23 Pumps and Pumping Stations

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23.1 General

Many agricultural areas are located along a river or in the vicinity of a lake or sea. Frequently, the required drainage water levels in these areas are lower than the water level of the river, lake, or sea. Under such circumstances water cannot be drained out of the area by gravity flow, but must be pumped out.

Thousands of years ago, the ancients had already developed water-lifting devices for use in irrigation. Men, beasts, and in some cases running water provided the driving force. With the water-powered types, lifts as high as 15 m were possible. Some of these devices, such as the water wheel and the Archimedean screw, are still in use today in their original form (see Figure 23.1).

For hundreds of years, the use of water-lifting devices was limited because they could not be connected to a pressure pipe. This limitation was overcome by the introduction of the impeller pump. The first known operational impeller pump was used in a Portuguese copper mine in the 15th century, it is now on exhibit in the Musée du Conservation National des Arts et Métiers in Paris.

The aim of this chapter is to provide the design engineer with basic information on pump performance characteristics; information that will enable him to make a

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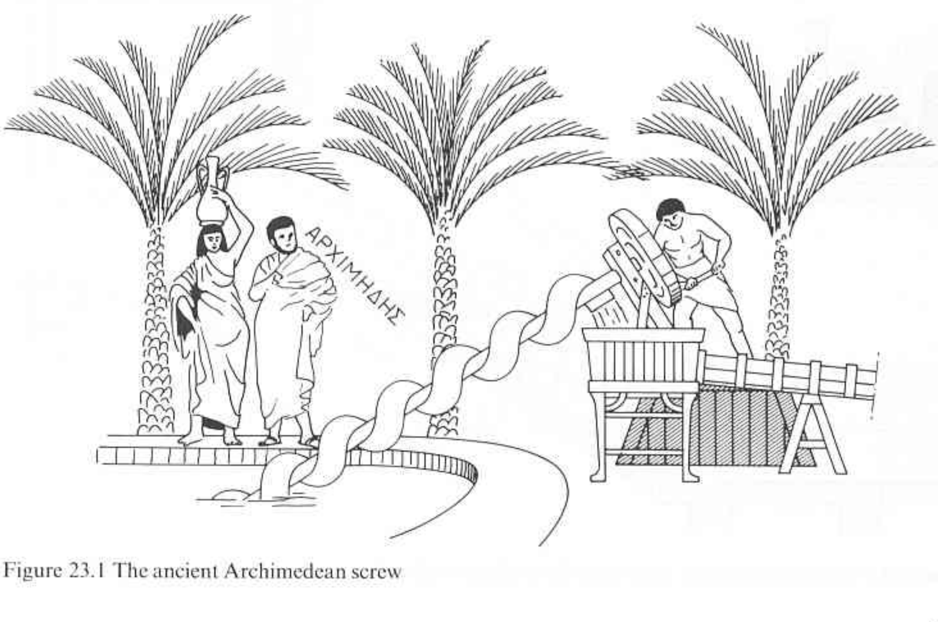

Figure 23.1 The ancient Archimedean screw

first estimate of the number and the dimensions of the pumps he will need, and will provide him with a sound basis for choosing the right pump for the job.

23.2 Pump Types

Essentially, every pump consists of two parts: a rotating part called a runner or impeller, and a stationary part called a casing or housing. When power is applied to the shaft of the runner, water can be displaced (as with an Archimedean screw), or can be forced into a rotary motion and led away under pressure (as with an impeller pump). Rotating impeller pumps are classified by the direction in which the water flows through the impeller. The three possible types are: the radial-flow (or centrifugal) pump, the mixed-flow pump, and the axial-flow pump.

Each type of water-lifting device has specific characteristics which fit it for specific operating conditions.

23.2.1 Archimedean Screw

Description

The modern Archimedean screw (Figure 23.2) is based on the ancient device. It can lift large volumes of water against low heads, and so is popular for use in drainage systems in flat countries like The Netherlands.

Basically, the screw consists of an inclined shaft to which one or more helically wound blades are attached. This spiral is fitted into a semi-circular casing. When the screw is rotated, the water confined between two successive blades, the wall of the casing, and the shaft is lifted.

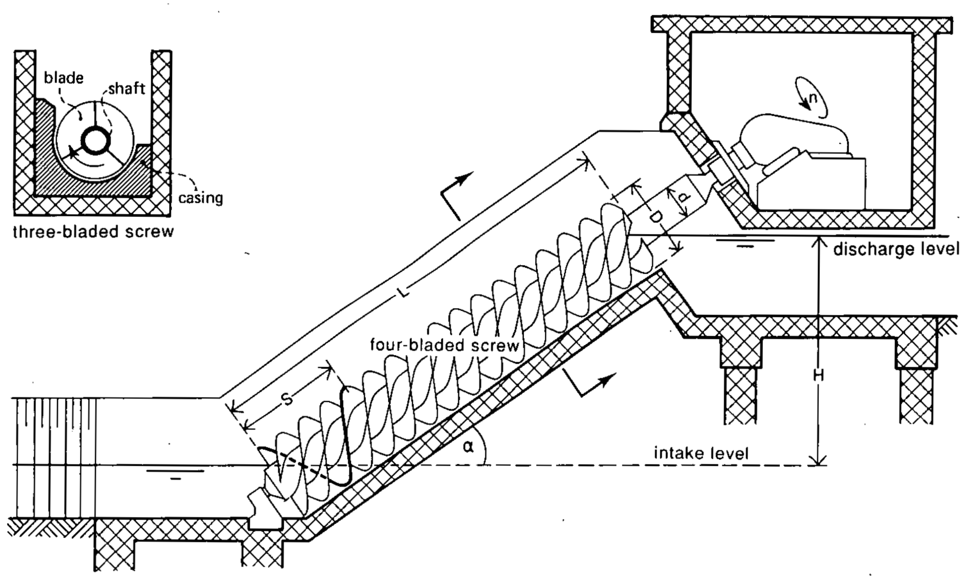

Figure 23.2 Archimedean screw, open type (from Hydro Delft 1972)

The screw has the following advantages over the impeller pump:
- The rotational speed of the screw is low, varying from 1/3 revolutions per second (rev/s) for large diameter screws to 2 rev/s for small diameter screws, hence wear is negligible and no cavitation will occur (see Section 23.4);
- The opening between two successive blades is relatively large, so the screw is able to handle relatively trash-laden water;
- The flow distribution on the suction side of the screw does not influence the hydraulic behaviour; hence simple suction basins are possible;
- Above a certain suction level (filling point), the discharge remains constant while efficiency remains favourable;
- Below a certain suction level, the discharge decreases while the efficiency remains favourable;
- The screw can turn continuously without risk of damage, even when the water supply stagnates;
- The screw has an open construction which makes it possible to inspect the entire lifting operation;
- In some instances the foundation of the screw need not be as deep as that of an impeller pump.

On the other hand, the screw has the following disadvantages compared with the impeller pump:
- The dimensions of the screw are larger than those of impeller pumps. This is necessary because the water pressure remains atmospheric during the lifting process;
- The screw's capabilities are restricted to pumping from one free surface 'reservoir' to another free surface 'reservoir'. Any connection to pressure piping is impossible;
- To operate with reasonable efficiency, the intake level should be above a certain value;
- Discharge levels must remain within a narrow range. Too high a level leads to backflow over the outer rim of the blades. Too low a level means that the water is lifted needlessly high;
- The whole structure must be rigid to keep the clearance between the rotating screw and the casing small, and thus prevent backflow;
- For reasons of safety the screw must be covered.

In general, the Archimedean screw is suitable as a water-lifting device provided that both the intake and discharge levels remain within narrow limits (Photo 23.1).

Discharge Characteristics
The discharge delivered by the screw is proportional to the quantity of water between two successive blades turned 360°, the number of blades on the srew (x), and the number of revolutions of the screw per unit of time (n). The quantity of water between two successive blades depends on the outer diameter of the screw (D), the diameter of the shaft (d), the pitch of the blades (S), and the angle of inclination of the shaft (α). This leads to the following basic formula for the discharge of a screw

$$Q = k \times n \times D^3 \tag{23.1}$$

Photo 23.1 Small Archimedean screw in a drainage canal

in which

Q = discharge (m^3/s)
n = number of revolutions (rev/s)
D = outer diameter of the screw (m)
k = constant, which depends on the shape of the screw characterized by S/D, d/D, and the inclination of the screw characterized by α (dimensionless)

Values of k for screws with three or four blades are given in Table 23.1. Such multi-bladed screws can achieve capacities of up to 5 m^3/s. A head of 10 m is possible with a modern screw. To attain more head, intermediate bearings can be applied or structures with two screws in series can be used.

The hydraulic behaviour of a screw is described by the discharge versus intake-level characteristic and the hydraulic-efficiency versus intake-level characteristic (see Figure 23.3). Both refer to an optimal discharge level as indicated above.

Table 23.1 k values for three- and four-bladed screws

d/D	$\alpha = 22°$		$\alpha = 26°$			$\alpha = 30°$	
	S = 1.0D	1.2D	0.8D	1.0D	1.2D	0.8D	1.0D
0.3	0.331	0.335	0.274	0.287	0.286	0.246	0.245
0.4	0.350	0.378	0.285	0.317	0.323	0.262	0.271
0.5	0.345	0.380	0.281	0.317	0.343	0.319	0.287
0.6	0.315	0.351	-	0.300	0.327	-	0.273

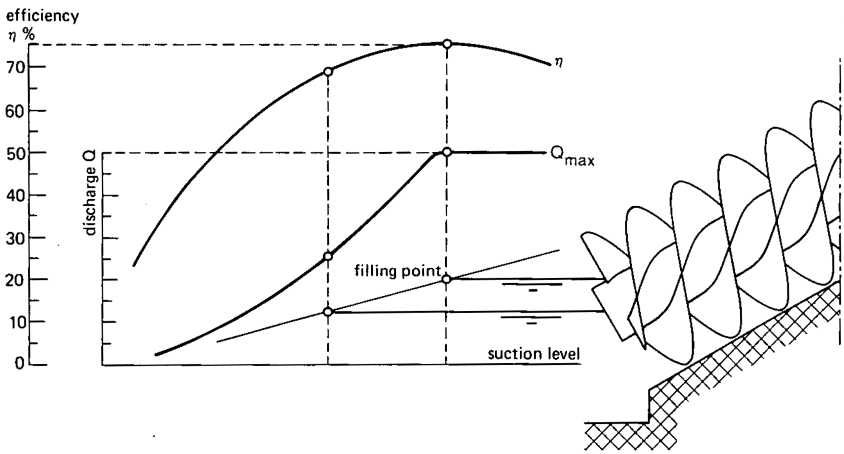

Figure 23.3 Discharge and efficiency versus intake level

The ratio between the hydraulic output power and the shaft mechanical input power is called the efficiency, which is hereby defined as

$$\eta = \frac{\rho g Q H}{N.2\pi n} \tag{23.2}$$

in which

'ρgQH'	=	hydraulic output power = transported mass per unit time × gravity acceleration × head (W)
ρ	=	density of the pumped water (about 1000 kg/m^3)
g	=	acceleration due to gravity (9.81 m/s^2)
Q	=	discharge (m^3/s)
H	=	head delivered by the pump (m)
'$N.2\pi n$'	=	shaft input power = torque on the shaft × angular speed of the shaft (W)
N	=	torque on the shaft (J)
n	=	speed of the shaft (rev/s)

Because the water velocities in a screw are low, the conversion of static suction energy via kinetic energy into static pressure energy can occur with low losses.

Efficiencies of 0.65 can be achieved for small diameter screws and of up to 0.75 for large diameter screws.

Tentative Dimensioning

For a required discharge and head, a tentative estimate of the screw dimensions can be made with the following four design formulas

$$D = \sqrt[3]{\frac{Q}{kn}} \tag{23.3}$$

$$n = \frac{0.85}{\sqrt[3]{D^2}} \tag{23.4}$$

$$L = \frac{H}{\sin \alpha} \tag{23.5}$$

$$d = \frac{L}{20} \tag{23.6}$$

in which

Q = discharge (m^3/s)
H = static head (m)
n = number of revolutions (rev/s)
k = constant (–)
D = diameter of the screw (m)
L = length of the screw (m)
α = inclination of the screw (degrees)
d = diameter of the shaft (m)

For three or four bladed screws with $22° < \alpha < 30°$, k values can be taken from Table 23.1. For definitive dimensioning screw manufacturers' specifications should be consulted.

23.2.2 Impeller Pumps

Description
Impeller pumps were developed from the need for one compact device to incorporate various lifting capabilities (combinations of discharge and head) as well as from the need to connect a pressure pipe to the outlet of the pump.

The hydraulic part of an impeller pump (see Figure 23.4) consists of an impeller, equipped with vanes, which forces the liquid into a relatively fast rotary motion, and a casing which directs the liquid from a suction opening to the impeller eye, and then leads it away from the outlet of the impeller to the pressure opening. Both openings can be connected to different devices: suction bells, steel piping, or concrete canals. The impeller is mounted on a shaft which is supported by bearings, and rotated by an engine through a flexible or rigid coupling. The impeller vanes and impeller side walls (shrouds) form the impeller canals. The pump casing has several functions: it encompasses the suction and discharge openings, supports the bearings, and houses the impeller assembly. The casing is sealed off around the shaft to prevent external leakage. Closely fitted rings (wearing rings) are mounted on the impeller and fitted in the casing to restrict leakage of high-pressure liquid from the impeller outlet to the impeller inlet.

As illustrated in Figure 23.5, different types of impeller and casing are possible. The three main types are: (i) the centrifugal pump with axial inflow, and a radial and a tangential outflow component; (ii) the mixed flow pump with axial inflow and axial,

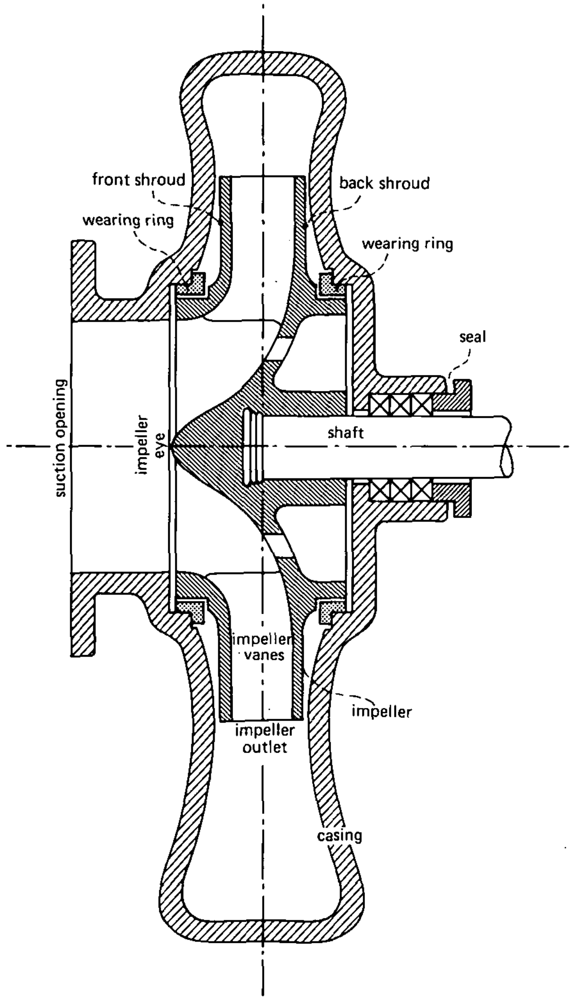

Figure 23.4 Section of a centrifugal pump (from Stepanoff 1957)

radial, and tangential outflow components; and (iii) the axial flow pump with axial inflow and axial and tangential outflow components.

Impeller pumps can be equipped with a vertical, inclined, or horizontal shaft, depending on the application, type of driver, or other requirements. Modern drivers are high-speed electric motors, internal-combustion engines, and steam turbines.

Hydraulic Behaviour of Impeller Pumps

The theory of lifting water by centrifugal force was first suggested by Leonardo da Vinci (1452-1519). The pump, as it is used today, is based on the invention of the French physicist Denis Papin (1647-1714). Both men noted that the pressure (head)

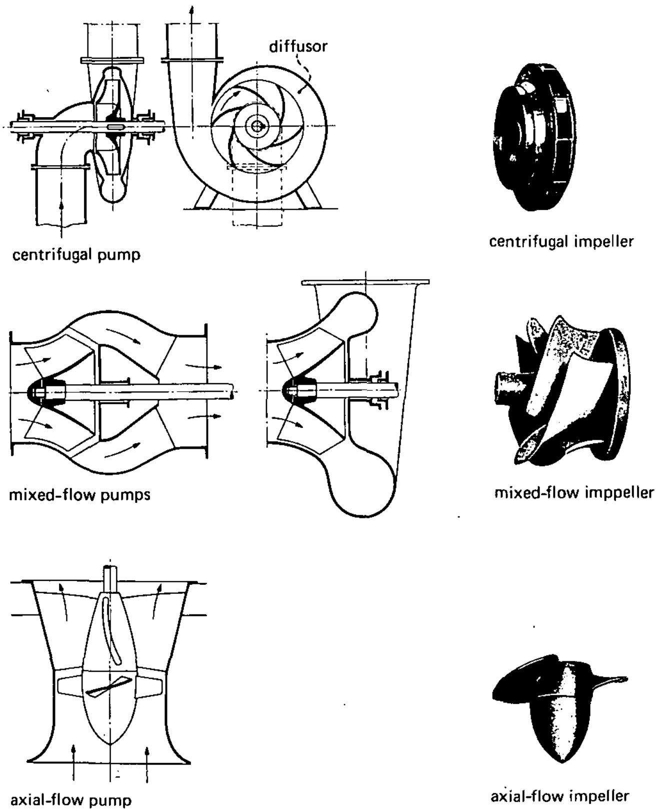

Figure 23.5 Examples of impeller pumps and impeller shapes (after Lazarkiewics and Troskolanski 1965)

of a pumped liquid could be raised if the liquid's momentum was increased by means of a rotating impeller as it flowed through the impeller canals.

Appropriately shaped impeller vanes cause both a reduction in pressure at the impeller eye and a rise in pressure at the impeller outlet. This, in turn, causes liquid to be drawn from the suction opening into the impeller canals and to be delivered from them, via an outlet element (volute or diffusor), to the pressure opening. Because the liquid flowing through the impeller is accelerated during this process, the kinetic energy of the liquid is raised. This kinetic energy is mainly transferred into pressure energy in the outlet element of the pump. As the liquid passes through the impeller and the outlet element, both friction losses and eddy losses occur. These form the major part of the power losses as the input power of the driver is transformed into the hydraulic power of the pump (see Figure 23.6).

For a given impeller pump running at a constant speed, there is only one combination of head and discharge at which the sum of the friction and eddy losses is minimal. In other words, there is only one point at which the given impeller pump can work with maximum efficiency.

The hydraulic behaviour of an impeller pump is described by the discharge versus head characteristic, the discharge versus shaft-power characteristic, and the discharge

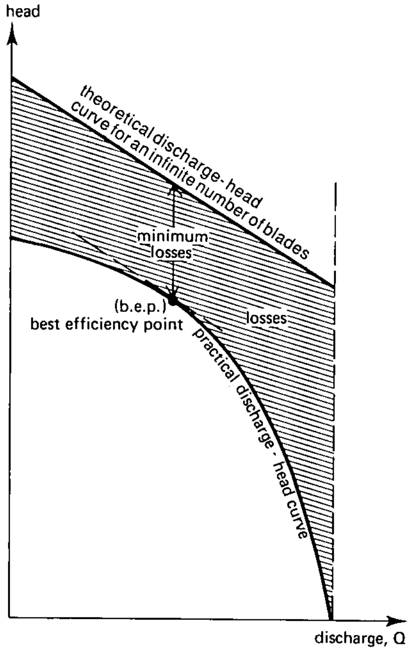

Figure 23.6 Theoretical and practical discharge-head curves for an impeller pump

versus hydraulic-efficiency characteristic (see Figure 23.7). The last-mentioned follows directly from the first two.

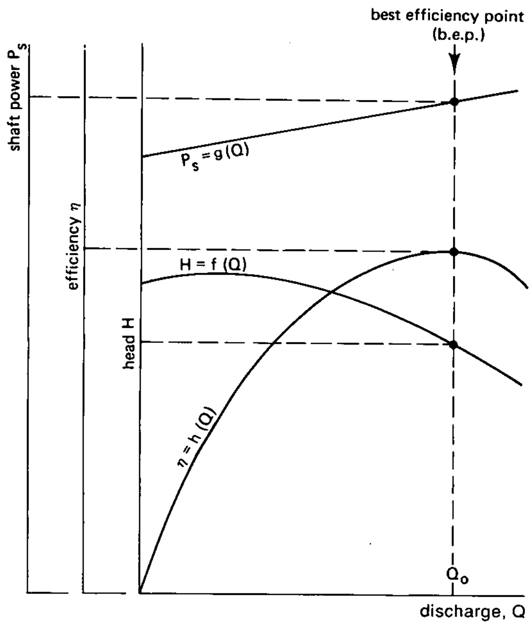

Figure 23.7 Hydraulic characteristics of an impeller pump

The efficiency, η, is hereby similar to that defined in Equation 23.2, i.e.

$$\eta = \frac{\rho g Q H}{N \times 2\pi n} \tag{23.7}$$

The power supply to the pump shaft is defined by

$$P_s = \frac{\rho g Q H}{\eta} \tag{23.8}$$

in which

P_s = power to be delivered to the shaft of the pump (Watt)

This power requirement should be determined on the basis of the least favourable hydraulic conditions under which the pump must run, i.e. under conditions that require the highest power consumption.

Each pump's particular characteristics suit it to a particular task. Under conditions of falling discharge, axial flow pumps show rather steeply rising discharge-head and discharge-power curves. An axial-flow pump is the logical choice for pumping high discharges at relatively low heads (see Figure 23.8). Under the same conditions of falling discharge, centrifugal pumps, on the other hand, show only a slight rise in head and a slight fall in power. Such pumps are suitable for pumping low discharges at relatively high heads. Mixed flow pumps, as their name indicates, are intermediate types. The typical operating range of the various pumps is illustrated in Figure 23.9 (Addison 1966; De Kovats and Desmur 1968).

Specific Speed

Since eddy losses and friction losses greatly influence the attainable efficiency, it is clear that a pump's best possible efficiency is governed by its shape, and thus by its type. The pump type can be characterized by the 'specific speed' related to the discharge and head at the best efficiency point. Presentation in dimensionless shape leads to

$$n_s = \frac{\omega\sqrt{Q_o}}{(gH_o)^{0.75}} \tag{23.9}$$

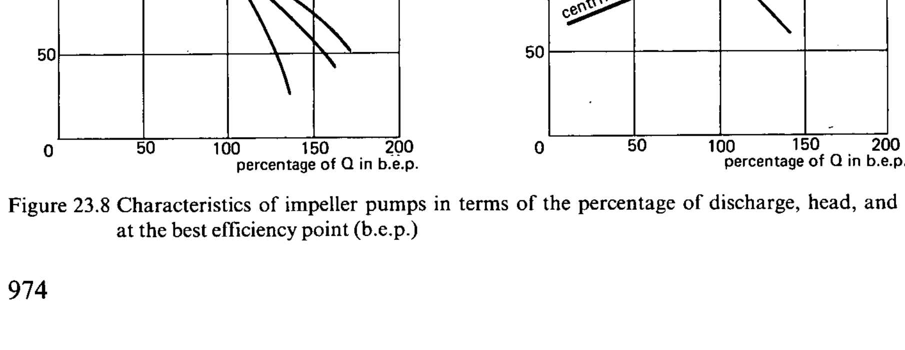

Figure 23.8 Characteristics of impeller pumps in terms of the percentage of discharge, head, and power at the best efficiency point (b.e.p.)

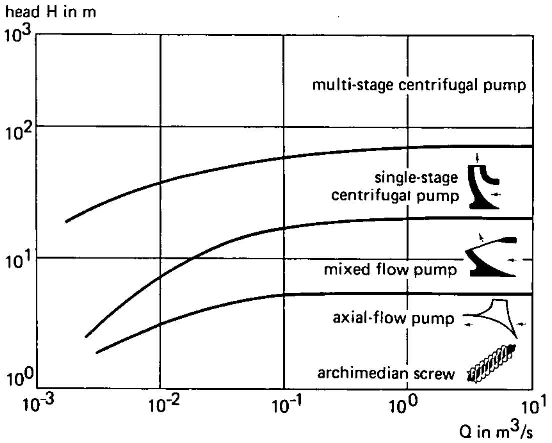

Figure 23.9 Approximate operating range of pumps (upper limits are shown!)

in which

n_s = specific speed (–)
ω = speed of the impeller (rad/s) ($\omega = 2\pi n$ with n in rev/s)
Q_o = discharge at the best efficiency point (m^3/s)
g = acceleration due to gravity (m/s^2)
H_o = head delivered by the pump at the best efficiency point (m)
($H = p/\rho g$ in which p is pressure rise of the water between suction and pressure opening in N/m^2)

Unfortunately, many dimensional definitions of the specific speed are applied in practice, including the much-used n_{sq}

$$n_{sq} = \frac{n\sqrt{Q_o}}{H_o^{3/4}} \tag{23.10}$$

in which

n_{sq} = specific speed ($m^{0.75}/\text{min}\ s^{0.50}$)
n = speed of the impeller in revolutions per minute (rev/min)
Q_o = discharge at the best efficiency point (m^3/s)
H_o = head delivered by pump at the best efficiency point (m)

For different pump types defined by their specific speed, n_{sq}, Figure 23.10 shows typical discharge-head and discharge-power characteristics which refer to the discharge, head, and power at the best efficiency point.

Shown in Figure 23.11 are half sections of typical impeller shapes and the height of their attainable efficiency (best efficiency points) as a function of the specific speed, n_{sq}. It is clear that the highest efficiencies are possible with pumps in the specific speed range of 40 to 120 $m^{0.75}/\text{min}\ s^{0.50}$.

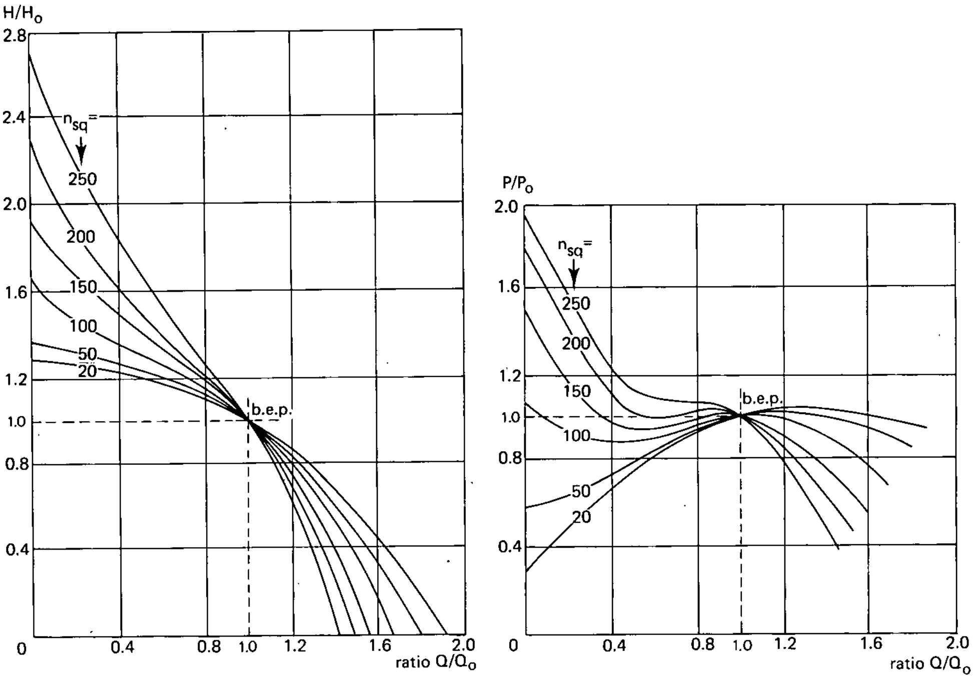

Figure 23.10 Discharge-head and discharge-power characteristics of impeller pumps (n_{sq} in metric units)

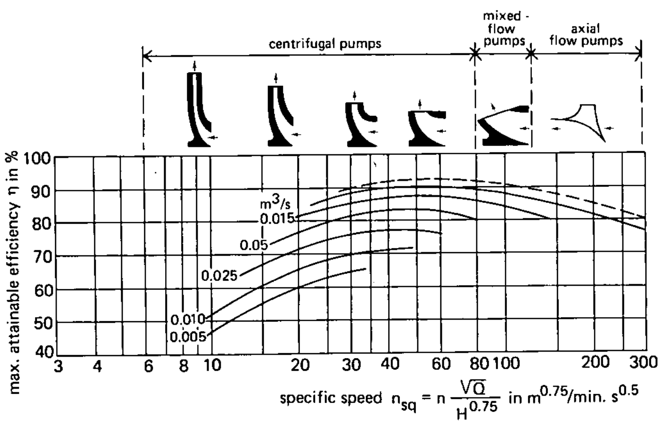

Figure 23.11 Maximum attainable pump efficiency as a function of n_{sq} (from Lazarkiewics and Troskolanski 1965)

23.3 Affinity Laws of Impeller Pumps

The hydraulic characteristics of two pumps of identical design and shape (conformable pumps), but of different size and with impellers that run at different revolutions per unit of time, are coupled through affinity laws. If one knows the discharge-head characteristic and the discharge-power characteristic of an 'existing' pump (a real or model pump as tested by the manufacturer or an independent laboratory), it is possible to estimate similar characteristics for a conformable pump of different dimensions and running at a different speed (see Figure 23.12).

The three dimensionless affinity laws are (Karassik et al. 1976; Schulz 1977)

$$\frac{Q_e}{Q_c} = \left(\frac{D_e}{D_c}\right)^3 \frac{n_e}{n_c} \tag{23.11}$$

$$\frac{H_e}{H_c} = \left(\frac{D_e}{D_c}\right)^2 \left(\frac{n_e}{n_c}\right)^2 \tag{23.12}$$

$$\frac{P_e}{P_c} = \left(\frac{D_e}{D_c}\right)^5 \left(\frac{n_e}{n_c}\right)^3 \tag{23.13}$$

in which

Q_e = discharge of the existing pump
Q_c = discharge of the conformable pump
D_e = the outlet diameter of the impeller of the existing pump
D_c = the outlet diameter of the impeller of the conformable pump
n_e = speed of the existing pump
n_c = speed of the conformable pump
H_e = head of the existing pump
H_c = head of the conformable pump
P_e = power consumption of the existing pump
P_c = power consumption of the conformable pump

These affinity laws hold for conformable working conditions in both pumps, or in

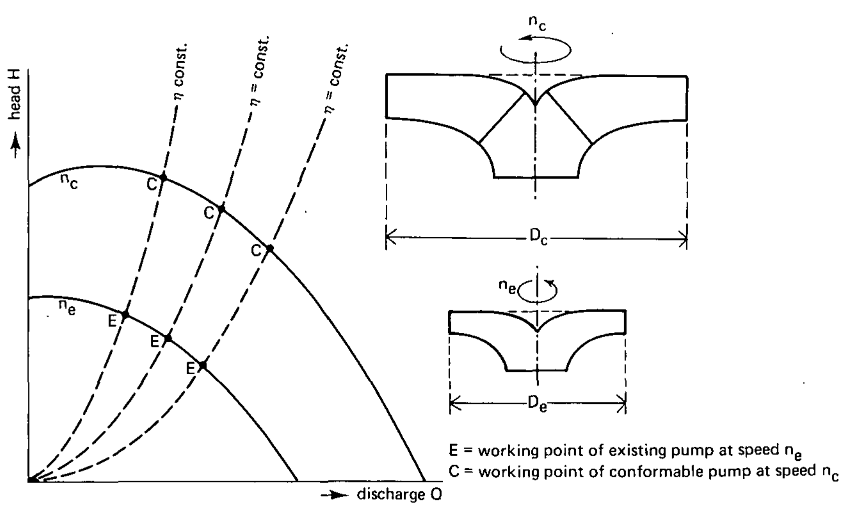

Figure 23.12 Discharge-head curves of an 'existing' and a conformable impeller pump

other words, for conformable flow patterns in both pumps. Because viscosity effects are neglected in these equations, both the existing and the conformable pump will operate at equal efficiencies ($\eta_c = \eta_e$).

If the pump dimensions differ, but the speed of the existing and conformable impeller are the same ($n_e/n_c = 1$). Equations 23.11 to 23.13 can be used to derive the hydraulic characteristics for the conformable impeller diameter, D_c.

From the affinity laws, a set of simplified rules can be derived to determine the hydraulic characteristics when the pump dimensions remain constant and only the impeller speed is changed.
They read

$$\frac{Q_e}{Q_c} = \frac{n_e}{n_c} \tag{23.14}$$

$$\frac{H_e}{H_c} = \left(\frac{n_e}{n_c}\right)^2 \tag{23.15}$$

$$\frac{P_e}{P_c} = \left(\frac{n_e}{n_c}\right)^3 \tag{23.16}$$

$$\eta_c = \eta_e \tag{23.17}$$

23.4 Cavitation

23.4.1 Description and Occurrence

As mentioned when the hydraulic behaviour of impeller pumps was being discussed, low pressures will occur at the impeller inlet. Generally, these low pressures are physically restricted to the vapour pressure of the pumped liquid at the prevailing temperature.

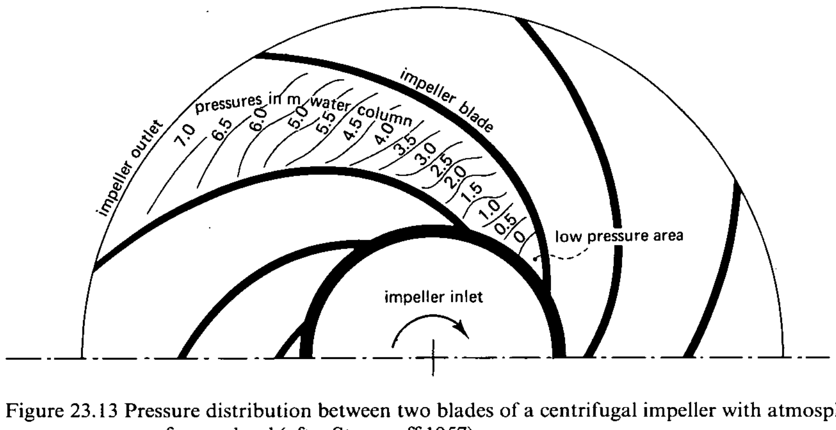

Figure 23.13 Pressure distribution between two blades of a centrifugal impeller with atmospheric pressure as reference level (after Stepanoff 1957)

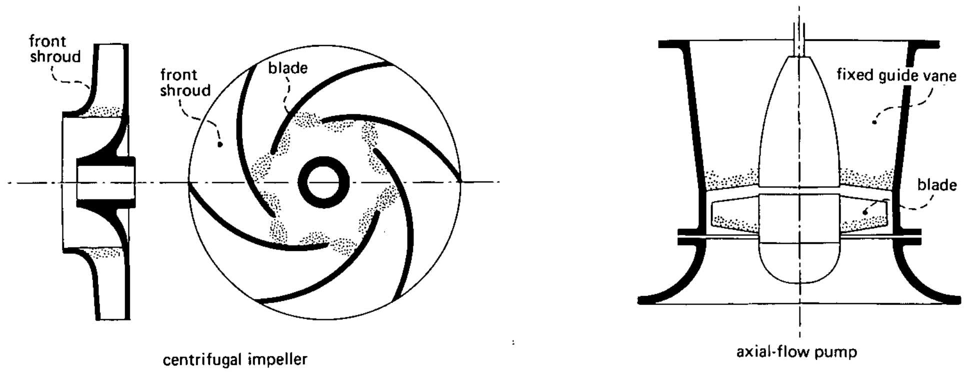

Figure 23.14 Pitting due to cavitation in impeller pumps (after Lazarkiewics and Troskolanski 1965)

An example of such pressure distribution is shown in Figure 23.13. Upon reaching the vapour pressure, the liquid starts to boil and forms bubbles which are then carried away with the liquid. When the bubbles reach areas of higher pressures, they collapse. This phenomenon, known as cavitation, is accompanied by typical noises. In small pumps, light vibrations due to light cavitation sound like 'the frying of bacon'; in large pumps heavy vibrations due to heavy cavitation sound like 'the pumping of boulders'.

Cavitation may lead to a change in the pump's hydraulic behaviour and to heavy wear on its structural parts, thus reducing its life. Simply put, cavitation batters the material. As early as 1933, pressure pulses up to 300 bar and with a frequency of 25 000 Hz were observed in experiments conducted by Haller. Figure 23.14 shows the points in impeller pumps where pitting due to cavitation is most likely to occur.

A cavitation attack will damage pumps of different materials at greatly different rates (see Figure 23.15). This damage is further influenced by the properties of the pumped water, e.g. vapour pressure, dissolved gas content, and free gas content.

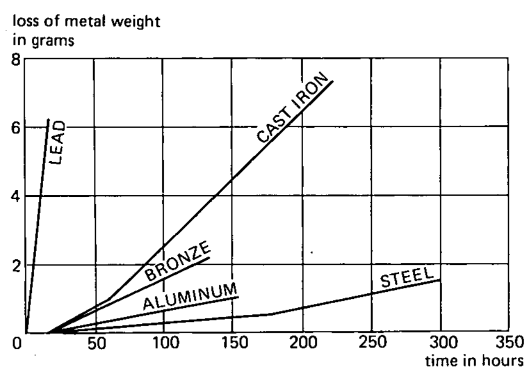

Figure 23.15 Rate of cavitation damage for various construction materials (from Stepanoff 1957)

23.4.2 Net Positive Suction Head (NPSH)

The cavitation behaviour in an impeller pump is described by the discharge-NPSH curve, an example of which is shown in Figure 23.16. NPSH stands for Net Positive Suction Head. It is defined as the margin between the absolute energy head (absolute static pressure head + velocity head) in the suction opening of the pump and the vapour pressure head of the pumped liquid at the prevailing temperature. Both of these heads refer to the centre of the pump's suction opening.

As illustrated in Figure 23.17, the NPSH thus equals

$$NPSH = H_A + h_o - h_v \tag{23.18}$$

in which

NPSH	=	Net Positive Suction Head (m)
$H_A + h_o$	=	absolute energy head in the suction opening of the pump (m)
H_A	=	energy head in the suction opening of the pump with atmospheric pressure as reference level (m) (see Figure 23.19)
h_o	=	absolute air pressure head above the suction reservoir (m)
h_v	=	vapour pressure head of the pumped liquid (m)

Absolute air pressure at sea level is about 10 m (1 bar = 100 kPa). The vapour pressure head of fresh water at temperatures of 15 °C to 30 °C is 0.4 m (0.04 bar = 4 kPa).

Besides the NPSH value, it is common practice to use the dimensionless Thoma number σ (see Figure 23.16)

$$\sigma = \frac{NPSH}{H} \tag{23.19}$$

The discharge-NPSH curve in Figure 23.16 gives an indication of the pressure head that is required in the pump's suction opening to keep pressures inside the pump above

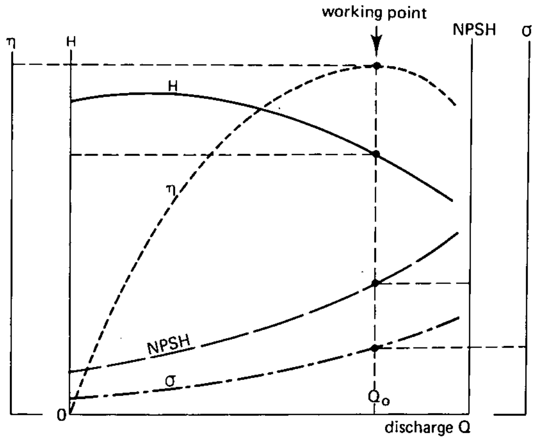

Figure 23.16 Illustration of discharge-NPSH curve and discharge-Thoma-number curve for an impeller pump

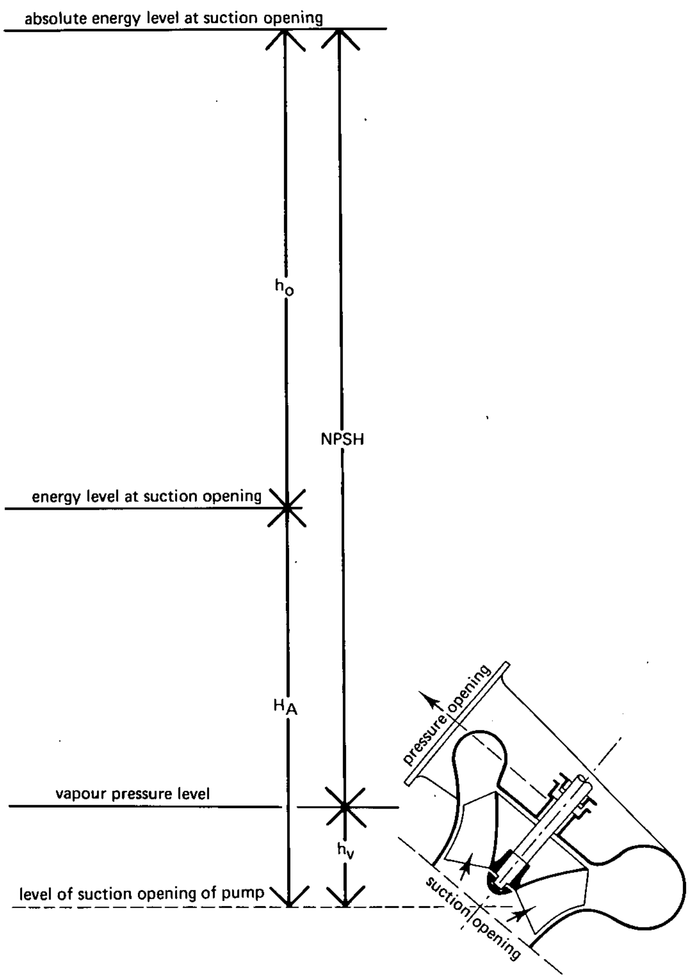

Figure 23.17 Illustration of Net Positive Suction Head (NPSH)

a particular level, and thus prevent what could be referred to as a 'certain amount of cavitation'. This 'certain amount of cavitation' may be defined as no cavitation at all, a certain level of vibrations and noise, a fall in pump head as compared with cavitation-free pumping (0.1 to 0.2% or 1 to 2%), or some erosion.

The amount of cavitation that can be tolerated is a question of economics. For example, the criterion of no cavitation at all leads to a high, required NPSH value. And, since the h_o and h_v values in Equation 23.18 cannot be altered, the criterion leads to a high H_A value (deep submergence of the pump, see also Figure 23.19A), or to a large noiseless pump, running at a low speed over a long life. Using the criterion of a 1 to 2% fall in pump head, on the other hand, would indicate a noisy, heavily cavitating pump with a relatively short life because of cavitation erosion.

Figure 23.18 is based on the results of numerous cavitation tests on various types of pumps. It shows a useful relation between the different pumps, characterized by