

PUMP DRIVES

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PUMP DRIVES

1. INTRODUCTION

A pump is a machine for converting rotary or reciprocating shaft power into fluid power. It therefore always requires a suitably rated prime mover. The principal selection criteria for matching the driver to the pump are speed and power but there are many other issues to be considered. The main purpose of this lecture is to introduce the fundamental principles underlying proper drive selection and operation.

2. PUMP SPEED

Pump speed is determined by considering pump size, efficiency and cavitation performance. Since for rotodynamic machines the total head generated at any specified flowrate is a function of impeller tip speed squared i.e.,

$$H = f(U_2^2)$$

and

$$U_2 \propto D_2 \times N_2$$

It can be seen that the higher the rotational speed the smaller the pump (as speed goes up, diameter goes down to maintain the same velocity and thus head rise). This should mean that the pump is lighter, cheaper and would occupy less space in the pumping installation.

For positive displacement pumps the fundamental relationships between flowrate, head (or pressure) and speed are different (basically $Q \propto N$) but the same conclusion can be drawn: a faster rotating pump is a smaller cheaper pump.

For most pumps there is a relationship between speed and efficiency, i.e., the speed may be optimised in order to obtain maximum pump efficiency. Maximum efficiency means that, firstly, the prime mover power rating is minimised and secondly that the energy costs of pumping are minimised.

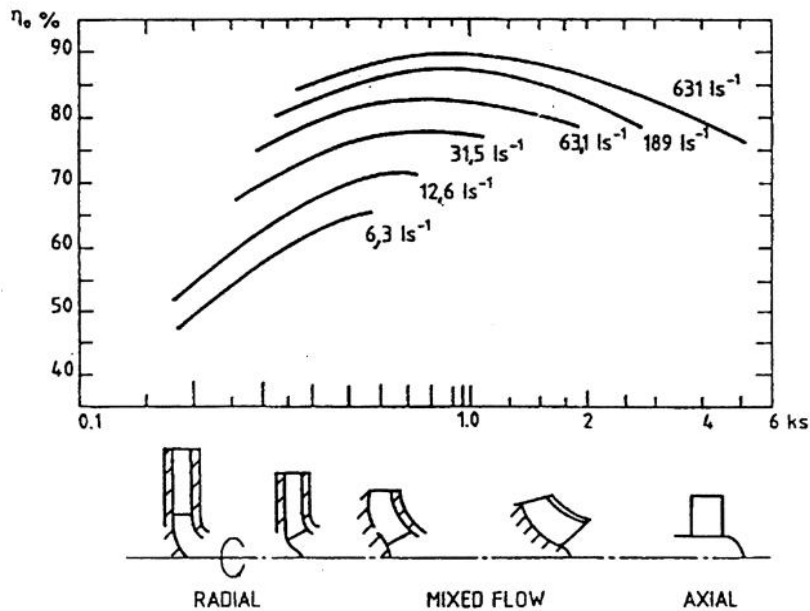


Figure 1 Variation of efficiency with flow and k_s

Fig.1 illustrates the relationship between pump efficiency and pump type number (or specific speed) where type number, k_s :

$$k_s = \frac{\omega \sqrt{Q}}{(gH)^{0.75}}$$

where $\omega = \frac{2\pi N}{60}$, and N is the rotational speed in rev/min.

[In UK or US units, it is called specific speed, $N_s = \frac{N \sqrt{Q}}{H^{0.75}}$]

It can be seen that k_s is directly proportional to rotational speed N (rev/min). The efficiency of positive displacement pumps is also influenced by speed.

The third major factor to be considered is the suction or cavitation performance of the pump. For rotodynamic pumps there is a relationship between the net positive suction head available ($NPSH_A$) and the pump speed, i.e.

$$k_{ss} = \frac{\omega \sqrt{Q}}{(g NPSH)^{0.75}}$$

[In UK and US units, it is defined as, $S = \frac{N \sqrt{Q}}{(NPSH)^{0.75}}$]

For pumps of a particular type operating at design point, k_{ss} has specific optimum values (e.g. 3.25 for single entry overhung impeller pumps). Hence transposing the equation for a specified flowrate Q and given $NPSH_A$ will determine the maximum acceptable running speed.

$$N = \frac{60 k_{ss} (g NPSH_A)^{0.75}}{2\pi \sqrt{Q}}$$

The Hydraulic Institute Standards lay down recommended maximum operating speeds based on this type of consideration, see Figs 2 and 3.

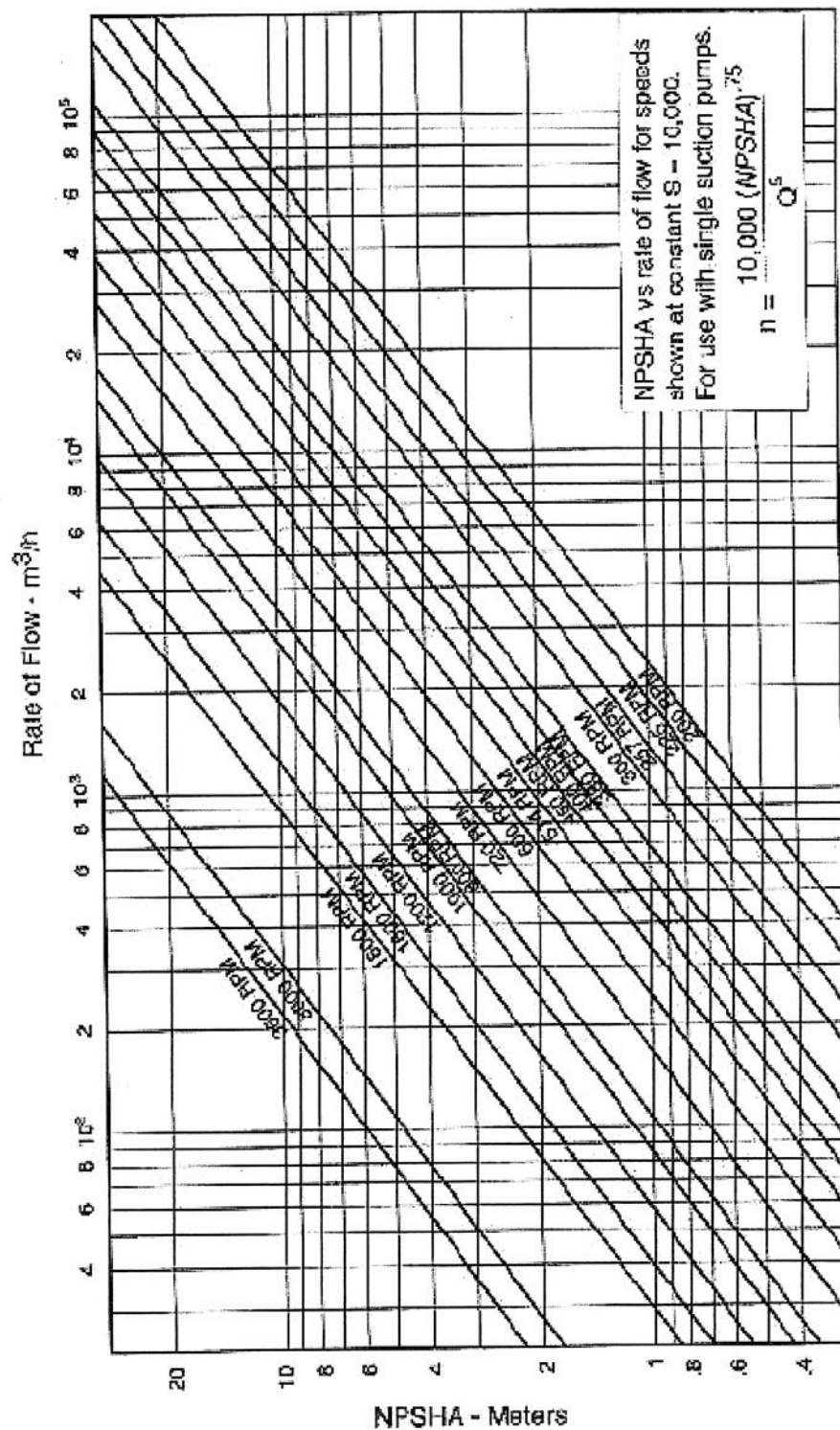


Figure 2 Recommended maximum operating speeds

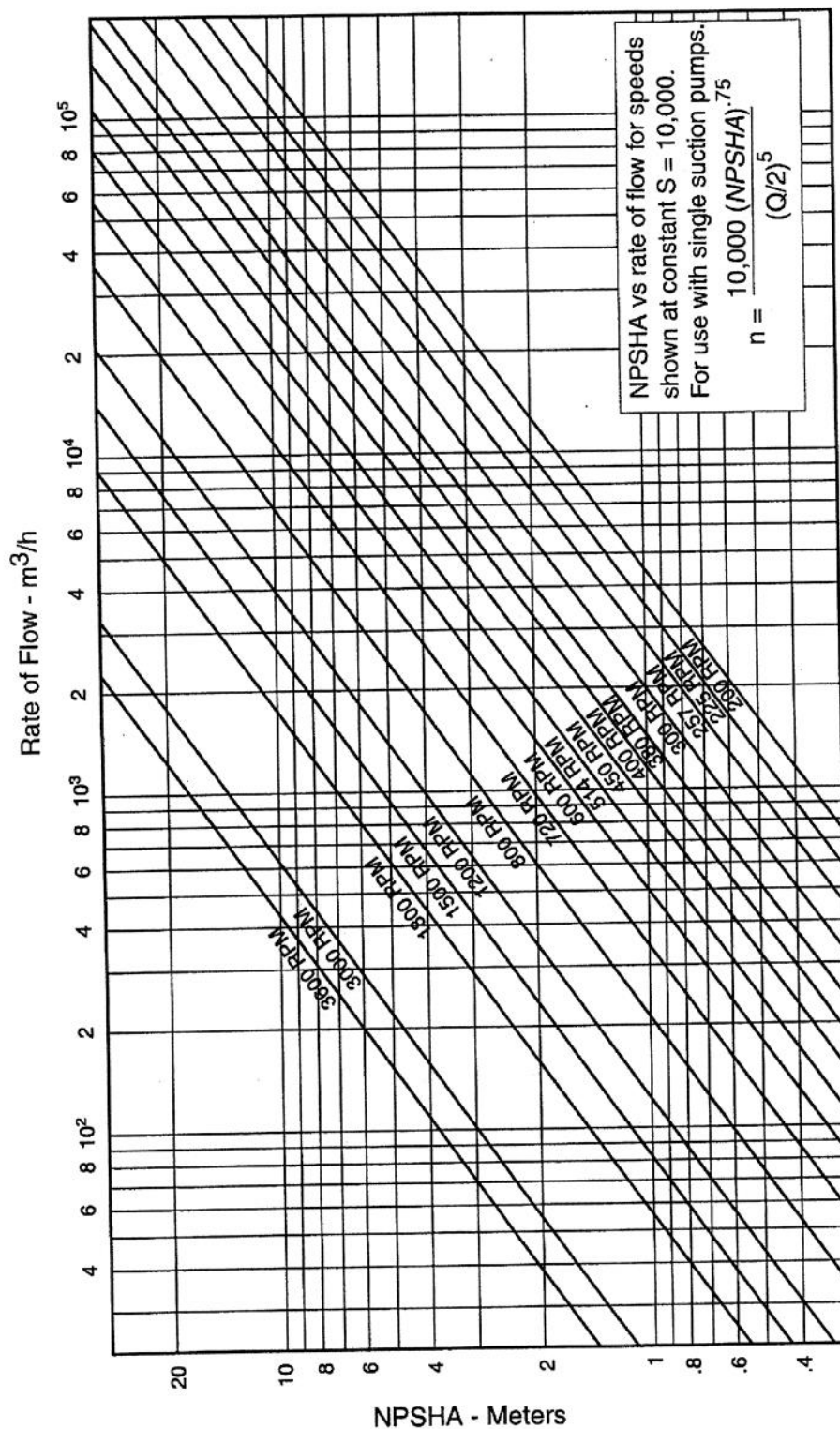


Figure 3 Recommended maximum operating speeds

It can be seen from the foregoing discussion that pump speed can be determined independently of driver considerations, although normally the decision is made partly based on the driver specification. Since many pumps are driven by squirrel cage a.c. induction motors, the direct drive speed is limited by a.c. electricity supply frequency, and the number of pole pairs in the motor (Table 1). In the UK with its 50 Hz a.c. grid most pumps would run at 980, 1480, or 2980 rev/min.

kW	No of poles	Supply frequency, Hz	
		50	60
0.75 to 2.2	2	2820	3380
	4	1405	1685
	6	930	1115
3 to 7.5	2	2855	3425
	4	1420	1700
	6	945	1130
11 to 22	2	2910	3490
	4	1445	1730
	6	960	1145
	8	720	860
30 to 75	2	2935	3520
	4	1475	1770
	6	975	1170
	8	735	880

Table 1 Motor size, number of poles, frequency and 'actual' speed

The speed of most prime movers is load dependent so unless there is an independent speed control, e.g. a compensating governor on an engine or steam turbine, the speed of the pump will vary with pump load. It is important to allow for this variation in pump system interaction calculation especially since most standard performance data is presented corrected to constant speed. For a.c. induction motors slip is almost directly proportional to load so that if full load and no load speeds are known for the motor an accurate estimate of the pump speed at any other load can be made.

3. PUMP SPEED VARIATION

Having established the maximum acceptable drive speed, the variation of speed should be considered. The reasons for selecting a variable speed system will be discussed later but the effect of speed change on pump performance has other important effects during transient conditions, e.g. starting.

The basic laws, sometimes known as the pump affinity laws, relating pump performance to speed are:

Flowrate	$Q \propto N$
Total Head	$H \propto N^2$
Power Input	$P \propto N^3$ (see below)
NPSHR	$NPSH_R \propto N^2$ (see below)

These laws relate similar operating points at different speeds hence any point on a pump characteristic at speed N_1 , can be translated to an equivalent point on the characteristic at speed N_2 . The relationships hold true for most practical speed changes but extra care should be taken when manipulating power and $NPSH_R$. Power is dependent on efficiency and efficiency does not remain constant as speed changes. Because the Reynolds Number of the flow reduces with reducing velocity (speed) then the flow losses tend to increase. This leads to a reduction in efficiency. Fig. 4 gives an estimate of the change of efficiency with speed and this change should be taken into account when calculating power input at reduced speed. The variation of $NPSH$ given is approximate for similar reason.

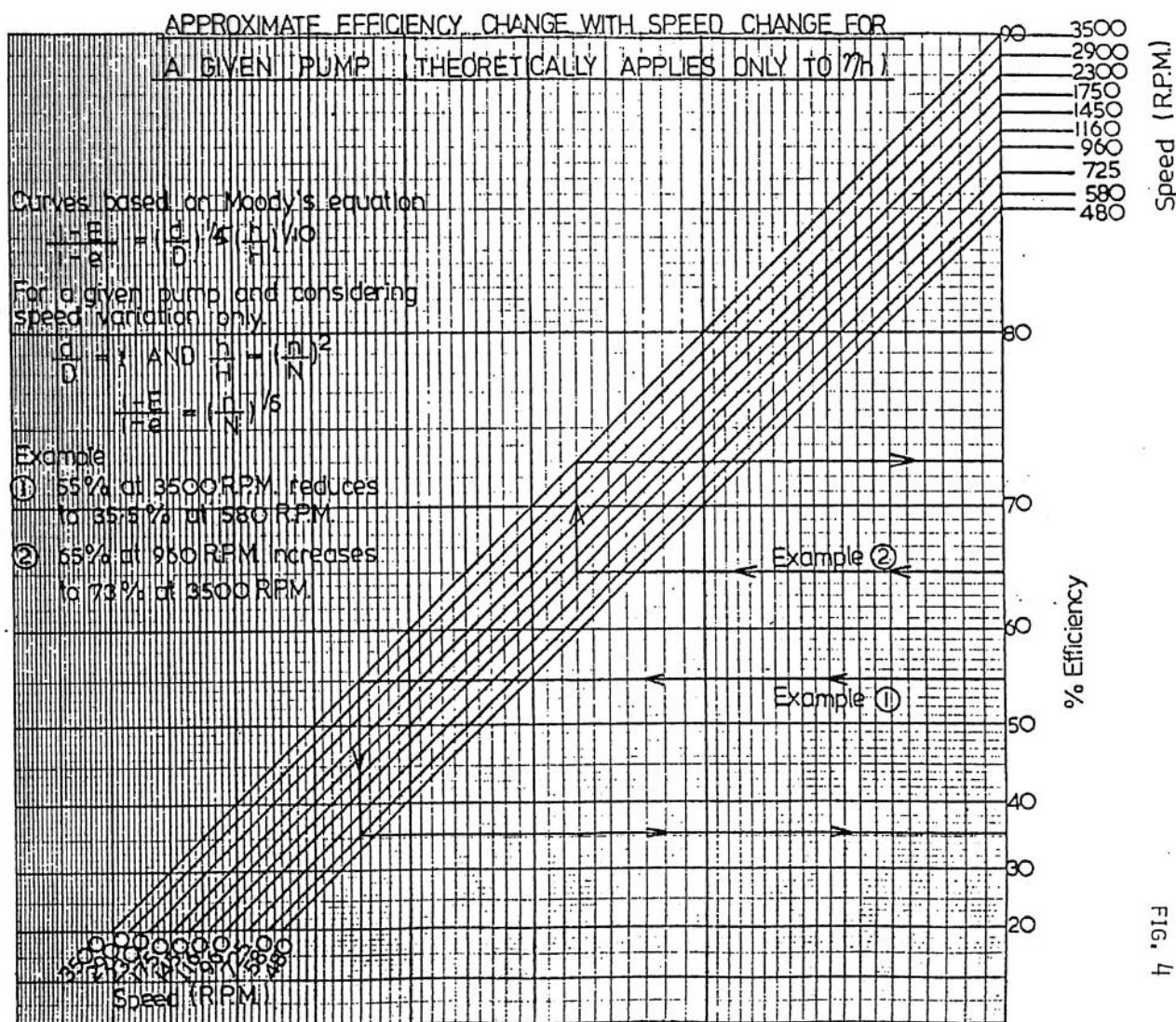


Figure 4 Approximate efficiency change with speed change for a given pump η_h

Curves based on Moody's equation:

$$\frac{1-E}{1-e} = \left(\frac{d}{D}\right)^{1/5} \left(\frac{h}{H}\right)^{1/10}$$

For a given pump

$$\frac{d}{D} = 1 \text{ and } \frac{h}{H} = \left(\frac{n}{N}\right)^2$$

Thus

$$\frac{1-E}{1-e} = \left(\frac{n}{N}\right)^{1/5}$$

Example 1 55% @3500rpm reduces to 35.5% @ 580rpm

Example 2 65% @ 960rpm increases to 73% @ 3500rpm

4. PUMP POWER INPUT

The basic relationship between pump duty, flowrate (Q), total head (H), and input power (P) is:

$$P(kW) = \frac{\rho g H Q}{\eta_{0a} \times 1000}$$

The power for any duty can be calculated if the efficiency is known. This in turn requires knowledge of the pump characteristics, especially where the pump operates over a range of flowrate or at different speeds, i.e. a variable speed pump set. Fig 5 illustrates a complete pump characteristic and Fig 6 shows the variation of shape of power characteristics over the range of type numbers, ks, for rotodynamic pumps (Diagrams after Stepanoff). Most positive displacement pumps exhibit a power characteristic, which increases with flowrate and/or pressure.

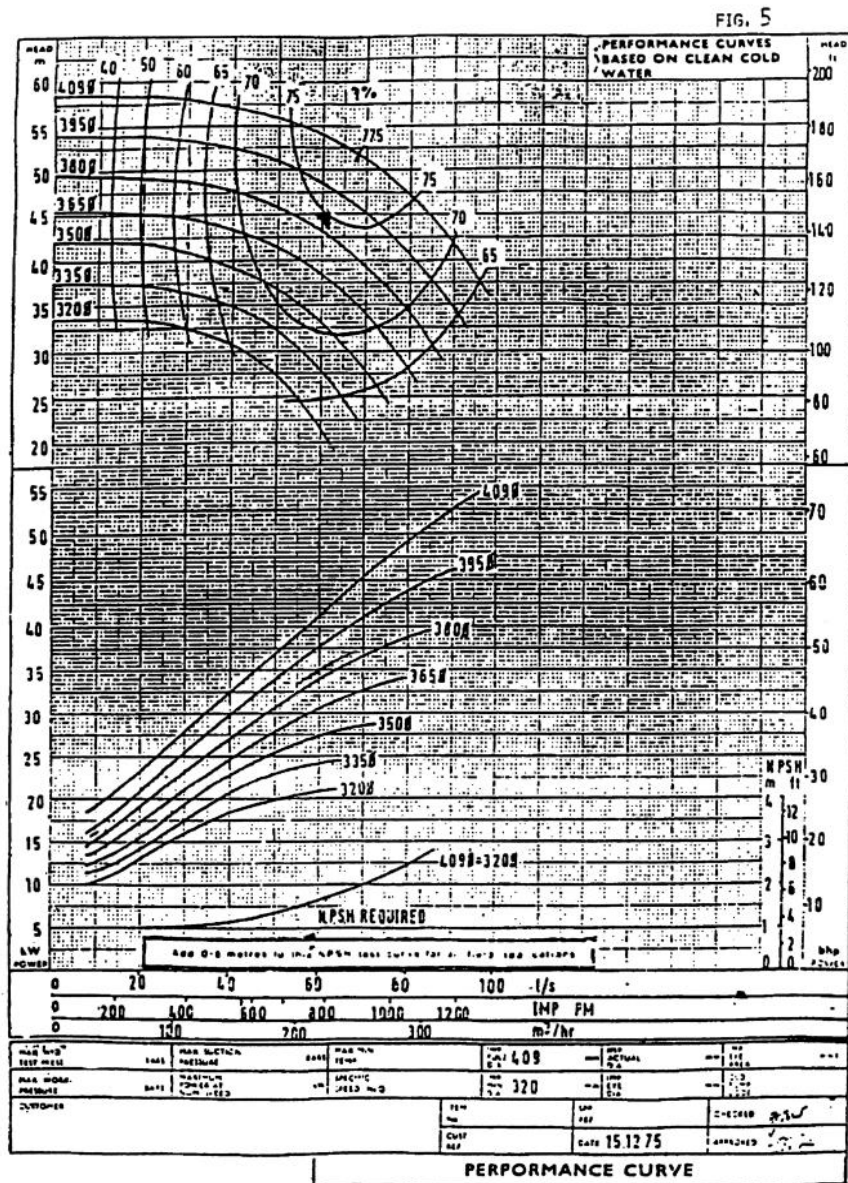


Figure 5 Pump characteristic

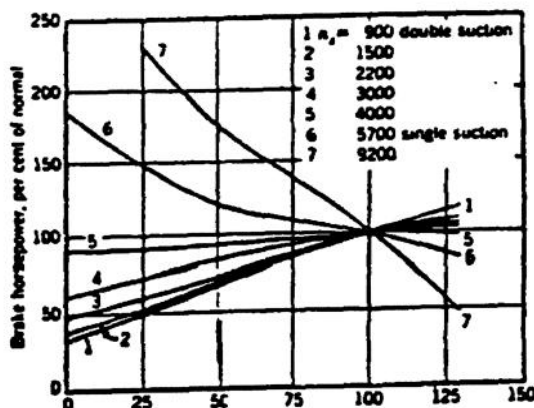


Figure 6a. Typical head capacity curve for several

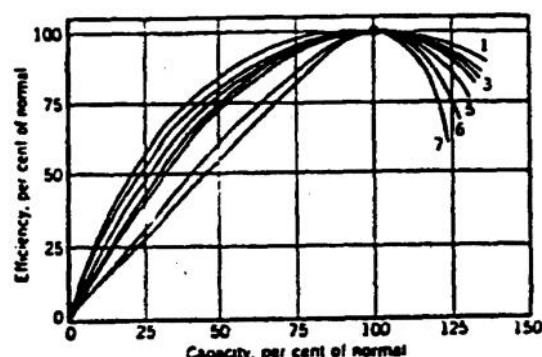


Figure 6a. Typical efficiency curve for several specific speed

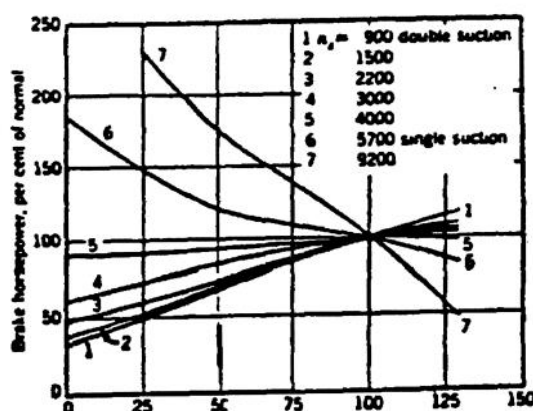


Figure 6c Typical brake horsepower curves for different specific speed

Various factors may influence the actual power absorbed, either initially or during operation. Viscosity has a significant effect and correction charts are given in Figs. 7 and 8, which will enable adjustment to be made to the "normal" water performance. Input power varies directly with liquid density which in turn is temperature dependent so full account must be taken of temperature variations. Note that in most cases viscosity also is temperature dependent.

Other factors to be taken into account are variations in pump and system characteristics, caused by inaccuracies in estimation or test, and changes of operating conditions, e.g., operating range. Low specific speed rotodynamic pumps and positive displacement pumps generally have a power characteristic of positive slope that is power increase with flowrate, whereas high specific speed axial flow pumps have a falling power curve. This variation in power must be considered when choosing the prime mover power rating, and usually margins would be added to pump duty input power.

Ideally, a pump with a non-overloading power curve would be chosen but this is not usually feasible, particularly for relatively high heads or low heads. Non-overloading means that

there is a maximum power absorbed at or near design flowrate so that when the motor is rated, usually with a margin of +10 to 15% on design pump input power, it will not be overloaded at any other flowrate pump. The pump designer can control to some extent the degree to which the pump power varies from design power by choosing suitable design parameters but once a pump is manufactured its power curve shape is fixed.

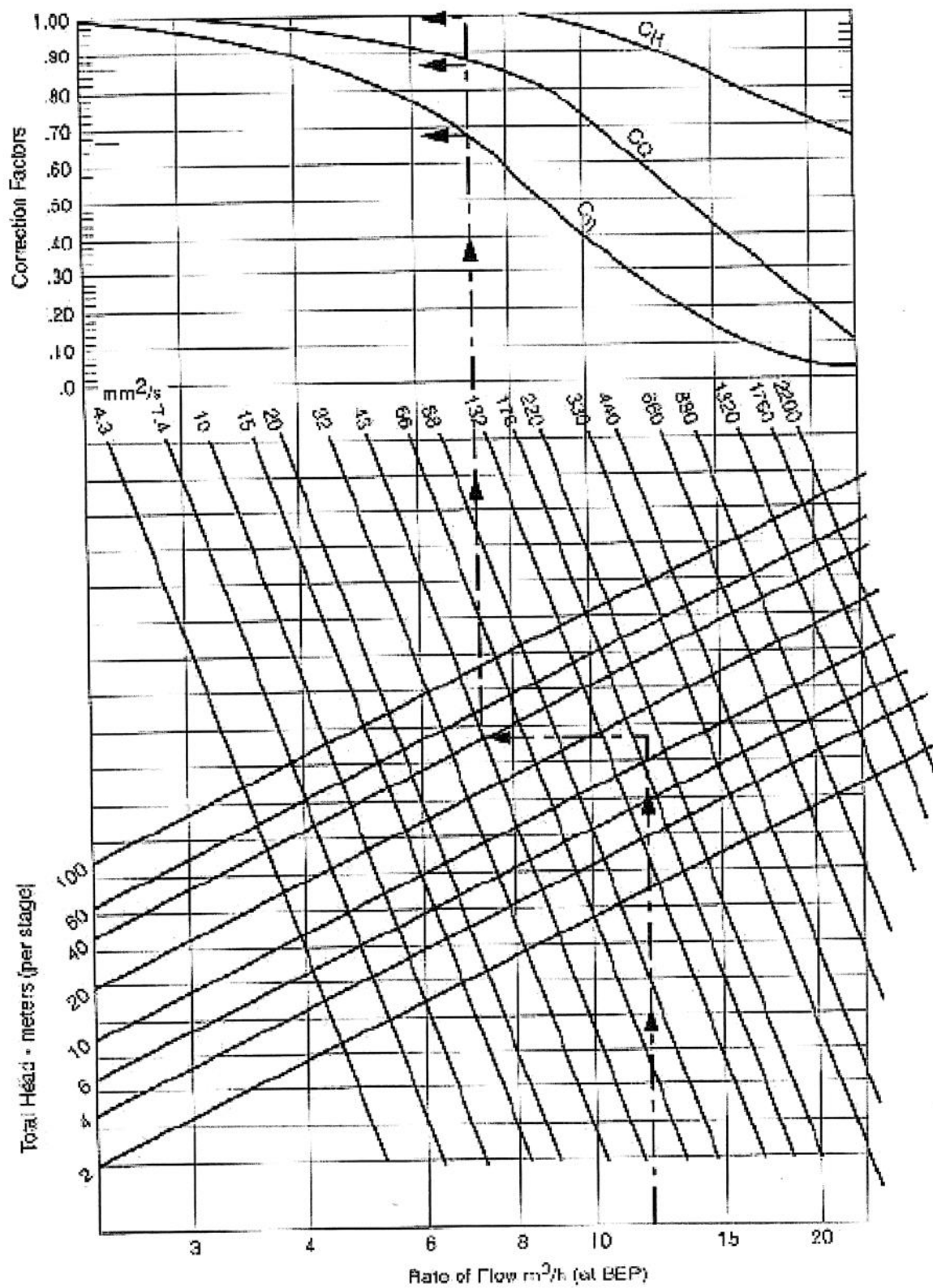


Figure 7 Performance correction chart for viscous liquids (metric)

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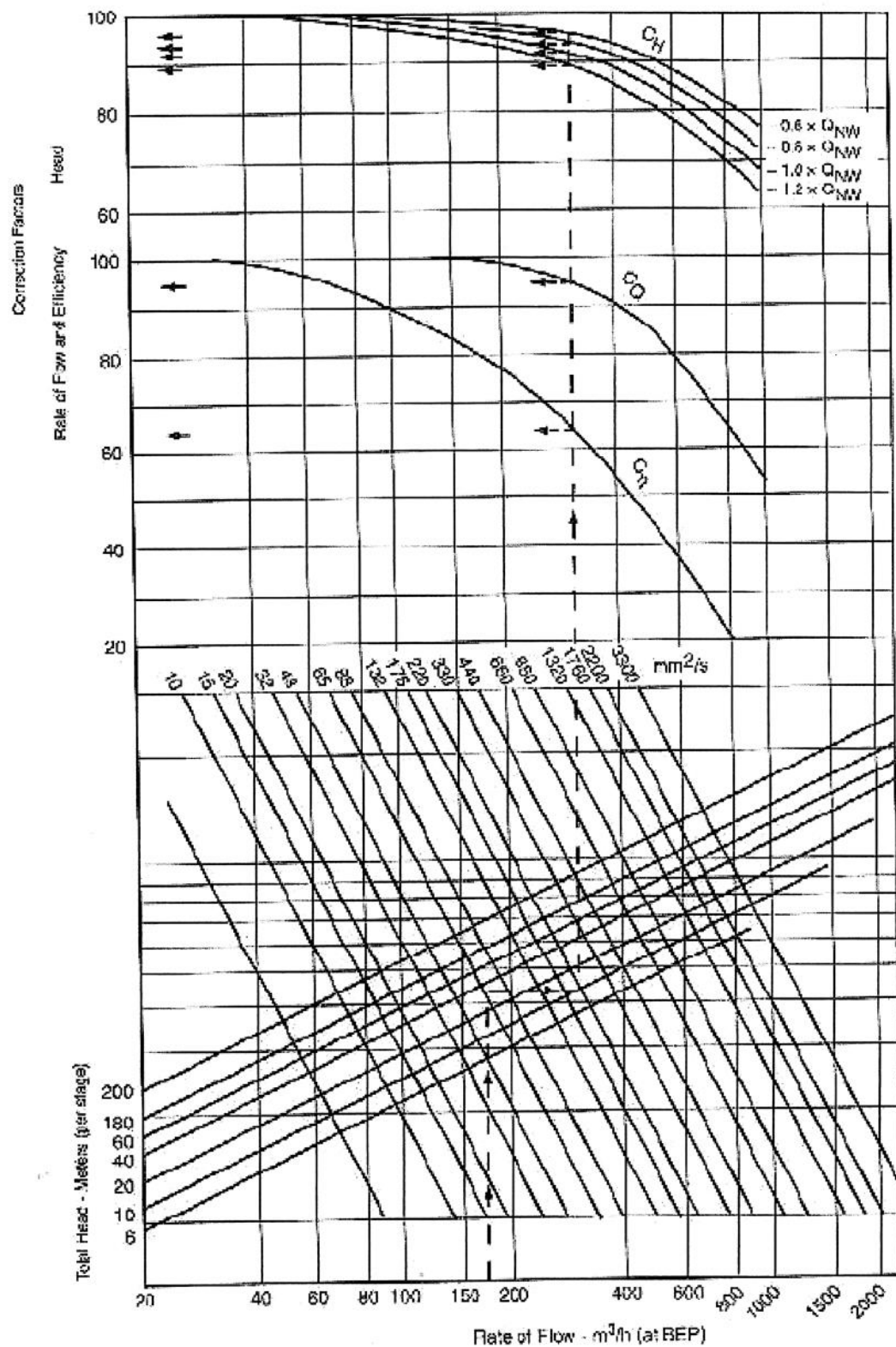


Figure 8 Performance correction chart for viscous liquids (metric)

5. HORIZONTAL OR VERTICAL

Whether the pumpset is to have a horizontal or vertical shaft axis is often decided by civil engineering considerations alone, that is, the type of pump house suitable for the location. Nevertheless, there are factors, which depend on the pumpset design and operation.

Firstly, consider the type of pump. Most low specific speed pumps have horizontal shafts, whereas most high specific speed designs are of the vertical axis, submerged pump type. Obviously for applications such as borehole pumping the orientation and layout are well defined in advance.

Secondly, consider the pump house design. For some types of submersible pump there will not be a pump house as such but merely a protective box for power supply connections and controls, hence pump house costs are minimised. For other types of pump, the pump house costs will be a major part of the whole pumping system costs. In general horizontal pumpset layout stations tend to be larger than those for vertical pumpsets and are therefore likely to produce higher pump house costs.

Thirdly, maintenance must be considered. Horizontal sets are easier to work on than vertical sets. Quite often the various sub-assemblies requiring removal for maintenance or replacement can be separately worked on in horizontal installations, where as in a vertical pumpset, everything above the assembly requiring attention has to be lifted.

Fourthly, the NPSH can often be maximised by using a suspended vertical axis pump arrangement. This allows a higher pump speed and hence the smallest pumpset size for the application.

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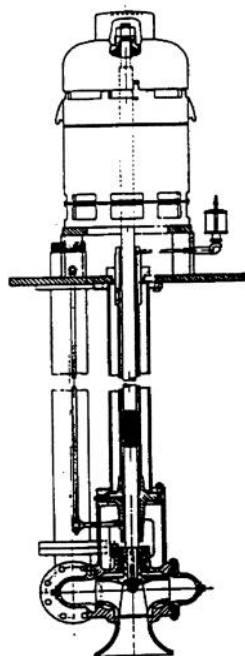


Figure 9 Vertical wet

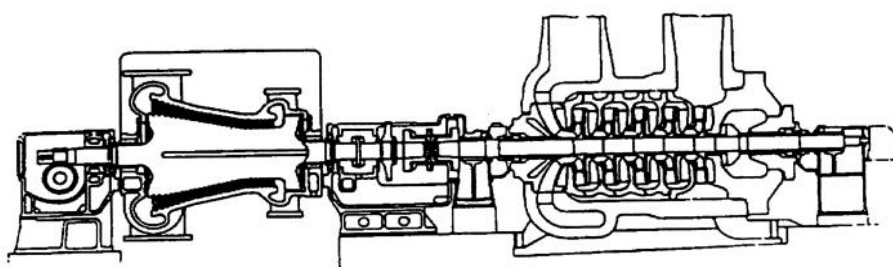


Figure 10 Horizontal

6. DRIVERS FOR INDUSTRIAL PUMPS

It is possible to use any form of rotary or reciprocating output prime mover for industrial pumps but the commonest, most obvious, choices are:

Electric motors	a.c. d.c.
Combustion engines	Petrol Diesel Gas (and Gas Turbine)
Turbines	Steam Water Gas (i.e. power recovery)
Hydraulic motors	

The primary factors in the selection of a pump drive are:

- Available forms of energy
- Cost of energy
- Capital costs of the driver and associated systems
- Security of drive

The a.c. electric motor is by far the most commonly used, due to its availability, standardisation and low cost (resulting from larger scale production than for other industrial drives). Electrical power is widely available in industrial and developing countries over a range of voltages (with suitable local transformers) and powers.

In most of Europe a.c. supply frequency is 50Hz giving a maximum synchronous speed of 3000 rev/min but in the USA and other countries on or near the American continent the frequency is 60Hz, giving maximum synchronous speed of 3600 rev/min.

Factors affecting the choice of type of electric motor are:

Environment	Ambient temperature, altitude, humidity, hazard conditions, local surroundings
Application	Type of driven machine and nature of load, rated duty, type of shaft power transmission system
Electrical power supply conditions	Stability, starting restrictions, fault levels
Maintenance	Accessibility, frequency of planned shutdowns, skill of personnel
Cost	Relative importance of capital vs running cost

D.C electric motors are not in common use now, tending only to be used where variable speed or special characteristics are required. They tend to be relatively large and expensive compared with a.c. machines.

Internal combustion engines tend to be used where electrical power is unavailable or unreliable. Also for portable sets. Engine driven pumpsets are often used for emergency pumping applications e.g. fire-fighting pumps. Both diesel and petrol engines are used, diesels tending to be larger, heavier and slower running, but having the advantage of wide fuel availability at advantageous costs. For water cooled engines the pumped water is often used but contamination levels need to be carefully controlled.

Gas turbines are available for a wide range of powers but tend to be expensive for pump drives. The high operating speed requires a gearbox in most cases and the fuel consumption cost is often uncompetitively high. They have the advantage of being small and light for a given power and they can also burn a wide range of fuels. They are reasonably tolerant of a poor operating environment. The main applications to date have been where fuel is readily available, such as oil or gas pipeline pumping overland or on oil platforms.

Steam turbines offer speeds up to 12,000 rpm and unlimited power for pumping applications but generation of steam in sufficient quantity at modern turbine pressures is becoming rare except in processes which use the steam for other purposes, e.g. power refinery and chemical industries. The economic generation of steam often requires boiler pressures and temperatures much higher than required by the process. By utilising the excess steam energy to drive a pump, expensive and wasteful throttling of control valves can be avoided. The value of the energy saved may well offset the additional cost of the steam turbine, particularly for continuous processes. An additional advantage is that a steam turbine can be used in hazardous (high explosion/fire risk) areas. Unfortunately, it also suffers because of relatively high speed, as for the gas turbine, in that a gearbox is often required for pump drives.

Hydraulic turbines are not often used for pump drives; their use is normally dictated by local conditions. Two examples are:

- Power recovery turbines on reverse osmosis (RO) water purification plants
- Downhole pumps for secondary oil recovery. In this case, where pumpsets are installed thousands of metres underground in a hot aggressive environment, the hydraulic turbine dispenses with the electric motor and its cable which have been found to be very vulnerable to damage during installation and service.

Of course a pump is reversed turbine and vice versa and so there is now an increasing interest in using pumps as low cost pump drives in areas where there is a supply of pressurised liquid. This may sound anomalous but the example of high head; low flow pumping using a high flow, lower head supply illustrates one case.

Hydraulic motors have the advantages of small size for given power and the ability to work in hazardous areas. They are not in common use and are relatively inefficient due to the need to generate and transmit the hydraulic power initially. Nevertheless, for some submersible applications, such as cavern storage pumping, they have been successful.

The main advantage of engine and turbine drives over the common squirrel cage induction motor has in the past been the relative ease with which variable speed and therefore control of pumping rate could be achieved.

7. DRIVE SYSTEMS

All of the prime movers listed can be used in conjunction with combinations of the following drive line components:

- Rigid couplings
- Flexible couplings
 - pin
 - gear
 - flexible
- Belt drives
- Gear boxes
- Clutches
- Fluid couplings
- Magnetic couplings

Rigid couplings are normally used to reduce shaft lengths, for manufacturing or assembly purposes, where accurate alignment can be established. Flexible couplings are used to connect two independent shafts when alignment is more difficult to achieve and maintain. Flexible couplings should tolerate some degree of parallel, axial and angular misalignment but operates best with least transmitted force and least vibration when the initial misalignment is minimised. Some types need to be lubricated, e.g. gear type, and all require regular checks and/or maintenance.

Belt drives are not in common use nowadays but are useful for matching pump/motor speeds, periodic speed change and remote drives. Speed changes are achieved by changing pulley sizes. Belts impose extra bearing loads due to belt tensioning requirements so some care/checking is necessary and occasionally a separate jack shaft is required.

Gear boxes are many and varied. They are commonly used to reduce a high drive output speed to suit a low speed pump (e.g. on a steam or gas turbine driven pump) or alternatively increase the pump speed where the driver speed is limited (eg. 2980 rev/min electric motor driving a small high speed pump). Their other main use is in right angle drive systems, such as engine driven vertical shaft fire pumps. The efficiency of oil lubricated gearboxes is normally high, circa 97/98%, but they are relatively expensive and require additional space and maintenance. An example of the successful integration of a high speed output step up gearbox in a competitive pumpset is the Sundyne pump.

Clutches are normally introduced when either the prime mover is unable to start with the pump load coupled or to allow pump disconnect in drive trains. Occasionally clutches are used for emergency pump isolation. For pumps having dual drives, e.g. electric motor and power recovery turbine, an over-running clutch is used so that the electric motor can drive

the pump alone without the drag of the de-energised turbine. Pumpset starting will be discussed in a later section but if a clutch is used for reasons associated with inadequate starting torque, of an engine say, then a centrifugal engagement type is often used.

Constant filling fluid couplings are often used for reasons similar to those applicable to clutches and to give a soft start facility. Variable filling fluid couplings are used when speed control is required.

Magnetic couplings are used in similar circumstances. They may be of the permanent magnet variety, either synchronous or induction, or separately electrically energised to give speed control. The permanent magnet type is often used for glandless applications as an alternative to the wet or canned electric motor.

8. PUMP STARTING

When considering the starting of a pumpset there are three main considerations.

- a) Will the driver overcome stiction ?
- b) Will the driver produce enough torque to accelerate the set to full speed and how long will it take?
- c) What will the effect on the drive and starting power supply system be?

Stiction is variable and depends on the construction of the pump and its condition. Factors such as gland friction, which is extremely variable, are difficult to estimate accurately. It is therefore good sense to ensure adequate margin between driver zero speed (or breakaway) torque and the estimated friction torque of the pump at zero speed.

Assuming that the driver overcomes stiction then the relationship between motor torque and pump torque at any speed has to be considered. The excess driver torque is the torque which will accelerate the pump and driver and any intermediate drive system components, such as couplings and gearboxes.

$$T = I\alpha$$

where T = excess motor torque

I = total pumpset rotor inertia

α = Instantaneous acceleration rate

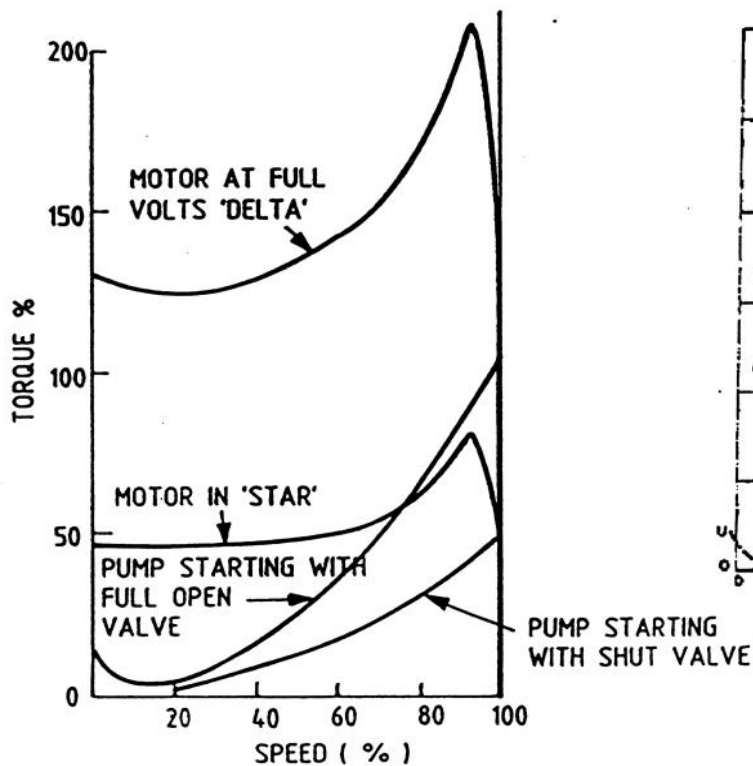


Figure 11 Typical pump and motor speed/torque curves

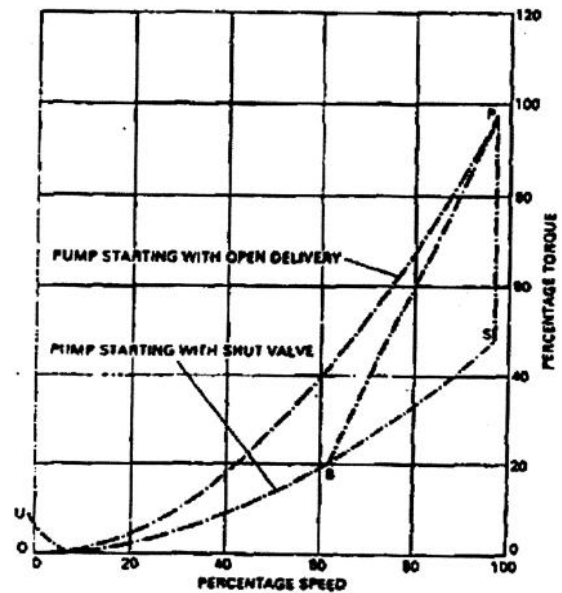


Figure 12 Torque/speed curves for a radial flow pump

(α is the instantaneous acceleration, because, as will be seen on the typical speed torque curve, Fig.11, torque, T , is normally speed dependent.) The torque available from the driver is normally obtained by test (and the data provided by the driver manufacturers).

Pump torque is calculated by matching the pump characteristic to the system characteristic, for the starting sequence and calculating the pump power at each intersection for speeds between zero and full speed.

Torque is then calculated for each speed, Figs. 12,13 and 14, and plotted on the driver speed/torque curve, See Fig 11. If there is excess driver torque available over the whole run of speed range then the pumpset will accelerate.

In order to calculate the run up time, it is necessary to calculate acceleration and speed attained for small increments of time, adjusting the torque for subsequent time increments according to the speed attained at the end of the last. If the excess torque was constant this time stepping calculation would be unnecessary but unfortunately torque normally varies. By adding all the time increments to full speed, the run up time is obtained.

It will be observed that the system characteristic during starting is very important and can, in some applications, be controlled. Generally low specific speed pumps are best started

against closed valve Fig 12 and high specific speeds pumps should always be started with the system open Fig 14. Very long discharge pipes which are empty on start up are undesirable for low specific speed pumps and long discharge pipes which are full are undesirable for high specific speed pumps. This type of problem is generally insignificant for pumps with non-overloading power characteristics (Fig 13).

The ability to start a pumpset depends on the driver characteristics. Consider an internal combustion engine first. Starting the engine with pump load coupled is a bit like operating the starter motor of a car with the car in gear. Success in getting the engine to take up load will depend on the size of the starter motor and the short-term power output of the battery. In many cases it is preferable to use a clutch, possibly of the centrifugal type, so that the engine starter motor starts against the smallest possible load.

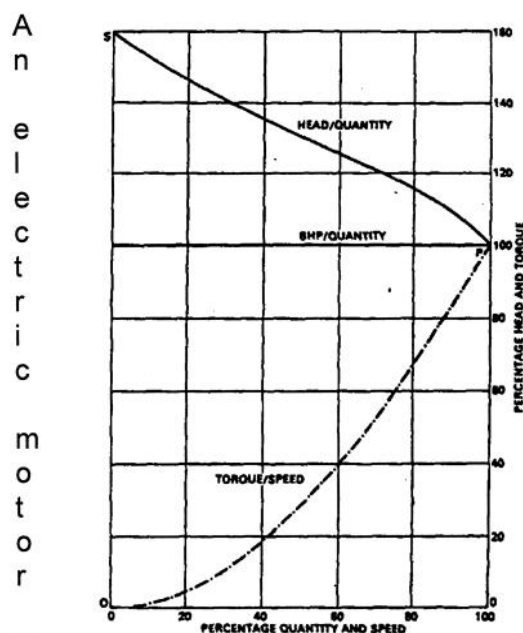


Figure 13 Starting characteristics of a mixed flow pump

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An electric motor presents a different case. Consider a squirrel cage induction machine. The current taken at constant voltage during start up depends on the torque characteristic. In general starting currents are much higher than (3 to 8 times) normal full load current. This level of current is usually unacceptable from a supply point of view, as it would lead to large voltage dips on the supply system, causing external grid problems. It would also lead to a voltage drop at the motor terminals resulting in reduced torque availability from the motor. For these reasons the starting currents drawn are restricted and it is necessary to find a motor starting system which not only gives acceptable pumpset run up but also acceptable starting currents.

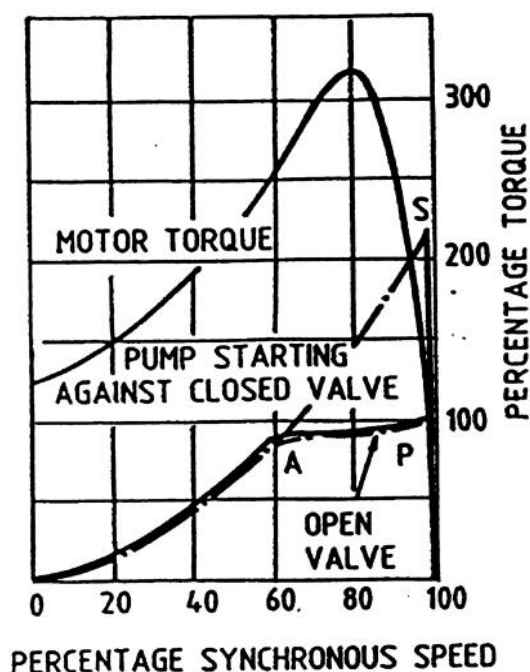


Figure 14 Starting characteristics of an axial pump

Some systems in common uses are:

- Dol
- Star delta
- Auto transformer
- Korndorffer
- Voltage regulating soft starters
- Inverter fed soft starters

Whichever system is used it will add to the cost of the installation – a sophisticated starter can cost almost as much as the basic squirrel cage motor. Note that in the Dol case, disconnecting the pump would have very little effect. Since the motor speed/torque and speed/current characteristics are inherent, the motor torque would merely be used to accelerate the motor faster so the motor run up would be reduced but the currents would be unaffected.

9. PUMPING CAPACITY CONTROL

There are a number of options available:

- Throttling one or more pumps
- On/off (stop/start) control of one or more pumps
- Variable speed drive for one or more pumps
- Bypass systems (sometimes used).

Throttling is by definition a wasteful process – the energy, proportional to dissipated head multiplied by the controlled flowrate, is converted to heat, which in almost all cases is not useful. The method uses an expensive valve which may deteriorate if high rates of energy breakdown are involved. However the method is in very common use due to its simplicity and relatively low capital cost.

On/off control, undesirable for most engineering equipment and pumpsets are no exception. The additional stress and strain during starting reduces the equipment life. For constant speed electric motor driven sets the higher currents required for starting cause increased heating within the motor carcass. If the motor starts frequently then the extra heat input at each start is not dissipated and there is a temperature build up. Eventually this causes trip out or failure to start. If the condition were allowed to persist, the motor insulation would be damaged, causing burn out necessitating an expensive rewind. Motors subject to frequent stop/start therefore need to be carefully selected. An imbalance between required flowrate and pump flowrate is implied by on/off operation and this suggests that liquid storage capacity, e.g. a sump, is necessary.

The third and most effective option is a variable speed drive. A typical variable speed pump characteristic is shown in Fig.15 and a composite multipump variable speed control characteristic is shown in Fig.16. Of course variable speed drives are more expensive in themselves and also need more sophisticated and expensive control equipment.

Arrangements for securing variable speed from fixed frequency a.c. motors can be classified according to whether they involve.

- a) dissipation of power proportional to reduction in speed
- b) no dissipation

The methods falling into these two categories are discussed later. If the method used involves no power dissipation, regulation of output by reduction of speed always yields a power saving comparison with on/off operation, throttling or bypassing, however, investigation is necessary as to whether the accruing saving justifies the extra cost of speed variation provisions.

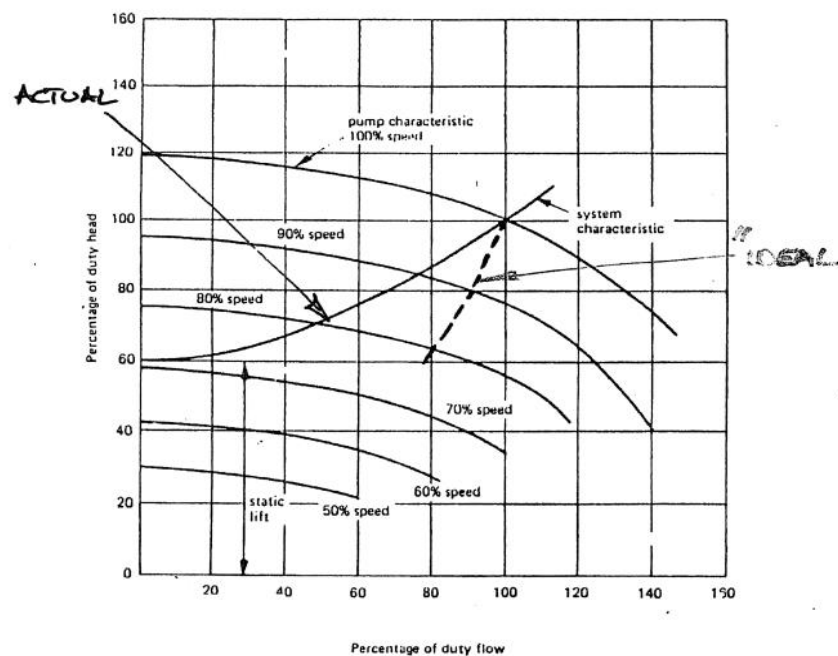


Figure 15

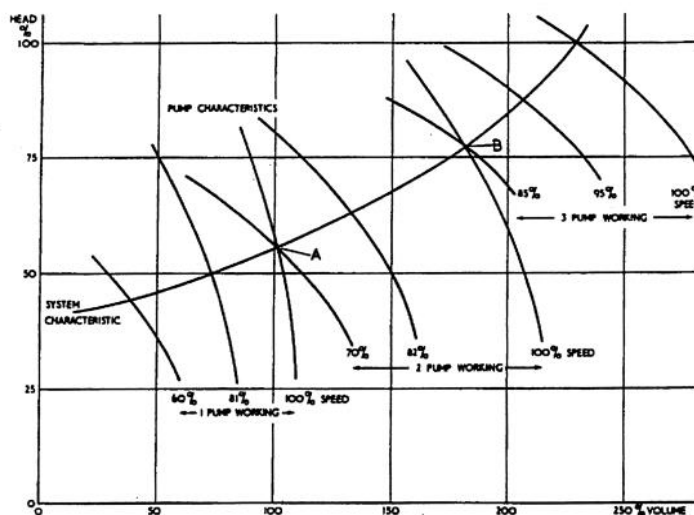


Figure 16 Single

10. VARIABLE SPEED DRIVES

This section is concerned with only variable speed drives from electrical supplies. The means of securing variable speed may be embodied in or associated with the driving motor itself, or be separate.

If separate, they may be either positive or 'slip' type. In the former there is direct mechanical connection between motor and driven unit shafts by means of either gearing or pulleys and belts of some kind, with no intervening medium which can absorb energy, so no dissipation of power or loss of efficiency. However, other than by belt drives, which are severely limited in power capacity, and often inconvenient, stepless speed variation is difficult and expensive to secure and is confined economically to exceptionally large units. Moreover, automation of mechanical positive variable speed drives is complicated.

10.1 "Slip" Couplings

In these there is no positive connection between input and output shafts, so no leverage, and the torque on the output shaft cannot exceed that of the input shaft. The engagement between the shafts is fluid or magnetic, and if there is a drop in speed ('slip') between the two, power is generated and dissipated in the linkage medium, so there is a loss of power and efficiency between input and output.

Slip couplings can be automated fairly readily, the magnetic type perhaps more so than the hydraulic and giving more rapid response.

However, in both this respect and efficiency they are no better, and even marginally less good, than slip-controlled electric motors (see below). Moreover, if substantial power is dissipated in the coupling, the need for artificial cooling may present problems or even be inadmissible, and the magnetic type may be of limited availability for vertical drives.

Advantages of Slip Couplings:

- They provide unlimited and stepless speed variation below maximum.
- They introduce resilience and lessen the transmission of sudden shocks and violent fluctuations in load between driving and driven machines.
- They enable heavy inertia loads to be run up to speed gradually while allowing normal motor run-up conditions.
- They make speed variation available where electrical methods would entail enclosure flame-proofing difficulties.
- They make independently variable speeds available on the same line of shaft.

10.2 Variable Speed Motors

Direct current motors give free choice of speed within practical design limits, and are inherently amenable to speed variation. Alternating current motors have fundamental speeds fixed by the frequency of the supply and the detail of the motor winding.

For synchronous and induction motors (of fixed constant speed and infrequent use) the maximum available speeds are given by the supply frequency in cycles per minute (i.e. cycles per second times 60, viz. 3000 for 50 c/s and 3600 for 60 c/s) divided by the successive integers 1,2 3...etc. The same applies for no-load speeds of normally used squirrel cage or slipring induction motors, but to obtain approximate full load r.p.m allowing (as must be done) for slip, the figures to be divided are respectively about 2900 and 3500.

10.2.1 A.C. Motor Speed Variation Possibilities

a. Pole-change Motors

These depend on the nature of variation required, whether stepless over a range or only certain set speeds.

If only a total of two set speeds are required and both are simple divisions of the appropriate above figure, they can be obtained with a single motor having separate windings for each speed. With corresponding starter provision including changeover rotor connections for slipring motors, which must have two rotor windings to correspond with stator pole changes. Squirrel cage motors (which do not involve rotor pole change windings) can be made pole amplitude modulated for three or four speeds if all are prime factors of the fundamental.

b. Variable Frequency Inverter – induction motor drive

With increased pressure for energy conservation and economic processes, making the pump match the system characteristic to produce exactly what is needed e.g. a flowrate or pressure, is becoming ever more important.

One way to address variable need is variable speed. To ensure that this is technically and economically the right approach requires a lifetime costs analysis, specifically capital costs versus running costs, for the whole system and its projected "demand" patterns for the foreseeable future.

Prior to the introduction of inverter technology for standard squirrel cage induction motor application, obtaining variable speed from an electric motor was a complex and expensive business. This sometimes resulted in non-electric drivers capable of easier speed control being used. Combustion engines with simple automated throttle control fit this bill. However because of costs, noise, and other environmental problems these solutions have never been popular and therefore the advantages of variable speed seldom utilised except on large power, continuous running applications (e.g., water supply, sewage) or critical applications where control is paramount. Perversely a fuel pump often controlled the engine speed!

Inverter technology has now become widely established in areas other than pumping (controlling mill roller or conveyor speeds for example). Advances in inverter technology generally have led to better inverters for pump drives. In fact most "standard" inverters on the market are far too sophisticated for pump and fan drives which represent a very easy motor load and, in general, the simplest of control requirements.

The principal inverter advances benefiting pumping applications are improvement in output waveform, simplified control and programmability.

With early inverters (which essentially take in ac electricity, rectify it and send it back out in pulses of dc configured to simulate ac at selected frequency) the output waveform was poor and caused additional heating effects in the motor. The motor therefore required to be “de-rated” to ensure that its operating temperatures were not excessive.

Because of electrical system dynamics complex harmonics were not only fed to the motor but also back into the supply system. Not unexpectedly this caused problems elsewhere. One such difficulty was pumpset electrical power input measurement. Most conventional analogue and simple digital electrical ‘meters’ are not capable of dealing with non-sinusoidal waveforms and give spurious data when asked to measure such power supplies. (This caused BPMA, BSI and CEN to commission a testing procedure standard especially for inverter-fed pumpsets undergoing guarantee tests).

Due to the improvements in the inverter simulated waveform, and associated control of the flux field in the motor by simultaneous voltage control, (voltage requires to be reduced approximately in line with frequency, hence speed, for ideal motor operation), it is now possible with careful matching to fit smaller rated motors than for fixed speed applications because the usual “safety factors” can be reduced, e.g. if the pump is a little over duty, the speed and hence power can be reduced to nearer the design level.

The combination of solid state power electronics and microprocessor technology has led to much simpler external control inputs and self-regulating motor inverter systems. The so-called sensorless inverter (it really means there is no shaft speed encoder) could, when matched to suitable pump and motor characteristics, potentially lead to some completely automated sensorless systems, i.e. no external measurements to feed back the inverter responding to changes in pump load. This could apply for example to systems with changes in static head or pressure, which normally tend to cause a change in load.

There has been such an increase in the demand for integrated motor/inverter systems that motors with inverters physically incorporated are now available. This had led to a drastic reduction in inverter space – the volume of inverter cubicles having been significantly large for cooling purposes. Integrated inverters now often use motor cooling fan air for forced cooling. It might be possible to use water in some applications. Motor cooling at the very low speed/frequencies attainable with inverters has been a concern in the past, to the extent that either minimum speeds have been applied or alternatively an independently powered cooled fan fitted to the motor. (The motor cooling fan is usually fitted to the motor shaft and hence reduces output i.e., cooling, as the motor speed drops.) It is often unnecessary to go to such low speeds in pumping applications.

In fact it can be dangerous if the speed is reduced such that the pump zero flowrate head (closed valve head) is reduced below the system static head or pressure. In these circumstances the pump will not “deliver” the energy it receives from the motor. Heat will build up according to the normal thermodynamic laws. This situation is doubly dangerous with an electro submersible pump where the motor often relies on pump flow to cool it.

In conclusion inverters should receive at least cursory consideration for inclusion in all pumping systems. There are many that would benefit from simple, reliable variable speed technology. As stated in the beginning the ultimate test is an economic one – do not be put off by the seemingly high first cost of the inverter and associated ancillaries – several manufacturers are offering lower cost, simpler, options for pump drives.

11. POWER FACTOR 'CORRECTION'

Electric supply tariffs often penalise low lagging power factor, because it costs a reduction in effective capacity of transmission and distribution systems, and has other undesirable consequences: conversely, therefore, there is a premium on improvement of power factor. This may be secured by

- a) Connecting static condensers across the motors.
- b) Commutator motors (these may be designed to give a high p.f. either alternatively – or in some cases additionally – to speed variation.
- c) Synchronous induction motors – only applicable to large constant speed long running machines with high value on power factor

12 USEFUL DATA

Several tables are attached, each giving electrical data useful in selecting (or designing) and operating electrical pumping plant.

kW	No of poles	Supply frequency, Hz	
		50	60
0.75 to 2.2	2 4 6	2820 1405 930	3380 1685 1115
3 to 7.5	2 4 6	2855 1420 945	3425 1700 1130
11 to 22	2 4 6 8	2910 1445 960 720	3490 1730 1145 860
30 to 75	2 4 6 8	2935 1475 975 735	3520 1770 1170 880

Table 1 Approximate full load speeds of squirrel cage

Method of starting	Starting torque (approx) % Full load torque	Starting current (approx) % Full load current
Direct	100% - 200%	350% - 700%
Star delta	33% - 66%	120% - 230%
Series parallel	25% - 50%	90% - 175%
Auto transformer	25% - 85%	90% - 300%

Table 2 Starting squirrel cage motors.

The above figures apply to Squirrel Cage motors of normal design and other types are available namely:

- High Torque Squirrel Cage machines will give approximately twice the above starting torques with unrestricted currents.
- Low current Squirrel Cage machines restrict the current but give a lower starting torque than the High Torque machines.

These two types can now be used in many cases where Slipring machines would have been necessary in the past.

- Slipring machines. All Slipring machines must be started by means of a rotor resistance starter. A starting torque of full load torque is obtainable with a starting current of approximately 1.25 full load current, this usually being sanctioned by Supply Authorities for any size of motor.

1/s	1	2	3	5	10	20	30	50	100	200
Average Efficiency %	34	44	50	58	66	72	75	78	81	83
Total head m										
5	0.15	0.23	0.30	0.43	0.75	1.47	1.96	3.15	6.05	11.8
7	0.20	0.31	0.41	0.59	1.04	1.91	2.75	4.40	8.48	16.6
10	0.29	0.45	0.59	0.85	1.49	2.73	3.92	6.30	12.1	23.6
20	0.58	0.89	1.18	1.69	2.97	5.45	7.84	12.6	24.2	47.3
30	0.87	1.34	1.77	2.54	4.46	8.18	11.8	18.9	36.3	71
50	1.44	2.23	2.94	4.23	7.43	13.6	19.6	31.5	60.5	118
70	2.02	3.12	4.12	5.92	10.4	19.1	27.5	44.0	84.8	166
100	2.88	4.46	5.88	8.45	14.9	27.3	39.2	63.0	121	236
200	5.77	8.92	11.8	16.9	29.7	54.5	78.4	126	242	473
300	8.65	13.4	17.7	25.4	44.6	81.8	118	189	363	710
500	14.4	22.3	29.4	42.3	74.3	136	196	315	605	1180

Table 3 Approximate KW absorbed in pumping water

The above average efficiencies are for one duty only with the best pump selected for that duty. These figures can be improved by using high efficiency pumps designed for the actual duty but allow reasonable variations obtainable from having a large range of sizes to give a good selection.

kW	Efficiency %			Power Factor			Full load amps on 3ph 415 v
	Full load	.75 load	.5 load	Full load	.75 load	.5 load	
0.75	74	73	69	0.72	0.65	0.53	2.0
1.5	79	78.5	76	0.83	0.78	0.69	3.2
3	82.5	82	80.5	0.85	0.80	0.73	6.0
5.5	84.5	84.5	83.5	0.87	0.87	0.75	10.5
7.5	85.5	85.5	84.5	0.87	0.83	0.76	14
11	87	87	85.5	0.88	0.84	0.77	20
18.5	88.5	88.5	87	0.89	0.85	0.79	33
30	90	89.5	88	0.89	0.86	0.80	52
45	91	90.5	89	0.89	0.86	0.80	77
75	92	91.5	90	0.90	0.87	0.81	126

Table 4 Typical Electric motor efficiencies and power factors.

13. APPENDICES

Appendix 1 Useful Electrical Formulae

To find the output kW of motors when volts, current, efficiency and power factor (PF) are known.

For three phase

$$kW = \frac{\text{Volts} \times \text{amps} \times \text{eff} \times \text{PF} \times 1.73}{1000 \times 100}$$

To find amperes to be carried by cable connections to a motor when output kW, volts, efficiency and power factor (PF) are known.

For three phase

$$\text{amps per phase} = \frac{kW \times 1000 \times 100}{\text{Volts} \times \text{eff} \times \text{PF} \times 1.73}$$

Appendix 2 Notation and units

D_2	Impeller tip diameter	m
H	Total Head	m
I	rotational inertia	kg m ²
N	Rotational speed	rev/min
NPSH	Net positive suction head	m
P	Power input	kW
Q	Flowrate	m ³ /s
T	Torque	Nm
U_2	Impeller tip speed	m/s
g	Gravitational acceleration	m/s ²
α	Angular acceleration	rad/s ²
η_{oa}	Pump overall efficiency	
ρ	Liquid density	kg/m ³

Match Driver and Driven Machine

- Steady State
- Load Changes
- Starting
- Other Transients eg, load loss

POWER

$$kW = \frac{\rho g H Q}{\eta_{pa} \times 1000}$$

$\rho_{\text{water}} \approx 1000 \text{ kg/m}^3$ { other! - }
 $g = 9.81 \text{ m/s}^2$ { CORRECT }
 $\eta_{pa} = \text{Pump Effy decimal}$
 eg., 0.8
but Correct for viscosity
 if $>> 1.0 \text{ cSt}$

Approximate kW absorbed to pump clean water

1/s	1	2	3	5	10	20	30	50	100	200
Average Efficiency %	34	44	50	58	66	72	75	78	81	83
Total head m										
5	0.15	0.23	0.30	0.43	0.75	1.47	1.96	3.15	6.05	11.8
7	0.20	0.31	0.41	0.59	1.04	1.91	2.75	4.40	8.48	16.6
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50	1.44	2.23	2.94	4.23	7.43	13.6	19.6	31.5	60.5	118
70	2.02	3.12	4.12	5.92	10.4	19.1	27.5	44.0	84.8	166
100	2.88	4.46	5.88	8.45	14.9	27.3	39.2	63.0	121	236
200	5.77	8.92	11.8	16.9	29.7	54.5	78.4	126	242	473
300	8.65	13.4	17.7	25.4	44.6	81.8	118	189	363	710
500	14.4	22.3	29.4	42.3	74.3	136	196	315	605	1180

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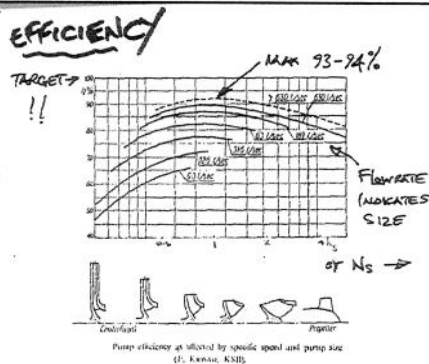


Fig. 1 P. 18

$$N_s = \frac{N \sqrt{Q}}{H^{3/4}} = \text{"SPECIFIC" SPEED}$$

K_s IS NOT DIMENSIONAL FORM

Speed

- Fixed or Variable
- Optimal or Maximum Control
- Pumps – Type, Efficiency, Cavitation
- Compressors – Type, Velocity, choking

PUMP ^{NB} SUCTION SPECIFIC SPEED

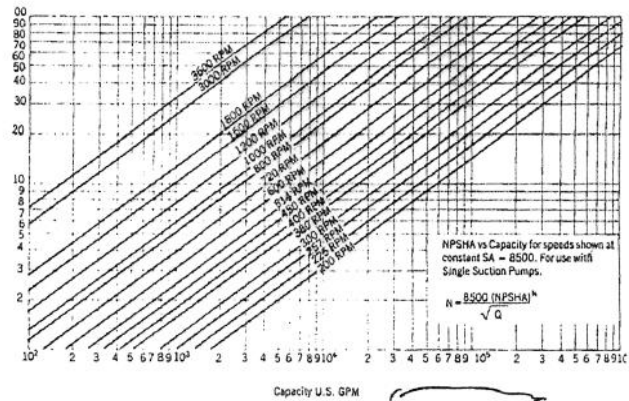
$$N_{ss} = \frac{N \sqrt{Q}}{NPSH^{3/4}}$$

IF Q FLOWRATE
NPSH
AND N_{ss} ARE KNOWN

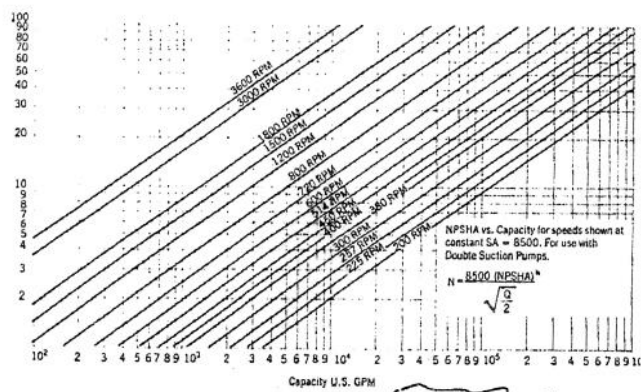
$$\text{THEN } N \text{ (RPM)} = \frac{N_{ss} \times NPSH^{3/4}}{\sqrt{Q}}$$

RUN FAST - MAKE SMALL
BUT WEAR, NOISE & VIBRATION

HYDRAULIC INSTITUTE



RECOMMENDED MAXIMUM OPERATING SPEEDS FOR SINGLE SUCTION PUMPS



RECOMMENDED MAXIMUM OPERATING SPEEDS FOR DOUBLE SUCTION PUMPS

Duty : $Q, H, NPSH$
 SELECT : N $N = \frac{60 K_{RS} (\sqrt[3]{NPSH})^{3/4}}{2\pi \sqrt{Q}}$
 SELECT : K_{RS} $K_{RS} = \frac{2\pi N \sqrt{Q}}{60 (\sqrt[3]{H})^{3/4}}$

A
OPTIMUM
EFFICIENCY

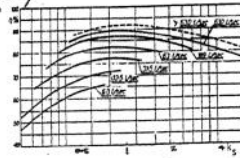


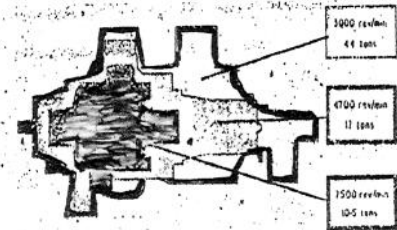
FIG. 1



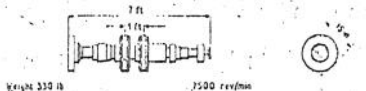
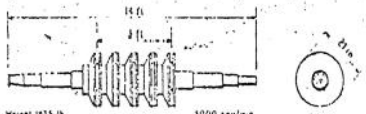
Pump efficiency as affected by specific speed and pump size
 (1) KENNEDY, K270

RUN FAST - MAKE SMALL - PUMP - DRIVER

FACTORS $\therefore$ SIZE/SPACE/WEIGHT
 EFFICIENCY
 CAVITATION

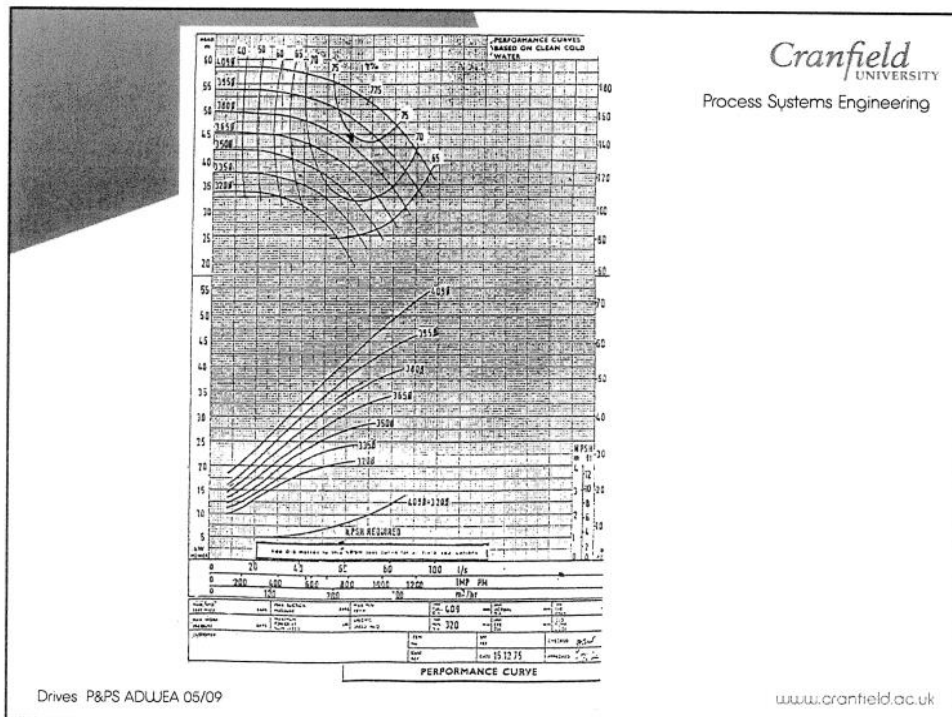
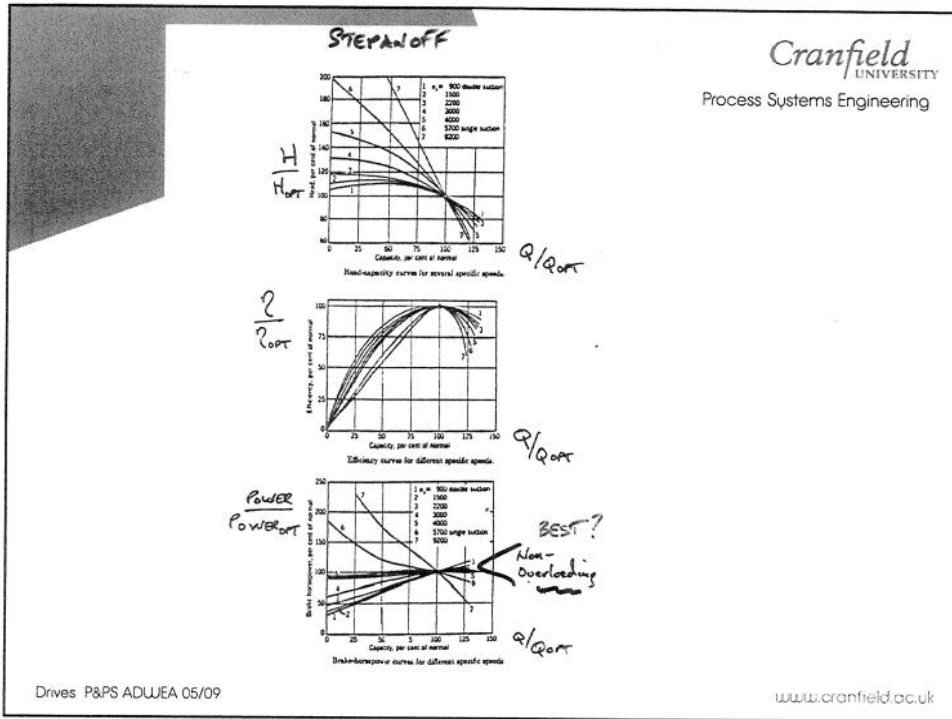


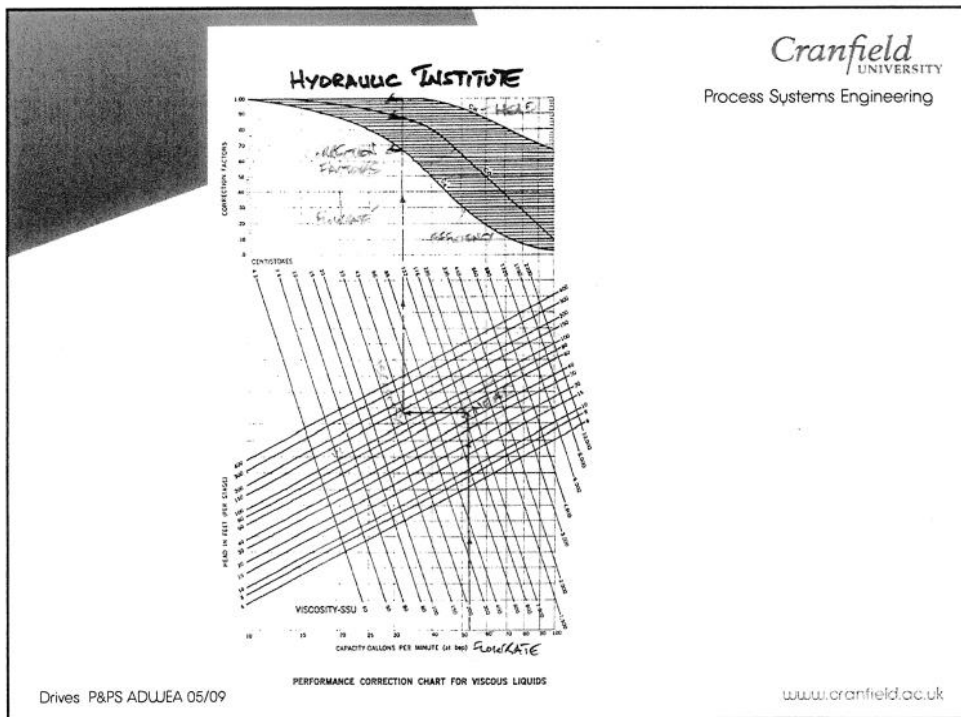
Reduction in pump size and weight consequent upon increasing rotational speed



Comparison of rotating elements of high-speed and low-speed pumps

M. H. ANDERSON





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Drivers

Primary Selection Factors

- Available Energy
- Cost of Energy
- Capital Cost
- Security

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Drivers

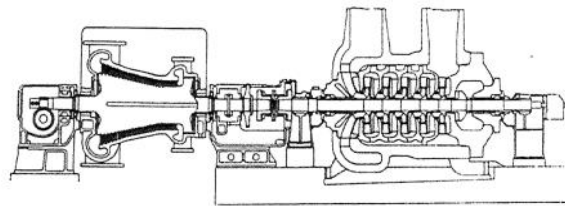
Electric Motors – ac
dc
Combustion Engines – Petrol
Diesel
Gas
Gas
Turbine
Turbines – Steam
Water
Gas
Hydraulic Motors – Air
Hydraulic

Rating Drivers

- Inaccuracy in Estimating
- Operating Differences
- Operating Changes
- Overspeed
- Safety Valve Rating
- Starting
- Service Factors

ARRANGEMENT

HORIZONTAL AXIS



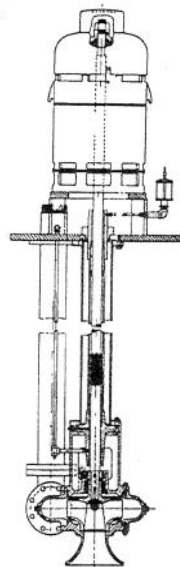
Steam turbine and pump for 550 MW unit, (Weir)

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ARRANGEMENT

VERTICAL
AXIS



Section of vertical wet-pit, nonclogging pump, (Warrington Pump)

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Capacity Control

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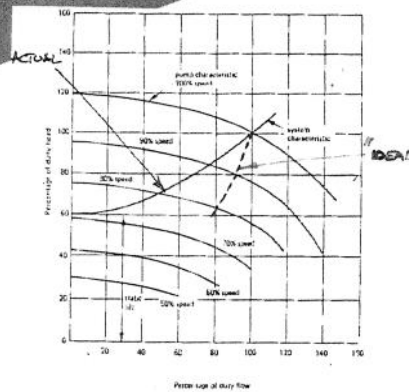
- Throttling
- Stop/Start (Sump)
- Bypass

VARIABLE SPEED?

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Variable speed pump characteristics

AFFINITY LAWS

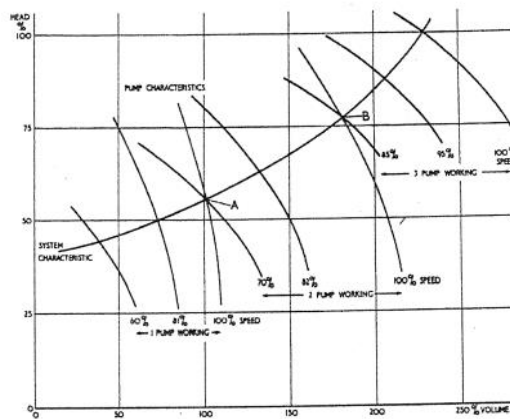
$Q \propto N$ $H \propto N^2$

@ CONSTANT EFFICIENCY

$P (KW) \propto N^3 \therefore T \propto N^2$

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Integral Motors

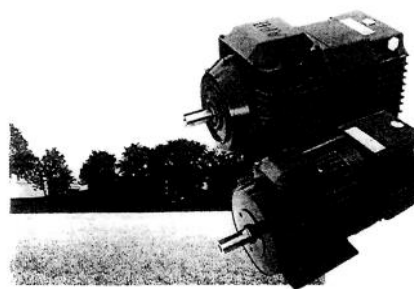


ABB Motors

ABB

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With the VLT® DriveMotor, you
won't need extra hardware

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The DriveMotor is a compact, integrated drive unit with a built-in motor and control unit. It is ideal for use in a wide range of applications.

The motor is rated:

- Noisy: 60-70 dB(A) at 1m
- For ambient temperatures up to 40°C
- For ambient humidity up to 95% RH

The DriveMotor is a compact, integrated drive unit with a built-in motor and control unit. It is ideal for use in a wide range of applications.

- The DriveMotor is a compact, integrated drive unit with a built-in motor and control unit. It is ideal for use in a wide range of applications.
- The DriveMotor is a compact, integrated drive unit with a built-in motor and control unit. It is ideal for use in a wide range of applications.

The DriveMotor VLT® series range now available from 0.37 to 100 kW

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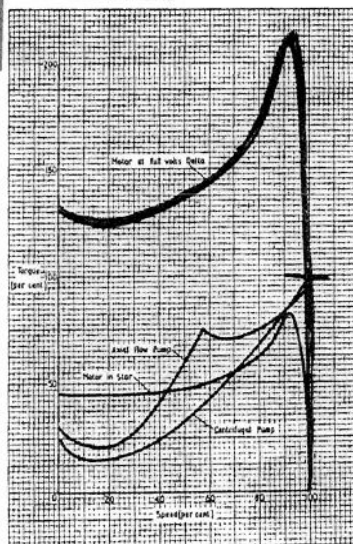
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Pump set
Starting

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A percentage speed/torque curve for axial-flow and centrifugal pumps related to induction motor characteristics

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TABLE 2
Starting - Squirrel Cage Motors

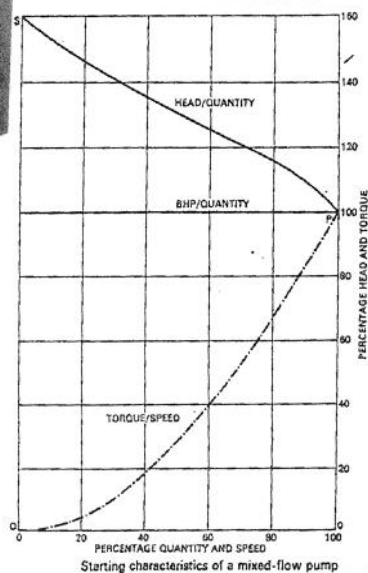
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Auto transformer	25% - 85%	90% - 300%

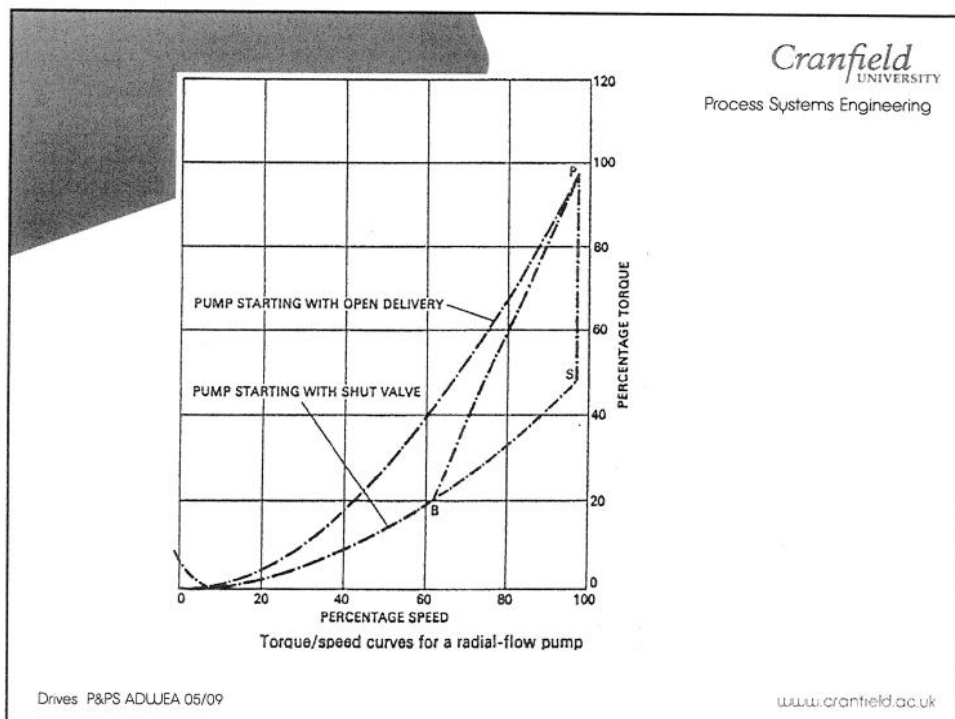
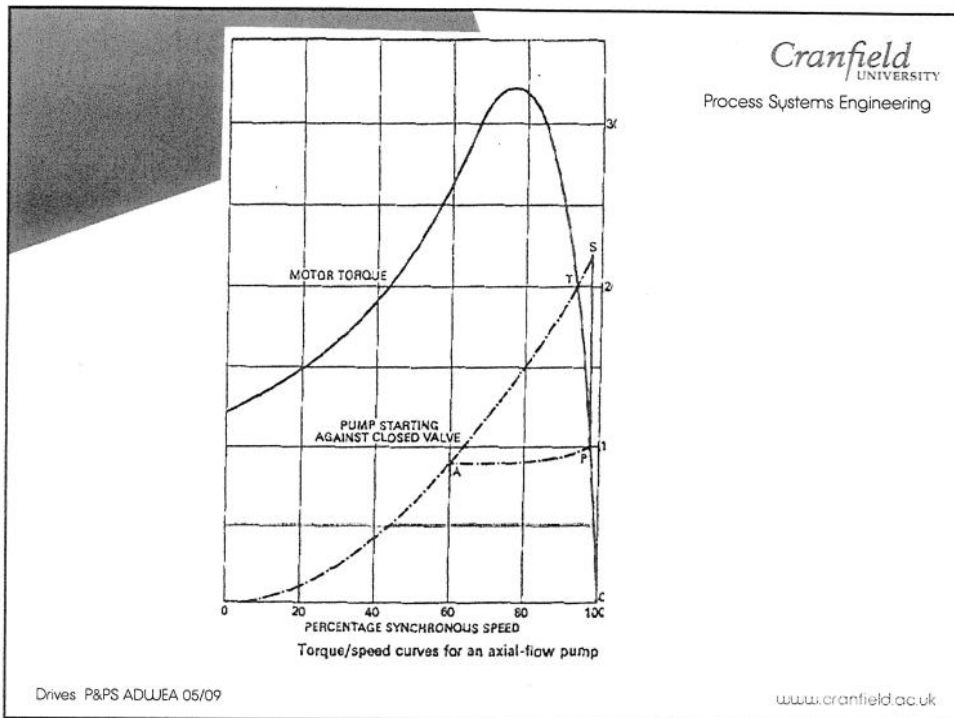
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Slipring machines. All Slipring machines must be started by means of a rotor resistance starter. A starting torque of full load torque is obtainable with a starting current of approximately 1.25 full load current, this usually being sanctioned by Supply Authorities for any size of motor.





Engine Starting

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Electric Start

- Size of Starter Motor
- Batteries
- Back-up
 - Hand crank (barring and valve lift)
 - Hydraulic
 - Air
- Clutches
 - Need?
 - Hand or
 - Auto (Centrifugal)

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SPECIFICATION AND TESTING

Mr Tony Salisbury

SPECIFICATIONS? SAFETY QUALITY CONTRACTUAL

1. WHY SPECIFICATIONS AND STANDARDS?

- Define Need
- Basis for Contract
- Set the Benchmarks
- Relate Statutory Requirements
- Reference other Relevant Specifications and Standards
- Define Acceptance Criteria

2. WHERE DO THEY COME FROM?

- International Law
- Government
- Regulatory Authorities
- Voluntary (Industry) Organisations
- The Customer
- His Consultants and Contractors
- Inspecting and Approvals Bodies

In the form of:

- EC Directives
- Safety
- The CE Mark
- ISO 9000 Quality
- Technical Standards
 - ISO
 - CEN
 - BS
 - API etc
-
- Industry Standards – Self Regulation & Benchmarks
- Project Specific

3. WHAT DO THEY MEAN FOR THE ENGINEER?

- Design and Construction Requirements
- Data Sheets
- Testing

4. TYPICAL PUMP STANDARDS

- ISO13709 (API610) Hydrocarbon Processing
- ISO5199 Class 2 Pumps
- ISO9906 Testing Rotodynamic Pumps

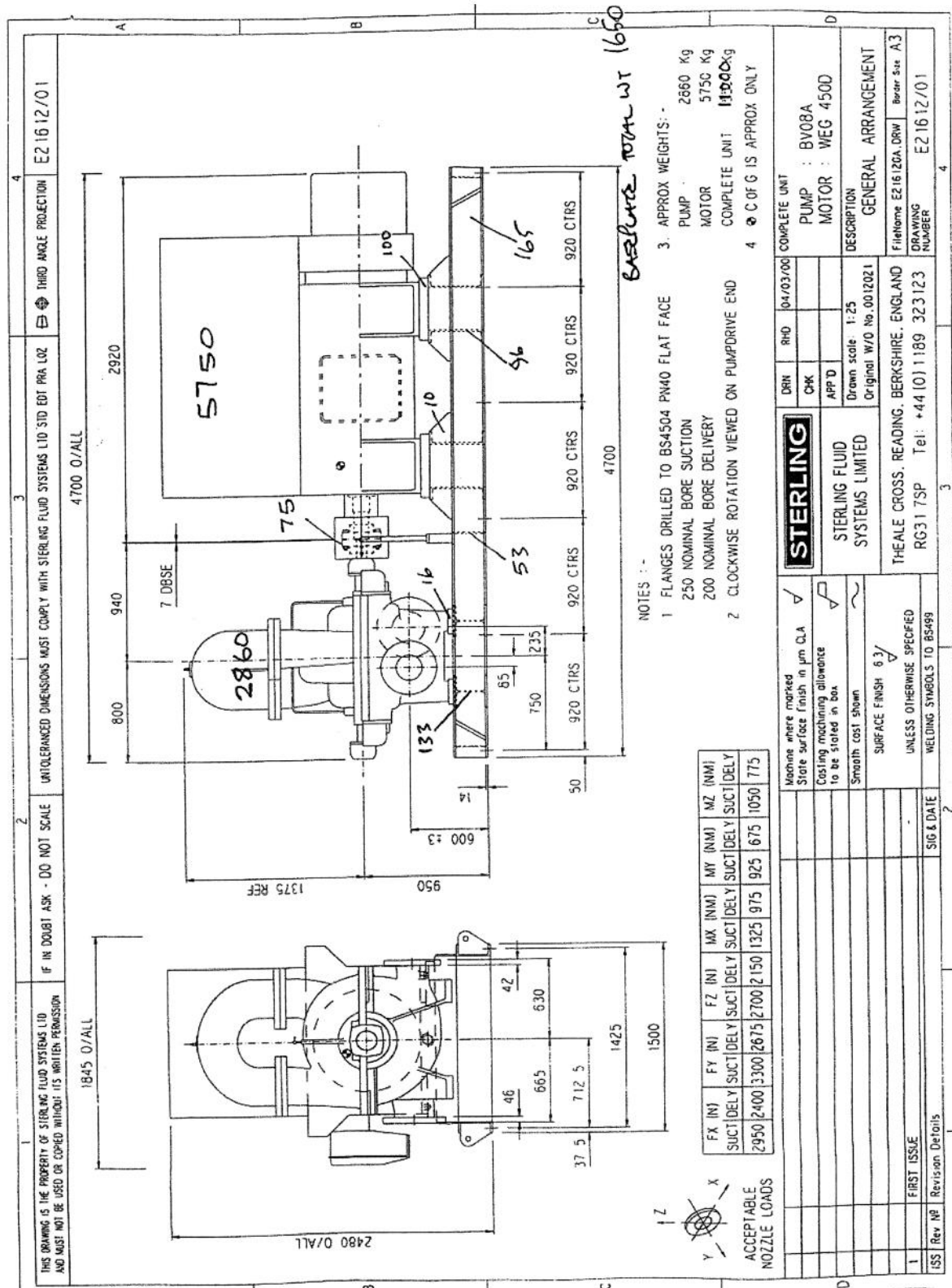
5. CONSIDERATIONS

- The Customer will not PAY if HE thinks you do not comply with his requirements
- You go to Jail without passing GO or collecting £200 if you knowingly break Safety Rules .

WATER INDUSTRY MECHANICAL AND ELECTRICAL SPECIFICATIONS

The Water Industry Mechanical and Electrical Specifications (WIMES) define the requirements for mechanical and electrical equipment used in the Water Industry. The specifications take the form of descriptive text coupled with data sheets.

WIMES	TITLE	ISSUE	DATE
1.01	Horizontal Axially Split Casing Pump for use in the Water Industry	4	July 99
1.02	Submersible Sewage Pump for Use in the Water Industry	3	Jan 96
1.03	Dry Well Mounted Sewage Pump for Use in the Water Industry	1	Apr 97
1.04	Progressing Cavity Pumps for Use in the Water Industry	1	Mar 98
1.06	Positive Displacement Pumps for Sludge Pumping in the Water Industry	1	Jan 99
2.01	Rotating Half-bridge Scrapers for Use in the Water Industry	1	Apr 97
3.01	Low Voltage Switchgear and Controlgear Assemblies for Use in the Water Industry	1	Aug 96
3.02	Low Voltage Electrical Installations	1	Oct 98
3.03	Low Voltage AC Motors for Use in the Water Industry	1	Mar 98
3.04	Electrical Requirements for Package Plant	1	July 99
4.01	Protective Coatings for Use in the Water Industry	1	May 96
5.03	Sewage Screens for Use in the Water Industry	1	May 99
5.04	Overflow Screens for Sewerage Systems and Sewage Treatment Works	1	Nov 99
6.01	Screenings Transfer Equipment	1	Sept 97
6.02	Screenings Treatment Equipment	1	Jan 98



CONTENTS

1.0 SCOPE

2.0 GENERAL

3.0 PROCESS INFORMATION

3.1 Location and Operating Environment

3.2 Properties of Pumped Liquid

4.0 PERFORMANCE INFORMATION

4.1 Mode of Operation

4.2 Hydraulic Performance, Efficiency and Adsorbed Power

4.3 Life and Maintainability

5.0 DESIGN INFORMATION

5.1 General

5.2 Mounting Arrangements

5.3 Casing

5.4 Impeller(s)

5.5 Pump Shaft

5.6 Seals

5.7 Rolling Element Bearings and Bearing Lubrication

5.8 Intermediate Drive Shaft

5.9 Couplings

5.10 Motors

5.11 Lifting Points

5.12 Information Plate

5.13 Guarding

5.14 Electrical Control Panel and Associated Electrical Installation

6.0 CONDITION MONITORING

7.0 TESTS

7.1 Test Regime

7.2 Test Details

APPENDIX A - DATA SHEET EXPLANATION

APPENDIX B - DATA SHEET

1.0 SCOPE

1. This Specification defines the requirements for the performance, design, condition monitoring and testing of horizontal and vertical, axially split casing, single and double entry pumpsets (including motor, driveshaft and baseplate) for flooded suction, water pumping duties in the Water Industry in the form of descriptive text and a DATA SHEET.
2. The number of pumpsets required shall be as specified on the DATA SHEET.

2.0 GENERAL

2.1 Definitions

1. The following definitions are used in this Specification:
 - a) The term 'Purchaser' shall mean the end user of the pumpset *or* the end user's nominated representative;
 - b) The term 'approved' shall mean 'approved by the Purchaser or his nominated representative'; and
 - c) Unless otherwise specified elsewhere (e.g. in a particular specification), the term 'Supplier' shall mean the party responsible for the design, construction, inspection and testing of the pumpset.

3.0 PROCESS INFORMATION

3.1 Location and Operating Environment

1. The location of the pumpset shall be as specified on the DATA SHEET.
2. Any special hazards associated with the operating environment shall be as specified on the DATA SHEET.
3. Unless otherwise specified on the DATA SHEET, the pumpset shall be capable of satisfactory operation within the temperature range -10 to 40°C and up to a relative humidity of 80 %.

3.2 Properties of the Pumped Liquid

1. The properties of the pumped liquid shall be as specified on the DATA SHEET.

4.0 PERFORMANCE INFORMATION

4.1 Mode of Operation

1. The mode of operation, type of duty and type of drive shall be as specified on the DATA SHEET.

4.2 Hydraulic Performance, Efficiency and Adsorbed Power

4.2.2 Fixed Speed Drives

1. Unless otherwise specified on the DATA SHEET, the maximum operating speed shall be 1500 rpm. The actual operating speed shall be as specified/stated on the DATA SHEET.
2. The hydraulic performance at the guaranteed duty point (GDP) shall be as specified on the DATA SHEET.
3. The reference for the system curve(s) (typically comprising H and NPSH_r vs. Q curves) shall be as specified on the DATA SHEET. The maximum and minimum static heads shall be as specified on the DATA SHEET. The pumpset shall be capable of continuous operation at all potential duty points.
4. Characteristic curves of pumpset head, efficiency, power (bare shaft) and NPSH required (NPSH_r) versus flow rate shall be supplied with the Tender.
5. The flow rate at the GDP shall be between X % and Y % of the best efficiency flow rate. X and Y shall be as specified on the DATA SHEET
6. The pump efficiency and adsorbed power (bare shaft) at the GDP and maximum flow rate (see clause 7) shall be as stated on the DATA SHEET.
7. The maximum flow rate shall be as stated on the DATA SHEET. This will correspond to the flow rate at which the pumpset H vs. Q curve intersects the lowest system H vs. Q curve.
8. The minimum flow rate that the pumpset is capable of operating continuously at without causing a significant temperature rise of the pumped liquid and/or significant vibration of the pumpset shall be as stated on the DATA SHEET.
9. If specified on the DATA SHEET, the pumpset shall be capable of discharging into an empty main.
10. The pumpset shall be capable of operating against a closed valve. The closed valve head and maximum allowable operating time at closed valve conditions shall be as stated on the DATA SHEET.
11. The NPSH available (NPSH_a) at the GDP and the NPSH required (NPSH_r) at both the GDP and the maximum flow rate (see clause 7) shall be as specified/stated on the DATA SHEET.

NPSH_r shall be taken as the 3 % head drop requirement. The pumpset shall be selected such that the NPSH_a always exceeds the NPSH_r (including the relevant safety margin) at all potential duty points.

At the maximum flow rate (see clause 7), the margin of NPSH required above NPSH_r for continuous operation without causing excessive pump noise, vibration, or damage due to cavitation shall be as stated on the DATA SHEET.

12. If a multi-stage pumpset is required/to be provided, the number of stages shall be as specified/stated on the DATA SHEET.

4.2.3 Dual/Variable Speed Drives

1. Unless otherwise specified on the DATA SHEET, the maximum operating speed of the pumpset shall be 1500 rpm. S1 and S2 shall be as specified/stated on the DATA SHEET ($S2 > S1$).
2. The hydraulic performance at the guaranteed duty point (GDP) associated with S2 shall be as specified on the DATA SHEET.
3. The reference for the system curve(s) (typically comprising H and $NPSH_r$ vs. Q curves) shall be as specified on the DATA SHEET. The maximum and minimum static heads shall be as specified on the DATA SHEET. The pumpset shall be capable of continuous operation at all potential duty points.
4. Characteristic curves of pumpset head, efficiency, power (bare shaft) and $NPSH_r$ required ($NPSH_r$) versus flow rate at S1 and S2 shall be supplied with the Tender.
5. The flow rate at the GDP associated with S2 shall be between X % and Y % of the best efficiency flow rate at S2. X and Y shall be as specified on the DATA SHEET.
6. The pump efficiencies and adsorbed powers (bare shaft) at the GDPs and maximum flow rates associated with S1 and S2 (see clause 7) shall be as stated on the DATA SHEET.
7. The maximum flow rate at S2 shall be as stated on the DATA SHEET. This will be the flow rate at which the pumpset H vs. Q curve at S2 intersects with the lowest system H vs. Q curve.
8. The minimum flow rate that the pumpset is capable of operating continuously at (at S2) without causing a significant temperature rise of the pumped liquid and/or significant vibration of the pumpset shall be as stated on the DATA SHEET.
9. If specified on the DATA SHEET, the pumpset shall be capable of discharging into an empty main.
10. The pumpset shall be capable of operating against a closed valve. The closed valve head and maximum allowable operating time at closed valve conditions at S2 shall be as stated on the DATA SHEET.
11. The $NPSH_a$ available ($NPSH_a$) at the GDP and the $NPSH_r$ required ($NPSH_r$) at both the GDP and the maximum flow rate associated with S2 (see clause 7) shall be as specified/stated on the DATA SHEET.

$NPSH_r$ shall be taken as the 3 % head drop requirement. The pumpset shall be selected such that the $NPSH_a$ always exceeds the $NPSH_r$ (including the relevant safety margin) at all potential duty points.

At the maximum flow rate associated with S2, the margin of NPSH required above NPSH_r for continuous operation without causing excessive pump noise, vibration, or damage due to cavitation shall be as stated on the DATA SHEET.

12. If a multi-stage pumpset is required/to be provided, the number of stages shall be as specified/stated on the DATA SHEET.

4.3 Life and Maintainability

1. The minimum expected service life of the pumpset shall be as specified/stated on the DATA SHEET. Details of the maintenance required to achieve this service life shall be provided with the Tender

5.0 DESIGN INFORMATION

5.1 General

5.1.1 General

1. The total combined weight of the pumpset and the weights of the major pumpset components shall be as stated on the DATA SHEET.
2. The size of the pumpset (length x width x height) shall be as stated on the DATA SHEET.

5.1.2 Materials Selection

1. Materials which will be in contact with potable water shall comply with Regulation 25A of the Water Supply (Water Quality) Regulations 1989 and the Water Supply (Water Quality) Amendment Regulations 1989 and 1991.
2. Where surfaces (especially stainless steel) may be subject to galling, such as renewable wear rings, materials shall be selected to minimise wear and have a minimum hardness differential of 50 H_B.

5.1.3 External Corrosion Protection

1. Unless otherwise specified on the DATA SHEET, if a protective coating system is required for the pumpset it shall comply with WIMES 4.01 (Protective Coatings for use in the Water Industry).

5.2 Mounting Arrangements

5.2.1 General

1. The orientation of the pumpset shall be as specified/stated on the DATA SHEET.
2. Mounting arrangements shall be designed to facilitate accurate alignment and prevent excessive distortion under all operating conditions.

5.2.2 Baseplates (Horizontally Mounted Pumpsets)

1. Baseplates shall be locally reinforced around foundation holes.
2. Baseplates shall be designed to prevent the formation of trapped air pockets during grouting and the collection of water and debris during operation.
3. The baseplate material shall be as stated on the DATA SHEET.
4. The baseplate shall be stiffened to prevent the pumpset exceeding the maximum vibration levels specified in clause 1, Section 7.1.

5.2.3 Support Frames (Vertically Mounted Pumpsets)

1. The ends of all sections shall be sealed and the whole support frame shall be designed to produce a crevice free assembly.
2. The support frame material shall be as stated on the DATA SHEET.
3. The support frame shall be stiffened to prevent the pumpset exceeding the maximum vibration levels specified in clause 1, Section 7.1.

5.3 Pump Casing

1. The casing shall be designed to resist abrasion and mechanical shock imposed by solids in the pumped flow.
2. The casing material shall be as specified/stated on the DATA SHEET.
3. To allow the removal of the rotating components, the casing shall be designed such that the top half-casing can be removed without disturbing the bottom half-casing and system pipework.

If the means provided to permit the lifting of the top half-casing is of insufficient strength to provide the means of lifting the whole weight of the pumpset, this must be clearly indicated.

4. The casing halves shall be accurately aligned during assembly to match the volute profiles and be doweled to ensure certain repositioning after disassembly. Means shall be provided to facilitate the separation of the casing halves. On larger castings this shall be by two or more stainless steel jacking screws.
5. The casing casting shall be dressed to a good commercial standard to expose any surface imperfections. All imperfections shall be brought to the attention of the Purchaser. After visual inspection the following criteria shall apply:

Imperfections of a depth less than 10 % of the local casting thickness or of a length less than 50 mm may be repaired at the discretion of the Supplier in accordance with an approved method.

Imperfections of a depth between 10 % and 25 % of the local casting thickness or of a length greater than 50 mm shall be brought to the attention of the Purchaser. The Purchaser may decide to reject the casting or agree to its repair in accordance with an improved method.

Imperfections of a depth greater than 25 % of the local casting thickness shall result in the casting being rejected.

6. If specified/stated on the DATA SHEET, connections shall be provided on the pumpset flanges or casing for connection of inlet and outlet pressure gauges, venting, drain and seal flushing pipework. These shall consist of bosses appropriately drilled and tapped. Venting connections shall be positioned on the highest practicable point on the casing.

If specified/stated on the DATA SHEET, for applications where the Supplier is also providing the inlet/outlet pipework, tappings shall be provided for the connection of performance monitoring sensors, positioned at least 2 x pipe diameters away from the inlet/outlet branches of the casing.

All unused, tapped holes shall be fitted with solid, corrosion resistant, metal plugs.

7. Unless otherwise specified on the DATA SHEET, the inlet and outlet branches of the casing shall terminate in accordance with flange type NP16, as detailed in BS 4504 : Part 3.
8. All casing surfaces having a fine clearance between fixed and rotating components shall be provided with renewable wear parts which shall be easily removable for refurbishment or replacement.
9. If specified/stated on the DATA SHEET, casing wear rings shall be fitted. These shall be secured by screwed dowels or locking pins. The wear ring and fixing materials shall be as specified/stated on the DATA SHEET.
10. The casing shall be specifically designed for baseplate/support frame mounting using support feet machined to a finish to suit the baseplate/support frame and incorporating holes drilled and spot faced for holding down bolts. The feet shall accommodate and transfer all loads to the baseplate/support frame.
11. If, as specified on the DATA SHEET, an internal coating is required for corrosion protection and/or efficiency enhancement, the type of coating and coating thickness shall be as specified/stated on the DATA SHEET. The % efficiency increase and the expected life of the coating shall be as specified/stated on the DATA SHEET.

5.4 Impeller(s)

1. The direction of rotation of the impeller shall be clearly and indelibly marked on the pump casing with an arrow.
2. The direction of rotation of the impeller (viewed from the non-drive end) shall be as stated on the DATA SHEET.
3. The dimensions of the impeller(s) shall be as stated on the DATA SHEET.

4. Each impeller shall be a one piece casting manufactured from the material specified/stated on the DATA SHEET. It shall be secured to the pump shaft and be retained against circumferential movement by keying. Impellers shall not be pinned to the shafts, neither shall shaft rotation be relied upon to ensure that the impeller is locked in position.
5. Each impeller shall be dynamically balanced in two planes to prevent the pumpset exceeding the maximum vibration levels specified in clause 1, Section 7.1. Balancing shall be achieved by machining, not addition of weights.
6. If specified/stated on the DATA SHEET, a machining allowance shall be provided on the impeller to accommodate wear/neck rings.
7. If, as specified/stated on the DATA SHEET, impeller wear rings are fitted, the wear ring and fixing materials shall be as specified/stated on the DATA SHEET.
8. If spare impellers are provided, they shall also be tested in the pumpset and characteristic curves be supplied.

5.5 Pump Shaft

5.5.1 General

1. The shaft material shall be as specified/stated on the DATA SHEET.
2. The shaft stiffness shall be such that, under the most severe dynamic conditions of operation, the total shaft deflection at the seal does not exceed the seal manufacturer's specified tolerances.
3. Wherever the shaft passes through a bearing housing, seals shall be provided to prevent the ingress of dust and moisture.
4. If the shaft is exposed to the pumped liquid, it shall be manufactured from a corrosion resistant material or be protected by the use of a sleeve.

5.5.2 Shaft Sleeves

1. A replaceable sleeve shall be fitted where the shaft passes through a packed gland seal.
2. Shaft sleeves shall be manufactured from a material which will not cause galvanic corrosion between themselves and the shaft. The shaft sleeve material shall be as specified/stated on the DATA SHEET.
3. Shaft sleeves shall be positively driven.
4. Seals shall be provided to prevent leakage of the pumped liquid between the shaft and sleeve.

5.6 Seals

5.6.1 General

1. The seal type and manufacturer shall be as specified/stated on the DATA SHEET.
2. If flushing, cooling and/or lubrication is required to maintain seal performance, the method of flushing, cooling and/or lubrication shall be as stated on the DATA SHEET. The flushing, cooling and/or lubrication requirements (pressure, flowrate and source) shall be as specified/stated on the DATA SHEET.

5.6.2 Mechanical Seals

1. Unless otherwise specified on the DATA SHEET, cartridge type mechanical seals shall be fitted.
2. Mechanical seal component materials shall be as specified/stated on the DATA SHEET.
3. If specified/stated on the DATA SHEET, spacer couplings shall be provided.

5.6.3 Packed Gland Seals

1. Packed gland seal component materials shall be as specified/stated on the DATA SHEET.

5.7 Bearings and Bearing Lubrication

5.7.1 General

1. All pump bearings shall have a minimum L_{10h} life of 40,000 hours at the guaranteed duty point (GDP).
2. If, at the GDP, the bearing re-lubrication interval (as calculated from the bearing/grease manufacturers data) is greater than 40,000 hrs, the pump shall be fitted with pre-packed, sealed for life, grease-lubricated bearings.
3. If, at the GDP, the bearing re-lubrication interval (as calculated from the bearing/grease manufacturers data) is less than 40,000 hrs, bearings shall be fitted with automatic, rechargeable grease dispenser units providing a clear indication of low grease content. The minimum capacity of the grease dispenser units shall be as specified/stated on the DATA SHEET.
4. If, at the GDP, the bearing re-lubrication interval (as calculated from the bearing/grease manufacturers data) is less than 2000 hrs, an oil lubrication system shall be fitted. Oil lubricated bearings shall be provided with sight glasses which give an accurate indication of oil level when the pumpset is operating.
5. Bearing housings shall be designed to prevent the ingress of dust and water. For pumpsets incorporating packed gland seals, the drive shaft shall be provided with a thrower ring fitted adjacent to the bearing housing. The bearing housing material shall be as specified/stated on the DATA SHEET.
6. Following alignment of the impeller in the casing, the bearing housing shall be doweled or spigoted to ensure accurate repositioning following maintenance.

7. The bearing housing shall be designed so that the bearing lubricant can be easily drained and replaced without excessive spillage.

5.7.2 Radial Bearings

1. The bearing manufacturer and manufacturer's bearing designations shall be as stated on the DATA SHEET.
2. The type and grade of bearing lubricant and the method of lubrication shall be as specified/stated on the DATA SHEET.

5.7.3 Thrust Bearings

1. The bearing manufacturer and manufacturer's bearing designations shall be as stated on the DATA SHEET.
2. The type and grade of bearing lubricant and the method of lubrication shall be as specified/stated on the DATA SHEET.

5.8 Intermediate Drive Shaft

1. If specified/stated on the DATA SHEET an intermediate drive shaft shall be provided.
2. The first critical shaft speed shall be at least 25 % above the maximum operating speed and shall be as stated on the DATA SHEET.
3. The drive shaft shall not excite natural vibrations in the support structure.
4. Details of the drive shaft arrangement shall be as stated on the DATA SHEET (length of shaft, shaft material, support bearings etc.).

5.9 Couplings

1. The coupling type and manufacturer shall be as stated on the DATA SHEET. If specified on the DATA SHEET, spacer type couplings shall be provided (for applications where mechanical seals are fitted).
2. Coupling component materials shall be as stated on the DATA SHEET.

5.10 Motors

1. Unless otherwise specified on the DATA SHEET, motors shall comply with WIMES 3.03 (Specification for LV AC Motors for Use in the Water Industry).
2. If the motor is not procured against WIMES 3.03, the electrical supply and motor details shall be as specified/stated on the DATA SHEET.

5.11 Lifting Points

1. The pumpset shall be provided with clearly identified, permanent lifting points located to give a safe, balanced, lift.
2. If the lifting points are not designed for lifting the whole pumpset, they shall be clearly marked accordingly.

5.12 Information Plate

1. The pumpset shall be provided with an information plate permanently fixed to one of the major pump components. The plate and its fixings shall be manufactured from corrosion resistant, metallic materials. The information plate shall include the information specified on the DATA SHEET.
2. If specified on the DATA SHEET, a duplicate information plate shall be provided, fixed/located in the position specified on the DATA SHEET.

5.13 Guarding

1. Guarding shall be in accordance with BS 5304 to achieve CE compliance/markings.
2. Interlocking of guards and covers shall comply with BS EN 1088.
3. Guarding fitted over parts of the pumpset which may require regular access for maintenance or condition monitoring purposes (e.g. bearing vibration) shall be provided with suitable provision for access.

5.14 Electrical Control Panel and Associated Electrical Installation

1. Unless otherwise specified on the DATA SHEET, the electrical control panel and associated electrical installation shall comply with WIMES 3.04 (Specification for Low Voltage Electrical Package Plant for Use in the Water Industry).

6.0 CONDITION MONITORING AND PROTECTION

1. The pumpset shall be provided with the types of condition monitoring specified on the DATA SHEET.
2. If the Supplier is not supplying the sensor equipment, the fixing arrangements shall be as specified on the DATA SHEET. If the Supplier is supplying the sensor equipment, suitable provision shall be made for the location of sensors.

7.0 TESTS

7.1 Test Regime

1. The test regime shall be as specified/stated on the DATA SHEET.

7.2 Test Details

7.2.1 Hydrostatic Pressure Test (If Specified)

1. The pump casing shall be hydrostatically pressure tested at an appropriate stage of manufacture prior to painting or coating of the components (this does not include coatings provided for the protection of raw parts against rust). The hydrostatic test pressure shall be at least 1.5 times the maximum working pressure. The test pressure shall be maintained for at least 30 minutes with no leakage or loss of pressure.

7.2.2 Noise Test (If Specified)

1. Unless otherwise specified on the DATA SHEET, noise levels for the pumpset shall not exceed 85 dB(A) at a distance of 1 m from the pumpset centre line (based upon the pumpset being mounted in 'semi-reverberant' conditions). If this level is exceeded, acoustic cladding or hoods shall be supplied with the pumpset.

APPENDIX A - DATA SHEET EXPLANATION

VERTICAL/HORIZONTAL, AXIALLY SPLIT CASING, SINGLE/DOUBLE ENTRY
PUMPSET FOR USE IN THE WATER INDUSTRY

1. The DATA SHEET should be partially filled in by the Purchaser, completed by the Supplier and returned to the Purchaser. Items to be completed by the Purchaser are marked (o) and by the Supplier (x). Items which the Purchaser may wish to specify, or alternatively may wish to leave for the Supplier to complete are marked (ox).
2. The Purchaser may customise the DATA SHEET included in Appendix B to provide for:
 - a) the Purchaser's name, issue reference, format for inclusion of Contract/Project title and references etc.
 - b) the Purchaser's standard technical and procedural requirements which apply to every Contract/Project carried out for the Purchaser provided that such inclusions comply with the requirements stated in 5 below.
3. References incorporated on the DATA SHEET in Appendix B relate to Sections or Clauses, as appropriate, within WIMES 1.01.
4. The technical content, its arrangement and any incorporated reference to WIMES 1.01 shall not be altered from the form shown on the DATA SHEET in Appendix B.
5. The person(s) responsible for completing the DATA SHEET shall only employ the form of DATA SHEET approved for use by the purchaser.
6. The DATA SHEET should be completed by ticking the feature(s) required or by entering requirements or information, as appropriate, in the column/row adjacent to the feature description.

Where there is a need to enter more information than space permits, a reference should be entered as to where the requirement is specified. This approach may be used where:

- a) further clarification or definition is required on a feature description, or
 - b) the feature description needs to be read in conjunction with another Standard or Specification applicable to the Contract/Project, or
 - c) a particular feature or arrangement is not listed on the DATA SHEET and the option 'Other (specify)' is used.
7. The referencing system employed shall be defined in the Contract/Project documentation.
 8. Any features or requirements, whether or not listed on the DATA SHEET, which are necessary to provide a safe and fully operational system fit for purpose, shall be deemed to be shown on the DATA SHEET as a requirement and be included as part of the Contract/Project.

APPENDIX B - DATA SHEET

VERTICAL/HORIZONTAL, AXIALLY SPLIT CASING, SINGLE/DOUBLE ENTRY PUMPSET FOR USE IN THE WATER INDUSTRY

PROJECT NUMBER	DOCUMENT REFERENCE
PROJECT TITLE	REVISION
PROJECT LOCATION	DATE
MANUFACTURER	REF DRAWINGS (included/not included)

Key

- o - Purchaser to specify
- x - Supplier to state
- ox - Purchaser/Supplier to specify/state
- * - Indicates Purchaser's default requirement

1.0 SCOPE

2	Number of Pumpsets (o)	
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3.0 PROCESS INFORMATION

3.1 Location and Operating Environment

1	Location (o)	
	Inland/coastal	
	Inside/outside	
	Other (specify)	
	Hazardous zone	
2	Special Hazards (o)	
3	Ambient Temperature and RH (o)	
	-10 to 40°C/up to 80% RH*	
	Other (specify)	

3.2 Properties of Pumped Liquid

1	Type of Pumped Liquid (✓) (o)	
	Potable water	
	Partially treated water	
	Raw water	
	Other (specify)	
1	Solids Content (o)	
	Wt %	
	Particle size range (µm)	
1	Hazards (✓) (o)	
	Toxic	
	Corrosive	
	Abrasive	

	Odorous	
	Other (specify)	
1	Other Properties (o) (if known)	
	pH	
	Vapour pressure (bar abs)	
	Density (kg/m³)	
	Temperature (inlet) (°C)	Min Max

4.0 PERFORMANCE INFORMATION

4.1 Mode of Operation

4.1.1 General

1	Mode of Operation (o)	
	Solo/parallel/series	
	Type of Duty (o)	
	Continuous/intermittent	
	Type of Drive (o)	
	Fixed/dual/variable	

4.2 Hydraulic Performance, Efficiency and Adsorbed Power (All Heads are Relative to the Centre Line of the Impeller)

4.2.1 Fixed Speed Drives

1	Operating Speed (rpm) (ox)	
	Maximum (1500*)	
	Actual	
2	Hydraulic Performance at the Guaranteed Duty Point (GDP) (o)	
	Flow rate (l/s)	
	Static head (m)	
	Dynamic head (m)	
3	System Details (o)	
	Reference for system curve(s) (specify)	
	Max/min static head (m)	

5	Q (GDP)/Q (BEP) - Limits (o)	
	X/Y (%)	
8	Pump Efficiency (%) and Adsorbed Power (Bare Shaft) (kW) (x)	
	Efficiency at GDP	
	Efficiency at max flow	
	Ads. power at GDP	
	Ads. power at max flow	
7	Maximum Flow Rate (l/s) (x)	
8	Minimum Flow Rate (l/s) (x)	
9	Pumpset to be Capable of Discharging into an Empty Main (Y/N) (x)	
10	Performance Against a Closed Valve (x)	
	Closed valve head (m)	
	Maximum operating time at closed valve (s)	
11	NPSH (m) (ox)	
	Available at GDP	
	Required at GDP	
	Required at max flow	
	NPSH Safety Margin (% NPSH _r) at max flow	
12	Number of Stages (ox)	

4.2.2 Dual/Variable Speed Drives

1	Operating Speed (rpm) (ox)	
	Maximum (1500*)	
	Actual	S1 S2
2	Hydraulic Performance at the Guaranteed Duty Point at S2 (GDP) (o)	
	Flow rate (l/s)	
	Static head (m)	
	Dynamic head (m)	
3	System Details (o)	
	Reference for system curve(s) (specify)	
	Max/min static head (m)	
5	Q (GDP)/Q (BEP) at S2 - Limits (o)	
	X/Y (%)	
6	Pump Efficiency (%) and Adsorbed Power (Bare Shaft) (kW) (x)	S1 S2
	Efficiency at GDP	
	Efficiency at max flow	
	Ads. power at GDP	
	Ads. power at max flow	
7	Maximum Flow Rate at S2 (l/s) (x)	
8	Minimum Flow Rate at S2 (l/s) (x)	
9	Pumpset to be Capable	

	of Discharging into an Empty Main (Y/N) (x)	
10	Performance Against a Closed Valve at S2 (x)	
	Closed valve head (m)	
	Maximum operating time at closed valve (s)	
11	NPSH at S2 (m) (ox)	
	Available at GDP	
	Required at GDP	
	Required at max flow	
	NPSH Safety Margin (% NPSH _r) at max flow	
12	Number of Stages (ox)	

4.3 Life and Maintainability

1	Minimum Expected Service Life of Pumpset (yrs) (ox)	
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5.0 DESIGN INFORMATION

5.1 General

5.1.1 General

1	Weight of Pumpset/Components (kg)(x)	
	Pumpset	
	Pump	
	Drive	
	Baseplate/frame	
	Other (specify)	
2	Size of Pumpset (m) (x)	
	Width/length/height	

5.1.3 External Corrosion Protection

1	Protective Coating Req'd (Y/N) (WIMES 4.01*) (o)	
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5.2 Mounting Arrangements

5.2.1 General

1	Orientation (ox)	
	Horizontal/vertical	

5.2.2 Baseplate (Horizontally Mounted)

3	Material (x)	
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5.2.3 Support Frame (Vertically Mounted)

2	Material (x)	
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5.3 Pump Casing

2	Material (ox)	
6	Tappings to be Supplied For: (✓) (ox)	
	Inlet/outlet pressure gauges	
	Venting pipework	
	Drain pipework	
	Seal flushing pipework	
	Performance monitoring sensors (if pipework is also supplied)	
7	Drilling of Flanges (o)	
	Inlet (mm)	
	Nominal bore diameter	
	Flange sched. (NP16*)	
	Outlet (mm)	
	Nominal bore diameter	
	Flange sched. (NP 16*)	
9	Casing Wear Rings (ox)	
	Required (Y/N)	
	Material	
	Fixing material	
11	Coatings (ox)	
	Required (Y/N)	
	Specification	
	Thickness (µm)	
	Efficiency increase (%)	
	Expected life (yrs)	

5.4 Impeller(s)

2	Direction of Rotation of Impeller (viewed from the non-drive end) (x)	
3	Impeller Dimensions (mm) (x)	
	Diameter fitted	
	Min. diameter	
	Max. diameter	
4	Material (ox)	
6	Machining Allowance Req'd (Y/N) (ox)	
7	Wear Rings (ox)	
	Required (Y/N)	
	Material	
	Fixing material	

5.5 Pump Shaft

5.5.1 General

1	Material (ox)	
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5.5.2 Shaft Sleeves

2	Material (ox)	
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5.6 Seals

5.6.1 General

1	Seal Type and Manufacturer (ox)	
	Mechanical/packed gland	
	Manufacturer	
2	Method of flushing/cooling/lubrication (x)	
2	Flushing/Cooling Water Reqmnts (x)	
	Supply Pressure (bar)	
	Flow rate (l/s)	
2	Source (o)	
	Potable water	
	Other (specify)	

5.6.2 Mechanical Seals (If Specified)

1	Type (Cartridge*) (o)	
2	Component Materials (ox)	
	Seal end plate	
	Throttle bush	
	Rotating ring	
	Stationary ring	
	Secondary seals	
	Spring	
	Other (specify)	
3	Spacer Couplings Req'd. (Y/N)	

5.6.3 Packed Gland Seals (If Specified)

1	Component Materials (ox)	
	Gland follower	
	Packing	
	Lantern ring	

5.7 Bearings and Bearing Lubrication

5.7.1 General

3	Min. capacity (weight) of grease lubricators (x)	
5	Bearing Housing Material (ox)	

5.7.2 Radial Bearings

1	Bearing Details (x)	
	Manufacturer	
	Designation	
2	Type of Lubricant (ox)	
	Oil/grease	
	Grade	
2	Method of Lubrication (ox)	
	Sealed for life	
	Re-lubrication facilities required/provided (Y/N)	

5.7.3 Thrust Bearings

1	Bearing Details (x)	
	Manufacturer	
	Designation	
2	Type of Lubricant (ox)	
	Oil/grease	
	Grade	
2	Method of Lubrication (ox)	
	Sealed for life	
	Re-lubrication facilities required/provided (Y/N)	

5.8 Intermediate Drive Shaft

1	Reqd/provided (Y/N) (ox)	
2	First critical speed (rpm) (x)	
3	Drive Shaft Arrangement (x)	

5.9 Couplings

1	Coupling Details (x)	
	Type (spacer for mech seals*)	
	Manufacturer	
2	Component Materials (x)	
	Hub/spacer	
	Flexible component	
	Guard	

5.10 Drive

1	Motor Specification (ox)	
	WIMES 3.03*	
	Other (specify)	
2	Supply Details (to be completed for motors not procured with WIMES 3.03) (ox)	
	Voltage (V)	

	No. of phases	
	Frequency (Hz)	
2	Motor Details (to be completed for motors not procured with WIMES 3.03) (ox)	
	Manufacturer	
	Motor Protection (IP55*)	
	Rating (kW)	
	Minimum efficiency (3/4 load and full load) (%)	
	Full load current (A)	

5.12 Information Plate

1	Data Required on Information Plate (✓) (o)	
	Manufacturer	
	Date of manufacture	
	Serial number	
	Pump type	
	Head at GDP (m)	
	Flow at GDP (m ³ /hr)	
	Efficiency at GDP(s) (%)	
	Operating speed(s) (rpm)	
	NPSHr at GDP (m)	
	Casing test press. (bar g)	
	Weight (Dry/wet) (kg)	
	Asset code	
	Other (specify)	
2	Duplicate Information Plate (o)	
	Required (Y/N)	
	Fixing position/location	

5.14 Electrical Control Panel and Associated Installation

1	Control Panel Specification (ox)	
	WIMES 3.04*	
	Other (specify)	

6.0 CONDITION MONITORING

6.1 Condition Monitoring

1	Types of Monitoring Required (✓) (o)	
	Seal leakage	
	Bearing vibration	
	Bearing temperature	
	Other (specify)	
2	Fixing Arrangements (o)	

7.0 TESTS

7.1 Test Regime (Clause 1)

Type of Test (o)	Test Location (o)		Witnessing Requirements (o)		Test Standard (ox)	Acceptance Levels & Tolerances (o)
	Works	Site	(Y/N)	Notice (Days)		
Hydrostatic						
Acceptance (Q, H, Power, Effy)					BS 5316 Class B*	
NPSH					BS 5316 Class B*	
Noise						dB(A)
Vibration					BS 7854 Part 1*	mm/s
Thermodynamic						
Other (specify)						

PUMP TESTING

Pumps have to be tested to satisfy the customer, and to check design (by the manufacturer). Parameters measured are pressure rise, flow rate, power consumption, NPSH (or NPSE) vibration levels, noise generated, and the duty point (which is usually but not always the best efficiency point). Both customer and maker are interested as the contracts will usually invoke such standards as SPI 610, ISO codes, ANSI Codes or other codes.

The customer needs to be satisfied that the unit performs to contract and often requires a performance plot at the rated driver speed. In some cases independent inspectors are used who witness the tests and the method of determining the data presented and then certify using an approval certificate. They sometimes insist that the system in which the pump is to be fitted be modelled for tests.

1. TEST CODES AND STANDARDS

Test codes are needed to establish test conditions and procedures and are normally invoked in contracts and the limits set by the code are part of the contract. There are several codes set by the industry, such as API610 which is invoked by the Petroleum Industry, and a number of ISO (or BSI) standards. International Power Test codes are also involved for such units as pumps for the Power Generating Industry.

1.1 ISO and BSI Codes

The International Standards Organisation based in Geneva produces ISO standards which combine national standards and are applied over the world. In general BSI (British Standard Institute) now issues standards that have dual numbering (ISO and BSI).

For most pumps used by chemical and petroleum industries, ISO 5199 – 1986 (BS 6836 : 1987) provides standards for what are called Class II pumps (end suction machines) and uses the provisions of BS 5316: Part 1: 1976 (ISO 2548: 1973) for acceptance tests. These were for acceptance tests for commercial pumps (old Class C). Tables 1 and 2 summarise the tolerances assigned by the standard. The work error defined as a value equal to twice the estimated deviation; it is assumed that the true error will not exceed twice the estimated standard deviation.

Equivalent codes in the USA are ANSI B73.1: 1977 and API 610 edition 7: 1989. ANSI uses the American Institute for Hydraulics Standards as a basis for obtaining and presenting performance data.

1.2 API 610 Edition 7 : 1989

This applies to centrifugal pumps, and is comprehensive code relating to dimensional standards, materials, running clearances, provision of seals, bearings, flanges, base plates, permissible loads on casings, noise and vibration limits, and provisions for test, inspection, and other matters. The code gives standards for presentation of data, and these will be compared with the other codes where relevant.

2. TEST LOOPS

2.1. Open Test Loops

Figure 1 illustrates an open loop circuit, a very common system used for routine pump tests as it allows one or more pumps to run off the sump while others are being stripped and rebuilt. It is useful as a heat sink where there is a large heat rejection from pumps, but cavitation tests are very difficult to conduct as the suction pressure is not easy to maintain during running over a range of duties.

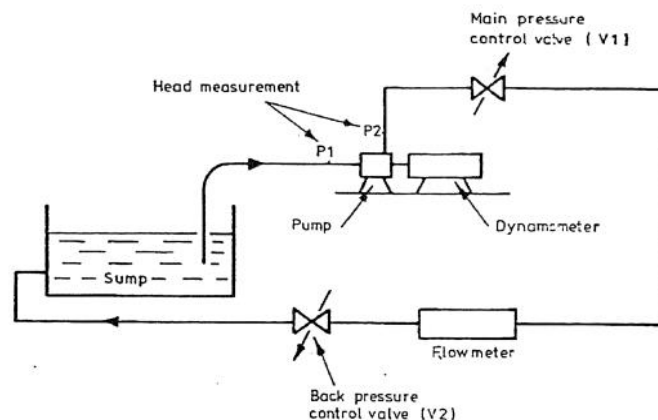


Figure 1 Diagram of typical open circuit test loop

2.2 Closed Circuits

Figure 2 shows a typical closed loop system. This allows the loop pressure to be controlled and allows good control for cavitation testing. Temperature rise is a problem, and a cooling coil will be required in some cases.

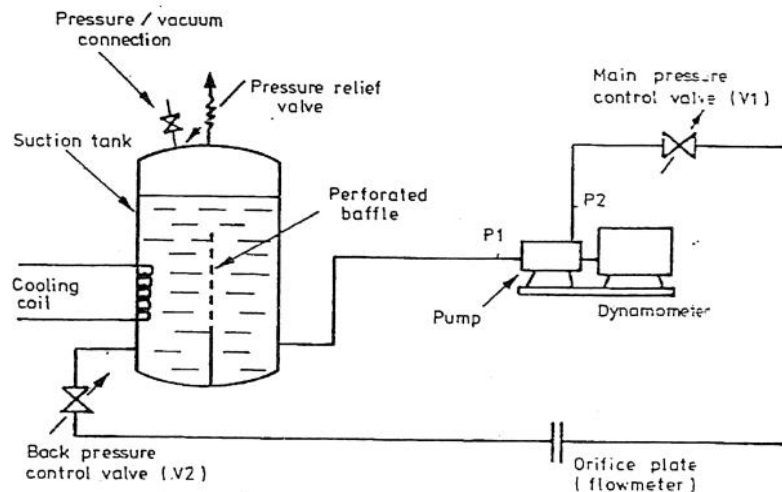


Figure 2 Diagram of typical closed circuit test loop

A formula for calculating the temperature rise is:

$$\text{Temperature rise } ^\circ\text{C / hour} = 0.86 \times \text{power in kW} / \text{loop volume in m}^3$$

where the sump or main tank is large and the pump small this may not be a problem. A rule of thumb approach (particularly for cavitation testing) is that the tank should be 30 times the pump test flow rate. This should give a minimum passage time of about 30 seconds so that air will be released from the system.

Which loop to go for is a matter of economics and of course space available.

3. TEST CIRCUIT PROVISION

3.1 Pipework and Pressure Tappings

BS 5316 part 1 specifies at least 6 diameters of straight pipe after a bend, with suction pressure tappings 2 diameters upstream from the flange of the pump and discharge tappings 2 diameters from the discharge flange, as shown in figure 3 and 4 taken from the standard. Where there is a considerable swirl in the inlet pipe, flow straighteners must be fitted. The influence of inlet systems on pump behaviour is discussed in this course and will be referred to during presentation of this section.

3.2 Flow Metering and Pressure Measuring Instruments

Figure 3 and 4 taken from the standard give clear information on how to use manometers or gauges. If transducers are used, correction is now usually needed. Pressure gauges must be calibrated. If transducers are used correction is not usually needed regularly, but, should be checked at intervals. Flow meters have varying accuracy, as Table 1 indicates, and must be used according to ISO 5167 for differential pressure devices, or to maker's specifications

for electro-magnetic meters and turbine meters. Turbine meters are very sensitive to swirl, but EM devices are relatively insensitive. All should be checked and recalibration at intervals, 6 months being common.

3.3 Provision for NPSH Tests

As figure 2 illustrates, a closed loop circuit provides good control by using a combination of vacuum pump, surface level variation and valving. A gate valve should not be used as it causes excessive disturbance when nearly closed and only really 'controls' over the last 20% of its travel, butterfly valves are also not suggested for the same reasons, and a well designed 'proportional' valve is best. A vacuum pump should be 'man enough' for the job and preferably run continuously with an air bleed back into tank to maintain pressure.

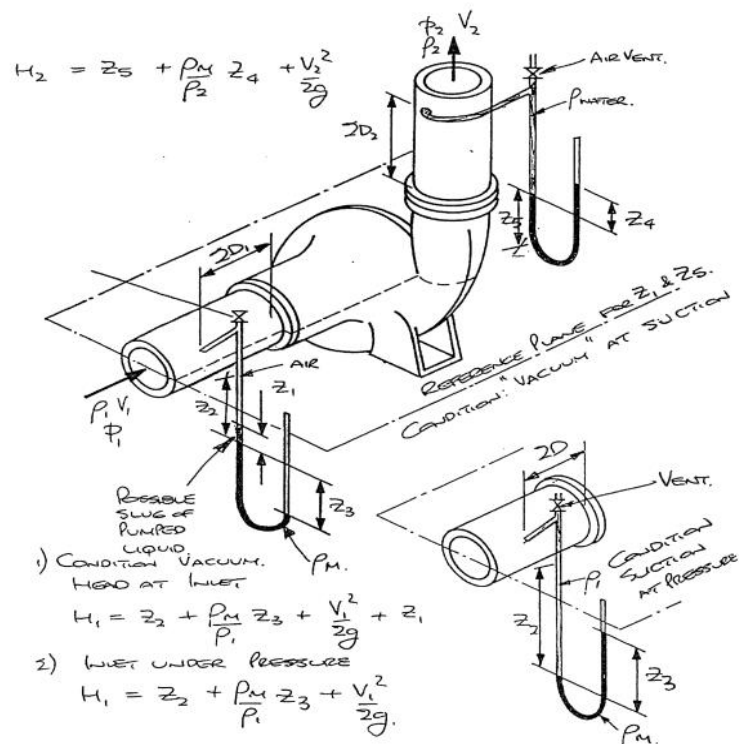


Figure 3a Pressure measurement using manometer

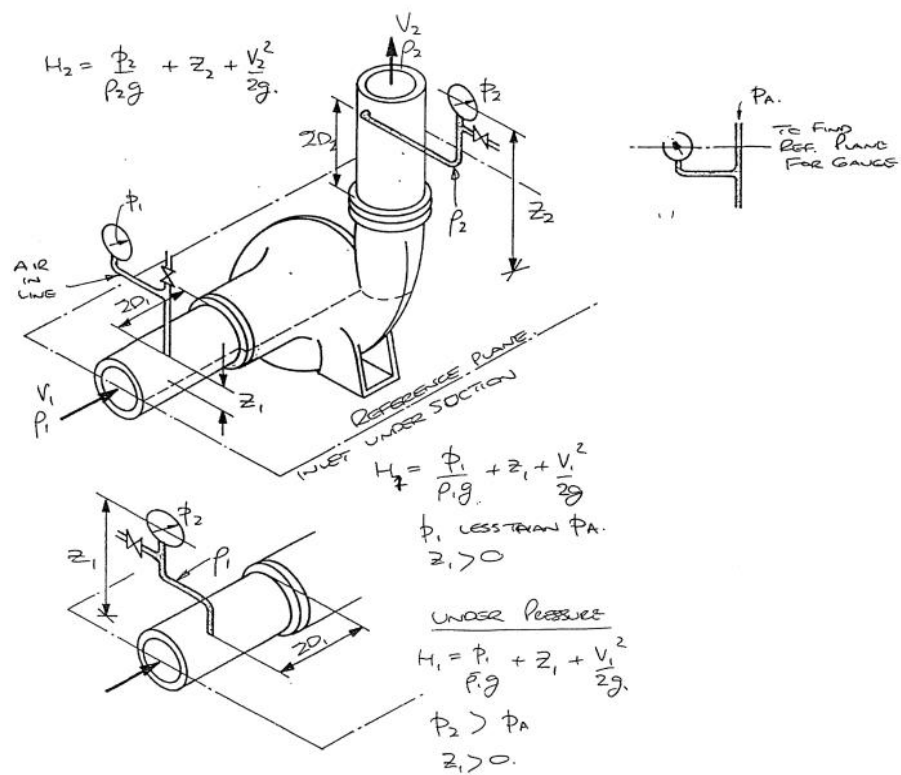


Figure 3b Pressure measurement using gauges

4. TEST PROCEDURES

4.1 General Performance Testing

This is routine and should follow good experimental practice, check plotting during tests, repeatability point checking, BS5316 does not specify the number of points to establish an H – Q curve. Though a specimen sheet shows 11 points, clearly a number of points (more than 4/5) are needed to establish a curve fit. API610 specifies at least 5 point of complete data, including head capacity and power, these should normally be shut-off, minimum continuous stable flow, midway between minimum and rated flows, rated flow and 110% of rated flow. BS5316 : part 1, Section 9.4 clearly lays down how to check if the guarantee duty is verified, and this is shown in Figure 4.

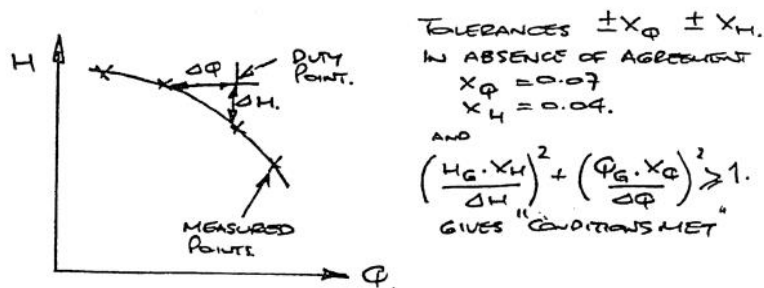


Figure 4

4.2 Cavitation Testing

As well as general performance, it is required that the NPSH to cause 'head' to fall off by 3% when the flow and speed are held constant be found.

Three methods for determining the cavitation performance of a pump are illustrated in Figure 5. Method (a) involves using a constant NPSH for fixed suction valve position, and adjusting the outlet valve until 3% 'head' loss is registered, this is then repeated for other flow rates. Method (b) is an open circuit uses inlet valve throttling or level change to adjust NPSE with the outlet valve fixed, until 3% 'head' loss is registered. Method (c) involves fixed suction and delivery valve settings and dropping the loop pressure slowly till a 3% drop in 'head' is achieved. (This needs small changes in the delivery valve when the head starts to drop to keep flow constant).

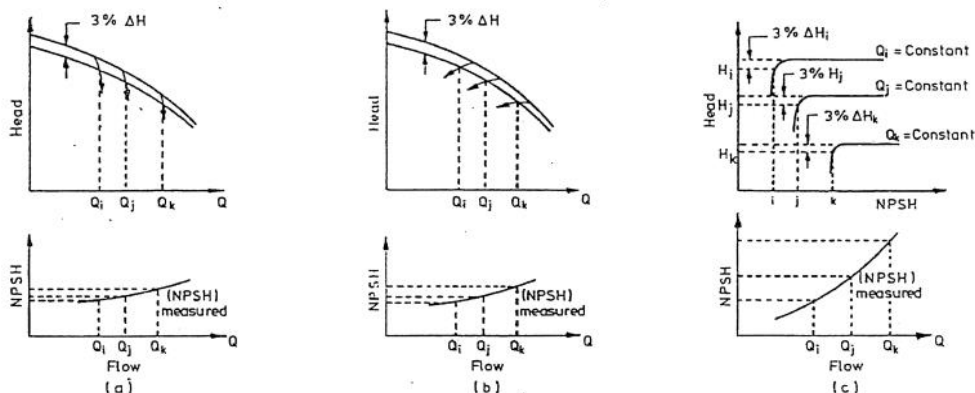


Figure 5 Method of determining required NPSH for 3% head loss ($3\% \Delta H$)

5. NOISE AND VIBRATION TESTING

API 610 specifies acceptable vibration limits and Figure 6 and 7 will be used to assist discussion. Limits specified for anti-friction bearing pumps the unfiltered vibration measured on the bearing housing at $\pm 10\%$ capacity shall not exceed a velocity of 0.3 in/s or a displacement of 63.5 micrometres peak to peak, and for sleeve bearings the corresponding values of 0.4 in/s and 63.5 micrometres.

Acoustic detection of cavitation is a very useful tool. For example, a hydrophone set to monitor noise levels at 40kHz (a frequency typical of bubble collapse will give a curve like that in Figure 8 indicating cavitation long before hydraulic performance fall off.

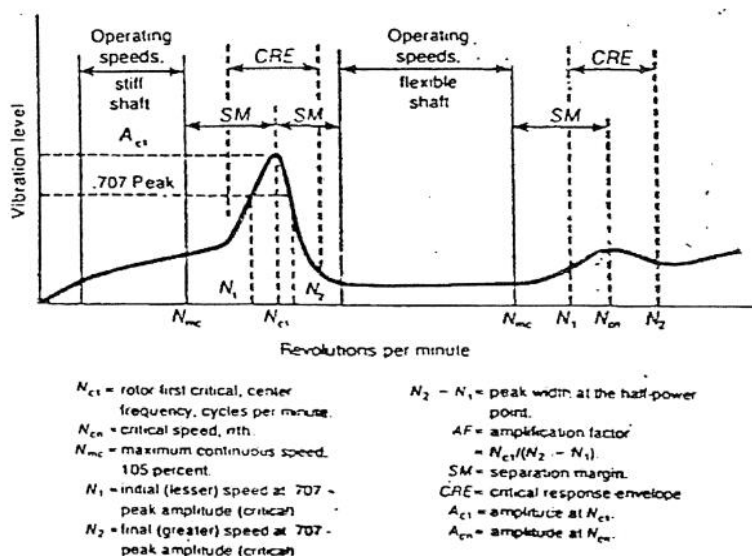


Figure 6

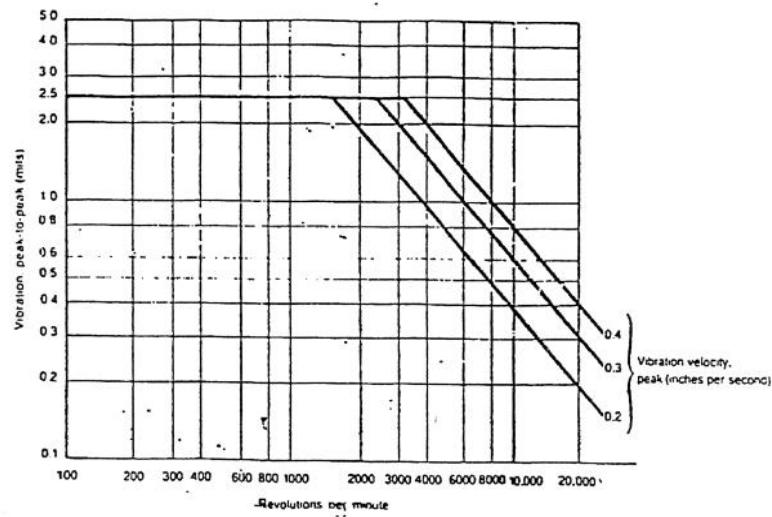


Figure 7 Vibration limit

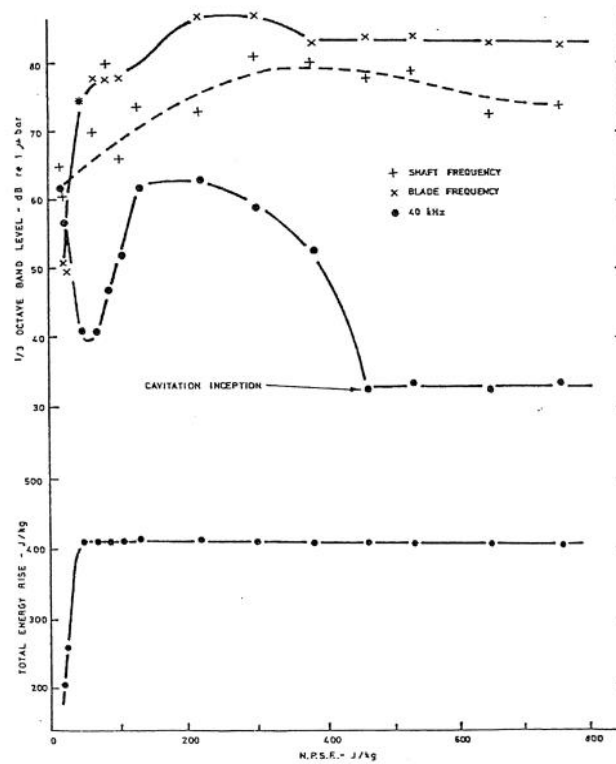


Figure 8

6. IN SITU TESTING

If the condition of a pump is needed once it is installed, difficulties emerge.

Pressure is 'easy' to monitor, but pipe work layout can cause problems, and give wrong or misleading values (a ring main as in Figure 3 helps to average readings).

Flow rate is more difficult in the average pump house or pump farm, as the specifications of BS1042 cannot usually be obeyed. Crude methods like reservoir draw down have been used but devices like clamp-on ultrasonic meters and Yates meter can give a fairly accurate value, and is not accepted by the water authorities.

Power similarly poses problems, and again the Yates meter is a solution which is also accepted.

7. REFERENCES

Flow measurement	BS1042 (ISO R541, ISO R781)
Bourdon gauges	BS 1780
Electrical instruments	BS 89
Motor efficiency determination	BS 4999 (BS 269)
Electric power measurement	BS 3435
Pump testing	BS 5316 Part 1 1976 (ISO 2598) BS 5316 Part 2 1977 (ISO 3555)

Table 1

**BS 5316 : Part 1 : 1976
Error Statement**

Quantity	Permissible systemic errors of measurement instrument	Permissible maximum limits of overall errors
Flow rate	+/- 2.5%	+/- 3.5%
Pump total head	+/- 2.5%	+/- 3.5%
Pump power input	+/- 2.5%	+/- 3.5%
Electrical power input	+/- 2.0%	+/- 3.5%
Speed of rotation	+/- 1.4%	+/- 2.0%
Motor efficiency	+/- 2.0%	
Overall efficiency		+/- 4.5%
Pump efficiency		+/- 5.0%

Table 2

API Tolerance

(Taken from Table 11 pp37 of API 610 – 7 edition)

Quantity	Rates Point (%)	Shut off (%)
Rated differential head		
0 – 500ft (0 – 152m)	-2 +5	-10 +10
501 – 1000ft (153-305m)	-2 +3	-8 +8
over 1000ft (305m)	-2 + 2	-5 + 5
Rated Power	+/- 1.4%	
NPSH required	0	

Table 3 A statement of probable metering errors

Flow Meters

venturi meter	+3% -1%
weigh tank	+/- 0.15%
electromagnetic	+/-1%

Pressure

manometer	+/-0.5%
Bourdon gauge	+/- 1%

Power meter

two wattmeter	+/- 1%
torque tube	+3.1% -1.1%
dynamometer	+/-0.3%

8. APPENDICES

APPENDIX 1 – TYPES OF TESTS

1. Pump Model Tests

Sometimes used where full size is large (e.g. cooling water pumps for power stations).

Tests using air or water.

Full size predicted using scaling laws

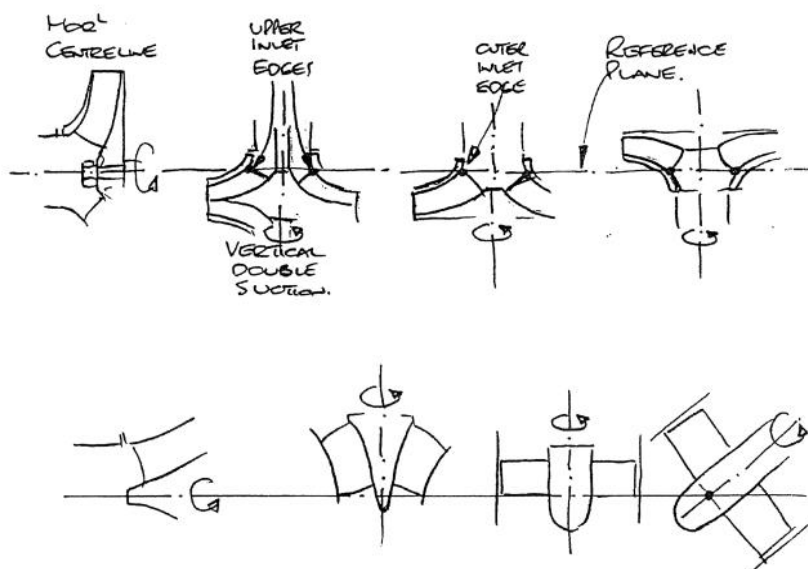
2. Sump or Intake Tests

Sometimes needed to ensure good intake conditions to pump.

Some customer have their own codes

APPENDIX 2 – REFERENCE PLANES

BS5316 recommends following reference planes



Specifications?

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- Safety
- Quality
- Contractual

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Why specifications and standards?

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- Define Need
- Basis for Contract
- Set the Benchmarks
- Relate Statutory Requirements
- Reference other Relevant Specs and Stds
- Define Acceptance Criteria

= *QUALITY*

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Specifications and Standards?

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Process Systems Engineering

- Where do they come from?
 - International Law
 - Government
 - Regulatory Authorities
 - Voluntary (Industry) Organisations
 - The Customer
 - His Consultants and Contractors
 - Inspecting and Approvals Bodies

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Specifications and Standards?

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- Where do they come from?
 - EC Directives
 - Safety
 - The CE Mark
 - ISO 9000 Quality
 - Technical Standards
 - ISO
 - CEN
 - BS
 - API etc
 - Industry Standards – Self Regulation & Benchmarks
 - Project Specific

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What do they mean for The Engineer?

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Process Systems Engineering

- Design and Construction Requirements
- Data Sheets
- Testing

Plus all the stuff on documentation, painting, packing,
etc, but watch out – sometimes there are interesting
bits, like seismic conditions!

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Typical Pump Standards

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- API 610, and equivalent ISO 13709
- BS EN 25199 (ISO5199)
- BS EN ISO 9906

And many many more!

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Considerations

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- The Customer doesn't PAY if HE thinks you don't comply with his requirements
- You go to Jail without passing GO or collecting £200 if you knowingly break Safety Rules

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Why Test?

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- Contractual
- Quality Control
- Product Performance Data

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The Standards

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- Typical working Standards for Production Testing

Pumps

ISO 9906

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What the Standard Says

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- Terms, Symbols and Units of Measure
- Time and Place of Test
- Qualifications and Responsibilities of Tester
- Layout of Test Set-up
- Instruments and Calibration
- Test Procedure and Taking Readings
- Allowable Variations/Fluctuations
- Interpretation of Results
- Presentation of Results
- Contractual Acceptance/Rejection

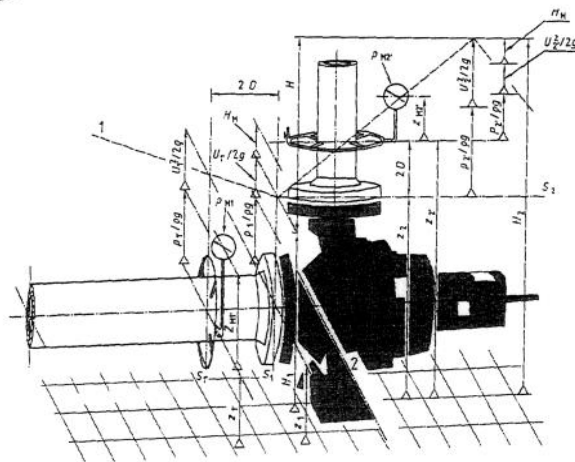
Measurements

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- Flowrate – Magnetic Flowmeter or Orifice/Nozzle
- Head or Pressure – Pressure Transducers
- Speed – Optical Pick-up
- Power – Electronic Power Meter

Advantage – all can feed digital data directly into a computer for immediate analysis of results

TEST LAYOUT



NOTE In this case for a horizontal shaft, $z_1 = z_2 = z_3$.

Figure 4 — Isometric illustration of the determination of pump total head

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Pre swirl Correction

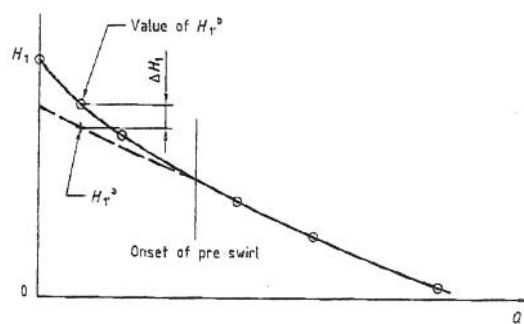


Figure 5 — Correction of inlet total head

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Variations and Fluctuations

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Table 3 — Permissible amplitude of fluctuations as a percentage of mean value of quantity being measured

Measured quantity	Permissible amplitude of fluctuations	
	grade 1 %	grade 2 %
Flow rate	+3	+6
Pump total head		
Torque		
Power input		
Speed of rotation	+1	+2

When using a differential pressure device to measure flow rate, the permissible amplitude of the fluctuations of the observed differential pressure is $\pm 0.5\%$ for grade 1 and $\pm 1.2\%$ for grade 2.

In the case of separate measurements of inlet total pressure and outlet total pressure, the maximum permissible amplitude of fluctuation shall be calculated on the pump total head.

Table 4 — Limits of variations between repeated measurements of the same quantity (based on 95 % confidence limits)

Conditions	Number of sets of readings	Permissible difference between largest and smallest readings of each quantity, related to the mean value			
		Flow rate, pump total head, torque, power input		Speed of rotation	
		grade 1 %	grade 2 %	grade 1 %	grade 2 %
Steady	1	0.6	1.2	0.2	0.4
	3	0.8	1.8	0.3	0.6
Unsteady	5	1.6	3.0	0.5	1.0
	7	2.2	4.5	0.7	1.4
	9	2.8	5.6	0.8	1.6
	13	3.8	7.8	1.0	2.0
	> 20	5.0	10.0	1.5	3.0

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Test Uncertainty

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Table 7 — Permissible values of systematic uncertainties

Quantity	Permissible value	
	grade 1 %	grade 2 %
Flow rate	± 1.5	± 2.5
Speed of rotation	± 0.35	± 1.4
Torque	± 0.9	± 2.0
Pump total head	± 1.0	± 2.5
Driver power input	± 1.0	± 2.0

Table 8 — Permissible values of overall measurement uncertainties

Quantity	Symbol	Grade 1 %	Grade 2 %
Flow rate	c_Q	± 2.0	± 3.5
Speed of rotation	c_n	± 0.5	± 2.0
Torque	c_T	± 1.4	± 3.0
Pump total head	c_H	± 1.5	± 5.5
Driver power input	c_{Pdr}		
Pump power input (computed from torque and speed of rotation)	c_P		
Pump power input (computed from driver power and motor efficiency)	c_P	± 2.0	± 4.0

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Acceptance Tolerances

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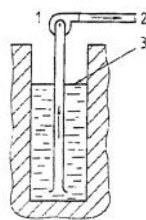
Quantity	Symbol	Grade 1 %	Grade 2 %
Flow rate	t_Q	$\pm 4,5$	± 8
Pump total head	t_H	± 3	± 5
Pump efficiency	t_η	$- 3$	$- 5$

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Cavitation Tests

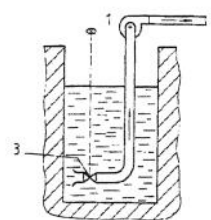
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Key

- 1 Test pump
- 2 To flow control valve and flowmeter
- 3 Adjustable water level

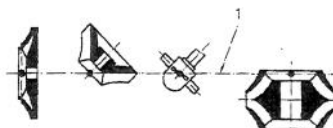
Figure 16 — Cavitation tests: Variation of NPSH by control of liquid level at pump inlet pump



Key

- 1 Test pump
- 2 To flow control valve and flowmeter
- 3 Inlet pressure control valve

Figure 17 — Cavitation tests: Variation of NPSH by means of an inlet pressure control valve



Key

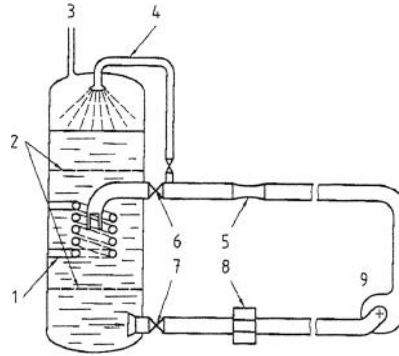
- 1 NPSH datum plane

Figure 1 — NPSH datum plane

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NOTE Cooling by means of a coil may be replaced by an injection of cool water above the liquid free surface and extraction of heated water.

Key

- | | | |
|---------------------------------|---------------------------------------|-----------------------------------|
| 1 Cooling or heating coils | 4 Spray nozzle for liquid de-aeration | 7 Isolating valve |
| 2 Stilling screens | 5 Flowmeter | 8 Measuring point for gas content |
| 3 To vacuum or pressure control | 6 Flow control valve | 9 Test pump |

Figure 15 — Cavitation tests: Variation of NPSH by means of a closed loop controlling pressure and/or temperature

Pumps Testing P&PS ADWJEA 05/09

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STERLING		PUMP TEST CERTIFICATE		Pump Type: TF20A	Works Number: 8/816003										
Sterling Fluid Systems Limited. Pease Cross, Pease Lane, Gilling, RG21 7SP UK Tel: 44 (0)18 932 3125 Fax: 44 (0)18 932 3302		Test Standard : Tested to BS5316 Part 1 (Class C)		Suction Diameter: 250 Discharge Diameter: 200 Number of Stages: 1											
Customer :		Customer Order Number : 1201-0215315(3555) Customer Item Number : WEST COMMON BOOSTER STN.													
Instrumentation		Site Duty		Test Conditions and Duty											
Delivery Gauge No. MW12 Suction Pipe Diam. 200 Suction Gauge No. P14 Suction Pipe Diam. 250 Wattmeter No. F20 Flowmeter No. W2 Voltmeter No. V2 Ammeter No. A2 Tachometer No. 1 Torquemeter No.		Fluid : WATER S.G. 1.00 Fluid Temp. AMB. Viscosity cS Speed rpm: 1423 Flow l/s: 200 Head m: 67.5 Abs. Power kW: 157.8 Efficiency %: 84 NPSH Required m: 4.9 Barom. Press. mbar:		WATER 1.00 l/s: m: kW:											
Impeller Details to Sterling SPP Standard EDT PRA 102				Driver Details											
Figure No: 1 D3: D4: A: B: L: R:		D1: 480.0 D5: E: T: T1: R1:		Driver Code: C Type: 7V-HAD355S Rated Power kW: 196.0 Rated Volt: 415											
TAD Code: ATMXN Y Pump Code: TF20AA1AQAA1 Other Test Codes: Hydro Test Pressure: 24.0 Bar				Make: BCP Serial Number: R499K-1 Speed rpm: 1425 F.L. Amps 313.0											
				Phase/Frequency: 3 / 50 Test Motor Number: Calibration Number:											
<table style="width: 100%; border-collapse: collapse;"> <tr> <td style="width: 60%;">Remarks</td> <td style="width: 40%; text-align: center;">0</td> </tr> <tr> <td>Reference Head Loss (m) :</td> <td style="text-align: center;">0</td> </tr> <tr> <td>Reference Flow (l/s) :</td> <td style="text-align: center;">0</td> </tr> <tr> <td>Constant Drive Loss (kW) :</td> <td style="text-align: center;">0</td> </tr> <tr> <td>Percentage Drive Loss :</td> <td style="text-align: center;">0</td> </tr> </table>						Remarks	0	Reference Head Loss (m) :	0	Reference Flow (l/s) :	0	Constant Drive Loss (kW) :	0	Percentage Drive Loss :	0
Remarks	0														
Reference Head Loss (m) :	0														
Reference Flow (l/s) :	0														
Constant Drive Loss (kW) :	0														
Percentage Drive Loss :	0														
Inspected By: AWA/SULZER Witnessed By: Test Date: 15-NOV-88 Certified:															

Pumps Testing P&PS ADLWEA 05/09

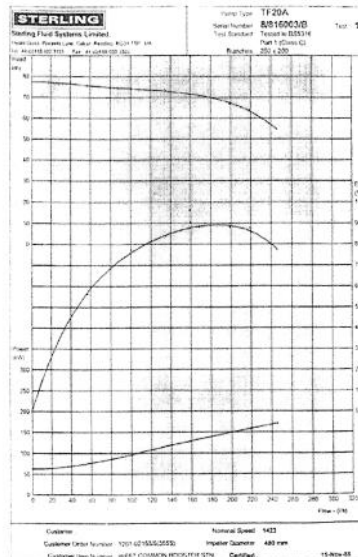
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STERLING		PUMP TEST CERTIFICATE			Pump Type:	TF20A	Works Number:			8/816003
Sterling Fluid Systems Limited		Test Standard:			Suction Diameter:	250	Pump ID Letter:			B
Theale Cross, Pinchot Lane, Calcot, Reading, RG31 7SP, UK		Tested to BS5316			Delivery Diameter:	200	Test Number:			1
Tel: 44 (0)118 922 3121 Fax: 44 (0)118 922 3102		Part 1 (Class C)			Number of Stages:	1	Sheet No:			2 of 2
Customer:		Customer Order Number:			1201-02153/5(3555)					
		Customer Item Number:			WEST COMMON BOOSTER STN.					
Test Point Number:		1	2	3	4	5	6	7	8	
Flow	l/s:	257.00	229.00	209.00	177.00	142.00	106.00	60.00	0.00	
Suction Head	m:	-3.40	-2.80	-2.60	-2.00	-1.60	-1.30	-1.00	-0.90	
Delivery Head	m:	53.30	65.10	68.60	74.10	77.80	79.10	81.80	83.90	
Velocity Correction	m:	2.01	1.60	1.33	0.96	0.61	0.34	0.11	0.00	
Pump Loss	m:	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Gauge Pos. Correction	m:	0.71	0.71	0.71	0.71	0.71	0.71	0.71	0.71	
Total Head	m:	58.42	70.21	73.24	77.77	80.52	81.45	83.62	85.51	
Volts		400.00	400.00	400.00	401.00	401.00	403.00	403.00	404.00	
Amps		327.00	302.00	286.50	259.00	230.00	193.00	161.00	137.00	
Power Factor		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Frequency	Hz:	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Motor Input	kW:	203.00	188.00	178.50	161.00	141.00	116.00	93.30	77.50	
Driver Output	kW:	194.42	179.81	170.56	153.53	134.06	109.72	87.62	72.24	
Drive Loss	kW:	0	0	0	0	0	0	0	0	
Input to Pump	kW:	194.42	179.81	170.56	153.53	134.06	109.72	87.62	72.24	
Speed	rpm:	1487	1489	1490	1492	1493	1495	1497	1498	
Pump Efficiency	%:	77.06%	87.72%	88.04%	87.95%	83.68%	77.20%	56.17%	0.00%	
Results Corrected to :	rpm:	1423.00	1423.00	1423.00	1423.00	1423.00	1423.00	1423.00	1423.00	
Flow	l/s:	245.94	218.85	199.60	168.81	135.34	100.89	57.03	0.00	
Total Head	m:	54.42	64.12	66.80	70.74	73.15	73.80	75.56	77.16	
Power Abs. SG 1	kW:	170.38	156.95	148.57	133.20	116.07	94.62	75.26	61.92	
Power Abs. SG 1	kW:	170.38	156.95	148.57	133.20	116.07	94.62	75.26	61.92	

Pumps Testing P&PS ADJWEA 05/09

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TEST CURVE



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Other Characteristics Tested

Cranfield
UNIVERSITY
Process Systems Engineering

- Noise
- Vibration
- Bearing Temperatures
- Seal Performance
- Shake down – extended run
- Cooling System