

Radial, mixed and axial flow pumps. Introduction

Endorsed by
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(Continued on inside back cover)

RADIAL, MIXED AND AXIAL-FLOW PUMPS. INTRODUCTION

CONTENTS

	Page
1. NOTATION AND UNITS	1
2. INTRODUCTION	3
2.1 Item Scope	3
2.2 Item Layout	3
3. HYDRAULIC ASPECTS OF PUMPS	4
3.1 How Pumps Work	4
3.1.1 Pump total head rise	4
3.2 Impeller Design and Specific Speed	5
3.3 Impeller Enclosures	9
3.4 Cavitation and Suction Specific Speed	10
3.4.1 Net positive suction head available, NPSHa	11
3.4.2 Net positive suction head required, NPSHr	12
3.4.3 Suction specific speed	13
3.5 Pump Similarity Laws	14
3.5.1 Effect of speed change	14
3.5.2 Effect of size change	16
3.6 Impeller Trimming	17
3.7 Practical Limitations	17
3.7.1 Head rise per stage and specific speed	17
3.7.2 Maximum and minimum continuous flow rates	20
3.7.3 NPSHr and suction specific speed	20
3.7.4 Maximum liquid viscosity	21
4. ASPECTS OF PUMP CONSTRUCTION	22
4.1 Installation Considerations	22
4.1.1 Orientation	22
4.1.2 Common configurations	22
4.2 Casing Design	26
4.3 Impeller Shaft Design	28
4.4 Shaft Sealing Arrangements	29
5. STANDARDS FOR PUMPS	30
5.1 Performance	30
5.2 Dimensional Standards	33

6.	SYSTEM CHARACTERISTICS	34
7.	PUMP AND SYSTEM MATCHING	39
7.1	Stable Operation	39
7.2	Margins for Error in Pump and System Head Loss Characteristics	40
7.3	Multi-Pump Arrangements	41
7.3.1	Two pumps in series	41
7.3.2	Two pumps in parallel	42
7.4	Pump Siting	44
7.5	Priming and Starting	44
7.5.1	Priming	44
7.5.2	Starting	44
7.6	Reflux Devices (Non-Return Devices)	45
7.7	Water Hammer Prevention	46
7.8	Pump Controls	49
8.	REFERENCES AND DERIVATION	52
8.1	References	52
8.2	Derivation	53

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.

		<i>Units</i>
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, $g \approx 9.81 \text{ m/s}^2$	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

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>i.e.</i> $\eta = 100 \times Q\rho g\Delta H/P$	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
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2. INTRODUCTION

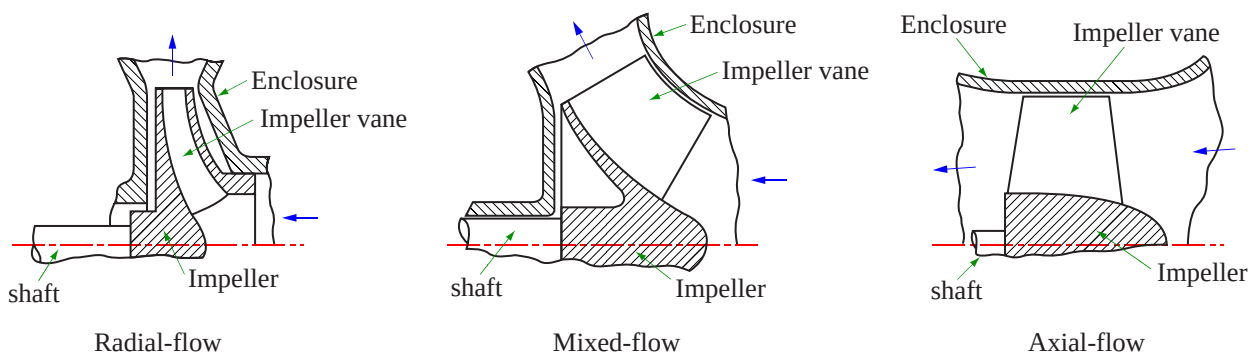
Radial, mixed and axial-flow pumps belong to a class of machines known as rotodynamic. As a class, they are suitable for the majority of liquid pumping applications but notable exceptions include metering, handling of highly viscous liquids (*e.g.* liquid with $\nu > 200$ cSt but see Item No. 80031 for 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.*

$$\Delta H = \Delta p / (\rho g), \quad (3.1)$$

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.

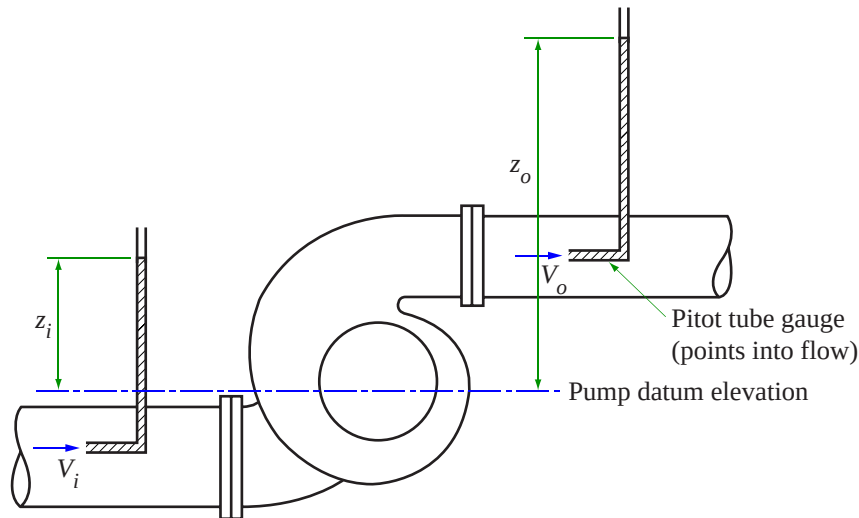
Note that total head, or total pressure, as used in this Item, refers to a quantity that is the sum of static, kinetic and elevation terms, *i.e.*

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

$$p = p_s + \frac{1}{2} \rho V^2 + \rho g z. \quad (3.3)$$

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 tapplings (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

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

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.81002²³.

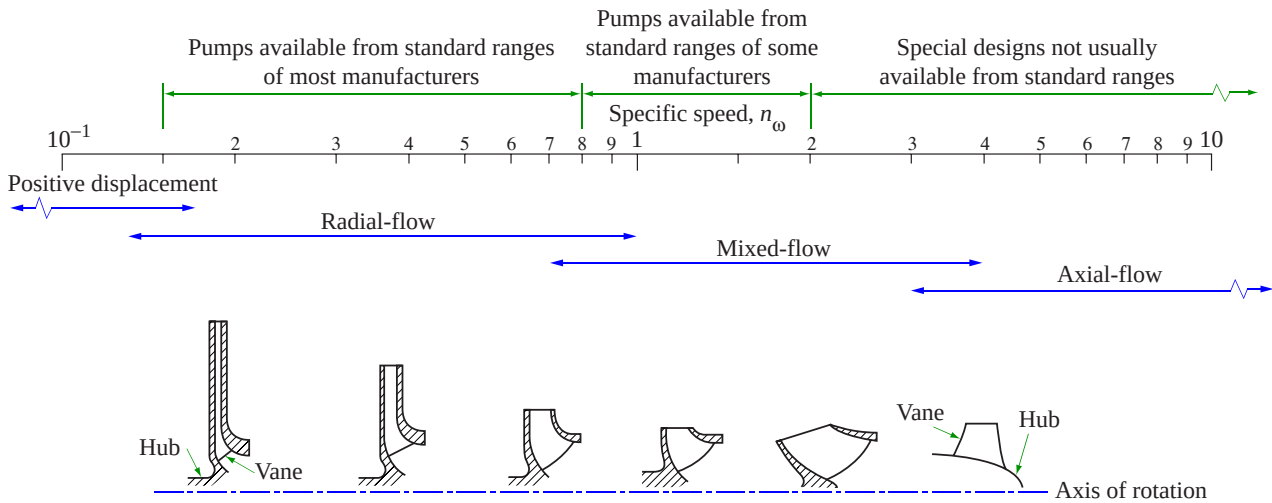
Specific speed conversion factors			
To convert from	rev/min (Imp gal/min) ^{1/2} ft ^{-3/4}	to n_{ω} , divide by	2494
	rev/min (US gal/min) ^{1/2} ft ^{-3/4}		2733
	rev/min (l/s) ^{1/2} m ^{-3/4}		1673
	rev/min (m ³ /h) ^{1/2} m ^{-3/4}		3175
	rev/min (m ³ /s) ^{1/2} m ^{-3/4}		52.92
Note these factors 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.



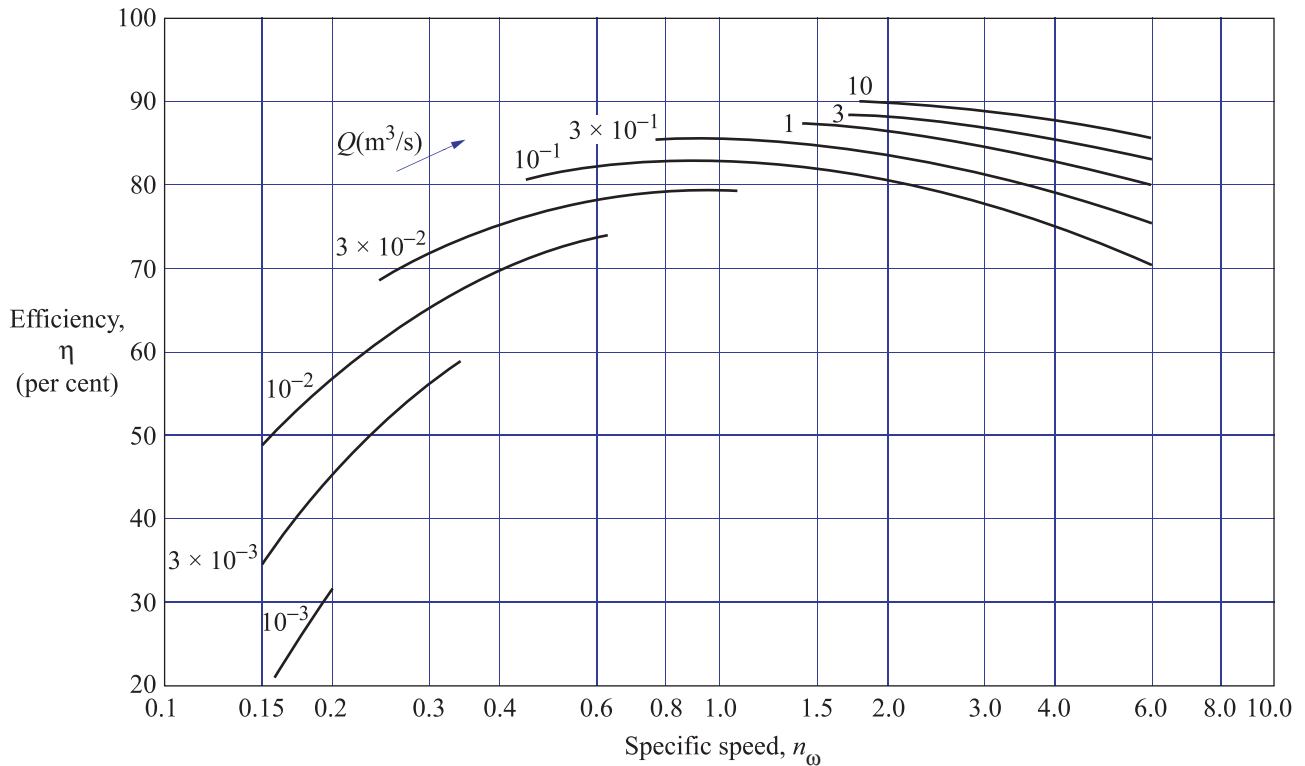
Sketch 3.2 Approximate variation of geometries of impellers having different specific speeds

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.

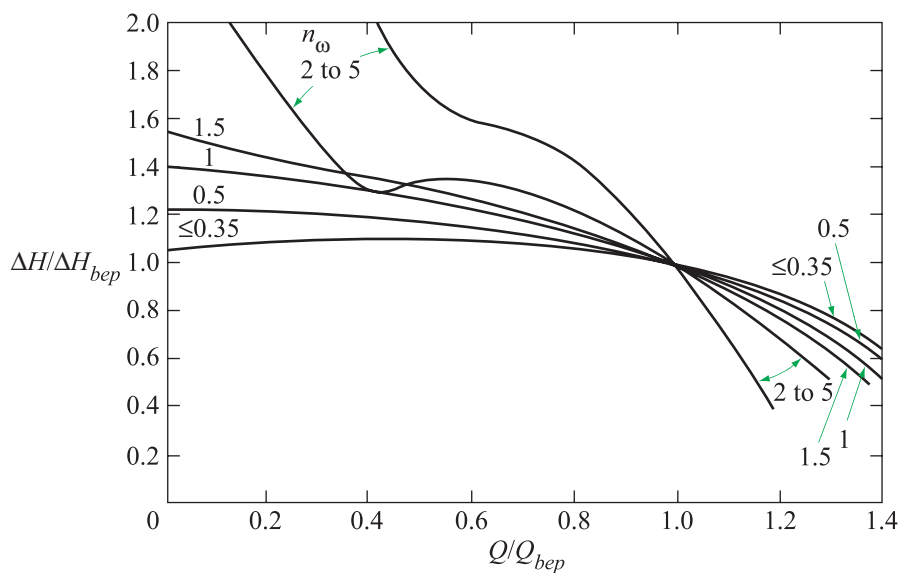
For pumps of $n_{\omega} > 2$ it is not possible to typify the shapes of pump characteristics. Such pumps are often of mixed or axial-flow design and are not generally available from manufacturers' standard ranges. The large scope for altering the geometry of such impellers means that pumps of the same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

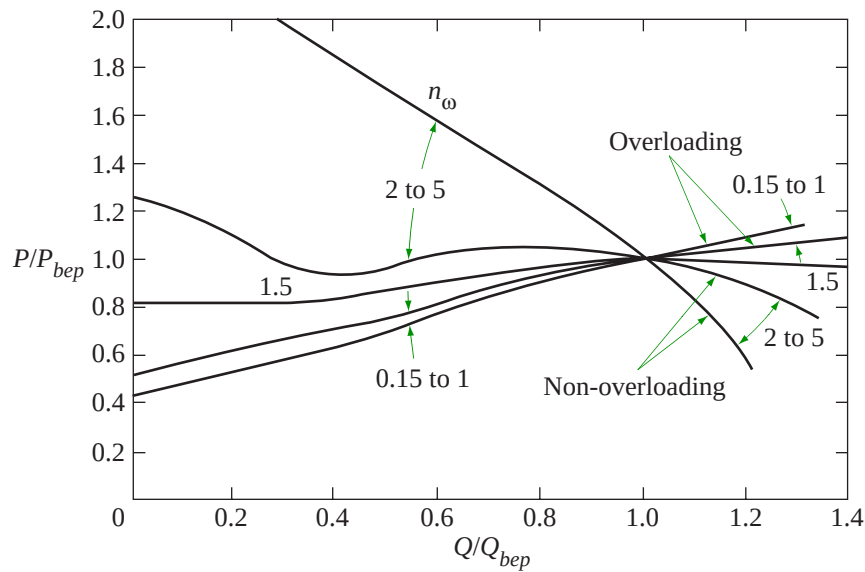
Knowledge of the shape of the power characteristic is also necessary to ensure that the driving unit will not be overloaded however the pump may be operated. Sketch 3.5 illustrates the shapes of typical power characteristics of pumps having different specific speeds. Those power characteristics that 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

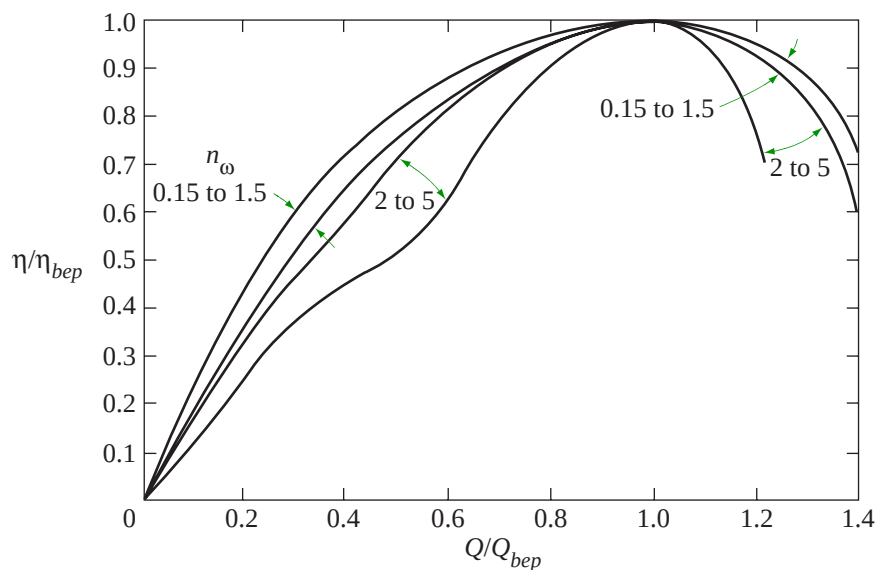


Sketch 3.4 Head rise characteristics of pumps having different specific speeds



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

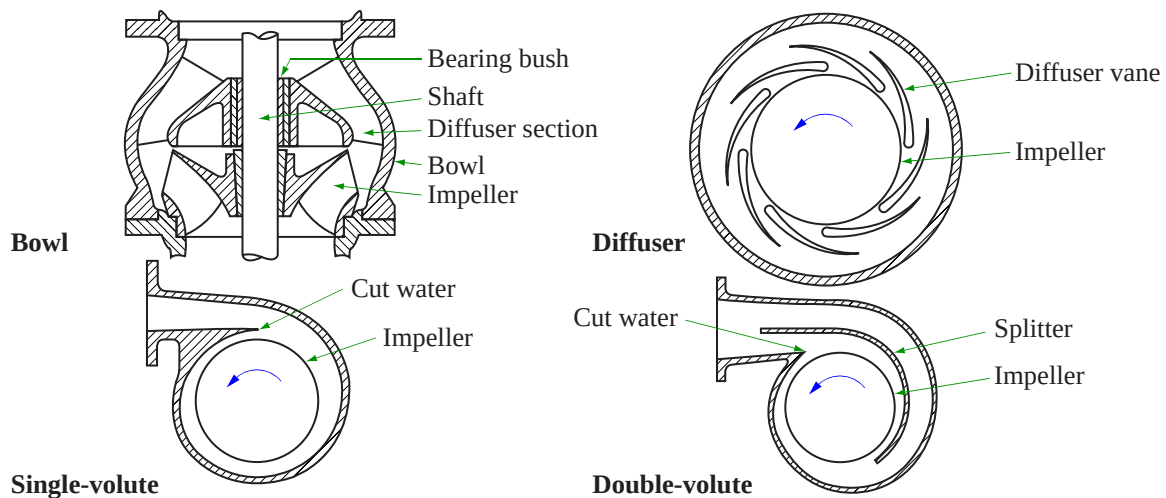
3.3 Impeller Enclosures

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.

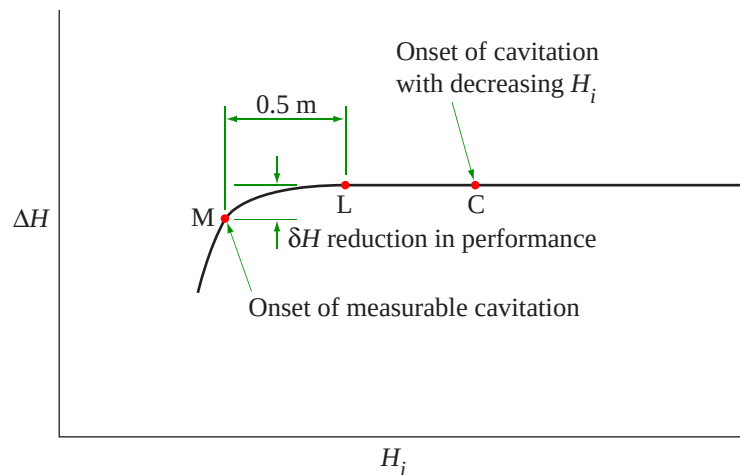
Diffuser enclosures offer the same advantages as double volutes and, because the position of the outlet is not governed by flow considerations, diffuser enclosures are frequently used on multi-stage radial-flow pumps where they discharge into an annular channel.

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 implussions, 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

$$NPSHa = (p - p_v) / (\rho g), \quad (3.5)$$

which may alternatively be expressed as

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

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

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

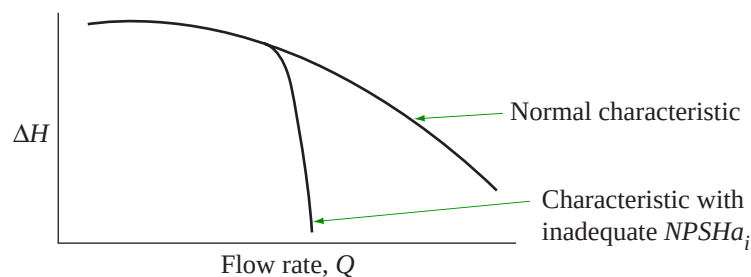
the static pressure is equal to the vapour pressure.

* 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.

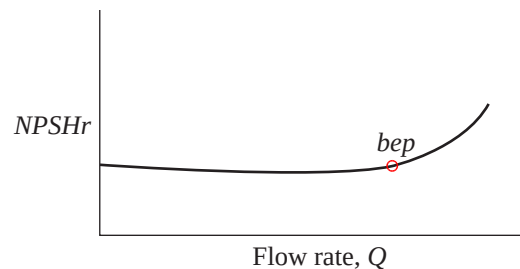
3.4.2 Net positive suction head required, $NPSRr$

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.



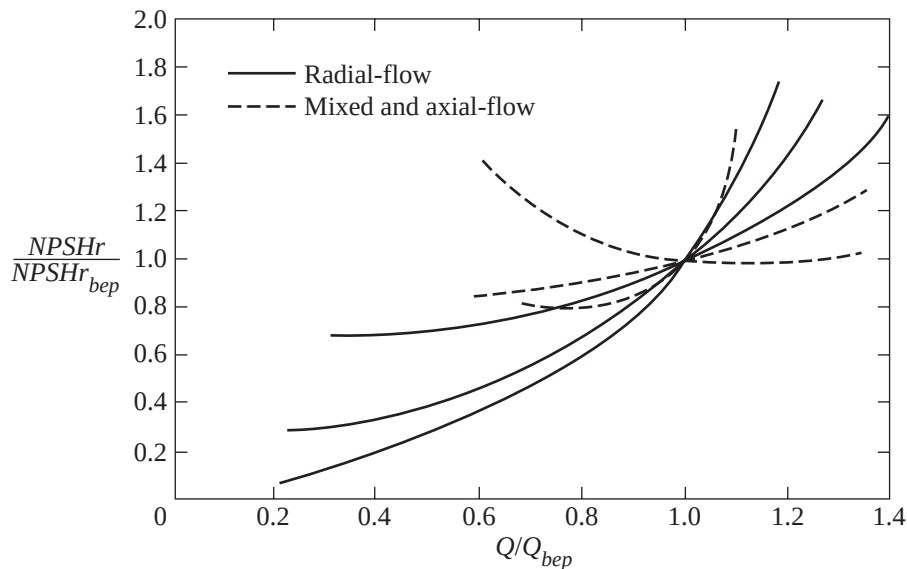
Sketch 3.10 Typical pump $NPSHr$ characteristic

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 available⁴⁴ 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

$$NPSHa_i \geq NPSHr + 0.5 \text{ m.} \quad (3.7)$$

3.4.3 Suction specific speed

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

$$S_\omega = \frac{\omega Q_{bep}^{1/2}}{[g NPSHr_{bep}]^{3/4}} \text{ rad,} \quad (3.8)$$

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:

$$\frac{Q_1}{D_1^3 N_1} = \frac{Q_2}{D_2^3 N_2} \quad (3.9)$$

and
$$\frac{\Delta H_1}{D_1^2 N_1^2} = \frac{\Delta H_2}{D_2^2 N_2^2} \quad (3.10)$$

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:

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

and
$$Q_2 = Q_1 N_2 / N_1 \quad (3.12)$$

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

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

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

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

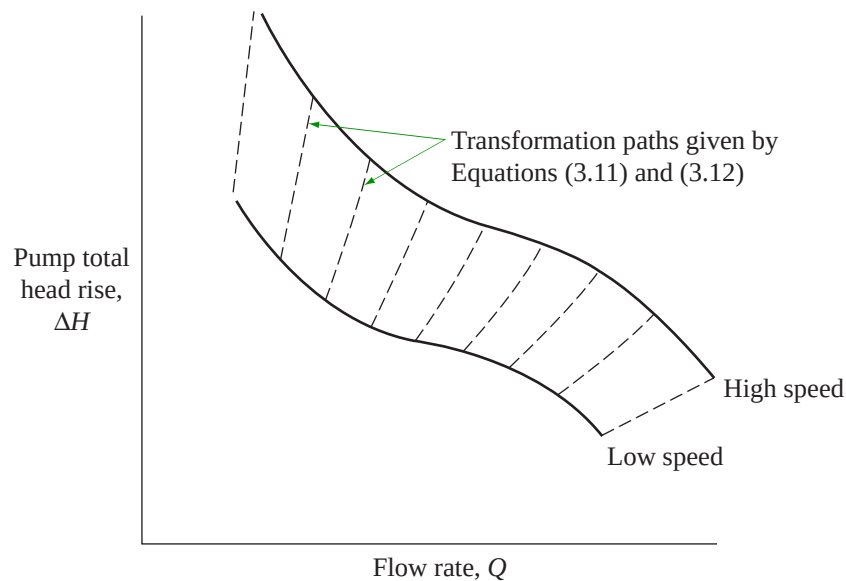
Equations (3.11) to (3.14) may be used to map a known characteristic (e.g. ΔH_1 versus Q_1) into a new characteristic (e.g. ΔH_2 versus Q_2) corresponding to a new speed. Sketch 3.12 illustrates this process for a head rise characteristic.

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:

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

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.