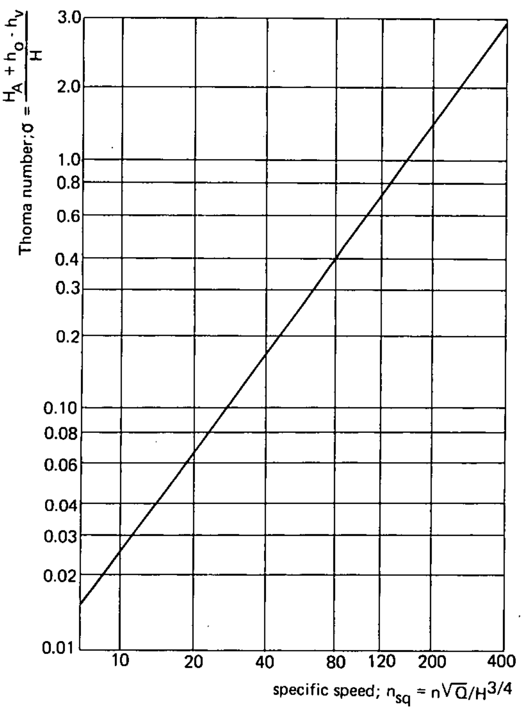

Figure 23.18 Relation between the minimum required Thoma number and the specific speed (from Stepanoff 1957)

the specific speed, n_{sq}, at their best efficiency points, and the minimum σ value at which proper performance of the relevant pump is still guaranteed. The graph is based on a small drop in head of 0.1 to 0.2% due to cavitation as compared with cavitation-free pumping.

To prevent the 'certain amount of cavitation' described above, it is important that the available NPSH for each pump and its pump system be higher than the NPSH required by the manufacturer for that typical criterion. In the preliminary stage of a project, required values of σ, and so for NPSH, can be taken from Figure 23.18.

23.5 Fitting the Pump to the System

23.5.1 Energy Losses in the System

Irrespective of the elevation of the pump, it must deliver a pump head, which actually consists of three heads (see Figure 23.19)

$$H = H_{st} + \Delta H_s + \Delta H_{pr} \tag{23.20}$$

in which

H = manometric or total head to be delivered by the pump (m)

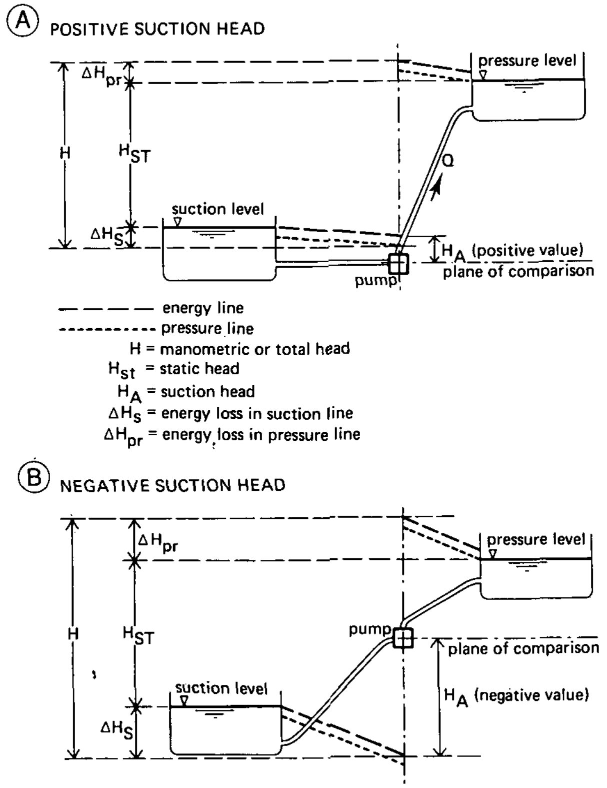

Figure 23.19 Schematic presentation of a pump in a simple system (after Wijdieks 1972)

H_{st} = static head, or the difference between the suction reservoir and the pressure reservoir (m)
ΔH_s = hydraulic loss on the suction side of the pump (m)
ΔH_{pr} = hydraulic loss on the pressure side of the pump (m)

In the above equation, ΔH_s and ΔH_{pr} can be expressed as

$$\Delta H_s = k_s \frac{v_s^2}{2g} + f_s \frac{L_s}{D_s} \frac{v_s^2}{2g} \tag{23.21}$$

and

$$\Delta H_{pr} = k_{pr} \frac{v_{pr}^2}{2g} + f_{pr} \frac{L_{pr}}{D_{pr}} \frac{v_{pr}^2}{2g} \tag{23.22}$$

in which
k_s = loss coefficient of the trash rack, suction piping, bends etc. (–)
v_s = reference velocity in the suction pipeline, if any (m/s)
g = acceleration due to gravity (m/s²)
f_s = friction factor of the suction pipe, if any (–)

L_s = length of the suction pipe, if any (m)
D_s = diameter of the suction pipe, if any (m)
k_{pr} = loss coefficient of the pressure piping, bends, valves etc. (–)
v_{pr} = reference velocity in the pressure piping (m/s)
f_{pr} = friction factor of the pressure piping (–)
L_{pr} = length of the pressure piping (m)
D_{pr} = diameter of the pressure piping (m)

Equation 23.20 can also be expressed in terms of pump head, H, and pump discharge, Q, leading to the following hydraulic characteristic for the system

$$H = H_{st} + CQ^2 \tag{23.23}$$

in which

C = constant, which depends on hydraulic losses in the system following from Equations 23.21 and 23.22 (s^2/m^5)

The static head, H_{st}, is equal to the difference in the water level between the pressure and suction side of the pumping station, and is entirely independent of the head losses in the system and the discharge delivered by the pump. The static head changes only if the suction level or pressure level changes. The term CQ^2 in Equation 23.23 can be calculated for several values of Q, enabling the head, H, to be plotted against the pumped discharge, as illustrated in Figure 23.20A.

23.5.2 Fitting the System Losses to the Pump Characteristics

As was shown in Figure 23.7, the hydraulic characteristics of a pump are given by two independent functions and one derived function. They are

$$H = f(Q) \tag{23.24}$$

$$P_s = g(Q) \tag{23.25}$$

$$\eta = h(Q) \tag{23.26}$$

These three functions and Equation 23.23 are plotted in Figures 23.20A, B and C. A further study of Figure 23.20A shows a point of intersection between the head-discharge curve of the system (Equation 23.23) and the head-discharge curve of the pump (Equation 23.24). This point is called the working point. If a pump running at a constant speed, with characteristic H = f(Q), is placed in the system, that pump will deliver a discharge, Q, at the related head, H, defined by the working point.

The required power of the driver, P_d, the pump efficiency, η, and the required NPSH can be read immediately from Figure 23.20B, C, and D, respectively.

23.5.3 Post-Adjustment of Pump and System

Because the hydraulic losses in the system, as calculated by Equations 23.21 and 23.22, are subject to errors in the estimate of coefficients and factors, the real energy losses

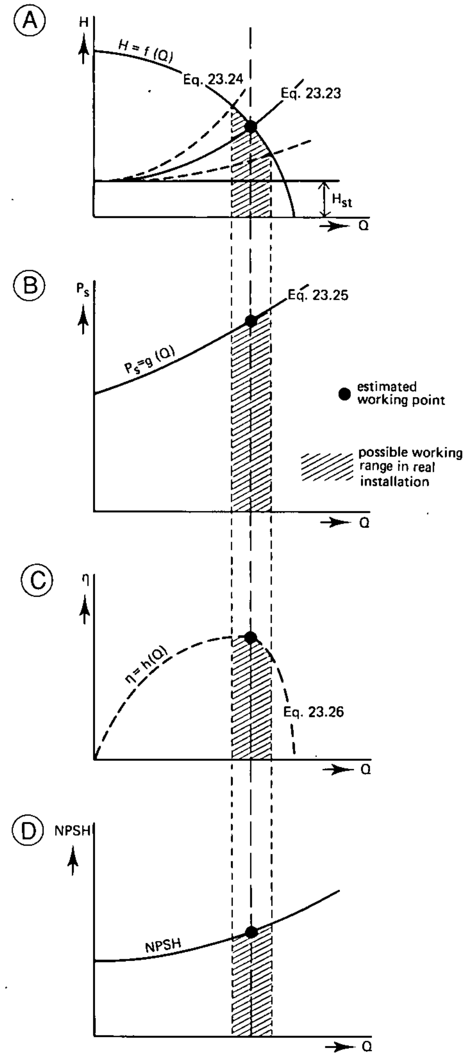

Figure 23.20 Combination of pump and pumping system

may be either lower or higher. As Figure 23.20A illustrates, the characteristic (Equation 23.23) of the real system intersects the pump characteristics at another operating point. This leads to a higher or lower pumped discharge, a different power consumption, and a lower efficiency than expected. Also, unexpected cavitation may occur because of higher required NPSH values (see Figure 23.20D) at an unplanned operating point in the pumping station.

Post-adjustment of the hydraulic characteristic of either the system or the pump may be needed. Obviously, the point of intersection of the two curves in Figure 23.20A will shift with a change in either curve. The shape of the system's curve can be changed only by making adjustments in its hydraulic design, which means that the energy loss coefficient will change as well.

The pump characteristic, H = f(Q) can be changed either by some cutting at the outlet end of the impeller vanes, or by replacing the whole impeller by a more suitable one. If cutting is done, the hydraulic characteristics after cutting can be estimated with the following three empirical equations

$$Q_{cut} = \left(\frac{D_{cut}}{D_{org}}\right)^2 \times Q_{org} \tag{23.27}$$

$$H_{cut} = \left(\frac{D_{cut}}{D_{org}}\right)^2 \times H_{org} \tag{23.28}$$

$$P_{cut} = \left(\frac{D_{cut}}{D_{org}}\right)^4 \times P_{org} \tag{23.29}$$

in which

Q_{cut} = discharge after cutting (m^3/s)
Q_{org} = original discharge (m^3/s)
D_{cut} = diameter of the impeller after cutting (m)
D_{org} = original diameter of the impeller (m)
H_{cut} = head after cutting (m)
H_{org} = original head (m)
P_{cut} = power consumption after cutting (W)
P_{org} = original power consumption (W)

These equations are only applicable to centrifugal pumps with $n_{sq} \leq 40\ m^{0.75}/min\ s^{0.50}$ and a cut of not more than 20%. The efficiency curve decreases somewhat because impeller and pump house are then no longer optimally matched.

23.6 Determining the Dimensions of the Pumping Station

23.6.1 General Design Rules

For a pumping station (pump in a system) to operate as planned, or as calculated by the pump manufacturer, the following general design rules should always be used:

- The NPSH available in the pumping station should either be better than the NPSH required by the manufacturer, or be better than the tentative value derived from Figure 23.18. This will prevent cavitation effects as described in Section 23.4;
- Flow velocity towards the pump suction opening should either remain constant or accelerate, but should never decelerate. This will ensure an even velocity distribution in the impeller eye of the pump;
- Hydraulic losses at the suction side of the pump (ΔH_s in Figure 23.19) should be as small as possible. This usually means using short suction pipes (if needed) without sharp corners or bends;
- Suction pipes with bends that traverse different planes should be avoided at all costs. They would introduce strong rotational flow in the suction opening of the pump. This rule is especially important for high specific-speed pumps;

- The suction opening should be sufficiently below the suction level to avoid air entrainment vortices (see Section 23.6.2);
- Pumps should operate at or near their best efficiency point.

Rotations and vortices disturb the uniformity of the flow pattern in the suction opening of the pump. Flow rotations moving in the same direction as the impeller rotation lead to a lower discharge and head than would be expected from the pump characteristics. Flow rotations moving against the impeller rotation lead to a higher discharge and head, and may also lead to overloading of the driver. Air entrainment can cause a low head and a low discharge, rough running, or even complete depriming of the pump. Moreover, air may concentrate in the pressure pipe, resulting in higher energy losses and a lower discharge.

23.6.2 Sump Dimensions

When one impeller pump is used, a simple rectangular sump as shown in Figure 23.21 suffices. The sump dimensions are given in terms of the bell-mouth diameter, D_b.

Most pump manufacturers use the following ratio of bell-mouth diameter to pump suction opening D_l

$$\frac{D_b}{D_l} = 1.5 \text{ to } 1.8 \tag{23.30}$$

The dimensions shown in Figure 23.21 can be used, provided that there is a steady, uniform flow through the approach channel section about $3D_b$ upstream of the bell-mouth centre line (Figure 23.21A).

Assuming a mean velocity of 4.0 m/s in the pump suction opening, bell-mouth entry velocity can reach 1.2 to 1.8 m/s and approach channel velocities about 0.25 to 0.35 m/s, depending on the ratio suction opening diameter over bell-mouth entry diameter of 1.5 to 1.8.

When several pumps are used in one wide approach channel, the design principle shown in Figure 23.22 is a good one. But any deviation from these basic design rules, for whatever reason, requires that sump model tests be performed (Wijdieks 1985). Such deviations often occur when water is delivered to the sump through a relatively narrow suction channel. This results in decelerating flow and related vortices and eddies in the sump. Special provisions are then needed to correct the flow pattern. Model testing should be entrusted only to skilled laboratories since scale effects can influence the results.

23.6.3 Parallel Pumping

The dimensions of a pumping installation depend greatly on the dimensions of the pump, particularly that of its suction opening. The first step in designing a pumping installation should therefore always be to estimate the number of pumps to be installed

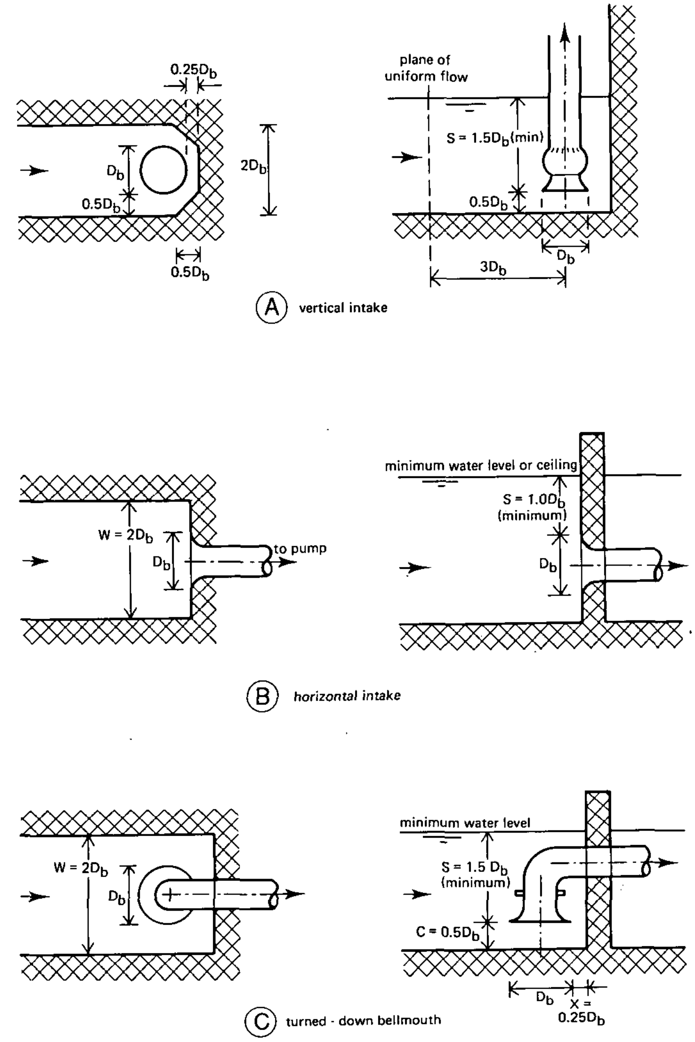

Figure 23.21 Single-pump sumps (after Prosser 1977)

and to characterize their dimensions by their suction opening, D_1.

Economically, it may seem wise to install a single pump unit instead of two, three, or more units which, together, have the same pumping capacity. However, the breakdown of a single unit leads to the complete cessation of pumping. To avoid this risk, and also to avoid inefficient pumping when the discharge is low, it is a good idea to spread the capacity of major pumping stations over several units. And, by choosing units of the same size, it is possible to keep the number of spare parts in stock to a minimum. When efficiency is the main consideration, as it so often is nowadays, several pumps with impellers that can run at variable speeds often are the best choice.

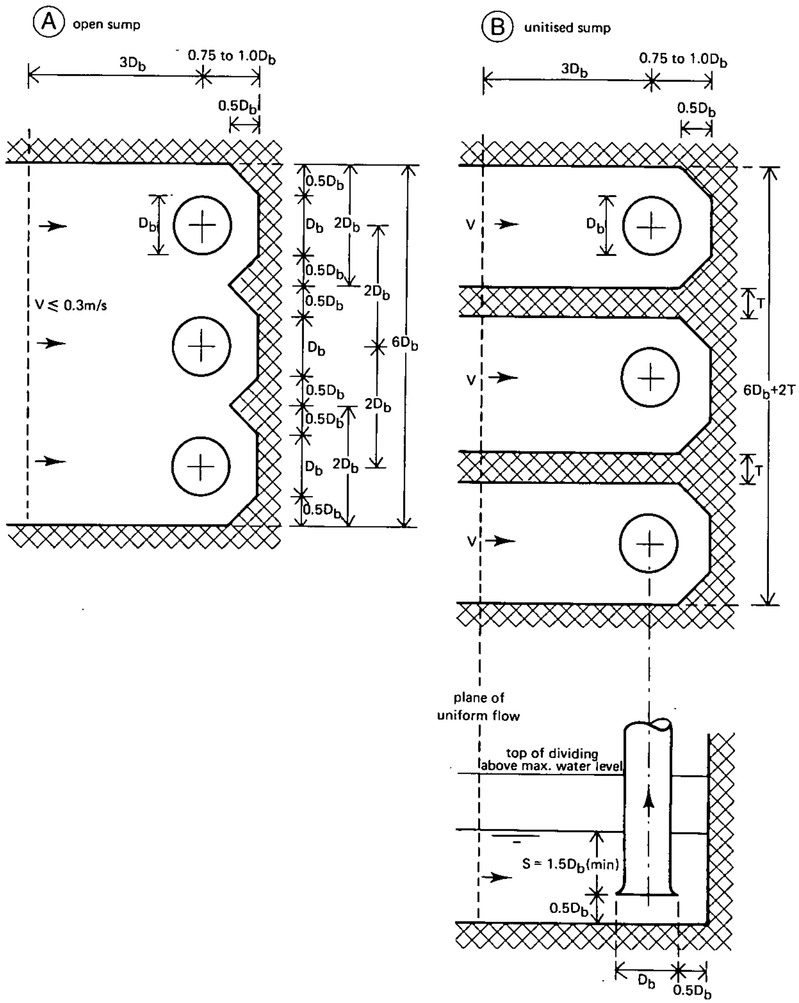

Figure 23.22 Basic sump dimensions for multiple pumps (after Prosser 1977)

Another advantage of spreading the pumping capacity over several units is that this will prevent large drawdowns in the main supply drain through which water flows to the pumping station. As it takes some time before a hydraulic energy gradient develops in the drain, any temporary shortage of water may cause such a drawdown that the stability of the earthen embankment is endangered. Shutting down a pumping station that is operating at full capacity will also constitute a serious attack on the canal embankments: the kinetic energy of the water flowing in the main drainage canal will generate a transition wave.

The total capacity of a pumping station can be made up of several units, each of them pumping through separate suction and pressure pipes, and each of them operating independently of the other. An alternative method is to have several pumps

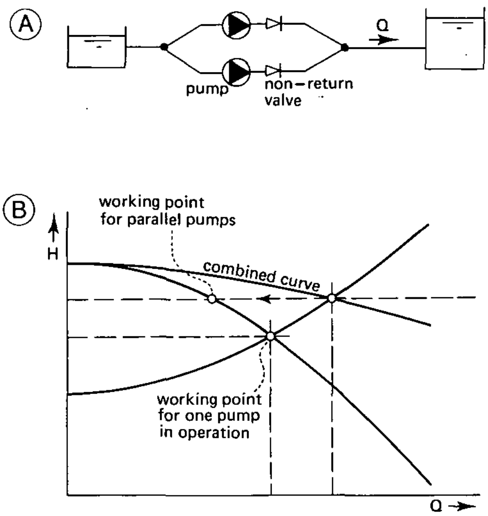

Figure 23.23 Parallel operating pumps; A: Schematic presentation; B: Determination of the working point(s) of parallel installed pumps

discharging simultaneously into a common pipe system; this is referred to as a parallel operation (see Figure 23.23).

The Q-H characteristic of the two parallel operating pumps of Figure 23.23 can be obtained by adding the two Q values for each point of the individual characteristics, while the head value remains constant.

To prevent water backflow through the pump units, each of them should have an automatic, non-return valve in its pressure conduit.

23.6.4 Pump Selection and Sump Design

Selecting the proper pump for a pumping station and designing a system around one or more pumps can be divided into a number of steps. Depending on the experience of the designer and the documentation available to him, these steps may have to be repeated one or more times to balance each part of the projected pumping station. The steps are:

1 Decide the number of pump units required to attain the total discharge capacity, basing this decision on the duration curve of the discharge to be pumped and on the considerations presented in Section 23.6.3;
2 Calculate the discharge, Q_o, of each unit at its best efficiency point;
3 Next, considering the practical restrictions imposed by the lowest suction level, pressure level, trash rack (Section 23.6.6), valves, bends, and so forth, make a first estimate of the total head, H_o, to be pumped at the best efficiency point. Use the flow velocities given in Section 23.6.2 to make the necessary calculations;
4 With both Q_o and H_o known for each pump unit, draw Line 1 in Figure 23.24.
This step represents a period of reflection. The combination of discharge and head can be delivered either by a big slow-running pump (Line a of Figure 23.24) or

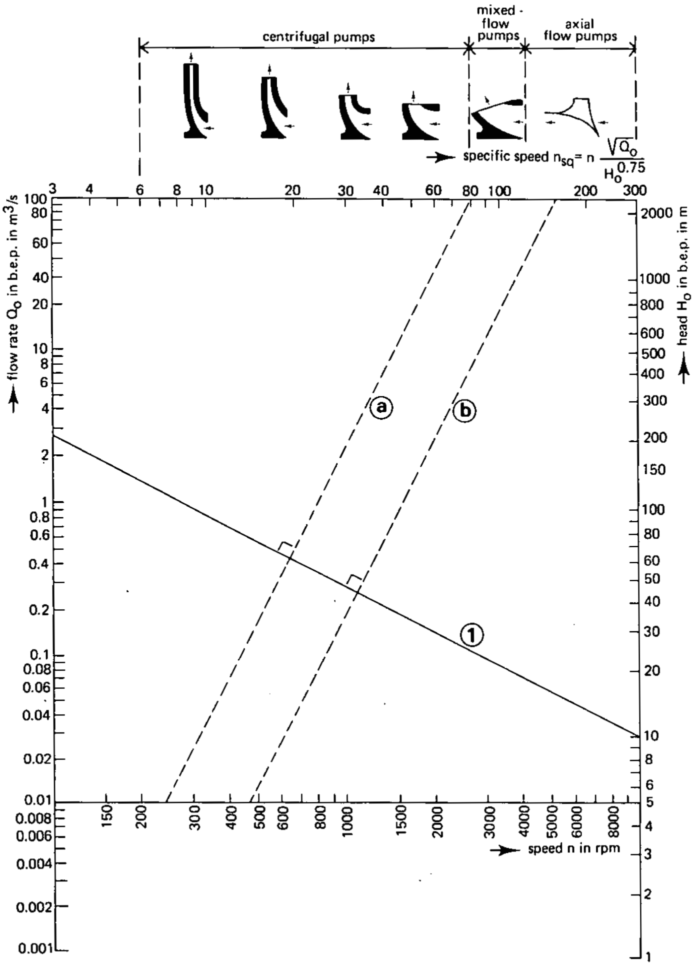

Figure 23.24 Relation diagram of Q_o, H_o and pump speeds (after De Gruyter 1971)

by a smaller faster-running pump (Line b of that Figure), while the position of these lines depends on the speed of the intended impeller. Hence, different specific speeds, and related pump types, are possible. Furthermore, the investment cost for a pumping station with small fast-running pumps is usually less than that for a comparable station with large slow-running pumps. On the other hand, the different pumps have different attainable maximum efficiencies, and also different NPSH and σ values. As follows from Figure 23.11, the pump on Line a has about 5% higher efficiency than the pump on Line b, and thus consumes 5% less energy. But, according to Figure 23.18, the pump on Line a has a σ value of 0.4 whereas the pump on Line b has a σ value of 0.8. This means that the foundation for the smaller pumps requires a deeper excavation than that for the larger pump. So, before Line 1 in Figure 23.24 can be drawn, the designer has to find a balance between the investment cost and the operating cost of the pumping station;

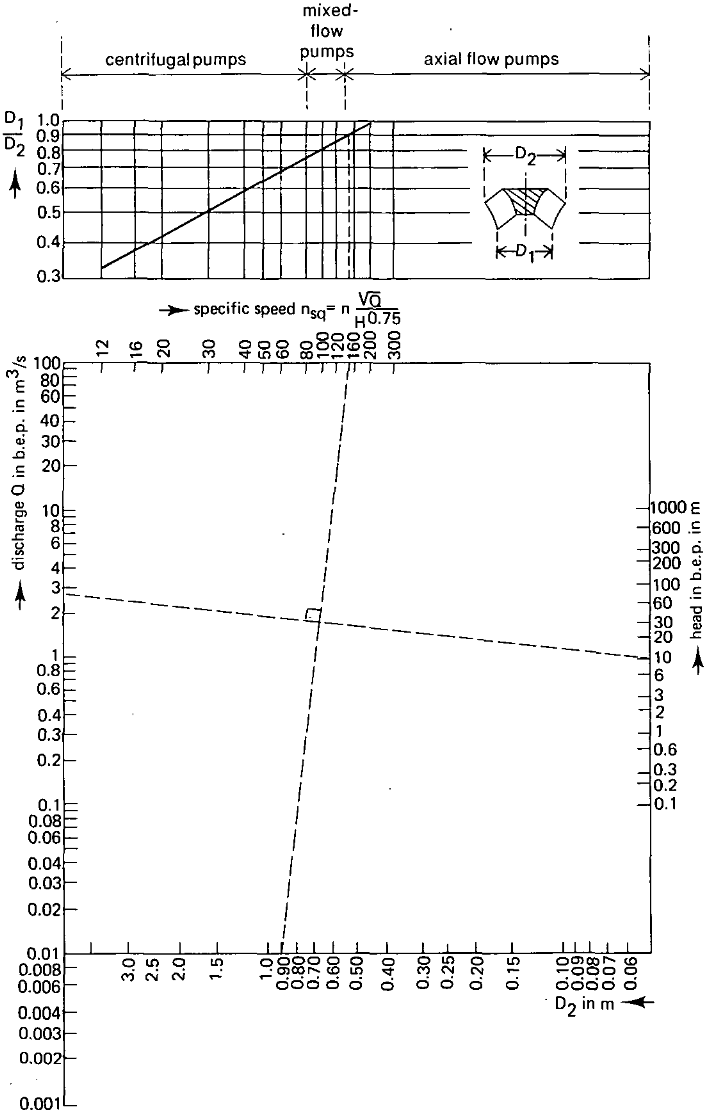

Figure 23.25 Nomograph to determine the impeller sizes, D_1 and D_2 (after De Gruyter 1971).

5 As was explained in Section 23.4, cavitation will influence a pump's allowable specific speed, n_{sq}, and cannot be disregarded. So, using Figure 23.18, calculate the tentative required NPSH and H_A values. On the basis of the practical and economic possibilities governing the minimum suction level, the cost of excavation, and the cost of the foundation, decide whether to choose a lower pump elevation or a lower specific speed;

6 Repeat steps 1 to 5 until one suitable n_{sq} value is found. Note that low n_{sq} values correspond to centrifugal pumps with low ratio D_1/D_2 and high n_{sq} values correspond with axial flow pumps with $D_1/D_2 = 1$;

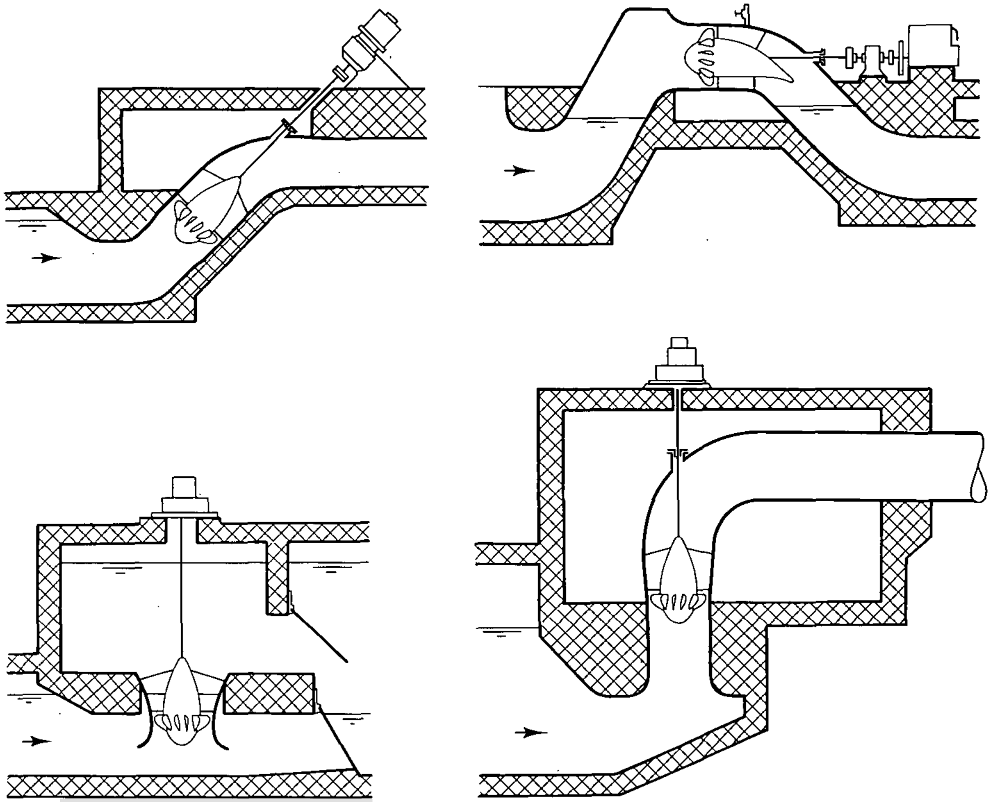

Figure 23.26 Sections of pumping stations with an axial-flow pump

7 From the lower part of the nomograph in Figure 23.25 first estimate the outlet diameter D_2, then from the upper part of it estimate the ratio D_1/D_2, which finally leads to an estimate of the suction opening D_1 ($\approx$ inlet diameter of impeller) and the bell-mouth opening D_b ($\approx$ 1.5 to 1.8 times D_1);
8 From Section 23.6.2 (Figures 23.21 and 23.22), determine the proper sump dimensions. This may mean that the tentative design of Step 3 has to be changed slightly. If so, repeat Steps 4 to 8.

The steps above should be followed until a satisfactory design is obtained. The shape of the resulting pumping station can vary greatly, as can be seen from Figures 23.26, 23.27, and 23.28.

23.6.5 Power to Drive a Pump

As was explained in Section 23.2.2 (Equation 23.8), the power to be delivered to the pump shaft is

$$P_s = \frac{\rho g Q H}{\eta} \tag{23.31}$$

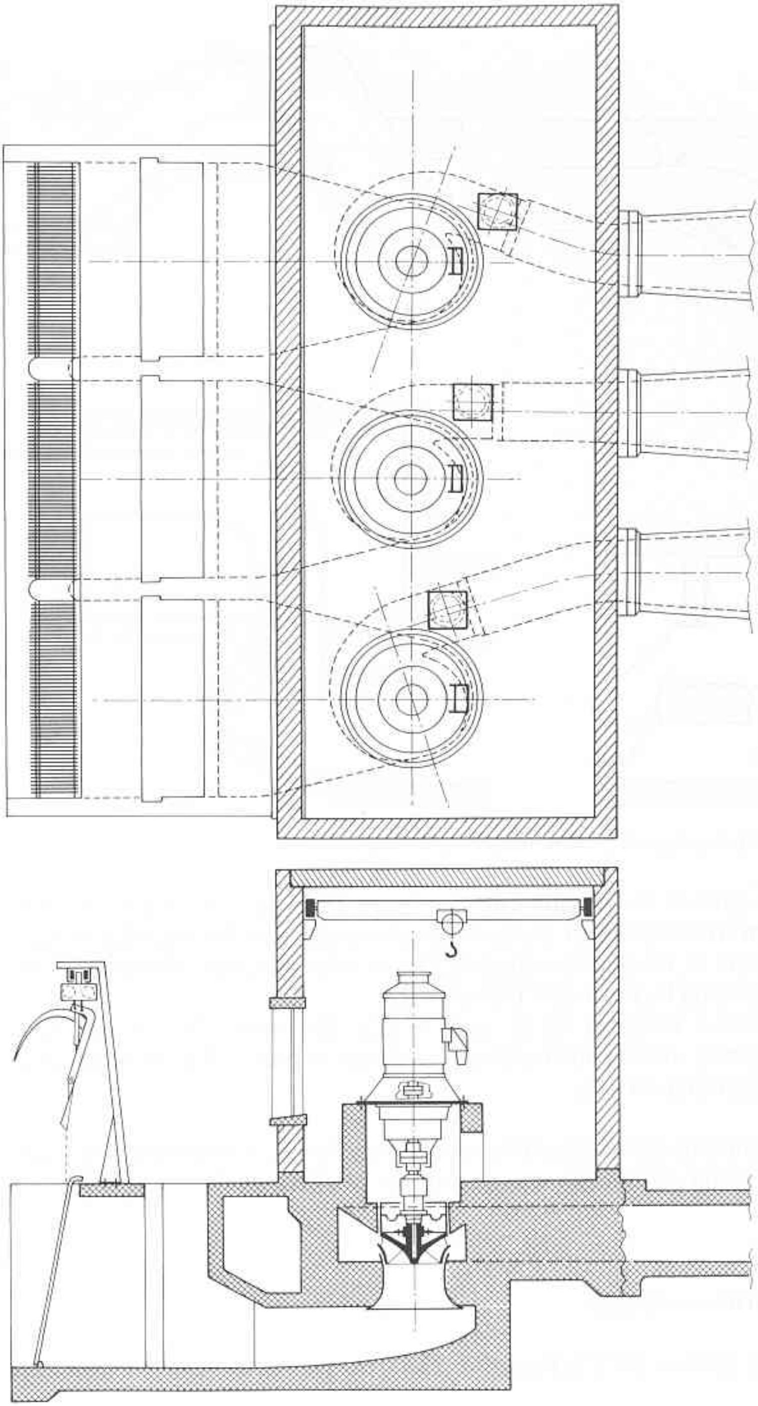

Figure 23.27 Example of assembly and drive for a pumping station with three mixed-flow pumps with concrete housing (Courtesy Stork)

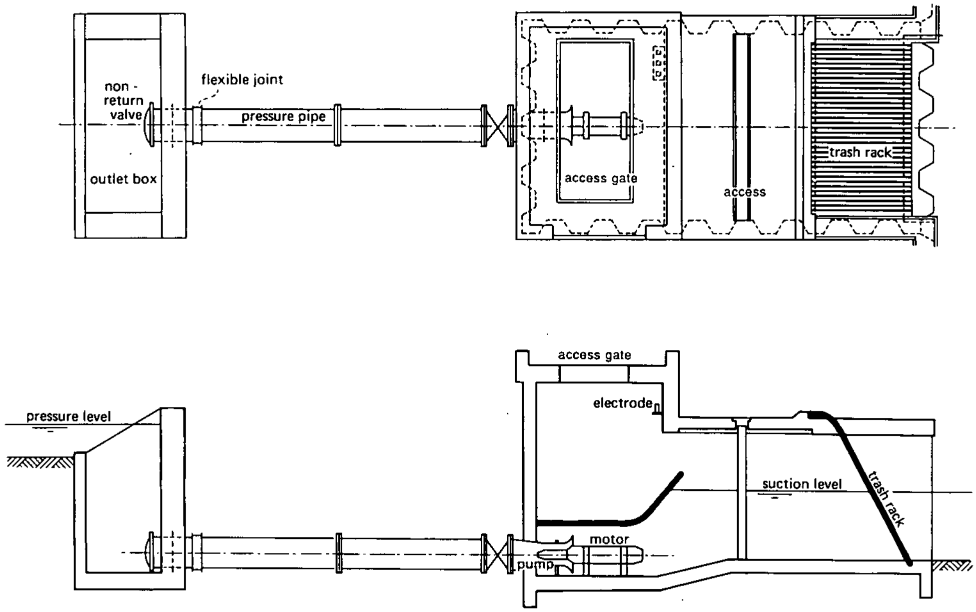

Figure 23.28 Section of a pumping station with underwater axial-flow pump

Any device that can produce a driving torque on the pump shaft can be used as a driver. The most common drivers are the electric motor and the internal combustion (diesel) engine. The choice of driver should be made on the basis of technical and economic considerations as well as on reliability. Some of these considerations are

- Is electric energy available? The construction of power lines can be costly.
- Is the energy supply reliable? Is it prone to failure? Is the transport of diesel fuel costly or uncertain? Are there any import restrictions on fuel?

Table 23.2 Percentage power reduction for an internal combustion engine

Altitude (m)	Temperature (°C)						
	20	25	30	35	40	45	50
100	1	2	4	6	8	10	11
500	6	8	9	11	13	15	17
1000	12	14	16	18	20	21	23
1500	19	21	23	24	26	28	30
2000	26	27	29	31	33	35	36
2500	32	34	36	37	39	41	43
3000	39	40	42	44	46	48	49
3500	45	47	49	51	52	54	56
4000	52	54	55	57	59	61	63
4500	58	60	62	64	66	67	69
5000	65	67	68	70	72	74	76

- What is the cost of energy? Is it possible to use off-peak electric power? Must energy be paid in local or foreign currency?
- Are electric motors or diesel engines produced locally? Are maintenance and repair services available?

The driver's power requirement should be calculated on the basis of the least favourable conditions. The actual efficiency of the pump will then usually be less than the maximum attainable values given in Figure 23.11. One should also realize that power losses in the driver can be caused by the use of a gear box (transmission) and by climatic factors. The power of internal combustion engines will decline as altitude and temperatures increase and as humidity decreases. The extent of the power decline must be specified by the manufacturer. To illustrate how significant a power reduction can be, Table 23.2 shows the effect of temperature and altitude on an internal combustion engine.

The driver should not operate continuously at its maximum capacity, but at an 85 to 90% load. The power of the driver can be calculated from

$$P_d = \frac{\rho g Q H}{\eta\, \eta_d\, \eta_t} \times \frac{100 + ac}{100} \times f_r \tag{23.23}$$

where, in addition to earlier defined terms

P_d = required power supply (W)
η = expected efficiency of the pump (–)
η_d = efficiency of the driver (electric motors) (–)
η_t = efficiency of the transmission (0.96 to 0.98) (–)
ac = percentage of power reduction due to altitude and climate (–)
f_r = factor to prevent the driver from running continuously at maximum capacity (internal combustion engines) (1.1 to 1.2)

23.6.6 Trash Rack

To prevent damage to the blades of an Archimedean screw or the impeller of a pump, as well as to avoid blockage of a pump's suction opening, a pumping station should be equipped with a trash rack. The spacing of the trash rack's bars varies according to the type and size of the water-lifting device. Pump and trash rack manufacturers can advise on the proper spacing.

Head losses over a clean trash rack are a function of the flow velocity and the shape and spacing of the bars. Head losses over various types of bars are given in Figure 23.29.

It is obvious that these head losses increase rapidly as trash collects against the rack. To avoid excessive head losses or clogging, the rack must be cleaned with a hand rake. Larger pumping stations are usually equipped with an installation like the one in Figure 23.30. Such cleaning equipment can be operated either manually or automatically. Cleaning should begin when the head loss over the trash rack exceeds a predetermined value.

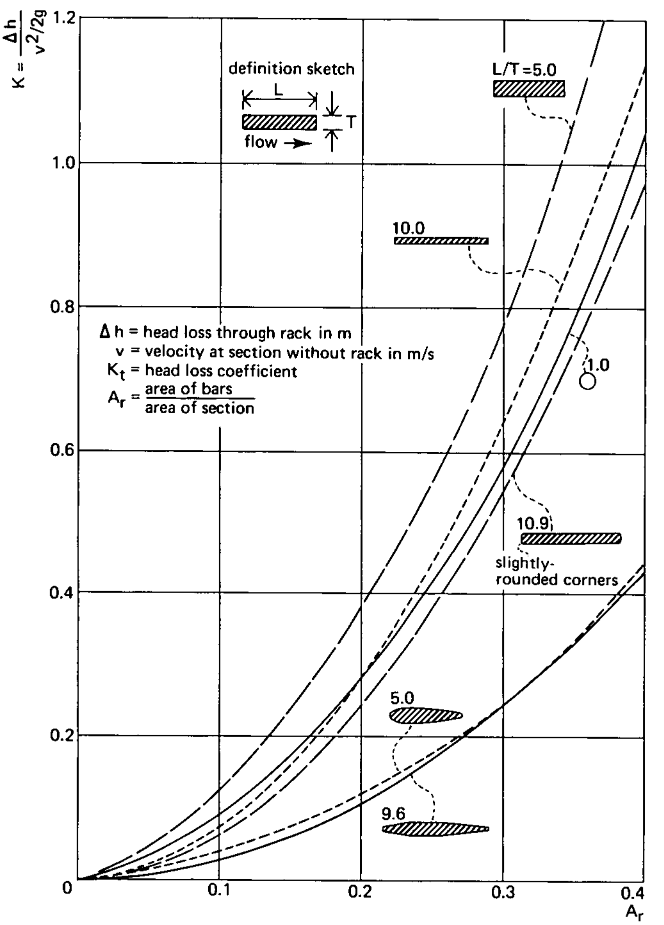

Figure 23.29 Trash rack losses for various types of bars (from Fellenius 1929 and Kirschmer 1926)

23.6.7 The Location of a Pumping Station

When the site for a pumping station is being selected, the following factors should be kept in mind:

- Drainage pumping stations almost always have to be located at the lowest point in the area. Soil conditions at such a site are usually poor. A foundation resting on different levels is not recommended because the bearing capacities of the soil may differ from one level to another;
- Groundwater levels will change after the canals and the pumping station become operational. It may be necessary to take measures to prevent excessive groundwater flow under the station;
- Pumping stations must be easily accessible. It must be possible to transport fuel by road or water, or to provide an easy link-up with the electric network;

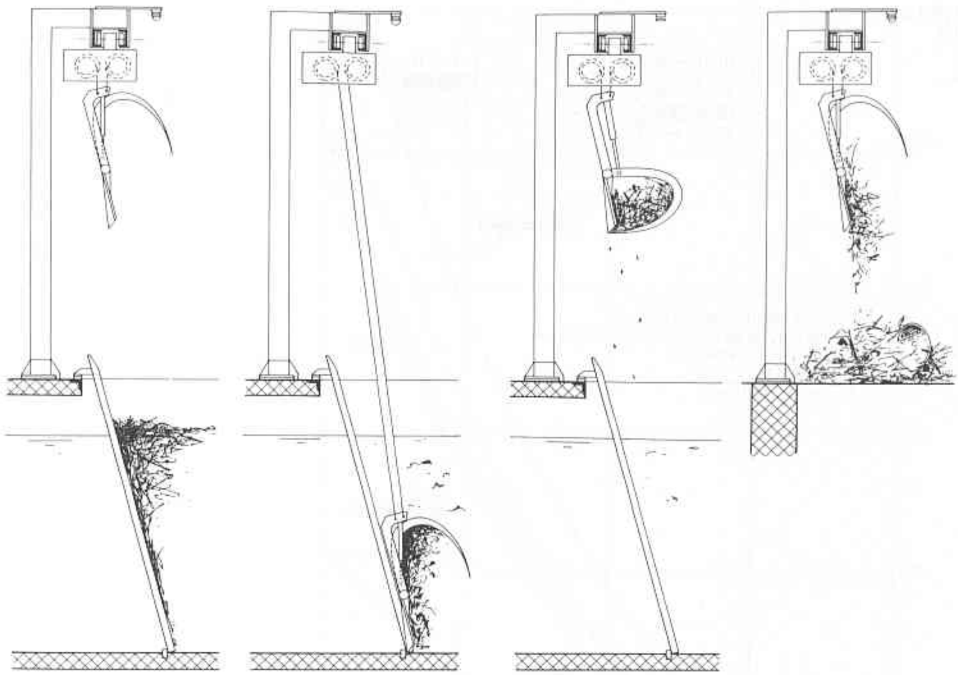

Figure 23.30 Cleaning the trash rack (courtesy Landustrie, Sneek)

- Pumping stations should never be placed on or close to dikes that contain layers of high permeability (e.g. sand); nor should they be built on old dikes;
- New dikes and newly drained lands are subject to varying degrees of subsidence, which are difficult to predict with accuracy. Pipe lines and concrete structures on or through new dikes should therefore be flexible;
- Trash and debris must be easily removable from the screens; a site must be available to deposit trash awaiting disposal.

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