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Radial, mixed and axial flow pumps. Introduction

Endorsed by
The Institution of Mechanical Engineers
The Institution of Chemical Engineers

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(*Continued on inside back cover*)

RADIAL, MIXED AND AXIAL-FLOW PUMPS. INTRODUCTION

CONTENTS

RADIAL, MIXED AND AXIAL-FLOW PUMPS. INTRODUCTION

1. NOTATION AND UNITS

The terminology and notation used in this Item largely follow those employed by pump specialists.

		Units
a	pressure wave velocity	m/s
C	constant in Equation (7.1)	
D	impeller diameter	m
d	pipe internal diameter	m
E	Young's modulus	N/m^2
g	acceleration due to gravity, $g \simeq 9.81$ m/s^2	m/s^2
H	total head referred to agreed vertical datum; this is by convention a gauge quantity unless otherwise defined	m
ΔH	total head rise produced by pump, $\Delta H = H_o - H_i$	m
K	liquid bulk modulus	N/m^2
k	arbitrary constant	
L	length of system	m
N	rotational speed of impeller	rev/min
$NPSHa$	net positive suction head available referred to agreed vertical datum	m
$NPSHr$	net positive suction head required referred to pump vertical datum	m
n_ω	specific speed	rad
P	pump shaft power	W
p	absolute total pressure referred to agreed vertical datum	Pa or N/m^2
p_s	static pressure	Pa or N/m^2
p_v	vapour pressure of pumped liquid	Pa or N/m^2
Δp	total pressure rise produced by pump, $\Delta p = p_o - p_i$	Pa or N/m^2
Q	volume flow rate through pump	m^3/s
S_ω	suction specific speed	rad

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t	thickness of pipe wall	m
V	mean flow velocity	m/s
z	vertical height, positive upwards measured from agreed vertical datum	m
η	pump efficiency expressed as a percentage, *i.e.* $\eta = 100 \times Q\rho g\Delta H/P$	per cent
μ	liquid dynamic viscosity	N s/m^2
ν	liquid kinematic viscosity $(\nu = \mu/\rho)$	m^2/s
ρ	density of pumped liquid	kg/m^3
τ	pump shaft torque	Nm
ω	angular speed of impeller	rad/s

Subscripts

a	denotes atmospheric value
bep	denotes value at pump best efficiency point
i	denotes value at pump inlet (suction) referred to pump vertical datum where relevant
o	denotes value at pump outlet (discharge) referred to pump vertical datum where relevant

Superscript

$'$	prime denotes value associated with water as pumped liquid

2. INTRODUCTION

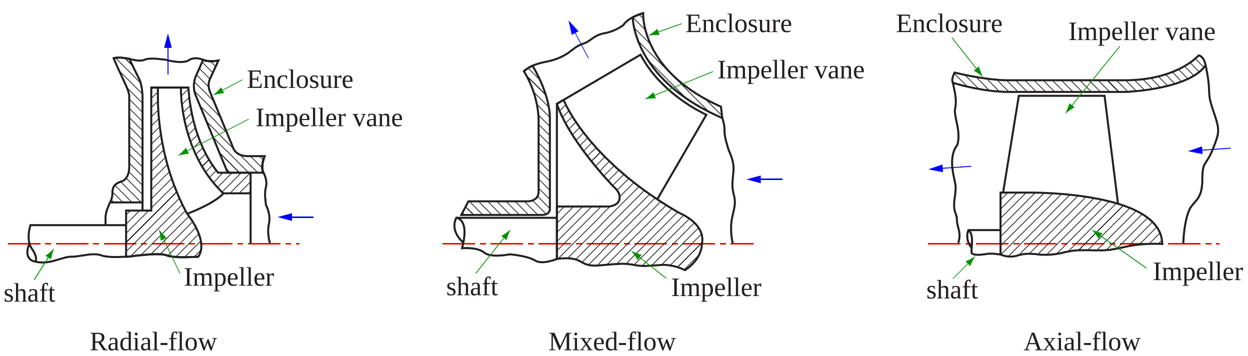
Radial, mixed and axial-flow pumps belong to a class of machines known as rotodynamic. As a class, they are suitable for the majority of liquid pumping applications but notable exceptions include metering, handling of highly viscous liquids (*e.g.* liquid with $\nu > 200$ cSt but see Item No. 80031 for further details) and very high pressure/low flow rate requirement (quantified in Section 3.7.1). For these applications, positive displacement pumps are normally used.

This Item, which is primarily intended for the non-specialist, presents pump characteristic properties and gives guidance on how they may be matched to system requirements, identifying the problem areas and thus providing the user with sufficient background for selecting the most appropriate type of pump, writing its specification and appraising tenders. It is intended for the use with the other Items in the series on pumps:

Radial, mixed and axial-flow pumps.	Size estimation and specification, Item No. 80031
Radial, mixed and axial-flow pumps.	Glossary of terms, Item No. 81001
Radial, mixed and axial-flow pumps.	Conversion factors, Item No. 81002.

2.1 Item Scope

The categories of pumps dealt with are single and multi-stage versions of radial-flow, mixed-flow and axial-flow pumps. These three categories of pumps are identified by the nature of the flow through the impeller, see Sketch 2.1. Radial-flow and mixed-flow pumps are commonly classed as centrifugal pumps.



Sketch 2.1 The major pump categories

The Item covers the pumping of clean, single-phase liquids only but it should be noted that rotodynamic pumps are available for handling slurries, sewage and other liquids containing solids or fibrous materials. Further information on this is given in Derivations 44 and 50 (Section 8).

2.2 Item Layout

In Sections 3 and 4, the Item leads the user through the hydraulic and mechanical aspects of pumps pertinent to their selection. Section 5 presents information on the various national and international standards relevant to pumps and in Sections 6 and 7 the Item identifies the features of systems which should influence pump selection.

Throughout the text, superscripts are used to denote references and derivations. They are listed in Section 8.

3. HYDRAULIC ASPECTS OF PUMPS

3.1 How Pumps Work

The impeller of a rotodynamic pump converts shaft power into fluid power by increasing the angular momentum of the flow. This gives rise to an increase in the energy of the flow, part of which appears as an increase in static pressure and part of which appears as an increase of kinetic pressure. The relative gains of static and kinetic pressures depend upon the design of the impeller. The gain in kinetic pressure is partially converted into static pressure in the impeller enclosure, see Section 3.3.

3.1.1 Pump total head rise

Although pressure has the dimensions of force per unit area, it may also be represented by a dimension of length (termed "head"). The term "head" implies the height of a column of liquid above a certain datum level and thus a change in head is related to a change in pressure by Equation (3.1), *i.e.*

$$\Delta H = \Delta p/(\rho g), \tag{3.1}$$

where g is the acceleration due to gravity.

The performance of rotodynamic pumps is, by convention, expressed in terms of a total head rise* rather than a pressure rise. Because of its rotodynamic action, the pressure rise produced by a rotodynamic pump is directly proportional to the density of the pumped liquid. However, since, at a given pressure, head is inversely proportional to density, the head rise from a rotodynamic pump is independent of density.

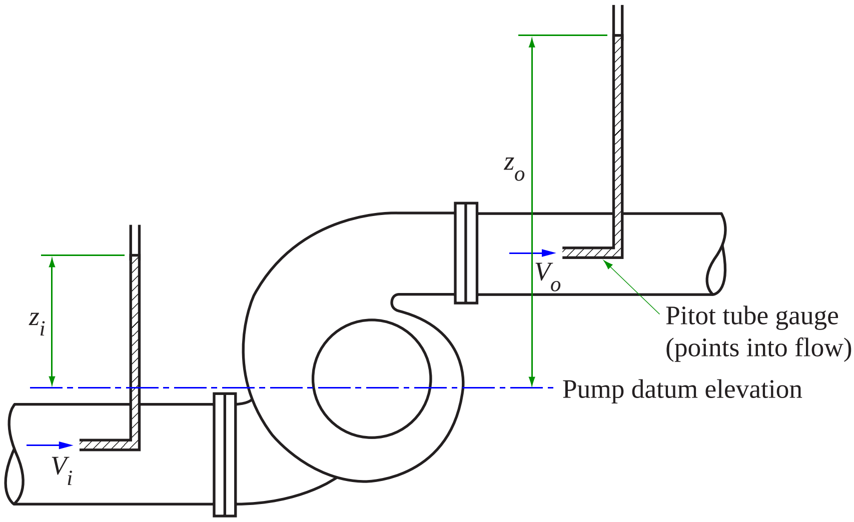
Note that total head, or total pressure, as used in this Item, refers to a quantity that is the sum of static, kinetic and elevation terms, *i.e.*

$$H = (p_s - p_a)/(\rho g) + V^2/(2g) + z, \tag{3.2}$$

$$p = p_s + ½\rho V^2 + \rho g z. \tag{3.3}$$

Note that H as defined in Equation (3.2) is a gauge quantity as is conventional for pumps but p is an absolute quantity insofar as it includes the atmospheric pressure in the term p_s. Both definitions, however, rely on a defined datum for the elevation term, z. The gauges illustrated in Sketch 3.1 measure gauge total head because, by pointing into the flow, they not only measure the static head of the flow but also the kinetic contribution, $V^2/2g$.

* In this Item, the term total head *rise*, denoted by ΔH, is used to describe the increase in total head produced by a pump. It should be noted, however, that, elsewhere, it is the usual practice to abbreviate this to pump total head, correspondingly denoted by H. Since H is used herein to denote the total head anywhere in the system, ΔH is used to denote the pump total head rise.



$H_i = z_i$ Note: (i) manometers must contain the pumped liquid

$H_o = z_o$

$\Delta H = z_o - z_i$ (ii) the arrangement shown here is purely schematic and should not be taken as sufficient for practical test measurements.

Sketch 3.1 Schematic illustration of head definitions

If the kinetic contributions, $V^2/(2g)$, were equal on both the inlet (suction) and outlet (discharge) flanges of the pump, manometers with static tappings (flush with the pipe wall) would indicate the same value of ΔH as the pitot tubes in Sketch 3.1. In practice, the inlet flange of a pump is often larger than the outlet and so the kinetic contribution in the inlet pipe, $V_i^2/(2g)$, is lower than $V_o^2/(2g)$ in the outlet pipe. The change in total head is, however, the correct representation of the increase in energy of the flow. Note that, because of the inclusion of the elevation terms in the definitions of total head and total pressure, values for ΔH (or Δp) are independent of the orientation of the pump.

3.2 Impeller Design and Specific Speed

Although alternative presentations are possible, the performance of a pump is usually characterised by a plot of total head rise, ΔH, versus flow rate, Q. For a given pump speed, there is a particular value of Q at which the pump operates most efficiently and, as far as possible, a pump is selected to fulfil its duty at or near that point. Values of pump variables corresponding to this "best efficiency point" are denoted here by the subscript "*bep*".

The shape of an impeller gives it a performance that may, or may not, suit its required duty and, although there are three main impeller categories (radial-flow, mixed-flow and axial-flow), the changes in impeller geometries between these categories are progressive.

The duties best suited to a given impeller geometry may be classified in a quantitative way through a non-dimensional group of terms known as "specific speed". Because it is dimensionless, specific speed can classify impeller geometries largely regardless of their size or operating speed.

The definition of n_ω in this Item requires the impeller rotational speed to be expressed in units of radians per unit time.

Specific speed is defined by

$$n_\omega = \frac{\omega Q_{bep}^{\,1/2}}{(g\Delta H_{bep})^{3/4}} \text{ rad,} \tag{3.4}$$

where Q_{bep} is the flow rate per impeller inlet* corresponding to the best efficiency point† and ΔH_{bep} is the corresponding total head rise per impeller*.

To assist conversion between n_ω and the many different non-coherent definitions of specific speed used in other publications, the following abbreviated table is provided. A fuller table of conversion factors is available in Item No.81002[23].

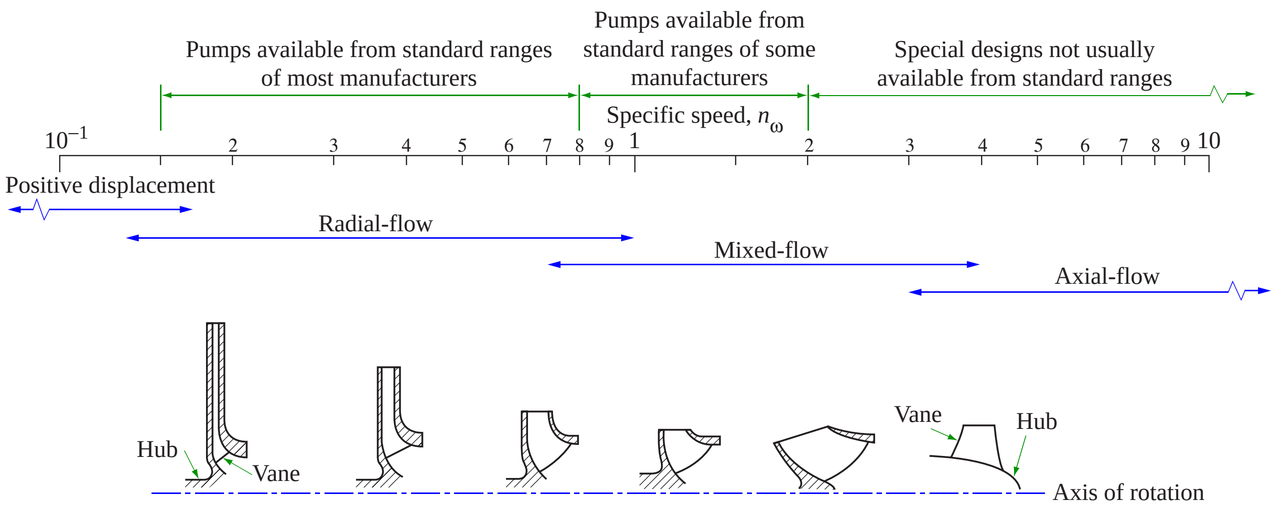
Specific speed conversion factors			
To convert from	rev/min (Imp gal/min)$^{1/2}$ ft$^{-3/4}$	to n_ω, divide by	2494
	rev/min (US gal/min)$^{1/2}$ ft$^{-3/4}$		2733
	rev/min (l/s)$^{1/2}$ m$^{-3/4}$		1673
	rev/min (m^3/h)$^{1/2}$ m$^{-3/4}$		3175
	rev/min (m^3/s)$^{1/2}$ m$^{-3/4}$		52.92
Note these factors also apply to suction specific speed, S_ω, which is dealt with in Section 3.4.			

To illustrate the significance of n_ω as a classifier of impeller types, Sketch 3.2 shows the geometries of impellers optimised for conditions corresponding to progressively higher specific speeds.

Since there is a useful correlation between specific speed and pump types, n_ω can also be used to generalise the performance of pumps. For example, Sketch 3.3 gives an approximate correlation of η_{bep} for various specific speeds and, later in this Section, other pump variables are classified in terms of n_ω. However, in Sketch 3.3, the trends of η_{bep} are not completely correlated by n_ω because n_ω alone is not capable of absorbing all effects of scale. Small pumps, *i.e.* those with low values of Q_{bep}, tend to be hydrodynamically less efficient than larger pumps as well as having disproportionately higher losses due to mechanical friction. The second variable, Q_{bep}, is therefore required to account for this effect of scale but see also Section 3.5.

* Some pumps, termed "multi-stage", consist of a series of impellers cascaded together. Usually the impellers are similar and share the duty total head rise equally. Other pumps, termed "double-suction", contain a symmetrical double impeller that has two inlets sharing the duty flow rate. Since specific speed is used to correlate the duties best suited to a given impeller shape, the value of Q_{bep} for use in Equation (3.4) should be that relating to each impeller inlet, and the value of ΔH_{bep} that for the impeller under consideration.

† Specific speed, n_ω, is further qualified by the need to use values of Q_{bep} and ΔH_{bep} associated with the full size of impeller; impellers machined down from their full size are often employed to alter the performance of a pump (see Section 3.6) and these "trimmed" impellers may exhibit characteristics that cannot be correlated by specific speed.



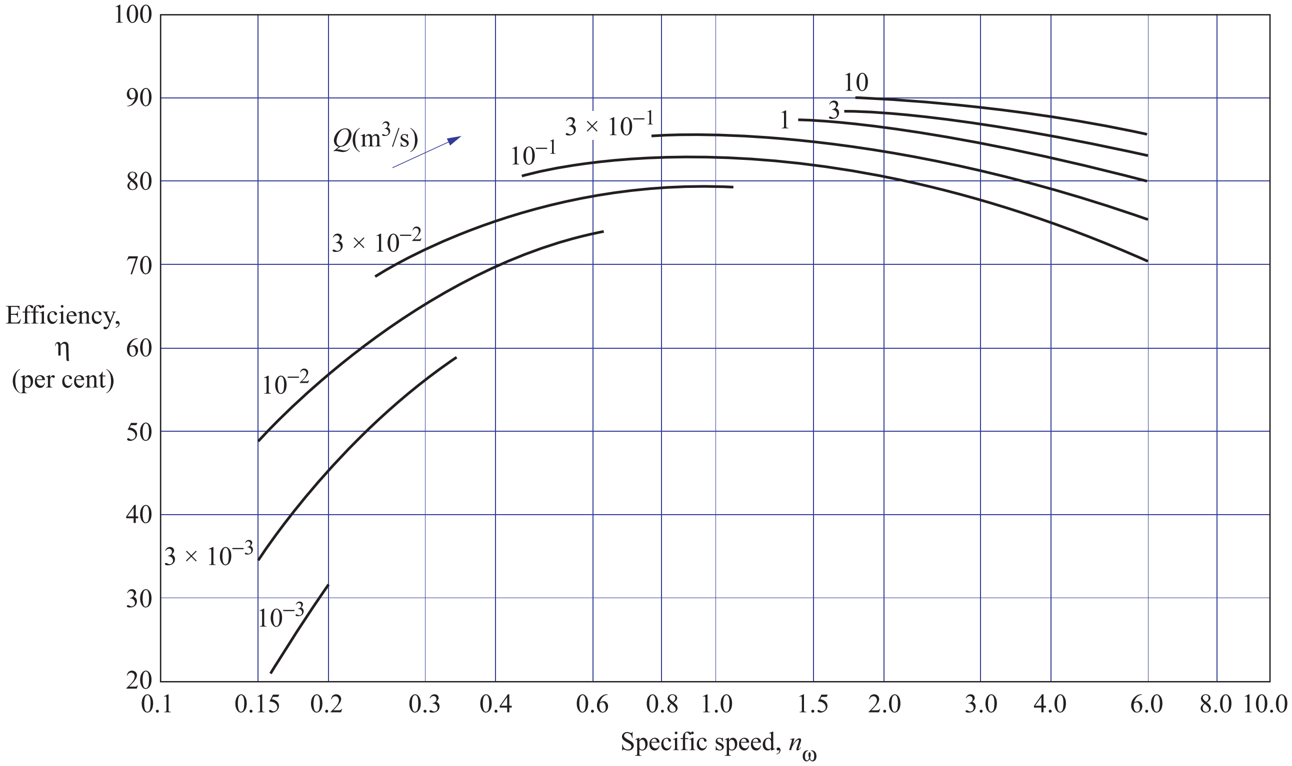
Sketch 3.2 Approximate variation of geometries of impellers having different specific speeds

Specific speed can also be used to typify the shape of pump performance curves represented non-dimensionally. Sketch 3.4 indicates the general shape of the head rise versus flow rate characteristics of different pumps having different specific speeds. The axes are labelled with ratios of Q/Q_{bep} and $\Delta H/\Delta H_{bep}$ such that values of unity correspond to the best efficiency point.

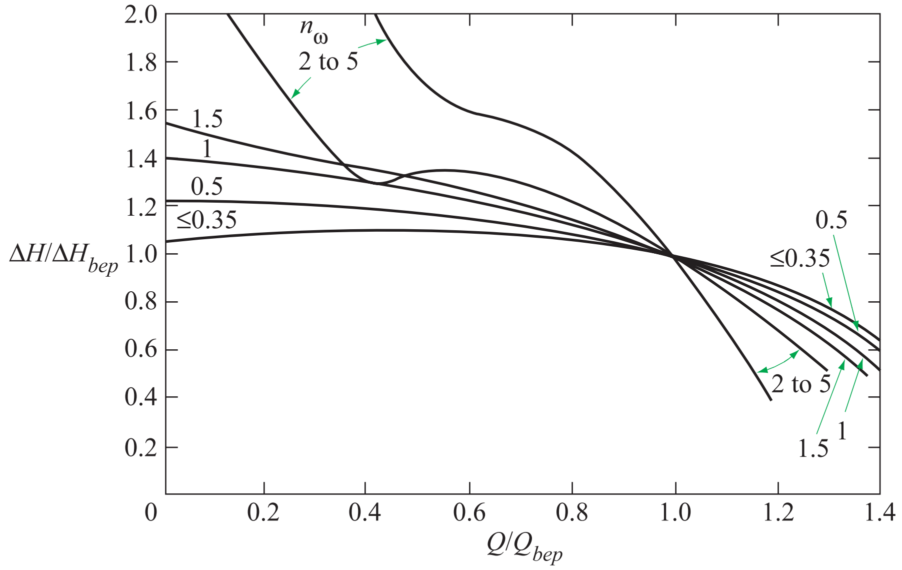
For pumps of $n_\omega > 2$ it is not possible to typify the shapes of pump characteristics. Such pumps are often of mixed or axial-flow design and are not generally available from manufacturers' standard ranges. The large scope for altering the geometry of such impellers means that pumps of the same specific speed may exhibit different shapes of characteristics.

Head rise characteristics having a shape similar to that of the curve labelled with $n_\omega < 0.35$ in Sketch 3.4 are called unstable on account of their positive slope at low flow rates. The significance of this, and of other features of the curve, is explained in Section 7.1.

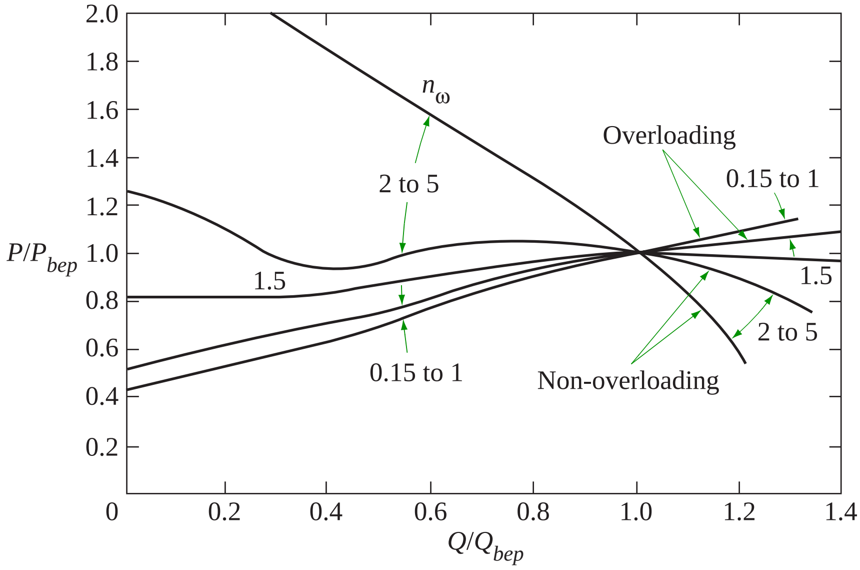
Knowledge of the shape of the power characteristic is also necessary to ensure that the driving unit will not be overloaded however the pump may be operated. Sketch 3.5 illustrates the shapes of typical power characteristics of pumps having different specific speeds. Those power characteristics that develop positive slope at $Q \geq Q_{bep}$ are called “overloading” whilst those of negative slope at $Q \geq Q_{bep}$ are called “non-overloading”. The significance of this in relation to the protection of the driving unit is explained in Section 7.3.



Sketch 3.3 Approximate correlation of peak efficiencies for various specific speeds and flow rates

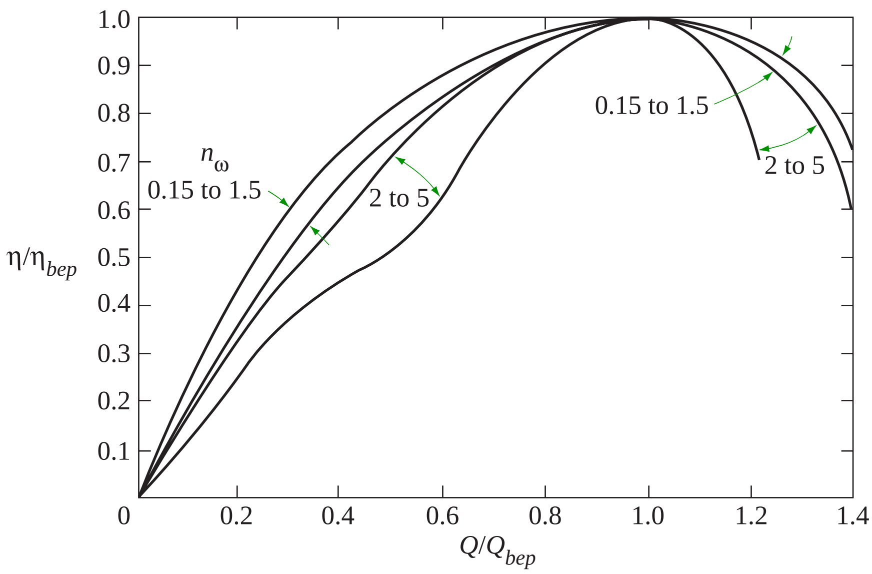


Sketch 3.4 Head rise characteristics of pumps having different specific speeds



Sketch 3.5 Power characteristics of pumps having different specific speeds

It is also possible to correlate very broadly the shape of the efficiency curves of pumps having different specific speeds. These are illustrated in Sketch 3.6.



Sketch 3.6 Efficiency characteristics of pumps having different specific speeds

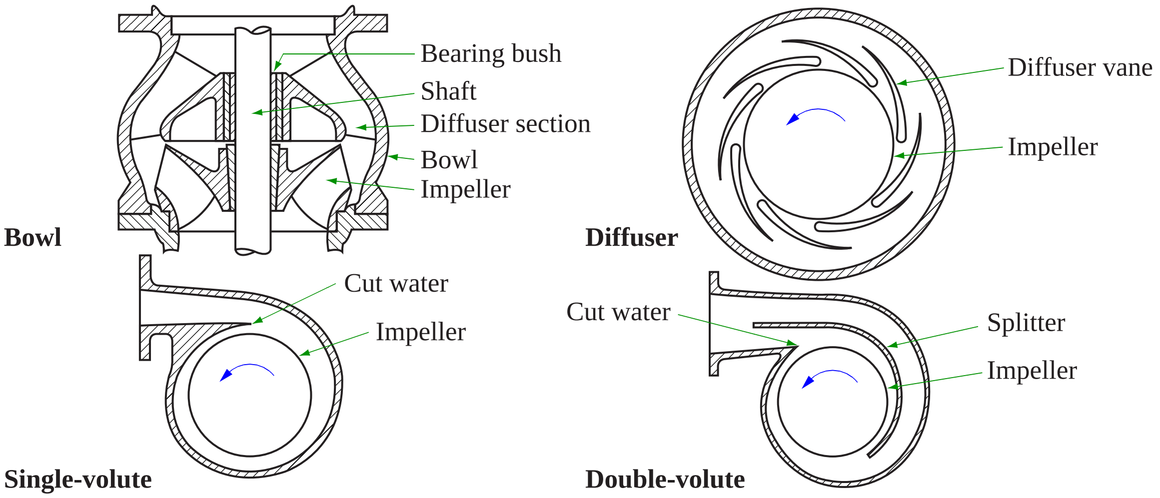
3.3 Impeller Enclosures

The impeller enclosure of a pump, which is often an integral part of its structural casing, not only collects the liquid as it is discharged from the periphery of the impeller but, on pumps other than axial flow, also slows down the high velocities of the discharged liquid thus diffusing the excess of kinetic pressure into a static pressure rise. Depending on the type of enclosure, this is achieved through its shape or through diffuser vanes manufactured inside the enclosure.

Volutes and bowls are the most often-used types of enclosures, see Sketch 3.7. Of these, volutes are the more common as they are used with the more common single-stage pumps in addition to several multi-stage designs.

Bowls are also used, but to a lesser extent, on single-stage pumps but their main application is in vertical multi-stage designs where the in-line arrangement of their suction and discharge nozzles allows simple multi-staging. For single-stage axial-flow designs, the enclosure is usually a simple cylindrical shroud.

Sketch 3.7 illustrates the different types of enclosures available.



Sketch 3.7 Bowl, single-volute, double-volute, and diffuser enclosures

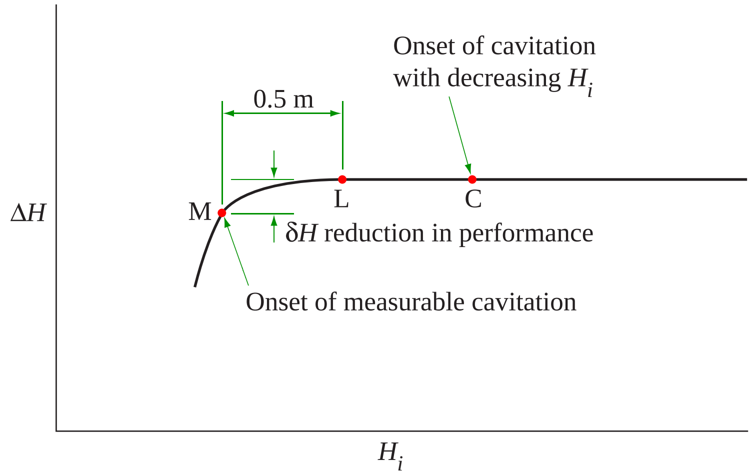
Single volutes are usual on small pumps but double volutes, although more costly to manufacture, are also used especially on larger sizes since the splitter reduces the radial loads on the pump bearings by preserving flow symmetry. In addition, the splitter contributes to the strength of the casing which can be a significant factor in large, high pressure, pumps. Note double-volute pumps are not the same as double-suction pumps which are described in Table 4.1.

Diffuser enclosures offer the same advantages as double volutes and, because the position of the outlet is not governed by flow considerations, diffuser enclosures are frequently used on multi-stage radial-flow pumps where they discharge into an annular channel.

3.4 Cavitation and Suction Specific Speed

Cavitation refers to the localised formation and collapse of bubbles in a liquid. Bubbles can form when the local pressure at a point falls below the vapour pressure of the liquid although they can also form at rather higher pressure due to dissolved gases coming out of solution. In either case, measurable cavitation as defined by a reduction in pump performance, see Sketch 3.8, should be avoided. Once formed, vapour cavities tend to collapse rapidly; the resulting implosions, which are usually noisy, erode surfaces local to the collapsing bubbles and induce vibrations that shorten the lives of bearings and seals.

Sketch 3.8 shows that cavitation starts at point C at a value of H_i that is considerably above that at point M corresponding to a measurable reduction in performance, *i.e.* δH. To allow pump manufacturers to present consistent data on the minimum inlet total head required, point M is usually defined with $\delta H/\Delta H = 0.03$ with cold water as the pumped liquid*. However, it should be noted that continuous operation at point M is not recommended because, apart from the 3 per cent loss in head rise, unacceptable rates of erosion and levels of noise and vibration may occur. It is common practice, therefore, to establish the minimum allowable inlet total head at point L which is at an inlet total head at least 0.5 m above point M. Although point L will usually be in a regime of some cavitation, the effect of cavitation on performance between points L and C is small and it is seldom economic to prevent all risk of cavitation by operating with an inlet total head above that at point C.



Sketch 3.8 Pump total head rise as a function of inlet total head for constant flow rate and speed

Note that the 0.5 m increment on $H_{i,M}$ is advisory and a different criterion may be more suitable for some installations. Critical applications should always be discussed with the pump manufacturer.

3.4.1 Net positive suction head available, NPSRa

The value of $NPSHa$ at any point in a system is equal to the increment of total pressure above the vapour pressure expressed in units of head. Hence

$$NPSHa = (p - p_v)/(\rho g), \tag{3.5}$$

which may alternatively be expressed as

$$NPSHa = H + (p_a - p_v)/(\rho g), \tag{3.6}$$

where p_v is the vapour pressure of the pumped liquid and p and H are the absolute total pressure and gauge total head respectively defined with respect to an agreed vertical datum. When

$$NPSHa = V^2/(2g) + z,$$

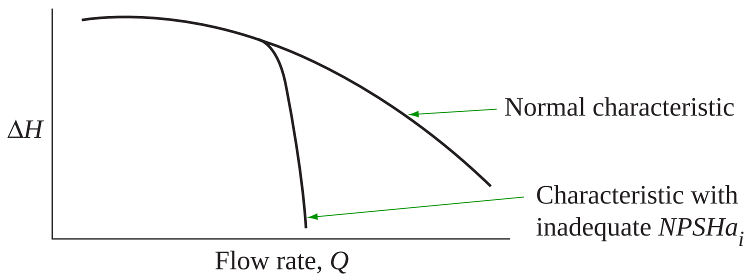
the static pressure is equal to the vapour pressure.

* Cold water is normally used for determining cavitation conditions since it has physical properties that allow cavitation to be recognised more clearly than with other liquids. A value for $\delta H/\Delta H = 0.03$ for defining measurable cavitation is an often-used rather than a universally accepted criterion.

Although $NPSHa$ is a system characteristic, it is a commonly-used tool for analysing the cavitation performance of pumps. In such a context, it is conventional to define the vertical datum as the "pump datum elevation". On small pumps, "pump inlet" will define this with sufficient precision for engineering purposes but, on larger units, a definition of the vertical datum to which the value of $NPSHa$ is referred is needed. Section 5.1 details the pump vertical datum positions adopted by several international standards. However, any position may be used provided it is clearly defined.

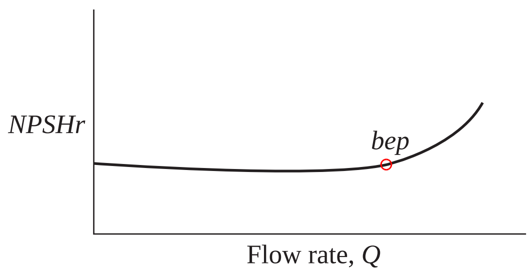
3.4.2 Net positive suction head required, NPSRr

The quantity $NPSHr$ is used to quantify the cavitation performance of pumps. It is numerically equal to that value of $NPSHa$ at the pump inlet, referred to the pump datum elevation, at which measurable cavitation occurs. Sketch 3.9 illustrates what happens to the pump total head rise characteristic as a result of an inadequate $NPSHa$ at inlet.



Sketch 3.9 Pump total head rise characteristic with inadequate $NPSHa_i$

The value of $NPSHa$ required to prevent measurable cavitation in a pump varies with flow rate as is shown by a typical $NPSHr$ characteristic in Sketch 3.10.



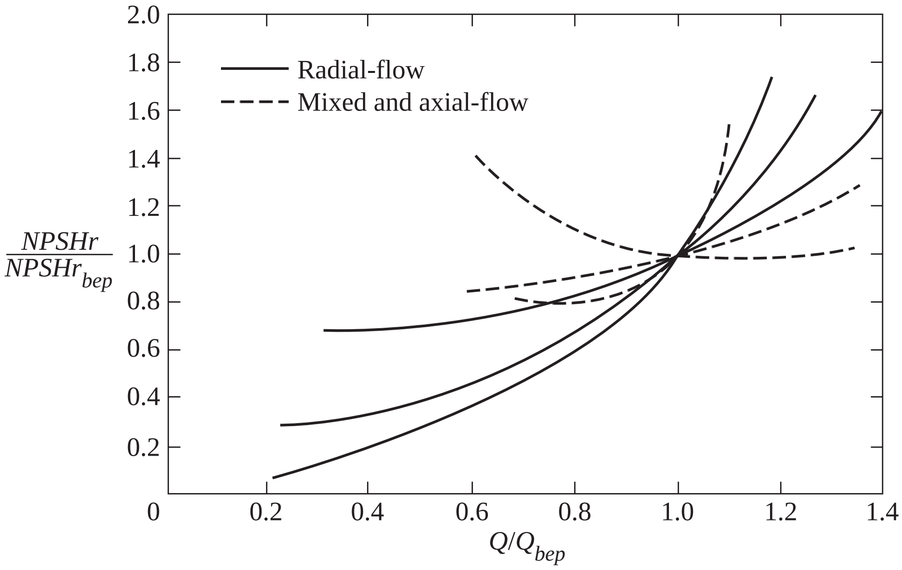
Sketch 3.10 Typical pump $NPSHr$ characteristic

Note that $NPSHr$ is a pump characteristic, *i.e.* it is largely independent of the pumped liquid because the liquid characteristics are already largely accounted for by the vapour pressure terms in Equations (3.5) and (3.6) but, as mentioned in the footnote to Section 3.4, cold water is the usual test liquid. Although the onset of cavitation, point C in Sketch 3.8, can be predicted whatever the liquid with a value of $NPSHa$, the difference between $NPSHa$ at points M and C depends to some extent on liquid properties other than vapour pressure. However, because cold water has properties that make points M and C closer together than for any other common liquid, it is conventional to regard a value of $NPSHr$ for cold water to be safe for any other liquid*. Note that in common with other pump characteristics expressed in units of head, $NPSHr$ is independent of the liquid density.

* Methods are available[44] to account for the difference between $NPSHr$ for cold water and other liquids. However, because of the uncertainties, not least of which is establishing the definition of $\delta H / \Delta H$ (point M in Sketch 3.8), it is recommended that, unless there is good reason to believe otherwise, water-derived values of $NPSHr$ should be used unfactored with other liquids.

It is possible to correlate very broadly the $NPSHr$ characteristics of pumps in terms of their impeller types. This is illustrated in Sketch 3.11.

For all types of pumps, Sketch 3.11 shows that $NPSHr$ increases with flow rate above Q_{bep}. Thus it is especially important when planning pumping provisions to ensure an adequate $NPSHa$ to cover all continuous modes of operation, including contingencies such as the failure of one of two or more pumps operating in parallel when the flow rate through the remaining pumps will increase, see Section 7.3.2.



Sketch 3.11 Typical $NPSHr$ characteristics for axial, mixed and radial-flow pumps

When using a manufacturer's $NPSHr$ catalogue data, it is important to recognise that continuous operation with $NPSHa_i = NPSHr$ is not recommended. As explained in Section 3.4, the minimum advisory value for $NPSHa$ at the pump inlet is

$$NPSHa_i \geq NPSHr + 0.5 \text{ m.} \tag{3.7}$$

3.4.3 Suction specific speed

Suction specific speed is a non-dimensional representation of $NPSHr$ and is defined by

$$S_\omega = \frac{\omega Q_{bep}^{\frac{1}{2}}}{[gNPSHr_{bep}]^{\frac{3}{4}}} \text{ rad,} \tag{3.8}$$

where Q_{bep} is the flow rate per impeller inlet*. As with specific speed, the constituent terms of S_ω are by convention evaluated at the best efficiency point. In other words both Q and $NPSHr$ in Equation (3.8) should be the best efficiency values for a given pump. Note that a high value for S_ω implies a good suction performance, *i.e.* low $NPSHr$.

* With multi-stage pumps, S_ω, may still be used to indicate the cavitation performance but of the first stage impeller only since this will clearly have the lowest $NPSHa$ and hence the greatest tendency to cavitate. For the case of a double-suction pump (symmetrical double impeller), the value of Q_{bep} for use in Equation (3.8) should be that relating to the flow rate through each impeller inlet, *i.e.* half the pump flow rate.

Again, just as specific speed is often quoted in non-coherent units in other publications, suction specific speed may also be quoted in such units. Thus the conversion table for n_ω in Section 3.2 may equally be applied to S_ω.

Most pumps have values of S_ω between about 2.5 and 4, a value of 2.8 being typical. Values of S_ω above these can be achieved by fitting an inducer to the impeller shaft which increases the pressure of the liquid before it enters the eye of the impeller giving values of S_ω up to about 8. However, a pump fitted with an inducer may be restricted to a narrow range of operation. Note that although these values for S_ω nominally apply to *NPSHr* referred to a pump datum consistent with BS 5316[11, 15], they are typical rather than definitive values and may therefore be taken as applicable to any of the pump datum definitions shown in Table 5.1.

3.5 Pump Similarity Laws

Geometrically similar* pumps have performance characteristics that are related by expressions known as "similarity" or "affinity" laws. For example, if the subscript "1" is used to denote one pump running at speed N_1 and the subscript "2" another, geometrically similar, running at speed N_2, then, if both operate on the same part of their performance curves (*e.g.* both at best efficiency), the following expressions apply:

$$\frac{Q_1}{D_1^3 N_1} = \frac{Q_2}{D_2^3 N_2} \tag{3.9}$$

and

$$\frac{\Delta H_1}{D_1^2 N_1^2} = \frac{\Delta H_2}{D_2^2 N_2^2}. \tag{3.10}$$

3.5.1 Effect of speed change

It is frequently necessary to predict the performance characteristics of a pump operating at a speed, N_2, different from that, N_1, at which it was tested. For this Equations (3.9) and (3.10) may be rearranged as follows:

$$\Delta H_2 = \Delta H_1 \left(\frac{N_2}{N_1}\right)^2 \tag{3.11}$$

and

$$Q_2 = Q_1 N_2 / N_1. \tag{3.12}$$

The shaft torque, τ_2, and power, P_2, corresponding to the new speed, N_2, are given respectively by

$$\tau_2 = \tau_1 \left(\frac{N_2}{N_1}\right)^2 \tag{3.13}$$

and

$$P_2 = P_1 \left(\frac{N_2}{N_1}\right)^3. \tag{3.14}$$

* Two pumps are geometrically similar when one pump and a scale factor completely defines the other.

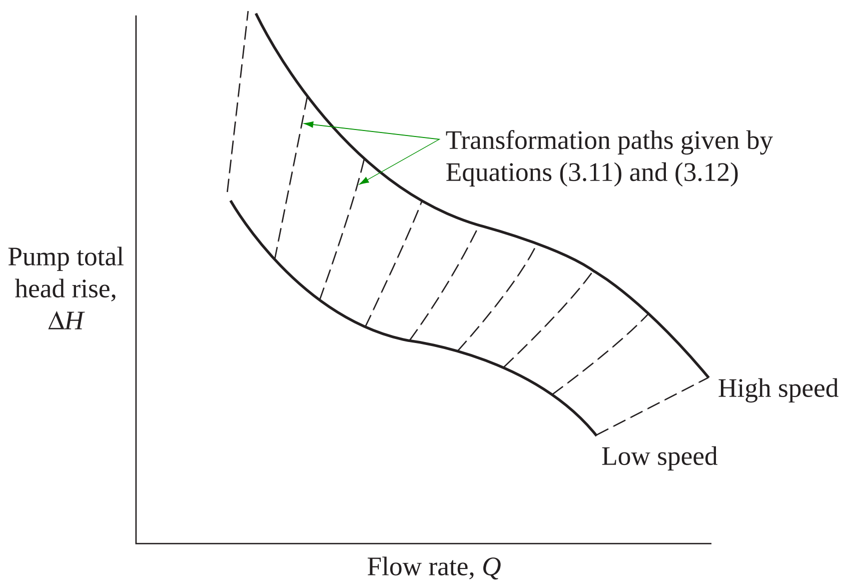
Equations (3.11) to (3.14) may be used to map a known characteristic (*e.g.* ΔH_1 versus Q_1) into a new characteristic (*e.g.* ΔH_2 versus Q_2) corresponding to a new speed. Sketch 3.12 illustrates this process for a head rise characteristic.

It should be noted that Equations (3.11) to (3.14) alone may not be used to predict operating points in a given application resulting from a change in speed. This depends also upon the system characteristic as is explained in Section 7.8.

The $NPSHr$ of a pump is affected by rotational speed in much the same way as the pump total head rise. To estimate the $NPSHr$ resulting from a limited change in speed*, *i.e.* $0.7 < N_2/N_1 < 1.5$, the following relationship may be used:

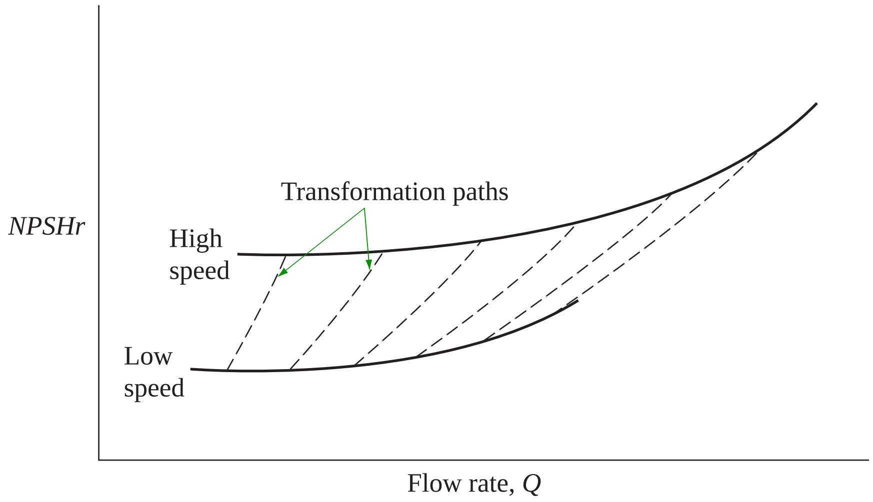
$$NPSHr_2 = NPSHr_1\left(\frac{N_2}{N_1}\right)^2 . \tag{3.15}$$

Equation (3.15) may also be used to map the known $NPSHr$ characteristic ($NPSHr_1$ versus Q_1) into a new characteristic ($NPSHr_2$ versus Q_2) corresponding to a new speed, see Sketch 3.13.



Sketch 3.12 Effect of change of speed on pump total head rise characteristic

* A restriction is necessary on the range of applicability of Equation 3.15 as it has only a limited theoretical justification particularly where low values of $NPSHr$ are involved.



Sketch 3.13 Effect of change of speed on pump *NPSHr* characteristic

3.5.2 Effect of size change

Because the full-scale testing of large pumps is costly, the results from model test are occasionally used to predict prototype performance. The similarity laws, Equations (3.9) and (3.10), are a reliable means of scaling up model tests provided both geometric and hydraulic similarity conditions are met between model and prototype.

To ensure geometric similarity, attention to detail is required and particular care is necessary when employing cast materials. To ensure hydraulic similarity, it is conventional to run the model at a speed sufficient to produce a total head rise close to that required from the prototype.

The rearranged forms of Equations (3.9) and (3.10) that may be used for predicting the total head rise and flow rate characteristic of a new (or prototype) pump from the known characteristic of another similar (or model) pump are as follows:

$$\Delta H_2 = \Delta H_1 \left(\frac{D_2 N_2}{D_1 N_1}\right)^2 \tag{3.16}$$

and

$$Q_2 = Q_1 \left(\frac{D_2}{D_1}\right)^3 \frac{N_2}{N_1} \quad . \tag{3.17}$$

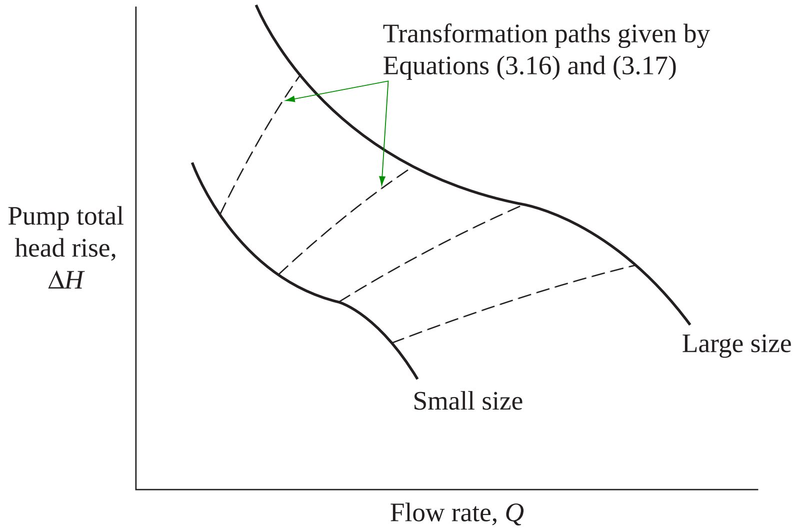
The shaft torque, τ_2, and power, P_2, corresponding to the new size and speed are given by

$$\tau_2 = \tau_1 \left(\frac{D_2}{D_1}\right)^5 \left(\frac{N_2}{N_1}\right)^2 \tag{3.18}$$

and

$$P_2 = P_1 \left(\frac{D_2}{D_1}\right)^5 \left(\frac{N_2}{N_1}\right)^3 \quad . \tag{3.19}$$

Equations (3.16) to (3.19) may be used to map the known characteristic (*e.g.* ΔH_1 versus Q_1) of the pump into a new characteristic corresponding to a new size of pump. This is illustrated in Sketch 3.14 for the special case of constant speed.



Sketch 3.14 Effect of change of size on pump characteristic at constant speed

It should be noted that Equations (3.16) to (3.19) alone may not be used to predict operating points in a given application corresponding to a change in pump size as these depend also upon the system characteristic, see Section 7.5. Furthermore, it is stressed that, because the similarity laws upon which Equations (3.16) to (3.19) are based are applicable only to geometrically similar pumps, they will not predict satisfactorily the effect of changing the impeller size within a given pump, see Section 3.6.

3.6 Impeller Trimming

If a radial or mixed-flow pump has been oversized for its required duty and a reduction in speed is not possible, the pumping efficiency can be kept near maximum by machining down the impeller to give a lower performance. This is preferable to throttling the pump which wastes power.

As a rule, impeller trimming somewhat lowers the pump efficiency and is only suitable for reducing the pump total head rise by, at most, 50 per cent. Typically this would correspond to approximately a 25 per cent reduction of the effective impeller diameter but this value varies according to the impeller type. In all cases therefore where impeller trimming is contemplated, the pump manufacturer should be consulted.

3.7 Practical Limitations

3.7.1 Head rise per stage and specific speed

The maximum total head rise that can be expected from a single-stage pump within a manufacturer's standard range is approximately 200 m at 3000 rev/min. Higher head rise requirements are usually met by multi-stage pumps. It should, however, be noted that extreme deviations do arise; for example, high speed single-stage boiler feed pumps can produce a total head rise of up to 1000 m.

Since the majority of pumps are directly driven by electric induction motors, their maximum speed is governed by the 2-pole motor speed. This is approximately 2900 rev/min for a 50 Hz supply and 3500 rev/min for a 60 Hz supply.

It follows from Equation (3.4) that pumps of a given specific speed produce a combination of best efficiency points. Sketch 3.15, which is derived from Equation (3.4) taking $N = 3000$ rev/min ($\omega = 314$ rad/s) as a representative maximum speed, illustrates the envelope of best-efficiency points that may be obtained from single-inlet, single-stage direct-drive pumps.

The boundaries illustrated in Sketch 3.15 envelop the duty points of the majority of pumps available from manufacturers' standard ranges. They are drawn up from the following practical considerations.

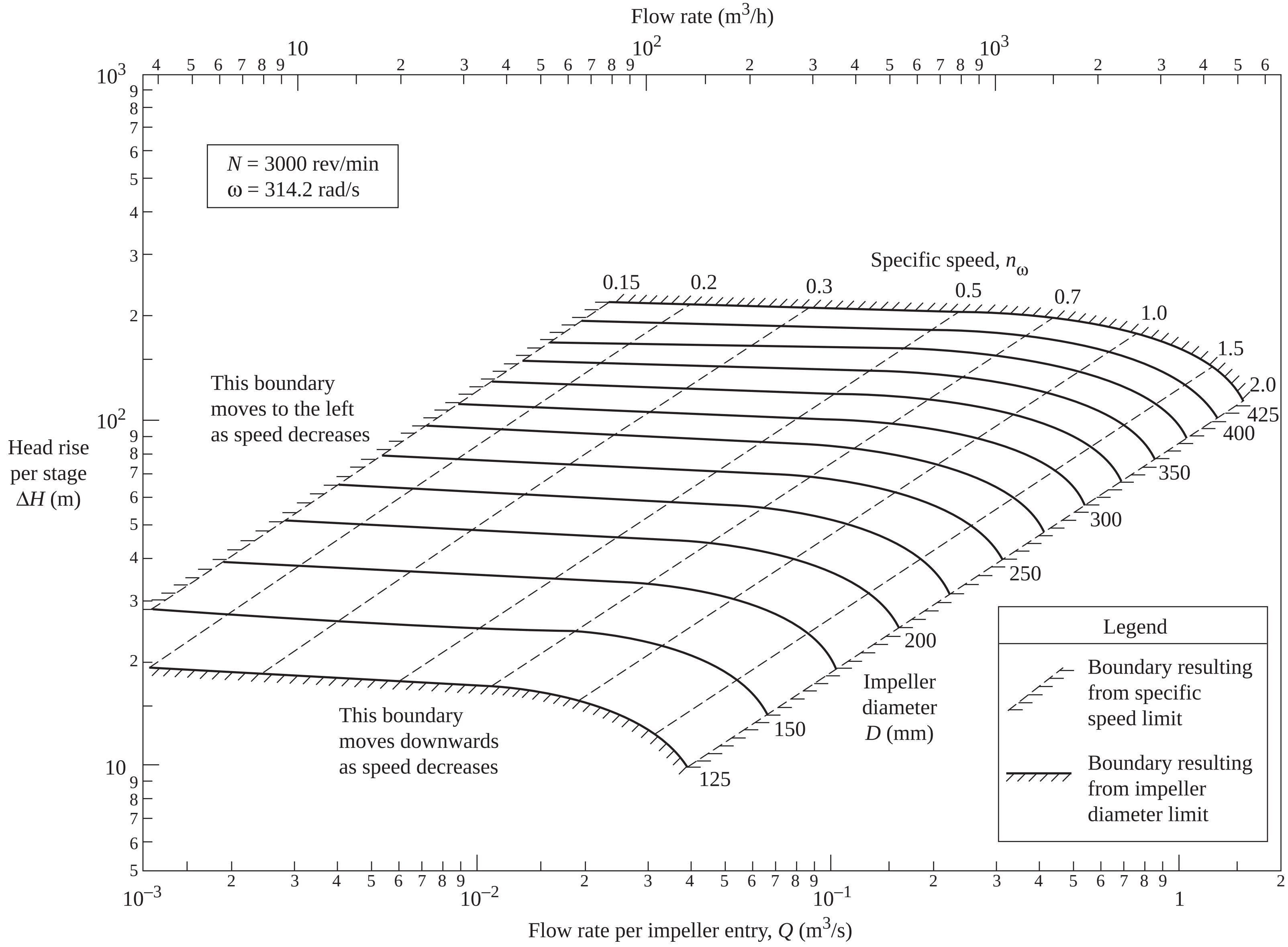
(1) A minimum specific speed, n_ω, of approximately 0.15; impellers, and specially small impellers, of lower specific speeds are difficult to manufacture and inspect because of their narrow blade passages. Note that for pumps running at slower speeds, this boundary will be positioned further to the left. If a duty point requires a specific speed very much less than 0.15 so that it cannot be met by a reasonable number of stages of standard impellers, a positive displacement pump should be considered.

(2) A maximum specific speed, n_ω, of approximately 2; impellers of higher specific speeds are available but, because they are best suited to pumping large flow rates at low total head rises, they are often large* and outside the standard ranges offered by most manufacturers. By using double-suction pumps (see Section 4.1.2) or pumps in parallel (see Section 7.3.2), the duty flow rate shared between each impeller inlet can usually be reduced sufficiently to allow pumps of lower specific speed to be used.

(3) A maximum impeller diameter of approximately 450 mm; this value arises from manufacturing, structural and marketing considerations. Higher head rises can be achieved by multi-stage pumps or by arranging pumps in series, see Section 7.3.1.

(4) A minimum impeller diameter of approximately 125 mm; smaller pumps are produced but are normally intended for the household market. Note that, for pumps running at slower speeds, this boundary will be positioned lower.

* It is not usual to specify high specific speed pumps for low flow rate duties because the $NPSHr$ of small pumps of high specific speed is often impracticably high. Instead, the same duties are met by one or more slower running pumps of lower specific speed.



Sketch 3.15 Envelope of duty points available from standard-range pumps

3.7.2 Maximum and minimum continuous flow rates

Some degree of flow variation through all pumps is inevitable either as a result of changes in system demands or as a result of aging. In addition, it should be recognised that errors in the estimation of system total head rise requirements may result in a pump operating away from its intended duty point, see Section 7.2.

There are five parameters that restrict the operating range of a given pump:

(1) $NPSHa$,

(2) system overpressure,

(3) shaft power required,

(4) hydraulic mismatching within the pump during off-design operation,

(5) temperature rise.

(1) The value of $NPSHr$ for all pumps increases for flow rates above Q_{bep} and for some pumps it increases below Q_{bep}, see Sketch 3.11. The satisfactory operation of all pumps away from Q_{bep} therefore depends on the provision of adequate $NPSHa$.

(2) System overpressure can arise on pumps of high specific speed from reducing the flow rate. The shape of the total head rise characteristic, illustrated in Sketch 3.4, for high specific speed pumps means that, with large pumps operating at a lower-than-design flow rate, a considerably higher-than-design total head will be available at the pump discharge flange. Thus the system must be protected from (or be able to withstand) the overpressure that could result from a decrease in flow rate. Under some circumstances, this may be most severe during start-up, see Section 7.5.2.

(3) Power overloading affects low specific speed pumps at flow rates above Q_{bep} and it also affects high specific speed pumps at flow rates below Q_{bep}, see Sketch 3.5. If operation at flow rates other than Q_{bep} is contemplated, the driving unit and transmission must be specified appropriately.

(4) Hydraulic mismatching with pumps running off design conditions gives rise to vibration and is responsible for increases in $NPSHr$. Typically, hydraulic mismatching problems are not severe on small or medium sizes of pumps provided the flow rates are kept within $0.3 \le Q/Q_{bep} \le 1.25$. However, much narrower limits are often necessary on large machines.

(5) The temperature rise across a pump is caused by the shaft power that is left unconverted into fluid power. Most of the heat is absorbed by the pumped liquid and, at Q_{bep}, the temperature rise is negligible on most pumps except on insulated closed systems. At lower flow rates, however, not only may the rate of heat generation increase because of lower efficiencies but a larger temperature gain will result from a given rate of heat generation. A simple heat balance may be calculated to determine approximate increases in temperature by assuming that all of the heat generated is absorbed by the pumped liquid.

3.7.3 NPSHr and suction specific speed

In general, suction specific speed, S_ω, gives a useful indication of the $NPSHr$ of a pump. A typical value for S_ω is 2.8 but values in excess of 4 can be achieved by well designed impellers. Furthermore, the fitting of an inducer may increase S_{ω} to around 8 but, as mentioned in Section 3.4.3, this may restrict the operating range.

It is inadvisable to design systems with $NPSHa < NPSHr + 0.5$ m even though this may lead to an apparently large margin on low $NPSHr$ pumps*. This is to allow for any dissolved gases which may start coming out of solution above the vapour pressure. Thus, depending on the amount of dissolved gas in the pumped liquid, cavitation may be dominated by dissolved gases in low $NPSHa$ systems.

3.7.4 Maximum liquid viscosity

Rotodynamic pumps are unsatisfactory for handling highly viscous liquids. For a given pump speed, the head rise and flow rate, but predominantly efficiency, all decrease as the liquid viscosity increases. Viscous liquids affect the performance of small pumps more severely than larger units. The effect is quantified in Figure 2 of Item No.80031[21] and where the ratio of the efficiency with the viscous liquid to the efficiency with water as the pumped liquid falls to 0.6, it is recommended that a positive displacement pump be considered. Figure 1 of Item No. 80031 correlates this criterion with the pump duty and liquid kinematic viscosity.

Although under normal running the efficiency penalty may be small, lower temperature operation, *e.g.* during start-up, may need special consideration when specifying the pump driving unit and transmission. If the efficiency ratio falls below 0.6, the pump manufacturer should be consulted to check that the higher shaft torque is acceptable. Some manufacturers are able to supply pumps with heating jackets to assist start-up with viscous liquids.

* Pumps for handling liquids near their boiling points are designed with very low $NPSHr$. The correspondingly low values of $NPSHa$ mean that the energy released during cavitation is also small. In such cases, cavitation is not damaging and the $NPSHa$ can be used to control the throughput of the pump. Further information on this subject may be found in Derivation 40.

4. ASPECTS OF PUMP CONSTRUCTION

4.1 Installation Considerations

4.1.1 Orientation

The choice between vertical and horizontal configurations is influenced by the following considerations.

Head room	–	horizontal pumps occupy less.
Floor space	–	vertical pumps occupy less.
Priming	–	submerged pumps may be preferred when the line empties on shutdown.
Maintenance	–	accessibility is important since pumps and especially their seals and bearings require periodic attention. Horizontal designs are usually preferable.
Locality of liquid	–	if the liquid is in a sump, a submerged pump will normally be selected.
Pipe connections	–	some vertical pumps have in-line inlet (suction) and outlet (discharge) flanges to enable the pump to be inserted directly into a line. In most horizontal designs, the suction and discharge flanges are at right angles.

4.1.2 Common configurations

Most pumps from manufacturers' standard ranges are driven directly by electric motors. These are illustrated in Table 4.1.

TABLE 4.1
Common Configurations of Single-stage Direct-drive Pumps

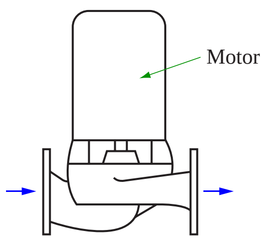
Pump type	Illustration	Comments
Vertical-axis, close-coupled		Motor bearings support impeller, straight through inlet-to-outlet (in-line) design allows simple installation in existing pipelines.

TABLE 4.1 (Continued)
Common Configurations of Single-stage Direct-drive Pumps

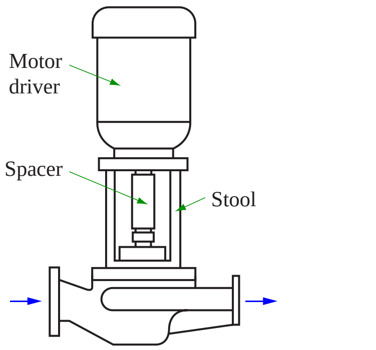
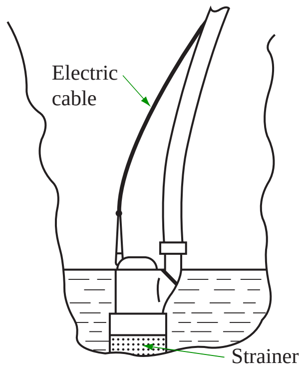
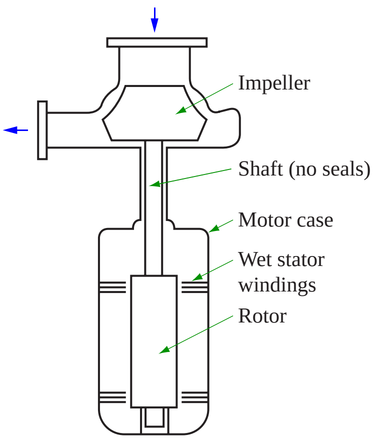
Pump type	Illustration	Comments
Vertical-axis spacer-coupled		Similar to vertical-axis close-coupled pump but spacer aids dismantling without disturbing motor.
Totally-submerged		Available for duties typically requiring less than 50 kW. Motor may run wet, see glandless pump, or more usually be sealed from liquid. Often of low cost design for short but maintenance-free life. Used for draining sumps, cellars, *etc*.
Glandless, wet rotor motor		Used with high temperature, high pressure, flammable or toxic liquids that must nevertheless be dirt free. Motor runs immersed in pumped liquid so no external rotary seals are required. Available for both vertical and horizontal mounting. Note: two different types of motors, canned and wet winding, are produced for glandless pumps. In the canned motor, the stator windings are kept dry from the pumped liquid by a ‘can’.

TABLE 4.1 (Continued)
Common Configurations of Single-stage Direct-drive Pumps

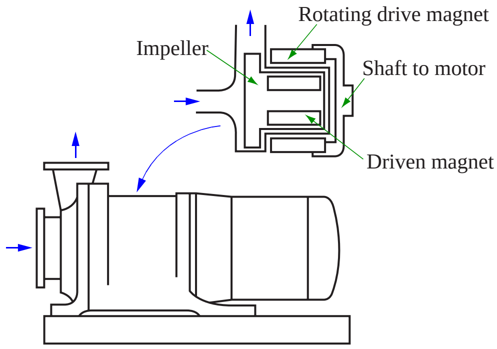
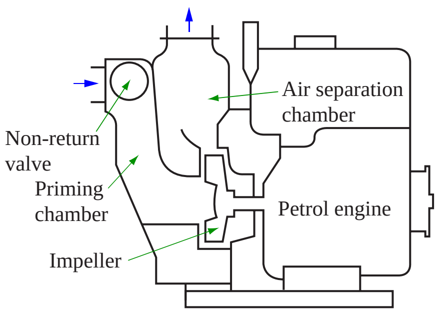
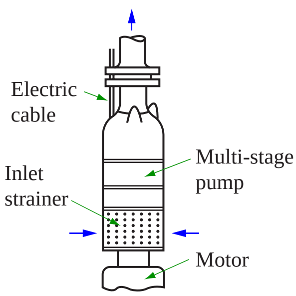
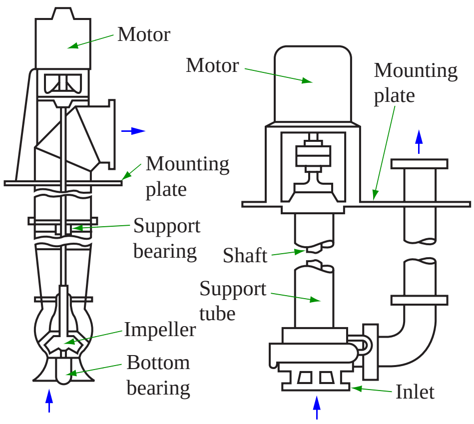
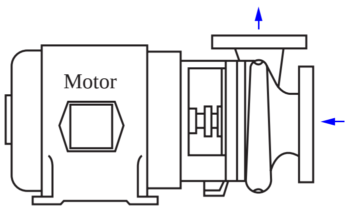
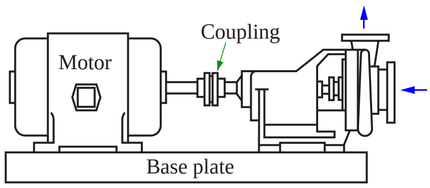
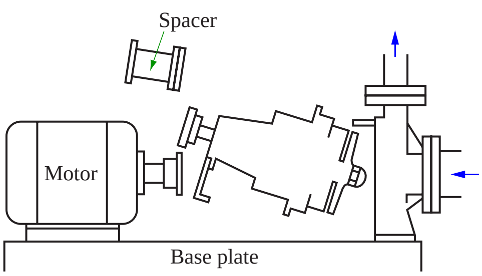
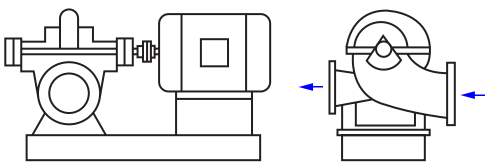
<table>
<tr><th>Pump type</th><th>Illustration</th><th>Comments</th></tr>
<tr><td>Glandless, magnetic drive</td><td>
</td><td>Used in similar, but lower pressure, applications to wet rotor motor glandless pumps. The need for shaft seals is obviated by the magnetic coupling which transmits power across the pump inner casing.</td></tr>
<tr><td>Self-priming pump</td><td>
</td><td>Type illustrated is a portable contractors pump with petrol engine. Check valve maintains liquid in casing when pump is stopped.</td></tr>
<tr><td>Bore-hole pump</td><td>
</td><td>Normally of small diameter to allow insertion down relatively narrow boreholes. When provided with a suitable suction vessel, can also be installed in a horizontal pipeline for boosting.</td></tr>
</table>

TABLE 4.1 (Continued)
Common Configurations of Single-stage Direct-drive Pumps

Pump type	Illustration	Comments
Top mounted submerged		Submerged casing ensures impeller priming making this type suitable for stand-by duties. Available in large sizes. Shaft is supported at intervals by guide bearings. Impeller bearings usually lubricated by pumped liquid to obviate sealing problems.
Horizontal-axis closed-coupled		More compact and lower cost version of horizontal spacer coupled pump. Motor bearings support impeller.
Horizontal-axis direct-coupled		Very common arrangement, available in powers up to 200 kW. Often referred to as horizontal end-suction pump.
Horizontal-axis, spacer-coupled		Available in powers up to 200 kW. Spacer enables impeller to be withdrawn without disturbing motor. Often referred to as a horizontal end-suction pump or back pull-out pump (see Table 4.2).
Double-suction		Symmetrical double inlet discharges into common volute. Popular design for larger flow rates.

4.2 Casing Design

Most small to medium sizes of radial and mixed-flow pumps have casings termed “radially-split”. Such casings are made up of two sections that are separable either side of a plane normal to the shaft axis. Larger sizes of pumps are sometimes constructed with “axially-split” casings. Here the casing separation plane coincides with the shaft axis.

Multi-stage pumps are subject to the same principles of axial and radially-split design as single-stage machines and are often constructed from a succession of identical single-inlet stages. It is nevertheless quite common for the first stage to be constructed with a double-inlet impeller in order to lower $NPSHr$. Another common variant employs an axi-symmetrical arrangement of impellers in order to balance the axial thrust that arises from the difference in average pressures on either side of single-inlet impellers.

The following tables illustrate some of the many different designs available.

TABLE 4.2
Radially-split Casings

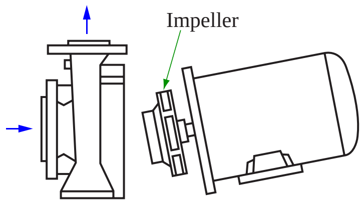
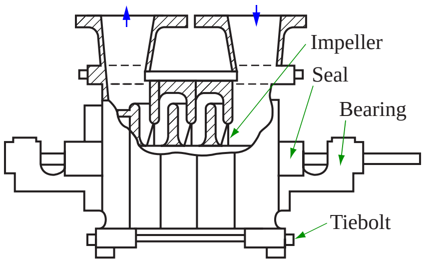
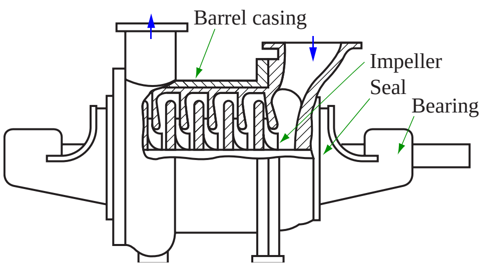
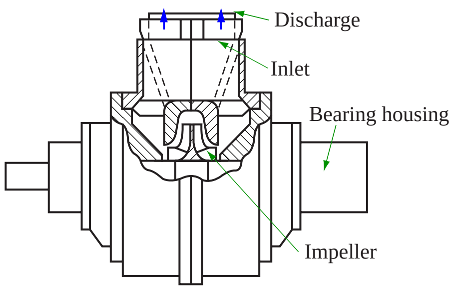
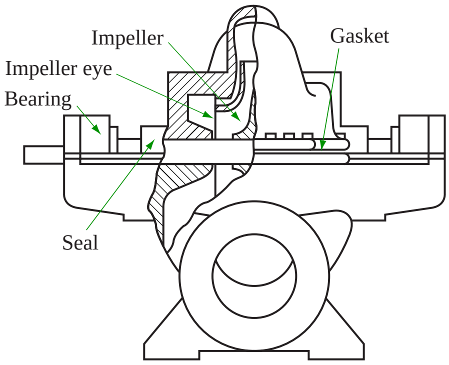
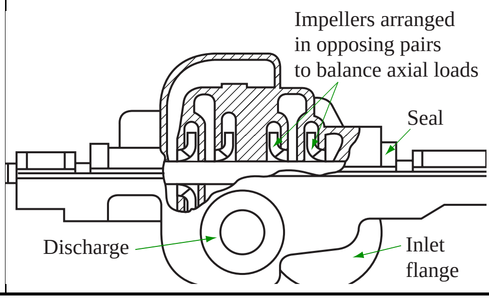
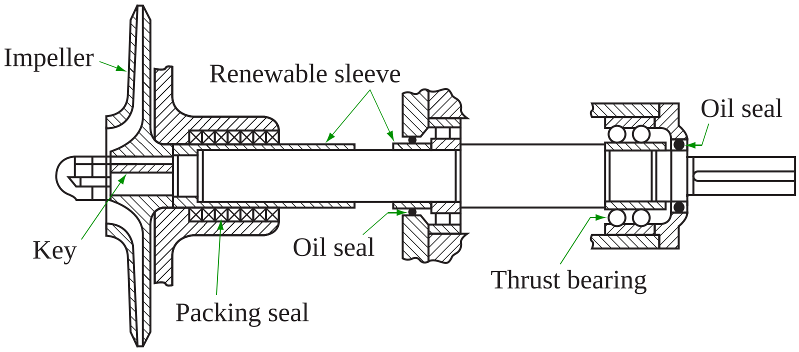
Pump type	Illustration	Comments
Back-pullout		Easy access to impeller assembly. Impeller can be removed without disturbing pipework if spacer coupling is fitted.
Tie-bolt multi-stage		Good pressure tightness due to well-distributed sealing loads. Stages often of modular design. Pipework must be disconnected before dismantling.
Barrel, multi-stage		Essentially a tie-bolt design with an extra outer casing that enables pipe work to remain undisturbed during pump servicing. Its inherent pressure tightness and convenience of servicing makes this a very popular design.
Double-suction pump. Note “double-suction” describes the number of impeller inlets. The casing has only one inlet.		Well suited to relatively high flow rates with moderate head rises. Less easy to service than corresponding axially split design but more satisfactory for high pressure duties.

TABLE 4.3 – Axially-split Casings

Pump type	Illustration	Comments
Double-suction pump. Note "double-suction" describes the number of impeller inlets. The casing has only one inlet.		Popular single-stage pump for relatively high flow rates. Axially split casing enables simple impeller and shaft removal. Less satisfactory than radially-split design for high pressure duties.
Multi-stage pump		Convenient impeller and shaft removal, pipework need not be disturbed. Although easier to manufacture, pressure tightness is inherently less satisfactory than with radially-split designs.

4.3 Impeller Shaft Design

Depending on the temperature, pressure and properties of the pumped liquid, pumps are designed with different bearing configurations. In most cases, because the absence of all dirt cannot be guaranteed*, bearings are isolated by the seal from the pumped liquid. For single-stage pumps, the most common configuration is the overhung impeller, see Sketch 4.1.



Sketch 4.1 Overhung impeller arrangement

* When the liquid is clean and has satisfactory lubrication properties, internal bearings may be used, for example, as in a glandless pump.

4.4 Shaft Sealing Arrangements

There are two widely-used alternative types of shaft seals: soft packing seals and mechanical seals. Whilst soft packing is inexpensive and is normally supplied by manufacturers unless an alternative is specified, mechanical seals are increasingly used where clean liquids are involved because of their very low leakage, low torque and low maintenance requirements. Both types of seals need to be protected from grit; this is normally achieved by flushing the seal housing (or stuffing box) with clean liquid.

For toxic or flammable liquids where leakage is unacceptable or for liquids at high temperatures and pressures where dynamic sealing is very difficult, glandless pumps may be specified, see Table 4.1.

5. STANDARDS FOR PUMPS

Standards are available for defining aspects of the performance of pumps and for defining their installation geometry.

5.1 Performance

Two tables are presented to compare the features offered by those standards concerned with performance. The first illustrates the datum elevations defined by the different standards and the second illustrates the performance guarantee tolerances set by the standards.

TABLE 5.1
Datum Elevation Definitions
(applies to first stage impeller in all cases)

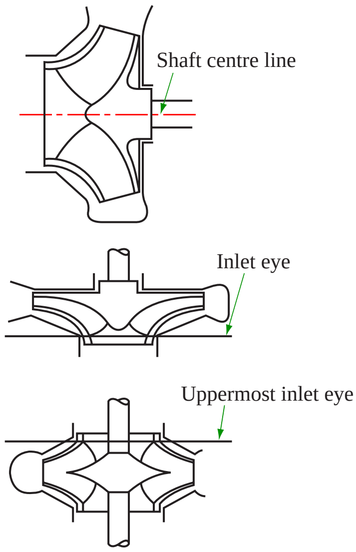
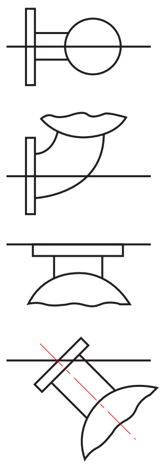
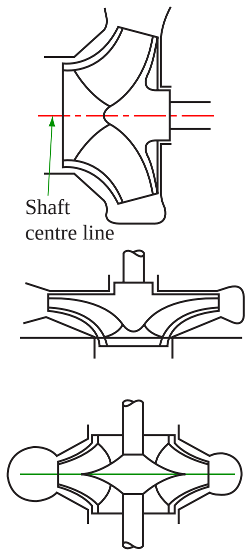
Standards	ISO 2548[6] BS 5316 Pt 1[11]	ISO 3555[14] BS 5316 Pt 2[15]	API 610[5] DIN 1944[3]	ASME PTC 8.2[1]
Comments			Relates to inlet flange centre line	
Definition				
The superscripted characters, 6, 11, *etc.*, refer to references in Section 8.				

TABLE 5.2
Performance Guarantee Tolerances
(percentage variations unless otherwise indicated, *i.e.* $Y_{actual} = Y_{quoted}(1 + X/100)$,
where X is the tolerance in per cent and Y is the pump performance variable)

Standard	ISO 3555[14] BS 5316 Pt 2[15]	ISO 2548[6] BS 5316 Pt 1[11]	API 610[5]
Comments	Class B	Class C	General
Flow rate, Q	± 5.66 *	± 9.9 *	± 0 †
Total head rise, ΔH	± 2.83 *	± 5.66 *	$\left.\begin{matrix}+5\\-2\end{matrix}\right\\}\ 0 < \Delta H \leq 152$ m $\left.\begin{matrix}+3\\-2\end{matrix}\right\\}\ 152 < \Delta H \leq 305$ m $\left.\begin{matrix}+2\\-2\end{matrix}\right\\}\ 305 < \Delta H$ m
Net positive suction head required‡ $NPSHr$	Guarantee $NPSHr$ must sustain Q and ΔH guarantees		+ 0 No specified lower limit
Efficiency, shaft to liquid, η	– 2.8 no specified upper limit	– 5 no specified upper limit	– 5 point** no specified upper limit
Efficiency, electricity to liquid	– 2.5 no specified upper limit	– 4.5 no specified upper limit	–
Power, shaft, P	+ 0 no specified lower limit within range defined by flow rate and head rise tolerances		+ 4 no specified lower limit
Power, electrical	as above	as above	–
The superscripted characters, 14, 15, *etc.*, refer to references in Section 8.			

* The ISO and BS codes are concerned with the proximity of the pump characteristic curve to the guarantee point. As the proximity depends on combinations of tolerance for Q and ΔH, the standards should be referred to for a full explanation. The values quoted are for a typical, but single, combination.

† The code API 610 relies on the interdependence of Q and ΔH. In API 610 a tolerance on ΔH only is given for the assumed flow rate.

‡ Guaranteed, as opposed to catalogue, values of $NPSHr$ correspond to point L rather than point M in Sketch 3.8.

** For example, a quoted efficiency of 79 per cent with -0.5 point tolerance means 78.5 per cent minimum allowable.

TABLE 5.2 Continued
PERFORMANCE GUARANTEE TOLERANCES
(percentage variations unless otherwise indicated, *i.e.* $Y_{actual} = Y_{quoted}(1 + X/100)$
where X is the tolerance in per cent and Y is the pump performance variable)

Standard	ASME PTC 8.2[1]	DIN 1944[3]		
Comments	General	Class I	Class II	Class III
Flow rate, Q	±2 *	+ 5 † − 5 †	+ 10 † − 5 †	+ 15 † − 5 †
Total head rise, ΔH	±2 *	+ 1 † − 1 †	+ 2 † − 1 †	+ 3 † − 1 †
Net positive suction head required‡ $NPSHr$	–	Guarantee $NPSHr$ must sustain Q and ΔH guarantees		
Efficiency, shaft to liquid, η	±1 (test reading fluctuation)	– 0 no specified upper limit	– 0 no specified upper limit	Quoted efficiencies are not guaranteed
Efficiency, electricity to liquid	–	– 0 no specified upper limit		Quoted efficiencies are not guaranteed
Power, shaft, P	–	Quoted efficiency must be reached or bettered within range defined by flow rate or head rise tolerances		+ 0 no specified lower limit
Power, electrical	–	–	–	–
The superscripted characters, 14,15, *etc.*, refer to references in Section 8.				

* The code PTC 8.2 relies on the interdependence of Q and ΔH. The tolerance on Q or ΔH but *not both* may be exploited.

† DIN 1944 discriminates between pumps with steep and flat characteristics at the guarantee point. For pumps with a steep characteristic, $|(Q/\Delta H)\ (\mathrm{d}H/\mathrm{d}Q)| > 0.2$, the tolerance on the guaranteed flow rate is required at a guaranteed head rise. For pumps with a flat characteristic, $|(Q/\Delta H)\ (\mathrm{d}H/\mathrm{d}Q)| \leq 0.2$, the tolerance on the guaranteed head rise is required at a guaranteed flow rate.

‡ Guaranteed, as opposed to catalogue, values of $NPSHr$ correspond to point L rather than point M in Sketch 3.8.

5.2 Dimensional Standards

Several standards are in existence for specifying the principal dimensions of pumps, usually in terms of a nominal duty point and a pressure rating. This promotes interchangeability of pumps and their components, such as seals and bearings, and it also enables installation details, *e.g.* pipe layouts, to be anticipated.

The standards are concerned with mass produced pumps and thus cover to date only the horizontal end-suction and the vertical in-line types. The more common standards are given in Table 5.3.

TABLE 5.3
Dimensional Standards

Horizontal end-suction pumps	Vertical in-line pumps
ISO 2858[8]	BS 4082 Pt.1[4]
ISO 3069[7]	BS 4082 Pt.2[4]
ISO 3661[12]	ANSI B73.2[9]
BS 5257[10]	
DIN 24255[17]	
DIN 24256[18]	
ANSI B73.1[16]	
The superscripted characters, 8, 9, *etc.*, refer to references in Section 8.	

6. SYSTEM CHARACTERISTICS

Before choosing a pump to satisfy the normal operating mode of a system, it is advisable to explore the possible extremes of operation. Operation away from the normal mode inevitably occurs because of, for example,

(1) changes in temperature affecting viscosity and hence friction losses,

(2) changes in load, *e.g.* items of plant being brought in and out of service affecting the flow rate,

(3) aging, *e.g.* scaling (or erosion) increasing (or decreasing) the friction loss,

(4) changes in level, or pressure, between supply and discharge vessels.

The total pressure rise (or total head rise) and flow rate requirements of the system corresponding to the mode of operation most exacting to the pump should be determined and, in most cases*, used for sizing and specifying the pump. In general, the most exacting operation for a pump in terms of its duty point is that corresponding to the highest flow rate with the largest probable total pressure or total head rise requirement, *e.g.* a cold system running at full load with scale-encrusted pipe emptying nearly-empty vessels into nearly-full vessels. Having sized a pump to satisfy such a duty point, however, a more exacting *NPSHa* condition will result from the higher rates that would be achieved using the same pump against the minimum probable total pressure or total head rise requirement of the system. Further guidance on this is given in Section 7.2.

The total pressure or head rise requirement of a system may be calculated by taking into account the following:

(1) the frictional resistance to flow through the system, for which data are available in Reference 19,

(2) the change in elevation between the inlet and outlet of the system,

(3) the difference in pressure between the inlet and outlet of the system.

System calculations may be undertaken either in terms of total pressure or total head. Total pressure calculations are advisable for high pressure systems such as steam generating plant where there may be significant changes in density, while total head calculations are more usual for open or low pressure systems. Total head methods, however, are inadvisable for flows with significant density changes. Both approaches are illustrated diagrammatically in Sketch 6.1.

The procedure undertaken in constructing the total pressure profile in Sketch 6.1 is represented algebraically by Equation (6.1),

$$p_a + \cancelto{0}{\tfrac{1}{2}V_1^2} + \rho g z_1 - (p_1 - p_2) - (p_2 - p_3) - (p_3 - p_4) + \Delta p - (p_5 - p_6) \ldots\ldots - p_{10} = 0 \quad (6.1)$$

where the terms in brackets, $(p_n - p_{n+1})$, are the total pressure losses due to friction exhibited by each section in the system. The first three terms in Equation (6.1) represent the total pressure p_1 at the surface

* On large systems, where the energy involved means that efficiency is at a premium, a pump may be specified to fulfil the normal operating mode only. Auxiliary pumps connected in parallel or series, see Section 7.3, may be used to cover the extremes of operation.

of the supply reservoir where the kinetic component, $\frac{1}{2}\rho V_1^2$ is negligible.

The term Δp in Equation (6.1) could be rewritten using the relationship of Equation (3.1), *e.g.*

$$\Delta p = \Delta H \rho g, \tag{6.2}$$

which, when rearranged in Equation (6.1), gives

$$\Delta H = \left[p_{10} + (p_9 - p_{10}) + (p_8 - p_9) + \ldots + (p_1 - p_2) - \left(p_a + \overset{0}{\cancel{\tfrac{1}{2}\rho V_1^2}} + \rho g z_1 \right) \right], \tag{6.3}$$

where the pump total head rise, ΔH, is that necessary to overcome the total head requirement of the system.

Alternatively, the total head profile in Sketch 6.1 may be represented by

$$H_1 - (H_1 - H_2) - (H_2 - H_3) - (H_3 - H_4) + \Delta H - (H_5 - H_6) \ldots . - H_{10} = 0, \tag{6.4}$$

where

$$H_1 = \overset{0}{\cancel{V_1^2/(2g)}} + z_1,$$

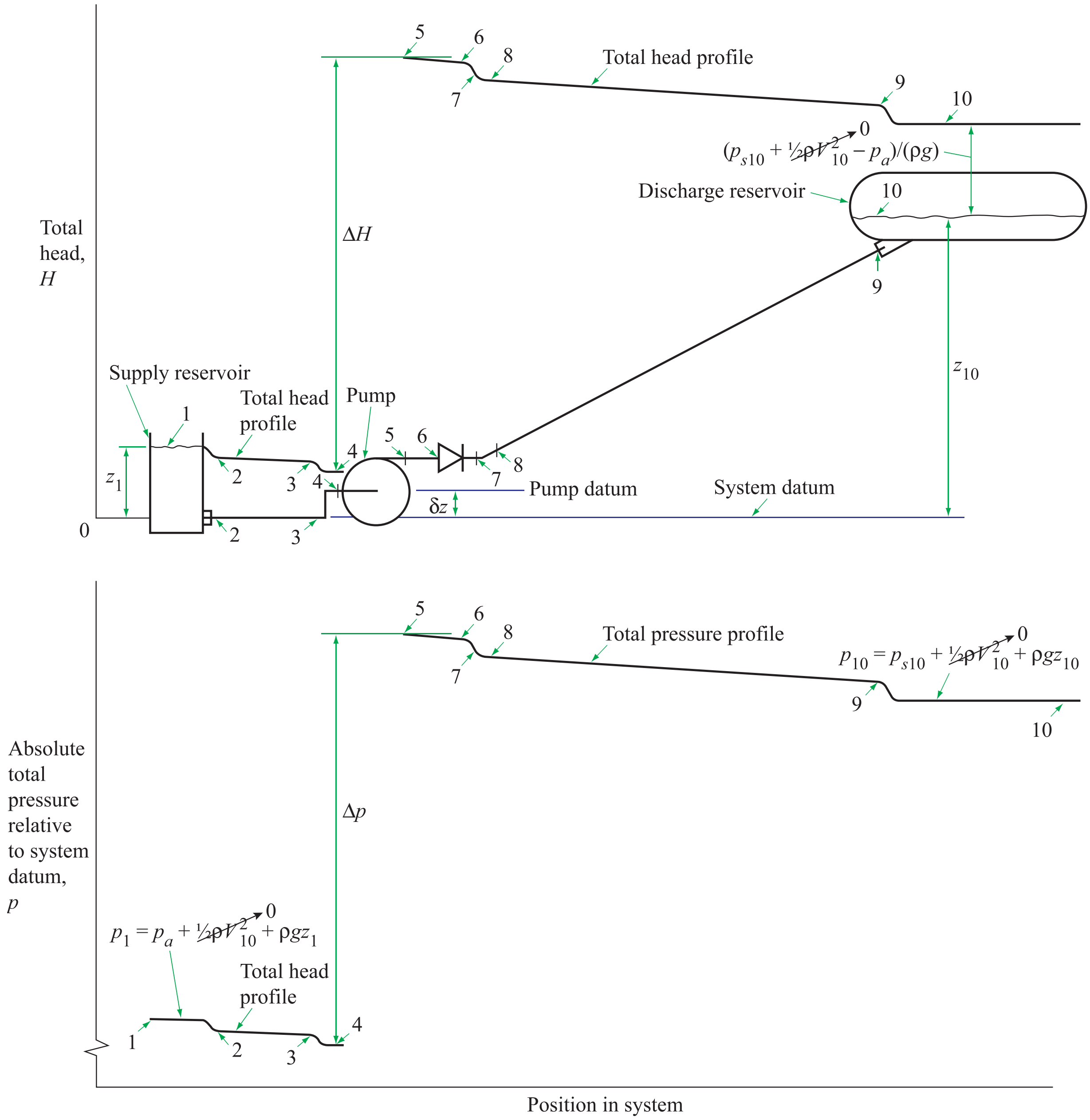
$$H_{10} = (p_{s10} - p_a)/(\rho g) + \overset{0}{\cancel{V_{10}^2/(2g)}} + z_{10}$$

and the terms in brackets, $(H_n - H_{n+1})$, are the total head losses due to friction exhibited by each section in the system. Again, the kinetic components, $V_1^2/(2g)$ and $V_{10}^2/(2g)$, of H_1 and H_{10} respectively are negligible.

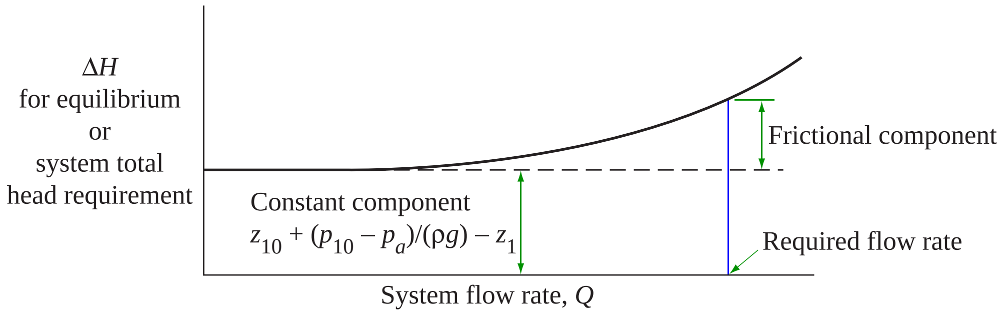
Equation (6.4) may be rearranged to give ΔH, the pump total head rise, *i.e.*

$$\Delta H = H_{10} + (H_9 - H_{10}) + (H_8 - H_9) + \ldots . + (H_1 - H_2) - H_1 . \tag{6.5}$$

Since the losses in the system depend on the flow rate, it is clear that ΔH, the total head rise required for equilibrium, is also a function of flow rate. It may thus be plotted against the flow rate to form a system total head requirement characteristic as illustrated in Sketch 6.2.



Sketch 6.1 Simple system illustrating total head and total pressure variations



Sketch 6.2 System total head requirement

In addition to determining the system total head loss, it is necessary to determine the $NPSHa$ at the pump inlet to ensure that this exceeds the pump $NPSHr$ by a satisfactory margin*. Note that it is essential that $NPSHa$ and $NPSHr$ are referenced to the same vertical datum for this purpose, *e.g.* the pump datum† since this is usual for $NPSHr$.

For calculating $NPSHa_i$, a value of p_i, the absolute total pressure at the pump inlet referred to the pump datum, is required. For the system in Sketch 6.1, this is given by

$$p_i = p_4 - \delta z \rho g, \tag{6.6}$$

where p_4 is obtained from the terms upstream of the pump, *i.e.*

$$p_4 = p_1 - (p_1 - p_2) - (p_2 - p_3) - (p_3 - p_4) \tag{6.7}$$

and where δz is the head adjustment allowing for the difference in elevation between the system datum (to which p_4 is referred) and the pump datum. Note δz is positive for a pump datum above a system datum.

The value for $NPSHa_i$ is then obtained from Equation (6.8) (*c.f.* Equation (3.5)),

$$NPSHa_i = (p_i - p_v)/(\rho g)\,. \tag{6.8}$$

Alternatively, the calculations may be expressed in terms of total head, *i.e.*

$$H_i = H_4 - \delta z\,, \tag{6.9}$$

where

$$H_4 = H_1 - (H_1 - H_2) - (H_2 - H_3) - (H_3 - H_4)\,. \tag{6.10}$$

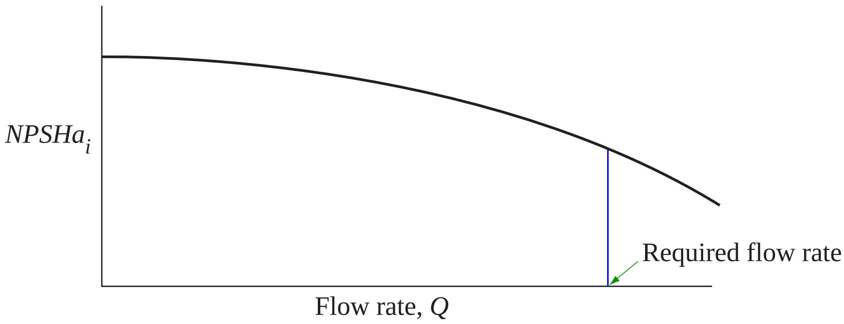
The value for $NPSHa_i$ is then given by Equation (6.11) (*c.f.* Equation (3.6)),

$$NPSHa_i = H_i + (p_a - p_v)/(\rho g)\,. \tag{6.11}$$

* Although the notes in this section refer specifically to the calculation of $NPSHa$ at the pump inlet, the same approach may be taken for other components in the system, *e g.* flowmeters, that have a minimum $NPSHa$ requirement.

† Section 5.1 details the pump vertical datum positions adopted by several international standards. However any position may be used provided it is clearly defined.

Since p_i (or H_i) depends on the system flow rate, $NPSHa_i$ can also be represented as a system characteristic as shown in Sketch 6.3.



Sketch 6.3 Characteristic for $NPSHa_i$ at the pump inlet

As explained in Section 3.4.2, for satisfactory pump operation it is advisable to ensure that

$$NPSHa_i \geq NPSHr + 0.5 \text{ m.}$$

7. PUMP AND SYSTEM MATCHING

For satisfactory operation, pumps and systems should be matched to fulfil the following criteria:

(a) stable operation, see Sections 7.1 to 7.4,

(b) satisfactory starting, see Section 7.5,

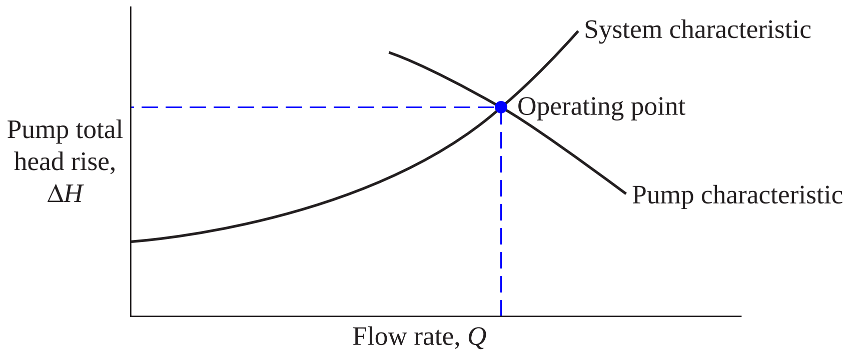
(c) fail safe, see Section 7.6,

(d) no water hammer, see Section 7.7,

(e) controllability, see Section 7.8.

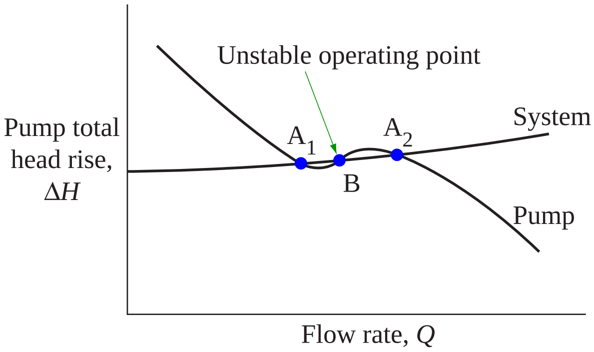
7.1 Stable Operation

A state of equilibrium between a pump and a system exists when the flow rate is such that the total head rise produced by the pump is equal to that necessary to maintain the system flow rate, *c.f.* Equation (6.3) or (6.5). This point of equilibrium is called the "operating point" and it exists at the intersection of the pump and system total head characteristic curves, see Sketch 7.1.

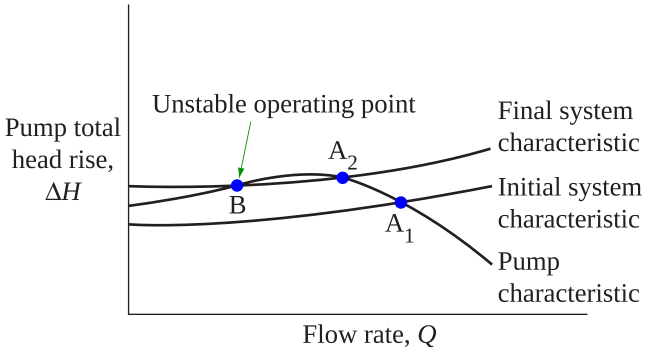


Sketch 7.1 Definition of operating point

The operating point in Sketch 7.1 is stable; if the flow were to fluctuate slightly, the relative slopes of the system and pump characteristics are such that the resulting imbalance would restore the flow rate to its equilibrium value. Sketch 7.2 shows two examples of unstable equilibrium at which the pump/system cannot operate even though the pump can supply and the system requires identical total head changes at a given flow rate.



(a) Instability from characteristic of high specific speed pump



(b) Instability from changing system characteristic

(a) Instability from characteristic of high specific speed pump

(b) Instability from changing system characteristic

Sketch 7.2 Examples of unstable equilibria

If, for the situations shown in Sketches 7.2, the flow rates were to deviate slightly from those at the unstable operating points (marked B), the slopes of the system and pump characteristics are such that the unbalanced heads would compound the deviations until the flow rates become established at one of the stable operating points (marked A).

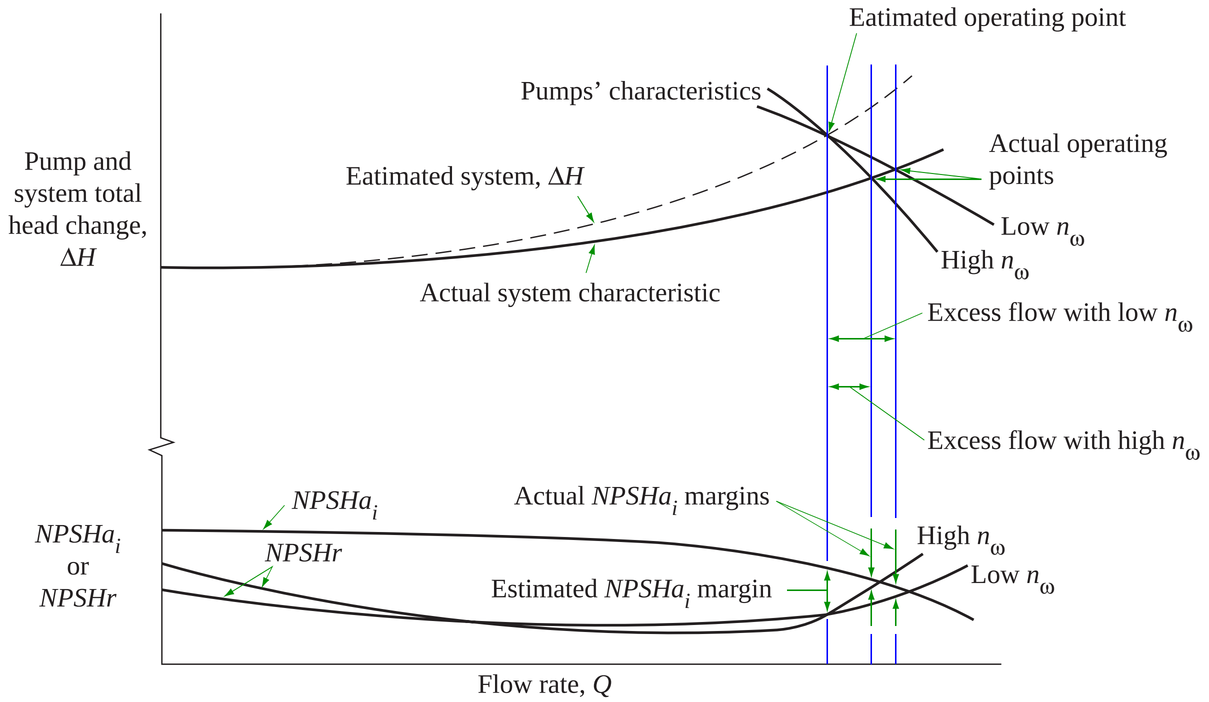
For the system shown in Sketch 7.2b, the initial system characteristic is lower than the final characteristic. This is a common property of systems that involve the filling of vessels which impose an increasing pressure on the system as they fill.

7.2 Margins for Error in Pump and System Head Loss Characteristics

Sketch 7.3 illustrates that flow rate is less susceptible to errors in the estimation of the system characteristic if the operating point is situated on a steep pump characteristic. It also shows that, although it is common practice to oversize a pump with the intention of gaining an adequate performance margin, unless the $NPSHa$ provisions have a corresponding margin, cavitation will occur. Such cases may be remedied by throttling the flow downstream of the pump, *e.g.* with an orifice plate, thus restoring the operating point to the pump best efficiency point, but see Section 7.8.

It should be recognised that pump characteristics are also subject to uncertainties due to manufacturing tolerances* and differences between the installation and the test set up. The pump and system uncertainties together define a region of probable operating points but, again, a smaller tolerance on the flow rate, Q, results from pumps with steep characteristics than from those with shallow characteristics.

* The uncertainty of a pump characteristic will be indicated by the standard to which the pump complies, see Section 5. Usually, however, this uncertainty is much less than the uncertainties of system characteristics.



Sketch 7.3 Influence of pump characteristics on flow rate margins

7.3 Multi-Pump Arrangements

There are reasons why it may be preferable to specify more than one pump to fulfil the requirements of a system, *e.g.*

(1) earlier delivery and often lower cost of several small standard pumps than a single large unit,

(2) more convenient installation of several small pumps than a single large unit,

(3) expansion of an existing system already containing a pump,

(4) lower $NPSHr$,

(5) pumps positioned at several stations along a pipeline to avoid excessive pressure build up,

(6) flow regulation by switching pumps in and out of service,

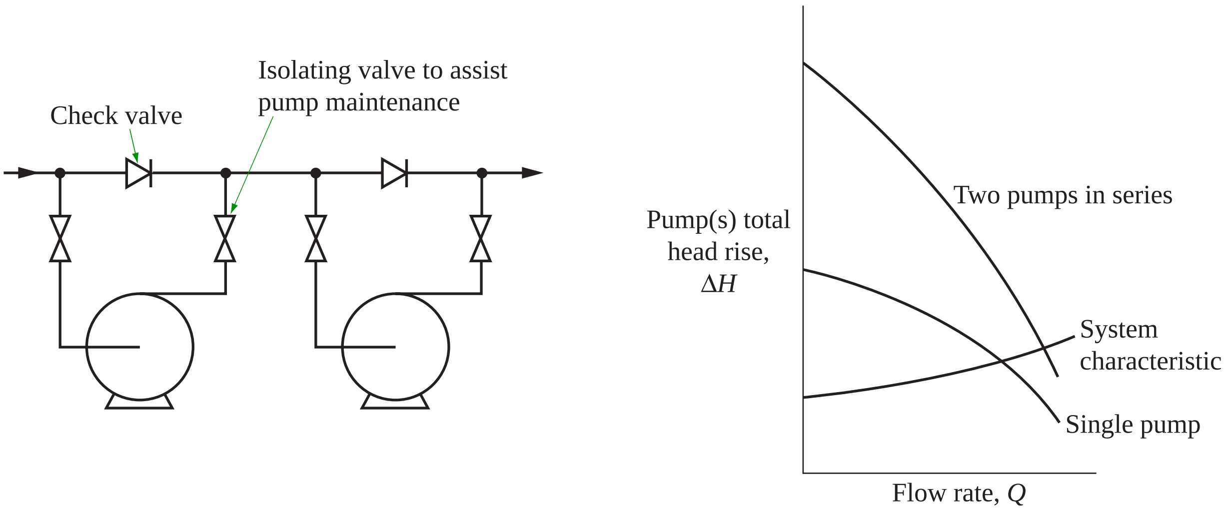
(7) applications where reserve pumps are necessary.

As examples of how combined characteristics can be calculated, Section 7.3.1 covers the case of two pumps operating in series and Section 7.3.2 covers two pumps operating in parallel. Note that in multi-pump applications where it is essential to maintain the flow in the event of a pump failure, each pump is sized with a higher-than-normal margin.

7.3.1 Two pumps in series

Pumps in series offer an increase in total head above that obtainable from a single pump. In general, however, a multi-stage pump would normally be specified for a high total head rise duty but there are circumstances where the system head rise requirement varies so that the flexibility offered by separate pumps connected in series is required.

Pumps in series all pass the same flow rate assuming no gains or losses in the system and the combined total head rise, ΔH, will approach the sum of the individual pump rises. Thus the combined total head rise versus flow rate characteristic for a number of pumps in series may be constructed from the individual characteristics by summing, for a series of values of Q, the values of total head rise from each of the pumps. This is illustrated in Sketch 7.4 for the case of two pumps.

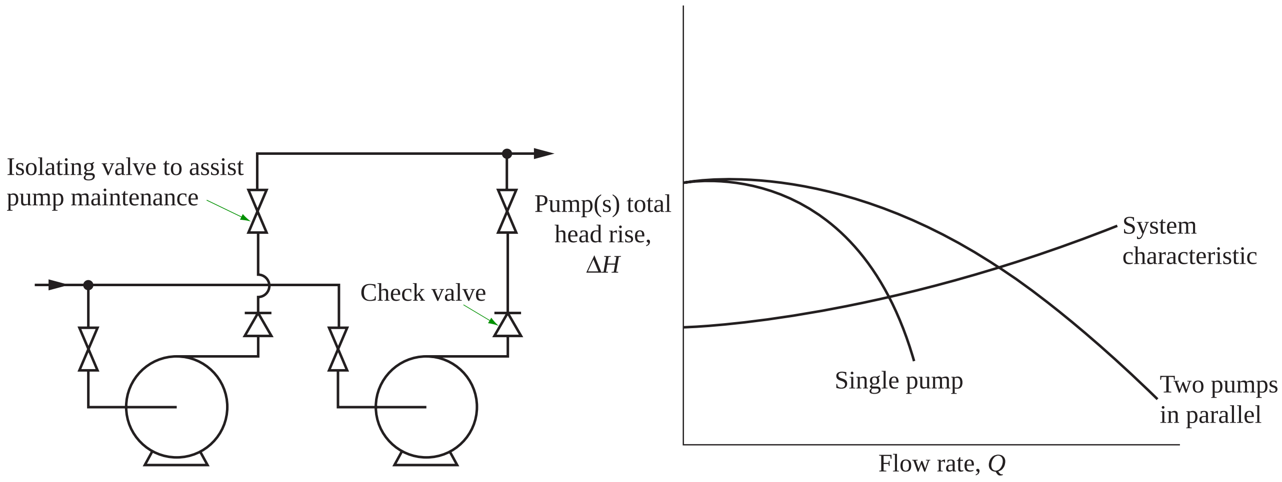


Sketch 7.4 Two pumps in series

Note that the downstream pump and its seals must be suitably constructed to withstand the pressure imposed upon it by the upstream pump. Furthermore, if the more downstream of the two pumps is ever operated singly, as perhaps in a stand-by mode, the resistance to flow through the inoperative portions of the system should be considered when calculating the $NPSHa$.

7.3.2 Two pumps in parallel

The combined characteristic of pumps in parallel is determined by adding all the possible combinations of flow rate, Q, that exist for a given total head rise, ΔH. Sketch 7.5 illustrates this procedure for two identical pumps.



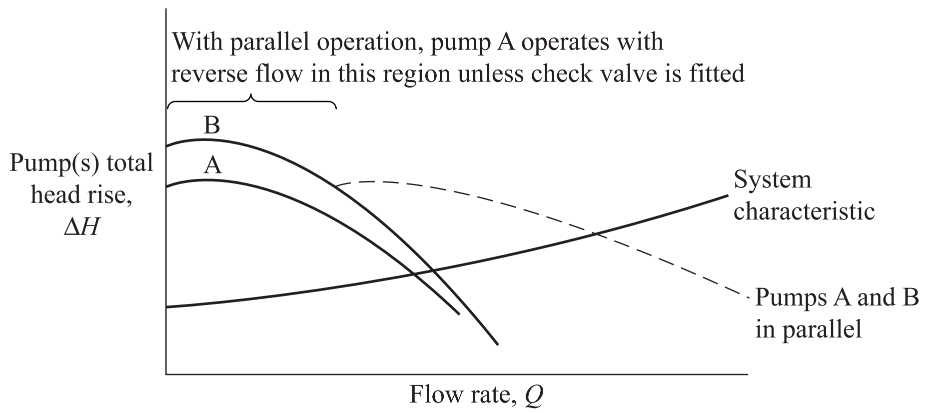
Sketch 7.5 Operation of two identical pumps in parallel

Pumps arranged for parallel operation should be fitted with check valves to avoid backflow which can damage a pump when the pumps are started or when a pump fails.

Note that in the event of one pump failing, the flow rate through the pump remaining in service will increase and if continuous operation is intended, appropriate $NPSHa$ provisions should be made otherwise a deficit between $NPSHa$ and $NPSHr$ may arise.

Problems of power overloading may also arise if pumps with an overloading power characteristic (low specific speed) are employed. In such cases the driving unit and its system must be protected from, or be able to withstand, the increased power absorption.

If two dissimilar pumps are arranged for parallel operation, care must be taken to ensure that the head rise applied by the running pump is not too great to prevent the other pump maintaining its normal operating condition. This imposes a limit on the differences in pump sizes that can be accommodated in parallel operation. Sketch 7.6 illustrates a characteristic of two dissimilar pumps arranged in parallel.



Sketch 7.6 Operation of two dissimilar pumps in parallel

7.4 Pump Siting

In general, the pump should be placed as near to the supply as practicable in order to maximise the pump $NPSHa$. If insufficient, the $NPSHa$ may be increased by increasing the pump inlet pressure, *e.g.* by raising the level of the supply, pressurising the supply or by inserting the pump in a pit sunk to a depth sufficient for an adequate $NPSHa$. Some pump manufacturers supply a specially designed chamber for siting underground in which the pump is already installed.

Adequate submergence of large pump inlets and correct pump inlet chamber design are necessary in order to avoid vortices[13]. Inlets should also be carefully designed to provide a uniform flow distribution. This is especially important for large pumps and those of high specific speed. These factors should be discussed with the pump supplier and may require model studies.

7.5 Priming and Starting

7.5.1 Priming

Most rotodynamic pumps are not self-priming since they develop their head rise by imparting kinetic energy to the flow. The kinetic energy is directly related to the fluid density and, due to the low density of air (or vapour), the pressure differential created by an unfilled (or unprimed) pump is very small.

Priming accessories* to overcome this problem on rotodynamic pumps that empty when shut down fall into three categories.

(1) Foot valve. This is a non-return valve fitted to the system inlet that allows the pump to be primed by filling the system with liquid from any available source. However, foot valves cause a pressure drop that reduces the pump $NPSHa$ and are susceptible to leakage and clogging.

(2) Vacuum priming requires a valve in the pump outlet line to be closed. The liquid is then drawn into the pump by evacuating air from the highest point in the pump. The disadvantage of this system is that it is complicated and requires special devices to prevent the liquid from being ingested into the vacuum line.

(3) Priming chamber. This is a tank that holds a large enough volume of liquid to keep the pump filled until pumping is sustained. A priming chamber can be integral with the pump casing to form a self-priming pump, see Table 4.1.

Because there are practical problems associated with pump priming, pumps should ideally be positioned in the system in such a way that they remain flooded upon shut down. For example, the time taken for priming usually means that flooded pumps are a prerequisite for stand-by duties. In addition, priming becomes more difficult (and operation less efficient) as seals wear and allow air to enter the pump casing.

7.5.2 Starting

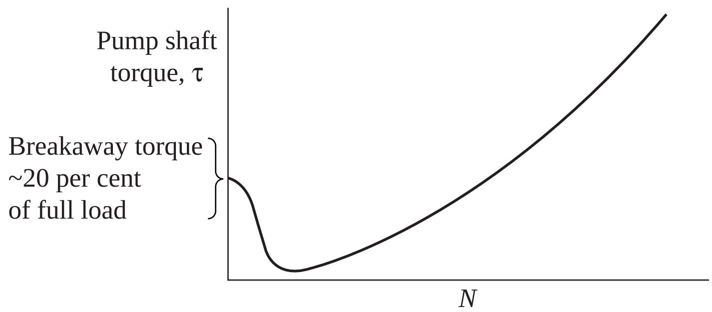
Starting problems, whilst unusual, can occur for both the system and the pump driving unit. System problems can be caused, for example, by the selection of a pump that produces a low total head rise at low flow rates for use in a system that requires a high total head rise for starting. System problems can also arise when pumping viscous liquids that are initially cold†. Sufficient flow rate must be achieved for the

* Many pumps can be supplied with integral priming equipment.

† Pumps are available fitted with integral heating jackets to alleviate problems due to liquids with high viscosities when cold or those that freeze under shut-down conditions.

system to warm up to its operating temperature safely. Figure 2 of Item No.80031[21] gives an approximate indication of the reduction in pumping performance that can be expected as a result of viscosity increases but the major effect of viscosity is likely to be on the system where it can cause large increases in friction. System starting problems can sometimes be satisfactorily resolved by employing more than one pump in order to provide the extra performance necessary for starting whilst leaving the normal operating conditions to a pump selected on the basis of optimum efficiency.

Driving unit problems are associated with the torque/speed history of the pump as it runs up to speed. Section 8.1of Item No.79037[20] gives information on the torque characteristics of electric motors and the likely problems that arise when they are coupled to fans. Radial-flow and mixed-flow pumps behave in a similar manner to fans except that the breakaway torque of a pump due to the seals may be considerably higher than that of a fan. Sketch 7.7 illustrates a typical pump torque/speed characteristic.



Sketch 7.7 A typical pump torque/speed characteristic

However an axial-flow pump cannot always be run up to speed without special precautions because, at low flow rates, the high total head rise and correspondingly high torques associated with this type of pump mean that the starting torque is a function of the mass of liquid in the whole pipe system that has to be accelerated. In other words if the system is long, *i.e.* contains a large mass of liquid, the effect on the pump at the moment of starting is virtually the same as if it were started with the discharge valve fully closed. Such problems may be safeguarded against by:

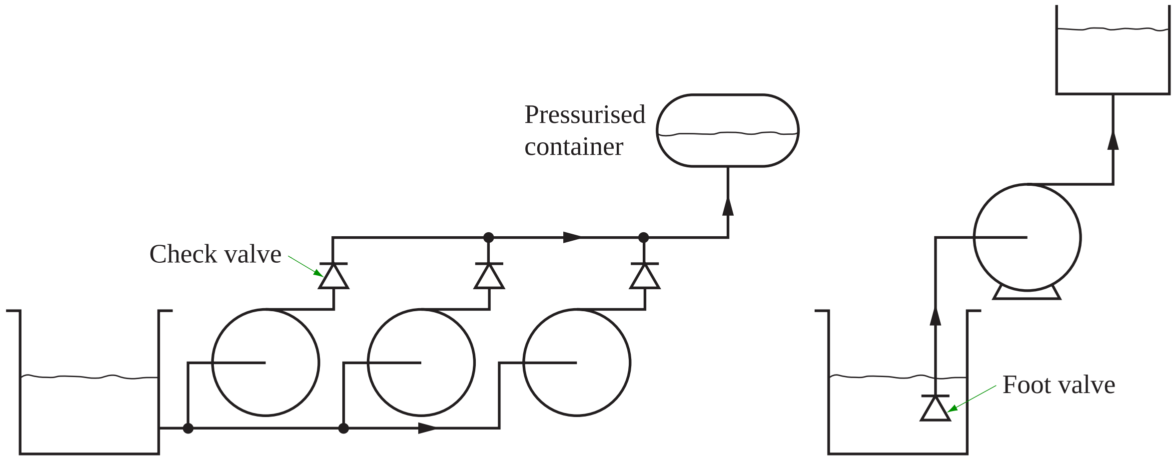
(1) fitting a recirculation bypass to the pump; the bypass can be progressively throttled until the system attains its normal flow rate, or

(2) accelerating the pump slowly for which a variable speed drive will usually be necessary, see Section 8.2 of Item No.79037[20].

7.6 Reflux Devices (Non-Return Devices)

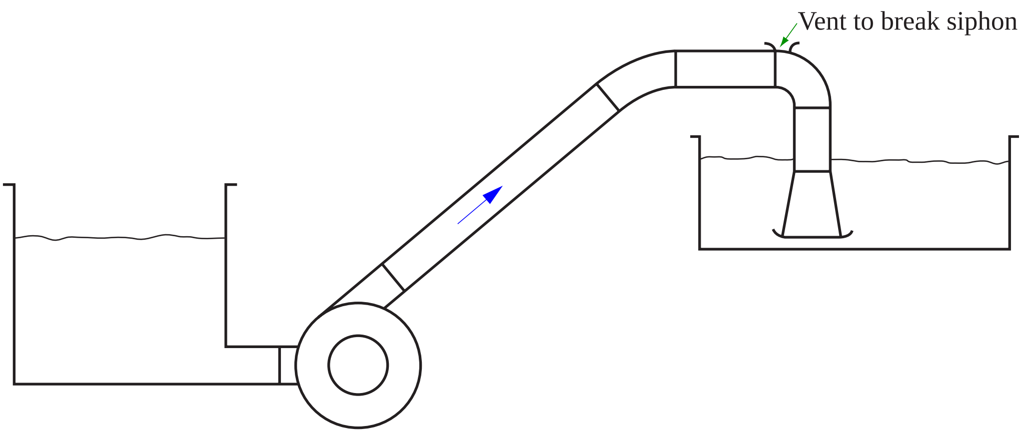
Reflux devices are necessary in systems where the flow direction would otherwise reverse once pumping ceased. Two such devices are check valves, which are common in most types of systems, and siphon breaks the use of which is generally restricted to large scale low head rise open systems.

Check Valves are necessary when the system contains pressurised reservoirs but they are also common on systems with free surface reservoirs, see Sketch 7.8.



Sketch 7.8 Pressurised and open systems fitted with check valves

<u>Siphon breaks</u> are usually restricted to large scale low head rise open installations such as those for pumping flow from one reservoir to a higher one, see Sketch 7.9.



Sketch 7.9 Siphon used as a reflux device

7.7 Water Hammer Prevention

When starting or stopping the flow through a system either by means of valve or a pump, the associated acceleration or deceleration of the flow gives rise to a change in pressure that travels up the line and is reflected at either end. A series of ESDU Data Items provides guidance on the estimation of wave speeds in fluids (Reference 24) and on the estimation, suppression and control of pressure surges (References 25 to 30). Significant enhancement of fluid transients can occur in flexible piping due to fluid/structure interaction (References 31 to 33).

A simplified sequence of events is illustrated in Sketch 7.10.

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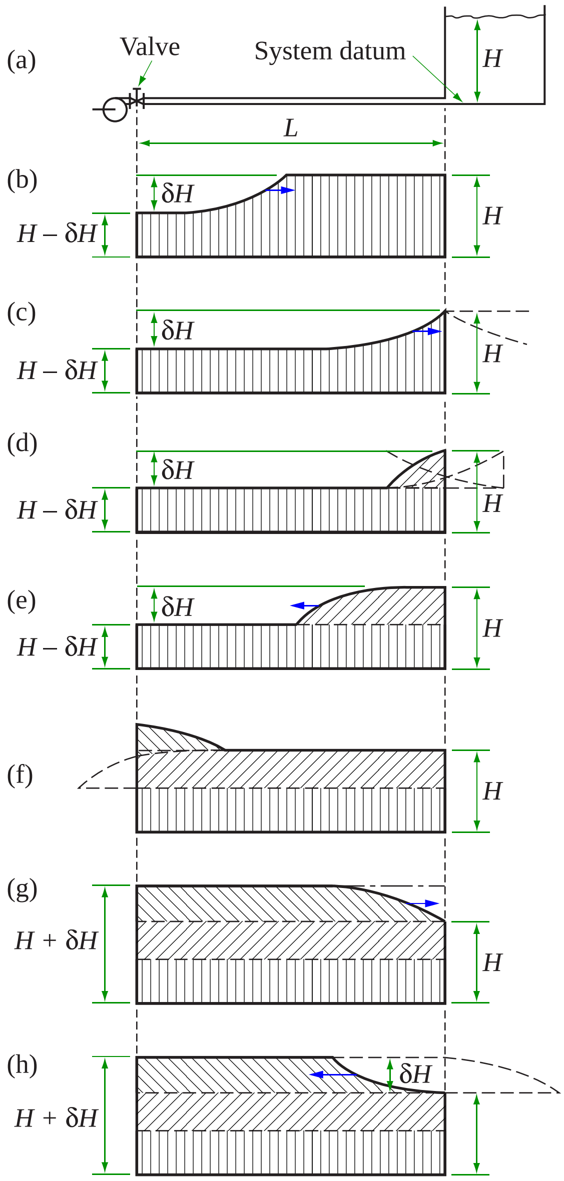

(a) The pump and system are in equilibrium and the system total head is assumed to be H throughout, *i.e.* the total head losses due to friction are neglected.

(b) Following the rapid closure of the valve, a low-pressure wave of magnitude δH travels away from the valve.

(c) The low-pressure wave reaches a sudden opening.

(d) It cannot continue past the opening because thereafter the total head is fixed by the level in the reservoir. Instead it is reflected from the reservoir opening as a high-pressure wave of the same magnitude, *i.e.* there is a 180 degree change of phase.

(e) The wave propagates back towards the valve.

(f) It cannot continue past the closed valve so it is reflected from the valve as a high-pressure wave of the same magnitude, *i.e.* there is no change of phase.

(g) The high-pressure wave travels away from the valve.

(h) At the reservoir, the wave is again reflected with a 180 degree change of phase. On arrival back at the closed valve, it is reflected with no change in phase and the cycle starts again from (b).

Sketch 7.10 Sequence of pressure changes following the closing of a valve

The velocity of the pressure wave in a typical pipeline is given by[25]

$$a = \frac{1}{\sqrt{\rho\left(\frac{1}{K} + \frac{dC}{tE}\right)}} \tag{7.1}$$

where t is the pipe wall thickness,

K is the bulk modulus for the liquid (2070×10^6 N/m^2 for water),

E is the Young's modulus for the pipe material and

C is a constant (usually assumed unity) depending upon Poisson's ratio and the pipe restraint and

d is the pipe internal diameter.

A typical value for the wave velocity in a water-filled steel pipe is 1200 m/s.

The magnitude of the pressure wave depends upon the rate at which the flow is changed[26]. At worst, if the change occurs in a time less than $2L/a$, the head change is given by

$$\text{Head change} = -\frac{a\Delta V}{g}, \tag{7.2}$$

where ΔV is the velocity change ($\Delta V = -V$ for closure).

For steel pipe carrying water with a wave velocity of 1200 m/s, the head change for a 1 m/s change in velocity would be 120 m or 12 bar. Pressure changes of this magnitude may cause the pressure after a pump or valve to fall to vapour pressure giving rise to vapour cavities. The subsequent collapse of cavities is a common source of pressure surges.

In most compact installations no special protective measures need to be taken against pressure surges because the time over which flow changes take place is long compared to the time for a pressure wave to travel through a system and be reflected back to the device causing the flow change. The following checklist identifies features of a system that could give rise to problems:

(1) long pipelines, where substantial flow changes occur at a device during the time that pressure waves take to travel to the end of the system and back,

(2) quickly acting valves or pumping units of low polar inertia (short run-down times),

(3) systems with a large gravitational or externally-applied head containing a check valve that may slam shut,

(4) incomplete priming of the system; because of the contrast in density between the liquid and air, pressure surges result from the rapid changes in flow rates as liquid and air pass through vents or restrictions in the system.

Pressure surges may be controlled by the methods illustrated in Sketch 7.11 and discussed in ESDU 84013[26]. Selection of a particular surge alleviation method is usually determined from economic, operating and maintenance considerations and should be decided on in conjunction with the pump supplier or a specialist in pressure surge problems.

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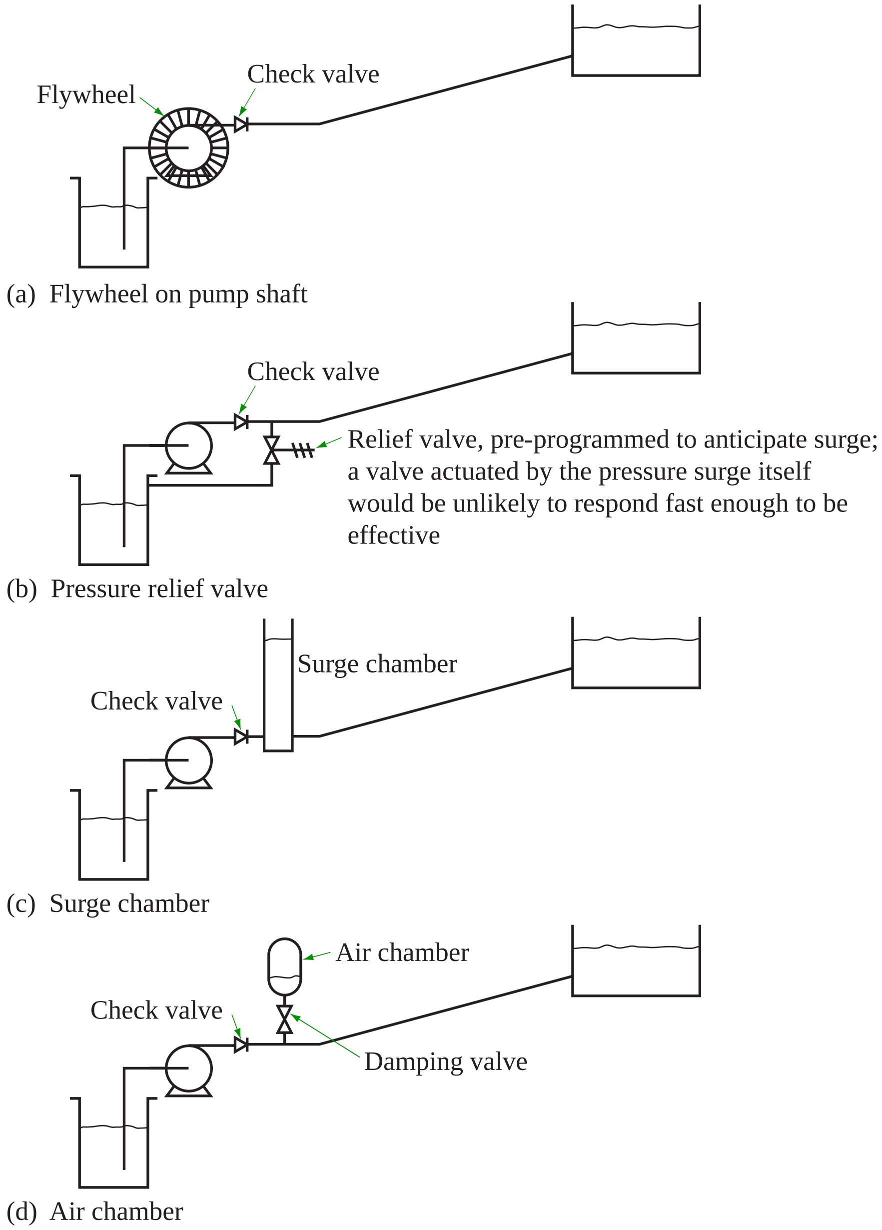


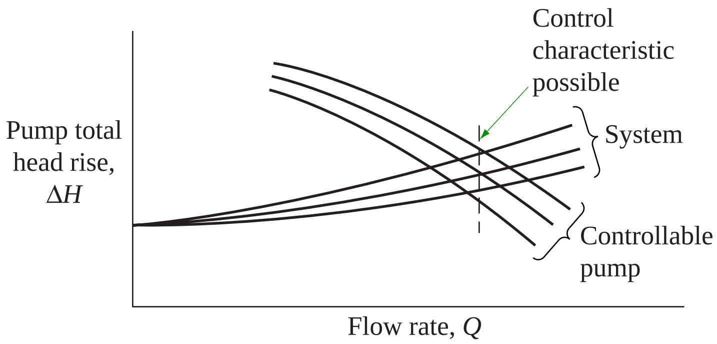
Sketch 7.11 Methods of alleviating pressure surges

7.8 Pump Controls

Pump controls are necessary when the system varies and its requirements cannot be adequately matched to the natural pump characteristic.

A controllable pump offers a flexibility of operation such that, over limited ranges of flow rate and head rise, most system requirements can be met. For example, a control characteristic that gives a truly constant volume flow rate with increased system resistance can be achieved, see Sketch 7.12.

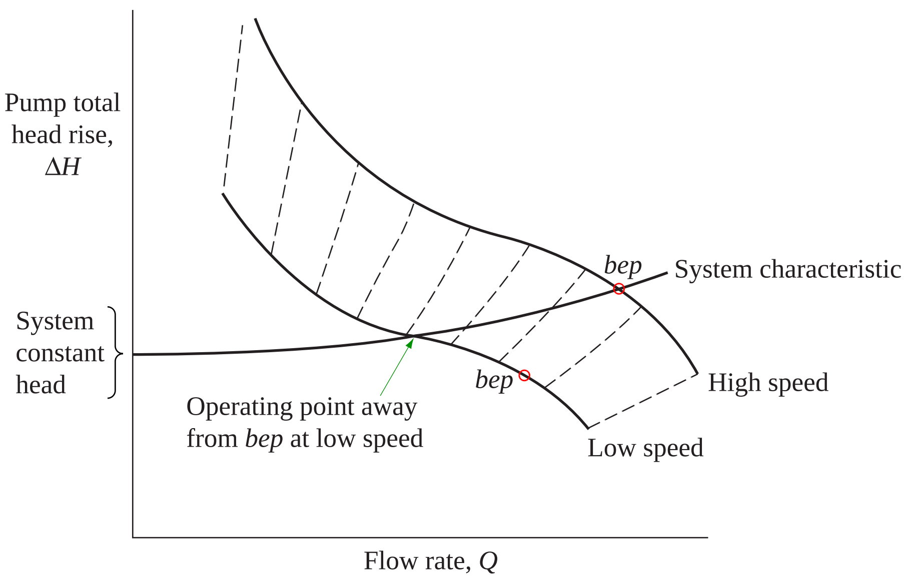
However, if such a control characteristic is to be maintained automatically, the consequences of failure should be investigated.



Sketch 7.12 Use of pump controls to achieve constant volume flow system requirements

Three types of controls are commonly used: speed control, flow control through throttling and flow control through by-passing. Variable pitch rotor or variable inlet swirl controls may be offered on sophisticated axial and mixed-flow pumps.

Speed control can be achieved by mechanical, hydraulic or electrical means. It is suitable for all categories of pumps and is hydrodynamically the most efficient means of flow control. It is also safe from a *NPSHr* aspect. Mechanical controls depend on friction and require regular maintenance and, although of high efficiency, are generally available only for powers less than 50 kW. Hydraulic couplings are a less efficient alternative that are designed to run with a controllable amount of slip. Electrical means of control may also incur efficiency penalties depending on their type. These are detailed in Section 8.2 of Item No.79037[20]. The consequences of speed control can be investigated using the similarity laws, Equations (3.11) to (3.15), in conjunction with the system characteristic, see Sketch 7.13.



Sketch 7.13 Effect of change of speed on operating point for system with high constant head

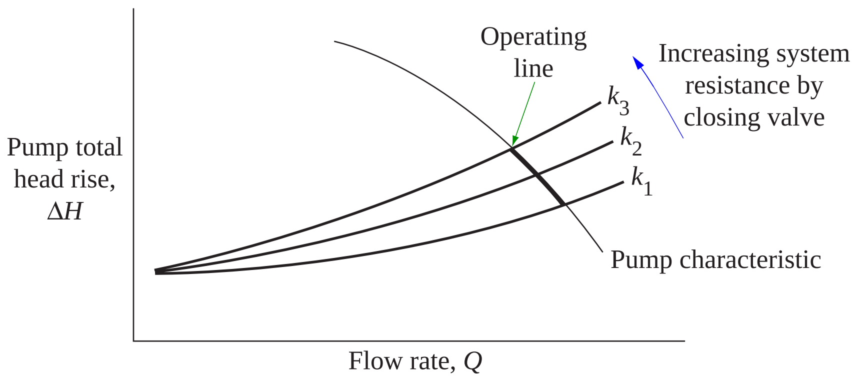
The intersections of the system characteristic with the pump characteristics occur at different positions on the two pump curves. In order to predict the operating point at a different speed, it is therefore necessary to produce a chart similar to Sketch 7.13.

Throttle flow control is the most often-used means of control. It can be applied to most systems except those employing axial-flow pumps. Throttle valves should always be positioned on the discharge side of a pump so that the $NPSHa$ to the pump is not affected but a large pump cannot operate continuously at flow rates much below its best efficiency point due to vibration caused by hydraulic mismatching. To control the flow from large pumps, therefore, it is often necessary to use a combination of throttling and bypassing.

Throttling devices waste power but are especially inefficient when used with high specific speed pumps. This is because the power requirements of such pumps increase as the flow rate is reduced, see Sketch 3.4. In addition, with high specific speed pumps, *i.e.* those whose $NPSHr$ increases as the flow rate decreases below Q_{bep}, cavitation problems can arise.

In order to provide an effective means of flow control, valves may need to dissipate total heads several times greater than the total head loss of the remainder of the system. However, the ability of valves to dissipate the flow total head is limited by cavitation and specially-designed control valves are required. Apart from having a high capital cost, these result in an increase in the system total head loss even when fully opened.

The range of control available with throttling devices can be investigated by adding a head loss proportional to Q^2 to the system characteristic as illustrated in Sketch 7.14. The constants of proportionality, k_1 to k_3, depicted in Sketch 7.14 are related to the loss coefficients and thus the position of the throttle. Item No. 69022[19] gives approximate data on the variation of loss coefficients with throttle position.



Sketch 7.14 Modification of system characteristic by throttling

Bypass flow control is a stable means of controlling the flow into a system but in commercial operations it is normally only applied to high specific speed pumps and to large sizes of other pumps that cannot tolerate large departures from best efficiency point operation. Because it is wasteful of power, it is usually only adopted if either speed control or variable pitch rotor control is not possible.

If the bypass allows higher-than-design flow rates to pass through the pump, the increase in $NPSHr$ should be noted and care taken to ensure an adequate $NPSHa$.

8. REFERENCES AND DERIVATION

8.1 References

The references given (listed in chronological order) are recommended sources of information supplementary to that in this Item.

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8.2 Derivation

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The work on this Item was carried out in the Internal Flow and Physical Properties Group of the Engineering Sciences Data Unit. The member of staff who undertook the technical work involved in the initial assessment of the available information and the construction and subsequent development of the Item was

Mr C.J.T.Clarke — Group Head, Internal Flow and Physical Properties.

The Internal Flow Panel, which took over the work on Internal flow previously monitored by the Fluid Mechanics Steering Group, first met in September 1979 and has the co-operation of many engineers and scientists in industry, research establishments and universities from whom much assistance and information is being received.

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