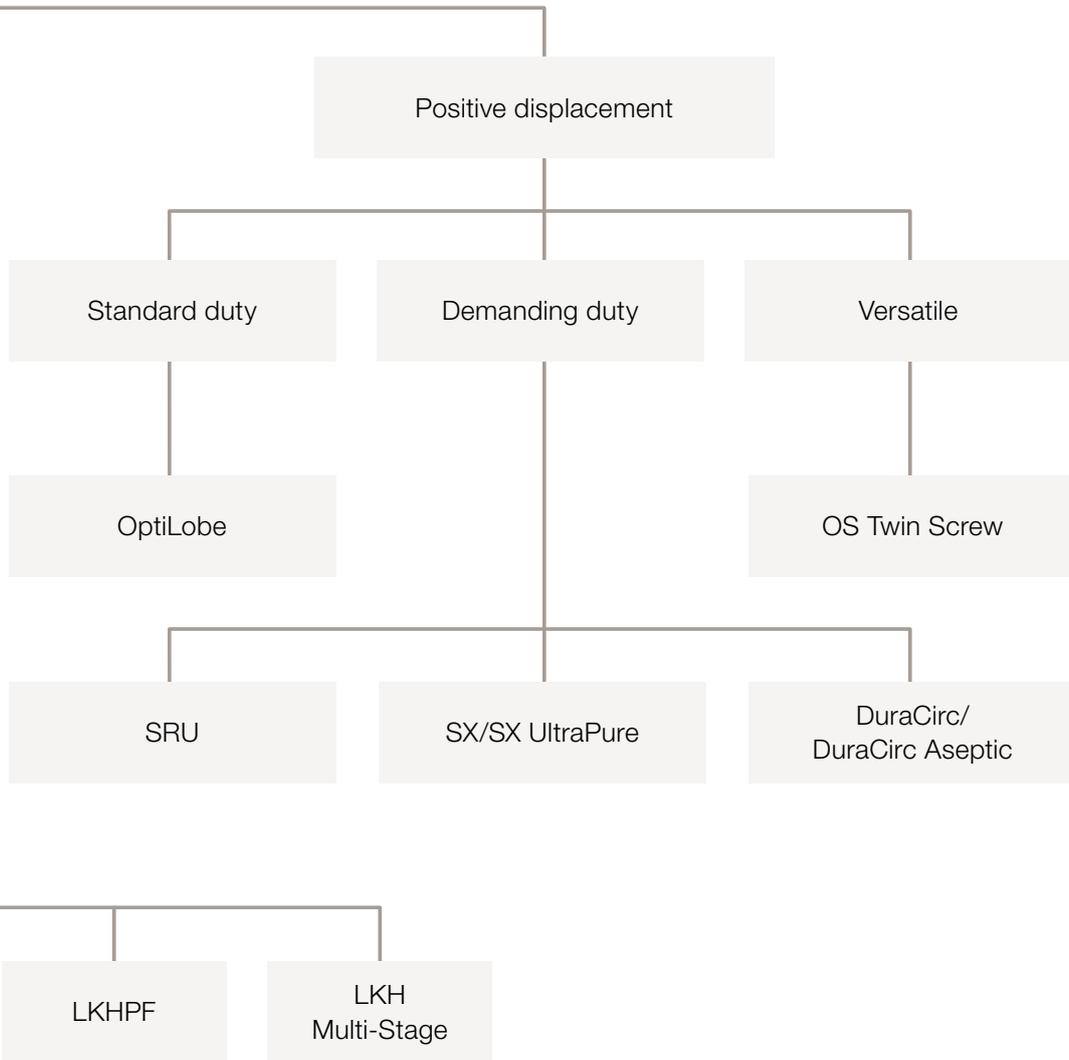
3.2 Pumps for Sanitary Applications

The following table illustrates which Alfa Laval pump ranges can be used in various sanitary application areas. A detailed description of these pump ranges is given in chapter 4.

Pump Type	Pump Range	Application Area										
		Brewery	Confectionary	Dairy	Prepared Foods	Oils / Proteins	Pharmaceutical	Personal/Homecare	Soap	Beverages	Sugar	Water
Centrifugal	LKH	✓	✓	✓	✓	✓		✓	✓	✓		✓
	LKH-Multistage	✓	✓	✓	✓	✓	✓	✓		✓		✓
	LKHPF			✓	✓		✓	✓				✓
	LKH Prime + LKH Prime UP	✓		✓	✓		✓			✓		✓
	LKHI	✓	✓	✓	✓			✓	✓	✓		✓
	LKH Evap	✓	✓	✓	✓	✓				✓		✓
	LKH-Ultra Pure	✓			✓	✓	✓	✓		✓		✓
	Solid C	✓		✓								✓
Rotary Lobe	SRU	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	
	Optilobe	✓	✓	✓	✓	✓		✓	✓	✓		
	SX + SX UP	✓	✓	✓	✓	✓	✓	✓	✓	✓		
Circumferential Piston	DuraCirc	✓	✓	✓	✓	✓			✓	✓	✓	✓
	DuraCirc Aseptic	✓	✓	✓	✓	✓		✓		✓		
Twin Screw	OS	✓	✓	✓	✓	✓		✓	✓	✓	✓	✓

Table 3.2a

Brewery

Alfa Laval Centrifugal and Positive Displacement pumps (PD) are used in most process stages of brewing, from wort handling to beer pasteurisation and filling. Generally, PD pumps best perform with higher fluid viscosity applications, such as liquid sugar tanker offloading and malt syrups, while low fluid viscosity applications, such as beer and water chilling, are mostly carried out using centrifugal pumps.

During the fermentation process, PD pumps such as rotary lobe or twin screw, with their gentle pumping action, are ideally used handling yeast holding delicate cells.

Confectionery

Alfa Laval is a long-standing supplier of pumping equipment to the confectionery industry, supplying pumps to all the major companies. Alfa Laval PD

pumps are to be found in many confectionery processes, where their reliable low shear flow characteristics are ideally suited to the transfer of media such as chocolate, glucose, biscuit cream and fondant. Circumferential piston pumps offer excellent NPSHr characteristics for applications where NPSHa is limited. Confectionery products that hold particulate matter, such as fruit pie fillings, can be handled by Alfa Laval's complete positive displacement pump range.

Alfa Laval's centrifugal pumps can be commonly found in fat and vegetable oil applications.

Dairy

Alfa Laval's whole pump portfolio, with their hygienic construction and conforming to 3-A standards (see chapter 11) with high efficiency and ease of service, are used extensively throughout the dairy industry. Alfa Laval's pumps have been used in milk processing, cream and cultured products such as yoghurt and quark for over 50 years.

Prepared Food

Generally Alfa Laval Positive Displacement pumps can be found on general transfer duties handling products such as pet food, baby food, sauces and flavourings. Alfa Laval's highly efficient centrifugal pump range can be found on products such as edible oils and soups ranging in the medium to low viscosity.

Pharmaceutical

Alfa Laval Centrifugal and Rotary Lobe pumps can be found on many applications within this industry where hygiene and corrosion resistance is paramount, such as cosmetic creams, protein solutions, toothpaste,

perfume, shampoo and blood products. Alfa Laval offers a specialised UltraPure (UP) line for the pharmaceutical industry, comprising of both centrifugal and rotary lobe pumps with industry leading documentation packages and fully traceable, electropolished components.

Personal/Homecare

Alfa Laval Centrifugal and Positive Displacement pumps can be found on many applications within this industry, handling products such as neat soap, sulphonic acid, fabric conditioner, dishwash liquid, fatty acid, SLES (Sodium Laureth Sulfate), CAPB (Cocamidopropyl Betaine), liquid detergents and surfactants. The gentle transference and high efficiency within Alfa Laval's pump portfolio help maintain optimal product integrity, keeping process time.

Beverages

Alfa Laval Centrifugal pumps are mainly used in applications handling thin liquid sugar solutions, water, soft drinks and flavourings. Alfa Laval Positive Displacement pumps are mainly used for applications handling fruit juice concentrates or wine to account for higher viscosity media and gentle handling.

Sugar

Alfa Laval Positive Displacement pumps, with their ability to handle highly viscous, abrasive products, can be found within many areas of sugar refined products requiring hygienic handling, such as high boiled sugars, glucose solutions and sugar syrups used in confectionery, bakery and brewing.

Water

Alfa Laval Centrifugal pumps provide a low-cost effective solution for high purity water and water like applications seen within general or pharmaceutical use.

3.3 ALiCE Configuration Tool

Pump selection for both Centrifugal and Positive Displacement Pumps can be made using Alfa Laval's ALiCE program (Alfa Laval intelligent Configurator Engine). This program prompts the user to enter pump duty conditions and generates a list of pumps most suited to their specific application. The program allows for selection for the whole pump portfolio via dedicated selection tabs and can accompany up to 3 varying duty conditions within calculations.

As well as performing the pump selection, ALiCE also extracts data from a comprehensive rheological database enabling it to suggest in-pump viscosity, Specific Gravity, maximum operating speeds, elastomer compatibility and primary seal configuration to users to streamline the process.

The extensive rheology database contained within ALiCE is based on rheological tests performed over decades on end users' liquids at Alfa Laval's laboratory and will be continually added as additional products are tested.

After the pump has been selected, the user will be aided to complete a pump unit design. This will include a wide scope of specification options such as connection types, heating/cooling devices and ancillaries that have been included in the program, driven by market demand.

ALiCE offers insight to the noted duty conditions within the ancillary section, detailing the operating speed, the power absorbed, and torque required for each duty

point and cross checking against any chosen motor drive to ensure all points are covered. This provides peace of mind to the user and ensures the most optimal selection for both capital and energy consumption costs.

After completion, the price of the pump will be displayed and its configuration code (item number) can be generated, simplifying the quotation and/or ordering process for all users.

In addition, ALiCE will also provide a detailed parts list for the pump with item numbers and recommended service kits identified and priced. Dimensional details in the form of general arrangement drawings can also be generated within the software and sent to an email of your choice.

A link to all technical information that may be needed to go with the quotation such as Operation manuals, generic or specific performance curves, and technical data sheets can also be provided.

Flexibility has been built into the software to enable specific enquiries to be answered without the need to complete a full pump selection. For example, recommended service kits can be extracted based on an existing configuration code or direct access to technical information relating to a specific pump technology is also possible.

All information is offered for guidance purposes only.

If you would like access to the Alfa Laval Configuration Tool, please contact your local Alfa Laval sales company.

This chapter gives a description of Alfa Laval pump ranges including design, principle of operation and pump model types.

4.1 Centrifugal Pumps

4.1.1 General

The Alfa Laval range of Centrifugal Pumps has been designed specially for use in the food, dairy, beverage, personal and home care, pharmaceutical and light chemical industries. Centrifugal pumps including self-priming, multi-stage and those for high inlet pressure, can handle most low viscosity applications.

Attributes include:

- Hygienic and cleanable

- High efficiency

- Low power consumption

- Low noise level

- Low NPSH requirement

- Easy maintenance

4.0

Pump Description
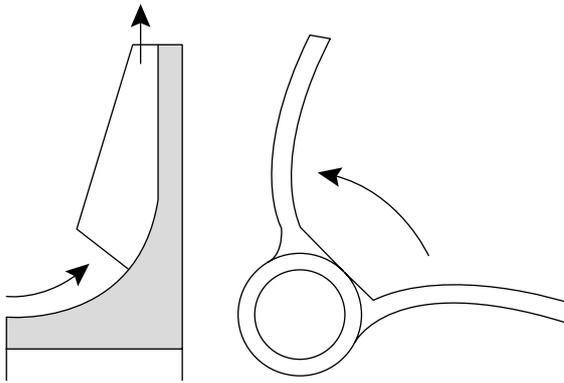
Fig. 4.1.2a Principle of operation
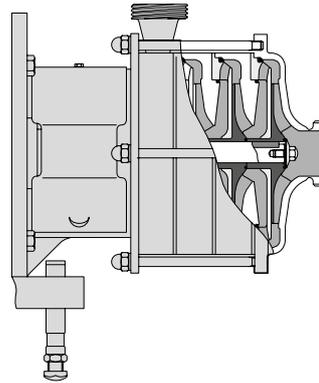
Fig. 4.1.2b Multistage centrifugal pump

4.1.2 Principle of Operation

Fluid is directed to the impeller eye and is forced into a circular movement by the rotation of the impeller vanes. As a result of this rotation, the impeller vanes transfer mechanical work to the fluid in the impeller channel, which is formed by the impeller vanes. The fluid is then pressed out of the impeller by means of centrifugal force and finally leaves the impeller channel with increased pressure and velocity (Fig. 4.1.2a). The velocity of the fluid is also partly converted into pressure by the pump casing before it leaves the pump through the outlet.

The principle of the multi-stage centrifugal pump is the same as the conventional centrifugal pump (Fig. 4.1.2b). The pump consists, however, of several impellers (several stages) which further develop the pressure from one stage to another but flow rate is unchanged. The multi-stage centrifugal pump operates as if several conventional centrifugal pumps are connected in series.

The impeller has two or multiple vanes depending on the type of centrifugal pump. The impeller diameter and width will vary dependent upon the duty requirements.

Fig. 4.1.3a Semi-open impeller

4.1.3 Design

In general, the Alfa Laval Centrifugal Pump does not contain many parts, with the pump head being connected to an electric motor. The impeller is fixed onto the pump shaft which is housed in a pump casing and back plate – these components are described in the following text:

Impeller

The impeller is of cast manufacture and semi-open type i.e., the impeller vanes are open in front (Fig. 4.1.3a). This type allows visual inspection of the vanes and the area between them. This design makes it easy to clean and suitable for polishing.
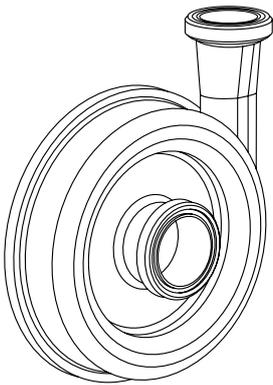
Fig. 4.1.3b Pump casing
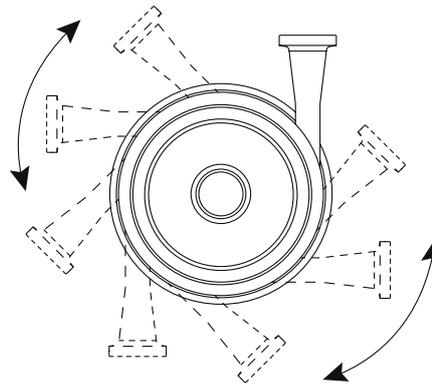
Fig. 4.1.3c 360° flexibility

Pump Casing

The pump casing is of rigid steel manufacture, complete with male screwed connections and can be supplied with fittings or clamp liners (Fig. 4.1.3b).

The pump casing is designed for multi position outlet, with 360° flexibility (Fig. 4.1.3c).
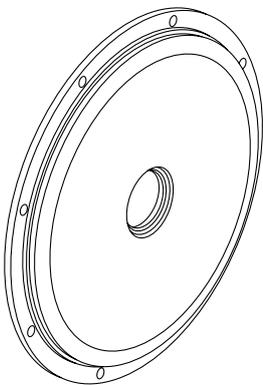
Fig. 4.1.3d Back plate

Back Plate

The back plate is of pressed steel manufacture, which together with the pump casing form the actual fluid chamber in which the fluid is transferred by means of the impeller (Fig. 4.1.3d).

Mechanical Seal

The connection between the motor shaft/pump shaft and the pump casing is sealed by means of a mechanical seal, which is described in chapter 6.
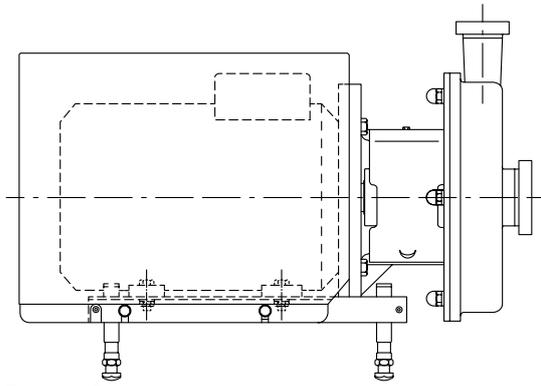
Fig. 4.1.3e Pump with shroud and legs
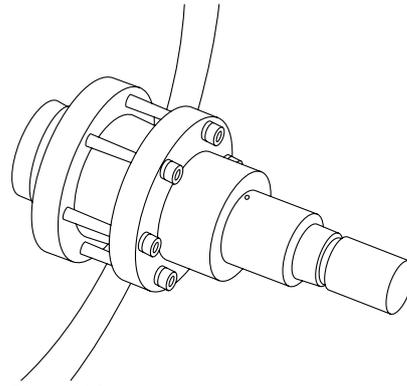
Fig. 4.1.3f Compression coupling

Shroud and Legs

Most pump types are fitted with shrouds and adjustable legs (Fig. 4.1.3e). The shroud maybe insulated to keep noise to a minimum and protect the motor against damage.

Please note Alfa Laval Centrifugal Pumps for the US market are supplied without shrouds in order to meet 3A requirements.

Pump Shaft/Connections

Most pumps have stub shafts that are fixed to the motor shafts by means of compression couplings, eliminating the use of keyways (Fig. 4.1.3f). The stub shaft assembly design provides a simple, yet secure method of drive that reduces vibration and noise. On the multistage centrifugal pump, the length of the pump shaft will differ depending upon the number of impellers fitted.
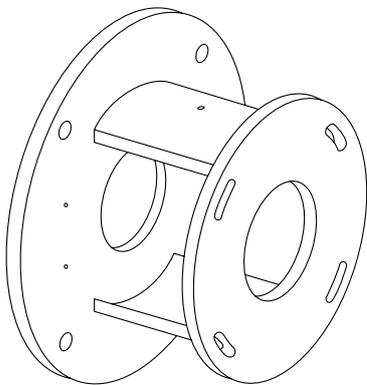
Fig. 4.1.3g Adaptor

Adaptor

The connection between the motor and back plate is made by means of an adaptor, which can be attached to any standard IEC or NEMA frame electric motor (Fig. 4.1.3g).

Fig. 4.1.4a LKH

Fig. 4.1.4b LKH (US version)

4.1.4 Pump Range

The Alfa Laval Centrifugal Pump portfolio comprises of different ranges as follows:

LKH Range

The LKH pump is a highly efficient and economical centrifugal pump, which meets the requirements of hygienic and gentle product treatment and chemical resistance (Fig. 4.1.4a).

The Alfa Laval LKH Centrifugal Pump is suited to demanding applications in a variety of industries such as dairy, food, beverage, home and personal care and additional chemical industries.

The LKH range is available in thirteen sizes: LKH-5, -10, -15, -20, -25, -35, -40, -45, -50, -60, -70, -75 (US) (Fig. 4.1.4b), -85 and -90.

Suitable for inlet pressures up to 10 bar (145 PSIG) and delivering flow rates for 50 Hz up to 500 m^3/hr (2200 US gal/min) and differential pressures up to 11.5 bar (165 PSIG) and for 60 Hz, 16 bar (230 PSIG).

Fig. 4.1.4c LKH Evap

Fig. 4.1.4d LKH Evap (US version)

LKH Evap Range

The Alfa Laval LKH Evap Centrifugal Pump is suited to evaporation type applications within the dairy, food, beverage, brewery, alcohol, ethanol, starch and chemical industries (Fig. 4.1.4c - 4.1.4d). As a low-NPSHr, high efficiency centrifugal pump, the LKH Evap is a tailored evaporator pump supported by strong and extensive documentation.

It is ideal for use in evaporation duties for applications, such as liquid concentration and powder processing as well as plant and equipment dewatering.

Additional port sizing options compared to standard LKH on some models provide for an improved suction performance, vital to applications where NPSHa is limited.

It features a special scraper impeller option, ClearFlow, that solves the product build-up problem in high solids applications, which can prolong production time between cleaning (Fig. 4.1.4e).

The LKH Evap pump is available in ten sizes, LKH Evap-10, -15, -20, -25, -35, -40, -45, -50, -60 and -70.

Flow rates for 50 Hz up to 280 m³/hr (1233 US gal/min) and differential pressures up to 10 bar (145 PSIG) and for 60 Hz up to 280 m³/hr (1233 US gal/min) and differential pressure up to 16 bar (230 PSIG).

Fig. 4.1.4e Clearflow impeller

Fig. 4.1.4f LKHex

LKHex Range

These pumps derived from the standard LKH are designed to meet the requirements of the ATEX directive 2014/34/EU group II, category 2G, temperature class T3 and T4 and can be used in potentially explosive environments (Fig. 4.1.4f).

The LKHex pump is available in 10 sizes, LKHex-10, -15, -20, -25, -35, -40, -45, -50, -60 and -70.

Flow rates for 50 Hz up to 280 m³/hr (1233 US gal/min) and differential pressures up to 10 bar (145 PSIG) and for 60 Hz up to 280 m³/hr (1233 US gal/min) and differential pressure up to 16 bar (230 PSIG).

This range is also available in an UltraPure version with the exception of LKHex-15 and -50.

Fig. 4.1.4g LKH-UltraPure

Fig. 4.1.4h LKH-UltraPure (US version)

LKH UltraPure Range

The Alfa Laval LKH UltraPure pump is designed to meet the stringent demands and regulations of high-purity applications across the biotechnology and pharmaceutical industries which require equipment with the highest material integrity (Fig. 4.1.4g - 4.1.4h).

Designed in accordance with ASME BPE and GMP, this pump range complies with the stringent requirements necessary for validation, qualification and process control providing consumer safety.

Typically supplied with a 45° casing outlet to ensure it is self-venting and options of improved surface finishes to prevent biofilm build up.

All pumps are delivered with a complete Alfa Laval Q-doc documentation package including material traceability. Q-doc provides easier validation, proof of origin and compliance for inspection according to Good Manufacturing Practice (GMP) and ASME BPE requirements.

The LKH-UltraPure range is available in eight sizes, LKH-UltraPure-10, -20, -25, -35, -40, -45, -60 and -70.

Flow rates up to 280 m³/hr (1233 US gal/min) and differential pressures for 50 Hz up to 10 bar (145 PSIG) and for 60 Hz, 16 bar (230 PSIG).

Fig. 4.1.4i LKHI

Fig. 4.1.4j LKHI (US version)

LKHI Range

This pump range is similar to the LKH range but is suitable for inlet pressures up to 16 bar (230 PSIG). The pump can withstand this high inlet pressure due to being fitted with an internal shaft seal (Fig. 4.1.4i - 4.1.4j).

The LKHI range is available in nine sizes, LKHI-10, -15, -20, -25, -35, -40, -45, -50 and -60.

Flow rates for 50 Hz up to 240 m³/hr (1056 US gal/min) with differential pressures up to 8 bar (115 PSIG). For 60 Hz, flow rates up to 275 m³/hr (1210 US gal/min) with differential pressures up to 11 bar (160 PSIG).

Fig. 4.1.4k LKH-Multistage

Fig. 4.1.4l LKH-Multistage (US version)

LKH Multistage Range

These pumps are primarily used in applications with high outlet pressure and low capacity requirements such as breweries, reverse osmosis and ultra-filtration. The pumps are available as two, three or four stage models (i.e., pumps fitted with two, three or four impellers respectively - Fig. 4.1.4k - 4.1.4.l).

Flow rates for 50 Hz up to 75 m³/hr (330 US gal/min) and discharge pressures up to 40 bar (580 PSIG) with boost pressures up to 19 bar (275 PSIG) and for 60 Hz up to 80 m³/hr (352 US gal/min) and boost pressures up to 26 bar (375 PSIG).

For inlet pressures greater than 10 bar (145 PSIG) a 'special' motor is used incorporating fixed angular contact bearings due to axial thrust.

The LKH Multistage range is available in six sizes			
Pump Size	Number of stages	Pump Size	Number of stages
LKH-112	2	LKH-122	2
LKH-113	3	LKH-123	3
LKH-114	4	LKH-124	4

Fig. 4.1.4m LKHPF-High Pressure

Fig. 4.1.4n LKHPF-High Pressure (US version)

LKHPF High Pressure Range

These pumps are designed to handle high inlet pressures built with reinforced pump casing and back plate. Application areas include reverse osmosis mono-filtration and ultra-filtration (Fig. 4.1.4m - 4.1.4n).

The LKHPF-High Pressure range is available in nine sizes, LKHP-10, -20, -25, -35, -40, -45, -50, -60 and -70.

The pump range is designed for inlet pressures up to 40 bar (580 PSIG). Flow rates for 50 Hz up to 280 m^3/hr (1233 US gal/min) with differential pressures up to 10 bar (145 PSIG) and for 60 Hz, differential pressures up to 16 bar (230 PSIG).

For these high inlet pressures a 'special' motor with fixed angular contact bearings is used due to axial thrust.

Fig. 4.1.4o SolidC

Fig. 4.1.4p SolidC (US version)

SolidC Range

The SolidC is the all-purpose Alfa Laval centrifugal pump for less demanding applications (Fig. 4.1.4o - 4.1.4p).

Designed for Cleaning-in-Place (CIP), it is ideal for basic duties across the dairy, food, beverage and personal care industries in which hygienic treatment is required. Typical applications are pumping of CIP solutions, utilities, cooling or heating water, and other simple transport duties.

The SolidC range is available in four sizes, SolidC-1, -2, -3 and -4.

Suitable for inlet pressures up to 4 bar (58 PSIG) and delivering flow rates for 50 Hz up to 75 m^3/hr (330 US gal/min) with differential pressures up to 8 bar (115 PSIG) and for 60 Hz, 11.5 bar (167 PSIG).

Fig. 4.1.4q LKH Prime

Fig. 4.1.4r LKH Prime (US version)

LKH Prime Range

Based on the Alfa Laval LKH pump, the Alfa Laval LKH Prime centrifugal pump is a versatile, highly efficient self-priming pump for use in hygienic applications, especially tank emptying and CIP return applications (Fig. 4.1.4q - 4.1.4r). With its combination of airscrew technology and advanced design, the pump can remove air or gas from the suction pipe.

The LKH Prime pump is designed to meet the stringent hygienic requirements across the food, dairy, beverage, and home-personal care industries. With verified and effective CIP cleanability, the LKH Prime can be used as a product pump as well.

The design of the LKH Prime is principally the same as LKH but with an additional priming chamber, airscrew and recirculation pipe.

1. Liquid ring is formed due to rotations of airscrew
2. Recirculation pipe
3. Water seal is created
4. Liquid/air mix passes through the canister
5. Liquid is transferred
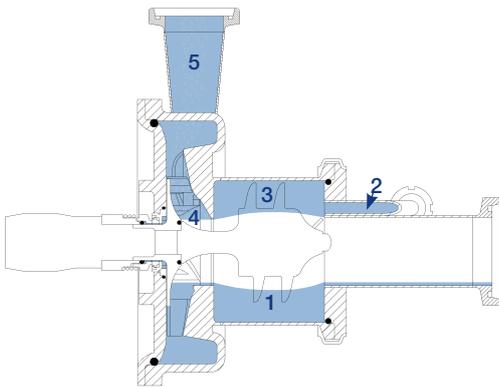
Fig. 4.1.4s Alfa Laval LKH Prime Pump

LKH Prime, Principle of Operation

Its principal of operation is that as the Alfa Laval LKH Prime pump starts up, the centrifugal force generated from the rotation of the airscrew and the initial priming liquid causes the formation of a liquid ring in the pump head canister (1). This also fills the recirculation pipe (2), thereby achieving the initial prime (Fig. 4.1.4s).

By design, the canister is offset from the centre of the airscrew and the resultant liquid ring creates a water seal between the airscrew hub and the top of the canister (3).

Due to the offset design, an air column is created between the airscrew hub and the liquid ring at the bottom of the canister (1). The rotating vanes of the airscrew separate the air column into air pockets, which are forced through the canister into the impeller's suction stage.

Some of the initial priming liquid recirculates from the casing discharge into the airscrew casing through the recirculation pipe (2). Until all the air has been evacuated, air pockets will continue to be generated.

When the air content is just a few percent, the air is contained as bubbles in the liquid. No air pockets are generated. Instead the liquid/air mix passes through the canister into the impeller's suction stage (4). Here, the pump acts as a traditional centrifugal pump, transferring the liquid through the discharge (5) at a higher velocity and pressure.

When there is no air present, the canister and recirculation loop have no function and are completely filled with liquid. The liquid passes through the canister into the impeller's suction stage. Here again, the pump acts as a traditional centrifugal pump, transferring the liquid through the discharge at a higher velocity and pressure.

The LKH Prime range is available in three sizes, LKH Prime-10, -20 and -40

Flow rates for 50 Hz up to 280 m³/hr (1233 US gal/min) and differential pressures up to 10 bar (145 PSIG) and for 60 Hz up to 280 m³/hr (1233 US gal/min) and differential pressure up to 16 bar (230 PSIG).

Fig. 4.1.4t LKH Prime -UltraPure

Fig. 4.1.4u LKH Prime-UltraPure (US version)

LKH Prime UltraPure Range

The Alfa Laval LKH Prime UltraPure is designed to meet the stringent demands and regulations of high-purity applications across the biotechnology and pharmaceutical industries. Where material integrity, high efficiency, exceptional cleanability, contamination safety, robust design and low maintenance are of paramount importance (Fig. 4.1.4t - 4.1.4u).

It is ideal for tank emptying and CIP return applications; having verified and effective CIP cleanability. The LKH Prime UltraPure can also be used as a product pump.

Designed in accordance with ASME BPE and GMP, this pump range complies with the stringent requirements necessary for validation, qualification and process control providing consumer safety.

All pumps are delivered with a complete Alfa Laval Q-doc documentation package including material traceability. Q-doc provides easier validation, proof of origin and compliance for inspection according to Good Manufacturing Practice (GMP) and ASME BPE requirements.

There are two sizes available, being LKH Prime-UltraPure 10 and 20.

Flow rates up to 70 m³/hr (308 US gal/min) and differential pressures up to 4 bar (58 PSIG) for 50 Hz, and 80 m³/hr (350 US gal/min) up to 5.5 bar (80 PSIG) for 60 Hz.

4.2 Rotary Lobe Pumps

4.2.1 General

The Alfa Laval range of Rotary Lobe Pumps with its non-contact pump element design has the ability to cover a wide range of applications in industry. The hygienic design, anti-corrosive stainless steel construction and smooth pumping action have long established these pumps in the food, beverage, dairy, personal/homecare, and pharmaceutical industries.

Attributes include:

- Gentle transfer of delicate suspended solids

- Bi-directional operation

- Compact size with high performance and low energy input

- Ability to pump shear sensitive media

- Easy maintenance

4.2.2 Principle of Operation

Alfa Laval ranges of Rotary Lobe pumps are of conventional design operating with no internal contacting parts in the pump head. The pumping principle is explained with reference to the diagram below, which shows the displacement of fluid from pump inlet to outlet. The rotors are driven by a gear train in the pump gear gearbox providing accurate synchronisation or timing of the rotors. The rotors contra-rotate within the pump head carrying fluid through the pump, in the cavities formed between the dwell of the rotor and the interior of the rotor case.

In hydraulic terms, the motion of the counter rotating rotors creates a partial vacuum that allows atmospheric pressure or other external pressures to force fluid into the pump chamber. As the rotors rotate an expanding cavity is formed which is filled with fluid. As the rotors separate, each dwell forms a cavity. The meshing of the rotor causes a diminishing cavity with the fluid being displaced into the outlet port.

Horizontally ported pump (top shaft drive)
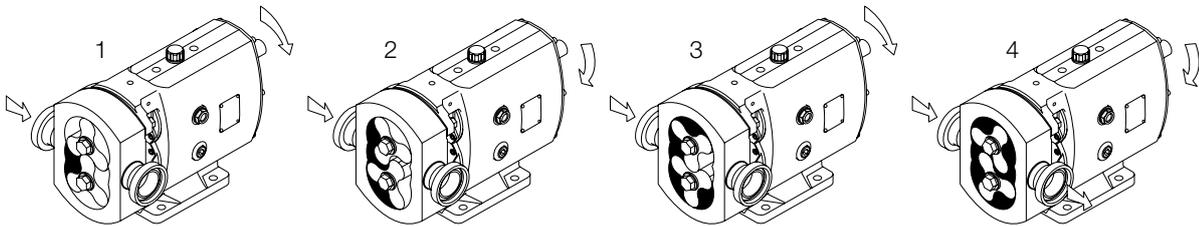
Vertically ported pump (left hand shaft drive)
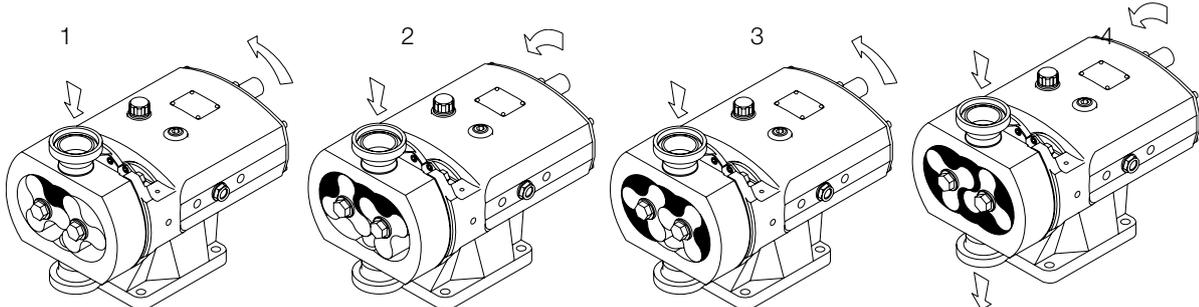
Fig. 4.2.2a Principle of operation

Fig. 4.2.3a SRU

4.2.3 Pump Range

Alfa Laval Rotary Lobe Pumps can be supplied bare shaft (without drive) or complete with geared electric motor (see section 8.2.7). Ranges primarily as follows:

SRU Range

The SRU pump range has been designed for use on general transfer duties throughout the beverage, dairy, food, home/personal care, and chemical manufacturing processes.

The SRU range is available in six gearbox sizes each having two pump head displacements and mostly two different shaft materials (Table 4.2.3a).

- Displacement is the theoretical amount of fluid the pump will transfer per revolution

- Duplex stainless steel shaft material used for higher pressures

The SRU pump range incorporates a universally mounted gearbox which gives the flexibility of mounting pumps with the inlet and outlet ports in either a vertical or horizontal plane by changing the foot and foot position. This pump range also incorporates full bore through porting complying with international standards BS4825/ISO2037, maximising inlet and outlet port efficiency and NPSH characteristics.

Flow rates up to 106 m³/hr (466 US gal/min) and pressures up to 20 bar (290 PSIG).

The SRU range dependent upon seal configuration conforms to US 3A requirements.

Pump Nomenclature

Frame Size	Model	Port Position	Shaft Material	Displacement		Max. Pressure (S or D)		Max. speed (rpm)
				litres/ 100 rev	US gal/ 100 rev	bar	PSI	
1	SRU1/005	L or H	D	5.3	1.40	8	115	1000
	SRU1/008	L or H	D	8.5	2.25	5	75	1000
2	SRU2/013	L or H	S or D	12.8	3.38	10/15	145/215	1000
	SRU2/018	L or H	S or D	18.1	4.78	7/10	100/145	1000
3	SRU3/027	L or H	S or D	26.6	7.03	10/15	145/215	1000
	SRU3/038	L or H	S or D	38.4	10.2	7/10	100/145	1000
4	SRU4/055	L or H	S or D	55.4	14.6	10/20	145/290	1000
	SRU4/079	L or H	S or D	79.0	20.9	7/15	145/215	1000
5	SRU5/116	L or H	S or D	116	30.7	10/20	145/290	600
	SRU5/168	L or H	S or D	168	44.4	7/15	145/215	600
6	SRU6/260	L or H	S or D	260	68.7	10/20	145/290	500
	SRU6/353	L or H	S or D	353	93.2	7/15	145/215	500

L - Horizontal Porting

H - Vertical Porting

S - Stainless Steel

D - Duplex Stainless Steel

Nomenclature example: SRU4/079/HD is the SRU4/079 with vertical ports and 15 bar (215 PSI) max. pressure.

Table 4.2.3a

Fig. 4.2.3b SX

SX Range

The SX pump range is designed for gentle transportation of process fluids in hygienic and ultra-clean applications (Fig. 4.2.3b). Suited for use in the home and personal care sector, and for demanding food and dairy applications. This pump range like the SRU range incorporates a universally mounted gearbox which gives the flexibility of mounting pumps with the inlet and outlet ports in either a vertical or horizontal plane by changing the foot and foot position. This pump range also incorporates full bore through porting complying with international standards BS4825/ISO2037, maximising the inlet and outlet efficiency of the pump and the NPSH characteristics.

The SX range has been certified by EHEDG (European Hygienic Equipment Design Group) as fully CIP cleanable to their protocol. In addition to being EHEDG compliant, the SX pump also conforms to the US 3A standard and all media contacting components are FDA compliant. All media contacting elastomers are controlled compression joints to prevent pumped media leaking to atmosphere (see section 6.2).

The SX range is available with seven gearboxes each having two pump head displacements. Flow rates up to 115 m^3/hr (506 US gal/min) and pressures up to 15 bar (215 PSIG) (Table 4.2.3b).

Pump Nomenclature

Frame Size	Model	Displacement		Max. Pressure (S or D)		Max. speed (rpm)
		litres/100 rev	US gal/100 rev	bar	PSI	
1	SX1/005	5	1.32	12	175	1400
	SX1/007	7	1.85	7	100	1400
2	SX2/013	13	3.38	15	215	1000
	SX2/018	18	4.78	7	100	1000
3	SX3/027	26	7.03	15	215	1000
	SX3/035	35	9.25	7	100	1000
4	SX4/046	46	12.2	15	215	1000
	SX4/063	63	16.7	10	145	1000
5	SX5/082	82	21.7	15	215	600
	SX5/115	115	30.4	10	145	600
6	SX6/140	140	37	15	215	500
	SX6/190	190	50.2	10	145	500
7	SX7/250	250	66.1	15	215	500
	SX7/380	380	100	10	145	500

Table 4.2.3b

Fig. 4.2.3c SX UltraPure

SX UltraPure Range

The SX UltraPure pump range is designed for gentle transportation of process fluids in hygienic and high purity applications (Fig. 4.2.3c). Suited for use in the pharmaceutical, biotechnology and personal care sector.

Designed in accordance with ASME BPE and GMP, this pump range complies with the stringent requirements necessary for validation, qualification and process control providing consumer safety.

This pump range also incorporates full bore through porting complying with international standards BS4825/ISO2037, maximising the inlet and outlet efficiency of the pump and the NPSH characteristics.

The SX UltraPure range has been certified by EHEDG (European Hygienic Equipment Design Group) as fully CIP cleanable to their protocol. In addition to being

EHEDG compliant all media contacting components are FDA compliant with USP Class VI elastomers. All media contacting elastomers are controlled compression joints to prevent pumped media leaking to atmosphere (see section 6.2).

All pumps are delivered with a complete Alfa Laval Q-doc documentation package including material traceability. Q-doc provides easier validation, proof of origin and compliance for inspection according to Good Manufacturing Practice (GMP) and ASME BPE requirements.

The SX UltraPure range is available with seven gearboxes each having two pump head displacements (Table 4.2.3c).

Flow rates up to 115 m³/hr (506 US gal/min) and pressures up to 15 bar (215 PSIG).

Pump Nomenclature

Frame Size	Model	Displacement		Max. Pressure		Max. speed (rpm)
		litres/100 rev	US gal/100 rev	bar	PSI	
1	SX UltraPure 1/005	5	1.32	12	175	1400
	SX UltraPure 1/007	7	1.85	7	100	1400
2	SX UltraPure 2/013	13	3.38	15	215	1000
	SX UltraPure 2/018	18	4.78	7	100	1000
3	SX UltraPure 3/027	26	7.03	15	215	1000
	SX UltraPure 3/035	35	9.25	7	100	1000
4	SX UltraPure 4/046	46	12.2	15	215	1000
	SX UltraPure 4/063	63	16.7	10	145	1000
5	SX UltraPure 5/082	82	21.7	15	215	600
	SX UltraPure 5/115	115	30.4	10	145	600
6	SX UltraPure 6/140	140	37	15	215	500
	SX UltraPure 6/190	190	50.2	10	145	500
7	SX UltraPure 7/250	250	66.1	15	215	500
	SX UltraPure 7/380	380	100	10	145	500

Table 4.2.3c

Fig. 4.2.3d OptiLobe

OptiLobe Range

The Alfa Laval OptiLobe Rotary Lobe Pump is a cost-effective alternative for general applications that require gentle product treatment and easy serviceability (Fig. 4.2.3d). Suited for use in applications across the dairy, food, beverage, home, and personal care industries. Easy to adapt to vertical or horizontal porting by changing the foot position on the gearbox provides good flexibility for a variety of installations.

The OptiLobe range has been certified by EHEDG (European Hygienic Equipment Design Group) as fully CIP cleanable to their protocol. In addition to being EHEDG compliant, the OptiLobe pump also conforms to the US 3A standard and all media contacting components are FDA compliant.

The OptiLobe range is available with five gearboxes each having two pump head displacements. Flow rates up to 77 m^3/hr (339 US gal/min) and pressures up to 8 bar (115 PSIG) (Table 4.2.3d).

Pump Nomenclature

Frame Size	Model	Displacement		Max. Pressure (S or D)		Max. speed (rpm)
		litres/100 rev	US gal/100 rev	bar	PSI	
10	OptiLobe 12	6	1.48	8	115	1000
	OptiLobe 13	10	2.61	8	115	1000
20	OptiLobe 22	17	4.49	8	115	1000
	OptiLobe 23	21	5.55	8	115	1000
30	OptiLobe 32	32	8.45	8	115	1000
	OptiLobe 33	40	10.57	8	115	1000
40	OptiLobe 42	64	16.91	8	115	1000
	OptiLobe 43	82	21.66	8	115	1000
50	OptiLobe 52	117	30.89	8	115	750
	OptiLobe 53	172	45.41	8	115	750

Table 4.2.3d

<div align="center">Suction   Discharge</div>
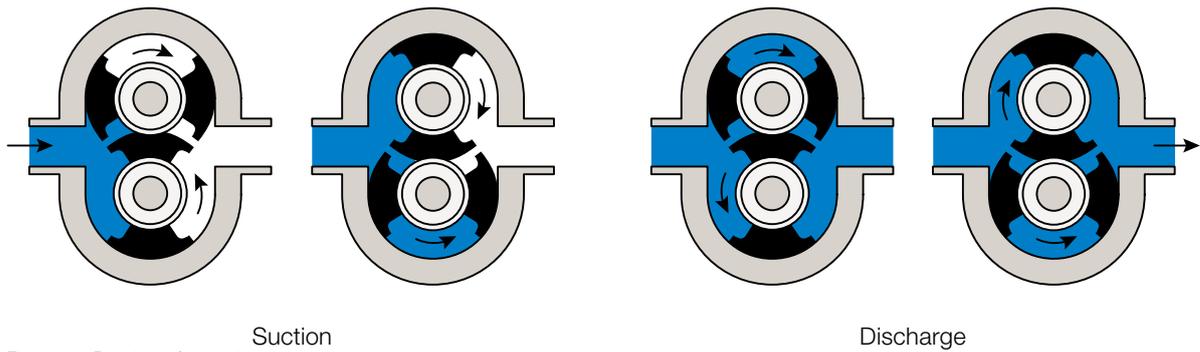
Fig. 4.3.1a Principle of operation

4.3 Circumferential Piston Pumps

4.3.1 General

The Alfa Laval range of Circumferential Piston Pumps are designed for hygienic applications within the dairy, food, beverage, home, and personal care industries. The highly efficient design with close clearances and long slip paths is particularly suited to applications that are low in viscosity with medium to high discharge pressures and require equipment that can be cleaned in place (CIP).

Attributes include:

- Gentle transfer of delicate suspended solids
- Excellent suction performance
- Higher pressure capabilities
- Easy maintenance
- Bi-directional operation

4.3.2 Principle of Operation

The Alfa Laval Circumferential Piston pumping principle is explained with reference to the diagram above (Fig. 4.3.1a), which shows the displacement of fluid from pump inlet to outlet. The rotors are driven by a gear train in the pump gear gearbox providing accurate synchronisation or timing of the rotors. The rotors contra-rotate within the pump head carrying fluid through the pump, in a channel between the rotor wings and the dwell of the rotors and the interior of the rotor case.

The rotor pistons with close running clearances rotate around the circumference of the channel in the pump casing. This continuously generates a partial vacuum at the suction port as the rotors un-mesh, causing fluid to enter the pump. The fluid is transported around the channel by the rotor pistons and is displaced into the outlet port as the rotor pistons re-mesh. The direction of flow is reversible.

Fig. 4.3.1b DuraCirc

DuraCirc Range

Designed for Cleaning-in-Place (CIP), the Alfa Laval DuraCirc is ideal for hygienic applications within the dairy, food, beverage, home and personal care industries (Fig. 4.3.1b). The highly efficient design is particularly suited to applications that are low in viscosity with medium to high discharge pressures and require equipment that can be cleaned in place.

DuraCirc has a variety of options available to suit may different processes and is designed to keep processes running with minimal maintenance requirements.

The DuraCirc range has been certified by EHEDG (European Hygienic Equipment Design Group) as fully CIP cleanable to their protocol. In addition to being EHEDG compliant, the DuraCirc pump also conforms to the US 3A standard, and all media contacting components are FDA compliant. All media contacting elastomers are controlled compression joints to prevent pumped media leaking to atmosphere (see section 6.2).

The DuraCirc range is available with five gearboxes and a total of thirteen pump head volumetric displacements. Flow rates up to 149 m^3/hr (650 US gal/min) and pressures up to 40 bar (580 PSIG) (Table 4.3.3a).

Pump Nomenclature

Frame Size	Model	Displacement		Max. Pressure		Max. speed (rpm)
		litres/100 rev	US gal/100 rev	bar	PSI	
30	DuraCirc 32	3	0.79	25	362	1000
	DuraCirc 33	6	1.58	25	362	1000
	DuraCirc 34	12	3.17	16	232	1000
40	DuraCirc 42	23	6.07	20	290	750
	DuraCirc 43	29	7.66	13	188	750
50	DuraCirc 52	38	10.03	37	536	750
	DuraCirc 53	59	15.57	25	362	750
	DuraCirc 54	96	25.3	16	232	750
60	DuraCirc 62	144	38.04	37	536	600
	DuraCirc 63	197	52.03	25	362	600
70	DuraCirc 72	192	50.7	40	580	500
	DuraCirc 73	286	75.55	25	362	500
	DuraCirc 74	414	109.4	16	232	500

Table 4.3.3a

Fig. 4.3.1c DuraCirc Aseptic

DuraCirc Aseptic Range

Designed for sterile flushing at all product media to atmosphere interfaces, as well as Cleaning-in-Place (CIP), the Alfa Laval DuraCirc Aseptic is ideal for aseptic processing within the dairy, food, beverage, home, and personal care industries. The highly efficient design is particularly suited to applications that are low in viscosity with medium to high discharge pressures and require equipment that can be cleaned in place (Fig. 4.3.1c).

As with DuraCirc pump the Alfa Laval DuraCirc Aseptic is certified with EHEDG and also conforms to the US 3A standard and all media contacting components are FDA compliant.

The DuraCirc Aseptic Circumferential Piston Pump is available with five different pump head displacements to handle flow rates up to 103 m³/hr (454 US gal/min) and differential pressures up to 25 bar (362 PSIG) (Table 4.3.3b).

Pump Nomenclature

Frame Size	Model	Displacement		Max. Pressure (S or D)		Max. speed (rpm)
		litres/100 rev	US gal/100 rev	bar	PSI	
40	DuraCirc Aseptic 42	23	6.07	20	290	750
50	DuraCirc Aseptic 53	59	15.57	25	362	750
	DuraCirc Aseptic 54	96	25.3	16	232	750
60	DuraCirc Aseptic 63	197	52.03	25	362	600
70	DuraCirc Aseptic 73	286	75.55	25	362	500

Table 4.3.3b

4.4 Twin Screw Pumps

4.4.1 General

The Alfa Laval Twin Screw Pump with non-contacting pump head, designed for handling sensitive, abrasive, and high and low viscosity fluids has the ability to cover a wide range of applications, providing process flexibility. It's hygienic design with smooth, low pulsation characteristics provide excellent solids handling capabilities reducing the risk of product damage. The ability to operate these pumps across a wide speed range makes it capable of handling both product transfer duties and Cleaning-in-Place (CIP) and is widely used in the dairy, food, beverage, home, and personal care industries.

Attributes include:

- Greater process flexibility

- Superior suction performance

- Gentle transfer of delicate suspended solids

- Low pulsation flow

- Ease of service

4.4.2 Principle of Operation

The Alfa Laval Twin Screw Pump is a rotating positive displacement pump which uses two intermeshed screws to convey product in the horizontal axis (Fig. 4.4.1a). As the pump rotates, the intermeshing of the two contra rotating screws along with the pump housing form volumetric chambers. These chambers fill with the pumped fluid and move it axially from the suction side, gradually building up the pressure across the chambers to the higher-pressure discharge side of the pump.

Typically driven via a direct drive motor using a frequency inverter for speed variation provides the flexibility on adapting the speed to meet each duty condition including cleaning in place (CIP) fluids.
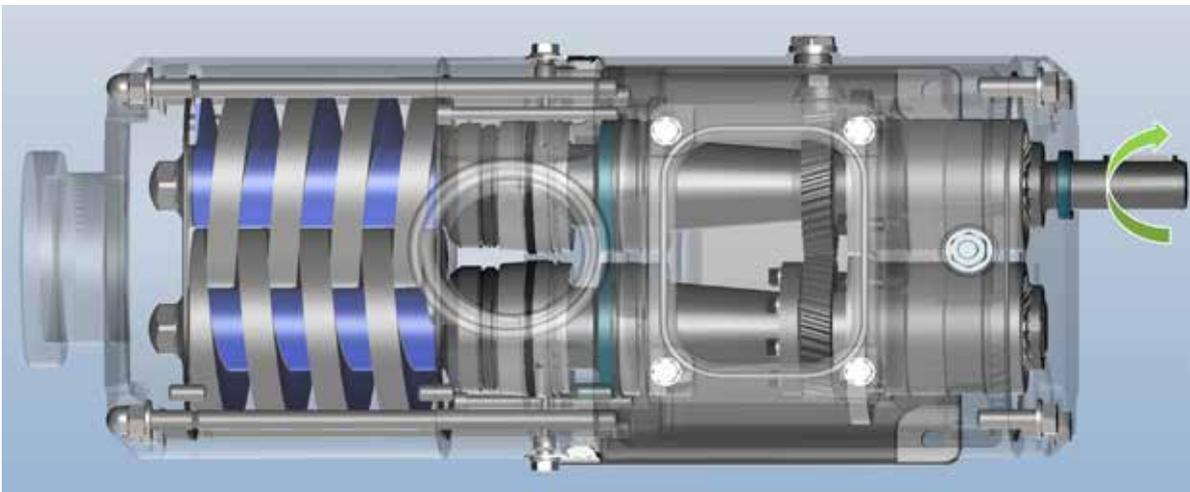
Fig. 4.4.1a Twin Screw axial flow transfer

Fig. 4.4.1b OS Twin Screw

OS Twin Screw Range

The Alfa Laval OS Twin Screw Pump is designed for handling sensitive, abrasive and high and low viscosity fluids, the Alfa Laval Twin Screw Pump is ideal for use in hygienic applications across the dairy, food, beverage, and home and personal care industries. Quiet and virtually pulse-free, the pump provides smooth and gentle operation, making it an excellent choice for handling sensitive products. Two-in-one operation provides easy handling of process media of varying viscosities as well as CIP fluids (Fig. 4.4.1b).

The OS Twin Screw features a clean, external stainless-steel finish with profiled elastomers and mechanical seals fully surrounded by the product. Designed for maximum cleanability using FDA-conforming materials, the pump is both EHEDG- and 3-A certified.

The OS Twin Screw is available in sixteen models based on four frame sizes. The OS10 and OS40 frames have three different screw profiles for varying pressure, flow and solids-handling capabilities, whilst the OS20 & OS30 frames have five different screw profiles. This provides a wide range of performance to enable the user to select the optimal pump for each application. Flow rates up to 150 m³/hr (660 US gal/min) and differential pressures up to 16 bar (232 PSIG).

This chapter describes the materials, both metallic and elastomeric, used in the construction of the Alfa Laval pump portfolio.

5.1 Main Components

Pumps can be manufactured from several varied materials, dependent upon the product being pumped and its environment.

For Alfa Laval, the pump ranges can be split into two main construction categories:

- Product Wetted Parts
  (i.e., metallic, and elastomeric parts in contact with the fluid being pumped)

- Non-product Wetted Parts
  (i.e., metallic, and elastomeric parts not in contact with the fluid being pumped)

5.0

Pump Materials of Construction

1. Casing
2. Adaptor
3. Shroud

Fig. 5.1a LKH Centrifugal pump

Centrifugal Pumps - LKH ranges		
Main Pump Component	Product Wetted Parts	Non-product Wetted Parts
Adaptor		AISI 304 or Werkstoff 1.4301
Backplate	AISI 316L or Werkstoff 1.4404	
Impeller	AISI 316L or Werkstoff 1.4404	
Pump Casing	AISI 316L or Werkstoff 1.4404	
Pump Shaft	AISI 316L or Werkstoff 1.4404	
Shroud and Legs		AISI 304 or Werkstoff 1.4301

Table 5.1a

1. Casing
2. Shroud
3. Legs

Fig. 5.1b SolidC Centrifugal pump

Centrifugal Pumps - SolidC + SolidC UP ranges		
Main Pump Component	Product Wetted Parts	Non-product Wetted Parts
Adaptor		AISI 304 or Werkstoff 1.4301
Backplate	AISI 316L or Werkstoff 1.4404	
Impeller	AISI 316L or Werkstoff 1.4404	
Pump Casing	AISI 316L or Werkstoff 1.4404	
Pump Shaft	AISI 316L or Werkstoff 1.4404	
Shroud		AISI 304 or Werkstoff 1.4301
Legs		Special - Plastic Coated

Table 5.1b

1. Casing
2. Adaptor
3. Backplate

Fig. 5.1c ICP2000 Centrifugal pump

Centrifugal Pumps - ICP2000 range (authorised channels only)		
Main Pump Component	Product Wetted Parts	Non-product Wetted Parts
Adaptor		Painted Zinc Coated Cast Steel
Backplate	AISI 316L or Werkstoff 1.4404	
Impeller	AISI 316L or Werkstoff 1.4404	
Pump Casing	AISI 316L or Werkstoff 1.4404	
Pump Shaft	AISI 316L or Werkstoff 1.4404	
Shroud		AISI 304 or Werkstoff 1.4301
Legs		Special - Plastic Coated

Table 5.1c

1. Product seal area
2. Rotor case
3. Rotor case cover
4. Ports
5. Gearbox
6. Drive shaft

Fig. 5.1d SRU Rotary lobe pump

Rotary lobe pumps - SRU + SX ranges				
Main Pump Component	SRU Models		SX + SX UP Models	
	Metallic Product Parts	Metallic Non-product Wetted Parts	Metallic Product Wetted Parts	Metallic Non-product Wetted Parts
Gear Case		AISI 304 or Werkstoff 1.4301		AISI 304 or Werkstoff 1.4301
Rotor	Werkstoff 1.4404 or AISI 316L/Non-Galling Alloy ASTM A-494		Werkstoff 1.4404 or AISI 316L	
Rotor Case	EN 1.4409 (CF-3) or AISI 316C12/Werkstoff 1.4404 or AISI 316L		EN 1.4409 or 316C12/ Werkstoff 1.4404 or AISI 316L	
Rotor Case Cover	Werkstoff 1.4404 or AISI 316L		Werkstoff 1.4404 or AISI 316L	
Shaft	Werkstoff 1.4404 or 316L Duplex stainless steel (AISI 329 or grade 1.4462)		Duplex stainless steel (AISI 329 or grade 1.4462)	

Table 5.1d

1. Product seal area
2. Gearbox
3. Ports
4. Rotor case cover

Fig. 5.1d Circumferential piston pump

Circumferential piston pumps - DuraCirc + DuraCirc Aseptic		
Main Pump Component	Metallic Product Wetted Parts	Metallic Non-product Wetted Parts
Gear Case		AISI 304 or Werkstoff 1.4301
Rotor	Non-Galling Alloy/ASTM A-494	
Rotor Case	Werkstoff 1.4404 or AISI 316L/EN 1.4409 (CF-3) or AISI 316C12	
Rotor Case Cover	Werkstoff 1.4404 or AISI 316L	
Shaft	Duplex stainless steel (AISI 329 or grade 1.4462)	

Table 5.1d

1. Product seal area
2. Gearbox
3. Ports
4. Rotor Case

Fig. 5.1e Twin Screw pump

Twin screw pumps - OS		
Main Pump Component	Metallic Product Wetted Parts	Metallic Non-product Wetted Parts
Gear Case		EN 1.4408 (CF-8) or 316 C16
Screws	Werkstoff 1.4404 or AISI 316L Optional: Diffusion hardened - 1092 HV0.05	
Casing	Werkstoff 1.4404 or AISI 316L Diffusion hardened - 1092 HV0.05	
Case Cover	Werkstoff 1.4404 or AISI 316L	
Shafts	Werkstoff 1.4404 or AISI 316L	

Table 5.1e

5.2 Stainless Steel

Hygienic demands have led stainless steel to be accepted as the top material choice within the food, beverage & pharmaceutical processing/services for product equipment.

This equipment must possess product integrity (no corrosion or leaks), cleanliness, ease of cleaning and ease of inspection. The equipment must also be able to resist chemical solutions used in cleaning, such as chloride-based sterilising agents, acids, and alkalis.

Stainless Steel Name	Composition	Characteristics	Application areas
Alloy 304	Also known as 18/8 for its composition of 18% chromium and 8% nickel	Excellent in a wide range of atmospheric environments and many corrosive media. Subject to pitting and crevice corrosion in warm chloride environments, and to stress corrosion cracking approximately above 60° C	Non to low chloride containing water applications, nitric acid, and oleum
Alloy 316	A composition of 18% chromium and 10% nickel and 2% of molybdenum	The 316 grade is used instead of 304 in cases where higher resistance to pitting and crevice corrosion is required, in chloride environments	Versatile material that is used in a very wide range of applications like; food, dairy, beverage, organic acids and pharmaceutical, to mention a few

Table 5.2a Stainless Steel Alloys

Stainless steel meets all these requirements for pump designs with the most commonly grades within hygienic applications being austenitic types; SS (Stainless Steel) 304 (1.4301), 316 (1.4401) and 316L (1.4404) (Table 5.2a).

Differences between SS 304 and SS 316:

- Both 304 and 316 are easily welded and formed

- Both types are non-magnetic

- 304 contains 18% chromium 8% nickel
  316 contains 17% chromium

- 304 contains no trace of molybdenum
  316 contains 2.1% molybdenum

The biggest difference between the grades is molybdenum is added to SS 316 to increase corrosion and pitting resistance.

SS 316L is almost identical to SS 316. The only difference is the lower carbon content with SS 316 having maximum value of 0.08% and SS 316L having maximum value of 0.03%.

Duplex Steel

Certain applications within targeted industries pose challenges that cannot be met by austenitic stainless steel alone.

Duplex stainless steels (AISI 329) are extremely corrosion resistant alloys. Their microstructures consist of a mixture of austenite and ferrite phases. As a result, duplex stainless steels display properties characteristic of both austenitic and ferritic stainless steels.

Strengths of duplex stainless steels can in some cases be double that for austenitic stainless steels, making it an ideal choice of alloy for the construction of pump shafts to handle higher pressure applications.

Whilst duplex stainless steels are considered resistant to stress corrosion cracking, they are not as resistant as ferritic stainless steels. However, the corrosion resistance of the least resistant duplex stainless steels is greater than that for the most used grades of stainless steels, i.e., 304 and 316.

Duplex stainless steels have good weldability. All standard welding processes can be used. They are not as easily welded as the austenitic grades, but low thermal expansion in duplex grades reduces distortion and residual stresses after welding.

Duplex steels are also magnetic, a property that can be used to easily differentiate them from common austenitic grades of stainless steel.

Please contact Alfa Laval on the possibility of providing a hygienic pump manufactured of Duplex steel or other exotic alloys suited for your requirements.

For description of elastomers used see section 5.3. For mechanical seal components see section 6.1.

Surface finish of product wetted steel components is a major factor in the food, pharmaceutical and biotechnology industries where hygiene and cleanability are of paramount importance.

5.3 Stainless Steel Surfaces

The 'standard' machined surface finish on pumps can be enhanced by the following methods:

- Rumbling
- Shot blasting
- Electropolishing
- Mechanical (Hand) polishing

Rumbling

This is achieved by vibrating the pump components with abrasive particulate such as stones and steel balls.

Shotblasting

This method involves blasting finished components with small metallic particles at great force to achieve the surface finish required. For Alfa Laval centrifugal, stainless steel pump components, fine particles of stainless steel are used in this process to avoid contamination.

Electropolishing

This is an electro-chemical process in which the stainless steel component is immersed into a chemical bath and subjected to an electrical current. A controlled amount of metal is removed from all surfaces evenly. The appearance is 'Semi bright.'

Mechanical (Hand)

This is required when it is necessary to reach a certain Ra surface finish for customers beyond that achieved by electropolishing only i.e., a 'Mirror finish'.

It typically involves:

- Fine grinding using felt and compound
- Brushing using bristle brushes and compound to remove any cutting marks left from fine grinding, and to reach any awkward areas
- Polishing using mops and compound to obtain a mirror polished effect

For Alfa Laval Centrifugal Pumps see table below:			
Pump surfaces	Standard surface roughness Ra (mm) by Rumbling method	Optional surface roughness (3A finish) Ra (mm) by Mechanical (Hand) method	Optional surface roughness (3A finish) Ra (mm) by shot blasting (Hand or Electropolished)
Product wetted surfaces	< 1.6 (64 Ra)	< 0.8 (32 Ra)	< 0.5 (20 Ra)
External exposed surfaces	< 1.6 (64 Ra)	< 1.6 (64 Ra)	< 1.6 (64 Ra)
Cast surfaces	< 3.2 (125 Ra)	} 3.2 (125 Ra)	} 3.2 (125 Ra)
Other surfaces	} 6.3 (250 Ra)	} 6.3 (250 Ra)	} 6.3 (250 Ra)

Table 5.3a
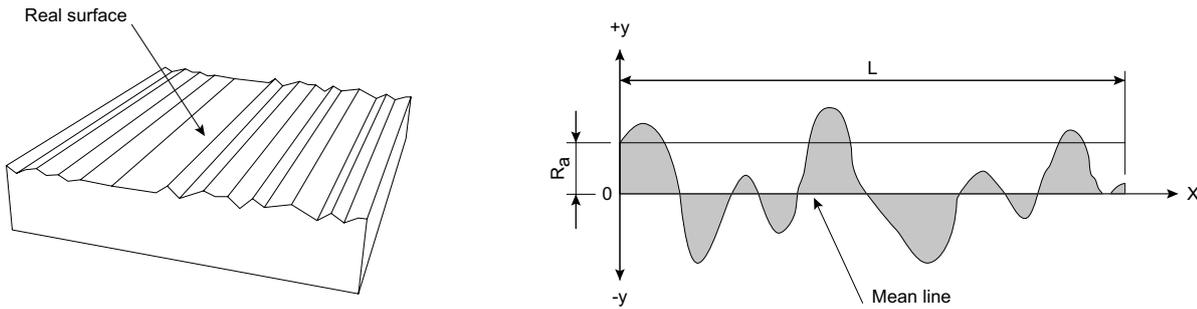
Fig. 5.3a Surface roughness

Surface Roughness

The most used surface roughness measurement is Ra and is defined as 'the arithmetic mean of the absolute value of the departure of the profile from the mean line' (Fig. 5.3a). Ra is measured in micron (μm). The surface roughness can alternatively be specified by a Grit value. The Grit value specifies the grain size of the coating of the grinding tool used.

The approximate connection between the Ra value and the Grit value is as follows:

Ra = 0.8 μm (32 Ra) ≈ 150 Grit (3A standard)

Ra = 1.6 μm (64 Ra) ≈ 100 Grit

Alfa Laval Centrifugal pumps supplied in the US have all product wetted surfaces and external exposed surfaces to 0.8 Ra.

For Alfa Laval Rotary Lobe Pumps the surface roughness on product wetted parts such as rotors, rotor case, rotor nuts and rotor case covers is as follows:

'Standard'	0.8	Ra
Electropolishing	0.8	Ra
Mechanical (Hand)	0.38	Ra

Passivation

The passivation of stainless steel is a process performed to make a surface passive, i.e., a surface film is created that causes the surface to lose its chemical reactivity.

Stainless steel passivation unipotentialises the stainless steel with the oxygen absorbed by the metal surface, creating a monomolecular oxide film. This process returns the stainless steel back to its original specifications. When a part is machined, various particles can permeate the surface of the base metal, weakening its resistance to corrosion and making the part more susceptible to environmental factors. Debris, dirt and other particles and residue such as free iron, grease, and machining oils all affect the strength of the natural surface and can become embedded in the surface during the machining process. These often go unseen to the human eye and are often the cause of the deterioration. The passivation process improves and purifies the surface of the part. The restored surface acts as a protective coating to environmental factors such as air, water, and other extreme environments.

It is important to mention that passivation does not change the outward appearance of the base metal.

Advantages of Passivation

- Improved corrosion resistance

- Uniform, smooth appearance & finish

- Cleanliness

- Improved & extended life of product

Note: Passivation is also accomplished by electropolishing.

5.4 Elastomers

Alfa Laval pump ranges incorporate elastomers of different material and characteristics dependent upon application within the pump and the fluid being pumped.

Various elastomer types are specified below. It is difficult to predict the lifetime of elastomers as they will be affected by many factors, e.g., chemical attack, temperature, mechanical wear etc.

It is important to note all Alfa Laval pumps are supplied with FDA, 3A EC1935/2004 and EC2023/2006 conforming elastomer grades as standard.

A selection guide is shown in section 14.10.

EPDM (Ethylene Propylene)

This is a gasket material with excellent heat resistance. It is resistant to oxidization, acids, bases, and tough CIP (Cleaning-in-Place).

- Used as static or dynamic seals
- Resistant to most products used within the food industry

- Resistant to ozone and radiation
- Temperature range - min. -40° C to max. +150° C (min. -40° F to max. +302° F)
- Not resistant to organic & non-organic oils/greases and aliphatic, aromatic, chlorinated hydrocarbons

FPM/FKM (Fluorinated rubber)

Alternatively known as Viton®. FPM is a fluoro rubber that has excellent chemical resistance to a very wide array of substances. It has excellent resistance to oils including aromatics, acids, oxidization, and heat.

- Used as static or dynamic seals
- Often used when other rubber qualities are unsuitable
- Resistant to most chemicals and ozone
- Temperature range - min. -20° C to max. +200° C (min. -4° F to max. +392° F)
- Not suitable for fluids such as steam, lye, acid, and high temperature alcohol

FFPM/FFKM (Perfluoroelastomer)

Perfluoroelastomers contain an even higher amount of fluorine than FKM and is typically used in applications where compatibility is limited to standard offerings.

- Used as static or dynamic seals
- Resistant to ozone and almost all products

- Temperature range - min. -20° C to max. +260° C (min. -4° F to max. +500° F)

- More elastic than PTFE (Polytetrafluoro Ethylene)

- Not suitable with molten alkali metals

PTFE (Polytetrafluoro Ethylene)

Polytetrafluoroethylene is a synthetic fluoropolymer of tetrafluoroethylene that has numerous application uses. It is also known by common trade name TEFLON™.

- Used as static seals

- Resistant to ozone and almost all products

- Temperature range - min. -30° C to max. +200° C (min. -22° F to max. +392° F)

- Not elastic, tendency to compression set

MVQ (Silicone)

MVQ is a silicone rubber with suitable properties for both high and low temperatures. Due to hydrolysis (cleavage of chemical bonds by the addition of water) its resistance is limited to acids, bases, and steam

- Used as static or dynamic seals

- Resistant to ozone, alcohol, glycols, and most products used within food industry

- Temperature range - min. -50° C to max. +230° C (min. -58° F to max. +446° F)

- Not resistant to steam, inorganic acids, mineral oils, or most organic solvents

FEP (Fluorinated Ethylene Propylene)

- FEP covered (vulcanised) FPM or MVQ O-rings

- Used as static or dynamic seals

- Resistant to ozone

- Resistant to almost all products

- Suitable for temperatures up to approx. 200° C (392° F)

- More elastic than PTFE

Alternative options upon request

Kalrez®/Chemraz® (Perfluoroelastomer)

Like FFPM, these Perfluoros are trademarked and have numerous material grades that can cover all conceivable application requirements.

- Used as static or dynamic seals

- Resistant to ozone

- Resistant to almost all products

- Temperature range – min. -40° C to max. +365° C (min. -40° F to max. +689° F) dependent on grade

- Elastic

This chapter describes the principle of pump sealing and illustrates the different sealing arrangements used on Alfa Laval pump ranges. A general seal selection guide is included, along with various operating parameters.

This chapter covers the primary shaft sealing devices used on Alfa Laval Centrifugal, Rotary Lobe, Internal Gear, Circumferential Piston and Twin Screw pumps. Other proprietary seals not detailed in this chapter, such as O-rings and lip seals can be found on the pump head and gear case.

"A Pump is only as good as its shaft seal"

A successful pump application largely depends upon the selection and application of suitable fluid sealing devices. Just as we know that there is no single pump that can embrace the diverse range of fluids and applications whilst meeting individual market require-ments and legislations, the same can be said of fluid sealing devices. This is clearly illustrated by the large range of shaft seal arrangements, such as mechani-cal, lip or O-ring seals, that are available to the pump manufacturer.

6.0

Pump Sealing

Shaft sealing devices used in the various pump technologies in the Alfa Laval pump portfolio include:

Mechanical Seals

- Single externally mounted

- Single internally mounted

- Single externally mounted for external flush

- Single internally mounted for product recirculation or external flush

- Double 'back-to-back' with the inboard seal externally mounted for flush

O-ring seals

- Single

- Single with flush

Centrifugal pumps only have one shaft seal whereas Rotary Lobe, Internal Gear, Circumferential Piston and Twin Screw pumps employ a minimum of two shaft seals (one per shaft). Generally, all shaft seals are under pressure with the pressure gradient across the seal being from pumped fluid to atmosphere. The exception to this is with a double seal, where the flush pressure can be greater than the differential pressure in the pump chamber. This results in the pressure gradient being reversed.

Mechanical seals meet the majority of application demands and of these, single and single flushed seals are most frequently specified. The application of double mechanical seals is increasing to meet both process demands for higher hygienic standards and legislation requirements, particularly those related to emissions.

The majority of proprietary mechanical seals available from seal manufacturers have been designed for single shaft pump concepts, for example Centrifugal pumps. Such pump types do not impose any radial or axial constraints on seal design. However, on Rotary Lobe, Circumferential Piston and Twin Screw pumps the need to minimise the shaft extension beyond the front bearing places significant axial constraints. If this were extended, the shaft diameter would increase introducing a radial constraint - because shafts on the above referenced pump technologies are in the same plane, the maximum diameter of the seal must be less than the shaft centres. Most designs therefore can only accommodate 'bespoke' or 'customised' seal design. This is not done to take any commercial advantage, rather it is as a consequence of these pumps design concept.

There is often more than one solution and sometimes no ideal solution, therefore a compromise may have to be considered.

Selection of shaft seals is influenced by many variables:

- Shaft diameter and speed

- Fluid to be pumped

Temperature	-	effect on materials?
	-	can interface film be maintained?
Viscosity	-	drag on seal faces?
	-	clogging of seal restricting movement?
	-	can interface film be established and maintained?
	-	stiction at seal faces?
Fluid behaviour	-	does product shear, thin, thicken or 'work' - balling/carbonise?
	-	can interface film be established and maintained?
Solids	-	size?
	-	abrasiveness?
	-	density?
	-	clogging of seal restricting movement?
	-	can interface film be established and maintained?
Thermal stability	-	what, if any change?
Air reacting	-	what, if any change?

- Pressure
  - within seal limits?
  - fluctuations?
  - peaks/spikes?
  - cavitation?

- Services
  - flush?
  - pressure?
  - temperature?
  - continuity?

- Health and safety
  - toxic?
  - flammable?
  - explosive?
  - corrosive?
  - irritant?
  - carcinogenic?

6.1 Mechanical Seals

General

Mechanical seals are designed for minimal leakage and represent the majority of Centrifugal, Rotary Lobe, Internal Gear, Circumferential Piston and Twin Screw pump sealing arrangements (Table 6.1a).

Mechanical seal selection must consider:

* The materials of seal construction, particularly the sealing faces and elastomers

* The mounting attitude to provide the most favourable environment for the seal

* The geometry within which it is to be mounted

A mechanical seal typically comprises of:

* A primary seal, comprising of stationary and rotary seal rings

* Two secondary seals, one for each of the stationary and rotary seal rings

* A method of preventing the stationary seal ring from rotating

* A method of keeping the stationary and rotary seal rings together when they are not hydraulically loaded i.e., when pump is stopped

* A method of fixing and maintaining the working length

The Primary Seal

Comprises of two flat faces, one rotating and one stationary, which support a fluid film, thus minimising heat generation and subsequent mechanical damage.

Commonly used material combinations are:

Carbon	-	Stainless Steel
Carbon	-	Silicon Carbide
Silicon Carbide	-	Silicon Carbide

The Secondary Seal

This is required to provide a seal between the primary seal rings and the components with which they interface. Also, it can provide a cushion mounting for the seat ring to reduce any effects of mechanical stress i.e., shock loads.

Types of secondary seal are:

* O-rings

* Cups

* Gaskets

* Wedges

For Alfa Laval pump ranges an O-ring or profiled elastomer is the most commonly type of secondary seal used. This is a simple and versatile solution and (dependent on range) is offered in the following comprehensive material options:

* NBR

* EPDM

* FPM

* PTFE

* FEP

* FFPM

* Silicone

Note:
USP class VI compliant variants are an available option on certain ranges, in certain materials.

These are fully described in chapter 11.

Mechanical Seal Face/'O' Ring Material Availability

Pump Type	Pump Range	Rotary Seal Face			Stationary Seal Face			Elastomer						
		Carbon	Stainless Steel	Silicon Carbide	Carbon	Stainless Steel	Silicon Carbide	NBR	EPDM	FPM	PTFE	FFPM	Silicone	FEP
Centrifugal	LKH/LKH Evap	✓		✓			✓	✓	✓	✓				✓
	LKH Prime	✓		✓			✓	✓	✓	✓				
	LKH Multistage	✓		✓			✓	✓	✓	✓				
	LKHPF			✓			✓	✓	✓	✓				
	LKHI	✓		✓			✓	✓	✓	✓				
	LKH UltraPure			✓			✓		✓	✓				✓
	LKH Prime UltraPure			✓			✓		✓	✓				✓
	SolidC	✓		✓			✓	✓	✓	✓	✓			
	SolidC UltraPure			✓			✓		✓	✓				✓
	FM	✓		✓			✓	✓	✓	✓				✓
	GM	✓		✓			✓	✓	✓	✓				
Rotary Lobe	OptiLobe	✓		✓		✓	✓		✓	✓				
	SRU		✓	✓	✓		✓	✓	✓	✓		✓		
	SX	✓		✓		✓	✓		✓	✓			✓	
	SX UltraPure			✓			✓		✓	✓			✓	
Circumferential Piston	DuraCirc			✓	✓		✓		✓	✓		✓		
	DuraCirc Aseptic			✓			✓		✓	✓				
Twin Screw	OS	✓		✓			✓		✓	✓			✓	
Internal Gear	M	✓	✓	✓	✓		✓		✓	✓				
Note 1: LKH Multistage 120 only has EPDM & NBR elastomer option														
Note 2: SX1 pump has tungsten carbide seal faces, not silicon carbide seal faces														

Table 6.1a
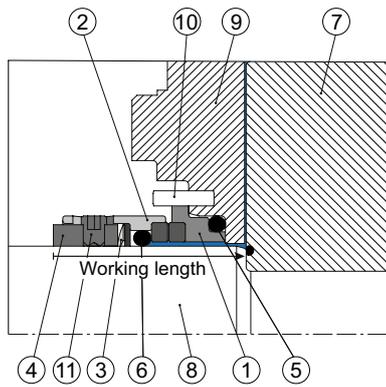
Fig. 6.1a Typical single mechanical seal used in rotary lobe pumps

Process media

1. Stationary Seal ring
2. Rotary Seal ring
3. Wave Spring
4. Rotary Seal drive ring
5. Stationary Seal drive O-ring
6. Rotary Seal O-ring
7. Rotor
8. Shaft
9. Rotorcase
10. Stationary Seal Ring
    Anti-rotation pin
11. Grub screw

Stationary Seal Ring Anti-Rotation

Ideally the selected device listed below will also allow for axial resilience.

- Flats

- Pins

- Elastomer resilience

- Press fit

- Clamps

Rotary Seal Ring Drive

Ideally the selected device listed below will allow for a degree of axial movement.

- Spring

- Bellows

- Physical positioning

- Elastomer resilience
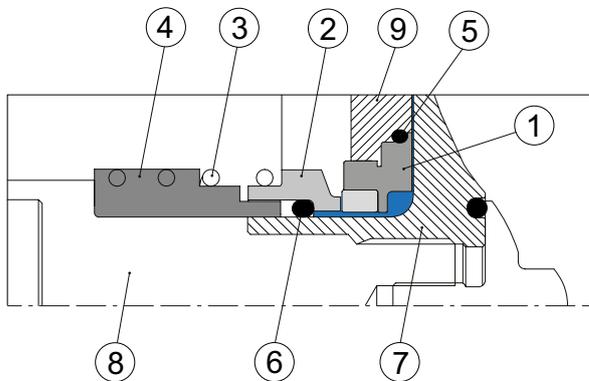
Fig. 6.1b Typical single mechanical seal used in centrifugal pumps

Process media

1. Stationary Seal ring
2. Rotary Seal ring
3. Spring
4. Drive ring
5. Stationary Seal elastomer
6. Rotary Seal elastomer
7. Impeller
8. Pump shaft
9. Backplate

Working Length

The ideal design should eliminate/minimise possibilities for error by incorporating (Fig. 6.1a):

- Physical position i.e., step on shaft

- Grub screws

Note:

Some Alfa Laval mechancial seals are self-setting by design, whereby working length is autoamtically set.

One of the main causes of seal failure is for the seal working length not being correctly maintained.
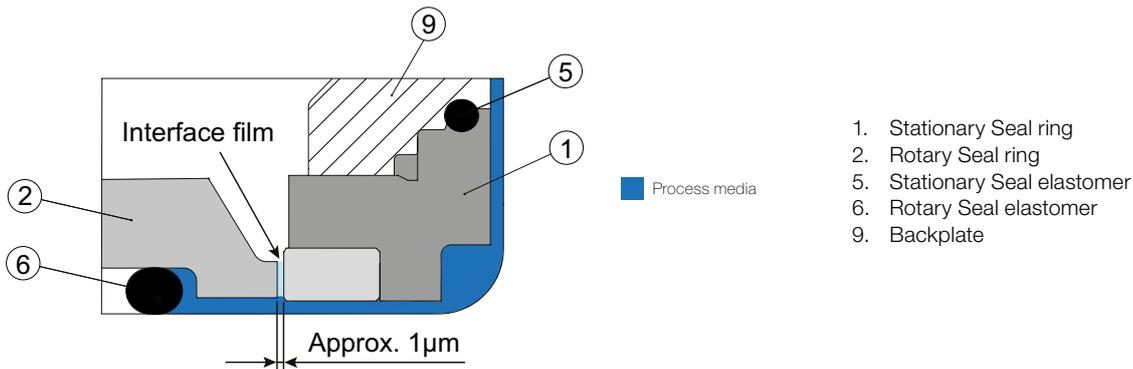
Fig. 6.1c Principle of mechanical seal operation

1. Stationary Seal ring
2. Rotary Seal ring
5. Stationary Seal elastomer
6. Rotary Seal elastomer
9. Backplate

Process media

Principle of Mechanical Seal Operation

During pump operation, hydraulic fluid forces combine with seal design features, including applied spring force, extremely flat sealing surfaces and seal face geometry, which push the seal faces together. This reduces the fluid interface thickness, also called the interface film, to a minimum, typically 1 µm, as shown in Fig 6.1c, whilst increasing pressure drop, therefore minimising pumped fluid leakage.

Mechanical Seal Mounting

Mechanical seals can be mounted externally or internally.
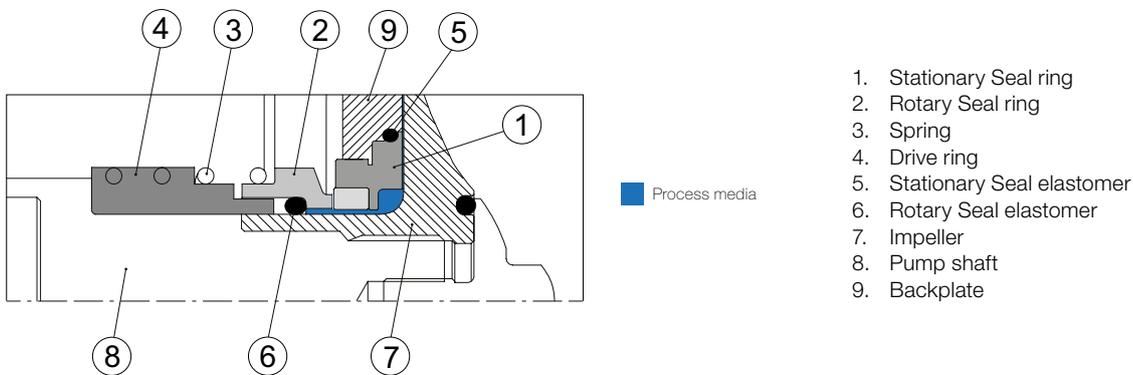
Fig. 6.1d Typical external shaft seal

1. Stationary Seal ring
2. Rotary Seal ring
3. Spring
4. Drive ring
5. Stationary Seal elastomer
6. Rotary Seal elastomer
7. Impeller
8. Pump shaft
9. Backplate

Process media

External Mounted Mechanical Seals

In an externally mounted mechanical seal, the seal is designed such that the direction of fluid flow across the seal faces is from the internal to external seal face diameter, i.e., the fluid is sealing inside to out, as shown in Fig. 6.1d.

Pumps having externally mounted mechanical seals include LKH, LKH UltraPure, LKHex, LKH Prime, LKH Prime UltraPure, Solid C, Solid C UltraPure and SRU.
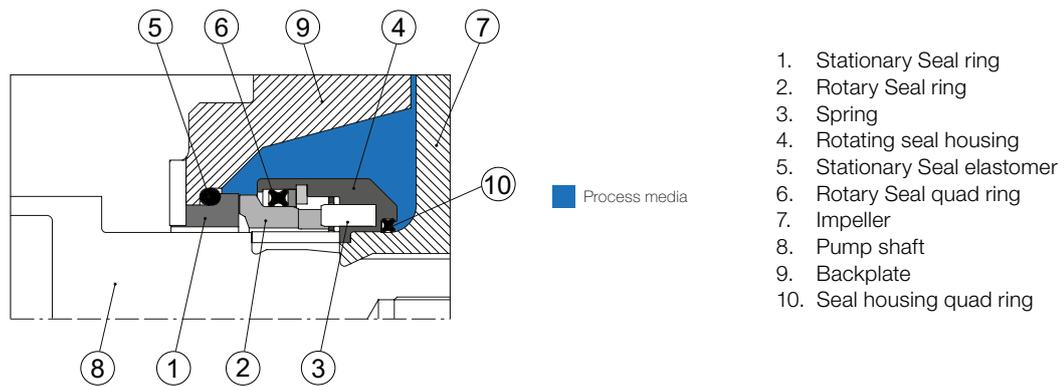
Fig. 6.1e Typical internal shaft seal

1. Stationary Seal ring
2. Rotary Seal ring
3. Spring
4. Rotating seal housing
5. Stationary Seal elastomer
6. Rotary Seal quad ring
7. Impeller
8. Pump shaft
9. Backplate
10. Seal housing quad ring

Process media

Internal Mechanical Seals

In an internally mounted mechanical seal, the seal is designed such that the direction of fluid flow across the seal faces is from the external to internal seal face diameter, i.e., the fluid is sealing outside to in, as shown in Fig 6.1e.

Pumps having internally mounted mechanical seals include LKHI, LKH Multistage, LKHPF, OptiLobe, SX, SX UltraPure, DuraCirc, DuraCirc Aseptic and OS Twin Screw.

In Alfa Laval hygienic pumps, dependent on range, both the externally and internally mounted types of mechanical seal are available as single and single flushed, as well as double versions.

The arrangements are described as follows:

- Single Mechanical Seal

- Single Flushed Mechanical Seal

- Double Flushed Mechanical Seal

1. Stationary Seal ring
2. Rotary Seal ring
3. Wave Spring
4. Rotary Seal drive ring
5. Stationary Seal drive O-ring
6. Rotary Seal O-ring
7. Rotor
8. Shaft
9. Rotorcase
10. Stationary Seal Ring
    Anti-rotation pin
11. Grub screw
12. Lip seal
13. Flush housing
14. Seal abutment spacer

Process media
Flush media
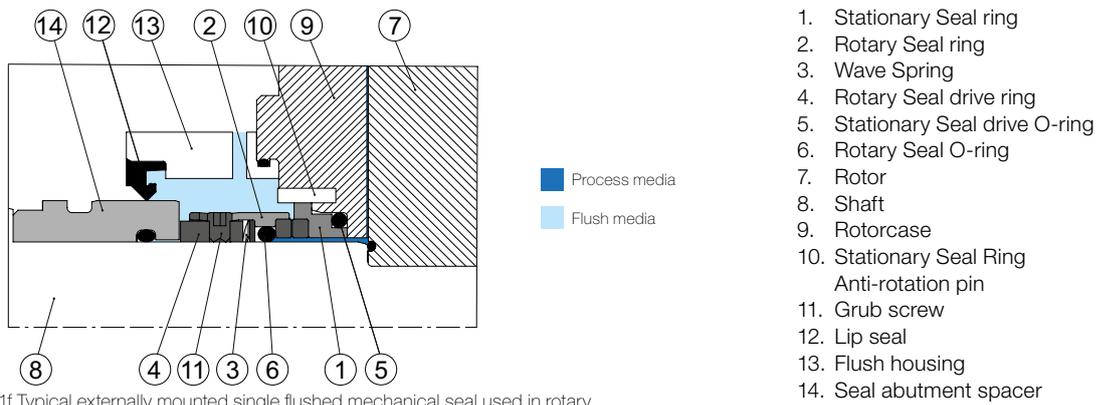
Fig. 6.1f Typical externally mounted single flushed mechanical seal used in rotary lobe pumps
Seal shown: SRU R90 single flushed seal

Single Mechanical Seal

This is the simplest shaft seal version, which has already been described previously in this chapter. This seal arrangement is generally used for fluids that do not solidify or crystallise in contact with the atmosphere and other non-hazardous duties (Fig. 6.1f). For satisfactory operation it is imperative the seal is not subjected to pressures exceeding the maximum rated pressure of the pump. Also, the pump must not be allowed to run 'dry', thus avoiding damage to the seal faces, which may cause excessive seal leakage.

Typical applications are listed below, but full product/ fluid and performance data must be referred to the seal supplier for verification.

- Alcohol
- Animal Fat
- Aviation Fuel
- Beer
- Dairy Creams
- Fish Oil
- Fruit Juice
- Liquid Egg
- Milk
- Shampoo
- Solvents
- Vegetable Oil
- Water
- Yoghurt

Fig. 6.1g Single Flushed Mechanical Seal

Process media

Flush media

1. Stationary Seal ring
2. Rotary Seal ring
3. Spring
4. Drive ring
5. Stationary Seal elastomer
6. Rotary Seal elastomer
7. Impeller
8. Pump shaft
9. Backplate
10. Flush housing lip seal
11. Flush housing

Single Flushed Mechanical Seal

The definition of 'flush' is to provide a liquid barrier or support to the selected seal arrangement. This seal arrangement is generally used for any of the following conditions:

- Where the fluid being pumped can coagulate, solidify, or crystallise when in contact with the atmosphere

- When cooling of the seals is necessary dependent upon the fluid pumping temperature

- In partial vacuum applications, where a barrier to atmosphere is required

This seal arrangement used on both externally as well as internally mounted seals requires the supply of liquid to the atmospheric side of the mechanical seal to flush the seal area (Fig. 6.1g). The characteristics of the fluid being pumped, and the duty conditions will normally determine if a flush is necessary. When selecting a flushing liquid, you must ensure that it is chemically compatible with the relevant materials of pump/seal construction and fully compatible with the fluid being pumped. Consideration should be given to any temperature limitations that may apply to the flushing liquid to ensure that hazards are not created (i.e., explosion, fire, etc.).

The flushing liquid is usually sealed from external atmosphere via a lip seal. In Alfa Laval pumps, the flushing liquid should enter the seal housing at a low pressure, with allowable pressure being up to 0.5 bar (7.5 PSI) maximum. Should the flush pressure be higher, there is a risk of blowing out the lip seal, thus allowing flush media leakage to atmosphere.

This most basic flush system, sometimes referred to as quench, provides liquid to the atmosphere side of the mechanical seal thereby flushing away any product leakage. For the majority of pump models, the flushed seal comprises of the same stationary and rotating parts as the single seal, with the addition of a seal housing having a flushing connection plus the afore-mentioned lip seal.

Note:
Flush housing on SX/SX-UP, DuraCirc as well as OS Twin Screw are integral to the rotor case/casing.

Typical applications are listed below, but full product/fluid and performance data must be referred to the seal supplier for verification.

- Adhesive

- Caramel

- Detergent

- Fruit Juice Concentrate

- Gelatine

- Jam

- Latex

- Paint

- Sugar Syrup

- Toothpaste

- Yeast

1. Stationary Seal ring inboard
2. Rotary Seal ring assembly inboard
3. Wave Spring
4. Drive ring
5. Stationary Seal O-ring inboard
6. Rotary Seal O-ring inboard
7. Rotor
8. Shaft
9. Rotorcase
10. Rotary Seal ring assembly outboard
11. Stationary Seal ring outboard
12. Rotary Seal O-ring outboard
13. Stationary Seal O-ring outboard
14. Flush housing

Process media

Flush media
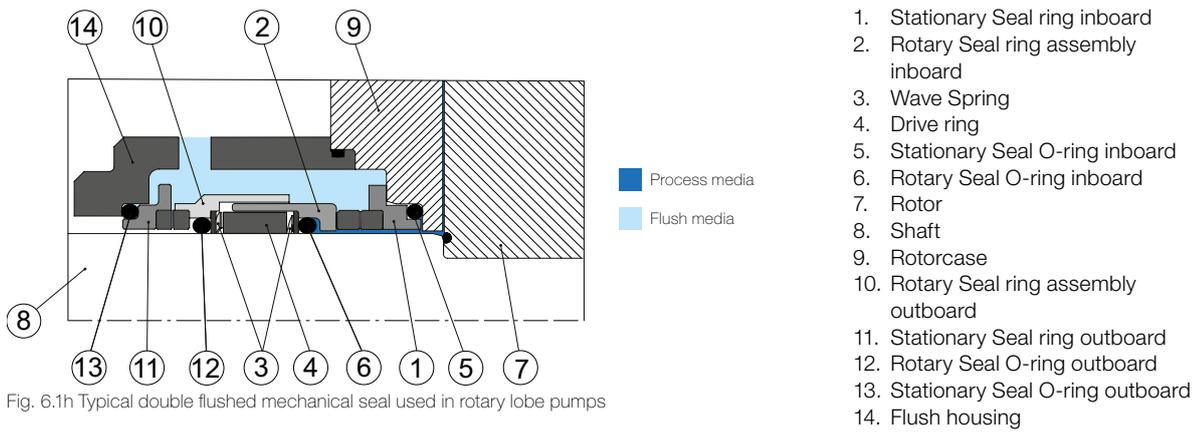
Fig. 6.1h Typical double flushed mechanical seal used in rotary lobe pumps

Double Flushed Mechanical Seal

This seal arrangement is generally used with hostile media conditions i.e., high viscosity, fluid is hazardous or toxic. Additionally, it can also be used on applications where cleaning is via a SIP process, where the steam condensate around the sealing area needs to be at a pressure higher than 0.5 bar (7.5 PSI). The double flushed seal used on Alfa Laval pump ranges is basically two single mechanical seals mounted 'back-to-back' (Fig. 6.1h). This seal generally comprises of the same stationary and rotating parts as the single seal for the majority of pump models, with the addition of a seal housing having a flushing connection and/or flushing tubes (dependent upon pump type).

Note:
Flush housing on SX/SX-UP, DuraCirc as well as Twin Screw are integral to the rotor case/casing).

A compatible flushing liquid is pressurised into the seal housing at a pressure of 1 bar (14 PSI) minimum above the discharge pressure of the pump. This results in the interface film being the flushing liquid and not the pumped liquid. Special attention is required in selecting seal faces and elastomers.

It is also possible for a double seal to operate in low pressure flush mode, i.e., where the flush pressure is at a similar pressure to that used with single flushed seals of 0.5 bar, or at a pressure lower than pump differential pressure. In this situation, the interface film will be pumped media, rather than flushed media (Fig. 6.1i).

The arrangement in contact with the pumped fluid is referred to as the 'inboard seal', and the seal employed for the flushing liquid is referred to as the 'outboard seal'. For Alfa Laval centrifugal pumps the design of the outboard seal differs to the inboard seal.

1. Inboard Stationary Seal ring
2. Inboard Rotary Seal ring
3. Spring
4. Outboard Rotary Seal ring
5. Stationary inboard
   Seal elastomer
6. Rotary Inbound Seal elastomer
7. Impeller
8. Pump shaft
9. Backplate
10. Outboard Stationary Seal ring
11. Flush housing
12. Rotary Outboard Seal elastomer
13. Stationary Outboard
    Seal elastomer
14. Flush housing elastomer

Process media
Flush media

Fig. 6.1i Typical double flushed mechanical seal used in centrifugal pumps

Typical applications are listed below, but full product/ fluid and performance data must be referred to the seal supplier for verification (Table 6.1b).

- Abrasive Slurries

- Chocolate

- Glucose

- Hazardous Chemicals

- PVC Paste

- Photographic Emulsion

- Resin

Viscosity	Seal Face Combination
Up to 4999 cP	Solid Carbon v Stainless Steel Solid Carbon v Silicon Carbide
Up to 24999 cP	Inserted Carbon v Stainless Steel Inserted Carbon v Silicon Carbide
Up to 149999 cP	Silicon Carbide v Silicon Carbide
Above 150000 cP	Consider Double Seals

Table 6.1b

Temperature	Seal Face Combination
Up to 150° C (302° F)	Inserted Carbon v Stainless Steel Inserted Carbon v Silicon Carbide Silicon Carbide v Silicon Carbide
Up to 200° C (392° F)	Solid Carbon v Stainless Steel Inserted Carbon v Silicon Carbide

Table 6.1c

General Seal Face Operating Parameters

The tables above show general parameters regarding viscosity and temperature, which should be noted when selecting a mechanical seal.

A: Pressure gauge
B: Control valve
C: Check valve
D: Isolation valve

* Double mechanical seal only

Flush Inlet

Flush outlet to waste

Suggested visible
indication of flow

1. Pressure gauge*
2. Control valve*
3. Suggested visible
   indication of flow
4. Pressure gauge
5. Control valve
6. Check valve
7. Isolation valve

* Double mechanical seal only
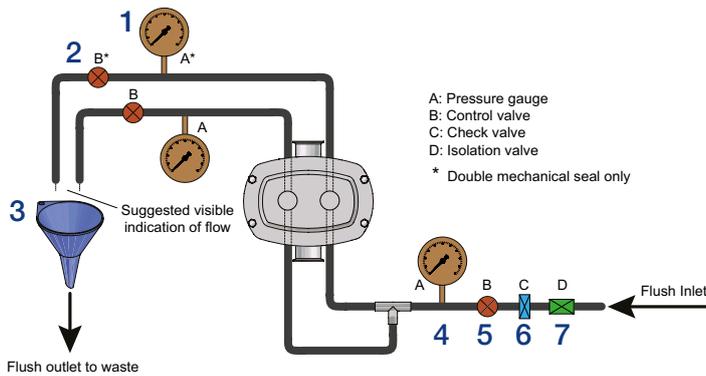
Fig. 6.1j Typical flushing pipework layout for a rotary lobe pump

Flushing Pipework Layout – PD pumps

It is recommended that seal flush pipework is designed such that seals are flushed in parallel. This is to ensure should one seal fail, then process media leakage does not enter the undamaged seal flush housing. The suggested arrangement above is for single mechanical seals (Fig. 6.1j). If the pump is fitted with double mechanical seals the pressure gauges and control valves should be fitted on the outlet side of the system. The choice of flushing liquid is dependent upon compatibility with the pumping media and overall duty conditions i.e., pressure and temperature. Usually, water is used for cooling and any water-soluble products.

On ATEX conforming pumps, seal flush pipework should be designed so that seals are flushed in series, rather than parallel. This should be done to eliminate the risk of one seal dry running should the flush pipework become blocked.

Flush Media
Outlet

Flush Media
Inlet
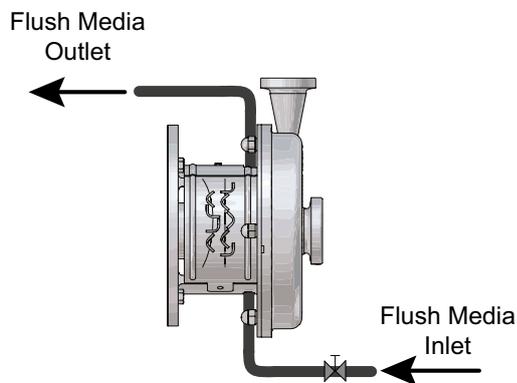
Fig. 6.1k Typical flushing pipework layout for a centrifugal pump

Flushing Pipework Layout – Centrifugal pumps

Centrifugal pump flush pipework should be designed such that inlet is at the bottom and outlet at the top, to allow air venting, thereby reducing the risk of air pockets developing, which could otherwise lead to localized seal dry running (Fig. 6.1k).

The below flow chart should be used for guidance purposes only, as actual seal selections should be verified by the seal suppliers.

Mechanical Seal Selection Process

The illustration describes the mechanical seal selection process with relevant questions to be answered.

| Obtain all Product/Fluid and Performance data |
| Select Seal Type |

- Does fluid crystallise?
- Is cooling required?
- Will pump run dry?
- Is aseptic barrier required?

Yes → Use Single Flushed Seal

No ↓

- Is fluid hazardous?
- Is fluid abrasive?
- Is fluid viscosity high?
- Is temperature high?
- Is aseptic barrier required?

Yes → Use Double Flushed Seal

No ↓

Use Single Seal

↓

Select Seal Materials

| Select Seal Faces | Select Elastomers |

Select Seal Faces
- Check viscosity limitations (see table 6.1b)
- Check temperature limitations (see table 6.1c)
- Is fluid abrasive?
- Check chemical compatibility

Select Elastomers
- Check elastomer compatibility (see guide in section 14.10)
- Check temperature limitations

Fig. 6.1l Seal selection process

6.2 Mechanical Seal Types
In Alfa Laval Pump Ranges

Seal Option Availability for Centrifugal Pumps

Pump Range	External Mounting			Internal Mounting	
	Single	Single Flushed	Double Flushed	Single	Single Flushed
LKH/LKH Evap	✓	✓	✓		
LKH Prime	✓		✓		
LKH Multistage				✓	✓
LKHPF				✓	✓
LKHI				✓	✓
LKH UltraPure	✓		✓		
LKH Prime UltraPure	✓		✓		
Solid C	✓	✓			
Solid C UltraPure	✓	✓			
FM	✓	✓			
GM	✓				

Table 6.2a

Seal Option Availability for Rotary Lobe Pumps

Mechanical Seal Type	Seal Name	Pump Range		
		OptiLobe	SRU	SX-UP/SX
Single externally mounted	R90		✓	
	Hyclean		✓	
Single flushed externally mounted	R90		✓	
	Hyclean		✓	
Single internally mounted	Easyfit	✓		
	R00			✓
Single flushed internally mounted	Easyfit	✓		
	R00			✓
Double flushed externally mounted	R90		✓	
Double flushed internally mounted	R00			✓

Note: R00 single flushed seal only available up to SX4

Table 6.2b

1. Stationary Seal ring
2. Rotary Seal ring
3. Wave Spring
4. Rotary Seal drive ring
5. Stationary Seal drive O-ring
6. Rotary Seal O-ring
7. Rotor
8. Shaft
9. Rotorcase
10. Stationary Seal Ring
    Anti-rotation pin
11. Grub screw
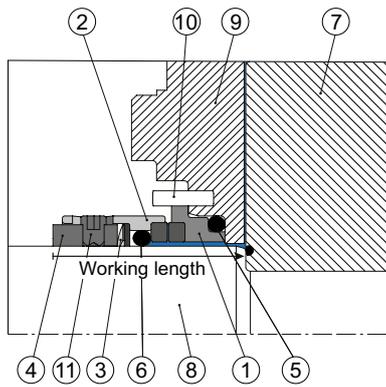
Fig. 6.2a R90 single mechanical seal

R90 Type Mechanical Seals

This seal type is the standard choice for many hygienic applications. A tried and trusted design, where easy conversion between single, single flushed and double seal variants is possible, without requiring any modification to pump head components.

Shown above in Fig 6.2a is the R90 single seal. The R90 single flushed and double seal variants are shown in Fig 6.1f & 6.1h respectively.

1. Stationary Seal ring
2. Rotary Seal ring
3. Wave spring
4. Washer
5. Stationary Seal O-ring
6. Rotary Seal O-ring
7. Rotor
8. Shaft
9. Rotor case
10. Rotor spline sealing O-ring
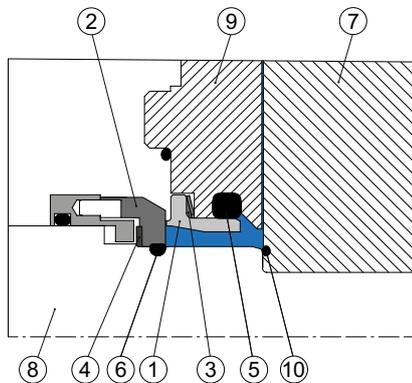
Fig. 6.2b Hyclean single mechanical seal

Hyclean Type Mechanical Seals

The design of this seal incorporates a self-cleaning feature in the form of the stationary seal element having an angled seal face (see Fig. 6.2b). This permits an enhanced cleaning action during CIP, as the cleaning media is thrown off tangentially due to the angle. Should EHEDG conformance be a user requirement, this seal type should be selected.
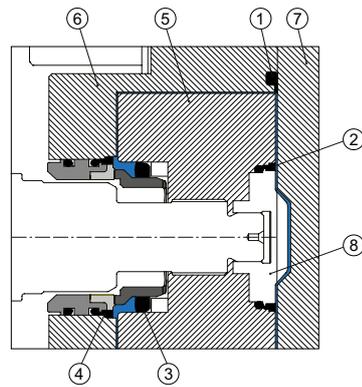
1. Front cover compression joint
2. Spline sealing cup seal
3. Rotary seal squad ring
4. Static seal cup seal
5. Rotor
6. Rotor case
7. Front cover
8. Rotor retainer

Process media

Fig. 6.2c SX pump head sealing

R00 Type Mechanical Seals

The R00 type mechanical seals (Fig. 6.2d), specifically designed for the SX and SX UltraPure rotary lobe pump ranges, are fully front-loading seals and fully interchangeable between seal variants, without the need for additional housings or pump component changes (Fig. 6.2c). Specialised seal setting of the mechanical seal is not required, as the seal is dimensionally set on assembly.

Seal faces are positioned directly in the fluid flow path, thereby seeing full liquid velocity, ensuring optimal cleaning during CIP cycle. All seals have controlled compression joint elastomers at fluid/atmosphere interfaces.
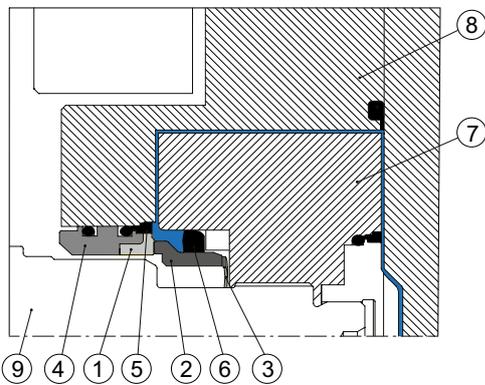
1. Stationary Seal ring
2. Rotary Seal ring
3. Wave Spring
4. Static seal ring
5. Static Seal cup seal
6. Rotary Seal Squad ring
7. Rotor
8. Rotorcase
9. Shaft

Process media

Fig. 6.2d R00 single mechanical seal
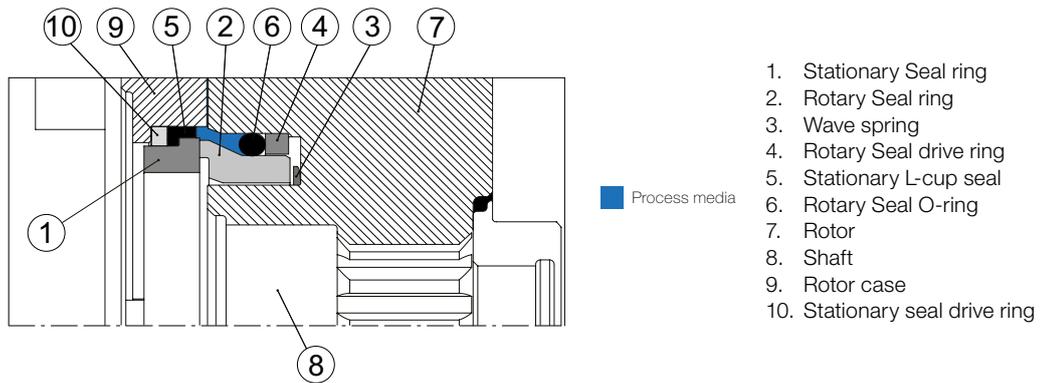
Fig. 6.2e OptiLobe Easyfit single mechanical seal in situ

Process media

1. Stationary Seal ring
2. Rotary Seal ring
3. Wave spring
4. Rotary Seal drive ring
5. Stationary L-cup seal
6. Rotary Seal O-ring
7. Rotor
8. Shaft
9. Rotor case
10. Stationary seal drive ring

EasyFit Mechanical Seals

This seal type is designed specifically for the OptiLobe rotary lobe pump range. Fully front-loading by design, simplifying service and interchangeable between variants, without pump modification, allows for increased process flexibility (Fig. 6.2e). Specialised seal setting of the mechanical seal is not required, as the seal is dimensionally set on assembly. Seal faces are positioned directly in the fluid flow path, thereby seeing full liquid velocity, ensuring optimal cleaning during CIP cycle.

1. Stationary Seal ring
2. Rotary Seal ring
3. Wave spring
4. Static assembly
5. Stationary seal squad ring
6. Rotary Seal square ring
7. Rotor
8. Shaft
9. Rotor case
10. Stationary seal ring
    anti rotation pin

Process media
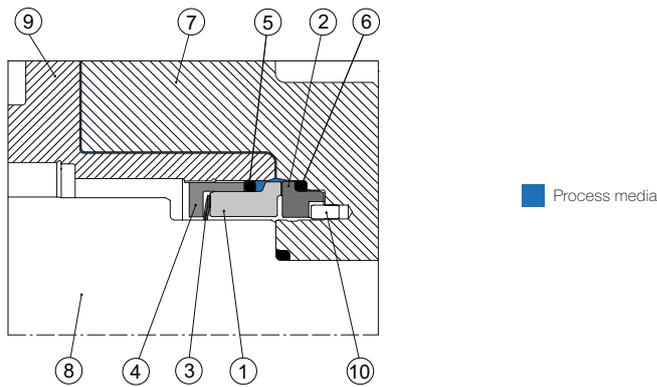
Fig. 6.2f DuraCirc single mechanical seal

Seal Option Availability for Circumferential Piston Pumps

DuraCirc is the name for Alfa Laval's range of circumferential piston pumps.

The DuraCirc is available with two main seal designs, the mechanical seal (Fig. 6.2f), and the O-ring seal (Fig. 6.2i).

The mechanical seal is designed specifically for the DuraCirc pump range. The single seal variant is fully front loading by design, simplifying service. The design of the DuraCirc is such that there is no separate flush housing, with the flush chamber instead integrated

as part of the rotor case. This makes the conversion from single-to-single flush seal via the simple addition of a lip seal and from single to double seal via the addition of outboard seal assembly. This makes seal interchangeability very simple without pump modification or the addition of housings, allowing for increased process flexibility. In all seal variants specialised seal setting is not required, as the seal is dimensionally set on assembly. Seal faces are positioned directly in the fluid flow path, thereby seeing full liquid velocity, ensuring optimal cleaning during CIP cycle.

1. Stationary Seal ring
2. Rotary Seal ring
3. Wave Spring
4. Static assembly
5. Stationary Seal squad ring
6. Rotary Seal squad ring
7. Rotor
8. Shaft
9. Rotorcase
10. Stationary Seal ring
    anti-rotation pin
11. Lip seal

Process media

Flush media
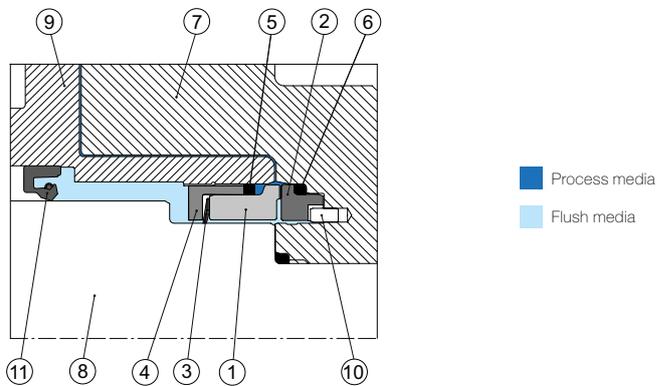
Fig. 6.2g DuraCirc single flushed mechanical seal

1. Stationary Seal ring inboard
2. Rotary Seal ring inboard
3. Wave Spring
4. Static assembly
5. Stationary Seal squad ring inboard
6. Rotary Seal squad ring inboard
7. Rotor
8. Shaft
9. Rotorcase
10. Stationary Seal ring
    anti-rotation pin
11. Stationary Seal O-ring outboard
12. Stationary Seal ring outboard
13. Rotary Seal ring outboard
14. Rotary Seal O-ring outboard

Process media

Flush media

Fig. 6.2h DuraCirc double flushed mechanical seal
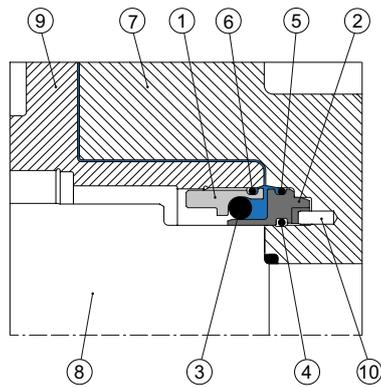
Fig. 6.2i DuraCirc O-ring seal

1. O-ring Seal housing
2. O-ring Seal sleeve
3. Dynamic O-ring
4. O-ring sleeve inner
5. O-ring sleeve outer
6. Housing O-ring
7. Rotor
8. Shaft
9. Rotor case
10. O-ring sleeve
    anti rotation pin

Designed for users more familiar with operating pumps fitted with O-ring seals, as with the mechanical seal equivalent, the O-ring seal assembly is designed specifically to fit in the DuraCirc pump range (Fig. 6.2i). Conversion from mechanical seal to O-ring seal variant is possible without any component modification.
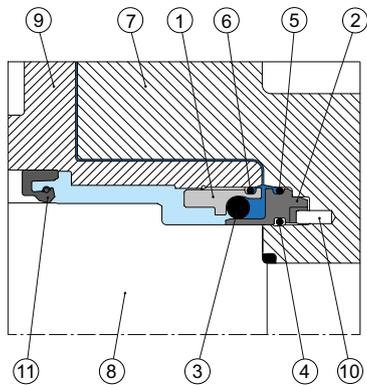
Fig. 6.2j DuraCirc flushed O-ring seal

1. O-ring seal housing
2. O-ring seal sleeve
3. Dynamic O-ring
4. O-ring sleeve inner
5. O-ring sleeve outer
6. Housing O-ring
7. Rotor
8. Shaft
9. Rotorcase
10. O-ring
    anti-rotation pin
11. Flush lip seal

The O-ring seal option also has the same advantages as with the mechanical seal equivalent, in that the single O-ring seal variant is fully front loading, there is easy conversion from single to flushed O-ring by the simple addition of a lip seal, without the need for housing or component modification (Fig. 6.2j). Additionally, the O-ring is positioned directly in the fluid flow path, thereby seeing full liquid velocity, ensuring optimal cleaning during CIP cycle.

O-ring seal material options in FDA conforming EPDM and FPM.

The O-ring in an O-ring seal is typically designed for more frequent change than pumps fitted with mechanical seals, however wear is typically limited to the O-ring itself. As it is an O-ring elastomer which is doing the actual sealing, as opposed to a set of mechanical seal faces, the recommended maximum running speed is 300 rpm and operating pressure is 7 bar (102 PSI).
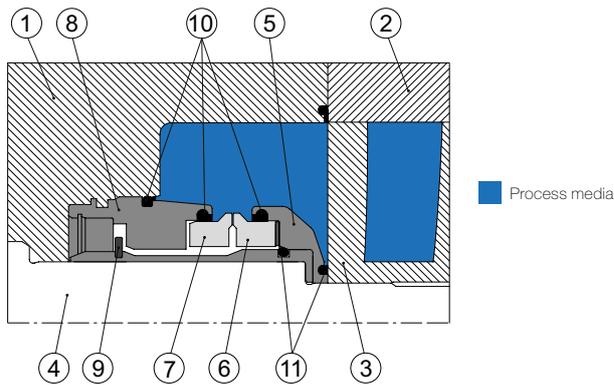
Fig. 6.2k Twin Screw Single Mechanical Seal

1. Seal housing
2. Pump casing
3. Screw
4. Shaft
5. Sleeve
6. Rotary Seal face
7. Static Seal face
8. Seal housing (incl. coil springs)
9. Circlip
10. P-ring
11. Washer

Seal Option Availability for Twin Screw Pumps

The OS Twin Screw pump has a cartridge style seal design, which means that the seal is fitted as a complete one-piece assembly, rather than as individual components.

The seal is truly front loading, making it very easy to fit. The self-setting design allows very easy installation of the seal, simply by sliding the seal onto the shaft until it can be pushed no further and locking in place with the seal retention pin. Additionally, the seal faces are positioned directly in the fluid flow path, thereby seeing full liquid velocity, ensuring optimal cleaning during CIP cycle.

Seal options include single (Flg. 6.2k), single flushed (Flg. 6.2l) and double variants (Flg. 6.2m).

The design of the OS Twin Screw pump is such that the flush housing is integrated into the seal housing. Conversion from single-to-single flushed or double seal is easy, with no additional housings or component re-work required.
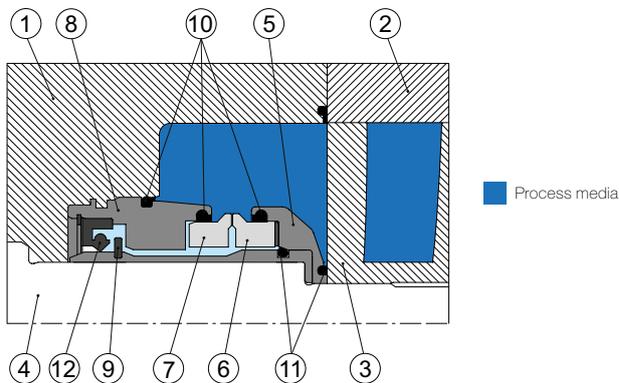
Fig. 6.2l Twin Screw Flushed Mechanical Seal

1. Seal housing
2. Pump casing
3. Screw
4. Shaft
5. Sleeve
6. Rotary Seal face
7. Static Seal face
8. Seal housing (incl. coil springs)
9. Circlip
10. P-ring
11. Washer
12. Flush lip seal

Fig. 6.2m Twin Screw Double Mechanical Seal
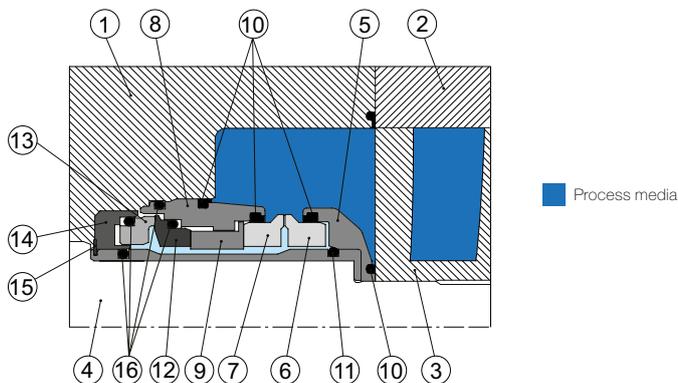
1. Seal housing
2. Pump casing
3. Screw
4. Shaft
5. Rotary holder
6. Rotary Seal face inboard
7. Static Seal face inboard
8. Housing
9. Drive ring
10. P-ring
11. Washer
12. Static Seal face onboard
13. Rotary Seal face onboard
14. Drive ring (incl. coil springs)
15. Circlip
16. O-ring

This section shows how to size an
Alfa Laval pump from product/fluid
and performance data given, sup-
ported by relevant calculations and
worked examples with a simple
step by step approach.

7.0

Pump Sizing

7.1 General Information Required

In order to correctly size any type of pump some essential information is required as follows:

Product/Fluid Data

- Fluid to be pumped

- Viscosity

- SG/Density

- Pumping temperature

- Vapor pressure

- Solids content (max. size and concentration)

- Fluid behavior (i.e., Newtonian or Pseudoplastic etc.)

- Is product hazardous or toxic?

- Does fluid crystallize in contact with atmosphere?

- Is CIP required and if so what temperature and performance data if needs to be carried out with our pump?

Performance Data

- Capacity (flow rate)

- Discharge head/pressure

- Suction condition (flooded or suction lift) NPSHa

Site Services Data

- Power source (electric, air, diesel, petrol or hydraulic). If electric – motor enclosure and electrical supply

- Seal flushing fluid

Specific requirements

- EHEDG/3A

- ATEX/Explosion zone

- Port Orientation

- Surface Finish

- Traceability
  – Use of frequency drive/VFD

In an ideal situation all the above criteria should be known before sizing a pump – however, in many instances not all this information is known and made available. In such cases to complete the sizing process, some assumptions may need to be made based upon application knowledge, experience etc. These should be subsequently confirmed, as they could be critical to satisfactory installation and operation.

If you have access to our Online ALiCE selection system (other systems as PUMP-FLO® can also be used) the acquired data together with assumptions can be
entered and the system will advise which pumps can be used for the specific duty and gives the option to sort by the most important factors for the customer whether this is efficiency, purchase price or other factors.

If you have access to ALiCE and want to select from there then please go through our Webinars on sizing and configuration in our learning portal.

For calculating the correct size pump and to make qualified assumptions the following will assist in optimal sizing.

7.2 Power

All of the system energy requirements and the energy losses in the pump must be supplied by a prime mover in the form of mechanical energy. For Alfa Laval pumps this will be an electrical motor, so it will be called motor from here. The rate of energy input needed is defined as power and is expressed in watts (W) – for practical purposes, power within this handbook is expressed in kilowatts (kW), i.e., watts x 10^3.

7.2.1 Hydraulic Power

The theoretical energy required to pump a given quantity of fluid against a given total head is known as hydraulic power, hydraulic horsepower or water horsepower.

This can be calculated as follows:

Hydraulic Power (W) $= Q \times H \times \rho \times g$

Where:

Q = Capacity (m^3/s)
H = Total Head/Pressure (m)
ρ = Fluid Density (kg/m^3)
g = Acceleration due to Gravity (m/s^2)

Other forms of this equation can be as follows:

Hydraulic Power (kW) $= \dfrac{Q \times H}{k}$

Where:

Q = Capacity
H = Total Head/Pressure
k = Constant (dependent upon units used)

Therefore

Hydraulic Power (kW) $= \dfrac{Q \times H}{k}$

Where:

Q = Capacity (l/min)
H = Total Head/Pressure (bar)
k = 600

or

Hydraulic Power (hp) $= \dfrac{Q \times H}{k}$

Where:

Q = Capacity (US gal/min)
H = Total Head/Pressure (PSI)
k = 1715
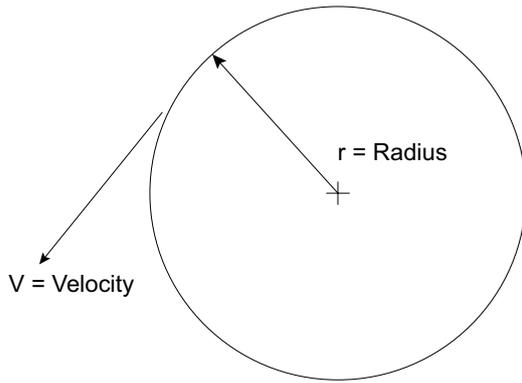
Fig. 7.2.2a Shaft angular velocity

7.2.2 Required Power

The required power or brake horsepower is the power needed at the pump shaft. This is always higher than the hydraulic power due to energy losses in the pump mechanism (friction loss, pressure loss, seals etc.) and is derived from:

Required Power $\quad = \quad \omega \times T$

Where:

ω = Shaft Angular Velocity
T = Shaft Torque

Shaft Angular Velocity $\quad = \quad \omega = V \times r$ (see Fig. 7.2.2a)

Where:

ω = Shaft Angular Velocity
V = Velocity
r = Radius

And is related to Hydraulic Power by:

Required Power $\quad = \quad \dfrac{\text{Hydraulic Power}}{\text{Efficiency (100\% = 1.0)}}$

The appropriate motor power must be selected. This will generally be the nearest motor rated output power above the required power.

7.2.3 Torque

Torque is defined as the moment of force required to produce rotation and is usually expressed in units of Nm (Newton meter), Kg-m (Kilogram meter) or ft lb (foot pound).

Torque can be calculated as follows:

Torque (Nm) $\quad = \dfrac{\text{Required power (kW) x 9550}}{\text{Pump speed (rev/min)}}$

or

Torque (Kg-m) $\quad = \dfrac{\text{Required power (kW) x 974}}{\text{Pump speed (rev/min)}}$

or

Torque (ft lb) $\quad = \dfrac{\text{Required power (hp) x 5250}}{\text{Pump speed (rev/min)}}$

It should be noted that positive displacement pumps are basically constant torque machines and therefore it is important that the transmission chosen is capable of transmitting the torque required by the pump. This is particularly important for variable speed drives which should be selected initially on torque rather than power.

Always be aware that torque requirement can be higher for the minimum speed than for the higher speed when you have a large span in speed.

The power requirements for mechanical devices such as pumps and pump drives are best expressed in terms of torque and speed.

7.2.4 Efficiency

Total Efficiency

Total efficiency is typically used on centrifugal pump to describe the relationship between input power at the pump shaft and output power in the form of water horsepower. The term 'mechanical efficiency' can also be used to describe this ratio. Total efficiency, designated by symbol η comprises of three elements, Hydraulic Efficiency (η_h), Mechanical Efficiency (η_m) and Volumetric Efficiency (η_v) which are described below:

Hydraulic Efficiency

The term hydraulic efficiency is used on centrifugal pumps to describe one of the three elements of centrifugal pump total efficiency as described above.

Where
Hydraulic Efficiency (η_h) =
$$\frac{\text{Pump Head Loss (m)}}{\text{Total Head (m)}} \times 100\%$$

The pump head losses comprise of the shock loss at the eye of the impeller, friction loss in the impeller blade and circulation loss at the outlet side of the impeller blades.

Mechanical Efficiency

This term is used on all centrifugal and positive displacement pump types, and is typically used to describe the losses associated with the transfer of energy from the motor through a mechanical system to the pumped liquid.

Where
Mechanical Efficiency (η_m) =
$$\frac{1 - \text{Pump mechanical losses}}{\text{Required power}} \times 100\%$$

Pump mechanical losses refers to the friction losses associated with bearings, seals and other contacting areas within the pump.

Volumetric Efficiency

This term is used on all centrifugal and positive displacement pump types. It is most commonly used to compare the performance of a number of pump types, where accurate geometric data is available.

For centrifugal pumps,
Volumetric Efficiency (η_v) = $\frac{Q}{Q + Q_L} \times 100\%$

Where: Q = Pump capacity
         Q_L = Fluid losses due to leakage through the impeller casing clearances

For positive displacement pumps the term volumetric efficiency (η_v) is used to compare the displacement of the pump against the capacity of the pump. The displacement calculation (q) per revolution for positive displacement pumps involves calculating the volume of the void formed between the rotating element and the fixed element of the pump. This is then multiplied by the number of voids formed by a rotating element per revolution of the pump's drive shaft and by the number of rotors in the pump.

For rotary lobe pumps,
Volumetric Efficiency (η_v) = $\frac{Q}{q} \times 100\%$

Where: Q = Pump capacity
         q = Pump displacement

Rotary lobe are generally efficient pumps and even at viscosity of 100 cP the volumetric efficiency of most pumps is approximately 90% for low pressure duties. At lower viscosities and/or higher pressures the volumetric efficiency will decrease due to slip as described in 7.6.1. Above 1000 cP, volumetric efficiency can be as high as 95–99% for the Rotary lobe pumps.

Looking at Circumferential Piston pumps these are highly efficient pumps with a volumetric efficiency of 95–99% at viscosities as low as 15 cP.

While Twin Screw pumps can easily handle viscosities from 1 to 1000000 cP the efficiency will be around 80% of the rotary lobe pumps for the higher viscous products.

The pump speed should always be selected according to the product and how gentle it needs to be treated, how high the viscosity is, how many particles it has etc.

When the maximum speed has been selected for the Rotary lobe/Circumferential Piston pump and the viscosity is high enough to have almost full volumetric efficiency, the capacity of the pump can be calculated according to the following formulas. Rearranging the formula will calculate the maximum speed. Full speed is rarely recommend as the mechanical losses inside the pump will cause vibration and noise.

$$Q = \frac{q \times \eta_v \times 60 \times n}{100}$$

Where:

n = Pump Speed (rev/min)
Q = Capacity (m3/h)
q = Pump Displacement (m3/100 rev)
η_v = Volumetric Efficiency (100% = 1.0)

$$n = \frac{Q \times 100}{q \times \eta_v \times 60}$$

Where:

n = Pump Speed (rev/min)
Q = Capacity (m^3/h)
q = Pump Displacement (m^3/100 rev)
η_v = Volumetric Efficiency (100% = 1.0)

or

$$Q = \frac{q \times \eta_v \times n}{100}$$

Where:

n = Pump Speed (rev/min)
Q = Capacity (US gal/min)
q = Pump Displacement (US gal/100 rev)
η_v = Volumetric Efficiency (100% = 1.0)

Pump Efficiency

The term pump efficiency is used on all types of pumps to describe the ratio of power supply to the drive shaft against water horsepower (the power available to move the liquid without losses).

$$\text{Pump Efficiency } \eta_p = \frac{\text{Water horsepower x 100\%}}{\text{Required power}}$$

or

$$\text{Pump Efficiency } \eta_p = \frac{Q \times H \times \rho \times g}{\omega \times T}$$

Where:

Q = Capacity (m^3/s)
H = Total Head/Pressure (m)
ρ = Fluid Density (lb/ft^3)
g = Acceleration due to Gravity (m/s^2)
ω = Shaft Angular Velocity (rad/s)
T = Shaft Torque (Nm)

or

$$\text{Pump Efficiency } \eta_p = \frac{Q \times H \times \rho \times g}{\omega \times T \times 3960}$$

Where:

Q = Capacity (US GPM)
H = Total Head/Pressure (ft)
ρ = Fluid Density (lb/ft^3)
g = Acceleration due to Gravity (ft/s^2)
ω = Shaft Angular Velocity (rad/s)
T = Shaft Torque (lb/ft)

Overall Efficiency

Overall efficiency is a term used to describe and compare the performance of all types of pumps. Overall efficiency considers the efficiency of both the prime mover and the pump, and is sometimes known as the wire to water/liquid efficiency where the prime mover is an electric motor.

$$\text{Overall Efficiency}_{oa} = \frac{\text{Water horsepower x 100\%}}{\text{Drive power}}$$

The higher the efficiency the less power will be lost to vibrations, heat generation etc. This makes the pump more sustainable and gives longer service intervals.
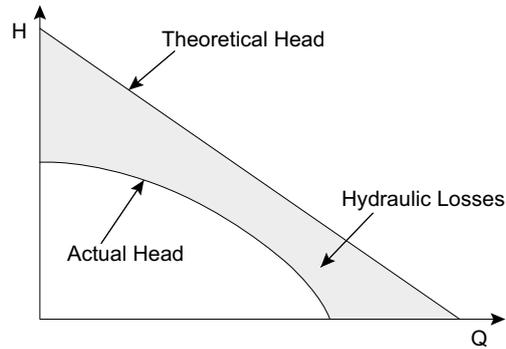
Fig. 7.3.1a Hydraulic losses

7.3 Centrifugal Pumps

7.3.1 Flow Curve

A centrifugal pump should always be sized from a pump flow curve or a pump selection program. Most pump flow curves are based on tests with water. It is difficult to determine general curves for fluids with viscosities different from water and therefore in these instances it is recommended to use a pump selection program.

A pump flow curve specifies the connection between Capacity Q, Head H, Required Power P, Required NPSH and Efficiency (η).

Hydraulic Losses

The connection between the capacity and the theoretical head of the pump is shown by means of a straight line, which decreases at a higher capacity (see Fig. 7.3.1a).

The actual head of a pump is, however, lower than the theoretical head due to hydraulic losses in the pump, which are friction loss, pressure loss and slip.

The connection between the capacity and actual head is consequently specified by means of a curve which varies depending on the design of the impeller.

Fig. 7.3.1b Curves for Q and H

Different Pump Characteristics

The Capacity Q and Head H curve of a centrifugal pump will vary depending upon the impeller vane design (see Fig. 7.3.1.b).

These fulfil different requirements and are well suited for flow control where only one parameter is to be changed (see section 7.3.2).

Curve 1 covers a wide range of heads without large changes to capacity.

Curve 3 covers a wide range of capacities without large changes to head.

Middle curve has a moderate change in both capacity and head.
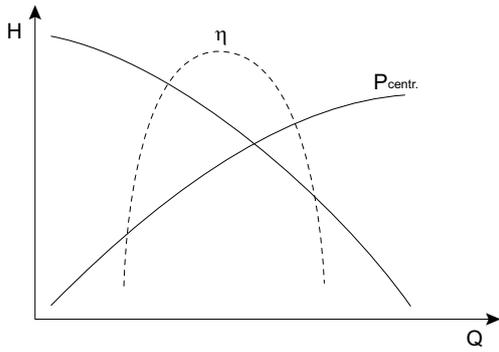
Fig. 7.3.1c Curves for Q, H, P and ρι
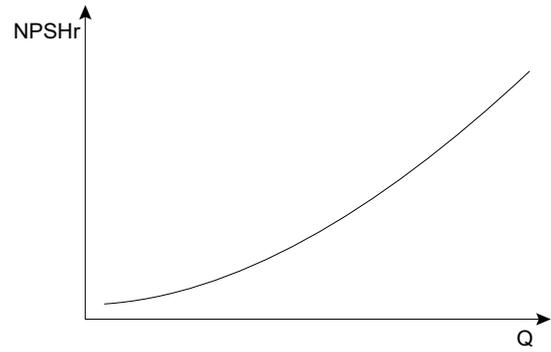
Fig. 7.3.1d NPSHr curve

Capacity Q, Head H, Power P and Efficiency Curves

In principle the duty point of a pump can be situated at any point on the Q – H curve. (Fig. 7.3.1c)

The efficiency of the pump will vary depending on where the duty point is situated on the Q – H curve. The efficiency is usually highest near the centre of the curve.

The power curve of the centrifugal pump increases at a higher capacity.

NPSHr Curve

(see section 2.2.4 on NPSHa calculation)

The NPSHr curve increases at higher capacity (see Fig. 7.3.1d). This should be used to ascertain the NPSHr of the pump. It is important that NPSHa of the system exceeds the NPSHr of the pump.
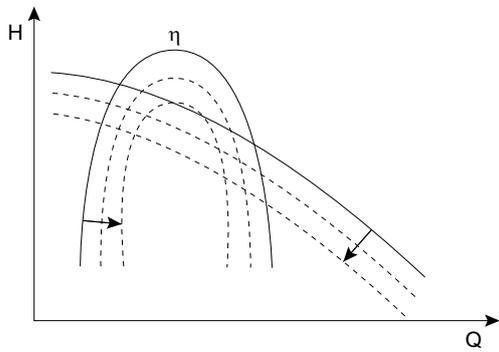
Fig. 7.3.1e Effects on Q, H and η
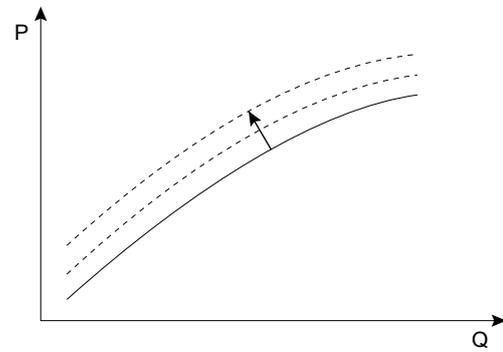
Fig. 7.3.1f Effects on Q, H and η

Viscosity Effect

Fluid viscosity will affect capacity, head, efficiency and power (see Fig. 7.3.1e).

• Capacity, head and efficiency will decrease at higher viscosities

• Required power will increase at higher viscosities

Density Effect

Fluid density will affect the head and required power which both increase proportionally at higher density (see Fig. 7.3.1f). When head increases capacity will go down if VFD is not used.

How to use the Flow Curve

The flow curve consists of three different curves:

- Head as a function of capacity (Q – H curve)

- Required power as a function of capacity (Q – P curve)

- Required NPSH as a function of capacity (Q – NPSHr curve)

Although illustrated here the efficiency is not shown on the published flow curves but can be determined from the required power on the flow curve and formula in section 7.2 when the duty point is known and the hydraulic power can be calculated and compared to the power from the curve. Efficiency is shown in the ALiCE sizing program.

The Q – H and Q – P curves are specified for different standard impeller diameters so that a correct duty point can be determined. This is not applicable to the LKH-Multistage pumps as the impeller diameters cannot be reduced.

The curves on the flow curve are based on tests with water at 20° C (68° F) with tolerances of ± 5%. It is recommended to select the pump by means of a pump selection program if the fluid to be pumped has other physical properties.

Example:

Product/Fluid Data:

Fluid to be pumped	-	Water
Viscosity	-	1 cP
SG	-	1.0
Pumping temperature	-	20° C

Performance Data:

Capacity	-	15 m³/h
Total head	-	25 m
Electrical supply	-	220/380v, 50 Hz

The optimum is to select the pump with the best return of investment (ROI), for the required duty point (15 m³/h, 25 m). This can also require information about running hours and expected lifetime so again assumptions might need to be made.

Step 1 – Find Appropriate Curve

Locate a flow curve for the required pump type that covers the duty point. For this particular example a flow curve of a centrifugal pump type LKH-10 with 3000 rev/min synchronous speed at 50 Hz is selected (see Fig. 7.3.1g).

Step 2 – Look at Q – H curve

• Locate the capacity (15 m^3/h) on the Q-scale

• Start from this point and follow the vertical line upwards until it intersects with the horizontal line indicating the required head (25 m) on the H-scale

• This duty point does not contact any curve corresponding to a certain impeller diameter. Therefore, the nearest larger size impeller diameter should be selected, in this case 150 mm. Alternatively ask to get the impeller reduced to 145 mm

• The head will then be 28 m

• The selected head (28 m) should be checked regarding the lower tolerance of the curve to ensure that it is at least the required 25 m

• In this case the head should be reduced by 5% being the curve tolerance

• The head will then be a minimum of 26.6 m greater than 25 m, thus satisfactory

Step 3 – Look at Q – P curve

• The next step in selecting the pump is to follow the vertical capacity line (15 m^3/h) downwards until it intersects with the power curve for the 150 mm impeller

• A horizontal line to the left of the intersection indicates a required power of 2.0 kW

• For a LKH centrifugal pump a safety factor of 5% for motor losses must be added, resulting in a total required power of 2.1 kW

• Consequently a 2.2 kW motor can be used

Step 4 – Look at Q – NPSHr curve

• Finally the vertical capacity line (15 m^3/h) is followed up to the NPSHr curve (green)

• The intersection corresponding to the 150 mm impeller is located

• A horizontal line to the right of the intersection indicates that NPSHr is approx. 0.8 m

A = 110   D = 140   G = 163
B = 120   E = 150
C = 130   F = 160

A = 110   D = 140   G = 163
B = 120   E = 150
C = 130   F = 160
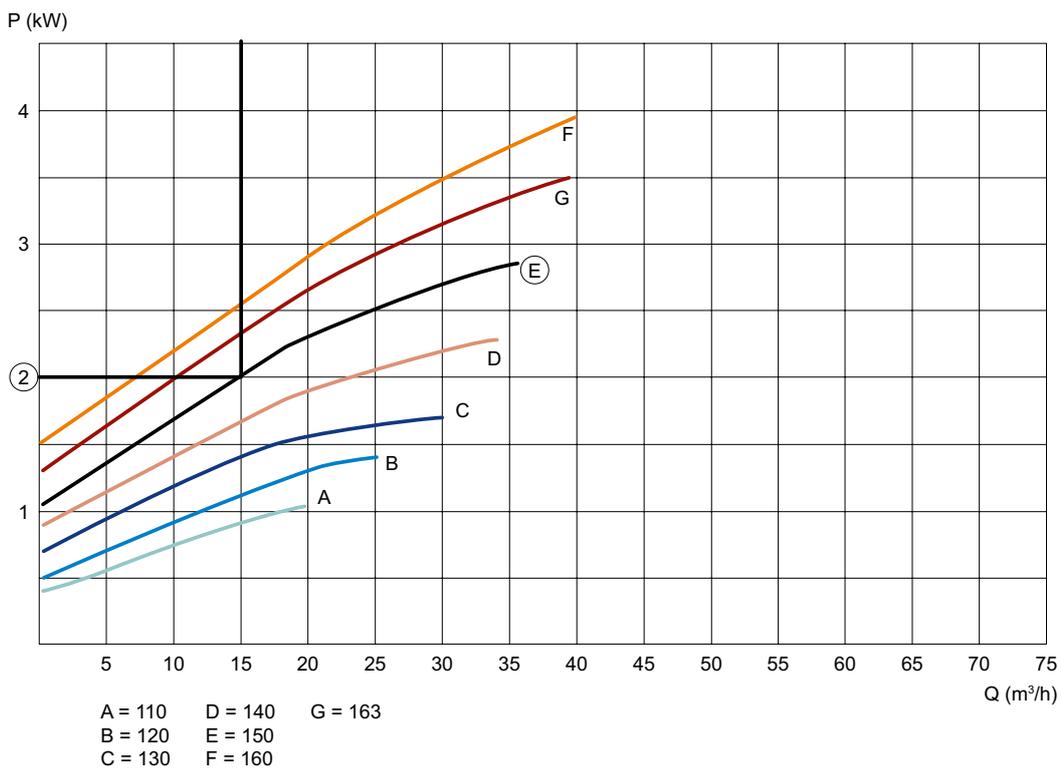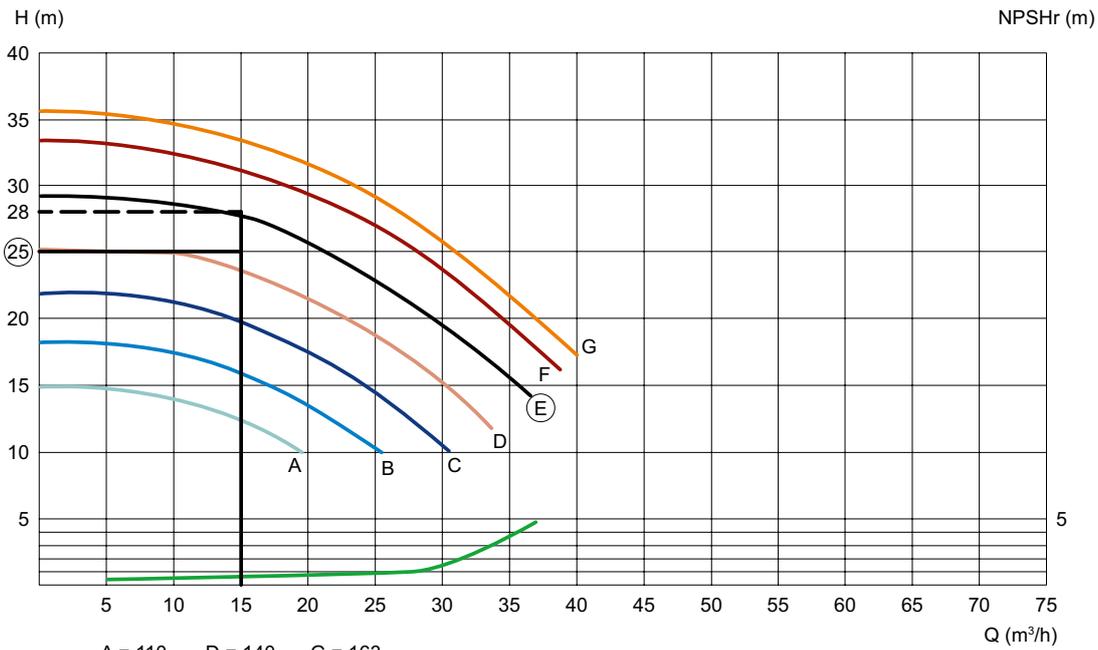
Fig. 7.3.1g Example
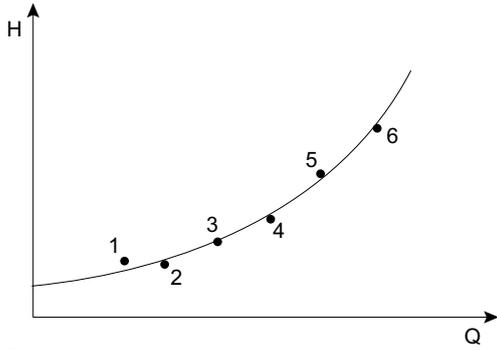
Fig. 7.3.2a Process curve

7.3.2 Flow Control

Duty Point

The duty point of a pump is the intersection point between the pump curve and the process curve.

Pump curve – this specifies the connection between head H and capacity Q (see section 7.3.1).

Process curve – this specifies the connection between the total pressure drop (L ΔH) in the process plant and the capacity (Q) (see Fig. 7.3.2a).

The process curve is determined by varying the capacity so that different pressure drop (ΔH) values are obtained. The shape of the process curve will depend on the process design (i.e., layout, valves, filters etc.).

Capacity:	Q_1	Q_2	Q_3	Q_4	Q_5	Q_6
Pressure drop:	DH_1	DH_2	DH_3	DH_4	DH_5	DH_6
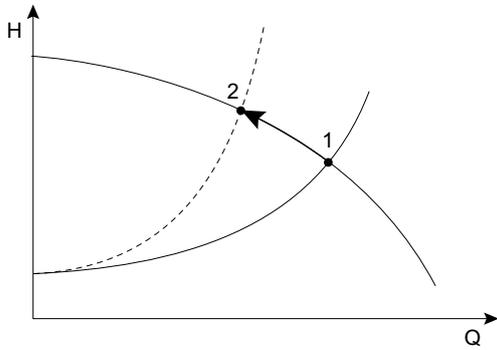
Fig. 7.3.2b Changes in pressure drop
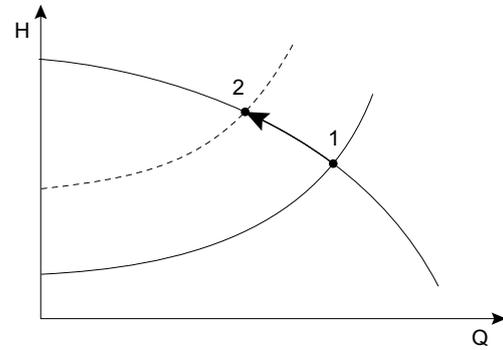
Fig. 7.3.2c Changes in required head

The duty point of a pump can change due to changes in the conditions of the process plant (changes in head, pressure drops etc.). The pump will automatically regulate the capacity to meet the new conditions (see Fig. 7.3.2b and 7.3.2c).

It is possible to compensate for the change of duty point by means of flow control that can be achieved as follows:

- Reducing the impeller diameter (not for Multistage pumps)

- Throttling the discharge line

- Controlling the pump speed

Due to flow control it is possible to achieve optimum pump efficiency at the required capacity resulting in the most economical pump installation.
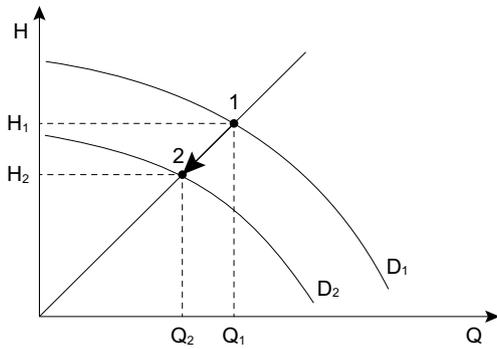
Fig. 7.3.2d Reducing impeller diameter

Reducing Impeller Diameter

Reducing the impeller diameter can only be carried out for centrifugal pumps. This will reduce the capacity and the head.

Centrifugal Pump

The connection between Impeller Diameter (D), Capacity (Q) and Head (H) is shown in Fig. 7.3.2d:

1. Before reducing

2. After reducing – the duty point moves towards point 2 when reducing the impeller diameter

If the impeller speed remains unchanged, the connection between Impeller Diameter (D), Capacity (Q), Head (H) and Required Power (P) is shown by the following formulas:

Speed/Capacity: $\dfrac{Q_1}{Q_2} = \dfrac{n_1}{n_2} \Rightarrow n_2 = n_1 \times \dfrac{Q_2}{Q_1}$ [rev/min]

Speed/Head: $\dfrac{H_1}{H_2} = \dfrac{n_1^2}{n_2^2} \Rightarrow n_2 = n_1 \times \sqrt{\dfrac{H_2}{H_1}}$ [rev/min]

Speed/Power $\dfrac{P_1}{P_2} = \dfrac{n_1^3}{n_2^3} \Rightarrow n_2 = n_1 \times \sqrt[3]{\dfrac{P_2}{P_1}}$ [rev/min]

Most pump flow curves show characteristics for different impeller diameters to enable the correct impeller diameter to be selected.

Reducing the impeller diameter by up to 20% will not affect the efficiency of the pump much. If the reduction in impeller diameter exceeds 20%, the pump efficiency will decrease.

The impeller diameter is reduced to D2 by means of the following formula:

$$D_2 = D_1 \times \sqrt{\dfrac{c-b}{a-b}} \quad [mm]$$

Where:

D_1 = Standard Diameter before Reducing
a = Maximum Duty Point
b = Minimum Duty Point
c = Required Duty Point

The formula is for guidance purposes only. It is recommended to add a safety factor of 10–15% to the new diameter.
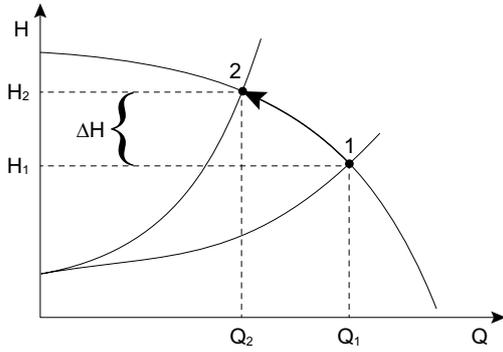
Fig. 7.3.2e Throttling discharge line
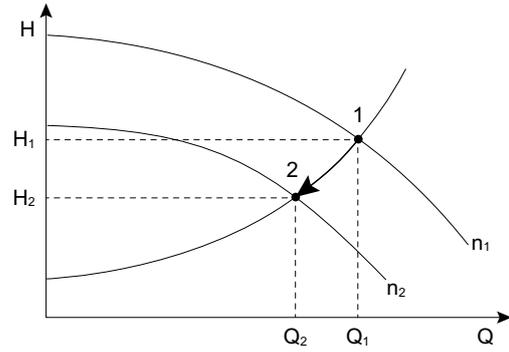
Fig. 7.3.2f Controlling pump speed

Throttling Discharge Line

Throttling the discharge line will increase the resistance in the process plant, which will increase the head and reduce the capacity.

The connection between Capacity (Q) and Head (H) when throttling is shown in Fig. 7.3.2e.

1. Before throttling
2. After throttling, the duty point moves towards point 2

Throttling should not be carried out in the suction line as cavitation can occur.

Also throttling will reduce the efficiency of the process plant ΔH shows the 'waste' of pressure at point 2 to overcome the throttling.

Controlling Pump Speed

Changing the impeller speed will change the centrifugal force created by the impeller. Therefore, the capacity and the head will also change.

The connection between Capacity (Q) and Head (H) when changing the impeller speed is shown in Fig. 7.3.2f.

1. Before reducing impeller speed

2. After reducing impeller speed. The working point moves towards point 2 when reducing the impeller speed

The most common form of speed control is by means of a frequency converter (see section 9.10).
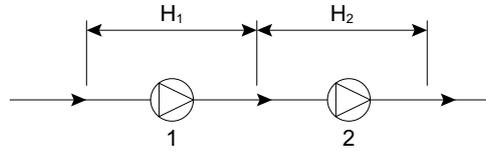
$$H = H_1 + H_2 \qquad Q = \text{Constant}$$
Fig. 7.3.3a Principle of connection

If the impeller dimensions remain unchanged, the connection between Impeller Speed (n), Capacity (Q), Head (H) and Required Power (P) is shown by the following formulas:

Speed/Capacity: $\dfrac{Q_1}{Q_2} = \dfrac{n_1}{n_2} \Rightarrow n_2 = n_1 \times \dfrac{Q_2}{Q_1}$ [rev/min]

Speed/Head: $\dfrac{H_1}{H_2} = \dfrac{n_1{}^2}{n_2{}^2} \Rightarrow n_2 = n_1 \times \sqrt{\dfrac{H_2}{H_1}}$ [rev/min]

Speed/Power $\dfrac{P_1}{P_2} = \dfrac{n_1{}^3}{n_2{}^3} \Rightarrow n_2 = n_1 \times \sqrt[3]{\dfrac{P_2}{P_1}}$ [rev/min]

As shown from the above formulas the impeller speed affects capacity, head and required power as follows:

- Half speed results in capacity x 0.5

- Half speed results in head x 0.25

- Half speed results in required power x 0.125

Speed control will not affect the efficiency much providing changes do not exceed 20%.

7.3.3 Alternative Pump Installations
Pumps Coupled in Series

It is possible to increase the head in a pump installation by two or more pumps being coupled in series (see Fig. 7.3.3a).

The Capacity (Q) will always be constant throughout the pump series (see Fig. 7.3.3b).

The head can vary depending on the pump sizes.

The outlet of pump 1 is connected to the inlet of pump 2.

Pump 2 must be able to withstand the outlet head from pump 1.
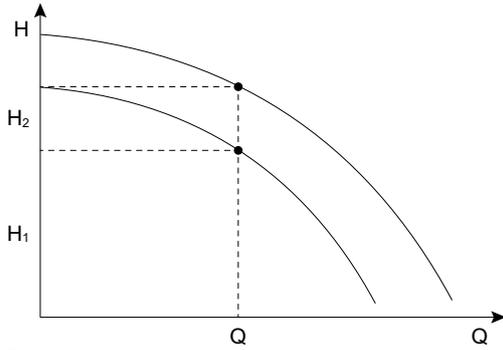
Fig. 7.3.3b Head of pumps in series
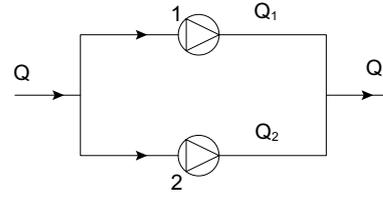
$Q_3 = Q_1 + Q_2$ H = Constant
Fig. 7.3.3c Principle of connection

If two different pumps are connected in series, the pump with the lowest NPSH value should be installed as the first pump (for critical suction conditions).

The capacity in the pump installation should not exceed the max. capacity of the smallest pump. Otherwise, there will be a pressure drop in the smallest pump.

A multi-stage centrifugal pump is in principle several pumps that are coupled in series but built together as one pump unit.
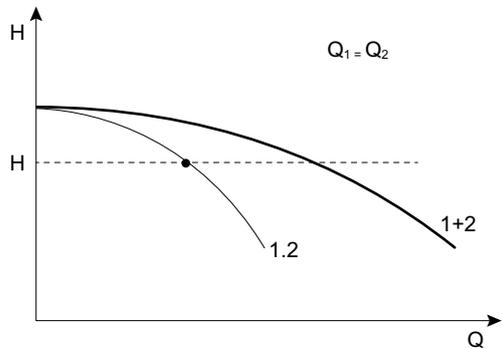
Fig. 7.3.3d Connection of two similar pumps
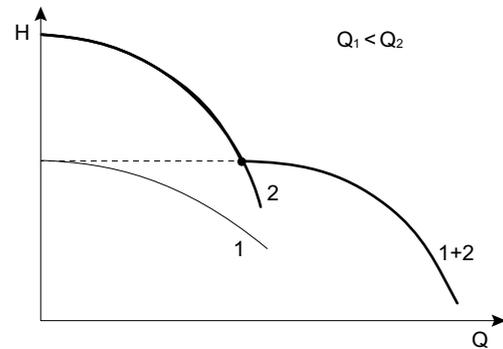
Fig. 7.3.3e Connection of two different pumps

7.3.4 Pumps Coupled in Parallel

It is possible to increase the capacity in a pump installation by two or more pumps coupled in parallel (see Fig. 7.3.3c).

The Head (H) will always be constant in the pump installation. The capacity can vary depending on the pump sizes. (See Fig. 7.3.3d)
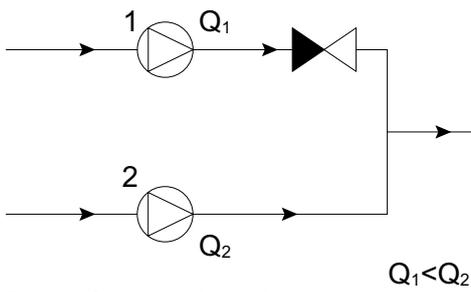
Fig. 7.3.3f Connection of two different pump sizes

The pumps receive the fluid from the same source and have a common discharge line.

When the capacity is increased by means of pumps coupled in parallel, the equipment and pressure drop in the installation must be determined according to the total capacity of the pumps (see Fig. 7.3.3f).

For two different pumps, If the capacity Q_1 is smaller than the capacity Q_2, it is possible to install a non-return valve in the discharge line of pump 1 to avoid pump 2 pumping fluid back through pump 1 (see Fig. 7.3.3e).

Equally it is important to install the pumps with same pipe size etc. so one pump does not take all the flow and "starve" the other which could result in cavitation.

7.4 Worked Examples – Centrifugal Pump Sizing

Metric units

7.4.1 Example 1

The following example in Fig. 7.4.1a shows a pump to be sized for a typical brewery process.

The pump is required to handle Wort from the Whirlpool to the Fermentation vessel.
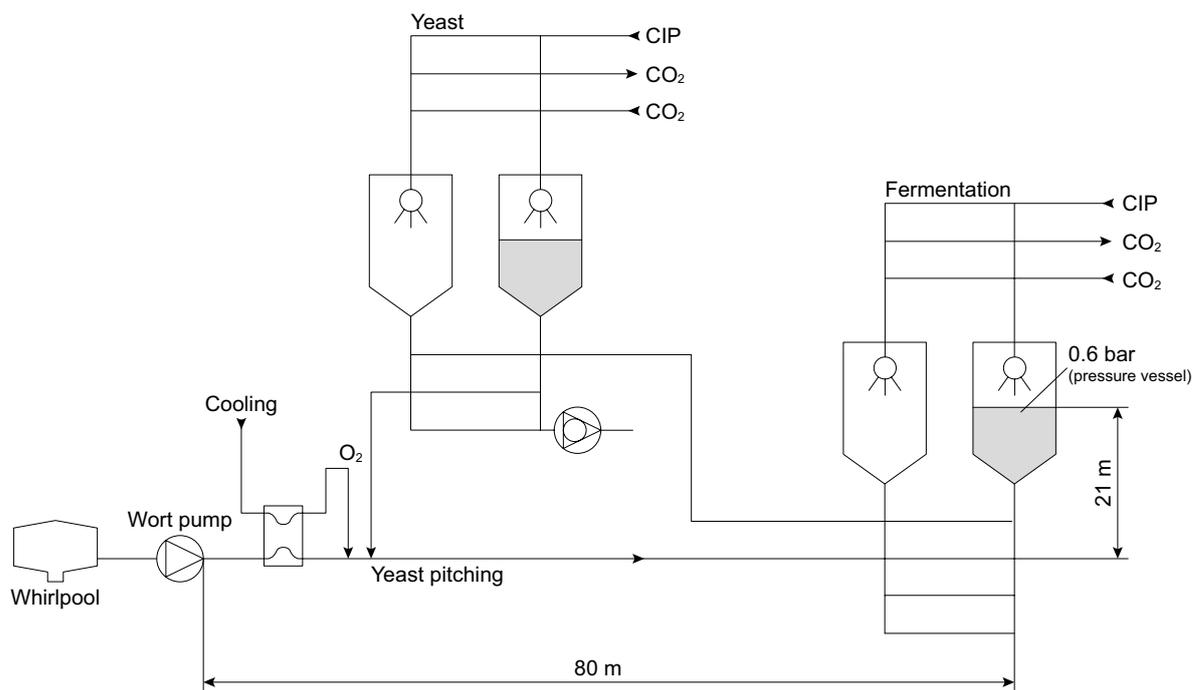
Fig. 7.4.1a Example 1

As described in section 7.1 in order to correctly size any type of pump, some essential information is required such as Product/Fluid data, Performance data and Site Services data.

Product/Fluid data:

Fluid to be pumped	-	Wort
Viscosity	-	1 cP
Pumping temperature	-	90° C

All the data has been given by the customer.

Performance data:

Capacity	-	40 m^3/h
Discharge	-	via 80 m of 101.6 mm dia. tube, plus a given number of bends, valves and a plate heat exchanger with Δp_{PHE} 1.6 bar. Static head in Fermenting vessel = 21 m. Pressure in Fermenting vessel = 0.6 bar
Suction	-	0.4 m head, plus a given number of bends and valves

Site Services data:

| Electrical supply | - | 400v, 50 Hz |
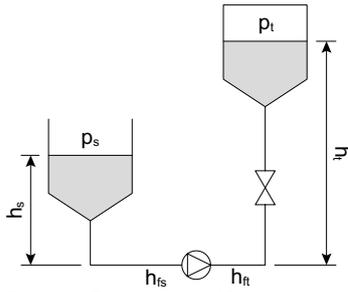
Fig. 7.4.1b Typical suction / Discharge Head set-up

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge Head H_t = $h_t + h_{ft} + p_t$

Where:

h_t = Static Head in Fermentation Vessel
h_{ft} = Total Pressure Drop in Discharge Line
p_t = Pressure in Fermentation Vessel

Therefore:

h_t = 21 m
h_{ft} = Pressure Drop in Tube Δp_{tube}
    + Pressure Drop in Bends and Valves Δp
    + Pressure Drop in Plate Heat Exchanger Δp_{PHE}
Δp_{tube} (from curve shown in 14.5) = 1.5 m
    (1.8 m loss per 100 m)
Δp is calculated to be 5 m
Δp_{PHE} is given as 1.6 bar = 16 m
h_{ft} = 1.5 + 5 + 16 m = 22.5 m
p_t = 0.6 bar = 6 m

$H_t = h_t + h_{ft} + p_t = 21 + 22.5 + 6$ m = 49.5 m (4.95 bar)

Total suction head H_s = $h_s - h_{fs} + p_s$

Where:

h_s = Static Suction Head in Whirlpool
h_{fs} = Total Pressure Drop in Suction Line
p_s = Pressure in Whirlpool (open tank)

Therefore:

h_s = 0.4 m
h_{fs} = Calculated to be 1 m
p_s = 0 (open tank)

$H_s = h_s - h_{fs} + p_s = 0.4 - 1 + 0$ m = - 0.6 m (- 0.06 bar)

Total head $H = H_t - H_s = 49.5 - (- 0.6) = 50.1$ m (5.01 bar)

NPSHa

NPSHa	=	$Pa + h_s - h_{fs} - Pvp$

Where:

Pa = Pressure Absolute above Level of Fluid in Whirlpool Tank

h_s = Static Suction Head in Whirlpool Tank

h_{fs} = Total Pressure Drop in Suction Line

Pvp = Vapour Pressure of Fluid

Therefore:

Pa = 1 bar (open tank) = 10 m

h_s = 0.4 m

h_{fs} = Calculated to be 1 m

Pvp = 0.70 bar a (from table 14.4a) = 7 m

NPSHa = 10 + 0.4 - 1 - 7 (m) = 2.4 m

Actual pump sizing can be made using pump performance curves or a pump selection program. The performance curves are, however, not suitable if the fluid to be pumped has physical properties (i.e., viscosity) different from water. In this particular example both the pump performance curves and pump selection program can be used. The performance curve selection procedure is more specifically described in section 7.3.1.

For this particular example, pump sized would be as follows:

Pump Model	-	LKH-25
Impeller size	-	200 mm
Speed	-	2940 rev/min
Capacity	-	40 m³/h
Head	-	50.1 m (5.01 bar)
Efficiency	-	63.1%
NPSHr	-	1.4 m
Motor size	-	11 kW

Cavitation check

NPSHa should be greater than NPSHr i.e., 2.4 m > 1.4 m, i.e., no cavitation will occur.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a double mechanical seal with carbon/silicon carbide faces and EPDM elastomers.

Fig. 7.4.1c Q - H charateristic when changing valves

Special Note

The discharge head (h_{t2}) is lower when the pump starts filling the fermenting vessel compared to the discharge head (h_{t1}) when the vessel is full. The reduction of the discharge head will result in higher flow. This will reduce the NPSHr and can therefore lead to cavitation. Another risk is that the larger capacity and thereby larger power consumption can cause overloading of the motor (see Fig. 7.4.1c).

Cavitation can be avoided by reducing the pump speed (reducing NPSHr), i.e., by means of a frequency converter, or by throttling the discharge line (increasing head). The flow control method is more specifically described in section 7.3.2.

Adjustment

In this example the pump is sized by the pump selection program which results in exact impeller diameter of 200 mm, so that the selected duty point is as close to the required duty point as possible.

The pump is sized with a standard impeller diameter if using the performance curve. In this case it may be necessary to adjust the selected duty point by means of flow control.

It is important to note that the selected head has a tolerance of ± 5% due to the tolerance of the pump curve. Consequently, there is a risk that the pump capacity will differ from the selected. If the required capacity is the exact value of the process, it is recommended to adjust to the required duty point by means of flow control. Flow control method is more specifically described in section 7.3.2.

7.4.2 Example 2

The following example in Fig. 7.4.2a shows a centrifugal pump to be sized for a typical dairy process.

Pump 'A' is a Raw Milk pump in connection with a pasteuriser. The raw milk is pumped from a Balance Tank to a Separator via the preheating stage in the plate heat exchanger.
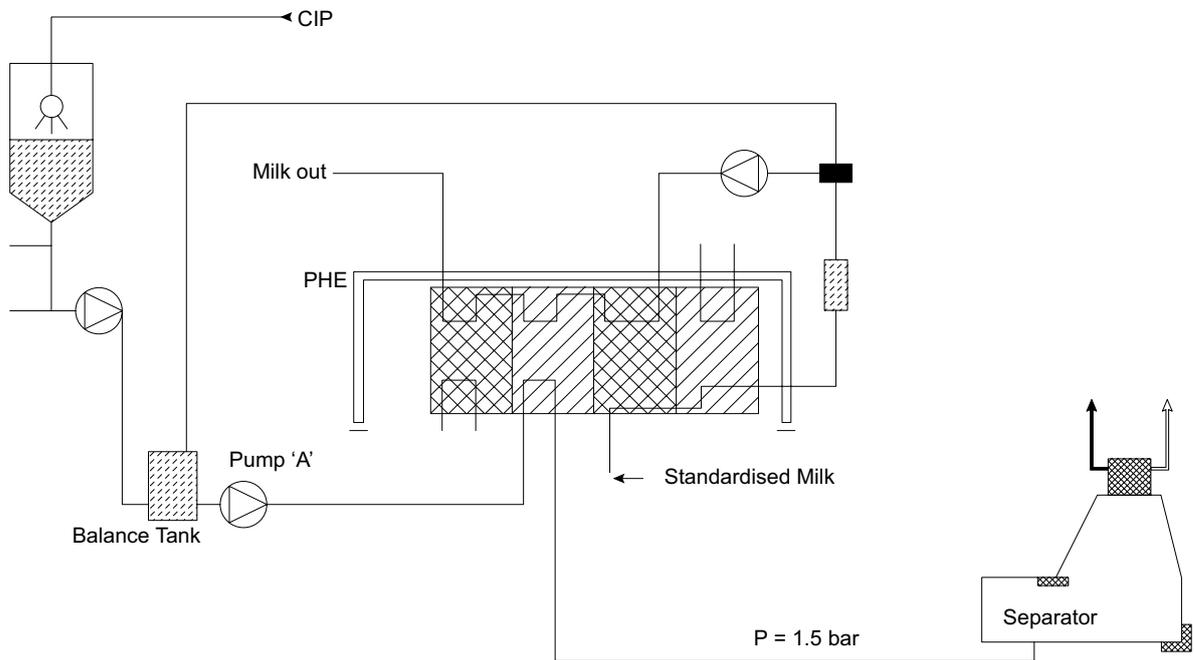
Fig. 7.4.2a Example 2

As described in section 7.1 in order to correctly size any type of pump, some essential information is required such as Product/Fluid data, Performance data and Site Services data.

Product/Fluid data:

Fluid to be pumped   -   Raw Milk

Viscosity   -   5 cP

Pumping temperature   -   5° C

Performance data:

Capacity   -   30 m³/h

Discharge   -   via 5 m of horizontal 76 mm dia. tube, plus a given number of bends, valves and a plate heat exchanger with Δp_{PHE} 1 bar. Inlet pressure for the separator = 1.5 bar

Suction   -   0.1 m head, plus a given number of bends and valves

Site Services data:

Electrical supply   -   400v, 50 Hz

All the data has been given by the customer.
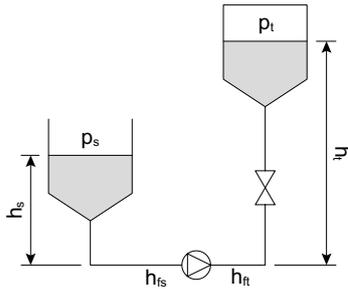
Fig. 7.4.2b Typical suction / Discharge Head set-up

Before sizing a pump, it will be necessary to determine the total head and NPSHa (Fig. 7.4.2b). The theory, including the different formula regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge Head $H_t = h_t + h_{ft} + p_t$

Where:

h_t = Static Head to Separator
h_{ft} = Total Pressure Drop in Discharge Line
p_t = Pressure in Separator

Therefore:

h_t = 0 m (no static head – only horizontal tube)
h_{ft} = Pressure Drop in Tube Δp_{tube}
    + Pressure Drop in Bends and Valves Δp
    + Pressure Drop in Plate Heat Exchanger p_{PHE}
Δp_{tube} (from curve shown in 14.5) = 0.2 m
    (4 m loss per 100 m)
Δp is calculated to be 0.1 m
Δp_{PHE} is given as 1.0 bar = 10 m
h_{ft} = 0.2 + 0.1 + 10 m = 10.3 m
p_t = 1.5 bar = 15 m

$H_t = h_t + h_{ft} + p_t = 0 + 10.3 + 15$ m = 25.3 m (2.53 bar)

Total suction head $H_s = h_s - h_{fs} + p_s$

Where:

h_s = Static Suction Head in Balance Tank
h_{fs} = Total Pressure Drop in Suction Line
p_s = Pressure in Balance Tank (open tank)

Therefore:

h_s = 0.1 m
h_{fs} = Calculated to be 0.4 m
p_s = 0 (open tank)

$H_s = h_s - h_{fs} + p_s = 0.1 - 0.4 + 0$ m = - 0.3 m (- 0.03 bar)

Total Head $H = H_t - H_s = 25.3 - (- 0.3) = 25.6$m (2.56 bar)

NPSHa

NPSHa	$=$ $Pa + h_s - h_{fs} - Pvp$

Where:

Pa	$=$	Pressure Absolute above Level of Fluid in Balance Tank
h_s	$=$	Static Suction Head in Balance Tank
h_{fs}	$=$	Total Pressure Drop in Suction Line
Pvp	$=$	Vapour Pressure of Fluid

Therefore:

Pa	$=$	1 bar (open tank) = 10 m
h_s	$=$	0.1 m
h_{fs}	$=$	Calculated to be 0.4 m
Pvp	$=$	At temperature of 5° C this is taken as being negligible i.e., 0 bar a (0.008 bar) = 0 m

NPSHa = 10 + 0.1 - 0.4 - 0 (m) = 9.7 m

As the fluid to be pumped has physical properties (i.e., viscosity) different from water, the pump performance curves should not be used, and actual pump sizing should be made using the pump selection program.

For this particular example, pump sized would be as follows:

Pump Model	-	LKH-20
Impeller size	-	144 mm
Speed	-	2883 rev/min
Capacity	-	30 m³/h
Head	-	25.6m (2.56 bar)
Efficiency	-	62.7%
NPSHr	-	1.4 m
Motor size	-	4 kW

Cavitation check

NPSHa should be greater than NPSHr i.e., 9.7 m > 1.4 m, i.e., no cavitation will occur.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a single mechanical seal with carbon/silicon carbide faces and EPDM elastomers.

7.4.3 Example 3

A CIP return pump is to be sized for an application for the following details given by the customer:

Product/Fluid data:

Fluid to be pumped	-	CIP return
Viscosity	-	1 cP
Pumping temperature	-	5° C – 90° C

Performance data:

Capacity	-	30 m³/h
Discharge	-	via 5 m of horizontal 76 mm dia. tube, plus a given number of bends, valves and a plate heat exchanger with Δp_{PHE} 1 bar as well as 1.5 bar over a separator
Suction	-	0.5 m static head, plus a given number of bends and valves all together with a friction loss of 0.5 m

Site Services data:

Electrical supply	-	400v, 50 Hz

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge head H_t = $h_t + h_{ft} + p_t$

Taken from Example 2 = 25.3 m (2.53 bar)

Total Suction Head H_s = $h_s - h_{fs} + p_s$

Where:

h_s =   Static Suction Head in Balance Tank
h_{fs} =   Total Pressure Drop in Suction Line
p_s =   Pressure in Balance Tank (open tank)

Therefore:

h_s =   0.5 m
h_{fs} =   Calculated to be 0.5 m
p_s =   0 (open tank)

$H_s = h_s - h_{fs} + p_s = 0.5 - 0.5 + 0 \text{ m} = 0 \text{ m} = (0 \text{ bar})$

Total head $H = H_t - H_s = 25.3 - 0 = 25.3$ m (2.53 bar)

NPSHa

NPSHa   = $P_a + h_s - h_{fs} - P_{vp}$

Where:

P_a =   Pressure Absolute above Level of Fluid in Balance Tank
h_s =   Static Suction Head in Balance Tank
h_{fs} =   Total Pressure Drop in Suction Line
P_{vp} =   Vapour Pressure of Fluid

Therefore:

P_a =   1 bar (open tank) = 10 m
h_s =   0.5 m
h_{fs} =   Calculated to be 0.5 m
P_{vp} =   At temperature of 90° C (important to check at highest temperature. Table 14.4) = 0.7 bar = 7 m

NPSHa = 10 + 0.5 - 0.5 - 7 (m) = 3 m

Using a sizing program we get the option of a LKH Prime-20:

Pump Model	-	LKH Prime-20
Impeller size	-	155 mm
Speed	-	2925 rev/min
Capacity	-	30 m³/h
Head	-	25.3 m (2.53 bar) - it is important not to oversize as air evacuation capability diminish below 2800 rpm
Efficiency	-	49.9%
NPSHr	-	4.1 m
Power absorbed	-	4.4 kW
Motor size	-	5.5 kW

Cavitation check

NPSHa should be greater than NPSHr i.e., 3 m > 4.1 m. This means that the pump would cavitate at the 90° C so we should look for a larger pump or maybe decrease the temperature.

Checking another size pump, LKH-40, the option is the below:

Pump Model	-	LKH Prime-40
Impeller size	-	162 mm
Speed	-	2903 rev/min
Capacity	-	30 m³/h
Head	-	25.3 m (2.53 bar) - it is important not to oversize as air evacuation capability diminish below 2800 rpm
Efficiency	-	34.6%
NPSHr	-	2.5 m
Power absorbed	-	6.1 kW
Motor size	-	7.5 kW

Cavitation check

NPSHa should be greater than NPSHr i.e., 3 m > 2.5 m.

This will work. However, the efficiency is lower resulting in higher power consumption so it is worth checking if temperature could be reduced to 80 – 85° C.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a single mechanical seal with carbon/silicon carbide faces and EPDM elastomers.

If any risk of dry running a flush should be added.

7.5 Worked Examples – Centrifugal Pump Sizing

US units

7.5.1 Example 1

The following example in Fig. 7.5.1a shows a pump to be sized for a typical brewery process.

The pump is required to handle Wort from the Whirlpool to the Fermentation vessel.
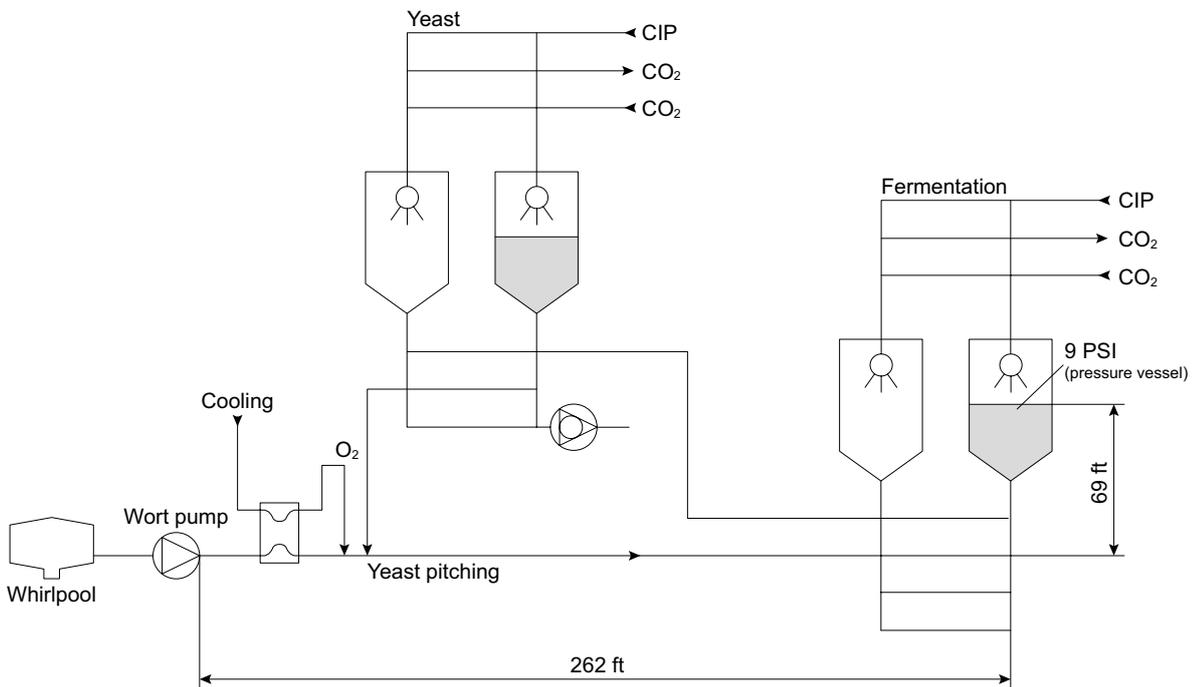
Fig. 7.5.1a Example 1

As described in section 7.1 in order to correctly size any type of pump, some essential information is required such as Product/Fluid data, Performance data and Site Services data.

Product/Fluid data:

Fluid to be pumped	-	Wort
Viscosity	-	1 cP
Pumping temperature	-	194° F

Performance data:

Capacity	-	176 US gal/min
Discharge	-	via 262 ft of 4 in dia. tube, plus a given number of bends, valves, and a plate heat exchanger with Δp_{PHE} 23 PSI. Static head in Fermenting vessel = 69 ft. Pressure in Fermenting vessel = 9 PSI
Suction	-	1.5 ft head, plus a given number of bends and valves

Site Services data:

Electrical supply	-	460v, 60 Hz

All the data has been given by the customer.
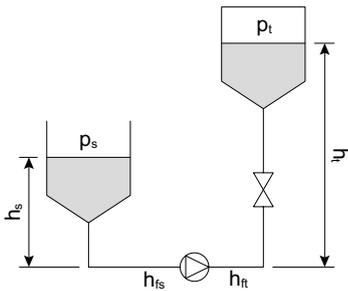
Fig. 7.5.1b Typical suction / Discharge Head set-up

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge Head $H_t = h_t + h_{ft} + p_t$

Where:

h_t = Static Head in Fermentation Vessel
h_{ft} = Total Pressure Drop in Discharge Line
p_t = Pressure in Fermentation Vessel

Therefore:

h_t = 69 ft
h_{ft} = Pressure Drop in Tube Δp_{tube}
    + Pressure Drop in Bends and Valves Δp
    + Pressure Drop in Plate Heat Exchanger Δp_{PHE}
Δp_{tube} (from curve shown in 14.5) = 4.7 ft
    (5.9 ft loss per 328 ft) = for 262 ft tube
    - loss 4.7 ft
Δp is calculated to be 16 ft
Δp_{PHE} is given as 23 PSI = 53 ft
h_{ft} = 4.7 + 16 + 53 ft = 73.7 ft
p_t = 9 PSI = 20 ft

$H_t = h_t + h_{ft} + p_t = 69 + 73.7 + 20$ ft = 162.7 ft (70.5 PSI)

Total suction head $H_s = h_s - h_{fs} + p_s$

Where:

h_s = Static Suction Head in Whirlpool
h_{fs} = Total Pressure Drop in Suction Line
p_s = Pressure in Whirlpool (open tank)

Therefore:

h_s = 1.5 ft
h_{fs} = Calculated to be 3 ft
p_s = 0 (open tank)

$H_s = h_s - h_{fs} + p_s = 1.5 - 3 + 0$ m = - 1.5 ft (- 0.6 PSI)

Total head $H = H_t - H_s = 162.7 - (- 1.5) = 164.2$ ft (71.2 PSI)

NPSHa

NPSHa	=	$Pa + h_s - h_{fs} - Pvp$	

Where:

Pa	=	Pressure Absolute above Level of Fluid in Whirlpool Tank
h_s	=	Static Suction Head in Whirlpool Tank
h_{fs}	=	Total Pressure Drop in Suction Line
Pvp	=	Vapour Pressure of Fluid

Therefore:

Pa	=	14.7 PSI (open tank) = 33.9 ft
h_s	=	1.5 ft
h_{fs}	=	Calculated to be 3 ft
Pvp	=	10 PSIA (from table 14.4a) = 23 ft

NPSHa = 33.9 + 1.5 - 3 - 23 (ft) = 9.4 ft

Actual pump sizing can be made using pump performance curves or a pump selection program. The performance curves are, however, not suitable if the fluid to be pumped has physical properties (i.e., viscosity) different from water. In this particular example both the pump performance curves and pump selection program can be used. The performance curve selection procedure is more specifically described in section 7.3.1.

For this particular example, pump sized would be as follows:

Pump Model	-	LKH-20
Impeller size	-	6.50 in
Speed	-	3500 rev/min
Capacity	-	176 US gal/min
Head	-	164.2 ft (71.2 PSI)
Efficiency	-	67.25%
NPSHr	-	7.5 ft
Motor size	-	15 hp

Cavitation check

NPSHa should be greater than NPSHr i.e. 9.4 ft > 7.5 ft, i.e., no cavitation will occur.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a double mechanical seal with carbon/silicon carbide faces and EPDM elastomers.

Fig. 7.5.1c Q - H charateristic when changing valves

Special Note

The discharge head (h_{t2}) is lower when the pump starts filling the fermenting vessel compared to the discharge head (h_{t1}) when the vessel is full. The reduction of the discharge head will result in higher flow. This will reduce the NPSHr and can therefore lead to cavitation. Another risk is that the larger capacity and thereby larger power consumption can cause overloading of the motor (see Fig. 7.5.1c).

Cavitation can be avoided by reducing the pump speed (reducing NPSHr), i.e., by means of a frequency converter, or by throttling the discharge line (increasing head). The flow control method is more specifically described in section 7.3.2.

Adjustment

In this example the pump is sized by the pump selection program which results in exact impeller diameter of 6.50 in, so that the selected duty point is as close to the required duty point as possible.

The pump is sized with a standard impeller diameter if using the performance curve. In this case it may be necessary to adjust the selected duty point by means of flow control.

It is important to note that the selected head has a tolerance of ± 5% due to the tolerance of the pump curve. Consequently, there is a risk that the pump capacity will differ from the selected. If the required capacity is the exact value of the process, it is recommended to adjust to the required duty point by means of flow control. Flow control method is more specifically described in section 7.3.2.

7.5.2 Example 2

The following example in Fig. 7.5.2a shows a centrifugal pump to be sized for a typical dairy process.

Pump 'A' is a Raw Milk pump in connection with a pasteuriser. The raw milk is pumped from a Balance Tank to a Separator via the preheating stage in the plate heat exchanger.
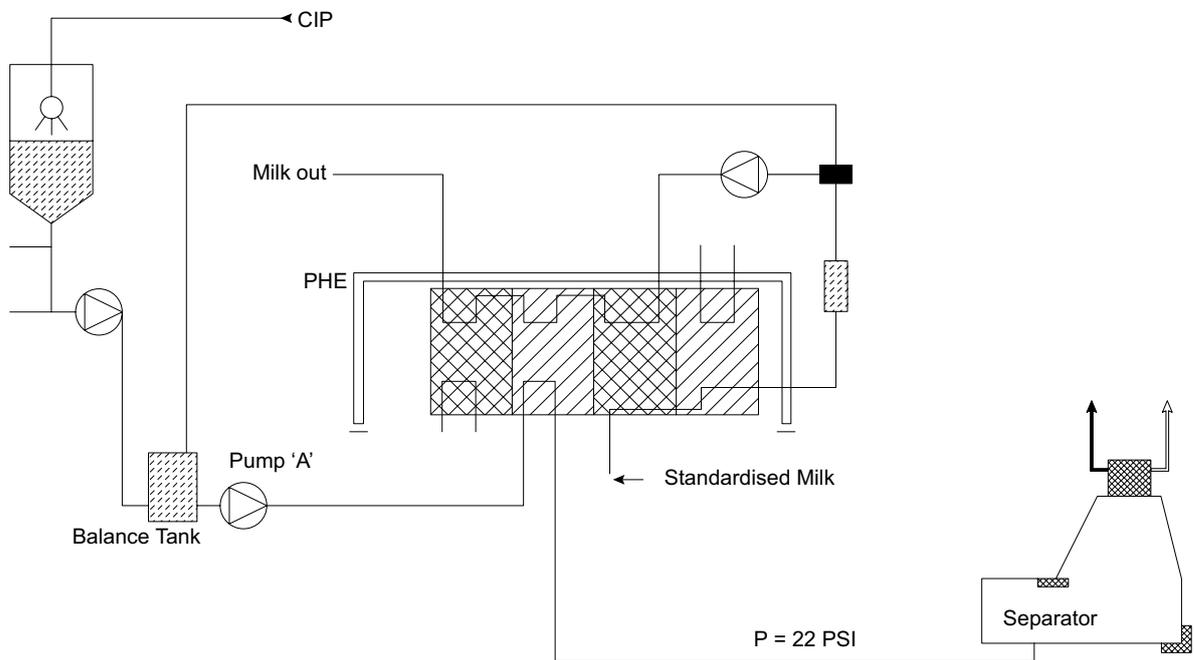
Fig. 7.5.2a Example 2

As described in 7.1 in order to correctly size any type of pump, some essential information is required such as Product/Fluid data, Performance data and Site Services data.

Product/Fluid data:

Fluid to be pumped	-	Raw Milk
Viscosity	-	5 cP
Pumping temperature	-	41° F

Performance data:

Capacity	-	132 US gal/min
Discharge	-	via 16 ft of horizontal 3 in dia. tube, plus a given number of bends, valves, and a plate heat exchanger with Δp_{PHE} 15 PSI. Inlet pressure for the separator = 22 PSI
Suction	-	0.3 ft head, plus a given number of bends, and valves

Site Services data:

Electrical supply	-	460v, 60 Hz

All the data has been given by the customer.
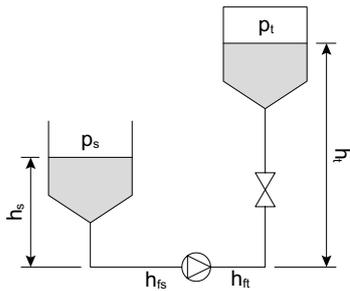
Fig. 7.5.2b Typical suction / Discharge Head set-up

Before sizing a pump, it will be necessary to determine the total head and NPSHa (Fig. 7.5.2b). The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge Head H_t $= h_t + h_{ft} + p_t$

Where:

h_t =   Static Head to Separator
h_{ft} =   Total Pressure Drop in Discharge Line
p_t =   Pressure in Separator

Therefore:

h_t =   0 ft (no static head - only horizontal tube)
h_{ft} =   Pressure Drop in Tube Δp_{tube}
        + Pressure Drop in Bends and Valves Δp
        + Pressure Drop in Plate Heat Exchanger Δp_{PHE}
Δp_{tube} (from curve shown in 14.5) = 0.6 ft
        (4 m loss per 100 m = 0.2 m = 0.6 ft)
Δp is calculated to be 0.3 ft
Δp_{PHE} is given as 15 PSI = 34 ft
h_{ft} =   0.6 + 0.3 + 34 ft = 34.9 ft
p_t =   22 PSI = 50 ft

$H_t = h_t + h_{ft} + p_t = 0 + 34.9 + 50$ ft = 84.9 ft (36.8 PSI)

Total suction head H_s $= h_s - h_{fs} + p_s$

Where:

h_s =   Static Suction Head in Balance Tank
h_{fs} =   Total Pressure Drop in Suction Line
p_s =   Pressure in Balance Tank (open tank)

Therefore:

h_s =   0.3 ft
h_{fs} =   Calculated to be 1.3 ft
p_s =   0 (open tank)

$H_s = h_s - h_{fs} + p_s = 0.3 - 1.3 + 0$ m = - 1 ft (- 0.4 PSI)

Total head $H = H_t - H_s = 84.9 - (- 1) = 85.9$ ft (37.2 PSI)

NPSHa

NPSHa	=	$Pa + h_s - h_{fs} - Pvp$

Where:

Pa	=	Pressure Absolute above Level of Fluid in Balance Tank
h_s	=	Static Suction Head in Balance Tank
h_{fs}	=	Total Pressure Drop in Suction Line
Pvp	=	Vapour Pressure of Fluid

Therefore:

Pa	=	14.7 PSI (open tank) = 33.9 ft
h_s	=	0.3 ft
h_{fs}	=	Calculated to be 1.3 ft
Pvp	=	At temperature of 41° F this is taken as being negligible i.e., 0 PSIA = 0 ft

NPSHa = 33.9 + 0.3 - 1.3 - 0 (ft) = 32.9 ft

As the fluid to be pumped has physical properties (i.e., viscosity) different from water, the pump performance curves should not be used, and actual pump sizing should be made using the pump selection program.

For this particular example, pump sized would be as follows:

Pump Model	-	LKH-10
Impeller size	-	5.51 in
Speed	-	3450 rev/min
Capacity	-	132 US gal/min
Head	-	84.9 ft (36.8 PSI)
Efficiency	-	65.4%
NPSHr	-	4.6 ft
Motor size	-	5.0 hp

Cavitation check

NPSHa should be greater than NPSHr i.e., 32.9 ft > 4.6 ft, i.e., no cavitation will occur.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a single mechanical seal with carbon/silicon carbide faces and EPDM elastomers.

7.5.3 Example 3

A CIP return pump is to be sized for an application for the following details given by the customer.

Product/Fluid data:

Fluid to be pumped	-	CIP return
Viscosity	-	1 cP
Pumping temperature	-	41° F - 194° F

Performance data:

Capacity	-	132 GPM
Discharge	-	via 16 ft of horizontal 3 in dia. tube, plus a given number of bends, valves, and a plate heat exchanger with Δp_{PHE} 15 PSI and 22 PSI over a separator
Suction	-	1.6 ft static head, plus a given number of bends and valves all together with a friction loss of 1.6 ft

Site Services data:

Electrical supply	-	460v, 60 Hz

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge head H_t = $h_t + h_{ft} + p_t$

Taken from Example 2 (same head) = 84.9 ft (36.8 PSI)

Total Suction Head H_s = $h_s - h_{fs} + p_s$

Where:

h_s = Static Suction Head in Balance Tank
h_{fs} = Total Pressure Drop in Suction Line
p_s = Pressure in Balance Tank (open tank)

Therefore:

h_s = 1.6 ft
h_{fs} = Calculated to be 1.6 ft
p_s = 0 (open tank)

$H_s = h_s - h_{fs} + p_s = 1.6 - 1.6 + 0$ m = 0 ft = (0 PSI)

Total head $H = H_t - H_s = 85.9 - 0 = 85.9$ ft (37.2 PSI)

NPSHa

NPSHa = $Pa + h_s - h_{fs} - Pvp$

Where:

Pa = Pressure Absolute above Level of Fluid in Balance Tank
h_s = Static Suction Head in Balance Tank
h_{fs} = Total Pressure Drop in Suction Line
Pvp = Vapour Pressure of Fluid

Therefore:

Pa = 14.7 PSI (open tank) = 33.9 ft
h_s = 1.6 ft
h_{fs} = Calculated to be 1.6 ft
Pvp = At temperature of 194° F (important to check at highest temperature. Table 14.4)
0.7 bar = 23.4 ft

NPSHa = 33.9 + 1.6 - 1.6 - 23.4 ft = 10.5 ft

Using a sizing program we get the option of a LKH Prime-20:

Pump Model	-	LKH Prime-20
Impeller size	-	5.31 in
Speed	-	3520 rev/min
Capacity	-	132 GPM
Head	-	85.9 ft (37.2 PSI) - it is important not to oversize as air evacuation capability diminish below 2800 rpm
Efficiency	-	44.91%
NPSHr	-	13.6 ft
Power absorbed	-	6.3 hp
Motor size	-	7.5 hp

Cavitation check

NPSHa should be greater than NPSHr i.e., 10.5 ft > 13.6 ft.

This means that the pump would cavitate at the 194° F so we should look for a larger pump or maybe decrease the temperature.

Checking another size pump it will be necessary to use a variable frequency drive if we would still want to have a LKH-Prime pump. Using a sizing program we can get the below:

Checking another size pump, LKH-40 the option is the below:

Pump Model	-	LKH Prime-40
Impeller size	-	6.3 in
Speed	-	2950 rev/min
Capacity	-	132 GPM
Head	-	85.9 ft (37.2 PSI) - it is important not to oversize as air evacuation capability diminish below 2800 rpm
Efficiency	-	36.15%
NPSHr	-	7.9 ft
Power absorbed	-	7.8 hp
Motor size	-	10 hp

Cavitation check

NPSHa should be greater than NPSHr i.e., 10.5 ft > 7.9 ft.
This would work. However, the efficiency goes down and higher power consumption so it is worth checking if temperature could be reduced to 176 – 185° F.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a single mechanical seal with carbon/silicon carbide faces and EPDM elastomers. If any risk of dry running add a flush.
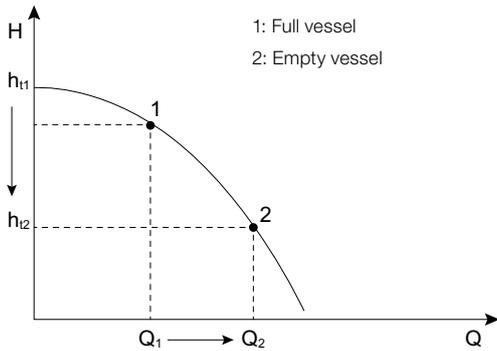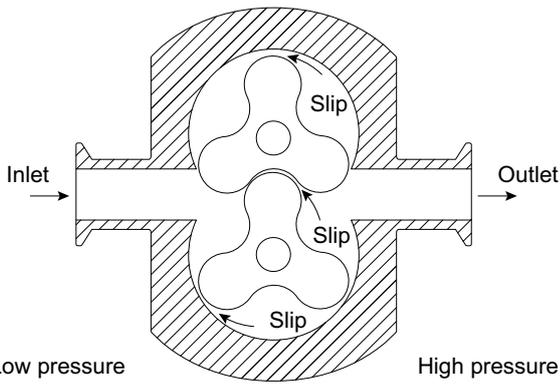
Fig. 7.6.1a Slip

7.6 Positive displacement Pumps

7.6.1 Slip

Slip is the fluid lost by leakage through the pump clearances. The direction of slip will be from the high pressure to the low pressure side of the pump i.e. from pump outlet to pump inlet (see Fig. 7.6.1a). The amount of slip is dependent upon several factors.

Clearance effect

Increased clearances will result in greater slip. The size and shape of the rotor will be a factor in determining the amount of slip.
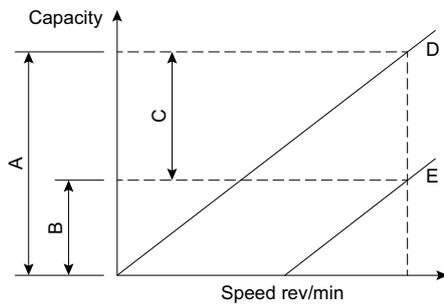
A = Capacity at '0' bar   D = '0' bar
B = Actual capacity at 'X' bar   E = 'X' bar
C = Slip

Fig. 7.6.1b Pressure effect
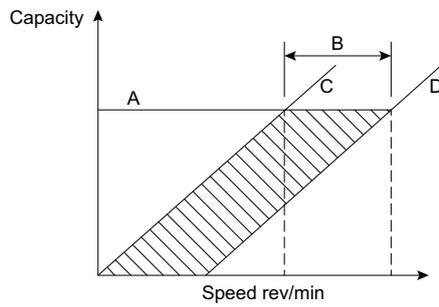
A = Required capacity   C = '0' bar
B = Speed increase to   D = 'X' bar
    maintain capacity

Fig. 7.6.1c Pressure effect

Pressure effect

The amount of slip will increase as pressure increases which is shown above. In Fig 7.6.1b for a given pump speed the amount of slip can be seen as the capacity at 'zero' bar less the capacity at 'X' bar. To overcome this amount of slip it will be necessary to increase the pump speed to maintain the capacity required as shown in Fig 7.6.1c.
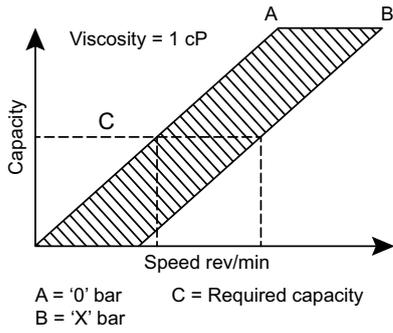
Fig. 7.6.1d Viscosity effect
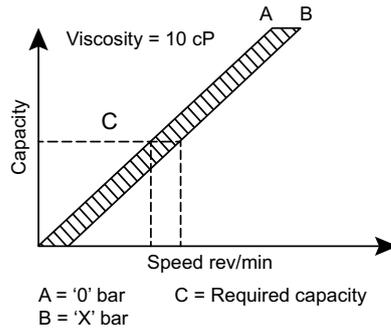
Fig. 7.6.1e Viscosity effect
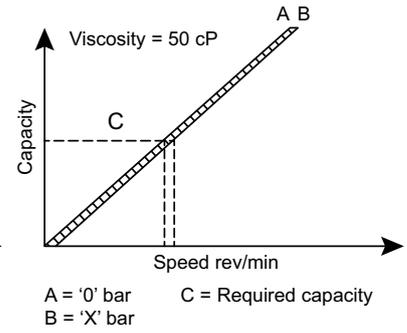
Fig. 7.6.1f Viscosity effect

Viscosity effect

The amount of slip will decrease as fluid viscosity increases. The effect of viscosity on slip is shown in Fig. 7.6.1d, 7.6.1e and 7.6.1f above. The pressure lines will continue to move towards the 'zero' pressure line as the viscosity increases.
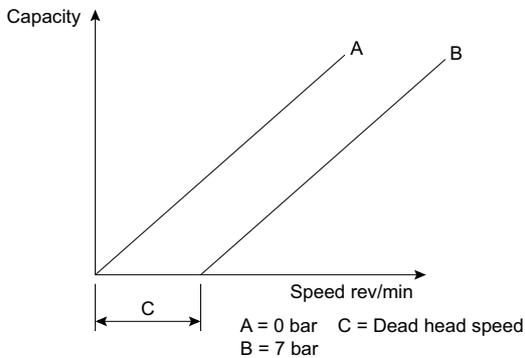
Fig. 7.6.1g Dead head speed

Pump Speed effect

Slip is independent of pump speed. This factor must be taken into consideration when operating pumps at low speeds with low viscosity fluids (Fig. 7.6.1g). For example, the amount of slip at 400 rev/min pump speed will be the same as the amount of slip at 200 rev/min pump speed providing pressure is constant.

The pump speed required to overcome slip is known as the 'dead head speed'.

It is important to note that flow will be positive after overcoming the dead head speed.

A summary of effects of different parameters on slip is shown below (Fig. 7.6.1j):

| Increases with Pressure | Increases with Clearances | Decreases with Viscosity |
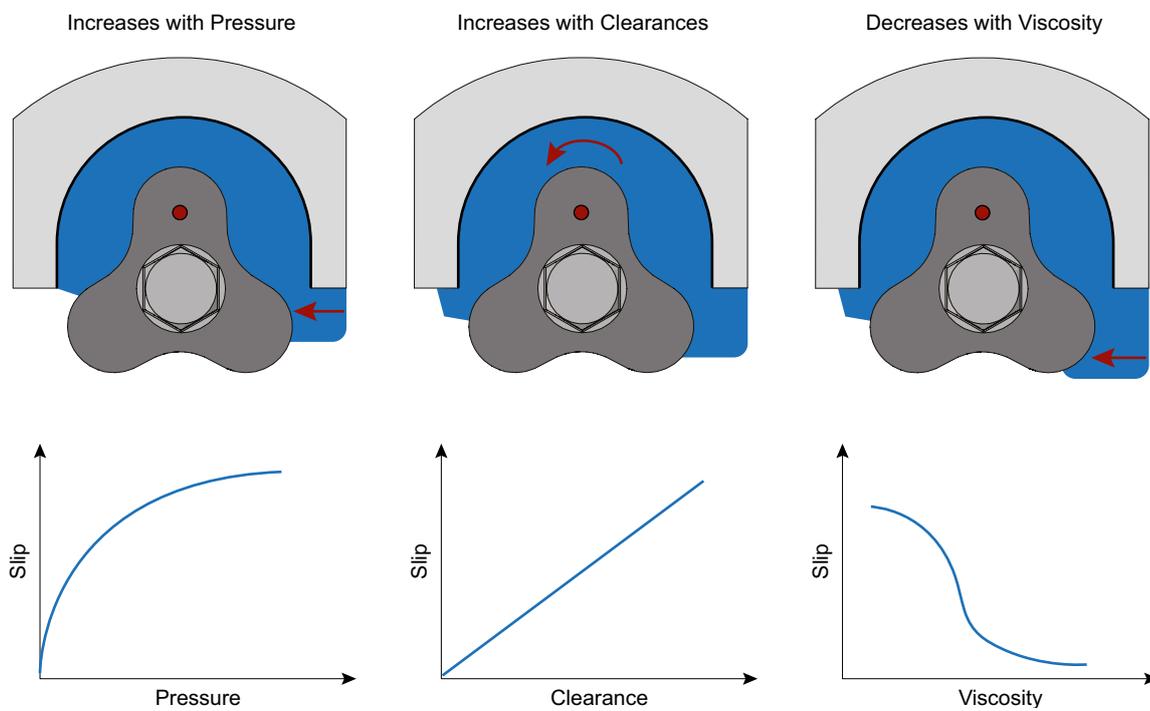
Fig. 7.6.1j

It is worth noticing that the clearances in a circumferential piston pump like the Alfa Laval DuraCirc pump is smaller than in a rotary lobe pump making the DuraCirc a lot less sensitive to slip.

The Twin Screw pump such as the Alfa Laval OS will have relatively larger clearances due to screw length and overhang. As such this means greater slip but this can be compensated with increased speed where the product allows for this.

7.6.2 Initial Suction Line Sizing

In general terms it is common to find the recommendation for the inlet pipe size to be the same diameter as the pump inlet connection.

For guidance purposes only on high viscosity duties, the suction line can be initially sized using the initial suction line sizing curve (see section 14.9) where the relationship between viscosity and flow rate provides an indication of pipe sizing.

For example, for a flow rate of 10 m³/h on a fluid with viscosity 900 cSt, a pump with 40 mm (1.5 in) diameter suction line would be initially selected.

It is important to note this is only an approximate guide and care should be taken not to exceed the pump's viscosity/speed limit.

7.6.3 Performance Curve

Alfa Laval positive displacement pumps can be sized from published performance curves or a pump selection program. Due to pump head clearances described in section 8.2.2, different performance curves are used for SRU pumps for the various temperature ratings for rotors i.e., 70° C (158° F), 130° C (266° F) and 200° C (392° F). The SX pump range has only 150° C (302° F) and the Optilobe pump range has only 130° C (266° F) temperature ratings. The Circumferential Piston Pump, DuraCirc, has 150° C (302° F) temperature rating.

For the Alfa Laval OS Twin Screw pump curves has not been published and should therefore always be selected using a pump selection program. This is to avoid any issues when several duties have to be considered.

If access to Anytime a selection guideline is avaiable here for the OS pump.

How to use the Performance Curve

There are two kinds of performance curves.

For the first Fig. 7.6.3a.

The performance curve consists of four different curves:

- Capacity as a function of speed, related to pressure and viscosity

- Power as a function of speed, related to pressure and viscosity of 1 cSt
  (see table 14.3.10 for viscosity conversion)

- Power as a function of viscosity greater than 1 cSt

- Speed as a function of viscosity

The curves are based on water at 20° C (68° F) but are shown with calculated viscosity correction data. Example shown refers to the SRU pump range, but the same sizing procedure is also used for the SX pump range.

Example

Product/Fluid data:

Fluid to be pumped	-	Vegetable Oil
Viscosity	-	100 cSt
Pumping temperature	-	30° C (86° F)

Performance data:

Capacity	-	3.6 m³/h (15.8 US gal/min)
Total Pressure	-	8 bar (116 PSIG)

The optimum is to size the smallest pump possible as hydraulic conditions dictate. However other factors such as fluid behaviour, solids etc. should be considered.

Typical Performance curve
SRU2/013/with mechanical seal

Rotors - St. Stl.
Trilobe
10 Bar (145 PSI)
70° (158 °F)
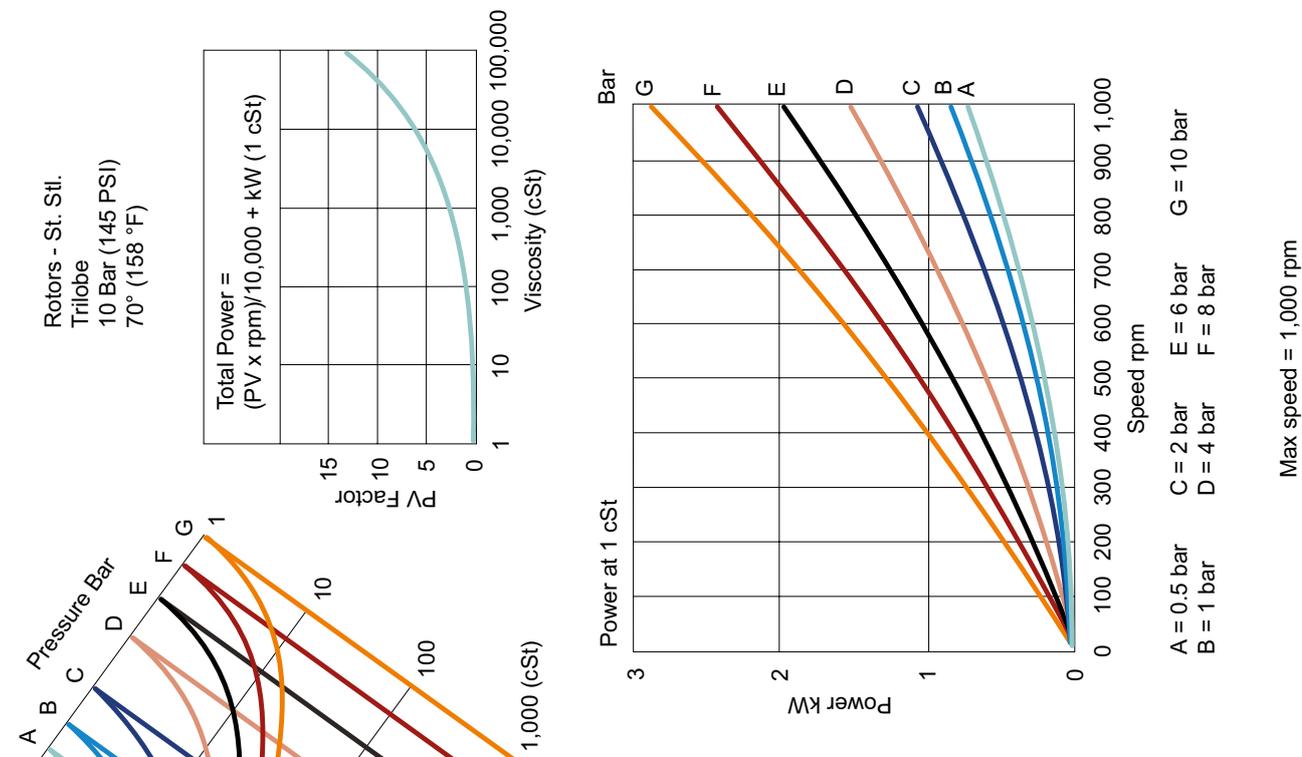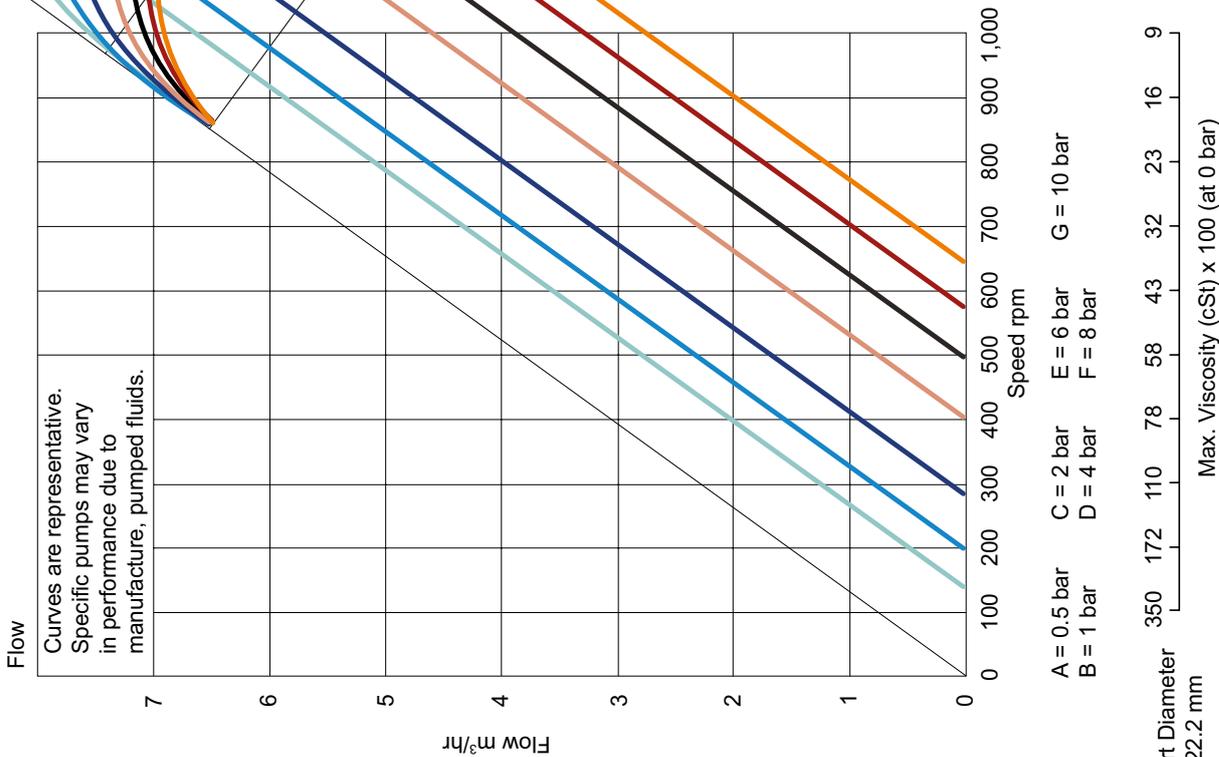
Fig. 7.6.3a SRU2/013/LS curves

Step 1 – Find Appropriate Curve

Locate a curve for the required pump model that covers the performance data. Due to the large number of curves available it is not practical to include all performance curves in this handbook. Curves can be obtained from the pump supplier, or downloaded from our Anytime program if you have access. However, the sizing process would be as follows:

From the initial suction line sizing curve (see section 14.9), a pump with a size 25 mm (1 in) inlet connection would be required. Although the smallest pump models SRU1/005 and SRU1/008 have 25 mm (1 in) pump inlet connections, the flow rate required would exceed the pumps speed limit on the performance curve. For this particular example, we therefore need to select a performance curve for the pump model SRU2/013/LS with 70° C (158° F) rotor clearances, as shown in Fig. 7.6.3a, being the next appropriate pump size.

Step 2 – Find Viscosity and Pressure

Begin with viscosity and find the intersection point with duty pressure.
From example – 100 cSt and 8 bar (115 PSIG).

Step 3 – Find Flow Rate

Move diagonally downward and find intersection with required flow rate.
From example – 3.6 m³/h (15.8 US gal/min).

Step 4 – Find Speed

Move vertically downward to determine necessary pump speed.
From example – 600 rev/min.

Step 5 – Viscosity/Port Size Check

Move vertically downward and check that maximum viscosity rating has not been exceeded against relevant inlet size.
From example – maximum viscosity rating 4300 cSt.

Step 6 – Find Power

The power required by a pump is the summation of the hydraulic power and various losses that occur in the pump and pumping system. Viscosity has a marked effect on pump energy losses. The losses being due to the energy required in effecting viscous shear in the pump clearances. Viscous power is the power loss due to viscous fluid friction within the pump (Pv factor).

It should be noted, this is the power needed at the pump shaft and the appropriate motor power must be selected, which in this instance would be 1.5 kW being the nearest motor output power above the required power.

Typically curves are used in conjunction with equation as follows:

Total Required Power (kW) =
$$\frac{Pv \times Pump\ speed\ (rev/min)}{10000} + Hydraulic\ power\ at\ 1\ cSt\ (kW)$$

Where: Pv = Power/Viscosity Factor

From example

- At speed 600 rev/min and 8 bar the hydraulic power at 1 cSt is 1.3 kW

- At viscosity 100 cSt the Pv factor is 1.0

Total Required Power (kW) =
$$\frac{Pv \times Pump\ speed\ (rev/min)}{10000} + Hydraulic\ power\ at\ 1\ cSt\ (kW)$$

$$= \frac{1.0 \times 600 + 1.3}{10000} = 1.36\ kW\ (1.82\ hp)$$

A = 100,000 cSt   D = 20,000 cSt   G = 2,500 cSt
B = 60,000 cSt   E = 10,000 cSt   H = 1,000 cSt
C = 30,000 cSt   F = 5,000 cSt   I = 1 cSt
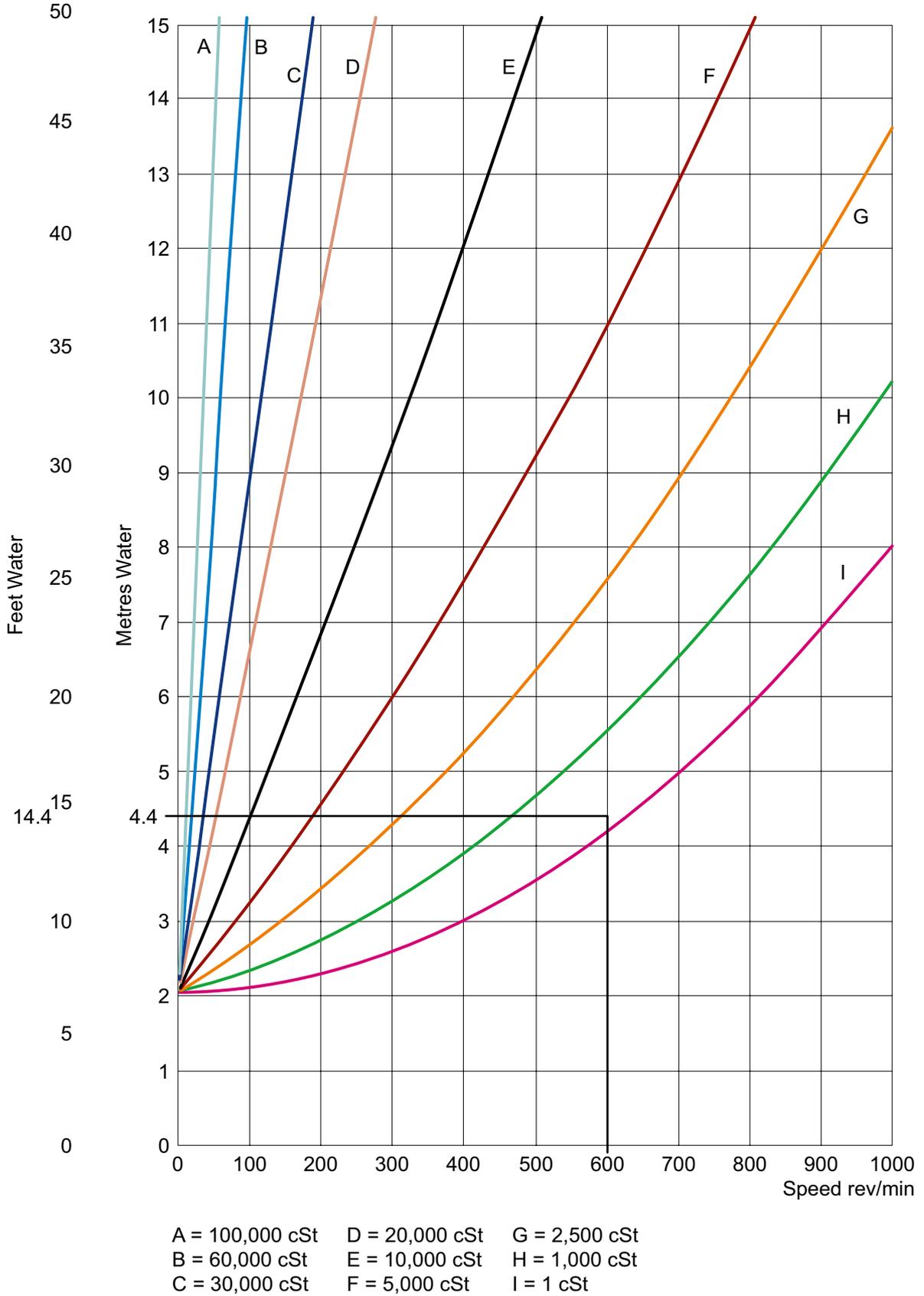
Fig. 7.6.3b SRU2 typical NPSHr curve based on water

Step 7 – Find NPSHr

NPSHr can be found by looking at the appropriate NPSH pump curve, which is a function of speed and expressed in metres water column (mwc) or feet (ft).

From example – at speed 600 rev/min and 100 cSt the NPSHr is 4.4 mwc (14.4 ft) (Fig. 7.6.3b).

For the newer curves used for Optilobe and DuraCirc (Fig. 7.6.3c).

The performance curve consists of five different curves

- Capacity as a function of speed, related to pressure at 1 cPs

- Viscosity correction factor in relation to viscosity pressure and speed

- Power as a function of speed, related to pressure and viscosity of 1 cPs
  (see table 14.3.10 for viscosity conversion)

- Power correction factor in relation to viscosity

- Speed as a function of viscosity

The curves are based on water at 20° C (68° F) but are shown with calculated viscosity correction data. Different curves exists for SI units and American units. Examples shown refers to the DuraCirc pump range, but the same sizing procedure is also used for the Optilobe pump range.
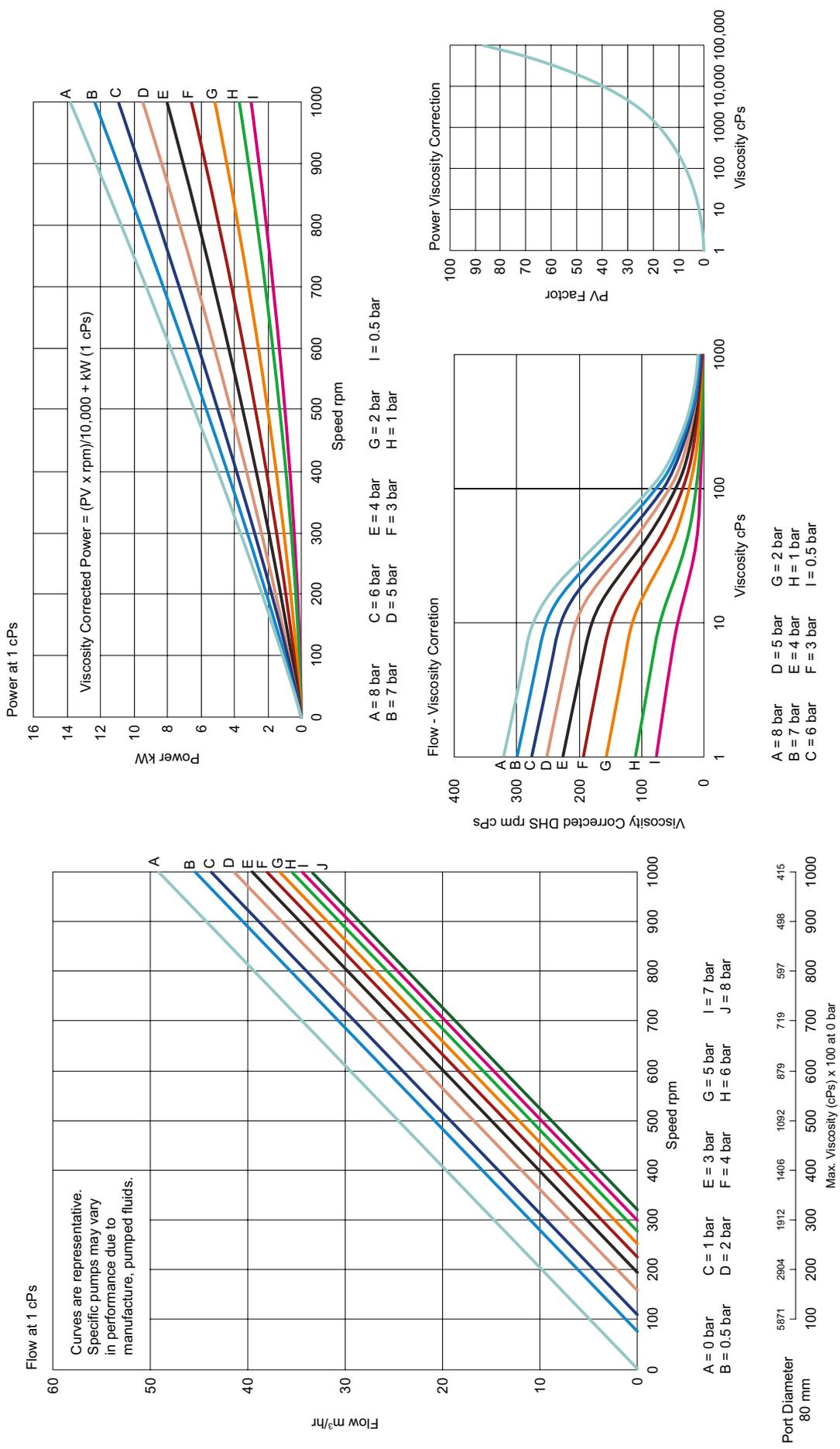
Fig. 7.6.3c OptiLobe 43 Typical performance curve based on water, with viscosity correction curves
With stainless steel 130° C TriLobe rotors and mechanical seal

Example SI units

Duty:

Viscosity: 10 cPs
Pressure: 10 bar
Flow rate: 4 m³/hr

Step 1 – to calculate speed:

- Using the Flow Viscosity Correction curve (Fig. 7.6.3d), draw a straight line up from the 10 cps viscosity point on the x-axis until it intersects the (10 bar) pressure line

- Draw a line across until it intersects the y-axis and read off the viscosity corrected dead head speed (DHS), in this case 165 rpm

Flow Viscosity Correction

A = 20 bar	D = 12 bar	G = 3 bar
B = 17 bar	E = 10 bar	H = 1 bar
C = 15 bar	F = 5 bar	I = 0.5 bar
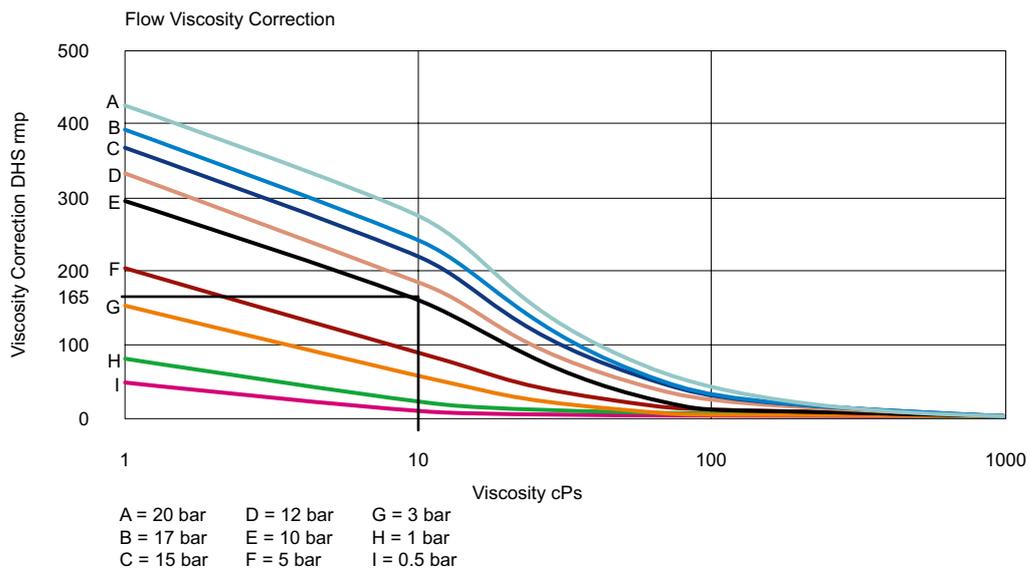
Fig. 7.6.3d

Step 2 – to calculate speed:

- Using the Flow at 1 cPs curve (Fig. 7.6.3e) starting at the viscosity corrected DHS speed value of 165 rpm, draw a line parallel to the pressure lines

- At the desired flow rate of 4 m³/hr, draw a line parallel to the x-axis, until it intersects the line drawn described below

- Then draw a line parallel to the y-axis down until it intersects the x-axis and read off the corresponding speed

- This speed (of 460 rpm) is the resulting duty speed

Flow at 1 cPs

Curves are representative. Specific pumps may vary in perdoemance due to manufacture, pumped fluids

A = 0 bar	D = 3 bar	H = 12 bar	K = 20 bar
B = 0.5 bar	F = 5 bar	I = 15 bar	
C = 1 bar	G = 10 bar	J = 17 bar	
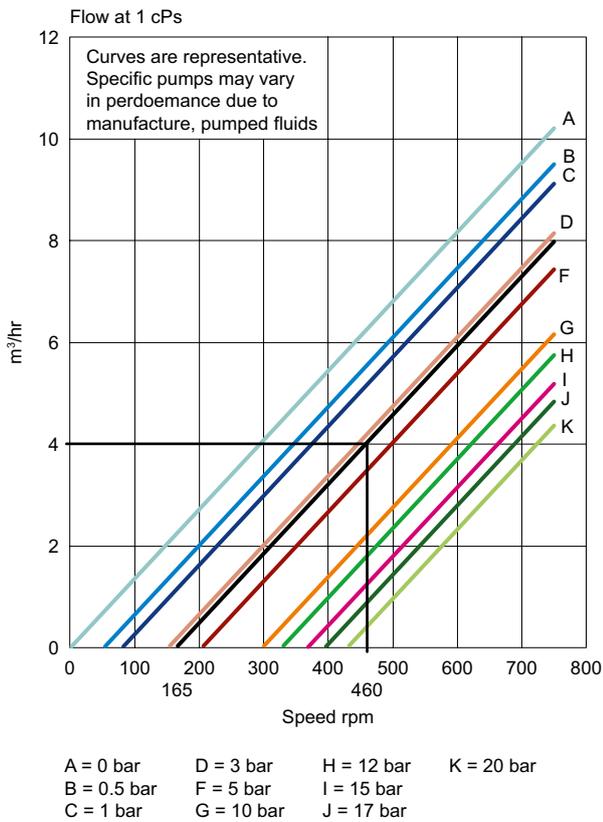
Fig. 7.6.3e

Step 3 – to calculate power:

- Using the Power at 1 cPs Curve (Fig. 7.6.3f), starting at the duty speed of 460 rpm, draw a line parallel to the y-axis, until it intersects the duty pressure line (10 bar)

- Draw a line parallel to the x-axis, until it insects the y-axis and read off the power value

- This value is the power absorbed requirement at 1 cPs (2.0 kW)

Power at 1 cPs

Viscosity Corrected Power = (PV x rmp)/10,000 + Power (1 cPs)

A = 20 bar	D = 12 bar	G = 3 bar
B = 17 bar	E = 10 bar	H = 1 bar
C = 15 bar	F = 5 bar	I = 0.5 bar
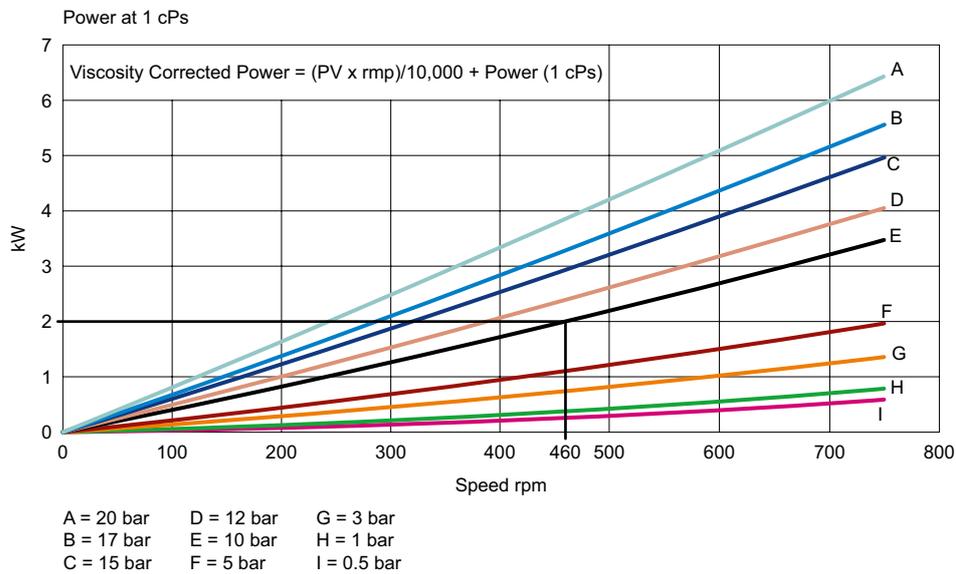
Fig. 7.6.3f

Step 4 – to calculate power:

- Using the Power Viscosity Correction curve
  (Fig. 7.6.3g), starting at the duty viscosity of 10 cPs,
  draw a line parallel to the y-axis, until it intersects
  the curve

- Draw a line parallel to the x-axis, until it insects the
  y-axis and read off the PV value (0.7)

- Using the Viscosity Corrected Power formula
  shown, calculate total power absorbed require-
  ment:

  Total power requirement
  $$= \frac{(0.7 \times 460)}{10000} + 2.0 \quad = 2.03 \text{ kW}$$

Power Viscosity Correction
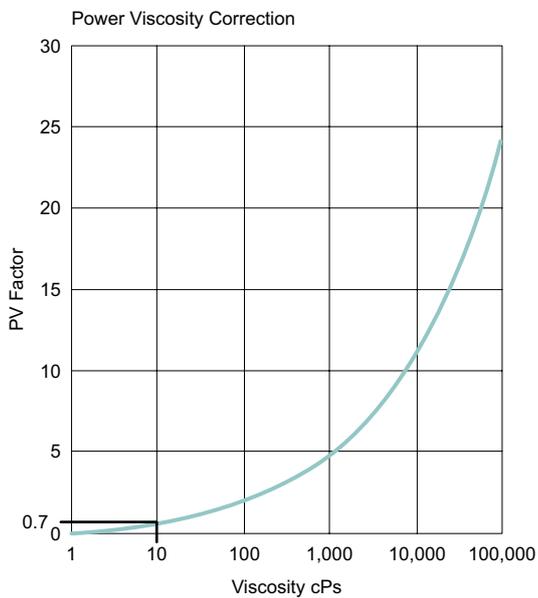
Fig. 7.6.3g

Example US units

Duty:

Viscosity: 10 cPs
Pressure: 130 PSI
Flow rate: 60 US GPM

Step 1 – to calculate speed:

- Using the Flow-Viscosity Correction curve (Fig. 7.6.3h), draw a straight line up from the 10 cPs viscosity point on the x-axis until it intersects the (130 PSI) pressure line

- Draw a line across until it intersects the y-axis and read off the viscosity corrected dead head speed (DHS), in this case 115 rpm

Flow - Viscosity Correction

A = 232.1 PSI	D = 130.5 PSI	G = 43.5 PSI
B = 188.5 PSI	E = 101.5 PSI	H = 14.5 PSI
C = 159.5 PSI	F = 72.5 PSI	I = 7.3 PSI
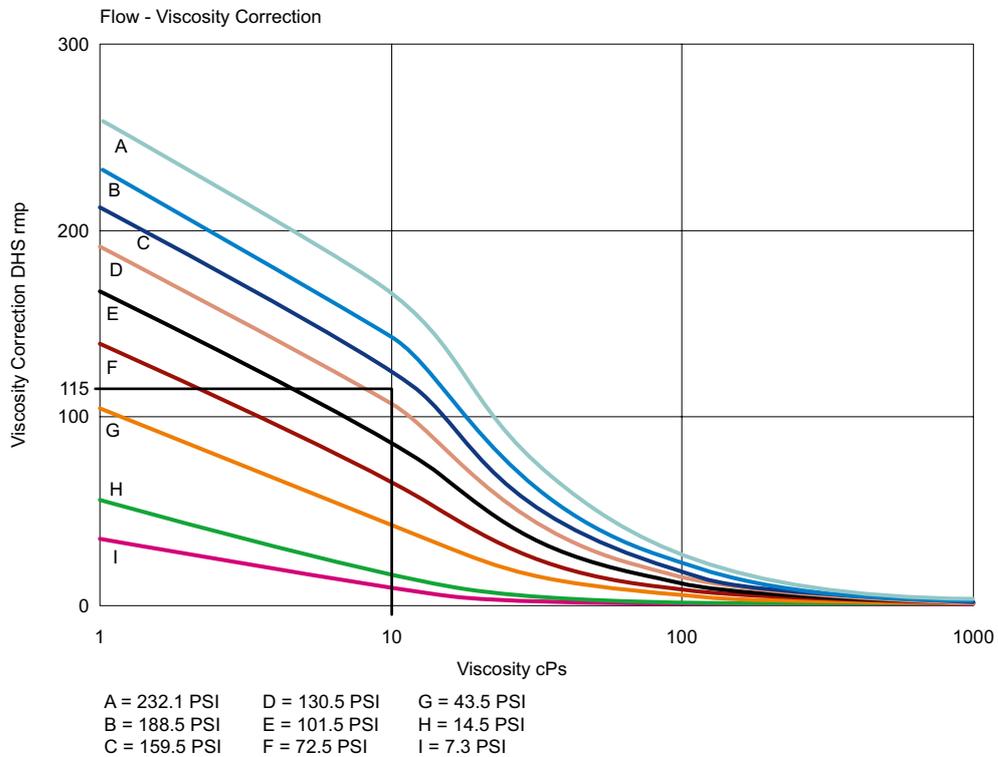
Fig. 7.6.3h

Step 2 – to calculate speed:

- Using the Flow at 1 cps curve (Fig. 7.6.3i), starting at the viscosity corrected DHS speed value of 115 rpm, draw a line parallel to the pressure lines

- At the desired flow rate of 60 US GPM, draw a line parallel to the x-axis, until it intersects the line drawn described above

- Then draw a line parallel to the y-axis down until it intersects the x-axis and read off the corresponding speed

- This speed (of 355 rpm) is the resulting duty speed

Flow at 1 cPs

Curves are representative. Specific pumps may vary in perdoemance due to manufacture, pumped fluids

A = 0 PSI	D = 43.5 PSI	H = 130.5 PSI	K = 232.1 PSI
B = 7.3 PSI	F = 72.5 PSI	I = 159.5 PSI	
C = 1.5 PSI	G = 101.5 PSI	J = 188.5 PSI	
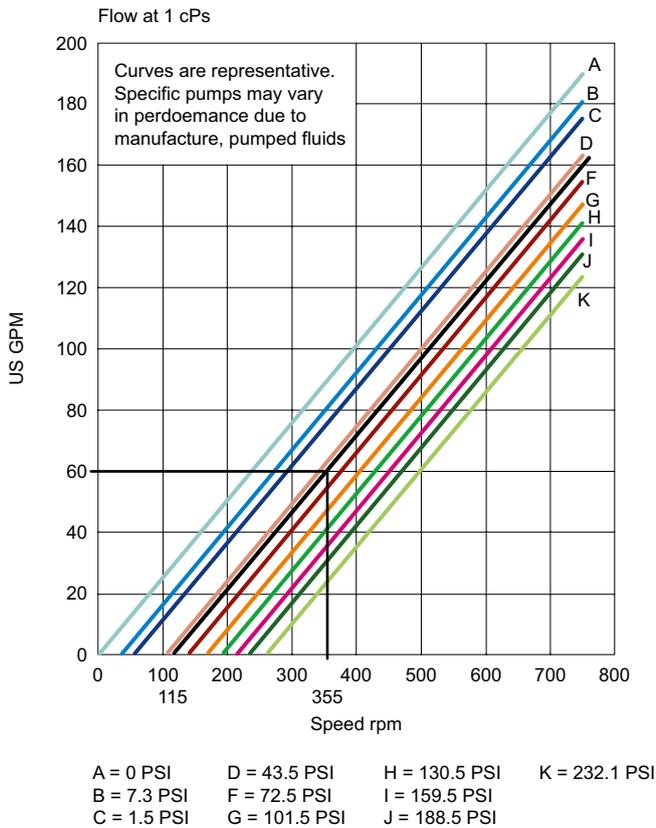
Fig. 7.6.3i

Step 3 – to calculate power:

- Using the Power at 1 cPs curve (Fig. 7.6.3j), starting at the duty speed of 355 rpm, draw a line parallel to the y-axis, until it intersects the duty pressure line (130 PSI)

- Draw a line parallel to the x-axis, until it insects the y-axis and read off the power value

- This value is the power absorbed requirement at 1 cPs (8 hp)

Power at 1 cPs

$$\text{Viscosity Corrected Power} = (PV \times rmp)/10{,}000 + \text{Power (1 cPs)}$$

A = 232.1 PSI	D = 130.5 PSI	G = 43.5 PSI
B = 188.5 PSI	E = 101.5 PSI	H = 14.5 PSI
C = 159.5 PSI	F = 72.5 PSI	I = 7.3 PSI
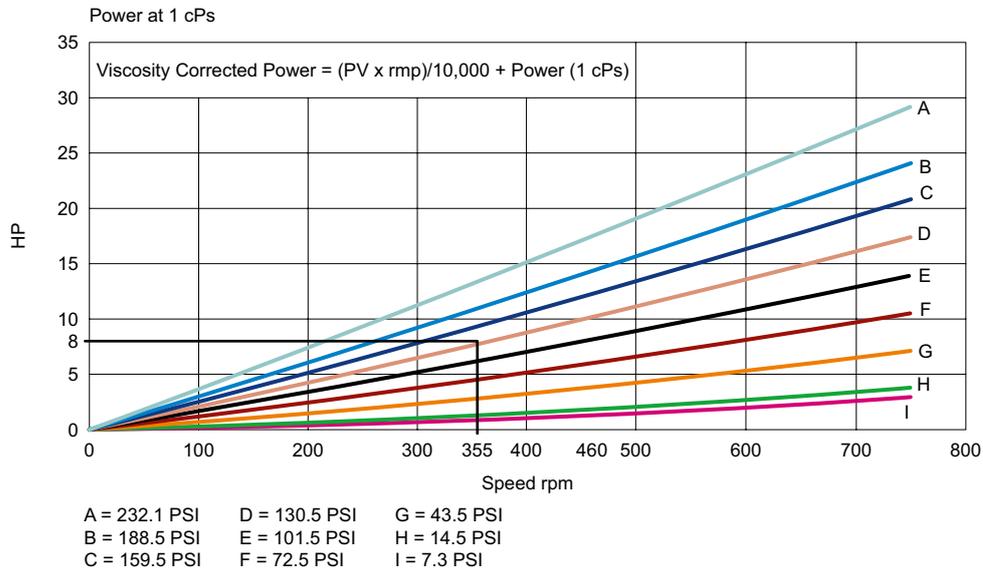
Fig. 7.6.3j

Step 4 – to calculate power:

- Using the Power Viscosity Correction curve
  (Fig. 7.6.3k), starting at the duty viscosity of 10 cPs,
  draw a line parallel to the y-axis, until it intersects
  the curve

- Draw a line parallel to the x-axis, until it intersects
  the y-axis and read off the PV value (3)

- Using the Viscosity Corrected Power formula
  shown, calculate total power absorbed require-
  ment:

  Total power requirement
  $$= \frac{(3 \times 355)}{10000} + 8.0 \qquad = 8.1 \text{ hp}$$

Power Viscosity Correction
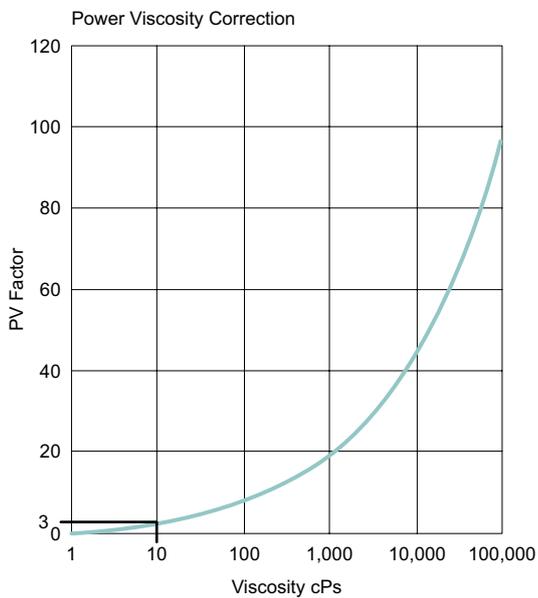
Fig. 7.6.3k

7.6.4 Pumps fitted with Bi-lobe Rotors (Stainless Steel)

These rotors, described in section 8.2.1, are available for SRU pumps and mainly used on high viscosity products containing solids where the pumps volumetric efficiency is high. When pumping such products optimum performance is obtained by using large slow running pumps.

Applications on water like viscosity fluids would result in slightly decreased efficiency over stainless steel Tri-lobe rotors. For this reason specific performance curves are available for Bi-lobe SS rotors where dead head speed is slightly higher than for Tri-lobes. Due to pump head clearances described in 8.2.2, different performance curves are used for the various temperature ratings of rotors i.e., 70° C (158° F), 130° C (266° F) and 200° C (392° F). The use of performance curves is as described in section 7.6.3.

NPSHr is slightly reduced when using Bi-lobe rotors.

7.6.5 Pumps fitted with Bi-lobe Rotors (Non Galling Alloy)

These rotors, described in section 8.2.1, have very small clearances resulting in increased volumetric efficiency over non Galling rotors when used on fluids with viscosities up to 50 cP. Pump sizing is achieved by referring to published performance curves or a pump selection program. Due to pump head clearances described in 8.2.2, different performance curves are used for the various temperature ratings of rotors i.e., 70° C (158° F), 130° C (266° F) and 200° C (392° F). The use of performance curves is as described in section 7.6.3.

7.6.6 Pumps with Electropolished Surface Finish

Pump performance will be affected by electropolish surface finish to the pump head internals. For sizing purposes a percentage increase on the 'dead head speed' (see tables below) should be applied to the performance curve for stainless steel Tri-lobe rotors and interpolated accordingly.

Pump Model	Percentage Increase Required on Stainless Steel Tri-lobe Rotor Dead Head Speed	
SRU range	Electropolishing only	Mechanical and Electropolishing
1/005	17.0	60.0
1/008	15.1	55.0
2/013	10.8	45.8
2/018	8.5	38.0
3/027	6.7	32.7
3/038	5.5	28.5
4/055	4.6	24.8
4/079	3.8	21.0
5/116	2.9	18.0
5/168	2.4	15.5
6/260	2.0	12.8
6/353	1.7	11.4

Table 7.6.6a

Pump Model	Percentage Increase Required on Multi-lobe Rotor Dead Head Speed	
SX range	Electropolishing only	Mechanical and Electropolishing
1/005	12.0	60.0
1/007	9.3	47.6
2/013	8.3	40.9
2/018	7.7	38.4
3/027	6.9	34.0
3/035	6.2	31.3
4/046	5.6	28.6
4/063	5.0	25.5
5/082	4.5	22.8
5/116	4.0	19.3
6/140	3.5	17.0
6/190	2.9	14.0
7/250	2.2	11.3
7/380	1.3	6.8

Table 7.6.6b

7.6.7 Guidelines for Solids Handling

The following criteria should be considered when deciding the pump's ability to handle large solids in suspension.

		Optimum Conditions
Solids form	-	Spherical
Solids physical properties i.e., hardness, resilience, shear, strength	-	Soft, yet possess resilience and shear strength
Solids surface finish	-	Smooth
Fluid/solids proportion	-	Proportion of solids to fluid is small
Relationship of fluid/solid SG	-	Equal
Flow velocity (pump speed)	-	Maintained such that solids in suspension are not damaged
Rotor form	-	Bi-lobe (If SRU)
Port size	-	Large as possible

Tables below show the maximum spherical solids size that can be satisfactory handled without product degradation, providing the optimum conditions are met. For non-optimum conditions these should be referred to Alfa Laval.

OptiLobe Model	Tri-lobe Rotors		SRU Model	Bi-lobe Rotors		Tri-lobe Rotors	
	mm	in		mm	in	mm	in
12	6	0.24"	SRU1/005	8	0.31"	6	0.24"
13	6	0.24"	SRU1/008	8	0.31"	6	0.24"
22	8	0.31"	SRU2/013	8	0.31"	6	0.24"
23	8	0.31"	SRU2/018	13	0.51"	9	0.34"
32	10	0.39"	SRU3/027	13	0.51"	9	0.34"
33	10	0.39"	SRU3/038	16	0.63"	11	0.44"
42	12	0.47"	SRU4/055	16	0.63"	11	0.44"
43	12	0.47"	SRU4/079	22	0.88"	15	0.59"
52	16	0.63"	SRU5/116	22	0.88"	15	0.59"
53	16	0.63"	SRU5/168	27	1.06"	18	0.72"
			SRU6/260	27	1.06"	18	0.72"
			SRU6/353	37	1.47"	24	0.94"

SX Model	Multi-Lobe Rotors	
	mm	in
SX1/005	7	0.28"
SX1/007	7	0.28"
SX2/013	10	0.39"
SX2/018	10	0.39"
SX3/027	13	0.51"
SX3/035	13	0.51"
SX4/046	16	0.63"
SX4/063	16	0.63"
SX5/082	19	0.75"
SX5/115	19	0.75"
SX6/140	25	0.98"
SX6/190	25	0.98"
SX7/250	28	1.1"
SX7/380	28	1.1"

Table 7.6.7a

Model	Max. Solids	
	mm	in
DuraCirc 32	8	0.3"
DuraCirc 33	8	0.3"
DuraCirc 34	13	0.5"
DuraCirc 42	13	0.5"
DuraCirc 43	13	0.5"
DuraCirc 52	17	0.7"
DuraCirc 53	21	0.8"
DuraCirc 54	25	1.0"
DuraCirc 62	25	1.0"
DuraCirc 63	34	1.3"
DuraCirc 72	34	1.3"
DuraCirc 73	51	2.0"
DuraCirc 74	51	2.0"

Table 7.6.7b Solid handling DuraCirc pump

Maximum Solids Handling mm (inch)											
OS1x			OS2x			OS3x			OS4x		
	mm	in		mm	in		mm	in		mm	in
OS12	6	0.24"	OS22	12	0.47"	OS32	16	0.63"	OS42	21	0.82
OS14	11	0.43"	OS24	16	0.63"	OS34	21	0.82"	OS44	29	1.14
OS16	17	0.67"	OS26	24	0.94"	OS36	32	1.26"	OS46	43	1.69
			OS27	15	0.59"	OS37	20	0.78"			
			OS28	32	1.26"	OS38	42	1.65"			

Table 7.6.7C Solid handling OS Twin Screw Pump

The OS Twin Screw pump can handle up to 43 mm (1.69") solids. The higher the screw pitch (represented by the last number in the model number), the larger the solid size. So an OS 36 can handle larger solids than the OS 34. Max. solid size should be entered in the selection program in order to get the right size pump.

7.6.8 Guidelines for Pumping Shear Sensitive Media

Special attention needs to be made to pumping shear sensitive media such as yeast and yoghurt where the cell structure needs to remain intact. Excess pump speed can irreversibly damage the cell structure. Therefore pump speeds need to be kept relatively low, in the range of 100 to 400 rev/min dependent upon the fluid being pumped, technology type, pump size/model and rotor form. For these types of applications refer to Alfa Laval.

7.7 Worked Examples – Positive Displacement Pump Sizing

Metric units

The following examples show two different positive displacement pumps to be sized for a typical sugar process and one pump to be sized for juice concentrate.

Pump 1
A low viscosity example handling sugar syrup

Pump 2
A high viscosity example handling massecuite x

Pump 3
A double duty example handling juice concentrate and CIP

As described in 7.1 in order to correctly size any type of pump, information is required such as Product/Fluid data, Performance data and Site Services data.

Pump 1 – Thin Sugar Syrup pump
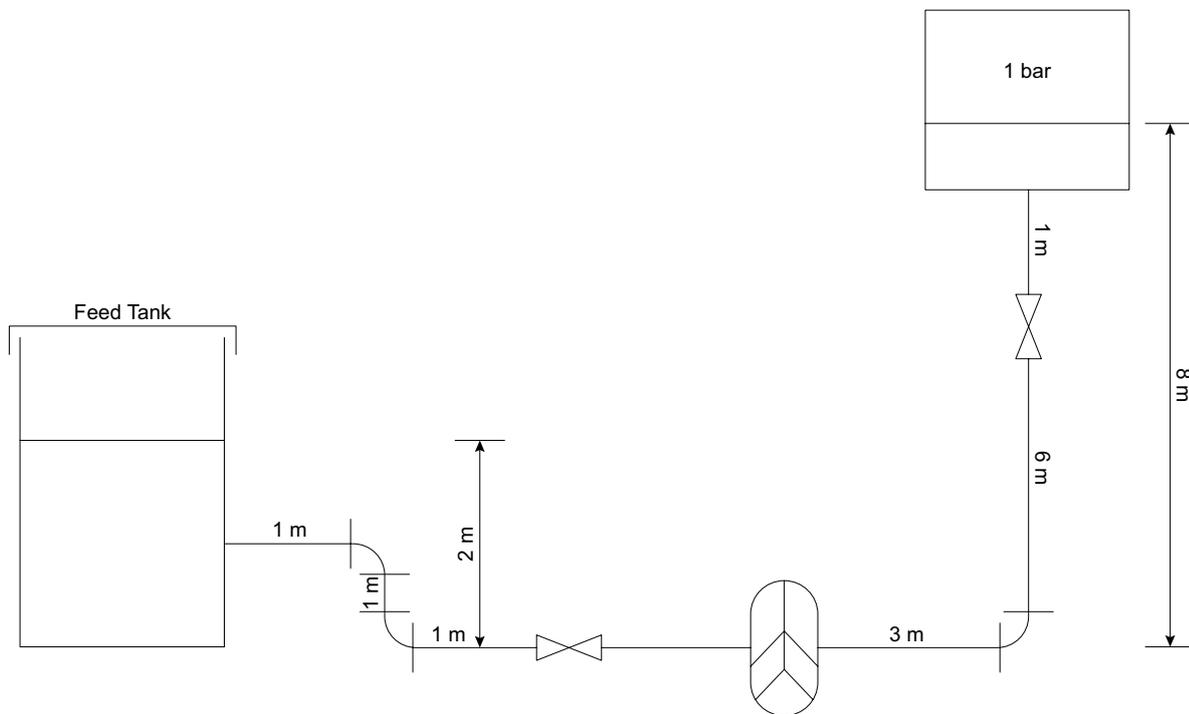
Fig. 7.7a Pump 1 – example

Product/Fluid data:

Fluid to be pumped	-	Sugar Syrup
Viscosity in pump	-	80 cP
SG	-	1.29
Pumping temperature	-	15° C
CIP temperature	-	95° C

All the data has been given by the customer.

Performance data:

Capacity	-	9 m³/h
Discharge	-	via 10 m of 51 mm dia. tube, plus 1 bend 90° and 1 butterfly valve. Static Head in Vessel = 8 m. Pressure in Vessel = 1 bar
Suction	-	via 3 m of 51 mm dia. tube, plus 2 bends 90° and 1 non-return valve. Static Head in Tank = 2 m

Site Services data:

Electrical supply	-	400v, 50 Hz
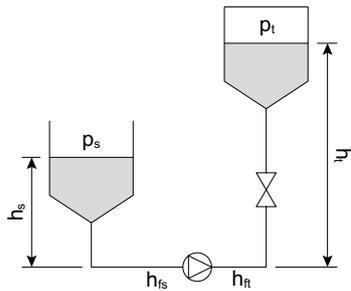
Fig. 7.7b Typical suction / Discharge Head set-up

Before sizing a pump, it will be necessary to determine the total head and NPSHa (Fig. 7.7b). The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge head H_t = $h_t + h_{ft} + p_t$

Where:

h_t = Static Head in Pressurised Vessel
h_{ft} = Total Pressure Drop in Discharge Line
p_t = Pressure in Vessel

Therefore:

h_t = 8 m x (SG = 1.29) = 10.3 m
h_{ft} = Pressure Drop in Tube Δp_{tube}
        + Pressure Drop in Bends and Valves Δp
        (calculated below)
p_t = 1 bar / (SG = 1.29) x 10 = 12.9 m

To ascertain h_{ft} the flow characteristic and equivalent line length must be determined as follows:

Flow Characteristic

Reynolds number Re	=	$\dfrac{D \times V \times p}{\mu}$	Where:

D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
p = Density (kg/m^3)
μ = Absolute Viscosity (cP)

Velocity V	=	$\dfrac{Q \times 353.6}{D^2}$	Where:

Q = Capacity (m^3/h)
D = Tube Diameter (mm)

$$= \frac{9 \times 353.6}{51^2}$$

$$= 1.22 \text{ m/s}$$

Density p   =   1290 derived from SG value 1.29 (see section 2.1.5)

Therefore Re   = $\dfrac{D \times V \times p}{\mu}$

$$= \frac{51 \times 1.22 \times 1290}{80}$$

$$= 1003$$

As Re is less than 2300, flow will be laminar so our calculations can continue for laminar flow.

Equivalent Line Length – Discharge Side

The equivalent lengths of straight tube for bends and valves are taken from table 14.7.1a. Since flow is laminar, the viscosity correction factor is 1.0 (see section 2.2.2).

Straight Tube Length = 3 + 6 + 1 = 10 m
1 bend 90° = 1 x 1 x 1.0 (corr. factor) = 1 m
1 butterfly valve = 1 x 1 x 1.0 (corr. factor) = 1 m
Total equivalent length = 12 m

Also as flow is laminar the friction factor f_D

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

$$= \frac{64}{1003}$$

$$= 0.064$$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{5 \times SG \times f_D \times L \times V^2}{D} \text{ (bar)}$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})
f_D = Friction Factor
L = Tube Length (m)
D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
SG = Specific Gravity

$$= \frac{5 \times 1.29 \times 0.064 \times 12 \times 1.22^2}{51} \text{ (bar)}$$

$$= 0.14 \text{ bar} = 1.4 \text{ m}$$

$H_t = h_t + h_{ft} + p_t = 10.3 + 1.4 + 12.9 \text{ m} = 24.6 \text{ m (2.46 bar)}$

Total Suction Head $H_s = h_s - h_{fs} + p_s$

Where:

h_s = Static Suction Head in Tank
h_{fs} = Total Pressure Drop in Suction Line
p_s = Pressure in Tank (open tank)

For this example:

h_s = 2 m x (SG = 1.29) = 2.6 m
h_{fs} = Calculated below
p_s = 0 (open tank)

Equivalent Line Length – Suction Side

The equivalent lengths of straight tube for bends and valves are taken from table 14.7.1. Since flow is laminar, the viscosity correction factor is 1.0 (see section 2.2.2).

Straight Tube Length = 1 + 1 + 1		= 3 m
2 bends 90°	= 2 x 1 x 1 (corr. factor)	= 2 m
1 non-return valve	= 1 x 12 x 1 (corr. factor)	= 12 m
	Total equivalent length	= 17 m

Also as flow is laminar the friction factor f_D

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

$$= \frac{64}{1003}$$

$$= 0.064$$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{5 \times SG \times f_D \times L \times V^2}{D} \text{ (bar)}$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})
f_D = Friction Factor
L = Tube Length (m)
D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
SG = Specific Gravity

$$= \frac{5 \times 1.29 \times 0.064 \times 1.7 \times 1.22^2}{51} \text{ (bar)}$$

$$= 0.2 \text{ bar} = 2 \text{ m}$$

$H_s = h_s + h_{fs} + p_s = 2.6 - 2 + 0 \text{ m} = 0.6 \text{ m (0.06 bar)}$

Total Head H $= H_t - H_s = 24.6 – 0.6 = 24 \text{ m} \longrightarrow \Delta p \text{ 24 m (2.4 bar)}$

NPSHa

NPSHa $\qquad = Pa + h_s - h_{fs} - Pvp$ Where:

Pa = Pressure Absolute above Fluid Level in Tank
h_s = Static Suction Head in Tank
h_{fs} = Total Pressure Drop in Suction Line
Pvp = Vapour Pressure of Fluid

Therefore:

Pa = 1 bar (open tank) (/1.29 x 10) = 7.75 m
h_s = 2.6 m
h_{fs} = Calculated to be 2 m
Pvp = At temperature of 15° C this is taken as being
    negligible i.e., 0 bar a = 0 m

NPSHa = Pa + h_s - h_{fs} - Pvp = 7.75 + 2.6 – 2 – 0 m = 8.35 m

Actual pump sizing can be made using pump performance curves or a pump selection program. The performance curve selection procedure is more specifically described in section 7.6.3.

From the initial suction line sizing curve (see section 14.9), a pump with a size 40 mm inlet connection would be required. As the duty is below 8 bar, and no special seal or options are needed – the Optilobe would be the first pump to check. As the sugar syrup can be quite abrasive, the pump should not run much faster than 450 rpm. Using a sizing program this gives the below OptiLobe:

Pump Model	-	OptiLobe 33
Connection size	-	50 mm
Speed	-	417 rev/min
NPSHr	-	2.1 m
Absorbed power	-	1.1 kW

Cavitation check

NPSHa should be greater than NPSHr i.e.,
8.35 m > 2.1 m.

Power calculation

The power requirement is mentioned in AnyTime but it is also possible to manually calculate as per below.

Total Required Power (kW) =
$$\frac{Pv \times Pump\ Speed\ (rev/min)}{10000} + Power\ at\ 1\ cSt\ (kW)$$

Where: Pv = Power/viscosity Factor

From example

- At speed 417 rev/min and total head 2.4 bar, the power at 1 cSt is 0.9 kW

- At viscosity 80 cP (62 cSt) the Pv factor is 3

Total Required Power (kW) =
$$\frac{Pv \times Pump\ speed\ (rev/min)}{10000} + Power\ at\ 1\ cSt\ (kW)$$

$$= \frac{3 \times 417}{10000} + 0.9$$

= 1.03 kW

It should be noted that this is the power needed at the pump shaft, and the appropriate motor power must be selected, which in this instance would be 1.5 kW being the nearest motor output power above the required power.

Since the viscosity is relatively low an alternative to this could be the DuraCirc pump, which is efficient at lower viscosity. Using the same data as above this can be selected either in a selection program or by means of the curves. Using a selection program the following pump is selected

Pump Model	-	DuraCirc 52
Connection size	-	50 mm
Speed	-	401 rev/min
NPSHr	-	0.7 m
Absorbed power	-	0.9 kW

The absorbed power is very close to that of the Optilobe pump and as the Optilobe will be the less expensive technology it would be best to go with this.

The recommended type of shaft seal based upon Alfa Laval application experience and guidelines would be a single flushed mechanical seal with silicon carbide / silicon carbide faces and EPDM or FPM elastomers.

- Hard silicon carbide seal faces due to the abrasive nature of sugar syrup

- Flushed version to prevent the sugar syrup from crystallising within the seal area

- EPDM or FPM elastomers for compatibility of both sugar syrup and CIP media

Pump 2 – Massecuite pump
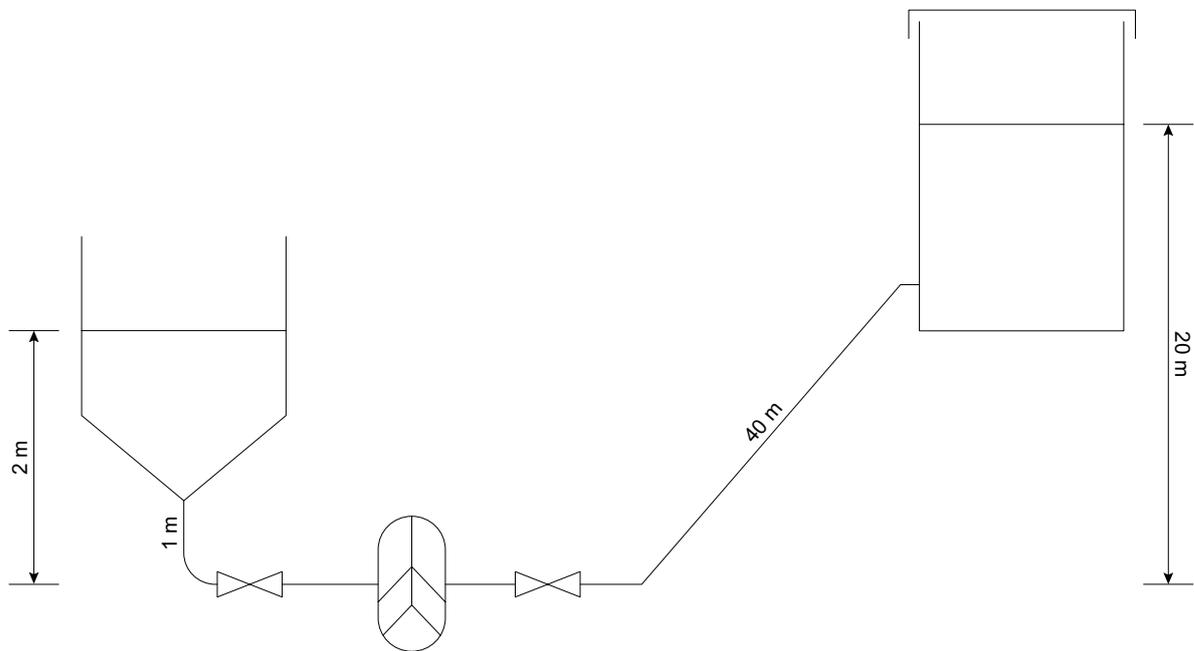
Fig. 7.7c Pump 2 - example

Product/Fluid data:

Fluid to be pumped	-	Massecuite
Viscosity in pump	-	25,000 cP
SG	-	1.35
Pumping temperature	-	65° C

All the data has been given by the customer.

Performance data:

Capacity	-	10 m³/h
Discharge	-	via 40 m of 76 mm dia. tube, plus 2 bends 45° and 1 butterfly valve. Static head in tank = 20 m
Suction	-	via 1 m of 101.6 mm dia. tube, plus 1 bend 90° and 1 butterfly valve. Static head in tank = 2 m

Site Services data:

Electrical supply	-	400v, 50 Hz

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Fig. 7.7d

Total head

Total Discharge head H_t = $h_t + h_{ft} + p_t$

Where:

h_t =   Static Head in Pressurised Vessel
h_{ft} =   Total Pressure Drop in Discharge Line
p_t =   Pressure in Vessel

Therefore:

h_t =   20 m x (SG = 1.35) = 27 m
h_{ft} =   Pressure Drop in Tube Δp_{tube}
        + Pressure Drop in Bends and Valves Δp
        (calculated below)
p_t =   0 bar (open tank) = 0 m

To ascertain h_{ft} the flow characteristic and equivalent line length must be determined as follows:

Flow Characteristic

Reynolds number Re   = $\dfrac{D \times V \times p}{\mu}$ Where:

D   =   Tube Diameter (mm)
V   =   Fluid Velocity (m/s)
p   =   Density (kg/m^3)
μ =   Absolute Viscosity (cP)

Velocity V   = $\dfrac{Q \times 353.6}{D^2}$ Where:

Q   =   Capacity (m^3/h)
D   =   Tube Diameter (mm)

= $\dfrac{10 \times 353.6}{76^2}$

=   0.61 m/s

Density p   =   1350 derived from SG value 1.35 (see section 2.1.5)

Therefore Re   = $\dfrac{D \times V \times p}{\mu}$

= $\dfrac{76 \times 0.61 \times 1350}{25,000}$

=   2.5

As Re is less than 2300, flow will be laminar.

Equivalent Line Length – Discharge Side

The equivalent lengths of straight tube for bends and valves are taken from table 14.7.1a. Since flow is laminar, the viscosity correction factor is 0.25 (see section 2.2.2).

Straight Tube Length		= 40 m
2 bends 45°	= 2 x 1 x 0.25 (corr. factor)	= 0.5 m
1 butterfly valve	= 1 x 2 x 0.25 (corr. factor)	= 0.5 m
	Total equivalent length	= 41 m

Also as flow is laminar the friction factor f_D

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

$$= \frac{64}{2.5}$$

$$= 25.6$$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{5 \times SG \times f_D \times L \times V^2}{D} \text{ (bar)}$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})
f_D = Friction Factor
L = Tube Length (m)
D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
SG = Specific Gravity

$$= \frac{5 \times 1.35 \times 25.6 \times 41 \times 0.61^2}{76} \text{ (bar)}$$

$$= 34.7 \text{ bar} = 347 \text{ m}$$

$H_t = h_t + h_{ft} + p_t = = 27 + 347 + 0 \text{ m} = 374 \text{ m} (37.4 \text{ bar})$

Total Suction Head $H_s = h_s - h_{fs} + p_s$

Where:

h_s = Static Suction Head in Tank
h_{fs} = Total Pressure Drop in Suction Line
p_s = Pressure in Tank (open tank)

Therefore:

h_s = 2 m x (SG = 1.35) = 2.7 m
h_{fs} = Calculated on the next page
p_s = 0 (open tank)

To ascertain h_{fs} the flow characteristic and equivalent line length must be determined as follows:

Flow Characteristic

Reynolds number Re	=	$\dfrac{D \times V \times p}{\mu}$	Where:

D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
p = Density (kg/m³)
μ = Absolute Viscosity (cP)

Velocity V	=	$\dfrac{Q \times 353.6}{D^2}$	Where:

Q = Capacity (m³/h)
D = Tube Diameter (mm)

$$= \frac{9 \times 353.6}{101.6^2}$$

= 0.34 m/s

Density p = 1350 derived from SG value 1.35 (see 2.1.5)

Therefore Re = $\dfrac{D \times V \times p}{\mu}$

$$= \frac{101.6 \times 0.34 \times 1350}{25000}$$

= 1.9

As Re is less than 2300, flow will be laminar.

Equivalent Line Length – Suction Side

The equivalent lengths of straight tube for bends and valves are taken from table 14.7.1a. Since flow is laminar, the viscosity correction factor is 0.25 (see section 2.2.2).

Straight Tube Length		= 1 m
1 bend 90°	= 1 x 2 x 0.25 (corr. factor)	= 0.5 m
1 butterfly valve	= 1 x 2 x 0.25 (corr. factor)	= 0.5 m
	Total equivalent length	= 2 m

Also as flow is laminar the friction factor f_D

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

$$= \frac{64}{1.9}$$

$$= 33.68$$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{5 \times SG \times f_D \times L \times V^2}{D} \text{ (bar)}$$

Where:

Pf = Pressure Loss due to Friction (h_{fs})
f_D = Friction Factor
L = Tube Length (m)
D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
SG = Specific Gravity

$$= \frac{5 \times 1.35 \times 33.68 \times 2 \times 0.34^2}{101.6} \text{ (bar)}$$

$$= 0.52 \text{ bar} = 5.2 \text{ m}$$

$H_s = h_s + h_{fs} + p_s = 2.7 - 5.2 + 0 \text{ m} = -2.5 \text{ m}$

Total Head $\quad H = H_t - H_s = 374 - (-2.5) = 376.5 \text{ m} \longrightarrow \Delta p \ 377 \text{ m (37.7 bar)}$

Because of the high total head the only pump which would be able to handle this would be the DuraCirc. Through the selection program the below is found.

Pump Model	-	DuraCirc 72
Connection size	-	100 mm
Speed	-	87 rev/min
NPSHr	-	0.7 m
Absorbed power	-	12.2 kW

It could however be an idea to consider reducing the head so a smaller pump can be suitably sized, consideration could be given to any or a combination of the following parameters:

• Reduce capacity

• Increase tube diameter

• Increase pumping temperature to reduce viscosity

Assuming the capacity is a definite requirement and the pumping temperature cannot be increased the customer could increase the discharge tube diameter i.e., from 76 mm to 101.6 mm.

The total head calculations are reworked, and for this particular example the fluid velocity (V) and friction factor (f_D) have already been established for 101.6 mm diameter tube. Also note, by referring to the equivalent tube length table 14.7.1a the values for bends 45° and butterfly valves remain unchanged.

Using the Miller equation to determine friction loss as follows:

$$Pf = \frac{5 \times SG \times f_D \times L \times V^2}{D} \text{ (bar)}$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})
f_D = Friction Factor
L = Tube Length (m)
D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
SG = Specific Gravity

$$= \frac{5 \times 1.35 \times 33.68 \times 41 \times 0.34^2}{101.6} \text{ (bar)}$$

$$= 10.6 \text{ bar} = 106 \text{ m}$$

Now $H_t = h_t + h_{ft} + p_t = 27 + 106 + 0 \text{ m} = 133 \text{ m (13.3 bar)}$

Now Total Head $\quad H = H_t - H_s = 133 - (-2.5) = 135.5 \text{ m} \rightarrow \Delta p\ 136 \text{ m (13.6 bar)}$

NPSHa

$$NPSHa = Pa + h_s - h_{fs} - Pvp$$

Where:

Pa = Pressure Absolute above Fluid Level in Tank
h_s = Static Suction Head in Tank
h_{fs} = Total Pressure Drop in Suction Line
Pvp = Vapour Pressure of Fluid

For this example:

Pa = 1 bar (open tank) = (1/1.35 x 10) = 7.4 m
h_s = 2.7 m
h_{fs} = Calculated to be 5.2 m
Pvp = At temperature of 65° C this is taken
        as being negligible i.e., 0 bar a = 0 m

$$NPSHa = Pa + h_s - h_{fs} - Pvp = 7.4 + 2.7 - 5.2 - 0 \text{ m} = 4.9 \text{ m}$$

With the new head an SRU pump or a smaller DuraCirc circumferential piston pump could be an option and using a pump selection program using stainless steel Tri-lobe rotors with 130° C rotor clearances would be as follows:

Pump Model	-	SRU5/168/LD
Connection size	-	100 mm (enlarged port)
Speed	-	100 rev/min
NPSHr	-	2.3 m
Absorbed power	-	5.1 kW

Pump Model	-	DuraCirc 62
Connection size	-	80 mm
Speed	-	116 rev/min
NPSHr	-	1.1 m
Absorbed power	-	5.2 kW

Note that by increasing the pipe size the energy consumption is reduced from 12.2 to 5.2 kW.

Cavitation check

NPSHa should be greater than NPSHr i.e.,
4.9 m > 2.3 m/1.1 m

Viscosity/Port Size check

The viscosity of 25000 cP at speed 100 rev/min is well within the pump's maximum rated figures.

It should be noted that this is the power needed at the pump shaft, and for a fixed speed drive the appropriate motor power must be selected, which in this instance would be 5.5 kW being the nearest motor output power above the required power.

As the SRU will be the least expensive technology and has the 100 mm inlet port option this would be best to go with.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a single flushed seal with SiC/SiC seal faces and FPM or EPDM elastomers.

It is important to notice that in the above we have had information about the in-pump viscosity. There can be a large difference between the viscosity at rest and the in-pump viscosity. In our selection system we have information about the typical in-pump viscosity for a variety of products. If in doubt about a product, it could be worth getting it tested in order to get the correct viscosity.

Alternative Pump Sizing Guide Using Volumetric Efficiency Calculation

Referring to the initial suction line sizing curve shown in 14.9, for the flow rate required of 10 m³/h with viscosity 25000 cP (18519 cSt), a pump having a 100 mm dia. inlet port would be selected.

For this example a Model SRU5/168 pump will be selected having 100 mm dia. enlarged ports. If a sanitary port is a definite requirement the Model SRU6/260 pump would be selected.

To calculate pump speed for the SRU5/168 pump selected the following formula is used as a general guide with volumetric efficiency of 99% (see section 7.2.4).

Pump speed (rev/min) $n \quad = \dfrac{Q \times 100}{q \times \eta_v \times 60}$

Where:

Q = Capacity (m³/h)
q = Pump Displacement (m³/100 rev)
η_v = Volumetric Efficiency (99% = 0.99)

$= \dfrac{10 \times 100}{0.168 \times 0.99 \times 60}$

$= 100$ rev/min

Pump 3 – Fruit Juice Concentrate and CIP

Product/Fluid data:

Fluid to be pumped	-	Fruit Juice Concentrate
Viscosity in pump	-	200 – 1500 cP
SG	-	1.1
Pumping temperature	-	30° C

Performance data:

Flow	-	28 m^3/h
Discharge	-	Through pipes/bends/valves calculated to 4 bar
Suction	-	Pump placed right by the tank so friction losses to be considered as 0 bar Static height in tank min. 0.5 m
p_t	-	0 bar (open tank) = 0 m

All the data has been given by the customer.

Also the pump should run CIP

Fluid to be pumped	-	CIP
Viscosity in pump	-	1 cP
Pumping temperature	-	80° C

Performance data:

Flow	-	90 m^3/h
Discharge	-	Through pipes/bends/valves calculated to 2.5 bar
Suction	-	Pump placed right by the tank so friction losses to be considered as 0 bar Static height in tank min. 0.5 m

Site Services data:

Electrical supply	-	400v, 50 Hz

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.
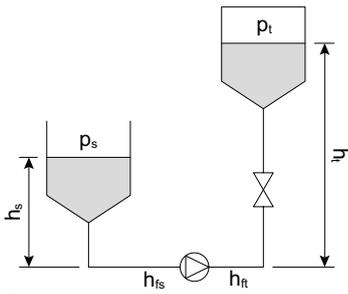
Total Discharge Head $H_t = h_t + h_{ft} + p_t$

Where:

h_t = Static Head in Pressurised Vessel
h_{ft} = Total Pressure Drop in Discharge Line
p_t = Pressure in Vessel

As H_t has already been informed from the customer at 4 bar we will not go further into the discharge pressure calculation or Reynold number calculation.

Total Suction Head $H_s = h_s - h_{fs} + p_s$

Where:

h_s = Static Suction Head in Tank
h_{fs} = Total Pressure Drop in Suction Line
p_s = Pressure in Tank (open tank)

Therefore:

h_s = 0.5 m x (SG = 1.1) = 0.55 m
h_{fs} = Considered to be 0
p_s = 0 (open tank)

$H_s = 0.55 - 0 + 0 = 0.55$ m

Total Head $H = H_t - H_s$

H product = (4 x 10 /1.1) – 0.55 = 35.81 m (3.94 bar)
H CIP = (2.5 x 10 / 1) – 0.5 = 24.5 m (2.45 bar)

NPSHa

NPSHa = Pa + h_s - h_{fs} - Pvp

Where:

Pa = Pressure Absolute above fluid level in Tank
h_s = Static Suction Head in Tank
h_{fs} = Total Pressure Drop in Suction Line
Pvp = Vapour Pressure of Fluid

Therefore:

Pa = 1 bar (open tank)
h_s = 0.5 m
h_{fs} = Assumed to be 0
Pvp = At temperature of 80^0C from table 14.4 = 4.75

NPSHa CIP = 1 x 10 + 0.5 - 0 - 4.75 = 5.75 m

(Calculated for CIP as this will be lowest value due to higher temperature)

Entering the data into the selection system
we have two options:

Pump Model	-	DuraCirc 73 Hi-Flow
Connection size	-	150 mm
Speed	-	163 / 561 rev/min
NPSHr	-	0.6 / 1.1 m
Absorbed power	-	4.2 / 10.6 kW - 15 kW motor

Pump Model	-	OS37
Connection size	-	100 mm
Speed	-	766 / 2619 rev/min (recommended speed max. 900 rpm for juice concentrate in a Twin Screw pump)
NPSHr	-	1.9 / 4.3 m
Absorbed power	-	5.5 / 11.37 kW - 15 kW motor

With a NPSHa of 5.75 both pumps can be used with NPSHr of 1.1 / 4.3 m.

Because the OS Twin Screw pump can run faster (in this case 2619 rpm for the CIP), a smaller size pump than for the DuraCirc can be chosen. In this case the smaller Twin Screw pump will be the least expensive pump to buy. Motor size will be the same and the DuraCirc will only be slightly more efficient than the Twin Screw.

Should the selection criteria be to have the most energy efficient pump an option could be to use a rotary lobe pump / circumferential piston pump for the product and the centrifugal pump for the CIP. In that case an Optilobe 53 with a 5.5 kW motor together with a LKH-45 with an 11 kW motor (Only running during CIP). This would save cost but also mean that there are two pumps to install, do service on etc.

For Fruit Juice Concentrate a single flushed seal with SiC/SiC seal faces would be recommended as this can be quite sticky with a high sugar content. For Twin Screw pumps we will generally recommend a flushed seal in case the pump is running when changing between products, so any dry running of the seal is avoided.

7.8 Worked Examples – Positive Displacement Pump Sizing

US units

The following examples show two different positive displacement pumps to be sized for a typical sugar process and one pump to be sized for juice concentrate.

Pump 1
A low viscosity example handling sugar syrup

Pump 2
A high viscosity example handling massecuite x

Pump 3
A double duty example handling juice concentrate and CIP

As described in 7.1 in order to correctly size any type of pump, information is required such as Product/Fluid data, Performance data and Site Services data.

As described in section 7.1 in order to correctly size any type of pump, information is required such as Product/Fluid data, Performance data and Site Services data.

Pump 1 – Thin Sugar Syrup pump

Feed Tank

3 ft

3 ft

6 ft

3 ft

10 ft

15 PSI

3 ft

26 ft

20 ft
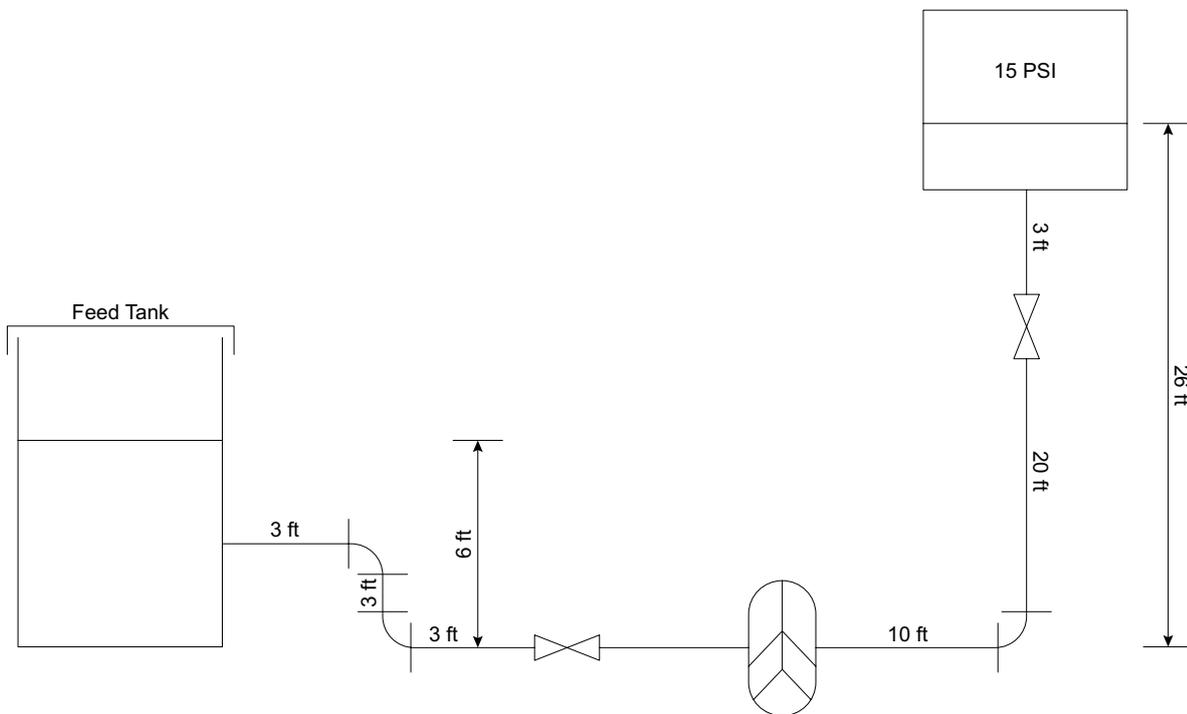
Fig. 7.8a Pump 1 – example

Product/Fluid data:

Fluid to be pumped	-	Sugar Syrup
Viscosity in pump	-	62 cSt (80 cP)
SG	-	1.29
Pumping temperature	-	59° F
CIP temperature	-	203° F

All the data has been given by the customer.

Performance data:

Capacity	-	40 US gal/min
Discharge	-	via 33 ft of 2 in dia. tube, plus 1 bend 90° and 1 butterfly valve. Static Head in Vessel = 26 ft. Pressure in Vessel = 15 PSI
Suction	-	via 9 ft of 2 in dia. tube, plus 2 bends 90° and 1 non-return valve Static Head in Tank = 6 ft

Site Services data:

Electrical supply	-	460v, 60 Hz
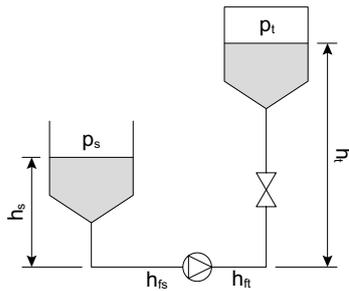
Fig. 7.8b Typical suction / Discharge Head set-up

Before sizing a pump, it will be necessary to determine the total head and NPSHa (Fig. 7.8b). The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

Total head

Total Discharge head H_t = $h_t + h_{ft} + p_t$

Where:

h_t = Static Head in Pressurised Vessel
h_{ft} = Total Pressure Drop in Discharge Line
p_t = Pressure in Vessel

For this example:

h_t = 26 ft x (SG = 1.29) = 33.5 ft
h_{ft} = Pressure Drop in Tube Δp_{tube}
        + Pressure Drop in Bends and Valves Δp
        (calculated below)
p_t = 15 PSI (x 2.31/1.29) = 26.86 ft

To ascertain h_{ft} the flow characteristic and equivalent line length must be determined as follows:

Flow Characteristic

Reynolds number Re   = $\dfrac{3162 \times Q}{D \times \nu}$

Where:

D = Tube Diameter (in)
Q = Capacity (US gal/min)
ν = Kinematic Viscosity (cSt)

$= \dfrac{3162 \times 40}{2 \times 62}$

= 1020

As Re is less than 2300, flow will be laminar.

Equivalent Line Length – Discharge Side

The equivalent lengths of straight tube for bends and valves are taken from table 14.7.2a. Since flow is laminar, the viscosity correction factor is 1.0 (see section 2.2.2).

Straight Tube Length = 10 + 20 + 3   = 33 ft

1 bend 90°   = 1 x 3 x 1.0 (corr. factor)   = 3 ft

1 butterfly valve   = 1 x 3 x 1.0 (corr. factor)   = 3 ft

Total equivalent length   = 39 ft

Also as flow is laminar the friction factor f_D

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

$$= \frac{64}{1020}$$

$$= 0.063$$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{0.0823 \times SG \times f_D \times L \times V^2}{D} \; (PSI)$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})

f_D = Friction Factor

L   = Tube Length (ft)

D   = Tube Diameter (in)

V   = Fluid Velocity (ft/s)

SG = Specific Gravity

$$\text{Velocity } V = \frac{Q \times 0.409}{D^2}$$

Where:

Q   = Capacity (US gal/min)

D   = Tube Diameter (in)

$$= \frac{40 \times 0.409}{2^2}$$

$$= 4.1 \text{ ft/s}$$

$$Pf = \frac{0.0823 \times 1.29 \times 0.063 \times 39 \times 4.1^2}{2} \; (PSI)$$

$$= 2.2 \text{ PSI} = 5 \text{ ft}$$

$H_t = h_t + h_{ft} + p_t = 33.5 + 5 + 26.86 \text{ ft} = 65.36 \text{ ft} \rightarrow \Delta p = 65 \text{ ft (28.17 PSI)}$

Total Suction Head H_s = $h_s - h_{fs} + p_s$

Where:

h_s =   Static suction head in Tank
h_{fs} =   Total pressure drop in suction line
p_s =   Pressure in Tank (open tank)

For this example:

h_s =   6 ft x (SG = 1.29) = 7.7 ft
h_{fs} =   calculated below
p_s =   0 (open tank)

Equivalent Line Length – Suction Side

The equivalent lengths of straight tube for bends
and valves are taken from table 14.7.2a. Since flow
is laminar, the viscosity correction factor is 1.0
(see section 2.2.2).

Straight Tube Length = 3 + 3 + 3   = 9 ft
2 bends 90°   = 2 x 3 x 1 (corr. factor)   = 6 ft
1 non-return valve   = 1 x 39 x 1 (corr. factor) = 39 ft
                              Total equivalent length   = 54 ft

Also as flow is laminar the friction factor f_D $= \dfrac{64}{Re}$

$= \dfrac{64}{1020}$

$= 0.063$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{0.0823 \times SG \times f_D \times L \times V^2}{D} \text{ (PSI)}$$

Where:

Pf =   Pressure Loss due to Friction (h_{ft})
f_D =   Friction Factor
L =   Tube Length (ft)
D =   Tube Diameter (in)
V =   Fluid Velocity (ft/s)
SG =   Specific Gravity

Velocity V $\quad = \dfrac{Q \times 0.409}{D^2}$ 　　　　Where:

$\quad$ Q $\quad=\quad$ Capacity (US gal/min)
$\quad$ D $\quad=\quad$ Tube Diameter (in)

$\qquad\qquad = \dfrac{40 \times 0.409}{2^2}$

$\qquad\qquad = 4.1$ ft/s

Pf $\quad = \dfrac{0.0823 \times 1.29 \times 0.063 \times 54 \times 4.12}{2}$ (PSI)

$\qquad = 3$ PSI $= 7$ ft

$H_s = h_s + h_{fs} + p_s = 7.7 - 7 + 0$ ft $= 0.7$ ft (0.3 PSI)

Total head $H = H_t - H_s = 65 - 0.7 = 64.3$ ft $\longrightarrow \Delta p = 64$ ft (27.74 PSI)

NPSHa

NPSHa $\quad = \quad Pa + h_s - h_{fs} - Pvp$ 　　　　Where:

$\quad$ Pa $\quad=\quad$ Pressure Absolute above Fluid Level in Tank
$\quad$ h_s $\quad=\quad$ Static Suction Head in Tank
$\quad$ h_{fs} $\quad=\quad$ Total Pressure Drop in Suction Line
$\quad$ Pvp $=\quad$ Vapour Pressure of Fluid

Therefore:

$\quad$ Pa $\quad=\quad$ 14.7 PSI (open tank) = 26.32 ft
$\quad$ h_s $\quad=\quad$ 7.7 ft
$\quad$ h_{fs} $\quad=\quad$ Calculated to be 7 ft
$\quad$ Pvp $=\quad$ At temperature of 59° F this is taken as being negligible i.e., 0 psia = 0 ft

NPSHa $= Pa + h_s - h_{fs} - Pvp = 26.32 + 7.7 - 7 - 0$ ft $= 27.02$ ft

Actual pump sizing can be made using pump performance curves or a pump selection program. The performance curve selection procedure is more specifically described in section 7.6.3.

From the initial suction line sizing curve (see section 14.9), a pump with a size 1.5 in inlet connection would be required. As the duty is below 8 bar, and no special seals or other options are needed – the Optilobe would be the first pump to check. As the sugar syrup can be quite abrasive with the pump we would not run much faster than 450 rpm. Using a sizing program this gives the pump sized as follows:

Pump Model	-	OptiLobe 33
Connection size	-	2 in
Speed	-	412 rev/min
NPSHr	-	6.8 ft
Absorbed power	-	1.2 hp – 1.5 hp drive

Cavitation check

NPSHa should be greater than NPSHr i.e., 27.02 ft > 6.8 ft.

Viscosity/Port Size check

The viscosity of 62 cSt at speed 412 rev/min is well within the pump's maximum rated figures.

Power calculation

The power requirement is mentioned in AnyTime but it is also possible to manually calculate as per below.

Total Required Power (kW) =
$$\frac{\text{Pv x Pump speed (rev/min)}}{10000} + \text{Power at 1 cSt (kW)}$$

Where: Pv = Power/viscosity Factor.

From example

- At speed 412 rev/min and total head 28 PSI, the power at 1 cSt is 1.2 hp

- At viscosity 62 cSt the Pv factor is 3

Total Required Power (kW) =
$$\frac{\text{Pv x Pump speed (rev/min)}}{10000} + \text{Power at 1 cSt (kW)}$$

$$= \frac{3 \times 412}{10000} + 1.1$$

$$= 1.22 \text{ hp}$$

It should be noted that this is the power needed at the pump shaft, and the appropriate motor power must be selected, which in this instance would be 1.5 hp being the nearest motor output power above the required power.

Since the viscosity is relatively low an alternative to this could be the DuraCirc pump, which is effecient at lower viscosity. Using the same data as above this can be selected either in a selection program or by means of the curves. Using a selection program the following pump is selected.

Pump Model	-	DuraCirc 52 Hi-Life
Connection size	-	2 in
Speed	-	407 rev/min
NPSHr	-	2.1 ft
Absorbed power	-	1.1 hp

The absorbed power is very close to that of the Optilobe pump and as the OptiLobe will be the least expensive technology it would be best to go with this.

The recommended type of shaft seal based upon Alfa Laval application experience and guidelines would be a single flushed mechanical seal with silicon carbide/silicon carbide faces and EPDM or FP; elastomers.

- Hard silicon carbide seal faces due to the abrasive nature of sugar syrup

- Flushed version to prevent the sugar syrup from crystallising within the seal area

- EPDM or FPM elastomers for compatibility of both sugar syrup and IP media

Pump 2 – Massecuite pump
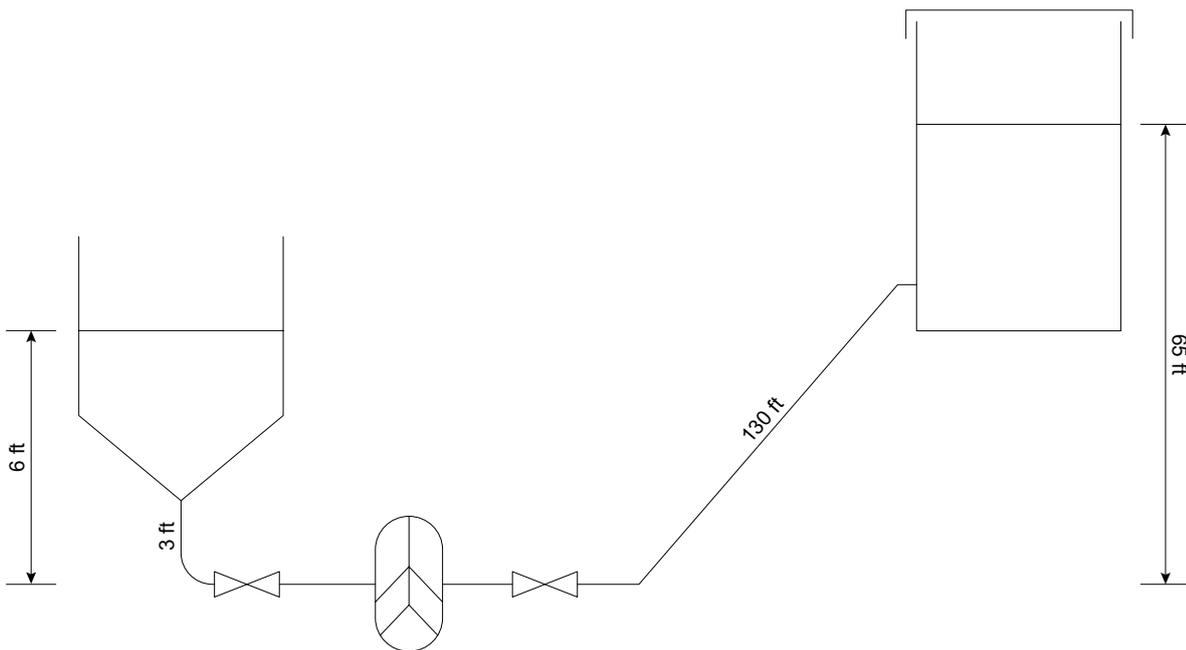
Fig. 7.8c Pump 2 – example

Product/Fluid data:

Fluid to be pumped	-	Massecuite
Viscosity in pump	-	18,519 cSt
SG	-	1.35
Pumping temperature	-	149° F

All the data has been given by the customer.

Performance data:

Capacity	-	44 US gal/min
Discharge	-	via 130 ft of 3 in dia. tube, plus 2 bends 45° and 1 butterfly valve Static head in tank = 65 ft
Suction	-	via 3 ft of 4 in dia. tube, plus 1 bend 90° and 1 butterfly valve. Static head in tank = 6 ft

Site Services data:

Electrical supply	-	460v, 60 Hz

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.

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Fig. 7.8d

Total head

Total Discharge head H_t = $h_t + h_{ft} + p_t$

Where:

h_t = Static Head in Pressurised Vessel
h_{ft} = Total Pressure Drop in Discharge Line
p_t = Pressure in Vessel

Therefore:

h_t = 65 ft x (SG = 1.35) = 88 ft
h_{ft} = Pressure Drop in Tube Δp_{tube}
$\quad\quad$ + Pressure Drop in Bends and Valves Δp
$\quad\quad$ (calculated below)
p_t = 0 PSI = 0 ft

To ascertain h_{ft} the flow characteristic and equivalent line length must be determined as follows:

Flow Characteristic

Reynolds number Re   = $\dfrac{3162 \times Q}{D \times \nu}$

Where:

D = Tube Diameter (in)
Q = Capacity (US gal/min)
ν = Kinematic Viscosity (cSt)

$= \dfrac{3162 \times 44}{3 \times 18519}$

$= 2.5$

As Re is less than 2300, flow will be laminar.

Equivalent Line Length – Discharge Side

The equivalent lengths of straight tube for bends and valves are taken from table 14.7.2a. Since flow is laminar, the viscosity correction factor is 0.25 (see section 2.2.2).

Straight Tube Length		= 130 ft
2 bends 45°	= 2 x 3 x 0.25 (corr. factor)	= 1.5 ft
1 butterfly valve	= 1 x 7 x 0.25 (corr. factor)	= 1.75 ft
	Total equivalent length	= 133 ft

Also as flow is laminar the friction factor f_D

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

$$= \frac{64}{2.5}$$

$$= 25.6$$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{5 \times SG \times f_D \times L \times V^2}{D} \ (PSI)$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})
f_D = Friction Factor
L = Tube Length (ft)
D = Tube Diameter (in)
V = Fluid Velocity (ft/s)
SG = Specific Gravity

$$Velocity\ V = \frac{Q \times 0.409}{D^2}$$

Where:

Q = Capacity (US gal/min)
D = Tube Diameter (in)

$$= \frac{40 \times 0.409}{3^2}$$

$$= 2\ ft/s$$

$$Pf = \frac{0.0823 \times 1.35 \times 25.6 \times 133 \times 2^2}{3} \ (PSI)$$

$$= 504\ PSI = 1163\ ft$$

$H_t = h_t + h_{ft} + p_t = 88 + 1163 + 0\ ft = 1251\ ft\ (542\ PSI)$

Total suction head $H_s = H_s - H_{fs} + p_s$

Where:

h_s = Static Suction Head in Tank
h_{fs} = Total Pressure Drop in Suction Line
Pvp = Pressure in Tank (open tank)

Therefore:

h_s = 6 ft x (SG = 1.35) = 8 ft
h_{fs} = Calculated below
P_s = 0 (open tank)

To ascertain h_{fs} the flow characteristic and equivalent line length must be determined as follows:

Flow Characteristic

Reynolds number Re $= \dfrac{3162 \times Q}{D \times \nu}$

Where:

D = Tube Diameter (in)
Q = Capacity (US gal/min)
ν = Kinematic Viscosity (cSt)

$= \dfrac{3162 \times 44}{4 \times 18519}$

$= 1.9$

As Re is less than 2300, flow will be laminar.

Equivalent Line Length – Suction Side

The equivalent lengths of straight tube for bends and valves are taken from table 14.7.2a. Since flow is laminar, the viscosity correction factor is 0.25 (see section 2.2.2).

Straight Tube Length		= 3 ft
1 bend 90°	= 1 x 7 x 0.25 (corr. factor)	= 1.75 ft
1 butterfly valve	= 1 x 7 x 0.25 (corr. factor)	= 1.75 ft
	Total equivalent length	= 6.5 ft

Also as flow is laminar the friction factor f_D

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

$$= \frac{64}{1.9}$$

$$= 33.68$$

The Miller equation is now used to determine friction loss as follows:

$$Pf = \frac{0.0823 \times SG \times f_D \times L \times V^2}{D} \; (PSI)$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})
f_D = Friction Factor
L = Tube Length (ft)
D = Tube Diameter (in)
V = Fluid Velocity (ft/s)
SG = Specific Gravity

$$\text{Velocity } V = \frac{Q \times 0.409}{D^2}$$

Where:

Q = Capacity (US gal/min)
D = Tube Diameter (in)

$$= \frac{44 \times 0.409}{4^2}$$

$$= 1.1 \; ft/s$$

$$Pf = \frac{0.0823 \times 1.35 \times 33.68 \times 6.5 \times 1.12}{4} \; (PSI)$$

$$= 7.4 \; PSI = 17 \; ft$$

$H_s = h_s + h_{fs} + p_s = 8 + 17 + 0 \; ft = -9 \; ft$

Total head $H = H_t – H_s = 65 - 0.7 = 64.3 \; ft \longrightarrow \Delta p = 64 \; ft \; (27.74 \; PSI)$

Because of the high total head the only pump which would be able to handle this would be the DuraCirc. Through the selection program the below is found.

Pump Model	-	DuraCirc 72 Hi-Life
Connection size	-	4 in
Speed	-	87 rev/min
NPSHr	-	2.1 ft
Absorbed power	-	16.4 hp

It could however be an idea to consider reducing the head so a smaller pump can be suitably sized, consideration could be given to any or a combination of the following parameters:

1. Reduce capacity
2. Increase tube diameter
3. Increase pumping temperature to reduce viscosity

Assuming the capacity is a definite requirement and the pumping temperature cannot be increased the customer could increase the discharge tube diameter i.e. from 3 inch to 4 inch.

The total head calculations are reworked, and for this particular example the fluid velocity (V) and friction factor (fD) have already been established for 4 in diameter tube. Also note, by referring to the equivalent tube length table 14.7.2a the values for bends 45^0 and butterfly valves remain unchanged.

Using the Miller equation to determine friction loss as follows:

$$Pf = \frac{0.00823 \times SG \times f_D \times L \times V^2}{D} \text{ (PSI)}$$

Where:

Pf = Pressure Loss due to Friction (h_{ft})
f_D = Friction Factor
L = Tube Length (m)
D = Tube Diameter (mm)
V = Fluid Velocity (m/s)
SG = Specific Gravity

$$= \frac{0.00823 \times 1.35 \times 33.68 \times 133 \times 1.12}{4} \text{ (PSI)}$$

$$= 150 \text{ PSI} = 346 \text{ ft}$$

Now $H_t = h_t + h_{ft} + p_t = 88 + 346 + 0 \text{ ft} = 434 \text{ ft (188 PSI)}$

Now Total Head $H = H_t - H_s = 434 - (- 9) = 443 \text{ ft (192 PSI)}$

NPSHa

NPSHa $= Pa + h_s - h_{fs} - Pvp$

Where:

Pa $=$ Pressure Absolute above fluid level in Tank
h_s $=$ Static Suction Head in Tank
h_{fs} $=$ Total Pressure Drop in Suction Line
Pvp $=$ Vapour Pressure of Fluid

For this example:

Pa $=$ 14.7 bar (open tank) = 25.15 ft (SG 1.35)
h_s $=$ 8 ft
h_{fs} $=$ Calculated to be 17 ft
Pvp $=$ At temperature of 149^0 F this is taken as being negligible i.e. 0 psia = 0 ft

NPSHa $= Pa + h_s - h_{fs} - Pvp = 25.15 + 8 - 17 - 0$ m = 16.15 ft

With the new head an SRU pump or a smaller DuraCirc circumferential piston pump could be an option and using a pump selection program using stainless steel Tri-lobe rotors with 130°C rotor clearances would be as follows:

Note that by increasing the pipe size the energy consumption is reduced from 16.4 to 6.8 hp.

Pump Model	-	SRU5/168/LD
Connection size	-	4 in (enlarged port)
Speed	-	100 rev/min
NPSHr	-	7.5 ft
Absorbed power	-	6.5 hp - 7.5 hp drive

Pump Model	-	DuraCirc 63 Hi-Life
Connection size	-	4 in
Speed	-	84 rev/min
NPSHr	-	2.1 ft
Absorbed power	-	6.8 hp - 7.5 hp drive

Cavitation check

NPSHa should be greater than NPSHr i.e., 16.15 ft > 7.5 ft / 2.1 ft.

Viscosity/Port Size check

The viscosity of 18519 cSt at speed 100 rev/min is well within the pump's maximum rated figures.

It should be noted that this is the power needed at the pump shaft, and for a fixed speed drive the appropriate motor power must be selected, which in this instance would be 7.5 hp being the nearest motor output power above the required power.

As the SRU will be the least expensive technology this would be best to go with in this case.

The recommended shaft seal type based upon Alfa Laval application experience and guidelines would be a single flushed seal with SiC/SiC seal faces and FPM or EPDM elastomers.

It is important to notice that in the above we have had information about the in-pump viscosity. There can be a large difference between the viscosity at rest and the in-pump viscosity. In our selection system we have information about the typical in-pump viscosity for a variety of products. If in doubt about a product, it could be worth getting it tested in order to get the correct viscosity.

Alternative Pump Sizing Guide Using Volumetric Efficiency Calculation

Referring to the initial suction line sizing curve shown in section 14.9, for the flow rate required of 44 US gal/min with viscosity 18519 cSt, a pump having a 100 mm dia. inlet port would be selected.

For this example a Model SRU5/168 pump will be selected having 4 in dia. enlarged ports.

If a sanitary port is a definite requirement the Model SRU6/260 pump would be selected.

To calculate pump speed for the SRU5/168 pump selected the following formula is used as a general guide with volumetric efficiency of 99% (see section 7.2.4).

Pump speed (rev/min) $\quad n \quad = \dfrac{Q \times 100}{q \times \eta_v}$

Where:

Q = Capacity (US gal/min)
q = Pump Displacement (US gal/100 rev)
η_v = Volumetric Efficiency (99% = 0.99)

$$= \frac{44 \times 100}{44.39 \times 0.99}$$

$$= 100 \text{ rev/min}$$

Pump 3 – Fruit Juice Concentrate and CIP

Product/Fluid data:

Fluid to be pumped	-	Fruit Juice Concentrate
Viscosity in pump	-	200 – 1500 cP
SG	-	1.1
Pumping temperature	-	86° F

Performance data:

Flow	-	123 GPM
Discharge	-	Through pipes/bends/ valves calculated to 58 PSI
Suction	-	Pump placed right by the tank so friction losses to be considered as 0 PSI. Static height in tank min. 1.64 ft
p_t	-	0 PSI (open tank) = 0 ft

All the data has been given by the customer.

Also the pump should run CIP

Fluid data:

Fluid to be pumped	-	CIP
Viscosity in pump	-	1 cP in pump
Pumping temperature	-	176° F

Performance data:

Flow	-	396 m³/h
Discharge	-	Through pipes/bends/ valves calculated to 36.26 PSI
Suction	-	Pump placed right by the tank so friction losses to be considered as 0 PSI. Static height in tank min. 1.64 ft

Site Services data:

Electrical supply	-	460v, 60 Hz

Before sizing a pump, it will be necessary to determine the total head and NPSHa. The theory, including the different formulae regarding these parameters is more specifically described in section 2.2.2 and 2.2.4.
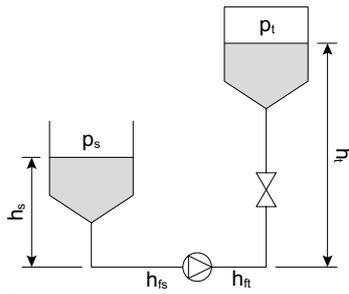
Total Discharge Head H_t = $h_t + h_{ft} + p_t$

Where:

h_t =   Static Head in Pressurised Vessel
h_{ft} =   Total Pressure Drop in Discharge Line
p_t =   Pressure in Vessel

As H_t has already been informed from the customer at 58 PSI / 36.26 PSI, calculations for discharge pressure and Reynold number will not be made.

Total Suction Head H_s = $h_s - h_{fs} + p_s$

Where:

h_s =   Static Suction Head in Tank
h_{fs} =   Total Pressure Drop in Suction Line
p_s =   Pressure in Tank (open tank)

Therefore:

h_s =   1.64 for CIP (for product x (SG = 1.1) = 1.8 ft)
h_{fs} =   Considered to be 0
p_s =   0 (open tank)

H_s = 1.8 – 0 + 0 = 1.8 ft

Total Head H = H_t - H_s

H product = (58 x 2.31 /1.1) – 1.8 = 120 ft (57 PSI)
H CIP   = (36.26 x 2.31) – 1.64 = 74.35 m (32.22 PSI)

NPSHa

$$NPSHa = Pa + h_s - h_{fs} - Pvp$$

Where:

Pa	=	Pressure Absolute above fluid level in Tank
h_s	=	Static Suction Head in Tank
h_{fs}	=	Total Pressure Drop in Suction Line
Pvp	=	Vapour Pressure of Fluid

Therefore:

Pa	=	14.5 PSI (open tank) = 33.5 ft
h_s	=	1.8 ft
h_{fs}	=	Assumed to be 0
Pvp	=	At temperature of 80^0 C / 176^0 F from table 14.4 = 47.5
kPa	=	15.9 ft

NPSHa CIP = 33.5 + 1.64 - 0 - 15.9 = 19.24 ft

(Calculated for CIP as this will be lowest value due to higher temperature)

Entering the data into the selection system
we have two options:

Pump Model	-	DuraCirc 73 Hi-Flow
Connection size	-	4 in / 3 in
Speed	-	163 / 559 rev/min
NPSHr	-	2 / 3.5 ft
Absorbed power	-	5.5 / 13.9 hp - 20 hp motor

Pump Model	-	OS37
Connection size	-	4 in
Speed	-	765 / 2619 rev/min (recommended speed max. 900 rpm for juice concentrate in a Twin Screw pump)
NPSHr	-	6.4 / 14.1 ft
Absorbed power	-	8.23 / 15.19 hp - 20 hp motor

With a NPSHr of 14.1 both pumps can be used with NPSHa of 19.24 ft.

Because the OS Twin Screw pump can run faster (in this case 2617 rpm for the CIP), a smaller size pump than for the DuraCirc can be chosen. In this case the smaller Twin Screw pump will be the least expensive pump to buy. Motor size will be the same and the DuraCirc will only be slightly more efficient than the Twin Screw.

Should the selection criteria be to have the most energy efficient pump an option could be to use a rotary lobe pump / circumferential piston pump for the product and the centrifugal pump for the CIP. In that case an Optilobe 53 with a 10 hp drive together with a LKH-45 with an 5 hp motor (Only running during CIP). This would save cost but also mean that there are two pumps to install, do service on etc.

For Fruit Juice Concentrate a single flushed seal with SiC/SiC seal faces would be recommended as it will have a high sugar content and can be very sticky.

This chapter gives descriptions of the various specification options available for the Alfa Laval pump ranges, such as port connections, heating/cooling jackets, pressure relief valves and other ancillaries.

8.0

Pump Specification Options

8.1 Centrifugal Pumps

8.1.1 Port Connections

Pumps are supplied with unions, clamp fittings and flanges to all main standards, i.e., SMS, DIN, ISO, ASME, BS, DS, bevel seat, DC and H-Line.

Pump Range	Pump Model	Nominal Connection Size			
		Inlet mm	Outlet mm	Inlet in	Outlet in
LKH	LKH 5	50	40	2	1.5
	LKH 10	65	50	2.5	2
	LKH 15	100	80	4	3
	LKH 20	65	50	2.5	2
	LKH 25	80	65	3	2.5
	LKH 35	65	50	2.5	2
	LKH 40	80	65	3	2.5
	LKH 45	100	80	4	3
	LKH 50	100	80	4	3
	LKH 60	100	100	4	4
	LKH 70	100	80	4	3
	LKH 75	100	100	4	4
	LKH 85	150	150	6.0	6.0
	LKH 90	150	150	6.0	6.0
LKH-Multistage	LKH 112	50	40	2.0	1.5
	LKH 113	50	40	2.0	1.5
	LKH 114	50	40	2.0	1.5
	LKH 122	80	65	3.0	2.5
	LKH 123	80	65	3.0	2.5
	LKH 124	80	65	3.0	2.5
LKHPF	LKHPF 10	65	50	2.5	2.0
	LKHPF 15	100	80	4.0	3.0
	LKHPF 20	65	50	2.5	2.0
	LKHPF 25	80	65	3.0	2.5
	LKHPF 35	65	50	2.5	2.0
	LKHPF 40	80	65	3.0	2.5
	LKHPF 45	100	80	4.0	3.0
	LKHPF 50	100	80	4.0	3.0
	LKHPF 60	100	100	4.0	4.0
	LKHPF 70	100	80	4.0	3.2

Pump Range	Pump Model	Nominal Connection Size			
		Inlet mm	Outlet mm	Inlet in	Outlet in
LKH Evap	LKH Evap 10	65	50	2.5	2.0
	LKH Evap 15	100	80	4.0	3.0
	LKH Evap 20	65	50	2.5	2.0
	LKH Evap 25	80	65	3.0	2.5
	LKH Evap 35	65	50	2.5	2.0
	LKH Evap 40	80	65	3.0	2.5
	LKH Evap 45	100	80	4.0	3.0
	LKH Evap 50	100	80	4.0	3.0
	LKH Evap 60	100	100	4.0	4.0
	LKH Evap 70	100	80	4.0	3.0
LKH UltraPure	LKH UltraPure 10	65	50	2.5	2.0
	LKH UltraPure 20	65	50	2.5	2.0
	LKH UltraPure 25	80	65	3.0	2.5
	LKH UltraPure 35	65	50	2.5	2.0
	LKH UltraPure 40	80	50	3.0	2.0
	LKH UltraPure 45	100	80	4.0	3.0
	LKH UltraPure 60	100	100	4.0	4.0
	LKH UltraPure 70	100	80	4.0	3.0
LKH Prime	LKH Prime 10	50	50	2.0	2.0
	LKH Prime 20	65	50	2.5	2.0
	LKH Prime 40	80	65	3.0	2.5
LKH Prime UltraPure	LKH Prime UltraPure 10	50	50	2.0	2.0
	LKH Prime UltraPure 20	65	50	2.5	2.0
SolidC	SolidC 1	50	40	2.0	1.5
	SolidC 2	65	40	2.5	1.5
	SolidC 3	80	40	3.0	1.5
	SolidC 4	80	50	3.0	2.0

Table 8.1.1a
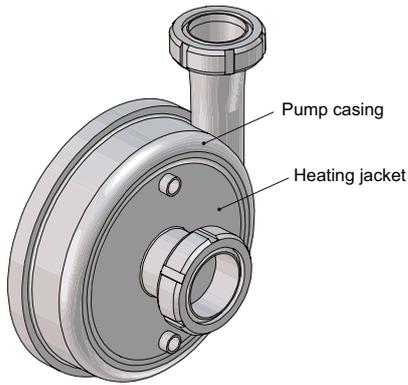
Fig. 8.1.2a Heating/Cooling jacket on the LKH pump
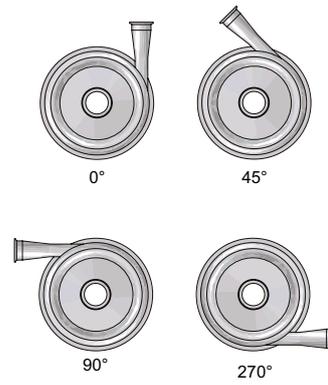
Fig. 8.1.3a Turned pump casing

8.1.2 Heated/Cooled Pump Casing

In some applications, heating of the fluid being pumped may be required to reduce the fluid viscosity so that satisfactory operation is achieved. Alternatively, it may be necessary to cool the fluid being pumped where heat is generated by means of the fluid repeatedly being passed through the pump. On such occasions most LKH pump models can be fitted with heating/cooling jacket (Fig. 8.1.2a).

8.1.3 Drainable Pump Casing

In applications where it is a requirement that no fluid should be left in the pump casing. This can be achieved by either turning the pump outlet downwards, fitting a drain connection or welding a valve to the bottom of the pump casing (Fig. 8.1.3a, 8.1.3b, 8.1.3c).
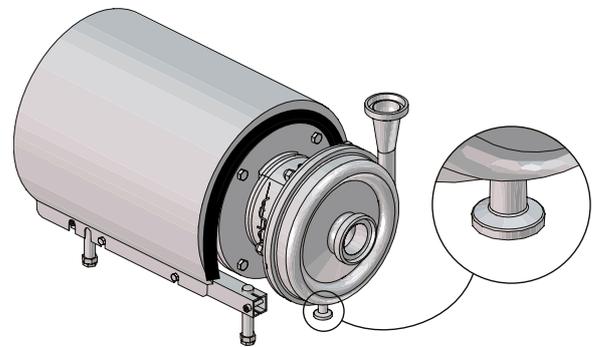
Fig. 8.1.3b Pump casing with drain connection
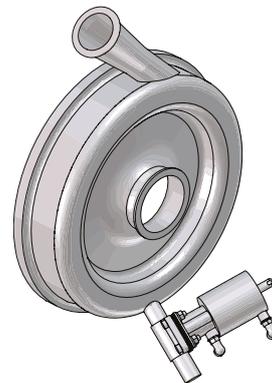
Fig. 8.1.3c Pump casing with Unique DVST valve
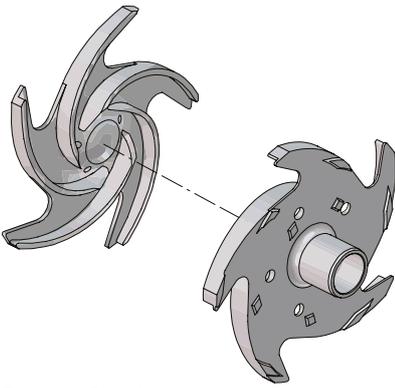
Fig. 8.1.4a Clear flow impeller

Fig. 8.1.4b LKH inducer

8.1.4 Clear Flow Impeller

In some applications, e.g. when using an LKH centrifugal pump as a booster pump in a cream pasteurisation unit, there is a risk that a hard layer of proteins will slowly build up between the backside of the impeller and the back plate. This will activate the thermal relay of the motor after a few hours of operation so that the pump stops.

The operating time of the pump can be increased by applying a clear flow impeller (Fig. 8.1.4a). The clear flow impeller is a special scraper impeller that solves the product build-up problem by increasing the standard gap width between the back of the impeller and the back plate. By introducing scrapers on the back of the impeller the hard layer is constantly removed.

When sizing pumps with clear flow impeller it must be taken into account that the head will be reduced by up to 10%. Furthermore, for this type of application, it is recommended to select a motor size with an output power one rating higher than the standard selection to avoid the motor thermal relay being constantly tripped.
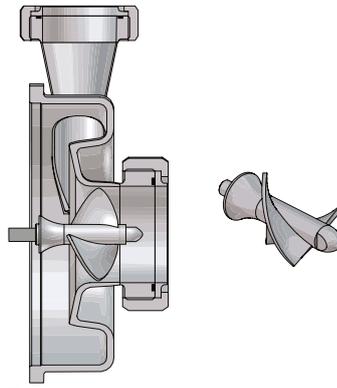
8.1.5 Inducer

In some applications, it may be necessary to improve suction conditions by means of fitting the pump with an inducer (Fig. 8.1.4b).

This has the effect of improving NPSHr for difficult applications and/or assisting the flow of a viscous fluid into the pump casing.

Alfa Laval's LKH pump ranges are acknowledged as having one of the best NPSHr characteristics on the market without the requirement of an inducer to the impeller; a more likely utilisation would be handling those higher viscosity applications where transition from inlet to impeller eye requires assistance.

8.1.6 Motor

Centrifugal pumps are generally available with 2 pole and 4 pole motors with synchronous speeds of 3000 and 1500 rev/min for 50 Hz and 3600 and 1800 rev/min for 60 Hz, respectively.

A stainless-steel protective shroud is standard with IEC motors. Electric motors are described in more detail in chapter 9.
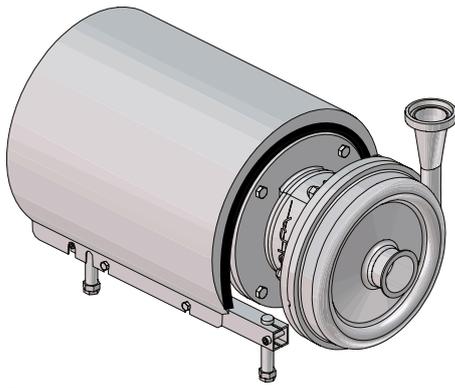
Fig. 8.1.7a Adjustable legs
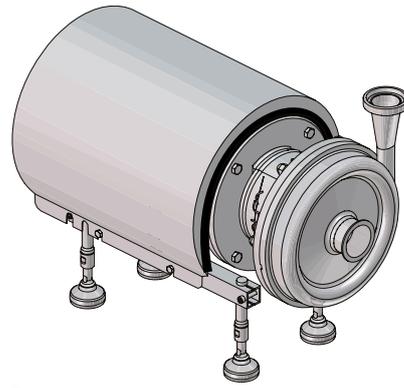
Fig. 8.1.7b Adjustable pads

8.1.7 Legs

As standard centrifugal pumps are supplied with adjustable feet and legs to enable easy installation and commissioning (Fig. 8.1.7a).

LKH pumps are also optionally available with adjustable pads with covered threads surfaces and sealing between all moveable parts of the legs and thereby reduce the cleaning effort to a minimum (Fig. 8.1.7b).

8.1.8 Other Centrifugal Pump Specification Options

There are also centrifugal pump options available to meet specific application demands related to internal surface finish (see chapter 5), elastomer material (see chapter 5) and shaft seal design (see chapter 6).

8.2 Positive Displacement Pumps

8.2.1 Rotor Form

Overview of Rotor form to positive displacement pumps:

	Pump Type	Rotor	Rotor Material
DuraCirc	Circumferential Piston	Bi-piston	Non galling alloy
OptiLobe	Rotary Lobe	Tri-lobe	Stainless steel
SRU	Rotary Lobe	Tri-lobe	Stainless steel
		Bi-lobe	Stainless steel
		Bi-lobe	Non galling alloy
SX	Rotary Lobe	Multi-lobe	Stainless steel
SX UltraPure	Rotary Lobe	Multi-lobe	Stainless steel
OS	Twin Screw	Screw	Stainless steel

Table 8.2.1a
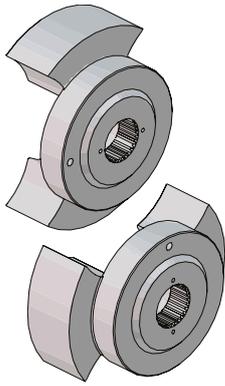
Fig. 8.2.1.1a DuraCirc Bi-piston Rotors

Model	Max. Solids	
	mm	in
DuraCirc 32	8	0.3"
DuraCirc 33	8	0.3"
DuraCirc 34	13	0.5"
DuraCirc 42	13	0.5"
DuraCirc 43	13	0.5"
DuraCirc 52	17	0.7"
DuraCirc 53	21	0.8"
DuraCirc 54	25	1.0"
DuraCirc 62	25	1.0"
DuraCirc 63	34	1.3"
DuraCirc 72	34	1.3"
DuraCirc 73	51	2.0"
DuraCirc 74	51	2.0"

Table. 8.2.1.1a The maximum spherical solids size that can be satisfactory handled without product degradation on DuraCirc circumferential piston pumps

8.2.1.1 Circumferential Piston Pumps

DuraCirc pumps have Bi-piston rotors manufactured from non-galling alloy to allow for very small clearances (see section 8.2.2), leading to very high efficiencies. High volumetric efficiency is particularly important in applications where there is a combination of low viscosity and high pressure. DuraCirc rotors are suitable for temperatures up to 150° C (302° F) and pressures up to 40 bar (580 PSIG) (Fig. 8.2.1.1a and Table 8.2.1.1a).

Fig. 8.2.1.2a Tri-lobe rotor

8.2.1.2 Rotary Lobe Pumps

Tri-lobe Rotors (Stainless steel)

Most duties can be accomplished by pumps fitted with stainless steel Tri-lobe rotors (Fig. 8.2.1.2a). The Tri-lobe rotor with its geometrically optimized profile and precision manufacture ensure interchangeability as well as smooth, high performance pumping action.

These are available on the SRU pump range with three temperature ratings:

• Up to 70° C (158° F)

• Up to 130° C (266° F)

• Up to 200° C (392° F)

And pressures up to 20 bar (290 PSIG)

On the OptiLobe range Tri-lobe rotors are suitable for temperatures up to 130° C (266° F) and pressures up to 8 bar (116 PSIG).
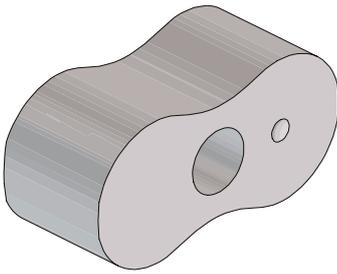
Fig.8.2.1.2b Bi-lobe rotor
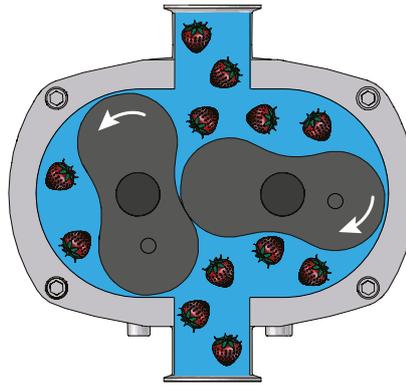
Fig.8.2.1.2c Bi-lobe rotors for solids handling

Bi-lobe Rotors (Stainless steel)

These are generally used for handling delicate suspended solids where minimum product damage is required (Fig. 8.2.1.2b - 8.2.1.2c). Typical applications are jam containing fruit pieces, sausage meat filling, petfood, soups and sauces containing solid matter.

Bi-lobe rotors in stainless steel are available on the SRU pump range with three temperature ratings:

• Up to 70° C (158° F)

• Up to 130° C (266° F)

• Up to 200° C (392° F)

And pressures up to 20 bar (290 PSIG)

SRU Model	Bi-lobe Rotors		Tri-lobe Rotors	
	mm	in	mm	in
SRU1/005	8	0.31"	6	0.24"
SRU1/008	8	0.31"	6	0.24"
SRU2/013	8	0.31"	6	0.24"
SRU2/018	13	0.51"	9	0.34"
SRU3/027	13	0.51"	9	0.34"
SRU3/038	16	0.63"	11	0.44"
SRU4/055	16	0.63"	11	0.44"
SRU4/079	22	0.88"	15	0.59"
SRU5/116	22	0.88"	15	0.59"
SRU5/168	27	1.06"	18	0.72"
SRU6/260	27	1.06"	18	0.72"
SRU6/353	37	1.47"	24	0.94"

Table. 8.2.1.2a The maximum spherical solids size that can be satisfactory handled without product degradation on SRU rotary lobe pumps

Bi-lobe Rotors (Non galling alloy)

Manufactured from non-galling alloy these rotors have an advantage over stainless steel, as smaller clearances (see section 8.2.2) can be used, leading to increased volumetric efficiency on lower viscosity applications.

These are available on the SRU pump range with 3 temperature ratings:

- Up to 70° C (158° F)

- Up to 130° C (266° F)

- Up to 200° C (392° F)

And pressures up to 20 bar (290 PSIG)
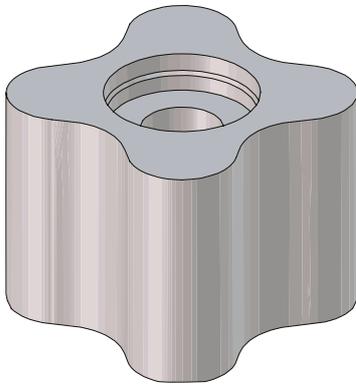
Fig.8.2.1.2d Multi-lobe rotor

Multi-lobe Rotors

This rotor is manufactured from stainless steel and as the name suggests has many lobes. For the SX and the SX UltraPure pump range these rotors have 4 lobes and are designed to maximise efficiency, reduce shear and provide a smooth pumping action (Fig. 8.2.1.2d and Table 8.2.1.2b). Rotors are suitable for temperatures up to 150° C (302° F) and pressures up to 15 bar (215 PSIG).

SX Model	Multi-lobe Rotors	
	mm	in
SX1/005	7	0.28"
SX1/007	7	0.28"
SX2/013	10	0.39"
SX2/018	10	0.39"
SX3/027	13	0.51"
SX3/035	13	0.51"
SX4/046	16	0.63"
SX4/063	16	0.63"
SX5/082	19	0.75"
SX5/115	19	0.75"
SX6/140	25	0.98"
SX6/190	25	0.98"
SX7/250	28	1.1"
SX7/380	28	1.1"

Table. 8.2.1.2b The maximum spherical solids size that can be satisfactory handled without product degradation on SX rotary lobe pumps
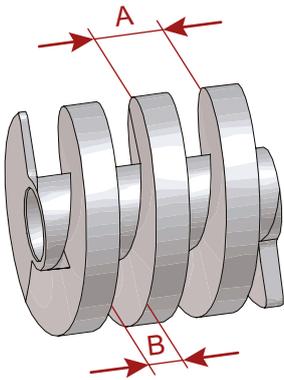
Fig. 8.2.1.3a Screw pitch (A) and chambers (B)

8.2.1.3 Twin Screw Pumps

OS Twin Screw Pumps have screws made from 316L stainless steel. Different screw profiles are available. Each screw has a specific pitch length which is basically the dimension from the rear face of one helix to the front of the next helix. A tighter pitch for the same length screw will have more closed chambers and as such less slip and greater pressure build up.

The chamber is the free distance between the rear of one helix and the front of the next helix in which product is moved, to this extent the chamber size for a given screw length determines the flow rate and the maximum particle size for solids handling (Fig. 8.2.1.3a.

Screws are suitable for pressures up to 16 bar. For continuous operation they are suitable for temperatures up to 100° C (212° F) and for intermittent operation up to 150° C (302° F).

Pitch	Chamber Size (mm) Defines max. solids size			
	OS1*	OS2*	OS3*	OS4*
*2	6	13	16.5	22.5
*4	11	17.5	23	31
*6	17	26	33.5	45.5
*7	-	15	20	-
*8	-	32	42	-

Chamber Size (in) Defines max. solids size			
OS1*	OS2*	OS3*	OS4*
0.24"	0.51"	0.65"	0.89"
0.43"	0.69"	0.9"	1.22"
0.67"	1.02"	1.32"	1.79"
-	0.59"	0.79"	-
-	1.26"	1.65"	-

Table. 8.2.1.3a The maximum spherical solids size that can be satisfactory handled without product degradation on OS twin screw pumps.
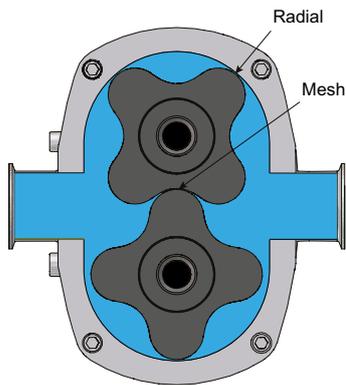
Fig. 8.2.2a Clearances in a rotary lobe pump

Clearances are necessary to avoid rotor to rotor, rotor to casing and rotor to casing cover contact (likewise for screws). The size of these clearances is related to the pressure and temperature of pump operation and rotor material.

8.2.2 Clearances

Within the pump head are clearances, which are the spaces between rotating components and between rotating and stationary components (Fig. 8.2.2a). The key clearances are as follows:

- Radial clearance
  (between rotor tip and casing)

- Mesh clearance
  (between rotors)

- Front clearance
  (between front of rotor and casing cover)

- Back clearance
  (between back of rotor and back face of casing)

- Root clearance
  (applicable to twin screw pumps; between the mesh of the screws)
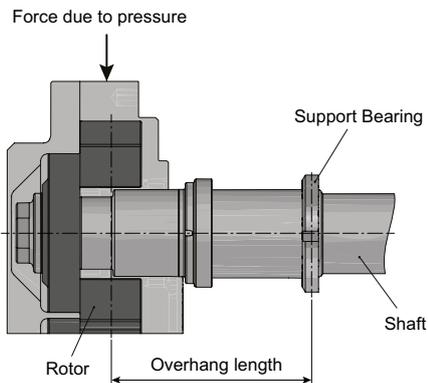
Fig. 8.2.2b Pressure effect on rotary lobe and circumferential piston pumps

Pressure effect

The design concept of the rotary positive displacement pump is to have no contacting parts in the pump head. This requires having the shaft support bearings mounted outside of the pump head, which results in an overhung load, caused by the rotors/screws fitted to the shafts (see Fig. 8.2.2b). The effect of pressure on the rotors will cause shaft deflection, which could result in contact between rotors, casing and casing cover. As product wetted parts of the rotary lobe and twin screw pump ranges are predominantly manufactured from stainless steel, any contact between rotating and stationary parts would cause 'galling' and possible pump seizure. To allow for this pressure effect, clearances are built into the pump head between surfaces that may contact. For the OptiLobe, SRU, SX, SX UltraPure and OS pump ranges there is only one pressure rating, which is the maximum differential pressure of the particular pump model. The pressure effect is less significant on pumps fitted with non-galling alloy rotors like the DuraCirc circumferential piston pump.

Temperature effect

Temperature change can be caused by the fluid being pumped, pump mechanism, drive unit and/or the environment. Any CIP operation required should also be taken into consideration (see chapter 10 for detailed explanation of CIP). Changes in temperature will cause expansion upon heating or contraction upon cooling, to the pump casing and gearcase components. The most significant result is movement between shaft and gearcase/pump casing causing the rotors to move forward/backward in the pump casing, thereby reducing the front clearance. To compensate for this, the SRU pump range has increased clearances as shown below. SRU pumps are designed for various temperature ratings for rotors i.e., 70° C (158° F), 130° C (266° F) or 200° C (392° F).

On other Alfa Laval rotary positive displacement pump ranges the design of the mechanical seal eliminates contact between the fluid being pumped and the shaft. This results in the shaft not being subjected to the full temperature variation and therefore only one temperature rating of 150° C (302° F) is necessary.

8.2.3 Port Connections

Alfa Laval rotary positive displacement pumps are supplied with connections to all major standards (please refer to Anytime for full listing of available port connection standard by pump range). All models have full bore through porting, conforming to International Sanitary Standards BS4825 / ISO2037. This provides effective CIP cleaning and maximises inlet and outlet port efficiency and NPSHr characteristics.

On SRU pump models there is an option of an enlarged port which can be chosen for high viscosity applications.

On DuraCirc pump models the Uni-Fit option is available whereby the pump is supplied with the same port to port and foot to port centre dimensions as the old Alfa Laval SCPP pump, enabling ease of replacement without design or pipework modification. On DuraCirc 34 and 42 the port size dimension is reduced when the Uni-Fit option is specified.

Pump Range	Pump Model	Nominal Connection Size			
		Standard		Optional	
		mm	in	mm	in
DuraCirc	DuraCirc 32	25	1"	-	-
	DuraCirc 33	40	1.5"	-	-
	DuraCirc 34	50	2"	40	1.5"
	DuraCirc 42	50	2"	40	1.5"
	DuraCirc 43	50	2"	-	-
	DuraCirc 52	50	2"	-	-
	DuraCirc 53	65	2.5"	-	-
	DuraCirc 54	80	3"	-	-
	DuraCirc 62	80	3"	-	-
	DuraCirc 63	100	4"	-	-
	DuraCirc 72	100	4"	-	-
	DuraCirc 73	150	6"	-	-
	DuraCirc 74	150	6"	-	-
OptiLobe	OptiLobe 12	25	1"	-	-
	OptiLobe 13	40	1.5"	-	-
	OptiLobe 22	40	1.5"	-	-
	OptiLobe 23	40	1.5"	-	-
	OptiLobe 32	50	2"	-	-
	OptiLobe 33	50	2"	-	-
	OptiLobe 42	65	2.5"	-	-
	OptiLobe 43	80	3"	-	-
	OptiLobe 52	80	3"	-	-
	OptiLobe 53	100	4"	-	-

Pump Range	Pump Model	Nominal Connection Size			
		Standard		Optional	
		mm	in	mm	in
SRU	SRU1/005	25	1"	-	-
	SRU1/008	25	1"	40	1.5"
	SRU2/013	25	1"	40	1.5"
	SRU2/018	40	1.5"	50	2"
	SRU3/027	40	1.5"	50	2"
	SRU3/038	50	2"	65	2.5"
	SRU4/055	50	2"	65	2.5"
	SRU4/079	65	2.5"	80	3"
	SRU5/116	65	2.5"	80	3"
	SRU5/168	80	3"	100	4"
	SRU6/260	100	4"	100	4"
	SRU6/353	100	4"	150	6"
SX	SX1/005	25	1"	-	-
	SX1/007	40	1.5"	-	-
	SX2/013	40	1.5"	-	-
	SX2/018	50	2"	-	-
	SX3/027	50	2"	-	-
	SX3/035	65	2.5"	-	-
	SX4/046	50	2"	-	-
	SX4/063	65	2.5"	-	-
	SX5/082	65	2.5"	-	-
	SX5/115	80	3"	-	-
	SX6/140	80	3"	-	-
	SX6/190	100	4"	-	-
	SX7/250	100	4"	-	-
	SX7/380	150	6"	-	-

Table 8.2.3b Port size on rotary lobe and circumferential piston pumps

Flanges for vertically ported circumferential piston and rotary lobe pumps are not fitted directly to the discharge port. In this instance an elbow bend is required to which the flange is fitted.

Due to the wide performance envelope with twin screw pumps there are multiple inlet/outlet combinations available. As with centrifugal pumps the basic recommendation is that the port size on the outlet should max. be the size of the inlet port.

Pump Range	Pump Model	Nominal Connection Size			
		Inlet mm	Outlet mm	Inlet in	Outlet in
OS	OS 12/14/16	40	25	1.5"	1"
		40	40	1.5"	1.5"
		50	25	2"	1"
		50	40	2"	1.5"
		50	50	2"	2"
		65	25	2.5"	1"
		65	40	2.5"	1.5"
		65	50	2.5"	2"
	OS 22/23/24	50	40	2"	1.5"
		50	50	2"	2"
		65	40	2.5"	1.5"
		65	50	2.5"	2"
		65	65	2.5"	2.5"
		80	40	3"	1.5"
		80	50	3"	2"
		80	65	3"	2.5"
	OS 32/34/36	80	65	3"	2.5"
		80	80	3"	3"
		100	65	4"	2.5"
		100	80	4"	3"
	OS 42/44/46	100	80	4"	3"
		100	100	4"	4"
		150	80	6"	3"
		150	100	6"	4"

Table 8.2.3c Port size options on twin screw pumps (conventional flow direction)

Fig. 8.2.4a DuraCirc circumferential piston pump with rectangular inlet

8.2.4 Rectangular Inlet

For handling extremely viscous products and/or large solids that would naturally bridge a smaller port, SRU rotary lobe pumps, DuraCirc circumferential piston pumps and OS twin screw pumps can be supplied with a rectangular inlet (Fig. 8.2.4a - 8.2.4b). Usually, the inlet port will be in vertical orientation to allow the product to flow into the pumping chamber under gravity from a hopper mounted directly above or mounted with an adaptor to facilitate connection to large diameter pipework.

As can be seen from the tables below for pumps with rectangular inlets, there is a percentage area increase when compared to a sanitary port connection. This increases the pump's ability to handle highly viscous products.

Pump Model	Sanitary Port Area (mm²)	Rectangular Inlet Area (mm²)	% Area Increase above Sanitary Port Diameter
SRU1/005	387	660	+71
SRU1/008	387	1260	+226
SRU2/013	387	1216	+214
SRU2/018	957	1976	+106
SRU3/027	957	2112	+121
SRU3/038	1780	3360	+89
SRU4/055	1780	2688	+51
SRU4/079	2856	4320	+51
SRU5/116	2856	5032	+76
SRU5/168	4185	8160	+95
SRU6/260	7482	13888	+86
SRU6/353	7482	18240	+144

Table 8.2.4a Port size impact on SRU rotary lobe pumps with rectangular inlet

Pump Size	Standard Port		Rectangular Port				
	Diameter mm	Area mm²	Length mm	Width mm	Radius mm	Area mm²	% Increase over Standard Port Area
DuraCirc 33	34.9	957	120	20	10	2314	+142
DuraCirc 42	47.6	1780	145	30	15	4157	+134
DuraCirc 53	60.3	2856	206	40	15	8047	+182
DuraCirc 54	73.0	4185	206	62	15	12579	+201
DuraCirc 62	73.0	4185	248	50	19	12090	+189
DuraCirc 63	97.6	7482	248	70	19	17050	+128
DuraCirc 72	97.6	7482	284	48	20	13289	+78
DuraCirc 73*	120.0	11310	284	70	20	19537	+73

 * DuraCirc 73 uses a 150 mm port with a 120 mm weld neck diameter

Table 8.2.4b Port size impact on DuraCric circumferential piston pumps with rectangular inlet
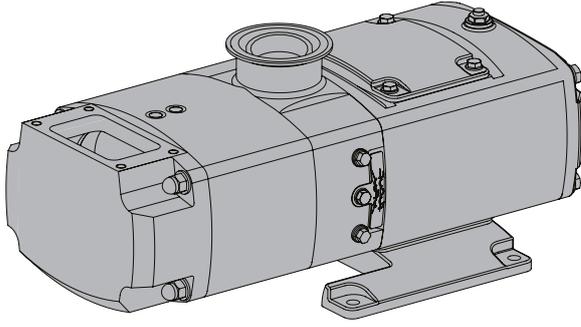
Fig. 8.2.4b OS Twin Screw pump with rectangular inlet

Pump Model	Default Port Size (cover)	Area of Default Port (mm²)	Dimensions Rectangle (mm)	Area of Rectangle (mm²)	% Area Increase
OS10	DN65	3421.19	65 x 48	3120	-8.8
OS20	DN80	5153.00	90 x 50	4500	-12.67
OS30	DN100	7853.98	110 x 65	7150	-8.96
OS40	DN150	17671.49	150 x 80	12000	-32.09

Table 8.2.4c Port size impact on OS twin screw pumps with rectangular inlet

8.2.5 Heated/Cooled Pump Casing

Rotary lobe, circumferential piston as well as twin screw pumps facilitate heating/cooling to improve handling of temperature sensitive products, maintain media viscosity and reduce risk of crystallization or solidification.

Solutions are designed to ensure a significant temperature impact on the pump casing. The exact solution depends on the specific technology in question, but as a general rule, heating/cooling devices should be in operation prior to pump start up and remain in operation for some time after pump shut down.

Typical applications include:

- Adhesive

- Chocolate

- Gelatine

- Jam

- Resin

	Operation prior to Pump Start Up	Operation after Pump Shut Down	Max. Temperature Heating Fluid	Max. Pressure Heating/Cooling Fluid
DuraCirc 30-70	15 minutes	15 minutes	150° C (302° F)	3.5 bar (50 PSIG)
SRU 1-6	15 minutes	15 minutes	150° C (302° F)	3.5 bar (50 PSIG)
SX 1-7	15 minutes	15 minutes	150° C (302° F)	3.5 bar (50 PSIG)
OptiLobe 10-30	30 minutes	30 minutes	150° C (302° F)	3.5 bar (50 PSIG)
OptiLobe 40-50	45 minutes	45 minutes	150° C (302° F)	3.5 bar (50 PSIG)
OS 10-40	15 minutes	15 minutes	150° C (302° F)	10 bar (145 PSIG)

Table 8.2.5a Operational data, heating/cooling devices

Fig. 8.2.5.1a Heating elements fitted to casing on DuraCirc

8.2.5.1 Circumferential Piston Pumps

DuraCirc pumps are pre-drilled to accept bolt-on heating/cooling jackets with O-ring elastomer sealing on the rear of the pump casing. Jackets are therefore retrofittable to installed pumps also (Fig. 8.2.5.1a).
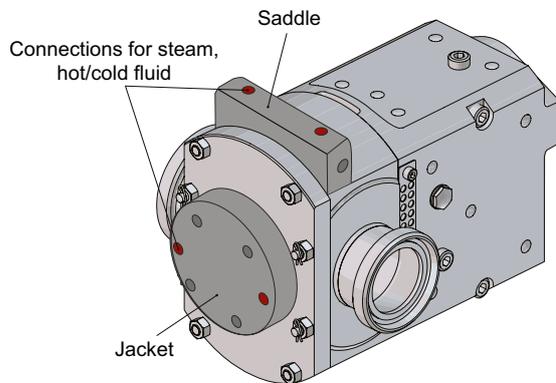
Fig. 8.2.5.2a SRU rotary lobe pump with heating/cooling jacket and saddle
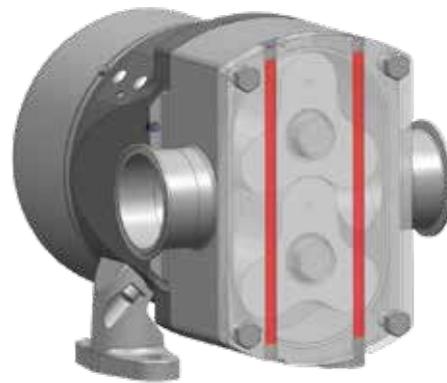
Fig. 8.2.5.2b OptiLobe rotary lobe pump with heating/cooling cover

8.2.5.2 Rotary Lobe Pumps

Heating/cooling jackets on the front cover are available on both the OptiLobe, SRU and SX pump ranges, but saddles are only available on the SRU pump range (Fig. 8.2.5.2a - 8.2.5.2b).
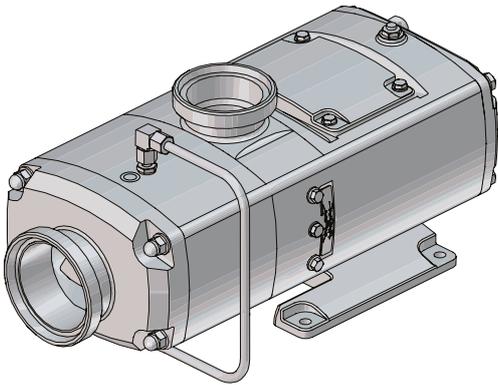
Fig. 8.2.5.3a OS Twin Screw pump with heating/cooling jacket

8.2.5.3 Twin Screw Pumps

On OS pumps the heating/cooling device option is fitted in the form of a modified casing including link piping between the heating/cooling chambers (Fig. 8.2.5.3a).
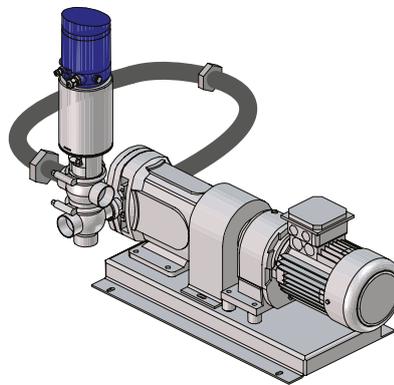
Fig. 8.2.6a Bypass/pressure relief configuration

8.2.6 Pump Overload Protection

Due to the positive action of the positive displacement pump any restriction on the outlet side of the pump, either partial or total, will result in excessive pressure developing in the pump casing. It is therefore recommended that some form of overload protection is installed to protect the pump, drive unit and also limit pressure build up within associated process equipment.

By-pass loop

With a by-pass loop, excess pressure will be relieved by bypassing the product through a loop back to the suction side of the pump (Fig. 8.2.6a). This protection will normally take the form of an external spring-loaded pressure relief valve fitted to the outlet side of the pump which will open under high pressure and allow fluid to return to the inlet side of the pump.

Pressure Relief Valve
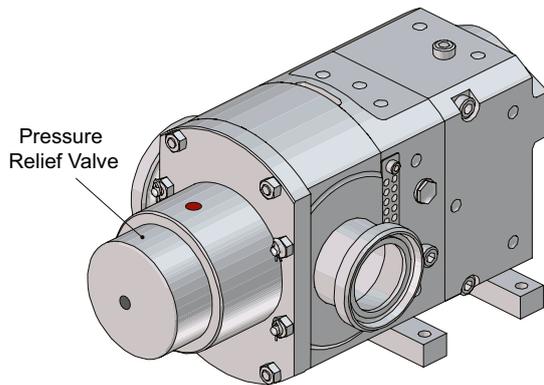
Fig. 8.2.6b SRU rotary lobe pump with pressure relief valve

Pressure Relief Valves

On SRU rotary lobe pumps it is also possible to supply a pressure relief valve as an integral part of the pump which means that external pipework is not required – It is important to note, this is a pump protection relief valve only, it is not to be utilized as a system relief valve, to which an external valve arrangement such as the by-pass loop would be required (Fig. 8.2.6b).

To fit a pressure relief valve a special rotor case cover is required. The valve will provide full pump protection for fluids having viscosities below 500 cP, above this figure Alfa Laval should be consulted with regard to specific flow rates in relation to viscosity and differential pressures. The design is such that the valve mechanism is isolated from the pumped fluid.

Product passes through open slip path

Piston opens when preset pressure reached
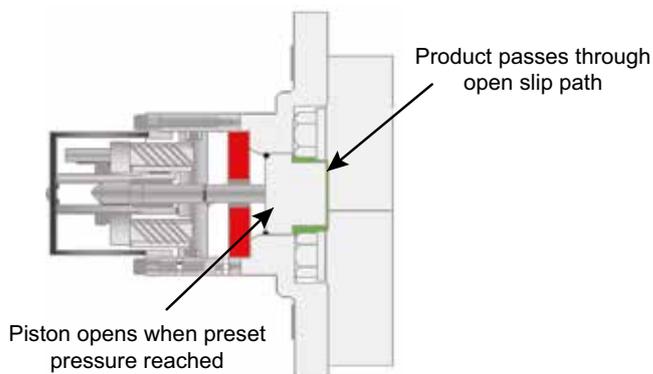
Fig. 8.2.6c Relief valve operation

As it is a mechanical device the relief valve does not operate instantaneously due to mechanical response time. The valve will begin to relieve at a pressure less than the fully open pressure (Fig. 8.2.6c). This 'accumulation' will vary depending upon the duty pressure, viscosity and pump speed. The accumulation tends to increase as pressure or pump speed decrease, and as viscosity increases. The valve is set to relieve at the required pressure by the correct choice of springs and can be adjusted on site to suit actual duty requirements.

Pressure relief valves are only available for SRU pumps fitted with metal rotors. They can be retrofitted to installed pumps and pumps will still be suitable for bi-direction operation.

The relief valve can be provided with the following options:

- Automatic with Pneumatic Override
  This valve may be pneumatically overridden for CIP conditions, and it may be remotely controlled if required. Air supply should be clean and dry at pressures of 4 bar (60 PSIG) minimum and 8 bar (115 PSIG) maximum

- Automatic with Manual Override
  This valve has a lever to enable manual override for CIP or certain tank filling applications

Valve Type	Pump Range - Availability	Normal Operating Pressure Range	
		bar	PSIG
Standard	SRU1-6	7-19	100-5
Pneumatic override	SRU1-6	7-19	100-5
Manual override	SRU1-3	19	5
	SRU4-5	7-10	100-145
	SRU6	7	100

Table 8.2.6a SRU pressure relief valve overview

8.2.7 Surface hardening

Care should be taken when handling abrasive media, i.e., products such as inks have very fine particles, whilst other products such as sugar slurries, can contain much larger particles which can lead to excessive pump wear.

To combat this issue, consideration needs to be given to pumping speed, temperature and differential pressure.

Depending upon the abrasion level of the product; consideration should be given to include for the additional hardened option to improve wear resistance.

8.2.7.1 Rotary lobe pumps

Alfa Laval SRU pumps can be specified with diffusion hardening to obtain a very high surface hardness rating between 1200-1400 HV0.05 with diffusion depth of 25 μm.

Hardening is applied to the pump-head which covers the casing, rotors, rotor nuts and front cover.

8.2.7.2 Twin Screw pumps

The Alfa Laval Twin Screw pump has hardened pump-head casing as the default.

Surface hardness measurement - typically 1092 HV0.05 with diffusion depth of 25.5 μm and can be an option to also harden the screws for the highly abrasive media.

Alfa Laval recommends an absolute operational limit of 600 rpm for twin screw pumps on any media that could be considered "abrasive".

It is important to note, diffusion hardening is a process that does not reduce material chemical resistance (not a coating and does not change the chemical composition of the surface), therefore unlike other traditional hardening processes such as plasma nitriding, which reduces chemical resistance, can be considered suitable for use in hygienic applications.

8.2.8 Ancillaries

Positive displacement pumps can be supplied with bare shaft (without drive) or mounted on a baseplate with an electric motor. Electric motors are described in more detail in chapter 9.

Drives

Rotary lobe and circumferential piston pumps generally operate at low to medium speeds i.e., 25 to 650 rev/min, and therefore some form of speed reduction is often required from normal AC motor synchronous speeds of 1500, 1000 and 750 rev/min for 50 Hz (1800, 1200 and 900 rev/min for 60 Hz). This is generally achieved by using a geared electric motor direct coupled to the pump drive shaft via flexible coupling.

When exact flow is not critical a fixed speed drive is generally used. The integral geared electric motor is the most commonly used type of fixed speed drive. This is a compact unit, which is easy to install, as it only requires one coupling and a safety guard. Complete ranges of drive speeds are available and usually one can be found within a few rev/min off the required speed.

As twin screw pumps are often selected due to the process flexibility they offer, they would typically not be selected for a duty requiring one fixed speed. In low-speed applications (<1000 rev/min) however, it can still be relevant to use a geared electric motor. For applications requiring higher speeds (e.g. CIP) a direct coupled motor (4, 6 or 8 pole) is the common choice with twin screw pumps.

To handle changing duty conditions or a number of different duties, it may be necessary to use a frequency converter (inverter) to obtain correct pump duty speeds. The frequency converter allows the operator to change the frequency of the electric motor, thereby changing pump speed and controlling flow (see chapter 9).

Fig. 8.2.8a DuraCirc circumferential piston pump, with shroud and geared motor on folded baseplate

Fig. 8.2.8b DuraCirc circumferential piston pump with shaft guard and geared motor baseplate with adjustable ball feet

Baseplates

The Alfa Laval 'standard' is a folded stainless-steel design (Fig. 8.2.8a) which is required to be bolted to the floor (see chapter 12).

In some application areas, such as dairy or brewing, it is normal practice to hose down pump units and floorings – in these circumstances ball feet can be fitted to baseplates, which can be a fixed or variable height and conform to 3A standard (see chapter 12), to raise baseplate above floor level (Fig. 8.2.8b). Baseplates can also be designed to meet specific customer standards when required.

Guards

All rotating machinery should be adequately guarded and when pumps are supplied complete with a drive, a guard is fitted over the flexible coupling which links the pump drive shaft to the output shaft of the electric motor.

The selection of guard material is important relative to its working environment. Non-sparking materials such as aluminium or brass are used with flameproof/explosion proof motors in hazardous areas. For non-hazardous applications stainless steel is generally used.

Shrouds

As an alternative to the guard a stainless-steel shroud covering both flexible coupling and complete electric motor is available. The purpose of the shroud is to protect the motor during washdown of the process area.

Fig. 8.3a - Q-doc

8.3 Q-doc

Alfa Laval Q-Doc is a complete documentation package, meeting the needs and demands of customers within the BioPharmaceutical industry (Fig. 8.3a).

Based on Good Documentation Practice (GDP), Q-doc documents every aspect from raw material to delivered equipment. With full transparency of sourcing, production, and supply chains it is a simple matter to trace even the slightest change in material or manufacturing procedures – even when it comes to spare parts.

Alfa Laval Q-doc comprises conformity declaration on EU food trace, elastomers and surface finish, material certificates, relevant test certificates and information about necessary Alfa Laval spare parts kits for standard components. The documentation package supports a smooth qualification and validation process and safeguards long-term peace of mind.

Q-doc declarations

* Compliance with Regulation (EC) No.: 1935/2004

* Compliance to EN 10204 type 3.1 (MTR)

* Compliance to the U.S. Food & Drug Administration CFR 21 (non-metallic parts)

* Compliance to the U.S. Pharmacopeia (Elastomers and Polymers)

* TSE (Transmissible Spongiform Encephalopathy)/ ADI (Animal Derivative Ingredient)

* Surface finish compliance to specification

* Passivation and electro polishing compliance to specification (if specified)

Availability of the Alfa Laval Q-doc is situated within the Alfa Laval UltraPure pump range:

Centrifugal pumps: LKH UltraPure, LKH Prime UltraPure

Positive Displacement pumps: SX UltraPure

The complete Q-doc documentation pack for each pump and/or service kit can be downloaded by the user from Alfa Laval's website, simply by typing in pump serial number or service kit lot number.

To download a Q-doc go to Alfa Laval FindMyCert.

Fig. 8.4.a Condition Monitor

Fig. 8.4.b Condition monitor connect - Cloud storage solution

8.4 Alfa Laval Condition Monitor

The Alfa Laval CM condition monitor is a quick and easy battery-operated device, to attach to rotating equipment and detect any change in the equipment behaviour; Industry 4.0 technology ready (Fig. 8.4a).

Providing users via Bluetooth radio with easy, safe data to enable them to optimise their process uptime, assist in maintenance scheduling and efficiency and reduce operating costs.

The Alfa Laval CM periodically measures the tri-axial vibration of the installed unit and the internal temperature storing 3 months of data for analysis, comparing it to the original baseline set-up values and pre-set warning and alarms, which if exceeded provide a visible indication via its LED and via the users compatible connected mobile device (Fig. 8.4b).

To add this to your pump configuration in the Alfa Laval Anytime configurator tool, use the "Condition Monitor" tab and select the CM Kit. This will add another line to your quote/order and the kit will be supplied in the box with the pump.

Fig.8.4.c Condition Monitor on LKH centrifugal pump

Fig. 8.4.d Condition Monitor on positive displacement pump

This chapter describes electric motors, including information on motor protection, methods of starting, motors for hazardous environments and speed control.

9.0

Motors
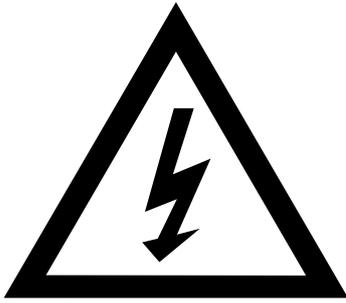
Fig. 9a Electrical hazard

All Alfa Laval pump ranges can be fitted with AC type Totally Enclosed Fan Cooled (TEFC) squirrel cage, three phase electric motors complying with various international standards and regulations such as IEC, CENELEC, VDE, DIN, BS and UL.

Electric motors supplied in the US are generally to NEMA (National Electrical Manufacturers Association) standard.
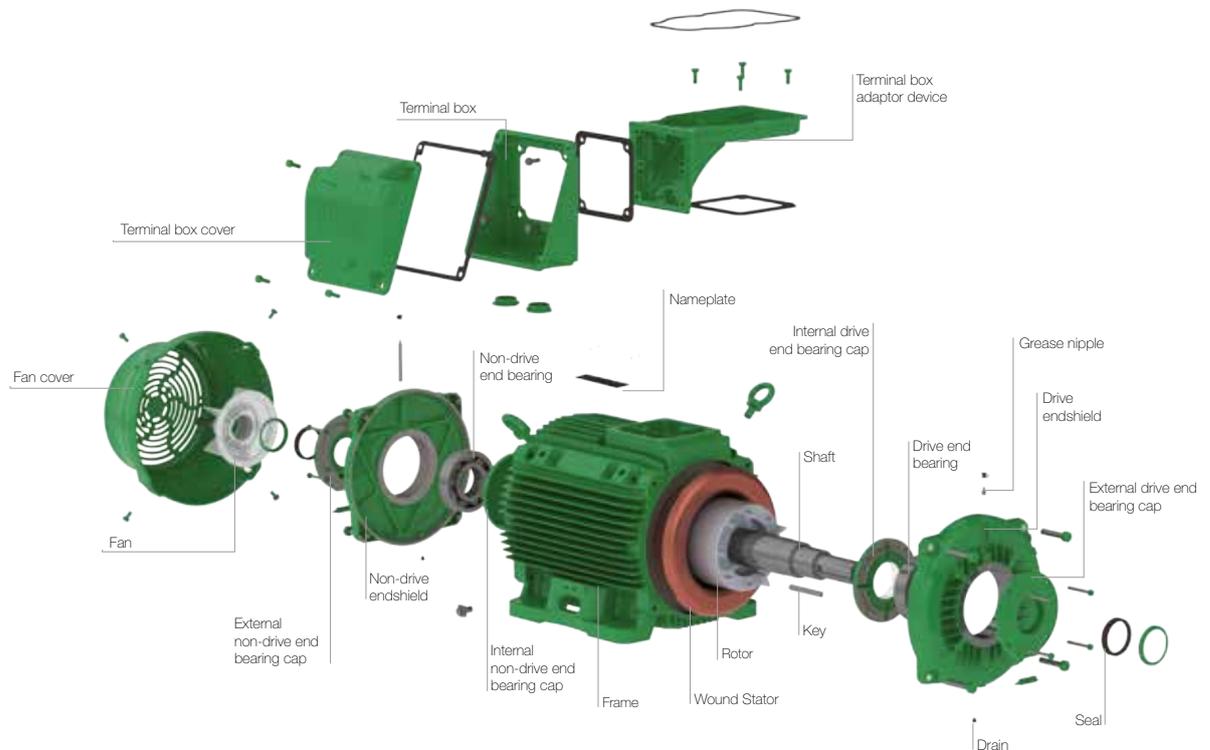
Fig. 9a - exploded view of a TEFC Induction motor

An AC (alternating current) motor is a type of electric motor that converts electrical energy into mechanical energy by utilizing the principles of electromagnetic induction. The standard design of an AC motor consists of several key components, each playing a crucial role in its operation (see Fig. 9a).

The motor is constructed as follows:

1. Stator

The stationary part of the motor is primarily responsible for generating a rotating magnetic field. It consists of a laminated core made of high-permeability magnetic material, typically stacked thin steel sheets. The core is designed to reduce eddy current losses and improve the efficiency of the motor. The stator also houses the stator windings, which are typically made of copper wire and are wound around the core in specific configurations.

2. Rotor

The rotor is the rotating part of the motor and is subjected to the rotating magnetic field generated by the stator. It is typically composed of a laminated iron core with conductive bars or coils embedded within it. The rotor windings are connected to a metal (iron) core, allowing the flow of current. When the stator's magnetic field interacts with the rotor's conductive elements, it induces an electromagnetic field that causes the rotor to rotate.

3. Bearings

AC motors incorporate bearings to support and facilitate smooth rotation of the rotor. These bearings are typically ball bearings or roller bearings and are positioned at each end of the motor's shaft. They provide low friction and support the rotor's weight, ensuring stable and efficient operation.

4. Shaft

The motor shaft connects the rotor to the external load or the driven equipment. It transfers the rotational motion generated by the rotor to the mechanical system being powered by the motor. The shaft is usually made of hardened steel and is carefully balanced to minimize vibration and ensure smooth operation.

5. Frame

The motor frame encloses and protects the internal components of the motor. It provides structural integrity and houses the stator, rotor, bearings, and other internal parts. The frame is typically made of cast iron, aluminium, or steel, depending on the motor's size, and intended application.

6. Cooling System

AC motors generate heat during operation, and it is important to dissipate this heat to maintain optimal performance and prevent overheating. AC motors employ various cooling methods such as forced air cooling, liquid cooling, or a combination of both. Cooling fins, fans, or cooling jackets may be integrated into the motor's design to ensure efficient heat dissipation.

7. Terminal Box

The terminal box is a housing located on the exterior of the motor and contains the electrical connections. It provides a convenient access point for connecting power supply cables to the motor windings. The terminal box often includes terminals, such as screw terminals or terminal blocks, for connecting the motor to the appropriate power source.

8. Protective Devices

To safeguard the motor from electrical faults or abnormal operating conditions, protective devices such as thermal overload relays or circuit breakers are often incorporated. These devices monitor motor parameters such as current, temperature, and voltage, and can interrupt the power supply to the motor in case of an overload or fault.

These are the basic components and features found in the standard design of an AC motor. The specific design and characteristics may vary depending on the motor's size, power rating, and intended application, but the fundamental principles remain the same.

9.1 Output Power

The output power of an AC motor is related to the active power, also known as the real power or true power, consumed by the motor. The rated output power of a motor refers to the maximum power it is designed to deliver continuously under normal operating conditions.

The rated output power is typically specified by the motor manufacturer and represents the motor's capacity or capability to perform work. It is usually expressed in units of watts (W), kilowatts (kW) or horsepower (HP).

In an ideal scenario where the motor operates at unity power factor (PF = 1), the active power is equal to the apparent power. In this case, the output power would be equal to the apparent power demanded by the motor from the power source.

However, in practical situations, the power factor (PF) of an AC motor is often less than 1 due to the presence of reactive power. Reactive power arises from the inductive or capacitive components of the motor's circuit, which can cause the current to lead or lag behind the voltage.

The power factor (PF) is defined as the ratio of active power (P) to apparent power (S):

PF = P/S

As the power factor decreases (i.e., PF < 1), the apparent power increases in relation to the active power. This means that for the same output power, a motor with a lower power factor will demand more apparent power from the power source.

To calculate the output power of a motor, you need to consider the power factor. The output power (P_out) can be determined using the following formula:

P_out = P × PF

Where:
P_out = Output power (watts or kilowatts)

P = Active power (watts or kilowatts)

PF = Power factor (between 0 and 1)

Therefore, the output power of an AC motor is directly dependent on the active power consumed by the motor, which, in turn, is influenced by the power factor. By optimizing the power factor of the motor, you can maximize its efficiency and ensure that the motor operates closer to its rated output power.

The table below shows output power that is specified in standard ratings.

Frequency	Output Power in kW									
50/60 Hz	0.37	0.55	0.75	1.1	1.5	2.2	3	4	5.5	7.5
	11	15	18.5	22	30	37	45	55	75	90

Table 9.1a (IEC motors)

Frequency	Output Power in HP								
60 Hz	0.5	0.75	1	1.5	2	3	5	7.5	10
	15	20	25	30	40	50	60	75	100

Table 9.1b (Nema motors)

9.2 Rated Speed

The rated speed of an electric motor refers to the speed at which the motor is designed to operate under normal operating conditions while delivering its rated output power. The rated speed is typically specified by the motor manufacturer and is an important parameter when selecting a motor for a specific application. The description of rated speed differs for asynchronous (induction) motors and synchronous motors, so let us explore each of them:

Asynchronous (Induction) Motors

Asynchronous motors are the most commonly used type of AC motors. They operate based on the principle of electromagnetic induction. In an asynchronous motor, the rotor rotates at a speed slightly lower than the synchronous speed, which is determined by the frequency of the power supply and the number of poles in the motor.

The rated speed of an asynchronous motor is specified as the synchronous speed (Ns) divided by a slip factor (s). The slip is the difference between the synchronous speed and the actual rotor speed, expressed as a percentage or a decimal. The rated speed (N_rated) can be calculated using the formula:

N_rated = Ns * (1 - s)

Example: 4pole - induction motor

Ns = 1500 rpm

s = 4%

N_rated = 1500 * (1 – 0.04) => 1440 rpm

Typically, the slip at the rated load is small, resulting in a rated speed close to the synchronous speed. However, the actual speed of an asynchronous motor varies with the load and may be lower than the rated speed under heavy loads.

To calculate the slip factor (s) we can use the following formula:

s = (Ns – N) / Ns

s - slip factor
Ns - synchronous speed
N – rotor speed

Synchronous Motors

Synchronous motors are designed to operate at a speed that is perfectly synchronized with the frequency of the power supply. The rotor of a synchronous motor rotates at the same speed as the rotating magnetic field generated by the stator. This synchronous speed (Ns) is determined by the frequency of the power supply and the number of poles in the motor.

The rated speed of a synchronous motor is equal to the synchronous speed (Ns) since the rotor always moves at this speed. In other words, the rated speed is the actual speed of the motor at its rated output power. Synchronous motors are often used in applications that require precise speed control or applications where a constant speed is essential.

It is important to note that synchronous motors require a separate excitation source to maintain synchronization with the power supply frequency. This can be achieved through permanent magnets or by providing DC excitation to the rotor windings.

In summary, the rated speed of an asynchronous motor is specified as the synchronous speed divided by a slip factor, while the rated speed of a synchronous motor is equal to the synchronous speed itself. The rated speed is an important parameter that helps determine the motor's performance characteristics and suitability for a particular application.

An example between synchronous speed, rated speed, frequency and poles is shown in the table below.

No. poles	2	4	6	8	12
No. pairs of poles	1	2	3	4	6
Synchronous speed at 50 Hz - rev/min	3000	1500	1000	750	500
Rated speed at 50 Hz - rev/min	2880	1440	960	720	480
Synchronous speed at 60 Hz - rev/min	3600	1800	1200	900	720
Rated speed at 60 Hz - rev/min	3460	1720	1150	860	690

Table 9.2a

9.3 Voltage

Three-phase motors

For three-phase motors operating at 50 or 60 Hz frequency, the common voltage ratings used worldwide are as follows:

208 V

This voltage rating is commonly used in North America, particularly in commercial and residential settings.

It is suitable for smaller motors with power ratings typically ranging from a few hundred watts to several kilowatts.

The voltage is distributed across the three phases, resulting in a line-to-line voltage of 208 V and a line-to-neutral voltage of approximately 120 V.

230 V

This voltage rating is prevalent in various regions, including parts of North America, Europe, and Asia.

It is commonly used for smaller motors in residential, commercial, and light industrial applications.

The voltage is distributed across the three phases, resulting in a line-to-line voltage of 230 V and a line-to-neutral voltage of approximately 132 V.

380 V

The voltage rating of 380 V is commonly used in several regions, including parts of Asia (China), Europe, and Africa.

It is often employed in industrial and commercial applications.

The voltage is distributed across the three phases, resulting in a line-to-line voltage of 380 V and a line-to-neutral voltage of approximately 220 V.

400 V

This voltage rating is widely used in Europe, Asia, and other regions as a standard for industrial applications.

It is common for motors, with up to several hundred kilowatts in power.

The voltage is distributed across the three phases, resulting in a line-to-line voltage of 400 V and a line-to-neutral voltage of approximately 230 V.

460 V

This voltage rating is predominantly used in North America for industrial applications.

The voltage is distributed across the three phases, resulting in a line-to-line voltage of 460 V and a line-to-neutral voltage of approximately 266 V.

575 V

This voltage rating is primarily used in North America, particularly in industrial settings.

The voltage is distributed across the three phases, resulting in a line-to-line voltage of 575 V and a line-to-neutral voltage of approximately 332 V.

Alfa Laval supplied motors at 400 or 460 V will generally operate satisfactorily with voltage variations of ± 10% from the rated voltage as per IEC 60038.

9.4 Cooling

Motor cooling

Motor cooling is specified by means of the letters IC (International Cooling) in accordance with standards. The most common is IC411 (Totally Enclosed Fan Cooled - TEFC) where an externally mounted fan cools the motor.

Some common methods of cooling for motors are shown below:

Code	Arrangement
IC411	Totally Enclosed Fan Cooled (TEFC) – motor cooled by an externally mounted fan
IC410	Totally Enclosed Non-Ventilated (TENV) – self cooling, no externally mounted fan
IC418	Totally Enclosed Air Over Motor (TEAOM) – motor cooled by airstream
IC416	Totally Enclosed Blower Cooled (TEBC) – motor cooled by an independent fan

Table 9.4a

9.5 Insulation and Thermal Rating

Insulation

Insulation plays a critical role in ensuring the safe and reliable operation of the motor. It helps to protect the motor windings and other internal components from electrical breakdown and thermal damage. The insulation materials and thermal rating of an AC motor are important considerations in motor design and selection. Here are some key details:

Insulation Classes

AC motor insulation systems are categorized into different insulation classes based on their thermal capabilities and temperature limits.

The insulation classes are standardized and designated by letters, such as Class A, Class B, Class F, and Class H, among others.

Each insulation class has a specific maximum allowable operating temperature, which indicates the maximum temperature the insulation system can withstand without significant degradation or loss of insulation properties.

Thermal Rating

The thermal rating of an AC motor specifies its maximum allowable operating temperature based on its insulation class.

The thermal rating is typically expressed as a temperature rise above the ambient temperature, measured at a specific point on the motor, such as the winding or the stator.

For example, a motor with a Class F insulation system may have a thermal rating of 105° C, meaning the maximum allowable temperature rise is 105° C above the ambient temperature.

Temperature Monitoring and Protection

AC motors often incorporate temperature monitoring devices, such as thermistors (PTC) or thermostats, to measure the actual winding temperature.

These temperature sensors provide feedback to the motor control system, allowing for temperature-based protection and control strategies.

If the motor temperature exceeds safe limits, protective measures like thermal overload relays or motor protective relays may be activated to shut down or protect the motor from damage.

It is important to select an AC motor with an insulation class and thermal rating that is suitable for the operating environment and the expected temperature conditions. Operating the motor within the specified temperature limits helps ensure the longevity, efficiency, and reliability of the motor's insulation system and prevents insulation breakdown or thermal degradation.

Standard Alfa Laval supplied motors will operate satisfactorily in an ambient temperature range of -20° C (-4° F) to 40° C (104° F) (Class B temperature rise) and at altitudes up to 1000 metres above sea level – Classified as Class F/Temperature Rise B.

Motors supplied with class F insulation system with class B temperature rise (80° C) (176° F) ensure an exceptional margin of safety and longer life even in abnormal operating conditions such as withstanding ambient temperatures up to 55° C (131° F) or 10% overload or adverse supply systems. Motors operating in ambient temperatures higher than 55° C (131° F) will have class H insulation. Some de-rating of the motor may be necessary for high ambient temperatures and high altitude.

9.0 Motors

9.6 Protection

The degree of motor protection is specified by means of the letters IP (International Protection) in accordance with standards. These state the method of determining degrees of ingress protection for both dust and water. The letters IP are followed by two digits, the first of which specifies the protection against contact and ingress of foreign bodies and the second digit specifies the protection against water.

Table showing degrees of protection is shown below:

Designation	1st Digit	2nd Digit
	Protection against contact and ingress of foreign bodies	Protection against water
IP44	Protection against contact with live or moving parts by tools, wires, or other objects of thickness greater than 1 mm (about 0.04 in). Protection against the ingress of solid foreign bodies with a diameter greater than 1 mm	Water splashed against the motor from any direction shall have no harmful effect
IP54	Complete protection against contact with live or moving parts inside the enclosure. Protection against harmful deposits of dust. The ingress of dust is not totally prevented, but dust cannot enter in an amount sufficient to interfere with satisfactory operation of the machine	Water splashed against the motor from any direction shall have no harmful effect
IP55		Water projected by a nozzle against the motor from any direction shall have no harmful effect
IP56		Motor protected against conditions on a ship's deck or powerful water jets
IP65	No ingress of dust	Water projected by a nozzle against the motor from any direction shall have no harmful effect
IP66	No ingress of dust - "Dust-tight"	Motor protected against high pressure and powerful water jets/sprays

Table 9.6a

Alfa Laval offers protection levels of IP55 as standard within the hygienic industries with an option for the improved IP66 when requested/conditions demand – Please contact Alfa Laval for more information.

9.6.1 Basic UL/CSA/Nema Enclosure Types

Type 3

An enclosure which is intended for outdoor use primarily to provide a degree of protection against windblown dust, rain, and damage from external ice formation.

Type 3R

An enclosure which is intended for outdoor use primarily to provide a degree of protection against falling rain and damage from external ice formation.

Type 3S

An enclosure which is intended for outdoor use primarily to provide a degree of protection against rain, sleet, windblown dust, and to provide for operation of external mechanisms when ice laden.

Type 4

An enclosure which is intended for indoor or outdoor use primarily to provide a degree of protection against windblown rain and dust, splashing water, hose directed water and damage from external ice formation.

Type 4X

An enclosure which is intended for indoor or outdoor use primarily to provide a degree of protection against corrosion, windblown rain and dust, splashing water, hose directed water and damage from external ice formation.

Type 6

An enclosure which is intended for indoor or outdoor use primarily to provide a degree of protection against falling dirt, hose directed water, the entry of water during occasional temporary submersion at a specified depth and damage from external ice formation.

Type 6P

An enclosure which is intended for indoor or outdoor use to primarily provide a degree of protection against falling dirt, hose directed water and the entry of water during prolonged submersion at a specified depth and damage from external ice formation.

Tropic Proof Treatment

Motors operating in tropical climates are invariably subjected to hot, humid, and wet conditions, which will produce considerable amounts of condensation on internal surfaces. Condensation occurs when the surface temperature of the motor is lower than the dew-point temperature of the ambient air. To overcome this, motors can be supplied with special tropic proof treatment. Failure to include this treatment and the resulting corrosion can cause irreparable damage to stator windings and moving parts.
If required, please contact Alfa Laval for more information.

Anti-Condensation Heaters

Where the motor is to be left standing for long periods of time in damp conditions it is recommended that anti-condensation heaters are fitted and energised to prevent condensation forming in the motor enclosure. These heaters are normally 110 V or 220 V.
If required, please contact Alfa Laval for more information.

Thermistors

(PTC – Positive Temperature Coefficient)

To protect the motor windings from overload due to hot temperature, motors can be fitted with thermistors, which are temperature-dependent semi-conductor devices embedded in the motor windings. Where motors can be allowed to operate at slow speed, i.e., being used with a frequency converter (see section 9.9), it is normal to fit thermistors to prevent the motor from overloading or to insufficient cooling from the motor fan.

Thermostats

Mechanical or electronic devices designed to monitor and control temperature in a system. They typically consist of a bimetallic strip and operates based on the principle of thermal expansion. The bimetallic strips are installed within the motor windings or in close proximity to them. The bimetallic strip is composed of two different metals bonded together that have different coefficients of thermal expansion. As the motor current flows through the windings, they generate heat due to the electrical resistance. The bimetallic strip in the motor thermal protector is designed to bend with temperature changes. As the motor temperature rises above a certain threshold, the bimetallic strip bends, tripping a switch and interrupting the power supply to the motor. This action protects the motor from further heat build-up and potential damage. Once the motor cools down, the bimetallic strip returns to its original position, allowing the motor to be restarted.

Note:

All motors supplied by Alfa Laval come with thermistors or thermostats (US) as standard to allow frequency converter operation.
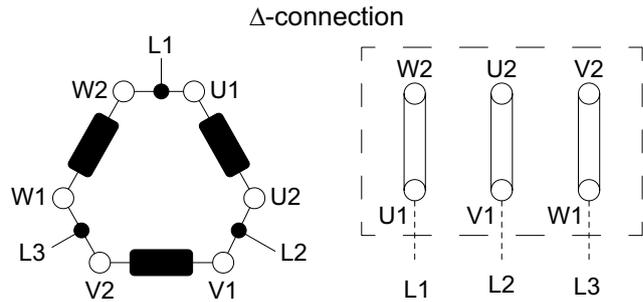
Δ-connection

Fig. 9.7a Connection of three-phase single speed motor

9.7 Methods of Starting

Choice of Starting Method

The choice of starting method depends on factors such as motor size, application requirements, and available power supply. Here are the main methods of starting an induction motor:

Direct-On-Line (DOL) Starting

This is the simplest and most common method of starting induction motors. In DOL starting, the motor is directly connected to the power supply, typically through a contactor or a circuit breaker. When the power is switched on, the motor receives the full supply voltage, causing it to start abruptly. DOL starting is suitable for small- to medium-sized motors but can cause high starting currents and mechanical stress.

Motors fitted to centrifugal and liquid ring pumps are normally directly started, as the moment of inertia of the motor is low due to pump design and the fluids being pumped having low viscosities. In this case the starting time with high starting current is incredibly low and it can consequently be ignored.

Star-Delta (Wye-Delta) Starting

Involves initially connecting the motor's stator windings in a star (wye) configuration during the starting period, which reduces the voltage across each winding. Once the motor reaches a predetermined speed, the windings are then switched to a delta (mesh) configuration for normal operation, where the voltage across each winding is higher. Star-delta starting reduces the starting current and torque, limiting the stress on the motor and the power supply (Fig. 9.7a).

If pumping viscous fluids or using a positive displacement pump, the starting time with the high starting current is longer and therefore requires some restriction of the starting current by using the star-delta method.
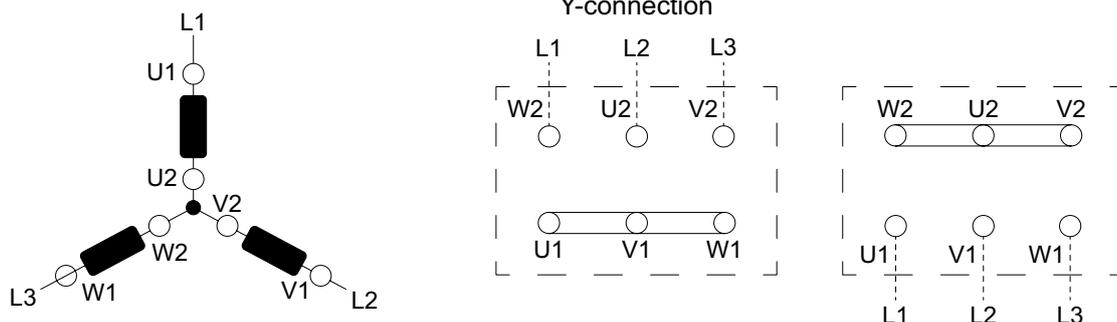
Fig. 9.7a Connection of three-phase single speed motor

Auto-Transformer Starting

This method is similar to star-delta starting and is used for larger motors with high starting currents. An auto-transformer is connected between the power supply and the motor's stator windings. Initially, the motor is connected to taps on the auto-transformer that provide a reduced voltage. As the motor accelerates, the taps are switched to provide a higher voltage. Auto-transformer starting helps reduce starting current and torque while providing a smooth acceleration.

Soft Starting

Soft starting methods are designed to gradually increase the voltage supplied to the motor during start-up, thus reducing the starting current and mechanical stress. This can be achieved using devices such as auto-transformers, solid-state soft starters, or variable frequency drives (VFDs). Soft starting minimizes voltage dips in the power supply and enables controlled acceleration of the motor. In many cases the soft starter saves energy by automatically adapting the motor voltage continually to the actual requirement. This is particularly important when the motor runs with a light load.

Frequency Converter Starting

Frequency converters, also known as variable frequency drives (VFDs) or inverters, allow precise control of the motor's speed and torque. They can be used for both starting and speed control of induction motors. VFDs convert the fixed-frequency AC power supply into variable frequency and voltage output, allowing smooth acceleration and deceleration of the motor. Frequency converter starting offers flexibility and energy savings but is more complex than other methods.

These are the main methods used for starting induction motors. The choice of method depends on factors such as motor size, application requirements, energy efficiency considerations, and cost constraints.

Alfa Laval recommends the use of a qualified electrician to best assess the optimal starting method and to carry out the required installation/set-up.

9.8 Motors for Hazardous Environments

ATEX

Short for "ATmosphères EXplosibles," is a European Union directive that outlines safety requirements for equipment used in potentially explosive atmospheres. It aims to protect workers and the environment from the risks associated with such environments. ATEX defines specific product categories and motor equipment protection methods to ensure safe operation in hazardous areas. ATEX classifies equipment into various categories based on their intended use in explosive atmospheres. The categories are as follows:

Zones

The degree of hazard varies from extreme to rare. Hazardous areas are classified into three Zones as follows:

Zone 0

An explosive gas-air mixture is continuously present or present for extended periods – No motors may be used in this zone.

Zone 1

An explosive gas-air mixture is likely to occur in normal operation.

Zone 2

An explosive gas-air mixture is not likely to occur in normal operation and if it occurs it will only be present for a brief time.

To ensure equipment can be safely used in hazardous areas, its gas group must be known, and its temperature class must be compared with the spontaneous ignition temperature of the gas mixtures concerned.

By implication, an area that is not classified Zone 0, 1 or 2 is deemed to be a non-hazardous or safe area.

Temperature class	Ignition temperature for gas/vapour	Max. permitted temperature of electrical equipment
T1	up to 450° C (842° F)	450° C (842° F)
T2	300 to 450° C (572 to 842° F)	300° C (572° F)
T3	200 to 300° C (410 to 572° F)	200° C (410° F)
T4	135 to 200° C (275 to 410° F)	135° C (275° F)
T5	100 to 135° C (212 to 275° F)	100° C (212° F)
T6	85 to 100° C (185 to 212° F)	85° C (185° F)

Table 9.8a

Group I	Equipment for coal mines susceptible to methane gas – Alfa Laval does not cover this group
Group II	Equipment for explosive atmospheres other than mines i.e., surface industries
IIA IIB IIC	Group II is subdivided according to the severity of the environment. IIC is the highest rating. A motor from one of the higher categories can also be used in a lower category

Table 9.8b

Flameproof Enclosure - Ex d

These motors are designated for operation in Zone 1 hazardous areas. The motor enclosure is designed in such a way that no internal explosion can be transmitted to the explosive atmosphere surrounding the machine. The enclosure will withstand, without damage, any pressure levels caused by an internal explosion.

The temperature of the motor's external enclosure should not exceed the self-ignition temperature of the explosive atmosphere of the installation area during operation. No motor device outside the flameproof area shall be a potential source of sparks, arcs, or dangerous overheating.

International standard IEC 60079-1

Suitable for Zones 1 and 2.

Increased Safety Design – Ex e/Ex ec

The design of this motor type prevents the occurrence of sparks, arcs, or hot spots in service, that could reach the self-ignition temperature of the surrounding, potentially explosive atmosphere, in all inner and outer parts of the machine.

International standard IEC 60079-7

Suitable for Zones 1 and 2.

Non-Sparking Design – Ex ec

These motors are designated for operation in Zone 2 hazardous areas. The motor construction is similar to standard TEFC motors, but with special attention to eliminate production of sparks, arcs, or dangerous surface temperatures.

International standard IEC 60079-15

Suitable for Zone 2 only.

When requiring use of an ATEX approved motor drive, Alfa Laval offers Zone 1, Ex d, IIB, 2G, and T4 as standard.

Please contact Alfa Laval for alternative options.

In North America, hazardous motors are classified and regulated by the National Electrical Code (NEC) and the Canadian Electrical Code (CEC). These codes provide guidelines for the safe installation and use of electrical equipment, including motors, in hazardous locations. The classifications and standards in North America differ from the ATEX system used in Europe.

Here is an overview of the North American standards and classifications:

Class and Division System (NEC)

The NEC makes use of a Class and Division system to classify hazardous locations based on the type of hazardous substance present. It categorizes hazardous locations into three classes and two divisions:

Class I: Locations where flammable gases, vapours or liquids are, or may be, present in sufficient quantities to cause a fire or explosion.

Class I is further divided into Divisions 1 and 2.

- Division 1: Hazardous substances are present under normal operating conditions or during abnormal situations, such as leaks or equipment failure

- Division 2: Hazardous substances are handled, processed, or stored, but are not present under normal operating conditions or during abnormal situations

Class II: Locations where combustible dust is or may be present in sufficient quantities to cause a fire or explosion. Class II is further divided into Divisions 1 and 2, similar to Class I.

Class III: Locations where easily ignitable fibers or flyings are or may be present in sufficient quantities to cause a fire or explosion. Class III is not divided into divisions.

Group Classification (NEC)

Within each class, hazardous substances are further classified into groups based on their properties. The group classification specifies the type of substance and its level of hazard. For example:

Group A: Acetylene
Group B: Hydrogen, butadiene, ethylene oxide
Group C: Ethylene, propylene, acrolein
Group D: Propane, gasoline, acetone

When requiring use of an NEC explosion proof approved motor drive, Alfa Laval offers Class 1, Div 1, and Group D as standard.

Zone System (CEC)

The Canadian Electrical Code (CEC) also provides a classification system for hazardous locations, known as the Zone System. The CEC divides hazardous locations into zones based on the likelihood and duration of the presence of flammable substances. The zones are categorized as Zone 0, Zone 1, Zone 2, and Zone 20, Zone 21, and Zone 22 for gases and dust, respectively.

9.9 Energy Efficient Motors

Motor energy efficiency plays a crucial role in reducing energy consumption and promoting sustainable practices in various industrial and commercial applications. In recent years, governments and regulatory bodies around the world have introduced minimum energy efficiency regulations to encourage the use of more efficient motors. Four widely recognised classification in this regard are IEC standards such as IE1, IE2, IE3, IE4 and IE5.

Motor energy efficiency refers to the ability of an electric motor to convert electrical energy into mechanical energy with minimal losses. Traditional motors are known to have significant energy losses due to factors such as resistive losses, mechanical losses, and stray losses. These losses result in wasted energy and increased operating costs.

IE1 (Standard Efficiency Standard)

IE1 is the basic energy efficiency standard for motors, defined by the International Electrotechnical Commission (IEC) under IEC 60034-30-1. Motors that meet the IE1 standard have relatively lower energy efficiency compared to more advanced motor designs. However, they are still widely used in applications where energy efficiency is not a primary concern.

IE2 (High Efficiency Standard)

IE2 is an intermediate energy efficiency standard, also defined by the IEC under IEC 60034-30-1. Motors that comply with the IE2 standard offer higher energy efficiency compared to IE1 motors. These motors are designed to reduce energy losses and are considered an improvement over IE1 motors.

IE3 (Premium Efficiency Standard)

IE3 is an international standard for energy-efficient motors, defined by the IEC under IEC 60034-30-1. According to this standard, motors must meet specific efficiency levels to be classified as IE3. IE3 motors have higher efficiency compared to IE1 and IE2 motors and are considered a significant improvement in energy performance.

IE4 (Super Premium Efficiency Standard)

IE4 is a more stringent energy efficiency standard introduced as an extension of the IE3 standard. Also known as super premium efficiency motors, IE4 motors have even higher efficiency levels than IE3 motors. These motors are designed to minimize energy losses and are typically used in applications where energy savings are critical.

IE5 (Ultra Premium Efficiency Standard)

IE5 is the highest energy efficiency standard currently defined by the IEC. IE5 motors offer the highest level of efficiency among all the standards. These motors are designed using advanced technologies and materials to achieve exceptional energy performance and are typically used in applications where maximum energy savings and performance are required.

Alfa Laval's supply of motors are also in accordance with the Minimum energy Efficiency Regulations (MEPs) with IE3 as the default efficiency level.

Please contact Alfa laval on alternative requests for higher efficiency levels such as IE4 or IE5.

9.9.1 Minimum Energy Efficiency Regulations (MEPs)

To promote the adoption of energy-efficient motors and reduce overall energy consumption, many countries have implemented minimum energy efficiency regulations. These regulations typically specify the minimum efficiency levels that motors must meet to

be legally sold and operated in those regions. The IE1, IE2, IE3, and IE4 standards are commonly used as benchmarks in these regulations and are under constant review and change to further push the most sustainable offerings in motor energy efficiency.

Benefits of High-Efficiency Motors

Energy Savings: High-efficiency motors significantly reduce energy consumption, leading to lower operating costs and decreased carbon emissions.

Cost Savings: Although high-efficiency motors may have a higher initial cost, the energy savings over the motor's lifetime usually outweigh the upfront investment.

Environmental Impact: Using energy-efficient motors helps reduce greenhouse gas emissions and contributes to environmental sustainability.

Enhanced Performance: Efficient motors often offer improved performance, including better speed control, reduced noise levels, and increased reliability.

Motor energy efficiency and minimum energy regulations such as IE1, IE2, IE3, and IE4 have transformed the motor industry by promoting the development and adoption of more energy-efficient technologies. These standards have played a vital role in reducing energy consumption, lowering operating costs, and mitigating the environmental impact of motor-driven systems. By adhering to these regulations, industries can contribute to a more sustainable future while reaping the benefits of energy and cost savings.

When required to supply motorised pumping units, Alfa Laval offers a comprehensive range of motors tailored to meet the diverse needs of customers across all countries, while adhering to the minimum energy efficiency requirements set forth by the respective regulatory bodies.

IEC 60034-30-1	IE1	IE2	IE3	IE4	IE5
NEMA MG1	Std	High	Premium		
NBR 17094-1		IR2	IR3		
GB 18613-2012		GB3	GB2	GB1	
GB 18613-2020			GB3	GB2	GB1

Table 9.9.1a Overview on common efficiency grades globally

9.10 Speed Control

The effective speed control of AC electric motors has long been regarded as an adaptable and economical means of reducing costs and saving energy.

Multi-Speed

Pole Change (Tapped or Dahlander).
These have a single winding and two speeds in a ratio of 2:1 and can be supplied for constant torque or variable torque applications.

PAM (Pole Amplitude Modulation)

Similar to above except that pole variations can be 4/6 or 6/8.

Dual Wound

Motors have two separate windings and can be supplied for any two speed combinations.

A combination of dual and pole change windings can give 3 or 4 speeds from one design.

Mechanical Speed Control

In some cases, mechanical speed control methods can be employed. These typically involve using adjustable sheaves or pulleys to change the effective diameter of the pump drive system. By adjusting the size of the pulleys, the rotational speed of the pump can be altered. This method is often used in older or simpler pump systems.

Throttling Control

Throttling involves partially closing a valve or using a bypass line to restrict the flow of the pumped fluid. By increasing the resistance to flow, the pump's operating point moves to a lower flow rate and pressure condition. However, throttling control is generally not an efficient method since it wastes energy and may cause excessive wear on the pump – Important not to be used within systems that include positive displacement pumps as closing the valve increases the pressure in the PD pump and the entire system. The pump will continue to work against the developed pressure until it reaches its maximum operating pressure, or the relief valve opens. This increased pressure will put additional strain on the pump, its components, and the piping system and if not corrected, leads to failure.

Variable Voltage

Variable voltage control provides a low capital cost means of varying the motor speed on centrifugal pumps. This form of speed control requires greater derating than for converter drives and is best suited to 4 pole machines of 2:1 speed reduction with close matching of motor output to absorbed pump load. These motors are of special design – standard motors being unsuitable.

Frequency Converter (Inverter)

The use of a frequency converter will allow speed control of a standard AC motor by adjusting the frequency, although some derating may be necessary. Basic frequency converters will permit operation over a typical speed range of 20:1. With increasing sophistication such as 'vector' control, e.g., field oriented control utilising closed loop feedback, the effective speed range can be increased to 1000:1.

For applications using variable torque loads such as centrifugal pumps, little derating will be required. For applications using constant torque loads such as positive displacement pumps, the level of derating will depend on the speed range required.

The motor ratings must take into account:

- Increased heating due to the harmonic content of the inverter waveforms

- Reduced cooling arising from motor speed reduction

- The power or torque requirements throughout the entire speed range

- Other limiting factors such as maximum motor speeds, ambient temperature, altitude etc.

- When using frequency inverters, it is important to consider the potential impact on power quality. VFDs can introduce harmonics into the electrical system, which may require additional measures like harmonic filters or power conditioning to mitigate their effects on other connected equipment.

As well as motors being remotely controlled by frequency converters, electric motors can be made available with the frequency converter already integrated to the motor. These arrangements have the advantage of not using any shielded motor cables, as there are no extra connections between the frequency converter and motor. Also providing room in a switch cabinet will not be necessary.

9.11 Motor Sizing Values

Values to consider when sizing for a motor

- Shaft power (kW/HP)

- Speed (rpm)

- Torque (Nm/(lb ft))

These 3 values have direct correlation that can be seen by below formula.

Metric units

Power, speed and torque:	Re-arrange for torque:
M = P * 9550 / n	M = P * 9550 / n
Where:	This arrangement shows how speed and power impacts on torque:
P = Power (kW) M = Torque	P = Power (kW) M = Torque (Nm) n = Speed (rpm)

Imperial US units

Power, speed and torque	Re-arrange for torque
M = P * 5252 / n	M = P * 5252 / n
Where:	This arrangement shows how speed and power impacts on torque:
P = Power (hp) M = Torque (lb ft) n = Speed (rpm)	P = Power (hp) M = Torque (lb ft) n = Speed (rpm)

9.11.1 Torque

The main consideration for selecting an electric motor is torque - not power.

Running a motor over the rated torque output will cause increased current flow which in most cases will cause the inverter safety features to stop the motor but could lead to permanent damage to the motor windings due to over-heating.

When it comes to the type of pump technology, a positive displacement pump is considered a constant torque application where a change in duty does not automatically correlate to a reduced torque requirement, therefore it is imperative the motor torque output covers the complete adjustment range of required duty points. Centrifugal pumps have a torque characteristic curve which increases in a quadratic function, which favours operation at low speeds as per the affinity laws but dramatically increases when increasing operation past the original selection point (see chapter 7 for more details).

Alfa Laval pumps can operate over speed ranges; therefore, consideration needs to be made to the impact the variable speeds have on the torque output of the motor and the varying power requirements of the pump at the different duty point to ensure sufficient motor power and torque is available over the full speed range.

9.11.2 Speed/Frequency

Frequency correlates directly to the motor speed and normally using a variable frequency drive or inverter.

- Decreasing frequency = Decreasing speed

- Increasing frequency = Increasing speed

The most common supply frequencies are 50 Hz or 60 Hz

	4–pole	6–pole	8–pole
50 Hz	1500 rpm	1000 rpm	750 rpm
25 Hz	750 rpm	500 rpm	375 rpm
10 Hz	300 rpm	200 rpm	150 rpm

- All motors supplied with Alfa Laval pumps are suitable for frequency inverter use

- Standard motor offering insulation class F/ Temperature rise B (80 °Kelvin)

- All applications involving operating speeds under 5Hz, please contact Alfa Laval Technical Support

	4–pole	6–pole	8–pole
60 Hz	1800 rpm	1200 rpm	900 rpm
30 Hz	800 rpm	600 rpm	450 rpm
10 Hz	360 rpm	240 rpm	180 rpm
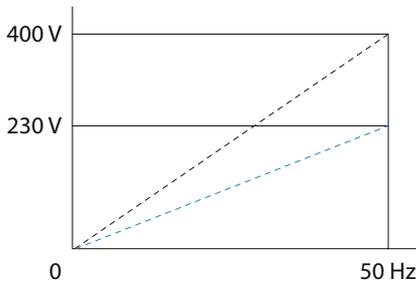
Fig. 9.11.3a Voltage to frequency for constant torque

9.11.3 Torque/Frequency

An electric motor in theory, will generate constant torque when frequency is reduced below rated value (50/60 Hz); This is handled by a Variable Frequency Drive (VFD) also known as an Inverter (see Fig. 9.11.3a).

A reduction of speed <50 Hz does not result in an increased torque, as is the case with adjustable gear units, but rather to a reduction in power. In this case, the current remains constant for a constant torque and the voltage reduces with the frequency.

Physical conditions for constant torque:

M = constant → Φ = constant → U/f = constant
Torque   Magnetic flux   Voltage/Frequency

The condition V/F = constant can maximally only be realised by the frequency inverter up to the nominal operating point. A further voltage increase above that of the mains voltage is technically impossible.

Physical factors can impact the torque output of the motor when speed is decreased, preventing constant torque.

For example, in self-cooled motors the torque output decreases as the motor speed reduces. This is caused by the reduced fan speed leading to reduced motor cooling causing an increase in motor temperature and therefore reduced power/torque rated output.

Torque derating for self cooling motors

A = Slower the speed, the greater the torque derates   E = 25 Hz
B = Constant Torque Range (No derating)   F = 50 Hz
C = Constant flux. Constant V/f. 50 Hz Supply
D = Note:
     1.0 represents 100%
     Frequency output: 50 or 60 Hz
     0.5 would represent 50%; 25 or 30 Hz
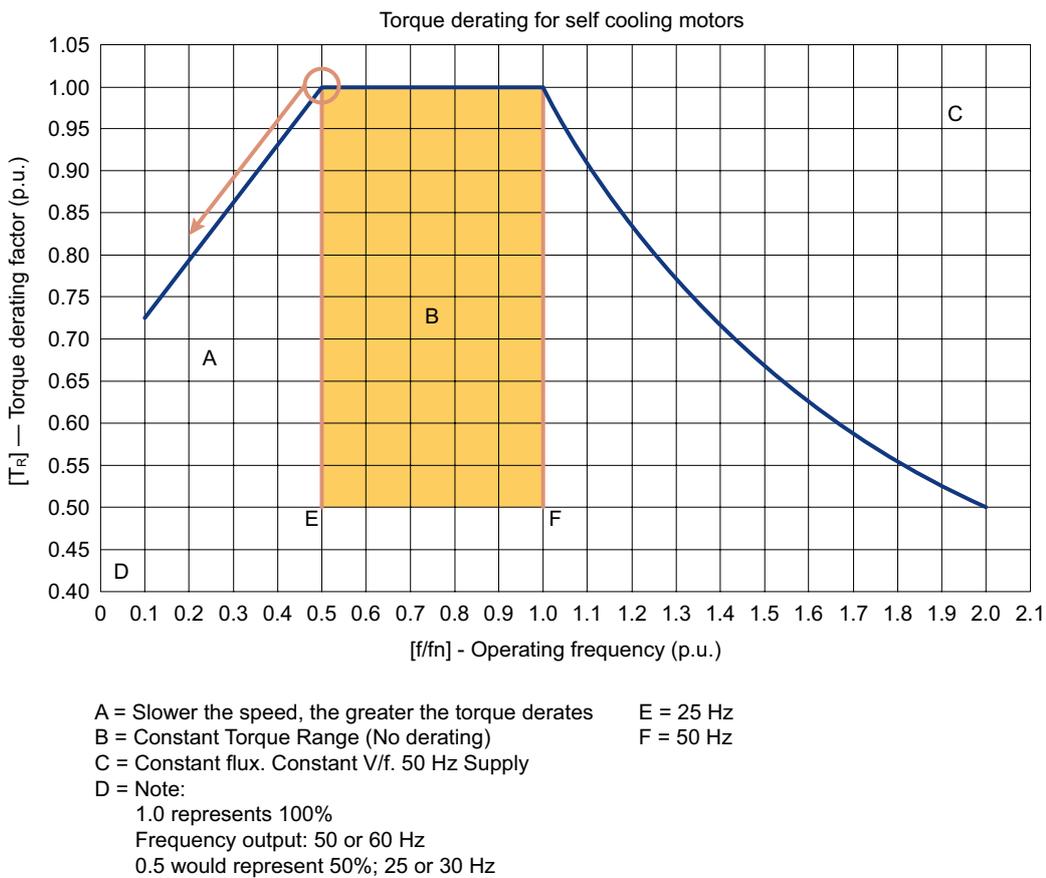
Fig. 9.11.3b Above shows the relationship of torque to frequency

With a self-cooled (TEFC) motor on a supply frequency of 50 Hz, there is a constant torque between 50 - 25 Hz (50% or 2:1) meaning no derating in torque output. Below 25 Hz, the torque output is derated due to the slower running speed of the integrated fan, in order to dissipate the additional heat generated (see Fig. 9.11.3b).

To counter torque derating at reduced speeds, the following two options can be considered:

1. Forced blower cooled motors (TEBC) are equipped with a separate fan driven by a separate motor thereby ensuring 100% airflow regardless of motor running speed and no derating is required due to increase motor temperature.

This results in 100% torque decreasing to 5 Hz (See Fig. 9.11.3c).

TEBC units can also be used to help optimise motor selections where duty points are below the 2:1 motor turn-down and require a larger motor to meet torque requirements after derating.

Torque derating for TEBC motors

A = Constant Torque Range (No derating)
B = Here we can see full torque from 1.0 (50/60 Hz) down to 0.1 with forced ventilation keeping the motor cool
C = Forced ventilation. 50 Hz Supply

D = Note:
  1.0 represents 100%
  Frequency output: 50 or 60 Hz
E = 5-50 Hz
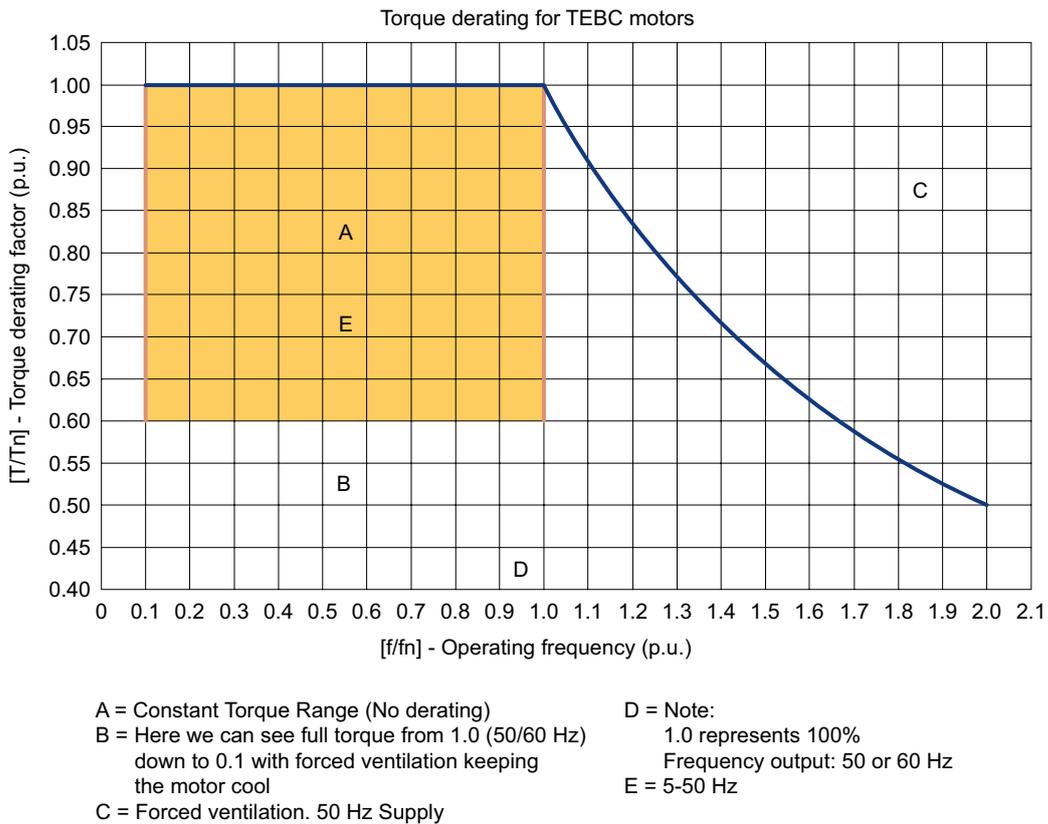
Fig. 9.11.3c Constant torque graph for TEBC motor
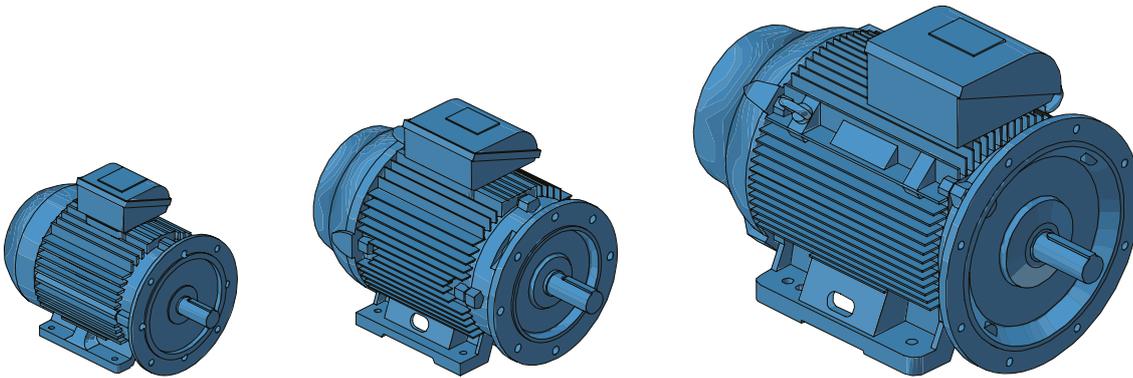
Fig.9.11.3d Range of IEC motor frame sizes

2. Increased motor size (see Fig. 9.11.3d). Increasing motor size is the alternative solution. With this, the motor is operated at a reduced load. Therefore, there is less power loss and an additional increased thermal reserve due to increased size of the motor.

Above the nominal frequency the available torque reduces, as the voltage is no longer increasing at higher frequencies (see Fig. 9.11.3a) the magnetic flux reduces. This range is known as the field weakening range. A further increase in frequency in the field weakening range therefore results in a torque reduction (See Fig. 9.11.3e).

Torque derating for self cooling motors

A = Constant Torque
B = Torque derating
   Here we can see full torque from
   1.0 (50/60 Hz) down to 0.1 with forced
   ventilation keeping the motor cool
C = Forced flux. Constant V/f. 50 Hz Supply

D = Note:
   1.0 represents 100%
   Frequency output: 50 or 60 Hz
   0.5 would represent 50%; 25 or 30 Hz
E = 50 Hz
F = >50 Hz
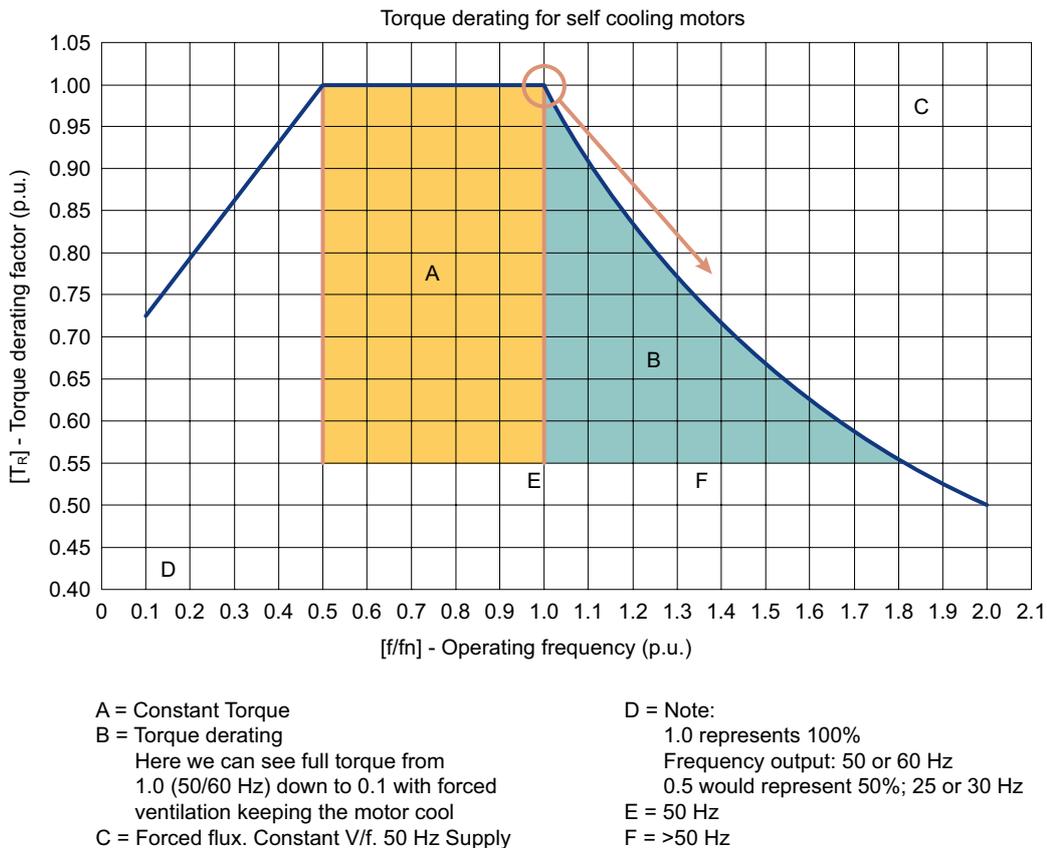
Fig. 9.11.3e - Constant torque graph for TEFC motor

In applications where running speed will be above supply frequency (50/60 Hz), motor torque will automatically be derated due to limitation available power/voltage.

Motor changes from constant torque to constant power, therefore as speed increases, available torque reduces.

The torque reduces according to the relationship
$M_{AB} / M_{NOM} = F_{NOM} / F_{AB}$

M = Torque
F = Frequency

Example - Frequency at 70 Hz

$$M_{70\,Hz} = \frac{F_{NOM}}{F_{AB}} * M_{NOM} = \frac{50\ Hz}{70\ Hz} * M_{NOM} => 71\% * M_{NOM}$$

Note:

Alfa laval pump frequency range recommendations:

Centrifugal pumps

Centrifugal pumps do not have any requirement for TEBC motors due to the operational principle of centrifugal force and the affinity laws, rendering the pump technology as variable torque machines.

Operating speeds down to 50% of the set frequency (50 or 60 Hz) is possible when adjusting an existing pump to new duty point is required.

25-60 Hz on 50 Hz motor
30-60 Hz on 60 Hz motor

Alfa Laval recommends for optimal selection all centrifugal pumps be sized as close to the best operating point as duty conditions allow which typically keeps closer.

Positive displacement pumps

Positive displacement pumps are designed to work within specific frequency ranges, ensuring optimal performance and longevity due to the operational principle of displacing fluid in fixed volumetric rotation, renders this pump technology as constant torque machines.

The frequency range depends on the pump's design, size, and intended application.

Alfa Laval recommends consideration in using TEBC (Force Blower Fan) on positive displacement pumps when motor frequency is below 20 Hz to offset the torque reduction and optimise efficiency and sustainability (duty dependent).

Alfa Laval recommends the use of TEBC (Force Blower Fan) on positive displacement pumps when motor frequency is below 10 Hz to offset the torque reduction (duty dependent).

- Self-cooled motors (TEFC) = 10 - 60 Hz (consider TEBC between 10-20 Hz operation)

- Forced blower cooled motors (TEBC) = 5 - 60 Hz

For speeds above set frequency, please consult with Alfa Laval to discuss further as this is very dependent on pump technology and intended applications.

This chapter provides cleaning guidelines for use in processes utilising CIP (Clean-In-Place) systems. Interpretations of cleanliness are given and explanations of the cleaning cycle.

10.1 CIP (Clean-In-Place)

Clean-In-Place (CIP) is a commonly established cleaning method in manufacturing operations associated within hygienic applications, such as Food & Beverage, Dairy, Home & Personal Care, Pharmaceutical & Biotechnology. CIP is designed to remove residual product and biofilms from processing lines and equipment using turbulent cleaning fluid, without the need to dismantle the equipment.

The following recommendations offer advice on how to maximise the CIP (Clean-In-Place) efficiency of the Alfa Laval ranges of centrifugal and positive displacement pumps. The guidelines incorporate references to internationally recognised cleaning detergents, velocities, temperatures, and pressures used to clean other types of flow equipment, such as valves and fittings, but have been specifically prepared to maximise the CIP effectiveness of our pumps.

10

Cleaning Guidelines

The perception of the word 'Clean' will vary from customer to customer and process to process. The four most common interpretations of 'Clean' are given below:

1. Physical Cleanliness

This is the removal of all visible dirt or contamination from a surface. This level of cleanliness is usually verified by a visual test only.

2. Chemical Cleanliness

This is defined as the removal of all visible dirt or contamination as well as microscopic residues, which are detectable by either taste or smell but not by the naked eye.

3. Bacteriological Cleanliness

This can only be achieved with the use of a disinfectant that will kill all pathogenic bacteria and the majority of other bacteria.

4. Sterility

This is the destruction of all known micro-organisms.

The following recommendations for CIP will address the first three definitions.

In most installations it is important to ensure the maximum recovery of pumped product residues from the production line at the end of each production run. Where this is a requirement, consideration should be given to mounting pumps such as the RLP (Rotary Lobe Pump) or Circumferential Piston Pump (CPP) with ports in the vertical plane to maximise drainability. This will minimise any product loss, ease the cleaning of the system and reduce the requirement to dispose or recycle the wash from the initial cleaning cycles. By maximising the recovery of product from the system both the efficiency of the production and cleaning processes will be increased.

Positive displacement pumps such as RLP or CPP are rarely used as the supply pump for CIP fluids. Centrifugal pumps are generally used during CIP for each phase of the cleaning cycle. In addition, the introduction of the Twin Screw (TS) pump, within hygienic industries, has increased flexibility in providing one pump for process media plus the use as a CIP liquid supply pump and possibility of performing CIP liquid return cycles.

For the majority of CIP cycles, it is recommended that a differential pressure of 2-3 bar is developed across any pump in a system not being utilised as CIP supply unit to promote efficient cleaning, whilst it is rotating at its normal operating speed. In many cases a valve is employed in the discharge line of the system to create the differential pressure across the pump(s) and a by-pass loop installed around said pump(s) to divert any excess of CIP liquid that the pump is unable to transfer. The valve(s) setting may be fluctuated during the CIP cycle to promote pressure/flow variations that may enhance the cleaning process.

During the CIP cycle there must always be sufficient flow of cleaning fluid being delivered by the CIP pump to make sure that the centrifugal or positive displacement pump(s) are neither starved of liquid at the inlet due to own flow capability, or over pressurised at inlet due to its tendency to act as a restriction if it is unable to transfer the full flow of the fluid being delivered.

Internationally accepted protocol for CIP suggests that during all phases of the CIP cycle a pipeline velocity of between 1.5 m/sec and 3.0 m/sec is required. Velocities within this range have proven to provide effective cleaning of Alfa Laval pumps, although as a rule, the higher the velocity the greater the cleaning effect.

Generally, the most effective cleaning processes incorporate five stages:

1. An initial rinse of clean, cold water (Pre-rinse)

2. Rinsing with an alkaline detergent (Caustic wash)

3. Intermediate rinse with cold water

4. Rinsing with an acidic disinfectant (Acid wash)

5. Final rinse with clean cold water

The cycle times, temperatures, cleaning mediums and concentrations of the detergents used will all influence the effectiveness of the cleaning cycle and care must be taken when defining these to ensure that they are suitable for use with the product being pumped. Of equal importance is the chemical compatibility between the cleaning detergents and the product wetted materials in the pump head. It is also crucial to ensure for pumps, that the maximum temperature profile is more than the intended CIP cycles(s) range. Consideration should also be given to the disposal or recycling of used cleaning liquids and the potential requirement for handling concentrated detergents.

Specialist suppliers should make the final selection of cleaning detergents/disinfectants.

Within these guidelines a typical cleaning cycle would be as follows:

1. Rinse with clean water (potable plant, deionized water) at ambient temperature to remove any remaining residue. 10-15 minutes are usually sufficient, but this will depend on the condition and volume of the residue to be removed.

This is a very important step to monitor as a well-executed pre-rinse will ensure the rest of the wash cycle is predictable and repeatable.

2. Rinse with an alkaline detergent, typically a 2.5% solution of Caustic Soda (NaOH) at between 70° C to 95° C (158° F to 203° F) for a period of 20-30 minutes would be used. It is common to add a wetting agent (surfactant) to lower the surface tension of the detergent to aid its cleansing ability. This phase of the cleaning cycle should dissolve and remove organic matter such as fats and proteins.

3. Intermediate rinse with clean water at ambient temperature for a period of 5-10 minutes. This phase should remove any residual detergents.

4. Rinse with an acidic disinfectant, typically a 2.5% solution of Nitric Acid (HNO3) at ambient temperature for a period of 10-15 minutes would be used. This phase of the cleaning cycle should remove proteins, mineral salts, lime, and other deposits.

5. Final rinse with clean water at ambient temperature for a period of 10-15 minutes or until all traces of the cleaning fluid have been removed.

Note:
In many systems, the final rinse water may be recovered and reused as the pre-rinse solution for the next cleaning cycle. The residual heat and chemicals it retains from the final rinse will help make the next pre-rinse more effective and economical.

During the CIP cycles it is important that the required concentration of cleaning detergents is maintained consistently. A significant increase in concentration could cause damage to pumps and other components in the system. A significant decrease in concentration could impact the detergents cleaning efficiency. A facility for monitoring and adjusting the detergent concentration should be considered.

Cautionary Notes:

1. Pumps and other equipment installed in CIP systems have components within them that will expand and contract at different rates. Care should be taken not to subject them to rapid temperature cycling.

2. Products containing particulate such as fibre and seeds have to be evaluated carefully and on an individual basis, as the nature of these will provide an increased cleaning challenge. These types of products may typically require increased cleaning cycle times and/or increased velocities and pressures during the cleaning cycle.

3. CIP detergent liquids and the elevated temperatures typically used for CIP processes can cause a potential health risk. Always adhere to site Health and Safety regulations.

4. Always store and dispose of cleaning agents in accordance with site Health and Safety regulations.

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 after CIP cycles.

The sterilisation can be carried out by means of chemicals, hot water, or steam. As an example, in the dairy industry the sterilisation temperature is approximately 145° C (293° F).

It is common practice for pumps to remain stationary when live steam is present during SIP to ensure no distortion within seals due to the gaseous state of the steam and the steam flow distribution. The temperature rise of the seal parts would not be even and therefore needs to be suitably cooled before pump can be restarted at low speeds (<100 rpm) to remove any trapped condensate build-up after SIP cycle.

Operation of pump is possible during steam cycles at low speeds (<100 rpm) if there is a quench/barrier/buffer present within seal housing to provide a lubricating fluid film.

Please note, a small volume of seal leakage is typically present on initial start-up as the pump seal faces rotate to realign back into place. This is temporary and not a cause of concern as long as pump was not in operation during any phase where liquid was not present to lubricate seal face.

If seal leakage remains after an intermediate period of operation, please contact Alfa Laval for guidance.

This chapter describes some of the international standards and guidelines applicable to Alfa Laval pump ranges.

11.1 Compliance with international Standards and Guidelines

Alfa Laval pump ranges are available with documented and certified compliance within a broad spectrum of relevant international and local hygiene standards, worldwide. This assists the user to significantly reduce the engineering costs of setting up and operating standard-compliant processing plants around the world.

11

Compliance

Authorized to carry
the 3A symbol

Please find below some examples of regulations, standards, and guidelines applicable to our products used in hygienic applications.

More information can be found in Instruction Manuals at alfalaval.com.

For special requests please contact your local Alfa Laval organization.

The mission of 3-A SSI is to enhance product safety for consumers of food, beverages, and pharmaceutical products through the development and use of 3-A Sanitary Standards and 3-A Accepted Practices. The 3-A symbol is a registered mark used to identify equipment that meets 3-A Sanitary Standards for design and fabrication.

ATEX-directive is the popular name for the European Directive 2014/34/EU setting the rules for equipment and protective systems intended for use in potentially explosive atmospheres.

Compliance to the Regulation (EC) No. 1935/2004.

The framework regulation (EC) No. 1935/2004 regulates food contact materials and articles within EU. It includes several requirements for materials and articles intended to come into contact with food to ensure material safety. The glass and fork symbol may be used to indicate that the relevant requirements stated in (EC) No. 1935/2004 are met.

CE marking is a mandatory conformity mark for products placed on the market in the European Economic Area (EEA). With the CE marking on a product the manufacturer ensures that the product conforms with the essential requirements of the applicable EC directives. The letters "CE" stand for "Conformité Européenne" ("European Conformity").

UKCA marking is a mandatory conformity mark for products placed on the market in Great Britain (England, Scotland, and Wales). With the UKCA marking the manufacturer ensures that the product conforms with the relevant requirements of the applicable legislations.

Within United States, requirements for food contact materials and articles are specified by the Food and Drug Administration (FDA) and are regulated under the Code of Federal Regulations, Title 21 "Food and drugs", Parts 170-199 "Food for human consumption".

The United States Pharmacopeia (USP) standards, chapter 87 and 88, and International Organization for Standardization (ISO) standard 10993, sections 5, 6, 10 and 11, specifies requirements to ensure biocompatibility of product contact parts intended to be used in pharma applications.

The American Society of Mechanical Engineers Bioprocessing Equipment (ASME BPE) is the Bioprocess Equipment group of the ASME that provides engineers and quality control professionals a measurable way to specify and purchase equipment for the Biotechnology, Pharmaceutical and Personal Care Products industries.

Fig. 11a CE

Fig. 11b 3-A

CE

The CE marking is to demonstrate to interested parties that goods or equipment with this mark comply with the appropriate directives of the European Community (Fig. 11a). The appropriate directives are those that are concerned with the design and manufacture of goods or equipment. Directives are intended to facilitate a Single Market in the European Union. With emerging European standardisation, conflicting national standards will eventually tend to disappear, as all EU member states will work to the same standard, with a few exceptions. Some national differences cannot be harmonised. In Europe many different languages are spoken, and some parts are prone to earthquakes, high winds, heavy snow and extremes of cold and heat. It is often uneconomic to design equipment that will withstand all these conditions.

All Alfa Laval pump ranges are CE marked and conform to the machinery directive 89/392/EEC as amended by 91/368/EEC, 93/44/EEC and 93/68/EEC and other relevant directives i.e., 'Electrical Equipment Low Voltage Directive 73/23/EEC' and 'Electromagnetic Compatibility Directive 89/336/EEC'.

Other applicable standards/specifications which Alfa Laval pump ranges comply to are as follows:

- EN 292 Parts 1 and 2: 1995 Safety of Machinery - Basic concepts, general principles for design

- EN 294: 1996 Safety distances to prevent danger zones being reached by the upper limbs

- EN 60204 Part 1: 2018 Safety of Machinery
  - Electrical equipment of machines
  - Specification for general requirements

- ISO 9001: 2015 Quality Management System

- ISO 14001: Environmental Management System

3-A

This standard has the purpose of establishing and documenting the material, fabrication, and installation (where appropriate) requirements for the engineering design and technical construction files for all products, assemblies, and sub-assemblies supplied by the manufacturer (Fig. 11b). The manufacturer has to be in compliance with the sanitary criteria found in 3-A Sanitary Standards or 3-A Accepted Practices. The 3-A Sanitary Standards and 3-A Accepted Practices are applied as suitable sanitary criteria for dairy and food processing equipment. 3A is subject to 3rd party validation according to 3A 02/10 guidelines.

The Alfa Laval pump ranges conform to this 3-A standard for certain configurations this can be selected and determined in the AnyTime Configurator.

Fig. 11c FDA

FDA

The Food and Drug Administration (FDA) in the US is the enforcement agency of the United States Government for food, drug and cosmetics manufacturing. It is responsible for new material approvals, plant inspections and material recalls (Fig. 11c). In the US, the 'Food, Drug and Cosmetic Act' requires food, drug and cosmetic manufacturers to prove that their products are safe. The FDA's primary purpose is to protect the public by enforcing this Act.

The FDA can:

- Approve plants for manufacturing

- Inspect plants at random

- Write general guidelines for good manufacturing processes

- Write specific criteria for materials in product contact

- Have certain expectations regarding design practices

The FDA cannot:

- Approve equipment outside of a particular use within a specific system

- Approve materials for use in pharmaceutical systems

- Write specific engineering or design requirements for systems

Fig. 11d USDA

USDA

The United States Department of Agriculture (USDA) is one of three Federal Agencies, along with the Environmental Protection Agency (EPA) and the U.S. Food and Drug Administration (FDA), primarily responsible for regulating biotechnology in the United States (Fig. 11d).

Products are regulated according to their intended use, with some products being regulated under more than one agency.

Agricultural biotechnology is a collection of scientific techniques, including genetic engineering, that are used to create, improve, or modify plants, animals, and micro-organisms. Using conventional techniques, such as selective breeding, scientists have been working to improve plants and animals for human benefit for hundreds of years. Modern techniques now enable scientists to move genes (and therefore desirable traits) in ways they could not before - and with greater ease and precision.

The Federal government has a well-coordinated system to ensure that new agricultural biotechnology products are safe for the environment and to animal and human health. While these agencies act independently, they have a close working relationship.

- USDA's Animal and Plant Health Inspection Service (APHIS) is responsible for protecting American agriculture against pests and diseases. The agency regulates the field testing of genetically engineered plants and certain micro-organisms. APHIS also approves and licenses veterinary biological substances, including animal vaccines that may be the product of biotechnology

- USDA's Food Safety and Inspection Service (FSIS) ensures the safety of meat and poultry consumed as food

- The Department of Health and Human Service's Food and Drug Administration (FDA) governs the safety and labelling of drugs and the nation's food and feed supply, excluding meat and poultry

- The Environmental Protection Agency (EPA) ensures the safety and safe use of pesticidal and herbicidal substances in the environment and for certain industrial uses of microbes in the environment

- The Department of Health and Human Service's National Institutes of Health have developed guidelines for the laboratory use of genetically engineered organisms. While these guidelines are generally voluntary, they are mandatory for any research conducted under Federal grants and they are widely followed by academic and industrial scientists around the world

USP (United States Pharmacopeia) class VI

USP (United States Pharmacopeia) class VI refers to a set of standards established by the USP that govern the biocompatibility and safety of materials used in the manufacturing of pharmaceutical and medical devices. Chapter 87 and Chapter 88 of USP class VI specifically address biological reactivity testing of in vitro and in vivo materials, respectively.

Chapter 87 focuses on the in vitro biological reactivity testing of materials used in medical devices. It outlines various tests and criteria to assess the potential adverse biological reactions caused by these materials. The chapter provides guidelines for evaluating the cytotoxicity (cellular toxicity), sensitization (allergic reactions), irritation, and other potential harmful effects of materials when they come into contact with living tissues.

On the other hand, Chapter 88 addresses the in vivo biological reactivity testing of materials. It covers the testing of materials using animal models to assess their potential adverse effects when implanted or introduced into living organisms. This includes tests for acute systemic toxicity, subchronic toxicity, and implantation effects.

Compliance with USP class VI standards, including chapters 87 and 88, is crucial for ensuring the safety and compatibility of materials used in medical devices and pharmaceutical products. Manufacturers and suppliers are expected to adhere to these standards to demonstrate that their materials are biocompatible and pose minimal risks to patients and users. By following these guidelines, the industry can maintain high standards of quality and safety in the design and production of medical devices and pharmaceutical products.

FDA/USP Class VI – ISO 10993

For all Alfa Laval Ultra Pure pumps are the product wetted Elastomers conforming with FDA USP Class VI.

- All elastomer types conform to USP class VI chapter 87 and 88

- Extractable testing undertaken to 121° C (250° F)

EN 10204 3.1

With the stringent demands of hygiene within new food and pharmaceutical plants being built, material traceability of equipment supplied is increasingly important. The EN 10204 standard defines the different types of inspection documents required for metallic products. In particular, 3.1 of this standard refers to inspection documents being prepared at each stage of manufacture and supervised tests performed by the quality department of the manufacture. The material certification has reference all the way back to original steel mill.

EN 10204 2.2

This standard defines documents supplied to the purchaser, in accordance with the order requirements, for the supply of metallic products such as pumps. This takes the form of a certificate of conformity and can be applied to all Alfa Laval pump ranges.

Fig. 11e EHEDG

EHEDG

EHEDG stands for European Hygienic Engineering and Design Group (Fig. 11e). It is a consortium of equipment manufacturers, food industries, research institutions, and public health authorities that promotes hygienic engineering and design principles in the food and beverage industry.

The primary goal of EHEDG is to provide guidelines and standards for the design, construction, and installation of food processing equipment and facilities to ensure hygienic production processes. By promoting hygienic engineering practices, EHEDG aims to improve food safety, minimize contamination risks, and enhance the overall quality of food products.

EHEDG develops and publishes guidelines, recommendations, and best practices for various aspects of food processing, including equipment design, materials selection, cleaning and disinfection procedures, and facility layout. These guidelines are based on scientific principles and are continuously updated to incorporate the latest advancements in food safety and hygiene.

Some key principles emphasized by EHEDG include:

- Cleanability: Equipment should be designed in a way that allows effective and efficient cleaning. Smooth surfaces, minimal joints and crevices, and the use of appropriate materials are crucial for easy cleaning and prevention of bacterial growth

- Hygienic design: Equipment should be designed to minimize the risk of product contamination. This involves considerations such as preventing the accumulation of product residues, avoiding dead spaces where bacteria can proliferate, and ensuring proper drainage of liquids

- Material selection: EHEDG provides guidance on suitable materials for food contact surfaces, taking into account factors such as resistance to corrosion, ease of cleaning, and compatibility with different food products

- Validation and verification: EHEDG emphasizes the importance of validating and verifying the hygienic performance of equipment and processes. This involves conducting tests and assessments to ensure that the equipment meets the desired hygiene standards and that cleaning and disinfection procedures are effective

Fig. 11f Ex

EHEDG and 3A

One significant difference between 3A SSI and EHEDG lies in their geographical focus. While 3A SSI primarily caters to the North American market, EHEDG has a broader international presence and addresses the needs of European and global industries. This difference in regional scope leads to variations in regulatory requirements, manufacturing practices, and market acceptance.

3-A certification requires only a theoretical review of design requirements while EHEDG certification reviews the design both theoretically and practically (using a standardised hygiene test)

Despite these differences, both 3A SSI and EHEDG share the common goal of ensuring hygienic equipment design and operation. Manufacturers and industry professionals consider relevant standards and guidelines from both organizations, along with other applicable local regulations, to achieve the highest level of hygiene and safety in their processes and products.

The consensus is 3A SSI and EHEDG are collaborating more and driving the future hygienic design standards and look to remove national or regional standards as non-trariff barriers for trade.

ATEX

ATEX stands for "ATmosphères EXplosibles," which translates to "Explosive Atmospheres" in English (Fig. 11f). ATEX is a European Union directive that outlines the requirements for equipment and protective systems intended for use in potentially explosive atmospheres.

For more information on ATEX see section 9.8).
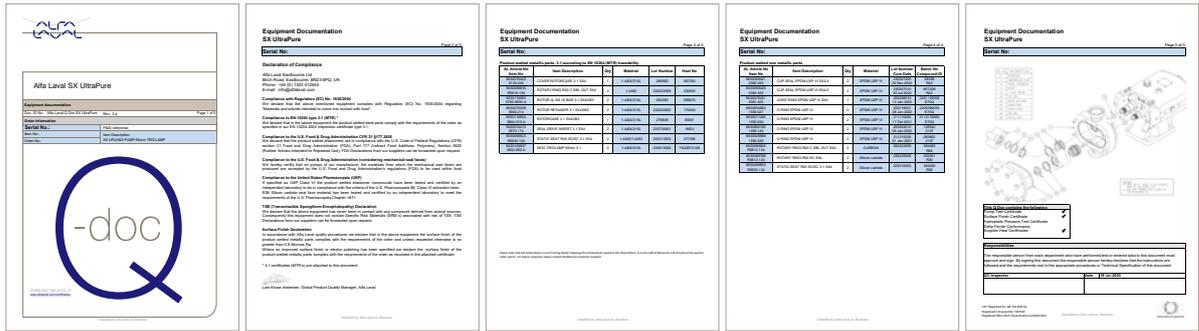
Fig. 11g Example of standard Q-Doc for SX UltraPure

Pharma documentation:

Alfa Laval Q-doc is our comprehensive documentation package for our UltraPure equipment (Fig. 11g). Based on Good Documentation Practice (GDP), Q-doc documents every aspect from raw material to delivered equipment. With full transparency of sourcing, production and supply chains it is a simple matter to trace even the slightest change in material or manufacturing procedures – even when it comes to spare parts.
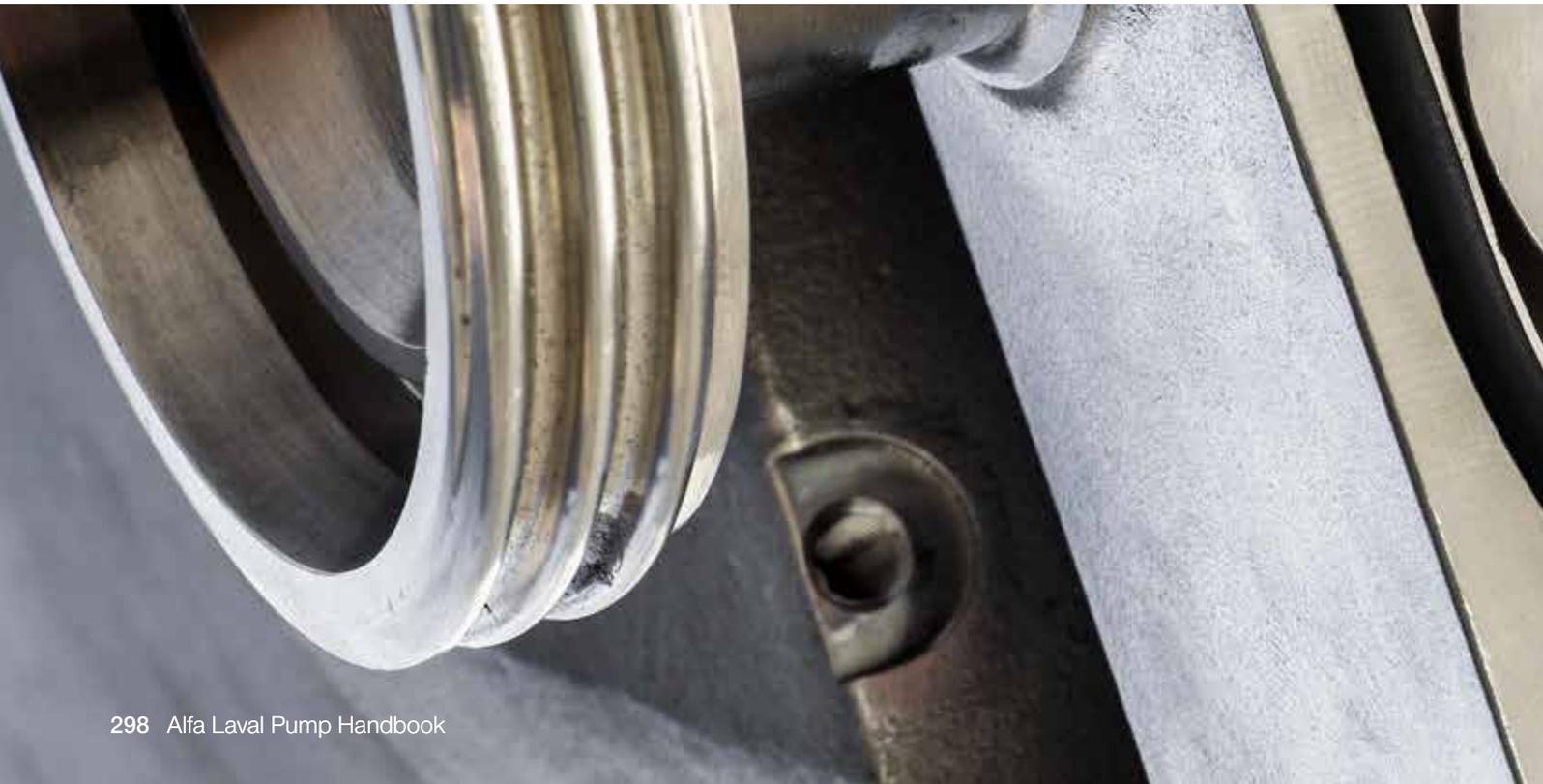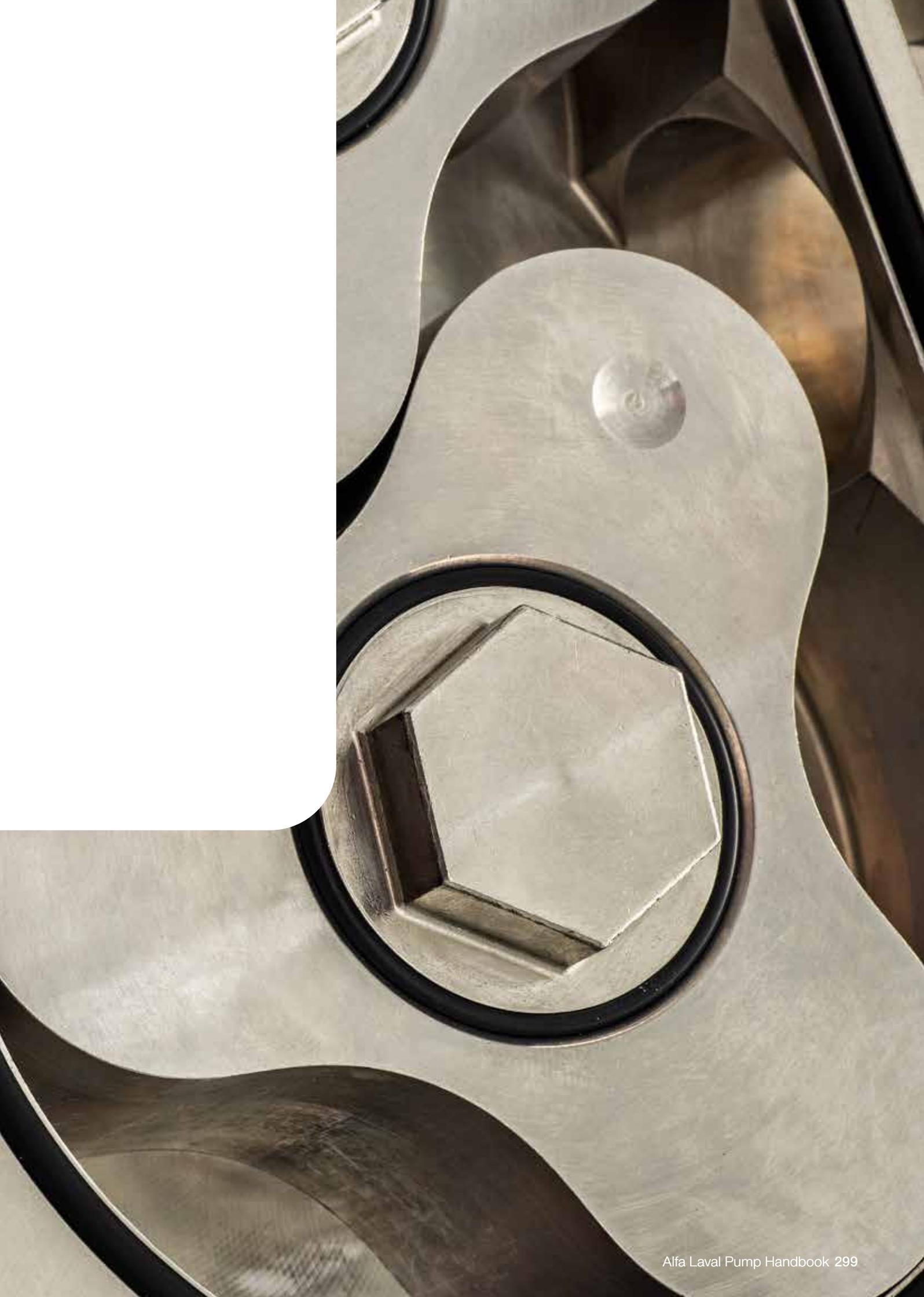
This chapter covers guidelines relating to pump installation, system design and pipework layout.

12.1 General

To ensure optimum pump operation it is important that any pump unit to be used is installed to the best engineering practice.

12

Installation Guide
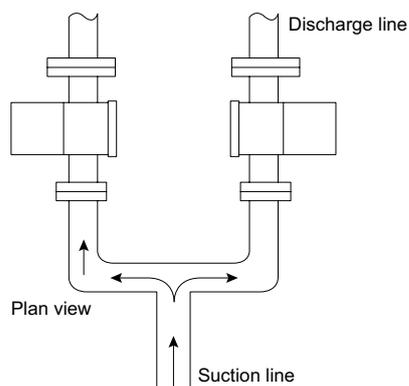
Fig. 12.1.1a Avoid common suction lines

12.1.1 System Design

When designing a pumping system, the following points should be taken into consideration:

- Confirm the Net Positive Suction Head (NPSH) available from the system exceeds the NPSH required by the pump, as this is crucial for ensuring the smooth operation of the pump and preventing cavitation

- Take care with designing suction lifts and manifold/common suction lines for two positive displacement pumps running in parallel, as this may cause vibration or cavitation (see Fig. 12.1.1a). It is important to note that each pump have their own NPSH required value so NPSH available on a shared suction line requires sufficient availability on both pumps. Recommendation is to take both NPSHr values for a total and add an additional 1 m lc margin to NPSHa for suitable operation

- Protect the pump against blockage from hard solid objects e.g., nuts, bolts etc. Also protect the pump from accidental operation against a closed valve by using relief valves, pressure switches and current limiting/tripping devices

- Fit suction and discharge pressure monitor points for diagnostic purposes

- Fit valves, if two pumps are to be used on manifold/common discharge lines

- Make the necessary piping arrangements as per individual requirements of the pump specification as detailed in the operating manual for flushing

- Allow at least 1 m for pump access/maintenance all around the unit

- Do not subject pumps to rapid temperature changes, as pump seizure can result from thermal shock

12.1.2 Pipework

Both the suction and discharge piping should be aligned and independently supported so that the pipework does not create any undesired strain against the connections of the pump. Pipe strain is a leading factor in causing shaft misalignment.

Pipework support should be balanced and secured so that when fluid is introduced, there is no movement that would cause pipe strain on the pump.

The maximum allowable forces and moments for an Alfa Laval pump can be found in the operating manual.

For optimal pipework design, provision needs to be made to the pipework to ensure unwanted effects such as hydraulic noise, vibration and cavitation are minimised.

The following should be taken into consideration

- Have short, wide inlet pipework to reduce friction losses in the pipework thereby improving the NPSH available

- Avoid bends, tees, and any restrictions close to either suction or discharge side of pump. Use long radius bends wherever possible

- Alfa Laval recommends x10 the pipe diameter of straight pipework before and after the pump to allow for steady flow conditions that reduce turbulence and unwanted vibration

- Provide isolating valves on each side of the pump when necessary

- Keep pipework horizontal where applicable to reduce air locks. Include eccentric reducers on suction lines and a positive gradient slope feeding to pump inlet being ideal

12.1.3 Weight

The weight of the pump and drive unit should be considered for lifting gear requirements.

Consult the pumping unit's operating manual on best methods for handling.

12.1.4 Electrical Supply

Ensure that there is an adequate electrical supply to the pump drive unit. This supply should be compatible with the electric motor selected and managed by a qualified electrician for installation purposes.
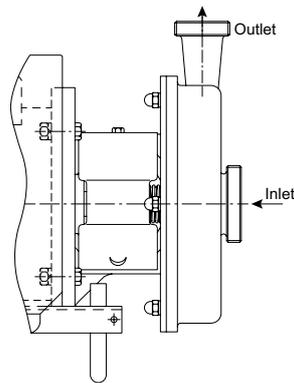
Fig. 12.2.1a Correct direction of flow
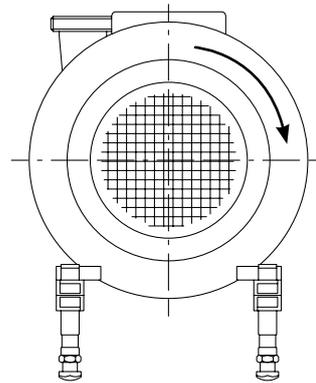
Fig. 12.2.1b Pump with impeller screw fitted

12.2 Flow Direction

12.2.1 Centrifugal Pumps

A centrifugal pump should never be operated in the wrong direction of rotation with fluid within the pump (Fig. 12.2.1a). It is possible to check this in two ways as follows:

1. Pump with impeller screw fitted

- Start and stop the motor momentarily (without fluid in the pump)

- Ensure that the direction of rotation of the motor fan is clockwise as viewed from the rear end of the motor (Fig. 12.2.1b)
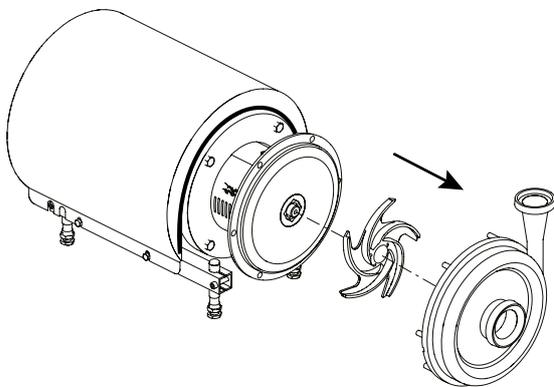
Fig. 12.2.1c Pump without impeller screw fitted

Stub shaft
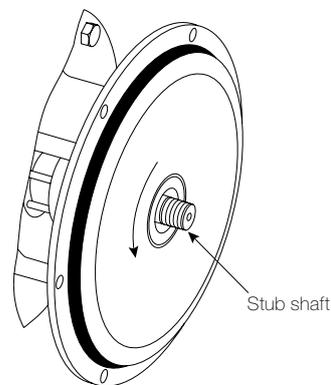
Fig. 12.2.1d Pump without impeller

2. Pump without impeller screw fitted

- With this method the impeller should always be removed before checking the direction of rotation (Fig. 12.2.1c)

- The pump should never be started if the impeller is fitted, and the pump casing has been removed

- If dealing with a LKH Prime with an airscrew, always remember to remove the air screw and impeller before checking the direction of rotation

- Start and stop the motor momentarily

- Ensure that the direction of rotation of the stub shaft is anti-clockwise as viewed from the pump inlet (Fig. 12.2.1d)
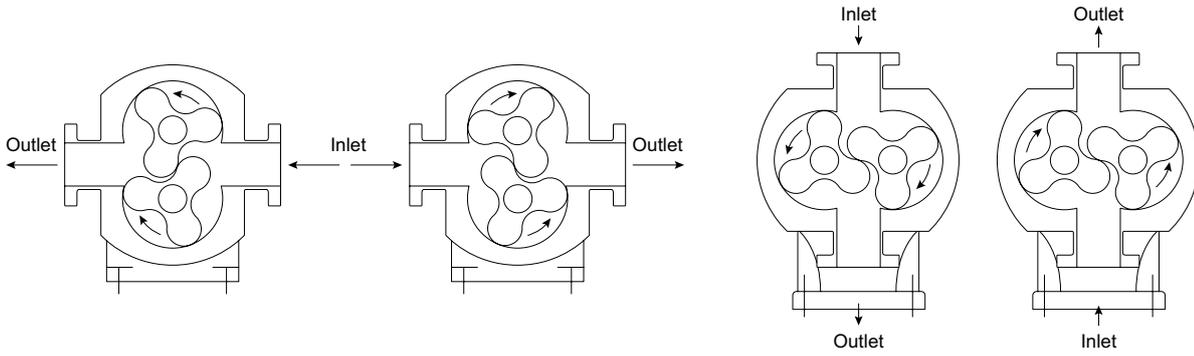
Fig. 12.2.2a Flow direction

12.2.2 Rotary Lobe & Circumferential Piston Pumps

The direction of flow is dictated by the direction of drive shaft rotation. Reversing the direction of rotation will reverse the flow direction (Fig. 12.2.2a).

A: Suction
B: Discharge

A: Suction
B: Discharge
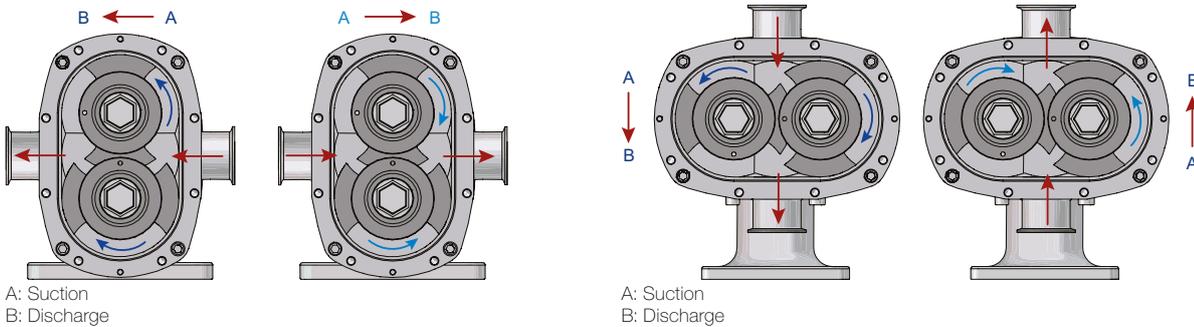
Both Rotary Lobe and Circumferential Piston pumps are capable of full bi-directional flow without configuration modification.
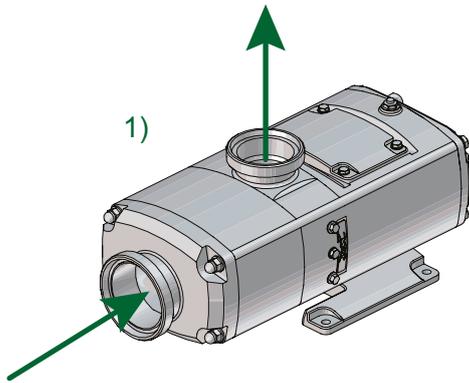
Fig: 12.2.3a 1) Standard flow direction - Front: Inlet/Top: Outlet
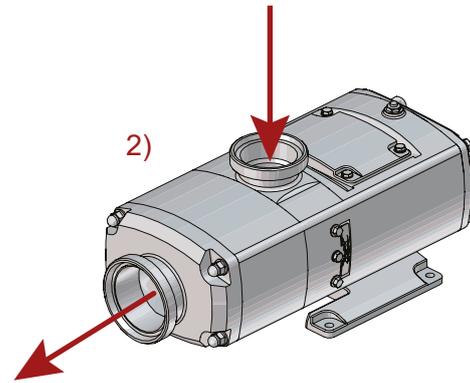
2) Reversed flow direction – Top: Inlet/Front: Outlet

12.2.3 Twin Screw Pumps

The direction of flow is dictated by the direction of drive shaft rotation. Reversing the direction of rotation will reverse the flow direction. It is important to note, the twin screw pump will have been built for a specific flow direction at time of order. This is highlighted by the arrow on the pump casing and the drive shaft as indication of which direction flow will occur via correct rotation (Fig. 12.2.3a).

Note:

The Alfa Laval Twin Screw pump may be operated in reverse to that originally intended but differential pressure limitations in accordance to pump configuration must be considered – Refer to instruction manual for further details.
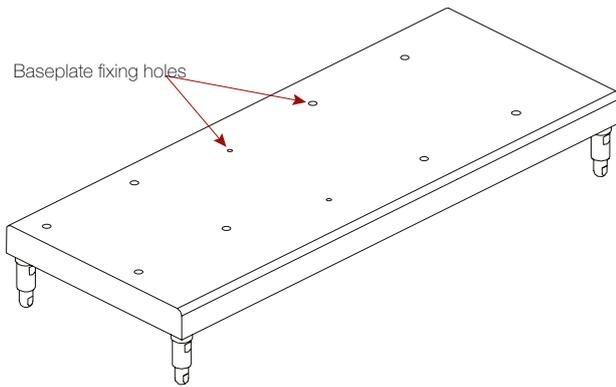
Fig. 12.3b Baseplate fixing

12.3 Baseplate Foundations

Positive Displacement Pumps

Success to pump longevity begins with a solid foundation. In general, a pump should be mounted on a strong baseplate and supported by a strong concrete foundation to absorb any vibration, strain or shock and form a permanent rigid support for the baseplate and pumping unit.

Positive displacement pumps when supplied with a drive unit are normally mounted on a baseplate. Alfa Laval standard baseplates have pre-drilled fixing holes to accept base retaining bolts (Fig. 12.3b).

Methods of anchoring the baseplate to the foundation are varied, they can be studs embedded in the concrete either at the pouring stage as shown below, or by use of epoxy type grouts. Alternatively mechanical fixings can be used.
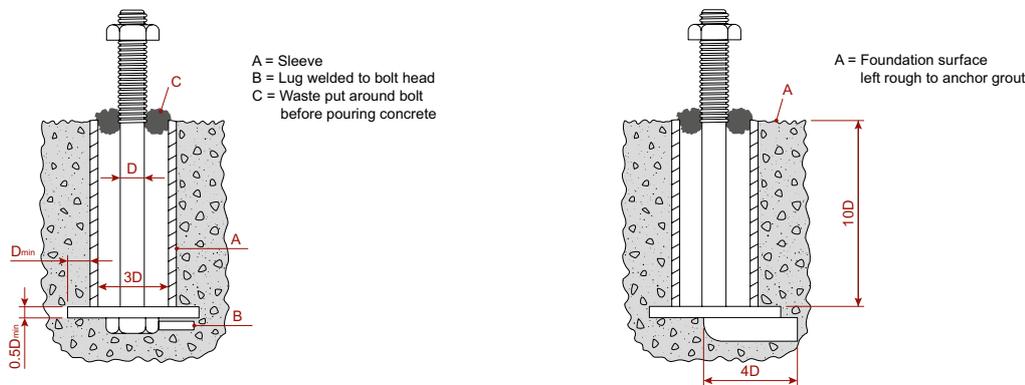
Fig. 12.3c Foundations

Optimal foundations should be approximately 150 mm longer and wider than the provided baseplate. In addition, the depth of the foundation should be proportional to the size of the complete pump unit with typical standard value being at a depth of x10 the foundation bolt diameter.

The drawing (Fig. 12.3b) above shows two typical methods for foundation bolt retaining. The sleeve allows for 'slight' lateral movement of the bolts after the foundation is poured. Rag or wastepaper can be used to prevent the concrete from entering the sleeve while the foundation is poured. A minimum of 14 days is normally required to allow the curing of the concrete prior to pump unit installation.
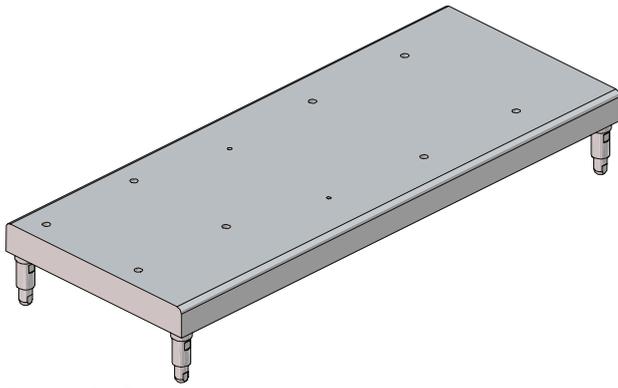
Fig. 12.3d Ball Foot baseplate

Ball Foot Baseplate with Adjustable Feet

The pumping unit can be supplied with an optional ball foot baseplate with height adjustable feet (Fig. 12.3d).

When used:

- Ensure the floor is level and can support the weight of the complete unit

- Ensure the unit is supported by all feet equally

Note:
These two points on suitable foundation are also applicable to centrifugal pumps when installed.

The 3A standard on positive displacement pumps requires a minimum clearance between the lowest part of the base, pump, motor, or drive and for the floor to be no less than 100 mm (4 in).
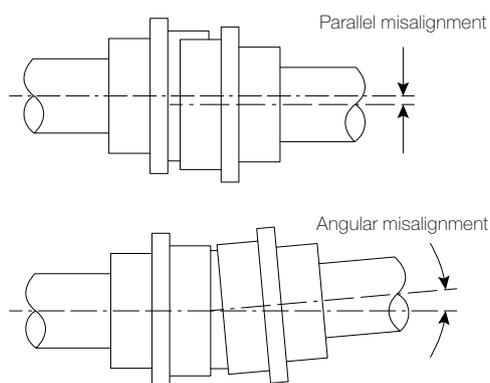
Fig. 12.4a Parallel and angular misalignment

12.4 Coupling Alignment
(Positive Displacement Pumps)

Before the pump unit is installed it is important to ensure that the mounting surface is flat to avoid distortion of the baseplate. This will cause pump/motor shaft misalignment and pump/motor unit damage (Fig. 12.4a).

Once the baseplate has been secured, the pump shaft to motor shaft coupling alignment should be checked and adjusted where necessary as it is possible for assembled units to shift out of tolerance during transit.

Coupling alignment is achieved by checking the maximum angular and parallel allowable misalignments for the couplings as stated by the coupling manufacturers that can be located within operating manuals.

Misalignment can lead to premature seal and bearing failure with increased noise output. Misalignment is often caused by the following:

* Improper mounting/shimming of the equipment when fitting/installing

* Pipe strain caused by improper alignment to the pump's fluid connections

* Pipe strain caused by improper support of the pipe
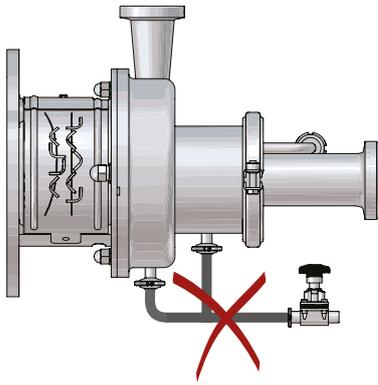
Fig. 12.5a Incorrect use
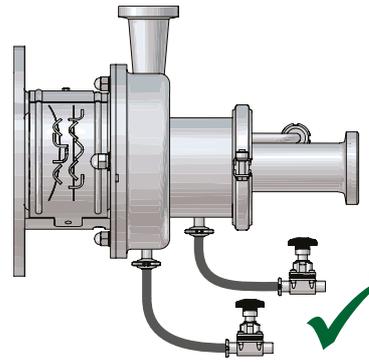
Fig. 12.5b Correct use

12.5 Considerations for LKH Prime Centrifugal Pump

Always use two drain valves

If the LKH Prime Pump is fitted with the drain option; never short circuit the drain connections as this will reduce the air release capacity (Fig. 12.5a).

To ensure optimal function of the self-priming capacity, LKH Prime must be installed in such a way that it ensures liquid is in the pump on start-up e.g., with a swan neck design as illustrated (Fig. 12.5b).

Note:
Max running time when releasing air should not exceed 15 min.
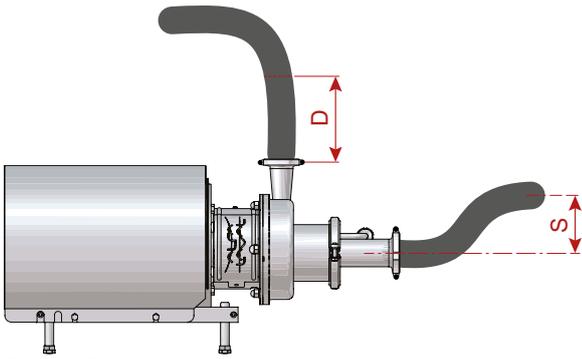
Fig. 12.5c Recommended start-up outlay

	^{S} min	^{D} min
LKH Prime 10	200 mm	1.5 m
LKH Prime 20		2" pipe
LKH Prime 40	200 mm	1.5 m 3" pipe or 2 m 2" pipe

Table 12.5.a

Installation guidelines

1. Suction considerations

- Ensure the suction line is designed so the pump is liquid filled at start up, e.g. swan neck (Fig. 12.5c)

- Design suction line with slope down toward the pump to avoid air entrapments

- Ensure NPSHa > NPSHr under all duty conditions including all temperatures

- Controlled start/stop of pump e.g. Level Switch (LS)

- Do not start the LKH Prime before tank bottom is liquid filled

- Stop the LKH Prime during phase changes

1. Suction line
2. LKH Prime Pump
3. Discharge line
4. Check valve
5. Level switch
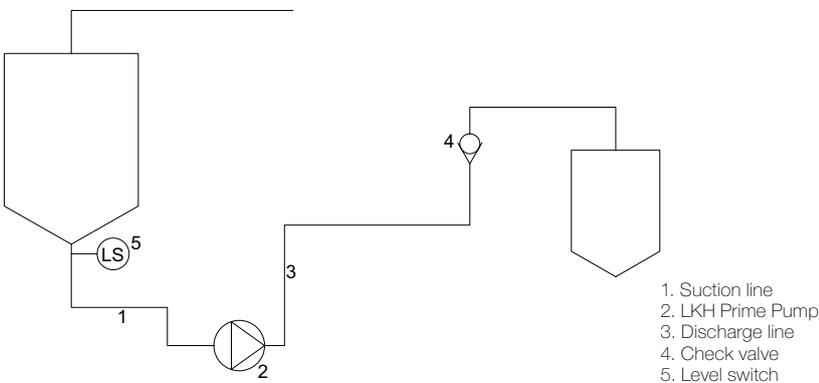
Fig 12.5d Installation example

2. General considerations

- Minimum speed for effective air evacuation is 2800 RPM

- Air evacuation ability improves with higher speed (Maximum speed 3600 RPM)

- The LKH Prime must be sized for the specific duty point

3. Discharge considerations

- Place check valve as far away from the pump as possible

- Replace check valve with automated valve, if possible

Note:
The LKH Prime is NOT a one-to-one replacement of the MR liquid-ring pump.

12.6 Pre-start Checklist

- Check the pipework system has been purged to remove debris

- Check all obstructions have been removed from pipework and pump

- Check pump connections and pipework joints are tight

- Check lubrication levels are correct

- Check seal flushing is connected if applicable

- Check all safety guards are in place

- Check that inlet and outlet valves are open

Note:
Where available, commissioning plugs should be used to reduce risk of any component damage because of debris becoming trapped between rotating element and casing.

12.6.1 Fastenings

If pump is to be disassembled for any reason it is imperative that upon assembly all fastenings are tightened to the recommended torque values as shown in the instruction manual.

This chapter offers possible causes and solutions to most common problems found in pump installation and operation.

13.1 General

Pumps are critical components in many industrial processes and applications. They are used to transfer fluids, gases, or other substances from one location to another. Problems with pumping equipment cause not only inconvenience, but also contribute to loss of production. An efficient operation depends on trouble-free pumping. Effective troubleshooting of pumps is crucial for minimising downtime and preventing further damage to equipment, ultimately saving time and money.

Pumps are likely to be the most vulnerable components and when trouble can arise, the symptoms frequently show the pump to be at fault regardless of what may be wrong. The problem is usually caused by inadequate control of the pumped fluid or a change in operating requirements of which the system or pump is not capable of handling or a component malfunction.

13

Troubleshooting

Diagnosis of problems will be greatly assisted by having pressure gauges fitted to both pump inlet and outlet.

Before starting to correctly identify the problem it is important to gather as much information relating to the process as follows:

- Reconfirm original duty conditions pump was sized towards

- What has changed in the process since operation was last satisfactory i.e., pressure, temperature, fluid viscosity etc.

- Was the system undergoing routine maintenance

- Were any new or repaired components omitted to be fitted

- When was the pump last serviced

- What was the appearance and condition of the pump internal components

- How long did the pump operate before the problem

- Any changes in pump noise or vibration

The most common problems found are generally as follows and explained in section 13.2:

- Loss of flow

- Loss of suction

- Low discharge pressure

- Excessive noise or vibration

- Excessive power usage

- Rapid pump wear

- Seal leakage

13.2 Common Problems

13.2.1 Loss of Flow

A simple cause of this could be incorrect direction of shaft rotation on a centrifugal pump, which although obvious is often overlooked.

Loss of flow can be caused by excessive discharge pressure and/or by a change in fluid viscosity.

In general terms:

*   For a positive displacement pump, if the viscosity is significantly reduced, the pump's rated flow will be reduced, more so for higher pressure operation

*   For a centrifugal pump if the viscosity is increased, the pump's rated flow will be decreased

13.2.2 Loss of Suction

Loss of suction can be minor, causing little, short-term damage or sufficiently major to catastrophic damage. Loss of suction means fluid is not reaching the pumping elements or not reaching them at a sufficiently high pressure to keep the fluid being pumped in a fluid state. Loss of suction can be interpreted as the inability to prime, cavitation or a gas content problem.

Positive displacement pumps can be classed as 'self-priming'. This means that within limits, unique to each technology, they are capable of evacuating (pumping) a modest amount of air from the suction side of the pump to the discharge side of the pump. Filling the inlet system with fluid or at least filling the pump (wetted pumping elements) will make a considerable improvement in the pump's priming capability.

The LKH Prime centrifugal pump range is specially designed to be self-priming with its air screw design.

Cavitation is caused by insufficient system inlet pressure to the pump. This can be caused by an inlet system restriction, excessive fluid viscosity or excessive pump speed. Inlet restrictions can include dirty or clogged inlet strainers, debris floating in the fluid supply that covers the inlet piping intake, or rags. If the fluid is cooler than design temperature, its viscosity may be too high causing excessive friction (pressure loss) in the inlet piping system. Cavitation is frequently accompanied by noise, vibration and significant increase in discharge pressure pulsation. If a pump is allowed to cavitate over long periods this will cause damage to the pump head components. The surface of these components are typically perforated and pitted.

Gas in the inlet pipework has the same impact on pump operation and creates the same symptoms as cavitation. This can occur under other circumstances such as a pump operating at an inlet pressure below local atmospheric pressure. In this instance it is quite likely that air is being drawn into the pipework through a loose pipe connection or pump casing joint, leaking inlet valve stem, defective or otherwise damaged joint gasket in the pipework system. In recirculating systems, such as a lubrication system where the fluid pumped is continuously returned to a supply source or tank, if the tank and return lines are not adequately designed, located and sized, air is easily entrained in the oil and immediately picked up by the pump inlet system. Be sure fluid level at its source is at or above minimum operating levels. Lines returning flow to a supply tank should terminate below minimum fluid level.

13.2.3 Low Discharge Pressure

Low discharge pressure can only be caused by loss of flow. Pump discharge pressure is caused only by the system's resistance to the flow provided by the pump. Either the pump is not providing the flow expected or the system is not offering the expected resistance to that flow and would not be a result of incorrect pump selection but rather system/process design. It is possible that flow is being restricted into the pump (cavitation), usually accompanied by noise and vibration, the pump is not producing its rated flow (pump worn or damaged), or the pump flow is bypassing rather than being delivered into the system as intended.

13.2.4 Excessive Noise or Vibration

Excessive noise and/or vibration can be a symptom of cavitation, mechanical damage to pump assembly, misalignment of drive or harmonics with other elements of the system. Cavitation is especially true if the discharge pressure is fluctuating or pulsating. Mechanical causes of noise and vibration include shaft misalignment, loose couplings, loose pump and/or driver mountings, loose pump and/or driver guards, worn or damaged driver or pump bearings or valve noise that seems to be coming from the pump. Valves, especially on the discharge side of the pump can sometimes go into a hydraulic vibration mode caused by operating pressure, flow rate and the valve design. Resetting or a change in an internal valve component is usually sufficient to solve the problem.

13.2.5 Excessive Power

Excessive power consumption can be caused by either mechanical or hydraulic problems. Mechanical causes include imminent bearing failure, pumping elements rubbing which can lead to a pump seizure and poor shaft alignments. Too high viscosity can result in the motor overloading.

- For a positive displacement pump, too high discharge pressure can cause the motor to overload

- For a centrifugal pump, too high capacity (too low discharge pressure) can cause the motor to overload

13.2.6 Rapid Pump Wear

Rapid wear of pump head components is either caused by abrasives being present in the fluid, chemical corrosion, loss of shaft support (bearing failure), or operation at a condition for which the pump is not suitable i.e., cavitation, excessively high pressure or high temperature. To avoid any abrasive foreign material entering the pump, strainers or filters should be employed wherever possible and practical. Rapid wear is sometimes not wear in the sense of a non-durable pump, but really a catastrophic pump failure that occurred very quickly. Looking at the pump's internal parts alone may not provide much help in identifying the cause, thus the importance of knowing what was occurring in the time period immediately preceding detection of the problem.

13.2.7 Seal Leakage

Mechanical seals fitted to pumps can be seen as the weakest point. Special care should be taken to ensure the correct seal configuration for the application is installed i.e., mounting attitude, seal face combination and elastomer selection.

Apart from mis-selection and poor servicing, seal leakage can be due to pump cavitation, too high discharge pressure, pumps being allowed to run dry without suitable liquid present for seal operation and unexpected solids in the fluid.

13.3 Problem Solving Table

The table shown offers probable causes and solutions to the most common problems encountered.

In the parenthesis ( ) next to the particular solution given you will find annotation relating to what pump type the solution is for.

i.e.,

- ce = Centrifugal Pump

- pd = Positive Displacement Pump

See table 13.3a on the following pages.

For further in-depth analysis into troubleshooting guidance on a particular pump technology within Alfa Laval's portfolio, please see documents "Troubleshooting PD Pumps" and "Troubleshooting Centrifugal Pumps".

No flow	Under capacity	Irregular discharge	Low discharge pressure	Pump will not prime	Prime lost after starting	Pump stalls when starting	Pump overheats	Motor overheats	Excessive power absorbed	Noise and vibration	Pump element wear	Syphoning	Seizure	Mechanical seal leakage	Probable Causes	Solutions
✓				✓											Incorrect direction of rotation	Reverse motor (ce, pd)
✓															Pump not primed	Expel gas from suction line and pumping chamber and introduce fluid (ce, pd)
✓		✓	✓		✓					✓					Insufficient NPSH available	Increase suction line diameter (ce, pd) Increase suction head (ce, pd) Simplify suction line configuration and reduce length (ce, pd) Reduce pump speed (pd) Decrease fluid temperature (ce) - check effect of increased viscosity
	✓		✓		✓					✓					Fluid vaporising in suction line	Increase suction line diameter (ce, pd) Increase suction head (ce, pd) Simplify suction line configuration and reduce length (ce, pd) Reduce pump speed (pd) Decrease fluid temperature (ce) - check effect of increased viscosity
✓		✓		✓						✓					Air entering suction line	Remake pipework joints (ce, pd)
	✓			✓						✓					Strainer or filter blocked	Service fittings (ce, pd)
	✓			✓		✓			✓					✓	Fluid viscosity above rated figure	Increase fluid temperature (ce, pd) Decrease pump speed (pd) Increase motor speed (ce) Check seal face viscosity limitations (ce, pd)
✓			✓												Fluid viscosity below rated figure	Decrease fluid temperature (ce, liq, rlp) Increase pump speed (pd)
							✓			✓	✓		✓	✓	Fluid temperature above rated figure	Cool the pump casing (ce, pd) Reduce fluid temperature (ce, pd) Check seal face and elastomer temperature limitations (ce, pd)
						✓		✓	✓						Fluid temperature below rated figure	Heat the pump casing (ce, pd) Increase fluid temperature (ce, pd)
										✓			✓	✓	Unexpected solids in fluid	Clean the system (ce, pd) Fit strainer to suction line (ce, pd) If solids cannot be eliminated, consider fitting double mechanical seals (ce, pd)
✓		✓			✓		✓		✓		✓		✓		Discharge pressure above rated figure	Check for obstructions i.e. closed valve (ce, pd) Service system and change to prevent problem recurring (ce, pd) Simplify discharge line to decrease pressure (ce, pd)

Table 13.3a - continues on the next page

No flow	Under capacity	Irregular discharge	Low discharge pressure	Pump will not prime	Prime lost after starting	Pump stalls when starting	Pump overheats	Motor overheats	Excessive power absorbed	Noise and vibration	Pump element wear	Syphoning	Seizure	Mechanical seal leakage	Packed gland leakage	Probable Causes	Solutions
																Seal flushing inadequate	Increase flush flow rate (ce, pd) / Check that flush fluid flows freely into seal area (ce, pd)
	✓							✓		✓						Pump speed above rated figure	Decrease pump speed (pd)
✓																Pump speed below rated figure	Increase pump speed (pd)
	✓						✓	✓		✓			✓			Pump casing strained by pipework	Check alignment of pipes (ce, pd) / Fit flexible pipes or expansion fittings (ce, pd) / Support pipework (ce, pd)
							✓			✓	✓		✓			Flexible coupling misaligned	Check alignment and adjust mountings accordingly (pd)
							✓	✓		✓			✓			Insecure pump driver mountings	Fit lock washers to slack fasteners and re-tighten (pd)
							✓	✓		✓			✓		✓	Shaft bearing wear or failure	Refer to pump maker for advice and replacement parts (pd)
								✓		✓		✓	✓			Insufficient gear case lubrication	Refer to pump maker's instructions (pd)
✓							✓	✓		✓			✓			Metal to metal contact of pumping element	Check rated and duty pressures (ce, pd) / Refer to pump maker (ce, pd)
✓			✓													Worn pumping element	Fit new components (ce, pd)
✓										✓						Rotor case cover relief valve leakage	Check pressure setting and re-adjust if necessary (SRU only) / Examine and clean seating surfaces (SRU only) / Replace worn parts (SRU only)
	✓									✓						Rotor case cover relief valve chatter	Check for wear on sealing surfaces, guides etc - replace as necessary (SRU only)
✓																Rotor case cover relief valve incorrectly set	Re-adjust spring compression - valve should lift approx. 10% above duty pressure (SRU only)
✓				✓												Suction lift too high	Lower pump or raise fluid level (ce, pd)
														✓		Fluid pumped not compatible with materials used	Use optional materials (pd)
												✓				No barrier in system to prevent flow passing back through pump	Ensure discharge pipework higher than suction tank (pd)
														✓		Pump allowed to run dry	Ensure system operation prevents this (ce, pd) / Fit single or double flushed mechanical seals (ce, pd)
								✓								Faulty motor	Check and replace motor bearings (ce, pd)
✓																Too large clearance between impeller and back plate/casing	Reduce clearance between impeller and back plate casing (ce)
✓																Too small impeller diameter	Fit larger size impeller - check motor size (ce)
✓																Pumping element missing e.g. after service	Fit pumping element (ce, pd)

Table 13.3a - continued

This chapter includes a summary of nomenclature and formulas used in this handbook. Various conversion tables and curves are also shown.

14

Technical Data

14.1 Nomenclature

Symbol	Description	Symbol	Description
A	Area	Q_L	Fluid Losses through Impeller Casing Clearances
D	Tube Diameter	q	Pump Displacement
F	Force	r	Radius
f_D	Darcy Friction Factor	Ra	Surface Roughness
g	Gravity	Re	Reynolds Number
H	Total Head	SG	Specific Gravity
H_s	Total Suction Head	T	Shaft Torque
H_t	Total Discharge Head	V	Fluid Velocity
h_{fs}	Pressure Drop in ALiCE	γ (greek letter 'gamma')	Specific Weight
h_{ft}	Pressure Drop in Discharge Line	δ (greek letter 'delta')	Total
h_s	Static Suction Head	ε (greek letter 'epsilon')	Relative Roughness
h_t	Static Discharge Head	η (greek letter 'eta')	Total Efficiency
L	Tube Length	$η_h$	Hydraulic Efficiency
n	Pump Speed	$η_m$	Mechanical Efficiency
Pa	Pressure Absolute above Fluid Level	$η_{oa}$	Overall Efficiency
P_f	Pressure Loss due to Friction	$η_v$	Volumetric Efficiency
P_s	Vacuum or Pressure in a Tank on Suction Side	μ (greek letter 'mu')	Absolute Viscosity
P_t	Pressure in a Tank on Discharge Side	ν (greek letter 'nu')	Kinematic Viscosity
Pv	Power/Viscosity Factor	ρ (greek letter 'rho')	Fluid Density
Pvp	Vapour Pressure	ω (greek letter 'omega')	Shaft Angular Velocity
Q	Capacity		

Table 14.1a

14.2 Formulas

Designation	Formula	Comments	Where to find
Product			
Viscosity	$\nu = \dfrac{\mu}{\rho}$	Where: ν = Kinematic Viscosity (mm²/s) μ = Absolute Viscosity (mPa.s) ρ = Fluid Density (kg/m³)	2.1.2
	or		
	$\nu = \dfrac{\mu}{SG}$	Where: ν = Kinematic Viscosity (cSt) μ = Absolute Viscosity (cP) SG = Specific Gravity	
	or		
	$\mu = \nu \times SG$	1 Poise = 100 cP 1 Stoke = 100 cSt	
Flow			
Velocity	$V = \dfrac{Q}{A}$	Where: V = Fluid Velocity (m/s) Q = Capacity (m³/s) A = Tube Area (m²)	2.1.7
	or		
	$V = \dfrac{Q \times 353.6}{D^2}$	Where: V = Fluid Velocity (m/s) Q = Capacity (m³/h) D = Tube Diameter (mm)	
	or		
	$V = \dfrac{Q \times 0.409}{D^2}$	Where: V = Fluid Velocity (ft/s) Q = Capacity (US gal/min) D = Tube Diameter (in)	
	or		
	$V = \dfrac{Q \times 0.489}{D^2}$	Where: V = Fluid Velocity (ft/s) Q = Capacity (UK gal/min) D = Tube Diameter (in)	

Table 14.2a - continues next page

Designation	Formula	Comments	Where to find
Reynolds Number (ratio of inertia forces to viscous forces)	$Re = \dfrac{D \times V \times \rho}{\mu}$	Where: D = Tube Diameter (m) V = Fluid Velocity (m/s) ρ = Density (kg/m³) μ = Absolute Viscosity (Pa.s)	2.1.7
	or		
	$Re = \dfrac{D \times V \times \rho}{\mu}$	Where: D = Tube Diameter (mm) V = Fluid Velocity (m/s) ρ = Density (kg/m³) μ = Absolute Viscosity (cP)	
	or		
	$Re = \dfrac{21230 \times Q}{D \times \mu}$	Where: D = Tube Diameter (mm) Q = Capacity (l/min) μ = Absolute Viscosity (cP)	
	or		
	$Re = \dfrac{3162 \times Q}{D \times \nu}$	Where: D = Tube Diameter (in) Q = Capacity (US gal/min) ν = Kinematic Viscosity (cSt)	
	or		
	$Re = \dfrac{3800 \times Q}{D \times \nu}$	Where: D = Tube Diameter (in) Q = Capacity (UK gal/min) ν = Kinematic Viscosity (cSt)	
Pressure/Head			
Pressure (total force per unit area exerted by a fluid)	$P = \dfrac{F}{A}$	Where: F = Force A = Area	2.2.2
Static Pressure/Head (relationship between pressure and elevation)	$P = \rho \times g \times h$	Where: P = Pressure/Head (Pa) ρ = Fluid Density (kg/m³) g = Acceleration due to Gravity (m/s²) h = Height of Fluid (m)	2.2.2
	or		
	$P = \dfrac{h \times SG}{10}$	Where: P = Pressure/Head (bar) h = Height of Fluid (m)	
	or		
	$P = \dfrac{h \times SG}{2.31}$	Where: P = Pressure/Head (PSI) h = Height of Fluid (ft)	
Total Head	$H = H_t - (\pm H_s)$	Where: H_t = Total Discharge Head H_s = Total Suction Head	2.2.2

Table 14.2a - continues next page

Designation	Formula	Comments	Where to find
Total Discharge Head	$H_t = h_t + h_{ft} + p_t$	Where: h_t = Static Discharge Head h_{ft} = Pressure Drop in Discharge Line $p_t > 0$ for Pressure $p_t < 0$ for Vacuum $p_t = 0$ for Open Tank	2.2.2
Total Suction Head	$H_s = h_s - h_{fs} + (\pm\, p_s)$	Where: h_s = Static Suction Head > 0 for Flooded Suction < 0 for Suction Lift h_{fs} = Pressure Drop in Suction Line $p_s > 0$ for Pressure $p_s < 0$ for Vacuum $p_s = 0$ for Open Tank	2.2.2
Friction Loss (Miller equation)	$Pf = \dfrac{f_D \times L \times \rho \times V^2}{D \times 2}$	Where: Pf = Friction Loss (Pa) f_D = Friction Factor (Darcy) L = Tube Length (m) V = Fluid Velocity (m/s) ρ = Fluid Density (kg/m³) D = Tube Diameter (m)	2.2.2
	or		
	$Pf = \dfrac{5 \times SG \times f_D \times L \times V^2}{D}$	Where: Pf = Friction Loss (bar) f_D = Friction Factor (Darcy) L = Tube Length (m) V = Fluid Velocity (m/s) SG = Specific Gravity D = Tube Diameter (mm)	
	or		
	$Pf = \dfrac{0.0823 \times SG \times f_D \times L \times V^2}{D}$	Where: Pf = Friction Loss (PSI) f_D = Friction Factor (Darcy) L = Tube Length (ft) V = Fluid Velocity (ft/s) SG = Specific Gravity D = Tube Diameter (in)	
Darcy Friction Factor	$f_D = \dfrac{64}{Re}$	Where: f_D = Friction Factor Re = Reynolds Number	2.2.2

Table 14.2a - continues next page

Designation	Formula	Comments	Where to find
NPSHa (Net Positive Suction Head available)	NPSHa = Pa ± h_s – h_{fs} – Pvp (+h_s for flooded suction) (– h_s for suction lift)	Where: Pa = Pressure Absolute above Fluid Level (bar) h_s = Static Suction Head (m) h_{fs} = Pressure Drop in Suction Line (m) Pvp = Vapour Pressure (bar a)	2.2.4
		or	
		Where: Pa = Pressure Absolute above Fluid Level (PSI) h_s = Static Suction Head (ft) h_{fs} = Pressure Drop in Suction Line (ft) Pvp = Vapour Pressure (PSIA)	
Power			
Hydraulic Power (theoretical energy required)	Power (W) = Q x H x ρ x g	Where: Q = Capacity (m³/s) H = Total Head (m) ρ = Fluid Density (kg/m³) g = Acceleration due to Gravity (m/s²)	7.2.1
		or	
	Power (kW) = $\dfrac{Q \times H}{k}$	Where: Q = Capacity (l/min) H = Total Head (bar) k = 600	
		or	
	Power (hp) = $\dfrac{Q \times H}{k}$	Where: Q = Capacity (US gal/min) H = Total Head (PSI) k = 1715	
		or	
	Power (hp) = $\dfrac{Q \times H}{k}$	Where: Q = Capacity (UK gal/min) H = Total Head (PSI) k = 1428	
Required Power (power needed at the pump shaft)	$\dfrac{\text{Hydraulic Power}}{\text{Efficiency (100\% = 1.0)}}$		7.2.2
Torque			
Torque	Torque (Nm) = $\dfrac{\text{Required Power (kW) x 9550}}{\text{Pump Speed (rev/min)}}$		7.2.3
		or	
	Torque (kgf m) = $\dfrac{\text{Required Power (kW) x 974}}{\text{Pump Speed (rev/min)}}$		
		or	
	Torque (ft lb) = $\dfrac{\text{Required Power (hp) x 5250}}{\text{Pump Speed (rev/min)}}$		

Table 14.2a - continues next page

Designation	Formula	Comments	Where to find
Efficiency			
Hydraulic Efficiency (η_h)	$\dfrac{\text{Pump Head Loss (m) x 100\%}}{\text{Total Head (m)}}$3		7.2.4
Mechanical Efficiency (η_m)	$\dfrac{\text{1 - Pump Mech. Losses x 100\%}}{\text{Required Power}}$		7.2.4
Volumetric Efficiency (Centrifugal and Liquid Ring Pumps)	$\eta_v = \dfrac{Q \times 100\%}{Q + Q_L}$	Where: η_v = Volumetric Efficiency Q = Pump Capacity Q_L = Fluid Losses due to Leakage through the Impeller Casing Clearances	7.2.4
Volumetric efficiency (Rotary Lobe Pumps)	$\eta_v = \dfrac{Q \times 100\%}{q}$	Where: η_v = Volumetric Efficiency Q = Pump Capacity q = Pump Displacement	7.2.4
Pump Efficiency (η_p)	$\dfrac{\text{Water Horse Power x 100\%}}{\text{Required Power}}$ or $\eta_p = \dfrac{Q \times H \times \rho \times g}{\omega \times T}$	 Where: η_p = Pump Efficiency Q = Capacity (m3/s) H = Total Head/Pressure (m) ρ = Fluid Density (kg/m^3) g = Acceleration due to Gravity (m/s^2) ω = Shaft Angular Velocity (rad/s) T = Shaft Torque (Nm)	 7.2.4
Overall Efficiency (η_{oa})	$\dfrac{\text{Water Horse Power x 100\%}}{\text{Drive Power}}$		7.2.4
Pump Speed - Rotary Lobe Pump			
Pump Speed	$n = \dfrac{Q \times 100}{q \times \eta_v \times 60}$ or $n = \dfrac{Q \times 100}{q \times \eta_v}$ or $n = \dfrac{Q \times 100}{q \times \eta_v}$	Where: n = Pump Speed (rev/min) Q = Capacity (m^3/h) q = Pump Displacement (m^3/100 rev) η_v = Volumetric Efficiency (100% = 1.0) Where: n = Pump Speed (rev/min) Q = Capacity (US gal/min) q = Pump Displacement (US gal/100 rev) η_v = Volumetric Efficiency (100% = 1.0) Where: n = Pump Speed (rev/min) Q = Capacity (UK gal/min) q = Pump Displacement (UK gal/100 rev) η_v = Volumetric Efficiency (100% = 1.0)	7.2.4

Table 14.2a - continues next page

Designation	Formula	Comments	Where to find
Flow Control - Centrifugal Pump			
Connection between Impeller Diameter and Capacity	$D_2 = D_1 \times \sqrt[3]{\dfrac{Q_2}{Q_1}}$	Where: D = Impeller Diameter (mm) Q = Capacity (m^3/h)	7.3.2
Connection between Impeller Diameter and Head	$D_2 = D_1 \times \sqrt{\dfrac{H_2}{H_1}}$	Where: D = Impeller Diameter (mm) H = Head (m)	7.3.2
Connection between Impeller Diameter and Power	$D_2 = D_1 \times \sqrt[5]{\dfrac{P_2}{P_1}}$	Where: D = Impeller Diameter (mm) P = Power (kW)	7.3.2
Reduction of Multi-stage Impeller Diameter	$D_2 = D_1 \times \sqrt{\dfrac{c-b}{a-b}}$	Where: D_1 = Standard Diameter (mm) a = Maximum Working Point (m) b = Minimum Working Point (m) c = Required Working Point (m)	7.3.2
Connection between Impeller Speed and Capacity	$n_2 = n_1 \times \dfrac{Q_2}{Q_1}$	Where: n = Impeller Speed (rev/min) Q = Capacity (m^3/h)	7.3.2
Connection between Impeller Speed and Head	$n_2 = n_1 \times \sqrt{\dfrac{H_2}{H_1}}$	Where: n = Impeller Speed (rev/min) H = Head (m)	7.3.2
Connection between Impeller Speed and Power	$n_2 = n_1 \times \sqrt[3]{\dfrac{P_2}{P_1}}$	Where: n = Impeller Speed (rev/min) P = Power (kW)	7.3.2

Table 14.2a

14.3 Conversion tables

14.3.1 Length

mm	m	cm	in	ft	yd
1	0.001	0.10	0.0394	0.0033	0.0011
1000	1	100	39.370	3.2808	1.0936
10	0.01	1	0.3937	0.0328	0.1094
25.4	0.0254	2.540	1	0.0833	0.0278
304.8	0.3048	30.48	12	1	0.3333
914.4	0.9144	91.441	36	3	1

Table 14.3.1a

14.3.2 Volume

m^3	cm^3	l	in^3	ft^3	UK gal	US gal
1	100×10^4	1000	61024	35.315	220	264
10×10^7	1	10×10^{-4}	0.0610	3.53×10^{-5}	22×10^{-5}	26.4×10^{-5}
0.0010	1000	1	61.026	0.0353	0.22	0.2642
1.64×10^{-5}	16.387	0.0164	1	58×10^{-5}	0.0036	0.0043
0.00283	28317	28.317	1728	1	6.2288	7.4805
0.0045	4546.1	4.546	277.42	0.1605	1	1.201
37.88×10^{-4}	3785.4	3.7853	231	0.1337	0.8327	1

Table 14.3.2a

14.3.3 Volumetric Capacity

m^3/h	l/min	hl/h	UK gal/min	US gal/min	ft^3/h	ft^3/s	m^3/s
1	16.667	10	3.6667	4.3999	35.315	9.81×10^{-3}	2.78×10^{-4}
0.060	1	0.60	0.22	0.2642	2.1189	5.88×10^{-4}	1.67×10^{-5}
0.10	1.6667	1	0.3667	0.4399	3.5315	9.81×10^{-4}	2.78×10^{-5}
0.2727	4.546	2.7270	1	1.201	9.6326	2.67×10^{-3}	7.57×10^{-5}
0.2273	3.785	2.2732	0.8326	1	8.0208	2.23×10^{-3}	6.31×10^{-5}
0.0283	0.4719	0.2832	0.1038	0.1247	1	2.78×10^{-4}	7.86×10^{-6}
101.94	1699	1019.4	373.73	448.83	3600	1	0.0283
3600	6×10^4	36000	13200	15838	127208	35.315	1

Table 14.3.3a

14.3.4 Mass Capacity

kg/s	kg/h	lb/h	UK ton/h	t/d (tonne/day)	t/h (tonne/hour)	lb/s
1	3600	7936.6	3.5431	86.40	3.6	2.2046
2.78×10^{-4}	1	2.2046	98.4×10^{-5}	0.024	0.001	6.12×10^{-4}
1.26×10^{-4}	0.4536	1	44.6×10^{-5}	0.0109	4.54×10^{-4}	2.78×10^{-4}
0.2822	1016.1	2240	1	24.385	1.0160	0.6222
11.57×10^{-3}	41.667	91.859	0.0410	1	0.0417	0.0255
0.2778	1000	2201.8	0.9842	24	1	0.6116
0.4536	1632.9	3600	1.6071	39.190	1.6350	1

Table 14.3.4a

14.3.5 Pressure/Head

bar	kg/cm^2	lb/in^2 (PSI)	ATM (water)	ft (water)	m	mm Hg	in Hg	kPa
1	1.0197	14.504	0.9869	33.455	10.197	750.06	29.530	100
0.9807	1	14.223	0.9878	32.808	10	735.56	28.959	98.07
0.0689	0.0703	1	0.0609	2.3067	0.7031	51.715	2.036	6.89
1.0133	1.0332	14.696	1	33.889	10.332	760	29.921	101.3
0.0299	0.0305	0.4335	0.0295	1	0.3048	22.420	0.8827	2.99
0.0981	0.10	1.422	0.0968	3.2808	1	73.356	2.896	9.81
13.3×10^{-4}	0.0014	0.0193	13.2×10^{-4}	0.0446	0.0136	1	0.0394	0.133
0.0339	0.0345	0.4912	0.0334	1.1329	0.3453	25.40	1	3.39
1.0×10^{-5}	10.2×10^{-6}	14.5×10^{-5}	9.87×10^{-6}	3.34×10^{-4}	10.2×10^{-5}	75.0×10^{-4}	29.5×10^{-5}	1

Table 14.3.5a

14.3.6 Force

kN	kgf	lbf
1	101.97	224.81
9.81×10^{-3}	1	2.2046
44.5×10^{-4}	0.4536	1

Table 14.3.6a

14.3.7 Torque

Nm	kgf m	lb ft	lb in
1	0.102	0.7376	8.8508
9.8067	1	7.2330	86.796
1.3558	0.1383	1	12
0.113	0.0115	0.0833	1

Table 14.3.7a

14.3.8 Power

W	kgf m/s	ft lbf/s	hp	kW
1	0.102	0.7376	1.34×10^{-3}	1000
9.8067	1	7.2330	0.0132	9806.7
1.3558	0.1383	1	1.82×10^{-3}	1355.8
745.70	76.040	550	1	74.6×10^{-4}
0.001	10.2×10^{-5}	73.8×10^{-5}	13.4×10^{-7}	1

Table 14.3.4a

14.3.9 Density

kg/m^3	g/cm^3	lb/in^3	lb/ft^3
1	10^{-3}	36.127×10^{-6}	62.428×10^{-3}
10^3	1	36.127×10^{-3}	62.428
27.680×10^3	27.680	1	1.728×10^3
16.019	16.019×10^{-3}	$0.578\ 70 \times 10^{-3}$	1

Table 14.3.9a

14.3.10 Viscosity Conversion Table

When SG = 1.0 Read Directly Across		When SG is other than 1.0											
cP	Poise	cSt	Stoke	Saybolt Universal SSU	Seconds Engler	Redwood Standard #1	Ford #3	Ford #4	Zahn #1	Zahn #2	Zahn #3	Zahn #4	Zahn #5
1	0.01	1	0.01	31	54	29							
2	0.02	2	0.02	34	57	32							
4	0.04	4	0.04	38	61	36							
7	0.07	7	0.07	47	75	44	8						
10	0.10	10	0.10	60	94	52	9	5	30	16			
15	0.15	15	0.15	80	125	63	10	8	34	17			
20	0.20	20	0.20	100	170	86	12	10	37	18			
25	0.25	25	0.25	130	190	112	15	12	41	19			
30	0.30	30	0.30	160	210	138	19	14	44	20			
40	0.40	40	0.40	210	300	181	25	18	52	22			
50	0.50	50	0.50	260	350	225	29	22	60	24			
60	0.60	60	0.60	320	450	270	33	25	68	27			
70	0.70	70	0.70	370	525	314	36	28	72	30			
80	0.80	80	0.80	430	600	364	41	31	81	34			
90	0.90	90	0.90	480	875	405	45	32	88	37	10		
100	1.0	100	1.0	530	750	445	50	34		41	12	10	
120	1.2	120	1.2	580	900	492	58	41		49	14	11	
140	1.4	140	1.4	690	1050	585	66	45		58	16	13	
160	1.6	160	1.6	790	1200	670	72	50		66	18	14	
180	1.8	180	1.8	900	1350	762	81	54		74	20	16	
200	2.0	200	2.0	1000	1500	817	90	58		82	23	17	10
220	2.2	220	2.2	1100	1650	933	98	62		88	25	18	11
240	2.4	240	2.4	1200	1800	1020	106	65			27	20	12
260	2.6	260	2.6	1280	1950	1085	115	68			30	21	13
280	2.8	280	2.8	1380	2100	1170	122	70			32	22	14
300	3.0	300	3.0	1475	2250	1250	130	74			34	24	15
320	3.2	320	3.2	1530	2400	1295	136	89			36	25	16
340	3.4	340	3.4	1630	2550	1380	142	95			39	26	17
360	3.6	360	3.6	1730	2700	1465	150	100			41	27	18
380	3.8	380	3.8	1850	2850	1570	160	106			43	29	19
400	4.0	400	4.0	1950	3000	1650	170	112			46	30	20
420	4.2	420	4.2	2050	3150	1740	180	118			48	32	21
440	4.4	440	4.4	2160	3300	1830	188	124			50	33	22
460	4.6	460	4.6	2270	3450	1925	200	130			52	34	23
480	4.8	480	4.8	2380	3600	2020	210	137			54	36	24
500	5.0	500	5.0	2480	3750	2100	218	143			58	38	25
550	5.5	550	5.5	2660	4125	2255	230	153			64	40	27
600	6.0	600	6.0	2900	4500	2460	250	170			68	45	30
700	7.0	700	7.0	3380	5250	2860	295	194			76	51	35
800	8.0	800	8.0	3880	6000	3290	340	223				57	40
900	9.0	900	9.0	4300	8750	3640	365	247				63	45
1000	10	1000	10	4600	7500	3900	390	264				69	49
1100	11	1100	11	5200	8250	4410	445	299				77	55

When SG = 1.0 Read Directly Across		When SG is other than 1.0											
cP	Poise	cSt	Stoke	Saybolt Universal SSU	Seconds Engler	Redwood Standard #1	Ford #3	Ford #4	Zahn #1	Zahn #2	Zahn #3	Zahn #4	Zahn #5
1200	12	1200	12	5620	9000	4680	480	323					59
1300	13	1300	13	6100	9750	5160	520	350					64
1400	14	1400	14	6480	10350	5490	550	372					70
1500	15	1500	15	7000	11100	5940	595	400					75
1600	16	1600	16	7500	11850	6350	635	430					80
1700	17	1700	17	8000	12600	6780	680	460					85
1800	18	1800	18	8500	13300	7200	720	490					91
1900	19	1900	19	9000	13900	7620	760	520					96
2000	20	2000	20	9400	14600	7950	800	540					
2100	21	2100	21	9850	15300	8350	835	565					
2200	22	2200	22	10300	16100	8730	875	592					
2300	23	2300	23	10750	16800	9110	910	617					
2400	24	2400	24	11200	17500	9500	950	645					
2500	25	2500	25	11600	18250	9830	985	676					
3000	30	3000	30	14500	21800	12300	1230	833					
3500	35	3500	35	16500	25200	14000	1400	950					
4000	40	4000	40	18500	28800	15650	1570	1060					
4500	45	4500	45	21000	32400	17800		1175					
5000	50	5000	50	23500	36000	19900		1350					
5500	55	5500	55	26000	39600			1495					
6000	60	6000	60	28000	43100			1605					
6500	65	6500	65	30000	46000			1720					
7000	70	7000	70	32500	49600			1870					
7500	75	7500	75	35000	53200			2010					
8000	80	8000	80	37000	56800			2120					
8500	85	8500	85	39500	60300			2270					
9000	90	9000	90	41080	63900			2350					
9500	95	9500	95	43000	67400			2470					
10000	100	10000	100	46500	71000			2670					
15000	150	15000	150	69400	106000								
20000	200	20000	200	92500	140000								
30000	300	30000	300	138500	210000								
40000	400	40000	400	185000	276000								
50000	500	50000	500	231000	345000								
60000	600	60000	600	277500	414000								
70000	700	70000	700	323500	484000								
80000	800	80000	800	370000	550000								
90000	900	90000	900	415500	620000								
100000	1000	100000	1000	462000	689000								
125000	1250	125000	1250	578000	850000								
150000	1500	150000	1500	694000									
175000	1750	175000	1750	810000									
200000	2000	200000	2000	925000									

Table 14.3.10a

14.3.11 Temperature Conversion Table

Minus 459.4 - 0			0 - 49			50 - 100			100 - 490			500 - 1000		
° C	to	° F	° C	to	° F	° C	to	° F	° C	to	° F	° C	to	° F
-273	-459		-17.8	0	32	10.0	50	122.0	38	100	212	260	500	932
-268	-450		-17.2	1	33.8	10.6	51	123.8	43	110	230	266	510	950
-262	-440		-16.7	2	35.6	11.1	52	125.6	49	120	248	271	520	968
-257	-430		-16.1	3	37.4	11.7	53	127.4	54	130	266	277	530	986
-251	-420		-15.6	4	39.2	12.2	54	129.2	60	140	284	282	540	1004
-246	-410		-15.0	5	41.0	12.8	55	131.0	66	150	302	288	550	1022
-240	-400		-14.4	6	42.8	13.3	56	132.8	71	160	320	293	560	1040
-234	-390		-13.9	7	44.6	13.9	57	134.6	77	170	338	299	570	1058
-229	-380		-13.3	8	46.4	14.4	58	136.4	82	180	356	304	580	1076
-223	-370		-12.8	9	48.2	15.0	59	138.2	88	190	374	310	590	1094
-218	-360		-12.2	10	50.0	15.6	60	140.0	93	200	392	316	600	1112
-212	-350		-11.7	11	51.8	16.1	61	141.8	99	210	410	321	610	1130
-207	-340		-11.1	12	53.6	16.7	62	143.6	100	212	414	327	620	1148
-201	-330		-10.6	13	55.4	17.2	63	145.4	104	220	428	332	630	1166
-196	-320		-10.0	14	57.2	17.8	64	147.2	110	230	446	338	640	1184
-190	-310		-9.4	15	59.0	18.3	65	149.0	116	240	464	343	650	1202
-184	-300		-8.9	16	60.8	18.9	66	150.8	121	250	482	349	660	1220
-179	-290		-8.3	17	62.6	19.4	67	152.6	127	260	500	354	670	1238
-173	-280		-7.8	18	64.4	20.0	68	154.4	132	270	518	360	680	1256
-169	-273	-459.4	-7.2	19	66.2	20.6	69	156.2	138	280	536	366	690	1274
-168	-270	-454	-6.7	20	68.0	21.1	70	158.0	143	290	554	371	700	1292
-162	-260	-436	-6.1	21	69.8	21.7	71	159.8	149	300	572	377	710	1310
-157	-250	-418	-5.6	22	71.6	22.2	72	161.6	154	310	590	382	720	1328
-151	-240	-400	-5.0	23	73.4	22.8	73	163.4	160	320	608	388	730	1346
-146	-230	-382	-4.4	24	75.2	23.3	74	165.2	166	330	626	393	740	1364
-140	-220	-364	-3.9	25	77.0	23.9	75	167.0	171	340	644	399	750	1382
-134	-210	-346	-3.3	26	78.8	24.4	76	168.8	177	350	662	404	760	1400
-129	-200	-328	-2.8	27	80.6	25.0	77	170.6	182	360	680	410	770	1418
-123	-190	-310	-2.2	28	82.4	25.6	78	172.4	188	370	698	416	780	1436
-118	-180	-292	-1.7	29	84.2	26.1	79	174.2	193	380	716	421	790	1454
-112	-170	-274	-1.1	30	86.0	26,7	80	176.0	199	390	734	427	800	1472
-107	-160	-256	-0.6	31	87.8	27.2	81	177.8	204	400	752	432	810	1490
-101	-150	-238	0.0	32	89.6	27.8	82	179.6	210	410	770	438	820	1508

Minus 459.4 - 0			0 - 49			50 - 100			100 - 490			500 - 1000		
° C	to	° F	° C	to	° F	° C	to	° F	° C	to	° F	° C	to	° F
-96	-140	-220	0.6	33	91.4	28.3	83	181.4	216	420	788	443	830	1526
-90	-130	-202	1.1	34	93.2	28.9	84	183.2	221	430	806	449	840	1544
-84	-120	-184	1.7	35	95.0	29.4	85	185.0	227	440	824	454	850	1562
-79	-110	-166	2.2	36	96.8	30.0	86	186.8	232	450	842	460	860	1580
-73	-100	-148	2.8	37	98.6	30.6	87	188.6	238	460	860	466	870	1598
-68	-90	-130	3.3	38	100.4	31.1	88	190.4	243	470	878	471	880	1616
-62	-80	-112	3.9	39	102.2	31.7	89	192.2	249	480	896	477	890	1634
-57	-70	-94	4.4	40	104.0	32.2	90	194.0	254	490	914	482	900	1652
-51	-60	-76	5.0	41	105.8	32.8	91	195.8				488	910	1670
-46	-50	-58	5.6	42	107.6	33.3	92	197.6				493	920	1688
-40	-40	-40	6.1	43	109.4	33.9	93	199.4				499	930	1706
-34	-30	-22	6.7	44	111.2	34.4	94	201.2				504	940	1724
-29	-20	-4	7.2	45	113.0	35.0	95	203.0				510	950	1742
-23	-10	14	7.8	46	114.8	35.6	96	204.8				516	960	1760
-17.8	0	32	8.3	47	116.6	36.1	97	206.6				521	970	1778
			8.9	48	118.4	36.7	98	208.4				527	980	1796
			9.4	49	120.2	37.2	99	210.2				532	990	1814
						37.8	100	212.0				538	1000	1832

Locate temperature in the middle column. If in° C read the° F equivalent in the right hand column. If in° F read° C equivalent in the left hand column.

$° C = (° F - 32) \times 0.5556$ $° F = (° C \times 1.8) + 32$

Table 14.3.11a

14.4 Water Vapour Pressure Table

Temperature (° C)	Density (p) (kg/m³)	Vapour Pressure (Pvp) (kPa)
0	999.8	0.61
5	1000.0	0.87
10	999.7	1.23
15	999.1	1.71
20	998.2	2.33
25	997.1	3.40
30	995.7	4.25
35	994.1	5.62
40	992.2	7.38
45	990.2	9.60
50	988.0	12.3
55	985.7	15.7
60	983.2	19.9
65	980.6	25.1
70	977.8	31.2
75	974.9	38.6
80	971.8	47.5
85	968.6	57.9
90	965.3	70.1
95	961.9	84.7
100	958.4	101.3
Vapour pressure: 1 bar = 100 kPa = 10^5 N/m²		

Table 14.4a

14.5 Pressure Drop Curve for 100 m ISO/DIN Tube

Pressure drop (m)

A: 25 mm	E: 51 mm	I: 76 mm	M: DN125
B: DN25	F: DN50	J: DN80	N: DN150
C: 38 mm	G: 63.5 mm	K: 101.6 mm	
D: DN40	H: DN65	L: DN100	1 bar ≈ 10 m (metre liquid column)

Note: A, C, E, G, I and K refer to ISO Tube - B, D, F, H, J, L, M and N refer to DIN Tube
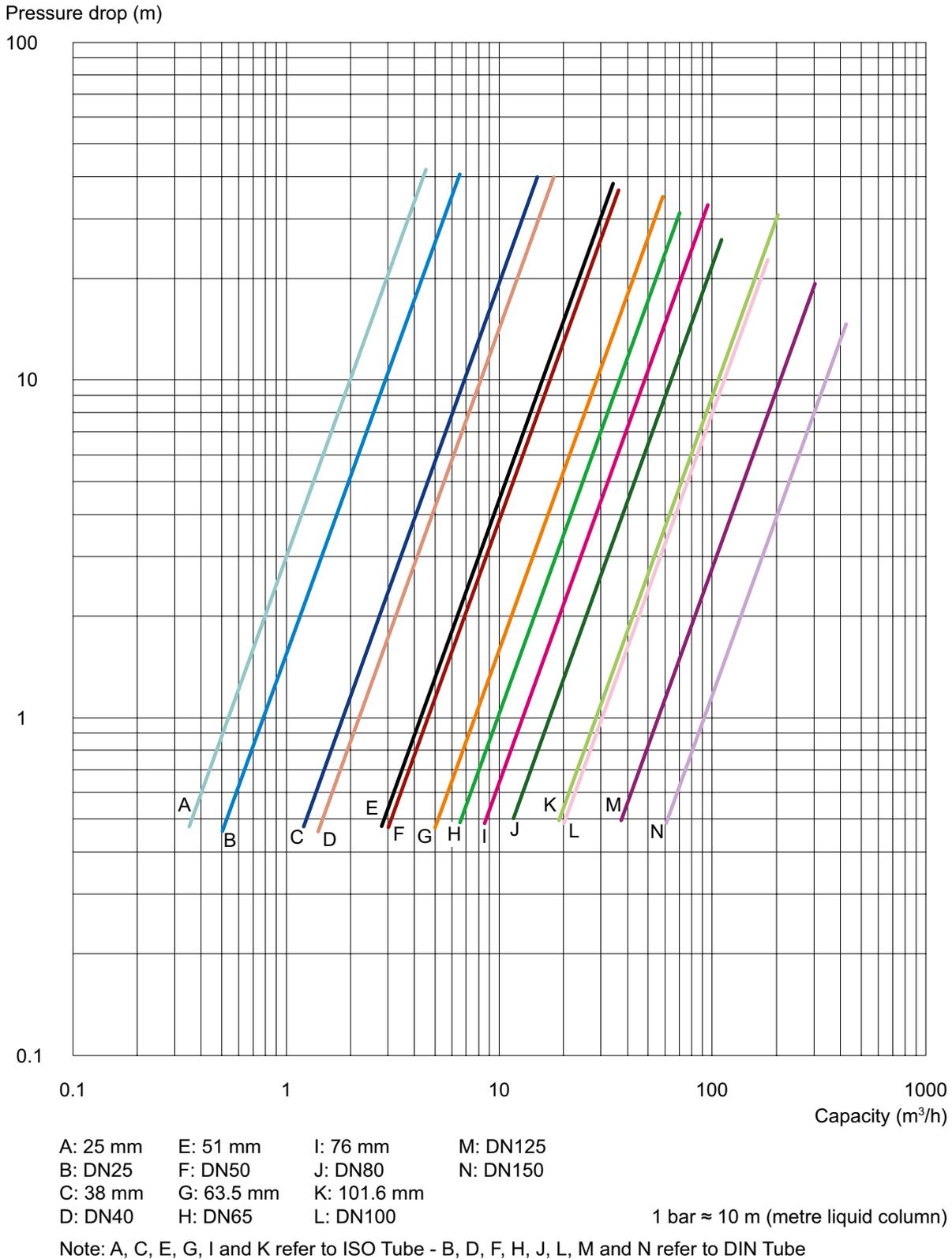
Fig. 14.5a Pressure Drop Curve

14.6 Velocity

(m/s) in ISO and DIN Tubes at various Capacities

A: 1"	E: 2"	I: 3"	M: DN125
B: DN25	F: DN50	J: DN80	N: DN150
C: 1½"	G: 2½"	K: 4"	
D: DN40	H: DN65	L: DN100	1 m³/h = 1000 l/h

Note: A, C, E, G, I and K refer to ISO Tube - B, D, F, H, J, L, M and N refer to DIN Tube

Fig. 14.6a Connection between velocity and capacity at different tube dimensions

14.7 Equivalent Tube Length Table

14.7.1 ISO Tube Metric for Water at 2 m/s

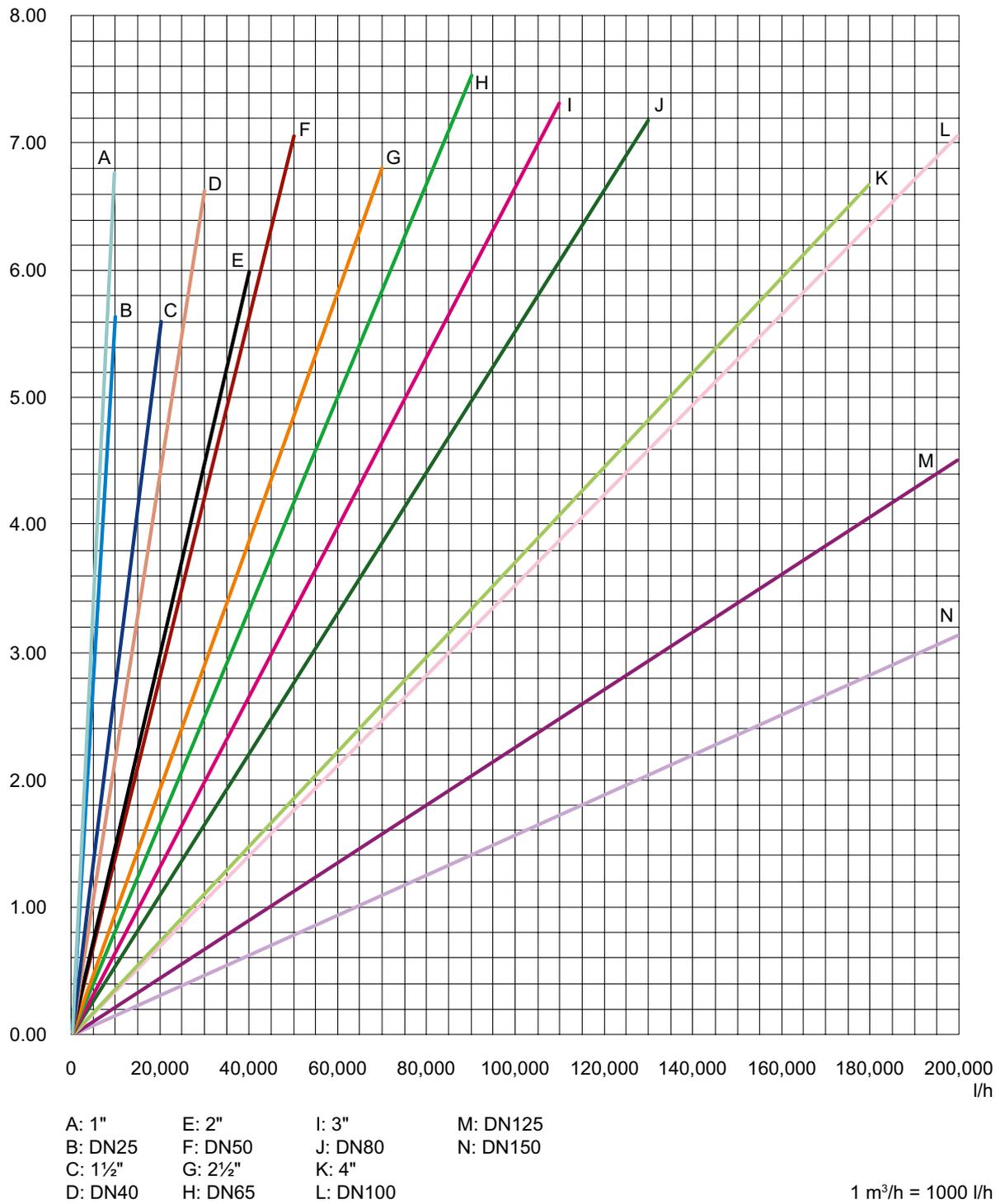| Unique SSV Standard		Equivalent tube length in metres per unit					
Unique SSV		25 mm	38 mm	51 mm	63.5 mm	76 mm	101.6 mm
	Shut-off	2	4	6	5	5	18
	Shut-off	2	3	5	5	4	18
	Change-over	2	4	7	5	4	22
	Change-over	2	6	9	8	5	20
	Change-over	3	7	11	10	8	33
	Change-over	4	7	11	15	12	31
Unique SSV Reverse Acting							
	RA	3	7	11	10	8	33
	RA	3	7	11	10	8	33
	RA	4	7	11	15	12	31
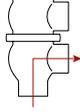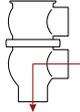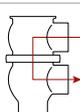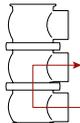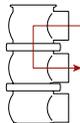
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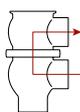| Unique SSV Standard | | Equivalent tube length in metres per unit | | | | | |
Unique SSV Long Stroke		25 mm	38 mm	51 mm	63.5 mm	76 mm	101.6 mm
	Shut-off		4	3	4	4	11
	Shut-off		1	2	3	3	4
	Change-over		1	2	2	3	4
	Change-over		1	2	2	2	4
	Change-over		2	3	4	4	7
	Change-over		1	5	7	7	6
Unique SSV Tangential							
	Shut-off			6	6	5	47
	Shut-off			5	5	4	17
	Change-over			7	5	5	22
	Change-over			9	8	5	20
	Change-over			11	10	8	33
	Change-over			11	15	12	31
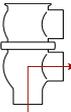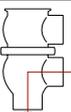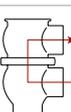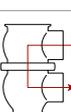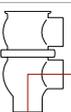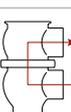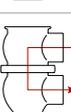
Table 14.7.1a - continues next page

Unique SSV Standard		Equivalent tube length in metres per unit					
Unique SSV Two step		25 mm	38 mm	51 mm	63.5 mm	76 mm	101.6 mm
	Shut-off		4	6	6	5	47
	Shut-off		3	4	5	4	17
	Change-over		3	7	5	4	22
	Change-over		6	9	9	5	20
	Change-over		7	11	10	8	33
	Change-over		7	11	15	12	31
Unique SSV Aseptic							
	Shut-off	2	5	8	9	8	28
	Shut-off	2	4	6	8	9	41
	Change-over	3	6	10	16	10	63
	Change-over	3	8	13	15	9	43
	Change-over	5	9	16	18	11	62
	Change-over	4	8	23	19	13	69

Table 14.7.1a - continues next page

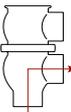| Unique SSV Standard	Equivalent tube length in metres per unit					
Unique SSV Tank Outlet	25 mm	38 mm	51 mm	63.5 mm	76 mm	101.6 mm
			4	5	4	17
			6	6	4	17
			11	10	8	33
			11	14	12	31
Other valves						
Non-return valve LKC-2						
	7	10	12	21	20	26
Butterfly valve LKB						
	1	1	1	1	2	2
Koltek MH						
	1	2	3	5	6	7
	1	2	4	6	9	10
Mixproof valves						
Unique*						
		14	14	27	25	26
		14	14	27	25	26
		5	4	6	5	4
		6	5	7	7	5
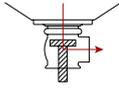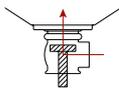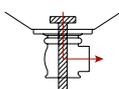
*Pressure drop/equivalent tube length is for unbalanced upper plug and balanced lower plug.

For other combinations use the Anytime Unique configuration tool.

Table 14.7.1a - continues next page

| Mixproof valves | Equivalent tube length in metres per unit | | | | | |
SMP-BC	25 mm	38 mm	51 mm	63.5 mm	76 mm	101.6 mm
		3	3	4	3	6
		3	6	11	8	18
		3	5	7	7	11
		7	11	13	15	32
		6	10	13	14	31
		9	12	34	25	101
		6	12	34	23	101
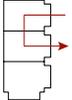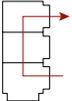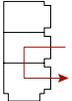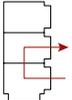
Table 14.7.1a - continues next page

Mixproof valves	Equivalent tube length in metres per unit					
SMP-BCA	25 mm	38 mm	51 mm	63.5 mm	76 mm	101.6 mm
		2	3	4	3	6
		5	10	18	29	84
		3	9	16	29	81
		6	18	30	41	104
		5	12	20	27	75
		5	14	41	41	152
		6	14	34	38	146
Unique Mixproof Tank Outlet						
			5	7	6	17
			12	21	15	35
			19	18	14	43
Tubes and fittings						
Bend 90°	0.3	1	1	1	1	2
Bend 45°	0.2	0.4	1	1	1	1
Tee (out through side port)	1	2	3	4	5	7
Tee (in through side port)	1	2	2	3	4	5

Table 14.7.1a

14.7.2 ISO Tube Feet for Water at 6 ft/s

Unique SSV Standard		Equivalent tube length in feet per unit					
Unique SSV		1"	1.5"	2"	2.5"	3"	4"
	Shut-off	7	13	20	16	16	59
	Shut-off	7	10	16	16	13	59
	Change-over	7	13	23	16	13	72
	Change-over	7	20	30	26	16	66
	Change-over	10	23	36	33	26	108
	Change-over	13	23	36	49	39	102
Unique SSV Reverse Acting							
	RA	10	23	36	33	26	108
	RA	10	23	36	33	26	108
	RA	13	23	36	49	39	102
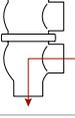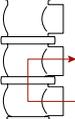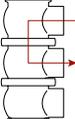
Table 14.7.2a - continues next page

Unique SSV Standard				Equivalent tube length in feet per unit			
Unique SSV Long Stroke		1"	1.5"	2"	2.5"	3"	4"
	Shut-off		13	10	13	13	36
	Shut-off		3	7	10	10	13
	Change-over		3	7	7	10	13
	Change-over		3	7	7	7	13
	Change-over		7	10	13	13	23
	Change-over		3	16	23	23	20
Unique SSV Tangential							
	Shut-off			20	20	16	154
	Shut-off			16	16	13	56
	Change-over			23	16	16	72
	Change-over			30	26	16	66
	Change-over			36	33	26	108
	Change-over			36	49	39	102

Table 14.7.2a - continues next page

| Unique SSV Standard | Equivalent tube length in feet per unit | | | | | |
Unique SSV Two step	1"	1.5"	2"	2.5"	3"	4"
Shut-off		13	20	20	16	154
Shut-off		10	13	16	13	56
Change-over		10	23	16	13	72
Change-over		20	30	30	16	66
Change-over		23	36	33	26	108
Change-over		23	36	49	39	102
Unique SSV Aseptic						
Shut-off	7	16	26	30	26	92
Shut-off	7	13	20	26	30	135
Change-over	10	20	33	52	33	207
Change-over	10	26	43	49	30	141
Change-over	16	30	52	59	36	203
Change-over	13	26	75	62	43	226

Table 14.7.2a - continues next page

Unique SSV Standard			Equivalent tube length in feet per unit			
Unique SSV Tank Outlet	1"	1.5"	2"	2.5"	3"	4"
			13	16	13	56
			20	20	13	56
			36	33	26	108
			36	46	39	102
Other valves						
Non-return valve LKC-2						
	23	33	39	69	66	85
Butterfly valve LKB						
	3	3	3	3	7	7
Koltek MH						
	3	7	10	16	20	23
	3	7	13	20	30	33
Mixproof valves						
Unique*						
		46	46	89	82	85
		46	46	89	82	85
		16	13	20	16	13
		20	16	23	23	16
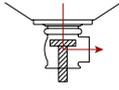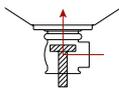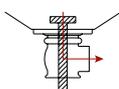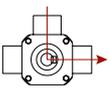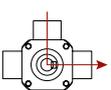
*Pressure drop/equivalent tube length is for unbalanced upper plug and balanced lower plug.

For other combinations use the Anytime Unique configuration tool.

Table 14.7.2a - continues next page

Mixproof valves	Equivalent tube length in feet per unit					
SMP-BC	1"	1.5"	2"	2.5"	3"	4"
		10	10	13	10	20
		10	20	36	26	59
		10	16	23	23	36
		23	36	43	49	105
		20	33	43	46	102
		30	39	112	82	331
		20	39	112	75	331

Table 14.7.2a - continues next page

Mixproof valves	Equivalent tube length in feet per unit					
SMP-BCA	1"	1.5"	2"	2.5"	3"	4"
		7	10	13	10	20
		16	33	59	95	276
		10	30	52	95	266
		20	59	98	135	341
		16	39	66	89	246
		16	46	135	135	499
		20	46	112	125	479
Unique Mixproof Tank Outlet						
			16	23	20	56
			39	69	49	115
			62	59	46	141
Tubes and fittings						
Bend 90°	1	3	3	3	3	7
Bend 45°	1	1	3	3	3	3
Tee (out through side port)	3	7	10	13	16	13
Tee (in through side port)	3	7	7	10	13	16

Table 14.7.2a

14.7.3 DIN Tube Metric for Water at 2 m/s

Unique SSV Standard		Equivalent tube length in metres per unit							
Unique SSV		DN25	DN40	DN50	DN65	DN80	DN100	DN125	DN150
	Shut-off	3	5	6	9	9	16	35	60
	Shut-off	3	4	5	6	8	16	25	70
	Change-over	3	5	7	7	9	21	30	65
	Change-over	3	7	9	12	10	19	45	75
	Change-over	4	8	12	14	17	31	60	150
	Change-over	5	8	11	20	25	29	40	75
Unique SSV Reverse Acting									
	RA	5	8	11	20	25	29		
	RA	5	8	11	20	25	29		
	RA	4	8	12	14	17	31		
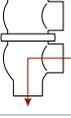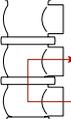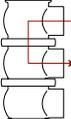
Table 14.7.3a - continues next page

Unique SSV Standard		Equivalent tube length in metres per unit							
Unique SSV Long Stroke		DN25	DN40	DN50	DN65	DN80	DN100	DN125	DN150
	Shut-off		5	3	5	8	10		
	Shut-off		1	2	3	5	5		
	Change-over		1	2	3	5	5		
	Change-over		1	2	3	4	5		
	Change-over		2	4	4	7	9		
	Change-over		1	6	8	12	8		
Unique SSV Tangential									
	Shut-off			6	8	9	44		
	Shut-off			5	6	8	16		
	Change-over			7	6	9	21		
	Change-over			9	12	10	19		
	Change-over			12	14	17	31		
	Change-over			11	20	25	29		

Table 14.7.3a - continues next page

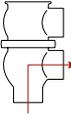| Unique SSV Standard		Equivalent tube length in metres per unit							
Unique SSV Two step		DN25	DN40	DN50	DN65	DN80	DN100	DN125	DN150
	Shut-off		5	6	8	9	44		
	Shut-off		4	5	6	8	16		
	Change-over		4	7	6	9	21		
	Change-over		7	9	12	10	19		
	Change-over		8	12	14	17	31		
	Change-over		8	11	20	25	29		
Unique SSV Aseptic									
	Shut-off	3	6	9	12	16	26		
	Shut-off	2	4	6	11	18	38		
	Change-over	3	7	11	22	20	59		
	Change-over	3	10	14	20	18	40		
	Change-over	7	11	17	25	23	59		
	Change-over	6	10	25	26	26	65		

Table 14.7.3a - continues next page

Unique SSV Standard			Equivalent tube length in metres per unit					
Unique SSV Tank Outlet	DN25	DN40	DN50	DN65	DN80	DN100	DN125	DN150
			5	6	8	16		
			6	8	9	16		
			12	14	17	31		
			11	20	25	29		
Other valves								
Non-return valve LKC-2								
	14	14	15	32	36	30		
Butterfly valve LKB								
	2	1	1	2	2	2	2	1
Koltek MH								
	2	2	5	9	10	8		
	2	2	5	9	14	13		
Mixproof valves								
Unique*								
		14	14	27	25	26	40	85
		14	14	27	25	26	40	85
		5	4	6	5	4	8	16
		6	5	7	7	5	10	20
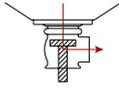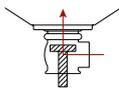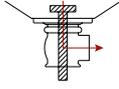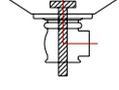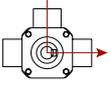
*Pressure drop/equivalent tube length is for unbalanced upper plug and balanced lower plug.

For other combinations use the Anytime Unique configuration tool.

Table 14.7.3a - continues next page

Mixproof valves	Equivalent tube length in metres per unit							
SMP-BC	DN25	DN40	DN50	DN65	DN80	DN100	DN125	DN150
		3	4	5	5	7	4	8
		4	7	13	15	21	38	78
		4	6	11	12	20	31	61
		9	17	22	24	40		
		7	13	22	23	37		
		10	15	52	44	114		
		9	15	52	44	114		
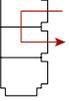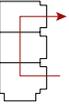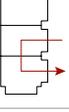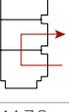
Table 14.7.3a - continues next page

Mixproof valves	Equivalent tube length in metres per unit							
SMP-BCA	DN25	DN40	DN50	DN65	DN80	DN100	DN125	DN150
		3	4	5	5	6		
		6	13	32	51	97		
		3	12	25	49	94		
		9	24	46	72	124		
		6	15	30	46	84		
		8	20	62	67	174		
		9	21	54	54	167		
Unique Mixproof 3 Body								
			25	37	48	55	40	85
			13	37	45	34	38	79
			8	15	21	28	40	85

Table 14.7.3a - continues next page

Unique Mixproof Tank Outlet								
		5	10	13	15			
		13	29	31	33			
		20	24	28	41			
Tubes and fittings								
Bend 90°	0.3	1	1	1	1	2	3	4
Bend 45°	0.2	0.4	1	1	1	1	2	2
Tee (out through side port)	1	2	3	4	5	7	9	10
Tee (in through side port)	1	2	2	3	4	5	7	8

Table 14.7.3a

14.8 Moody Diagram
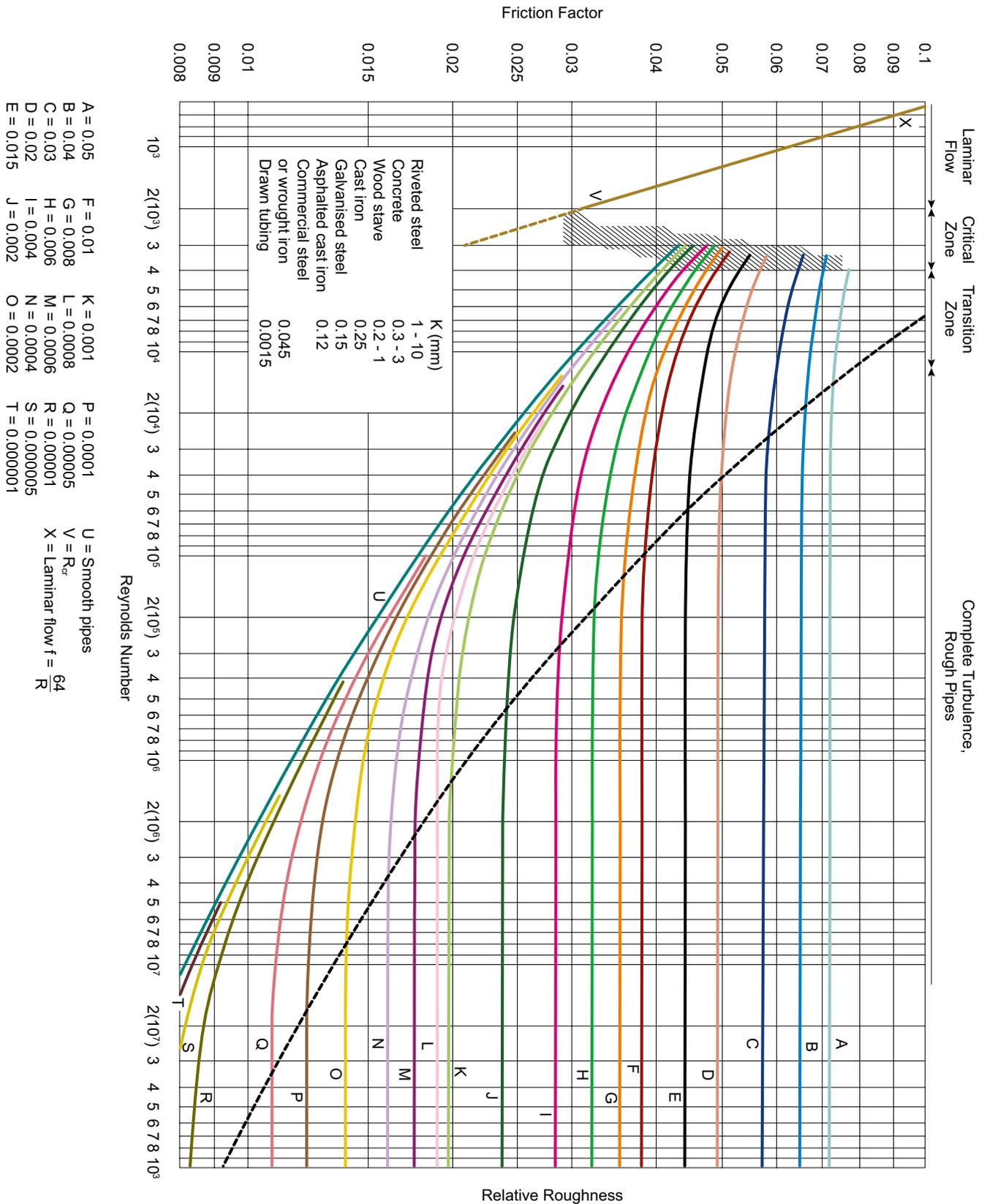
Fig. 14.8a Moody diagram for f$_D$ (after Miller)

14.9 Initial Suction Line Sizing

A = ½" - DN15   D = 1½" - DN40   G = 3" - DN80   J = 6" - DN150
B = ¾" - DN20   E = 2" - DN50   H = 4" - DN100   K = 8" - DN200
C = 1" - DN25   F = 2½" - DN65   I = 5" - DN125
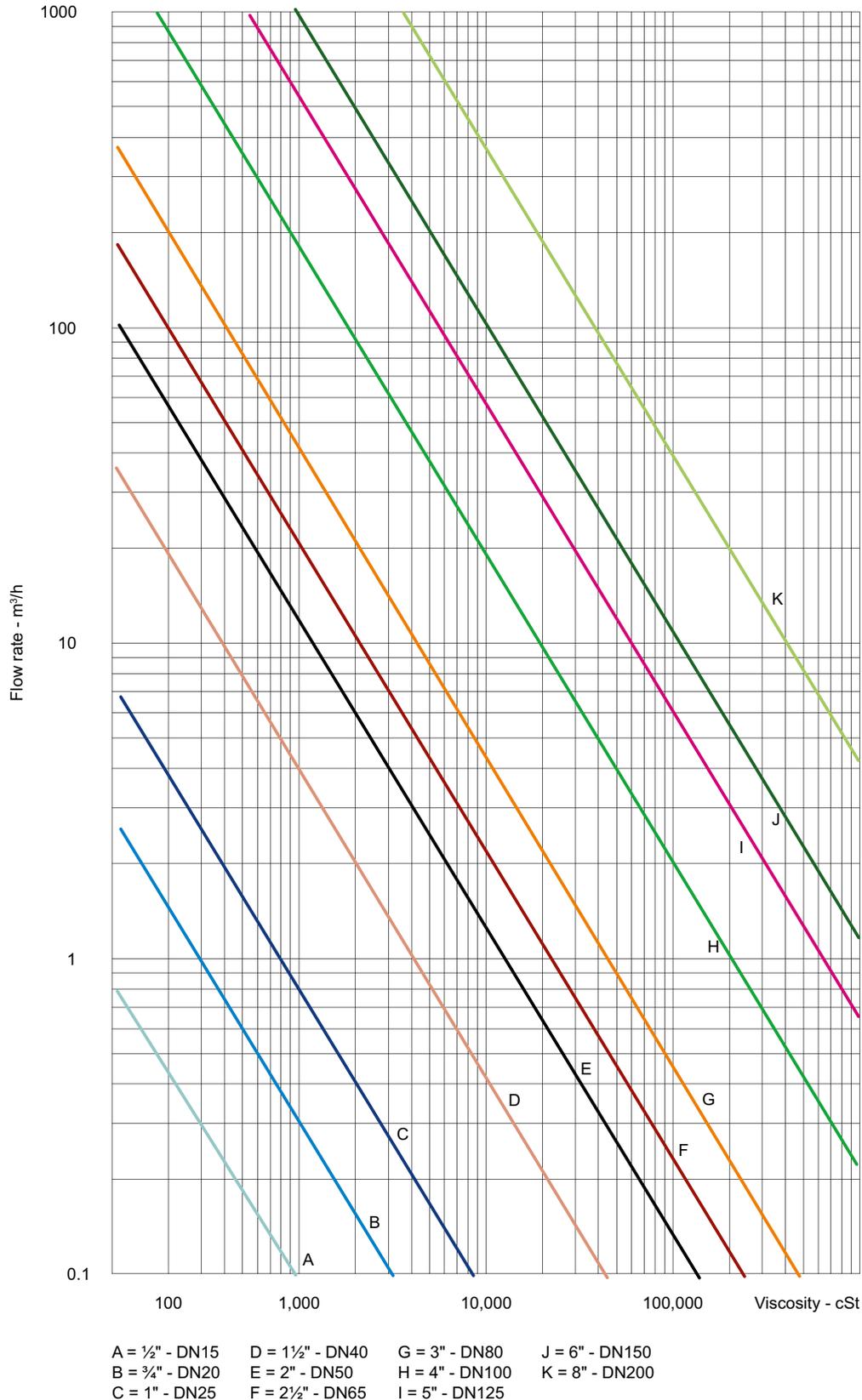
Fig. 14.9a Initial suction line sizing

14.10 Elastomer Compatibility Guide

Listed below are fluids commonly pumped.

The elastomer compatibility is for guidance purposes only as this may be affected by temperature.

The fluid viscous behaviour type shown relates to general terms - in some instances Pseudoplastic fluids can have Thixotropic tendencies.

(†) - Fluid can become Dilatant at high concentration and high shear rate.

(‡) - If low concentration, this can be Newtonian.

Name of Fluid Pumped	Elastomer Material				Viscous Behaviour Type
	NBR	EPDM	FPM	PTFE/ FFPM	
ACETIC ACID		✓		✓	Newtonian
ACETONE		✓		✓	Newtonian
ADHESIVE - SOLVENT BASED				✓	Pseudoplastic
ADHESIVE - WATER BASED		✓		✓	Pseudoplastic
ALUM SLUDGE	✓	✓	✓	✓	Pseudoplastic
AMMONIUM HYDROXIDE		✓		✓	Newtonian
ANIMAL FAT			✓	✓	Newtonian
BABY BATH			✓	✓	Pseudoplastic
BABY LOTION			✓	✓	Pseudoplastic
BABY OIL			✓	✓	Newtonian
BATH FOAM			✓	✓	Pseudoplastic
BATTER		✓	✓	✓	Pseudoplastic
BEER		✓	✓	✓	Newtonian
BENTONITE SUSPENSION	✓	✓	✓	✓	Pseudoplastic (†)
BISCUIT CREAM			✓	✓	Pseudoplastic
BISULPHITE	✓	✓	✓	✓	Newtonian
BITUMEN	✓		✓	✓	Pseudoplastic
BLACK LIQUOR			✓	✓	Newtonian
BLEACH		✓	✓	✓	Newtonian
BLOOD		✓	✓	✓	Newtonian
BODY LOTION			✓	✓	Pseudoplastic
BODY SCRUB			✓	✓	Pseudoplastic
BRINE	✓	✓	✓	✓	Newtonian
BUTTER		✓	✓	✓	Pseudoplastic
CALCIUM CARBONATE SLURRY	✓	✓	✓	✓	Pseudoplastic
CARAMEL - COLOURING		✓	✓	✓	Newtonian
CARAMEL - TOFFEE		✓	✓	✓	Pseudoplastic
CASTOR OIL	✓		✓	✓	Newtonian

Table 14.10a Elastomer compatibility guide - continues next page

Name of Fluid Pumped	Elastomer Material				Viscous Behaviour Type
	NBR	EPDM	FPM	PTFE/ FFPM	
CELLULOSE ACETATE DOPE				✓	Pseudoplastic
CELLULOSE SUSPENSION	✓	✓	✓	✓	Pseudoplastic
CERAMIC SLIP	✓	✓	✓	✓	Pseudoplastic (†)
CHEESE		✓	✓	✓	Pseudoplastic
CHEWING GUM				✓	Pseudoplastic
CHINA CLAY SLURRY	✓	✓	✓	✓	Pseudoplastic (†)
CHOCOLATE			✓	✓	Pseudoplastic
CHROMIC ACID			✓	✓	Newtonian
CHUTNEY		✓	✓	✓	Pseudoplastic
CITRIC ACID	✓	✓	✓	✓	Newtonian
COAL TAR			✓	✓	Newtonian
COCOA BUTTER			✓	✓	Newtonian
COCOA LIQUOR			✓	✓	Pseudoplastic
COCONUT CREAM			✓	✓	Pseudoplastic
COLLAGEN GEL		✓	✓	✓	Pseudoplastic
CONDENSED MILK		✓	✓	✓	Pseudoplastic
COPPER SULPHATE		✓	✓	✓	Newtonian
CORN STEEP LIQUOR		✓	✓	✓	Newtonian
CORN SYRUP	✓	✓	✓	✓	Newtonian
COSMETIC CREAM			✓	✓	Pseudoplastic
COUGH SYRUP		✓	✓	✓	Pseudoplastic
CRUDE OIL			✓	✓	Pseudoplastic
CUSTARD		✓	✓	✓	Pseudoplastic
DAIRY CREAM		✓	✓	✓	Pseudoplastic
DETERGENT - AMPHOTERIC			✓	✓	Newtonian
DETERGENT - ANIONIC	✓	✓	✓	✓	Pseudoplastic (‡)
DETERGENT - CATIONIC			✓	✓	Newtonian
DETERGENT - NONIONIC		✓	✓	✓	Newtonian
DIESEL OIL	✓		✓	✓	Newtonian
DODECYL BENZENE SULPHONIC ACID			✓	✓	Newtonian
DRILLING MUD	✓	✓	✓	✓	Pseudoplastic
DYE		✓	✓	✓	Newtonian
EGG		✓	✓	✓	Pseudoplastic
ENZYME SOLUTION		✓		✓	Newtonian
ETHANOL	✓	✓		✓	Newtonian
ETHYLENE GLYCOL	✓	✓	✓	✓	Newtonian
FABRIC CONDITIONER			✓	✓	Pseudoplastic
FATS			✓	✓	Newtonian
FATTY ACID			✓	✓	Newtonian
FERRIC CHLORIDE	✓	✓	✓	✓	Newtonian
FERTILISER	✓	✓	✓	✓	Pseudoplastic
FILTER AID	✓	✓	✓	✓	Pseudoplastic

Table 14.10a Elastomer compatibility guide - continues next page

Name of Fluid Pumped	Elastomer Material				Viscous Behaviour Type
	NBR	EPDM	FPM	PTFE/ FFPM	
FININGS		✓	✓	✓	Pseudoplastic
FIRE FIGHTING FOAM			✓	✓	Pseudoplastic
FISH OIL			✓	✓	Newtonian
FONDANT		✓	✓	✓	Pseudoplastic
FORMIC ACID		✓		✓	Newtonian
FROMAGE FRAIS		✓	✓	✓	Pseudoplastic
FRUCTOSE		✓	✓	✓	Newtonian
FRUIT JUICE CONCENTRATE		✓	✓	✓	Pseudoplastic
FRUIT PUREE		✓	✓	✓	Pseudoplastic
FUDGE		✓	✓	✓	Pseudoplastic
GELATINE		✓	✓	✓	Pseudoplastic
GLUCOSE		✓	✓	✓	Newtonian
GLYCERINE	✓	✓	✓	✓	Newtonian
GREASE	✓		✓	✓	Pseudoplastic
GYPSUM SLURRY	✓	✓	✓	✓	Pseudoplastic
HAIR CONDITIONER			✓	✓	Pseudoplastic
HAIR GEL			✓	✓	Pseudoplastic
HAND CLEANSER			✓	✓	Pseudoplastic
HONEY		✓	✓	✓	Pseudoplastic
HYDROCHLORIC ACID			✓	✓	Newtonian
HYDROGEN PEROXIDE			✓	✓	Newtonian
ICE CREAM MIX		✓	✓	✓	Pseudoplastic
INK - PRINTING			✓	✓	Pseudoplastic
INK - WATER BASED		✓	✓	✓	Newtonian
ISOBUTYL ALCOHOL		✓	✓	✓	Newtonian
ISOCYANATE				✓	Newtonian
ISOPROPANOL		✓	✓	✓	Newtonian
JAM		✓	✓	✓	Pseudoplastic
KEROSENE	✓		✓	✓	Newtonian
LACTIC ACID			✓	✓	Newtonian
LACTOSE		✓	✓	✓	Newtonian
LANOLIN			✓	✓	Newtonian
LATEX		✓		✓	Pseudoplastic
LECITHIN			✓	✓	Newtonian
LIPSTICK			✓	✓	Pseudoplastic
LIQUORICE			✓	✓	Pseudoplastic
MAGMA		✓	✓	✓	Pseudoplastic
MAIZE STARCH SLURRY	✓	✓	✓	✓	Pseudoplastic
MALT EXTRACT		✓	✓	✓	Pseudoplastic
MANGANESE NITRATE			✓	✓	Newtonian
MASCARA			✓	✓	Pseudoplastic
MASHED POTATO		✓	✓	✓	Pseudoplastic

Table 14.10a Elastomer compatibility guide - continues next page

Name of Fluid Pumped	Elastomer Material				Viscous Behaviour Type
	NBR	EPDM	FPM	PTFE/ FFPM	
MASSECUITE		✓	✓	✓	Pseudoplastic
MAYONNAISE				✓	Pseudoplastic
MEAT PASTE		✓	✓	✓	Pseudoplastic
METHANOL	✓	✓		✓	Newtonian
METHYL ETHYL KETONE SOLVENT		✓		✓	Newtonian
METHYLATED SPIRIT	✓	✓		✓	Newtonian
METHYLENE CHLORIDE			✓	✓	Newtonian
MILK		✓	✓	✓	Newtonian
MINCEMEAT		✓	✓	✓	Pseudoplastic
MINERAL OIL	✓		✓	✓	Newtonian
MOLASSES		✓	✓	✓	Newtonian
MUSTARD		✓	✓	✓	Pseudoplastic
NEAT SOAP		✓		✓	Pseudoplastic
NITRIC ACID			✓	✓	Newtonian
PAINTS - SOLVENT BASED				✓	Pseudoplastic
PAINTS - WATER BASED	✓	✓	✓	✓	Pseudoplastic
PAPER COATING - CLAY		✓		✓	Pseudoplastic (†)
PAPER COATING - PIGMENT	✓	✓	✓	✓	Pseudoplastic (†)
PAPER COATING - STARCH	✓	✓	✓	✓	Pseudoplastic
PAPER PULP	✓	✓	✓	✓	Pseudoplastic
PEANUT BUTTER			✓	✓	Pseudoplastic
PERACETIC ACID				✓	Newtonian
PETFOOD		✓	✓	✓	Pseudoplastic
PETROLEUM	✓		✓	✓	Newtonian
PHOSPHORIC ACID		✓		✓	Newtonian
PHOTOGRAPHIC EMULSION		✓	✓	✓	Pseudoplastic
PLASTISOL		✓		✓	Newtonian
POLYETHYLENE GLYCOL		✓	✓	✓	Newtonian
POLYVINYL ALCOHOL		✓	✓	✓	Pseudoplastic
POTASSIUM HYDROXIDE		✓		✓	Newtonian
PROPIONIC ACID				✓	Newtonian
PROPYLENE GLYCOL	✓	✓	✓	✓	Newtonian
QUARG		✓	✓	✓	Pseudoplastic
RESIN			✓	✓	Newtonian
RUBBER SOLUTION				✓	Pseudoplastic
SAUCE - CONFECTIONERY			✓	✓	Pseudoplastic
SAUCE - VEGETABLE		✓	✓	✓	Pseudoplastic
SAUSAGE MEAT		✓	✓	✓	Pseudoplastic
SEWAGE SLUDGE	✓	✓	✓	✓	Pseudoplastic
SHAMPOO			✓	✓	Pseudoplastic
SHAVING CREAM			✓	✓	Pseudoplastic
SILICONE OIL	✓	✓	✓	✓	Newtonian

Table 14.10a Elastomer compatibility guide - continues next page

Name of Fluid Pumped	Elastomer Material				Viscous Behaviour Type
	NBR	EPDM	FPM	PTFE/ FFPM	
SODIUM HYDROXIDE		✓		✓	Newtonian
SODIUM SILICATE		✓	✓	✓	Newtonian
SORBIC ACID				✓	Newtonian
SORBITOL	✓	✓	✓	✓	Newtonian
STARCH		✓	✓	✓	Pseudoplastic
SUGAR PULP - BEET		✓	✓	✓	Pseudoplastic
SUGAR PULP - CANE		✓	✓	✓	Pseudoplastic
SUGAR SYRUP		✓	✓	✓	Newtonian
SULPHURIC ACID			✓	✓	Newtonian
TALL OIL			✓	✓	Newtonian
TALLOW			✓	✓	Newtonian
TITANIUM DIOXIDE	✓	✓	✓	✓	Pseudoplastic (†)
TOBACCO FLAVOURING		✓			Newtonian
TOLUENE			✓	✓	Newtonian
TOMATO KETCHUP		✓	✓	✓	Pseudoplastic
TOMATO PUREE		✓	✓	✓	Pseudoplastic
TOOTHPASTE			✓	✓	Pseudoplastic
TRUB	✓	✓	✓	✓	Pseudoplastic
UREA		✓	✓	✓	Newtonian
VARNISH				✓	Newtonian
VASELINE	✓		✓	✓	Pseudoplastic
VEGETABLE GUM		✓	✓	✓	Pseudoplastic
VEGETABLE OIL			✓	✓	Newtonian
VITAMIN SOLUTION		✓	✓	✓	Newtonian
WATER	✓	✓	✓	✓	Newtonian
WAX			✓	✓	Newtonian
WHEY		✓	✓	✓	Newtonian
WHITE SPIRIT			✓	✓	Newtonian
WINE		✓	✓	✓	Newtonian
WORT		✓	✓	✓	Newtonian
XYLENE			✓	✓	Newtonian
YEAST		✓	✓	✓	Pseudoplastic
YOGHURT		✓	✓	✓	Pseudoplastic
ZEOLITE SLURRY	✓	✓	✓	✓	Pseudoplastic (†)
ZIRCONIA SLURRY	✓	✓	✓	✓	Pseudoplastic (†)

Table 14.10a Elastomer compatibility guide

This chapter explains the various terms found in this handbook.

15

Glossary of Terms

A

Absolute Pressure	Total pressure exerted by a fluid i.e., atmospheric pressure plus gauge pressure
Absolute Viscosity	Measure of how resistive the flow of a fluid is between two layers of fluid in motion
Adaptor	Connection piece between the motor and back plate on a centrifugal and liquid ring pump
Anti-thixotropic	Fluid viscosity increases with time under shear conditions
Air-screw	A screw type Impeller fitted in the offset priming chamber to assist in evacuating air or gas

B

Back Plate	Part of a centrifugal and liquid ring pump, which together with the pump casing forms the fluid chamber

C

Cartridge seal	NO EXPLANATION in Word
Cavitation	Vacuous space in the inlet port of a pump normally occupied by fluid
CPP	Circumferential Piston Pump (CPP) has a pair of rotating winged rotors (pistons) moving liquid around the circumference of the casing channel
Centrifugal	Tending to move out from the center
Chamber	For Twin Screw pumps: The chamber is the free distance between the rear of the helix and the front of the next helix in which product is moved, to this extent the chamber size determines the maximum particle size for solids handling
CIP	Cleaning In Place - ability to clean pump system without dismantling pump and system
Clearflow	Special Impeller design on LKH Evap pumps to prevent crystalline build up on pump backplate
CM	Condition Monitor – measures vibrations and temperature
CM Connect	Gateway for sending data from up to 10 CM, condition monitors, to the GSM, Gateway for sending data to the Cloud, from network on CM's

D

Dead Head Speed	Pump speed required to overcome slip for a rotary lobe pump
Density	Fluids mass per unit of volume
Differential Pressure	Total absolute pressure differences across the pump during operation i.e., discharge pressure minus suction pressure
Diffusion Hardening	A process used in manufacturing that increases the hardness of steels. Diffusion only happens through a small thickness of a piece of steel, so only the surface is hardened while the core maintains its original mechanical properties
Dilatant	Fluid viscosity increases as shear rate increases
Discharge Pressure	Pressure at which fluid is leaving the pump
Duty Point	Intersection point between the pump curve and the process curve
Dynamic Head	Energy required to set fluid in motion and to overcome any resistance to that motion

E

Elastomer	Non-metallic sealing device that exhibits elastic strain characteristics
Electropolishing	Method of surface finishing achieved by an electro-chemical process

F

Feed Screw	Helical Screw rotor for the Twin Screw pump
Flooded Suction	Positive inlet pressure/head
Friction Head	Pressure drop on both inlet and discharge sides of the pump due to frictional losses in fluid flow

G

Gauge Pressure	Pressure within a gauge that exceeds the surrounding atmospheric pressure, using atmospheric pressure as a zero reference

H

Hydraulic Power	Theoretical energy required to pump a given quantity of fluid against a given total head

I

Impeller	Pumping element of a centrifugal and liquid ring pump
Inlet Pressure	Pressure at which fluid is entering the pump

K

Kinematic Viscosity	Measure of how resistive the flow of a fluid is under the influence of gravity

L

Laminar Flow	Flow characteristic whereby the fluid moves through the pipe in concentric layers with its maximum velocity in the center of the pipe, decreasing to zero at the pipe wall

M

Multi-stage	A pump with more than one impeller mounted on the same shaft and connected so as to act in series

N

Newtonian	Fluid viscosity is constant with change in shear rate or agitation
NPSH	Net Positive Suction Head describing the inlet condition of a pump and system
NPSHa	Net Positive Suction Head available in a system
NPSHr	Net Positive Suction Head required from a pump
NIPA	Net Inlet Pressure Available in a system
NIPR	Net Inlet Pressure Required from a pump
Non-Product Wetted	Metallic and elastomeric components not in contact with the fluid being pumped

O

Outlet Pressure	Pressure at which fluid is leaving the pump

P

Pitch	The pitch length which is basically the dimension from the rear face of the of the helix to the front of the next helix
Positive Displacement	Pump type whereby the fluid pumped is directly displaced
Pressure Drop	Result of frictional losses in pipework, fittings, and other process equipment
Pressure Shock	Result of change in fluid velocity
Product Wetted	Metallic and elastomeric components in contact with the fluid being pumped
Profiled Seal Ring	Seal ring with optimized hygienic design minimizing crevasses for easy CIP cleaning
Pseudoplastic	Fluid viscosity decreases as shear rate increases
Pump Casing	Part of a centrifugal and liquid ring pump, which together with the back plate forms the fluid chamber

R

Required Power	Power needed at the pump shaft
Reynolds Number (Re)	Ratio of inertia forces to viscous forces giving a value to determine type of flow characteristic
Rheology	Science of fluid flow
Rheomalactic	Fluid viscosity decreases with time under shear conditions but does not recover
Rotodynamic	A machine to transfer rotating mechanical energy into kinetic energy in the form of fluid velocity and pressure
Rotor	Pumping element of a rotary lobe pump
Rotor Case	Part of a rotary lobe pump, which together with the rotor case cover forms the pump chamber
Rotor Case Cover	Part of a rotary lobe pump, which together with the rotor case forms the pump chamber
Rumbling	Method of surface finishing achieved by vibrating components with abrasive particulate

S

Shotblasting	Method of surface finishing achieved by blasting finished components with small metallic particles at great force
SIP	Steam or Sterilisation In Place - ability to steam clean or sterilise pump system without dismantling pump and system
Slip	Fluid lost by leakage through the pump clearances of a rotary lobe pump
Specific Gravity	Ratio of a fluid's density to the density of water
Specific Weight	Fluid's weight per unit volume

S

Static Head	Difference in fluid levels
Static Discharge Head	Difference in height between the fluid level and the centre line of the pump inlet on the discharge side of the pump
Static Suction Head	Difference in height between the fluid level and the centre line of the pump inlet on the inlet side of the pump
Suction Lift	Fluid level is below the centre line of the pump inlet
Suction Pressure	Pressure at which fluid is entering the pump

T

Thermal Shock	Rapid temperature change of pump head components
Thixotropic	Fluid viscosity decreases with time under shear conditions
Timing Gear	Setting the timing between the Rotors or the Feed Screws in the gearbox
TLA	Timing gear location via Torque Locking Assembly (TLA) providing full 360° uniform loading and easy time setting
Torque	Moment of force required to produce rotation
Total Discharge Head	Sum of the static discharge and dynamic heads
Total Efficiency	Relationship between the input power at the pump shaft and output power in the form of water horsepower
Total Head	Total pressure difference between the total discharge head and the total suction head of the pump
Total Static Head	Difference in height between the static discharge head and the static suction head
Total Suction Head	Static suction head less the dynamic head
Transitional Flow	Flow characteristic combining both laminar and turbulent flow tendencies
TS Pump	Twin Screw Pump
Turbulent Flow	Flow characteristic whereby considerable mixing of the fluid takes place across a pipe section with velocity remaining fairly constant

V

Vacuum	Pressure in a pumping system below normal atmospheric pressure
Vapour Pressure	Pressure at which a fluid will change to a vapour, at a given temperature
Velocity	Distance a fluid moves per unit of time
Viscosity	Measure of how resistive a fluid is to flow
Viscous Power	Power loss due to viscous fluid friction within the pump
Volumetric Efficiency	Ratio of actual capacity against theoretical capacity

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