11

PUMPING INSTALLATIONS

11.1 Introduction

Pumping installations are intrinsic elements of water supply and wastewater disposal systems. Essentially, they transfer energy to the through-flow of water by effecting a step-increase in head or pressure. In water supply applications, pumps are typically running continuously and, in most cases, the required step increase in head is substantial. In wastewater applications, pumps are typically running in a start/stop mode and the required wastewater lift is usually not large, the pump installation being typically used to augment gravity flow systems.

11.2 Pump types

The following pump types, differentiated by their mode of pumping action, are in general use in the water industry.

(1) Positive displacement pumps e.g. helical rotor, diaphragm and piston pumps. The use of positive displacement pumps is confined to specialist applications e.g. the dosing of chemical solutions where a high level of flow control is required; the pumping of viscous fluids such as sewage slugs; high-pressure applications.

(2) rotodynamic pumps, which may be classified as centrifugal, mixed flow and axial flow. Rotodynamic pumps are by far the most widely used pump type for bulk-pumping of clean water and wastewaters.

(3) air-lift pumps: airlift pumps may be of the submerged air injection type, in which compressed air is injected into a vertical riser pipe, thus airlifting liquid in a two-phase flow, or the air ejector type, where compressed air is used to displace liquid from a closed vessel. The latter operates in a batch mode.

(4) the Archimedean screw pump.
The Archimedean screw pump is an inclined screw conveyer, which is used for low-lift applications, particularly in the wastewater treatment field, where it is used for raw sewage and activated sludge pumping.

11.2.1 Positive displacement pumps

Positive displacement pumps can be divided into reciprocating and rotary types. Reciprocating pumps include piston, plunger (ram) and diaphragm types. They have the common characteristic of a discontinuous pulsed delivery, the magnitude of which is effectively independent of delivery head, hence the description positive displacement. The peak pressure generated during the delivery stroke can be reduced by the installation of an air vessel on the pump delivery. This will also have a smoothing effect on the pump discharge. The sequential filling and emptying of the pump body requires non-return valves on both the suction and delivery sides of the pump.

Rotary pumps may be of the gear, lobe, helical rotor and sliding vane types. They combine the continuous discharge characteristic of rotodynamic pumps and the positive displacement characteristic of reciprocating pumps.

Positive displacement pumps are particularly suited to chemical dosing applications in water and wastewater treatment plants, where their accurate and controllable discharge characteristics can be used to advantage. Helical rotor or progressive cavity pumps and diaphragm pumps are widely used to pump viscous liquids including sewage sludges.

11.2.2 Rotodynamic pumps

The active element of a rotodynamic pump is the rotating impeller or propeller, which imparts a momentum to the fluid, that, on deceleration, is converted to a pressure rise. If geometrically similar impellers of differing size are driven at appropriate speeds, such that the exiting fluid has the same flow direction in all cases, the impellers are described as a homologous set. Using the method of dimensional analysis, it has been shown in Chapter 9 that such a homologous set can be categorised by the compound non-dimensional parameter, known as the specific speed N_s, which is a function of the pump speed N, the pump head H and the pump discharge Q:

$$N_s = \frac{NQ^{0.5}}{(gH)^{0.75}} \qquad (9.12)$$

In pump technology literature, the gravity constant is often omitted, resulting in the following form of the specific speed function:

$$N_s = \frac{NQ^{0.5}}{H^{0.75}} \qquad (9.13)$$

As N_s in this form are not dimensionless, its numerical value is dependent on the system of units used for N, H and Q.

On the basis of their specific speed value, rotodynamic pumps can be broadly classified in the following types:

Pump type	Specific speed
Centrifugal	≤ 80
Mixed flow	80-150
Axial flow	150-300

based on parameter units: N (rpm), Q (m^3s^{-1}), H (m)

The typical geometric form of the impeller component of each of these rotodynamic pump types is illustrated in Fig 11.1.

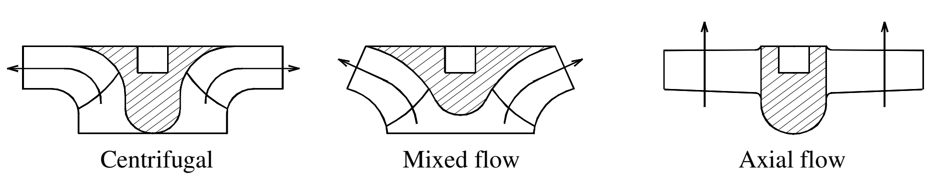

Fig 11.1 Rotodynamic pump impeller types

The manometric head H is defined as the step change in total head across the pump:

$$H = \left(\frac{p_d}{\rho g} + \frac{v^2}{2g}\right) - \left(\frac{p_s}{\rho g} + \frac{v^2}{2g}\right) \qquad (11.1)$$

where the subscript s and d relate to the pump suction and delivery sides, respectively. Pump efficiency η is defined as the ratio of the hydraulic power transferred to the fluid, to the shaft power P:

$$\eta = \frac{\rho g H Q}{P} \qquad (11.2)$$

Efficiencies up to 90% can be achieved in large centrifugal pump units pumping clean water. The performance characteristics of individual pumps are generally presented graphically as plots of manometric head (H), power (P) and efficiency (η), as functions of discharge (Q), as illustrated on Fig 11.2.

It will be noted that head and power increase sharply as the discharge approaches zero in axial flow pumps. For this reason, such pumps should never be started up against a closed valve in the delivery line. A centrifugal pump, on the otherhand, has its minimum power demand at zero discharge rate.

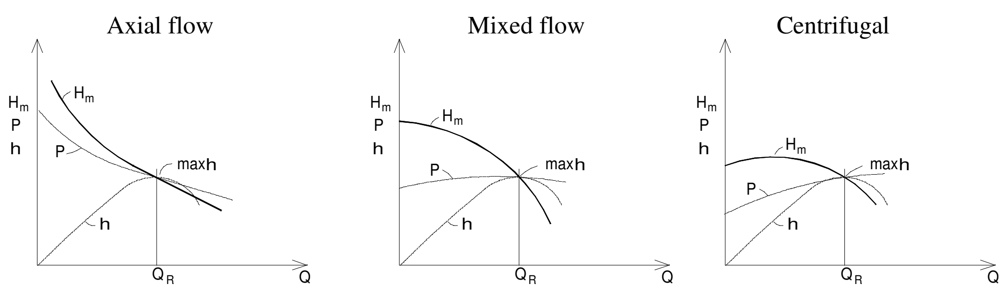

Fig 11.2 **Rotodynamic pump characteristics; H_m is the manometric head; P is the power, and η is the efficiency**

The respective ranges of practical application for the three types of rotodynamic pump, in terms of head/discharge capacity, are presented on Fig 11.3.

As shown in Chapter 5, the H/Q relationship for rotodynamic pumps, driven at their rated speed NR, may be expressed in quadratic equation form as follows:

$$H_R = A_0 + A_1 Q_R + A_2 Q_R^{\,2} \qquad (5.9)$$

where A_0 is the manometric head at zero flow, A_1 and A_2 are constants. As shown in Chapter 6, the H/Q equation for any other speed N, as derived from the rated speed, has the following form:

$$H_N = A_0\left(\frac{N}{N_R}\right)^2 + A_1\left(\frac{N}{N_R}\right) Q_N + A_2 Q_N^{\,2} \qquad (6.30)$$

This modification of equation (5.9) is based on the fact that for a given impeller, $Q \propto N$ and $H \propto N^2$.

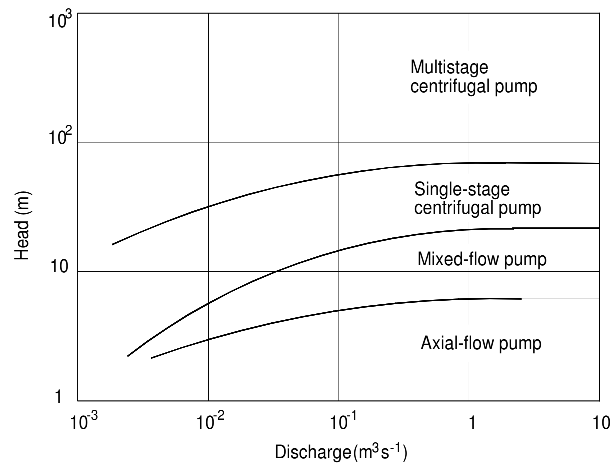

Fig 11.3 The range of application of pump types

11.2.3 The air-lift pump

The mode of operation of the airlift pump is illustrated on Fig 11.4. Air is injected into the vertical pipe at a depth h below the free water surface. The resulting rising stream of air bubbles within the riser pipe produces a two-phase fluid with a lower composite density than the liquid on its own and thus enables the riser to discharge at a height h above the free water surface.

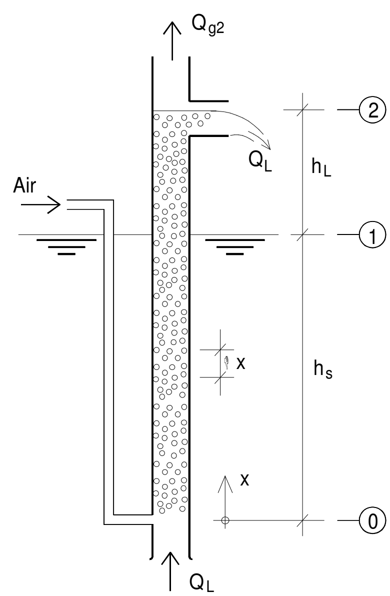

Fig 11.4 Air-lift pump schematic

It is clear that the upward driving force at the bottom of the two-phase column is the static liquid pressure at that point and hence the average driving force per unit column length is the ratio $h_s/(h_s + h_L)$, which is defined as the submergence ratio Sr. The fluid efficiency η_a of an air-lift pump is expressed as the ratio of the work done in lifting the liquid to the work done by the expanding air stream:

$$\eta_a = \frac{\rho_L g h_L Q_L}{P_2 Q_{g2} \ln(P_0 / P_2)} \tag{11.3}$$

where ρ_L is the liquid density, P_0 and P_2 are the absolute pressures at levels 0 and 2, respectively, Q_L and Q_{g2} are the volumetric flow rates for liquid and gas, respectively (Q_{g2} is the volumetric gas flow rate at the discharge pressure P_2, which would normally be atmospheric pressure, as in Fig 11.4).

Computation of air-lift pump discharge

The two-phase flow in an airlift riser pipe can be analysed by application of the momentum principle (Nicklin, 1963; Clark and Daybolt, 1986). Consider a control volume of length Δx as shown in Fig 11.4. Since the flow is steady, the sum of the acting forces - pressure, gravity and friction - must be zero. Neglecting the weight of the air phase, the momentum equation is written as

$$\Delta P\, A + \rho_L g A(1-\varepsilon)\Delta x + F_w \Delta x = 0 \tag{11.4}$$

where ΔP is the pressure change over the length Δx, A is the riser cross-sectional area, ε is the void (air) fraction and F_w is the frictional drag force per unit length.

The void fraction is influenced by the relative motion or 'slip' of the air past the liquid in the riser. This is minimised if the injected air remains as dispersed bubbles. In airlift pipes the air bubbles typically coalesce to form air 'slugs' or elongated round-nosed bubbles. Nicklin has shown that the average rise velocity of air slugs within a riser can be represented as

$$\frac{Q_g}{\varepsilon A} = \frac{1.2\left(Q_g + Q_L\right)}{A} + 0.35\sqrt{gD} \tag{11.5}$$

where D is the pipe diameter ($0.35\sqrt{gD}$ is the theoretical rise velocity under still water conditions). The variation in gas volume with reduction in pressure over the column height is given by the isothermal relation

$$Q_g = Q_{g2}\frac{P_2}{P} \tag{11.6}$$

Clarke and Daybolt have shown that the friction loss in the two-phase airlift flow (F_w) can be approximately related to the friction loss $F_{w(L)}$ of the liquid phase, flowing on its own, as follows:

$$F_w = F_{w(L)}\left(1 + 15\varepsilon\right) \tag{11.7}$$

The liquid flow frictional resistance per unit length can be expressed in accordance with the Darcy-Weisbach equation, as

$$F_{w(L)} = \frac{\rho_L f Q_L{}^2}{2AD} \tag{11.8}$$

Hence

$$F_w = \frac{\rho_L f Q_L{}^2}{2AD}\left(1 + 1.5\varepsilon\right) \tag{11.9}$$

Inserting the foregoing expression for F into the momentum equation (11.4) and combining with (11.5) and (11.6) leads to the following expression in P and x:

$$-\Delta P = \left[\rho_L g \left\{ 1 - \frac{Q_{g2} P_2}{1.2\left(Q_{g2} P_2 + Q_L P\right)} \right\} + K_f \left\{ 1 + \frac{1.5 Q_{g2} P_2}{1.2\left(Q_{g2} P_2 + Q_L P\right) + Q_D P} \right\} \right] \Delta x \quad (11.10)$$

where

$$K_f = \frac{\rho_L f Q_L^{\,2}}{2A^2 D} \qquad \text{and} \qquad Q_D = 0.35 A \sqrt{gD}$$

Integrating equation (11.10) between the limits P = P_2 and P = P_0 and x = (h_s + h_L and x = 0 yields the following:

$$\frac{P_2 - P_0}{\rho_L g + K_f} - \left(\frac{(1.5K_f - \rho_L g) Q_{g2} P_2}{(\rho_L r + K_f)^2 S} \right) \ln \left(\frac{R + (\rho_L g + K_f) S P_2}{R + (\rho_L g + K_f) S P_0} \right) + (h_s + h_L) = 0 \qquad (11.11)$$

where $R = Q_{g2} P_2 (0.2 \rho_L g + 2.7 K_f)$ and $S = 1.2\ Q_L + Q_D$.

In typical water engineering applications the pump outlet discharges to atmosphere and hence the outlet pressure Pg2 is atmospheric pressure. In practice, the injection pressure P_0 will be less than the local external hydrostatic value by an amount equal to the total head loss in the pipe between the pipe inlet and the point of air injection into the pipe. In general, the major energy loss in airlift pumps is due to slippage between the air and water flows and hence the friction loss component tends not to be significant, except in circumstances where the lift is small and the suction pipe is long.

The efficiency of airlift pumps, as earlier defined, varies with the submergence ratio S_r, the liquid and gas flow rates and the riser pipe diameter. Nicklin has shown that the best achievable efficiency, based on the foregoing theory, is in the approximate range of 45% to 55%, depending on pipe diameter, the magnitude increasing marginally with increase in diameter. The following may be used as an approximate guide to the most efficient operating regions for liquid and gas flow rates:

Liquid: $$\frac{Q_L}{A\sqrt{gD}} = 0.5 - 1.5$$

Gas: $$\frac{Q_{g2}}{A\sqrt{gD}} = 0.1 - 1.0$$

The foregoing theory is based on the assumption that the two-phase flow remains within the slug flow mode. For practical design purposes, this can be assumed to be the case provided that the superficial air velocity does not exceed twice the superficial liquid velocity.

The air-flow rate required to generate a specific superficial water velocity in the airlift riser is inversely related to the submergence ratio. The ratio of superficial air velocity (Q_{g2}/A) to superficial water velocity (Q_L/A) is plotted in Fig 11.5 as a function of the superficial water velocity. The computation (based on equation (11.11)) relates to a 200mm diameter airlift riser operating at submergence ratios of 0.7, 0.8 and 0.9. The computation assumed free discharge to atmosphere, that is, $P_{g2} = P_a$, where P_a is atmospheric pressure. In computing the air injection point pressure P_0, account was taken of the entry head loss, the velocity head and the friction loss in the suction pipe upstream of the injection point.

The **air-ejector** type of airlift pump is used in association with a closed vessel sump, which fills, by gravity inflow. Its contents are displaced into a rising main by the admission of compressed air. This mode of operation requires non-return valves on the inflow and discharge lines. Air ejectors are suitable for low-volume wastewater pumping duties.

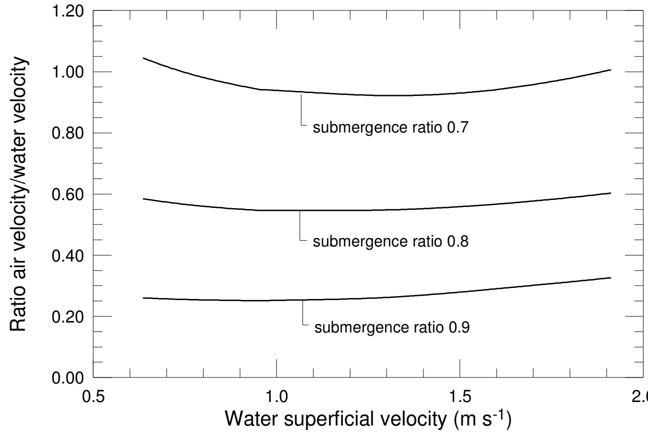

Fig 11.5 Correlation of air and water superficial velocities in a 200mm diameter airlift riser at differing submergence ratios

11.2.4 Archimedean screw pump

The Archimedean screw pump is illustrated in Fig 11.6. It consists of an inclined screw conveyer shaft rotating within a lose-fitting open-top cylindrical conduit. In rotation, the helical screw blades force the liquid up along the inclined conduit, achieving a liquid lift equal to the vertical projection of the inclined rotor shaft. The discharge capacity is a function of the rotor diameter and its rotational speed. The operational blade tip speed is usually in the range of 2.0 to 3.5 ms^{-1}, with a corresponding diameter (D) range of 0.4 m to 3.0 m and a pumping capacity range of 0.02 to 3.2 m^3s^{-1}. Archimedean screw pumps retain their pumping efficiency over a wide flow range. A best overall efficiency of about 75% is claimed for pumping at rated capacity, reducing to about 65% at 30% of rated capacity.

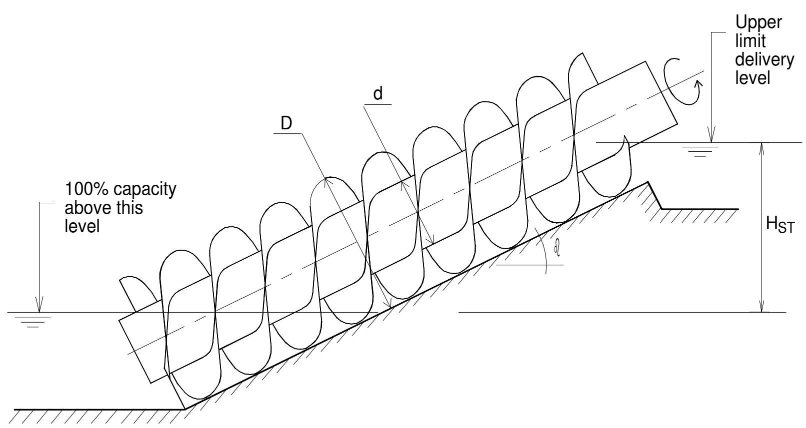

Fig 11.6 Schematic outline of the Archimedean screw pump (courtesy of Spaans Babcock, Hoofdorp, The Netherlands). β is in the range 22-28 degrees.

11.3 Hydraulics of rotodynamic pump/rising main systems

Fig 11.7 illustrates a typical pump/rising main system. As already defined by equation (11.1), the pump manumetric head is the total differential head across the pump. In steady state pump operation the manometric head H is equal to the pipe system head:

$$H = H_{ST} + h_L \tag{11.12}$$

where H_{ST} is the static head and is the sum of the suction lift H_{SU} and the delivery lift H_{DY}; h_L is the total head loss between suction and delivery reservoirs.

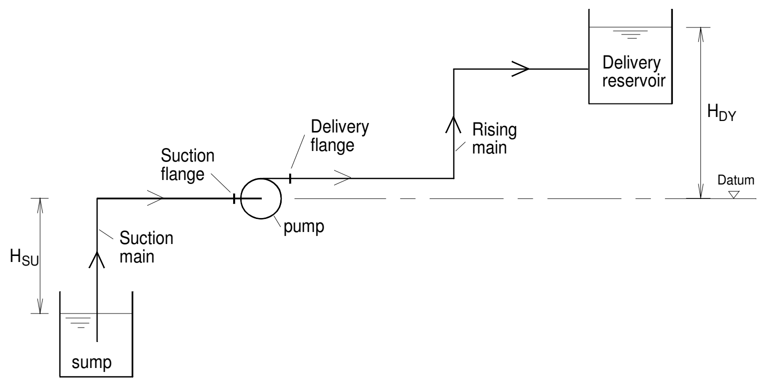

Fig 11.7 Pump and pipe system

The maximum lift for a rotodynamic pump is limited by the necessity to avoid cavitation. This phenomenon is caused by a drop in fluid pressure to vapour level and the consequent formation of vapour cavities - hence the term cavitation. Cavitation is manifested by noise and vibration as the vapour pockets implode on moving to regions of higher pressure. It reduces pump efficiency and may cause erosion of the pump impeller and housing. It is avoided by setting a lower limit to the allowable pressure at the suction flange. This limit is conventionally expressed as the 'net positive suction head' or NPSH for a pump. NPSH is defined as

$$\text{NPSH} = \left(\frac{P_s}{\rho g} - \frac{P_v}{\rho g} \right) \tag{11.13}$$

where P_s is the absolute pressure at the suction flange and P_v is the prevailing vapour pressure. The NPSH value can be related to the specific N_s through the Thoma cavitation number σ, which is defined as the ratio of NPSH to manometric head H:

$$\sigma = \frac{\text{NPSH}}{H} \tag{11.14}$$

Based on empirical evidence (Wijdieks, 1971), σ can be approximately correlated with the specific speed N, as follows:

(1) for single suction pumps: $\sigma = 0.001 N_s^{1.36}$

(2) for double suction pumps: $\sigma = 0.0006 N_s^{1.36}$

While such correlations may be used as a general guide for preliminary design, the individual pump NSPH specification should be used in final design computations.

Pumps may be operated in parallel or in series. The resulting H/Q characteristics are shown on Figs 11.8(a) and 11.8(b), respectively.

The duty point at which a pumped system operates is determined by solution of the pump equation (6.33) and the system equation (11.12) for H and Q. This solution is illustrated graphically on Fig 11.9. The duty point may also be computed using the ARTS hydraulic design software from Aquavarra Research Limited. This program determines (a) the pump equation coefficients, given three sets of H/Q values from the rated characteristic pump curve and (b) solves the system and pump equations to find the duty point value for the operating speed.

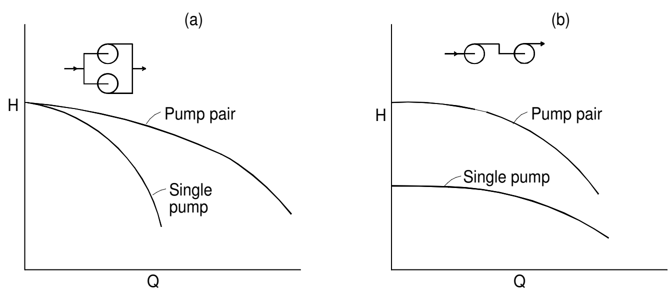

Fig 11.8 Characteristic H/Q curves for pump combinations
(a) in parallel (b) in series

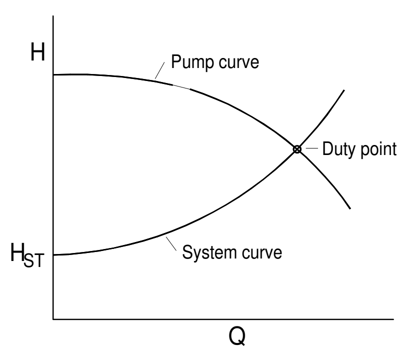

Fig 11.9 **Determination of pump duty point**

11.4 Economics of pump/rising main systems

The cost of water pumping includes the capital cost of the water transportation system including structures, pumps, power supply, pipelines, reservoirs etc. In selecting the type and size of rising mains, particularly where the friction head represents a significant fraction of the total head, the influence of pipe size on the unit cost of pumping merits examination. A larger pipe incurs less friction head loss and hence requires smaller pumps and lower power input; thus the problem is one of trading off increased capital costs against reduced running costs or vice versa.

Assuming the following unit costs (arbitrary currency units)

Pipeline: C_{pi}/m length/m diameter
Pumps: C_{pu}/installed kW
Energy: C_e/kWy (kWy = kW x 1 year)
Capital: P% annual charge (interest + capital repayment)

and using the following system variables

Maximum pumping rate: Q (m^3s^{-1})
Pipe length: L (m)
Pipe diameter: D (m)
Pump + motor efficiency: η

Then, allowing for some standby capacity, the installed pump power P_i can be expressed in the form

$$P_i = \frac{S\rho gHQ}{\eta}(\mathrm{W}) = \frac{SgHQ}{\eta}(\mathrm{kW}) \qquad (11.15)$$

where S is a standby factor, for example, where three pumps of equal size are installed, one of which is a standby unit, the value of S is 1.5.

Using the foregoing data, an annual cost function Ca can be written as follows:

$$C_a = \frac{P}{100}\left[C_{pu}\left(\frac{SgHQ}{\eta}\right) + C_{pi}L\right] + C_e\frac{gHQ}{\eta} \qquad (11.16)$$

For minimum cost:

$$\frac{dC_a}{dD} = 0 = \frac{P}{100}\left[C_{pu}\left(\frac{SgQ}{\eta}\frac{dH}{dD}\right) + C_{pi}L\right] + C_e\frac{gQ}{\eta}\frac{dH}{dD} \qquad (11.17)$$

The duty point head H can be written in the form:

$$H = H_{ST} + h_f$$

where H_{ST} is the static head and $h_f = KQ^2/D^5$ is the system friction head, where K is a system constant. Then

$$\frac{dH}{dD} = \frac{dh_f}{dD} = -\frac{5KQ^2}{D^6}$$

Insertion of this value for dH/dD in equation (11.17) gives the following expression for the minimum cost diameter D_{mc}:

$$D_{mc} = \left\{\frac{5KQ^2\left[(C_e gQ) + (C_{pu}SgQP/100)\right]}{\frac{p}{100}C_{pi}L\eta}\right\}^{1/6} \qquad (11.18)$$

11.5 Pumping station design

The stages in pumping station design include:

(1) selection of type, sizes and number of pumps
(2) design of general layout
(3) design of the wet well or sump
(4) selection of a pumping control system

11.5.1 Pump selection

Invariably, rotodynamic pumps are used for pumping clean water and wastewaters. In general, pump type selection is largely determined by the required duty, from which the specific speed can be calculated, thus indicating whether the pump should be of the centrifugal, mixed flow or axial flow type. The number and size of pumps is normally selected to match the pattern of flow variation. Where the pumping demand is more or less constant, there is an obvious maintenance advantage in using a single size and type of pump. Except in very high head installations, pumps are used in parallel configuration. The pumping efficiency will vary, depending on whether one, two or more pumps are working. Energy efficiency is obviously an important consideration in pump set selection, the normal design objective being the achievement of the highest possible operational efficiency, taking all modes of operation into account. While pump units of best efficiency in a particular category may be chosen for clean water pumping, the designer has generally to settle for a lesser efficiency in pumping wastewater, where the necessity to pass suspended solids calls for large flow passages and clog-free impeller geometry. The provision of standby capacity is essential. The minimum standby capacity is that which will allow the station to operate at design load with any one of its pumps down for maintenance.

11.5.2 General layout

The underground structure of a pumping station typically consists of a wet well or sump, the volume and shape of which are determined by the factors, outlined in the following section, and a dry well in which the pump set is housed, as illustrated schematically on Figs 11.10 (a) and 11.10 (b). It will be noted that the use of a vertical shaft pump allows a more compact dry well and has the environmental advantage of drive motor placement at ground level. The use of submersible pumps eliminates the necessity for a separate dry well, as shown on Fig 11.10 (c).

11.5.3 Pump sump design

Pump sumps are preferably designed to provide flooded pump suction (positive gauge pressure at the suction flange) at start-up and thus obviate priming problems. Also, as discussed later, the geometry of the sump and submergence of the inlet should be such as to avoid vortex formation and air intake.

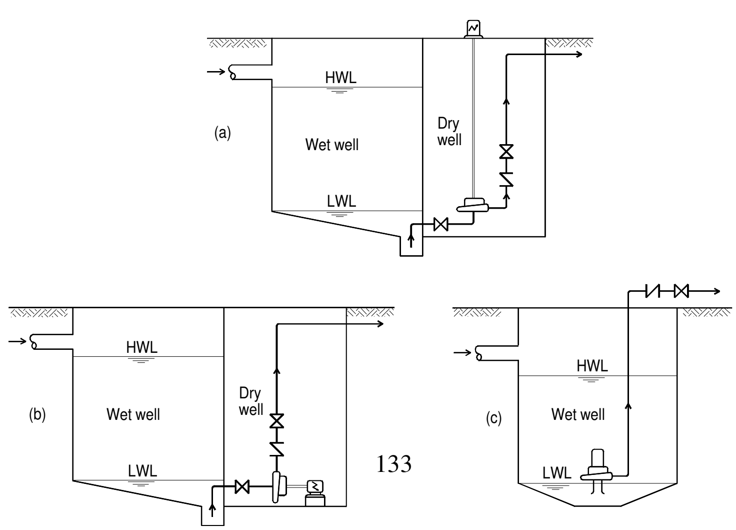

Fig 11.10 Schematic of underground pumping station structure

Where the inflow rate to the sump is variable, as in sewage and stormwater pumping, it is necessary to provide a storage volume in the sump to avoid too frequent pump starting, which would lead to motor-starter burn-out. At the same time, it is also desirable to minimise solids deposition in pump sumps and the attendant septicity problems, hence the need to avoid unduly large sumps. The sump volume required to satisfy starting frequency criteria may be calculated on the following basis, where P is the pumping rate, Q is the inflow rate to the sump and V is the effective sump volume:

$$\text{Time to empty sump} = \frac{V}{P-Q}$$

$$\text{Time to fill sump} = \frac{V}{Q}$$

Interval between starts I = time to empty + time to fill. Hence

$$I = \frac{PV}{Q(P-Q)} \tag{11.19}$$

$$N = \frac{1}{I} = \frac{Q(P-Q)}{PV} \tag{11.20}$$

where N is the frequency of starting. To obtain a maximum value for N:

$$\frac{dN}{dQ} = \frac{P-2Q}{PV} = 0$$

giving $N = N_{max}$ when $Q = P/2$. Inserting this value for Q in eqn (11.20), the corresponding value for V, which is the minimum sump volume, is found to be

$$V_{min} = \frac{P}{4N_{max}} \tag{11.21}$$

For a typical value for N_{max} of 15 h^{-1}, V_{min} = P/60 or a volume corresponding to a pumping duration of one minute.

The foregoing minimum sump volume refers to the volume between pump cut-in level and pump cutout level, as illustrated on Fig 11.11. It is normal practice to separate the cut-in and cut-out levels for individual pumps, as shown on Fig 11.11. Where this is the case, the appropriate value for P in equation (11.21) is the output capacity of one pump. It should also be noted that the separation of cut-in and cut-out levels for pumps is desirable on the grounds of minimisation of waterhammer effects associated with starting and stopping.

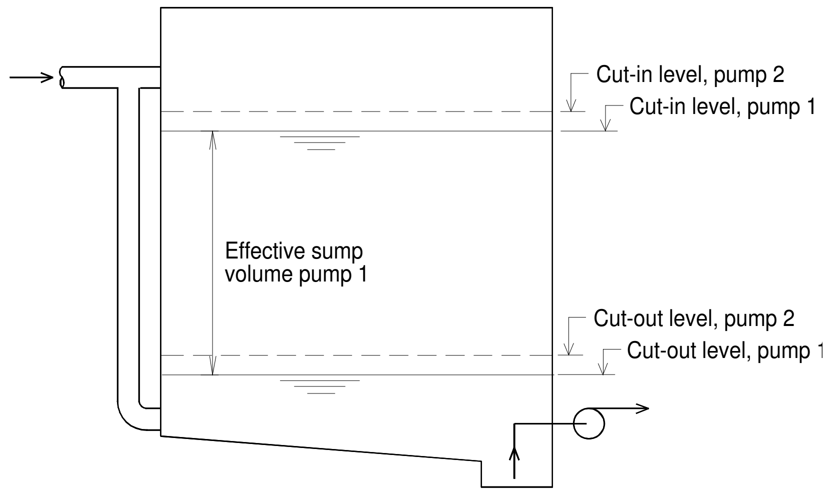

Fig 11.11 **Minimum sump volume for start/stop pumping**

The selected pump cut-out level is significant in the separate contexts of cavitation and air entrainment, both of which reduce pumping efficiency. Cavitation problems are avoided by meeting the NPSH pump specification.

Air entrainment adversely affects pump performance. Denny (1956) reported that 1% free air could reduce the efficiency of a centrifugal pump by 5 to 15%. Air entrainment may result from the formation of air-entraining vortices in the vicinity of the pump suction, as illustrated in Fig 11.12, or may be caused by a cascading inflow to the pump sump resulting in the dispersion of air bubbles in the fluid bulk.

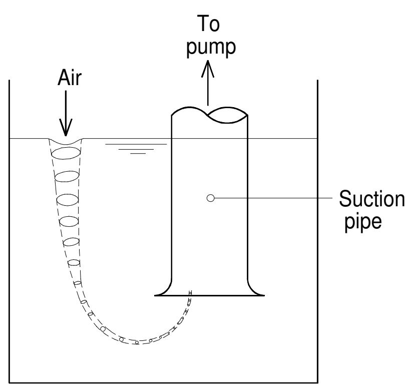

Fig 11.12 Air-entrainment vortex

The tendency towards vortex formation can be effectively controlled by appropriate geometric design of pump sumps. Boundary discontinuities, which cause boundary layer separation or flow obstruction, which give rise to vortex shedding, should be avoided. Where feasible, flows should be symmetrically directed to individual pump intakes. For detailed discussion related to the geometric aspects of pump sump design the reader is referred to the publications of Hattersley (1965), Prosser (1977) and Sweeney et al. (1982).

Design guidelines (Prosser, 1977) for a single pump intake are presented in Fig 11.13. For this type of intake layout a length of straight approach channel of at least 10 D_b is recommended. Iversen (1953) has shown that pump efficiency is adversely if the bellmouth floor clearance is less than about 0.5 D_b.

For large multiple intake sumps and sumps of unusual geometry the design process can be greatly aided by physical model studies.

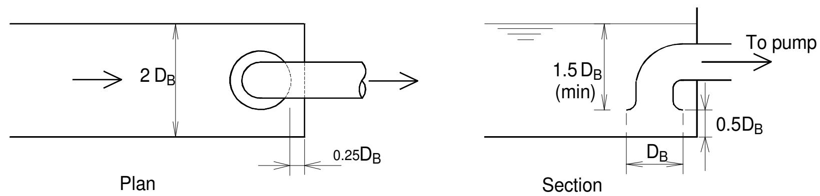

Fig 11.13 Single pump intake layout

11.6 Control of pumping

Pumping systems are normally automatically controlled, responding to signals generated by the particular duty regime. In wastewater pumping installations, the control signal is normally generated by the water surface level in the pump sump. A variety of level-sensing devices are available, including electrode rods, floats, ultrasonic devices, pressure sensors, etc. These sensors produce a signal which switches on a pump when the water level in the sump reaches the upper set point for that pump or switches off a pump when the water level drops to the cut-out level for that pump. Typically, each pump has its own cut-in and cut-out level, as illustrated on Fig 11.11. This means that only one pump is involved in any switching operation, thus minimising the resulting step change in flow in the rising main. The minimum height difference between two adjacent water surface control levels should be such as top avoid simultaneous switching of these pumps due to water surface wave action.

Pump control may also be exercised in response to signals generated on the delivery side. This type of control is typical of water supply installations, where the requirement may, for example, be the maintenance of a fixed pressure at a specified location in the distribution pipe system. The use of variable speed electric motor pump drives offers considerable control flexibility in this category of application, allowing a close matching of supply and demand and thus reducing energy costs.

Pump installations are typically fitted with a valve set, as illustrated on Fig 11.10, that is, with a gate or sluice valve on the suction side and with a gate valve plus a non-return valve on the delivery side. The gate valves allow the isolation of the pump and non-return valve for repair/maintenance purposes, while the non-return valve prevents the emptying of the rising main when the pump is not operating.

All pumping installations should be checked at the design stage for waterhammer effects and provided with appropriate control devices, if required; see Chapter 6.

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