Irrigation Manual
Module 5

Irrigation Pumping Plant

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List of abbreviations

AC	Asbestos Cement
ASAE	American Society of Agricultural Engineers
BP	Brake Power
d	inside pipe diameter
e	vapour pressure of water
E	Efficiency
E	Elasticity of pipe material
EFF	Efficiency
fps	feet per second
g	gravitational force
g	gallon
H	Head
HP	Horse Power
Kpa	Kilopascal
kW	kilowatt
L	Length
N	Speed
NPSHA	Net Positive Suction Head Available
NPSHR	Net Positive Suction Head Required
P	Pressure
PE	Polyethylen
PVC	Polyvinyl Chloride
Q	Discharge
rpm	revolutions per minute
SEC	State Electricity Commission
t	pipe wall thickness
T	Time
TDH	Total Dynamic Head
uPVC	unplasticized Polyvinyl Chloride
V	Velocity
WP	Water Power
Z	Elevation
ZITC	Zimbabwe Irrigation Technology Centre

Chapter 1
Introduction

Most irrigation pumps fall within the category of pumps that use kinetic principles, that is centrifugal force or momentum, in transferring energy. This category includes pumps such as centrifugal pumps, vertical turbine pumps, submersible pumps and jet pumps. Most of these pumps operate within a range of discharge and head where the discharge will vary as the head fluctuates.

The second category of pumps is that of positive displacement pumps, whereby the fluid is displaced by mechanical devices such as pistons, plungers and screws. Mono pumps, treadle pumps and most of the manual pumps fall into this category.

Allahwerdi (1986) calls the first category of pumps turbo pumps and depending on the type of discharge subdivides these pumps into:

- Radial flow pumps (centrifugal action)
- Axial flow pumps (propeller-type action)
- Mixed flow pumps (variation of both)

It should be noted that Allahwerdi's classification does not include positive displacement pumps.

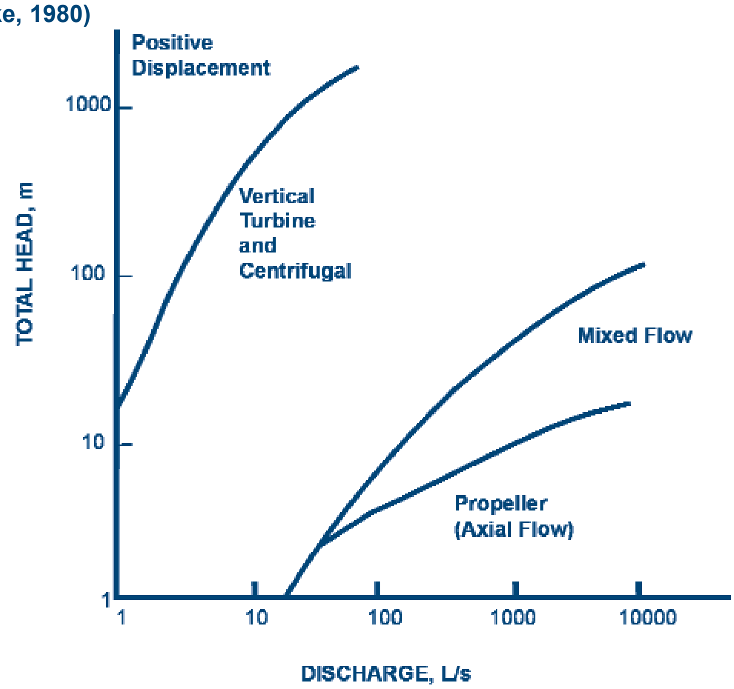
Longenbaugh and Duke (1980) classify pumps into:

- Vertical turbine and centrifugal pumps
- Propeller or axial flow pumps
- Mixed flow pumps
- Positive displacement pumps

Figure 1 shows this classification as a function of the total operating head and discharge. The schematic classification employed by the State Electricity Commission (SEC) is shown in Figure 2 and the one employed by the Hydraulic Institute in Figure 3.

Positive displacement pumps are as a rule suitable for small discharges and high heads and the head is independent of the pump speed. Some types of these pumps should only be used with water free of sediments. The vertical turbine and the centrifugal pumps fit conditions of moderately small to high discharges and moderately low to high heads. These are the most commonly used pumps in irrigation. They can operate with reasonable amounts of sediments, but periodic replacement of impellers and volute casing should

Figure 1
Sub-classification of pump types as a function of operating head and discharge (Adapted from Longenbaugh and Duke, 1980)



be anticipated. Turbine pumps are more susceptible to sediments than centrifugal pumps. Mixed flow pumps cover a good range, from moderately large to large discharges, and moderately high heads. They have the same susceptibility to sediments as do centrifugal pumps. Axial flow pumps are suitable for low heads and large discharges.

Figure 2

Schematic classification of pump types by the State Electricity Commission in 1965 (Source: T-Tape, 1994)

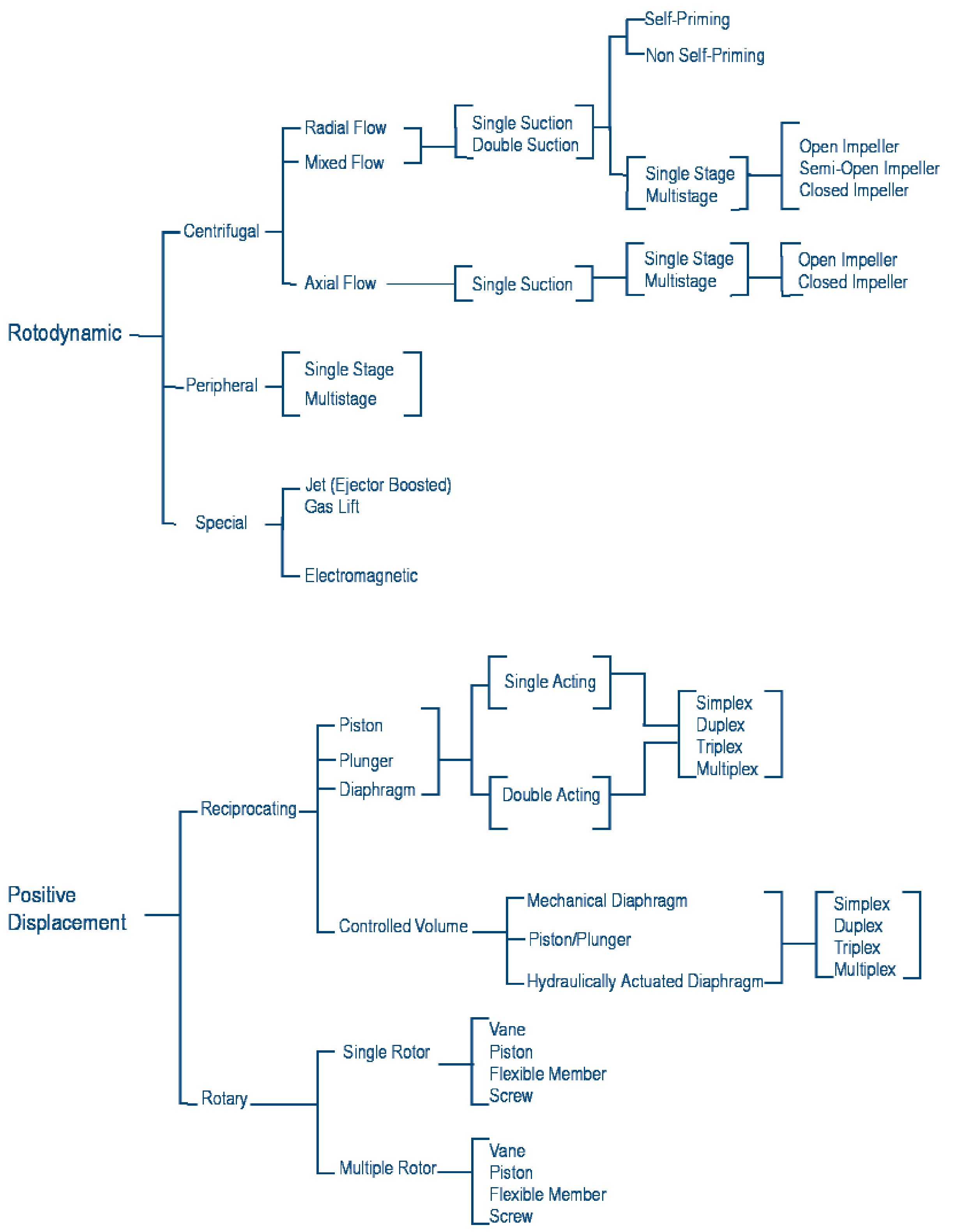
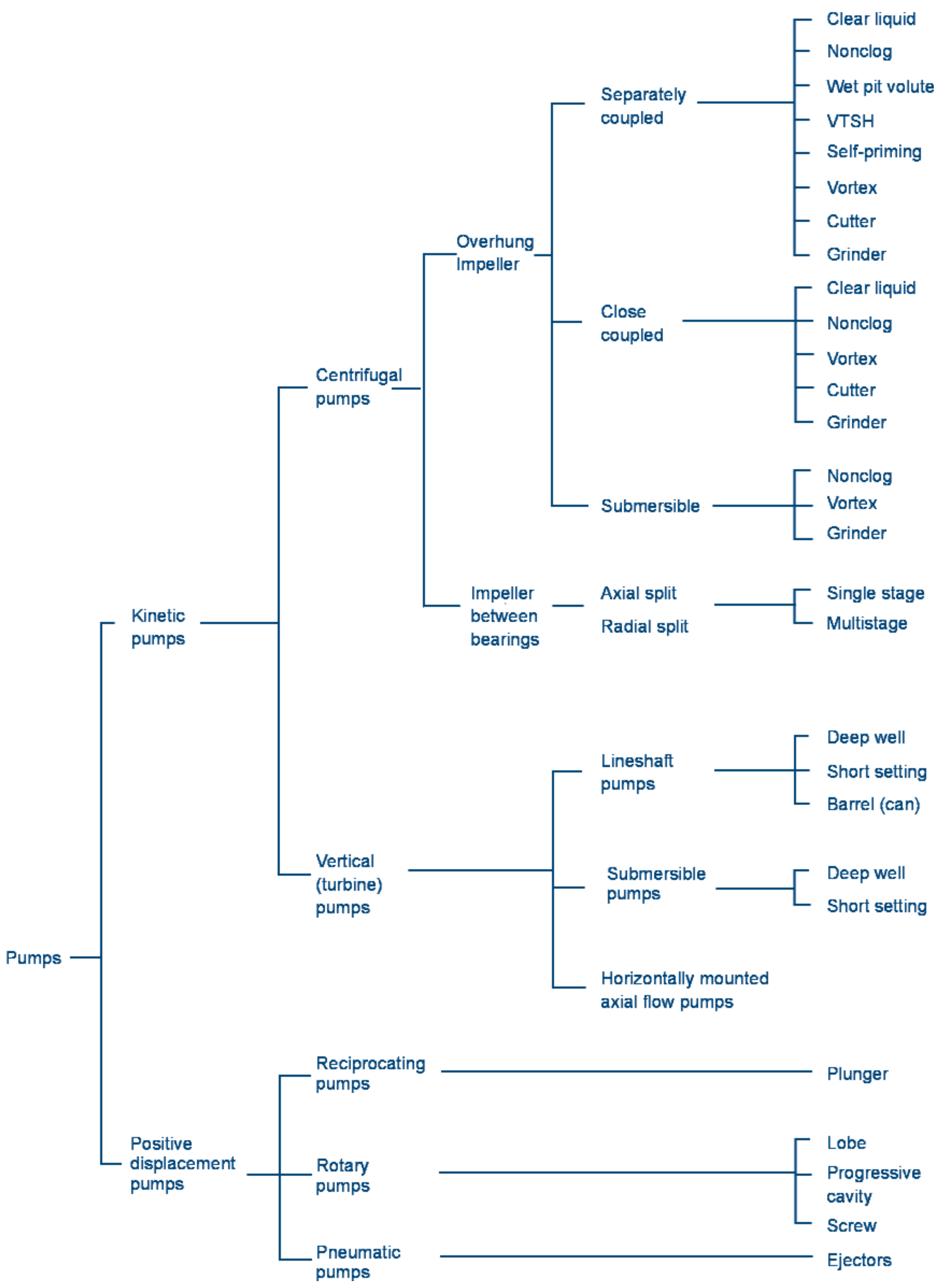


Figure 3

Schematic classification of pump types by the Hydraulic Institute in 1983 (Source: T-Tape, 1994)



Chapter 2
Total dynamic head or total pumping head

Head is the expression of the potential energy imparted to a liquid to move it from one level to another. Total dynamic head or total pumping head is the head that the pump is required to impart to a fluid in order to meet the head requirement of a particular system, whether this be a town water supply system or an irrigation system. The total dynamic head is made up of static suction lift or static suction head, static discharge head, total static head, required pressure head, friction head and velocity head. Figure 4 shows the various components making up the total dynamic head.

2.1. Static suction head or static suction lift

When a pump is installed such that the level of the water source is above the eye of the impeller (flooded suction), then the system is said to have a positive suction head at the eye of the impeller. However, when the pump is installed above the water source, the vertical distance from the surface of the water to the eye of the impeller is called the static suction lift.

2.2. Static discharge head

This is the vertical distance or difference in elevation between the point at which water leaves the impeller and the point at which water leaves the system, for example the outlet of the highest sprinkler in an overhead irrigation system.

2.3. Total static head

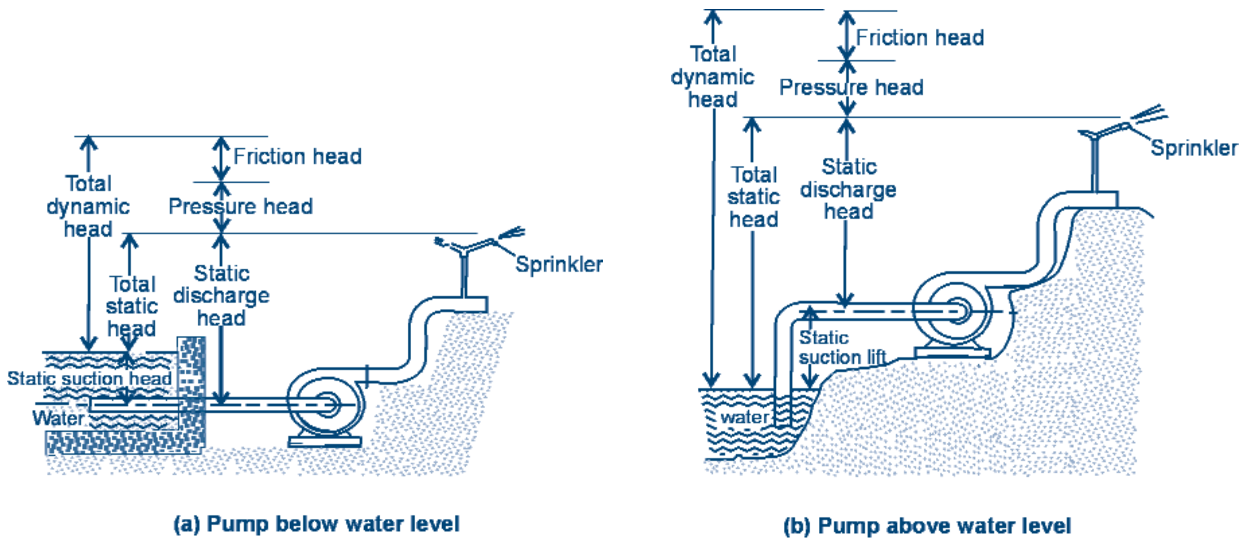
When no water is flowing (static conditions), the head required to move a drop of water from a (water source) to b (the highest sprinkler or outlet point) is equal to the total static head. This is simply the difference in elevation between where we want the water and where it is now.

For systems with the water level above the pump, the total static head is the difference between the elevations of the water and the sprinkler (Figure 4a).

Total Static Head = Static Discharge Head – Static Suction Head

For systems where the water level is below the pump, the

Figure 4
Components of total dynamic head (Source: Australia Irrigation Association, 1998)



total static head is the static discharge head plus the static suction lift (Figure 4b).

Total Static Head = Static Discharge Head + Static Suction Lift

2.4. Friction head

When water flows through a pipe, the pressure decreases because of the friction against the walls of the pipe. Therefore, the pump needs to provide the necessary energy to the water to overcome the friction losses. The losses must be considered both for the suction part and the discharge part of the pump. The magnitude of the friction head can be calculated using either hydraulic formulae or tables and graphs.

2.5. Pressure head

Except for the cases where water is discharged to a reservoir, or a canal, a certain head to operate an irrigation system is required. For example, in order for a sprinkler system to operate, a certain head is required.

2.6. Velocity head

This energy component is not shown in Figure 4. It is very small and is normally not included in practical pressure calculations. Most of the energy that a pump adds to flowing water is converted to pressure in the water. Some of the energy is added to the water to give the velocity it requires to move through the pipeline. The faster the water is moving the larger the velocity head. The amount of energy that is needed to move water with a certain velocity is given by the formula:

Equation 1

$$\text{Velocity Head} = \frac{V^2}{2g}$$

Where:

V = the velocity of the water (m/s)

g = the gravitational force which is equal to 9.81 (m/s^2)

Keller and Bliesner (1990) recommend that for centrifugal pumps the diameter of the suction pipe should be selected such that the water velocity $V < 3.3$ m/s in order to assure good pump performance. Assuming this maximum velocity for the flow and applying the above formula, then the velocity head corresponding to the minimum diameter of the suction pipe that can be selected to satisfy this condition is 0.56 m/sec ($3.3^2/(2 \times 9.81)$).

2.7. Drawdown

Usually, the level of the water in a well or even a reservoir behind a dam does not remain constant. In the case of a well, after pumping starts with a certain discharge, the water level lowers. This lowering of the water level is called drawdown. In the case of a dam or reservoir, fluctuation of the water level is common and depends on water inflow, evaporation and water withdrawal. The water level increases during the rainy season, followed by a decrease during the dry season because of evaporation and withdrawal of the stored water. This variation in water level will affect the static suction lift or the static suction head and, correspondingly, the total static head.

Chapter 3
Types of pumps and principles of operation

3.1. Radial flow pumps

Radial flow pumps are based on the principles of centrifugal force and are subdivided into volute pumps and diffuser (turbine) pumps.

3.1.1 Volute pumps

The well-known horizontal centrifugal pump is a volute pump. The pump consists of two main parts, the propeller that rotates on a shaft and gives the water a spiral motion, and the pump casing that directs the water to the impeller through the volute and eventually to the outlet. The suction entrance of the casing is in such a position that the water enters the eye of the impeller. The water is then pushed outwards because of the centrifugal force caused by the rotating impeller. The centrifugal force, converted to velocity head and thus pressure, pushes the water to the outlet of the volute casing. Figure 5 shows the components of a typical centrifugal pump.

Figure 6 shows the impeller inside the volute casing and the three types of impellers commonly used in centrifugal pumps. Closed impellers develop higher efficiencies in high-pressure pumps. The other two types are more able to pass solids that may be present in the water.

Volute pumps may be classified under three major categories (Figure 7):

- ❖ Low head, where the impeller eye diameter is relatively large compared with the impeller rim diameter
- ❖ Medium head, where the impeller eye diameter is a small proportion of the impeller rim diameter
- ❖ High head, where the impeller rim diameter is relatively much larger than the impeller eye diameter

3.1.2. Diffuser or turbine pumps

The major difference between the volute centrifugal pumps and the turbine pumps is the device used to receive the water after it leaves the impeller.

In the case of the turbine pumps, the receiving devices are diffuser vanes that surround the impeller and provide diverging passages to direct the water and change the velocity energy to pressure energy. Deep well turbine pumps and submersible pumps use this principle.

Figure 5
Cross-section of a centrifugal pump (Source: Miller, 1991)

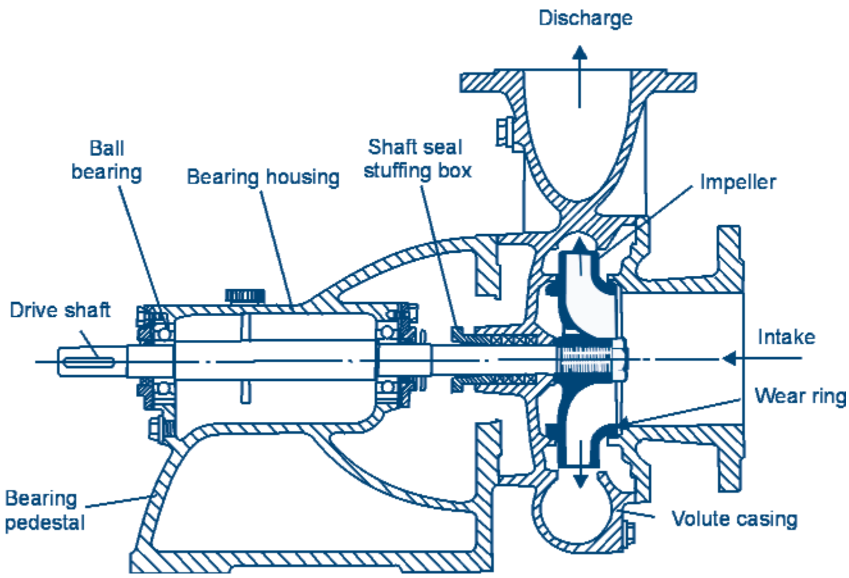
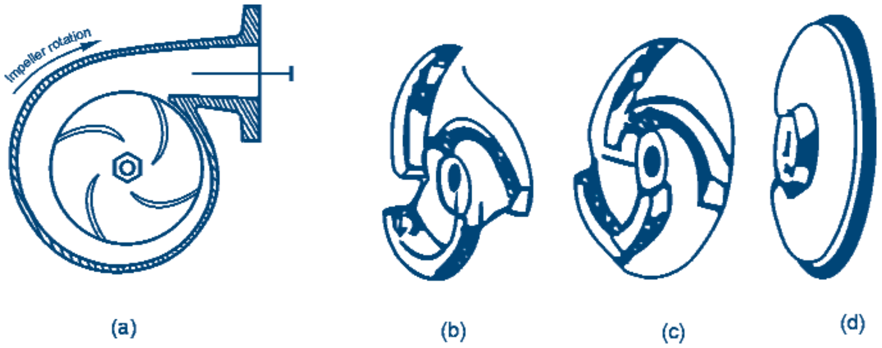


Figure 6

Pump impellers and volute casing (Source: T-Tape, 1994)



(a) Impeller inside volute casing

(b) Open impeller

(c) Semi-open impeller

(d) Closed impeller

Figure 7

Classification of volute pumps based on the impeller proportions (Source: T-Tape, 1994)

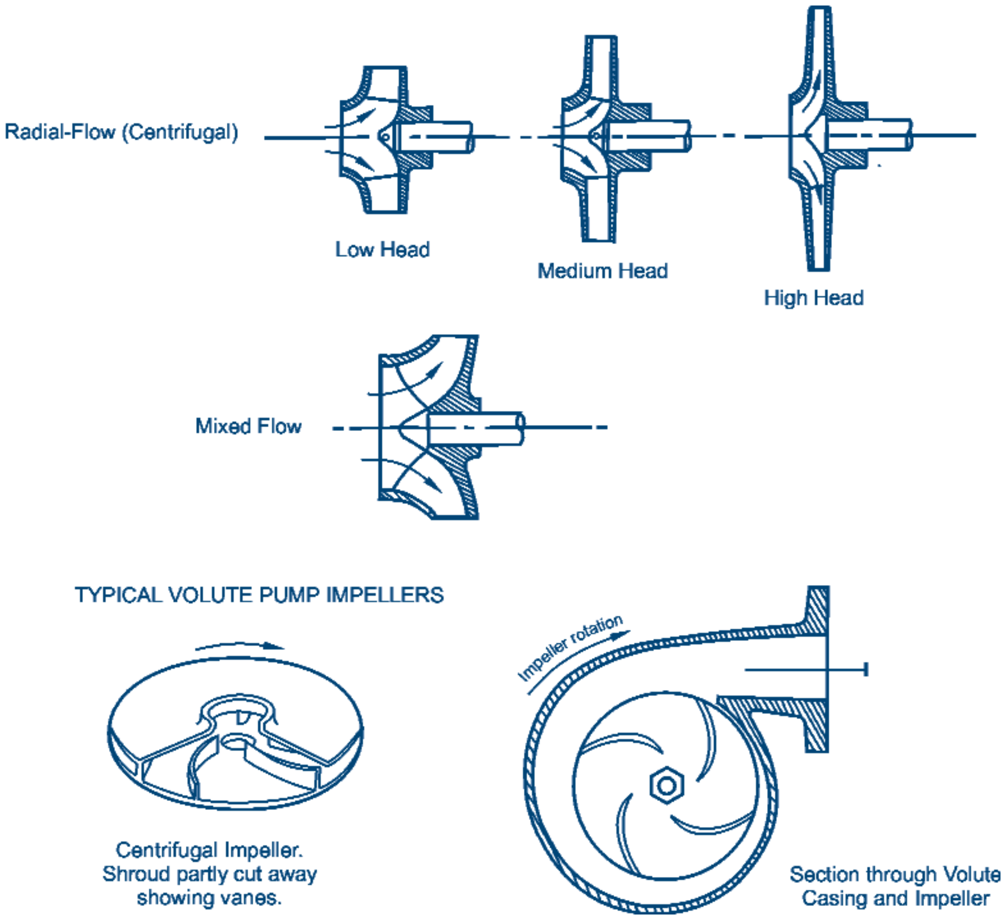


Figure 8
Parts of bowl assembly (Source: Grundfos, undated)

Figure 9

Different drive configurations (Source: Grundfos, undated)

Electric Motor Drive

The electric motor is a vertical flange mounted, totally enclosed fan-cooled squirrel-cage induction type manufactured in accordance with (IEC) international standards. Up to and including 30 kW (40 HP) the motors are fitted with special bearings to absorb the axial thrust. For 37 kW (50 HP) and larger motors a special intermediate housing with coupling and bearings is fitted between the pump head and the motor for the same purpose.
The GRUNDFOS deep well turbine pumps are available with electric motors up to 45 kW (60 HP) in standard design and motors up to 75 kW (100 HP) in special design.

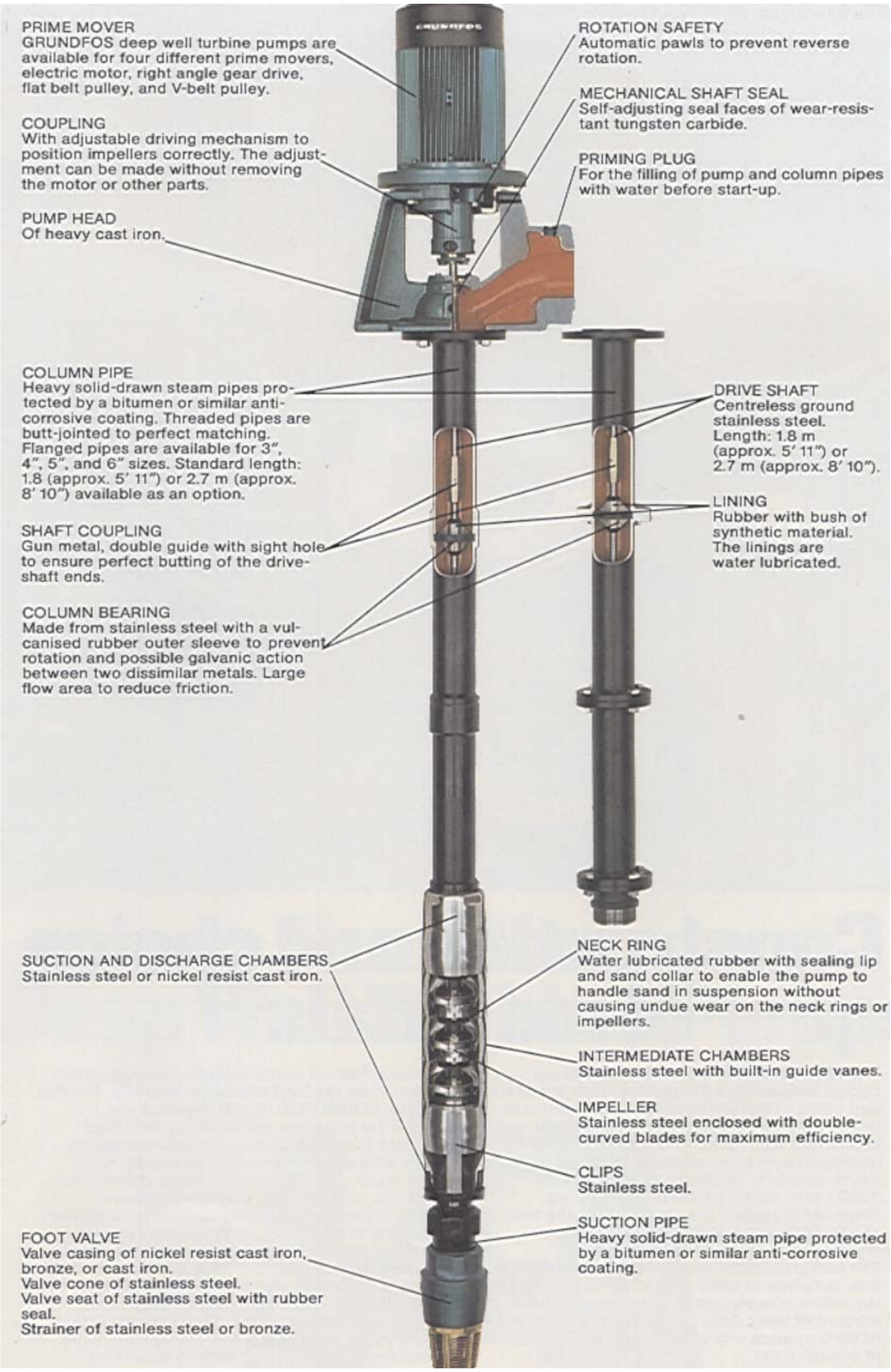
Right Angle Gear Drive

The right angle gear drive is made from high specification materials to ensure a long and trouble free life.
The gear housing is made from rigid cast iron and constructed to ensure constant correct alignment of the gear wheels at maximum load. The hardened, ground, and paired gear wheels are kept in the correct position by angular contact bearings. In the construction of the gear wheels and the selection of the bearings, the aim has been to ensure long life with quiet operation.
The gear wheels and the bearings are oil lubricated by means of an oil pump positioned at the bottom of the gear housing. The largest gear types are water cooled as standard, and the smaller types are available with water cooling by request.
The vertical main shaft in the smaller types is of a solid construction, and the coupling between the main shaft and the column pipe shaft is positioned in the pump head. This means that the adjustment of the pump impellers can be made in the pump head without removing the gear dome.
In the larger types the main shaft is of a hollow construction, and the coupling between the main shaft and the column pipe shaft is positioned in the gear top, where the adjustment of the impellers is also made.
The right angle gears are available in four sizes from max. 22 kW (30 HP) to max. 92 kW (125 HP) nominal power at a ratio of 1:2 and a max. speed of 3500 r.p.m. For other ratios and pump speeds, the nominal power is reduced accordingly.

Pulley Drive

The pulley head and the bearings are made from high quality materials to obtain a long and trouble free life and smooth quiet operation.
The flat belt pulley drive is available for pumps up to 22 kW (30 HP) and for pump speed up to 3450 r.p.m.
The pulley head drive is also available in a design for V-belt drive for special purposes.

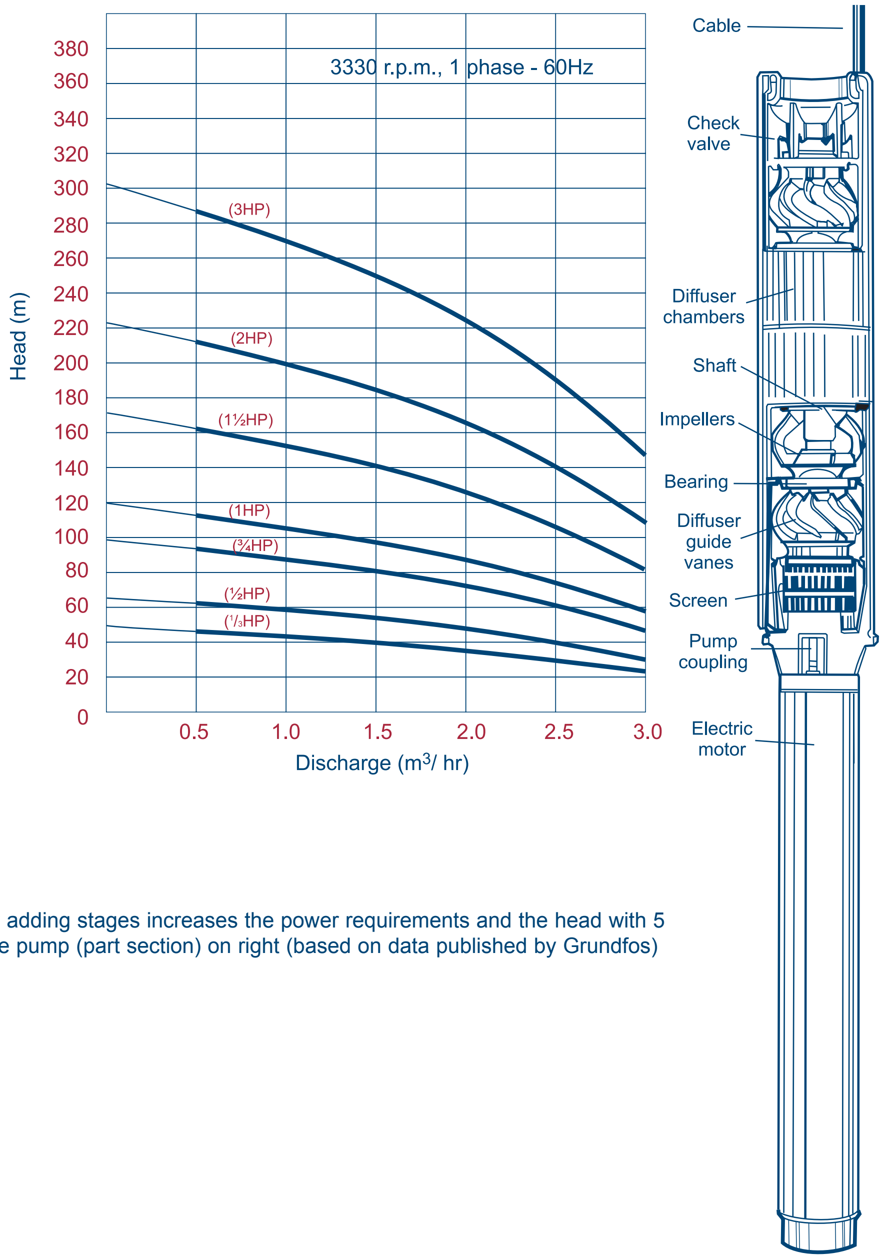
Figure 10
Electrically driven turbine pump (Source: Grundfos, undated)



Depending on the required head, these pumps have a number of impellers, each of which is enclosed with its diffuser vanes in a bowl. Several bowls form the bowl assembly that must always be submerged in water. Figure 8 shows parts of the bowl assembly. A vertical shaft rotates the impellers. In the case of turbine pumps the shaft is located in the centre of the discharge pipe. At intervals of usually 2-3 m, the shaft is supported by rubber lined water lubricated bearings. Figure 9 shows different drive configurations. Figure 10 shows a complete electrically driven turbine pump.

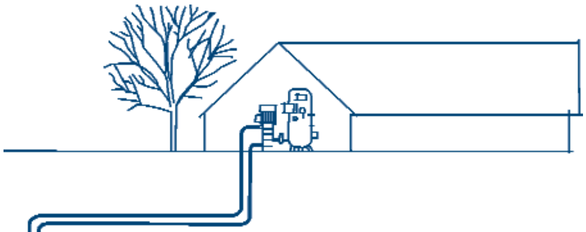
Electro-submersible pumps are turbine pumps with an electric motor attached in the suction part of the pump, providing the drive to the shaft that rotates the impellers. Therefore, there is no shaft in the discharge pipe. Both the motor and pump are submerged in the water. They are especially suitable for installation in deep boreholes. Submersible electrically driven pumps depend on cooling via the water being pumped, and a failure of the water supply can result in serious damage to the unit. For this reason submersible pumps are protected with water level cut-off switches. Figure 11 shows a complete submersible pump.

Figure 11
Cross-section through a submersible pump and submersible motor (Source: FAO, 1986)



How adding stages increases the power requirements and the head with 5 stage pump (part section) on right (based on data published by Grundfos)

Figure 12
An example of a jet pump (Source: Grundfos, undated)



This deep well pumping system is deal for small water supply plants that are to take water from depths of more than 6/8 metres.

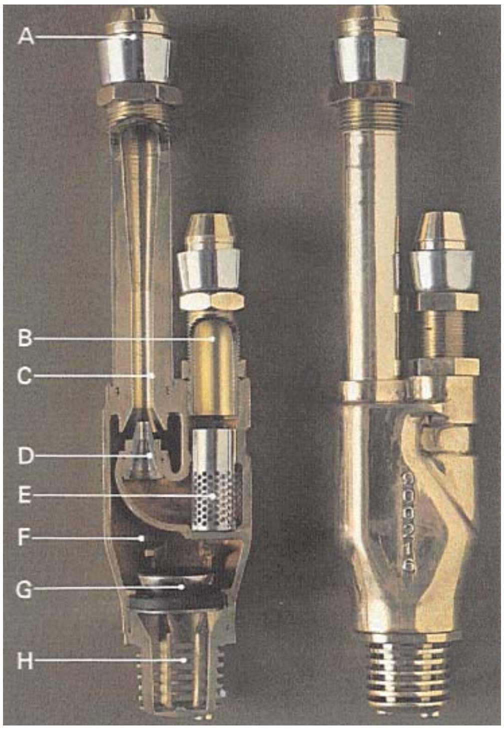
1. The ejector pump system is inexpensive to purchase initially and is quick and easy to install.
2. A separate pump house is not normally required over the borehole/well, as the pump can be installed in the top of the well or in an existing adjacent building.
3. Electric cables are not needed in the well.
4. Comparatively low noise level – an even flow of water.
5. Suitable tank pressure irrespective of depth.
6. Easy adjustment of pump output to match the well capacity.
7. The pump is easily accessible for overhaul.
8. Sturdy and reliable even where there are long horizontal pipe runs and great depths.

Operation

The **Grundfos** ejector system consists of a vertical multistage centrifugal pump connected by two pipes to ejector (see illustration) which is situated below the water level in the well. The pump has a third connection, the discharge port and its position on the pump can be varied to give varying discharge pressure to suit the application.

The method of operation is as follows. The pump supplies water at high pressure down the pressure pipe **B**, through the strainer **E** and into the nozzle **D**. In the nozzle the high pressure is converted into high velocity water jet which passes through the chamber into the diffusor **C**. The chamber is connected via the foot valve **G** and the strainer **H** to the well water.

The water in the chamber **F** is picked up by the high velocity water jet passing from the nozzle into the diffusor. Here the two water flows are mixed and the high velocity is converted into pressure, which forces the water up the riser pipe **A** into the pump suction chamber.



The use of a multistage centrifugal pump enables the discharge port to be positioned at a suitable stage to give the correct discharge pressure at maximum water output. This ensures optimum operating efficiency. At the same time the stages of the pump above the discharge port maintain the required pressure for the ejector, even when the discharge pressure falls too zero when the consumption is momentarily larger than the well capacity.

Grundfos have developed this ejector system and the present range of pumps and ejectors have evolved from many years experienced under varying conditions ranging from the far North of Scandinavia to the far South of Australia.

The ejector body is made of bronze and fitted with a wear-resistant stainless steel nozzle, which is protected against blockage by the strainer **E**. The built-in foot valve has a cone of stainless steel, seating on rubber and the strainer is made of bronze.

The wide range of **Grundfos** centrifugal pumps, ejector pumps and submersible pumps are still being enlarged and improved and on the basis of extensive research are **THE RIGHT PUMPS** for water supply.

3.2. Axial flow pumps

While the radial flow type of pump discharges the water at right angles to the axis of rotation, in the axial flow type water is propelled upwards and discharged nearly axially. The blades of the propeller are shaped somewhat like a ship's propeller. Axial flow type pumps are used for large discharges and low heads (see Figure 1).

3.3. Mixed flow pumps

This category includes pumps whereby the pressure head is developed partially through the centrifugal force and partially through the lift of the vanes on the water. The flow is discharged both axially and radially. These pumps are suitable for large discharges and medium head.

3.4. Jet pumps

This pump is a combination of a centrifugal pump and a nozzle converting high pressure into velocity (Figure 12). As such it cannot fit into one of the above categories. A high-pressure jet stream is ejected through a suitable nozzle to entrain a large volume of water at low pressure and force it to a higher level within the system. The pump has no moving parts in the well or beneath the water surface. It is composed of a multistage centrifugal pump installed above ground, an ejector installed below the water surface and connecting pipes. The disadvantage of these units is that when they are used in high head situations, the discharge and efficiency are greatly reduced. Basically such units are categorized as:

- Low head, large discharge – most efficient
- High head, low discharge – least efficient

3.5. Positive displacement pumps

3.5.1. Manual pumps

For all practical purposes, water is incompressible. Consequently, if a close-fitting piston is drawn through a pipe full of water it will displace water along the pipe (Figure 13). Similarly, raising a piston in a submerged pipe will draw water up behind it to fill the vacuum that is created, and water is actually displaced by atmospheric pressure on its external surface. Two examples of manual pumps employing these principles are described below.

Piston or bucket pumps

The most common and well-known form of displacement pump is the piston pump, also known as the bucket, hand or bush pump. A common example is

Figure 13
Basic principles of positive displacement pumps (Source: FAO, 1986)

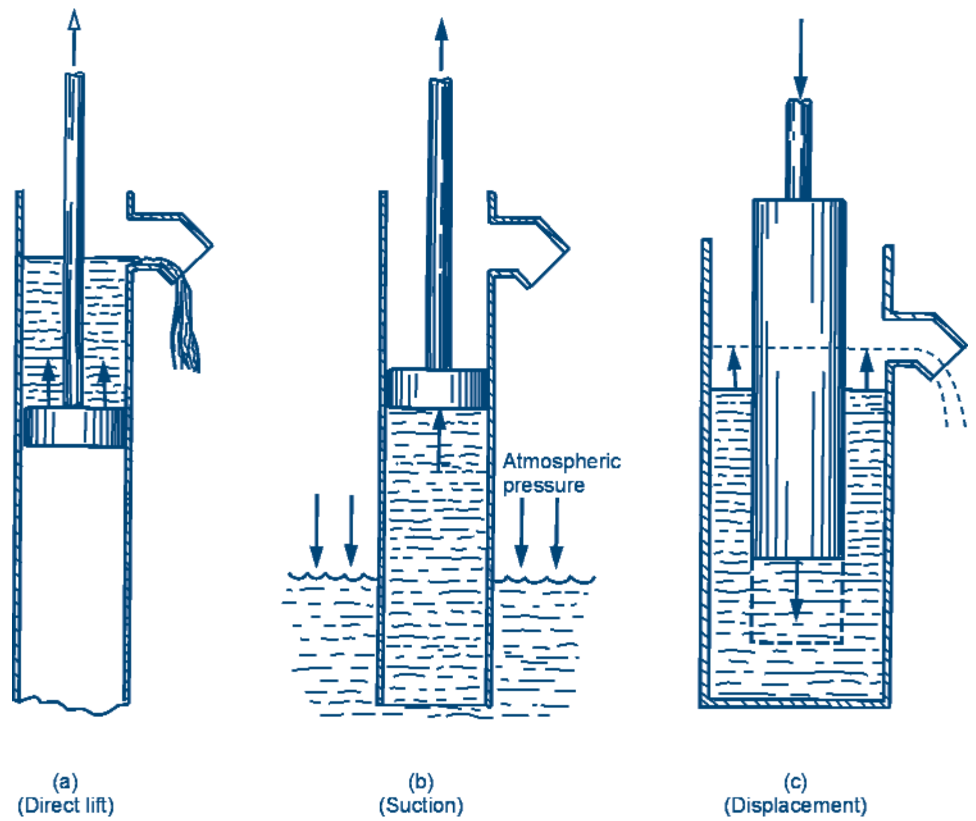
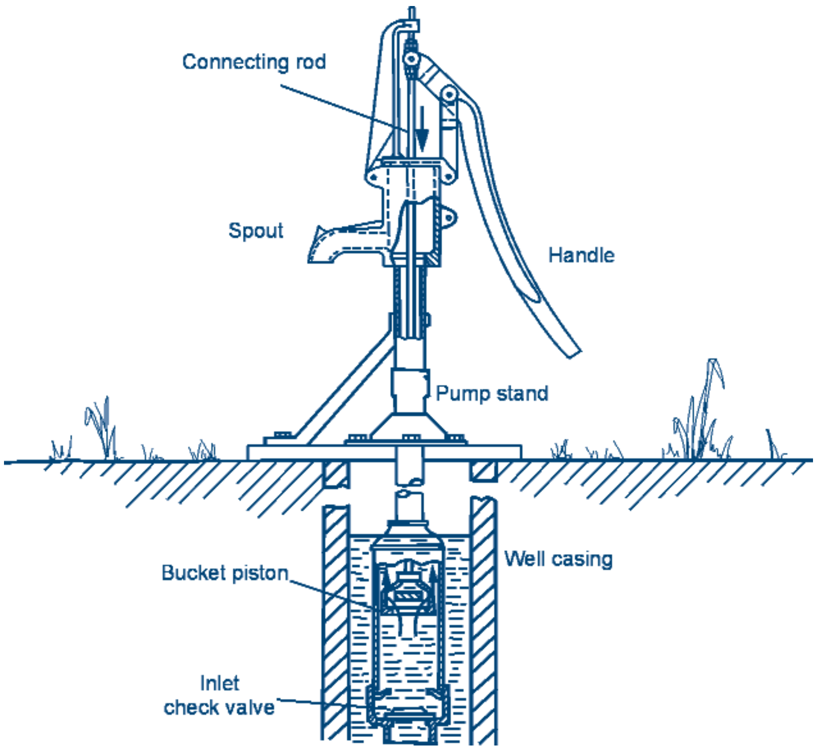


Figure 14

Hand pump with single acting bucket and piston (Adapted from FAO, 1986)



illustrated in Figure 14. Water is sucked into the cylinder through an inlet check valve or non-return valve on the upstroke, which is opened by the vacuum created. This vacuum also keeps the piston valve closed. On the down stroke, the check valve is held closed by both its weight and the water pressure. As this happens the piston valve is forced open as the trapped water is displaced through the piston ready for the next upstroke.

The piston valve has two leather cup washer seals. The outer casing and fittings are normally cast iron. While this pump is widely used in Zimbabwe for domestic water supplies, it is also used to irrigate gardens, but to a limited extent. These pumps have wide operating head ranges of 2 to 100 m depending on construction of the pump. Discharges of 15 to 25 m^3/hr or 4 to 7 l/s could be realized.

Treadle pumps

A treadle pump is another form of a positive displacement pump where the feet are used to treadle. Most treadle pumps are double acting, meaning that there is discharge on both the upstroke and downstroke. Figure 15 shows a typical double acting pressure treadle pump.

Tests carried out at the Zimbabwe Irrigation Technology Centre (ZITC) revealed that suction heads exceeding 3 m make the pump quite difficult to operate. In a similar argument, delivery heads in excess of 6 m are also not recommended. This shows that treadle pumps can only be used where there are shallow water tables. In semi arid regions, their use could be confined to vleis or dambos, where the water tables are shallow, or to draw water from dams or rivers.

Table 1 shows results of the tests carried out at ZITC on a pressure treadle pump. The data are plotted in Figure 16.

Table 1

Pressure treadle pump test analysis

Total Dynamic Head (m) (= suction head + delivery head)	Discharge (m^3/hr)
3.5	6.9
5.0	4.9
6.0	3.7

Other models of treadle pump, based on the same principles but delivering water without pressure, have been used extensively in the Indian Sub-continent, as typified by

Figure 17, and have recently been introduced in eastern and southern Africa. These types of treadle pump are also composed of two cylinders and two plungers. The pumped water, instead of being delivered at the lower part of the pump through a valve box, is delivered at the top through a small channel. Figure 17 gives the details.

Figure 15
Double acting pressure treadle pump (Source: ZITC, 1997)

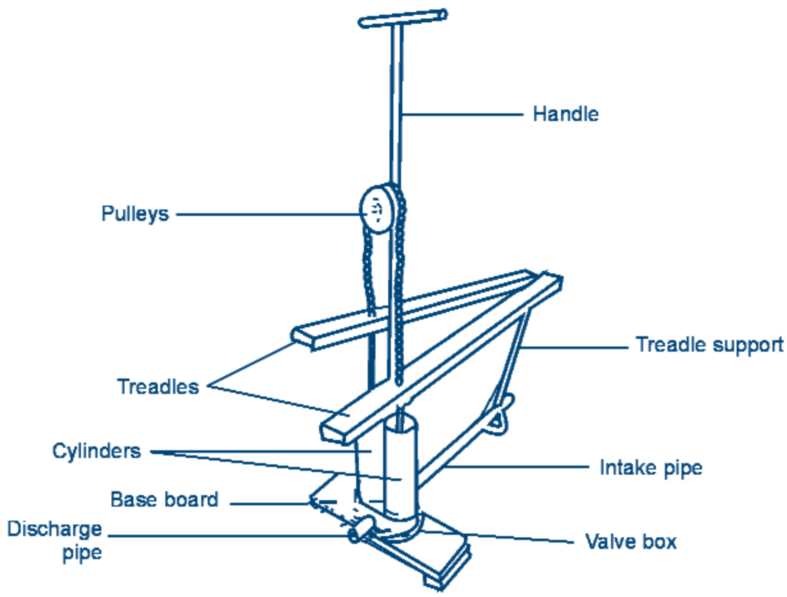


Figure 16
Discharge-head relationship for pressure treadle pump (based on Table 1)

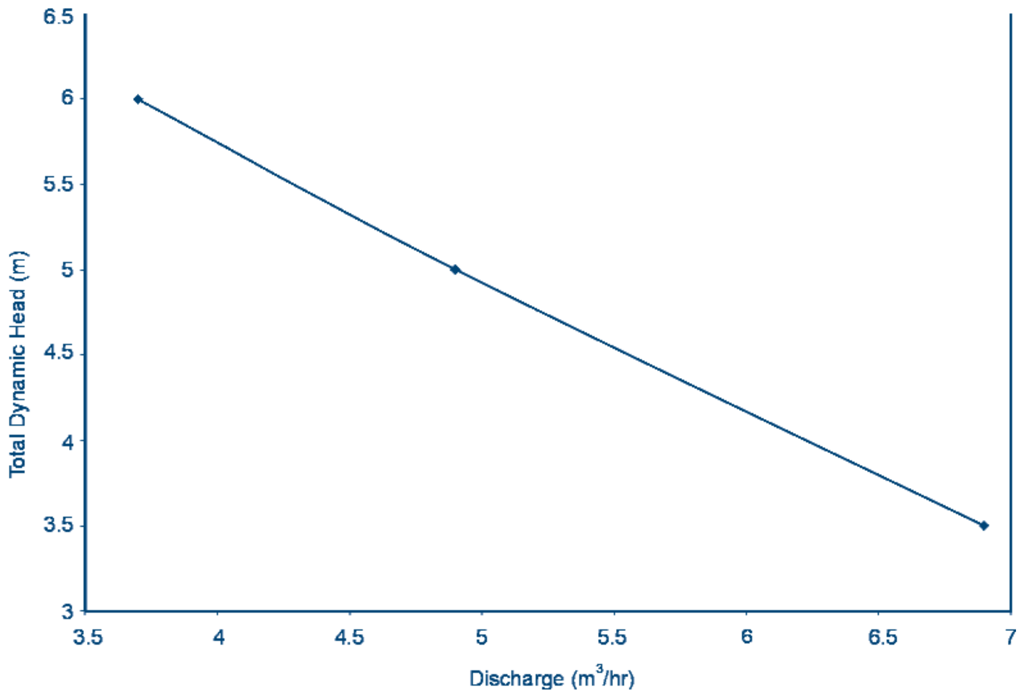
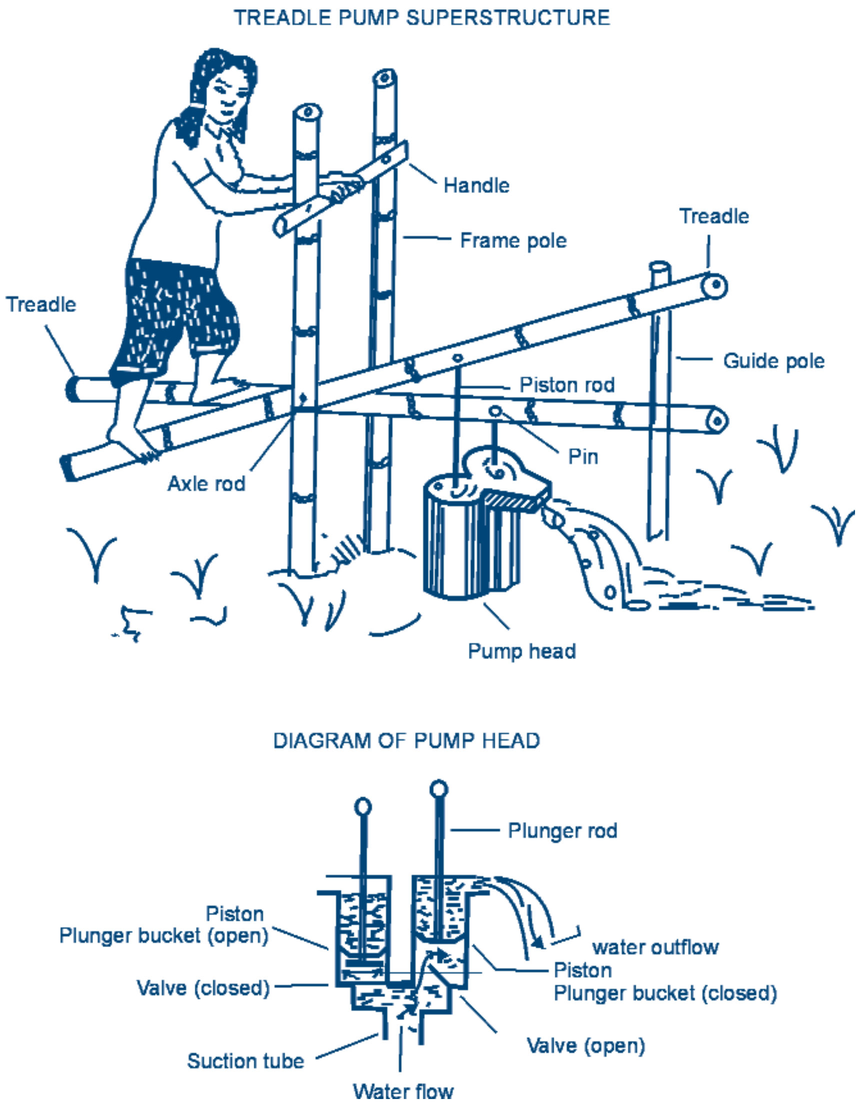


Figure 17
Double acting non-pressure treadle pump (Source: FAO, 1986)



3.5.2. Motorized pumps

Mono pumps

Mono pumps are motorized positive displacement pumps. Water is displaced by means of a screw type rotor that moves through the stator. As mono pumps fall in the positive displacement category the head is independent to the speed. However, the flow is about proportional to the speed. Figure 18 shows the individual components of a mono pump.

Figure 18

Mono pump (Source: Mono Pump, undated)

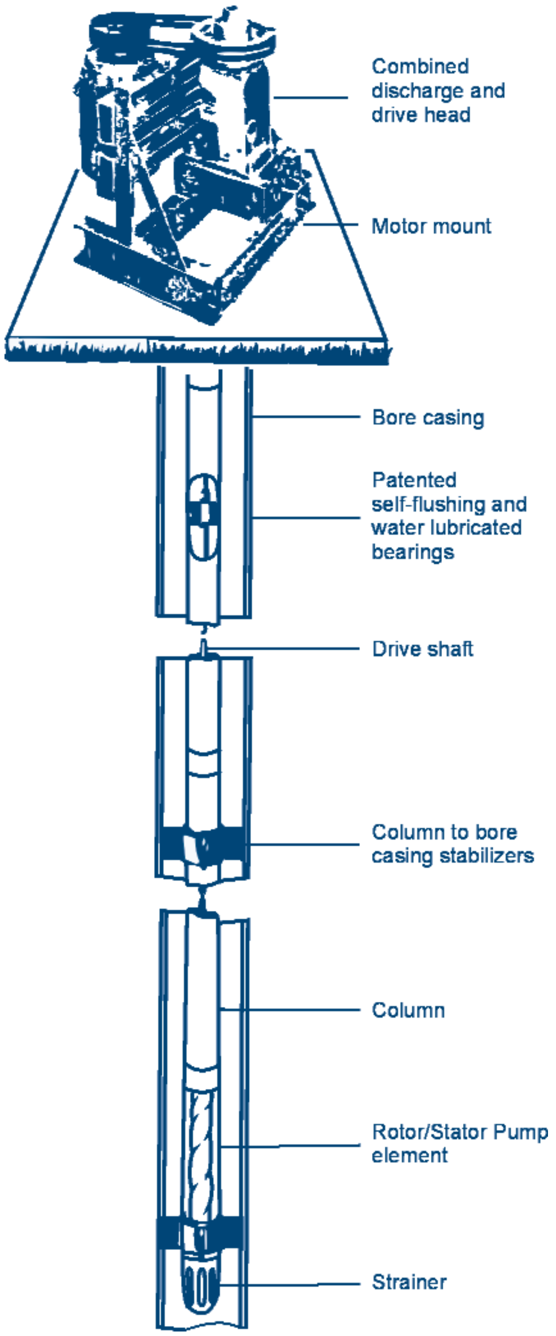
CHARACTERISTICS OF THE MONO BOREHOLE PUMP

1. SELF PRIMING. Due to the material used there is an interference fit between Rotor and Stator. This close contact with the absence of valves or ports makes a very effective air exhauster as long as a lubricating film of water is present.
2. STEADY FLOW. Due to the line of seal which is a curve of constant shape moving through the stator at a constant axial velocity the rate of displacement is uniform and steady without any pulsation, churning or agitation.
3. POSITIVE DISPLACEMENT. As the Mono unit is a positive displacement pump the head developed is independent of the speed and the capacity approximately proportional to the speed.
4. SIMPLICITY. As the mono unity consists of a fixed stator with a single rotating element it is an extremely simple mechanism.
5. EFFICIENCY. Because of the continuous steady delivery coupled with the positive displacement the Mono pump has an extremely high efficiency.
6. COMPACTNESS. Although the Mono Pump is constructed on very robust lines the simplicity of its pumping principle and the absence of valves or gears makes a very compact and light weight unit.
7. ABRASION RESISTANCE. Due to the design of the stator and rotor, the position of the seal line is continuously changing both on the rotor and on the stator. This fact is the chief reason for the remarkable ability of the Mono Pump to handle water containing some sand. If, for instance, a piece of grit is momentarily trapped between the rotor and the stator, the resilient rubber stator yields to it without damage in the same way as a rubber tyre passes over a stone, and, owing to the instant separation of the two surfaces, the particle is at once released again and swept away by the water. There is no possibility of pieces of grit being embedded or dragged along between the two surfaces, which is the chief cause of the heavy wear of most other pumps when gritty water is being handled. The low velocity of the water through the pump and its steady continuous motion also contribute to freedom from wear.
8. VERSATILITY. The pump is suitable for electric motor or engine drive.

SPECIFICATIONS OF THE MONO BOREHOLE PUMP

1. Discharge head which also incorporates the pulley housing consists of a cast iron body with gland from which the column is suspended. The pulley bearing assembly contains two pre-packed ball bearings, one being an angular contact thrust bearing.
2. Column piping is standard galvanized medium class water piping to British standard 1387/1967 with squared ends and B.S.P. thread.
3. Bobbin Bearing are styrene butadiene compound which grip the column pipe walls and support the drive shaft every 1.6m for the full length of the drive shaft. These bearings are water lubricated and the bearing piece of stainless steel.
4. Stabilizers are also a rubber compound stabilizing the column in the borehole every 13m.
5. The drive shafting is of high tensile carbon steel which allows for a minimum area usage in the pipe column but retains its strength as a positive drive.
6. The pump unit consists of a strainer, the element and the body.
7. The element is a stationary stator of a resilient neoprene based compound in the form of a compound internal helix vulcanised to the outer casing. A rotor with a hard chrome finish in the form of a single of double helix turns inside the stator. This maintains a full seal across the travelling constantly up the pump giving uniform positive displacement.

SCHEMATIC DIAGRAM OF UNIT



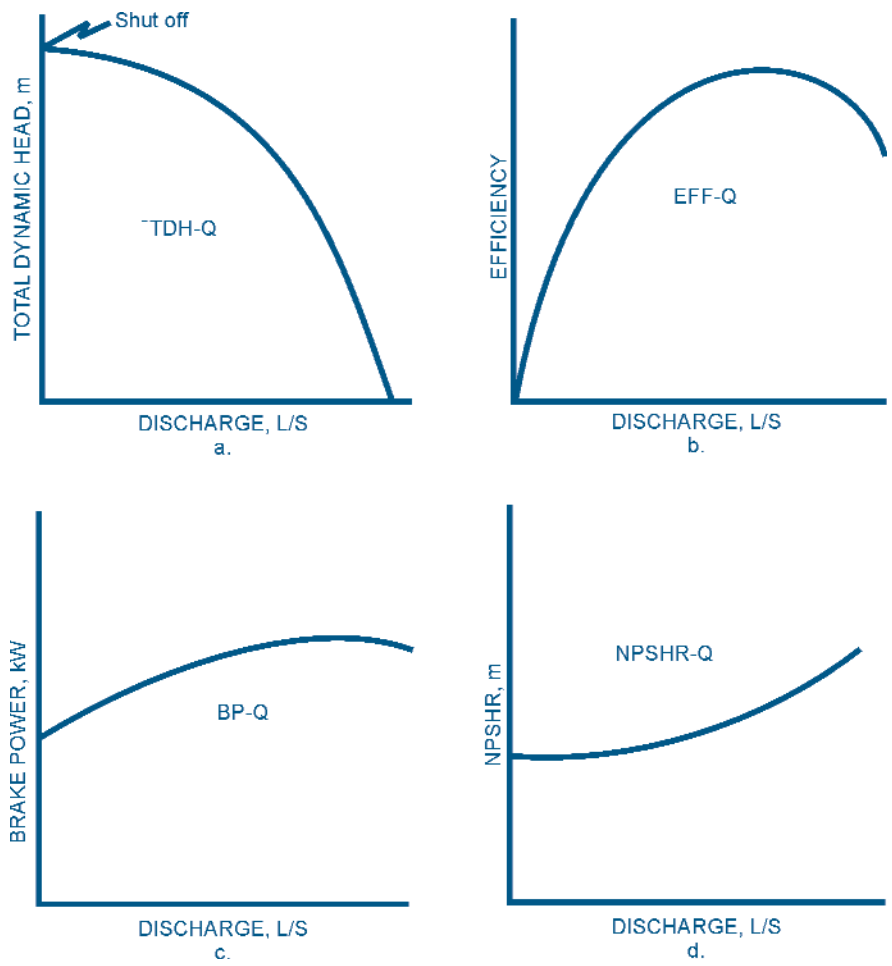
Chapter 4
Pump characteristic curves

Most manufacturers provide four different characteristic curves for every pump: the Total Dynamic Head versus Discharge or TDH-Q curve, the Efficiency versus Discharge or EFF-Q curve, the Brake Power versus Discharge or BP-Q curve and Net Positive Suction Head Required versus Discharge or NPSHR-Q curve. All four curves are discharge related. Figure 19 presents the four typical characteristic curves for a pump, with one stage or impeller.

4.1. Total dynamic head versus discharge (TDH-Q)

This is a curve that relates the head to the discharge of the pump. It shows that the same pump can provide different combinations of discharge and head. It is also noticeable that as the head increases the discharge decreases and vice versa.

Figure 19
Pump characteristic curves (Adapted from Longenbaugh and Duke, 1980)



The point at which the discharge is zero and the head at maximum is called shut off head. This happens when a pump is operating with a closed valve outlet. As this may happen in the practice, knowledge of the shut off head (or pressure) of a particular pump would allow the engineer to provide for a pipe that can sustain the pressure at shut off point if necessary.

4.2. Efficiency versus discharge (EFF-Q)

This curve relates the pump efficiency to the discharge. The materials used for the construction and the finish of the impellers, the finish of the casting and the number and the type of bearings used affect the efficiency. As a rule larger pumps have higher efficiencies.

Efficiency is defined as the output work over the input work.

Equation 2

$$E_{pump} = \frac{\text{Output work}}{\text{Input work}} = \frac{WP}{BP} = \frac{Q \times TDH}{C \times BP}$$

Where:

E_{pump} = Pump efficiency

BP = Brake power (kW or HP = 1.34 x kW): energy imparted by the prime mover to the pump

WP = Water power (kW): energy imparted by the pump to the water

Q = Discharge (l/s or m^3/hr)

TDH = Total Dynamic Head (m)

C = Coefficient to convert work to energy units – equals 102 if Q is measured in l/s and 360 if Q is measured in m^3/hr

4.3. Brake or input power versus discharge (BP-Q)

This curve relates the input power required to drive the pump to the discharge. It is interesting to note that even at zero flow an input of energy is still required by the pump to operate against the shut-off head. The vertical scale of this curve is usually small and difficult to read accurately. Therefore, it is necessary that BP is calculated using Equation 3, which can be found by rearranging Equation 2:

Equation 3

$$BP = \frac{Q \times TDH}{C \times E_{pump}}$$

4.4. Net positive suction head required versus discharge (NPSHR-Q)

At sea level, atmospheric pressure is 100 kPa or 10.33 m of water. This means that if a pipe was to be installed vertically in a water source at sea level and a perfect vacuum created, the water would rise vertically in the pipe to a distance of 10.33 m. Since atmospheric pressure decreases with elevation, water would rise less than 10.33 m at higher altitudes.

A suction pipe acts in the manner of the pipe mentioned above and the pump creates the vacuum that causes water to rise in the suction pipe. Of the atmospheric pressure at water level, some is lost in the vertical distance to the eye of the impeller, some to frictional losses in the suction pipe and some to the velocity head. The total energy that is left at the eye of the impeller is termed the Net Positive Suction Head.

The amount of pressure (absolute) or energy required to move the water into the eye of the impeller is called the Net Positive Suction Head Requirement (NPSHR). It is a pump characteristic and a function of the pump speed, the shape of the impeller and the discharge. Manufacturers establish the NPSHR-Q curves for the different models after testing. If the energy available at the intake side is not sufficient to move the water to the eye of the impeller, the water will vaporize and the pump will cavitate (see Section 4.4.1). In order to avoid cavitation the NPSHA should be higher than the NPSHR required by the pump under consideration.

4.4.1. Cavitation

At sea level water boils at about 100°C and its vapour pressure is equal to 100 kPa. When water boils, air molecules dissolved in water are released back into air. The vapour pressure increases rapidly with temperature increase (Table 2) while atmospheric pressure decreases with altitude increase.

Table 2

Variation of vapour pressure with temperature (Source: Longenbaugh and Duke, 1980)

Temperature (°C)	0	5	10	15	20	25	30	35	40	45	50
Vapour pressure of water, e (m)	0.06	0.09	0.13	0.17	0.24	0.32	0.43	0.58	0.76	0.99	1.28

In the eye of the impeller of a pump, pressure may be reduced to such a point that the water will boil. As the water is carried to areas of higher pressure in the pump, the vapour bubbles will collapse or explode at the surface of the impeller blades or other parts of the pump, resulting in the material erosion. The phenomenon described here is known as cavitation. Cavitation makes itself noticeable by an increase in noise level (rattling sound), irregular flow, a drop in pump efficiency and sometimes in head. Heavy cavitation, especially in larger pumps, sounds like the roar of thunder.

In order to determine the possibilities of the occurrence of cavitation, the water pressure at the pump's entrance is determined and compared with the vapour pressure at the temperature of the water to be pumped. For this purpose the NPSHA is calculated as follows:

Equation 4

NPSHA = atmospheric pressure at the given altitude – static suction lift – friction losses in pipe – vapour pressure of the liquids at the operating temperature

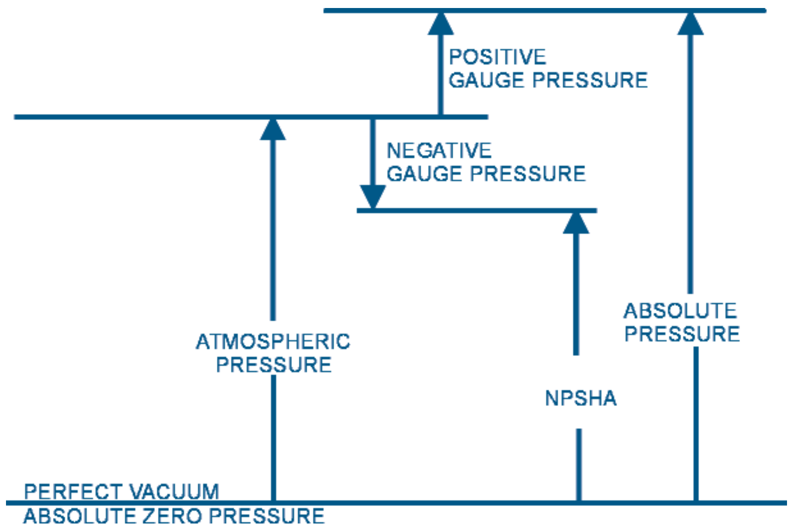
Where:

- Atmospheric pressure at the given altitude,
  Pb = 10.33 – 0.00108 Z (Barometric pressure)
  Z = elevation (m) can be measured
- Static suction lift (m) can be measured
- Friction losses h_l, in metres, can be calculated from graphs and tables or formulae
- Vapour pressure e (m) can be estimated from Table 2

Gauge pressure (Figure 20) = static suction lift + friction losses in pipe + vapour pressure

If the NPSHA is less than the NPSHR, the NPSHA will have to be increased. This can be achieved by reducing the friction losses in the pipe by using a wider suction pipe, although this is not very effective. Generally, decreasing the static suction lift increases the NPSHA, which can be obtained by positioning the pump nearer to the water level (see Figure 4).

Figure 20
Schematic presentation of Net Positive Suction Head Available (NPSHA) (Source: T-Tape, 1994)



Example 1

Calculate the NPSHA for a pump to operate at an elevation of 2 000 m, under 35°C temperature. The friction losses in the suction pipe were calculated to be 0.7 m and the suction lift to be 2 m.

Pb = 10.33- 0.00108 x 2 000 = 8.17 m

e = 0.58 m (from Table 2)

Therefore, using Equation 4:

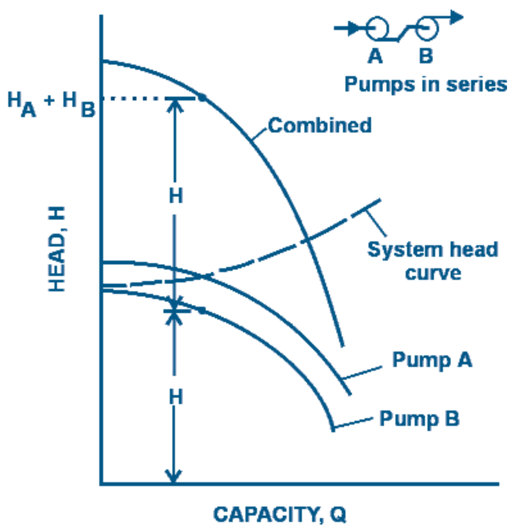
NPSHA = 8.17 - 2.0 - 0.7 - 0.58 = 4.89 m

4.5. Pumps in series

A good example of connecting pumps in series is where a centrifugal pump takes water from a dam and pumps it to another pump, which in turn boosts the pressure to the required level. Another example is the multistage turbine pump. In fact, each stage impeller represents a pump. In general, connecting pumps in series applies to the cases where the same discharge is required but more head is needed than that which one pump can produce.

For two pumps operating in series, the combined head equals the sum of the individual heads at a certain discharge. Figure 21 shows how the combined TDH-Q curve can be derived. If pumps placed in series are to operate well, the discharge of these pumps must be the same.

Figure 21
TDH-Q curve for two pumps operating in series (Adapted from Longenbaugh and Duke, 1980)



The following equation from Longenbaugh and Duke (1980) allows the calculation of the combined efficiency at a particular discharge.

Equation 5

$$E_{series} = \frac{Q \times (TDH_a + TDH_b)}{C \times (BP_a + BP_b)}$$

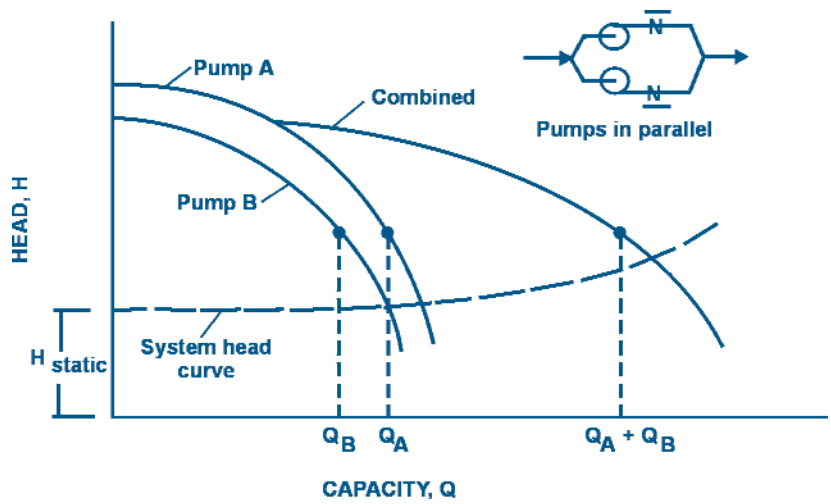
Where:

E	=	Efficiency
Q	=	Discharge (l/s)
TDH	=	Total Dynamic Head (m)
C	=	102 (coefficient to convert work to energy units)
BP	=	Brake power (kW)

4.6. Pumps in parallel

Pumps are operated in parallel when, for roughly the same head, variation in discharge is required. A typical example would be a smallholder pressurized irrigation system with many users. In order to provide a certain degree of flexibility when a number of farmers cannot be present, due to other unforeseen obligations (for example funerals), several smaller pumps are used instead of one or two larger pumps. This has been practiced in a number of irrigation schemes in Zimbabwe. Figure 22 shows the TDH-Q combined curve, for two pumps in parallel.

Figure 22
TDH-Q curve for two pumps operating in parallel (adapted from Longenbaugh and Duke, 1980)



The equation for the calculation of the combined efficiency is as follows:

Equation 6

$$E_{series} = \frac{(Q_a + Q_{b)}) \times TDH}{C \times (BP_a + BP_b)}$$

Where:

E	=	Efficiency
Q	=	Discharge (l/s)
TDH	=	Total Dynamic Head (m)
C	=	102 (coefficient to convert work to energy units)
BP	=	Brake power (kW)

It should be noted from this equation that each of the pumps used in parallel should deliver the same head and this has to be a criterion when selecting the pumps.

At times, engineers are confronted with a situation where pumping is required from a number of different sources at different elevations. In this case each pump should deliver its water to a common reservoir and not a common pipe in order to avoid the flow of water from one pump to another.

Chapter 5
Speed variation

In discussing pump characteristic curves, no mention of speed was made. Figure 23, a typical manufacturer's characteristic curve, provides several TDH-Q, EFF-Q and BP-Q curves. This is because the same pump can operate at different speeds. A change in the impeller speed causes a shift of the Q-H characteristics in the diagram. It is a shift upwards and to the right with increasing speed and downwards and to the left when the speed is decreased. The BP required power also changes.

The relationship between speed, on the one hand, and discharge, head and power on the other is described by Euler's affinity laws in the Hydraulics Handbook of Colt Industries (1975) as follows (see also Figure 24):

❖ The discharge Q varies in direct proportion to the speed:

Equation 7

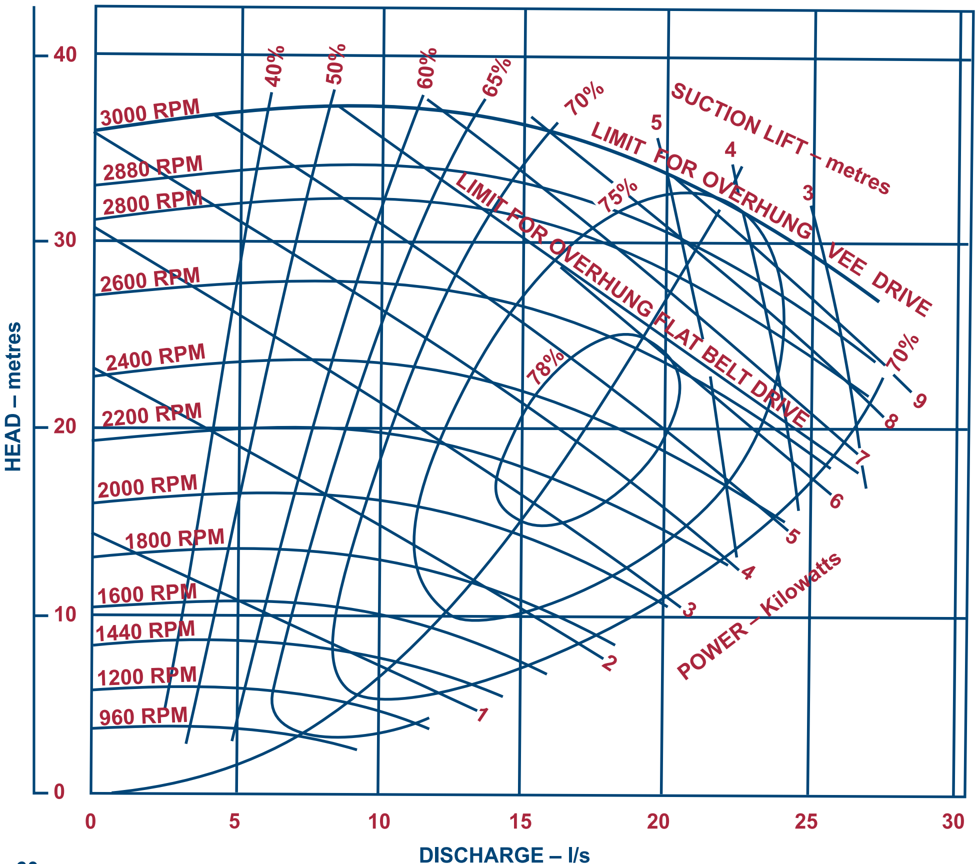
$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

❖ The head H varies directly with the square of the speed:

Equation 8

$$\frac{H_1}{H_2} = \left[\frac{N_1}{N_2}\right]^2$$

Figure 23
Pump characteristic curves (Source: Irrigation Association, 1983)



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3000 R.P.M. (MAXIMUM) FOR DIRECT DRIVE

- The break power BP varies approximately with the cube of the speed:

Equation 9

$$\frac{BP_1}{BP_2} = \left[\frac{N_1}{N_2}\right]^3$$

Where:

Q_1 = discharge and
H_1 = head and
BP_1 = brake power at N_1 speed in revolutions per minute (rpm)
Q_2 = discharge and
H_2 = head and
BP_2 = brake power at N_2 speed in revolutions per minute (rpm)

As a rule, most pump characteristic curves are presented with one speed only. Hence the need to use Euler's affinity laws in deriving performance at different speeds. Example 2 clarifies the process.

If the speed of the pump is changed from 1 200 rpm to 2 000 rpm, the discharge, head and brake power will change from 40 l/s to 66.7 l/s, 32 m to 88.9 m, and 16.8 to 77.7 kW respectively. However, the affinity laws make no reference as to how the pump efficiency is affected by speed changes. As a rule, pumps that are efficient at one speed would be efficient at other speeds.

Example 2

If a pump delivers 40 l/s at a head of 32 m and runs at a speed of 1200 rpm, what would be the discharge and head at 2000 rpm? What would the brake power of the pump be if it were 16.78 kW at 1200 rpm?

Using Equation 7 the new discharge Q_2 would be:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \Rightarrow \frac{40}{Q_2} = \frac{1\ 200}{2\ 000} \Rightarrow Q_2 = 40 \times \left[\frac{2\ 000}{1\ 200}\right] = 66.7 \text{ l/s}$$

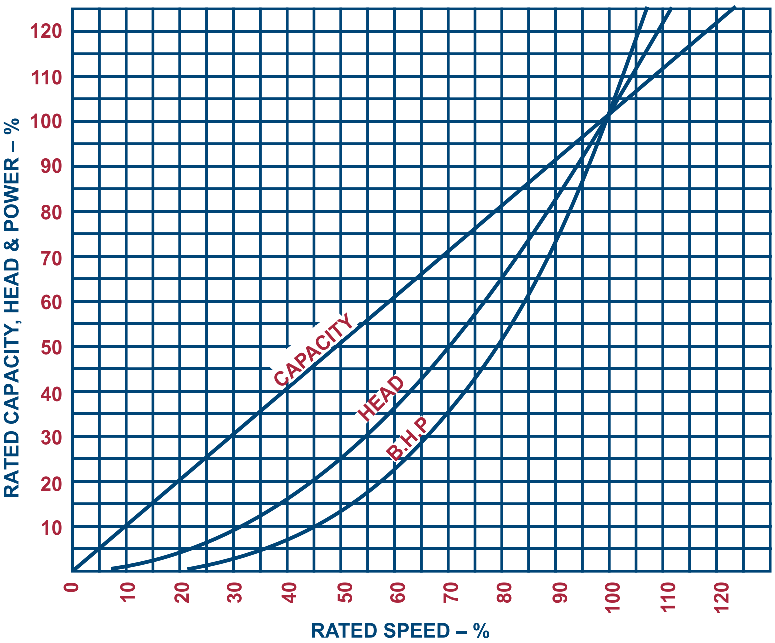
Using Equation 8, the new head would be:

$$\frac{H_1}{H_2} = \left[\frac{N_1}{N_2}\right]^2 \Rightarrow \frac{32}{H_2} = \left[\frac{1\ 200}{2\ 000}\right]^2 \Rightarrow H_2 = 32 \times \left[\frac{2\ 000}{1\ 200}\right]^2 = 88.9 \text{ m}$$

BP_2 is calculated using Equation 9 as follows:

$$\frac{BP_1}{BP_2} = \left[\frac{N_1}{N_2}\right]^3 \Rightarrow \frac{16.78}{BP_2} = \left[\frac{1\ 200}{2\ 000}\right]^3 \Rightarrow BP_2 = 16.78 \times \left[\frac{2\ 000}{1\ 200}\right]^3 = 77.7 \text{ kW}$$

Figure 24
Effect of speed change on centrifugal pump performance (Adapted from Colt Industries, 1975)



Chapter 6
Pump selection

The selection of pumps requires the use of manufacturers' pump curves. As a first step, by looking at the various pump curves we can identify a pump that can provide the discharge and head required at the highest possible efficiency. Following the identification of the pump, the NPHSR-Q curve is checked and evaluations are made to ensure that its NHPSA is higher than the NPHSR.

Example 3

Let us assume that a designed sprinkler system would require a Q = 40 m^3/hr at an H = 60 m. What would be the best pump to select?

Looking at various performance curves provided by manufacturers (Figures 25a and 25b) the curve of Figure 25b was selected, as it appears to provide the highest efficiency (65%) for the required discharge and head requirements, compared to an efficiency of 45% given by curves of Figure 25a. Ideally we would have preferred a pump where the required head and flow combination falls on the right-hand side of the efficiency curve. With age, the operating point will move to the left, then we would be able to operate with higher efficiency. This pump should be equipped with the 209 mm impeller, as shown in the curve.

Looking at the NPSH-Q curve in Figure 25b, the NPSHR of this pump is 1.2 m.

Assuming the following data for the site:

- Elevation: 2 000 m
- Static suction: 2 m
- Suction pipe friction losses: 0.5 m
- Maximum temperature: 35°C

Using Equation 4, NPSHA = (10.33 - 0.00108 x 2000) - 2.0 - 0.5 - 0.58 = 5.09 m

Since NPSHA (5.09 m) is higher than the NPSHR (1.2 m) of the selected pump no cavitation should be expected.

Example 4

Assuming a surface irrigation scheme requires a pump with a Q = 70 m^3/hr delivered at an H = 23 m. In this case the pump of Figure 25a would be more suitable. It can provide the required Q and H at an efficiency of 68%, using an impeller of 140 mm diameter.

If we opted to use a high pressure pump (Figure 25b) instead of a high volume low pressure pump (Figure 25a), the required Q of 70 m^3/hr with 23 m head would fall outside the range of the pumps. Hence the efficiency would be very low.

Figure 25a
Performance curve of a pump (Source: Stork Pumps, undated)

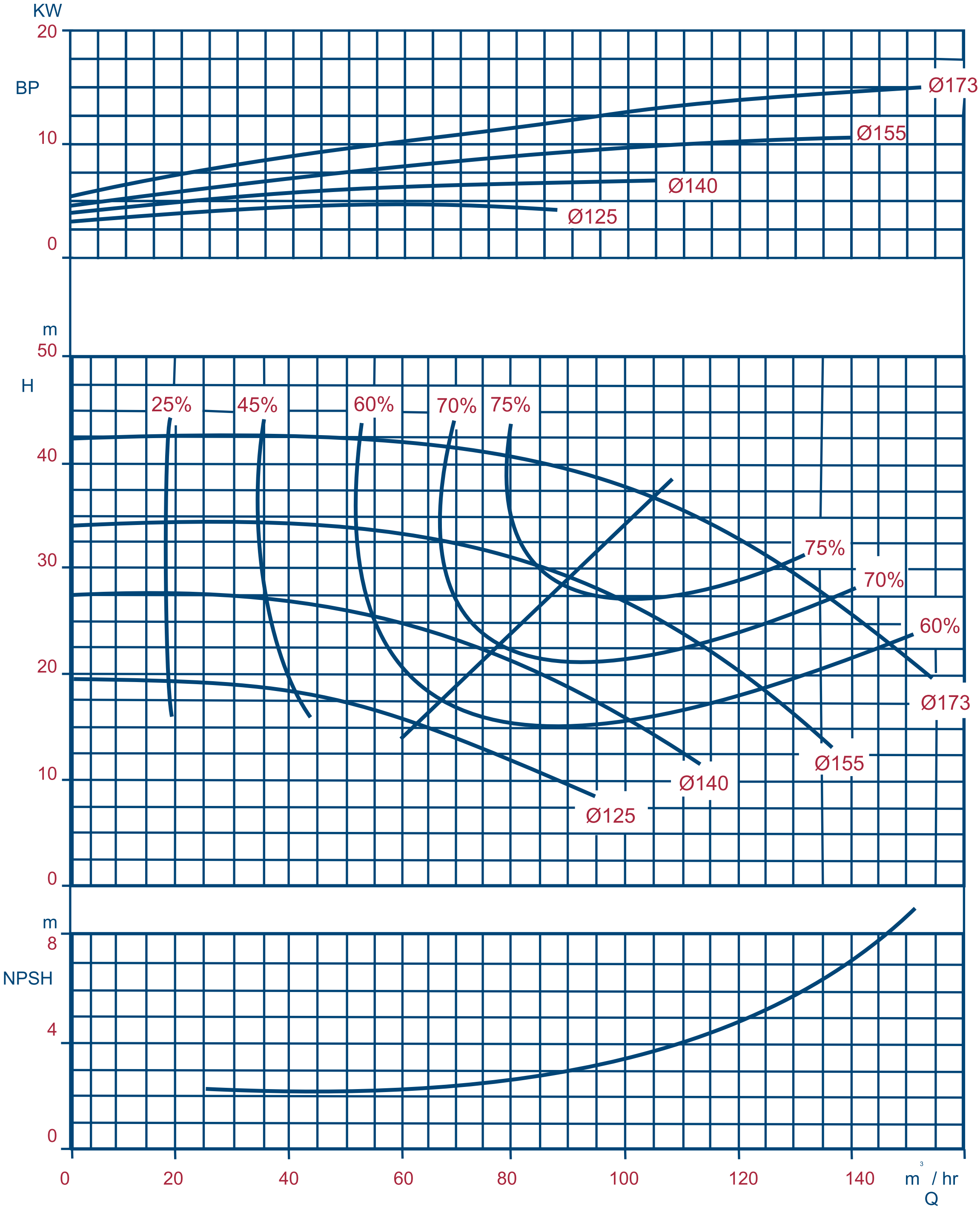
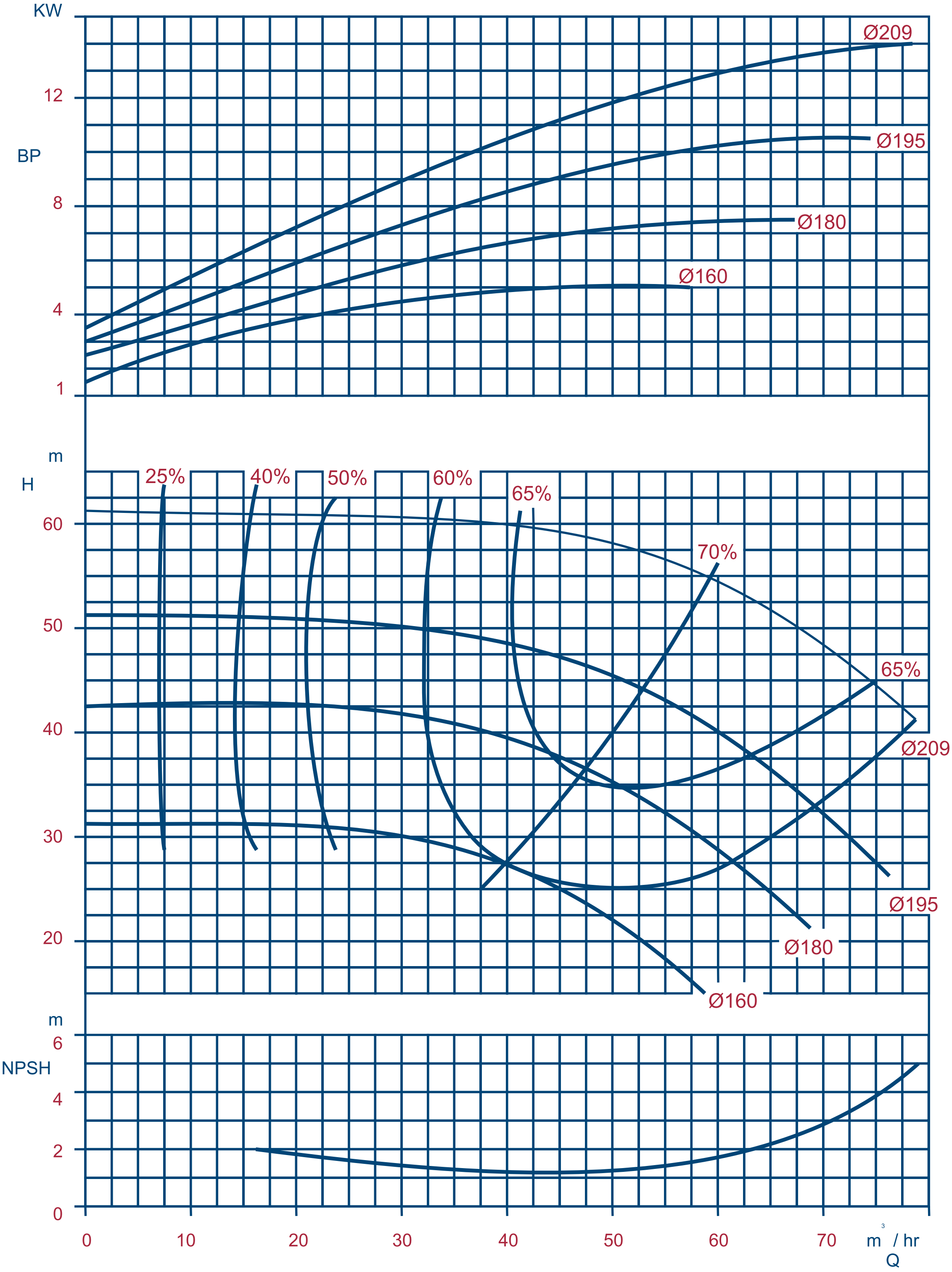


Figure 25b
Performance curve of a pump (Source: Stork Pumps, undated)



When the required Q and H combination falls outside the performance curve or when it falls at the fringes of the performance curve, that type of pump should not be selected.

Another important consideration in selecting a pump is the size of the pump impeller. If the required Q and H combination falls between two impeller sizes, then the larger impeller will have to be used, but only after it is trimmed down by the manufacturers so that it matches the requested Q and H.

Chapter 7
Power units

Most irrigation pumps are powered either with electric motors or diesel engines. In some countries, natural gas, propane, butane and gasoline engines are also used to drive pumps. Wind and solar driven pumps are also used for pumping water, mostly for human and animal purposes.

Chapter 4 described how to compute the size of the power unit. For centrifugal pumps and turbine pumps up to 20 m deep it is not necessary to compute the energy required to overcome bearing losses in the pump. For turbine pumps that are more than 20 m deep, the manufacturer's literature should be consulted on line shaft bearing losses.

7.1. Electric motors

For most centrifugal pumps the motors are directly coupled to the pump. This results in the elimination of belt drives and energy loss due to belt slippage, and safety hazards. Most centrifugal pumps used in Eastern and Southern Africa are coupled to the motor shaft through a flexible coupling.

In the past it was common practice to overload motors by 10-15% above the rated output without encountering problems. However, because of the materials currently used, motors can no longer stand this overloading. Therefore, they should be sized to the needed and projected future output.

For sustained use of a motor at more than 1 100 m altitude or at temperatures above 37°C derating may be necessary. Manufacturer's literature should be consulted for the necessary derating. An example of the derating of diesel engines is shown in the following section.

7.2. Diesel engines

As a rule, petrol engines drive very small pumps. For most irrigation conditions, the diesel engine has gained popularity. It is more robust, requires less maintenance and has lower overall operation and maintenance costs.

Most literature on engines uses English units of measurement. To convert kilowatts to horsepower a conversion factor of 1.34 can be applied. Horsepower versus speed curves (Figure 26) illustrate how output power increases with engine speed. However, there is a particular speed at which the engine efficiency is highest. This is the point at which the selected engine should operate. The continuous rated curve indicates the safest continuous duty at which the engine can be operated. Care should be taken to use the continuous rated output curve and not the intermittent output curve.

Manufacturer's curves are calculated for operating conditions at sea level and below 30°C. It is therefore necessary to derate the engines for different altitudes and temperatures where the operating conditions are different. According to Pair et al. (1983), derating is approximately 1% per 100 m increase in altitude and 1% per 5.6°C increase in air temperature from the published maximum output horsepower curve. On the top of that, an additional 5-10% for reserve should be deducted. If the continuous output curves are used, only the 5-10% deduction is applied.

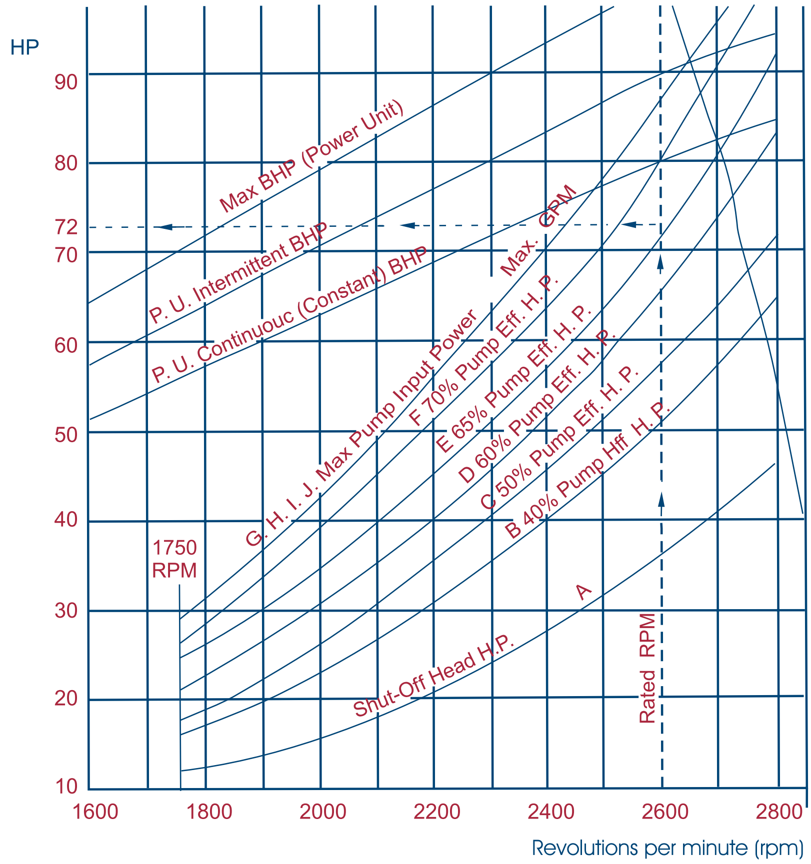
Example 5

What will be the output of a diesel engine with a speed of 2 600 rpm at 2 000 m altitude and a temperature of 35°C?

Referring to Figure 26, the maximum output at 2 600 rpm, by interpolation, would be around 114 hp, which falls outside the limits of this curve. By applying the above rule for 2 000 m altitude and 35°C, a deduction of 20% should be applied for elevation and 1% for temperature. An additional 10% should be applied for reserve. Therefore, the total deduction should be 114 x 0.31 = 35.3 HP, resulting in an output of 78.7 HP (= 114 - 35.3).

If we apply the 10% deduction on the continuous rating curve then the output will be 80 - 8 = 72 HP. This is a more conservative approach.

Figure 26
Rating curves for engine (Source: Irrigation Association, 1983)



Tractors can also be used to drive pumps. However, it may not be an economically sound approach to permanently attach a tractor to a pump in view of the high capital cost of a tractor.

7.3. Power transmission

There are four types of transmission usually applied to irrigation pumps: direct coupling, flat belt, V-belt and gear. Direct coupling generally implies negligible or no loss of power. The loss of power through flat belt varies from 3-20%. Transmission losses for V-belt and gear drive, as a rule, do not exceed 5%.

Referring to our example, if we use direct coupling of engine to pump, the HP would remain 72 HP. If we use gear or V-belt drive then the power available to the pump would be 68.4 HP (0.95 x 72). This should satisfy the input power requirements of the pump as calculated using Equation 3 and multiplying the result by 1.34 to convert to horsepower.

7.3.1. Overall derating

Most engineers multiply the result of Equation 2 by a factor of 1.2 and use the engine continuous output rating curve. In other words, they derate an engine by 20%.

Going back to the approaches described earlier in this chapter, the total derating on the continuous output curve is 10-15% for V-belt or gear (5-10% derating for continuous output and 5% for the transmission losses).

Chapter 8
Energy requirements

Energy requirements are proportional to the discharge, head and efficiency of the pumping system as demonstrated by the formula used to calculate the kW power requirements (Equation 3):

$$BP = \frac{Q \times TDH}{C \times E_{pump}}$$

Where:

Q	=	discharge in l/s with C = 102 or in m^3/hr with C = 360
TDH	=	Total Dynamic Head (m)
E_{pump}	=	pump efficiency

The annual or seasonal energy requirements increase with the increase of the total volume of water pumped annually or seasonally, and are therefore affected by the overall irrigation efficiency.

Motor efficiency also has a bearing on energy requirement calculations. According to Longenbaugh and Duke (1980), motor efficiencies are in the range of 0.88 - 0.92. Motors of 7.5 kW or less have motor efficiencies usually below 0.88. For motors of 75 kW or larger the efficiency is 0.9 - 0.92. Hence, there is the tendency to use 0.88 for motor efficiency in small size irrigation schemes.

From the three examples below, localized irrigation would have the lowest energy requirements (2 741 kW/ha per year) followed by surface (3 743 kW/ha per year) and sprinkler (4 485 kW/ha per year), in that order. This is the result of higher irrigation efficiency combined with low operating pressure, in the case of drip irrigation. In the case of surface irrigation the lack of operating pressure puts it in the second place (before sprinkler) in terms of energy requirements, irrespectively of its low irrigation efficiency. The high operating pressure of the sprinkler system (30 m) makes this system the highest energy user.

Example 6

A 14 ha drag-hose sprinkler irrigation scheme, designed to satisfy 20 hours/day pumping at peak demand, requires a discharge of 57 m^3/hr. Its TDH is 56 m (20 m static lift, 30 m sprinkler operating head and 6 m friction losses). The net irrigation requirements are 131 250 m^3/year. What are the energy requirements?

The total gross annual irrigation requirements at 75% irrigation efficiency are:

$$\frac{131\ 250}{0.75} = 175\ 000 \text{ m}^3 \text{ / year}$$

From the performance curves (Figure 25b) the best pump to satisfy this discharge and head has an efficiency of 0.69. Considering an overall derating of 20%, the power requirement is:

$$BP = \frac{57 \times 56 \times 1.2}{360 \times 0.69} = 15.4 \text{ kW}$$

Looking at the sizes commonly marketed (7.5 kW, 11 kW, 15 kW, 18 kW, 22kW, 30 kW, 40 kW, 55 kW, etc.), it appears that the 18 kW motor is the best choice for this scheme.

In order to pump the 175 000 m^3 annually the motor will be in operation for 3 070 hours (175 000/57). If the motor efficiency is 0.88, the annual energy requirements would then be:

$$\frac{3\ 070 \times 18}{0.88} = 62\ 795 \text{ kW / year for 14 ha or 4 485 kWh/ha per year}$$

Example 7

Assuming that the availability of water is not a constraint and that, instead of a sprinkler irrigation system, a surface irrigation system with 40% irrigation efficiency and a pumping lift of 25 m (assumed to be 20 m static lift plus 5 m friction losses), operating for 10 hours per day, is used. What would the energy requirements be?

Total gross annual irrigation water requirements:

$$\frac{131\ 250}{0.4} = 328\ 125 \text{ m}^3 \text{ / year}$$

Converting the discharge of 57 m^3/hr for the drag-hose sprinkler system to a discharge for the surface irrigation system gives a discharge of 213.5 m^3/hr (57 x 20/10 x 0.75/0.4). Assuming the same efficiency of 69% for the best pump to satisfy the discharge and TDH, the power requirements will be:

$$BP = \frac{213.5 \times 25 \times 1.2}{360 \times 0.69} = 25.8 \text{ kW}$$

From the standard sizes of motors available on the market, a 30 kW motor will be selected.

In order to pump the 328 125 m^3 of water annually the motor will have to operate for 1 537 hrs (328 125/213.5). The energy requirements would then be:

$$\frac{1\ 537 \times 30}{0.88} = 52\ 398 \text{ kW / year for the 14 ha or 3 743 kWh/ha per year}$$

Example 8

If, instead of a sprinkler or surface irrigation system, a localized irrigation system with 90% irrigation efficiency and a pumping lift of 40 m (static lift of 20 m, friction losses and operating head assumed to be 20 m), operating for 20 hours per day, is used. What would the energy requirements be?

Total gross annual irrigation water requirements:

$$\frac{131\ 250}{0.9} = 145\ 833 \text{ m}^3 \text{ / year}$$

Converting the discharge of 57 m^3/hr for the drag-hose sprinkler system to a discharge for the localized irrigation system gives a discharge of 47.5 m^3/hr (57 x 20/20 x 0.75/0.9). Again assuming a pump efficiency of 69%, the power requirements would be:

$$BP = \frac{47.5 \times 40 \times 1.2}{360 \times 0.69} = 9.2 \text{ kW}$$

Although a 9.2 kW motor is required, from the standard size motors available on the market an 11 kW motor will be selected.

In order to pump the 145 833 m^3 of water annually the motor will have to operate for 3 070 hrs (145 833/47.5). The total annual irrigation energy requirements would then be:

$$\frac{3\ 070 \times 11}{0.88} = 38\ 375 \text{ kW / year for the 14 ha or 2 741 kWh/ha per year}$$

The picture changes when the static lift increases to 35 m, as demonstrated in Example 9.

Example 9

Assuming all figures of the previous three examples remain the same, except for the static lift, which increases from 20 m to 35 m. What would the energy requirements be?

From Examples 6, 7 and 8

Type of irrigation system	Total gross annual irrigation required (m^3)	Discharge (m^3/hr)	Hours of operation per year
Sprinkler	175 000	57.0	3 070
Surface	328 125	213.5	1 539
Localized	145 833	47.5	3 070

Sprinkler power requirements $= \frac{57 \times 71 \times 1.2}{360 \times 0.69} = 19.6$ kW $\Rightarrow$ motor size: 22 kW

Surface power requirements $= \frac{213.5 \times 40 \times 1.2}{360 \times 0.69} = 41.3$ kW $\Rightarrow$ motor size: 45 kW

Localized power requirements $= \frac{47.5 \times 55 \times 1.2}{360 \times 0.69} = 12.6$ kW $\Rightarrow$ motor size: 15 kW

Sprinkler kWh $= \frac{3\,070 \times 22}{0.88} =$ 76 750 kW / year for 14 ha or 5 482 kW/ha per year

Surface kWh $= \frac{1\,539 \times 45}{0.88} =$ 78 699 kW / year for 14 ha or 5 621 kW/ha per year

Localized kWh $= \frac{3\,070 \times 15}{0.88} =$ 46 050 kW / year for 14 ha or 3 289 kW/ha per year

In Example 9 surface irrigation is the most energy inefficient, because of the combined low irrigation efficiency and high static head. Localized irrigation again has lowest demand or highest efficiency.

Table 3 presents a comparison of energy requirements for sprinkler, surface and localized irrigation systems for different static lifts and operating pressures.

Following the same procedures described in Examples 6, 7, 8 and 9, the comparison of energy requirements of Table 3 was prepared. This comparison is based on net annual water requirements of 9 375 m^3/ha per year and an efficiency of 75% for sprinklers, 40% for surface and 90% for localized irrigation. The total area is assumed to be 14 ha. The flow rate used for sprinklers is 57 m^3/hr, for surface irrigation 213.5 m^3/hr and for localized 47.5 m^3/hr. For all systems a pump efficiency of 69% and a motor efficiency of 88% were assumed. It should be noted that no adjustment of the kW requirements was made to match the availability of motors in the market, because the sizes of motors available vary from country to country.

For surface irrigation, the head losses for conveying the water to the night storage reservoir were assumed to be 5 m. In the case of sprinkler irrigation, the sprinkler operating pressure was assumed to be 30 m and the head losses 6 m. For localized irrigation, the operating pressure plus the head losses were assumed to be 20 m.

Table 3

Comparison of the energy requirements for the three irrigation systems under different levels of static lift

	Power Requirements (kW)			Annual Energy Requirements for 14 ha (kWh)			Annual Energy Requirements per hectare (kWh/ha)		
Static lift (m)	Surface irrigation	Sprinkler irrigation	Localized irrigation	Surface irrigation	Sprinkler irrigation	Localized irrigation	Surface irrigation	Sprinkler irrigation	Localized irrigation
5	10.3	11.3	5.7	17 989	39 426	19 887	1 285	2 816	1 421
10	15.5	12.7	6.9	27 071	44 310	24 074	1 934	3 165	1 720
20	20.6	15.4	9.2	35 988	53 731	32 099	2 571	3 838	2 293
30	36.1	18.2	11.5	63 049	63 500	40 124	4 503	4 536	2 866
40	46.4	20.9	13.8	81 038	72 920	48 148	5 788	5 209	3 439
50	56.7	23.7	16.1	99 027	82 689	56 173	7 073	5 906	4 012
55	61.9	25.1	17.2	108 108	87 574	60 011	7 722	6 255	4 286
60	67.0	26.4	18.4	117 016	92 110	64 198	8 358	6 579	4 586
65	72.2	27.8	19.5	125 748	96 994	68 036	8 982	6 928	4 860
70	77.4	29.2	20.7	135 179	101 879	72 222	9 656	7 277	5 159
75	82.5	30.6	21.8	144 086	106 736	76 060	10 292	7 626	5 433
80	87.7	31.9	22.8	153 168	111 299	79 549	10 941	7 950	5 682
85	92.8	33.3	24.1	162 075	116 184	84 085	11 577	8 299	6 006
90	98.0	34.7	25.2	171 157	121 068	87 923	12 226	8 648	6 280
95	103.1	36.1	26.4	180 064	125 953	92 110	12 862	8 997	6 579
100	108.3	37.4	27.5	188 622	130 489	95 946	13 473	9 321	6 853

The energy requirements comparison presented in Table 3 demonstrates the following:

- ❖ The break-even point between sprinkler and surface irrigation in terms of energy requirements occurs when both systems operate with a static lift of about 30 m.
- ❖ As the static lift increases, the difference in the energy requirements between surface and sprinkler irrigation increases substantially. The latter system requires less energy. This is attributed to the higher efficiency of sprinkler irrigation, which after the 30 m static lift point compensates for the higher pressure required by this system for its operation.
- ❖ The break-even point between surface and localized irrigation in terms of energy requirements falls somewhere between 5 and 10 m static lift (about 8 m). In this respect, it should be noted that low-pressure drip systems operating with 1-3 heads were not considered in this comparison.
- ❖ As the static lift increases, the difference in energy requirements between surface and localized systems increases, with the latter requiring less energy. This is attributed to the much higher efficiency of localized systems, which after the 8 m static lift point compensate for the higher pressure requirements of the localized systems.
- ❖ Localized systems are less energy demanding than sprinkler systems irrespective of static lift. This is attributed to the higher efficiency and lower operating pressure of the localized systems.

When electricity is not available and diesel engines are used for pumping, fuel requirements should be based on the manufacturer's catalogues as they vary according to the speed at which an engine operates. For example, a TS2 LISTER engine would consume 241 g/kWh at 1500 rpm or 266 g/kWh at 3000 rpm. As a rule a good estimate can be obtained by basing the diesel consumption on 0.25 litres/kWh.

Chapter 9
Siting and installation of pumps

9.1. Siting of pumping station

The careful selection of a suitable location for a pumping station is very important in irrigation development. Several factors have to be taken into consideration when choosing the site.

Firstly, one has to find out whether the flow is reliable in the case of a river or whether the amount of water stored in the dam is enough to fulfil the annual irrigation requirements for the proposed cropping programme. This information is often obtained from the water authority or from the local farmers' experiences.

Secondly, in the case of river abstraction one has to check the maximum flood level of the river and preferably site the pumping station outside the flood level. With the limitations often imposed by the length of the suction pipe necessary to cater for the net positive suction head, where there are fluctuating flood levels, a portable pumping station is preferable. Such a site, however, should be on stable soil and have enough of water depth for the suction pipe. For permanent pumping stations pumps are installed on concrete plinth or foundation, the size of which varies in relation to the size of the pumping unit. Figure 27 shows a typical plinth and its reinforcement for pumps up to 50 kW.

Thirdly, the abstraction point should not be sited in a river bend where sand and silt deposition may be predominant. Otherwise, the sand would clog both the suction pipe and pump. Where the river is heavily silted, a sand abstraction system can be developed.

Fourthly, where water is to be pumped from a dam or weir, the site should be outside the full supply level in case of upstream abstraction. In the case of downstream abstraction, the site should neither be too close to nor in line with the spillway.

Finally, as a rule, before a final decision is made on the location of the pumping station, a site visit has to take place to verify the acceptability of the site, taking into consideration the above requirements. It is generally helpful to talk to the local people to get information on the site.

The cost of a pumping station will have to be divided into investment costs, costs of operation and costs of maintenance and repair. These costs will have to be carefully estimated during the various stages of the design process in order to make comparisons for the different options more meaningful.

9.2. Installation of pump

When the correct type of pump has been selected it must be installed properly to give satisfactory service and be reasonably trouble-free. Pumps are usually installed with the shaft horizontal, occasionally with the shaft vertical (as in wells).

9.2.1. Coupling

Pumps are usually shipped already mounted, and it is usually unnecessary to remove either the pump or the driving unit from the base plate. The unit should be placed above the foundation and supported by short strips of steel plate and wedges. A spirit level should be used to ensure a perfect levelling. Levelling is a prerequisite for accurate alignment.

To check the alignment of the pump and drive shafts, place a straightedge across the top and side of the coupling, checking the faces of the coupling halves for parallelism. The clearance between the faces of the couplings should be such that they cannot touch, rub or exert a force on either the pump or the driver.

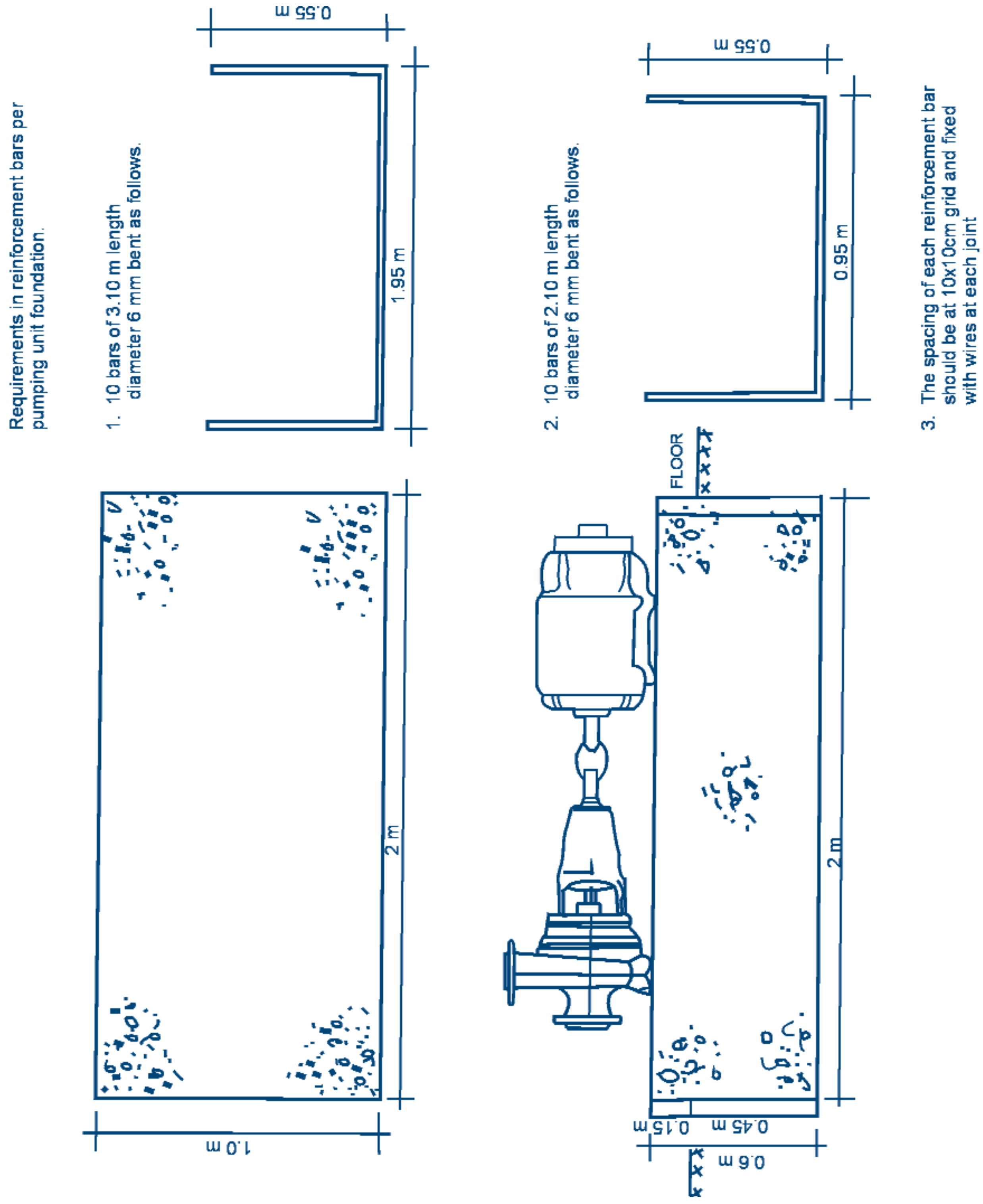
9.2.2. Grouting

The grouting process involves pouring a mixture of cement, sand and water into the voids of stone, brick, or concrete work, either to provide a solid bearing or to fasten anchor bolts. A wooden form is built around the outside of the bedplate to contain the grout and provide sufficient head for ensuring flow of mixture beneath the only bedplate. The grout should be allowed to set for 48 hours; then the hold-down bolts should be tightened and the coupling halves rechecked.

9.2.3. Suction pipe

The suction pipe should be flushed out with clear water before connection, to ensure that it is free of materials that might later clog the pump. The diameter of the suction pipe should not be smaller than the inlet opening of the

Figure 27
Foundation of a pumping unit and the reinforcement requirements



pump and it should be as short and direct as possible. If a long suction pipe cannot be avoided, then the diameter should be increased. Air pockets and high spots in a suction pipe cause trouble. After installation is completed, the suction pipe should be blanked off and tested hydrostatically for air leaks before the pump is operated.

A strainer should be placed at the end of the inlet pipe to prevent clogging. Ideally the strainer should be at least four times as wide as the suction pipe. A foot valve may be installed for convenience in priming. The size of the foot valve should be such that frictional loses are very minimal.

9.2.4. Discharge pipe

Like the suction pipe, the discharge pipe should be as short and free of elbows as possible, in order to reduce friction. A gate valve followed by a check valve should be placed at the pump outlet. The non-return valve prevents backflow from damaging the pump when the pumping action is stopped. The gate valve is used to gradually open the water supply from the pump after starting and to avoid overloading the motor. The same valve is also used to shut off the water supply before switching off the motor.

Chapter 10
Water hammer phenomenon

Water hammer is the name given to the pressure surges caused by some relatively sudden changes in flow velocity. This can be caused by valve opening or closing, pump starting or stopping, cavitation or the collapse of air pockets in pipelines, filling empty pipelines or, of most concern to irrigation applications, a power outage, which suddenly shuts down all the electric pumps on the pipeline.

When the velocity in the pipeline is suddenly reduced, the kinetic energy (velocity head) of the moving column of water is converted into potential energy (pressure head), compressing the water and stretching the pipewalls. These disturbances then travel up and down the pipeline as water hammer waves. The reader has probably experienced the banging and rattling of household pipework resulting from opening and closing a tap too rapidly – this is water hammer.

The pressure surges may either be positive or negative, i.e. the pressure may either rise above or fall below the operating pressure (static pressure, Po) by an amount equal to the maximum surge pressure, or surge head. The total pressure (Pt) rise due to water hammer is given by Joukowsky's Law (T-Tape, 1994), stated as:

Equation 10

$$Pt = Po + \frac{0.07 \times V \times L}{T}$$

Where:

Pt = The total pressure developed in the system due to water hammer (psi)
Po = The static pressure (psi)
L = Length of pipe on the pressure side of the valve (feet) (3.28 feet = 1 m)
V = Velocity of water at the time the reduction occurred (fps) (3.28 fps = 1 m/s)
T = Valve closing time (s)

Another expression for the same purpose is provided through Equation 11. This equation takes the elasticity of the pipe material into consideration. It does not, however, take into account the valve time closure.

Equation 11

$$P = 1\,423 \times V \times \sqrt{\frac{E}{E + 294\,000\,\frac{d}{t}}}$$

Where:

P = The excess pressure above normal (kPa)
V = The velocity of flow (m/s)
E = Modulus of elasticity of the pipe material (for steel, cast iron, concrete and uPVC, E = 21 x 10^7, 0.5 x 10^7, 2.1 x 10^7 and 0.28 x 10^7 respectively)
d = Pipe inside diameter (mm)
t = Thickness of pipe wall (mm)
1423 = A constant for metric units

In Equation 11, V = V1-V2 where V1 is the upstream velocity and V2 is the downstream velocity of water in the pipe. As can be seen, the most severe case occurs when V2 becomes zero due to a sudden valve closure or similar action.

This equation calculates the surge pressure that would theoretically occur were the velocity instantaneously changed from V1 to V2. If a valve is closed slowly, the actual surge pressure will be less than this value. Thus, using this equation with V2 equal to zero (or V = V1) provides a safety factor.

Characteristics of the pipe, such as temperature, pipe material and the ratio of the diameter of the pipe to its wall thickness, affect the elastic properties of the pipe and will ultimately have an impact on the speed at which the shock waves travel up and down the pipe.

Example 10

An irrigation system has a uPVC mainline with a pressure rating of 125 psi. The velocity of flow is 5.29 fps. The system operating pressure (static) is 50 psi.

1. The longest length of uninterrupted piping between the source and valve is 100 feet and valve closure time is 10 seconds.

$$Pt = 50 + \frac{0.07 \times 5.29 \times 100}{10} = 50 + 3.7 = 53.7 \text{ psi}$$

2. The longest length of uninterrupted piping between the source and valve is 100 feet and valve closure time is 1 second.

$$Pt = 50 + \frac{0.07 \times 5.29 \times 100}{1} = 50 + 37.0 = 87.0 \text{ psi}$$

3. The longest length of uninterrupted piping between the source and valve is 1000 feet and valve closure time is 10 seconds.

$$Pt = 50 + \frac{0.07 \times 5.29 \times 1000}{10} = 50 + 37.0 = 87.0 \text{ psi}$$

4. *The longest length of uninterrupted piping between the source and valve is 1000 feet and valve closure time is 1 second.*

$$Pt = 50 + \frac{0.07 \times 5.29 \times 1000}{1}$$ = 50 + 370.3 = 420.3 psi. This is well above the pipe pressure rating of 125 psi

⇒ **Severe water hammer damage**

Example 11

The irrigation system in Example 10 has a uPVC mainline with a pressure rating of 125 psi. The velocity of flow is 5.29 fps. The system operating pressure (static) is 50 psi.

$$P = 1\,423 \times V \times \sqrt{\frac{E}{E + 294\,000 \frac{d}{t}}}$$

V = 5.29 fps = 1.61 m/s

E = 0.28×10^7 for uPVC

d = 151.4 mm

t = 4.59 mm

$$P = 1\,423 \times 1.61 \times \sqrt{\frac{0.28 \times 10^7}{0.28 \times 10^7 + 294\,000 \times \frac{151.4}{4.59}}}$$

P = 2291.03 x 0.47333

P = 1084.41 Kpa = 154.9 psi

Pt = Po + P

Pt = 50 psi + 154.9 psi

Pt = 204.9 psi

The resultant water pressure is well above the pipe pressure rating of 125 psi ⇒ **Severe water hammer damage**

10.1. Effect of temperature

As water temperature increases the pipe material will become more ductile (elastic). The normal pressure rating is quoted for specific temperature conditions normally prescribed by the manufacturer. This pressure rating will need to be derated with a service factor for higher temperature conditions to provide for safe operation.

Table 4 shows the service factors for PVC and PE pipes for temperatures higher than 23°C.

Table 4

Temperature service rating factors for PVC and PE pipes (Source: Seipt, 1974)

Temperature °C	Service Factors	
	PVC Pipe	PE Pipe
23.0	1.00	1.00
26.7	0.88	0.92
32.2	0.75	0.81
37.8	0.62	0.70
43.3	0.50	-
48.9	0.40	-
54.4	0.30	-
60.0	0.22	-

The pressure rating of the pipe given by the manufacturer should be multiplied by the appropriate service factor from Table 4 to obtain the temperature compensated pressure rating of the pipe. For example, PVC Class 6 has a rating of 6 bars at 23°C, but when the water temperature is increased to 26.7°C the pressure rating decreases to 5.28 bars (0.88 x 6 bars)

10.2. Effect of pipe material and the relationship between pipe diameter and wall thickness

Figure 28 shows the relationship between the ratio of the diameter of the pipe to its wall thickness.

From Figure 28 it can be seen that the wave velocities are generally much less for uPVC than AC, cast iron and steel. This means that the pressure surges will be less in PVC for a given sudden change in velocity than for the other types.

However, valve closure times that will cause the maximum surge will be much longer than for the other types producing higher surge wave velocities. Minimum valve closure times are given by the formula:

Equation 12

$$\mu = \frac{2L}{a}$$

Where

μ = The return period of the surge waves or the time taken for the surge wave to travel the length of the pipe and return to the source /valve (s)

L = Pipe length (m)

a = Surge wave velocity (m/s)

Maximum surges will occur for any change in flow velocity that takes place within this minimum time period. This includes valve opening, pump start-up and pump stopping, as well as valve closure. The surges are reduced progressively for periods longer than this "instantaneous" period.

As a rule, when closing valves, the last tenth of travel to complete valve closure should not be less than 10 μ or 10 times the return period of the surge wave. This can amount to more than 60 s/km of pipeline where uPVC is used.

Hardies (1985) recommends that for PVC pipelines, the maximum pressure variation in total surge pressure, that is maximum to minimum, should not exceed 50% of the normal pressure rating of the pipe. For example, the maximum surge head for PVC Class 6 should be ± 15 m. Table 5 gives the recommended maximum surge heads and corresponding changes in flow velocity for each class of pipe.

Table 5

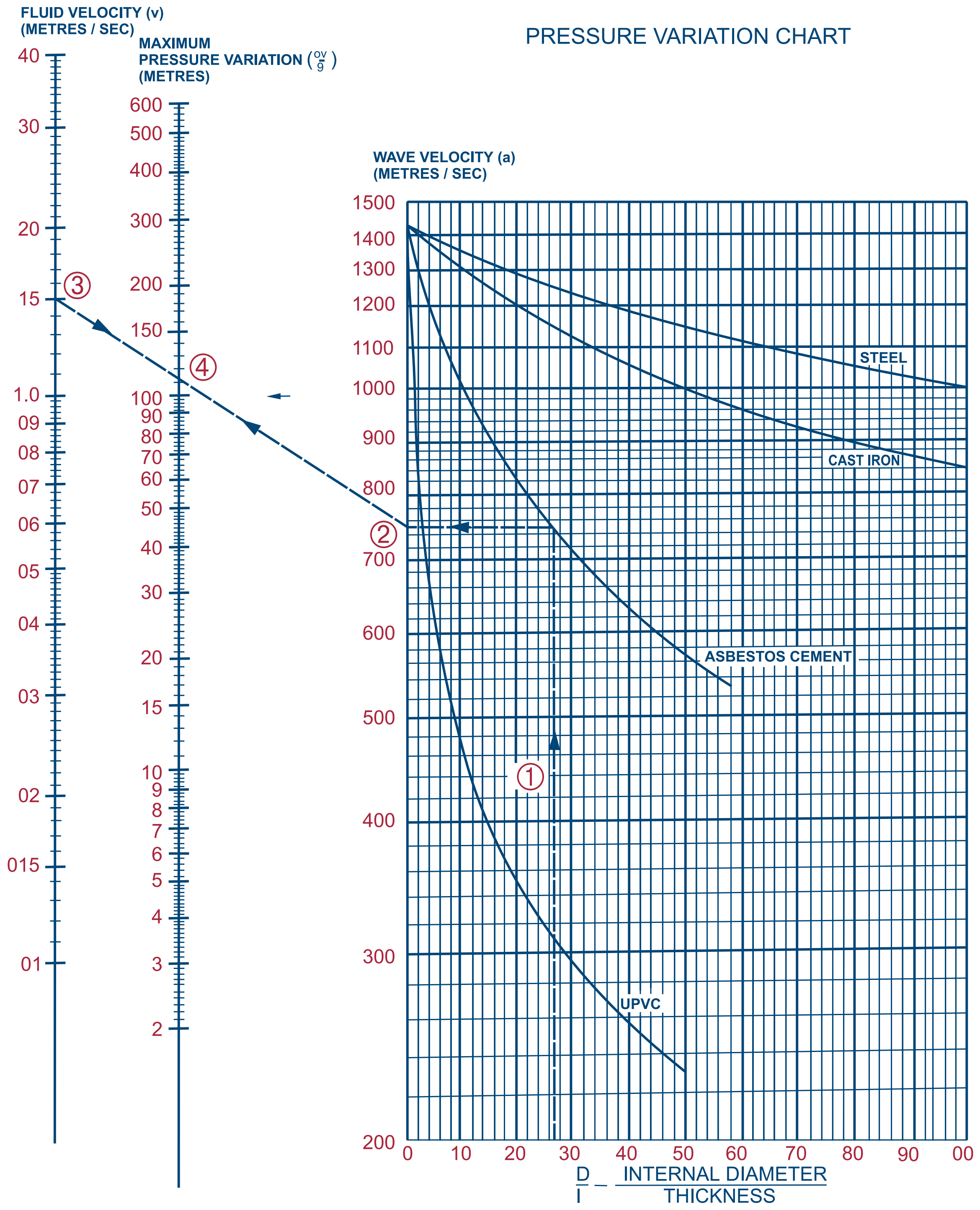
Recommended maximum surge heads for PVC pipes of different classes (Adapted from T-Tape, 1994)

Class	d/t	H (m)	ΔH (m)	ΔV (m/s)
6	39	60	±15	±0.55
9	23	90	±22.5	±0.65
12	18	120	±30	±0.75

The d/t ratios vary only with pipe class. Thus the ratios used in this table apply for different pipes sizes, i.e. 160 mm, 200 mm and 250 mm.

Figure 28

Pressure surge caused by water hammer after sudden valve closure (Source: T-Tape irrigation Training Seminar, 1994)



To use the chart:

1. Determine the D/t ratio of the pipe and draw a vertical line from that point to intercept the pipe material curve – Line 1
2. Read across to establish the wave velocity (a) – Point 2
3. Determine the fluid velocity (v) from the relevant pipe flow resistance chart and plot – Point 3
4. Connect points 3 and 2 to establish pressure variation – Point 4

In Table 5:

d = Internal diameter of pipe (mm)

t = Thickness of pipe wall (mm)

H = Nominal rated head of pipe (m)

ΔH = Maximum recommended variation in head due to cyclic surges (m)

ΔV = Maximum recommended instantaneous variation in flow velocity (m/s)

Polyethylene pipe is not affected by cyclic pressures to the same extent as is PVC, being more resilient material. However, surges still occur, as in any other pipeline, thus the need to control them to minimize damage to the pipeline.

10.3. Design and management considerations in dealing with water hammer

In the design process, pipe should be selected with a pressure rating equal to or greater than the combination of operating plus surge pressures.

To help minimize surge pressures the maximum velocity of water in the pipeline should be limited. General recommendations given by the Irrigation Association (1983) are to limit maximum operating velocities to 1.5 m/s and in no case should velocity exceed 3 m/s.

Column separation is the division of the column of water in a pipeline into two or more segments, due to the effects of a negative surge wave and flow velocity in the pipe. The separated water columns produce significant surge pressures due to the high velocities encountered when the column rejoins. A large positive surge results when the columns rejoin, which very often results in the fracture of the pipeline at that point.

Entrapped air in a pipeline can cause problems. Air is very compressible and can compress and expand in the pipeline, resulting in varying velocity conditions and significant pressure variations.

Preventing air from accumulating in the system can minimize problems associated with air entrapment and positive surges. This can be accomplished by positioning air relief valves at the high points on the pipeline.

For negative surges a suction by-pass is placed between the pump suction, upstream of the eccentric reducer, and the pump discharge, just downstream of the butterfly valve or non-return valve. A non-return valve is installed on this by-pass system, which is then used to control the flow of water from the suction side of the pump. This valve opens when the pump is shut down and when a negative pressure develops in the mainline. When the valve opens water bypasses the pump and flows into the pipeline, relieving the negative pressure build-up, thus reducing the subsequent positive surge.

The diameter of the suction by-pass should be the same or bigger than the diameter of the discharge non-return valve.

Additional design considerations include:

- Surge arrestors (devices such as pressure tanks which can absorb shock waves) or automatic pressure reducing valves at flow regulators and at pump discharge.
- Flow controllers used to minimize the rate of filling and to reduce start-up surge in filled lines.
- In cycling systems, design pipelines, if possible, to keep out all air, and then to restart with filled system.

Water hammer can not be completely eliminated in an economical design, but, by taking precautions during management and operation, the effects can be minimized. Start-up is critical, especially when pipelines are empty. Empty lines should be filled as slowly as possible to allow entrapped air to escape. In addition, the following cautions should be observed:

- Never completely open the gate valve on the discharge side of a deep-well turbine. This prevents excessive shut-off heads from developing.
- Open all manual valves leading to the zones to be irrigated except the one at the pump discharge. The pump discharge valve should be opened slowly to allow the slow filling of the pipeline. Caution should be observed when filling is interrupted and restarted because a quick surge may develop during the restart, which could slam into a stationary or slow-moving body of water. This situation could result in damaging water hammer pressures, especially if air becomes entrapped between the waterfronts. Therefore, follow the same precautions on restart as during the initial starting of the system.
- Make sure all air has been discharged from the system before operating the system at full throttle.
- Close all manual valves slowly. No valve should ever be closed in less than 10 seconds; 30 seconds or more is preferable.
- Use the same precautions in stopping the irrigation system as used in start-up and general operation.

Water hammer is a complex process, the mechanics of which are not fully understood (T-Tape, 1994). However, it can be readily analyzed and many effective methods of control are available. A basic understanding of the effects of water hammer and some simple means of control are essential to system designers if their schemes are to operate effectively and be trouble-free.

Chapter 11
Operation and maintenance of pumping units

There are several types of pumps available on the market. All pump manufacturers provide users' operation and maintenance manuals specific to their pumps. These have to be closely adhered to in order to ensure the most efficient operation of the pump and avoid unnecessary pump breakdowns. In view of the wide variety of operational instructions, which can be expected for different pumps, only general guidelines can be provided here.

Manual pumps are operated by people or animals, whereas motorized pumps are operated by prime movers, engines and electric motors. In general, the principles of operation of pumps are the same. The discharge and pumping head relationship of all pumps is dependent on the type of pump and the amount of energy that the manual operator or prime mover can transfer to the pump, among other factors. Since the principles of pump operation are the same, this section will deal with the general aspects of pump operation, but with specific reference to motorized pumps.

11.1. Pump start-up and shut-down

There are certain procedures that are recommended by pump manufactures before any pump start-up. Some of the pre-start-up inspections recommended immediately after pump installation are checking for correct pump-motor wiring connections, valve connections, shaft and gland clearance. It has to be remembered that starting a pump dry will cause seizing or destructive wear between the pump components. Therefore, pumps that are not self-priming or those with a positive suction lift should be primed before they are started. Different manufacturers also have specific instructions for pump shut down after operation. These have to be adhered to strictly.

11.1.1. Priming

While deep well pumps, such as submersible pumps, are submerged into the water and have no need for priming, the well-known horizontal centrifugal pump usually needs priming. Priming is the process of removing sufficient air from the pump and the suction pipe so that the atmospheric pressure can cause the flow of water inside the pump.

The simplest way of doing this is to displace the air in the system by filling the pump and suction pipe with water. For this purpose, a tank is connected to the pump and a foot valve to the suction pipe. The tank is filled with water when the system is operating. Before the system is switched on, the water from the tank is diverted to the pump and suction pipe via a valve.

However, the most popular priming method is the use of a manually operated vacuum pump. Other means are also available for priming, such as mechanically operated vacuum pumps and exhaust primers. At times, horizontal centrifugal pumps are installed at a dam outlet. In this case no priming is required since the water level inside the dam is higher than the level of the impeller, which forces the water to remove all the air from the suction pipe and the volute of the pump.

The pump must not be run unless it is completely filled with liquid, otherwise there is danger of damaging some of the pump components. Wearing rings, bushings, seals or packing and internal sleeve bearings all need liquid for lubrication and may seize if the pump is run dry.

11.1.2. Starting the pump

The pump is started with the gate valve closed. This is because the pump operates at only 30-50% of full load when the discharge gate valve is closed. In cases where the pump is below the water source, the pump can be started with an open gate valve. To avoid water hammer, the gate valve has to be opened gradually until it is fully open.

11.1.3. Stopping the pump

The first step is to close the gate valve. This eliminates surges that may occur in case of an abrupt closure. When this has been done, the prime mover is then closed or shut down. If the pump remains idle for a long time after it is stopped, it gradually looses its priming. Thus the operator should re-prime the pump every time before start-up.

11.2. Pump malfunctions, causes and remedies (troubleshooting)

Following are some general causes of pump malfunctioning and their remedies that can be used for on-spot trouble-shooting when pump problems are encountered. Cornell pump manufacturers provide useful information, presented in Table 6, for identifying and rectifying problems with pumping plants.

Table 6
Pump problems, causes and corrections

SYMPTOMS	CAUSES	CORRECTIONS
Failure to pump	1. Pump not properly primed	1. Prime pump correctly
	2. Speed too low or high	2. Check speed, check calculations, consult with manufacturer
	3. Not enough head to open check valve	3. Check speed, check calculations, consult with manufacture
	4. Air leak	4. Check and rework suction line
	5. Plugged section	5. Unplug section
	6. Excessive suction lift	6. Check NPSH and consult manufacturer
Rapid wear of coupling cushion	7. Misalignment	7. Align
	8. Bent shaft	8. Replace
Reduced performance	9. Air pockets or small air leaks in suction line	9. Locate and correct
	10. Obstruction in suction line or impeller	10. Remove obstruction
	11. Insufficient submergence of suction pipe	11. Extend suction line to deeper water to the extent that NPSH allows you or excavate and deepen the area where the suction basket is located
	12. Excessively worn impeller or wear ring	12. Replace impeller and/or wear ring
	13. Excessive suction lift	13. Calculate NPSH, consult with manufacturer
	14. Wrong direction of rotation	14. Ask contractor to rectify
Driver overloaded	15. Speed higher than planned	15. Reduce speed
	16. Water too muddy	16. Raise suction
	17. Too large an impeller diameter	17. Trim impeller
	18. Low voltage	18. Consult power authority
	19. Stress in pipe connection to pump	19. Support piping properly
	20. Packing too tight	20. Loosen packing gland nuts
Excessive noise	21. Misalignment	21. Align all rotating parts
	22. Excessive suction lift	22. Check NPSH, consult with manufacturer
	23. Material lodged in impeller	23. Dislodge obstruction
	24. Worn bearings	24. Replace bearings
	25. Impeller screw loose or broken	25. Replace
	26. Cavitation	26. Check NPSH, correct suction piping
	27. Wrong direction of rotation	27. Ask contractor to rectify
Premature bearings failure	28. Worn wear ring	28. Replace
	29. Misalignment	29. Align all rotating parts
	30. Suction or discharge pipe not properly supported	30. Correct support
	31. Bent shaft	31. Replace shaft
Electric motor failure	32. High or low voltage	32. Check voltage and consult power authority
	33. High electric surge	33. Monitor voltage and consult power authority
	34. Poor electric connection	34. Turn power off, clean and check connections
	35. Overloads	35. Check amperage, do not exceed full load amperage
	36. Bearing failure	36. Change motor bearing
	37. Cooling vent plugged (rodent, dirt, leaves)	37. Install proper screen
	38. Moisture or water in motor	38. Send for blow-dry and protect from environment

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